Stress analysis methods for underground pipe lines

Elements include pipe movement, anchorage force, lateral soil force, soil friction, soil-pipe interaction

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ANALYZING an underground pipe line is quite different from analyzing plant piping. Special problems are involved because of the unique characteristics of a pipe line, code requirements and techniques required in analysis. Elements of analysis include pipe movement, anchorage force, soil friction, lateral soil force and soil-pipe interaction.

Unique characteristics. To appreciate pipe code requirements and visualize problems involved in pipe line stress analysis, it is necessary to first distinguish a pipe line from plant piping. Unique characteristics of a pipe line include:

- High allowable stress. A pipe line has a rather simple shape. It is circular and very often runs several miles before making a turn. Therefore, the stresses calculated are all based on simple static equilibrium formulas which are very reliable. Since stresses produced are predictable, allowable stress used is considerably higher than that used in plant piping.

- **High yield strength pipe.** To raise the allowable, the first obstacle is yield strength. Although a pipe line operating beyond yield strength may not create structural integrity problems, it may cause undesirable excessive deformation and possibility of strain follow up. Therefore, high test line pipe with a very high yield to ultimate strength ratio is normally used in pipe line construction. Yield strength in some pipe can be as high as 80 percent of ultimate strength. All allowable stresses are based only on yield strength.

- **High pressure elongation.** Movement of a pipe line is normally due to expansion of a very long line at low temperature difference. Pressure elongation, negligible in plant piping, contributes much of the total movement and must be included in the analysis.

- **Soil-pipe interaction.** The main portion of a pipe line is buried underground. Any pipe movement has to overcome soil force, which can be divided into two categories: Friction force created from sliding and pressure force resulting from pushing. The major task of pipe line analysis is to investigate soil-pipe interaction—which has never been a subject in plant piping analysis.

Code requirements. Pipe lines normally are designed, constructed, inspected and operated according to minimum federal safety standards stipulated in Title 49 of Code of Federal Regulations.[1,2] The standards base for the analysis are ANSI B31.4, "Liquid Petroleum Transportation Piping Systems,"[3] and ANSI B31.8, "Gas Transmission and Distribution Piping Systems."[4]

Because it is more economical to ship gas at the lowest temperature possible, the stress problem involved in a gas line is less severe than that in an oil line. The following discussion will be based mainly on ANSI B31.4 which is made a part of 49-CFR Part 195, but the philosophy presented should be applicable to gas pipe lines as well. This section covers only the rules that are pertinent to stress analysis, however, and requirements are revised frequently to reflect results of new developments.

Wall thickness. The first step in stress analysis is to calculate wall thickness required. (The diameter of the pipe is generally determined by a different discipline of engineering.)

According to the code, nominal wall thickness of straight sections of steel pipe shall be equal to or greater than t determined in accordance with the following equation:

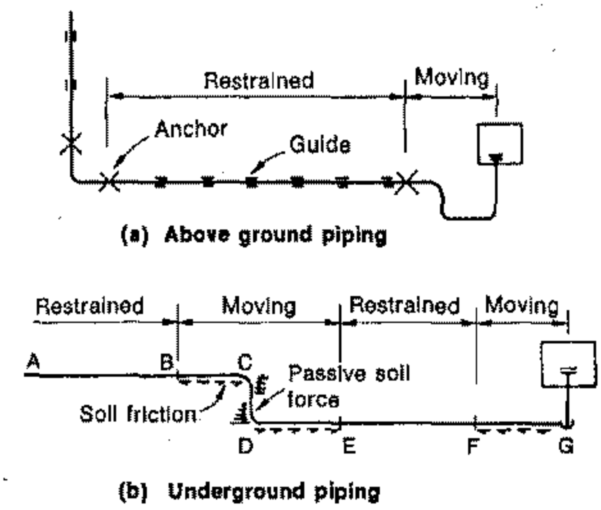

Fig. 1—Restrained and moving portions of a pipe line.

Conversely, restrained portions do not have significant bending stresses.

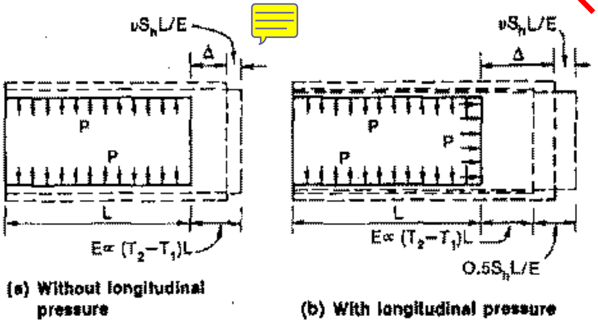

Fig. 2—Free expansion of pipe.

$$t = \frac{PD}{2ES} + A \qquad (1)$$

where

t = Nominal wall thickness, inch
P = Internal design pressure, psig
D = Nominal outside diameter of pipe, inch
E = Weld joint factor (efficiency)
S = 72% of specified minimum yield strength (SMYS) for new pipe of known specification, psi
A = Sum of allowance for threading, grooving, corrosion and others as required, inch. Corrosion allowance is not required if pipe and components are protected against corrosion in accordance with code requirements.

Equation 1 is well known. What makes it unusual is the definition of t. The thickness determined is nominal thickness, which is quite different from minimum thickness calculated by other piping codes. Subtracting underthickness allowed in the approved specification, remaining minimum thickness can be as low as 87.5% of nominal thickness. True allowable stress is then equal to

$$0.72\ SMYS/0.875 = 0.823\ SMYS$$

Adding the 10 percent surge allowance, final allowable stress is 90 percent of *SMYS*.

This allowable stress is considerably higher than allowable stress for plant piping in view of the high yield-ultimate strength ratio of the steel normally used in pipe line construction. API Grade X52 pipe, for example, has an *SMYS* of 359 MPa (52,000 psi) and a minimum ultimate strength of 455 MPa (66,000 psi). The true allowable stress is 0.823 *SMYS* = 295 MPa (42,796 psi), which has a safety factor of 1.5 based on the ultimate strength. While this factor is much lower than 3.0 or 4.0 used in plant piping, the high level of safety is well maintained. Because of simplicity of the pipe line configuration, the 1.5 factor is a true factor including little uncertainty. Remoteness of pipe line location is also a consideration.

Expansion and flexibility. Flexibility analysis is the stress analyst's main task. The code classifies a pipe line into two categories—restrained lines and unrestrained lines—which conflicts with a widespread misconception that a whole pipe line project is a line and the pipe is always more or less restrained.

A pipe line, buried or above ground, has both fully restrained portions and moving portions. The moving portions, which are equivalent to the code's unrestrained lines, will generally create significant bending stresses.

As shown in Fig. 1, in an above ground line restrained portions are always prevented from moving by installing anchors and guides, but in a buried line a large portion is fully restrained by soil friction only.

When a line is pressurized and heated, corners C, D and G will start moving. The movement creates a soil friction force proportional to the length of the moving portion of the pipe. If total friction force developed along the pipe is sufficient to suppress expansion, the movement will stop.

Points B, E and F where the movement stops are called virtual anchor points. Non-moving portions AB and EF are called fully restrained lines.

Restrained portions. To prevent movement, a force is required to bring the pipe from its free expanded or contracted position to the original position. As shown in Fig. 2(a), in a fully restrained line longitudinal pressure stress is absorbed by the anchor or soil friction and does not come into the picture.

The figure uses the following symbols:

L = Length of a pipe section, inch
T_1 = Temperature at time of installation, °F
T_2 = Maximum or minimum operating temperature, °F
α = Linear coefficient of thermal expansion, inch/inch/°F
ν = Poisson's ratio (0.3 for steel)
S_h = Hoop stress due to fluid pressure, psi
E = Modulus of elasticity of pipe, psi
Δ = Net free expansion, inch
S_L = Longitudinal stress in the pipe, psi
t = Nominal wall thickness of the pipe, inch.

When temperature reaches T_2 the pipe section will expand $\alpha(T_2 - T_1)L$, but the hoop tensile stress will make it to shrink $\nu S_h L/E$. This shrinkage due to hoop tension is similar to the common phenomena seen in stretching a rubber band: when stretched in the longitudinal direction, the sidewise dimension will shrink. If steel is stretched one inch in one direction, it will shrink 0.3 inch each in both perpendicular directions. This 0.3 is called Poisson's ratio and the shrinkage is commonly referred to as Poisson shrinkage. After subtracting the Poisson shrinkage from the expansion, net expansion becomes,

$$\Delta = \alpha(T_2 - T_1)L - \nu S_h L/E \qquad (2)$$

Longitudinal stress produced is equivalent to the stress required to squeeze Δ back to the original position. Since $S_L = -E\Delta/L$, then

$$S_L = -E\,\alpha(T_2 - T_1) + \nu S_h \qquad (3)$$

Equation 3 is the same formula as shown in the code, except the sign has been reversed so that a minus (−) will mean a compressive stress. Note that the longitudinal pressure stress is not acting on a fully restrained section. The net longitudinal stress becomes compressive for a moderate increase of T_2.

The code does not have a special allowable for longitudinal stress. It requires, however, that combined equivalent stress shall not exceed 90 percent of pipe *SMYS*. Fig. 3 shows stresses acting on the pipe wall. For the biaxial stresses shown, the code uses maximum shear theory of failure which says that pipe yields when maximum shear reaches shear yield stress. Maximum shear stress τ_{max} in this case can be easily shown[5] as

Restrained
NO BENDING

$$\tau_{max} = \sqrt{\frac{(S_h - S_L)^2}{4} + \tau^2} \qquad (4)$$

where τ is shear stress in the principle axes of the pipe. Since shear yield stress equals one half of tensile yield stress, an equivalent tensile stress defined as twice maximum shear stress is used to compare with tensile yield stress. The equivalent tensile stress is therefore equal to

$$S_e = 2.0\,\tau_{max} = \sqrt{(S_h - S_L)^2 + 4\tau^2} \qquad (5)$$

S_e is to be limited to 0.9 *SMYS*. The correct sign should be used fo S_L in substituting Equation 5. In cases where direct shear stress is negligible, the absolute sum of hoop stress and compressive longitudinal stress should not exceed the 0.9 *SMYS* limit.

In a restrained pipe line, anchors are frequently needed to reduce end movement. Because longitudinal stress has been calculated as shown in Equation 3, designers are often tempted to calculate anchor load by multiplying S_L with the pipe cross sectional area. This is incorrect because pressure end force has been ignored.

anchor
force includes
P x A

As shown in Fig. 4, an anchor is installed to limit end movement of the pipe. The anchor, therefore, separates the restrained portion from the moving portion of the line. Anchor force comes from both sides, longitudinal stress from the restrained side and pressure force from the moving side. Since longitudinal pressure stress equals to $0.5\,S_h$, the anchor force can be expressed as

$$F = A\,(0.5\,S_h - S_L)$$

or

$$F = A\,[(0.5 - \nu)\,S_h + E\,\alpha(T_2 - T_1)] \qquad (6)$$

where $A = \pi Dt$ is the cross sectional area of the pipe.

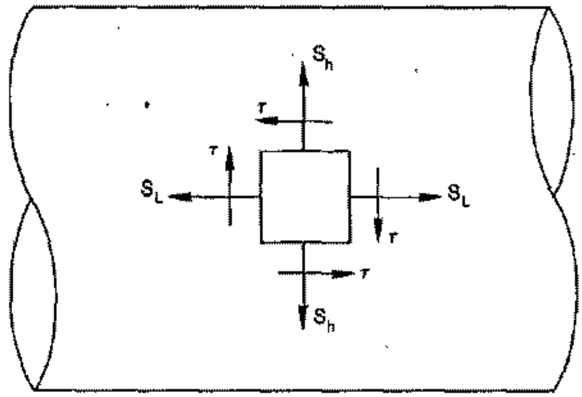

Fig. 3—Stresses acting on pipe wall.

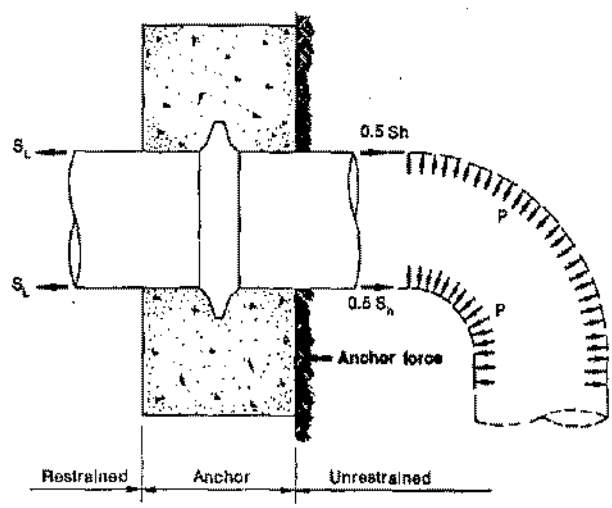

Fig. 4—Anchor force, from anchor installed to limit end movement of pipe.

REMEMBER: UNRESTRAINED portions
can have significant bending stress

Moving portions (Unrestrained lines). For the moving portion of the line, the code groups stresses into two categories: self-limiting stress and sustained stress.

Self-limiting stress resulting from thermal expansion and other strains shall be combined in accordance with the following equation:

$$S_E = \sqrt{S_b^2 + 4\,S_t^2} \qquad (7)$$

where $\boxed{S_b =} \sqrt{\dfrac{(i_i M_i)^2 + (i_o M_o)^2}{Z}}$ = Bending stress, psi

$S_t = M_t/(2Z)$ = Torsional shear stress, psi
M_i = Bending moment in plane of member, in.-lb.
M_o = Bending moment out of plane of member, in.-lb.
M_t = Torsional moment, in.-lb.
i_i = In plane stress intensification factor
i_o = Out of plane stress intensification factor
Z = Section modulus of pipe, in.[3]

Both i_i and i_o are to be taken from Fig. 419.6.4(c) of the code. The maximum computed expansion stress range S_E, without regard for fluid pressure stress, based on cold modulus of elasticity, shall not exceed the allowable stress range, S_A, where $S_A = 0.72$ *SMYS* of the pipe.

Equation 7 in fact is a modified form of Equation 5 by setting $\boxed{S_h = 0.0}$ and ignoring all direct shear stresses. Expansion stress, S_E, generally is calculated by using a computer program. Approximate formulas, tables and charts are also frequently used in estimation of the stress in simple configurations, but they are becoming more and more obsolete.

Because temperature change in a pipe is generally not very high, expansion to pressure (not to be confused with pressure stress which is not regarded in expansion stress calculation) effect is significant and should not be ignored. Fig. 2(b) shows the pipe expansion including longitudinal pressure effect. When the pipe is heated up to operating temperature, T_2, it expands in every direction.

In the longitudinal direction, thermal expansion is $\alpha(T_2 - T_1)L$. Applying longitudinal pressure, the pipe will expand $0.5S_hL/E$ in longitudinal direction but shrink somewhat in diametrical direction. Finally, adding radial pressure (hoop stress), the pipe expands fully in diametrical direction but shrinks $0.3S_hL/E$ in longitudinal direction due to Poisson effect. Net longitudinal expansion, Δ, is therefore equal to:

$$\Delta = \alpha(T_2 - T_1)L + 0.5S_hL/E - 0.3S_hL/E$$

Since the strain $\varepsilon = \Delta/L$ we have:

$$\varepsilon = \alpha(T_2 - T_1) + 0.2S_h/E \qquad (8)$$

The calculated expansion rate, ε, is in inch per inch unit, and a proper conversion may be required before being input to a computer program. It should be noted however that some computer programs calculate expansion rate in accordance with Equation 8 automatically, and care should be exercised to prevent a double penalty.

By comparing Equation 8 with Equation 6 it is clear that net expansion rate is equivalent to strain resulting from a pull by a force having the same magnitude as the anchor force. Anchor force is therefore referred to as potential expansion force in many discussions.

For sustained stress, the code requires that the sum of longitudinal stresses due to pressure, weight and other sustained external loadings shall not exceed 0.5 *SMYS*. This is undoubtedly very low in view of the high longitudinal pressure stress already used. However, this tight restriction on other loadings is also one of the justifications for allowing high hoop stress.

The entire code requirements follow well planned logic. Some of the logic is apparent and some is not too apparent. Therefore a code should be adopted in its entirety. Cross use of different codes generally is not acceptable.

About the author

Liang-Chuan Peng *is a registered mechanical engineer in California and Texas. He is currently employed by AAA Technology and Specialties, Houston, responsible for flange and pressure vessel computer program development. Before joining AAA he worked with Foster-Wheeler, Brown & Root, Nuclear Services, Bechtel, Fluor Pioneer and other companies on design, construction and operation of power plants, pipe lines and process plants. He has authored or coauthored SIMFLEX, NUPIPE, PIPERUP and many other computer programs, and an ASME paper on nuclear piping analysis. He graduated from Taipei Institute of Technology in 1960 and received an M.S. in mechanical engineering from Kansas State University in 1966.*

Sample calculations. Suppose a crude line of 20-in. diameter is to be designed with an operating pressure of 1,200 psi and an operating temperature of 180° F. It is decided that API 5LX Grade X52 electric resistance welded pipe will be used for the main portion of the construction. Construction temperature is expected to be 50° F.

Table 402.3.1(a) of the code shows the pipe has an *SMYS* of 52,000 psi and a weld joint factor of 1.0.

Therefore:

1. Wall thickness required is

$$t = \frac{PD}{2ES} + A = \frac{1200 \times 20.0}{2 \times 1.0 \times 0.72 \times 52000} + 0.0$$
$$= 0.3205 \text{ in., use } 0.344 \text{ in.}$$

The main portion of the pipe requires a nominal wall thickness of 0.344 in. However, a thicker wall pipe is generally required at station piping and river or road crossings.

2. Hoop stress check

$$S_h = \frac{PD}{2(t\text{-}A)} = \frac{1200 \times 20.0}{2 \times (0.344\text{-}0.0)} = 34{,}883 \text{ psi}$$

3. Compressive longitudinal stress at the fully restrained portion

$$S_L = -E\,\alpha(T_2 - T_1) + 0.3S_h$$
$$= -27.9 \times 10^6 \times 6.5 \times 10^{-6}\,(180\text{-}50) + 0.3 \times 34883$$
$$= -13{,}110 \text{ psi}$$

4. Equivalent tensile stress at the fully restrained portion

$$S_e = \sqrt{(S_h - S_L)^2 + 4\tau^2}$$
$$= \sqrt{(34883 + 13110)^2 + 0.0} = 47993 \text{ psi} > 0.9\ SMYS\text{, no good.}$$

Since the equivalent tensile stress exceeds the allowable of 0.9 *SMYS*, the design needs to be revised. There are several ways to make the design work: a) use semi-restrained construction such as placing offsets at regular intervals; b) increase installation temperature by burying pipe at midday or running hot air through it before back-filling; c) increase wall thickness. In this particular case because it is only slightly overstress, either b or c can be used. However, assume that in order to simplify construction scheduling, it is decided to increase wall thickness to 0.375 inch. After recalculating for 0.375 inch wall, we have:

$$S_h = 32{,}000 \text{ psi}$$
$$S_L = -13{,}975 \text{ psi}$$
$$S_e = 45{,}975 \text{ psi} < 0.9\ SMYS = 46{,}800 \text{ psi, O.K.}$$

5. Anchor force to fully restrain the line is

$$F = \pi Dt\,[0.2S_h + E\,\alpha(T_2 - T_1)]$$
$$= 3.1416 \times 20.0 \times 0.375\ [0.2 \times 32000 + 27.9 \times 6.5\ (180\text{-}50)]$$
$$= 706{,}280 \text{ lbs.}$$

6. Thermal expansion rate is

$$\varepsilon_t = \alpha(T_2 - T_1) = 6.5 \times 10^{-6}\ (180\text{-}50)$$
$$= 8.45 \times 10^{-4} \text{ in./in.} = 1.014 \text{ in./100 ft. pipe}$$

7. Pressure expansion rate will be

$$\varepsilon_p = 0.2\, S_h/E = 0.2 \times 32000/27.9 \times 10^6$$
$$= 2.29 \times 10^{-4} \text{ in./in.} = 0.275 \text{ in./100 ft. pipe}$$

8. Total expansion rate is therefore

$$\varepsilon = \varepsilon_t + \varepsilon_p = 1.014 + 0.275 = 1.289 \text{ in./100 ft. pipe}$$

Part 1 conclusion. From the above presentation the following conclusions can be made:

- The wall of a pipe line is very thin compared with plant piping for the same internal pressure. Wall thickness calculated by code formula is sufficient to ensure structural integrity of the main line. However, the code formula includes little uncertainty factor, therefore at complicated places such as station piping and river or road crossings, thicker pipe is generally required.

- At lines fully restrained either by soil friction or mechanical anchors, longitudinal stress will become compressive for a moderate temperature change of about 65° F for 52,000 *SMYS* pipe. If longitudinal stress is compressive, it should be added absolutely to hoop pressure stress to obtain equivalent tensile stress. This equivalent tensile stress, rather than longitudinal stress, should be limited to 0.9 *SMYS*.

- For a temperature rise of about 130° F, equivalent tensile stress will start to govern pipe wall thickness. Pipe thickness determined by pressure alone may not be sufficient.

- Although internal pressure will reduce longitudinal compressive stress at the fully restrained section of the line, it also increases expansion rate at the unrestrained portion. This pressure elongation is significant, especially in lines with lower temperature raise such as in gas transmission lines.

- The anchor force required to anchor the fully restrained pipe should be equal to the sum of the force required to resist longitudinal stress at the restrained side plus pressure end force at the unrestrained side. The anchor force can also be called potential expansion force.

LITERATURE CITED

[1] Part 195, Title 49, Code of Federal Regulations, "Minimum Federal Safety Standards For Liquid Pipelines."

[2] Part 192, Title 49, Code of Federal Regulations, "Minimum Federal Safety Standards For Gas Lines."

[3] ANSI B31.4, "Liquid Petroleum Transmission Piping Systems," published by the American Society of Mechanical Engineers.

[4] ANSI B31.8, "Gas Transmission and Distribution Piping Systems," published by the American Society of Mechanical Engineers.

[5] Timoshenko, S., *Strength of Materials, Part I*, page 51, 3rd edition, 1955. ■

Coming in May: Part 2—Soil-pipe interaction analysis, says the author, is the most important part of pipe line stress analysis. Part 2 includes discussion of soil forces and longitudinal and lateral pipe movement, plus further calculations on the same hypothetical pipe line covered in Part 1.