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Revision Memo

- 03/04** References have been updated. The more noteworthy revisions reflect the following:
- Piping connections for double-wall tanks.
 - Refrigerated liquefied nitrogen and oxygen information.
 - LNG design temperatures.
 - Confined space consideration for high concrete dikes.
 - Compressor back-up considerations.
 - Pump heat input and vent lines considerations.
 - Safety valve handling equipment.
 - Slug flow considerations.
 - Pressure and level instrument considerations.

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SCOPE

This section covers refrigerated storage for refineries and chemical plants. Included are descriptions of a variety of types of refrigerated storage facilities, along with basic selection criteria. Also included are design procedures and considerations for a "typical" installation, having an aboveground steel tank and a vapor recovery type of refrigeration system.

REFERENCES

DESIGN PRACTICES (BESIDES OTHER SECTIONS OF SECTION XXII)

Section I	Index and Introduction
Section II	Design Temperature, Design Pressure and Flange Rating
Section V	Drums
Section IX	Heat Exchange Equipment
Section X	Pumps
Section XI	Compressors
Section XII	Instrumentation
Section XIV	Fluid Flow
Section XV	Safety in Plant Design
Section XVI	Thermal Insulation

GENERAL PRACTICES

GP 3-2-3	Firewater Systems
GP 3-2-4	Pressure Relieving Systems
GP 3-5-1	Fill and Discharge Lines, and Auxiliary Piping for Storage Tanks and Vessels
GP 3-10-1	Piping Selection and Design Criteria
GP 4-8-1	Tank Foundations
GP 5-1-1	Pressure Vessels
GP 9-1-1	Spacing and Dikes for Storage Vessels and Tanks
GP 9-6-1	Refrigerated Storage Tanks
GP 14-1-1	External Insulation
GP 15-1-2	Instrumentation for Compressors and Drivers
GP 15-1-3	Instruments for Storage Tanks and Vessels
GP 18-10-1	Additional Requirements for Materials

API STANDARDS / PUBLICATIONS

API STD 620	Design and Construction of Large, Welded, Low-Pressure Storage Tanks (Tenth Edition, 02/02)
API STD 2000	Venting Atmospheric and Low-Pressure Storage Tanks, Nonrefrigerated and Refrigerated (Fifth Edition, 04/98)
API STD 2510	Design and Construction of LPG Installations (Eighth Edition, 05/01)
API PUBL 2510A	Fire-Protection Considerations for the Design and Operation of Liquefied Petroleum Gas (LPG) Storage Facilities (Second Edition, 12/96)

BRITISH STANDARDS INSTITUTE (BSI) STANDARDS

BSI BS 7777	Flat-bottomed, Vertical, Cylindrical Storage Tanks for Low Temperature Service, Parts 1, 2, 3 and 4(01/93)
BSI PD 7777	Alternative Steel Selection and Its Effect on Design and Testing of Tanks to BS 7777 (05/00)

NATIONAL FIRE PROTECTION ASSOCIATION (NFPA) STANDARDS

NFPA 58	Liquefied Petroleum Gas Code (02/01)
NFPA 59	Utility LP-Gas Plant Code (02/01)

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NFPA 59A Standard for the Production, Storage, and Handling of Liquefied Natural Gas (LNG) (02/01)

OTHER LITERATURE

1. Costner, H. H., *Investment Costs for Refrigerated LPG Terminals*, Esso Engineering Report No. EE.5LD.71 (April 29, 1971).
2. Costner, H. H., *Underground Storage of LPG*, Esso Engineering Report No. EE.54E.72 (April 13, 1972).
3. Mestanas, N. M., *Safety Review of Refrigerated Liquefied Gas Facilities*, Exxon Engineering Report No. EE.91E.80 (November 6, 1980).
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5. Gas Processes Suppliers Association, *Engineering Data Book* (1987).
6. Lieberwirth, Robert A., *Refrigerated LPG Storage*, Exxon Engineering Memorandum 95PD 195 (June 6, 1995).
7. Chemical Industries Association Code of Practice for Large Scale Storage of Fully Refrigerated Anhydrous Ammonia in the U.K. (1992).
8. Engineering Equipment and Materials Users Association (EEMUA), *Recommendations for the Design and Construction of Refrigerated Liquefied Gas Storage Tanks*, (December 15, 1986).
9. W.B. Bedesem et al, *Additional Design Considerations for Refrigerated Liquefied Gas Facilities*, Exxon Engineering Report No. EE.49E.79 (July, 1979).
10. Nazario, F. N., *Rollover Protection Guidelines for Refrigerated Liquefied Gas Storage Tanks*, Exxon Engineering Report No. EE.42E.83 (June 1983).
11. The American Society for Testing and Materials, *Standard Guide for Designing Systems for Oxygen Service*, ASTM G 88 (1997).
12. The American Society for Testing and Materials, *Standard Guide for Control of Hazards and Risks in Oxygen Enriched Systems*, ASTM G 128 (1995).

BACKGROUND

Within Exxon, refrigerated storage is sometimes used for liquefied hydrocarbon gases, nitrogen, oxygen, and ammonia. The liquefied hydrocarbon gases include methane, ethylene, propane, propylene, butane, isobutane, butadiene, pentane, isopentane, and mixtures of the above hydrocarbon gases, such as propane/butane. By refrigerating these gases, they can be stored as liquids in containment that is slightly above atmospheric pressure to prevent the leakage of air into the container and the formation of explosive hydrocarbon/air mixtures. Refrigerated liquefied gases such as nitrogen and oxygen are special cases typically requiring pressurized storage in the 100-150 psig design pressure range.

The refrigerated storage temperatures depend on the component(s) being stored, and for mixtures, their exact compositions. Note that most liquefied hydrocarbon gases are actually mixtures containing a primary component mixed with other components and/or impurities. The other components and impurities and their potential range must be fully understood in setting the bases for the design of refrigerated storage facilities.

Because the environment is warmer than the refrigerated storage, provisions must be made for heat leak into the containment facility. Those provisions might be as simple (and low cost) as disposing boiled off gases to a fuel or flare system or as complex (and expensive) as re-refrigerating the boiled off gases. Refrigeration systems, which recover boil-off resulting from heat leak and which minimize product loss, are quite common and preferred from an environmental standpoint.

DEFINITIONS

(IN ADDITION TO THOSE COVERED IN SECTION XXII-A)

Liquefied Natural Gas (LNG)

Natural gas that has been liquefied at extremely cold temperatures. It is typically stored and transported at refrigerated temperatures [-270°F (-168°C) design] and at pressures slightly above atmospheric.

Liquefied Petroleum Gas (LPG)

Propane, butane, or (more commonly) a mixture of propane and butane obtained from natural gas or from refinery processes. LPG has a vapor pressure sufficiently low to permit compression and storage as a liquid at moderate pressures and normal ambient temperatures.

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EQUIPMENT TYPES AND APPLICATIONS

Refrigerated storage requires the use of expensive alloy materials to ensure ductility, extensive insulation and barrier protection to minimize heat and moisture leak, and significant engineering effort to achieve safe and reliable operations. Due to its specialized nature and relative expense, refrigerated storage vendors are continually developing proprietary design and construction technology in their efforts to secure a competitive advantage. It should be noted that features which make a particular design economic in one location might make it uneconomic in another due to differences in material, engineering, and labor costs. Considering these factors, vendors and other EME specialists should be consulted early in establishing the siting and design bases of any refrigerated storage facility. The following information is intended to provide the necessary background to facilitate that consultation.

ABOVEGROUND STEEL VESSELS

Aboveground steel vessels represent the most common type of refrigerated storage containment. They can be in the shape of drums, spheres, or vertical cylindrical tanks. Design and operating conditions are established based on an economic evaluation of the investment cost for the containment system, which is greatly affected by the alloy content of the materials used, and the investment and operating cost of the required refrigeration system.

Drums find application primarily in the storage of small and intermediate quantities of very high vapor pressure, and therefore very low storage temperature, stocks (LNG and other cryogenic gases). A practical maximum diameter, to avoid excessive costs for structural reinforcement, is about 15 ft (4.6 m). This is also a common limitation for shop-fabricated vessels due to transportation clearance limitations. However, larger diameters are possible where there is an economic incentive.

Spheres are commonly used for storing intermediate volumes of refrigerated and partially refrigerated product. For partially refrigerated storage, the storage temperature is not cold enough to permit atmospheric or near atmospheric pressure storage. Rather, the storage is at some temperature above the boiling point at atmospheric pressure, requiring storage pressure significantly above atmospheric pressure to prevent the liquid contents from boiling. Vertical cylindrical tanks are used for storing large volumes of refrigerated product at or near atmospheric pressure.

The maximum currently feasible vessel size is about 900,000 barrels. Vertical cylindrical tanks are classified as single containment, double containment, or full containment tanks.

SINGLE CONTAINMENT TANKS

A single containment tank can be either a single tank or a combination of an inner tank and an outer container designed and constructed so that only the inner tank is required to meet the low temperature ductility requirements for storage of the refrigerated product.

The outer container (if any) of a single containment tank serves primarily to retain and protect insulation and to contain the vapor purge gas pressure, but it is not designed to contain the refrigerated liquid in the event of leakage of the inner tank. Since it is subject only to ambient temperatures, it is generally constructed of ordinary carbon steel and designed for the mechanical loads imposed by the insulation and the purge gas pressure in the annular space between it and the inner tank.

The annular space is typically 2 to 3 ft (70 to 100 cm) to facilitate personnel access. A 4 to 8 in. (10 to 20 cm) resilient blanket of textile fiberglass is often installed immediately adjacent to the inner tank to absorb the expansion generated during warm-up cycles when a tank is taken out of service. The remainder of the annular space, which is 19 to 32 in. (50 to 80 cm), is filled with insulation (typically perlite). Generally, this type of tank is not economic unless the storage temperatures require such thickness of insulation. See [Figure 1](#) (from British Standard BS-7777) for examples of single containment tanks.

DOUBLE CONTAINMENT TANKS

A double containment tank is a double tank designed and constructed so that both the inner tank and the outer tank (or wall) are capable of independently containing the full volume of refrigerated liquid stored. To minimize the pool of any liquid product leakage from the inner tank, the outer tank (or wall) is located at a distance not exceeding 20 ft (6 m) from the inner tank. Under normal operating conditions, the inner tank contains the refrigerated product. Since the inner tank supports the roof, the outer tank (or wall) is intended only to contain any refrigerated liquid product leakage from the inner tank; it is not intended to contain any vapor resulting from the liquid product leakage. See [Figure 2](#) (from British Standard BS-7777) for examples of double containment tanks.

FULL CONTAINMENT TANKS

A full containment tank is a double tank designed and constructed so that both the inner tank and the outer tank are capable of independently containing the full volume of refrigerated liquid stored. The space between the inner tank and outer tank is typically 3.3 to 6.6 ft (1 to 2 m). Under normal operating conditions the inner tank contains the refrigerated product. Since the outer tank supports the roof, the outer tank is intended to be capable both of containing the refrigerated liquid and controlling

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venting of the vapor resulting from product leakage from the inner tank. See [Figure 3](#) (from British Standard BS-7777) for examples of full containment tanks.

SELECTION OF SINGLE, DOUBLE, OR FULL CONTAINMENT

The failure of any tank to contain its liquid content, particularly a flammable hydrocarbon liquid, is an extremely hazardous contingency. The large volume of flammables coupled with the relative ease with which ignition may be realized, makes primary containment security of paramount importance. This importance has led ExxonMobil, and the industry, to follow a number of procedures and standards in the design, construction, and operation of these facilities.

Refrigerated liquid storage presents an added dimension to the urgency for containment security in that a release of cold liquid will produce a gas cloud due to the environment (earth, sea, structures) providing a heat sink for the vaporization of the cold liquid. The gas cloud can easily mix with air, producing an explosive mixture of hydrocarbons and air.

In addition, containment failure in refrigerated service can be catastrophic if caused by brittle fracture, a phenomenon that can be described as the high speed propagation of a crack at nominal stresses well below the allowable stress. It takes place without yielding of the steel.

Refrigerated oxygen has additional inherent risks that must be considered. Examples are the potential for metal fires, changes in flammable limits, lower auto-ignition temperatures, and lower ignition energies. If refrigerated liquid oxygen use or storage is considered, the EME Safety and Risk Section should be contacted.

Due to these added dimensions associated with refrigerated storage, a number of additional safeguards, represented by additional procedures and standards, are followed by ExxonMobil and the industry to further increase the integrity of the storage vessel or tank. These include full tank height hydrotesting, lower permissible foundation settlements than allowed for conventional tanks, enhanced reliability for instrumentation to prevent tank overfilling, and emergency shutoff on loading lines at marine terminals. For tanks in congested sites, consideration should be given to high concrete dikes or walls or remote impoundment to reduce potential vapor cloud travel from leaks. If concrete dikes higher than 6 ft are considered, then the site Safe Operations Committee (SOC) should be consulted regarding confined space classifications. When these standards and safeguards are followed, a single containment design is considered to be technically acceptable for refrigerated storage and it represents the most cost-effective method of aboveground refrigerated storage.

Notwithstanding the extra emphasis in obtaining integrity for the storage vessel of refrigerated liquids, some international standards and local laws, especially of some European and Mid-East countries, require additional containment safeguards for certain refrigerated liquids. These additional safeguards may include the requirement for double or full containment.

As can be readily imagined, additional containment is costly. Compared to single containment, some authors of papers presented at LNG/NGL conferences have indicated additional costs for full containment of 115% to 215% for propane storage and 70% to 150% for LNG storage. In addition, quotes have been received from refrigerated storage vendors indicating even higher differentials for full containment versus single containment. Because of the higher costs, double or full containment is usually specified only when local regulations and/or site specific factors require its use.

In considering the bases for refrigerated storage, the standards and local regulations that are to govern the design and operation of the refrigerated storage facilities must first be understood. Those local regulations, which are specific to the plant location and set the containment requirements, will also define design criteria/severity of events such as fire, blast, flying objects, and other situations that have to be considered and designed for.

CONCRETE TANKS

Prestressed concrete tanks and reservoirs have been designed and constructed for use in a wide variety of storage services for a number of years. Their most extensive use has been for ambient temperature liquid storage tanks and reservoirs, both aboveground and inground. Techniques for such storage are well established. Prestressed concrete tanks have also found application in storage of refrigerated products, primarily LNG, but only for use as an outer wall of a storage tank designed for double or full containment.

Reinforced concrete tanks have also found application in storage of refrigerated product, but only for use as an outer wall of a storage tank designed for double or full containment and with an earth embankment.

When there appears to be an economic incentive for the use of concrete tanks, the Civil Engineering Group within ExxonMobil Engineering - Technology Department (EMETD) should be consulted for specific construction details and requirements.

NON-CONVENTIONAL STORAGE

Mined caverns and frozen earth cavities are examples of non-conventional refrigerated storage. This type of storage may be competitive with conventional storage in larger volume applications, i.e., over approximately 60,000 barrels (9,540 m³). However, inground refrigerated storage should presently be limited to temperatures warmer than -50°F (-46°C). An inground refrigerated storage cavern is shown conceptually in [Figure 4](#).

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Because of the very specialized nature of these types of storage, they require examination on an individual and specific basis and are outside the scope of this Design Practice. Report No. EE.54E.72 provides some guidelines for use in early planning feasibility studies of underground LPG storage. In addition, the Civil Engineering Group within EMETD can be consulted on specific applications.

REFRIGERATION SYSTEMS

A typical offsite refrigerated storage facility will employ an autorefrigeration system to recover boil-off as described below. Other refrigeration schemes to recover boil-off are used which may be necessary or desirable under some circumstances. Design details for these other systems will not be specifically addressed below, although the basic considerations and procedures presented herein apply to, and establish guidelines for, these alternative refrigeration schemes.

Autorefrigeration - This type of system is usually used when the stored product can be used as a refrigerant. The desired storage temperature is maintained through autorefrigeration of the product, i.e., heat leaking into the vessel and absorbed by the liquid product is dissipated through boil-off of a part of the liquid. The resulting vapor is withdrawn from the top of the vessel and compressed to a pressure at which the corresponding saturation temperature (i.e., condensing temperature at that pressure) is above that of the available cooling medium. This limits the applicability of an autorefrigeration recovery system to the use of stored products where the available cooling medium, typically air and cooling water, can condense the compressed gas at a reasonable pressure, and of course, to products that can be safely or easily used as a refrigerant. Propane, butane, and propane/butane refrigerated storage products are amenable to an autorefrigeration recovery system. The compressed vapor is condensed in a heat exchanger. The condensed liquid is expanded, providing refrigeration, and the resulting cool liquid is returned to the tank. The flashed vapor is returned to the compression system. A typical two-stage vapor compression autorefrigeration system is shown in [Figure 5](#).

External Refrigerant - When the stored product cannot be used as a refrigerant, as is the case with some chemical products such as butadiene, and with light products such as ethylene and methane where the vapor cannot be condensed at a reasonable pressure by air or water, it is necessary to use an external refrigerant separated from the stored product in a closed system. In such a system, the stored liquid product is recirculated through a heat exchanger to maintain the desired storage temperature. The closed independent refrigeration system provides the refrigeration to that heat exchanger. A refrigeration system employing an external refrigerant is usually more expensive than an autorefrigeration system.

Recirculated systems using external refrigerant may, however, be appropriate for small installations where packaged refrigeration units can be readily specified, purchased, and installed. A packaged, off the shelf, refrigeration unit using external refrigerant may be lower in cost than an autorefrigeration system for small installations due to the high engineering cost for a tailor made autorefrigeration system.

Integrated Refrigeration Systems - Where two or more refrigerated products having different boiling points are being stored in a common storage area, integration of the refrigeration systems for the various individual products may be possible and may afford system economies. Such systems combine the autorefrigeration and external refrigerant type systems. An autorefrigeration system is used to refrigerate the product with the lowest boiling point. Cold liquid product from that system is then used as a coolant to refrigerate, in an external refrigerant recirculated system, the other higher boiling products. A schematic diagram of an integrated refrigerated system for propane and butane is given in [Figure 6](#).

The vapor compression refrigeration system used for the lower-boiling product must, of course, be sized and designed to handle the maximum duty resulting from expected heat leakage into all products of the integrated systems.

REFRIGERATION COMPRESSORS

Compressors for refrigeration systems are selected, designed, and specified in accordance with considerations presented in [Design Practices Section XI](#) and the work is usually coordinated with the Machinery Engineering Section of EMETD.

Type - Many types of compressors have been used in refrigerated storage refrigeration systems and the selection of type of compressor has to take into account the capacity or flow, the head or compression ratio, the operating temperature, turndown requirements, investment cost, and efficiency.

Reciprocating compressors are frequently used in small autorefrigeration systems handling propane or butane because the small flow and required head to condense the compressed gas product are frequently a good match to a reciprocating compressor's capability. The operating temperature, above -50°F (-46°C), is also within the compressor's lubrication and materials capability. In addition, the reciprocating compressor can easily be turned down to lower than design flow capacity at reasonable efficiency.

Centrifugal compressors are used for the larger refrigerant flow situations and where the head developed by a centrifugal machine results in a discharge pressure sufficient to condense the compressed gas with air and/or cooling water. Centrifugal compressors are also used where the operating temperatures are colder than -50°F (-46°C). High speed impeller wheels and multiple wheels, or stages, have been used to get the head of the centrifugal machine up to required levels for the intended service.

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Turndown - The turndown requirements of the refrigeration system have to be established taking into account variations in both ambient conditions and product storage levels. The selected compressor system must meet those turndown requirements.

Reciprocating compressors can meet turndown requirements down to zero flow, in stages, utilizing "clearance pockets" on the machine that can be manually or automatically controlled (loaded/unloaded).

Centrifugal compressors are limited to turndown in the 60 to 70% of design flow range because of aerodynamic stall (surge) on the impeller wheels when the flow is insufficient. The surging can cause vibration and failure of the wheel(s). For net refrigerant flow below about 70%, the centrifugal refrigeration compressor must go on recycle sufficient to keep the flow through the machine above the flow that can cause surge. This is obviously inefficient and may be uneconomical where substantial recycle is required. Alternatively, a throttling valve can be employed on the compressor suction, however, its use is usually limited by the desire to keep the compressor suction pressure from going below atmospheric pressure.

Maintenance and Run Lengths - Generally speaking, reciprocating compressors require more maintenance and have shorter run lengths than centrifugal machines, typically averaging about 3,000 hours. Centrifugal machines offer the advantage of uninterrupted run lengths approaching 24,000 hours and are favored by facilities owners and operators for that reason.

The differences in maintenance requirements and run lengths between overhauls must be taken into account in the design of a refrigerated storage facility. Multiple machines may be required to meet the boil-off recovery requirements of the refrigerated storage facility, particularly when reciprocating compressors are used. Taking into account the machine maintenance requirements(s), consideration may be given to alternative boil-off gas disposal, however, the low boil-off gas pressure which makes alternative boil-off gas disposal difficult must also be considered.

Compressor Back-up - In addition to maintenance and run length considerations in design of the boil-off gas compressors, provisions for temporary/emergency back-up should also be considered. Use of ejectors directed to the flare or other safe dispositions have proven effective as backup for single compressor services. Note that if steam or nitrogen is used as motive force for an ejector, the design shall include check valves to prevent backflow and freeze-up concerns. Consideration shall also be given to use of quick-opening blinds to ensure ongoing steam or nitrogen leakage does not create icing, freeze-up, or product contamination problems.

DESIGN CONSIDERATIONS

REFRIGERATION SYSTEMS

Product Drying - Warm stocks sent to refrigerated storage must be dried to minimize hydrate (a hydrocarbon/water slush) formation, which may require limiting the water content to 10 wppm maximum. Heat regenerated desiccant dryers are usually specified for this purpose. The drying of product rundown is usually an onsite design function and is therefore not covered in this Design Practice. However, it is mentioned here to remind the offsite designers of the dryness requirement particularly since the stocks may come from a variety of sources such as ships, trucks, rail cars, etc.

Cooling Load - The refrigeration load results from various heat leaks into the system, product loading operations, and feed conditions. Sources of vapor generation or boil-off typically include:

- Hot product rundown.
- Heat leak from the environment into lines and vessels.
- Heat input from pump work.
- Cooldown of ship, lines, and/or storage facility.
- Vapor displaced during transfer and/or rundown operations.
- Atmospheric pressure changes (usually significant for LNG only).

The heat gain by a refrigerated storage system may be calculated using procedures and correlations given in [Design Practice Section XVI, Thermal Insulation](#). Depending on tank construction, either expanded perlite or polyurethane is generally used as insulation. Heat pickup should be considered for all system components which contain refrigerated product (tanks, lines, drums).

Since a dome roof tank is frequently used for refrigerated product storage and since its roof area available for heat transfer is not easy to calculate, the following empirical correlations may be used to estimate a roof area for the purpose of calculating heat leak from the roof.

$$\text{If tank diameter} < 60 \text{ ft (18.3 m),} \quad A = 0.8418 D^2$$

$$\text{If tank diameter} > 60 \text{ ft (18.3 m),} \quad A = 0.8822 D^2$$

where: A = Area of dome for heat transfer purposes, ft² (m²)
D = Diameter of tank, ft (m)

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As in the case of hot insulation, insulation thickness is a matter of economics. Insulation investment is justified on the basis of reduced refrigeration investment and reduced operating costs.

Product loading operations add heat to the system in addition to normal atmospheric heat leak into the storage tank(s), loading lines, and other equipment. Pump work during tank filling or tank recirculation also contributes to the overall refrigeration load. This pump work heat load may be approximated by the following equation:

$$Q = CWH \left[\frac{1 - e}{e} \right]$$

where: Q = Heat generated from pumping, Btu/h (kW)
C = Conversion factor, 1.286×10^{-3} (2.722×10^{-6})
W = Mass flow rate, lb/h (kg/h)
H = Net differential head, ft (m) of liquid pumped
e = Pump efficiency, fraction

Heat input from pump minimum flow recycle streams should be considered for all operating scenarios, especially where multiple product/delivery/shipping pumps are involved. Full kickback flows should be used as a design basis. Use of pump autostart can be problematic in cryogenic services and is not recommended as a means to minimize pump kickback vapor load. Product shipping schedule influences refrigeration load to some extent. Frequent shipping of small parcels tends to favor maintaining pumps and lines cold. Infrequent loading of large cargoes, on the other hand, would favor allowing these facilities to warm up between loadings, thus saving energy costs. In this latter case, a cooldown period is often required prior to shipping. The length of the cooldown period depends on the temperature level (the mechanical stress needs to be limited) and the capability of the refrigeration facilities to accommodate the boil-off rate.

Feed properties and conditions have a major impact on refrigeration duty. The feed composition and storage pressure determine the required product storage temperature and the feed temperature significantly influences vapor generation which must be handled by the refrigeration system. Hot liquid feeds, such as those from light ends units, will require large refrigeration work to achieve storage conditions; whereas cold feeds will require less refrigeration. Therefore, it is essential to consider feed composition, feed properties, and conditions, along with possible variations in those properties and conditions, to design a suitable refrigerated storage facility.

Normal tank pressure controllers and safety valves discharging to the atmosphere work on the difference between atmospheric pressure and tank pressure. A falling ambient pressure will result in a higher differential pressure and be seen as a higher tank pressure by a normal pressure controller resulting in more boil-off to be accommodated by the refrigeration system. Therefore, where the rate and degree of ambient pressure change can influence refrigeration size, the use of an absolute pressure controller should be considered. An absolute pressure controller is not influenced by atmospheric changes. However, when an absolute pressure controller is used to control the tank pressure, the tank safety valve set pressure and the tank design pressure need to be adjusted upward accordingly because safety valves discharging to the atmosphere and containers restraining contents from the atmosphere only see differential pressure.

Some typical atmospheric pressure ranges which generally hold for "non storm prone" climates are as follows:

Normal Range: 14.49 to 14.98 psia [99.91 to 103.28 kPa (abs)]

Maximum Range: 14.00 to 15.23 psia [96.53 to 105.01 kPa (abs)]

Maximum Short-Term Rate of Change: 0.147 psi in 1 hour (1.01 kPa in 1 hour)

Maximum Long-Term Rate of Change: 0.433 psi in 4 hours (2.99 kPa in 4 hours)

Of course, storm conditions, which are more severe, must also be recognized and designed for.

Full Versus Partial Refrigeration - In designing storage facilities for a liquefied hydrocarbon gas, one has the options to use pressure storage, full refrigeration storage or partial refrigeration storage. Pressure storage (covered in [Section XXII-C](#)) contains the liquid/gas at ambient temperature, while full refrigeration stores the product sufficiently cold that it remains liquid at or slightly above ambient pressure. Partial refrigeration is between the two, i.e., it maintains the product liquid at a pressure above atmospheric pressure but below the vapor pressure at ambient temperature. The choice among these storage options is made consistent with the economic guidelines of the project, taking into account not only storage investment/operating cost optimization but also the intended shipping and receiving requirements.

Partial refrigeration facilities utilize insulated pressure vessels, that is, vessels operating below ambient temperature and at pressure significantly above atmospheric pressure. Refrigerated drums or spheres would be required for these pressure levels, subject to the volume limitations previously discussed.

The major refrigeration loads are chilling the rundown or tankage feed to storage conditions and holding storage temperature in the absence of any rundown. With a warm liquid rundown, rundown refrigeration accounts for most of the total chilling load when the liquefied gas is stored at atmospheric pressure. Partial refrigeration, or storing the liquid above atmospheric pressure, would have a significant impact on reducing refrigeration load. In addition, the complexity of the refrigeration system

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is reduced, since single-stage compressors might be acceptable. However, the investment to contain the pressure is significant and heavily dependent on volume.

Compressor Sparing - Basically, from an operating standpoint, the design should accommodate the situation where one compressor is out of service for whatever reason. This may require a spare machine, vent facilities, intermediate tankage, or combinations of these. Each design must be evaluated independently and sufficient weight must be given to environmental, safety, and economic factors. Some configurations that have been found acceptable in the past are:

1. **Configuration A**
 - a. Hot rundown to storage.
 - b. One large compressor for rundown and holding refrigeration.
 - c. One small compressor for holding refrigeration.
 - d. Hot rundown bypass to pressurized storage.
2. **Configuration B**
 - a. Hot rundown to storage.
 - b. One large compressor for rundown and holding refrigeration.
 - c. One small compressor for holding refrigeration.
 - d. Controlled overpressure vent to flare.
3. **Configuration C**
 - a. Cold rundown to storage.
 - b. Two 100% compressors for holding refrigeration.

From a safety standpoint, the loss of all refrigeration must be anticipated, considered, and provided for in the integrity of the storage facilities.

Compressor Driver Selection - Driver selection for the refrigeration compressors is generally based on the major driver study for the project. Where the refrigeration facilities in question represent an independent project, an evaluation should be carried out consistent with considerations and methods presented in [Section XI-K](#).

Reacceleration after power outages is generally not specified as a requirement for electric motor drivers in refrigeration service. Short-term pressure build-up and/or venting, which results from voltage dips and temporary power outages, can usually be tolerated.

Compressor Inlet Conditions - The suction pressure of compressors in hydrocarbon service should normally be maintained positive to prevent air leakage into the compressor, thereby reducing the risk of forming an explosive mixture. Also, a slight superheat of the inlet gas should be considered to preclude condensation and liquid droplets from damaging the compressor. This would be particularly pertinent when handling a product whose storage temperature is sometimes above ambient temperature or a product that has unusual thermodynamic properties. While centrifugal compressors are more tolerant of liquid droplets than reciprocating compressors, they too can be damaged by excessive droplet formation and/or entrainment. Superheat of as little as 10°F to 20°F (6 to 11°C) has frequently been found to be sufficient to prevent condensation.

In some cases, such as ethylene storage, the suction inlet has been heated above -50°F (-46°C) by exchange with the compressor product to permit utilization of reciprocating machines.

Compressor Cooling - Although reciprocating compressors in refrigeration service are generally provided with cylinder jackets and compressor cylinder cooling, the jacket cooling medium may or may not be circulated depending on the operating temperature levels within the compressor. Where it can be determined that circulation is not required, the jackets should be filled with a non-freezing fluid, such as ethylene glycol, and provided with a standpipe to permit expansion, level checks, and filling. When circulation is required, a closed cooling water system should be considered. In areas of low ambient temperature, the cooling water system may have to be provided with heat tracing and insulation to guard against freezing. See [Section XI-H](#).

Lube Oil Separation Facilities - Reciprocating compressors should generally be of the non-lubricated type. Although they are more expensive than lubricated compressors and require more maintenance, they avoid the (usually intolerable) product contamination resulting from lube oil injected for lubricating the rubbing parts of the cylinder. If other considerations require the use of lubricated reciprocating compressors, then lube oil separation facilities must be provided on the compressor discharge.

Refrigeration Condenser - The condenser must remove heat equivalent to the various heat leaks into the system, the heat entering with the feed, and the work of compression. The condensing medium may be water, air, a combination of air/water, or another refrigeration system.

The condenser temperature affects both the required heat exchanger surface area and the required compressor horsepower. Lowering the condenser temperature increases the required surface area, but reduces the compressor horsepower. Conversely, raising the temperature lowers the required surface area but raises the horsepower requirements. The designer must select the condensing temperature to fit the economic guidelines for the project.

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For low boiling point materials, such as ethylene, it will probably be necessary to provide other, separate, refrigeration facilities to handle the condenser duty and minimize compressor work.

Accumulator - An accumulator drum is provided downstream of the condenser. Its purpose is to provide for an inventory of liquid refrigerant sufficient for stable refrigeration operations during changes in refrigeration operation and to purge non-condensibles from the refrigeration cycle. These non-condensibles which originate from startup of the new system, and which come in as impurities in the product or in the refrigerant make-up, would otherwise gradually accumulate and eventually overpressure the system. Material vented from this drum should be released to a safe location, preferably an elevated vent or flare stack. Generally, the refrigeration system is designed so that there is no need to continuously release inerts from the accumulator during normal operation.

TANKS

Vessel Foundations - The design of foundations for refrigerated storage vessels does not differ significantly from that for any other service and follows standard foundation practices. Spread footings are frequently used for spheres, while reinforced concrete ringwall foundations and piled foundations are used for vertical cylindrical tanks depending on the soil conditions. Piled foundations may also be used for spheres.

For tanks with ringwall foundations, the tank bottom will rest directly on the compacted soil tank pad. In order to prevent freezing and heaving of the soil, it may be necessary to provide heaters in the tank pad to maintain the soil temperatures above 32°F (0°C). The Civil Engineering Group within ExxonMobil Engineering - Technology Department (EMETD) should be consulted regarding the susceptibility of any given type of soil to heaving. Such heaving will progress continuously and eventually result in unstable support for the tank bottom.

Any heating medium may be used, although electric heaters are generally most favored, due to their ease of installation and control. With suitable insulation, the tank heat gain from the heaters will not exceed that of an equivalent area of the shell. Consideration should be given to installation of heaters within conduits to facilitate removal and repair as required. Where a piled foundation is used, the pile cap should be elevated 3 ft (90 cm) to allow for the circulation of air which eliminates the need for any additional heat source.

Wind loadings on the side of vertical cylindrical tanks, high internal gas pressure, and/or low liquid inventory can cause "uplift" on the shell sufficient to lift the weight of the roof and shell. To provide a reasonable factor of safety against uplift, anchor bolts are required to secure the vessel to the foundation. Consequently, the foundation must be designed to resist this force.

The foundation for a refrigerated vessel must also be protected against the thermal effects of a cold liquid spill. The Civil Engineering Group within ExxonMobil Engineering - Technology Department (EMETD) should be consulted for specific recommendations.

Typical foundation details for cylindrical tanks are shown in **GP 4-8-1**. Foundations for double-wall cylindrical tanks are similar.

The foundation insulation material for refrigerated tanks must, in addition to limiting heat leak into the tank, transmit the liquid load to the foundation. The most common insulations used are load bearing polyurethane, foamglass blocks, and perlite concrete. A layer of sand is usually furnished above and below the insulation, to provide bedding for the insulation and the tank bottom plate, and to prevent abrasion of the insulation. Where several layers of insulation material are required, it may be necessary, depending on the insulation employed, to provide a material, such as kraft paper, between the blocks to prevent abrasion.

In standard ringwall foundation designs, a vapor barrier is necessary immediately below the insulation to prevent migration of moisture into the insulation through the foundation. For an elevated piled foundation, an impervious membrane is not required, since the pile cap itself generally functions to prevent moisture from reaching the insulation.

Precast structural members or timbers surrounded by loose insulation are frequently used in foundation design. With this type of foundation, wood decking usually has to be used between the tank bottom and the insulation to provide uniform support for the tank bottom, since the loose fill will not support the load. Either treated dried wood or perlite concrete blocks are used directly under the tank shell. These materials combine the high load bearing capacity necessary to transmit the shell load to the foundation with their effectiveness as insulators.

Innage and Outage Allowances

- Outage** - Several factors must be considered by the designer to arrive at a safe allowable liquid level in any refrigerated vessel. These include:
 - Thermal expansion of liquid between minimum storage temperature and maximum operating temperature.
 - Tank gauging system accuracy and reliability.
 - Vessel fill rate.
 - The time required for emergency shutdown of the filling stream and consequently the equipment from which the product is received.
 - Local code requirements.

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Maximum inventory in any storage tank should be established from thermal expansion considerations, then reduced below this as dictated by operational/processing considerations and appropriate prudence. The primary concern is to prevent overfilling of the vessel, which can cause not only immediate structural damage but also the formation of a hydrocarbon vapor cloud from the vaporization of the cold liquid product.

Typical outage values have been 1 to 2 ft (30 to 60 cm) below the roof tangent line for refrigerated tanks and 10 to 15% of capacity for spheres and drums.

2. **Innage** - The minimum operating level is generally based on the following:

- Minimum safety stock to prevent runouts.
- Pump NPSH requirements.
- Pumpout rates.
- Gauging system.

Pump damage can result if the level is allowed to get low enough to create suction cavitation when the minimum NPSH requirement is not met. Typical innage values range from 18 to 24 in. (460 to 610 mm) for tanks. For spheres, the minimum levels should generally correspond to not less than 1% of the sphere volume.

Boiling Surface - Sufficient vapor space is required in refrigerated vessels with "open type" refrigeration systems to provide adequate surface for boil-off and to prevent liquid entrainment in the vapor drawoff. For additional information on entrainment of liquid droplets, see [Section V](#) on Drums. Usually the outage requirements for tanks are sufficient to satisfy the requirements to prevent liquid entrainment; however, drums and spheres may require special design features. For dome-roof tanks, the roof volume in itself is generally sufficient.

♦ **Piping Connections** - Piping connections to the inner tank of a single containment tank, consisting of an inner tank surrounded by an outer container, and all double and full containment tanks are welded. Shell connections are prohibited for double-wall tanks. Expansion joints are only used for secondary components. **GP9-6-1** provides additional guidance. If possible, tank drain connections should come off of the pump suction or other bottom line downstream of the tank EBV.

STRATIFICATION AND ROLLOVER

Refrigerated liquids of unequal density, due either to composition or temperature differences, may stratify in the storage tank. The upper layer will be in or near equilibrium with the vapor space above it while the lower layer(s) may not be in or near equilibrium with the vapor space in the tank as a result of being "capped" by the upper layer(s). The capping results because natural convection circulation is limited to within a layer of near equal density. This circulation limitation can prevent a lower heavier layer from releasing any heat contained in the feed or rundown to the tank or heat absorbed while stored in the tank.

"**Rollover**" may occur when new material is added to a refrigerated tank. The conditions leading to a rollover are initiated by stratifying the material added during the filling operation. Stratification may occur if the new material is top-filled and is less dense than the material in the tank (the heel) and sufficient mixing is not generated in the tank. Stratification may also occur if the new material is bottom-filled and is more dense than the tank heel and adequate mixing is not achieved. Once stratified, little or no natural mixing occurs between the stratified layers.

The conditions leading to a rollover can also be initiated by self-stratification of the refrigerated liquid tank. This mechanism is only of concern in LNG or similar material where some components, such as nitrogen, have higher volatility than components with lower molecular weight. Thus, when nitrogen boils off the LNG tank surface, the resulting liquid may be lighter in density than the remainder of the tank and will tend to collect near the surface. Over time, this mechanism may lead to a stratified tank.

Two adjacent layers can reach the same density over time as a result of compositional changes in the upper layer (it is continuously vaporizing the most volatile components into the vapor space), temperature changes in either the upper and/or lower layer resulting from a complex mass and heat transfer phenomenon between the upper layer and its vapor space above it, heat transfer between layers, and heat transfer between the wall(s) and/or tank bottom and the layer(s). If the lower layer becomes essentially equal in density to the top layer, an unstable equilibrium will develop. Any disturbance of this unstable equilibrium may result in vaporization of the lower layer, which had been suppressed by the hydrostatic head of the upper layer. This phenomenon is known as rollover.

The rollover phenomenon has occurred in various LNG tanks when a heavy high vapor pressure material was introduced into a tank containing a heel of light low vapor pressure material. There have also been unconfirmed reports of sudden vapor evolution in ammonia storage tanks. The potential for stratification and rollover also exists in LPG installations.

Rollover in refrigerated liquefied gas storage can be extremely serious due to the potentially high rate of vapor evolution that may occur. Although all refrigerated liquefied gas storage tanks are protected against overpressure by safety valves, these facilities cannot be relied on for protection against rollover since there are no currently approved methods to determine the maximum vapor evolution resulting from rollover to use as a sizing basis for the safety valves. Primary protection against rollover must therefore be to provide adequate facilities to ensure the conditions that create a rollover potential do not exist. Accordingly, installation of the following equipment should be considered to prevent stratification and potential rollover.

- A recirculation system to continuously circulate material from the bottom of the tank to the top of the tank.

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- A jet mixing nozzle on the feed or rundown to the tank designed and sized to mix the heaviest foreseeable feed at the lowest possible rate with the lightest foreseeable and largest size tank heel. Note that any contained superheat in the feed will cause vaporization when the feed is mixed with the existing tank heel and allowed to come to equilibrium at the pressure in the vapor space in the tank. Some of that vaporization will be shock condensed and cause noise (hammer) and vibration that is not desirable because of the potential for mechanical failure. To preclude or minimize the vaporization, condensation, noise, and vibration, the feed superheat should be removed before entry to the tank. This can be done by cooling the feed using refrigeration. It can also be done by flashing the tank feed in a pre-flash drum whose vapor space is connected to the tank vapor space and whose flash liquid is pumped into the tank through the aforementioned jet mixing nozzle.
- Where the input to the tank is expected to contain some vapor, a centrally located ring sparger should be used in lieu of a jet mixing nozzle.

The *Reactor and Fluid Dynamics Section* of ExxonMobil Engineering - Process Engineering Department and/or the *Safety and Risk Section* of the ExxonMobil Engineering -Technology Department should also be consulted for specific recommendations.

MATERIALS OF CONSTRUCTION

Considerations associated with materials for low-temperature applications, with the exception of oxygen, are covered in [Section II](#). All component materials for storage vessels subject to the subcooled temperatures must satisfy requirements of **GP 9-6-1**. Materials for equipment in the refrigeration system are covered by the applicable General Practice, e.g., **GP 3-10-1** for piping. Requirements for materials in an oxygen rich environment can be found in ASTM G 88, *Standard Guide for Designing Systems for Oxygen Service*.

Where multiple products are stored in a common facility, consideration should be given to making some or all tanks suitable for the colder/higher vapor pressure product than what they might normally handle. This provides flexibility for future changes in tankage allocations due to any maintenance or operational requirements.

PUMPS

Pumps should be specified in accordance with considerations present in [Section X](#). Generally, the pumps selected for refrigerated storage service are of the vertical centrifugal "deep well" type because of the low NPSH available with the product at or near its bubble point and the tank at minimum elevation above grade. Since the NPSH available is generally very low, it is very important that the NPSH available, which is specified, be carefully evaluated since pump type, model selection, and cost are very sensitive to variations in available NPSH in this low NPSH range.

Special attention should be given in layout with respect to length, size, and routing of suction piping. In particular, high points in suction lines must be avoided to prevent accumulation of vapor when the pumps are not operating. Vapor generated by heat leak should be allowed to migrate back to the storage tank. Pumps should be located as closely as possible to the storage vessel, but outside the firewalls.

Pumps should be vented back to the storage vessel to facilitate cool-down and pump starting. Pump vent lines should be insulated to reduce potential vapor generation if liquid filled. Vent lines that are vapor when the pump is running, such as seal balance lines, are often full of liquid on the standby or idle pump and these lines can add considerable vapor load.

Recirculation back to the tank will be necessary where the pumps may be operated at less than say 30% of design capacity, since pump heating of the pumped fluid will be significant and cavitation will rapidly occur in this situation.

An alternative pumping arrangement frequently used on double and full containment tanks is submersible pumps placed in a pump column within the tank. This arrangement eliminates the need for a pump suction nozzle on the tank shell close to the bottom of the tank, which is considered a potential source of a leak. This arrangement is generally not recommended due to concerns with support for the required tank internals to accommodate thermal expansion and pump accessibility for operational monitoring and maintenance. Where this arrangement is used, the reader is referred to Appendix E of ExxonMobil Engineering Report No. EE.49E.79 for a more detailed discussion of concerns and recommendations pertaining to submerged pumps.

SAFETY FACILITIES

Refer to [Section XV](#) for guidance on the design of safety facilities for refrigerated storage.

Safety Valves - Normally, safety valves installed on refrigerated storage tanks release to the atmosphere due to little tolerance for either superimposed or built-up backpressure. The tank design pressure and safety valve set pressure are only slightly above atmospheric pressure, typically 0.5 to 1.5 psig [6.9 to 10.3 kPa (g)]. Safety valves in other higher pressure segments of the refrigeration systems, however, can discharge to a safety valve header because backpressure is usually tolerable.

When storage tank safety valve releases directly to the atmosphere are not permitted, and a closed relief system is mandatory, the water seals of the relief system must be protected against freezing. This is normally accomplished with steam sparging under temperature control. Alternatively, a gas purge or dry seal per [Section XV](#) may replace the water seal. These latter seals have the advantage of imposing a relatively small backpressure on the tank safety valve.

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Pilot-operated safety valves with remote sensing are preferred for low set point pressures such as are required on storage tanks. Remote sensing for the pilot operated safety valve is necessary because the pressure drop limitation for inlet piping cannot usually be satisfied. However, actual pressure at the valve inlet should be used in determining valve area. Installed spares are recommended to permit periodic valve maintenance without reducing the relieving capacity of the system.

An alternative pressure relief device that is sometimes used is a P-V vent with a weighted pallet to achieve the correct pressure.

Relief valve capacity must be consistent with the single contingency that produces the largest relief requirements. Possible causes of overpressure or vacuum that should be considered in determining the valve capacity per API STD 2000, *Venting Atmospheric and Low-Pressure Storage Tanks - Nonrefrigerated and Refrigerated*, are:

- Liquid movement into and out of the tank.
- Atmospheric temperature and/or barometric changes.
- Fire exposure.
- Other circumstances resulting from equipment failures and/or operating errors.

Safety valve installations should include overhead chain hoists and trolley facilities to allow safety valve removal from the tank roof without the need for large cranes. Large crane use over refrigerated tankage adds unnecessary risk for the periodic testing and maintenance of safety valves.

Vacuum Protection - A vacuum may be created within a refrigerated tank in several ways. The product may be withdrawn at an excessive rate, such that natural boil-off does not accommodate the additional volume created by the falling tank level.

Secondly, the vapor space within the tank may be chilled too rapidly by cold feed. Thirdly, the vapor in the tank may be withdrawn excessively, due to a failure of the automatic controls. A rapid rise in barometric pressure could also cause vacuum conditions in a tank.

A vacuum breaker must be used as a last resort protection against a negative pressure. Other facilities are always provided to minimize the need for the vacuum breaker to function. The alternatives that are used alone or together include:

- Use of an external gas supply and associated pressure control system to maintain the tank at a positive pressure.
- The use of the stored material in conjunction with a vaporizer to maintain a positive tank pressure.
- The use of vapor return lines for product loading facilities.

In cases where refrigerated ships deliver product to a terminal, blowers are provided at the terminal to return to the ship some of the vapors generated by the filling operation. In this way, vacuum conditions are avoided on the ship.

Refer to API STD 2000 for procedures used in determining required overpressure/vacuum venting capacity for the above contingencies.

TOP AND BOTTOM FILLING

The refrigerant return connection for a refrigerated storage tank should be to the tank vapor space and it is frequently located 12 to 18 in. (305 to 460 mm) above the high liquid level. For large tanks, the return refrigerant is sprayed through a distribution ring to distribute the chilling of the vapor space.

It is generally recommended that the feed to a refrigerated storage tank from upstream process production rundown, or from receipts, be directed to the bottom of the storage tanks. Bottom filling is low in cost and is usually amenable to the use of a jet mixing nozzle. Where the feed to the tank may be mixed phase (liquid and vapor), a centrally located ring sparger should be used in lieu of a jet mixing nozzle.

When a tank is filled from the bottom, and the feed is superheated, there can be shock condensation of the vaporized feed, resulting in noise and vibration. This can be minimized by refrigerating the feed before entry to the tank or by flashing the feed in a pre-flash drum that is vented to the vapor space of the tank and whose flash liquid is pumped into the tank bottom through a jet mixing nozzle.

Where the feed or rundown rate to the tank with bottom filling is or can at times be low, the rate may be inadequate to provide the necessary mixing energy with a jet mixing nozzle. This can be overcome by the installation and use of an external tank recirculation system where the feed and recirculating stream are both fed through the same jet mixing nozzle.

Storage tanks with double or full containment are generally fed from the top to minimize tank shell connections below the maximum liquid level consistent with the desire to maximize the integrity of the storage containment. In addition, most double or full containment tanks (and some single containment tanks) are provided with submerged pumps, placed in a pump shaft, inside the tank near the tank bottom. This results in a tank shell without any pipe connections, so that the possibility of major leakage from a pipe to shell connection is eliminated.

If a tank is filled from the top, and the rundown or receipt feed is superheated, the rundown/feed may flash in the filling line, resulting in slug flow, noise, and vibration especially in the vertically oriented part of the filling line. While the vibration can be minimized by pipe bracing, the noise and residual vibration is undesirable. In addition, there is the potential for stratification

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and rollover because of the poor mixing of the flashed feed liquid and existing tank contents. This may require the installation and use of an internal recirculation system to reduce the risk of stratification and subsequent rollover to an acceptable level.

Slug flow can also cause flow instability and overloading of the boil-off gas compressor. All two-phase lines entering the top of the tank should be analyzed for slug flow.

In summary, for tanks with single containment, bottom filling should usually be employed. For tanks with double or full containment, top filling should usually be employed. For all tanks, consideration must be given to handling flashing feed and mixing feed with the existing tank contents.

VENTS

Vents are provided to avoid vapor locking the various sub-systems by vapors generated by heat leak. These vents should characteristically discharge to the tank vapor space via a common header. The individual vent lines, as well as the common header, should slope continuously up to the tank vapor space. Vents should be considered for the following sub-systems:

- Pump suction lines.
- Pump casings.
- Loading lines.
- Refrigerated rundown lines.

Manual vents are suitable for most of the sub-systems. Automatic vents are used where the primary purpose of the vent is to maintain the system cold. The automatic vent features a standpipe or drum with a level controller actuating a vent valve. High and low level alarms warn of any abnormalities. Where tank liquid level is insufficient to drive off vapors that are to be vented, a pressurizing or cool-down pump is required.

Pump suction and casing/stuffing box vent lines will often be full of liquid. If tied together in multiple pump installations, they could create bypasses around individual pump suction EBVs. Care should be exercised in the design of these vent lines to ensure that liquid flow will not bypass emergency isolation valves. In general, separate pumping services should have separate vent line headers.

VALVE FREEZEUP PROTECTION

As stated previously, liquefied hydrocarbon gases are dried to a 10 ppm water content to avoid freezups due to hydrate formation. Even with this low water content, there is the possibility that additional hydrate formation may occur in control valves in flashing service below 50°F (10°C). To restore movement in the event these valves plug, methanol injection connections should be provided immediately upstream and downstream of the valve, in the valve body, and in the valve bonnet.

CONTROLS AND INTERLOCKS

A typical, basic control and safety interlock scheme for a refrigeration system employing a reciprocating compressor is shown on [Figure 5](#). As described in [Section XI](#), flow control of reciprocating compressors is accomplished through discrete flow steps using suction unloaders. The unloaders for the first stage suction are regulated by a pressure controller located on the storage tank. The second stage unloaders are actuated by an interstage pressure controller located on the primary flash drum.

For centrifugal compressors, a common method of control is inlet throttling via a control valve in the compressor suction line controlling the tank pressure. This permits smooth load control between compressor design and surge points but only to the point where the compressor suction pressure is near atmospheric. The compressor is protected against surging by flow-controlled recycling of gas from the compressor discharge. The recycled gas must be cooled to maintain proper compressor operating temperature. In-house pressure monitoring should be provided at the lowest pressure point in the compressor system, often directly on the compressor cylinder and/or downstream of pulsation dampening orifices.

Safety interlocks consist of a series of high-level cutouts that interrupt feed and stop compressors on high level. The cutouts are located on the accumulator, primary flash drum, and storage tank. Tank high level alarm and/or cutout devices must be capable of on-line testing and removal for maintenance/repair. Tank high level alarms must consist of at least two different types of devices for redundancy.

COMMISSIONING

All necessary connections and requirements for commissioning the storage facility must be specified in the design specification because of the special features of a refrigerated storage system. A typical commissioning procedure (post tightness test) for an aboveground storage vessel is described below.

Dryout - All moisture must be removed to avoid ice formation during commissioning. The entire system is purged, usually with dry instrument air or preferably nitrogen, until the dew point is sufficiently low. In most cases, a dew point of -40°F to -50°F (-40°C to -46°C) is acceptable. If the system contains equipment that would be adversely affected by even small amounts of

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ice (e.g., plate fin exchangers), consideration must be given to attaining even lower dew points for the colder materials. Note that during dryout all low point drains and high point vents in the system piping must be purged.

Purging of Air - This is usually done with nitrogen. This should be continued until the oxygen concentration is well below the combustible limit of the storage material. In some situations, oxygen content of less than 1% may be warranted.

Purging of Nitrogen - This is usually done with warm product vapor.

Cool-down - The cool-down of equipment is done using the refrigeration compressors. The cold liquid is introduced in the storage vessel via the spray ring. The cool-down rate should proceed in accordance with the vessel vendor's recommendation so as not to expose the vessel to excessive stress. Note tank pad heaters should be commissioned before cool-down begins.

Refrigerated Oxygen - Commissioning of refrigerated oxygen systems present some unique hazards. ASTM documents such as ASTM G 128, *Standard Guide for Control of Hazards and Risks in Oxygen Enriched Systems*, should be consulted.

DESIGN PROCEDURE

The design of refrigerated storage facilities involves selecting the type of refrigerated storage vessel(s), the method of refrigeration to be employed, and the number and sizes of storage vessels for each service. When that has been done, duty specification type information is prepared for each vessel and its accessories recognizing that specialist vendors design those equipment items. The refrigeration system, however, is frequently specified in much more detail and specification sheets are usually prepared for all of the components of the refrigeration system with the exception of the instruments.

The purpose of this section is not to provide step-by-step detailed instruction for preparing all of the duty or design specification information required for all of the components of a refrigerated storage system. Instead, this section covers the system as a whole and design of individual components is treated elsewhere; for example:

Component	Section
Tanks	DP XXII-B, Atmospheric Storage, or DP XXII-C, Pressure Storage
Drums	DP V, Drums
Exchangers	DP IX, Heat Exchange Equipment
Pumps	DP X, Pumps
Compressors	DP XI, Compressors
Instruments	DP XII, Instrumentation
Piping	DP XIV, Fluid Flow
Safety Facilities	DP XV, Safety in Plant Design
Insulation	DP XVI, Thermal Insulation

A computer program may be used for preparing the various heat and material balances and for describing the stream properties required for the design of individual components of the refrigerated storage system. The program will normally have to be run several times to optimize the compressor horsepower considering the following variables:

- Condenser temperature.
- Interstage pressure.
- Insulation thickness on tanks and cold lines.

The results of the computer runs will provide data from which to size lines and other equipment items. The results will also set the mechanical requirements (temperature, pressure) of equipment. Therefore, variations in feed properties must also be covered in the analysis and they must include the entire range of possible variations, such as other than normal operation, upstream upsets, possible impurities, etc.

Heat gain is one of the primary inputs for the computer simulation. All of the heat gains mentioned earlier in this section should be included. Where equipment size is unknown, estimates should be made for the initial runs and updated later for final runs. As with any computer calculation, the results should be compared with those of other calculation/estimation techniques for reasonableness and to better understand primary influences on the design of the facility.

An illustration of the basic computer calculation operations involved and the organization of these operations are presented in the following Sample Problem and [Figure 7](#). A two-stage refrigeration system is assumed, since it is a relatively common type. The simulation represents the following equipment items as vapor/liquid flashes.

Flash No. 1	Interstage flash drum
Flash No. 2.	Storage tank
Flash No. 3.	Condenser/accumulator drum

Heat gains are input for the following equipment and/or operation:

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Storage Tank

Product loading operations

Tankage heat gain

Heat gains in lines, pumps, etc.

Tank or line cooldown

Interstage Flash Drum

Feed

The refrigeration compressor is divided into two distinct compressor operations with a split function combining vapor at the interstage point. If a centrifugal compressor were being simulated, the split function would not be required since interstage feed can be added directly to the centrifugal compressor operation in the computer simulation.

Note that the arrangement shown on [Figure 7](#) that is used for the Sample Problem does not necessarily represent some preferred arrangement. It is shown to illustrate calculations for a Sample Problem. Other arrangements should be considered as well, such as top filling of refrigerant return, bottom filling of feed to tank, flashing tank feed at the tank pressure, etc.

Required stream specifications at various points in the system are indicated on [Figure 7](#), along with typical stream identification names in parentheses. Input and output data are shown in the [Appendix](#).

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SAMPLE PROBLEM

Given: Refrigerated storage system depicted in [Figure 7](#). The feed composition is as follows:
(dimensions of feed composition must be consistent with the feed rate dimensions):

Component	LV%
C ₂	2.7
C ₃	94.6
iC ₄	1.7
C ₄	<u>1.0</u>
	100.0

Other feed conditions:

Feed rate:	2,820	B/SD
Feed temperature:	100	°F
Total heat leak:	200,000	Btu/h
Cooling water temperature:	80	°F

Find: Compressor horsepower, condenser duty, and stream properties for the system. Condenser duty is not calculated directly, but is obtained by difference in the heat contents of entering and leaving streams of the FLASH 3 calculation.

Solution: The calculation procedure for defining the system involves using the computer simulation program. Estimates are made of the compressor second stage discharge pressure and overall heat leakage for the initial run. The discharge pressure is altered in subsequent runs until the accumulator vent stream approaches a value of zero.

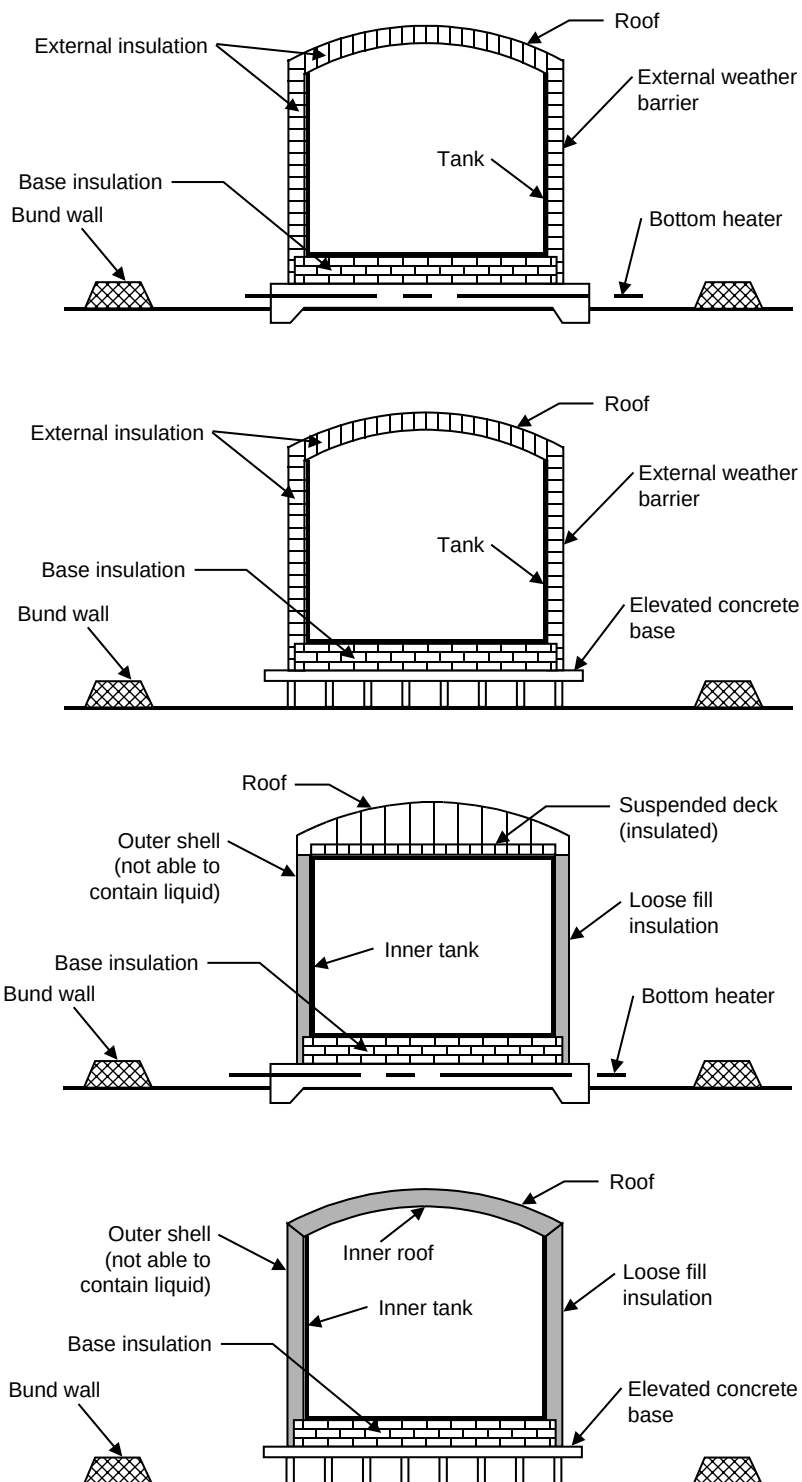
The forms that define the sample problem comprise the [APPENDIX](#) of this section. For the most part, the forms are self-explanatory. However, it is useful to know that the program adjusts the recycle stream rate in converging. Therefore, if the program does not converge during the initial computer run, the second run should include the recycle stream composition data computed from the initial run, but the stream flow rate should be set higher. These data are added to the Stream Information Card. Condenser temperature (FLASH 3) should be selected consistent with cooling water temperature to avoid a temperature cross in the condenser.

The results of the computer run are shown on the computer printouts immediately following the input forms. The printouts contained herein begin with the last iteration of the convergence operation. As shown, thermodynamic properties, flow rates, and other physical data are printed out for every stream identified in [Figure 7](#).

STORAGE FACILITIES/SYSTEMS

REFRIGERATED STORAGE

Figure 1 Examples of Single Containment Tanks



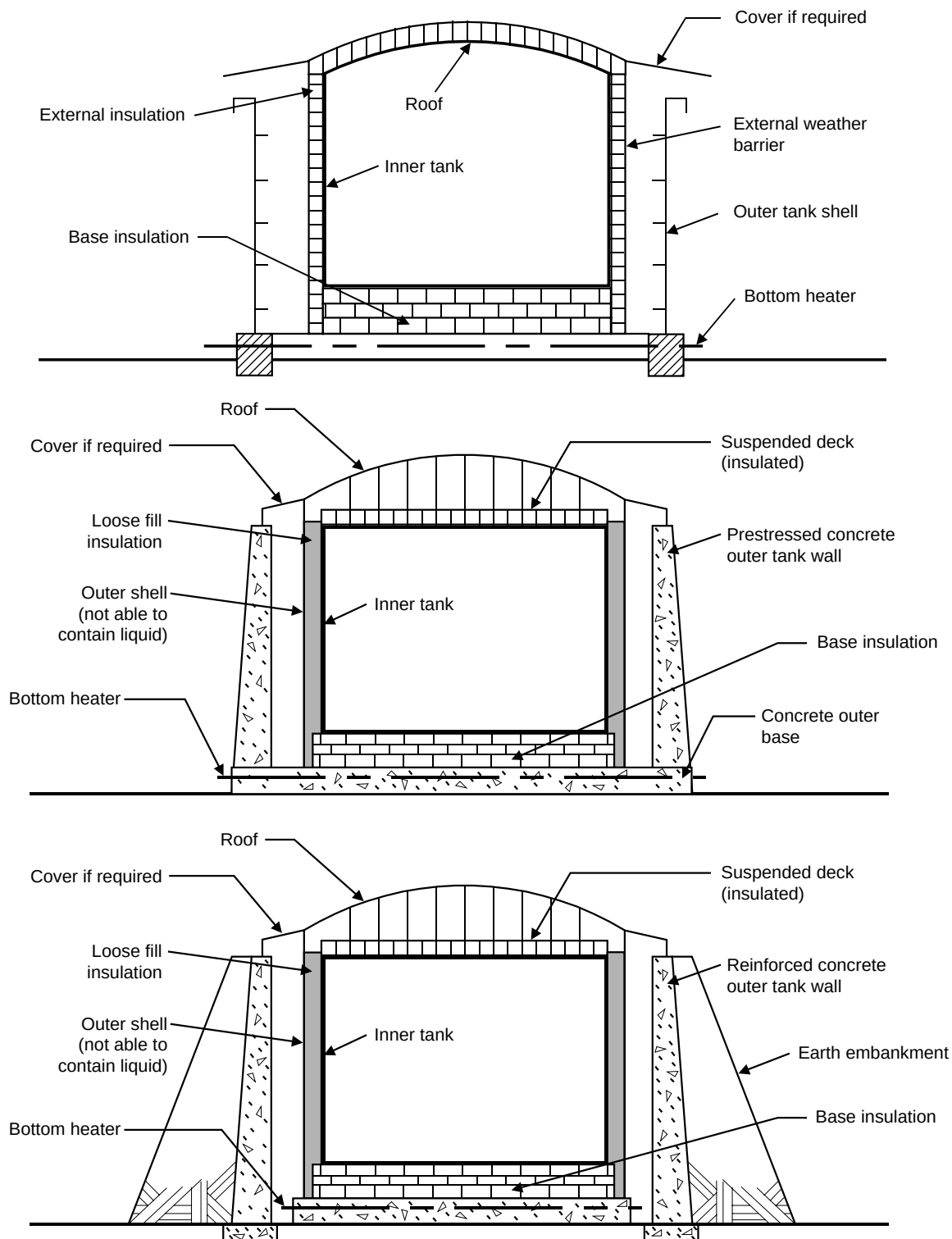
DP22D11

Source: BS 7777, Part 1

STORAGE FACILITIES/SYSTEMS

REFRIGERATED STORAGE

Figure 2 Examples of Double Containment Tanks



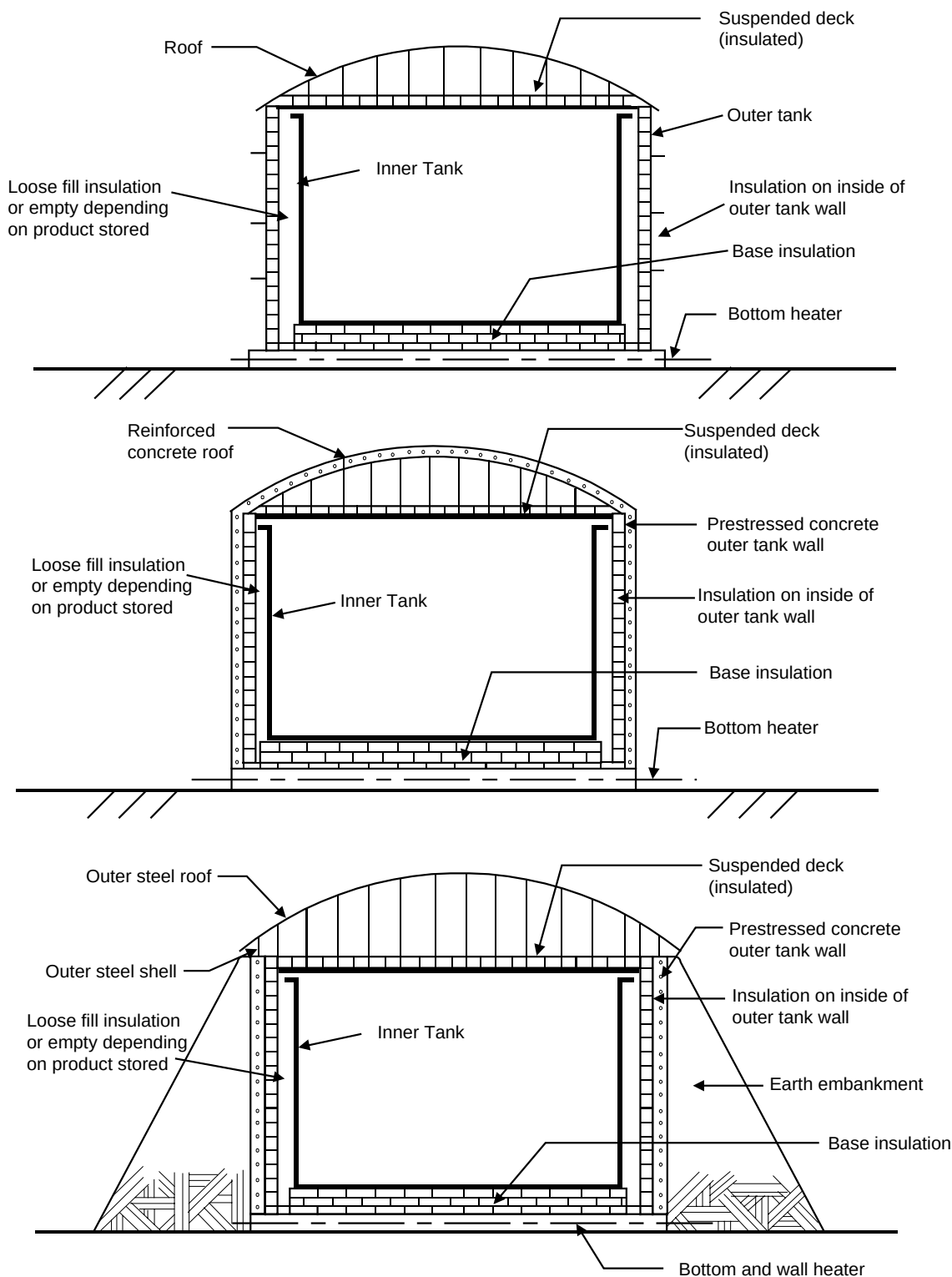
DP22DF2

Source: BS 7777, Part 1

STORAGE FACILITIES/SYSTEMS

REFRIGERATED STORAGE

Figure 3 Examples of Full containment Tanks

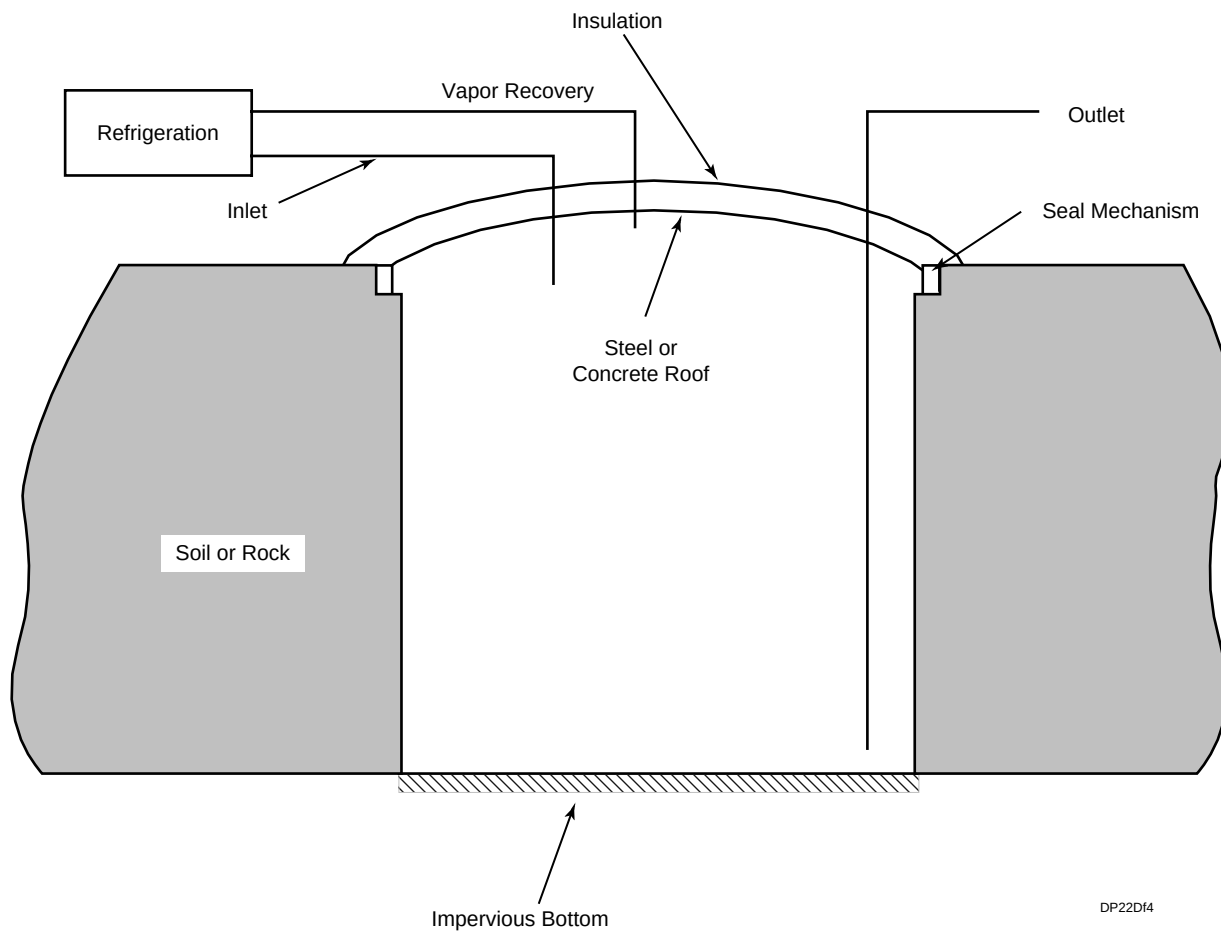


DP22DF3

Source: BS 7777, Part 1

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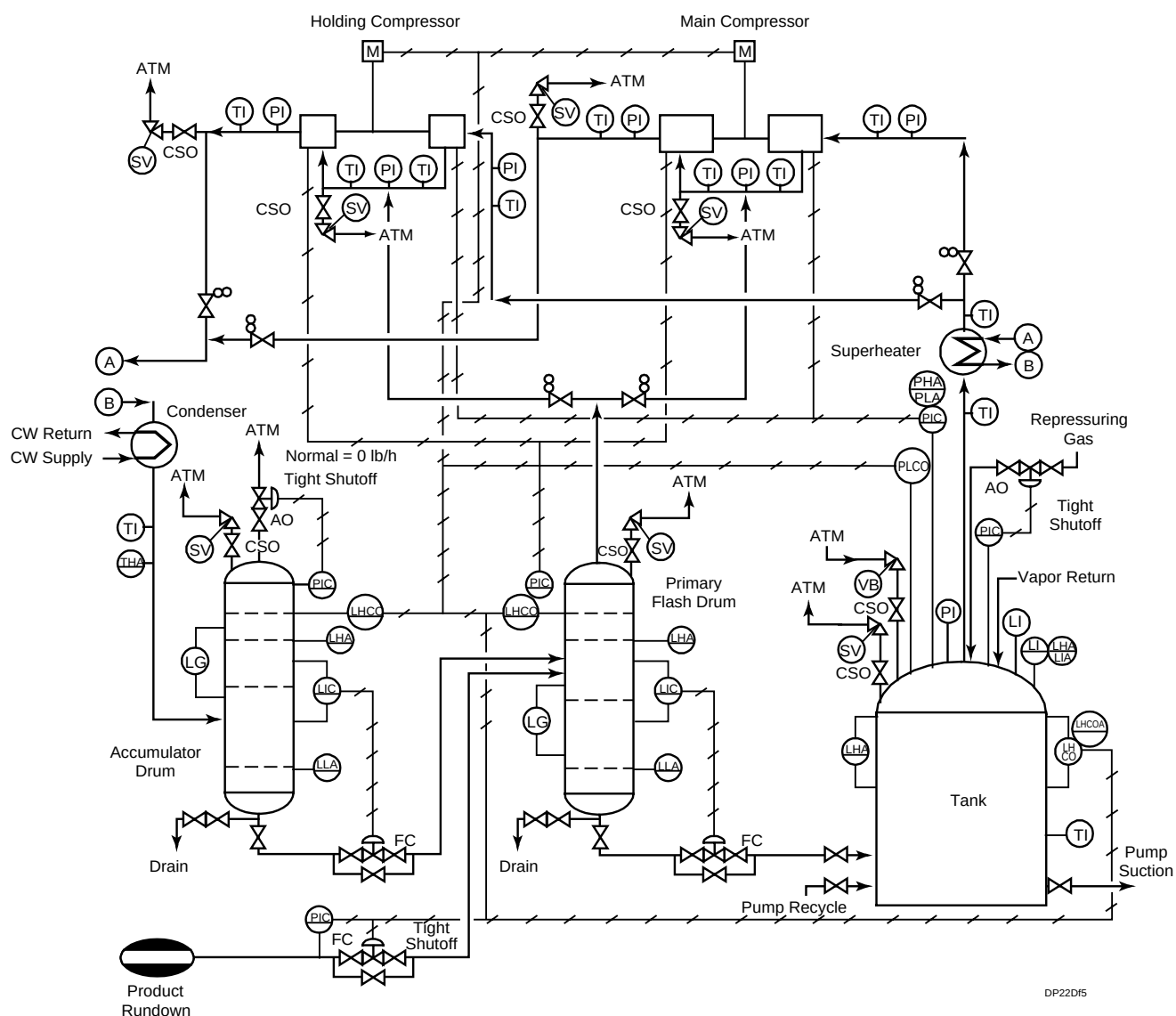
Figure 4 InGround Refrigerated Storage



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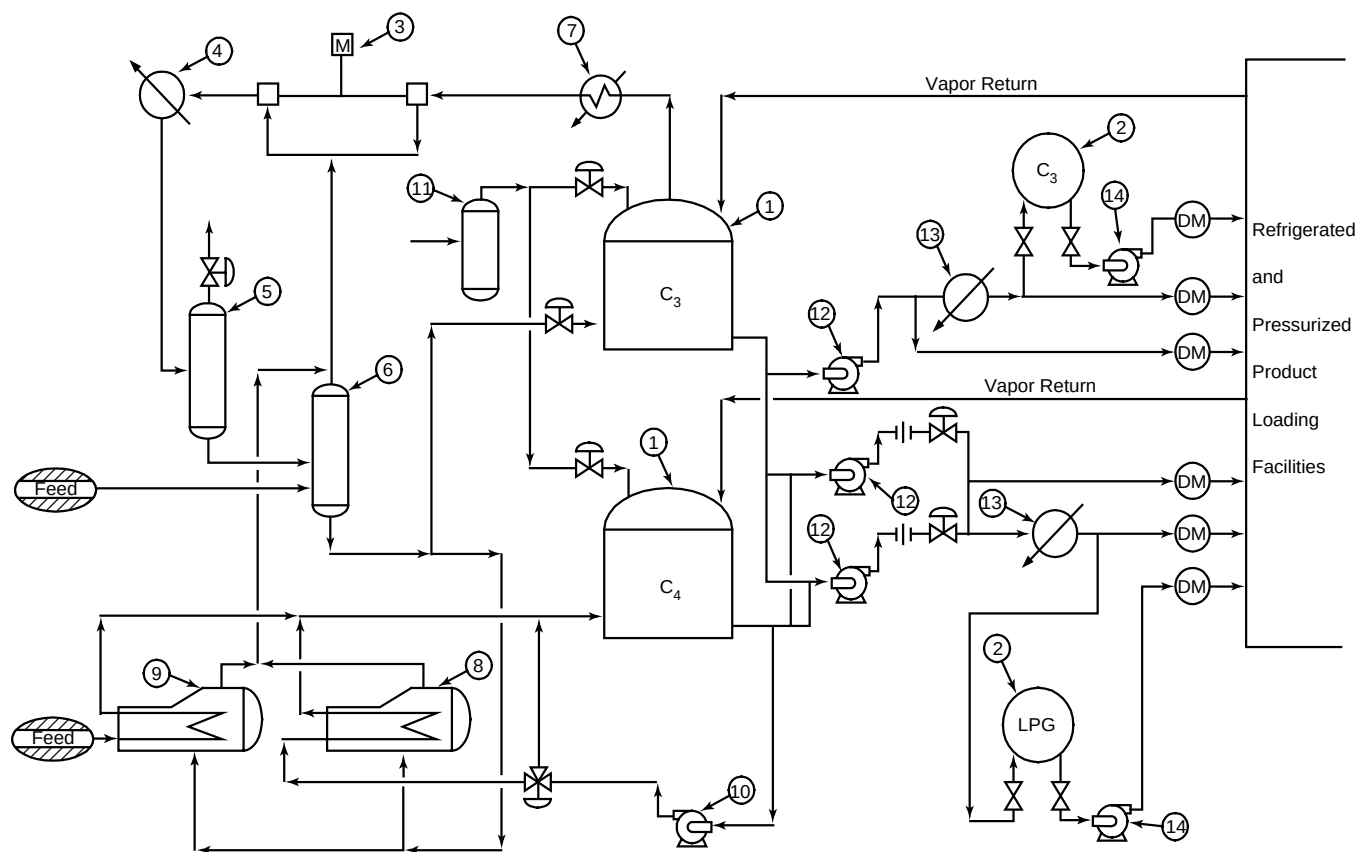
Figure 5 Typical Refrigerated Storage Facility With Vapor Compression Autorefrigeration System



STORAGE FACILITIES/SYSTEMS

REFRIGERATED STORAGE

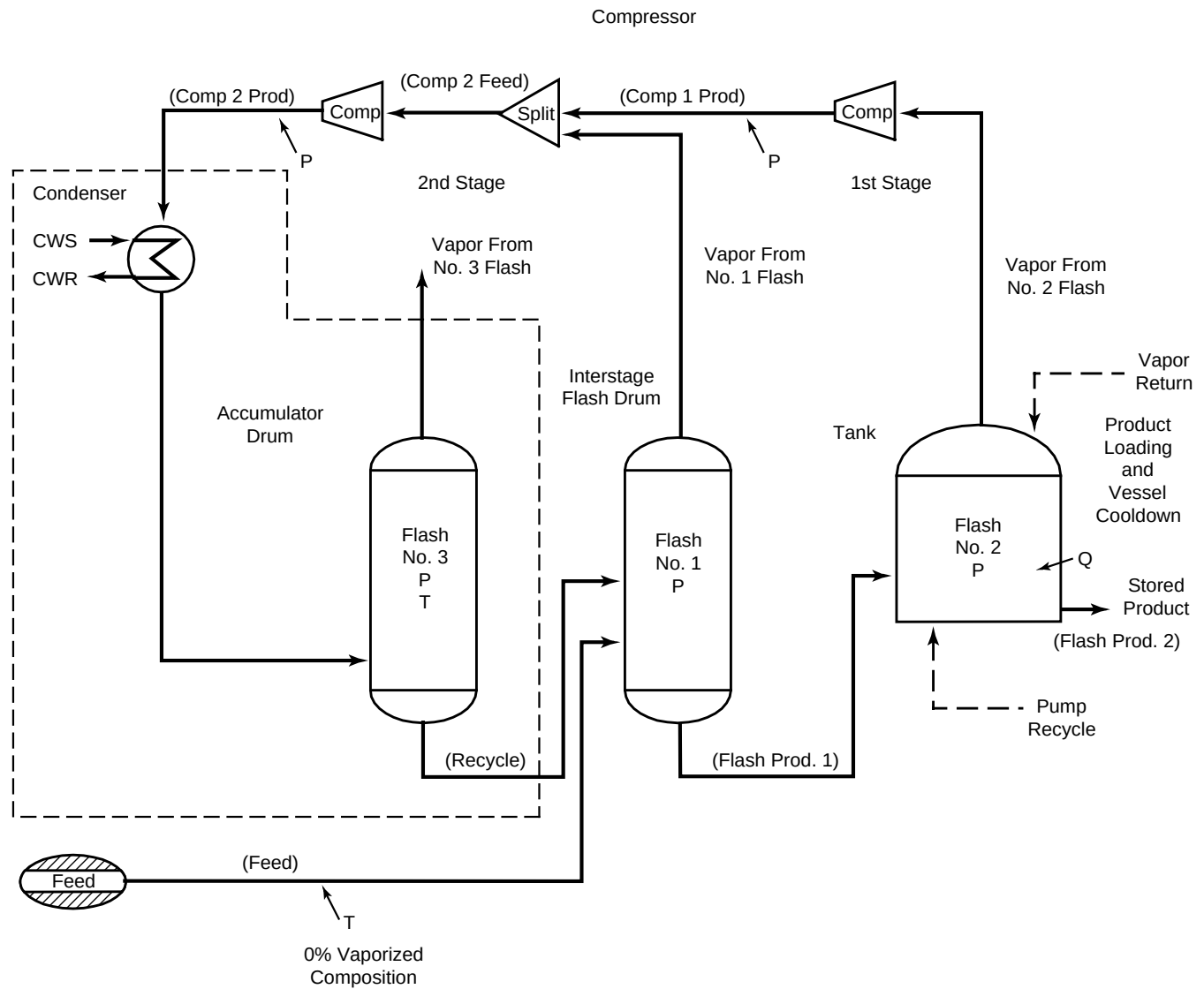
Figure 6 Schematic Diagram of an Integrated Refrigeration System

**LEGEND**

- | | | |
|-----------------------------|----------------------|---------------------------|
| ① Refrigerated Storage | ⑥ Flash Drum | ⑪ Pressurization Gas Drum |
| ② Pressure Storage | ⑦ Superheater | ⑫ Loading/Transfer Pumps |
| ③ Refrigeration Compressors | ⑧ Pumparound Chiller | ⑬ Product Heaters |
| ④ Refrigeration Condenser | ⑨ Rundown Chiller | ⑭ Loading Pumps |
| ⑤ Accumulator | ⑩ Pumparound Pumps | ⑮ Displacement Meters |

DP22Df6

Figure 7 Simulation Diagram for Refrigerated Storage System



Legend

T = Temperature
P = Pressure
Q = Heat Input
 ΔP = Pressure Drop

DP22Df7

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APPENDIX SAMPLE PROBLEM USING PRO / II

\$ Generated by PRO / II Keyword Generation System <version 2.50 - 12-15-94>

\$ Generated on: Fri Mar 14 12:27:01 1997

TITLE

This is the Sample Problem, in English Units
for DP XXII-D, Refrigerated Storage
Compressor Horsepower can be found on Pages P-5 and P-6
Condenser Duty can be found on Page P-3, in the Accumulator Drum Column

PRINT INPUT=PART, STREAM=ALL, RATE=LV, FRACTION=LV, PERCENT=LV, LV
DIMENSION ENGLISH, LIQVOL=GAL
OUTDIMENSION REPLACE, ENGLISH, TIME=DAY, LIQVOL=BBL, DUTY=BTU/HR
SEQUENCE SIMSCI

COMPONENT DATA

LIBID 1,ETHANE/2,PROPANE/3,BUTANE/4,2MPROPANE

THERMODYNAMIC DATA

METHOD SYSTEM=DCUBE, TRANSPORT=EDL, SET=DCUB01, DEFAULT

STREAM DATA

PROPERTY STREAM=FEED, TEMPERATURE=100, PHASE=L, RATE(LV)=4934.99,
COMPOSITION(LV)=1,2.7/2,94.6/3,1/4,1.7
PROPERTY STREAM=RECYCLE, TEMPERATURE=95, PHASE=L, RATE(LV)=4934.99,
COMPOSITION(M)=1,20/2,80
NAME FEED,Feed/RECYCLE,Recycle/FLASHPROD1,Flash 1 Product/ &
FLASHPROD2,Stored Product/VAPFLASH2,Vapor From No. 2 Flash/ &
COMP1PROD,Compressor 1 Product/ &
VAPFLASH1,Vapor from No. 1 Flash/COMP2FEED,Compressor 2 Feed/ &
COMP2PROD,Compressor 2 Product/ &
VAPFLASH3,Vapor from No. 3 Flash

UNIT OPERATIONS

COMPRESSOR UID=C1, NAME=1st Stage Compressor
FEED VAPFLASH2
PRODUCT M=COMP1PROD
OPERATION CALCULATION=GPSA, EFF=80
DEFINE PRES(Psia) AS FLASH=Fl, PRES(Psia)
COMPRESSOR UID=C2, NAME=Compressor - 2nd Stage
FMM COMP2FEED
PRODUCT V=COMP2PROD
OPERATION CALCULATION=GPSA, PRES=262, EFF=80
FLASH UID=Fl, NAME=Interstage Flash Drum
FEED FEED,RECYCLE
PRODUCT L=FLASHPROD1, V=VAPFLASH1
ADIABATIC TESTIMATE=-25.001, PRESSURE=25
FLASH UID=F2, NAME=Tank
FEED FLASHPROD1
PRODUCT L=FLASHPROD2, V=VAPFLASH2
ADIABATIC TESTIMATE=-47.999, PRESSURE=15.2, DUTY=0.2
FLASH UID=F3, NAME=Accumulator Drum
FEED COMP2PROD
PRODUCT L=RECYCLE, V=VAPFLASH3
ISO TEMPERATURE=95
MIXER UID=M1, NAME=Mixer
FEED VAPFLASH1,COMP1PROD
PRODUCT M=COMP2FEED

END

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*** PROBLEM SOLUTION BEGINS

FEED FLASH COMPLETE

LOOP	1	BEGINS	- 'LOOP1	'
UNIT	1	SOLVED	- 'F1	'
UNIT	2	SOLVED	- 'F2	'
UNIT	3	SOLVED	- 'C1	'
UNIT	5	SOLVED	- 'M1	'
UNIT	4	SOLVED	- 'C2	'
UNIT	6	SOLVED	- 'F3	'

MAX ERRORS: X=.00535 (FLASHPROD2) T=-.0193 (FLASHPROD1)
P=0.

LOOP 1 SOLVED AFTER 1 TRIALS

*** PROBLEM SOLUTION REACHED

*** SIMSCI ROYALTY FOR THIS PROBLEM IS 38.50 PRO / II CHARGE UNITS

*** RUN STATISTICS

STARTED	12:26:41	03/14/97	NO ERRORS
FINISHED	12:26:47	03/14/97	NO WARNINGS
RUN TIMES			NO MESSAGES
INTERACTIVE	0 MIN,	2.47 SEC	
CALCULATIONS	0 MIN,	3.35 SEC	
TOTAL	0 MIN,	5.82 SEC	

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2	CALCULATION SEQUENCE AND RECYCLES
3	FLASH DRUM SUMMARY
4	MIXER SUMMARY
	COMPRESSOR SUMMARY
5	UNIT 3, 'C1', '1st Stage Compressor'
6	UNIT 4, 'C2', 'Compressor - 2nd Stage'
7	STREAM DRY LV COMPONENT RATES
9	STREAM SUMMARY

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COMPONENT DATA
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	COMPONENT	COMP. TYPE	PHASE	MOL. WEIGHT	API
1	ETHANE	LIBRARY	VAP/LIQ	30.069	265.526
2	PROPANE	LIBRARY	VAP/LIQ	44.096	147.208
3	BUTANE	LIBRARY	VAP/LIQ	58.123	110.629
4	2MPROPANE	LIBRARY	VAP/LIQ	58.123	119.788

	COMPONENT	NBP F	CRIT. TEMP. F	CRIT. PRES. PSIA	CRIT. VOLM. GAL/LB-MOL
1	ETHANE	-127.480	89.924	706.479	17.5972
2	PROPANE	-43.744	206.060	615.975	23.9635
3	BUTANE	31.082	305.618	550.563	30.3444
4	2MPROPANE	11.066	274.460	527.937	30.6560

	COMPONENT	ACEN. FACT.	HEAT FORM. BTU/LB-MOL	GFORM. BTU/LB-MOL
1	ETHANE	.09890	-36027.52	-13682.72
2	PROPANE	.15210	-45012.90	-10450.13
3	BUTANE	.19950	-53998.28	-7037.83
4	2MPROPANE	.18510	-57695.61	-8876.61

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CALCULATION SEQUENCE

SEQ	UNIT ID	UNIT TYPE	SEQ	UNIT ID	UNIT TYPE
1	F1	FLASH	4	M1	MIXER
2	F2	FLASH	5	C2	COMPRESSOR
3	C1	COMPRESSOR	6	F3	FLASH

RECYCLE LOOPS

LOOPS			TEAR STREAMS		
ID	FIRST UNIT	LAST UNIT	ID	FROM UNIT	TO UNIT
LOOP1	F1	F3	RECYCLE	F3	F1

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FLASH DRUM SUMMARIES
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FLASH ID	F1	F2	F3
NAME	Interstage Flash Drum	Tank	Accumulator Drum
FEEDS	FEED RECYCLE	FLASHPROD1	COMP2PROD
PRODUCTS MIXED			VAPFLASH3
VAPOR	VAPFLASH1	VAPFLASH2	
LIQUID	FLASHPROD1	FLASHPROD2	RECYCLE
TEMPERATURE, F	-28.522	-47.532	95.000
PRESSURE, PSIA	25.000	15.200	262.000
PRESSURE DROP, PSI	177.077	9.800	.000
MOLE FRAC VAPOR	.44962	.10794	.00000
MOLE FRAC LIQUID	.55038	.89206	1.00000
DUTY, MM BTU/HR	.00000	.20000	-3.36947
FLASH TYPE	ADIABATIC-P	ADIABATIC-P	ISOTHERMAL

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MIXER SUMMARY

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```
=====
MIXER  ID                      Ml
      NAME                      Mixer
      FEEDS                     VAPFLASH1
                                COMPlPROD
```

```

      PRODUCTS VAPOR           COMP2FEED
```

```

TEMPERATURE, F                 -26.329
PRESSURE, PSIA                 25.000
PRESSURE DROP, PSI             .000
MOLE FRAC VAPOR                1.00000
MOLE FRAC LIQUID               .00000
```

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COMPRESSOR SUMMARY
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UNIT 3, 'Cl', '1st Stage Compressor'

FEEDS VAPFLASH2

PRODUCTS VAPOR COMPLPROD

OPERATING CONDITIONS

	INLET	ISENTROPIC	OUTLET
	-----	-----	-----
TEMPERATURE, F	-47.53	-16.63	-10.14
PRESSURE, PSIA	15.20	25.00	25.00
ENTHALPY, MM BTU/HR	-2.6072	-2.5841	-2.5783
ENTROPY, BTU/LB-MOL-F	-64.3636	-64.3635	-64.1363
CP, BTU/LB-MOL-F	14.5556		15.6822
CV, BTU/LB-MOL-F	12.2353		13.2653
CP/(CP-R)	1.1580		1.1450
CP/CV	1.1896		1.1822
MOLE PERCENT VAPOR	100.0000	100.0000	100.0000
MOLE PERCENT LIQUID	.0000	.0000	.0000
ACT VAP RATE, M FT3/MIN	.2665		
ADIABATIC EFF, PERCENT			80.0000
POLYTROPIC EFF, PERCENT			80.8147
ISENTROPIC COEFFICIENT, K			1.1733
POLYTROPIC COEFFICIENT, N			1.2236
HEAD, FT			
ADIABATIC			7501.05
POLYTROPIC			7577.44
ACTUAL			9376.31
WORK, HP			
THEORETICAL			9.07
POLYTROPIC			9.16
ACTUAL			11.34

NOTE: POLYTROPIC AND ISENTROPIC COEFFICIENTS
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COMPRESSOR SUMMARY

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UNIT 4, 'C2', 'Compressor - 2nd Stage'

FEEDS COMP2FEED

PRODUCTS VAPOR COMP2PROD

OPERATING CONDITIONS

	INLET	ISENTROPIC	OUTLET
	-----	-----	-----
TEMPERATURE, F	-26.33	143.70	167.07
PRESSURE, PSIA	25.00	262.00	262.00
ENTHALPY, MM BTU/HR	-22.0357	-20.9975	-20.7379
ENTROPY, BTU/LB-MOL-F	-63.6760	-63.6757	-62.8124
CP, BTU/LB-MOL-F	15.1722		22.5761
CV, BTU/LB-MOL-F	12.7018		18.0310
CP/(CP-R)	1.1506		1.0964
CP/CV	1.1945		1.2521
MOLE PERCENT VAPOR	100.0000	100.0000	100.0000
MOLE PERCENT LIQUID	.0000	.0000	.0000
ACT VAP RATE, M FT3/MIN	1.4385	.2665	
ADIABATIC EFF, PERCENT			80.0000
POLYTROPIC EFF, PERCENT			84.3763
ISENTROPIC COEFFICIENT, K			1.1842
POLYTROPIC COEFFICIENT, N			1.2260
HEAD, FT			
ADIABATIC			39888.89
POLYTROPIC			42070.97
ACTUAL			49861.11
WORK, HP			
THEORETICAL			408.04
POLYTROPIC			430.36
ACTUAL			510.05

NOTE: POLYTROPIC AND ISENTROPIC COEFFICIENTS
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STREAM DRY LV COMPONENT RATES
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```

STREAM	ID	COMPLPROD	COMP2FEED	COMP2PROD	FEED
	NAME	Compressor 1	Compressor 2	Compressor 2	Feed
		Product	Feed	Product	
	PHASE	VAPOR	VAPOR	VAPOR	LIQUID

FLUID RATES, BBL/DAY

1	ETHANE	52.0307	553.3161	553.3161	76.1399
2	PROPANE	284.6214	2328.3200	2328.3200	2667.7150
3	BUTANE	.4579	3.9223	3.9223	28.1999
4	2MPROPANE	1.3096	10.9386	10.9386	47.9399

TOTAL RATE, BBL/DAY	338.4196	2896.4970	2896.4970	2819.9950
---------------------	----------	-----------	-----------	-----------

TEMPERATURE, F	-10.1435	-26.3287	167.0746	99.9999
PRESSURE, PSIA	25.0000	25.0000	262.0000	202.0770
ENTHALPY, MM BTU/HR	-2.5783	-22.0357	-20.7379	-24.2249
MOLECULAR WEIGHT	41.9464	41.4119	41.4119	44.0293
MOLE FRAC VAPOR	1.0000	1.0000	1.0000	.0000
MOLE FRAC LIQUID	.0000	.0000	.0000	1.0000

STREAM	ID	FLASHPROD1	FLASHPROD2	RECYCLE	VAPFLASH1
	NAME	Flash 1	Stored	Recycle	Vapor from
		Product	Product		No. 1 Flash
	PHASE	LIQUID	LIQUID	LIQUID	VAPOR

FLUID RATES, BBL/DAY

1	ETHANE	125.4063	73.3759	553.3161	501.2854
2	PROPANE	2951.4710	2666.8500	2328.3200	2043.6990
3	BUTANE	28.6595	28.2016	3.9223	3.4643
4	2MPROPANE	49.2531	47.9434	10.9386	9.6290

TOTAL RATE, BBL/DAY	3154.7900	2816.3700	2896.4970	2558.0770
---------------------	-----------	-----------	-----------	-----------

TEMPERATURE, F	-28.5215	-47.5316	95.0000	-28.5215
PRESSURE, PSIA	25.0000	15.2000	262.0000	25.0000
ENTHALPY, MM BTU/HR	-28.8485	-26.0413	-24.1074	-19.4574
MOLECULAR WEIGHT	43.8171	44.0434	41.4119	41.3413
MOLE FRAC VAPOR	.0000	.0000	.0000	1.0000
MOLE FRAC LIQUID	1.0000	1.0000	1.0000	.0000

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STREAM DRY LV COMPONENT RATES
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=====

STREAM ID	VAPFLASH2
NAME	Vapor From
	No. 2 Flash
PHASE	VAPOR

FLUID RATES, BBL/DAY

1	ETHANE	52.0307
2	PROPANE	284.6214
3	BUTANE	.4579
4	2MPROPANE	1.3096

TOTAL RATE, BBL/DAY	338.4196
---------------------	----------

TEMPERATURE, F	-47.5316
PRESSURE, PSIA	15.2000
ENTHALPY, MM BTU/HR	-2.6072
MOLECULAR WEIGHT	41.9464
MOLE FRAC VAPOR	1.0000
MOLE FRAC LIQUID	.0000

THE FOLLOWING ZERO-RATE STREAMS OR STREAMS WITH
ZERO TEMPERATURE OR PRESSURE WERE NOT PRINTED
VAPFLASH3

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STREAM SUMMARY
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=====

STREAM ID	COMP1PROD	COMP2FEED	COMP2PROD	FEED
NAME	Compressor 1	Compressor 2	Compressor 2	Feed
PHASE	Product	Feed	Product	
	VAPOR	VAPOR	VAPOR	LIQUID
----- TOTAL STREAM -----				
RATE, LB-MOL/DAY	1369.469	11734.130	11734.130	11332.990
MLB/DAY	57.444	485.933	485.933	498.983
STD LIQ RATE, BBL/DAY	338.420	2896.497	2896.497	2819.995
TEMPERATURE, F	-10.143	-26.329	167.075	100.000
PRESSURE, PSIA	25.000	25.000	262.000	202.077
MOLECULAR WEIGHT	41.946	41.412	41.412	44.029
ENTHALPY, MM BTU/HR	-2.578	-22.036	-20.738	-24.225
BTU/LB	-1077.205	-1088.333	-1024.236	-1165.163
MOLE FRACTION LIQUID	.0000	.0000	.0000	1.0000
REDUCED TEMP (KAYS RULE)	.6940	.6736	.9742	.8424
PRES (KAYS RULE)	.0397	.0395	.4137	.3277
ACENTRIC FACTOR	.1439	.1418	.1418	.1515
WATSON K (UOPK)	15.231	15.369	15.369	14.762
STD LIQ DENSITY, LB/BBL	169.743	167.766	167.766	176.945
SPECIFIC GRAVITY	.4848	.4791	.4791	.5053
API GRAVITY	160.399	163.839	163.839	148.519
----- VAPOR -----				
RATE LB-MOL/DAY	1369.469	11734.130	11734.130	N/A
LB/DAY	57.444	485.933	485.933	N/A
FT3/DAY	251.998	2071.412	247.380	N/A
STD VAP RATE(1), M FT3/DAY	519.692	4452.919	4452.919	N/A
SPECIFIC GRAVITY (AIR=1.0)	1.448	1.430	1.430	N/A
MOLECULAR WEIGHT	41.946	41.412	41.412	N/A
ENTHALPY, BTU/LB	-1077.205	-1088.333	-1024.236	N/A
CP, BTU/LB-F	.374	.366	.545	N/A
DENSITY, LB/M FT3	227.955	234.590	1964.310	N/A
Z (FROM DENSITY)	.9536	.9490	.8212	N/A
TH COND, BTU/HR-FT-F	.00797	.00750	.01600	N/A
VISCOSITY, CP	.00701	.00679	.01019	N/A

(1) STANDARD VAPOR VOLUME IS 379.49 FT3/LB-MOLE (60 F AND 14.696 PSIA)

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STREAM SUMMARY

LICENSED EXCLUSIVELY FOR USE BY: EXXON Corporation
SPECIAL VERSION WITH EDL III
VERSION NUMBER 1.7

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=====
STREAM  ID          COMP1PROD  COMP2FEED  COMP2PROD  FEED
        NAME        Compressor 1 Compressor 2 Compressor 2  Feed
                        Product      Feed      Product      LIQUID
                        VAPOR        VAPOR     VAPOR
        PHASE

----- LIQUID -----
RATE,  LB-MOL/DAY      N/A        N/A        N/A        11332.990
        M LB/DAY        N/A        N/A        N/A        498.983
        BBL/DAY        N/A        N/A        N/A        3020.991
        GAL/MIN        N/A        N/A        N/A        88.112
STD LIQ RATE, BBL/DAY  N/A        N/A        N/A        2819.995
SPECIFIC GRAVITY (H2O=1.0) N/A        N/A        N/A        .5053
MOLECULAR WEIGHT      N/A        N/A        N/A        44.029
ENTHALPY, BTU/LB      N/A        N/A        N/A        -1165.163
CP, BTU/LB-F          N/A        N/A        N/A        .686
DENSITY, LB/BBL       N/A        N/A        N/A        165.172
Z (FROM DENSITY)      N/A        N/A        N/A        .0504
SURFACE TENSION, DYNE/CM N/A        N/A        N/A        5.2933
THERMAL COND, BTU/HR-FT-F N/A        N/A        N/A        .05250
VISCOSITY, CP         N/A        N/A        N/A        .08766

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STREAM SUMMARY

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=====
STREAM ID          FLASHPROD1  FLASHPROD2  RECYCLE      VAPFLASH1
NAME              Flash 1      Stored      Recycle      Vapor from
                  Product    Product
                  LIQUID    LIQUID      LIQUID      No. 1 Flash
                  PHASE      LIQUID      LIQUID      VAPOR

----- TOTAL STREAM -----
RATE, LB-MOL/DAY      12687.520    11318.050    11734.130    10364.660
M LB/DAY              555.930     498.485     485.933     428.489
STD LIQ RATE, BBL/DAY 3154.790    2816.371    2896.497    2558.077
TEMPERATURE, F        -28.522     -47.532     95.000     -28.522
PRESSURE, PSIA        25.000     15.200     262.000     25.000
MOLECULAR WEIGHT      43.817     44.043     41.412     41.341
ENTHALPY, MM BTU/HR   -28.849     -26.041     -24.107     -19.457
BTU/LB                -1245.416   -1253.780   -1190.652   -1089.824
MOLE FRACTION LIQUID  1.0000     1.0000     1.0000     .0000
REDUCED TEMP (KAYS RULE) .6506     .6203     .8622     .6708
PRES (KAYS RULE)      .0405     .0247     .4137     .0394
ACENTRIC FACTOR       .1507     .1516     .1418     .1416
WATSON K (UOPK)       14.808     14.759     15.369     15.387
STD LIQ DENSITY, LB/BBL 176.218    176.996    167.766    167.504
SPECIFIC GRAVITY      .5032     .5055     .4791     .4784
API GRAVITY           149.674    148.438    163.839    164.300

----- VAPOR -----
RATE, LB-MOL/DAY      N/A         N/A         N/A         10364.660
LB/DAY                N/A         N/A         N/A         428.489
FT3/DAY               N/A         N/A         N/A         1819.093
STD VAP RATE(1), M FT3/DAY N/A         N/A         N/A         3933.227
SPECIFIC GRAVITY (AIR=1.0) N/A         N/A         N/A         1.427
MOLECULAR WEIGHT      N/A         N/A         N/A         41.341
ENTHALPY, BTU/LB      N/A         N/A         N/A         1089.824
CP, BTU/LB-F          N/A         N/A         N/A         .365
DENSITY, LB/M FT3     N/A         N/A         N/A         235.550
Z (FROM DENSITY)      N/A         N/A         N/A         .9483
TH COND BTU/HR-FT-F   N/A         N/A         N/A         .00743
VISCOSIW CP           N/A         N/A         N/A         .00676

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(1) STANDARD VAPOR VOLUME IS 379.49 FT3/LB-MOLE (60 F AND 14.696 PSIA)

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STREAM SUMMARY
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=====

STREAM ID	FLASHPROD1	FLASHPROD2	RECYCLE	VAPFLASH1
NAME	Flash 1	Stored	Recycle	Vapor from
PHASE	Product	Product		No. 1 Flash
	LIQUID	LIQUID	LIQUID	VAPOR
----- LIQUID -----				
RATE, LB-MOL/DAY	12687.520	11318.050	11734.130	N/A
M LB/DAY	555.930	498.485	485.933	N/A
BBL/DAY	2784.078	2438.763	3041.469	N/A
GAL/MIN	81.202	71.131	88.709	N/A
STD LIQ RATE, BBL/DAY	3154.790	2816.371	2896.497	N/A
SPECIFIC GRAVITY (H2O=1.0)	.5032	.5055	.4791	N/A
MOLECULAR WEIGHT	43.817	44.043	41.412	N/A
ENTHALPY, BTU/LB	-1245.416	-1253.780	-1190.652	N/A
CP, BTU/LB-F	.550	.536	.711	N/A
DENSITY, LB/BBL	199.682	204.401	159.769	N/A
Z (FROM DENSITY)	6.6569E-03	4.1577E-03	.0641	N/A
SURFACE TENSION, DYNE/CM	14.1524	15.6165	4.5120	N/A
THERMAL COND, BTU/HR-FT-F	.07563	.07967	.05166	N/A
VISCOSITY, CP	.18552	.20901	.07901	N/A

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STREAM SUMMARY
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STREAM	ID	VAPFLASH2
	NAME	Vapor From
		No. 2 Flash
	PHASE	VAPOR

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----- TOTAL STREAM -----
```

RATE,	LB-MOL/DAY	1369.469
	M LB/DAY	57.444
STD LIQ RATE,	BBL/DAY	338.420
TEMPERATURE,	F	-47.532
PRESSURE,	PSIA	15.200
MOLECULAR WEIGHT		41.946
ENTHALPY,	MM BTU/HR	-2.607
	BTU/LB	-1089.264
MOLE FRACTION LIQUID		.0000
REDUCED TEMP (KAYS RULE)		.6363
PRES (KAYS RULE)		.0241
ACENTRIC FACTOR		.1439
WATSON K (UOPK)		15.231
STD LIQ DENSITY,	LB/BBL	169.743
	SPECIFIC GRAVITY	.4848
	API GRAVITY	160.399

```
----- VAPOR -----
```

RATE,	LB-MOL/DAY	1369.469
	LB/DAY	57.444
	FT3/DAY	383.738
STD VAP RATE(1),	M FT3/DAY	519.692
SPECIFIC GRAVITY (AIR=1.0)		1.448
MOLECULAR WEIGHT		41.946
ENTHALPY,	BTU/LB	1089.264
CP,	BTU/LB-F	.347
DENSITY,	LB/M FT3	149.697
Z (FROM DENSITY)		.9630
TH COND,	BTU/HR-FT-F	.00673
VISCOSITY,	CP	.00641

(1) STANDARD VAPOR VOLUME IS 379.49 FT3/LB-MOLE (60 F AND 14.696 PSIA)

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STREAM SUMMARY

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=====
STREAM ID          VAPFLASH2
NAME              Vapor From
                  No. 2 Flash
PHASE             VAPOR

```

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----- LIQUID -----
RATE, LB-MOL/DAY      N/A
M LB/DAY              N/A
BBL/DAY               N/A
GAL/MIN               N/A
STD LIQ RATE, BBL/DAY N/A
SPECIFIC GRAVITY (H2O=1.0) N/A
MOLECULAR WEIGHT      N/A
ENTHALPY, BTU/LB      N/A
CP, BTU/LB-F          N/A
DENSITY, LB/BBL       N/A
Z (FROM DENSITY)      N/A
SURFACE TENSION, DYNE/CM N/A
THERMAL COND, BTU/HR-FT-F N/A
VISCOSITY, CP         N/A

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THE FOLLOWING ZERO-RATE STREAMS OR STREAMS WITH
ZERO TEMPERATURE OR PRESSURE WERE NOT PRINTED
VAPFLASH3