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LINER LONG-TERM PERFORMANCE LIFE PREDICTION USING CRITICAL BUCKLING STRAIN

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ABSTRACT: In North America, close-fitting structural pipe liners are designed according to the American Society for Testing and Materials (ASTM) F1216 design appendix. This design standard is based on the structural buckling theory. A long-term material elastic modulus is introduced to accommodate for material creep for a long-term design. This paper discusses the theoretical incorrectness of applying a long-term elastic modulus in the instantaneous buckling equation, and the technical difficulties in determining the value of the modulus. Based on the critical buckling strain theory introduced in the 1960's, the authors propose a new liner design method. In the new design method, the instantaneous buckling equation is used for short-term liner buckling design, then the creep strain and the critical buckling strain theory are used to validate the long-term design.

Compared with the current design method, the proposed design method:
1) is theoretically sound,
2) utilizes mature creep theory and simple creep tests to predict liner long-term performance, and
3) overcomes the over-conservativeness in the current design method.

Design examples are given to illustrate the proposed design procedure. Discussions are presented to address some key issues in current plastic liner/pipe design, tests, and long-term performance.

1. INTRODUCTION

Early pipe lining technologies were invented in the 1970's. During the past 30 years, as the trenchless pipe rehabilitation market has continued to develop, numerous new technologies and innovations have been introduced. Concurrently, a significant amount of research work has been devoted to helping the industry to better understand liner behavior and to achieve a better and more accurate design.

The ASTM F1216 design appendix (1993) has been used as the liner structural design standard in North America for more than 15 years. It has been widely criticized for its theoretical incorrectness almost since the start of its application (Gumbel (2001), McAlpine (2005)). The ASTM F1216 design method is based on a free-ring buckling theory, which can be traced back to Von Mises and was well presented in a textbook by Timoshenko & Gere (1961). The adoption of a free-ring buckling theory in the liner design equation is theoretically incorrect because the liner is encased in the existing pipe. The existing pipe functions as a rigid boundary to the movement of the pipe liner, and effectively enhances the geometric stiffness of the liner.

An enhancement factor K is introduced into the equation in an attempt to compensate for the support provided by the existing structure to the liner. The K value must be experimentally determined for a specific liner sample. K = 7 is normally applied based upon the results from the experiments done by

Aggarwal & Cooper (1984). However, K=7 was determined to be a very conservative value and K is actually highly variable to a liner's dimension ratio (DR) and material properties (Guice et al. 1994). A liner with a thinner wall (i.e. a lower geometric stiffness), as opposed to a thicker wall, will have greater opportunity for geometric stiffness enhancement, and will generally test with a much higher K value. A tighter fitting liner will have a higher K factor than will a liner with a larger gap (micro-annulus). To further complicate the matter, the relative flexibility versus rigidity of the liner may also influence the ring stability enhancement generated by the sidewall support, because the relative rigidity will influence the strain concentration and the strain limit of the ring.

The theory of circular linings in a rigid cavity by Glock (Glock, 1977) has been introduced into the liner buckling design since the 1990s. The Glock equation was developed with proper theoretical consideration of the stiffening influence of the rigid encasement. As might be expected, a much-improved match between experimental data and theoretical calculations has been found using the Glock method. Significant efforts have been employed to introduce the modified Glock design methodology into actual industry design practice. The authors are aware of the Glock theory having been applied in Germany (Falter (2001)) and France (Thepot, 2000) for liner structural designs. An ASCE PINS Group committee has also recommended the use of the modified Glock design methodology in North America (ASCE, 2007).

A further limitation and additional recent criticism of the ASTM F1216 design method is that it doesn't accommodate for the ring stability reduction generated by some important and common geometric imperfections, which may have significant influence upon liner performance. Falter (2001) summarized geometric imperfections into three (3) categories:

1) **local intrusion / curvature reversal** which are local imperfections distributed over a part of the circumference;
2) **ovality** which is a global imperfection distributed over the whole circumference; and
3) **annular gap** which generates local strain at the point of liner separation from the sidewall support.

The influences of these geometric imperfection categories have been analyzed:

1) **theoretically** (Thepot (2000));
2) **numerically** (El Sawy (1997), Falter (2001), Lu (1999), Omara (1997), and Zhao (2004); and
3) **experimentally** (Moore (1998), Seemann (2000)).

Although the ASTM F 1216 design method has an ovality reduction factor, the selected factor is considered too conservative (Gumble (2001)), and the other common imperfections are not adequately considered.

Another key issue in applying the ASTM F1216 design method, as well as other design methods, is the determination of the long-term (time corrected) material modulus of elasticity, E_L. According to the ASTM F 1216 design appendix, the choice of the value of E_L depends on the material, the load, the design life, and the temperature. The E_L should be based upon all issues that influence the apparent long-term modulus of a material and not merely upon the apparent long-term creep modulus. Based upon encased buckling data collected on a widely used liner material, design engineers typically take E_L as one half of the short-term elastic modulus, E, as a conservative performance estimate for a 50 year design life. It should be noted, however, that in a long-term (up to 10,000hrs) encased buckling test, the E_L has been determined as the combined influence of not only creep, but also material specific physical and chemical aging. As a material ages and has a rise in E, the strain limit can simultaneously decline. Corrosion can also lead to a change in the strain limit over time. With a time dependent change in a material's strain limit, there can be a time dependent decline in the strain induced rupture of the liner wall. Thus, for materials that may ultimately exceed their strain limit, the E_L as determined by long-term encased buckling tests can also be substantially influenced by such a change in the strain limit.

Falter et al. (1996) provided an expression for the variation in the elastic modulus by assuming that the short-term modulus corresponds to a time of 0.1 hours and the long-term modulus corresponds to 50 years. Fitting an equation using these two modulus-time pairs on a log E versus log time plot results in

$E(t) = \frac{E_{0.1h}}{(10 \cdot T)^{0.0453}}$. Applying this equation to predict long-term modulus results in Figure 1 (Zhao, et al. 2004):

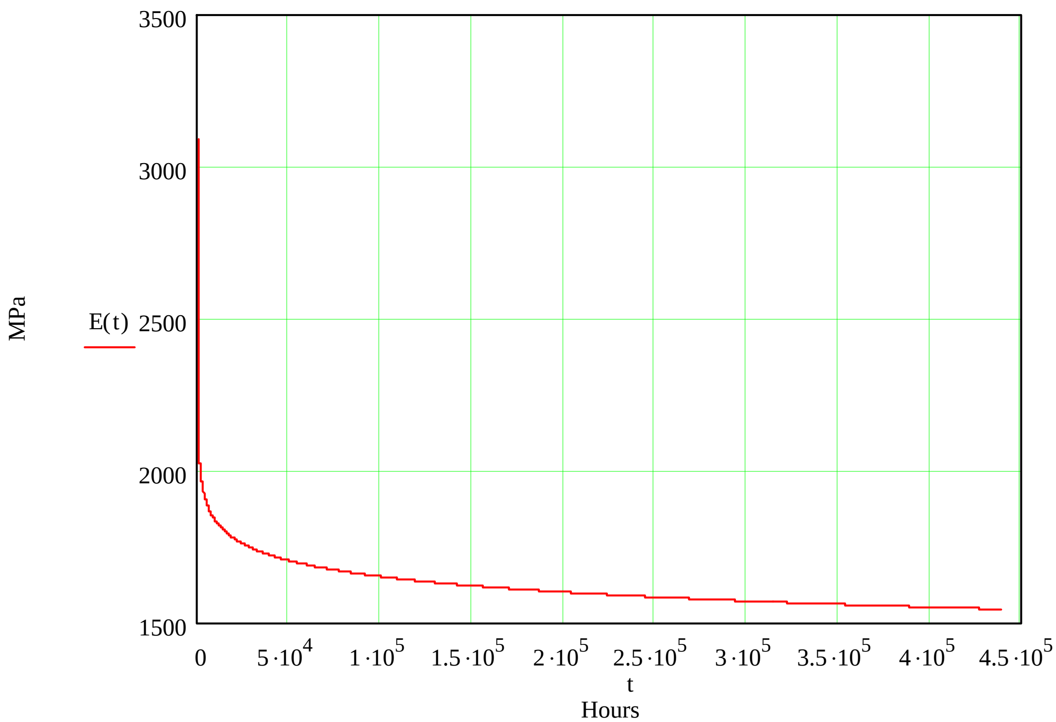

Fig. 1 Time corrected long-term modulus of elasticity defined by Falter et al. (1996)

2. CRITICAL BUCKLING STRAIN

Liner Buckling

From the perspective of the pipe mechanics, the four main aspects in studying the performance of a structure are stress, strain, deformation, and stability. Buckling is a typical stability failure, which is usually sudden and catastrophic. Since buckling may occur even though stress has not reached yield strength, it has been used as the design standard for most structures, especially long and slender structures like pipes and liners. Both the ASTM F1216 design method and the Glock method are based on buckling theory.

Buckling theory was first introduced by Euler in 1741. Geometry and boundary condition are two major factors in studying the buckling resistance capability of a structure. The ASTM F1216 design appendix is based on a free-ring buckling theory. Since the applied boundary conditions are incorrect, it has been widely considered as faulty.

Non-uniformities or imperfections are factors that initiate buckling. A perfectly round pipe with no geometric or material imperfections doesn't buckle under compressive pressure. However, once there is an imperfection within either the geometry or the material property, buckling is the most likely failure mode. Perfection is a myth for a non-existent "ideal" world; underground pipes obviously don't live up to this ideal, particularly not the pipes selected for rehabilitation. Most pipes scheduled for rehabilitation are out of round after being buried for years, or else frequently out of round from the original poor installation practice. Therefore, tight-fitting liners at least have ovality as an inherent geometric imperfection caused by conforming to the inside of the pipe. Under compressive pressure (like ground water pressure), the liner deforms under compression and bending (caused by geometric out-of-roundness). The work done by the compressive pressure is stored by the liner as strain energy as long as the structure is in equilibrium. When the strain energy in the liner reaches its maximum, the equilibrium is broken, and the liner buckles. This is in accordance with the "stationary energy theory."

Strain in Buckling

Since a structure buckles when its strain energy reaches maximum, it is interesting to investigate the state of the strain when a structure loses its stability (buckles). Menges & Gaube (1969) did some significant research on the correlation between buckling and strain on HDPE. The critical buckling strain for bars under longitudinal compressive stress is

$$\varepsilon_{cr} = \frac{\sigma_{cr}}{E} = \frac{1}{E}\frac{\pi^2}{l_e^2}\frac{EI}{A_0} = \frac{\pi^2}{(l_e/\sqrt{I/A_0})^2} \quad [1]$$

and the critical buckling strain for a circular cylinder under radial pressure is

$$\varepsilon_{cr} = \frac{\sigma_{cr}}{E} = \frac{1}{E}\frac{E}{4(1-\nu^2)}(\frac{t}{R})^2 = \frac{\pi^2}{(2\pi\frac{R}{t}\sqrt{1-\nu^2})^2} \quad [2]$$

The term of slenderness ratio was introduced as λ. For a bar $\lambda = l_e/\sqrt{I/A_0}$, and for a cylinder $\lambda = (2\pi\frac{r}{t}\sqrt{1-\nu^2})$. Bring the correspondent λ into Equations 1 and 2, respectively. Equations 1 and 2 are both taking the form

$$\varepsilon_{cr} = \frac{\pi^2}{(\lambda)^2} \quad [3]$$

From Equation 3, the critical buckling strain is a function of the slenderness ratio λ only, since π is a constant. The slenderness ratio λ is a parameter that represents the geometric dimension property of a structure (where λ is the function for the effective length, the moment of inertia, and the cross-section area in case of a bar; or mainly a function of radius and thickness for a cylinder); thus, Equation 3 shows that the critical buckling strain of a structure is a function of its geometric characteristics only. The material properties, such as elastic modulus, have no effect on the critical strain. To prove their theory, Menges & Gaube carried out a series of buckling tests, both short term and long-term, on bars and cylinders. The test results showed that buckling occurred as soon as the critical buckling strain was exceeded, regardless of material stiffness.

Critical Strain in Liner Buckling

For a liner failed in a one-lobe buckling mode, and without considering geometric imperfections, the Glock equation (Glock, 1977) for the critical buckling pressure is

$$P_{cr} = E*(\frac{t}{D})^{2.2} \quad [4]$$

Applying stationary energy theory, Thepot (2000) derived a series of critical values when a liner buckles, which include critical buckling pressure, critical buckling angle, critical buckling bending moment, critical buckling axial force, etc. Critical buckling bending moment and critical buckling axial force are presented as Equation 5 and Equation 6, respectively

$$M_{cr} = 1.2\frac{EI}{R} \quad [5]$$

$$N_{cr} = 1.26 P_{cr} R \quad [6]$$

Apply Equation 4 to Equation 6

$$N_{cr} = 1.26 E*(\frac{t}{D})^{2.2} * R \quad [7]$$

From Equation 5, the maximum bending strain at buckling is

$$\varepsilon_b = \frac{M_{cr}}{EI}\frac{t}{2} = 1.2\frac{EI}{REI}\frac{t}{2} = 1.2\frac{t}{D} \quad [8]$$

From Equation (7), the maximum axial strain at buckling is

$$\varepsilon_a = \frac{N_{cr}}{EA} = 0.63(\frac{t}{D})^{1.2} \quad [9]$$

Thus, the total strain when a liner buckles is the sum of the maximum bending strain and the maximum axial strain

$$\varepsilon_t = 1.2\frac{t}{D} + 0.63(\frac{t}{D})^{1.2} \quad [10]$$

From Equation 10, for an encased liner, the critical buckling strain is the function of its dimensional ratio (DR) only, independent of its material property.

Given a liner with an elastic modulus of E = 1000 Mpa (145000 psi), and thickness of t = 9.2 mm (0.363 inch), Table 1 and Figure 2 show, with different pipe diameters (i.e., different DR), the calculated critical buckling pressure using the ASTM and the Glock methods, respectively. The critical buckling strain is also presented for each case.

Table 1 Critical buckling pressure and critical buckling strain for a liner with different DR

D (inch)	DR (D/t)	Pcr (psi) (ASTM)	Pcr (psi) (Glock-Thepot)	Critical buckling strain (Glock-Thepot)
8	23	87.9	78.0	6.58%
10	29	48.1	51.0	5.22%
12	35	27.8	34.2	4.31%
14	41	17.5	24.3	3.68%
16	47	11.7	18.1	3.20%
18	52	8.2	14.0	2.83%
20	58	6.0	11.1	2.54%
22	64	4.5	9.0	2.30%
24	70	3.5	7.4	2.10%
26	76	2.7	6.2	1.93%
28	82	2.2	5.3	1.79%
30	87	1.8	4.6	1.67%

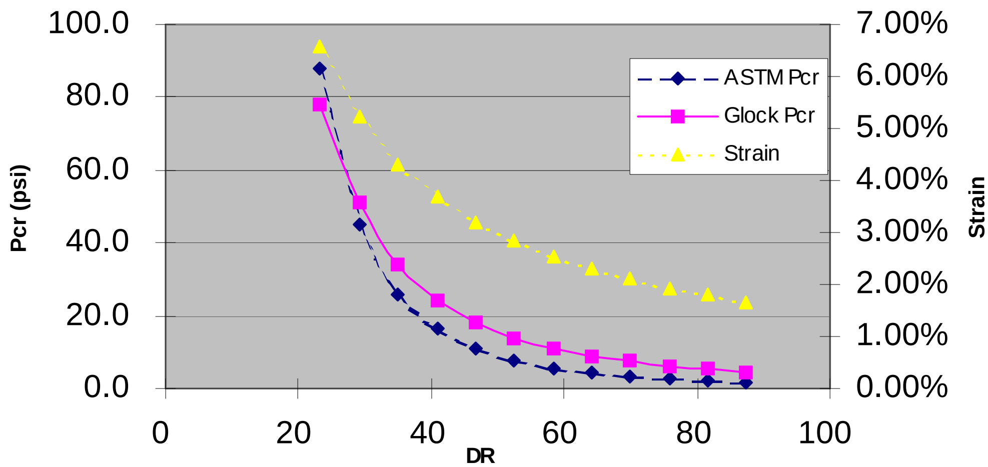

Figure 2 The variation of critical buckling pressure and strain with DR

Note that in Table 1, Equation 11 is used in calculating the ASTM critical buckling pressure

$$P_{cr} = \frac{2 \cdot K \cdot E_L}{1-\nu^2} \times \frac{1}{(SDR-1)^3} \times \frac{1}{N} \qquad [11]$$

and Equation (12) is used in calculating the Glock critical buckling pressure

$$P_{cr} = \frac{E}{1-\nu^2} * (\frac{t}{D})^{2.2} \frac{1}{N} \qquad [12]$$

In both equations, Poisson's ratio ν =0.3 is used for plain strain assumption. In both cases, the safety factor N =2 is used. Since the critical buckling strain is calculated based on the Glock equation, a safety factor of 2 is applied to the calculated values as well.

Creep vs. Modulus Reduction

Creep is defined as continuing deformation with time when the material is subjected to a constant load. In both the ASTM design appendix and the Glock method, the creep of the liner material is handled by introducing a long-term modulus reduction factor. Assuming the deformation doubled in 50 years because of creep, a long-term reduction factor of 2 is applied. Thus the long-term modulus is presumed to be half of the instantaneous value $E_L = \frac{1}{2}E$. When the stress is a constant, the calculated strain at 50 years is twice as much as the instantaneous strain.

However, this "sounds reasonable" approach at conservatism has been questioned by several researchers. Moser et al. (1990) reported a test at Utah State University. The stiffness of a PVC pipe at one hour and after buried for 13 years was compared. The results showed that pipe stiffness and modulus for PVC pipe did not decrease with time. They concluded that apparent or creep modulus was an inappropriate property to predict long-term deflection of buried PVC sewer pipe. Watkins and Anderson (2000) also stated that, if not exposed to a chemical degradation environment, properties of many plastic materials remain pristine over time. The most pertinent properties are strength, modulus of elasticity, and Poisson's ratio. The long-term creep modulus is termed as virtual modulus by Watkins and Anderson. It is just virtually the ratio of stress over strain over time, which doesn't represent the elastic modulus at the specific time. Watkins and Anderson also emphasized that the virtual modulus does not apply to collapse. Creep is just a continuous deformation over time, instead of a degrading of material properties.

Creep Test and Data Analysis

Findley (1987) reported a 26-year creep and recovery test on thermoplastics at Brown University. The long-term strain caused by creep is determined using the following equation

$$\varepsilon = \varepsilon^0 + \varepsilon^+ T^n \quad [13]$$

where $\varepsilon^0, \varepsilon^+,$ and n are creep constants for a material under a certain stress level. In his study, the data for the first 1900 hrs were used to determine the constants. The predicted creep curves were quite close to the actual observed 26-year strain.

Given a thermoplastic material with an instant elastic modulus E = 1000 Mpa (145,000 psi), Table 2 shows creep data at stress level 2.8 Mpa (410 psi) up to 2136 hrs (3 months).

Table 2 Creep data for a thermoplastic material at 410 psi stress

Time (hr)	Strain (%)	Time (hr)	Strain (%)
0.000	0.30%	451.667	0.56%
0.017	0.32%	475.667	0.56%
0.100	0.34%	499.667	0.57%
0.200	0.35%	523.667	0.57%
0.500	0.37%	547.667	0.58%
1.000	0.39%	595.667	0.58%
2.000	0.41%	643.667	0.58%
5.000	0.43%	691.667	0.59%
19.667	0.45%	739.667	0.59%
43.667	0.46%	787.667	0.59%
67.667	0.48%	835.667	0.60%
91.667	0.49%	883.667	0.61%
115.667	0.50%	931.667	0.61%
139.667	0.50%	979.667	0.61%
163.667	0.51%	1075.667	0.62%
187.667	0.52%	1171.667	0.62%
211.667	0.52%	1267.667	0.63%
235.667	0.53%	1363.667	0.63%
259.667	0.53%	1459.667	0.63%
283.667	0.54%	1555.667	0.64%
307.667	0.54%	1651.667	0.64%
331.667	0.54%	1747.667	0.64%
355.667	0.55%	1843.667	0.65%
379.667	0.55%	1939.667	0.65%
403.667	0.55%	2035.667	0.65%
427.667	0.56%	2136.000	0.66%

Figure 3 shows the result of the creep curve and the experimental data. Equation 13 is used for the data fitting. The following parameters are obtained $\varepsilon_0 = 0.0030$, $\varepsilon^+ = 9.2 * 10^{-4}$, and n = 0.173.

For a linear viscoelastic creep, at an arbitrary time t, the strains at two different stresses can be expressed as

$$\frac{\varepsilon_1(t)}{\sigma_1(t)} = \frac{\varepsilon_2(t)}{\sigma_2(t)} \qquad [14]$$

Using Equation 14, the strain of a material under any other stress level can be calculated using the creep parameters at stress 410 psi as long as they are in the linear viscoelastic creep range. The strain at any time under 205 psi stress is one half of the strain under 410 psi stress according to Equation 14. Figure 4 shows the strain vs. time experimental data for both 205 psi and 410 psi. A typical linear characteristic is observed.

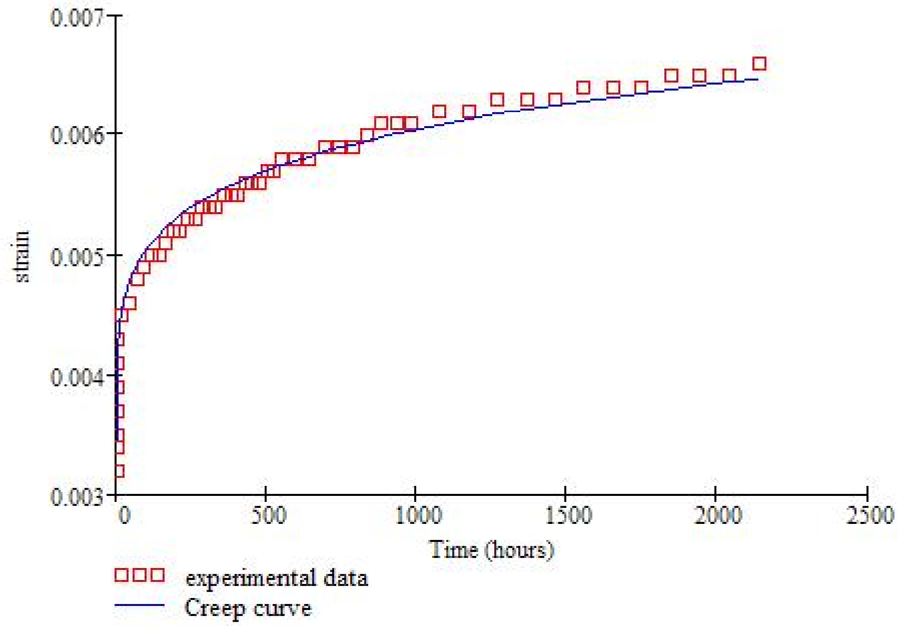

Figure 3 Data fitting of the creep data from Table 2

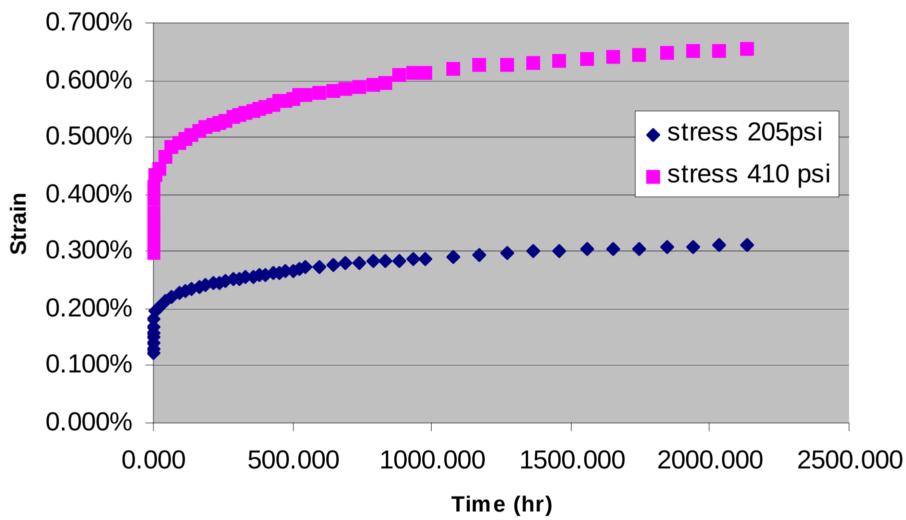

Figure 4 Strain vs. Time for stress 205 psi and 410 psi

3. RECOMMENDED LINER BUCKLING DESIGN METHOD USING CRITICAL BUCKLING STRAIN

The Glock method is widely-accepted by academics in contrast to the more widely used ASTM design method because of the scientific correctness of the Glock method. However, as explained in the previous section, the application of the creep reduced long-term modulus in the instantaneous buckling design is faulty. Therefore, the following design procedures are recommended by the authors.

1) *Instantaneous Buckling Design*
Apply the Glock equation utilizing the material's **short-term modulus** to calculate the required thickness of the liner
2) *Critical Buckling Strain*
Apply Equation 10, $\varepsilon_t = 1.2\frac{t}{D} + 0.63(\frac{t}{D})^{1.2}$ with the calculated thickness t from step 1, to estimate the critical strain ε_t
3) *Long-term Strain*
Apply Equation 13 $\varepsilon = \varepsilon^0 + \varepsilon^+ T^n$ to calculate the strain level at the end of the design life. Constants $\varepsilon^0, \varepsilon^+,$ and n can be obtained by appropriate long-term material characteristic experiments. T is the design life
4) *Long-term Design Validation*
Compare the calculated strain level at the end of the design life, ε, and the critical buckling strain, ε_t; if $\varepsilon < \varepsilon_t$, the liner thickness calculated in step 1 is valid; otherwise, a smaller DR (thicker liner wall) is chosen from Figure 4 using the calculated ε in step 3. Note that Figure 4 is from Equation 10.

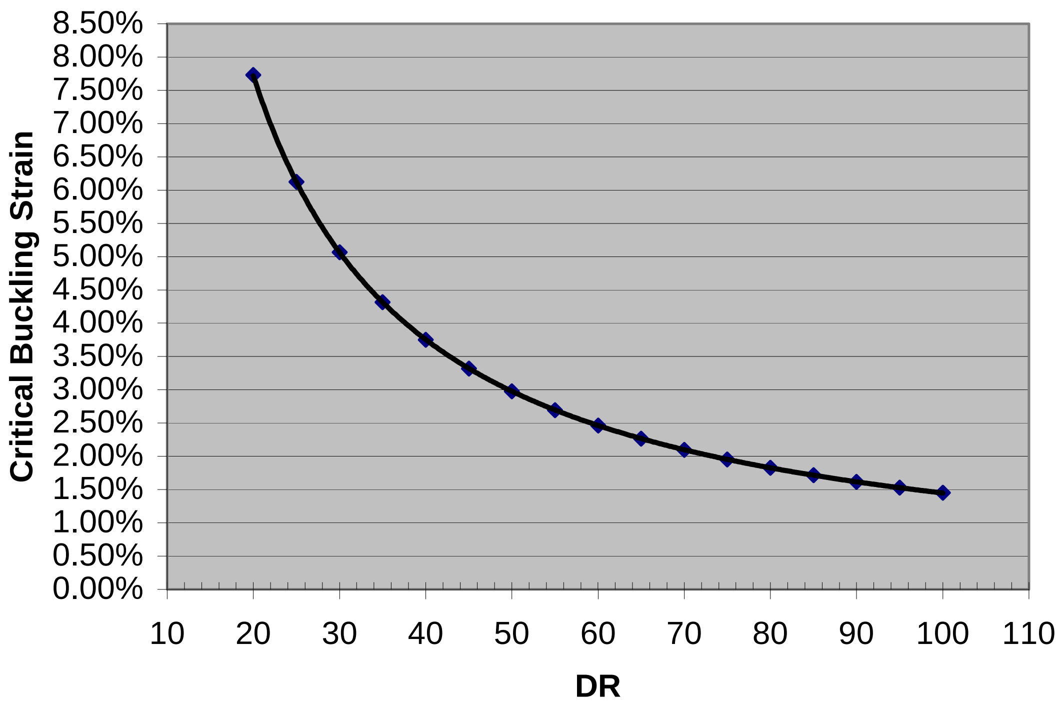

Figure 4 Critical Buckling Strain vs. Dimensional Ratio (DR)

Design Examples

Example 1

Pipe condition: h = 2.75 m (9 feet), i.e., P = 0.027 Mpa (3.9 psi). ID = 203 mm (8 inch), ovality is q = 5% (i.e., $c = (\frac{1-5\%}{(1+5\%)^2})^{2.2} = 0.72$)

Material: E = 1,000 Mpa (145,000 psi), at 2.8 Mpa (405 psi) stress level: n = 0.173, $\varepsilon^+ = 9.2*10^{-4}$, $\varepsilon^0 = 0.0030$ (creep test data is shown in table 2); $\nu = 0.3$

Design: T = 50 years (438,000 hrs), N = 2

Design:

Step 1) *Instantaneous Buckling Design*

Using the Glock equation $P_{cr} = \frac{E}{1-\nu^2} * (\frac{t}{D})^{2.2} \frac{c}{N}$, t = 2.59 mm (0.102 inch), DR = 78.

Step 2) *Critical Buckling Strain*

Applying Equation 10, $\varepsilon_t = 1.2\frac{t}{D} + 0.63(\frac{t}{D})^{1.2}$, since DR = $\frac{D}{t}$ = 78, the critical buckling strain is $\varepsilon_t = 1.88\%$

Step 3) *Long-term Strain*

Apply Equation 13 $\varepsilon = \varepsilon^0 + \varepsilon^+ T^n$, at stress level 405 psi $\varepsilon^0 = 0.0030$, $\varepsilon^+ = 9.2*10^{-4}$, n = 0.173, and T = 438,000 hrs (50 years), the expected strain at design life is $\varepsilon_{(410 psi)} = 1.17\%$. With provided liner and loading situation (DR = 78, P = 3.9 psi), the maximum stress is 5.6 Mpa (810 psi) according to the Finite Element (FE) modeling. Based on the linear viscoelastic assumption, the expected long-term creep strain at 810 psi stress is $\varepsilon_{(810 psi)} = \frac{810}{405}\varepsilon_{(405 psi)} = 2.34\%$

Step 4) *Long-term Design Validation*

Compare ε with ε_t (2.34% > 1.88%) the design with liner thickness t = 2.59 mm (0.102 inch), DR = 78 is not valid. Using $\varepsilon = 2.34\%$, from Figure 4, DR = 60 is chosen.

Comparison with the ASTM and the Glock design method

Table 3 lists the calculated DRs using both the ASTM (Equation 11) and the Glock (Equation 12) design methods. The long-term modulus reduction factor ranges from 10% to 70%. The objective of this table is to investigate the appropriate reduction factor in each case by comparing the DR value resulting from the proposed new design method with the ASTM design method, and the Glock equation, respectively. Note that all of the DRs are reduced to the closest number which is divisible by five (5).

Table 3 Design results using the ASTM and the Glock methods with different E_L

Long term reduction factor	E_L (psi)	DR by Glock	DR by the ASTM
0%	145000	75	55
10%	130500	75	55
20%	116000	70	50
30%	101500	65	50
40%	87000	60	45
50%	72500	55	45
60%	58000	50	40
80%	43500	45	35

From Table 3, using the Glock method and assuming $E_L = 40\%E = 87000\,psi$, the calculated DR is 60, which is the same as using the proposed new design method. However, if the assumed long-term reduction factor is smaller than 40%, i.e., the E_L is larger than 87000 psi, the calculated DR is larger than 60, which results in a thinner liner wall, and a less conservative design. Therefore, for this material, at this stress level, a long-term reduction factor of 40% is required for a comparably conservative design if choosing the Glock method. By comparison, even without using any long-term modulus reduction at all, the ASTM design method provides a more conservative long-term design calculation (DR 55) at this stress level than is shown to be required by the proposed new design method (DR 60).

Example 2
Pipe condition: h = 6.10 m (20 feet), i.e., P = 0.059 Mpa (8.6 psi). ID = 203 mm (8 inch), ovality is q = 5% (i.e., $c = (\frac{1-5\%}{(1+5\%)^2})^{2.2} = 0.72$)

Material: This example uses the PVC material tested by Findley (1987). The specimen was initially under stress of 27.6 Mpa (4000 psi), and the test lasted for 230,000 hrs (26 yrs). The following creep parameters were given by Findley: E = 3447 Mpa (500,000 psi), n = 0.305, $\varepsilon^+ = 4.6*10^{-4}$, $\varepsilon^0 = 0.0081$, $\nu = 0.3$.

Design: T = 50 years (438,000 hrs), N = 2

Design:
Step 1) *Instantaneous Buckling Design*

Using the Glock equation $P_{cr} = \frac{E}{1-\nu^2} * (\frac{t}{D})^{2.2} \frac{c}{N}$, t = 2.03 mm (0.08 inch), DR = 96.

Step 2) *Critical Buckling Strain*

Applying Equation 10, $\varepsilon_t = 1.2\frac{t}{D} + 0.63(\frac{t}{D})^{1.2}$, since DR = $\frac{D}{t}$ = 96, the critical buckling strain is $\varepsilon_t = 1.51\%$

Step 3) *Long-term Strain*

Apply Equation 13 $\varepsilon = \varepsilon^0 + \varepsilon^+ T^n$, at stress level 27.6 Mpa (4000 psi), $\varepsilon^0 = 0.0081$, $\varepsilon^+ = 4.6*10^{-4}$, n = 0.305, and T = 438,000 hrs (50 years), the expected strain at the design life is $\varepsilon_{(4000\,psi)} = 3.2\%$. With the liner properties and the loading conditions (DR = 96, P = 8.6 psi), the maximum stress is 17.25 Mpa (2500 psi) according to the FE modeling. Based on the linear viscoelastic assumption, the expected long-term creep strain at 2500 psi stress is $\varepsilon_{(2500\,psi)} = \frac{2500}{4000}\varepsilon_{(4000\,psi)} = 2.0\%$

Step 4) *Long-term Design Validation*

Compare ε with ε_t, 2.0% > 1.51%, the design with a liner thickness t = 2.03 mm (0.08 inch), DR = 96 is invalid. Using $\varepsilon = 2.0\%$, from Figure 4, DR = 70 is chosen.

Comparison with the ASTM and the Glock design method
Compare with the Glock and the ASTM design methods using the same process as described in Example 1. For a comparably conservative design, a reduction factor of 50% is required when using the Glock method, while no reduction is required when using the ASTM design method.

4. DISCUSSION

1) **Applying a long-term modulus reduction is theoretically incorrect and significantly flawed in practice**

Needless to say, the theoretical correctness of a design equation is crucial for any design. In the pipe rehabilitation industry, tremendous efforts have been exerted towards replacing the current ASTM F1216 design equation with the Glock equation. However, the theoretical incorrectness of applying a long-term modulus reduction factor in the design equation is relatively ignored. Its incorrectness can be summarized as follows:

a. For most plastics used as pipe liners, the material modulus doesn't degrade with time. On the contrary, for some plastic materials, modulus increases with time because of material aging.
b. Virtual (creep) modulus doesn't apply to buckling analysis. The original derivations of the Timoshenko and the Glock equations utilize the short-term modulus. A long-term modulus reduction factor was improperly incorporated into the equations in an attempt to accommodate for time-dependent changes in measured buckling or strain limits.
c. For the short-term design, it is obvious that a thicker liner wall increases the instantaneous buckling resistance capacity. However, why is a thicker liner wall required to compensate for the long-term material creep? It is hard to find a logical explanation from the instantaneous design equations—both the ASTM and the Glock equations are based on instantaneous buckling theory. The critical buckling strain theory provides a reasonable answer to this question: A larger critical buckling strain resulting from a thicker liner wall is more difficult to exceed within the design life, thus providing a safer, more conservative design, regardless of the material creep rate. Furthermore, a thicker liner wall leads to a lower stress level, thus a lower creep rate. Therefore, a thicker liner wall is desirable for enhancing long-term stability. Applying a long-term reduction factor to the material modulus in the instantaneous design equation does lead to a thicker liner wall, and it is simple in application. This is probably the reason it has been widely accepted and used. However, this doesn't imply its theoretical correctness.
d. In many cases, the choice of the value of the reduction factor is based on experience and pre-assumption, which can be risky. Long-term modulus testing protocols such as ASTM D-2990 do not adequately or properly reflect the loads and strains of a liner in a rigid encasement. Encased buckling tests calculate a long-term reduction factor from samples subjected to failure at strains far exceeding real world applications. Such tests also include the influences of material aging upon the data; material aging can create time dependent changes to the strain limit (as modulus rises the strain to failure declines) that cannot be readily separated from the creep data. In other words, there is limited scientific basis for the current selection of a long-term modulus reduction factor.

2) **The new design method is theoretically correct**

a. It is obvious that the long-term material behavior plays a significant role in a structure's long-term performance. Creep is a universal material behavior; even steel and concrete experience creep and have to be designed accordingly. Creep of plastics has been well studied and the theories are mature both in terms of experimental methods and data analysis. Theoretically correct equations exist to avoid the use of theoretically improper modulus reduction factors. The derivation of the critical buckling strain makes it feasible to apply the relatively simple material characteristic test results to the liner design. Although any preferred buckling equation can be utilized in Step 1, the proposed design method applies the Glock equation for buckling design, and the creep theory for long-term design validation. This makes both the stability and material properties applications in the design procedure theoretically correct.
b. The preferred buckling equations in step 1 can be further augmented with the inclusion of improved variables for handling the influence of local imperfections, ovality, and annular gap.

3) **Critical buckling strain is not a design for the limit of strain**

   a. Critical buckling strain is the strain level that a structure reaches when it buckles. It is a measurement of when the structure loses its stability. Moser (1981) and Moser et al. (1990) carried out a series of tests at Utah State University on the structural performance of PVC sewer pipes. Their research shows that it is unwarranted to design PVC sewer and drainage pipe using a *strain limit*. As a clarification, a structure designed using *critical buckling strain* is a design using stability instead of strain as the performance limit; i.e., it is still a design for buckling, not a design for a strain limit.
   b. Note that in Table 1, for an eight (8) inch pipe with DR = 23, the critical buckling strain is 6.58%. For some materials, such a high critical buckling strain may exceed the strain limit which may therefore prove to be the controlling failure mechanism. Materials which are stiffer, more rigid, and more brittle may have a lower long-term (post-aging) strain limit. The proposed design methodology assumes that the strain limit will not be the controlling failure mechanism.
   c. Additionally, material aging and chemical degradation can cause materials to become more brittle over time. Some materials age more and are more sensitive to chemical degradation than others. Some materials, due to the time dependent effects of aging and chemical degradation, can eventually exceed their strain limit (as their strain to failure declines as their modulus rises). This phenomenon can also exhibit itself as a time-dependent decline in maximum encased "buckling" pressure over time and can be easily confused with the effects of the virtual (creep) modulus. Therefore, the long-term strain limit of some liner materials may need to be checked after the critical buckling strain as well.

4) **Material stiffness vs. geometrical stiffness**

   a. In both the Glock equation and the ASTM equation, the geometric stiffness, represented by (t/D), is in higher power than material stiffness, represented by E—in the Glock equation (Equation 12), (t/D) is exponentially influenced at a power of 2.2, while in the ASTM design equation, (t/D) is at a power of 3 (Equation 11 where DR = D/t); E is at a power of 1 in both cases. It is obvious that an increase in the geometric stiffness has a much greater effect on increasing the liner's buckling resistance capacity. Just from observing Equations 11 and 12, a material with higher E theoretically results in a lower required t, which in turn results in a reduced maximum critical buckling strain (ε_t). The current ASTM and Glock design practices don't check for the maximum critical buckling strain. This may lead to long-term failure caused by the long-term strain exceeding the critical buckling strain for some high modulus and low thickness liners.
   b. Additionally, imperfections are the factors which initiate buckling; a liner with a thinner wall is more imperfection sensitive. A thicker walled liner will retain greater ring stability as influenced by local imperfections, ovality, and gap.

5) **Consideration on water table variations**

In the current design practice, the maximum water table during the rainy season is normally applied as the constant water pressure the liner has to withstand for its entire service life. The recession of the water table during the dry season, which permits partial recovery of the strain in the liner, is not given consideration. The authors believe the variation of the water table has a significant positive effect on the long-term liner performance. The recovery which can occur during the dry season can justify the use of a much less conservative long-term material modulus reduction factor in some cases. While it is recognized that such differences result in seemingly negligible changes in wall thickness and material costs, importantly, they greatly extend the expected performance life of the pipeliners. In this proposed design method, the record high water table (50 year or 100 year rain event level) is recommended in step 1 for the instantaneous buckling pressure design; more research work is required to cover the effects of water table variations on the long-term strain design in step 3.

6) **Constructability may actually control the design**

In the smaller diameters (18” and smaller), constructability issues (such as insertion forces) as well as the relative influence of host pipe geometric imperfections will rarely lead a design engineer to select a DR higher than 50, and will frequently encourage the selection of a DR as low as 35. However, in liner installations, thicker walls start creating additional constructability issues that increase the risk exposure for a failed installation, so pipe manufacturers typically have maximum recommended thicknesses as well. Similar to direct burial design of plastic pipes, long-term material structural response is not necessarily the controlling design factor in the selection of the DR. Issues pertaining to constructability may encourage a much more conservative wall thickness selection than the long-term structural design equations actually deem necessary.

7) Implications for long-term service life of liners and also direct burial plastic pipes

- **a.** The relatively high critical buckling strain of the typically employed DRs would indicate a substantially longer service life expectation for liners than suggested by the improper use of the long-term modulus reduction factor in current industry design methods. In fact, it may be reasonable to expect that creep may not actually govern the ultimate service life limit of a typically designed and properly installed liner.
- **b.** Although back-filling and handling issues generally control the design of direct burial plastic pipes, rather than the long-term loading response, the comparable use of a long-term creep reduction factor in direct burial pipe design likewise under-estimates the true long-term service life of direct burial plastic pipes. The predicted controlling structural failure mechanism of the direct burial pipe can be considered as Step 1, followed by a comparable analysis of the long-term strain response to provide a scientifically correct long-term performance analysis of direct burial plastic pipes. Such an analysis could assist in identifying a more correct long-term service life expectation than current design methodologies, which are far too conservative. Some other failure mechanism, such as the degradation failure of gasketed joints, may prove to be the controlling parameter for a conservative long-term design. Even if long-term creep failure still proves to be the controlling failure mechanism, the extra wall thickness which results from considering back-filling and handling in direct burial application, leads to a substantially high critical buckling strain. The actual service life can be expected to far exceed the design life using current design methods.
- **c.** It has been widely recognized that properly designed and installed buried plastic pipes and liners are not behaving under real world conditions in the manner predicted using a creep reduction factor in their design. The proposals herein offer a scientifically based explanation for understanding the true long-term service life limits of plastic pipes and liners using the correct creep theory.

5. CONCLUSIONS

A modified liner design method is proposed based on the theory of critical buckling strain. The validation of critical buckling strain after the short-term buckling design is required. The proposed design methodology eliminates the theoretically incorrect long-term modulus from the buckling design equation, while enabling a more accurate long-term design life validation. The long-term liner performance is studied using the correct creep theory. Simple material characterization experiments are employed to obtain required creep constants for the long-term design. While the proposed design method is theoretically accurate, it should be recognized that the resulting wall thicknesses may not be sufficient in consideration of the risks associated with unknown field conditions and the as yet inadequately defined imperfection sensitivity of thin wall pipeliners.

Table of symbols

E_t	Time corrected material modulus (MPa/psi)	D	Diameter (mm/inch)
$E_{0.1h}$	Material modulus at 0.1 hour (MPa/psi)	DR	Dimensional ratio, or D/t
T	Time, design life (hours)	M_{cr}	Critical buckling moment (N*mm/Lbf*inch)
ε_{cr}	Critical strain	N_{cr}	Critical buckling axial force (N/Lbf)
σ_{cr}	Critical stress (MPa/psi)	ε_b	Maximum bending strain
E	Modulus of elasticity (MPa/psi)	ε_a	Maximum axial strain
I	Moment of inertia (mm^3 / in^3)	ε_t	Critical buckling total strain
l_e	Effective length for buckling (mm/inch)	P_{cr}	Critical buckling pressure (MPa/psi)
A_0	Original cross-section area (mm^2 / in^2)	K	Enhancement factor
μ	Passion's ratio	E_L	Long term material modulus (MPa/psi)
t	Width, wall thickness (mm/inch)	ε	Strain
R	Radius (mm/inch)	ε^0	Initial strain
λ	Slenderness ratio	ε^+	Creep constant
A	Area	c	Ovality reduction factor

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