A REFRIGERATION SCREW COMPRESSOR PACKAGE – WHAT FEATURES DO I NEED FOR MY GAS PROCESSING FACILITY

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ABSTRACT

Oil flooded screw compressors are the most commonly used type of compression equipment in the refrigeration service at the midstream gas processing facilities. They are compressors of choice for gas dew point control plants and gas pre-cooling in liquid recovery facilities. They also serve in fractionation plants and export or import NGL/LPG terminals. Despite being quite standardized, the screw compressor packages can be provided with variety of design features, some of which may or may not be required by the facility owners and operators.

The discussion will start with an overview of the components of a screw compressor train and presentation of various features of a typical compressor such as capacity control, discharge port size adjustment, role of the side port and its capacity, types of compressor casing materials, designs of radial and thrust bearings and the role of the oil injection in compressor performance. The paper will also discuss various types of available compressor shaft seals with emphasis on their suitability for use in the refrigeration service. Additional discussion will focus on the differences between drivetrain design utilizing gas engines and electric motor drives.

The paper will also present typical features of the compressor lubrication system design including the types of the oil separators and their capabilities for oil removal. The oil management methods within the refrigeration loop and various methods for removing and returning to the compressor of the residual oil not captured by the oil separator will be presented. The discussion will include an overview of oil filters types and methods of lube oil cooling including air, thermosyphon and liquid injection.

The paper will also discuss special applications in which screw compressor may operate. This includes elevated or varying evaporating temperatures, refrigerated condensing at elevated ambient temperatures or utilizing a single screw compressor for multiple evaporators.

The paper describes the integral role of the screw compressor as a part if a complete gas processing facility. The paper concludes with a case study.

Introduction

Typical midstream gas processing facility receives feed gas which needs to be conditioned in various ways before it can be provided to a customer as a final product. The processing may include separating free liquids such as water and heavy hydrocarbons from the process gas, sweetening of the gas, gas dehydration, etc. After the initial conditioning, the feed may have to be further processed to remove heavier components such as condensate, LPG and in some cases ethane. This process will in majority of the cases require mechanical refrigeration system, quite often described as MRU or Mechanical Refrigeration Unit, either as a stand-alone installation or equipment working in conjunction with a gas expansion device such as turboexpander or Joule-Thompson valve. Majority of the refrigeration systems in the midstream industry utilize various grades of propane as a refrigerant.

A conventional mechanical refrigeration system requires compression equipment to remove flash gas and vaporized liquid, compress it and direct it to in most cases an air cooled condenser, in which compressed vapors dissipate heat and get converted into liquid, which then is returned back within the closed refrigeration loop.

There are multiple types and sizes of compressors that could provide compression in the refrigeration systems. These can be reciprocating compressors for very small systems or centrifugal compressors for very large installations. However, most of the midstream facilities utilize oil flooded screw compressors in the refrigeration plants, mainly because they fit well into typical midstream gas flows and associated with the refrigeration system duties.

Oil flooded screw compressor drivetrains

An oil flooded screw compressor drivetrain consists typically of the compressor with a directly coupled electric motor driver, compressor lubrication system complete with lube oil separator, oil pumps, filters and lube oil cooling equipment, compressor inlet and outlet piping, instrumentation and controls including Unit Control Panel all mounted on a common baseplate built from structural steel components. The electric motor driver is in most cases self-lubricated. Optional engine drives directly coupled or connected to the compressor through the gearbox have also been utilized. Occasionally, motor driven packages utilizing gearboxes to speed up the compressor above the motor driver rotational speed and increase compressor refrigeration capacity have also been supplied to the midstream industry.

Compressor refrigeration capacity

Compressor refrigeration capacity is given in terms of heat removal capacity at a given evaporating, condensing and flash interstage temperatures. It can be expressed in Tons of Refrigeration, British Thermal Units per Hour, kilowatts, etc. Once the refrigeration duty is stated the compressor selection programs calculate the required mass flow rate required for the cooling duty, take into consideration the amount of flash gas generated in the cycle and convert it into the compressor volumetric flow rate at suction and side port. Once the volumetric flow rates have been established and the compressor rotational speed selected the machine size is determined, volumetric and adiabatic efficiencies established and the estimated compressor brake horsepower calculated.

Oil flooded screw compressors – general overview

The oil flooded screw compressors are positive displacement type machines which typically utilize twin meshing rotors for gas compression. In general, they are the most common compressors used by the refrigeration industry. The rotary oil flooded screw compressors offer many advantages. They are proven in service. The off-the shelf design allows for quick compressor delivery. The sophisticated design of the rotors profiles combined with injection of the lubrication oil to the compression chamber, which provides cooling and seals the rotors results in the screw compressors having high volumetric efficiency and being able to operate at high compression ratios. The same oil is also used for lubrication of the compressor bearings and the shaft seal. A properly designed lube oil separator reduces the lube oil carryover below 10 to 15 parts per million on weight basis. If lower carryover is desired, additional secondary coalescing lube oil separator can be installed which will reduce the oil carryover to less than 1 part per million weight. Single seals are most common but dual wet/dry and wet/wet shaft seals are also available if positive containment is required in the event of the primary shaft seal failure. The compressors are typically fitted with a side port allowing effectively for a two stage compression. They are also equipped with a slide valve which allows for the infinite steps of compressors capacity control. Most of the compressors operate at rotational speeds equivalent to rotational speed of two or four pole induction motors.

Majority of the refrigeration systems supplied to the midstream industry operate on propane with a typical operating envelope having suction close to atmospheric pressure and discharge between 200 and 300 psig (14 to 21 bar gauge). The majority of the systems are designed for single stage-economized cycles. The MRUs can also be designed for 2-stage compression if two (2) compressors in Booster – High Stage arrangement are applied. A third stage can be added by economizing the High Stage compressor. The slide capacity control allows for compressor turn down to approximately 10-20 percent of its maximum capacity providing for operational flexibility and a significant reduction in drive train power draw at the reduced refrigeration load.

Current compressor designs for most practical cases are limited to maximum design pressures of 350-400 psig (24-28 bar gauge) although higher compressor design pressures are also available. The maximum suction gas flow of a single compressor is limited currently to approximately 6000-7000 actual ft3/min (10000-12000 m3/hr). The maximum driver power will be less than 8000 HP (6000 kW) with typical compressors utilizing drivers in a 1000 to 5000 HP (750 to 4000 kW) range.

Some compressor manufacturers can also offer screw compressors in compound configuration. This design features two sets of twin rotors internally coupled within a single compressor body.

Typically, the Booster or Low Stage rotors are one size larger in diameter than the set of the rotors of the High Stage machine. Various rotor lengths are available for each pair of rotors. The compound design concept can be compared to a multistage centrifugal compressor, in which as the gas get compressed the compressor wheel width decreases. The compound designs offer improved efficiency as compared to a single compressor especially for the applications requiring very high compression ratios. Their disadvantage is in increased cost, because the compressor is in reality a set of two machines, increased delivery and more complex drivetrain design. Also, a similar design can be accomplished by utilizing two individual screw compressors with booster or low stage machine discharging directly into the suction of the highs stage compressor.

Oil flooded screw compressors – compressor casings and rotors

The screw compressor casings ate typically constructed of three, vertically split parts, which are bolted together. The standard compressor case materials are grey cast iron. Ductile iron and cast steel materials are also available but usually require extended delivery. Ductile iron and cast steel casing are typically available as special orders unless the compressor manufacturer decided to standardize their design around ductile iron case materials. This would be typical for very large compressor frames. The advantage of iron materials is their relative low cost and ease of manufacturing. Steel castings are more expensive and more difficult to pour. The premium required for ductile versus cast iron case would be around 10-15 percent of the compressor cost. Steel casing can double the compressor cost for small and medium frames. The price premium for large steel frames would be between 20-50 percent of the compressor cost. Cast iron compressor casing are typically suitable for pressures up to 350-400 psig, Nodular and cast steel casing can be rated for pressures up to 600 psig or higher.

The majority of the screw compressors, whether dry or oil flooded utilize twin, male and female rotors. In case of the oil flooded designs majority of the designs feature rotors that operate without the need of the timing gears. The male rotor is connected the driver and the female rotor rotates due to very close clearances between male and female rotor and inter-lobe lubrication. The most common rotor profiles feature 4 x 6 and 5 x 7 designs, i.e. male rotors with four or five male lobes and six or seven lobes on the female rotor. The rotor length is typically expressed in term of the ratio of the rotor length to its diameter, with common L/D ratio ranging between 1.1 for very short and 2.4 for very long rotors. For a given rotor diameter and fixed operating speed, the compressor refrigeration capacity increases with an increase in the rotor length; however, very long rotors are not suitable high compression ratios due to increased rotor deflection. The long rotors may only be used for compressors in booster applications in two-stage refrigeration systems.

The compressor rotors are typically fabricated from steel bar in case of the small and medium size machines. Large rotors are typically made from steel forgings. Large forgings are more economical to fabricate compared to rotors machined from large steel billets. Some screw compressor manufacturers offer ductile iron rotors as standard.

The ANSI/API standard 619 specifies the requirements for "Rotary-type Positive Displacement Compressors for Petroleum, Petrochemical and Natural Gas Industries". The document is intended to cover "dry and oil-flooded, helical-lobe rotary compressors" that are in "special-purpose applications". Furthermore, the standard defines the special-purpose application as an "application for which the equipment is designed for uninterrupted continuous operation in critical service and for which there is usually no installed spare equipment".

API 619 requires compressor casing materials to be made from steel if compressor rated discharge pressure exceeds 400 psig, discharge temperature is greater than 500 deg F or gas handled by the compressor is flammable or toxic.

API 619 also stipulates that compressor shafts be made from forged steel. Considering that the shaft is an integral part of the rotor, following this API requirement results in forged steel rotors furnished with the compressor.

Unlike the equipment supplied to refineries and chemical facilities, the end users in the midstream industry typically do not require compliance to API 619 standard as a design requirement. Midstream facilities are commonly equipped with multiple compressor drivetrains and critically requirement is not applied to the facility design. Majority refrigeration compressors in midstream facilities have cast iron casings and manufacturer standard rotors.

<u>Oil flooded screw compressors – compressor bearings</u>

Oil flooded screw compressors are equipped with radial and thrust bearings. The radial bearings are also called journal bearings. The compressor consists of two (2) meshing rotors and each of the rotors requires minimum two (2) radial bearings to keep the rotor in place. The radial bearings can be sleeve hydrodynamic or antifriction, which in most cases will be cylindrical or tapered roller type. The type of radial bearings depends on the compressor manufacturer.

The sleeve bearings are usually steel backed babbitt type. Babbitt is sometimes referred to as white metal. There are various types of babbitt alloys on the market. The most common babbitt consists of majority of tin with balance of antimony, copper and lead. Some babbitt alloys are composed mainly of lead; however, these alloys are mainly applied to slow speed applications, with surface speeds below 1000 feet per minute. Tin based babbitt materials are suitable for speeds above 1000 and below 2400 feet per minute. Babbitt surfaces are common in sleeve bearings because the exhibit resistance to galling or wear of sliding surfaces caused aby adhesion. Babbitt alloys are soft and form metal matrix composites, which allows for lubricant to flow as the softer metal erodes. The babbitt sleeve bearings require pressurized supply of lube oil of correct viscosity that has been filtered and cooled. The lubricant oil creates hydrodynamic oil film which has to maintain correct bearing surface temperature. The required oil flow will assure proper oil distribution between the sleeve and the rotating surface.

The anti-friction radial bearings are constructed of inner and outer races with rolling elements in-between. The rolling elements roll between the races along their axis. The roller bearings also require supply of clean, filtered and pressurized oil but some of the compressor manufacturer do not require oil pressure to be above the compressor discharge pressure for bearing lubrication. This eliminates the need for the lube oil pumps as long as the compressor discharge pressure is sufficiently above the compressor suction pressure and lube oil circuit pressure losses are small. Unlike the sleeve type journals, which have theoretically infinite life, the anti-friction rolling element bearing life is expressed in terms of L10 life. The L10 life represents the probability of 10 percent of the bearings failure in hours of operation under given conditions.

The compressor thrust bearings typically provided with oil flooded screw compressors are in most cases angular contact ball and provided on both rotors. Additionally, the thrust load is handled by a balance piston, typically fitted on a male rotor. The angular contact thrust bearings are anti-friction type bearings with rotating balls located between inner and outer race of the bearing. Most of the designs utilize four (4) point contact bearings in single or double row configuration. Large compressors may require multiple set of the bearings on each rotor. The design of the bearing allows them to handle thrust forces in active and reverse direction. Optionally and exclusively on very large frames, the compressor manufacturers provide hydrodynamic tilting pad thrust bearings. The tilting pad bearings are built with bearing collar and multiple pads located radially in a carrier ring on both sides of the collar, which also allows the bearing to handle both active and revers loads. The tilting pads have tin-based babbitt surfaces that are steel backed, which is similar to the he construction of the hydrodynamic sleeve bearings. The tilting pad bearing advantage is in essentially no wear and capability to handle larger loads. They require pre-lube and post-lube cycle during compressor start-up and shutdown.

API 619 standard allows both types of bearings to be used in oil flooded screw compressors. It provides limits on the application of anti-friction bearings based on the compressor rotational speed and its shaft diameter, specifies a minimum L10 life of 50,000 hours in continuous operation and 32,000 hours at maximum loads and effectively limits their use for compressor break horsepower above 1,500 HP at rotational speed of 3600 rpm and 1,800 HP at 3000 rpm, with break horsepower values twice as high and half those speeds.

A typical screw compressor cross-sectional arrangement depicting casing, rotors and bearings is presented in Figure 1.

<u>Oil flooded screw compressors – compressor shaft seals</u>

A standard shaft seal supplied with the screw compressor is a single, mechanical type. It consists of a stationary and a rotating component which are lubricated with the lube oil supplied to the compressor bearings and injected into the compression chamber. The seal is designed for lubricating oil to leak through the seal into an external container. The estimated oil leakage rates vary with the compressor shaft sizes, operating speeds, operating pressures and typically range between 0.5 up to 5 drops per minute. Usually, the seal designs are based on API 682 standard and specifically designed for a particular compressor. These seals are the most common type used on screw compressors in the refrigeration industry.

Other shaft seal arrangements are available. In particular, double wet/dry and double wet/wet configurations can be offered, with latter one designed for tandem or back to back arrangement.

The double wet/dry seal consists of a primary wet running seal lubricated by the compressor lube oil and a secondary, dry running seal, providing a containment. If primary wet seal fails, the secondary seal will prevent the process fluid from spilling into the environment. John Crane 28SC type seal or other manufacturer equal is utilized as a secondary type seal for this arrangement. At a minimum, the dual wet/dry running seal arrangement requires API plan 75 plan to collect the oil from the primary seal into the liquid accumulator. The vessel collecting the oil will be equipped with a level switch, alerting the operator that the vessel needs to be drained. The vent line connecting the vessel and the seal housing to the flare will be provided with a flow restriction orifice and a high pressure switch, which provides an indication of primary seal failure in the event of high pressure. The increase in the pressure will indicate that the process gas is flowing from within the compression chamber, through the primary seal into the flare. This seal arrangement can also be equipped with API plan 72 providing buffer gas for circulation and sweep of the cavity between the primary and secondary seals.

The wet/wet tandem seal arrangement consists of a primary seal lubricated by the compressor lubrication oil and secondary seal lubricated by the external console designed to API plan 52 or API plan 55. The purpose of this arrangement is to provide similar protection to the wet/dry dual seal arrangement described above.

The wet/wet seal in back to back arrangement requires lubrication oil to be supplied to primary and secondary seal via API plan 53. The oil pressure has to be maintained above compressor lube oil pressure to allow the seal oil supplied to the primary seal to leak into the compression chamber. The flow path on the primary seal is reversed in this case and the oil supplied into the seal drains into the compressor rather than away from the machine. This seal provides the highest degree of protection but it is rarely used in the refrigeration systems.

Similarly, rarely used but available from some of the compressor manufacturers is a single dry gas seal that uses a buffer gas supplied to the seal at the pressure above the compressor discharge pressure via an API plan 74. The buffer gas flows into the seal and vents into the compression chamber.

API standard 619 requires for process gas that is toxic or flammable a separation seal in addition to a primary seal. It allows the secondary seal to act as a temporary back-up in case of the primary seal failure. The wet/dry gas seals with API plans 75/72 are common for compressors operating on hydrocarbon gases on projects that require API 619 compliance. The compressors supplied to midstream and industrial refrigeration industry utilize for most part single mechanical shaft seals.

Figure 2 depicts a drawing of a typical dual wet/dry mechanical shaft seal. The API 72/75 plans are presented in Figure 3.

<u>Oil flooded screw compressors – compressor capacity control</u>

The oil flooded screw compressors installed at midstream facilities are typically large machines operating at fixed speed. The most efficient capacity control of the machine can be achieved by varying the operating speed of the compressor. It allows for essentially linear reduction in compressor power with reduction of compressor speed and its capacity. However, medium voltage Variable Frequency Drives designed for motor full power are expensive and majority of the facilities do not utilize speed for compressor capacity control.

In most cases the refrigeration capacity of a screw compressor is controlled via a device called a slide valve. The slide valve is built into the compressor case and operated by a hydraulic cylinder attached to the compressor. The slide valve effectively "shortens" the length of the compressor rotors as it moves towards the discharge end of the compressor. The "shortening" of the rotors reduces the amount of the gas that the compressor is capable of compressing, with some of the gas getting recirculated back to the inlet. Typical screw compressor capacity can be controlled by the slide valve from its full capacity to about 20 percent of the total design throughput. The slide valve position is determined by the compressor suction pressure. As the refrigeration load decreases, the compressor suction pressure will also decrease. The control built into the Unit Control Panel will, in response, move the slide valve and "shorten" the length of the rotor available for compression, which will reduce the compressor capacity. This will allow the compressor suction pressure to return back to the set-point. Alternatively, if the refrigeration load increases, the compressor suction pressure will raise, the slide valve will move back and the compressor's capacity to remove vapors increase, which in turn will decrease the compressor suction pressure back to set point.

For the typical compression ratios between 2.5 and 4.5, i.e. the ratio between compressor discharge and suction pressure expressed in absolute terms, and the screw machine compressing propane vapors at 20 percent of its capacity will be require approximately 40-45 percent of its full load power. At 50 percent turndown, the screw compressor power will be 60-65 percent of the full load and at 80 percent turndown, the compressor will draw about 85 percent of its full load power. In general, the lower the compression ratio at which the screw compressor operates, the greater the reduction of the full load power at a turndown capacity.

<u>Oil flooded screw compressors – slide stop Volume Index (Vi) adjustment</u>

The screw compressors are provided with two (2) separate discharge ports through which the compressed gas can exit the compressor. The axial port is always open to the discharge gas. The radial port size and location can be adjusted be the extension of the slide valve called a slide stop.

The Volume Index, called Vi represents the ratio of volume of gas swept by the compressor at suction (Vs), i.e. volume of gas trapped in the threads of the rotors and volume of the compressed gas just before it exits the compressor at discharge (Vd), i.e. the volume of gas trapped in the threads of the rotors at that point. Considering the compression process within the compression chamber follows the equation

$$PsVs^{Ks} = PdVd^{Kd}$$

where Ps, Ks, Pd, Kd represent pressures and gas specific heat ratios at suction and discharge conditions, respectively. Assuming that the specific heat ratios are approximately the same at both suction and discharge conditions, i.e. Ks=Kd=K and after rearranging the equation becomes:

$$V_i = \frac{Vs}{Vd} = \left(\frac{Pd}{Ps}\right)^{1/k}$$

Depending on suction and discharge pressures and temperatures and type of the refrigerant the Volume Index can range between 2 and 8, for low and high compression ratios, or difference between evaporating and condensing temperature, respectively.

In practical terms, the Vi determines the required size and the location of the compressor discharge port, which can be adjusted by varying the size of the radial port using the slide stop. For low compression ratios, i.e. low Vi applications the size of the port has to be large. For high compression ratios, i.e. high Vi conditions the discharge port has to be small. Unlike the reciprocating compressor, the screw compressor is not provided with the discharge check valves which would open once the pressure inside the compression chamber exceeded the pressure in the discharge piping. If the screw compressor discharge port is too small for the application, the compressor will tend to over-compress and discharge port will act as a restriction orifice. If the discharge port is too big, the gas from the discharge piping will flow back into the compression chamber. Both situations are undesirable from the operation stand point and can result in compressor excessive vibrations, thrust bearing and shaft seal failures and excessive power consumption.

It is important to note that as the refrigeration load decreases and compressor slide valve unloads the compressor, the Vs component in the Vi equation decreases and the Vi becomes smaller. Therefore, the main impact of the compressor Vi is for the compressor operating at its full capacity. Considering that for various other reasons the compressor discharge pressure is kept within a narrow range, the adjustment of Vi would be recommended if the compressor is expected to operate at full capacity at greatly different evaporating temperatures. These temperatures determine the compressor suction pressure. This would be the case for refrigeration plants consisting of multiple compressors providing a pre-cooling upstream of the cryogenic plant. Typically, the evaporating temperature in ethane rejection mode is significantly higher than the evaporating temperature in the ethane recovery mode. At the same time, the condensing temperature, thus compressor discharge pressure, will be the same for both modes of operation. If the compressor is expected to operate at its full capacity during both mode of operations, the Vi adjustment between higher and lower Vi would be desirable. Most of the compressor manufactures provide means for VI adjustment within certain range. Typical Vi range for refrigeration applications starts with Vi of 2-2.2 for low compression ratio applications and 5.0 Vi for high differential pressures with some manufacturers offering fixed Vi as high as 5.8.

Figure 4 depicts the vertical section of the screw compressor, the slide valve, slide stop and the hydraulic cylinder.

<u>Oil flooded screw compressors – side port and its role in the refrigeration cycle</u>

Most of the screw compressors are provided with a side inlet connection, which in the refrigeration systems is called as an economizer port. The port is used for improvement of the refrigeration system efficiency by providing an additional suction capacity of the compressor. The port location, determined by the compressor manufacturer sets the port pressure, which in turn determines the economizer pressure. Typically, the compressor port would be set at the point, in which the suction gas volume has been reduced by 30 to 40 percent. Using adiabatic compression equation depicted describing the concept of Vi, the port pressure can be determined based on the type of gas and compressor suction pressure. The port pressure is then dependent on the compressor suction pressure and independent of the compressor discharge pressure.

The refrigeration cycles can be single, in which condensed refrigerant liquid is supplied directly to the evaporator or economized. The economized cycle can be based on utilizing a flash vessel or shell and tube or shell and tube (coil) type design. In systems with flash economizers, the condensed liquid refrigerant is supplied to a vessel that is maintained at the reduced pressure by the compressor economizer port. The liquid from the vessels, which is now cold and at reduced pressure is then directed to the system evaporator. The shell and tube (coil) economizer utilize a heat exchanger which cools the high pressure, warm liquid coming from the condenser using cold liquid at reduced pressure maintained by the compressor economizer port. In this case high pressure, cold liquid is supplied to the system evaporator.

The economized refrigeration systems are more efficient than simple cycles, because a portion of the flash gas, which inherently is a part of the refrigeration cycle is removed and compressed at higher pressure rather than at system evaporator pressure. This adds very little to the system efficiency for the low compression ratios applications but can result in overall power reduction of up to 20-30 percent for systems operating at low evaporating and high condensing temperatures, which is the case for majority of the midstream applications. The shell and tube (coil) applications are slightly less efficient, because liquid temperature leaving the exchanger has to be higher than the flashed liquid temperature, which cools it. However, they are ideal for applications having remote or elevated evaporators, because liquid supply to those exchangers is available at essentially condensing pressure. The net effect of the economizing the refrigeration cycle is reduction of the compressor size for the fixed system load or additional refrigeration capacity available from a given compressor frame.

Figure 5 depicts the refrigeration cycles described above on pressure/enthalpy diagram.

The compressor side port can also be used as a suction for a secondary evaporator within the refrigeration cycle. This can be achieved in addition to economizer vessel or heat exchanger. The secondary chiller's evaporating temperature has to be high enough for the economizer port pressure to be sufficient for proper exchanger performance. The refrigeration load has to be small enough for the economizer port pressure to handle the flow of the boiled off vapors. The larger the secondary evaporator's load the more vapor has to be removed by the compressor's side port to handle this load. This in effect will reduce the amount of vapor that can be removed from the flash economizer, which will increase the economizer vessel's pressure and temperature. This in turn will require the compressor to remove more flash gas from the main plant's evaporator. Therefore, adding a secondary evaporator to the refrigeration cycle handled by a single screw compressor will reduce the compressor main suction capacity in the economized refrigeration cycle.

Overview of Oil flooded screw compressor drivetrains

The majority of the screw compressor packages utilize induction motors to drive the screw compressors. The compressors are design to be able to handle rotational speeds developed by two-pole induction motors and in general are more efficient at higher speeds because compressor volumetric and adiabatic efficiencies decrease at much reduced speeds. The compressors and motors are coupled together using typically a flexible disk coupling speed coupling. The motor drivers are self-lubricated and utilize sleeve or anti-friction style radial bearings. In very hot climates the self-lubricated motors may not be suitable and forced lube motor bearing design may be required, which would necessitate the need of a separate lube oil consoles.

For midstream facilities the motors manufactured to vendor standard having WP II enclosures rather than API 541 or API 546 are typically supplied. They provide sufficient reliability, relatively fast delivery and competitive pricing. Medium voltage power is utilized for motor power above 300-400 HP.

Important consideration in selection of the motor driver is its starting torque capability. The screw compressors are fairly forgiving considering that the screw compressor will not be allowed to start unless the slide valve is located at the location allowing the compressor to develop only the minimum refrigeration capacity. On shutdown, the compressor pressure will equalize at the pressures typically close to an average of the system suction and discharge pressure unless compressor suction or discharge check valve do not hold tight. Thus, the compressor will be started fully unloaded at no differential pressure across it and will not develop the system discharge pressure before the motors reaches its full speed. However, if the compressor is very large and the expected motor starting time long a starting by-pass from discharge to suction may be required.

Figure 6 depicts typical electric motor driven screw compressor drivetrain.

Occasionally, a midstream facility may not have sufficient electric power available on site to drive the refrigeration compressors. The refrigeration compression represents typically the largest user of electric power at the facility. The lack of power could be the case at any of the Early Production

Facilities or plants installed in the remote locations. In those situations, gas engines, typically encountered with gas gathering reciprocating compressors have been successfully provided as drivers of the refrigeration screw machines.

The gas engines operate at slow speeds. Very small engines are available at operating speeds up to 1800 rpm but as the power requirements increase the engine speed decreases. In most cases the engines required to drive the refrigeration machinery will operate at 1200-1400 rpm for power in 1000-2000 HP range or 1000 rpm for greater engine power. Considering that the screw compressor design, power ratings and the highest efficiency are based on rotational speeds equivalent to two pole or four pole motor drivers, the engine driven refrigeration screw compressor drivetrains are typically provided with gearboxes installed between the engine and the compressor. The smaller horsepower applications utilize gearboxes bolted directly to the drive end of the engine or built into the screw compressor. The large horsepower designs require stand-alone gearboxes, attached to the baseplate assembly and connected to the compressor and the engine via high and low speed couplings, respectively.

The drivetrains utilizing gearboxes bolted to the engine or being part of the compressor are relatively straight forward. The stand-alone gearboxes, however introduce much higher degree of complexity. The base plates have to be machined to allow for proper alignment between the three pieces of rotating equipment, the torsional vibration are much more complex and typically the slow speed coupling required between the driver and the gearbox will have to be torsionally soft. The gearbox requires a separate lubrication system and depending on level of specifications may require variety of additional instrumentation for general performance monitoring as well as machinery protection, Gas engines will obviously require typical auxiliary equipment such as engine jacket and turbocharger cooling water cooler, inlet air filter and manifold as well as exhaust piping, noise attenuation, emission reduction catalyst and personnel protection insulation.

Figure 7 shows engine driven compressor drivetrain with gearbox built into the compressor. In Figure 8, a large drivetrain with a stand-alone gearbox and remotely mounted utility cooler is depicted.

The stationary process equipment will consist of pressure vessels, heat exchangers, piping and valves. The vessels and heat exchangers will be designed to ASME Boiler and Pressure Vessel Code, Section VIII and provided with a U-stamp. The refrigerants are clean and non-corrosive and corrosion allowance will be limited to 0.0625” (1.6 mm). The refrigerant piping will be designed in accordance with AME B31.3 Process Piping Code. The ASME B31.5 Refrigerant Piping Code could also be applied, but considering that the rest of the gas plant piping will be designed to B31.3 and the in case of hydrocarbon refrigerants both codes requirements become very similar, the B31.5 code is rarely used. Based on the code requirements, the process piping will be provided at a minimum with five (5) percent non-destructive examination above full visual.

The valves provided could be API style, gate and globe or specialty refrigerant provided by refrigeration suppliers. There are some advantages of using refrigeration valves. In particular, they come with socket or butt-welded bodies that minimize system leaks and most of them are rated for low temperatures, down to minus 55 or minus 60 deg F (minus 48 to minus 51 deg C). They are equipped with soft seats, which can provide a tight shut-off but also can be damaged during the assembly, testing, etc. The API valves offer the familiarity advantage; the rest of the gas plant will utilize them within the process piping. They also offer robust design features including metal seats. Standard valves are rated for minus 20 deg F (minus 28.9 deg C) minimum process temperatures. Optional "low temperature valves" are available for minimum process temperatures of minus 50 deg F (minus 45.6 deg C) and can be rated for temperatures even lower based on ASME B31.3 Process Piping Code rules.

Oil flooded screw compressor drivetrains – structural steel skid support system

Small screw compressor drivetrains are commonly mounted on top of the horizontal oil separators. The suction piping consisting of the isolation and check valves as well as strainer are supported off the compressor suction flange. Short discharge piping connects the compressor outlet to the oil separator inlet. Outlet of the oil separator is provided with a discharge isolation and check valves. The lube oil piping including pumps and oil cooler if necessary, oil filters, etc. as well as Unit Control Panel, are supported from the oil separator. This arrangement makes for a very compact design, which does not require any additional structural steel support and can be mounted directly onto a concrete foundation. The disadvantage is in rotating equipment mounted fairly high.

Larger compressors drawing substantial amount of power require mounting on the structural steel baseplates to enable a proper alignment of the rotating equipment and placement of all the auxiliary equipment, piping and piping supports. Unlike commonly encountered with large reciprocating machinery, the unbalanced forces and moments generated by the screw compressors are limited. However, high rotational speeds and large rotating masses in addition to gas pulsations at frequencies four or five times the rotational frequency, depending on rotor configuration, may result not only in excessively noisy drivetrains but also in vibration levels that could be detrimental to the equipment. A robust, flat, well designed and having full depth structural steel members underneath the rotating machinery baseplate, which is then properly anchored and grouted at site will significantly reduce any field alignment or vibration and noise issues.

The baseplates can also be equipped with deckplate and drip rim, which will provide a positive containment in case of the lubricant spills. Obviously, this option is not available for compressor mounted on horizontal oil separators unless the entire horizontal drivetrain is then mounted on a structural steel base frame.

The expected Sound Pressure Level of large refrigeration screw compressor trains will exceed 85 dBA and in majority of the cases exhibit noise levels above 90 dBA. The noise levels could be somewhat reduced by installing a sound blanket around the compressor and insulating compressor discharge piping and the oil separator. In some cases, mounting of the entire compressor drivetrain inside a sound reducing enclosure may be the only solution to effectively reduce the noise from the equipment. This solution commonly encountered in installations within the residential areas is not common for packages supplied to midstream facilities.

Oil flooded screw compressor drivetrain lubrication system and lube oil cooling method

The screw compressor requires lube oil supply to the bearings, balance piston and compressor shaft seal. The bearing oil and majority of the lube oil supplied to the shaft seal drain back into the compression chamber. In most cases additional amount of lube oil is also injected into the compression chamber. There are two reasons for oil injection; to cool the gas during the compression process and seal the rotor tips and the compressor casing and limit the blowback of the compressed gas back to suction and improve volumetric efficiency. The injection oil also helps male rotor to drive the female rotor without the need of timing gears, reduces the compressor noise and gas pulsations. The oil injection allows to control compressor discharge temperature by varying the temperature and the amount or of oil injected to the compressor. The compressor discharge temperature has to be maintained above the compressed gas dew point temperature at discharge pressure. In case of refrigeration systems this means that the gas discharge temperature has to be above the system condensing temperature by a certain margin of safety. On the other hand, the discharge temperature has to be below the compressor mechanical limits. The oil flooded screw compressors have a fair amount of flexibility when it comes to the amount of oil injected to the compressor chamber. In some cases, they can operate without oil injection and only use the oil supplied to the bearings and drained to the casing to seal the rotors. They can also accept very large amounts of injection oil, if necessary.

The screw compressor lubricant selection requires careful consideration. The type of lube oil is typically determined by the type of compressed gas. The lubricant has to have properties that will limit the dilution of oil by the refrigerant at discharge conditions. Otherwise, lubricant/refrigerant mix supplied to the machine would flash upon entering the compressor, which would not desirable. For that reason, mineral oils are not used in hydrocarbon refrigeration system. The recommended lube oil for screw compressors in propane service would be synthetic PAG. In propylene service, the PAO oil is recommended. The oil viscosity is selected based on the lubricant injection and discharge gas temperature, mainly from bearing and shaft seal operating viscosity requirements.

Typical lubrication system of the screw compressor requires lube oil separator capable of separating the oil discharged by the compressor from the compressed gas, lube oil pump unless the compressor can simply utilize differential pressure between discharge and suction for oil supply pressure, lube oil filtration to remove any particulates and lube oil cooling to reduce the oil temperature from compressor discharge temperature to the design injection temperature.

Oil separators can be designed as vertical or horizontal vessels. Compressor drivetrains requiring large power, typically in excess of around 750 horsepower, will be provided with vertical separators. Smaller drivetrains will feature horizontal vessels, with compressor and the driver mounted on top of the separator. All large and engine driven packages will be provided with vertical oil separators. A hybrid design, mainly used to minimize the height of the package feature horizontal oil separator and the compressor and the driver mounted next to it directly on the structural steel base.

The industry standard oil separator features include bulk separation by impingement and change of direction of the gas which is further enhanced by fine oil removal utilizing coalescing elements. The expected oil carryover from a properly designed oil separator will be below 10 to 15 parts per million on weight basis. A lower oil carryover is achievable if a secondary coalescing lube oil separator is installed downstream of the primary oil separator. The secondary oil coalescer will reduce the expected oil carryover to 1 part per million or less. The separator has to be designed to allow for oil residence time within the vessel to assure proper liquid seal as well as enough time for degassing of the entrained gas from the lubrication oil. Typical residence time varies between 30 seconds to 2 minutes. The design of the piping between the compressor discharge and the inlet of the oil separator has to provide for the oil from the compressor to drain into the vessel upon the shutdown of the system. This can be achieved by either elevating the compressor above the oil level in the separator or installing an equalizing line at the centerline of the compressor drivetrain and the separator. This equalizing line will allow the compressor to drain the oil from the upper half of the machine, which is sufficient for the purpose of restarting of the machine. Otherwise the compressor would stay clogged with oil and experience excessive torque and vibration upon start-up.

The oil separator will be equipped with a single or multiple level gauges and immersion heaters. Oil level transmitter or switch is required for systems utilizing differential pressure for oil supply to the compressor, i.e. those without the lube oil pump. A proper drainage of the oil collected by the coalescing elements back to compressor suction is also required.

If lube oil pumps are provided they will be positive displacement type, electric motor driven. The pumps are typically gear or screw type and operate at speeds as high as 3600 rpm. Some pumps may require a speed decreasing gear between the driver and the pump. In most cases the pumps, whether direct driven or coupled via a gear, operating at reduced speeds will exhibit longer life compared to pumps operating at higher speeds. Some pumps will be equipped with an internal relief regulator, allowing the lube oil to by-pass from the discharge to suction of the pump at elevated differential pressures. The pumps are typically provided with a mechanical shaft seal which require careful selection to assure that the seal can handle high oil separator/pump suction pressure. The suction to the pump should be provided with a strainer. The discharge will feature a check valve and a relief regulator allowing to the excess pump oil to return back to the oil separator. The oil pump should be selected for excess flow of 30 to 50 percent over the required bearing and injection oil flow. The oil pumps can be provided as single or dual, with one pump installed as spare or both of them controlled in a lead-lag arrangement.

The oil before entering the compressor will have to be filtered and cooled. The oil supplied to the compressor bearings will have to pass through fine filters, with efficiencies of at least 95 percent for particle sizes of 5 micron or greater. The elements have to have sufficient surface area to prevent excessive pressure drop at the design flow rates and dirt hold capacity to reduce the necessity of frequent changes of the elements. The filters can be supplied in single or dual arrangement depending on customer specifications. For installations having multiple compressors a single filter can be considered sufficient. If only a single compressor is installed, then dual filter assembly would be recommended.

The oil injected directly into the compression chamber can be filtered together with the bearing oil or routed via a strainer. The latter arrangement will reduce the overall size of the lube oil filters provided on the compressor skid and will not adversely impact the performance of the compressor, especially in closed loop, clean refrigeration installations.

The lubricant oil can be cooled externally. The most common type of lube oil cooling methods in the midstream industry include utilization of the air cooled heat exchangers. The slip stream or the entire oil flow is circulated through the exchanger which the dissipates the excess heat to ambient. The amount of the oil circulated through the cooling coil is controlled typically by a thermostatic mixing valve, which maintains the desired oil temperature downstream of the valve. In cold climates the oil is routed through a shell and tube or plate and frame (shell) exchanger. The cooling medium is in this case a propylene or ethylene glycol mix, which in turn is circulated through the air cooled exchanger. This arrangement is mainly done to assure that the oil is not exposed to very cold ambient temperatures, which can approach its pour point causing the exchanger to exhibit excessive pressure drop due to very high lube oil viscosity at low temperatures.

Another method of cooling is via a thermosyphon heat exchanger, which utilizes warm liquid refrigerant to cool the oil. Condensed refrigerant is routed to a shell and tube or plate and frame (shell) type oil cooler. A portion of the liquid supplied to the exchanger gets vaporized as it removes heat from the oil and returns to refrigerant condenser, where it is converted back to liquid. This is a very efficient way to cool the oil, especially that typically the oil cooling load increases as the refrigerant compressor capacity decreases allowing ample condenser capacity to handle the oil cooling load. The drawback of this method is in a requirement for a relative close proximity of the refrigerant condenser to the compressor drivetrains and additional refrigerant piping between the compressors and the condenser. The thermosyphon cooling systems also require a small receiver elevated above the coolers providing for liquid refrigerant seal.

The internal oil cooling method is available via injection of liquid refrigerant to the compression chamber to maintain the compressor discharge temperature below the required lube oil supply temperature. This arrangement eliminates the need for the lube oil cooler. The lube oil is supplied via pumps and filters into the bearings and the compression chamber. A separate liquid refrigerant line is installed, which feeds refrigerants into the compressor via a temperature controlled valve. There are separate ports provided on the compressor casing for the purpose of liquid injection.

Typically, the compressor will be equipped with multiple ports located along the rotor length, which allow for optimization of the actual liquid injection location.

The popularity of the liquid injection oil cooling method was revived in recent years with improvements of the control devices and system controls. The amount of liquid refrigerant injected to the compressor chamber has to be precisely controlled to prevent excessive amount liquid from entering the machine. The goal is to inject just enough of liquid, which is going to remove some of the heat of compression and get vaporized during the process. On the other hand, if not enough liquid is provided the compressor, the discharge and lube oil temperature will rise and exceed the maximum allowed by compressor mechanical or lube oil viscosity requirements. Also, the careful selection of the lubricant and evaluation of the oil refrigerant dilution levels need to considered in addition to marinating relatively stable compressor discharge pressure. The sudden changes in condenser pressure will cause the compressor discharge pressure to fall, which could lead to refrigerant liquid coming out of the solution with the lube oil. This can lead to lube oil foaming in the oil separator and excessive oil carry-over.

The liquid injection oil cooling method simplifies the lube oil circuit and eliminates the need for the external lube oil cooler and the lube oil cooler air cooler fan motor. It requires additional refrigerant liquid line with a control valve. Even though the liquid is not injected at the compressor suction but later during the compression process, liquid inject oil cooling method reduces the compressor refrigeration capacity, because some of the compressor displacement is used to compressor refrigerant vapors used to cool the lubricant. The net refrigeration efficiency also decreases because a portion of the compressor power is used to compressor the vapors that are not used to refrigerate the process but utilized the keep the compressed gas and oil mixture at the desired temperature levels. On the other hand, this method may reduce the total lube oil circulation rate.

The lube oil piping will typically follow the same piping and vessel/heat exchanger design codes and principles as the refrigerant piping.

A typical P&ID of the refrigeration compressor employing liquid injection oil cooling method is depicted in Figures 9a and 9b.

<u>Oil flooded screw compressor drivetrain oil carryover and oil recovery within the refrigeration loop</u>

The majority of the refrigeration systems operate in a closed loop, with the refrigerant circulating between the condensed liquid supplied to the evaporator and refrigerant vapor delivered back to the condenser via a refrigerant compressor. The oil carried over from a conventional oil separator will not typically exceed more than 10 parts per million or 1 part per million if a secondary lube oil separator is included. The oil that was not captured by the oil separator will form a mixture with the condensed refrigerant and travel with liquid refrigerant through various control valves to the system evaporator. The oil supplied for systems utilizing synthetic refrigerants such as various type of freon mixes will tend to form layers located close to the top of the liquid bath, due to the fact that the lubricant is lighter than the refrigerant. In the chiller utilizing propane the oil will collect at the bottom of the vessel or exchanger. Oil can then be drained to a collection vessel located underneath the chiller and pumped or pushed back to the compressors. This will minimize possible oil losses during the operation of the system.

For the evaporators operating as thermosyphon chillers, which may be the case for some shell and tube and majority of the plate and frame (shell) or brazed aluminum exchangers, liquid refrigerant will be supplied to the exchanger tank above. For those designs the liquid refrigerant should be supplied from a nozzle located above the bottom of the vessel, while the bottom of the vessel should be connected to the oil distiller for collection of the lubricant and return of the oil back to the compressors.

Other components of the screw compressor drivetrain

Outside the compressor and the electric motor or engine driver, the lubrication system and basic suction and discharge piping, some packages may also include suction scrubbers installed upstream of the compressor. In general, suction scrubbers are not required for the refrigeration systems provided with properly designed single or multiple evaporators located in a relatively closed proximity to the machinery and connected with piping that does not consist of any traps that could collect liquid.

Otherwise, a properly designed suction scrubber may be required. The biggest challenge in properly designing the scrubber is in determining the liquid holding capacity and liquid removal t method. Under no circumstances, the scrubber should act as a slug catcher – otherwise the design methods used to design gas plant slug catchers should be applied, which would make the vessel prohibitively large. Suction scrubber can handle occasional and limited refrigerant liquid carryover from the evaporators. Once liquid collects in the vessel it needs to be returned to the system. A common method is to vaporize the collected liquid using external source of heat, which can be provided by an electric heater, a coil containing flowing warm refrigerant or sparging liquid with compressor discharge gas supplied underneath the liquid level. A very large system consisting of a horizontal scrubber may be provided with a shell and tube exchanger connected directly underneath the vessel having warm refrigerant circulated through the exchanger. Otherwise, the collected liquid needs to be pumped to the refrigerant accumulator or pushed using the discharge gas to a vessel operating at intermediate pressure, considering that the compressor discharge pressure is just slightly above the pressure of the refrigerant accumulator and there is not enough of differential pressure available for liquid transfer back to the accumulator.

Oil flooded screw compressor drivetrain controls and overpressure protection

The screw compressor drivetrains require Unit Control Panels for proper operation. These panels can be very simple and limited to start/stop/load/unload pushbuttons interfacing with a remotely located Plant Control System or complete control PLC/PAC panels equipped with a microprocessor, input/output cards, etc. In former arrangement or local control devices located on the skid will be wired to a junction box or Unit Control Panel, from which they will be further connected to the Plant Control System. The latter arrangement would connect all devices mounted on the skid to the local controller and interface to the Plant Control System would be limited to a remote start/stop signals and perhaps mapping of the local devices via a communication link.

The local instrumentation will include at least four temperature transmitters and/or RTDs monitoring suction, discharge, oil supply and separator oil temperatures as well as four pressure transmitters for monitoring of suction, discharge, lube oil supply, and lube oil upstream of the oil filters pressure. The control system could also include basic vibration monitoring such as compressor casing velocity or acceleration measurement in addition to male and female rotor axial position monitoring. As mentioned before, compressor operating without full time running pump will also require oil level measurement in the oil separator.

Occasionally, separate instrumentation for pressure and temperature shutdown may be required if plant's control philosophy requires a separate Safety Instrumented Shutdown system in addition to Plant Control System. In this case, duplicate and separate instruments will be installed on the skid and wired to a single or multiple junction boxes.

Availability of other devices, especially related to machinery monitoring is impractical and for most of the time not offered. Radial bearing temperature probes or rotating shaft radial X/Y position measuring devices are difficult to install due to a very compact design of the compressor as well as the fact that all bearings and shafts, unlike with centrifugal or reciprocating machinery, operate pressurized. The only exception are axial proximity probes for large machines.

Electrical wiring and all devices should be suitable for the electrical area classification, which typically in the midstream industry is Class 1, Division 2, Group C &D. The instrumentation from reputable suppliers will be available with the third party certificates confirming instruments suitability for the area. The wiring will be run in rigid conduit or utilizing armored cable installed in a tray. The Unit Control Panel will be weatherproof NEMA 4 or NEMA 4X (IP 55 or 56) and may require a purge to reduce the area classification inside the panel if the electrical devices mounted inside the box are not rated for the area classification. The junction boxes do not require a purge, because they will not include any sparking devices inside. The installations requiring Class 1, Division 1 rating will most likely use a combination of intrinsically safe and explosion proof devices and wiring. Similar rules will apply for installations in IEC/ATEX or IEC/IEC-EX areas, typically encountered at overseas locations.

A well designed drivetrain will be provided with a compressor discharge pressure relief valve designed in accordance with API 520 and API 521 for a compressor full flow at the relieving pressure. A typical control system will prevent the compressor from loading once the compressor discharge pressure alarm is reached, however a full flow relieving devices provides relatively inexpensive assurance that the compressor and system are properly protected. The valve will typically be mounted on the oil separator to make sure it is exposed only to gas rather than two-phase flow. Alternatively, if overloading the plant's flare system in the event of valve discharge is a concern, a smaller relief valve could be employed and high reliability Safety Instrumented System employed to assure compressor shutdown at high pressure event.

Summary and case study

A well specified and designed refrigeration system and the refrigeration compressors are essential for a well operating gas processing facility. The refrigeration system is required at most of the gas dewpoint control plants, in which gas is chilled to allow heavy components to condense and drop out in the Low Temperature Separator. The lean gas is then returned to the pipeline via a gas-to-gas exchanger in which the refrigerated gas exchanges heat with the incoming rich gas feed. This not only allows the gas to warm to pipeline operating temperature but also reduces the refrigeration system required duty.

Similarly, unless the gas very lean, refrigerant precooling will be required upstream of the turboexpander in high propane or ethane recovery gas plants. Most of the ethane recovery plants can operate in ethane recovery or ethane rejection modes. Typically, the refrigeration system's evaporating temperature will be much lower in the recovery than rejection case. Despite the fact that the actual refrigeration duty in ethane recovery case will be less than in ethane rejection mode, the difference in the evaporating temperature makes the recovery case controlling for the size and the design of the compressor and the rest of the refrigeration system. In fact, the system requirements in the recovery case will determine the size of the refrigeration compressor, which then can be rated for operation at the rejection case at reduced load, despite removing larger amount of heat from the system. Also, considering that the compressor discharge pressure will remain fairly constant whether in rejection or recovery cases, the compression ratio across the compressor will vary depending on the mode of operation. This needs to be taken into consideration when selecting and adjusting the compressor variable volume ratio during the plant life,

The refrigeration systems have also been applied to chill the process gas and/or solvent in amine and/or mixed solvent hydrogen sulfide and carbon dioxide removal plants This application is more common outside North America, where design ambient temperatures may approach temperatures above 122 deg F (50 deg C). Finally, the refrigeration systems are applied in NGL and LPG export or import facilities and storage.[i]

A refrigeration plant associated with 200-300 MMSCFD ethane recovery plant typically require multiple screw compressor drivetrains, each needing drivers in 2000 HP to 3500 HP range. The plants are designed with multiple compressors, each providing between 33 and 50 percent of the total refrigeration capacity. For the case study, the 3500 HP midstream style compressor drivetrains were compared to the machines specified to more stringent API requirements, typically encountered in downstream industry. Table 1 summarizes the difference between the scope of supply between both options and it can be applied to all other equipment sizes.

The basic compressor package would consist of a cast iron compressor equipped with manufacture standard rotors, casing connections and include a single mechanical shaft seal. The driver would be a WP II motor with self-lubricated bearings. The lubrication system would include a single lube oil pump, dual filters and a single air cooled lube oil cooler. The drivetrain components would be mounted on a compact structural steel baseplate. The process piping would include package outlet isolation and check valves. The inlet of the compressor would be provided with compressor inlet strainer and quick closing automated valve which would act as a check and isolation valve. All vessels and heat exchangers would be built to ASME code, provided with 1/16" corrosion allowance and spot radiography. The piping would be designed to ASME B31.3 code, also with 1/16" corrosion allowance and 5 percent radiography. Valves would be API style globe and gate type. The instrumentation and control system would include SMART pressure and temperature transmitters and a basic PLC based Unit control panel with a touch screen display.

The equivalent downstream package would include cast steel compressor with additional documentation and testing. The rotors would be forged steel. The compressor shaft seal would include double mechanical shaft seal in a wet/dry arrangement complete with API 72/75 purge and drain system. The driver would be built to API 541 and have TEAAC enclosure. The air cooled lube oil cooler would be designed to API 661 standard. Dual lube oil pumps built to API 676 standard would be provided. Equipment would be mounted on a large base plate. Vessels and piping would have similar design to the basic package but would feature 1/8" corrosion allowance and full or 100 percent radiography. Ball valves would be used for isolation and in lube oil piping. The instrumentation and controls would include separate devices for instrumentation and shutdown, extensive machine monitoring equipment and a higher end PLC Unit Control Panel.

Both units would provide identical refrigeration duty and power draw. However, the downstream package would be over 40 percent more expensive and have about 3-month longer delivery.

Similar comparison was done for a basic unit configuration versus a design utilizing liquid injection oil cooling method. The unit capacity decreased by 3 percent and power increased by 4 percent when the compressor was supplied without an external lube oil cooler. The overall unit cost was about 10 percent less.

In general, liquid injection oil cooling method decreases the compressor refrigeration capacity and increases the compressor power. The review of performance of various sizes of the machines shows a decrease in capacity between 2 and 6 percent with corresponding brake horse power increase between 3 and 5 percent for the same operating conditions if liquid injection oil cooling is utilized. This may not appear high, however the decrease in the refrigeration capacity may result in the drivetrain requiring a larger compressor to meet the required refrigeration duty. This will cause the drivetrain cost to increase and could result in the refrigeration plant having oversized machinery.

The selection and the level of applied specifications to the refrigeration machinery for a midstream facility require careful considerations.

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[1] The reader interested in refrigeration applications for NGL/LPG export terminals should reference the papers presented by this author at 2014 and 2106 GPA conventions under the titles "Refrigeration and boil off compression system considerations for NGL products export facilities" and "Compression power requirements and cooling equipment size estimates for LPG export facilities", respectively.

Table 1 – Comparison between API and non-API style refrigeration screw compressor packages for Midstream Industry[1]

Design features	API 619 style unit	Midstream standard unit
Compressor design		
Casing material	Cast Steel for flammable gases	Cast Iron for small compressors Nod Iron for large compressors
Rotors	Forged steel	Ductile iron, Steel bar or forged steel based on suppliers' manufacturing process
Mechanical seal	Double mechanical type with seal support systems	Single mechanical
Lube oil connections for compressor casing	NPT or optional flanged with stub pipe	NPT
Condition monitoring	X-Y casing vibration transmitters, male/female rotor axial position proximity probes, optional radial bearing RTD's	Single casing vibration transmitters
Compressor testing	<u>Witnessed or Unwitnessed</u> API 619 4 hr performance test Casing hydraulic test Rotor balance test Air under water leak test <u>Additional optional tests</u> Helium under water leak test Casing RT Casing MPI UT of casting and rotor stock Noise test Vibration spectrum	<u>Unwitnessed</u> Vendor standard mechanical run test Casing hydraulic test Rotor balance test Air under water leak pressure test
Studies	Torsional and lateral study Gas correlation or performance test	None (except for special drive train configuration)
Package design		
Baseplate	Multi-runner heavy duty structural skid with deckplate and drip rim	Multi-runner structural steel skid with optional deckplate and drip rim.
Process Piping	B31.3 piping with minimum 5% RT, 1/8" corrosion allowance, API style isolation valves	B31.3 piping 5% RT, 1/16" corrosion allowance, Refrigeration or API style isolation valves.
Oil separator design	2-minute lube oil retention time minimum	Packager standard lube oil retention time, typically in a range between 30 seconds and 1 minute.
Lube oil pumps	Dual (2x100%) Steel casing (option)	None or Single Cast iron casing
Lube oil coolers	Dual (2x100%)	Single or Liquid Injection
Lube oil filters	Dual	Single or dual
Lube oil piping	Carbon steel piping upstream of the filters, stainless steel piping downstream of the filters. Optional stainless steel piping upstream of the filters. Stainless or carbon steel valves.	Carbon steel piping upstream of the filters, carbon steel or optional stainless steel piping downstream of filters with carbon steel valves.
Coupling	API 671	Non API, flexible disc
Relief Valves	API 520/521, optional dual	API 520/521, single
Valves	API valves	API or refrigeration valves
Control Instruments and wiring	Separate Instruments for control and shutdown (SIS), both having SIL 2/SIL 3 rating, SMART, Class 1, Div.2, Group C &D (or IEC Zone 2, Group IIB)	Single instruments for control and shutdown, SMART Class 1, Div.2, Group C &D (or IEC Zone 2, Group IIB) for most cases, optional blind non-rated
Control Panel	PLC or DCS control panel. Separate panel for shutdown (SIS) functionality	Single PLC control panel. Optional microprocessor panels with blind instruments.

[1] Courtesy of Santosh Yarlagadda – private communication

Figure 1

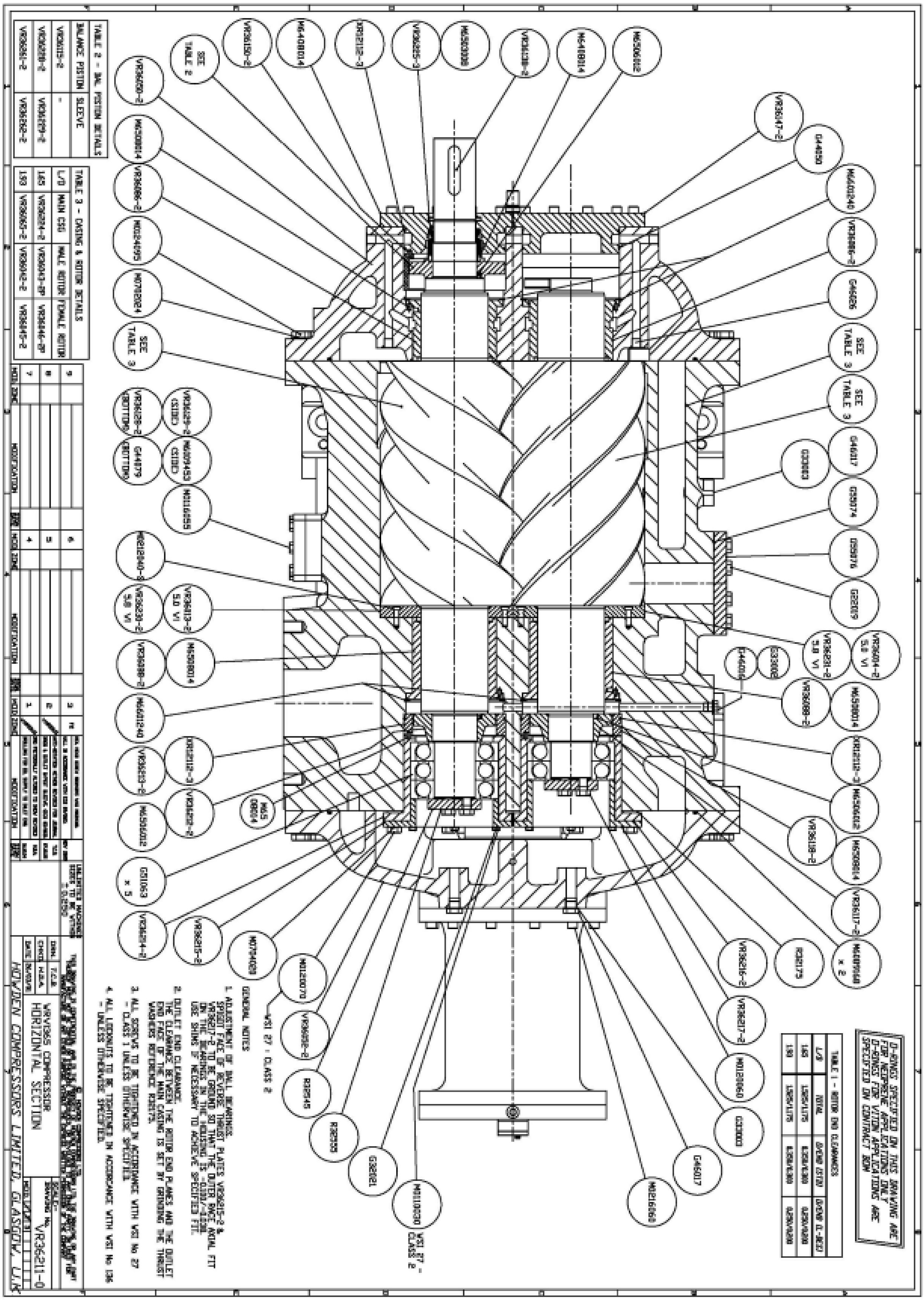


Figure 2

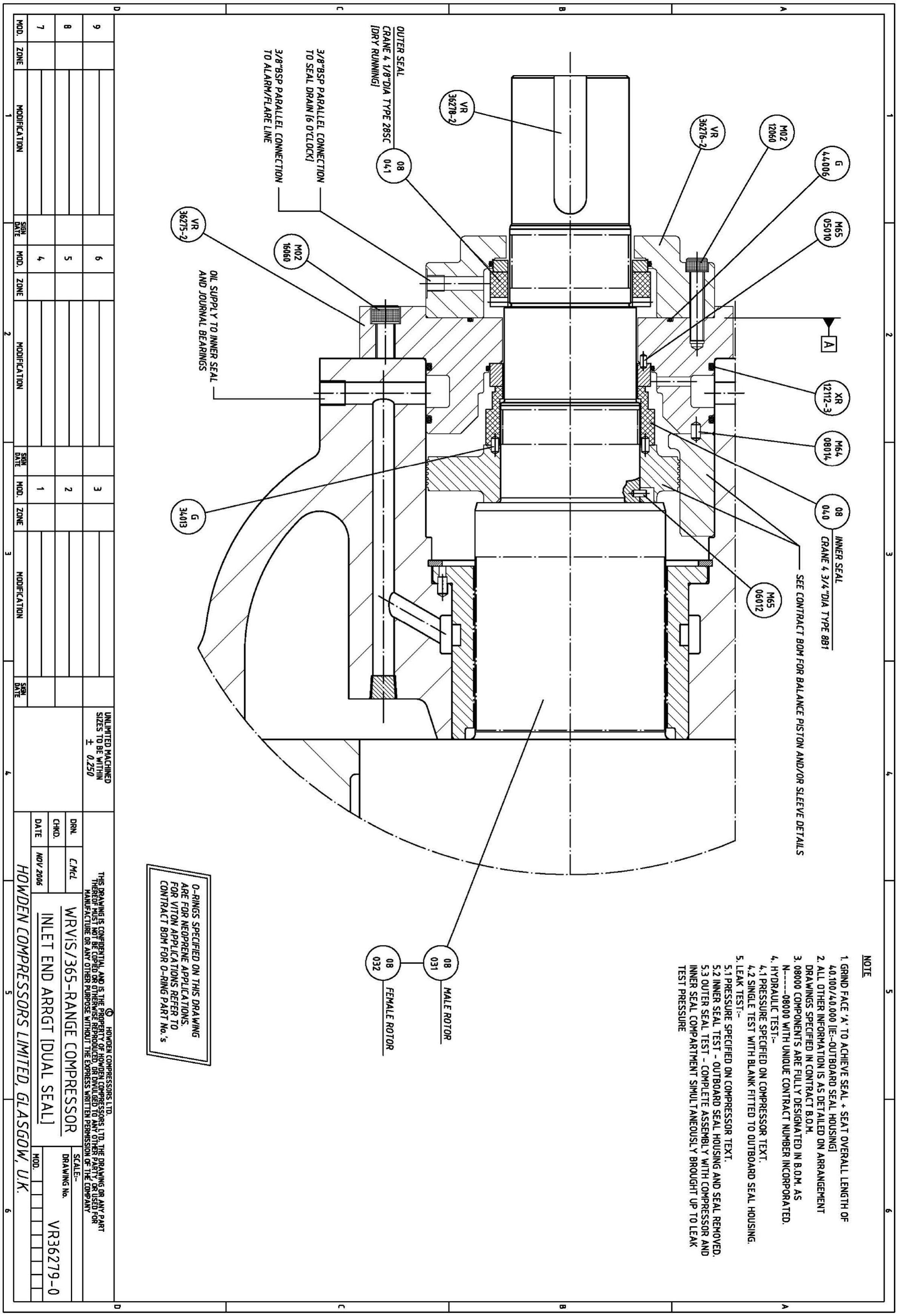
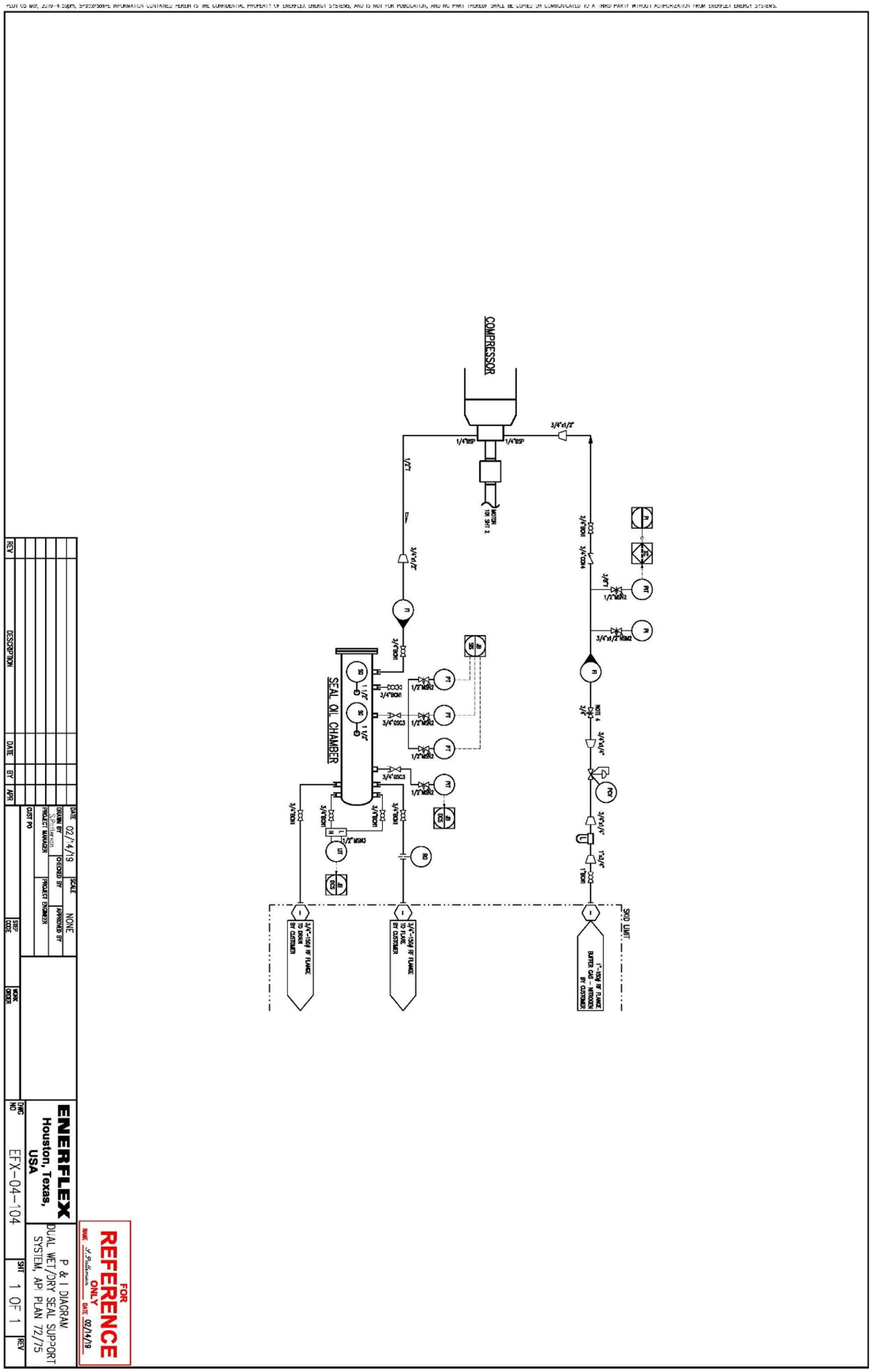

NOTE
1. GRIND FACE 'A' TO ACHIEVE SEAL + SEAT OVERALL LENGTH OF 40.100/40.000 [IE:-OUTBOARD SEAL HOUSING]
2. ALL OTHER INFORMATION IS AS DETAILED ON ARRANGEMENT DRAWINGS SPECIFIED IN CONTRACT B.O.M.
3. 08000 COMPONENTS ARE FULLY DESIGNATED IN B.O.M. AS N----08000 WITH UNIQUE CONTRACT NUMBER INCORPORATED.
4. HYDRAULIC TEST:-
4.1 PRESSURE SPECIFIED ON COMPRESSOR TEXT.
4.2 SINGLE TEST WITH BLANK FITTED TO OUTBOARD SEAL HOUSING.
5. LEAK TEST:-
5.1 PRESSURE SPECIFIED ON COMPRESSOR TEXT.
5.2 INNER SEAL TEST - OUTBOARD SEAL HOUSING AND SEAL REMOVED.
5.3 OUTER SEAL TEST - COMPLETE ASSEMBLY WITH COMPRESSOR AND INNER SEAL COMPARTMENT SIMULTANEOUSLY BROUGHT UP TO LEAK TEST PRESSURE
OUTER SEAL CRANE 4 1/8"DIA TYPE 285C [DRY RUNNING]
3/8"BSP PARALLEL CONNECTION TO SEAL DRAIN [6 O'CLOCK]
3/8"BSP PARALLEL CONNECTION TO ALARM/FLARE LINE
OIL SUPPLY TO INNER SEAL AND JOURNAL BEARINGS
INNER SEAL CRANE 4 3/4"DIA TYPE 8B1
SEE CONTRACT BOM FOR BALANCE PISTON AND/OR SLEEVE DETAILS
VR 36278-2
VR 36276-2
M02 12060
G 44006
M65 05010
A
XR 12112-3
M64 08014
08 040
M65 06012
G 34013
M02 16080
VR 36275-2
08 041
08 031
MALE ROTOR
08 032
FEMALE ROTOR
O-RINGS SPECIFIED ON THIS DRAWING ARE FOR NEOPRENE APPLICATIONS. FOR VITON APPLICATIONS REFER TO CONTRACT BOM FOR O-RING PART No.'s
UNLIMITED MACHINED SIZES TO BE WITHIN ± 0.250
DRN. C.McL
CHKD.
DATE NOV 2006
WRVIS/365-RANGE COMPRESSOR
INLET END ARRGT [DUAL SEAL]
HOWDEN COMPRESSORS LIMITED, GLASGOW, U.K.
SCALE:-
DRAWING No.
VR36279-0
MOD. ZONE MODIFICATION SIGN DATE

Figure 3


COMPRESSOR
SEAL OIL CHAMBER
SKID LIMIT
ENERFLEX
Houston, Texas,
USA
P & I DIAGRAM
DUAL WET/DRY SEAL SUPPORT
SYSTEM, API PLAN 72/75
EFX-04-104
1 OF 1
FOR
REFERENCE
ONLY
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BY
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Figure 4

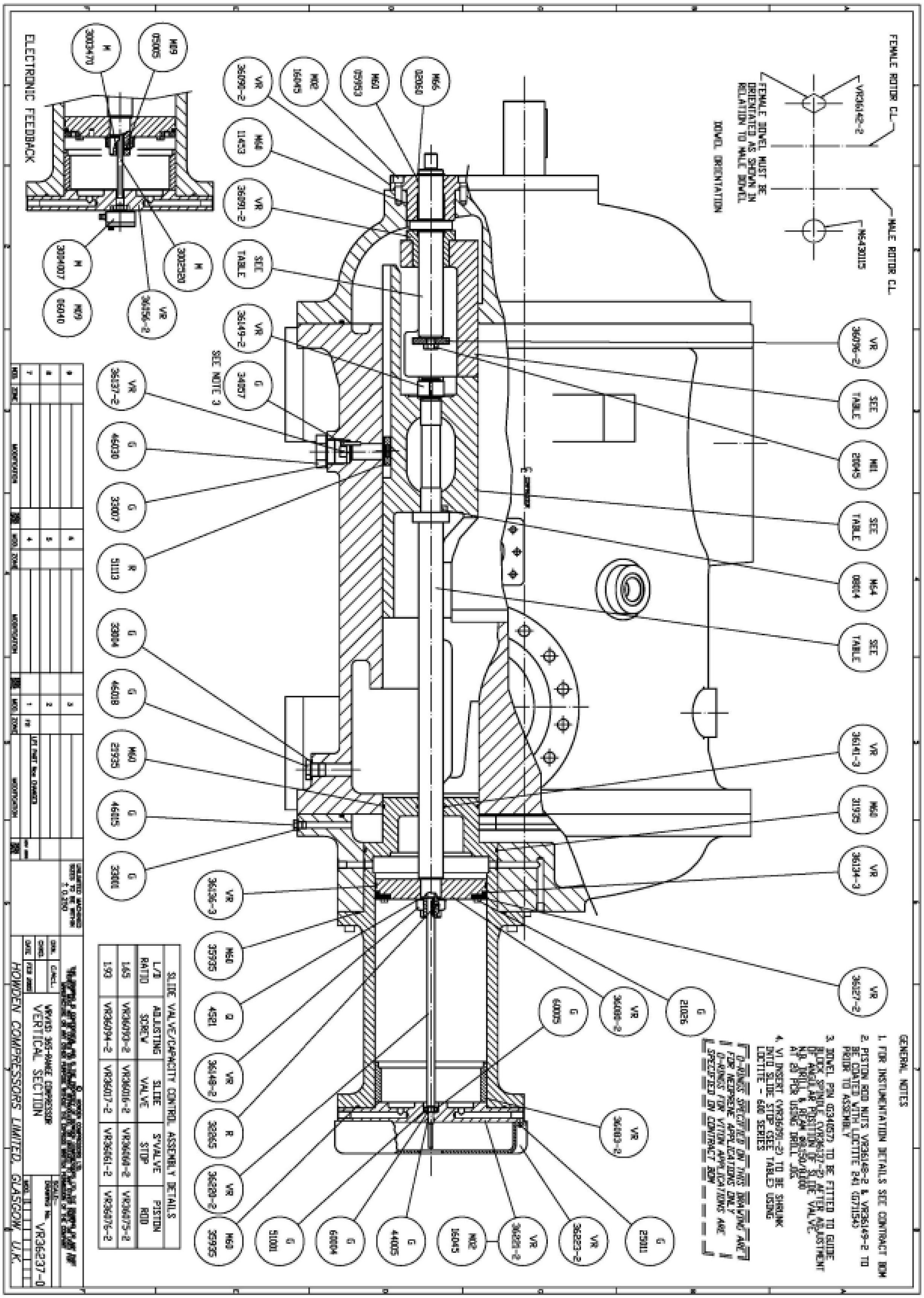
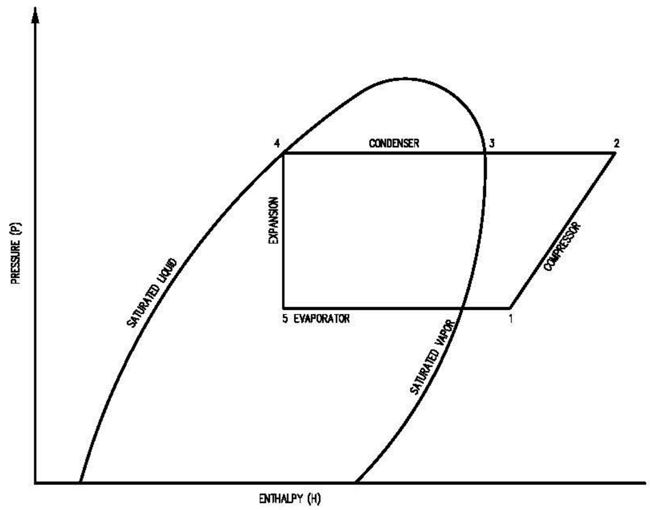
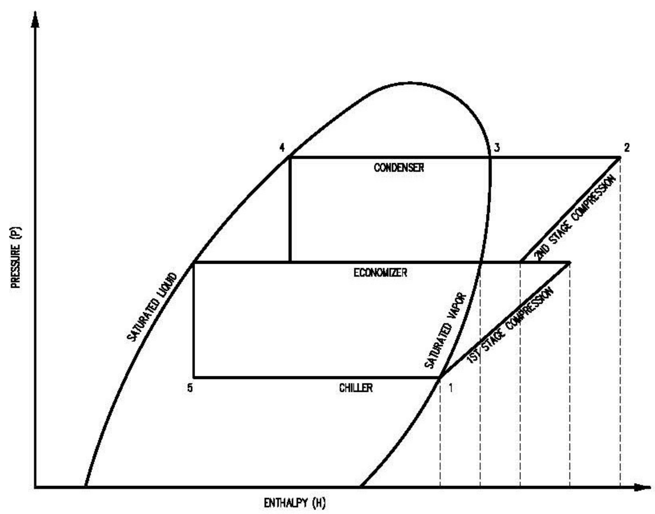

FEMALE ROTOR C.L.
MALE ROTOR C.L.
DOWEL ORIENTATION
GENERAL NOTES
ELECTRONIC FEEDBACK
SLIDE VALVE/CAPACITY CONTROL ASSEMBLY DETAILS
VERTICAL SECTION
HOWDEN COMPRESSORS LIMITED, GLASGOW U.K.

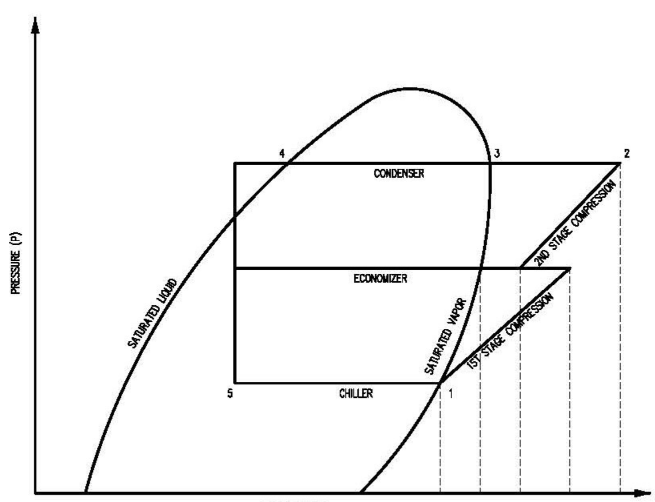
Figure 5



NON-ECONOMIZED



ECONOMIZED – FLASH



ECONOMIZED – SHELL & TUBE

Figure 6

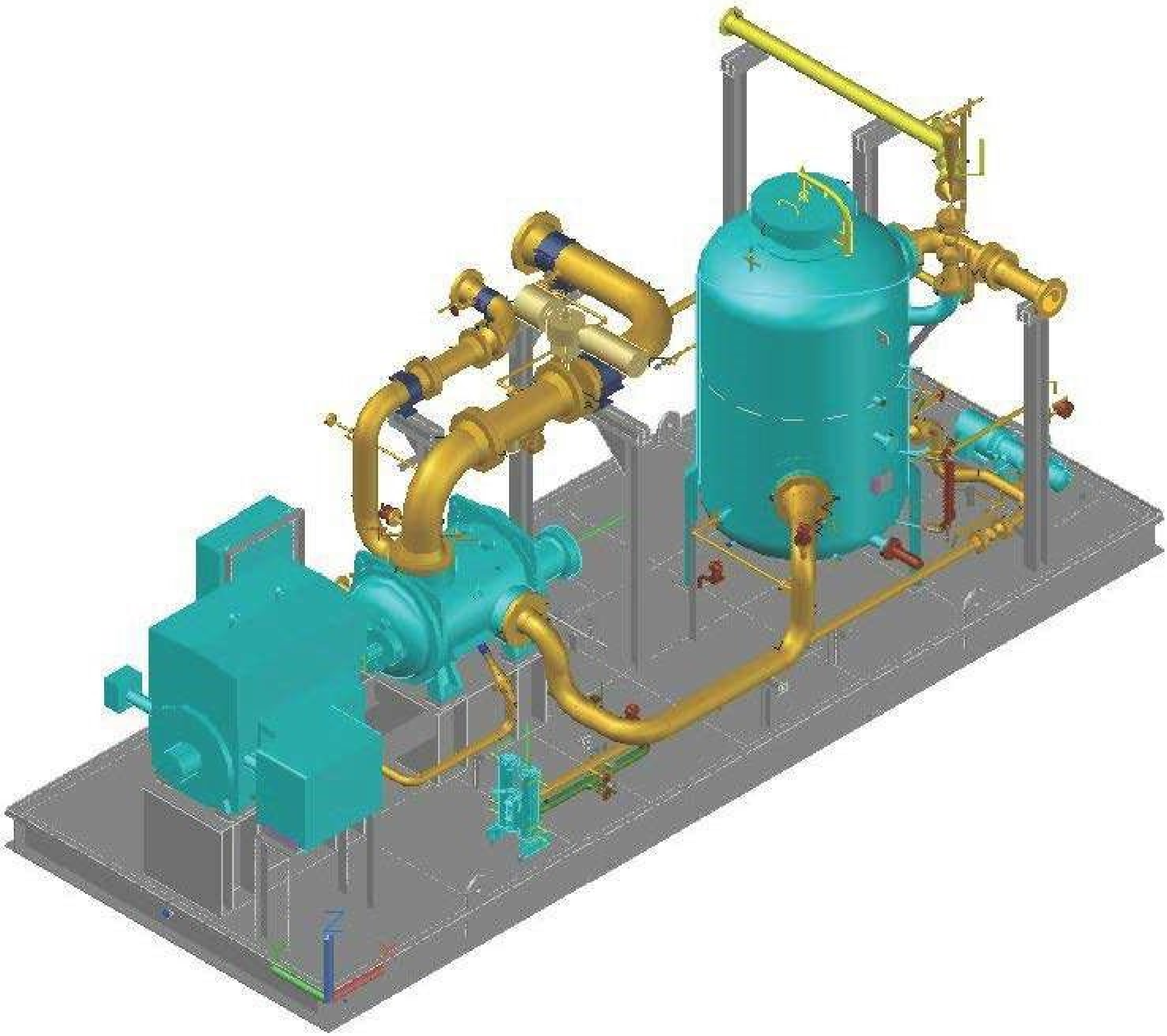


Figure 7

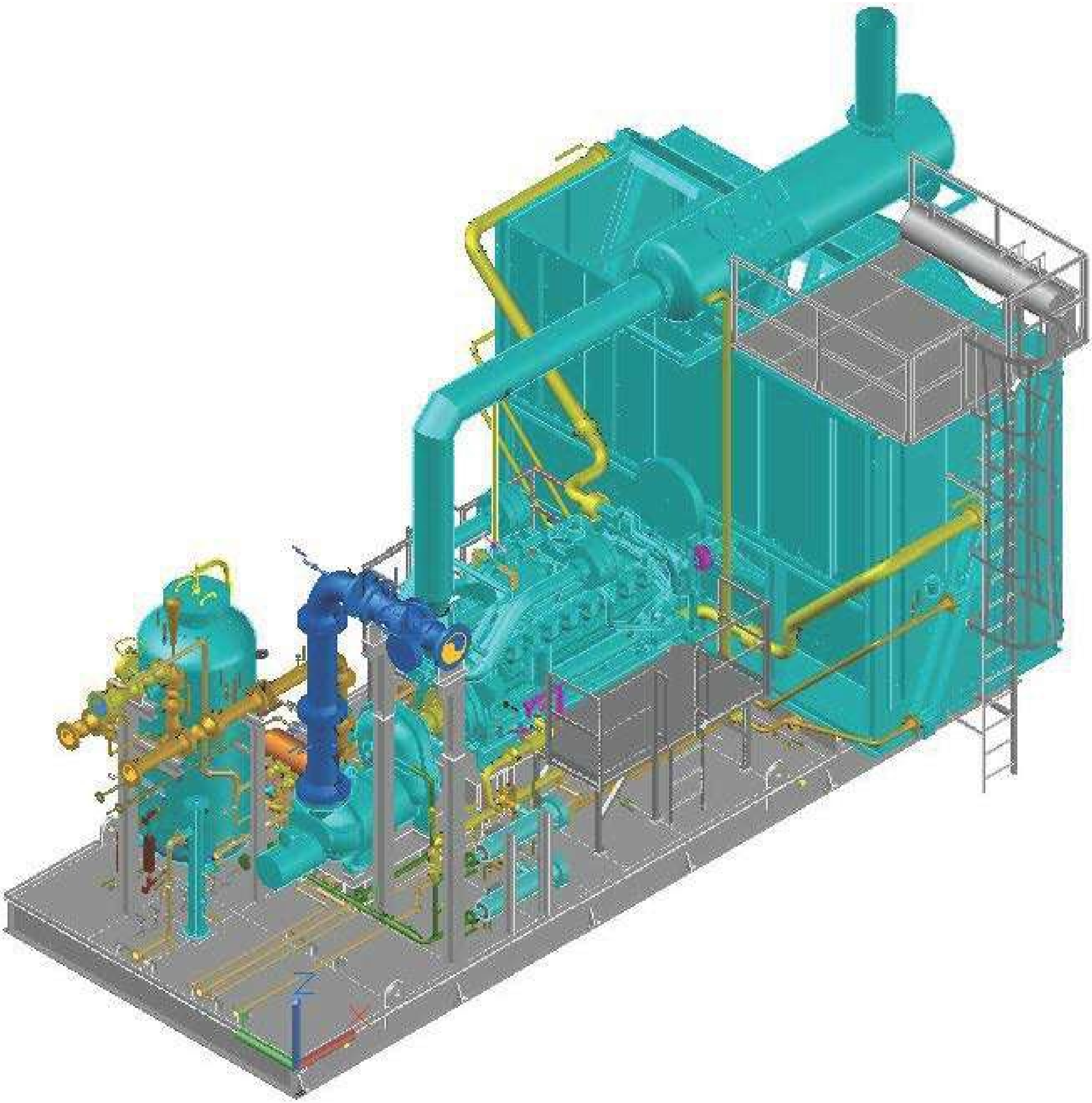


Figure 8

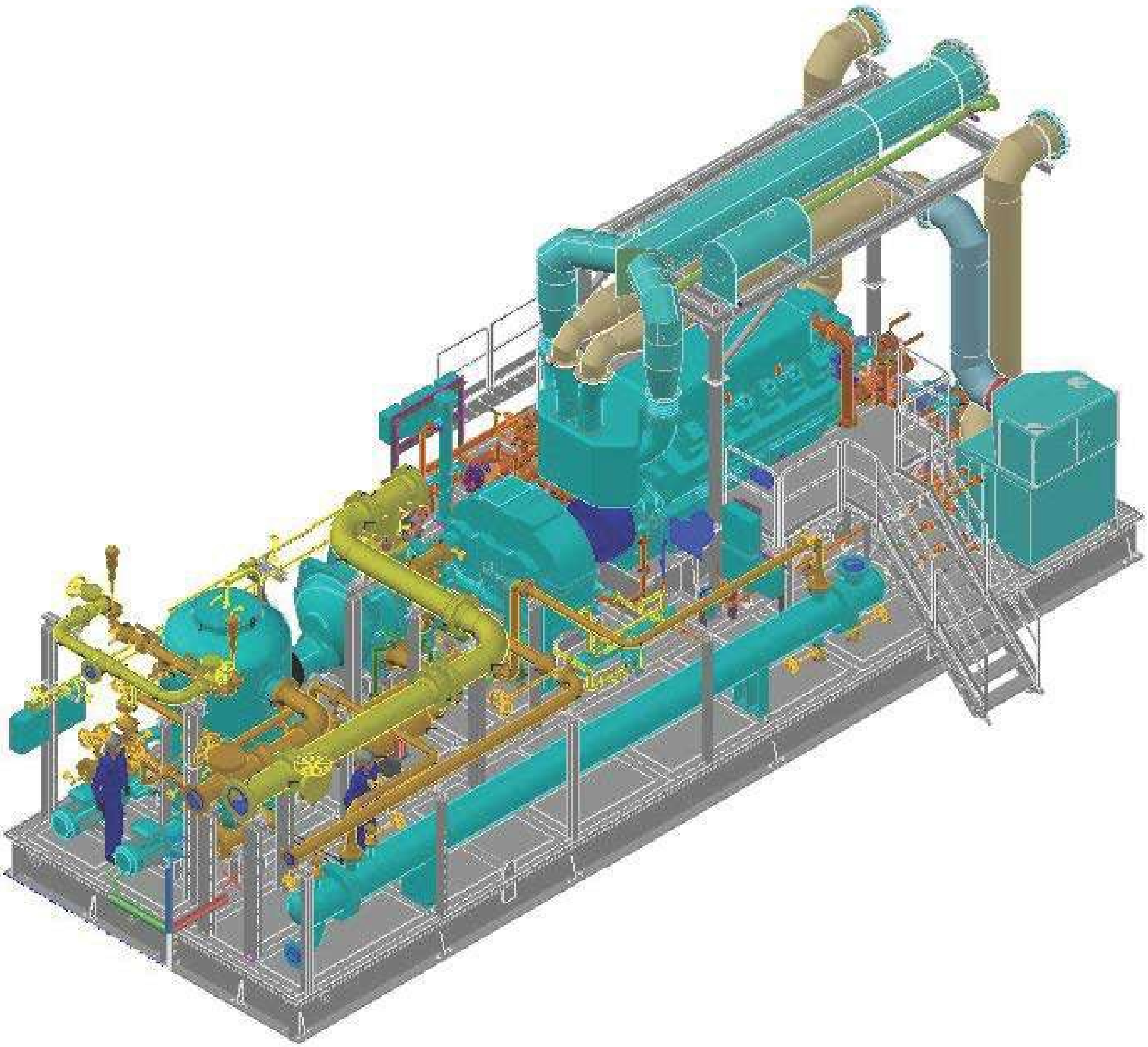


Figure 9a

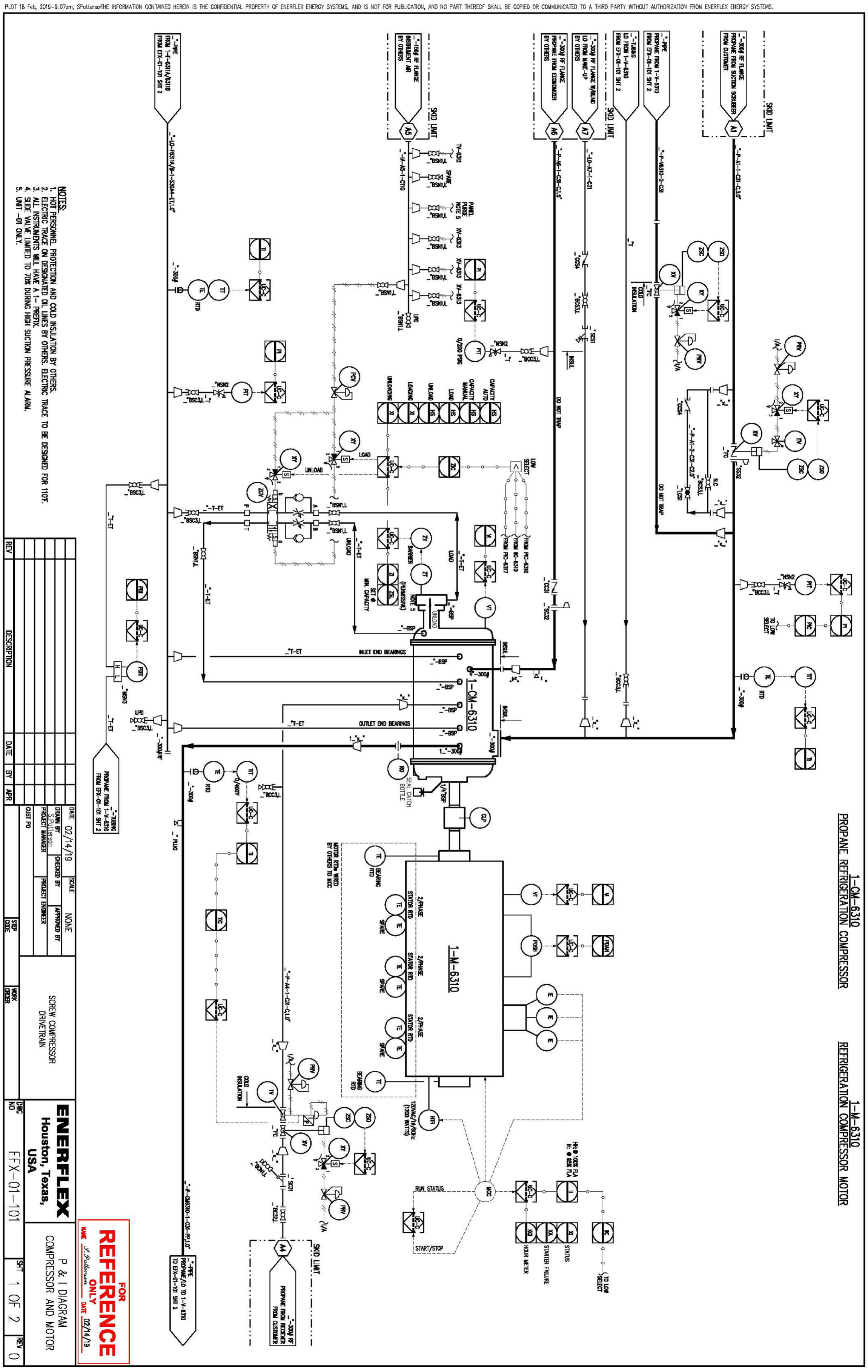
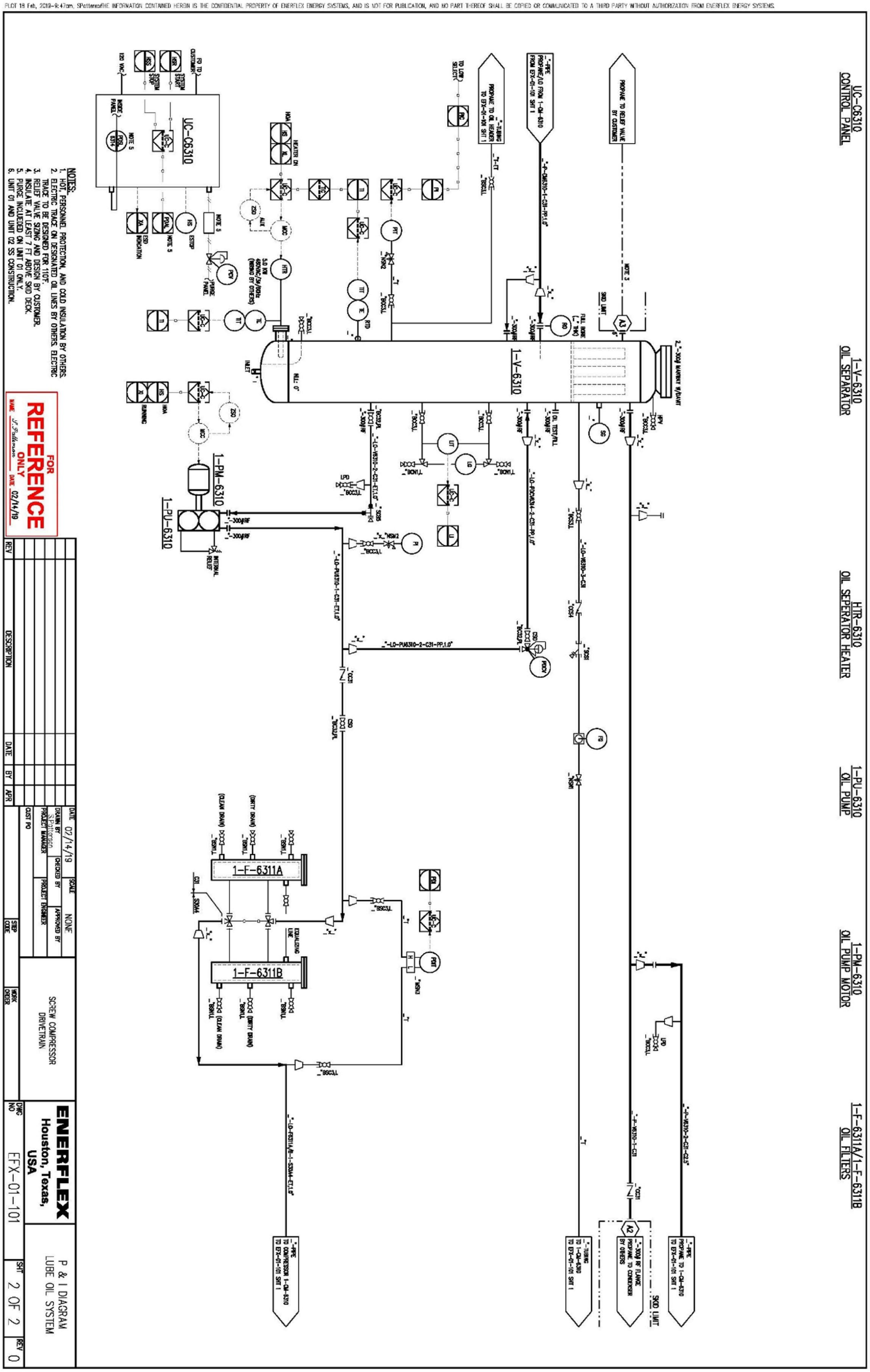

1-CM-6310
PROPANE REFRIGERATION COMPRESSOR
1-M-6310
REFRIGERATION COMPRESSOR MOTOR
NOTES:
1. HOT PERSONNEL PROTECTION AND COLD INSULATION BY OTHERS.
2. ELECTRIC TRACE ON DESIGNATED OIL LINES BY OTHERS. ELECTRIC TRACE TO BE DESIGNED FOR 110F.
3. ALL INSTRUMENTS WILL HAVE A 1- PREFIX.
4. SLIDE VALVE LIMITED TO 70% DURING HIGH SUCTION PRESSURE ALARM.
5. UNIT -01 ONLY.
SCREW COMPRESSOR DRIVETRAIN
ENERFLEX
Houston, Texas, USA
P & I DIAGRAM
COMPRESSOR AND MOTOR
EFX-01-101
1 OF 2
FOR REFERENCE ONLY

Figure 9b


UC-C6310 CONTROL PANEL
1-V-6310 OIL SEPARATOR
HTR-6310 OIL SEPERATOR HEATER
1-PU-6310 OIL PUMP
1-PM-6310 OIL PUMP MOTOR
1-F-6311A/1-F-6311B OIL FILTERS
UC-C6310
1-V-6310
1-PM-6310
1-PU-6310
1-F-6311A
1-F-6311B
NOTES:
1. HOT, PERSONNEL PROTECTION, AND COLD INSULATION BY OTHERS.
2. ELECTRIC TRACE ON DESIGNATED OIL LINES BY OTHERS. ELECTRIC TRACE TO BE DESIGNED FOR 110F.
3. RELIEF VALVE SIZING AND DESIGN BY CUSTOMER.
4. INSULATE AT LEAST 7 FT ABOVE SKID DECK.
5. PURGE INCLUDED ON UNIT 01 ONLY.
6. UNIT 01 AND UNIT 02 SS CONSTRUCTION.
FOR REFERENCE ONLY
DATE 02/14/19
SCALE NONE
SCREW COMPRESSOR DRIVETRAIN
ENERFLEX
Houston, Texas, USA
P & I DIAGRAM
LUBE OIL SYSTEM
EFX-01-101
SHT 2 OF 2
REV 0