

**Baker**

Department of Transportation  
Research and Special Programs Administration  
Office of Pipeline Safety

---

**SSD, Inc.**

**BEAR**  
Berkeley Engineering And Research, Inc.  
PURES WELLDENS MODORUM MACHINATORUM

*TTO Number 11*

*Integrity Management Program  
Delivery Order DTRS56-02-D-70036*

*Pipe Wrinkle Study*

***FINAL REPORT***

*Submitted by:  
Michael Baker Jr., Inc.  
October 2004*

**ChallengeUs.**

This report is intended to serve as a technical resource for OPS and State pipeline safety inspectors evaluating operators' integrity management (IM) programs. Inspectors consider information from a number of sources in determining the adequacy of each IM program. Development of this report was funded via a Congressional appropriation specifically designated for implementation of IM oversight. This and other similar reports are separate and distinct from the work products associated with and funded via OPS's R&D Program.

*This page intentionally left blank*

# *TTO Number 11 Pipe Wrinkle Study*

## *Table of Contents*

<b>EXECUTIVE SUMMARY.....</b>	<b>1</b>
<b>1 INTRODUCTION .....</b>	<b>3</b>
<b>2 BACKGROUND.....</b>	<b>5</b>
<b>3 ILI TECHNOLOGY EVALUATION .....</b>	<b>7</b>
3.1 SCOPE STATEMENT .....	7
3.2 ILI TECHNOLOGY.....	7
3.2.1 UT Tools .....	7
3.2.2 MFL Tools.....	8
3.2.3 TFI Tools .....	9
3.2.4 EMAT Tools .....	9
3.3 EFFECTS OF RIPPLES, WRINKLES AND BUCKLES ON ILI DETECTION OF METAL LOSS.....	9
<b>4 SUMMARY OF DEMAND-CAPACITY FRAMEWORK FOR CORRODED WRINKLES .....</b>	<b>11</b>
4.1 ILLUSTRATIVE EXAMPLE .....	12
4.1.1 Problem Parameters.....	12
4.1.2 Pressure Capacity of Corroded Section.....	13
4.1.3 Fatigue Demand Capacity Evaluation .....	14
<b>5 CONCLUSIONS AND RECOMMENDATIONS .....</b>	<b>19</b>
5.1 CONCLUSIONS REGARDING ILI CAPABILITIES .....	19
5.2 CONCLUSIONS REGARDING PIPELINE INTEGRITY AT CORRODED WRINKLES .....	19
5.3 RECOMMENDATIONS .....	21
<b>6 REFERENCES .....</b>	<b>23</b>
<b>APPENDIX A.....</b>	<b>A-1</b>
<b>APPENDIX B.....</b>	<b>B-1</b>
B.1 SCOPE STATEMENT .....	B-3
B.2 CAPACITY EVALUATION FOR BOUNDING CASES .....	B-5
B.2.1 Corrosion.....	B-5
B.2.2 Fatigue .....	B-8
B.3 STRESS CONCENTRATION FACTORS.....	B-11
B.3.1 Wrinkled Pipe.....	B-12
B.3.2 Corroded Pipe.....	B-12
B.3.3 Combined Wrinkling and Corrosion.....	B-13
B.4 RELATIONSHIPS BETWEEN SIFs (I-FACTORS) AND SCFs.....	B-14
B.5 FRAMEWORK FOR EVALUATION OF CORRODED WRINKLES .....	B-15
B.6 EXAMPLE APPLICATION OF DEMAND-CAPACITY FRAMEWORK .....	B-22

B.6.1	<i>Description of Example Problem Parameters</i> .....	B-22
B.6.2	<i>Step-by-Step Application of the Demand-Capacity Framework Parameters</i> .....	B-22
B.7	SUMMARY AND CONCLUSION .....	B-26
B.8	REFERENCES .....	B-29

## List of Figures

FIGURE 3.1	MFL ILI TOOL .....	7
FIGURE 3.2	UT TOOL IN A LIQUID BATCH (PIPETRONIX).....	8
FIGURE B.1	PRESSURE-CORROSION SPACE.....	B-3
FIGURE B.2	FATIGUE-CORROSION SPACE.....	B-4
FIGURE B.3	PRESSURE & FATIGUE-CORROSION SPACE.....	B-4
FIGURE B.4	COMPARISON OF REMAINING STRENGTH CALCULATION METHODS .....	B-8
FIGURE B.5	CONCEPTUAL PROCEDURE FOR CORRODED WRINKLE EVALUATION.....	B-21
FIGURE B.6	MAOP OF CORRODED 24-INCH DIAMETER, 0.266-INCH WALL, X60 PIPE WITH 50% METAL LOSS .....	B-23

## List of Tables

TABLE 4-1	EXAMPLE PROBLEM PARAMETERS .....	13
TABLE 4-2	MODIFIED B31G CALCULATIONS .....	14
TABLE 4-3	PRESSURE CYCLE SPECTRUM OVER TYPICAL ONE YEAR TIME PERIOD.....	14
TABLE 4-4	TEMPERATURE CYCLE SPECTRUM OVER TYPICAL ONE YEAR TIME PERIOD.....	14
TABLE 4-5	PRESSURE CYCLE FATIGUE RESULTS.....	16
TABLE 4-6	TEMPERATURE CYCLE FATIGUE RESULTS.....	17
TABLE B-1	COMPARISON OF PARAMETERS USED IN DIFFERENT MAOP AND BURST PRESSURE CALCULATION METHODS .....	B-7
TABLE B-2	PRESSURE CYCLE SPECTRUM OVER TYPICAL ONE YEAR TIME PERIOD.....	B-23
TABLE B-3	TEMPERATURE CYCLE SPECTRUM OVER TYPICAL ONE YEAR TIME PERIOD.....	B-23

## Definitions

### **Buckle**

A **buckle** can be described as a wrinkle that has advanced well into the post-wrinkling stage of deformation. Buckles that form under high-pressure conditions are typically characterized by severe distorted outward bulges. However, under low-pressure conditions, buckles can take on an inward/outward “diamond” lobe pattern around the pipe circumference. With very severe buckles, a “folding over” of the outward bulge of the pipe wall has been observed.

### **Ripple**

A localized waveform deformation pattern in the pipe wall, typically consisting of several low-amplitude, alternating inward/outward lobes, is referred to as a **ripple**. It is not uncommon to observe mild ripples along the intrados of field cold bends or along the extreme compression fibers of a pipe during the early stages of full-scale pipe bending tests. Ripples are permanent features that result from plastic deformation of the pipe wall.

### **Wrinkle**

A **wrinkle** is defined as a localized deformation of the pipe wall, usually characterized by a dominant outward bulge. A wrinkle is more severe than a ripple and is usually formed at one of the outward lobes of a previously rippled section of pipe. Wrinkles formed under low-pressure conditions can be characterized by significant inward distortions. For a pipe subject to bending, a wrinkle forms on the compression side of the pipe. For a pipe with only axial force, the wrinkle may be axi-symmetric.

*This page intentionally left blank*

## Executive Summary

This report examines the effects of corrosion metal loss on wrinkles and buckles in steel pipelines. The report focuses on the ability of in-line inspection (ILI) to detect corrosion-related defects within the deformed pipe section and evaluates the possibility of developing a demand-capacity criteria framework for evaluation of wrinkles and buckles with general metal loss due to corrosion.

While current ILI tools can accurately detect localized pitting and general metal loss in cylindrical pipe sections (i.e., in sections without wrinkles or buckles), the ability of ILI tools to accurately characterize metal loss due to corrosion in the vicinity of wrinkle bends and buckles is uncertain.

In areas where the pipe wall's radius of curvature is small, the sensors on both types of tools commonly used for detection of metal loss—magnetic flux leakage (MFL) and ultrasonic (UT)—will not conform properly to the pipe surface and the minimum detection level can be seriously impacted.

Thus, though it is possible the severity of metal loss can be accurately reported in pipe containing mild ripples, the more severe the deformation, the more likely it will be that metal loss will not be accurately detected.

There are numerous acceptable methods available for evaluating the pressure capacity of cylindrical pipe sections containing corrosion-related metal loss (ASME B31G, modified B31G, RSTRENG, etc.). Likewise, there are methods available for evaluating the fatigue life of deformed pipe sections, though these methods are typically more complicated and not as widely used or standardized. These fatigue life evaluation methods typically rely on some form of finite element analysis (FEA). However, there is currently no well-established method for combining the effects of general metal loss at a wrinkle or buckle with fatigue effects from pressure and/or temperature cycling.

Developed specifically for this report, the framework for evaluating the effects of corrosion metal loss on wrinkles or buckles consists of the following steps:

- Evaluate the pressure capacity of the section based on measurements of the corrosion alone, using one of the widely accepted methods and assuming the pipe is cylindrical (ignoring the presence of the wrinkle or buckle).
- If the results of the pressure capacity evaluation indicate an acceptable condition, evaluate the fatigue integrity of the wrinkle or buckle by performing the following:
  - Develop representative annual “histograms” of pressure and temperature cycles for the pipeline at the location of interest.
  - Develop and analyze a case-specific “global” buried pipe model at the location of interest to produce estimates of the global loads and nominal stresses at the wrinkle.
  - Develop and analyze a case-specific “local” FEA model of the wrinkle geometry of interest to establish estimates of the stress concentration factors (SCFs) for internal pressure and bending moment loads.

- Combine the pressure and temperature cycle histograms with the corresponding nominal stresses and the pressure and bending moment SCFs to obtain the localized fatigue stress demands at the wrinkle.
- Use fatigue stress versus cycle (“S-N”) curves to compute estimates of the annual fatigue damage at the wrinkle based on a fatigue usage factor. The fatigue life in years is equal to the inverse of the annual usage factor.

Once the fatigue integrity of the wrinkle, disregarding the presence of corrosion, has been considered, the fatigue analysis can be extended to consider the effects of corrosion within the wrinkle. The only change to the evaluation approach is that the detailed “local” FEA model of the wrinkle is modified to include a characterization of the corrosion. The corrosion is typically represented as a rectangular patch. Depending on the geometry of the corrosion (e.g., its length, width and depth and its location with respect to the peak of the wrinkle), the SCFs are likely to increase relative to those of the un-corroded wrinkle.

The fatigue analysis aspects of the proposed framework are far less well established and more time consuming than the procedures used to evaluate pressure integrity. However, the application of FEA methods is very well established in the pipeline and piping research industry. The use of FEA as a tool for performing pipeline structural integrity and serviceability assessments is becoming much more common. FEA methods used in combination with additional experimental data represent the most promising means of evaluating complex pipe stress and deformation problems such as assessing the fatigue behavior of corroded wrinkles.

Based on the combined experience of the project team and discussions with industry experts, pipeline failures due to fatigue in corroded ripples, wrinkles or buckles could not be identified. Moreover, there is a lack of full-scale experimental evaluations of corroded pipes designed to produce fatigue failures in the corrosion; most corroded pipe tests are aimed at evaluating burst pressure. However, pipelines that have experienced external corrosion at elbows were identified during the research. In this case, there was concern that the corrosion within the elbow would increase the flexibility and stress intensification effects with a potential reduction in the fatigue capacity of the elbow. Detailed (proprietary) FEA and fatigue testing of both uncorroded and corroded elbows led to the conclusion that evaluation of the corrosion pressure capacity by any established methodology (e.g., B31G, RSTRENG), as well as derating or repair if the corrosion is severe enough, should take precedence over fatigue concerns. Using established pressure integrity methods should result in derating or repairing the pipeline long before fatigue becomes a concern for all but the most extreme cases of cyclic stress demand. The same conclusion can be applied to corroded wrinkles.

The proposed framework presented in this report is based in large part on theoretical information. With additional research data on fatigue in corroded pipe and corroded wrinkled pipe and burst capacity of corroded wrinkled pipe, this framework could likely be further enhanced. Even though the apparent lack of any in-service pipeline fatigue failures related to corroded wrinkles or buckles may indicate that further research on this subject is not warranted, a better understanding of the interaction between corrosion and fatigue at wrinkles and buckles would be useful to help ensure industry experience to date is correct.

## 1 Introduction

This report was prepared in accordance with the Statement of Work and proposal submitted in response to RFP for Technical Task Order Number 11 (TTO 11) entitled “*Pipe Wrinkle Study.*”

A complex integrity management issue is uncovered when one combines the uncertainties associated with in-line inspection (ILI) tools’ ability to accurately characterize metal loss, in particular that caused by corrosion, near wrinkle bends and buckles with the current lack of a definitive understanding of how best to evaluate the pressure integrity of wrinkles and buckles containing corroded regions. The issue of thermal and pressure cycling of cold bent sections of pipe containing ripples, wrinkles or buckles with localized corrosion introduces separate concerns for fatigue damage due to high localized stress/strain cycling in addition to the pressure integrity issues.

This report presents the results of a review of current in-line inspection (ILI) technology related to detecting general metal loss from corrosion in ripples, wrinkles and buckles. An engineering approach for developing a failure criterion for metal loss on wrinkle bends and buckles is also presented.

*This page intentionally left blank*

## 2 Background

Wrinkle bends and buckles in buried pipelines may be susceptible to metal loss caused by corrosion. Severely distorted and wrinkled sections of pipe can also be subject to localized metal loss due to impacts, or “dings,” from the passage of in-line tools. While current ILI tools can accurately detect localized pitting and general metal loss in cylindrical pipe sections (i.e., in sections without wrinkles or buckles) and standardized procedures are available to assess the pressure integrity of the pipe accounting for metal loss, it is unclear whether current ILI technology can accurately detect these same defects if they occur on or near a wrinkle or buckle because the effects of the pipe wall local curvature on the ILI tool signals can cause inaccuracies.

The standard methods used to assess the pressure integrity of a cylindrical pipe section containing pitting or general metal loss (e.g., ASME B31.G, Modified B31.G, RSTRENG, etc.) are not necessarily appropriate (or at least not thoroughly proven) for evaluating the pressure integrity of pipe sections containing wrinkles or buckles. However, it may be possible to evaluate the pressure integrity of wrinkled pipe sections that contain corrosion using similar methods to those used for evaluating cylindrical pipe. One possibility is by modifying the “calibration factor” that accounts for bulging of the corroded section near burst. This is because the residual stress and strain pattern associated with the wrinkle distortion/deformation will tend to “wash out” as the pipe is strained to near burst pressures (it is well known that wrinkles can tend to flatten when the pipe is subjected to very high pressures).

In the absence of corrosion, the primary integrity concern associated with sections of pipe that have stable ripples, wrinkles or buckles is fatigue damage or failure when the pipeline is subject to pressure and/or temperature cycling. The fatigue demands due to pressure and temperature cycling are increased at locations where the pipe undergoes a change of direction (e.g., at a field bend or a location of high curvature) which is where pipe ripples, wrinkles or buckles are most likely to be found. The most appropriate way to evaluate fatigue damage at these locations is through the use of formal fatigue calculations that consider the geometry of the ripples, wrinkles or buckles, the gross geometry and orientation of the bend, the depth of soil cover and soil type, and the location-specific pressure and temperature differential history.

When sections of pipe containing ripples, wrinkles or buckles also contain corrosion patches, it is clear that a pipe integrity assessment should be based on demand capacity calculations that consider both the pressure integrity and the fatigue failure limit states. One of the aims of this scope of work is to develop a framework for pipe integrity assessment that considers both the pressure integrity and the fatigue failure limit states. The goal of the framework is such that operators are able to assess corroded pipe sections with or without ripples, wrinkles or buckles, as well as pipe containing ripples, wrinkles or buckles with or without corrosion.

*This page intentionally left blank*

### 3 ILI Technology Evaluation

#### 3.1 Scope Statement

“Perform a detailed review to evaluate current ILI technologies used to identify corrosion-related anomalies for their ability to detect such defects in pipe bends containing ripples, wrinkles or buckles. If defect detection is possible, attempt to characterize the accuracy of the resulting defect geometry (lengths, widths and depths) as a function of bend and wrinkle geometry parameters (i.e., bend radius, circumferential extent, wavelength, amplitude and number of lobes present in the ripple/wrinkle/buckle) and as a function of various ILI tool parameters (number of sensors, sensor resolution, sampling rate, tool travel speed, etc.)”

#### 3.2 ILI Technology

Several different ILI tools are available for assessing the integrity of a pipeline. However, selection of these tools must be made carefully based on the particular defect type of interest and the level of accuracy required.

For detection of internal and external metal loss, ultrasonic (UT) and magnetic flux leakage (MFL) tools are most commonly used. Transverse field inspection (TFI), a relatively recent development in ILI technology, has also proven effective in detecting metal loss. Application of electromagnetic acoustic transducers (EMAT) for use in ILI has been available only for a short time and there is relatively little actual field data available. However, EMAT is expected to be applicable to detection of metal loss. Figure 3.1 shows an example of an MFL ILI tool<sup>1</sup>.



**Figure 3.1 MFL ILI Tool**

##### 3.2.1 UT Tools

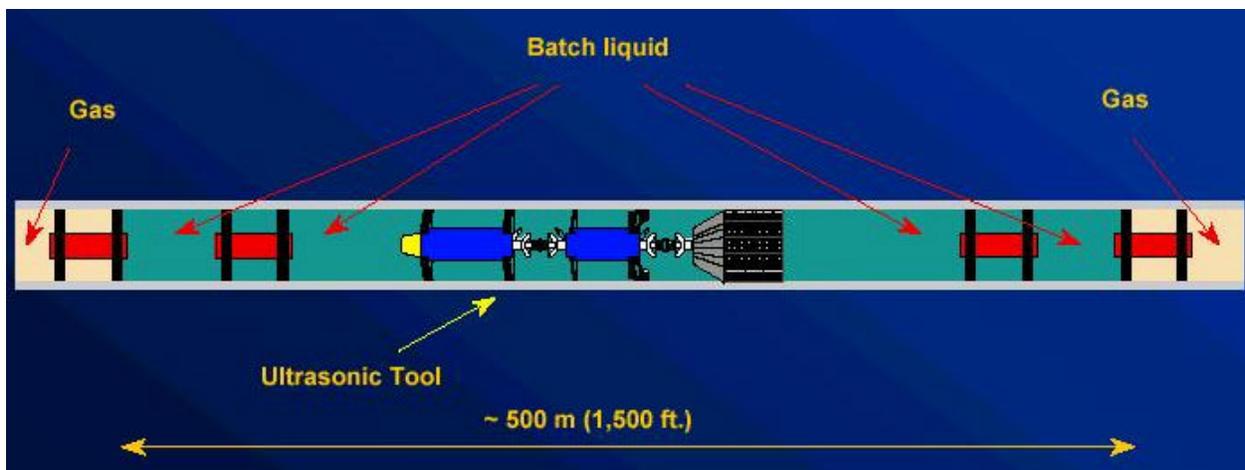
UT tools directly measure the remaining wall thickness as the tool travels through the pipeline. UT tools have transducers that generate ultrasonic signals perpendicular to the pipe wall. An echo is received from both the

---

<sup>1</sup> Vectra MFL by BJ Services

inside and outside surface of the pipe. By timing these return signals, the tool measures the distance from the pipe wall and the pipe wall thickness. Because the transducers require a liquid couplant to transmit the sound wave, UT tools work best in liquid pipelines.

UT tools can be used in gas pipelines, but usually present an increased level of difficulty. Using a UT tool in a gas pipeline requires a liquid slug to be introduced into the pipeline to act as a couplant (see Figure 3.2). This procedure usually requires several pigs in front of the UT tool to hold back the liquid and several pigs behind the ILI tool to help remove the liquid when the tool run is complete. This may require modification to existing pig launchers and receivers in order to accommodate staged launching of the pigs. And, since the couplant is usually water, which is a prime contributor to internal corrosion in gas pipelines, the liquid must be removed when the run is complete. Disposal of the liquid used as the couplant also can present various environmental and cost concerns.



**Figure 3.2 UT tool in a liquid batch<sup>2</sup> (Pipetronix)**

### 3.2.2 MFL Tools

MFL was the first method fully developed for pipeline ILI and has been the most widely used. (Bickerstaff, 2002 and NACE, 2000). The MFL tool induces an axial magnetic flux into the pipe wall between two poles of a magnet. A uniform homogeneous steel pipe without defects creates a uniform distribution of magnetic flux. Metal loss causes a disturbance in the magnetic flux, which, in a magnetically saturated pipe wall, “leaks” out, and sensors detect this leakage. Because the measurement of metal loss is indirect, only limited quantification using complex interpretation techniques is possible. MFL can be used to measure metal loss in both gas and liquid pipelines. Based on the testing needs, varying levels of sensitivity can be used. These levels are:

- Standard, or low resolution
- High resolution

<sup>2</sup> Source Pipetronix

- Extra high resolution (high number of sensors)

Low-resolution tools can size anomalies to a minimum of 20% wall loss with 15-20% accuracy. High-resolution tools can size anomalies to within 10% of wall loss with 10-15% accuracy. Extra high-resolution tools can detect lower levels of corrosion to less than 10% (Bickerstaff, 2002).

### 3.2.3 TFI Tools

TFI tools are a variation of MFL in that a magnetic field is introduced into the pipe wall, however, the direction in which the field is introduced is circumferential as opposed to axial as with traditional MFL. While relatively new, TFI has been used successfully on pipelines for detection of metal loss. These tools operate equally well in liquid and gas pipelines, and are more sensitive to longitudinal anomalies than standard or high-resolution MFL. However, because the tool does not differentiate various defects well, often other tools, such as UT (shear wave), are used to supplement the data gathered. The tool also has difficulty in sizing defects after identification.

### 3.2.4 EMAT Tools

EMAT has recently been developed primarily for the detection of cracks, however, it can also detect internal and external metal loss. The basic principle of EMAT is the generation of an ultrasound compression wave using a magnetic field at the pipe wall's internal surface. Alternating current placed through the coil induces a current in the pipe wall, causing Lorentz forces (Bickerstaff, 2002). After the compression wave has been generated, it travels through the pipe wall and reflects from the surfaces. The returning echo produces a pulse in the transducer. As with traditional UT, the time between firing pulses and the echoes determines the remaining pipe wall thickness.

EMAT tools do not require a couplant and therefore can be used in both liquid and gas pipelines.

## 3.3 Effects of Ripples, Wrinkles and Buckles on ILI Detection of Metal Loss

MFL and UT tools should perform reasonably well in detecting metal loss (within their capability in straight pipe) in areas of relatively smooth deformation. However, in areas where the pipe wall's radius of curvature is small, the sensors will not conform properly to the pipe surface and the minimum detection level can be seriously impacted.

Thus, it is possible that severity of metal loss can be accurately reported in pipe containing mild ripples. However, since wrinkles and buckles are more severely deformed than ripples and tend to exhibit areas of extreme pipe wall curvature, the probability of one of the metal loss tools being able to perform well within these discontinuities is relatively low.

In the smoothest wrinkles and buckles, it is possible to get a metal loss signal, but it should not be relied upon for evaluation of the metal loss. As wrinkles and buckles become deeper, they become more and more abrupt and the probability of accurate metal loss detection becomes lower. In these situations, evaluation of any metal loss signal received is not practical. UT devices perform even worse than MFL devices in these situations because of the loss of the return signal.

In summary, the more extreme the deformation, the more serious the defect, and the more probable that metal loss is coexisting. Further, it is more probable that metal loss will not be detected by metal loss devices when the deformation is severe. In no event can any of the metal loss ILI devices be reliably used to determine the presence of metal loss in pipe deformations.

## 4 Summary of Demand Capacity Framework for Corroded Wrinkles

The primary concern for corrosion in a pipeline is how it will affect the pressure capacity of the pipe. The pipeline industry has well-accepted procedures in place for evaluating the pressure capacity of corroded pipelines. These procedures are supported by a database of hundreds of burst test results. Pipeline operators and consultants have a wealth of experience with this type of evaluation.

Once a wrinkle is discovered in a pipeline, the primary concern is the stability of the wrinkle (e.g., are the wrinkle deformations likely to increase due to continued settlement?). If it is unlikely that the deformations in a wrinkle will increase (i.e., the wrinkle is stable), the primary concern becomes the potential for fatigue damage in or near the wrinkle. There are currently no universally accepted guidelines or specific criteria that can be used to limit the geometry of pipeline wrinkles based on fatigue considerations. However, it is understood that the B31.8 Code Committee is presently considering an agenda item allowing for wrinkles with peak-to-trough heights of up to 1% of the pipe diameter based on recent research (Rosenfeld et. al 2002). It is also believed that the B31.4 Code Committee is likewise considering an agenda item related to the acceptance of mild wrinkles.

When stable wrinkles in pipelines are found to contain corrosion, the concerns should be the same as those expressed above:

- Is the pressure integrity of the pipeline at risk?
- Is the corroded wrinkle at risk of experiencing fatigue damage or failure?

The first and most important step in the recommended framework is to evaluate the pressure integrity of the corroded wrinkle. It is believed that the geometry of the wrinkle is unlikely to have a significant effect on the burst capacity of the corroded pipe section since the plastic strains in the wrinkle will tend to “wash out” from the large strains associated with the burst pressure. For this reason, it is recommended that the corrosion evaluation be performed by treating the pipe as if it was cylindrical (i.e., neglecting the wrinkled geometry). We are aware of some proprietary burst tests on wrinkled pipe specimens that support this analysis approach. If the pressure integrity of the pipe is affected by the corrosion, then the operator should proceed based on the appropriate CFR integrity management rules.

Once the pressure integrity has been evaluated, the next step is to evaluate the fatigue integrity of the wrinkle, disregarding the presence of corrosion. The analysis approach for this step is far less established and more time consuming than the procedures used to evaluate pressure integrity. The highlights of the fatigue evaluation are summarized as follows:

- Develop representative annual “histograms” of pressure and temperature cycles for the pipeline at the location of interest.
- Develop and analyze a case-specific “global” buried pipe model at the location of interest to develop estimates of the global loads and nominal stresses at the wrinkle.

- Develop and analyze a case-specific “local” FEA model of the wrinkle geometry of interest to establish estimates of the stress concentration factor (SCF) for internal pressure and bending moment loads.
- Combine the pressure and temperature cycle histograms, with the corresponding nominal stresses and the pressure and bending moment SCFs to obtain the localized fatigue stress demands at the wrinkle.
- Use fatigue “S-N” curves to compute estimates of the annual fatigue damage at the wrinkle using a fatigue usage factor (where 0.0 corresponds to zero fatigue damage and 1.0 corresponds to fully consumed fatigue life). The fatigue life in years is equal to the inverse of the annual usage factor. Compare the *design* fatigue life (computed using a “design” fatigue curve containing a significant safety factor on stress or cycles) to the design life of the pipeline. If the design fatigue life is longer than the design life of the pipeline, the wrinkle satisfies the type of fatigue criteria that would be used for the design of a new pipeline, including a significant safety factor (as opposed to performing a serviceability assessment of an existing pipeline). If the design fatigue life is shorter than the design life of the pipeline, the wrinkle may still be considered as acceptable depending on the safety factor in the design S-N curve.

Once the fatigue integrity of the wrinkle has been considered, the fatigue analysis can be extended to consider the effects of corrosion within the wrinkle. The only change to the evaluation approach is that the detailed “local” FEA model of the wrinkle is modified to include characterization of the corrosion. The corrosion is typically characterized as a rectangular patch. Depending on the geometry of the corrosion (e.g., its length, width and depth and its location with respect to the peak of the wrinkle), the SCFs are likely to increase relative to those of the un-corroded wrinkle.

As noted above, the fatigue analysis aspects of the proposed framework are far less established and more time consuming than the procedures used to evaluate pressure integrity. However, the application of FEA methods is very well established in the pipeline and piping research industry, and the use of FEA as a tool for performing pipeline structural integrity and serviceability assessments is becoming more common. FEA methods used in combination with additional experimental data represent the most promising means of evaluating complex pipe stress and deformation problems such as assessing the fatigue behavior of corroded wrinkles.

#### **4.1 Illustrative Example**

Application of the demand capacity framework for corroded wrinkles described above is demonstrated in the following example.

##### **4.1.1 Problem Parameters**

For this example, it is assumed that a corroded wrinkle has been detected on a liquids pipeline having the basic parameters presented in Table 4-1. The pipeline has been in operation for 10 years and the normal operating pressure at the location of the corroded wrinkle is approximately 700 psi. It has been determined that the soil support at the wrinkle is stable (i.e., no ongoing ground movement) and that the maximum

corrosion depth is 50% of the wall thickness with a maximum length of 10 inches. The wrinkled geometries have inward deformations of approximately 1 inch and a wavelength of approximately 9 inches.

**Table 4-1 Example Problem Parameters**

Parameter	Value
Diameter	24 Inches
Wall thickness	0.266 inches
SMYS	60,000 psi
MAOP	960 psi
$\Delta T$ (tie-in to operating)	+80°F
Design life	25 years

#### 4.1.2 Pressure Capacity of Corroded Section

The method chosen for evaluating the pressure capacity of the corroded section in this example is the modified B31G procedure as defined by the following formula:

$$MAOP = \left( \frac{2 \cdot t \cdot F}{D} \right) \cdot S' \cdot \left( \frac{1 - 0.85 \frac{a}{t}}{1 - 0.85 \frac{a}{t} \cdot M^{-1}} \right)$$

where:

$a$  is the defect depth

$D$  is the pipeline diameter

$F$  is the design factor

$S'$  is the flow stress of the pipe material (SMYS + 10 ksi)

$t$  is the wall thickness of the pipe

$M$  is Folias' bulge factor given by:

$$M = \sqrt{1 + 0.6275 \cdot \frac{L^2}{D \cdot t} - 0.00375 \cdot \left( \frac{L^2}{D \cdot t} \right)^2} \quad \text{for } \frac{L^2}{D \cdot t} \leq 50$$

$$M = 3.3 + 0.32 \cdot \frac{L^2}{D \cdot t} \quad \text{for } \frac{L^2}{D \cdot t} > 50$$

Values for the parameters not presented in Table 4-1, as well as the results of intermediate calculations are presented in Table 4-2.

**Table 4-2 Modified B31G Calculations**

Parameter/Calculation	Value
$\frac{a}{t}$	0.5
$F$	0.72
$S'$	70,000 psi
$\frac{L^2}{D \cdot t}$	15.66
$M$	3.15
$MAOP$	743 psi

Since the result of the modified B31G calculation indicates that the MAOP for the corroded section is less than the original MAOP of the line, either a derating of the line to the lower pressure or repair of the location are required. For this example, derating is considered a viable option, thus further evaluation is warranted to determine whether fatigue of the corroded wrinkle is a concern.

#### 4.1.3 Fatigue Demand Capacity Evaluation

For the purposes of this example, pressure cycle and temperature cycle spectra as given in Table 4-3 and Table 4-4 have been postulated. In addition, these events have been considered to be non-coincident and thus will each be evaluated separately.

**Table 4-3 Pressure Cycle Spectrum Over Typical One Year Time Period**

Pressure Range (psi)	Number of Cycles (n)
700	5
500	50
300	500
100	5000

**Table 4-4 Temperature Cycle Spectrum Over Typical One Year Time Period**

Temperature Differential (degrees F)	Number of Cycles (n)
80	5
60	25
40	250
20	2500

A detailed shell finite element analysis was undertaken of a representative wrinkle geometry. The height of the wrinkle was approximately 1 inch (inward) and the wavelength of the wrinkle was about 9 inches. The wrinkle was assumed to extend over approximately 50% of the pipe circumference, and was well separated from the nearest girth weld. The peak of the wrinkle was located at the intrados location in a side bend and did not span across the longitudinal seam. Elastic analysis of the FEA mesh of this wrinkle for internal pressure loading indicated that the stress concentration factor or *SCF* (i.e., the ratio of the maximum local stress to the nominal hoop stress  $PD_i/2t$ ) for internal pressure load was 2.54. Elastic analysis of the FEA model of this wrinkle for bending moment loading indicated that the *SCF* (i.e., the ratio of the maximum local stress to the nominal bending stress  $M/Z$ ) for bending moment loads was 2.72.

To illustrate the factor of safety of the design versus mean fatigue relationships, the fatigue evaluation was undertaken using both the mean and design fatigue S-N relationships developed in Appendix A. The *mean* fatigue S-N relationship is summarized as follows:

$$i \cdot S = 490 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 20 \quad \text{for } N > 8.8 \times 10^6$$

Applying a factor of safety of 2.0 on stress range leads to the following *design* S-N relationship:

$$i \cdot S = 245 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 10 \quad \text{for } N > 8.8 \times 10^6$$

In these relationships, *S* is the nominal stress range (in ksi), *N* is the number of stress reversals to failure, and *i* is the fatigue effective stress intensification factor (*SIF*). The “*C*” term in these equations is equal to Markl’s material constant, which can be taken as 245 ksi for carbon steels. As discussed in Appendix B, Section B.4, the fatigue effective *SIF* can be taken as:  $i = SCF/2$ . The steps for evaluating the fatigue damage due to pressure cycles at this wrinkle are as follows:

1. Compute the nominal hoop stress due to the various pressure ranges using the formula:

$$S_H = \frac{P \cdot D_i}{2 \cdot t}$$

where:

*D<sub>i</sub>* is the inside diameter of the pipe,

*P* is the pressure range, and

*t* is the wall thickness of the pipe.

2. Compute the localized fatigue demand measure  $i \cdot S = i \cdot S_H = SCF/2 \times S_H$ .
3. Since pressure cycles result in stress-controlled loading (see Appendix B, Section B.2.2), use the stress-controlled material constant *C* equal to 2/3 of the displacement-controlled material constant *C* ( $C = 2/3 \cdot C$ ) in the *mean* and *design* fatigue curves. The endurance limits (20 ksi for the mean curve and 10 ksi for the design curve) are also scaled by the 2/3 factor. For localized fatigue

demand measure values ( $i \cdot S$ ) below the endurance limits, the corresponding  $N$  value is 8. For  $i \cdot S$  values above the endurance limits, solve for the number of cycles  $N$  on the mean and design fatigue S-N curves using:

$$N = \frac{1}{\left(\frac{i \cdot S}{490 \cdot 2/3}\right)^5} \quad \text{for the mean curve}$$

$$N = \frac{1}{\left(\frac{i \cdot S}{245 \cdot 2/3}\right)^5} \quad \text{for the design curve}$$

The results from these evaluation steps are presented in Table 4-5.

**Table 4-5 Pressure Cycle Fatigue Results**

Pressure Range (psi)	Annual Number of Cycles, n	Hoop Stress, $S_H$ (ksi)	Localized Fatigue Demand (ksi)	Mean N Value	Design N Value	Mean n/N	Design n/N
700	5	30.9	39.2	40,188	1,256	0.00012	0.00398
500	50	22.1	28.1	212,321	6,635	0.00023	0.00754
300	500	13.2	16.8	2,779,571	86,682	0.00018	0.00576
100	5000	4.4	5.6	$\infty$	$\infty$	0	0
Annual Usage Factor (Sn/N)						0.00054	0.01727
Fatigue Life (years)						1,863	58

The steps for evaluating the fatigue damage due to thermal cycles are as follows:

- The nominal longitudinal stress demand ( $S_L$ ) in a buried pipe subject to a temperature change should be computed based on buried pipe stress analysis of the configuration of interest. (Note: For this example, buried pipe analysis results published in “*Development of Acceptance Criteria for Mild Ripples in Pipeline Filled Bends*” (Rosenfeld, et. al 2002) for a 24-inch diameter buried pipe were used.)
- Compute the localized fatigue demand measure  $i \cdot S = i \cdot S_L = SCF/2 \cdot S$ .
- Since thermal cycles result in displacement- or strain-controlled loading, the basic C factor in the S-N relationships defined in Section B.2.2 is used to represent the fatigue capacity. For localized fatigue demand measure values ( $i \cdot S$ ) below the endurance limits (20 ksi for the mean curve and 10 ksi for the design curve), the corresponding  $N$  value is 8. For  $i \cdot S$  values above the endurance limits, solve for the number of cycles  $N$  on the mean and design fatigue curves corresponding to the above  $i \cdot S$  values using:

$$N = \frac{1}{\left(\frac{i \cdot S}{490}\right)^5} \quad \text{for the mean curve}$$

$$N = \frac{1}{\left(\frac{i \cdot S}{245}\right)^5} \quad \text{for the design curve}$$

The results from these evaluation steps are presented in Table 4-6.

**Table 4-6 Temperature Cycle Fatigue Results**

Temperature Differential (degrees F)	Annual Number of Cycles, n	Longitudinal Stress, $S_L$ (ksi)	Localized Fatigue Demand (ksi)	Mean N Values	Design N Values	Mean n/N	Design n/N
80	5	26.2	35.6	494,004	15,438	0.000010	0.000324
60	25	19.6	26.7	2,081,729	65,054	0.000012	0.000384
40	250	13.0	17.7	$\infty$	508,117	0	0.000492
20	2500	6.5	8.8	$\infty$	$\infty$	0	0
Annual Usage Factor (Sn/N)						0.000022	0.001200
Fatigue Life (years)						45,455	833

For this example, the pressure cycles are the dominant source of fatigue damage. When the annual fatigue usage ratios due to pressure cycles and thermal cycles are combined, the *mean* and *design* fatigue lives of this wrinkle are 1,779 and 54 years, respectively.

If the FEA of the wrinkle described above is extended to include characterization of the 10-inch long, corrosion patch with 50% wall loss, it is postulated that the *SCFs* for pressure and moment loading were both increased by 15% (the *SCF* for internal pressure load was increased from 2.54 to 2.92 and the *SCF* for bending moment loads was increased from 2.72 to 3.13). Steps 1 through 6 described above were repeated using the increased *SCF* values associated with the corroded wrinkle (in effect the localized fatigue demand measure  $i \cdot S$  was increased by 15%). For the corroded wrinkle, the resulting *mean* and *design* fatigue lives are 929 and 29 years, respectively.

Several points can be made based on this example fatigue evaluation:

- For cases where the stresses are above the endurance limit, the ratio of “mean” fatigue life to the “design” fatigue life is equal to 32. This factor of 32 represents the factor of safety on cycles and is equal to the factor of safety of 2 on stress raised to the power 5:  $32=2^5$ .
- The presence of corrosion in the wrinkle resulted in an increase in the localized stresses in the wrinkle, which were already larger than the nominal stresses in the pipe. A 15% increase in the localized stresses

due to corrosion, resulted in an approximate factor of 2 reduction in both the mean and design fatigue lives. This approximate factor of 2 corresponds to the increased stress raised to the power 5:  $2 \sim 1.15^5$ .

- Evaluation of the postulated wrinkles with and without corrosion using a design fatigue curve resulted in design fatigue lives of 27 and 54 years, respectively (this assumes that the evaluated anomaly has been present in the pipeline since startup). Both of these design fatigue lives exceed the 25-year design life of the pipeline. This means that even the corroded wrinkle would satisfy the type of fatigue design criteria that would be used for the design of a new pipeline, including a significant safety factor.

For this example, it would be concluded that fatigue of the wrinkle (with or without corrosion) does not pose a greater hazard than pressure alone. In other words, evaluation of the corrosion using established industry procedures for pressure capacity (and derating the MAOP or repairing the corrosion if necessary) would take precedence over fatigue concerns for this case.

## 5 Conclusions and Recommendations

### 5.1 Conclusions Regarding ILI Capabilities

While current in-line inspection (ILI) tools can accurately detect localized pitting and general metal loss in cylindrical pipe sections (i.e., in sections without wrinkles or buckles), the ability of ILI tools to accurately characterize metal loss due to corrosion in the vicinity of wrinkle bends and buckles is uncertain.

MFL and UT tools should perform reasonably well in detecting metal loss (within their capability in straight pipe) in areas of relatively smooth pipe wall deformations. However, in areas where the pipe wall's radius of curvature is small, the sensors will not conform properly to the pipe surface and the minimum detection level can be seriously impacted.

Thus, it is possible that severity of metal loss can be accurately reported in pipe containing mild ripples. However, since wrinkles and buckles are more severe than ripples and tend to exhibit areas of extreme pipe wall curvature, the probability of one of the metal loss tools being able to perform well within these discontinuities is relatively low.

In the smoothest wrinkles and buckles, it is possible to get a metal loss signal but it should not be relied upon for evaluation of the metal loss. As wrinkles and buckles become deeper, they become more and more abrupt and the probability of accurate metal loss detection becomes lower. In these situations, evaluation of any metal loss signal received is not practical. The UT devices perform even worse than the MFL devices in these situations because of loss of the return signal.

In summary, the more severe the deformation, the more serious the defect, and the more probable that metal loss is coexisting. Further, it is more probable that metal loss will not be detected by metal loss devices when the deformation is severe. In no event can any of the metal loss ILI devices be reliably used to determine the presence of metal loss in deformation.

### 5.2 Conclusions Regarding Pipeline Integrity at Corroded Wrinkles

The primary concern for corrosion in a pipeline is how it will affect the pressure capacity of the pipe. The pipeline industry has well-accepted procedures in place for evaluating the pressure capacity of corroded pipelines. These procedures are supported by a database of hundreds of burst test results. Pipeline operators and consultants have a wealth of experience with this type of evaluation.

When a wrinkle is discovered in a pipeline, and subsequently verified, the primary concern is the stability of the wrinkle. If an unknown wrinkle is identified, an evaluation must be conducted to determine whether the wrinkle deformations are likely to increase due to ongoing settlement or other causes. If it is unlikely that the deformations in a wrinkle will increase (i.e., the wrinkle is stable), the primary concern becomes the potential for fatigue damage in/near the wrinkle. Although some significant research and development efforts have been undertaken, there are currently no universally accepted guidelines or specific criteria that can be used to limit the geometry of wrinkles in pipelines based on fatigue considerations. The most appropriate approach for evaluating pipeline wrinkles is a formal fatigue damage assessment that considers the pressure

and temperature cycling and the soil conditions at the location of the wrinkle, the geometry of the wrinkle, and the fatigue resistance of the pipe (usually characterized based on a S-N curve).

When (stable) wrinkles in pipelines are found to contain corrosion, the concerns should be the same as those expressed above:

- Is the pressure integrity of the pipeline at risk?
- Is the corroded wrinkle at risk of experiencing fatigue damage or failure?

The first and most important step in the recommended framework is to evaluate the pressure integrity of the corroded wrinkle. It is unlikely the geometry of the wrinkle will have a significant effect on the burst capacity of the corroded pipe section since the plastic strains in the wrinkle will tend to “wash out” at the large strains associated with the burst pressure. For this reason, it is recommended that the corrosion evaluation be performed by treating the pipe as if it was cylindrical (i.e., neglecting the wrinkled geometry). We are aware of some proprietary burst tests on wrinkled pipe specimens that support this analysis approach. If the pressure integrity of the pipe is affected by the corrosion, then the operator should proceed based on the appropriate CFR integrity management rules.

Once the pressure integrity has been evaluated, the next step is to evaluate the fatigue integrity of the wrinkle, neglecting the presence of corrosion. The analysis approach for this step is far less established and more time consuming than the procedures used to evaluate pressure integrity. The highlights of the fatigue evaluation (see Appendix B for more details) are summarized as follows:

- Develop representative annual “histograms” of pressure and temperature cycles for the pipeline at the location of interest.
- Develop and analyze a case-specific “global” buried pipe model at the location of interest to develop estimates of the global loads and nominal stresses at the wrinkle.
- Develop and analyze a case-specific “local” FEA model of the wrinkle geometry of interest to establish estimates of the stress concentration factor (SCF) for internal pressure and bending moment loads.
- Combine the pressure and temperature cycle histograms, with the corresponding nominal stresses and the pressure and bending moment SCFs to obtain the localized fatigue stress demands at the wrinkle.
- Use fatigue “S-N” curves to compute estimates of the annual fatigue damage and the fatigue life at the wrinkle. Compare the *design* fatigue life (computed using a “design” fatigue curve containing a significant safety factor on stress or cycles) to the design life of the pipeline. If the design fatigue life is longer than the design life of the pipeline, the wrinkle satisfies the type of fatigue criteria that would be used for the design of a new pipeline, including a significant safety factor. If the design fatigue life is shorter than the design life of the pipeline, the wrinkle may still be considered as acceptable depending on the safety factor included in the design S-N curve.

Once the fatigue integrity of the wrinkle has been considered, the fatigue analysis should be extended to consider the effects of corrosion within the wrinkle. The only change to the evaluation approach is that the detailed “local” FEA model of the wrinkle is modified to include a characterization of the corrosion. The corrosion is typically characterized as a rectangular patch. Depending on the geometry of the corrosion (e.g., its length, width and depth and its location with respect to the peak of the wrinkle), the SCFs are likely to increase relative to those of the un-corroded wrinkle.

The fatigue analysis aspects of the proposed framework are far less established and more time consuming than the procedures used to evaluate pressure integrity. However, the application of FEA methods is very well established in the pipeline and piping research industry and the use of FEA as a tool for performing pipeline structural integrity and serviceability assessments is becoming much more common. FEA methods used in combination with additional experimental data represents the most promising means of evaluating complex pipe stress and deformation problems such as assessing the fatigue behavior of corroded wrinkles.

Based on the combined experience of the project team and upon discussions with industry experts, pipeline failures due to fatigue in corroded ripples, wrinkles or buckles could not be identified. Moreover, there is a lack of full-scale experimental evaluations of corroded pipes that were designed to produce fatigue failures in the corrosion; most corroded pipe tests are aimed at evaluating burst pressure. However, pipelines that have experienced external corrosion at elbows were identified in the research. In this case, there was concern that the corrosion within the elbow would increase the flexibility and stress intensification effects with a potential reduction in the fatigue capacity of the elbow. Detailed proprietary FEA and fatigue testing of both uncorroded and corroded elbows led to the conclusion that evaluation of the pressure capacity of the corrosion by any established methodology (e.g., B31G, RSTRENG), and derating or repairing if the corrosion is severe enough should take precedence over fatigue concerns. Using established pressure integrity methods should result in derating or repairing the pipeline long before fatigue should be a concern for all but the most extreme scenarios of cyclic stress demand. The same conclusion can be applied to corroded wrinkles.

### **5.3 Recommendations**

The proposed framework presented in this report is based in large part on theoretical information. With additional research data on fatigue in corroded rippled or wrinkled pipe and burst capacity of corroded wrinkled pipe, this framework could likely be enhanced. Even though the apparent lack of any fatigue failures related to corroded wrinkles or buckles on in-service pipelines may indicate that further research is not warranted, a better understanding of the interaction between corrosion and fatigue at wrinkles and buckles would be useful to help ensure that experience to date is not biased in some manner.

*This page intentionally left blank*

## 6 References

Bickerstaff, Robert, Mark Vaughn, Gerald Stoker, Michael Hassard and Mark Grrett, “*Review of Sensor Technology for In-Line Inspection of Natural Gas Pipelines*”, Sandia National Laboratories, 2002.

Rosenfeld, M. J., Hart, J. D., Zulfiqar, N., Gailing, R., “*Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends*”, IPC02-27124, ASME International Pipeline Conference, Calgary, Alberta, Canada, September 29-October 3, 2002.

*This page intentionally left blank*

# **APPENDIX A**

## **FATIGUE DESIGN CURVE RECOMMENDATIONS BY BERKELEY ENGINEERING AND RESEARCH, INC.**

*This page intentionally left blank*



May 7, 2004

Dr. Jim Hart  
SSD, Inc.  
6119 Ridgeview Court, Suite 400  
Reno, NV 89509

**Re: Design Fatigue Curve for the OPS Pipe Wrinkle Study**

Dear Dr. Hart:

We have investigated the available fatigue curves that could be used for design in the OPS Pipe Wrinkle Study. A recommended composite fatigue curve for full penetration weld metal is developed and provided.

This study revisits, combines and organizes prior work performed by BEAR for the Trans-Alaska Pipeline System (TAPS) [A.1] and the ASME Mechanical Design Technical Committee (MDC) B31 Code for Pressure Piping. Comparisons were made between design fatigue curves given in the American Society of Mechanical Engineers (ASME) Section III Pressure Vessel Code, the B31 Code for Pressure Piping (MarkI), and the American Welding Society (AWS). All of these curves make specific provisions for weld metal except the ASME curve. However, all of these fatigue curves can be reasonably reconciled as equivalent when adjusted for their different assumed: (1) safety factors, (2) weld or base metal, (2) mean and biaxial stress states and (4) elastic or elastic-plastic analysis.

The Trans-Alaska Pipeline System (TAPS) work required the development of “design” and “decision” S-N fatigue curves for the evaluation of dents in both weld and base metal. The fatigue curves developed were based on a combination of the AWS and ASME design fatigue curves [A.2,A.3]. Despite numerous declarations to the contrary by Civil and Mechanical Engineering Code Committee members, it was found that these fatigue curves give essentially the same values when properly adjusted for surface roughness, differences in applied safety factors, etc.

The same is true for the MarkI and ASME design fatigue curves. Safety factors removed and adjusted for differences described above, these curves can be shown to give almost identical results. Thus, sufficient understanding of fatigue data exists such that a MarkI based fatigue curve can be used with confidence in the OPS wrinkle study for elastic and strain based analysis and can be modified to cover new materials only characterized by other fatigue curves.

## The Markl and ASME Fatigue Curves

The Markl curve is considered valid from 20 to 2 million cycles by its' author [A.4]. A comparison of the Markl, ASME and AWS-X curves follows with the associated calculations given in the attached Mathcad worksheet. On page 1 of the calculations, the values for N (cycles) used in the ASME code to define their fatigue curve [A.3] are determined with Equation 1 and the corresponding stress values determined without the safety factors applied with Equation 2 [A.5].

Reasonably approximate stress values for weld metal are determined with Equation 3 by dividing the stress values determined in Equation 2 by a factor of 2 [A.6] and applying a linear mean stress adjustment in the high cycle region above  $10^5$  cycles as shown in Equation 10 [A.7]. Equation 5 passes through the mean of the Markl fatigue data without safety factors.

As shown in Figure 1, the Markl fatigue curve falls well below the ASME curve (adjusted for weld metal [A.6] and mean stress [A.7]), particularly in the low-cycle fatigue (LCF) region. This comparison is invalid because of the large plastic strains that occur in the LCF region. The ASME fatigue data is based on actual elastic-plastic strain multiplied by the elastic modulus [A.5], whereas the Markl data stresses are based on nominally elastic moment values [A.4].

## ASME Elastic Plastic Adjustment

The ASME code provides a simplified adjustment factor to approximate an elastic-plastic analysis with the results of an elastic analysis [A.7,A.8]. An elastic-plastic factor,  $K_e$ , is determined and multiplied by the elastically determined stress prior to entering the ASME design fatigue curve. Applying this factor to the Markl curve, Equations 6 and 7, almost bring it into agreement with the ASME curve, as shown in Figure 1.

Close examination of  $K_e$  (see Figures 1, 2 and 3) shows that its adjustment is conservatively held constant below approximately 100 cycles. Numerous fatigue curves in the literature indicate that cyclic stresses should continue to increase in the LCF region as cycles decrease all the way down to  $\frac{1}{4}$  cycles. Furthermore, the value of  $m$  for carbon steels (3.0) used in the determination of  $K_e$  is a conservative lower bound value based on comparison with bi-axial fatigue test data [A.9]. This is appropriate for a design curve as a lower bound value for  $m$  gives an upper bound value for  $K_e$  and provides a conservatively high equivalent elastic-plastic stress to enter the ASME code fatigue curve with.

Depending on the ratio of biaxial stress, the value of  $m$  can be shown to vary between 3 and 5. Choosing a value for  $m$  of 3.5 and assuming a continuous correction, causes the adjusted Markl curve to almost perfectly fall on top of the ASME curve as shown in Figure 4. This is consistent with a range of  $m$  between 3 and 5. Furthermore, the Markl and ASME fatigue curves are supported by considerable fatigue data. Thus, multiplying the Markl fatigue curve by  $K_{en}$  based on an  $m$  value of 3.5 should provide an excellent fatigue curve for an elastic equivalent stresses determined from an elastic-plastic analysis. For an elastic analysis, the stresses determined should be compared directly to the Markl curve.

In practice, the ASME code fatigue curve is assumed to reach an endurance limit at  $10^7$  cycles. Prior proprietary high cycle fatigue test work on full size pipe welds and data provided in Reference 11 indicates that an endurance limit range of 20 ksi (no safety factor) mean for butt welded piping connections. This stress range value occurs at  $8.8 \times 10^6$  cycles for the Markl curve, very close to  $10^7$  cycles.

### Comparison with the AWS X-curve

The Markl curve provides similar results to the ASME fatigue curve when compared in a consistent manner. However, the ASME curve data is based on the testing of base metal material. Thus, a second comparison to a weld metal based fatigue curve is appropriate. Both the Markl and AWS X-curve require full penetration welding. Assuming the endurance limit discussed above for the Markl curve, 20 ksi, and a safety factor of 2 on stress (Equation 11), a comparison is shown in Figure 5. For weld metal fatigue data, a factor of 2 on stress corresponds to approximately 2 standard deviations from the mean [A.10].

The endurance limit adjusted Markl design curve and AWS X-curve compare well in the high cycle region. In the low cycle region the Markl curve is significantly lower. However, piping and large tubular structure fatigue test data in this region more closely match the Markl design curve than the AWS X curve.

### Recommended Design Fatigue Curve

Based on the above assessment and reasonable agreement with both the ASME and AWS X fatigue curves, the author recommends using the Markl fatigue curve as given in Equation 11 with a fatigue endurance limit of 20 ksi range ( $8.8 \times 10^6$  cycles) as a mean fatigue curve for use with elastic analysis results. A safety factor of 2 on stress is suggested for a design curve, giving a fatigue endurance limit of 10 ksi range. Welds evaluated with the recommended fatigue curve should be held to the detailing and undercut limitations given in the AWS structural welding code [A.2].

The user may wish to apply alternate factors of safety depending on the application, flaw inspection criteria and corrosion environment. To allow for significant flaw sizes, an adjustment based on fracture mechanics is recommended. An adjustment to the endurance limit can be determined based on a stress intensity threshold value [A.11] and applied as shown in Equation 12. Prior proprietary work by BEAR indicates the endurance limit should be reduced by 42% for maximum flaws that are 2 inches long and one-quarter wall in depth.

For use with elastic-plastic analysis results, elastic equivalent stresses should be determined from the strain results and used to enter the (MarklKen) fatigue curve generated by multiplying the Markl curve by *Ken* based on an *m* value of 3.5. Note, cyclic stresses are given in terms of range in all the above cited equations and figures.

For base metal piping material, the same recommended Markl curve can be used by increasing the stress range by a factor of 2 [A.6]. This corresponds to an *i* value of  $\frac{1}{2}$  in B31 Piping Code fatigue equations

[A.12]. This method can be shown to be conservative. A less conservative biaxial base metal design fatigue curve derived for the Alaska Pipeline is given in Reference [A.1].

To evaluate weld or base metal material below 20 cycles of life, a log-linear interpolation is recommended between the Markl curve stress range at  $N=20$  cycles and stress (or strain) values determined at  $N=1/4$  cycle (e.g., via burst testing of pipe or burst analysis). Burst testing and/or analysis take into account the significant material properties and the biaxial piping stresses. Burst analysis based stresses can be determined using RSTRENG [A.13], B31G [A.14], API 579 [A.15] or equivalent biaxial plastic instability analysis methods.

If you have any questions or comments, please contact me at 510-549-3300, extension 1.

Best Regards,

BERKELEY ENGINEERING AND RESEARCH, INC.

Glen Stevick, Ph.D., P.E.

## References

- A.1. Stevick, G. R., Hart, J. D., Flanders, B., “Fatigue Curves for Damage Calculations For a Dented and Ovalled Section of the Trans-Alaska Pipeline System”, 1998 International Pipeline Conference, Calgary, Alberta Canada.
- A.2. D1.1-1999 Structural Welding Code - Steel, AWS, 550 NW LeJeune Rd., Miami, FL.
- A.3. ASME Boiler and Pressure Vessel Code, Section III and Section VIII, Division 2, 1999.
- A.4. Markl, A. R. C., “Piping-Flexibility Analysis”, Trans of the ASME, pp. 127-49, 1955.
- A.5. “Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Section III and Section VIII, Division 2”, ASME, 1969.
- A.6. "Relationships Between Stress Intensification Factors And Stress Concentration Factors", Draft Report Prepared For: Offshore And Onshore Design Application Supervisory Committee PRC International (PRCI) and Gas Research Institute, By SSD, Inc. and Kiefner And Associates, Inc., December, 2003
- A.7. Cooper, W. E., “The Initial Scope and Intent of the Section III Fatigue Design Procedures,” PVRC Workshop on Cyclic Life and Environmental Effects in Nuclear Power Plants, Clearwater Beach, Florida, January 1992.
- A.8. Jaske, C. E., “White Paper on Weld Fatigue-Strength Reduction and Stress Concentration Factors”, Project R235, PVRC, Columbus, OH, October 7-9, 1996.
- A.9. Rahka, K, “Review of Strain Effects on Low-Cycle Fatigue of Notched Components”, PVP-Vol. 263, High Pressure – Codes, Analysis and Applications, ASME 1993.
- A.10. “Evaluation of Fatigue Tests and Design Criteria on Welded Details”, National Cooperative Highway Research Program Report 286, Project 12-15(5) 1982.
- A.11. Lindley, T.C., “Near Threshold Fatigue Crack Growth: Experimental Methods, Mechanisms, and Applications”, Elsevier Applied Science Publishers, England, 1984.
- A.12. American ASME, Piping Code B31.3-2004 Process Piping.
- A.13. Vieth, Patrick H.; Kiefner, John F., “A Modified Criteria for Evaluating the Remaining Strength of Corroded Pipe.”, PRCI, December 1989.
- A.14. ASME B31G – 2004 Manual for Determining Remaining Strength of Corroded Pipelines: Supplement to B31 Code-Pressure Piping.

A.15. API 579 Recommended Practice on Fitness for Service, 2004.

**Fatigue Curve Comparisons**

**1.0 ASME Pressure Vessel Code**

**Equation (1)**  
Set N index values

$$\begin{array}{l}
 i := 1 \dots 25 \quad \text{ORIGIN} = 1 \quad Nn_i := 1 \cdot \left\{ \begin{array}{l}
 10^{\frac{\text{trunc}\left(\frac{i-1}{3}\right)}{3}} \quad \text{if } \frac{i-1}{3} - \text{trunc}\left(\frac{i-1}{3}\right) \leq .001 \\
 2 \cdot 10^{\frac{\text{trunc}\left(\frac{i-1}{3}\right)}{3}} \quad \text{if } \frac{|i-2|}{3} - \text{trunc}\left(\frac{i-2}{3}\right) < .001 \\
 5 \cdot 10^{\frac{\text{trunc}\left(\frac{i-1}{3}\right)}{3}} \quad \text{if } \frac{|i-3|}{3} - \text{trunc}\left(\frac{i-3}{3}\right) < .001 \\
 0 \quad \text{otherwise}
 \end{array} \right. \\
 E := 30 \cdot 10^6 \cdot \text{psi} \quad A := 68.5 \\
 \text{ksi} = 1000 \cdot \text{psi} \quad B := 21645 \cdot \text{psi}
 \end{array}$$

$$\text{Sasme}_i := \left( \frac{E}{4 \cdot \sqrt{Nn_i}} \cdot \ln\left(\frac{100}{100 - A}\right) + B \right) \cdot 2 \quad \text{Equation (2)}$$

Carbon Steel "Best Fit" Curve (Range)

$$\text{SasmeW}_i := \left( \frac{E}{4 \cdot \sqrt{Nn_i}} \cdot \ln\left(\frac{100}{100 - A}\right) + B \right) \quad \text{Equation (3)}$$

Adjust for weld metal, divide by 2 (Still Range)

Mean Stress Adjustment for ASME curve

$$\begin{array}{l}
 R1 := 1 \quad N1 := 2 \cdot 10^5 \quad R2 := .75 \quad N2 := 10^7 \\
 C := \frac{\log\left(\frac{R2}{R1}\right)}{\log\left(\frac{N2}{N1}\right)} \quad C = -0.074 \quad D := \log(R1) - C \cdot \log(N1) \quad D = 0.39
 \end{array}$$

$$\text{Rmn}_i := \left\{ \begin{array}{l}
 1 \quad \text{if } Nn_i \leq N1 \\
 10^{C \cdot \log(Nn_i) + D} \quad \text{if } N1 < Nn_i < N2 \\
 .75 \quad \text{otherwise}
 \end{array} \right.$$

$$\text{SasmeWM}_i := \left( \frac{E}{4 \cdot \sqrt{Nn_i}} \cdot \ln\left(\frac{100}{100 - A}\right) + B \right) \cdot \text{Rmn}_i$$

**Equation (4)** ASME curve adjusted for weld metal and mean stress

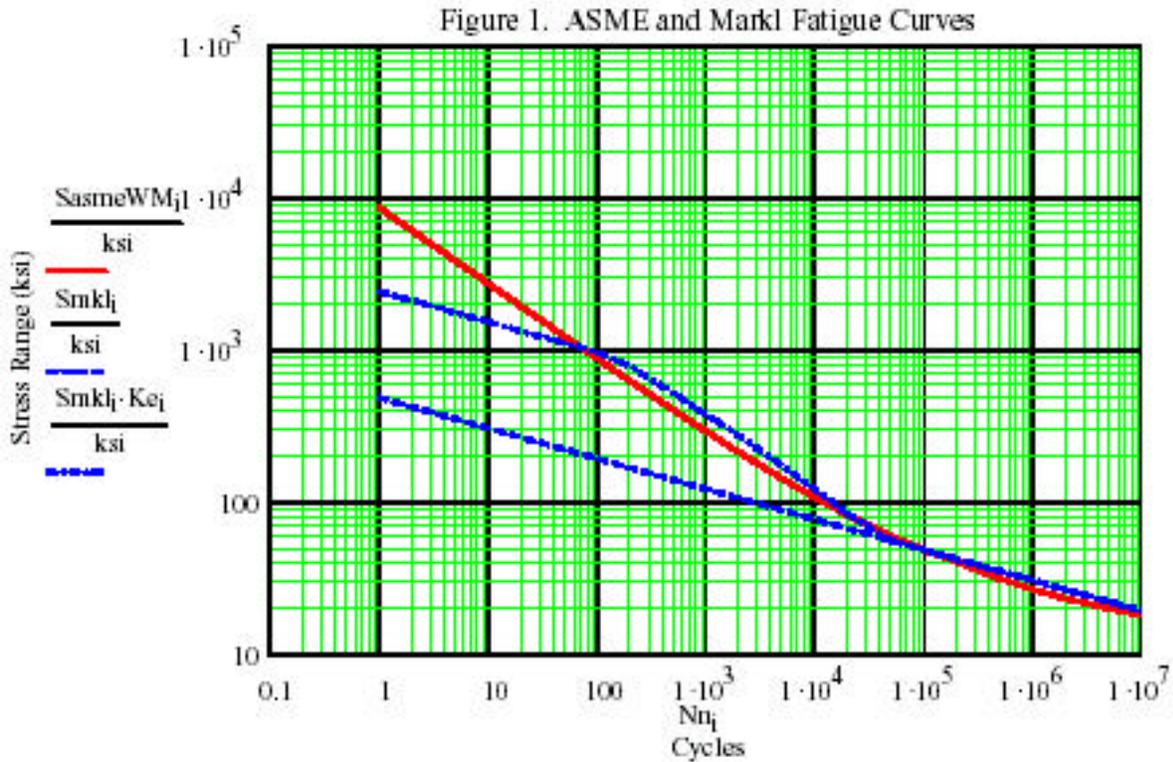
2.0 Markl Curve

$$Smkl_i := 490 \cdot (Nn_i)^{-.2} \cdot ksi$$
**Equation (5)**  
 Carbon Steel Mean Data fit, Room Temp (Range)

$m := 3.0$        $n := 0.2$        $Sm := 20 \cdot ksi$       Adjust for Plastic Range

$$Ke_i := \begin{cases} 1 & \text{if } \frac{490 \cdot (Nn_i)^{-.2} \cdot ksi}{3 \cdot Sm} \leq 1 \\ \left[ 1 + \frac{1-n}{n \cdot (m-1)} \cdot \left[ \frac{490 \cdot (Nn_i)^{-.2} \cdot ksi}{3 \cdot Sm} - 1 \right] \right] & \text{if } 1 < \frac{490 \cdot (Nn_i)^{-.2} \cdot ksi}{3 \cdot Sm} < m \\ \frac{1}{n} & \text{otherwise} \end{cases}$$
**Equation (6)**  
 Adjustment for Elastic Analysis of Plastic Deformation

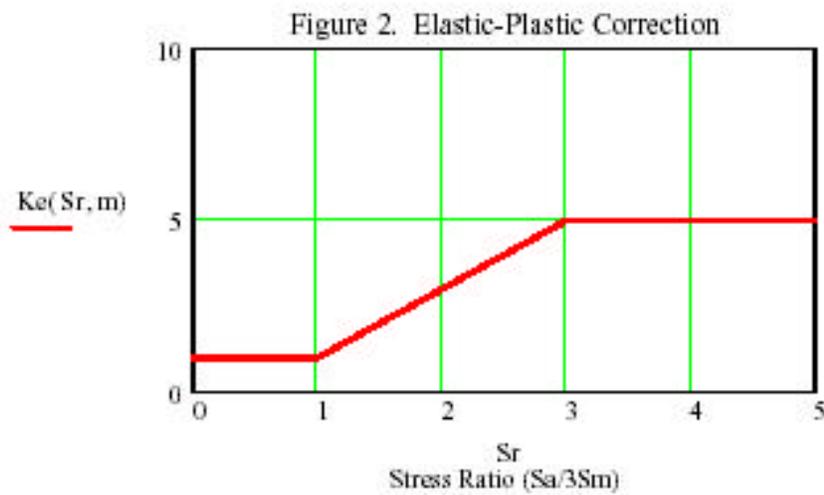
$$SmkKe_i := 490 \cdot (Nn_i)^{-.2} \cdot ksi \cdot Ke_i$$
**Equation (7)** Markl adjusted via Equation 6



### 3.0 Elastic-Plastic Adjustments

$$K_e(Sr, m) := \begin{cases} 1 & \text{if } Sr \leq 1 \\ \left[ 1 + \frac{1-n}{n \cdot (m-1)} \cdot (Sr-1) \right] & \text{if } 1 < Sr < m \\ \frac{1}{n} & \text{otherwise} \end{cases}$$

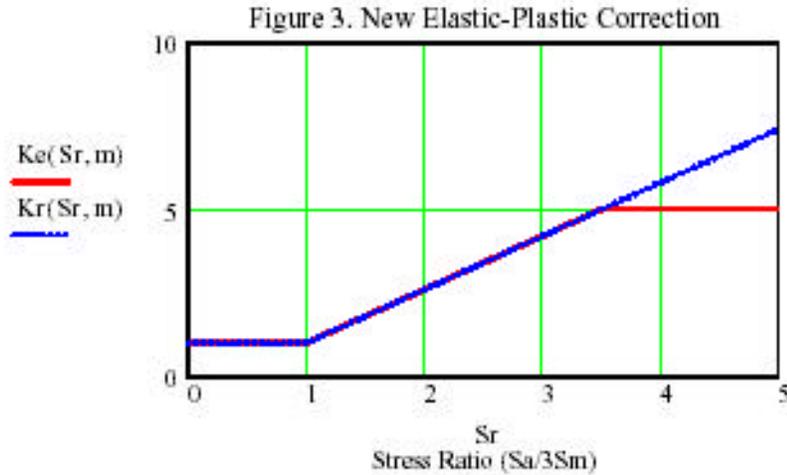
Equation (8) ASME Code  
Elastic-Plastic Correction



$$Kr(Sr,m) := \begin{cases} 1 & \text{if } Sr \leq 1 \\ \left[ 1 + \frac{1-n}{n \cdot (m-1)} \cdot (Sr-1) \right] & \text{otherwise} \end{cases}$$

**Equation (9)**  
New continuous Elastic-Plastic Correction

New Adjustment for Plastic Range  $m := 3.5$

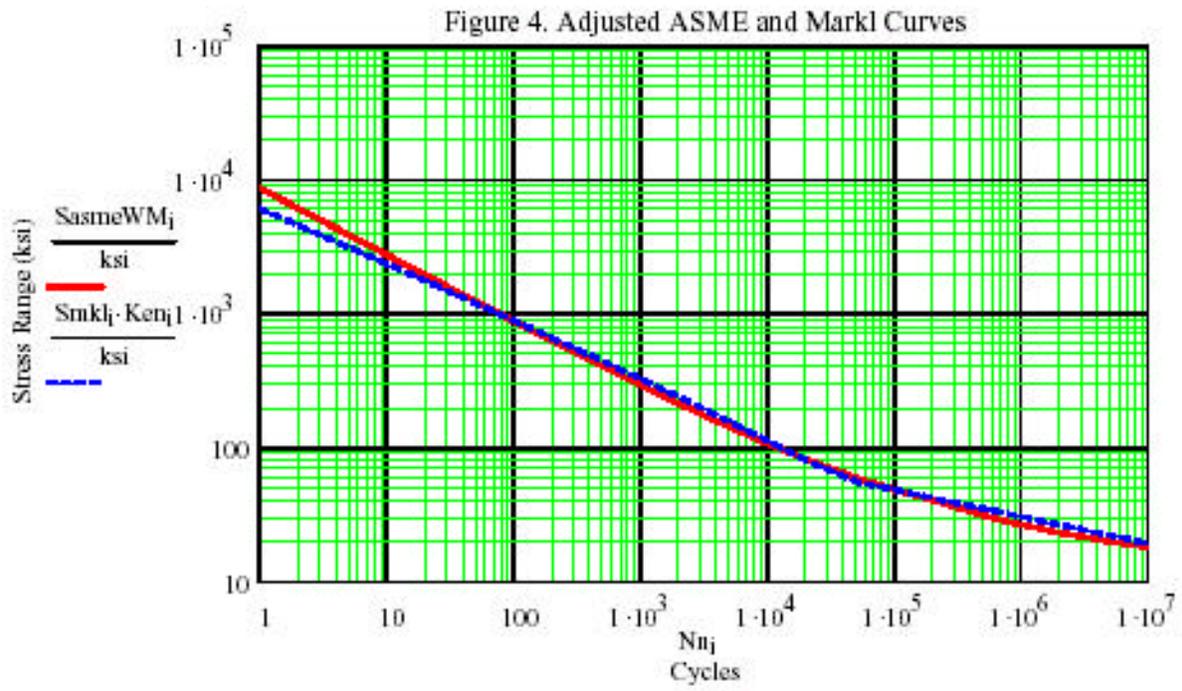


**4.0 Apply New Adjustments**

$m := 3.5$

$$Ken_i := \begin{cases} 1 & \text{if } \frac{490 \cdot (Nn_i)^{-0.2} \cdot \text{ksi}}{3 \cdot Sm} \leq 1 \\ 1 + \frac{1-n}{n \cdot (m-1)} \cdot \left[ \frac{490 \cdot (Nn_i)^{-0.2} \cdot \text{ksi}}{3 \cdot Sm} - 1 \right] & \text{otherwise} \end{cases}$$

**Equation (10)**  
New Ke factor, continued to highest strain levels (applied to Markl curve).



## 5.0 Proposed Markl Design Curve

SF := 2      Safety Factor of 2 on stress, retain temperature adjustment

$$\frac{490}{SF} \cdot (8.826 \cdot 10^6)^{-.2} \cdot \text{ksi} = 10 \text{ ksi}$$

\* Normalize by:  $\frac{S_c + S_h}{2 \cdot S_c}$

$$S_{mkld_i} := \begin{cases} 10 \cdot \text{ksi} & \text{if } \frac{490}{SF} \cdot (N_{n_i})^{-.2} \cdot \text{ksi} < 10 \cdot \text{ksi} \\ \frac{490}{SF} \cdot (N_{n_i})^{-.2} \cdot \text{ksi} & \text{otherwise} \end{cases}$$

**Equation (11)**  
Proposed Markl based design  
fatigue curve

Adjustment for Significant Flaws ( $2c = 2$  in,  $a/t = 1/4$ )

$$R1 := 1 \quad N1 := 2 \cdot 10^1 \quad R2 := .58 \quad N2 := 8.826 \cdot 10^6$$

$$C := \frac{\log\left(\frac{R2}{R1}\right)}{\log\left(\frac{N2}{N1}\right)} \quad C = -0.042 \quad D := \log(R1) - C \cdot \log(N1) \quad D = 0.055$$

$$A_{f_i} := \begin{cases} 1 & \text{if } N_{n_i} \leq N1 \\ 10^{C \cdot \log(N_{n_i}) + D} & \text{if } N1 < N_{n_i} < N2 \\ R2 & \text{otherwise} \end{cases}$$

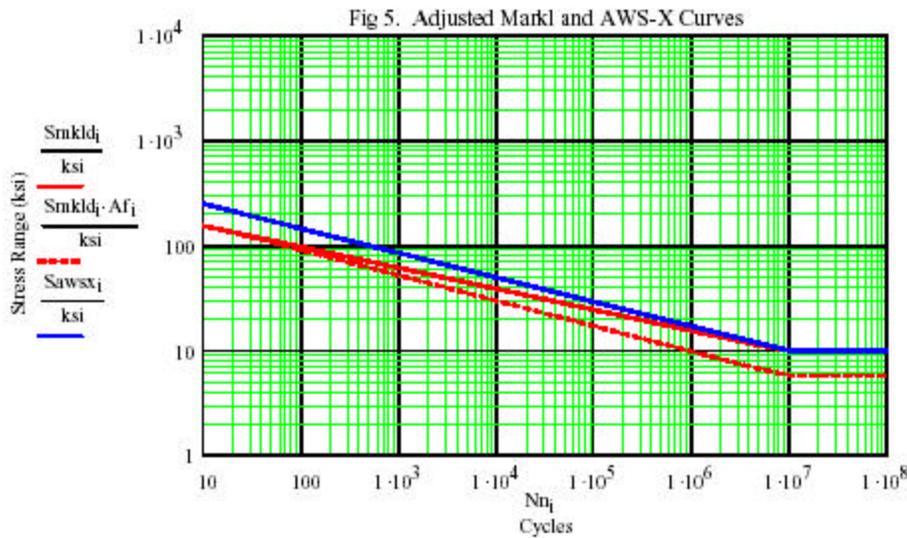
**Equation (12)**  
Example fracture adjustment

AWS X-Curve for Comparison

$$S1 := 50\text{-ksi} \quad N1 := 10^4 \quad S2 := 10\text{-ksi} \quad N2 := 10^7$$

$$A := \frac{\log\left(\frac{S2}{S1}\right)}{\log\left(\frac{N2}{N1}\right)} \quad A = -0.233 \quad B := \log\left(\frac{S1}{\text{ksi}}\right) - A \cdot \log(N1) \quad B = 2.631$$

$$Saw_{sx_i} := \begin{cases} 10\text{-ksi} & \text{if } Nn_i > N2 \\ \left(10^{A \cdot \log(Nn_i) + B}\right) \cdot \text{ksi} & \text{otherwise} \end{cases} \quad \begin{matrix} \text{Equation (13)} \\ \text{AWS X design fatigue curve} \end{matrix}$$



## **APPENDIX B**

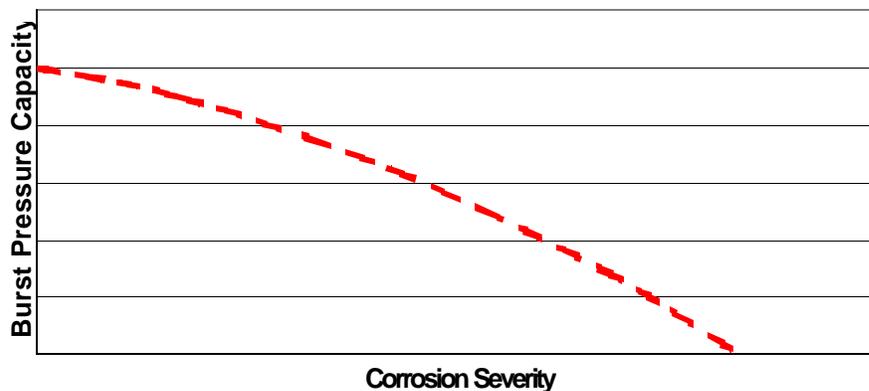
### **DEVELOPMENT OF DEMAND-CAPACITY FRAMEWORK BY SSD, INC.**

*This page intentionally left blank*

### B.1 Scope Statement

Evaluate the phenomenon of corrosion in rippled, wrinkled or buckled sections of pipelines and develop a framework of rational, quantitative criteria for evaluating such wrinkles in terms of those that can continue to remain in service and those that must be removed from service. This evaluation framework will aim to consider both pressure integrity limits and fatigue damage limits for pipelines. Determine the appropriate method to verify the proposed framework (i.e., correlation with existing pipe burst and fatigue test databases, computer modeling, additional physical testing, or a combination of these). It is proposed to start with a calculation method that is similar to well accepted simple corrosion evaluation procedures such as ASME B31G or Modified B31G and to extend this procedure to consider fatigue damage in addition to the burst pressure limit state. Ideally, the resulting calculation framework will accept measures of the corrosion geometry, various wrinkle geometry parameters, and the pressure and temperature differential loads as input and will evaluate both the burst pressure and fatigue failure limit states.

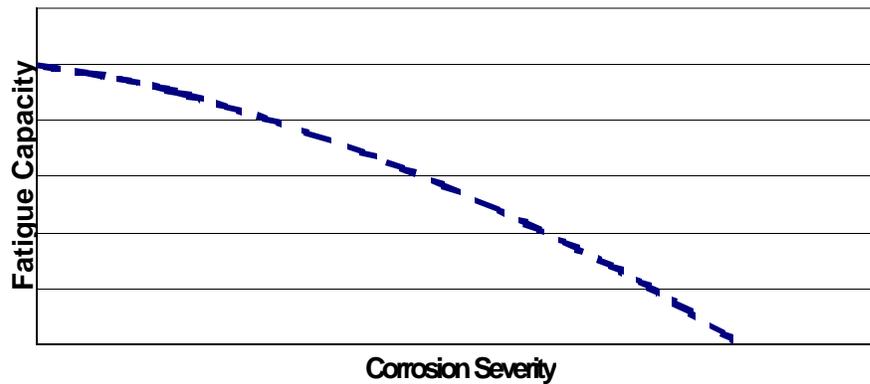
The components of the framework are illustrated schematically in Figure B.1, Figure B.2 and Figure B.3. A two-dimensional illustration of how the burst pressure capacity decreases with increasing corrosion severity is shown in Figure B.1. This aspect of the framework would be based on the existing burst test database for cylindrical, corroded pipe specimens (hundreds of tests) in the absence of bending moment loads and wrinkles. The principle illustrated in Figure B.1 can be rationally extended to consider increasing levels of bending moment and axial force (i.e., longitudinal stresses) including representative wrinkle geometries based on elastic finite element analyses of pipe sections which include representative idealized corrosion “patches” (based on stress concentration factor (SCF) analyses).



**Figure B.1 Pressure-Corrosion Space**

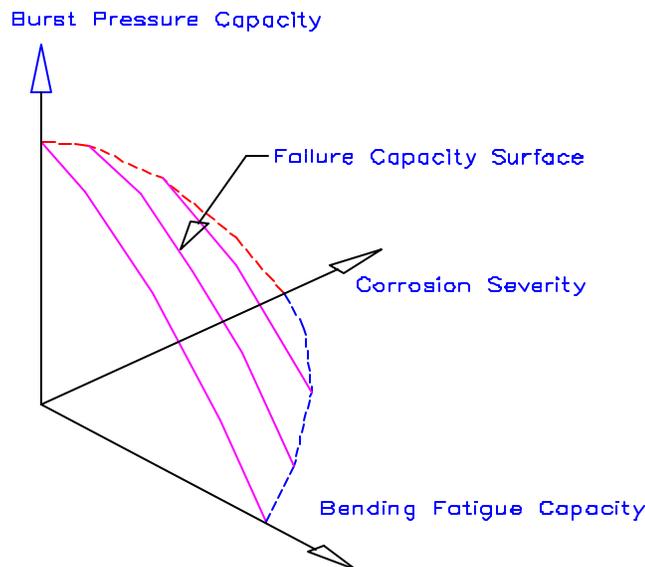
The manner in which fatigue capacity of pipe would tend to decrease with increasing corrosion severity is illustrated in Figure B.2. At zero levels of corrosion severity, this aspect of the framework could be related to the fatigue testing of pipe components (e.g., the Markl fatigue (stress versus number of cycles or  $S-N$ ) relationship and the ASME Section VIII, Division 2 fatigue relationship which are based on hundreds of fatigue tests). These  $S-N$  relationships can be applied to the evaluation of wrinkled pipe sections by relating an elastically computed SCF to a fatigue based stress-intensification factor (SIF) (i.e., the B31  $i$ -factor). The principle illustrated in Figure B.2 can be rationally extended into increasing levels of internal pressure

and corrosion severity based on elastic finite element analyses of piping components including representative idealized corrosion “patches” (again based on SCF analyses).



**Figure B.2 Fatigue-Corrosion Space**

The overall framework in the burst pressure capacity – fatigue failure capacity – corrosion severity “space” is illustrated in Figure B.3. If the wrinkle geometry is “cylindrical”, then the computed burst pressure capacity would decompose to be consistent with the burst pressure database for corroded straight pipe sections. If the corrosion geometry is “uncorroded”, then the fatigue capacity would decompose to be reasonably consistent with uncorroded pipe fatigue test data. Combinations of wrinkle and corrosion geometries and pressures and bending moment combinations that are between these bounding cases in effect would be considered with respect to the failure capacity surface illustrated in this space.



**Figure B.3 Pressure & Fatigue-Corrosion Space**

The “capacity surface” concept described above could be used for pipe integrity assessments by comparing it to different measures of location specific demand on the pipe (e.g., maximum pressure demand, cyclic pressure demand, and the cyclic stress demands due to temperature differential cycling). The pressure

demand measures can be established based on the pipeline design basis and representative pressure history samples from previous years of operation. The global temperature differential demand measures would best be established based on charts that relate temperature differential, bend angle, soil cover depth and soil type, to maximum nominal stress range which would be scaled by a SCF associated with the corroded wrinkle under investigation.

The deliverable from this task will be an outline of a formal procedure for performing integrity assessments of corroded, wrinkled sections of pipe. This will include (a) formulas that can be used to develop the capacities for the bounding cases (e.g., modified B31G for evaluating the burst pressure capacity of unwrinkled, corroded pipe, a *S-N* capacity curve for evaluation of uncorroded, wrinkled pipe), (b) an introduction to the use of stress concentration factors for wrinkled pipe, corroded pipe, and wrinkled and corroded pipe sections, (c) guidance for developing pipe demand measures based on pressure and temperature differential, and (d) an example of how the procedure can be applied.

## ***B.2 Capacity Evaluation for Bounding Cases***

The following sections describe the procedures available for evaluating the burst pressure and maximum allowable operating pressure (MAOP) of corroded sections of pipeline (i.e., the pressure capacity), and relationships that are used to evaluate pipe and piping components for fatigue damage (i.e., the fatigue capacity).

### ***B.2.1 Corrosion***

The pipeline industry has long recognized that some sections of high-pressure pipelines may experience corrosion. Based on industry experience, experimental evaluations and theoretical considerations, it is known that some amount of metal loss due to corrosion can be tolerated without impairing the ability of pipelines to operate safely. Methods for evaluating safe operating pressure levels for pipes affected by corrosion have received wide attention within the pipeline industry to the point that well-accepted procedures have been directly implemented into the ASME B31.4 and B31.8 pipeline codes (e.g., ASME B31G), and more importantly, directly referenced in Title 49 of the Code of Federal Regulations Parts 192 and 195 (49 CFR 192 and 195). Since the development of B31G, the evaluation methods have continued to evolve in efforts to remove excess conservatism that results in unnecessary pipeline repairs. Kiefner and Vieth provide an excellent discussion of the basis of the B31G method including its assumptions and limitations (Kiefner and Vieth 1989). This document also provides a useful introduction to the modified B31G criterion including the refinements to the flow stress and the Folias factor, and the 0.85-d·L and effective area representations of metal loss used in the industry accepted computer program for determining the remaining strength of corroded pipe, RSTRENG.

Several methods are available for the evaluation of the burst pressure or the maximum allowable operating pressure of corroded pipelines including B31G, modified B31G, RSTRENG, KAPA, API 579, KOGAS, NG-18 Log Secant, etc. The report “*ASME B31G: Determining the Remaining Strength of Corroded Pipelines*” provides a good overview of the methods for determining remaining strength of corroded pipelines currently in use by the pipeline industry (ASME 2003). The basic formula used in most of these methods is of the following form:

$$S = S^* \cdot \left[ \frac{1 - \frac{A}{A_o}}{1 - \left( \frac{A}{A_o} \right) \cdot M^{-1}} \right]$$

where:

$A$  is the area of the defect in the longitudinal plane through the wall thickness

$A_o$  is  $L \cdot t$

$S$  is the hoop stress level at failure

$S^*$  is the flow stress of the pipe material

$M$  is Folias' original bulging factor for a through-wall axial flaw, a function of  $L$ ,  $D$  and  $t$

where:

$L$  is the axial extent of the defect

$t$  is the nominal wall thickness of the pipe

$D$  is the diameter of the pipe

In terms of the pipe's maximum allowable operating pressure, this expression is often presented in the following form:

$$MAOP = \left( \frac{2 \cdot t \cdot F}{D} \right) \cdot S^* \cdot \left[ \frac{1 - \frac{A}{A_o}}{1 - \left( \frac{A}{A_o} \right) \cdot M^{-1}} \right] \leq \left( \frac{2 \cdot F \cdot (t - CA)}{D} \right) \cdot SMYS$$

where:

$F$  is the design factor (e.g. 0.72),

$SMYS$  is the specified minimum yield stress of the pipe material, and

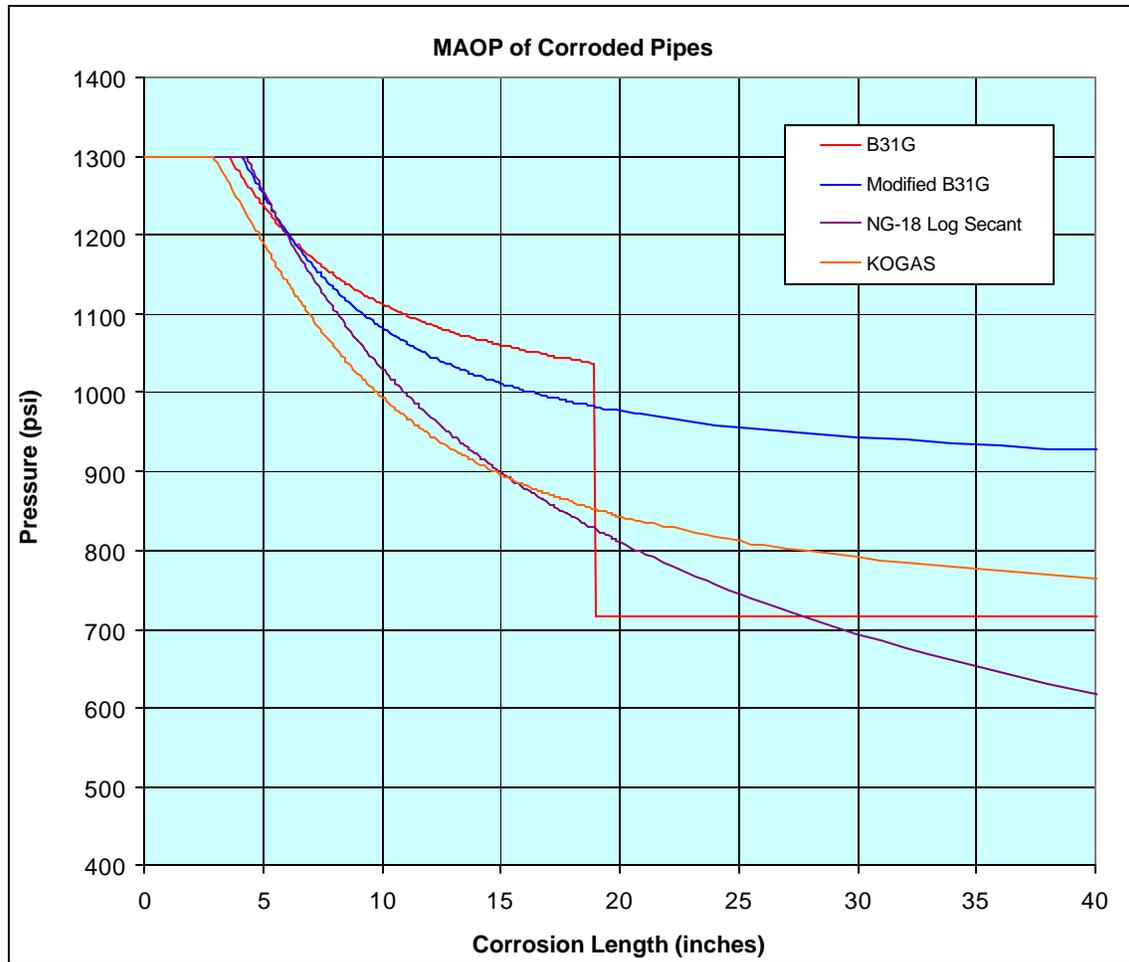
$CA$  is the corrosion allowance.

The key terms in these calculations for several of the methods mentioned above are summarized in Table B.1

**Table B-1 Comparison of Parameters Used in Different MAOP and Burst Pressure Calculation Methods**

Method	S*	A/A <sub>0</sub>	Folias Factor, M
ASME B31G	1.1·SMYS	$\frac{2}{3} \cdot a/t$ for $z = 20$ $a/t$ for $z > 20$	$\sqrt{1 + 0.8 \cdot z}$ for $z \leq 20$ $\infty$ for $z > 20$
Modified B31G	SMYS + 10 ksi	$0.85 \cdot a/t$	$\sqrt{1 + 0.6275 \cdot z - 0.00375 \cdot z^2}$ for $z \leq 50$ $3.3 + 0.032 \cdot z$ for $z > 50$
DNV Level 1	SUTS	$a/t$	$\sqrt{1 + 0.31 \cdot z}$
KOGAS	0.9·UTS	$a/t$	$\sqrt{1 + 0.31 \cdot z}$
NOTE: $z = L^2/(D \cdot t)$ SUTS is the Specified Ultimate Tensile Strength of the pipe material. UTS is the Ultimate Tensile Strength of the pipe material			

In each of these methods, the calculation is based on a characterization of the corrosion defect based on a depth “*a*” and a length “*L*”. The circumferential extent of the corrosion is not included in the formulas. A comparison of the MAOP computed using several methods for a 36-inch diameter, 0.5-inch thick, X-65 pipe with a 50% wall loss (i.e.,  $a/t=0.5$ ) for corrosion defect lengths ranging from 0 to 40 inches is presented in Figure B.4. Note that: (a) the B31G method exhibits an undesirable discontinuity at around 19 inches (i.e., at  $z = 20$ ), and (b) the modified B31G method is the least conservative of the continuous methods. The authors believe that the modified B31G method with the  $0.85 \cdot d \cdot L$  metal loss area is the most appropriate hand calculation method without resorting to the more complicated corrosion grid processing used for RSTRENG (or KAPA) computations.



**Figure B.4 Comparison of Remaining Strength Calculation Methods**

A corrosion defect is considered acceptable when the computed failure stress is equal to or greater than the hoop stress at the MAOP multiplied by a suitable Safety Factor. The minimum recommended Safety Factor is equal to the ratio of the minimum hydrostatic test pressure required for the given type of pipeline construction to the MAOP, but no less than 1.25 in general. Greater factors of safety may be appropriate in some cases, for example in areas of greater risk to the public or the environment. Lesser factors of safety may be justified in some circumstances, for example for short time periods or in remote locations. In establishing the Safety Factor for a given pipeline segment, the pipeline operator should give consideration to the accuracy of corrosion measurements (particularly if the corrosion is internal or is indicated by in-line inspection, and has not been verified physically), the characteristics of the pipe, etc.

### B.2.2 Fatigue

The calculation of fatigue damage is an inexact science and there is always a significant scatter in experimental fatigue data. For the purposes of new design, it is usual to make conservative assumptions in order to ensure that if the design satisfies the design criteria, then the probability of fatigue failure is extremely small. This is done by using *design S-N* curves that ensure a very low probability that fatigue failure will occur. This is typically accomplished by selecting a *design S-N* curve that provides a near lower

bound envelope to the experimental ( $S-N$ ) data points. Some fatigue design codes (e.g., British Standard BS 7608) provide  $S-N$  relationships in terms of the mean and standard deviations of their basis data. A *design*  $S-N$  curve is typically selected based on the mean minus two standard deviations. Assuming a normal distribution, the calculation of a usage factor (i.e., cumulative damage ratio) of 1.0 using a *design*  $S-N$  relationship based on the mean minus two standard deviations corresponds to a 2.3% nominal probability of failure. On the other hand, calculation of a usage factor of 1.0 using the *mean*  $S-N$  relationship corresponds to a 50% nominal probability of failure (i.e., half of the  $S-N$  data points would lie above the mean  $S-N$  relationship and half of the  $S-N$  data points would lie below the mean  $S-N$  relationship). This means that analyses attempting to predict an actual fatigue failure should utilize the mean  $S-N$  relationship. This illustrates how the use of the fatigue data statistics can provide a framework for quantifying the factor of safety associated with a given fatigue assessment.

For the purposes of this work, a fatigue capacity relationship was developed based on three different sources of fatigue data namely, (a) the Markl fatigue data, (b) the ASME Boiler and Pressure Vessel Code Section VIII, Division 2 design fatigue data, and (c) the American Welding Society X curve.

A. R. C. Markl (and his colleagues) performed an extensive series of fatigue tests during the early 1950s (Markl 1952 and 1955). The “Markl tests” form the basis for the fatigue design rules in today’s ASME B31 piping codes. Markl generated  $S-N$  curves over a range of  $2 \times 10^2$  to  $1 \times 10^6$  cycles for the round bar specimens, plain straight pipe, straight pipe containing a girth butt weld, short- and long-radius elbows, fabricated miter bends, forged welding tees, un-reinforced branches, pad-reinforced branches, welding neck flanges, slip-on flanges, and socket-welded flanges. Markl found that within a scatter band, the  $S-N$  curves of each type of component lay essentially parallel to the round-bar  $S-N$  curve. When plotted on log-log paper, the entire body of data was essentially parallel and could be reasonably approximated by the formula:

$$i \cdot S = 2 \cdot C \cdot N^{-0.2}$$

where:

$i$  is the SIF (the stress intensification factor associated with a given piping component),

$S$  is the nominal stress range,

$N$  is the number of stress reversals to failure, and

$C$  is a material constant (a value of  $2 \cdot C = 490$  ksi is used to represent the mean of the carbon steel fatigue test data).

The fact that the  $S-N$  curves for different components are essentially parallel allows for the use of a single curve scaled by the  $i$  factor to compute fatigue damage for a range of different piping components.

There is substantial pipeline and piping industry experience with the use and application of the ASME Boiler and Pressure Vessel design fatigue curve. As described in *Criteria of the ASME Boiler and Pressure Vessel Code for Design Analysis in Sections III and VIII, Division 2*, this is a “*design*” fatigue curve derived from base metal fatigue tests which has a significant factor of safety with respect to its basis data. Comparison of the design curve and its basis data indicates that the factor of safety for the *design* curve is a

factor of 2 on stress (i.e.,  $S/2$ ) or 20 on life (i.e.,  $N/20$ ) relative to the basis data points, whichever is more conservative at each point. The best fit (mean) S-N curve for carbon steel base metal is based on the following relationship:

$$S = \frac{E}{4\sqrt{N}} \ln \frac{100}{100 - A} + B$$

where:

$$A = 68.5\%,$$

$$B = 21,645 \text{ psi, and}$$

$$E = 30,000 \text{ ksi.}$$

The AWS X design curve for full penetration welds has been considered because it provides a clearly defined endurance limit in the high cycle region (a stress range  $S=10$  ksi at  $10^7$  cycles).

Berkeley Engineering and Research, Inc. (BEAR) performed a detailed comparison of the above mentioned fatigue (S-N) curves making the necessary adjustments for weld metal versus base metal, etc. (See Appendix A for more detail.) Based on this comparison, BEAR developed a recommended fatigue S-N relationship for the purposes of assessing corroded wrinkles. The recommended *mean* fatigue S-N relationship is summarized as follows:

$$i \cdot S = 490 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 20 \quad \text{for } N > 8.8 \times 10^6$$

where:

$S$  is the stress range (in ksi)

$N$  is the number of stress reversals to failure, and

$i$  is the fatigue effective SIF (note:  $i=1.0$  for a girth butt weld and  $i=0.5$  for pipe base metal).

Applying a factor of safety of 2.0 on stress range leads to the following *design* S-N relationship:

$$i \cdot S = 245 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 10 \quad \text{for } N > 8.8 \times 10^6$$

Note that in the above fatigue relationships, the term on left-hand side of the equation represents the fatigue demand while the term on the right-hand side represents the fatigue capacity. The fatigue demand is compared to the fatigue capacity based on the cumulative fatigue damage (usage ratio) that is established by application of Miner's linear damage theory (Miner 1945). The fatigue rules used in the ASME B31 piping codes also provide a basis for accumulating damage due to cycles with different stress ranges. Accepted algorithms such as the "rainflow" counting method can be used as a basis for accurately counting equivalent stress cycles based on experimental time history measurements (e.g., piping temperature or pressure time histories).

### B.2.2.1 *Modification for Stress-Controlled Loading*

All of the discussion on fatigue presented in the previous section is related to fatigue testing wherein the material is cycled under displacement-controlled (secondary) loading conditions, such as those that would result from thermal expansion and contraction or other cyclic secondary loads. As discussed in the report, “*Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends* (Rosenfeld et. al, 2002), comparatively few Markl-type SIF tests have been reported in which the loading environment was load controlled (i.e., primary or stress controlled). The following definitions may be helpful to make a distinction between displacement-controlled (secondary) and load-controlled (primary) stress conditions:

**Primary Stresses** are developed by imposed loadings and are necessary to satisfy equilibrium of internal and external forces and moments. Primary loads cannot be relieved by local yielding or distortion. The basic characteristic of a primary stress is that it is not self-limiting, which means that the stress will be present as long as the load is applied. Primary stresses which considerably exceed the yield strength will result in failure or, at least, in gross distortion. Stresses caused by the following loads are considered primary stresses: internal pressure, external pressure and overburden, dead and live loads. In ASME B31 design terminology, sustained loads produce primary stresses.

**Secondary Stresses** are developed by the self-constraint of the structure or by the constraint of adjacent material and are developed to satisfy strain/displacement compatibility. Secondary stresses can be relieved by local yielding and distortion. The basic characteristic of a secondary stress is that it is self-limiting. In reality, a secondary stress is a strain that is converted to a stress. Stresses caused by the following loads are considered secondary stresses: temperature differential, differential settlement, and fault displacement. In ASME B31 design terminology, secondary stresses are referred to as displacement limited stresses.

Fatigue curves presented in “*Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*” (ASME) indicate that the stress range for stress controlled tests is less than the stress range for displacement controlled tests. Rodabaugh (Rodabaugh, 1988) discussed work by Lane and Rose (Lane and Rose, 1961) on pulsating pressure tests of pressure vessel nozzles (pressure loading creates primary stress and hence is load controlled). Manipulation of the Lane and Rose fatigue relationship into a form analogous to Markl’s equation leads to an effective material constant for stress-controlled tests of  $C' = 165.4$  ksi. This value agreed very well with an effective  $C' = 163.3$  derived from API rippled pipe pressure cycling test results of rippled pipe (Rosenfeld et. al 2002). Therefore, for the purposes of developing this analysis framework, for fatigue analysis of stress-controlled conditions (i.e., pressure loads), it is recommended that a modified material constant  $C'$  equal to essentially 2/3 of the displacement-controlled material constant  $C$  ( $C' = 2/3 \cdot C = 163.3$ ) be utilized.

### B.3 *Stress Concentration Factors*

As discussed previously, the results of finite element analysis (FEA) of piping are frequently expressed in terms of a scalar factor that characterizes the stress concentration effects due to the geometry of the piping component or the anomaly being evaluated by the FEA model. This factor is the *SCF*, which is typically taken as the ratio of the maximum local stress or the maximum stress intensity to the nominal stress. The *SCFs* discussed in this report are based on elastic FEA results for a stub of pipe containing a mesh of

rippled or wrinkled profile. The results from finite element analysis of rippled pipe stubs subjected to bending moment and pressure loads presented in “*Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends*” (Rosenfeld et. al 2002) serves as a useful starting point for this work.

### B.3.1 Wrinkled Pipe

The wrinkle geometry parameters varied in the analyses reported in the paper noted above are:

1. Parameter  $h/D$ : the ratio of the wrinkle depth, “ $h$ ”, to the pipe diameter, “ $D$ ”.  $h/D$  ratios of 1%, 2%, 4% and 6% were considered.
2. Parameter  $\lambda/h$ : the ratio of the wrinkle trough-to-trough wavelength, “ $\lambda$ ”, to the wrinkle depth, “ $h$ ”.  $\lambda/h$  ratios of 9, 12 and 15 were considered.
3. Parameter  $D/t$ : the ratio of the pipe diameter, “ $D$ ”, to the wall thickness, “ $t$ ”. Pipe  $D/t$  ratios of 60, 90 and 120 were considered.
4. Parameter  $c/pD$ : the ratio of the circumferential extent of the wrinkle, “ $c$ ”, to the total circumference, “ $pD$ ”. This parameter was taken as a linear function of  $h/t$ .

Additional wrinkle geometry parameters include whether or not the wrinkle deformation is dominantly inward or outward and the number of lobes in the wrinkle wave form. The parametric studies were carried out on a pipe model with a diameter of 36 inches. A total of 36 different ripple geometries are required to evaluate the  $h/D$ ,  $\lambda/h$  and  $D/t$  ratios listed above (4  $h/D$  ratios x 3  $\lambda/h$  ratios x 3  $D/t$  ratios = 36 cases). For each wrinkle geometry, a finite element mesh was generated and each model was analyzed for two load cases; a pressure vessel loading corresponding to a pressure of 1,000 psi and a uniform bending moment of 5,000 kip-inches. A total of 72 cases were considered (36 for internal pressure and 36 for bending moment). Based on these cases, regressions of the following form were developed to estimate the SCFs at the peak of the wrinkle for pressure and bending moment loads:

$$SCF_P = \mathbf{a}_0 \cdot \left(\frac{D}{t}\right)^{a_1} \cdot \left(\frac{h}{D}\right)^{a_2} \cdot \left(\frac{h}{t}\right)^{a_3} \cdot \left(\frac{\lambda}{h}\right)^{a_4} \cdot \left(\frac{c}{p \cdot D}\right)^{a_5}$$

$$SCF_M = \mathbf{b}_0 \cdot \left(\frac{D}{t}\right)^{b_1} \cdot \left(\frac{h}{D}\right)^{b_2} \cdot \left(\frac{h}{t}\right)^{b_3} \cdot \left(\frac{\lambda}{h}\right)^{b_4} \cdot \left(\frac{c}{p \cdot D}\right)^{b_5}$$

In order to evaluate a wrinkled section of pipe in the absence of corrosion, the geometric parameters  $D/t$ ,  $h/D$ ,  $h/t$ ,  $\lambda/h$  and  $c/pD$  would be established based on physical or smart pig measurements. Using these wrinkle geometry parameters, the SCFs for internal pressure and bending moment could then be estimated using the above regression formulas. As a more direct method, the analyst could develop a case-specific finite element model of a stub of pipe containing a mesh of the wrinkle with the measured geometry and directly determine case specific SCFs based on elastic FEA.

### B.3.2 Corroded Pipe

The metal loss due to significant corrosion obviously has the potential to cause stresses to concentrate locally in and near the corroded region. The increased local stress plays a role in reducing the pressure capacity of the pipe. Based on industry experience and discussions with industry experts, pipeline failures

due to fatigue in corroded ripples, wrinkles or buckles were not identified. Nevertheless, if a detailed finite element analysis (or strain gauged experiment) is performed on a corroded section of pipe, the local stresses in or near the corroded region would be larger than the nominal stress and hence an *SCF* could be established. Based on the research, most FEA of corrosion defects characterize the corrosion as a uniformly thinned “patch” region. In this case, the local stresses in the patch may be roughly approximated by calculating the nominal stress (i.e.,  $PD/2t$  or  $M/Z$ ) using the “thinned” cross section properties.

### B.3.3 Combined Wrinkling and Corrosion

As discussed in the previous two sections, it is possible to develop finite element models based on the geometry of wrinkled sections of pipe as well as corroded sections of pipe (e.g., where the corrosion is usually idealized as a uniformly thinned patch). It follows then that these types of models could be combined to simulate a wrinkled and corroded section of pipe. To investigate this, preliminary pilot analyses were undertaken on several of the FEA models of wrinkles described in Section B.3.1. The wrinkled pipe models were extended to consider idealized uniform depth corrosion patches. In all cases, the corrosion patch was assumed to be axi-symmetric and centered on the center of the wrinkle profile. Corrosion patches with up to 50% metal loss were evaluated with lengths of 6, 18 and 36 inches (i.e.,  $D/6$ ,  $D/2$  and  $D$ ). Based on these analyses, it was observed that the wrinkle plus corrosion *SCFs* were increased relative to the wrinkle *SCFs* without corrosion, as expected. In general the *SCFs* were found to increase with increasing corrosion length and depth. Based on these observations, it should be possible, given a much more comprehensive matrix of analysis cases, to extend the previous *SCF* regressions to include a metal loss parameter (e.g.,  $A/A_o$ ) as well as a parameter defining the circumferential and longitudinal location of the corrosion with respect to the peak of the wrinkle such that the *SCF* is increased when the corrosion is significant and the *SCF* decomposes to the un-corroded value when the metal loss reduces to zero. The form of a possible modified regression is as follows:

$$SCF_p = \mathbf{a}_0 \cdot \left(\frac{D}{t}\right)^{a_1} \cdot \left(\frac{h}{D}\right)^{a_2} \cdot \left(\frac{h}{t}\right)^{a_3} \cdot \left(\frac{I}{h}\right)^{a_4} \cdot \left(\frac{c}{p \cdot D}\right)^{a_5} \cdot \left(1 - \frac{A}{A_o}\right)^{g_1}$$

$$SCF_M = \mathbf{b}_0 \cdot \left(\frac{D}{t}\right)^{b_1} \cdot \left(\frac{h}{D}\right)^{b_2} \cdot \left(\frac{h}{t}\right)^{b_3} \cdot \left(\frac{I}{h}\right)^{b_4} \cdot \left(\frac{c}{p \cdot D}\right)^{b_5} \cdot \left(1 - \frac{A}{A_o}\right)^{g_2}$$

In order to evaluate a wrinkled and corroded section of pipe, the geometric parameters  $D/t$ ,  $h/D$ ,  $h/t$ ,  $?/h$  and  $c/pD$  and the geometry of the corrosion (e.g.,  $A/A_o$ ) would be established based on smart pig or physical measurements. Using these measured parameters, the *SCFs* for internal pressure and bending moment could then be estimated using the above modified regression formulas. However, it should be noted that the general consideration of corrosion in addition to wrinkles adds a significant number of geometric parameters to the FEA analysis matrix (i.e., in addition to the numerous parameters required to characterize the wrinkles such as  $h/D$ ,  $h/t$ ,  $?/h$  and  $c/pD$ ). For example, even if the corrosion is characterized as a simple rectangular patch, the additional geometric parameters include the patch length, width and depth, the transition length between the full and corroded thickness, as well as the longitudinal and circumferential locations of the center of the patch relative to the peak of the wrinkle. Based on the pilot analyses undertaken as part of this work, it is not considered practicable to develop a general regression since it

would take literally thousands of parametric FEA analysis cases to bound the possible ranges of the pertinent combined wrinkle and corrosion geometries. As an alternative and much more practicable approach, the analyst could develop a case-specific finite element model of a stub of pipe containing a mesh of the measured corroded wrinkle geometry of interest and use this model to determine case specific *SCFs* based on elastic FEA.

#### **B.4 Relationships Between SIFs (*i*-factors) and SCFs**

One of the primary objectives of this work is to develop a framework for evaluating the potential for fatigue damage and failure in corroded wrinkles subjected to pressure and bending loads. A key element of this assessment is the proper evaluation of the left-hand side demand term (*i*·*S*) described in Section B.2.2. The *i* term is the fatigue effective stress intensification factor. Markl formally defined the stress intensification factor *i* as “...the ratio of the bending moment required to produce fatigue failure in a straight girth butt welded pipe of nominal dimensions, to that producing fatigue failure in the same number of cycles in the component under consideration...” It should be clearly stated and understood that Markl’s, and thus the B31 codes, definition for the “butt weld in straight pipe” stress intensification factor is  $i = 1.0$ . For pipe base metal,  $i = 0.5$ . The B31 codes provide tables and formulas for the *i*-factors for a range of piping components. The “*S*” term is well understood – it represents the nominal stress range in the pipe. For bending moment loads, this term would normally be taken as  $S = M/Z$  where *M* is the nominal bending moment range (for the cycle under consideration) and *Z* is the pipe section modulus. For a pressure cycle, *S* is normally taken as the nominal hoop stress in the pipe wall:  $S = P \cdot D_i / (2 \cdot t)$  where *P* is the internal pressure change for the cycle under consideration, *D<sub>i</sub>* is the pipe’s inside diameter and *t* is the nominal (uncorroded) wall thickness.

As noted above, the B31 codes provide tables and formulas for the *i*-factors for a range of piping components allowing relatively straightforward application of the B31 fatigue rules for piping design. However, the effective *i*-factors for pipe anomalies such as corrosion, wrinkles or corroded wrinkles are not readily available and procedures used to evaluate the fatigue performance of these anomalies are typically undertaken in the realm of case-specific fitness for service assessments or research projects. In most instances, the pipeline anomaly will be evaluated based on the analysis of detailed shell or solid finite element model meshes that approximate the geometric features of the anomaly. The analysis typically consists of the application of a bending moment or an internal pressure to a “stub” of pipe containing the anomaly. The finite element analysis results are typically expressed in terms of an *SCF*, which is taken as the ratio of the maximum local stress or the maximum stress intensity to the nominal stress. *SCFs* can be thought of as general local response quantities since they can be developed for different types of loading (e.g., bending moment, internal pressure, axial load, torsional moment, etc.). It is very important to point out that *SCFs* (which are defined based on FEA or theory of elasticity solutions) are not equal to the “fatigue effective” SIFs (*i* factors) as used in the B31 pipeline and piping codes (which are defined based on fatigue testing). However, as detailed in “*Relationships Between Stress Intensification Factors and Stress Concentration Factors*” (SSD, Inc. and Kiefner & Associates, Inc. 2003), it is possible to develop useful relationships between theoretically developed *SCFs* and experimentally determined SIFs.

For the purposes of this work, it is assumed that the localized stress field in the vicinity of a pipe anomaly (i.e., a corrosion patch, a wrinkle or ripple or a corroded wrinkle or ripple) that results from globally applied loads (such as bending moment and internal pressure) can be reasonably represented using an *SCF* from an

elastic finite element analysis of a pipe stub model containing a characterization of the anomaly. If the anomaly does not contain a weld, then it can be shown that the fatigue effective B31  $i$ -factor to be used in the fatigue assessment for bending loads is related to the analytical  $SCF$  as follows (SSD Inc. and Kiefner & Associates, Inc. 2003):

$$i = SCF/2$$

If the anomaly were located on a weld, which is not explicitly included in the FEA model (i.e., the typical case) the B31  $i$ -factor would be related to the analytical  $SCF$  as follows (SSD, Inc. and Kiefner & Associates, Inc. 2003):

$$i = SCF K_2 / 2$$

where:

$K_2$  is the peak (highly-localized) bending stress index which results from localized discontinuities, such as welds (used for Class 1 nuclear power plant piping per ASME Code Section III – see “*Stress Indices and Stress Intensification Factors of Pressure Vessel and Piping Components*” (ASME PVP, 1981) for more details).

If the fatigue performance of the weld detail is considered to be equivalent to a full penetration butt weld without any external sharp corners, then the value of the  $K_2$  index in the above expression could be taken as 2.0.

### **B.5 Framework for Evaluation of Corroded Wrinkles**

The primary concern for corrosion in pipelines is how it will affect the pipe’s pressure capacity. As previously discussed, the pipeline industry has well-accepted procedures in place for evaluating the pressure capacity of corroded pipelines and this area continues to be the subject of extensive research.

Once a wrinkle has formed in a pipeline, the primary concern is the stability of the wrinkle (e.g., are the wrinkle deformations likely to increase due to continued settlement?). If it is unlikely that the deformations in a wrinkle or ripple will increase (i.e., the wrinkle or ripple is stable), the next concern is the potential for fatigue damage in or near the wrinkle. There are currently no universally accepted guidelines or specific criteria that can be used to limit the geometry of wrinkles or ripples in pipelines based on fatigue considerations. However, it is our understanding that the B31.8 Code Committee is presently considering an agenda item allowing for ripples with peak-to-trough heights of up to 1% of the pipe diameter based on recent research (Rosenfeld et. al 2002). It is also understood that the B31.4 Code Committee is considering an agenda item related to the acceptance of mild ripples, as well.

When wrinkles (or ripples) in pipelines are found to contain corrosion, the concerns are the same as those described above:

- a. Is the pressure integrity of the pipeline at risk?
- b. Is the wrinkle stable?
- c. Is the corroded wrinkle at risk of fatigue damage or failure?

The development of a consistent analysis framework for accessing corroded wrinkle serviceability would provide a very important step forward for the pipeline industry. Based on experience with piping and pipeline analysis, a corroded wrinkle analysis framework is presented below. The level of complexity of the serviceability analysis depends on numerous issues including how much is known about the corrosion and wrinkle/ripple geometry, the pipeline operating history, the pipe material properties, the soil conditions and other variables. A flow chart of the analysis framework is presented in Figure B.5. The overall steps of the framework are outlined as follows:

- 1. Develop Measurements of the Corrosion and the Wrinkle.** Ideally, the geometry of both the corrosion and the wrinkle can be established with reasonable accuracy based on smart pig measurements. If this is not the case, it may be necessary to excavate the pipe and physically take the necessary measurements (e.g., a grid of the corrosion and its location with respect to the peak of the wrinkle and measurements of the wrinkle height, length, circumferential extent, etc.).
- 2. Evaluate the Pressure Capacity of the Corroded Section.** Evaluate the corrosion using industry-accepted procedures (e.g., B31G, Modified B31G or RSTRENG) assuming that the pipe is cylindrical (i.e., disregarding the wrinkle). If the remaining strength is not sufficient to resist the MAOP with an adequate Safety Factor, repair the pipe (or reduce the MAOP of this segment – see appropriate CFR integrity management rules).
- 3. Check the Stability of the Wrinkle or Ripple.** Assuming that the corrosion has “passed” the remaining strength check performed under item 2, the next step is to determine if the wrinkle is stable. If the wrinkle formed as a result of pipe settlement or other sources of permanent ground displacement (e.g., earthquake fault movement, landslide, etc.), the analyst must make a determination if the pipe deformations are likely to increase from continued ground movement or other causes. If additional distortion is anticipated, a decision to repair the wrinkled section or to stabilize the pipeline in the vicinity of the wrinkle must be made, depending on the level of deformation in the wrinkle. If the wrinkle is determined to be stable and not likely to be subject to additional distortion (e.g., mild wrinkles or ripples on the intrados of a cold bend that have been in place since construction), then the wrinkle should be screened as a potential source of fatigue damage.
- 4. Evaluate the Fatigue Demands vs. the Fatigue Capacity of the Wrinkle.** Assuming that the corrosion has “passed” the remaining strength check performed in item 2, and that the wrinkle is determined to be stable as described in item 3, the next step is to evaluate the fatigue demands on the wrinkle with respect to the fatigue capacity (neglecting the presence of corrosion). The simplest approach involves the use of screening “rules of thumb” namely:
  - Australian Standard AS 2885.1 1997: Height ( $h$ ) = 5% of Peak-to-Peak Length ( $L$ ).
  - Proposed Criteria: Height ( $h$ ) =  $1.5t$  and Aspect Ratio  $L/h > 12$  (Olson et. al 1996).

- *Hazardous Liquid Pipelines:* Shallow ripples having crest-to-trough dimensions ( $h$ ) up to 0.5% of the pipe outside diameter ( $D$ ) for operation at (hoop) stress levels in excess of 47 ksi, increasing to 2% of the pipe outside diameter for operation at (hoop) stress levels at less than 20 ksi (Rosenfeld et. al 2002).
- *Gas Pipelines:* Shallow ripples having crest-to-trough dimensions ( $h$ ) up to 1% of the pipe outside diameter ( $D$ ) for operation at (hoop) stress levels in excess of 47 ksi, increasing to 2% of the pipe outside diameter for operation at (hoop) stress levels at less than 37 ksi (Rosenfeld et. al 2002).

In addition to, or as an alternative to the wrinkle screening guidelines described above, a detailed multi-step evaluation process may be undertaken. The components are described as follows:

- Develop Representative Pressure and Temperature Cycle Histograms.** Based on the operating history of the pipeline and the corresponding pipeline pressure and temperature profiles, it should be possible to develop a count of pressure and temperature differential cycles at a specific location (station) of interest over a given time interval (e.g., one year) or over the entire operating history of the pipeline. If digital time histories of the pressure and temperature are available, the cycles can be counted using numerical procedures such as the rain flow cycle counting method (this can be used to group the cycle counts in pressure or temperature “bins,” e.g., 100 pressure cycles with a stress range between 600 and 700 psi, 400 pressure cycles with a stress range between 500 and 600 psi, etc.). The key result of this step is a table or histogram that identifies the pressure range or temperature differential range and the corresponding number of cycles “n” at this range.
- Develop Buried Pipe Models.** Develop “global” pipe-soil interaction finite element models of the buried pipe section containing the wrinkled pipe section of interest. This step requires an analyst with experience in buried pipe analysis to develop and analyze a model of the pipeline containing the region of interest. The buried pipe model geometry can be developed based on as-built pipeline drawings or geometry pig survey data. The section of the model containing the wrinkle and the associated angle change is modeled as a series of short, straight pipe (or beam) elements. The ripples or wrinkles are *not* explicitly considered in this model. The state-of-practice for modeling buried pipe-soil interaction is through the use of a “beam on foundation model” where the centerline of the pipe (beam) is supported by a nonlinear Winkler foundation. Well-established procedures are available for computing elastic-perfectly plastic pipe-soil spring properties in the longitudinal, transverse horizontal, uplift and bearing directions (e.g., see American Lifelines Alliance 2001 and ASCE 1984) based on the cover depth, soil density, friction angle and cohesion. The yield strength is the key calculated pipe-soil spring parameter while the displacement required to mobilize the full strength is usually based on a simple rule of thumb. Due to the variability of soil properties, they are often specified as a range (e.g., the friction angle is between 30 to 35 degrees). It is common practice to develop both a “soft-weak” spring based on lower range strength and largest yield displacement as well as a “stiff-strong” spring based on upper range strength and smallest yield displacement. Analyses should be performed for both the “soft-weak” and “stiff-strong” soil spring assumptions in order to bound the expected pipe-soil interaction behavior.

- c) **Analyze the Buried Pipeline Models for Global/Nominal Demand Measures.** The buried pipe models described above are analyzed for pressure and temperature differential loadings based on a pre-defined pipeline design basis that specifies the MAOP and temperature differential. Depending on how much is known about the pipeline operating hydraulics and heat transfer, it may be possible to develop estimates of location-specific maximum pressures and temperature differentials (see item 4a). The results from these analyses include the displacements at the pipe nodes, the forces and deformations in the pipe-soil springs, as well as the “global” axial force ( $F$ ) and bending moment ( $M$ ) “demands” in the pipe elements and the corresponding nominal pipe stress ( $S$ ). The pipe-soil interaction analysis is advantageous since it provides estimates of the longitudinal and transverse pipe movements that can occur at bends and it can capture the pipe axial force variation between fully restrained sections (e.g., long straight runs) and partially restrained sections (e.g., bends) of the pipeline.
- d) **Analyze Representative Wrinkle Geometries for Local Demand Measures.** The purpose of this step is to develop estimates of the degree of local stress concentration associated with a given wrinkle geometry (i.e. the  $SCFs$  for pressure and moment loads). Although it may be possible to use closed-form solutions or regressions for this purpose, probably the most practicable approach is to develop a detailed, case-specific elastic shell or solid finite element models of a pipe stub containing the wrinkle geometry of interest. This step requires detailed measurements of the wrinkle geometry and an analyst with experience in FEA to develop and analyze the model of the pipe section containing a detailed mesh of the wrinkle. As discussed in Section B.3.1, the fundamental wrinkle geometry parameters are the wrinkle height, the wrinkle wavelength and the circumferential extent of the wrinkle. Although it is possible to include the effects of soil restraint in the “local” analyses, it is most practicable to neglect soil restraint in this detailed FEA model. The main advantages of using linear elastic pipe material are that the material model is relatively simple (the material stiffness is defined by the elastic modulus,  $E$ , and Poisson’s ratio,  $\nu$ ) and that the results from a given “unit” load case are scalable to any load level. The stress localization can be adequately represented by performing linear elastic, small displacements analysis of the FEA stub model for unit pressure and unit bending moment load cases. For a given wrinkle geometry, evaluate the pressure and bending moment stress concentration factor ( $SCF$ ) defined as the maximum computed local stress (or stress intensity) in or near the wrinkle to the corresponding nominal stress. In the absence of the ability to perform a case-specific FEA for a wrinkle of interest, Section B.3 provides  $SCF$  regression formulas established based on FEA over a range of ripple/wrinkle geometries for pressure and bending moment loads based on the following dimensionless geometry parameters:  $D/t$ ,  $h/D$ ,  $h/t$ ,  $\lambda/h$  and  $c/pD$ . These geometric parameters would be established based on physical or smart pig measurements for the wrinkle of interest.
- e) **Combine the Nominal and Local Stress Demand Measures.** For a given wrinkled pipe configuration of interest, use the pressure and temperature histograms to establish location-specific pressure and temperature differentials ranges. Use these ranges to establish the corresponding nominal pressure stress demands and the nominal bending stress demands computed from the pipe-soil interaction models. Scale the nominal stress demand measures ( $S$ )

by the *SCFs* computed from the local finite element analyses of the wrinkled stub models to compute the overall localized fatigue stress demand measures (*i*-*S*) that are associated with a pressure or temperature differential cycle at or near the wrinkle(s) of interest. Note that, as discussed in Section B.4, the analytically established *SCFs* must be appropriately adjusted to develop the equivalent fatigue effective *SIFs* (*i* factors).

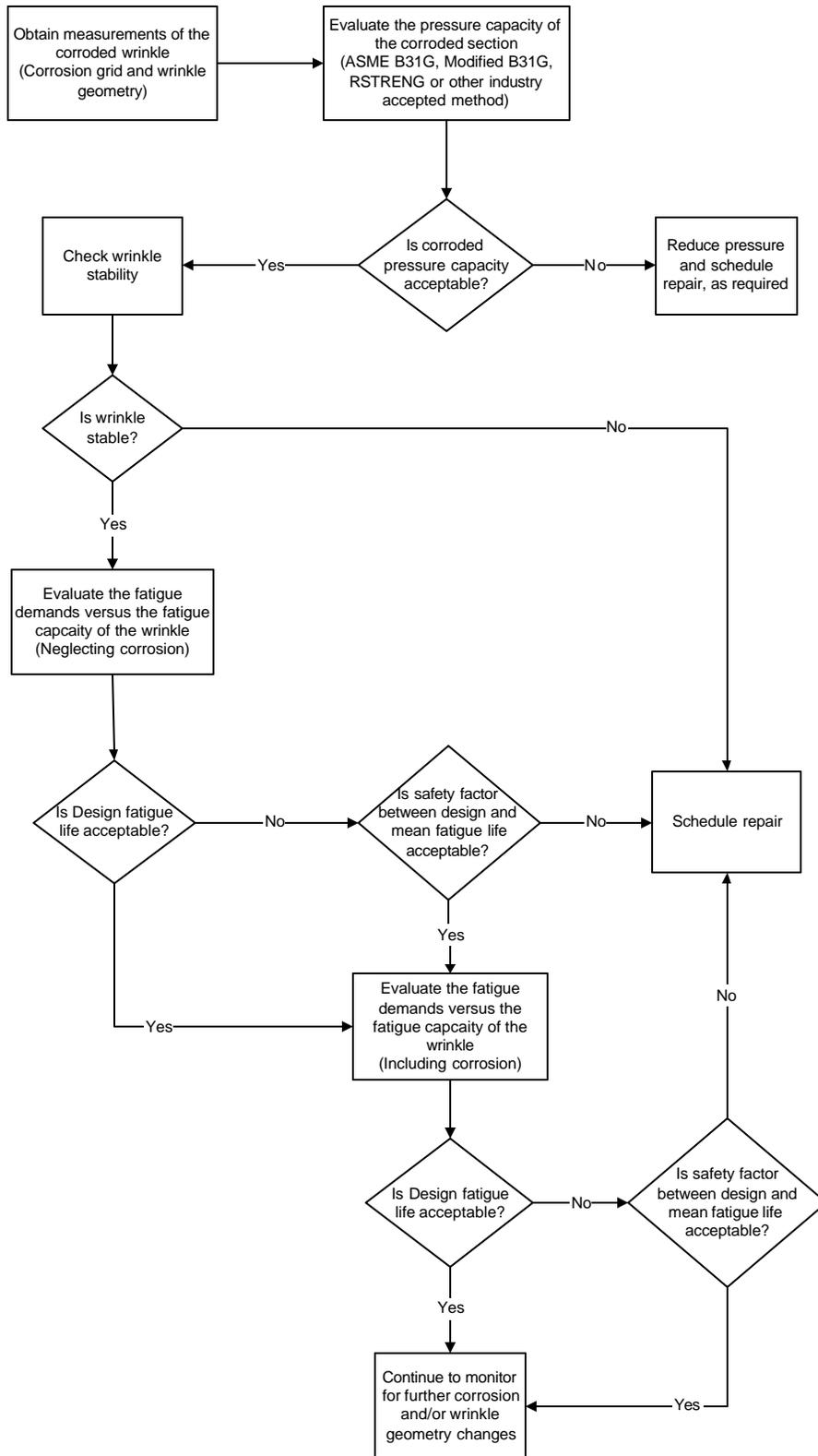
- f) **Perform Fatigue Damage Calculations.** Use the intensified stress demand measures at or near the wrinkles with appropriate fatigue “capacity” curves (i.e., S-N curves) to compute the number of operating cycles to failure for each wrinkle. In order to estimate the actual number of cycles to failure, a fatigue S-N curve that passes through the mean of the S-N data should be used. A “design” S-N curve including a Safety Factor would be used to estimate the design life of the wrinkle under consideration based on the type of criteria that would be used for the design of a new pipeline (as opposed to performing a serviceability assessment of an existing pipeline). Section B.2.2 and Appendix A provide a recommended *mean* and *design* curve for piping and pipeline materials. Operating cycles that may not produce the full design pressure or temperature differential can be included using an appropriate cycle counting algorithm (e.g., rainflow) and a cumulative damage rule (e.g., Miner’s rule (Miner, 1945)).

This analysis framework is attractive because it is a relatively simple approach that includes (a) the “global” effects of pipe-soil interaction, (b) the “local” effects of unrestrained stress concentration at wrinkles and (c) a basis for estimating the fatigue damage at wrinkles for a given pressure or temperature differential cycle. This framework is also reasonably consistent with the analysis procedures presently used to evaluate piping and pipeline designs per the ASME B31 Codes (e.g., elastic pipe, inelastic supports, evaluation of pressure and temperature differential load cases, a fatigue stress check, etc.). The procedure can be used to determine appropriate limits on wrinkle geometry. The wrinkle acceptance criteria presented in “*Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends*” (Rosenfeld et. al 2002) is based essentially on the application of this approach to a set of generic pipeline configurations.

The end result of sub-steps (a) through (f) is an estimate of the number of pressure and/or temperature differential cycles required to produce a fatigue failure in the wrinkle. In order for this information to be useful, the analyst must obtain detailed information regarding the operating pressure and temperature cycles of the pipeline at the location of the wrinkle (i.e., operating scenarios). By combining this historical information, with the “damage per cycle” results from the fatigue calculations, estimates of the (mean and design) fatigue “life” can be established in terms of anticipated forward-looking operations. If the fatigue lives established using this approach are shorter than the design life of the pipeline, then, decisions may need to be made to schedule a repair of the wrinkle, depending on the acceptability of the safety factor associated with the design curve.

5. **Evaluate the Fatigue Demands vs. the Fatigue Capacity of the Corroded Wrinkle.** Assuming that the corrosion has “passed” the remaining strength check performed in item 2, the wrinkle is determined to be stable as described in item 3, and the uncorroded wrinkle has “passed” the fatigue check performed under item 4, an additional step in the framework is to evaluate the fatigue demands on the *corroded* wrinkle with respect to the fatigue capacity. As previously noted, based on experience and discussions with industry experts, pipeline failures due to fatigue in

corroded ripples, wrinkles or buckles were not identified. Nevertheless, the presence of corrosion in or near a wrinkle may increase the local stress field beyond the increased local stress levels that exist due to the wrinkle alone. In order to evaluate corroded wrinkles, sub-steps (a) through (f) under item 4 above could be repeated except that in this case, the *SCFs* would be established based on the combination of the wrinkle and the corrosion. Ideally, the appropriate *SCF* could be estimated based on a case-specific FEA model of the corroded wrinkle. As discussed in Section B.3.3, it may be possible to develop generalized *SCFs* for corroded wrinkles based on regression analysis on the results of FEA for a range of corroded wrinkle configurations, provided a wide range of the important wrinkle and corrosion geometries could be considered. In addition to the geometric parameters describing the wrinkle (e.g.  $D/t$ ,  $h/D$ ,  $h/t$ ,  $\lambda/h$  and  $c/pD$ ), the regression would include a metal loss parameter (e.g.,  $A/A_o$ ) and a means of defining the location of the corrosion relative to the peak of the wrinkle such that the *SCF* is increased when the corrosion is significant and the *SCF* decomposes to the uncorroded value when the metal loss reduces to zero.



**Figure B.5 Conceptual Procedure for Corroded Wrinkle Evaluation**

## **B.6 Example Application of Demand Capacity Framework**

The previous section provides an overview of the recommended demand capacity comparison procedure for evaluation of pipeline segments containing wrinkles and corrosion. This section provides a simple illustrative example of how the framework could be applied to a pipeline containing wrinkles and corrosion.

### **B.6.1 Description of Example Problem Parameters**

This example problem considers a 24-inch diameter, liquid products pipeline with a 0.266-inch wall thickness ( $D/t=90$ ). The pipe steel is X-60. The pipeline has been in operation for a period of 10 years and it has a design life of 25 years. The MAOP of the line is 960 psi and the normal operating pressure at the subject location is approximately 700 psi. The estimated temperature differential (between operating and tie-in) at the subject location is  $+80^{\circ}\text{F}$ . The line is buried under 3 feet of sandy soil cover. The pipe was found to contain several wrinkles, and furthermore, some of the wrinkles had experienced significant external corrosion. Based on geometry pig surveys of the pipeline in the vicinity of the wrinkles that showed no evidence of pipe or ground movement, the wrinkles in the pipe were believed to be stable and were most likely formed during cold bending. The individual wrinkles are located in joints containing cold side bends with a bend angle of approximately  $10^{\circ}$  and with a bend radius equal to the minimum radius permitted by the B31.4 code ( $R=30D=60$  feet).

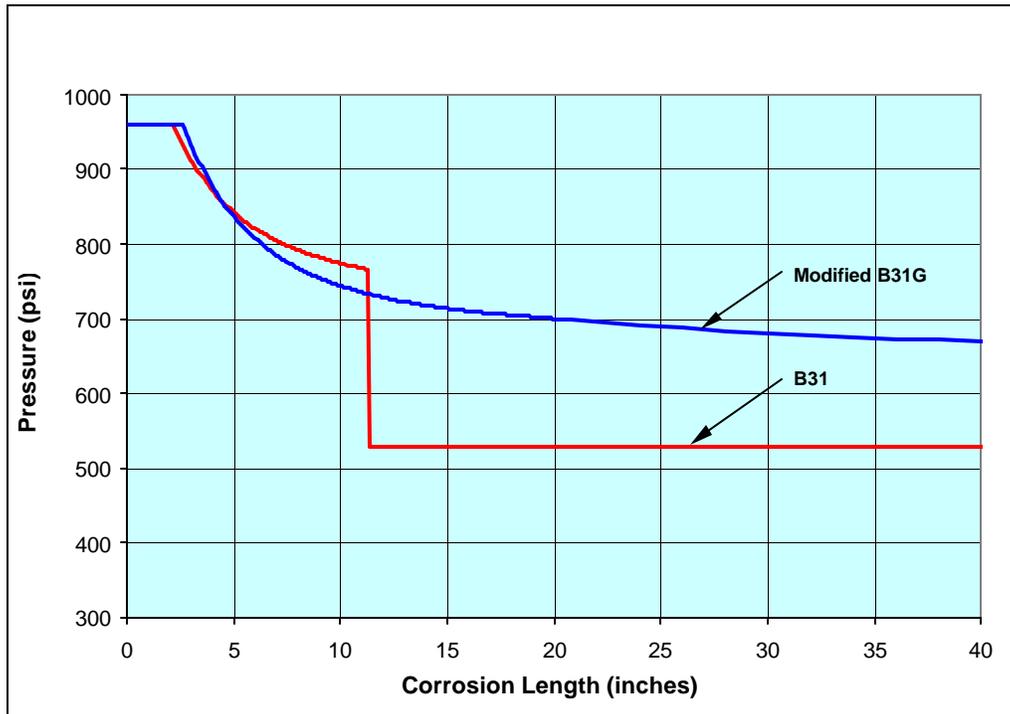
### **B.6.2 Step-by-Step Application of the Demand Capacity Framework Parameters**

Following the framework outline, the first step in the evaluation is to establish the geometry of the corrosion and the geometry of the wrinkles. The grid of the most significant corrosion location was found to contain a maximum corrosion depth of 0.133 inches (50% of the nominal wall thickness) with a maximum along the pipe corrosion length of about 10 inches. Measurements of the wrinkles revealed wrinkled geometries with inward deformations of the pipe wall of approximately 1 inch (i.e., a  $d/D$  ratio of about 4%) and a wavelength of approximately 9 inches.

The next step is to evaluate the pressure capacity of the corroded sections of pipe. This was accomplished using both the B31G and modified B31G procedures. Figure B.5 presents a plot of the resulting MAOP for a corrosion depth equal to 50% of the wall thickness for this pipe over a range of corrosion lengths. For a corrosion length of 10 inches, the MAOP is reduced to 772 psi based on B31G while the MAOP is reduced to 743 psi based on modified B31G. Notice that if the corrosion length is increased to about 11.3 inches (corresponding to  $z = 20$ ), the B31G method drops the MAOP to 527 psi (due to its discontinuous function) while the MAOP of the modified B31G method is about 732 psi. The fact that the MAOP of the corroded pipe is less than the original design MAOP requires a reduction in pressure and scheduling of a corrosion repair (see appropriate CFR integrity management rules).

The next step is to perform a fatigue demand capacity comparison of the wrinkles. The procedure used herein is based upon results presented in *“Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends”* (Rosenfeld 2002). Based on a review of the operating history of the subject pipeline, it was determined that the location of interest was subjected to a fairly wide range of operating pressure and temperature cycles within a typical one-year period. The pressure cycles result from operating pressure changes during batching and delivery of product while the temperature cycles are mainly related to seasonal temperature variations of the pipeline contents. Therefore, for this pipeline, the pressure and temperature

cycles were considered as non-coincident. The characterized “spectrum” of pressure and temperature differential cycles are summarized in Table B-2 and Table B-3.



**Figure B.6 MAOP of Corroded 24-inch Diameter, 0.266-inch Wall, X60 Pipe with 50% Metal loss**

**Table B-2 Pressure Cycle Spectrum Over Typical One Year Time Period**

Pressure Range (psi)	Number of Cycles (n)
700	5
500	50
300	500
100	5000

**Table B-3 Temperature Cycle Spectrum Over Typical One Year Time Period**

Temperature Differential (degrees F)	Number of Cycles (n)
80	5
60	25
40	250
20	2500

For this illustration, a detailed shell finite element analysis was undertaken of a representative wrinkle geometry. The height of the wrinkle was approximately 1 inch (inward) and the wavelength of the wrinkle

was about 9 inches. The wrinkle, which extended over about 50% of the pipe circumference, is well separated from the nearest girth weld. The peak of the wrinkle was located at the 3 o'clock position of the pipe, which was the intrados location in a side bend. The wrinkle did not span over the longitudinal seam. Elastic analysis of the FEA mesh of this wrinkle for internal pressure loading indicated that the stress concentration factor or *SCF* (i.e., the ratio of the maximum local stress to the nominal hoop stress  $PD_i/2t$ ) for internal pressure load was 2.54. Elastic analysis of the FEA model of this wrinkle for bending moment loading indicated that the *SCF* (i.e., the ratio of the maximum local stress to the nominal bending stress  $M/Z$ ) for bending moment loads was 2.72.

To illustrate the factor of safety of the design versus mean fatigue relationships, the fatigue evaluation was undertaken using both the mean and design fatigue S-N relationships developed in Appendix A. The *mean* fatigue S-N relationship is summarized as follows:

$$i \cdot S = 490 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 20 \quad \text{for } N > 8.8 \times 10^6$$

Applying a factor of safety of 2.0 on stress range leads to the following *design* S-N relationship:

$$i \cdot S = 245 \cdot N^{-0.2} \quad \text{for } 20 = N = 8.8 \times 10^6$$

$$i \cdot S = 10 \quad \text{for } N > 8.8 \times 10^6$$

In these relationships, *S* is the nominal stress range (in ksi), *N* is the number of stress reversals to failure, and *i* is the fatigue effective stress intensification factor (*SIF*). The constant 245 is the Markl material constant “*C*” and the constant 490 is equal to  $2C$ . As discussed in Section B.4, the fatigue effective SIF can be taken as:  $i = SCF/2$ . The steps for evaluating the fatigue damage due to pressure cycles at this wrinkle are as follows:

1. Compute the nominal hoop stress due to pressure ranges of 700, 500, 300 and 100 psi using  $S = PD_i/2t$  — the corresponding values are 30.9, 22.1, 13.2 and 4.4 ksi.
2. Compute the localized fatigue demand measure  $i \cdot S = SCF/2 \cdot S = 2.54/2 \cdot S$  — the values corresponding to pressure ranges of 700, 500, 300 and 100 psi are 39.2, 28.1, 16.8 and 5.6 ksi.
3. Pressure cycles result in stress-controlled loading. Therefore, as discussed in B.2.2.1, use the stress-controlled material constant  $C_c$  equal to  $2/3$  of the displacement-controlled material constant  $C$  ( $C_c = 2/3 \cdot C$ ) in the mean and design fatigue curves. The endurance limits (20 ksi for the mean curve and 10 ksi for the design curve) are also scaled by the  $2/3$  factor. For localized fatigue demand measure values ( $i \cdot S$ ) below the endurance limits, the corresponding *N* value is 8. For  $i \cdot S$  values above the endurance limits, solve for the number of cycles *N* on the mean and design fatigue S-N curves using:

$$N = \frac{1}{\left(\frac{i \cdot S}{490 \cdot 2/3}\right)^5} \quad \text{for the mean curve}$$

$$N = \frac{1}{\left(\frac{i \cdot S}{245 \cdot 2/3}\right)^5} \quad \text{for the design curve}$$

The *mean*  $N$  values corresponding to pressure ranges of 700, 500, 300 and 100 psi are 40,188, 212,321, 2,779,571 and 8, respectively and the *design*  $N$  values corresponding to pressure ranges of 700, 500, 300 and 100 psi are 1,256, 6,635, 86,682 and 8, respectively. The annual fatigue usage ratio is computed as  $S(n/N)$  and the fatigue life in years is equal to the inverse of the usage ratio. Using this approach, the mean fatigue life due to pressure cycling of this wrinkle is estimated as 1,863 years, while the design fatigue life is estimated as 58 years.

The steps for evaluating the fatigue damage due to thermal cycles are as follows:

4. The nominal stress demand in a buried pipe subject to a temperature change should be computed based on buried pipe stress analysis of the configuration of interest. For this case, the buried pipe analysis results for a 24-inch diameter pipe published in Reference (IPC, 2002) have been used. The pipe is buried under 3 feet of cohesionless sand cover with a friction angle of 30° and an in-situ density of 100 pcf. Buried pipe bend configurations with bend angles of 10°, 20°, 30° and 40° were subjected to a pipe temperature of 120°F and the resulting pipe stresses at the apex of the bend was obtained at 10°F increments. The nominal longitudinal stress ( $S = F/A \pm M/Z$ ) due to temperature differentials of 80°, 60°, 40° and 20° F for the subject bend angle of 10° are 26.2, 19.6, 13.0 and 6.5 ksi, respectively.
5. Compute the localized fatigue demand measure  $i \cdot S = SCF/2 \cdot S = 2.72/2 \cdot S$  — the values corresponding to temperature differentials of 80°, 60°, 40° and 20° F are 35.6, 26.7, 17.7 and 8.8 ksi.
6. Thermal cycles result in displacement or strain-controlled loading. Therefore, the basic C factor in the S-N relationships defined in Section B.2.2 is used to represent the fatigue capacity. For localized fatigue demand measure values ( $i \cdot S$ ) below the endurance limits (20 ksi for the mean curve and 10 ksi for the design curve), the corresponding  $N$  value is 8. For  $i \cdot S$  values above the endurance limits, solve for the number of cycles  $N$  on the mean and design fatigue curves corresponding to the above  $i \cdot S$  values using:

$$N = \frac{1}{\left(\frac{i \cdot S}{490}\right)^5} \quad \text{for the mean curve}$$

$$N = \frac{1}{\left(\frac{i \cdot S}{245}\right)^5} \quad \text{for the design curve}$$

The *mean*  $N$  values corresponding to temperature differentials of 80°, 60°, 40° and 20° F are 494,004, 2,081,729, 8, and 8, respectively and the *design*  $N$  values corresponding to temperature differentials of 80°, 60°, 40° and 20° F are 15,438, 65,054, 508,117 and 8, respectively. The annual fatigue usage ratio is computed as  $S(n/N)$  and the fatigue life in years is equal to the inverse

of the usage ratio. Therefore the mean fatigue life due to thermal cycling of this wrinkle is estimated as 45,455 years while the design fatigue life is estimated as 833 years.

For this example, the pressure cycles are the dominant source of fatigue damage. The annual fatigue usage ratios due to pressure cycles and thermal cycles can be combined to develop estimates of the mean and design fatigue life of this wrinkle. The mean fatigue life is estimated as 1,779 years, while the design fatigue life is estimated as 54 years.

The finite element analysis of the wrinkle described above was extended to include a characterization of the 10-inch long, corrosion patch with 50% wall loss. Elastic analysis of a mesh of the corroded wrinkle indicated that the *SCFs* for pressure and moment loading were both increased by 15% (the *SCF* for internal pressure load was increased from 2.54 to 2.92 and the *SCF* for bending moment loads was increased from 2.72 to 3.13). Steps 1 through 6 described above were repeated using the increased *SCF* values associated with the corroded wrinkle (in effect the localized fatigue demand measure  $i \cdot S$  was increased by 15%). For the corroded wrinkle, the resulting mean fatigue life is estimated as 929 years while the design fatigue life is estimated as 29 years.

Several points can be made based on this fatigue evaluation:

- For cases where the stresses are above the endurance limit, the ratio of “mean” fatigue life to the “design” fatigue life is equal to 32. This factor of 32 represents the factor of safety on cycles and is equal to the factor of safety of 2 on stress raised to the power 5:  $32=2^5$ .
- The presence of corrosion in the wrinkle resulted in an increase in the localized stresses in the wrinkle, which were already larger than the nominal stresses in the pipe. A 15% increase in the localized stresses due to corrosion, resulted in an approximate factor of 2 reduction in both the mean and design fatigue lives. This approximate factor of 2 corresponds to the increased stress raised to the power 5:  $2 \sim 1.15^5$ .
- Evaluation of the wrinkles with and without corrosion using a design fatigue curve resulted in design fatigue lives of 55 and 27 years, respectively (this assumes that the evaluated anomaly has been present in the pipeline since startup). Both of these design fatigue lives exceed the 25-year design life of the pipeline. This means that even the corroded wrinkle would satisfy the type of fatigue design criteria that would be used for the design of a new pipeline, including a significant safety factor.
- For this example, it would be concluded that fatigue of the wrinkle (with or without corrosion) does not pose the same or greater hazard that pressure alone. In other words, evaluation of the corrosion using established industry procedures for pressure capacity (and derating the MAOP or repairing the corrosion if necessary) would take precedence over fatigue concerns for this case.

### ***B.7 Summary and Conclusion***

Based on the combined experience of the project team and discussions with industry experts, pipeline failures due to fatigue in corroded ripples, wrinkles or buckles could not be immediately identified. Moreover, there is a lack of full-scale experimental evaluations of corroded pipes that were designed to produce fatigue failures in the corrosion; most corroded pipe tests are aimed at evaluating burst pressure. However, pipelines that that have experienced external corrosion at elbows have been identified during the

research. In this case, there was concern that the corrosion within the elbow would increase the flexibility and stress intensification effects with a potential reduction in the fatigue capacity of the elbow. Detailed proprietary FEA and fatigue testing of both uncorroded and corroded elbows led to the conclusion that evaluation of the pressure capacity of the corrosion by any established methodology (e.g., B31G, RSTRENG), and derating or repairing if the corrosion is severe enough should take precedence over fatigue concerns. Using established pressure integrity methods should result in derating or repairing the pipeline long before fatigue becomes a concern for all but the most extreme scenarios of cyclic stress demand. The same conclusion applies to corroded wrinkles.

This Appendix contains an overview of an analysis framework for the evaluation of corrosion in wrinkled pipe sections, including a review of methods used to evaluate the pressure capacity of corroded pipe and a detailed discussion about the fatigue capacity of pipe steels and a rational basis for evaluation of fatigue damage in corroded wrinkled pipe. The key aspects of this Appendix are summarized as follows:

- The primary concern for corrosion in a pipeline is how it will affect the pressure capacity of the pipe. The pipeline industry has well-accepted procedures in place for evaluating the pressure capacity of corroded pipelines. These procedures are supported by a database of hundreds of burst test results. Pipeline operators and consultants have a wealth of experience with this type of evaluation.
- Once a wrinkle is discovered in a pipeline, the primary concern is the stability of the wrinkle (e.g., are the wrinkle deformations likely to increase due to continued settlement?). If it is unlikely that the deformations in a wrinkle will increase (i.e., the wrinkle is stable), the primary concern becomes the potential for fatigue damage in or near the wrinkle. There are currently no universally accepted guidelines or specific criteria that can be used to limit the geometry of wrinkles in pipelines based on fatigue considerations. However, it is understood that the B31.8 Code Committee is presently considering an agenda item allowing for wrinkles with peak-to-trough heights of up to 1% of the pipe diameter (Rosenfeld et. al 2002). It is also believed that the B31.4 Code Committee is considering an agenda item related to the acceptance of mild wrinkles, as well.
- When stable wrinkles in pipelines are found to contain corrosion, the concerns should be the same as those expressed above:
  - Is the pressure integrity of the pipeline at risk?
  - Is the corroded wrinkle at risk of experiencing fatigue damage or failure?
- The first and most important step in the recommended framework is to evaluate the pressure integrity of the corroded wrinkle. It is believed that the geometry of the wrinkle is unlikely to have a significant effect on the burst capacity of the corroded section of pipe since the plastic strains in the wrinkle will tend to “wash out” at the large strains associated with the burst pressure. For this reason, it is recommended that the corrosion evaluation be performed by treating the pipe as if it was cylindrical (i.e., neglecting the wrinkled geometry). We are aware of some proprietary burst tests on wrinkled pipe specimens that support this analysis approach. If the pressure integrity of the pipe is affected by the corrosion, then the operator should proceed based on the appropriate CFR integrity management rules.

- Once the pressure integrity has been evaluated, the next step is to evaluate the fatigue integrity of the wrinkle, neglecting the presence of corrosion. The analysis approach for this step is far less well established and more time consuming than the procedures used to evaluate pressure integrity. The highlights of the fatigue evaluation are summarized as follows:
  - Develop representative annual “histograms” of pressure and temperature cycles for the pipeline at the location of interest.
  - Develop and analyze a case-specific “global” buried pipe model at the location of interest to develop estimates of the global loads and nominal stresses at the wrinkle.
  - Develop and analyze a case-specific “local” FEA model of the wrinkle geometry of interest to establish estimates of the stress concentration factor (SCF) for internal pressure and bending moment loads.
  - Combine the pressure and temperature cycle histograms, with the corresponding nominal stresses and the pressure and bending moment SCFs to obtain the localized fatigue stress demands at the wrinkle.
  - Use fatigue “S-N” curves to compute estimates of the annual fatigue damage at the wrinkle using a fatigue usage factor (where 0.0 corresponds to zero fatigue damage and 1.0 corresponds to fully consumed fatigue life). The fatigue life in years is equal to the inverse of the annual usage factor. Compare the *design* fatigue life (computed using a “design” fatigue curve containing a significant safety factor on stress or cycles) to the design life of the pipeline. If the design fatigue life is longer than the design life of the pipeline, the wrinkle satisfies the type of fatigue criteria that would be used for the design of a new pipeline, including a significant safety factor (as opposed to performing a serviceability assessment of an existing pipeline). If the design fatigue life is shorter than the design life of the pipeline, the wrinkle may still be considered as acceptable depending on the safety factor in the design S-N curve.
- Once the fatigue integrity of the wrinkle has been considered, the fatigue analysis can be extended to consider the effects of corrosion within the wrinkle. The only change to the evaluation approach is that the detailed “local” FEA model of the wrinkle is modified to include a characterization of the corrosion. The corrosion is typically characterized as a rectangular patch. Depending on the geometry of the corrosion (e.g., its length, width and depth and its location with respect to the peak of the wrinkle), the SCFs are likely to increase relative to those of the uncorroded wrinkle.

As noted above, the fatigue analysis aspects of the proposed framework are far less established and more time consuming than the procedures used to evaluate pressure integrity. However, the application of finite element analysis methods is very well established in the pipeline and piping research industry and the use of FEA as a tool for performing pipeline structural integrity and serviceability assessments is becoming much more common. We believe that FEA methods used in combination with additional experimental data represents the most promising means of evaluating complex pipe stress and deformation problems such as assessing the fatigue behavior of corroded wrinkles.

## B.8 References

- American Lifelines Alliance (ALA) “*Guidelines for the Design of Buried Steel Pipe*,” Published by the ASCE American Lifelines Alliance, [www.americanlifelinesalliance.org](http://www.americanlifelinesalliance.org), July 2001.
- ASCE, “*Guidelines for the Seismic Design of Oil and Gas Pipeline Systems*,” Committee on Gas and Liquid Fuel Lifelines, 1984.
- ASME, “*Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2.*”
- ASME B31G Committee, “*ASME B31G: Determining the Remaining Strength of Corroded Pipelines*,” Proposed DRAFT Revision, Submitted to B31 Mechanical Design Committee, September 10, 2003.
- ASME PVP publication, “*Stress Indices and Stress Intensification Factors of Pressure Vessel and Piping Components*,” ASME PVP publication, Volume 50, 1981.
- Kiefner, J. F., and P. H. Vieth, “*A Modified Criterion for Evaluating the Remaining Strength of Corroded Pipe*,” PRCI, December 22, 1989.
- Lane, P. H. R., and Rose, R. T., “*Comparative Performance of Pressure Vessel Nozzles Under Pulsating Pressure*,” Proc. Inst. Mech. Eng., Vol. 175, 1961.
- Markl, A. R. C., “*Fatigue Tests of Piping Components*,” Paper 51-PET-21, *Trans. ASME*, 17 pages, 1952.
- Markl, A. R. C., “*Piping-Flexibility Analysis*,” Paper No. 53-A-51, *Trans. ASME*, 23 pages, February 1955.
- Miner, M. A., “*Cumulative Damage in Fatigue*,” *Journal of Applied Mechanics*, ASME Vol. 12, 1945.
- National Association of Corrosion Engineers (NACE) Technical Committee Report, “*In-Line Nondestructive Inspection of Pipelines*,” Publication 35100, December 2000.
- Olson, R., Clark, T., Odom, T., “*Evaluation of the Structural Integrity of Cold Field Bent Line Pipe*,” Proceedings of the Ninth Symposium on Line Pipe Research, September 1996.
- Rodabaugh, E. C., “*A Review of Area Replacement Rules for Pipe Connections in Pressure Vessels and Piping*,” WRC Bulletin 335, August 1988.
- Rosenfeld, M. J., Hart, J. D., Zulfiqar, N., Gailing, R., “*Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends*,” IPC02-27124, ASME International Pipeline Conference, Calgary, Alberta, Canada, September 29-October 3, 2002.
- SSD, Inc. and Kiefner & Associates, Inc., “*Relationships Between Stress Intensification Factors And Stress Concentration Factors*,” Draft Report Submitted to PRCI and GRI, December, 2003.
- “Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2,” ASME.