Collective effects of leakage, temperature changes, and entrapped air during hydrostatic testing

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DURING HYDROSTATIC TESTING of pipelines, leaks in the test section will result in a loss of test pressure, and a reduction in water temperature during the test hold time will also typically result in a pressure decrease. This paper reports an analysis of the comparative effects of temperature, leakage, and trapped air on the pressure during hydrostatic testing.

Equations were developed to quantify the effects of temperature changes and leakage rates on the pressure during hydrostatic testing. Additional analysis was performed to investigate the effect of trapped air in the system on the test pressure. By comparing the relative magnitudes of these factors for a given pipeline geometry, insight can be gained into the minimum size of a leak that can be reasonably detected using hydrostatic testing.

The results of this analysis show that small leaks are most easily detected in short test sections with stable temperatures and when very little air remains trapped the piping. The magnitudes of pressure changes resulting from temperature variations and leaks depend on the starting temperatures and test pressures. The variations of the compressibility and the thermal expansion of the fluid are often neglected in test calculations, but can be important. The effect of temperature changes on the test pressure are minimized for cold temperature tests with water near 40°F, and increase as the test temperature increases. The presence of relatively small volumes of trapped gas can significantly affect the apparent compressibility of the test fluid. As a result, if a small leak is present, the pressure loss due to a leak may be less than expected. This effect becomes smaller at higher test pressures.

HYDROSTATIC TESTING IS commonly used as a means to verify the integrity of a pipeline. Testing prior to placing a pipeline in service is intended to demonstrate the adequacy of the materials and construction methods used, and periodic retesting can be used to show that the pipeline maintains adequate strength. While not the intended purpose, it is tempting to consider using hydrostatic testing as a means for detecting leaks in a pipeline segment. This study explores the sensitivity of the pressure drop due to leakage compared to other factors that cannot be completely controlled during hydrostatic testing.

A decrease in pressure during hydrostatic testing does not necessarily indicate the presence of a leak. It is well known that temperature variations will also result in pressure changes during testing, and that temperature changes can mask the effects of small leaks. Trapped air can also mask the presence of a leak, because gas pockets can significantly affect the apparent compressibility of the water in the pipeline. If a small leak is present, the pressure loss due to the leak may be less than expected because the expanding air in the pipeline will tend to keep the pressure constant. Trapped air also reduces the effect of temperature changes on pressure.

Hydrostatic testing of pipeline requires proper planning. Most companies have hydrostatic test manuals that detail the procedures to be followed for testing, and will also have acceptance criteria do determine the success or failure of testing. Usually, this work is performed by a hydrostatic testing contractor hired by the pipeline owner, or hydrostatic testing may be included as a part of the main construction contract.[1]

While the pressure measurements taken during hydrostatic testing are representative of the entire pipeline (elevation must be considered), the temperatures used to characterize the tests are much more difficult to specify. Typically, measurements are collected of the air temperature and the pipe temperature at a location close to the water injection point. A ground temperature measurement may also be recorded for buried pipelines, and is often taken at a single point in the backfill near the pipeline. Without a distribution of temperature measurements along the pipe, the assumption must be made that the temperatures are constant along the length of the test section of pipeline. For buried pipe, a period of time is typically provided between filling the pipeline with water and beginning the pressurization to allow the temperatures to equilibrate as much as practical. When unburied pipelines are tested, the temperature distribution becomes much more susceptible to variations in ambient temperature, sunlight exposure, and other weather conditions.

The presence of trapped air in the test section may mask the presence of a leak, but the amount of air present may be difficult to ascertain. An indication of the amount of entrained air can be obtained during pressurization. The presence of a volume of highly compressible air will reduce the rate of pressure rise in the test section. If a leak is present, the expansion of the air may prevent the detection of the leak. Even without air present in the system, small leaks may be difficult to detect, and since expanding air within the pipeline will result in slower changes in pipeline pressure, trapped air will make detection of leaks through hydrostatic testing even more difficult.

In the following sections of this paper, the effects of temperature changes, leakage, and trapped air on the pressure in a pipeline during hydrostatic testing will be examined. First, equations will be developed to calculate the pressure change in pipes as a function of the temperature and mass exchange. How the variation of water properties with temperature and pressure affects the results of the equations will be discussed. Next, several example calculations and a comparison with measured hydrostatic test data are provided. A method is introduced to account for the presence of trapped air in the pipeline during testing and the effects on the behavior are discussed. Finally, some conclusions of the analysis are presented.

Theoretical development

Effects of pressure and temperature on the volume of a pipe section

A section of a pipeline containing a fixed mass of liquid can be treated as a closed but variable size control volume. While no liquid is added to or removed from the system, heat can flow across the boundaries, allowing the temperature of the fluid to change. Because the temperature can change, the pressure and density of the contents can also change. The pipe walls are flexible and subject to thermal expansion, so the volume of the system will change as a result of heat transfer.

An increase in internal pressure causes a section of pipe to stretch, which increases its volume. An increase in temperature will also cause the pipe to expand and increase in volume. How much the pipe expands depends on the restraint of the pipe. For a short section of pipe in a lab, for example, the ends of the pipe will be capped and the internal pressure will act upon the close ends. Here, the pipe is acting as a cylindrical pressure vessel. In this case, the expansion occurs in two directions – the pipe gets longer and at the same time the diameter increases. This condition is referred to as “unrestrained”. For a long section of buried pipeline, however, the friction of the soil prevents the pipe from expanding along its axis. This is the “restrained” case, and the pipe is assumed to expand only in the radial direction due to the axial restraint provided by the soil. Any resistance to radial expansion of the pipe by the soil is neglected, and so the calculated expansion in this case is only approximate. The formulas for the effect of pressure and temperature on the volume of the pipe for the restrained and unrestrained cases are:

Restrained pipe:

$$\Delta V = V_1 \cdot \left[\frac{\Delta p\ D_1\ (1-\nu^2)}{t\ E} + 2(1+\nu)\alpha_L\ \Delta T\right] \qquad 2.1$$

Unrestrained pipe:

$$\Delta V = V_1 \left[\frac{\Delta p\ D_1 \left(\frac{5}{4}-\nu\right)}{t\ E} + 3\alpha_L\ \Delta T\right] \qquad 2.2$$

where ΔV is the change in volume, Δp is the change in pressure, ΔT is the change in temperature, V_1 is the initial volume, D_1 is the initial pipe diameter, t is the wall thickness, ν is Poisson's ratio, E is elastic modulus, and α_L is the linear thermal expansion coefficient. In both cases, it can be seen that an increase in either pressure or temperature will result in an increase in the pipe volume.

It should be noted that these equations were developed using linear elastic theory, and are therefore only valid until the pipe walls begin to yield. These equations as presented are not applicable when testing is to be performed above the (actual) yield stress of the pipe, and further analysis would be required.

Effect of leakage

The development of Eqns. 2.1 and 2.2 is based on the volume of the pipe itself, and assumes that the volume of the fluid is that same as the volume of the pipe that contains it. Therefore, these equations are valid with or without leakage, but they do not explicitly include the effects of mass loss from the volume. In this section, we will bring the effects of leakage (and at the same time, pumping) into the analysis by explicitly equating the volume of the pipe with the volume of the contained fluid.
For this analysis, the effects of a leak will be determined by the change in fluid mass contained in the pipe. Since we are considering the change in volume of the water with pressure, it is more straightforward to keep track of its mass than its volume. We will define the amount of mass leaked from the pipe as Δm. Assuming the amount leaked is much smaller than the amount of fluid originally contained in the pipe, the leaked mass can be linearized as:

$$\Delta m = \rho_1 \Delta V + \Delta\rho V_1 \qquad 2.3$$

where ρ_1 is the initial fluid density in the pipe and $\Delta\rho$ is the change in density of the fluid in the pipe over the duration of the leak. The error is on the order of the change in density times the change in volume, which should be very small.

With a goal of determining the change in pressure as a function of Δm and ΔT, we express the change in density as a function of the changes in pressure and temperature as follows:

$$\Delta\rho = \Delta p \left[\frac{\partial\rho}{\partial p}\right]_T + \Delta T \left[\frac{\partial\rho}{\partial T}\right]_p \qquad 2.4$$

Although water and other liquids are often considered incompressible, all materials have some amount of compressibility. The bulk modulus of a material refers to the reciprocal of compressibility of the material. The isothermal bulk modulus can be considered as a measure of how resistant a material is to compression when held at a constant temperature, and can be defined as

$$B = \rho_1 \left[\frac{\partial p}{\partial\rho}\right]_T \qquad 2.5$$

A large value of bulk modulus indicates a stiff, only slightly compressible fluid and a low bulk modulus indicates a fairly compressible fluid.

Making matters somewhat complex, at any given temperature and pressure, there are four different values of bulk modulus available, and they can be considerably different.[2] Two values are available for the bulk modulus of a fluid based on different thermodynamic processes, an isentropic value and an isothermal value. Then each of these values is commonly given as a "secant" value and a "tangent" value, based on how the testing was performed. The differences between the tangent and secant values can be found in various references, but from a thermodynamic point of view, the tangent bulk modulus is the "true" bulk modulus at given conditions.[3] The isentropic values are most commonly provided in references and so caution should be used to obtain the correct values for the analysis.

Using this definition, Eq. 2.4 can be rewritten as:

$$\Delta \rho = \rho_1 \frac{\Delta p}{B} + \Delta T \left[\frac{\partial \rho}{\partial T}\right]_p \qquad 2.6$$

where B is the isothermal bulk modulus and $\left[\frac{\partial \rho}{\partial T}\right]_p$ will be referred to here as the fluid thermal expansion factor, which can be determined from tables or an equation of state. The expansion due to temperature could be treated in similar manner as the isothermal bulk modulus term, resulting in a term functionally equivalent to the volumetric thermal expansion coefficient.

Solving the above equations for the change in pressure gives the following two equations:

For a restrained pipe:

$$\Delta p = \frac{\frac{\Delta m}{V_1} - \Delta T \left[2\rho_1 (1+\nu)\alpha_L + \left[\frac{\partial \rho}{\partial T}\right]_p\right]}{\rho_1 \left[\frac{D_1(1-\nu^2)}{t\,E} + \frac{1}{B}\right]} \qquad 2.7$$

For an unrestrained pipe:

$$\Delta p = \frac{\frac{\Delta m}{V_1} - \Delta T \left[3\rho_1 \alpha_L + \left[\frac{\partial \rho}{\partial T}\right]_p\right]}{\rho_1 \left[\frac{D_1}{t\,E}\left(\frac{5}{4} - \nu\right) + \frac{1}{B}\right]} \qquad 2.8$$

These equations can be used to calculate the change in pressure in a section of pipe due to temperature changes and the loss (or addition) of fluid mass to the pipe. As noted, the coefficients for the fluid properties vary with the temperature and pressure of the fluid. Therefore, the best accuracy can be achieved in the calculations for small changes in temperature and mass, and the resulting small changes in pressure. For significant changes in the temperature or mass of the fluid in the pipeline, it is recommended that the equations be solved in small steps, recalculating the coefficients at each step.

If the change in mass is set to zero, Eq. 2.8 becomes equivalent to the equation developed by Vejahati et al.[4] In their paper, the goal was to investigate how to detect leaks in shut-in pipelines filled with oil. However, the analysis does not explicitly include the mass loss. Instead they propose leak detection based on identifying pressure variations that are not consistent with the expected behavior of a pipe without leaks.

Software is available to calculate the properties of water as a function of the state. For the examples provided in this paper, the fluid properties were calculated using the REFPROP dynamic-link library [5,6,7] and the calculations were performed using software developed in-house.

Behavior of the developed equations

The results suggest that the effect of soil restraint is relatively small. For the pipe stiffness terms in the denominators of the equations, using a value of ν = 0.3 for steel pipe, $(1-\nu^2) = 0.91$ and $\left(\frac{5}{4}-\nu\right) = 0.95$. And for the pipe thermal expansion terms in the numerators, $2(1+\nu) = 2.6$ in the restrained case corresponds to a value of 3 in the unrestrained case.

Some general observations about the behavior of Eqns. 2.7 and 2.8 can be made. First, in both cases, the denominator is independent of the volume (and therefore the length) of the pipe. The temperature dependent terms in the numerators are also independent of the length. The result is that pressure change due to a change in pipeline temperature (assuming the pipe and the test fluid are the same temperature and that temperature is the same everywhere along the pipe) does not depend on the length of the pipe.

Somewhat more intuitively, the equations show that for a specified mass of water leaked, the pressure change is related to the volume, and therefore the length, of the pipe being tested. The mass loss terms in Eqns. 2.7 and 2.8 are inversely proportional to the volume of the test section, so the longer the pipe the smaller the pressure change. In other words, the longer the section of pipeline being tested the more difficult it becomes to identify a leak based on the resultant pressure change.

The value of the thermal expansion factor for a fluid will generally be negative (at a fixed pressure, the density decreases as the temperature increases). An important and relevant exception to this rule is the behavior of water between 32°F and 39°F at pressures near atmospheric. Some implications of this peculiar behavior will be discussed in the following paragraph. When the thermal expansion factor is negative, the equations show that the pressure can increase or decrease as a result of a change in temperature, depending on the relative values of the thermal expansion coefficient for the pipe material and the factor for the test fluid. If the pipe expands faster than the fluid as the temperature increases, the pressure will decrease. Typically, however, the fluid will expand faster than the pipe, and the pressure will increase with temperature.

At pressures relevant to pipeline testing, liquid water has a maximum density at approximately 39°F. This is why a pool of water begins to freeze at the top, since the colder water floats on the warmer water. This anomaly means that liquid water has a positive thermal expansion factor between approximately 32°F and 39°F (at low pressures). Above this point, the thermal expansion factor changes sign. Importantly, at temperatures near 39°F the thermal expansion factor of water is relatively small, and changes in the temperature will have less of an effect on the pressure of a hydrostatic test than when tests are performed with warm water.

Additionally, the equations show that the pressure change for a given mass loss or temperature change will be greater for smaller pipe diameters, thicker pipe walls, and larger values of bulk modulus (less compressible fluids in the piping).

Over the relatively limited temperature ranges which hydrostatic tests are typically conducted, the properties of the steel pipe used in the equations (i.e., the elastic modulus E, the Poisson's ration ν, and the thermal expansion coefficient α_L) will not vary significantly, and therefore can be treated as constants. On the other hand, variations in the density ρ, bulk modulus B, and the thermal expansion factor $\left[\frac{\partial \rho}{\partial T}\right]_p$ with temperature and pressure can have more significant effects on the results.

Figure 2-1 shows the bulk modulus of water as a function of the temperature and pressure. The figure shows that the bulk modulus increases with pressure and temperature over the range shown. As pressure increases, the bulk modulus of a fluid will increases as the size of intermolecular gaps decreases.[8] The increase in collision rates increases the effective stiffness of the fluid. Since the pressures encountered during testing will range from atmospheric to the maximum test pressure but the temperatures will usually not vary significantly, hydrostatic test calculations that fail to account for the variation in bulk modulus with pressure may result in significant errors. While the bulk modulus of most fluids is known to decrease with an increase in temperature, the isothermal bulk modulus of water increases with temperature over the range anticipated for hydrostatic testing purposes. The behavior of pipelines filled with oil may be considerably different from that during hydrostatic testing.

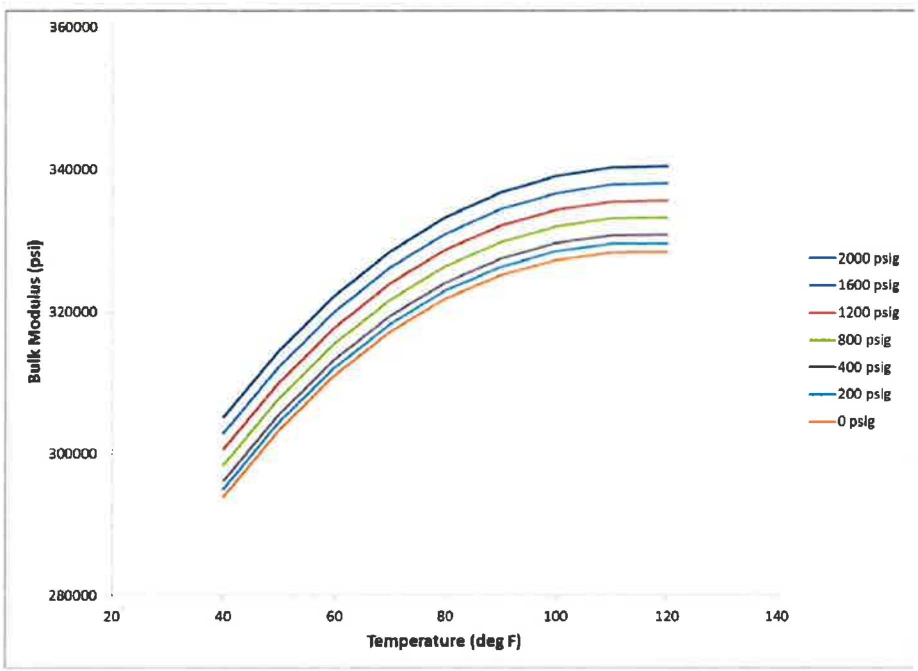

Figure 2-1. Effect of temperature and pressure on the isothermal bulk modulus of water.

The variation of the density of water with pressure and temperature is shown in Figure 2-2. While the density does appear in the equations, the variation of density with temperature and pressure over the ranges typically seen during hydrostatic testing are not very large. Of more significance is the thermal expansion factor $\left[\frac{\partial \rho}{\partial T}\right]_p$, which is the slope of the curves shown in Figure 2-2. The thermal expansion factor is plotted as a function of pressure and temperature in Figure 2-3.

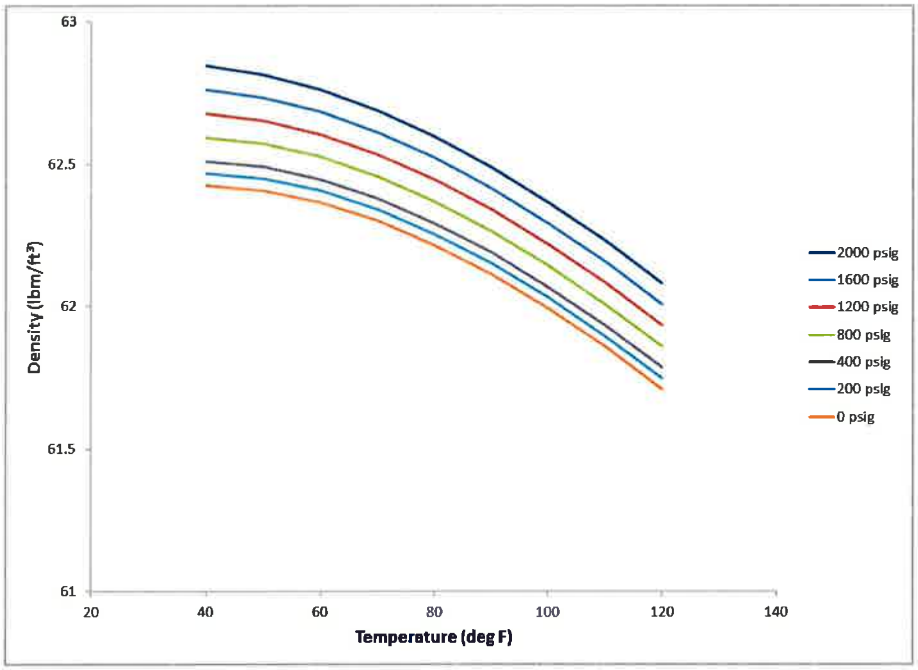

Figure 2-2. Effect of temperature and pressure on water density.

The thermal expansion factor determines how quickly the test fluid expands (or contracts) due to changes in temperature at constant pressure. If the term is negative, which is normal for most materials, increasing the temperature decreases the density (i.e., the fluid expands). Figure 2-3 shows that the thermal expansion term is not a strong function of pressure, but varies significantly with temperature. This figure shows that near 40°F, the thermal expansion term approaches 0. This is a result of water's unusual behavior of having a maximum density slightly above the freezing point. As mentioned above, the result is that for temperatures near 40°F water does not expand significantly as the temperature increases (and depending on the temperature may even contract). For hydrostatic tests performed at temperatures near the density maximum, a small change in temperature during testing will have a minimal effect on the test pressure and an increase in temperature may even result in a decrease in test pressure.

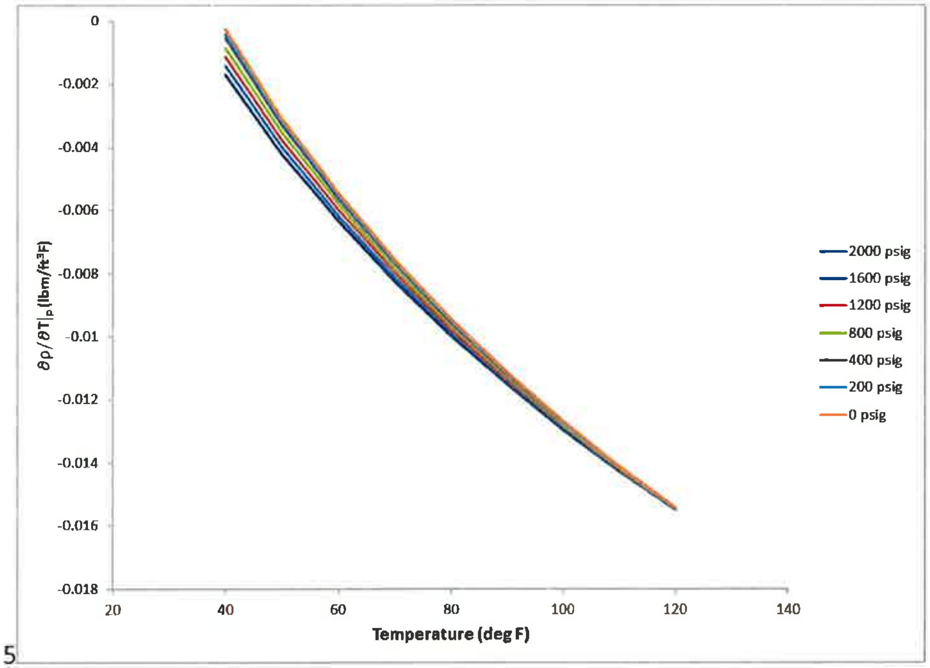

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Figure 2-3. Effect of temperature and pressure on the thermal expansion factor $\left[\frac{\partial \rho}{\partial T}\right]_p$.

Example calculations

To illustrate what the equations show about the detectability of leaks using hydrostatic tests, some examples will be provided. First, some example calculations will be used to demonstrate the behavior of the equations for leakage and small temperature changes under different test conditions. Second, the results of the developed equations will be compared to some actual pipeline test data.

Demonstration of the equation behavior

In this section, the behavior of the equations for leakage and small temperature changes under different test conditions will be investigated. The examples in this section are calculated for a pipe with the following initial properties:

- D_1 = 16 inch nominal
- t = 0.312 inch
- P_1 = 1720 psi
- T_1 = 70 F
- No trapped air in pipe

Recall that the change in pressure due to a change in temperature is not a function of the length of the pipe. For the 16-inch pipe and the water properties assumed in this example, the calculated change in pressure for a change in temperature is 24.6 psi/°F for restrained pipe and 23.7 psi/°F for unrestrained pipe. As anticipated, the effect of soil restraint on the results is small.

Next, assume that the example pipe under test leaks 8.3 lbm of water (approximately 1 gallon) during the test. Figure 3-1 shows a graph of the resulting magnitude of the pressure change as a function of the length of the test section. The results shown are calculated for a constant temperature. The results show that for a 100 ft section of pipe, the pressure will decrease approximately 205 psi during the test – which will clearly be obvious in the test results. However, for a section of pipe 10,000 ft long, the pressure drop due to the leakage is calculated to be just 2.1 psi. Due to the necessary allowance for temperature effects (recall that a 1°F temperature change can result in a 24 psi change in pressure), a leak of this size could not reasonably be detected in this case. The results are shown for both restrained and unrestrained pipe, and the difference is not significant.

Note that assuming a constant mass loss for different lengths of pipe isn't really realistic. For a fixed size hole, the leak rate will decrease with the internal pressure, and the pressure will decrease much faster in a short length of pipe, so less mass will be lost through the same size leak in the same time. Still, it makes for an easy comparison and provides a helpful example.

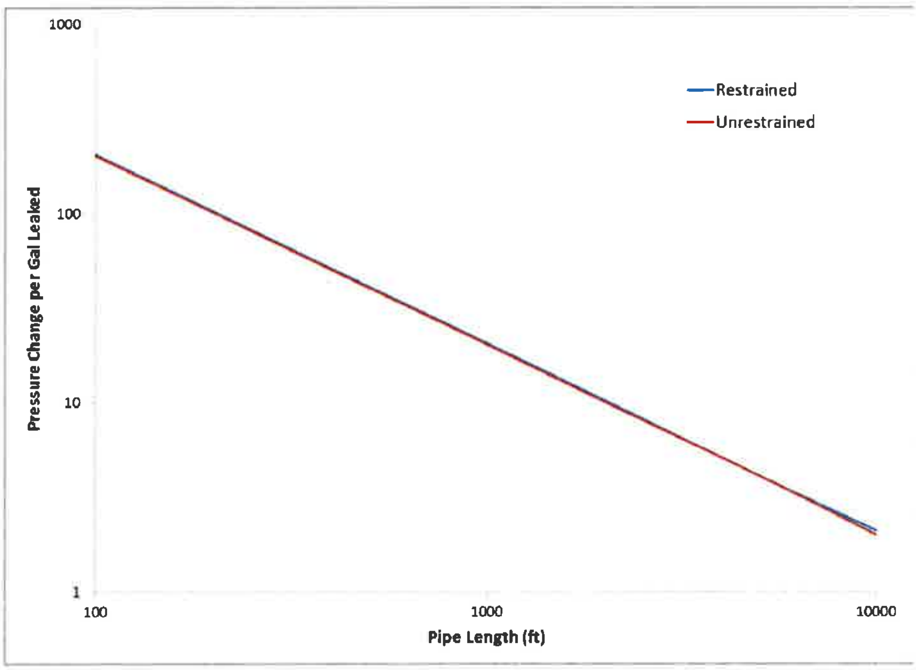

Figure 3-4. Variation in pressure for 8.3 lbm (approx. 1 gal) leak from the example pipe as a function of the test section length.

Since the thermal expansion factor varies significantly with temperature, the change in pressure that results from temperature changes during testing is also a function of the starting temperature of the water. To illustrate this, we calculated the pressure change that would result from a *1°F temperature increase* as a function of the starting temperature. The results are shown in Figure 3-2. The graph shows, for example, that a change from 70°F to 71°F increases the pressure approximately 24 psi. The graph also shows that for a test with water starting at 40°F, increasing the temperature 1°F during the test will increase the pressure only about 1 psi. Note that the calculated values are for 16-inch x 0.312-inch pipe starting at 1720 psig, and the graph will change for different pipelines and conditions. For this calculation, there was no trapped air in the pipe or water leakage.

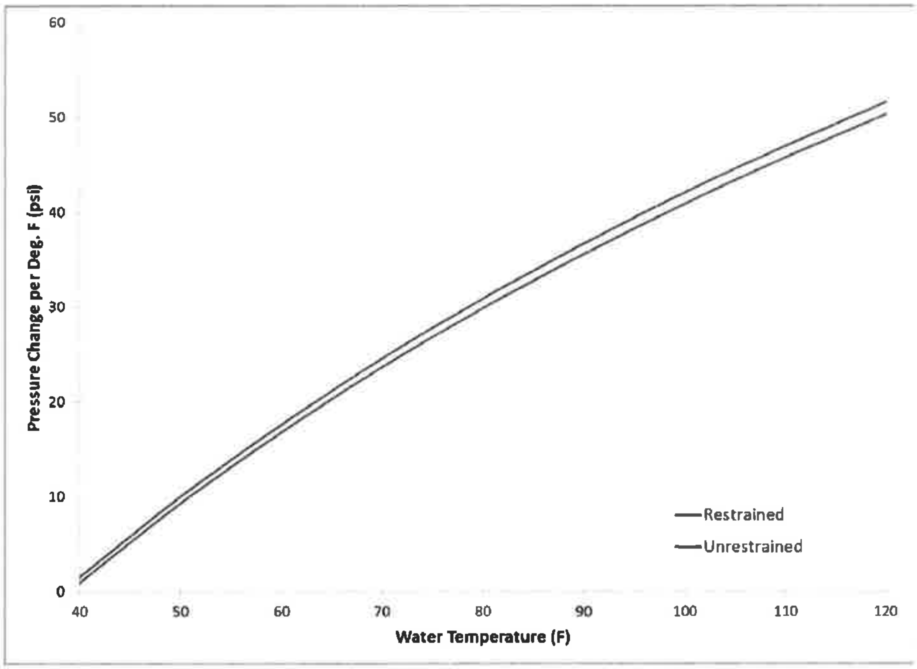

Figure 3-5. Pressure change due to 1°F change for different water temperatures in a 16-inch x 0.312-inch pipe at 1720 psig.

Comparison to test data

The measured data shown in this section was collected during hydrostatic testing of a buried pipeline. The pipeline was 30-inch OD x 0.312-inch wall thickness and grade X-65. The test section reported here was approximately 113,800 ft in length, and only 150 ft was exposed during testing. The temperature measurements are provided; air temperature, pipe temperature, and ground temperature, though no details were provided as to where these temperatures were recorded.

Pressure and volume of water added were recorded during pressurization of the test section. The volume of water added was calculated based on the number of injection pump strokes. During pressurization, the temperatures of the pipe and the ground are listed as 48.7 and 46.3, respectively. Figure 3-3 shows a plot of the injected water volume vs the pressure for a portion of the pressurization process. From the recoded volume and pressure data, values of the rate of change of pressure with the mass of water added were calculated and are shown on the plot. While one would expect that the calculated value would exceed the measured value due to the likely presence of air trapped in the pipe (discussed later in the paper), these values are quite close, considering that many assumptions and approximations were made in the analysis. A potential reason for the measured volume estimate to be high is that the pump may have been operating at less than 100% efficiency, but without investigating at the time of testing, this is only speculative.

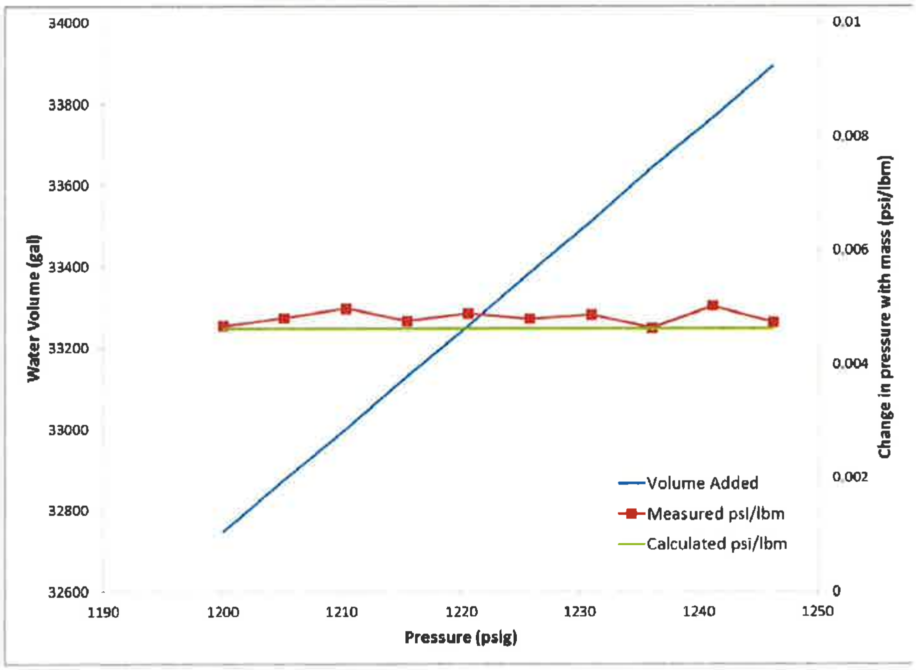

Figure 3-6. Relationship of pressure and water addition during pressurization phase of testing.

Another comparison was made with the data acquired during the test hold period. Figure 3-4 shows the measured pressure over time, along with the measured ground and pipe temperatures. During the period of the test shown, the ground temperature stayed fairly constant, while the reported pipe temperature varied nearly 6°F during the test. An analysis was performed to predict the change in pressure that would be anticipated during testing based on the measured temperatures. Calculations were performed starting at the initial measured pressure using both the measured ground and pipe temperatures. The calculations based on the reported pipe temperature indicate that if this was an accurate representation of the temperature of the pipeline and the contained water, a pressure change on the order of 50 psi would be expected. The measured ground temperature changed much less, and therefore the predicted change in pressure is also small. The calculated pressure change based on the ground temperature provides a much better math to the measured pressure. Note that it is not known if either of these temperatures is representative of the temperature of the pipeline and test fluid over the entire length of the pipeline.

It is also interesting to note that the measured pressure in the pipeline during the hold period decreased slightly, by approximately ½ psi. This is in contradiction to the trends of the temperature plots, which suggest that the pressure should have increased over the test period if the mass of fluid was constant. Since the change in pressure was within test tolerance, the hydrostatic test was considered to be successful, but the pressure loss was not explained.

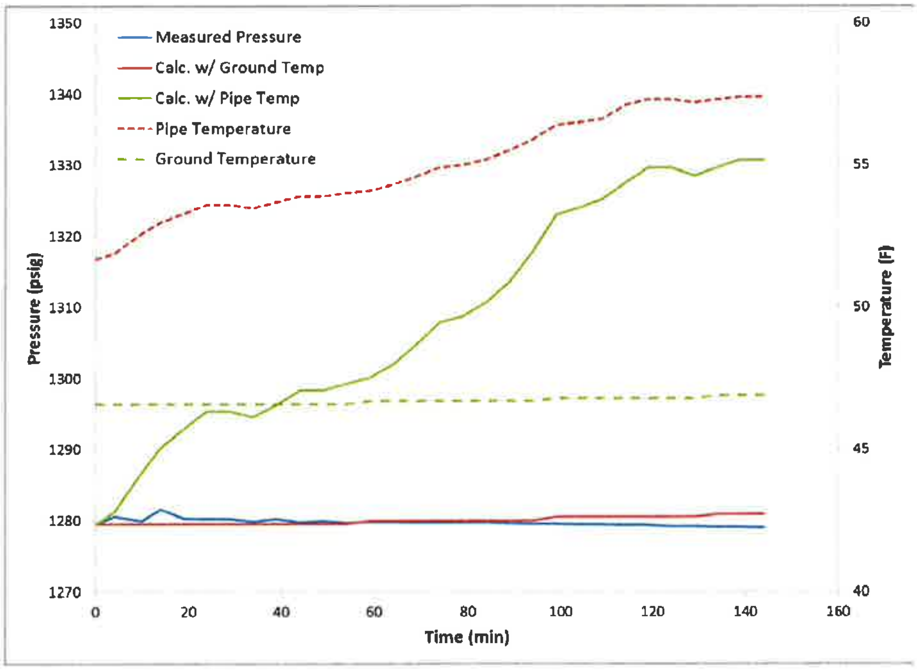

Figure 3-7. Comparison of measured pressure during testing and calculations using the measured pipe and ground temperatures.

Effects of trapped air

Air trapped in the pipes will significantly change the effective compressibility of the fluid in the pipe. The effects of any trapped air needs to be accounted for in the calculation of pressure changes due to temperature changes and leakage. Rather than developing the equations to explicitly account for the presence of gas in the pipeline, the effect of trapped air will be brought into the pressure calculation approximately by considering the effective properties of the combined water and air in the pipe.

The first way that the effect of trapped air will influence Eqs. 2.7 and 2.8 is through the isothermal bulk modulus term B in the denominator. Air is much more compressible than water, so the presence of air pockets causes an increase in the effective value of B. Larger values of B reduce the effect of both temperature changes and leaks on pressure in the pipe.

Air can be present in the pipeline in three forms [9]:

- Free air pockets trapped in part of the piping system. These can often be removed from the piping by accurate venting at high points.
- Entrained air bubbles that are distributed through the piping. Unless an emulsion is formed, these will usually coalesce over time to form free air pockets.
- Dissolved air spread throughout the fluid. Dissolved air is not visible, and the fluid is all liquid phase.

The presence of free or entrained air in a pipe filled with liquid can significantly reduce the effective bulk modulus compared to the bulk modulus of the pure liquid. On the other hand, test data indicate that as long as the air is in solution, it does not affect the fluid bulk modulus.[9]

The solubility of gases in liquids is a function of the pressure and temperature. At a given pressure and temperature, dissolved air should remain dissolved. However, increasing the temperature or lowering the pressure may result in air coming out of solution, and this process occurs at some finite rate. Therefore, depending on the operating conditions in which the fluid is subjected, it is possible for the dissolved air to become entrained (and vice versa). Air pockets that exist when a pipeline is filled for hydrostatic testing may become dissolved at test pressures, causing the effective bulk modulus to vary during the test period. This potential effect has not been investigated for this paper.

The second way that the effect of trapped air will affect the calculations is through the thermal expansion factor $\left[\frac{\partial \rho}{\partial T}\right]_p$. As with the bulk modulus, the thermal expansion of the gas portion of the fluid may vary significantly from the expansion of the liquid phase portion. It is not as clear if dissolved gases will alter the thermal expansion factor, but since the quantity of dissolved air is typically low, the effect of air dissolved in hydrostatic test water will not be considered in this analysis. A method of approximating the effect of free air pockets and entrained air on the effective thermal expansion factor will be presented later in this paper.

In order to investigate the effect of free air pockets and entrained air on hydrostatic testing, it is first necessary to define how the vapor fraction will be handled for the analysis. The volume of air pockets and bubbles will change significantly compared to the water due to pressurization of the pipeline. The volumetric fraction can be defined at test conditions or it can be defined at atmospheric pressure and corrected to test conditions. For convenience of the analysis, the volumetric fraction at test conditions will be used in this paper. The volumetric fraction is defined here as:

$$X = \frac{V_g}{V_l + V_g} = \frac{V_g}{V_c} \qquad 4.1$$

where V_g is the volume of gas in the pipeline at test conditions, V_l is the liquid volume, and V_c is the combined total volume of the test section.

In this analysis, air pockets and entrained air are treated as identical for the purposes of analysis. This is considered a reasonable assumption, since the pressure changes are typically slow, and the pressure and temperature are considered to be uniform throughout the volume of the piping at all times. Local variations in vapor fraction and compressibility will become important if fast transients or acoustic phenomena are being investigated.

Effective bulk modulus

During hydraulic testing of a pipeline, the test fluid is typically a mixture of water, dissolved air, and air bubbles and pockets (air bubbles and pockets are functionally the same in this analysis). The relative amounts of water and air as well as the test pressure and temperature will determine the effective isothermal bulk modulus. To account for the effects of these variables, different models of varying complexity have been developed. For mixtures of liquids and gases, these models can be divided in two types [3]:

- "Compression only" models that only consider the volumetric compression of the air; and
- "Compression and dissolve" models that consider both the volumetric compression of the air and the volumetric reduction of the air due to air dissolving into solution.

As stated above, the process of air coming in and out of solution is beyond the scope of this analysis, and therefore a model of the "compression only" type will be used.

The approach taken here roughly follows Wylie and Streeter[10] and Merritt[11]. Starting from the volumetric definition of the isothermal bulk modulus, we define an effective bulk modulus for the fluid mixture contained in the volume as

$$B_{eff} = -V_c \left[\frac{\partial p}{\partial V_c}\right]_T \qquad 4.2$$

Exchanging small finite changes for the differentials and inverting the equations gives:

$$\frac{1}{B_{eff}} \cong -\frac{1}{V_c}\frac{(\Delta V_l+\Delta V_g)}{\Delta p} \qquad 4.3$$

Because we defined the volumetric fraction at the test conditions, it is simple to separate the right hand side into the terms of the component values of bulk modulus. The effective bulk modulus for the two phase test fluid mixture can then be written:

$$B_{eff} \cong \left[\frac{X}{B_g} + \frac{1-X}{B_l}\right]^{-1} \qquad 4.4$$

where B_g and B_l are the isothermal bulk modulus values of the gas and liquid phases, respectively, at the current conditions. Note that because the effective bulk modulus depends on the volumetric fraction at the current conditions as well as the gas and liquid phase bulk moduli, the volumetric fraction should be updated in steps to preserve accuracy if the calculated pressure change is large.

Figure 4-1 shows the effect of trapped air in the piping system on the effective bulk modulus. The results are graphed based on the volumetric fraction of air at the specified pressure. Note that changing the pressure changes the density of the air and changes the volume fraction of air in the pipeline. Therefore, a large volumetric fraction of air at atmospheric pressure may represent a much smaller fraction at test pressure. For low pressures, even a small amount of air in the system has a large effect on the bulk modulus of the system. At typical hydrostatic test pressures of 1,000 to 1,500 psi, the effect of a small volume percentage of air is still significant, though the effect decreases with increasing fluid pressure. The results have been plotted at 40°F and 80°F to show that temperature also has a significant effect on the effective bulk modulus.

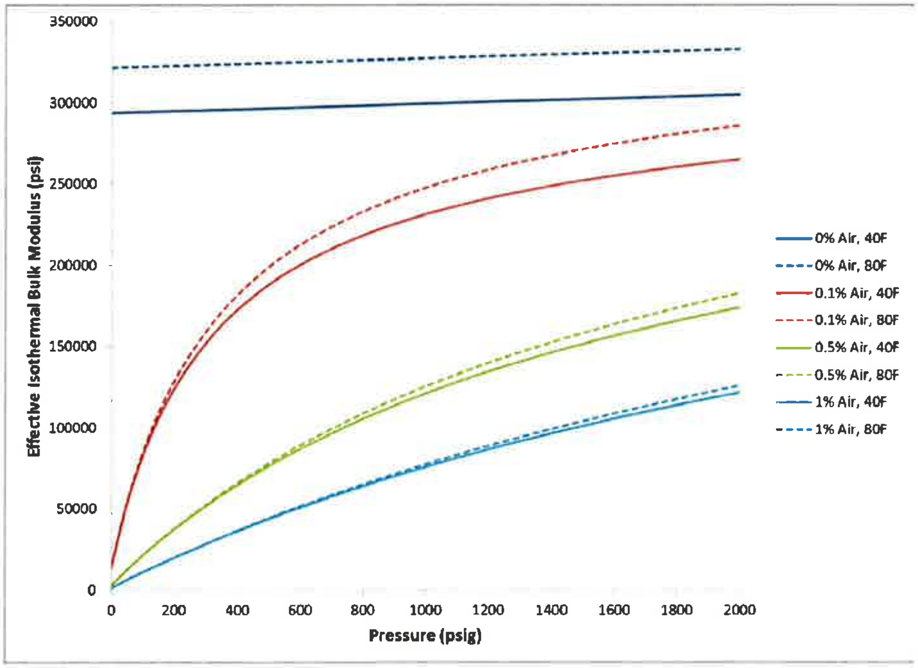

Figure 4-8. Effect of small amounts of trapped air on the effective bulk modulus of the test fluid.

Effect of air on the effective fluid thermal expansion

Like the compressibility and the effective bulk modulus, the presence of air bubbles in a pipeline during hydrostatic testing will also have an effect on the thermal expansion properties of the fluid mixture. This effect will influence the test pressure through the fluid thermal expansion factors in the numerators of Eqns. 2.6 and 2.7.

Since an approach to combining the fluid thermal expansion factors to determine an effective factor for the combined water and air in the pipeline is not obvious, we will start by casting the thermal expansion factor in terms of the volumetric expansion factor, which can then be combined in a straightforward manner. The isobaric volumetric thermal expansion coefficient is defined as:

$$\alpha_V = \frac{1}{V}\left[\frac{\partial V}{\partial T}\right]_p = -\frac{1}{\rho}\left[\frac{\partial \rho}{\partial T}\right]_p \qquad 4.5$$

This shows that the volumetric thermal expansion coefficient is equal to the negative of the isobaric thermal expansion factor divided by the fluid density.

Using an approach similar to that used for the bulk modulus, a method for developing an effective thermal expansion coefficient to account for the present of gaseous air can be obtained. Exchanging small finite changes for the differentials and inserting the volumes of the gas and liquid phases gives:

$$\alpha_{eff} = \frac{1}{V}\left[\frac{\Delta V_l + \Delta V_g}{\Delta T}\right]_p \qquad 4.6$$

which can then be separated to show:

$$\alpha_{eff} = \frac{V_l}{V}\left[\frac{\Delta V_l}{V_l\,\Delta T}\right]_p + \frac{V_g}{V}\left[\frac{\Delta V_l}{V_g\,\Delta T}\right]_p = (1-X)\alpha_l + X\alpha_g \qquad 4.7$$

where α_g and α_l are the isobaric volumetric thermal expansion coefficients of the gas and liquid phases, respectively, at the current conditions. This demonstrates the intuitively obvious result that the volumetric expansion of a system with two separate phases will be equal to the sum of the expansions of the individual phases. The effects of air dissolved in water or water vapor in the air have not been included in this simple model.

Figure 4-2 shows how a small amount of gaseous air will affect the effect isobaric volumetric thermal expansion coefficient for the test fluid. The graph shows that trapped air in the system increases the thermal expansivity of the test fluid, which tends to increase the pressure response to a change in temperature. However, the effect is not large for small amounts of trapped air, and the effect of the air on the effective bulk modulus will dominate the response. The calculation has been performed at 40°F and 80°F, and the results show that the effect of temperature on the thermal expansion coefficient may be more significant than the effect of small amounts of trapped air.

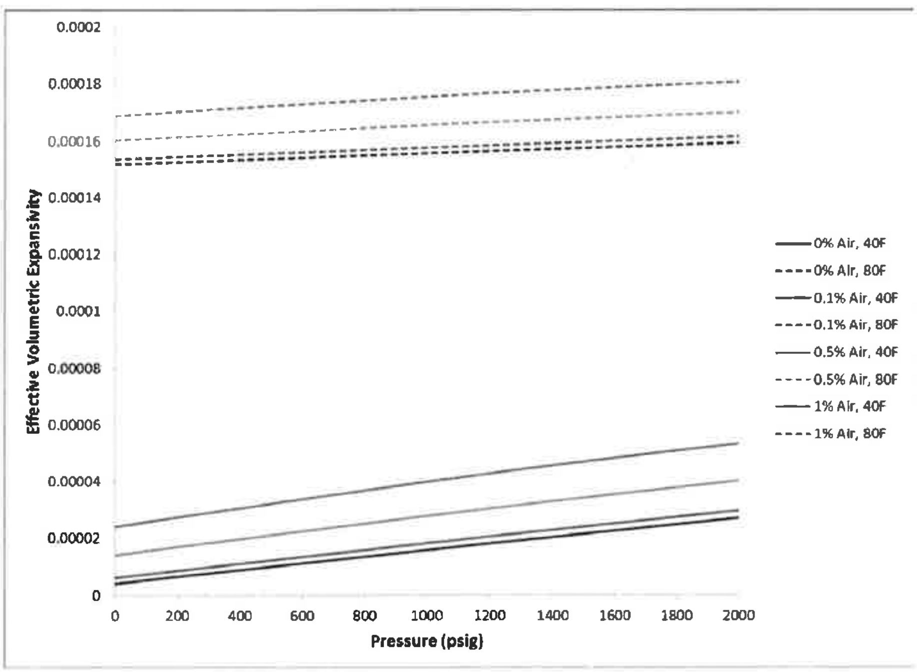

Figure 4-9. Effect of small amounts of trapped air on the thermal expansion coefficient of the test fluid.

To demonstrate the effect of trapped air on how the pressure will respond to leakage, the results for the example case with 8.3 lbm of water leaking from a pipe (from the examples above and shown in Figure 3-1) were recalculated for 0.1% and 1% air trapped in the system. As before, the temperature was assumed to remain constant during the leak. Figure 4-3 shows the results. Having 0.5% air trapped in the system reduces the change in pressure by nearly a factor 2, and as little as 0.1% trapped air reduces the change in pressure by approximately 10%. The effect of trapped air on the pressure change due to temperature variations will be similar. This shows that small amounts of air can significantly affect the results, reducing the pressure change due to a leak, and making it more difficult to detect.

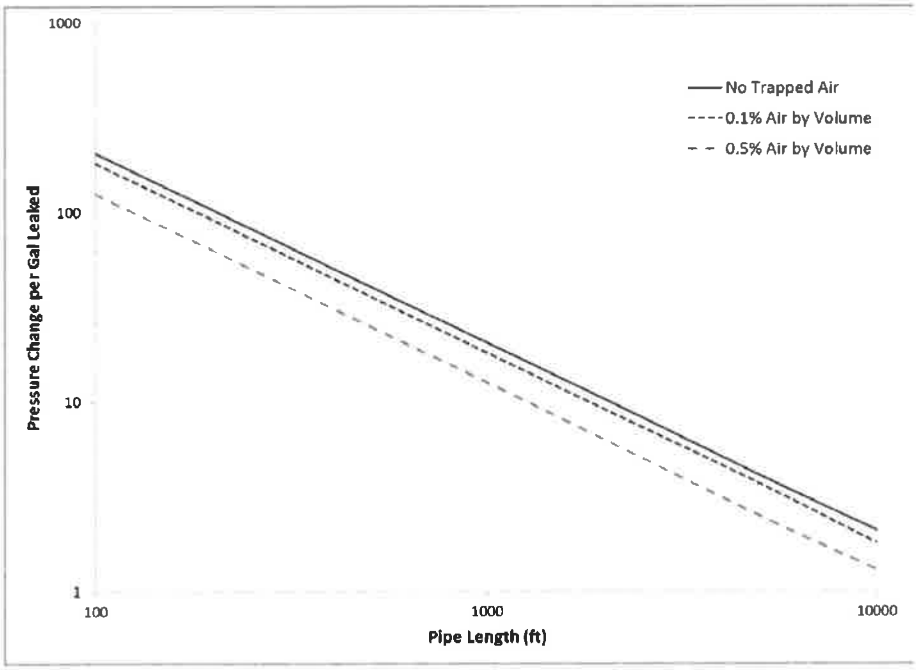

Figure 4-10. Effect of trapped air on the pressure drop due to a 8.3 lbm (approx. 1 gal) leak from the example pipe as a function of length.

Conclusions

Equations that can be used to calculate the effects of temperature changes and leakage on the pressure inside the pipe during hydrostatic testing have been developed. Also, the significance of the changes in properties of water with temperature and pressure on the calculations has been investigated. The equations can be used to evaluate the loss in pressure in the pipe due to a leak during testing.

The results show that small leaks become more difficult to detect as the length of the hydrostatic test section increases. Whether or not a small leak can be detected is not necessarily determined by the precision of the pressures, because the effects of leakage can be masked by temperature changes and the effects of trapped air in the piping. In addition, temperature variations over the length of the pipeline may be impossible to determine and have an influence on the accuracy of the results. Small leaks are most easily detected in short test sections with stable (and uniform) temperatures and when as much air has been removed from the piping as possible. Test procedures must provide for some variation between the initial and final pressures, and this will ultimately determine the detectable size of a leak.

The pressure changes resulting from temperature changes and leaks depend on the starting temperatures and test pressures. For accurate calculations, it is important to consider the effects of the test temperature and pressure on the bulk modulus and the thermal expansion coefficient of the test fluid. The simplified approach to calculating these effects often used for hydrostatic test calculations often neglect to accurately account for these properties. The effect of temperature changes on the test pressure are minimized for cold temperature tests with water near 40°F.

Trapped air can mask the presence of a leak because the presence of undissolved gas significantly affects the apparent compressibility of the water in the pipeline. If a small leak is present, the pressure loss due to the leak may be less than expected because the expanding air in the pipeline will tend to keep the pressure constant. Trapped air also reduces the effect of temperature changes on pressure.

References

1. Kiefner, J.F. and Maxey, W.A., "The Benefits and Limitations of Hydrostatic Testing", In: McAllister, E.W., Ed., *Pipeline Rules of Thumb Handbook*, 7th Ed., Boston, MA, Gulf Prof. Pub., pp. 145-168, 2009.
2. Hayward, A. T. J., "Compressibility Measurements on Hydraulic Fluids", Hydraulic Pneumatic Power, Vol. 11(131), 1965, pp. 642-646.
3. Gholizadeh, H., "Modeling and Experimental Evaluation of the Effective Bulk Modulus for a Mixture of Hydraulic Oil and Air", PhD Thesis, University of Saskatchewan Saskatoon, 2013.
4. Vejahati, F., Rostamy, N., and Noroozi, N., "Leak Detection for Shut-In Pipelines", IPC2016-64675, Proceedings of the 2016 11th International Pipeline Conference, Calgary, Alberta, September 26-30, 2016.
5. Lemmon, E.W., Huber, M.L., McLinden, M.O. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP*, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, 2013.
6. Wagner, W. and Pruss, A., "The IAPWS Formulation 1995 for the Thermodynamic Properties of Ordinary Water Substance for General and Scientific Use," J. Phys. Chem. Ref. Data, 31(2):387-535, 2002.
7. Lemmon, E.W., Jacobsen, R.T, Penoncello, S.G., and Friend, D.G., "Thermodynamic Properties of Air and Mixtures of Nitrogen, Argon, and Oxygen from 60 to 2000 K at Pressures to 2000 MPa," J. Phys. Chem. Ref. Data, 29(3):331-385, 2000.
8. Temperley, H. N. V. and Trevena, D. H. *Liquids and their Properties: A Molecular and Macroscopic Treatise with Applications*, Chichester, England, Ellis Horwood, 1978.
9. Magorien, V. G., "Air in Fluid Systems - Its Effect, Measurement and Removal", BHRA (Br Hydromech Res Assoc) Fluid Eng, Vol. 12(1), 1978, pp. 1-18.
10. Wylie, E. B. and Streeter, V. L., *Fluid Transients*, Ann Arbor, Michigan, USA, Feb Press, 1983.
11. Merritt, H. E., *Hydraulic Control Systems*, New York, U.S.A, Wiley, 1967.