Comparative Risks of Static Tests for Pipelines

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1. INTRODUCTION

When a pipeline, gathering line, or well flow line is constructed an important step in the process is to verify that the line is fit for purpose using some form of non-destructive testing. Many projects employ x-ray or magnetic-particle testing during construction to confirm that welds meet design conditions. Virtually all projects include an after-construction strength test in addition to any other non-destructive tests. These strength tests are intended to ensure that the line can be safely operated at pressures approaching the Maximum Allowable Working Pressure (MAWP, also referred to as Maximum Allowable Operating Pressure (MAOP) and Design Pressure or (DP)).To conduct a strength test you fill the piping up with a test fluid, raise the pressure to some multiple of the MAWP (the multiple is described in ASME B31.3 for liquid hydrocarbon lines and ASME B31.8 for gas hydrocarbon lines, working through the standards often ends up with gathering systems and well sites tested to 1.5 times MAWP, and mainlines tested to values between 1.05 and 1.25 times MAWP).

The test fluid tends to be water or some available gas with a very strong bias in the industry towards water. This bias manifests itself in informal reluctance to consider another fluid (one engineer was asked “what is a safe distance from a pneumatic test?” and he answered with “I don’t know about you, but I’ll be in a different time zone”); Company Policy prohibiting pneumatic tests; and even regulations putting rigid restrictions and very large exclusion zones around pneumatic tests. This bias, like so many arbitrary exclusions, stems from an improper extrapolation of very good research.

2. STATIC TESTS

When setting up a static test there are 5 basic steps: (1/2) fill the line with test fluid; (2/1) remove any foreign fluids; (3) add fluid mass to raise the pressure; (4) hold the test pressure for a specified period of time; and (5) depressurize and remove test fluid. For tests with water, you fill the line and then remove foreign fluids (i.e., dissolved gases). For tests with natural gas, you remove foreign fluids (i.e., air) and then fill the line. For tests with air or nitrogen, you fill the line and there is no requirement to remove foreign fluids.

2.1. Determining test pressure

Before you can set a test pressure you must first assign an MAWP to the piping. The procedure and equations for assigning an MAWP varies somewhat from piping-code to piping-code, and the following is based on ASME B31.8-2010 (Gas Transmission and Distribution Piping Systems), but other codes yield similar results.

MAWP is determined based on the stresses that the pipe will undergo in operation. Typically stresses considered include: hoop stress from internal pressure; longitudinal stress from internal pressure (both for constrained and unconstrained pipes); thermal expansion; cyclical loading; local stresses; and stresses at joints and supports or anchors. Combined stress from internal pressure is taken as the difference between hoop stress and longitudinal stress (i.e., longitudinal stresses tend to cancel or offset hoop stress). Longitudinal stresses from internal pressure are generally less than half the magnitude of the hoop stresses from internal pressure. It has become standard to compute hoop stress and then not subtract longitudinal stresses (which creates a safety factor equal to the magnitude of longitudinal stress).

$$P_{mawp} = \left(\frac{2 \cdot S \cdot (t_{pipe} - t_{corr})}{D}\right) \cdot F \cdot E_{joint} \cdot T_{adj} \qquad \textit{Equation 1}$$

Equation 1 is a version of the "Steel Pipe Design Formula" (the version in ASME B31.8 does not reduce the pipe wall thickness for the corrosion allowance, but verbiage within the text indicates that this is proper).

The phrase in the parentheses of Equation 1 is known as "Barlow's Equation" (rearranged to calculate pressure and replacing "hoop stress" with Specified Minimum Yield Strength or SMYS) and the remaining terms are factors that are used to adjust the allowable working pressure for observed conditions. These terms are:

- **F**: Basic Design Factor (Table 841.1.6-1 and Table 841.1.6-2). This factor adjusts the allowable working pressure based on both service (e.g., the factor is higher for crossing a private road than crossing a railroad) and location (e.g., the adjustment is smaller if the pipe is running across a field than if it is running under a high-rise apartment). Basic design factor ranges from 0.8 to 0.4.
- **E_{joint}**: Longitudinal Joint Factor (Table 841.1.7-1) which derates pipe based on any longitudinal joints. For example, Seamless or Electric Resistance Welded (ERW) pipe has a Joint Factor of 1.0 while Furnace-Butt Welded pipe has a Joint Factor of 0.6.
- **T_{adj}**: Temperature Derating Factor (Table 841.1.8-1) has factors for the expected operating temperature of the pipe. If the pipe is operating below 250°F (121°C) then this factor is 1.0. The factor reduces by steps until an operating temperature of 450°F (232°C) has a factor of 0.867.

As can be seen from the Nomenclature table, the diameter term ("D") is outside diameter. For 12-inch pipe (DN300) and smaller, the outside diameter will be approximately the nominal pipe size plus twice the pipe wall thickness. For example, 8-inch (DN200) pipe has an inside diameter of 7.981 inches (202.7 mm), a wall thickness of 0.322 in (8.179 mm) for an outside diameter (D) of 8.625 in (219.08 mm). For pipe that is 14 inches (DN400) and larger, the outside diameter (D) is the nominal pipe size. For example, 20-inch (DN500) standard wall pipe has an outside diameter of 20 inches (508 mm), a wall thickness of 0.375 inches (9.525 mm), so the inside diameter is 19.25 inches (488.95 mm).

The Specified Minimum Yield Strength (*SMYS, "S"* in Equation 1) is provided by the pipe manufacturer. ASME B31.8 provides a table in the Mandatory Appendix D (table D-1) showing SMYS for various pipe types. A common pipe specification is API 5L, and in that specification it is common to use "Grade B" which table D-1 says has an SMYS of 35,000 psig (241 MPag). The API 5L standard has been around for a long time, and as steel-fabrication quality control has improved, it has become common for pipe to be dual stamped "Grade B/X-42", in that case the SMYS is 42,000 psig (290 MPag). For the other "X grades" in API 5L, the SMYS is the X number (e.g., X-70 has an SMYS of 70,000 psig (483 MPag)).

At the end of the day, the Steel Pipe Design Formula for modern steel pipe will most often reduce to Barlow's formula for the specific pipe size and grade reduced by the Location Class. Location class comes up many times in ASME B31.8, so it is useful to discuss it a bit further. According to Section 840.2.2 (when the text says "1 mile" it means "1 square mile, 1 mile along the pipe and ½ mile either side of the pipe"):

- ***Location Class 1.*** A Location Class 1 is any 1-mile (1.6-km) section that has 10 or fewer buildings intended for human occupancy. A Location Class 1 is intended to reflect areas such as wasteland, deserts, mountains, grazing land, farmland, and sparsely populated areas.
    - *Class 1, Division 1.* This Division is a Location Class 1 where the design factor of the pipe is greater than 0.72 but equal to or less than 0.80 and has been hydrostatically tested to 1.25 times the maximum operating pressure. (See Table 841.1.6-2 for exceptions to design factor.)
    - *Class 1, Division 2.* This Division is a Location Class 1 where the design factor of the pipe is equal to or less than 0.72 and has been tested to 1.1 times the maximum operating pressure. (See Table 841.1.6-2 for exceptions to design factor.)
- ***Location Class 2.*** A Location Class 2 is any 1-mile (1.6-km) section that has more than 10 but fewer than 46 buildings intended for human occupancy. A Location Class 2 is intended to reflect areas where the degree of population is intermediate between Location Class 1 and Location Class 3, such as fringe areas around cities and towns, industrial areas, ranch or country estates, etc.

- ***Location Class 3.*** A Location Class 3 is any 1-mile (1.6-km) section that has 46 or more buildings intended for human occupancy except when a Location Class 4 prevails. A Location Class 3 is intended to reflect areas such as suburban housing developments, shopping centers, residential areas, industrial areas, and other populated areas not meeting Location Class 4 requirements.
- ***Location Class 4***. Location Class 4 includes areas where multistory buildings are prevalent, where traffic is heavy or dense, and where there may be numerous other utilities underground. Multistory means four or more floors above ground including the first or ground floor. The depth of basements or number of basement floors is immaterial.

The term "intended for human occupancy" of course includes all homes, apartments, and house trailers, but it also includes workplaces where someone could normally be expected to occupy the space for more than 8 hours per day multiple days per week. It does not include structures such as free-standing residential garages, storage sheds, and hay barns—while these structures can be occupied, they are not normally occupied for extended periods multiple days per week.

In normal usage the phrase "has been hydrostatically tested to 1.xx times the maximum operating pressure" in Class 1 does not often have any impact, because strength tests are normally done to a multiple of MAWP which will always be greater than maximum normal operating pressure, so the real impact of Division 1 vs. Division 2 is the requirement in Division 1 that a hydrostatic test be performed. This is really the first place that the bias in favor of hydrostatic tests over pneumatic strength tests shows up—if you do a hydrostatic test, you can rate your pipeline to a higher pressure.

As can be seen in Figure 1, it is allowed by the code to use water or air for many (but not all) tests, and for tests in Location 1 Division 2 you can test with "gas" meaning you can use a hydrocarbon gas to test with. Notice that the maximum test pressure for any gas test is 1.25 times calculated MAWP (referred to as "DP" or "Design Pressure" in this table).

Notice that Table 841.3.3-1 column 6 re-defines MAWP from the constraints of Equation 1 to a real-world value based on actual test results. If your test pressure on an air test in Class 1 Div 2 is 1.25 times calculated MAWP, then MAWP equals the value you calculated. On the other hand, if you based your test on 1.25 times maximum operating pressure (MOP), then your pipe MAWP is equal to your expected MOP. This can result in using thinner walled pipe and/or a lower pipe grade, but if conditions change in the future the pipe may not be fit for purpose without a new static test.

Table 841.3.2-1 Test Requirements for Steel Pipelines and Mains to Operate at Hoop Stresses of 30% or More of the Specified Minimum Yield Strength of the Pipe

1	2	3	4	5	6
			Pressure Test Prescribed		
Location Class	Maximum Design Factor, *F*	Permissible Test Medium	Minimum	Maximum	Maximum Allowable Operating Pressure, the Lesser of
1 Division 1	0.8	Water	1.25 × MOP	None	TP ÷ 1.25 or DP
1 Division 2	0.72	Water	1.25 × MOP	None	TP ÷ 1.25 or DP
	0.72	Air or Gas [Note (1)]	1.25 × MOP	1.25 × DP	TP ÷ 1.25 or DP
2	0.6	Water	1.25 × MOP	None	TP ÷ 1.25 or DP
	0.6	Air [Note (1)]	1.25 × MOP	1.25 × DP	TP ÷ 1.25 or DP
3 [Note (2)]	0.5	Water [Note (3)]	1.50 × MOP	None	TP ÷ 1.5 or DP
4	0.4	Water [Note (3)]	1.50 × MOP	None	TP ÷ 1.5 or DP

DP = design pressure
MOP = maximum operating pressure (not necessarily the maximum allowable operating pressure)
TP = test pressure

GENERAL NOTES:

(a) This Table defines the relationship between test pressures and maximum allowable operating pressures subsequent to the test. If an operating company decides that the maximum operating pressure will be less than the design pressure, a corresponding reduction in the prescribed test pressure may be made as indicated in the Pressure Test Prescribed, Minimum, column. If this reduced test pressure is used, however, the maximum operating pressure cannot later be raised to the design pressure without retesting the line to a higher test pressure. See paras. 805.2.1(d), 845.2.2, and 845.2.3.

(b) Gas piping within gas pipeline facilities (e.g., meter stations, regulator stations, etc.) is to be tested and the maximum allowable operating pressure qualified in accordance with para. 841.3 and Tables 841.3.2-1 and 841.3.3-1 subject to the appropriate location class, design factor, and text medium criteria.

(c) When an air or gas test is used, the user of this Code is cautioned to evaluate the ability of the piping system to resist propagating brittle or ductile fracture at the maximum stress level to be achieved during the test.

NOTES:

(1) When pressure testing with air or gas, see paras. 841.3.1(c), 841.3.2(a) through (c), and Table 841.3.3-1.

(2) Compressor Station piping shall be tested with water to Location Class 3 pipeline requirements as indicated in para. 843.5.1(c).

(3) For exceptions, see paras. 841.3.2(b) and (c).

Figure 1: ASME B31.8 Table 841.3.2-1

Published values for API 5L pipe MAWP such as "Cameron Hydraulic Data" [10] that don't specify SMYS or area class tend to use Grade "A" (24,000 psig (165 MPag)) and a Class 1 Division 2 Design Factor of 0.72. Selecting a value from one of these tables will be ultra-conservative.

Pipe MAWP is rarely the limiting factor in a pipeline design. The limiting factor is nearly always the published flange rating of the flanges the designer uses. For example, many gathering systems are designed to use ASME B16.5 Class 300 flanges using metallurgy in Material Group 1.1 (Carbon steel) which has a MAWP of 720 psig. Couple Class 300 flanges with 16-inch (DN 400), 0.375-inch wall (9.55 mm) with 3/64-inch (1.2 mm) corrosion allowance, X-42 grade ERW, tested with water in a rural setting with a maximum operating temperature under 100°F (37.8°C) would have an MAWP of 1378 psig by Equation 1, but the flanges limit the system MAWP to 720 psig (5.0 MPag) or less (600 psig (4.1 MPag) is common).

Steel pipe is not perfectly rigid under pressure. The amount of stretch in a pipe is given by Hook's Law (Equation 2). The Modulus of Elasticity or "Young's Modulus" (Y) for steel pipe is on the order of 30×10^6 psi, and the initial length (L_0) is the circumference of the center of the pipe wall. For example, 1 mile (1.6 km) of 16-inch (DN400) Schedule 30 pipe (wall thickness 0.375 in (9.525 mm)) has a volume of 1193 bbl (190 m^3); raising the pressure to 900 psig (a common test pressure for ASME B16.5 Class 300 flanges) would have a ΔL term of 0.000469 inches (11.9μm), resulting in a change in pipe inside diameter of 0.00014 inches (3.79 μm) and a volume change of 3.1 gallons per mile (7.245 L/km) or 0.0061 percent of the pipe volume.

$$\Delta L = \left(\frac{P_{test}}{Y}\right) \cdot L_0 \qquad \textit{Equation 2}$$

2.2. Fluid Energy storage, transportation, and conversion

The risks associated with the internal energy of the fluid converting to an explosive failure requires a discussion of where the internal energy comes from, how it is transported, and how it can be converted to kinetic energy.

At any temperature and pressure, all fluids have internal energy. At atmospheric pressure and ambient temperature, the fluid contains some base level of energy. At elevated pressure the fluid contains: (1) the difference between the internal energy of the entire mass of fluid at the base conditions and the internal energy of the entire mass of fluid at elevated conditions; plus (2) the total energy in the fluid that was added to raise the pressure. The amount of fluid that must be added is a function of the "compression resistance" of the fluid. For liquids this compression resistance is the "Bulk Modulus", for gases it is the "compressibility".

$$V_{added} = \left(\frac{1.01}{E}\right) \cdot P_{tes\times t} \cdot V_{system} \cdot 1.01 = \frac{P_{test} \cdot V_{system}}{315842psig} \qquad \textit{Equation 3}$$

Liquid bulk modulus (defined as, the amount of pressure required to reduce the liquid volume by 1 percent) is very large, so even in the most aggressive tests the liquid will have relatively little added fluid volume (e.g., the bulk modulus (E) of water is on the order of 319,000 psi (2,200 Mpa)). To compress the test fluid, one must either reduce the volume (as in a piston moving in a closed cylinder) or add fluid. Equation 3 [2] shows how much water volume must be added to reach a desired test pressure with a constant system volume.

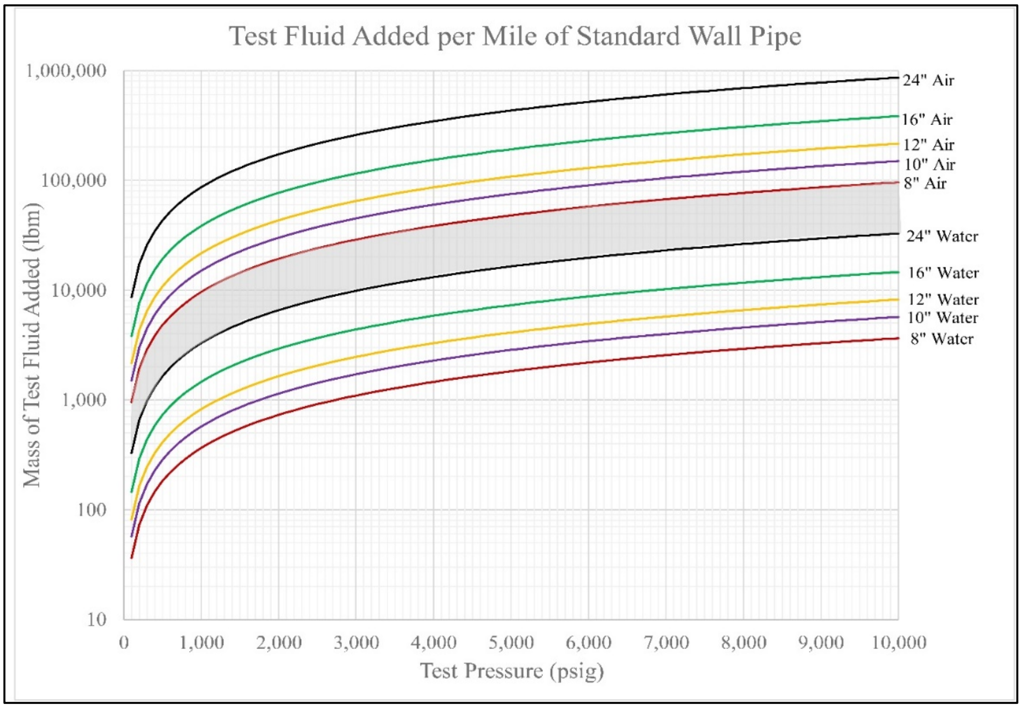

Figure 2. Mass added in a Static Test

Dealing with gas, you have to start with an Equation of State (Equation 4). It has become normal in many industries to refer to gas "volumes" referenced back to some "standard" conditions. This allows calculating the mass of gas in a system from the gas density at standard conditions without having to calculate exact values for temperature, pressure, and compressibility at flowing conditions. It is important to note that stating the gas "volume" in standard terms does not change the mass in the system, so dividing Equation 4 at flowing conditions by Equation 4 at standard conditions allows mass and the gas constant to cancel. In this paper I will refer to gas "volume" at standard condition in terms of SCF (SCm) with the prefix "M" taking the value "times 1000" for FPS units and "MM" as "times 10^6", while gas volumes at actual conditions and liquids as ft^3 (m^3). For a pneumatic test, the standard volume added can be determined by the real gas law stated in terms of density (Equation 5).

$$P \cdot V = m \cdot R_{gas} T \cdot Z \qquad \text{Equation 4}$$

$$V_{std} = P_{test} \cdot V_{system} \cdot \left(\frac{T_{std} \cdot Z_{std}}{T_{test} \cdot Z_{test} \cdot P_{std}} \right) \qquad \text{Equation 5}$$

Figure 2 shows the difference in fluid mass required to be added for various test pressures. Notice how much larger the required mass of gas is than the required mass of water.

This concept is laid out as an example in *Table 1* (fluid enthalpy values in *Table 1* come from NIST's REFPROP.exe) [9]. For the example used above with one mile of 16-inch 0.375 wall pipe tested to 900 psig the added volume of water for a hydrostatic test is 19 ft^3 (0.5 m^3) or 3.4 bbl (538 L)—a 0.2 percent increase in fluid mass. For the test in Table 1, you would have to add 62 times as much air as was in the line at atmospheric pressure.

	Air	Water
Volume of system under test (ft^3)	6,697	6,697
Fluid Specific Enthalpy (BTU/lbm)		
0 psig (14.7 psia) and 60°F	178.51	28.14
900 psig (914.7 psia) and 60°F	172.33	30.68
Fluid volume (MSCF for gas ft^3 for water)		
0 psig (14.7 psia) and 60°F	6.7	6,697
900 psig (914.7 psia) and 60°F	409.0	6,716
Fluid mass (lbm)		
0 psig (14.7 psia) and 60°F	512	447,367
900 psig (914.7 psia) and 60°F	31,303	448,642
Total fluid energy added (MMBTU)	5.30	1.18
Total fluid energy added (lbm of TNT)	2,772	615

Table 1. Example of fluid energy (1 mile of 16-inch, 0.375 wall pipe tested to 900 psig)

Notice in *Table 1* that the specific enthalpy of water goes up when you apply pressure and the specific enthalpy of air goes down when you apply pressure. The direction of these changes is very common, most liquids increase in specific enthalpy when pressure is applied, while most gases see a decrease in their specific enthalpy when pressures increase. You can see the results of this phenomena in vehicle braking.

- For a hydraulic braking system applying the brakes increases pressure (and therefore) the energy in the brake fluid, and that increased energy allows the slave cylinders to move with minimal mass transfer. Releasing the brake pedal lowers the pressure in the system and allows springs to disengage the slave cylinders, again with minimal mass transfer.
- For an air-brake system, applying the brakes allows a mass of high-pressure air from on-board air tanks to enter the brake lines and force the slave cylinders to move. When the breaks are released, the motive air is vented. While each unit mass of air has less energy at high pressure than it did at low pressure, the added mass from the tank more than makes up for loss in energy per unit mass.

The total fluid energy numbers in *Table 1* are large, and if that energy were rapidly converted from potential energy to kinetic energy the result would be an explosion. This type of event is called "explosive decompression" and is defined as "the rapid conversion of stored energy within a fluid into kinetic energy". Using the conversion 1 lbm TNT = 1913 BTU (2018 kJ) [4], one finds that the change in energy of the base fluid volume plus the total volume of the ejected fluid would have significantly more energy present with a gas test than with a liquid test if all of the energy was converted from stored to kinetic during the explosive event.

Experiments done at the University of Nebraska-Lincoln for the Department of Energy in 2012 [6] show that the gas temperature in an explosive decompression drops very rapidly to a minimum, and then increases to approximately initial temperature over the next few seconds. This minimum can be taken to be the end of "explosive decompression" and the start of "depressurization". The University of Nebraska-Lincoln paper does not identify the duration of this nearly vertical temperature transient. Other, less formal sources indicate it occurs at 20-50 mS after the creation of an opening large enough that pipe wall thickness is irrelevant to flow patterns (typically taken as an area larger than 6 times the square of the magnitude of the wall thickness, e.g, for 0.375-inch wall pipe, $0.375^2 \times 6 = 0.8435$ in^2 or the equivalent of a 1-inch diameter drilled hole). This very short time period explains why simply opening a vent valve doesn't result in an explosion—valve travel is on the order of seconds, not milliseconds.

$$v_{sonic} = \sqrt{k \cdot R_{gas} \cdot T} \qquad \textit{Equation 6}$$

Natural events within a fluid volume are limited to the speed of sound (Mach 1.0, Equation 6 [1]). This limitation comes about because of the creation of standing "shock waves" in the flow that inhibit communication from downstream to upstream. For fluids with velocities slower than Mach 1.0, the existence of lower pressure downstream is communicated upstream through a failure of the downstream fluid to support the higher upstream pressure.

At Mach 1.0 and above the shock wave is adequate to support the upstream pressure and fluid will only allow flow at the speed of sound. In 60°F (15.6°C) air, the speed of sound is 1118 ft/s (341 m/s), so one assumes the maximum range of the duration of an explosive decompression event, it would take 25 mS for the failure to communicate upstream and establish standing shock waves, and another 25 mS for the last fluid which can participate in the explosion to reach the failure point—27 ft (8.52 m). Assuming that there is an infinite (i.e., more than about 30 ft) length of pipe each direction from the failure, we can use 50 ft (15.24 m) as the "length" term in the available storage volume as a conservative approximation since an extreme time period is assumed.

	Air	Water
Entire System Added Energy (lbs TNT)	2,772	615
Ejected mass during explosive decompression (lbm)	296	4,236
Energy available to explosive decompression (lbm of TNT)	27	68
Adiabatic explosive decompression (lbm of TNT)	10	68

Table 2: Energy released in explosive decompression

Adiabatic energy can be calculated by Equation 7 which results in smaller values than a simple change in system enthalpy due to various inefficiencies in the energy conversion (this explains the difference in the "air" column between the last two rows in Table 2).

$$W_{gas} = \left(\frac{P_{test} \cdot V_{evacuated}}{k-1}\right) \cdot \left[1 - \left(\frac{P_{std}}{P_{test}}\right)^{\frac{k-1}{k}}\right] \qquad \textit{Equation 7}$$

In all of this discussion, it has been implicit that the magnitude of an explosive decompression is the sum of the internal energy of the mass ejected ± the change in internal energy of the remaining fluid during the ejection event (about 50 mS). For hydrostatic tests, the ejection event will normally include all of the water added during the pressurization phase, and since high pressure water contains more internal energy than low pressure water, and release of that excess energy does not require a mass transfer, as the volume of the system increases the magnitude of the explosion will increase.

For pneumatic static tests, only a small fraction of the added mass will be ejected, plus high-pressure gas contains less energy than low-pressure gas so increasing pipe length after a certain point does not increase the magnitude of an explosive decompression.

As you can see from Figure 3, for a given pipe size the energy that can be converted to an explosive decompression in a pneumatic test quickly reaches a maximum (due to the limitations of the speed of sound). After that maximum, adding pipe length does not change the magnitude of an explosive decompression. The hydrostatic test has a different driver for energy conversion, pressurizing a gas lowers its specific enthalpy while pressurizing a liquid increases its specific enthalpy. Thus, the force of the explosion includes a source of energy that is transported without requiring mass transfer and the "water" lines in Figure 3 increase without bound.

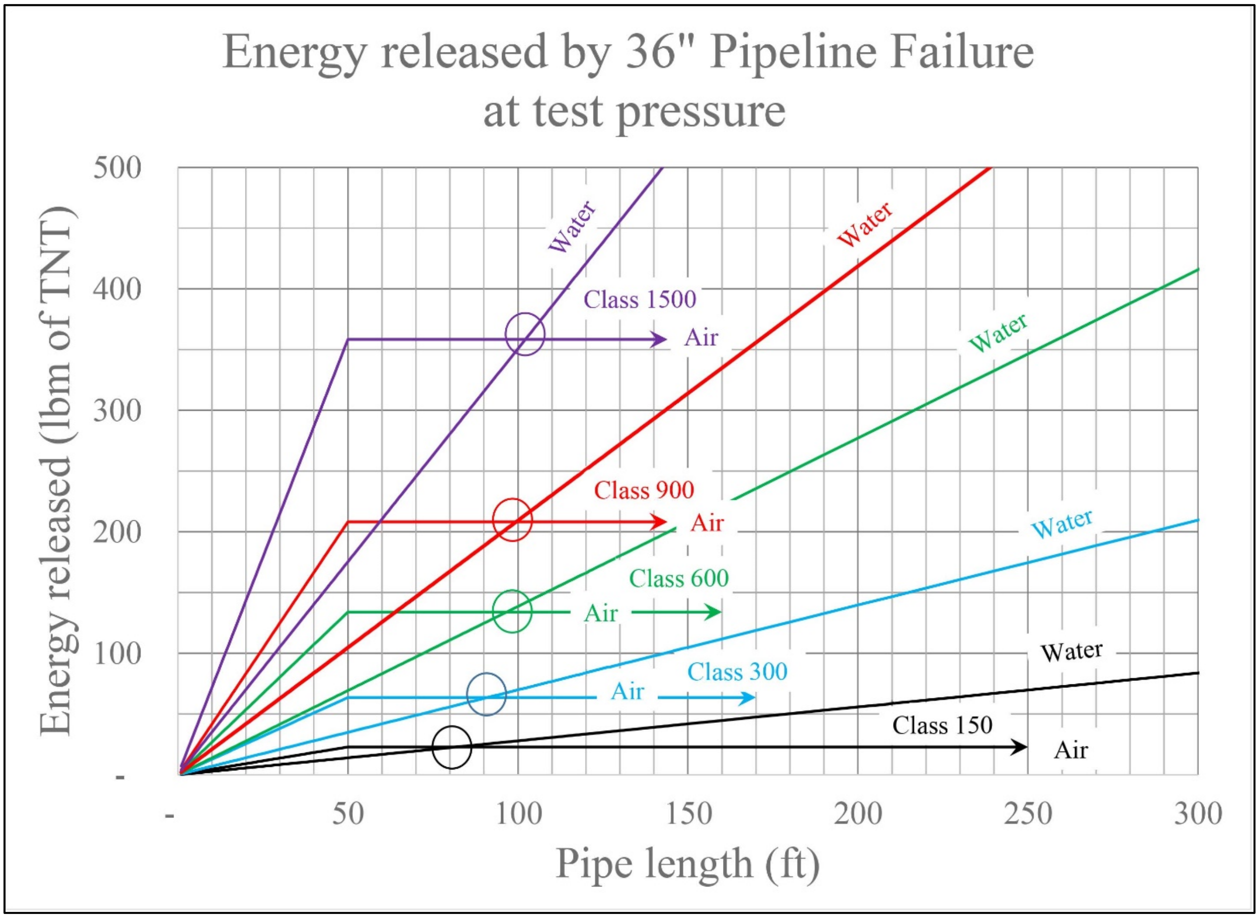

Figure 3. Pipe length vs. Energy released

2.3. Test failures

A static test procedure should include the conditions of success. For example, it is common to define "success" as "pressure at the end of the test greater than MAWP" and disallow adding any water during the test. Others will allow some small volume to be added (usually a number of injection pump strokes amounting to a very small volume per stroke) and "success" is defined as "maintain test pressure by adding fewer than x strokes". Either definition works fine, but the test must have a pre-defined success criteria to account for temperature changes and insignificant leaks at bolted connections. The static test is looking for actual points in the pipe subject to catastrophic failure.

At room temperature, most grades of steel are “ductile” (i.e., able to be deformed without losing toughness; pliable, not brittle) and an increasing level of stress will tend to tear the steel without losing any metal mass (left hand image in Figure 4). As temperature drops, steel will transition to “brittle” (i.e., having hardness and rigidity, but less tensile strength; breaking readily with a comparatively smooth fracture, right hand image in Figure 4). The point where a given steel makes this transition is called the “Ductile to Brittle Transition Temperature (DBTT)”. The DBTT is a function of the steel’s metallurgy and post-processing. For API 5L, Grade B pipe the DBTT tends to be around -15°F (-26°C), other pipe grades would be expected to have different DBTT values (with higher grades having a higher DBTT).

Figure 4. Steel Failure Modes

In a ductile failure, the metal around the failure point thins prior to failing, and the resulting failure in a pipe can look like the “fish mouth” failure (left hand image in Figure 4). Virtually all of the metal is still attached to the pipe and except in rare cases, no shrapnel is ejected. The case where shrapnel can be formed is a dead-end component such as a vent line or a tee with a blind flange on the branch where a circumferential ductile failure can release the vent line or blind flange. Even with these specific cases, the chance that the pipe will fail evenly around the entire circumference of the pipe and launch a projectile is exceedingly small. It is much more common for something less than 300° of the circumference to fail and fold over like a hinge. The likelihood of shrapnel being generated in a ductile failure is low enough to be disregarded.

A brittle failure (right hand image in Figure 4) is quite different. The edges of the failed steel tend to be full thickness with little or none of the thinning to be expected in a ductile failure. It can be expected that portions of the failed area can become projectiles, exiting the area at high speed. These breaks are most common along grain boundaries. Places where the grains are not homogenous can cause the crack propagation mechanics to turn the crack, even back on itself. When a crack creates an unsupported island, that island then becomes a projectile. The missing piece of the vessel in Figure 4 is approximately 2 ft^2 (0.186 m^2), 0.75 in (19.1 mm) thick, located 10 ft (3.1 m) above the ground. The mass of this projectile is 6.6 lbm (3 kg). The initial acceleration is (disregarding gravity and wind resistance for the acceleration period) is nearly 600 miles/s^2 (952 km/s^2). That acceleration results in a velocity 50 mS after the failure of 160,000 ft/s (48 km/s) or 137 Mach (the acceleration period ends prior to the establishment of the choked-flow mechanism so speed greater than 1.0 Mach can be achieved). For comparison, 30-06 round from a high-powered rifle with a 220-grain load has an initial velocity on the order of 2400 ft/s (731 m/s). A smaller or larger projectile would have the same acceleration and initial velocity since changing the impact area causes a proportional change to the mass.

At the end of explosive decompression, the projectile will have outrun the choked flow which will have been established at this point so there is no more acceleration in the horizontal (x) direction, and gravity provides acceleration in the vertical (y) direction. The kinematics equations describe the motion of this 6.6 lbm object that started its motion 10 ft above the ground. This projectile has a high likelihood of hitting something, if that collision coincidentally happened in 50 mS after separation, it would exert 635,000 lbf (2900 kN).

If a given project determines the DBTT is 32°F (0°C), then the test procedures could specify that the test cannot be started if the predicted ambient temperature has a reasonable chance of falling below 50°F (10°C) while the pipe is under test to provide confidence that the chance of brittle failure will be essentially zero.

3. HYDROSTATIC TESTS

For a hydrostatic test, you must acquire the water, fill the system in such a way to eliminate air pockets as much as possible, allow the water to "rest" to cause as much dissolved gas as possible to evolve out of the water to be vented, raise the pressure to the test pressure, hold the test pressure for a specified time period, depressurize, and dewater. Skipping the rest period is a common cause of a test failing when it the pipe integrity is intact and the line should have passed.

Water is classified as an "incompressible fluid", but that is not exactly correct. The bulk modulus of a fluid is defined as "the amount that pressure must be increased to reduce the volume by 1 percent". For water the bulk modulus (E) is 319,000 psig (2200 Mpag). Equation 3 results in a fairly small number, but not zero.

Figure 2 shows the amount of water that you would need to add to account for compressibility of the water. The density of pure water is 350.5 lbm/bbl (1000 kg/m^3), so in the 900-psig test of 16-inch 0.375 wall pipe in the example above you would need to add about 3000 lbm (1360 kg) or 8.6 bbl (1367 L) which 0.266 percent of the pipe volume or 45 times larger than the volume that had to be added due to pipe stretch.

Since liquid water is an excellent heat transfer medium and it freezes well above the DBTT discussed above, sensitivity to converting a ductile pipeline into the range where brittle failure is much less of a problem than it is with gas tests.

3.1. Hydrostatic test issues

Many thousands of hydrostatic tests have been conducted on Oil & Gas piping and the industry has grown very comfortable with the issues surrounding these tests, but that doesn't mean that the issues are trivial or can be ignored.

3.1.1. Acquiring water

The first issue is acquiring water which includes sourcing it, hauling it, and putting it into the line. The quality of the water is generally not a major issue with mild-steel pipe since it will only be in the line for a short time and the steel is very resistant to corrosion over short periods. Higher grades of steel and all stainless steels are quite susceptible to Chloride Stress Corrosion and other salt-specific corrosion modalities and using high salt-content water (e.g., produced water from Oil & Gas operations) should be avoided with higher grades of steel. It has been common to get water from city fire hydrants if the job is close enough to a town for this to be economic (and if the city has not passed ordinances to prevent taking water from a fire hydrant as many cities have done), from a river (if your project can purchase rights from the water-rights owners), from a stock pond, or produced water from an oil or gas well. The amount of water you need per 1000 ft (304.8 m) of pipeline length can be estimated by Equation 8. For the 16-inch 0.375 wall pipe above, you would need 232 bbl/1000 ft (121 L/m) or 1228 bbl/mi.

$$V_{system} = \left(\frac{\pi}{4}\right) \cdot ID^2 \cdot L_{pipe} = \left(\frac{\pi \cdot 1000ft}{4 \cdot (144)\frac{in^2}{ft^2}}\right) \cdot \left(\frac{bbl}{5.615ft^3}\right) \cdot ID^2$$
$$= 0.971\frac{bbl}{in^2} \cdot ID^2 \approx (ID^2)\frac{bbl}{1000ft} \qquad \textit{Equation 8}$$

Hauling water from the source to the test can be a major logistical effort. If a water truck holds 80 bbl (12719 L), and you are testing 15 miles (24 km) of 16-inch 0.375 wall pipe you would need 223 truckloads of water to fill the line. Depending on the distance required to travel to/from a water source it can easily require several days and considerable expense to shift this water. Then you have to get the water from the trucks into the pipeline. When you stack water vertically, the pressure at the bottom of the column is shown in Equation 9.

$$P_{bottom} = \rho \cdot \gamma \cdot \frac{g}{g_c} \cdot h = \left(62.428\frac{lbm}{ft^3}\right) \cdot \gamma \cdot \frac{\left(32.2\frac{ft}{s^2}\right)}{\left(\frac{32.2ft \cdot lbm}{s^2 \cdot lbf}\right)}\left(\frac{ft^2}{144 \cdot in^2}\right) \cdot h$$
$$= \left(0.433\frac{psi}{ft}\right) \cdot \gamma \cdot h \qquad \textit{Equation 9}$$

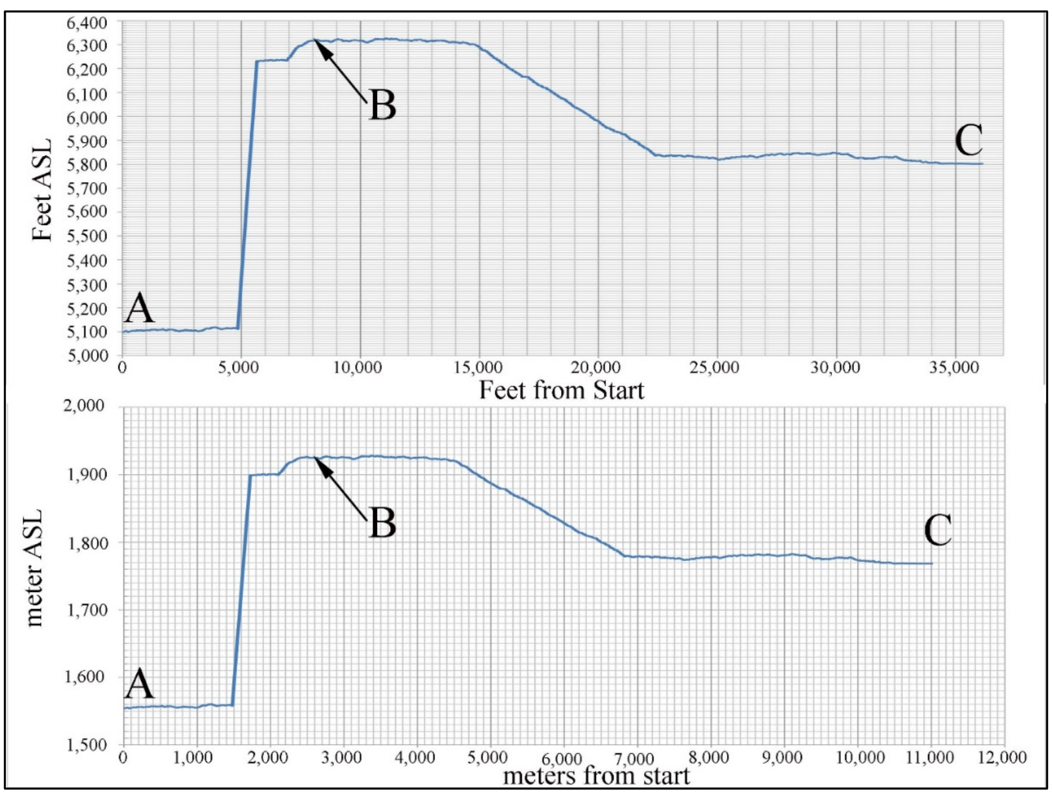

Figure 5. Pipeline profile

Maximum discharge pressure of the pumps on most oil-field water trucks is generally around 15 psig, so if the highest elevation of the pipeline is more than about 30 ft (9 m), then an auxiliary pump will be required to actually fill the line.

3.1.2. Pressure at the top/bottom of hills

Equation 9 shows the pressure gradient imposed by a column of water. No pipeline will ever be perfectly flat, and it is common in hill country for there to be a significant difference in elevation from the low point to the high point in a pipeline. If that elevation change is more than about 100 ft (30.5 m) then the designer of a hydrostatic test must begin being careful. Looking at Figure 5, the test designer has the option of placing the pressure recorder at "Point A", "Point B", or "Point C".

- If the recorder is placed at Point A (elevation 5,100 ft ASL (1554 m ASL)), the line is filled with 1.07 SG water, and the pressure is raised to 900 psig (6.2 Mpag) as used in the examples above, then the pressure at Point B (elevation 6,320 ft ASL (1,926 m ASL) will be 335 psig (2.31 Mpag), so the new MAWP must be limited to 267 psig (1.84 Mpag) by Figure 1 instead of the 600 psig (4.13 Mpag) that the system was designed for. Pressure at Point C (elevation 5,800 ft ASL (1768 m ASL)) would be 575 psig (3.96 Mpag).
- If the recorder is placed at either Point B or Point C and the pressure raised to 900 psig (6.2 Mpag), the pressure at Point A would be 1432 psig (9.87 Mpag) or 239 percent of MAWP.

This issue has caused a number of pipe failures at low points in mountainous terrain, and there is no solution that doesn't cause more additional problems than it solves.

3.1.3. Weight of water

In addition to the pressure that a column of water will apply to the constraining vessel at the bottom of the column, water itself has weight. The standard wall 16-inch pipe we have been using for examples weighs 62.77 lbm/ft (28.4 kg/m) and holds 9.5 gallons (36L) of water weighing (assuming 1.07 specific gravity) 45.7 lb (20.7 kg) for a combined weight of 108.5 lbm (49.2 kg). For buried lines this weight is not a factor, but pipe racks and fabrications designed to carry a pipe full of gas may very well collapse with nearly double the design weight.

3.1.4. Degas

Virtually any water source will have a non-trivial mass of dissolved gases. River or pond water will have dissolved air. City water will have gaseous biocides (chlorine is most common). Produced water will have dissolved natural gas. If the water is allowed to sit for a period of time, then the gasses will tend to evolve out of the water and collect at high points. Hopefully the pipeline design allowed for vents at major high points to allow the accumulated gas to be vented. Large pockets of gas can react badly to standing at high pressure.

- Increasing pressure increases the amount of gas that can dissolve in water, so over a period of hours large volumes of gas can be re-dissolved into the water, significantly lowering the pressure in the line and often causing a test to fail.
- Gas pockets have a larger change in their volume in response to a temperature change than the liquid-filled portions which can drop pressure more than expected, which can cause a test to fail.

For most source waters, a 24-hour soak period is adequate, but for city water it is better to allow it to degas for 48 hours.

3.1.5. Temperature changes

In a water filled system with no gas pockets and minimal dissolved gas, the water will try to expand (constrained by the pipe walls), the pressure will increase to a value proportional to the Coefficient of Thermal Expansion (β, which is temperature dependent, at about 60°F (15.5°C) it is 300×10^{-6} psi/°F) and the water will compress to a value proportional to the Bulk Modulus (E) that we discussed before. Equation 10 assumes that the pipe is infinitely rigid which is not quite true, but as we've seen above the pipe expansion is very small relative to the water expansion and can be safely disregarded.

Temperature-related expansion is a serious problem on any exposed piping or yard-tests of fabrications. For a buried pipeline, temperature increases on exposed piping will generally dissipate to the buried volumes and raise much more slowly.

$$\Delta P_{temp} = E \cdot \beta \cdot \Delta T \approx 70 \frac{psi}{°F} \cdot \Delta T \qquad \text{Equation 10}$$

3.1.6. Water disposal

Loading potable or near potable water into a pipeline will cause the water to pick up grease, oil, and mill scale. You can no longer just dump this water in a bar ditch, it must be properly disposed of. This water is now "industrial waste" not "Oil & Gas waste". Oil & Gas waste can be disposed of in wells classified as "UIC Class II" that are common in oil and gas fields and are generally quite inexpensive. UIC Class I wells are harder to find and much more expensive.

4. PNEUMATIC TESTS

Pneumatic tests are less common than hydrostatic tests, partly because of regulatory bias against them and partly because many engineers look at the amount of mass that must be added and are concerned about that mass contributing to a large explosion.

4.1. Pneumatic test issues

Gas tests avoid many of the issues discussed above. Acquiring air for a test simply requires deploying readily available rental compressors. A column of gas is compressible and the increase in pressure at the bottom of a fluid column is very low (Equation 11; for the example in Figure 5 if you put the recorder at Point "A", a 900 psig test would have 861 psig (5.9 Mpag) at Point "B"; and setting the recorder at Point "B" would have pressure at Point "A" of 940 psig (6.5 MPAg); both values quite acceptable). Gas is light and does not present a load issue on above-ground piping. There is no need to degas a gas test. Gas is a poor conductor of heat and gas tests do not have the heating/cooling issues of a water test. You dispose of the gas in most tests by venting the air into the air (or the nitrogen into the 80% nitrogen in the air), and if you do a natural gas test you can sell the gas you used for the test.

$$P_{bottom} = P_{testAtTop} \cdot exp\left(\frac{0.01875 \cdot \gamma_{gas} \cdot h}{T_{avg} \cdot Z_{avg}}\right)$$

$$P_{top} = P_{testAtBottom} \cdot exp\left(\frac{-0.01875 \cdot \gamma_{gas} \cdot h}{T_{avg} \cdot Z_{avg}}\right) \qquad \text{Equation 11}$$

While pneumatic tests have none of the issues that we see in hydrostatic tests, pneumatic tests have their own issues. A big one that gas is a poor heat transfer medium so very cold gas will chill the pipe without much ability to absorb heat from the bulk gas in the line. The risk of brittle failure is significant with many types of gas tests.

4.1.1. Choice of gases

Pneumatic tests are generally done with:

- Compressed air. Because of the temperature increase inherent in compressing air, it is generally injected into the pipeline at temperatures well above ambient and compressed air does not tend to contribute to a risk of brittle failure.
- Bulk nitrogen. Bulk nitrogen arrives on location as a saturated liquid at relatively low pressure and temperature around -319°F (-195°C) and it is heated to achieve the required pressure. The gas at pressure is superheated on the truck to the required injection temperature. If this parameter is not specified and carefully monitored then it is too common for the gas to enter the line well below the ductile-to-brittle-transition temperature (DBTT) discussed above—this failure to specify has resulted in many failed tests and too many injuries and even deaths. Nitrogen tests are not inherently more dangerous than other tests as long as proper engineering controls are in place, but monitoring the injection temperature is an extra task that has to be built into the test procedure.
- Natural gas (called "gas" in Figure 1). Natural gas tests are fairly rare because it is uncommon to have a source of gas at pressures above test pressure located in close proximity the line being tested. When those conditions are satisfied you still have to have a purchase agreement with the owner of the gas to take it for the test. If you have all of the necessary conditions, then you need to make sure that your test procedure includes a detailed purge procedure to ensure that the line is clear of oxygen prior to pressurizing it. At the end of the test, you can generally sell the test gas back to the supplier as long as there is a way to measure the sales gas (since you are going to use a significant quantity for the purge). Natural gas tests do not increase the risk of DBTT.

4.1.2. Soak periods

Because of the poor heat-transfer capabilities of gases, it is important to periodically allow a pause in the pressurization to allow stresses to move towards equilibrium. Tests will often have language like "Raise pressure to 50 psig at a rate of 5 psig/min; hold at 50 psig for 30 minutes while the line is inspected for leaks; raise pressure to 150 psig at a rate of 10 psig/min, hold at 150 psig for 30 minutes; etc." with soak points about every 10 bar.

4.1.3. Blowdown

The primary risk during a blowdown is Joule-Thomson cooling [1]. Equation 12 shows the J-T effect for a region that does not include an inversion point. The Joule-Thomson coefficient for air (μ_{jt}) at 50°F and 900 psig is 0.025078 R/psi (0.002021 K/kPa). Dropping the pressure on an air test from 900 psig to atmospheric pressure lowers the temperature of the fluid to 27°F (-2.8°C). This will often cause ice to form on the outside of the pipe and can cause higher grades of steel pipe to transition from ductile to brittle. There have been many reported cases of the connection point of vent lines, the vent pipe itself, and the vent valve becoming projectiles during an aggressive depressurization. This risk can be easily managed by specifying a rate of blowdown, but the procedure has to include this maximum blowdown rate, and the people executing the procedure must adhere to that limitation.

$$T_{final} = T_{initial} - \mu_{jt} \cdot \Delta P \qquad \textit{Equation 12}$$

5. BIAS IN FAVOR OF HYDROSTATIC TESTS

The 2010 edition of ASME B31.8 section 841.3.1c has the caution:

"The user is cautioned that the release of stored energy in a gas test failure can be significantly more hazardous than a similar failure with water. When testing with gas, a formal risk assessment is recommended, wherein risks are identified and appropriate mitigating measures and practices are identified and implemented to minimize these additional risks."

Previous editions of this document did not have a similar caution, and the design parameters for a pneumatic test in the 1999 edition are identical to the design parameters of the 2010 edition. This leads one to ask "What changed between 1999 and 2010?"

Considerable research has been done on the energy conversion in pressure vessels in a rupture, and this research has been independently repeated any number of times since the First World War. Work on quantifying the explosive force resulting from a failed gas-filled pressure vessel is quite robust [5] and the conclusions are:

- A failure of a high pressure, gas filled vessel can properly be called an "adiabatic" process (i.e., it occurs at constant entropy and is reversible).
- An adiabatic decompression results in a rapid conversion of the fluid potential energy to kinetic energy that is explosive [3].
- Essentially all of the gas involved in the explosive event will exit the vessel during the event.

- The dominant factor in the magnitude of the stored energy in a static test is the fluid added to elevate the system pressure.

These conclusions are specific to pressure vessels, where all of mass of the pressurized fluid is physically in relatively close proximity to any given position within the vessel. The last point applies to both gas and liquid tests, but the lion's share of the research has assumed that the added mass in a liquid test was de minimis and could be ignored.

NASA published a document towards the end of the 20th Century which has come to be known as the "NASA Glenn Research Center Methodology". This document was really the first time that anyone had made an effort to quantify the risk of pressurized-gas static strength-testing of pipelines as opposed to pressure vessels. This paper was on NASA's web site for several years but recent attempts to locate it have proven unsuccessful. Several regulations and many company policies have been written based on the NASA document. Basically this 2-page document said:

- A gas pipeline failure could properly be called an "adiabatic" process (i.e., it occurs at constant entropy and is reversible)
- An adiabatic decompression of a pipeline results in a significant energy release.
- All of the material in the pipeline will participate in the explosive decompression

Since these were fundamentally the same conclusions that researchers had reached for decades, there didn't seem to be much push-back from within NASA, but again all of the documents that might have shown the quality of review have been deleted from online sources and as far as the author can determine, few paper sources ever existed.

The problem with the NASA Glenn Research Methodology is that, as discussed above, an explosive decompression event is very short duration and far-distant fluids simply cannot arrive at the failure point in time to participate. If the transportation requirement was ever discussed prior to the publication of the Glenn Research Methodology, documentation of the discussions is not currently available.

If one assumes that the entire mass of the fluid within a pipeline system will participate in an explosive decompression, then it is prudent to specify exclusion areas and restricted distances to protect personnel from an explosive decompression event. The equations that the Methodology recommended (Equation 13) could be used to calculate a "restricted distance" (i.e., the closest safe point of approach while under test) have been repeated in numerous government regulations, industry standards, and company policies.

$$D_{NASA} = \left[\frac{P_{test} + P_{atm}}{1252.08} + \frac{(P_{test} + P_{atm})^2}{5.509 \times 10^6} - 2.0287 \cdot ln(P_{test} + P_{atm}) + 1.0812 \cdot ln(P_{test} + P_{atm})^2\right] \cdot \left(V_{system}^{1/3}\right) \cdot ft \qquad \text{Equation 13}$$

For our 16-inch 0.375 wall 1-mile test to 900 psig, Equation 13 works out to requiring an exclusion zone 886 ft (270 m) on either side of the pipe or 170 acres (69 hectare) for a pneumatic test, but no exclusion zone at all is required for a hydrostatic test.

This calculation demonstrates the fallacy of the Glenn Research approach—if a 100-mile (160 km) pipeline was operating at 300 psig (2.1 Mpag) (half of a 600 psig (4.2 Mpag) MAWP) the closest anyone could ever approach the line in operation would be 525 ft (160m) and it would cover 12,700 acres (5,150 hectares) that could never be approached by personnel while in operation. If the NASA Glenn Research Methodology had any validity at all, then every time a PSV lifted, or a rupture disk failed we would have a kiloton-range explosion on location since a pipe failure is simply opening the line to atmosphere very rapidly.

It has become impossible to identify why this bit of documentation from NASA has entered into codes, standards, regulations, and company policy, but it is clear that it has. One could conjecture that the prestige that affixes to NASA (i.e., "they put a man on the moon, after all") caused enough people to accept this flawed document unquestioningly at face value and when regulations and such were proposed they did not meet with significant resistance since anyone familiar with testing vessels would see it as obviously "correct". Regardless of the reason, these exclusion zones, cautionary statements, and strict policies have become ubiquitous. However, they have the unintended consequence that they actually increase the real risk of static testing.

6. CONCLUSION

Working with pressurized fluids has risks. It is incumbent on all of us to avoid any risks that can be avoided, minimize the risks that can be minimized, and design reasonable precautions around the remaining mix of risks.

For example, <u>avoiding risks</u> can be as simple as understanding the code prior to following it. ASME B31.8-2010 has the statement [7] (section ¶841.3.2(d), emphasis added):

> (d) Before being placed in service, a newly constructed pipeline system shall be strength tested for a minimum period of two hours at a **minimum pressure** equal to or greater than that specified in Table 841.3.2-1 after stabilization of temperatures and surges from pressuring operations has been achieved. **The minimum pressure shall be obtained and held at the highest elevation in the pipeline system.**

The discussion of hydrostatic head above should lead a test-designer to realize that the bold text in the quote says "put the test gauge at the highest point" which puts an unrecorded pressure on the low points—significantly increasing the risk to the community. A designer interested in actually lowering risk can avoid the risk of significantly over-stressing the pipe at the low points in a test in the mountains by performing a pneumatic test instead of a hydrostatic test which is contrary to the advice of ASME B31.8-2010.

One can minimize the risk of brittle failure by ensuring that the test fluid and ambient temperature are well above the ductile to brittle transition temperature (DBTT).

A reasonable precaution for any test is to define soak periods where the pressure is held at a designated intermediate value for a period of time (ranging from several minutes to long enough to check all of the key points on the line) to allow stresses to equilibrate. These soak periods are quite common in pneumatic test and generally unheard of in hydrostatic tests, but the reasons that they are used in pneumatic tests are just as valid in hydrostatic tests.

There are a number of flow charts from various industry organizations and consulting engineering firms that show a distinct bias for hydrostatic testing over pneumatic testing. Figure 6 is a flow chart that reverses this bias and can be used to actually help minimize the risks of testing.

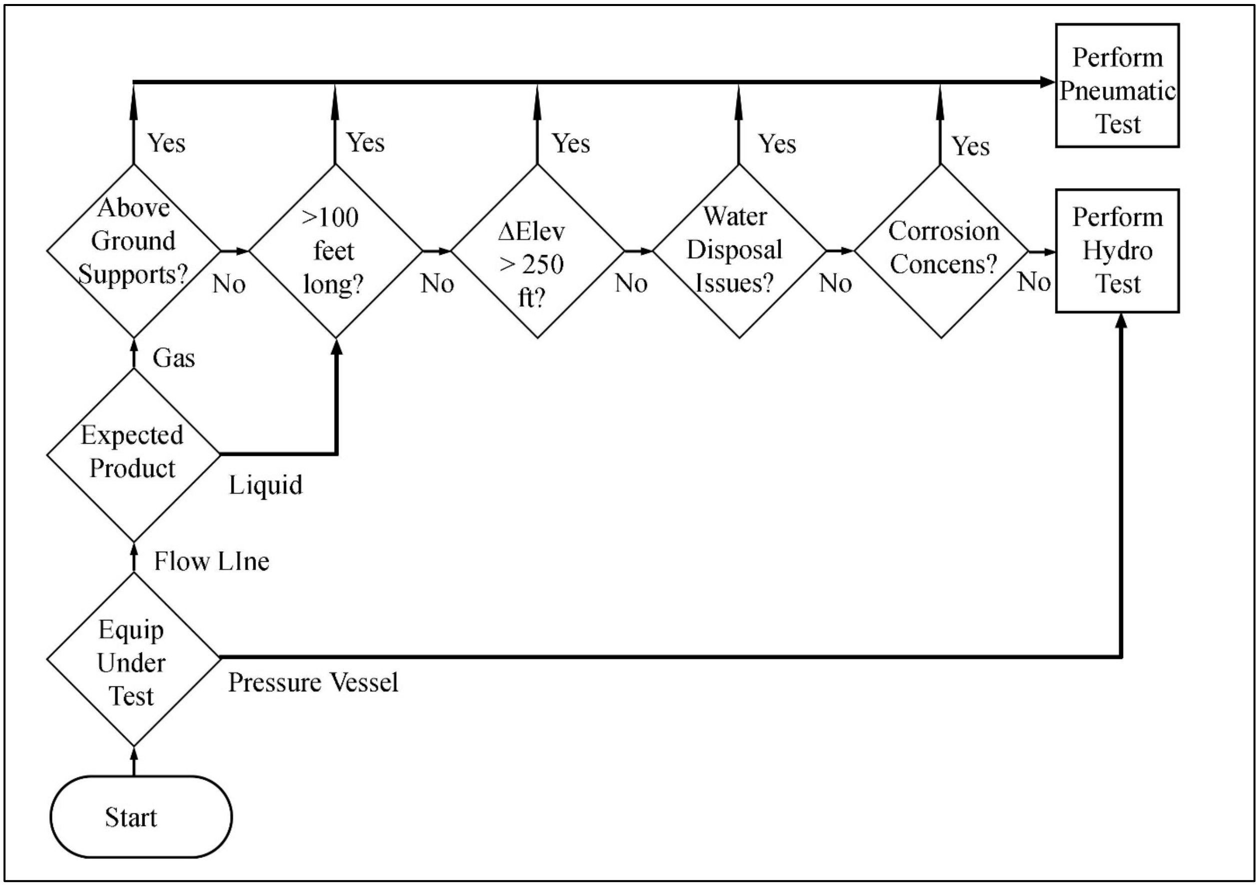

Figure 6.Static Testing selection flow chart

The current tendency to assume that hydrostatic testing is inherently safe and that pneumatic static testing is high risk is based more on an incomplete understanding of the possible forces related to stored energy than on actual risks. Any strength testing approached without proper respect for the mass and energy involved has a high potential for personal injury and property damage. Any strength testing approached with an adequate understanding of the risks and the important issues can reduce the potential for personal injury and property damage to an acceptably low level. These statements are true for any test fluid, and an arbitrary bias for one over the other limits the test-designer's options and can significantly increase total risks.

Regulations and standards that impose "exclusion zones" and "minimum approach distances" for pneumatic testing tend to provide a false sense of security for hydrostatic tests that can easily lead to dire consequences. As the examples in this paper have tried to illuminate, a pre-defined exclusion zone assumes a failure mode that may be very different from the more likely failure mode. There is no substitute for a competent analysis performed with a complete understanding of the issues followed by procedures that lead to a safe test.

There is some hope that the actual risks are beginning to be considered in standards documents. There was a change to ASME PCC2-2015 (Repair of Pressure Equipment and Piping) from 2015 to PCC2-2018 [8] (new material is emphasized):

> (e) The maximum calculated stored energy of any vessel or piping system being pneumatically pressure tested should not be greater than 271 000 000 J (200,000,000 ft-lb). **When calculating the stored energy for a vessel, the total volume shall be considered. When calculating the stored energy of a piping system, a maximum volume based on a length of 8 pipe diameters may be considered for any single failure analyzed.**

The source of the "8 pipe diameters" is not included in the document, and it seems short for small-diameter pipe, but it is decidedly a step in the right direction.

7. NOMENCLATURE

Symbol	Description	FPS	MKS
a	Acceleration	ft/s^2	m/s^2
A	Area	in^2	m^2
ASL	Elevation above sea level	ft	m
c_p	Gas specific heat capacity held at constant pressure	BTU/(lbm×R)	kJ/(kg×K)
c_v	Gas specific heat capacity held in a constant volume	BTU/(lbm×R)	kJ/(kg×K)
D	Pipe outside diameter	in	mm
D_{NASA}	Exclusion area from a pneumatic test	ft	m
E	Bulk modulus	psi	kPa
E_{joint}	Pipe Joint Factor		
g	Acceleration due to gravity	ft/s^2	m/s^2
g_c	Conversion from/to lbm/lbf	32.2 (ft×lbm)/(s^2×lbf)	
h	Height of fluid column	ft	m
ID	Pipe inside diameter	in	mm
k	Adiabatic constant (c_p/c_v)		
F	Location Factor		
f	Force of impact	lbf	N
L_0	Unstretched length	in	mm
L_{pipe}	Length of pipe in the system	ft	m
MW	Gas molecular weight	lbm/lb-mole	kg/kg-mole
OD	Pipe Outside Diameter	in	mm
P	Pressure	psia	kPaa
P_{atm}	Local atmospheric pressure	psia	kPaa
P_{bottom}	Pressure at the bottom of a column	psia	kPaa

P_{mawp}	Maximum allowable working pressure	psia	kPaa
P_{std}	Pressure at standard temperature and pressure	psia	kPaa
P_{test}	Designated test pressure	psig	kPag
$P_{testAtBottom}$	Pressure at test instrument located low in the system	psig	kPag
$P_{testAtTop}$	Pressure at test instrument located high on the system	psig	kPag
P_{top}	Pressure at the top of a column	psia	kPaa
$\bar{R}$	Universal gas constant	1545.3 ft×lbf/ R/lb-mole	8.314 J/ K/mole
R_{air}	Specific gas constant for air	53.355 ft^3×lbf/ R/lbm	287.068 J/K/kg
R_{gas}	Specific gas constant ($\bar{R}/_{MW}$)	R_{air}/γ_{gas}	R_{air}/γ_{gas}
S	Specified Minimum Yield Strength	psig	Mpag
S_H	Hoop stress	psig	kPag
t_{corr}	Pipe corrosion allowance	in	mm
t_{pipe}	Pipe wall thickness	in	mm
T	Temperature	R	K
T_{std}	Temperature of a gas at standard conditions (normally 60°F or 15.6°C)	R	K
T_{adj}	Temperature adjustment factor		
T_{test}	Temperature of a fluid at test pressure	R	K
v_{sonic}	Velocity of sound in the referenced medium	ft/s	m/s
V	Volume	ft^3	m^3
V_{added}	Volume added to a system to reach an elevated pressure	ft^3	m^3
$V_{evacuated}$	Volume of pipe that is emptied during an explosive decompression event	ft^3	m^3
V_{std}	Gas volume determined at a standard pressure and temperature	SCF	SCm
V_{system}	Volume of a system	ft^3	m^3
W_{gas}	Work done on or by a gas	BTU	J
Y	Young's Modulus (also Modulus of Elasticity)	psi	kPa
Z	Compressibility		

Z_{std}	Compressibility of a gas at standard temperature and pressure (1.0 for air)		
Z_{test}	Compressibility of a gas at test pressure and temperature (approx. 1.0 for air)		
ΔL	Change in characteristic length under stress	in	mm
ΔP	Change in pressure	psi	kPa
β	Coefficient of Thermal Expansion	psi/R	kPa/K
ρ	Fluid Density	lbm/ft^3	kg/m^3
γ	Fluid specific gravity		
μ_{jt}	Joule-Thomson coefficient	R/psi	K/kPa

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