University of Strathclyde Engineering

Department of Mechanical and Aerospace Engineering

Project
Title
ELECTRICITY GENERATION FROM AMBIENT HEAT

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A thesis submitted in partial fulfilment for the requirement of the degree
Master of Science
Sustainable Engineering: Renewable Energy Systems and the Environment
2012

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Abstract

Fossil fuels are becoming scarce and their use is associated with greenhouse gas emissions. As the population increases rapidly and the economy grows, demand will outstrip supply. Current power supply and demand patterns do not match and that has caused a lot of grid instability. The energy infrastructure is overbuilt and over capacitated to maintain a spinning reserve for maintaining grid stability. The need for storage is thus important for any modern energy infrastructure. Storage will help in reserving excess capacity (spinning reserve) from off-peak periods or intermittent sources for utilization during peak times.

This thesis involves the thermodynamic analysis of a closed system cryogenic nitrogen power cycle using ambient heat as a source of energy. The aims are to evaluate the possibility of identifying a cycle configuration which can help in utilizing ambient heat as a source of power while at the same time providing an avenue for energy storage. The analysis explores the thermodynamic properties and behaviour of nitrogen at different pressures and temperatures aiming to balance off all the work and heat requirements of specifically identified cycle configurations. Heat and work balance is carried out to establish the net-work output and thermal needs of the cycles before introducing any external work or heat to create equilibrium at steady state. The sources of equilibrium heat are liquid nitrogen for cooling and ambient heat for heating. External work is introduced by the use of SHTS to supply supplemental cooling needs. All properties are evaluated on a reliable software platform called Engineering Equation Solver (ESS). The cycle operates by use of equipment's which enable the occurrence of four basic thermodynamic processes of Compression(C), Heating (H), Expansion (E) and Cooling(C). The cycle is identified by the acronym CHEC which stands for the initials of the four key processes. Cooling and heating are done in heat exchangers, expansion in turbine expanders or expansion valves and compression in done by a compressor or liquid compression pumps. All configurations were based on performance criteria of equipments available in the market. Information obtained from the websites of two leading cryogenic companies {[Linde Engineering], [Ebara International Corp]} was handy in ensuring that the cycles configurations used in this study remained within practical limits.

The results of ideal thermodynamic analyses proved nitrogen's suitability for use as a fluid medium in ambient heat electricity generation. Several cycles were analysed on an ideal basis and the final verdict was that a high pressure condensation cycle operating on liquid pumping at sub-cooled temperatures is fit for the purposes outlined in this thesis.

Acknowledgements

I would like to thank all my colleagues in the renewable energy course and all members of the academic staff for their relentless efforts in enabling this work come to a successful completion. I appreciate the support I received from my family members which has helped me endure through all the ups and downs of this struggle. I would like to express many thanks to my academic supervisor, Paul Tuohy for having put me to task in doing thorough investigations to the latter. Thanks to Ebara International Corporation for clarifications on cryogenic equipment operational limits.

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List of Abbreviations

AH	Ambient Heater
ASU	Air Separation Unit
CHEC	Compression Heating Expansion Cooling
CLS	Compressed liquid Stream
CE	Condensation enthalpy at constant pressure and temperature
CP	Condensation pressure at constant temperature
CT	Condensation temperature at constant pressure
COMPEX	Compression and Expansion Unit
DCAR	Direct Cooling Ability Ratio
ESS	Engineering Equation Solver
FHRU	Forced Heat Rejection Unit
FSCP	Final Stage Condensation Pressure
GTE	Gas Turbine Expander
GTOS	Gas Turbine Outlet Stream
GTOP	Gas Turbine Outlet Pressure

GTOT	Gas Turbine Outlet Temperature
GTIP	Gas Turbine Inlet Pressure
GTIT	Gas Turbine Inlet Temperature
H	Enthalpy [KJ/Kg]
HPC-CHEC	High Pressure Condensation CHEC power cycle
hr	Hour (Unit for time)
K	Kelvin (Temperature scale)
KE	Kinetic Energy
Kg	Kilogram (Units for mass)
KJ	Kilo Joules (Units for Work)
KW	Kilowatt (1000W) or KJ/s, unit for Power
LCP	Liquid Compression Pump
LNBT	Liquid Nitrogen Bulk Tank
LNG	Liquefied Natural Gas
LNGR	Liquefied Natural Gas Regasification
LNGRS	Liquefied Natural Gas Regasification Stream
m	Mass flow
OMFR	Operation Mass Flow Ratio
PE	Potential Energy
PFHE	Plate and Fin Heat Exchanger
Q	Heat transfer
S	Entropy [KJ/Kg-K]
s	Seconds (units for time)
SFEE	Steady Flow Energy Equation
SHTS	Secondary Heat Transfer Systems
SNC	Specific Nitrogen Consumption
VE	Vaporisation enthalpy at constant pressure and temperature
W	Work transfer
X_0	State of a substance at the saturated liquid line
X_1	State of a substance at the saturated vapour line
X_{FG}	Degree of dryness/wetness of a substance at point between the saturated liquid and vapour lines.
1SCP	1st Stage Condensation Pressure

2PTE	Two Phase Turbine Expander
2PTOS	Two Phase Turbine Outlet Stream
2PTOP	Two Phase Turbine Outlet Pressure
2PTOT	Two Phase Turbine Outlet Temperature
2PTIP	Two Phase Turbine Inlet Pressure
2PTIT	Two Phase Turbine Inlet Temperature
2SCP	2nd Stage Condensation Pressure
3SCP	3rd Stage Condensation Pressure

CHAPTER 1

INTRODUTION

Chapter contents

1.1: General background

Ambient heat is available everywhere on the planet. It is a low heat source and that’s why it has been difficult to harness it effectively for power generation. Most of the current modes of power generation operate at higher temperatures and therefore require a lot of heat which cannot be supplied from ambient. The onus of this thesis is to try and create a cycle that will operate with ambient heat state being the higher temperature. The process is therefore cryogenic in order to have a meaningful temperature difference between the ambient and the lowest temperature. Cryogenic technologies have been around for quite some time and the technologies involved are ripe for practical exploitation in defining how ambient heat can be harnessed. The basis of this thesis is to identify the processes, systems and ways or clues on how that ambient heat may become an integral part of our energy system. Given that ambient heat is an infinite and non-polluting form of energy, every effort towards its utilization has to be pursued keenly. The chapters and section that follow have identified the path which was executed effectively and results gave a promising outlook towards harnessing ambient heat. The power cycle being envisioned has the acronym CHEC which stands for the initials of four key thermodynamic processes of compression (C), heating (H), expansion (E) and cooling (C). Please refer to the list of abbreviations for the full meaning of the acronyms used in this thesis.

1.2: Area of focus and its importance

This thesis focuses on the design of a power cycle or heat engine [2.2] operating with nitrogen as its fluid medium. The overall power cycle will operate with one primary/main nitrogen power cycle assisted by one or several secondary heat transfer systems (SHTS) [2.3]. The SHTS help the cycle in maintaining a thermal and work balance. It is therefore paramount to state that the design of all associated SHTS systems forms part of the theme of this thesis.

1.3: Research question statement

1.3.1: Research Questions

1. Can a nitrogen power cycle be used to effectively harness ambient heat for electricity generation?
2. Can a nitrogen power cycle serve any needs for energy storage and grid stabilization in today’s energy infrastructure?
3. Can a nitrogen power cycle be used in utilising LNG regasification cold for power generation?
4. Can the CHEC power cycle be used for combined cooling and power operations?

1.3.2: Gaps addressed by this thesis

This thesis aims to fill voids which exist in the current energy systems and patterns of electricity generation with regard to resource availability, environmental concerns, political and socio-economical factors affecting today’s society. Establishing a new way of harnessing energy, mainly renewable energy to serve for electricity or mechanical energy needs of both industry and domestic consumers is of essence to the world of commerce and industry. The effects of bringing on board energy sources such as passive solar (ambient air/water) in to mainstream reliable power will serve to help in addressing some of the concerns. The gaps addressed by this thesis are as mentioned below:

- Diversification of renewable energy sources.
- Energy storage and grid stabilization
- Tame fuel poverty.
- Establish energy security and reliability.
- Reduction of Greenhouse gas emissions and pollution associated with fossil fuels.
- Address the intermittency and stochastic nature of renewable energy.
- Reduce world oil dependency and stabilize the price of oil.
- Diffuse tensions among nations scuffling for the meagre energy reserves in the Middle East, gulf cost and other parts of the world.
- Lowering the foot print of LNG handling process.

1.3.3: Motivation

1.3.3.1: General views

Dwindling fossil fuel reserves and raising energy demand are a catalyst for future supply shortages and raising prices. As the world population grows, so does energy demand, this will lead to either a price hike or severe scarcity of fossil fuel resources or a tremendous increase in greenhouse gas emissions and associated pollution?

There is need to broaden the supply and scope of renewable energy by using innovative means of utilizing currently available technological resources. The need to do this is immediate and that demands a technology which can emerge strongly without having to put a strain on the need for new hardware’s to be designed and manufactured. A technology that can utilize the currently market available equipments with very minimal modifications or without any need for new production lines to be set up is ideal for this purpose. The CHEC power cycle does all this.

This technology is mean on resources with competing uses such as water and so has little strain on the environment.

Addressing the intermittency of renewable energy has always been an elusive dream. This technology addresses that effectively.

Advantages of the nitrogen power plant.

1. It has no emissions of any kind.
2. Does not require any special geographical features. (but favoured by areas with higher ambient temperatures, lower relative humidity and windy areas.)
3. Has limited impacts on the environment.
4. Can be located near load centers thereby reducing transmission losses.
5. Provides Energy security
6. Use of locally available resources (ambient/waste heat and atmospheric nitrogen) thereby saving in foreign expenses of fuel importation.
7. Can be a precursor to the establishment of a hydrogen economy, (Hydrogen from electrolysis).
8. Can put emphasis on the use of electric cars as clean electricity will be available.
9. Long life of equipments due and low downtime/maintenance. {The fluid medium (N_2) used is inert and very clean and is non-fouling to equipment's}, [Linde Engineering], [Ebara International Corp].
10. Can provide stability and reliability in the provision of renewable power at all times by storing intermittent power from renewables.
11. Provides an energy reservoir for spare capacity if enough cryogen facilities are availed.

1.3.3.2: Personal views

I come from the developing world where the resources addressed in this thesis are abundant in supply but technology and research capability is lacking. It is with hope that the concept in this thesis will make it to the operational stage in order to address the acute energy poverty in developing countries and the world as a whole. The price of crude oil has escalated drastically and that has eaten heavily into the development budgets of most developing countries. Important projects on social services, healthcare and education have taken a back foot at the expense of raising and volatile crude oil prices in the world market. Budgets have to be re-adjusted upwards quite often and businesses tend to operate on an unreliable and unpredictable unforeseeable and an unknown future. Persistent civil strife and potential for war is closely associated with energy resources, this has led to a build-up in the stock of military arsenal to defend or perhaps conquer areas rich in energy resources. A new resource which is available in abundance will help neutralize some of the existing unnecessary tension and the associated resources could be used for other development purposes and the welfare of humanity.

The use of LNG is increasing rapidly thereby making it a must have part of any energy infrastructure. That provides an opportunity to utilize the cold given out from the regasification process to run power plants which will infuse ambient energy into the grid thus helping to reduce the prevailing foot print of LNG. It is my hope that this thesis may lead to the introduction of a nitrogen based energy economy and that the concept will grow into a feasible prototype in the same manner as that of Comrat [University of Strathclyde] thus making it the next online possible venture to emanate from Esru [University of Strathclyde] before finding its way to the Technology Innovation Centre [University of Strathclyde] for further scrutiny.

1.4: Literature review

1.4.1: Thermodynamics definitions and basics

Applied thermodynamics is the science of the relationship between heat, work and the properties of systems. It is concerned with the means necessary to convert heat energy from available sources such as fossil fuels into mechanical work (Ref: Eastop & McConkey).

A heat engine is the name given to a system which by operating in a cyclic manner produces net work from the supply of heat (Ref: Eastop & McConkey).

Heat is a form of energy which can be transferred from one body to another at a lower temperature by virtue of the temperature difference between the bodies (Ref: Eastop & McConkey).

A system is a collection of matter within prescribed and identifiable boundaries (Ref: Eastop & McConkey).

Work is defined as the product of a force and the distance moved in the direction of the force (Ref: Eastop & McConkey).

1.4.2: The laws of Thermodynamics

1.4.2.1: First law

The first law of thermodynamics states that energy can be neither created nor destroyed during a process, it can only change forms. The first law of thermodynamics is also known as the conservation of mass principle. [Cengel & Boles].

1.4.2.2: Second law

"It is impossible to construct a system which will operate in a cycle, extract heat from a reservoir, and do an equivalent amount of work on the surroundings" (Ref: Rogers & Mayhew[4th Ed]).

1.4.3: Steady flow Energy Equation

Energy balance:

The net change (increase/decrease) in the total energy of the system during a process is equal to the difference between the total energy entering and the total energy leaving the system during that process. Thus for a system,

$$(Total\ Energy\ in) - (Total\ Energy\ out) = (Change\ in\ Total\ Energy\ of\ system)$$

This relation is often referred to as the energy balance and is applicable to any kind of system undergoing any kind of process.

Energy change of a system: (where *E* stands for Energy)

$$Energy\ Change = Energy\ at\ Final\ State - Energy\ at\ Initial\ State$$

$$\Delta E_{system} = E_{final} - E_{initial} = E_2 - E_1$$

In absence of electric, magnetic and surface tension effects (i.e., for simple compressible systems) the change in total energy (E) of a system during a process is the sum of the changes in its internal (U), kinetic (KE) and potential (PE) energies and can be expressed as

$\Delta E = \Delta U + \Delta KE + \Delta PE$

Where:

$$\Delta U = m(u_2 - u_1)$$

$$\Delta KE = \frac{1}{2} m(V_2^2 - V_1^2)$$

$$\Delta PE = mg(z_2 - z_1)$$

(Ref: Cengel & Boles)

The steady flow energy equation is thus (where Q and W stand for heat and work respectively).

$Q + W = m\left\{(\mathrm{h2} - \mathrm{h1}) + \frac{1}{2}(C_2^2 - C_1^2) + g(z_2 - z_1)\right\}$ (Ref: Rogers & Mayhew).

1.4.4: Mechanisms of Energy transfer (Cengel & Boles, Chap 2; pg 73)

Energy can be transferred to or from a system in three forms: heat, work and massflow. Energy interactions are recognized at the system boundary as they cross it, and they represent the energy gained or lost by a system during a process. The only two forms of energy interactions associated with a fixed mass or closed system are heat transfer and work.

1.4.4.1: Heat Transfer (Q)

Heat transfer to a system (heat gain) increases the energy of the molecules and thus internal energy of the system and heat transfer from a system (heat loss) decreases it since the energy transferred out as heat comes from the energy of the molecules of the system.

1.4.4.2: Work Transfer (W)

An energy interaction that is not caused by a temperature difference between a system and its surroundings is work. Work transfer to (work done on) a system increases the energy of the system, and work transfer from (work done by) a system decreases it since the energy transferred out as work comes from the energy contained in the system.

1.4.4.3: Mass Flow (m)

Mass flow in and out of a system serves as an additional mechanism of energy transfer. When mass enters a system, the energy of the system increases because mass carries energy with it. Likewise when mass leaves the system, the energy contained within the system decreases because the leaving mass takes out some energy with it.

$$\Delta E_{system} = E_{in} - E_{out} = (Q_{in} - Q_{out}) + (W_{in} - W_{out}) + (E_{mass.in} - E_{mass.out})$$

1.4.5: Heat Transfer and heat Exchangers

Heat is energy in transition under the motive force of temperature difference (Ref: Rogers & Mayhew). Heat is a form of energy that is transferred between two systems (or systems and its surroundings) by virtue of a temperature difference (Ref: Cengel & Boles). A heat exchanger is a piece of equipment which allows the transfer of heat from one substance/system to another.

1.4.6: Thermodynamic Processes

Thermodynamic processes are shown on property plot profiles. A property plot is basically a graphic presentation of one thermodynamic property plotted on the (horizontal) x-axis against another plotted on the (vertical) y-axis.

- T-S diagrams (Temperature-Entropy diagram)
- P-V diagram (Pressure-Volume diagram)
- P-H diagram (Pressure-Enthalpy diagram)
- T-H diagram (Temperature-Enthalpy diagram)

1.4.6.1: Constant pressure processes (heating and cooling either in single phase or involving condensation/vapourisation)

These are processes which occur while the fluid is under constant pressure. These processes mainly involve the movement of heat to or from a system.

1.4.6.2: Isentropic process (adiabatic processes (pumping/compression and expansion)

This is a process which occurs at constant entropy. Most isentropic processes are adiabatic in nature which means no heat is absorbed or given out during the process.

1.4.7: Cryogenic energy storage

In cryogenic energy storage, excess electricity is utilized in operating a plant that produces and stores a cryogenic liquid. The fluid involved is mainly air. The cryogenic air would then be pumped to high pressure, heated by ambient or waste heat before being expanded in a turbine hooked with a generator to produce recovered electricity. Highview power storage company Ref: 1.4.7.1 is a pioneer in cryogenic energy storage and details of their process can be found in the sections that follow.

1.4.7.1: Highview Power Storage

"Highview is a developer of utility scale energy storage and power systems to optimise energy resources and help decarbonise the grid. Its proprietary process uses cryogenic (liquefied) air or its principle component, liquid nitrogen, as the working fluid and the media for storing and/or transporting energy'' (Ref: Highview Power Storage [1]).

1.4.7.1.1: Highview cryogenic energy storage Process

(Ref: Highview Power Storage [2]).

Figure: 1.4.7.1.1 is an outline of the process of Highview power storage technology. The process is shown alongside the key stages of liquefaction, storage and power recovery. The process flow makes the figure self-explanatory and further details are given in part 1.4.7.1.3. The process outlined here is the basis of operation of the two Highview products mentioned in part 1.4.7.1.2 and 1.4.7.1.3.

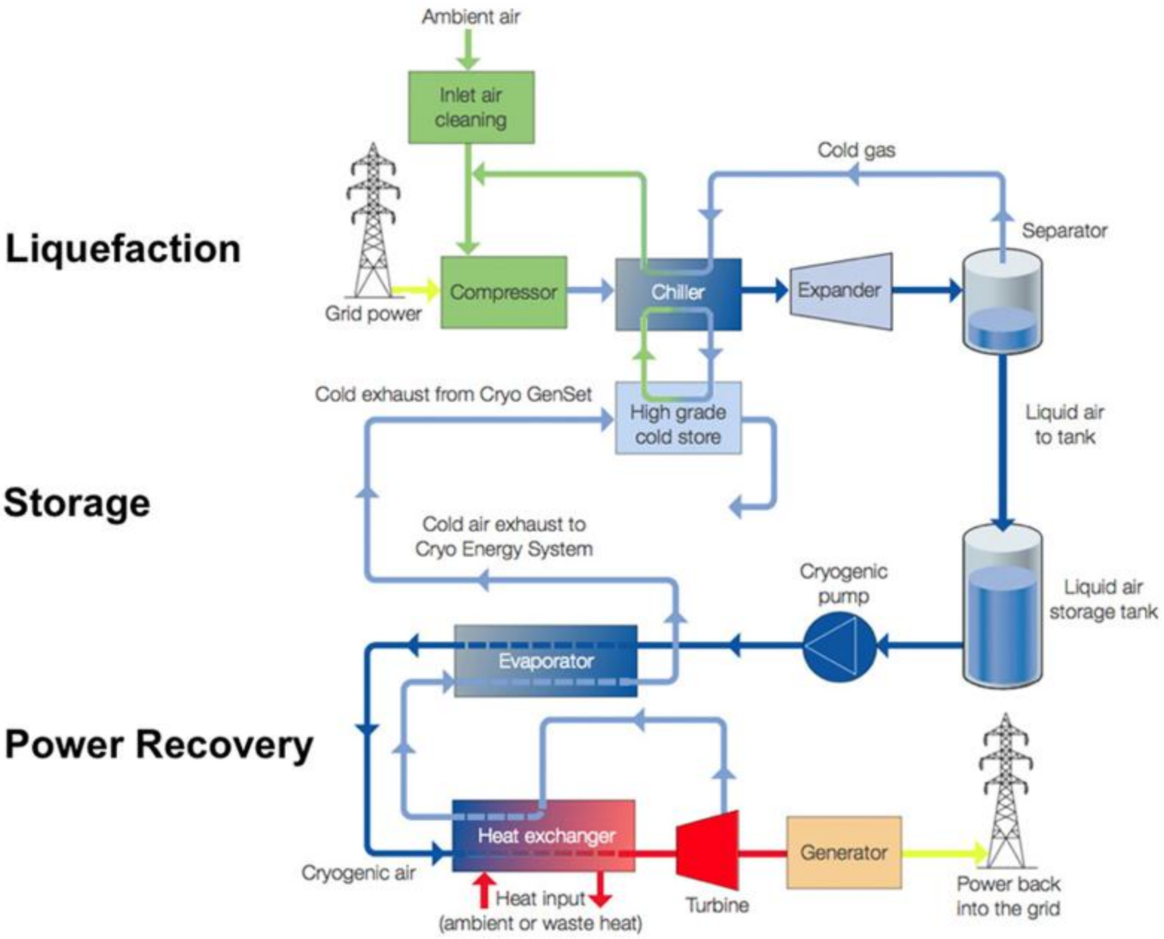


Figure: 1.4.7.1.1: Outline of Highview Power Storage process (Ref: Highview Power Storage [2]).

The process is briefly explained in the Cryo energy section below.

1.4.7.1.2: Cryo Energy System

(Ref: Highview power storage [3])

This is one of Highview's products which operate with an embedded cold store for cold recovery. The cold recovery helps in reducing the energy required for liquefaction by 50%, thereby increasing the overall efficiency of the system.

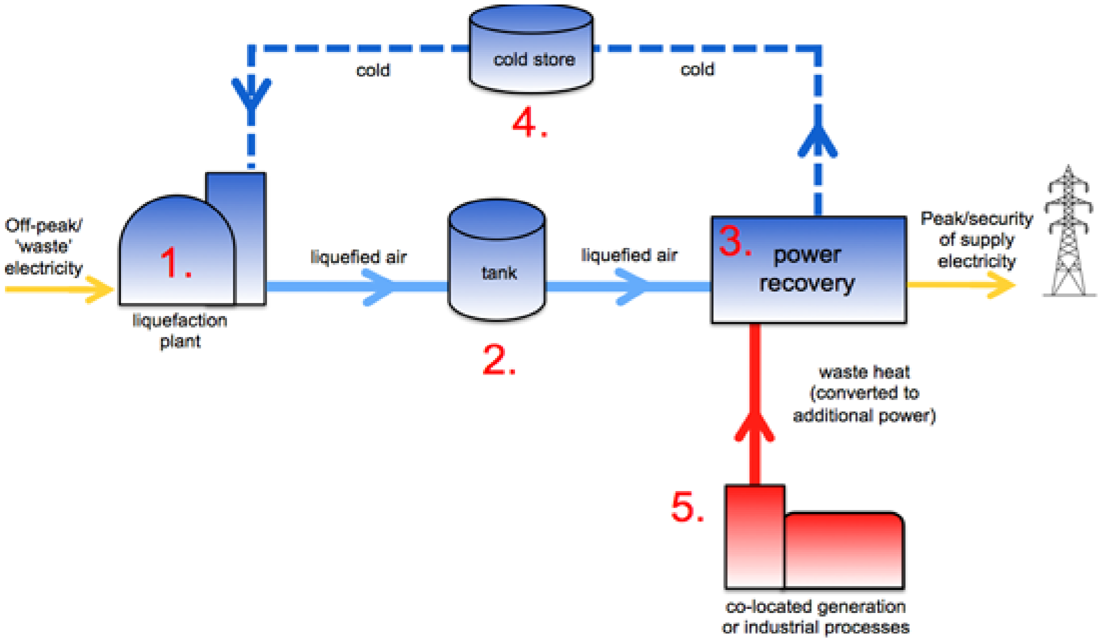


Figure: 1.4.7.1.2: Highview Power Storage, Cryo Energy System (Ref: Highview power storage [3])

The system uses excess electricity/off-peak electricity to produce the cryogenic liquid from air. The cryogenic liquid is then stored in specially designed properly insulated tanks at low pressure.

A 2000tonne tank=200MWhrs equivalent while a 100000tonne tank =10GWhrs.

Power is recovered when needed by pumping the cryogenic liquid to high pressure and then heating it by using ambient or waste heat before expanding the vapour in a turbine-generator to produce electricity. The cold gas exhausted during power recovery is recycled for use in the liquefaction process thus lowering the energy needed for cryogenic liquid production and thus raising the overall efficiency. If waste heat is used (stage 5), the overall efficiency increases due a higher initial enthalpy.

1.4.7.1.3: Cryogensate

(Ref: Highview power storage [4])

This is another product from Highview power Storage. Liquefaction is done at a different site while nitrogen is delivered to the generating station via tankers as demanded. This product is not designed to recycle any cold but it avails it for any industrial or commercial uses such as data centre cooling. The unit has capabilities of harnessing low grade waste heat with some increase in efficiency.

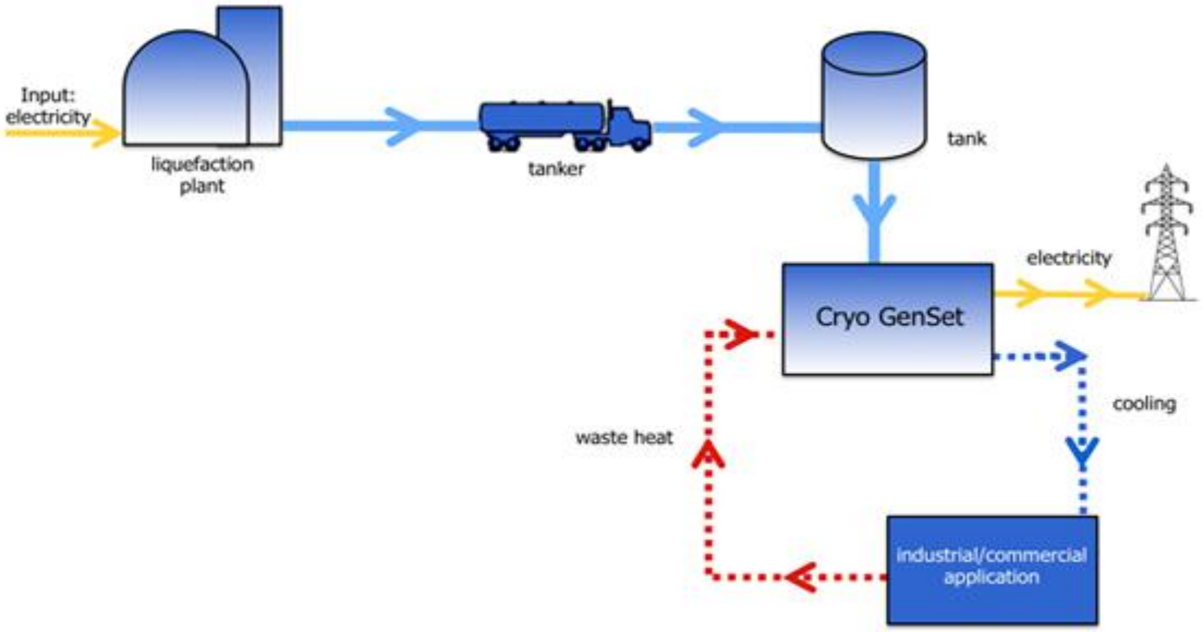


Figure: 1.4.7.1.3: Highview Power Storage, Cryo Gen Set (Ref: Highview power storage [4])

1.4.8: Liquid Nitrogen Regasification (LNGR)

LNG is liquefied natural gas in its liquid form. It is formed by chilling natural gas to around 113K, which makes its volume shrink 600 times from its gaseous state volume. It is a convenient way of storing and transporting bulk gas volumes from producer countries to the rest of the world. Here in the UK, two major LNG regasification terminals are

- South Hook LNG Terminal Company
- National Grid-Grain LNG terminal

“The UK’s demand for natural gas is increasing and is expected to rise by at least 15% over the next ten years. The UK’s traditional gas sources from the North Sea are declining. In order to meet this shortfall, it had been estimated that by 2010 around 50% of the UK’s gas will have to be imported. LNG importation is also playing a major role in increasing the security and diversity of energy supplies to the UK’’. (Ref: National grid GLNG).

Functions of an LNG terminal:

- Receiving and unloading LNG ships
- Storing of LNG in cryogenic tanks
- Managing gas quality
- Regasification of LNG to meet consumer demand
- Supply gas to the national transmission system

It is in the process of regasification as thermodynamically analysed in different parts of this thesis that the need to utilise the cold is emphasised. Regasification usually involves using a pump to increase the pressure of the liquid and them heating the compressed liquid by using any of the following means;

- Ambient air vaporisers
- Ambient water vaporisers
- Direct fired heaters
- Heated water vaporisers

This thesis aims to line-up the use of the cold given off during the regasification process with the contribution of heating up the LNG stream as it gains temperature. This is a relationship which may prove to be mutually beneficial, taking the cold while giving the heat. (Ref: National grid GLNG).

1.5: Research method statement

The next steps form an outline which defines the steps which will be followed in order to answer the research question in part 1.3.1.

1.5.1: Basic Thermodynamic analysis

The thermodynamic analysis for this cycle will be carried out on the ESS software platform [2.2]. Thermodynamic properties of nitrogen will be identified, evaluated and analysed in terms of work and heat for each and every of the processes it undergoes on an ideal CHEC power cycle at steady state [2]. The specific thermodynamic suitability of nitrogen as a medium of heat and work transfer for the CHEC power cycle is described in part [2.4].

The analysis will be carried out systematically in the following order:

- Introduction and assumptions
- Basic description of how ambient heat can be used for power generation
- A short introduction to the EES software
- Basic description of the nitrogen CHEC power cycle
- An outline of the reasons for choosing nitrogen as the cycle's fluid medium
- A general description of the CHEC power cycle operation
- Details of process thermodynamics and calculations
- Secondary Heat Transfer Systems
- Auxiliary units

1.5.2: Analyses of base case operational systems C-120-30-1, C-120-34-30@113.1K and C-120-34-30@123.6K

1.5.3: Further analyses of C-120-30-1 with liquid nitrogen and LNGR cooling

1.5.4: Further analyses of high pressure condensation cycle (C-120-34-30@113.1K) with liquid nitrogen and LNGR cooling

1.5.5: Further analyses of high pressure condensation cycle (C-120-34-30@123.6K) with liquid nitrogen and LNGR cooling

1.5.6: Results of analyses and discussion

1.5.7: Practical Implementation

This part will outline ways in which the CHEC power cycle is practically suitable for the envisioned applications of:

- Electricity generation from ambient heat
- Energy storage and grid stabilization
- Utilisation of LNGR cold for power generation
- Combined cooling and power operations

1.5.8: Future Work

- SHTS based on fluids other than nitrogen
- SHTS optimization
- Higher temperature CHEC cycles
- Effects of wind and humidity variations on CHEC power cycle performance
- The actual plant
- Financials
- Environmental impact assessment
- Carbon foot printing
- Power and cold from high gas pressure let down stations
- Establishing the cold potential of regasification facilities
- Examining the possibility of providing a huge sink for nuclear power plants.

1.5.9: Conclusion

A General conclusion on the work done in the whole exercise.

CHAPTER 2

BASIC THERMODYNAMIC ANALYSIS

Chapter contents

Page

2.1: Introduction and Assumptions

Notes:

1. The T-S diagram is not drawn in proportion to the real T-S diagram for nitrogen. Some parts have been enlarged visual clarity and ease of explanation.
2. The CLS pressure of the cycle for this study is 120 bars.
3. All cycles are considered to be in their ideal state.
4. Heat transfer from ambient to the cycle and heat transfer between the different streams is assumed to be perfect (Ideal).

5. Please refer to the list of abbreviations for the full meaning of the acronyms used in this thesis.

2.1.1: How ambient heat can be utilized for power generation

Ambient heat exists at low temperatures and that has made it an abundant but less effective form of energy which cannot be easily utilized for power generation. In many power plants existing today, ambient conditions form the lower side of the heat cycle, where heat has to be rejected in order to attain its original status.

This is opposite to what the CHEC power cycle is aiming to achieve, ambient conditions become the highest enthalpy positions of the cycle. In order to achieve this, the cycle must start at a lower than ambient enthalpy position where any exposure to ambient conditions will invoke a natural flow of energy from the ambient into the cycle fluid stream thus enabling other processes which transform of the absorbed ambient energy into mechanical power useable in the generation electricity. While that happens, the cycle’s ability to regain its original lower than ambient enthalpy position has to be maintained in order to sustain a cyclic system for continuous power production.

2.2: Engineering Equation Solver (ESS)

“EES (pronounced 'ease') is a general equation-solving program that can numerically solve thousands of coupled non-linear algebraic and differential equations. The program can also be used to solve differential and integral equations, do optimization, provide uncertainty analyses, perform linear and non-linear regression, convert units, check unit consistency, and generate publication-quality plots. A major feature of EES is the high accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability” (Ref: F-Chart Software [1]).

Program Details

“There are two major differences between EES and other equation-solving programs. First, EES allows equations to be entered in any order with unknown variables placed anywhere in the equations; EES automatically reorders the equations for efficient solution. Second, EES provides many built-in mathematical and thermophysical property functions useful for engineering calculations. For example, the high accuracy steam tables are implemented such that any thermodynamic property can be obtained from a built-in function call in terms of any two other properties. Similar capability is provided for most refrigerants, ammonia, methane, carbon dioxide and many other fluids. Air tables are built-in, as are psychrometric functions and JANAF data for many common gases. Transport properties are also provided for all substances" (Ref: F-Chart Software [2]).

"The library of mathematical and thermophysical property functions in EES is extensive, but it is not possible to anticipate every user's need. EES allows the user to enter his or her own functional relationships in three ways. First, a facility for entering and interpolating tabular data is provided so that tabular data can be directly used in the solution of the equation set. Second, the EES language supports user-written functions, procedures, modules and subprograms. Functions and procedures are similar to those in Pascal and FORTRAN in which assignment statements rather than equalities are employed. EES modules and subprograms are callable EES routines that use equalities. Functions, procedures, modules, and subprograms can be saved in library files that are automatically read in when EES is started. Third, compiled functions and procedures, written in a high-level language such as Pascal, C, or FORTRAN, can be dynamically-linked with EES. These three methods of adding functional relationships provide very powerful means of extending the capabilities of EES" (Ref: F-Chart Software [2]).

2.3: The basic nitrogen CHEC power cycle

The basic CHEC power cycle involves thermodynamic processes of Compression, Heating, Expansion and Cooling, with Vaporization and Condensation processes being part of the heating and cooling. Given that the first four processes are the main ones, the acronym CHEC is derived from the first letter of each of the processes.

- C---Compression
- H---Heating
- E---Expansion
- C---Cooling

The envisioned power cycle will henceforth be referred to as the CHEC power cycle. The ambient conditions shall be at 273K, 1bar with wind and humidity conditions assumed to be of no effect to the system. Though wind and humidity have some effects on the performance of the CHEC power cycle, it is apparent that they be considered at a later stage as further work of this study [9].

2.4: Why Nitrogen and not any other fluid?

Quite a number of other fluids can do the same but nitrogen beats them all due to its unlimited abundance in air and a wide temperature difference between its liquid state at 1bar and ambient gaseous state. The table below shows some of the other fluids as well as stating their merits and shortfalls.

Nitrogen is cheap and readily available. It is the main constituent of air at 78% of air by volume. It may have to undergo some preparatory processes before being introduced into the cycle. Unlike many fluid mediums which have competing uses, this is mainly used as an industrial gas. The initial supply for start-up of the plant is to be supplied by road tankers (in liquid form) after which the levels shall be maintained by a continuous make-up from a localized N_2 plant incorporated to utilize the extra cold generated by the cycle.

The preparatory process involved is not as intense as those required for pretreatment of boiler feed water for steam generation in steam plants. No consumable chemicals are used in the nitrogen pre-treatment process.

Nitrogen as a fluid medium has no corrosive properties as compared to steam and the equipments involved are expected to incur less downtime and maintenance costs. Equipments operating on nitrogen are expected to last longer because of its cleanliness and non-corrosiveness (inertness). Other fluid mediums are not to be ruled out completely and have been placed as a focus area for further work on the CHEC power cycle studies in chapter 9.

Table 2.4: Merits and demerits of different fluids suitability for use on the CHEC cycle

Fluid	**Cold Status**	**ΔT to Ambient**	**Merits**	**Demerits**
Nitrogen	77	196	Free and highly abundant in air	Need to separate from Air mixture
Hydrogen			Low freezing point	Has to be processed
CO2			Has to be purchased	Sublimes at higher temperature
Air			Free and highly abundant in air	The problem of preferential boiling between O2 and N2.
Helium			Low freezing point	Rare gas
Argon			Low freezing point	Rare gas
O2			Free and highly abundant in air	Need to separate from Air mixture and the danger of fire
Propane				Flammable, fossil fuel based

2.4.1: General description of the CHEC cycle operation

Figure 2.4.1 is a simple plot of a basic ideal CHEC power cycle on a T-S diagram. The cycle can be broken down into 7 nodes as described in table 2.4.1 below; (Please refer to the table of acronyms)

Table: 2.4.1: Description of the nodes on a typical CHEC power cycle

Node	Description
1	Liquid nitrogen from the LNBT (bulk tank) at the LCP(Pump) inlet (always at 1bar, -122.2KJ/Kg ,77.24K) unless stated otherwise
2	LCP discharge (always at 120bar unless stated otherwise)
3	GTE inlet (always at 255.2KJ/Kg, 273K) unless stated otherwise
4	GTE outlet (dependent on the GTOS pressure)
5	GTOS at saturated vapour state (dryness fraction/wetness index=1)
6	GTOS at saturated liquid state (dryness fraction/wetness index=0)
7	GTOS at 1 bar, 77.24K (dryness fraction/wetness index dependent on entropy)

Liquid nitrogen from the LNBT is pumped isentropicaly by the LCP from node 1(1bar) to node 2 (120bar) thus creating the high pressure Compressed Liquid Stream (CLS) [2.4.2.1]. The CLS flows through an arrangement of heating systems where the stream is heated at constant pressure to reach the presumed ambient temperature of 273K at node 3 .At node 3, the high pressure CLS will be a superheated high enthalpy gas which is expanded isentropicaly (adiabatically) by a GTE from the GTIP of 120bar to the respective GTOP pressure at node 4. The expansion is adiabatic thus no heat flows into or out of the system while the process is isentropic and produces some work output [2.4.2.3]. All the heat will be sourced from within the fluid itself thus creating a temperature drop which becomes very useful in cold creation required by the GTOS. The GTOS needs to be cooled further between nodes 4 to 1 where it is expected to be completely condensed back to its original liquid state at the LNBT. The GTOS at node 4 is cooled at constant pressure (GTOP) the saturated vapour state at node 5 [2.4.2.4]. At the saturated vapour state the stream undergoes some condensation at constant pressure (GTOP) and temperature (GTOT) to reach the saturated liquid state node 6 [2.4.4.5].

After attaining the saturated liquid state at node 6 the stream is further expanded in a 2PTE from the GTOP to the FSCP at node 7. The pressure (FSCP) at node 7 is 1bar@ 77.24K but the quality is that of a wet vapour which requires further cooling to bring it back to node 1 where the cycle will be complete (Figure 2.4.1). The stream at node 7 will undergo the final condensation at 1bar to reach its original state at node 1 in the LNBT or the pump suction vessel ready to begin another cycle [2.4.2.7].

The process is then repeated again and the cycle runs continuously providing a reliable Net power output from the GTE and the 2PTE after deducting all the power input from the LCP and other associated secondary systems.

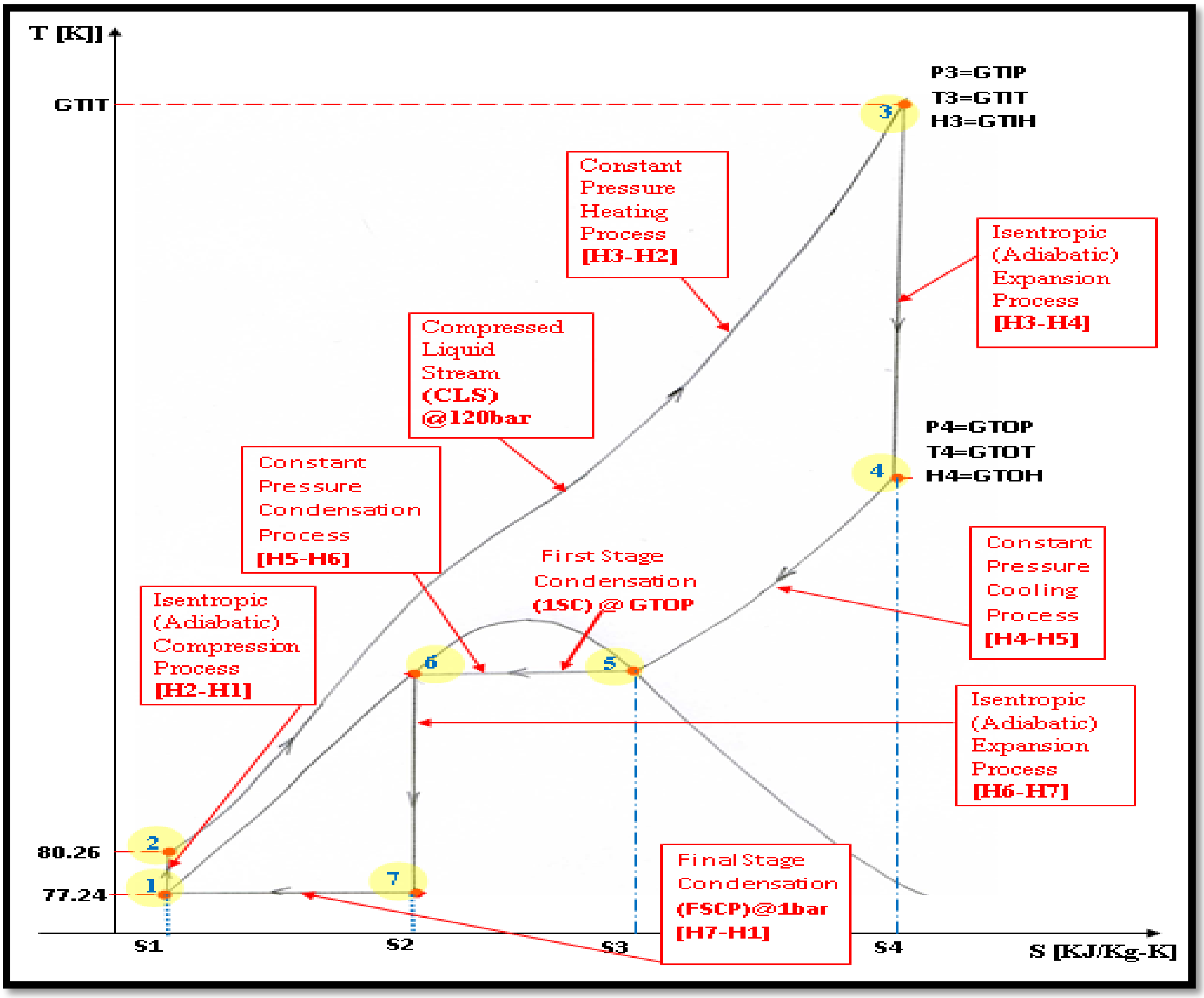


Figure 2.4.1: T-S diagram of a typical CHEC Power cycle.

2.4.1.1: CHEC Cycle nomenclature

The general CHEC power cycle does not have a fixed configuration in terms of its operational parameters. All thermodynamic property variables depend on the operational pressures of the individual cycles. It is therefore important to have a systematic way of identifying each individual power cycle in terms of the pressures at different stages of operation.

A system which starts with the letter “C’’ in caps to denote the word CHEC, followed by the cycle’s pressures in descending order of expansion from the highest to the lowest. The highest pressure being the LCP discharge pressure or CLS pressure while the lowest being the FSCP. In case of a multi-stage expansion, the intermediate pressures between the GTOP and the FSCP will be referred by the number of stage at which they occur. As an example, 1SCP will be the pressure at which condensation of the GTOS occurs first. The next condensation pressure shall be referred to as 2SCP which occurs after the second (GTOS) expansion. The rest are named in that order until the FSCP is attained.

Thus the generic cycle on figure 2.4.1 can be identified by this nomenclature as C-P2-P4-P7, where;

- C- Stands for CHEC
- P2- The LCP discharge/CLS pressure
- P4- The GTOP
- P7- The FSCP

As an example, the cycle on figure 3.1.1.a can be identified as C-120-30-1, where;

- C- Stands for CHEC
- 120- The LCP discharge/CLS pressure (120bar)
- 30- The GTOP (30bar)
- 1- The FSCP (1bar)

In case different systems have the same pressure configurations but differ in any other way, the lowest temperature of the cycle at which liquid collects before being compressed shall be marked as @T^{O}K, where T is the liquid temperature in the accumulator.

As an example, the cycle on figure 3.2.2.a can be identified as C-120-34-30@113.1K, where;

- C- Stands for CHEC
- 120- The LCP discharge/CLS pressure (120bar)
- 34- The GTOP (34bar)
- 30- The FSCP (30bar)
- 113.1- The liquid temperature at the accumulator(113.1K)

2.4.1.2: The key Flow streams of a CHEC power cycle

The CHEC power cycle is made up two main streams generally referred to as compressed liquid stream (CLS) and the gas turbine outlet stream (GTOS). These are further described in parts 2.4.1.2.1 and 2.4.1.2.2 below.

2.4.1.2.1: Compressed Liquid Stream (CLS)

This is the stream which stretches from the discharge of the LCP all the way to the gas turbine inlet. On the T-S diagram of figure 2.4.1, this can be represented by the stretch between nodes 2 and 3. The pressure of this flow stream in this thesis is 120bar [2.1], however that can vary depending on the cycle configuration and equipment specification. The flow pattern may also vary in order to match specific cooling enthalpy requirements of the GTOS or any of the secondary heat transfer systems. The flow may be split into parallel or a series of several parallel streams for that very purpose. At some point where the enthalpy is required is less, a flow controlled matching side stream may be diverted to balance the enthalpy before rejoining the mainstream. The first stage of flow immediately after the LCP discharge will be through heat exchangers of the FHRU and COMPEX units which facilitate the cooling of the GTOS at lower temperatures [2.5]. The stream acts as a heat sink for the SHTS [2.5]. The final stretch before entering the gas turbine is ambient heating which is done with the help of well-designed ambient heaters (AH) [2.4.3.2]; the compressed liquid stream undergoes a constant pressure heating process [1.4.6.1] as it gains heat while its temperature rises towards ambient.

2.4.1.2.2: Gas Turbine Exhaust Stream (GTOS)

This appears as the stretch from node 4 all the way down to node 1 on figure 2.4.1. It starts at the turbine outlet down to the LNBT or LCP suction vessel. The pressure of this stream depends on the designated GTOS pressure of the system and any further expansions on the downstream path towards the FSCP. The GTOS will undergo the processes of constant pressure cooling or constant pressure condensation as it loses heat down to the node 1 status.

2.4.2: Details of CHEC Process thermodynamics and calculations

The individual thermodynamic processes which the fluid undergoes can be classified in the following order based of the pattern of occurrence in the cycle’s T-S diagram [Figure 2.4.1]:

2.4.2.1: Isentropic (Pumping) liquid compression (LCP); nodes 1 and 2

$Q21 + W21 = (H2 - H1)\ KJ/Kg$

$Q21 = 0$

$Therefore: W21 = (H2 - H1) KJ/Kg \ldots\ldots Pump\ Work\ input$

2.4.2.2: Isobaric (constant pressure) vaporization (PFHE); nodes 2 and 3

$Q32 + W32 = (h3 - h2) KJ/Kg$

$W32 = 0 \ldots no\ work\ is\ done\ during\ vaporization$

$Therefore:\ Q32 = (h3 - h2) KJ/Kg \ldots\ldots\ldots Heat\ Supplied$

2.4.2.3: Isentropic Gas Expansion: (GTE); nodes 3 and 4

$Q34 + W34 = (H3 - H4) KJ/Kg$

$Q34 = 0 \ldots..The\ expansion\ is\ adiabatic$

$Therefore{:}\ \ W34 = (H3 - H4)KJ/Kg \ldots .GTE\ Work\ out$

2.4.2.4: Isobaric (constant pressure) Cooling: (PFHE); nodes 4 and 5

$Q45 + W45 = (H4 - H5)KJ/Kg$

$W45 = 0 \ldots .Since\ no\ work\ is\ done\ in\ the\ cooling\ process$

$Therefore{:}\ Q45 = (H4 - H5)KJ/Kg \ldots \ldots .Heat\ Rejected$

2.4.2.5: Isobaric (constant pressure) Condensation: (PFHE): nodes 5 and 6

$Q56 + W56 = (H5 - H6)\ KJ/Kg$

$W56 = 0 \ldots .Since\ no\ work\ is\ done\ during\ condensation$

$Therefore{:}\ \ Q56 = (H5 - H6)KJ/Kg \ldots ..Heat\ Rejected$

2.4.2.6: Isentropic (Adiabatic) Two Phase Expansion: (2PTE); nodes 6 and 7

$Q67 + W67 = (H6 - H7)$

$Q67 = 0 \ldots ..The\ expansion\ is\ adiabatic$

$Therefore{:}\ \ W34 = (H3 - H4)KJ/Kg \ldots .2PTE\ Work\ out$

2.4.2.7: Isobaric (constant pressure) Condensation: (PFHE): nodes 7 and 1

$Q71 + W71 = (H7 - H1)\ KJ/Kg$

$W71 = 0 \ldots .Since\ no\ work\ is\ done\ during\ condensation$

$Therefore{:}\ \ Q71 = (H7 - H1)KJ/Kg \ldots ..Heat\ Rejected$

2.4.2.8: Cycle analysis and performance criteria

2.4.2.8.1: Work Ratio (WR) or Coefficient of performance (COP)

Work ratio is the ratio of work output to work input

$$\frac{Work\ Output}{Work\ input}$$

Work Output:

$$Work\ output = [GTE\ Work.out(H3 - H4) + 2PTE\ Work.out(H6 - H7)]\ KJ/Kg$$

$$Work\ output = [(H3 - H4) + (H6 - H7)]\ KJ/Kg$$

Work input:

$$Work\ input = [LCP\ Work.in(H2 - H1) + Secondary\ Systems\ Work.in]\ KJ/Kg$$

Thus Work Ratio:

$$WR = \frac{Work\ Output}{Work\ input} = \left\{\frac{[(H3 - H4) + (H6 - H7)]}{[LCP\ Work.in(H2 - H1) + Secondary\ Systems\ Work.in]}\right\}$$

2.4.2.8.2: Specific Nitrogen Consumption (SNC)

This is the mass flow of cooling cryogenic liquid expressed per unit Kg of the main circulating fluid.

2.4.3: Property values of different processes as obtained from ESS

The following analyses involve getting the property values at the respective nodes so as to enable the calculation of heat and work of the process which the fluid has undergone. The resulting status is of importance for evaluation of the ability of the fluid to maintain its constant flow to the next stage of the process within the cycle. The flow of heat is dependent on the temperature difference between the streams and thus the importance of knowing the temperature at every node is of essence.

The processes/equipments have been categorized as shown below in order to create a systematic flow from node 1 to node 7 and back to node 1 in that cyclic order as appearing on Figures 2.4.1. Starting from node 1 at the pump inlet, the processes flow as follows:

- Pumping (LCP) liquid compression.
- SHTS heat sinking by FHRU and COMPEX (PFHE)
- Ambient heating (AH)
- Gas Turbine Expander (GTE)
- SHTS heat sourcing by COMPEX (PFHE)
- Two Phase Turbine Expander (2PTE)
- SHTS heat sourcing by FHRU.

The heat sourcing and sinking will be explained further in part 2.5 while the rest are dealt with in the section below.

2.4.3.1: Pump (LCP)

Table 2.4.3.1 shows the values of various thermodynamic properties of nitrogen undergoing an isentropic (pumping) liquid compression process from node 1 to state 2. Figure 2.4.3.1 is a graphic plot of the increase in pressure (bars) against enthalpy change (KJ/Kg) which resulted into a linear relationship as expected for any ideal isentropic process. Thermodynamically, the amount of work input for any pressure increase can be determined by applying the associated property values in the equation given in part [2.4.2.1].

Note:

1. H1 (-122.2 KJ/Kg) is the enthalpy at (P1)1bar while H2 changes as the pressure increases.
2. T2 (pump outlet temperature) is the initial temperature at which the CLS is available for heat exchange with either the TOS or the secondary heat transfer systems.

Table 2.4.3.1: Values of ideal enthalpy changes of nitrogen undergoing liquid compression from H1-H2.

Pressure	Entropy (S)	Enthalpy (H)	Enthalpy Change (H1-H2)	(T1) Pump inlet Temperature	(T2) Pump outlet Temperature
[Bar]	[KJ/Kg-K]	[KJ/Kg]	[KJ/Kg]	[K]	[K]
1	2.832	-122.2	0	77.24	77.24
10	2.832	-121.1	1.1	77.24	77.51
20	2.832	-119.8	2.4	77.24	77.77
30	2.832	-118.6	3.6	77.24	78.03
40	2.832	-117.4	4.8	77.24	78.29
49	2.832	-116.3	5.9	77.24	78.52
50	2.832	-116.1	6.1	77.24	78.54
60	2.832	-114.9	7.3	77.24	78.79
70	2.832	-113.7	8.5	77.24	79.04
80	2.832	-112.5	9.7	77.24	79.29
90	2.832	-111.2	11	77.24	79.54
100	2.832	-110	12.2	77.24	79.78
110	2.832	-108.8	13.4	77.24	80.02
120	2.832	-107.6	14.6	77.24	80.26

2.4.3.2: Ambient heater (AH)

Ambient heating is assumed to be any residual heating required to make the fluid attain the GTIT (which is set to be the atmospheric ambient temperature of the area) after all the SHTS have utilised the CLS as a heat sink for their operations. The operation of this part of the cycle is assumed to be a perfect ideal heat gain by the fluid sourced from surrounding ambient air or water or any renewable source of heat.

2.4.3.3: Gas Turbine Expander (GTE)

Table: 2.4.3.3.a: Thermodynamic values of the ideal isentropic expansion process at the GTE

(GTOP)-Turbine outlet pressure	(CE)-Condensation Enthapy at GTOP	(CT)-Condensation Temperature at GTOP	Enthalpy after Isentropic Expansion from 120bar/273K	(GTOT)-Turbine Outlet Temperature after Isentropic Expansion from 120bar/273K	Work Out 1 (Gas turbine Expansion)
[Bar]	[KJ/Kg]	[K]	[KJ/KgK]	[K]	[KJ/Kg-K]
1	199.27	77.24	63.35	77.24	191.85
2	190.52	83.63	78.29	83.63	176.91
3	183.96	87.91	87.85	91.11	167.35
4	178.35	91.23	95.29	99.12	159.91
5	173.32	93.99	101.5	105.8	153.7
6	168.63	96.38	106.8	111.7	148.4
7	164.19	98.49	111.6	116.9	143.6
8	160.05	100.04	115.9	121.6	139.3
9	156.19	102.1	119.8	125.9	135.4
10	152.53	103.7	123.4	129.9	131.8
11	148.91	105.2	126.8	133.7	128.4
12	145.75	106.6	129.9	137.2	125.3
13	142.04	108	132.9	140.5	122.3
14	137.32	109.2	135.7	143.6	119.5
15	133.26	110.4	138.4	146.6	116.8
16	129.17	111.5	141	149.5	114.2
17	125.11	112.6	143.4	152.2	111.8
18	122.52	113.6	145.8	154.8	109.4
19	118.49	114.6	148	157.4	107.2
20	113.04	115.6	150.2	159.8	105
21	109.3	116.5	152.3	162.1	102.9
22	104.89	117.4	154.4	164.4	100.8
23	101.6	118.3	156.3	166.6	98.9
24	97.41	119.1	158.2	168.8	97
25	92.85	119.9	160.1	170.8	95.1
26	88.08	120.7	161.9	172.9	93.3
27	83.07	121.5	163.7	174.8	91.5
28	77.736	122.2	165.4	176.7	89.8
29	71.971	122.9	167.1	178.6	88.1
30	65.626	123.6	168.7	180.4	86.5
31	58.516	124.5	170.3	182.2	84.9
32	49.912	125	171.9	184	83.3
33	38.27	125.6	173.4	185.7	81.8
34			174.9	187.4	80.3
35			176.4	189	78.8

Note:

- The gas turbine inlet conditions are, H=255.2 KJ/Kg; GTIT=273K; S= 5.234 KJ/Kg-K and GTIP=120Bar.
- The gas turbine outlet conditions are shown in the table with respect to the different outlet pressures from 1bar to 35bar.

Table: 2.4.3.3.b: Values of condensation enthalpy of GTOS between 34 and 1bar.

(GTOP)-Turbine outlet pressure	Entropy at GTOP and X=0	Enthalpy of GTOS at GTOP and X=0	Entropy at GTOP and X=1	Enthalpy of GTOS at GTOP and X=1	(CE)-Condensation Enthapy at GTOP	(CT)-Condensation Temperature at GTOP	Enthalpy after Isentropic Expansion from 120bar/273K	(GTOT)-Turbine Outlet Temperature after Isentropic Expansion from 120bar/273K	Cooling Demand from Turbine Outlet to Sat Vapour Line	Cooling Demand from Turbine outlet to Sat Liquid line
[Bar]	[KJ/Kg-K]	[KJ/Kg]	[KJ/Kg-K]	[KJ/Kg]	[KJ/Kg]	[K]	[KJ/Kg]	[K]	[KJ/Kg]	[KJ/Kg]
1	2.831	-122.2	5.412	77.07	199.27	77.24	63.35	77.24	-13.72	185.55
2	2.994	-109	5.273	81.52	190.52	83.63	78.29	83.63	-3.23	187.29
3	3.098	-100	5.19	83.96	183.96	87.91	87.85	91.11	3.89	187.85
4	3.176	-92.85	5.131	85.5	178.35	91.23	95.29	99.12	9.79	188.14
5	3.24	-86.81	5.084	86.51	173.32	93.99	101.5	105.8	14.99	188.31
6	3.294	-81.49	5.044	87.14	168.63	96.38	106.8	111.7	19.66	188.29
7	3.342	-76.69	5.009	87.5	164.19	98.49	111.6	116.9	24.1	188.29
8	3.385	-72.29	4.979	87.76	160.05	100.4	115.9	121.6	28.14	188.19
9	3.424	-68.17	4.953	88.02	156.19	102.1	119.8	125.9	31.78	187.97
10	3.461	-64.26	4.931	88.27	152.53	103.7	123.4	129.9	35.13	187.66
11	3.495	-60.5	4.91	88.41	148.91	105.2	126.8	133.7	38.39	187.3
12	3.523	-57.4	4.89	88.35	145.75	106.6	129.9	137.2	41.55	187.3
13	3.553	-53.99	4.869	88.05	142.04	108	132.9	140.5	44.85	186.89
14	3.59	-49.79	4.848	87.53	137.32	109.2	135.7	143.6	48.17	185.49
15	3.619	-46.43	4.827	86.83	133.26	110.4	138.4	146.6	51.57	184.83
16	3.647	-43.19	4.805	85.98	129.17	111.5	141	149.5	55.02	184.19
17	3.673	-40.07	4.784	85.04	125.11	112.6	143.4	152.2	58.36	183.47
18	3.686	-38.51	4.764	84.01	122.52	113.6	145.8	154.8	61.79	184.31
19	3.71	-35.58	4.743	82.91	118.49	114.6	148	157.4	65.09	183.58
20	3.746	-31.3	4.723	81.74	113.04	115.6	150.2	159.8	68.46	181.5
21	3.757	-28.81	4.703	80.49	109.3	116.5	152.3	162.1	71.81	181.11
22	3.79	-25.73	4.684	79.16	104.89	117.4	154.4	164.4	75.24	180.13
23	3.804	-23.87	4.663	77.73	101.6	118.3	156.3	166.6	78.57	180.17
24	3.825	-21.21	4.643	76.2	97.41	119.1	158.2	168.8	82	179.41
25	3.848	-18.31	4.622	74.54	92.85	119.9	160.1	170.8	85.56	178.41
26	3.871	-15.36	4.601	72.72	88.08	120.7	161.9	172.9	89.18	177.26
27	3.894	-12.34	4.578	70.73	83.07	121.5	163.7	174.8	92.97	176.04
28	3.918	-9.216	4.554	68.52	77.736	122.2	165.4	176.7	96.88	174.616
29	3.943	-5.951	4.529	66.02	71.971	122.9	167.1	178.6	101.08	173.051
30	3.97	-2.476	4.501	63.15	65.626	123.6	168.7	180.4	105.55	171.176
31		1.324	4.469	59.84	58.516	124.3	170.3	182.2	110.46	168.976
32		5.678	4.431	55.59	49.912	125	171.9	184	116.31	166.222

Table: 2.4.3.3.c: Values showing the ideal power yield of expansion in the two phase region

2PTOP	Enthalpy of 2PTOS at 2PTOP and Xo	CT of 2PTOS at 2PTOP	Isentropic 2PTE Power yield from 30bar to 2PTOP	Cold Generated by the Isentropic Expansion from 30Bar to 2PTOP	Cold required for Condensation of 2PTOS to Saturated liquid line at 2PTOP	Work Out 2 from 2PTE Turbine
[Bar]	[KJ/Kg]	[K]	[KJ/Kg]	[KJ/Kg]	[KJ/Kg]	[KJ/Kg]
1	-122.2	77.24	-34.29	31.814	87.91	31.814
2	-109	83.63	-27.42	24.944	81.58	24.944
3	-100	87.91	-23.33	20.854	76.67	20.854
4	-92.85	91.23	-20.43	17.954	72.42	17.954
5	-86.81	93.99	-18.19	15.714	68.62	15.714
6	-81.49	96.38	-16.37	13.894	65.12	13.894
7	-76.69	98.49	-14.85	12.374	61.84	12.374
8	-72.29	100.04	-13.55	11.074	58.74	11.074
9	-68.17	102.1	-12.41	9.934	55.76	9.934
10	-64.26	103.7	-11.41	8.934	52.85	8.934
11	-60.5	105.2	-10.52	8.044	49.98	8.044
12	-57.4	106.6	-9.721	7.245	47.679	7.245
13	-53.99	108	-8.997	6.521	44.993	6.521
14	-49.79	109.2	-8.337	5.861	41.453	5.861
15	-46.43	110.4	-7.734	5.258	38.696	5.258
16	-43.19	111.5	-7.18	4.704	36.01	4.704
17	-40.07	112.6	-6.669	4.193	33.401	4.193
18	-38.51	113.6	-6.198	3.722	32.312	3.722
19	-35.58	114.6	-5.759	3.283	29.821	3.283
20	-31.3	115.6	-5.348	2.872	25.952	2.872
21	-28.81	116.5	-4.968	2.492	23.842	2.492
22	-25.73	117.4	-4.61	2.134	21.12	2.134
23	-23.87	118.3	-4.277	1.801	19.593	1.801
24	-21.21	119.1	-3.963	1.487	17.247	1.487
25	-18.31	119.9	-3.667	1.191	14.643	1.191
26	-15.36	120.7	-3.39	0.914	11.97	0.914
27	-12.34	121.5	-3.128	0.652	9.212	0.652
28	-9.216	122.2	-2.881	0.405	6.335	0.405
29	-5.951	122.9	-2.647	0.171	3.304	0.171
30	-2.476	123.6	-2.476	0	0	0

Note:

1. The 2PTE inlet conditions are, H=-2.476 KJ/Kg; 2PTIT=123.6K; S= 3.97 KJ/Kg-K and 2PTIP= 30Bar; X=0
2. The 2PTE outlet conditions are shown in the table with respect to the different outlet pressures from 1bar to 30bar.

2.4.3.4: Heat exchanger

The extent of heat exchange will be considered as ideal, meaning that heat exchange is perfect and possible provided that the temperature difference between the streams allows the flow of available enthalpy between them. That is not the practical approach but because this thesis is exploring the ideal cycle, heat transfer will be assumed as perfect between different streams.

2.5: Secondary Heat Transfer Systems

These are a set of systems which will assist the CHEC system to meet its heat balance requirements. The units operate by sourcing heat from the different streams of the CHEC cycle. This section briefly described the thermodynamic aspects of two systems referred to as the COMPEX unit and FHRU.

2.5.1: Compression-Expansion Unit (COMPEX)

This is basically an arrangement of a set of equipment's playing the role of a heat pump. The COMPEX system uses the GTOS as its heat source and the CLS as its heat sink. The main components of this system are a compressor, and a GTE that is hooked with an oil brake and a compressor on the same shaft. The arrangement of the set-up is shown in the sketch of Figure 2.5.1a while the T-S diagram is shown in Figure 2.5.1.b.

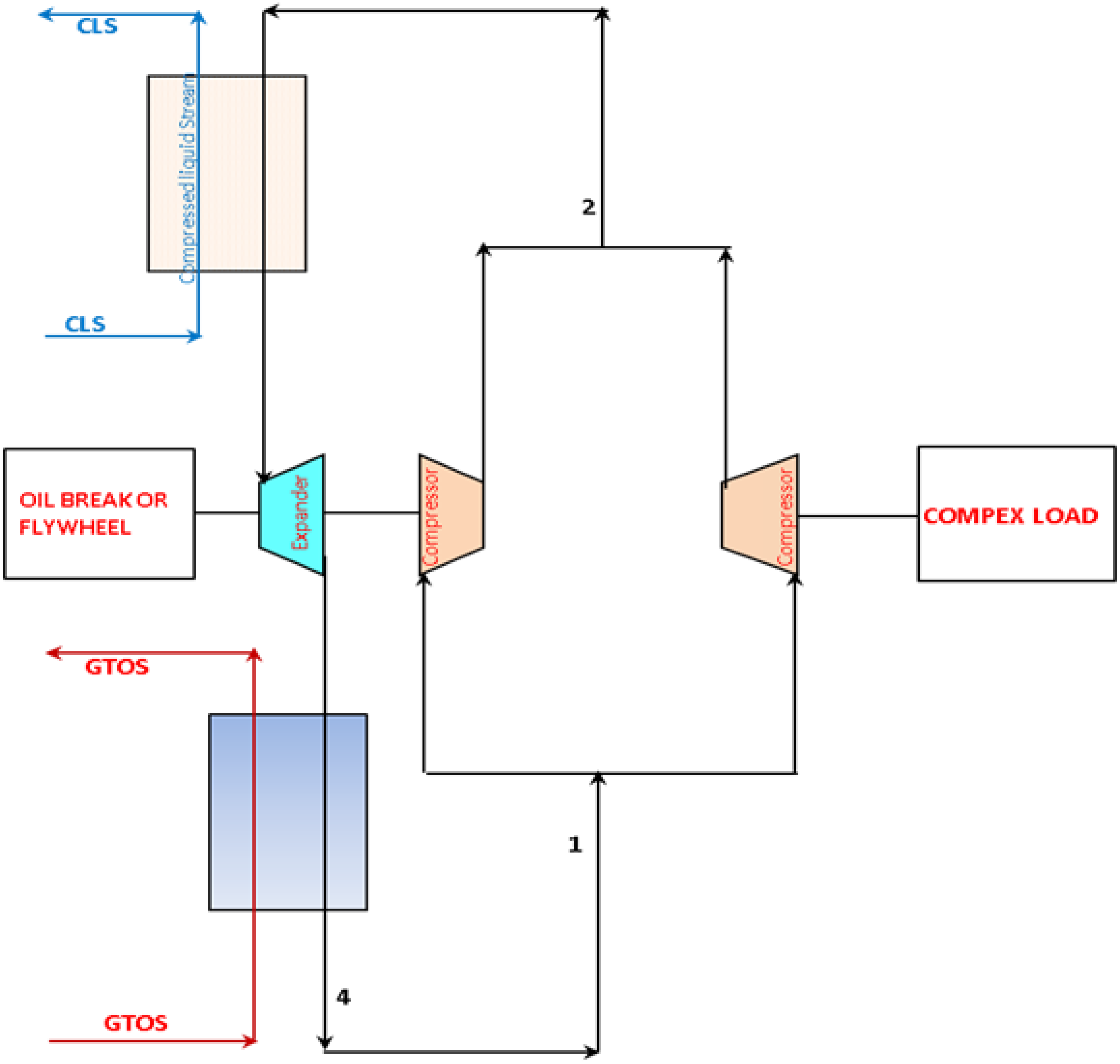


Figure: 2.5.1.a: A sketch of a COMPEX system with all its primary components

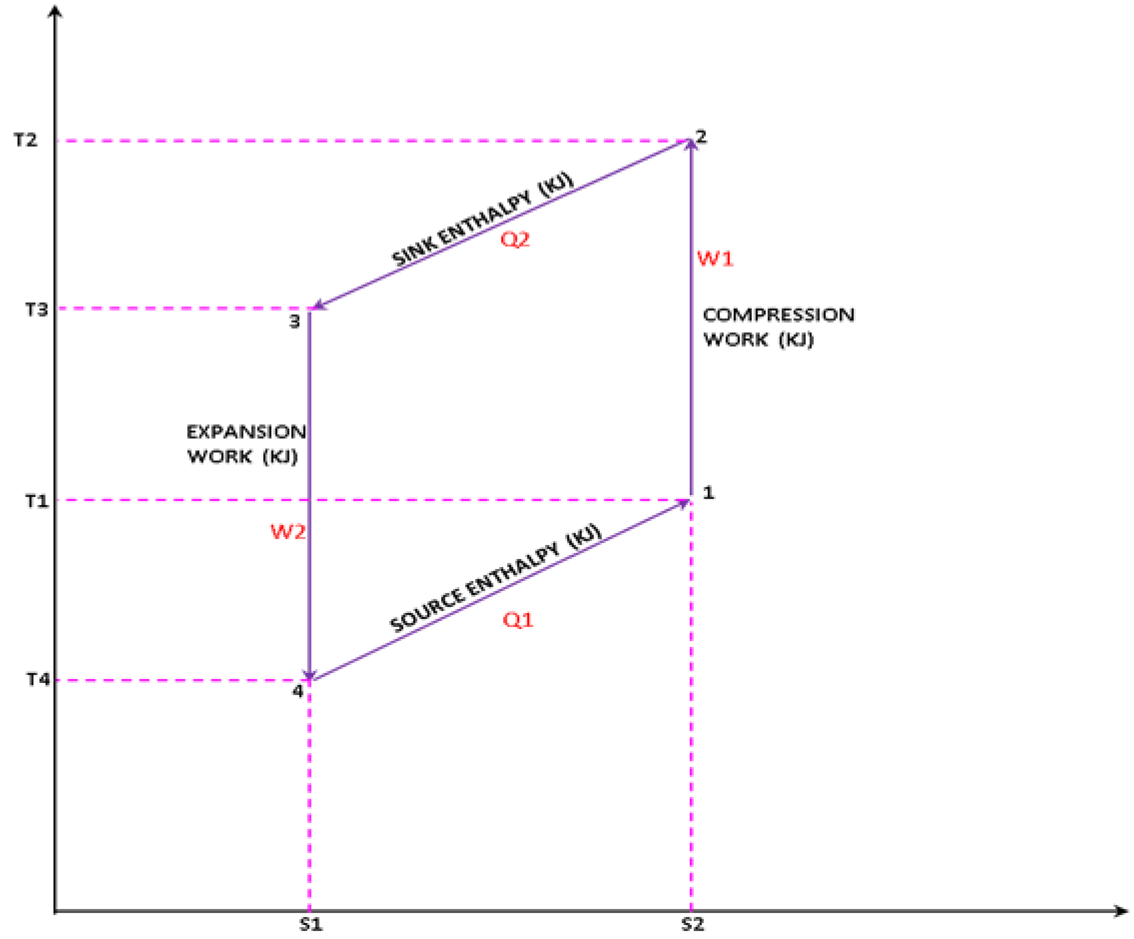


Figure 2.5.1.b: T-S diagram of a COMPEX unit

2.5.1.1: Thermodynamic analysis of a COMPEX system (Units in KJ/Kg)

Referring to figures 2.5.1a and b respectively:

$$Compressor\ Work\ (W1) = (H2 - H1)$$

$$Expander\ Work\ (W2) = (H3 - H4)$$

$$Heat\ Source\ (Q1) = (H4 - H1)\ \ldots.\ from\ GTOS\ or\ 2PTOS$$

$$Heat\ Sink\ (Q2) = (H2 - H3)\ \ldots.\ into\ CLS$$

$$Net\ Work\ input\ (W1 - W2) = \{(H2 - H1) - (H3 - H4)\}\ \ldots.\ at\ steady\ state$$

2.5.1.2: Automatic Load uptake (COMPEX operation)

This involves the automatic transfer of some of the motor driven compressor load to the expander driven compressor. At the beginning, the motor driven compressor is started while the expander driven compressor is in no load mode. In this situation the input to the expander is totally supplied by the motor driven compressor and cold is generated by the oil brake being fully engagement to the expander. Once the cold cycle is established, the oil-brake starts to disengage automatically while the expander driven compressor engages in providing the brake by taking up some of the motor driven compressor load without affecting the expander input. This happens with a concurrent automatic operation of the inlet and outlet valves of the both compressors.

The oil-brake continues to disengage relative to the expander driven compressor increase in load uptake until when it has taken up its maximum share of the load. The maximum share of the expander driven compressor is the power provided by the expander itself operating on the same shaft. The oil break becomes completely disengaged when the expander driven compressor if fully loaded thus providing the required braking load for the expander’s cold creation process. The whole process can be summarily described in Figure 2.5.1.2.

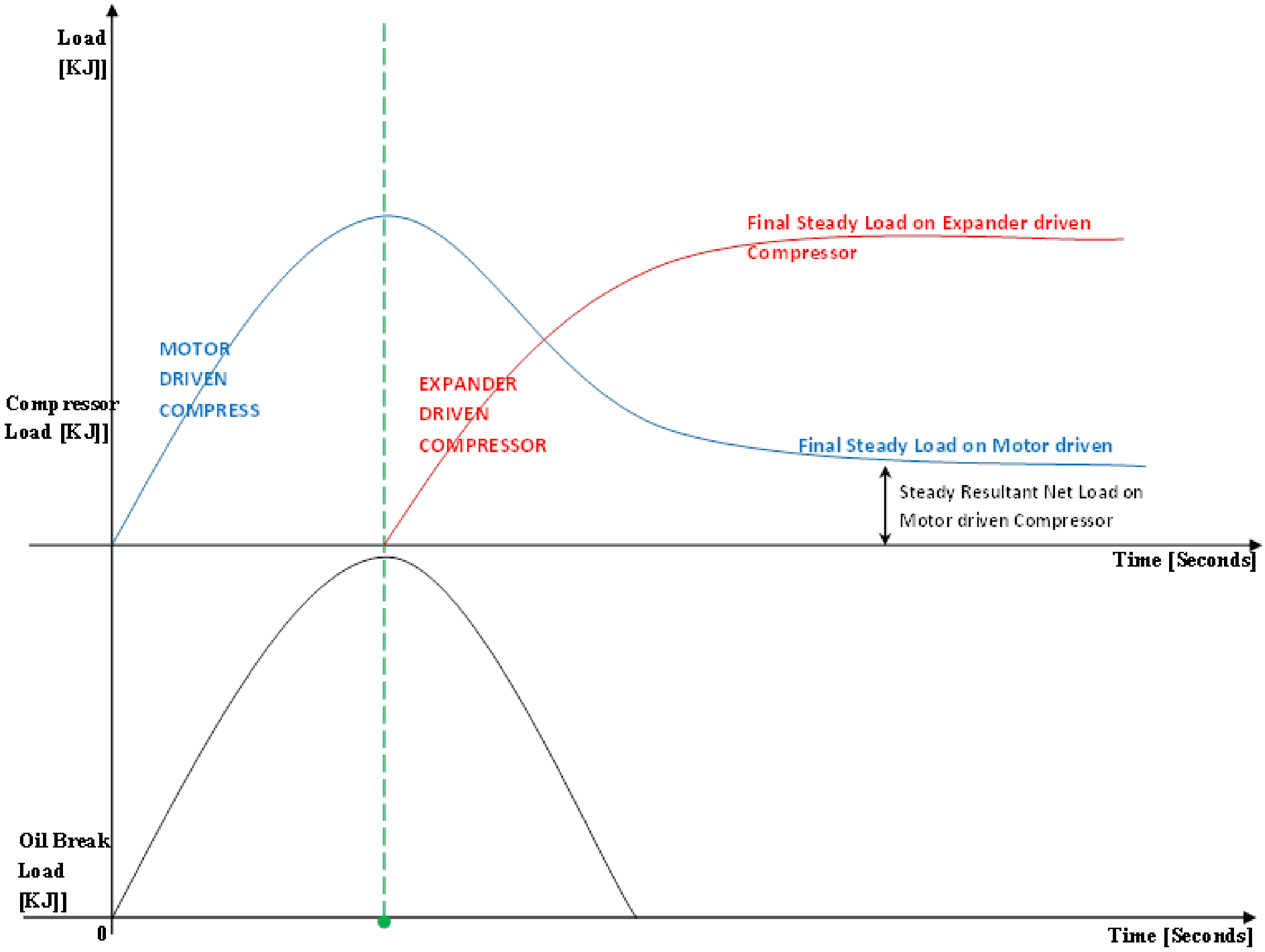


Figure 2.5.1.2: Trends showing the automatic load uptake by the expander driven compressor

2.5.1.3: Filling of fluid to the COMPEX unit

High pressure nitrogen gas will be introduced into the COMPEX unit as shown below. Ambient heaters will be utilised in the vaporisation of liquid nitrogen under constant pressure (the required COMPEX suction drum pressure) to give a mass of fluid required by the system during start-up. Once the COMPEX unit is in full operation, the start-up feed pump switches to make-up mode where it shall regularly supply make-up liquid for mass losses from the system.

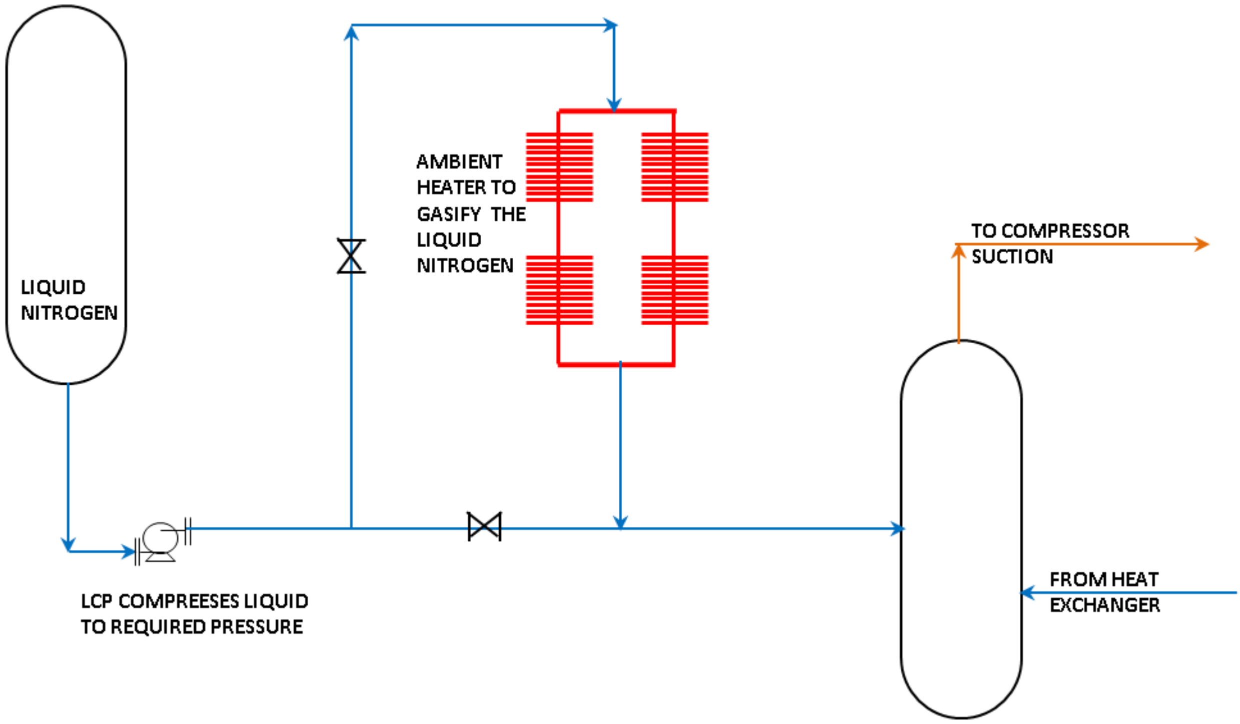


Figure 2.5.1.3: Flow diagram of a SHTS initial filling

2.5.1.4: COMPEX system nomenclature

A general COMPEX system does not have a fixed configuration in terms of its operational parameters. All other thermodynamic property variables depend on the systems operational pressures. It therefore becomes necessary to have a naming system for the identification of individual COMPEX units in terms of the highest and lowest operational pressures.

A naming system which starts with the letters “EX’’, followed by the highest then the lowest unit pressures representing the pressure cycle in which the unit operates and that serves in identifying different systems based on their configurations. The highest pressure being the pressure at which the COMPEX unit transfer heat to the sink (CLS) while the lowest being the pressure at which the unit sources heat from the GTOS. The cycle on figure 2.5.1.a can therefore be easily identified by this system as EX-P2-P1, where;

- EX- Stands for COMPEX
- P2- The highest unit operational pressure
- P1- The lowest unit operational pressure

2.5.2: Forced Heat Rejection Unit (FHRU)

The FHRU is another SHTS which operates similar to the COMPEX unit, but in this case, the GTE is replaced with a 2PTE in order to accommodate the two phase status of the fluid undergoing expansion. The expansions final state is always within the wet vapour region where the fluid will be a mixture of both liquid and vapour. The liquid fraction is forced to vaporise by the effect of a compressor suction draw-off hooked on the downstream of the 2PTE. The process occurs in a heat exchanger where the heat of vaporisation is sourced from the GTOS. As the FHRU stream is forced to vaporise by the low pressure created by the suction effect of the compressor, the GTOS loses heat equivalent to the amount of liquid vaporised and thus condenses to the same degree. The compressor suction node is usually set to be in a state of saturated vapour and beyond to avoid any liquid being sucked by the compressor. The net amount of cold sourced from GTOS shall be the enthalpy difference between the compressor suction (H1) and that of the 2PTE outlet (H4). Thermodynamic analysis an ideal FHRU can be found in section 2.5.2.1 with the help of Figures 2.5.2.a and b.

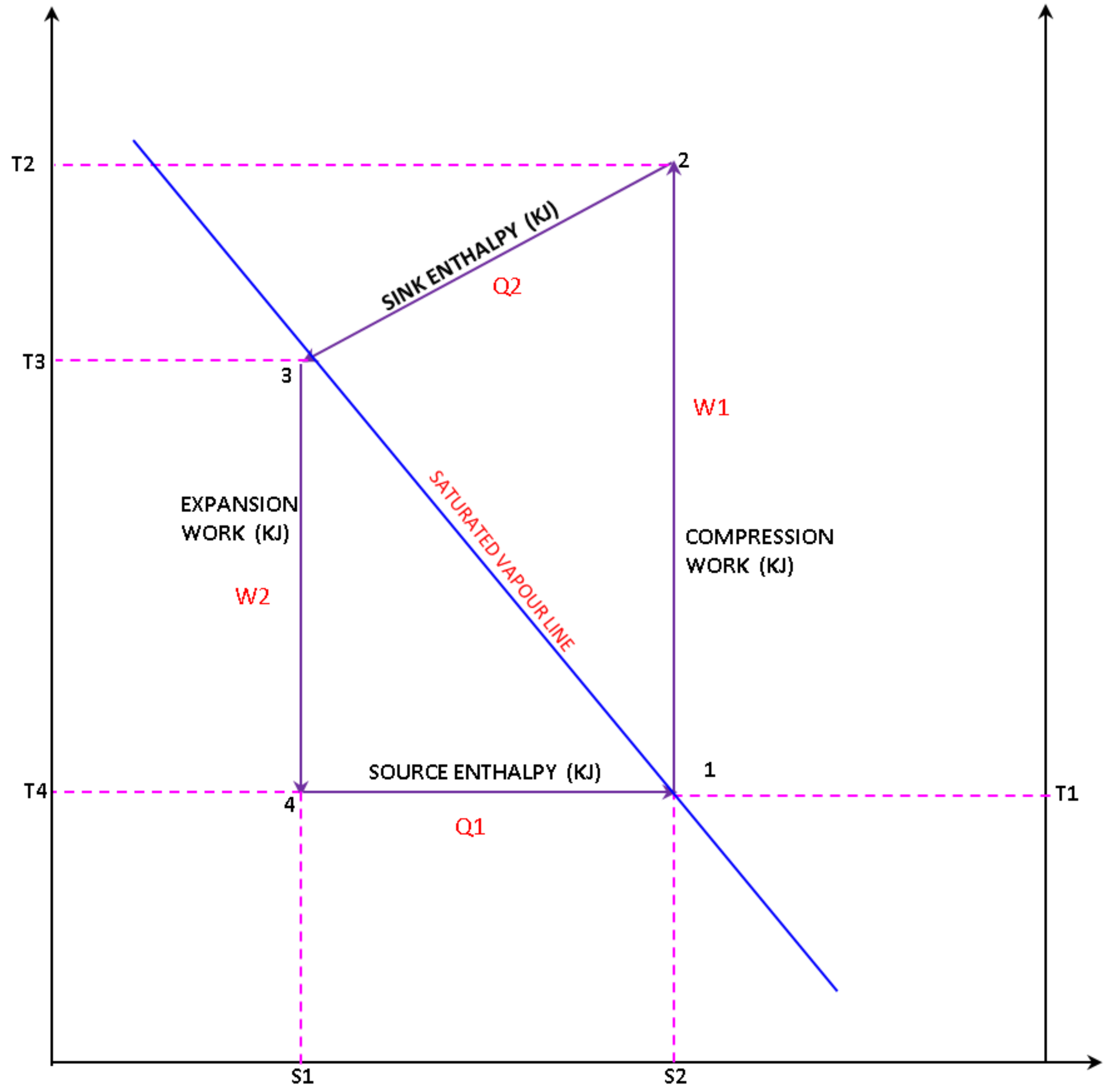


Figure 2.5.2.a: T-S of an ideal FHRU

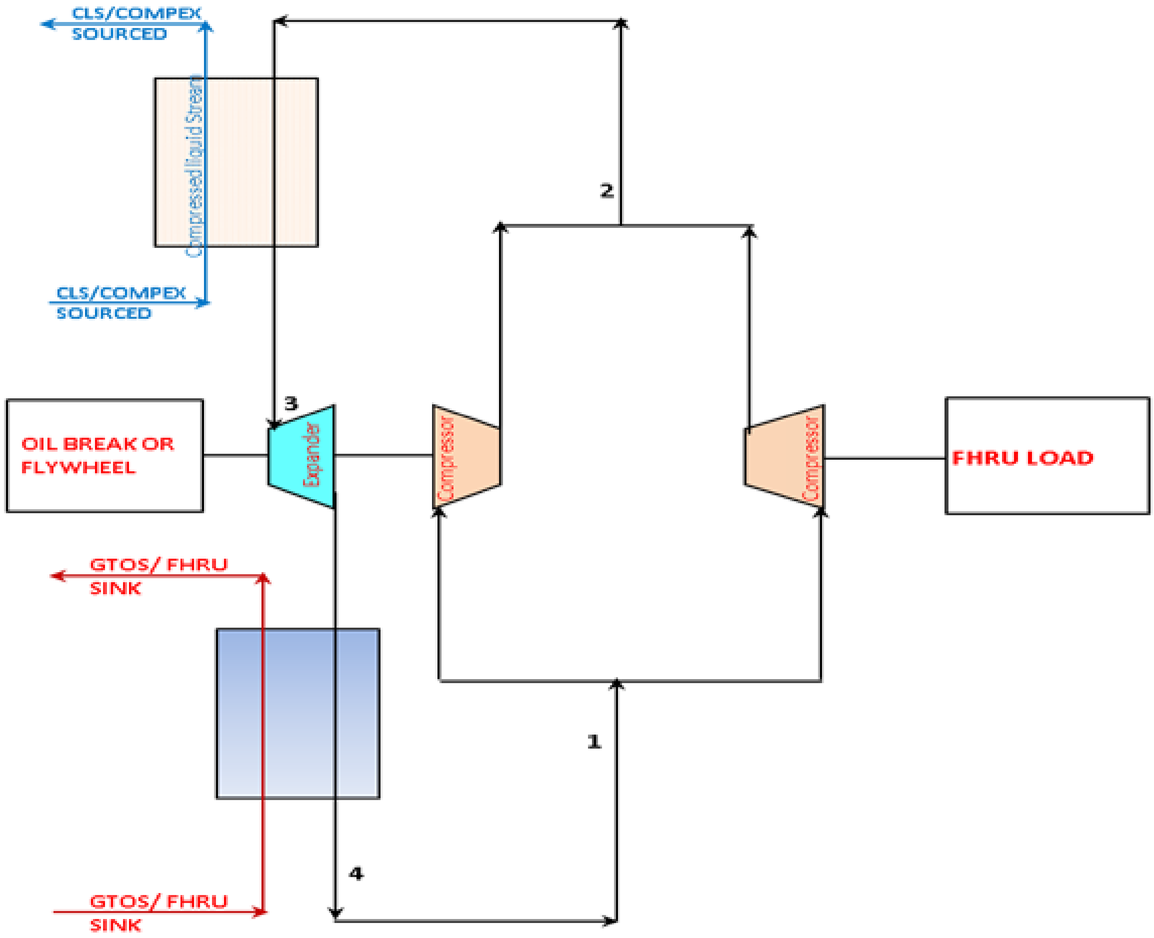


Figure 2.5.2.b: Flow diagram of an ideal FHRU showing all its primary components

2.5.2.1: Thermodynamic analysis of a FHRU system (Units in KJ/Kg)

Referring to figures 2.5.2.a and b respectively:

$Compressor\ Work\ (W1) = (H2 - H1)$

$Expander\ Work\ (W2) = (H3 - H4)$

$Heat\ Source\ (Q1) = (H4 - H1) \ldots.from\ GTOS\ or\ 2PTOS$

$Heat\ Sink\ (Q2) = (H2 - H3) \ldots.into\ CLS$

$Net\ Work\ input\ (W1 - W2) = \{(H2 - H1) - (H3 - H4)\} \ldots.at\ steady\ state$

2.5.2.2: Automatic Load uptake (FHRU operation)

Apart from using a different expansion turbine, the operation of this unit is similar to that of a COMPEX unit as described in part 2.5.1.2. Refer to this part for more details.

2.5.2.3: Filling of fluid to the FHRU

The process of filling up is the same as that described in part 2.5.1.3 for the COMPEX unit. Refer to that section for more details.

2.5.2.4: FHRU system nomenclature

As in part 2.5.1.4, a general FHRU system does not have a fixed configuration in terms of its operational parameters. All other thermodynamic property variables depend on the systems operational pressure limits. It is prudent to have a system of naming each individual FHRU system in terms of its highest and lowest operational pressures.

A naming system which starts with the acronym “FHRU’’, followed by the highest then the lowest unit pressures representing the pressure cycle in which the unit operates. That serves to effectively identify different systems based on their operational pressure configurations. The highest pressure being the pressure at which the FHRU unit transfer heat to the sink (CLS) while the lowest being the pressure at which the unit sources heat from the GTOS. The cycle on figure 2.5.2.a can therefore be easily identified by this system as FHRU-P2-P1, where;

- FHRU- Units acronym
- P2- The highest unit operational pressure
- P1- The lowest unit operational pressure

2.6: Auxiliary units

These are support units for the power plant operational activities. The ASU is of extra importance as an integral part of the power cycle because it provides make-up nitrogen to the power plant. The ASU has been given much attention in part [2.6] because of the key role it plays in the CHEC power cycle while the rest have been briefly described for knowledge purposes. The energy demand by both the IAU and the UWU are not much and as such have not been accounted for in the energy balancing of the CHEC systems in this study. The auxiliary units have been classified into the following and each is described in the sections below.

- Air Separation Unit (ASU)
- Instrument Air Unit (IAU)
- Utility Water Unit (UWU)

2.6.1: Air Separation Unit (ASU)

(Ref: Linde-Engineering; ASU process)

The ASU as an integral part of the CHEC power cycle has been analysed thermodynamically to establish the extent of heat and work demands it may impart in its operation to provide the required amount liquid nitrogen. The analysis done is on an ideal open air cycle with ambient air intake, oxygen vaporisation back to ambient with liquid nitrogen yield fed to the LNBT.

2.6.1.1: Purpose of an ASU

The main purpose is to provide a source of liquid nitrogen for continuous make-up to offset fluid losses of between 1% and 5% of the mass of nitrogen fluid in circulation.

2.6.1.2: Description of ASU operation

The unit sucks ambient air which is first filtered and then inter-cooled at several stages of compression by utilizing the residual CLS cold left behind by the SHTS. In this way, ambient air is compressed from 1 to 10bars and then precooled to condense any moisture before passing it through a purification unit where CO2, residual moisture and any unwanted impurities are removed. It is further cooled to 200K (assuming the residual cold is from 200K and above) before being expanded in a COMPEX unit to 2 bars. The expander outlet stream is further cooled by an expanded stream of liquid oxygen extracted from the bottom of the liquid air separator vessel as it blows off to the atmosphere. This condenses the oxygen fraction of the air stream as it enters the air separator vessel where nitrogen vapour separates from liquid oxygen.

The liquid oxygen is expanded to 1bar and vented off to the atmosphere as a stream which serves to cool the incoming air stream from the expander unit before it enters the air separator vessel. The cooling provided to the expanded air stream is to be at 1K below the condensation temperature of oxygen at 2bars to ensure that the oxygen is subcooled and completely liquefied. This assures that there will be no oxygen escape into the nitrogen vapour stream.

The nitrogen vapour stream is allowed off the air separator vessel from the top and is further subcooled by heat exchange with a side stream from the FHRU. Finally, the subcooled nitrogen stream is expanded to the requisite LNBT or make up vessel ready to be utilised for make-up against system losses.

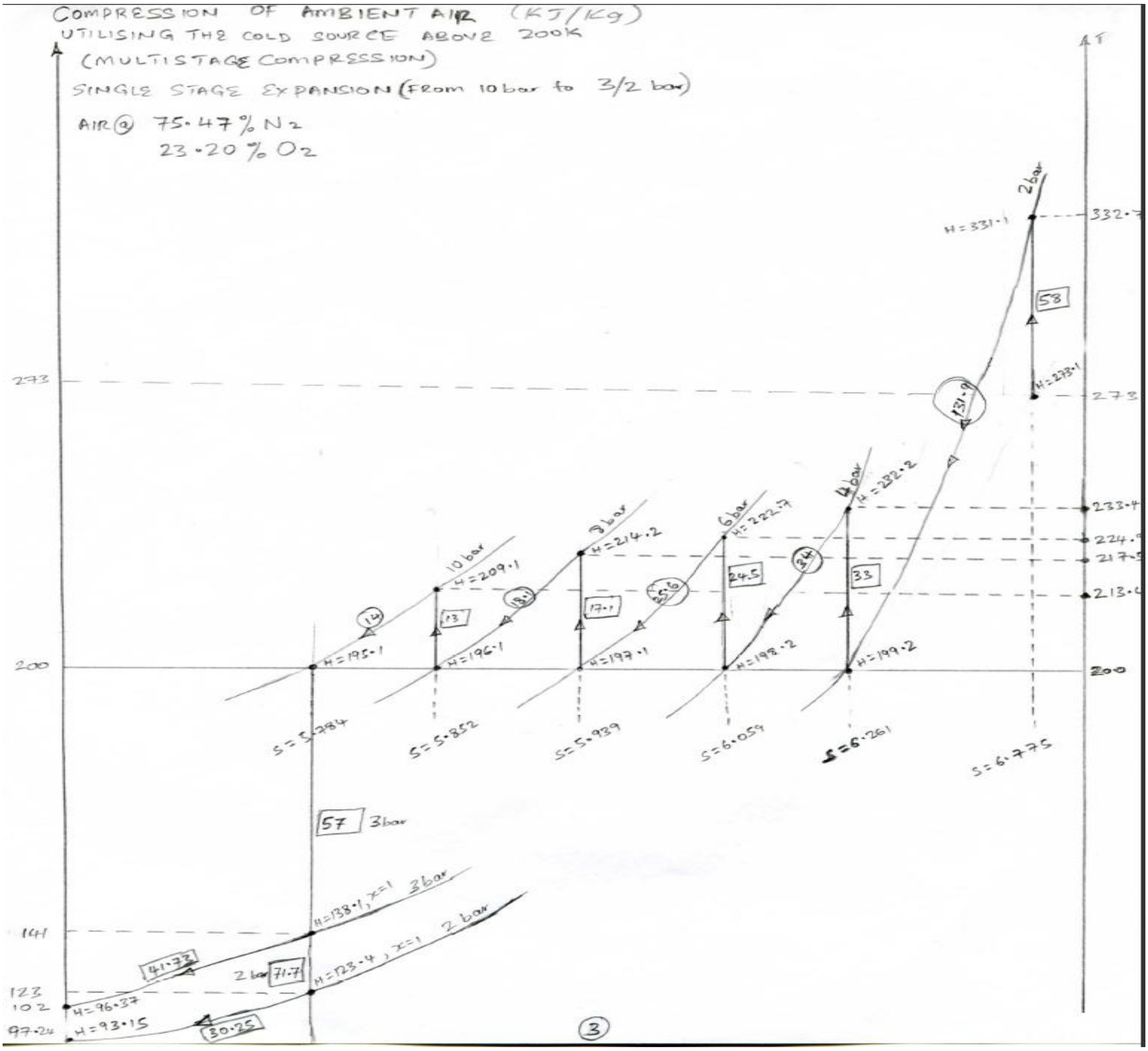


Figure 2.6.1.2.a: T-S diagram of a typical ASU

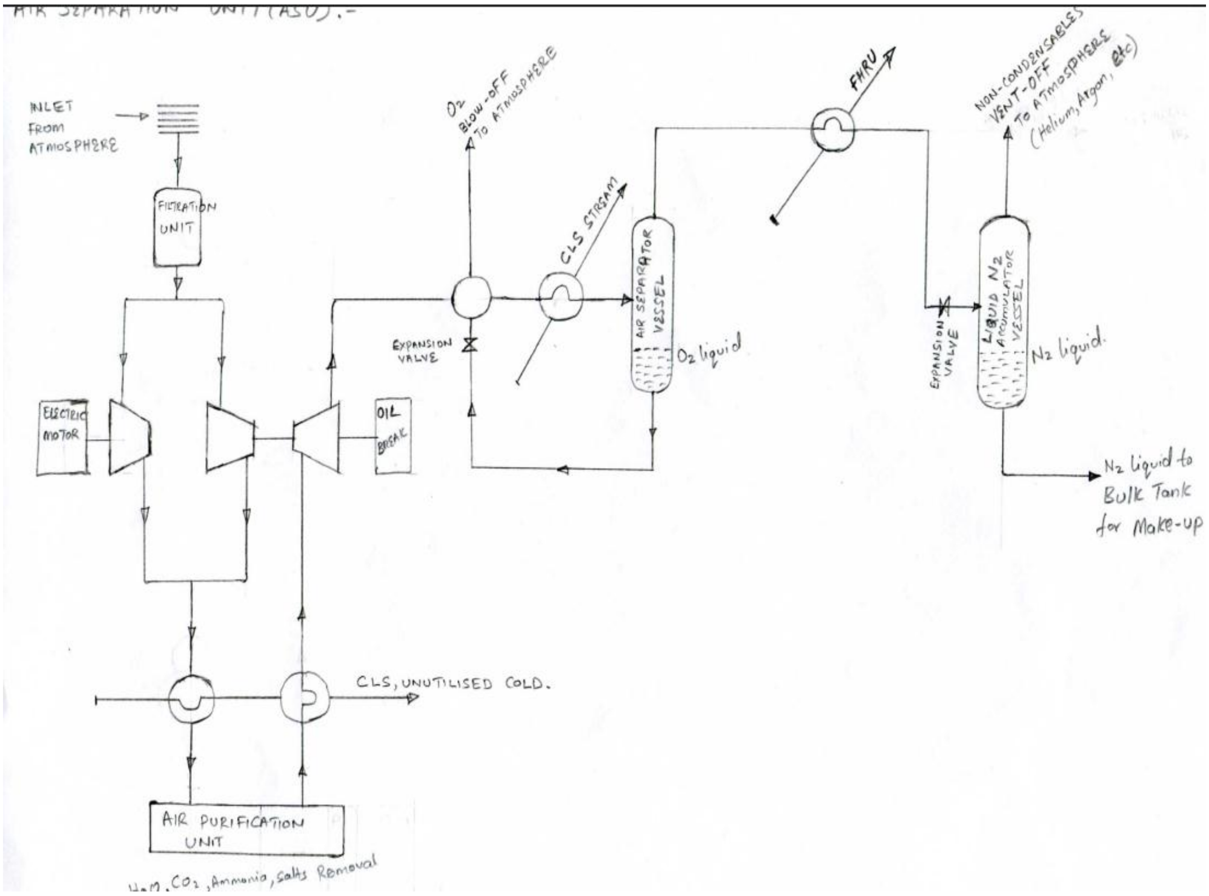


Figure 2.6.1.2.b: Flow diagram of a typical ASU

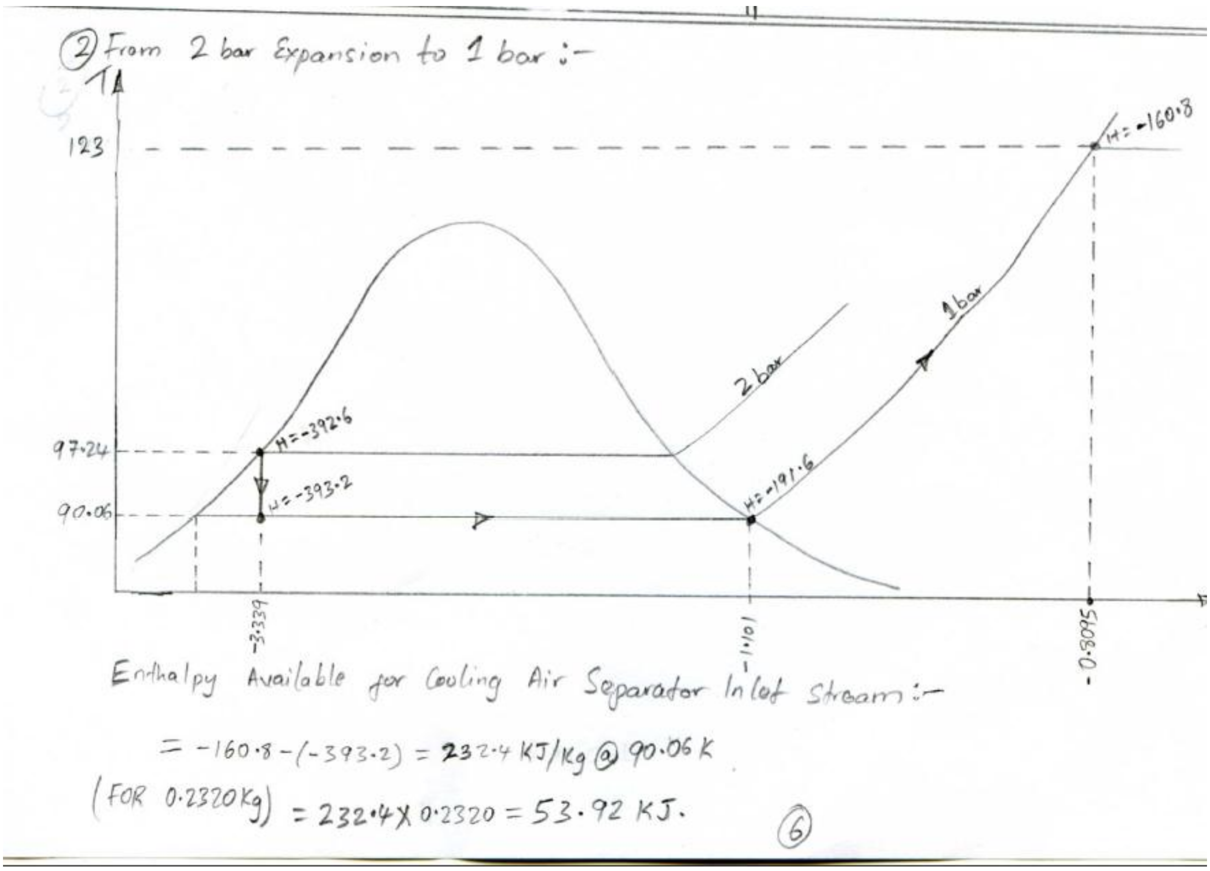
2.6.1.3: Thermodynamic calculations for the ASU Processes

2.6.1.3.1: The air stream

Refer to figure 2.6.1.2.a and b

2.6.1.3.2: The oxygen stream

Refer to figure 2.6.1.3.2 and 2.6.1.2.b



② From 2 bar Expansion to 1 bar :-

Enthalpy Available for Cooling Air Separator Inlet Stream :-

$= -160.8 - (-393.2) = 232.4$ KJ/Kg @ 90.06 K

(FOR 0.2320 Kg) $= 232.4 \times 0.2320 = 53.92$ KJ.

⑥

Figure: 2.6.1.3.2: T-S diagram of the ideal ASU oxygen stream

2.6.1.3.3: The nitrogen stream

Refer to figure 2.6.1.3.3 and 2.6.1.2b

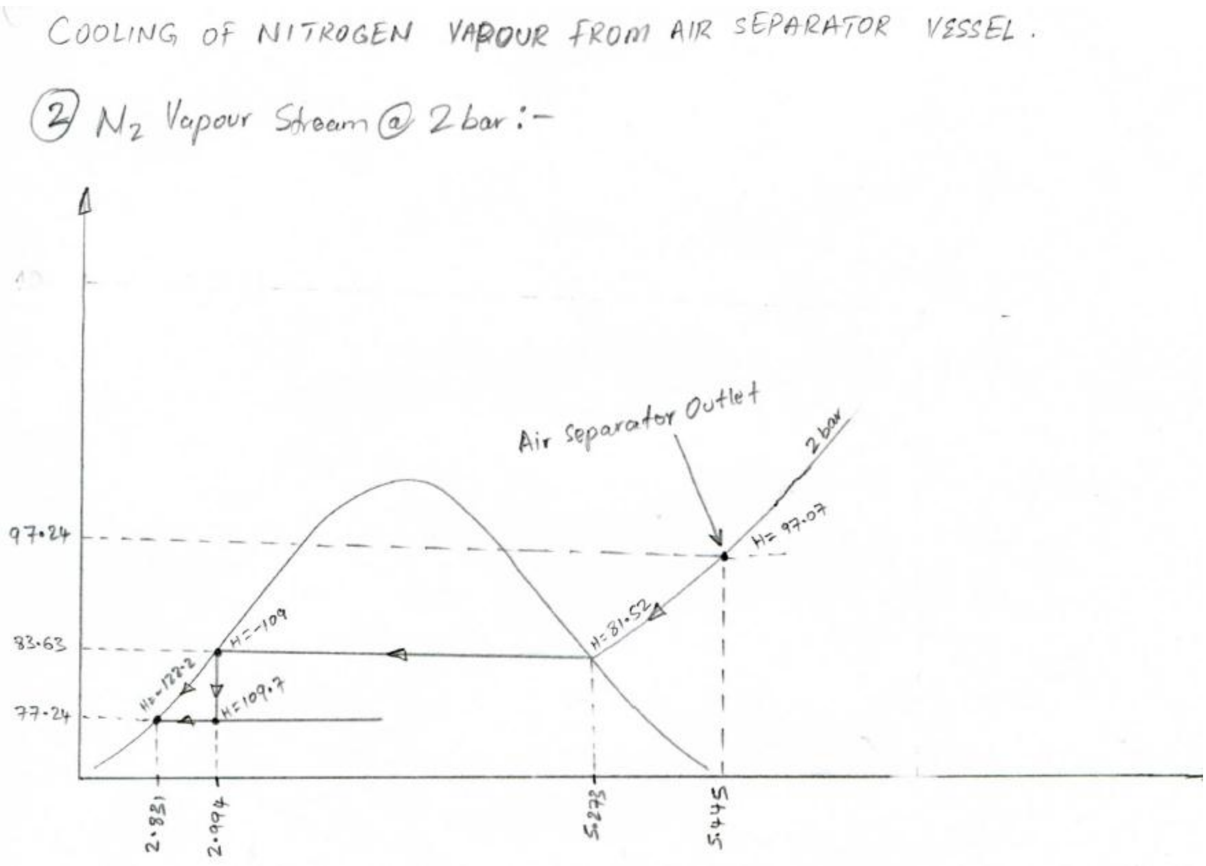


Figure: 2.6.1.3.3: T-S diagram of the ideal ASU nitrogen stream)

2.6.2: Instrument air Unit

This is a small but necessary unit required in order to enable the operation of pneumatic instruments and valves. The operation of a CHEC power plant demands a lot of automatic operation which can only be actualised by the presence of this unit at site. This unit may be embedded as a function of the air purification section of the ASU or may be independent. The design of this unit is site specific and cannot be predicted from theory. The unit operates by filtration, compression, cooling and drying of ambient air, which is usually the fluid medium in any instrument air unit. The air supply pressure of this unit depends on the plant specifications.

The role of this unit is to ensure a reliable and secured supply of instrument air for start-up, operation and safe shutdown of the plant. Tailor made instrument air supply units are available in the market as complete packages and a choice of any configurations can be sourced from suppliers and manufactures without duress. The energy consumption of this unit is not much and it hasn't been considered in any the calculation in this thesis.

2.6.3: Utility water Unit

This is a small unit set-up to ensure the site has adequate supplies of water to meet the industrial water requirements of the site where a CHEC power cycle to operate. This unit is being considered as an essential part for the operation of any industrial activity such as the one under this study. Its worth to mention it but its energy consumption is not of much significance for this study. However, as part of any future work, it may be worthy to include it as part of the units that contribute to work input to the system.

CHAPTER 3

ANALYSIS OF BASECASE CHEC OPERATIONAL SYSTEMS

Chapter contents

Chapter overview

The thermodynamic properties and evaluation of enthalpy changes in part [2.4.3] bring this study to a point where the ideal heat and work of demands of a specified CHEC power cycle can be determined by directly referencing to the figures appearing on the summary tables. Ideal CHEC power cycles of different configurations can be analysed effectively to establish the specific demands of heat and work before adjusting the demands by using the appropriate SHTS in forming a reliable cyclic power system. There are many CHEC power cycle configurations which can be analysed, this thesis the will analyse three cycles described by the acronyms C-120-30-1, C-120-34-30@113.1K and C-120-34-30@123.6K. These cycles will be analysed in detail in parts 3.1(C-120-30-1), 3.2 (C-120-34-30@113.1K) and 3.3(C-120-34-30@123.6K) respectively.

Please refer to the list of abbreviations for the full meaning of the acronyms used in this thesis.

3.1: Analysis of base case C-120-30-1

Overview of C-120-30-1 basecase analysis

This analysis is carried out to check and determine the heat and work characteristics of C-120-30-1 at its base state. The aim is to determine the cooling gap which remains after the designated SHTS have worked to their maximum. Results are briefly shown below and discussed further in chapter 7.

Table: 3.1.a: Summary of C-120-30-1 base case analysis

SYSTEM C-120-30-1	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+86.5	
2PTE	+31.814	
Total Work output		117.414
Work input		
LCP pump	-14.5	
EX-80-29	-26.46 (Ref: 3.1.1.1)	
FHRU-100-30	-15.936(Ref: 3.1.1.2)	
FHRU-30-1	-21.669(Ref: 3.1.1.3)	
Total Work input		-78.565
C-120-30-1 Net Work		**38.849**
Heat Supplied (from ambient) (255.2-185.6) Ref 3.1	69.6	
Heat Rejected at 1 bar (balanced from liquid nitrogen reserve or LNGR) {(-122.2-[-34.29])+58.06}	-29.85	
Net heat Supplied to cycle		39.75
COP/Work ratio		**1.494**
Direct cooling ability ratio ([30.8+105.55] / [105.55+65.626+87.91])=(136.35/259.384)		0.526
SNC(Kg)		0.15

Figure 3.1.a below shows the marked heat zones of the base case C-120-30-1

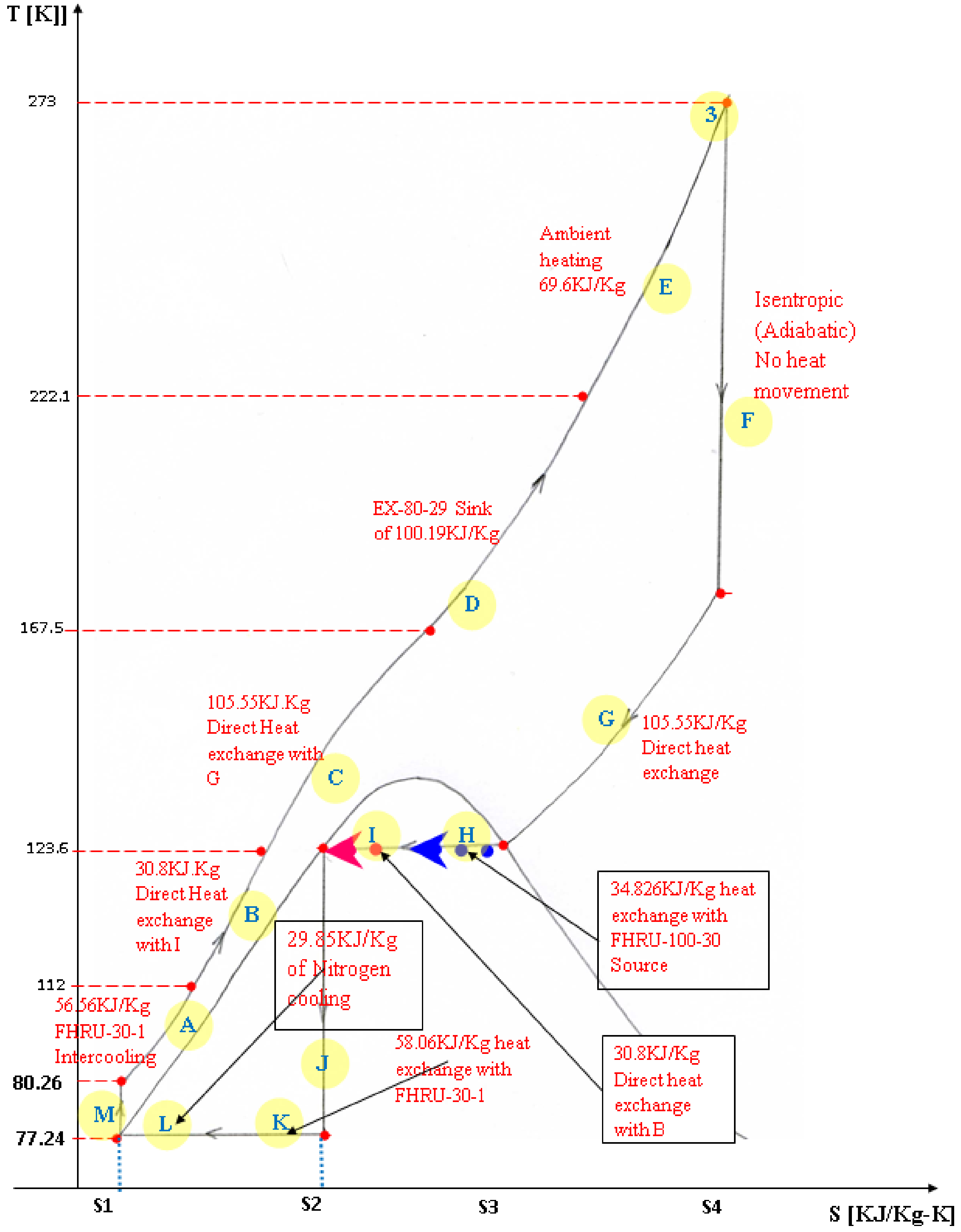


Figure: 3.1.a: T-S diagram of profiled heat zones for C-120-30-1 base case

The total enthalpy absorbed from available CLS is 100.19KJ but the useful output at 123.6K is 57.795KJ.

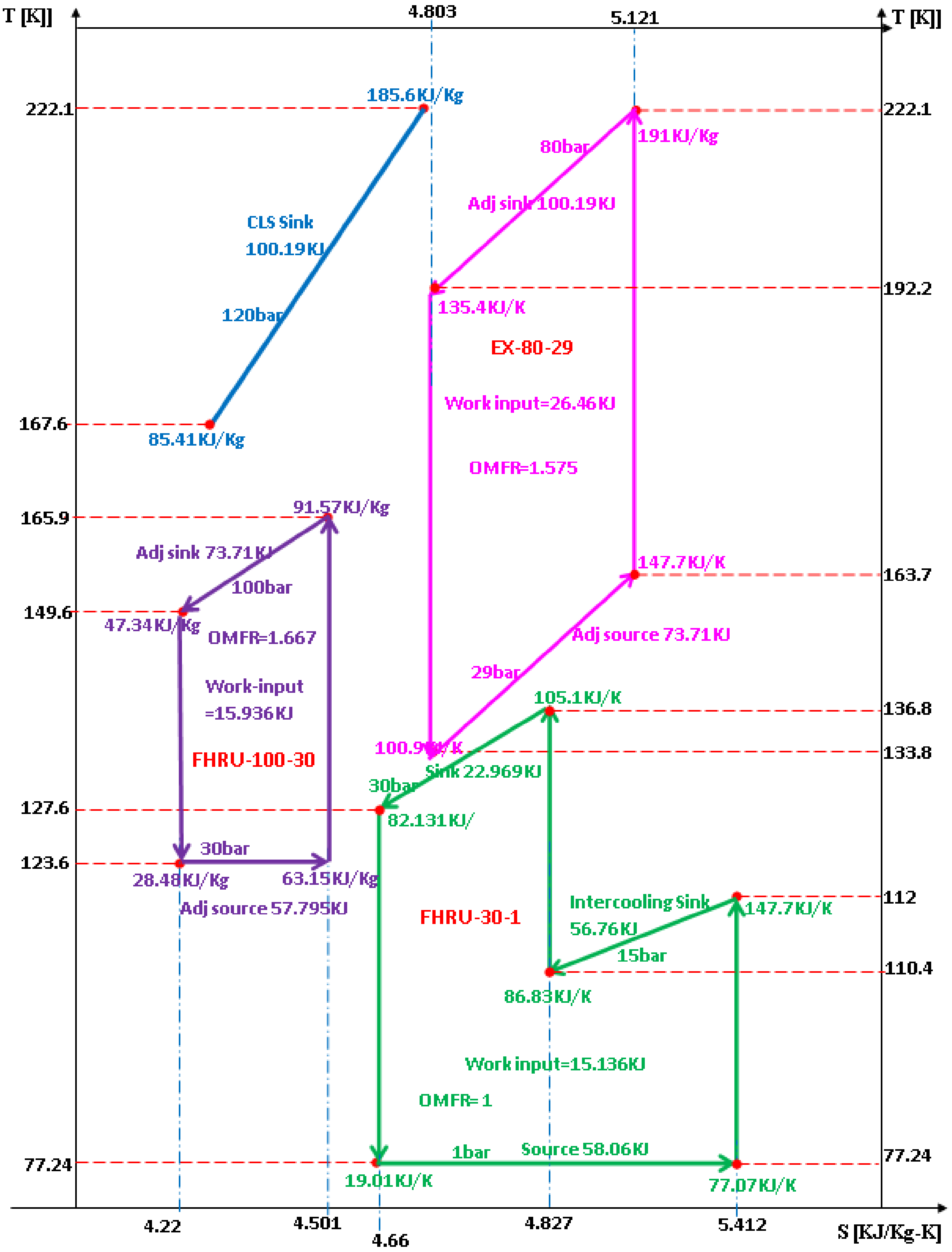


Figure: 3.1.b: Combined T-S diagram of all SHTS showing the chain of enthalpy pull down for C-120-30-1 base case.

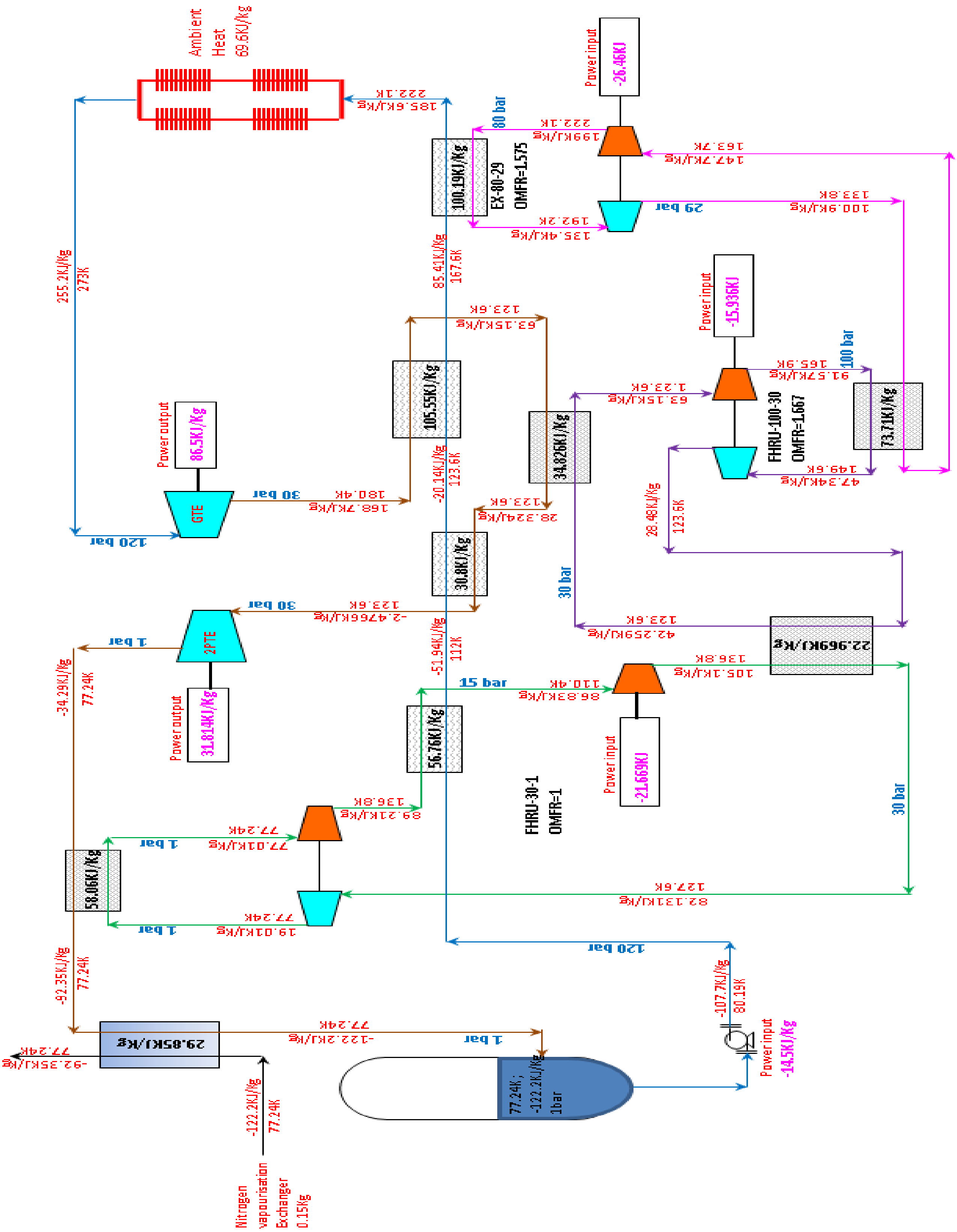


Figure: 3.1.c: Flow diagram of a balanced base case C-120-30-1.

3.1.1: C-120-30-1 heat and work balance at steady state

Table: 3.1.1.a; System heat and work balance for C-120-30-1 base case

Location of heat/work demand (node)	Pressure (bar)	Heat demand (KJ/Kg)	Work done (KJ/Kg)	Temperature Conditions (K)	Thermodynamic Process (Ideal)
1>2	1>120	0	-14.5	77.24>80.19	Isentropic(adiabatic) Compression
2>3	120	+362.9	0	80.19>273	Constant Pressure Heating
3>4	120>30	0	+86.5	273>180.4	Isentropic(adiabatic) Expansion
4>5	30	-105.55	0	180.4 to 123.6	Constant Pressure Cooling
5>6	30	-65.626	0	123.6	Constant Pressure Condensation
6>7	30>1	0	+31.814	123.6>77.24	Isentropic(adiabatic) Expansion
7>1	1	-87.91	0	77.24	Constant Pressure Condensation

The system requires the support of SHTS to enable it meet its heat balance. Each of the SHTS is considered independently in the sections that follow. Table 3.1.1.b is a summary of results obtained after calculating the work input to all SHTS of the cycle. The sections have been classified under the following headings in order to have a smooth follow-up of the heat and work balance;

- COMPEX unit EX-80-29
- FHRU-100-30
- FHRU-15-1

Table: 3.1.1.b; Summary of SHTS work input for C-120-30-1 base case

SHTS type	Work input (KJ/Kg)	Reference
EX-80-29	-26.46	3.1.1.1
FHRU-100-30	-20.325	3.1.1.2
FHRU-30-1	-21.669	3.1.1.3

3.1.1.1: COMPEX unit EX-80-29

Table: 3.1.1.1.a: Design configuration of EX-80-29 with adjustment for C-120-30-1 basecase operation

	(KJ/Kg)	(KJ/Kg)	CLS Sink available (KJ/Kg)	System adjustment for maximum utilization of available CLS sink(KJ/Kg)
Compression Work (199-147.7)	51.3			
Expansion Work (135.4-100.9)	34.5			
Net Work (51.3-34.5)		16.8		26.46 (Ref 3.1.1.1.3)
Heat Sink (119-135.4)		63.6	100.19 (Table 3.1.1.1.d)	100.19 (Ref 3.1.1.1.3)
Heat Source (147.7-100.9)		46.8		73.71 (Table 3.1.1.1.1)]

Note:

CLS sink available is calculated as the enthalpy at 120 bars between the temperatures of 222.1K and 167.6K, where167.6K is the temperature at which the ideal cooling demand for GTOS (zone G) of C-120-30-1 is fully matched by the heating demand of zone C. This temperature range allows for heat exchange between CLS and the high pressure stream (@80bar) of EX-80-29. The values are shown in table 3.1.1.1.b.

Table: 3.1.1.1.b: Sink enthalpy available from CLS at the stated temperatures for C-120-30-1 basecase

CLS temperature (K)	CLS Enthalpy (KJ/Kg)	CLS Sink available (KJ/Kg)
222.1	185.6	
167.6	85.41	
		100.19

3.1.1.1.1: EX-80-29 System Heat source demand:

The heat source of EX-80-29 is determined by the sink demand of FHRU-100-30. Table 3.1.1.1.1 below summarises how the heat source for EX-80-29 is arrived at.

Table: 3.1.1.1.1: Table of sink capacity of FHRU-100-30.

SHTS	**Sink Demand (KJ/Kg)**	**Adjusted Sink (KJ/Kg)**
FHRU-100-30	46.8 (Table 3.1.1.2)	44.429

Note:

$$Adjusted\ (FHRU-100-30)\ Heat\ Sink = Heat\ source\ demand\ (EX-80-29)$$

$$Heat\ source\ demand\ (EX-80-29) = 44.429\ KJ/Kg$$

3.1.1.1.2: EX-80-29 System adjustment:

The design of EX-80-29 has a source capacity of 46.8 KJ/Kg [Figure 3.1.1.1.a] which is not exactly matching with the sink demand of 44.429 KJ/Kg [Table 3.1.1.1.1] from FHRU-100-30. It is therefore apparent that an adjustment of mass flow rate must be done so as to enable it generate a heat source which exactly matches the required FHRU-100-30 sink demand. The operating system of EX-80-29 is therefore adjusted to utilize the available CLS sink of 100.19KJ/Kg so as to provide the required heat source of 44.429 KJ/Kg and extra cold for of 38.882KJ/Kg for FHRU-30-1 [Table 3.1.1.3] as shown in the calculations below [3.1.1.1.3].

3.1.1.1.3: EX-80-29 calculations

Note: Figures used in these calculations have been extracted from Tables 3.1.1.1.a, b and c. Determination of OMFR therefore is based on the maximum utilization of available sink.

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(EX-80-29) design\ sink\ capacity}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{100.19}{63.6} = 1.575$$

$$Adjusted\ Net\ Work = OMFR \times (EX - 80 - 29)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 1.575 \times 16.8 = 26.46\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (EX - 80 - 29)\ design\ source\ (KJ/Kg)$$

$$Adjusted\ Heat\ Source = 1.575 \times 46.8 = 73.71 KJ/Kg$$

3.1.1.2: Forced Heat Rejection Unit FHRU-100-30

The heat required for complete condensation of GTOS at 30 bar/123.6K is calculated as the balance after deducting the sink provided by direct heat exchange with CLS zone C (108.9K and 123.6K [Table 3.1.1.2.1]). The 30.8KJ/Kg sink enthalpy available from CLS is used in condensation by direct heat exchange in a PFHE. The remaining balance of 34.826KJ/Kg is supplied by FHRU-100-30 as part of a dropped down enthalpy from EX-80-29.

Table: 3.1.1.2: Design configuration of FHRU-100-30 with adjustment for C-120-30-1 basecase operation.

	(KJ/Kg)	(KJ/Kg)	EX-80-29 Sink available (KJ/Kg)	System adjustment to match demand (KJ/Kg)
Compression Work	28.42			
Expansion Work	18.86			
Net Work		9.56		15.936 (Ref 3.1.1.2.2)
Heat Sink		44.23	73.71 (Table 3.1.1.1.1)	73.71
Heat Source		34.67		57.795 (Ref 3.1.1.2.2)

3.1.1.2.1: FHRU-100-30 System adjustment:

The design of FHRU-100-30 has a source capacity of 34.67 KJ/Kg which is not matching with the condensation demand of 34.826 KJ/Kg [Table 3.1.1.2.1]. It is therefore apparent that an adjustment of mass flow rate must be done so as to enable it match the required demand. The operating system of FHRU-100-30 is therefore adjusted to supply 34.826 KJ/Kg heat source as shown in the calculations below [Ref 3.1.1.2.2].

Table: 3.1.1.2.1: Condensation cooling demand @30bar

CLS Temperature (K)	CLS Enthalpy (KJ/Kg)	KJ/Kg	Description
108.9	-50.94		After Multistage compression
123.6	-20.14		Up to condensation temperature @30bar
CLS sink available for direct heat exchange with GTOS		30.8	
Balance to complete condensation		34.826	To be supplied by FHRU-100-30
Enthalpy required for complete condensation at 30bar/123.6K		65.626	Table 2.4.3.3.b (63.13-[-2.476])= 65.626

3.1.1.2.2: FHRU-100-30 calculations

Note: Figures used in these calculations have been extracted from Tables 3.1.1.2 and 3.1.1.2.1 respectively. Determination of OMFR is therefore based on FHRU-100-30 heat source demand. The mass of the system is adjusted for maximum utilisation of available CLS sink of 73.71 KJ/K [Table 3.1.1.1.c].

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(FHRU-100-30)\ design\ sink}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{73.71}{44.23} = 1.667$$

$$Adjusted\ Net\ Work = OMFR \times (FHRU - 100 - 30)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 1.667 \times 9.56 = 15.936\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (FHRU - 100 - 30)\ design\ source\ (KJ/Kg)$$

$$Adjusted\ Heat\ Source = 1.667 \times 34.67 = 57.795\ KJ/Kg$$

3.1.1.3: Forced Heat Rejection Unit FHRU-30-1

The FHRU has no adjustment to be made and therefore its configuration will remain intact. This is due to the fact that the enthalpy available for intercooling in multistage compression is limited to the CLS sink available [Appendix C].

Table: 3.1.1.3: Design configuration of FHRU-30-1 for C-120-30-1 basecase operation

		KJ/Kg	KJ/Kg	KJ/Kg	Pressure range (Bar)	Temperature range (K)
Compression Work	Multistage	66.52			1-15	77.24-112
	Single stage	18.27			15-30	110.4-136.8
Net Compression Work			84.79		1-30	77.24-136.8
Expansion Work			63.121		30-1	123.9-77.24
Net Work (FHRU-30-1)				21.669	1-30-1	77.24-136.9-77.24
Heat Sink	Multistage		56.76		1-15	77.24-112
	Single stage		22.969		15-30	110.4-136.8
Net Heat Sink (FHRU-30-1)				79.729	1-30	77.24-136.8-77.24
Heat Source				58.06	1	77.24

3.2: HIGH PRESSURE CONDENSATION CYCLES (HPC-CHEC)

The high pressure condensation cycles are a modified form of the basic CHEC power cycle in which condensation takes place at a higher than atmospheric pressure. The fluid medium is completely condensed at a point on the saturated liquid line or sub-cooled beyond the liquid line within the liquid phase region. Although the HPC-CHEC cycles have low power generation capacity than the complete cycle [3.1], they do have the advantage of less SHTS requirements or even non in some cases and thus not equipment intensive as the basic cycle C-120-30-1. This thesis will examine two high pressure condensation cycles in part 3.2(C-120-34-30@113.3K) and 3.3(C-120-34-30@123.6K) respectively.

3.2.1: Analysis of C-120-34-30@113.1K basecase

Overview of C-120-34-30@113.1K basecase Analysis

This analysis is carried out to check and determine the heat and work characteristics of C-120-34-30@113.1K at its base state. The aim is to determine the cooling gap which remains after the designated SHTS have worked to their maximum. Results are briefly shown below and discussed further in chapter 7.

Table: 3.2.1.a: Summary of C-120-34-30@113.1K base case analysis (high pressure condensation cycle)

SYSTEM C-120-34-30@113.1K	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-14.26	
EX-80-29	-15.355 (Ref:3.2.4.1)	
FHRU-55-15	-12.150 (Ref:3.2.4.2)	
Total Work input		-41.765
C-120-34-30@113.1 Net Work		**+38.535**
Heat Supplied (from ambient) (255.2-185.6)	+69.6	
Heat Rejected (balanced from liquid nitrogen reserve or LNGRS) (-41.47-[-11.342])	+30.128	
Net heat Supplied to cycle		+39.472
COP/Work ratio		**1.923**
SNC		**0.12**
Direct cooling ability ratio(DCAR) ([154.71] / [176.474+38.999])= (154.71/215.473)		0.718

Heat Zones:

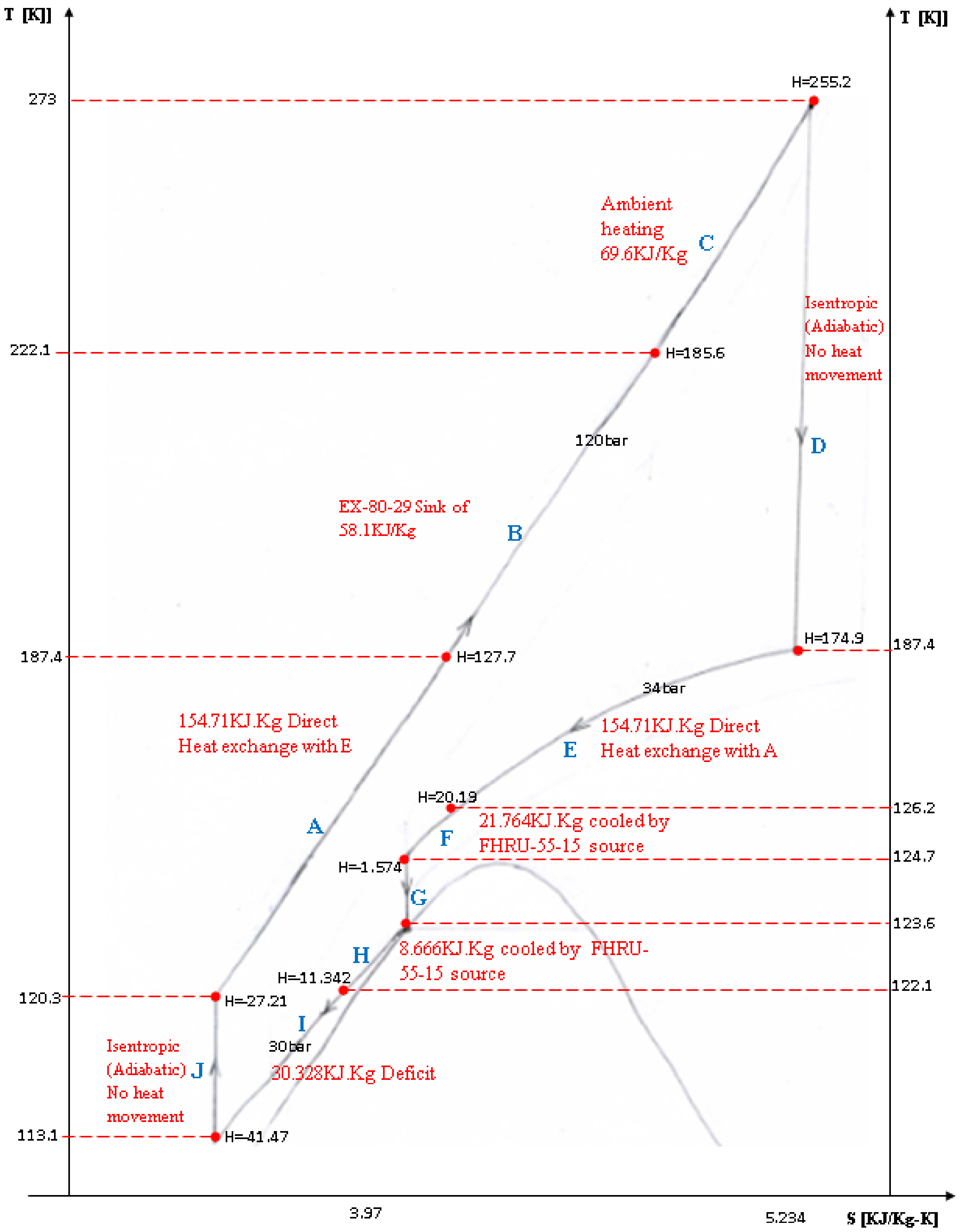


Figure: 3.2.1.a: C-120-34-30@113.1K T-S diagram of profiled heat zones.

Enthalpy flow on SHTS of C-120-34-30@113.1K:

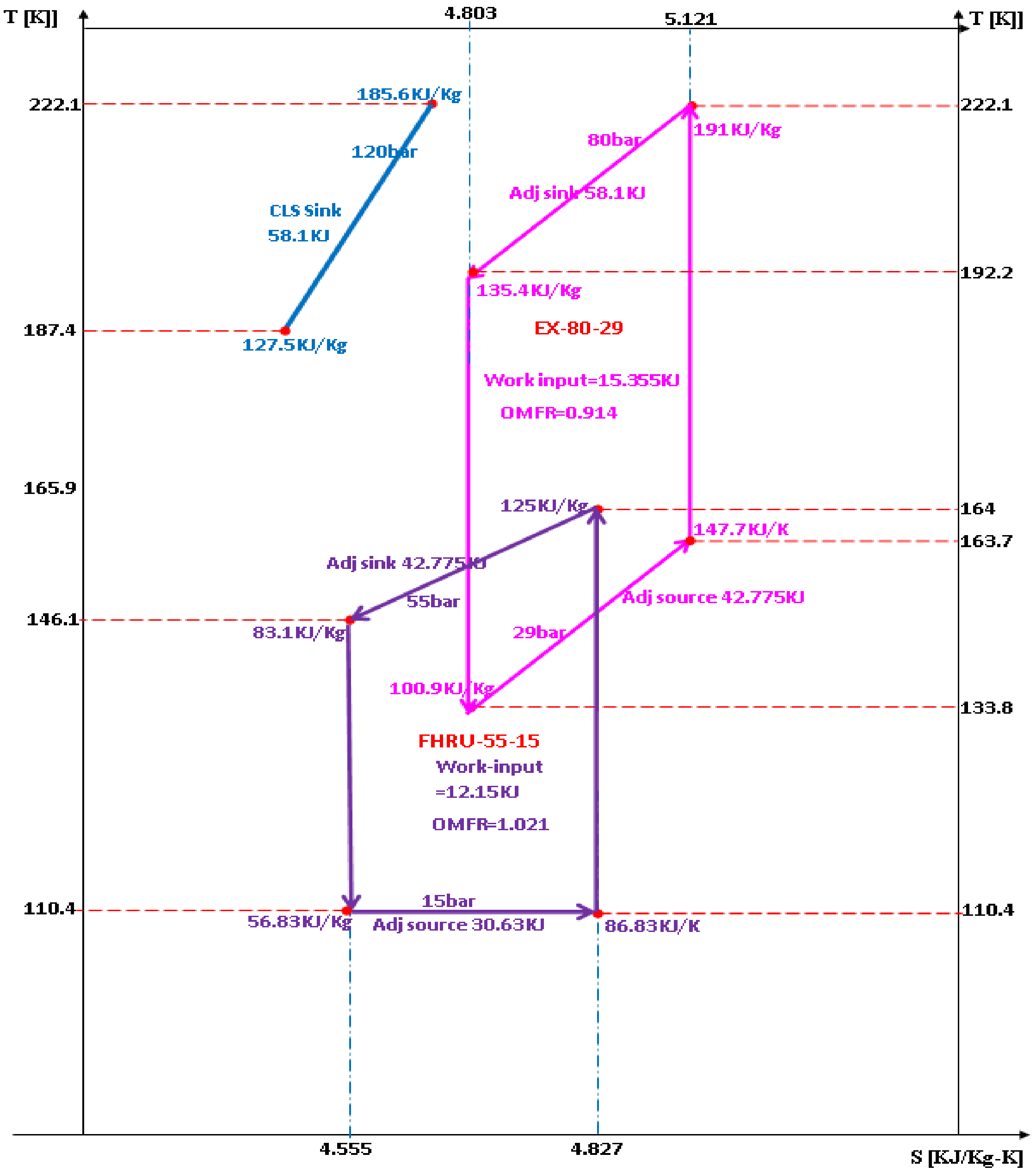


Figure 3.2.1.b: Combined T-S diagram of all SHTS showing the chain of enthalpy pull down

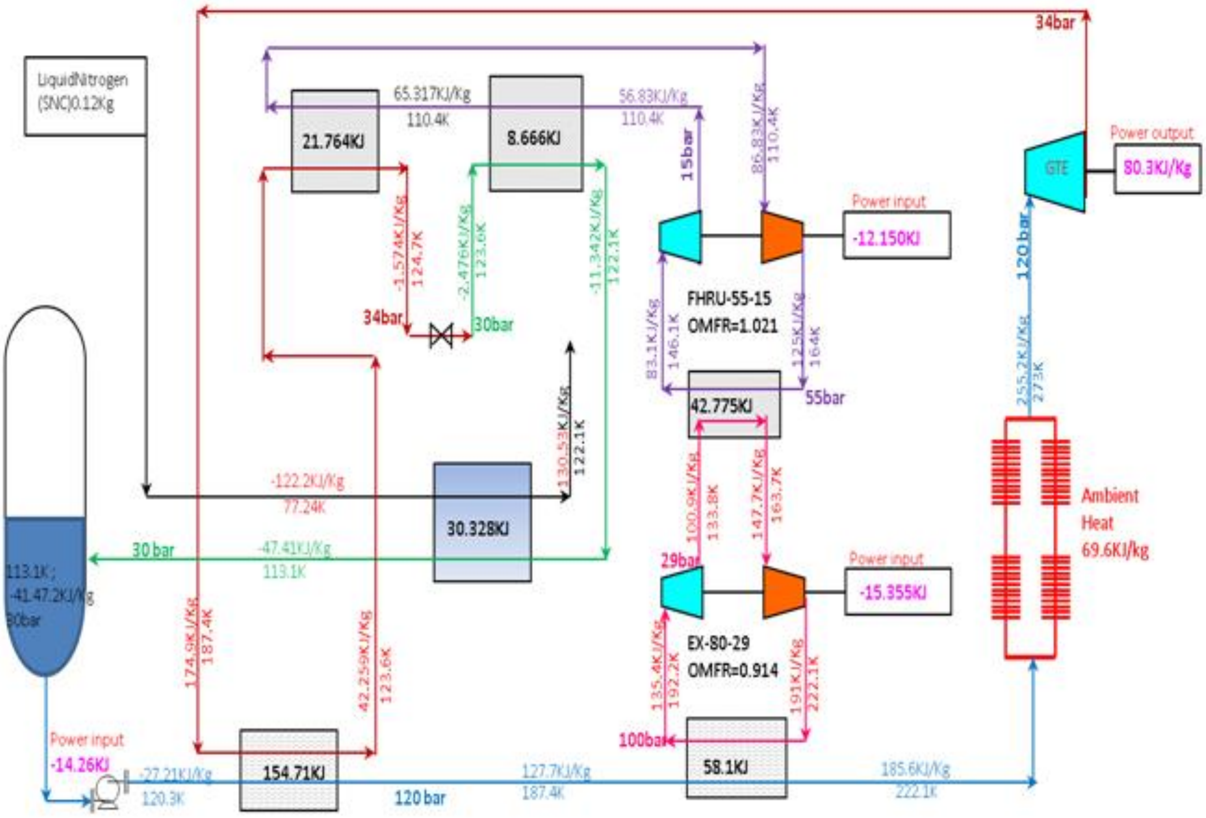


Figure: 3.2.1.c: Flow diagram of a balanced base case C-120-34-30@113.1K cycle.

3.2.2: C-120-34-30@113.1K heat and work balance at steady state

Table: 3.2.2: System heat and work balance for C-120-34-30@113.1K

Location of heat/work demand (node)	Pressure (bar)	Heat demand (KJ/Kg)	Work done (KJ/Kg)	Temperature Conditions (K)	Thermodynamic Process (Ideal)
11>3	30>120	0	-14.26	113.1>120.3	Isentropic(adiabatic) Compression
3>6	120	+282.41	0	120.3>273	Constant Pressure Heating
6>7	120>34	0	+80.3	273>187.4	Isentropic(adiabatic) Expansion
7>8	34	-176.474	0	187.4 to 124.7	Constant Pressure Cooling
8>9	34>30	0	0	124.7>123.6	Isentropic(adiabatic) Expansion
9>11	30	-38.994	0	123.6>113.1	Constant Pressure Sub-cooling

3.2.3: Assessment C-120-34-30@113.1K direct cooling ability

This is done by determining the CLS sink available for direct heat transfer with the GTOS. Table 3.2.3.a gives a breakdown of the CLS sinks available within the requisite temperatures for this cycle.

Table: 3.2.3.a: Sink enthalpy available from CLS at the stated temperatures

CLS temperature (K)	CLS Enthalpy (KJ/Kg)	CLS Sink available (KJ/Kg)	Cooling demand (KJ/Kg)	Cooling deficit (KJ/Kg)
			Ref Table 3.2.2	
120.3	-27.21			
187.4	127.5			
		154.71	-215.468	60.758

The cooling deficit of 60.758KJ/Kg demands the SHTS to utilise the CLS sink from higher temperatures towards meet this cooling requirement. Table 3.2.3.b shows how much CLS sink is available for any possible SHTS.

Table: 3.2.3.b: Sink enthalpy available from CLS beyond direct heat transfer limit but within reach of EX-80-29

CLS temperature (K)	CLS Enthalpy (KJ/Kg)	CLS Sink available (KJ/Kg)
222.1	185.6	
187.4	127.5	
		58.1

Table: 3.2.3.c: Comparing of Enthalpy's of CLS at GTOS at different temperatures

Stream	Pressure	Enthalpy@ 187.5K	Enthalpy@ 124.7K	Enthalpy@ 123.6K	Enthalpy@ 120.3K
CLS	120	127.5	-17.75		-27.21
GTOS	34	174.9	-1.574	-2.476	

The CLS enthalpy of 154.71KJ/Kg is transferred by direct heat exchange to the GTOS in an arrangement shown in Table 3.2.3.d.

Table: 3.2.3.d: Heat exchange between CLS and GTOS.

Stream		Inlet	Outlet	Heat Transfer (KJ/Kg)	Temperature Change
CLS@120bar	Temperature(K)	120.3	187.4		67.1
	Enthalpy(KJ/Kg)	-27.21	127.5	154.71	
GTOS@34bar	Temperature(K)	187.4	126.2		-61.2
	Enthalpy(KJ/Kg)	174.9	20.19	-154.71	

3.2.4: Evaluation of SHTS assisted cooling for C-120-34-30@113.1K

This is done so as to identify the requisite SHTS design which would appropriately help the cycle to match its cooling requirements at the expense of doing some external work. The evaluation is done in the order of temperature profiles of the SHTS from higher to lower levels. This usually starts with the COMPEX unit [3.2.4.1] followed by any respective FHRU.

3.2.4.1: COMPEX unit for the supply of 60.758KJ/Kg for GTOS cooling at lower temperatures.

Table 3.2.4.1 is derived from Table 3.1.1.1.a where the basic COMPEX system design configuration has remained but an adjustment has been done to match the requirements of a high pressure condensation cycle. The same procedure as in [Ref 3.1.1.1] has been followed in calculating the adjusted figures shown in Table 3.2.4.1. The system mass flow rate is adjusted to operate pegged on the available CLS sink of 58.1KJ/Kg [Table 3.2.3.b]

Table: 3.2.4.1: Design configuration of EX-80-29 as adjusted for the high pressure cycle C-120-34-30@113.1K.

	(KJ/Kg)	(KJ/Kg)	CLS Sink available (KJ/Kg)	System adjustment for maximum utilization of available CLS sink(KJ/Kg)
Compression Work (199-147.7)	51.3			
Expansion Work (135.4-100.9)	34.5			
Net Work (51.3-34.5)		16.8		15.355 (Ref 3.2.4.1.1)
Heat Sink (119-135.4)		63.6	58.1 (Table 3.2.3.b)	58.1 (Table 3.2.3.b)
Heat Source (147.7-100.9)		46.8		42.775 (Ref 3.2.4.1.1)

3.2.4.1.1: EX-80-29 calculations in adjusting configuration for high pressure condensation cycle (C-120-34-30@113.1K)

Note: Figures used in these calculations have been extracted from Tables 3.1.1.1.a, b and c. Determination of OMFR therefore based on the heat source demand.

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(EX-80-29)\ design\ sink}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{58.1}{63.6} = 0.914$$

$$Adjusted\ Net\ Work = OMFR \times (EX-80-29)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 0.914 \times 16.8 = 15.355\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (EX-80-29)\ design\ source\ (KJ/Kg)$$

$$Adjusted\ Heat\ Source = 0.914 \times 46.8 = 42.775 KJ/Kg$$

3.2.4.2: Forced Heat Rejection Unit FHRU-55-15

The role of this unit is to transfer the enthalpy from EX-80-29 further down to 110.4K where it can be utilized in cooling the GTOS. The configuration is designed to be able to source heat from the GTOS and sink it to the heat source provided by EX-80-29.

Table: 3.2.4.2: Design configuration of FHRU-55-15

	(KJ/Kg)	(KJ/Kg)	Sink available from EX-110-20 (KJ/Kg)	System adjustment to match sink demand of [54.199] (KJ/Kg)
Compression Work (125-86.83)	38.17			
Expansion Work (83.1-41.9)	26.27			
Net Work		11.9		12.150 (Ref 3.2.4.2.1)
Heat Sink		41.9	42.775 (Table 3.2.4.1)	42.775 (Ref 3.2.4.1.1)
Heat Source		30		30.63 (Ref 3.2.4.2.1)

3.2.4.2.1: FHRU-55-15 calculations

Note: Figures used in these calculations have been extracted from Tables 3.2.4.1 and 3.2.4.2 respectively. Determination of OMFR is therefore based on FHRU-55-15 heat source demand.

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(FHRU-55-15)\ design\ source}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{42.775}{41.9} = 1.021$$

$$Adjusted\ Net\ Work = OMFR \times (FHRU-55-15)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 1.021 \times 11.9 = 12.150\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (FHRU-55-15)\ design\ sink\ (KJ/Kg)$$

$$Adjusted\ Heat\ Sink = 1021 \times 30 = 30.63\ KJ/Kg$$

3.2.4.2.2: Heat exchange between FHRU-55-15 source stream and GTOS

This occurs after the GTOS has been cooled by direct heat exchange with CLS [Table 3.2.3.d]. The GTOS outlet from that heat exchanger becomes the inlet of this one as shown in the Table 3.2.4.2.2.

The balance of cooling enthalpy required to complete the power cycle for C-120-34-30@113.1K is 60.758KJ/Kg [Table 3.2.3.a], out of that requirement, the SHTS brought down 30.63KJ/Kg to 110.4K flat. The heat exchange in 3.2.4.2.2 is limited to this amount only.

Table: 3.2.4.2.2: Heat exchange between FHRU-55-15 source stream and GTOS.

Stream		Inlet	Outlet	Adjusted Heat Transfer (KJ/Kg)	Temperature Change (K)
FHRU-55-15 (source stream)	Temperature(K)	110.4	110.4		
	Enthalpy(KJ/Kg)	56.83	86.83	(30*1.021) =30.63	
GTOS @34bar	Temperature(K)	126.2	124.7		-1.5
	Enthalpy(KJ/Kg)	20.19	-1.574	-21.764	
GTOS@30bar	Temperature(K)	123.6	122.1		-1.5
	Enthalpy(KJ/Kg)	-2.476	-11.342	-8.666	
GTOS@30bar	Temperature(K)	122.1	113.1		9
	Enthalpy(KJ/Kg)	-11.342	-41.47	-36.068	

3.3: Analysis of base case C-120-34-30@123.3.3.1

3.3.1: Overview of C-120-34-30@123.6K Analysis

This analysis is carried out to check and determine the heat and work characteristics of C-120-34-30@123.6K at its base state. The aim is to determine the cooling gap which remains after the designated SHTS have worked to their maximum. Results are briefly shown below and discussed further in chapter 7.

Table: 3.3.1.a: Summary of C-120-34-30@123.6K base case analysis (high pressure condensation

SYSTEM C-120-34-30@123.6	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-17.816	
EX-80-29	-26.023 (Ref 3.3.4.1)	
FHRU-100-30	-15.669 (Ref 3.3.4.1)	
Total Work input		-59.508
C-120-34-30@123.6K Net Work		**+20.792**
Heat Supplied (from ambient) (255.2-185.6)	+69.6	
Heat Rejected (balanced from liquid nitrogen reserve or LNGRS) (-1.574-[-46.276])	-47.85	
Net heat Supplied		+21.75
COP/Work ratio		**1.349**
SNC(Kg)		**0.180**
Direct cooling ability ratio (DCAR) (71.8 / 176.474)		0.407

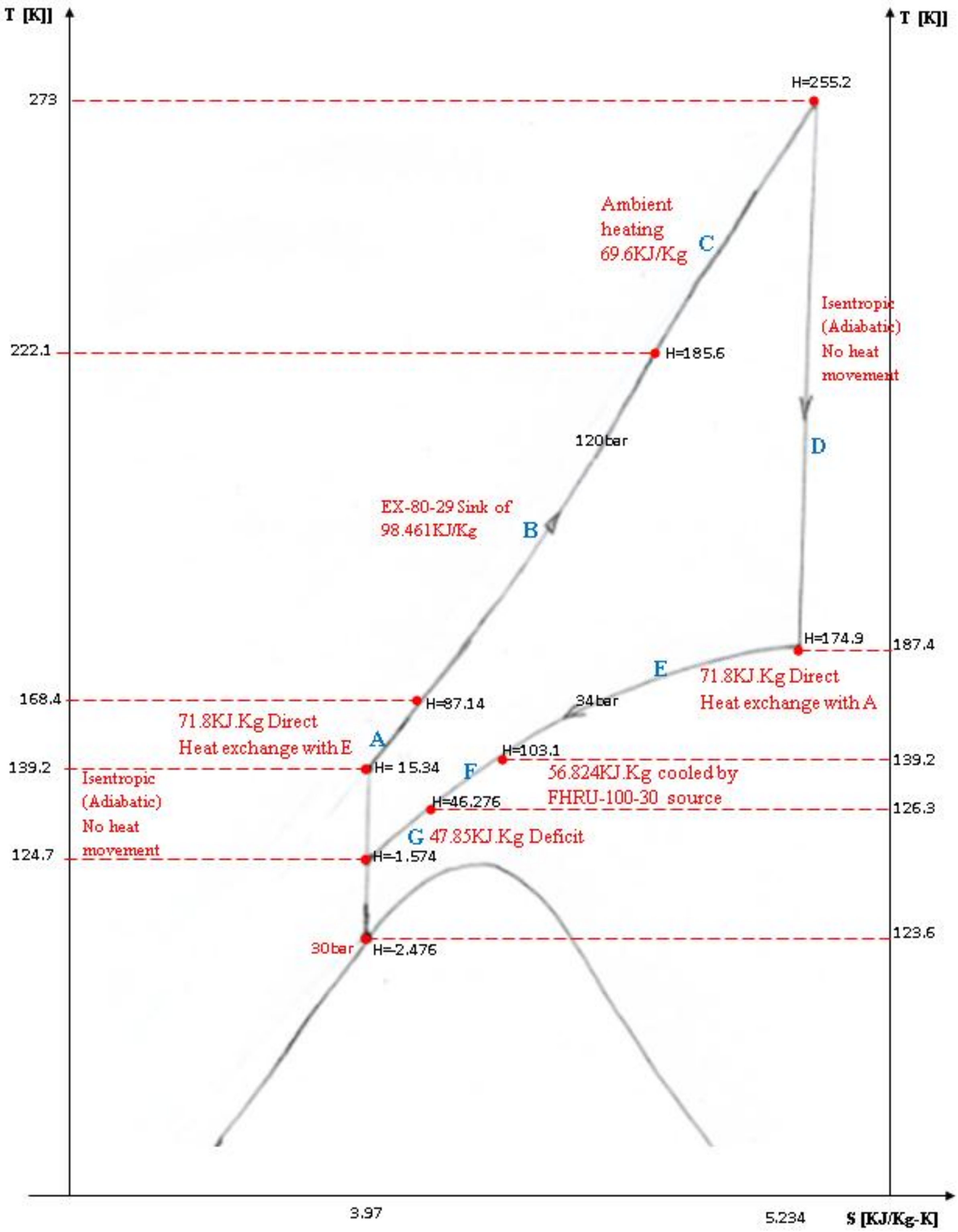


Figure: 3.3.1.a: T-S diagram of C-120-34-30@123.6K profiled heat zones of.

Enthalpy flow on SHTS of C-120-34-30@123.6K:

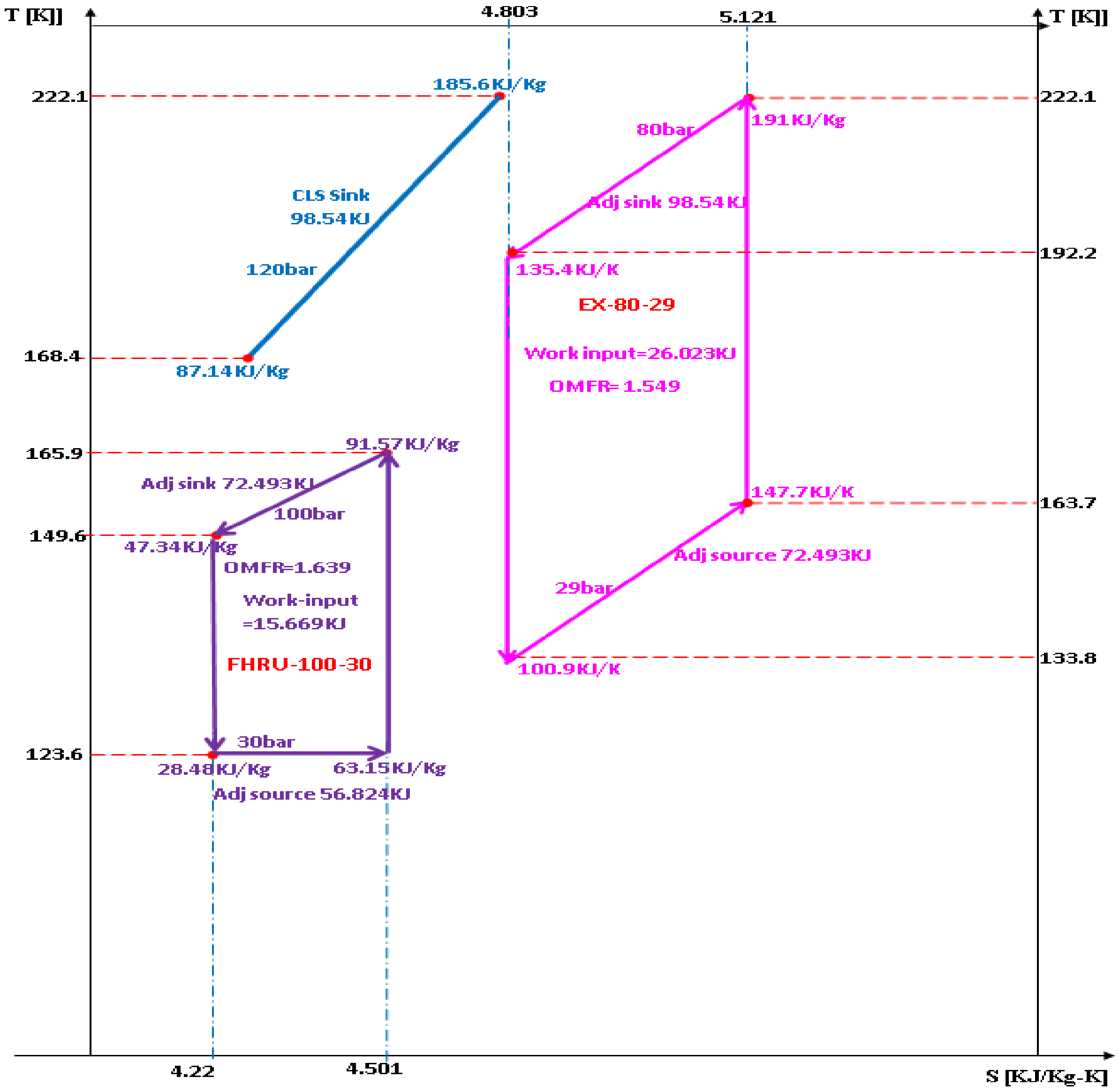


Figure 3.3.1.b: Combined T-S diagram of all SHTS showing the chain of enthalpy pull down

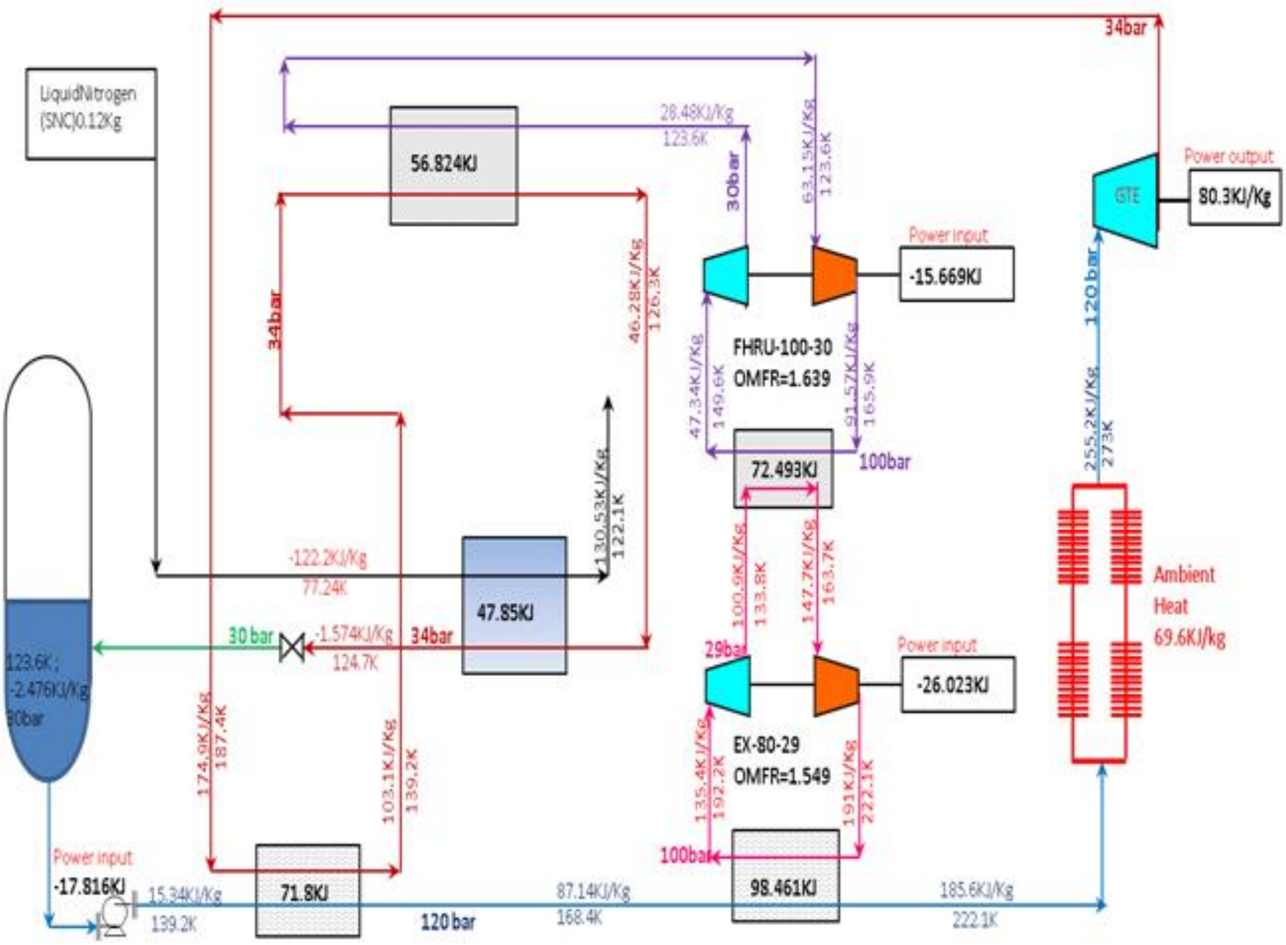


Figure 3.3.1.c: Flow diagram of a balanced base case C-120-34-30@123.6K cycle

3.3.2:C-120-34-30@123.6K heat and work balance

Table: 3.3.2: System heat and work balance for C-120-34-30@123.6K

Location of heat/work demand (node)	Pressure (bar)	Heat demand (KJ/Kg)	Work done (KJ/Kg)	Temperature Conditions (K)	Thermodynamic Process (Ideal)
9>5	30>120	0	-17.816	123.6>139.2	Isentropic(adiabatic) Compression
5>6	120	+239.86	0	123.6>273	Constant Pressure Heating
6>7	120>34	0	+80.3	273>187.4	Isentropic(adiabatic) Expansion
7>8	34	-176.474	0	187.4 to 124.7	Constant Pressure Cooling
8>9	34>30	0	0	124.7>123.6	Isentropic(adiabatic) Expansion

3.3.3: Assessment C-120-34-30@123.6K direct cooling ability

This is done by determining the CLS sink available for direct heat transfer with the GTOS. Table 3.3.3.a gives a breakdown of the CLS sinks available within the requisite temperatures for this cycle.

Table: 3.3.3.a: Sink enthalpy available from CLS at the stated temperatures

CLS temperature (K)	CLS Enthalpy (KJ/Kg)	CLS Sink (KJ/Kg)	CLS sink available for Direct heat transfer (KJ/Kg)	Cooling demand (KJ/Kg)	Cooling deficit (KJ/Kg)
			Ref Table 3.3.3.d	Ref Table 3.3.2	
139.2	15.26				
187.4	127.5				
		112.24	71.8	-176.474	104.674

Since the pump discharge is at 139.2K, the amount of CLS available for direct heat exchange with GTOS is limited to 71.8KJ/Kg [Table 3.3.3.d]. The GTOS available for direct heat transfer is the stream above 139.2K and thus heat transfer will be restricted to be within these temperature limits. The cooling deficit of 104.674KJ/Kg demands need some SHTS in order to utilise the CLS sink from higher temperatures towards meeting this cooling requirement. Table 3.3.3.b shows how much CLS sink is available for any possible SHTS design.

Table: 3.3.3.b: Sink enthalpy available from CLS beyond direct heat transfer limit but within reach of EX-80-29

CLS temperature (K)	CLS Enthalpy (KJ/Kg)	CLS Sink available (KJ/Kg)
222.1	185.6	
167.5	(15.34+71.8)= 87.14	
		98.46

Table: 3.3.3.c: Comparing of Enthalpy's of CLS at GTOS at different temperatures

Stream	Pressure	Enthalpy@ 187.4K	Enthalpy@ 167.5K	Enthalpy@ 139.2K	Enthalpy@ 124.7K	Enthalpy@ 123.6K
CLS	120	127.5	85.06	15.26		
GTOS	34	174.9		103.1	-1.574	-2.476

CLS enthalpy of 71.8KJ/Kg is transferred by direct heat exchange to the GTOS in an arrangement shown in Table 3.3.3.d.

Table: 3.3.3.d: Direct Heat exchange between CLS and GTOS.

Stream		Inlet	Outlet	Heat Transfer (KJ/Kg)	Temperature Change
CLS@120bar	Temperature(K)	139.2	167.5		+28.3
	Enthalpy(KJ/Kg)	15.34	127.5	+71.8	
GTOS@34bar	Temperature(K)	187.4	139.2		-48.2
	Enthalpy(KJ/Kg)	174.9	103.1	-71.8	

3.3.4: Evaluation of SHTS cooling for C-120-34-30@123.6K

This is done so as to identify the requisite SHTS which would appropriately help the system in meeting its cooling requirements at the expense of doing some external work. The evaluation is done in the order of temperature profiles of the SHTS from higher to lower levels. This starts with a COMPEX unit [3.3.4.1] followed by any respective FHRU [Part 3.3.4.2].

3.3.4.1: A COMPEX unit designed for the supply of 104.674KJ/Kg for GTOS cooling at lower temperatures of C-120-34-30@123.6K.

Table 3.3.4.1 is derived from Table 3.1.1.1.a where the basic COMPEX system design configuration has remained but an adjustment has been done to match the requirements of this high pressure condensation cycle with the above configurations. The same procedure as in [Ref 3.1.1.1] has been followed in calculating the adjusted figures shown in Table 3.3.4.1. The system mass flow rate is adjusted to operate pegged on the available CLS sink of 98.46KJ/Kg [Table 3.3.3.b]. The associated FHRU would also share part of the CLS by direct heat transfer.

Table: 3.3.4.1: Design configuration of EX-80-29 as adjusted for high pressure cycle C-120-34-30@123.6K

	(KJ/Kg)	(KJ/Kg)	CLS Sink available (KJ/Kg)	System adjustment for maximum utilization of available CLS sink(KJ/Kg)
Compression Work (199-147.7)	51.3			
Expansion Work (135.4-100.9)	34.5			
Net Work (51.3-34.5)		16.8		26.023 (Ref 3.3.4.1.1)
Heat Sink (119-135.4)		63.6	98.54 (Table 3.3.3.b)	98.54 (Table 3.3.3.b)
Heat Source (147.7-100.9)		46.8		72.493 (Ref 3.3.4.1.1)

3.3.4.1.1: EX-80-29 calculations in adjusting configuration for high pressure condensation cycle (C-120-34-30@123.6K

Note: Figures used in these calculations have been extracted from Tables 3.1.1.1.a, b and c. Determination of OMFR therefore based on the heat source demand.

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(EX-80-29)\ design\ sink}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{98.54}{63.6} = 1.549$$

$$Adjusted\ Net\ Work = OMFR \times (EX-80-29)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 1.549 \times 16.8 = 26.023\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (EX - 80 - 29)\ design\ source\ (KJ/Kg)$$

$$Adjusted\ Heat\ Source = 1.549 \times 46.8 = 72.493KJ/Kg$$

3.2.4.2: Forced Heat Rejection Unit FHRU-100-30 for C-120-34-30@123.6K

The role of this unit is to transfer the enthalpy from EX-80-29 further down to 123.6K where it can be utilized in cooling the GTOS. The configuration is designed to be able to source heat from the GTOS and sink it to the heat source of EX-80-29.

Table: 3.2.4.2: Design configuration of FHRU-100-30 with adjustment for C-120-34-30@123.6K.

	(KJ/Kg)	(KJ/Kg)	EX-80-29 Sink available (KJ/Kg)	System adjustment to match demand (KJ/Kg)
Compression Work	28.42			
Expansion Work	18.86			
Net Work		9.56		15.669 (Ref 3.3.4.2.1)
Heat Sink		44.23	72.493 (Table 3.3.4.1)	72.493 (Ref 3.3.4.2.1)
Heat Source		34.67		56.824 (Ref 3.3.4.2.1)

3.3.4.2.1: FHRU-100-15 calculations for C-120-34-30@123.6K

Note: Figures used in these calculations have been extracted from Tables 3.2.4.1 and 3.2.4.2 respectively. Determination of OMFR is therefore based on FHRU-100-30 heat s sink available.

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Heat\ Sink\ available}{(FHRU - 100 - 30)\ design\ source}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{72.493}{44.23} = 1.639$$

$$Adjusted\ Net\ Work = OMFR \times (FHRU - 100 - 30)\ design\ Net\ Work\ (KJ/Kg)$$

$$Adjusted\ Net\ Work = 1.639 \times 9.56 = 15.669\ KJ/Kg$$

$$Adjusted\ Heat\ Source = OMFR \times (FHRU - 100 - 30)\ design\ source\ (KJ/Kg)$$

$$Adjusted\ Heat\ Source = 1.639 \times 34.67 = 56.824\ KJ/Kg$$

3.3.4.2.2: Heat exchange between FHRU-100-30 source stream and GTOS

This occurs after the GTOS has been cooled by direct heat exchange with CLS [Table 3.3.3.d]. The GTOS outlet from that heat exchanger becomes the inlet of this one as shown in the Table 3.3.4.2.2.

Table: 3.3.4.2.2: Heat exchange between FHRU-100-30 source stream and GTOS.

Stream		Inlet	Outlet	Adjusted Heat Transfer (KJ/Kg)	Temperature Change (K)
FHRU-100-30 (source stream)	Temperature(K)	123.6	123.6		
	Enthalpy(KJ/Kg)	28.48	63.15	(34.67*1.639) = +56.824	
GTOS @34bar	Temperature(K)	139.2	126.3		-12.9
	Enthalpy(KJ/Kg)	103.1	46.276	-56.824	
GTOS @34bar	Temperature(K)	126.3	124.7		-1.6
	Enthalpy(KJ/Kg)	46.276	-1.574	-47.85	

CHAPTER 4

FURTHER ANALYSIS OF A COMPLETE C-120-30-1 CYCLE WITH LIQUID NITROGEN AND NATURAL GAS COOLING

Chapter overview

As observed from the analysis of the overall system [Ref 3.1], The CHEC power cycle cannot generate enough cold to sustain its heat and work balance in a cyclic order without the input of cold from an outside source. This has to be done by using either liquid nitrogen or cold from a natural gas regasification facility (site specific), or any other supply of cold. This thesis will consider the liquid nitrogen [Ref 4.1] and liquefied natural gas regasification [Ref 4.2] options as viable sources of cold for the operation of C-120-30-1.

Chapter contents

4.1: Analysis of C-120-30-1 cycle assisted by liquid nitrogen cooling

Overview on analysis of C-120-30-1 with liquid nitrogen assisted cooling

In this section, C-120-30-1 is analysed again for heat and work balance while it’s base case cooling deficit is provided for by liquid nitrogen. The introduction of liquid nitrogen completed the required cooling and provide a further 7.49KJ/Kg to which was utilised for GTOS condensation at 30bar. This led to a reduction of work input to the SHTS and thus a slightly improved COP. This analysis has established that this configuration has the ability to run with liquid nitrogen supply of 15% mass of main circulating mass flow. At this mass flow rate, it has an ideal COP of 1.6 as compared to 1.4 of base case. This cycle has a lower demand for nitrogen but very poor cooling DCAR. Much of the nitrogen is mainly useful at 1 bar but not beyond. The flat condensation sections on the T-S diagram contribute towards it’s poor DCAR. More results of the analysis are given in Table 4.1.a and further discussed in chapter 7.

Summary of C-120-30-1 after introducing liquid nitrogen cooling

Table 4.1.a summarises the performance of the effect of using nitrogen for supplementing the cooling requirements of C-120-30-1 in comparison with the original base case cycle where cooling was incomplete.

Table: 4.1.a: Summary of C-120-30-1 after introducing liquid nitrogen cooling

SYSTEM C-120-30-1	**KJ/Kg**	**(KJ/Kg)**	**After Nitrogen Cold supply (KJ/Kg)**		
Work output					
GTE	+86.5		+86.5		
2PTE	+31.814		+31.814		
Total Work output		+117.414			+117.414
Work input					
LCP pump	-14.5		-14.5		
EX-80-29	-26.46		-23.020		
FHRU-100-30	-15.936		-13.864		
FHRU-30-1	-21.669		-21.669		
Total Work input		-78.565			-73.053
C-120-30-1 Net Work		**+38.849**			**+44.361**
Heat Supplied (from ambient) (255.2-185.6)	+69.6		(255.2-172.575) Figure: 4.1.1.a	82.625	
Heat Rejected (balanced from liquid nitrogen reserve) ((-58.06-[-122.2])-34.29}	-29.85		(29.891+7.49) Table 4.1.1	-37.381	
Net heat Supplied to cycle		+39.75			+53.736
COP/Work ratio[Electrical]		**1.494**			**1.607**
SNC(Kg)					1.5

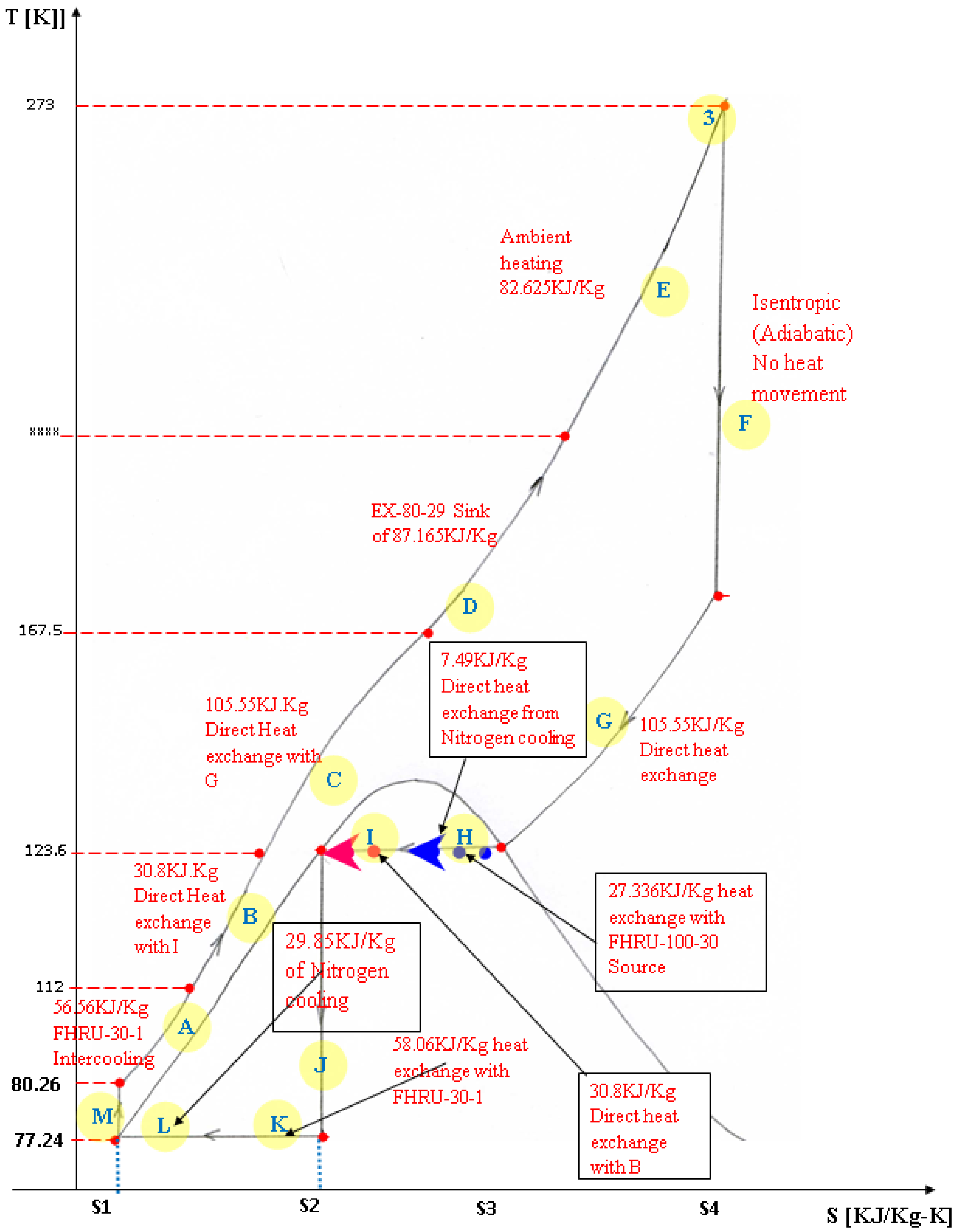


Figure 4.1.a: T-S diagram of C-120-30-1 profiled heat zones after nitrogen cooling

Enthalpy flow on SHTS of C-120-30-1:

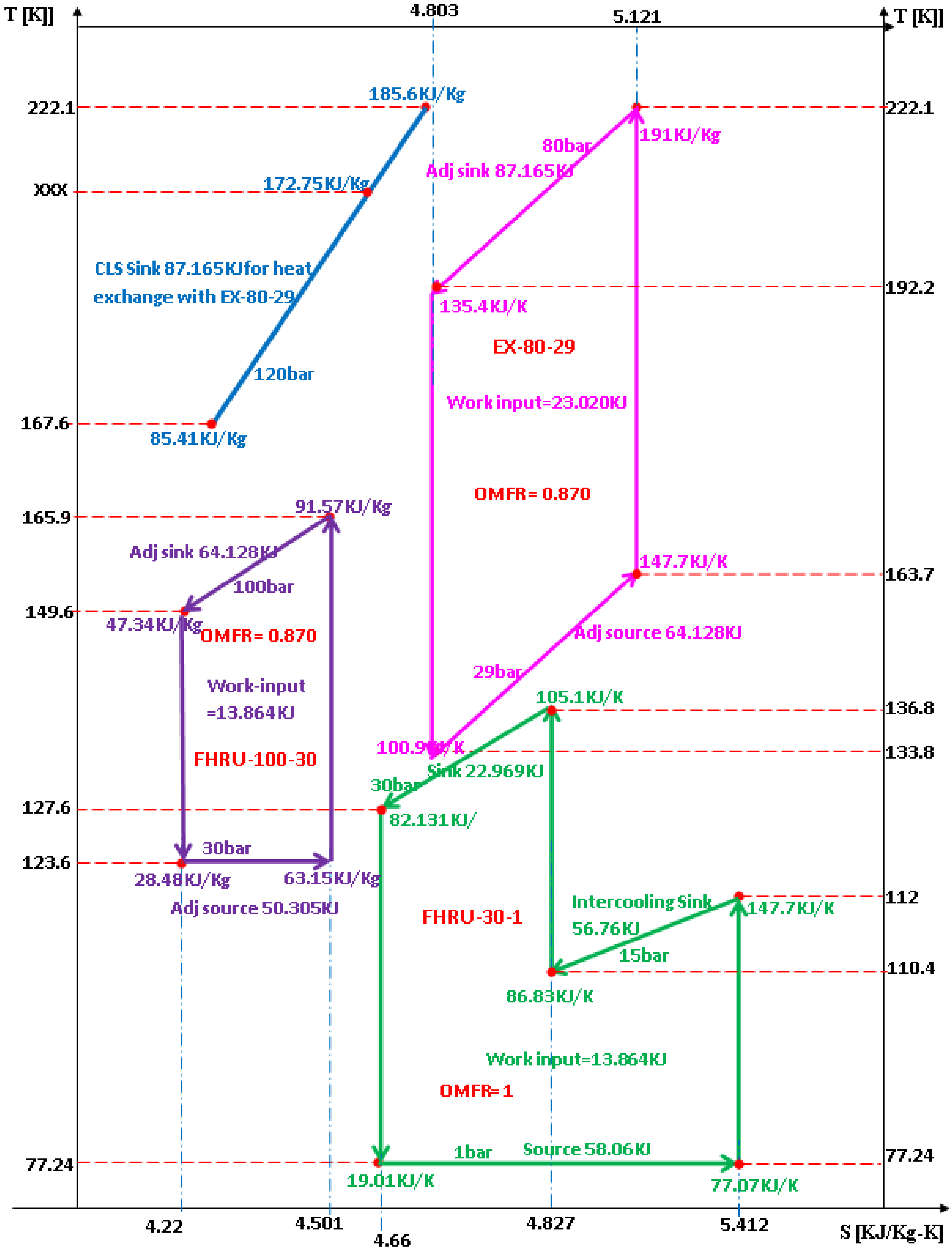


Figure 4.1.b: Combined T-S diagram of all SHTS showing the chain of enthalpy pull down after nitrogen cooling

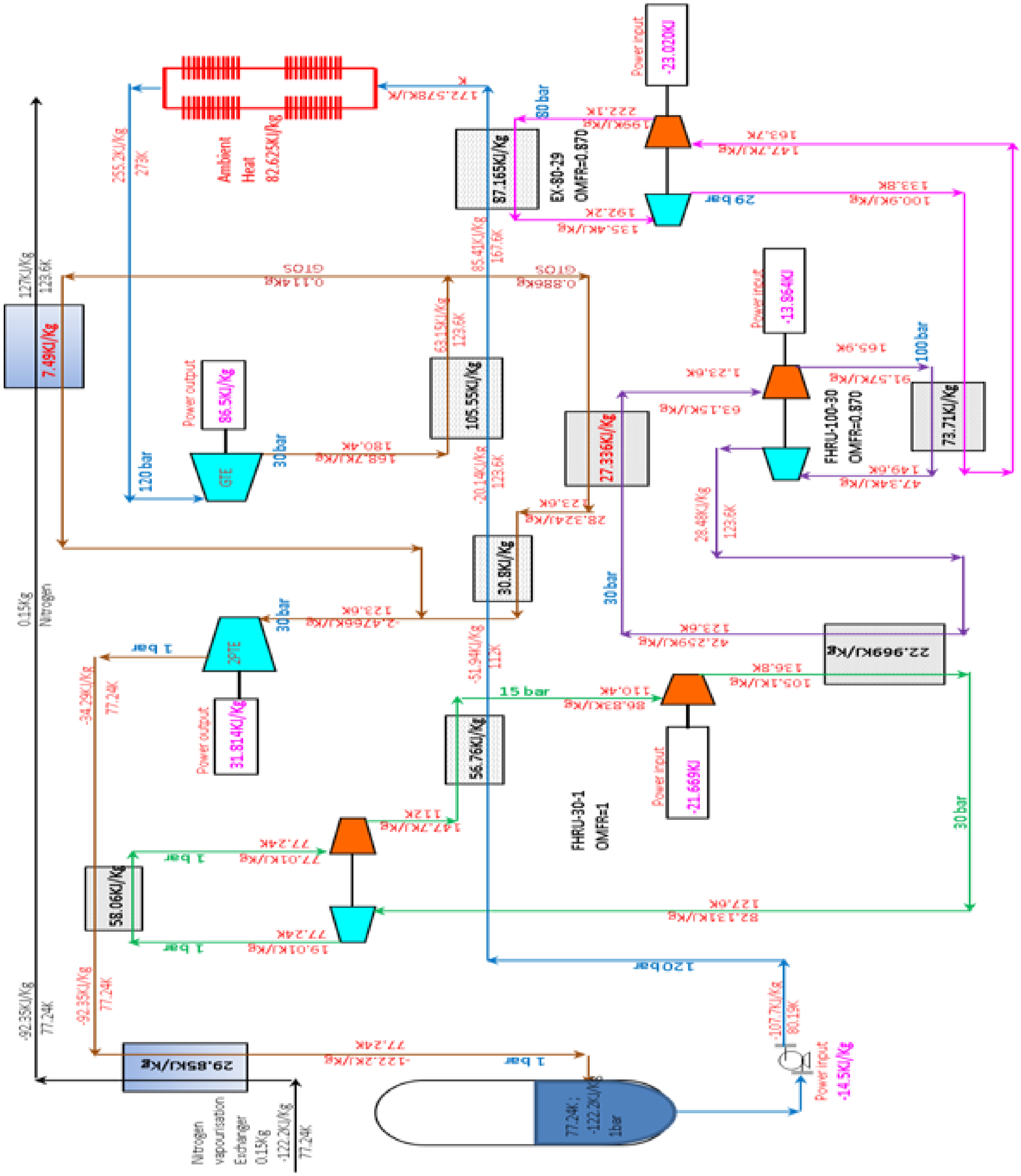


Figure 4.1.c: Flow diagram of a balanced base case C-120-30-1 cycle after nitrogen cooling.

4.1.1: Assumptions

The liquid nitrogen for this purpose shall be sourced from excess electricity or intermittent renewables.

4.1.1.1: Establishing the net amount of cooling from liquid nitrogen for C-120-30-1

Note: This analysis is done with reference to figures 2.4.1, 3.1.1.a, Table 2.4.3.3.b and the exercise carried out in part 3.1.

In part 3.1.2, C-120-30-1 required some cooling of 29.85KJ/Kg at 1 bar/77.24K [Table 3.1.2]. This is to be supplied by vaporization of liquid nitrogen. Table 4.1.1 gives details of the enthalpy available for the cycle when the required amount of liquid nitrogen is vaporised.

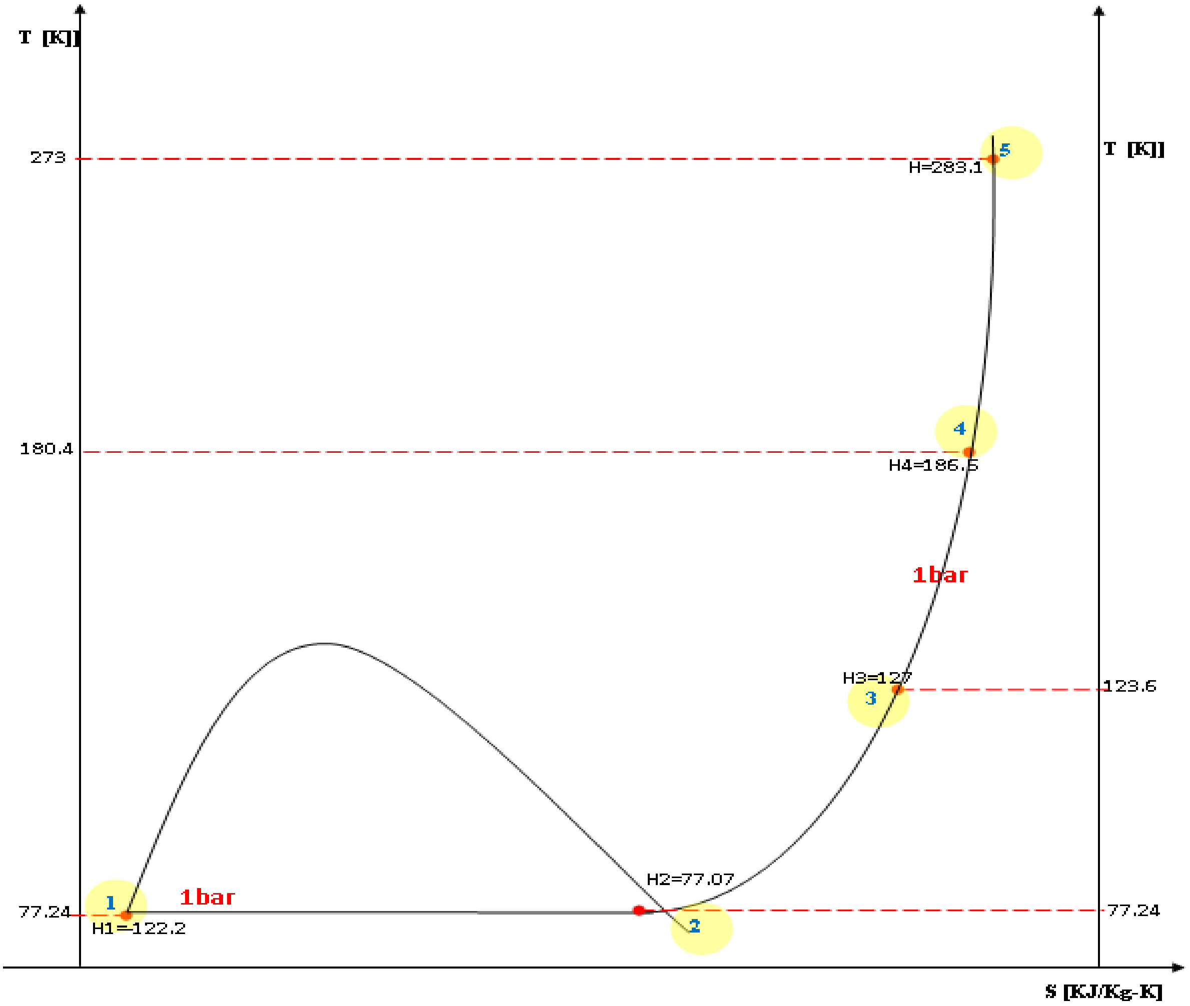


Figure 4.1.1.1: T-S diagram of nitrogen vaporisation at 1bar for C-120-30-1 cooling

Table: 4.1.1: Values of Enthalpy of Liquid nitrogen vaporisation at 1 bar for C-120-30-1

Vaporisation of liquid nitrogen(Node)		Pressure/ Temperature (Bar/K)	Enthalpy (KJ/Kg)	Cooling availed by 0.15Kg/s [Ref 4.2.1.1] of liquid nitrogen to C-120-30-1
1	-122.2	1 / 77.24		
2	77.07	1 / 77.24		
1-2			199.27	29.891
2	77.07	1 / 77.24		
3	127	1 / 123.6		
2-3			49.93	7.49
3	127	1 / 123.6		
4	186.5	1 / 180.4		
3-4			59.5	8.925
4	186.5	1 / 180.4		
5	283.1	1 / 273		
4-5			96.6	14.49
Net cold available from liquid nitrogen vaporisation (1-5)			405.3	60.796

Nitrogen cooling calculations

$$Operating\ Mass\ flow\ = \frac{Cooling\ required\ by\ [C-120-30-1]}{Available\ cold\ from\ Nitrogen\ vaporisation}$$

$$Operating\ Mass\ flow\ = \frac{29.85}{199.27} = 0.14979 \cong 0.15Kg/s$$

This means for every 1Kg flow on the CHEC cycle, 0.15Kg of liquid nitrogen have to vaporise. This establishes some specific nitrogen consumption (SNC) requirement of 15% of the total fluid medium flow.

4.1.1.2: The complete C-120-30-1 power cycle (with nitrogen assisted cooling)

The cycle will have to be reconfigured in order to take advantage of the extra cooling provided by the vaporising stream of nitrogen at different temperature stages as it loses enthalpy. The nitrogen cooling provides an opportunity for reducing the work input from some or all of the SHTS. This can be done by providing some side streams of GTOS to undergo direct heat transfer in a PFHE as the nitrogen gains enthalpy towards ambient. There are three options of how the nitrogen extra cooling can be utilized, these are

- FHRU intercooling so as to increase the circulating mass flow for more condensation at FSCP.
- GTOS condensation at 30bars / zone I
- GTOS gas cooling at 30bars / zone G
- ASU intake for economical nitrogen recovery

This study will consider the second option as it is the most viable out of the above three, the other three option will be a subject for further work.

4.1.1.2.1: Determining the mass flow of GTOS Condensation side stream (@ 30bar / 123.6K) for direct heat exchange with nitrogen stream.

From table 4.1.1, 7.49KJ/Kg can be used by direct heat transfer because the temperature difference allows the flow of enthalpy between the two streams. A calculation is performed to determine the mass flow of GTOS which is to be channelled to exchange heat with the 0.15Kg/s nitrogen flow. The enthalpy required for complete condensation of nitrogen vapour at 30bar/123.6K is 65.626KJ/Kg [Ref Table 2.4.3.3.b]

$$Operating\ Mass\ flow\ = \frac{Available\ cold\ from\ Nitrogen\ vaporisation(nodes\ 2-3)}{Condensation\ Enthalpy\ @30bar/123.6K}$$

$$Operating\ Mass\ flow\ = \frac{7.49}{65.626} = 0.114Kg/s$$

A GTOS side-stream of 0.114Kg/s at node 5 of C-120-30-1 will be channelled to fully utilize enthalpy of nitrogen before re-joining the mainstream at node 6 [Ref Figure 3.1.1.a]. This will have the effect of reducing the work and heat loads of FHRU-100-30 and EX-80-29 respectively, and that will require some adjustments to correct the heat and work balance of the system. This will lead to a reduction in work input to the two SHTS and an increase in the work output from the system. The cycle COP will also be higher.

4.1.1.2.1.1: Adjustment of FHRU-100-30 to account for the impact of nitrogen assisted cooling

The new heat source for FHRU-100-30 will therefore reduce by 7.49KJ/Kg. Table 3.1.1.2 shows the initial FHRU-100-300 source demand to be 34.826KJ/Kg.

$$\therefore New\ heat\ source\ demand(FHRU-100-30) = 57.795 - 7.49 = 50.305KJ/Kg$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{New\ Heat\ Source\ Demanded}{(FHRU-100-30)Basecase\ \ source}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{50.305}{57.795} = 0.870$$

$$Nitrogen\ Adjusted\ Net\ Work = OMFR \times (FHRU - 100 - 30)\ Basecase\ Net\ Work\ (KJ/Kg)$$

$$Nitrogen\ Adjusted\ Net\ Work = 0.870 \times 15.936 = 13.864\ KJ/Kg$$

$$Nitrogen\ Adjusted\ Sink = OMFR \times (FHRU - 100 - 30)\ Basecase\ sink\ (KJ/Kg)$$

$$Nitrgen\ Adjusted\ Sink = 0.870 \times 73.71 = 64.128\ KJ/Kg$$

All results on adjustments made on FHRU-100-30 are summarised in table 4.2.1.2.1.1

Table: 4.1.1.2.1.1; Adjustments on FHRU-100-30 after utilizing 7.49KJ/Kg nitrogen cooling for 30 bar condensation.

	(KJ/Kg)	(KJ/Kg)	Base case adjusted configuration (KJ/Kg)	Nitrogen Adjusted configuration (KJ/Kg)
Compression Work	28.42			
Expansion Work	18.86			
Net Work		9.56	15.936 (Ref 3.1.1.2.2)	13.864 (Ref 4.1.12.2.1.1)
Heat Sink		44.23	73.71	64.128 (Ref 4.1.12.2.1.1)
Heat Source		34.67	57.795 (Ref 3.1.1.2.2)	50.305 (Ref 4.1.12.2.1.1)

The above changes have the effect of triggering further changes on the EX-80-29 due to the change in its source demand as result of the changes in FHRU-100-30 sink. From Table 4.2.1.2.1.1 above, the new overall adjusted sink on FHRU-100-30 is 64.128KJ/Kg, which is less than the original 73.71KJ/Kg.

$$\therefore Heat\ source(EX - 80 - 29) = Heat\ sink(FHRU - 100 - 30)$$

$$New\ Heat\ source\ (EX - 80 - 29) = 64.128 KJ/Kg$$

This figure is then used in the calculations to adjust EX-80-29 shown below.

4.1.1.2.1.2: EX-80-29 calculations after nitrogen cooling as affected by reduced FHRU-100-30 sink demand

$$New\ Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{New\ Heat\ Source\ demand}{(EX-80-29)\ Basecase\ source}$$

$$Operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{64.128}{73.71} = 0.870$$

$$Nitrogen\ Adjusted\ Net\ Work = OMFR \times (EX-80-29)\ Basecase\ Net\ Work\ (KJ/Kg)$$

$$Nitrogen\ Adjusted\ Net\ Work = 0.870 \times 26.46 = 23.020 KJ/Kg$$

$$Nitrogen\ Adjusted\ Sink = OMFR \times (EX-80-29)\ Basecase\ sink\ (KJ/Kg)$$

$$Nitrogen\ Adjusted\ Sink = 0.870 \times 100.19 = 87.165 KJ/Kg$$

4.1.1.2.1.3: Adjustment of EX-80-29 heat and work balance after nitrogen cooling.

Table: 4.1.1.2.1.3; EX-80-29 heat and work balance after nitrogen cooling.

System Design configuration	KJ/Kg	KJ/Kg	Basecase adjusted configuration (KJ/Kg)	Nitrogen Adjusted configuration (KJ/Kg)
Compression Work (199-147.7)	51.3			
Expansion Work (135.4-100.9)	34.5			
Net Work (51.3-34.5)		16.8	26.46 (Ref 3.1.1.1.3)	23.020 (Ref 4.1.1.2.1.2)
Heat Sink (119-135.4)		63.6	100.19 (Ref 3.1.1.1.3)	87.165 (Ref 4.1.1.2.1.2)
Heat Source (147.7-100.9)		46.8	73.71 (Table 3.1.1.1.1)	64.128 (Ref 4.1.1.2.1.1)

4.2: Analysis of C-120-30-1 cycle assisted by LNGR cooling

Overview on analysis of C-120-30-1 with LNGR assisted cooling

This analysis is carried out to check and determine the heat and work characteristics of C-120-30-1 while its basecase cooling deficit is provided by LNGR. Results are briefly shown below and discussed further in chapter 7.

Table: 4.2.a: Summary of C-120-30-1 after using LNGRS cooling

SYSTEM C-120-30-1	**(KJ/Kg)**	**(KJ/Kg)**	**After Cooling with LNGRS (KJ/Kg)**		
Work output					
GTE	+86.5		+86.5		
2PTE	+31.814		+31.814		
Total Work output		+117.414			+117.414
Work input					
LCP pump	-14.5		-14.5		
EX-110-20	-26.46		0		
FHRU-60-30	-15.936		0		
FHRU-40-1	-21.669		-49.36		
Total Work input		-78.672			-63.86
C-120-30-1 Net Work		**38.742**			**+53.554**
Heat Supplied (from ambient)	+69.6		(255.2-28.57)	+226.63	
Heat Rejected (balanced from liquid nitrogen reserve)	-29.58				
		39.75			
COP/Work ratio[Electrical]		**1.494**			**1.836**

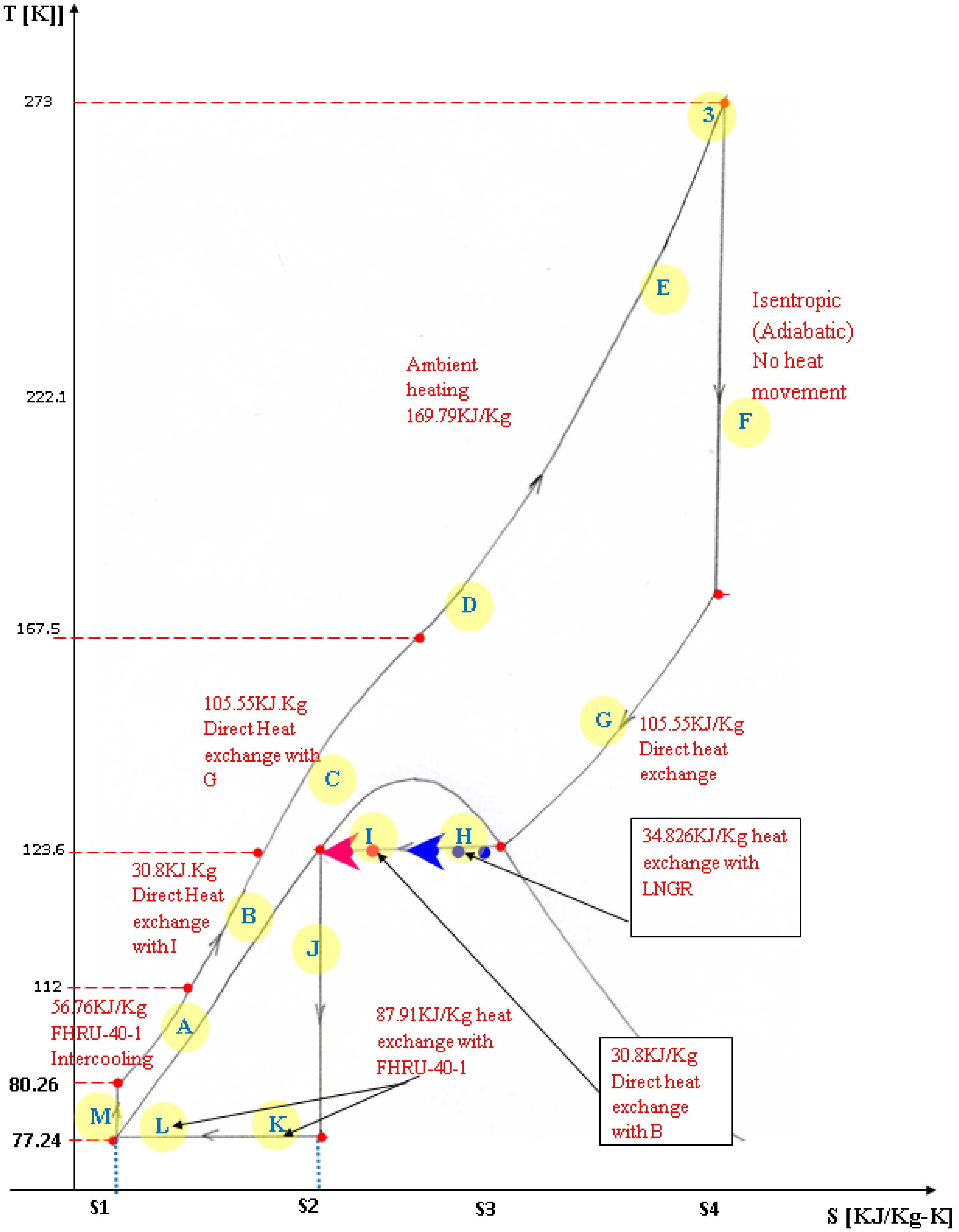


Figure 4.2.a: T-S diagram of C-120-30-1 profiled heat zones after LNGR cooling

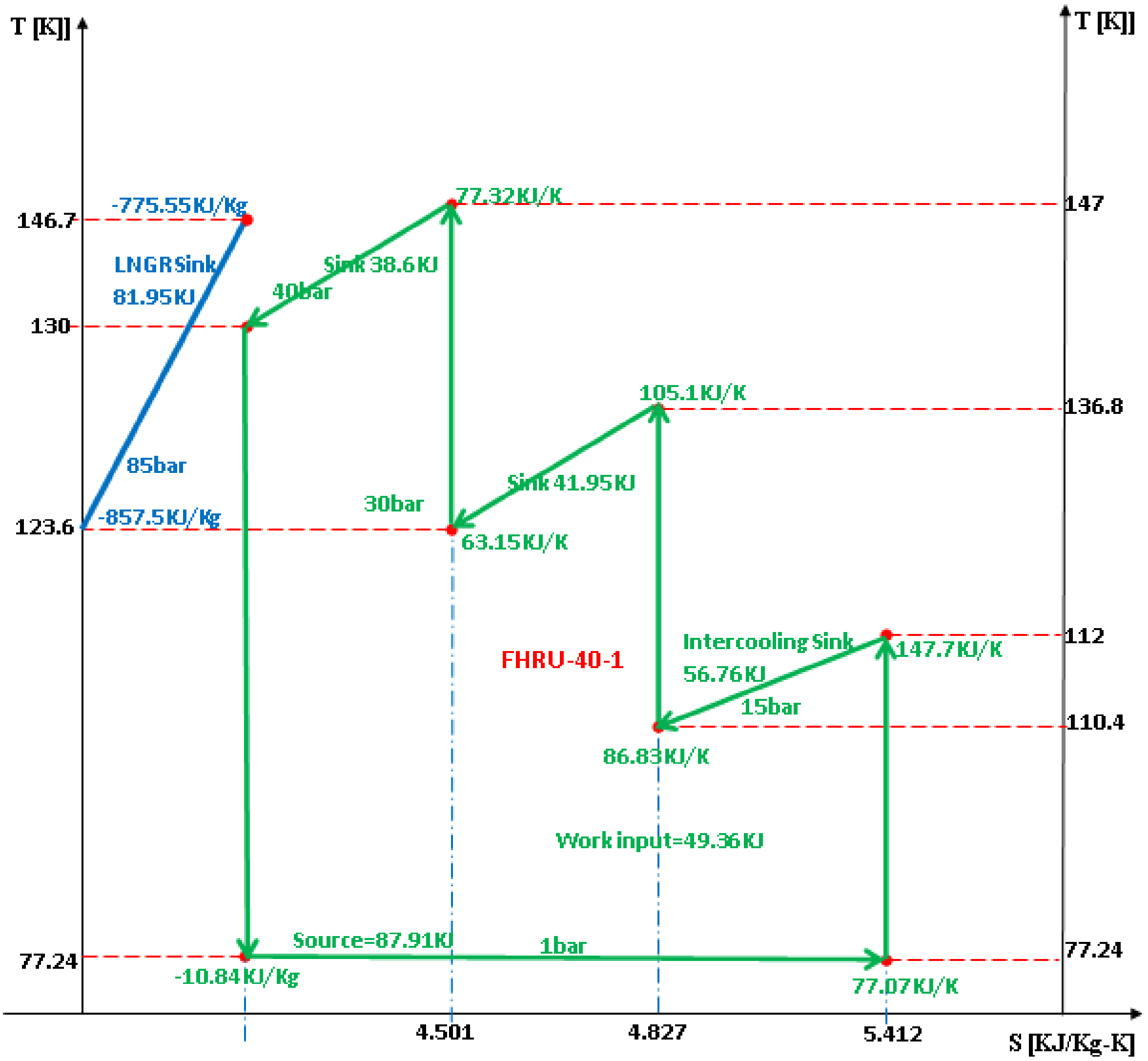


Figure 4.2.b: Combined T-S diagram of all SHTS showing the chain of enthalpy pull down after LNGR cooling

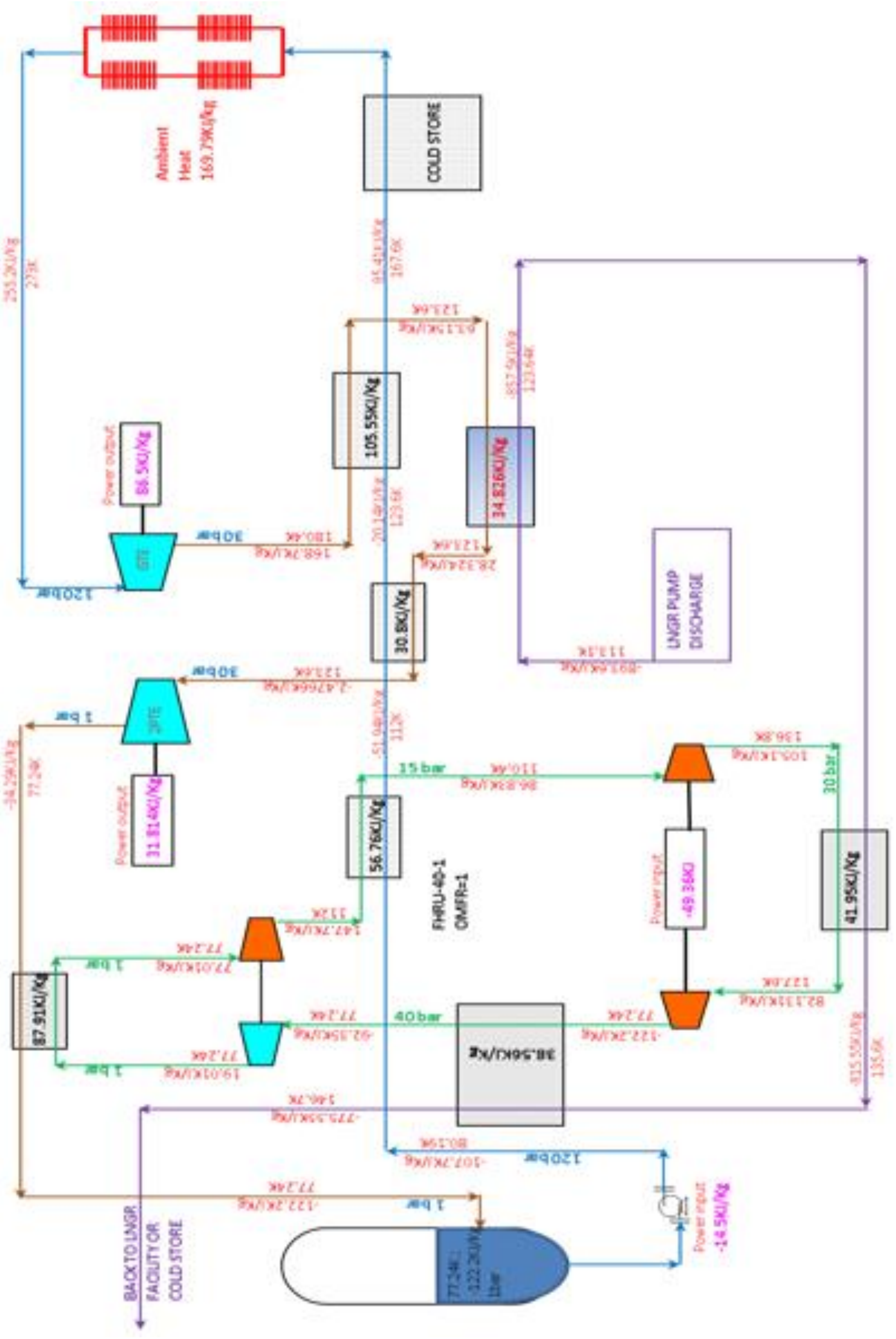


Figure 4.2.c: Flow diagram of a balanced LNGR cooled C-120-30-1 cycle

4.2.1: Assumptions

Note: The EES platform does not have LNG properties but since LNG is mostly over 90% methane, this study will use the properties of methane to simulate those of LNG. Although there will be some slight differences on the properties of LNG and methane, the limit of their differences shall be acceptable for the purposes of this study and all properties will be assumed to be the same. The cold associated with any LNG regasification process is described thermodynamically in part 4.2.1.1. .

4.2.1.1: LNG heat balance calculations in the regasification process

LNG is natural gas which has been cooled to its liquefaction temperature of 111K at 1 bar. The process of regasification involves heating the gas with ambient heat sourced from air, water or waste heat. In order to utilize the cold generated by the LNG regasification process to address the cooling needs of the CHEC power cycle, the GTOS acts to provide some of the heat required for LNG gasification.

LNG is pumped from storage at 1bar / 111K to UK's mains gas supply pressure of 85bar before undergoing a constant pressure heating process until it attains a temperature of 278K. The heat balance is summarized with respect to C-120-30-1 temperature configuration as shown in table 4.2.1.

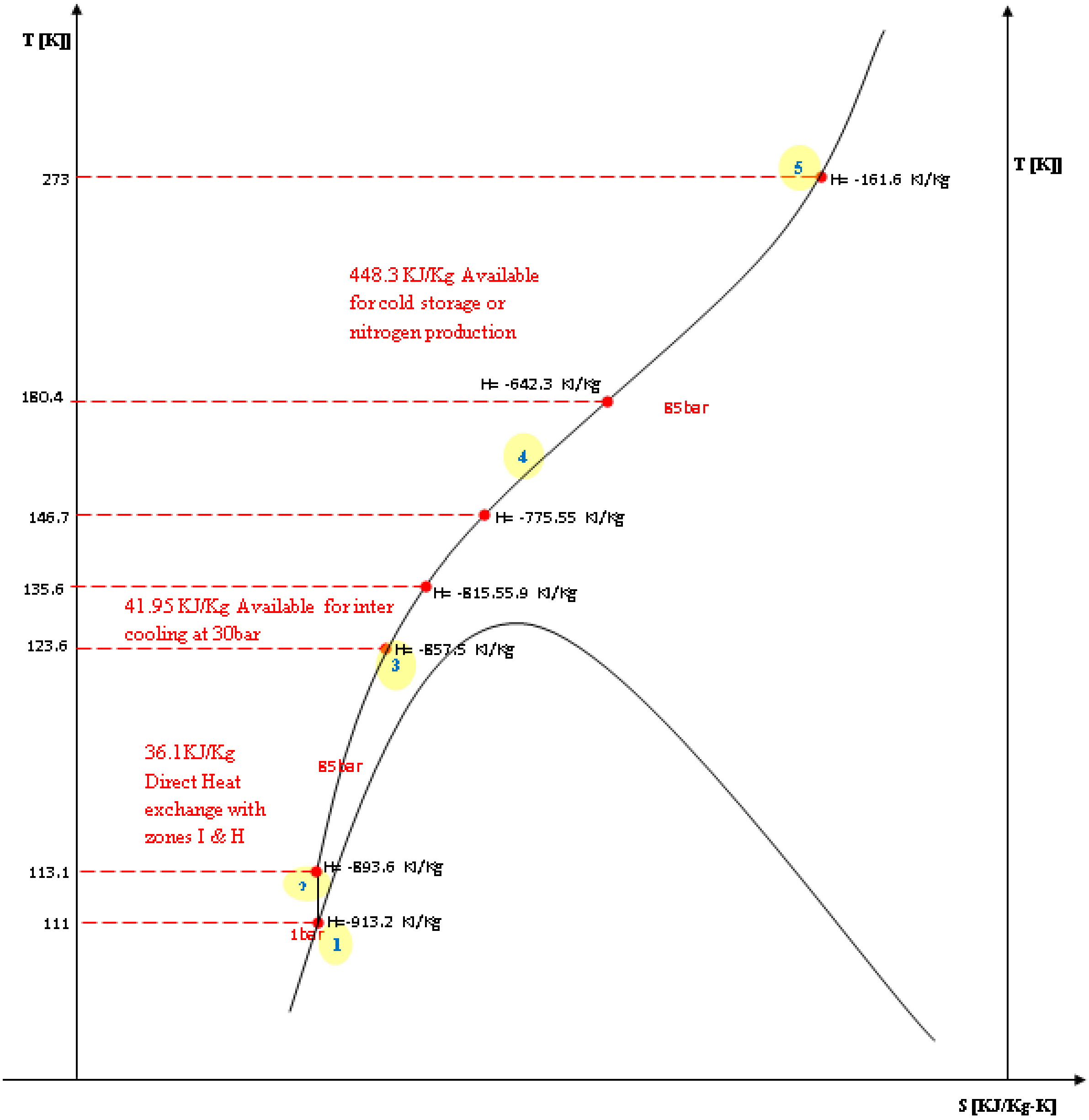


Figure 4.2.1.1: T-S diagram of LNGR Stream for C-120-30-1

Table: 4.2.1.1; LNG heat balance analysis in the regasification process

LNG Temperature (K)	Enthalpy (KJ/Kg)	Enthalpy available for C-120-30-1 (KJ/Kg)	Description
111	-913.2	0	Initial state
113.1	-893.6	0	Isentropic compression (Regasification pump discharge)
123.6	-857.5	(-857.5-[-893.6])= 36.1	Available for GTOS condensation @30bar
130	-835.3		
135.6	-815.55	(-815.55-[-857.5])= 41.95	Available for intercooling @30bar
140	-799.9		
146.7	-775.55	(-775.55-[-815.55])= 40	Available for intercooling @40bar before expansion to 1bar.
150	-763.5		
160	-725.9		
170	-686.4		
180.4	-642.3	(-642-[-857.5])= 215.2	Available for GTOS cooling between nodes 4 and 5 of figure 3.1.1.a
190	-597.1		
200	-541.2		

4.2.1.1.1: Establishing the amount of cooling from liquid LNGR for C-120-30-1

Note: This analysis is done with reference to figures 2.4.1, 3.1.1.a and the exercise carried out in part 3.1

In part 3.1.2, C-120-30-1 required some cooling of 29.85KJ/Kg at 1 bar/77.24K. This is to be supplied from the LNG regasification process when liquefied natural gas is vaporised back into a gaseous state before being supplied to customers. Table 4.2.1.1 gives details the enthalpy available for the cycle when LNG is being vaporised

4.2.1.2: Adjustments on C-120-30-1 to accommodate the LNGR cooling

In this section, C-120-30-1 is considered afresh with much of its cooling sink being availed from the LNGR and partly by CLS. The FHRU will be redesigned to match the cooling requirements of 87.91KJ/Kg at FCSP. The sections that follow will demonstrate the following;

- Incorporation of the LNGR sink of 36.1KJ/Kg into C-120-30-1
- Adjusting the operation of the FHRU in providing the required cooling.

4.2.1.2.1: Incorporation of 36.1KJ/Kg cooling from LNGR for condensation (30 bars / 123.6K) by direct heat exchange

The GTOS enthalpy requirement for complete condensation between nodes 5 and 6 (Figure 3.1.1.a) is 65.626KJ/Kg [Ref Table 2.4.3.3.b]. This heat requirement is fully matched by the combined heat sink of 66.9KJ/Kg from CLS and LNGR as shown in Table 4.2.1.2.1. This eliminates the need for FHRU-60-30 whose main purpose was to ensure complete condensation at 30bar/ 123.6K. The GTOS will be divided in to two streams at node 5 for direct heat transfer with the LNGR stream and the CLS before converging together at node 6.

Table: 4.2.1.2.1; Available cooling enthalpy’s from CLS and LNGR.

Source of cold for condensation at 30bar	Enthalpy available for GTOS condensation @30bar (KJ/Kg)	Total heat sink for condensation @30bar (KJ/Kg)
CLS	30.8 (Ref 3.1.1.2)	
LNG (Ref 4.2.1.1)	36.1 (Table 4.2.1.1)	
		66.9

4.2.1.2.2: A redesign of FHRU to provide the required cold of -87.91KJ/Kg at 1bar [Ref: Table 2.8.1]

This is necessary because the LNG outlet where heat exchange with GTOS is set to begin is at 113.1K [4.2.1.1] while GTOS final condensation requires cooling at 77.24K. A FHRU will have to be improvised to transfer the LNGR sink from the temperatures above down to 77.24K. A new FHRU with the following compression stages is considered;

- 1-15 bar
- 15-30bar
- 30-40bar

From part 2.5.2.4, the new FHRU can be assigned the acronym FHRU-40-1 whose thermodynamic properties are analysed in table 4.2.1.2.2. Figures 4.2.1.2.2.a and 4.2.1.2.2b give a further graphic description of the FHRU-40-1. The design features higher pressure compression in order to be able to access much of the LNGR sink which is mainly concentrated on higher temperatures. Thus FHRU-40-1 is specifically designed to provide enough cooling (87.91KJ/Kg) for complete condensation at 1bar.

Table: 4.2.1.2.2: FHRU-40-1 system configuration

		KJ/Kg	KJ/Kg	KJ/Kg	Pressure range (Bar)	Temperature range (K)
Compression Work	Multistage / intercooled	66.52			1-15	77.24-112
	Single stage	18.27			15-30	110.4-136.8
	Single stage	14.17			30-40	123.6-147
Net Compression Work			98.96		1-40	77.24-130
Expansion Work(38.76-[-10.84])=			49.6		40-1	130-77.24
Net Work (FHRU-40-1)				49.36	1-40-1	77.24-147-77.24
Heat Sink	Multistage		56.76		1-15	77.24-112
	Single stage		41.95		15-30	110.4-136.8
	Single stage		38.56		30-40	123.6-147
Net Heat Sink (FHRU-40-1)				137.27	1-30	77.24-136.9-77.24
Heat Source				87.91	1	77.24

CHAPTER 5

FURTHER ANALYSIS OF C-120-34-30@113.1K CYCLE WITH NITROGEN AND NATURAL GAS COOLING

Chapter contents

Chapter overview

As observed from the analysis of the basecase system [Ref 3.2], C-120-34-30@113.1K power cycle cannot generate enough cold to sustain its heat and work balance in a cyclic order without the input of cold from an outside source. This has to be done by using either liquid nitrogen or cold from a natural gas regasification facility (site specific), or any other supply of cold. This thesis will consider the liquid nitrogen [Ref 5.1] and liquefied natural gas regasification [Ref 5.2] options as viable and possible sources of cold for the operation of the plant.

5.1: Analysis of C-120-34-30@113.1K cycle assisted by liquid nitrogen cooling

Overview on analysis of C-120-34-30@113.1K with liquid nitrogen assisted cooling

The analysis has established that this configuration has the ability to run with liquid nitrogen cooling as supplied by a mass flow which is 24% of main circulating flow. At this mass flow rate, it attains an ideal COP of 5.631. This cycle has a lower demand for nitrogen cooling and its DCAR is good at 0.718(71.8%). More results of the analysis are given in table 5.1.a.

Summary of C-120-34-30@113.1K after introducing liquid nitrogen cooling

Table 5.1.a summarises the performance of the effect of using nitrogen in supplementing the cooling requirements of C-120-34-30@113.1K with liquid nitrogen in comparison with the original cycle where cooling was incomplete. This is a practical cycle because the cooling deficit has been filled. The cycle in part 3.2 (base case) had not been charged with nitrogen.

Table: 5.1.a: Summary of C-120-34-30@113.1K after introducing liquid nitrogen cooling

SYSTEM C-120-34-30@113.1K	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-14.26	
EX-80-29	0	
FHRU-55-15	0	
Total Work input		-14.26
C-120-34-30 Net Work		**+66.04**
Heat Supplied (from ambient) (255.2-127.7)	+127.5	
Heat Rejected (balanced from liquid nitrogen reserve or LNGRS) (-41.47-[-11.342])	-60.758	
Net heat Supplied from ambient		+66.922
COP/Work ratio		**5.631**
Specific nitrogen consumption per Kg (SNC)		0.24
DCAR		0.718

Figure 5.1.a, the below T-S diagram shows the profiled heat zones after nitrogen cooling is applied.

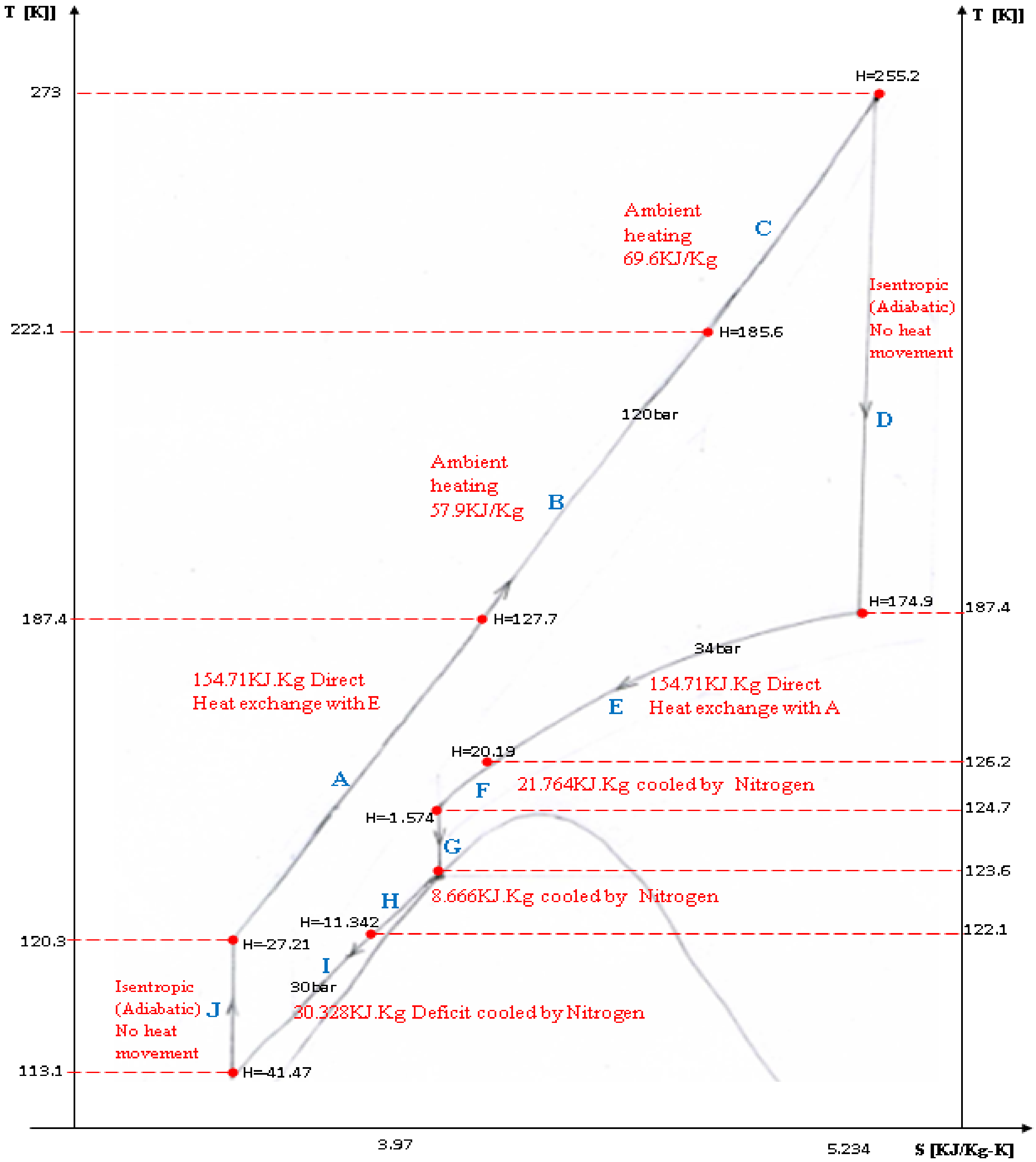


Figure 5.1.a: T-S diagram of C-120-34-30@113.1K profiled heat zones after nitrogen cooling.

Figure 5.1.b shows the reconfigured flow diagram after using nitrogen as the source of cold. Note that the SHTS are not featured in this case because they aren't playing any part in cold supply.

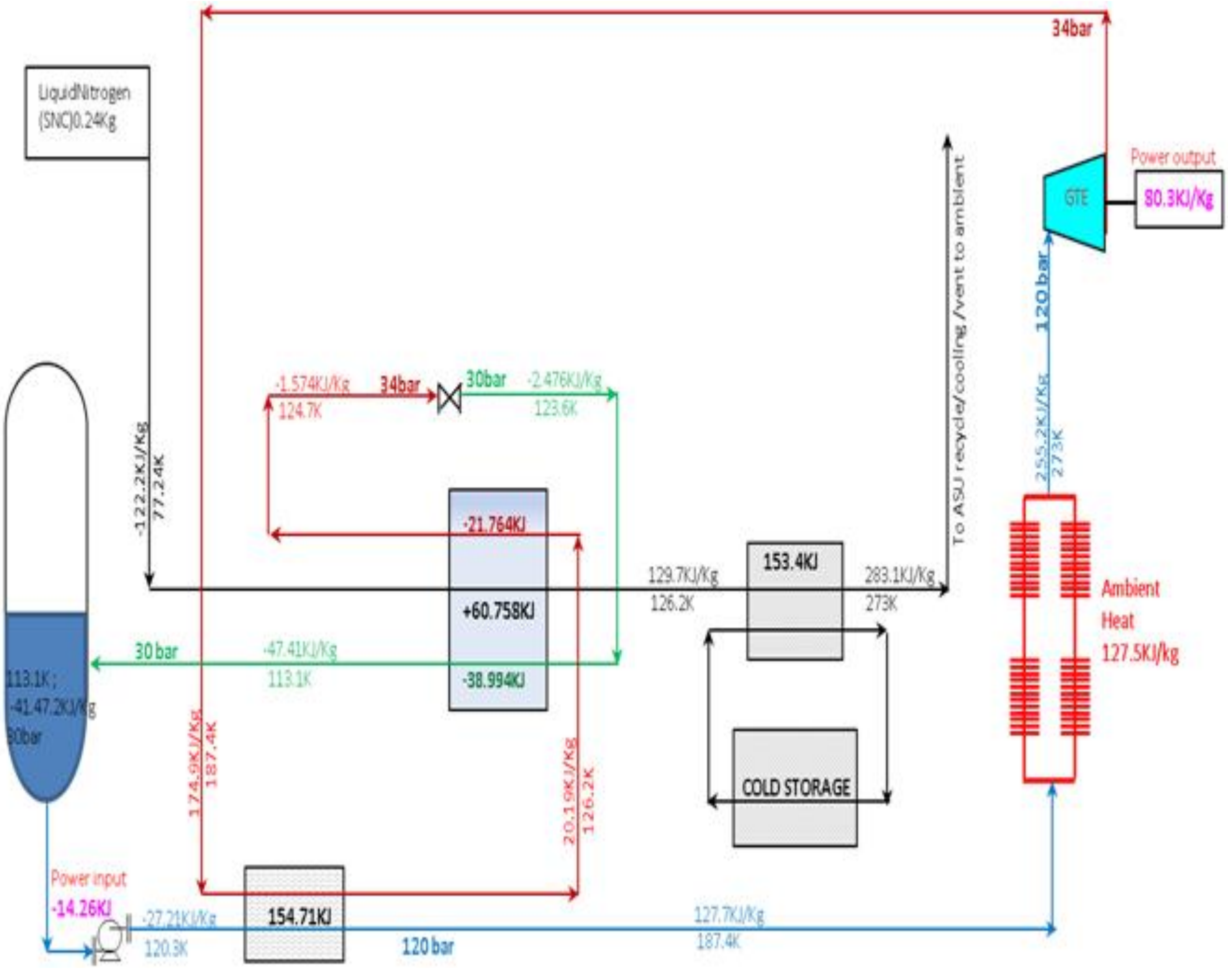


Figure 5.1.b: Flow diagram of a balanced C-120-34-30@113.1K cycle operating on nitrogen cooling

5.1.1: Assumptions

The liquid nitrogen for this purpose shall be sourced from excess electricity of off-peak/intermittent renewables.

5.1.1.1: Establishing the net amount of cooling from liquid nitrogen for C-120-34-30@113.1K

Note: Please refer to figures 2.4.1, 3.1.1.a, Table 2.4.3.3.b and the exercise carried out in part 3.2.

In part 3.2.2, C-120-34-30@113.1K required some cooling of 60.758 KJ/Kg at 30 bar from 113.1K upwards [Table 3.2.1.a]. This is to be supplied by vaporization of liquid nitrogen. Table 5.1.1.1 gives details of the enthalpy(KJ/Kg) available to the cycle when the required amount of liquid nitrogen is vaporised.

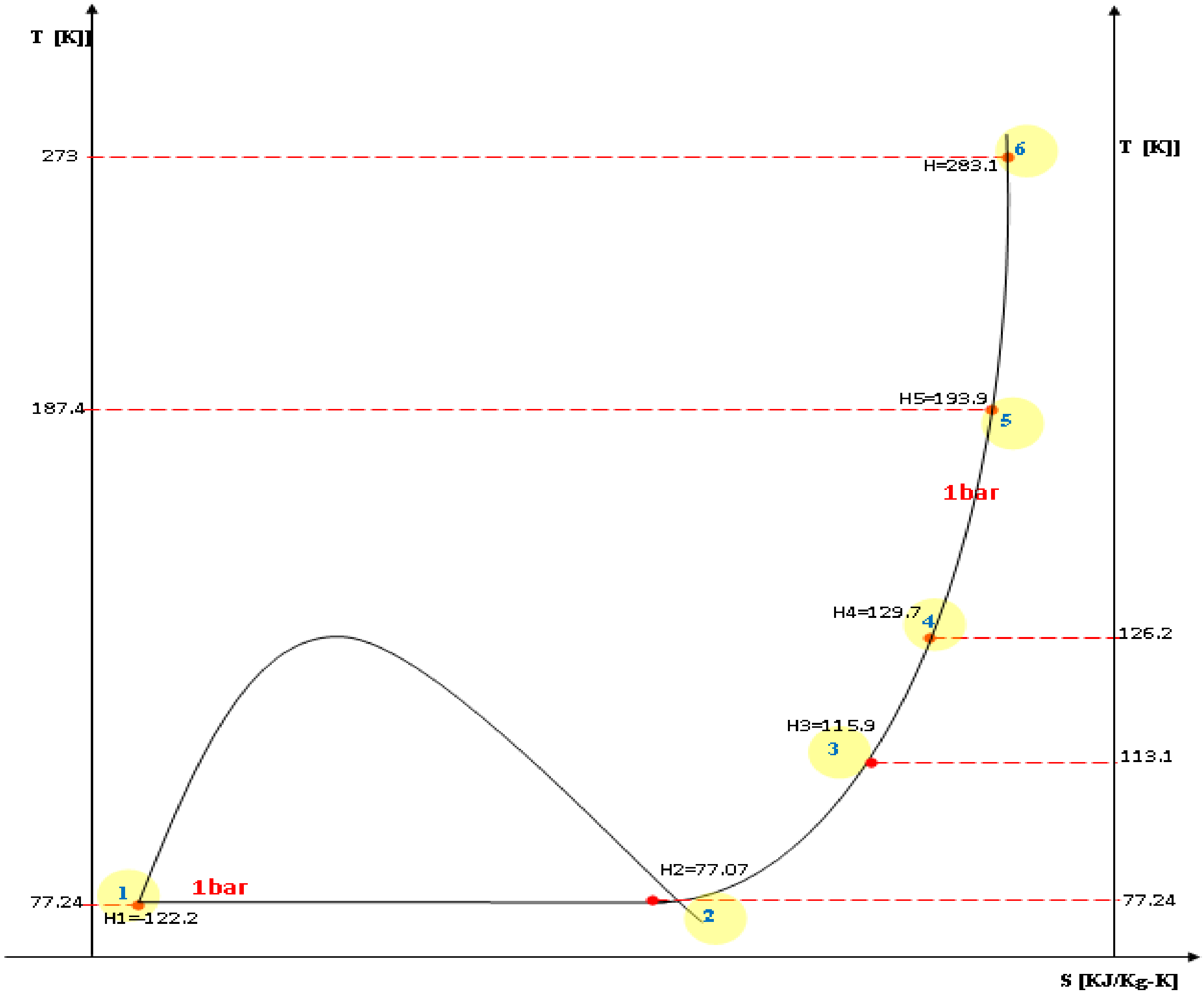


Figure 5.1.1.1: A profiled T-S diagram of nitrogen vaporisation at 1bar for C-120-34-30@113.1K cooling

Table: 5.1.1.1: Values of Enthalpy/Pressure and temperature of liquid nitrogen vaporisation at 1bar.

Vaporisation of liquid nitrogen		Pressure/ Temperature (Bar/K)	Enthalpy (KJ/Kg)	Cooling availed by 0.24Kg/s [Ref 5.2.1.1] of liquid nitrogen to C-120-34-30@113.1K
Node	Enthalpy			
1	-122.2	1 / 77.24		
2	77.07	1 / 77.24		
1-2			199.27	47.825
2	77.07	1 / 77.24		
3	115.9	1 / 113.1		
2-3			38.83	9.3192
3	115.9	1 / 113.1		
4	129.7	1 / 126.2		
3-4			13.8	3.312
4	129.7	1/126.2		
5	193.9	1/187.4		
5-4			65.2	15.648
Cold available to 120-34-30@113.1K operational profile				
5	193.9	1 / 187.4		
6	283.1	1 / 273		
5-6			89.2	21.408
Net cold available from liquid nitrogen vaporisation to ambient (1-6)			405.3	97.272

Nitrogen cooling calculations:

$$Operating\ Mass\ flow\ \ = \frac{Cooling\ required\ by\ [C - 120 - 34 - 30@113.1K]}{Available\ cold\ from\ Nitrogen\ vaporisation}$$

$$Operating\ Mass\ flow\ = \frac{60.578}{251.9} = 0.24048 \cong 0.240Kg/s$$

This means for every 1Kg flow on the CHEC cycle will require 0.240Kg of liquid nitrogen vaporisation to match the cycle’s cooling requirements. This leads to specific nitrogen consumption (SNC) requirement of 24% of the total fluid medium flow.

5.1.1.2: The complete 120-34-30@113.1K power cycle (with nitrogen assisted cooling)

The cycle will have to be reconfigured in order to take advantage of the extra cooling provided by the vaporising stream of nitrogen at different temperature stages as it loses enthalpy while gaining temperature towards ambient. The nitrogen cooling provides an opportunity for reducing the work input from some or all of the SHTS in case they are used. This can be done by providing some side streams of GTOS to undergo direct heat transfer in a PFHE as the nitrogen gains enthalpy towards ambient. There are two options of how the nitrogen extra cooling can be utilized, these are

- Extra GTOS gas cooling correction from ideal at 34bars / zone E
- ASU intake for economical nitrogen recovery above 187.4K.
- Easy accessible cold sink for SHTS in reducing or regulating the amount of nitrogen consumption verses power production.

5.2: Analysis of C-120-34-30@113.1K cycle assisted by LNGR cooling

Overview on analysis of C-120-34-30@113.1K with LNGR assisted cooling

The analysis has established that this configuration has the ability to run with an LNGR mass flow ratio of 1.563 per Kg of N2 circulation. At this mass flow rate, it has a COP of 5.631. The cold demand flow rate is higher because of the need to sub-cool the liquid. The DCAR is good, at 0.718. More results of the analysis are given in table 5.2.a

Table 5.2.a summarises the performance of the effect of using LNGR to supplement the cooling requirements of C-120-34-30@113.1K in comparison with the original cycle(base case) where cooling was incomplete. The cooling deficit that existed in the basecase profile [3.2] is provided for by LNGR cooling in this part.

Table: 5.2.a: Summary of C-120-34-30@113.1K after using LNGRS cooling

SYSTEM C-120-34-30@113.1K	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-14.26	
EX-80-29	0	
FHRU-55-15	0	
Total Work input		-14.26
C-120-34-30 Net Work		**+66.04**
Heat Supplied (from ambient) (255.2-127.7)	+127.5	
Heat Rejected (balanced LNGRS)	-60.578	
Net heat Supplied from ambient		+66.922
COP/Work ratio		**5.631**
LNGR OMFR (1.08 for 30bar and 0.481 for 34bar)		1.563

Figure 5.1.a below shows the marked heat zones after nitrogen cooling is applied.

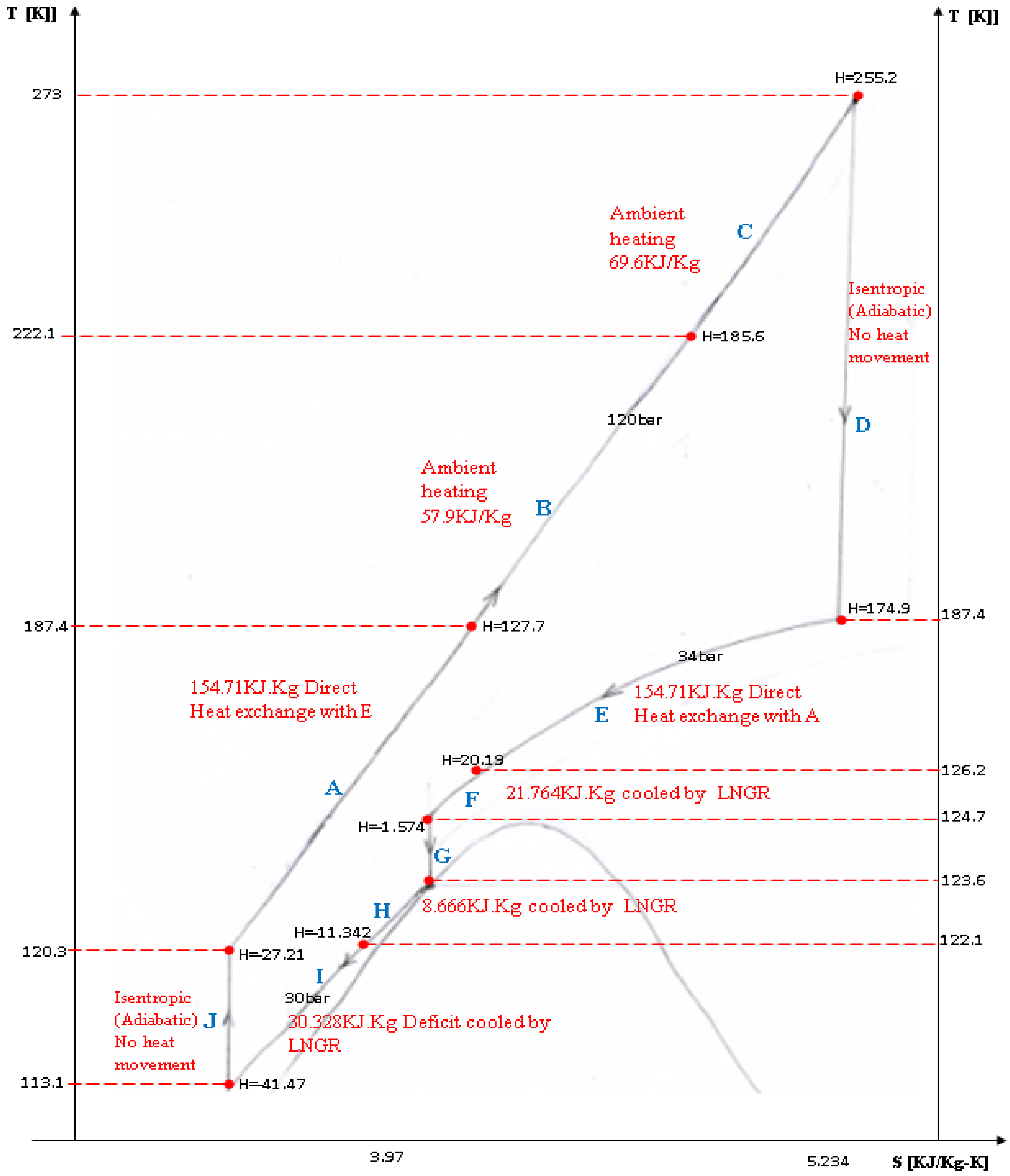


Figure 5.2.a: T-S diagram of C-120-34-30@113.1K profiled heat zones after LNGR cooling.

Figure 5.2.b shows the reconfigured flow diagram after using LNGR as the source of cold. Note that the SHTS are not featured in this case because they aren't playing any part in cold supply.

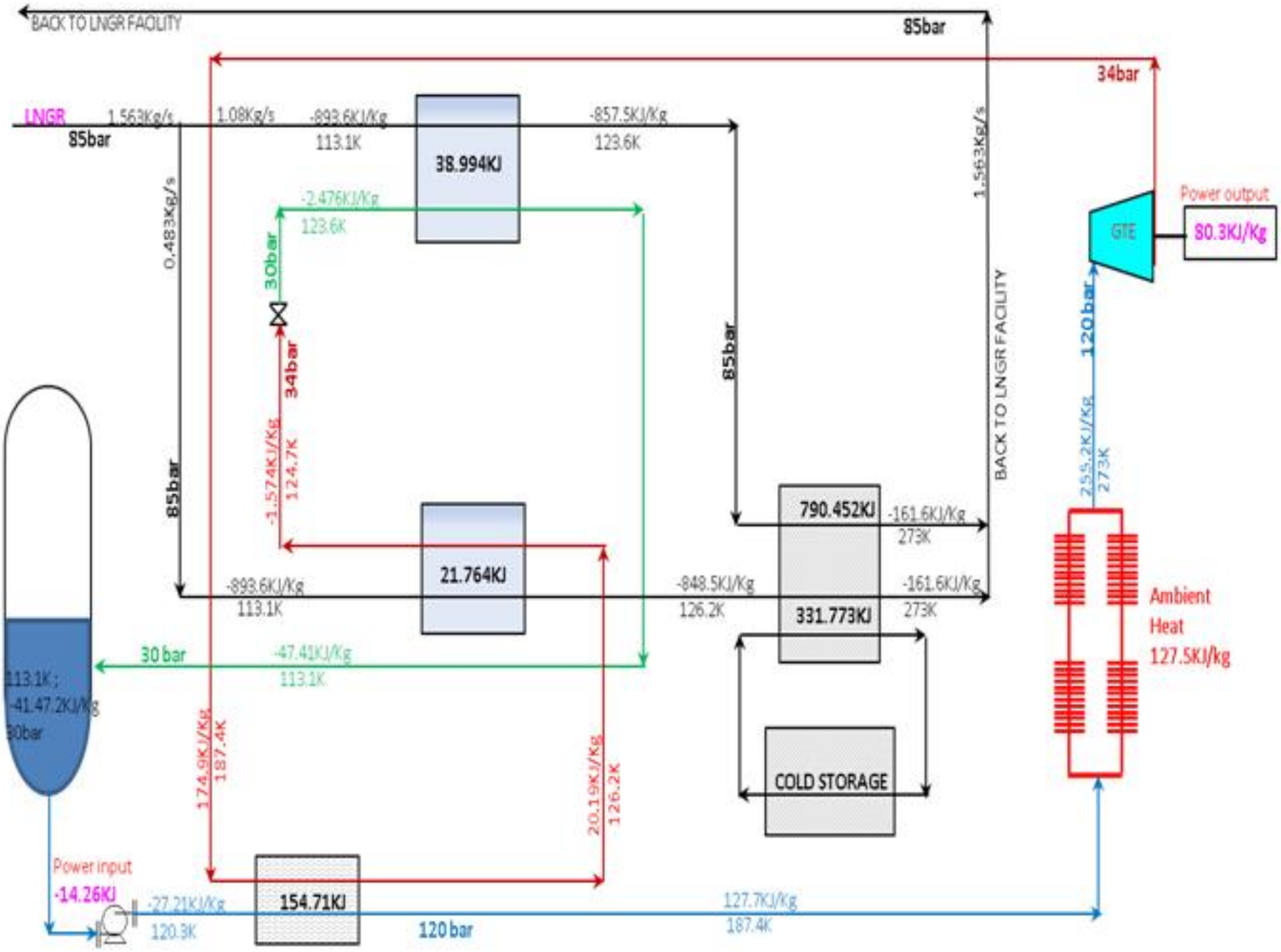


Figure 5.2.b: Flow diagram of a balanced LNGR cooled C-120-34-30@113.1K

5.2.1: Assumptions

Note: The EES platform does not have LNG properties but because its main constituent (@90%) is mostly methane, this study will use the properties of methane to simulate those of LNG. Although there will be some slight differences on the properties of LNG and methane, the limit of their differences shall be acceptable for the purposes of this study and they shall be assumed to be the same. The cold associated with any LNG regasification process is described thermodynamically in part 5.2.1.1.

5.2.1.1: LNG heat balance calculations in the regasification process for cold provision to C-120-34-30@113.1K.

LNG is natural gas which has been cooled to its liquefaction temperature of 111K at 1 bar. The process of regasification may involve heating the gas with ambient heat sourced from air, water or waste heat from combustion of fossil fuel plants of direct firing by fossil fuel based heaters. In order to utilize the cold generated by the LNG regasification process to address the cooling needs of the CHEC power cycle, the GTOS acts to provide some of the heat required for LNG gasification while gaining the cold required for condensation.

LNG is pumped from storage at 1bar / 111K to UK’s mains gas supply pressure of 85bar before undergoing a constant pressure heating process until it attains a temperature of 278K [National grid-LNG]. The heat balance is summarized with respect to C-120-34-30@113.1K temperature configuration as shown in table 5.2.1.1.

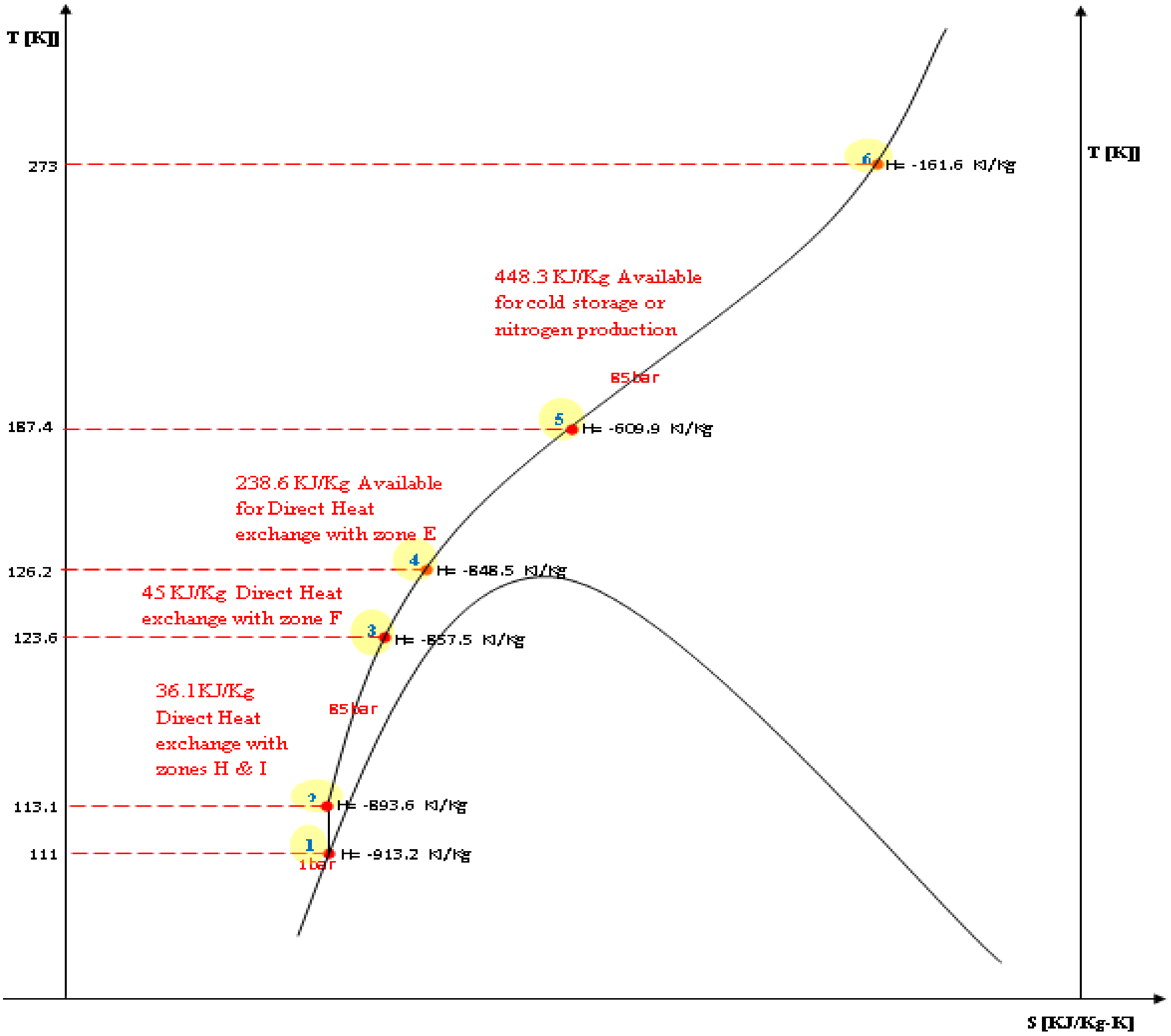


Figure 5.2.1.1: A profiled T-S diagram of LNGR cooling on C-120-34-30@113.1K

Table: 5.2.1.1; LNG heat balance analysis in the regasification process for cold utilization by C-120-34-30@113.1K

Node	LNG Temperature (K)	Enthalpy (KJ/Kg)	Enthalpy available for C-120-34-30@113.1K (KJ/Kg)	Description
1	111	-913.2	0	Initial state
2	113.1	-893.6	0	Isentropic compression (Regasification pump discharge)
3	123.6	-857.5	(-857.5-[893.6])=36.1	Available for GTOS cooling at Zone H and I
	124.7	-853.7		
4	126.2	-848.5	(-848.5-[-893.6])=45	Available for GTOS cooling at Zone F.
	130	-835.3		
	140	-799.9		
	150	-763.5		
	160	-725.9		
	170	-686.4		
5	187.4	-609.9	(-609.9-[-848.5])= 238.6	Available for GTOS ideal cooling correction Zone E or Cold storage
	190	-597.1		
	200	-541.2		
6	273	-161.6	(-161.6-[-609.9])=448.3	Available to Cold storage or nitrogen production.

5.2.1.1.1: Establishing the amount of cooling from LNGR to C-120-34-30@113.1K

Note: Please refer to figures 2.4.1, 3.1.1.a and the exercise carried out in part 3.3

In Table 3.2.3, C-120-34-30@113.1K required some cooling of 60.578KJ/Kg between 124.7K down to 113.1K. This is to be supplied by the LNG regasification process where liquefied natural gas is vaporised back into a gaseous state before being supplied to customers. Table 5.2.1.1 gives details of the enthalpy available for the cycle when LNG is being vaporised.

5.2.1.2: Adjustments on C-120-34-30@113.1K in order to utilize the LNGR cooling

In this section, C-120-34-30@113.1K is considered afresh with much of its cooling sink being availed as direct cooling by CLS (154.71KJ/Kg) and the deficit filled by LNGRS (60.758KJ/kg). The main reason of using CLS is to reduce the heating load at the ambient heaters. The cold requirements can be calculated and the flow of LNGR be adjusted to provide the required enthalpy. The enthalpy quantities will be matched by the number of transfer unit between the two flow streams. The enthalpy on the LNGR stream between the temperatures is balanced in the heat exchanger

Table: 5.2.1.2: Heat exchange between LNGR and GTOS streams at 30 bar

Stream		Inlet	Outlet	Heat Transfer (KJ/Kg)	Temperature Change
LNGR@85bar	Temperature(K)	113.1	123.6		+10.5
	Enthalpy(KJ/Kg)	-893.6	-857.5	36.1	
GTOS@30bar	Temperature(K)	123.6	113.1		-10.5
	Enthalpy(KJ/Kg)	-2.476	-41.47	-38.994	

From the above table, the flow of LNG will have to be higher so as to match the cold requirements of GTOS. The required LNGR stream mass flow ratio is calculated below as the ratio of enthalpy’s between the two streams.

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Enthalpy\ of\ GTOS\ stream}{Enthalpy\ of\ LNGR\ stream}$$

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{38.994}{36.1} = 1.080$$

This means the mass flow of LNGR stream has to be higher by a factor of 1.08 or +8% for matching the cooling requirements of zone H and I.

Table: 5.2.1.3: Heat exchange between LNGR and GTOS streams at 34 bar.

Stream		Inlet	Outlet	Heat Transfer (KJ/Kg)	Temperature Change
LNGR@85bar	Temperature(K)	113.1	126.2		+13.1
	Enthalpy(KJ/Kg)	-893.6	-848.5	+45.1	
GTOS@30bar	Temperature(K)	126.2	124.7		-1.5
	Enthalpy(KJ/Kg)	20.19	-1.574	-21.764	

The mass flow ratio is calculated to balance the enthalpies of the two streams

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Enthalpy\ of\ GTOS\ stream}{Enthalpy\ of\ LNGR\ stream}$$

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{21.764}{45.1} = 0.483$$

This means the mass flow of LNGR stream has to be less by a factor of 0.483 or (51.7% less) for zone H to be matched.

The total LNG mass flow is 1.08+0.483=1.563Kg/s

CHAPTER 6

FURTHER ANALYSIS OF C-120-34-30@123.6K CYCLE WITH NITROGEN AND NATURAL GAS COOLING

Chapter overview

As observed from the analysis of the basecase system [3.3], C-120-34-30@123.6K power cycle cannot generate enough cold to sustain its heat and work balance in a cyclic manner without the input of cold from an outside source. This has to be done by using either liquid nitrogen or cold from natural gas regasification (site specific), or any other supply of cold. This thesis will consider the use of liquid nitrogen [Ref 6.1] and liquefied natural gas regasification [Ref 6.2] as options of cold provision to the power cycle.

Chapter contents

6.1: Analysis of C-120-34-30@123.6K cycle cooled by liquid nitrogen

Overview on analysis of C-120-34-30@123.6K with liquid nitrogen cooling

This analysis has established that this configuration has the ability to run with liquid nitrogen supply of 39.4% mass of main circulating mass flow. At this mass flow rate, it has a COP of 4.507. This cycle has a higher demand for nitrogen cooling and its DCAR is poor. More results of the analysis are given in table 6.1.a.

Summary results of C-120-34-30@123.6K analysis after introducing liquid nitrogen cooling

Table 6.1 summarises the performance of the effect of using nitrogen to supplement the cooling requirements of C-120-34-30@123.6K with liquid nitrogen in comparison with the original cycle (base case) where cooling was deficient. It therefore becomes a practical cycle once the cooling gap has been filled. The cycle in part 3.3 had not been charged with nitrogen but this one has.

Table: 6.1.a: Summary of C-120-34-30@123.6K after introducing liquid nitrogen cooling

SYSTEM C-120-34-30@123.6K	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-17.816	
EX-80-29	0	
FHRU-55-15	0	
Total Work input		-17.816
C-120-34-30@123.6K Net Work		**+62.484**
Heat Supplied (from ambient) (255.2-85.06)	+170.14	
Heat Rejected (balanced from liquid nitrogen reserve) (-1.574-103.1)	-104.674	
Net heat Supplied from ambient		+65.466
COP/Work ratio		**4.507**
Specific nitrogen consumption per Kg (SNC)		0.394

Figure 6.1.a below shows the marked heat zones after nitrogen cooling is applied.

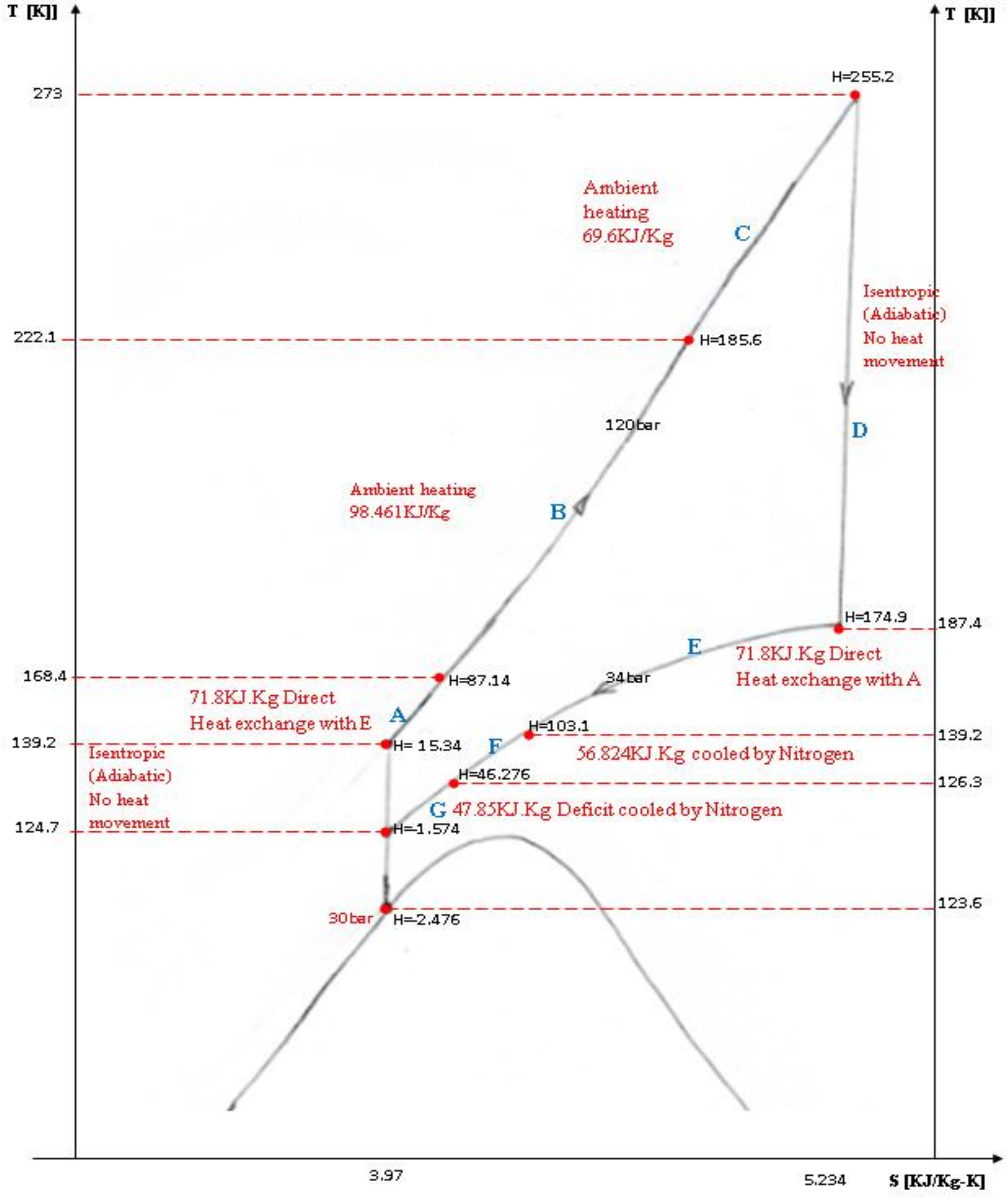


Figure 6.1.a: C-120-34-30@123.6K profiled heat zones after nitrogen cooling

Figure 6.1.b shows the reconfigured flow diagram after using nitrogen as the source of cold. Note that the SHTS are not featured in here because they aren't playing any part in cold supply.

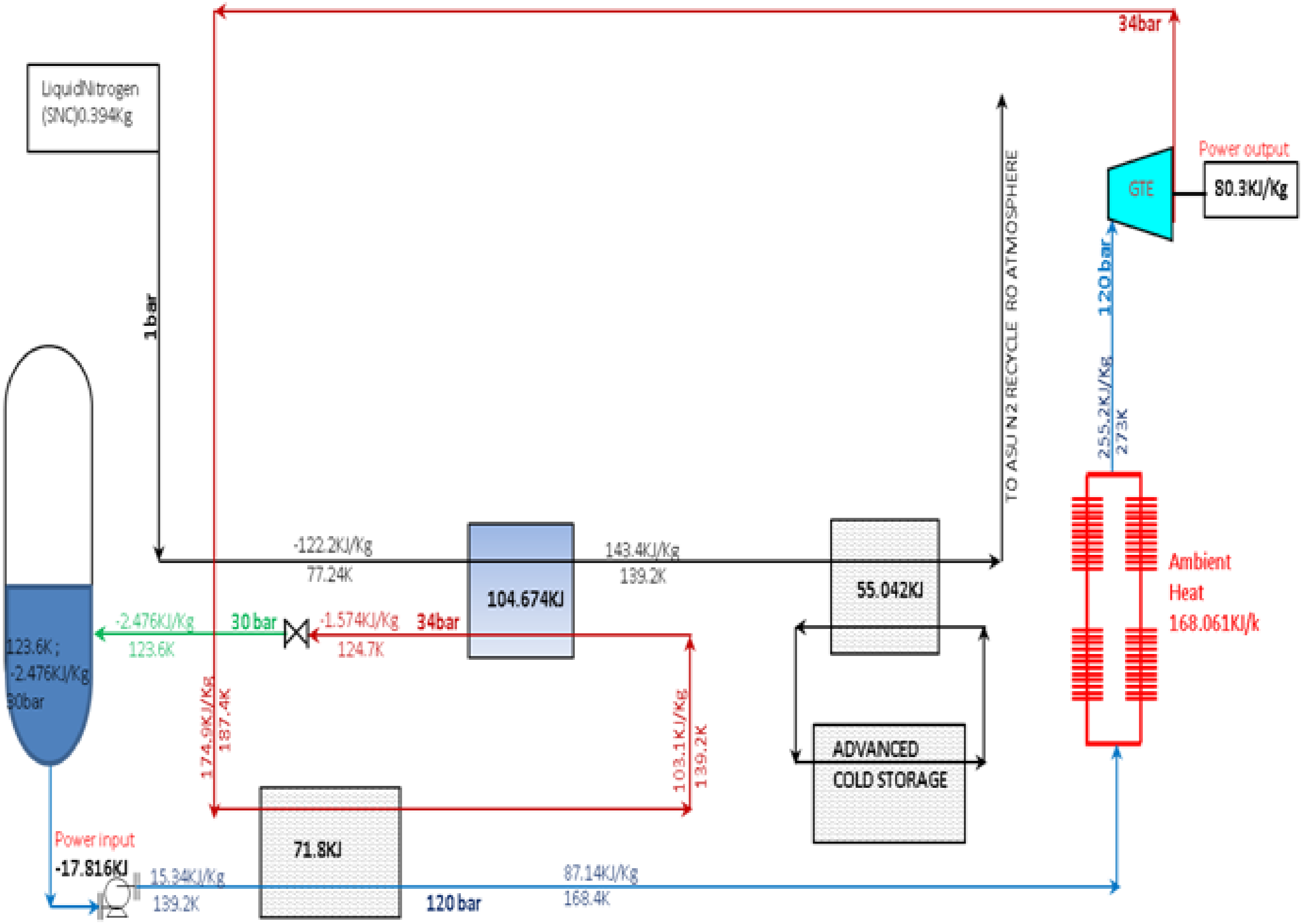


Figure 6.1.b: Flow diagram of a balanced base case C-120-34-30@123.6K cycle after nitrogen cooling.

6.1.1: Assumption

The liquid nitrogen for this purpose shall be sourced from excess/ off-peak electricity and or intermittent renewables.

6.1.1.1: Establishing the net amount of cooling from liquid nitrogen for C-120-34-30@123.6K

Note: This analysis is done with reference to figures 2.4.1, 3.1.1.a, Table 2.4.3.3.b and the exercise carried out in part 3.3.

In part 3.3.2, C-120-34-30@123.6K required some cooling of 104.674 KJ/Kg at 30 bar from 113.1K upwards [Table 3.2.1.a]. This is to be supplied by vaporization of liquid nitrogen. Table 6.1.1.1 gives details the enthalpy available for the cycle when the required amount of liquid nitrogen is vaporised.

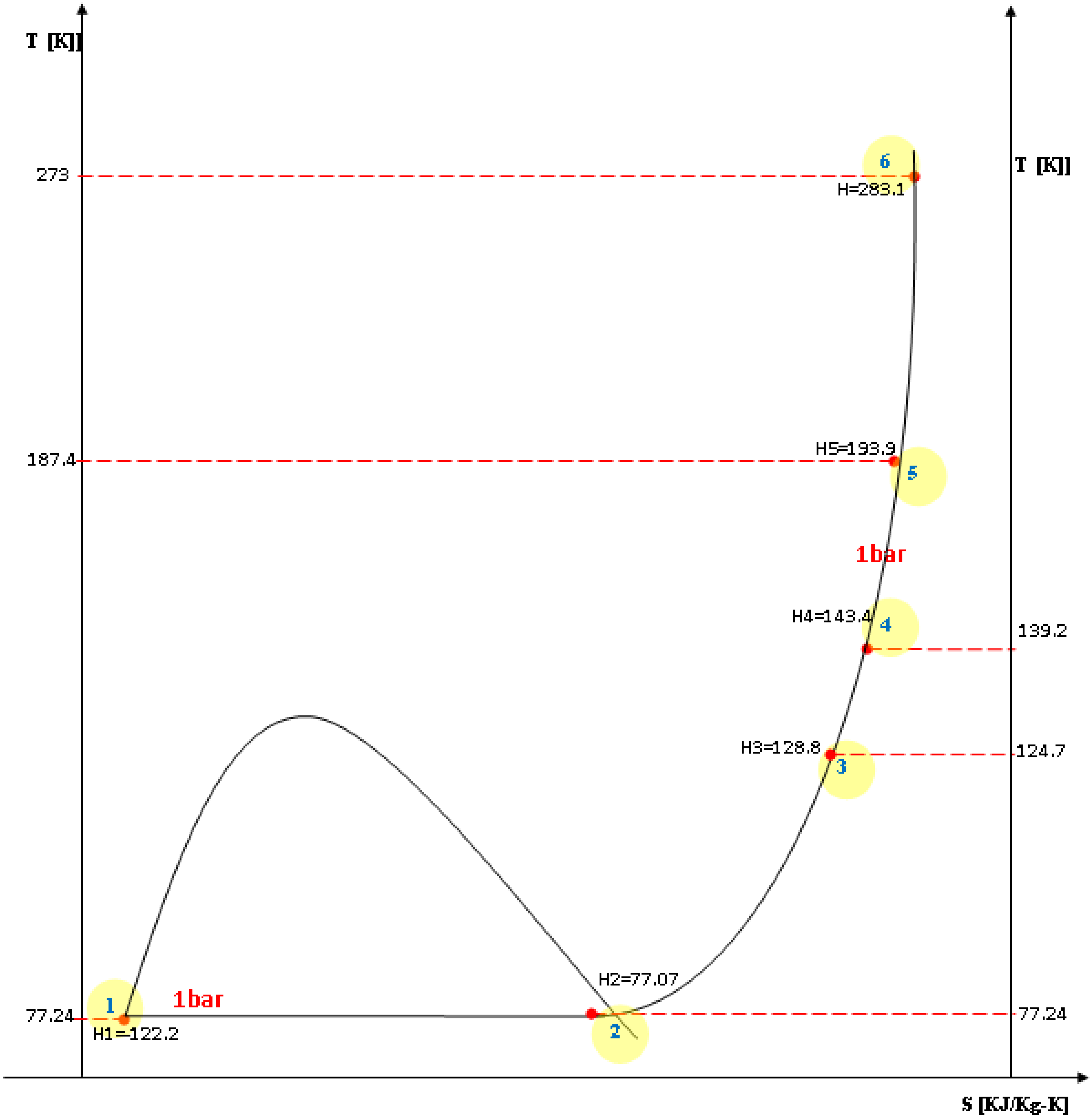


Figure 6.1.1.1: A profiled T-S diagram of nitrogen vaporisation at 1bar for C-120-34-30@123.6K cooling

Table: 6.1.1.1: Values of Enthalpy/pressure and temperature of liquid nitrogen vaporisation at 1 bar for C120-34-30@123.6K cooling

Vaporisation of liquid nitrogen		Pressure/ Temperature (Bar/K)	Enthalpy (KJ/Kg)	Cooling availed by 0.394Kg/s [Ref 6.2.1.1] of liquid nitrogen to C-120-34-30@123.6K
Node	Enthalpy			
1	-122.2	1 / 77.24		
2	77.07	1 / 77.24		
1-2			199.27	78.512
2	77.07	1 / 77.24		
3	128.1	1 / 124.7		
2-3			51.03	20.106
3	128.1	1 / 124.7		
4	143.4	1 / 139.2		
3-4			15.3	6.028
4	143.4	1/139.2		
5	193.9	1/187.4		
5-4			65.2	25.689
Cold available to 120-34-30@113.1K operational profile				
5	193.9	1 / 187.4		
6	283.1	1 / 273		
5-6			89.2	35.145
Net cold available from liquid nitrogen vaporisation to ambient (1-6)			405.3	159.688

Nitrogen cooling calculations:

$$Operating\ Mass\ flow \ = \frac{Cooling\ required\ by\ [C-120-34-30@123.6K]}{Available\ cold\ from\ Nitrogen\ vaporisation}$$

$$Operating\ Mass\ flow \ = \frac{104.674}{265.6} = 0.39410 \cong 0.394Kg/s$$

This means for every 1Kg flow on the CHEC cycle, 0.394Kg of liquid nitrogen will be needed to match the cooling requirements. This establishes specific nitrogen consumption (SNC) requirement of 39.4% of the total fluid flow on the main cycle.

6.1.1.2: A balanced C-120-34-30@123.6K power cycle (with liquid nitrogen cooling)

The cycle will have to be reconfigured in order to take advantage of the extra cooling provided by the vaporising stream of nitrogen at different temperature stages as it loses enthalpy. The nitrogen cooling provides an opportunity for reducing the work input from some or all of the SHTS. This can be done by providing some side streams of GTOS to undergo direct heat transfer in a PFHE as the nitrogen gains enthalpy towards ambient. There are two options of how the nitrogen extra cooling can be utilized, these are

- Extra GTOS gas cooling correction from ideal at 34bars / zone E
- ASU intake for economical nitrogen recovery above 187.4K.
- Easy accessible cold sink for SHTS in reducing or regulating the amount of nitrogen consumption verses power production.

6.2: Analysis of C-120-34-30@123.6K cycle cooled by LNGR regasification

Overview on analysis of C-120-34-30@123.6K with LNGR assisted cooling

This analysis has established that this configuration has the ability to run with an LNGR mass flow ratio of 1.162 per Kg of N2 circulation. At this mass flow rate, it has a COP of 4.507. The demand flow rate is higher because of the poor DCAR. More results of the analysis are given in table 6.2.a

Table 6.2.a summarises the performance of the effect of using nitrogen for supplementing the cooling requirements of C-120-34-30@123.6K with LNGR in comparison with the original cycle where cooling was incomplete. The cooling deficit that existed in the basecase profile [3.3] is completed by LNGR cooling in this part.

Table: 6.2.a: Summary of C-120-34-30@123.6K after using LNGRS cooling

SYSTEM C-120-34-30@123.6K	**(KJ/Kg)**	**(KJ/Kg)**
Work output		
GTE	+80.3	
Total Work output		+80.3
Work input		
LCP pump	-17.816	
EX-80-29	0	
FHRU-55-15	0	
Total Work input		-17.816
C-120-34-30 Net Work		**+62.484**
Heat Supplied (from ambient) (255.2-127.7)	+170.14	
Heat Rejected (balanced LNGRS)	-104.674	
Net heat Supplied		+65.466
COP/Work ratio		**4.507**
LNGR OMFR(per Kg of N2 circulation)		1.162

Figure 6.1.a below shows the marked heat zones after nitrogen cooling is applied.

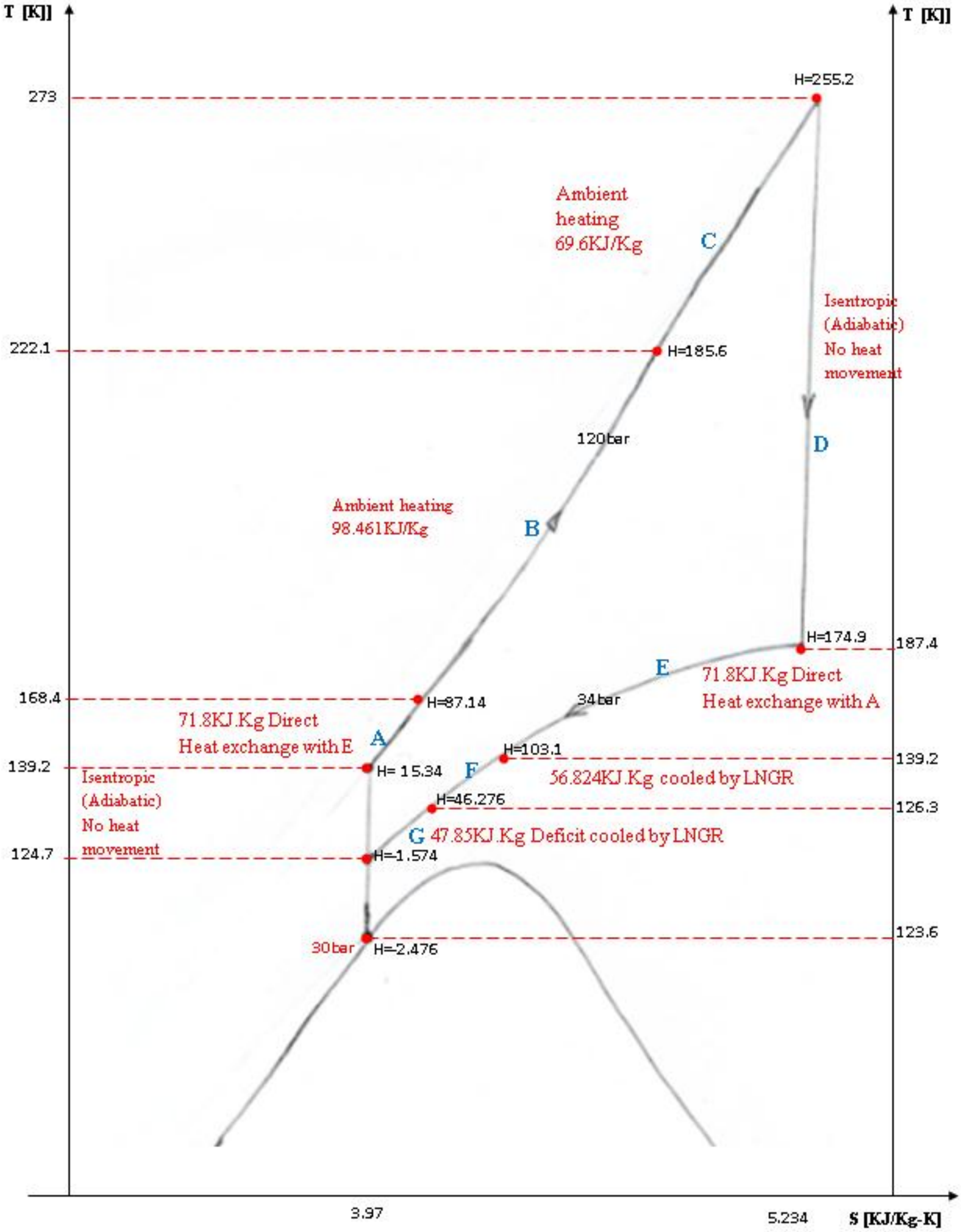


Figure 6.2.a: C-120-34-30@123.6K profiled heat zones after LNGR cooling

Figure 6.2.b shows the reconfigured flow diagram after using LNGR as the source of cold. Note that the SHTS are not featured in here because they aren't playing any part in cold supply.

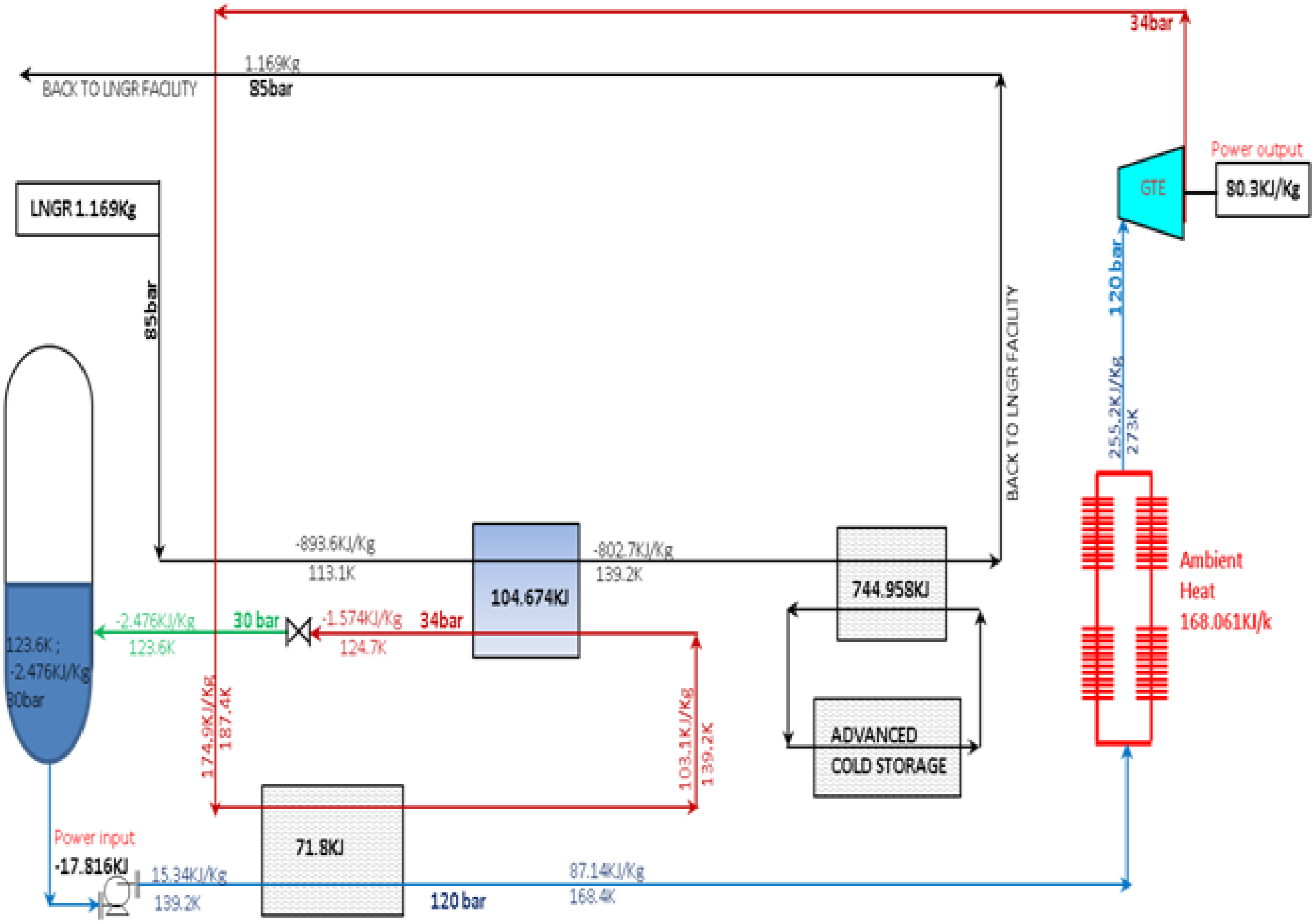


Figure 6.2.b: Flow diagram of a balanced LNGR cooled C-120-34-30@123.6K

6.2.1: Assumptions in the analysis

Note: The EES platform does not have LNG properties but since is main constituent is mostly methane (@ more than 90%), this study will use the properties of methane to simulate those of LNG. The differences on the properties of LNG and methane shall be assumed to be negligible. The cold associated with any LNG regasification process is described thermodynamically in part 6.2.1.1.

6.2.1.1: LNG heat balance calculations in the regasification process for cold provision to C-120-34-30@123.6K

LNG is natural gas which has been cooled to its liquefaction temperature of 111K at 1 bar. The process of regasification involves heating the gas with ambient heat sourced from air, water or waste heat. In order to utilize the cold generated by the LNG regasification process to address the cooling needs of the CHEC power cycle, the GTOS acts to provide some of the heat required for LNG gasification.

LNG is pumped from storage at 1bar / 111K to UK's mains gas supply pressure of 85bar before undergoing a constant pressure heating process until it attains a temperature of 278K. The heat balance is summarized with respect to C-120-34-30@123.6K temperature configuration as shown in table 6.2.1.1.

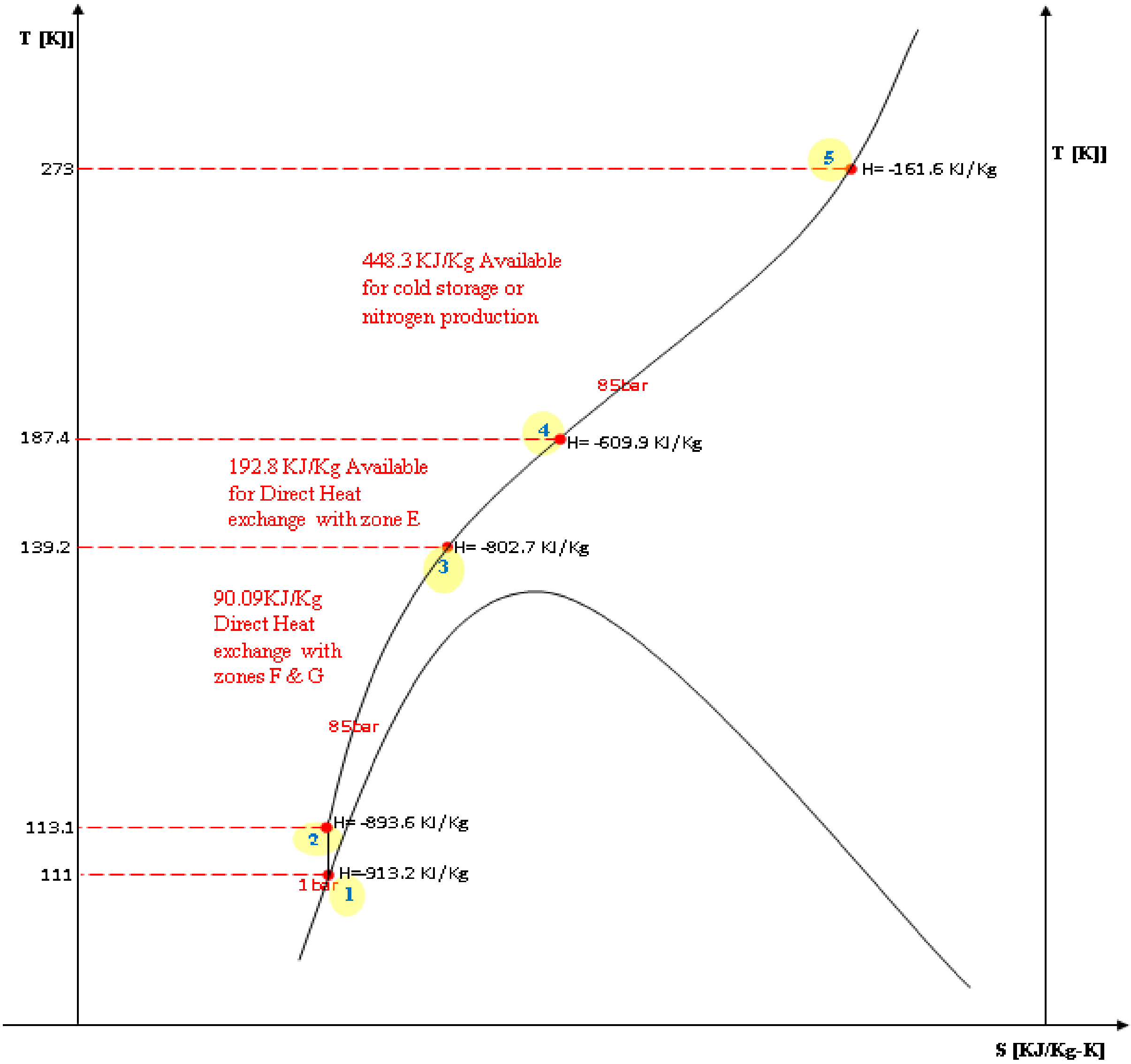


Figure 6.2.1.1: A profiled T-S diagram of LNGR vaporization at 85 bar.

Table: 6.2.1.1: LNGR heat balance analysis in the regasification process for cold utilization by C-120-34-30@123.6K

LNG Temperature (K)	Enthalpy (KJ/Kg)	Enthalpy available for C-120-34-30@123.6K (KJ/Kg)	Description
111	-913.2	0	Initial state
113.1	-893.6	0	Isentropic compression (Regasification pump discharge)
123.6	-857.5		
124.7	-853.7		
126.2	-848.5		
130	-835.3		
139.2	-802.7	(-802.7-[-893.6])=90.09	Available for GTOS cooling at Zone G and F.
140	-799.9		
150	-763.5		
160	-725.9		
170	-686.4		
187.4	-609.9	(-609.9-[-802.7])= 192.8	Available for GTOS ideal cooling correction between Zone E
190	-597.1		
200	-541.2		
273	-161.6	(-161.6-[-609.9])= 448.3	Available for cold storage or nitrogen production

6.2.1.1.1: Establishing the amount of cooling from LNGR for C-120-34-30@123.6K

Note: This analysis is done with reference to figures 2.4.1, 3.1.1.a and the exercise carried out in part 3.3

In part Table 3.3.3, C-120-34-30@123.6K required some cooling of 104.674KJ/Kg between 124.7K down to 113.1K. In this case it is assumed to be supplied from the LNG regasification process where liquefied natural gas is vaporised back into a gaseous state before being supplied to customers. Table 6.2.1.1 gives details the enthalpy available for the cycle when LNG is being vaporised

6.2.1.2: Adjustments on C-120-34-30@123.6K so as to utilize the LNGR cooling

In this section, C-120-34-30@123.6K is re-balanced afresh with much of its cooling sink being availed as direct cooling by CLS (71.8KJ/Kg) and the deficit supplied by LNGRS (104.674KJ/kg). The main reason of using CLS is to reduce the heating load at the ambient heaters. The cold requirements are evaluated and the flow of LNGR adjusted proportionally to provide the required (cooling) enthalpy. The enthalpy quantities will be matched by the number of transfer units between the two flow streams. The enthalpy on the LNGR stream between the temperatures is balanced in the heat exchanger shown in Table 6.2.1.2.

Table: 6.2.1.2: Heat exchange between LNGR and GTOS streams at 34 bar

Stream		Inlet	Outlet	Heat Transfer (KJ/Kg)	Temperature Change
LNGR@85bar	Temperature(K)	113.1	139.2		+26.1
	Enthalpy(KJ/Kg)	-893.6	-857.5	90.09	
GTOS@30bar	Temperature(K)	139.2	124.7		-14.5
	Enthalpy(KJ/Kg)	103.1	-1.574	-104.674	

From the above table, the flow of LNG will have to be higher so as to match the cold requirements of GTOS. The required LNGR stream mass flow ratio is calculated below as the ratio of enthalpy's between the two streams.

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{Enthalpy\ of\ GTOS\ stream}{Enthalpy\ of\ LNGR\ stream}$$

$$LNGR\ operating\ Mass\ flow\ Ratio\ (OMFR) = \frac{104.674}{90.09} = 1.162$$

This means the mass flow of LNGR stream has to be higher by a factor of 1.162 or 16.2% for zone H cooling requirements to be matched.

CHAPTER 7

RESULTS OF ANALYSES AND DISCUSSION

Chapter contents:

Chapter overview

Note: All cycles discussed in this chapter are ideal, just as it is in the whole thesis.

The key factors which determine the favourability of a CHEC power cycle configurations are to be found in [Ref: 7.4];

- Higher direct cooling ability
- Not having initial nor final operations near the fluid medium's critical point in most thermodynamic properties.
- The liquid point should not be on or too close to the saturated liquid line, instead, it should be way far into the sub-cooled region.

The results have demonstrated that C-120-30-1 is not suitable for a closed system nitrogen power cycle. It is difficult to afford and sustain cooling requirements at very low temperature/pressure for fluid medium condensation. In order to satisfy the cooling requirements of such a cycle, a lot of energy and a great deal of equipment will need to be installed. This makes it an unlikely choice as a cycle for power production under both nitrogen/LNGR cooling. Parts 7.1 and 7.1.1 have elaborated on the details of its thermodynamic unsuitability as a power cycle of choice.

The higher pressure condensation cycles have proved to be superior in many performance criteria's as shown in table 7.2 and 7.3. However, C-120-34-30@123.6K has the risk of being unstable at its lowest point is too close to the critical pressure and temperature of nitrogen. The fact that liquid pumping is done at 30bar saturated liquid point makes it vulnerable to cavitation due to vapour formation as the position may easily drift in to the two phase region whenever there may be plant upsets.

The cycle of choice is C-120-34-30@113.1K because it has the advantages of being off from the unstable region due to sub-cooling of liquid. Though its pressure is not far off from the critical pressure of nitrogen, its temperature is pulled down to counter that closeness and also enhance the liquid phase stability status. Its liquid pumping/compression position on the T-S diagram is way off the liquid line and that assures a guarantee of sustaining its liquid status in the accumulator vessel. In part 7.5, C-120-34-30@113.1K results have been explored further for performance variation while using different amounts of SNC so as to understand the position of its flexibility in power output with lesser nitrogen cooling. Details of full results and discussions on the high pressure condensation cycles are given in parts 7.2, 7.2.1, 7.3, and 7.3.1 respectively.

7.1: C-120-30-1 results

This part derives all its data from parts 3.1, and chapter 4. The results in table 7.1 are those of an ideal cycle.

Table: 7.1: Results from the analysis of an ideal C120-30-1 cycle.

	Base case	**Nitrogen cooled**	**LNGR cooled**
Net Work output	**+38.849**	**+44.361**	**+53.554**
Total work out	**+117.414**	**+117.414**	**+117.414**
Total work in	**-78.565**	**-73.053**	**-63.86**
SHTS work input	**-64.065**	**-58.553**	**-49.36**
COP(Electrical)	**1.494**	**1.607**	**1.838**
SNC	**0.15**	**1.5**	**N/A**
LNGR-(OMFR)	**N/A**	**N/A**	**1**
Ambient heat Load	**+69.6**	**+82.625**	**+169**
Cooling Load	**-29.85**	**-37.381**	**N/A**
Net heat	**+39.75**	**+53.736**	**N/A**
Direct cooling ability ratio (DCAR)	**0.526**		

7.1.1: Discussion on C-120-30-1 results

This configuration requires a lot of SHTS input even after providing external cooling with either liquid nitrogen or LNGR. The ideal COP is relatively lower (1.4 to 1.6) which means that it will be very difficult to produce any work output in case of an actual cycle. The massive involvement of SHTS entails a heavy burden on capital investment and makes it unlikely to be a viable power producing cycle. In the actual power plant, the COP is expected to fall further and that makes it an unfavourable option. The greatest hurdle is the sustenance of the thermodynamic status at of the circulating fluid at its final condensation point at 1bar/77K. This point is beyond the LNGR initial cooling point of 113.1K and will consume a lot of liquid nitrogen as well as a lot of SHTS work input. The fact that direct cooling cannot be utilised for condensation makes it mandatory to have SHTS and thus the lower COP. SHTS are subject to thermodynamic efficiencies which will further increase the power input in the actual cycle and that results in a further fall in COP.

This configuration is therefore not practically viable unless it is adjusted to higher pressures where condensation can be accomplished at higher temperatures within the regions of considerable direct cooling by either liquid nitrogen or LNGR.

7.2: C-120-34-30@113.1K results

This part derives all its data from parts 3.2, and chapter 5. The results in table 7.2 are those of an ideal cycle of this configuration..

Table: 7.2: Results from the analysis of an ideal C120-34-30@113.1K cycle

	Base case	**Nitrogen cooled**	**LNGR cooled**
Net Work output	**+38.585**	**+66.04**	**+66.04**
Total work out	**+80.3**	**+80.3**	**+80.3**
Total work in	**-41.765**	**-14.26**	**-14.26**
SHTS work input	**-27.505**	**0**	**0**
COP(Electrical)	**1.923**	**5.631**	**5.631**
SNC (per Kg of N2 circulation)	**0.12**	**0.24**	**N/A**
LNGR-(OMFR) (per Kg of N2 circulation)	**N/A**	**N/A**	**1.563**
Ambient heat Load	**+69.6**	**+127.5**	**+127.5**
Cooling Load	**-30.128**	**-60.578**	**-60.578**
Net heat	**+39.472**	**+66.578**	**+66.922**
Direct cooling ability ratio (DCAR)	**0.718**		

7.2.1: Discussion on C-120-34-30@113.1K results

The results for this cycle configuration are very impressive. The higher COP after introducing both liquid nitrogen and LNGR cooling makes it a viable power producing cycle under both circumstances. The rise in work output is relative to the increase in ambient load which means that ambient energy is being utilised for the production of work output. This configuration has a short cyclic pathway with the highest DCAR which enhances its ability to operate without any input from SHTS if the cooling load is matched by either liquid nitrogen or LNGR. The SNC is lower and thus it can be used as a power plant operating on stored cryogenic energy. The configuration is pulled off from the liquid line thus having a stable sub-cooled stream which is ideal for liquid compression by the LCP. Although the pressure is closer to its critical point, the temperature is off quite far down which further enhances a stabilised liquid phase status. The lowest point of the cycle is positioned to coincide with the outlet temperature of the LNGR (at 113.1K) thus allowing for direct heat exchange without deploying any SHTS.

This configuration is therefore highly promising to serve the aims of this thesis. It goes well with both liquid nitrogen and LNGR, but some adjustment may be needed to improve the SNC whenever liquid nitrogen is used [Ref: 7.4.1].

7.3: C-120-34-30@123.6K results

This part derives all its data from parts 3.3, and chapter 6. The results in table 7.3 are those of an ideal cycle.

Table: 7.3: Results from the analysis of an ideal C120-34-30@123.6K cycle

	Base case	**Nitrogen cooled**	**LNGR cooled**
Net Work output	**+20.792**	**+62.484**	**+62.484**
Total work out	**+80.3**	**+80.3**	**+80.3**
Total work in	**-59.508**	**-17.816**	**-17.816**
SHTS work input	**-41.692**	**0**	**0**
COP(Electrical)	**1.349**	**4.507**	**4.507**
SNC **(per Kg of N2 circulation)**	**0.18**	**0.394**	**N/A**
LNGR-(OMFR) **per Kg of N2 circulation**	**N/A**	**N/A**	**1.162**
Ambient heat Load	**+69.6**	**+170.14**	**+170.14**
Cooling Load	**-47.85**	**-104.674**	**-104.674**
Net heat	**+21.75**	**+65.466**	**+65.466**
Direct cooling ability ratio (DCAR)	**0.407**		

7.3.1: Discussion on C-120-34-30@123.6K results

This configuration is better than C-120-30-1 but cannot march the ability of C-120-34-30@113.1K. The direct cooling ability (0.407) is the poorest amongst these three and that's why it has a higher SNC than all others in this thesis. The lowest point of the cycle (30bar; 123.6K) is close to the critical point both in terms of temperature and pressure. The final condensate collection and pumping point falls right on the liquid line at 30bar which is a very vulnerable position in terms of phase stability. The poor DCAR makes it apparent that the cycle will have to engage some input from a SHTS in order to be at par with C-120-34-30@113.1K if they are to operate with the same SNC of 0.24.

This configuration cannot be a reliable cycle for producing power from ambient energy. It is vulnerable to phase slip-off because its too close to the critical point of nitrogen and lies on the saturated liquid line at 30bar.

7.4: Important points to note from the discussions in 7.1, 7.2 and 7.3.

The discussion on parts 7.1 to 7.3 concentrated on the individual configurations as described there but the key issues to note are:

- A better cycle will always be the one with the highest ability for direct cooling.
- The lowest point of the cycle should not be too close to the critical point in both temperature and pressure so as to ensure a stable phase status.
- The fluid thermodynamic properties in the liquid accumulator should not be at or very close to the saturated liquid line. A position safely far enough from the saturated liquid line is preferable to avoid the possibility of vulnerable to phase change.

7.5: Further general discussions and results interpretation

The operation of the CHEC power cycles analysed in this thesis will not ideally operate with external cooling from liquid nitrogen or LNGR. A balance between utilising the SHTS and external cooling will have to be the norm because the amount if liquid cryogen may not be available in the same quantities all the time. Engaging the SHTS will therefore be used in regulating the demand for external cooling from liquid nitrogen and LNGR as a counter measure to the variability in the availability of cryogenic liquid. There be less cryogen regulation on the part of an LNGR cooled cycle due to the nature of its availability, but liquid nitrogen has because it is produced from electricity from variable sources.

This part will look into the possible combinations of engaging combined operations of running a mix of external cooling and SHTS in addressing the cooling needs of the GTOS in C-120-34-30@113.1K. The aim is to gauge and define the relationship that exist between the different measured performance indicators such as SNC, COP, Cycle work output and SHTS work input.

7.5.1: Operation of combined SHTS with nitrogen/LNGR cooling on C-120-34-3@113.1K cycle.

This cycle configuration has a minimum external cooling requirement of 30.128KJ/Kg [Table 3.2.1.a]. This marks the end limit at which the SHTS as configured in part 3.2 can assist the cycle in meeting its cooling demand. The cooling demand with no SHTS in operation is 60.758KJ/Kg [Table 5.1.a] this puts the cold input from the SHTS at 30.63KJ/Kg since that is he maximum the SHTS can avail. The enthalpy drop down from the SHTS as graphically shown on Figure 3.2.1.c is 30.63KJ with a total SHTS work input of 27.505KJ.

In its base case state when the SHTS are fully operational, C-120-34-30@113.1K has a COP of 1.923 but while fully on external cooling the COP increases to 5.631. At its maximum COP the SNC is 0.24 while at the lowest COP the SNC 0.12. The sections that follow are a statistical definition of the relationships between the performance parameters and how they affect the final work output of the cycle or the liquid cryogen demand.

7.5.1.1: Relationship between liquid cryogen demand with the work output and SHTS work input (Table 7.5.1.1 and Figure 7.5.1.1)

The maximum amount for nitrogen consumption relates to the cycle's work output through a linear relationship described by the equation Y= [212.1]3x + 13.13 where Y is the work output; x is the SNC mass consumption.

The equation can thus be rewritten as;

$$C-120-34-30@113.1K\ Work\ output = 212.13(SNC) + 13.13 \ldots\ldots(KJ/Kg)$$

Similarly, for SHTS, work input can be estimated by the $SHTS\ work\ input = -229.21(SNC) + 55.01 \ldots\ldots(KJ/Kg)$

Table 7.5.1.1 and Figure 7.5.1.1 give a visual description of this relationship.

Table: 7.5.1.1: Values of C-120-34-30@113.1K [SNC] nitrogen Mass flow, work output and SHTS work input at its base case and while fully on nitrogen/LNGR cooling.

	[SNC]Liquid Nitrogen Mass flow(Kg)	C-120-34-30@113.1K Work output (KJ/Kg)	SHTS Work input (KJ)
Base case	0.12	38.585	27.505
Complete Nitrogen/LNGR cooled	0.24	64.04	0

The values are carefully plotted on a graph and the linear relationship is defined by an x, y relationship.

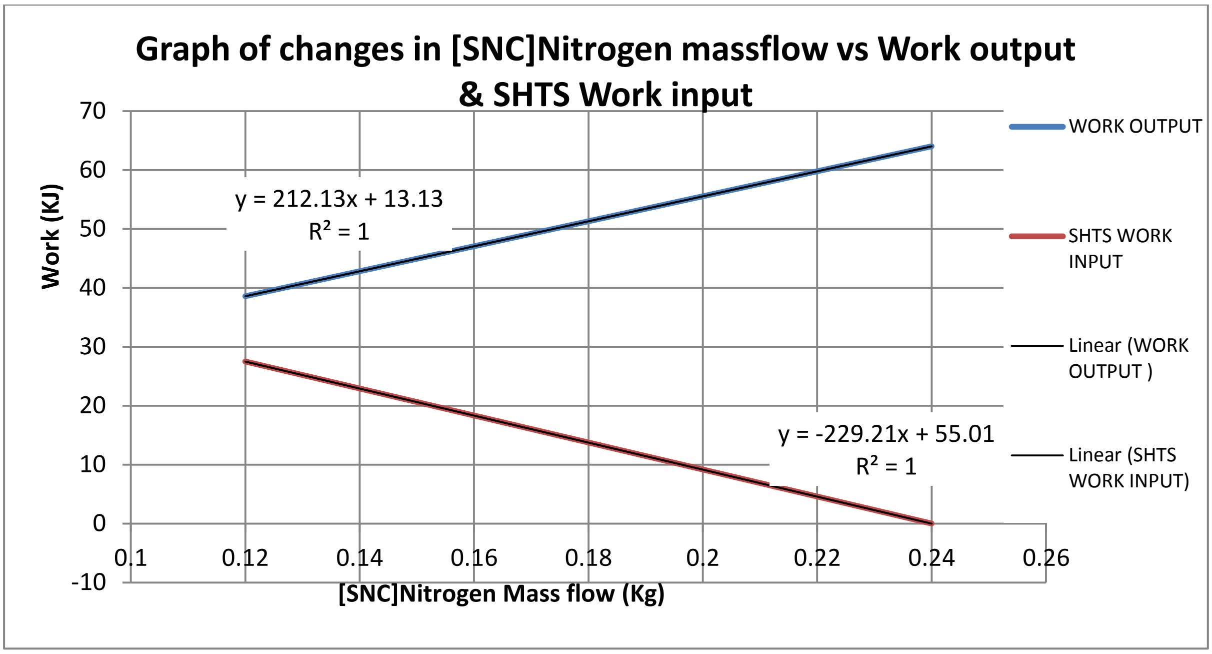


Figure: 7.5.1.1: Graph of changes in [SNC] nitrogen mass flow against C-120-34-30@113.1K Work output & SHTS Work input

7.5.1.2: Relationship between COP with the work output and SHTS work input (Table 7.5.1.2 and Figure 7.5.1.2)

The COP of C-120-34-30@113.1K being a key performance indicator has the following relationship to work output and the SHTS work input.

$$C-120-34-30@113.1K\ Work\ output = 6.6489(COP) + 25.384 \ldots\ldots(KJ/Kg)$$

Similarly, for SHTS, work input can be estimated by the

$$SHTS\ work\ input = 7.4177(COP) + 41.769 \ldots\ldots(KJ/Kg)$$

Table: 7.5.1.2: Values of C-120-34-30@113.1K COP, work output and SHTS work input in its base case and while fully on nitrogen/LNGR cooling.

	COP	C-120-34-30@113.1K Work output(KJ/Kg)	SHTS Work input (KJ)
Base case	1.923	38.585	27.505
Complete Nitrogen/LNGR cooled	5.631	64.04	0

The values are carefully plotted on a graph and the linear relationship is defined by an x, y relationship.

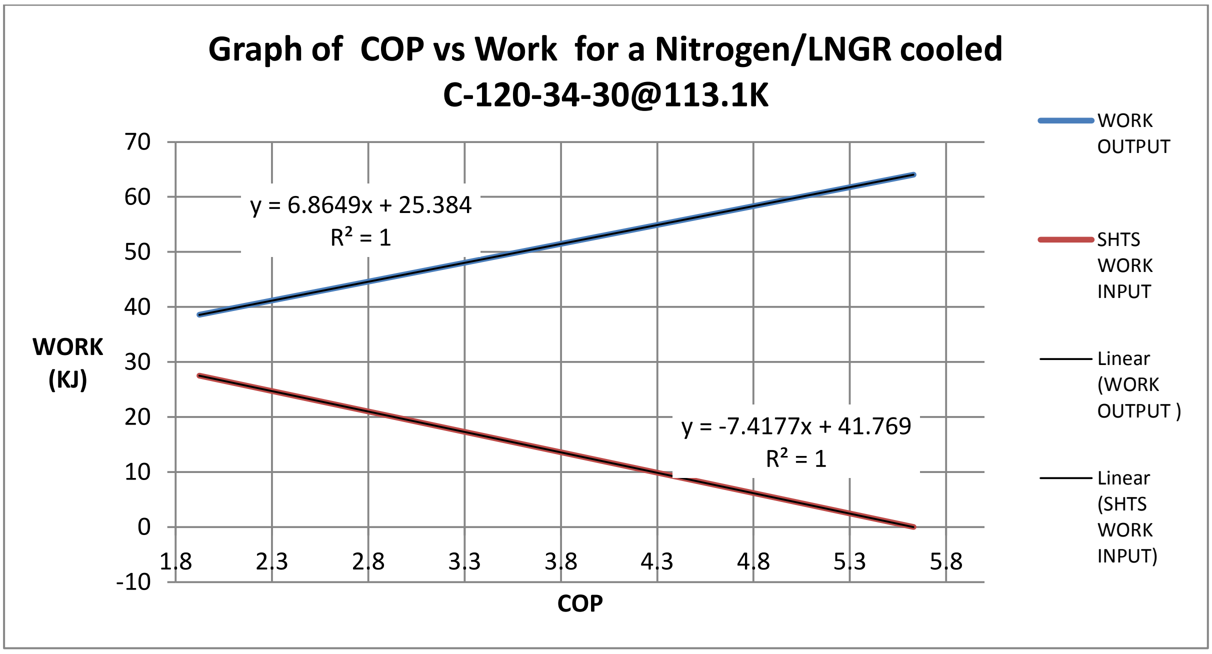


Figure: 7.5.1.2: Graph of changes in C-120-34-30@113.1K COP against Work output & SHTS Work input.

7.5.1.3: Relationship between SNC with the COP of C-120-34-30@113.1K (Table 7.5.1.3 and Figure 7.5.1.3)

This relationship will give the position of plant efficiency brought about by changes in the consumption of liquid cryogen. This will help in determining how to effectively utilise any cold resource.

$$C - 120 - 34 - 30@113.1K(COP) = 30.9(SNC) + 1.785$$

Table: 7.5.1.3: Values of C-120-34-30@113.1K [SNC] nitrogen mass flow and COP, in its base case and while fully on nitrogen/LNGR cooling.

	[SNC]Nitrogen Mass flow(Kg)	COP
Base case	0.12	1.923
Complete Nitrogen/LNGR cooled	0.24	5.631

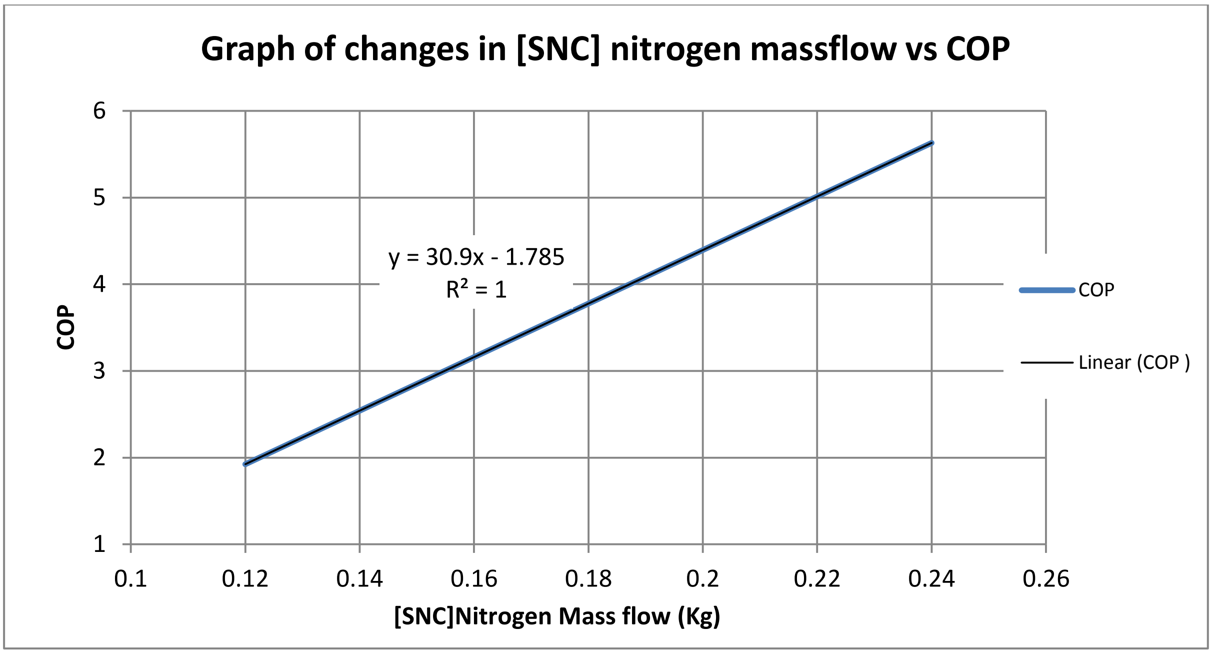


Figure: 7.5.1.3: Graph of changes in C-120-34-30@113.1K SNC against COP

7.6: Operation of CHEC power cycle with an embedded ASU.

The ASU is an essential utility to the CHEC power plant because it is the source of liquid nitrogen. An embedded ASU will have the advantage of operating with a lower energy input because of some easily accessible cooling from the ambient heating zone of a CHEC cycle. This kind of arrangement would be mutually beneficial to both the CHEC cycle because of reduction in ambient heat load and the ASU cooling needs. Further beyond that, the heat transfer between the two streams will be much efficient than ambient heating for the CHEC power cycle. The effects of moisture frost (air sourced heat) and algae build-up heating tubes (water sourced heat) will have been evaded.

It is expected that any CHEC power plant will operate in a multiple combination of modes because of grid demand/supply fluctuations. This is described in detail in chapter 8. In this case the ASU will be central to the operation of any power plant of this nature by allowing the plant to utilize ambient heat in generating electricity and storing excess power by running on maximum ASU mode. The maximum ASU mode allows the build–up of cryogenic liquid reserve during of-peak hours while minimum/zero mode will be set for sending power to the grid during peak demand. The ASU is thus a beneficial unit for any CHEC power plant operations.

7.7: Answers to the research questions.

7.7.1: The questions (recap)

In part 1.3.1 of this thesis, three research questions needed to be answered. The questions were:

1. Can a nitrogen power cycle be used to effectively harness power from ambient heat for electricity generation?
2. Can a nitrogen power cycle serve any needs for energy storage in today's society?
3. Can a nitrogen power cycle be used in utilising LNG regasification cold for power generation?
4. Can the CHEC power cycle be used for combined cooling and power operations?

7.7.2: Answers to the questions

7.7.2.1: Can a nitrogen power cycle be used to effectively harness power from ambient heat for electricity generation?

The answer is yes, provided the cycle has a configuration with the features described in this thesis. A power cycle configured to the likes of C-120-34-30@113.1K has all the potential to effectively generate electricity from ambient heat or any low heat source. According to the analysis done in this thesis, ambient energy can potentially be utilized effectively for power generation.

7.7.2.2: Can a nitrogen power cycle serve any needs for energy storage and grid stability within the existing energy infrastructure?

The answer is yes, if the power cycle is operated with an embedded ASU with a facility to store cryogenic liquid.

7.7.2.3: Can a nitrogen power cycle be used in utilizing LNGR cold for power generation?

The answer is yes, provided there is an LNGR facility. This is the most effective of all provided the cycle is configured to fit in to the heating needs of an LNGR process.

7.7.2.4: Can the CHEC power cycle be used for combined cooling and power operations?

The answer is yes, provided there is a nearby demand for either cooling or power.

CHAPTER 8

PRACTICAL APPLICATION

Chapter contents

Chapter overview

This chapter has taken the results of chapter seven a step further in defining the possible options of how the CHEC power cycle can become part and parcel of the electrical energy infrastructure. Parts 8.2 and 8.3 are the proposed pathway towards practical CHEC power plant operations. In order to be specific to the analyses of earlier chapters, the practicability has been considered on the basis of LNGR and nitrogen cooling provisions. All possible practicable applications seem to point towards the requirement of a combined CHEC, ASU and Cryogen storage facilities as the common denominator of any CHEC power plant operations. The LNGR cooled CHEC facility demonstrates enormous potential for ambient power to play a bigger role in electricity generation despite being site specific. The nitrogen cooled CHEC has a demonstrated potential for enhancing ambient heat energy usage in distributed generation in conjunction with grid stability(spinning reserve). The LNGR facility is therefore seen to be a nitrogen CHEC facility placed within a regasification (LNG) facility. The impacts of CHEC operations seem to have a beneficial advantage of creating some energy amplifications [8.5]. The scenarios are presumptions based of predicted performance of the ideal CHEC power cycle explored in this thesis.

8.1: Practicability of the CHEC power cycle

This chapter will outline ways in which the CHEC power cycle is practically suitable for the envisioned applications of:

- Electricity generation from ambient heat
- Energy storage and grid stabilization
- Utilisation of LNGR cold for power generation
- Combined cooling and power operations

The cycle that shall be deemed as ideal for practical application is C-120-34-30@113.1K. It is obvious to state that the applicability of the LNGR cooled power cycle is site specific and limited to where LNG regasification is being carried out. The nitrogen cooled power plant on the other hand has some limit of flexibility to wherever there is a grid connection and adequate facilities for liquid nitrogen storage. The applicability can therefore be considered in two scenarios, where a regasification facility (cold from LNGR) is available and secondly where liquid nitrogen reserve is available (mainly from off-peak electricity/erratic renewable supply, or nitrogen being supplied from an external source [Ref: 8.2.4]). It is important to note that the liquid nitrogen reserve is an important energy reservoir and may feature in any CHEC power facility because of its ability to store bulk cold which is very useful in anabling a stable operation. This means that the ASU and a nitrogen storage facility will feature prominently even at an LNGR cooled CHEC power plant. Parts 8.2 and 8.3 have specifically described the practicability of a CHEC power facility within the context of LNGR and liquid nitrogen availability respectively.

8.2: CHEC power plant operations based on the availability of a Regasification facility

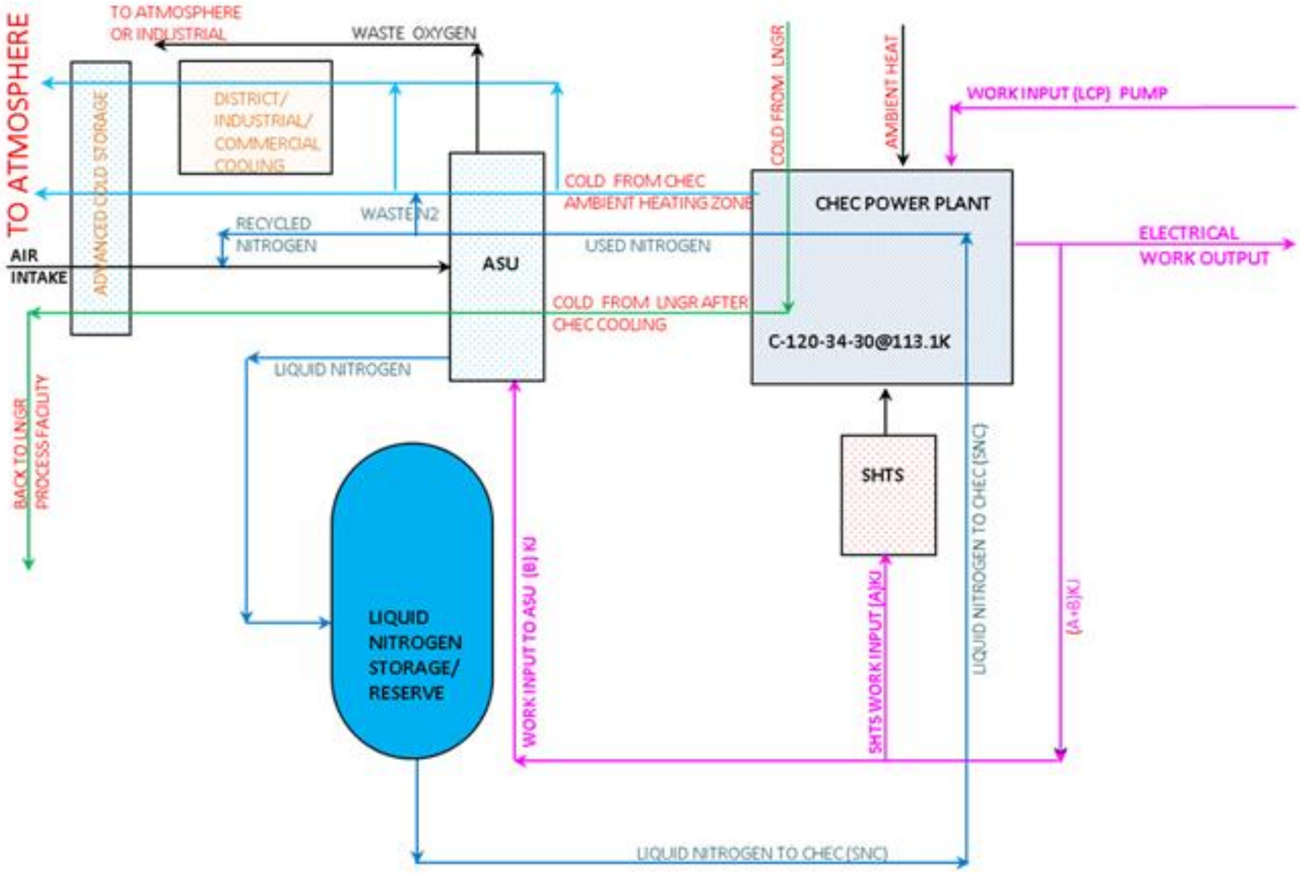


Figure: 8.2: Outline of an integrated CHEC power cycle operations at a regasification facility.

Note:

1. For this facility, external power will be required for start-up since the unit will not have any means of running the pump before it gets to the point of producing power.
2. The cold from LNG will be supplied as a pass through from the regasification facility to the CHEC main heat exchanger and back to the regasification facility having gained some heat by losing its cold to the GTOS.

The outline in Figure 8.2 is a brief overview of a typical CHEC power cycle in operation at an LNGR facility. The set up system has the ability to;

- Generate electricity from ambient heat by LNGR cooling (LNGR mode.)

- Store energy in the form of liquid nitrogen (Energy storage mode)
- Store cold energy in the advanced cold storage reserve
- Generate electricity from ambient heat by nitrogen cooling (Nitrogen mode)
- Supply outpost CHEC power facilities with liquid nitrogen (If enough levels of nitrogen exist)
- Directly provide for industrial or commercial cooling needs (all modes)

8.2.1: Generate electricity from ambient heat by LNGR cooling (LNGR mode)

The system in this mode will generate electricity by multiplying the electrical input by the COP of the cycle. The electrical output will be diverted in proportions of grid send off and supply for internal consumption for the SHTS, ASU and other necessary demands. This mode allows power to be exported to the grid or be used to store energy or both in proportions as per demand.

8.2.2: Storing energy in the form of liquid nitrogen (Energy storage mode)

This mode allows for the build-up of nitrogen levels in stock for peak time electricity demand or supply of liquid nitrogen to any outpost generation points. The operation of the ASU will be very economical because electricity will have been sourced from ambient heat and cold needs will mainly be sourced from the ambient heating zones of both the CHEC and LNGR streams.

8.2.3: Storing cold energy in the advanced cold storage reserve

The system has a lot of cold which would otherwise be released to the ambient in the course of its operations, this is reserved in the advanced cold reserve to be used for ASU operations in producing more nitrogen efficiently. The capture of cold has the ability to reduce the energy required in nitrogen production and thus an increase in the overall efficiency of the cycle (Ref: Highview Power Storage [3]).

The cold stored may find other uses such as supplementing for industrial or commercial cooling needs for nearby customers.

As described in [9.3], the cold store may be used for the provision of cooling for a higher temperature CHEC cycle operation in generating more electricity from ambient heat or waste heat. Higher temperature CHEC cycles operating with fluids which condense at higher temperatures such as ammonia can chip in to utilise this cold as their GTOS cooling source [9.3] thus increasing the amount and range of power production from for the overall system. This will increase the overall efficiency and allow for effective utilization of waste heat and other low heat sources for electricity production.

8.2.4: Generate electricity from ambient heat by nitrogen cooling (Nitrogen mode)

This mode will help to boost up generation at peak times when demand outstrips the capacity of the LNGR stream cooling ability. At this time, more CHEC units can be brought online to operate in nitrogen mode so as to match up the peak demand. This mode will come to play when demand of electricity has increased and thus the need to boost up supply. This will see the see the volume of nitrogen fall while power output increases. The increase in power output will bring about less recycling and more cold storage in the advanced cold reserve. There will be less recycling of nitrogen which means much of it will be vented to atmosphere after the cold store. The ASU will go to zero or minimum operation in order to allow more power to be exported to the grid.

This mode will also come into play when the amount of LNGR flow has reduced for whatever reason, so as to compensate the LNGR cold lost and to maintain a steady power output.

8.2.5: Supply to outpost CHEC power facilities with liquid nitrogen (If enough levels of nitrogen exist)

The build-up of a liquid nitrogen reserve is also meant to supplement the need to supply emergency power particularly to data centres and industrial facilities which may have both power and cold demands. The deliveries will normally be done by trucks as the usual means of bulk nitrogen transport just as in the case of Highview Power Storage (Ref: Highview power storage [4]).

8.2.6: Directly provide for industrial or commercial cooling and power needs (all modes)

This will be site specific and will depend on whether industrial or commercial cooling demand exists within the vicinity. In case of no cold demand nearby, the option is to store it in the advanced cold reservoir before releasing venting to the ambient atmosphere.

8.3: CHEC power plant operations based on the availability of a liquid nitrogen facility

The CHEC power cycle may operate with nitrogen cooling for two reasons depending on how liquid nitrogen is availed. Liquid nitrogen may be produced on site or supplied from an outside source such as from a CHEC power plant at a LNGR facility. The reasons as to why a power plant may operate with nitrogen cooling are:

- Energy storage/grid stabilization
- Electricity and cold generation

It is of utmost importance to note that the plant will be able to perform both functions if it operates for the purposes of energy storage and grid stabilization. If used for cold and power generation, the functions may not be flexible.

8.3.1: CHEC power plant operations for energy storage and grid stabilization

The CHEC power system for this purpose has a large ASU throughput/nitrogen storage capacity to enable it stock pile nitrogen from off-peak, spinning reserves and intermittent (renewable) sources of electricity. This type of power plant will have to be situated at locations where there is grid connection to allow it serve its functions effectively. The set up can be used as a source of liquid nitrogen supply to outpost locations [8.2.5].

The unit's functions for this purpose are similar to those of a Highview's Cryo Energy system [1.4.7.1.2] with the difference being the cycle design and system configuration.

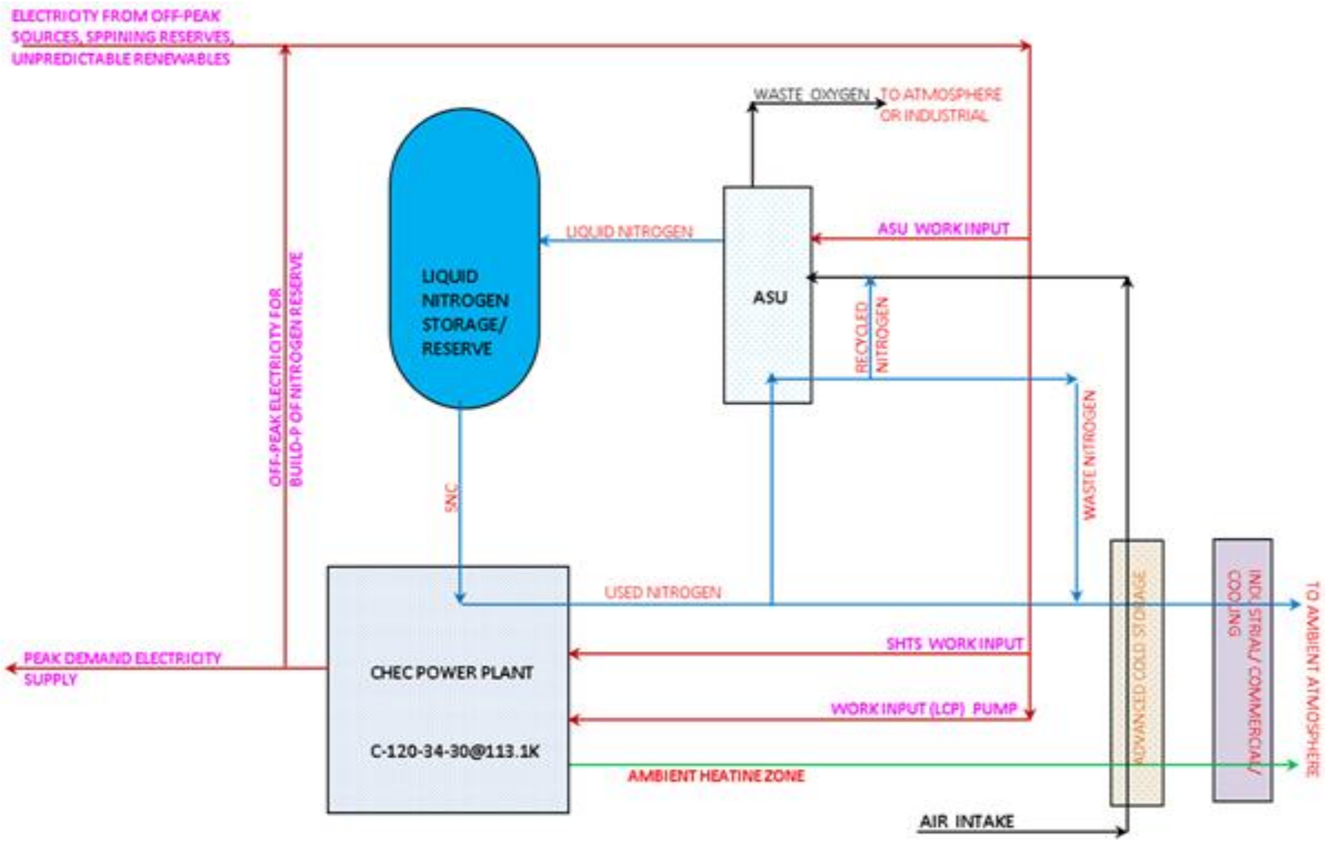


Figure: 8.3.1: Outline of an integrated ASU-CHEC power cycle operating on nitrogen mode [8.2.4]

8.3.2: CHECK power plant operations for electricity and cold generation

This set up is a design to cater for small outposts with power and cold needs such as data centres or cold stores. The unit's functions for this purpose are similar to those of a Highview's Cryo Gen Set [1.4.7.1.3] with the difference being the cycle design and configurations.

The consumption of nitrogen will be based of the cycle SNC specification. The ability to regulate power and cold by controlling the nitrogen mass consumption gives the CHEC design an upper hand than other designs. This will allow effective scheduling of liquid nitrogen deliveries.

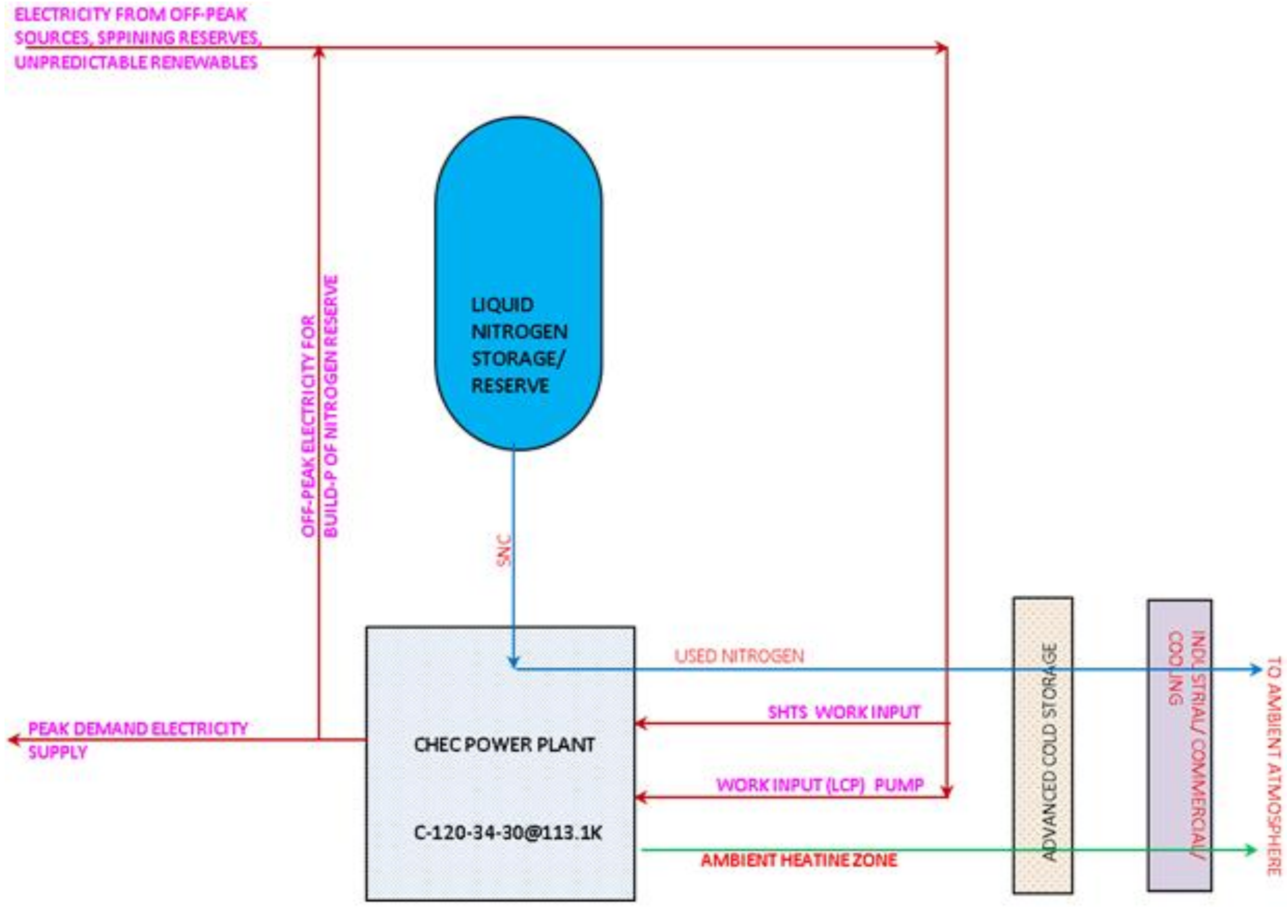


Figure: 8.3.2: Outline of a CHEC power cycle operating on nitrogen mode at an outpost location [8.2.6]

8.4: Start-up and operational strategy

8.4.1: Start-up strategy

The operation of any power plant is designed to co-exist with the available electrical supply/distribution infrastructure and the patterns of power demand and supply. It is therefore important to mention how the CHEC power plant operational circumstances may effectively be fitted in to existing norms of other power generation facilities so as to fulfil its intended role of power generation and grid stabilization. A CHEC power plant requires a source of power for start-up operations. This could be either grid supply or any portable source of power in case of remote or isolated locations. The amount of power needed should be enough for all initial operations including the running of auxiliary units and all associated control functions. Portable power could be supplied by emergency diesel generators, small gas turbines or battery reservoirs.

The start-up strategy is based on the fact that a CHEC power facility will operate with several units in parallel in order to have flexibility in operations on electricity generation capacity. The start-up power demands for a smaller unit are much less and therefore more manageable and easily controllable.

The proper way to start is to run a single unit with external power until it's fully operational and ready to export power, and then recycle the power generated and use it to start the other units. As more units attain power export status, more power becomes available for starting the other units. Once all internal power requirements are matched, the source of start-up power can be isolated and power export to grid or nitrogen production can be initiated.

8.4.2: Operational strategy

The CHEC power plant is cryogenic in nature unlike other conventional systems, and this puts the strain of ensuring that the plant is cooled down to operational temperatures in the cold sections before any start-up. This may take some time and the counter measure will be to put all units in operational readiness by running all the units at lower loads instead of a single unit at full capacity during low demand sessions. If a single unit is run at full capacity to provide power during off-peak periods, the other units will have to be maintained in cold readiness condition which will increase the use of the cryogen. In order be economical in cryogen utilisation, the low peak power demand can be supplied by several or all units running at reduced output. This keeps all the units in a state of being able to chip in more power by rising there output instantly without having to undergo the cool down start –up procedure. This will enable CHEC power plant operations be able to increase power promptly and thus stabilize the grid.

A stable operation will need the power plant to be versatile in grid surges due to demand and supply. It should be able to raise or reduce its production within reasonable time in order to be useful in grid power stabilisation. In order to be able to fit in to this role, the individual power units can be operated at a lower output load capacity of 25% so as to keep the balance of 75% for variable production at any time of need.(the stated capacities are assumptions).

8.4.3: Example of a CHEC power plant start-up operation on James Weir Power Company (JWPC)

Let's assume a power company by the name of James Weir operates six CHEC units at its generation facility. The units are arrangement similar to Figure 8.4.3, and have a total generation capacity of 6000KW. Each unit has the ability to generate 1000KW at full capacity and all have a minimum operational condition of 200KW with a COP of 4 (assuming the COP remains constant at all production levels). This means that a unit cannot operate below the stated minimum capacity. The whole power plant facility consumes 500KW of power for other internal uses. The units have the acronym PU with a number for identification purposes. PU1 stands for Power Unit 1 and the rest are named in a similar manner up to the last unit PU6.

Assuming that the power facility has to generate 1450KW during low peak demand, discuss how the plant will be started-up and maintained in operation to provide for this demand and be able to increase its power to 4000KW at short notice?

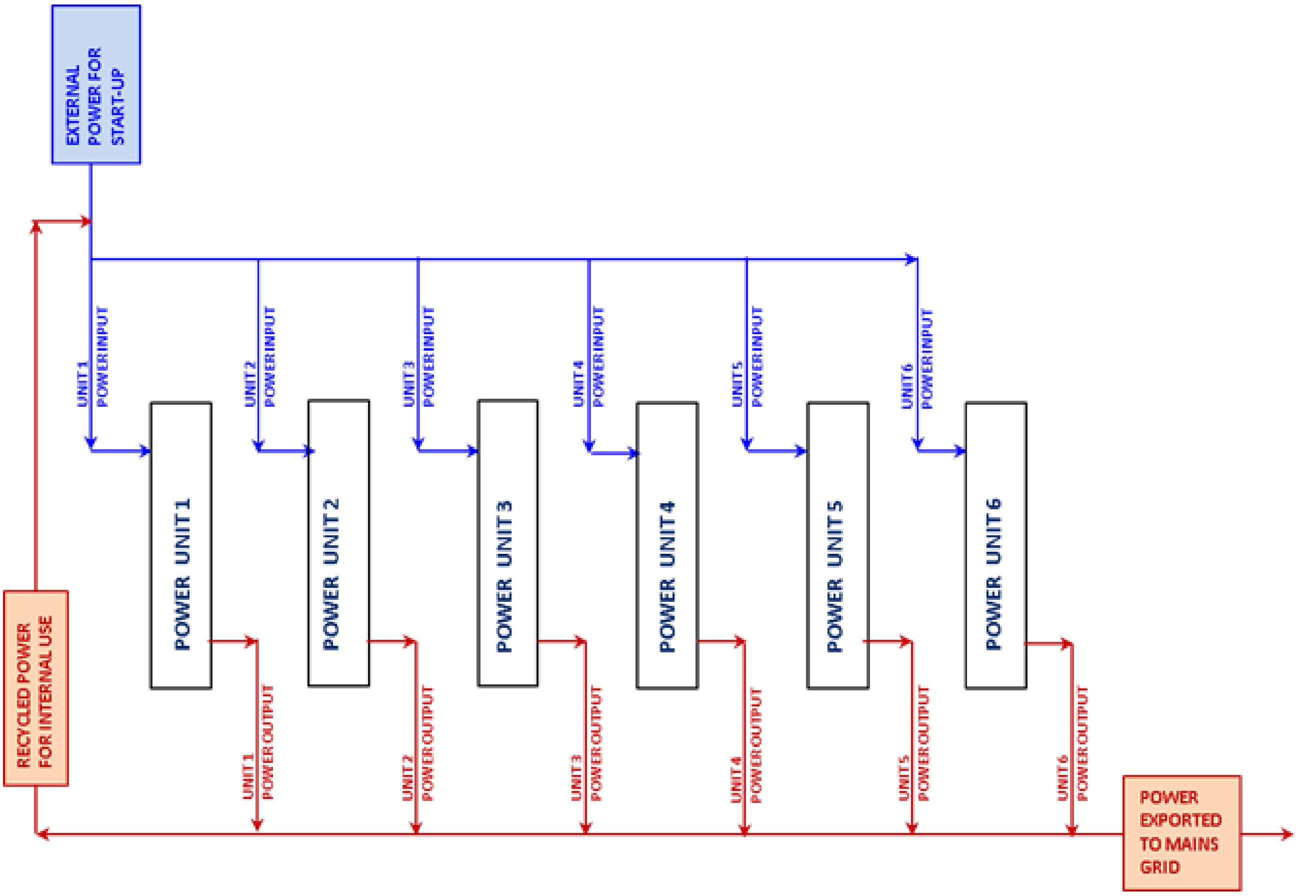


Figure: 8.4.3: An arrangement of CHEC units at the presumed JWPC

8.4.3.1: Start-up of JW power plant

The start-up operation will take place in the following steps;

1. Use the nitrogen reserve(from storage) to cool down PU1 until its ready for start-up.
2. Import power (550KW) from the grid or any other portable source to start PU1. Note that 500KW is required for other internal power needs while 50KW is for start-up of PU1.
3. Run PU1 with an input of 50KW to operate it at minimum output capacity of 200KW.
4. Recycle PU1 output (200KW) for start up of the next three units (PU2, PU3 and PU4) and run them at minimum capacity as well. Each unit will consume 50KW and produce an output of 200KW. Four units in operation will generate a total of 800KW while consuming a total of 200KW.

5. Stabilize operations of the four units at this level and gradually reduce power input until a level of self sustained supply is attained, then switch the external power into emergency cut in mode. Note when imported power is on emergency cut in mode, it remains as a standby as a source of power but won’t be consumed by the operation.
6. With four units giving 800KW out which is recycled to provide for internal needs of 500KW and 200KW input to the units an extra 100KW shall be available for start-up of the two remaining units.
7. Use the 100KW to start-up PU5 and push up its output to 400KW (COP of 4). At this juncture, the system will have a total output of 1200KW with a total consumption of 800KW.
8. Start-up PU6 and put it at maximum capacity of 1000KW. The power facility will now have all its units in operation producing a total output of 2200KW at a consumption of 1050KW giving a net output of 1050KW.
9. Recycle more power (100KW) to PU5 and with a new input of 200KW its output to increases to 800KW. The final standing will be a total output of 2600KW at a consumption of 1150KW. The final status of JWPC is shown in Table 8.5.3.1.

Table: 8.4.3.1: JWPC stable power generation at 1450KW power export to grid during low peak demand.

Power Unit	Power input (KW)	Power output (KW)	Net Power output exported to Grid (KW)
Internal use	500	N/A	
PU1	50	200	
PU2	50	200	
PU3	50	200	
PU4	50	200	
PU5	200	800	
PU6	250	1000	
Total	1150	2600	1450

8.4.3.2: Normal operation of JWPC power plant

The demand of 1450KW at low peak is well serviced as shown in table 8.5.3.1 while a spare capacity of 2550KW exists to push up generation to 4000KW in case of need. Table 8.5.3.2 shows the status of JWPC at maximum capacity. At this point, it is important to note that the plant may go to maximum capacity generation even during low peak demand periods so as to build up its nitrogen reserves [8.3.2 & 8.4.1]. The spare capacity is multiplied by the COP due to intake of ambient energy and then used to produce more nitrogen for peak time generation. During normal operations, JWPC will be trading off between the grid demand and spare capacity in relation to the cryogen storage capability in order to optimize operations and fulfil obligations associated with electricity generation, energy storage and grid stability.

Table: 8.4.3.2: JWPC operating at maximum generation

Power Unit	Power input (KW)	Power output (KW)	Net Power output exported to Grid (KW)
Internal use	500	N/A	
PU1	250	1000	
PU2	250	1000	
PU3	250	1000	
PU4	250	1000	
PU5	250	1000	
PU6	250	1000	
Total	2000	6000	4000

8.5: CHEC Energy multiplier effect on an existing power facility

This part is a build-up of likely scenarios which may be brought in by bringing in CHEC power plant operations to current electrical power infrastructure. The scenarios are based on assumptions which are envisioned for the ideal power cycle investigated in this thesis. Parts 8.5.1 and 8.5.2 are exploring the likely outcome of CHEC power plant operations during off-peak and peak periods.

8.5.1: The Energy multiplier effect from CHEC power plant operations during off-peak periods

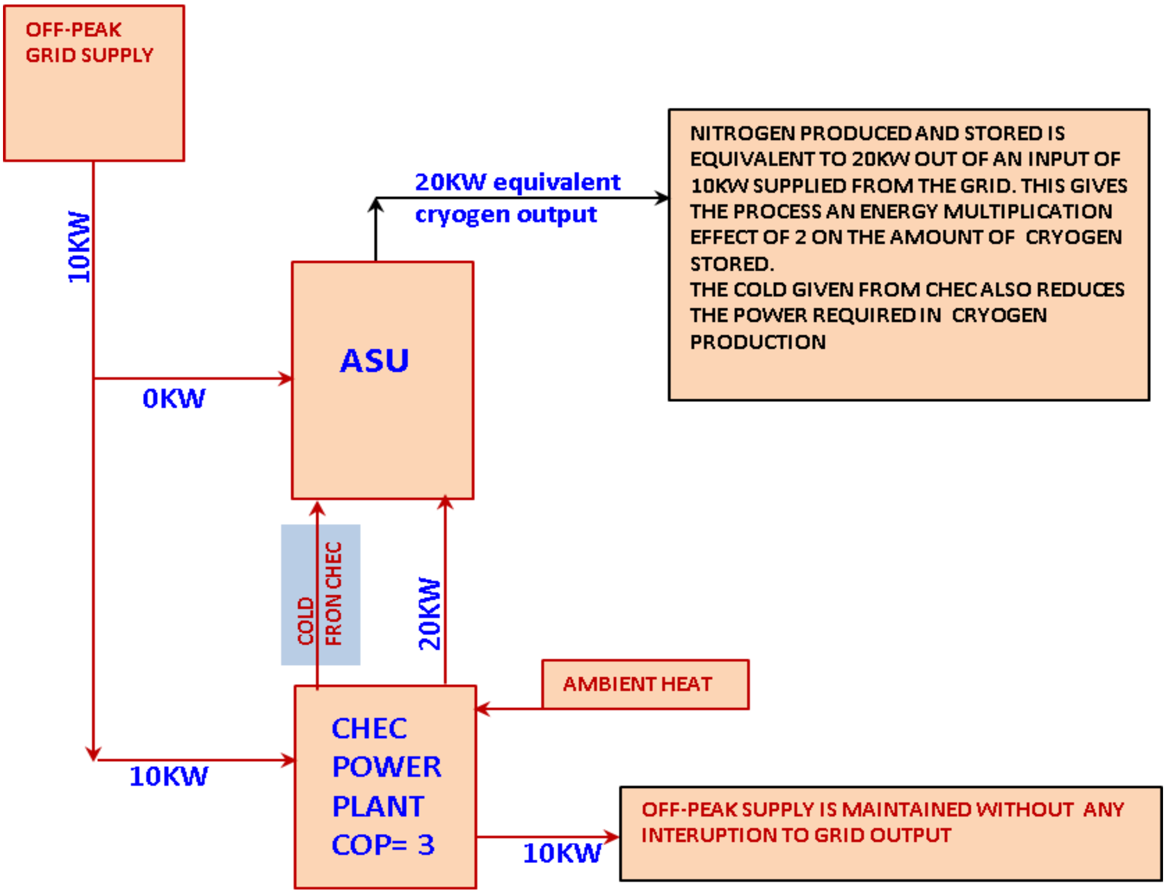


Figure: 8.5.1: CHEC Energy multiplier effect during off-peak periods

The operation of a CHEC power cycle has an energy multiplier effect if used for the production of cryogen for the storage of off peak electricity. This is captured in Figure 8.5.1 where the electricity flow pattern and transformation into cryogen is captured. The input of 10KW if used directly in cryogen production would have produced 10KW equivalent of cryogen. The circumstances are completely different if the electricity is channelled via the CHEC system. The example of Figure 8.5.1 shows a CHEC power plant with a COP of 3 thus helping the unit increase the electrical power being fed to the ASU by a factor of two. The CHEC unit has therefore increased the amount of power to the ASU and thus the increased cryogen production at the expense of ambient energy intake while maintain the generation units in operational readiness at minimum capacity. The overall throughput of the ASU is improved by a factor of 2 as shown in Figure 8.5.1. This demonstrates that as a whole, the system can act to improve on power yields by operating strategically based on patterns of off-peak demand and supply to give a multiplying effect on the input power.

8.5.2: The Energy multiplier effect from CHEC power plant operations during peak periods

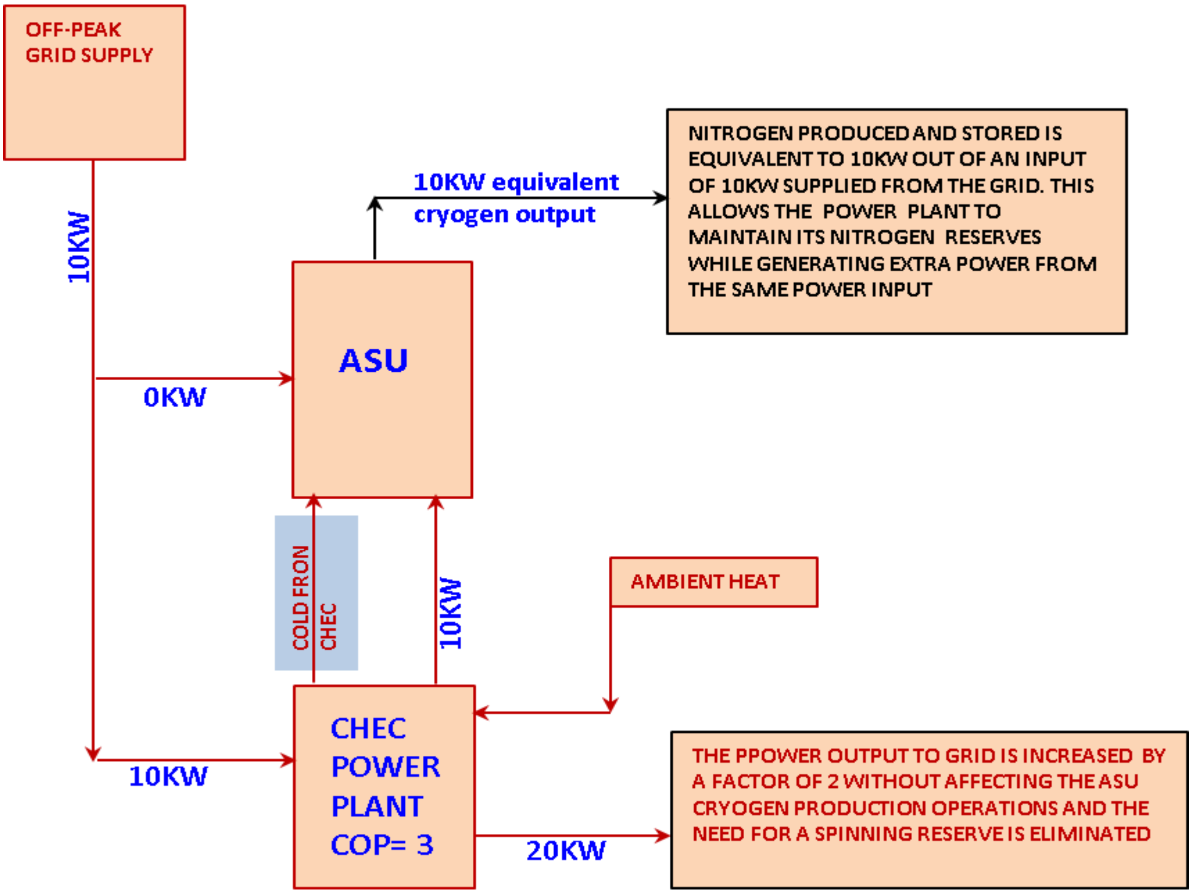


Figure: 8.5.2: CHEC Energy multiplier effect during peak periods

At times of peak electricity demand, the amount of electrical input is increased without effecting the operations of cryogen production from the ASU. Figure 8.5.2 is self-explanatory in showing the power multiplication effect. This demonstrates how ambient energy can be infused into the energy system effectively and thereby becoming part and parcel of the existing energy infrastructure without the need to over build generation capacity in order to have spinning reserves.

CHAPTER 9

FUTURE WORK

Chapter Contents

Chapter overview

This chapter briefly points to areas where the CHEC power cycle can be examined further to establish how it may be evaluated, investigated, improved, criticised or appraised. The future work mentioned in here concerns the CHEC power cycle and all its associated systems and processes. The following list is a summary of the targeted future work topics;

- SHTS based on fluids other than nitrogen
- SHTS optimization
- Higher temperature CHEC cycles
- Effects of wind and humidity variations on CHEC power cycle performance
- The actual plant

- Financials
- Environmental impact assessment
- Carbon foot printing
- Power and cold from high gas pressure let down stations
- Establishing the cold potential of regasification facilities

9.1: SHTS based on fluids other than nitrogen

Analysis of all SHTS based on different fluids other than nitrogen need to be carried out in order to establish their suitability in performing the same functions better. Fluids such as helium and hydrogen may have the potential to be effective in doing that because they have a much lower boiling point.

9.2: SHTS optimization

The compression and expansion processes have positions at which power input and output can the minimized or maximized. Study on the identification of optimal power positions on the T-S diagram to map out regions of high performance for different pressures and temperatures are essential.

9.3: Higher temperature CHEC cycles

Substances which have higher condensation temperatures such as ammonia, CO2, and others, may be studied to check their suitability of operating in a cycle similar to CHEC power cycle. There is potential of using such fluids to operate on waste heat or any low heat sources by utilizing the ambient heating zone as their GTOS coolant. Such systems will be mutually beneficial to the CHEC cycle by helping to reduce the ambient heating load thus improving the overall efficiency.

9.4: Effects of wind and humidity variations on CHEC power cycle performance

The effects of wind and humidity on CHEC power plant performance need to be investigated. Although this will be site specific, the brief knowledge of general weather influence on plant performance remains very important in ascertaining the performance of ambient heaters.

9.5: The actual power plant

The actual plant needs to be modelled and analysed based actual performance of all the equipment's involved. The effect of temperature losses and effectiveness of heat transfer equipment need to be evaluated and analysed thoroughly. The performance criteria as specified by manufactures of available equipment can be modelled to different configurations of the CHEC power cycles in order to establish their expected actual performance.

9.6: Financial evaluation

Getting on the capital costs of plant and operational cost of running the plant is a key area for economic evaluation on the viability of the CHEC power cycle to become a product in the energy market. Financials will also need to be evaluated in the context of the need to compare the economic performance the CHEC power cycle with that of other modes of power generation currently in existence. Normal business performance indicators such as ROE (return on equity), ROOCE,(return on owners capital employed), Net profit margin and many others will set the pace for a thorough business evaluation.

9.7: Environmental impact assessment (EIA)

The whole system needs a proper EIA evaluation to establish its impact on the environment in which it shall operate. Effects on land, air and water have to be put on the rudder to check how this power plant operation will impact on them. Other important areas such as impact on livelihoods and biodiversity fall under this category. The CHEC power cycle needs a proper EIA to be done on it based on established standards.

9.8: Carbon foot printing

The carbon footprint based of everything consumed in bringing the plant to existence need to be evaluated to establish the amount of embedded carbon. It is important that the ability of the power plant to recoup its carbon foot print during its operations is evaluated. The plant is to be evaluated further on the ability to create carbon saving through its operations of utilising excess electricity and using it to reduce the amount of fossil fuels which could have been used to generate peak demand power. The investigations on carbon foot printing are to be based on the following:

- Carbon in the purchased equipment’s
- Carbon foot print of plant construction
- Carbon foot print from power plant operations

9.9: Power and cold from high gas pressure let down stations

The fact that gas pressure let down stations create a lot of cold which may necessitate heating, it is worth to investigate and to know whether the flow pattern can allow CHEC power plant operations to be carried out effectively.

9.10: Establishing the cold potential of regasification facilities

It is important to note from this thesis that cold has power. Studies need to be carried out to establish the cold potential and cold reliability of each regasification facility in order to understand how an LNGR cooled configuration may be optimized.

9.11: Examining the possibility of providing a huge sink for nuclear power plants

There may be a possibility to operate near nuclear plants so as to act as sink for the excessive heat which the plants have to reject. This is important as it may enable the development of nuclear plant in places where the usual huge water sinks do not exist.

CHAPTER 10

CONCLUSION

The world faces many challenges because resources a dwindling and the population is increasing. Energy is key to all aspects of life in every part of the world. The current energy resources are based on fossil fuels which are finite and environmentally polluting. The current energy infrastructure struggles to cope with the demand and supply patterns resulting into more wastage and pollution. Lack of reliable bulk energy storage systems is an impediment towards grid stabilization and utilization of intermittent sources of power. As fossil fuels become scarce with each passing day, the use of renewable energy is set to increase. This may lead to a much unstable grid as renewable are intermittent in nature and thus cannot be relied upon as the salvage of this quagmire. Energy has become a political hot spot and many governments take steps towards having strategic reserve in order to have energy security.

It is with all these in mind that this thesis targeted to work towards proposing a viable solution which could help in addressing many of the above mentioned issues. This thesis had set to come up with ways and means which would help in bringing ambient energy in to the light as a reliable, dependable and renewable source of power. Nitrogen was the fluid of choice because it can be sourced freely from the atmosphere and has the ability to operate at lower (cryogenic) temperatures. Three ideal nitrogen cycles (closed systems) of different configurations were analysed for this purpose and the results indicated that a particular set of consideration need to be in place in for the cycle to work [7.4]. One of the cycles which had closely matched that criterion of design and configuration has shown that ambient heat can indeed be part of our energy resource with a lot of gains towards addressing the issues mentioned earlier. This study settled on a cycle configuration of C-120-34-30@113.1K because of its outstanding performance in providing answers not only to the research questions but also to the many problems facing our energy infrastructure and future. The use of LNG as a clean source of power is set increase and within it lay a huge reserve of cold which is usually given off to the ambient but mostly done with by gas firing to remove the cold. This thesis has found out that a CHEC power facility co-existing with an LNGR facility can make a huge impact towards ambient heat electricity generation in a manner which can be beneficial to both. The potentiality of the CHEC power cycle towards grid stabilization and energy storage was demonstrated in chapter 8 as very promising. A reserve of cryogen from off-peak/intermittent electricity can serve to help in amplifying the power during peak hours with the outcome of low investment in extra generation facilities for spinning reserves. A CHEC power plant operated strategically has the ability to increase or reduce its power output within a short period thus injecting the needed stabilizing power potential to the grid.

It is apparent that the world of energy is in a dilemma and it will take innovation beyond invention to sort out the current problems. All is not lost because ambient heat is a resource which in the end, we may not ignore. This thesis has clearly exposed the path towards that direction of saying we can explore ambient heat to benefit the society economically, socially, politically and environmentally. Therefore, let further steps be taken towards ambient energy power generation so as to sort out some of the problems and reduce the carbon foot print of our energy infrastructure.

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Appendices:

Appendix A: Composition of air (Courtesy of UIGI http://www.uigi.com/air.html Ref: 4)

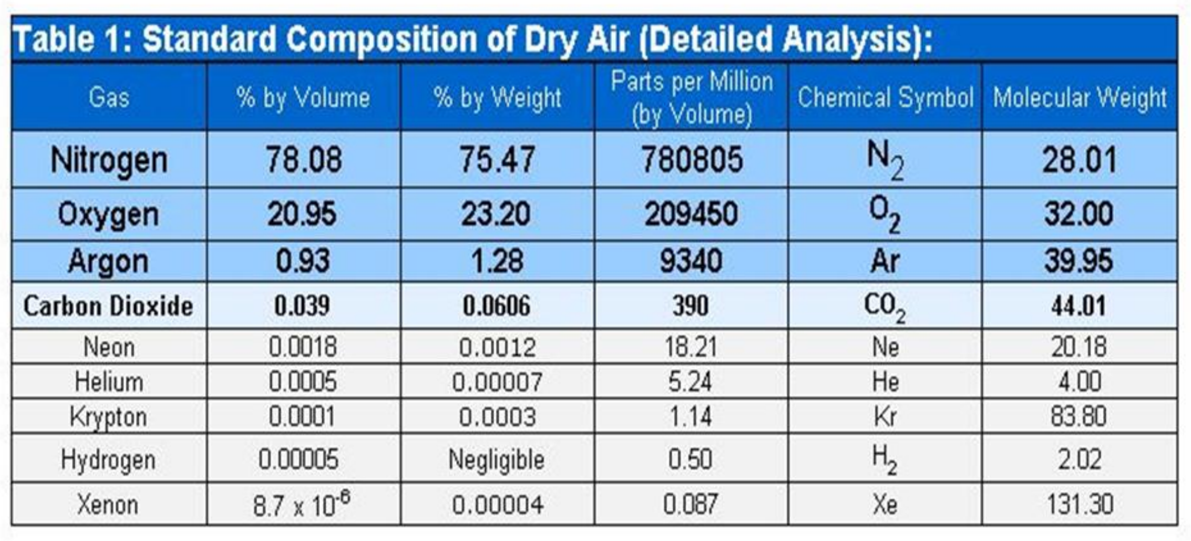
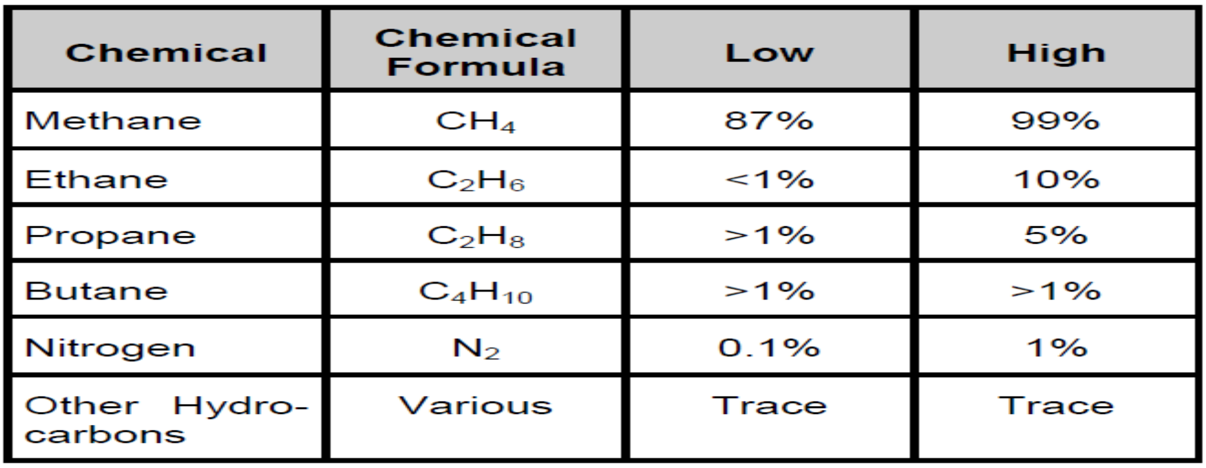


Table 1: Standard Composition of Dry Air (Detailed Analysis):

Gas	% by Volume	% by Weight	Parts per Million (by Volume)	Chemical Symbol	Molecular Weight
Nitrogen	78.08	75.47	780805	N_2	28.01
Oxygen	20.95	23.20	209450	O_2	32.00
Argon	0.93	1.28	9340	Ar	39.95
Carbon Dioxide	0.039	0.0606	390	CO_2	44.01
Neon	0.0018	0.0012	18.21	Ne	20.18
Helium	0.0005	0.00007	5.24	He	4.00
Krypton	0.0001	0.0003	1.14	Kr	83.80
Hydrogen	0.00005	Negligible	0.50	H_2	2.02
Xenon	8.7×10^{-6}	0.00004	0.087	Xe	131.30

Appendix B: Typical chemical composition of LNG (Courtesy of: Centre for Energy Economics; Ref 5)



Chemical	Chemical Formula	Low	High
Methane	CH_4	87%	99%
Ethane	C_2H_6	<1%	10%
Propane	C_2H_8	>1%	5%
Butane	C_4H_{10}	>1%	>1%
Nitrogen	N_2	0.1%	1%
Other Hydro-carbons	Various	Trace	Trace

Appendix C: Details of Multistage compression (1 to 15bar) with intercooling.