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HIGH-PRESSURE GAS AND CRYOGENIC SYSTEMS

DEPARTMENTS OF THE ARMY AND THE AIR FORCE

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HIGH-PRESSURE GAS AND CRYOGENIC SYSTEMS

*This manual supersedes TM 5-810-7/AFM 88-12, Chap 4, 3 May 1966.

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CHAPTER 1
INTRODUCTION

1-1. Purpose and scope.

a. This manual provides guidance applicable to the design of high-pressure gas and cryogenic systems. The intent is to supply the user with a relevant and current list of references to handbooks and codes and to highlight the design considerations. Where possible, data and guidance are offered in tabular form for easy access. Examples are specifically excluded so that the designer will be forced to use the latest versions of the cited references which contain, among other things, detailed solutions to standard problems displaying limitations and caveats. A reference list is given in appendix A. Trade names for various products have been used in the manual to afford the designer the descriptive information generally associated with such names. A table equating trade and generic names is shown in appendix B. A selected bibliography is also provided in appendix C.

b. For the purposes of this manual high-pressure gases are taken to be those with pressures generally greater than 1000 psi. However, for completeness, some data will be offered pertaining to pressures less than that value.

1-2. Typical systems. For ease of understanding, two block diagrams representing the elements found in the system under consideration are offered in figures 1-1 and 1-2. The first displays the high-pressure gas systems while the second employs cryogenic fluids. The chapters in the manual develop the various elements shown.

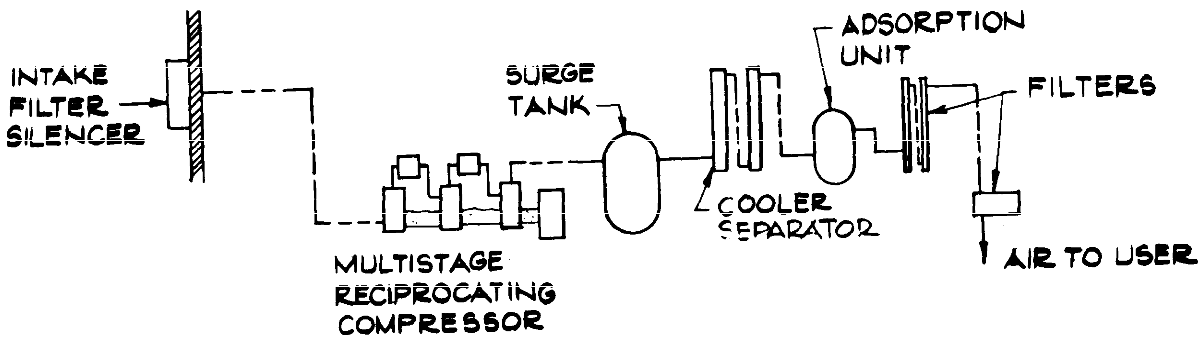

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Figure 1-1. Example of a high-pressure air system.

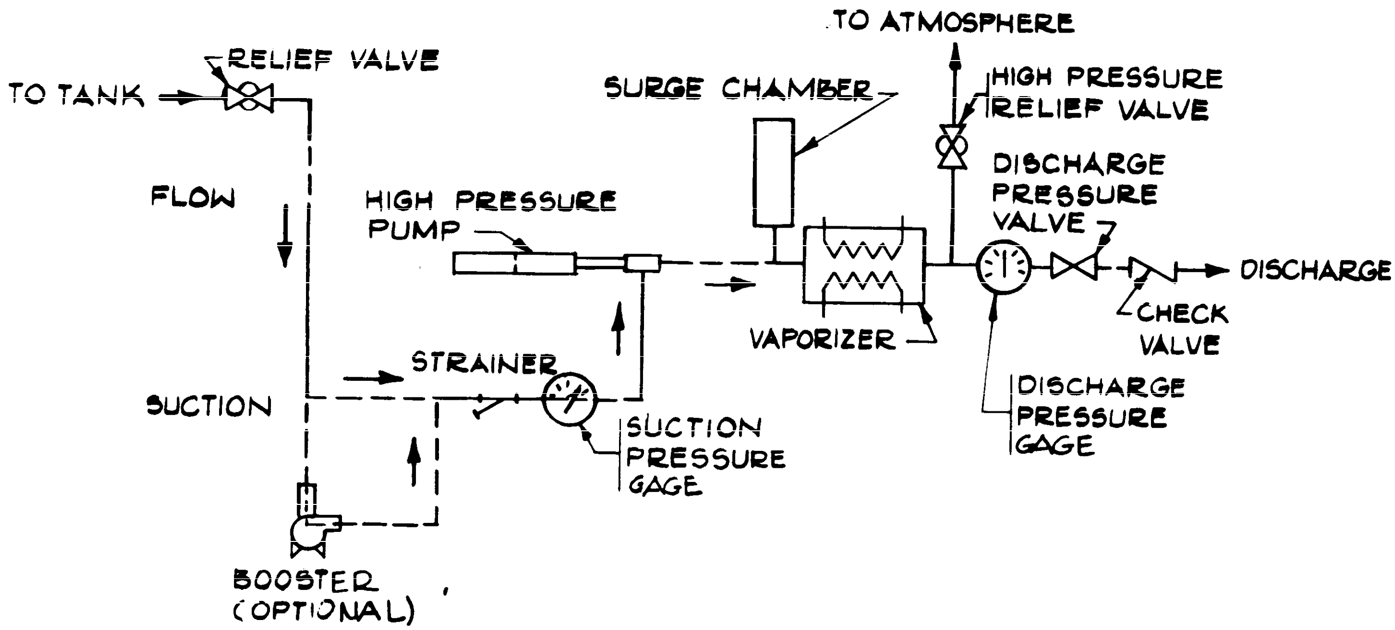

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Figure 1-2. Cryogenic liquid to high-pressure gas conversion system.

CHAPTER 2
FLUID PROPERTIES

2-1. General. This chapter briefly discusses sources of basic data for the fluids used in the high-pressure gas and cryogenic systems. Such data precedes the discussion of the system components to familiarize the designer with fluid properties.

2-2. Definition of cryogenics. Refrigeration in the ordinary sense and cryogenics are difficult to separate. The National Bureau of Standards considers the field of cryogenics to be that involving temperatures below −240°F (220°R) or −150°C (123°K). This definition of cryogenics will be used in this manual. The so-called permanent gases, helium (He), hydrogen (H), nitrogen (N), and oxygen (O), boil at temperatures lower than −240°F.

2-3. Fluid properties.

a. A great variety of sources of fluid property data exist. After the working fluid is selected, the designer will obtain thermodynamic property data. For most fluids, *Marks Standard Handbook for Mechanical Engineers* and the *Piping Handbook* can offer first cuts at such data and can lead the designer to additional reference material. Property data generally required are:

—Conductivity
—Critical point properties
—Density
—Entropy
—Gas law applicability
—Internal energy
—Mixture properties of the working fluid, with for example, water vapor
—Molecular weight
—Ratio of specific heats
—Saturated state properties
—Specific heat at constant pressure
—Specific volume
—Viscosity

b. It should be noted that the above data often come in one of several formats: tabular, equations (derived from curve fits of tables), or chart form. The designer will often find one or more of these convenient to use, depending on the required calculations. The designer should also be aware that standard thermodynamic texts can be used effectively for some data access, but more importantly, for terminology problems and sample calculations.

c. Cryogenic fluids commonly used in the application considered here and some of their properties are displayed in table 2-1. Naturally, the designer will also need many of the same properties cited in paragraph 2-3*a*. Again, *Marks Standard Handbook for Mechanical Engineers* and *Piping Handbook* may be used, and for cryogenics, specifically, property data may be obtained in part from *Cryogenic Systems.*

Table 2-1. Selected Properties of Cryogenic Liquids at the Normal Boiling Point.

Saturated-liquid property at 1 atm	Liquid helium 3	Liquid helium 4	Liquid hydrogen	Liquid neon	Liquid nitrogen	Liquid air	Liquid flourine	Liquid argon	Liquid oxygen	Liquid methane
Normal boiling point, °R	5.74	7.57	36.7	48.8	139.2	142	153.6	157.1	162.4	201.1
Critical temperature, °R	5.08	9.36	59.74	79.9	227.0	240	212.7	271.2	278.3	343.3
Critical pressure, atm	1.15	2.26	12.98	26.84	33.5	38.7	55.0	48.3	50.1	45.8
Triple-point temperature, °R	...	...*	25.1	34.21	113.7	...	96.37	150.8	98.0	159.7
Pressure at triple point, atm	...	...†	0.0711	0.426	0.1268	...	0.00218	0.679	0.00150	0.099
Density, lb_m/ft^3	3.68	7.80	4.43	75.17	50.61	54.56	93.96	87.56	70.8	26.53
Heat of vaporization, Btu/lb_m	3.65	8.92	191.9	37.26	85.32	88.2	71.6	69.5	91.63	219.2
Specific heat c_p, Btu/lb_m-°R	1.10	1.09	2.336	0.44	0.487	0.470	0.37	0.272	0.406	0.825
Viscosity, lb_m/ft-hr	0.00392	0.00864	0.0316	0.30	0.382	0.195	0.592	0.610	0.455	0.287
Thermal conductivity, Btu/hr-ft-°F	0.0099	0.01565	0.0683	0.075	0.0804	...	0.078	0.0712	0.0855	0.0642
Dielectric constant	...	1.0492	1.226	...	1.434	...	1.43	1.52	1.4837	1.6758
Speed of sound, fps	376	593	3900	...	2810	...	...	2780	2960	4630

* Lambda-point temperature = 3.91°R.
† Lambda-point pressure = 0.0497 atm.

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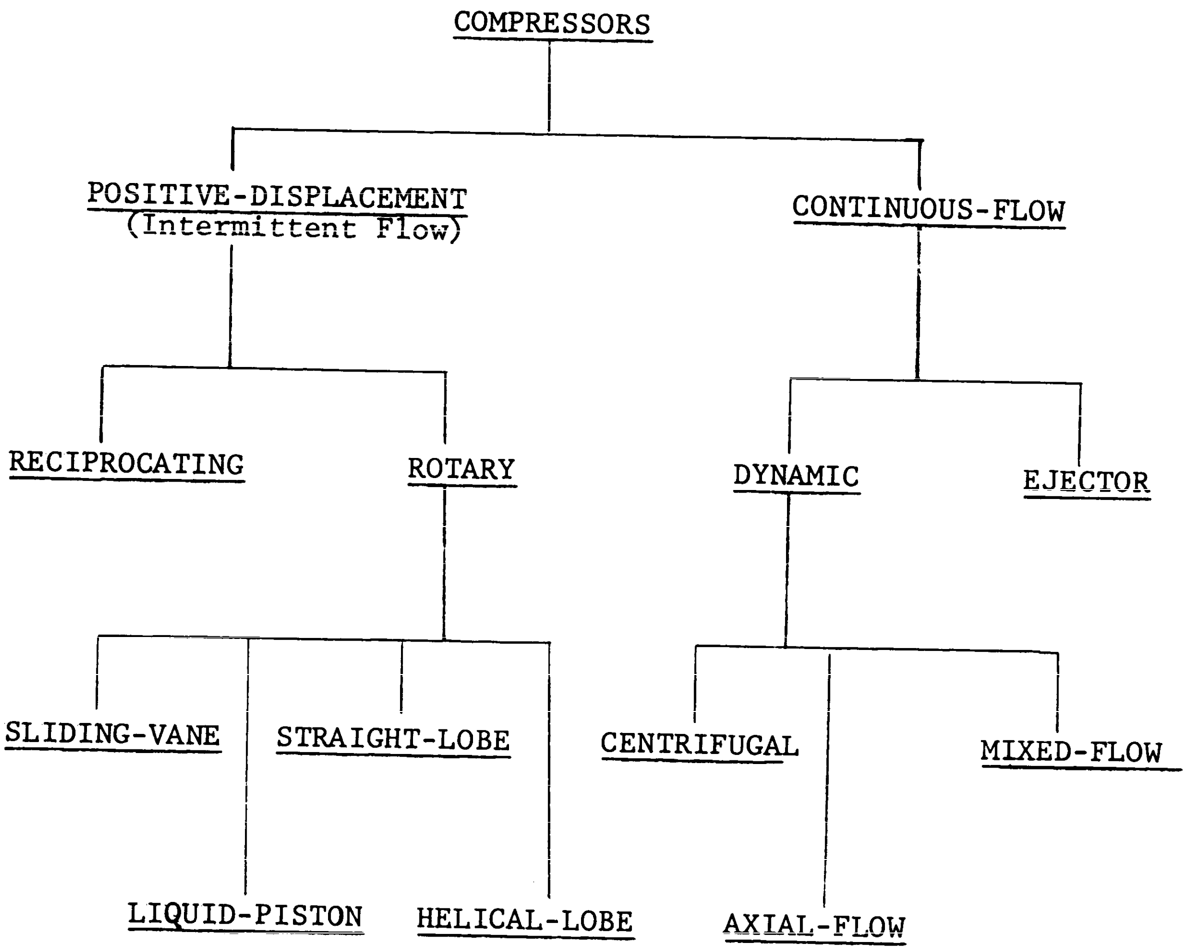
CHAPTER 3
COMPRESSORS, PUMPS, AND RELATED SYSTEMS

Section I. COMPRESSORS FOR HIGH-PRESSURE GAS SYSTEMS

3-1. General. There are several types of compressors that should be considered in high-pressure gas systems. Table 3-1 shows the maximum pressure obtainable with each type and the maximum power required. *Marks Standard Handbook* and *Compressed Air and Gas Data* offer complete discussions of compressor operation. The following paragraphs describe the types that produce high pressures, calculation suggestions for power required, and the compatible power sources. Of the five types listed in table 3-1, only the reciprocating and centrifugal are capable of producing pressures greater than 1000 psi. These two will be discussed in some detail with brief additional notes about a Lysholm type that can be used as a preliminary stage, and a diaphragm compressor that can be used for low-volume, high-pressure application or as a booster stage for high-volume, high-pressure application.

Table 3-1. Compressor Types and Approximate Maximum Pressure



Compressor Type	Approx. Max. BHP	Approx. Max. Pressure-psig
Reciprocating	Over 12,000	100,000
Vane Type Rotary	(Twin Unit) 860	400
Helical Lobe Rotary	6,000	250
Centrifugal Dynamic	Over 35,000	5,500
Axial Flow Dynamic	Over 100,000	500

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3-2. Reciprocating compressor.

a. The multistage, reciprocating compressor is generally the first choice to furnish high-pressure gases. The horizontal, tandem style is the simplest and least costly and is used for pressures to 5000 psig and powers to 150 hp. The next power step, to 500 hp, is met by a radialcylinder type with all stages operating from a single-throw crank. The largest machines, with pressures to 15,000 psig, power to 10,000 hp, and rotating speeds from 150 to 450 rpm, are "horizontally opposed"; a multi-throw crankshaft operates all stages from a common crankcase. Large capacity is obtained by two or more first-stage cylinders.

b. For continuous service, first consideration should be given to machines that are water-cooled, heavy-duty, long life, low maintenance types. Lighter, higher speed, air-cooled machines are acceptable for special purposes in capacities from 15 to 100 scfm and pressures to 5000 psig. These machines are radially or horizontally opposed types and suitable for mobile, intermittent duty in which weight and space conservation are more important than frequency of maintenance work or accessibility.

c. The total work of compression is usually divided equally between stages, the number of stages being determined by considerations of the maximum allowable compression temperature. A 250°F compression-temperature limitation is typical for all air compressors. Higher temperature limits are used for the compression of stable gases free of oxygen.

d. Nonlubricated compressors are available and should be considered where oil-free air is desired. The cost of removing oil from air compressed by a lubricated compressor must be balanced against the extra costs of a non-lubricated compressor. Such extra costs include higher initial cost, higher power cost, higher maintenance cost, and more downtime for maintenance. Oil removal is specifically discussed in section III.

3-3. High-pressure centrifugal compressor.

a. Table 3-2 shows the types of centrifugal compressors with their capabilities and power requirements. The multi-stage horizontally split unit will reach 1000 psig, but the most common high-pressure centrifugal compressors are those with the vertically split case. The more recently developed high-pressure centrifugal compressors are designed for pressures to 5000 psig, although service at half this pressure is more representative of current installations. A discharge capacity of about 200 cfm limits the use of the high-pressure centrifugal compressor because the compressor becomes less efficient as the clearance volume approaches the volume of the gas being compressed.

b. The most difficult design problem of the high-pressure centrifugal compressor is that of the shaft seal which controls gas leakage between points of unequal pressure and prevents the lubricant from contaminating the gas and vice versa. Several current designs are in use; they are selected according to the shaft speed, pressure differentials, and kind of gas being compressed. The designer should therefore investigate the different types: the labyrinth, the oil-film, and contact or carbon ring seals.

c. One method of producing high-pressure air that generally has been overlooked, but should be considered by the designer, consists of combining lower stage centrifugal- or rotating-type compression with reciprocating higher pressure stages. For head pressures over 3000 psig, an air capacity of 20,000 scfm would be required to utilize centrifugal compression for three stages. For air capacities of less than 20,000 scfm, a rotary compressor of the Lysholm type (see para 3-5*a*) may be used for the lower stages. Both combinations have the advantages of lower weight, smaller space, and good efficiency in comparison with an all-reciprocating machine for large horsepower installation.

d. For both structural and power-economy reasons, the discharge temperature is a design limitation on the compression ratio. A typical pressure ratio is 2.5/1 or less, and the number of impellers required depends on the design shaft speed and impeller diameter, which determine peripheral velocity. From pressure and weight considerations, a small-diameter case is advantageous because it tends to increase the number of wheels employed for a given compression ratio.

e. Fixed guide vanes are provided at the inlet of each impeller to establish the correct velocity vector of the gas flow for the design wheel speed. A capacity change or large temperature change will affect the pressure of a pipe system and alter this vector. A correction can be made by changing the shaft revolutions per minute if the driver, such as a turbine, has a variable speed; for the constant-speed driver, adjustable guide vanes are employed to vary capacity efficiently with demand changes. The operating point relative to the surge point may also be controlled in this manner.

3-4. Diaphragm-type compressor.

a. The diaphragm compressor will process all types of gases or liquids to 15,000 psig; however, maximum capacities are relatively low (between 5 and 60 cfm) except when the machine is used with high intake pressure as a booster. The large diameter of the diaphragm compared to its small stroke allows oil to cool the gas under compression to a degree approaching the isothermal case; consequently, delivery pressures to 250 psi per stage from an ambient intake are realized. The higher overall pressures are obtained by staging.

b. An important design advantage is that the gas does not come in contact with any lubricant (unless the diaphragm should leak) in its path through the machine. Oxygen compressors are specially designed so that oil is used in the crankcase only, and water operates the diaphragm. The gas under compression cannot leak from the machine since there is no stuffing box or rod packing.

3-5. Lysholm-type compressor.

a. The Lysholm compressor is a positive-displacement, axial-flow, rotary machine in which gas is compressed between meshing rotors. There are no rings or valves and the rotors are not in contact; a small clearance is maintained mechanically by the gearing, although most of the power is transmitted aerodynamically between the lobes by gas under compression. The compressor casing is water-cooled and the rotors are oil-cooled by internal shaft passages. The lobes are not lubricated, and the gas is discharged without being contaminated by oil in passing through the compressor. Standard capacity is 500 scfm at a compression ratio of 3.5:1 with one stage. Staging permits 2000 to 13,000 scfm at a compression ratio up to 11:1 with two intercooled stages. The relative advantages of this type of compressor are its low specific weight and space requirements, compression without contamination of gas or lubricant, low orders of vibration or pulsation, positive displacement without surge limitations (compared to other rotary types), low order of maintenance, good adiabatic efficiency, and low cost.

b. The built-in compression ratio is a peculiarity of rotor design. A variation of the outlet pressure from the design condition introduces a loss from backflow in an over-pressure system and unnecessary compression when the outlet pressure is too low. The latter effect is by far the more serious, and the machine should not be operated with the system below the design compressor pressure ratio for efficient results.

3-6. Power requirement calculation.

a. For the compressors discussed here, the basic analysis for power requirement is available from many references. The designer should consult *Marks Standard Handbook* and *Compressed Air and Gas Data*. Additional details may be obtained from other basic source books mentioned in those references.

b. With the use of the relations from these basic references, the designer should keep in mind effects of variables such as altitude, ambient temperature, and humidity. These factors, and others, all affect compressor capacity and thereby compressor performance.

3-7. Prime movers. Compressors may be driven by many prime movers: electric motors, steam or gas turbines, reciprocating engines, or water wheels. The choice of power may well be governed by the resources available at the operating site and the type of compressor selected. Direct drive is usually the optimum choice. Table 3-3 lists the common compressor drives used by the reciprocating and centrifugal compressors. Table 3-4 lists the characteristics of such drives.

3-8. Lubrication. All the major components of the compressor system must be adequately lubricated. The system designer will check with the manufacturer to be sure the proper specifications have been set. A summary of components, lubricant types, and some comments thereon are presented in table 3-5 to offer the designer a starting point. Typical references in the field such as *Compressed Air and Gas Data* should be consulted for a more detailed selection process for lubricant distribution for various types of compressors. While it is traditional to use the Society of Automotive Engineers (SAE) classification for lubricating oils, it should be noted that normal compressor oils do not usually carry an SAE rating. When SAE numbers are referred to, in practice, then, it is solely to define viscosity range and it is not to be taken as a general recommendation for grade.

Table 3-2. Types of centrifugal compressors.

		Approximate Maximum Ratings		
	Casing Type	*Pressure psig*	*Capacity Inlet scfm*	*bhp*
A.	Sectionalized			
	Usually multistage	10 *	20,000 *	600 *
B.	Horizontally split			
	Single stage			
	(double suction)	15 *	650,000 *	10,000 *
	Multistage	1,000	200,000 *	35,000
C.	Vertically split			
	Single stage			
	(single suction)			
	Overhung	30 *	250,000 *	10,000 *
	Pipeline	1,200	25,000	20,000
	Multistage	over 5,500	20,000	15,000

* Based on air at atmospheric intake conditions.

Table 3-3. Common compressor drives.

Compressor Type	*Reciprocating*	*Dynamic*
Drive Type		
Induction Motor	B-C-F-G	C-G
Synchronous Motor	C-E-F-G	C-G
Steam Engine	I	. . . .
Steam Turbine	G	C-G
Gas Turbine	. . . .	C-G
Hydraulic Turbine	G	. . . .
Gas Engine (Heavy-Duty)	C-I	G
Diesel Engine (Heavy-Duty)	C-I	G
Industrial Engine	B-C	B-G
Expander	I-G	C

Key

B — Belted
C — Coupled
E — Engine-Type Direct-Connected
F — Flange-Mounted
G — Geared
I — Integral

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Table 3-4. General Characteristics of Compressor Drives

Driver	Horsepower Range	Available Speed (rpm) (60-cycle power)	Possible Speed Variation	Efficiency	Starting Torque and Amperes % Full-Load	Stalling Torque % Full-Load
Induction Motor	1 to 5,000 (or larger)	3,600/N less 2% N = 1 thru 8 ***	Constant Speed	10 hp - 85% 100 hp - 91% 1000 hp - 94%	60 to 100% Torque 550 to 650% Amperes	150% min. (or more)
Synchronous Motor	100 to 20,000 (or larger)	3,600/N N = 2 thru 20 ***	Constant Speed	93% to 97%	40% Torque under 514 rpm, 40 to 100% Torque 514 rpm and over 300 to 500% Amperes	150%
Steam Engine	55 to 4,000	400 to 140	100% down to 20%	50% to 75% RCE	About 120%	About 115%
Steam Turbine	To 20,000 (or larger)	34,000 to 1,800	*100% down to 25%	35% to 82% RCE	175 to 300%	Up to 300%
Combustion Gas Turbine	3,000 hp at 10,000 rpm to 20,000 hp at 3,000 rpm (1,000 ft altitude and 80°F)		100% down to 55%	16% to 25% overall Thermal Efficiency for simple open cycle, 27 to 30% with regenerator.	Both single- and two-shaft designs require a sizeable starting motor or turbine. The single shaft design has poor part load torque characteristics and requires a larger starter. Two-shaft turbines have good torque characteristics.	
Integral Gas Engine	From 85 hp at 600 rpm to 5000 to 6000 at approx. 300 rpm.		*100% down to 60%	Up to 40% overall Thermal Efficiency	Nil - Started with compressed air.	About 120%
Integral Diesel Engine	From 100 hp at 600 rpm to approx. 1300 at lower rpm.		*100% down to 60%	32% (HHV) **	Nil - Started with compressed air.	About 120%
Coupled Gas or Diesel Engine	From 100 hp at 600 rpm to 5000 hp or more at approx. 350 rpm.		*100% down to 60%	41% on gas (LHV) and 36.6% on oil (HHV) **	Nil - Started with compressed air.	About 120%

Tabulation is limited to the commercially applied ranges of hp and rpm.

* Depends on design and existence of critical shaft speeds.
** High heat value or low heat value.
*** N is number of pairs of poles.

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Table 3-5. Lubrication Requirements.

Component	*Type of Lubricant*	*Comments*
Crankcase	Automotive-type of either naphthenic or paraffinic base, with or without detergents.	Water sludge accumulation when operating temperature low; pump usually employed for forced feed to bearings and crosshead wristpins.
Cylinder	Naphthenic base.	Critical selection problems; used to resist heat and oxidation; excessive carbon residue reduces discharge areas, causes leakage; pump requirements and maintenance severe; rate control important; low temperature problems.
	Phosphate esters.	Better resistance to combustion; high load carrying capacity; higher cost than naphthenic based oils; pump control important; low temperature problems.

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Section II. COOLING SYSTEM

3-9. General. The heat exchangers (see *Marks Standard Handbook* and *Compressed Air and Gas Data*) in the compressed gas system have three functions: the control of compression temperatures within safe limits, the reduction of compression power requirements, and the removal of moisture from the gas. Two classes of heat exchangers are used to accomplish these tasks: the heat exchanger between stages or intercooler, and the heat exchanger after the final stage or aftercooler. Two types are commonly used, the tube and shell for lower pressure intercoolers and the coiled tube for higher pressure (i.e., for pressure over 2000 psig). The coiled tubing must be secured by coil separators, usually wood blocking, so that the compressor pulsations cannot cause tube failure by erosion and fatigue. Materials far apart in the galvanic series scale should not be combined in tube and shell because of galvanic corrosion.

3-10. Discharge temperature. For large air compressors a maximum compression temperature of 250°F is recommended as a design consideration for operational safety. This limit allows some safety margin for abnormal operating conditions such as high intake temperature, leaking valves or rings, and fouled heat exchanger surfaces, all of which cause higher discharge temperatures. Note that an increase in the intake temperature results in a greater increase in the discharge temperature (see *Marks Standard Handbook* and *Compressed Air and Gas Data*). For gases such as hydrogen, helium, and nitrogen, higher design discharge temperatures of about 350°F are tolerable and sometimes tend to reduce carbon deposits; however, the power requirement factor to handle such temperature must be considered.

3-11. Moisture removal. A cooler-separator collects condensed moisture, oil, and heavy solid particles entrained in the gas stream. The separation is usually accomplished by reducing the velocity of the gas through expansion and directing the flow so as to obtain some centrifuging effect. The effectiveness of a well-designed separator depends on its operating temperature; therefore, it should not be located where it is subject to being heated by radiation or convection. This is an especially important consideration for dried-air requirements since the separators remove more than 90 percent of the moisture in the high-pressure air. An evaluation of the separator design is obtained by measuring the separator water discharge, operating temperature, and stage compression ratio (sec III).

3-12. Design considerations. In the design and use of effective heat exchangers several items are of prime importance for effective operation: location, cooling water supply, cooling tower requirements, and instrumentation. Guidance for these elements is summarized in table 3-6 and details may be found in *Marks Standard Handbook* and *Compressed Air and Gas Data.*

Table 3-6. Heat exchanger considerations.

Element	*Comments*
Location........................	Near compression discharge points; drainage required; maintenance should be considered.
Cooling water	Optimum water rates should be investigated; hardness and resulting lack of cooling capability creates a problem in a closed system; usually designed to handle entire heat-load equivalent of compressor horsepower input.
Cooling tower	Several types suitable: forced, induced or natural draft; suction must be carefully watched; evaporation and wind entrainment

Table 3-6. Heat exchanger considerations—Continued.

Element	*Comments*
	must be studied; flow rates call for careful design; lowest economical temperature is 5 degrees above ambient wet bulb temperature.
Instrumentation and controls	Requirements are head or stage pressure, water control valve, sight glass, pressure relief valve, gas intake temperature at each stage, final temperature after last cooler, safety valves for coolers; vibration isolation consideration is important near compressor.

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Section III. AIR TREATMENT

3-13. General. Atmospheric air contains dust, pollen, water vapor and other gases that must be disposed of before, during, or after the compression process so that the final product is clean and dry for its intended use. In addition, lubricated compressors introduce carbon and oil "mist" into the stream; these must also be removed. In this section the various stages of air treatment are discussed.

3-14. Air treatment stages.

a. Intake air must be filtered before it reaches the compressor. In each stage of the compressor air is heated, and after each stage it is cooled. During the cooling process, moisture is precipitated and drained off. Depending on the intended use, the air may then go through absorption, adsorption and, possibly, chiller units to lower the amount of water vapor being carried by the air. The oil mist must then be removed. Finally particles must be eliminated. Table 3-7 treats these various stages and offers the designer guidance for system development.

b. Additional design considerations for some of these steps, beyond those offered in the table, are in order:

(1) *Filtration.* Note that filters must be cleaned and serviced on a regular schedule; a clogged filter is worse than no filter at all.

(2) *Moisture removal.* In the removal process by adsorption the capacity of the drying unit is affected by the following items: inlet humidity, temperature of the desiccant bed, exit conditions required, residual moisture in bed after regeneration, bed depth and fluid velocity, shape of bed, packing of desiccant, and contact time. The designer will consider all these in the selection of the process and the desiccant. Further, cost will be a factor. As noted in table 3-7, silica gel excels at high humidity while the molecular sieve is best for low humidity; a composite bed is possible.

(3) *Oil removal.* A scrubber can be used to supplement the filtration process, but only at considerable extra cost. Measurements of amounts of oil vapor on the gas stream are difficult to perform with high accuracy and thus, great care must be taken here.

(4) *Particle removal.* Particles arise in the system due to dirt left over from the manufacturing process, abrasion as a result of vibration, attrition of the filter and, of course, the fluid quality itself. Careful attention to details is called for.

Table 3-7. Air treatment stages.

Stage	*Treatment Method Used*	*Comments*
1. Intake Filtration	dryfilters: -felted cloth	Always used with non-lubricated machines; maintained by vacuuming or dry cleaning.
	-oil treated paper	Subject to shrinkage under high humidity; removed and replaced when dirty.
	viscous-impingement filters	Woven or packed wire in cells or frames; not used for dusty areas or non-lubricated machines; frames oil-coated with oil, viscosity carefully selected.
	oil bath filters	Uses a scrubbing action; dirt collected in a sump; viscosity must be carefully selected.
2. Moisture Removal	compression and cooling	Operates on basis of condensation during the compression process; efficiency depends on gases obeying Dalton's law.
	adsorption	Solid surfaces attract and condense water vapor by adsorption and then capillary action; capacity varies with vapor pressure of fluid.
	-silica gel	Can adsorb up to 50% of its dry weight of moisture from saturated air; temperature has important effects; can be reactivated easily; good for high humidity.
	-activated alumina	Chemically inert; non-toxic; shock, abrasion and crushing resistant; acts like silica gel; readily regenerated.

Table 3-7. Air treatment stages—Continued.

Stage	*Treatment Method Used*	*Comments*
	-molecular sieve	High capacity at high temperature and low humidities; derived from a combination of silica gel and aluminum oxides; readily regenerated; for low humidity, capacity far exceeds other desiccants; most expensive.
	non-regenerative drying	Adsorptive desiccant discarded after one cycle; generally sealed in a cartridge.
3. Oil Removal	coarse screening	First stage used to break up slugs; filter out gross contamination; also used to level separating load.
	dropout chamber for gravity separation with wire mesh	De-entrains oil mist; can remove up to 90% in droplet sizes larger than 20 microns.
	oil adsorption	Filter should have high affinity for oil; should be unaffected by water vapor; can use same desiccants as in moisture removal; can use wool felt, fullers earth and glass wool filters.
4. Particle Removal	Filters	Necessary when particles can interfere with system operations; see Military Specifications MIL-F-5504 and MIL-F-8815; can use paper fiber and woven wire filters.

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Section IV. CRYOGENIC FLUID PUMPS AND VAPORIZERS

3-15. General. Much of the information presented in the earlier sections applies to cryogenic systems as well. Here, some pertinent pump and vaporizer information specific to low-temperature systems is discussed.

3-16. Pumps. Two types of commercially available cryogenic-fluid pumps are the centrifugal or turbine pump and the axial-piston pump

a. Centrifugal pumps. This type is generally used when large discharge rates must be handled with small discharge heads. Some of the problems associated with the centrifugal or turbine pump are cavitation and premature failure of pump shaft bearings and seals. The centrifugal pump may be run submerged in the fluid to provide the required net positive suction head to prevent cavitation. The critical component of ball bearings in pumps used at cryogenic temperatures is the retaining ring. One of the most satisfactory materials for it is a fabric-reinforced phenolic of glass-filled Teflon. The sealing problem can be alleviated by locating the seal at ambient temperatures instead of cryogenic temperatures if possible. (See chap 4 for more information on seals and gaskets.)

b. Axial-piston pumps. The axial-piston pump has been used for very high pressures (up to 10,000 psig) and for fairly small flow rates. The problems associated with this type of pump include sealing the piston and removing the frictional energy dissipated. For the former, see chapter 4. For the latter, the energy dissipated because of friction may be removed by using a heat exchanger between the discharge fluid and the piston cylinder.

3-17. Vaporizers. A vaporizer is simply a heat exchanger which supplies the heat of vaporization to the cryogenic fluid as it is discharged from the liquid storage vessel. There are many types which use the heat from ambient air (natural convection and forced air), electricity, gas or steam, which are capable of operating at pressures up to 15,000 psig.

a. Ambient air vaporizers. Ambient air vaporizers work well, although bulky, but have some important problems. Since the fins tend to ice over, the gas does not reach ambient temperatures, causing piping to ice over and perhaps become brittle beyond the vaporizer. They should be considered for use only as part of a system containing other vaporizers.

b. Electric vaporizers. These are the most compact vaporizers for high flow rates (up to 500,000 scfh) and they are constructed by casting heating coils and the cryogenic fluid tubing in aluminum. These provide the simplest operation of any vaporizers requiring power input.

c. Other vaporizers. Forced flow ambient air vaporizers with automatic electric defrost may be considered, but they are bulky. Jet engines are used as the heat source for very high flow rates, but are not efficient. Steam heat exchangers are available, but are complicated when compared to the electric vaporizers and may freeze if the steam supply is inadequate. These should be considered only for applications with special requirements.

Section V. BUILDING REQUIREMENTS

3-18. Protection.

a. The lubricating and cooling requirements make protection from cold weather mandatory. The pour points of compressor oils range from zero to 10°F and differential contraction of dissimilar metals employed in sleeve bearings must be considered. The water-cooled compressor must be protected against jacket freezing, and cooling systems employing an atmospheric cooling device are piped so that the cooling can be controlled in freezing weather. Moisture in air under compression will freeze if the process air temperature is sufficiently lowered at any point in the cycle such as the coolers. Difficulties encountered from this include the failures of intake valves and check valves, and freezing of moisture in separators and drain valves leading to eventual blockage of plumbing and a forced shutdown.

b. A building, then, is required to maintain an environmental temperature above freezing. Likewise, a structure of some kind is necessary to protect against other unfavorable working conditions such as hot surfaces caused by solar heat absorbed by unprotected machinery, exposure to high winds, and blowing sand. Finally, and equally pertinent, a practical environment must be maintained so that maintenance and operational personnel can function with a reasonable degree of efficiency.

3-19. Induction. The air to the compressors must be inducted from outside the building; otherwise it becomes impossible to heat the building in winter. The reduction in pressure within a tight building brought about by inside induction could cause its collapse. The air induction point should be located in a space free of concentration of combustible gases or vapors and protected by filters for the exclusion of sand, dust, or other solids. Where the noise usually made by the suction of the compressor or the noise occasionally encountered in compressor discharge lines is objectionable, a silencer should be used. Sometimes where air contamination is a function of wind direction, it is advisable to have multiple air induction points. The air induction piping must not be collocated with piping conducting engine exhaust or combustible gases or liquids. The designer will refer to TM 5-805-4 for further details.

3-20. Cranes. Commercial compressors of substantial horsepower are massive, and maintenance or repair work requires a lifting device to handle the heavy parts. An overhead hoist capable of movement in two lateral directions will greatly facilitate the work of repair, or major overhaul, and add significantly to the safety of all such operations. Adequate laydown space should be provided near the compressor. The crane, or hoist, should have adequate vertical clearance to lift the uppermost parts of the machine without resorting to makeshift handling, and its area of operation should include the heat exchangers.

Section VI. FOUNDATION AND MACHINE BASES

3-21. General. Any stationary compressor must be anchored to a solid mounting, which may mean bolting a small compressor to the floor or building a large, special foundation. The soil strata must be carefully examined including drilling for core samples over the bearing area down to bedrock. If soft soil is encountered in depth, piling is often necessary. Instruction books and foundation plans should be consulted in the design of the foundation. For special cases a foundation specialist may be required. Even manufacturers' recommendations cannot be guaranteed for a specific installation and the responsibility for a successful foundation rests with the designer. The foundation is not the place to economize.

3-22. Basic design objectives.

a. The large compressor and its driver should be maintained in proper alinement and elevation by the foundation; the vibration should be minimized and isolated from the building structural members. Reference will be made to TM 5-805-4 for other design considerations.

b. Soil has two load capacities, static and dynamic. Foundation design for compressors is concerned with the allowable dynamic loads, which are considerably smaller than the allowable static loads. These loads should be distributed over the entire base area. Sliding caused by unbalanced horizontal forces can be prevented by having sufficient mass and bearing area. To avoid warping, the temperature of the base should not vary internally. The resultant of the vertical machine and blockloads and the unbalanced inertia force must fall within the base area.

c. Minimizing vibration depends on the soil, the compressor type, the foundation bulk, and the soil bearing area. Soil moisture may vary seasonally and core specimens analysis should take into account these variations. The compressor type governs the decision for separate bases or combined bases for compressors. Reciprocating compressors tend to balance forces and a combined base is recommended. Dynamic compressors require separate bases and rotary compressors can have either. The foundation should rest only on one type of supporting soil or rock. Foundation bulk is never the sole answer to foundation problems and bearing area is considered to be of more value than foundation mass in damping vibrations. Therefore, the area of the base should be increased if there is any question of compressor vibrations causing sliding or rocking.

d. The manufacturers will supply weight, center of gravity, and unbalanced force data for compressors and will give suggested foundation dimensions for firm dry soil backed up by bedrock.

3-23. Design Guidance. Some basic guidance for foundation follows:

a. The manufacturer's standard foundation is designed to absorb the particular machine's unbalanced forces and provide rigidity to maintain good alinement in very firm soil.

b. Sand, clay and alluvial soils are considered good support for static loads, but not for dynamic loads. If the soil at the site is anything less than bedrock or well-cemented sand and gravel in thick beds, or if the soil is wet at times, special precautions must be taken. In such cases, the foundation should be extended or a mat used (a subsurface support for the foundation). The support area should be increased several times if necessary. No mistake can be made by spreading the foundation, but going deeper will increase the tendency to rock.

c. The foundation should not be deeper than its length unless bedrock can be reached easily. This applies where there are unbalanced horizontal forces involved. The long dimension of the foundation should be in the direction of the unbalanced forces. If the firm ground is anything less than bedrock, the foundation should be extended or a mat should be used.

d. The foundation should not be placed in ground that has been disturbed or in ground where part of the area has been disturbed. Any disturbed ground will allow the compressor to settle, or tip and go out of alinement.

e. Piling should be used only for large installations and should only be used by the experienced foundation designer.

3-24. Alinement of machinery. The installation and alinement of large compressors are usually accomplished either by the manufacturer, as supervised by the representative, or under contract by firms specializing in this field and recommended by the manufacturer. The designer should be aware that the extensive dimensions of large horizontal machines require great accuracy in the alinement of pistons, crossheads, and cranks. Separate crankcases connected to a common driver must have centerlines in the same horizontal plane and be parallel. Cylinders, flange-connected to a common crankcase, must be drawn tight so that their centerlines are also in the same horizontal plane and parallel. Coupling-connected shafts must be alined carefully by shimming for dependable operation. Although some couplings are designed to connect imperfectly alined shafts, the tolerance is usually quite limited, and relative motion in the coupling causes wear and requires lubrication.

Section VII. SAFETY

3-25. Design considerations. The question of safety should be of great concern to the designer. General guidance and areas to look for will be highlighted here. This, however, should not be considered all-inclusive but rather suggestive of problems. These items are summarized in table 3-8.

Table 3-8. Safety considerations.

Area of Concern	*Potential Problems*
1. General operation	Purity of compressor intake air; lubricating oil feed rate; compressor discharge temperatures; compressor pressure ratio; location, use, and operation of "panic buttons."
2. Inspection	Oil build-up must be prevented; rusting and pitting; constant observation of fluid properties; filling of compressed gas cylinder; excessive torque on valve handwheels; valve leaks; proper placement and location of equipment to prevent personnel hazards.
3. Tools	Overhead cranes or floor hoists for heavy equipment; isolated welding area with proper equipment; working platforms for elevated machinery; tool board and beds carefully stocked and labelled; ventilated storage rooms; appropriate personnel uniforms; positive electrical control.
4. Vibration	Excessive vibration cannot be tolerated: use mass change, anchorage alteration for remedy; gage vibration causes instrument failures; safety valves must not vibrate.
5. Noise	Severe personnel hazard; building design can assist; control by changing position of sound and listener, acoustic environment, and separation; bearings must be watched; compressor intake and outlet must be watched.

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Section VIII. SUMMARY

3-26. Basic specifications. Since there are so many detailed components to be specified in compressor systems, it is worthwhile to summarize here the minimum data required for the design and bid process. These are grouped in various logical categories and displayed in table 3-9; much of this information is obtained from *Compressed Air and Gas Data.*

Table 3-9. Basic specifications.

(These are intended as a minimum guide only.)

Item	*Typical Data Required*
Gas to be handled	Analysis by mole, volume or weight; composition variation, corrosive expectations, discharge temperature limits.
Quantity to be handled	Differentiate between wet and dry; pressure and temperature reference points.
Inlet conditions	Barometer; pressure and temperature at compressor flanges; humidity; ratio of specific heats; compressibility.
Discharge condition	Pressure at compressor flange, temperature references; compressibility.
Interstage conditions	Temperature difference between gas out of cooler and water into cooler; interstage removal of gas requirements with permissible pressure range.
Variable conditions	Expected variations in intake conditions (pressure and temperature) and their interrelations.
Flow diagram	Flow sheet showing controls involved is useful.
Regulation	What is to be controlled—pressure, flow, temperature; variations allowed; manual or automatic regulation.
Cooling water	Temperature—maximum and minimum; pressure; open or closed cooling system; source and nature of cooling water.
Heat exchangers	Use of intercoolers and/or aftercoolers; mounting of equipment; construction codes; special materials requirements.
Driver	Type; electric motor-type, current condition, power factor, enclosure, service factor, temperature variations; steam-inlet and exhaust pressure and temperature, water rate; fuel gas—gas analysis, pressure, heating value.
General	Lubrication requirements; installation location floor space available; soil and foundation character; accessories.

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CHAPTER 4
PIPING AND FITTINGS

Section I. INTRODUCTION

4-1. General. This chapter deals with the two types of pipe systems: high-pressure gas and cryogenic. Design information regarding materials, fluid mechanics, joining methods, valves, supports, guides, and anchors will be discussed.

4-2. High-pressure gas systems. Design problems for high-pressure gas systems are treated first. It should be recognized that the design process is, at least in part, an iterative scheme. The steps that must be taken follow the pattern shown in figure 4-1. Note that one must either specify a material and iterate or consider the fluid mechanics and iterate.

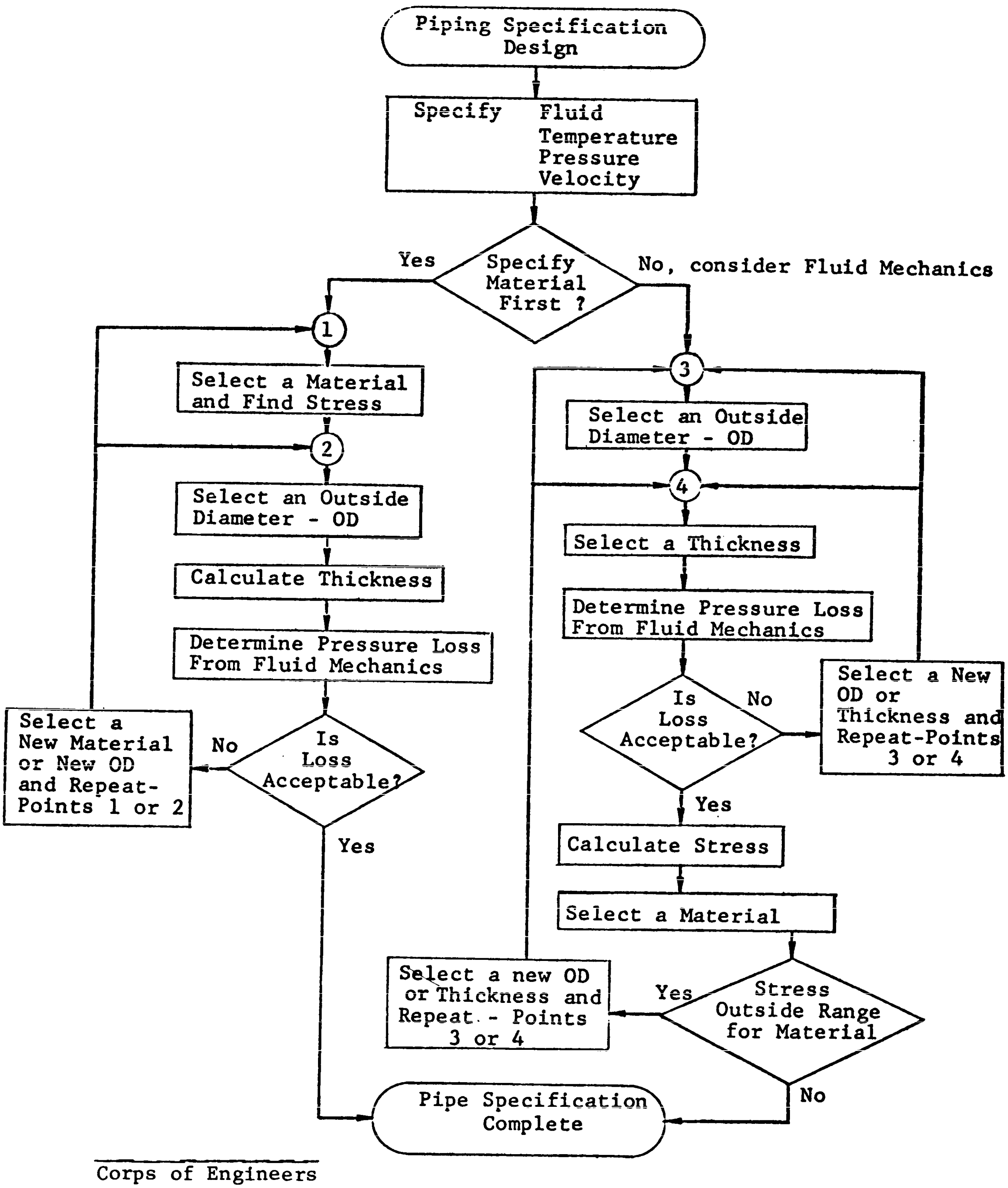


Figure 4-1. Typical design specification steps.

4-3. Cryogenic systems. The steps for the design of piping systems for cryogenic fluids are similar to above, but because of the extremely low fluid temperatures and their boiling points at temperatures below ambient, additional design factors must be considered. These additional factors are covered below in section III.

Section II. HIGH-PRESSURE GAS SYSTEMS

★ 4-4. Materials and sizing.

a. Considerable data is available from the American National Standards Institute (ANSI) Code for Power Piping, B31.1; contained therein are data for piping, tubing, bolting, fittings, valves and flanges. Table 4-1 offers a sampling of the material specifications for piping use in high-pressure, high-temperature systems. The relevant ANSI B31.1 Codes prescribe minimum requirements for the design, fabrication, installation, materials, and testing of piping systems. Furthermore, this same information is also presented and available in the Piping Handbook.

b. As part of the design procedure, working stresses must be determined. In table 4-2 values are given for the stated class and grades of materials listed in table 4-1. Again, for more complete data, reference should be made to ANSI B31.1, or to the *Piping Handbook.*

Table 4-1. Typical Material Specifications for Piping

ASTM Specification	Type	Service	Pipe Sizes	Weld	Wall thickness tolerance	OD tolerances (max)
A53	Carbon steel pipe Wrought iron pipe	General (steam, water, gas, etc.)	No limits ($\frac{1}{8}$"-24")	Seamless, furnace butt welded, electric resistance welded	Not more than 12$\frac{1}{2}$% under nominal	To 1$\frac{1}{2}$" OD + $\frac{1}{64}$", − $\frac{1}{32}$" 2" OD and over ± 1%
A106	Carbon steel pipe Wrought iron pipe	High-temperature service	No limits ($\frac{1}{8}$"-24")	Seamless	Not more than 12$\frac{1}{2}$% under nominal	*
	*For details see Code					

From American National Standards Institute (ANSI) Code B31.1. Used with permission of American Society of Mechanical Engineers.

c. Once the material has been selected (this is design following the left hand path in figure 4-1), and the working pressure for the piping has been noted, the thickness of the wall can be determined. ANSI B31.1 specifies an equation for thickness in terms of maximum internal service pressure and allowable stress at the operating temperature, outside diameter of the pipe, and coefficients generally a function of material type and pipe size. The code shall be consulted for details. If pipe is ordered by nominal wall thickness, as is customary in practice, the manufacturing tolerance must be taken into account.

d. The code equations may be inverted and solved for pressure or allowable stress if the designer wishes, instead of specifying a nominal wall thickness. This is the right hand path in figure 4-1.

e. From a cost point of view, it is always less expensive to order standard commercial pipe; thus, the designer shall consult current standards after use of the code equations.

Table 4-2. Maximum Allowable Stresses, psi

ASTM Specification	Grade	− 20° to 100°F	200°F	400°F	600°F	800°F	1000°F
A53	A	16000	15300	13800	12350	9300	2500
	B	20000	19100	17250	15500	10800	2500
A106	A	16000	15300	13800	12350	9300	2500
	B	20000	19100	17250	15500	10800	2500
	C	23350	22250	20150	18050	12000	

From American National Standards Institute (ANSI) Code B31.1. Used with permission of American Society of Mechanical Engineers.

4-5. Fluid mechanics.

a. The most common problem is the pressure loss in the pipe due to frictional effects. Since the relations for pressure loss are well known and the attendant correlations for friction factors (the Moody diagram), for roughness, and for losses through valves, elbows, tees, and bends are well established, only problem areas are addressed here. Fluid mechanics texts or the *Piping Handbook* will be consulted for the actual equations.

b. In turbulent flow, determine the friction factors from a Moody diagram. Note that they are a function of the Reynolds number and the ratio of pipe roughness to pipe inside diameter. Several important considerations are necessary. First, in determining friction factors the designer must be sure that the flow is indeed turbulent. In the transition range, the experimental data is not clear and engineering judgment should be used to estimate values. Furthermore, in arriving at any appropriate value for roughness, there is generally a considerable range for many commercial materials. Also, when evaluating the performance of a system over a long period of time, the roughness can be expected to grow and, thus, affect operation to some extent. Hence, some judgment in choosing its value is necessary.

c. Compressible flow is another problem area to be investigated. For most engineering and design purposes, if the pressure loss, as calculated from the standard equations, is less than 10 percent of the initial pressure, it is reasonable to assume incompressibility. A further criterion may be established by investigating the Mach number of the flow. From standard fluid mechanics texts, or from the *Piping Handbook*, incompressibility may be assumed provided that

$$kM^2 << 1 \qquad \textit{(Equation 4-1)}$$

where k is the ratio of specific heats, M the Mach number. Note that

$$M = V/a \qquad \textit{(Equation 4-2)}$$

where a is the local speed of sound. It can be shown that

$$kM^2 = \rho V^2/P \qquad \textit{(Equation 4-3)}$$

where ρ is the density and P, the pressure of the stream. The designer must, therefore, apply equation 4-1 to the conditions under consideration.

4-6. Joining methods.

★ High-pressure gas systems have both mechanical (flanges and gaskets) and welded joints. The applicable ANSI B31.1 Codes will be followed for design information. Some additional guidance is offered here.

a. Flanges, gaskets, and bolting. Flanged joints will be specified where pipe lines, components, or equipment must be disassembled for maintenance work. The elements of flange design are:

- type of flange facing
- finish of contact surface
- gasket type
- bolting
- flange proportions

Some guidance for these is offered in figure 4-2 and in tables 4-3, 4-4, and 4-5. For flange design, all these items must be specified. The designer should be aware that a number of combinations will be available for the same or similar applications and that the trade-offs relate predominantly to cost.

b. Welded joints. Welding design procedures and inspection are described in detail in the ANSI B31.1 Codes and in TM 5-805-7.

4-7. Valves.

Valves are used to control the flow or protect equipment. Each type of valve has its own particular function. Selection is governed by functional characteristics generally described by:

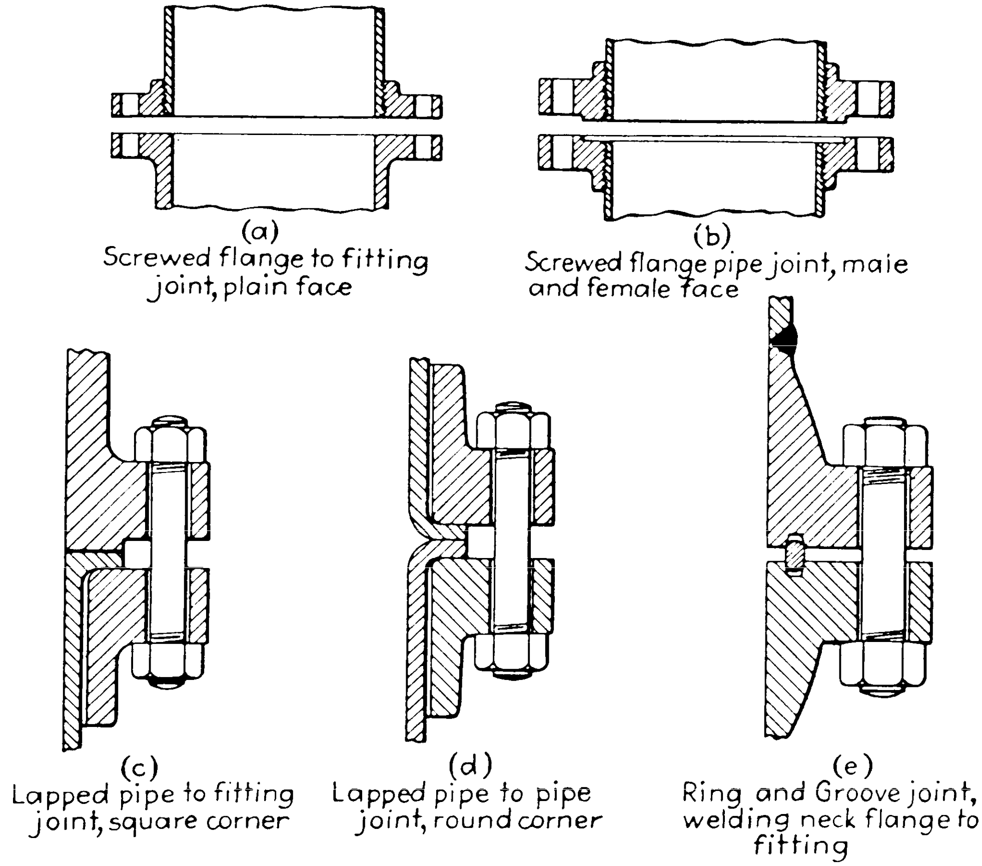
- on-off service
- throttling service
- back flow prevention
- pressure control

Rather than offer the typical array of valve types

Table 4-3. Elements of Flange Design

Element	Types	Comments
Flange facing	See figure 4-2	Recess type preferred: figure 4-2 (b) or (e): plain facing satisfactory to 220° F; serrated raised-face to 750° F; high contact force required; proprietary facings should be investigated.
★ Surface finish	64 or 32 are usual	Varies with facing and gasket used; consult ANSI B16.5; bolting must be matched to finish.
Gaskets	See table 4-4*	May be metallic, non-metallic; must match with facing, finish, bolt loads.
Bolting	See table 4-5	Carbon, low-alloy steel satisfactory to 700° F; consult ASTM A 309, A 449, A 354; bolt relaxation must be considered.
★ Proportions	loose type is usual	Consult ANSI B16.5 for standard dimensions; cylindrical hubs added to loose type; standards offer safe and economically sized flanges.

*Complete data in *Handbook of Industrial Pipework Engineering.*



From Piping Handbook by Reno C. King, Copyright 1973, McGraw-Hill Book Company. Used with permission of McGraw-Hill Book Company.

Figure 4-2. Commonly used flanged joints.

and mechanisms readily available elsewhere (e.g., ★ *Piping Handbook* or ANSI B16.5 and B31.1, table 4-6 presents a guide to valve selection. Design specification can be obtained by reference to this table.

Table 4-4. Some Gasket Materials For Different Services

Fluid	Application	Gasket Material
Air	Temp up to 1000°F	Spiral-wound comp. asbestos
	Temp up to 750°F	Comp. asbestos
	Temp up to 220°F	Red rubber
Gases such as	Temp up to 1000°F	Asbestos, metallic
Nitrogen and	Temp up to 750°F	Comp. asbestos
Oxygen	Temp up to 600°F	Woven asbestos
	Temp up to 220°F	Red rubber
Ammonia	Temp up to 1000°F	Asbestos, metallic
	Temp up to 700°F	Comp. asbestos
	Weak solutions	Red rubber
	Hot	Thin asbestos
	Cold	Sheet lead
Hydrogen	Temp up to 1000°F	Comp. asbestos
	Temp up to 300°F	Synthetic rubber bonded cork

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*Table 4-5. AISI Steel Recommended For Bolts**

	Proof strength† psi		
	75,000	100,000	125,000
1/4 - 3/4	1038	4037	4037
3/4 - 1 1/4	1038	4140	4140
1 1/4 - 2	4140	4140	4145

*All selections are based on a minimum tempering temperature of 850°F.
†At room temperature.

Table 4-6. Guide to Valve Selection

General Function	Valve type	Guide lines	Service	Comments
THROTTLING SERVICE	Globe valve	Most commonly used valve for efficient regulation of a critical service. Used extensively for Automatic Process Control. Not normally used for ON-OFF service. Wide size, pressure/temp. range. Available in an extensive material range	Sizes range 1/8 in. to 16 in.; pressures to 6000 psi; temperature to 1500°R.	Can be designed with wiping gear for cleaning
	Angle valve	Similar to conventional globe valve but ends are at right angles to each other. Provides less pressure drop and reduces number of pipe fittings in a system. Care should be taken at design stage to ensure that valve is not subjected to stresses often present in pipe bends.	Sizes like "Y" valve; pressures to 5000 psi.	
	Needle valve	Restricted to smaller sizes and used where close manual regulation of flow is required. Suited to high pressures.	Sizes from 1/8 in. to 6 in.; pressure to 3000 psi; can go to 10,000 psi.	
	Butterfly valve	Extremely simple in construction. Specially suited for large flows of gases, liquids, and slurries at low pressures. Also suited for ON-OFF conditions. Low pressure drop, low priced, clean, and quick acting. Ordinarily not 'bubble-tight' but can give tight shut-off according to design.	Sizes from 1/4 in. to 180 in.; pressure to 1200 psi; temperature to 3000°R.	
	Diaphragm valve	Glandless type of valve for corrosive, volatile, and toxic fluids, particularly where leakage must be avoided. Although choice limited by diaphragm materials a very wide range exists to handle many corrosive fluids. Temp. pressure imposes limitations. Valve versatile and also used for ON-OFF service.	Sizes from 1/2 in to 2 in.; note that pressures only to 800 psi.	Can be fitted with quick acting lever for rapid operation

Table 4-6. Guide to Valve Selection

General function	Valve type	Guide lines	Service	Comments
ON-OFF SERVICE	Gate valve	Wide variety of designs. Straight-through flow when open. Minimum pressure drop. If used for throttling there is tendency to erode quickly with consequent leakage. Wide use range including corrosive duties. Extensive size and pressure/temp. range. Quick opening types available. Tapered disc type can be used for throttling.	Sizes range from 1/8 in to 72 in., pressures as high as 6000 psi; temperature to 1000°R.	Obtainable in practically any type machineable material; pressure loss very small compared to globe valves
	Slide valve	As above. Fluid pressure provides effective closing. Has a limited use for throttling.		
	Lubricated plug valve	For critical service requiring packing under pressure. Lubricant prevents leakage, reduces friction, wear, and turning effort. Lubricant can cause contamination of highly pure products and its use sets the maximum temperature of operation.		
	Non-lubricated plug valve	Inexpensive materials can be elastomer or plastic coated. Excellent for corrosive conditions. Relatively inexpensive as high cost metals are avoided. Pressure/temp. conditions limited by lining materials.		
	Ball valve	Widely used in industry where conditions of corrosion and high pressure, high temperature exist. Simple compact, and quick opening. Very wide size and temp. pressure range. Good for conditions which require to be fire-safe. Conventional type has poor throttling characteristics. Can be used for throttling if special ball fitted (i.e. V. Notch)	Sizes range from 1/4 in to 60 in; pressures up to 3000 psi.	

Table 4-6. Guide to Valve Selection (continued)

General Function	Valve type	Guide lines	Service	Comments
PREVENTION OF BACKFLOW	Check valve	Automatically prevents reversal of flow in lines. Kept open by fluid in piping system. Various types exist to meet different conditions, e.g. pulsating flow, horizontal or vertical mounting, etc.	Wide sizes and pressure and temperature ranges available.	
AUTOMATIC PROCESS CONTROL	Automatic control valves	Wide range of valves for automatic regulation and control of process temperature, pressures, flow rates, liquid levels, etc. Covers a vast field of operation including that via a computer. Continuously operated plants have led to growth. Reliability is essential. Actuated from a signal received from the flowing medium itself of from a programme having pre-set conditions. Operation can be hydraulic, pneumatic, electric, or combinations of the same.	Wide range of service conditions available.	Uses many of the globe, angle and "Y", balls, plug, butterfly as above. (See Chapter 6 for additional details)
SAFETY AND PROTECTION	Safety valve	For protection against excessive pressure. Used with gases	Selected with the maximum pressure expected; available in all ranges of interest.	Generally should be located near vessels; should be provided where compressors can discharge into a dead end vessel.
	Relief valve	For protection against excessive liquid pressures.		
	Safety-relief valve	Can be used for either gas or liquid. All types require periodic inspection to ensure that they can function. Types exist to meet all conditions, including corrosion, met within industry.		
	Bursting disc	For protection of plant systems where extremely rapid pressure rise may be encountered and where safety or relief valves might not be effective. Gives instantaneous unrestricted relief. Discs are available in many materials and require to be replaced after each operation.		

Section III. CRYOGENIC SYSTEMS

4-8. Design principles. Cryogenic systems are those operated at very low temperatures, colder than refrigeration systems. The dividing line between the two is not exact, but the two are commonly separated at −150°C or −238°F.

4-9. Design. Designers of cryogenic systems cannot rely entirely on experience gained on steam or refrigeration systems. The low temperatures and cryogenic materials present certain problems not commonly encountered in other systems. Some of the frequently overlooked and unrecognized design problems are—

a. Long runs of piping that will contain a liquid cryogen must be vented at intervals along the entire length of the piping. Two-phase flow first occurs when the liquid is introduced into a warm pipeline, then all of the liquid vaporizes and the resulting gas accelerates through the pipe to the nearest exit. If the pipe is sized for the liquid, the pressure drop of the unvented gas can easily be great enough to slow or even stop the advances of the liquid through the pipe. Venting at intervals of approximately 50 feet for small lines and at approximately 100 feet for 4-inch and larger lines will avoid many problems that occur on initial cooldown. The size of the vents should be calculated to match the flow rate of the entering liquid.

b. Lines 6 inches and larger that are partly filled with a liquid cryogen are subject to upward bowing. This takes place when the wetted bottom of the pipe is much colder than the unwetted top portion, and differential thermal expansion causes the pipe to rise between supports. This causes an upward thrust on intermediate restraints and may result in support failure if the restraints are not designed for a load in the upward direction.

c. When stainless steel pipe is selected for a cryogenic system, the pipe weight is usually Schedule 5S or 10S. This thin wall pipe frequently must withstand large anchor reaction from thermal expansion loads. When the loads exceed one or two kips, the pipe wall should be reinforced at the anchor attachment points with a doubler plate or a stainless steel saddle to prevent the pipe from bending or deforming at the anchor points.

d. Schedule 5S and 10S stainless steel pipe is usually specified to conform to ASTM A 269 Type 304 or 316. Tolerances allowed by this standard are such that it is possible for large pipe sizes not to be matched up. One pipe length could actually slip inside the other if each were on different ends of the tolerance extremes. This can be avoided by requiring tighter tolerances on large sizes of 5S or 10S stainless steel piping.

e. Concrete tends to spall when it is subjected to cryogenic liquids. To dissipate a liquid spill rapidly, it is good practice to place 6 inches or more of crushed stone in any area where a cryogenic spill could occur.

f. Liquid hydrogen piping is sometimes pressure tested with water, and liquid oxygen lines are frequently cold-shocked with liquid nitrogen. Loads and thermal expansion stresses must be calculated for the most severe service.

g. Cryogenic valves have long extended stems and heavy motor operators. The weight of these items must be taken into consideration when designing supporting elements.

4-10. Materials, lines and insulation. As noted in chapter 3, careful consideration must be given to the low temperature properties of the material in contact with the cryogenic fluid. Furthermore, the pipeline construction and the type of insulation is also of prime importance.

a. Materials. Generally wall thickness can be determined according to ANSI B31.1. An exception is the vacuum system discussed below. Minimum operation temperatures are obviously critical for allowable stress, and reference to ANSI B31.3 is necessary. As a guide, typical ferrous and non-ferrous metals in various temperature ranges are shown in tables 4-7 through 4-10. Note that these tables also cover elements to be discussed later, mainly flanged bolting and gaskets. It is also important to consider cost very carefully in the design of cryogenic systems. Typical data on comparative costs for uninsulated pipe are shown in table 4-11, which indicate the great variations possible in system design.

Table 4-7. -20 to -50°F Material Specification

Bolting	ASTM A193 Grade B7 bolts ASTM A194 Grade 2H nuts
Gasketing	Compressed asbestos sheet
Pipe	ASTM A333 Grade C, seamless*
Butt-welding fittings	ASTM A420 Grade WPLC
Castings	ASTM A352 Grade LCB
Forgings	ASTM A350 Grade LF1 Flanges to ANSI B16.5; S-W fittings to ANSI B16.11

* allowable stress 18,350 psi

From article by H.M. Howarth. *Petro/Chem Engineering*.

Table 4-8. -50 to -150°F Material Specifications

Bolting	ASTM A320 Grade L7 (4140) bolts ASTM A194 Grade 4 (C-Mo) nuts
Gasketing	Type 304 spiral-wound
Pipe	ASTM A333 Grade 4 (Cr-Cu-Ni)*
Butt-welding fittings	ASTM A420 Grade WPL4 (Cr-Cu-Ni)
Castings	ASTM A351 CF8 (304) or CF8M (316) or ASTM A352 Grade LC3 (3-1/2% Ni)
Forgings	ASTM A350 LF4 (Cr-Cu-Ni) Flanges to ANSI B16.5 S-W fittings to ANSI B16.11

*allowable stress 20,000 psi

From article by H.M. Howarth, *Petro/Chem Engineering*.

Table 4-9. Below -150°F Material Specification

Bolting	ASTM A193 (orA320) Grade B8 (304) bolts ASTM A194 Grade 8 (304) nuts
Gasketing	Type 304 spiral-wound
Pipe	ASTM A312 TP304 seamless*
Butt-welding fittings	ASTM A403 Grade WP304 ANSI B16.9
Castings	ASTM A351 Grade CF 8 (304) or CF8M (316)
Forgings	ASTM A182 Grade F304 Flanges to ANSI B16.5; S-W fittings to ANSI B16.11

* Allowable stress 18,350 psi

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Table 4-10. Aluminum Specification

Bolting	ASTM A193 Grade B8 (304) bolts ASTM B98 nuts
Gasketing	Type 304 spiral-wound
Pipe	ASTM B241 alloy M1A (3003) *
Butt-welding fittings	Aluminum conforming to ASTM A234, as applicable
Castings	ASTM A351 Grade CF8 (304) or CF8M (316)
Forgings	ASTM B247 alloy GS11A (6061-T6) Flanges to ANSI B16.5 dimensions

* Allowable stress 9,500 psi

Table 4-11. Approximate Comparative Pipe Costs

Material	ASTM Spec. and Grade	Type	Schedule	Comparative Costs (approx.) 3 in.	8 in.
Carbon steel	A 53 A or B	Seamless	40	1.0	1.0
	A 106 A or B	Seamless	40	1.10	1.10
	A 333 C	Seamless	40	1.3-1.6	1.6-2.05[2]
3½ Ni.......	A 333 3	Seamless	40	2.8-3.5	4.35-5.35[2]
Cr-Cu-Ni....	A 333 4	Seamless	40	1.65-1.85	2.6-2.9[3]
5 Ni........	A 333 5	Seamless	40	3.3-4.05	5.1-6.3[2]
9 Ni........		Seamless	40	5.65-6.8	8.75-10.5[4]
Red brass...	B 43	Seamless	Reg.	6.6	9.5[6]
			XS	10.1	14.9[6]
Copper......	B 42	Seamless	Reg.	7.6	9.5[6]
			XS	10.3	14.9[6]
Aluminum....	B 241 M1A(3003)	Seamless	40	1.7-1.9	1.9-2.1
			10	1.2	1.3
			5	0.9	1.0
	B 241 GS11A (6061-T6)	Seamless	40	2.0-2.25	2.3-2.6
			10	1.4	1.65
			5	1.1	1.2
Monel.......	B 165	Seamless	40	15.8	17.6
304.........	A 312 TP304	Seamless	40	10.1-10.7	12.9-13.6
		Seamless	10	6.0-6.35	
		Seamless	5	4.6-4.85	
		Welded and annealed	10	5.25	
			5	3.95	
		Welded and annealed[5]	40	10.3-10.9	12.3-12.9
			10		4.35-4.7
			5		3.5-3.8
304L........	A 312 TP304L		21% more than 304		
316.........	A 312 TP316		64% more than 304		
316L........	A 312 TP316L		85% more than 304		
321.........	A 312 TP321		18% more than 304		
347.........	A 312 TP347		32% more than 304		

1 Comparative costs are approximate. Comparative cost ranges indicate how price varies with the size of the order (from 20 tons to 5 tons).
2 Small extra charge for impact tests.
3 Minimum order 13 tons.
4 Minimum order 11 tons.
5 A-312 except filler metal added.
6 Based on 2,000 lb. or ft.

b. Lines. A cryogenic-fluid transfer line is generally one of three basic types: uninsulated, porous-insulated, or vacuum-insulated. The designer must select among these. The selection process includes determination of the heat transfer rate expected from bare uninsulated lines. This rate is generally available from the manufacturer. Detailed heat transfer calculation procedures are available for the complex flows to be expected in cryogenic systems (see *Cryogenic Systems* and *Analysis of Heat and Mass Transfer*); however, the aforementioned manufacturer data is usually sufficient. Some simple design guidance is offered here.

(1) *Uninsulated lines.* For short-time and short-distance transfer of liquid air, oxygen and nitrogen, uninsulated lines are the most economical and are the most frequently used. Air condensate in hydrogen lines causes an elevated heat-transfer rate. Uninsulated hydrogen lines are used only rarely and in special application.

(2) *Porous-insulated lines.* At times insulation can be applied to bare lines to reduce heat in-flow and yet still be fairly inexpensive. Typical insulations selected are glass-wool or polystyrene or polyurethene foams. A vapor barrier must be applied to the outer surfaces of the insulation to prevent water vapor diffusion into the insulation.

(3) *Vacuum-insulated lines.*

(*a*) These lines consist of an inner liquid line and an outer line with a vacuum space between the two. The annular space may be filled with additional insulation or no insulation other than the evacuated space. For long-distance, long-time transfer, the vacuum line is usually preferable, although the most expensive.

(*b*) Sections of vacuum-jacketed transfer lines require spacers to center the inner line and to support it from the outer line and to reduce stresses in the inner line produced by the weight of fluid in the line. Spacers are usually made of micarta or other insulation material.

(*c*) Desirable spacer properties are low thermal conductivity, high strength, low specific heat (to reduce cooldown losses), and a low outgassing rate.

c. Insulation. Insulation is clearly a key issue in the effective performance of cryogenic equipment. A great variety is available; the different types with their advantages and disadvantages are presented in table 4-12. Performance of these insulations is best typified by two basic parameters: apparent mean thermal conductivity and heat losses for similar insulation thicknesses. Some results are shown in table 4-13. Generally, heat flux can be calculated using the apparent thermal conductivity (see *Cryogenic Systems*):

$$\dot{Q} = \bar{k}_t (T_h - T_c)/t_i \qquad \textit{(Equation 4-4)}$$

where $\dot{Q}$ = heat flux rate in Btu/hr-ft^2;

$\bar{k}_t$ = apparent thermal conductivity in Btu/ft-°R-hr;

t_i = insulation thickness in feet;

T_h, T_c = absolute temperature of hot and cold surfaces of the insulation in °R.

Table 4-12. Cryogenic Insulation Materials

Type	Examples	Advantages	Disadvantages	Comments
Expanded foam	polyurethane; polystyrene; rubber; silicone; glass	low cost; no need for a rigid jacket	high thermal contraction; time dependent properties; highest thermal conductivity	cracking during operation
Gas-filled powders and fibrous materials	fiberglass, powdered cork, rockwool; perlite	low cost; good for irregular shaped surfaces	require a vapor barrier; powder can pack reducing effectiveness	reduces convection due to small gas voids; thermal conductivity approaches that of gas in insulation
Vacuum	evacuated annular space	lower heat flux than most others for small thicknesses; small cool-down loss; complex shapes easily insulated	requires permanent high vacuum; surfaces must have low emissitivities	radiation only mode of heat transfer; radiation shields sometimes interposed
Evacuated powder and fibers	perlite	lower heat flux than vacuum above for thicknesses greater than 4 inches; vacuum requirement less stringent; good for complex shapes	powder may pack; vacuum filters must be used to prevent system fouling	combines characteristics of gas-filled powders and vacuum above
Opacified-powder	copper flakes; aluminum flakes	better performance than evacuated powders; vacuum requirement less stringent than vacuum above; good for complex shapes	explosion hazard; higher cost than powders; settling of flakes	reduces radiation mode of heat transfer best performance when mixture is 40 to 50% metal powder by weight
Multilayer	alternate layers of reflective materials (aluminum, copper foil); low conductivity spacers such as fiber glass	best performance; low weight, lower cool-down losses; no stability problems	highest cost per unit volume; difficult for complex shapes; stringent vaccum requirements	performance depends on local evacuated pressure

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Table 4-13. Comparison of Cryogenic Insulation Performance *

Type	Apparent Thermal Conductivity (Btu/ft-°R-hr)		Heat Flux (Btu/hr-ft^2)	
	Between 540 and 140 °R	Between 140 and 36 °R	Between 540 and 140 °R	Between 140 and 36 °R
Polystyrene foam	0.019	0.0086	15.3	1.78
Gaseous conduction for He gas at 1 atm.	0.066	0.255	53	53
Gas-filled powder (perlite with He)	0.073	0.033	58.4	6.9
Evacuated perlite powder	0.00063	0.00011	0.50	0.022
High vacuum (P=10^{-6}mm)	1.0036	0.00006	2.84	0.013
Opacified powder (Cu flakes in Santocel)	0.00013		0.10	
Multilayer insulation (70 layers/inch)	0.00001	0.00001	0.0080	0.0023

* (Heat flux calculated for an insulation thickness of 6 inches in all cases)

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4-11. Fluid mechanics.

a. With single-phase flow, the relations and methodology indicated in paragraph 4-4 are applicable to cryogenic liquids. The only exception to this is that the generally short lines in cryogenic systems must be recognized. There is increased turbulence (and loss) due to entrance and exit effects through joints and valves. For this reason, the friction factor as obtained from the Moody diagram may have to be increased by approximately 25 percent in calculations.

b. Two-phase flow is not uncommon in these systems. For cryogenic liquids during the period when the piping system is being cooled, the liquid is boiling and both liquid and vapor phases are present. Once the system is cool, steady-state single-phase flow can be maintained by having the static pressure greater than the vapor phase of the liquid. This is generally achieved by increasing the pressure by means of a pump or other pressuring medium.

c. If two-phase flow does exist within the system (and there can be designs where the attendant pressure loss is acceptable) prediction of performance is difficult. Remarkably different flow patterns occur: laminar, turbulent, or mixed regimes may be present and the pattern may in fact change over the length of the piping system (fig 4-3). For detailed descriptions of these patterns and for computational techniques, the designer should consult *Piping Handbook, Cryogenic Systems,* and *Analysis of Heat and Mass Transfer.*

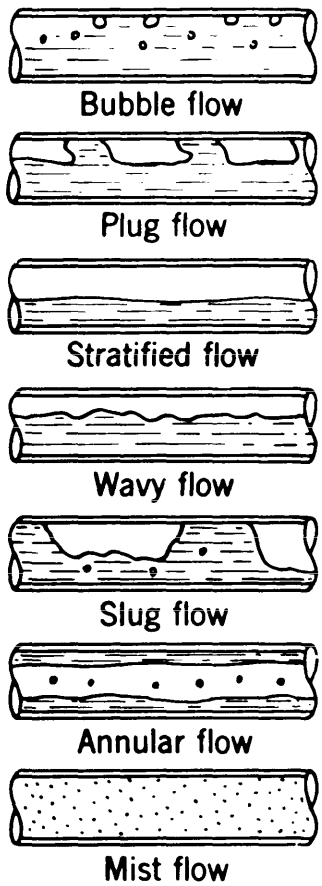

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Figure 4-3. Types of two-phase flow.

4-12. Joining methods.

Generally, what is applicable at high temperature-high pressure is also applicable for cryogenics. Thus, the design guidance offered in section II can be used except for special considerations for cryogenic systems given here. Design suggestions will be offered; the designer, however, should use these are springboards to the particular application.

a. Flanges, gaskets, and bolting.

(1) The major difference between high-pressure and cryogenic systems revolves around the selection of gasket material and type. Brittleness at low temperature becomes of great significance. Notwithstanding these problems, for runs longer than about 40 feet, it is impractical to construct the line in a single section; therefore, some means of connecting sections of vacuum-jacketed line is required. Welds are generally to be employed except where disassembly is required; then flange joints must be considered. A simple low performance connection, shown in figure 4-4, consists of a bolted-flange connection with fiberglass, foam, or powder insulation and stainless steel, aluminum, or fluoro-carbon O-ring seals.

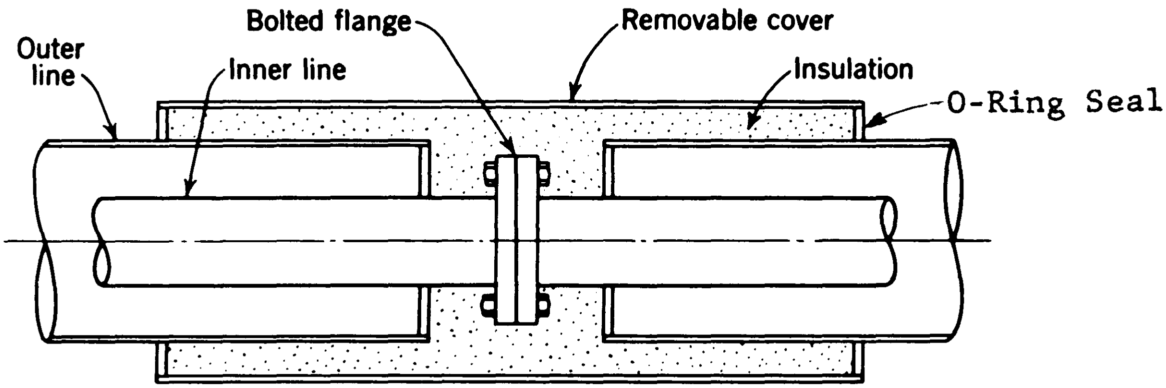

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Figure 4-4. Bolted-flange joint for vacuum-jacketed transfer lines.

(2) For better performance and for those conditions where the introduction of the cover shown in figure 4-4 is impractical, gasket materials for traditional flange types must be selected with care. A summary of gaskets in use in cryogenic systems is shown in table 4-14.

Table 4-14. Gasket Materials for Cryogenic Flanges

Material	Flange facing	Comments
Natural rubber, nitrile rubber, Viton-A, neoprene, Buna N	Tongue and groove	O-ring type gasket; compressed to 80% of original thickness; needs strong flanges, high bolt loads
Nylon, Mylar	Flat	10 mil initial thickness, compressed to 3 mils; small scratches or dents result in leaks
Metals spirotallics with 304SS using Teflon or Kel-F as filler		Large bolt loads not required
Asbestos cloths: Teflon impregnation		Generally die-cut gaskets
Cryol: sintered Teflon with glass fibers		Ring gaskets, cut from sheets.

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(3) The flange bolting is not generally a problem provided that the contraction rates of the bolting and flange are similar. Therefore, the designer will specify bolting of the same or similar material as that of the flange.

(4) A special high-performance joint, the bayonet joint (see *Cryogenic Systems*), has been developed for cryogenic service. It is used for lines which must be dismantled frequently or for applications in which the heat transfer leak of the low-performance joint cannot be tolerated. This joint has several advantages: it introduces no additional pressure drop for the flowing fluid; the heat transfer leak through the joint is minimal; the joint is easily assembled and disassembled without special tools; and there is no leakage from the joint, even under thermal cycling conditions. As with many high-performance components, the disadvantage of the bayonet joint is its high cost.

b. Welded joints.

(1) For general application, what applies to high-pressure design, likewise applies here. A special field-welded joint for vacuum-jacketed cryogenic lines has been developed and is shown in figure 4-5.

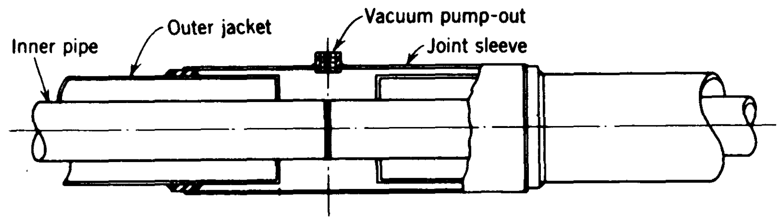

Figure 4-5. Field-welded joint assembly for vacuum-jacketed transfer lines.

(2) Lines equipped with this connection are shipped to their destination in standard sections and are welded together at the site. The welded joint has all the advantages of the bayonet joint except the ease of disassembly. It does have the significant additional advantage that the joint is not very expensive.

c. Vacuum seals. Seals are required on vacuum systems to join components, to allow electrical leads to penetrate the vacuum or to seal rotating shafts projecting into the vacuum space with minimum heat influx. Elastomer seals are widely used because of their availability and low cost. The main disadvantages of the elastomer seals are their higher permeability and outgassing and their temperature limitation. For ultra-high vacuum systems, metallic seals have been used to overcome these objections of elastomer seals. An O-ring seal (fig 4-6) is a popular type for general vacuum work. A single O-ring may be used, or a double O-ring with a pump-out connection between the O-rings could be used for better performance. For systems which require bakeout, a seal equivalent to that shown in figure 4-7 should be used. It consists of a flat copper gasket compressed between two toroidal sealing surfaces and provides a vacuum connection even after prolonged and repeated bakeout at 900°F.

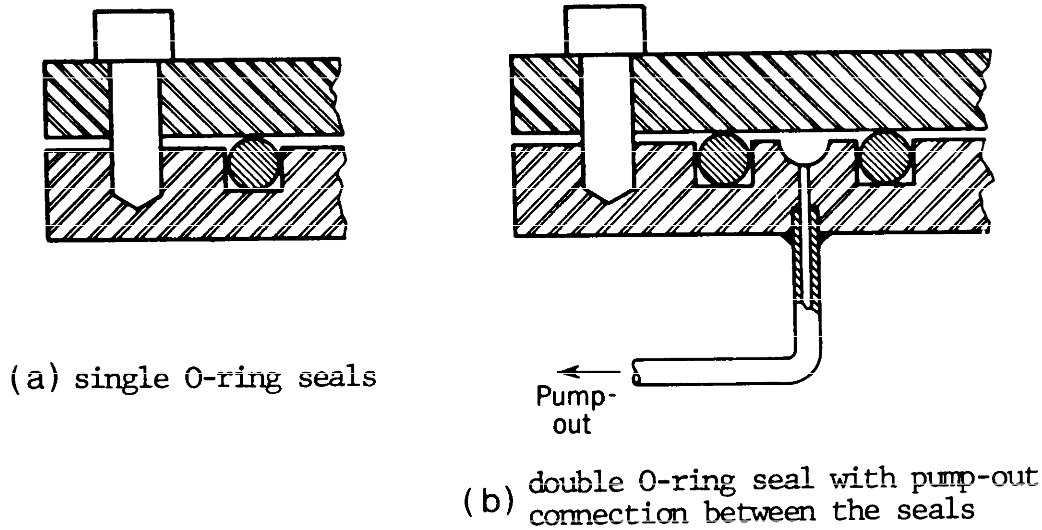

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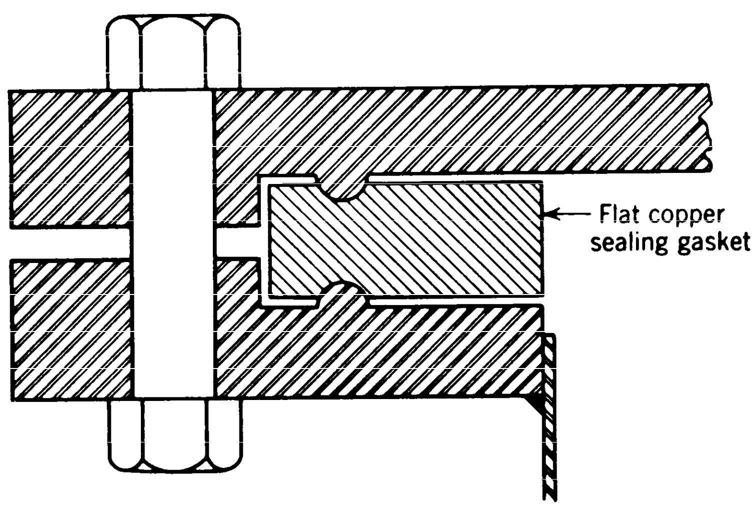
Figure 4-6. O-ring vacuum seals.



Courtesy of Ulteck Corporation, Palo Alto, California.
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Figure 4-7. Copper gasket vacuum seal.

4-13. Valves.

a. Cryogenic valves are off-the-shelf components and generally counterparts exist to those cited in table 4-6. There are special cryogenic design considerations which should be accounted for: materials should remain ductile at low temperature, materials should minimize heat influx, gaskets should be selected with the cryogenic temperature in mind. In general, cryogenic valves are of two types (see *Cryogenic Systems*): the extended-stem valve and the vacuum-jacketed valve.

b. The extended-stem valve resembles an ordinary valve, except that the valve stem is modified by extending the stem about 10 or 12 inches through the use of a thin-walled stainless steel tube. The extension of the valve stem serves two purposes: the valve handle is maintained at ambient temperature to protect the operator from low temperatures, and the valve stem may be sealed at ambient temperatures instead of cryogenic temperatures, thereby eliminating a severe sealing problem and improving the reliability of the valve.

c. The vacuum-jacketed valve is an extended-stem valve with a vacuum jacket around the extended stem and valve body to reduce heat transfer. This high-performance valve usually has a specially made body with thin-walled sections in order to minimize the cooldown loss for the valve. The valve is designed so that the valve stem, valve plug, and valve seal can be removed for inspection and servicing without breaking the vacuum of the line. The valve body is also insulated with multilayer insulation to further reduce heat transfer.

Section IV. FLEXIBILITY AND SUPPORT

4-14. Flexibility.

a. Single-plane, two-anchor systems can be analyzed for thermal expansion stresses, movements, and anchor reactions by any accepted method. Multiplane systems are best analyzed by one of the available computer programs.

b. Under the effect of changes in temperature of the pipeline, or of movement of support reaction, or both, the determination of the stress distribution in a pipe or pipeline leads to the nature of the support required and to the necessary flexibility required of the pipeline system. The following design principles should be adopted when laying out the original system (see *Marks Standard Handbook*):

—Avoid expansion bends where possible, and design the entire pipeline to take care of its own expansion.

—The movement of the equipment to which the ends of the pipeline are attached must also be included in the analysis.

—Maximum flexibility is obtained by placing supports and anchors so that they will not interfere with the natural movement of the pipe.

—The shape that is most efficient is one in which the maximum length of pipe is working at the maximum safe stress.

—Excessive bending moment at valves and like components is more likely to cause trouble than excessive stresses in pipe walls. Hence, components should be kept away from point of high moment.

—Reactions and stresses are greatly influenced by flattening of the cross section of the curved portions of the pipeline.

c. Pipeline flexure stresses that result from movement of supports or from the tendency of the pipes to move under temperature change often may be avoided entirely through the use of expansion joints. Their use may simplify both the design of the pipeline and the support structure. When using expansion joints, the following suggestions will be considered:

(1) Select expansion joints carefully for maximum temperature range (and deflection) expected so as to prevent damage to expansion fitting.

(2) Provide guides to limit movement at expansion joint to direction permitted by joint.

(3) Provide adequate anchors at one end of each straight section or along their mid-length, forcing movement to occur at expansion joint yet providing adequate support for pipeline.

(4) Mount expansion joints adjacent to an anchor point to prevent sagging of the pipeline under its own weight. Expansion joints cannot be depended upon for stiffness; they are intended to be flexible.

(5) Give consideration to effects of corrosion, since the corrugated character of expansion joints makes cleaning difficult.

d. Furthermore, piping systems must be designed so that they will not fail because of excessive stresses, will not produce excessive thrusts or moments at connected equipment, and will not leak at joints because of expansion of the pipe. Because of geometry of the layout, it is often unnecessary to use either expansion joints or corrugated sections of piping to provide sufficient flexibility to absorb thermal strains.

e. Because of plastic flow of the piping material, hot stresses tend to decrease with time while cold stresses tend to increase with time; their sum, called the stress range, remains substantially constant.

f. An allowable stress range, S_A, is calculated from

$$S_A = f(1.25 S_c + 0.25 S_h) \qquad \textit{(Equation 4-5)}$$

where S_c and S_h are the stress values for the cold and hot conditions and may be found in table 4-2 and tables 4-7 through 4-10, or in the ANSI B31.1. The stress reduction factor, f, is a function of the number of full temperature cycles (hot-to-cold-to-hot) anticipated over the life of the system; some typical values are shown in table 4-15 and other values may be found in the *Piping Handbook* and

ANSI B31.1. The actual expansion stress, S_E, on the system may be found, as noted earlier, either from a complex calculation (see *Marks Standard Handbook* and *Piping Handbook*) or from manufacturer guidance. Then, the design criteria is simply that

$$S_E \leq S_A \qquad \textit{(Equation 4-6)}$$

Table 4-15. Values of Stress Reduction Factor

Total number of full temp cycles over expected life	Stress-reduction factor, f
7,000 and less	1.0
★ 7,000 – 14,000	0.9
14,000 – 22,000	0.8
22,000 – 45,000	0.7
45,000 – 100,000	0.6
100,000 – over	0.5

From American National Standards Institute (ANSI) Code B31.1, Power Piping. Used with permission of American National Standards Institute.

g. For vacuum-jacketed cryogenic systems, the differential rates of expansion and contraction between inner and outer pipes must be taken into account. This can be treated in cryogenic transfer-line design through the use of expansion bellows and U bends. Unless other considerations make inner-line bellows necessary, it is good practice to locate the expansion bellows only in the outer line and to attain flexibility of the inner line through the use of U bends, especially for large-diameter lines with high working pressures.

h. Some useful simple criteria have been developed over the years to determine if a pipeline has adequate flexibility and if additional analysis may be required. While the theoretical bases for these formulas are weak and while they may fail to give the correct result (that is, suggest inadequate flexibility when the pipeline does in fact have sufficient capability), the ANSI Code does recommend their employment.

(1) According to the first of these, a piping configuration may be considered to be adequately flexible to prevent damage to the pipe itself if the inequality

$$DY/(L - U)^2 \leq 0.03 \qquad \textit{(Equation 4-7)}$$

is satisfied. In this formula:

L = total length of pipe centerline, in feet.
U = straight-line distance between anchors, in feet
D = nominal pipe diameter, in inches.
Y = geometrical resultant of expansion and terminal movements, in inches.

The different units here should be noted. Note further that this deals only with the integrity of the pipe and cannot be used to estimate reaction ranges or reactions on terminal equipment.

★ (2) Another criterion has been cited in the Piping Handbook. Within a wide range of geometrical ratios, a fairly inclusive variety of two-anchor configurations may be conservatively evaluated by use of the criterion

$$L/U = 1 + 6\,(L^*/U)^{1/2} \qquad \textit{(Equation 4-8)}$$

where, in addition to the symbols given above, the quantity L^* is a fictitious length defined by the formula

$$L^* = (E/S)\,(Y/U)\,(D/12) \qquad \textit{(Equation 4-9)}$$

E and S denote Young's modulus and/allowable stress, which, within the framework of the ANSI B31.1 Code, should be interpreted as Young's modulus at the cold temperature, and allowable stress range S_A. Then as long as the actual total length is at least as great as $U\,[1 + 6\,(L^*/U)^{1/2}]$, the piping is not overstressed.

(3) Both these criteria generally give quite conservative results. Accordingly, it is a common experience that more exact analyses of systems which fail to satisfy one or both may still reveal that the system is indeed quite satisfactory. In order to employ any of these criteria, it is necessary to know the sum of the expansion and terminal movements (equations 4-7, 4-8, and 4-9). These can generally be estimated from thermal expansion data, typical of which are presented in table 4-16, and from manufacturer specifications.

Table 4-16. Unit Linear Thermal Expansion (in/100 ft) †

Material	-325	-150	-50	70	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400
				Temperature range 70 F to													
Carbon steel; carbon-moly steel; low-chrome steels (through 3% Cr)	-2.37	-1.45	-0.84	0	0.99	1.82	2.70	3.62	4.60	5.63	6.70	7.81	8.89	10.04	11.10	12.22	13.34
Intermediate alloy steels (5 Cr Mo through 9 Cr Mo)	-2.22	-1.37	-0.79	0	0.94	1.71	2.50	3.35	4.24	5.14	6.10	7.07	8.06	9.05	10.00	11.06	12.05
Austenitic steels	-3.85	-2.27	...	0	1.46	2.61	3.80	5.01	6.24	7.50	8.80	10.12	11.48	12.84	14.20	15.56	16.92

† These data are for information, and it is not to be implied that materials are suitable for all the temperature ranges shown.

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Table 4-17. Maximum Spacing of Pipe Supports*

Nominal pipe size, in............	1	1-1/2	2	3	4	6	8	10	12	14	16	18	20	24
Maximum span ft..................	7	9	10	12	14	17	19	22	23	25	27	28	30	32

*This tabulation assumes that concentrated loads, such as valves and flanges, are separately supported. Spacing is based on a combined bending and shear stress at 1,500 psi when pipe is filled with water; under this condition, sag in pipeline between supports will be approximately 0.1 in.

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4-15. Supports, guides and anchors.

a. After the design of the system for flow rate, pressure losses and appropriate materials handling, the supports, guides and anchors must be selected. They must support the weight of the pipe, the fittings, insulation, the weight of the fluid being carried, and be capable of controlling the direction and amount of distortion arising from the analysis of pipeline flexibility. Of prime consideration in hanger design is the determination of hanger location. This depends on the pipe size and configuration, the location of heavy valves and fittings, and the structure that is available for support. No firm rules or limits exist which fix the location of each support on a piping system. Suggested maximum spacings are displayed in table 4-17. These do not apply where concentrated weights (such as valves and fittings) or changes in direction of the system occur. Note that supports should be placed as close as possible to concentrated loads in order to keep bending stresses to a minimum. Where changes in direction occur between hangers, it is generally good practice to keep the total length between supports less than three-fourths the spaces listed in table 4-17.

b. The selection of hangers for a particular job depends on the loads and movements to be absorbed by the system. The movement at each point dictates the basic type required. The design should take full advantage of commercially available load-rated and load-tested components. Most manufacturers offer load capacities as part of their design specification. The actual calculation for the movements and loads are complex; some comments are in order. First, a considerable number of examples are available in the *Piping Handbook*. Second, simple beam analysis is sufficiently conservative to give useful results to the designer.

c. Manufacturers Standardization Society (MSS) of the Value and Fittings Industry Standard Practices, MSS SP-58 and SP-69, list requirements for supports, hangers, saddles, inserts, clamps, and pipe rolls including rods, bolts, turnbuckle bases and protection shields. The designer will consult the above MSS Standard Practices or the *Piping Handbook* for the applicability of these various types.

CHAPTER 5
STORAGE VESSELS

Section I. INTRODUCTION

5-1. General. In all cases the American Society of Mechanical Engineers (ASME) Code for Boiler and Pressure Vessels will be used as the standard for design, manufacture, and installation of storage vessels. Pressure vessels are produced to Code specifications by a variety of manufacturers and, thus, should be purchased to specifically meet the needs of the design.

5-2. Loading. The loading conditions which must be considered in the design will include but not be limited to:

—Internal and external pressure, including static head.

—Weight of vessel and normal contents under operating or test conditions.

—Superimposed loads such as other vessels, operating equipment, insulations, corrosion-resistant or erosion-resistant lining and piping.

—Wind loads, snow loads and earthquake loads.

—Reactions of supporting lugs, rings, saddles, or other types of vessel supports.

—Impact loads, including rapidly fluctuating pressure.

—Temperature conditions, introducing differential loadings and reactions resulting from expansion or contraction of attached piping or other parts.

Section II. HIGH-PRESSURE GAS SYSTEM

5-3. General. Although all high-pressure gas vessels shall be designed, manufactured, and installed in accordance with the American Society of Mechanical Engineers (ASME) Code for Boiler and Pressure Vessels, several important design considerations are noted.

a. The first item of importance to consider is the minimum thickness requirements for the pressure vessels. The Code specifies thickness as a function of design pressure, shell radius, maximum allowable stress, and weld efficiency.

b. Specific requirements on nozzle design are developed within the Code. These requirements are concerned with shapes of openings, sizes of openings, strength, reducer sections, and reinforcement. The designer of such transition sections is urged either to closely follow Code requirements or specify such requirements for the manufacturer.

c. All vessels must be supported and the supporting members arranged and attached to the vessel wall in such a way as to provide for the maximum imposed loadings. The design should conform to good structural practice keeping in mind, among other things, prevention of excessive localized stresses due to temperature deformations, stiffening effects, prevention of high secondary stresses from column supports, and skirt reaction.

Section III. CRYOGENIC SYSTEMS

5-4. General.

a. Cryogenic liquid vessels consist of an inner vessel which encloses the cryogenic fluid to be stored. This is enclosed by a vessel or vacuum jacket which contains the high vacuum necessary for the effective insulation and which serves as a vapor barrier. The essential elements are shown in figure 5-1.

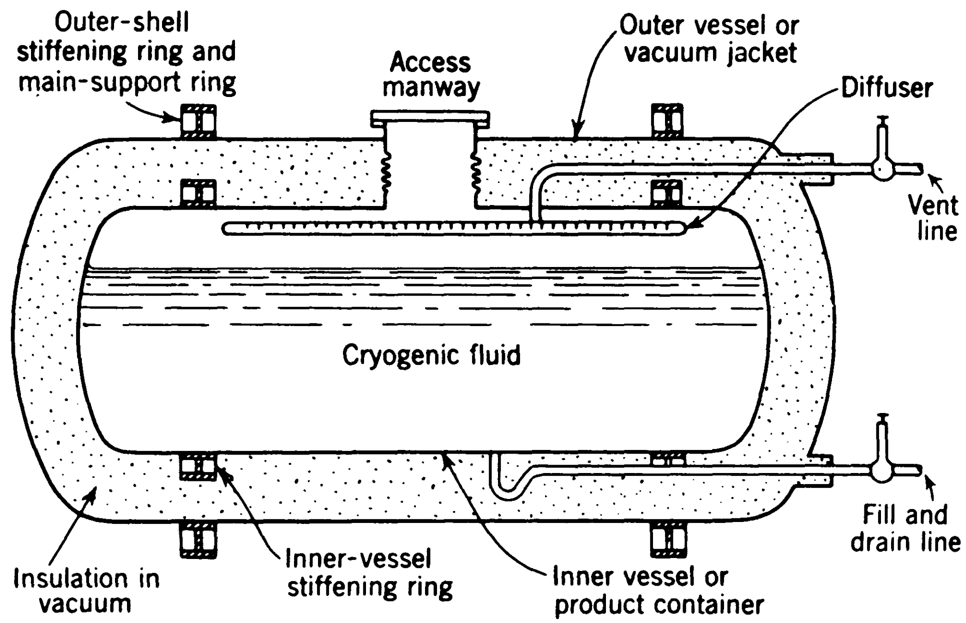

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Figure 5-1. Typical cryogenic vessel.

b. This section on cryogenic systems is ordered somewhat differently than the parallel work on high-pressure gases for several reasons. First, the ASME Pressure Vessel Code does not explicitly call out cryogenics.

c. As will be noted later, outer vessel calculations should be made under the rules generally described by "vessels with external pressure." Second, concepts and design guidance are usually offered in the specialized references on cryogenic systems. The primary source here will be *Cryogenic Systems*. However, the Codes ultimately control the design.

5-5. Inner vessel design.

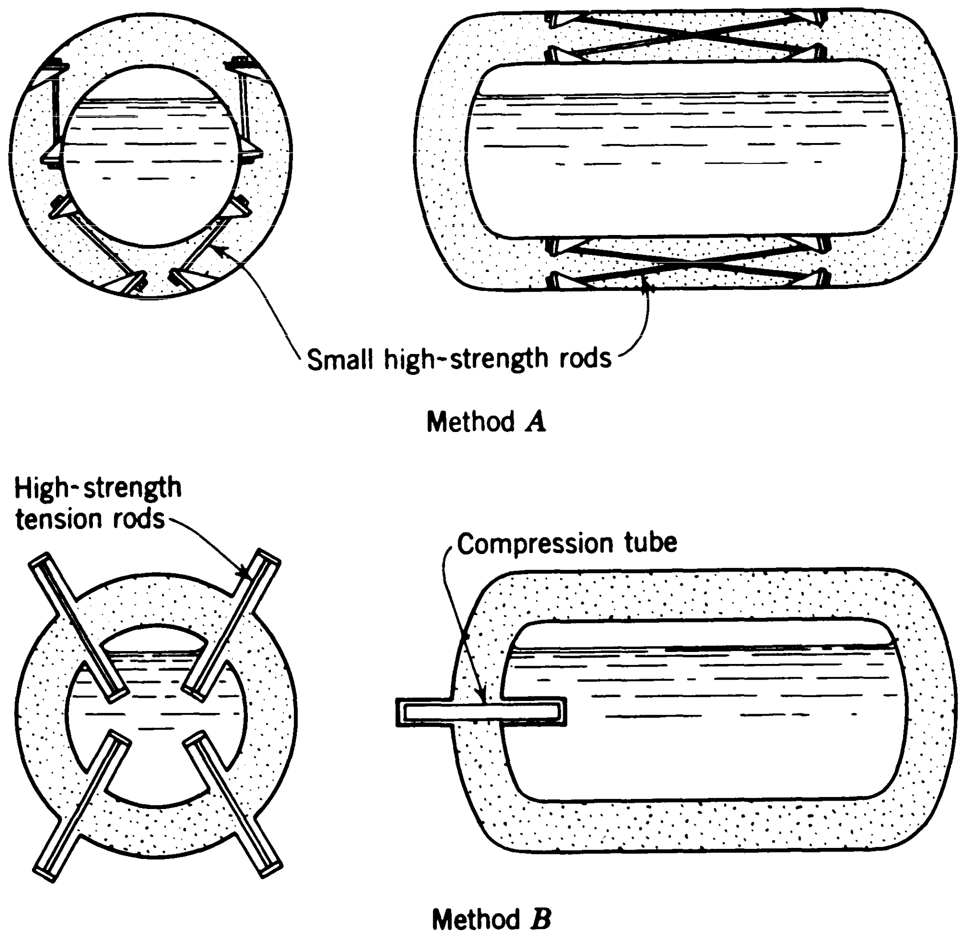
a. The inner vessel is designed according to the rules for vessels under internal pressure loading. The only difference is in capacity specification. Cryogenic fluid storage vessels are not designed to be completely filled. Heat transfer to the inner vessel is always present; therefore, the vessel pressure would rise quite rapidly because of liquid evaporation if no vapor space were provided. Further, inadequate cooldown during a rapid filling results in additional boil-off, and the liquid percolates through the vent tube if no ullage is provided. This generally occurs when the vessel is approximately 90 percent full. For this reason, a 10 percent ullage is customarily used for large storage vessels.

b. Insulation to be provided between the inner and outer vessel is discussed in paragraph 4-10. The design guidance offered for piping systems there applies equally well to the pressure vessels.

5-6. Outer shell design. The outer shell design will follow the rules for failure by elastic instability rather than excessive stress (as in the inner pressure vessel). In either Division 1 or Division 2 thickness calculations are an iterative procedure. In Division 1 special attention is given to jacketed vessels. Within this part of the Code, joints and closures are also described. Some simplified relations satisfying the Division 1 rules have been developed in *Cryogenic Systems*. The designer should consult these in conjunction with the Code.

5-7. Suspension systems.

a. Some commonly used suspension systems are tension rods of high-strength stainless steel, saddle bands of metal or plastic, plastic compression blocks, multiple-contact supports (stacked disks), compression tubes, and wire cables or chains. The inner vessel suspension system loading results from a combination of the weight of the vessel and its contents and from acceleration forces arising from, for example, transportation and earthquake. The support members should be constructed of a material with a low thermal conductivity so that heat transferred down the support member is minimized, and the length of the support members should be as long as possible for the same reason. If the available space within the vacuum space is not large enough to accommodate a long suspension member, standoffs may be added to the outer and/or inner vessels. Design ideas for suspension are shown in figure 5-2. Method B uses a combination of high-strength tension rods and compression tubes.



From Cryogenic Systems by Randall Barron, Copyright 1966, McGraw-Hill Book Company. Used with permission of McGraw-Hill Book Company.

Figure 5-2. Typical methods of supporting the inner vessel within the outer vessel.

b. Suspension materials must take account of both thermal conductivity (which should be low) and strength (which should be high). The material with the largest strength-conductivity ratio should be selected as a first choice. Other considerations, such as ease of fabrication and cost, could affect the final choice for a support-member material; however, a material with a fairly large strength-conductivity ratio should still be selected to minimize the heat transfer down the member. Some values of strength-conductivity ratio for some materials which could be used as support members are given in table 5-1.

Table 5-1 Strength-Conductivity Ratio for Various Materials.

	Yield Strength psi	Thermal conductivity,* Btu/hr-ft-°F	Strength-conductivity ratio
Teflon	3,000	0.895	3,350
Nylon	10,000	0.913	11,930
Mylar	40,000	0.566	70,700
Dacron fibers	88,000	0.566	155,500
Kel F oriented fibers	30,000	0.224	134,000
Glass fibers	130,000	2.87	45,300
304 stainless steel (c.d.) †	91,000	7.30	12,479
316 stainless steel (c.d.) †	119,000	7.30	16,300
347 stainless steel (c.d.) †	128,000	7.30	17,540
1100-H16 aluminum	19,700	143.0	138
2024-0 aluminum	11,000	52.0	212
5056-0 aluminum	20,000	67.6	296
K Monel (45% c.d.)+	93,000	10.96	8,480
Hastelloy C (annealed)	55,000	7.27	7,560
Inconel (c.d.)+	45,000	7.85	5,730

* Values of thermal conductivity are mean values between room temperature and liquid-oxygen temperature.

† Cold-drawn

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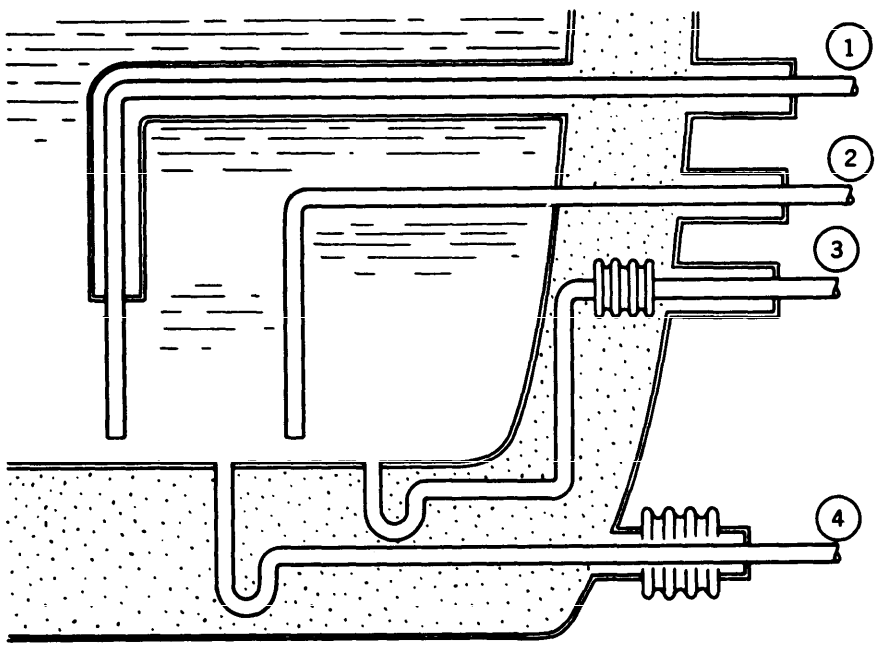
c. In addition to minimizing heat transfer down the support members, the designer must also be aware of the problem of thermal contraction of the support members. Both the inner vessel and the support members will contract during cooldown; differential rates may cause undue stress in the system.

5-8. Transfer.

a. Special considerations are necessary for access to the liquid through the double-vessel typical of cryogenic systems. Four arrangements are shown in figure 5-3.



From Cryogenic Systems by Randall Barron, Copyright 1966, McGraw-Hill, Book Company. Used with permission of McGraw-Hill Book Company.

Figure 5-3. Cryogenic-fluid storage vessel piping arrangement.

b. Arrangement 1 is commonly used for multilayer-insulated vessels. Arrangement 2 is unsatisfactory. Condensation of vapor will take place along the top of the horizontal portion of the line which is exposed to fluid within the inner vessel. This liquid will flow along the horizontal section of the line to the warmer portion of the line, at which point the liquid will evaporate. Heat will be transferred to the inner container by the quite efficient boiling-condensation convection process, and a large heat-transfer rate will result. In addition, no provision is made for thermal contraction of the line; therefore, high thermal stresses will develop in the line. In arrangement 3 a vertical rise in the vacuum space is used to prevent the convection problem present in arrangement 2, and an expansion bellows is introduced to allow for thermal contraction. Arrangement 3 presents a serious problem in reliability of the container, however, and it should be avoided if possible. If a leak into the vacuum space should occur, it would most likely occur in the bellows because of fatigue loading of the bellows during repeated cooldown. With the bellows joint located as in arrangement 3, it is quite difficult and costly to repair. The best piping arrangement is shown in arrangement 4, in which the expansion bellows is located in the standoff, where the bellows is easily accessible for repair. A vertical rise is also incorporated in arrangement 4 to prevent percolation of the liquid within the line.

5-9. Access manways.

a. Access manways are frequently installed in large (500 gallon or more) cryogenic-fluid storage vessels in order that the inner shell can be inspected, cleaned, repaired, or modified without cutting through the outer shell. There are two common types of access manways: the unconnected manway and the connected manway.

b. For the unconnected manway, the vacuum must be broken and some of the insulation removed to gain access into the inner vessel. This arrangement has the advantage that the manway introduces no additional heat transfer into the inner vessel. The unconnected manway is used for long-term storage applications in which the manway will seldom be used. The connected manway has the advantage that persons can enter the inner vessel without breaking the insulating vacuum. An expansion bellows is used to seal the vacuum space, increase the heat-transfer length of the manway connection, and allow for thermal contraction of the inner vessel. This arrangement is employed for applications in which the manway is expected to be used frequently.

5-10. Draining the vessel. Cryogenic system maintenance generally requires draining the vessel periodically for cleaning and inspection. One of three methods is commonly used to force the liquid from the inner vessel: self-pressurization of the inner vessel, external gas pressurization of the inner vessel, or pump transfer. Self-pressurization involves removing some of the liquid from the product container and boiling the liquid in an external evaporator or pressurization coil. External gas pressurization involves introducing high-pressure gas from an external source into the inner vessel to increase the vessel internal pressure. In the case of pump transfer, the storage vessel is not pressurized, but a cryogenic pump in the liquid drain line is used to remove the liquid. Pump transfer is often used when large liquid flow rates are desired. Design of the system must provide for draining the vessel.

Section IV. SAFETY DEVICES

5-11. Introduction. All pressure vessels designed according to Section VIII of the Code, irrespective of sizes or pressure, must be provided with safety devices according to very rigid requirements. In this regard high-pressure gas and cryogenic designs are similar. Furthermore, Division 1 and 2 rules are similar enough so that the discussion herein can be presented without distinction between the two.

5-12. Requirements.

a. With regard to set or operating pressure, the Code requires protection from pressures rising more than 10 percent above the maximum working allowable pressure. With multiple valves, the second or third valve can be rated higher. Where additional hazards can be created from unexpected sources, supplemental devices rated generally at 21 percent above the maximum allowable pressure should be installed. These devices should be constructed, located, and installed so that they are readily accessible for inspection and repair. The Code specifies spring tolerances within the several classes of devices. Seals must be considered both from the point of view of operation of the pressure vessel and the safety device.

b. The Code requirements on materials are quite severe and properly so. For example:

—Cast-iron seats and disks are not permitted.
—Sliding surfaces should be of corrosion-resistant material.
—Springs should be of corrosion-resistant material.
—ASME Code Section II and Section VIII materials are to be used in both bonnets and yokes.

If the designer is to generate new devices, a copy of the Code is essential; if the designer is to purchase on-the-shelf items, then Code certification is the key to utilization.

5-13. Types.

a. There are two general classes of safety devices: pressure-relief valves and rupture disks. Within each class is a variety of particular types for individual use.

b. For high-pressure gas systems the arrangement with the inner shell burst-disk assembly and the pressure-relief valve located between the inner vessel and the vent value in the vent line is usually adequate.

c. The inner-vessel pressure-relief valve is generally a spring-loaded safety valve set so that the pressure within the inner vessel can never exceed the design pressure by more than 10 percent as noted earlier. This safety device is installed so that excessive pressures within the inner vessel can be relieved before damage to the vessel occurs. Pilot valve control for these safety valves it not permitted unless special features are built into the system.

d. The inner shell rupture disk is an additional or redundant safety device which is used in the event that the safety valve does not open or does not have sufficient capacity to relieve the inner-vessel overpressure. The burst-disk assembly is placed in parallel with the safety valve and is selected so that the disk ruptures at a pressure approximately 20 percent above the design pressure of the inner vessel. If the safety valve opens, it may be reset after the overpressure has been relieved; however, the rupture disk must be replaced after it has been ruptured.

e. Of importance in the design is the fact that the system, with the rupture disk and relief and vent valves, acts in sequence. Considerations then must include assurance of sufficient flow area to the relief valve even after disk rupture; of control pressure build-up between the disk and valve; of pressure readings in the area between the two devices; and of provision for operation if leaks should occur in either of the housings. Further details are available in the Code.

f. The annular-space rupture disk is used to protect the outer vessel from excessive internal pressure and to prevent collapsing of the inner vessel due to excessive external pressure. Excessive annular-space pressure occurs if a leak develops in the inner or outer vessel. When the vessel is warmed up to make repairs, the cold gas in the annular space is warmed and expands, thereby increasing the pressure within the annular space. Note then that rupture disks can be used to relieve excessive pressure within an inner vessel or to limit the build-up of pressure within the annular space designed as a vacuum chamber.

CHAPTER 6
INSTRUMENTATION AND CONTROL

Section I. INTRODUCTION

6-1. General.

a. In the processes under consideration, measurement and control of temperature, pressure and flow rate determine the quality and efficiency of operation. The purpose of instrumentation and process control devices is to maintain the operation of the system at an efficient and safe level. Only general guidance is offered here since instrument selection is vitally dependent on the application and since the system designer will be required to make very specific choices.

b. The devices or sensors provide qualities and quantities of interest, and regulate signals for control. Control can be manual or automatic. Automatic control is now more standard. Obviously, the control systems are dependent on the measured signal; the quality of this measurement is highly dependent on the tapping process. A number of general features regarding tapping (see *Marks Standard Handbook*) into the piping system should be enumerated. The tapping point should be at a convenient location with respect to access. Poor location leads to difficult servicing. Temperature sensors are preferably installed where turbulence is at a maximum. Pressure sensors, conversely normally require a straight length of pipe, with relatively quiet flow.

c. Response of the instruments is crucial since no control mechanism can act until it has received instructions. The longer the response time, the greater the departure from normal which the control system has to correct.

d. With these introductory remarks the designer can begin to understand some of the instrumentation and control problems. It should be noted, however, that manufacturers will often provide complete data on selected instrumentation. Generalized data are available, for example, in the *Process Instruments and Controls Handbook.*

Section II. DESIGN GUIDANCE FOR MEASUREMENT DEVICES

6-2. Temperature. Several classes of temperature indicating devices are available: thermocouples, thermoresistive systems, filled-system thermometers, liquid-in-glass thermometers, pyrometric systems, and bi-metal thermometers. All have different applicability, ranges, and accuracy. For ready reference, a summary of pertinent quantities of interest is presented in table 6-1. Table 6-2 compares these various classes of devices.

Table 6-1. Temperature Indicating Devices.

Type	Practical Range (°F)	Accuracy	General Remarks (See reference 10 for specific data)
Thermocouples			
General	-450 to 4000	1°F	No ideal construction available; characteristics of concern: temperature and instrument range, corrosion, output, linearity, insulation.
Cryogenic			
Gold Cobalt/Copper	-450 to 90	0.2°F	
Constantan/Copper	-420 to 600	0.2°F	
Resistance Thermometers			
General	-420 to 1500	0.1%-0.5%	Resistance varies significantly with temperature and range; uses bridge circuits; cost of wire significant; stability, response time, and self-heating important.
Cryogenic			
Platinum	-440 to 1200	10^{-2}-10^{-4}°F	
Carbon	-455 to -410	10^{-2}-10^{-3}°F	
Germanium	-455 to -280	10^{-2}-10^{-3}°F	
Thermistors	-450 to 600	0.5°F	Semi-conductor metal oxides; uses bridge circuits; cost of wire significant; stability, response time, and self-heating important.
Filled System Thermometers (Gas Only)			
General	-300 to 1200	2%	Several classes of systems available; see Scientific Apparatus Manufacturers Association standards for different compensation schemes.
Cryogenic			
Helium	-455 to -450	10^{-3}°F	
Hydrogen	-435 to -400	10^{-2}°F	
Nitrogen	-345 to -233	10^{-2}°F	
Oxygen	-360 to -180	10^{-2}°F	
Liquid-in-Glass Thermometers	-200 to 600	.5%	Temperature and range important; immersion characteristic leads to design type.

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Table 6-2. Comparison of Temperature Instruments.

Type	Advantages	Disadvantages
Thermocouples	Good reproducibility Small size Wide variety of design Easily adaptable to output or control devices High response speed Low cost Good accuracy Easy to calibrate Transmission distance can be long	Non-linear temperature/voltage relationship Accuracy less than resistance thermometer Stray voltages are a problem Wide temperature spans Require amplifiers Aging of junctions Accessories expensive
Resistance Thermometers	High accuracy Narrow spans Good reproducibility Remain stable and accurate for years Temperature compensation not necessary Fast response time	Relatively expensive compared to thermocouples Bulb sizes longer than thermocouple Self heating can be a problem Mechanical abuse or vibration a problem
Thermistors	Fast response Good for narrow spans Low cost Cold junction compensation not necessary Negligible leadwire resistance Stability increases with age Available in small sizes Adaptable to various electrical readout devices	Nonlinear temperature versus resistance curve Not suitable for wide temperature spans Interchangeability of individual elements often a problem Experience limited for process application
Filled System Thermometers	Simple, time-proven measurement method Relatively low cost No outside source of power required Good selection of calibrated charts available Ruggedly constructed Presents no electrical hazards in hazardous atmospheres	Limited to measurements below 1,500°F Relatively slow response (fast enough for most applications, however) Bulb failure requires replacement of entire thermal system Transmission distances more limited than for electrical systems
Liquid-In-Glass Thermometers	Low cost Simplicity of use Ease of checking for physical damage No need for auxiliary power	Very fragile Range limitation at high end Cannot be adapted to remote readout

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6-3. Pressure. Within high-pressure systems, pressure can exceed 10,000 psig and within the vacuum insulation of cryogenics, pressure can be lower than 10^{-3} torr. Thus, a wide variety of instruments must be considered. Two basic types are available and within these two are several classes particularly applicable to the system at hand:

—Mechanical pressure elements: liquid column devices, elastic elements, diaphragms, bellows, and bourdon springs.

—High-vacuum measurement: some from the mechanical pressure types listed plus McLeod, Knudsen, conductivity and ionization gauges.

A summary chart is displayed in table 6-3; comparisons are shown in table 6-4.

Table 6-3. Pressure Indicating Devices.

Type	Practical Range	Accuracy (of full span)	General Remarks (See reference 10)
Liquid Column Devices	1 to 120 psia	1%	Traditional manometer; best for differential pressure.
Diaphragms			
High pressure	1 to 10^3 psia	1%	Direct conversion to electric signal possible.
Vacuum	10^{-4} to 100 torr	0.5%	
Bellows			
High pressure	1 to 10^3 psia	0.5%	Adaptable to control because of movement.
Vacuum	10^{-1} to 100 torr	2%	
Bourdon (C, Spiral, Helical)			
High pressure	1 to 10^5 psia	2%-5%	Excellent range; can be converted to force balance; number of turns in windings, spring material, corrosion are critical characteristics.
Vacuum	1 to 100 torr	10%	
McLeod	10^{-5} to 10^{-1} torr	1%	Pressure read in absolute terms; gas must obey Boyle's Law; intermittent.
Knudsen	10^{-5} to 10^{-1} torr; can go as low as 10^{-8} torr.	10%	Independent of gas composition.
Conductivity	10^{-3} to 1 torr	0.5%	Contact surface problems crucial
Ionization	10^{-11} to 10^{-3} torr	0.5%	Similar to a triode.

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Table 6-4. Comparison of Pressure Instrumentation.

Type	Advantages	Disadvantages
Liquid Column Devices	Simple and time proven High accuracy and sensitivity Wide range of filling fluids of varying specific gravities Moderate cost Particularly suitable for low pressure and low differential pressure applications	Lack of portability Need of leveling Generally large and bulky Measured fluid must be compatible with the manometer fluid used No overrange protection Condensation may present problems
Diaphragms	Moderate cost High overrange characteristics Linearity good Adaptability to absolute and differential pressure measurement Available in several materials for good corrosion resistance Small in size Adaptable to slurry services	Does not have good vibration and shock resistance Difficult to repair Limited to relatively low pressures (except for button diaphragm type) May require pumping system
Bellows	High force delivered Moderate cost Adaptable for absolute and differential pressure use Good in the low-to-moderate pressure range	Need ambient temperature compensation Not suited for high pressures Limited in the availability of metal - some tend to work-harden Require spring for accurate characterization
Bourdon	Low cost Simple construction Years of experience in application Available in a wide variety of ranges, including very high range Adaptable to transducer designs for electronic instruments Good accuracy, especially when considered in relation to cost	Low spring gradient below 50 psig Susceptible to shock and vibration Subject to hysteresis Poor accuracy at low pressures
McLeod	Read absolute pressure Good for calibration of other instruments	Fluid being measured must obey Boyle's Law Intermittent readings only Remote readout difficult Seals very important
Knudsen	Useful for extremely low pressure	Electronics complex Expensive Extremely delicate
Conductivity	Excellent accuracy below 1 torr	Surface conditions are problems Non-linearity Calibration problems Compensation for ambient temperature needed
Ionization	Excellent range for vacuum pressure	Extremely expensive Composition dependent Must have low pressure Corrosion of filaments Electronics very complex

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6-4. Flow. Flow meters may be classified in numerous ways. For practical purposes the major categories may be enumerated in terms of head, area, positive-displacement, and velocity. As in earlier sections, for convenience, the basic characteristics of these devices are displayed in table 6-5 and a comparison given in table 6-6.

Table 6-5. Flow Meters

Type	Rangeability	Accuracy	General Remarks
Head	3:1 to 5:1	1/4% to 2%	Careful line tapping required; flange taps generally used; calibration very important; placement generally at end of long straight runs of pipe.
Area (Rotameter, orifice, piston)	10:1	1/2% to 10%	Float design and material important considerations; fluid properties must be taken into account for operation and calibration.
Positive Displacement			
General	Up to 30:1	1% to 2%	Requires addition of time base; can be used with all types of fluids; can be used as a controller as well.
Nutating Disk	Up to 30:1	1%	
Oscillating piston	Up to 30:1	1/4% to 1/2%	
Velocity			
Turbine	10:1 to 20:1 can go to 100:1	1/4% to 1/2%	Calibration of rotating element very important; fluid properties must be taken into account.
Vortex	100:1	1/2% to 1%	Rotor material important.

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Table 6-6. Flow Meter Comparison.

Type	Advantages	Disadvantages
Head	Relatively low cost, especially for large lines Proven accuracy and reliability, well known and predictable flow characteristics Easily removed without shutting down the process The secondary or differential device easily isolated for zero check and/ or calibration Adaptable to any pipe size Cost remains almost the same with increasing pipe size - the cost of the primary elements (orifice plates) and flanges are the primary variables	Relatively high permanent pressure loss (considering the thin edge orifice plate) Difficult to use for slurry services. Square root rather than linear characteristic Meter run lengths required for accurate measurement sometimes present difficulties Connecting piping sometimes presents problems-freezing, condensation, etc. Low flow rates not easily measured except through the use of integral orifice devices where in-line mounting is required Accuracy dependent on many fixed fluid characteristics such as temperature, pressure, specific gravity, compressibility, etc. Pulsating flow difficult to measure
Area	Good rangeability Relatively low cost Good for metering small flows Easily equipped with alarm switches No restrictions in regard to inlet and outlet piping requirements (other than a vertical flow requirement) Viscosity-immune designs available Low pressure drop requirement Can be used in some light slurry services	Glass tube types subject to breakage Not good in pulsating services Must be mounted vertically Generally limited to small pipe sizes (unless bypass rotameter is used) Limited to relatively low temperatures. Fair accuracy Require in-line mounting (except bypass type)
Positive Displacement General	High rangeability Ease of calibration Linear readout Flexibility of readout devices Good to excellent accuracy	Relatively high pressure drop Very little overrange protection Susceptible to damages from gas or liquid slugs and from dirty fluids In-line mounting Relatively high cost, especially for high flow rate applications For most meters, problems presented by materials that tend to "plate" or "coat"
Nutating disk	Relatively low cost Moderate pressure loss Applicable to liquid batching systems Several construction materials available	Limited as to pipe size and capacity Fair accuracy for PD meters Fluids should be clean
Oscillating piston	Good accuracy, especially at low flow rates. Easily applied to liquid batching systems Good repeatability Moderate cost Easy to install and maintain	Available only in small sizes, normally 2 inches or less Limited power for driving accessories Fluids must be clean
Velocity Turbine	Good accuracy Excellent repeatability Excellent rangeability Low pressure drop Easy to install and maintain Can be compensated for viscosity variations Adaptable to flow totalizing Adaptable to digital blending systems Good temperature and pressure ratings	In-line mounting required Relatively high cost Limited use for slurry applications Nonlubricating fluids sometimes present problems Straight runs of pipe (15 diameters) required ahead of the meter Strainers recommended, except for the special slurry meters
Vortex	Excellent rangeability Digital readout lends itself to blending applications and flow totalization No moving parts Within the linear range, it is relatively immune to density, temperature, pressure and viscosity variations Very low pressure drop	Limited application data In-line mounting required Limitations imposed on upstream and downstream piping requirements in some cases preceded by 10 pipe diameters on the upstream side. Operation impaired if the temperature elements coat enough to affect adversely the rate of heat exchange between the elements and the flowing liquid Relatively high cost

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6-5. Liquid level. Measurements with liquid level instruments can be made with an accuracy and simplicity rivaling all others. The designer has a wide choice and guidance offered in the tables is basically with respect to type and class. Table 6-7 summarizes the various instruments and table 6-8 compares them.

Table 6-7. Liquid Level Instrumentation.

Type	Accuracy	General Remarks
Float Mechanisms	1% - 2%	Variations in devices relate to sensing instrumentation; magnetic type offers minimum seal problems.
Hydrostatic	1/2% - 2%	Variations in sensors; measures either pressure differentials or forces.
Sonic	3%	Pulse transit time determines liquid level.
Rotating Paddle	Within paddle diameter	Paddle material is important.
Electronic	1-1/2%	Antenna output.
Capacitance	1% - 3%	Fluid property dependent.

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Table 6-8. Comparison of Liquid Level Instruments.

Type	Advantages	Disadvantages
Float Mechanisms	Simple to operate Inexpensive	Difficult to service Clean surface required Requires direct mechanical hookup Cannot generally handle surface sloshing
Hydrostatic	Good accuracy Adaptable to wide level ranges Available in many construction materials and/or can be purged for corrosive services Can be purged for use in slurry service Moderate cost Can be isolated and zeroed in place	Errors caused by density variations Low pressure lead line often undesirable in other than atmospheric applications Heating of lead lines sometimes necessary Operating and maintenance problems often presented by purged lines Purging materials often present process difficulties
Sonic	Essentially no moving parts Utilizes solid-state circuitry requiring little maintenance Accuracy good where application is suitable Applicable to some difficult-to-measure streams such as powders, solids, solids-containing fluids and slurries Easy to install	Insufficient application data Tendency to bridge for some sensor types and for some materials Relatively high cost Difficulty in fixing distance between transmitting and receiving units in two-element systems--this objection is overcome by fixed distances in some models
Rotating Paddle	Inexpensive Easy to install	Need protection from incoming stream Seal problems
Electronic	Not affected by splashing, surging, etc.	Complex electronics Expensive Calibration problem
Capacitance	Rugged No moving parts	Somewhat expensive Fluid dependent Insulation important

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Section III. CONTROLS

6-6. General. Controls serve three purposes: protection of equipment, protection of personnel, and most efficient production of the high-pressure gas. Most of the instruments mentioned in this chapter can be adapted to remote readout. Control systems require the transmitting of signals to a controller. This can be done with pneumatic, electric or electronic, or hydraulic systems. These several types are briefly discussed below. In terms of control, the systems range from manual to complete automation. In control systems, there are several elemental operations. They include sensing, monitoring, protecting, and sequencing. Monitors warn or remind with lights or sounds when undesirable conditions occur. Protection means that the condition is not only sensed but the system corrects out of normal conditions or shuts itself down. Sequencing means that related procedures or operations are linked together to perform a process such as starting or shutting down a compressor. The above elemental operations can be used to regulate or optimize a process. The designer will have to decide on the degree of automation to achieve the desired output and write specifications accordingly.

6-7. Pneumatic control. All controllers operate in a similar fashion, with the primary difference being the medium in which they function. Pneumatic controllers offer several modes and responses: on-off, proportional, integral, derivative. These are obtained by appropriate choice of feedback elements and networks. In general, pneumatic systems are less expensive than their electrical counterparts, but are not capable of long transmission, fast response or ease of computer adaptability.

6-8. Electric and electronic control. These are widely used because of their sensitivity and versatility. Modes of response are the same as those available with pneumatic control. Because of the nature of the hardware involved, these controllers can be considerably more expensive than their pneumatic counterparts.

6-9. Hydraulic control. The principal difference between hydraulic and the other means of control is the use of a liquid as the control medium as opposed to air for pneumatic or electricity for electric and electronic controllers. Basic advantages of hydraulic controls include reasonably high speed response, high power gain, simplicity, and long life. Limitations include maintenance of control fluid, system leakage, and power supply difficulties.

6-10. Actuators.

a. Actuators are the link between the control system and the control valves. They can be either electric, pneumatic, or hydraulic, depending on the control system. The pneumatic types can be air or gas motors or cylinders. Normally, valve manufacturers will have actuators available for their valves. The designer should be certain that the actuators are compatible with the control system. For variable flow, actuators should be capable of positioning the valve at any point from fully open to fully closed. The desired opening and closing speeds should be included for actuator selection. It is at this point that the designer may wish to change the valve choice in order to reduce costs of the actuator-valve assembly. Only carefully written specifications will allow control systems suppliers/designers to provide the best but least expensive control system. For remote operation of high-pressure systems, it should be determined in which position (open or closed) it will be fail-safe and the appropriate actuator selected.

b. The general design problem is primarily one of meeting requirements for positioning accuracy, speed of response, adequate force level, and stable operation. Secondary factors include behavior on loss of signal or auxiliary power supply, service life, and first cost and operating cost.

CHAPTER 7
INSTALLATION AND TESTING

7-1. General. The inherent hazards in cryogenic and high-pressure gas systems make it imperative that special design emphasis be placed on cleaning, inspection, and testing. The hazards of cryogenic systems may be divided into several categories as follows: personnel exposure to extremely cold materials, personnel exposure to inert or toxic gases, material failure, and flammability of cryogenic fuel or oxidant. The primary hazard of high-pressure gases is the amount of available energy stored in the system under pressure.

7-2. Cleaning. The degree of cleaning will depend upon service of the cryogenic or high-pressure gas systems. In general, the following procedures should be included as a minimum:

a. Blowdown.
b. Flush.
c. Degrease and clean.
d. Dry.
e. Pressurize for standby.

7-3. Testing and inspection. The various codes under which the components are designed include rigorous testing and inspection procedures. Nondestructive testing, qualification of nondestructive examination personnel, and leak tests are defined. The designer shall insure that appropriate tests are defined for the system and any special tests such as "cold shock" of cryogenic systems are specified.

CHAPTER 8
SOURCES FOR DESIGN CALCULATIONS

8-1. Source description. As noted in chapter 1, rather than offering detailed design examples, sources for such calculations are displayed in table 8-1. To effectively carry out any design, detailed equations with their limitations, caveats, and applicability must be accounted for. Further, standard and current data, only alluded to within the text of the manual, must be employed to insure appropriate design. The examples cited in table 8-1 provide the designer with ready reference material from the basic source books which contain the information noted above with regard to equations and tables.

8-2. Other resources. In addition to the reference list (app A), a bibliography is presented. This literature will assist the designer in locating standards and codes, and will provide additional resources for his use.

Table 8-1. Sources for Standard Design Calculations.

Type	*Author or Source**	*Chapter, Section Page or Paragraph*
Fluid properties:		
high pressure gas	Baumeister	Section 4
	King	Chapter 3
cryogenics	Baumeister	18-36
	King	Chapter 11
	Barron	57, 633
Compressor systems:		
power requirements,	Baumeister	14-42
sizing, fluid quality	Gibbs	Chapters 3, 4, 5
heat exchanger	Baumeister	9-83
	Barron	129
	Gibbs	Chapter 19
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Corps of Engineers

APPENDIX A
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★ A 194-83	Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service
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APPENDIX B
EQUIVALENCE BETWEEN TRADE AND GENERIC NAMES

Trade Name	*Generic Name*
Buna-N	Butadiene copolymer with acrylonitrile.
Hastelloy C	Nickel-base alloy with specific amounts of molybdenum, chromium, manganese, copper, silicon and iron.
Inconel	Nickel-base heat- and oxidation-resistant alloy; approximately 13 percent chromium, 6 percent iron, small amounts of manganese, silicon, and copper.
Kel-F	Chlorotrifluoroethylene.
K-monel	Age-hardened, non-ferrous alloy containing 60-70 percent nickel, 25-35 percent copper, 1-3 percent iron, 2-4 percent aluminum, traces of manganese, silicon, and carbon.
Lucite	Acrylic resin.
Mylar	Polyethylene terephthalate.
Perlite	An inert siliceous volcanic rock (see Fed. Spec. HH-1-574).
Santocel	Light porous form of silica aerogel.
Teflon	Tetrafluoroethylene.
Viton-A	Synthetic rubbers.

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HIGH-PRESSURE GAS AND CRYOGENIC SYSTEMS

This change revises dates and titles of referenced publications, and corrects range of temperature cycles for stress reduction factors. TM 5-810-7/AFM 88-12, Chap. 4, 15 July 1983, is changed as follows:

1. New or revised material is indicated by a star.
2. Remove old pages and insert new pages as indicated below:

Remove pages	*Insert pages*
4-3 through 4-10	4-3 through 4-7
4-27 and 4-28	4-27 and 4-28
A-1	A-1 through A-2

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