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Petroleum and natural gas industries — Design and installation of piping systems on offshore production platforms

Industries du pétrole et du gaz naturel — Conception et installation de systèmes de tuyauterie sur les plates-formes de production en mer

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Foreword

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International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this International Standard may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 13703 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, Subcommittee SC 6, *Processing equipment and systems*.

Annexes A, B and C of this International Standard are for information only.

Introduction

This International Standard is based on API RP 14E, 5th edition, October 1991.

Petroleum and natural gas industries — Design and installation of piping systems on offshore production platforms

1 Scope

This International Standard specifies minimum requirements and gives guidance for the design and installation of new piping systems on production platforms located offshore for the petroleum and natural gas industries. It covers piping systems up to 69 000 kPa (ga) maximum, within temperature range limits for the materials meeting the requirements of ASME B31.3.

NOTE For applications outside these pressure and temperature ranges, this International Standard may be used but special consideration should be given to material properties.

Annex A gives some worked examples for solving piping design problems.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 13623, *Petroleum and natural gas industries — Pipeline transportation systems.*

API RP 520-2[1)], *Recommended practice for design and installation of pressure-relieving systems in refineries — Part 2.*

ASME[2)], *Boiler and pressure vessel code: Section VIII: Pressure vessels, Division 1.*

ASME B 31.3, *Process piping.*

NACE MR0175[3)], *Sulfide stress cracking resistant metallic materials for oil field equipment.*

NACE TM0177, *Laboratory testing of metals for resistance to specific forms of environmental cracking in H_2S environments.*

NACE TM0284, *Evaluation of pipeline and pressure vessel steels for resistance to hydrogen-induced cracking.*

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2) American Society of Mechanical Engineers, 345 East 47th Street, New York, N.Y. 10017, U.S.A.

3) National Association of Corrosion Engineers, P.O. Box 218340, Houston, Texas 77218-8340, U.S.A.

3 Terms, definitions, symbols and abbreviated terms

For the purposes of this International Standard, the following terms, definitions, symbols and abbreviated terms apply.

3.1 Terms and definitions

3.1.1
chloride stress-corrosion cracking service
service in which the process stream contains water and chlorides in a sufficient concentration, and at a high enough temperature, to induce stress-corrosion cracking of susceptible materials

NOTE Other constituents present, such as oxygen (O_2), may contribute to such chloride stress-corrosion cracking.

3.1.2
choke
device specifically intended to restrict the flow rate of fluids

3.1.3
corrosion-erosion
eroding away of a protective film of corrosion product by the action of the process stream, exposing fresh metal which then corrodes

NOTE Extremely high metal mass loss can occur under these conditions.

3.1.4
corrosive gas
gas which, when dissolved in water or other liquid, causes corrosion of metal

NOTE Corrosive gases usually contain hydrogen sulfide (H_2S), carbon dioxide (CO_2) and/or oxygen (O_2).

3.1.5
corrosive hydrocarbon service
service in which the process stream contains water or brine and carbon dioxide (CO_2), hydrogen sulfide (H_2S), oxygen (O_2) or other corrosive agents under conditions which cause corrosion of metal

3.1.6
expansion bellows
corrugated piping device designed to absorb expansion and contraction

3.1.7
expansion bend
piping configuration designed to absorb expansion and contraction

3.1.8
flowline
piping that carries well fluid from wellhead to manifold or first process vessel

3.1.9
flow regime
flow condition of a multi-phase process stream

EXAMPLES Slug flow, mist flow or stratified flow.

3.1.10
fluid
gas, vapour, liquid or combinations thereof

3.1.11
header
part of a manifold that directs fluid to a specific process system

See Figures 5 and 6.

3.1.12
hydrocarbon wettability
ability of the process stream to form a protective hydrocarbon film on metal surfaces

3.1.13
manifold
assembly of pipe, valves and fittings by which fluid from one or more sources is selectively directed to various process systems

3.1.14
nipple
section of threaded or socket-welded pipe, shorter than 300 mm, used as an appurtenance

3.1.15
nominal pipe size
nominal size
NPS
DN
designation of size in inches which is common to all components in a piping system other than those components designated by outside diameter

NOTE Nominal pipe size is designated by the letters NPS (when relating to inches) or DN (when relating to millimetres) followed by a number; it is a convenient number for reference purposes and it is normally only loosely related to manufacturing dimensions.

3.1.16
non-corrosive hydrocarbon service
service in which the process stream conditions do not cause significant metal mass loss, selective attack, chloride stress-corrosion cracking or sulfide stress-cracking

3.1.17
normal conditions
absolute pressure of 101,325 kPa and temperature of 0 °C

3.1.18
platform piping
any piping intended to contain or transport fluid on a platform

3.1.19
pressure rating
number relating to the pressure for which a system is suitable

NOTE The number may relate directly to the rated working pressure (e.g. ISO 10423 [1] pressure rating 13,8 MPa and API pressure rating 2 000 psi) or may have a more indirect correlation (e.g. ASME class 300).

3.1.20
pressure sensor
device designed to detect a predetermined pressure

3.1.21
process component
single functional piece of production equipment and associated piping

EXAMPLES Pressure vessel, heater, pump, etc.

3.1.22
riser
vertical portion of a pipeline (including the bottom bend) arriving on or departing from a platform

3.1.23
shutdown valve
automatically-operated valve used for isolating a process component or process system

3.1.24
sulfide stress-cracking service
service in which the process stream contains water or brine and contains a sufficient concentration of hydrogen sulfide (H_2S) to induce sulfide stress-cracking of susceptible materials

3.1.25
wellhead pressure
maximum shut-in surface pressure that may exist in a well

3.2 Symbols and abbreviated terms

3.2.1 Symbols

A minimum pipe cross-sectional flow area required per unit volume flowrate, expressed in square millimetres per cubic metre per hour ($mm^2/m^3/h$)

B mean coefficient of thermal expansion at operating temperatures normally encountered, expressed in millimetres per kelvin (mm/K)

C empirical constant, dimensionless

C_e sum of corrosion, mechanical strength and thread allowance, expressed in millimetres (mm)

C_v valve coefficient, dimensionless

NOTE 1 This value is equal to the water flowrate in US gpm at 60 °F required to generate a pressure drop of 1 psi (US Customary units only are used in this instance to maintain alignment with other published data).

D_i pipe inside diameter, expressed in metres (m)

D_o pipe outside diameter, expressed in millimetres (mm)

d_i pipe inside diameter, expressed in millimetres (mm)

d_g gas relative density (air = 1), dimensionless

d_L liquid relative density (water = 1), dimensionless

E longitudinal weld joint factor, dimensionless

E_m modulus of elasticity of piping material in the cold condition, expressed in newtons per square millimetre (N/mm^2)

f Moody friction factor, dimensionless

g gravitational constant, expressed in metres per second per second (m/s^2)

h_a acceleration head, expressed in metres (m) of liquid

h_f friction head, expressed in metres (m) of liquid

h_p absolute pressure head, expressed in metres (m) of liquid

h_{st} static head, expressed in metres (m) of liquid

h_{vh} velocity head, expressed in metres (m) of liquid

h_{vpa} absolute vapour pressure, expressed in metres (m) of liquid

h_W pressure loss, expressed in kilopascals (kPa)

K acceleration factor, dimensionless

L developed pipe length, expressed in metres (m)

L_m pipe length, expressed in kilometres (km)

m manufacturing wall thickness tolerance, expressed as a percentage (%)

$NPSH_a$ available net positive suction head, expressed in metres (m) of liquid

p operating pressure, expressed in kilopascals [kPa (abs)]

NOTE 2 Also denoted in text as “flowing pressure”.

p_i internal design pressure, expressed in kilopascals [kPa (ga)]

q_g gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions

q_L liquid flow rate, expressed in cubic metres per hour (m^3/h)

q_m total liquid plus vapour mass flowrate, expressed in kilograms per hour (kg/h)

R gas/liquid volume ratio, dimensionless

Re Reynolds number, dimensionless

R_p pump speed, expressed in revolutions per minute (r/min)

S allowable stress, expressed in newtons per square millimetre (N/mm^2)

T operating temperature, expressed in kelvin (K)

NOTE 3 Also denoted in text as “flowing temperature”.

t pressure design thickness, expressed in millimetres (mm)

t_{nom} minimum nominal pipe wall thickness, expressed in millimetres (mm)

U anchor distance (straight line distance between anchors), expressed in metres (m)

v_e fluid erosional velocity, expressed in metres per second (m/s)

v_g average gas velocity, expressed in metres per second (m/s)

NOTE 4 Also denoted in text as “gas velocity”

v_L average liquid velocity, expressed in metres per second (m/s)

y resultant of total displacement strains, expressed in millimetres (mm)

Y temperature factor, dimensionless

Z gas compressibility factor, dimensionless

ΔL expansion to be absorbed by pipe, expressed in millimetres (mm)

Δp pressure drop, expressed in kilopascals (kPa)

ρ_g gas density at operating pressure and temperature, expressed in kilograms per cubic metre (kg/m^3)

ρ_L liquid density at operating temperature, expressed in kilogram per cubic metre (kg/m^3)

ρ_m gas/liquid mixture density at operating pressure and temperature, expressed in kilograms per cubic metre (kg/m^3)

ΔT temperature change, expressed in kelvin (K)

μ_g gas viscosity at flowing pressure and temperature, expressed in pascal seconds (Pa·s)

μ_L liquid viscosity, expressed in pascal seconds (Pa·s)

3.2.2 Abbreviated terms

ERW Electric Resistance Weld

PWHT Post-Weld Heat Treatment

RF Raised Face

RTJ Ring Type Joint

SAW Submerged Arc Weld

SMYS Specified Minimum Yield Strength

4 General considerations

4.1 Materials

Carbon steel materials are suitable for many of the piping systems on production platforms, although stainless steels and other materials are also widely used. The following should be considered when selecting piping materials:

a) type of service;

b) compatibility with other materials;

c) mechanical strength, ductility, elasticity and toughness;

d) need for special welding procedures, or other jointing techniques;

e) need for special inspections, tests, or quality control;

f) possible misapplication in the field;

g) corrosion and erosion caused by internal fluids and/or marine environments;

h) need for performance in a fire situation.

4.2 Code of pressure piping

4.2.1 The design and installation of platform piping shall be in accordance with ASME B31.3, as modified by this International Standard. Risers for which ASME B31.3 is not applicable should be designed and installed in accordance with 4.2.2 to 4.2.6.

4.2.2 Design, construction, inspection and testing of risers shall be in accordance with ISO 13623 and governmental rules and regulations as appropriate to the application, using a design stress no greater than 0,6 times SMYS. Pipeline design codes may be used from pig trap to pig trap, except where precluded by national regulations.

4.2.3 One hundred percent radiography of welding should be performed on riser piping. The non-destructive testing of platform piping complying with ASME B31.3 should as a minimum satisfy Table 10 of this International Standard.

4.2.4 Impact tests shall be performed as specified by ASME B31.3. The design of high-pressure piping systems (i.e. above ASME class 2500) needs special consideration and shall be in accordance with the high-pressure piping requirements of ASME B31.3.

4.2.5 Valves, fittings and flanges should be manufactured in conformance with International and/or National Standards. Pressure/temperature ratings and material compatibility should be verified.

4.2.6 In determining the transition between risers and platform piping to which these practices apply, the first incoming and last outgoing valve that block pipeline flow are the limits of this International Standard's application, except for riser wall thickness calculations and material selection which may be to a pipeline code to permit a constant bore for pigging. Recommended practices included in this International Standard may be utilized for riser design when factors such as water depth, batter of platforms legs, potential bubbling area etc. are considered. National regulations may require the pipeline code to be continued to/from the pig launcher/receiver.

4.2.7 It is also common practice for a pipeline code to apply through the riser up to the pig trap and to include the piping and the first valve on each branch on the riser/pipeline.

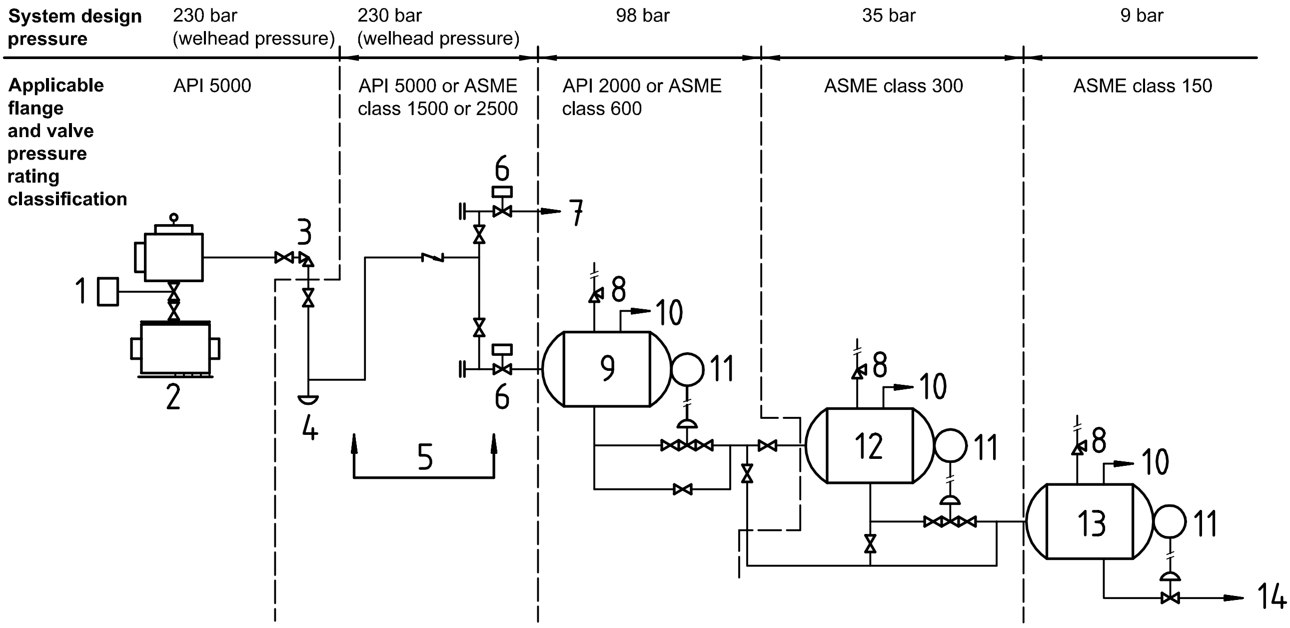
4.3 Demarcation between systems with different pressure ratings

4.3.1 Normally, after the well-stream leaves the wellhead the pressure is reduced in stages.

After the pressure is reduced, process components of lower pressure ratings may be used. A typical example is shown in Figure 1.

4.3.2 A pressure-containing process component shall either be designed to withstand the maximum internal pressure which can be exerted on it under any conditions, or shall be protected by a pressure-relieving device. In this case, a pressure-relieving device means a safety relief valve or a rupture disc. In general, when determining if pressure-relieving devices are needed, high-pressure shutdown valves, check valves, control valves or other such devices should not be considered as preventing overpressure of process components.

4.3.3 Pressure-rating boundaries shall be indicated on piping and instrument diagrams. Each system component (vessels, flanges, pipe or accessories) shall be designed to withstand the highest pressure to which it could be subjected under any foreseeable conditions, or it shall be protected by a pressure-relieving device. Abnormal pressure conditions shall be considered, e.g. start-up, shutdown, surge, etc.



NOTE 1 Design temperature is 65 °C throughout.

NOTE 2 Required shutdown sensors are not shown.

NOTE 3 Flowline and manifold are designed for wellhead pressure.

NOTE 4 System design pressures may be limited by factors other than the flange and valve pressure classifications (i.e. pipe wall thickness, separator design pressure, etc.).

NOTE 5 Only where spare relief valves are installed may upstream isolation valves be installed, and then it is essential that all isolation valves are interlocked to ensure that the pressurized system is protected at all times.

Key

1 Upper master gate valve
2 Wellhead
3 Wing choke
4 Flow tee
5 Manifold
6 Shutdown valve
7 To other systems
8 Pressure safety valve
9 High pressure separator
10 Gas outlet
11 Level controller
12 Medium pressure separator
13 Low pressure separator
14 Treating, storage or sales

Figure 1 — Example of a process system, denoting flange and valve pressure-rating changes

4.4 Corrosion considerations

4.4.1 General

Detailed corrosion-control practices for platform piping systems are outside the scope of this International Standard. Such practices should, in general, be developed by corrosion control specialists. Platform piping systems should, however, be designed to accommodate and be compatible with the corrosion control practices described below. Recommendations for corrosion-resistant materials and mitigation practices are given in the appropriate clauses of this International Standard.

The corrosivity of process streams may change over time. The possibility of changing conditions should be considered at the design stage.

4.4.2 Mass loss corrosion

Carbon steel platform piping systems may corrode under some process conditions. Production process streams containing water, brine, carbon dioxide (CO_2), hydrogen sulfide (H_2S) or oxygen (O_2), or combinations of these, may be corrosive to metals used in system components. The type of attack (uniform metal loss, pitting, corrosion-erosion, etc.) as well as the specific corrosion rate can vary in the same system, and can vary over time. The corrosivity of a process stream is a complex function of many variables, including:

a) hydrocarbon, water, salt and corrosive gas content;

b) hydrocarbon wettability;

c) flow velocity, flow regime and piping configuration;

d) temperature, pressure and pH;

e) solids content (sand, mud, bacterial slime and microorganisms, corrosion products, and scale).

Corrosivity predictions are very qualitative and may be unique for each system. Some corrosivity information for corrosive gases found in production streams is shown in Table 1.

Table 1 is intended only as a general guide for corrosion mitigation considerations and not for specific corrosivity predictions. Corrosion inhibition is an effective mitigation procedure when corrosive conditions are predicted or anticipated (see 5.1.2).

Table 1 — Qualitative guideline for mass loss corrosion of steel

Corrosive gas	Solubility[a] ratio $\times 10^{-6}$	Limiting values in brine	
		Non-corrosive ratio $\times 10^{-6}$	Corrosive ratio $\times 10^{-6}$
Oxygen (O_2)	8	< 0,005	> 0,025
Carbon dioxide (CO_2)	1 700	< 600	> 1 200
Hydrogen sulfide (H_2S)	3 900	See Note	See Note

NOTE No limiting values for mass loss corrosion by hydrogen sulfide are shown in this table because the amount of carbon dioxide and/or oxygen greatly influences the metal loss corrosion rate. Hydrogen sulfide alone is usually less corrosive than carbon dioxide due to the formation of an insoluble iron sulfide film which tends to reduce metal mass loss corrosion.

[a] Solubility ratio by volume. Solubility at 20 °C in distilled water at 1 atm partial pressure. Oxygen (O_2) is for 1 atm air pressure. Source: [3].

4.4.3 Chloride stress-corrosion cracking

Careful consideration shall be given to the effect of stress and chlorides, if alloy and stainless steels are selected to prevent corrosion by hydrogen sulfide and/or carbon dioxide. Process streams that contain water with chlorides may cause cracking in susceptible materials, especially if oxygen is present and the temperature is above 60 °C. High alloy and stainless steels, such as the AISI 300-series austenitic stainless steels, precipitation-hardening stainless steels, and “A-286” (ASTM A 453 [2] grade 660), should not be used unless their suitability in the proposed environment has been adequately demonstrated. Consideration should also be given to the possibility that chlorides may be concentrated in localized areas in the system.

4.4.4 Sulfide stress-cracking

Process streams containing water and hydrogen sulfide may cause sulfide stress-cracking of susceptible materials. This phenomenon is effected by a complex interaction of parameters including metal chemical composition, hardness and microstructure, heat treatment condition and factors such as pH, hydrogen sulfide concentration, stress and temperature. Materials used for process streams containing hydrogen sulfide should be selected to accommodate these parameters.

Testing of these materials should be in accordance with NACE TM0177.

4.4.5 Application of NACE MR0175

Materials selected for resistance to sulfide stress-cracking should be in accordance with NACE MR0175. Corrosion-resistant alloys not listed in NACE MR0175 may exhibit such resistance and may be used if it can be demonstrated that they are resistant in the proposed environment of use (or in an equivalent laboratory environment). Caution should be exercised in the use of materials listed in NACE MR0175. The materials listed may be resistant to sulfide stress-cracking, but may not be suitable for use in chloride stress-corrosion cracking service.

4.4.6 Hydrogen-induced cracking

Process streams containing water and hydrogen sulfide may cause hydrogen-induced cracking (HIC) of susceptible materials, particularly to carbon steel plate fabrications or pipe made from plate. Consideration shall be given to HIC-testing such materials, which should be in accordance with NACE TM0284. Specialist advice should be sought in this area.

5 Piping design

5.1 Pipe material grades

5.1.1 Non-corrosive hydrocarbon service

The two most commonly used material grades of carbon steel pipe are ASTM A 106 [4] grade B, API 5L [5] grade B and ISO 3183-1 [6]. Seamless pipe is generally preferred due to its consistent quality. ASTM A 106 is only manufactured in seamless, while API 5L is available in seamless, ERW and SAW. If use of grade B requires excessive wall thickness, use of pipe with higher allowable design stress such as API 5L grade X52, may be required; however, special welding procedures and close supervision are necessary when using API 5L grade X46 or higher. It should be noted that the use of high yield materials such as 5L X-grades, will not result in a proportional increase in allowable stress values when used in accordance with ASME B31.3.

Many of the grades of pipe listed in ASME B31.3 are suitable for non-corrosive hydrocarbon service. The following types or grades of pipe are specifically excluded from hydrocarbon service in accordance with ASME B31.3:

a) furnace lap-welded and furnace butt-welded;

b) fusion-welded per ASTM A 134 [7] and ASTM A 139 [8];

c) spiral-welded, except API 5L spiral-welded.

5.1.2 Corrosive hydrocarbon service

Design for corrosive hydrocarbon service should provide for one or more of the following corrosion-mitigating practices:

a) chemical treatment;

b) corrosion-resistant alloys;

c) protective coatings (see 9.5.2).

Of these, chemical treatment of the fluid in contact with carbon steels has been common practice. Corrosion-resistant alloys that have proven successful in similar applications (or by suitable laboratory tests) may be used, however careful consideration should be given to welding procedures. Consideration shall also be given to the possibility of sulfide stress-cracking and chloride stress-corrosion cracking (see 4.4.3 and 4.4.4). Adequate provisions should be made for corrosion monitoring (coupons, probes, spools, etc.) and chemical treating.

Because welding can significantly alter the corrosion-resistance of otherwise resistant materials, careful consideration shall be given to the development of welding procedures.

5.1.3 Sulfide stress-cracking service

The following guidelines should be used when selecting pipe if sulfide stress-cracking is anticipated.

a) Only seamless pipe should be used unless specifications and quality control applicable to this service have been exercised in manufacturing ERW or SAW pipe.

b) Carbon and alloy steels and other materials that meet the property, hardness, heat treatment and other requirements of NACE MR0175 may be used in sulfide stress-cracking service.

The most commonly-used pipe grades that meet the above guidelines are ASTM A 106 grade B; ASTM A 333 [9] grade 6 and API 5L grade B seamless. API 5LX grades may also be used but their welding presents special problems.

5.1.4 Resistance to brittle fracture

To ensure adequate resistance to brittle fracture, the selected pipe material grade shall have adequate notch toughness at its design thickness and design temperature combination.

Non-impact-tested carbon steel pipe (materials) should at least be supplied normalized for services below 0 °C; and welded components may require PWHT depending on minimum (design) service temperature and thickness of weldment.

CAUTION — PWHT may reduce mechanical properties of API 5L X pipe material grades.

ASTM A 333 grade 6 is an impact-tested carbon steel suitable for cold service and should have adequate notch toughness down to – 46 °C. PWHT may be required for certain minimum design temperature and weldment thickness combinations.

5.1.5 Utilities services

Materials other than carbon steel are commonly used in utilities service. If, however, steel pipe is used that is of a type or grade not acceptable for hydrocarbon service in accordance with 5.1.1, some definite marking system should be established to prevent such pipe from accidentally being used in hydrocarbon service.

5.1.6 Tubing (instrumentation and hydraulic/air systems)

Solution-annealed austenitic stainless steel (AISI 316 or AISI 316L) tubing, either seamless or ERW, should be used in chloride environments and for all hydrocarbon service and air service.

5.2 Sizing criteria — General

5.2.1 In determining the diameter of pipe to be used in platform piping systems, both the flow velocity and pressure drop should be considered. 5.3, 5.4 and 5.5 present equations for calculating pipe diameters (and graphs for rapid approximation of pipe diameters) for liquid lines, single-phase gas lines, and gas/liquid two-phase lines, respectively. These equations may be used for preliminary sizing and for lines where pressure drop is not critical. For critical lines, more detailed calculations shall be performed. Many companies also use computer programs, often using the Colebrook equation, to facilitate piping sizing; see ref. [10] for further information.

5.2.2 When determining line sizes, consideration should be given to the range of conditions that give rise to the largest line size as well as just initial flow rates. These conditions may be higher liquid flow rates, or lower gas pressures, that may apply some time after facility start-up. It is often advisable to add a surge factor of between 20 % and 50 % to the anticipated normal flow rate, unless surge expectations have been more precisely determined by pulse pressure measurements in similar systems or by specific fluid hammer calculation. Table 2 presents some typical surge factors that may be used if more definite information is not available.

In large diameter flow lines producing liquid-vapour-phase fluids between platforms through riser systems, surge factors have been known to exceed 200 % due to slug flow. Liquid-vapour slug flow programs are generally available and may be consulted for the evaluation of slug flow.

Table 2 — Typical surge factors

Service	Surge factor
Facility handling primary production from its own platform	20 %
Facility handling primary production from another platform or remote well in less than 45 m (150 ft) of water	30 %
Facility handling primary production from another platform or remote well in greater than 45 m (150 ft) of water	40 %
Facility handling gas-lifted production from its own platform	40 %
Facility handling gas-lifted production from another platform or remote well	50 %

5.2.3 Determination of pressure drop in a line should include the effect of valves and fittings. Manufacturer’s data or an equivalent length as given in Table 3 may be used.

5.2.4 Calculated line sizes may need to be adjusted in accordance with good engineering judgement.

5.3 Sizing methods for liquid lines

5.3.1 General

Single-phase liquid lines should be sized primarily on the basis of flow velocity. For lines transporting liquids in a single phase from one pressure vessel to another by pressure differential, the flow velocity should not exceed 5 m/s at maximum flow rates, to minimize flashing ahead of the control valve. Note that where “softer” materials are used, e.g. cupro-nickel, lower velocity limits apply. If practical, flow velocity should not be less than 1 m/s to minimize deposition of sand and other solids. At these flow velocities, the overall pressure drop in the piping will usually be small. Most of the pressure drop in liquid lines between two pressure vessels will occur in the liquid dump valve and/or choke.

Table 3 — Equivalent length of 100 % opening valves and fittings

Dimensions in metres

NPS	DN	Globe valve or ball check valve	Angle valve	Swing check valve	Plug valve, gate valve or ball valve	45° ell		Short radius ell		Long radius ell		Branch of tee		Run of tee		Enlargement					Contraction				
						Weld	Thread	Weld	Thread	Weld	Thread	Weld	Thread	Weld	Thread	Sudden			Standard reducer		Sudden			Standard reducer	
																Equivalent length in terms of small diameter									
																d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4	d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4
1½	40	16,8	7,92	3,96	0,305	0,305	0,610	0,914	1,52	0,610	0,914	2,49	2,74	0,610	0,914	1,52	0,914	0,305	1,22	0,305	0,914	0,610	0,305	10,305	
2	50	21,3	10,1	5,18	0,610	0,610	0,914	1,22	1,52	0,610	1,22	3,05	3,35	0,914	1,22	2,13	1,22	0,305	1,52	0,305	0,914	0,914	0,305	10,305	
2½	65	24,4	12,2	6,10	0,610	0,610		1,52		0,914		3,66		0,914		2,44	1,52	0,610	1,83	0,610	1,22	0,914	0,610	20,610	
3	80	30,5	15,2	7,62	0,610	0,610		1,83		1,22		4,17		1,22		3,05	1,83	0,610	2,44	0,610	1,52	1,22	0,610	20,610	
4	100	39,6	19,8	9,75	0,914	0,914		2,13		1,52		5,79		1,52		3,66	2,44	0,914	3,05	0,914	1,83	1,52	0,914	3,914	0,305
6	150	61,0	30,5	14,6	1,22	1,22		3,35		2,44		8,53		2,44		5,49	3,66	1,22	4,27	1,22	2,74	2,13	1,22	1,22	
8	200	79,2	38,1	19,5	1,83	1,83		4,57		2,74		11,3		2,74		7,62	4,88	1,52	5,79	1,52	3,66	2,74	1,52	1,52	0,610
10	250	101	48,8	24,4	2,13	2,13		5,49		3,66		14,3		3,66		9,45	6,10	2,13	7,32	2,13	4,57	3,66	1,83	1,83	0,610
12	300	122	57,9	29,0	2,74	2,74		6,71		4,27		16,8		4,27		11,3	7,32	2,44	9,53	2,49	5,49	4,27	2,13	2,13	0,610
14	350	137	64,0	32,0	3,05	3,05		7,92		4,88		18,9		4,88		12,8	7,92	2,74			6,10	4,88	2,44		
16	400	152	73,2	36,6	3,35	3,35		8,84		5,49		21,9		5,49		14,3	9,14	3,05			3,32	5,49	2,74		
18	450	168	85,3	42,7	3,66	3,66		10,1		6,10		25,0		6,10		16,2	10,7	3,35			7,92	6,10	3,05		
20	500	198	91,4	42,7	42,7	42,7		11,0		7,01		27,4		7,01		18,3	11,6	3,96			9,14	7,01	3,35		
22	550	210	102	51,8	4,57	4,57		12,2		7,62		30,5		7,62		19,8	12,8	4,27			9,75	7,62	3,66		
24	600	229	113	56,4	4,88	4,88		13,4		8,23		33,5		8,23		21,3	14,0	4,57			10,7	8,23	3,96		
30	750				6,40	6,40		16,8		12,2		42,7		12,2											
36	900				7,62	7,62		20,1		14,3		51,8		14,3											
42	1 050				9,14	9,14		23,5		16,8		61,0		16,8											
48	1 200				10,7	10,7		26,8		19,8		67,1		19,8											
54	1 350				12,2	12,2		30,2		21,3		76,2		21,3											
60	1 500				13,7	13,7		33,5		24,4		79,2		24,4											

NOTE 1 Source: GPSA Data Book, 1987 Revision, ref. [11].

NOTE 2 d is inside diameter of small outlet. D is inside diameter of larger outlet.

5.3.1.1 Flow velocities in liquid lines may be read from Figure 2 or may be calculated using the following derived equation:

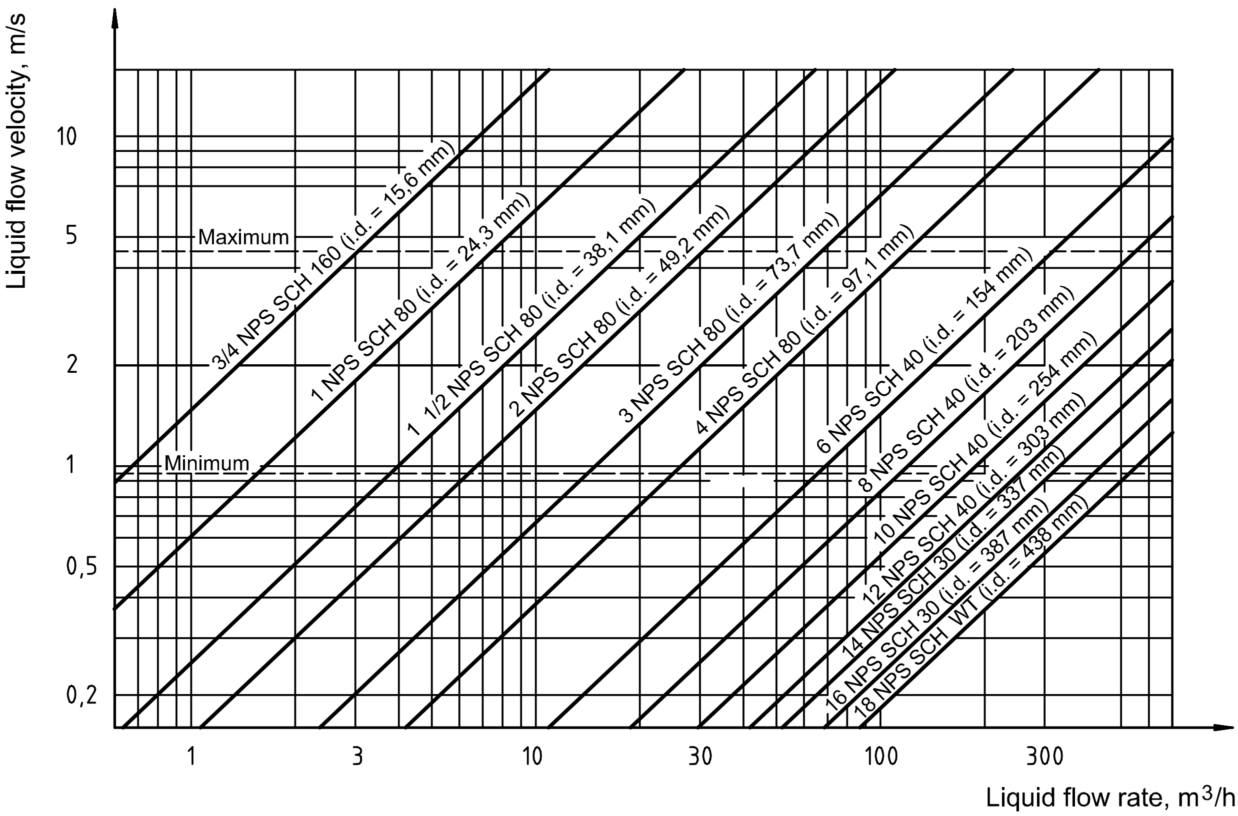
$$v_\text{L} = \frac{353{,}7\, q_\text{L}}{d_\text{i}^{\,2}} \tag{1}$$

where

v_L is the liquid flow velocity, expressed in metres per second (m/s);

q_L is the liquid flow rate, expressed in cubic metres per hour (m³/h);

d_i is the pipe inside diameter, expressed in millimetres (mm).



NOTE These curves were determined using equation (2).

Figure 2 — Velocity in liquid lines

5.3.1.2 Pressure drop per 100 m pipe length for single-phase liquid lines may be calculated using the following (Fanning) equation:

$$\Delta p = \frac{6\,270 \times 10^{6} f \cdot q_{\mathrm{L}}{}^{2} \cdot d_{\mathrm{L}}}{d_{\mathrm{i}}{}^{5}} \tag{2}$$

where

Δp is the pressure drop per 100 m of pipe, expressed in kilopascals (kPa);

f is the Moody friction factor, dimensionless;

q_{L} is the liquid flow rate, expressed in cubic metres per hour (m^3/h);

d_{L} is the liquid relative density (water = 1), dimensionless;

d_{i} is the pipe inside diameter, expressed in millimetres (mm).

5.3.1.3 The Moody friction factor, f, is a function of the Reynolds number and the surface roughness of the pipe. The modified Moody diagram, Figure 3, can be used to determine the friction factor once the Reynolds number is known. The Reynolds number is determined by the following equation:

$$Re = \frac{\rho_{\mathrm{L}} D_{\mathrm{i}} v_{\mathrm{L}}}{\mu_{\mathrm{L}}} \tag{3}$$

where

Re is the Reynolds number, dimensionless;

ρ_{L} is the liquid density, expressed in kilograms per cubic metre (kg/m^3);

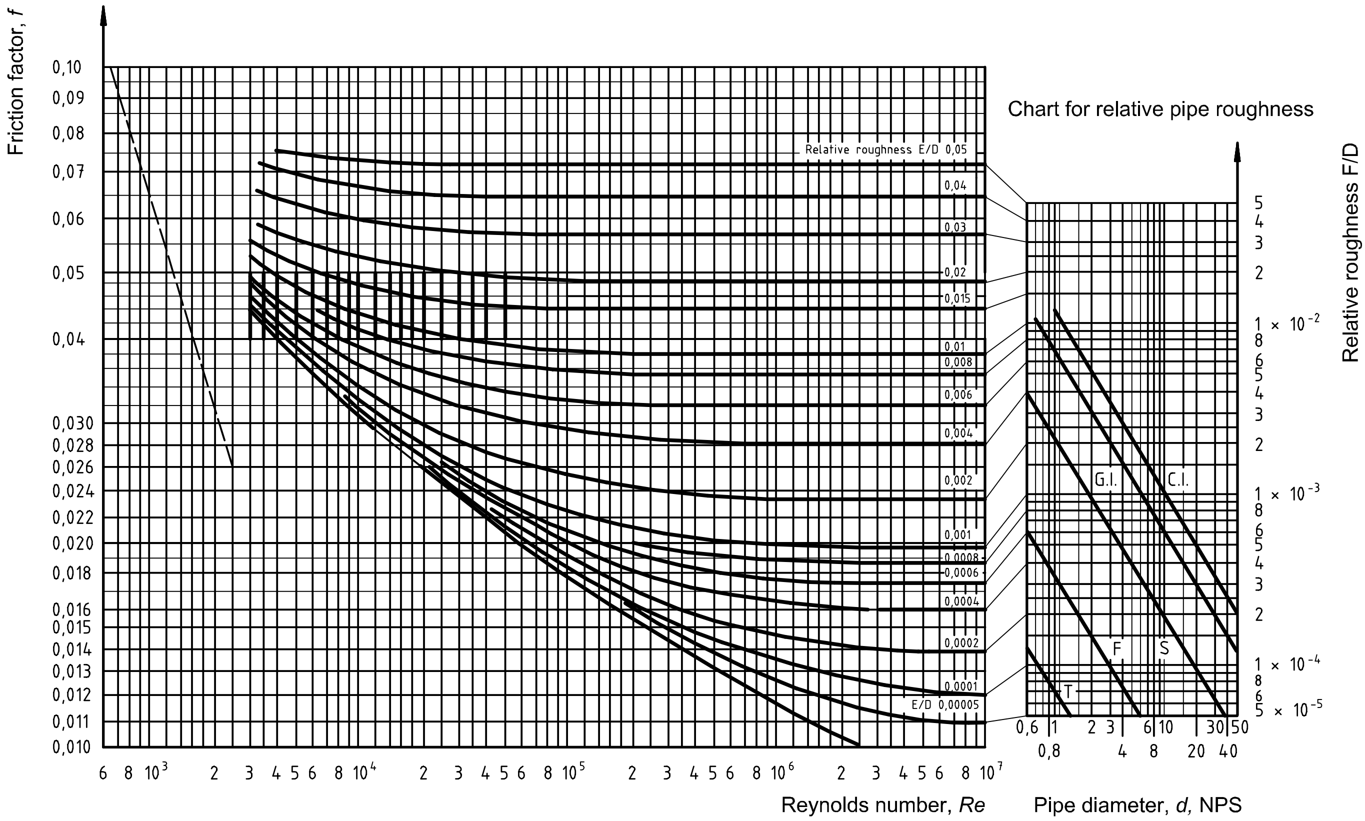
D_{i} is the pipe inside diameter, expressed in metres (m);

v_{L} is the liquid flow velocity, expressed in metres per second (m/s);

μ_{L} is the liquid viscosity, expressed in pascal seconds (Pa·s), or

the quantity in centipoises divided by 1 000, or

the (quantity in centistokes $\times$ relative density) divided by 1 000.



Key

C.I. Cast iron, clean

F Fibreglass epoxy

G.I. Galvanized iron

S Steel pipe, clean commercial (use also for cement, asbestos or cement-lined steel pipe)

T Drawn tubing

Figure 3 — Friction factor chart (modified Moody diagram)

5.3.2 Pump piping

5.3.2.1 Reciprocating, rotary and centrifugal pump suction piping systems should be designed so that the available net positive suction head ($NPSH_a$) at the pump inlet flange exceeds the NPSH required by the pump. Additionally, provision should be made in reciprocating pump suction piping to minimize pulsations. Satisfactory pump operation requires that essentially no vapour be flashed from the liquid as it enters the pump casing or cylinder.

5.3.2.2 In a centrifugal or rotary pump, the liquid pressure at the suction flange shall be high enough to overcome the pressure drop between the flange and the entrance to the impeller vane (or rotor) and maintain the pressure on the liquid above its vapour pressure, otherwise cavitation will occur. In a reciprocating unit, the pressure at the suction flange shall meet the same requirement but the pump's required NPSH is typically higher than for a centrifugal pump because of pressure drop across the valves and pressure drop caused by pulsation in the flow. Similarly, the available NPSH at the pump suction shall account for the acceleration in the suction piping caused by the pulsating flow, as well as the friction, velocity and static head.

5.3.2.3 The necessary available pressure differential over the pumped fluid vapour pressure may be defined as net positive suction head available ($NPSH_a$) which is the total head in metres absolute determined at the suction nozzle, minus the vapour pressure of the liquid in metres absolute. Available NPSH should always equal or exceed the pump’s required NPSH. Available NPSH for most pump applications may be calculated using equation (4). It is normal practice to add a safety factor of up to 1 m to the NPSH calculated.

$$NPSH_a = h_p - h_{vpa} + h_{st} - h_f - h_{vh} - h_a \tag{4}$$

where

h_p is the absolute pressure head due to pressure, atmospheric or otherwise, on surface of liquid going to suction, expressed in metres (m) of liquid;

h_{vpa} is the absolute vapour pressure of the liquid at suction temperature, expressed in metres (m) of liquid;

h_{st} is the static head, positive or negative, due to liquid level above or below datum line (centreline of pump), expressed in metres (m) of liquid;

h_f is the friction head, or head loss due to flowing friction in the suction piping, including entrance and exit losses, expressed in metres (m) of liquid;

h_{vh} is the velocity head, $v_L^2/2g$, expressed in metres (m) of liquid;

h_a is the acceleration head, expressed in metres (m) of liquid;

v_L is the velocity of liquid in piping, expressed in metres per second (m/s);

g is the gravitational constant (usually 9,81 m/s^2).

5.3.2.4 For a centrifugal or rotary pump, the acceleration head, h_a, is zero. For reciprocating pumps, the acceleration head is critical and may be determined by the following equation:

$$h_a = \frac{L v_L \cdot R_p \cdot C}{K \cdot g} \tag{5}$$

where

h_a is the acceleration head, expressed in metres (m) of liquid;

L is the length of suction line (actual length, not equivalent length), expressed in metres (m);

v_L is the average liquid velocity in suction line, expressed in metres per second (m/s);

R_p is the pump speed, expressed in revolutions per min (r/min);

C is the empirical constant for the type of pump:

C = 0,200 for simplex double-acting;

C = 0,200 for duplex single-acting;

C = 0,115 for duplex double-acting;

C = 0,066 for triplex single- or double-acting;

C = 0,040 for quintuplex single- or double-acting;

C = 0,028 for septuplex single- or double-acting;

NOTE The constant C varies from these values for unusual ratios of connecting rod length to crank radius.

K is the liquid compressibility factor representing the reciprocal of the fraction of the theoretical acceleration head which shall be provided to avoid a noticeable disturbance in the suction piping:

K = 1,4 for liquid with almost no compressibility (deaerated water);

K = 1,5 for amine, glycol, water;

K = 2,0 for most hydrocarbons;

K = 2,5 for relatively compressible liquid (hot oil or ethane);

g is the gravitational constant (usually 9,81 m/s^2).

It should be noted that there is not universal acceptance of equation (5) or the effect of acceleration head (see [12], [13] and [14] for further information). However, equation (5) is believed to be a conservative basis that will assure adequate provision for acceleration head.

5.3.2.5 When more than one reciprocating pump is operated simultaneously on a common feed line, all crankshafts are at times in phase and, to the feed system, the multiple pumps act as one pump of that type with a capacity equal to that of all pumps combined. In this case, the maximum instantaneous velocity in the feed line would be equal to that created by one pump having a capacity equal to that of all the pumps combined.

5.3.2.6 If the acceleration head is determined to be excessive, the following should be evaluated.

a) Shorten the suction line. Acceleration head is directly proportional to line length, L.

b) Use a larger suction pipe to reduce velocity. This is very helpful since velocity varies inversely with the square of pipe inside diameter. Acceleration head is directly proportional to fluid velocity, v_L.

c) Reduce the required pump speed by using a larger-size piston or plunger, if permitted by the pump rating. Speed required is inversely proportional to the square of piston diameter. Acceleration head is directly proportional to pump speed, R_p.

d) Consider using a pump with more plungers. For example:

   C = 0,040 for a quintuplex pump. This is about 40 % less than a triplex pump;

   C = 0,066 for a triplex pump. Acceleration head is directly proportional to C.

e) Consider using a pulsation dampener if the above remedies are unacceptable. The results obtainable by using a dampener in the suction system depend on the size, type, location and charging pressure used. A good, properly-located dampener, if kept properly charged, may reduce L, the length of pipe used in the acceleration head equation, to a value of 5 to 15 nominal pipe diameters. Dampeners should be located as close to the pump suction as possible.

f) Use a centrifugal booster pump to charge the suction of the reciprocating pump.

5.3.2.7 The following guidelines may be useful in designing suction piping:

a) suction piping may be one or two pipe sizes larger than the pump inlet connection;

b) suction lines should be short with a minimum number of elbows and fittings, and should fall continuously to the pump suction nozzle;

c) eccentric reducers should be used near the pump, with the flat side on top, to keep the top of the line level. This eliminates the possibility of gas pockets being formed in the suction piping. If potential for accumulation of debris is a concern, means for removal should be provided;

d) for reciprocating pumps, provide a suitable pulsation dampener (or make provisions for adding a dampener at a later date) as close to the pump suction as possible;

e) in multi-pump installations, size the common feed line so that the velocity will be as close as possible to the velocity in the laterals going to the individual pumps. This avoids velocity changes and thereby minimizes acceleration head effects.

5.3.2.8 Reciprocating, centrifugal and rotary pump discharge piping should be sized on an economical basis. Additionally, reciprocating pump discharge piping should be sized to minimize pulsations. Pulsations in reciprocating pump discharge piping are also related to the acceleration head, but are more complex than suction piping pulsations. The following guidelines may be useful in designing discharge piping:

a) discharge piping should be as short and direct as possible;

b) discharge piping may be one or two pipe sizes larger than pump discharge connection;

c) the velocity in discharge piping should not exceed three times the velocity in the suction piping. This velocity will normally result in an economical line size for all pumps, and will minimize pulsations in reciprocating pumps;

d) for reciprocating pumps, include a suitable pulsation dampener (or make provisions for adding a dampener at a later date) as close to the pump discharge as possible.

5.3.2.9 Table 4 may be used to determine preliminary suction and discharge line sizes.

Table 4 — Typical flow velocities

Type of pump	Suction velocity m/s	Discharge velocity m/s
Reciprocating		
Speeds up to 250 r/min	0,6	1,8
Speeds 251 r/min to 330 r/min	0,45	1,4
Speeds above 330 r/min	0,3	1,0
Centrifugal	0,6 to 1	1,8 to 2,7

5.4 Sizing criteria for single-phase gas lines

5.4.1 Sizing considerations

Single-phase gas lines should be sized so that the resulting end pressure is high enough to satisfy the requirements of the next piece of equipment. Excessive noise may result from restrictions such as control valves, orifices etc. or from velocities exceeding 25 m/s; however, the velocity of 25 m/s should not be interpreted as an absolute criteria. Higher velocities are acceptable if pipe routing, valve choice and placement are done to minimize or isolate noise.

The design of any piping system where corrosion inhibition is expected to be utilized shall consider the installation of additional wall thickness in piping design and/or reduction of velocity to less than 15 m/s to reduce the effect of stripping inhibitor film from the pipe wall. In such systems, a wall thickness monitoring method should be instituted.

5.4.2 General pressure drop equation

$$p_1^2 - p_2^2 = 6 \times 10^4 \left(\frac{S_g \cdot q_g{}^2 \cdot Z \cdot T_1 \cdot f \cdot L}{d_i{}^5} \right) \tag{6}$$

where

p_1 is the upstream pressure, expressed in kilopascals [kPa (abs)];

p_2 is the downstream pressure, expressed in kilopascals [kPa (abs)];

d_g is the gas relative density (air = 1,0), dimensionless;

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

Z is the compressibility factor for gas (see [11] for further information);

T_1 is the flowing temperature, expressed in kelvin (K);

f is the Moody friction factor, dimensionless (refer to Figure 3);

d_i is the pipe inside diameter, expressed in millimetres (mm);

L is the length, expressed in metres (m).

Rearranging equation (6) and solving for q_g:

$$q_g = 0{,}004\,08 \left[\frac{d_i{}^5 \left(p_1^2 - p_2^2 \right)}{Z \cdot T_1 \cdot f \cdot L \cdot d_g} \right]^{1/2} \tag{7}$$

An approximation of equation (6) can be made when the change in pressure is less than 10 % of the inlet pressure. If this is true, the assumption can be made:

$$p_1^2 - p_2^2 \cong 2 p_1 \left(p_1 - p_2 \right) \tag{8}$$

Substituting in equation (6):

$$\Delta p = 300 \frac{d_g \cdot q_g{}^2 \cdot Z \cdot T_1 \cdot f \cdot L}{p_1 \cdot d_i{}^5} \tag{9}$$

5.4.3 Empirical pressure drop

5.4.3.1 Equation assumptions

Several empirical equations have been developed so as to avoid solving for the Moody friction factor. All equations are patterned after the general flow equation with various assumptions relative to the Reynolds number. The most common empirical pressure-drop equation for gas flow in production facility piping is the Weymouth equation described below.

5.4.3.2 Weymouth equation

This equation is based on measurements of compressed air flowing in pipes ranging from 20 mm to 300 mm in the range of the Moody diagram where the friction factor curves are horizontal (i.e. high Reynolds number). In this range the Moody friction factor is independent of the Reynolds number and dependent on the relative roughness.

The Weymouth equation can be expressed as:

$$q_g = 0{,}013\,1\, d_i^{\,2{,}67} \left[\frac{p_1^2 - p_2^2}{L \cdot d_g \cdot Z \cdot T_1} \right]^{1/2} \tag{10}$$

where

q_g is the flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

d_i is the pipe inside diameter, expressed in millimetres (mm);

p_1 is the pressure at point 1, expressed in kilopascals [kPa (abs)];

p_2 is the pressure at point 2, expressed in kilopascals [kPa (abs)];

L is the length of pipe, expressed in metres (m);

d_g is the gas relative density (air = 1,0), dimensionless;

T_1 is the temperature of gas at inlet, expressed in kelvin (K);

Z is the compressibility factor of gas (see [11] for further information).

It is important to remember the assumptions used in deriving this equation and when they are appropriate. Short lengths of pipe with high pressure drops are likely to be in turbulent flow (high Reynolds Numbers) and thus the assumptions made by Weymouth are appropriate. Industry experience indicates that the Weymouth equation is suitable for most piping within the production facility. However, the friction factor used by Weymouth is generally too low for large-diameter or low-velocity lines where the flow regime is more properly characterized by the sloped portion of the Moody diagram.

5.4.3.3 Panhandle equation

This equation assumes that the friction factor can be represented by a straight line of constant negative slope in the moderate Reynolds number region of the Moody diagram.

The Panhandle equation can be written:

$$q_g = 0{,}001\,15 \times E \left[\frac{p_1^2 - p_2^2}{d_g^{\,0{,}961} Z \cdot T_1 \cdot L_m} \right] 0{,}51\, d_i^{\,2{,}53} \tag{11}$$

where

p_1 is the upstream pressure, expressed in kilopascals [kPa (abs)];

p_2 is the downstream pressure, expressed in kilopascals [kPa (abs)];

d_g is the gas relative density (air = 1,0), dimensionless;

Z is the compressibility factor for gas (see [11] for further information);

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

T_1 is the flowing temperature, expressed in kelvin (K);

L_m is the length, expressed in kilometres (km);

d_i is the pipe inside diameter, expressed in millimetres (mm);

E is the efficiency factor:

E = 1,0 for brand new pipe;

E = 0,95 for good operating conditions;

E = 0,92 for average operating conditions;

E = 0,85 for unfavourable operating conditions.

In practice, the Panhandle equation is commonly used for large-diameter [greater than DN 250 (10 NPS)], long pipelines (usually in the order of kilometres in length rather than metres) where the Reynolds number is on the straight-line portion of the Moody diagram. It can be seen that neither the Weymouth nor the Panhandle equation represents a “conservative” assumption. If the Weymouth equation is assumed, and the flow has a moderate Reynolds number, the friction factor will actually be higher than assumed (the sloped line portion is higher than the horizontal portion of the Moody curve), and the actual pressure drop will be higher than calculated.

If the Panhandle equation is used and the flow actually has a high Reynolds number, the friction factor will actually be higher than assumed (the equation assumes the friction factor continues to decline with increased Reynolds number beyond the horizontal portion of the curve), and the actual pressure drop will be higher than calculated.

5.4.3.4 Spitzglass equation

This equation is used for near-atmospheric-pressure lines. It is derived by making the following assumptions in equation (7):

a) $f = \left(1 + \frac{91,4}{d_i} + 1,18 \times 10^{-3} d_i\right)\left(\frac{1}{100}\right)$

b) T = 289 K

c) p_1 = 103,4 kPa

d) Z = 1,0 for ideal gas

e) $\Delta p < 10$ % of p_1

With these assumptions, and expressing pressure drop in pascals, the Spitzglass equation can be written:

$$q_g = 0{,}001\,08 \left[\frac{h_w \cdot d_i^5}{d_g L \left(1 + \frac{91,4}{d_i} + (1,18 \times 10^{-3}) d_i\right)} \right]^{1/2} \tag{12}$$

where

h_W is the pressure loss, expressed in pascals (Pa);

d_g is the gas relative density (air = 1,0), dimensionless;

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

L is the length, expressed in metres (m);

d_i is the pipe inside diameter, expressed in millimetres (mm).

5.4.4 Gas velocity equation

Gas velocities may be calculated using the following derived equation:

$$v_g = \frac{131\, Z \cdot q_g \cdot T}{d_i{}^2 \cdot p} \tag{13}$$

where

v_g is the gas velocity, expressed in metres per second (m/s);

d_i is the pipe inside diameter, expressed in millimetres (mm);

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

T is the operating temperature, expressed in kelvin (K);

p is the operating pressure, expressed in kilopascals [kPa (abs)];

Z is the gas compressibility factor (see [11] for further information).

5.4.5 Compressor piping

Reciprocating and centrifugal compressor piping should be sized to minimize pulsation, vibration and noise. The selection of allowable velocities requires an engineering study for each specific application.

NOTE 1 When using gas flow equations for old pipe, the build-up of scale, corrosion, liquids, paraffin etc., can have a large effect on gas flow efficiency.

NOTE 2 For other empirical equations, refer to [11] for further information.

5.5 Sizing criteria for gas/liquid two-phase lines

5.5.1 Erosional velocity

Flowlines, production manifolds, process headers and other lines transporting gas and liquid in two-phase flow should be sized primarily on the basis of flow velocity. Experience has shown that loss of wall thickness occurs by a process of corrosion-erosion. This process is accelerated by high fluid velocities, presence of sand, corrosive contaminants such as CO_2 and H_2S, and fittings such as elbows which disturb the flow path.

The following procedure for establishing an “erosional velocity” can be used where no specific information as to the erosive/corrosive properties of the fluid is available. The erosional velocity is based on a simplified model which does not take into account sand production and other factors (e.g. CO_2, H_2S and chloride levels).

NOTE The approach outlined in 5.5.1 uses an empirical formula, is simplistic and will normally result in a conservative velocity limit. However, experience shows that this limit can be overly conservative for many conditions. Erosion and the related corrosion-erosion are complex problems with many variables, including flow geometry, material type, solids properties, gas and liquid velocities, fluid viscosity, fluid density and fluid corrosivity. More recently, a number of computer-based models taking into account such variables have been generated. These may be used as an alternative to the present approach.

The velocity above which erosion may occur can be determined by the following empirical equation:

$$v_e = \frac{c}{\sqrt{\rho_m}} \tag{14}$$

where

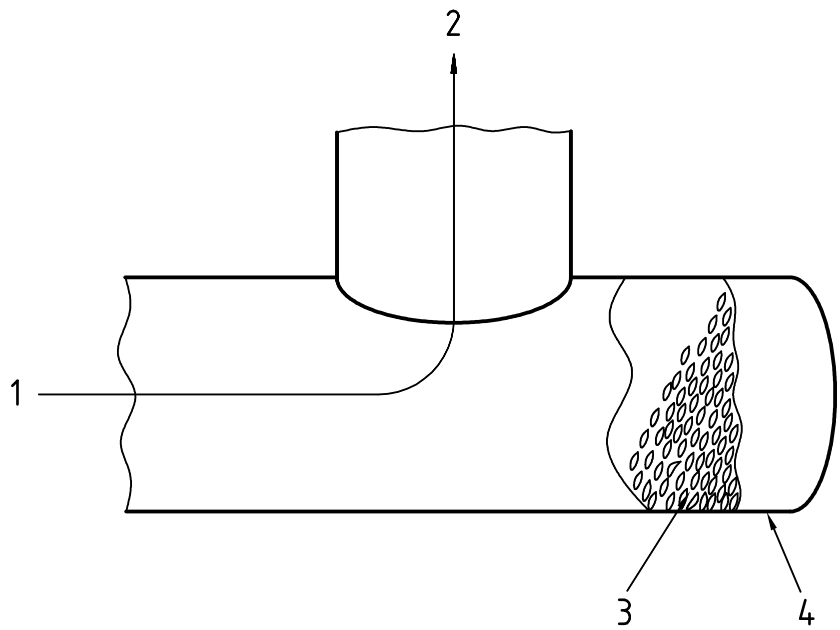
v_e is the fluid erosional velocity, expressed in metres per second (m/s);

c is the empirical constant;

ρ_m is the gas/liquid mixture density at flowing pressure and temperature, expressed in kilograms per cubic metre (kg/m^3).

Industry experience to date indicates that for solids-free fluids, values of c = 122 for continuous service and c = 152 for intermittent service are conservative. For solids-free fluids where corrosion is not anticipated or if corrosion is controlled by inhibition or by employing corrosion-resistant alloys, values of c between 180 and 245 may be used for continuous service, and values of c up to 300 have been used successfully for intermittent service. If solids production is anticipated, fluid velocities should be significantly reduced. Different values of c may be used where specific application studies have shown them to be appropriate.

Where solids and/or corrosive contaminants are present or where c values higher than 122 for continuous service are used, periodic surveys to assess pipe wall thickness should be considered. The design of any piping system where solids are anticipated should consider the installation of sand probes, capped tee or target tee, see Figure 4, and a minimum length of 1 m of straight piping downstream of choke outlets.



Key

1 Inlet

2 Outlet

3 Solids build-up

4 Weld cap or blind flange

Figure 4 — Target tee

5.5.2 Density

The density of the gas/liquid mixture can be calculated using the following derived equation:

$$\rho_{\mathrm{m}} = \frac{28\,833\, d_{\mathrm{L}} \cdot p + 37{,}22\, R \cdot d_{\mathrm{g}} \cdot p}{28{,}82\, p + 10{,}68\, R \cdot T \cdot Z} \tag{15}$$

where

- p is the operating pressure, expressed in kilopascals [kPa (abs)];
- d_{L} is the liquid relative density (water = 1, use average value for hydrocarbon-water mixtures) at standard conditions, dimensionless;
- R is the gas/liquid ratio at normal conditions;
- T is the operating temperature, expressed in kelvin (K);
- d_{g} is the gas relative density (air = 1), dimensionless;
- Z is the gas compressibility factor, dimensionless.

Once v_{e} is known, the minimum cross-sectional area required to avoid fluid erosion can be determined from the following derived equation:

$$A = \frac{277{,}6 + (103\, Z \cdot R \cdot T / p)}{v_{\mathrm{e}}} \tag{16}$$

where A is the minimum pipe cross-sectional flow area required per unit volume flowrate, expressed in square millimetres per cubic metre per hour ($\mathrm{mm^2/m^3/h}$).

5.5.3 Minimum velocity

If possible, the minimum velocity in two-phase lines should be about 3 m/s to minimize slugging of separation equipment. This is particularly important in long lines with elevation changes.

5.5.4 Pressure drop

The pressure drop in a two-phase flow, steel piping system may be estimated using a simplified Darcy equation from [11] (1981 Revision), as follows:

$$\Delta p = \frac{6\,253\,000\, q_{\mathrm{m}}^{2} \cdot f}{d_{\mathrm{i}}^{5} \cdot \rho_{\mathrm{m}}} \tag{17}$$

where

- Δp is the pressure drop per 100 m of pipe, expressed in kilopascals (kPa);
- d_{i} is the pipe inside diameter, expressed in millimetres (mm);
- f is the Moody friction factor, dimensionless;
- ρ_{m} is the gas/liquid density at flowing pressure and temperature, calculated as shown in equation (15), expressed in kilograms per cubic metre ($\mathrm{kg/m^3}$);
- q_{m} is the total liquid plus vapour mass flow rate, expressed in kilograms per hour (kg/h).

The use of this equation should be limited to a 10 % pressure drop due to inaccuracies associated with changes in density.

If the Moody friction factor is assumed to be an average of 0,015, this equation becomes:

$$\Delta p = \frac{93\,795\, q_m^{\,2}}{d_i^{\,5} \cdot \rho_m}$$

q_m can be calculated using the following derived equation:

$$W = 1{,}29\, q_g\, d_g + 1\,000\, q_L\, d_L \qquad (18)$$

where

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

d_g is the gas relative density (air = 1), dimensionless;

q_L is the liquid flow rate, expressed in cubic metres per hour (m^3/h);

d_L is the liquid relative density (water = 1), dimensionless.

NOTE This pressure drop calculation is an estimate only.

5.6 Pipe wall thicknesses

5.6.1 The pipe wall thickness required for a particular piping service is primarily a function of internal operating pressure, temperature and the allowable hoop stress of the selected material. The standards under which pipe is manufactured permit a variation in wall thickness below nominal wall thickness. It is usually desirable to include a minimum corrosion/mechanical strength allowance of 1,25 mm for carbon steel piping. A calculated corrosion allowance should be used if the corrosion rate can be predicted.

5.6.2 The pressure design thickness required for a particular application may be calculated by the following equation, and shall be in accordance with ASME B31.3:

$$t = \frac{p_i \cdot D_o}{20\left(100\, S \cdot E + 0{,}1\; p_i \cdot Y\right)} \qquad (19)$$

where

t is the pressure design thickness, expressed in millimetres (mm);

NOTE t is the minimum wall thickness excluding corrosion/mechanical strength allowance, thread allowance and allowance for manufacturer's mill tolerance (see 5.6.3).

p_i is the internal design pressure, expressed in kilopascals [kPa (ga)];

D_o is the pipe outside diameter, expressed in millimetres (mm);

E is the longitudinal weld joint factor (E shall be in accordance with ASME B31.3);

E = 1,00 for seamless pipe;

E = 0,85 for ERW pipe;

Y is the temperature factor (0,4 for austenitic and other ductile materials at 482 °C or below) when $t < D_o/6$. (Y shall be in accordance with ASME B31.3 requirements for the design of straight pipe under internal pressure);

S is the allowable stress which shall be in accordance with ASME B31.3, expressed in newtons per square millimetre (N/mm^2).

5.6.3 The minimum nominal pipe wall thickness required may be calculated by the following equation:

$$t_{nom} \geqslant \frac{t + C_e}{1 - (m/100)} \tag{20}$$

where

t is the pressure design thickness, expressed in millimetres (mm);

C_e is the sum of corrosion, mechanical strength and thread allowances, expressed in millimetres (mm);

m is the manufacturing negative wall thickness tolerance, expressed as a percentage (%).

5.6.4 Guidelines for corrosion allowances are as follows:

low/moderate duty = 1,0 mm, 1,25 mm or 1,5 mm

moderate/high duty = 3,0 mm

5.6.5 The maximum allowable working pressures for most of the nominal wall thicknesses in sizes DN 50 (2 NPS) through DN 600 (24 NPS) are given in Table 5 for ASTM A 106 grade B, seamless pipe, using a corrosion/mechanical strength allowance of 1,25 mm and a manufacturer's wall thickness under-tolerance of 12,5 %. The maximum working pressures in Table 5 were computed from equations (19) and (20), for values of $t < D_o/6$. For clarification, using Table 5 nominal wall thickness (t_{nom}), equation (20) is applied to obtain the pressure design thickness (t); this value is used in equation (19) to obtain the internal design pressure (p_i), i.e. maximum allowable working pressure. For values of $t > D_o/6$, the Lamé equation shall be used in accordance with ASME B31.3. Table 5 considers internal pressure and temperature only. These wall thicknesses may have to be increased in cases of unusual mechanical or thermal stresses. The maximum allowable working pressure of stainless steel tubing may be calculated using equation (19) with a corrosion/mechanical strength allowance of zero.

5.6.6 Small-diameter, thin-wall pipe is subject to failure from vibration and/or corrosion. In hydrocarbon service, nipples DN 20 (3/4 NPS) or smaller should be schedule 80 minimum. Completely threaded nipples should not be used.

5.6.7 Special attention shall be given to thin-wall piping, e.g. stainless steel, in vibrating service to avoid fatigue failure.

Table 5 — Maximum allowable working pressures for platform piping, ASTM A 106 grade B, seamless pipe (stress values from ASME B 31.3)

NPS in	Outside diameter mm	Nominal wall thickness mm	Nominal mass kg/m	Mass class	Schedule No.	Temperature range °C – 29 to 204 [a]	205 to 260	261 to 315	316 to 343
						Maximum allowable working pressure kPa (ga)			
2	60,3	5,54	7,48	XS	80	17 166	16 221	14 848	14 586
		8,74	11,10	-	160	31 848	30 097	27 545	27 069
		11,07	13,45	XXS	-	43 345	40 959	37 490	36 841
2,5	73,0	7,01	11,41	XS	80	19 407	18 345	16 786	16 497
		9,53	14,90	-	160	28 924	27 331	25 021	24 586
		14,02	20,41	XXS	-	47 241	44 641	40 862	40 152
		19,05	25,33	-	-	67 393	69 486	59 483	58 455
3	88,9	7,62	15,25	XS	80	17 607	16 634	15 228	14 966
		11,13	21,30	-	160	28 434	26 869	24 593	24 166
		15,24	27,65	XXS	-	42 000	39 690	36 331	35 697
4	114,3	6,02	16,07	STD	40	9 924	9 379	8 586	8 434
		8,56	22,31	XS	80	15 697	14 834	13 579	13 338
		11,13	28,30	-	120	21 717	20 524	18 786	18 455
		13,49	33,53	-	160	27 441	25 931	23 738	23 324
		17,12	41,02	XXS	-	36 600	34 579	31 662	31 110
6	168,3	7,11	28,26	STD	40	8 317	7 855	7 193	7 069
		10,97	42,56	XS	80	14 221	1 344	12 303	12 090
		14,27	54,20	-	120	19 428	18 366	16 807	16 517
		18,26	67,55	-	160	25 862	24 503	22 428	22 041
		21,95 [b]	79,18	XXS	-	32 138	30 372	27 800	23 717
8	219,1	7,04	36,76	-	30	6 262	5 917	5 421	5 324
		8,18	42,53	STD	40	7 572	7 159	6 552	6 441
		10,31	53,07	-	60	10 048	9 497	8 690	8 538
		12,70	64,63	XS	80	12 855	12 152	11 117	10 924
		15,09	75,79	-	100	15 710	14 848	13 586	13 351
		18,26	90,43	-	120	19 572	18 497	16 931	16 641
		20,62 [b]	100,88	-	140	22 503	21 269	19 469	19 131
		22,22 [b]	107,87	XXS	-	24 517	23 166	21 207	20 841
		23,01 [b]	111,18	-	160	25 517	24 110	22 069	21 690
10	273,0	6,35	41,77	-	20	4 386	4 145	3 793	3 731
		7,09	46,47	-	-	5 055	4 779	4 372	4 297
		7,80	51,00	-	30	5 703	5 386	4 931	4 848
		9,27	60,29	STD	40	7 055	6 669	6 103	5 993
		12,70	81,54	XS	60	10 241	9 675	8 855	8 703
		15,09	95,84	-	80	12 490	11 807	10 807	10 621
		18,26	114,74	-	100	15 531	14 676	13 434	13 200
		21,44 [b]	132,85	-	120	18 621	17 6,00	16 110	15 828
		25,40 [b]	154,96	XXS	140	22 559	21 317	19 510	19 172
		28,58 [b]	172,11	-	160	25 772	23 452	22 290	21 903

Table 5 *(continued)*

NPS in	Outside diameter mm	Nominal wall thickness mm	Nominal mass kg/m	Mass class	Schedule No.	Temperature range °C – 29 to 204 [a]	 205 to 260	 261 to 315	 316 to 343
						Maximum allowable working pressure kPa (ga)			
12	323,8	6,35	49,72	-	20	3 690	3 490	3 193	3 138
		8,38	65,20	-	30	5 241	4 959	4 538	4 455
		9,52	73,82	STD	-	6 124	5 786	5 297	5 207
		10,31	79,74	-	40	6 731	6 366	5 828	5 724
		12,70	97,44	XS	-	8 593	8 117	7 434	7 303
		14,27	108,96	-	60	9 828	9 290	8 503	8 359
		17,48	132,01	-	80	12 372	11 690	10 703	10 517
		21,44 [b]	159,66	-	100	15 572	14 710	13 469	13 234
		25,40 [b]	186,75	XXS	120	18 828	17 786	16 283	16 000
		28,58 [b]	207,87	-	140	21 476	20 297	18 579	18 255
		33,32 [b]	238,60	-	160	25 517	24 110	22 069	21 690
14	355,6	6,35	54,68	-	10	3 359	3 172	2 903	2 855
		7,92	67,94	-	20	4 448	4 207	3 848	3 786
		9,52	81,28	STD	30	5 566	5 262	4 814	4 731
		11,13	94,49	-	40	6 697	6 324	5 793	5 690
		12,70	107,38	XS	-	7 807	7 379	6 752	6 634
		15,09	126,51	-	60	9 510	8 986	8 228	8 083
		19,05	158,08	-	80	12 372	11 697	10 703	10 517
		23,83 [b]	194,90	-	100	15 890	15 014	13 745	13 503
		27,79 [b]	224,36	-	120	18 855	17 821	16 310	16 028
		31,75 [b]	253,31	-	140	21 869	20 669	18 917	18 593
		35,71 [b]	281,49	-	160	24 938	23 566	21 572	21 200
16	406,4	6,35	62,63	-	10	2 931	2 772	2 538	2 497
		7,92	77,86	-	20	3 890	3 676	3 366	3 303
		9,52	93,21	STD	30	4 862	4 593	4 207	4 131
		12,70	123,29	XS	40	6 814	6 441	5 897	5 793
		16,66	160,72	-	60	9 276	8 766	8 028	7 883
		21,41 [b]	203,88	-	80	12 276	11 600	10 621	10 434
		26,19 [b]	245,55	-	100	15 345	14 503	13 276	13 041
		30,94 [b]	287,21	-	120	18 448	17 434	15 959	15 683
		36,50 [b]	333,35	-	140	22 152	20 938	19 165	18 834
18	457,2	6,35	70,59	-	10	2 607	2 462	2 255	2 214
		7,92	87,79	-	20	3 455	3 262	2 986	2 931
		9,52	105,14	STD	-	4 317	4 076	3 731	3 669
		11,13	122,36	XS	30	5 186	4 897	4 483	4 407
		12,70	139,19	-	-	6 041	5 710	5 228	5 138
		14,27	155,91	-	40	6 903	6 524	5 972	5 869
		18,24	197,93	-	60	9 097	8 593	7 869	7 731
		23,80 [b]	254,48	-	80	12 214	11 545	10 566	10 3 86
		29,36 [b]	309,54	-	100	15 393	14 545	13 317	13 083
		34,11 [b]	355,67	-	120	18 152	17 152	15 703	15 428

Table 5 *(continued)*

NPS in	Outside diameter mm	Nominal wall thickness mm	Nominal mass kg/m	Mass class	Schedule No.	Temperature range °C – 29 to 204 [a]	 205 to 260	 261 to 315	 316 to 343
						Maximum allowable working pressure kPa (ga)			
20	508	6,35	78,56	-	10	2 343	2 214	2 027	1 991
		9,52	117,03	STD	20	3 876	3 683	3 353	3 295
		12,70	155,13	XS	30	5 428	5 129	4 695	4 613
		15,06	183,08	-	40	6 588	6 226	5 699	5 600
		20,62	247,84	-	60	9 353	8 838	8 090	7 950
		26,19	311,19	-	80	12 166	11 497	10 524	10 341
		32,54	381,55	-	100	15 429	14 580	13 346	13 115
		38,10	441,52	-	120	18 335	17 327	15 860	15 585
24	609,6	6,35	94.47	-	10	1 950	1 843	1 687	1 658
		9,52	140,89	STD	20	3 224	3 047	2 789	2 740
		12,70	186,95	XS	-	4 511	4 263	3 902	3 834
		14,27	209,51	-	30	5 150	4 867	4 455	4 378
		17,48	255,25	-	40	6 464	6 109	5 591	5 494
		24,61	355,04	-	60	9 418	8 900	8 147	8 005
		30,96	441,80	-	80	12 091	11 426	10 59	10 277
		38,88	547,23	-	100	15 482	14 630	13 392	13 160
		46,02	639,62	-	120	18 595	17 572	16 084	15 805

[a] See 5.1.4 concerning resistance to brittle fracture.

[b] All welds on wall thickness greater than 19 mm shall be stress-relieved.

NOTE Includes corrosion/mechanical strength allowance of 1,25 mm and includes 12,5 % variation below nominal wall thickness (manufacturer tolerance) of pipe and fittings.

5.7 Joint connections

5.7.1 Pipe joint connection types

Commonly-accepted types of pipe joint connections include butt-welded, socket-welded and threaded-and-coupled. The following guidelines indicate where particular types of joints should be used.

5.7.2 Hydrocarbon piping

Butt welding should be used for joint fabrication.

Piping ⩾ DN 50 (2 NPS) shall be butt-welded.

Piping ⩽ DN 40 (1½ NPS) diameter may be socket-welded.

Socket-welded joints should not be used where there is:

a) risk of severe crevice corrosion; or

b) severe mechanical vibration.

All sizes of piping in pressure ratings ASME class 600 and higher should be butt-welded.

The slip-on type of flange may only be used in non-cyclic, non-shock service for ASME class 300 and lower.

5.7.3 Utility piping (non-steam service)

Pressurized utility piping $\geqslant$ DN 80 (3 NPS) should be butt-welded.

5.7.4 Threaded piping systems

Screwed pipe joints and fittings have lower integrity than butt-welded and socket-welded types, and shall only be used if:

a) the fluid is non-flammable, non-toxic, non-hazardous and non-erosive;

b) the duty is non-cyclic and not subjected to significant mechanical vibration;

c) the design pressure is less than 2 000 kPa (ga);

d) the design temperature is between – 29 °C and 200 °C.

Occasionally, it may not be possible to observe the guidelines given above, particularly when connecting to equipment or instruments. In such cases a risk assessment of the jointing method shall be performed to consider the hazards to equipment and all personnel. Threads should be tapered, concentric with the pipe, clean-cut and without burrs. The inside of the pipe on all field-cut threads should be reamed.

Particular attention shall be given to layout of piping which incorporates screwed joints, ensuring that the joints are not overloaded, nor the piping configured so that it can become unscrewed in service.

ISO 10422 [15] or ASME B1.20.1 [16] may be consulted for further details on threading, ISO 13678 [17] may be consulted for information on thread compounds.

5.8 Expansion and flexibility

5.8.1 Piping systems may be subjected to many, diversified loadings. Generally, stress caused by the following factors are significant in the flexibility analysis of a piping system:

a) pressure and pressure variations;

b) mass of pipe, fittings, valves, fluid and insulation;

c) thermal expansion or contraction due to process conditions, temperature cycling, venting and ambient changes;

d) external loads and dynamic effect such as hydraulic shocks, wind, discharge reactions, vibration and earthquake. Protection from blast over-pressure may need consideration in certain cases;

e) terminal loads and movements, and other restraints in the piping system.

Piping systems shall have sufficient flexibility to ensure that they do not leak, distort, fail, or overload valves and connected strain-sensitive equipment such as vessels, compressors and turbines.

Flexibility and stress analysis of the piping systems should be carried out to satisfy the criteria in accordance with ASME B31.3.

5.8.2 A formal stress analysis should be carried out if the following approximate criteria, in accordance with ASME B31.3, are not satisfied.

For a piping system which is of uniform size, has not more than two points of fixation, no intermediate restraints, and falls within the limitations of empirical equation (21).

$$\frac{D_o \cdot y}{(L - U)^2} \leqslant 208{,}3 \qquad (21)$$

where

D_o is the pipe outside diameter, expressed in millimetres (mm);

y is the resultant of total displacement strains to be absorbed by the piping system, expressed in millimetres (mm);

L is the developed pipe length, expressed in metres (m);

U is the anchor distance (straight line between anchors), expressed in metres (m).

No general proof can be offered that this equation will provide accurate or consistently conservative results. It is not applicable to systems used under severe cyclic conditions. It should be used with caution in configurations such as unequal leg U-bends ($L/U > 2{,}5$) or near-straight "sawtooth" runs, or for large-diameter, thin-wall pipe (stress intensity factor $i > 5$, and should be in accordance with the stress intensification factors specified in ASME B31.3), or where extraneous displacement (not in the direction connecting anchor points) constitute a large part of the total displacement. There is no assurance that terminal reactions will be acceptably low, even if a piping system falls within the limitations of equation (21).

5.8.3 ASME B31.3 does not require a formal stress analysis in systems that meet one of the following criteria:

a) the systems are duplicates of successfully-operating installations or replacements of systems with a satisfactory service record;

b) the systems can be judged adequate by comparison with previously analysed systems.

5.8.4 Pipe movement can be accommodated by expansion bends (including loops and U-, L- and Z-shaped piping), swivel joints or expansion bellows. Expansion bends should be used wherever practical. If expansion bends are not practical, swivel joints should be used. Swivel joints can be subject to leakage and shall be properly maintained. Expansion bellows can be subject to failure if improperly installed and should be avoided. However, expansion bellows are often used in engine exhaust systems and other low-pressure utility systems.

5.9 Start-up provisions

Temporary start-up, cone-type screens should be provided in all pump and compressor suction lines. Screens (with the cone pointed upstream) should be located as close as possible to the inlet flanges, with consideration for their later removal. Sometimes a set of breakout flanges are required to remove the screens. The screens should be checked during start-up and removed when sediment is no longer being collected. Caution should be exercised in screen selection and use in order to avoid creating NPSH problems. Consideration should be given to the need for small valves required for hydrostatic testing, venting, draining and purging.

6 Selection of valves

6.1 General

6.1.1 Ball, gate, plug, butterfly, globe, diaphragm, needle and check valves have all been used in platform production facilities. Brief discussions of the advantages, disadvantages and design features for each type of valve are given below. Based on these considerations, specific recommendations for the application of certain types of valves are given in the following clauses. Valve catalogues contain basic design features, materials, drawings and photographs of the various valve types. Valves should comply with relevant National and International valve standards.

6.1.2 As a general guideline, manually operated ball valves and plug valves should be provided with manual gear operators as follows:

ASME class 150 through class 400	DN 250 (10 NPS and larger);
ASME class 600 and class 900	DN 250 (6 NPS and larger);
ASME class 1500 and higher	DN 100 (4 NPS and larger).

Excessive lever lengths (> 750 mm) should be avoided owing to offshore space constraints.

6.1.3 As a general guideline, the following valves should be equipped with power actuators:

a) all shutdown valves;

b) centrifugal compressor inlet and discharge valves. These valves should close automatically on shutdown of the prime mover;

c) divert, blowdown and other automatic valves;

d) valves of the following sizes, if frequently operated:

ASME class 150	DN 400 (16 NPS) and larger;
ASME class 300 and class 400	DN 300 (12 NPS) and larger;
ASME class 600 and class 900	DN 250 (10 NPS) and larger;
ASME class 1500 and higher	DN 200 (8 NPS) and larger.

6.2 Types of valves

6.2.1 Ball valves

Ball valves are suitable for most manual on-off hydrocarbon or utilities service when operating temperatures are between – 29 °C and 200 °C. Application of soft-seated ball valves above 100 °C should be carefully considered due to the temperature limitations of some soft-seat materials. For very abrasive or high temperature service, metal-seated ball valves should be considered, but it should be noted that the operating torque will be increased.

— Ball valves are available in both floating-ball and trunnion-mounted designs. Valves of the floating-ball design, develop high operating torques and seat load and therefore are not suitable for high pressures and large diameters.

— Ball valves are generally not suitable for throttling because, in the partially-open position, seats and sealing surfaces are exposed to abrasion by process fluids.

— In critical service, consideration should be given to purchasing ball valves with sealant-injection fittings for the ball seats as well as for the stem, since sealant is sometimes necessary to prevent minor leaks or reduce operating torques. If a body cavity-bleed capability is desired, a body bleed-port independent of the sealant injection fittings should be provided.

6.2.2 Gate valves

Gate valves are suitable for most on-off, non-vibrating hydrocarbon or utilities services for all temperature ranges. In vibrating service, gate valves may move open or closed from their normal positions unless the stem packing is carefully adjusted. Gate valves have better torque characteristics than ball or plug valves but are larger and do not have the advantage of quarter-turn action.

— In sizes DN 50 (2 NPS) and larger, manually-operated gate valves should be equipped with flexible discs or through-conduit expanding gates. Wedge gate valves do not normally have body-cavity over-pressure protection.

— Gate valves with unprotected rising stems should not be used since the marine environment can corrode exposed stems and threads, making the valves hard to operate and damaging stem packing.

— Gate valves can be used with simple push-pull actuators, but for emergency shutdown isolation duty the valve should be a reverse-acting, slab-gate type. All moving parts on gate valves with power actuators should be enclosed, eliminating fouling by paint or corrosion products.

— Gate valves should not be used for throttling service. Throttling, especially with fluids containing sand, can damage the sealing surfaces.

— The operating temperature of soft-seated gate valves may be limited by the seating material.

6.2.3 Plug valves

Plug valves are suitable for the same applications as ball valves (see 6.2.1) and are also subject to similar temperature limitations. Plug valves are available with quarter-turn action in either lubricated or non-lubricated designs. Lubricated plug valves shall be lubricated on a regular schedule to maintain a satisfactory seal and ease of operation, the frequency of lubrication depending on the type of service. The lubrication feature provides a remedial means for freeing stuck valves. In the non-lubricated design, the seal is accomplished by a soft material, unless the design is metal-to-metal pressure-balanced. They do not require frequent maintenance lubrication but may be more difficult to free after prolonged setting in one position. The application circumstances will generally dictate a selection preference based on these characteristics.

6.2.4 Butterfly valves

Butterfly valves with a resilient lining are suitable for coarse throttling and low-pressure, non-hydrocarbon, non-hazardous service. They are not suitable as primary block valves for vessels, tanks, etc. For temperatures above 65 °C, or for pressure ratings above ASME class 150, or for hydrocarbon or hazardous fluids, a high-performance, non-lined type of butterfly valve should be used. Because low-torque requirements permit butterfly valves to vibrate open, handles with detents should be specified.

6.2.5 Globe valves

If good throttling control is required (e.g. in bypass service around control valves, or for small vents), globe valves are the most suitable.

6.2.6 Diaphragm (bladder) valves

In this valve design, a diaphragm made of an elastomer is connected to the valve stem. Closure is accomplished by pressing the diaphragm against a metal weir which is a part of the valve body.

Diaphragm valves are used primarily for low pressure water service [1 400 kPa (ga) or less]. They are especially suitable for systems containing appreciable sand or other solids.

6.2.7 Needle valves

Needle valves are basically miniature globe valves. They are frequently used for instrument and pressure-gauge block valves, for throttling small volumes of instrument air, gas or hydraulic fluids and for reducing pressure pulsations in instrument lines. The small passageways through needle valves are easily plugged, and this should be considered in their use.

6.2.8 Check valves

Check valves are manufactured in a variety of designs, including swing-check valves, lift-plug check valves, ball check valves, piston check valves, non-slam check valves for compressor service, and dual-plate check valves. Of these, a full-opening, swing-check type is suitable for most non-pulsating applications. Swing-check valves may be used in vertical pipe runs (with flow in the upward direction) only if a stop is included to prevent the clapper from opening past top-dead-centre. Swing-check valves shall not be used in a downward direction in a vertical piping run. If used where there is pulsating flow or low flow velocities, swing-check valves will chatter, and eventually the sealing surfaces will be damaged. The clapper may be faced with stellite for longer life. To minimize leakage through the seat, a resilient seal should be used. Removable seats should be used since they make repair of the valve easier and also facilitate replacement of the resilient seal in the valve body. Swing-check valves should have a screwed or bolted bonnet to facilitate inspection or repair of the clapper and seats. In many cases, for a high-pressure swing-check valve to be in-line repairable, the minimum size is in the order of DN 65 (2,5 NPS) or DN 80 (3 NPS).

— Swing-check valves in a wafer design (which saves space) are available for installation between flanges. This type of check valve is normally not full-opening and requires removal from the line for repair.

— The dual-plate wafer-type check valve is normally used offshore because of its space- and mass-saving. However, spring failure can occur due to fatigue and therefore dual-plate check valves should not be used in pulsating services; non-slam check valves or swing check valves should be considered for such services.

— Lift-plug check valves should only be used in small, high pressure lines up to DN 40 (1,5 NPS), handling clean fluids. Lift-plug check valves can be designed for use in either horizontal or vertical lines, but the two are not interchangeable. Since lift-plug check valves usually depend on gravity for their operation, they may be subject to fouling by paraffin or debris.

— Ball check valves are very similar to lift-plug check valves. Since the ball is lifted by fluid pressure, this type of check valve does not have a tendency to slam as does a swing-check valve. They should therefore be the type chosen in sizes DN 50 (2 NPS) or smaller for clean services that have frequent flow reversals.

— Self-balancing, axial-piston, non-slam check valves should be used in pulsating flow, such as in reciprocating compressor or pump discharge lines. They should not be used for sandy or dirty fluid service. Piston check valves are equipped with an orifice to control the rate of movement of the piston. Orifices used for liquid services are considerably larger than orifices used for gas services. A piston check valve designed for gas service should not be used in liquid service unless the orifice in the piston is changed.

6.3 Fire resistance of valves

Not all valves perform satisfactorily under fire conditions and the user should consider this when selecting valves for process or critical duty. In hydrocarbon, flammable or toxic service, valves and valve materials should be inherently firesafe by design. Valves with soft seats or seals shall be of a certified fire type-tested design.

6.4 Valve sizing

6.4.1 In general, valves should correspond to the size of the piping in which the valves are installed. Unless special considerations require a full-opening valve (e.g. for sphere launching or receiving, minimum pressure drop required, meter proving, pump suction, etc.), regular-port valves are acceptable subject to process considerations (e.g. downstream of pressure safety valves require full-port valves).

6.4.2 In calculating the overall pressure drop in a piping system, it is common practice to add the equivalent length of valves to the length of straight pipe. Valve manufacturers usually publish data on their valves, either directly in terms of Equivalent Length of Straight Pipe, in metres, or as length/diameter ratios. If such data are not available for a particular valve, approximate values may be read from Table 3. Block valves and bypass valves, used in conjunction with control valves, should be sized in accordance with API RP 550 [18].

6.4.3 The pressure drop across a valve in liquid service may be calculated from the following equation from the Fluid Controls Institute, ref. [19]:

$$\Delta p = 133{,}7\ d_{\mathrm{L}} \left(\frac{q_{\mathrm{L}}}{C_{\mathrm{v}}} \right)^2 \tag{22}$$

where

Δp is the pressure drop, expressed in kilopascals (kPa);

q_{L} is the liquid flow rate, expressed in cubic metres per hour (m^3/h);

C_{v} is the valve coefficient, dimensionless;

d_{L} is the liquid relative density (water = 1), dimensionless.

6.4.4 For a valve in gas service, the following (Fluid Controls Institute) equation may be used:

$$\Delta p = 646{,}2 \times 10^{-6} \left(\frac{q_{\mathrm{g}}}{C_{\mathrm{v}}} \right)^2 \frac{d_{\mathrm{g}} \cdot T}{p} \tag{23}$$

where

Δp is the pressure drop, expressed in kilopascals (kPa);

d_{g} is the gas relative density (air = 1), dimensionless;

T is the flowing temperature, expressed in kelvin (K);

p is the flowing pressure, expressed in kilopascals [kPa (abs)];

q_{g} is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

C_{v} is the valve coefficient, dimensionless.

6.4.5 Values of C_{v} are usually published in valve manufacturers' catalogues.

6.5 Valve pressure and temperature ratings

6.5.1 Steel valves are manufactured in accordance with various International and national Standards, see ISO 10434 [20], API Std 602 [21], ISO 10423, API Spec 6D [22] or ASME B16.34 [23] for further information. In addition see API specifications for complete manufacturing details, and ASME B16.5 [24] for pressure-temperature ratings and dimensional details.

6.5.2 Most steel valves used in platform facilities are designed to the pressure-temperature ratings for steel pipe, flanges and flanged fittings given in ASME B16.5. For face-to-face and end-to-end dimensions for steel valves, see ISO 5752 [25] and ASME B16.10 [26] for further information. The allowable working pressure for an ASME B16.34, an ISO 10434, an API Std 602 or an API Spec 6D valve is a function of the operating temperature and body material.

6.5.3 Steel valves built to ISO 10423 are used primarily on wellheads and flowlines and they have pressure ratings of 13,8 MPa (2 000 psi), 20,7 MPa (3 000 psi), 34,5 MPa (5 000 psi), 69,0 MPa (10 000 psi), 103,5 MPa (15 000 psi) and 138,0 MPa (20 000 psi). The ISO 10423 designation numerically denotes the allowable working pressure in megapascals (pounds per square inch) for temperatures between –29 °C and 120 °C. It shall be taken into account that although ISO 10423 and ASME B16.5 flanges are similar dimensionally, they are fabricated from different materials and consequently have different pressure ratings, corrosion resistance and weldability.

6.5.4 The allowable working pressures and temperatures described above consider the metal valve parts only. For valves utilizing resilient sealing materials, the maximum allowable operating temperature is normally limited by the resilient material. The maximum permissible temperatures for valves are indicated in valve catalogues and should be included in the pipe, valves, and fittings tables described in annex B.

6.6 Valve materials

6.6.1 Non-corrosive service

For non-corrosive services, carbon steel should meet the requirements of ISO 10434, ISO 10423, API Std 6D. Valve bodies should meet the requirements of ASME B16.5 in terms of strength, ductility and resistance to damage by fire. See also clause 5.

Cast iron and ductile iron valves should not be used offshore; instead, steel valves (including cast steel valves) should be used. Non-ferrous valves are not suitable for process hydrocarbon service because they may fail in a fire; bronze body valves may be used for water services.

Resilient soft-sealing materials should be carefully selected to be compatible with the process fluids, temperature and pressure rating.

6.6.2 Corrosive service

Generally, carbon steel valve bodies with corrosion-resistant internal trims are used for corrosive service. AISI 410 type stainless steel is often used for internal trims. Austenitic stainless steels, such as AISI 316 and higher alloys may also be used for internal trims. For more corrosive service where corrosion-resistant piping materials are selected, the valve body should match the material of the piping system.

For low-pressure seawater services, butterfly valves with lined carbon steel body and corrosion-resistant trim are satisfactory. For seawater application aluminium-bronze gate valves also give good service.

6.6.3 Chloride stress-corrosion cracking service

Consideration should be given to chloride stress-corrosion cracking when selecting materials, see 4.4.3.

6.6.4 Sulfide stress-cracking service

Valve bodies and internal trim shall be in accordance with NACE MR0175 or constructed of materials which can be demonstrated to be resistant to sulfide stress-cracking in the environment for which it is proposed.

6.6.5 Hydrogen-induced cracking (HIC)

HIC can occur in services containing hydrogen sulfide, particularly in carbon steel plate fabrications. HIC-testing such materials shall be in accordance with NACE TM0284 using a test solution in accordance with NACE TM0177.

7 Fittings and flanges

7.1 General

Welded, screwed and flanged piping connections are acceptable for use in platform piping within the application limitations discussed in 5.7 and clause 7.

In this clause only carbon steel materials compatible with those in clause 4 are considered; if high strength pipe materials are used, fittings and flanges should have similar properties. The operator should select other materials, as needed, on an engineering basis (see clause 1 and 4.1).

Many of the fittings and flanges described in clause 7 should be manufactured using materials in accordance with ASTM A 105 [27], ASTM A 350 [28] and ASTM A 420 [29], and some grades require heat treatment (normalizing etc.) which would apply to all nominal pipe sizes. Fittings and flanges in this category should be marked HT, N or with some other appropriate marking to designate the heat treatment applied.

7.2 Welded fittings

7.2.1 The most common material grades for butt-weld fittings are seamless (see ASTM A 23 grade WPB or ASTM A 420 grade WPL6 for further details), although seam-welded fittings may also be used subject to a satisfactory joint efficiency and radiography of the weld.

7.2.2 For dimensions and tolerances for butt-weld, long-radius elbows and tees see ASME B16.9 [30] for further details. For butt-weld, short-radius elbows see ASME B16.28 [31] for further information. If space permits, long-radius elbows should be used in platform piping systems. Short-radius elbows are subject to high stress concentrations in the throat of the sharp-curvature bend, and should be derated to 80 % of the calculated allowable working pressure.

7.2.3 The pressure rating of steel butt-weld fittings should be equal to that of seamless pipe of the same wall thickness (see ASME B16.9 for further information). Butt-weld fittings should, therefore, be purchased with a wall thickness to match the pipe to which they will be connected, except if thicker walls are required for short-radius elbows which have been derated as outlined in (7.2.2). Otherwise, the pressure rating of the system will be limited by the lesser wall thickness in the fittings. Due consideration should also be given to the difference in manufacturing tolerance between fittings made from seamless pipe or plate. If it is necessary to weld fittings or pipe with unequal wall thicknesses, the procedure outlined in annex C should be used to design the joint.

7.2.4 For the manufacture of socket welding elbows and tees see ASME B16.11 [32] for further information. Normally, ASTM A 105 and ASTM A 350 grade LF2 steel forgings are used. Socket welding fittings are supplied in pressure ratings of 2 000 psi, 3 000 psi, and 6 000 psi for non-shock water, oil and gas (WOG) service.

7.2.5 The pressure drop due to welded fittings may be calculated by including their equivalent length in the total length of the piping system. Equivalent lengths for welded elbows and tees are included in Table 3.

7.3 Screwed fittings

Screwed fittings, if permitted by 5.7.4, should be forged steel and should be used for utility service only. Forged steel screwed fittings are normally manufactured in accordance with ASTM A 105 in 2 000 psi, 3 000 psi and 6 000 psi pressure ratings (see ASME B16.11 for further information).

7.4 Branch connections

7.4.1 Branch connections in welded lines should be butt-weld straight tees or reducing tees if the branch line is DN 50 (2 NPS) or larger, and is no smaller than one half of the nominal run size. If the branch line is DN 50 (2 NPS) or larger, but smaller than one half of the nominal run size, welded nozzles shall be in accordance with ASME Section VIII, Division 1, Part UW or other appropriate national pressure vessel codes. Branch lines DN 40 (1½ NPS) and smaller should be connected to runs DN 40 (1½ NPS) and smaller with, for example, socket-weld tees, and to runs DN 50 (2 NPS) and larger with socket-weld branch fittings. Table 6 illustrates the application of this.

7.4.2 Stub-in connections should not be used. The disadvantages of an unreinforced stub-in connection are numerous. Sharp changes in section and direction at the junction introduce severe stress intensifications. Reinforcement using a pad or saddle improves the situation somewhat; however, the finished connection is difficult to examine for weld defects and other defects. The likelihood that stress-intensifying defects will escape detection make stub-ins a poor choice for critical hydrocarbon services or those with severe cyclic operating conditions or loadings.

7.4.3 Branch connections in screwed piping systems should be made using straight tees and reducers, or reduced-outlet tees. All screwed piping systems should be isolated from welded piping systems by block valves.

Table 6 — Typical branch connection schedule — Welded piping

Nominal branch size NPS	Nominal run size (NPS)																
	½	**3/4**	**1**	**1½**	**2**	**2½**	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**	**18**	**20**	**24**
½	SWT	SWT	SWT	SWT	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL
3/4		SWT	SWT	SWT	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL
1			SWT	SWT	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL
1½				SWT	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL	SOL
2					T	RT	RT	RT	WOL	WOL	WOL	WOL	WOL	WOL	WOL	WOL	WOL
2½						T	RT	RT	WOL	WOL	WOL	WOL	WOL	WOL	WOL	WOL	WOL
3							T	RT	RT	WOL	WOL	WOL	WOL	WOL	WOL	WOL	WOL
4								T	RT	RT	WOL	WOL	WOL	WOL	WOL	WOL	WOL
6									T	RT	RT	RT	WOL	WOL	WOL	WOL	WOL
8										T	RT	RT	RT	RT	WOL	WOL	WOL
10											T	RT	RT	RT	RT	RT	WOL
12												T	RT	RT	RT	RT	RT
14													T	RT	RT	RT	RT
16														T	RT	RT	RT
18															T	RT	RT
20																T	RT
24																	T

T - Straight tee (butt-weld)

RT - Reducing tee (butt-weld)

WOL - Welded nozzle or equivalent (schedule of branch pipe)

SOL - Socket-weld branch fittings (forged steel)

SWT - Socket-weld tee

7.5 Flanges

7.5.1 General

7.5.1.1 Welding-neck flanges should be used for piping DN 50 (2 NPS) and larger. Slip-on flanges should not normally be used. ASME-type flanges are used in most applications (see ASME B16.5 for further details for sizes up to and including DN 600 (24 NPS) and see ASME B16.47 [33] for further information on larger sizes). Flanges used primarily near the wellhead should be manufactured in accordance with ISO 10423. Clamped connectors and proprietary compact flanges may be considered for mass and space savings.

7.5.1.2 ASME flanges are supplied in raised face (RF) and ring type joint (RTJ) designs. RF flanges offer easier maintenance and equipment replacement than RTJ flanges. RTJ flanges are commonly used in higher pressure services (ASME class 900 and higher) and may also be used for ASME class 600 piping systems that are subject to vibrating service. RTJ flanges should also be considered for services with special temperature or hazard problems. Where RTJ flanges are used, the piping configuration should be designed to allow component removal since additional flexibility is required to remove the ring gasket. For materials for carbon steel ASME flanges, see ASTM A 105 or ASTM A 350 for further information.

7.5.1.3 Materials for flanges should be in accordance with ISO 10423. Some material types require special welding procedures. Flanges specified in ISO 10423 are available in pressure ratings of 13,8 MPa (2 000 psi), 20,7 MPa (3 000 psi), 34,5 MPa (5 000 psi), 69,0 MPa (10 000 psi), 103,5 MPa (15 000 psi) and 138,0 MPa (20 000 psi). The maximum working pressure ratings are applicable for temperatures between – 29 °C and 121 °C. Flanges rated 13,8 MPa (2 000 psi), 20,7 MPa (3 000 psi), and 34,5 MPa (5 000 psi) are designated Type 6B, and require Type R or RX gaskets, except in a few sizes where Type BX gaskets are necessary. Flanges rated 69,0 MPa (10 000 psi), 103,5 MPa (15 000 psi) and 138,0 MPa (20 000 psi) are designated Type 6BX and require BX ring gaskets. Type 6B flanges should have a full-face contour. Type 6BX flanges should have a relieved-face contour.

7.5.2 Gaskets

7.5.2.1 For ASME raised face (RF) flanges, spiral-wound asbestos or graphite gaskets with stainless steel windings should be used because of their strength and sealing ability. For flat-face ductile iron or cast iron valves used in water service, full-face compressed asbestos gaskets or arimid or glass-reinforced, elastomer-bound gaskets should be used. If less than full-face gaskets are used on flat faces, cast iron flanges may be broken when the flange bolts are tightened. If gaskets containing asbestos are unavailable or cannot be used in a particular service, other gasket materials are available. Flexible-graphite sheet, laminated with stainless steel reinforcement and graphite-filled spiral wound gaskets are an acceptable alternate for asbestos for hydrocarbon and steam services. Use of graphite or other asbestos substitutes for other services should be preceded by service-specific studies.

WARNING — Materials containing asbestos may be subject to legislation that requires precautions to be taken when handling them to ensure that they do not constitute a hazard to health. In a growing number of countries the use of products containing asbestos is discouraged or banned.

7.5.2.2 Ring gaskets for API and ASME RTJ flanges should be manufactured in accordance with ISO 10423. API ring-joint gaskets are made of either soft iron, low carbon steel, AISI 304 stainless steel or AISI 316 stainless steel. Unless otherwise specified by the operator, ring-joint gaskets made of soft iron or low carbon steel are plated with cadmium or other suitable metal. For ASME and API Type 6B RTJ flanges, either API Type R or RX gaskets shall be used. Type R ring-joint gaskets are made with either an octagonal or oval cross-section. Type RX ring joint gaskets are pressure-energized and have a modified octagonal cross-section. Type R and RX gaskets are interchangeable; however, RX gaskets have a greater ring height so the distance between made-up flanges is greater and longer flange bolts are required.

7.5.2.3 Type R ring-joint gaskets of soft iron should be used in ASME class 600 and ASME class 900 services (see 7.5.1). Type RX ring-joint gaskets of low carbon steel provide a better seal at high pressure and should be used in ASME class 1500 and higher pressure ratings and in 2 000 psi, 3 000 psi and 5 000 psi pressure ratings.

7.5.2.4 Type 6BX flanges require Type BX pressure-energized ring gaskets. These gaskets, made of low carbon steel, should be used for 69,0 MPa (10 000 psi), 103,5 MPa (15 000 psi) and 138,0 MPa (20 000 psi) flanges.

7.5.2.5 For gaskets in potable water systems, see 8.6.3.

7.5.3 Flange protectors

Various methods (painting, wrapping with tape, etc.) have been tried to protect gaskets, bolts and flange faces from corrosion but none have been completely satisfactory. Potential solutions include:

— installing soft rubber flange protectors (150 °C limit) when the flange is made up, and

— stainless steel or polymer bands with a grease fitting.

For H_2S service, the bolts should be left open to atmosphere for ventilation, or bolts suitable for H_2S service and the flange pressure rating should be used.

7.5.4 Bolts and nuts

For flanged piping systems, stud bolts, threaded over their length in accordance with ASTM A 193 [34] grade B7, should be used. Nuts should be heavy hexagon, semi-finished, in accordance with ASTM A 194 [35] grade 2H. However, for low temperature service, bolting to ASTM A 320 [36] grade L7 and heavy hexagon nuts to ASTM A 194 grade 4 or grade 7 should be considered (with impact testing in accordance with supplementary requirement S4). Bolts and nuts should be protected from corrosion; current methods include cadmium plating, hot dip galvanizing, and resin coating.

WARNING — Hazards are involved in the process of cadmium-plating, this is resulting in a greater use of alternative coatings.

7.6 Proprietary connectors

Several proprietary connectors are available to replace flanges. Such connectors are satisfactory if they have sealing, strength and fire-resistance qualities comparable to flanges. Specially-machined hubs and ring gaskets are required on valves and fittings to mate with the connectors. Alignment may be critical.

7.7 Special requirements for sulfide stress-cracking service

Fitting and flange materials, as normally manufactured, are generally satisfactory for sulfide stress-cracking service with the additional stipulation that they should be modified to conform to the requirements of NACE MR0175. ASTM A 194 grade 2HM nuts, and ASTM A 193 grade B7M bolts or bolting to ASTM A 320 grade L7M and heavy hexagon nuts to ASTM A 194 grade 7M (with impact testing to ASTM A 194, S4) for low temperature service, are generally satisfactory for pipe flanges. Consideration should also be given to torque requirements during installation. Type R and RX rings should be made of annealed AISI 316 stainless steel, or soft iron in full conformance with NACE MR0175, dependent on the process (see also 4.4.5 and 4.4.6).

7.8 Erosion prevention

To minimize erosion where sand production is expected, short-radius pipe elbows should not be used. All turns in flowlines should be made with tees and weld caps (or blind flanges), capped or target tees, or long-radius bends (minimum bending radius normally 3D or 5D). For maximum velocities to minimize erosion, see 5.5.1.

8 Design considerations for particular piping systems

8.1 General

Clause 8 presents considerations for flow schemes, piping layouts and specific design requirements for particular piping systems.

8.2 Wellhead accessory items

8.2.1 Sampling and injection connections

Connections may be desired near the wellhead for chemical injection and for obtaining samples (see Figures 5 and 6). If installed, they should be DN 15 (½ NPS) minimum nominal size and include a close-coupled block valve. Associated piping shall be in suitable material for the process and injection fluid duty and consist of heavy-wall pipe and should be well protected to minimize the possibility of damage. On injection lines, a spring-loaded ball check valve should be close-coupled to the block valve on the injection side. On sampling connections, a block valve should be close-coupled with a throttling valve on the downstream side. Consideration should be given to double-block isolation on the sampling connection.

8.2.2 Chokes

Chokes are normally installed to control the flow from oil and gas wells. The types of choke include adjustable chokes, positive chokes and combination chokes. The number and location of chokes depend on the amount of pressure drop taken, well fluid, flow rate, and solids in the wellstream. Usually, if only one choke is used, it should be located near the wellhead. Additional chokes may be located near the manifold, entering the separation units. This may be in conjunction with line heaters, as appropriate, for gas wells.

The following general guidelines should be considered regardless of the number of chokes or their location:

a) choke bodies should be installed in a manner that will permit easy removal and trim changes;

b) the downstream flow passage within 10 to 20 nominal pipe diameters should be free of abrupt changes in direction to minimize flow cutting due to high velocity;

c) outlet connections should be examined to determine if their bore should be tapered to improve flow patterns;

d) suitable provisions should be provided to isolate and depressure the choke body when changing trim, removing trash, etc.;

e) materials shall be suitable for the temperatures downstream of chokes.

8.3 Flowline and flowline accessories

8.3.1 Flowline design considerations

When designing flowlines, consideration should be given to pressure, temperature, velocity, erosive and corrosive effects on the pipe, etc. (See clauses 5, 6 and 7). The accessories discussed below may be used as necessary.

8.3.2 Flowline pressure sensor

The installation of a flowline sensor should be in accordance with API RP 14C [37]. Further, the connection should be DN 15 (½ NPS) minimum nominal size and located to minimize the possibility of plugging and freezing. Connections on the bottom of the line or in bends should be avoided. Sensors should be installed with an external test connection and block valve. Double-block or double-block-and-bleed isolation valve arrangements should be considered to permit on-line maintenance. Sensing lines should be stainless steel or other material suitable for the fluid service and secured to prevent whip in the event they are severed.

8.3.3 Flowline flowmeter

A flowmeter, as shown on Figures 5 and 6, may be desirable in gas well service either for a well monitoring aid or as a means of production allocation.

8.3.4 Flowline heat exchanger

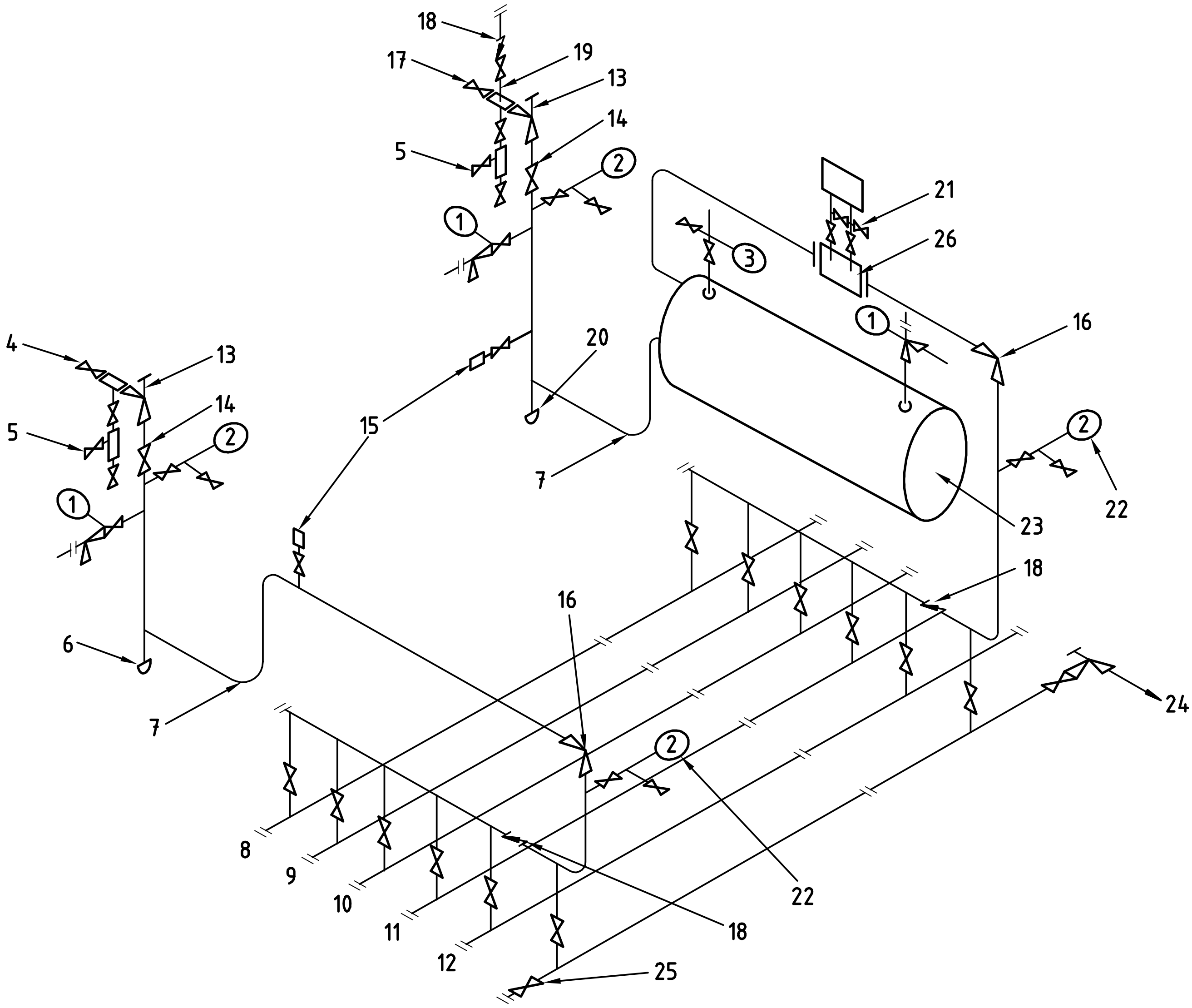
If a flowline heat exchanger is used (see Figures 5 and 6), the following provisions should be considered.

a) Connections should be arranged so that the exchanger bundle may be pulled without having to cut or weld on inlet and outlet piping.

b) On the flanged end of the shell, exchanger U-bends, if used, should either be exposed to the exterior or be easily accessible for non-destructive testing.

c) A flanged-end heat exchanger shell of a standard dimension should be used so that bundles can be cleaned, pulled, repaired or interchanged.

d) A relief system in accordance with 8.8 should be provided.



Block valves under relief valves shall be locked open.

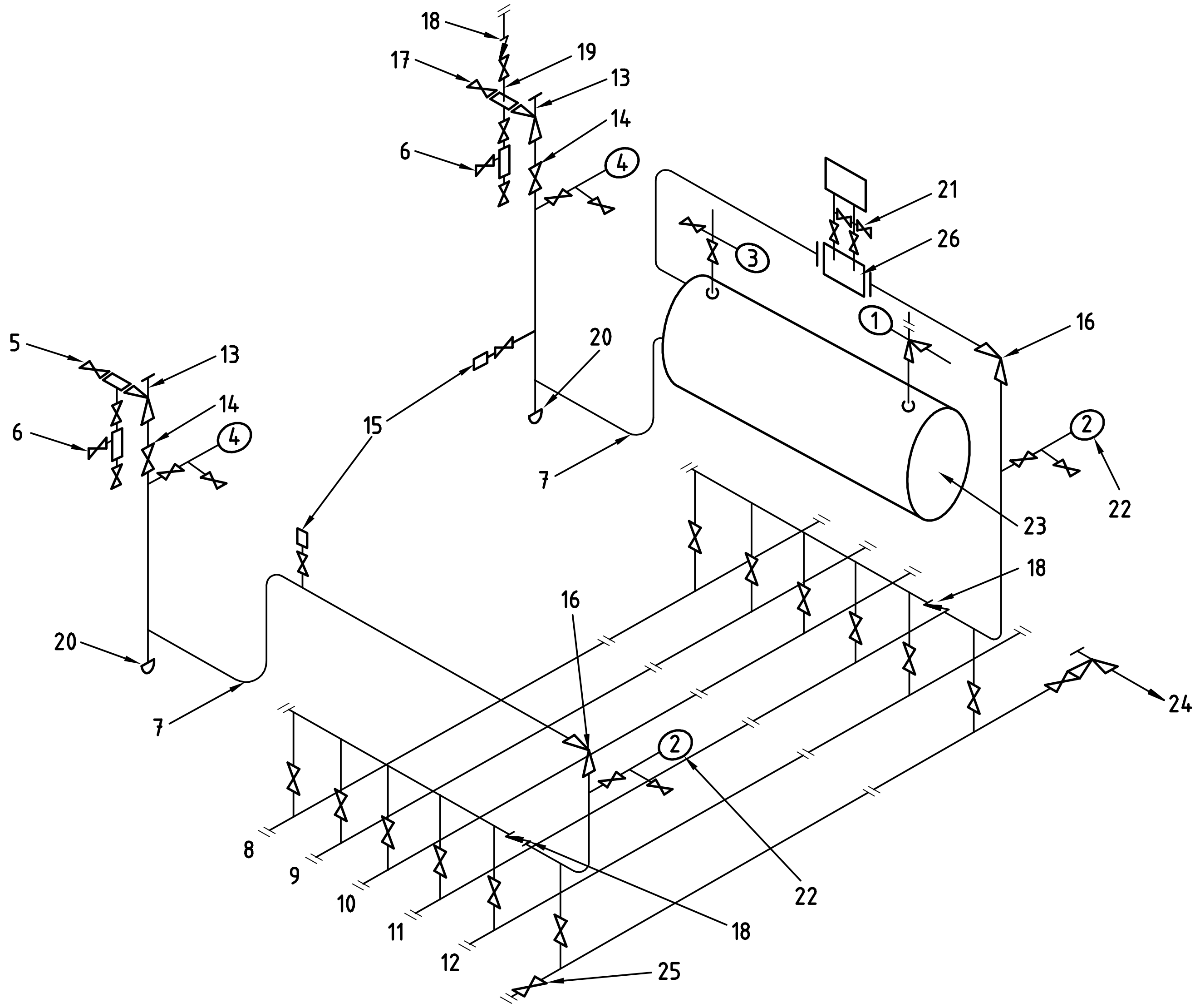
NOTE 1 This figure is included to illustrate points made in the text. The accessories and piping arrangements should not be interpreted as being typical or recommended.

NOTE 2 Relief valves vent to appropriate disposal systems (see 8.8).

Key

1 Pressure safety valve
2 Pressure switch high-low
3 Pressure switch high
4 Oil well Xmas tree wing
5 Sample catcher
6 Capped tee
7 Long radius bend
8 Atmosphere
9 Low pressure test
10 High pressure test
11 Low pressure prod.
12 High pressure prod.
13 Wing choke
14 Flowline valve
15 Sand monitoring equipment
16 Secondary choke
17 Gas well Xmas tree wing
18 Check valve
19 Chemical injection line
20 Flow tee
21 Flowmeter
22 (Used in conjunction with secondary choke)
23 Flowline heat exchanger (needed only for externally heated gas well streams)
24 To atmospheric blowdown separator or sump
25 From auxiliary service pump
26 Fluid or gas meter

Figure 5 — Example schematic drawing of piping and accessories for flowlines and manifolds that are not rated for wellhead pressure



Block valves under relief valves shall be locked open.

All piping upstream of the example headers shown, including manifold block valves and block and relief valves on the example auxiliary service pump line, shall be designed for wellhead pressure.

NOTE 1 This figure is included to illustrate points made in the text. The accessories and piping arrangements should not be interpreted as being typical or recommended.

NOTE 2 Relief valves vent to appropriate disposal systems (see 8.8).

Key

1 Pressure safety valve
2 Pressure switch high-low
3 Pressure switch high
4 Pressure switch low
5 Oil well Xmas tree wing
6 Sample catcher
7 Long radius bend
8 Atmosphere
9 Low pressure test
10 High pressure test
11 Low pressure prod.
12 High pressure prod.
13 Wing choke
14 Flowline valve
15 Sand monitoring equipment
16 Secondary choke
17 Gas well Xmas tree wing
18 Check valve
19 Chemical injection line
20 Flow tee
21 Flowmeter
22 (Used in conjunction with secondary choke)
23 Flowline heat exchanger (needed only for externally heated gas well streams)
24 To atmospheric blowdown separator or sump
25 From auxiliary service pump
26 Fluid or gas meter

Figure 6 — Example schematic drawing of piping and accessories for flowlines and manifolds that are rated for wellhead pressure

8.3.5 Flowline sand monitors

Where large quantities of sand are envisaged, sand-monitoring equipment should be provided on the flowlines.

8.3.6 Flowline check valve

A flowline check valve should be installed to minimize back-flow due to inadvertent switching of a low-pressure well into a higher-pressure system, or in case of line rupture. Provisions should be made for blowdown of the flowline segment between the wellhead and check valve to facilitate periodic testing at the check valve. See 6.2.8 for check valve details.

8.3.7 Flowline support

Flowlines shall be supported and secured to minimize vibration and to prevent whip. See 9.4 for piping support details. When designing flowline supports, it should be recognized that even though the wellhead may be fixed to the platform, there is a possibility of independent wellhead movement due to wave action, wind forces, thermal, pressure change, settlement, etc. on the conductor pipe and casing.

8.4 Production manifolds

8.4.1 General

For illustration purposes, Figures 5 and 6 show a six-header manifold. The actual number and function of headers depend upon the specific application. The pipe routing to the production headers should be the shortest and least tortuous, with adequate flexibility, see 5.8. Each part of the manifold shall be designed to limit maximum velocity in accordance with 5.5. Fabricated components should be given PWHT, if required, in accordance with ASME B31.3. Provisions should be made to allow non-destructive testing of headers. Quarter-turn valves should be used in manifold service due to their ease of operation. Gate valves are generally more available for working pressures above 41 400 kPa (ga).

8.4.2 Manifold branch connections

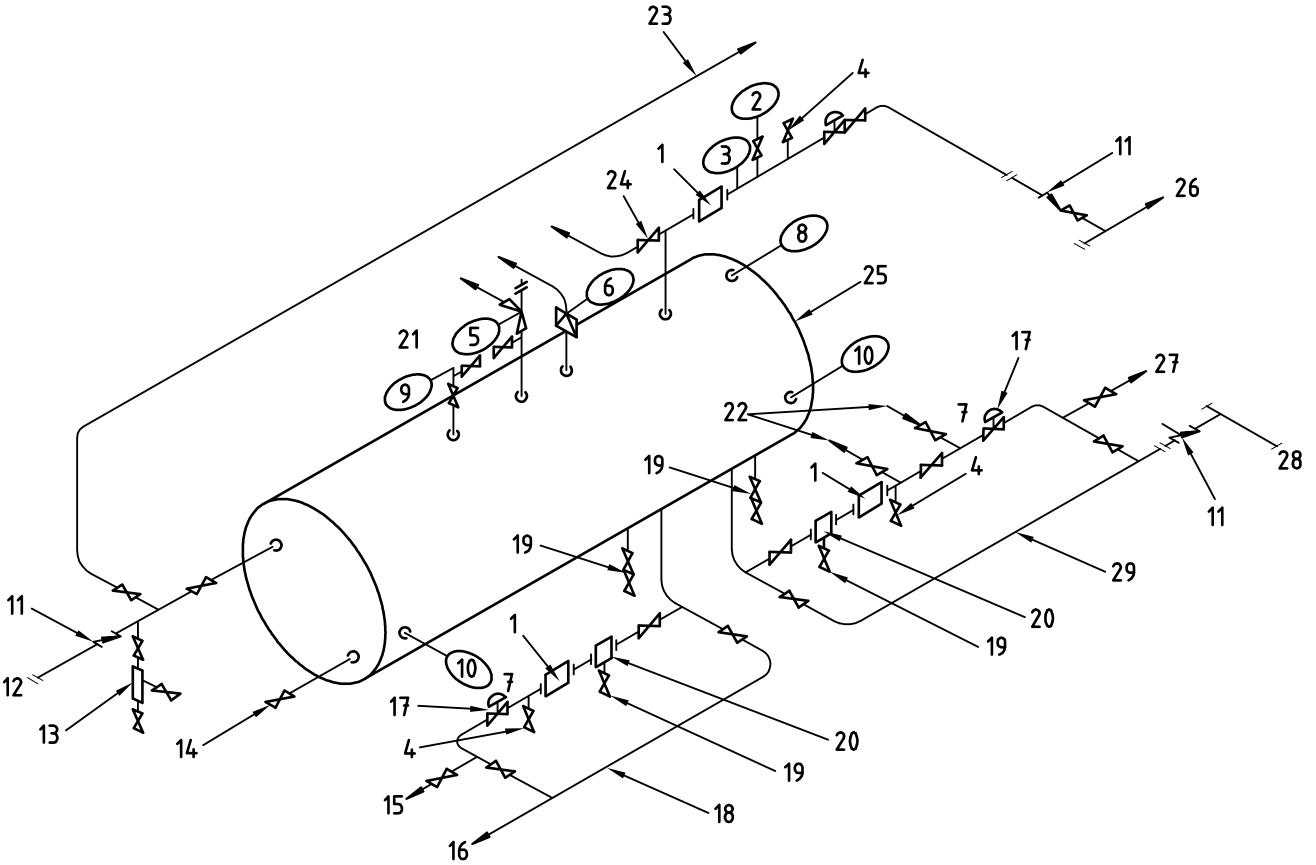
Manifold branch connections should be in accordance with Table 6. If proprietary branch connections are used, care should be taken to ensure the entrance hole is smooth and free of burrs after it is welded in place. The ends of the manifold runs should be blind-flanged to provide a fluid cushion area and for possible future expansion.

8.4.3 Manifold valve installation

The manifold arrangement should provide easy access to each valve for operational purposes and easy removal. In initial design of the manifold, provisions for the installation of actuated valves should be considered.

8.5 Process vessel piping

A typical three-phase process vessel with standard accessories and many optional items is shown in Figure 7. Different vessels are required for different functions in processing, however, all the flow streams to and from a vessel are generally handled in a similar manner. All the items shown in Figure 7 may not be needed, but are shown where they should be located when required. Accessories should be installed to permit easy removal for repair or replacement. Safety devices should be capable of being tested in place.



Block valves under relief valves shall be locked open.

NOTE 1 This figure is included to illustrate points made in the text. The accessories and piping arrangements should not be interpreted as being typical or recommended.

NOTE 2 Relief valves vent to appropriate disposal systems (see 8.8).

Key

1 Fluid or gas meter
2 Pressure indicator
3 Temperature indicator
4 Sample connection
5 Pressure safety valve
6 Pressure safety element
7 Pilot-operated control valve
8 Level switch high
9 Pressure switch high-low
10 Level switch low
11 Check valve
12 Inlet
13 Sample catcher
14 Sand jet
15 To meter-prover
16 To water process
17 Control valve
18 Bypass
19 Drain
20 Strainer
21 Example pressure-relief installation
22 Meter-prover connections
23 Bypass to alternative process vessel or to pipeline
24 Blowdown
25 Example three-phase process vessel
26 To dehydration, sales, etc.
27 Alternative meter-prover connection
28 Storage, etc.
29 Oil and/or condensate outlet bypass

Figure 7 — Example schematic drawing of three-phase process vessel showing various accessories

8.6 Utility systems

8.6.1 Pneumatic systems

8.6.1.1 Piping and fittings

Pneumatic systems shall provide a dependable supply of air or gas for pneumatically-operated components. All pipe, tubing and fittings DN 9 (3/8 NPS) nominal size and smaller should be AISI 316 or AISI 316L stainless steel, with 0,889 mm minimum wall thickness where exposed to the atmosphere. Piping should be installed in a manner that will minimize low points or traps for liquid. Outlets from vessels and piping should be from the top of the system and drains from the bottom. Blowdown provisions should be included in the piping system to allow removal of condensation. If pneumatic systems are to be pneumatically tested, see 10.3.

8.6.1.2 Air systems

Main air headers should utilize corrosion-resistant material such as threaded-and-coupled galvanized steel or stainless steel (AISI 316, or AISI 316L). Piping drops to instruments should be in accordance with 8.6.1.1. Care shall be exercised in locating air compressor suctions to preclude the introduction of gas or hydrocarbon vapours into the system. Crossovers, whereby air and gas could be intermixed, shall not be allowed anywhere in the system.

8.6.2 Gas systems

For gas systems, vents and relief valves should be piped to a safe location if it is determined that the volumes capable of being vented could create an abnormal condition. The gas source chosen should be the driest gas available on the platform. The following guidelines may be helpful in designing an instrument gas system.

a) Consideration should be given to the necessity of dehydrating the gas prior to taking a significant pressure drop. An external heat source may be required to prevent freezing if the gas is not dehydrated.

b) The gas should be expanded into a separator to prevent hydrates and liquids from entering the system piping.

c) The inlet and outlet pressure rating of pressure-reduction devices should be carefully considered. If the outlet pressure rating is less than the inlet source pressure, a relief device shall be close-coupled to the reduction device.

d) Parallel reduction devices should be considered to maintain system operation in the event of the primary device failing.

8.6.3 Fire-water systems

Fire-water systems are normally constructed of corrosion-resistant materials such as 90/10 copper nickel alloy and high alloy stainless steel, or carbon steel with or without coating. The fire-water system may require sectionalizing valves so that if one portion of the system is inoperable it can be closed off and the remainder of the system can remain in service. Accessibility during a fire shall be considered when locating fire-hose stations and/or turrets (monitors). In the determination of required flowrates, consideration shall be given to the surface area, the location of the equipment and the maximum number of discharge nozzles which could be in use simultaneously. See NFPA 6 [38], NFPA 8 [39] and API RP 14G [40] for further information.

8.6.4 Potable-water systems

Potable-water systems may be constructed of galvanized carbon steel, type 316 stainless steel, copper or copper alloys, or fibre-reinforced plastic. Toxic joint compounds should not be used. If water makers are used, consideration shall be given to potential contamination of the water from heating sources. When potable water is supplied to other facilities such as engine jacket water makeup, etc., care should be exercised to prevent contamination from back-flow.

8.6.5 Sewage systems

Interior sewage piping, such as in living quarters areas, should be galvanized carbon steel, coated carbon steel, copper nickel pipe or non-metallic pipe, properly supported in accordance with ASME B31.3. Exterior piping may be galvanized carbon steel, coated carbon steel, copper nickel pipe, or non-metallic pipe (if properly supported and protected from sunlight). All piping should be well-supported and have a minimum slope of 1 %. The system should be designed with adequate clean-out provisions. Discharge lines from sewage treatment plants should terminate below water level and contain readily-accessible sampling connections. Care should be exercised in locating odorous vents.

8.7 Heating fluid and glycol systems

8.7.1 The paramount safety consideration in the design of heating fluid systems is containment of the fluid for personnel protection and fire prevention. All piping, valves and fittings should be in accordance with clauses 5, 6 and 7 except that flanges for other than low pressure steam and hot water systems should be minimum of ASME class 300 in order to minimize leakage. Piping shall be designed for thermal expansion (see 5.8). Thermal insulation should be in accordance with 9.6.

8.7.2 If the process side of a shell-and-tube heat exchanger has a higher operating pressure than the design pressure of the heating fluid side, the heating fluid side shall be protected by a relief device. The location of the relief device depends on the actual design of the system. If possible, the relief device should be located on the expansion (surge) tank which will serve as a separator. All heating fluid should pass through an expansion (surge) tank to allow venting of any gas present. A relief device may also be required at the heat exchanger. Consideration shall also be given to tube failure in heat exchangers if the design pressure of the low pressure side is less than two thirds the design pressure of the high pressure side.

8.7.3 The effect of mixing hot fluids with cold fluids (see API RP 521 [41] for further information) should be considered when determining how to dispose of the discharge of a relief device on a heat exchanger. A separate scrubber may be required.

8.7.4 Heating systems (except hot water or steam) should be pneumatically tested in accordance with 10.3.3. If hydrostatically tested, provisions shall be made for removal of all water from the system before placing in service. Additionally, any water remaining after draining should be removed at start-up, by slowly bringing the system to 100 °C and venting the generated steam. Care should be taken to ensure that each branch of the system has circulation during this period.

8.7.5 The exhaust stream from a glycol reboiler contains steam and hydrocarbon vapours. Caution shall be exercised in the design of reboiler exhaust piping to prevent excessive back-pressure, ignition and condensation problems.

8.8 Pressure relief and disposal systems

8.8.1 General

8.8.1.1 Pressure relief and disposal systems are required to prevent over-pressure of equipment and piping systems and to dispose of the relieved product in a safe manner. Some possible causes of over-pressure are downstream blockage, upstream control valve malfunction and external fire.

8.8.1.2 The commonly used safety relief devices are the conventional spring-loaded relief valve, the balanced-bellows spring-loaded relief valve, the pilot-operated relief valve, the pressure-vacuum relief valve and the rupture disc. The description, operation, sizing, pressure setting and application shall be in accordance with ASME Boiler and Pressure Vessel Code, Section VIII or appropriate national pressure vessel codes. In addition, see API RP 520-1 [42], API RP 521 and ISO 10418 [43] for further information on pressure relief systems.

8.8.1.3 Relief devices in gas or vapour service should normally be connected to either the vessel vapour space or the outlet piping. They should be located upstream of wire-mesh mist extractors. Liquid-relief devices should be located below the normal liquid level.

8.8.1.4 If vessels with the same operating pressure are in series, a relief device set at the lowest design pressure in the system may be installed on the first vessel. If any remaining vessel can be isolated, a relief device sized for fire or thermal expansion shall be installed. Relief devices shall be located so they cannot be isolated from any part of the system being protected.

8.8.2 Relief device piping

8.8.2.1 This subclause applies to systems covered by ASME Section VIII, Division 1. For heating boilers, ASME Boiler and Pressure Vessel Code Section IV [44] and other appropriate national pressure vessel codes may be consulted for further information.

8.8.2.2 If the relief device has to be removed, process systems connected to a common relief header shall be shut down. Alternatively, a full-opening block valve may be installed downstream of relief devices if connected to a common relief header. All block valves installed either upstream or downstream of relief devices shall be equipped with locking devices and operated in accordance with ASME Section VIII annex M or other national pressure vessel codes.

8.8.2.3 Piping on the exhaust side of relief devices should be designed to minimize stress on the devices. The piping shall also be designed to withstand the maximum back-pressure to which it could be subjected. API Std 526 [45] may be consulted for further information on the allowable working pressure of relief valves.

8.8.2.4 Relief devices shall be located (as far as practicable) so that outlet piping is self-draining in order to prevent any build-up of liquids/condensed vapours in the device or the outlet piping.

8.8.2.5 There are two types of flame arrestor.

a) Explosion-protection flame arrestor.

   This type of flame arrestor shall not be used if

   1) the length of the connecting line to the potential ignition point is more than 20 times the pipe internal diameter, or

   2) fittings that can provoke an acceleration of the flame front (elbows, restrictions) are installed between the flame arrestor and the potential ignition point.

b) Detonation-protection flame arrestor.

   This type of flame arrestor shall be used in the cases cited in a) above, and it shall include a dumping chamber. Where this type is used, the length of the connecting line to the potential ignition point shall be more than 30 times the pipe inside diameter, the flame arrestor being as close as possible to the equipment being protected.

The pressure loss of the flame arrestor shall be checked against the operating conditions of the line.

The installation and testing of flame arrestors should be to recognized standards, see prEN 12874 [46] for further information.

8.8.3 Relief (disposal) system piping

8.8.3.1 The relief system and piping shall be designed to dispose of the relieved product in a safe and reliable manner. The system and piping should be designed to prevent liquid accumulation so that the design relieving capacity of any of the pressure-relieving devices is not reduced. Additionally, the design should limit back-pressure. The maximum possible back-pressure at each relief point should be determined. This is particularly important where two or more relief devices may relieve simultaneously into the same disposal system. The materials, fittings, welding and other design criteria shall be in accordance with the respective parts of this International Standard and API RP 520-2.

8.8.3.2 Vent or flare structures shall be designed to prevent buckling caused by wind moment. Vent or flare structures should be installed on the downwind side of the platform, taking into account helicopter and boat approaches, etc. In determining the height and distance from the platform, consideration shall be given to accidental ignition due to lightning, falling burning fluid and heat radiation.

8.8.3.3 When hydrocarbon vapours are discharged into the atmosphere, mixtures within the flammable range will occur downstream of the outlet. To determine the location of this flammable mixture, and the intensity of the heat should the mixture become ignited, API RP 521 and ref. [47] may be consulted. If toxic vapours are discharged into the atmosphere, systems shall be designed in accordance with local and national regulations; EPA AP-26 [48] may be consulted for further information.

8.8.3.4 If feasible, all relief systems should be designed for a minimum pressure of between 300 kPa (ga) and 500 kPa (ga) in order to contain flashback. In most cases, vents from atmospheric pressure equipment should be equipped with flame arrestors for flashback protection. Flame arrestors are subject to plugging with ice and should not be used in cold climates without suitable protection against freezing. Flame arrestors should be inspected periodically, e.g. for paraffin build-up.

8.9 Drain systems

8.9.1 Design considerations

Drain systems should be designed to collect and dispose of contaminants from all sources. The drain system prevents contaminants from spilling overboard, prevents the accumulation of flammable liquids on the deck or pans and promotes good housekeeping practices.

8.9.2 Closed drains

If closed drains from pressure vessels are used, they shall be sloped and piped directly to the disposal facilities, independent of the gravity drains, to prevent the introduction of fluids from the pressure drains into the gravity drains. The design pressure of the inter-connecting piping and drain valve on each process component shall be not less than the highest pressure to which the system could be subjected and shall correspond to the highest working pressure process component in the system. Piping should be in accordance with clauses 5, 6 and 7. A separate closed-drain system should be provided for hydrogen sulfide service to permit safe disposal of the fluids. Also, a separate closed-drain system should be provided for discharge from cold product lines to prevent freezing.

8.9.3 Gravity drains

Platform decks and skid pans are usually drained by gravity to the disposal facilities. A wide variety of materials may be satisfactory for this service. If steel pipe that is not satisfactory for hydrocarbon service is used, it shall be marked in accordance with 5.1.5. Consideration should be given to minimizing the number of bends and flow restrictions in the system. The gravity drain system shall be equipped with one or more vapour traps or other means to prevent gas migration from the collection vessel to the open drains. Piping should be installed with a downward slope in the order of 1 %. In some cases, it may be necessary to install runs in a horizontal plane, but there shall be no upslopes under any circumstances. Clean-out connections should be provided.

8.10 Bridge piping between platforms

Design of bridge piping is similar to that of other piping except that platform movement shall also be considered. Once the maximum platform movement has been determined, design shall be in accordance with 5.8.

8.11 Risers

Risers shall be designed for maximum wave loading, internal pressure, marine traffic and other environmental conditions. Risers should be within the supporting structure; if they are not they shall be protected by bumpers if they are on the exterior of the side of the jacket that is normally accessed by boats. Corrosion protection in the splash-zone is specified in 9.5.1.6. Further design criteria may be found in national pipeline codes.

8.12 Sampling valves

Valves for sampling process streams should be provided at appropriate locations. Valves should be located so that representative samples will be obtained. Sample valves may be used in conjunction with sample catchers or with sample tubes which extend into the centre of the pipe. Consideration should be given to the quality and condition of the stream at each location. Valve design and piping should allow for cleaning or rodding of valves that may become plugged with solids. Valves subject to large pressure drops may be quickly eroded. Doubling the valving and proper sampling procedures can minimize such problems. Sample valves are usually DN 15 (½ NPS), made from austenitic stainless steels or other material suitable for the process conditions.

9 Considerations of related items

9.1 General

This clause contains miscellaneous considerations related to piping systems.

9.2 Layout

The following items shall be considered when planning piping layout on production platforms:

a) safety of personnel;

b) compatibility with vessel, equipment and skid layout (see API RP 2G [49] for further information);

c) accessibility to equipment and maintenance;

d) pipe supporting;

e) walkways and escape routes;

f) operability access.

9.3 Elevations

Piping elevation on platforms shall be determined by factors such as the fluid transported, the temperature, drainage needs and personnel access. Piping shall not be installed on grating or flooring and there shall be adequate clearance for maintenance. Overhead piping should be arranged to provide sufficient head-clearance for personnel.

9.4 Piping supports

Platform piping should be supported on racks, stanchions or individual stand-offs. The location and design of supports are dependent upon routing, media, mass, diameter, shock loads, vibration, etc. Consideration should also be given to seal-welding supports to minimize corrosion and to providing sufficient clearance to permit painting. Valve supports should not interfere with removal of the valve for repair or replacement. Piping flexibility and support requirements should be in accordance with ASME B31.3. See 5.8 and 8.3.7 for further information.

9.5 Other corrosion considerations

9.5.1 Protective coatings for external surfaces

9.5.1.1 Coating standards

Design, selection and inspection of external coating of piping system should be in accordance with NACE RP0176 [50], sections 10, 12, 13 and 14. Additional points to consider with external coatings are provided below.

9.5.1.2 Types of platform piping coating systems

All steel piping should be considered for protection with a coating (painting) system which has been proven acceptable in a marine environment by prior performance or by suitable tests.

9.5.1.3 Selection of platform piping coating systems

When selecting coating systems, the following points should be considered:

a) surface preparation;

b) temperature range (maximum and minimum) expected;

c) dry and/or wet surfaces;

d) chemical contamination expected;

e) location of equipment under consideration (height above splash-zone, maximum wave height, etc.);

f) abrasion expected;

g) application of coating system;

h) maintenance.

Tapes and extruded coatings should generally not be used on platform piping. Experience has shown that severe corrosion can occur beneath these protective materials without any visible evidence. Because of the difficulty in sandblasting on platforms, it is desirable to sandblast, prime and/or paint pipe and accessories onshore.

9.5.1.4 Corrosion protection of insulated piping

Piping materials can suffer severe corrosion under insulation, and susceptible materials shall be protected by a paint system. Insulation should be kept to a minimum and the paint system should be inspected and maintained.

9.5.1.5 Chloride stress-corrosion cracking of stainless steel

Austenitic stainless steel can be susceptible to chloride stress-corrosion cracking above 60 °C in a saliferous atmosphere, and consideration shall be given to its prevention.

9.5.1.6 Risers

Risers shall be protected in the splash-zone, e.g. by coating or cladding.

9.5.2 Corrosion protection for internal surfaces

9.5.2.1 General

The best method to alleviate internal metal loss depends on the type and severity of the corrosion. Each case should be considered individually. See NACE RP0176 for guidelines and further information on detecting and controlling piping internal corrosion.

9.5.2.2 Process piping

Protective-type coatings should not normally be applied on the inside surfaces of process piping. Alternative methods of minimizing internal corrosion include:

a) dehydration of process streams;

b) use of corrosion inhibitors;

c) sizing of pipe for optimum velocity;

d) use of corrosion-resistant metals.

9.5.2.3 Water piping

Potential methods of minimizing internal corrosion of water-service piping include:

a) oxygen removal and/or exclusion;

b) chemical treatment (corrosion inhibition, biocides, scale and pH control);

c) protective coatings and linings (plastics, etc.);

d) non-metallic piping;

e) corrosion-resistant alloys;

f) control of solids (sand, mud and sludge).

9.5.2.4 Protective coatings

If internal coatings are used, the piping should be designed to facilitate fabrication without damage to the coatings.

9.5.3 Compatibility of materials

When different metals are coupled together in the presence of an electrolyte, galvanic action will occur. The greater the difference in electromotive force of the metals in the galvanic series, the more severe the corrosion. The possibility of galvanic corrosion shall be taken into account when selecting the materials of construction for piping, valves and fittings. Some general guidelines follow.

a) Whenever possible, use the same metal throughout, or metals that are close together on the galvanic series.

b) The following alternatives should be considered if it is necessary to utilize differing metals:

   1) break the couple with a non-conducting material;

   2) exclude the electrolyte by the application of a protective coating on one or both metals;

   3) keep the less noble, anodic member large in comparison with the more noble metal;

   4) fit a sacrificial spool piece.

9.5.4 Non-destructive erosion and/or corrosion surveys

Access should be provided to points in the piping system where erosion and/or corrosion is anticipated. These points should be examined by radiographic or ultrasonic methods and documented for comparison with future surveys. Corrosion and/or erosion may be monitored by coupons inserted in a piping branch. Special emphasis shall be given to well flowlines, bends and production manifolds.

9.5.5 Cathodic protection

If the submerged portion of a pipeline is to be cathodically protected by a system other than that used to protect the platform, its riser shall be electrically isolated from the platform. This may be accomplished with an insulating flange joint above the water and insulating material inserts within the support clamps below the insulating flange.

9.6 Thermal insulation

9.6.1 Temperature considerations

Thermal insulation shall be used on platform piping for prevention of combustion from hot surfaces, heat conservation, freeze protection and prevention of moisture or ice on piping. For personnel protection, all readily-accessible surfaces operating above 70 °C shall either be insulated or be fitted with wire shields or perforated shields. Surfaces with a temperature in excess of 205 °C shall be protected from liquid hydrocarbon spillage, and surfaces in excess of 480 °C shall be protected from combustible gases. The auto-ignition temperature of liquid and gas mixtures that could be present shall be established and the piping shall be protected if the surface temperature is in excess of the auto-ignition temperature. Typical values of auto-ignition temperature are 200 °C for liquids and 300 °C for gas mixtures.

9.6.2 Guidelines for proper insulation

a) Piping should be properly cleaned and painted before applying insulation.

b) Insulating material should be selected for the particular temperature conditions. Insulating material, such as magnesia, that could deteriorate or cause corrosion of the insulated surface if wet, should not be used. Some commonly-used insulation materials are calcium silicate, mineral wool, glass fibre, cellular glass.

c) A vapour barrier should be applied to the outer surface of the insulation on cold piping.

d) Insulation should be protected by sheet-metal jacketing from weather, oil spillage, mechanical wear, or other damage. If aluminium sheet-metal is used for this purpose, it should be protected by an internal moisture barrier.

e) To prevent H_2S from concentrating around the bolts, flanges shall not be insulated in H_2S service unless botting materials are used that are suitable for H_2S service in accordance with NACE MR0175.

f) Certain heating fluids are not compatible with some insulating materials and auto-ignition can occur. Caution should be exercised in selecting materials.

g) Certain insulation materials can be flammable or can give off toxic fumes in a fire.

h) Preformed insulation and jacketing are available for various sizes of pipe and fittings.

i) Corrosion of piping under insulation can occur and can remain undetected. Where alternatives to insulation are available (e.g. cages installed for personnel protection), they should be used instead (see also 9.5.1.4).

9.6.3 Typical insulation thickness

Tables 7, 8 and 9 provide typical minimum insulation thicknesses for various conditions of temperature and service.

Table 7 — Typical hot insulation thickness

Maximum temperature °C	**NPS**							
	⩽ 1½	**2**	**3**	**4**	**6**	**8**	**10**	**⩾ 12**
	Insulation thickness mm							
120	25	25	25	35	35	35	35	35
260	25	35	35	35	50	50	50	50
315	35	35	50	50	50	65	65	65
400	50	50	50	50	65	75	75	75

Table 8 — Typical cold insulation thickness

Min. temp. °C	NPS ½	3/4	1	1½	2	2½	3	4	6	8	10	12	14	16	18	20	24	30	Flat surf.
	Insulation thickness mm																		
5	25	25	25	25	25	25	25	25	25	35	35	35	35	35	35	35	35	35	35
0	25	25	25	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
– 5	35	35	35	35	35	35	35	35	50	50	50	50	50	50	50	50	50	50	50
– 10	35	35	35	35	50	50	50	50	50	50	65	65	65	65	65	65	65	65	65
– 20	35	50	50	50	50	50	50	65	65	65	65	65	65	65	65	65	75	75	75
– 25	50	50	50	50	50	65	65	65	65	65	75	75	75	75	75	75	75	75	90
– 30	50	50	50	65	65	65	65	65	70	70	70	70	70	90	90	90	90	90	100

Table 9 — Personnel protection temperature range for typical insulation thicknesses

NPS	Insulation thickness mm 25	35
	Hot surface temperature range °C	
½	70 to 400	401 to 560
¾	70 to 340	341 to 500
1	70 to 390	391 to 515
1½	70 to 350	351 to 470
2	70 to 340	341 to 465
2½	70 to 325	326 to 515
3	70 to 315	316 to 430
4	70 to 315	316 to 420
6	70 to 290	291 to 400
8	—	70 to 400
10	—	70 to 400
12	—	70 to 400
14	—	70 to 370
16	—	70 to 365
18	—	70 to 365
20	—	70 to 365
24	—	70 to 360
30	—	70 to 360
Flat surface [a]	70 to 270	271 to 350

[a] Application range also applies to piping and equipment with NPS ⩾ 30.

9.7 Noise

9.7.1 In the design of platform piping systems, provisions shall be made to protect personnel from harmful noise. See API Medical Research Report EA 7301 [51] for further information on problems and solutions. A general discussion of noise related to piping systems is included below.

9.7.2 Noise in a piping configuration is caused by the turbulence of a fluid passing through the system. Turbulence is created downstream of restricted openings and it increases as the fluid velocity increases. Most noises in piping systems may be attributed to the various types of control valve. The sound pressure level may be calculated for control valves from formulae and data supplied by the various manufacturers.

9.7.3 The fundamental approach to noise control in piping systems should be to avoid or minimize the generation of harmful noise levels. Methods that may be effective in avoiding such levels in piping systems include:

a) minimizing fluid velocities; the noise levels generated by the velocities that should be applied according to clause 5 should be acceptable;

b) selecting control valves of a type or with special trim to minimize noise;

c) using extra-heavy-wall pipe and fittings to attenuate sound and vibration (see API Medical Research Report EA 7301 for further information);

d) using acoustic insulation and/or shielding around pipe and fittings to absorb or isolate sound;

e) using flow-stream silencers for extreme cases.

9.8 Pipe, valves and fittings tables

Detailed information concerning pipe, valves and fittings may be readily shown using tables. Information that may be shown in the tables includes size ranges, general specifications, valve selection, pressure ratings, temperature limitations, pipe schedules, material specifications, special notes etc. Such tables are generally prepared for use as a company guide. An example of a pipe, valves and fittings table is provided in annex B.

9.9 Inspection, maintenance, repair and modification

Inspection, maintenance, repairs and alterations are important considerations after a platform piping system has been commissioned. API RP 510 [52], although written for pressure vessel applications, contains guidelines on these topics that can also be applied to piping systems.

10 Installation and quality control

10.1 General

Fabrication, assembly, erection, inspection, testing and related operations associated with the installation and quality control of metallic and non-metallic platform piping systems should be in accordance with ASME B31.3, except as modified herein.

10.2 Welding

10.2.1 Safety precautions

Welding on offshore platforms shall be subject to a work permit system and shall not be permitted in hazardous areas unless made safe for welding operations.

10.2.2 Welding procedure qualification

Tests shall be conducted to qualify the welding procedures to be utilized. Procedure qualification methods and requirements shall be in accordance with ASME B31.3.

10.2.3 Welder qualification

Welders should be qualified using the specific qualified welding procedure for the proposed job. These qualifications shall be in accordance with ASME B31.3.

10.2.4 Welding records

Test records of the welding procedure qualification and the welder performance qualification shall be maintained and shall be in accordance with ASME B31.3.

10.2.5 Welding requirements

Welding requirements shall be in accordance with ASME B31.3. In addition, welding should not be done if there is danger that the quality of the weld may be affected by atmospheric conditions.

10.2.6 Heat treatment

Heat treatment, including pre-heat and PWHT, shall be in accordance with ASME B31.3.

10.2.7 Non-destructive examination

Visual, radiographic and ultrasonic examination should be in accordance with ASME B31.3. Additionally, platform piping in hydrocarbon service, regardless of service temperature and pressure, should be radiographically examined in accordance with Table 10.

Table 10 — Minimum extent of radiographic weld examination for carbon steel piping

Pressure rating	Percent of welds
ASME class 150 to 600	10 %
ASME class 900 and 1500	20 %
ASME class 2500	100 %
5 000 psi and higher	100 %

10.3 Pressure testing

10.3.1 Safety considerations

Prior to being placed in operation, piping shall be pressure tested for structural integrity and leaks. Preparation for, and performance of, hydrostatic and pneumatic testing shall be in accordance with ASME B31.3, except as modified below.

The temperature of the material under test shall be safely above that at which brittle fracture may be initiated. When testing with air or nitrogen, rapid depressurization shall be avoided in order to prevent excessive auto-refrigeration lowering metal temperatures to a level where brittle fracture is possible.

10.3.2 Hydrostatic tests

Water should be used in all hydrostatic testing unless it would have an adverse effect on the piping or operating fluid. Any flammable liquids used for testing should have a flash point above 65 °C.

Equipment and instruments unable to withstand the test pressure shall be removed or isolated. Typically:

a) pumps, turbines and compressors;

b) rupture discs and relief valves;

c) rotameters and displacement meters;

d) indicating pressure gauges, if the test pressure exceeds the scale range;

e) instrumentation and sight-glasses.

The following equipment should be tested to design pressure and then isolated:

a) indicating pressure gauges, if the test pressure exceeds the scale range;

b) external-float-type level shutdown devices and controllers, if the float is not rated for the test pressure. The float should be subjected to design pressure and then the float chamber should be isolated from the system;

c) check valves should be held open and other valves should be in the half-open position during testing.

10.3.3 Pneumatic tests

Pneumatic tests may be performed in accordance with ASME B31.3 if hydrostatic tests are undesirable, such as in instrument air, heating fluid and refrigeration systems. Because a pneumatic test may create an unsafe condition, special precautions should be taken and careful supervision shall be provided during testing. Only air or nitrogen (with or without tracers) should be used as the test fluid. Each test system should be kept as small as practical.

a) The test pressure should be 1,1 times the maximum design pressure. Testing using pressures above 700 kPa should be avoided. To guard against brittle fracture hazards, the minimum metal temperature for all components during the test should be 10 °C.

b) Pressure should be gradually increased to not more than 170 kPa, and held until all joints have been inspected for leaks with soap solution. If no leaks are found, the pressure should be increased in increments of approximately 200 kPa until the final test pressure is reached. The pressure should then be reduced to 90 % of test pressure and held for a sufficient length of time to permit inspection of all joints, welds and connections with soap solution.

10.3.4 Leak tests

When required, leak testing is performed to confirm the pressure-tightness of mechanical joints and seals in piping systems that are mechanically complete following assembly of piping spools and components that have previously been hydrotested, or upon reinstatement after maintenance activities.

Water is normally used for leak testing unless it would have a detrimental effect on the piping or operating process. Where required, leak testing may be performed using nitrogen, with or without a tracer gas such as helium.

The test pressure should not exceed 110 % of the maximum allowable working pressure.

10.4 Test record

Test records including welding procedure qualifications, welder performance qualifications and pressure testing shall be in accordance with ASME B31.3.

Annex A
(informative)

Example problems

A.1 Introduction

This annex demonstrates, by means of solutions to example problems, applications of the piping design guidelines presented in this International Standard.

A.2 Flowline piping design example

A.2.1 Problem

A.2.1.1 Design a flowline for a gas-condensate completion.

A.2.1.2 The completion is expected to have the following initial characteristics:

a) shut-in wellhead pressure = 37 900 kPa (ga);

b) maximum test and production flow rate expected (including surge):

q_g = 16 740 m^3/h at normal pressure [d_g = 0,65 (air = 1)];

R = 3 370 m^3/m^3 of gas condensate at normal conditions [d_L = 0,80 (water = 1)];

c) flowing tubing pressure = 31 000 kPa (ga);

d) flowing temperature = 49 °C.

A.2.1.3 The completion is expected to have the following characteristics at depletion:

a) flowing tubing pressure = 10 300 kPa (ga);

b) q_g = 11 160 m^3/h at normal conditions;

c) R = 8 425 m^3/m^3 of gas condensate at normal conditions [d_L = 0,80 (water = 1)];

d) 9,94 m^3/h of produced water [d_L = 1,08 (water = 1)].

A.2.1.4 Equivalent length of flowline = 15 m.

A.2.1.5 Flowline to be designed for wellhead pressure.

A.2.2 Solution

A.2.2.1 General

The following items should be considered:

a) erosional velocity;

b) pressure containment;

c) noise;

d) pressure drop.

A.2.2.2 Erosional velocity

A.2.2.2.1 Since the well will flow continuously and little or no sand production is anticipated, equation (A.1), with an empirical constant of 122, is used to calculate the maximum allowable erosional velocity (see 5.5.1).

$$v_\mathrm{e} = \frac{c}{\sqrt{\rho_\mathrm{m}}} \tag{A.1}$$

where

v_e is the fluid erosional velocity, expressed in metres per second (m/s);

c is the empirical constant:

c = 122 for continuous service with minimum solids;

ρ_m is the gas/liquid density at operating pressure and temperature, expressed in kilograms per cubic metre (kg/m^3).

A.2.2.2.2 ρ_m will be calculated for initial and final flowing conditions, to determine which conditions control, using equation (A.2).

$$\rho_\mathrm{m} = \frac{28\,833\, d_\mathrm{L} \cdot p + 37{,}22\, R \cdot d_\mathrm{g} \cdot p}{28{,}82\, p + 10{,}68\, R \cdot T \cdot Z} \tag{A.2}$$

where

p is the operating pressure, expressed in kilopascals [kPa (abs)];

d_L is the liquid relative density (water = 1; use average value for hydrocarbon-water mixtures) at standard conditions, dimensionless;

R is the gas/liquid ratio at normal conditions;

T is the operating temperature, expressed in kelvin (K);

d_g is the gas relative density (air = 1), dimensionless;

Z is the gas compressibility factor, dimensionless.

a) For initial conditions:

d_L = 0,80

p = 31 000 kPa (ga) + 100 = 31 100 kPa (abs)

R = 3 370 m^3/m^3 of gas condensate at normal conditions

d_g = 0,65

T = 49 + 273 = 322 K

Z = 0,91

Inserting in equation (A.2):

$$\rho_{\mathrm{m}} = \frac{(28\ 833 \times 0{,}8 \times 31\ 100) + (37{,}22 \times 3\ 370 \times 0{,}65 \times 31\ 100)}{(28{,}82 \times 31\ 100) + (10{,}68 \times 3\ 370 \times 322 \times 0{,}91)} = 285\ \mathrm{kg/m^3}\ \text{(initial)}$$

b) For final conditions:

$$d_{\mathrm{L}} = \frac{(11\ 160/8\ 425) \times 0{,}8 + (9{,}94 \times 1{,}08)}{(11\ 160/8\ 425) + 9{,}94} = 1{,}047$$

p = 10 400 kPa (abs)

$$R = \frac{11\ 160}{11\ 160/8\ 425 + 9{,}94} = 991\ \mathrm{m^3/m^3}$$

d_{g} = 0,65

T = 322 K

Z = 0,81

Inserting in equation (A.2):

$$\rho_{\mathrm{m}} = \frac{(28\ 833 \times 1{,}04 \times 10\ 400) + (37{,}22 \times 991 \times 0{,}65 \times 10\ 400)}{(28{,}82 \times 10\ 400) + (10{,}68 \times 991 \times 322 \times 0{,}81)} = 184\ \mathrm{kg/m^3}\ \text{(final)}$$

A.2.2.2.3 Using the above values in equation (A.1) gives:

$$v_{\mathrm{e}}\ (\text{initial}) = \frac{122}{\sqrt{285}} = 7{,}2\ \mathrm{m/s}$$

$$v_{\mathrm{e}}\ (\text{final}) = \frac{122}{\sqrt{184}} = 9{,}0\ \mathrm{m/s}$$

A.2.2.2.4 Using equation (A.3), the minimum allowable cross-sectional areas can be determined:

$$A = \frac{277{,}6 + (103\ Z \cdot R \cdot T)/p}{v_{\mathrm{e}}} \tag{A.3}$$

where A is the minimum pipe cross-sectional flow area required per unit volume flowrate, expressed in square millimetres per cubic metre per hour ($\mathrm{mm^2/m^3/h}$).

a) For initial conditions:

$$A = \frac{277{,}6 + (103 \times 3\ 370 \times 322 \times 0{,}91)/31\ 100}{7{,}2} = 492{,}8\ \mathrm{mm^2/m^3/h}$$

$$A = 492{,}8 \times \left(\frac{16\ 740}{3\ 370}\right) = 2\ 448\ \mathrm{mm^2}\ \text{(initial)}$$

b) For final conditions:

$$A = \frac{277{,}6 + (103 \times 991 \times 322 \times 0{,}81)/10\,400}{9{,}0} = 311\ \text{mm}^2/\text{m}^3/\text{h}$$

$$A = 311 \times \left(9{,}94 + \frac{11\,160}{8\,425}\right) = 3\,503\ \text{mm}^2\ (\text{final})$$

A.2.2.2.5 Although the allowable erosional velocity is higher for the final conditions, the line size will still be controlled by the final conditions since the liquid volume is higher.

A.2.2.2.6 A is converted into an inside pipe diameter as follows:

$$A = \frac{\pi \cdot {d_\text{i}}^2}{4} \tag{A.4}$$

$$d_\text{i}\ (\text{initial}) = \sqrt{\frac{4 \times 2\,448}{\pi}} = 55{,}8\ \text{mm inside diameter}$$

$$d_\text{i} = \sqrt{\frac{4 \times 3\,503}{\pi}} = 67{,}1\ \text{mm (final)}$$

A.2.2.3 Pressure containment

A.2.2.3.1 A preliminary line size can now be selected using Table 5. The required pressure rating for the line shall be greater than 37 900 kPa (ga). The two obvious choices are listed below:

Nominal pipe size	Schedule	Inside diameter	API 5L grade B pipe, max. working pressure
DN 80 (3 NPS)	XXS	58,4 mm	42 000 kPa (ga)
DN 100 (4 NPS)	XXS	80,1 mm	36 600 kPa (ga)

The DN 80 (3 NPS) has the required pressure rating, but the internal diameter is too small for the final conditions. The DN 100 (4 NPS) has sufficient internal diameter for the final conditions, but the required working pressure is slightly deficient if grade B pipe is used.

A.2.2.3.2 The final selection requires engineering judgement. The following alternatives should be evaluated.

a) Recheck the source of the shut-in wellhead pressure requirements. Often, nominal values somewhat higher than actual requirements are supplied to piping designers. If the actual shut-in pressure was less than 36 600 kPa (ga), the DN 100 (4 NPS) XXS would be the proper choice.

b) Check availability of DN 100 (4 NPS) XXS grade X42 pipe which will meet the required 37 900 kPa (ga) pressure requirement.

c) Consider using a pressure relief device on a DN 100 (4 NPS) XXS line until the wellhead shut-in pressure declines to 36 600 kPa (ga).

d) Recheck the final well conditions prediction to determine if slightly lower flows might exist which would allow a DN 80 (3 NPS) XXS line to suffice for the life of the well. The amount of surge included in the maximum flow rate should be examined in detail to see if it is reasonable.

e) Consider installing a DN 80 (3 NPS) XXS line initially, then replacing it with a larger (inside diameter) line later in the life.

f) Exceptionally, consideration may be given to exceeding the design pressure on an infrequent basis, in accordance with ASME B31.3.

A.2.2.4 Noise

A.2.2.4.1 The velocity (and relative indication of the noise) in the line will be determined for initial and final conditions to determine which conditions control.

The velocity can be determined as follows.

a) For initial conditions:

Mass flow of gas:

$$= \frac{16\ 740 \times 0{,}65 \times 29}{3\ 600 \times 22{,}414} = 3{,}9 \text{ kg/s}$$

Mass flow of condensate:

$$= \frac{16\ 740 \times 2{,}97 \times 10^{-4} \times 0{,}8 \times 998{,}5}{3\ 600} = 1{,}1 \text{ kg/s}$$

Total mass flow of wellstream (initial) = 5,0 kg/s

b) For final conditions:

Mass flow of gas:

$$= \frac{11\ 160 \times 0{,}65 \times 29}{3\ 600 \times 22{,}414} = 2{,}61 \text{ kg/s}$$

Mass flow of condensate:

$$= \frac{11\ 160 \times 1{,}187 \times 10^{-4} \times 0{,}8 \times 998{,}55}{3\ 600} = 0{,}294 \text{ kg/s}$$

Mass flow of water:

$$= \frac{9{,}94 \times 998{,}55}{3\ 600} = 0{,}276 \text{ kg/s}$$

Total mass flow of wellstream (final) = 5,66 kg/s

A.2.2.4.2 The flowing velocity will be determined as above for erosional velocity, the total volume flows would be:

$$\text{final flow volume} = \frac{5{,}66}{184{,}06} = 0{,}030\ 3 \text{ m}^3\text{/s}$$

$$\text{initial flow volume} = \frac{5{,}0}{284{,}52} = 0{,}020\ 2 \text{ m}^3\text{/s}$$

Thus the final conditions control the maximum velocity.

A.2.2.4.3 The flowing velocity will be determined for DN 80 (3 NPS) and DN 100 (4 NPS) nominal XXS line sizes.

$$\text{Velocity [DN 80 (3 NPS)]} = \frac{0{,}030\,3}{(p/4) \times (58{,}42/1\,000)^2} = 11{,}3 \text{ m/s}$$

$$\text{Velocity [DN 100 (4 NPS)]} = \frac{0{,}030\,3}{(p/4) \times (80{,}01/1\,000)^2} = 6{,}03 \text{ m/s}$$

A.2.2.4.4 Since the velocity is less than 25 m/s (see 5.4) in both cases, noise should not be a problem and would not influence the line size selection.

A.2.2.5 Pressure drop

A.2.2.5.1 Pressure drop in the line may be determined using equation (A.5).

$$\Delta p = \frac{6\,253\,000\; q_\mathrm{m} \cdot f}{{d_\mathrm{i}}^5 \cdot \rho_\mathrm{m}} \qquad \text{(A.5)}$$

where

Δp is the pressure drop per 100 m of pipe, expressed in kilopascals (kPa);

d_i is the pipe inside diameter, expressed in millimetres (mm);

f is the Moody friction factor, dimensionless;

ρ_m is the gas/liquid density at flowing pressure and temperature, expressed in kilograms per cubic metre (kg/m^3), calculated as shown in equation (A.2);

q_m is the total liquid plus vapour rate, expressed in kilograms per hour (kg/h).

A.2.2.5.2 q_m may be determined using equation (A.6).

$$q_\mathrm{m} = 1{,}29\; q_\mathrm{g} \cdot d_\mathrm{g} + 1\,000\; q_\mathrm{L} \cdot d_\mathrm{L} \qquad \text{(A.6)}$$

where

q_g is the gas flow rate, expressed in cubic metres per hour (m^3/h) at normal conditions;

d_g is the gas relative density (air = 1), dimensionless;

q_L is the liquid flow rate, expressed in cubic metres per hour (m^3/h);

d_L is the liquid relative density (water = 1), dimensionless.

q_m (initial):

$$= 1{,}29 \times 16\,740 \times 0{,}65 + 1\,000 \times \left(\frac{16\,740}{3\,370} \times 0{,}8 \right) = 18\,035 \text{ kg/h}$$

q_m (final):

$$= 1{,}29 \times 11\,160 \times 0{,}65 + 1\,000 \times \left(\frac{11\,160}{8\,425} + 9{,}94 \right) = 21\,165 \text{ kg/h}$$

A.2.2.5.3 Using the above values in equation (A.6) gives:

$$\Delta p\ [\text{initial DN 80 (3 NPS)}] = \frac{6{,}253 \times 10^{6} \times 0{,}019 \times (18\ 035)^{2}}{(58{,}42)^{5} \times 284{,}52} = 200\ \text{kPa}/100\ \text{m}$$

$$\Delta p\ [\text{final DN 100 (4 NPS)}] = \frac{6\ 253\ 000 \times 0{,}019\ 6 \times (18\ 035)^{2}}{(80{,}01)^{5} \times 284{,}52} = 43\ \text{kPa}/100\ \text{m}$$

$$\Delta p\ [\text{final DN 80 (3 NPS)}] = \frac{6\ 253\ 000 \times 0{,}020 \times (21\ 165)^{2}}{(58{,}42)^{5} \times 184{,}06} = 448\ \text{kPa}/100\ \text{m}$$

$$\Delta p\ [\text{final DN 100 (4 NPS)}] = \frac{6\ 253\ 000 \times 0{,}019\ 6 \times (21\ 165)^{2}}{(80{,}01)^{5} \times 184{,}06} = 91\ \text{kPa}/100\ \text{m}$$

Refer to Figure 3 for Moody friction factor (f).

A.2.2.5.4 Since the line is only 15 m long, the total pressure drop would not be critical in most cases and probably would not influence line size selection.

A.3 Pump suction piping design example

A.3.1 Problem

A.3.1.1 A single reciprocating pump will be used to transfer crude oil from a production separator to a remote oil treating facility. Select a suction line size for this pump application. Data for the pumping system is listed below.

A.3.1.2 Operating conditions of separator:

a) operating pressure = 415 kPa (ga);

b) inlet oil volume = 33,1 m^3/h;

c) inlet water volume = zero;

d) oil gravity = 40° API (d_L = 0,825);

e) oil viscosity, μ = 1,5 × 10^{-3} Pa·s at pumping temperature;

f) level in separator will be controlled constant by bypassing pump discharge back to separator;

g) vessel outlet nozzle = DN 200 (NPS 8) ASME class 150.

A.3.1.3 Pump data:

a) volume handled, 150 % of oil inlet (to ensure ability to maintain liquid level during surges) = 49,7 m^3/h;

b) type pump = Triplex;

c) pump speed = 200 r/min;

d) suction connection = DN 150 (NPS 6) ASME class 150;

e) discharge connection = DN 80 (NPS 3) ASME class 600;

f) required NPSH = 27,6 kPa (abs) at operating condition;

g) discharge pressure = 3 450 kPa (ga);

h) pump datum located 4,57 m below the separator fluid level.

A.3.1.4 The suction line is 15,2 m long and contains one tee, four 90° elbows, two full-open ball valves and one DN 200 (NPS 8) × DN 150 (NPS 6) standard reducer.

A.3.2 Trial solution

A.3.2.1 From Table 4, a suction velocity of 0,6 m/s is selected to determine a preliminary line size. From Figure 2, for 49,7 m^3/h flow rate a DN 150 (NPS 6) Sch 40 line gives a velocity of 0,75 m/s; and a DN 200 (NPS 8) Sch 40 line gives a velocity of 0,43 m/s. From this, DN 200 (NPS 8) line (203 mm inside diameter) is selected for the first trial.

A.3.2.2 Determine line equivalent length using Table 3.

Elbow equivalent length	= 4 × 2,74 m	= 11,0 m
Ball valves equivalent length	= 2 × 1,83 m	= 3,66 m
Tee run equivalent length	= 1 × 2,74 m	= 2,74 m
Reducer equivalent length	= 1 × 0,610 m	= 0,610 m
Vessel outlet contraction	= 1 × 3,66 m	= 3,66 m
DN 200 (8 NPS) line length	= 15,2 m	
Equivalent line length	= 36,9 m	

A.3.2.3 Next calculate line friction losses using equation (A.7):

$$\Delta p = \frac{6\ 270 \times 10^6\ f \cdot q_L{}^2 \cdot d_L}{d_i{}^5} \tag{A.7}$$

where

Δp is the pressure drop per 100 m of pipe, expressed in kilopascals (kPa);

f is the Moody friction factor, dimensionless;

q_L is the liquid flow rate, expressed in cubic metres per hour (m^3/h);

d_L is the liquid relative density (water = 1), dimensionless;

d_i is the pipe inside diameter, expressed in millimetres (mm).

A.3.2.4 The friction factor, f, may be found from Figure 3 using the Reynolds number calculated from equation (A.8) as follows:

$$Re = \frac{\rho_L \cdot D_i \cdot v_L}{\mu_1} \tag{A.8}$$

where

Re is the Reynolds number, dimensionless;

ρ_L is the liquid density, expressed in kilograms per cubic metre (kg/m^3);

D_i is the pipe inside diameter, expressed in metres (m);

v_L is the liquid flow velocity, expressed in metres per second (m/s);

μ_L is the liquid viscosity, expressed in pascal seconds (Pa·s), or

= centipoises divided by 1 000, or

= (centistokes times relative density) divided by 1 000.

For this problem:

ρ_L = 1 000 × 0,825 = 825 kg/m^3

D_i = 0,203 m

v_L = 0,43 m/s (Figure 2)

μ_L = 1,5/1 000 = 0,001 5 Pa·s

$$Re = \frac{825 \times 0{,}203 \times 0{,}43}{0{,}0015} = 48\ 010$$

Using Re and the steel pipe curve, f may be found from Figure 3.

f = 0,023

A.3.2.5 All the required values for equation (A.7) are now known:

q_L = 49,7 m^3/h

d_L = 0,825

d_i = 203 mm

$$\Delta p / 100 \text{ m} = \frac{6\ 270 \times 10^6 \times 0{,}023 \times 49{,}7^2 \times 0{,}825}{203^5} \text{ kPa}/100 \text{ m} = 0{,}852 \text{ kPa}/100 \text{ m}$$

$$\Delta p \text{ total} = \frac{0{,}852 \times 36{,}9}{100} = 0{,}314 \text{ kPa}$$

A.3.2.6 Next determine available NPSH from equation (A.9):

$$NPSH_a = h_p - h_{vpa} + h_{st} - h_f - h_{vh} - h_a \qquad \text{(A.9)}$$

where

h_p is the absolute pressure head due to pressure, atmospheric or otherwise, on surface of liquid going to suction, expressed in metres of liquid (m);

h_{vpa} is the absolute vapour pressure of the liquid at suction temperature, expressed in metres of liquid (m);

h_{st} is the static head, positive or negative, due to liquid level above or below datum line (centreline of pump), expressed in metres of liquid (m);

h_f is the friction head, or head loss due to flowing friction in the suction piping, including entrance and exit losses, expressed in metres of liquid (m);

h_{vh} is the velocity head, $v_L^2/2g$, expressed in metres (m) of liquid;

h_a is the acceleration head, expressed in metres of liquid (m);

v_L is the velocity of liquid in piping, expressed in metres per second (m/s);

g is the gravitational constant, expressed in metres per second squared (usually 9,81 m/s^2).

A.3.2.7 Since the oil is in equilibrium with the gas in the separator, the vapour pressure of the oil will be 415 kPa (ga) also. Thus:

$$h_{vpa} = h_p = \frac{(415 + 100) \times 0{,}101\,97}{0{,}825} = 63{,}65 \text{ m}$$

$$h_{st} = 4{,}57 \text{ m (given)}$$

$$h_r = \frac{0{,}314 \times 0{,}101\,97}{0{,}825} = 0{,}039 \text{ m}$$

$$h_{vh} = \frac{(0{,}43)^2}{2 \times 9{,}81} = 0{,}009 \text{ m}$$

A.3.2.8 h_a may be determined from equation (A.10):

$$h_a = \frac{L \cdot v_L \cdot R_p \cdot C}{K \cdot g} \qquad \text{(A.10)}$$

where

h_a is the acceleration head, expressed in metres of liquid (m);

L is the length of suction line, expressed in metres (m) (actual length, not equivalent length);

v_L is the average liquid velocity in suction line, expressed in metres per second (m/s);

R_p is the pump speed, expressed in revolutions per minute (r/min);

C is the empirical constant for the type of pump:

= 0,066 for triplex single- or double-acting;

K is the liquid compressibility factor, representing the reciprocal of the fraction of the theoretical acceleration head which shall be provided to avoid a noticeable disturbance in the suction piping:

= 2,0 for most hydrocarbons;

g is the gravitational constant expressed in metres per second squared (usually 9,81 m/s^2).

Substituting known values into equation (A.10) yields:

$$h_a = \frac{15{,}2 \times 0{,}43 \times 200 \times 0{,}066}{2{,}0 \times 9{,}81} = 4{,}40 \text{ m}$$

A.3.2.9 The available net positive suction head is:

$$\begin{aligned} NPSH_\mathrm{a} &= 63{,}65 - 63{,}65 + 4{,}57 - 0{,}039 - 0{,}009 - 4{,}40 \\ &= 0{,}122\ \mathrm{m} \end{aligned}$$

A.3.2.10 $\text{NPSH required} = \dfrac{27{,}6 \times 10^5}{0{,}825 \times 10 \times 9{,}81} = 3{,}41\ \mathrm{m}$

A.3.2.11 **Conclusion:** The pump would not operate under these conditions.

A.3.3 Alternative solutions to increase $NPSH_\mathrm{a}$ (referring to 5.3.2.6)

A.3.3.1 Shorten suction line

Although it might be possible to shorten the line length somewhat, the acceleration head needs to be reduced by at least:

$$\left[1 - \frac{4{,}40 - 3{,}41 - 0{,}122}{4{,}40}\right] 100\ \% = 80\ \%$$

so this alternative would not be feasible.

A.3.3.2 Use larger suction pipe to reduce velocity

If a DN 250 (NPS 10) pipe were used rather than a DN 200 (NPS 8) pipe, the velocity would be reduced from 0,43 m/s to 0,27 m/s (Figure 2). Likewise a DN 300 (NPS 12) pipe would reduce the velocity to 0,19 m/s. Since neither of the pipe sizes would reduce the velocity by 80 % (and thereby reduce the acceleration head by 80 %), this alternative would not be feasible.

A.3.3.3 Reduce pump speed

A pump speed of 200 r/min is already very low, so this alternative would not be feasible.

A.3.3.4 Consider a pump with a larger number of plungers

The reasonable pump alternative would be to use a quintuplex, rather than a triplex, which would reduce the acceleration head by 40 %. Since a greater percentage reduction is required, this alternative is not feasible.

A.3.3.5 Use a pulsation dampener

A.3.3.5.1 A properly installed pulsation dampener may reduce the length of line used to 15 (or less) nominal pipe diameters in equation (A.10).

$$\left(\frac{15 \times 200}{1000} = 3{,}0\ \mathrm{m}\right)$$

A.3.3.5.2 Recalculate the acceleration head.

$$h_\mathrm{a}\ (\text{dampener}) = \frac{3{,}0 \times 0{,}43 \times 200 \times 0{,}066}{2{,}0 \times 9{,}81} = 0{,}868\ \mathrm{m}$$

A.3.3.5.3 By using a pulsation dampener, the available NPSH would be:

$$NPSH_\mathrm{a} = 63{,}65 - 63{,}65 + 4{,}57 - 0{,}039 - 0{,}009 - 0{,}868 = 3{,}65\ \mathrm{m}$$

A.3.3.5.4 Since the conservative approach was used in determining the line length to recalculate the acceleration head, the available NPSH should be adequate if a pulsation dampener is used.

A.3.3.5.5 If a greater margin of available NPSH over required NPSH is desired, then one of the alternatives discussed above could be included in the system design in addition to the pulsation dampener.

Annex B
(informative)

Examples of pipe, valves and fittings tables

B.1 Example of index

An example of an index for pipe, valves and fittings tables is shown in Table B.1.

Table B.1 — Typical index for pipe, valves and fittings tables

Table	Service	Pressure rating
A	Non-corrosive hydrocarbons and glycol	ASME class 150
B	Non-corrosive hydrocarbons and glycol	ASME class 300
C	Non-corrosive hydrocarbons and glycol	ASME class 400
D	Non-corrosive hydrocarbons and glycol	ASME class 600
E	Non-corrosive hydrocarbons and glycol	ASME class 900
F	Non-corrosive hydrocarbons and glycol	ASME class 1500
G	Non-corrosive hydrocarbons and glycol	ASME class 2500
H	Non-corrosive hydrocarbons	13,8 MPa (2 000 psi)
I	Non-corrosive hydrocarbons	20,7 MPa (3 000 psi)
J	Non-corrosive hydrocarbons	34,5 MPa (5 000 psi)
K	Non-corrosive hydrocarbons	69,0 MPa (10 000 psi)
L	Air	ASME classes 150 and 300
M	Water	ASME class 150
N	Steam and steam condensate	ASME class 150 ASME class 300 ASME class 600
O	Drains and sewers	Atmospheric
P (Spare)		
Q (Spare)		
R (Spare)		
SV	Valves for corrosive service	General
AA	Corrosive hydrocarbons	ASME class 150
BB	Corrosive hydrocarbons	ASME class 300
CC (Not prepared)	Corrosive hydrocarbons	ASME class 400
DD	Corrosive hydrocarbons	ASME class 600
EE	Corrosive hydrocarbons	ASME class 900
FF	Corrosive hydrocarbons	ASME class 1500
GG	Corrosive hydrocarbons	ASME class 2500

B.2 Specific example

An example of an ASME class 150 pipe, valves and fittings table is shown in Table B.2.

Table B.2 — Typical pipe, valves and fittings table for a specific application

Service:	Non-corrosive	
Material:	Carbon steel 1,25 mm corrosion allowance	
Temperature range:	0 °C to 200 °C	
Maximum pressure:	As per ASME B16.5 material group 1.1	
Size range (NPS)	**Description**	**Notes**
Pipe:	ASTM A 106 grade B	
½ - 1½	Schedule 80, seamless, plain ends	
2 - 3	Schedule 80, seamless, bevelled ends	
4 - 6	Schedule 40, seamless, bevelled ends	
8 - 18	Schedule 30, seamless, bevelled ends	
Fittings:		
½ - 1½	ASTM A 105, normalized	
Socket weld	3 000 psi pressure rating, forged steel	
2 - 18	ASTM A 234 grade WPB	
Butt weld	Seamless to match pipe	
‘O’ Lets		
½ - 1½	ASTM A 105 normalized	
Branch nipple	3 000 psi pressure rating, forged steel	
2 - 8	ASTM A 105, normalized	
Reinforced branch outlet to match pipe		
Flanges:		
½ - 1½	ASTM A 105, normalized	
Socket weld	ASME class 150, raised face	
2 - 18	ASTM A 105, normalized	
Weld neck	ASME class 150, raised-face, bored to suit pipe schedule	
Bolting:		
Studs	ASTM A 193 grade B7	
Nuts	ASTM A 194 grade 2H	
Gaskets:	Compressed non-asbestos fibre 1,5 mm thick	
½ - 18	ASME class 150, flat ring to ASME B16.21	
½ - 18	Carbon steel body, 13 % chrome trim, bolted bonnet	NOTE 2

Table B.2 *(continued)*

Service:	Non-corrosive	
Material:	Carbon steel 1,25 mm corrosion allowance	
Temperature range:	0 °C to 200 °C	
Maximum pressure:	As per ASME B16.5 material group 1.1	
Size range (NPS)	**Description**	**Notes**
Valves:	API 6D / ASME B16.34	
Gate	ASME class 150, flanged, raised-face	
Ball		
½ - 10	ASME class 150, flanged, raised-face, reduced-bore, seat-supported ball. Bolted carbon steel body, AISI 316 ball and stem, PTFE seats, firesafe	NOTE 1
12 - 18	Reduced bore, trunnion-mounted, Bolted carbon steel body, AISI 4130. Ball and stem electroless-nickel-plated. PTFE seats firesafe	NOTE 1
Globe	ASME class 150, flanged, raised face	
½ - 6	Carbon steel body, 13 % chrome trim, plug or ball type disc, bolted bonnet	
Check	ASME class 150, flanged, raised face	
½ - 1½	Horizontal piston-lift-type, carbon steel body, 13 % chrome trim, bolted cover	
2 - 18	ASME class 150, dual-plated wafer type	
	Carbon steel body, 13 % chrome discs and seats, nickel alloy springs	
NOTE 1 Maximum service temperature of soft-seated ball valves is 120 °C.		
NOTE 2 Ball valves should be used; only use gate valves where required by service/temperature.		

The valve manufacturers and valve figure numbers have not been shown in example B.2. One or more valve manufacturers and figure numbers should be included in the tables.

Valve equivalency tables are available from several manufacturers. Using these tables, with one manufacturer's figure number as a base, it is possible to determine the different valve manufacturer's equivalent figure numbers. By having an equivalent figure number, a valve can be quickly located in a manufacturer's catalogue. This procedure allows an operator to compare manufacturing details and materials of alternate valves.

Annex C
(informative)

Acceptable butt-welded joint design for unequal wall thicknesses

C.1 Figure C.1 illustrates acceptable preparations for butt-welding pipe ends that have different wall thicknesses (whether or not the materials have different SMYS).

C.2 The wall thicknesses of the pipes to be joined, each side of the joint design area, should be in compliance with the design requirements of ASME B31.3.

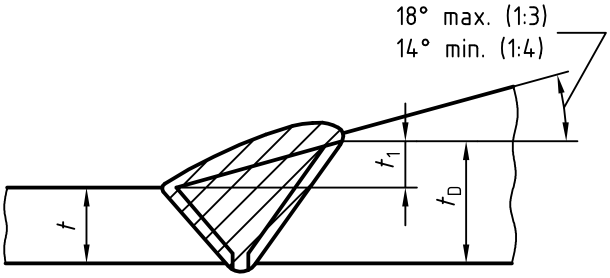
C.3 If the SMYS of the pipes to be joined are unequal, the deposited weld metal should have mechanical properties at least equal to those of the pipe having the higher strength.

C.4 The transition between ends of unequal thickness may be accomplished by tapering or welding as illustrated in Figure C.1 a), or by means of a prefabricated transition piece not less than one-half pipe diameter in length, or 60 mm, whichever is the greater.

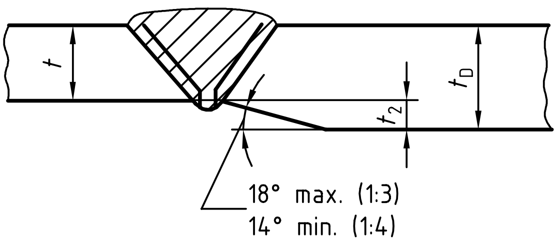
C.5 Sharp notches or grooves at the edge of the weld where it joins a slanted surface should be avoided.

C.6 This annex is also applicable for joining pipes of unequal wall thicknesses and equal SMYS, except that there is no minimum limit to the taper angle.

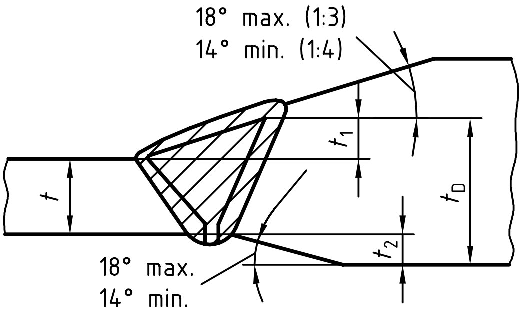
C.7 For the welding of steels of dissimilar chemical composition, specialist advice should be obtained.



a) Bevel for outside thickness



b) Bevel for inside thickness



c) Bevel for combined thickness

NOTE 1 If the materials joined have equal SMYS, there is no restriction on the minimum slope.

NOTE 2 Neither t_1, t_2, nor their sum ($t_1 + t_2$), should exceed 0,5 t.

NOTE 3 If the SMYS of the sections to be joined are unequal, the value of t_D should at least equal t times the ratio of SMYS of the pipe to SMYS of the flange.

NOTE 4 Welding should be in accordance with the applicable code.

Figure C.1 — Acceptable butt-welded joint design for unequal wall thickness and/or additional thickness for welding to higher strength pipe

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