Liquefied Natural Gas (LNG) - Thermodynamics and Liquefication Systems
Part 1

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Steven Vitale Ph.D., P.E.

CED CANADA
engineering.ca
CPD FOR THE PROFESSIONAL

Continuing Education and Development, Inc.

P: (877) 322-5800
info@cedengineering.ca

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A Safety Moment

This learning document is meant to be at a technical level document mostly on refrigeration systems. The most important intention of every document I produce is to give you the basic technical knowledge, that you need to start your study, on how to continue to make 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 a someone who loves, and needs them, to come home at the end of their shift.

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

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

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

Figure 1. Beautiful Granddaughter

Source: Self-Made photo

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 that of the vapor compression – condensation – pressure drop evaporation system. This very same system is used in household refrigerators and in massive liquefaction plants around the world. The massive systems used in LNG export facilities use the very same principles as most of the 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 are different from those used in the home air-conditioning and refrigeration systems.

We take for granted that when we want a cold soda, it is readily available from the refrigeration system in our house. The amazing thing is that the systems are so well perfected that typically the only reason they are replaced is that they go out of style, and not because they are malfunctioning. These systems run for many decades with little if any maintenance. One reason for these smaller systems being so reliable for so many years is that 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 not efficient. These toxic refrigerants included ammonia, methyl chloride, and sulfur dioxide. Refrigeration systems were often installed outside to avoid death from a refrigerant leak.

I myself had chemical pneumonia for a month due to exposure to anhydrous ammonia from a small refrigeration system. Many have been killed by such exposures.

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 the production, use and reuse of Freons. Concurrent with phasing out the 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 environmental concerns associated with them 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 size 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.

In this work, in Phases 1 and 2, we will start our study with understanding thermodynamics. Then in Phase 3 we will apply this knowledge to include the understanding of simple systems like that in your automobile air conditioning system and your house household refrigerator/freezer system. Finally, in Phase 3 we will expand this learning to include the understanding of the ultra-large systems used to liquefy natural gas for export terminals (in the 300,000 hp range).

There are billions of small-size air conditioning/refrigeration systems in operation in the world today, but only a few hundred of the behemoth-size systems are used in the LNG Liquefaction Industry. There are many other gas liquefying industries, but we will limit our 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 not complex to achieve the 35 F temperature needed to cool down a soda. If you want to store frozen food at 0 F it takes more energy but still the technology is simple.

However, if the outside environment is 80 F it will take a significantly large amount of energy and a 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 folds 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 extremely power intensive. Keep in mind absolute zero temperature is -459.67 F.

To better respond to anomalies during plant operation, engineers need some understanding of thermodynamics. 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 of 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 a thermodynamic software should be used in concern with this learning.

At the end of this Phase 1, a thermodynamic overview of an open expansion gate station-type liquefaction plant process is studied. This open expansion plant overview study is not rigorous in that it is based on a simplified plant process without pressure drops, it does not take into account the pinch points of process heat exchangers, and it treats natural gas as a pure substance of 100% methane. Although this is not rigorous enough to depict the actual process, it serves the intent of this publication by exercising and reinforcing the concepts presented herein.

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, fluid velocity energy, head pressure due to elevation, unintended heat and mass leaks etc. associated with flows through piping, exchangers, and other process equipment. Such effects are not taken into account 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 and not get bogged down in numerical detail.

In this Phase 1 course, we will use a thermodynamic chart for methane for computations and illustrations. A separate copy of this chart accompanies this course to allow the learner to zoom in for better clarity and accuracy in reading the chart.

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 consensus code, National Fire Prevention Association (NFPA) 59A. 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)
- 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.

Terms and Units of Measure for Natural Gas and LNG

1. 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’)

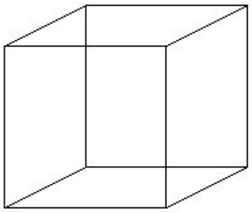


Figure 2: Cubic Foot is a volume 1’ x 1’ x 1’

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 different countries use a different standard temperature of the water being heated. The heat capacity of water differs with temperature.

HHV	Higher heating value is the amount 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

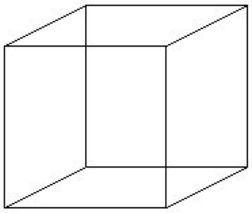


Figure 3: Cubic Foot is a volume 1' x 1' x 1'

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 Industry and the Petrochemical Industry. For this training, the Petrochemical Industry Barrel will be used. 1 barrel = 42 gallons

2. Approximations used in this training:

$1\ m^3 \sim 35.3\ ft^3$

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

One cu.ft. ~ 7.48 gallons

1 gallon of LNG ~ 80 SCF of vapor

Definitions

Before we can talk about the thermodynamics of refrigeration, we must define some of the terms that we will use. Care must be taken because improper use of terms can lead to confusion. The definitions in this section are not rigorous but are shortened in keeping with the intent of this training document. Some of the terms we will use in here are given below. Refer to these terms regularly to ensure that you have a clear understanding of how to use them. These terms are listed in an order that allows each definition to build on an earlier definition.

Pure Substance: A substance that can be treated as one homogeneous material with definite chemical and physical properties. Example: Carbon (as in graphite alone) is a pure substance, which consists of one, or more carbon atoms. Carbon, mixed in two allotrope forms of diamond and graphite, is not a pure substance. Methane is also a pure substance, which consists of (one or more methane molecules).

Element: The simplest type of pure substance which cannot be separated into simpler substances by a chemical change. Example: Carbon is an element because it cannot be made simpler. Methane is not an element because it can be made simpler by breaking it into its elements of carbon and hydrogen.

Atom: The smallest particle of an element that characterizes that element. (Example: Carbon is an element; the smallest particle of carbon is an atom. A cup of pure carbon is a cup of the element carbon, which consists of millions of carbon atoms. The letter C is the symbol used to express a carbon atom.

Molecule: A neutral (not electrically charged) group of atoms bound together. Example: CH_4 is the symbol used to express a methane molecule. This symbol represents one carbon atom bound to 4 hydrogen atoms.

Compound: Molecules that contain at least two “different” elements in a fixed proportion. Example: A methane molecule (CH_4) is a compound of 1 carbon atom and 4 hydrogen atoms. A hydrogen molecule (H_2) is a molecule made up of 2 hydrogen atoms. H_2 is a molecule but is “not” a compound because it does not contain two or more “different” atoms.

Hydrocarbon: A compound that contains only hydrogen and carbon. (Example: Methane (CH_4) is a pure substance, and is a molecule, and is a compound and, is a hydrocarbon. Table 3.1 will show that as more carbon and hydrogen atoms are added to a hydrocarbon compound, the atomic mass increases. Such compounds are often referred to as “heavy hydrocarbons.” Methanol (CH_4O) and Ethanol (C_2H_6O) are not hydrocarbons because although they contain carbon and hydrogen, they also contain oxygen).

Phase: Matter that is uniform throughout. Phases can be solid, liquid or, gas. A gas is a quantity of matter that does not have a definite shape or volume. A liquid has a definite volume but not a definite shape. A solid phase has a definite shape and a definite volume.

Mixture: A group of more than one substance in a volume. Example: Air is a mixture of oxygen and nitrogen and other gasses. Natural Gas is a mixture of methane, ethane, propane, and many other trace gasses. Methane (CH_4) is not a mixture since it is only one substance. Water (H_2O) is not a mixture since it is only one substance. However, colored water is a mixture since it is water mixed with some coloring agent. Mixtures contain different molecules but they are not in themselves molecules. Another definition of a mixture is a volume of two phases of the same pure substance.

The following table might help to assist you in understanding the above definitions.

Table 1: Typical Substances and Their Classification

Substance	Pure Substance	Element	Atom(s)	Molecule(s)	Compound	Hydro-carbon(s)	Mixture
Carbon (C)	Yes	Yes	Yes	No	No	No	No
Methane (CH_4)	Yes	No	No	Yes	Yes	Yes	No
Carbon Dioxide (CO_2)	Yes	No	No	Yes	Yes	No	No
Hydrogen Atom (H)	Yes	Yes	Yes	No	No	No	No
Hydrogen Molecule (H_2)	Yes	Yes	No	Yes	No	No	No
Propane (C_3H_8)	Yes	No	No	Yes	Yes	Yes	No
Natural Gas	No	No	No	No	No	Mostly	Yes
Air*	No	No	No	No	No	No	Yes

Pure Water (H_2O)	Yes	No	No	Yes	Yes	No	No
Colored Water (H_2O)	No	No	No	No	No	No	Yes
Water and Steam (H_2O)	Yes	No	No	Yes	Yes	No	Yes
Methanol (CH_4O)	Yes	No	No	Yes	Yes	No	No

* Note that substances such as air do contain molecules of oxygen (O_2)—with each oxygen molecule containing two atoms of oxygen—and do contain molecules of nitrogen (N_2)—with each molecule of nitrogen containing two atoms of nitrogen. These molecules of oxygen and nitrogen make up air, but air itself is not a molecule. Air by nature is a mixture of many different molecules, some of which are compounds.

Boiling: The change of phase from a liquid to a gas.

Condensation: The change of phase from a gas to a liquid.

Heat: A form of energy that transfers from one body to another driven by a difference in temperature. Heat will travel from a hotter body to a colder body.

Temperature: A property of a substance that determines how it transfers heat with its surroundings. It is typically expressed in degrees Fahrenheit (deg F or °F).

Absolute Temperature: Another way to express temperature is degrees Rankine (deg R). We will rarely use this term. Degrees Rankine is a measure of temperature from the reference point of absolute zero temperature.

Latent Heat of Vaporization: The amount of heat needed to change a unit of mass from a liquid to a vapor.

Sensible Heat: The heat needed to change the temperature of a unit of mass without changing its phase.

Energy: The capacity to do work or transfer heat. It is typically expressed, as British Thermal Units (Btu). In this booklet we will use the term energy to refer to thermodynamic energy.

Enthalpy: A term used to describe the thermodynamic energy of a substance, related to its pressure and temperature.

Entropy: For the limited scope of this learning, we will define entropy as a property that is used to help define the reversibility of a process. If an expansion or compression process is reversible (perfectly efficient) the process occurs without a net change in entropy. A process that is not perfect results in a net overall increase in entropy. In the real world, there is no such thing as a truly reversible compression or expansion, however, it is used as a comparison for computing efficiency.

Pressure: A force per unit area exerted on a surface. It is typically measured in pounds of force per square inch (lbf/sq. in). Gauge pressure (psig) is a measurement of pressure above atmospheric pressure. Absolute pressure is a measure of pressure from absolute zero. (See Figure 4) **Absolute pressure (psia) is gauge pressure plus the atmospheric (barometric) pressure.** Atmospheric pressure is approximately 15 psia.

Equation 1: Relationship between psig and psia (Note: 15 was used as a rounded-up number – actual atmospheric pressure (known as standard pressure) is different for different agencies and industries. The Natural Gas Industry uses 14.73 psi as atmospheric pressure).
psia = psig + 15 psi

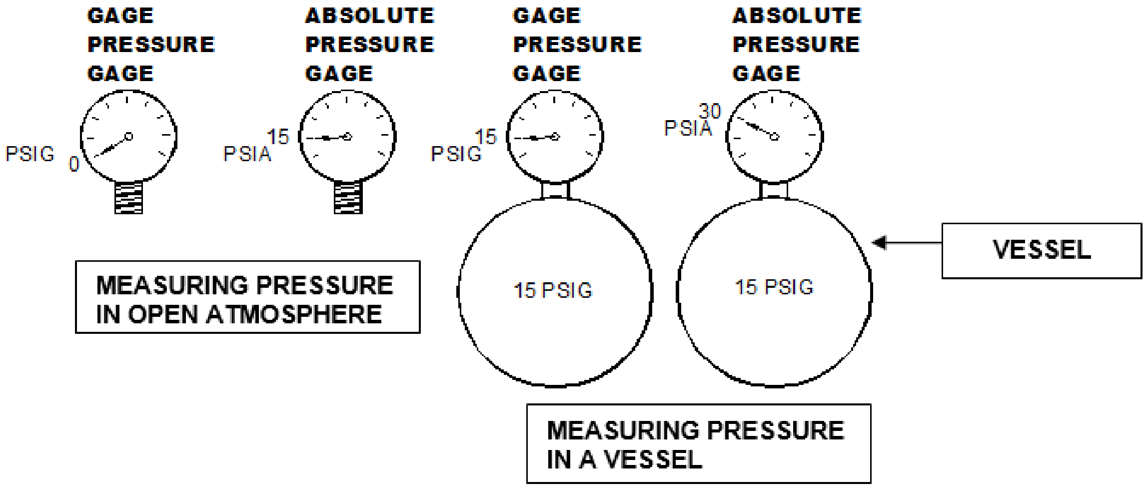


Figure 4: Relationship between psig and psia (note: 15 was used as a rounded-up number – actual atmospheric pressure (known as standard pressure) is different for different agencies and industries. The Natural Gas Industry uses 14.73 psi as atmospheric pressure.

Source: Self-Made

Mass: The quantity of a substance (this is directly related to the number of atoms/molecules of the substance). It is usually measured in pounds mass (lbm). Do not get this mixed up with weight. A pound mass (lbm) of a substance will weigh (push down on a scale) one pound force (lbf) here on earth. That same pound mass (lbm) of the substance will weigh (push down on a scale) zero-pound force (lbf) in outer space. It is still one pound mass (lbm) in space, but it just does not push on any scale in outer space. To understand this better, just realize that when you get on a scale, the force that you exert down on the scale is nothing more than the earth’s pull on the mass of your body. If you are not near the earth, the mass of your body still exists, but the force on it due to the earth no longer exists. For our study, realize that all substance has mass.

Weight: The force that gravity exerts on a mass. It is usually measured in pounds force (lbf). Do not get this mixed up with mass (see mass above). In this booklet pounds mass will be designated as (lbm) and pounds force will be designated as (lbf). It is important to not get these two very different terms confused. Below is a picture that may help you remember the difference.

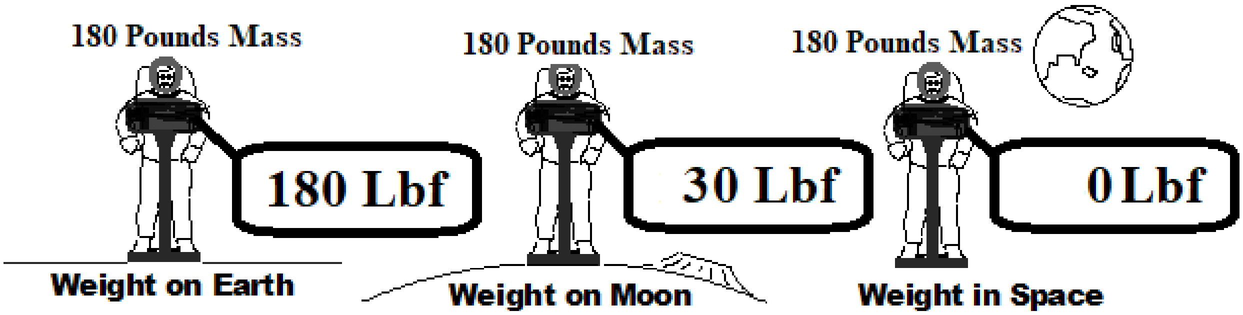


Figure 5: Conceptual Spring scale image describing the difference between mass and weight

Source: Self-Made

Density: The density of a substance is the amount of mass of that substance in a given volume. (Example: the density of methane at atmospheric pressure (14.73 psia is used by the Natural Gas Industry) and 60 deg F is approx. 0.042 pounds mass per cubic foot (0.042 lbm/cu.ft.). This means that every cubic foot of methane at that temperature and pressure has a mass of 0.042 lbm.

Specific Volume: The specific volume of a substance is the volume that one unit of mass occupies. Let’s again take methane at atmospheric pressure and 60 deg F. We already said that one cubic foot has a mass of approx. 0.042 lbm. Well, then the volume of one pound mass is nothing more than the number 1 divided by the density. In the case of methane at atmospheric

pressure and 60 deg F, it is (1/0.042) cubic feet per pound mass (cu.ft./lbm). For this temperature and pressure methane has a specific volume of 23.81 cubic feet per pound mass (cu.ft./lbm). A simple formula to remember is as follows:
Equation 2: Relationship between density and specific volume
specific volume = 1/density
or
density = 1/specific volume

Control Volume: The volume created by an imaginary enclosure (control surface) that one places around a device to be studied. If we were to study what happens to the fluid flowing through a valve, we would place an imaginary box (control volume) around the valve and analyze the flow into the control volume the flow leaving the control volume, and the fluid inside the control volume. In this book, dashed lines (-----) are used to designate control volume surfaces.

Fluid Property: A characteristic of the fluid that is measurable or able to be identified by the use of measured parameters. Pressure and temperature are measured fluid properties. Density or specific volume can be measured, but are more easily identified by using more easily measured properties such as pressure or temperature. Enthalpy and entropy cannot be measured directly but are identified via measured properties such as temperature and pressure.

It is important to note, that the properties of enthalpy and entropy are based on a value assigned to a reference point selected by the engineer doing the analysis. Thus, for a specific set of fluid properties, the values found by two different engineers might be different, however, the change in entropy and enthalpy across a process is always, the same. As an example, imagine two separate engineers compute the enthalpy value across a compressor. Engineer 1 shows the inlet enthalpy to be 300 Btu/lbm and the outlet enthalpy to be 600 Btu/lbm. Engineer 2 shows the inlet enthalpy to be -900 Btu/lbm and the outlet enthalpy to be -600 Btu/lbm. Note the difference in both cases is a positive 300 Btu/lbm, even though the inlet and outlet properties for enthalpy were different. They only used different reference points in the calculation. Most people today use software for such calculations.

The same can be said about degrees Fahrenheit (F) and degrees Rankine (R). The difference between 459.67 R and 469.67 R is exactly the same as the difference between 0-degree F and 10-degree F. The difference in temperature is 10 R which is equal to 10 F.

State: The state of a fluid is defined by its properties. If we define two or more intensive properties of a fluid (such as the pressure and temperature), we have defined its state. An intensive property is one that is not dependent on the mass of the fluid. Examples of intensive properties are pressure, temperature, and density. Examples of other properties, which are not intensive, are the total mass and the total volume of a substance. Although total mass and total volume are not intensive, the density (mass per unit volume) and specific volume (volume per unit mass) are intensive. This is because the density and specific volume are not dependent on mass. Both a small amount and a large amount of mass of a substance, at the same state, will have the very same density and the very same specific volume.

Steady-State Process: A process where the state of the fluid at every point in the process being studied does not change with time. If we put pointers at every location in the flow stream and watch them as we analyze the process, the process is steady-state if the properties of the fluid at every location do not change with time. An example would be a valve that has flowing LNG at 415 psia and –160 deg F at the inlet and a mixture of liquid and vapor at 30 psia and –242 deg F at the outlet. If these pressures and temperatures do not change over time, this flow process can be treated as a steady-state process.

Steady-Flow Process: A process that does not have changes in flow rates or changes in fluid inventory over the time period when the process is studied. If we wanted to study the flow of water out of a sink faucet, assuming the supply pressure is constant, we can treat the analysis as a steady flow process as long as we are not changing the valve opening. An example of a steady-flow process would be a knock-out pot that has a constant liquid level and constant and equal inlet and outlet flows. Such a knock-out pot would not be operating at steady flow if the flow rate into or out of the pot was fluctuating or if the liquid level was changing over time.

Steady-State-Steady-Flow Processes (SSSF): A process that is both steady-state and steady-flow. **All the analyses in this learning document are steady-state-steady-flow.** This is the simplest type of analysis to make. It assumes that the process is both steady-state (pressure and temperatures do not change at any point over time) and steady-flow (flow rates at any point do not change over time).

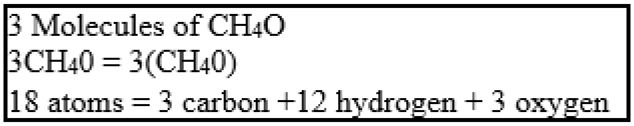
What is Natural Gas? What is LNG?

Natural gas is a combination of many gasses. The natural gas delivered to end-use customers comes from deep in the ground. There are several theories on how this gas was formed. The most popular theory is that it was formed by the decay of animal and plant life over millions of years and high pressure and elevated temperature.

Natural gas is a combination of several hydrocarbons and some non-hydrocarbon gases. In natural gas there is usually a very high percentage of methane (CH_4), some ethane (C_2H_6), some propane (C_3H_8), and small amounts of other hydrocarbons.

Natural gas also contains small amounts of non-hydrocarbon gases like nitrogen (N_2), oxygen (O_2), carbon dioxide (CO_2), and trace amounts of other gases. Note that we gave the molecular formula for each of these components. The molecular formula tells you the proportions of the elements that make up each molecule (Example: A propane (C_3H_8) molecule contains 3 carbon atoms and 8 hydrogen atoms). Take note that the number following the element symbol tells you how many atoms of that element are in that molecule. Thus, in an oxygen molecule (O_2) there are 2 oxygen atoms. In an ozone molecule (O_3) there are three oxygen atoms.

In a chemical equation, if you want to show 2 oxygen molecules you would write $2(O_2)$. This means 2 oxygen molecules for a total of 4 oxygen atoms. There is no importance to the brackets in this learning other than to try to add clarity. If we use $2O_2$ or $2(O_2)$ or $(2O_2)$ it all means the same thing (2 molecules each having 2 atoms of oxygen). See Figure 6 below for a deeper understanding.



3 Molecules of CH_4O

$3CH_40 = 3(CH_40)$

18 atoms = 3 carbon +12 hydrogen + 3 oxygen

Figure 6: Three Molecules of Methanol

You will hear the terms “light hydrocarbon” and “heavy hydrocarbon.” These terms refer to the mass of the hydrocarbon molecule. The atomic mass is a measure of the mass of a molecule. The more carbon atoms in a hydrocarbon, the greater the mass of the molecule. Natural gas usually contains over 90% methane, a few percent ethane, and very small amounts of propane, butane, and other heavier hydrocarbons.

Because natural gas is mostly methane, for the rest of this basic learning we will often talk about natural gas as if it were 100% pure methane—to make understanding the concepts easier. However, caution must be maintained because, in some process analyses, this can introduce significant error.

A methane molecule is made up of one carbon atom, bound to four hydrogen atoms. Methane is referred to as CH_4 (molecular formula). When methane and oxygen combine, if the combustion is complete, it forms carbon dioxide (CO_2) and water (H_2O) and it gives off heat (exothermic reaction). The equation describing this is $CH_4 + 2(O_2) = CO_2 + 2(H_2O)$ + heat release. This means that one methane molecule combines with two oxygen molecules (each oxygen molecule contains two oxygen atoms) to form one carbon dioxide molecule (each carbon dioxide molecule contains one carbon atom and two oxygen atoms) and two water molecules (each water molecule contains two hydrogen atoms and one oxygen atom) and gives off heat. Count the number of carbon, hydrogen, and oxygen atoms on each side of the equation. They are equal.

Table 2: Substance Formulas may be of use in understanding these concepts

Substance	Chemical Formula	Atomic Mass	Number of Carbon Atoms	Number of Hydrogen Atoms	Number of Oxygen Atoms
Methane	CH_4	16	1	4	0
Ethane	C_2H_6	30	2	6	0
Propane	C_3H_8	44	3	8	0
Butane	C_4H_{10}	58	4	10	0
Water	H_2O	18	0	2	1
Carbon Dioxide	CO_2	44	1	0	2

LNG is nothing more than Liquefied Natural Gas. It is made by cooling down natural gas until it liquefies. At atmospheric pressure, natural gas liquefies at approximately –260 deg F. The composition of LNG is usually slightly different than that of the natural gas brought into the LNG liquefaction plant because—

1) certain of the molecules in natural gas do not liquefy and
2) certain heavy hydrocarbons drop out of the natural gas during the liquefaction process and
3) certain molecules are intentionally removed and discarded during the liquefaction process.

In particular, almost all of the nitrogen in the natural gas does not liquefy, heavy hydrocarbons condense out of the process stream first, and carbon dioxide and water are typically removed in the process because they can freeze out on the heat exchangers and degrade the liquefaction process. In our analysis here, as mentioned above, we will often treat natural gas as if it were the pure substance methane. Thus, we will often treat LNG as if it were liquid methane.

Understanding Pressure, Temperature, and Boiling

1. Pressure:

We deal with pressure all the time but just what is pressure? As you know fluids contain molecules. These molecules exert a force on surfaces. Pressure is nothing more than force on a surface. A common set of units for measuring pressure is pounds force per square inch. We know that as psi. A pressure gauge is the most common device used to measure pressure. There are many types of gauges that operators use on a daily basis.

We live in a pressurized environment. The air we breathe is at approximately 15 psia. Note that I am using “psia”. This is an abbreviation for “pounds per square inch absolute”. That means that on every 1 inch by 1 inch area there is 15 pounds force pushing on it. The “a” at the end of the term psia refers to “absolute” pressure. In outer space where there are no air molecules, there is zero absolute pressure. If we read a gauge calibrated for reading absolute pressure, here on earth, without it being connected to anything, it would read approximately 15 psia. In outer space it would read zero.

The gauges we use on our pipes typically “do not” measure psia but instead measure “psig”. The “g” in the term psig refers to “gauge” pressure. This gauge is calibrated to read 0 at 15 psia. When we read a pipeline pressure as 28 psi, we are really reading 28 psig. However, since most times we do not use the term “g” in our everyday talk, it is easy to forget that when we say psi, we are really referring to psig (gauge pressure). If we were to unscrew that gauge from the pipe, it would read zero pressure. If we were then to take that gauge into outer space, if it did not have a pin to prevent the needle from reading below zero, it would read a -15 psig! Take note the gauge would read “negative” in outer space. This is because here on earth all our surroundings are at approximately 15 psia, which is atmospheric pressure. **In order to convert psig to psia, we must add 15 to the psig gauge reading (See Equations 1 and 3).**

Equation 3: Same as equation 1 except including an example (note atmospheric pressure is rounded up to 15 psia)

psia = psig + 15

Example: A gauge reading of 62 psig is measuring 77 psia.
In the study of thermodynamics, we typically use “absolute” pressure. Thus, if we say we have a methane stream flowing at –160 deg F and 100 psig, we need to convert the psig pressure into psia pressure before doing a thermodynamic analysis. In order to analyze this flow stream, we

must look up the properties of the methane stream using –160 deg F and 115 psia. (Remember: psia = psig +15). In our case, it is 100 psig + 15 psi.

Look again at the pressure gage examples shown in Figure 7 (reproduced below):

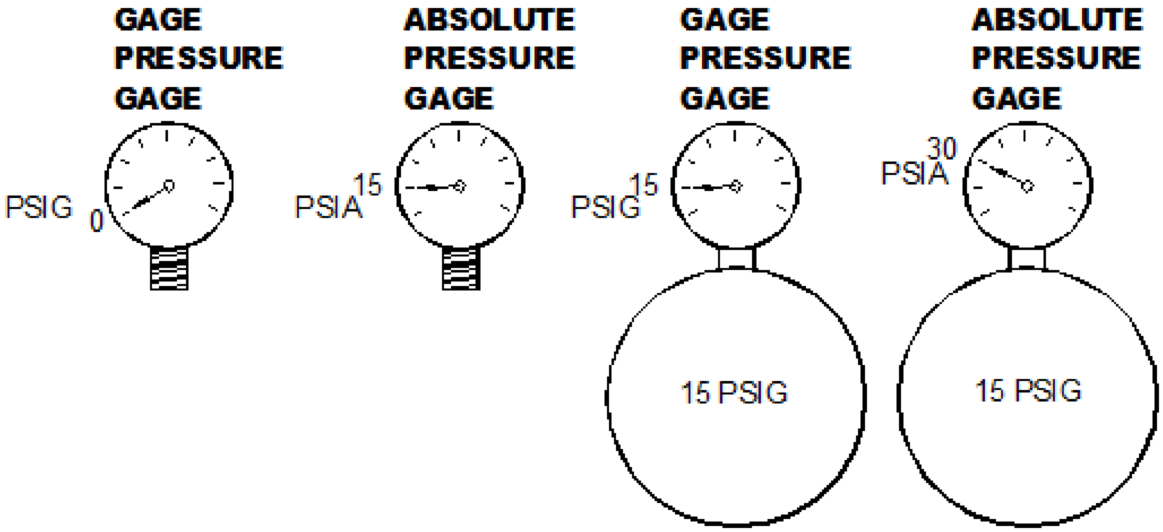


Figure 7: Conceptual graphic showing the difference between psig and psia

Source: Self-Made

In the study of thermodynamics, we typically use "absolute" pressure. Thus, if we say we have a methane stream flowing at –160 deg F and 100 psig, we need to convert the psig pressure into psia pressure before doing a thermodynamic analysis. In order to analyze this flow stream, we must look up the properties of the methane stream using –160 deg F and 115 psia. (Remember: psia = psig +15) In our case it is 100 psig + 15 psi = 115 psia.

Look again at the pressure gauge examples shown in Figure 7 (reproduced below):

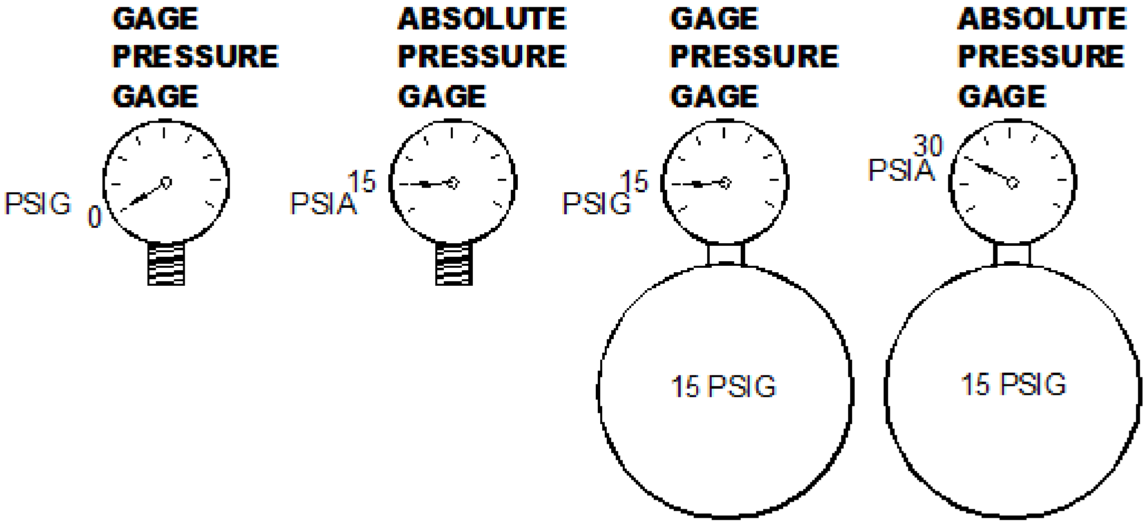


Figure 7 above, is shown a 2nd time to bring emphasis to it.

Source: Self-Made

2. Temperature:

Temperature is a difficult term to define. We say something is hot or something is cold because it feels that way to our senses. We will give a weak definition of temperature as the property of a substance that causes heat flow. Heat flows from a hot substance to a cold substance. Devices that measure temperature (thermometers) allow us to determine which substances are colder and which are hotter compared with each other. We could give a more precise definition relating temperature to the kinetic energy of the molecules within a substance, but for this level of study the above definition will suffice.

3. Boiling and Condensation:

A word about boiling and condensation is needed before we proceed with this study. Let's start by talking about boiling water. If we take an open pot of water at 60 deg F and we put it on the stove over a lit stove burner, the water will heat up. The open pot is at 0 psig (15 psia). The heat needed to raise the temperature of the water is called *sensible heat*. As we continue to heat the water, it becomes hotter and hotter until it reaches 212 deg F. Look at Figure 8 below.

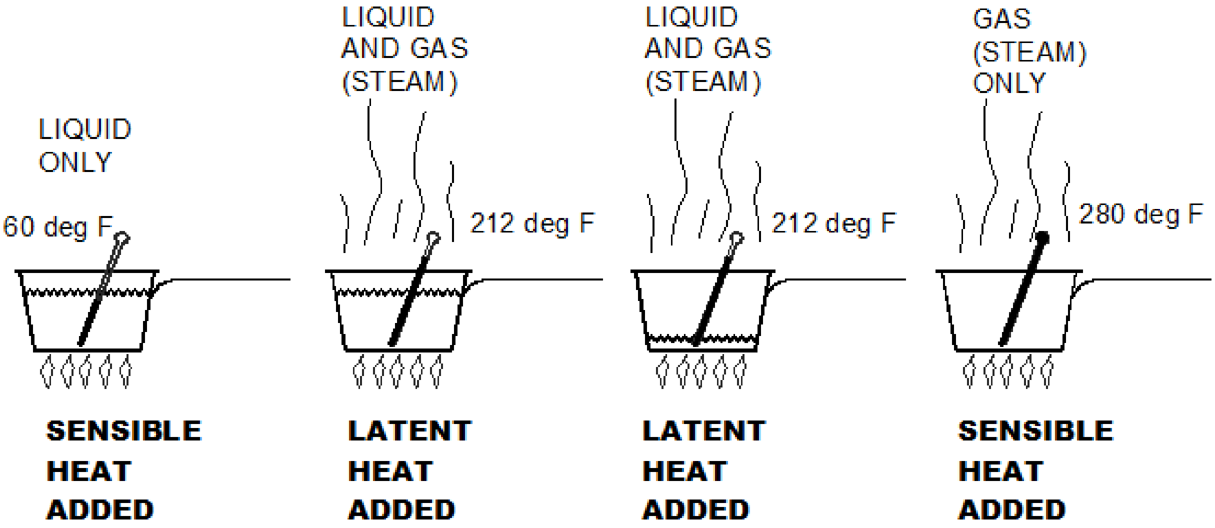


Figure 8: Example of Warm Water Being Heated to Superheated Vapor at 0 psig

Source: Self-Made

All the heat added until this point is called "sensible heat"—the heat used to raise the temperature of a substance without changing its phase. For water at atmospheric pressure (our example here), all the heat needed to raise the temperature of the water from 60 deg F to 212 deg F is sensible heat. This water that is below its boiling point is called a *subcooled liquid*. Now, as we continue to add heat, something different happens. At 212 deg F, the water starts to boil (this water at its boiling point is called a *saturated liquid*). Even if we turn the gas burner up really high, the water stays at 212 deg F, and the water just boils faster. The steam that leaves the pot is also at 212 deg F (this vapor at 212 deg F is called a *saturated vapor).* The heat that is added while changing the phase from liquid to vapor is called "latent heat".

Once the last drop of water is converted from a liquid to a vapor, the process changes again. If we continue to add heat after the last drop of water is turned to steam, the contents of the pot (now all steam) will again begin to rise in temperature. This hot steam, which is at a temperature higher than the boiling point of water, is called a *superheated vapor*. The added heat that raised the temperature of the steam above the boiling point is again called "sensible heat".

Look again at Figure 8. In summary, sensible heat is used to raise the temperature of the liquid, then latent heat is used to change the phase of the fluid from a liquid to a vapor, and then sensible heat is used to raise the temperature of the vapor. Just remember that sensible heat results in a temperature change and latent heat results in a phase change. If you were to reverse the process by removing heat, the steam vapor would cool down to 212 deg F (sensible heat leaving) and then condense back to a liquid at 212 deg F (latent heat leaving) and then continue to cool down to a cool liquid (sensible heat leaving).

It is important to note that as long as any liquid remains in the pot, the temperature in the pot does not exceed 212 deg F.

The boiling experiment described above was done with an open pot of water. If we conducted the same experiment with a pressure cooker set with a relief valve to maintain a pressure of 15 psig (30 psia), the boiling would take place at 250 deg F instead of 212 deg F. (See Figure 9 below).

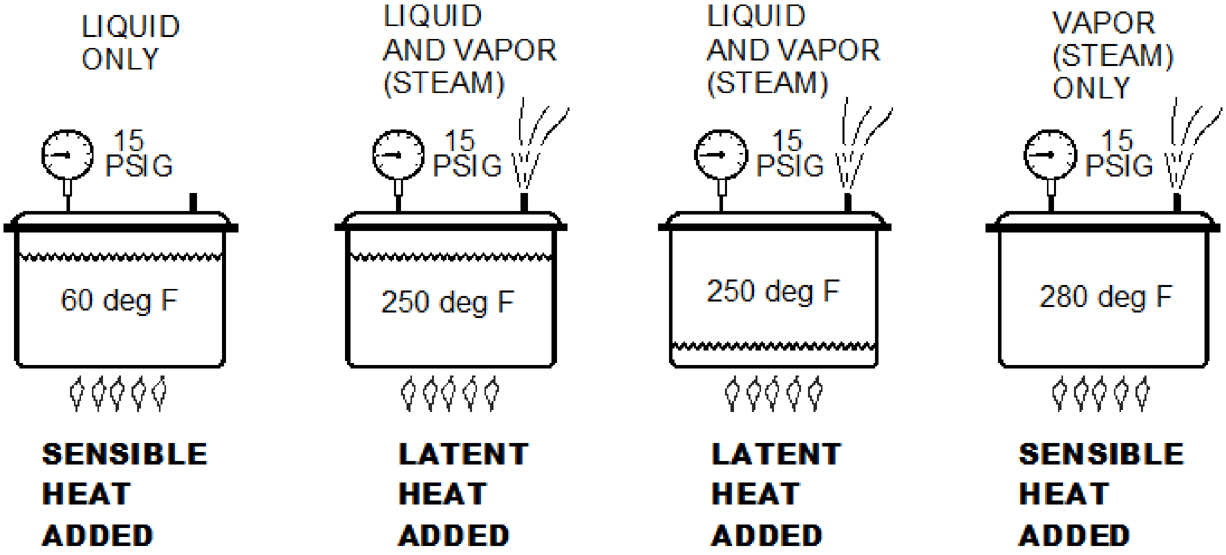


Figure 9: Example of Warm Water Being Heated to Superheated Vapor at 15 psig

Source: Self-Made

We can do the very same experiment with methane. If we take an open pot of liquid methane at -270 deg F and we put it on the stove over a lit stove burner, the methane will heat up (of course one would not put a flammable substance over an open flame, but this is shown for concept proposes only). The open pot is at 0 psig (15 psia). The heat needed to raise the temperature of the liquid methane is sensible heat used to raise the temperature of a subcooled liquid. As we continue to heat the liquid methane, it becomes hotter and hotter until it reaches -260 deg F. All the heat added until this point is called sensible heat—the heat used to raise the temperature of a substance without changing its phase (See Figure 10).

For liquid methane at atmospheric pressure (our example here), all the heat needed to raise the temperature of the liquid methane from –270 deg F to –260 deg F is sensible heat added to a subcooled liquid. Now, as we continue to add heat something different happens. At –260 deg F the liquid methane starts to boil (the liquid is now a saturated liquid being converted to a saturated vapor). Even if we turn the gas burner up really high, the liquid methane stays at -260 deg F and just boils faster. The vapor that leaves the pot is also at -260 deg F. The heat that is added while changing the phase from liquid to vapor is called “latent heat”.

Once the last drop of liquid methane is converted from a liquid to a vapor, the process changes again. If we continue to add heat after the last drop of liquid methane is turned to vapor, the contents of the pot (now all vapor) will again begin to rise in temperature. This hot vapor, which is at a temperature higher than the boiling point of liquid methane, is called a *superheated vapor*. The added heat that raised the temperature of the vapor above the boiling point is again called “sensible heat”. The gas delivered to our homes is a superheated vapor, even thought to use it is at room temperature.

In fact, while using the terminology of superheated gases, do realize that the air around us is a superheated gas containing mostly nitrogen and oxygen.

In summary (see Figures 9, through 11), sensible heat is used to raise the temperature of the liquid, then latent heat is used to change the phase of the fluid from a liquid to a vapor, and then sensible heat is used to raise the temperature of the vapor. This is what occurs in LNG vaporizers but at higher pressure.

Just remember that sensible heat results in a temperature change and latent heat results in a phase change. If you were to reverse the above process by removing heat, the vapor would cool down to –260 deg F (sensible heat leaving) and then condense back to a liquid at -260 deg F (latent heat leaving) and then continue to cool down to a cool liquid (sensible heat leaving). This is what we do in a liquefaction plant.

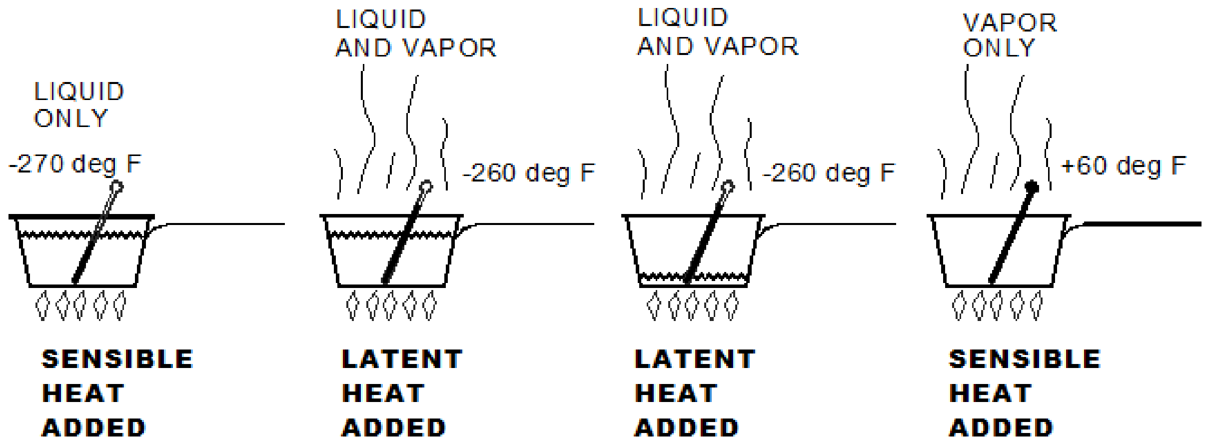


Figure 10: Example of Liquid Methane Being Heated to Superheated Vapor at 0 psig

Source: Self-Made

The boiling experiment we did above was done with an open pot of liquid methane. If we did the same experiment with a pressure cooker set with a relief valve to maintain a pressure of 15 psig (30 psia) the boiling would take place at -242 deg F instead of -260 deg F (See Figure 11).

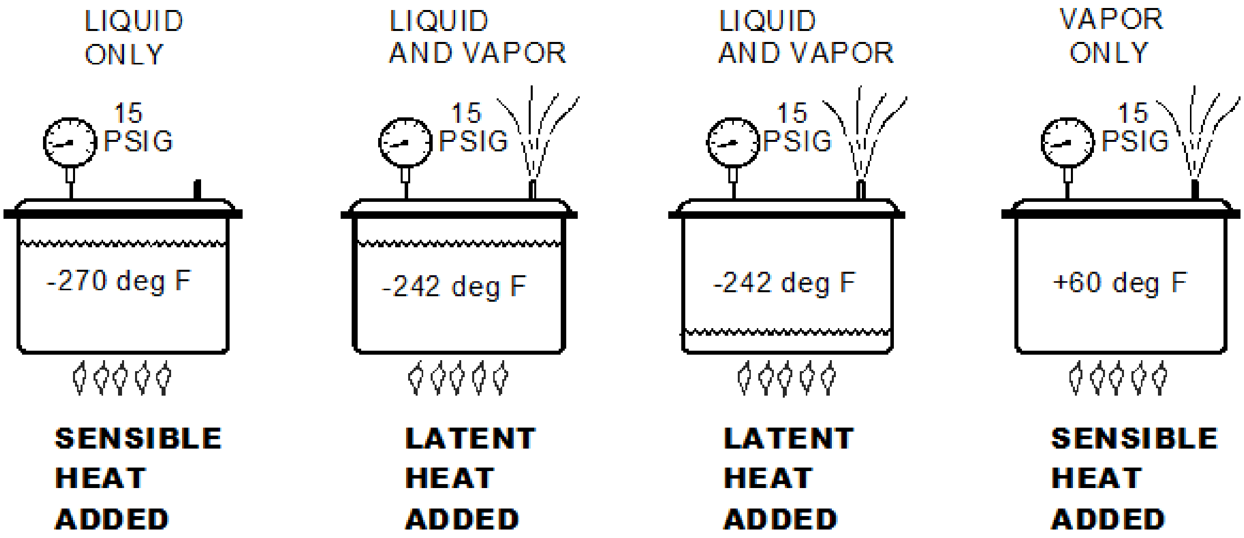


Figure 11: Example of Liquid Methane Being Heated to Superheated Vapor at 15 psig

Source: Self-Made

If we were to do this experiment at several different pressures, we could plot the data as shown in Figure 12, a hand sketch of a pressure enthalpy chart.

If we were to do this experiment many times and plot the fluid properties for many pressures, we would produce the pressure enthalpy chart shown in several parts of this learning.

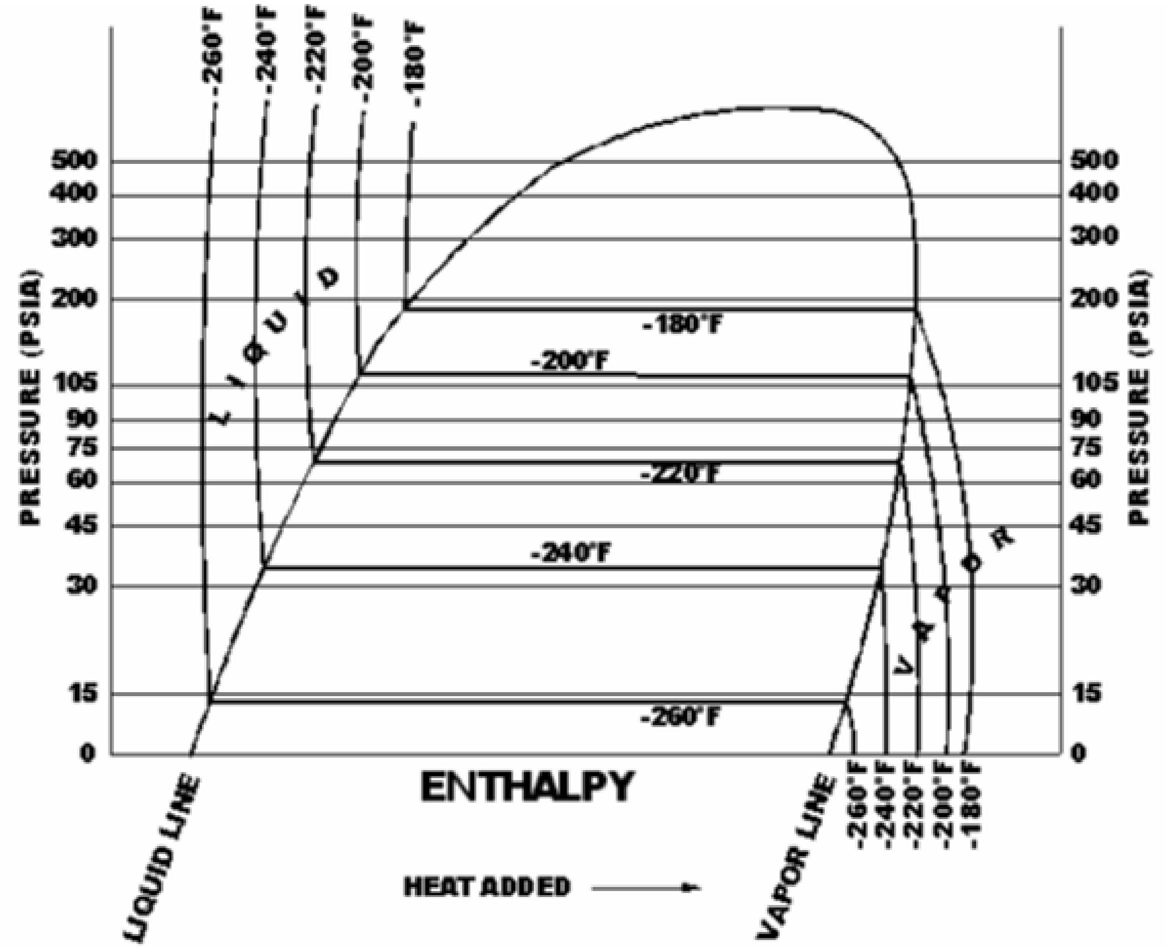


Figure 12: Hand Sketch of Pressure Enthalpy Chart

Source: Self-Made

Understanding Properties and Mass Flow Rate

1. Properties:

In Section 7 (Definitions), we defined a property as a characteristic of a fluid that is measurable or able to be identified by the use of measured parameters. By this definition, pressure, mass, temperature, density, enthalpy per unit mass, total enthalpy, volume, and specific volume are just some of the properties that could be used to define a fluid. Some of these properties are intensive and some are extensive.

An *intensive* (independent of mass) property is one that is not dependent on the mass of the fluid. Examples of intensive properties are pressure, temperature, and density. Examples of other properties, which are *extensive* (not intensive), are the total mass and total volume of a substance. Although total mass and total volume are not intensive, the closely related properties of density (mass per unit volume) and specific volume (volume per unit mass) are intensive. This is because the density and specific volume are not dependent on mass. Both a small amount and a large amount of a substance, at the same state, will have the same density and the same specific volume.

Why is this important to us in the study of thermodynamics? It is important because, with two intensive properties, we can define the state of a fluid. This cannot be done with extensive properties unless the properties are converted to the related intensive properties. Further, as we learn the first law of thermodynamics later in this learning, we will see that we need to understand and properly use both extensive and intensive properties to properly apply the first law of thermodynamics.

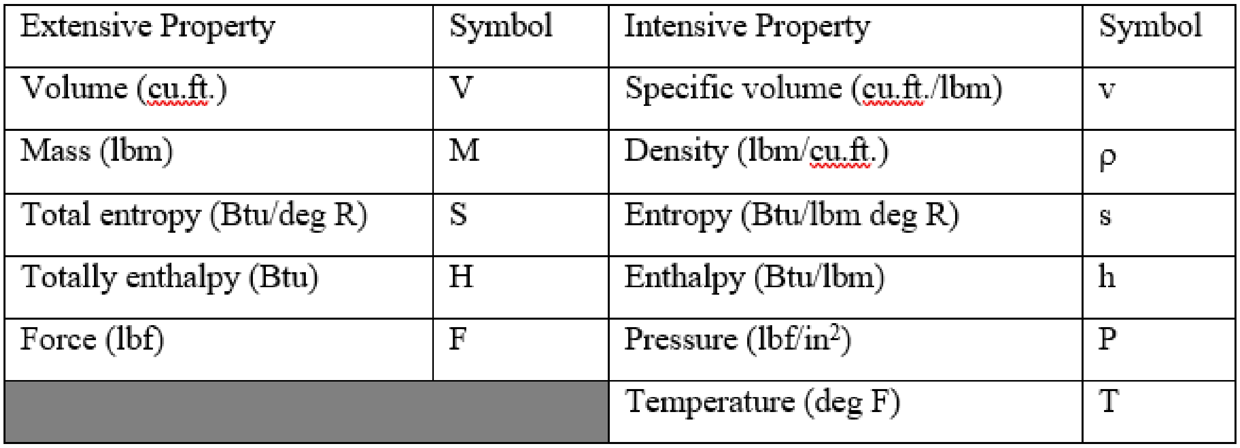
It is also important to note that if we know the mass of a fluid being studied and if we know an extensive property, we can then define a related intensive property. The following example may help you understand this better.

Example: Suppose you have 10 cubic feet of methane at 60 deg F.

The 10 cubic feet is an extensive property and the 60 deg F is an intensive property. With this information, we cannot tell the state of the fluid. However, if we also know that the mass of the fluid is 0.42 pounds mass (lbm), by using the mass and the volume we can define the density. Density is the measure of mass "per unit volume." In this example, the density is (0.42/10) lbm/cu.ft.; the density is 0.042 lbm/cu.ft.. With the two intensive properties of density (0.042

lbm/cu.ft.) and temperature (60 deg F) we can completely define the state of this fluid, as you will see in the next section. Table 3 may make this easier to understand.

Table 3: Extensive and Intensive Properties and Their Symbols



Extensive Property	Symbol	Intensive Property	Symbol
Volume (cu.ft.)	V	Specific volume (cu.ft./lbm)	v
Mass (lbm)	M	Density (lbm/cu.ft.)	ρ
Total entropy (Btu/deg R)	S	Entropy (Btu/lbm deg R)	s
Totally enthalpy (Btu)	H	Enthalpy (Btu/lbm)	h
Force (lbf)	F	Pressure (lbf/in^2)	P
		Temperature (deg F)	T

Note that, for the above, we can convert volume, total entropy, and total enthalpy to intensive properties by dividing them by the mass of the fluid. We can also convert the total mass into the intensive property of density by dividing the total mass by the total volume.

Although force is not usually defined or used as a property, it has been included in Table 3 for completeness. If we were to divide the force on a vessel wall by the surface area of the vessel wall, we would get pressure. We have not shown an extensive counterpart for the intensive property of temperature. One could argue that total molecular kinetic energy is the extensive counterpart; however, generally in the study of thermodynamics, no extensive property is used to convert to the intensive property of temperature.

Finally, to close this section on Properties, we will define **a test to tell if a property is intensive or extensive.** If we take the sample of fluid we are studying and cut it into two equal samples and ask, “Does the property change in each of the samples?” If it does change, the property is extensive (mass, volume, total enthalpy, etc.). If it does not change, the property is intensive (temperature, pressure, etc.).

2. Mass Flow Rate:

In the examples used in this publication, we use mass flow rates to show how much fluid is flowing in or out of a device. A mass flow rate is an amount of mass flowing per unit time. For our problems, we will use pounds mass per hour (lbm/hr.). We will use the term m° to designate mass flow rate. The dot over the m is there to remind us the term refers to a “rate” of mass flow.

In our open expansion gate station plant (which we will study later in this publication), the flow entering the plant is nominally 2,000,000 standard cubic feet per hour. Although this appears to be given as a volumetric flow rate, it is really a mass flow rate. If it had been stated as 2,000,000 cubic feet per hour—without the word “standard”—that would be purely a volumetric flow rate. But because it is stated as “standard,” it is referring to a specific number of molecules of mass flowing every hour. A “standard” cubic foot is a cubic foot of gas at the fixed state defined by 60 deg F and one atmosphere of pressure. At 60 deg F and one atmosphere of pressure, methane gas has a fixed density and thus a fixed mass per cubic foot. Thus, we conclude that the term “standard cubic foot” is a mass measurement.

Let’s compute (using methane) the amount of mass flowing into a plant when 2,000,000 standard cubic feet of mass are flowing. The exact density of methane at 60 deg F and one atmosphere is 0.04234 lbm/cu.ft.. This tells us that each standard cubic foot of methane has a mass of 0.04234 lbm. If we are moving 2,000,000 standard cubic feet of methane through the plant every hour we are then moving (2,000,000 std cu.ft./hr.) multiplied by (0.04234 lbm/std cu.ft.) through the plant. **This number, which we will use for our open expansion plant analysis (later in this publication) is 84,680 lbm/hr.**

Understanding the Pressure Enthalpy Chart

For this section, refer to the pressure enthalpy chart (Figure 13)[1].

So far, we have discussed many different parameters without effectively relating them to each other. The pressure enthalpy (PH) chart is nothing more than a graphical display (a road map) of many intensive properties, most often, for a pure substance on a single graph. Many more parameters could be plotted on the PH chart, but if you plot too many parameters, the plot becomes difficult to read due to clutter.

PH charts can also be made for mixtures, but these are more complex and difficult to use accurately, and thus, are typically used only to show concepts because as boiling and condensation take place for mixtures, the composition of the fluid changes. For the following discussion, refer to the chart shown in Figure 13, which applies to pure methane.

Since the chart has been reduced for this learning, you may want to also refer to the chart that accompanies this learning. It is a larger version of the identical chart. On the chart in Figure 13, there are horizontal lines that are lines of constant pressure. On the right of the chart, the line 150 psia has been underlined for display. (Note: If you are seeing the chart "sideways," use the Rotate View function in the View menu in Acrobat Reader to be able to properly view the chart.)

- There are vertical lines that are lines of constant enthalpy. On the chart, the line –1640 Btu/lbm has been highlighted for display.
- There are curved lines that define constant temperature. On the chart, the line – 60 deg F has been highlighted for display.
- There are dashed curved lines of constant specific volume. On the chart, the line 1.25 cu.ft./lbm has been highlighted for display.
- There are curved lines of constant entropy. On the chart, the line 2.12 Btu/lbm deg R has been highlighted for display.

You will also see a dome on the left side of the chart. That dome is the graphical representation of the change of phase from a liquid to a vapor. To the left of the dome is pure liquid and to the right of the dome is pure vapor. A point inside the dome represents a mixture of both liquid and vapor. For mixtures within the dome, the properties of the liquid portion are given as a pure

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[1] All the PH diagrams used in this learning are edited copies of Canjar, Lawrence N.; Tejada, Victor M.; Manning, Francis S. "Thermodynamic Properties of Hydrocarbons. Part 2: Thermodynamic properties of methane". Hydrocarbon Processing, vol. 41, no.9, Sep. 1962. pp 255-256.

liquid (on the liquid line on the left side of the dome (that line is called the saturated liquid line)). For mixtures within the dome, the properties of the vapor portion are given as a pure vapor (on the vapor line on the right side of the dome (that line is called the saturated liquid line)).

The question might be asked, “So, who cares?” We care, because with any two of these properties we can use the chart to find any of the other three properties. For example, if you have a vapor and you know pressure and temperature, you can look up the specific volume, entropy, and enthalpy of the vapor. Once you know the specific volume, you can calculate the density using Equation 2. If other properties had been displayed, they also could be read off of this chart.

Of particular interest is enthalpy. For example, if you know the pressure and temperature of methane vapor, you can use the chart to find the enthalpy of the vapor. As indicated in Section 7, Definitions, this enthalpy is the energy contained in the methane at a specific temperature and pressure.

We will use the pressure enthalpy chart extensively once we learn the use of the first law of thermodynamics.

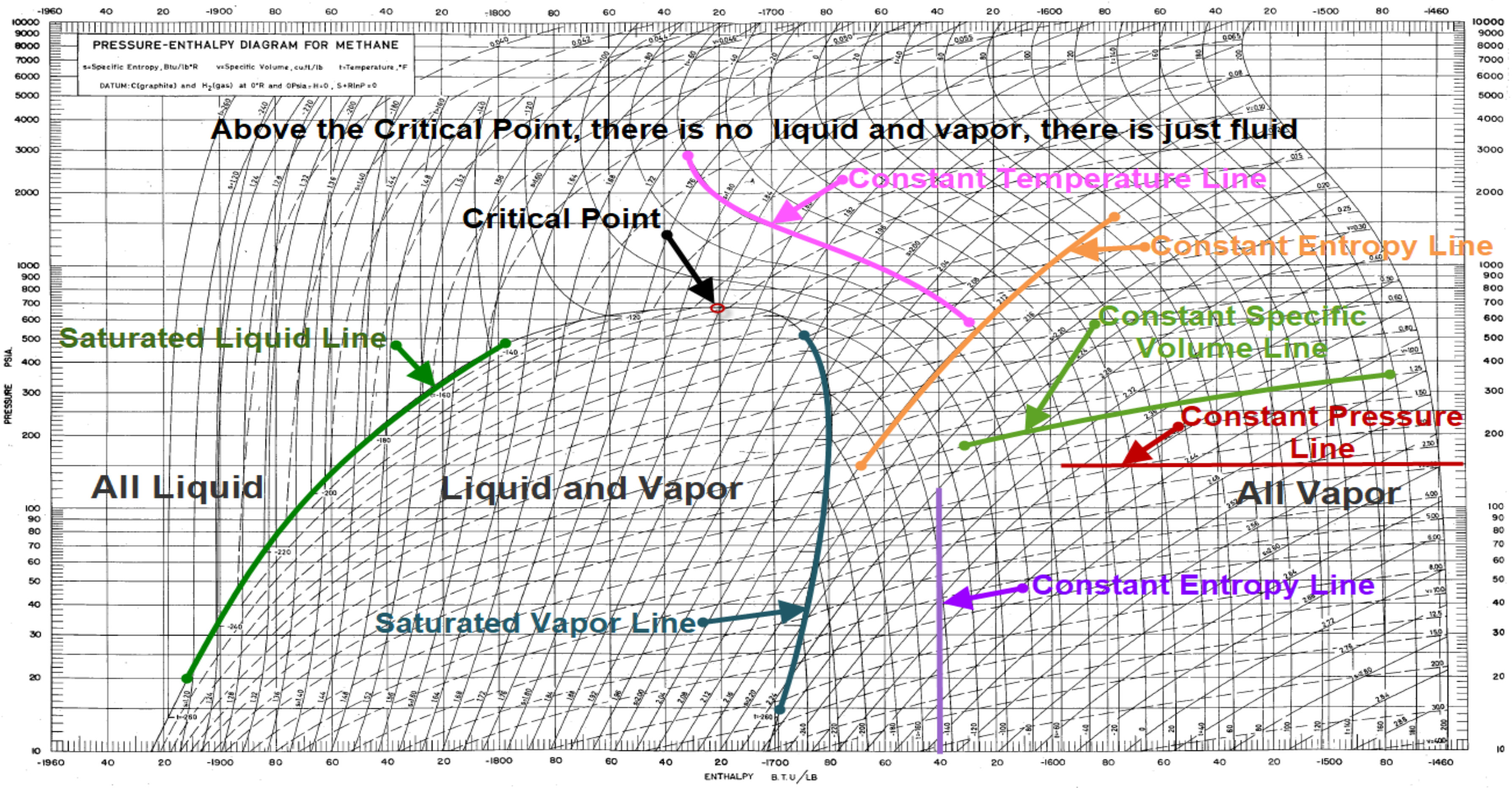

Figure 13: Pressure Enthalpy (PH) Diagram (use accompanying chart to duplicate accurately)

Source: Self-Made

The Conservation of Mass

The concept of conservation of mass as we will apply it to either steady-flow (SF) or steady-state-steady-flow (SSSF) problems means that the mass flowing into a control volume must equal the mass flowing out of the control volume. A *control volume* is the volume created by an imaginary enclosure (control surface) that one places around a device to be studied. This means that, for SSSF problems, "the flow rate of all the fluids that go in is the same as the flow rate of all the fluids that come out!" In equation form this can be written as:

Equation 4: Conservation of mass for a steady state, steady flow process
Mass flow rate in = Mass flow rate out (For SSSF)

Consider a knock out pot in which we separate liquid and vapor. At the inlet to a knock out pot, a valve controls the flow into the pot. For the moment, we will assume that the mass flow rate through the valve and the liquid level in the pot do not change with time. To analyze the knock out pot, first draw a control surface around the knock out pot that creates a control volume.

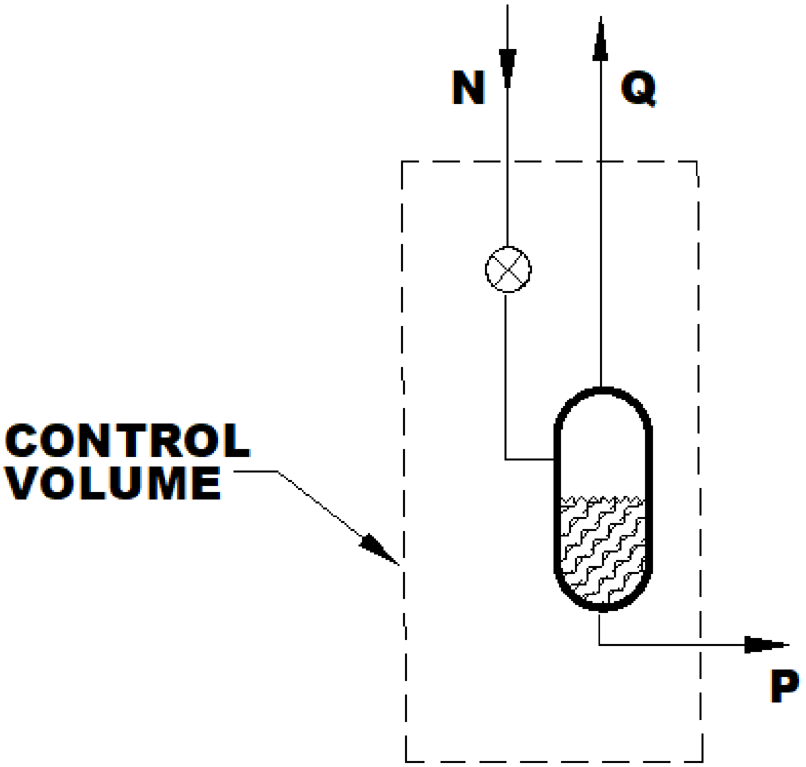

Figure 14: Conservation of mass example

Source: Self-Made

If we assume that the device (everything inside the control volume) is operating at steady-state-steady-flow conditions, we can write the conservation of mass as Equation 4:

Mass flow rate in = Mass flow rate out

For the knock out pot shown above this can be written as:

$$\mathbf{mN^{\circ} = mQ^{\circ} + mP^{\circ}}$$

For this example, if 1000 lbm/hr. enter via (N) and 300 lbm/hr. leaves via (Q) then 700 lbm/hr. must leave from (P).

1000 lbm/hr. = 300 lbm/hr. + 700 lbm/hr.

Although this may seem trivial, it is not, and we will use this concept as we as we learn the first law of thermodynamics and analyze the open expansion liquefaction plant later in this publication

The First Law of Thermodynamics

The word “thermodynamics” can be intimidating to many people, but it is not really complicated. It is something you already understand and use in your everyday life. Here, we will use the first law of thermodynamics in its simplest form—stated slightly differently from the way it is stated in thermodynamics textbooks—but for our cases it will mean the same thing.

The first law of thermodynamics is the conservation of energy. We will look at what are called steady-state-steady-flow (SSSF) problems only. Steady-state means nothing more than that, at every point in the flow, the state of the fluid does not change with time. (“State” can be defined by temperature and pressure.) This means that if we are looking at 200 deg F and 400 psig in front of a valve at one point in time, it is still at that temperature and pressure an hour later. Steady-flow means that the rate of flow is not changing with time. This means that if we are flowing at159 lbm/hr. now, we are still flowing at that rate an hour from now.

Now that we have defined all of our problems as steady-state and steady-flow (SSSF), we can state the SSSF first law of thermodynamics as the conservation of energy. But first we must remember the definition of a “control volume.” A control volume is an imaginary box that defines the limits of what we are studying. If we are studying a valve, then draw a box around the valve. The box you draw will be your control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy

Equation 5: First Law of Thermodynamics for a SSSF process (Conservation of energy)

Energy in = Energy out
This can be written in more detail as:

Equation 6: First Law of Thermodynamics for a SSSF process with more detail

Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out
This can be written symbolically as follows:

Equation 7: First Law of Thermodynamics for a SSSF process written with symbols

$$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}}$$

Equation 8: First Law of Thermodynamics for a SSSF process showing mass quantities

$$\mathbf{m° h_{IN} + m° w_{IN} + m° q_{IN} = m° h_{OUT} + m° w_{OUT} + m° q_{OUT}}$$

Remember that the terms $W°_{IN}$, $Q°_{IN,}$ $W°_{OUT}$, and $Q°_{OUT}$ consists of all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid we would have to the lower-case terms like $w_{IN,}$ $q_{IN,}$ w_{OUT}, q_{OUT}.

Keep in mind the relationship that $W°_{IN} = m° (w_{IN})$ a $Q°_{IN} = m°(q_{IN})$.

Note: The symbol ° above the term [W°,Q°, H° etc.] is to remind us that the term is a “rate” term with units of Btu/hr.

We will define the term “enthalpy” as a measure of the energy in a fluid as compared with some reference point. In the above equations, the total enthalpy (energy associated with total mass flow) is shown as H**°,** and specific enthalpy (energy associated with each lbm of flow) is shown as h. When you refer to Figure 13 or the other pressure enthalpy charts in this publication, don’t be confused by the fact that the enthalpy is shown as negative. It is shown as negative only because of the reference point used for the development of the chart. What really is important is the change in enthalpy as you move through a process and not the actual numerical value of enthalpy.

As an example, let’s say the enthalpy of a fluid changes from -1650 Btu/lbm to -1750 Btu/lbm. This means that the fluid at an enthalpy of -1750 Btu/lbm has less energy in it than the fluid at an enthalpy of -1650 lb/lbm. The reason is that -1750 Btu/lbm is 100 Btu/lbm “more negative” than -1650 Btu/lbm. If it a different reference point had been selected for making the chart, the same temperature and pressure that resulted in and enthalpy of -1750 Btu/lbm, might have been shown as 600 Btu/lbm. However, then, using the new reference point, the same temperature and pressure that had resulted in an enthalpy of -1650, would not be shown as 500 Btu/lbm.
Note that the enthalpy becomes more negative the farther left you are on the chart. Thus, enthalpy increases as you move from the left to the right on the chart.

The remainder of this section is devoted to exercising our newly learned first law of thermodynamics for a SSSF process. Remember: Every time we identify two intensive properties like enthalpy and pressure, we can then look up the temperature and density and entropy on the pressure enthalpy chart. To keep the analysis simple, we will not use this lookup process in the next several examples. In later sections, we will be doing the lookup of all the parameters using the pressure enthalpy chart.

1. Example: Flow Through a Restricted Valve.

Suppose we are flowing 10 lbm/hr. of methane at 400 psig and 80 deg F into a pipe that contains a valve without any work or heat entering or leaving the pipe. Figure 15 shows this. Apply the first law of thermodynamics to this problem and find the energy content of the fluid leaving the pipe. If we look at the pressure enthalpy chart provided accompanying this learning, we can look up the value of the inlet enthalpy as -1535 Btu/lbm. Process of Pressure Drop Across a Valve

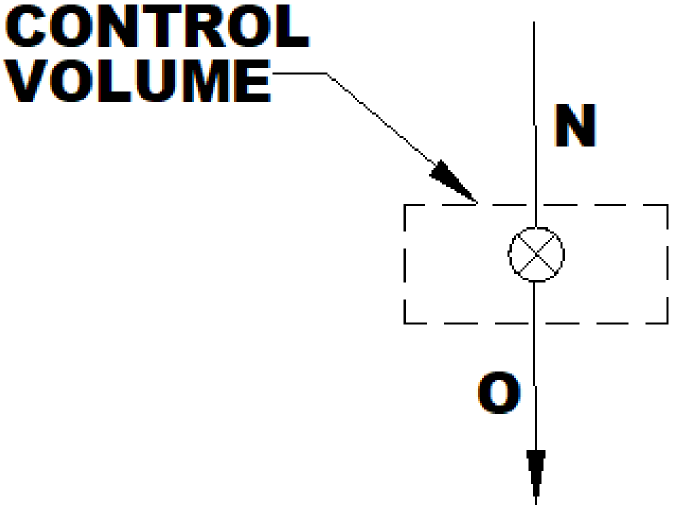

Figure 15: Process of pressure drop across a valve

Source: Self-Made

The amount of fluid flowing is 10 lbm/hr. We will call this m°.
The energy of fluid into the control volume is –1535 Btu/lbm. We will call this h_{IN}
Work into the control volume is "0" (we did no work on the fluid). We will call this $W°_{IN}$.
Heat into the control volume is "0" (we did not transfer any heat to the fluid). We will call this $Q°_{IN}$.

Work out of the control volume is "0" (the fluid did not do any work on the outside world). We will call this $W°_{OUT}$.

Heat out of the control volume is "0" (no heat was transferred to the outside world). We will call this $Q°_{OUT}$.
Energy of fluid leaving the control volume is what we must find. We will call this h_{OUT}.

Now let's plug these values into Equation 5 through 8 and solve for the energy of the fluid out of the control volume.
First Law of Thermodynamics (SSSF) = Conservation of Energy
Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.
This can be written symbolically as follows:
$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ **Eq. 7**

$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:
$m°h_{IN} + W°_{IN} + Q°_{IN} = m°h_{OUT} + W°_{OUT} + Q°_{OUT}$

Remember that the terms $W°_{IN}$, $Q°_{IN,}$ $W°_{OUT}$, and $Q°_{OUT}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used the lower-case terms like $w_{IN,}$ $q_{IN,}$ w_{OUT}, q_{OUT}. **Keep in mind the relationship that $W_{IN} = m° (w_{IN})$ and that $Q_{IN} = m°(q_{IN})$.**

Now let's plug the values into the last equation above.
10 lbm/hr. (-1535 Btu/lbm) + 0 + 0 = 10 lbm/hr. (h_O) + 0 + 0
10 lbm/hr. (-1535 Btu/lbm) = 10 lbm/hr. (h_O)
Divide both sides of the equation by 10 lbm/hr.
h_O = Energy of the fluid out = -1535 Btu/lbm **ANSWER** (energy of fluid leaving)

If you solve this equation by dividing each side of the equation by 10 lbm/hr. (as done above), you find that the energy of the fluid out (O) is exactly the same as the energy of the fluid in (N). This may seem trivial, but this is one of the most important concepts in making liquid methane in an LNG plant. When you flow a fluid across a restricted valve, as long as you do not transfer heat or work across the control surface, the enthalpy of the fluid out stays exactly the same as the enthalpy of the fluid in.

In LNG plants, you may have heard this type of pressure drop across a valve referred to as a *Joule-Thomson pressure drop*. We will see later that the use of a valve (sometimes called a Joule Thomson valve) allows us to convert high-pressure LNG (100% liquid near its boiling temperature at the inlet) into much colder, lower-pressure LNG and gas vapor (liquid and vapor at the outlet).

On each side of the Joule Thomson valve, the fluid energy is the same. When the pressure drop across a valve involves a change of phase from a liquid to a liquid and a vapor, this is called *flashing across a valve*. The term “flashing” means that some of the liquid was flashed (boiled off) into vapor. When flashing occurs, a significant temperature drop occurs. In our open-expansion liquefaction plant, which we will study in Section 17, flashing occurs across the valve just above point N. This flashing occurs as we drop pressure from approximately 400 psig to approximately 15 psig.

In our simplified plant process, a 15-psig outlet was selected to simplify the example (so we could easily examine the thermodynamics of the process). In an actual plant, this flashing occurs in several steps. Accordingly, in a real plant, the liquid that is collected in the pot is then sent to the tank at approximately 60 psig, where it is usually flashed one more time from the plant outlet pressure to the tank pressure.

Liquid LNG may also be flashed in subcoolers, where some of the liquid from the knock-out pot is flashed to 15 psig and perhaps even to 2 psig to subcool the liquid exiting the knock-out pot before sending it to the tank. Subcooling lessens the amount of flash gas that is produced at the tank. (This subcooling is not shown in this publication’s example.)

In warmer gas streams, as in a gate station, we drop from high pressure to low pressure across a valve called a *regulator*. A regulator is a valve that has an automatic operator that modulates the position of the valve to hold a fixed pressure or a fixed flow at its outlet. Again, as above, the energy of the fluid (enthalpy) across the regulator stays the same, but the temperature drops. This is why we need heaters at some of our gate stations. You will see in the next section that, as the gas becomes denser (is colder or at higher pressures), the amount of temperature change for a given pressure drops increases.

Let’s do another problem. This time we’ll study a heat exchanger. First, we’ll study a heat exchanger that removes heat from a flowing fluid. This could be an aftercooler.

2. Example: Flow Through an Aftercooler.

Suppose we have a mass flow rate of minlet° going through the heat exchanger. Let minlet° designate the mass flowrate with the units of lbm/hr. and let minlet designate only the coefficient of that flowrate.

For this problem, let minlet° = 84,680 lbm/hr. and minlet = 84,680. Let the fluid be methane with an inlet enthalpy of -1483 Btu/lbm. Cool the methane in the exchanger by removing 52 Btu from each lbm of methane.

What is the energy (enthalpy) of the fluid leaving the exchanger, and what is the total amount of heat removed per hour?

We start our analysis by drawing a control surface around the devise to be studied.

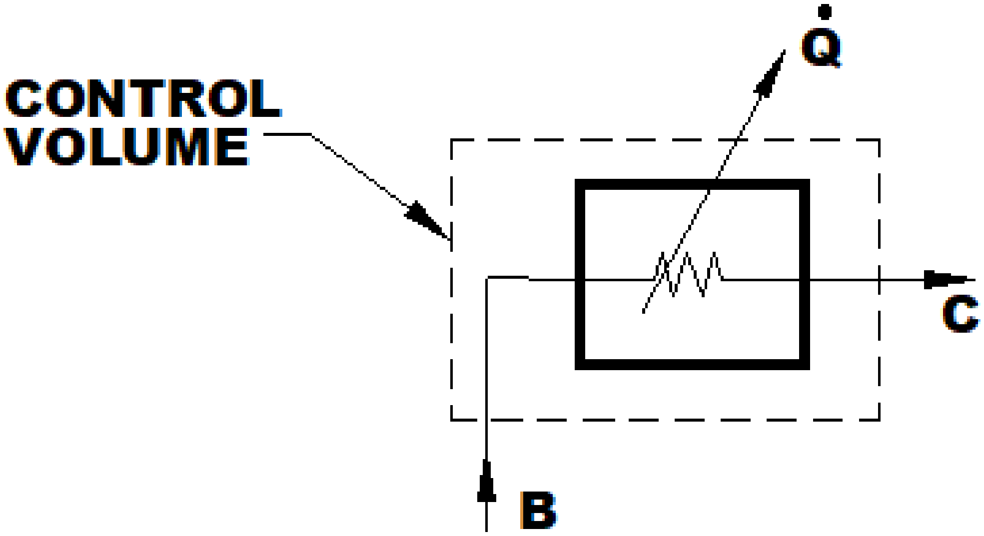

Figure 16: Heat Exchanger Like an Aftercooler

Amount of fluid flowing is minlet° lbm/ hr.

Energy of fluid into the control volume is –1483 Btu/lbm. We will call this h_{IN}.

Work into the control volume is “0” (we did no work on the fluid). We will call this $W°_{IN}$.

Heat into the control volume is "0" (we did not transfer any heat to the fluid). We will call this $Q°_{IN}$.
Work out of the control volume is "0" (the fluid did not do any work on the outside world). We will call this $W°_{OUT}$.

Heat out of the control volume is 52 Btu/lbm (heat was transferred to the outside world). We will call this $q°_{OUT}$. Notice that we are using lower case "q" because this is a heat "per lbm," not total heat out.

Energy of fluid leaving the control volume is what we must find. We will call this h_{OUT}.
Now let's plug these values into equation 5 thorough 8 (repeated here for convenience) and solve for the energy of the fluid out of the control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy
Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.

This can be written symbolically as follows:
$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ **Eq. 7**
$m°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:
$m°h_{IN} + W°_{IN} + Q°_{IN} = m°h_{OUT} + W°_{OUT} + Q°_{OUT}$

Remember that the terms $W°_{IN}$, $Q°_{IN}$, $W°_{OUT}$, and $Q°_{OUT}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used the lower-case terms like w_{IN}, q_{IN}, w_{OUT}, q_{OUT}.

Keep in mind the relationship that $W_{IN} = m°\,(w_{IN})$ and that $Q_{IN} = m°(q_{IN})$.

Now let's plug in the enthalpy values into the last equation above.
$m°\,(-1483\text{ Btu/lbm}) + 0 + 0 = m°\,(h_{OUT}) + 0 + m°\,(52\text{ Btu/lbm})$

m° (-1483 Btu/lbm) = m° (h_{OUT}) + m° (52 Btu/lbm)
Divide both sides of the equation by m°
–1483 Btu/lbm = h_{OUT} + 52 Btu/lbm
h_{OUT} = Energy of the fluid out = -1483 Btu/lbm – 52 Btu/lbm = -1535 Btu/lbm
h_{OUT} = -1535 Btu/lbm **ANSWER** (energy of the fluid leaving)
The total amount of heat removed is the amount of heat removed per pound mass multiplied by the pounds mass flowing per hour.

If m° = 84,680 lbm/hr. then:
$Q°_{OUT}$ = m° (q_{OUT})
$Q°_{OUT}$ = 84,860 lbm/hr. (52 Btu/lbm)
$Q°_{OUT}$ = 4,412,720 Btu/hr. **ANSWER** (total heat removed per hour)

This could also be written as $Q°_{OUT}$ = (52 minlet) Btu/hr.

Note that the enthalpy of the fluid exiting the aftercooler is lower than that of the fluid entering the aftercooler. This means that the energy content of the fluid is less. This makes sense because heat left the fluid. **Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out**.

Let's do another problem and study another heat exchanger. Now we'll study a heat exchanger that adds heat to a fluid flow. This could be a boil-off heater.

3. Example: Flow Through a Boil-Off Heater.

Suppose, we have 900-lbm/hr. of methane at an enthalpy of -1691 Btu/lbm heated by adding a total of 131,400 Btu/hr. to the fluid. What is the energy added to each lbm of fluid, and what is the energy of the fluid leaving the heat exchanger?
We start our analysis by drawing a control surface around the device to be studied.

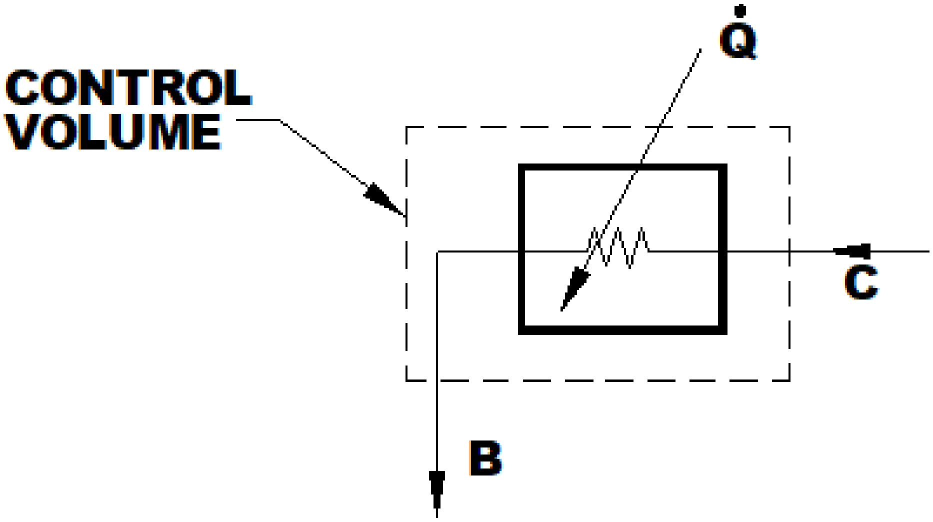

Figure 17: Heat Exchanger Like a Boil-Off Heater

Amount of fluid flowing is 900 lbm/hr. We will call this m°.
Energy of fluid into the control volume is –1691 Btu/lbm. We will call this $h_{IN.}$
Work into the control volume is “0” (we did no work on the fluid). We will call this $W°_{IN}$.
Heat into the control volume is 131,400 Btu/hr. We will call this $Q°_{IN}$.
Work out of the control volume is “0” (the fluid did not do any work on the outside world). We will call this $W°_{OUT}$.
Heat out of the control volume is “0” (heat was not transferred to the outside world). We will call this $Q°_{OUT}$.
Energy of fluid leaving the control volume is what we must find. We will call this h_{OUT}.

Of the two questions asked, the first question is easiest to answer, so we will first compute the amount of energy added to each lbm of fluid.

We have a mass flow rate of 900 lbm/hr. and we are adding 131,400 Btu/hr. to that flow. To find how much heat goes into each pound mass (lbm) per hour, we merely divide the total heat in per hour by the number of pounds mass that flows per hour.

$Q°_{IN} = m° q_{IN}$ What we are looking for is q_{IN}
$q_{IN} = Q°_{IN} / m°$

q_{IN} = 131,400 Btu/hr./900 lbm/hr.
q_{IN} = 146 Btu/lbm **ANSWER** (heat added per lbm per hour)
Now let's plug the values into equations 5 through 8 (repeated here for convenience) and solve for the energy of the fluid out of the control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy
Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.

This can be written symbolically as follows:

$$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT} \quad \textbf{Eq. 7}$$

$$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT} \quad \textbf{Eq. 8}$$

If we merge parts of equation 7 and 8 for our problem we get:

$$m°\ h_{IN} + W°_{IN} + Q°_{IN} = m°\ h_{OUT} + W°_{OUT} + Q°_{OUT}$$

Remember that the terms $W°_{IN}$, $Q°_{IN}$, $W°_{OUT}$, and $Q°_{OUT}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used the lower-case terms like w_{IN}, q_{IN}, w_{OUT}, q_{OUT}.

Keep in mind the relationship that $W_{IN} = m° (w_{IN})$ and that $Q_{IN} = m°(q_{IN})$.

Now let's plug the values into the last equation above.
900 lbm/hr. (-1691 Btu/lbm) + 0 + 131,400 Btu/hr. = 900 lbm/hr. (h_{OUT}) + 0 + 0
900 lbm/hr. (-1691 Btu/lbm) + 131,400 Btu/hr. = 900 lbm/hr. (h_{OUT})

Divide both sides of the equation by 900 lbm/hr.
–1691 Btu/lbm +146 Btu/lbm = h_{OUT}
h_{OUT} = Energy of the fluid out = -1691 Btu/lbm +146 Btu/lbm = -1545 Btu/lbm
h_{OUT} = -1545 Btu/lbm **ANSWER** (energy of the fluid leaving)

Note that the enthalpy of the fluid leaving the control volume is higher than that of the fluid entering the control volume. This means that the energy content of the fluid is greater. This makes sense since heat was added the fluid. **Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out**.

Now let's study a compressor with no heat transfer.

4. Example: Flow through a compressor.

Suppose we have a fluid with an enthalpy of –1541 Btu/lbm being compressed from a low pressure to a high pressure. In doing the compression, we are putting 58 Btu/lbm work energy into the fluid. What is the enthalpy of the fluid leaving? Since we are not saying how many lbm/hr. we are studying, let's do the problem for 1 lbm, and then we can later multiply by the actual flow rate, if necessary.

We start our analysis by drawing a control surface around the device to be studied.

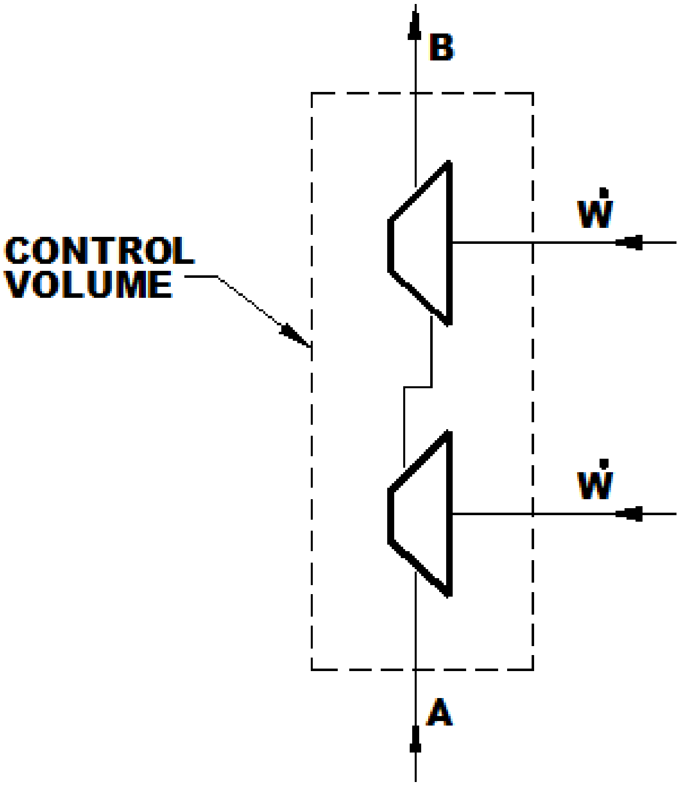

Figure 18: Compressors, typical of some high and low-level plant compressors

Note that we have two work inputs into the control volume. However, since both work inputs enter into the control volume, we will look at the total work entering the control volume as $w°_{IN}$ for work per unit lbm or as $W°_{IN}$ for total work into the control volume.

Amount of fluid flowing is 1 lbm/hr. We will call this m°.
Energy of fluid into the control volume is –1541 Btu/lbm. We will call this $h_{IN.}$
Work into the control volume is 58 Btu/lbm (we did work on the fluid). We will call this $w°_{IN}$.

Notice we are using lower case "w" to designate work in per lbm.
Heat into the control volume is "0" Btu/hr. (we did not transfer any heat to the fluid). We will call this $Q°_{IN}$.

Work out of the control volume is "0" (the fluid did not do any work on the outside world). We will call this $W°_{OUT}$.
Heat out of the control volume is "0" (heat was not transferred to the outside world). We will call this $Q°_{OUT}$.
Energy of fluid leaving the control volume is what we must find. We will call this $h_{OUT.}$

Now, let's plug these values into equation 5 through 8 (repeated here for convenience) and solve for the energy of the fluid out of the control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy
Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.

This can be written symbolically as follows:
$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ **Eq. 7**
$m°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT}$ **Eq. 8**
If we merge parts of equation 7 and 8 for our problem we get:
$m°\, h_{IN} + W°_{IN} + Q°_{IN} = m°\, h_{O} + W°_{OUT} + Q°_{OUT}$

Remember that the terms $W°_{IN}$, $Q°_{IN,}$ $W°_{OUT}$, and $Q°_{OUT,}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used the lower-case terms like w_{IN}, q_{IN}, w_{OUT}, q_{OUT}.

Keep in mind the relationship that $W_{IN} = m°(w_{IN})$ and that $Q_{IN} = m°(q_{IN})$.

Now let's plug in the values into the last equation above.
1 lbm/hr. (-1541 Btu/lbm) + 1 lbm/hr. (58 Btu/lbm) + 0 = 1 lbm/hr. (h_{OUT}) + 0 + 0
1 lbm/hr. (-1541 Btu/lbm) + 1 lbm/hr. (58 Btu/lbm) = 1 lbm/hr. (h_{OUT})
Divide both sides of the equation by 1 lbm/hr.
-1541 Btu/lbm + 58 Btu/lbm = h_{OUT}
h_{OUT} = Energy of the fluid out = -1541 Btu/lbm + 58 Btu/lbm = -1483 Btu/lbm
h_{OUT} = -1483 Btu/lbm **ANSWER** (energy of the fluid leaving)

Now let's study a turbine with no heat transfer.

5. Example: Flow through a turbine (sometimes called and expander).

Suppose we have a fluid with an enthalpy of –1622 Btu/lbm being expanded from a high pressure to a low pressure. In doing the expansion, we are taking 36.25 Btu/lbm work energy out of the fluid. What is the enthalpy of the fluid leaving? Since we did not say how many lbm per hour we are studying, let's do the problem for 1 lbm/hr. and then we can later multiply by the actual flow rate.

We start our analysis by drawing a control surface around the device to be studied.

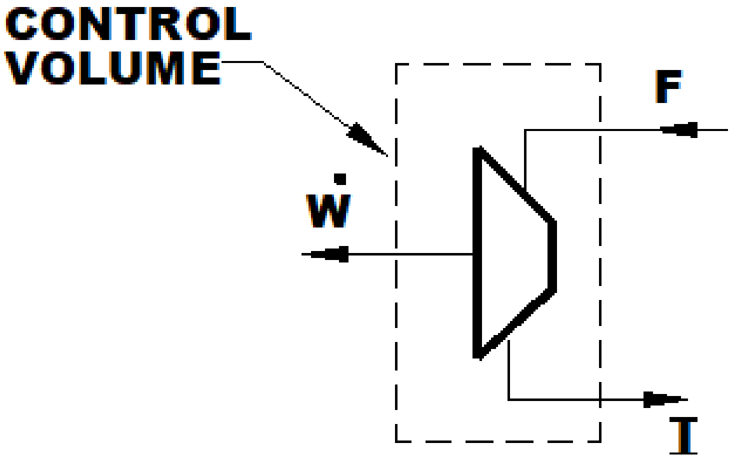

Figure 19: Turbine (Expander), Typical of some high- or low-level expanders

Amount of fluid flowing is 1 lbm/hr. We will call this m°.

Energy of fluid into the control volume is –1622 Btu/lbm. We will call this h_{IN}.

Work into the control volume is "0" (we did no work on the fluid). We will call this $W°_{IN}$.

Heat into the control volume is "0" Btu/hr. (we did not transfer any heat to the fluid). We will call this $Q°_{IN}$

Work out of the control volume is 36.25 Btu/lbm (the fluid did do work on the outside world). We will call this $w°_{OUT}$.

Heat out of the control volume is "0" (heat was not transferred to the outside world). We will call this $Q°_{OUT}$.

Energy of fluid leaving the control volume is what we must find. We will call this h_{OUT}.

Now let's plug these values into equations 5 through 8 and solve for the energy of the fluid out of the control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy
Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.

This can be written symbolically as follows:

$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ **Eq. 7**

$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:

$m°\ h_{IN} + W°_{IN} + Q°_{IN} = m°\ h_{OUT} + W°_{OUT} + Q°_{OUT}$

Remember that the terms $W°_{IN}$, $Q°_{IN}$, $W°_{OUT}$, and $Q°_{OUT}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used the lower-case terms like w_{IN}, q_{IN}, w_{OUT}, q_{OUT}.

Keep in mind the relationship that W_{IN} = m° (w_{IN}) and that Q_{IN} = m°(q_{IN}).

Now let's plug the values into the last equation above.
1 lbm/hr. (-1622 Btu/lbm) + 0 + 0 = 1 lbm/hr. (h_{OUT}) + 1 lbm/hr. (36.25 Btu/lbm) + 0
1 lbm/hr. (-1622 Btu/lbm) = 1 lbm/hr. (h_{OUT}) + 1 lbm/hr. (36.25 Btu/lbm)
Divide both sides of the equation by 1 lbm/hr.
-1622 Btu/lbm = $h_{OUT\,+}$ 36.25 Btu/lbm
h_{OUT} = Energy of the fluid out = -1622 Btu/lbm –36.25 Btu/lbm = -1658.25 Btu/lbm
h_{OUT} = -1658.25 Btu/lbm **ANSWER** (energy of the fluid leaving)

Let's do another problem.

6. Example: Flow through a heat exchanger with multiple flow streams (typical of many plant heat exchangers).

We will study the C heat exchanger below (exchanger with multiple flows). This exchanger transfers heat from one flow stream to another. This is not much different than a boil-off heater or an aftercooler. However, earlier, we did not concern ourselves with the other stream (the air in the case of the aftercooler or the steam in the case of the boil-off heater).

Suppose stream #1 flows 0.2 minlet lbm/hr. with an inlet enthalpy of –1822.5 Btu/lbm and unknown outlet enthalpy. The other stream (we'll call it stream #2) flows 0.0659 minlet lbm/hr. The inlet of stream 2 has an enthalpy of –1691 Btu/lbm and an outlet enthalpy of –1658 Btu/lbm.

Let minlet° designate the mass flowrate with the units of lbm/hr. and let minlet designate only the coefficient of that flowrate (in other words, for this problem, let minlet°= 84,680 lbm/hr. and minlet = 84,680). Find the outlet enthalpy of stream #1.

We start our analysis by drawing a control surface around the device to be studied.

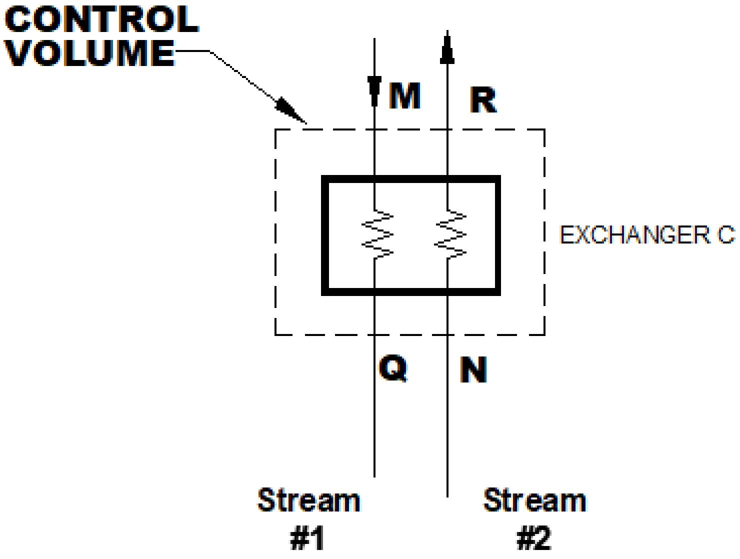

Figure 20: Heat Exchanger with Multiple Streams

Amount of fluid flowing is 0.2 minlet lbm/hr. for stream #1. We will call this $m_1°$.
Amount of fluid flowing is 0.0659 minlet lbm/hr. for stream #2. We will call this $m_2°$.
Energy of fluid #1 into the control volume is –1822.5 Btu/lbm. We will call this $h_{1\ IN}$.
Energy of fluid #2 into the control volume is –1691 Btu/lbm. We will call this $h_{2\ IN}$.
Work into the control volume is “0” (we did no work on the fluid). We will call this $W°_{IN}$.
Heat into the control volume is “0” (we did not transfer any heat to the fluid). We will call this $Q°_{IN}$.

Work out of the control volume is “0” (the fluid did not do any work on the outside world). We will call this $W°_{OUT}$.
Heat out of the control volume is “0” (heat was not transferred to the outside world). We will call this $Q°_{OUT}$
Energy of fluid (1) leaving the control volume is unknown. We will call this $h_{1\ OUT}$.
Energy of fluid (2) leaving the control volume is –1658 Btu/lbm. We will call this $h_{2\ OUT}$.

Now, let’s plug these values into equation 5 through 8 and solve for the energy of the fluid out of the control volume.

First Law of Thermodynamics (SSSF) = Conservation of Energy

Energy in = Energy out

This can be written in more detail as:
Energy of the fluid in + work in + heat in = Energy of fluid out + work out + heat out.

This can be written symbolically as follows:
$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ **Eq. 7**
$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}$ **Eq. 8**
If we merge parts of equation 7 and 8 for our problem we get:
$m_1°\ h_{1IN} + m_2°\ h_{2\ IN} + W°_{IN} + Q°_{IN} = m_1°\ h_{1\ OUT} + m_2°\ h_{2\ OUT} + W°_{OUT} + Q°_{OUT}$

Remember that the terms $W°_{IN}$, $Q°_{IN}$, $W°_{OUT}$, and $Q°_{OUT}$ are all total flow terms. This means that they refer to the work and heat added to or taken away from the total system (all the mass flowing in each hour) and not to each lbm of mass of the problem. If we wanted to refer to the amount of heat or work that was added to or taken away from each pound mass of fluid, we would have used lower case terms like w_{IN}, q_{IN}, w_{OUT}, q_{OUT}
Keep in mind the relationship that $W_{IN} = m° (w_{IN})$ and that $Q_{IN} = m°(q_{IN})$.

Now let's plug the values into the last equation above.
0.2 minlet lbm/hr. (-1822.5 Btu/lbm) + 0.0659 minlet lbm/hr. (-1691 Btu/lbm) + 0 + 0 = 0.2 minlet lbm/hr. ($h_{1\ OUT}$) + 0.0659 minlet lbm/hr. (-1658 Btu/lbm) + 0 + 0
-364.500 Btu/hr. –111.437 Btu/hr. = 0.2 lbm/hr. ($h_{1\ OUT}$) –109.262 Btu/hr.
-475.937 Btu/hr. = 0.2 lbm/hr. ($h_{1\ OUT}$) – 109.262 Btu/hr.
-475.937 Btu/hr. + 109.262 Btu/hr. = 0.2 lbm/hr. ($h_{1\ OUT}$)
-366.675 Btu/hr. = 0.2 lbm/hr. ($h_{1\ OUT}$)
-366.675 Btu/hr. / 0.2 lbm/hr. = ($h_{1\ OUT}$)
$h_{1\ OUT}$ = -1833.375 Btu/lbm **ANSWER** (energy of the fluid #1 leaving)

Note that the enthalpy of fluid #1 is lower at the outlet of the exchanger. This means that the energy content of the fluid is lower. This makes sense since in this exchanger we are subcooling the liquid LNG before it is flashed across the JT valve (shown in the example of the open gate station expansion liquefaction plant analyzed later in this learning (section 17)).

In Summary, the energy of the fluid in + work in + heat in = the energy of fluid out + work out + heat out (first law of thermodynamics for a SSSF process).

Now we have studied a valve, an aftercooler, a boil-off heater, a compressor, a turbine, and a multi-stream heat exchanger. We have all the basic components to understand the workings of a liquid methane plant.

Pressure Enthalpy Chart Applications

For this section, refer to the pressure enthalpy charts shown in Figures 21 and 22, at the end of the section.

Let's start by taking a pound mass of methane at 80 deg F and 400 psig. Note that we said 400 psig (gauge). That is actually 415 psia (absolute).

On the pressure enthalpy chart in Figures 21, look at the vertical axis at the far left or far right of the chart. You can see that this is the absolute pressure axis. Estimate where a line at 415 psia would be, and mark it on the psia axis. Follow the horizontal line of 415 psia until you get to where it intersects the 80 deg F temperature curved line. The temperature lines are the lines that start at the bottom of the chart, and as you follow them upward, they curve to the left and then start to move straight up again. Place a pencil mark at the intersection point where 415 psia and 80 deg F coincide. This is labeled as point C on the chart.

Now read the enthalpy from the horizontal axis at the top or bottom of the chart. You need to estimate the number by eyeing where the point lies. It lies between –1520 Btu/lbm and –1540 Btu/lbm. We "eyeball" it to read around –1535 Btu/lbm. We can also read the specific volume at this point by reading the nearly horizontal dashed lines. The specific volume is read as being between 0.8 and 1.0 cu.ft./lbm. We estimate it by eyeing the chart to be approximately 0.84 cu.ft./lbm. With this specific volume and equation 2, we can compute the density, which is 1.19 lbm/cu.ft.. We can also read the entropy to be approximately 2.346 Btu/lbm deg R. The entropy lines are the solid lines on the chart that traverse almost diagonally from the lower left to the upper right part of the chart. We have now identified a point on the chart and have determined many of the properties associated with the state of the fluid.

With this point identified we can do many things that are exercised in the next section. These include the following processes, all of which start at the point defined immediately above. Figures 21 and 22 show the paths for the first 5 processes.

We can do a constant pressure heat removal as in an aftercooler. Let's say we were to remove 95 Btu/lbm. To find the new state, we would look at a new point at 415 psia pressure and –1630 Btu/lbm enthalpy (–1535 Btu/lbm –95 Btu/lbm = –1630 Btu/lbm). This is point D on Figures 21 and 22.

Or we could do a constant pressure heat addition as in a gas heater. Let's say we added 52 Btu/lbm. To find the new state, we would look at a new point at 415 psia pressure and –1483 Btu/lbm enthalpy (–1535 Btu/lbm + 52 Btu/lbm = –1483 Btu/lbm). This is point B on Figures 21 and 22.

Or we could do a constant enthalpy pressure reduction as in a Joule-Thomson valve. Let's say we expanded via constant enthalpy to a new pressure of 105 psia. To find the new state, we would look at a new point at an enthalpy of –1535 Btu/lbm and a pressure of 105 psia. This is point E on Figures 21 and 22.

Or we could do a constant entropy expansion without heat transfer as in a hypothetical turbine. Let's say we expanded via an imaginary* constant entropy process to 105 psia. To find the new state, we would look at a new point at 105 psia pressure and 2.346 Btu/lbm degree R entropy. This is point F on Figures 21 and 22.

Or we could do a constant entropy compression without heat transfer as in a hypothetical compressor. Let's say we compress via an imaginary* constant entropy process to 605 psia. To find the new state, we would look at a new point at 605 psia pressure and 2.346 Btu/lbm degree R entropy. This is point G on Figures 21 and 22.

Or we could do any one of many multiple path processes which will result in a new end state that can be identified by two intensive properties.

* You will learn more about why we have called this "imaginary" in Section 16.

There are many processes that could be studied, but the first five above are all that will be needed to understand the basic thermodynamics of our LNG plants.

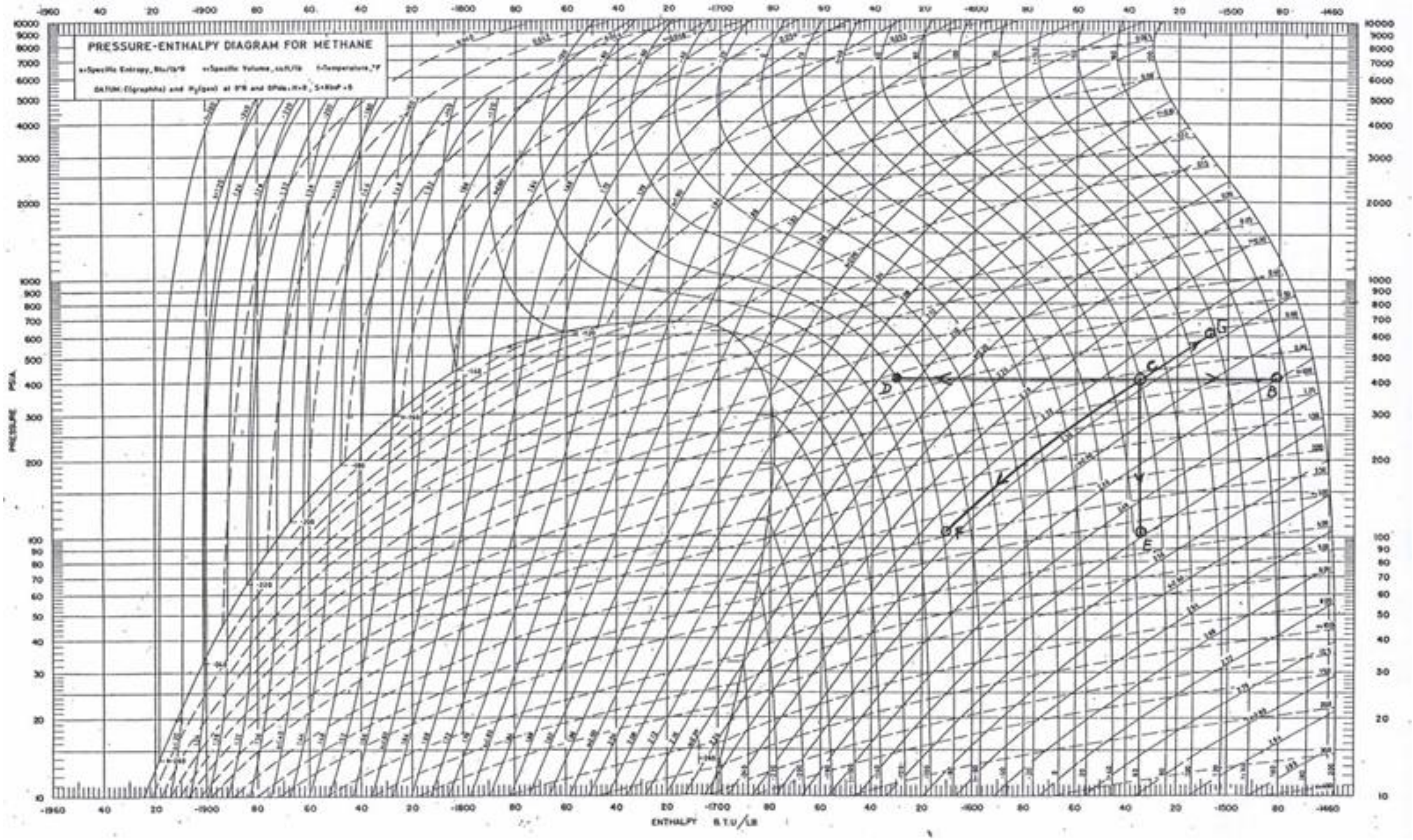

Figure 21: Pressure Enthalpy Paths (use accompanying chart to duplicate accurately)

Source: Self-made curves over footnote 1 table

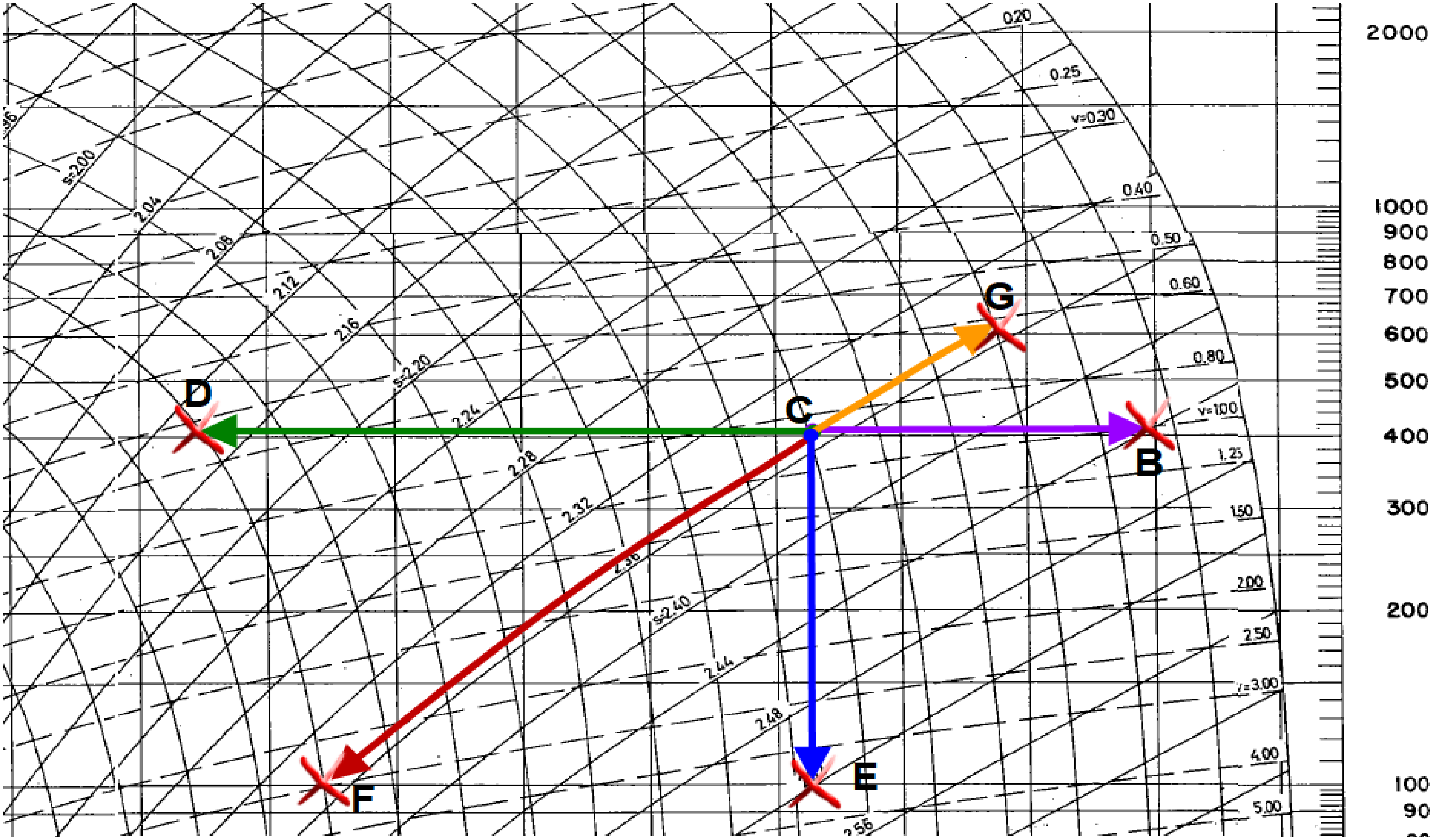

Figure 22: Pressure Enthalpy Paths (closer look at Figure 21) (use accompanying chart to duplicate accurately)

Source: Self-made curves over footnote 1 table

Using a Pressure Enthalpy Chart for Everyday Problems—Examples

For this section, refer to the pressure enthalpy chart accompanying this learning and referenced in footnote 1.

1. Example: Constant pressure heat removal

Let’s do a Constant Pressure Heat Removal process (called an isobaric process). Take 1 lbm/hr. of methane at 170 deg F and 400 psig and remove 52 Btu/lbm from it while maintaining the pressure at 400 psig. List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet and outlet of this heat exchanger.

We start our analysis by drawing a control surface around the device to be studied.

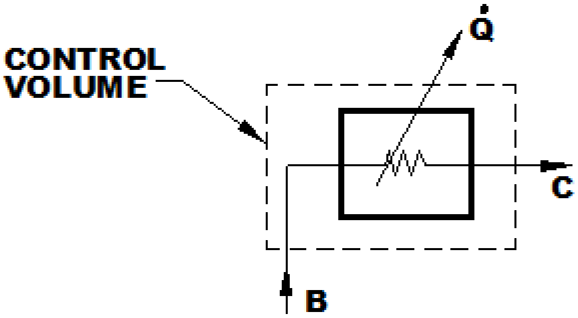

Figure 23: Constant Pressure Heat Removal

Source: Self-made

Note that we said 400 psig (gauge). That is actually 415 psia (absolute). Take out the pressure enthalpy chart (accompanying this course but also shown in Figure 29). Look at the vertical axis at the far left or far right of the chart. You can see that this is the absolute pressure axis. Follow the horizontal line of 415 psia until you get to the point at which it intersects the 170 deg F temperature curved line. The temperature lines are the lines that start at the bottom of the chart, and as you follow them upward, they curve to the left and then start to move straight up again. Place a pencil mark at the intersection point where 415 psia and 170 deg F coincide. Label this as point B on the chart.

Now read the enthalpy from the horizontal axis. You need to estimate the number by eyeing where the point lies. It lies between –1500 Btu/lbm and –1480 Btu/lbm. We “eyeball” it to read about –1483 Btu/lbm. We can make a similar reading for specific volume and entropy. We can read the specific volume of this mass flow by reading the nearly horizontal dashed lines. The specific volume is read as being 1.0 cu.ft./lbm (point B appears to fall right on the 1.0 cu.ft./lbm line). According to Equation 2, the density is nothing more than 1/specific volume, so the density is 1 lbm/cu.ft.. We can also read the entropy to be approx. 2.436 Btu/lbm deg R.

Now let's hold the pressure constant at 415 psia and cool the gas by removing 52 Btu/lbm of heat from the gas. We can apply the first law of thermodynamics as follows:

$$\mathbf{H^\circ_{IN} + W^\circ_{IN} + Q^\circ_{IN} = H^\circ_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^\circ\, h_{IN} + m^\circ\, w_{IN} + m^\circ\, q_{IN} = m^\circ\, h_{OUT} + m^\circ\, w_{OUT} + m^\circ\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of equations 7 and 8 for our problem we get:
$m^\circ h_{IN} + W^\circ_{IN} + Q^\circ_{IN} = m^\circ h_{OUT} + W^\circ_{OUT} + m^\circ\, q_{OUT}$
1 lbm/hr. (-1483 Btu/lbm) + 0 + 0 = 1 lbm/hr.(h_{OUT}) + 0 + 1 lbm/hr. (52 Btu/lbm)
h_{OUT} = -1483 Btu/lbm – 52 Btu/lbm
h_{OUT} = -1535 Btu/lbm (At point C)

The new enthalpy at point C is –1535 Btu/lbm (-1483 Btu/lbm – 52 Btu/lbm). What is our new temperature, new specific volume, and new entropy? Look at the chart in Figure 29 where 415 psia intersects with an enthalpy of –1535 Btu/lbm. Place a new pencil mark at this point, and label it as point C. At the new point, we read the temperature to be approximately 80 deg F and the new specific volume as approximately 0.84 cu.ft./lbm, and the new entropy as approximately 2.346 Btu/lbm deg R. Using the new specific volume, we can then compute the new density of 1.19 lbm/cu.ft. via Equation 2.

ANSWERS:

At Point B

Initial pressure	415 psia
Initial temperature	170 deg F
Initial enthalpy	–1483 Btu/lbm
Initial specific volume	1 cu.ft./lbm
Initial density	1 lbm/cu.ft.
Initial entropy	2.436 Btu/lbm deg R

At Point C

Final pressure	415 psia
Final temperature	80 deg F

Final enthalpy	–1535 Btu/lbm
Final specific volume	0.84 cu.ft./lbm
Final density	1.19 lbm/cu.ft.
Final entropy	2.346 Btu/lbm deg R

2. Example: Constant pressure heat removal process

Let’s do another Constant Pressure Heat Removal process (called an isobaric process). Take 1 lbm/hr. of methane at 80 deg F and 400 psig and remove 95 Btu/lbm from it at while maintaining the pressure at 400 psig. List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet and the outlet of this heat exchanger.

We start our analysis by drawing a control surface around the device to be studied.

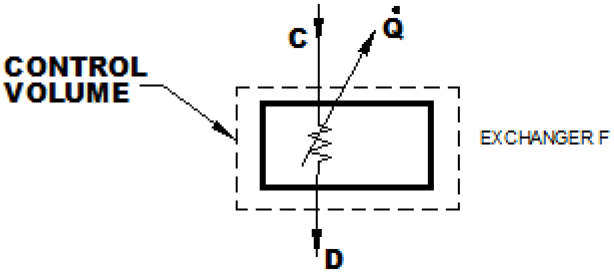

Figure 24: Constant Pressure Heat Removal (See plant example problem later in this learning.)

Source: Self-made

Note that, in Figure 20, we pointed out that in the C exchanger, heat was transferred between streams rather than to the “outside world” (air or steam). The F exchanger studied in this example also transfers heat between streams. However, for purposes of this example, we examine only one-half of the F exchanger (from the plant shown later in this publication). To simplify the analysis, we assume that heat from the single stream is transferred to air rather than to a second stream. Later, we will analyze this F exchanger more rigorously.

Note that we said 400 psig (gauge). That is actually 415 psia (absolute). Take out the pressure enthalpy chart. Look at the vertical axis at the far left or far right of the chart. You can see that this is the absolute pressure axis. Follow the horizontal line of 415 psia till you get to the point at which it intersects the 80 deg F temperature curved line. The temperature lines are the lines that start at the bottom of the chart, and as you follow them upward, they curve to the left and then start to move straight up again. Place a pencil mark at the intersection point where 415 psia and 80 deg F coincide. Label this as point C.

Now read the enthalpy from the horizontal axis. You need to estimate the number by eyeing where the point lies. It lies between –1520 Btu/lbm and –1540 Btu/lbm. We "eyeball" it to read about –1535 Btu/lbm. We can make a similar reading for specific volume and entropy. We read the specific volume of this mass by reading the nearly horizontal dashed lines. The specific volume is read as being between 0.8 and 1.0 cu.ft./lbm. We estimate it by eyeing the chart to be approximately 0.84 cu.ft./ lbm. According to Equation 2, the density is nothing more than 1/specific volume, so the density is 1.19 lbm/cu.ft.. We can also read the entropy to be approximately 2.346 Btu/lbm deg R.

Now let's hold the pressure constant at 415 psia and cool the gas by removing 95 Btu/lbm of heat from the gas. We can apply the first law of thermodynamics as follows:

$$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + m^{\circ}\, q_{IN} = m^{\circ}\, h_{OUT} + m^{\circ}\, w_{OUT} + m^{\circ}\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of equation 7 and 8 for our problem we get:

$m^{\circ}h_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = m^{\circ}h_{OUT} + W^{\circ}_{OUT} + m^{\circ}\, q_{OUT}$

1 lbm/hr. (-1535Btu/lbm) + 0 + 0 = 1 lbm/hr. (h_{OUT}) + 0 +1 lbm/hr. (95 Btu/lbm)

h_{OUT} = -1535 Btu/lbm – 95 Btu/lbm

h_{OUT} = -1630 Btu/lbm (At Point D)

The new enthalpy at point D is –1630 Btu/lbm (-1535 Btu/lbm – 95 Btu/lbm). What is the new temperature, the new specific volume, and the new entropy? Look at the chart where 415 psia pressure intersects with an enthalpy of –1630 Btu/lbm. Place a new pencil mark at this point. At the new point, we read the temperature to be approximately –80 deg F and the new specific volume as approximately 0.51 cu.ft./lbm, and the new entropy as approximately 2.135 Btu/lbm deg R. Using the new specific volume, we then compute the new density as 1.96 lbm/cu.ft. via Equation 2.

ANSWERS:

At point C

Initial pressure	415 psia
Initial temperature	80 deg F

Initial enthalpy	–1535 Btu/lbm
Initial specific volume	0.84 cu.ft./lbm
Initial density	1.19 lbm/cu.ft.
Initial entropy	2.346 Btu/lbm deg R

At point D

Final pressure	415 psia
Final temperature	80 deg F
Final enthalpy	–1630 Btu/lbm
Final specific volume	0.51 cu.ft./lbm
Final density	1.96 lbm/cu.ft.
Final entropy	2.135 Btu/lbm deg R

3. Example: Constant enthalpy expansion

Let’s do a Constant Enthalpy Expansion of 1 lbm/hr. from 400 psig (415 psia) and 80 deg F (point C on the chart) to a pressure of 100 psi (115psia). List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet an outlet of the valve.

We start our analysis by drawing a control surface around the device to be studied.

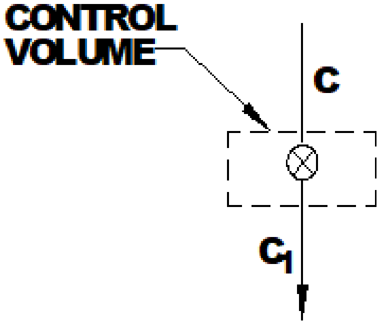

Figure 25: Constant Enthalpy Expansion

Source: Self-made

When we let pressure down through a valve or a regulator, we do no work and we exchange no heat, so by the first law of thermodynamics (Equations 7 and 8) the enthalpy stays constant. If we now drop from 415 psia and 80 deg F to 115 psia (100 psig), we get a new temperature, a new specific volume, and a new entropy**, but the enthalpy remains unchanged**.

This is exactly what happens in a gate station when pressure is dropped across a regulator. Gas operators use a rule of thumb that says *for every 100-psig decrease in pressure we drop approximately 7 deg F*. In our example (415 psia to 115 psia), we can read off the chart the actual temperature drop. At 415 psia and 80 deg F (shown as point C on the chart), we read the enthalpy from the chart as being –1535 Btu/lbm. For that same point, we read the specific volume to be 0.84 cu.ft./lbm. According to Equation 2, the density is nothing more than 1/specific volume, so the density is 1.19 lbm/cu.ft.. We can also read the entropy to be approximately 2.346 Btu/lbm deg R.

Now let's reduce the pressure from 415 psia to 115 psia without heat transfer or work crossing the control surface. We can apply the first law of thermodynamics as follows:

$\mathbf{H^\circ_{IN} + W^\circ_{IN} + Q^\circ_{IN} = H^\circ_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT}}$ **Eq. 7**

$\mathbf{m^\circ\, h_{IN} + m^\circ\, w_{IN} + m^\circ\, q_{IN} = m^\circ\, h_{OUT} + m^\circ\, w_{OUT} + m^\circ\, q_{OUT}}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:

$m^\circ h_{IN} + W^\circ_{IN} + Q^\circ_{IN} = m^\circ h_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT}$

1 lbm/hr. (-1535 Btu/lbm) + 0 + 0 = 1 lbm/hr. (h_{OUT}) + 0 + 0

h_{OUT} = -1535 Btu/lbm (at Point C1)

We know from the earlier studies in this publication and from the above exercise that after a constant enthalpy pressure drop from 415 psia to 115 psia, the enthalpy remains unchanged. In our case, it remains at –1535 Btu/lbm. If we now read the chart for an enthalpy of –1535 Btu/lbm and a pressure of 115 psia, we can place a pencil mark at the new point C1. At point C1, we read the new temperature to be approximately 65 deg F and the new specific volume to be 3 cu.ft./lbm, and a new entropy to be 2.505 Btu/lbm deg R. Using the new specific volume, we then compute the new density of 0.333 lbm/cu.ft. via Equation 2.

Note that, for the 300 psi drop in pressure, the gas decreased 15 deg F via a constant enthalpy process. That is about 5 deg F for every 100-psi pressure drop—which means that our rule of thumb (7 deg F per 100 psi) is not very accurate.

ANSWERS:

At Point C

Initial pressure	415 psia
Initial temperature	80 deg F
Initial enthalpy	–1535 Btu/lbm
Initial specific volume	0.84 cu.ft./lbm

Initial density	1.19 lbm/cu.ft.
Initial entropy	2.346 Btu/lbm deg R

At Point C1

Final pressure	115 psia
Final temperature	65 deg F
Final enthalpy	–1535 Btu/lbm
Final specific volume	3 cu.ft./lbm
Final density	0.333 lbm/cu.ft.
Final entropy	2.505 Btu/lbm deg

4. Example: Constant enthalpy expansion

Let’s do the same Constant Enthalpy Expansion of 1 lbm/hr., but this time we will use our point D that was at 415 psia and –80 deg F (enthalpy of –1630 Btu/lbm) as our starting point. Let’s drop the pressure from 415 psia (400 psig) to 115 psia (100 psig). List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet and outlet of the valve.

We start our analysis by drawing a control surface around the device to be studied.

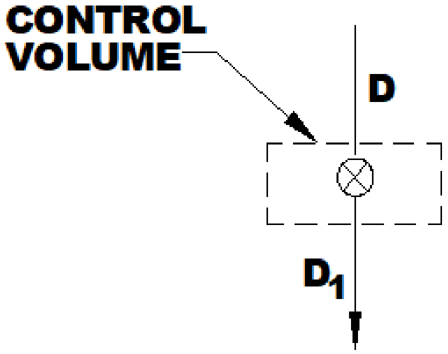

Figure 26: Constant Enthalpy Expansion

Source: Self-made

In our earlier Figure 24 we looked up the properties of point D on the chart shown in Figure 29 (415 psia and –80 deg F).

Now let’s reduce the pressure from 415 psia to 115 psia without heat transfer or work crossing the control surface. We can apply the first law of thermodynamics as follows:

$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ Eq. 7
$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}$ Eq. 8

If we merge parts of equation 7 and 8 for our problem we get:
$m°h_{IN} + W°_{IN} + Q°_{IN} = m°h_{OUT} + W°_{OUT} + Q°_{OUT}$
1 lbm/hr. (-1630 Btu/lbm) + 0 + 0 = 1 lbm/hr. (h_{OUT}) + 0 + 0
$h_{OUT} =$ -1630 Btu/lbm (at Point D1)

We know from the earlier studies we did here and from the above exercise that after a constant enthalpy pressure drop from 415 psia to 115 psia, the enthalpy remains unchanged. In our case, it remains at –1630 Btu/lbm. If we now read the chart in Figure 29 for an enthalpy of –1630 Btu/lbm and a pressure of 115 psia, we can place a pencil mark at point D1. At point D1 we read the new temperature to be about –115 deg F and the new specific volume to be 1.8 cu.ft./lbm, and a new entropy of 2.275 Btu/lbm deg R. Using the new specific volume, we then compute the new density as 0.555 lbm/cu.ft. via Equation 2.

Note that for the 300 psi drop in pressure in this example the gas decreased 35 deg F in temperature via a constant enthalpy pressure drop. That is about 12 deg F for every 100-psi pressure drop. This means that our rule of thumb (7 deg F temperature drop per 100 psi pressure drop) is not very accurate. This is significantly different than the earlier finding in Example 10.3.

In Example 10.3, we started at 80 deg F and dropped 5 deg for every 100-psig pressure drop, and in Example 10.4, we started at –80 deg F and dropped 12 deg F for every 100-psig pressure drop. Why is it a different temperature drop for the same pressure drop? The answer is that the denser the gas, the less it behaves like an ideal gas. An ideal gas does not drop in temperature when it undergoes a constant enthalpy pressure drop. However, in a denser gas (higher pressure or colder temperature or both), molecular interaction causes the gas to deviate from ideal gas behavior. In short, the closer one is to the lower right-hand corner of the pressure enthalpy chart (where density is lowest), the less temperature drop one experiences in a constant enthalpy pressure drop (the more the gas acts like an ideal gas). This is readily seen by the fact that the temperature lines are nearly vertical at the lower right side of the pressure enthalpy chart. In keeping with the limited scope of this publication, we will just accept this as fact without further discussion.

ANSWERS:

At Point D

Initial pressure	415 psia
Initial temperature	–80 deg F
Initial enthalpy	–1630 Btu/lbm
Initial specific volume	0.51 cu.ft./lbm
Initial density	1.96 lbm/cu.ft.
Initial entropy	2.135 Btu/lbm deg R

At Point D1

Final pressure	115 psia
Final temperature	–115 deg F
Final enthalpy	–1630 Btu/lbm
Final specific volume	1.8 cu.ft./lbm
Final density	0.555 lbm/cu.ft.
Final entropy	2.275 Btu/lbm deg R

5. Example: Constant enthalpy expansion

Before we end the topic of constant enthalpy expansion, we must do one special type of problem. This is a Constant Enthalpy Expansion with Phase Change. This type of expansion occurs in all standard home refrigerators and in LNG plants (across the Joule-Thomson valve). Let’s take liquid methane at 415 psia and –165 deg F (point N on the chart in Figure 29) and expand 1 lbm/hr. across a Joule Thomson valve to a new pressure of 30 psia.). List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet and outlet of the valve. Also list the mass flow of liquid and the mass flow of vapor that exits the control volume.

We start our analysis by drawing a control surface around the device to be studied.

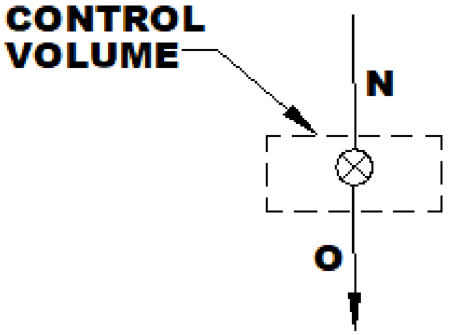

Figure 27: Constant Enthalpy Expansion with Phase Change

On the chart, we can see that the methane at point N is 100% liquid, and we can read the enthalpy of point N to be –1833 Btu/lbm. At the same point, we read the specific volume to be 0.048 cu.ft./lbm and the entropy to be 1.5 Btu/lbm deg R. According to Equation 2, the density is nothing more than 1/specific volume, so the density is computed to be 20.83 lbm/cu.ft..

Now let's reduce the pressure from 415 psia to 30 psia without heat transfer or work crossing the control surface. We can apply the first law of thermodynamics as follows:

$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ Eq. 7

$m°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT}$ Eq. 8

If we merge parts of Equations 7 and 8 for our problem, we get:

$m°h_{IN} + W°_{IN} + Q°_{IN} = m°h_{OUT} + W°_{OUT} + Q°_{OUT}$

1 lbm/hr. (-1833 Btu/lbm) + 0 + 0 = 1 lbm/hr. (h_{OUT}) + 0 + 0

h_{OUT} = -1833 Btu/lbm (At point O)

If we expand down to 30 psia, the enthalpy remains at –1833 Btu/lbm. With these two intensive properties, we can place a pencil mark at point O on the chart. From this point, we can read that the new temperature is –242 deg F and we can see that we are now within the dome on the chart—which means that we have a mixture of vapor and liquid. The liquid at –242 deg F and 30 psia must have an enthalpy of –1903Btu/lbm (read this off the liquid line at 30 psia), and the vapor must have an enthalpy of –1691 Btu/lbm (read this off the vapor line at 30 psia). To find out how much is liquid and how much is vapor, we simply apply the conservation of mass, as shown in Section 12 and the first law of thermodynamics as shown in Section 13.

mass flow rate in = mass flow rate out (For SSSF)

1 lbm/hr. (into valve) = (x) lbm/hr. (liquid out of valve) + (1-x) lbm/hr. (vapor out of valve)

where x = mass flow of liquid (lbm/hr.)

and

(1-x) = mass flow of vapor (lbm/hr.)

$$m^\circ h_{IN} + m^\circ w_{IN} + m^\circ q_{IN} = m^\circ h_{OUT} + m^\circ w_{OUT} + m^\circ q_{OUT} \qquad \textbf{Eq. 8}$$

Rewriting this in its liquid and vapor components we can write:

$$m^\circ h_{IN} + 0 + 0 = m^\circ_{liq}\, h_{OUTliq} + m^\circ_{vapor}\, h_{OUTvapor} + 0 + 0$$

1 lbm/hr. (-1833 Btu/lbm) = x lbm/hr. (-1903Btu/lbm) + (1-x) lbm/hr. (-1691 Btu/lbm)

where x is the mass flow of liquid (lbm/hr.)
-1833 Btu/hr. = -1903(x) Btu/hr. - 1691 Btu/hr. + 1691(x) Btu/hr.

The units can be all canceled out and we can solely solve for the coefficient (x).
-1833 + 1691 = (-1903 + 1691) (x)
-142 = -212(x)
212 (x) = 142
x = 0.67
1 – x = 0.33

From this we find that x = 0.67 lbm/hr. (mass flow rate of Liquid)
From this we find that 1-x = 1-0.67 lbm/hr. = 0.33 lbm/hr. (mass flow rate of Vapor)

ANSWERS:
At Point N

Initial mass flow	1 lbm/hr.
Initial pressure	415 psia
Initial temperature	–165 deg F
Initial enthalpy	–1833 Btu/lbm
Initial specific volume	0.048 cu.ft./lbm
Initial density	20.83 lbm/cu.ft.
Initial entropy	1.5 Btu/lbm deg R

At Point O (Liquid portion – see point P on the chart)

Final mass flow of liquid	0.67 lbm/hr.
Final pressure liquid	30 psia
Final temperature liquid	–242 deg F
Final enthalpy liquid	–1903 Btu/lbm
Final specific volume liquid	0.038 cu.ft./lbm
Final density liquid	26.32 lbm/cu.ft.
Final entropy liquid	1.24 Btu/lbm deg R

At Point O (Vapor portion – see point Q on the chart)

Final mass flow of vapor	0.33 lbm/hr.
Final pressure vapor	30 psia
Final temperature vapor	–242 deg F
Final enthalpy vapor	–1691 Btu/lbm
Final specific volume vapor	4.65 cu.ft./lbm
Final density vapor	0.215 lbm/cu.ft.
Final entropy vapor	2.21 Btu/lbm deg R

6. Example: Isentropic and real-world expansion

Next, let’s work with a turbine that expands 1 lbm/hr. of methane from 105 psia and –100 deg F to 30 psia via an Expansion Without Heat Transfer process. List the pressure, temperature, enthalpy, specific volume, density, and entropy at both the inlet and outlet of the turbine for both the adiabatic isentropic case and the actual case. In the actual case, use an end state temperature of –180 deg F.

The word *adiabatic* means without heat transfer and the word *isentropic* means having no change in entropy. We will talk more about constant entropy expansion processes without heat transfer in the next section. Such a process does not really exist in nature because it is an imaginary, ideal, perfectly efficient process. We do, however, study such imaginary processes in order to measure the efficiency and to know the limits of our real processes.

We start our analysis by drawing a control surface around the device to be studied.

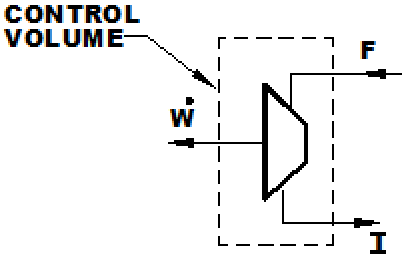

Figure 28: Expansion through a turbine without heat transfer

Constant Entropy Expansion Without Heat Transfer
actual (path F to I) and adiabatic isentropic (path F to I_S).

The first law of thermodynamics for this problem reads as follows:

$$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT} \qquad \textbf{Eq. 7}$$

$$m°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT} \qquad \textbf{Eq. 8}$$

If we merge parts of equation 7 and 8 for our problem we get:

$m°\, h_{IN} + 0 + 0 = m°\, h_{OUT} + W°_{OUT} + 0$

$m°\, h_F = m°\, h_I + W°_{OUT}$ Real-World case

$m°\, h_F = m°\, h_{IS} + W°_{OUTs}$ Adiabatic Isentropic case

In Example 13.5, we dropped the pressure from point F to point I via a real expansion process. In doing that, we found that we were able to extract 36.25 Btu of energy from every lbm of mass as the enthalpy dropped from –1622 to –1658.25 (see Example 13.5). This is the real-world process examined in this problem in more detail and shown on the pressure enthalpy chart as occurring along path F to I (see Figure 29).

Locate point F on the pressure enthalpy chart where 105 psia and –100 deg F coincide. Mark it with a pencil. We now have enough data to answer the question in Example 10.6 for point F. These properties are listed below.

Locate point I on the pressure enthalpy chart where 30 psia and –180 deg F coincide. Mark it with a pencil. We now have enough data to answer the question in Example 10.6 for point I. These properties are listed below.

The actual process follows the path from point F to point I.

If, however, this had been an adiabatic isentropic expansion, the path of the process would have followed a line or constant entropy from point F to point I_S.

Locate point I_S on the pressure enthalpy chart where 30 psia and 2.31 Btu/lbm deg R entropy line (same as for point F) coincides. Mark it with a pencil. We now have enough data to answer the question in Example 10.6 for point I_S. These properties are listed below. The concept of entropy will be discussed in the next section in more detail.

Note that the temperature at point I_S is approximately –194 deg F and the enthalpy is –1665 Btu/lbm. This means that, if this had been an ideal turbo expander, more energy would have come out of the turbine. Instead of getting only 36.25 Btu/lbm, the machine would have delivered 43 Btu/lbm (1665 Btu/lbm–1622 Btu/lbm). We will talk about the meaning of this adiabatic isentropic process in the next section.

ANSWERS:

At Point F

Initial pressure	105 psia
Initial temperature	–100 deg F
Initial enthalpy	–1622 Btu/lbm
Initial specific volume	2.15 cu.ft./lbm
Initial density	0.465 lbm/cu.ft.
Initial entropy	2.31 Btu/lbm deg R

At Point I

Final pressure actual	30 psia
Final temperature actual	–180 deg F
Final enthalpy actual	–1658.25 Btu/lbm
Final specific volume actual	6 cu.ft./lbm
Final density actual	0.167 lbm/cu.ft.
Final entropy actual	2.335 Btu/lbm deg R

At Point I_S

Final pressure isentropic	30 psia
Final temperature isentropic	–194 deg F
Final enthalpy isentropic	–1665 Btu/lbm
Final specific volume isentropic	5.7 cu.ft./lbm

Final density isentropic	0.175 lbm/cu.ft.
Final entropy isentropic	2.31 Btu/lbm deg R

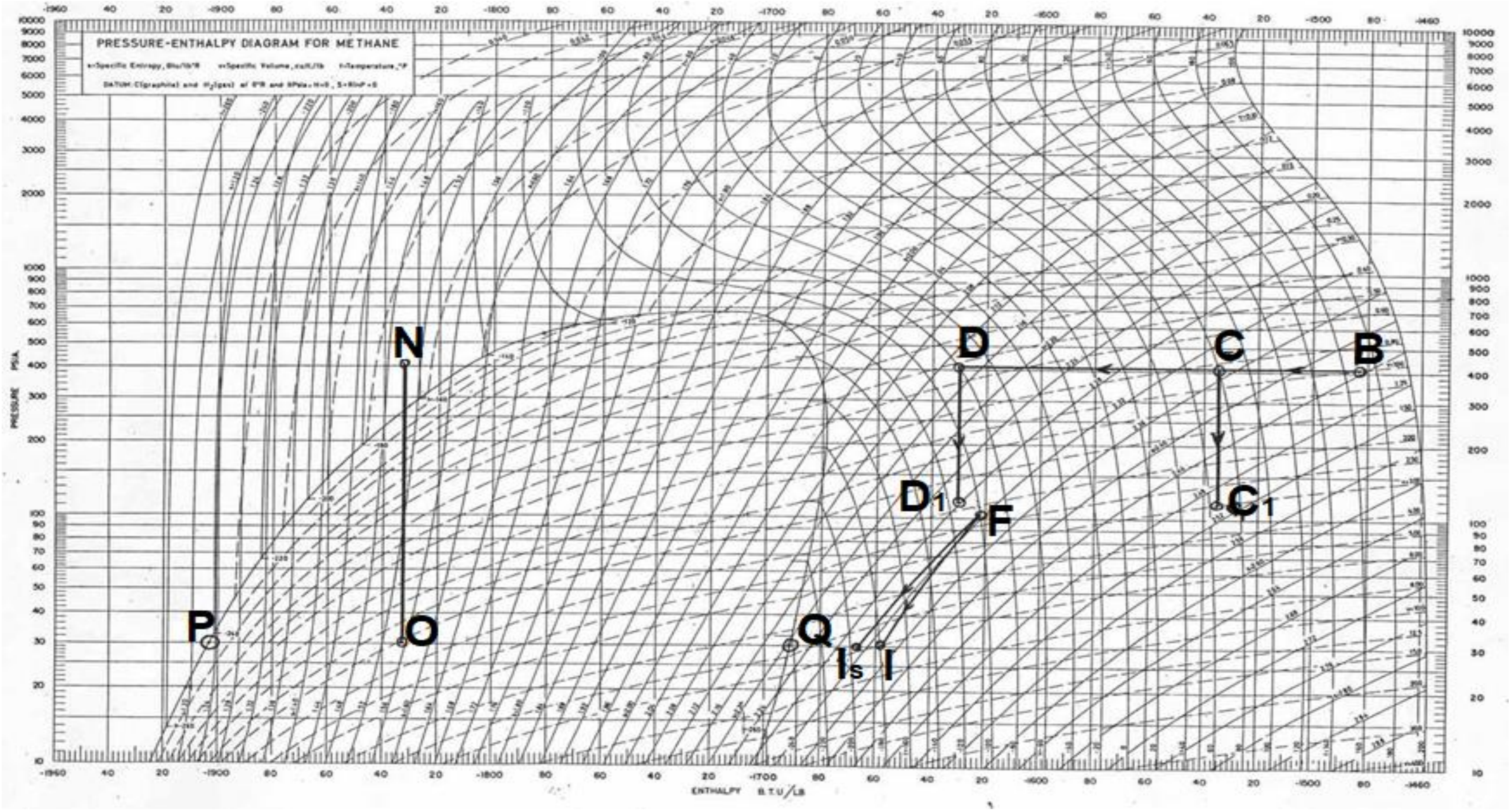

Figure 29: PH diagram with various process's plotted on the graphic (use accompanying chart to duplicate accurately)

Source: Self-made curves over footnote 1 table

A More Advanced Concept—Entropy

For this section, refer to the pressure enthalpy chart in Figure 30, shown at the end of the section.

The word *entropy* has been known to bring beads of sweat to the armpits of engineering students and has often been linked to digestive irregularity in both students and engineers alike. But it isn't as scary as its reputation may imply.

We will not get into a rigorous definition of entropy because that is beyond the limited scope of this publication. Instead, we will look at how we can apply the property of entropy to the analysis of SSSF compressors and turbines. For our analysis we will use the property of entropy to determine the efficiency of a compressor or a turbine.

In short, in a compressor or a turbine, if there is no heat transfer to outside the control volume, and if the compression or expansion occurs as a 100% efficient process, the entropy of the fluid does not change. If the efficiency is anything less than 100%, there will be an increase in entropy. Absolutely all processes we know in our world are not 100% efficient, so they all result in an overall net increase in entropy. Although we will never have a perfectly efficient compression or expansion process, we often compare our actual compression or expansion to a perfectly efficient process. We use this comparison to define the efficiency of our compression or expansion. In typical operating plants, we have experienced compression and expansion efficiencies of between 75% and 85%.

In Example 8.4, we examined a real-world compression process in which fluid was compressed from point A to point B. It required adding 58 Btu/lbm to achieve this compression. Let us now examine this same process in greater detail.

At point A, the inlet pressure is 215 psia and the inlet temperature is 60 deg F. Locate this point on the pressure enthalpy chart (shown in Figure 30). Reading off the chart, note that the enthalpy at point A is –1541 Btu/lbm and that the entropy is 2.415 Btu/lbm deg R.

We now compress the fluid in a real-world compressor to outlet point B. The pressure is 415 psia and the temperature is 170 deg F.

Locate point B on the pressure enthalpy chart. Reading off the chart, note that the enthalpy at point B is –1483 Btu/lbm and the entropy is 2.435 Btu/lbm deg R. Note also that just as in

Example 8.4, the fluid enthalpy increased by 58 Btu/lbm (1541–1483) as it followed the path from point A to point B.

If the compression process had been ideal (100% efficient), the path would have followed a constant entropy line from point A to point Bs.

Locate point Bs on the pressure enthalpy chart where 415 psia coincides with entropy of 2.415 Btu/lbm deg R (same as point A). Reading off the chart at point Bs, note that the enthalpy is –1494 Btu/lbm and the temperature is 148 deg F.

Note also that the ideal compressor required less work into the compressor, only 47 Btu/lbm (1541–1494) compared with 58 Btu/lbm in the actual compressor. Also, the outlet temperature is lower (148 deg F compared with 170 deg F).

Efficiency for a compressor or a turbine is defined as the percentage that results when you compare the actual work required to compress or expand the gas with the theoretical work required to compress or expand that gas if it had been compressed or expanded in an adiabatic isentropic process.

The easiest way to remember this is to realize it is a ratio, which always has the smaller number in the numerator (the top of the fraction).

1. For a compressor

In a compressor, you are putting work into the compressor, and you must always put more work in with an actual compressor than with an ideal compressor, so the equation is written as:
% Efficiency = 100 x [(Theoretical work for a 100% efficient compressor) / (actual work required to compress the gas)]

Writing this slightly differently, in short hand, we could say:

Equation 9: Equation for efficiency of a compressor

$$\textbf{\% Efficiency}_{\textbf{Compresser}} = \textbf{100 [ideal work/actual work]}$$

In this example the actual work was 58 Btu/lbm and the ideal work was 47 Btu/lbm. The % efficiency is 100 x 47/58 or 81%.

2. For a turbine or and expander

In a turbine, you are taking work out of the turbine, and you always get less work out of an actual turbine than an ideal turbine, so the equation is written as:

% Efficiency = 100 x [(actual work from the turbine) / (theoretical work from a 100% efficient turbine)]

Writing this slightly differently, in short hand, we could say:

Equation 10: Equation for efficiency of a turbine or expander

% $\text{Efficiency}_{\text{Turbine}}$ = 100 [actual work/ideal work]

In the analysis of our simplified open expansion plant, we will see in section 17 that the compressors operate at an efficiency of 81%, as shown above, and the turbines operate at approximately 84.3% efficiency.

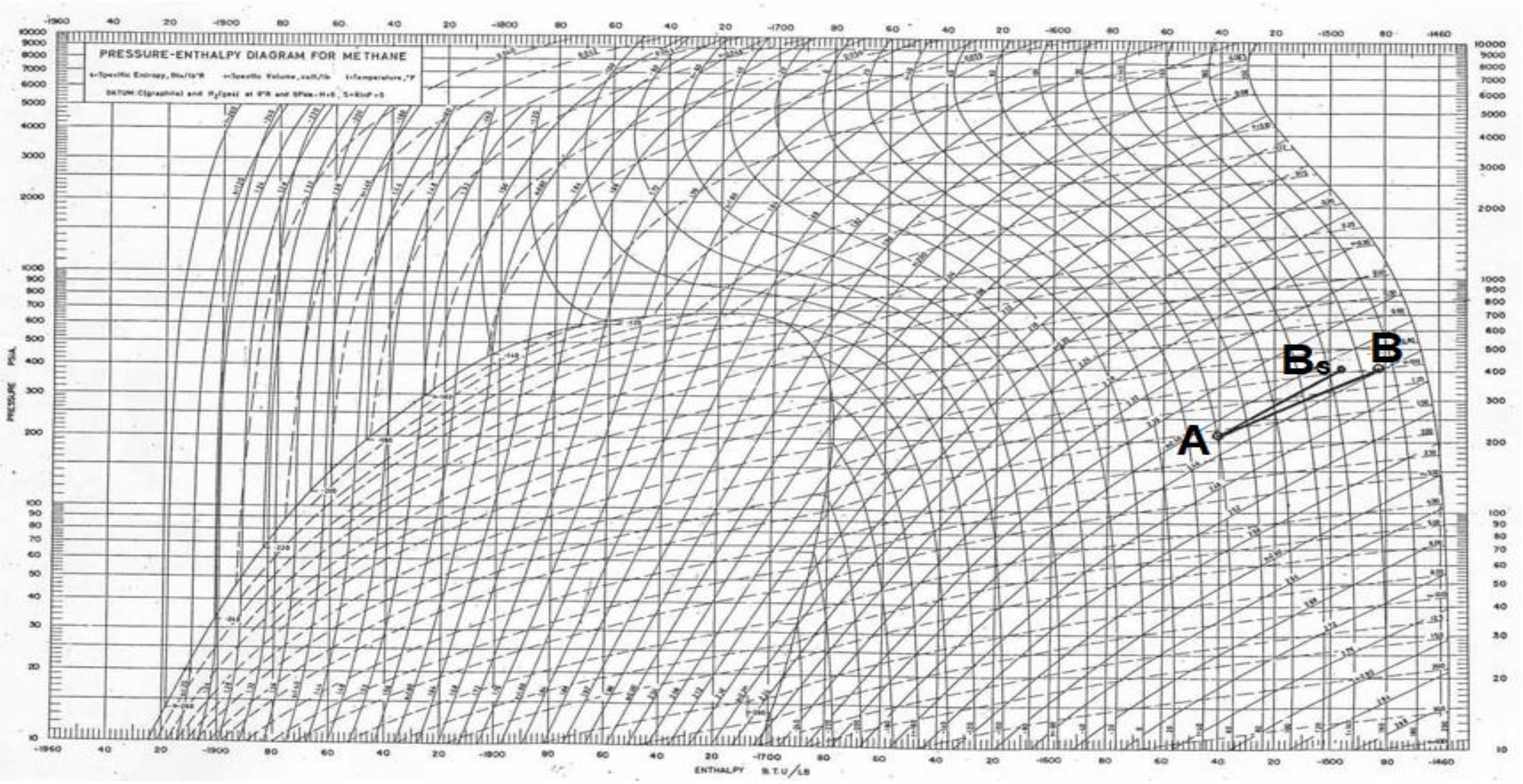

Figure 30: Example of actual and isentropic compression (use accompanying chart to duplicate accurately)

Source: Self-made curves over footnote 1 table

Simplified Overview Analysis of an Open Expansion Liquefaction Plant

For this section, refer to the pressure enthalpy chart shown in Figure 44 at the end of the section.

Before we do an analysis of the simplified open expansion liquefaction plant, we must understand that the plant discussed in the following pages is an oversimplified process intended to teach basic principles. Actual plants will differ from the process analyzed significantly at various points. As an example, the knock out pot in the simplified plant is operating at 15 psig. In an actual plant, it operates at a higher pressure to supply enough head to send liquid to the tank (perhaps at 60 psig, with a subsequent flash). The real plant has many differences not accounted for, such as—but not limited to—heat leak into the process, different performing heat exchangers, pressure drops, and phase changes of non-methane fluids. Also, the process described here uses the pure substance methane for analysis. (See also the cautionary note on the of this publication.)

The open expansion plant is based on a very simple process. As we recall from the first law of thermodynamics, for a steady-state, steady-flow (SSSF) process, the energy into a control volume must equal the energy out of a control volume. This means that if we have a SSSF process with a flow into a control volume and either heat or work energy leaving the control volume, the only way the first law is satisfied is if the fluid leaving the control volume has a lower energy content. Also recall that the thermodynamic energy of a fluid is defined by its enthalpy. Thus, the enthalpy of a fluid is reduced if we extract heat or work from that fluid.

In our open expansion plant, the heart of the plant is the turbo-expander. A turbo-expander is a device that has a high-pressure fluid enter it and work and a low-pressure fluid leave it. When we apply the first law of thermodynamics to the turbo-expander control volume, we find that the outlet fluid has a relatively low enthalpy and thus has a low temperature. (See Example 10.6.) This low enthalpy fluid (low temperature) is used to remove heat from and eventually liquefy the natural gas.

On the next page, you will find a schematic diagram of a very simplified process. We are using a simplified process to provide a conceptual understanding of the thermodynamics involved and to avoid being burdened with a rigorous process analysis. The theory exercised in analyzing this simplified plant can be applied to understanding open- and closed-cycle plants, the operation of vaporizers at plants, and truck loading and unloading, boil-off, and pumping phenomena.

Along with the schematic diagram is a data page. We have eliminated certain data so that we can use the pressure enthalpy chart in Figure 44 and the theory covered here to find the missing data. This will exercise all the theory covered in the earlier sections and solidify our understanding of how our plants function.

We will start at the high-pressure facility gas inlet to the plant and close the analysis with the delivery of liquid methane to the tank and tail gas to the gas system. Finally, we will place a control volume around the entire plant and do an overall mass and energy balance to check the earlier calculations and findings.

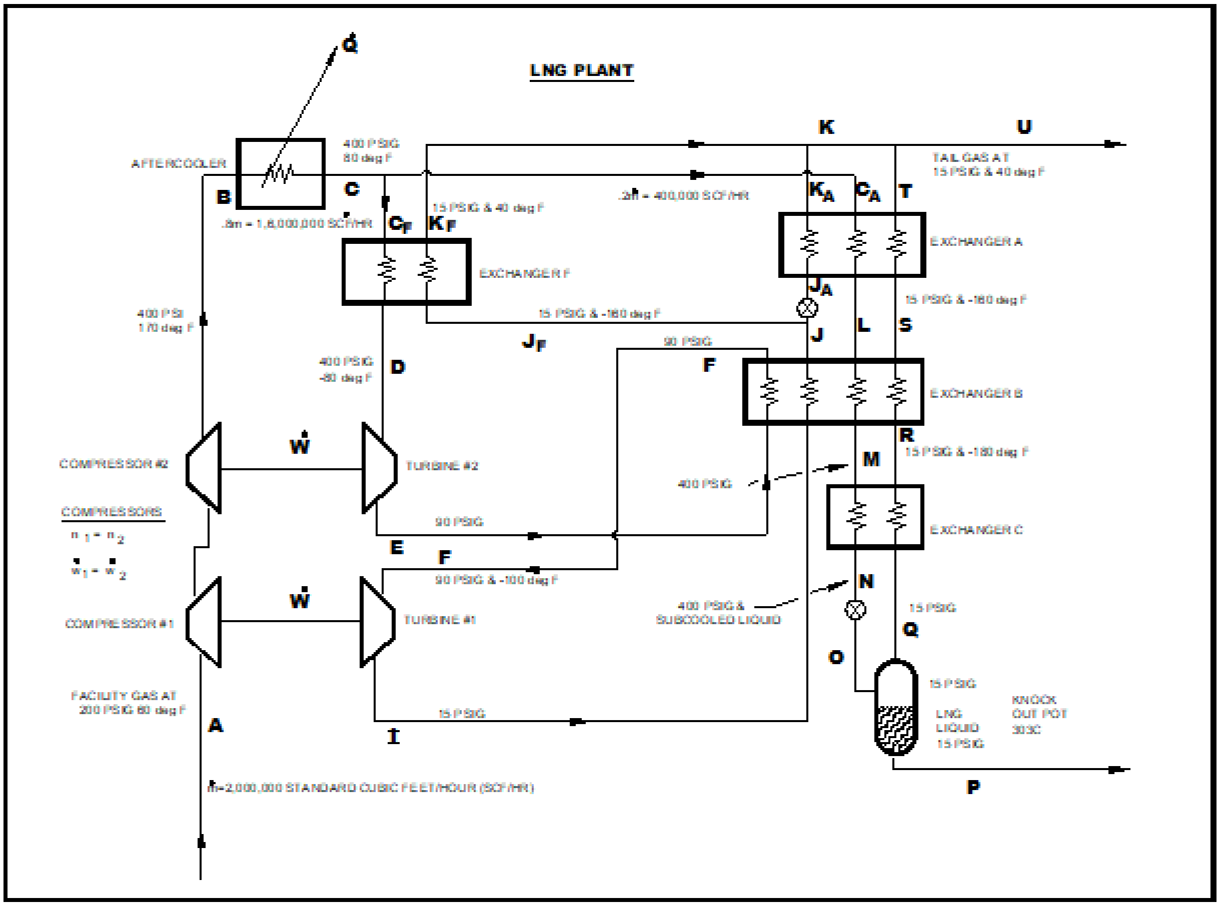

Figure 31: Simplified Schematic of an Open Expansion Liquefaction Plant

Source: Self-made

Notes:

1. n_1 and n_2 are compressor efficiencies.
2. $w°_1$ & $w°_2$ are shaft work from the turbines to the compressors.
3. After point C, the flow splits into flows C_F and C_A. The intensive properties stay the same, but the mass flow splits, with 80% of the flow at point C going to point C_F and 20% of the flow at point C going to point C_A.
4. After point J, the flow splits into flows J_F and J_A. The intensive properties stay the same, but the mass flow splits, with some percentage of the flow at point J going to point J_F and some percent of the flow at point J going to point J_A.
5. The valve shown between the A Exchanger and the B Exchanger is used for flow balancing purposes. For this analysis, there is no pressure drop across this valve.

In order to exercise the materials explained so far, take the three tables below and fill in the blanks using the conservation of mass equation (Section 12); the first law of thermodynamics (Section 8); the data on the schematic diagram above; and the pressure enthalpy chart. Also, find the shaft work and efficiency of each compressor and each turbine. Then find heat rejected by the aftercooler, and do an overall energy balance for the entire plant. As we do the analysis, we will highlight the answers to fill in the table below with the word "**ANSWER**" in bold type.

Table 4: Table of values given and to be found by this exercise

Location	Temp, deg F	Abs Pressure, psia	Enthalpy, Btu/lbm	Mass Flow, lbm/hr.
A	60	215		
B	170	415		
C	80	415		
C_A	80	415		
C_F	80	415		
D	–80	415		
E		105		
F	–100	105		
I		30		
J_A	–160	30		
J_F	–160	30		
J	–160	30		
K_A	40	30		
K_F	40	30		

L		415		
M		415		
N		415		
O		30		
P		30		
Q		30		
R	–180	30		
S	v160	30		
T	40	30		
U	40	30	**Note 6**	Note 6

Note 6: Values for point U will not be tabulated until later in this section.

Table 5: Table of values to be found by this exercise

Aftercooler	**Heat Rejected per lbm,** Btu/lbm	Total Heat Rejected, Btu/hr.

Table 6: Table of values to be found by this exercise

Machine	Work per lbm, Btu/lbm	Total Work, Btu/hr.	Efficiency, %
Compressor #1			
Compressor #2			
Turbine #1			
Turbine #2			

If you would like to learn more about the use of the pressure enthalpy diagram, we suggest that you add three more columns to the first table, above, and that you look up the entropy and specific volume at each point and compute the density for each point using Equation 2.

1. Inlet

Let’s start at the inlet of the plant where facility gas at 200 psig and 60 deg F is supplied to the plant. Referring to Section 5, you will see that our flow of 2,000,000 standard cu.ft. of gas per hour equates to a mass flow rate of 84,680 lbm/hr. We can see by inspection of the schematic diagram on Figure 44 that this is the mass flowrate for points A, B, and C.

For the following analyses in most of Section 12, for simplicity, let minlet° designate the mass flowrate into the plant with the units of lbm/hr. and let minlet designate only the coefficient of that flowrate (in other words, minlet° = 84,680 lbm/hr. and minlet = 84,680). We will use the actual value of 84,680 lbm/hr. only where needed to find the total numerical value of a quantity. If, at any point in the analysis, you need to find out the actual value of the data, just plug in the value for minlet°.

2. Compressors

Let's start our analysis by placing a control volume around the two compressors that compress the gas from point A to point B.

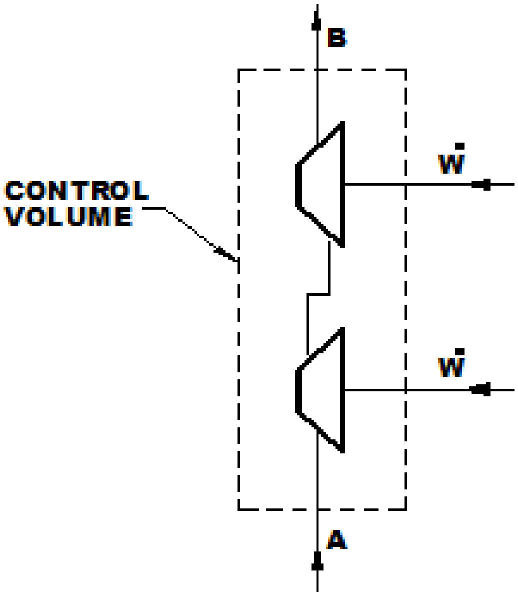

Figure 32: Control Volume Placed Around the Compressors

Source: Self made

The properties at the inlet and outlet of the compressors are as follows:

At point A:

Pressure =	200 psig (given)
Pressure absolute =	215 psia (see equation 1 for conversion to absolute pressure)
Temperature =	60 deg F (given)

Enthalpy = -1541 Btu/lbm (look up on pressure enthalpy chart) **ANSWER**
Entropy = 2.415 Btu/lbm deg R (look up on pressure enthalpy chart)
Mass flow rate minlet° (determined earlier) **ANSWER**

At point B:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see equation 1 for conversion to absolute pressure)
Temperature = 170 deg F (given)
Enthalpy = -1483 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**
Mass flow rate minlet° (determined earlier) **ANSWER**

At point B_S (Adiabatic Isentropic Compression)
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see equation 1 for conversion to absolute pressure)
Temperature = 148 deg F (look up on pressure enthalpy chart)
Enthalpy = -1494 Btu/lbm (look up on pressure enthalpy chart)
Entropy = 2.415 Btu/lbm deg R (same as at point A)

Let's apply the first law of thermodynamics to these compressors as follows:

$$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + m^{\circ}\, q_{IN} = m^{\circ}\, h_{OUT} + m^{\circ}\, w^{\circ}_{OUT} + m^{\circ}\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of Equations 7 and 8 for our problem we get:

Real-world case

$m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + 0 = m^{\circ}\, h_{OUT} + 0 + 0$ Real-world case
$h_{IN} + w_{IN} = h_{OUT}$ Real-world case
$w_{IN} = h_{OUT} - h_{IN}$ Real-world case
$w_{IN} =$ -1483 Btu/lbm – (-1541 Btu/lbm) Real-world case
$w_{IN} =$ 58 Btu/lbm Real-world case
$W^{\circ}_{IN} = m^{\circ}\, w_{IN}$ Real-world case
$W^{\circ}_{IN} =$ minlet° (58 Btu/lbm) Real-world case
$W^{\circ}_{IN} =$ 58 (minlet) Btu/hr. (total for both compressors) Real-world case

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

Before we leave the topic of the real-world compressors, let's state that, for this simplified plant, there are two shafts of input power into the control volume, and it was a given for this plant that the work of shaft #1 = the work of shaft #2 (given on the schematic diagram). Thus, the work into each compressor is (58/2) Btu/lbm or 29 Btu/lbm. **ANSWER**

The total work into each compressor is 29 (minlet) Btu/hr. **ANSWER**

Adiabatic Isentropic case

$m° h_{IN} + m° w_{INs} = m° h_{OUTs}$	Adiabatic Isentropic case
$w_{INs} = h_{OUTs} – h_{IN}$	Adiabatic Isentropic case
$w_{INs} =$ -1494 Btu/lbm – (-1541 Btu/lbm)	Adiabatic Isentropic case
$w_{INs} =$ 47 Btu/lbm	Adiabatic Isentropic case
$W°_{INs} = m° w_{IN}$	Adiabatic Isentropic case
$W°_{INs} =$ minlet° (47 Btu/lbm)	Adiabatic Isentropic case
$W°_{INs} =$ 47 (minlet) Btu/hr.	Adiabatic Isentropic case

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

The above means that if this had been a perfectly efficient compressor, it would have required only 47 Btu of energy to compress each lbm of gas. However, our real compressor required 58 Btu of energy to compress each lbm of gas.

To find the efficiency we use the equation in Section 16.1:

% $\text{Efficiency}_{\text{Compresser}}$ = 100 [ideal work/actual work] Eq. 9

% $\text{Efficiency}_{\text{Compresser}}$ = 100 [47 Btu/lbm / 58 Btu/lbm]

% $\text{Efficiency}_{\text{Compresser}}$ = 81 % **ANSWER**

3. Aftercooler

To analyze the aftercooler, we need only to evaluate points B and C and then to apply the first law of thermodynamics.

We start by placing a control volume around the aftercooler.

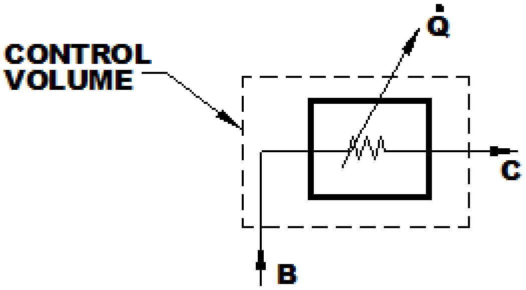

Figure 33: Control Volume Placed Around the Aftercooler

Source: Self made

The properties of the inlet and outlet of the aftercooler are as follows:

At point B:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = 170 deg F (given)
Enthalpy = –1483 Btu/lbm (look up on pressure enthalpy chart) (determined earlier)
Mass flow rateminlet° (determined earlier)

At point C:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = 80 deg F (given)
Enthalpy = –1535 Btu/lbm (look up on pressure enthalpy chart) **ANSWER**
Mass flow rateminlet° (determined earlier) **ANSWER**

Let's apply the first law of thermodynamics to this heat exchanger as follows:

$$\mathbf{H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we use Equation 8 for our problem we get:

$m°\, h_{IN} + 0 + 0 = m°\, h_{OUT} + 0 + m°\, q_{OUT}$

$q_{OUT} = h_{IN} – (h_{OUT})$

$q_{OUT} = - 1483 – (-1535)$

q_{OUT} = heat out = 52 Btu/lbm **ANSWER**

$Q_{OUT}° = m°q_{OUT}$
$Q_{OUT}° = minlet° (52 Btu/lbm)$
$Q_{OUT}° = 52(minlet) Btu/hr.$ **ANSWER**

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

4. F Exchanger

To analyze the F exchanger, we need to apply the first law of thermodynamics. Here we have the properties of all the inlets and outlets but we do not have the flow rate of the stream that travels from point J_F to K_F.

Let's start by drawing a control volume around the F exchanger.

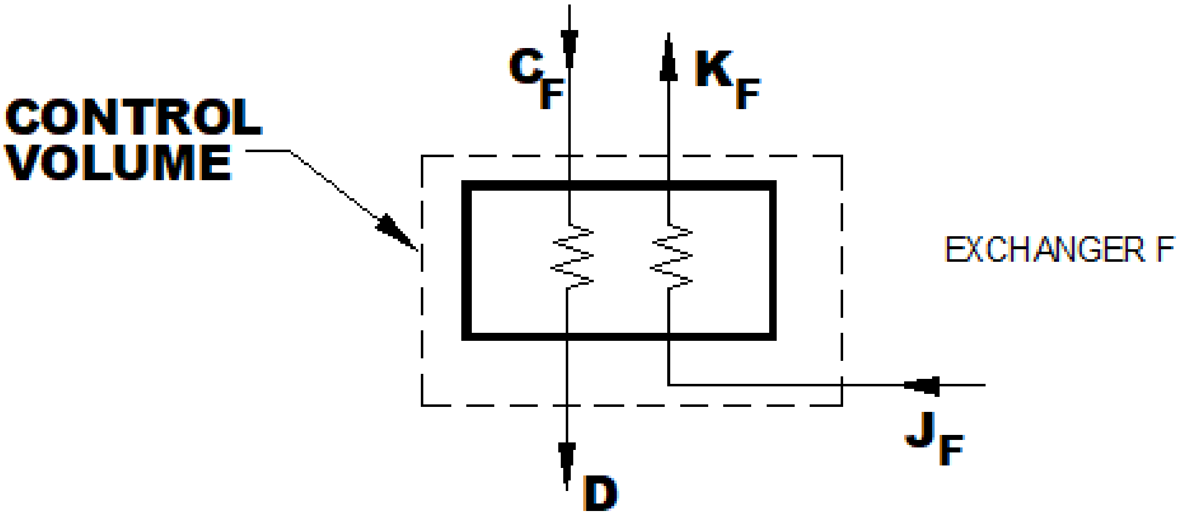

Figure 34: Control Volume Placed Around the F Exchanger

Source: Self made

The properties of the inlet and outlet of the F Exchanger are as follows:

At point C_F:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = 80 deg F (given)
Enthalpy = –1535 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

Mass Flow = 0.8(minlet°) = 67,744 lbm/hr. (It was shown on the schematic that the 20%-80% split occurred before the F exchanger. The flowrate to the F exchanger is 0.8(84,680 lbm/hr.). By inspection we observe that this is the flow rate for points C_F D, E, F, I, J, and K). **ANSWER**

At point D:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = –80 deg F (given)
Enthalpy = –1630 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**
Mass Flow rate = 0.8(minlet°) (determined earlier)

At point J_F:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

At point K_F:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = 40 deg F (given)
Enthalpy = –1545 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

But inspection point J has the same intensive properties of JF. This means that the enthalpy of point J is –1648 Btu/lbm and the temperature of point J is –160 deg F. **ANSWER**

Let’s apply the first law of thermodynamics to the F exchanger as follows:

$$H^\circ_{IN} + W^\circ_{IN} + Q^\circ_{IN} = H^\circ_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT} \qquad \textbf{Eq. 7}$$

$$m^\circ\, h_{IN} + m^\circ\, w_{IN} + m^\circ\, q_{IN} = m^\circ\, h_{OUT} + m^\circ\, w_{OUT} + m^\circ\, q_{OUT} \qquad \textbf{Eq. 8}$$

If we use equation 8 for our problem we get:

$$m^\circ\, h_{IN} + 0 + 0 = m^\circ\, h_{OUT} + 0 + 0$$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet.

$$m^\circ_{CF}\, h_{IN\ CF} + m^\circ_{JF}\, h_{IN\ JF} = m^\circ_{D}\, h_{OUT\ D} + m^\circ_{KF}\, h_{OUT\ KF}$$

By the conservation of mass equation, we know that $m^{\circ}_{CF} = m^{\circ}_{D}$. **(See Equation 4)**

$m^{\circ}_{CF} = m^{\circ}_{D} = 0.8(\text{minlet}^{\circ})$

$0.8(\text{minlet}^{\circ})$ (–1535 Btu/lbm) + m°_{JF} (-1648 Btu/lbm) = $0.8(\text{minlet}^{\circ})$ (–1630 Btu/lbm) + m°_{KF} (–1545 Btu/lbm)

By the conservation of mass equation, we know that $m^{\circ}_{JF} = m^{\circ}_{KF}$. **(See Equation 4)**

$m^{\circ}_{JF} = m^{\circ}_{KF}$ = unknown

Since $m^{\circ}_{JF} = m^{\circ}_{KF}$ we can rewrite the above as:

$0.8(\text{minlet}^{\circ})$ (–1535 Btu/lbm) + m°_{JF} (–1648 Btu/lbm) = $0.8(\text{minlet}^{\circ})$ (–1630 Btu/lbm) + m°_{JF} (–1545 Btu/lbm)

–1228(minlet) Btu/hr. + m°_{JF} (–1648 Btu/lbm + 1545Btu/lbm) = –1304 (minlet) Btu/hr.

m°_{JF} –103 Btu/lbm) = –76 (minlet) Btu/hr.

m°_{JF} = 0.7378 (minlet) lbm/hr. (which is 74% of the flow that entered the plant) **ANSWER**

Via conservation of mass Equation 4, we also know that:

m°_{KF} = 0.7378 (minlet) lbm/hr. (which is 74% of the flow that entered the plant) **ANSWER**

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

5. Turbo Expander #2

To analyze the turbo expander #2, we need to apply the first law of thermodynamics. Here, we have the flowrate and properties of the inlet from our F exchanger analysis, and we have the work output from our compressor analysis (work out of the turbine is equal to the work into the compressor), and we have the pressure at the outlet from the schematic. All we need to find is another intensive property of the outlet, and then we will be able to determine all the properties of the outlet (at point E). Then we can also do the adiabatic isentropic analysis to find the efficiency of the expander.

Let's start by drawing a control volume around the expander #2.

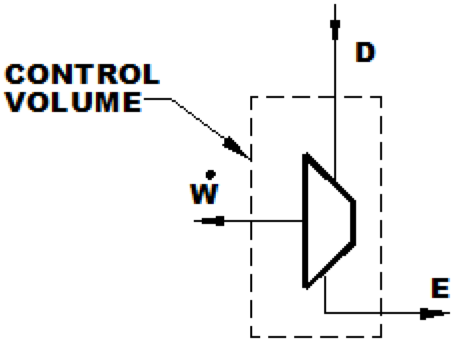

Figure 35: Control Volume Placed Around the Turbo Expander #2

Source: Self made

The properties of the inlet and outlet of the turbo expander are as follows:

At point D:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = –80 deg F (given)
Enthalpy = –1630 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Entropy = 2.135 Btu/lbm deg R (Look up on pressure enthalpy chart)
Mass flow rate = 0.8(minlet°) (determined earlier)

At point E:
Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics and chart
Enthalpy = we need to find via first law of thermodynamics and chart

At point E_S (Adiabatic Isentropic Expansion)
Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –191 deg F (look up on pressure enthalpy chart)
Enthalpy = –1673 Btu/lbm (look up on pressure enthalpy chart)
Entropy = 2.135 Btu/lbm deg R (same as at point D)
Work out of turbine = W_{OUT} = 29 (minlet) Btu/hr. per our analysis of the compressors (above)

ANSWER

We recall that:

$W°_{OUT} = m°w_{OUT}$

$w_{OUT} = W°_{OUT} / m°$

$w_{OUT} = 29\ (\text{minlet})\ \text{Btu/hr.} / 0.8\ (\text{minlet}°)$

$w°_{OUT} = 36.25\ \text{Btu/lbm}$ **ANSWER**

Let's apply the first law of thermodynamics to the turbine as follows:

$\mathbf{H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}}$ **Eq. 7**

$\mathbf{m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}}$ **Eq. 8**

If we merge parts of Equations 7 and 8 for our problem we get:

Real-world case

$m°\ h_{IN} + 0 + 0 = m°\ h_{OUT} + W°_{OUT} + 0$ Real-world case

$m°\ h_{IN} = m°\ h_{OUT} + W°_{OUT}$ Real-world case

$m°\ h_{OUT} = m°\ h_{IN} - W°_{OUT}$ Real-world case

$0.8(\text{minlet}°)\ (h_{OUT}) = 0.8(\text{minlet}°)\ (-1630\ \text{Btu/lbm}) – 29(\text{minlet})\ \text{Btu/hr.}$ Real-world case

$h_{OUT} = -1630\ \text{Btu/lbm} –36.25\ \text{Btu/lbm}$ Real-world case

$h_{OUT} = -1666.25\ \text{Btu/lbm}$ (Point E) Real-world case **ANSWER**

Using the pressure enthalpy chart, we can find the temperature at point E (105 psia pressure and –1666.25 Btu/lbm enthalpy) to be –179 deg F

ANSWER

Adiabatic Isentropic case

$m°\ h_{IN} + 0 + 0 = m°\ h_{OUTs} + W°_{OUTs} + 0$ Adiabatic Isentropic case

$W°_{OUTs} = m°\ h_{IN} - m°\ h_{OUTs}$ Adiabatic Isentropic case

$W°_{OUTs} = 0.8(\text{minlet}°)\ (-1630\ \text{Btu/lbm} – (-1673\ \text{Btu/lbm}))$ Adiabatic Isentropic case

$W°_{OUTs} = 34.4(\text{minlet})\ \text{Btu/hr.}$ Adiabatic Isentropic case

(Reminder minlet° = 84,680 lbm/hr. and minlet = 84,680)

The above means that if we had a perfectly efficient turbine, it would have produced 34.4 (minlet) Btu/hr. However, our real-world turbine produces only 29 (minlet) Btu/hr.

To find the efficiency, we use the equation in Section 16.2.
% Efficiency$_{\textbf{turbine}}$ = 100 [actual work/ideal work]Eq. 10

% Efficiency $_{\text{turbine}}$= 100 [29(minlet) Btu/hr. / 34.4(minlet) Btu/hr.]

% Efficiency$_{\text{turbine}}$= 84.3 % **ANSWER**

6. Turbo Expander #1

To analyze the turbo expander #1, we need to apply the first law of thermodynamics. Here, we have the flowrate and properties of the inlet, we have the work output from our earlier compressor analysis (the work into the compressor is equal to the work out of the turbine), and we have the pressure at the outlet from the schematic. All we need to find is another intensive property of the outlet and we will be able to then determine all the properties of the outlet (at point I). Then we can also do the adiabatic isentropic analysis to find the efficiency of the expander.

Let's start by drawing a control volume around the expander #1.

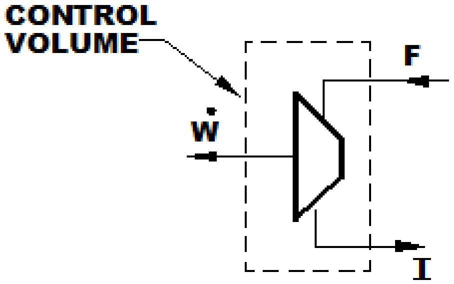

Figure 36: Control Volume Placed Around the Turbo Expander #1

The properties of inlet and outlet of the turbo expander are as follows:

At point F:

Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –100 deg F (given)

Enthalpy = –1622 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**
Entropy = 2.315 Btu/lbm deg R (look up on pressure enthalpy chart)
Mass flow rate = 0.8(minlet°) (determined earlier)

At point I:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see equation 1 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics and chart
Enthalpy = we need to find via first law of thermodynamics and chart

At point I$_S$ (Adiabatic Isentropic Expansion)
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –195 deg F (look up on pressure enthalpy chart)
Enthalpy = –1665 Btu/lbm (look up on pressure enthalpy chart)
Entropy = 2.315 Btu/lbm deg R (same as at point F)

Work out of turbine = W_{OUT} = 29 (minlet) Btu/hr. per our analysis of the compressors (see above). **ANSWER**

We recall that:
$W^\circ_{OUT} = m^\circ w_{OUT}$
$w_{OUT} = W^\circ_{OUT} / m^\circ$
w_{OUT} = 29 (minlet) Btu/hr. / 0.8(minlet°)
w°_{OUT} = 36.25 Btu/lbm **ANSWER**

Let's apply the first law of thermodynamics to these compressors as follows:

$$\mathbf{H^\circ_{IN} + W^\circ_{IN} + Q^\circ_{IN} = H^\circ_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^\circ\, h_{IN} + m^\circ\, w_{IN} + m^\circ\, q_{IN} = m^\circ\, h_{OUT} + m^\circ\, w_{OUT} + m^\circ\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of equation 7 and 8 for our problem we get:

Real-world case
$m^\circ\, h_{IN} + 0 + 0 = m^\circ\, h_{OUT} + W^\circ_{OUT} + 0$ Real-world case
$m^\circ\, h_{IN} = m^\circ\, h_{OUT} + W^\circ_{OUT}$ Real-world case
$m^\circ\, h_{OUT} = m^\circ\, h_{IN} - W^\circ_{OUT}$ Real-world case
0.8(minlet°) (h_{OUT}) = 0.8(minlet°) (–1622 Btu/lbm) – 29 (minlet) Btu/hr. Real-world case

h_{OUT} = –1622 Btu/lbm –36.25 Btu/lbm Real-world case
h_{OUT} = –1658.25 Btu/lbm (point I) Real-world case **ANSWER**
Tout = –80 deg F (look up on pressure enthalpy chart) Real-world case **ANSWER**

Adiabatic Isentropic case
m° h_{IN} + 0 + 0 = m° h_{OUTs} + W°$_{OUTs}$ + 0 Adiabatic Isentropic case
W°$_{OUTs}$ = m° h_{IN} - m° h_{OUTs} Adiabatic Isentropic case
W°$_{OUTs}$ = 0.8(minlet°) (-1622 Btu/lbm – (-1665 Btu/lbm)) Adiabatic Isentropic case
W°$_{OUTs}$ = 34.4 (minlet) Btu/hr. Adiabatic Isentropic case

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

The above means that if we had a perfectly efficient turbine, it would have produced 34.4 (minlet) Btu/hr. of energy. However, our real-world turbine produces only 29 (minlet) Btu/hr. of energy.

To find the efficiency we use the equation in Section 16.2.

% Efficiency$_{turbine}$ = 100 [actual work/ideal work] **Eq. 10**

% Efficiency$_{turbine}$= 100 [29 (minlet) Btu/hr. / 34.4 (minlet) Btu/hr.]

% Efficiency$_{turbine}$ = 84.3 % **ANSWER**

7. Analysis of A, B, C Exchangers and Knock Out Pot

Next, we will place a control volume around the A, B, C exchangers and the knock out pot. We have selected this set of exchangers and knock out pot for analysis because this selection of control volume allows us to write the first law of thermodynamics and the conservation of mass with only two unknowns. With two equations and two unknowns, we can solve the equations.

Let's start by drawing a control volume around the A, B, and C exchangers and knock out pot.

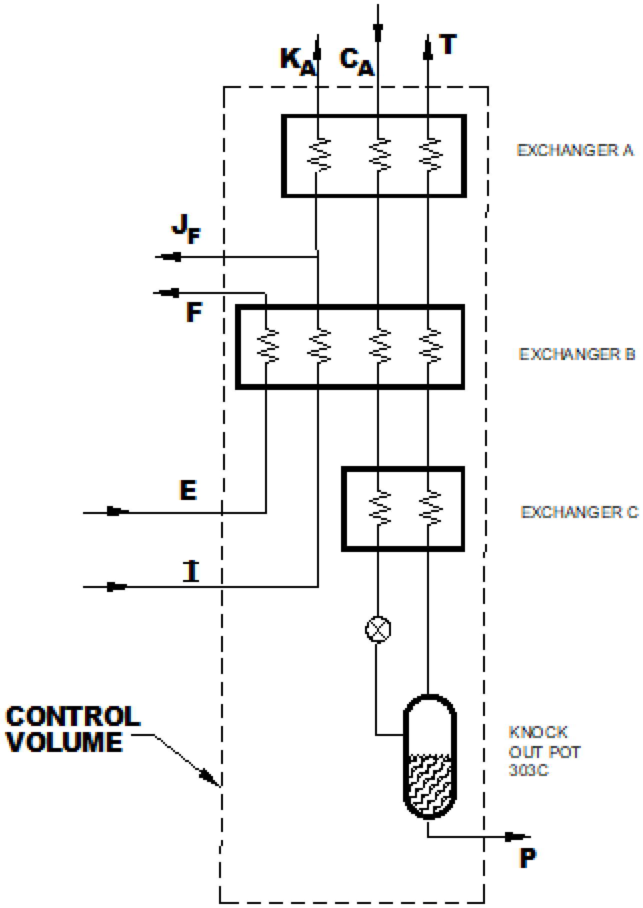

Figure 37: Control Volume Placed Around the A, B, C Exchangers and Knock Out Pot

Source: Self made

The properties of the inlet and outlet of the heat exchanger are as follows:

At point I:

Pressure =	15 psig (given)
Pressure absolute =	30 psia (see Section 4 for conversion to absolute pressure)
Temperature =	–180 deg F (determined earlier)
Enthalpy =	–1658.25 Btu/lbm (determined earlier)
Mass flow rate =	0.8 (minlet°) (determined earlier)

At point E:

Pressure =	90 psig (given)

Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –179 deg F (determined earlier)
Enthalpy = –1666.25 Btu/lbm (determined earlier)
Mass flow rate = 0.8 (minlet°) (determined earlier)

At point F:
Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –100 deg F (given)
Enthalpy = –1622 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.8(minlet°) (determined earlier)

At point J_F:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.7378 (minlet°) (determined earlier)

At point K_A:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = 40 deg F (given)
Enthalpy = –1545 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

At point C_A:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = 80 deg F (given)
Enthalpy = –1535 Btu/lbm (look up on pressure enthalpy chart) **ANSWER**
Mass Flow = 0.2(minlet°) = 16,936 lbm/hr. (It was shown on the schematic that the 20% - 80% split occurred before the F exchanger.) The flowrate to the A exchanger is 0.2 (84,680 lbm/hr.). By inspection we observe that this is the flow rate for points C_A L, M, N, and O).

ANSWER

At point T:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = 40 deg F (given)
Enthalpy = –1545 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

At point P:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –242 deg F (look on pressure enthalpy chart, liquid line) **ANSWER**
Enthalpy = –1903 Btu/lbm (look on pressure enthalpy chart, liquid line) **ANSWER**

Do not let this analysis scare you. It is nothing we have not done before. First, let's apply the conservation of mass to the flow stream from point I to points J_F and K_A. I and J_F are already known, so point K_A can be computed as follows:

$m^{\circ}_{I} = m^{\circ}_{JF} + m^{\circ}_{KA}$
$m^{\circ}_{KA} = m^{\circ}_{I} - m^{\circ}_{JF}$
$m^{\circ}_{KA} = 0.8(\text{minlet}^{\circ}) - 0.7378(\text{minlet}^{\circ})$.
$m^{\circ}_{KA} = 0.0622\ (\text{minlet}^{\circ})$ **ANSWER**

Let's apply the first law of thermodynamics to the exchangers as follows:

$$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + m^{\circ}\, q_{IN} = m^{\circ}\, h_{OUT} + m^{\circ}\, w_{OUT} + m^{\circ}\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of Equations 7 and 8 for our problem, we get:

$m^{\circ}\, h_{IN} + 0 + 0 = m^{\circ}\, h_{OUT} + 0 + 0$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet

$m_I^{\circ}h_I + m_E^{\circ}h_E + m_{CA}^{\circ}h_{CA} = m_F^{\circ}h_F + m_{JF}^{\circ}h_{JF} + m_{KA}^{\circ}h_{KA} + m_T^{\circ}h_T + m_P^{\circ}h_P$

0.8 (minlet°) (–1658.25 Btu/lbm) + 0.8 (minlet°) (–1666.25 Btu/lbm) + 0.2 (minlet°) (–1535 Btu/lbm) = 0.8 (minlet°) (–1622 Btu/lbm) + 0.7378 (minlet°) lbm/hr. (–1648 Btu/lbm) + 0.0622 (minlet°) (–1545 Btu/lbm) + m_T°(–1545 Btu/lbm) + m_P°(–1903 Btu/lbm)

Let's multiply out the various terms to make the equation simpler.
–1326.6 (minlet) Btu/hr. – 1333 (minlet) Btu/hr. – 307 (minlet) Btu/hr. = –1297.6 (minlet) Btu/hr. – 1215.9 (minlet) Btu/hr. – 96.1 (minlet) Btu/hr. + m_T°(–1545 Btu/lbm) + m_P°(–1903 Btu/lbm)
– 2966.6 (minlet) Btu/hr. = - 2609.6 Btu/hr. + m_T°(–1545 Btu/lbm) + m_P°(–1903 Btu/lbm)
– 357 (minlet) Btu/hr. = + m_T°(–1545 Btu/lbm) + m_P°(–1903 Btu/lbm)

By the conservation of mass equation, we know that: **(See equation 4)**
$m_{CA}^\circ = m_T^\circ + m_P^\circ$
$m_T^\circ = m_{CA}^\circ - m_P^\circ$
$m_T^\circ =$ 0.2 (minlet°) - m_P°

Plug into the above equation: $m_T^\circ =$ 0.2 (minlet°) – m_P°
– 357 (minlet) Btu/hr. = + (0.2 (minlet°) – m_P°) (–1545 Btu/lbm) + m_P°(–1903 Btu/lbm)

Notice that there is only one unknown in the above equation….. solve for it!
– 357 (minlet) Btu/hr. = – 309 (minlet) Btu/hr. + (1545 – 1903) m_P°
– 48 (minlet) Btu/hr. = (–358 Btu/lbm) m_P°
m_P° = 0.1341 (minlet) lbm/hr. **ANSWER**

[This is the net LNG production (approx. 13.41% of the gas input into the plant). The actual LNG plant, from which this example was modeled, converts approximately 10% of inlet gas to LNG.]

By the conservation of mass equation, we know that: **(See equation 4)**
$m_{CA}^\circ = m_T^\circ + m_P^\circ$
$m_T^\circ = m_{CA}^\circ - m_P^\circ$
$m_T^\circ =$0.2 (minlet)lbm/hr. - 0.1341 (minlet) lbm/hr.
m_T° = 0.0659 (minlet) lbm/hr. **ANSWER**

(This is the flash gas produced in pot (after the flash across the valve) (approx. 6.59% of the gas input into the plant))

By inspection, we observe that this is also the flow rate for points Q, R & S. **ANSWER**
(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

8. Analysis of A Exchanger

Next, we will analyze the A Exchanger. We can write the first law of thermodynamics and solve the equation.

Let's start by drawing a control volume around the A exchanger.

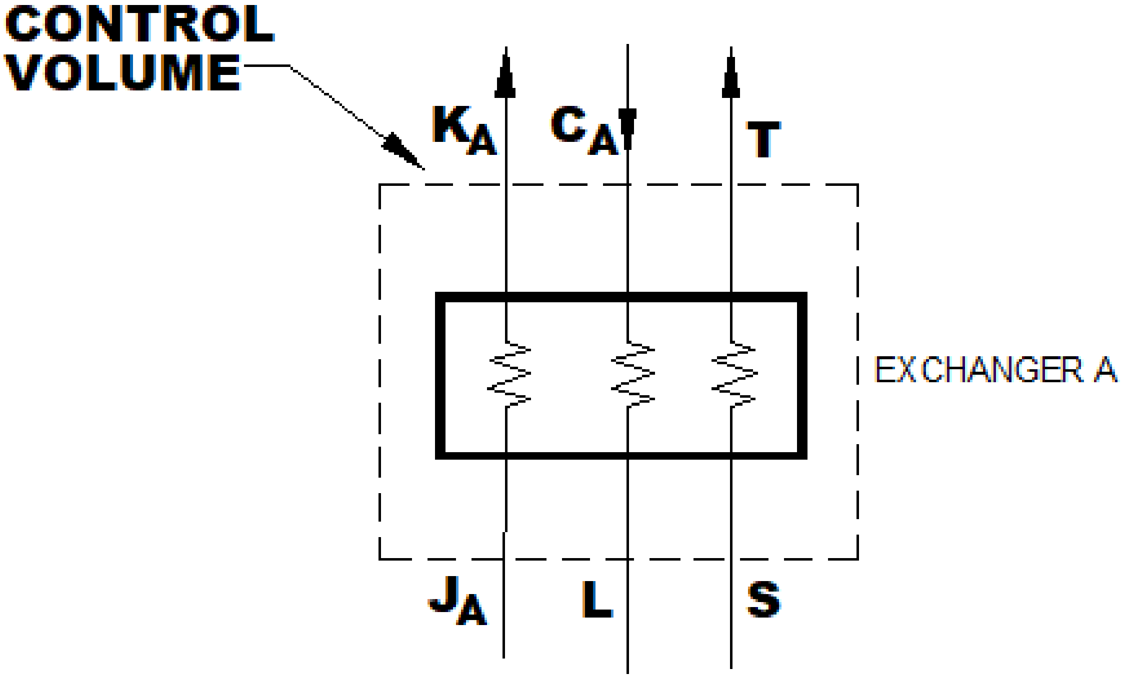

Figure 38: Control Volume Placed Around the A Exchanger

The properties of the inlet and outlet of the heat exchanger are as follows:

At point K_A:

Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)

Temperature = 40 deg F (given)
Enthalpy = –1545 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.0622 (minlet°) (determined earlier)

At point C_A:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = 80 deg F (given)
Enthalpy = –1535 Btu/lbm (look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point T:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = 40 deg F (given)
Enthalpy = –1545 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.0659 (minlet°) (determined earlier)

At point J_A:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

At point L:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics
Enthalpy = we need to find via first law of thermodynamics
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point S:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see equation 1 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

Mass flow rate = 0.0659 (minlet°) (determined earlier)

By the conservation of mass equation, we know that: (**See Equation 4)**
$m°_{JA} = m°_{KA} = 0.0622$(minlet°)

Let's apply the first law of thermodynamics to the exchangers as follows:
$\mathbf{H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}}$ **Eq. 7**

$\mathbf{m°\, h_{IN} + m°\, w_{IN} + m°\, q_{IN} = m°\, h_{OUT} + m°\, w_{OUT} + m°\, q_{OUT}}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:
$m°\, h_{IN} + 0 + 0 = m°\, h_{OUT} + 0 + 0$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet
$m_{JA}°h_{JA} + m_{CA}°h_{CA} + m_S°h_S = m_{KA}°h_{KA} + m_L°h_L + m_T°h_T$
0.0622 (minlet°) (–1648 Btu/lbm) + 0.2 (minlet°) (–1535Btu/lbm) + 0.0659 (minlet°) (-1648 Btu/lbm) = 0.0622 (minlet°) (–1545 Btu/lbm) + 0.2 (minlet°) (h_L) + 0.0659 (minlet°) (-1545 Btu/lbm)
–102.5 (minlet) Btu/hr. –307 (minlet) Btu/hr. –108.6 (minlet) Btu/hr. = -96.1 (minlet) Btu/hr. + 0.2 (minlet°) (h_L) – 101.82 (minlet) Btu/hr.
–518.1 Btu/hr. = –197.92 Btu/hr. + 0.2 lbm/hr. (h_L)
–320.18 Btu/hr. = 0.2 lbm/hr. (h_L)
h_L = -1601 Btu/lbm **ANSWER**

Look up on the pressure enthalpy chart the temperature using the pressure of 415psia and the enthalpy of 1601 Btu/lbm and we find a temperature of: T_L = -33 deg F. **ANSWER**

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

9. Analysis of B Exchanger

Next, we will place a control volume around the B exchanger. We can write the first law of thermodynamics and solve the equation.

Let's start by drawing a control volume around the B exchanger.

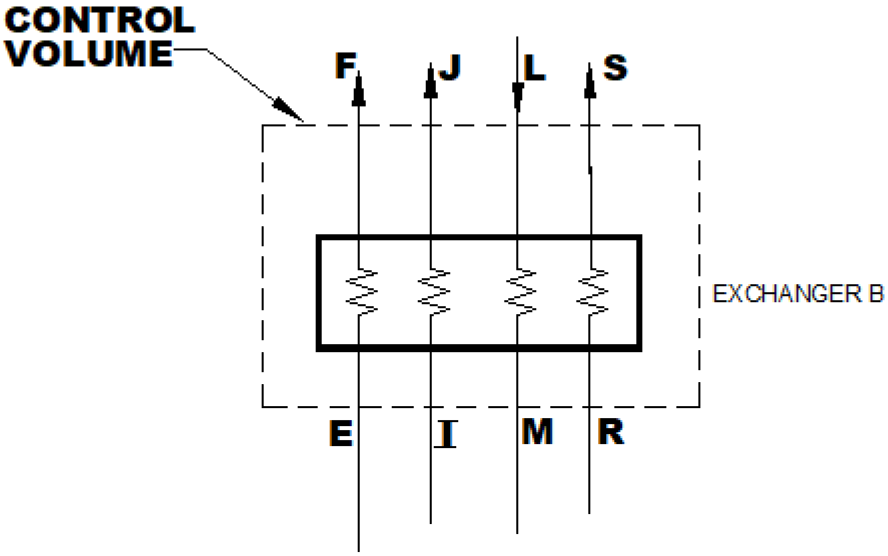

Figure 39: Control Volume Placed Around the B Exchanger

The properties of the inlet and outlet of the heat exchanger are as follows:

At point F:
Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –100 deg F (given)
Enthalpy = –1622 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Entropy = 2.135 Btu/lbm deg R
Mass flow rate = 0.8 (minlet°) (determined earlier)

At point J:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.8 (minlet°) (determined earlier)

At point L:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = –33 deg F (determined earlier)

Enthalpy = –1601 Btu/lbm (determined earlier)
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point S:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –160 deg F (given)
Enthalpy = –1648 Btu/lbm (Look on pressure enthalpy chart) (determined earlier)
Mass flow rate = 0.0659 (minlet°) (determined earlier)

At point E:
Pressure = 90 psig (given)
Pressure absolute = 105 psia (see Section 4 for conversion to absolute pressure)
Temperature = –179 deg F (determined earlier)
Enthalpy = –1666.25 Btu/lbm (determined earlier)
Mass flow rate = 0.8 (minlet°) (determined earlier)

At point I:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –180 deg F (determined earlier)
Enthalpy = –1658.25 Btu/lbm (determined earlier)
Mass flow rate = 0.8 (minlet°) (determined earlier)

At point M:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics and chart
Enthalpy = we need to find via first law of thermodynamics and chart
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point R:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –180 deg F (given)
Enthalpy = –1658 Btu/lbm (Look up on pressure enthalpy chart) **ANSWER**

Mass flow rate = 0.0659 (minlet°) (determined earlier)

Let's apply the first law of thermodynamics to the exchangers as follows:
$H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}$ Eq. 7

$m^{\circ} h_{IN} + m^{\circ} w_{IN} + m^{\circ} q_{IN} = m^{\circ} h_{OUT} + m^{\circ} w_{OUT} + m^{\circ} q_{OUT}$ Eq. 8

If we merge parts of equation 7 and 8 for our problem we get:
$m^{\circ} h_{IN} + 0 + 0 = m^{\circ} h_{OUT} + 0 + 0$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet
$m_E{}^{\circ}h_E + m_I{}^{\circ}h_I + m_L{}^{\circ}h_L + m_R{}^{\circ}h_R = m_F{}^{\circ}h_F + m_J{}^{\circ}h_J + m_M{}^{\circ}h_M + m_S{}^{\circ}h_S$
0.8 (minlet°) (–1666.25 Btu/lbm) + 0.8 (minlet°) (-1658.25Btu/lbm) + 0.2 (minlet°) (-1601 Btu/lbm) + 0.0659 (minlet°) (–1658 Btu/lbm) = 0.8 (minlet°) (–1622 Btu/lbm) + 0.8 (minlet°) (-1648 Btu/lbm) + 0.2 (minlet°) (h_M) + 0.0659 (minlet°) (–1648 Btu/lbm)
–1333 (minlet) Btu/hr. –1326.6 (minlet) Btu/hr. –320.2 (minlet) Btu/hr. –109.3 (minlet) Btu/hr. = -1297.6 (minlet) Btu/hr. –1318.4 (minlet) Btu/hr. + 0.2 (minlet°) (h_M) – 108.6 (minlet) Btu/hr.
–3089.1 (minlet) Btu/hr. = –2724.6 (minlet) Btu/hr. + 0.2 (minlet°) (h_M)
–364.5 (minlet) Btu/hr. = 0.2 (minlet°) (h_L)
h_M = -1822.5 Btu/lbm **ANSWER**

Look up on the pressure enthalpy chart the temperature using the pressure of 415 psia and the enthalpy of -1822 Btu/lbm and we find a temperature of: TM = -156 deg F. **ANSWER**

(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

10. Analysis of C Exchanger

Next, we will place a control volume around the C exchanger. We can write the first law of thermodynamics and solve the equation.

Let's start by drawing a control volume around the C exchanger.

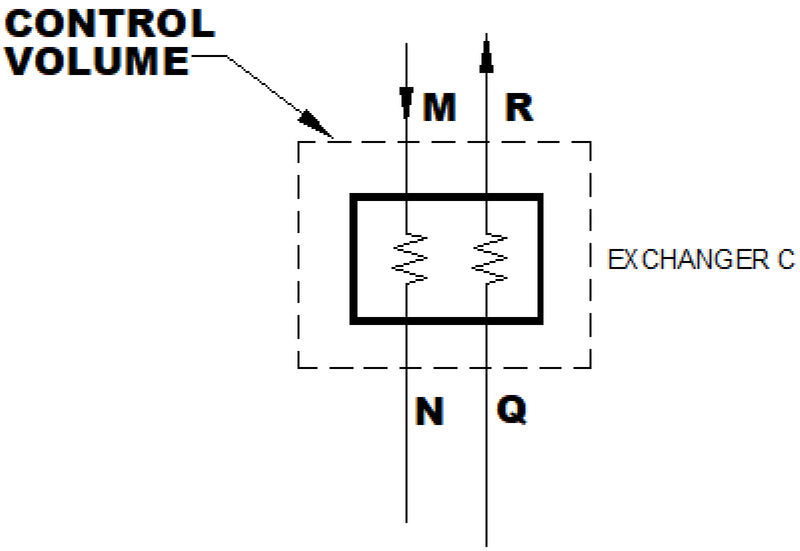

Figure 40: Control Volume Placed Around the C Exchanger

The properties of the inlet and outlet of the heat exchanger are as follows:

At point M:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = –156 deg F (determined earlier)
Enthalpy = –1822.5 Btu/lbm (determined earlier)
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point R:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –180 deg F (given)
Enthalpy = –1658 Btu/lbm (Look up on pressure enthalpy chart) (determined earlier)
Mass flowrate = 0.0659 (minlet°) (determined earlier)

At point N:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics and chart
Enthalpy = we need to find via first law of thermodynamics and chart

Mass flow rate = 0.2 (minlet°) (determined earlier)

At point Q:

Pressure = 15 psig (given)

Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)

Temperature = –242 deg F (Look on pressure enthalpy chart, vapor line) **ANSWER**

Enthalpy = –1691 Btu/lbm (Look on pressure enthalpy chart, vapor line) **ANSWER**

Mass flow rate = 0.0659 (minlet°) (determined earlier)

Let's apply the first law of thermodynamics to the exchangers as follows:

$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}}$ **Eq. 7**

$\mathbf{m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + m^{\circ}\, q_{IN} = m^{\circ}\, h_{OUT} + m^{\circ}\, w_{OUT} + m^{\circ}\, q_{OUT}}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:

$m^{\circ}\, h_{IN} + 0 + 0 = m^{\circ}\, h_{OUT} + 0 + 0$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet

$m_M{}^{\circ}h_M + m_Q{}^{\circ}h_Q = m_N{}^{\circ}h_N + m_R{}^{\circ}h_R$

0.2 (minlet°) (–1822.5 Btu/lbm) + 0.0659 (minlet°) (–1691 Btu/lbm) = 0.2 (minlet°) (h_N) +0.0659 (minlet°) (–1658 Btu/lbm)

–364.5 (minlet) Btu/hr. – 111.44 (minlet) Btu/hr. = 0.2 (minlet°) (h_N) – 109.26 (minlet) Btu/hr.

–475.94 (minlet) Btu/hr. = 0.2 (minlet°) (h_N) – 109.26 (minlet) Btu/hr.

–366.68 (minlet) Btu/hr. = 0.2 (minlet°) (h_N)

(h_N) = 1833.4 Btu/lbm **ANSWER**

Look up on the pressure enthalpy chart the temperature using the pressure of 415 psia and the enthalpy of –1833.4 Btu/lbm and we find a temperature of: T_N = -165 deg F. **ANSWER**
(Reminder: minlet° = 84,680 lbm/hr. and minlet = 84,680)

11. Analysis of Joule-Thomson Valve

Next, we will place a control volume around the Joule Thomson Valve. We can write the first law of thermodynamics and solve the equation.

Let’s start by drawing a control volume around the Joules Thompson valve.

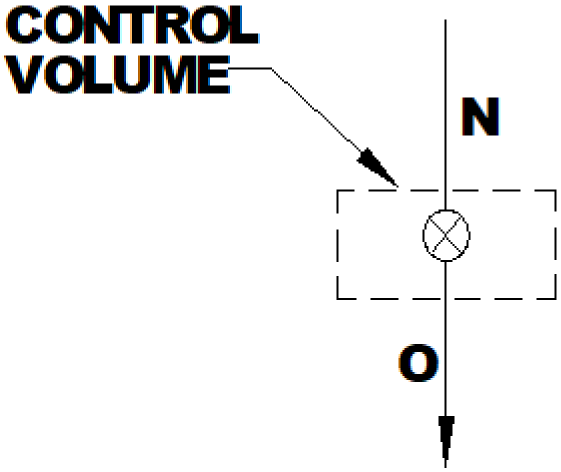

Figure 41: Control Volume Placed Around Joule-Thomson Valve

The properties of the inlet and outlet of the valve are as follows:

At point N:
Pressure = 400 psig (given)
Pressure absolute = 415 psia (see Section 4 for conversion to absolute pressure)
Temperature = –165 deg F (determined earlier)
Enthalpy = –1833.4 Btu/lbm (determined earlier)
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point O:
Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = we need to find via first law of thermodynamics and chart
Enthalpy = we need to find via first law of thermodynamics and chart
Mass flow rate = 0.2 (minlet°) (determined earlier)

Let’s apply the first law of thermodynamics to the exchangers as follows:

$$\mathbf{H^{\circ}_{IN} + W^{\circ}_{IN} + Q^{\circ}_{IN} = H^{\circ}_{OUT} + W^{\circ}_{OUT} + Q^{\circ}_{OUT}} \qquad \textbf{Eq. 7}$$

$$\mathbf{m^{\circ}\, h_{IN} + m^{\circ}\, w_{IN} + m^{\circ}\, q_{IN} = m^{\circ}\, h_{OUT} + m^{\circ}\, w_{OUT} + m^{\circ}\, q_{OUT}} \qquad \textbf{Eq. 8}$$

If we merge parts of equation 7 and 8 for our problem we get:

$m° h_{IN} + 0 + 0 = m° h_{OUT} + 0 + 0$
$m_N°h_N = m_O°h_O$

Since $m_N° = m_O°$, we can divide both sides of the equation by $m_N°$ and get:
$h_N = h_O$
$(h_O) = 1833.4$ Btu/lbm **ANSWER**

Look up on the pressure enthalpy chart the temperature using the pressure of 30 psia and the enthalpy of –1833.4 Btu/lbm, and we find a temperature of: T_O = –242 deg F. **ANSWER**
Note that the new point O is inside the liquid vapor dome. This means that the flow consists of both vapor and liquid.

In the next analysis, we will compute the mass of vapor and the mass of liquid. Since we already did compute this mass of vapor and liquid when we analyzed the A, B, and C exchangers with the knock out pot, this computation should come out to be the same values. This new calculation should serve as a check to confirm the earlier calculations as correct.

We have now computed all the values that were needed to answer the questions originally asked in this exercise. The next two analyses will be computed to check (confirm) our answers as correct. We will analyze the knock out pot and then the entire plant using some of the data already computed, and the mass energy analysis should balance.

12. Calculation Check by Performing a Mass Energy Balance Around the Knock Out Pot

Next, we will place a control volume around the knock out pot. We can write the first law of thermodynamics and solve the equation.

Let’s start by drawing a control volume around the knock out pot.

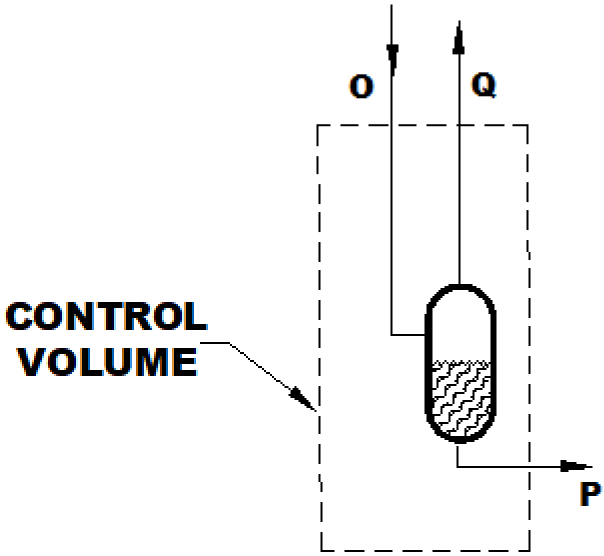

Figure 42: Control Volume Placed Around the Knock Out Pot

The properties of the inlet and outlet of the knock out pot are as follows:

At point O:

Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –242 deg F (determined earlier)
Enthalpy = –1833.4 Btu/lbm (determined earlier)
Mass flow rate = 0.2 (minlet°) (determined earlier)

At point Q:

Pressure = 15 psig (given)
Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = –242 deg F (Look on pressure enthalpy chart, vapor line) (determined earlier)
Enthalpy = –1691 Btu/lbm (Look on pressure enthalpy chart, vapor line) (determined earlier)
Mass flow rate = we need to find via first law of thermodynamics and chart

At point P:

Pressure = 15 psig (given)

Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)
Temperature = (determined earlier) –242 deg F (look up on pressure enthalpy chart, or liquid line)
Enthalpy = (determined earlier) –1903 Btu/lbm (Look up on pressure enthalpy chart, on liquid line)
Mass flow rate = we need to find via first law of thermodynamics and chart

Let's apply the first law of thermodynamics to the exchangers as follows:
$H^\circ_{IN} + W^\circ_{IN} + Q^\circ_{IN} = H^\circ_{OUT} + W^\circ_{OUT} + Q^\circ_{OUT}$ **Eq. 7**

$m^\circ h_{IN} + m^\circ w_{IN} + m^\circ q_{IN} = m^\circ h_{OUT} + m^\circ w_{OUT} + m^\circ q_{OUT}$ **Eq. 8**

If we merge parts of equation 7 and 8 for our problem we get:
$m^\circ h_{IN} + 0 + 0 = m^\circ h_{OUT} + 0 + 0$

Since there are multiple inlets and outlets, we will use the subscript of each port to describe each inlet and outlet
$m_O{}^\circ h_O = m_Q{}^\circ h_Q + m_P{}^\circ h_P$

By the conservation of mass equation, we know that: **(See equation 4)**
$m_O{}^\circ = m_Q{}^\circ + m_P{}^\circ$
$m_P{}^\circ = m_O{}^\circ - m_Q{}^\circ$
$m_P{}^\circ = 0.2$ (minlet)lbm/hr. – $m_Q{}^\circ$

Plug the above into the earlier equation.
$m_O{}^\circ h_O = m_Q{}^\circ h_Q + (0.2$ (minlet)lbm/hr. - $m_Q{}^\circ)\ h_P$
0.2 (minlet°) (–1833.4 Btu/lbm) = $m_Q{}^\circ$ (–1691 Btu/lbm) + (0.2 (minlet)lbm/hr. – $m_Q{}^\circ$) (–1903 Btu/lbm)
–366.68 (minlet) Btu/hr. = $m_Q{}^\circ$ (–1691 Btu/lbm) – 380.6 (minlet) Btu/hr. +1903 Btu/lbm ($m_Q{}^\circ$)
13.92 (minlet) Btu/hr.= 212 Btu/lbm ($m_Q{}^\circ$)
$(m_Q)^\circ = 0.0657$ (minlet) lbm/hr.

Check. This value is almost identical to the value we computed earlier. The slight difference is due to round-off error. Thus, this confirms the earlier calculations as correct.

From our earlier conservation of mass work above we restate that:

$m_P^\circ = m_O^\circ - m_Q^\circ$

Plug in the newly found value for (m_Q) and we get:

$m_P^\circ = 0.2$ (minlet)lbm/hr. – 0.0657 (minlet) lbm/hr.

$m_P^\circ = 0.1343$ (minlet) lbm/hr. Check. This value is almost identical to the value we computed earlier. The slight difference is due to round-off error. Thus, this confirms the earlier calculations as correct.

13. Overall Check by Performing a Mass Energy Balance Around the Entire Plant

Next, we will place a control volume around the entire plant. We can write the first law of thermodynamics and solve the equation.

Let's start by drawing a control volume around the entire plant.

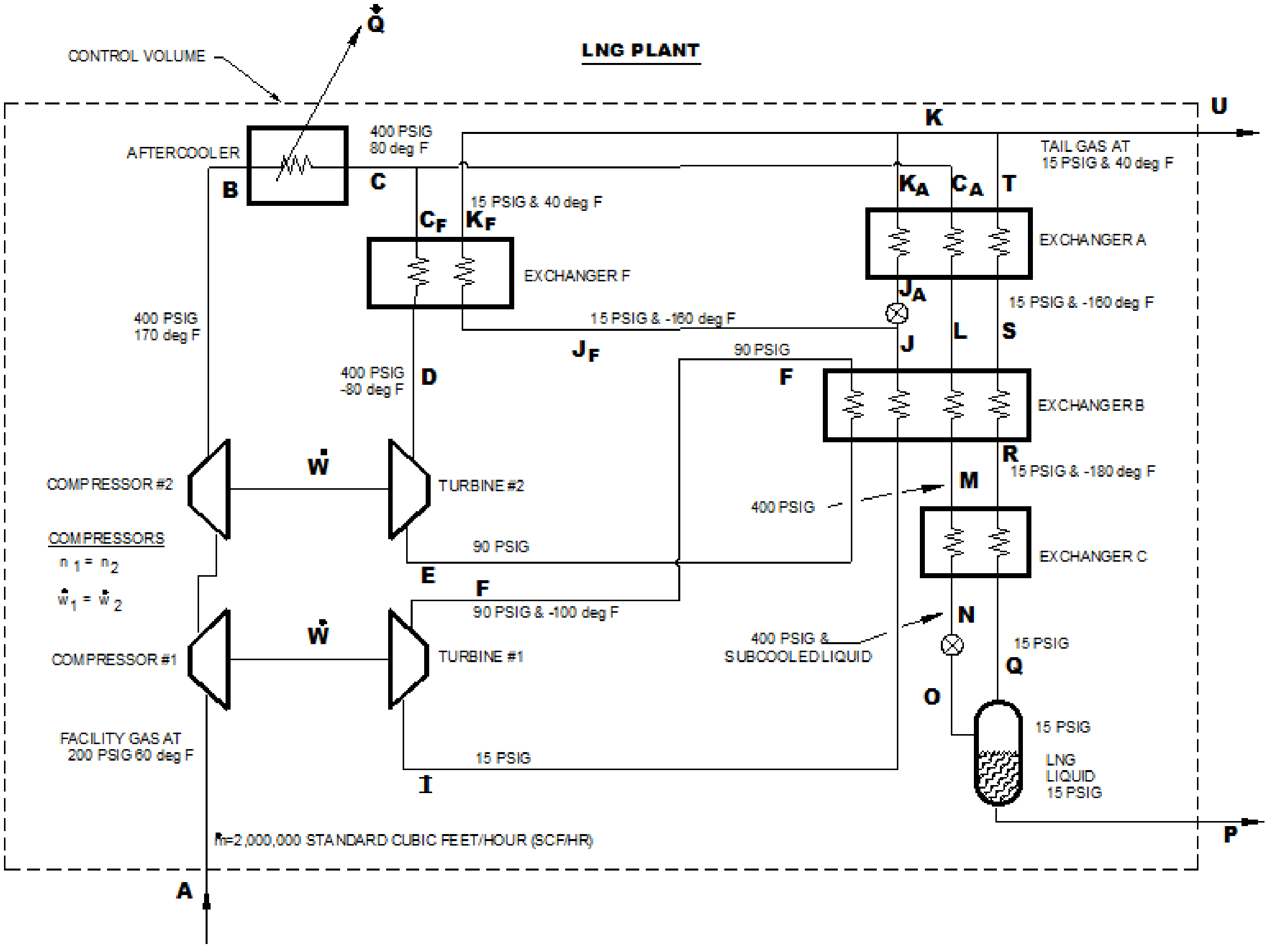

Figure 43: Control Volume Placed Around the Entire Plant

The properties of the inlet and outlets of the plant are as follows:

At point A:

Pressure = 200 psig (given)

Pressure absolute = 215 psia (see Section 4 for conversion to absolute pressure)

Temperature = 60 deg F (given)

Enthalpy = –1541 Btu/lbm (look on pressure enthalpy chart) (determined earlier)

Entropy = 2.415 Btu/lbm deg R (look up on pressure enthalpy chart) (determined earlier)

Mass flow rate minlet° (determined earlier)

At point U:

Pressure = 15 psig (given)

Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)

Temperature = 40 deg F (given)

Enthalpy = –1545 Btu/lbm (look on pressure enthalpy chart) **ANSWER**

At point P:

Pressure = 15 psig (given)

Pressure absolute = 30 psia (see Section 4 for conversion to absolute pressure)

Temperature = –242 deg F (look on pressure enthalpy chart, liquid line) (determined earlier)

Enthalpy = –1903 Btu/lbm (look on pressure enthalpy chart, liquid line) (determined earlier)

Mass flow rate = 0.1342 minlet° (Here we used the average of the finding on pages 120 and pages 137)

Let's apply the first law of thermodynamics to the exchangers as follows:

$H°_{IN} + W°_{IN} + Q°_{IN} = H°_{OUT} + W°_{OUT} + Q°_{OUT}$ Eq. 7

$m°\ h_{IN} + m°\ w_{IN} + m°\ q_{IN} = m°\ h_{OUT} + m°\ w_{OUT} + m°\ q_{OUT}$ Eq. 8

If we merge parts of equation 7 and 8 for our problem we get:

$m°\ h_{IN} + 0 + 0 = m°\ h_{OUT} + 0 + Q°_{OUT}$

Since there are multiple outlets, we will use the subscript of each port to describe each inlet and outlet

$m_A°h_A = m_U°h_U + m_P°h_P + Q°_{OUT}$
From our analysis of the aftercooler, we know that $Q°_{OUT}$ = 52(minlet) Btu/lbm
By the conservation of mass equation, we know that: **(See equation 4)**
$m_A° = m_U° + m_P°$
$m_U° = m_A° - m_P°$
$m_U°$ = (minlet)lbm/hr. - 0.1342 (minlet)lbm/hr. **ANSWER**
$m_U°$ = 0.8658 (minlet)lbm/hr.

Check: This confirms that the value of $m_U°$ is approximately the sum of K_F+K_A+T. The slight difference is due to round-off error. Thus, this confirms the earlier calculations as correct.

Plug values into the earlier energy balance equation. For the following equations, we will use the symbol ≈ instead of an equal sign because we realize that due to round-off errors, the left side of the equation will not exactly be equal to the right side of the equation.

(minlet°) (–1541 Btu/lbm) ≈ + 0.8658 (minlet°) (–1545 Btu/lbm) + 0.1342 (minlet°) (–903 Btu/lbm) + 52 (minlet°) Btu/lbm

–1541 ≈ -1337.66 - 255.38 + 52
–1541 ≈ –1541.04

Check. This equation almost balances. The slight difference is due to round-off error. Thus, this confirms the earlier calculations as correct.

Final Answer Page

Table 7: Answer table

Location	Temp, Deg F	Abs Pressure, psia	Enthalpy, Btu/lbm	Mass Flow, lbm/hr.
A	60	215	-1541	minlet
B	170	415	-1483	minlet
C	80	415	-1535	minlet
C_A	80	415	-1535	0.2 (minlet)
C_F	80	415	-1535	0.8 (minlet)
D	-80	415	-1630	0.8 (minlet)
E	-179	105	-1666.25	0.8 (minlet)
F	-100	105	-1622	0.8 (minlet)
I	-180	30	-1658.25	0.8 (minlet)
J_A	-160	30	-1648	0.0622 (minlet)
J_F	-160	30	-1648	0.7378 (minlet)
J	-160	30	-1648	0.8 (minlet)
K_A	40	30	-1545	0.0622 (minlet)
K_F	40	30	-1545	0.7378 (minlet)
L	-33	415	-1601	0.2 (minlet)
M	-156	415	-1822	0.2 (minlet)
N	-165	415	-1833.4	0.2 (minlet)
O	-242	30	-1833.4	0.2 (minlet)
P	-242	30	-1903	0.1341 (minlet)
Q	-242	30	-1691	0.0659 (minlet)
R	-180	30	-1658	0.0659 (minlet)
S	-160	30	-1648	0.0659 (minlet)
T	40	30	-1545	0.0659 (minlet)
U	40	30	-1545	0.8658 (minlet)

Table 8: Answer table

Aftercooler	**Heat Rejected per lbm** Btu/lbm	Total Heat Rejected Btu/hr.
	52	52 (minlet)

Table 9: Answer

Machine	Work per lbm, Btu/lbm	Total Work, Btu/hr.	Efficiency, %
Compressor #1	29	29 (minlet)	81
Compressor #2	29	29 (minlet)	81
Turbine #1	36.25	29 (minlet)	84.3
Turbine #2	36.25	29 (minlet)	84.3

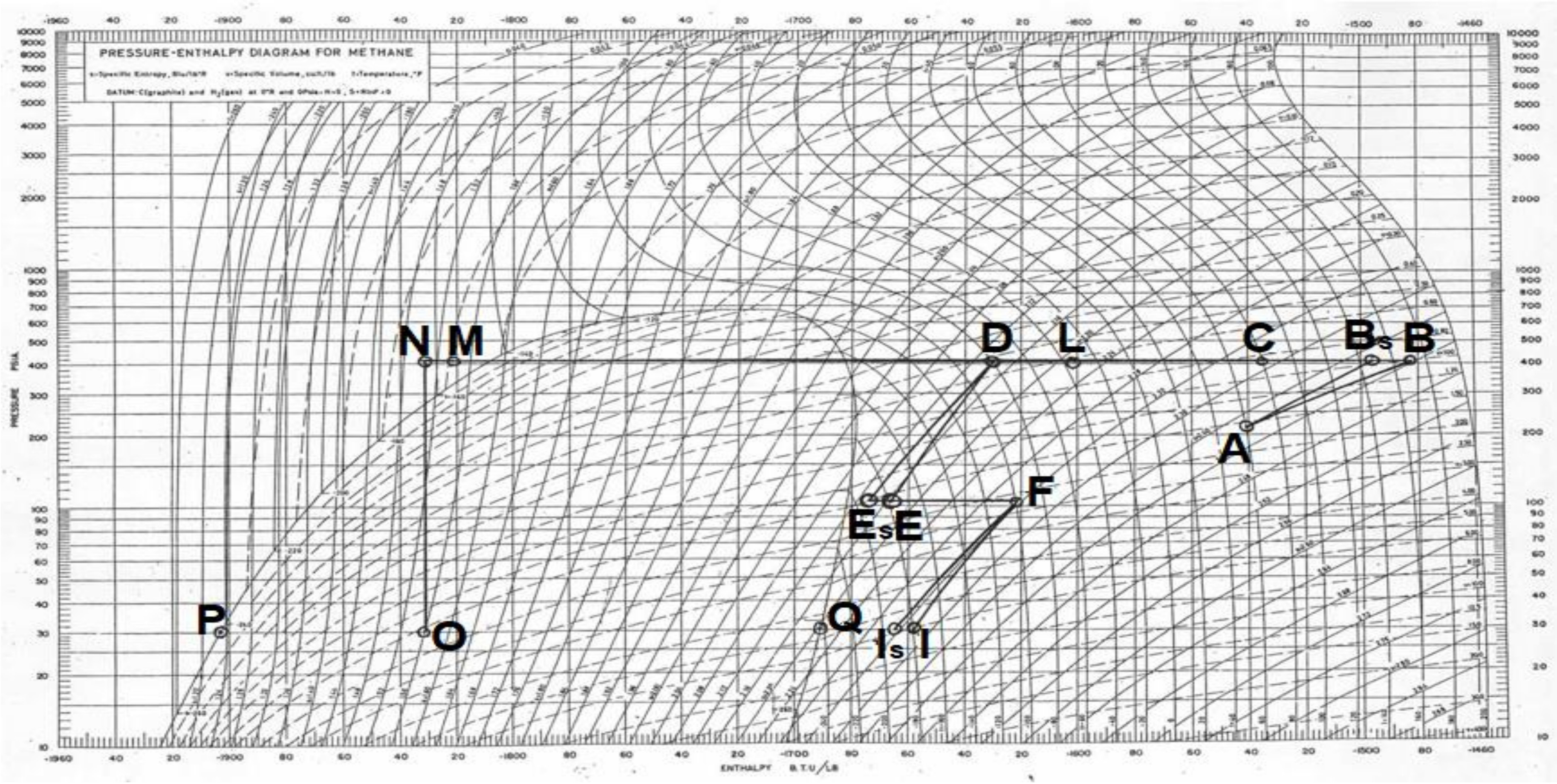

Figure 44: Plant process points (use accompanying chart to duplicate accurately

Source: Source: Self-made curves over footnote 1 table

Conclusion

In this publication, we have used the conservation of mass and the first law of thermodynamics to understand the basics of liquid methane and methane gas. It is understood for the reasons stated earlier that the analysis was simplified with the intent of teaching concepts. As you—the engineers of our LNG and propane plants—exercise the concepts described here, it is hoped that you will gain a hunger for a deeper understanding of thermodynamic concepts. I encourage you to do the supplemental problems in Section 18.

From this point on, I encourage you to see everything in your day-to-day life with an understanding of thermodynamics:

As you make a cup of coffee, you are transferring heat and changing phase (boiling and evaporation) (increasing the enthalpy of the water).

As that cup of coffee cools, it is transferring heat to the surroundings (its enthalpy is dropping), and its temperature is dropping.

When you press on the accelerator of your car, there are so many processes that are taking place, each of which has its own thermodynamic phenomena associated with it.

When you turn on a methanol or an LNG pump, realize that you are increasing the enthalpy of the fluid as you add work energy to that fluid. As that fluid then flows through the plant piping, realize that, as it gains or loses heat, the fluid is changing in enthalpy. As you vaporize LNG to send gas out to your customers, realize that you are changing phase and then warming the vapor to an acceptable temperature. As a liquid is flowing in a pipe there is a pressure drop that is due to viscous dissipation (friction losses) which result in a very slight heating of the fluid.

Our LNG and propane plants are essential to our industry's future. Your skill and ability in managing these assets is a critical part of the future. This publication ends with appreciation of you for your technical ability and your dedication to excellence. You are the Energy Industry's greatest asset.

Supplemental Problems

The following problems are intended to further exercise some of the concepts learned in the earlier sections of this publication. The answers are listed on the following page.

1) The mass equivalent of 2 million standard cubic feet per hour is fed into a vaporizer. The inlet conditions are (100% liquid at 300 psia with properties as shown on the liquid line of the pressure enthalpy chart). The liquid methane is vaporized and then heated to 60 deg F. How much heat must be added to each lbm? How much total heat must be added to the flow each hour?

2) The mass equivalent of 20,000 standard cu.ft./hr. of boil off at 15 psia and –240 deg F is heated in a boil-off heater to 60 deg F. How much heat must be transferred into each lbm of methane? How much total heat must be added to the flow each hour?

3) Methane gas is compressed adiabatically from 30 psia and 60 deg F to 60 psia and 180 deg F. What is the efficiency of the compressor? How much work is needed to compress 1,000 standard cubic foot per hour?

4) In a take station methane is dropped in pressure from 600 psia and 40 deg F to 350 psia. What is the expected temperature downstream of the pressure drop regulator? If the station flow is 10,000,000 standard cubic feet per hour, how much heat must be added to the methane to maintain it temperature at 40 deg F?

5) At one of our facilities, we wish to extract energy by using an expansion turbine to expand gas from our facility pressure to our 60-psig system. Assume that the flow through the turbine is 2 million standard cubic feet per hour, the inlet pressure is 200 psia, and the outlet pressure is 70 psia. How much energy could be extracted from an 80% efficient expander if the inlet temperature was at 60 deg F? What would be the temperature of the outlet gas? How much energy could be extracted from an 80% efficient expander if the inlet temperature was at 200 deg F? What would be the temperature of the outlet gas?

6) If a tank of liquid methane maintaining a constant pressure of 15 psia boils off at a rate of 20,000 standard cubic feet per hour, how much heat per hour is leaking into the tank?

1. Supplemental Problems—Answer Page

Table 10: Supplemental problems answer table page.

Problem	Answers
1	284 Btu/lbm 24,049,120 Btu/hr.
2	154 Btu/lbm 130,407 Btu/hr.
3	76.6% 2,710 Btu/hr.
4	22 deg F 4,234,000 Btu/hr.
5	4,064,640 Btu/hr. –38 deg F 5,080,800 Btu/hr. 87 deg F
6	185,449 Btu/hr.