TURBOMACHINERY & PUMP SYMPOSIA | HOUSTON, TX
SEPTEMBER 13-15, 2022
SHORT COURSES: SEPTEMBER 12, 2022

Refrigeration Compressor Basics For the Oil And Gas Industries

Medhat Zaghloul
Regional Technology Manager
Compressor Controls Corporation
Abu Dhabi, U.A.E.

Wayne Jacobson
Global Technology Manager
Compressor Controls Corporation
Des Moines, Iowa, U.S.A.

Medhat Zaghloul is CCC's Regional Technology Manager for Europe, the Middle East and Africa, based in Abu Dhabi. Medhat joined CCC in 1993, after a 15-year career in instrumentation and controls in the petrochemical industry. His responsibilities include providing technical guidance, supporting Sales, and developing technical solutions and control applications for CCC. Medhat has over 43 years of controls experience in a variety of up-, mid- and down-stream Oil & Gas facilities. Medhat holds a B.Sc. in Electrical Engineering from Helwan University, Egypt.

Wayne Jacobson is CCC's Global Technology Manager in Des Moines, IA.. Wayne joined CCC in 1996 as a Systems Engineer with a BS in Mechanical Engineering from the University of Wisconsin – Platteville. His responsibilities include providing technical guidance and support throughout the organization, develop and enhance control applications, maintain, and improve standard engineering and overseeing the overall technical development of engineering staff.

ABSTRACT

This tutorial discusses the design basics for refrigeration compressors, using the 2-section Propane refrigeration configuration as an example.

The benefits of incorporating a flash gas Economizer are discussed, as are the various methods of controlling the performance these machines, with the pros and cons for each method analyzed.

The proper antisurge control design is developed and correct recycle line arrangements are presented.

A case study of incorrect recycle line arrangements for a Single Mixed Refrigeration (SMR) Compressor, and the proposed modifications, is also presented.

INTRODUCTION

Many midstream and downstream oil and gas facilities utilize centrifugal compressors to drive process refrigeration circuits. This tutorial does NOT focus on the process refrigeration requirement and operation. Nor is it intended to be a guide on how to select or size a suitable centrifugal refrigeration compressor for a particular process duty.

Rather, this tutorial is intended to discuss the control aspects of centrifugal compressors used for process refrigeration duty. This tutorial will attempt to address:

- How the centrifugal compressor works as part of a process refrigeration cycle, with continuous reference to a pressure-enthalpy chart for that particular refrigerant.
- Why the thermodynamics of the refrigeration cycle influence the design and implementation of the compressor's control system, and,
- What are the best piping practices that are suitable for refrigeration centrifugal compressors.

The Basic Industrial Refrigeration Cycle

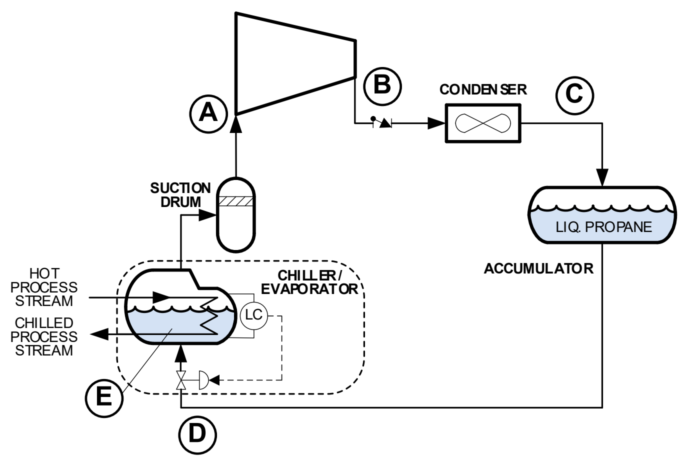

Figure 1 – Basic Process Refrigeration Circuit - Process Schematic

The above Figure 1 depicts a simplified process refrigeration circuit, using propane as the refrigerant. Propane vapor is collected from the process chiller (Evaporator) and sent to a centrifugal compressor (point A), which compresses the vapor to a pressure high enough (point B) so that the vapor will condense completely if cooled using available utilities (air cooler or water cooler) in a Condenser.

The liquid refrigerant that is compressed is collected in an Accumulator (point C) and then sent to the Evaporator. In most industrial applications a "kettle" type evaporator is used, where the liquid refrigerant is introduced, and its level maintained by a level control valve (point D).

As the liquid refrigerant passes through the level control valve, its pressure is dropped to the value where it commences to flash or partially, further lowering the temperature of the liquid fraction.

It is important to note that only the latent heat of vaporization of the liquid refrigerant in the Evaporator is useful to chill or cool the hot process stream and the refrigeration duty is therefore a function of the mass of refrigerant that vaporizes and its latent heat of vaporization. The portion of the liquid refrigerant that flashes (vaporizes) across the level control valve does not absorb heat from the process stream and therefore is not utilized to chill the process.

By adjusting the level control loop set-point, it is possible to completely or partially immerse all the process stream piping in the liquid refrigerant in the Evaporator – thus providing some degree of temperature control. Partially exposing some of the process tubing may serve to provide some amount of superheat to the vapor produced, but this is minimal and usually neglected. In this Tutorial, therefore, only the liquid refrigerant level completely immersing the hot process piping will be considered, i.e. the operating scenario which produces the maximum process chilling effect.

All the vapor produced in the Evaporator and its associated level control valve is collected in the compressor's suction drum and any entrained liquid is trapped, with the vapor being compressed again in the compressor (point 1) and thus the cycle repeats.

Introduction to the Pressure-Enthalpy (Mollier) Diagram

To better visualize and understand the refrigeration cycle, a Pressure/Enthalpy diagram of the refrigerant – also called a Mollier diagram – is used. This is illustrated in the following Figure 2.

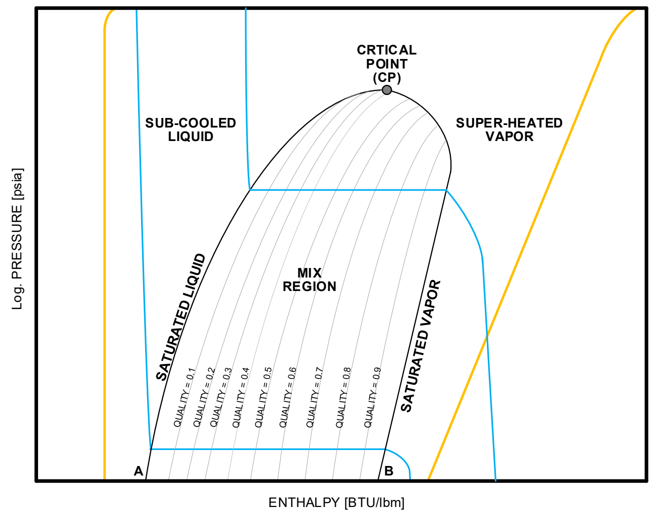

Figure 2 – Basic Pressure-Enthalpy (P-h) Diagram

Above Figure 2 shows the basic structure of a Pressure-Enthalpy (P-h) diagram, indicating the selected refrigerant's various thermodynamic states.

The area above and to the left of the saturation line for liquid (from points A to CP) is the area where the refrigerant is sub-cooled, i.e. its temperature is lower than the saturation temperature for that pressure.

The area above and to the right of the saturation line for gas (from points CP to B) is the area where the gas is superheated, i.e. the gas has a higher temperature than the saturation temperature at that pressure.

The area inside the "dome" (from points A to CP to B) represents the conditions where the refrigerant can change its state of aggregation from liquid to gas or vice versa. Hence, there is a mixture of gas and liquid. Depending on the amount of vapor inventory compared to the total mix inventory it is possible to obtain the "quality" of the mixture, which represents the mass fraction of the vapor content of the mix. For example, a quality of 0.2 signifies that 20% of the total refrigerant inventory is available as vapor, while 80% is available as liquid.

The practical meaning of the critical point (CP) is that at temperatures higher than this, the refrigerant cannot be condensed, no matter how high the pressure. Also, above the critical point the substance in question exhibits properties of both vapors and liquids in varying degrees, but no separate phases can exist.

Therefore, compression refrigeration systems normally operate at temperatures below the critical point.

Lines of constant temperature (isotherms) are almost vertical in the sub-cooled liquid region, horizontal (i.e. parallel to the constant pressure lines) in the liquid + vapor mixture region, and drop steeply towards the enthalpy axis in the superheated gas region. When these isotherms exhibit the steep drop after the saturated vapor curve, and become nearly vertical, at higher degrees of superheat and below the critical point, they are considered to represent near ideal gas behavior in the sense of being a function of temperature changes only and unaffected by pressure changes. These are represented by the blue lines in Figure 2.

Lines of constant entropy are vertical in the sub-cooled liquid region, and slope more and more as one moves to the right. These are represented by the orange lines in Figure **2**.

The pressure values are usually expressed in a logarithmic scale, while the enthalpy values are usually depicted in a linear scale.

Example of a Simple Refrigeration Cycle (Summer Case)

As an example, consider a simple industrial refrigeration system, comprising a centrifugal compressor, a suitably sized Condenser (using ambient air or available sea or fresh water as a coolant), an Accumulator, an Evaporator with its associated level control valve, and using propane (C_3H_8, and with a molecular weight of 44.097) as the refrigerant.

It is assumed that the suction pressure of the compressor, and hence the operating pressure of Evaporator is 24.0 psia. Referring to the P-h diagram for propane, it is found that at that pressure, propane evaporates at -22.6 degF.

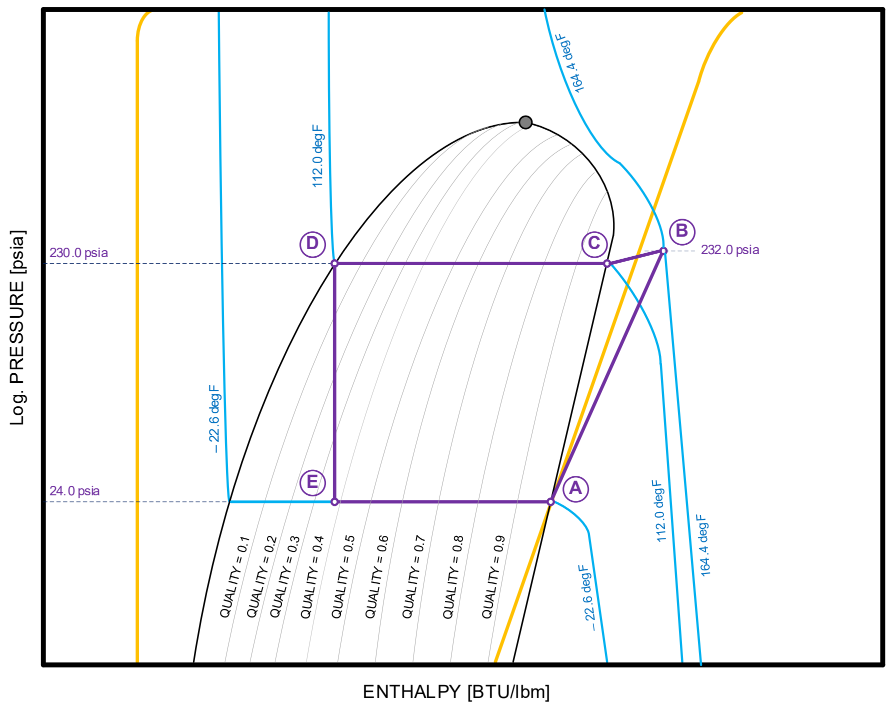

Figure 3 – Theoretical Propane Refrigeration Simple Cycle (Summer Conditions) in a P-h Diagram

The propane refrigerant entering the compressor is depicted at point A in the above Figure **3**. The propane vapor is compressed polytropically in the centrifugal compressor, i.e. across several lines of entropy, until it reaches point B, to the right of the saturated vapor boundary, and therefore “superheated”.

The superheated vapor then enters the Condenser, where its superheat is removed, and the vapor commences to change state to liquid at point C. This happens at a constant pressure (from point C to D) and it shall be assumed that all of the vapor condenses to liquid at point D; in this example at a temperature of 112.0 degF.

In a real system there will be some pressure losses in the piping, but this may be ignored. The saturated liquid refrigerant (at 230.0 psia and 112.0 degF) is then reduced in pressure isenthalpically (i.e. at the same enthalpy value, without any change in heat content per unit of mass) by passing through the level control valve, and so enters the Evaporator at point E, vertically below point D, at a pressure of 24.0 psia.

As may be seen on the P-h diagram, point E lies at a quality value that signifies that some of the refrigerant has already vaporized, by flashing through the level control valve, and the remainder of the liquid collected in the Evaporator is available to absorb the heat from the hot process stream. This occurs at the constant pressure of 24.0 pia until all the liquid refrigerant vaporizes, producing a saturated vapor at the phase boundary at point A. The cycle is now complete.

The amount of heat (in BTU/h) that may be removed from the hot process stream may be calculated as:

$$heat_{absorbed} = \dot{m} \cdot \Delta h \qquad (1)$$

With:

Δh being the difference in enthalpy between points E and A, and

$\dot{m}$ being the mass flow rate through the Evaporator.

The work equivalent (in hp) of the Evaporator duty may be calculated as:

$$Work_{evap} = heat_{absorbed} \cdot 0.000393 \qquad (2)$$

And the maximum refrigeration duty (in TR – tonnes of refrigeration) of the Evaporator estimated as:

$$Ref_{duty} = Work_{evap} \,/\, 4.7161 \qquad (3)$$

The compressor polytropic head (in ft-lb/lbm) may be calculated as:

$$H_p = R_o \cdot \frac{Z_{avg} \cdot T_s}{MW} \cdot \frac{R_c^{\sigma} - 1}{\sigma} \qquad (4)$$

With:

(σ) being the polytropic exponent, and may be calculated as:

$$\sigma = \frac{k-1}{k \cdot \eta_p} \qquad (5)$$

And (R_c) being the pressure ratio of the compressor, or $\left(\frac{P_d}{P_s}\right)$.

It should be noted that this tutorial is not intended to be a sizing guideline for the process equipment used in the refrigeration circuit but shall focus on the control aspects of these circuits. Therefore, several simplifications have been made, such as ignoring piping frictional losses.

Basic Industrial Refrigeration Example (Summer Case)

It is proposed to use in this example a process chilling application that requires 200,000 lb/h of propane refrigerant. This simple refrigeration circuit now has the following values associated with it:

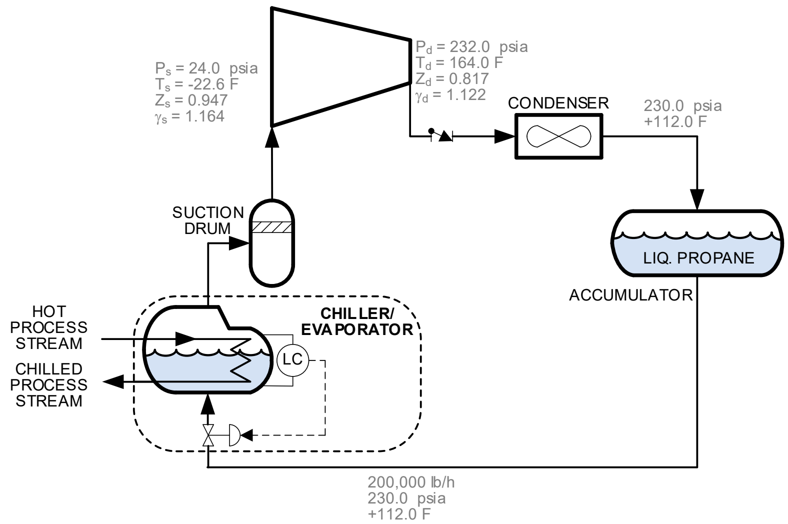

Figure 4 – Simple Refrigeration Circuit for a Typical Summer Operating Case - Process Schematic

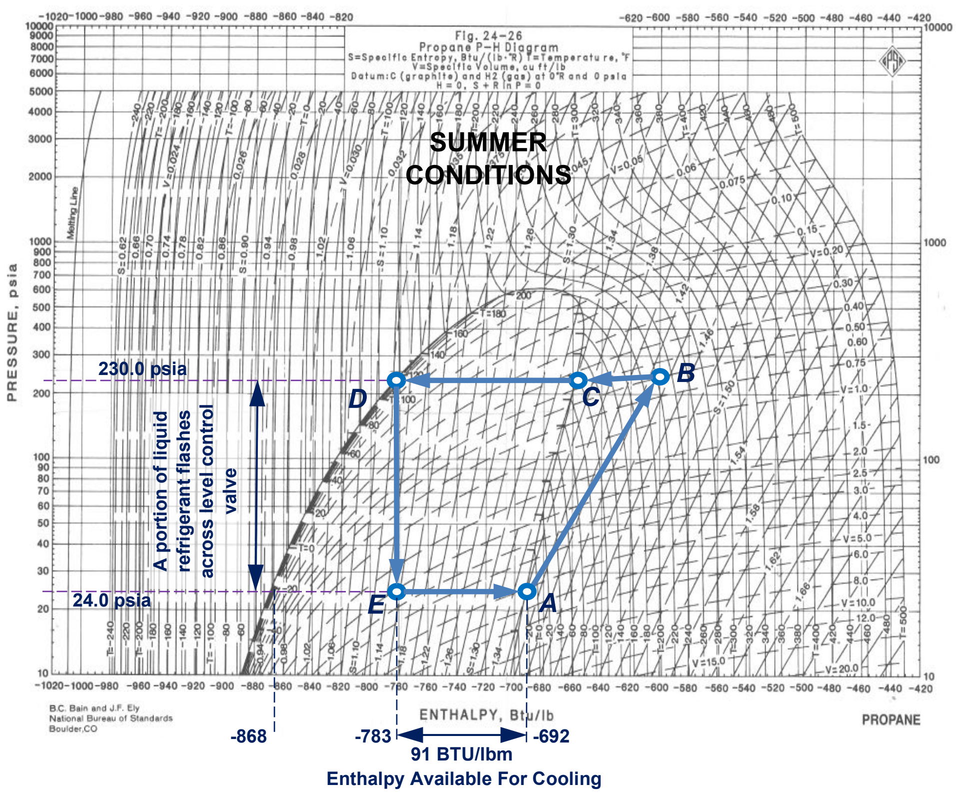

Figure 5 – Actual P-h Diagram for the Summer Operating Case

The gas compressibility factors and the specific heat ratio values for the refrigerant vapor as it enters and discharges from the compressor are obtained from NIST tables. It is further assumed that the process Condenser is capable of cooling the compressed refrigerant to a temperature of 112.0 degF, to ensure that it condenses fully.

By plotting the operating conditions of the compressor on to a P-h diagram for propane, it is possible to obtain, graphically, the heat of vaporization the Evaporator will be able to absorb from the hot process stream, as well as the amount of liquid refrigerant inventory that flashes through the level control valve (without significantly cooling the process), and thus be able to derive the amount of remaining liquid refrigerant that vaporizes inside the Evaporator that is in contact with the hot process stream (thus providing the required refrigeration duty).

As may be seen from the following Figure 5, the heat that may be absorbed from the hot process stream is approx. 91.0 BTU/lbm of the refrigerant, considering that some refrigerant has flashed across the level control valve.

For a total refrigerant flow rate of 200,000 lb/h, the total heat absorbed in this circuit from the hot process stream may therefore be calculated as 18,200,000 BTU/hr, or 7,153 hp which is the equivalent of 1,516.7 Tonnes of Refrigeration duty.

The polytropic head developed by the compressor to circulate the refrigerant through the circuit may be calculated as 38,805 ft-lb/lbm and the gas power demand on the compressor driver may be estimated as 4,657 hp.

This produces a coefficient of performance (COP) of $\left(\frac{\mathbf{7{,}153}}{\mathbf{4{,}657}}\right)$ = 1.536 with the coefficient of performance being the amount of process refrigeration obtainable (in hp equivalent) divided by the gas power demand (also in hp) of the compressor.

Example of a Simple Refrigeration Cycle (Winter Case)

If the previous refrigeration circuit for the winter operating conditions is re-examined, a couple of points may be observed:

- The ambient air or water temperature that serves as the cooling fluid in the Condenser is significantly lower than in summer, and
- A continuous process fluid chilling duty, unlike a commercial or residential air conditioning unit, will probably require the same refrigeration duty in winter as in summer.

Therefore, in the presented stylized and theoretical P-h diagram, it is expected that point A remains at the same pressure of 24.0 psia and a temperature of -22.6 degF. The compressor, however, no longer needs to compress the vapor up to 232.0 psia to condense. It will suffice for the compressor to raise the refrigerant pressure up to just 138.0 psia so it will condense completely at a temperature of 75.0 degF (Winter conditions).

Point B is thus at a lower pressure value and the condenser completely condenses the compressed vapor to a saturated liquid in the Accumulator at a pressure of 135.0 psia. This is significantly lower than the Summer operating case and so it is expected that the compressor will work less in Winter.

At this reduced Winter operating temperature and pressure (135.0 psia and 75.0 degF), point D is located at a higher enthalpy differential than in Summer, and so the available amount of latent heat of vaporization is larger; also the amount of flashing across the level control valve is less, making more mass of liquid refrigeration available for process cooling duty. It can be expected to require less refrigerant inventory to circulate in the circuit in Winter than in Summer for the same process refrigeration duty. This is illustrated in the following Figure 6.

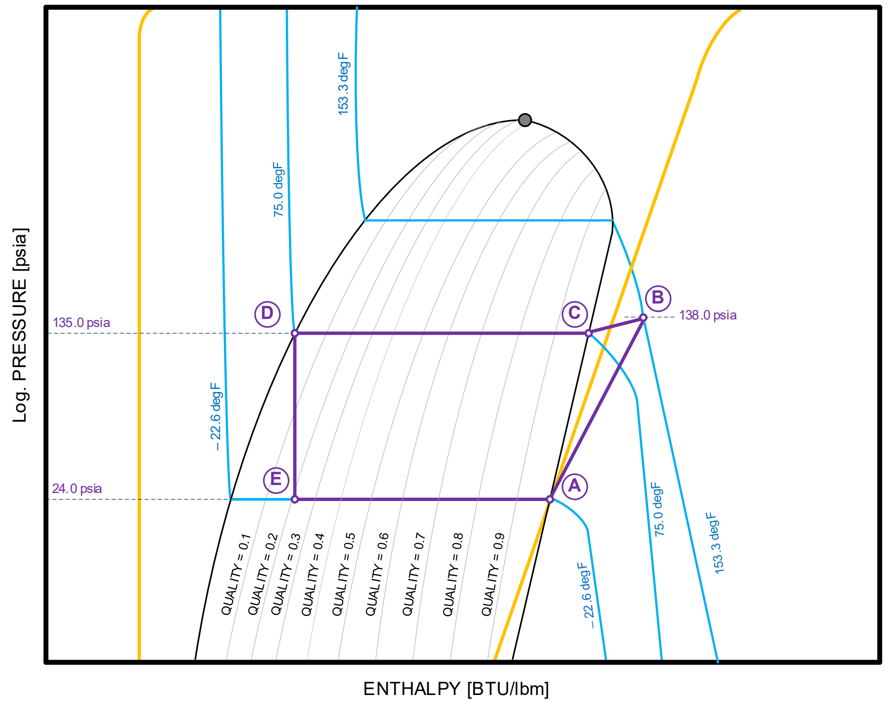

Figure 6 – Theoretical Propane Refrigeration Simple Cycle (Winter Conditions) in a P-h Diagram

Basic Industrial Refrigeration Example (Winter Case)

As mentioned in the previous section, it is considered that it is needed to keep the process chilling duty at the same value as in Summer, i.e. 1516.7 TR, or 7,153 hp, which is the equivalent of 18,200,000 BTU/hr of heat capable of being absorbed by the Evaporator from the hot process stream.

Values from the actual P-h diagram for propane indicate that, for point E for Winter operating conditions, the available enthalpy differential (Δh) is 116 BTU/lbm of liquid refrigerant, and the quality is lower, which means more liquid is available for vaporization in the Evaporator, as opposed to being flashed across the level control valve. It is therefore possible to calculate the mass of liquid propane required in the circuit for Winter conditions ($\dot{m}$) as 18,200,000 ÷ 116 = 156,897 lbm/h.

The compressor needs to develop a polytropic head of only 28,017 ft-lb/lbm in Winter; and the gas power demand on the compressor driver may therefore be estimated as 2,779 hp. This produces a coefficient of performance (COP) for Winter of $\left(\frac{\mathbf{7,153}}{\mathbf{2,779}}\right)$ = 2.574; which is a significant improvement over Summer conditions. The simple refrigeration circuit schematic now has the following values associated with it:

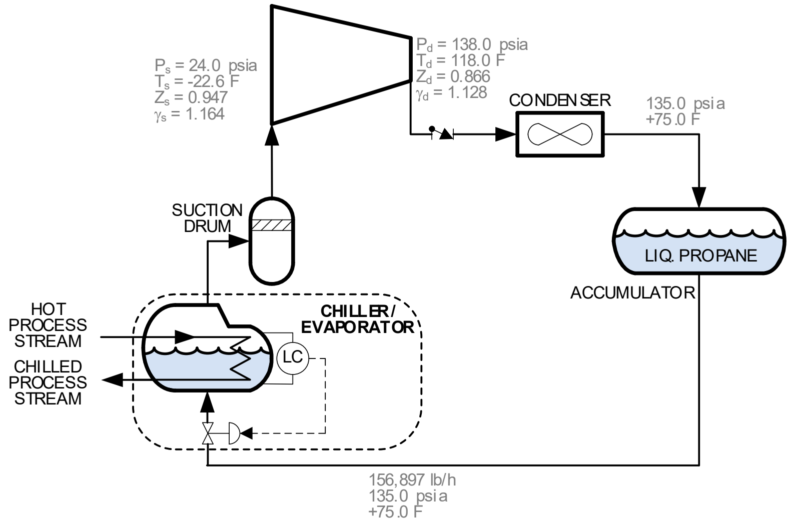

Figure 7 – Simple Refrigeration Circuit for a Typical Winter Operating Case - Process Schematic

And the actual P-h diagram associated with the Winter Operating case becomes:

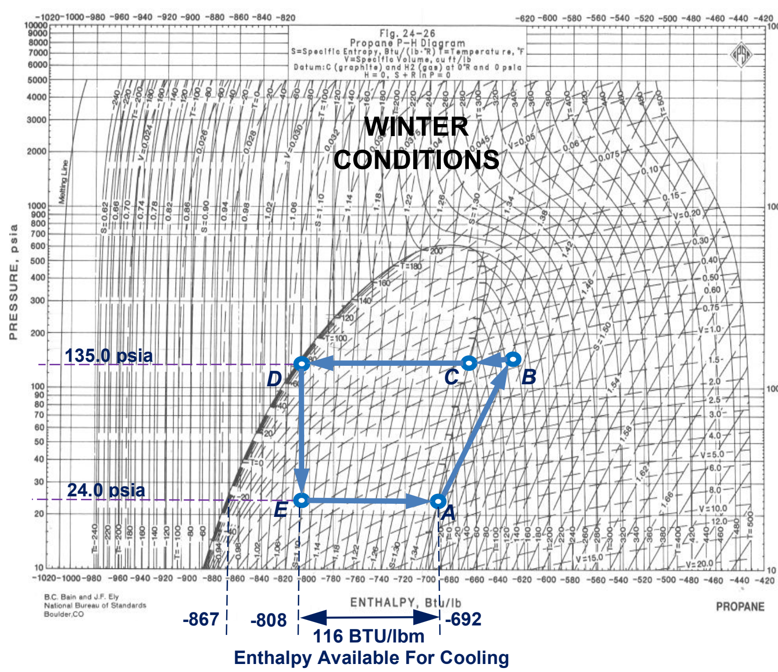

Figure 8 – Actual P-h Diagram for the Winter Operating Case

The Industrial Refrigeration Cycle Tends to be Self-Regulating

It may be observed from the Summer operating case of a single speed refrigeration compressor, per Figure 5, there are two locations in the circuit where a phase change takes place:

- At the discharge end – in the Condenser – there is a change of state from vapor to liquid, with the pressure being "fixed" by the condensing temperature on a seasonal basis, and

- At the suction end of the compressor – in the Evaporator – there is another change of state, with the pressure also "fixed" by process design considerations.

It follows that the suction pressure would vary as a function of the liquid propane refrigerant evaporated by the process load. The higher the process load, the more vapor is formed, and the higher the suction pressure becomes.

This will produce a lower pressure ratio (R_c) in the compressor, which, in turn, means that the compressor's operating point would ride down and to the right of the performance curve, and the compressor would be capable of delivering more throughput without any external manipulation.

Thus, the illustrated single speed refrigeration compressor would be self-regulating, to a certain degree, for small changes in process loads.

The Single Casing Simple Refrigeration Compressor is Rarely Implemented

The basic refrigeration cycle as depicted in Figure 4, above, is rarely implemented. Many basic propane refrigeration compressors are designed to split the total required pressure ratio (in the previous example equivalent to $\frac{232.0}{24.0} = 9.667$) over two or more compressor sections, each containing one or more sections of compression.

From a mechanical design aspect, it is likely that splitting the overall pressure ratio that is required over two or more compressor sections would result in lower values of vibration and mechanical stresses than would be expected in one high pressure ratio section.

It is therefore common engineering practice to split the overall pressure ratio required for propane refrigeration duty (as depicted in Figure 4) into two sections of compression.

For this Tutorial, it is possible to assume an "equitable" pressure ratio split among the two sections of compression, or $\sqrt{9.667} = 3.11$.

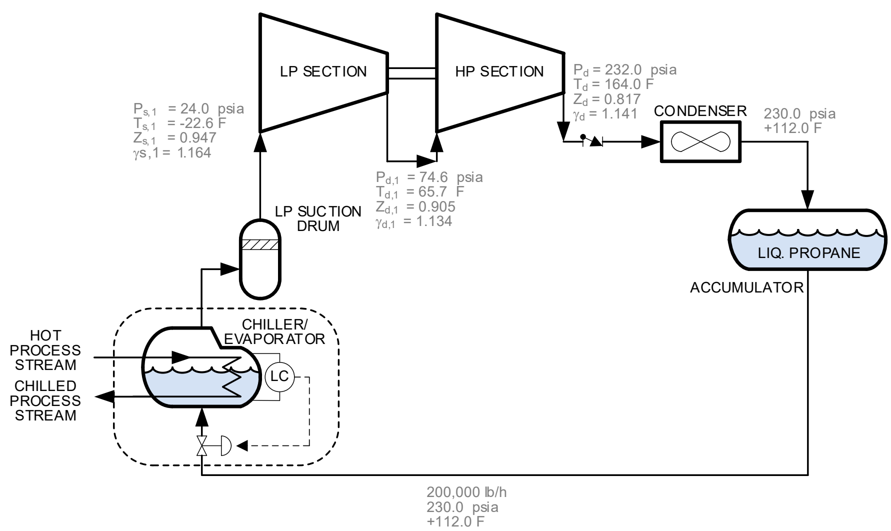

Figure 9 – Two-Section Refrigeration Circuit for a Typical Summer Operating Case - Process Schematic

The above Figure 9 illustrates the new 2-section compressor configuration used for the summer operating case, which provides the same process refrigeration duty of 18,200,000 BTU/hr, or 7,153 hp which is the equivalent of 1,516.7 Tonnes of Refrigeration. The same pressure-enthalpy diagram of Figure 5 may thus be applied.

If the refrigeration cycle is represented on the theoretical P-h diagram for propane, it will be observed that the only change is the addition of the intermediate point A' on the polytropic line of compression (from points A to B) that represents the newly created interstage at 74.6 psia and 68.9 degF. The process refrigeration duty remains unchanged at 1,526.7 Tonnes of Refrigeration.

This is illustrated in the following Figure 10.

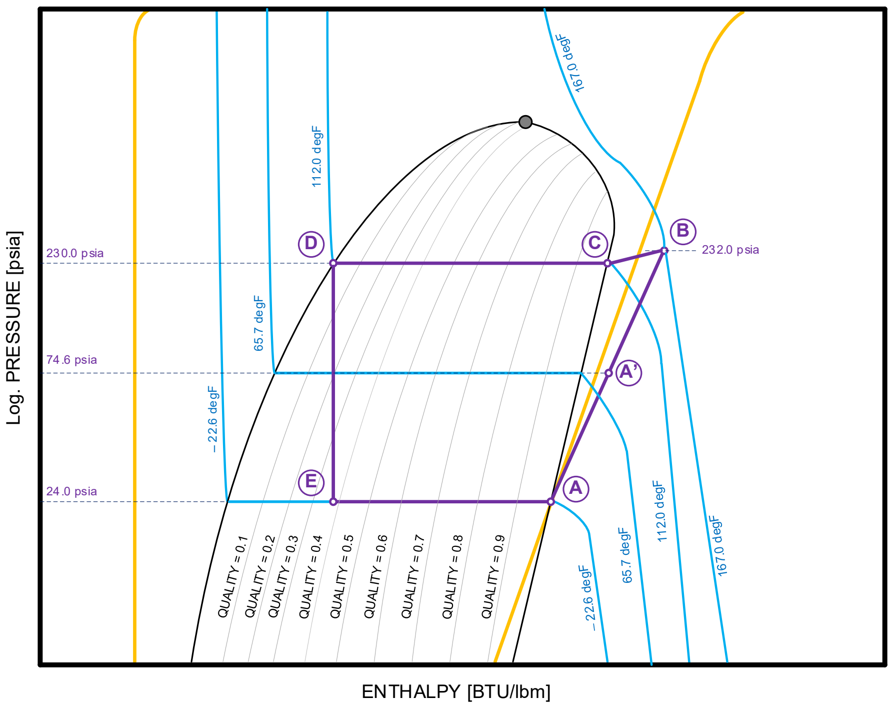

Figure 10 – Two-Section Propane Refrigeration Cycle (Summer Conditions) in a Theoretical P-h Diagram

Adding a Flash Gas Economizer to the Compressor Interstage

It is common practice nowadays to add a Flash Drum, called an "Economizer" to the compressor interstage, which allows a portion of the total liquid refrigerant collected in the Accumulator to be completely vaporized and the resulting vapor directed to the 2nd section of the Compressor.

This serves two simultaneous goals:

- The vapor produced by the Economizer is at a significantly lower temperature than the Compressor's 1st section discharge vapor and when it mixes with the 1st section discharge vapor, it cools it to entering the 2nd section. This results in lowering the resultant 2nd section discharge temperature which reduces the required heat load and duty of the Condenser for the same total mass flow as before.
- Because of the portion of liquid refrigerant that has evaporated or "flashed" in the Economizer, the remaining liquid exits the Economizer at a lower temperature and a correspondingly lower Enthalpy than in the simple single section refrigeration cycle. To maintain the same process cooling as in the previous example, a reduced mass flow rate of refrigerant is required in the Evaporator and the associated 1st section of compression, which reduces its required compression power.

In the theoretical P-h diagram of Figure **11**, the total refrigerant vapor mass flow is de-superheated and then condenses completely in the Condenser from points B to C to D, just as in the case of the simple cycle without the Economizer.

The entire liquid refrigerant enters the Economizer where like in the process chiller, a portion of the liquid refrigerant is flashed across the level control valve and part of the liquid evaporates. The total vapor phase is sent to the 2nd section of the Compressor where it combines with the vapor from the 1st section.

Because of the combined effects of flashing across the Economizer's level control valve and the vaporization of a portion of the liquid inventory in the Economizer, the remaining liquid in the Economizer exits at a temperature of approx. +35.0 degF. This is illustrated by the point D'' in the Figure 11.

The remaining liquid refrigerant (now at a temperature of approximately 35 degF, instead of 112 degF as in the previous configuration, is then sent to the Evaporator (process chiller) where the pressure drop across its level control valve is much lower than in the simple cycle (because of the intermediate pressure at which the Economizer operates), and this results in a significantly higher proportion of remaining liquid refrigerant being available for process cooling duty.

Also, the fact that point E on the P-h diagram of Figure 11 is "pushed" to the left means that a higher enthalpy differential is available per unit mass of liquid refrigerant in the Evaporator for process cooling.

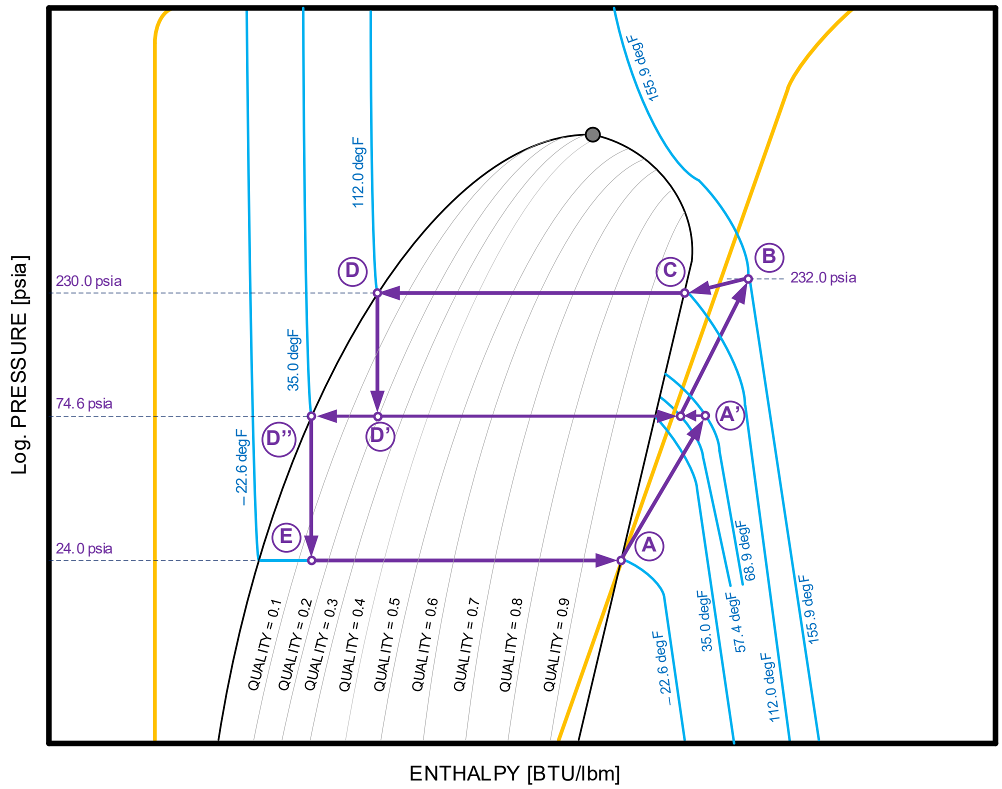

Figure 11 – Two-Section Propane Refrigeration Cycle – with an Economizer (Summer Conditions) – P-h Diagram

This produces a process chilling application that requires a total mass flow rate of 200,000 lb/h of propane refrigerant and has the following values associated with it:

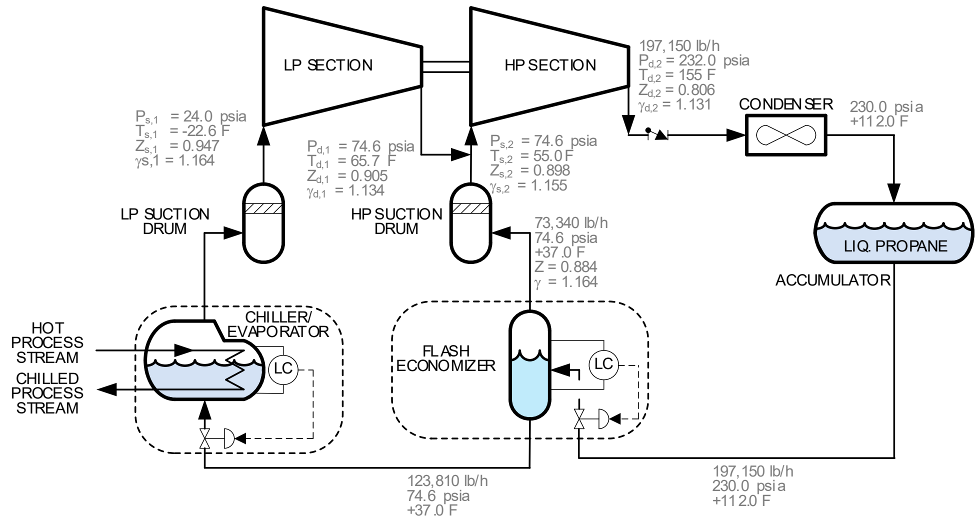

Figure 12 – Two-Section Propane Refrigeration Cycle – with an Economizer (Summer Conditions) – Process Schematic

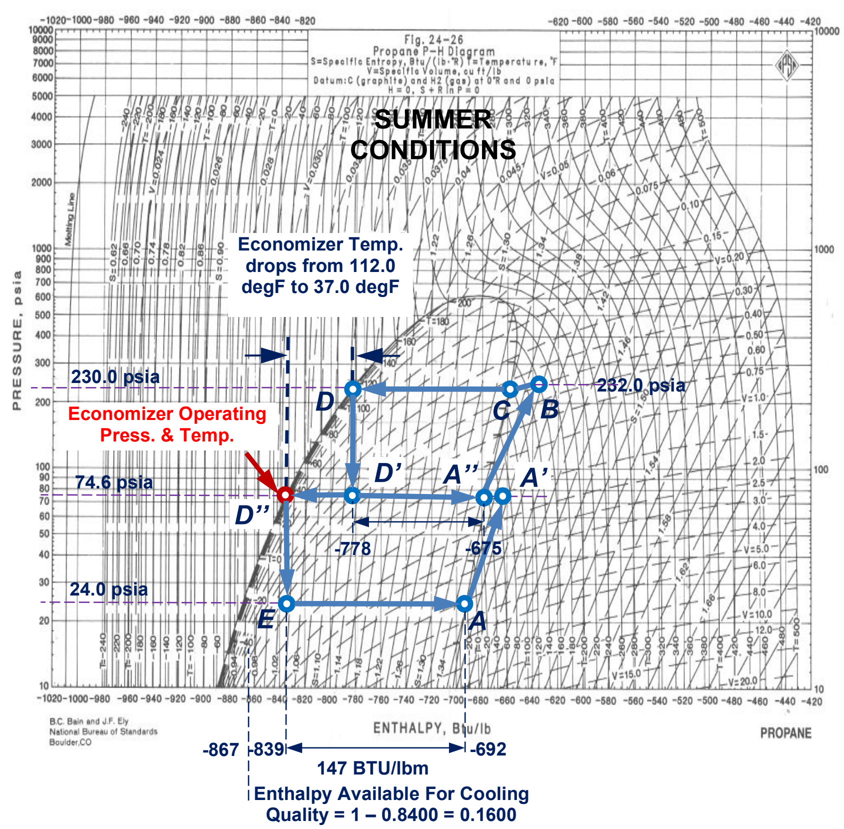

Figure 13 – Actual P-h Diagram for the Summer Operating Case – with an Inter-section Flash Economizer @ 74.6 psia

As may be observed from Figure 13, at the new inlet conditions of the Evaporator (74.6 psia and 37.0 degF), or point E, the quality of the refrigerant in the Evaporator becomes 0.1600, indicating that the amount of liquid refrigerant that is flashed across the Evaporator level control valve is quite small, leaving more liquid refrigerant for process cooling duty.

It is also observed from Figure 13 that the enthalpy differential available for process cooling in the Evaporator is approximately 147 BTU/lbm of refrigerant; which means that in order to have the same refrigeration duty of 1,516.7 TR (or 18,200,000 BTU/hr) as before, a total amount of (18,200,000 ÷ 147) = 123,810 lb/hr must be vaporized in the Evaporator.

The mass flow (in units lb/hr) through the Economizer may be calculated as:

$$\dot{m}_{ECON} = \dot{m}_{EVAP} \div (1 - Qual_{ECON}) \qquad (6)$$

From above Figure 13, the quality of the Economizer may be estimated as:

$Qual_{ECON} = \left(\frac{-839+778}{-839+675}\right) = 0.372$

$\dot{m}_{ECON} = 123{,}810 \div (1 - 0.372) = 197{,}150$ lb/h

The amount of refrigerant that is vaporized in the Economizer may be estimated as (197,150 – 123,810) = 73,340 lb/hr.

It is possible to develop the following compressor audit:

	Symbol	Units	LP Section	HP Section
Molecular Weight	MW	---	44.097	
Inlet pressure	P_s	psia	24.0	74.6
Inlet temperature	T_s	degF	-22.6	55.0
Inlet compressibility factor	Z_s	---	0.947	0.898
Inlet specific heat ratio	k_s	---	1.164	1.155
Mass flow rate	W	lbm/h	123,810	197,150
Inlet density	ρ_s	lb/ft^3	0.2383	0.6632
Volumetric flowrate	Q_s	ACFH	519,596	297,224
Outlet pressure	P_d	psia	74.6	232.0
Outlet temperature	T_d	degF	65.7	155.0
Outlet compressibility factor	Z_d	---	0.9050	0.806
Outlet specific heat ratio	k_d	---	1.134	1.131
Estimated polytropic efficiency	η_p	---	0.800	0.800
Polytropic head	H_p	ft-lb/lbm	17,660	19,079
Estimated gas power	P_g	hp	1,382	2,376

Table 1 – Two-section Compressor Operating Data – Summer Conditions Without Economizer Throttle Valve

The total polytropic head developed by the two compressor sections is calculated as 36,739 ft-lb/lbm; and the gas power demand on the compressor driver may be estimated as 3,758 hp.

To produce the same process refrigeration duty as before, the addition of the interstage flash gas Economizer has resulted in a gas power demand reduction of (4,657 – 3,758) = 899 hp, or 19.3% lower than the configuration without the Economizer.

Considering that the average price of electricity for industrial use in the U.S.A is $0.07 per kw/h, this would lead to an annual reduction of approximately:

899 hp * 0.7457 hp/kw-h * 24 hr/day * 360 days/year * 0.07 $/kw-h = $ 405,448 per year.

The coefficient of performance (COP) now becomes $\left(\frac{\mathbf{7{,}153}}{\mathbf{3{,}758}}\right) = 1.903$.

Adding a Throttle valve to the Flash Gas Economizer

Many refrigeration compressor design layouts include a throttle valve installed between the flash gas economizer and the inlet flange for the 2nd section sidestream inlet.

The modulation of this throttle valve is usually based on controlling the pressure in the flash gas economizer, as may be seen in Figure 15.

However, based on the previous analysis, it becomes evident that any partial opening of this newly introduced throttle valve can only raise the pressure of the flash gas economizer beyond the operating pressure of the 2nd section of the compressor, with the throttle valve absorbing the resultant pressure differential.

This means that the more the throttle valve closes, the higher the operating pressure of the flash gas economizer becomes which reduces its effect of lowering the temperature of the subcooled refrigerant that is directed to the process Evaporator.

If the previous compressor, with the same sectional performance curves and the same inter-section operating pressure is maintained, is now considered with a throttle valve added between the Economizer and the 2nd-section of the compressor, it is possible to illustrate this in the following Figure 14:

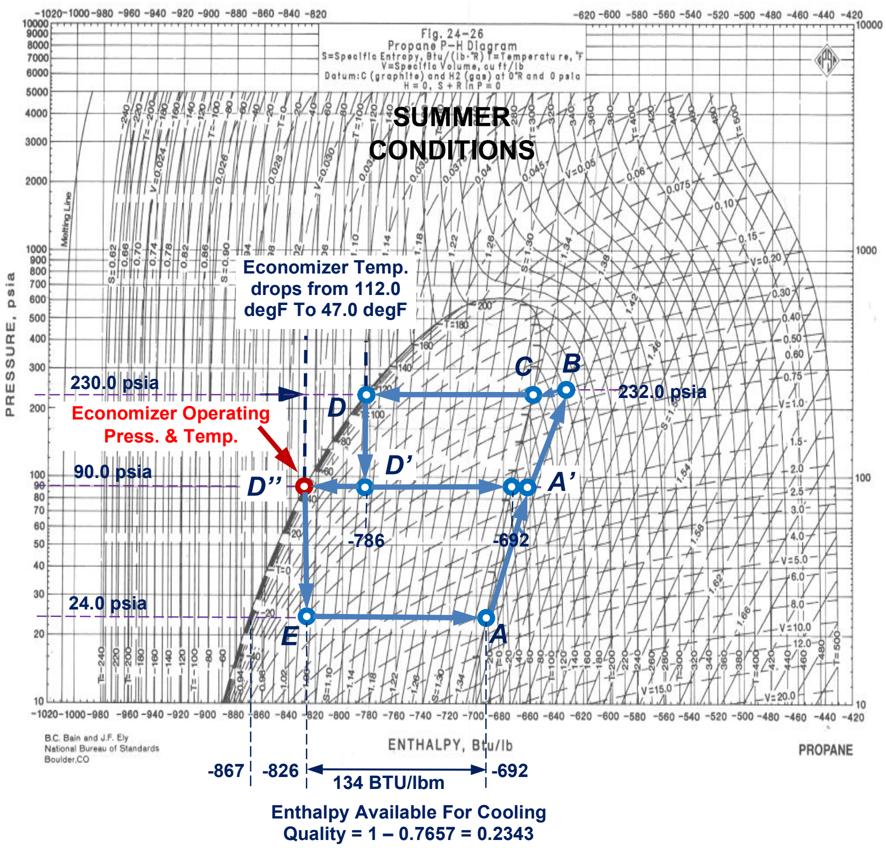

Figure 14 – Actual P-h Diagram for the Summer Operating Case – With an Inter-section Flash Economizer @ 90 psia

As may be observed from Figure 14, at the new inlet conditions of the Evaporator (24.0 psia and -22.6 degF), or point E, the quality of the refrigerant in the Evaporator becomes 0.2343, indicating that the amount of liquid refrigerant that is flashed across the Evaporator level control valve has increased compared to the scenario where the inlet temperature is lower. The Evaporator will require a higher total liquid refrigerant mass flow rate to produce the same process cooling duty.

In this scenario, the enthalpy differential available for process cooling in the Evaporator is approximately 134 BTU/lbm of refrigerant which means that in order to have the same refrigeration duty of 1,516.7 TR (or 18,200,000 BTU/hr) as before, a total amount of (18,200,000 ÷ 134) = 135,820 lb/hr must be vaporized in the Evaporator.

The mass flow through the Economizer may be calculated as:

$$\dot{m}_{ECON} = \dot{m}_{EVAP} \div (1 - Qual_{ECON})$$

From above Figure 14, the quality of the Economizer may be estimated as:

$$\text{Qual}_{\text{ECON}} = \left(\frac{-826+786}{-826+692}\right) = 0.299$$

$$\dot{m}_{ECON} = 135{,}820 \div (1 - 0.299) = 193{,}701 \text{ lb/h}$$

The amount of refrigerant that is vaporized in the Economizer may be estimated as (193,701 – 135,820) = 57,881 lb/hr.

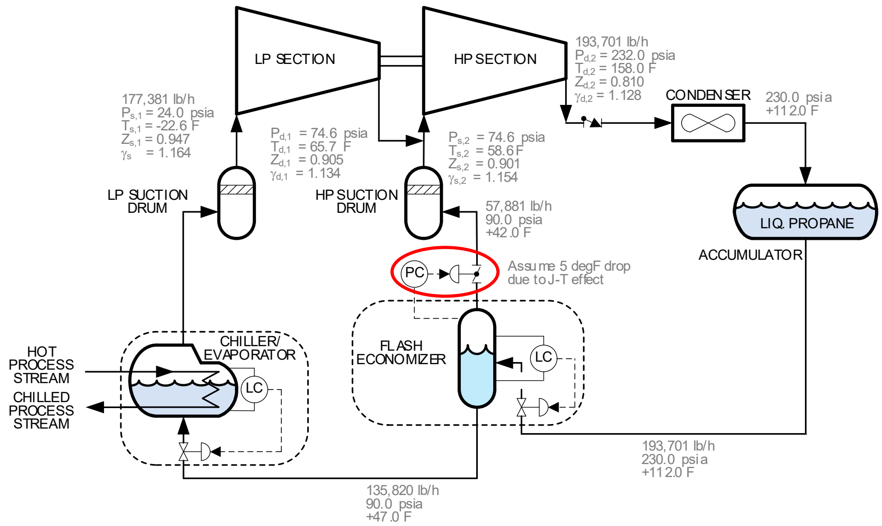

Figure 15 – Two-Section Propane Refrigeration Cycle – with an Economizer and Throttle Valve – Process Schematic

It is observed that without the throttle valve, or fully open valve, the Economizer operates at 74.6 psia and +37.0 degF (see Figure 12):

- 123,810 lb/h of liquid refrigerant is sent to the Evaporator where approx. 84.0% is utilized for process chilling, producing a refrigeration duty of 1,516.7 Tonnes.

When, on the other hand, the throttle valve is present and modulated so that the Economizer operates at 90.0 psia and +47.0 degF:

- 135,820 lb/h of liquid refrigerant is sent to the Evaporator where approx. 76.6% is utilized for process chilling, producing the same refrigeration duty of 1,516.7 Tonnes.

By assuming a 5 degF temperature drop (due to the Joules-Thomson effect as the refrigerant vapor produced in the Economizer traverses the throttling valve) and recalculating the two-section compressor audit for these new conditions, produces the following audit:

	Symbol	Units	LP Section	HP Section
Molecular Weight	MW	---	44.097	
Inlet pressure	P_S	psia	24.0	74.6
Inlet temperature	T_S	degF	-22.6	58.6
Inlet compressibility factor	Z_S	---	0.947	0.901
Inlet specific heat ratio	k_S	---	1.164	1.154
Mass flow rate	W	lbm/h	135,820	193,701
Inlet density	ρ_S	lb/ft^3	0.2383	0.6564
Volumetric flowrate	Q_S	ACFH	569,998	295,049
Outlet pressure	P_d	psia	74.6	232.0
Outlet temperature	T_d	degF	65.7	158.0
Outlet compressibility factor	Z_d	---	0.905	0.810
Outlet specific heat ratio	k_d	---	1.134	1.128
Estimated polytropic efficiency	η_p	---	0.800	0.800
Polytropic head	H_p	ft-lb/lbm	17,660	19,271
Estimated gas power	P_g	hp	1,516	2,359

Table 2 – Two-section Compressor Operating Data – with Economizer Throttle Valve & 90.0 psia Economizer Operating Pressure

The total polytropic head developed by the two sections of the compressor is calculated as 36,931 ft-lb/lbm – a rise of only 0.5 % over the previous scenario where there was no throttle valve downstream of the Economizer. The gas power demand on the compressor driver, however, has increased to 3,875 hp – a rise of 3.1% over the example with no throttle valve.

The additional horsepower required to drive the compressor (3,875 – 3,758 = 117 hp) would cost the asset owner of the facility on an annual basis:

177 hp * 0.7457 hp/kw-h * 24 hr/day * 360 days/year * 0.07 $/kw-h = $ 52,767 per year

Also, the coefficient of performance (COP) now degrades to $\left(\frac{\mathbf{7{,}153}}{\mathbf{3{,}875}}\right)$ = 1.846, or a reduction of approximately 3 % compared to operating the Economizer without a throttle valve.

It may be concluded that the usage of a throttling valve between the Economizer and the refrigeration compressor's 2nd section, for the same 2-section compressor configuration and unchanged sectional performance curves, can result in only lowering the coefficient of performance, without producing any process benefit whatsoever.

Converting the Economizer to a Process Pre-cooler

Instead of improving the coefficient of performance by the addition of an Economizer, some refrigeration compressor process designs operate by vaporizing some of the refrigerant inventory collected in the Accumulator in a second process heat exchanger (or Pre-cooler).

In this tutorial it is assumed that in such a configuration, a 50 - 50 % split occurs between the two process Evaporators, in terms of handling the total liquid refrigerant inventory.

This would produce a schematic layout as per Figure 16.

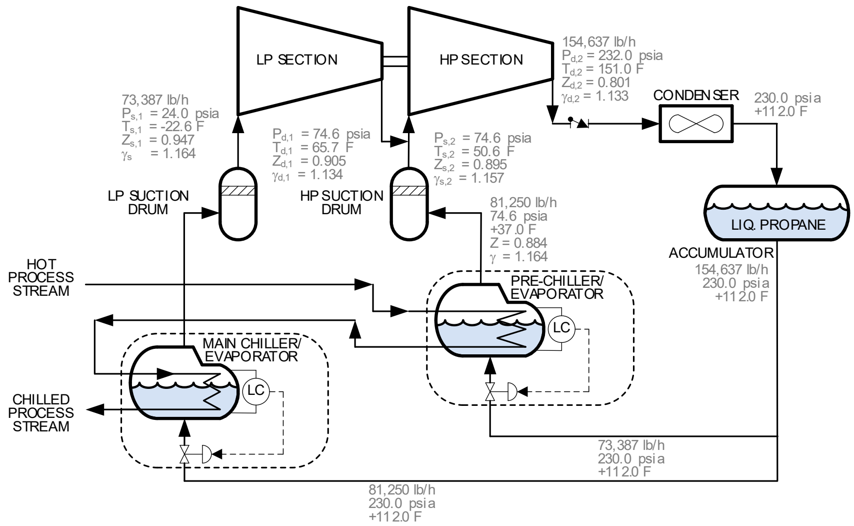

Figure 16 – Two-Section Propane Refrigeration Cycle – with Dual Process Chillers in Series – Process Schematic

The Summer conditions P-h diagram for the Dual Chiller configuration is given in Figure 17.

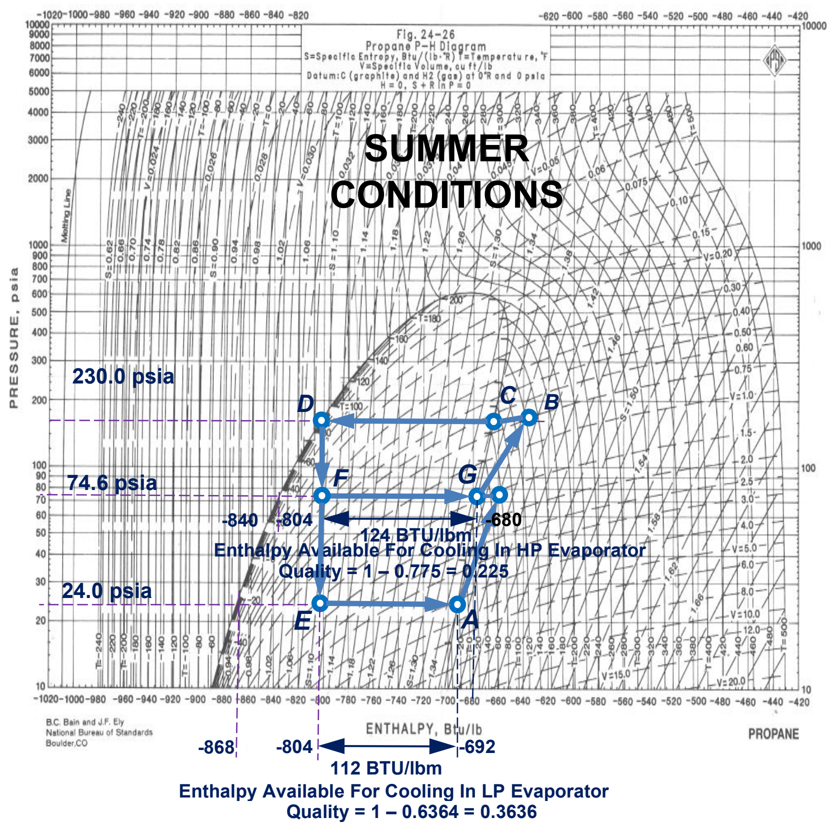

Figure 17 – Actual P-h Diagram for the Summer Operating Case – with an Interstage HP Evaporator

The trajectory (point D to point F to point G) represents the path of the refrigerant from the shared Accumulator to the HP Evaporator where flashing across the level control valve takes place between points D and F; and evaporation of the remaining liquid as it cools the process stream takes place between points F and G.

If this configuration was to be designed to produce the same refrigeration duty as the previous scenarios (Summer conditions without an Economizer; and Summer conditions with an Economizer), then the total refrigeration duty of 1,516.7 TR may be split 50 – 50 % across the LP and HP Evaporators.

As may be seen, from the above Figure 17, there is approximately 124 BTU/lbm of enthalpy available for process chilling duty in the HP Evaporator. To produce 758.35 TR from the HP Evaporator, or 9,100,000 BTU/hr, a mass flow rate of 73,387 lb/hr is required to be routed to HP Evaporator.

The trajectory (point D to point E to point A) represents the path of the refrigerant from the shared Accumulator to the LP Evaporator, where flashing across the level control valve takes place between points D and E. Evaporation of the remaining liquid as it cools the process stream takes place between points E and A.

There is approximately 112 BTU/lbm of enthalpy available for process chilling duty in the LP Evaporator. To produce 758.35 TR from the LP Evaporator, or 9,100,000 BTU/hr, a mass flow rate of 81,250 lb/hr is required to be routed to LP Evaporator.

In terms of the refrigeration compressor operation, we may consider the following audit:

	Symbol	Units	LP Section	HP Section
Molecular Weight	MW	---	44.097	
Inlet pressure	P_s	psia	24.0	74.6
Inlet temperature	T_s	degF	-22.6	50.6
Inlet compressibility factor	Z_s	---	0.947	0.895
Inlet specific heat ratio	k_s	---	1.164	1.157
Mass flow rate	W	lbm/h	81,250	154,637
Inlet density	ρ_s	lb/ft^3	0.2383	0.6712
Volumetric flowrate	Q_s	ACFH	340,983	230,366
Outlet pressure	P_d	psia	74.6	232.0
Outlet temperature	T_d	degF	65.7	151.0
Outlet compressibility factor	Z_d	---	0.905	0.801
Outlet specific heat ratio	k_d	---	1.134	1.133
Estimated polytropic efficiency	η_p	---	0.800	0.800
Polytropic head	H_p	ft-lb/lbm	17,660	18,847
Estimated gas power	P_g	hp	907	1,840

Table 3 – 2-section Compressor Operating Data – with an Interstage Evaporator

The total polytropic head developed by the two sections of the compressor is calculated as 36,507 ft-lb/lbm – a slight drop of only 0.6 % below the Economizer configuration. However, the gas power demand on the compressor driver has decreased to 2,747 hp – a drop of 26.9 %.

The reduced horsepower required to drive the compressor (being 1,011 hp) would save the asset owner of the facility, compared to the scenario where there is an Economizer, on an annual basis:

1,011 hp * 0.7457 hp/kw-h * 24 hr/day * 360 days/year * 0.07 $/kw-h = $ 455,960 per year.

Also, the coefficient of performance (COP) now improved to $\left(\frac{7{,}153}{2{,}747}\right)$ = 2.604.

The following table summarizes the coefficient of performance (COP) for the different Compressor Configurations:

Compressor Configuration	COP
2-Section configuration without Economizer (Summer)	1.536
2-Section configuration with Economizer (Summer)	1.903
2-Section configuration with LP & HP Evaporators (Summer)	2.604

Table 4 – COP values for Different Compressor Configurations

Driving The Two-Section Refrigeration Compressor

There are basically two methods utilized in industry for driving the two-section refrigeration compressor:

Fixed speed drivers – usually an induction or synchronous electric motor. In some cases, a fixed-speed gas turbine driver is employed.

In these cases, a suction throttle valve may be installed just before the LP section suction drum (to prevent any condensate forming by the temperature drop due to any J-T effect across the valve from entering the compressor) and is used to modulate the overall performance of the two-section compressor.

In some cases the throttling valve is installed downstream of suction drum and may be used to reduce starting motor power demand.

Generally the throttling valve provides the means to maintain Evaporator and/or suction drum pressure within the desired range. For parallel units, each train's throttling valve may be used to adjust the load distribution and allow for starting and stopping units with the minimum process disturbance.

This valve is usually modulated to maintain the process Evaporator operating pressure at the design value, in this tutorial 24.0 psia, which will produce a saturation temperature of approximately -22.6 degF in the Evaporator.

This configuration is represented by the following Figure 18.

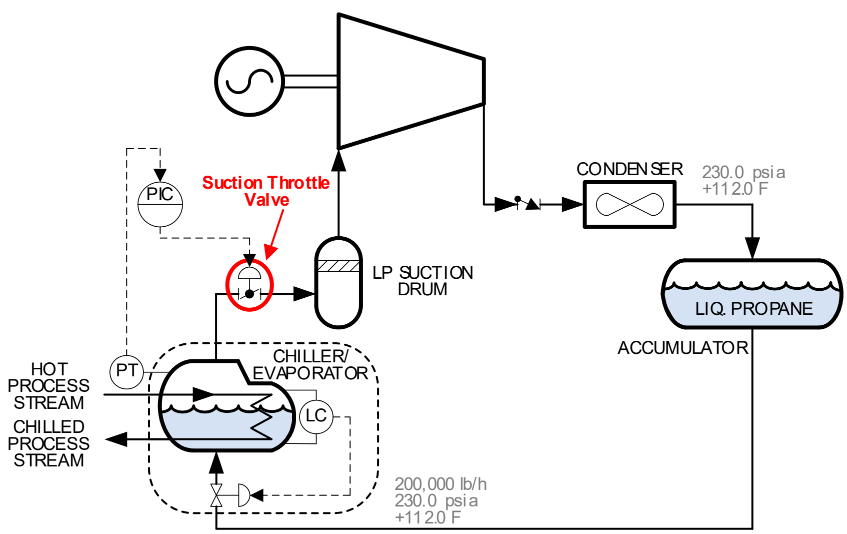

Figure 18 – Fixed Speed Drive with a Suction Throttle Valve Configuration for the Refrigeration Compressor

If the process heat load drops, the Evaporator will produce less vapor and if the suction throttle valve is not present or is present and does not modulate, then the Evaporator operating pressure will tend to drop as does the compressor suction pressure. But the compressor discharge pressure is "fixed" by the operating temperature of the Condenser (which is assumed to be unchanged), and the higher pressure ratio across the compressor will result in the compressor handling less volumetric flow. Ultimately, a new equilibrium point is reached where the reduced amount of vapor produced by the Evaporator is matched by the reduced flow rate of the compressor.

If the reduction in vapor produced by the Evaporator is excessive, there will be a risk of surging the compressor and the antisurge control mechanisms will force the compressor to operate with recycle.

To illustrate how the suction throttle valve modulates the overall performance of the compressor, consider the single performance curve in the coordinates of discharge pressure vs. suction volumetric flowrate for a fixed speed compressor as shown in the following Figure 19.

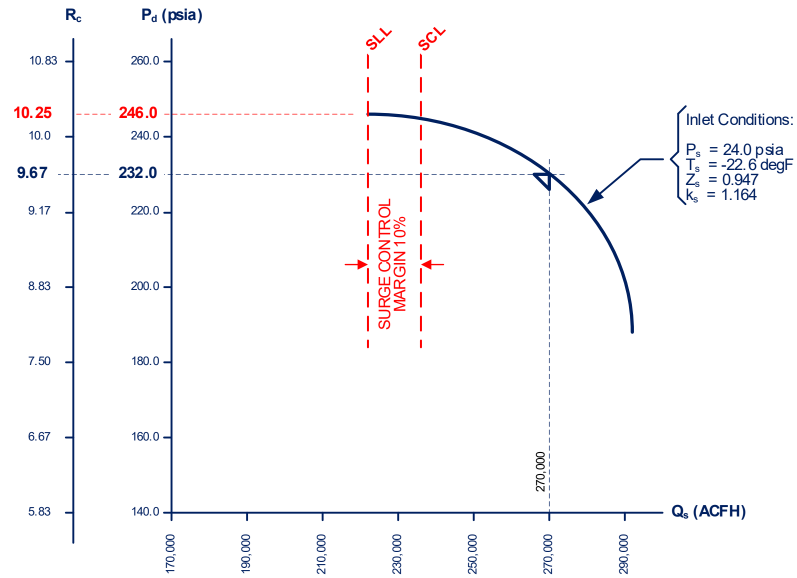

Figure 19 – Fixed Speed Drive Compressor Performance Curve at Design Conditions with the Suction Throttle Valve Wide Open

As may be seen, the design operating point lies at 270,000 ACFH and 232.0 psia, which represents a pressure ratio (R_c) value of

$\left(\frac{232.0}{24.0}\right) = 9.67$. In the example shown, the Surge Control Line (SCL) is located at approximately 235,000 ACFH and a discharge pressure of 246.0 psia, or an R_c value of $\left(\frac{246.0}{24.0}\right) = 10.25$.

When the suction throttle valve modulates in the closed direction, it creates a pressure drop across it which further lowers the suction pressure of the compressor. The discharge pressure of the compressor, however, tends to remain the same as it is dependent on the cooling medium temperature in the Condenser.

The performance curve and the associated R_c scale both then shift downwards, as shown in the following Figure 20, to reflect the new 1st section suction pressure of $\left(\frac{232.0}{10.25}\right) = 22.63$ psia. Note that the operating temperature of the Evaporator remains at its original value and the pressure drop across the suction throttle valve is only (24.0 – 22.63) = 1.37 psi.

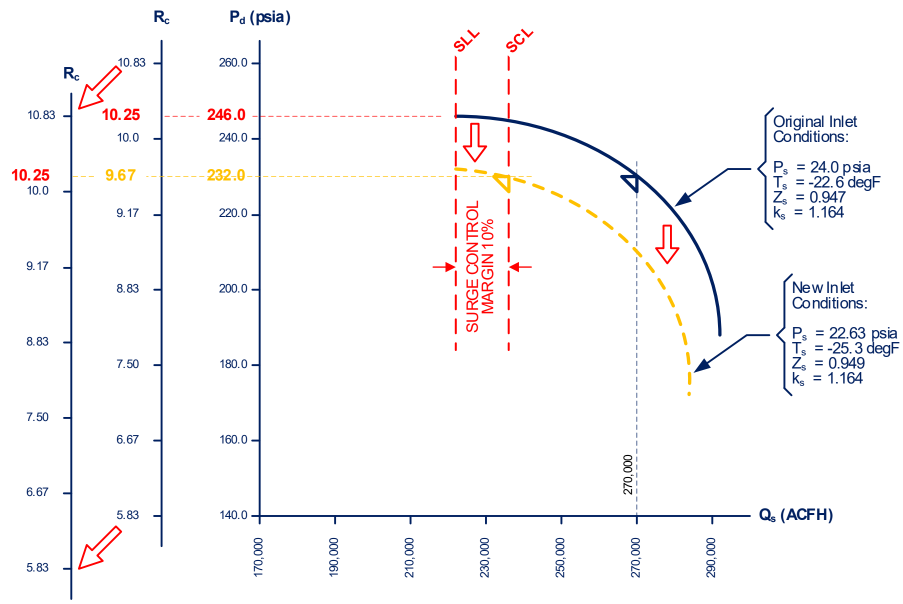

Figure 20 – Fixed Speed Drive Compressor Performance Curve Shifts Lower as the Suction Throttle Valve Modulates

The suction throttle valve should not modulate such that the pressure drop across it causes the 1st section suction operating pressure to become sub-atmospheric.

Also, the shift in the performance curve due to the pressure drop across the suction throttle valve should be limited as not to cause the R_c value of the original Surge Control Line (in the illustrated example an R_c value of 10.25) to drop further than compressor's surge point, as the compressor would not be capable of providing enough head to stay out of surge.

Since many two-section process refrigeration compressors are designed such that the LP section suction pressure is close to atmospheric pressure, and the rise to surge from the design operating point to the Surge Control Line is quite limited, this method proves to be of limited efficacy in terms of modulating the capacity of the machine.

Variable speed drivers – usually a steam or gas turbine, or a variable speed electric motor. A recent development uses a fixed speed electric motor with a variable speed and torque hydraulic/mechanical gear assembly between it and the compressor, providing variable speed operation of the compressor, while retaining the simplicity of the electric motor.

All of these methods leverage the Affinity Laws (aka the "Fan Laws") which state, that for a constant geometry centrifugal compressor:

- The change in flow is proportional to the change in speed,
- The change in head is proportional to the square of the change in speed, and
- The change in gas power demand is proportional to the cube of the change of speed.

For example, if a compressor decreases its speed by 10%, it is expected that a 10% decrease in flow, will produce approximately a 19% drop in head and approximately a 27% decrease in gas power demand. This makes speed variation much more efficient than a suction throttle valve for refrigeration compressor capacity modulation. In addition, the energy penalty associated with the pressure drop across a suction throttle valve is eliminated. Finally, refrigeration compressors may be designed with the 1st section suction pressure just slightly above atmospheric pressure, which will help to operate the Evaporator at a lower temperature of refrigerant liquid vaporization, hence obtaining a higher enthalpy differential and refrigeration duty. This configuration may be illustrated in Figure 21, below.

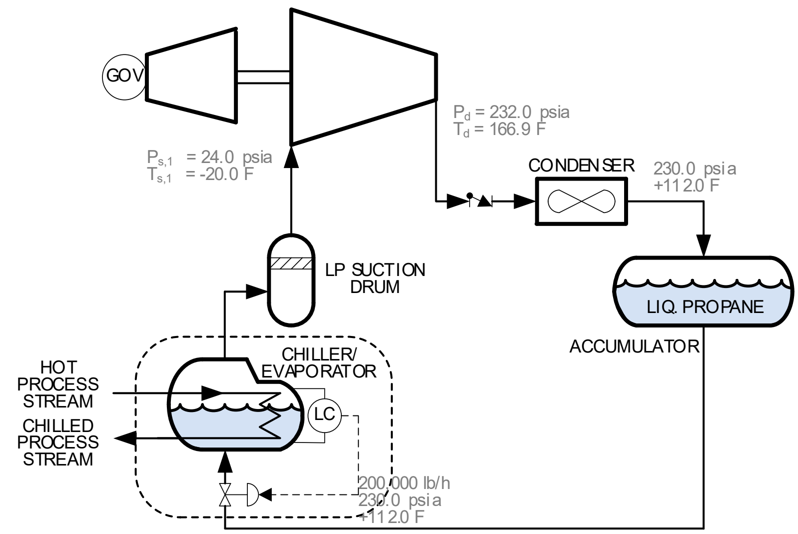

Figure 21 – Variable Speed Drive for the Two-Section Refrigeration Compressor

The performance map in the coordinates of discharge pressure vs. suction volumetric flowrate for a variable speed compressor is shown in Figure 22.

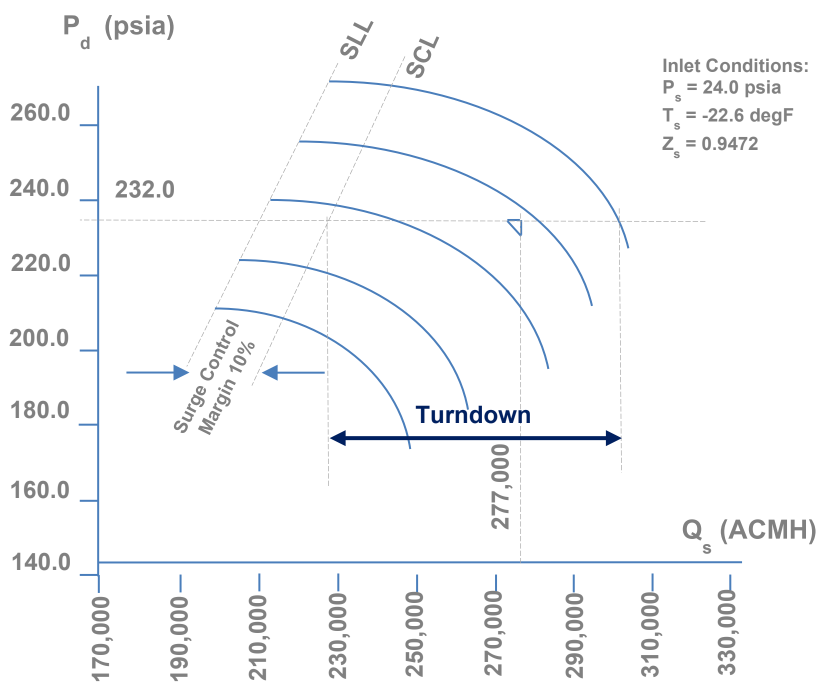

Figure 22 – Variable Speed Drive Compressor Performance Curve at Design Conditions

Performance Control of the Two-Section Refrigeration Compressor

Performance control of any centrifugal compressor ensures matching its throughput with the process demand. For the drivers discussed above, this means modulating the suction throttle valve for fixed speed configurations or varying the speed of the driver for variable speed configurations.

There are three theoretical process variable (PV) choices on which to base the Performance Control of a refrigeration compressor:

- Suction pressure
- Discharge pressure
- Mass flow (throughput) of the compressor.

Selecting the compressor's discharge pressure as the PV for the Performance control arrangement would be illustrated as per the following Figure 23.

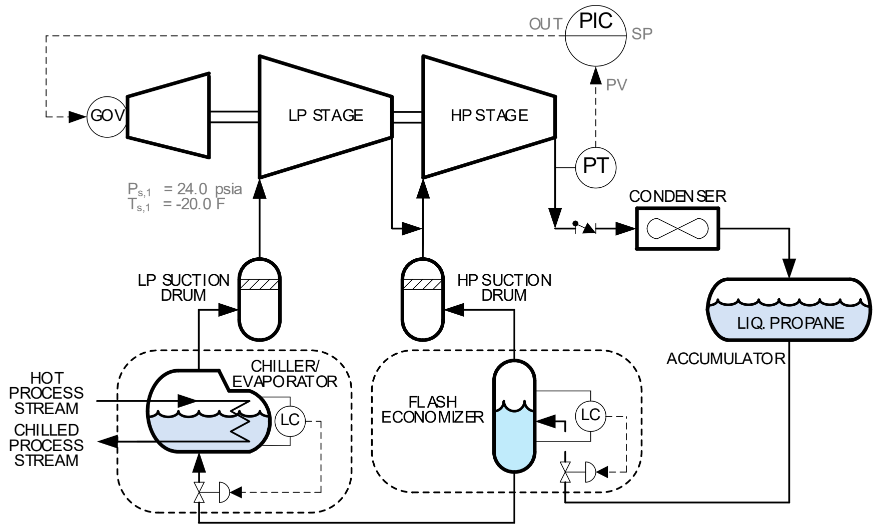

Figure 23 – Selecting Discharge Pressure for the Compressor Performance Control Scheme

As has been observed, the Chiller/Evaporator is usually designed to cater to a known process chilling duty and should operate practically all year round at a fixed and stable temperature and pressure of vaporization. On the discharge side, however, the pressure at which the compressed refrigerant will condense depends on the cooling medium's (ambient air or available river- or sea-water) temperature, which fluctuates significantly in a seasonal manner.

Here the designer has two choices, either have the discharge pressure controller's set-point always be slightly above the condensing temperature of the day, which usually entails the operator to manually adjust it continuously, or select a fixed pressure set-point that is slightly above the highest annual expected ambient temperature.

This would ensure that the discharge pressure is always high enough so that the refrigerant would condense on the hottest day of the year and eliminate the need for continuous operator adjustments. But this would also mean that for all the days when the ambient temperature is lower than this value, the pressure to speed cascade control loop would tend to drive the speed of the train to maximum, i.e. tend to saturate the output of the controller, which is not an acceptable method of control.

In conclusion, selecting the compressor discharge pressure as the performance control variable is not advisable.

Selecting the mass flow throughput as the performance control variable is relatively straightforward in terms of instrumentation, but the problem lies in setting the appropriate mass flow set-point value in the Performance controller. As we have seen in this tutorial, this mass flow set-point must be selected based on the cooling duty required at any moment by the Evaporator, and this depends on the process conditions, as well as the ambient condensing temperature and hence compressor discharge pressure.

Therefore, it is also not advisable to select mass flow as the Performance control variable.

Modulating the refrigeration compressor based on its suction pressure measurement is the preferred method to implement Performance control. With this approach, once an appropriate set-point value has been selected and assigned to the Performance controller, whenever the process load creates less refrigerant vapor in the Evaporator/Chiller, the measured suction pressure will tend to drop and the Performance controller will react by decreasing the compressor's throughput. Likewise, compressor throughout will be increased whenever more vapor is produced in the Evaporation/Chiller by the process load.

With suction pressure being selected as the Performance control variable, the compressor's discharge pressure will be left "floating" and will rise or drop according to the seasonal changes in the temperature of the Condenser's cooling medium. This is illustrated in the following Figure 24 and Figure 25:

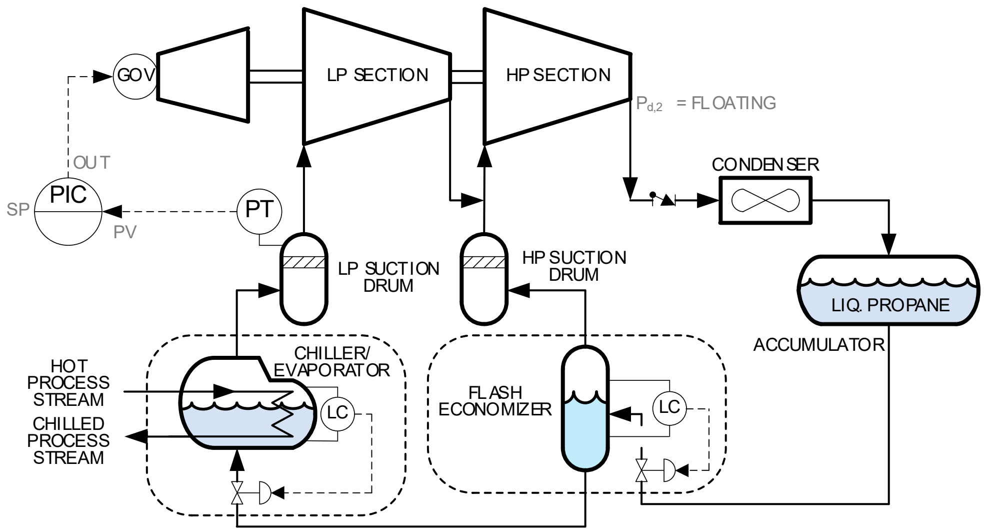

Figure 24 – Selecting Suction Pressure for the Performance Control Scheme (Variable Speed Driver)

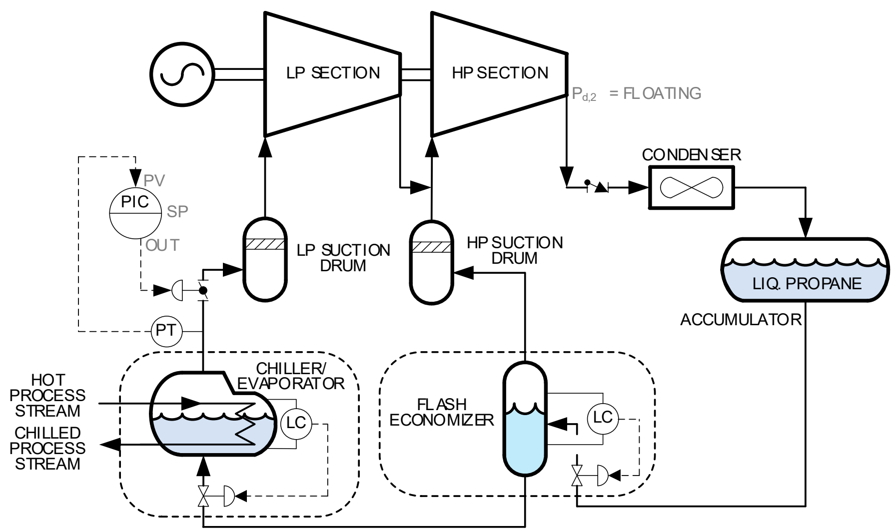

Figure 25 – Selecting Suction Pressure for the Performance Control Scheme (Fixed Speed Driver)

If one takes the previous example of a two-section propane Refrigeration compressor in Corpus Christi, TX, the average annual ambient temperature (according to meteorological records) is 84.0 degF. The compressor designed for this service would then have an average floating discharge pressure of approximately 168.0 psia, with a corresponding refrigerant condensing temperature of 90 degF.

Under these operating conditions, the compressor with a polytropic efficiency on 0.800, would require approximately 20.5 hp of gas power to compress every 1,000 lb/h of propane refrigerant from 24.0 psia to 168.0 psia; which is less by 3.2 hp per 1,000 lb/h of refrigerant for the same compressor configured for discharge pressure control.

Recycle Piping for the Two-Section Refrigeration Compressor

The optimum recycle piping design for the two-section Refrigeration compressor should include a dedicated recycle path and antisurge valve for each section, as per the following Figure 25.

The recycle line should commence from the 2nd section discharge and will therefore be recycling “hot” gas. It is not advisable to locate the recycle take-off downstream of the condenser, as the fluid there is mostly, if not all, liquid. Some plant designs split the Condenser functionality into two separate process coolers, a “desuperheater” to remove the superheat from the discharge gas, and the Condenser proper, to condense the compressed vapor. It is theoretically possible to commence the common recycle line just downstream of the Desuperheater, so the recycle gas is not as hot as it would be at the compressor immediate discharge, but this is somewhat risky in the sense that over-cooling in the Desuperheater would also cause liquid condensate to form and be sent to recycle, which should be prevented.

It is highly recommended to install a process check valve just downstream of the recycle line take-off to reduce the discharge volume that the antisurge valve(s) must evacuate during surge events to the minimum possible. This will ensure the effectiveness of the antisurge control system under all conditions, including the emergency trip of the compressor.

The fact that the recycle gas will be “hot” is quite problematic in the sense that allowing full recycle for even short periods of time, and partial recycle for longer time periods, will inevitably lead to the suction temperature of each section rising to the mechanical integrity trip point. This is especially true during compressor start-up where good operating practice dictates that the antisurge valves be in the fully open position initially.

Clearly, to sustain continuous total or partial recycle, a solution must be provided to cool the recycle gas to design suction temperatures. Some compressor manufacturers consider that cool process gas will be available at all times and by mixing with the hot recycle stream, the mixture temperature entering compressor section will be cooler than at the hot recycle take-off. While this might permit long term operation of the compressor with some amount of partial recycle, at higher-than-design suction temperatures, but lower than the mechanical integrity trip value, higher recycle rates will, at some point, cause the compressor to overheat.

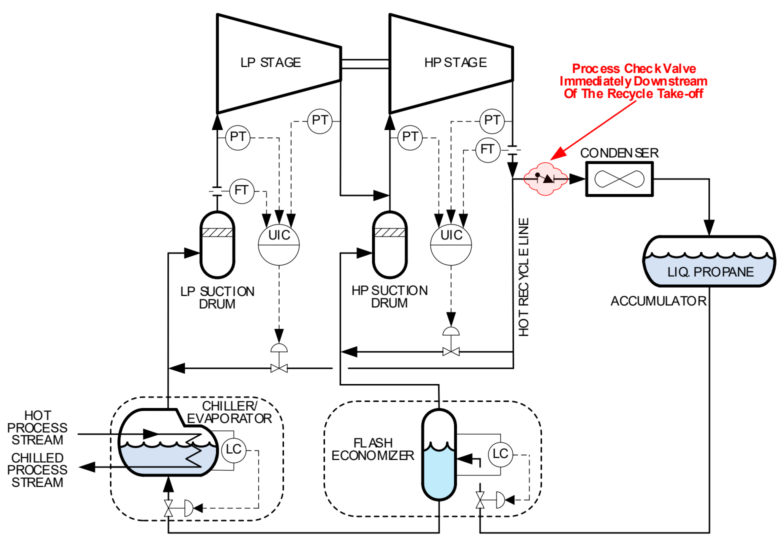

Figure 26 – Optimum Recycle piping for the Two-Section Refrigeration Compressor

Recycle Cooling by Means of Liquid Quench

Traditionally, recycle gas cooling for a refrigeration compressor was achieved by introducing an atomized spray of the refrigerant into the hot recycle gas stream after it passes through the antisurge valve. The vaporization of the fine droplets of liquid refrigerant is a highly effective method of cooling the hot recycle gas. This is illustrated in the following Figure 27.

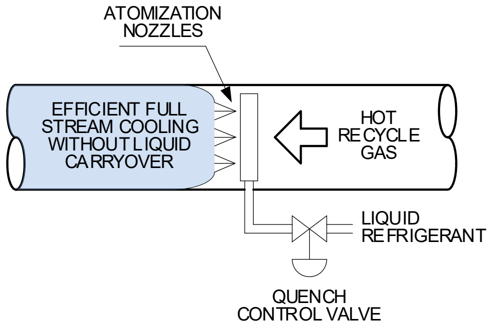

Figure 27 – A Properly Designed Quench Nozzle Ensures Full Atomization of Liquid Quench

To ensure the proper atomization of the quench liquid, specially designed spray nozzles should be used so that all of the liquid quench that is introduced is indeed vaporized and no (or very little) liquid quench refrigerant is allowed to remain.

Each recycle path's quench valve is modulated by a Quench controller, that keeps the mix temperature at the designated value that is usually selected equal or slightly above the design suction temperature of each section.

Traditionally, the Quench controller was nothing more than a temperature controller, using the recycle/quench mix temperature as its Process Variable (PV). It is important to force the liquid quench valve to be completely closed when the compressor is running but when the antisurge valve itself is closed, otherwise liquid quench would be introduced in the absence of hot recycle.

Because the recycle/mix temperature sensor must be of industrial robustness, it will typically be designed to be placed in a thermowell and will be considerably slower than the typically high-speed antisurge control loop. It can be expected that the quench/recycle mix temperature control will be sluggish when called into action.

It is recommended that the quench valve be dynamically de-coupled with the antisurge valve so that every opening or closing of the antisurge valve simultaneously produces a proportional and suitable movement of the liquid quench valve. The much slower temperature control modulating action can then "trim" the quench valve to the desired steady-state position.

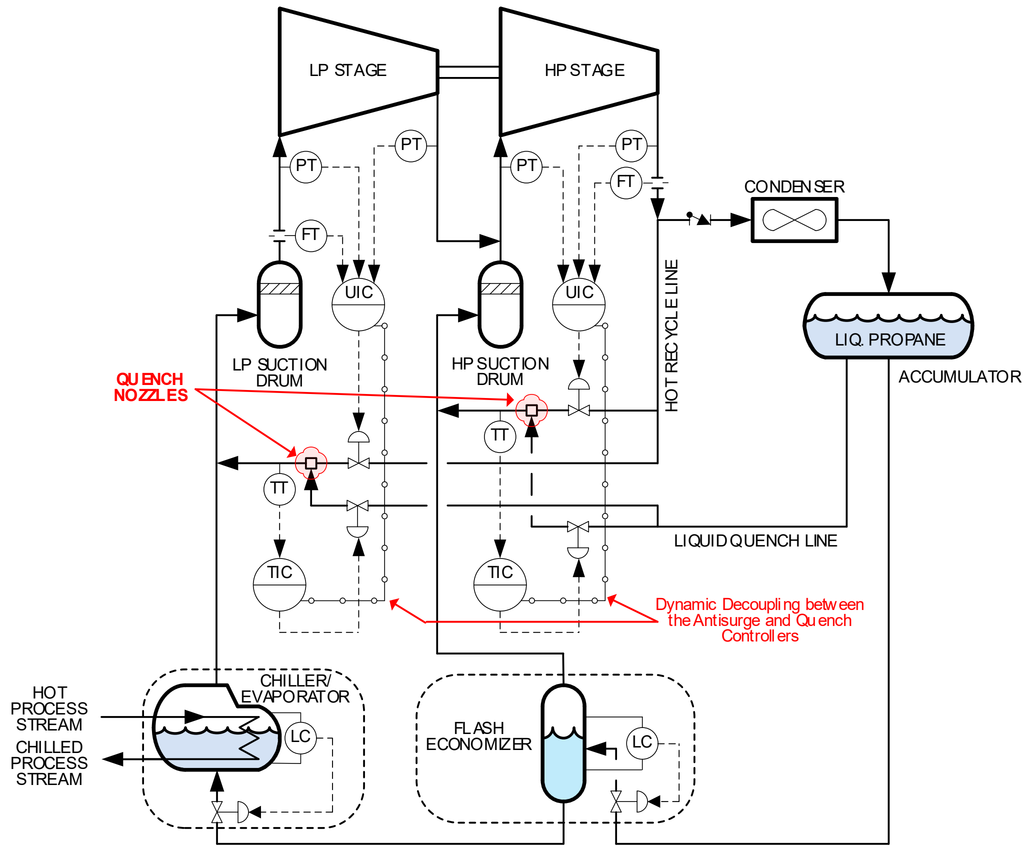

Figure 28 – Quench Control Arrangement Showing Dynamic Decoupling with the Antisurge Controllers

Another potential problem is the assumption that the designated recycle/quench mix temperature set-point is the same for all operating conditions. It is possible to change the suction pressure of the two-section refrigeration compressor for process requirements, which means that the optimum recycle/quench temperature set-point value must also change as a function of suction pressure for each section.

A "fixed" temperature set-point for the Quench controller would then be either too high, which increases the risk of overheating the compressor, or too low, which consumes liquid quench unnecessarily.

It is therefore recommended to assign the recycle/quench mix temperature set-point based on the actual suction pressure of each section and the corresponding saturation pressure of the refrigerant, with a suitable offset.

It is also recommended that the selected quench control valves be designed to fail closed upon an actuation failure, or control signal loss. This would prevent flooding the recycle lines in the event of a quench valve failure.

Recycle Cooling by Means of Sparging

A more modern and highly effective alternative method for cooling the hot recycle involves the sending the hot recycle gas to the low point of a suitable liquid inventory in the refrigeration circuit. The hot recycle gas for the LP section may be sent to the LP process Evaporator/Chiller, and for the HP section to the Economizer.

In each of these vessels, the hot recycle is allowed to permeate through the liquid refrigerant inventory in the shape of small bubbles by means of a suitably designed sparger, as per the following Figure 29.

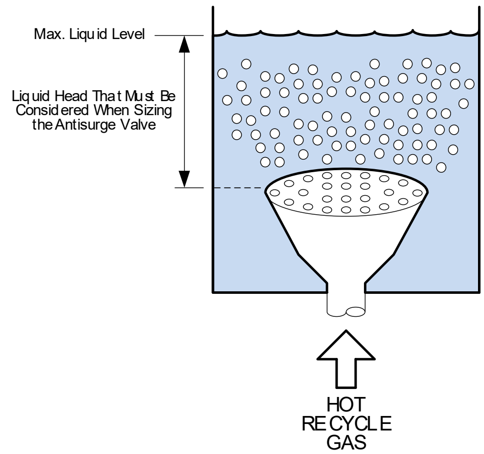

Figure 29 – Example of a Sparger Design

In this arrangement the hot recycle vapor bubbles are cooled by the liquid inventory and tend to release additional cool vapor from the surface of the liquid level that may be sent to the compressor suction.

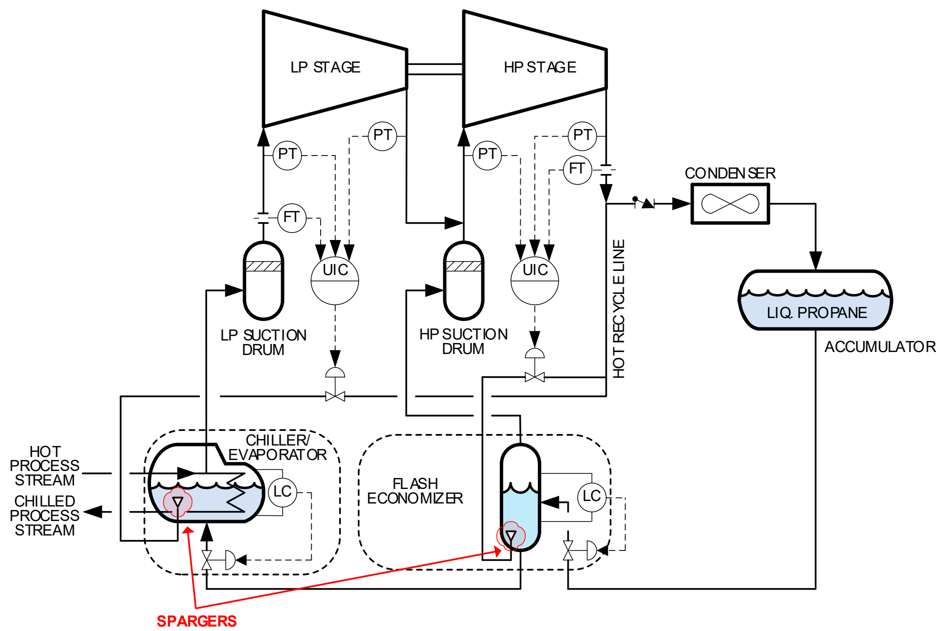

Figure 30 – Cooling the Hot Recycle with Spargers – Compressor Layout

The spargers must be carefully designed to handle the amount or recycle and avoid “jetting” the hot recycle so that it tends to pass straight through the liquid inventory without cooling. Furthermore, the liquid head between the sparger and the maximum level of the stored liquid inventory represents a significant back-pressure to the antisurge valve operation and must be taken into account when sizing the antisurge valve.

In a well-designed system, opening a compressor section’s antisurge valve produces an immediate increase of recycle flow that is cooled to a value that is close to the design operating temperature of the compressor suction.

The only drawback to this scheme is that a sufficient inventory of liquid refrigerant must be available to fill the Evaporator and Economizer vessels prior to compressor start-up, so that the recycle gas from the wide-open antisurge valves may be cooled as the machine is started.

Conclusions

Because of seasonal ambient temperature variations, that affect the operation of the Condenser, and hence the discharge pressure of the refrigeration compressor, the gas power demand on the compressor's driver in Summer could easily be 1.6 ~ 1.7 times the gas power demand for Winter conditions. It is recommended that this be considered when sizing the driver.

The basic industrial refrigeration cycle, in terms of controllability, exhibits a degree of self-regulation. Considering the refrigeration cycle at a certain ambient temperature, which "fixes" the compressor's discharge pressure, a higher process cooling load in the Evaporator will produce more vapor and the compressor's suction pressure will tend to rise, for the same speed of operation. This, in turn, means that the compressor's pressure ratio will drop, and the compressor's operating point will move to the right and down the single-speed performance curve, resulting in a higher compressor throughput – without any external manipulation.

A common two-section refrigeration compressor configuration utilizes a Flash Gas Economizer in the compressor's interstage. This significantly increases the available enthalpy differential for process cooling, and simultaneously lowers the total gas demand on the driver, for a given ambient temperature on the Condenser. Hence the Coefficient of Performance sees a significant improvement over the configuration without an Economizer. Converting the Economizer to a Process Pre-Cooler has the potential to further improve the Coefficient of Performance above what may be achieved by using an Economizer.

Generally, the recommended control strategy is to allow the economizer pressure float with the process demand, while controlling the suction pressure of the 1st section. There are, however, process designs where the refrigerant flow into high pressure sections must be maintained within certain limits, which is not possible to achieve by merely controlling compressor 1st section suction pressure. In these cases, a throttling valve between the Economizer and 2nd section may be installed, although it may decrease the COP.

Adding a throttle valve between the Economizer and the refrigeration compressor's 2nd section only raises the Economizer's operating pressure, and hence lowers the Coefficient of Performance. It is the author's recommendation to not install such a valve, unless the specific compressor configuration results in the 2nd section being sized for a significantly higher gas power demand than that of the 1st section, and a fixed speed electric motor driver is chosen. In this case, the addition of a throttle valve between the Economizer and the 2nd section may be warranted as it may be throttled during train startup, to lower the total gas power demand and mitigate the risk of tripping the motor driver on startup overload. In this scenario, the throttle valve is forced to fully open once the startup is successfully completed, so that the COP may be maximized.

A variable speed driver is the most efficient method of driving a centrifugal process refrigeration compressor. Since many of these applications are designed to operate with a 1st section suction pressure that is only slightly above atmospheric pressure, selecting a fixed speed driver and a 1st section suction throttle valve provides quite limited controllability and hence compressor turndown.

Because of the presence of a Condenser on the discharge end of the refrigeration compressor, whose operating pressure varies seasonally with the changes in ambient temperature, it becomes problematic to select compressor discharge pressure as the control variable for the Performance Control system. Suction pressure is recommended to be used as the Performance Control variable. If a variable speed driver is used and there is no suction throttle valve present, the location of the suction pressure transmitter may be on either the compressor suction drum, or the Process Evaporator. If a fixed-speed driver and suction throttle valve combination is used, it is recommended to install the suction pressure transmitter ***upstream*** of the suction throttle valve.

To allow for operating the compressor continuously with partial or full recycle, an efficient method must be incorporated to cool the recycle lines, to remove the heat acquired through the compression process.

Case Study: SMR Compressors in Small-Scale LNG Plants

Many small-scale LNG plants (capacities between 50,000 and 500,000 gallons per day or gpd) utilize the two-section single mixed-refrigerant (SMR) compressor technology. The majority of such compressors are driven by a variable-speed gas turbine driver and have two sections of compression.

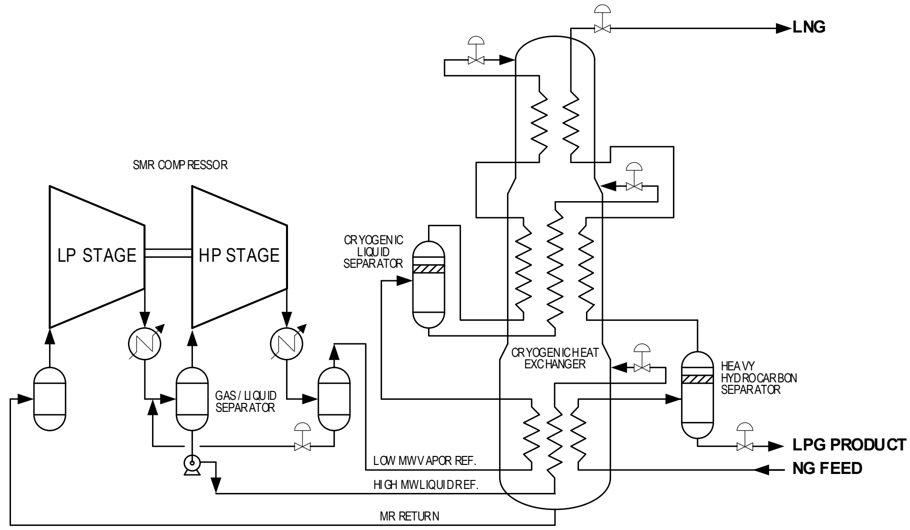

Figure 31 – SMR Compressor in a Typical Small-Scale LNG Plant – Process Schematic

Mixed refrigerant vapor (MW = 37.2) returns from the Cryogenic Heat Exchanger to the L.P. Suction Drum and is compressed in the L.P. Section of the compressor. The compressed mixed refrigerant vapor is cooled in the inter-stage cooler, which produces a mixed-phase stream, comprising the higher molecular weight liquid (approx. MW = 48.1) and a lower molecular weight vapor (approx. MW = 29.4).

This is routed to the H.P. Suction Drum where the high molecular weight liquid is collected, and the remaining lower molecular weight gas (MW = 29.4) is sent to the HP section for further compression. The outlet of the H.P. compression section is cooled in the discharge after-cooler, where, again, a mixed-phase stream is produced, comprising a higher molecular weight liquid (approx. MW = 33.2) and a lower molecular weight vapor (approx. MW = 27.7), which is sent to the H.P. Discharge Drum.

Here the higher molecular weight liquid is separated from the lower molecular weight vapor, and the liquid component is sent to the H.P. Suction Drum, where it is mixed with the liquid stream condensed after the L.P section and producing a liquid mixture of approx. MW = 43.6, and, in turn, sent to the Cryogenic Heat Exchanger as liquid refrigerant.

The lower molecular weight vapor separated in the H.P. Discharge Drum is also sent to the Cryogenic Exchanger, as a vapor refrigerant.

The typical SMR compressor control system consists of a dedicated antisurge controller, with associated antisurge valve and recycle piping, for each compressor section, to prevent that section from surging when the compressor is incapable of producing sufficient head to drive the compressed vapor forwards.

A Performance controller that determines what the compressor throughput needs to be, based on comparing a suitable process variable (almost always the L.P. section suction pressure) to a suitable set-point value. The required throughput modulation is constrained between identified minimum and maximum operating speeds and sent in a cascade control arrangement as a remote set-point to the driver’s governor control system.

A processor licensor for small-scale LNG plants proposed the following SMR compressor recycle configuration:

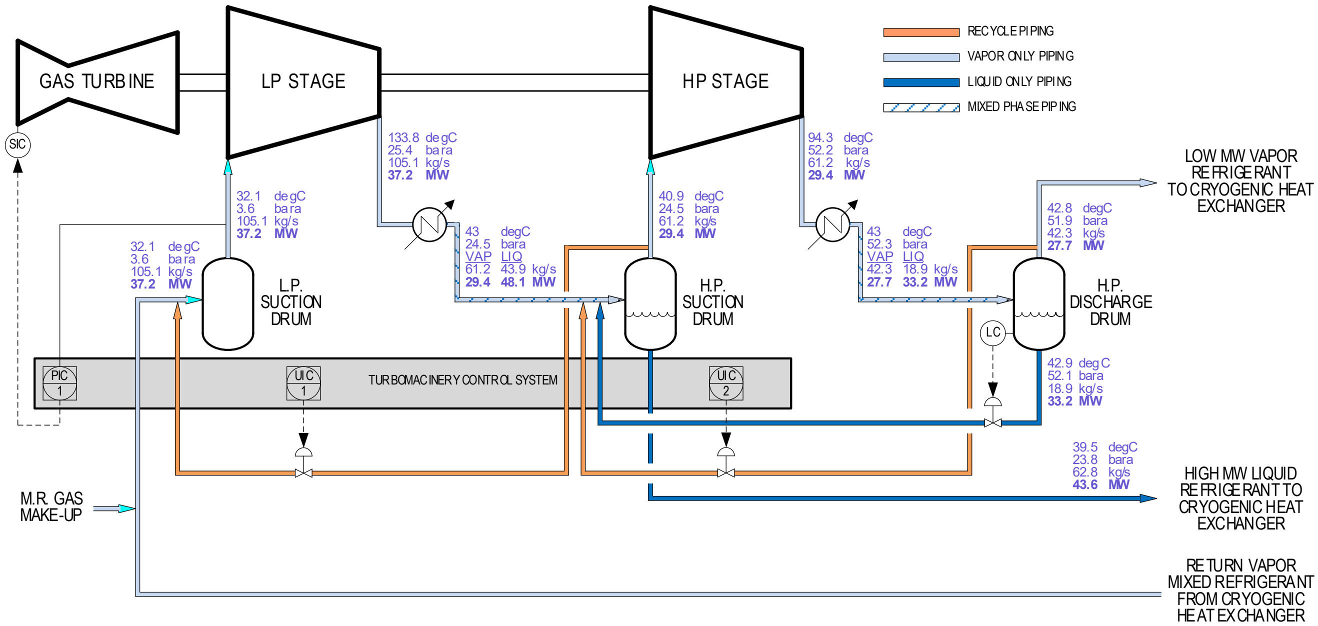

Figure 32 – Proposed Single Mixed-Refrigerant (SMR) Compressor Recycle Arrangement

As may be seen from the above Figure 32, the 1st Section antisurge valve was designed to recycle low MW vapor from the HP Suction Drum (MW = 29.4) back into the LP Suction Drum which normally feeds return vapor (MW = 37.2) to the 1st section of the SMR compressor. The introduction of significantly lower MW vapor to the 1st Section is quite problematic because it will initially drive its operating point, for a given speed of operation, deeper into the surge region. Whereas opening the antisurge valve is intended to have the opposite effect.

Unless the surge control margin for the 1st Section is set to an unacceptably large value, this sudden introduction of lower molecular weight gas will greatly increase the risk of several surge cycles until a new speed of operation is reached that is appropriate for the lighter vapor.

The situation is much less noticeable for the HP Section, where the vapor that is recycled has a molecular weight of approx. MW = 27.7 as compared to the design operating molecular weight of the HP Section of MW = 29.4. Here just a slightly wider-than-normal Surge Control Margin should prove sufficient to prevent the sudden decrease of H.P. Section vapor molecular weight and the associated risk of surging that compressor section.

Another potential problem is that operating the two sections of the SMR compressor under heavy or full recycle – with lower than design MW vapor – increases the risk that the compressor cannot generate sufficient polytropic head at design speeds to sustain forward flow.

Finally, the original compressor layout did not include for the installation of process check valves located just downstream of the recycle line take-offs.

To address these potential issues, the following modifications to the recycle line layouts were recommended.

For the LP section, it is recommended to take-off the LP recycle line immediately after the LP section discharge flange, and before the interstage cooler. This will ensure that the recycle vapor that is introduced into the LP section will have the same composition, and hence the same MW as the LP's design conditions.

Obviously, this recycle gas will be heated as it is compressed in the LP section and must be cooled to the design inlet temperature. It was therefore recommended to provide for a dedicated recycle cooler, as shown in Figure 32.

It is also recommended to install a non-return (check) valve immediately after the proposed recycle line take-off.

Concerning the HP section recycle, it was recommended that it begins also immediately after the HP section discharge flange and before the HP after-cooler, and to have it deliver the recycle vapor between the LP discharge check valve and the existing interstage cooler. As for the LP section, it was also recommended to install a check valve immediately after the

HP recycle line take-off.

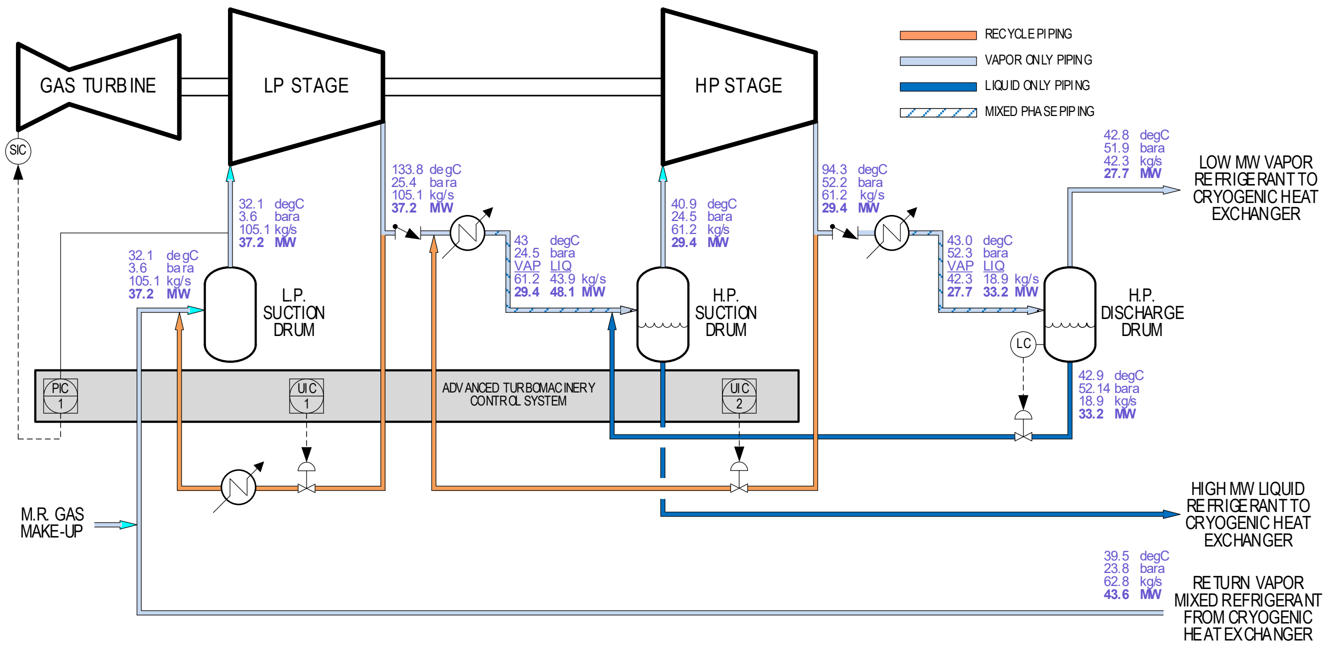

Figure 33 – Recommended & Modified Single Mixed-Refrigerant (SMR) Compressor Recycle Arrangement

NOMENCLATURE

$\dot{m}$	= mass flowrate (lbm/h)
h	= enthalpy (BTU/lbm)
W	= work equivalent (hp)
Ref_{duty}	= refrigeration duty (Tonnes of refrigeration – or TR)
H_p	= polytropic head (ft-lb/lbm)
R_o	= Universal gas constant (1545.349 lbf-ft/(lbmol-R))
Z	= gas compressibility factor
T	= absolute temperature (degR)
P	= absolute pressure (psi)
MW	= gas molecular weight
ρ	= gas density (lbm/ft^3)
R_c	= compressor pressure ratio
σ	= polytropic exponent
k	= real gas specific heat ratio, estimated using CCC proprietary software.
η_p	= polytropic efficiency
Q	= suction volumetric flow (ACFH)
COP	= coefficient of performance = $\frac{\text{Refrigeration Duty (hp)}}{\text{Gas Power Demand (hp)}}$
P_{gas}	= gas power (hp)

Subscripts:

s	= suction
d	= discharge
1	= 1st Section
2	= 2nd Section

REFERENCES

(1) Zaghloul, M. – “Resolving Single Mixed Refrigeration (SMR) Compressor Control Challenges” – LNG Industry Magazine, November 2016.

ACKNOWLEDGEMENTS

(1) The authors would like to acknowledge Serge Staroselsky, Chief Technology Officer with Compressor Controls Corporation, for providing the real gas estimated specific heat ratio factors, checking the power calculations and reviewing the Tutorial.