Liquefied Natural Gas (LNG) - Thermodynamics and Liquefication Systems Part 3B

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A Safety Moment

This learning document is meant to be at a technical level, mostly on refrigeration systems. The most important intention of every document I produce is to give you the basic technical knowledge you need to start your study on how to continue making the Liquid Natural Gas Industry a "Safe and Reliable Industry." You need technical knowledge to do that.

The image to the right is placed here as a reminder that everyone has someone who loves and needs them to come home at the end of their shift.

Reliability is also related to safety because if the LNG plant cannot make LNG into vapor when needed, the consuming public may be out of gas during the worst of cold weather times. This would put the public at severe health risk.

We, as engineers, need to ensure that our designing, planning, operating, and maintenance of LNG facilities help assure safety and reliability.

This will help ensure that everyone comes home at the end of their shift and that gas is supplied when needed by the end-user customers.

*Figure **Error! No text of specified style in document.**.1: Beautiful Granddaughter*

Source: Self-Made photo

1. Introduction

Two hundred years ago, ice was harvested from frozen rivers in the winter, and large chunks of it were stored in buildings insulated by large bales of hay and sawdust. During the spring and summer, that ice was consumed, but those who could afford it, and once it ran out, there was no way to produce the heat extraction needed to make ice. The only source of that cold product was to wait for the winter weather to produce it!

In 1834 Jacob Perkins invented the first vapor compression system for refrigeration, and in 1876, Carl von Lined patented a new process for liquefying gases. Today nearly every car and home are air-conditioned via various technologies; the most common is the vapor-compression–condensation–pressure-drop-evaporation system. This system is used in household refrigerators and massive liquefaction plants worldwide. The massive systems used in LNG export facilities use the same principles as most home refrigeration systems, except that some enhancements are used to make the large-scale systems more energy efficient. Also, the refrigerants needed to produce very cold temperatures differ from those used in home air-conditioning and refrigeration systems.

It is taken for granted that when a cold soda is desired, it is readily available from the refrigeration system in the house. The amazing thing is that the refrigeration systems are so well perfected that typically the only reason they are replaced is that they go out of style and not because they need to be fixed. These systems have been running for many decades with little maintenance. These smaller systems have been so reliable for so many years because, since the late 1920s, these small-scale refrigeration systems have been hermetically sealed. That means the motor and compressor are in a sealed case without the need for shaft seals that could leak and without any possibility of refrigerant contamination.

Over the period of the past 189 years, since Jacob Perkins invented the vapor compression refrigeration system, refrigerant technology has also developed significantly. In the 1800s, refrigerants were extremely toxic, and some were inefficient. These toxic refrigerants included ammonia, methyl chloride, and sulfur dioxide. Refrigeration systems were often installed outside to avoid death from a refrigerant leak.

I had chemical pneumonia for a month due to exposure to anhydrous ammonia from a small refrigeration system. Such exposures have killed many.
In 1928 halogenated hydrocarbons such as chlorofluorocarbons (Trade named Freons) were invented. This revolutionized refrigeration, and various Freon compounds were developed for

various temperature applications. However, in the late 1900s, as it became apparent that Freon was harmful to the environment, legal restrictions were placed on using and reusing Freons. With the phasing out of older traditional Freons, newer refrigerants (hydrofluorocarbons, also trade name Freons) were introduced, such as R-134A for automobile applications and R-410a for home air conditioning applications. Even these refrigerants have associated environmental concerns, and newer refrigerants are under development.

For very low-temperature refrigeration systems (LNG production), Freons cannot be used because they cannot achieve the very low temperatures needed to liquefy natural gas. Instead, mixtures of nitrogen, methane, ethane, propane, and iso-pentane are some of the most common refrigerant mixes used for attaining ~ -260 F. In a large LNG production facility, the downside of these refrigerants is that tons of them are needed in these systems that cannot be hermetically sealed, and all but nitrogen are highly flammable. Any system that contains large quantities of pressurized flammable liquids and vapors poses a risk to the plant and its operators.

There are billions of small-size air conditioning/refrigeration systems in operation today, but only a few hundred behemoth-size systems are used in the LNG liquefaction industry. There are many other gas liquefying industries, but this work will limit its focus to understanding small, simple systems and then learn about the larger systems used for liquefying natural gas. Although the basic technology is the same between the small-size units and the large units, the complexity of the systems and the refrigerants used differs as the desired temperatures become colder and as the capacity of the units becomes larger.

If the outside environment is at 80 F, it takes little energy, and the technology is simple to achieve the 35 F temperature needed to cool down a soda. It takes more energy to store frozen food at 0 F, but the technology is still simple.

However, if the outside environment is 80 F, it will take a significantly large amount of energy and more complex technology to achieve the ~ -260 F temperature needed to make Liquid Natural Gas (LNG). To achieve a temperature of ~ -424 F to liquefy hydrogen, the energy and technology required increases many-fold over that needed to make LNG. To take this to the extreme, liquid helium (the very coldest gas liquefied) can be produced at ~ -452 F and is extremely difficult and power intensive. Keep in mind that the absolute zero temperature is -459.67 F.

Engineers need some understanding of thermodynamics to better respond to anomalies during plant operation. The thermodynamics presented in this publication are basic and based on

application rather than theory. The cases studied are all steady-state (the properties of the fluid at any point do not change with time) and steady-flow (the flow rate does not change with time) type problems. All the solutions are based on some simple calculations and/or on the use of the pressure-enthalpy chart or thermodynamic software. A large-size pressure enthalpy chart for methane or thermodynamic software should accompany this publication.

This section 3B is the last of a 3-part series (with section 3 being made of two parts 3A and 3B) on the thermodynamics of producing LNG. In earlier sections, vapor compression systems and open expansion systems were explained. In this section, nitrogen expansion systems will be explained.

Cautionary Note

This document is intended to teach basic concepts. To accomplish this, a simplified approach is taken to explain thermodynamic processes.

Real plants have pressure drops associated with flows through piping, exchangers, and other processing equipment. Such pressure drops are not considered in the simplified examples given herein. Parts of the plant, such as the CO_2 and water removal systems, were not included in the analysis because their study is outside the scope of this document.

Also, rounded-off numbers are often used throughout to allow the reader to focus on the concept without getting bogged down in numerical detail.

Facilities in the United States and Codes that Govern them

The Federal Energy Regulatory Commission (FERC) governs most permanent LNG facilities in the U.S. via the federal code of regulations (code 49CFR193). This code requires the governed facilities to abide by the National Fire Prevention Association (NFPA) 59A consensus code. Many countries around the world also conform to NFPA 59A.

The U.S. Pipeline and Hazardous Material Association (PHMSA) collects data on LNG facilities annually. According to PHMSA, the inventory of LNG facilities as of 10/1/2022 for the 2021 annual reporting year is as follows:
https://www.phmsa.dot.gov/data-and-statistics/pipeline/gas-distribution-gas-gathering-gas-transmission-hazardous-liquids

- 71 Peak Shaver (PS) LNG Facilities (48 with liquefiers) **(Almost 70 %)**
- 23 Satellite (Sat) LNG Facilities (1 with liquefier – that is how it is reported)
- 26 Base Load LNG (liquefiers - not counted)
- 40 Mobile or temporary LNG facilities
- 8 Other LNG facilities
- Of the 94 Peak Shaver and Satellite LNG facilities, 44 (47%) facilities in the Northeast
- Of the 94 Peak Shaver and Satellite LNG facilities, 72(77%) 1960's – 1970's vintage
- Of the 48 PS and Sat in the Northeast U.S., 40 (83%) 1965 – 1975 vintage
- Of the 48 PS and Sat in the Northeast U.S, 12 (25%) have liquefiers

The Northeast is emphasized because it is the country's area where local distribution companies (LDCs) are heavily dependent on LNG, as many of these facilities receive LNG via tanker truck from the Everett LNG import terminal.

2. Terms and Units of Measure for Natural Gas and LNG

Abbreviations

The following abbreviations, terms and units will be used in this document for natural gas and LNG:

U.S.	United States
Peak Shaver	An LNG facility used to supplement the supply of natural gas during times of high gas demand
Satellite	An LNG facility used to supply gas to a localized area
Sendout	Natural gas or vaporized LNG sent out, by a gas utility, via pipelines to customers
Boil-off gas	The gas that boils off from an LNG tank as heat is leaked into the tank from the environment (sometimes just called boil-off)
F	Degree Fahrenheit (a measure of temperature)
Psia	Pounds force per square inch absolute (above absolute zero)
Psig	Pounds force per square inch above atmospheric pressure
Lbm	Pound mass
Lbf	Pound force (the force exerted due to 1 lbm accelerated at 32.2 ft./sec^2)
Cu.ft.	A cubic foot of volume 1 foot x 1 foot x 1 foot (1' x 1' x 1')
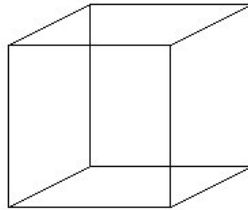
*Figure **Error! No text of specified style in document.**.2: Cubic Foot is a volume 1' x 1' x 1'*

Source: Self-Made

BOG	Boil-off gas
SCF	A standard cubic foot is a 1' x 1' x 1' volume of gas at a standard temperature and pressure. For this document, the American Gas Association (A.G.A.) definition of standard pressure and temperature of 14.73 psia and 60 F is used.
BTU	Btu is the amount of energy needed to raise 1 lbm of water 1 deg F. This is not a precise measure of energy because different industries and countries use a

	different standard temperature for the water being heated. The heat capacity of water differs with temperature.
HHV	Higher heating value is the heat released from burning a SCF of natural gas at 60 F with air at 60 F and bringing the combustion products down to 60 F.
Therm	By definition, a Therm is 100,000 Btu.
Dekatherm	Deca means 10, so a dekatherm is 1,000,000 Btu.
MSCF	In the U.S. Gas Industry, "M" means 1,000. Thus, MSCF is 1,000 SCF.
MMSCF	In the U.S. Gas Industry, "M" means 1,000. Thus, MMSCF is 1,000,000 SCF (one million standard cubic feet)
BSCF	In the U.S. Gas Industry, "B" means billion. Thus, BSCF is 1,000,000,000 SCF (one billion standard cubic feet)
Gallon	Is an imperial unit measure of volume.
m	A meter is a measure of length in the International System of Units
m^3	A volume measurement 1 meter x 1 meter x 1 meter
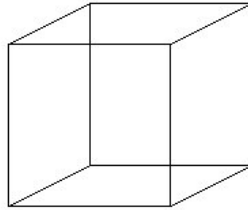
*Figure **Error! No text of specified style in document.**.3: Cubic Foot is a volume 1' x 1' x 1'*

Source: Self-Made

Nm^3	Normal cubic meter is a cubic meter volume of gas at a defined "Normal" temperature and pressure. Different countries and different industries use different values for "Normal" properties.
Barrel	There are two different barrel terms, one for the Alcohol and Petrochemical industries. For this training, the Petrochemical Industry Barrel will be used. 1 barrel = 42 gallons

Approximations used in this training:

1 m^3 ~ 35.3 ft^3

One cu. ft. of LNG ~ 600 SCF of vapor

One cu.ft. ~ 7.48 gallons

1 gallon of LNG ~ 80 SCF of vapor

3. Nitrogen Expansion system

3.1 General

The standard nitrogen expansion system is depicted in Figure 3.1 below. At first glance, it is a bit complex. Thus, for this study, a much simpler nitrogen expansion system will be evaluated first, and then more sophisticated systems will be studied until an understanding of the more complex system shown in Figure 3.1 below is achieved.
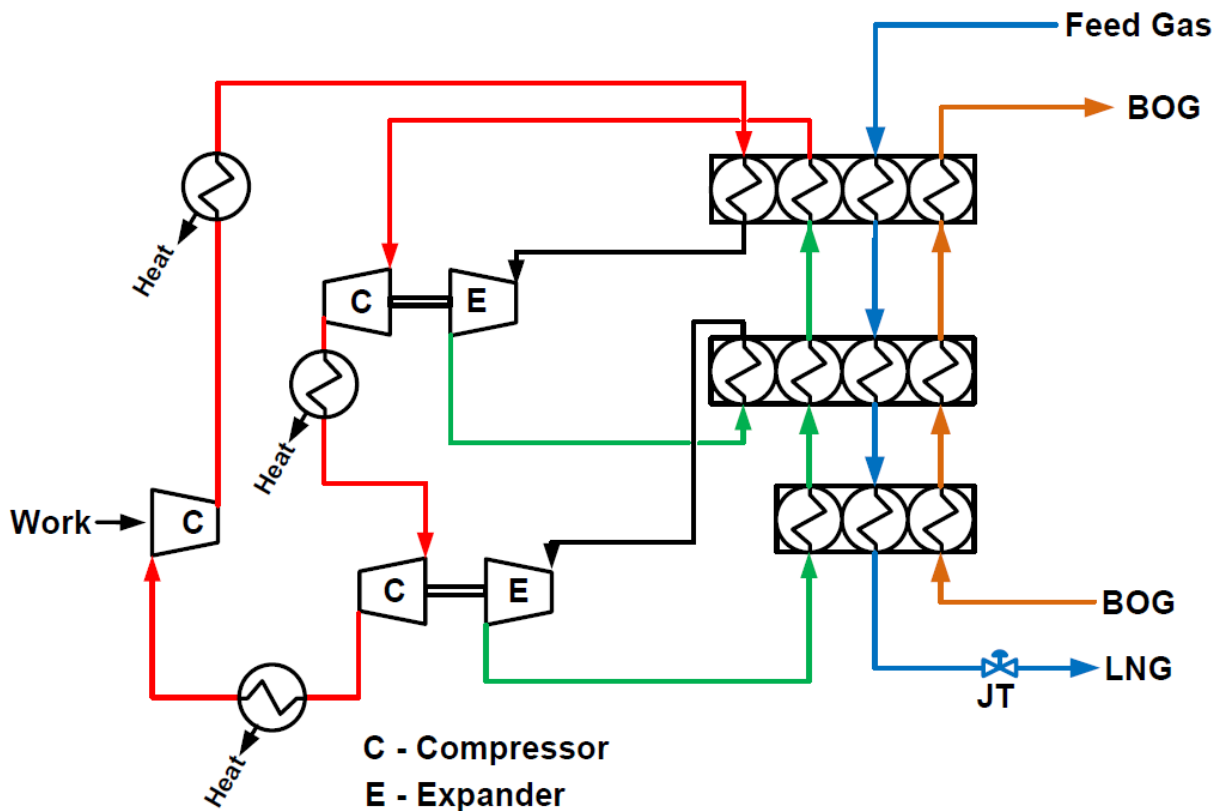
Figure Error! No text of specified style in document..4: Nitrogen expansion process drawing

In all this section's work, only steady state and steady flow problems will be evaluated, and pressure drops will be neglected to learn concepts.

From earlier studies (sections 1, 2, and 3A of this series) it was learned that when a fluid is dropped in pressure across a JT valve, it is a constant enthalpy process. It was also explained

that when that pressure drop occurs across an expander shaft work is produced. The amount of work produced is a function of the efficiency of that expander. If the expander were to be ideal and if there was no heat transfer to or from the expander (no such ideal expander exists in the universe), it would be an adiabatic (no heat transfer) isentropic (constant entropy) (ideal– reversible) expander. If such an adiabatic isentropic expander did exist, one could expand a gas and then use the shaft work of the expander to drive an adiabatic isentropic compressor to bring the gas back up to its original pressure and temperature. That would be a perpetual motion machine that violates the 2nd law of thermodynamics.

Since such ideal expanders and compressors do not exist, a measure of efficiency defines the work produced or consumed by real expanders and compressors. For an expander, a percent of the ideal work output of an ideal expander is used. And a measure of the efficiency of a compressor is the amount of work required to drive an ideal compressor divided by the actual work of the real compressor.

Thus, if a pressure drop between two pressures occurs across an ideal expander would produce 100 Btu/lbm of fluid flow, and it only produced 80 Btu/lbm of fluid flow, that real expander would be 80% efficient. If a block of wood is shoved into the rotor so that it would not spin, then no work would come out of the expander. It would be a zero-efficient expander. It would become a constant enthalpy process, identical to dropping the pressure across a JT valve.

The same holds for a compressor. If an ideal compressor would require 100 Btu/lbm to increase the pressure from pressure one to pressure two, then an 80% efficient compressor would require 100/0.8 or 125 Btu/lbm to compress the gas from pressure 1 to pressure 2.

This relationship can be written in a logical mathematical form as follows:

Efficiency (compressor) = ideal work $_{in}$ / actual work $_{in}$ = isentropic work $_{in}$ /actual work $_{in}$

Efficiency (expander) = actual work $_{out}$ / ideal work $_{out}$ = actual work $_{out}$ / isentropic work $_{out}$

When a piece of wood is shoved into an expander, no work comes out, so where does the energy of the fluid go? It goes into the fluid as heat. That heat is caused by viscous dissipation

of the fluid molecules turbulently crashing past each other with friction forces resulting in heat being liberated by these collisions and shear forces.

Consider this: some of the fluid's energy (its enthalpy) going into an expander can be converted into shaft work. If it is converted into shaft work, by the first law of thermodynamics, the fluid leaving the expander must be lower by the amount that has left as shaft work. E.g., If the fluid went into the expander with a fluid energy of 200 Btu/lbm flowing through the expander and 100 Btu/lbm flowing through the expander leaves the fluid as shaft work, then the fluid leaving the expander must have only 100 Btu/lbm left in it when it leaves the expander.

That means that since the fluid gave up 100 Btu/lbm, its enthalpy has dropped. Associated with this enthalpy drop is a drop in temperature. The more efficient the expander, the colder the outlet gas will be for a given pressure drop and the higher the shaft work will be. However, one cannot reach or exceed a 100% efficient expander. A 100% efficient expander would be an isentropic compressor, violating the 2nd law of thermodynamics. Within the scope of this learning, the 2nd law of thermodynamics will not be covered other than to say that it cannot be violated (expanders cannot be at or greater than 100% efficient).

The same logic can be used to explain what happens in a compressor. If an ideal compressor (adiabatic isentropic) took 100 Btu/lbm (shaft work) to compress a gas from pressure one to pressure two, the fluid's enthalpy leaving the compressor would increase by the amount of shaft work (100 Btu/lbm) put into the fluid. Associated with this enthalpy increase is an increase in the fluid's temperature. The more inefficient the compressor is, the more work it takes to compress the gas, resulting in a higher enthalpy of the outlet fluid—the higher enthalpy, the higher the compressor outlet temperature for a given delta pressure.

3.2 Expanders

Figure 3.2 shows an expander expanding nitrogen.
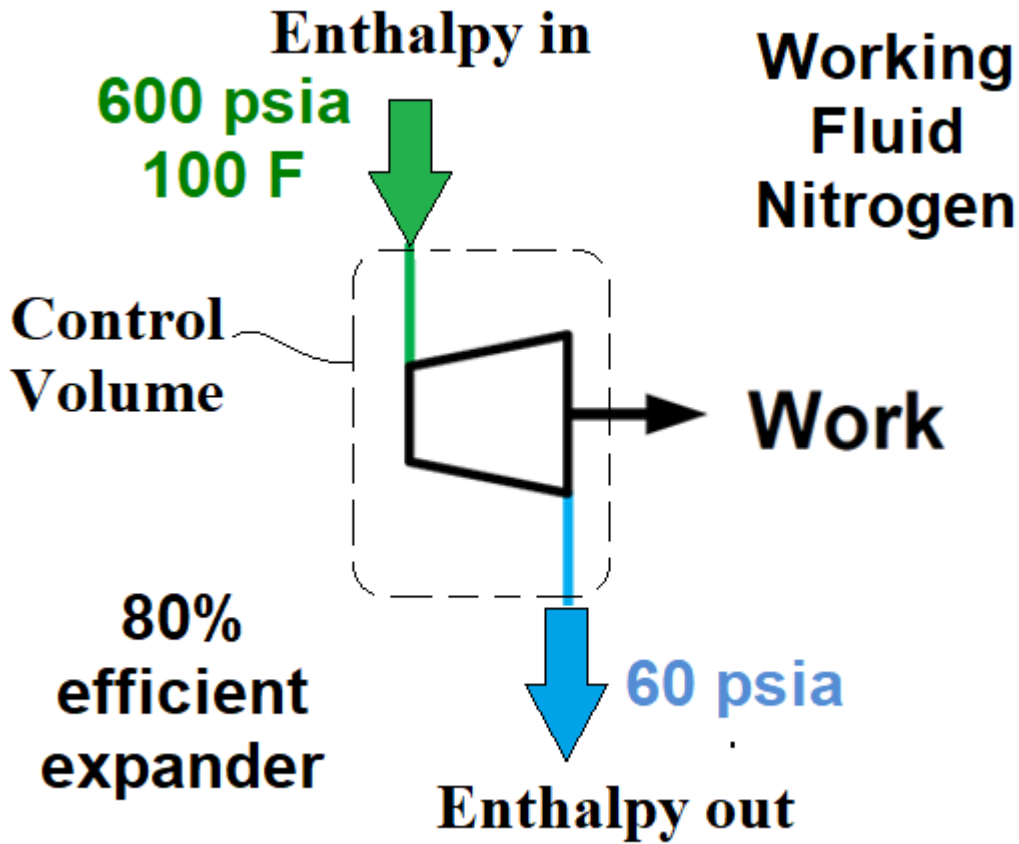
*Figure **Error! No text of specified style in document.**.5: Expander (enthalpy out = enthalpy in – shaft work out)*

For this example, the expander is assumed to be 80% efficient, and the inlet fluid state is defined by 600 psia and 100 F. The outlet fluid pressure is given as 60 F.

In Figure 3.3, the data is computed using REFPROP software.

Since the state at the inlet fluid is known by two intensive properties in row 1, the rest of the table populates after 100 F and 600 psia are input.

Next, in row 2, the isentropic process data is calculated by setting two intensive properties. One of the properties is the pressure of 60 psia, and the other property is the entropy. Since an

isentropic expander is being analyzed, the entropy is identical to that of row 1 (1.3751). Once these two values are input the rest of the data on the table populates.

Using rows 1 and 2, the shaft work can be calculated as the change in enthalpy (135.40 Btu/lbm – 69.733 Btu/lbm) = 65.667 Btu/lbm of shaft work.

Now apply 80% efficiency. The real shaft work is 65.667 Btu/lbm x 0.8 = 52.5336 Btu/lbm which is the Real Shaft Work. Thus, the real enthalpy at the outlet of the expander is 135.40 Btu/lbm (from row) 1 – 52.5336 Btu/lbm (just computed) = 82.866 Btu/lbm.

Now input into row 3 the two known intensive properties of the pressure of 60 psia and enthalpy of 82.866. When these values are input, the rest of the table populates. Take note of the temperature of the isentropic and the real expander at -173.04 F and 121.51 F

16: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	600.00	2.7999	135.40	1.3751
2	-173.04	60.000	0.55747	69.733	1.3751
3	-121.51	60.000	0.46840	82.866	1.4172

*Figure **Error! No text of specified style in document.**.6: Expander Data from REFPROP*

Figure 3.4 shows this on a pressure enthalpy diagram with the isentropic (ideal expander), the 80% efficient compressor, and the JT valve process flow plotted on it. As can be seen, the 80% efficient compressor results in a warmer outlet temperature because of losses due to viscous dissipation friction losses within the expander. As improved machining methods have been developed, closer tolerances and smoother surfaces on the rotors and volutes have resulted in higher-efficiency expanders and compressors.

As shown in Figure 3.4, the isentropic expander would be a 100% efficient expander, the JT valve would be the same as a 0% efficient expander, and the real-world expander would be somewhere in between. It is typical to expect expanders to be 80 – 85% efficient.
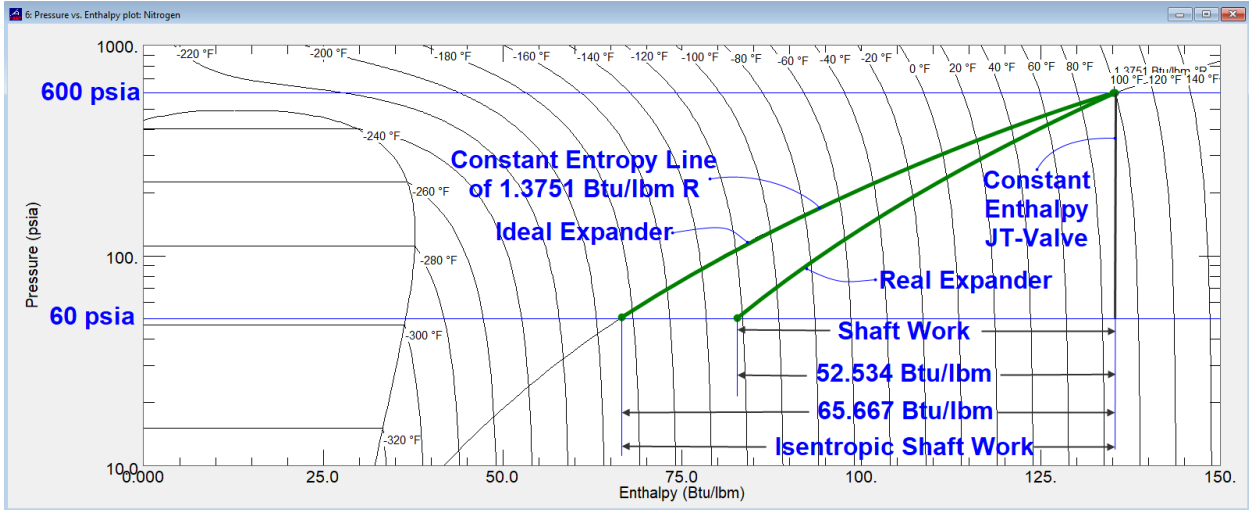
*Figure **Error! No text of specified style in document.**.7: Ph diagram showing an isentropic expansion, an 80% efficient expander and a JT valve expansion.*

3.3 Compressors

The very same logic can be used for compressors. Figure 3.5 shows a compressor used to compress nitrogen from 30 F and 60 psia to 600 psia. Figure 3.6 shows in row 1 and 2 the adiabatic isentropic compression consumes (234.82 -120.99 = 113.83 Btu/lbm) of work. Note that the entropy is the same for row 1 and row 2 at 1.5104 Btu/lbm R.

If this is a real compressor at 80% efficiency, the actual work is 113.83/0.8 = 142.29 Btu/lbm. Thus, the new real-world enthalpy would be 120.99 + 142.29 = 263.28. In row 3 input the enthalpy of 263.28 Btu/lbm and pressure of 600 psia into the table, and since two intensive properties have been input to define the state, the rest of the table populates. Note that the outlet temperature of the real-world compressor is significantly hotter than that of the ideal compressor. That is because the additional work produced heat.
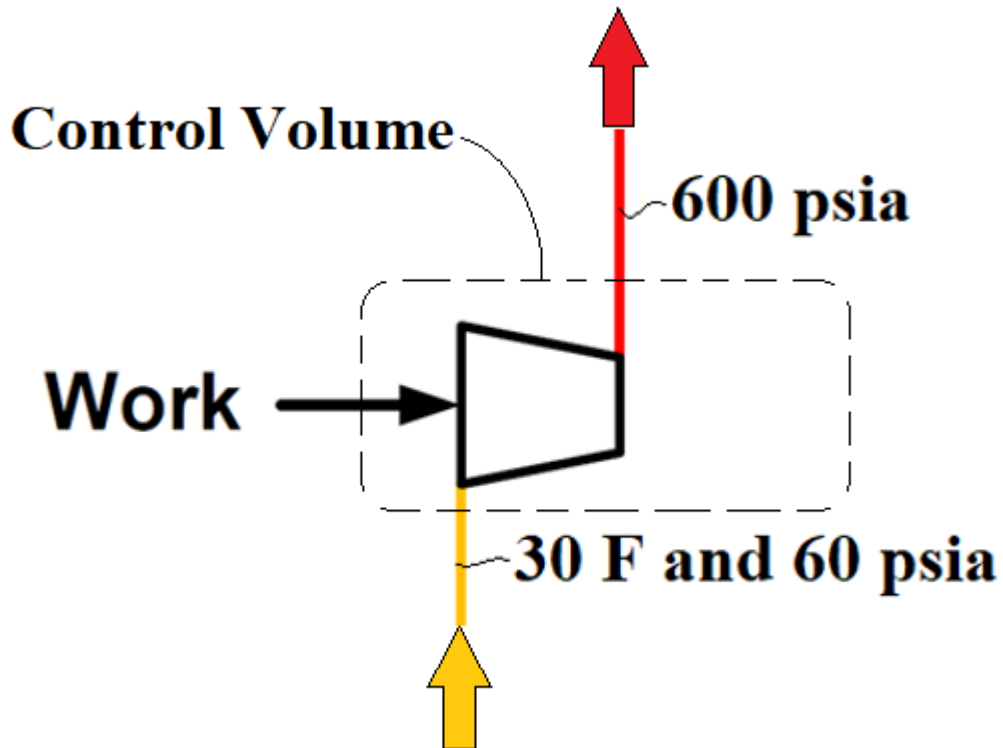
*Figure **Error! No text of specified style in document.**.8: An 80% efficient nitrogen compressor showing work in, state of fluid in and outlet pressure*

*Figure **Error! No text of specified style in document.**.9: REFPROP data related to compressor shown in Figure 3.5*

Figure 3.7 shows the compression process on a Ph diagram. Note that the work required to drive a real-world compressor is greater than that required to drive an isentropic compressor.

*Figure **Error! No text of specified style in document.**.10: Ph diagram showing an isentropic compressor, and an 80% efficient compressor*

3.4 Apply the use of nitrogen compressor and expander to produce refrigeration

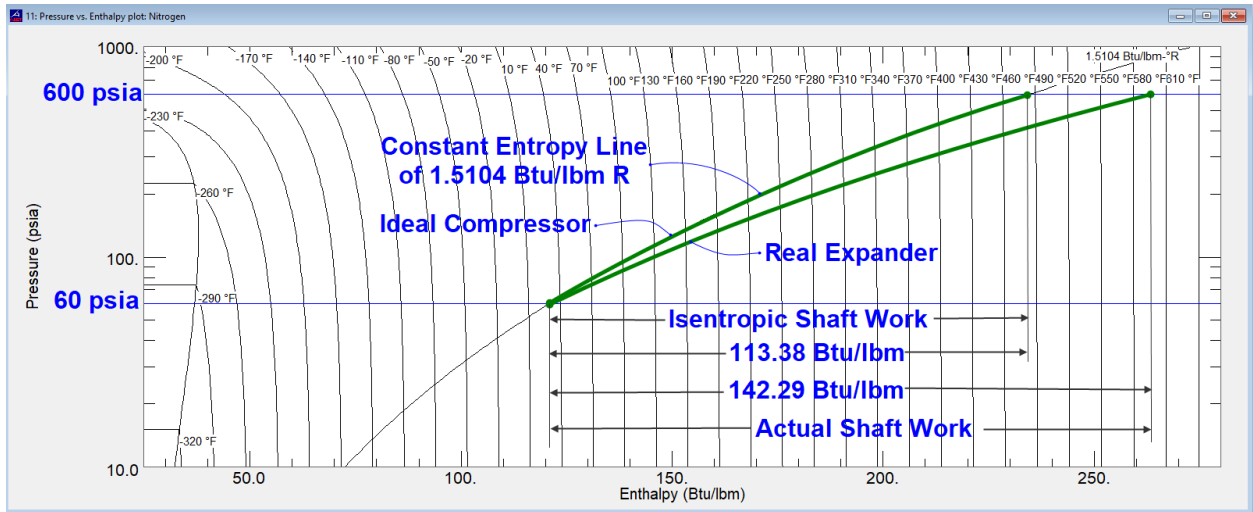Figure 3.8 depicts a very simple compressor expander refrigeration system. The compressor compresses gas to a high pressure and high temperature. That high enthalpy gas is cooled in a heat exchanger called an aftercooler. The aftercooler might be a **fin-fan exchanger** that transfers heat to the atmosphere. As shown in Figure 3.8, the aftercooler rejects heat, which lowers the gas's enthalpy.

The fluid is then expanded in an expander. Since the expander extracts work from the fluid, according to the first law of thermodynamics, the fluid exiting the expander has a lower enthalpy. This lower enthalpy is associated with a lower temperature. That low-temperature fluid removes heat from whatever needs to be cooled down. As heat is extracted from whatever needs to be cooled, the working fluid increases enthalpy as it increases in temperature. The fluid leaves the chiller and is returned to the inlet of the compressor, where the process starts all over again.
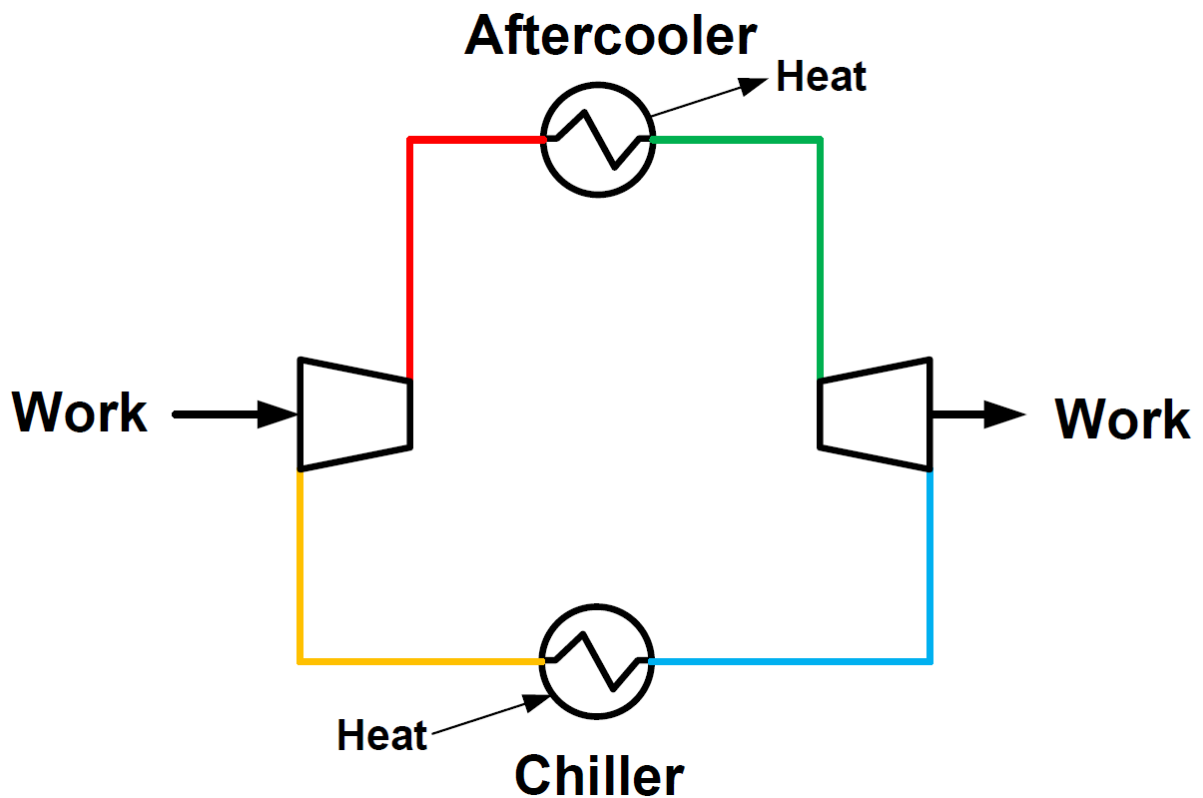

*Figure **Error! No text of specified style in document.**.11: A simple compressor – expander refrigeration system*

Now define the working fluid (nitrogen) and apply some numerical values to identify the state of the fluid at one part of the process. Assume the pressure at the outlet of the compressor and the temperature at the outlet of the aftercooler (typically fin-fan type). Do realize that if the aftercooler transfers heat to the environment, then the temperature at the outlet of the aftercooler is directly related to the ambient temperature. In this case, it was assumed that the ambient temperature was 90 F and that the nitrogen in the aftercooler was cooled down to 100 F. Another assumption is that the compressor and the expander are 80% efficient. This is all shown in Figure 3.9.
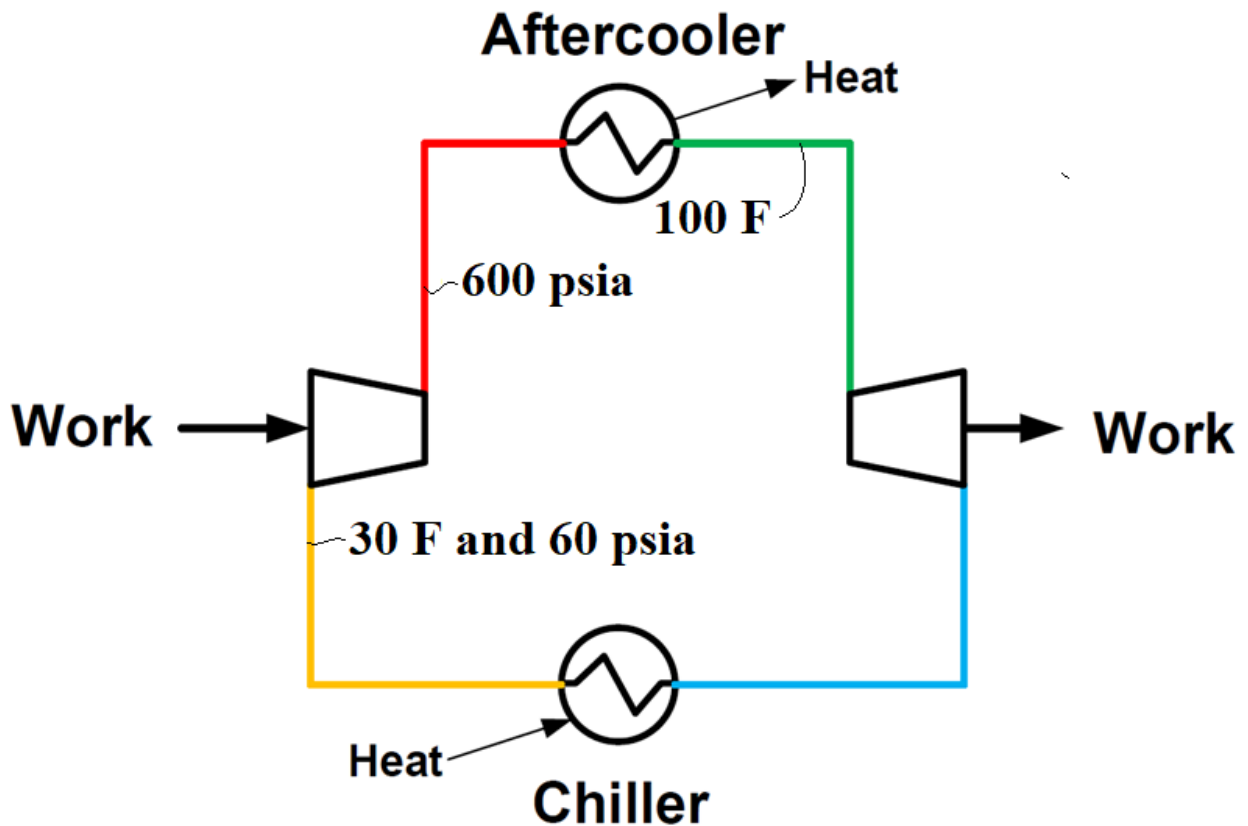

*Figure **Error! No text of specified style in document.**.12: Nitrogen expansion system used for chilling*

The first analysis is that of the compressor. A control volume is drawn around the compressor and the values avail are shown on the drawing shown in Figure 3.10. This happens to be the exact same compressor considered earlier so the details of the analysis are the same.

In the real world this compression would be done with multiple compressors with intercooling to improve efficiency. This will be done later in this study.

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

Adiabatic Isentropic first analysis $s_{in} = s_{out}$ (s stands for entropy)

Assume $\dot{m} = 1$ lbm/hr. for simplicity $\dot{m}h_{in} + \dot{m}w_{in} = \dot{m}h_{out}$ (h stands for enthalpy) (m stands for mass)
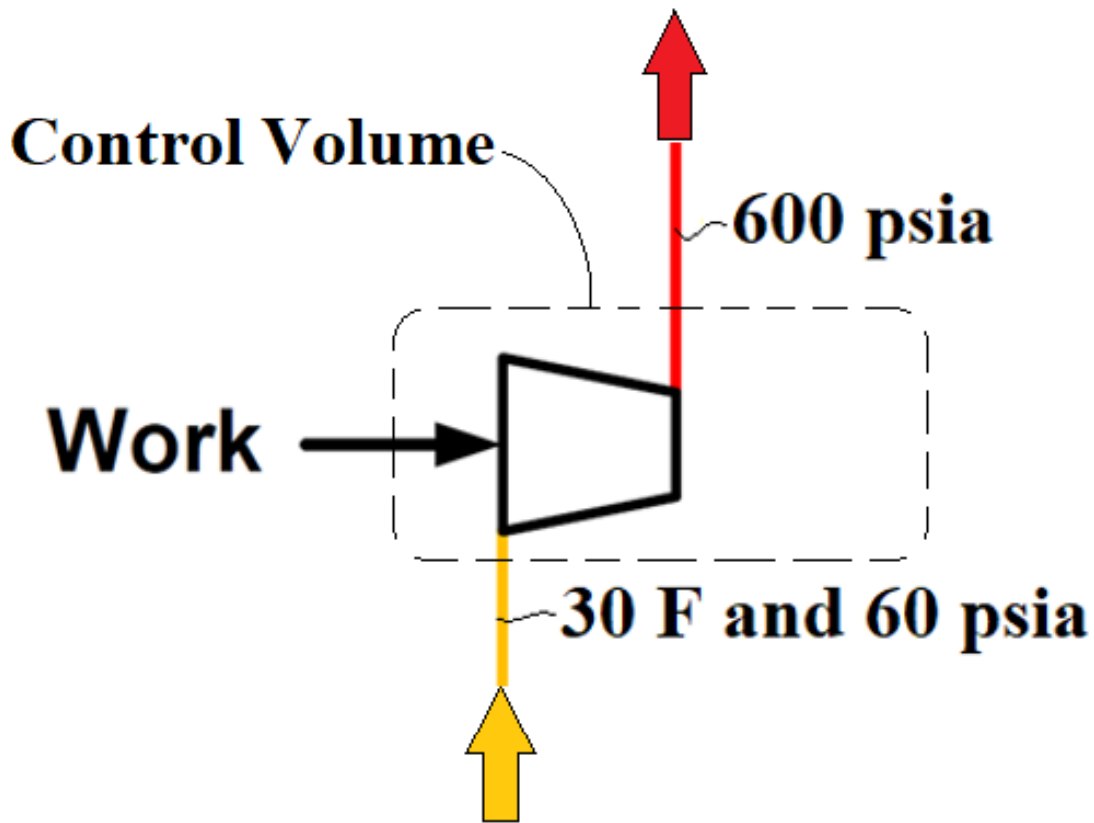
$w_{in} = h_{out} - h_{in}$ (W stands for work)

*Figure **Error! No text of specified style in document.**.13: The compressor used in Figure 3.9*

Figure 3.11 shows the data related to the compressor shown in Figure 3.10. Note that the outlet temperature is 595.38 F.

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	30.000	60.000	0.32046	120.99	1.5104
2	485.26	600.00	1.6286	234.82	1.5104
3	595.38	600.00	1.4583	263.28	1.5389
4					

*Figure **Error! No text of specified style in document.**.14: REFPROP data related to compressor shown in Figure 3.9*

From Figure 3.11 it can be seen that:
Adiabatic Isentropic work = 234.82 – 120.99 = 113.83 Btu/hr.

Actual work at 80% efficiency = 263.28 – 120.99 = 142.29 Btu/hr.

The data from Figure 3.11 is used as input data for the analysis of the aftercooler (typically fin-fan type) shown in Figure 3.12.
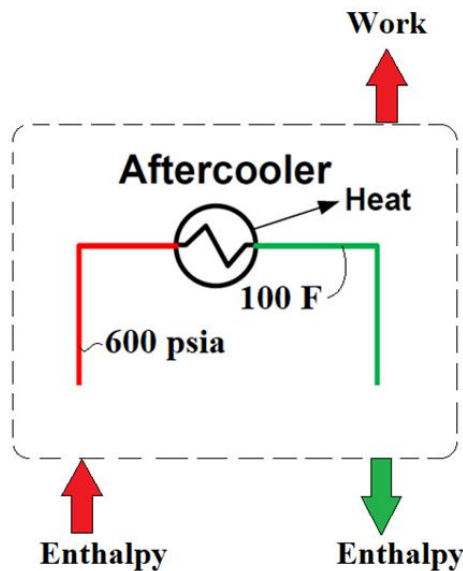
*Figure **Error! No text of specified style in document.**.15: Aftercooler shown in Figure 3.9*

Once the data from Figures 3.11 and 3.13 are input into the REFPROP table shown in Figure 3.13, the table populates. From this 263.28 Btu/lbm – 135.40 Btu/lbm = 128.88 Btu/lbm are removed from nitrogen (rejected from the system).

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	595.38	600.00	1.4583	263.28	1.5389
2	100.00	600.00	2.7999	135.40	1.3751
3					

14: Nitrogen: Specified state points

*Figure **Error! No text of specified style in document.**.16: REFPROP data related to compressor shown in Figure 3.12*

From Figure 3.13 it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$$\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}q_{out}$$

$\dot{m} = 1$ lbm/hr.

$q_{out} = h_{in} - h_{out}$ (q is heat)

$q_{out} = 263.28 - 135.40 = 127.88$ Btu/hr.

Next Analyze the expander shown in Figure 3.14. In the real world this expansion would be done with multiple expanders with interheating (by removing heat from natural gas to be liquefied to improve efficiency (this will be shown later in this learning). The term interheater or interheating is use to mean (expand then absorb heat then expand and the absorb heat again).

SSSF Process, First Law of Thermodynamics– Energy in = Energy out
Adiabatic Isentropic first analysis $s_{in} = s_{out}$
Assume $\dot{m} = 1$ lbm/hr. for simplicity $\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}w_{out}$
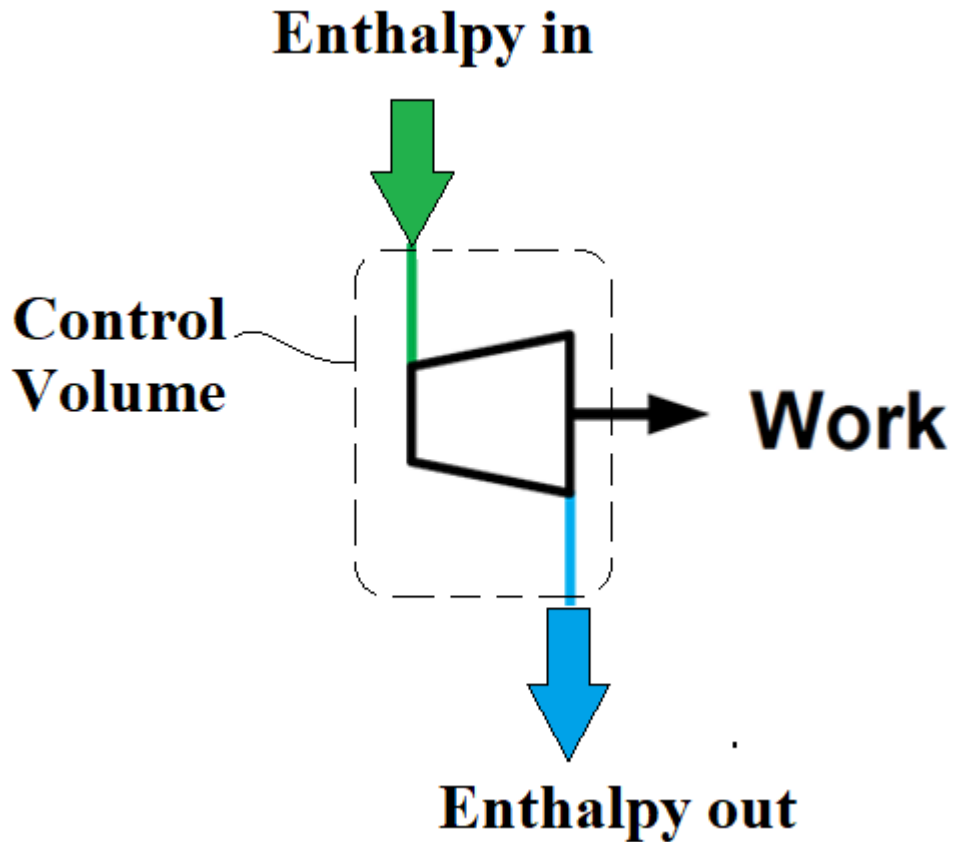$w_{out} = h_{in} - h_{out}$

*Figure **Error! No text of specified style in document.**.17: Expander used in Figure 3.9*

*Figure **Error! No text of specified style in document.**.18: REFPROP data related to compressor shown in Figure 3.14*

From Figure 3.15 it can be seen that:

w_{out} = h_{in} − h_{out} =135.4 − 82.866 = 52.534 Btu/hr. for each N_2 lbm

Adiabatic Isentropic Expansion s_{in} = s_{out}, Temp outlet = -173.04 F
Real world Temp outlet = -121.51 F
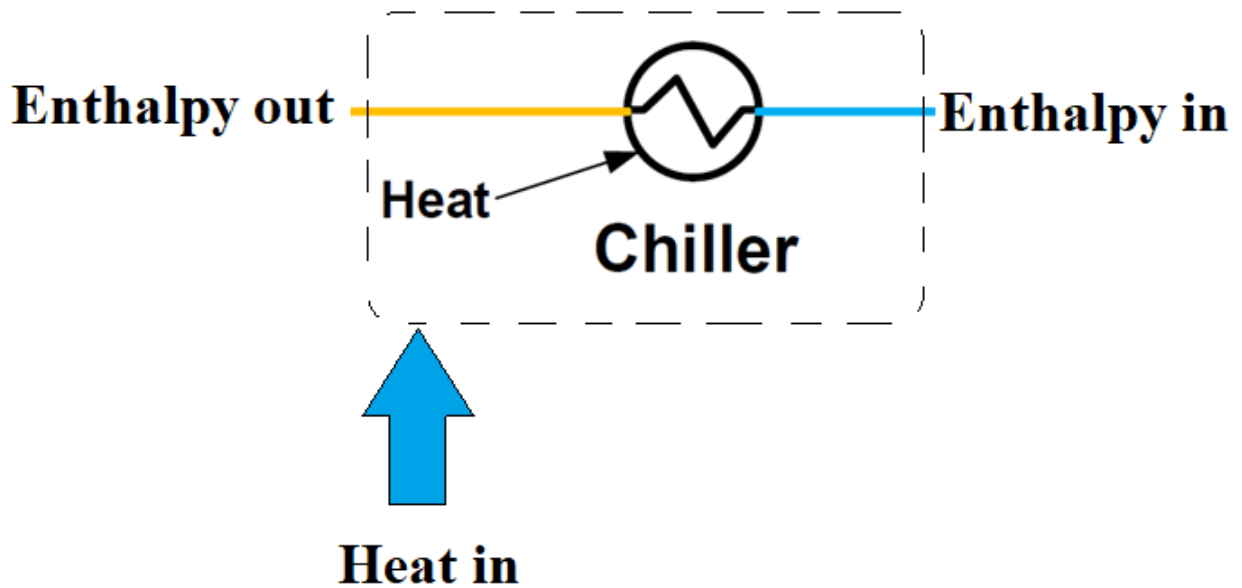
Next Analyze the chiller shown in Figure 3.9.

*Figure **Error! No text of specified style in document.**.19: Chiller shown in Figure 3.9*

*Figure **Error! No text of specified style in document.**.20: REFPROP data related to compressor shown in Figure 3.16*

From Figure 3.17 it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} + \dot{m}q_{in} = \dot{m}h_{out}$

$\dot{m}$ = 1 lbm/hr.
$q_{in} = h_{out} - h_{in}$
q_{in} = 120.99 – 82.866 = 38.124 Btu/hr.

Now analyze how much heat must be removed from 5 MMSCFd to liquefy it.
Assume pretreated feed gas is available at 40 F and 400 psia and assume storage temperature is saturated at 16.5 psia tank pressure. Assume a LNG composition of 94% methane, 5% ethane, and 1% propane.

8: Methane/Ethane/Propane: Saturation points (at equilibrium) (0.94/0.05/0.01)

	Liquid Phase Temperature (°F)	Vapor Phase Temperature (°F)	Liquid Phase Pressure (psia)	Vapor Phase Pressure (psia)	Liquid Phase Density (lbm/ft³)	Vapor Phase Density (lbm/ft³)	Liquid Phase Enthalpy (Btu/lbm)	Vapor Phase Enthalpy (Btu/lbm)	Liquid Phase Entropy (Btu/lbm-°R)	Vapor Phase Entropy (Btu/lbm-°R)
1	-254.7431	-254.7431	16.50000	16.50000	27.40162	0.1251812	-4.371164	221.4899	0.01281178	1.088110
2										

7: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	60.00000	14.73000	0.04506871	370.3947	1.515155
2	-256.8269	400.0000	27.64808	-4.371164	0.0001582132
3	40.00000	400.0000	1.367036	346.2096	1.089738
4					

*Figure **Error! No text of specified style in document.**.21: REFPROP data showing how much heat must be removed to produce LNG from feed gas.*

It can be seen from Figure 3.12 that:

Each SCF has a mass of 0.04506871 Lbm/SCF. Thus, 5 MMSCFD = 225,343.55 lbm/day which is equal to 9,389.3 lbm/hr. NG.

Heat removal from the pretreated natural gas =
9,389.3 lbm/hr. x (346.2096 – (– 4.3712164)) = 3,291,708 Btu/hr.
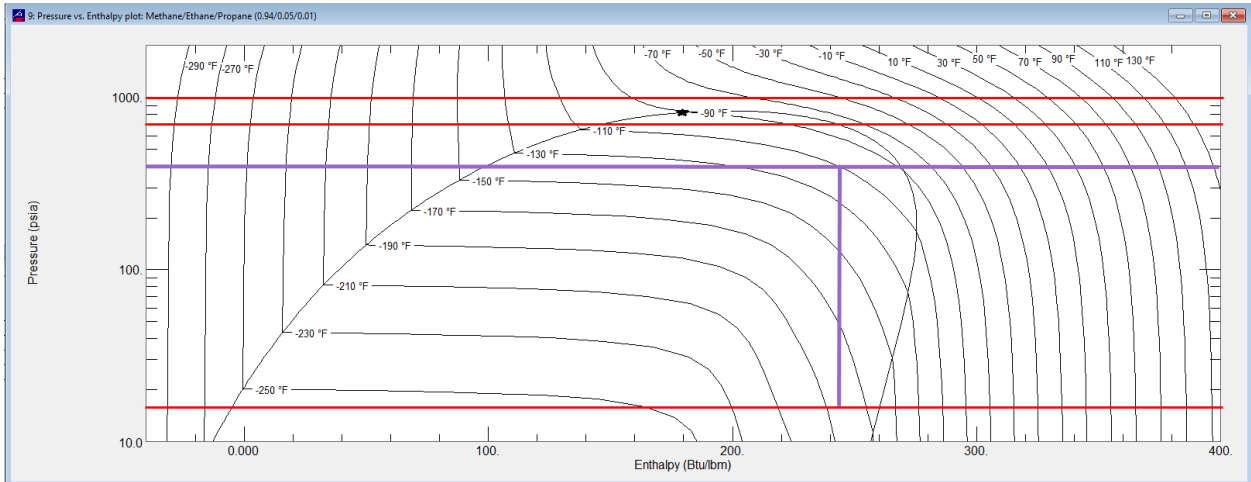
Plot this on a PH diagram as shown in Figure 3.19.

*Figure **Error! No text of specified style in document.**.22: Ph diagram for a 94% methane, 5% ethane, 1% propane gas mixture*

From Figure 3.19 the present conditions of 400 psia feed gas being brought down to -110 F would not be practical to operate because it would produce a very low yield of an LNG that would be very high in propane and ethane and thus not be usable. If the feed gas were to be 700 psia, it would produce a better grade of LNG and if it were at 1,000 psia, it would produce an even better grade of LNG. However, the present design as shown in Figure 3.9 and conditions of 400 psig feed gas is not viable as the LNG produced would have only 6.6% methane.

6.6% methane not usable LNG

*Figure **Error! No text of specified style in document.**.23: Composition of LNG produced by an LNG of -110 F and 400 psia*

Even if the feed gas were to be 600 F and brought down to -110 F as shown in Figure 3.21 the LNG produced would not be usable as it would have only 63% methane.

Not usable LNG

*Figure **Error! No text of specified style in document.**.24: Composition of LNG produced by LNG of -110F and 600 psia*

The analysis can be checked by putting a control volume around the entire plant and confirming that the first law of thermodynamics confirms that the energy in is equal to the energy out. This is shown in Figure 3.22.

Put control volume around the entire plant
No mass crossing the control volume shown
Heat is extracted from natural gas stream

Energy in = Energy out

$w_{in} + q_{in} = w_{out} = q_{out}$

For each lbm/hr. of N2

$Work_{in} + Heat_{in} + = Work_{out} + Heat_{out}$

142.29 Btu/lbm + 38.124 Btu/lbm = 52.534 Btu/lbm + 127.88 Btu/lbm

180.414 Btu/Lbm = 180.414 Btu/Lbm CHECK

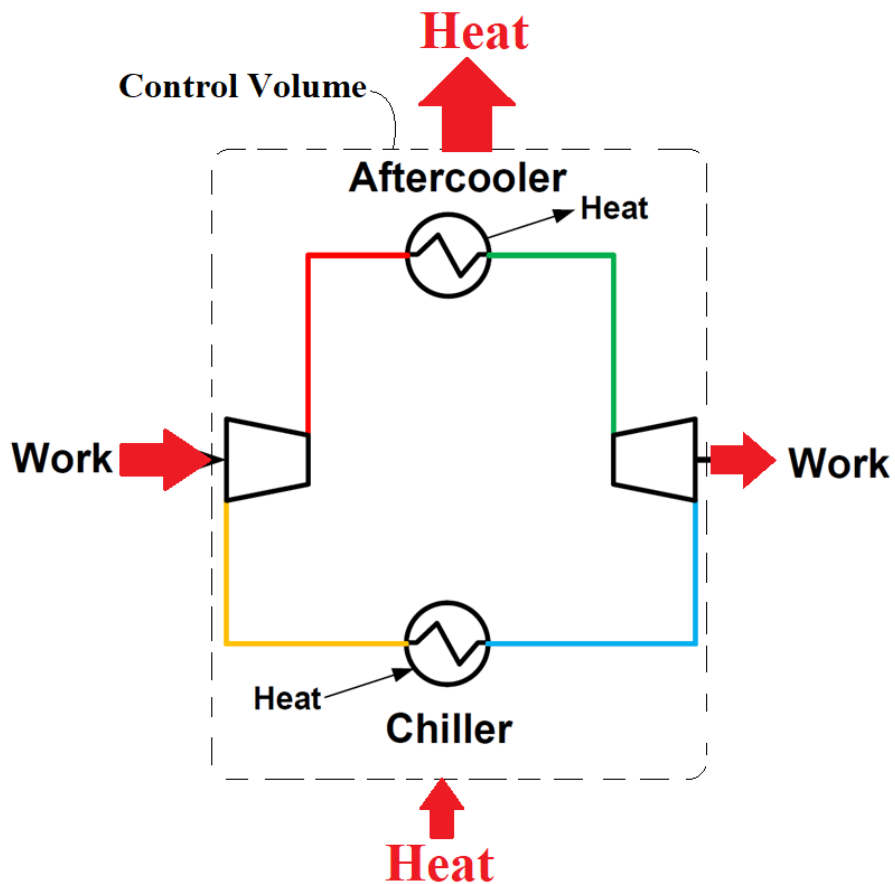**The process works to produce a lot of cooling. A lot of cold flash gas but just not much LNG**

It should be noted that in a nitrogen expansion system the compressor requires shaft work, and the expander puts out shaft work. Clearly as stated earlier, the expander puts out less work than the compressor requires, but it is typical to put the expander and the compressor on the same shaft so that the shaft work output of the expander lessens the amount of work that needs to be put into the compressor via a prime mover (electric or fuel driven shaft work producer). This is called "Net-work Required".

In the learning above the Net-work was:
Use expander work to offset compressor work to determine the Net-work needed to run the process.

Net-work in = (142.29 – 52.534) Btu/lbm$_{N2}$ = 89.756 Btu/lbm$_{N2}$

3.5 How to use the nitrogen compressor and expander to produce refrigeration, while driving the temperature lower

This process is not viable as it requires a very high feed gas pressure and recovery of cooling from a large amount of flash Gas and even with that the LNG is unusable as it does not contain enough methane in the LNG produced.

There are several methods to lower the temperature of the chiller. One method is to increase the pressure drop across the expander. This could be accomplished by raising the inlet pressure to the expander. Another method would be to lower the outlet pressure of the expander. Lowering the outlet pressure is more effective than raising the inlet pressure because the temperature drop is a function of the pressure ratio (using the absolute pressure units). Thus, lowering the outlet pressure of the expander by say 30 psia has a much more profound effect on the pressure ratio than raising the inlet pressure by 30 psia. There are problems with doing that that we will not cover in this learning.

Another method of lowering the temperature of the expander is to lower the temperature of the nitrogen entering the expander. This is the method that will be used in the next analysis.

One way to lower the temperature of the inlet of the expander is to transfer heat from the inlet of the expander to the chiller. That is the method that will be used.

Bring the expander inlet down to 0 F by giving off heat to the chiller, as shown in Figure 3.23

Assume Compressor and Expander are 80% Efficient and Neglect Pressure Drops
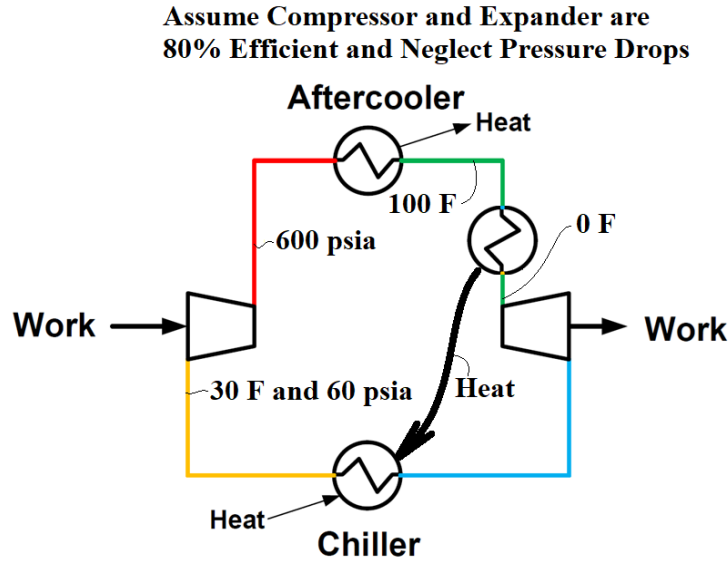
Figure Error! No text of specified style in document..26: Nitrogen expansion unit used to obtain lower temperature by bringing the inlet temperature of the expander down

For this new process analyze the compressor. It has not changed from the earlier analysis. See Figures 3.10 and 3.24.

In the real world this compression would be done with multiple compressors with intercooling to improve efficiency.

SSSF Process, First Law of Thermodynamics is: Energy in = Energy out

Adiabatic Isentropic first analysis $s_{in} = s_{out}$

Assume $\dot{m} = 1$ lbm/hr. for simplicity $\dot{m}h_{in} + \dot{m}w_{in} = \dot{m}h_{out}$
$w_{in} = h_{out} - h_{in}$
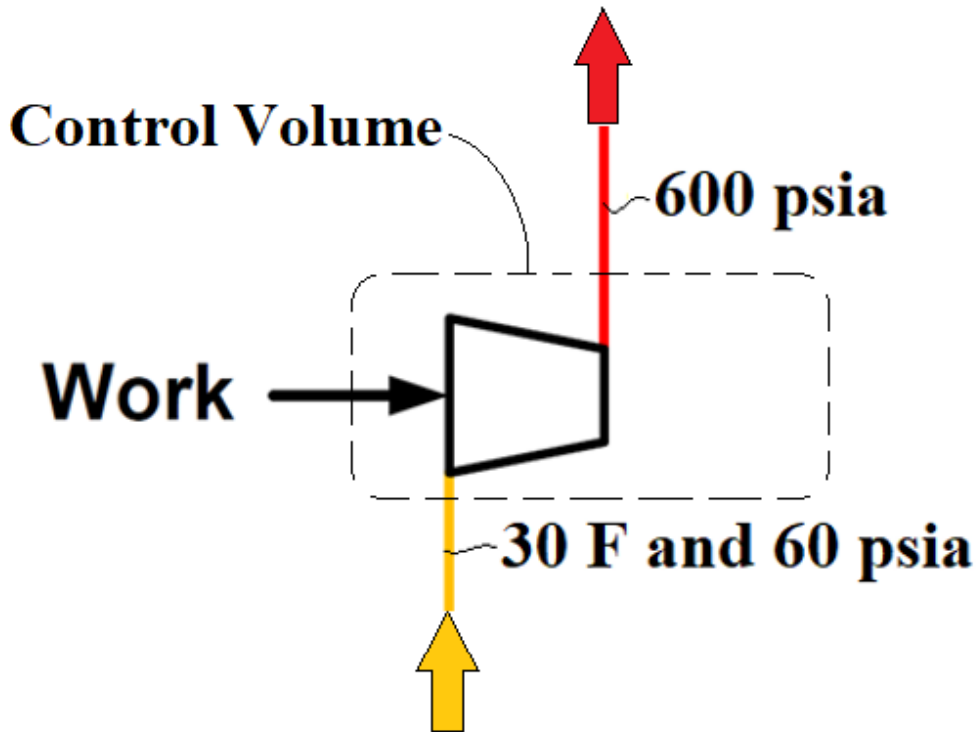
*Figure **Error! No text of specified style in document.**.27: Nitrogen compressor same as from Figure 3.23*

From Figure 3.25 it is seen that:

Adiabatic Isentropic work = 234.82 – 120.99 = 113.83 Btu/hr.

Actual work at 80% efficiency = 263.28 – 120.99 = 142.29 Btu/hr.

13: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	30.000	60.000	0.32046	120.99	1.5104
2	485.26	600.00	1.6286	234.82	1.5104
3	595.38	600.00	1.4583	263.28	1.5389
4					

*Figure **Error! No text of specified style in document.**.28: REFPROP data associated with Figure 3.24*

Next Analyze the aftercooler. This too is the same as the analysis performed for the earlier example. See Figure 3.26.
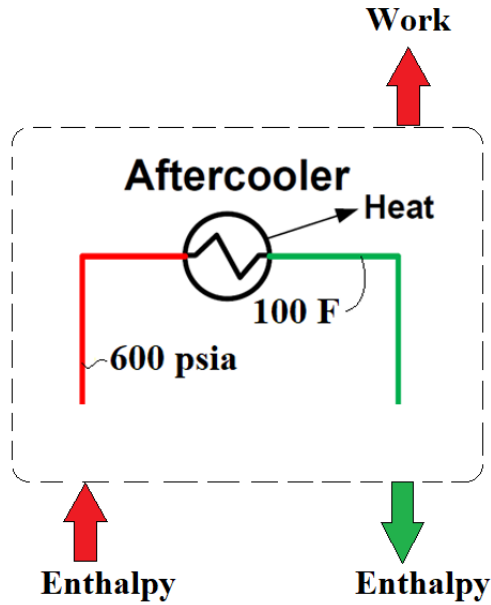
Figure Error! No text of specified style in document..29 Control volume around Aftercooler

From Figure 3.27 it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}q_{out}$

$\dot{m} = 1$ lbm/hr.

$q_{out} = h_{in} - h_{out}$

$q_{out} = 263.28 - 135.40 = 127.88$ Btu/hr.

14: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	595.38	600.00	1.4583	263.28	1.5389
2	100.00	600.00	2.7999	135.40	1.3751
3					

Figure Error! No text of specified style in document..30:REFPROP data related to Figure 3.26

Next Analyze the new heat exchanger that has been added to obtain colder inlet temperature to the inlet of the expander. This new heat exchanger is shown in Figure 3.28.
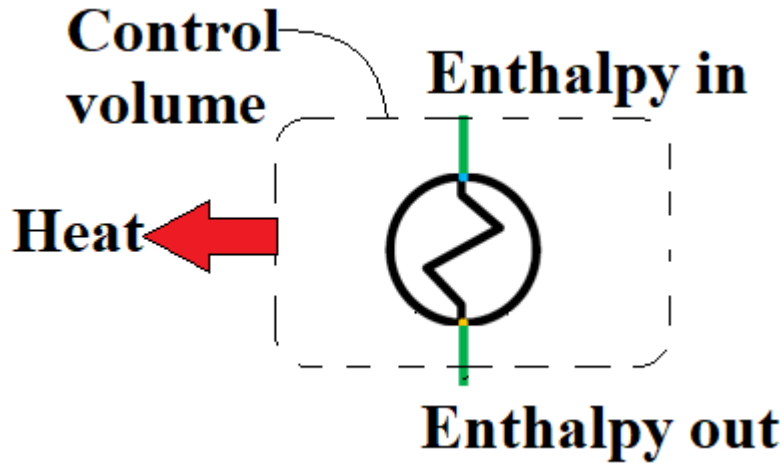
*Figure **Error! No text of specified style in document.**.31: New heat exchanger used to lower the temperature of the inlet to the expander*

From Figure 3.29 it can be seen that:
SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}q_{out}$

$\dot{m} = 1$ lbm/hr.

$q_{out} = h_{in} - h_{out}$

$q_{out} = 135.4 - 108.66 = 26.74$ Btu/hr.

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	600.00	2.7999	135.40	1.3751
2	0.00000	600.00	3.4893	108.66	1.3224
3					

*Figure **Error! No text of specified style in document.**.32: REFPROP data related to Figure 3.28*

Next Analyze the expander shown in Figure 3.30 with the lowered inlet temperature.

SSSF Process, First Law of Thermodynamics– Energy in = Energy out
Adiabatic Isentropic first analysis $s_{in} = s_{out}$
Assume $\dot{m} = 1$ lbm/hr. for simplicity $\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}w_{out}$
$w_{out} = h_{in} - h_{out}$
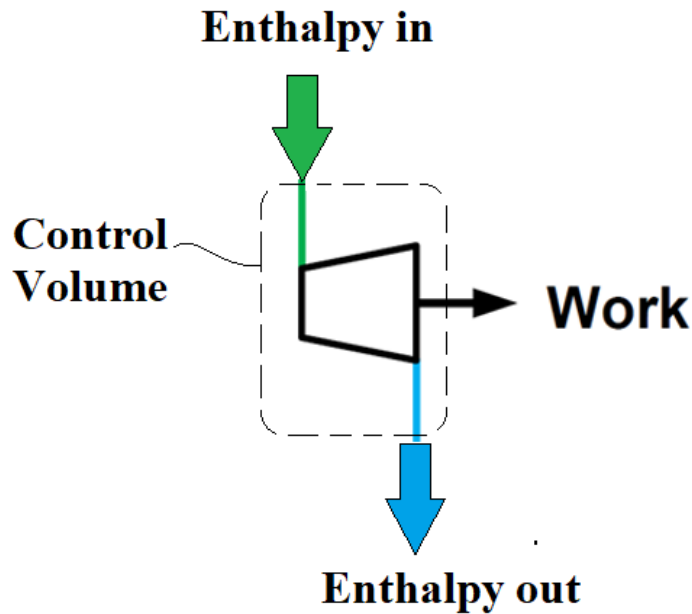
Adiabatic Isentropic Expansion $S_{in} = S_{out}$, Temp outlet = -225.69 F

Real world Temp outlet = -185.28 F
Real world output work is (108.66 – 66.586) = 42.074 Btu/lbm.

As can be seen the outlet temperature of the expander is much colder than the previously studied system (-185 F vs. -122 F)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	600.00	2.7999	135.40	1.3751
2	0.00000	600.00	3.4893	108.66	1.3224
3	-225.69	60.000	0.69577	56.068	1.3224
4	-185.28	60.000	0.58415	66.586	1.3639
5					

*Figure **Error! No text of specified style in document.**.33: REFPROP data related to Figure 3.29*

Next Analyze the chiller shown in Figure that is not absorbing heat from the inlet to the expander heat exchanger and from the LNG to be liquefied.

*Figure **Error! No text of specified style in document.**.34: Chiller which is absorbing heat from both the inlet to the expander and from the feed gas that is being liquefied*
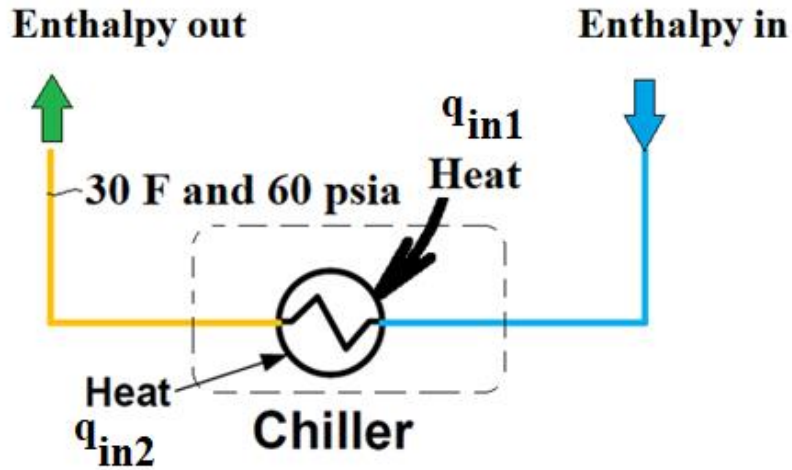
From the data shown in Figure 3.32 it can be seen that:
SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} + \dot{m}q_{in1} + \dot{m}q_{in2} = \dot{m}h_{out}$ (where q_{in1} is the heat removed from the inlet to the expander and q_{2in} is the heat extracted from the feed gas that is to be liquefied)

$\dot{m} = 1$ lbm/hr.
$q_{in2} = h_{out} - h_{in} - \dot{m}q_{in1}$
$q_{in2} = 120.99 – 66.586 – 26.74 = 27.664$ Btu/hr.

At a temperature of – 185 F, this can be used to liquefy pretreated natural gas at 400 psia feed gas pressure.

*Figure **Error! No text of specified style in document.**.35: REFPROP data related to Figure 3.29*

As done earlier analyze how much heat must be removed from 5 MMSCFD to liquefy it.

Assume pretreated feed gas is available at 40 F and 400 psia and assume storage temperature is saturated at 16.5 psia tank pressure. Use the data from Figure 3.33.

*Figure **Error! No text of specified style in document.**.36: Data related to how much energy is needed to liquefy treated feed gas*

Each SCF has a mass of 0.04506871 Lbm/SCF. Thus, 5 MMSCFD = 225,343.55 lbm/day which is equal to 9,389.3 lbm/hr. of NG.

Heat removal from the pretreated natural gas =
9,389.3 lbm/hr. x (346.2096 – (– 4.3712164)) = 3,291,708 Btu/hr.

Each lbm N$_2$ removes 27.664 Btu.

Lbm of N₂ circulated is 3,291,708 Btu/hr./27.664 Btu/lbm = 118,989 lbm/hr.
A plot of the feed gas to liquid storage is shown is shown on the PH diagram of Figure 3.34.

The chiller operates at -185 F, but the chilled NG is brought down to -175 F due to needed Δ T
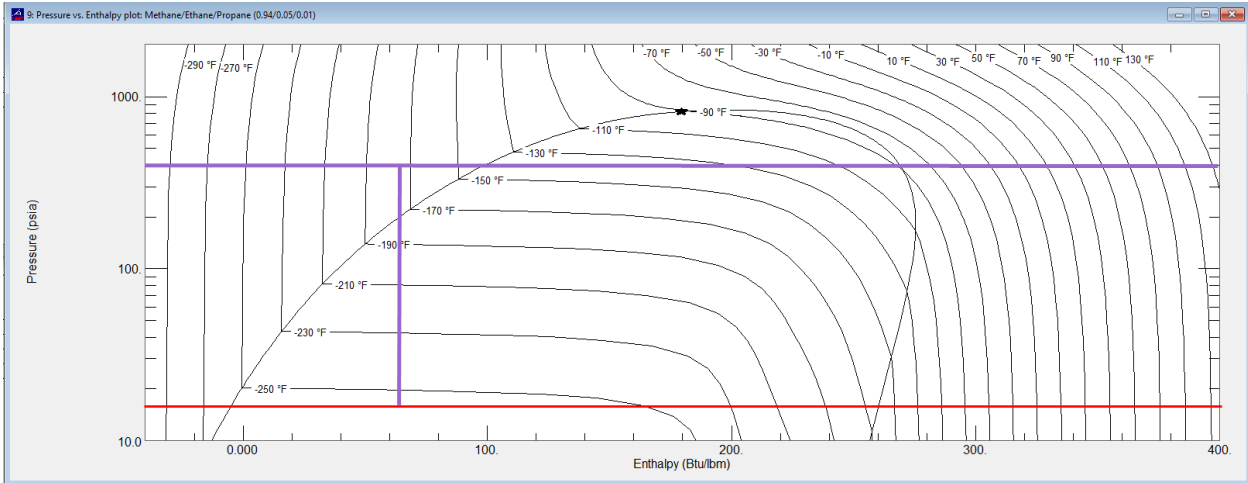
*Figure **Error! No text of specified style in document.**.37: Ph diagram showing the feed gas to liquid storage process*

Figure 3.35 below shows the composition of the LNG produced. As can be seen with a methane composition of 91% methane it is a suitable LNG, but the Btu content is higher than desirable. The FERC guideline for suitable interchangeable LNG is 1110 Btu/SCF and this LNG produced is 1089 Btu/SCF. Stored LNG weathers with time and becomes higher in Btu value. Thus, starting with a 1089 Btu value LNG is of some concern as it does not allow for long-term storage before such LNG would become too high in Btu value to be within the FERC interchangeability guidelines.

The LNG produced is acceptable but of high Btu value of 1089 Btu/scf

*Figure **Error! No text of specified style in document.**.38: LNG produced from feed gas at 400 psia and -175 F*

After such an analysis it is a good practice to place a control volume around the entire plant to assure that the heat balance is correct for the entire plant. It is a form of double checking the calculations. This is shown in Figure 3.36

Now that this process produces usable LNG compute the energy requirements

Motor work would be the (specific compressor work – the specific expander work) x nitrogen mass flow work

(142.29 – 42.074) Btu/lbm x **118,989** lbm/hr. = 11,924,602 Btu/hr.

1 Btu/hr. = 0.000393 hp, Thus, motor shaft output size needs to be > 4,686 hp. Nominal 5,000 hp motor when running at steady state.

Sounds reasonable.
Electric requirement
4,686 hp = (3596 kW/0.9) = 3,884 kW (0.9 used for motor efficiency)

NOTE: THIS SYSTEM DID PRODUCE LNG, HOWEVER, A LOT OF THE COOLING LEFT THE SYSTEM AS FLASH GAS. IN ORDER FOR IT TO PRODUCE 5 MMSCF OF LNG/DAY ALL OF THE REFRIGERATION FROM THE FLASH GAS WOULD NEED TO BE RECOVERED BY CHILLING FEED GAS.

The analysis can be checked by putting a control volume around the entire plant and confirming that the first law of thermodynamics confirms that the energy in is equal to the energy out. This is shown in Figure 3.36.
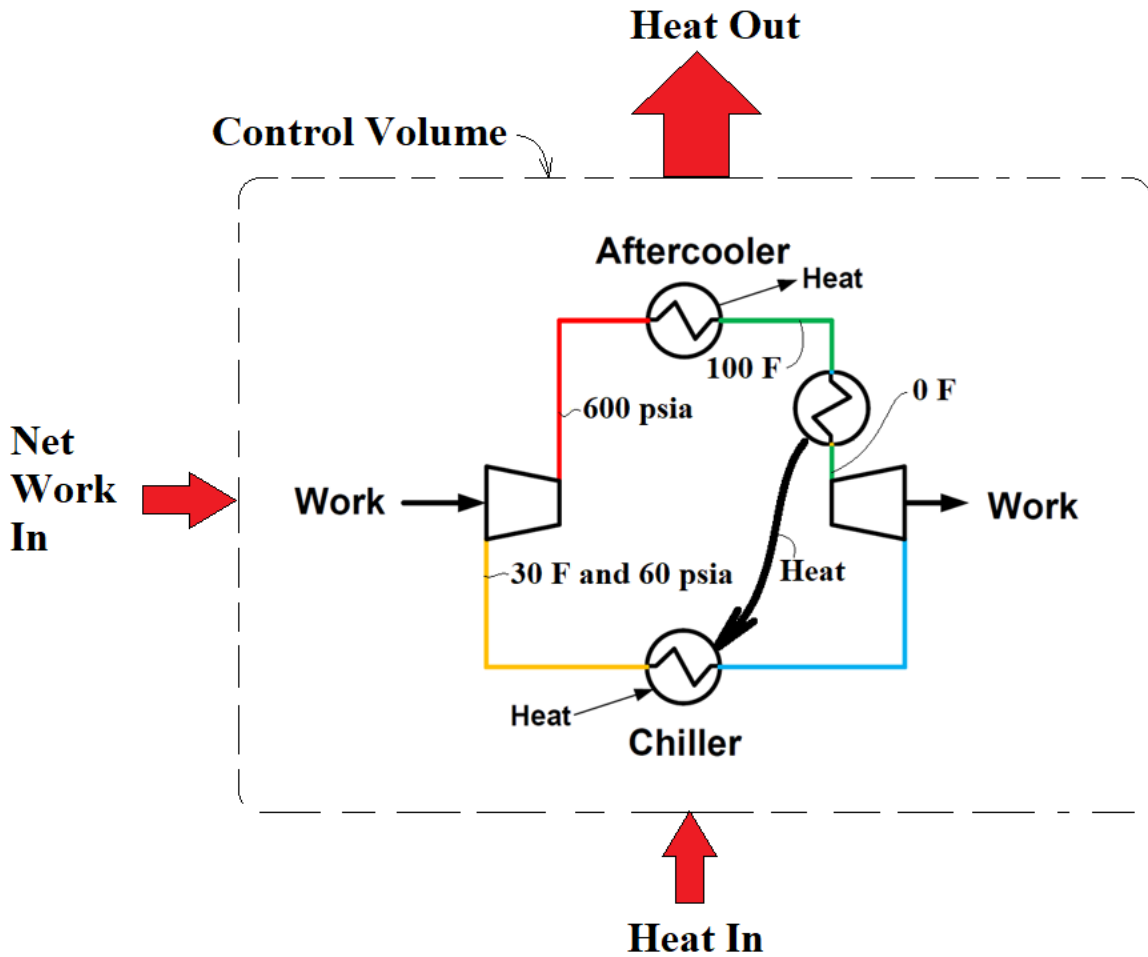
*Figure **Error! No text of specified style in document.**.39 Check on calculations by calculating heat balance of entire plant*

From the data shown in Figure 3.36 it can be seen that:

Work$_{in}$ + Heat$_{in}$ + = Work$_{out}$ + Heat$_{out}$

142.29 Btu/Lbm + 27.664 Btu/lbm = 42.074 Btu/Lbm +127.88 Btu/lbm

169.954 Btu/Lbm = 169.954 Btu/Lbm CHECK!

3.6 For the next analysis transfer more heat from the inlet of the expander to cause the expander outlet to be at -262F.

Start by drawing a control volume around the compressor as shown in figure 3.37.
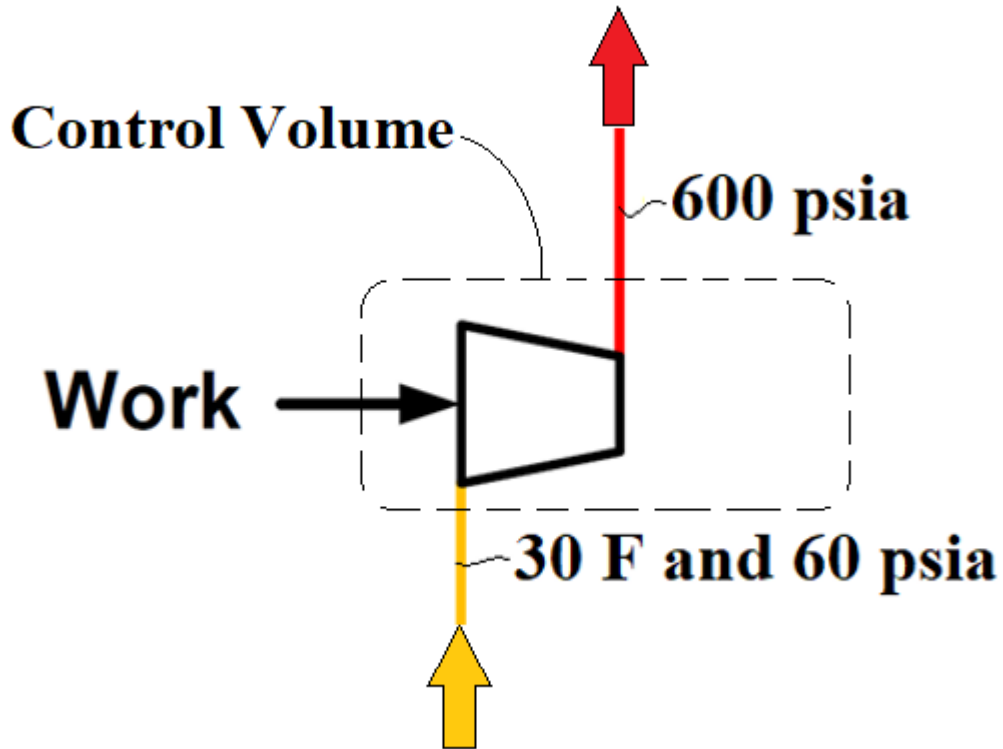
*Figure **Error! No text of specified style in document.**.40: Nitrogen compressor same as from Figure 3.23*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	30.000	60.000	0.32046	120.99	1.5104
2	485.26	600.00	1.6286	234.82	1.5104
3	595.38	600.00	1.4583	263.28	1.5389
4					

*Figure **Error! No text of specified style in document.**.41: REFPROP data associated with Figure 3.37*

In the real world this compression would be done with multiple compressors with intercooling to improve efficiency, which will be shown later in this learning.

From the data shown in Figure 3.37 it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

Adiabatic Isentropic first analysis $s_{in} = s_{out}$

Assume $\dot{m}$ = 1 lbm/hr. for simplicity $\dot{m}h_{in} + \dot{m}w_{in} = \dot{m}h_{out}$

$w_{in} = h_{out} - h_{in}$

Adiabatic Isentropic work = 234.82 – 120.99 = 113.83 Btu/Lbm

Actual work at 80% efficiency = 263.28 – 120.99 = 142.29 Btu/Lbm

Next Draw a control volume around the aftercooler.
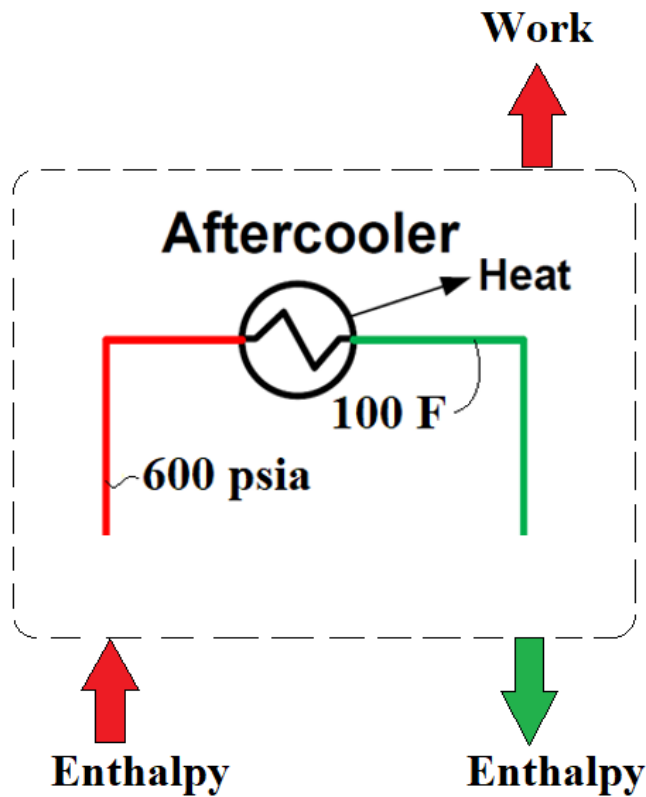
*Figure **Error! No text of specified style in document.**.42: Control volume around the Aftercooler*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	595.38	600.00	1.4583	263.28	1.5389
2	100.00	600.00	2.7999	135.40	1.3751
3					

Figure Error! No text of specified style in document..43: REFPROP data associated with Figure 3.39

From the data shown in Figure 3.39 it can be seen that:
SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}q_{out}$

$\dot{m} = 1$ lbm/hr.
$q_{out} = h_{in} - h_{out}$
$q_{out} = 263.28 - 135.40 = 127.88$ Btu/hr.

Normally it would be the next to draw a control volume around the heat exchanger upstream of the expander. However, this analysis cannot be performed until the analysis of the expander is performed to determine what the temperature at the inlet to the expander needs to be in order to get an outlet temperature of -262 F.
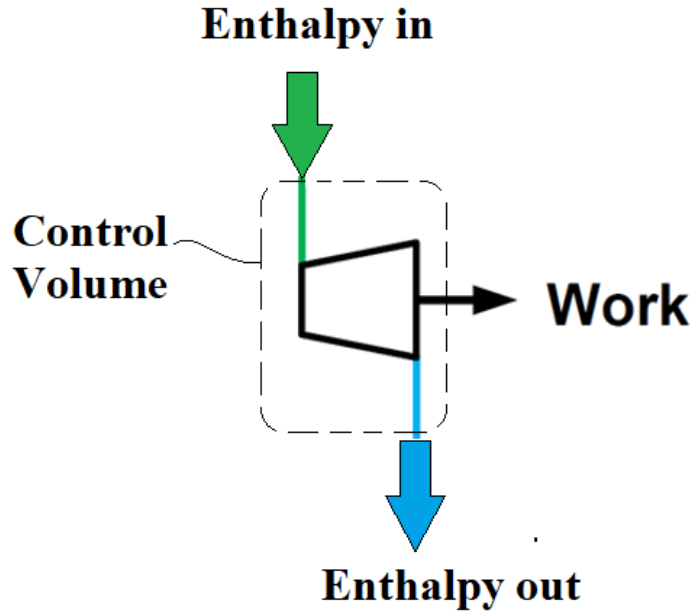
*Figure **Error! No text of specified style in document.**.44: Control volume around the expander with the expander outlet temperature of -262 F and 60 psia.*

*Figure **Error! No text of specified style in document.**.45: REFPROP data associated with Figure 3.41*

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

Adiabatic Isentropic first analysis $s_{in} = s_{out}$

Assume $\dot{m}$ = 1 lbm/hr. for simplicity $\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}w_{out}$

$w_{out} = h_{in} - h_{out}$

Adiabatic Isentropic Expansion $s_{in} = s_{out}$, Temp outlet = -287.42F

Real world Temp outlet = -262.16 F

Real world output work is (75.052-46.264) = 28.788 Btu/lbm.

Now that the inlet to the expander temperature has been determined to be -117 F an analysis of the heat exchange upstream of the expander can be made.
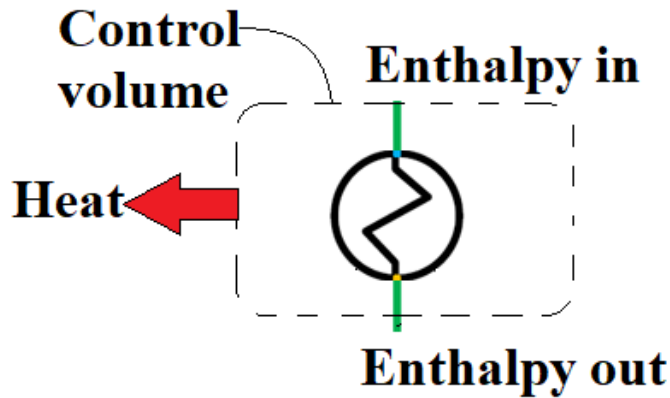
*Figure **Error! No text of specified style in document.**.46: Control volume placed around the heat exchanger before the expander*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	600.00	2.7999	135.40	1.3751
2	-117.00	600.00	5.0896	75.052	1.2378
3	-287.42	60.000	1.0102	39.067	1.2378
4	-262.16	60.000	0.84754	46.264	1.2768
5	30.000	60.000	0.32046	120.99	1.5104
6					

*Figure **Error! No text of specified style in document.**.47: REFPROP data associated with Figure 3.43*

From Figure 3.44 it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} = \dot{m}h_{out} + \dot{m}q_{out}$

$\dot{m}$ = 1 lbm/hr.

$q_{out} = h_{in} - h_{out}$

q_{out} = 135.4 – 75.052 = 60.348 Btu/hr.

Next Analyze the chiller which produces the LNG by placing a control volume around the chiller as shown in Figure 3.45.
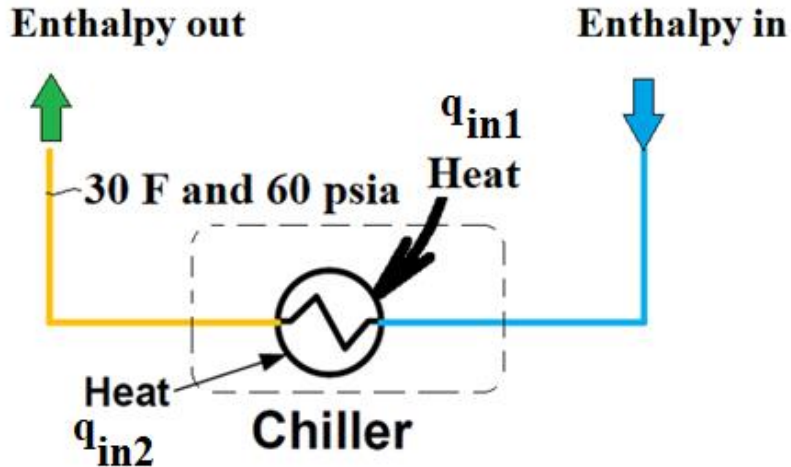
Figure Error! No text of specified style in document..48: Control volume placed around the chiller

Figure Error! No text of specified style in document..49: REFPROP data associated with Figure 3.45.

From Figure 3.46, it can be seen that:

SSSF Process, First Law of Thermodynamics– Energy in = Energy out

$\dot{m}h_{in} + \dot{m}q_{in1} + \dot{m}q_{in2} = \dot{m}h_{out}$

$\dot{m} = 1$ lbm/hr.

$q_{in2} = h_{out} - h_{in} - \dot{m}q_{in1}$

q$_{in2}$ = 120.99 – 46.264 – 60.348 = 14.378 Btu/hr.

At a temperature of – 262.16 F, this can liquefy pretreated natural gas.

Next, analyze the amount of heat removal needed to liquefy 5 MMSCFD

How much heat must be removed from 5 MMSCFD to liquefy it?

Assume pretreated feed gas is available at 40 F and 400 psia and assume storage temperature is saturated at 16.5 psia tank pressure.

8: Methane/Ethane/Propane: Saturation points (at equilibrium) (0.94/0.05/0.01)

	Liquid Phase Temperature (°F)	Vapor Phase Temperature (°F)	Liquid Phase Pressure (psia)	Vapor Phase Pressure (psia)	Liquid Phase Density (lbm/ft³)	Vapor Phase Density (lbm/ft³)	Liquid Phase Enthalpy (Btu/lbm)	Vapor Phase Enthalpy (Btu/lbm)	Liquid Phase Entropy (Btu/lbm-°R)	Vapor Phase Entropy (Btu/lbm-°R)
1	-254.7431	-254.7431	16.50000	16.50000	27.40162	0.1251812	-4.371164	221.4899	0.01281178	1.088110
2										

7: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	60.00000	14.73000	0.04506871	370.3947	1.515155
2	-256.8269	400.0000	27.64808	-4.371164	0.0001582132
3	40.00000	400.0000	1.367036	346.2096	1.089738
4					

*Figure **Error! No text of specified style in document.**.50: LNG produced from feed gas at 400 psia and -175 F*

Each SCF has a mass of 0.04506871 Lbm/SCF. Thus, 5 MMSCFD = 225,343.55 lbm/day, which is equal to 9,389.3 lbm/hr. Of NG.

Heat removal from the pretreated natural gas =

9,389.3 lbm/hr. x (346.2096 – (– 4.3712164)) = 3,291,708 Btu/hr.

Each lbm N$_2$ removes 27.664 Btu. Lbm of N$_2$ circulated is 3,291,708 Btu/hr./14.378 Btu/lbm = 228,941 lbm/hr.

Plot the liquefaction process on a PH diagram, as shown in Figure
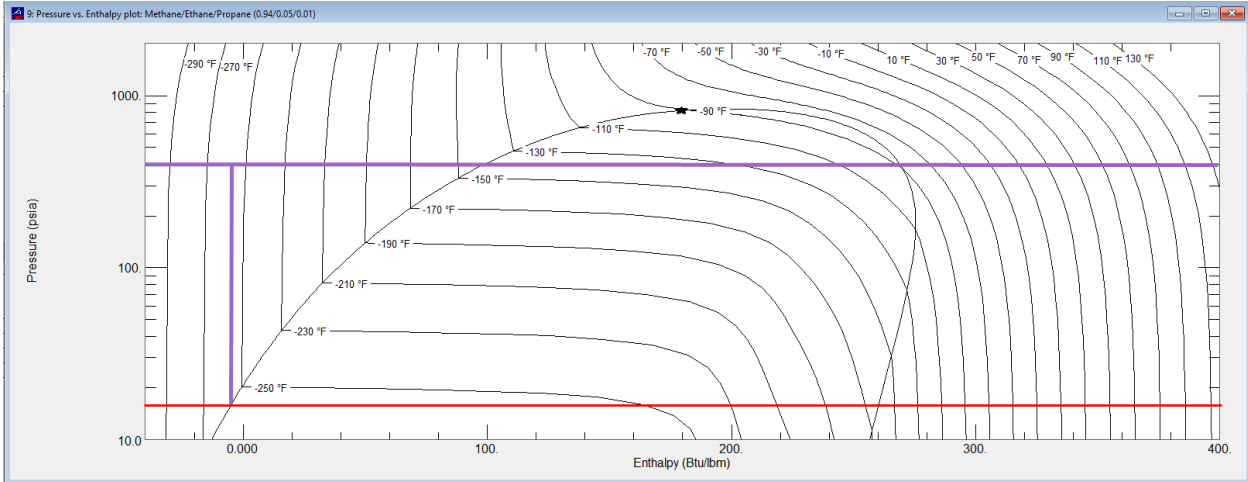
*Figure **Error! No text of specified style in document.**.51: Ph diagram of feed gas cooled, and JT valve reduced in pressure not producing any flash gas.*

NOTE: THIS SYSTEM, DUE TO ITS TEMPERATURE, DID NOT PRODUCE LNG WITHOUT PRODUCING FLASH GAS.

Thus, this closed N2 Expansion Liquefaction would work but could be better optimized.

What size motor would be needed to drive the nitrogen compressor?
Motor work would be the (specific compressor work – the specific expander work) x nitrogen mass flow work.

(142.29 – 42.074) Btu/lbm x **228,941** lbm/hr. = 2,294,355 Btu/hr.,
1 Btu/hr. = 0.000393 hp, Thus, the motor shaft output size needs to be 9,017 hp—nominal 10,000 hp motor when running at steady state.

Electric requirement
9,017 hp = (3596 kW/0.9) = 10,019 kW (0.9 used for motor efficiency)
Efficiency 10,019 kW/9389 lbm/hr.

Liquefaction efficiency is 1.07 kW hr./(lbm of LNG produced)

Today's higher efficiency plants have an efficiency of ~ 0.3 kW hr./(lbm of LNG produced). This above efficiency is very poor!

Next, perform an energy balance of the entire plant to help assure there were no mistakes in the individual computations.
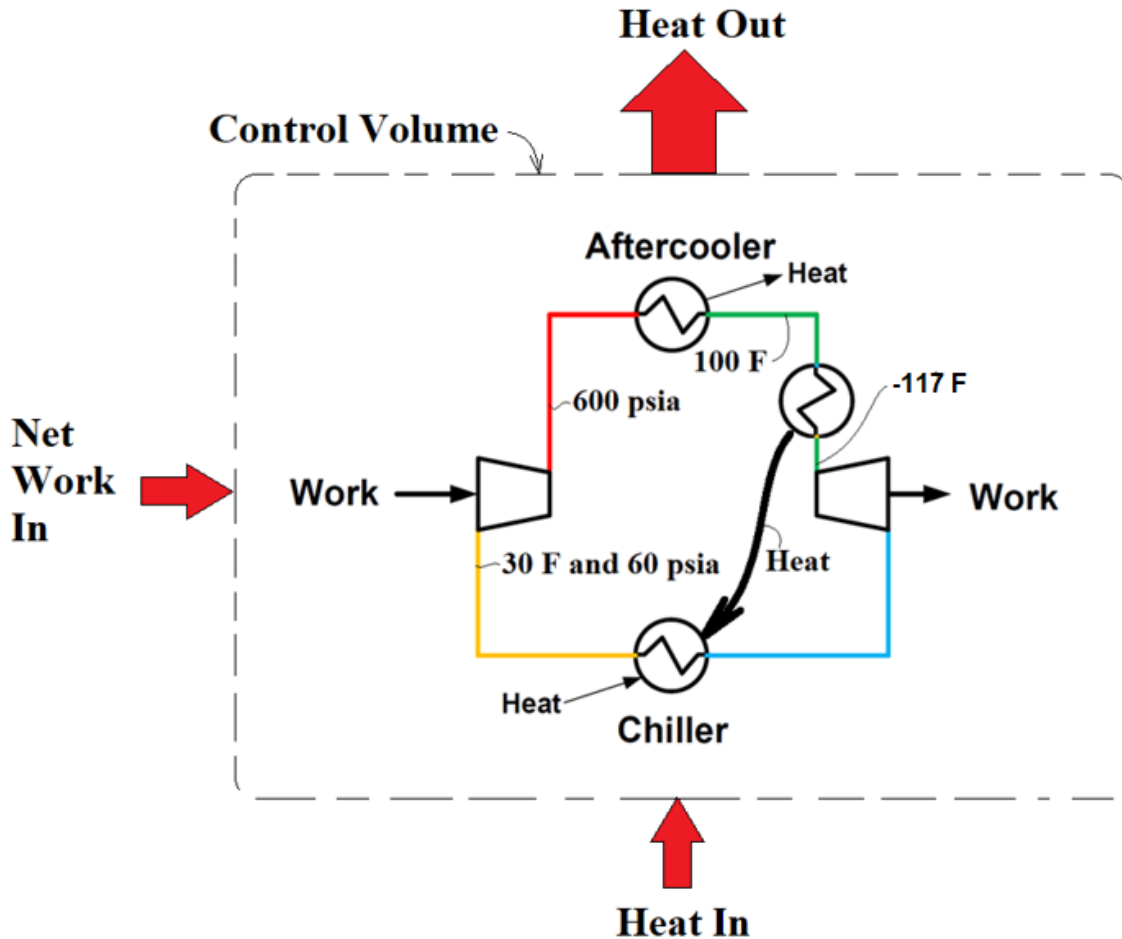
*Figure **Error! No text of specified style in document.**.52: Check on calculations by calculating the heat balance of entire plant*

The analysis can be checked by putting a control volume around the entire plant and confirming that the first law of thermodynamics confirms that the energy in is equal to the energy out. This is shown in Figure 3.49.

$Work_{in} + Heat_{in} = Work_{out +} Heat_{out}$

142.29 Btu/lbm + 14.378 Btu/lbm = 28.788Btu/Lbm + 127.88 Btu/lbm

156.668 Btu/Lbm = 156.668 Btu/LBM CHECK!

3.7 New analysis for better efficiency

Multiple stages of compression with intercooling increases system efficiency. (See Figure 3.50)

Multiple stages of expansion with reheating the expanding gas by extracting heat from the NG to be liquefied increases the system efficiency (see Figure 3.50)
The graphic shows capturing cooling from BOG. However, later, for simplicity, it is assumed that BOG has no cooling contribution.

Note that in Figure 3.50, the 100 F is highlighted with a star because it depends on the ambient temperature, which was assumed to be 90 F.

Note that when an expander is directly connected to a compressor so the work out of the expander drives the compressor, it is called a Compander.
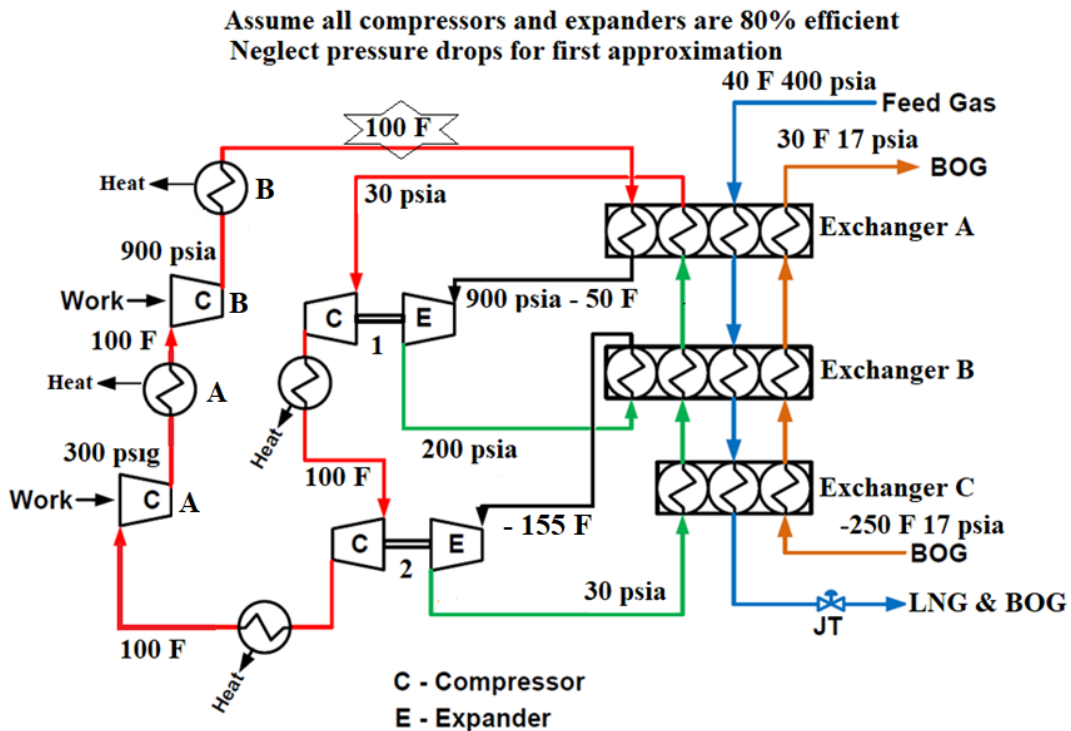
*Figure **Error! No text of specified style in document.**.53: More optimized nitrogen expansion process with multiple compressors with intercooling and two expanders.*

Next, analyze the B main compressor, as shown in Figure 3.51, and the B aftercooler, shown in Figure 3.53
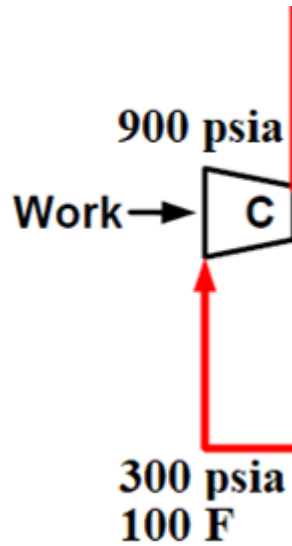
Figure Error! No text of specified style in document..54: B compressor is shown in Figure 3.50

Work into main compressor $w_{in} = h_{out} - h_{in} = 201.83 - 137.09 =$ **64.74 Btu/hr. Based on one lbm/hr. N$_2$ flowrate**

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	300.00	1.4008	137.09	1.4272
2	308.78	900.00	2.9873	188.88	1.4272
3	358.64	900.00	2.8006	201.83	1.4435
4	100.00	900.00	4.1893	133.81	1.3435
5					

Figure Error! No text of specified style in document..55 REFPROP data associated with Figures 3.51 and 3.53

Next, Analyze the heat exchanger downstream of the main compressor shown in Figure 5.53.

Heat removed in main compressor aftercooler (typically fin-fan type)
$q_{out} = h_{in} - h_{out}$
q_{out} = 201.83 – 133.81 = 68.02 Btu/lbm
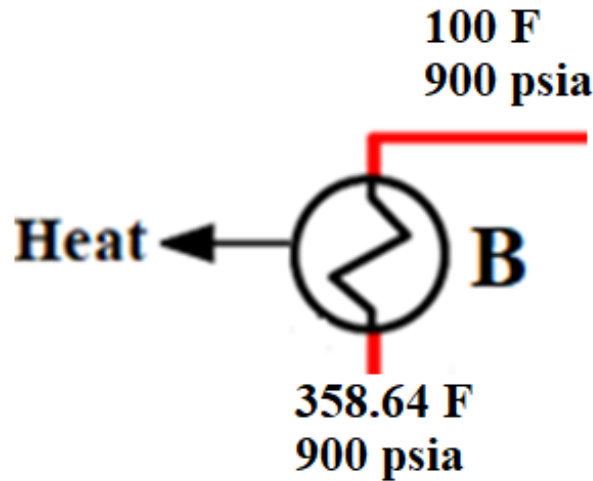
*Figure **Error! No text of specified style in document.**.56: B aftercooler after the main compressor*

Next, follow and analyze the flow stream into the next heat exchanger leaving the B aftercooler.
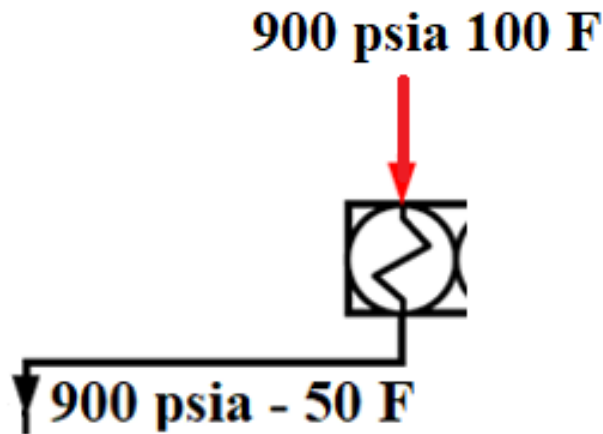
*Figure **Error! No text of specified style in document.**.57: Heat exchanger downstream of the B aftercooler*

*Figure **Error! No text of specified style in document.**.58: REFPROP data associated with Figure 3.54*

Heat exchange from nitrogen to cooling streams $q_{in} = h_{in} - h_{out}$

$q_{in} = (133.81 - 91.495) = 42.315$ Btu/lbm

Next, follow the flow and analyze the first pressure drop expander.
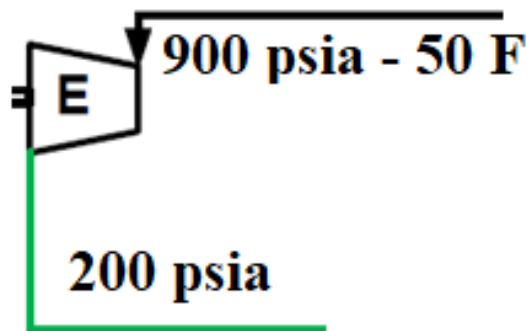
Figure **Error! No text of specified style in document.**.59: First expansion from 900 psia to 200 psia

*Figure **Error! No text of specified style in document.**.60: REFPROP data associated with Figure 3.56*

Work output **expander 1** = q_{out} = h_{in} − h_{out} = 91.495 – 65.695 **= 25.8 Btu/lbm**
The temperature of **expander one outlet = -175.99 F**
 the work output of the 2nd expander
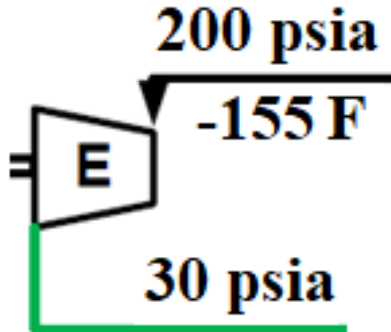
Figure Error! No text of specified style in document..61: Second expander from 200 psia to 30 psia

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	-155.00	200.00	1.8119	71.507	1.2987
2	-284.71	30.000	0.46875	41.687	1.2987
3	-262.09	30.000	0.40914	47.651	1.3308
4					

Figure Error! No text of specified style in document..62: REFPROP data

Work output **expander 2** = q_{out} = h_{in} − h_{out} = **71.507 – 47.651** = **23.856 Btu/lbm**
Temperature of expander 2 = –262.09 F

Next, Analyze the work output of the cluster of heat exchangers to the right of Figure 3.6 (which is the same as Figure 3.50. Note that the 100 F is highlighted with a star because it depends on the ambient temperature which was assumed to be 90 F.
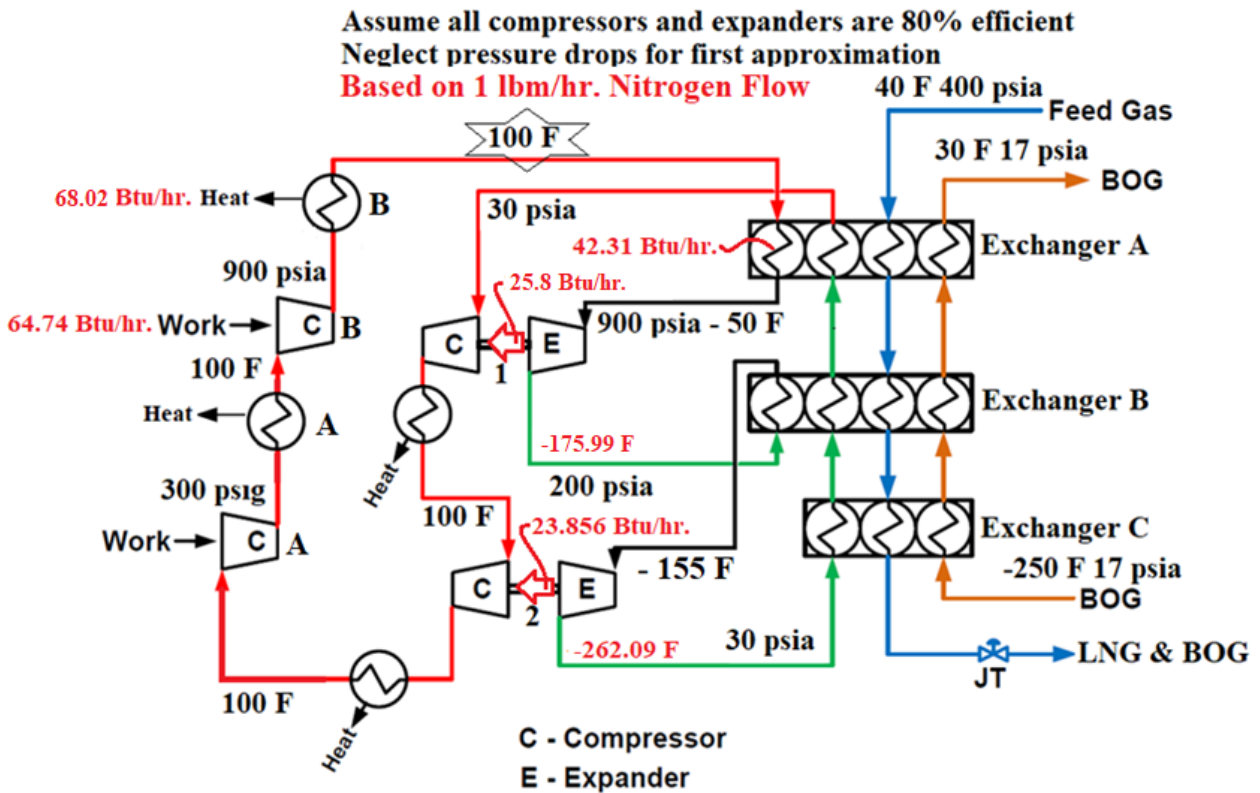
*Figure **Error! No text of specified style in document.**.63: Total plant process flow diagram (same as Figure 3.50)*

Note that when an expander is directly connected to a compressor so the work out of the expander drives the compressor, it is called a Compander.

As shown in Figure 3.61, Put a control volume around all the heat exchangers to the right of Figure 3.60. and number flows for analysis.
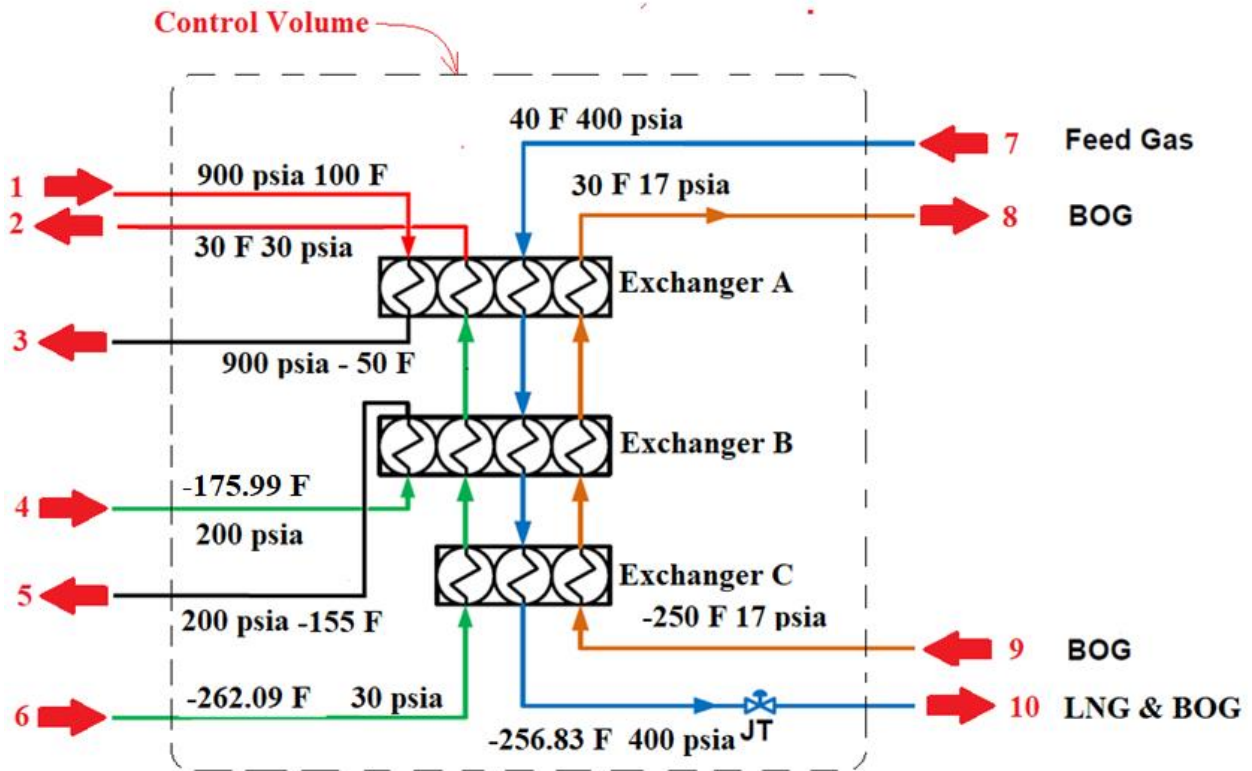
*Figure **Error! No text of specified style in document.**.64: Control volume placed around right cluster of heat exchangers*

The first law of thermodynamics is applied to the control volume shown in Figure 3.61 which results in the following equation.

First Law for SSSF Energy in = Energy out

$H_1 + H_4 + H_6 + H_7 + H_9 = H_2 + H_3 + H_5 + H_8 + H_{10}$

For the purposes of learning, to simplify the analysis the BOG circuit will be removed from Figure 3.61 which results in the drawing shown in Figure 3.62.
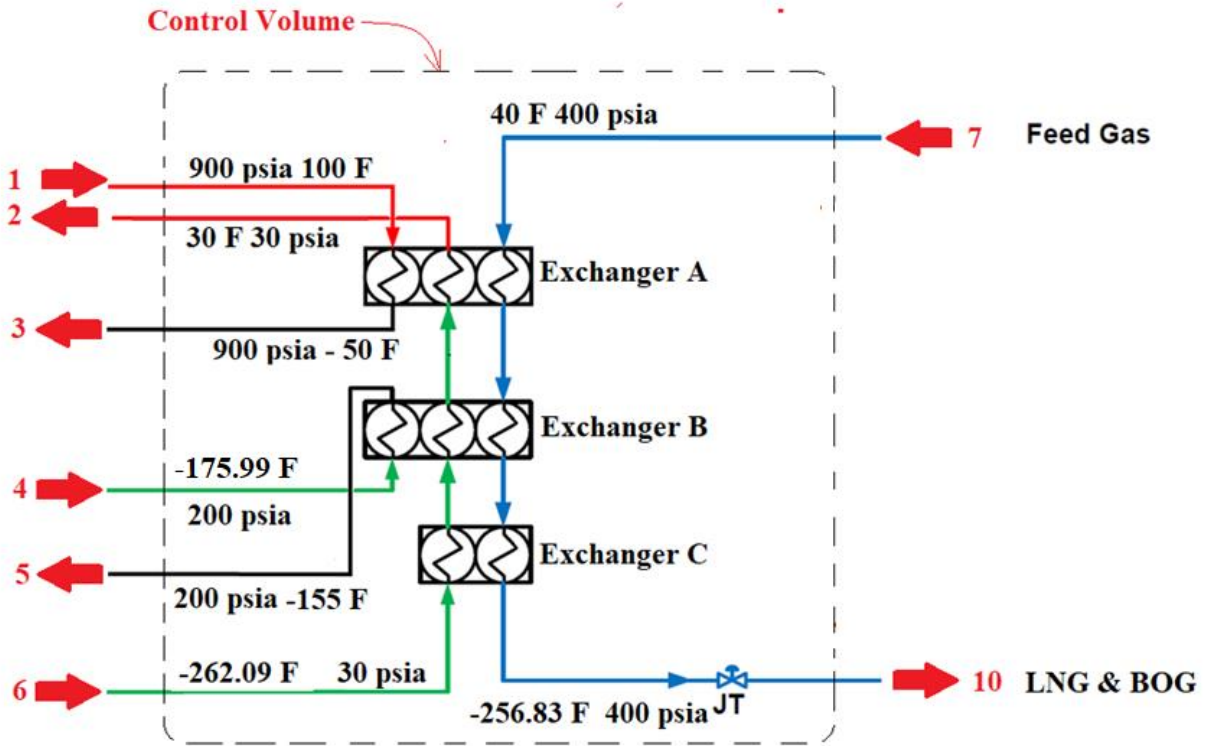
*Figure **Error! No text of specified style in document.**.65: Heat exchanger cluster with the BOG circuit removed*

The simplification results in the following equation when the first law of thermodynamics is applied.

First Law for SSSF Energy in = Energy out

$H_1 + H_4 + H_6 + H_7 = H_2 + H_3 + H_5 + H_{10}$

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	900.00	4.1893	133.81	1.3435
2	30.000	30.000	0.16008	121.23	1.5600
3	-50.000	900.00	6.1166	91.495	1.2552
4	-175.99	200.00	1.9782	65.695	1.2789
5	-155.00	200.00	1.8119	71.507	1.2987
6	-262.09	30.000	0.40913	47.652	1.3308
7					

Figure Error! No text of specified style in document..66: REFPROP data associated with Figure 3.62

Analysis of the heat exchanger cluster using the first law of thermodynamics results in the following:

First Law for SSSF Energy in = Energy out

For each lbm/hr. of nitrogen flow

$H_1 + H_4 + H_6 + H_7 = H_2 + H_3 + H_5 + H_{10}$

1 lbm/hr. h_1 + 1 lbm/hr. h_4 + 1 lbm/hr. h_6 + $\dot{m}h_7$ = 1 lbm/hr. h_2 + 1 lbm/hr. h_3 + 1 lbm/hr. h_5 + $\dot{m}h_{10}$

Next **For 1 lbm/hr. of Nitrogen flow, find out how much cooling is available for liquefaction of natural gas**

	Liquid Phase Temperature (°F)	Vapor Phase Temperature (°F)	Liquid Phase Pressure (psia)	Vapor Phase Pressure (psia)	Liquid Phase Density (lbm/ft³)	Vapor Phase Density (lbm/ft³)	Liquid Phase Enthalpy (Btu/lbm)	Vapor Phase Enthalpy (Btu/lbm)	Liquid Phase Entropy (Btu/lbm-°R)	Vapor Phase Entropy (Btu/lbm-°R)
1	-254.74	-254.74	16.500	16.500	27.402	0.12518	-4.3712	221.49	0.012812	1.0881
2										

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
10	-256.83	400.00	27.648	-4.3712	0.00015804
2	-254.74	16.500	27.402	-4.3712	0.012812
7	40.000	400.00	1.3670	346.21	1.0897

Figure Error! No text of specified style in document..67: REFPROP data needed to examine how much cooling is available for liquefaction of feed gas for each lbm/hr. of nitrogen is circulated.

1 lbm/hr. h_1 + 1 lbm/hr. h_4 + 1 lbm/hr. h_6 + $\dot{m}h_7$ = 1 lbm/hr. h_2 + 1 lbm/hr. h_3 + 1 lbm/hr. h_5 + $\dot{m}h_{10}$

133.81 Btu/hr. + 65.695 Btu/hr. + 47.652 Btu/hr. + $\dot{m}h_7$ = 121.23 Btu/hr. + 91.495 Btu/hr. + 71.507 Btu/hr. + $\dot{m}h_{10}$

-37.075 Btu/hr. = $\dot{m}$ (h_{10} - h_7) = $\dot{m}$(-4.371 – 346.21) = $\dot{m}$(- 350.581) Btu/lbm

$\dot{m}$ = 0.105753 lbm/hr. I.e., for every lbm of N_2 circulated, 0.105753 lbm of NG is liquefied

It was found out earlier that 5 MMSCFD is 9,389.3 lbm/hr.

Thus, the analysis shows that 88,785 lbm/hr. of N_2 needs to be circulated.
Plot the liquefaction process on a Ph diagram, as shown in Figure 3.65.
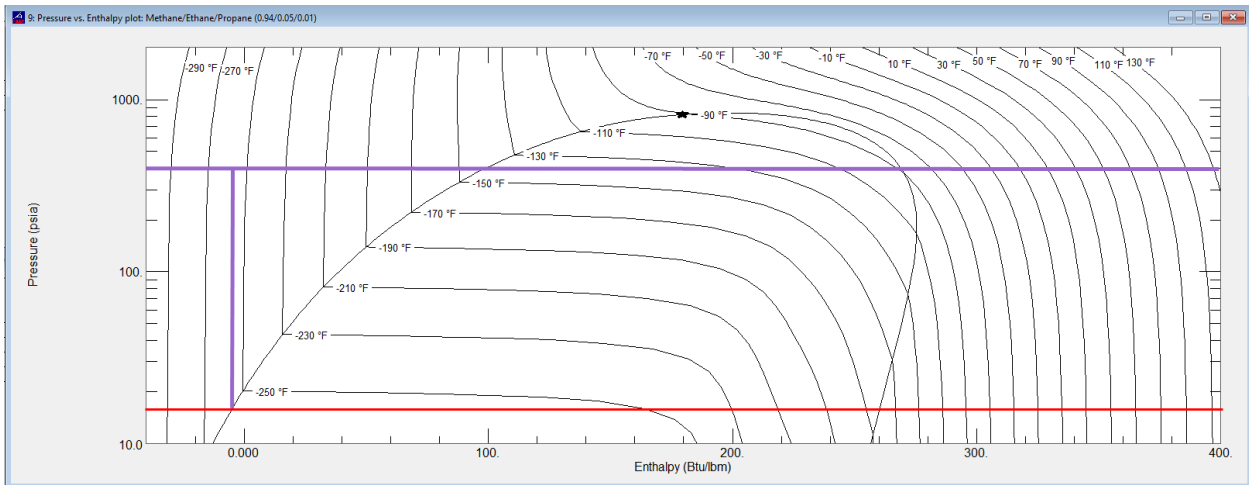
*Figure **Error! No text of specified style in document.**.68: Ph diagram showing the liquefaction process*

Next, analyze compressor 1 since it is now known what the quantity of work in is.

*Figure **Error! No text of specified style in document.**.69: Compressor 1 has the same work input as the work output of expander 1*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	30.000	30.000	0.16008	121.23	1.5600
2	113.29	51.967	0.23679	141.87	1.5600
3	133.95	51.967	0.22850	147.03	1.5688
4	100.00	51.967	0.24246	138.55	1.5541
5					

*Figure **Error! No text of specified style in document.**.70 REFPROP data associated with Figures 3.66 and 3.68*

Pressure at the outlet of compressor 1 is 51.967 psia, temperature is 133.95 F

Now analyze the heat exchanger downstream of compressor 1 aftercooler.
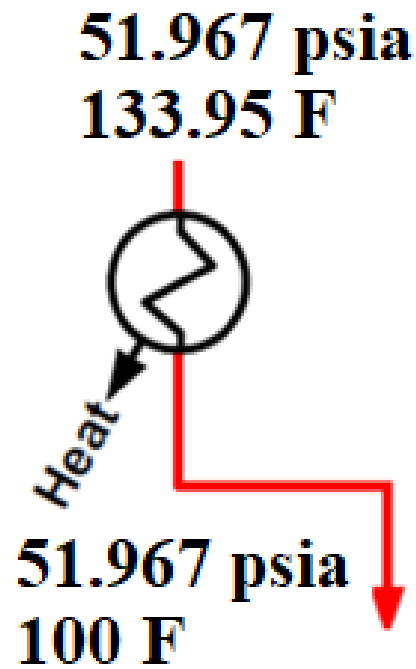
*Figure **Error! No text of specified style in document.**.71: Aftercooler of compressor 1*

Using the data from Figure 3.67, it is found that:
Compressor 1 aftercooler heat output $h_{in} = q_{out} + h_{out}$

For 1 lbm/hr.
$q_{out} = h_{in} - h_{out} = (147.03 - 138.55) = $ **8.48 Btu/hr.**

Next Analyze Compressor 2 and its aftercooler.

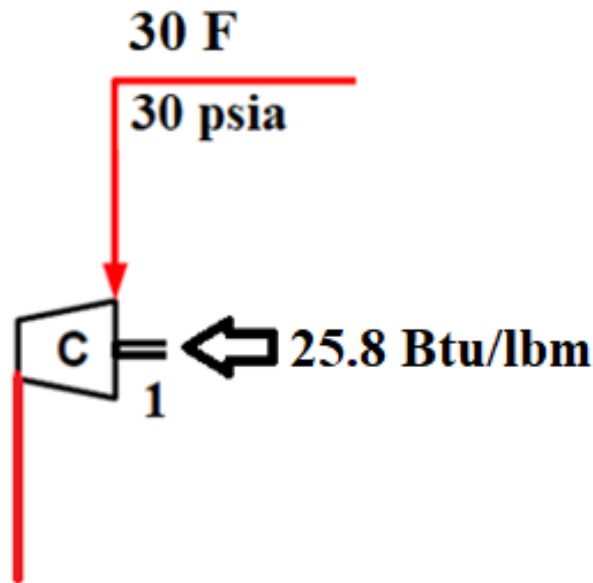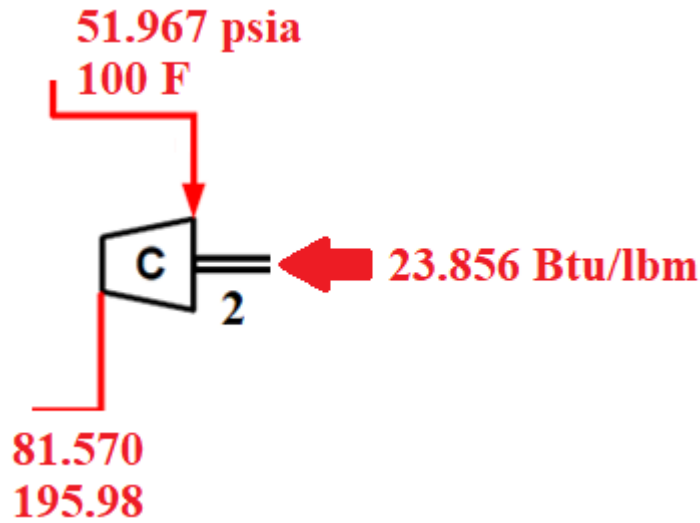*Figure **Error! No text of specified style in document.**.72: Compressor 2 has the same work input as the work output of expander 2*

Pressure at the outlet of compressor 2 is 81.570 psia, temperature is 195.98 F

*Figure **Error! No text of specified style in document.**.73: REFPROP data associated with Figures 3.69 and 3.71*

Refer to Figure 3.71 and 3.70 and it can be seen that:

Compressor 2 aftercooler heat output $h_{in} = q_{out} + h_{out}$

For 1 lbm/hr.

$q_{out} = h_{in} − h_{out} = (162.41 − 138.37) = 24.04$ Btu/hr.

*Figure **Error! No text of specified style in document.**.74: Aftercooler of compressor 2*

Next Analyze Compressor 2 and its aftercooler
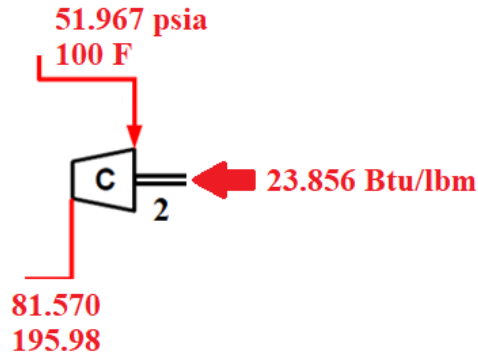
*Figure **Error! No text of specified style in document.**.75: Compressor 2*

Pressure at the outlet of compressor 2 is 81.570 psia, temperature is 195.98 F

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	51.967	0.24246	138.55	1.5541
2	176.92	81.570	0.33421	157.63	1.5541
3	195.98	81.570	0.32443	162.41	1.5615
4	100.00	81.570	0.38064	138.37	1.5219
5					

*Figure **Error! No text of specified style in document.**.76: REFPROP data associated with Figures 3.72 and 3.74*

*Figure **Error! No text of specified style in document.**.77: Aftercooler downstream of compressor 2*

Compressor 2 aftercooler heat output $h_{in} = q_{out} + h_{out}$
For 1 lbm/hr.
$$q_{out} = h_{in} - h_{out} = (162.41 - 138.37) = 24.04 \text{ Btu/hr.}$$
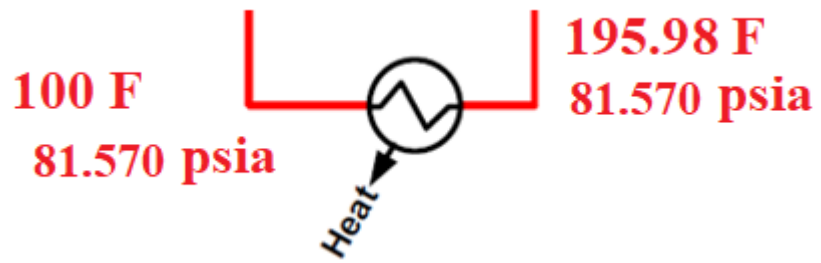
Next Analyze Compressor A and its downstream aftercooler.
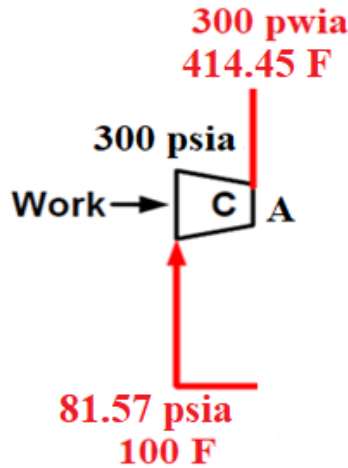
*Figure **Error! No text of specified style in document.**.78: Compressor A*

Work into main A compressor $w_{in} = h_{out} - h_{in} =$ **216.98 – 138.37 = 78.61 Btu/hr. Based on 1 lbm/hr. N$_2$ flowrate**

4: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	81.570	0.38064	138.37	1.5219
2	352.56	300.00	0.95677	201.26	1.5219
3	414.45	300.00	0.88851	216.98	1.5406
4	100.00	300.00	1.4008	137.09	1.4272
5					

*Figure **Error! No text of specified style in document.**.79: REFPROP data associated with Figures 3.75 and 3.77*

Heat removed in main compressor A aftercooler

$q_{out} = h_{in} - h_{out}$

q$_{out}$ = 216.98 – 137.09 = 79.89 Btu/lbm

*Figure **Error! No text of specified style in document.**.80: Aftercooler of compressor A*

Heat removed in main compressor A aftercooler

$q_{out} = h_{in} - h_{out}$

$q_{out} = 216.98 - 137.09 = 79.89$ Btu/lbm

*Figure **Error! No text of specified style in document.**.81: More optimized nitrogen expansion process with multiple compressors with intercooling and two expanders process with data values populated*

Note that when an expander is directly connected to a compressor so the work out of the expander drives the compressor, it is called a Compander.

The analysis can be checked by putting a control volume around the entire plant and confirming that the first law of thermodynamics confirms that the energy in is equal to the energy out. This is shown in Figure 3.49.
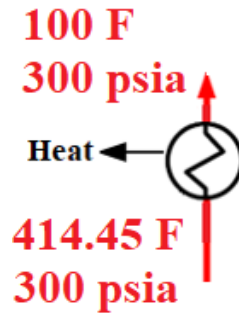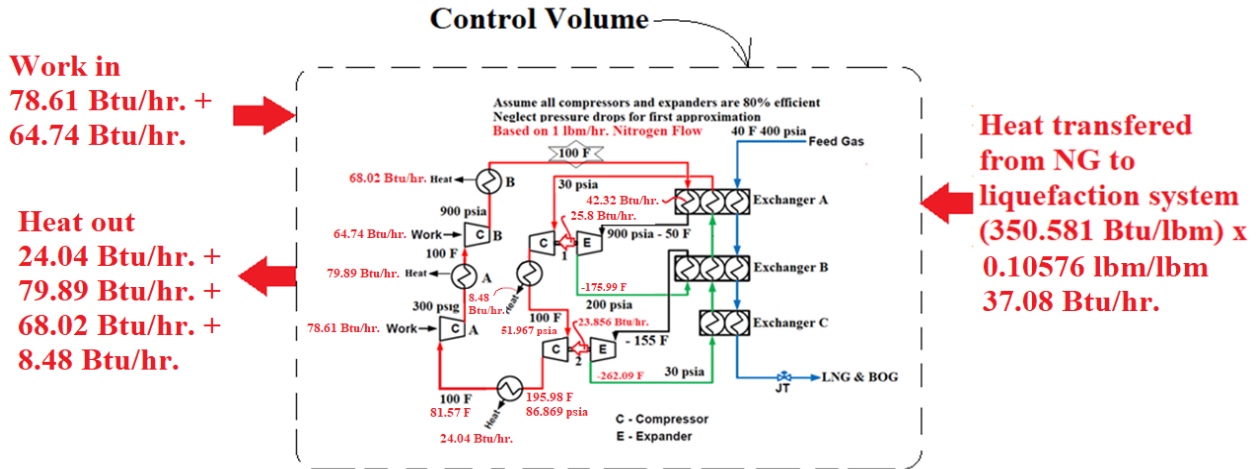

*Figure **Error! No text of specified style in document.**.82: Heat and work balance for entire plant*

First Law SSSF Energy in = energy out   Work in + heat in = heat out

(78.61 + 64.74) Btu/hr. + 37.08 Btu/hr. = (24.04 + 79.89 + 68.02 + 8.48) Btu/hr.

180.43 = 180.43   CHECK

Thus, this closed N_2 Expansion Liquefaction is much better optimized than the earlier configuration.

What size motor would be needed to drive the nitrogen compressor?

Motor work would be the (motor driven specific compressor work) x nitrogen mass flow work = **(74.03 + 64.74) Btu/lbm x 88,785 SLIDE 49 lbm/hr. = 12,320,694 Btu/hr. Note: nitrogen circulation is 39% of the previous system analyzed.**

1 Btu/hr. = 0.000393 hp, Thus, motor shaft output size needs to be > 4,842 hp. Nominal 5,000 hp motor when running at steady state.

Sounds reasonable.

4,842 hp = (3,611 kW/0.9)/9389.3 lbm$_{NG}$/hr. (0.9 used for motor efficiency) = 0.4273 kW hr./lbm of LNG produced.

Liquefaction efficiency is 0.427 kW hr./(lbm of LNG produced) (compared to 1.07 kW hr./lbm in non-optimized analysis)

Today's higher efficiency plants have an efficiency of ~ 0.3 kW hr./lbm of LNG produced

For this 0.3kW hr./lbm of LNG produced it is not known at what ambient temperature this was computed, nor is it known if this is with BOG cooling recovery. Further it is not known what feed gas conditions were used. **Thus, for this analysis the results are considered reasonable!**
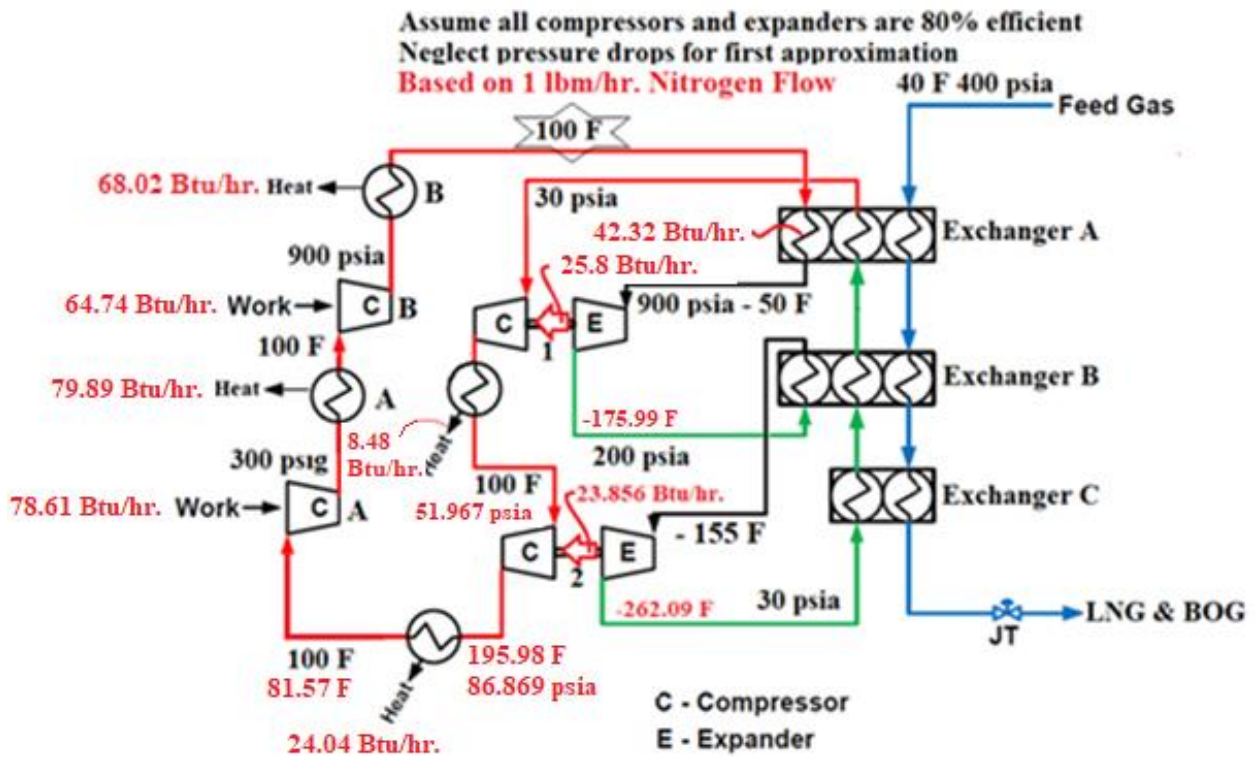
Next Analyze Exchangers A, B and C
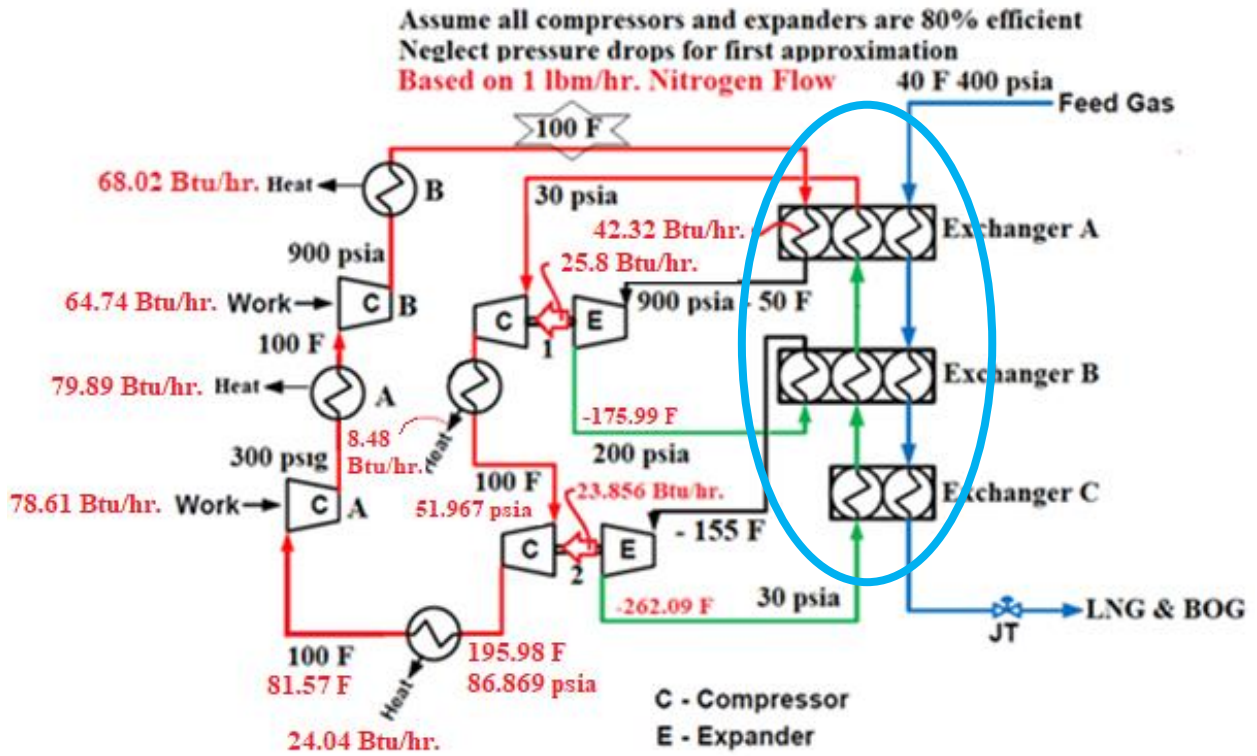
*Figure **Error! No text of specified style in document.**.83: More optimized nitrogen expansion process with multiple compressors with intercooling and two expanders process with data values populated*

Next Analyze the A exchanger
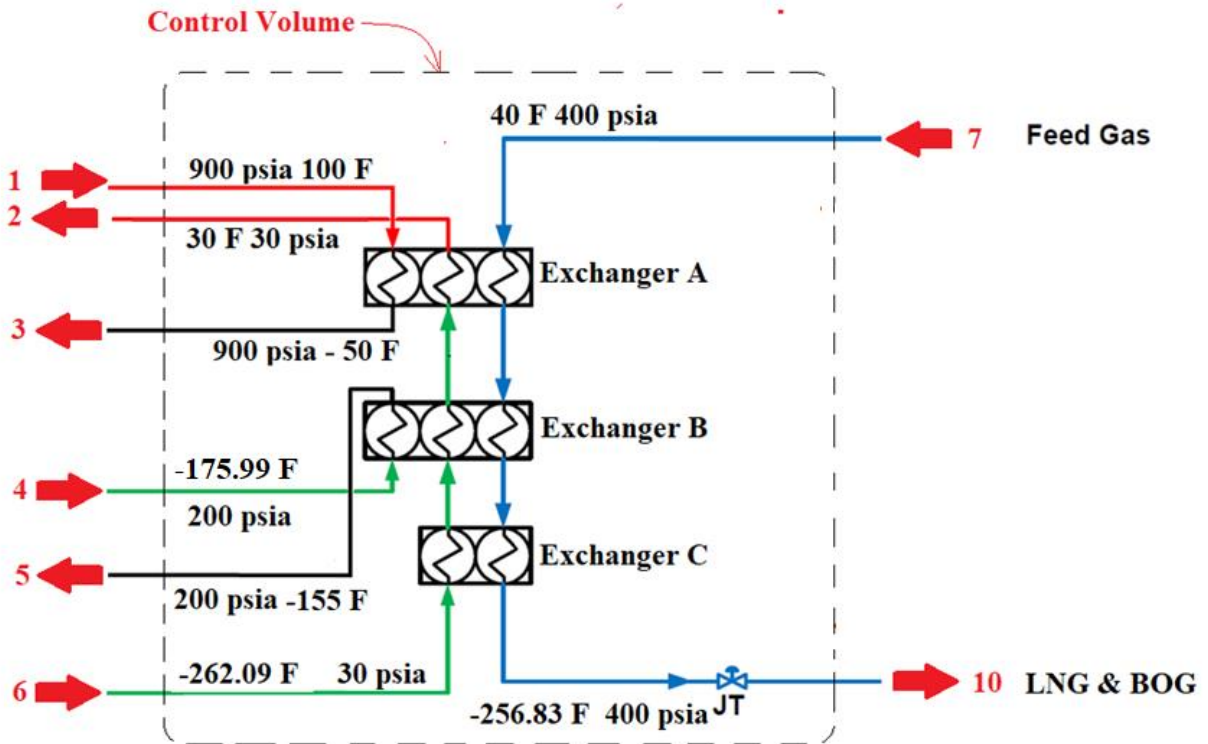
*Figure **Error! No text of specified style in document.**.84: Graphic for analyzing the A, B and C exchangers*
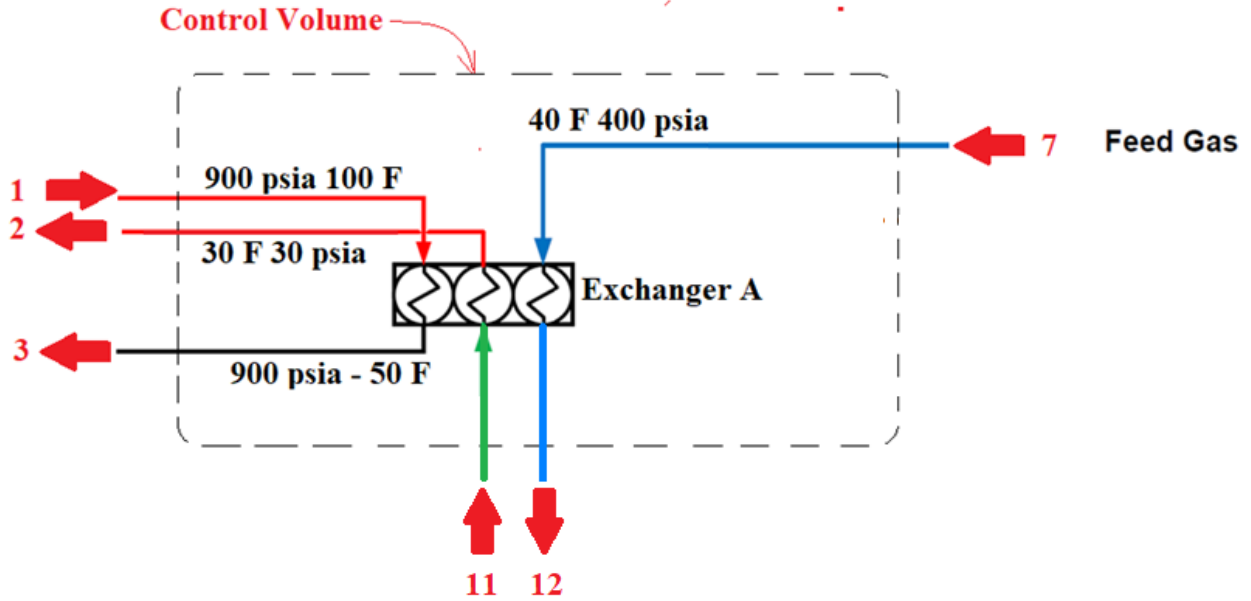
*Figure **Error! No text of specified style in document.**.85: Graphic for analyzing the An Exchanger*

Perform analysis for each Lbm/hr. of nitrogen circulated.

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	900.00	4.1893	133.81	1.3435
2	30.000	30.000	0.16008	121.23	1.5600
3	-50.000	900.00	6.1166	91.495	1.2552
4	-175.99	200.00	1.9782	65.695	1.2789
5	-155.00	200.00	1.8119	71.507	1.2987
6	-262.09	30.000	0.40913	47.652	1.3308
7					

*Figure **Error! No text of specified style in document.**.86: REFPROP data used along with REFPROP data of Figure 3.84 and graphic of Figure 3.82 to analyze A exchanger*

	Liquid Phase Temperature (°F)	Vapor Phase Temperature (°F)	Liquid Phase Pressure (psia)	Vapor Phase Pressure (psia)	Liquid Phase Density (lbm/ft³)	Vapor Phase Density (lbm/ft³)	Liquid Phase Enthalpy (Btu/lbm)	Vapor Phase Enthalpy (Btu/lbm)	Liquid Phase Entropy (Btu/lbm-°R)	Vapor Phase Entropy (Btu/lbm-°R)
1	-254.74	-254.74	16.500	16.500	27.402	0.12518	-4.3712	221.49	0.012812	1.0881
2										

1: Methane/Ethane/Propane: Saturation points (at equilibrium) (0.94/0.05/0.01)

2: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
10	-256.83	400.00	27.648	-4.3712	0.00015804
2	-254.74	16.500	27.402	-4.3712	0.012812
7	40.000	400.00	1.3670	346.21	1.0897

*Figure **Error! No text of specified style in document.**.87: REFPROP data used along with REFPROP data of Figure 3.83 and graphic of Figure 3.82 to analyze A exchanger*

From heat balance, the following can be determined.

$\dot{m}_{N2}$ = 1 lbm/hr.

$\dot{m}_{Feed\ Gas}$ = 0.105753 lbm/hr.

Now determine temperature at station 12.

Energy in = Energy out

1 lbm/hr. (h_1) + 1 lbm/hr. (h_{11}) + 0.10576 lbm/hr. (h_7) = 1 lbm/hr. (h_2) + 1 lbm/hr. (h_3) + 0.10576 lbm/hr.(h_{12})

Select a reasonable design temperature for flow 11. Needs to be colder than   -50 and cold enough to bring point 3 down to -50 F. Select -155F (same as "5")

133.81 Btu/hr. + 74.939 Btu/hr. + 0.10576 lbm/hr. (346.21) = 121.23 Btu/hr. + 91.495 + 0.10576 lbm/hr. (h_{12})

0.10576 lbm/hr. (h_{12}) = 32.639Btu/lbm/(0.10576 lbm/hr.) = 308.614

Temp of flow 12 = -25.573 F

Next analyze the B exchanger.
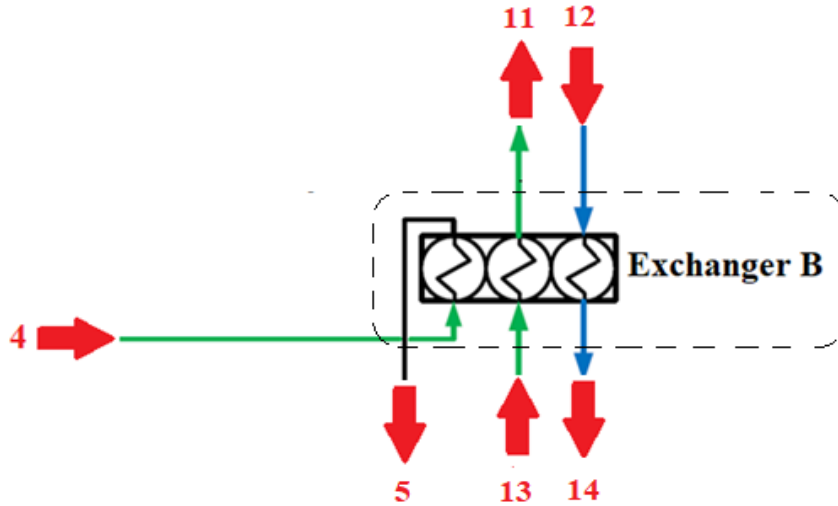
*Figure **Error! No text of specified style in document.**.88: Graphic for analyzing exchanger B*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
11	-155.00	30.000	0.25912	74.939	1.4413
2					

*Figure **Error! No text of specified style in document.**.89: REFPROP data used to analyze the B exchanger*

1: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
12	-25.573	400.00	1.6566	308.61	1.0090
2					

*Figure **Error! No text of specified style in document.**.90: REFPROP data used to analyze the B exchanger*

*Figure **Error! No text of specified style in document.**.91: REFPROP data used to analyze the B exchanger*

Perform energy balance around the B exchanger.

Energy in = Energy out

1 lbm/hr. (h₄) + 0.10576 (h₁₂) + 1 lbm/hr. (h₁₃) = 1 lbm/hr. (h₅) + 1 lbm/hr. (h₁₁) + 0.10576 lbm/hr.(h₁₄)

Select -175 F for temperature of flow 13

1 lbm/hr. (65.695) + 0.105753 (308.61) + 1 lbm/hr. (69.898) = 1 lbm/hr. (71.507) + 1 lbm/hr. (74.939) + 0.105753 lbm/hr.(h₁₄)

h₁₄ = 21.783/0.105753 = 205.984 Btu/lbm

T₁₄ = -128.53 F

*Figure **Error! No text of specified style in document.**.92: REFPROP data used to analyze the B exchanger*

1: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
12	-25.579	400.00	1.6567	308.61	1.0090
14	-128.53	400.00	3.3491	205.98	0.72867
3					

*Figure **Error! No text of specified style in document.**.93: REFPROP data used to analyze the B exchanger*

Next perform the analysis for Exchanger C.
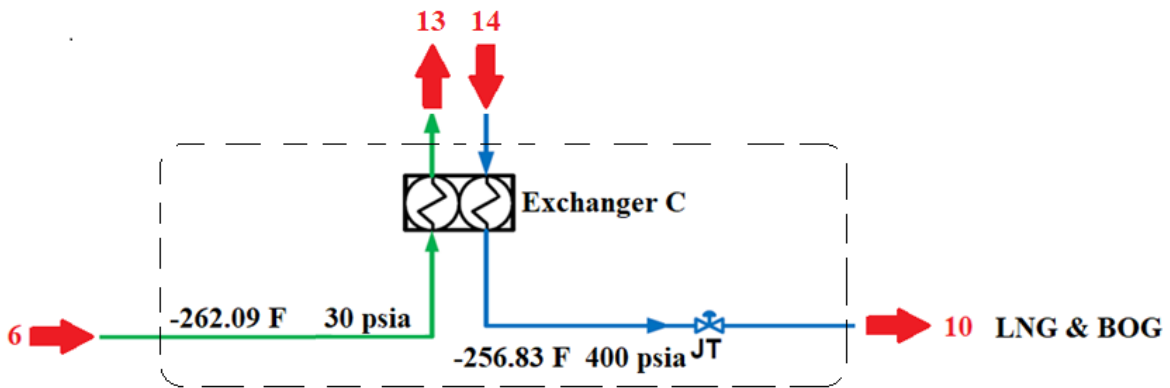
*Figure **Error! No text of specified style in document.**.94: Graphic of the C exchanger*

1: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
1	100.00	900.00	4.1893	133.81	1.3435
2	30.000	30.000	0.16008	121.23	1.5600
3	-50.000	900.00	6.1166	91.495	1.2552
4	-175.99	200.00	1.9782	65.695	1.2789
5	-155.00	200.00	1.8119	71.507	1.2987
6	-262.09	30.000	0.40913	47.652	1.3308
7					

*Figure **Error! No text of specified style in document.**.95: REFPROP data used to analyze the C exchanger*

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
10	-256.83	400.00	27.648	-4.3712	0.00015804
2	-254.74	16.500	27.402	-4.3712	0.012812
7	40.000	400.00	1.3670	346.21	1.0897

*Figure **Error! No text of specified style in document.**.96: REFPROP data used to analyze the C exchanger*

1: Nitrogen: Specified state points

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
13	-175.00	30.000	0.27790	69.898	1.4241
2					

*Figure **Error! No text of specified style in document.**.97: REFPROP data used to analyze the C exchanger*

1: Methane/Ethane/Propane: Specified state points (0.94/0.05/0.01)

	Temperature (°F)	Pressure (psia)	Density (lbm/ft³)	Enthalpy (Btu/lbm)	Entropy (Btu/lbm-°R)
12	-25.579	400.00	1.6567	308.61	1.0090
14	-128.53	400.00	3.3491	205.98	0.72867
3					

*Figure **Error! No text of specified style in document.**.98: REFPROP data used to analyze the C exchanger*

Perform an energy balance around the entire exchanger.

Energy in = Energy out

1 lbm/hr. (h_6) + 0.105753 (h_{14}) = 1 lbm/hr. (h_{13}) + 0.105753 lbm/hr.(h_{10})

1 lbm/hr. (47.652 Btu/lbm) + 0.105753 (205.98 Btu/lbm) = + 1 lbm/hr. (69.898 Btu/lbm) + 0.105753 lbm/hr.(-4.3712)

69.435 Btu/hr. = 69.436 Btu/hr. Δ= <0.001 = <0.001 % This is round off error. Thus, Calculations CHECK
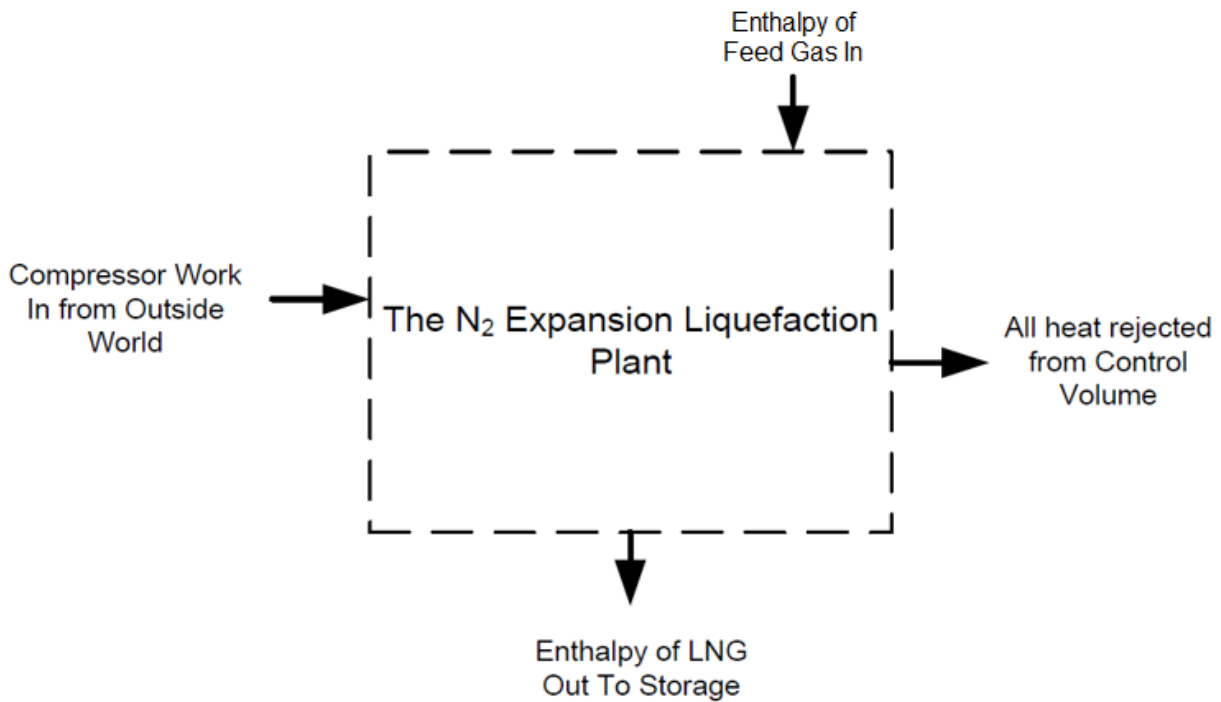

Figure **Error! No text of specified style in document.**.99: Graphic of a control volume around the entire plant

Sign Convention Energy in is positive +
Sign Convention Energy out is negative –

For the basis of analysis use 1 lbm of N_2/hr. circulation producing 0.10576 lbm of subcooled LNG/hr.
The sum of the energy in and out needs to be zero!

Below is per lbm of LNG per hour.
Compressor work in from outside word   =   + 78.61 Btu + 64.74 Btu **=**
+143.35 Btu
Enthalpy of 1 lbm of feed-gas   = + 346.21 Btu x (0.10576) =
+ 36.62 Btu
Sum of all heat rejected to the outside world = – 79.89(A)Btu   – 68.02(B)Btu – 8.48 (1st compander)Btu – 24.04 (2nd compander)Btu = **– 180.43 Btu**

Enthalpy of subcooled LNG sent to storage   = –(– 4.3712 Btu) x (0.10576) =

+ 0.46 Btu

Sum of energy in and out = 180.43 – 180.43 = 0 Thus, Calculations   CHECK

Shortcomings of analyses
- Cooling available from BOG not considered -   approximately 1 MMSCF of BOG available for heat extraction
- Pressure drops were not accounted for
- Adding additional compression stages with intercooling would improve efficiency
- Optimization of pressure selection was not performed
- We could have selected more efficient expanders and compressors (82% instead of 80% efficient)
- Equipment standard sizes were not analyzed
- Only analysis made was for 90 F Day and 40 F gas at 400 psia
- Efficient pre-chilling with propane JT system not considered

3.8 Open expansion liquefaction system

The open expansion system was analyzed in detail in section 1 of this three-part thermodynamic learning series. Thus, for this part, only the drawings shown in part 1 of this learning series will be shown below. For more detailed analysis see "What Every Energy Engineer Needs to Know about Thermodynamics and Liquefaction Systems Part (1 of 3)".
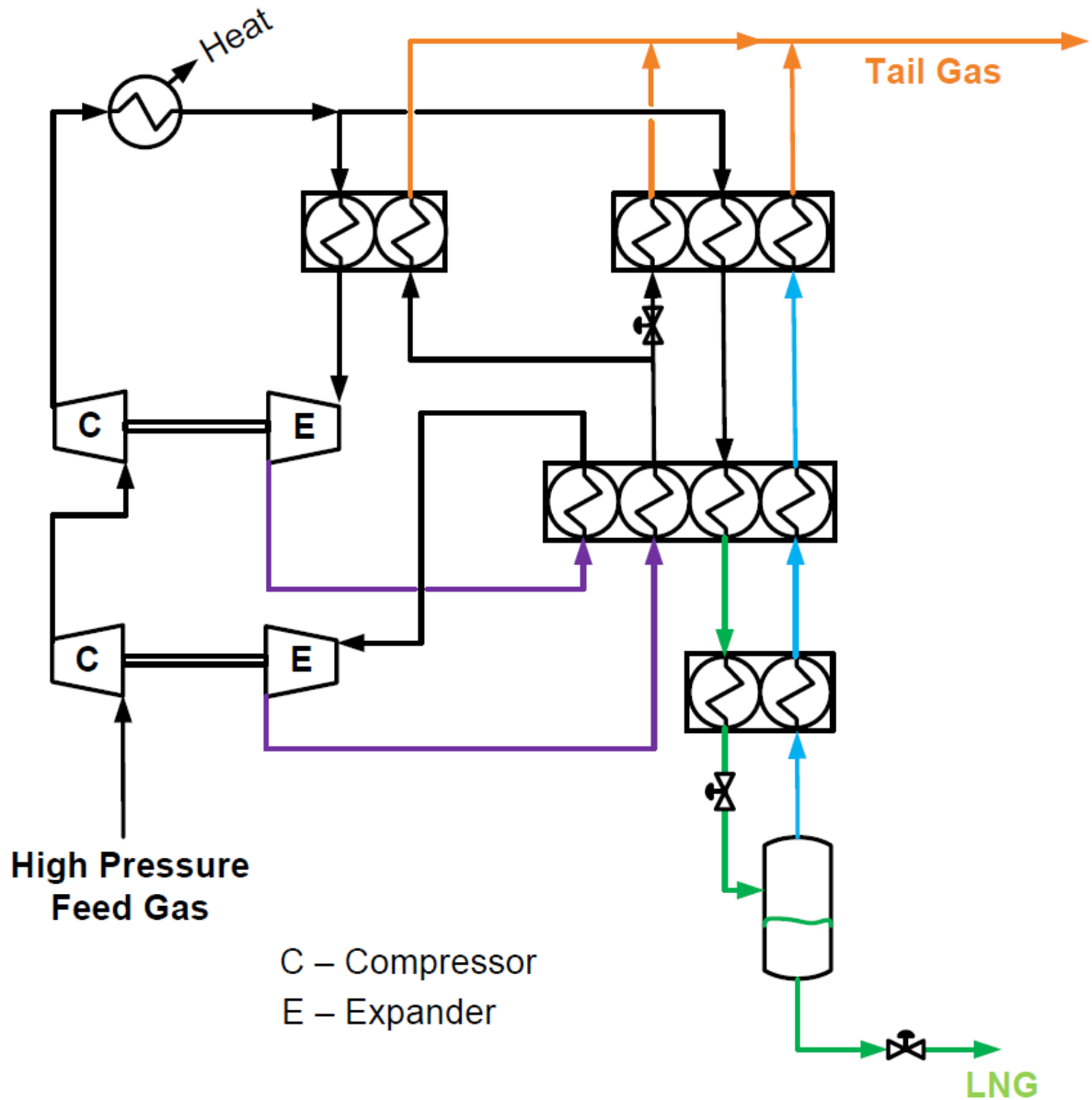

*Figure **Error! No text of specified style in document.**.100: Graphic of an Open Expansion Liquefaction System*
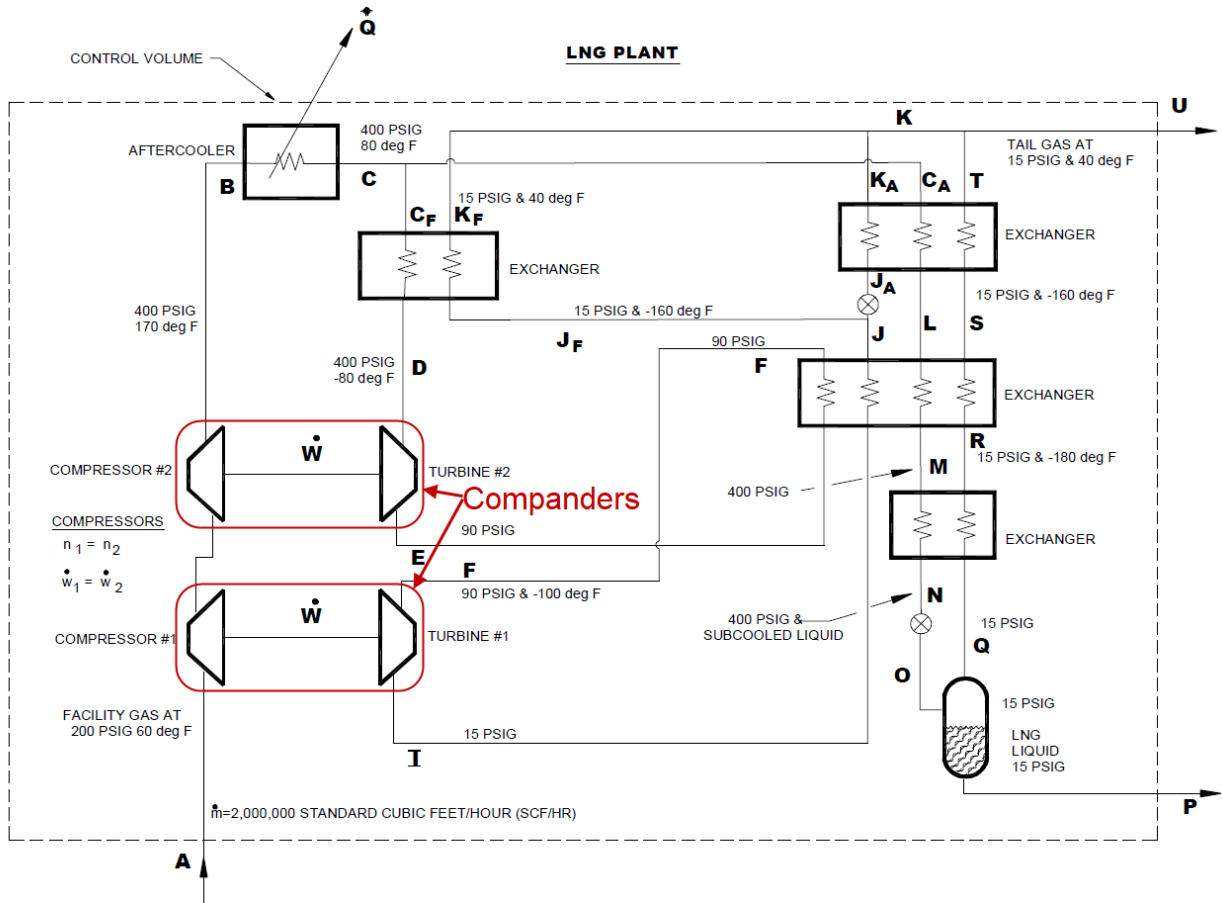
*Figure **Error! No text of specified style in document.**.101: Graphic of an Open Expansion Liquefaction System*
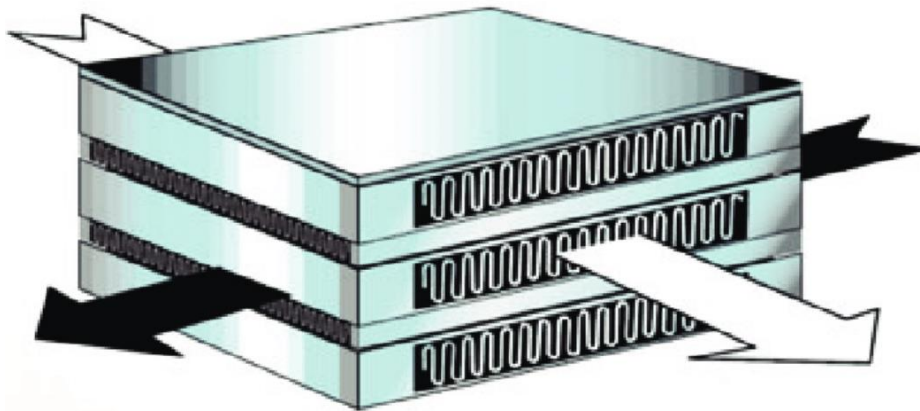

*Figure **Error! No text of specified style in document.**.102: Plate fin heat exchanger Source: https://www.researchgate.net/figure/Basic-structure-of-plate-fin-heat-exchanger-PFHE_fig7_283325770 (typically used for cryogenic service)*

*Figure **Error! No text of specified style in document.**.103: Plate fin heat exchangers typically made of aluminum and brazed together Source: http://webwormcpt.blogspot.com/2010_07_01_archive.html (typically used for cryogenic service)*
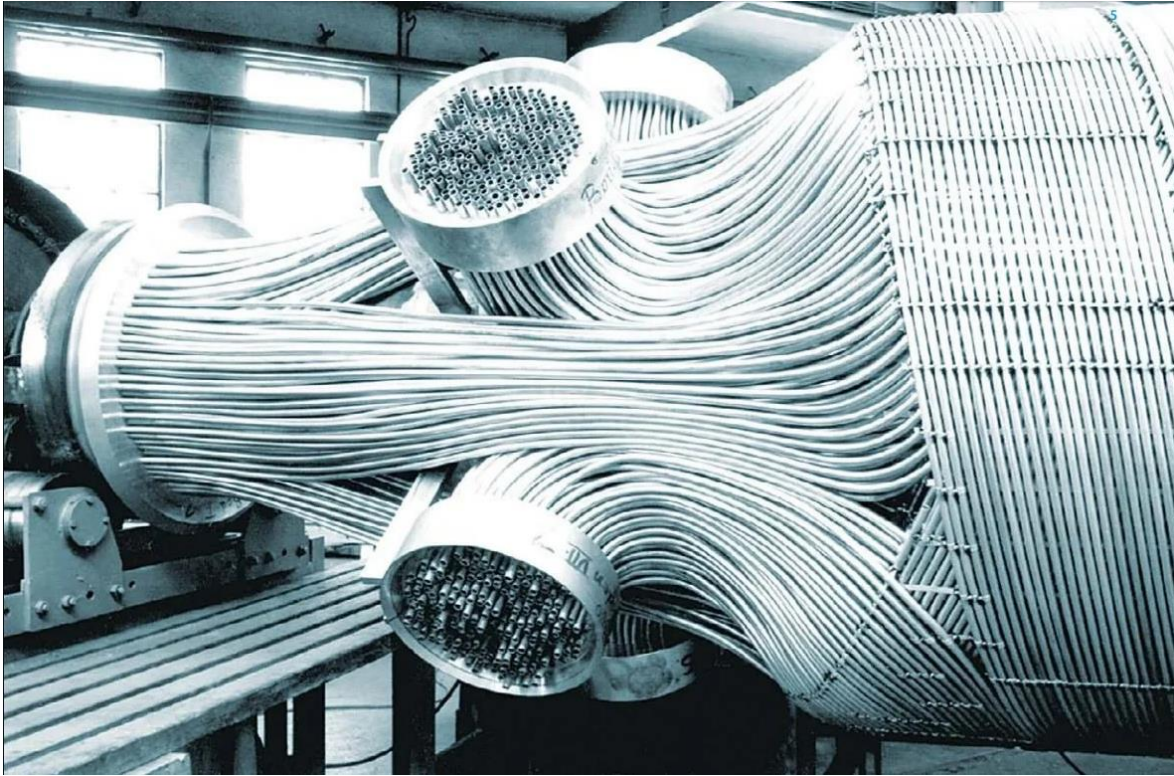
*Figure **Error! No text of specified style in document.**.104: Spiral wound heat exchanger Source: https://www.reddit.com/r/engineering/comments/3xdhg9/the_end_of_a_coil_wound_heat_exchanger/ (typically used for cryogenic service)*

Closing Remarks

In this 3-part series (Parts 1, 2, 3A, 3B) many topics on thermodynamics were covered. A recommended prerequisite to this 3-part series of study is the "Introduction to LNG Study" and the "LNG Plant Safety Study".

This 3-part series was elementary in nature as only Steady State, Steady Flow (SSSF) examples were examined. Most of the analyses were performed using the first law of thermodynamics which states that for a SSSF analysis the energy into a control volume must equal the energy out of a control volume. The highest math used was addition, subtraction, multiplication, and division.

In the analysis of natural gas liquefaction systems, the open expansion liquefaction system was examined showing how the energy within the pipeline pressure was let down using expanders to produce cold temperatures. These cold temperatures were used to produce Liquefied Natural Gas (LNG). This is essentially a no energy required process (except for auxiliary systems).
In the analysis of natural gas liquefaction systems single component, mixed refrigerant (MR) and C3MR systems were analyzed. These systems have the potential to be very efficient. However, that efficiency comes only if the operators are effectively adjusting the MR mixture to properly compensate for the feed gas composition changes and for the temperature changes of both the ambient environment and the feed gas. The C3MR process made this process easier as it made sure that both the feed gas and the MR entered the main cryogenic heat exchanger (MCHE) at a fixed temperature regardless of ambient temperature.

In this last section 3B, the closed nitrogen cycle expansion system was reviewed from initial concepts to understand the workings of compression and expansion to optimization of complex nitrogen expansion systems.

Nitrogen Expansion Natural Gas Liquefaction plants are becoming increasingly popular because they are easy to operate, easy to understand, easy to maintain and being produced with higher efficiencies as new expander and compressor machining processes are resulting in higher efficiency compressors and expanders.

In all the analyzes presented, the goal of the work was to deliver an understanding and not to perform detailed design nor optimization. Thus, pressure drops were neglected and other real-world issues like heat leak from the environment were neglected.

These learnings were technical in nature, and thus, were not focused on plant safety and reliability. This author reminds the learner that the only reason technical learning is being presented is to provide you with the technical knowledge needed to operate your plant safely and reliably.

The greatest asset an LNG plant has is its human resources. A culture of Safety, Reliability, Continuous Learning and Continuous Improvement are strongly encouraged.