CHAPTER C5
OIL PIPELINE SYSTEMS

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INTRODUCTION

This chapter on oil pipeline systems has been prepared as a basic guide to the design of cross-country pipelines for liquid petroleum and related products. It focuses on the fundamentals of pipeline design, emphasizing practical guidelines for real systems. It provides a general overview of the system approach to design, which integrates the hydraulic, mechanical, and operations and maintenance aspects in the design of a system, along with project economic analysis, in determining the preferred pipeline system.

This chapter also includes discussion of design topics for related pipeline-system components such as pump station and location and sizing, material selection for pipe, metering, leak detection, and system control. Aspects of petroleum-system design related to the special characteristics of some petroleum commodities are also addressed, in particular topics related to high-vapor pressure systems, multiproduct systems, and systems requiring consideration of variable thermal properties of the fluid, i.e., hot oil systems. Finally, design considerations for seismic and underwater design of pipeline systems are outlined.

Scope

There are three basic codes developed by the American Society of Mechanical Engineers (ASME) which govern the design of piping systems in chemical, petroleum liquid, and gas usage.

Piping inside the boundaries of a chemical plant, refinery, or gas processing plant falls under the scope of ASME B31.3 and is covered separately in this handbook (see Chap. C7). Likewise, ASME B31.8, covering gas transmission and distribution piping systems, is specifically addressed in Chap. C6 of this handbook.

This chapter specifically addresses oil transportation systems as covered by the ASME B31.4 Code—*Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids*.[1] In this chapter, references to specific sections of the ASME B31.4 Code will be cited as ''Sec. of the Code,'' or in some cases ''the Code.''

Figures C5.1a and C5.1b are schematic diagrams illustrating the scope of ASME B31.4 for liquid petroleum piping systems other than carbon dioxide and carbon dioxide respectively.

Code Compliance

The ASME Codes set forth the practices required for design and operation of safe pipeline systems. Section 400.1.1 of the Code states its scope, in part, as follows:

> This code prescribes requirements for the design, materials, construction, assembly, inspection, and testing of piping transporting liquids such as crude oil, condensate, natural gasoline, natural gas liquids, liquified petroleum gas, carbon dioxide, liquid alcohol, liquid anhydrous ammonia, and liquid petroleum products between producers' lease facilities, tank farms, natural gas processing plants, refineries, stations, ammonia plants, terminals (marine, rail, and truck), and other delivery and receiving points.

While the Code gives guidelines for pipeline system design, it is not intended to provide complete specifications for all phases of design and operation. Furthermore, there may be additional federal (or country), state, and local regulations governing pipeline design, construction, and operation.

It is the intent of this chapter to supplement the Code with discussion of the principles governing the design of a petroleum transportation system, identifying the analytical and design tools and procedures an engineer might use. This chapter in itself may not address all the design problems which will arise in the real world, and there are some cases where even the best design guide or reference cannot replace experience and judgement. Specific engineering and operating companies may also have guidelines which they require an engineer to follow in the course of designing a pipeline transportation system.

The responsibility rests with the engineer or designer to identify the specific requirements and applicable codes for a given system with regard to design and operational conditions, as well as to follow specific guidelines mandated by the engineering or operating company and any relevant additions to the Code, and other regulations which may govern the design.

Codes, Standards, Specifications, and Recommended Practices

In general, pipelines which are designed in accordance with the Code will meet the requirements in the USA for liquid petroleum pipelines and associated facilities.

In addition to the ASME B31.4 Code, the following codes, specifications, standards, regulations, and recommended practices may be applicable to a proposed pipeline system or component thereof (this list is representative but is not comprehensive):

Design

U.S. Code of Federal Regulations, Title 49, Part 195—Transportation of Hazardous Liquids by Pipeline (known as 49 CFR 195)

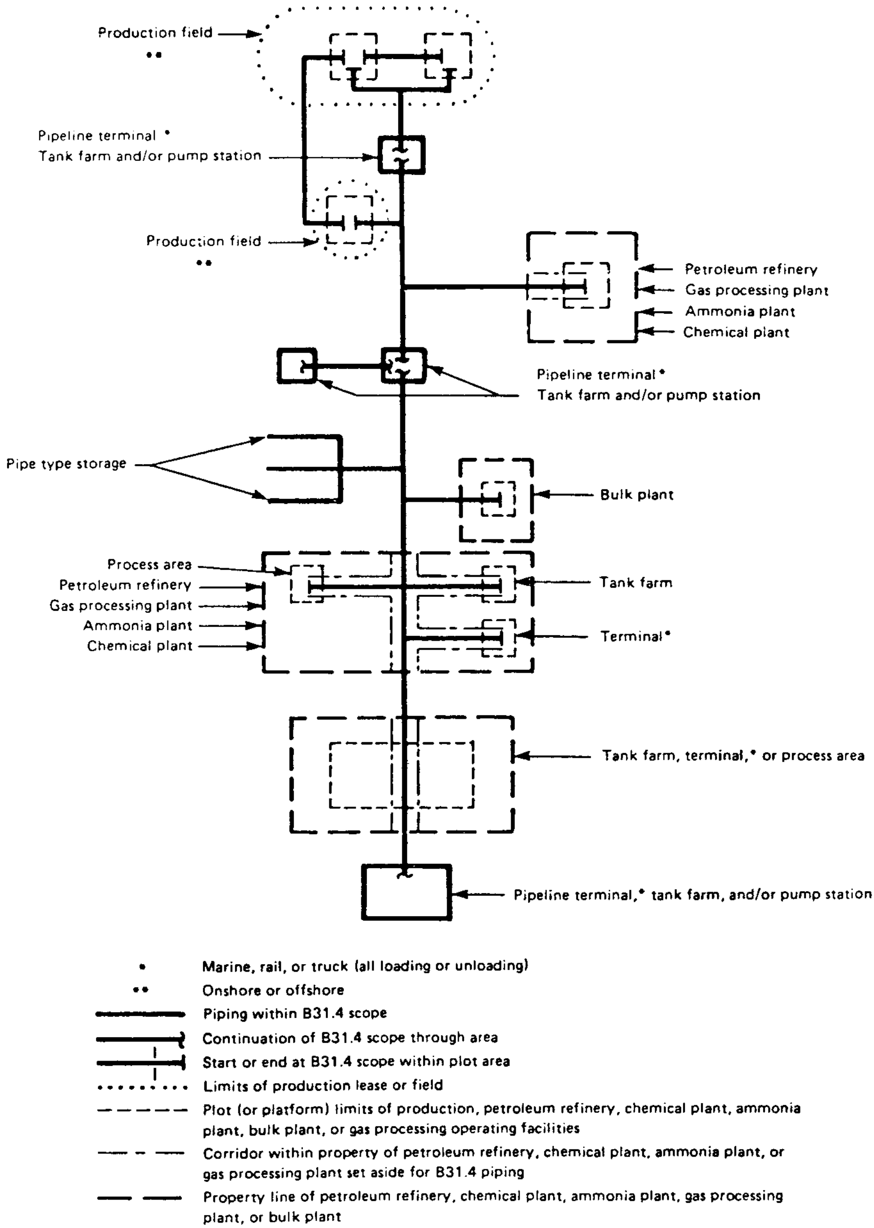


FIGURE C5.1a Scope of ASME B31.4, excluding carbon dioxide pipeline systems.

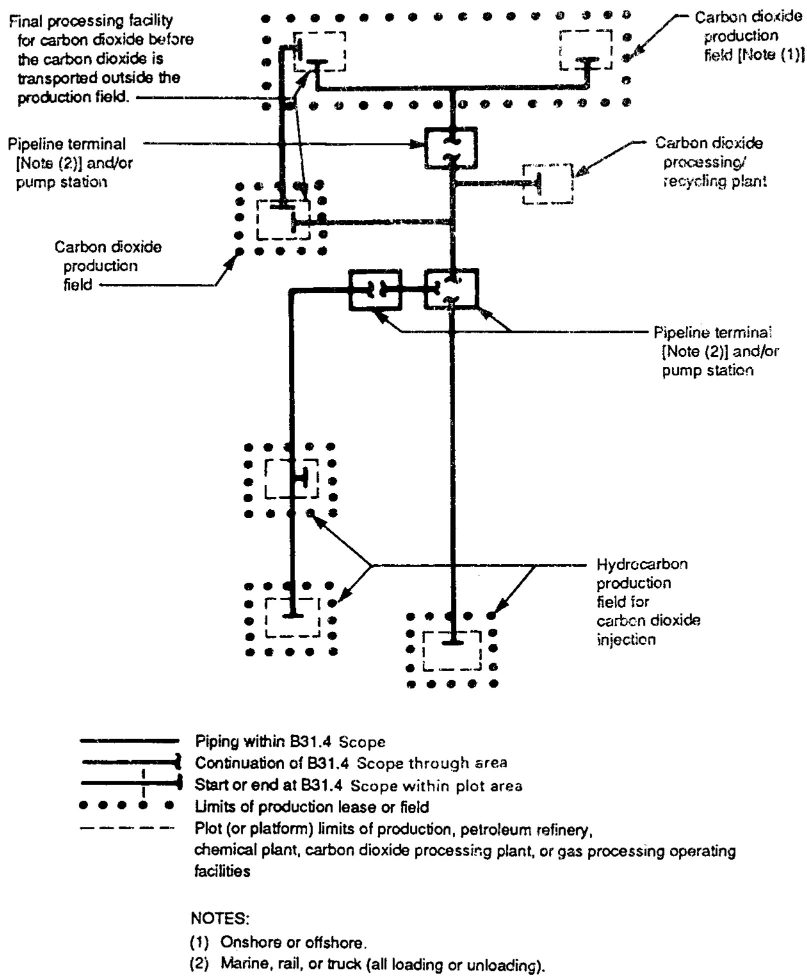


FIGURE C5.1b Scope of ASME B31.4 for carbon dioxide pipeline systems.

American Petroleum Institute (API) RP 1102—Recommended Practice for Liquid Petroleum Pipelines Crossing Railroads and Highways

American Petroleum Institute (API) RP 1111—Recommended Practice for Design, Construction, Operation and Maintenance of Offshore Hydrocarbon Pipelines

Australian Standard–1997: AS 2885.1 Pipelines—Gas and Liquid Petroleum—Part 1: Design and Construction

Australian Standard–AS 1978–1987: Pipelines—Gas and Liquid Petroleum—Field Pressure Testing

BS 8010: Part 1: 1989—Pipelines, Part 1. Pipelines on land: general Canadian Standard: CAN/CSA-Z662–96 Oil and Gas Pipeline Systems

API STD 1104, 1994: Welding of Pipelines and Related Facilities ASME Section IX—Welding and Brazing Qualifications

Material

API 5L—Specifications for Line Pipe[2]

API 6D—Pipeline Valves (Gate, Plug, Ball, and Check Valves)[3]

ASME B16.5—Pipe Flanges and Flanged Fittings[4]

ASME B16.34—Valves—flanged, Threaded, and Welding End[5]

ISO 3183−1, −2, and −3, 1996: Petroleum and natural gas industries—Steel pipe for pipelines—Technical delivery conditions

These codes, specifications, et cetera, also cross reference additional standards and recommended practices that apply to various aspects of petroleum pipeline system components.

Systems of Units. The metric units used herein are adopted from reference codes ASME B31.4a (1994); ASTM E380–93 Standard Practice for International System of Units (SI); ANSI/ASTM D1250 Manual of Petroleum Measurement Standards (MPMS)—Chap. 11.1, Volume Correction Factors[6]; ANSI/ASTM D1298 API Manual of Petroleum measurement Standards; API 5L Specifications for Line Pipe; and API 6D Pipeline Valves (Gate, Plug, Ball, and Check Valves). Each of these codes presents metric values, originally standardized in the English system, to the accuracy with which the conversions warrant. The reader should refer to these references for additional detail.

The engineer is concerned with physical dimensions and mechanical properties of materials comprising the pipeline system to an accuracy limited by repeatability and error of measurement. Similarly, the quantity of commodity transported is determined by the combined accuracy and repeatability of quantity measurement and determination on a weight basis at the time of delivery for commercial transaction, relative to standard temperature and pressure. The Manual of Petroleum Measurement Standards, Chap. 15—Guidelines for the use of the International System of Units (SI) in the Petroleum and Allied Industries (API Publication 2564),[7] establishes both SI units and acceptable exceptions.

Within these guidelines, SI units will be used herein with the exceptions °C for °K, and limiting expressions of pressure, head, and flow rate to the range 0.1 to 1,000 by powers of 10. Additionally, the period will be used for the radix as in customary English usage rather than the comma. The examples illustrating pipeline design methods are shown in metric units, and example calculations and tables include both customary English and metric units.

Conversion factors between English and metric systems are based on six significant figures to enhance repeatability. However, fewer significant figures are used for exact values, or when precision of the determinable value is limited. The Manual of Petroleum Measurement Standards expresses volume and density measurements to four significant figures. API 5L tabulates nominal metric pipe dimensions to 0.1 mm and pipe minimum yield and ultimate tensile strengths in MPa as integers to three significant figures. API Specification 6D, Pipeline Valves, designates pressure class as nominal pressure (PN) and nominal pipe diameter as (DN). For NPS 4 and larger sizes, multiply the NPS by 25 to obtain the corresponding DN.

This chapter of the *Piping Handbook* includes at the end of this chapter three tables of conversion factors useful in pipeline engineering practice but may not be inclusive of all used. The range of operating pressures, flow rates, fluid characteristics, size and strength of materials used in designing and fabricating pipeline systems introduces a requirement for numerical presentation in consistent scales that include the range of a particular application without introducing varying powers of 10. The examples in this chapter follow the practice of showing calculations in the same numerical form used in the illustrations.

References

General references and supplements for the material covered in this chapter are supplied at the end of the chapter. The majority of the material covered in this chapter is also available in numerous petroleum reference books and technical papers through trade publications. No attempt has been made to reference all sources of relevant data available. Only those most pertinent to the discussions are included here, or those considered most useful.

LIQUID-PETROLEUM PIPELINE SYSTEMS

It is interesting to note that liquid-petroleum pipeline systems have been in operation since the late 1800s; and, as illustrated in the title of the Code, a number of different commodities are transported via pipeline, with widely varying properties. Therefore, to begin the discussion of liquid-petroleum pipeline systems, it is useful to illustrate some of the characteristics of fluids covered by the B31.4 Code, based on examples of real pipeline systems. Then, as this chapter is intended to be an aid in the selection of the preferred pipeline design, the concept of the *system approach* to design is introduced, wherein the hydraulic, mechanical, and operations and maintenance aspects of design are integrated and evaluated with economic analysis to select the most economically attractive system.

Characteristics of Transported Commodities

ASME B31.4 covers a wide range of petroleum liquid commodities, including crude oils, residuals, and products of refining such as diesel oil, jet fuel, gasoline, natural gas liquids, oil-water emulsions, anhydrous ammonia, alcohols, carbon dioxide, and others. The physical properties of these commodities are also variable, and each pipeline system design is based on specific properties of an identified commodity, or group of commodities in the case of multiproduct pipeline systems.

Table C5.1 identifies general characteristics of some fluid commodities covered under the Code. The API gravity, viscosity, and temperature values shown (where available) are specific examples of operating pipeline systems and should only be considered as examples. The table also identifies specific considerations for different commodities which can have an effect on the design and operation of a pipeline sytem.

Table C5.1 includes reference to API gravity, which relates specific gravity of the petroleum commodity at 60°F to water at 60°F (60/60°F). While API gravity and specific gravity are related, API gravity characterizes boiling-point fraction range and, historically, has been indicative of the commercial value of the commodity and vapor pressure.

For pipeline hydraulics, the engineer is concerned with density, viscosity, vapor pressure, and pour point. Section 15.4.10 of the Manual of Petroleum Measurement Standards (MPMS) advises that API gravity and specific gravity are to be eliminated for use. API gravity is to be replaced with absolute density in kg/m^3, and specific gravity(sg) by relative density(rd) to water, both at standard conditions 15°C and 101.325 kilopascals. Vapor pressure is determined by ASTM D323-Reid Method, for volatile crudes and products except liquified petroleum gas (LPG). Vapor pressure for LPG is determined by ASTM D1267-LP Gas Method.

System Approach to Design

In the design of an oil transportation system, it is necessary to consider many aspects of design and operation as well as project economics in determining the preferred pipeline system to transport a commodity, or commodities, from a source to a destination. On a technical or engineering level, there are three aspects of design which are interrelated in the system approach to design:

- Hydraulic
- Mechanical
- Operations and Maintenance

Decisions in one area of design directly affect, or limit, the options in another area. For example, it may be necessary to locate a pump station such that it is accessible, for example, on a main road, near an electrical power source. Thus the pipeline route will have an intermediate location point set, in addition to the origin and terminal points. Likewise, preliminary design and cost estimating are not separate and independent procedures but are instead closely related and proceed concurrently.

The hydraulic design is the process of evaluating the physical characteristics of the commodity or commodities to be transported, the quantities to be transported, the pipeline route and topography, and the range of pressures, temperatures, and environmental conditions along the route. Identifying the number and location of pump stations with respect to the hydraulic characteristics of the system is also part of the hydraulic design. There may be several viable hydraulic designs for any given pipeline-design basis and route. The most feasible is identified in conjunction with the owner or operator of the system, giving consideration to early use requirements and future capacity plans for the system.

For any one hydraulic design there are a number of mechanical system designs that can be developed to meet the criteria of the design basis and deliver the commodity from origin to destination. The mechanical design is governed by the codes and standards developed from experience in operating petroleum pipelines systems, and focuses on selection of pipe material and the specification of physical line-pipe properties such as pipe diameter and wall thickness as required by the stresses imposed on the system by the hydraulic and thermal conditions, yet within the limits set by the Code. Other aspects of the mechanical design include the type, size, and power required of pumps and other equipment or ancillary facilities required to meet the hydraulic-thermal design, such as heating stations, and the support or burial requirements for the pipeline.

The final aspect of design takes into consideration the day-to-day tasks of operating and maintaining the functional integrity of the system. These include the

TABLE C5.1 Petroleum Commodity Characteristics for Pipelines

Commodity	Temp range F/(C)	Relative density	API gravity	Viscosities cSt at F/(C)	 cSt at F/(C)	Pour point F/(C)	Vapor press psi (kPa)	Remarks: flow regime, rheology, general conditions
Residuals:	150 to 250 (65 to 121)	1.02	7.2	50,000 at 130 (54)	330 at 250 (121)	130 (54)	—	Common design temperature in the range of 150 to 250°F (65–105°C). [One design for 800°F (427°C) is operating.] Thermal design required. Laminar flow.
	150 to 250 (65 to 121)	0.96	15.9	1,000 at 100 (38)	45.7 at 210 (99)	90 (32)	—	
Crudes:								
General	40 to 160 (4 to 70)	0.84	12–40	11 at 68 (11 at 20)	4.1 at 122 (4.1 at 50)	55 (13)	15 (103)	Generally newtonian in range of operating temperature. May require thermal design considerations. Transition to fully turbulent flow.
High wax content	70 to 140 (20 to 60)	0.81	35–45	7.4 at 122 (7.4 at 50)	3.3 at 140 (3.3 at 60)	95 (35)	15 (103)	Generally newtonian above cloud point; develops yield stress and nonnewtonian flow characteristics after static cooling. Transition to fully turbulent flow.
Shale oil	40 to 120 (4 to 50)	Data still emerging in shale oil systems—state of the art						Fluid properties vary with method of extraction and preparation for transport. Require considerations for vapor pressure, interfacial mixing and fluid properties when setting operations requirements.
Products:								
No. 2 furnace	30 to 80 (−1 to 27)	0.82–0.84	39	5.7 at 30 (5.7 at −1)	2.6 at 100 (2.6 at 38)	—		Newtonian fluids. Fully turbulent flow. May require thermal design.
Diesel–No. 1	30 to 80 (−1 to 27)	0.83		2.8 at 30 (2.8 at −1)	1.4 at 100 (1.4 at 38)	—		Multiproduct hydraulic design is based on the combination of fluid properties in the system that produces the maximum system stress and required pumping power at stations.

TABLE C5.1 Petroleum Commodity Characteristics for Pipelines *(Continued)*

Commodity	Temp range F/(C)	Relative density	API gravity	Viscosities: cSt at F/(C)	Viscosities: cSt at F/(C)	Pour point F/(C)	Vapor press psi (kPa)	Remarks: flow regime, rheology, general conditions
Diesel–No. 3	30 to 80 (−1 to 27)	0.88		10 at 30 (10 at −1)	3.6 at 100 (3.6 at 38)			
Jet fuel	30 to 80 (−1 to 27)	0.78		2.2 at 30 (2.2 at −1)	1.3 at 100 (1.3 at 38)			
Gasoline	30 to 80 (−1 to 27)	0.71–0.73	65	0.8 at 30 (0.8 at −1)			30@100 (207@38)	Require considerations for vapor pressure, interfacial mixing, and fluid properties when setting operations requirements, selecting pumps.
Natural gas liquids (NGL) and other high–vapor pressure petroleum liquids		30 to 130 (−1 to 55)	0.5	0.23 at 30 (0.23 at −1)	0.2 at 100 (0.2 at 38)	—	200 at 60 (1380 at 15) to 700 at 120 (4830 at 50)	Pressure is maintained above the critical pressure to avoid two-phase flow. Special considerations for loading pumps and blowing down pipeline sections are required. Fully turbulent flow.
Other:								
Oil-water emulsions		There is no specific example pipeline system for this commodity, or the information is proprietary or confidential.						May be either an oil or water suspension; fluid properties are extremely variable.
Alcohols		There is no specific example pipeline system for this commodity, or the information is proprietary or confidential.						Toxic; flame may be invisible in daylight.
Anhydrous ammonia		There is no specific example pipeline system for this commodity, or the information is proprietary or confidential.						Toxic and corrosive—Code requires >0.2 percent water by weight to inhibit stress corrosion cracking.
Carbon dioxide		There is no specific example pipeline system for this commodity, or the information is proprietary or confidential.						(See high–vapor pressure liquids.) Dehydration is required for pipeline quality. Heavier than air, toxic in elevated concentrations.

necessary control systems to operate the system within its design parameters and to promote safe and continuous operation.

The preferred pipeline system for a given set of conditions is selected through an economic comparison of several systems, seeking to identify the system that yields the best economic return on the investment dollar, depending on the initial and subsequent capital costs, the method of financing, and the operating and maintenance costs for the economic life of the investment. If alternatives require capital investments, e.g., for pumping stations, at different future dates, then these costs should be compared on a present-value basis, discounted at a real (after inflation) interest rate, to ensure a valid, unbiased selection of the preferred system. The details of economic analysis are addressed in financial analysis references.[22]

Programs are available for hand-held calculators, personal computers and mainframe computers for separately performing the hydraulic, thermal, mechanical, and economic analyses and design of petroleum pipeline systems.[28,29]

Concurrent with the hydraulic, mechanical, and operations and maintenance designs, the pipeline project team will also be performing many tasks related to the construction of the pipeline. These include technical and environmental surveys of the pipeline route and surrounding areas, preparation of environmental impact reports, acquisition of permits and rights-of-way, procurement of construction materials, development of construction costs considering pipe diameter, wall thickness, grade of steel and welding procedures, and preparation of contract specifications and bidding papers. These and other topics related to construction of pipelines are covered in other handbooks.

The investigation of any pipeline system begins by establishing the design basis for the commodity, then making a preliminary selection of pipe diameters and cost estimates for comparative economic attractiveness. If the preliminary estimates indicate further consideration is desirable, then preliminary feasibility considerations begin by selecting possible routes and developing a preliminary design.

The preliminary cost estimates developed in Tables C5.7, C5.7M, C5.8, and C5.8M, are examples of initial order-of-magnitude cost estimates for selecting alternative pipe diameters and overall cost feasibility. The pipe diameters used in Table C5.8M are the basis for the discussion of the hydraulic designs illustrated in the section that follows. The pipe diameters used have been selected to illustrate considerations in the hydraulic analysis and effect on mechanical design and cost rather than to select a preferred design for the example system discussed.

HYDRAULIC DESIGN

The hydraulic design integrates the physical characteristics of the transported commodity along a given pipeline route, within specified operating conditions as established in the design basis. The result of the hydraulic design is identification of the total system energy required to meet the design criteria. In addition, the hydraulic calculations indicate a range of feasible pipe diameters and preliminary spacing of pump stations along the route.

When the design is finalized, i.e., the route selected, pipe-line size determined, and type of pipe selected, the hydraulic calculations are refined to determine the conditions for over-pressure control during line shutoff and surges during operation. Hyraulic calculations can also be made for the variables in the operating conditions (temperature, ranges of viscosities for products pipelines, et cetera) and for future expansion of system capacity.

Route Selection

Given the task of transporting a liquid commodity from one point to another—whether it is from the point of production or storage to a processing plant, or from the process plant to distribution facilities—the first selection of route will logically be the shortest course, or a straight line.

While a straight-line route is a reasonable first approximation of the pipeline route, there are several common-sense reasons for deviation, including:

- Significant natural obstacles such as mountain ranges, rivers, swamps, et cetera.
- Minimizing of control points in the hydraulic profile (discussed later).
- Access for construction equipment and materials.
- Permitting restrictions.

A preliminary route is determined using suitable maps of the area which need to show geographic features such as contour lines as well as towns, roads, rivers, railroads, existing pipelines and utility corridors, et cetera. World Aeronautical Charts are available for most parts of the world, on different scales, for this purpose. U.S. Geological Survey maps are particularly useful for pipeline routing in the United States. Aerial photographs are also useful.

Several of the factors which will influence the selection of the design route may not be readily identifiable or resolved until later phases of design, in particular, environmental and permitting requirements and land acquisition. However, a preliminary route can be selected and later modified when more information on the specific and final route is available. Once an initial route is identified, the ground profile is plotted for use in the hydraulic design.

Example C.5.1. Figure C5.2 is a potential pipeline-route profile for a crude-oil pipeline design illustrated in this chapter. This example is a simplified profile. Real

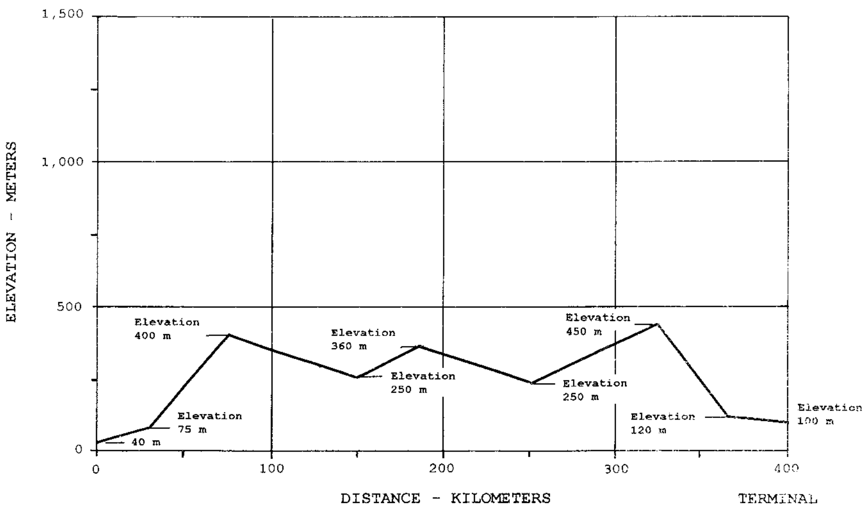


FIGURE C5.2 Example pipeline route profile.

pipeline systems typically have much more detail, showing river crossings, mountain ranges, and other geographical points.

It should be emphasized that this is not a cross-section of a straight-line course from origin to terminal but represents the real route, avoiding major obstacles mentioned earlier. The selected route results in a length of 248 miles (400 kilometers). Elevations of significant points along the route are shown.

If more than one reasonable alternative route has been identified, they are also plotted, and designed and analyzed simultaneously. Following sufficient identification, the alternatives will be evaluated on the basis of construction, operation, and maintenance cost, and a comparative economic analysis spanning the effective project or system life is performed to identify the preferred alternative. This topic is covered in more detail later in this chapter.

Design Basis

When beginning the design of a pipeline system, it is necessary to define the basis of the design as completely as possible. The general parameters which are required for the design of the system include:

- System operating parameters, such as design *throughput* or flow rate; operating temperature of the system
- Environmental conditions, such as ambient ground and air temperature (average and extremes)
- Properties of the transported fluid(s), or commodity, such as viscosity, relative density, vapor pressure, and pour point temperature.

System Parameters. There are a number of system parameters which are typically defined by the operating company or owner of the system. It is helpful for the design flow rate for the system to be defined as closely as possible. Maximum, minimum, and forecast future daily or annual throughputs of the pipeline system are required for good design, resulting in selection of the economically preferred line size as early as possible. This limits the iterations of the design as well as the range of alternatives.

The design throughput of an oil pipeline may vary by year and is usually expressed as the average daily flow rate in barrels per day (BPD) or 1000 m^3 per calendar day (1,000 m^3/cd), or in million tonnes[a] per annum (MTA), which requires conversion to daily rates for computation. The actual flow rate that a system must be capable of attaining to compensate for lost capacity from shutdowns and reduced flow conditions is the flow rate per operating day BPOD, or (1,000 m^3/d), which is greater than the flow rate per calendar day. The ratio of flow rate per calendar day to operating day is the load factor.

$$\text{load factor} = \text{rate per calendar day/rate per operating day} \qquad \text{(C5.1)}$$

A well-operated pipeline can be expected to have a load factor of 92 to 95 percent. For domestic pipelines, this may be used in the design procedure unless special circumstances dictate a lower factor. Pipelines to be located in remote areas, more complex systems with many pump stations, or pipelines operated with expected flow variations would be more reasonably designed to a lower load factor, 85 to

[a] A tonne is equal to 1,000 kg (2205 lbs).

90 percent, to account for greater system downtime, i.e., from interruptions in service as a result of operations and maintenance.

Environmental Parameters. The critical environmental parameter for the hydraulic design is the ambient temperature of the ground, for buried pipelines, or the air, for aboveground systems. Most locations will have seasonal variations, and long pipeline systems may have variations over the length of the system. It is important to identify the mean or average ambient temperature as well as the seasonal and local extremes.

Properties of the Commodity. Specification of the commodity to be transported includes identification of viscosity, density, vapor pressure, and pour-point temperature. Some of these properties will have to be determined from laboratory tests on specific commodity samples. However, design may proceed on the basis of a typical commodity and include flexibility for a specified range of variation.

Viscosity is the physical property of fluids which resists flow and, for liquids, varies inversely with temperature. Besides density, viscosity is a key characteristic of the fluid to be considered in the design of liquid pipelines, having a significant effect on determining line size, station spacing, and pumping-power requirements. A discussion of viscosity, including the definitions of kinematic and absolute viscosity, has been included in Chap. B8, Flow of Fluids. Several of the references may also be consulted for viscosity data on specific hydrocarbons and other fluids.[7,8,9,10,11,12]

Example C5.2. Figure C5.3 shows the approximate viscosity-temperature (for a limited range of temperatures) relationship for a crude oil sample, having a viscosity of 6.0 centistokes (cSt) at 129.2°F (54°C), and 23.7 cSt at 59°F (15°C). This will be used for the basis of the example.

The viscosity-temperature is not truly linear, unless plotted on a special ASTM graph paper designed for this purpose. Figure C5.3 is plotted for the range 32 to 140°F (0 to 60°C) to the coordinates of ASTM D-341, Chart VII: kinematic viscosity, middle range, degrees Celsius, for the temperature range of −40 to 302°F (−40 to 150°C).

Reference ASTM D-341[11] for further discussion.

For crude oils, the pour point of the oil, i.e., the temperature at which viscosity of a cooling oil abruptly increases, needs to be considered to determine if special measures are required to move the oil when ambient ground and air temperatures are below this temperature. An oil with a pour-point temperature above the ambient condition will require dilution with a lighter stock oil (sometimes referred to as cutter stock), addition of a pour-point depressant, or a heated pipeline system.

Isothermal Systems

Oil transportation pipelines typically will have some variation in temperature over the entire system. As this then affects viscosity and has other design impacts, the discussion of the hydraulic design will continue based on isothermal—constant temperature—systems, with the assumption that temperature variations are limited. Special considerations for nonisothermal systems are discussed later.

Example C5.3. Crude-oil pipeline system, isothermal flow, 248 ground miles (400 km) over a route with a maximum elevation of 1476 ft (450 m). Figure C5.2, included earlier, shows the route profile for this example. The initial elevation is 131 ft (40 m) at the origin, and a liquid head level of 492 ft(150 m) with respect to the datum is maintained at the terminal.

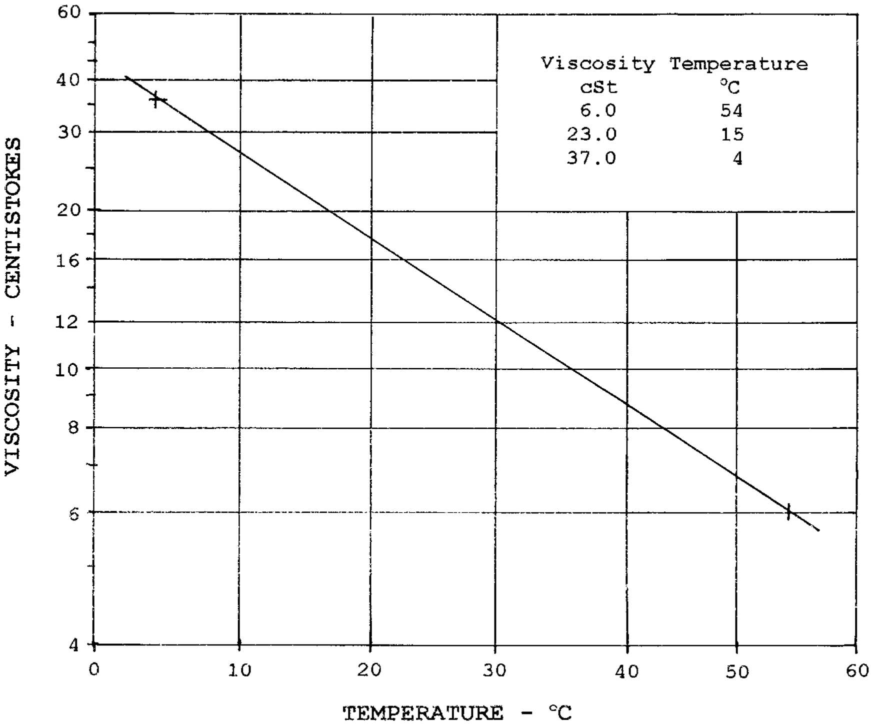


FIGURE C5.3 Example viscosity versus temperature, crude oil.

Design Basis:

API gravity at 60°F	32.6
Specific gravity (relative density) based on water @60°F (15°C)	0.8623
Ambient design temperature	39.2°F (4.0°C)
Viscosity @ 59°F (15°C)	23.0 cSt
Viscosity @ 129.2°F (54.0°C)	6.0 cSt
Flow rate (average)	169,800 BPCD (27,000 m^3/d)
Load factor	0.9

The design temperature for average operation differs from the measurement standard of the design crude. Consequently, it is necessary to make adjustments from the values given here at standard conditions @ 60°F (15°C) to the operating conditions.[7,8,10,11]

MPMS Chap. 11.1[7] Tables 5.3A and 5.4A tabulate correction factors for density and volume that adjust standard conditions to the design conditions for generalized crude oils, which will be used here. Additionally, Tables 5.3B and 5.4B tabulate correction factors for generalized products.

The correction factors are applied to the standard conditions to adjust density and volume to the design conditions. Density-and volume-correction factors are reciprocals of each other and can be applied by multiplication or division as required.

$$\text{Relative density } (rd) = \text{standard density kg/m}^3\ /1{,}000\ \text{kg/m}^3 \tag{C5.2}$$

$$\text{Relative density @ } 4^\circ\text{C} = 862.3/1{,}000 \times 1.0091 = 0.8701$$

$$\text{Design flow rate} = 27{,}000/(0.9 \times 1.0091) = 29{,}790\text{m}^3/\text{d}$$

adjusted for both load factor and volume correction factor.

Use 188,700 BPOD (30,000 m^3/d) (rounded)

Viscosity @ 4°C = 37 cst, see Fig. C5.3

System Energy

Both head, H, and pressure, P, are used in discussing system energy. The conversions from head to pressure, and vice versa, are given by the following formulas, for standard conditions 60°F (15°C), with correction for specific gravity (sg) or relative density (rd) of the commodity, which adjusts the pressure or head to operating conditions.[7,8,10,11]

$$H,\ \text{ft} = 2.31 \times P,\ \text{psi}/sg \tag{C5.3}$$

or

$$H,\ \text{m} = 9.807 \times P,\ \text{kPa}/rd$$

$$P,\ \text{psi} = H,\ \text{ft} \times 0.433 \times sg \tag{C5.4}$$

or

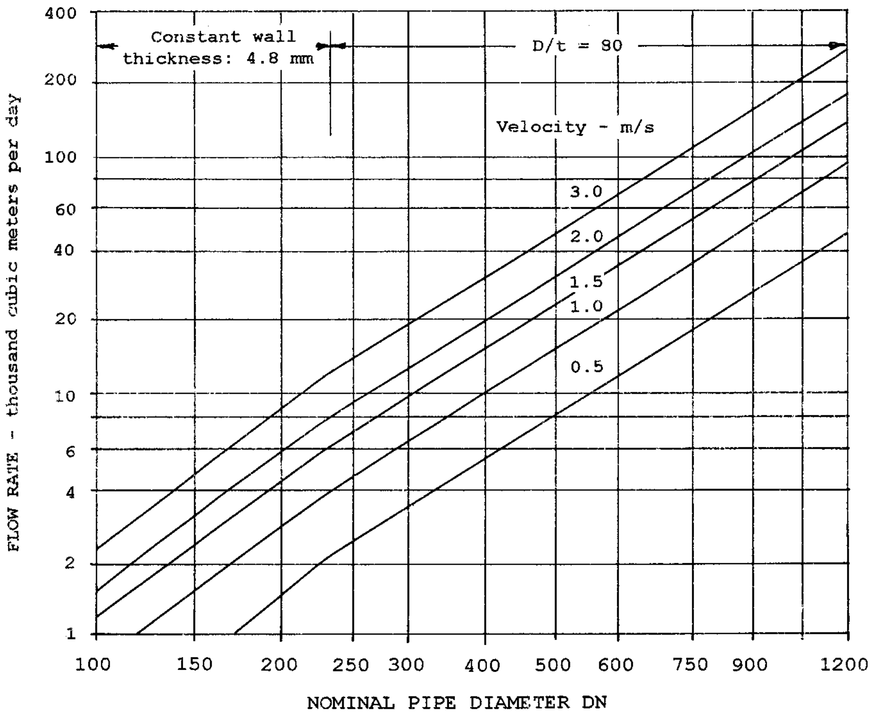
$$P,\ \text{kPa} = H,\ \text{m} \times 0.102 \times rd$$

The total pressure drop in a pipeline system comprises three components:

1. Static pressure drop, due to changes in elevation
2. Acceleration pressure drop, due to changes in pipe geometry or phase
3. Friction pressure drop, due to flow rate, fluid properties, and pipe characteristics

When using the term *head* of fluid to denote system energy, the three components of head are typically referred to as static, or elevation head, H_s, velocity head, H_v, and friction head, H_f. These are the terms of the Bernoulli equation discussed in Chap. B8. Head loss expressed per unit of pipe length uses the lower case h, with the same subscripts.

The elevation head, H_s, is the difference between the inlet and outlet elevations between points on a system. The velocity-head component for long pipeline systems,



Note: DN (metric) = 25 x NPS (English)

FIGURE C5.4 Flow rate versus nominal pipe diameter for velocities of 0.5, 1.0, 2.5, 2.0, and 3.0 m/s.

and generally for systems with high head requirements, is a small percentage of the total head and is often assumed to be negligible in energy calculations.

Friction-head loss, on the other hand, is the dominant effect in most liquid pipeline systems and can be calculated using one of the following equations previously discussed in Chap. B8, Flow of Fluids:

- Darcy-Weisbach
- Fanning
- Hazen-Williams

The former two equations require a determination of friction factor as a function of the Reynolds number. The Hazen-Williams equation accounts for friction in a coefficient *C*, which is an empirically determined factor for the specific commodity.

Friction-loss tables based on the Darcy-Weisbach formula are available in many engineering reference books for water and viscous liquids.[9,13] Generally these tables, as illustrated in App. E, give friction loss in units of head loss, ft of head per 100 ft of pipe. Multiplying by 10 gives pipe friction loss in ft/1000 ft, or m/1000 m of pipe length. Each of these unit friction losses can be converted to pressure loss per unit of length using the appropriate conversion factor and density of fluid.

Hydraulic calculations are not complex for pipelines with a single commodity having little variation in viscosity. The first task is to specify the nominal pipeline diameter, or range of diameters, based on the design flow rate.

An approximation of line size can be made for the design flow rate using the tables available in handbooks, standard fluid-design manuals, or engineering reference books. The selection of line size is based on a preliminary economic analysis of alternatives discussed in Table C5.7 or C5.7M. Depending upon commodity, location, distance, and energy costs, velocities for preliminary estimates of line size are typically in the range of 2 to 10 ft/s (1 to 3 m/s). Figure C5.4 shows daily flow rate in 1000 m^3/d versus nominal pipe diameter, DN, in millimeters for velocities between 0.5 to 3 m/sec.

At this point a preliminary choice of one or more wall thicknesses is necessary, as eventually a cost of pipe will be found and because wall thickness will also be a component of the internal pressure and stress limitations. For a given diameter, *D*, also referred to as the nominal pipe size (NPS) or diameter (DN), a range of pipe Specified Minimum Yield Strength (SMYS) and wall thickness, *t*, is available.[1,2] Standard pipe wall thickness, depending on diameter, such as 0.250 in (6.4 mm) or 0.375 in (9.5 mm), may be used for the early design, and later adjusted if special commodity or system requirements are identified, such as high pressure or limitation on the allowable pipe stress in a section of the system.

The examples that follow use the internal design pressure, calculated by Code formula for the pipe grade and wall thicknesses, to determine the maximum allowable operating pressure (MAOP). After converting MAOP in units of pressure to units of head (MAOH) in feet or meters for the particular density of the fluid, the calculated slopes of hydraulic gradients between the MAOH at a pump station and the intersection with the ground profile determine pump-station locations for the given diameter and flow rate.

Example C5.4. Returning to the example, for the flow rate of 188,700 BPOD (30,000 m^3/d), assume that there are three diameters of pipe which are viable to illustrate the effect of pipe diameter on design. The following table can be constructed using a tabulation of the Darcy-Weisbach formula and a minimum wall thickness of 0.250 in (6.4 mm), for the three diameters:

TABLE C5.2 Example: Pipeline System Parameters

Parameter	NPS 16		NPS 20		NPS 24	
O.D. and wall thickness, in	16.0 × 0.250		20.0 × 0.250		24 × 0.250	
Flow rate, MBPD nominal	126.8	251.6	126.8	251.6	126.8	251.6
Roughness, in	0.0018	0.0018	0.0018	0.0018	0.0018	0.0018
Reynold's number	20,385	40,450	16,200	32,150	13,445	26,680
Darcy-Weisbach friction factor	0.026	0.022	0.027	0.023	0.029	0.024
Unit friction loss, ft/mi	65.37	220.42	21.88	73.34	9.01	30.05
Velocity, ft/s	6.64	13.29	3.97	7.95	2.73	5.47

TABLE C5.2M Example: Pipeline System Parameters

Parameter	DN 400		DN 500		DN 600	
O.D. and wall thickness, mm	406.4 × 6.4		508.0 × 6.4		610.0 × 6.4	
Flow rate, cubic meters per day	20,000	40,000	20,000	40,000	20,000	40,000
Roughness, mm	0.0457	0.0457	0.0457	0.0457	0.0457	0.0457
Reynold's number	20,240	40,480	16,085	32,170	13,335	26,670
Darcy-Weisbach friction factor	0.026	0.022	0.028	0.023	0.029	0.024
Unit friction loss, m/km	12.22	41.78	4.09	13.90	1.68	5.67
Velocity, m/s	1.90	3.80	1.20	2.40	0.825	1.65

Figure C5.5 is graphed using the data from Table C5.2M. The use of log/log scales results in an approximately linear relationship between flow rate and unit friction loss, h_f.

Two of the diameters also show the effect of adjusted viscosity to the operating condition of 39.2°F (4°C).

It is important to point out the adjustments made for the viscosity at operating temperature. If using a reference table based on water (sg = 1.0) to determine unit friction loss, or if the friction loss chart is based on viscosity at 60°F, corrections would need to be made for the design temperature and viscosity of the specific commodity at operating conditions.

Summing the friction losses for a specific set of pipeline diameters along the system results in the net friction head that must be overcome by the pump stations of the system. In addition, the positive or negative static head due to elevation difference between the inlet and outlet must be considered in determining the pumping requirements of the system. Minor losses at valves and fittings between stations are normally ignored, initially, or an allowance of an added length of pipe may be added to the scaled length to determine the estimated total system friction head loss.

Hydraulic Gradient

The hydraulic gradient is a profile representing the static head at any point in the pipeline system, relative to a common datum elevation, which is usually mean sea level. Ground elevation is represented by the route-elevation profile to the same datum. Energy added to the system through a pump station is plotted above the elevation profile. Head losses from friction, et cetera, are also shown graphically. For a pipeline system with constant parameters along the system, such as viscosity, relative density, and diameter, the hydraulic gradient will be a straight line with a slope equal to the friction loss per unit of length, h_f, for a specific flow rate. Therefore, the actual pressure in the pipeline at any point along the route is the difference between the hydraulic gradient and the ground elevation.

Figure C5.6 illustrates the slope of hydraulic gradients based on the 188,700 BPOD (30,000 m^3/d) flow rate and the three diameters tabulated in Table C5.2M, where an assumed pump-station discharge head of 3281 ft (1000 m) is shown at the origin. The hydraulic gradient, being the available head at any point along the pipeline route, cannot be less than the elevation of the pipeline-route profile. Therefore, where the gradient intersects the pipeline elevation, a pump station is required. It is obvious from Fig. C5.6 that unrealistically high station discharge

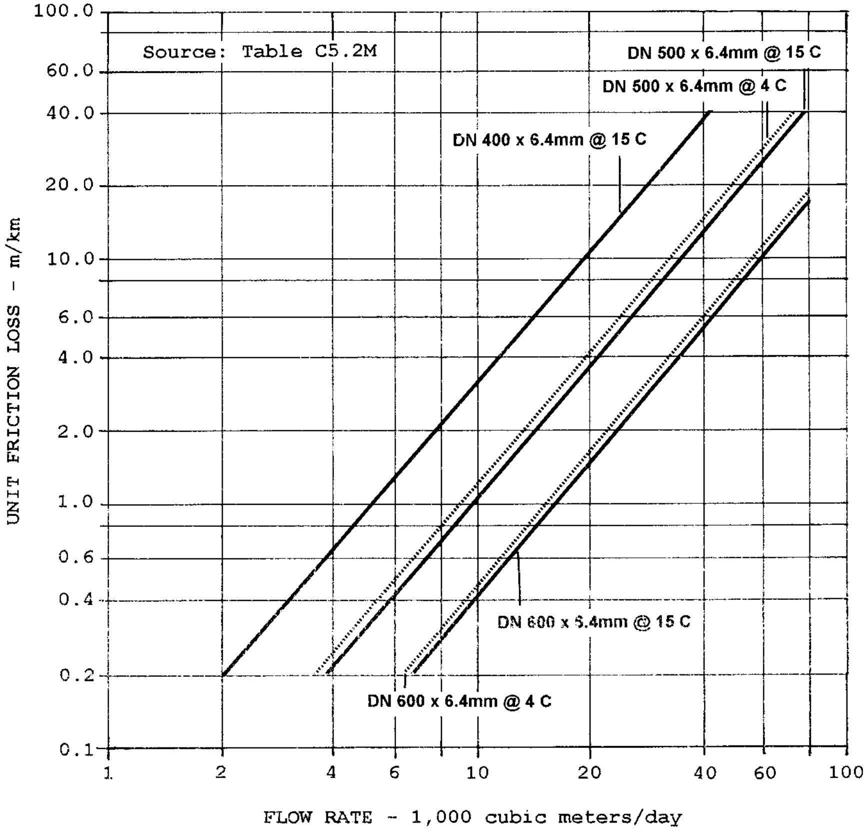


FIGURE C5.5 Unit friction loss versus flow rate.

heads would be required for only one pump station using NPS 16 (DN 400) or NPS 20 (DN 500) pipe. However, one station could be used on the NPS 24 (DN 600) pipeline by elevating the hydraulic gradient at the station discharge sufficiently to clear the ground profile at mile post 202 (kilometer post 325). The station discharge head is usually selected on the basis of the maximum allowable pressure rating for valves and fittings, or the MAOP of the selected pipe, whichever provides the required operating flexibility, and is the most attractive economically.

The example above for the NPS 24 (DN 600) pipeline illustrates an important use of the hydraulic gradient for identifying hydraulic control points on a pipeline route. A hydraulic control point is a high elevation point that will govern the inlet head for a section of pipeline as discussed above. In the example, the maximum profile elevation of 1476 ft (450 m) at milepost 199 (kilometer post 320) is a control point. It is logical that the hydraulic gradient must clear the ground elevation control point, but in doing so, two situations may result downstream as shown in Fig. C5.7 for the hydraulic gradient of the NPS 24 (DN 600) line.

1. Either the line is designed to flow full, requiring that back pressure control is provided at the terminal, or

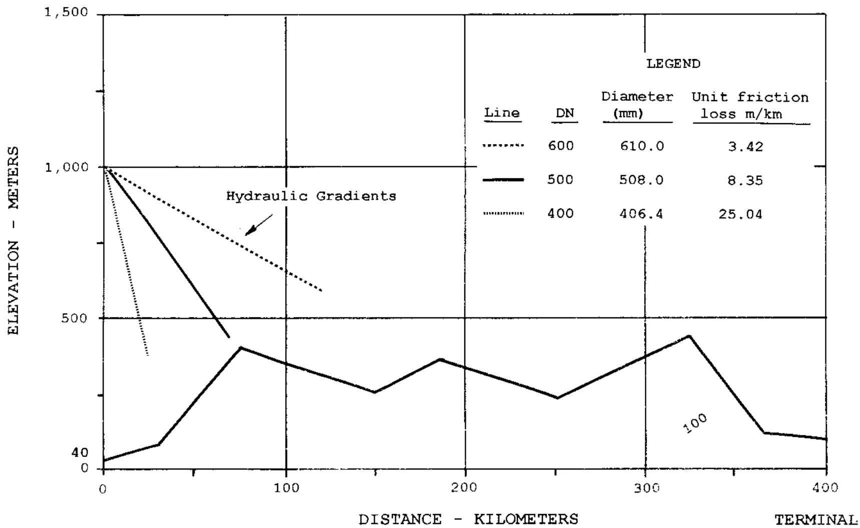


FIGURE C5.6 Example hydraulic gradient for 30,000 m^3/d for DN 400, DN 500, and DN 600 nominal pipe size.

2. Without back pressure control, the length of line downstream of the control point will flow partially full in a cascade or slack-line condition. Slack flow may not be a desirable condition in some pipeline systems; however, systems transporting crudes having low vapor pressures can, and do, operate successfully with slack flow.

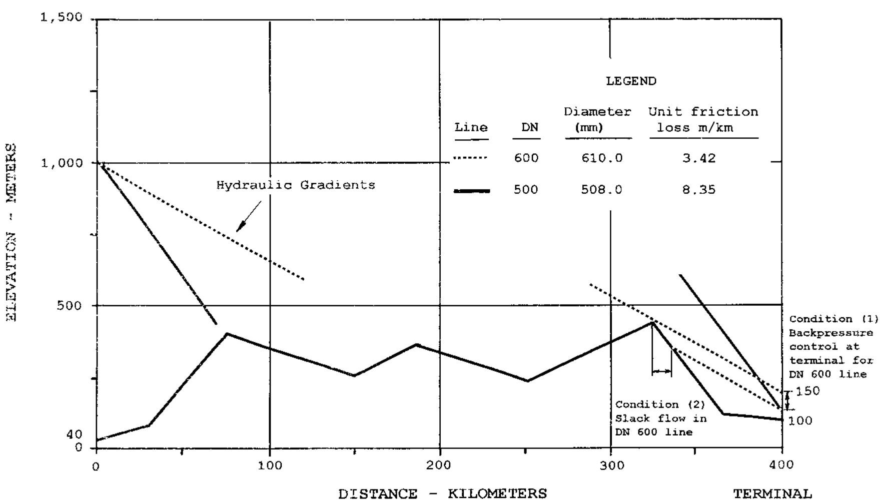


FIGURE C5.7 Example for clearance of a control point illustrating backpressure control and slack flow for 30,000 m^3/d.

In this example, the hydraulic gradient of the NPS 20 (DN 500) line clears the elevation of the control point, and the considerations for back-pressure control or slack flow only occur at lower flow rates.

Throughout this section, the example which will be discussed is a system with constant flow rate and one delivery point. In general, pipeline systems may have intermediate injection or delivery points along the route such that flow in the main line is increased or decreased. Hydraulic design of this type of system requires that the gradients for each section of the system with different flow rates be calculated and plotted in succession along the pipeline.

Maximum Allowable Operating Pressure

Section 404.1.2 of the Code expresses the required wall thickness for the internal design pressure for straight pipe by Eq. C5-5.

$$t = P_i \times D/(2 \times S) \tag{C5.5}$$

where P_i = maximum allowable internal gauge pressure, psig (SI and metric units, see Eq. C5.6)
D = outside diameter, in (mm)
S_A = allowable stress, psig (MPa)—see sec. 4.2
t = wall thickness, in (mm)

Expressing Eq. C5-5 in the following form for calculating the internal design pressure,

$$P_i = B \times S_A \times t/D \tag{C5.6}$$

The constant B is determined by the units of the allowable stress S_A, and the units of pressure to be determined using the ratio of t/D in consistent units.

Units	SI	bar	Metric	English
Allowable stress S_A	MPa	MPa	MPa	psi
Pressure	MPa	bar	kPa	psi
Constant B	2.0	20	2000	2.0

Wall thickness, *t*, for calculation of the MAOP excludes additional thickness for corrosion allowance, or imposed stresses such as concentrated loads at supports, thermal expansion or contraction, and bending. Determination of additional required wall thickness for these considerations is discussed later in this chapter.

At this point, the reader is referred to the tabulations for specified minimum yield strength and allowable stress which can also be found in API Specification 5L, Specification for Line Pipe,[2] which is based on the restrictions on allowable stress, according to the Code.

The following example uses the tabulated allowable stress values of API Specification 5L, and solves Eq. C5.6 for P_i based on an assumed diameter, wall thickness, and material.

Example C5.5. For the example pipeline system, assume the steel pipe is API grade 5L-X60, having a specified minimum yield strength of 60,000 psi (413 MPa).

TABLE C5.3 Maximum Allowable Operating Pressure

Pipe nominal size, NPS	16		20		24	
Pipe wall thickness, in	0.250	0.312	0.250	0.312	0.250	0.312
MAOP, psig	1,350	1,684	1,080	1,347	900	1,123
MAOH, ft[2]	3,580	4,468	2,864	3,574	2,387	2,978

TABLE C5.3M Maximum Allowable Operating Pressure

Pipe nominal diameter, DN	400		500		600	
Pipe wall thickness, mm	6.4	7.9	6.4	7.9	6.4	7.9
MAOP, kPa[1]	9,366	11,561	7,493	9,249	6,240	7,702
MAOH, m[2]	1,097	1,354	878	1,083	731	902

[1] API wall thickness in millimeters is approximately 0.6% greater than direct conversion of API wall thickness in inches, which yields a corresponding increase of MAOP when converted from wall thickness in English units. Maximum allowasble operating pressures are gauge.

[2] Allowable head is based on 0.8701 specific gravity (rd).

Tables C5.3 and C5.3M tabulate the MAOP and MAOH for grade X60 pipe for three diameters and two wall thicknesses.

(Calculations are based on Eq. C5.5, using actual pipe diameter, mm, and tabulated value of SMYS from API 5L. Allowable stress, S_A, calculated using Eq. C5-14, is 74.93 bar (7,493 kPa) with design factor of 0.72, and SMYS of 413 MPa per tables in API Spec 5L.)

The MAOP is used in the development of the system hydraulic design as a limit on the internal pressure component of the hydraulic gradient. When plotted above the route profile, the hydraulic gradient may not exceed this limit and still be designed to the B31.4 Code. MAOP is also used in determining the approximate number of pump stations.

Pump Stations

The calculated total system head required to achieve a given flow rate through a pipeline system with a selected diameter, which includes the total friction component and the static elevation difference between inlet and outlet, determines the pumping requirements of the system. As seen in Fig. C5.6, at least one intermediate pump station is required for the example system and pipe diameters illustrated. This section discusses a method of determining how many pump stations are required and locating them on the basis of hydraulic balance and a graphical method.

Number of Pump Stations. A rough number of pumping stations is found by dividing the total system pressure or head required to overcome elevation changes and friction, by the maximum allowable operating pressure or head, MAOP or MAOH, for a specific diameter, wall thickness and pipe material, using consistent units. For example:

$$\text{No. of pump stations} = \text{Total System Head}/(\text{MAOH} - \text{NPSH}) \qquad \text{(C5.7)}$$

Example C5.6. The first step to determining the number of pump stations is to determine the pressure or head required to overcome the frictional resistance caused by the flow of oil. Using a NPS 20-(DN 500) diameter pipe size with 0.250 in- (6.4 mm) wall thickness (this is usually written 20 × 0.250 in (508.0 × 6.4 mm) and a hand-held calculator programmed with the Darcy-Weisbach equation, the unit friction head loss, h_f, for the design flow of 188,700 BPOD (30,000 m^3/d) is 44.06 ft/mi (8.35 m/km).

Summing the friction head loss over the 248 miles (400 kilometers) measured length of the system yields 10,958 ft (3,340 m) of total head loss due to friction, H_f.

There is also a static head, H_s, of 361 ft (110 m) to be pumped against due to a change in elevation between the initial point and the head required at the terminal.

The total head, H, which is required from all pump stations on the system, is the sum of the static head and the friction head:

$$\begin{aligned} H,\ \text{ft} &= (H_f + H_s)\ \text{ft} & H,\ \text{m} &= (H_f + H_s)\ \text{m} \\ &= 10{,}958 + 361 & &= 3{,}340 + 110 \\ &= 11{,}319\ \text{ft} & &= 3{,}450\ \text{m} \end{aligned} \tag{C5.8}$$

The MAOP of API 5L Grade X60 pipe NPS 20 × 0.250 in wall (DN 500 × 6.4 mm), expressed in units of head is 2 864 ft (873 m). Assuming 82 ft (25 m) of head loss in station piping, the available head per station is

$$\begin{aligned} \text{Number of pump stations} &= 11{,}319\ \text{ft}/(2{,}864 - 82)\ \text{ft} = 4.07 \\ &= 3{,}450\ \text{m}/(878 - 25)\ \text{m} = 4.04 \end{aligned}$$

Fractional stations may be accounted for by using heavier wall pipe for a limited length, or by installing a booster station. These alternatives are evaluated on the basis of economics.

Location of Pump Stations. One pump station is located at the initial point of the pipeline. Downstream stations are initially located such that each of the pipeline sections will be in hydraulic balance, i.e., with each station having approximately the same differential head or pressure, thereby distributing the energy load equally.

$$\begin{matrix} \text{Pump Station} & & \text{Total} & \text{Number of} \\ \text{Differential Head} & = & \text{System Head} & /\text{Pump Stations} \end{matrix} \tag{C5.9}$$

In level terrain, this procedure for determining the number of pump stations would space stations equally along the route. However, in uneven terrain, such as in the example, the stations can be balanced using a graphical technique.

Figures C5.8 and C5.9 illustrate stations located graphically for the NPS 20 (DN 500) and NPS 24 (DN 600) pipelines by beginning at the elevation required at the terminal, then drawing the hydraulic gradient upstream to the intersection of the MAOH of the pipe, which is plotted above the ground-elevation profile. This location establishes the Actual Discharge Head (ADH) at the discharge valve of the pump station. The ADH is used here to avoid confusion with the term Total Discharge Head (TDH), which refers to pump performance relative to the centerline elevation of a pump.[13]

The elevation of the hydraulic gradient into the station is plotted above the ground elevation by the allowance to provide station losses to the pumps, and provide the required Net Positive Suction Head (NPSH) at the pumps. In this

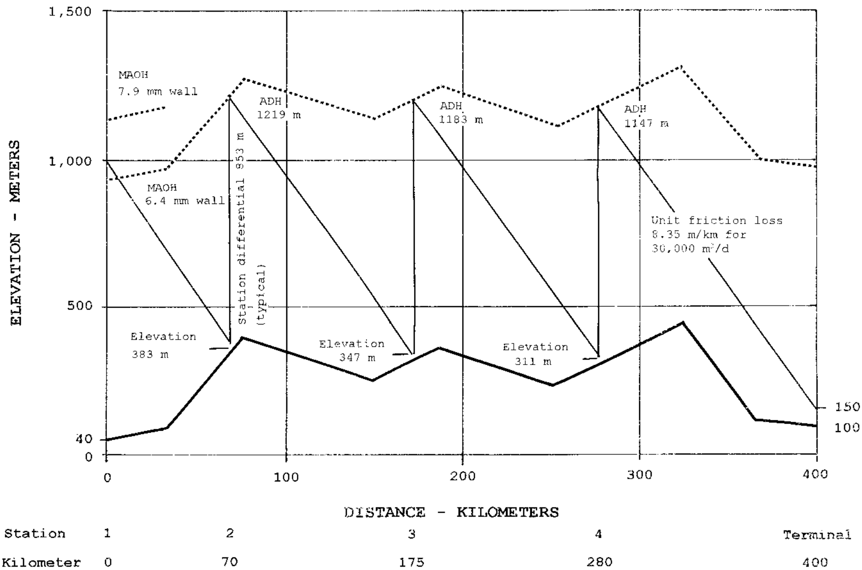


FIGURE C5.8 Hydraulic gradient for DN 500 pipeline example, 30,000 m^3/d.

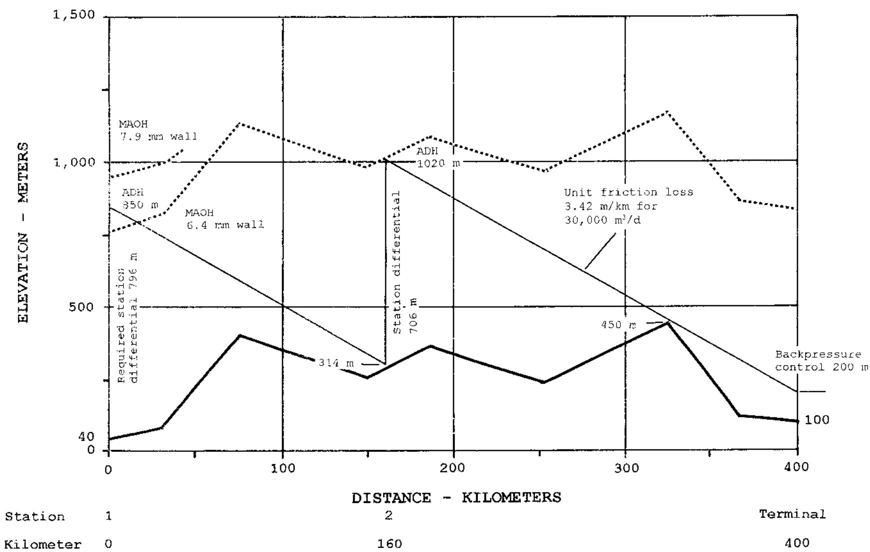


FIGURE C5.9 Hydraulic gradient for DN 600 pipeline example, 30,000 m^3/d with profile control point and backpressure control at terminal.

example the allowance is 25 meters. The difference between the ADH and the elevation of the incoming head is the difference between the head at the station discharge valve and the station suction valve, and is the station differential. In this example, the station differential is the MAOH of the pipe, minus the allowance for station suction losses and NPSH. The example below shows the calculation for the station differentials for the NPS 20 (DN 500) and NPS 24 (DN 600) systems.

Example C5.7: Station Differentials

Nominal pipe size	NPS 20 (DN 500)	NPS 24 (DN 600)
OD and wall, in (mm)	$20 \times 0.250(508.0 \times 6.4)$	$24 \times 0.250(610.0 \times 6.4)$
MAOH, ft(m)	2863(873)	2387(731)
Station loss, ft(m) and NPSH	82(25)	82(25)
Station differential, ft(m)	2781(848)	2305(706)

The actual differential head required of the pumps, however, is greater than the station differential by adding the total of all station losses including a discharge pressure-control valve if one is included in the station discharge piping.

Generally, an allowance of 50 to 100 feet (15 to 30 meters) of head at the intake of the pump station will account for the suction requirements of the pumps, station valves, and fittings losses. The estimate may be revised once specific pumps at a given station—station piping and control discharge head and the associated suction head requirements—are determined. Subsequent pump stations are located similarly with the use of a parallel-rule drafting tool to draw the gradients.

Alternatively, in more complicated terrain, pump-station locations may be determined beginning at the initial point and the MAOH, then progressing downstream using the same graphical procedure. Using this procedure, the station is located at the elevation required for the incoming hydraulic gradient above the ground profile.

While it is not required that the pump stations along a pipeline system be equal in energy, or discharge head, doing so results in a better operating environment. Other considerations may preclude a pure balance between the stations. For example, it may be necessary to fix or shift the location of intermediate pump stations due to unsuitable terrain, population and infrastructure criteria, or operations and maintenance considerations. Many designs, especially in rugged terrain or populated regions, may involve fixing certain key stations in accessible locations, i.e., near power lines, then locating other stations by the trial and error method.

The graphical technique described here allows leeway to make these adjustments. A considerable amount of adjustment can be made with little effect on the total hydraulic design, with respect to pipe diameter or total pump station power. However, moving stations to accommodate unequal station differentials may require using heavier wall pipe on the discharge of some stations. Figures C5.8 and C5.9 illustrate using 0.312-in (7.9 mm) wall pipe for the NPS 20 (DN 500) and NPS 24 (DN 600) systems at the discharge of station 1 to accommodate the hydraulic gradients that exceed the MAOH of 0.250-in (6.4 mm) wall pipe used farther downstream. In this case, the total requirement for heavier wall pipe to avoid a fractional station is installed at the initial station rather than distributed in shorter sections to the discharge of the remaining stations.

In the cases illustrated, using 0.312-in (7.9 mm) wall pipe increases the friction loss between station 1 and station 2, and the pumping head at station 1.

Example C5.8: Incremental Friction Loss

Nominal pipe size	NPS 20 (DN 500)	NPS 24 (DN 600)
Pipe diameter, in (mm)	20.0(508.0)	24.0(610.0)
Wall thickness, in (mm)	0.250(6.4)0.312(7.9)	0.250(6.4)0.312(7.9)
Friction loss, ft/mi	44.06 45.43	18.09 18.55
(m/km)	(8.35) (8.58)	(3.42) (3.50)
Incremental friction, ft/mi	45.43 − 44.06 = 1.37	18.55 − 18.09 = 0.46
due to increase of		
wall thickness, (m/km)	8.58 − 8.35 = 0.23	3.50 − 3.42 = 0.08
Length of pipe, mi (km)	19.3(31)	24.9(40)
Total incremental friction		
loss, ft(m)	26.4(7.1)	11.5(3.2)

In cases where the hydraulic gradient crosses deep defiles in the terrain, heavier wall pipe may be required to prevent exceeding the MAOH at these locations. Increased friction loss is determined for these locations similar to this example, or by accumulating the distance and friction loss for each of the selected wall thicknesses.

System Growth and Station Bypass. Many pipeline systems are designed to operate, initially, at a reduced flow rate, with increments added in subsequent years. The design basis for these systems is usually the economic-maximum future flow rate, considering oil characteristics, pipe diameter, wall thickness, and grade of steel.

During the early years of operation, only the stations necessary to transport the initial flow rate are installed. Stations are then added as flow rate is increased until the pipeline is fully developed. Intermediate pumping stations are installed at locations determined from the design for the maximum flow rate. As part of the system design, each station, except the first, is omitted from the system to establish the maximum flow rate possible through the system for that configuration. In other words, each omitted station creates a bottleneck in the system at some maximum flow rate.

Example C5.9. The bypass operating conditions for the NPS 20 (DN 500) and NPS 24 (DN 600) pipeline systems are shown in Fig. C5.10 and C5.11.

The hydraulic gradient bypassing a single station is drawn, and the unit head loss determined, graphically or numerically. Then, using the unit friction-loss graph constructed earlier for the specific crude oil transported in the system, Fig. C5.5, the limiting flow rate is found. In Fig. C5.10, the section between stations 1 and 3 is a bottleneck section that limits the flow rate to 116,800 BPOD (20,800 m^3/d.)

Designing for bypassing station 2 in the NPS 20 (DN 500) and NPS 24 (DN 600) systems would use 19 and 24 (nominal) miles (31 and 40 km) of 0.312-in (7.9 mm) wall pipe respectively, as illustrated in Figs. C5.10 and C5.11. Pipe tonnage would be adjusted upward as shown in Tables C5.8 and C5.8M to

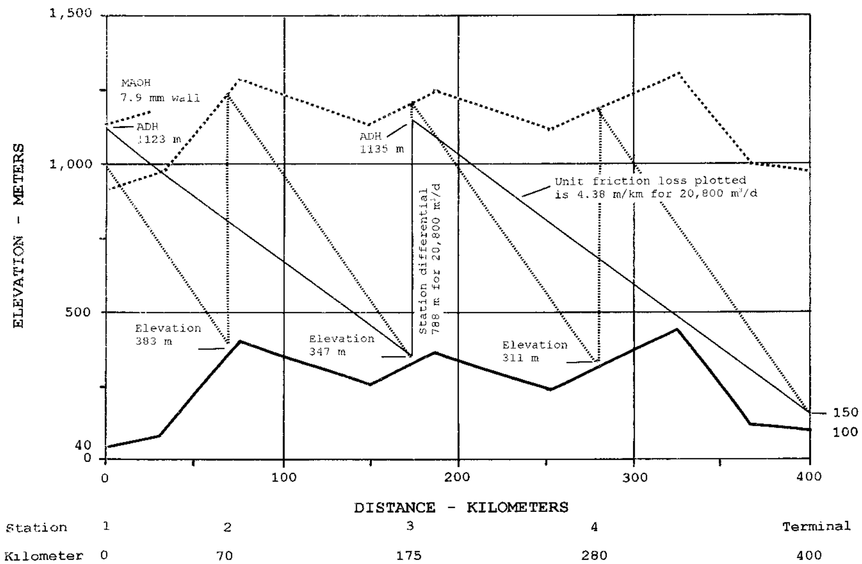


FIGURE C5.10 Hydraulic gradient for DN 500 pipeline example, 20,800 m^3/d with bypass at stations 2 and 4.

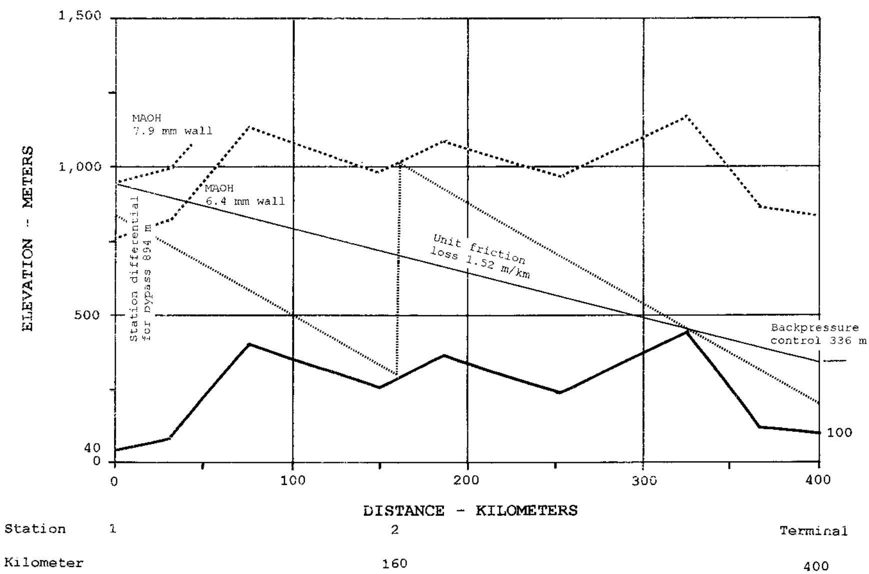


FIGURE C5.11 Example hydraulic gradient for DN 600 pipeline, 18,900 m^3/d with bypass at station 2 and profile control point at kilometer 325.

reflect the effect of ground-elevation profile on the mechanical design and cost. Additional heavier wall pipe would be inserted in other locations where the MAOH would lie below the hydraulic gradient to accumulate the appropriate pipe tonnage. Additionally, the increasing wall thickness adds to the friction loss. In this case, the added friction loss would appear at station 1 and increase the pumping power at this station.

Figures C5.9 and C5.11 for the NPS24 (DN 600) pipeline illustrate a hydraulic control point establishing the elevation of the hydraulic gradient out of a station. The elevation added to clear a hydraulic control point adds to the static head difference for pumping. In Fig. C5.9, the added head for the design flow rate shifted the intermediate pump station closer to the hydraulic control point than for a level line. The added head was then moved to station 1 by limiting the discharge head at station 2 to the MAOP of NPS 20 (DN 500) wall pipe.

In this case, 0.312-in (7.9 mm) wall placed at station 1 for the design flow rate is available for increasing flow rate when bypassing station 2. The incremental length of 0.312-in (7.9 mm) wall pipe at station 1 to bypass station 2 is 25 mi (40 km) to utilize the available MAOH of 0.312-in (7.9 mm) wall pipe at station 1, which is illustrated in Fig. C5.11 Placing any length of 0.312-in (7.9 mm) wall pipe at station 2 would not be available for improving the flow rate to bypass station 2.

Depending upon the ground profile and the length of the system, bypassing one station may also allow bypassing other stations and redistributing units of pumping energy among the remaining stations, as illustrated for the NPS 20 (DN 500) system in Fig. C5.10. This same analysis can also be used to establish the logical growth pattern for installation of intermediate stations.

Once the logical growth pattern has been established, and the hydraulic gradients plotted for each stage of growth, it is possible to identify sections of the pipeline where different wall thicknesses, material with higher yield strength, or reductions in line diameter may be economical. When the pipeline system is fully commissioned and operating, this design becomes the operating design for the system when any station is bypassed.

System Curves. The basis for selecting the actual type and size of pump unit(s) at a station is aided by plotting the resistance curve for the section the station must deliver energy to and superimposing it on the characteristic operating, or performance, curve of the pump(s) being considered.

The system-resistance curve is constructed using the unit head loss curve, such as Fig. C5.5. For several flow rates, h_f is multiplied by the distance to the next station, resulting in the friction loss in that section of pipeline for each of the several flow rates. A graph can be constructed this way for a range of operating flow rates. Superimposed on this is the head required to overcome elevation changes between two stations, or the elevation of a critical profile point, plus an allowance for station losses resulting from flow-through fittings, control valves, et cetera. The resulting curve is the system-resistance curve and is concave upward, i.e, the total friction head increases at a faster rate than flow rate increases.

Example C5.10. Figure C5.12 is the system-resistance curve for Station 1 of the NPS 20-(DN 500) diameter example.

A pump unit of a given model and size can deliver a specific quantity of flow at a given discharge pressure or head. Pump manufacturers publish this information in the form of head-capacity curves, otherwise known as pump-performance curves. Performance curves for centrifugal pumps are characteristically concave downward, i.e., delivered head decreases as capacity increases for a given model.

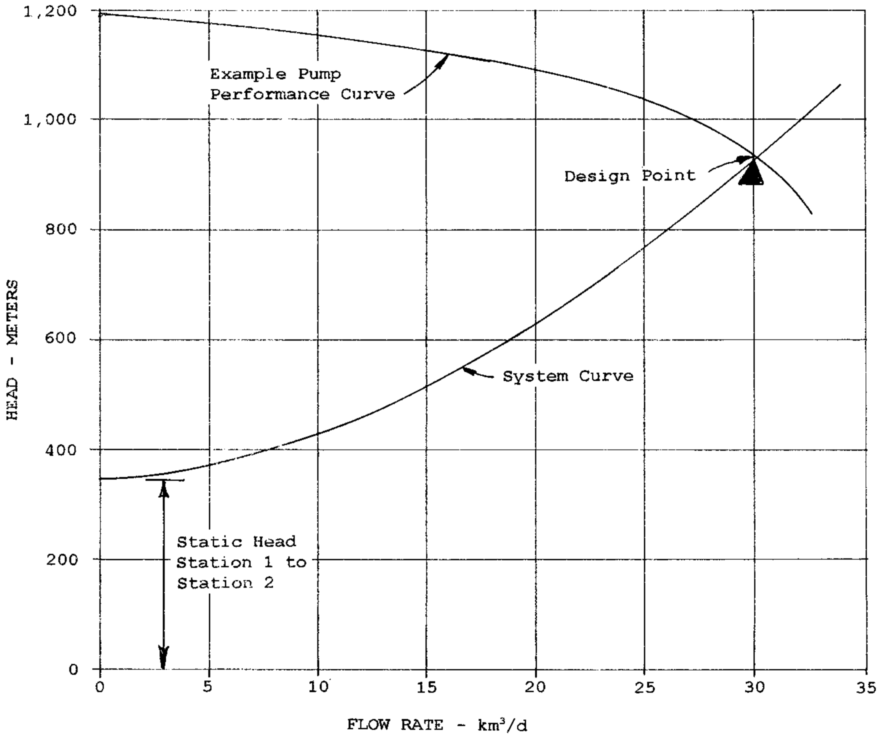


FIGURE C5.12 System curve for Station 1 with example pump-performance curve.

Superimposing a pump-performance curve on the system-resistance curve results in an intersection of the two curves, which is the operating point of the pump units at the station. The configuration of multiple pumps at a station is also considered. For example, pumps in parallel result in increased capacity without a similar increase in head delivered. Pumps in series yield increased head at a given capacity, which is roughly additive.[13,14] Therefore, the net performance curve of the combination of pumps at the station and the combination of pump units that satisfies the required pumping head and power requirements most efficiently are selected for that station.

Nonisothermal Systems[15,16]

Up to this point, the discussion of hydraulic design has focused on isothermal pipeline systems. However, in general, the physical properties of petroleum commodities are temperature dependent, and the significance of temperature change varies with the commodity. Therefore, the true nature of liquid petroleum systems is nonisothermal.

Temperature Profiles. The basis for nonisothermal hydraulic design is an analysis of the thermal nature of the system, considering variations of external, or environmental, temperature as well as systemic factors, i.e., flow rate and its effects on friction, including heat generated by friction. The analysis of a particular system's hydraulic design is based on an analysis of the temperature profile for the range of operating conditions.

The temperature profile can be calculated by the following equation (holds for any set of consistent units):

$$T(x) = T(a) + [T(0) - T(a)] \times e^{[-(uAx)/(wcv)]} \tag{C5.10}$$

where $T(x)$ = temperature at distance x along the pipeline, °F (°C)
$T(a)$ = ambient temperature, °F (°C)
$T(0)$ = temperature at the initial point, °F (°C)
u = overall heat transfer coefficient per unit of area, Btu/ (hr-ft-°F) (W/m$^2 \cdot$ °C)
A = pipe surface area per unit of length, ft^2/ft (m^2/m)
x = distance from initial point, ft (m)
w = weight of fluid per unit of length, lb$_f$/ft (kg$_f$/m)
c = specific heat of fluid, Btu/lb$_f \cdot$ °F (J/kg$_f \cdot$ °C)
v = velocity of fluid, ft/s (m/s)
J = joule, N · m
W = watt, J/s

Hydraulic designs for heavier commodities, such as heavy crude oils and residual fuel oils, are influenced largely by the effect of temperature on viscosity and related friction losses. For lighter commodities, such as gasoline and natural gas liquids (NGL), hydraulic friction losses diminish continuously with declining flow rate, but for some heavy commodities, continuing reduction of flow rate may cause an increase in pump-station discharge pressure to overcome the fluid's resistance to motion by increasing viscosity. Hydraulic design for the lighter commodities, although relatively independent of viscosity, is more dependent on vapor pressure, which is also a function of temperature.

In the discussion of isothermal oil systems, Fig. C5.5 illustrated that at constant flow rate, unit friction loss increases with declining temperature, and that at constant temperature, unit friction loss reduces with declining flow rate. In nonisothermal systems, temperature typically decreases at increasing distance from a heating station, increasing the unit friction in the direction of flow, which is illustrated in Fig. C5.13. Generally, total friction loss in a hotoil pipeline decreases with declining flow rate as in isothermal systems. However, for some commodities, friction loss reaches a minimum, then increases at lower flow rates. Figure C5.14 shows that pipeline friction for 113,200 BPD (18,000 m^3/d) lies above the pipeline friction for 150,950 BPD (24,000 m^3/d) at distances greater than 53 mi (85 km), and above 188,700 BPD (30,000 m^3/d) at distances greater than 65 mi (105 km). Figure C5.14 includes the beneficial effect of heat of friction. Friction loss would be greater if heat of friction were ignored.

Heat of Friction. In a flowing fluid, the pressure dissipated by friction becomes heat. The effect of friction heating generally increases with flow rate, viscosity, insulation, and line length. Heat of friction should be considered at high flow rates to assure that overheating will not occur. Pipeline insulation to reduce heat loss during cold weather may contribute to overheating in summer.

When heat of friction is considered, the limit of cooling while the line is operating can be obtained by adding the temperature offset by heating to the ambient tempera-

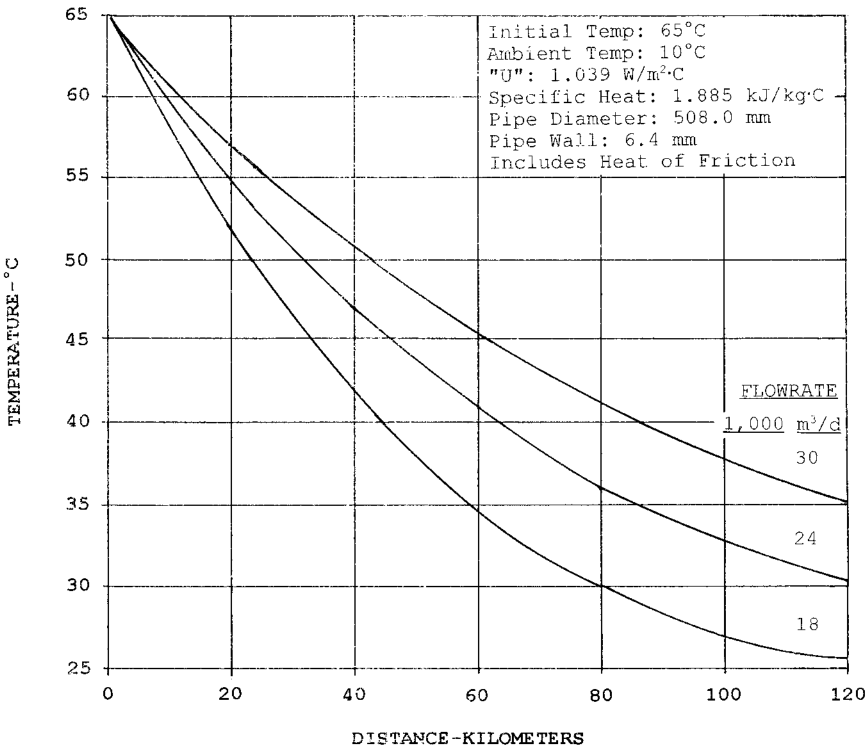


FIGURE C5.13 Example temperature profiles for DN 500 hot-oil system, 18-, 24-, and 30-thousand m^3/d.

ture. The temperature offset is calculated by solving the heat-flow equation for temperature difference where the heat flow equals the frictional heat generated by a unit length of fluid flowing in the pipe:

$$(T - T_A) = Q/(u \times A) \tag{C5.11}$$

where Q = unit heat of friction, Btu/ (h · ft) (J/s · m)
u = heat transfer coefficient per unit of area, Btu/ (h · ft² · °F) (W/m² · °C)
A = pipe surface area per unit of length, ft²/ft (m²/m)
T = flowing temperature, °F (°C)
T_A = ambient temperature, °F (°C)
J = joule, N · m
W = watt, J/s

The heat of friction, Q, can be calculated by converting the unit friction loss to heat, and multiplying by the weight of fluid per unit length and velocity in m/sec.

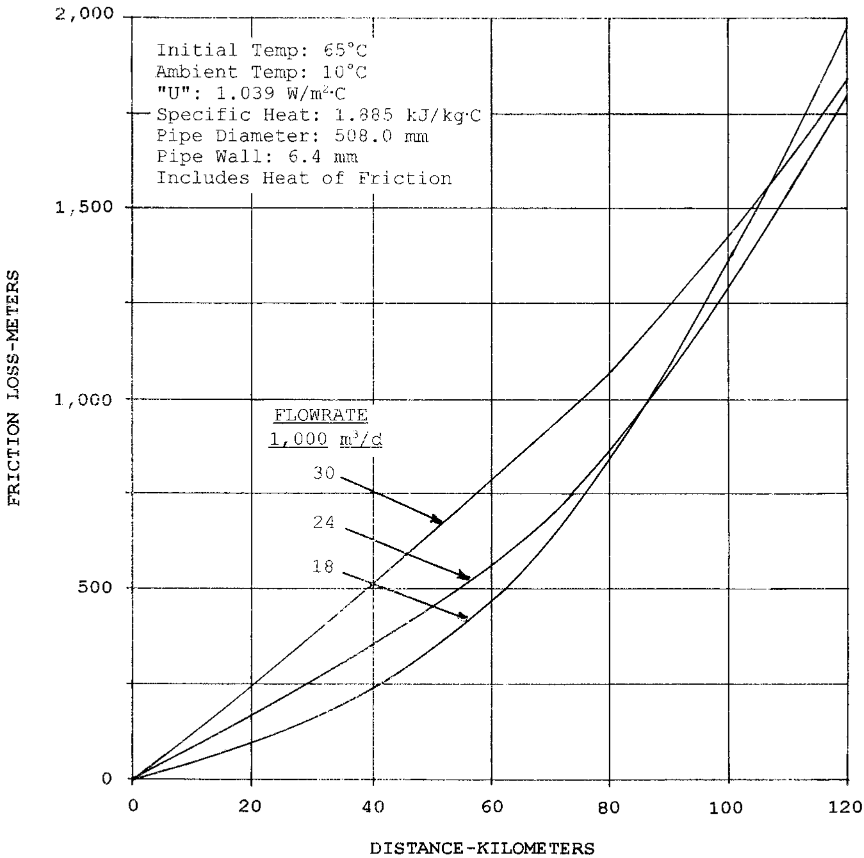


FIGURE C5.14 Example DN 500 pipeline friction versus temperature, 18-, 24-, and 30-thousand cubic meters per day.

For large diameter pipelines and high flow rates, heat generated by friction loss should also be included in the temperature profile, and the following equation results:

$$T(x) = [T(a) + Q/uA] + \{T(a) - [T(a) + Q/uA]\}e^{[-(uAx)/(wcv)]} \quad \text{(C5.12)}$$

where Q = heat generated per linear unit of flowing liquid in a unit of time by converting the unit friction loss to thermal units as previously described.

Example C5.11. Figure C5.13 shows calculated temperature profiles versus distance, including heat of friction, for a 0.976-specific gravity- (976.0 kg_f/m^3) density crude oil, flowing between 113,200 BPD (18,000 m^3/d) and 188,700 BPD (30,000 m^3/d) in a NPS 20-(DN 500) diameter pipeline.

Figure C5.14 shows friction loss as a function of the distance from a pumping and heating station, with an initial temperature of 149°F (65°C) to correspond with the temperature profiles of Fig. C5.13.

Thermal-Hydraulic Gradients

For initial hydraulic designs, once the temperature profile is determined, the system can be divided into discrete sections and the thermal-hydraulic gradient calculated for appropriate sections using the average viscosity of the successive sections. Station locations and sizes are then determined in the manner described for the isothermal case.

For a more detailed hydraulic design, it is recognized that the hydraulic gradients of nonisothermal systems are curves of increasing friction loss as a result of cooling, as the distance from the pumping and heating station increases, unless heat of friction is a significant factor, such as in a large diameter line at high flow rates and with a low initial temperature, wherein temperature may actually increase with distance from the station.

Example C5.12. Table C5.4M and Fig. C5.15 illustrate the hydraulic gradients of a pipeline system for 0.976 sg–(976.0 kg_f/m^3) density crude oil along the given route, for three flow rates—113,200 BPD (18,000 m^3/d), 150,950 BPD (24,000 m^3/d), and 188,700 BPD (30,000 m^3/d). The discharge temperature at each station is 149°F (65°C), and the ground temperature is 50°F (10°C).

Table C5.4M illustrates the effect of increasing distance between heating stations by comparing the difference in friction loss between Station 1 and the remaining stations as flow-rate increases.

This example illustrates, by comparison with Fig. C5.8, the effect of characteristics between a 0.8701 and 0.976-relative-density (32.6 and 13.5 API gravity) crude oil on the hydraulic, mechanical, and operating designs that differ between an isothermal and nonisothermal system for a given pipeline route.

The hydraulic and mechanical designs of nonisothermal systems are more detailed than systems within the isothermal range at usual ambient temperatures including the additional effects of temperature on stresses and of materials and components for the operating conditions. Mechanical design is discussed in the following section.

Intermediate pumping and pumping and heating stations would be investigated in this example to determine the effect increasing the average flowing temperature would have on reducing average viscosity and the total friction loss, and on establishing the required wall thickness between pumping stations. Further analysis would also include increasing the temperature of the heated crude leaving the stations and insulating the pipeline and analyzing other pipe diameters.

With some commodities such as heavy and waxy crudes, the effect of friction heating can be significant, and it results in a decrease in the energy required for pumping. The decrease in pumping energy is a factor of how much the viscosity decreases by increasing temperature and how sensitive the flow regime is to changes in viscosity, i.e., pressure drop in laminar flow is a stronger function of viscosity than in transition flow and is a relatively insignificant function of viscosity in the fully turbulent flow regime.

Shutdown and Restart. During periods when flow ceases, fluid in the pipeline cools statically without the heat of friction until flow resumes, or the system reaches

TABLE C5.4 Example: Hot Oil System Station Discharge Heads, NPS 20 System

	Unit	Station 1 at origin			Station 2 MP 46.6			Station 3 MP 108.7			Station 4 MP 174.0		
Flow rate	MBPD	113	151	189	113	151	189	113	151	189	113	151	189
Friction loss	feet	2,067	2,677	4,301	4,451	4,580	4,961	4,455	4,580	4,967	6,522	5,892	6,066
Head at inlet downstream	feet	1,257	1,257	1,257	1,138	1,138	1,138	1,020	1,020	1,020	492	492	492
Head out of station	feet	3,323	3,524	4,301	5,594	5,719	6,106	5,476	5,600	5,988	7,014	6,385	6,558
Station elevation	feet	131	131	131	1,175	1,175	1,175	1,056	1,056	1,056	938	938	938
Suction loss + NPSH	feet	82	82	82	82	82	82	82	82	82	82	82	82
Station differential	feet	3,310	3,310	4,088	4,337	4,462	4,849	4,337	4,462	4,849	5,994	5,364	5,538
Station loss	feet	82	82	82	82	82	82	82	82	82	82	82	82
Pumping head	feet	3,192	3,392	4,170	4,419	4,544	4,931	4,419	4,544	4,931	6,076	5,446	5,620

TABLE C5.4M Example: Hot Oil System Station Discharge Heads, DN 500 System

	Unit	Station 1 at origin			Station 2 km 75			Station 3 175			Station 4 km 280		
Flow rate	1,000 m^3/d	18	24	30	18	24	30	18	24	30	18	24	30
Friction loss	meters	630	691	1,311	1,358	1,396	1,514	1,358	1,396	1,514	1,988	1,796	1,849
Head at inlet downstream	meters	383	383	383	347	347	347	311	311	311	150	150	150
Head out of station	meters	1,013	1,074	1,311	1,705	1,743	1,861	1,669	1,707	1,825	2,138	1,946	1,999
Station elevation	meters	40	40	40	358	358	358	322	322	322	286	286	286
Suction loss + NPSH	meters	25	25	25	25	25	25	25	25	25	25	25	25
Station differential	meters	948	1,009	1,246	1,322	1,360	1,478	1,322	1,360	1,478	1,827	1,635	1,688
Station loss	meters	25	25	25	25	25	25	25	25	25	25	25	25
Pumping head	meters	973	1,034	1,271	1,347	1,385	1,503	1,347	1,385	1,503	1,852	1,660	1,713

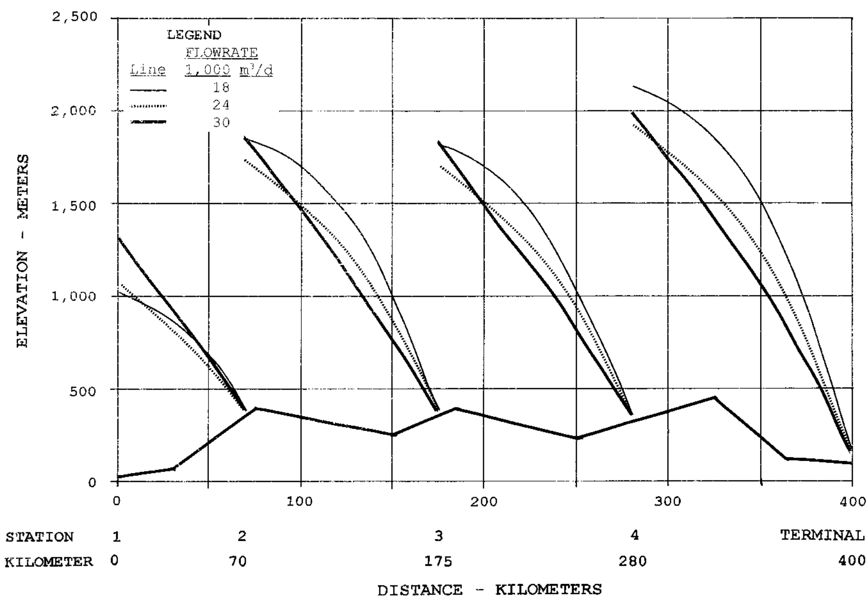


FIGURE C5.15 Example plot of DN 500 hot-oil pipeline hydraulic gradients for 18, 24, and 30 thousand m^3/d.

the ambient temperature. During static cooling, the temperature may be calculated by

$$T(t) = T(a) + [T(0) - T(a)] \times e^{-(uAt)/(wc)} \tag{C5.13}$$

where $T(t)$ = temperature at time t, °F (°C)
$T(a)$ = ambient temperature, °F (°C)
$T(0)$ = temperature at the initial point, °F (°C)
t = time from start of static cooling, s
e = base of natural logarithms
u = heat transfer coefficient, Btu/h · ft² · °F (W/m² · °C)
w = weight of fluid per unit length, lb_f/ft (kg_f/m)
c = specific heat, Btu/lb_f · °F (J/kg_f · °C)

On restart, the viscosity, determined by the local temperature, determines the local friction loss. The restart flow rate and pressure may be determined by dividing the pipeline into segments, then summing the friction losses of the segments. Repeating this process stepwise as sequential segments are displaced with heated oil determines the calculated startup pumping rate and restart time. Waxy crudes may develop a yield stress on cooling, which may require additional pressure to reestablish flow.

Figure C5.14 showed that a flow rate can be reached where friction loss can increase for decreasing flow rates. At lower flow rates, where friction loss increases, the friction loss may increase beyond the head capacity of the pumps or the pressure rating of the pipeline leaving the station. This condition would plug the pipeline and must be avoided. Methods of preventing plugging a pipeline that cools below the limit of pumpability, either by low flow rate or static cooling include:

- maintaining the flow rate above the temperature at which the viscosity effect increases friction loss
- blending with a lighter oil having a lower pour point (a cutter stock) to reduce the viscosity at lower temperatures
- displacing with a displacement fluid before static cooling
- insulating and heat tracing

Pipelines using the displacement method generally require preheating before refilling the line with the commodity. These systems are preheated by circulating heated displacement fluid until the system is sufficiently warm to accept the heated commodity and allow time to redisplace if a false start should occur during the refill and restart operation.

MECHANICAL DESIGN

Mechanical design of a pipeline system is the selection of materials, including type of steel and diameter and wall thickness of pipe. Selected also are methods of support and restraint for the system in response to the loadings and stresses imposed on the pipeline system by physical pressures and forces such as the internal and external design pressures; static loadings and weight effects of the pipe, fluid, and soil; dynamic loadings from wind, waves, earthquake, and other natural forces; and relative motion of connected components. These factors impose loadings on the pipe and result in longitudinal, hoop, and radial stresses which must be evaluated in the mechanical design of the piping system.

In addition to the mechanical factors that affect the allowable stress levels for design, the grade of steel and wall thickness determine welding procedures and affect construction cost. Section 434.8 of the Code prescribes the requirements for welding. Section 434.8.3 specifies welding qualifications with reference to API 1104 and Sec. IX of the ASME Boiler and Pressure Vessel Code. The engineer should refer to these references for welding requirements that may determine the maximum diameter, wall thickness, or grade steel for the system on the basis of cost.

The mechanical design of the pipeline, with respect to restraint against longitudinal, or axial, and radial motions, considers the pipeline as a unit and must provide sufficient flexibility in the system to ensure that expansion or contraction, as a result of the internal or external loadings, do not cause excessive stresses in the piping material, bending moments at joints, or excessive forces or moments at points of connection to equipment or at supports.

Mechanical piping-system design primarily utilizes computer programs, many of which operate on personal computers. In most cases, the code requirements are built into the programs, so for a set of internal pressures and external loadings, the program will give the optimum wall thickness, pump locations, and maximum stress values on the basis of parameters input by the engineer. However, this does not preclude the possibility of error. The engineer must be able to determine accurately the required loadings and pressure and to analyze the computer results, verifying their validity to the overall system.

Line Pipe, Fittings, and Valves

Specifications for line pipe and for fittings, valves, and flanges are given in various API, ANSI, and ASME standards and specifications including:

- ANSI/ASME B36.10M, Welded and Seamless Steel Pipe
- ANSI/ASME B36.19M, Stainless Steel Pipe
- API 5L, 5LU, Line Pipe
- ANSI B16.5, B16.9, B16.10, B16.11, B16.25, B16.28, Flanges, Fittings, Valves
- API 6D Pipeline Valves, Gate, Plug, Ball, and Check Valves
- API 600, 602, 603, Valves

Additional information on components of piping can be found in Chap. A2, Piping Components. Chapter A10 deals with selection and application of valves.

Line pipe is manufactured by several methods, the most common being seamless (SMLS), electric resistance welded (ERW), and submerged arc welded (SAW) in the form of longitudinal and helical (spiral) welds. Table C5.5 summarizes some of the characteristics for these four types of pipe manufacture. Each has advantages and disadvantages for different uses, and there are also economic and availability considerations which enter into the decision on the type specified and supplied on a particular project.

With respect to the mechanical design of a pipeline, the characteristic of line pipe which is of critical interest is the specified minimum yield strength (SMYS) of the material. API 5L, Specifications for Line Pipe, is available in various strength grades, ranging from Grade B, rated at 35,000 psi (241 MPa) to Grade X80, rated at 80,000 psi (551 MPa), where the Grade X80 refers to the SMYS in ksi, which is defined as kips per square inch, a kip being 1,000 pounds.

There is some advantage to the higher strength grades, principally that wall thickness may be reduced. In some cases this may have an economic impact on the project, as thinner walls translate into lower steel tonnage for the entire pipeline system, and this may be a significant factor, even though higher grades of pipe cost more per ton. Cost savings can also result from reduced time required to field weld the thinner wall sections. There are other considerations which will affect the decision to use higher strength, thinner wall pipe. These include aspects of construction which are the result of experience in the field, such as the way the pipe handles with regard to field bending, laying stresses, tendency to go *out of round*, et cetera. In addition, there may be limitations placed on the grade of pipe and wall thickness used for a particular project, particularly for a system which will be in *sour* or corrosive service.

Valves, pipe flanges, and fittings are described by class ratings, which are not the same as the rated pressure for a particular class. For example, API 6D, Specification for Pipeline Valves, Class 300 (PN 50) valves, standard flanged end and standard weld end, have a nominal pressure rating of 720 psig at −20 to 100 F° (−29 to 38°C), not 300 psig (91.2MPa). Metric designations, PN for nominal pressure and DN for nominal pipe size replace class and nominal pipe diameter respectively. Refer to Chaps. A1 and A10.

However, API 6D, App. B, states that English units are preferred and are the standard. Section 402.2.1 of the Code specifies for pressure-temperature ratings of piping components as follows: ‘‘Within the metal temperature limits of −20°F (−30°C) to 250°F (120°C), pressure ratings for components shall conform to those stated for 100°F (40°C) in material standards listed in Table 423.1.’’

TABLE C5.5 Availability and Usage for Types of Line Pipe

	Seamless	ERW	SAW, longitudinal welds	SAW, helical (spiral) welds
Minimum diameter	2.375 in (DN 60) or less.	2.375 in (DN 60) or less.	NPS 16 to 20 (DN 400 to 500).	NPS 16 (DN 400).
Maximum diameter	NPS 16 to 26 possible. (DN 40 to 650 possible.)	NPS 24 to 26 possible. (DN 600 to 650 possible.)	64 to 84 in. (DN 1,600 to DN 2,100.)	80 to 100 + in. (DN 2,000 to DN 2,500.)
Maximum wall thickness	0.75 to 2.00 in (19.1 to 51 mm).	0.312 to 0.750 in (7.9 to 19.1 mm).	0.625 to 1.500 in (15.9 to 38.1 mm).	0.500 to 1.500 in (12.7 to 38.1 mm).
Grades	B through X-80. Grades X-60 and higher are heat treated.	B through X-70. X-52 and higher are made from controlled rolled skelp.	B through X-80. X-52 and higher are made from controlled rolled plate.	B through X-70. X-52 and higher are made from controlled rolled skelp.
Service	All services: on and off shore.	All services: generally not used for offshore.	All services: on and offshore.	Experience limited to less critical service; used as equivalent to SAW in other countries.
Relative cost	More expensive than ERW. Cost premium for larger diameters/wall thickness, higher grades.	Less expensive than seamless.	Less expensive than seamless, more than ERW (within size overlap range).	May be less expensive than SAW (depends on manufacturer).

TABLE C5.6 Pressure Ratings of Valves

	Class					
	150	300	400	600	900	1500
Temperature °F	Pressure rating, psig					
−20 to 100	275	720	960	1440	2160	3600
150	270	705	940	1415	2120	3540
200	260	675	900	1350	2025	3375
250	255	665	885	1330	1995	3325

Interpolation is permitted for intermediate temperatures.

TABLE C5.6M Pressure Ratings of Valves

Class (PN)	150 20	300 50	400 68	600 100	900 150	1500 250	2500 420
Temperature °C	Rating, bar						
−29 to 38	19.0	49.6	66.2	99.3	149	248	414
50	18.8	49.2	65.6	98.5	148	246	411
66	18.6	48.6	64.8	97.6	146	244	407
75	18.4	47.9	63.9	96.0	144	240	400
93	17.9	46.5	62.1	93.1	140	233	388
100	17.7	46.4	61.8	92.7	139	232	387
121	16.9	45.9	61.0	91.7	138	229	382

For temperatures below −29°C the rating shall be not greater than rating for −29°C.

Few pipelines now in operation utilize fittings heavier than Class 600 (PN 100), rated at 1,440 psig (9,930 kPa), although the trend is toward higher pressure ratings. Deep-water subsea pipelines which have heavier wall thickness due to laying stresses and high external pressures may have fittings and flanges rated at Class 1,500 (PN 250), rated at 3,600 psi (24,000kPa). Tables C5.6 and C5.6M list maximum operating temperatures and pressures for standard flanged-end and standard weld-end valves by nominal pressure rating, Class (PN), from API 6D, App. B, Table B2.1.

Allowable Pipe Stress

Paragraph 402.3.1 of the Code establishes the allowable stress value, S_A in psi (MPa), to be used in the temperature range −20°F to 250°F (−30°C to 120°C) for design calculations:

$$S_A = \text{design factor} \times \text{SMYS} \times \text{joint weld factor} \qquad \text{(C5.14)}$$

Paragraph 402.3.1 describes application of Eq. C5-11 for allowable stress values of S_A depending on new, used, or reclaimed straight pipe of known or unknown specification. Sub paragraphs 402.3.1 (a) through 402.3.1 (d) are summarized below.

Reference paragraph	Application for straight pipe	Formulations and reference paragraphs
402.3.1(a)	New pipe of known specification	$S_A = 0.72 \times E \times$ SMYS psi(MPa) where E = joint factor, Table 402.3.1(a).
402.3.1(b)	Used or reclaimed pipe of known specification	$S_A = 0.72 \times E \times$ SMYS psi(MPa) in accordance with paragraph 405.2.1(b), Table 423.1, and limitations in paragraphs 437.4.1, 437.6.1, 437.6.3, and 437.6.4.
402.3.1(c)	New, used, or reclaimed pipe of unknown or ASTM A 120 specification	$S_A = 0.72 \times E \times$ minimum yield strength of pipe, psi (MPa) [24,000 psi (165 Mpa) or yield strength determined in accordance with paragraphs 437.6.6 and 437.6.7], and limitations in paragraph 405.2.1(c).
402.3.1(d)	Pipe cold worked in order to meet specified minimum yield strength and subsequently heated to 600°F (300°C) or more except welding.	S_A = 75% of applicable allowable stress value as determined by paragraphs 402.3.1(a), (b), or (c) for design calculations of wall thickness in paragraph 404.1.2 (see Eqs. C5.12 and C5.12M).

Table 402.3.1(a) of the Code tabulates allowable stress values for various grades of material specifications and manufacturing processes.

Specified minimum yield strength of a pipe material is used as the basis of design because it is a property which can be determined for a specific material. Furthermore, steel generally behaves elastically below this stress level. Ultimate yield strength has also been used as the basis for design and may still be used in other countries.

A particular batch of steel line pipe may be tested and determined that its elastic yield strength is higher than the nominal value; however, Section 402.3.1 (g) of the Code specifies, ''In no case where the Code refers to the specified minimum value of a physical property shall a higher value of the property be used in establishing the allowable stress value.''

Table 402.3.1 (a) also includes the joint weld factor E in calculating allowable stress as a consideration to the way the line pipe is manufactured. In most cases, the joint weld factor is 1.00, but it may be 0.80 or 0.60 for specific grades of steel and welding method of the manufacturing process.

It is useful at this point to clarify a question that arises periodically as to whether the mechanical stress calculations are based on a particular design temperature, i.e., if S_A is temperature dependent. Section 401.3.1 of the Code states ''it is not necessary to vary allowable design stress for metal temperatures between $-20°F$ ($-30°C$) and 250°F (120°C).'' However, for applications where ground or air temperature is expected to be extremely low, seasonally or locally, the properties of pipe-component materials at low temperatures should be considered to verify that the design will be adequate.

Allowable stress limits for shear and bearing are given in Sec. 402.3.1(e) of the Code. For shear, S_A shall not exceed 45 percent of SMYS; for bearing, S_A is limited to 90 percent of SMYS. Limits on calculated stresses due to sustained loads and thermal expansion and due to occasional loads in operation and test conditions are specified in Sec. 402.3.2 and 402.3.3 of the Code. In general, these limits fall within the definition of allowable stress given previously; however, special circumstances may apply, and the engineer is encouraged to verify the stresses and relevant limitations.

Pipe Diameter

In the hydraulic design of a pipeline system, line size is initially based on a preliminary choice of diameter and wall thickness from experience and from simplified charts. Further calculations are needed to verify the selection and finalize the system design based on the Code requirements as well as on considerations for project cost and material availability.

For most pipeline systems, the pipe cost, which is based on the diameter and wall thickness, will be the highest material cost in the system. In addition, the size of pipe will have a direct effect on the cost of installation. Therefore, total project cost is impacted by the selection of pipe size. For this reason, it is important to optimize the pipe diameter, wall thickness, and grade of steel to be used so that the overall project cost is contained.

As discussed earlier in the hydraulic-design section of this chapter, the diameter of pipe is based on the design flow rate, and mechanical-design considerations have little effect on diameter selection. However, internal and external pressure, allowable stress, and other considerations do affect the final design of the wall thickness for the selected diameter.

Wall Thickness

In the hydraulic design, a preliminary determination of wall thickness is based on experience for the preliminary selection of pipe diameter and grade of steel. The actual design of a system must reflect Code requirements for the wall thickness, which is based on internal design pressure and additional loads at the design temperature.

Internal Design Pressure. By definition, Sect. 401.2.2 of the Code prescribes the following:

> The piping component at any point in the piping system shall be designed for an internal design pressure which shall not be less than the maximum steady state operating pressure at that point, or less than the static head pressure at that point with the line in a static condition. The maximum steady state operating pressure shall be the sum of the static head pressure, pressure required to overcome friction losses, and any required back pressure. Credit may be given for hydrostatic external pressure, in the appropriate manner, in modifying the internal design pressure for use in calculations involving the pressure design of a piping component (see Para. 404.1.3). Pressure rise above maximum steady state operating pressure due to surges and other variations from normal operations is allowed in accordance with para. 402.2.4.

The earlier discussion of MAOP in the hydraulic design defined it as a function of diameter, wall thickness, and allowable stress for a material, per Code restrictions. Section 402.2.4 of the Code provides for an additional allowance of 10 percent over the internal design pressure for surges and other variations from the normal operation.

Additional Loads. Additional loads for determining wall thickness include loadings applied on a pipeline system from pipeline material and commodity weight, wind, hydrostatic, and other external forces such as impact loads. In some applications, external pressure may be a significant factor in pipeline wall thickness determination. One such example is a subsea pipeline. Section 401.2.3 of the Code specifies that a component of the pipeline system shall be designed to withstand the maximum differential between external and internal design pressures.

Wall Thickness Calculation. Minimum wall thickness, t, is a function of internal pressure, P_i, nominal diameter, D, and the allowable stress, S_A, as specified by Sec. 404.1.2 of the Code by solving Eq. C5-4 for wall thickness. Refer to Eq. C5.4 for determination using various pressure units.

$$t = P_i \; D/(2 \; S_A) \text{ English units} \tag{C5.15}$$

$$P_i, \text{ psig}, \; D \text{ and } t, \text{ in}$$

$$t = P_i \; D/(20 \; S_A) \text{ metric units} \tag{C5.15M}$$

$$P_i \text{ gauge pressure, bar}, \; D \text{ and } t, \text{ mm}$$

Nominal wall thickness, t_n, includes an allowance for manufacturing tolerance.

$$t_n = t + \text{allowance}(s) \tag{C5.16}$$

The actual wall thickness used in the system will be equal to or greater than this calculated value. API 5L, Specifications for Line Pipe, Table 6.2, lists commonly manufactured wall thickness for various grades of materials.

With respect to construction and installation of a pipeline, there is also a practical minimum wall thickness, based on handling during installation, as the pipe wall must be able to resist damage and maintain roundness during construction. Good judgement should be used in balancing higher strength steel materials versus heavier wall thickness.

It is not a requirement that wall thickness be constant for the entire length of the system. Specific sections of the system may have different wall-thickness requirements as determined by the internal pressure and other imposed stresses, and making use of the hydraulic gradient developed in the hydraulic design. Thinner wall may be installed at some distance downstream of pumping stations as the operating pressure in the system declines. There are economic benefits to utilizing the minimum wall thickness allowed under Code design; however, there are other considerations, such as complications in construction, i.e., field welding, of a system with frequent variations in wall thickness. Practically speaking, changes in wall thickness should be limited. Furthermore, anticipated growth or expansion of the system capacity should be considered carefully.

Design of Restrained and Unrestrained Pipelines

A pipeline system is subjected to static and dynamic loads due to local environmental and operating conditions, and provision must be made for the system to have flexibility and expansion capability to prevent excessive stresses in the pipe or components, excessive bending or unusual loads at joints, or undesirable forces or moments at points of connection to equipment. The types of loadings which will affect the flexibility and expansion of the pipeline as a system include:

- Thermal expansion and contraction
- Internal pressure
- Fluid expansion
- Pipe and soil–friction interaction
- External pressure
- Bending (sag or uplift) due to

  Dead Loads, including weight of the pipe, coatings, backfill, and unsupported pipe appurtenances

  Live Loads such as liquid transported, wind, snow, earthquake, waves, or currents

The stresses that may develop in the pipe are functions of both the loadings on the pipeline system and the degree of restraint against motion of or in the pipeline system or a section of the system. Stresses may be reduced to acceptable levels by a combination of anchors; extra depth of burial of piping; use of bends, loops, or offsets in the piping; heavier wall-piping components; or other interventions. The reader is referred to Chap. B4, Stress Analysis of Piping Systems, and Chap. B5, Piping Supports, for additional information on stress and supports.

Long, cross-country pipelines are generally buried for the following reasons:

1. Surface use of pipeline corridor
2. Protection from intentional or accidental damage
3. Protection against expansion and contraction from ambient-temperature changes and radiant-energy gains and losses
4. Minimizes variations of ambient temperature and resultant effects on fluid viscosity

5. Provides restraint longitudinally along pipeline length

6. Regulations restrict aboveground installation

Although a buried pipeline can be considered restrained against expansion or contraction radially and longitudinally as a result of the overburden of soil and soil-pipe friction, expansion calculations are necessary, per Sec. 419.6.4 (b) of the Code, if significant temperature changes are expected, as in a heated oil system. Furthermore, thermal expansion of buried lines may cause movement at a transition to an aboveground section and at the termination points of the pipeline or a section of line. There may also be movement where the buried line changes direction, i.e., when there is insufficient soil restraint, therefore either the system should be designed with adequate flexibility in these areas or anchors against motion should be provided.

Pipeline systems may be partially or wholly installed aboveground for reasons of economy of construction, maintenance, et cetera. Likewise, installation on the surface may be practical for pipelines requiring insulation or heat tracing. An aboveground pipeline can be designed with longitudinal restraints at certain support locations such that expansion or contraction due to temperature or pressure changes is absorbed by axial, or longitudinal, compression or tension stress, in addition to radial expansion. The additional consideration in aboveground pipeline systems is that beam-bending stresses in spans between and at supports must be evaluated.

Thermal Expansion and Contraction. Thermal expansion and contraction calculations are necessary for buried and aboveground systems if a substantial temperature change is expected between installation and operation, such as when the line is to carry a hot oil, or where significant variations in local environmental temperatures will occur. Thermal expansion may cause movement where the line changes direction or terminates or where a discontinuity occurs, such as a change in size. In some pipeline systems, these motions may not be restrained by anchors or supports, i.e., absorbed by direct axial stress of the pipe, in which case flexibility must be provided using loops, bends, or expansion joints.

The total maximum range of temperature is used to determine thermal expansion, with the linear coefficient of thermal expansion, α, for carbon and low alloy high tensile steel being 6.5×10^{-6} in/in °F (11.7×10^{-6} mm/mm °C) up to 250°F (120°C). For a section of pipe restrained at both ends, longitudinal stress, $S_{L/T}$, due to a temperature change, ΔT, is given by

$$S_{L/T}, \text{ psi (MPa)} = \mathrm{E}\alpha\,\Delta T \tag{C5.17}$$

where E is the modulus of elasticity for steel 30×10^6 psi (2×10^5 MPa). An increase in temperature results in a compressive longitudinal stress, and vice versa. The general convention is that tensile stress is positive and compressive stress is negative.

Internal Pressure. Internal pressure imposes a radial and longitudinal component of stress in the pipe material. The hoop stress, which is in the radial direction due to internal pressure, was discussed earlier in Eq. C5.4. In a free, or unrestrained, section of pipe material, or pipeline system, as internal pressure increases, the length of a section will decrease due to the Poisson effect.

Restrained systems will develop a component of longitudinal stress as a result of the internal pressure, $S_{L/IP}$, given by

$$S_{L/IP} = \nu\, S_H \quad \text{psi (MPa)}$$
$$= \nu\, P_I\, D/2\,t \tag{C5.18}$$

where ν = Poisson's ratio, 0.30 for carbon steel

This is generally a tensile (positive) stress in the longitudinal direction. For most liquid lines, maximum internal pressure will most likely occur during the hydrostatic test prior to operation.

Fluid Expansion. The expansion of the fluid in a liquid pipeline due to increase in temperature will impose an additional component of pressure which will have both a radial and a longitudinal component, resulting in stress in the pipe material. This stress, $S_{L/E}$, is calculated in the same manner as the stresses from internal pressure after conversion of the expansion of the fluid to equivalent pressure terms.

Pipe and Soil–Friction Interaction. The combined longitudinal effects of temperature, pressure, and expansion of the fluid can cause significant stresses in long sections of pipe. For buried pipelines, the interaction of the pipe material and the soil will have an opposing friction effect, with the maximum stress due to soil friction occurring at the midlength of the section. The friction stress is calculated as follows, with any set of consistent units:[17]

$$S_{L/F} = L \times H \times \rho \times \sigma/(2t) \tag{C5.19}$$

where L = length of pipe section, ft (m)
H = depth of cover above the pipe, ft (mm)
ρ = unit weight of the soil cover, lb_f/ft^3 (kg_f/m^3)
σ = coefficient of friction between pipe and soil
t = wall thickness, in (mm)

External Pressure. For sections of pipe laid underwater, external hydrostatic pressure needs to be considered in the stress analysis. The hoop and longitudinal stresses resulting from hydrostatic pressure are calculated in the same way as the internal pressure stresses are calculated and are opposite in sign. A similar condition arises when thin-walled pipe is drained and a vacuum is produced inside the pipe.

The critical collapse pressure, P_{cr}, for round thin-walled pipe with a diameter-to-wall thickness is calculated by the equation:[18]

$$P_{cr} = 2 \times \mathrm{E} \times (t/D)^3/(1 - \nu^2) \tag{C5.20}$$

where E = modulus of elasticity, 30×10^6 psi (2×10^5 MPa)
ν = Poisson's ratio, 0.3

Eq. C5.20 assumes the stress is within the elastic range and the ratio of length to radius is greater than 20. If the pipe section is out-of-round, the resistance to collapse is reduced by flattening. Stiffening rings or heavier wall pipe may be used to prevent or limit collapse due to external pressure. Collapse of buried pipe can also be controlled by careful preparation of the bedding and compaction of the backfill around the pipe. Additional detail is beyond the purpose here, and the reader should refer to appropriate references or specialists where collapse by external pressure is a concern.

Considering all the longitudinal components of stress discussed to this point, the net longitudinal stress, S_L, imposed on a restrained pipeline due to the combined effects of internal pressure, external pressure, temperature, fluid expansion, and soil and pipe friction is the algebraic sum of Eqs. C5.14, C5.15, and C5.16, where tensile stress is the positive convention:

$$S_L = S_{L/IP} + S_{L/EP} S_{L/T} + S_{L/E} + S_{L/F} \tag{C5.21}$$

where S_L = net longitudinal stress, psi (MPa)
$S_{L/IP}$ = internal pressure, psi (MPa)
$S_{L/EP}$ = external pressure, psi (MPa)
$S_{L/T}$ = thermal expansion/contraction, psi (MPa)
$S_{L/E}$ = fluid expansion, psi (MPa)
$S_{L/F}$ = pipe/soil friction, psi (MPa)

The net longitudinal stress is generally tensile, as the internal-pressure and fluid-expansion components are tensile, countered by the compressive temperature component and by the soil and friction interaction in buried pipeline systems; the external hydrostatic pressure component is only applicable to deepwater crossings. Concentrated loadings at highway and railroad crossings require special consideration and may require casing. Refer to Sec. 434.13.4 of the Code.

It is important to note that in an axially restrained line, if the increase in temperature between placement of the line and operation is great enough, the compressive stress from the restraint to pipe ''growth'' by thermal expansion from Eq. C5.17 will exceed the tensile stress due to internal pressure.

The numerical sum of the longitudinal stress and the hoop stress is called the equivalent tensile stress. In the case where net S_L is negative, i.e., compressive, then the absolute values are used for pipe stresses, and the equivalent tensile stress is the sum of the absolute value of S_H and S_L. Section 419.6.4(b) of the Code specifies that the equivalent tensile stress for restrained lines is not to exceed 90 percent of the SMYS calculated for the nominal pipe wall thickness. Stresses may be reduced by burial, anchors, heavier pipe wall components, or expansion provisions such as loops, offsets, or bends. The following section discusses the beam-bending stresses which are included in longitudinal stress calculations for aboveground portions of restrained as well as unrestrained lines.

Bending or Beam Stresses. For most cases, welded pipelines are very stiff longitudinally and make good beams capable of sustaining typical loadings from the weight of the pipe itself, the commodity within, and any coatings applied. However, a bending-stress analysis is a standard part of pipeline system design, in particular aboveground sections, and is generally carried out by an expert, using one of several computer programs available.

When considering the structural nature of a pipeline system, one of two types of *beam* or structural models can be assumed. A buried pipeline or section of a longer pipeline system can be considered continuously supported as long as the bedding it is laid on is uniform. Even if not perfectly uniform, the stiffness of the pipe will allow reasonable spans across low or soft areas in the soil bedding or rock protrusions.[19]

An aboveground pipeline system and a buried pipeline system with nonuniform bedding can be modeled as a continuous beam system with multiple supports. At each support there will be reactions to the loadings on the pipeline due to pipe material, commodity, and coating weights, as well as loadings from environmental factors such as snow, earthquake, and wind.

Beam-bending stresses resulting from sagging (deflection) are included in the analysis of maximum stresses. This means evaluating the pipeline for different loading cases and combinations of loadings. Section 419.6 of the Code provides for the possibility for bending moments in the plane of the pipe (M_i); transverse to the plane (M_o); and torsionally, i.e., about the axis of the pipe (M_t); with the resultant stress due to expansion from bending for unrestrained pipelines in the expansion stress as follows:

$$S_E = \sqrt{S_b^2 + 4 \times S_t^2} \tag{C5.22}$$

where S_E = stress due to expansion
S_b = equivalent bending stress, psi (MPa)
$S_b = \sqrt{(i_i \times M_i)^2 + (i_o \times M_o)^2}/Z$
S_t = torsional stress, psi (MPa)
$= M_t/2 \times Z$
i_i = stress intensification factor (in-plane)
i_o = stress intensification factor (transverse)
Z = section modulus of pipe, in^3 (mm^3)

For straight pipe, the stress intensification factors (in-plane and transverse) have a value of 1, therefore the S_b component reduces to the flexure formula ($S_b = M/Z$). Stress intensification factors for elbow, miter bends and tee sections of pipe can be found in Fig. 419.6.4© in the ASME B31.4 Code. This figure also includes a correction of pressure (Note 7) to be applied to large-diameter, thin-walled pipe fittings, as pressure can significantly affect the stress intensification and flexibility of these components.[1]

Code Stress Limits.[1] Sections 402.3.2 and 402.3.3 of the Code specify that the calculated stresses due to sustained loads, thermal expansion, and occasional loads are to be limited by:

- Internal pressure stresses—not to exceed S_A as discussed in the Code paragraph 402.3 (see Sec. 4.2 of this chapter).
- External pressure stresses—considered within bounds where the minimum wall thickness has been calculated based on the code formulas.
- Expansion stresses—allowable stress, S_A, values for the equivalent tensile stress for restrained lines is not to exceed 90 percent of SMYS; for unrestrained lines, it shall not exceed 72 percent of SMYS.
- Additive longitudinal stresses—the sum of longitudinal stresses due to pressure, weight, and other external sustained loadings shall not exceed 75 percent of S_A calculated for the expansion cases stated above.
- Additive circumferential stresses—the sum of circumferential stresses due to internal design pressure and external loads is not to exceed allowable stress calculated for internal pressure.
- Longitudinal stresses from occasional loads—the sum of the longitudinal stresses produced by pressure, live and dead loads, and occasional loads such as wind or earthquake (not considered concurrently), shall not exceed 80 percent of SMYS.

Pump Selection

The concept of pump curves and preliminary pump selection was discussed earlier. Expanding on that discussion, it is assumed that the hydraulic and pipe design are essentially complete. The next step is to choose the pumps and drivers. The first decision is what type of driver—diesel, turbine, or electric. This is primarily an economic decision influenced by the availability of fuel and cost of electric power. When electric power is readily available, electric motors provide simple operation, low cost, and low maintenance; where natural gas or other unrefined fuels are available, gas turbines may give low fuel costs; and occasionally a steam turbine or some combination of machines proves the best choice. It may even be possible to draw off commodity from the system, to fuel the pump drivers themselves, for example, with a diesel transport system.

Pumping power requirements for the entire system can be calculated using the following formula, where H_T and P_T refer to the total system head and pressure requirements:

$$\text{Power, hp} = \text{Flow, BPOD} \times H_T,\ \text{ft} \times \text{sg}/(136{,}000 \times \text{eff}) \tag{C5.23}$$

$$= \text{Flow, GPM} \times P_T,\ \text{psi}/(1{,}714 \times \text{eff}) \tag{C5.24}$$

$$\text{power, kW} = \text{Flow, m}^3/d \times H_T,\ \text{m} \times \text{rd}/(8{,}810 \times \text{eff}) \tag{C5.25M}$$

Pump efficiency in percent typically ranges from 70 to 80 percent for centrifugal pumps to 90 percent for reciprocating pumps. Determination of pumping power for individual stations and pumps is similar, using the head or pressure required for the downstream section between one station and the next. As motor power is provided in standardized increments, the power provided by the motor will exceed the required pumping power.

Example. In the example for the NPS 20 (DN 500) pipeline, assuming 82 percent efficiency for the pumps, the required system operating power at the design temperature is:

$$\text{power, hp} = 188{,}700,\ \text{BPOD} \times 10{,}984,\ \text{ft} \times .8701,\ \text{sg}/(136{,}000 \times .82)$$

$$= 16{,}200 \text{ (rounded)}$$

$$\text{power kW} = 30{,}000,\ \text{m}^3/d \times 3{,}450,\ m \times 0.8701,\ rd/(8{,}810 \times 0.82)$$

$$= 12{,}460 \text{ (rounded)}$$

Specification for the main line pumps should stress performance, efficiency and ease of maintenance, because over the life of a pipeline, the cost of fuel and power will be the major operating or annual expense. Therefore, a percent or two in pump or driver efficiency has a major impact. Consideration should also be given to variable speed contrifugal pumps, on the basis of economics, in order to satisfy varying flow and pressure or head requirements over the life of a pipeline system. It may be that the added cost of a variable speed pumping unit is favorable versus controlling station discharge pressure by throttling.

Valve Spacing

The location and spacing of section block valves along an oil pipeline is a matter of design procedure and may be dependent on factors such as the terrain that the pipeline is crossing. In general, valves should be installed at locations where they will contribute to the safe operation of the line and enhance the safety of the system. Section 434.15.2 of the Code provides the details for these considerations.

Typically, valves are installed at the origin and termination of a pipeline and at branch points to provide isolation of a section and to facilitate hydrostatic testing, i.e., anywhere that the test pressure is differentiated such as sections of higher operating pressure or a change in wall thickness. Section block valves should be located in easily accessible positions, e.g., aboveground on a buried pipeline or in an impervious pit where the groundwater level is high.

OPERATIONS AND MAINTENANCE DESIGN

Operating Conditions

The operating conditions for a pipeline system are defined by the operator of the system and are part of the design basis. Many of the design decisions are directly related to the operating philosophy of the system, both in an intermediate stage, as with a phased or growth system, and in the final system. For example, if the pipeline will be transporting a hot oil, it is critical that the system not be underdesigned, i.e., that the operating temperature be reasonably set, thereby leading to a viable design of the pumping and heating station requirements. Maintenance aspects of the system will also influence the design with regard to location of facilities (near fuel or power sources) and spacing of stations for cleaning facilities (pigging). The aspect of operator training and required level of supervision to monitor and verify the function of the pipeline system may dictate the location of controls and instrumentation readout panels, spacing of head operator and control function locations, and required personnel to operate the facilities.

The following sections discuss some of the specific operation and maintenance considerations for liquid pipeline design.

Surge

An important consideration in the design of liquid pipelines is surge, also known as water hammer. It is comprised of the pressure wave and reflected wave which travel through the fluid up and down a length of pipe when the velocity of the flowing column of fluid is altered or stopped suddenly. For example, water flowing at 10 ft/s (3 m/s) can generate a surge pressure of 500 psi (3400 kPa).

If a valve is closed against a flowing stream or if the velocity of the stream is slowed, as when a pump station is stopped, the kinetic energy of flow is converted to pressure energy and a positive wave is sent upstream at the velocity of sound in the medium:

$$V_s = \sqrt{\frac{BK/\rho}{1 + (KdC)/(Et)}} \tag{C5.26}$$

where V_s = speed of sound through commodity, ft/s (m/s)
B = 4,637 in^2/ft.s^2 for Eng units (1.0 for metric units)
K = bulk modulus of liquid,[8,10] psi (kPa)

ρ = density of liquid, lb_m/ft^3 (kg/m^3)
d = inside diameter of pipe, in (mm)
E = modulus of elasticity, psi (kPa)
t = wall thickness, in (mm)
C = constant of pipe fixity (0.91 for an axially restrained line, 0.95 for unrestrained)

In the case of a main-line block valve closing instantaneously some distance downstream of a pump station, flow is stopped in the vicinity of the valve but continues to flow from the pump station. The continued pumping packs the line section between the pump and the valve. The surge wave beginning at the valve travels up the hydraulic gradient reaching the pump several seconds later, raising the discharge pressure high enough to reach the shutoff head or setpoint of the pump, causing the discharge check valve to close. This closed valve reflects the wave back toward the valve, reinforcing the incoming wave and resulting in line pressure much higher than normal. This case is exaggerated as instantaneous, while in reality valve closure is not. However, in a real situation such as described, the effect is measurable and can be dramatic.

The surge wave travels upstream and is reflected downstream, oscillating back and forth until its energy is dissipated in pipe wall friction. The amplitude of the surge wave, or the magnitude of the pressure surge, P_{surge}, is a function of the change in velocity and the steepness of the wave front and is the inverse of the time it took to generate the wave:

$$P_{\text{surge}} = Bv_s\rho\Delta V \qquad \text{for } T < 2L/v_s \tag{C5.27}$$

where P_{surge} = pressure, psig (kPa)
B = 2.157E−04 for Eng units; (1.0 E−03 for SI units;)
v_S = speed of sound through commodity, f/s (m/s)
ρ = density of liquid, lb_m/ft^3 (kg_m/m^3)
ΔV = total change in velocity of fluid, ft/s (m/s)
$2L/v_S$ = propagation time, s
T = valve closing time, s
L = distance from inception, ft (m)

This is only an approximation of the surge pressure magnitude for cases limited by the stated time of closure criteria.

Computer programs are utilized to give specific and accurate analysis of the maximum surge pressure, location of critical points in the system, and other factors.

The surges are attenuated by friction, and the surge arriving at any point on the line is less than at the origin of the surge wave. Nevertheless, when flow velocity is high and stoppage is complete, or when a pump station is bypassed suddenly, the surge energy generated can produce pressures high enough to burst pipe, sometimes at points distant from the point of origin of the event.

Another case of surge is when a pump station is shut down suddenly, as in a power failure at an electrically powered station, causing the station to be bypassed automatically. This produces a drastic change in velocity of the flow but not a complete stoppage. As in any other surge, a pressure wave is sent upstream and a refraction wave is sent back downstream from some point in the system where there is sufficient discontinuity to permit total or partial reflection of the wave, i.e., a valve, or major bend in the pipe. The positive surge wave, traveling upstream, reaches the next upstream station, raising both its discharge and suction pressure (with the wave effectively passing counter to flow through the pump since the check valve does not close). It is possible that this situation may shut down the upstream station on high discharge pressure, thereby starting a new surge wave which effectively adds to the first. This pyramiding of surges can travel hundreds of miles knocking down pump station after station in domino style and producing higher and higher pressures upstream. In the downstream direction, the negative wave may shut down stations downstream of the event by starving their suction pressure.

Severe surge problems can be mitigated through the use of quick-acting relief valves, tanks, and gas-filled surge bottles. These facilities tend to be expensive single-purpose devices which are seldom needed and are often inadequately maintained by operators. An inexpensive feature of modern computer-based control systems is the *permissive* circuit, which can be rigged so that the system cannot be operated at a rate above an intrinsically safe level unless all outside parameters are satisfied. The loss of a signal saying for example, that a given station is operating will cause other stations, set points to be backed off to a level consistent with any surge situation.

Design for the control of surges requires a thorough understanding of the particular pipe, pump, valve, and tank systems and their instrumentation. Therefore the total evaluation of the surge problems on a new system must await a fairly complete design (pipe size, wall thickness, taper, profile, flow velocity, tankage requirements, block-valve spacing, number, size and arrangement of pumping units, type of devices, and static-relief systems must be defined). Certain rudimentary surge considerations can be observed from the beginning which will help surge-proof the system. These include:

- Provide interlocks such that all pumping stops before main-line block valves can be closed.
- Low flow velocities ensure that changes in velocity cannot be too great.
- Long station spacing ensures maximum surge attenuation.
- Multiple pumping units at each pump station minimize the opportunity for a complete station failure.
- Looped feeders for electric stations.
- A margin of error between the hydraulic gradient and the MAOP based on wall thickness.
- SCADA communications to prevent upstream pump stations from over pressuring the downstream section.

Section 402.2.4 of the Code addresses the topic of surge pressure, stating that ''. . . the level of pressure rise due to surges and other variations from normal operations shall not exceed the internal design pressure at any point in the piping system and equipment by more than 10 percent.''

Corrosion Protection[20,21]

Corrosion of a pipeline can be both external and internal. Internal corrosion, apart from exceptional cases of corrosive fluid components such as H_2S, is usually a gradual process resulting in a lowering of pipeline efficiency and is characterized by indentations and pits. Regular line cleaning with scraper *pigs*, discussed later, can be utilized to care for the internal surface of most installations. Internal corrosion can also be controlled by injecting a corrosion inhibitor into the transported fluid.

Another method of reducing internal corrosion is by internally lining the pipe. While there are few examples of existing liquid hydrocarbon pipeline systems using epoxy coatings, they give good protection, long life, and have a low friction factor.

External corrosion is a major factor in the design and operation of a pipeline system; external corrosion can reduce the life of a pipeline and impair its safety. External corrosion is mitigated by application of a pipe coating and the installation of a cathodic protection system. The external coating increases the pipe-to-soil electro-chemical resistance, and the cathodic protection (impressed current or galvanic anodes) system makes the pipe cathodic with respect to the surrounding soil.

There are a number of methods and materials available for external coatings, offering different benefits and a range of cost. Methods include over-the-ditch after welding, and precoating pipe joints at the pipe mill before shipping or in plants located near the jobsite before stringing.

Materials for over-the-ditch include cleaning and priming followed by one or more sequential layers of hot-applied asphalt or coal-tar enamel, felt wrap, and a final wrap of kraft paper. The most recent applications of plant-applied materials are fusion-bonded, thin-film epoxy (FBE), followed by an extruded copolymer adhesive and an extruded polyethylene jacket. (Refer to Chap. B10 for more details on FBE lining and coating.) Consideration for selection of a coating should include, in addition to installation and shipping costs, quality and integrity of the completed coating system as installed and other concerns, such as chemical resistance to soil conditions, stray electrical currents, maximum service temperature, storage and handling, et cetera. Specialty coating systems for thermally insulated pipelines in high-temperature service may require an exterior jacket impervious to intrusion by water vapor.

Further discussion of internal and external coatings and of cathodic protection systems can be found in Chap. C6, Gas Systems Piping. The same material is applicable to liquid petroleum systems.

ASME B31.4 does not specify that an allowance be made for corrosion in determining nominal wall thickness for fluids and services covered by the Code, with the provision that internal and external corrosion control is provided as directed by Chap. VIII of the Code.

Metering

Early methods for monitoring the volume of flow through a liquid pipeline relied on tankage gauge readings at different points along the system. With the development of computer monitoring systems, liquid pipeline systems are now monitored continuously. Metering devices used in the pipeline business today fall into three groups:

- Pressure drop
- Positive displacement
- Turbines
- Miscellaneous, including sonic and vortex meters

The first group measures the pressure drop created when flow is restricted either across an orifice plate or a venturi tube. This type of meter is unable to identify variations in density and viscosity, which may be variable in liquid petroleum services, and therefore is not used for fine measurements except in conjunction with a viscosity or density meter.

The positive displacement group of meters includes a wide variety of mechanical devices which entrap a discrete quantity of fluid and move it physically from one side of the meter to the other. Screws, pistons, buckets, gears, and sliding vanes have been used successfully, especially with viscous fluids, but are subject to wear in continuous, high-capacity low-lubricity situations.

Turbine meters consist of a turbine wheel with tiny magnets mounted axially between sleeve bearings inside a short length of pipe. As the fluid turns the turbine wheel, the rotations are recorded and counted electronically by the passage of the magnets. These meters have been successful because they are simple in design, essentially having only one moving part. The turbine wheel, when in motion, tends to center itself in the pipe so there is almost no bearing friction.

All three of these meter types have their uses. The orifice, because it is rugged and low in cost, is used where exact measurement is not required. Positive displacement meters are used where the fluid is viscous or a high range of viscosity is expected and close measurements are needed for custody transfer. Turbine meters are used for the continuous, high-capacity bulk movement of fluids because they are reliable and accurate over a wide range of flow rates.

The fourth group has more specialized meters. Information on a particular type of meter or installation is available from manufacturers.

Leak Detection

Detection of major leaks is a major concern of pipeline operating companies. Large leaks can be detected with relatively simple instrumentation; however, to detect small leaks requires computer systems that can also monitor and account for variations in temperature, pressure, density, and composition. To illustrate the problem of leak detection, with an order of magnitude comparison, consider that for an NPS 8 (DN 200) line transporting 24,000 BPD (3,815 m^3/d), a detectable deficiency of $+/-$ 2 percent of flow is a 14 gpm (3.18 m^3/h) leak. A 0.2 percent deficiency, detectable only with a sophisticated leak detection system, on a NPS 48 (DN 1,200) line pumping 2.4 MMBPD (381,000 m^3/d), is a 140-gpm (31.8 m^3/h) leak. The hole in the pipe which would leak 140 gpm (318 m^3/h) at gage pressure of 500 psig (34.5 bar) is more that 0.50 in (12.7 mm) in diameter. The hole which would leak 14 gpm (3.18 m^3/h) at the same pressure is less than 0.0625 in (1.59 mm) in diameter.

For years, basic leak detection has been based on the principle of line balance, based on the continuity of flow in the pipeline, i.e., flow in equals flow out. In its simplest form, leak detection can rely solely on readings of flow meters or tank gauges at periodic intervals, with a recurring discrepancy indicating a leak. For line balance to be viable, accurate flow measurements at both ends of the pipeline or section of pipeline must be made and reconciled regularly. In early pipeline systems, minor differences in reading time, temperature, line pack et cetera, made the results somewhat erratic, but a continuous shortage in delivery for several hours was sufficient cause for a line patrol to be sent out or for the system to be shut down and pressure tested. Large leaks or line breaks were detected by comparing suction and discharge pressures with flow at pump stations, e.g., falling discharge pressure combined with increased flow meant a leak downstream; falling suction pressure combined with decreased flow meant a leak upstream.

In modern pipeline systems, computer-based monitoring systems compare flow rates as well as total flow and record the variables of the flow (temperature, viscosity, et cetera) The determination of whether there is a leak in the system is based on the readout differentials. For reliable pipeline surveillance, both short-term (minutes) and long-term (days) calculations should be maintained and compared against respective threshold values.

Perhaps the more important problem in leak detection is location of the leak once it has been detected. While line balance will indicate the magnitude of the leak, other means are required to pin-point the location between two monitoring positions.

A general method for location of a leak is to compare the hydraulic gradient for the flow measured upstream with that of the flow measured downstream, with the intersection of the hydraulic gradients being the approximate leak location. Methods for location of rapidly formed leaks include monitoring the rate of change of pressure and flow (dp/dt and dq/dt, respectively) and the pressure-wave differential which occurs as the result of a rapid leak, propagating from the leak in both directions, with the velocity of sound in the liquid (see surge discussion). Similarly, the transient pressure wave can be monitored with transducers which will detect the transient pressure wave, its magnitude being the generated pressure wave at the leak location delayed exponentially as a function of the distance to the leak and the velocity of sound in the liquid.

More discussion on the theory of these leak detection and location methods can be found in standard hydraulic texts, manufacturer's literature, and in some of the references and bibliography.

Pipeline Pigs

A ''smart'' pig is an instrumented device which travels internally along a pipeline, monitoring the operating parameters (flow, temperature, etcetera) and the physical condition of the pipe (wall thickness, corrosion, out-of-roundness, et cetera). Pigs are also used to ''listen'' for the acoustical traces of leaks. Simple pigs, on spheres, are sometimes used to mark the transition between two commodities in a multiproduct pipeline.

The information the pig collects must be relayed to the master control center, and the pumping, heating, or pumping-heating stations along the route must have facilities designed for the handling of pigs, including launching and receiving traps. Additionally, the design for use of pigs may influence the specifications for elbows, tees and valves.

SCADA[23]: Supervisory Control and Data Acquisition

Maintaining the integrity of a pipeline system which spans hundreds of miles (kilometers) is a complicated task, involving the monitoring, measurement, and analysis of a continuous flow of data from meters, pipeline pigs, transducers, and other collecters. Just as critical as collecting the data from source points along the length of the system is the transmission of the data to the facilities of the pipeline operating company, which may be remote from the pipeline itself. There is also the aspect of control and coordination of multiple stations along the system so that they operate in conjunction with each other rather than in opposition.

Early pipeline systems relied on analysis of data at the local station and communication of information by voice over telephone or wireless. The operating philosophy of many pipeline systems today is to limit the number of manned control stations. Today's technology relies instead on SCADA (Supervisory Control and Data Acquisition) systems to collect data from monitor points by fiber optics, microwaves, and satellite-communications technology to transmit the data to control stations. Here, high-speed computers analyze the data and perform on-line control functions required to maintain system parameters within their operating limits. Application of these technologies requires consideration early in the mechanical design to include the operating components for the intended modes of operation and control.

The result of this technology has been to maximize the safety and operating efficiency of the system. Additionally, the collection and storage of operational information over the life of a pipeline system facilitates new pipeline design and operation through the verification of computer models which are used in the design phase to:

- Identify out-of-range operating variables
- Select preferred operations at and among pumping stations to control use of pumping units and cost of energy
- Identify limits for pumping rates and fluid properties of commodities transported
- Predict effects of modifying system facilities, operations, or changing characteristics of commodities transported
- Schedule tenders, or contracts, for transporting commodities by various fluid properties, ownership, source and destination

SYSTEM COST ANALYSIS[22]

The first several sections of this chapter have shown that for a given proposed pipeline, there are many possible pipeline systems which can be assigned to transport the commodity. Selection of the preferred pipeline route, diameter, material, wall thickness, pump-station location, pump units, and operational equipment or facilities is typically the result of economic analysis and investment-capital evaluation of the most reasonable scenarios developed through the design phase.

Typically, even before the detailed design of a pipeline systems has begun, an order-of-magnitude cost study will be performed, with the goal of determining the feasibility of continuing to invest time and capital in the design phase of the project. At this point, a preliminary route may have been selected; however, other possibilities may still exist, and further developments may indicate a change in the assumptions or information available during the preliminary analysis. In conjunction with the hydraulic, mechanical, and operational and maintenance designs, the economic analysis progresses, at times steering the decisions made in the design.

For a typical cross-country pipeline project, the cost of pipe and its associated construction and installation costs can be as much as 80 percent of the capital investment; therefore, the selection of the pipe, with regard to type of material, size, et cetera is very important. Trade-offs can be made during the design process with regard to diameter and wall thickness, grade of steel, and method of manufacture, as discussed in various earlier sections of this chapter. The cost of the pipe itself generally represents 25–50 percent of the total line cost, and the use of a reliable cost, based on current industry information, is significant. Annual cost indexes are published in a number of trade periodicals. Another source of costs for line pipe, as well as component fittings, valves, and installation factors are manufacturing associations and the construction industry publications.

Using the information developed through the earlier example, a comparison of system designs can be made, as shown in Tables C5.7, C5.7M, C5.8, and C5.8M. These tables illustrate the general procedure for selecting an appropriate pipeline system for a specific route. The same comparisons can be made for different routes.

TABLE C5.7 Isothermal Crude Oil System

Comparison of alternative diameters			
188,700 BPOD			
Pipe OD × wall thickness, in	16 × 0.250	20 × 0.250	24 × 0.250
Unit friction loss, ft/mile	132.1	44.1	18.1
Length, miles	248	248	248
Pipe friction loss, ft	32,756	10,934	4,486
Static head, ft	50	50	50
Total head, ft	32,806	10,984	4,536
Total pressure, psi	12,357	4,137	1,709
MAOH, ft	3,580	2,864	2,387
MAOP, psi	1,350	1,080	900
Number of pump stations	9.15	3.83	1.90
Operating horsepower @ 82% efficiency	48,382	16,200	6,690
Preliminary comparative cost comparison			
Capital cost ($1,000)			
Pipeline	16 × 0.250	20 × 0.250	24 × 0.250
Pipe unit weight, tons/mile	111.0	139.2	161.4
Pipe weight, tons	27,528	34,522	40,027
Pipe cost @ $700/ton	19,270	24,165	28,019
Unit installation cost, $/ft	35	40	45
Pipeline installation	45,830	52,378	58,925
Total pipeline cost	65,100	76,543	86,944
Pump stations			
Installed horsepower per station	5,000	4,250	3,500
Base cost per station	1,000	1,000	1,000
Installed horsepower @ $750/hp	3,750	3,188	2,625
Cost per station	4,750	4,188	3,625
Total pump station cost	47,500	16,750	7,250
Total system capital cost	112,600	93,293	94,194
Annual costs			
Insurance @ 1% of installed cost	1,126	933	942
Operating cost			
Pipeline @ 1.5% of installed cost	977	1,148	1,304
Stations @ 3% of installed cost	1,425	503	218
Financing cost (assuming 100% @ 7% of total cost)	7,882	6,530	6,594
Annual power @ $0.08 per kw	23,985	8,031	3,317
Total annual cost	35,395	17,145	12,374

TABLE C5.7M Isothermal Crude Oil System

Comparison of alternative diameters			
30,000 cubic meters per day			
Pipe size × wall thickness, mm	DN 400 × 6.4	DN × 6.4	DN 600 × 6.4
Unit friction loss, m/km	25.04	8.35	3.42
Length, km	400	400	400
Pipe friction loss, m	10,016	3,340	1,368
Static head, m	110	110	110
Total head, m	10,126	3,450	1,478
Total pressure, kPa	86,406	29,439	12,612
MAOP, kPa	9,366	7,493	6,240
MAOH less NPSH, m	1,073	853	706
Number of pump stations	9.44	4.04	2.09
Operating power @ 82% efficiency, kW	36,584	12,464	5,340
Preliminary comparative cost comparison			
Capital cost ($1,000)			
Pipeline	DN 400 × 6.4	DN 500 × 6.4	DN 600 × 6.4
Pipe unit weight, tonnes/km	62.5	78.4	94.3
Pipe weight, tonnes	25,000	31,360	37,720
Pipe cost @ $775/tonne	19,375	24,304	29,233
Unit installation cost, $/m	115	131	148
Pipeline installation	46,000	52,400	59,200
Total pipeline cost	65,375	76,704	88,433
Pump stations			
Installed power per station, kW	3,900	3,300	2,700
Base cost per station	1,000	1,000	1,000
Installed power @ $1,000/kW	3,900	3,300	2,700
Cost per station	4,900	4,300	3,700
Total pump station cost	49,000	17,200	7,400
Total system capital cost	114,375	93,904	95,833
Annual costs			
Insurance @ 1% of installed cost	1,144	939	958
Operating cost			
Pipeline @ 1.5% of installed cost	981	1,151	1,326
Stations @ 3% of installed cost	1,470	516	222
Financing cost (assuming 100% @ 7% of total cost)	8,006	6,573	6,708
Annual power @ $0.08 per kw	23,074	7,862	3,368
Total annual cost	34,675	17,040	12,583

TABLE C5.8 Isothermal Crude Oil System

Revised comparison for station bypass condition		
188,700 BPOD		
Pipe OD, in	20.0	24.0
Length of pipe (nominal), mi		
0.0250 in wall	229	223
0.312 in wall	19	25
Unit friction loss, m/km		
0.250 in wall	44.06	18.09
0.312 in wall	45.43	18.55
Pipeline friction, ft		
0.250 in wall	10,090	4,034
0.312 in wall	863	464
Total pipeline fiction, ft	10,953	4,498
Static head, ft	360	360
Number of pump stations	4	2
Station losses @ 15 m per station	200	100
Total pumping head, ft	11,513	4,958
Total pumping pressure, psi	4,341	1,869
Pumping power @ 82% efficiency, hp	16,979	7,312
Adjusted comparative cost		
Capital cost ($1,000)		
Pipeline OD, in	20.0	24.0
Pipe unit weight, tons/mi		
0.250 in wall	139.2	167.4
0.312 in wall	173.2	208.4
Pipe weight, tons		
0.250 in wall	31,877	37,330
0.312 in wall	3,291	5,210
Total pipe weight, tons	35,168	42,540
Pipe cost @ $700/ton	24,617	29,778
Unit installation cost, $/ft	40	45
Pipeline installation	52,378	58,925
Total pipeline cost	76,995	88,703
Pump stations		
Installed power per station, hp	4,300	3,700
Base cost per station	1,000	1,000
Installed power @ $750/hp	3,225	2,775
Cost per station	4,225	3,775
Total pump station cost	16,900	7,550
Total system capital cost	93,895	96,253
Annual costs		
Insurance @ 1% of installed cost	939	963
Operating cost		
Pipeline @ 1.5% of installed cost	1,155	1,331
Stations @ 3% of installed cost	507	227
Financing cost (assuming 100% @ 7% of total cost)	6,573	6,738
Annual power @ $0.08/kw	7,989	3,440
Total annual cost	17,162	12,698

TABLE C5.8M Isothermal Crude Oil System

Revised Comparison for station bypass condition		
30,000 cubic meters per day		
Pipe size	DN 500	DN 600
Length of pipe, km	400	400
6.4 mm wall	369	360
7.9 mm wall	31	40
Unit friction loss, m/km		
6.4 mm wall	8.35	3.42
7.9 mm wall	8.58	3.50
Pipeline friction, m		
6.4 mm wall	3,081	1,231
7.9 mm wall	266	140
Total pipeline friction, m	3,347	1,371
Static head, m	110	110
Number of pump stations	4	2
Station losses @ 25 m per station, m	100	50
Total pumping head, m	3,557	1,531
Total pumping pressure, kPa	30,355	13,067
Pumping power @ 82% efficiency, kW	12,852	5,532
Adjusted comparative cost		
Capital cost ($1,000)		
Pipeline size	DN 500	DN 600
Pipe unit weight, tonnes/km		
6.4 mm wall	78.4	94.3
7.9 mm wall	97.5	117.4
Pipe weight, tonnes		
6.4 mm wall	28,930	33,948
7.9 mm wall	3,023	4,696
Total pipe weight, tonnes	31,952	38,644
Pipe cost @ $775/tonne	24,763	29,949
Unit installation cost, $/m	131	148
Pipeline installation	52,400	59,200
Total pipeline cost	77,163	89,149
Pump stations		
Installed power per station, kW	3,300	2,800
Base cost per station	1,000	1,000
Installed power @ $1,000/kW	3,300	2,800
Cost per station	4,300	3,800
Total pump station cost	17,200	7,600
Total system capital cost	94,363	96,749
Annual costs		
Insurance @ 1% of installed cost	944	967
Operating cost		
Pipeline @ 1.5% of installed cost	1,157	1,337
Stations @ 3% of installed cost	516	228
Financing cost (assuming 100% @ 7% of total cost)	6,605	6,772
Annual power @ $0.08/kW	9,007	3,878
Total annual cost	18,230	13,183

These comparisons also illustrate that consideration should be given to the method of financing and appropriate economic factors, such as system growth and annual operating costs.

The reader should refer to a handbook on financial decision making to learn more about the process of optimum system selection on economic grounds, including the processes of discounted cash-flow analysis.[22]

SPECIAL OIL PIPELINE DESIGN TOPICS

Special Hydraulic Conditions

Pipeline systems which fall under the scope of the Code have a wide variety of physical properties, as was shown in Table C5.1. Furthermore, two of the most important properties—viscosity and vapor pressure—vary with temperature. Earlier sections have discussed pipeline systems having no fluctuation in temperature, therefore the hydraulic system design was simplified. Hydraulic design of nonisothermal systems for heavier commodities, such as heavy crude and residual fuel oils are influenced by the effect of temperature on viscosity and related friction losses. Hydraulic design for the lighter commodities, such as gasoline and natural gas liquids (NGL), is relatively independent of viscosity, but more dependent on vapor pressure considerations.

This section summarizes the special considerations which will arise in the design of multiproduct, high-vapor pressure, hot oil, and non-Newtonian fluid pipeline systems, without attempting to be comprehensive. The reader is encouraged to consult specialists and specific references for more detailed discussions of the design of these systems than can be described here.

Multi-product Pipelines. If the pipeline system is to transport fluids of differing properties such as in a batch-type operation of a multiproduct system, the design fluid should be considered as the fluid producing the greatest friction loss, i.e., with the greatest viscosity at the design operating temperature. Using this fluid assures that all pumping stations have adequate power and all sections between stations have adequate wall thickness to sustain the design pressure and flow rate. Furthermore, multiproduct systems may have intermediate deliveries to the system and discharges from the system, i.e., different flow rates for each pipeline section may need to be considered.

To avoid excessive mixing of products, the system should be designed for flow in the turbulent region. Batching pigs can be used to minimize interface mixing at low flow rates.

A special pipeline application uses a batch-flow technique to transport unlike petroleum products sequentially in a single pipe without significant deterioration of the quality of the products by contamination. An explanation of the principle involves a discussion of laminar and turbulent flow regimes, as given in Chap. B8, Flow of Fluids.

In laminar flow, the molecules travel forward in parallel. Velocity is maximum at the center of the pipe, with decreasing velocity to zero at the pipe wall caused by viscous drag among the molecules and friction at the pipe wall. The standard velocity profile is parabolic over the cross-section of the pipe. Therefore, in the laminar flow region, an interfacial plane between adjacent products would rapidly deteriorate into a bulge, and the trailing product would tend to push through the leading product travelling more slowly near the pipe wall.

In turbulent flow, however, the molecules are in random motion, bouncing off other molecules as well as the pipe wall, and tend to remain in relative position. Therefore the profile of velocity is almost constant over the cross-section of the pipe and the fluid flows more like a ''plug.'' The interfacial plane between adjacent batches in the pipeline tends to remain in place as the products proceed down the pipeline. Some mixing occurs, principally by diffusion, and is more a function of length (time) of transport than velocity. The amount of contamination of the leading product by the trailing is the same as the contamination of the following by the leading. The transition from one product to another, as the interfacial zone passes a point, takes the shape of a sine wave. At some point as the interface wave passes the switching manifold, the flow is diverted to separate storage or distribution facilities. The contamination of one product in another is controlled by timing the *cut* or diversion. One way this is done is to place a gravitometer some distance upstream of the receiving terminal and to record its signal at the terminal so that the receiving operator has a preview of exactly what the interface will look like on arrival. From a design point, a batch flow pipeline has several hydraulic gradients, at least one for each product, as well as for the mixing zone.

High Vapor-Pressure Pipelines. High vapor-pressure systems are characterized by low density, low viscosity, and the necessity to operate the system at elevated pressure to maintain the fluid as a single phase in the pipeline. Single phase is maintained throughout the pipeline by maintaining the elevation of the hydraulic gradient above the ground profile by more than the head equivalent of the local vapor pressure. Back-pressure regulators may be installed in terminals to maintain the required elevated gradient. In this sense, the design of high vapor-pressure lines differs from crude-oil pipelines in that the parameter governing design is the vapor pressure, directly related to temperature, rather than viscosity, which is on maximum temperature, where maximum vapor pressure occurs, versus the maximum viscosity or minimum temperature point which is the design basis of highly viscous fluid systems.

Waxy and Heavy Crude Oils. One treatment for waxy and heavy crude oils is to heat the commodity as part of the transport operation. This concept was discussed earlier under nonisothermal hydraulics.

Chemical additives have also been used in oil pipeline systems to improve flow, i.e., reduce the drag of heavy and waxy crudes. Initially developed to enhance oil-well fracturing, these additives are hydrocarbon-based polymers. The additives are solutions of a high molecular–weight copolymer or polyolefin in a hydrocarbon solvent. Being a hydrocarbon, it distills according to the volatility of its fractions and is not distinguishable from the hydrocarbons originally present in crude oil. The additive reduces the turbulent flow, effectively expanding the transition flow region. It is injected into the flowing pipeline stream by a compact, skid-mounted chemical injection module consisting of injection pumps, positive displacement flowmeters with totalizers, and miscellaneous instrumentation. The polymer itself is so viscous that an inert gas, such as nitrogen, is used to help transfer the material from tank to pumps.

Injection of the additive must be at every station on a system, because the polymer is degraded by the action of centrifugal pumps. Initially it was thought that only very large diameter pipelines could benefit from the use of drag reducers.

Widespread tests have demonstrated that smaller crude and product lines can also benefit.

Multiphased Flow. The presence of gas and liquid phases of a product in sections or in the entire pipeline length has been mentioned in the discussion of slack flow. Multiphased flow is particularly a factor in flow lines used to gather the produced gas-oil-water mixture from wells in a production field and transport it to separation facilities. Detailed knowledge of phase-equilibrium conditions and related product properties as well as specific multiphased flow energy and hydraulic conditions is required for the design of these systems. Computer programs have been developed to include the possibility of multiphased flow and are commonly employed for the solution of the hydraulic design of this type of system.

Seismic Considerations

The entire topic of designing facilities and systems, including pipeline systems for earthquakes, is of major importance in many parts of the world. With consideration to pipeline design, three major seismic hazards for buried pipelines are landslides, liquefaction of soil under and around the pipe, and differential fault movement and ground rupture. Ground motion (shaking) itself is a major consideration in the design of stations and terminal facilities and aboveground supports of a pipeline, but has less effect on buried pipelines.[24,25] Differential settlement and faultline shifts with vertical and horizontal displacement will impose additional longitudinal stresses on the pipe material.

In route selection, a survey of alternate routes should consider evidence of past landslides, i.e., movement of the ground triggered by a seismic occurrence. Slopes showing signs of recent instability or movement may be candidates for further disturbance in event of an earthquake. If slope instability involves deep translations and rotational displacement, the potential ground movements in the area may be large, and relocation of the pipeline may be more cost beneficial than expensive stabilization measures. If instability involves slumps and shallow slides, slope stabilization may be effective. In any case, seismic and geologic specialists should be consulted.

In saturated, cohesionless soils, for example, nonplastic silts and medium-dense sands, liquefaction may be the major hazard wherein the soil temporarily is transformed into a liquid state, losing shear strength and bearing capacity. Liquefaction leads to lateral spreading, loss of bearing capacity, and uplift of buried pipe due to increased buoyancy. Several measures can be taken to design a pipeline system crossing areas susceptible to liquefaction, including designing for moderate deformations as a result of uplift during an event, limited burial in the area, and adding weight coatings, thereby limiting uplift. Operationally, additional line block valves for shut-off in case of a failure following an earthquake may need to be provided.

Fault movement is the most dramatic earthquake occurrence, and potentially catastrophic for a pipeline. Pipeline alignment in fault zones should be such that the expected fault movement will produce tensile stresses in the pipe. If compressive stresses result, buckling may occur. The line should be laid in relatively straight sections, crossing the fault at an angle of 60 to 90° and without abrupt changes in direction or elevation, a procedure which might serve as an anchor during an earthquake. Depth of cover should be minimal, thereby reducing soil restraint, and the backfill material should be granular, medium-range soft sand without large stones, placed well around the pipe to allow relatively free movement.

Two of the references, "Guidelines for the Seismic Design of Oil and Gas Pipeline Systems,"[24] and "Seismic Design of Oil Pipeline Systems"[25] have additional material on the location and operation of pipelines across earthquake fault zones.

Underwater Pipeline Design[26,27]

The subject of underwater pipeline design, in particular ocean or deepwater pipelines, is worthy of a separate chapter. Yet it is important to reference some of the related topics here, particularly about the section of a pipeline system that may cross a river or lake, requiring placement underwater. There are generally two types of underwater pipelines—those that cross a relatively short distance and represent a minimal portion of the entire system design, e.g., a river crossing; and the submarine pipeline, which is primarily underwater. River crossings of a pipeline can be made by pulling the pipe string previously welded onshore, or by laying the pipeline into a trench beneath the depth of scour, then covering with protective backfill. Directional drilling techniques to drill under the riverbed are available where soil conditions, distance, construction space, and pipe properties are amenable. However, the design of an underwater, or submarine, pipeline has several special considerations, which may or may not also be major considerations in river crossings of terrain pipeline systems. The reader is referred to specific handbooks on the design of ocean pipelines for more specific information on this topic.

If a body of water is to be crossed as part of a terrain pipeline and the decision is made to place the line underwater, a hydrographic survey of the area should be initiated. In addition to as much information as possible about the potential route, other hydrographic data are required, such as current and tidal data, weather records, wave height and patterns, possible underwater obstructions along the route, and sources of impact such as dragging anchors. Selection of the landfall location is also an important element in the route selection.

External pressure, discussed earlier, will be greatest on the empty pipeline after placement. The tendency of the pipe to buckle under external pressure should always be checked and design of buckle arresters, i.e., stiffer sections, included in the design to limit buckle propagation.

Selection of the material for the pipeline should take into consideration that the stresses during the laying operation may govern the mechanical design. Note that these stresses do not occur in conjunction with internal pressure. The laying

TABLE C5.9 Conversion Factors, Length

To obtain → Multiply ↓ by ↘	mm	in	ft	m	mi	km
Millimeter	1	0.03937	0.003281	0.001	6.2137E-7	1.000E-6
Inch	25.4	1	0.08333	2.540E-2	1.5783E-5	2.540E-5
Foot	304.8	12	1	3.0480E-1	1.8939E-4	3.048E-4
Meter	1000	39.37	3.2808	1	6.2137E-4	1.000E-3
Mile	1.6093E6	6.336E4	5.280E3	1.6093E3	1	1.6093
Kilometer	1.0000E6	3.937E4	3.2808E3	1.000E3	6.2139E-1	1

TABLE C5.10 Conversion Factors, Hydraulic Gradients

To obtain → Multiply ↓ by ↘	$kg_f/cm^2/km$	kPa/km	psi/mi	ft_{hd}/mi	m_{hd}/km
kg_f/cm^2 per km	1	98.0665	22.8892	52.828/rd	10/rd
kPa per kilometer	0.0102	1	0.2334	0.5387/rd	0.10976/rd
psi per mile	0.04369	4.2843	1	2.3077/rd	0.4371/rd
$foot_{hd}$ per mile	0.01893*rd	1.8564*rd	0.4333*rd	1	0.1894
$meter_{hd}$ per kilometer	0.10*rd	9.8067*rd	2.2879*rd	5.280	1

rd = relative density: absolute density of fluid divided by density of water @15°C and 101.325 kPa, where density of water is taken as 1000 kg/m^3. Subscript hd = head in column of liquid at the density of the fluid.[6,7]

stresses can be categorized as direct pulling stresses, from welding onshore and pulling into location or from the tension of laying from a laybarge; bending stresses as the pipe is lowered to the riverbed or seabed; torsional stresses induced during the laying operation; and current and wave stresses during tow and placement.

Pipeline stability, once in place, is another consideration to ensure security of the pipeline. If additional weight is required, it can be provided by weights or anchors placed at intervals or by continuous weight coating. Placement of the pipeline in a trench and covering with protective backfill is another alternative.

TABLE C5.11 Conversion Factors, Pressure

To obtain → Multiply ↓ by ↘	lb_f/sq in	lb_f/sq ft	kg_f/sq cm	bar (b)	kPa
$Pound_f$ per square inch	1	144	0.070307	6.89476E-2	6.89476
$Pound_f$ per square foot	6.943E-3	1	4.883E-4	4.787E-4	0.04787
$Kilogram_f$ per square centimeter	14.223	2048.1	1	0.9807	98.0665
Bar	14.504	2088.6	1.0197	1	100
Kilopascal	0.14504	20.89	0.010197	0.0100	1

Unit Conversion Tables. Tables C5.9, C5.10 and C5.11 provide unit conversion factors for length, hydraulic gradient, and pressure, respectively. Refer to App. E1 also.

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