



**Basics of Metal Fatigue in Natural Gas
Pipeline Systems — A Primer for Gas Pipeline Operators**

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EXECUTIVE SUMMARY

Occasionally and under certain circumstances, fatigue may constitute a potential threat to the integrity of some pipelines. Part 192, Subpart O, “Implementing Integrity Management”, Paragraph 192.917(e)(2) requires an operator to address the threat of “cyclic fatigue”. This document explores the questions “What is fatigue?”, “Where can it occur in a gas pipeline facility?”, and “What can be done about it?” The objective is to provide operators and others interested in natural gas pipeline safety with a useful understanding of the extent to which fatigue could pose a legitimate and actionable safety threat, as well as to demonstrate the authors’ opinion that in most respects, fatigue remains a comparatively minor risk component of the overall spectrum of threats to natural gas pipeline safety.

Basic considerations of fatigue, physical testing of pipe, and operating experience have shown that fatigue due to pressure cycles is not a limiting factor on the service life of sound pipe free of gross defects. Fatigue in longitudinal seams due to the effects of pressure cycles acting on longitudinal seam defects has only been observed where initial flaws were sufficiently large, and even then only in certain liquid pipelines that operate in a cycle-intensive manner. Many liquid pipelines are not subject to this threat, and no failures of this type have been identified in gas pipelines. The reason for this is that gas pipelines do not experience a sufficient number of large pressure cycles to cause fatigue crack growth to failure within the expected service life of the facility, from initially present flaws that are small enough to survive a hydrostatic pressure test to the usual margins above the MAOP.

In the infrequent occasions where fatigue does occur in gas pipeline facilities, it is generally for reasons other than operational pressure cycles or where unusual circumstances were present. Some scenarios involving fatigue affecting gas pipelines include: effects of pressure pulsations or mechanical vibration associated with the operation of reciprocating compressor units, inadequate bracing of above-ground piping subject to vibration, or vortex-shedding on pipe exposed to water currents. Details of construction in fabricated assemblies, such as welding quality and design choices, often affect susceptibility, as may damage to the surface of the pipe.

A hydrostatic pressure test or in-line inspection is unlikely to serve as an effective assessment for the presence of these threats on the pipeline, or as a threat mitigation technique. The reasons for this have to do with the nature of some susceptible facilities (e.g., piping or appurtenances in compressor stations), the random nature of precipitating conditions or events

(e.g., exposure of pipe to floodwaters), or the mode of fracture (e.g., circumferentially). Most situations involving fatigue are best managed through a process involving observation of conditions affecting a facility that could be conducive to fatigue, engineering evaluation on a case-by-case basis, and corrective action as appropriate. A sample process for identifying systems potentially susceptible to “cyclic fatigue” and other threats identified in paragraph 192.917(e)(2) is included.

BASICS OF METAL FATIGUE IN NATURAL GAS PIPELINE SYSTEMS — A PRIMER FOR GAS PIPELINE OPERATORS

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INTRODUCTION

The natural gas pipeline industry is rapidly implementing comprehensive integrity management practices to meet the demands of new regulatory imperatives and public interests. These new demands require formal integrity management planning programs be developed and applied where pipeline failures could affect “High Consequence Areas”. A formal integrity management plan (IMP) incorporates some process for identifying threats to a pipeline’s integrity. Such threats come in many forms and are uniquely dependent on a wide range of attributes associated with an individual pipeline segment. Once such threats are identified, the pipeline operator must characterize the degree of risk associated with the threat as a means of prioritizing responses, identify suitable methods to assess the presence of the threat, and develop appropriate mitigations.

Interest (or concern) has arisen regarding metal fatigue as one such possible integrity threat. We know from some pipeline failures that occasionally and under certain circumstances, fatigue may constitute a potential threat. More to the point, 49 CFR Part 192, Subpart O, “Implementing Integrity Management”, Paragraph 192.917(e)(2) requires an operator to address the threat of “cyclic fatigue”. In order to meet the objectives and requirements of this rule, a pipeline operator must be able to discern what types of pipe or piping construction are susceptible to fatigue, what modes of pipeline operation or situations are conducive to fatigue, the consequences of a fatigue-related incident, and what actions could be taken to mitigate the threat. At the request of the Pipeline Research Council International (PRCI), the Gas Technology Institute (GTI), and the Interstate Natural Gas Association of America (INGAA) this review was undertaken to provide fundamental information to gas pipeline operators to enable them to address issues of fatigue as they pertain to natural gas pipelines. In so doing, the document will explore the questions “What is fatigue?”, “Where can it occur in a gas pipeline facility?”, and “What can be done about it?”

The cumulative body of knowledge derived by theory, test, and experience on the subject of fatigue and its effects on piping, pressurized equipment, and welded structures is vast in scope and detail, and it is not the intention of this document to summarize that. Rather, it is intended that this document provide natural gas pipeline operators and others interested in natural gas pipeline safety with a useful understanding of the extent to which fatigue could pose a legitimate and actionable safety threat, as well as to demonstrate the authors' opinion that in most respects, fatigue remains a comparatively minor risk component of the overall spectrum of threats to natural gas pipeline safety.

FATIGUE FUNDAMENTALS AND CONCEPTS

What is Fatigue?

Fatigue is a process of structural degradation caused by fluctuations or cycles of stress or strain. Such stresses or strains are typically concentrated locally by structural discontinuities, geometric notches, surface irregularities or damage, defects, or metallurgical nonhomogeneities. Fatigue may occur in three sequential stages: the formation of a crack, called "initiation"; the stable incremental enlargement of the crack in service, called "propagation"; and rapid unstable fracture, i.e., failure. Fatigue arises as a result of accumulated cycles of applied stress in service. The term "cycles" implies a repetitive loading condition or a randomly fluctuating load. Fatigue is not caused by a steady load or a one-time loading event. The phases of initiation, propagation, and final failure, though sequential, are distinct and governed by separate considerations.

Fatigue Initiation

Initiation of fatigue occurs at microstructure-scale nucleation sites within the material such as inclusions, pores, or soft grained regions, or as they become generated through microvoid coalescence by the straining process.[1] The presence of macro-scale stress concentrators, or more accurately strain concentrating features, enhances this process. Examples of stress concentrators are:

- grooves or notches
- threads
- abrupt transitions in metal thickness
- weld bead toes

- welding workmanship flaws
- manufacturing flaws in the pipe seam or pipe body
- pipe deformations such as dents or buckles
- mechanical damage such as gouges
- sites of environmental attack such as corrosion pits or stress-corrosion cracks.

Stress concentrators generically are characterized by a notch or notch-like geometry. A sharp notch will be more prone to form a fatigue crack than a blunt notch when subjected to the same cyclical loading conditions, because the sharper notch produces a more severe concentration of strain locally at its root. While it is true that fatigue can occur eventually if stress cycles are sufficiently numerous and large in magnitude, even where the material surface is free of gross stress-concentrating features, so many cycles of stress would be required for fatigue to initiate in the absence of a stress-concentrating feature that this is not a scenario of concern in a gas pipeline context, as will be demonstrated subsequently in this report. Conversely, the onset of fatigue is promoted by the presence of stress-concentrating features in proportion to their severity. This has important implications for pipe affected by mechanical damage.

The fatigue-crack-initiation behavior of a material is described by an “S-N” curve, which is a graph of the magnitude of cyclical stress amplitude, S (amplitude being half the total range of stress cycle or variation), plotted against the number of cycles, N , in which the cyclic stress amplitude will cause a failure. Some S-N data for plain carbon steel and low-alloy steel are shown in Figure 1(a).[2] The S-N curves are empirically derived from large numbers of tests in which polished round bars of material are cyclically loaded at specific nominal stress levels until fracture occurs. The data in Figure 1(a) were used as the basis for the fatigue design S-N curve in the ASME Boiler and Pressure Vessel Code (BPVC), Section VIII, Division 2, Appendix 5 shown in Figure 1(b). The curve is drawn well below the data points to provide a factor of safety. S-N curves for specific materials or structural weldment configurations can be found in other design standards and fitness-for-purpose standards as well.

The S-N curve demonstrates that larger cycles of stress result in failure in fewer cycle occurrences, while smaller cycles of stress result in failure after a greater number of cycles. At the left end of the curve, the stress amplitudes may greatly exceed the yield strength of the material, because what is shown as “stress” is actually a computed pseudo-elastic stress quantity

based on local strain that could fall well into the plastic range locally. The stress includes the concentrating effect of any notch. Large amounts of plasticity hasten the fatigue initiation process, resulting in failure in fewer cycles. This region of fatigue performance is sometimes referred to as “low-cycle fatigue” because it pertains to failure in a relatively low number of load cycles. At the right end of the curve, on the order of 10^5 or more cycles of stress are tolerable, if the magnitude of stress cycles is very low. This region of fatigue performance is sometimes referred to as “high-cycle fatigue” because it pertains to failure in a large number of load cycles. If stress amplitude is sufficiently low, the S-N curve flattens out and the fatigue life is infinite for practical purposes. This stress level is referred to as the “endurance limit”. As-finished pipe and welds may not exhibit an endurance limit.[15]

The S-N curves are developed from tests to failure, so the number of cycles includes the propagation and final fracture phases of fatigue without explicitly describing them. It is presumed that in the absence of an initial crack, these phases comprise a very small proportion of the overall fatigue life.

The fatigue-initiation characteristics of a given material, design feature geometry, surface finish characteristics, and loading level are of great importance to the design of rotating machinery, vehicles, aircraft, and highway bridges because such structures rapidly accumulate millions of individual stress cycles. In contrast, the initiation phase of fatigue is of little concern with the pressure design of pipe, because the magnitude of hoop stress cycles due to pressure variation is in the range where 10^5 to 10^6 pressure cycles from 0-MAOP-0 would be required to cause failure, and this is far more than most pipelines would be expected to experience. The initiation phase of fatigue is a significant consideration in the design of piping systems that are free to flex in response to changes in operating temperature, such as piping systems located in refineries, power plants, or other process facilities. Here, the problem is not that stress cycles due to changes in operating temperature are particularly numerous, but rather that they are large in magnitude. The magnitude of flexural stress cycles in piping components, such as elbows and tees, are magnified by their geometries such that the range of stress cycle may be much greater than the yield strength of the material. Appropriate design for such circumstances is achieved by performing a piping “flexibility analysis” in accordance with the design rules for above-ground piping systems contained in standards such as the ASME B31 Code for Pressure Piping.

The resistance to fatigue crack initiation is generally proportional to ultimate tensile strength properties. However, the range of ultimate tensile strengths in line pipe does not vary over a sufficiently large range for this to be a significant factor. Resistance to fatigue initiation is enhanced by improvements in surface finish quality (smoother being better) and by treatments that impart compressive residual surface stresses (e.g., peening) or hardened surface microstructures (e.g., induction case hardening). Such treatments may be important to rotating machinery because they are initiation-sensitive owing to their high-cycle loading environment, but are not generally of value with pipe.

It is often assumed by equipment designers that the effects of fatigue are cumulative, in accordance with Miner's Rule of Linear Cumulative Damage. This rule of thumb states that the sum of fatigue-life fractions of various stress ranges, perhaps associated with different loadings or modes of operation, can be summed. Failure would be expected when the life fraction sum equals a value of 1. (Note that Miner's Rule cannot be assumed to apply when performing the explicit fatigue propagation analysis discussed in the following section, because incremental crack growth is not necessarily linear.)

Fatigue Propagation

The initiation process described above causes the formation of a crack in otherwise sound, uncracked metal. As load cycles accumulate, initiation is followed by propagation or enlargement of the crack in service. Fatigue failures frequently exhibit prominent concentric features on the exposed fracture surfaces, such as what is shown in Figure 2. These marks, referred to as "beach marks", indicate the incremental enlargement of the crack with continued cycles of loading in service. These fracture features are often somewhat elliptical in profile and typically are seen to emanate from the initial flaw, notch, area of local damage, or other stress concentration.

Propagation necessarily concerns a crack that is already present, so it is most useful to consider propagation in terms of parameters related to fracture mechanics. The crack-tip stress-intensity is an expression of the theoretical stress at the tip of a crack, derived from linear elastic fracture mechanics as $K = f[\text{geometry}] \times \sigma \times (\pi a)^{1/2}$, where σ is a nominal applied stress, a is the crack size, and K is expressed in U.S. Customary units of $\text{ksi} \cdot (\text{in})^{1/2}$ or in metric units of

$\text{MPa}(\text{mm})^{1/2}$. The geometry factor accounts for the crack's configuration and its orientation in the plate. The geometry factor may change as the flaw enlarges.

The idealized configuration of a surface-breaking crack having a semi-elliptical shape, Figure 3, is the principal one of interest in dealing with seam susceptibility issues in line pipe, since the concern is for features having configurations similar to this. Various expressions for the crack-tip stress intensity at the tip of a semi-elliptical surface flaw in a plate have been developed, with some variations or alternatives. [3,4]

The stresses in service fluctuate over a range, $\Delta\sigma$, so the fluctuation in stress-intensity is $\Delta K = f[\text{geometry}] \times \Delta\sigma \times (\pi a)^{1/2}$. An operating pressure spectrum for a natural gas pipeline may look something like what is shown in Figure 4. Typically the largest cyclical component is seasonal, which means it occurs once per year. (Some pressure signals appear to go toward zero, implying a large pressure cycle. Most likely these are the result of instruments being taken off line for calibration or other events, rather than actual pressure swings.) The pressure signal is “stochastic”, meaning it consists of an apparently random mix of signal amplitudes. In order to usefully account for the variations in stress, the operating history must be decomposed in terms of the number of cycles of various magnitudes. This is normally accomplished by performing a “rainflow” cycle-counting algorithm.[5] (The term “rainflow” is used because the analysis captures the effects of larger cycles widely separated in time, somewhat analogously to how a multilevel roof sheds rain to a wider lower-level roofline.)

Propagation or growth of a fatigue crack in service is governed by the “Paris Law”, given as $da/dN = C [\Delta K]^n$ where da/dN is the increment of crack extension per load cycle, ΔK is the magnitude of the range of alternating crack-tip stress intensity associated with a given load cycle acting on the crack of size a , and C and n are material properties. The size of the crack, a , thus increases incrementally by Δa with each load cycle ΔN while the magnitude of the stress-intensity range, ΔK , increases with each increment of crack growth. The form of the Paris Law results in an exponential increase in crack growth rate and an acceleration of crack size as load cycles accumulate, as illustrated by Figure 5.[3] The practical implication of this is that a small crack may remain small for a long time, and by the time it is detectable, either by means of in-service examination (e.g., crack detection ILI) or proof load testing (e.g., hydrostatic pressure test), the remaining safe service life could be very short. The effect of a larger initial flaw is to

move the curve to the left, resulting in failure in fewer cycles. This suggests that achieving the largest possible margin between the test pressure and the operating pressure is of value to maximizing the retest interval where pressure testing is used as the method of assessment.

The value of C can vary by several orders of magnitude, while n has been observed to vary from 2 to 4, though for most pipeline materials n usually seems to fall between 2.5 and 3. A higher C and lower n will result in a faster initial crack growth rate that does not accelerate as greatly toward failure compared to a lower C and higher n , which results in very flat initial crack growth rate and more rapid acceleration toward final failure. If only an initial and a final flaw size are known, there is no one combination of C and n that uniquely defines the crack growth curve between initial and final flaw sizes for any given operating spectrum. “Typical” values reported for C and n in plain carbon steel are $C = 3.6 \times 10^{-10}$ and $n = 3.0$ for ΔK expressed in units of $\text{ksi}(\text{in})^{1/2}$, though any given steel might exhibit very different values for the crack growth rate parameters. This “typical” relationship between da/dN and ΔK is shown in Figure 6.[6] Some fitness-for-purpose standards (e.g., API 579) also recommend higher C values specifically for welds in order to account for residual stresses.

The values of C and n are influenced somewhat by load cycle frequency and stress ratio (R , the ratio of minimum to maximum stress in a cycle), and may be influenced strongly by the chemistry environment that the crack tip becomes exposed to (e.g., dry versus aqueous, or the presence of oxygen, chlorides, sulfur, or hydrogen).[6] The exposure of the fracture to environments at the soil interface, under coatings, or in the pipe interior could enhance crack growth rates compared to those indicated by the “typical” coefficients.

“Retardation” (of the crack growth) occurs where an infrequent overload cycle blunts the crack tip and introduces a large plastic zone ahead of the crack. When the proof load is released, the residual stress field in the plastic zone is compressive, causing a delay in subsequent crack growth. Theoretically, beneficial retardation effects might be expected to occur in conjunction with high hydrostatic proof test pressures. While retardation is a proven phenomenon, it may not occur to a significant degree where the proof test is not greatly above the normal operating stress. (It does appear to play a role in delaying continued extension of near-neutral-pH SCC if the pressure test exceeds 100% of SMYS.) The effect of retardation is usually disregarded when

performing incremental fatigue crack growth computations for pipelines because assuming that it may have been operative when it actually may not have would be unconservative.

Accurate crack-growth life prediction methods do not exist in closed form. It should be apparent from the foregoing discussion that performing fatigue crack-growth life predictions is a highly technical process that requires specialized knowledge and some computational capabilities. It is not the sort of analysis that many pipeline operators are in a position to readily perform on a routine basis.

Final Fracture

The final stage of fatigue crack growth occurs when the crack-growth rate accelerates under the influence of ductile tearing and the crack grows to such size that it could fail at the next applied load cycle. The critical flaw size depends on the nominal stress, the material strength, and the fracture toughness. The crack configuration most relevant to the concern for pressure-cycle-induced fatigue is a longitudinal defect in pressurized pipe, for which accepted models exist.[7]

PRESSURE CYCLE FATIGUE IN PIPELINE LONGITUDINAL SEAMS

Susceptible Longitudinal Seams

Although it will be demonstrated subsequently that fatigue in longitudinal seams would not be expected to be an issue in any gas pipeline, it is worth reviewing which types of longitudinal seams have demonstrated susceptibility to fatigue, at least in a few liquid pipelines, as this appears to be a basis for the so-called “material threat” contained in Paragraph 192.917(e). The reader should keep in mind that fatigue crack growth in longitudinal seams as a result of pressure cycles has been experienced only in a subset of liquid products pipelines in which the pipe was affected by certain species of seam defect conditions, and the lines operated with relatively intensive pressure cycles. It would be quite incorrect to project this susceptibility to all liquid pipelines, or to all pipelines having a particular form of longitudinal seam.

ERW Type Seams

Autogeneous weld seams (e.g., electric resistance-welded and electric flash-welded

seams) are potentially susceptible to the occurrence of various types of defects, but not all have been associated with fatigue from pressure cycles. In every case involving fatigue in autogenous seams, the initial flaws are artifacts of the manufacturing process that escaped detection by the inspection process in the pipe mill and that were also small enough to pass the hydrostatic test at the mill or in the field prior to commissioning. Cold weld or lack-of-fusion defects that lie on the bondline of low-frequency-welded or dc-welded ERW seams have not been seen to grow by fatigue because the extremely low toughness of the bondline means that defects large enough to be affected by service stresses simply cannot survive and will therefore not be present.[8] Theoretically, bondline lack-of-fusion defects could grow by fatigue where the seam was made by the high-frequency ERW process, because the seam is sufficiently tough to permit large enough starter cracks to exist. However, the high-frequency welded ERW seams have typically not exhibited the large bondline lack-of-fusion defects that affected some earlier types of ERW pipe, due mainly to the more reliable bonding of the skelp edges afforded by the high-frequency current and perhaps also to improved inspection methods in the pipe mill.

Hook cracks occur as a result of soft manganese sulfide inclusions that are rolled parallel to the plate surface. When the inclusions intersect the skelp edge, they turn toward the plate surface as the seam upset is formed. When the seam flash is trimmed off, the inclusions effectively result in longitudinal cracks at the pipe surface. This is evident in the cross-section in Figure 7. Hook cracks are unaffected by the frequency of the welding current or the seam normalization heat-treatment practices because they result fundamentally from high sulfur content in the skelp. Newer high-frequency ERW pipe is often manufactured using skelp having low sulfur content, a necessity for making high-strength, high-toughness line pipe. On the other hand, low-strength grades of pipe might not use low-sulfur skelp because it is unnecessary and costs more, in which case hook cracks could still occur even with a high-frequency ERW seam.

Fatigue crack growth due to pressure cycles has occurred at hook cracks because they actually lie off the bondline in plate material, which usually has adequate ductility to allow subcritical flaws to enlarge without immediately failing. A fatigue crack that developed from a hook crack in a liquid products pipeline is shown in Figure 7 (exposed fracture surface above, and in cross-section below). It has been the industry experience that hook cracks that survive a hydrostatic test have not experienced fatigue in gas pipelines.

Transportation Fatigue Defects

Pipe that is stacked incorrectly prior to transportation by rail or ship can experience a problem known as “rail shipment fatigue”, wherein pipe leaves the mill in sound condition but develops cracks due to the cyclical inertial loads experienced while in transit. An example is shown in Figure 8 (exposed fracture surface in upper figure, and in cross-section in the lower figure). This problem was first discovered in new line pipe that failed during a hydrostatic test following construction. The exposed fracture surfaces exhibited evidence of fatigue from a large number of loading cycles, even though the pipe had never been in service. Higher D/t pipe is more susceptible to the problem than lower D/t pipe, although it has been observed in pipe having D/t below 20.[9] Rail shipment fatigue has been primarily associated with DSAW pipe, because the long seam reinforcement protrudes from the pipe surface. It has been identified on ERW pipe, but only where the pipe was sitting on a rivet head or other object pressing against the pipe body. The analysis of pressure-cycle-induced fatigue crack growth of rail shipment flaws is essentially similar to that for hook cracks in ERW seams.

Rail shipment fatigue is avoided by correctly stacking pipe on the rail car or in the vessel hold in a manner such that pipe on the bottom of the stack is not excessively stressed, and so that seam welds are not located at areas of high local stress adjacent to cradles or supports. This can be accomplished by diligent adherence to appropriate specifications such as API 5L1 for shipment by rail or 5LW for shipment by vessel. Widespread use of these standards seems to have eliminated transportation-induced fatigue as a significant threat in new pipe, although failures due to that cause still occur in older pipe.[22]

Concern is occasionally expressed over the fact that no standard currently exists for shipment of pipe over the road by truck.[22] Simple economics dictate that pipe is unlikely to be transported by truck over distances as far as it would by rail or water, while practical considerations (e.g., bridge clearances or load limits) usually limit trucks to lower stack height. Therefore, the exposure to a fatigue-inducing environment over the road is likely to be much less intense in both magnitude and duration.

Lap-Welded Pipe

Lap-welded seams fail from two possible causes unique to that seam type. One cause is

poorly-bonded seams due to oxides trapped along the bond line, and the other is embrittlement and cracking in or near the seam due to “burnt metal”, a metallurgical condition caused by overheating during seam formation. Sometimes corrosion becomes involved in a poorly-bonded lap joint. None of the types of materials tests, theoretical flaw assessment techniques, or fatigue life estimating processes described above are in any way useful for evaluating the susceptibility to seam related failures in lap-welded pipe. There is no evidence from failures that metal fatigue plays a direct role in lap-welded seam failures. Failures in lap-welded seams primarily occur when the pipe is subjected to a historically high level of pressure that is usually in excess of 50% of SMYS (which is why lap-welded pipe was historically assigned seam joint efficiency factors less than 1.0 by piping design codes). The lack of an apparent fatigue susceptibility in lap-welded seams means that hydrostatic testing can be expected to be effective for removing flawed seams and that it is unnecessary to periodically retest lap-welded seams for the purpose of finding flaws that have enlarged by a pressure-cycle induced fatigue process.

Why Gas Pipelines do not Experience Fatigue in Longitudinal Seams

Liquid pipelines have occasionally suffered from fatigue failure in the longitudinal seam, usually originating from manufacturing-related or other types of flaws present initially in the seam of some types of line pipe. The same types of pipe materials and seams, the same initial hydrostatic test requirements, and the same maximum operating stress levels prevalent in liquid pipelines are widely used in gas pipeline service as well. Are gas pipelines susceptible to the seam fatigue problem? As it turns out, the answer to this question is “No”, as will be demonstrated below.

First consider the matter of fatigue initiation. Even for a liquids pipeline, this is unlikely to occur within the normal operating life *in the absence of gross initial defects*. To demonstrate this, consider an X60 pipeline having an MAOP corresponding to a hoop stress of 72% of SMYS, equivalent to a hoop stress of 43.2 ksi. If the line pressure cycled frequently from 0 to the MAOP, the stress amplitude would be 21.6 ksi, which corresponds to a cycle life in excess of 10^5 cycles according to the ASME BPVC design curve in Figure 1. Even with daily 0-MAOP-0 cycles, this corresponds to a fatigue life of 274 years, suggesting that initially sound pipe operated in a more usual manner will fail by other mechanisms (e.g., corrosion) before it fails due to metal fatigue caused by pressure cycles.

This result is substantiated by test and experience. Pressure cycle fatigue tests were performed on sound ERW and seamless line pipe for the PRCI in order to determine whether or not sound ERW pipe was more susceptible to fatigue than pipe containing no seam at all.[10] The pipes tested were 2 samples of 12.75-inch OD x 0.188-inch WT X42 ERW pipe, 2 samples of 12.75-inch OD x 0.250-inch WT X42 ERW pipe, and 1 sample of 12.75-inch OD x 0.250-inch X42 seamless pipe. Each pipe was cyclically pressurized to produce a cyclical hoop stress range equal to 46% of the ultimate tensile strength, which corresponded to hoop stress ranges between 29.3 ksi and 35.1 ksi. The alternating stresses were then between 14.7 ksi and 17.6 ksi. All specimens, including the ERW pipes, failed by fatigue in the pipe body, with cycle lives between 1.07×10^5 and 4.27×10^5 cycles. In fact, the seamless pipe had the second shortest life proving that seam-welded pipe can perform on a par with seamless pipe. (It is noted also that these results agree closely with the ASME BPVC design S-N curve in Figure 1 rather than the raw fatigue data on which the design curve was based. This is because the raw data were produced using polished specimens, whereas the pressure tests used pipe in the as-finished condition with minute pits, mill scale, and other asperities present. The as-finished condition lowers the endurance limit by a factor of about 0.65 compared to polished bar data, which is consistent with what was observed with these tests.)

The tests described above are consistent with operating experience. There are no known cases of fatigue failures due to pressure cycle effects in gas or liquid pipelines in the absence of some sort of significant initial flaw, damage, or ill-considered design feature that concentrates stresses locally. Matters can change significantly if pronounced geometric features are present that concentrate the stress locally, such as weld toes that exhibit undercut or abrupt profiles, or structural discontinuities associated with pad-reinforced branches or heavy-walled self-reinforcing weld-on branch fittings (e.g., weld-o-lets) attached to thin-walled highly-stressed headers. Such features may have the effect of concentrating stresses by factors of 2 to 5. Locally concentrated stresses may then be great enough to result in failures in a matter of hundreds or thousands of cycles, as suggested by the S-N curves.

What if a crack-like flaw is initially present? Then the initiation phase is bypassed and fatigue life is governed by the fatigue crack propagation life according to the Paris Law. In fact, this has been experienced in some liquid pipelines. Why has it not been an issue for gas pipelines, considering that the types of pipe, the minimum hydrostatic test conditions, and the

maximum design pressures are similar for both gas and liquid pipelines? Figure 4 represented a cycle-intensive operating pressure spectrum for a gas pipeline.[11] Despite the intensiveness, gas pressure fluctuations are not particularly large, with the largest pressure differential occurring on a seasonal basis. Figure 9 shows an operating pressure spectrum from a liquids pipeline that is cycle-intensive. The pressure fluctuations are a large proportion of the maximum operating pressure. The fluctuations in operating pressures in a liquids pipeline will typically be larger and more numerous than those in a gas pipeline due to the incompressible nature of most transported fluids in a liquid state. Initially the pressure spectra in Figures 4 and 9 appear similar, until the gas spectrum is replotted on the same pressure and time scale as the liquid spectrum to produce Figure 10. Upon comparing Figures 9 and 10, it becomes obvious why pressure-cycle fatigue failures have so far not occurred in the longitudinal seams of gas pipeline even though they have been observed on several occasions in liquid pipelines.

The comparison of operating conditions explains why seam-related fatigue failures have not yet occurred in gas pipelines, but how long might be expected before they do? That can be answered by performing fatigue crack-growth life estimates in accordance with the Paris Law discussed earlier. When that is done, the results indicate that a Class 1 pipeline operated with a spectrum similar to what was shown in Figure 4 for a gas pipeline and containing an initial flaw that is 30% of the wall thickness in depth and that passes the minimum required hydrostatic test (1.1 times MAOP) would not be expected to experience a fatigue failure in less than 100 years, assuming moderately aggressive crack growth rate coefficients (C and n in the Paris Law). In a Class 4 pipeline, this increases to 500 years. On the other hand, a liquids pipeline operated according to the liquid pressure spectrum above would be expected to experience a fatigue failure in less than 10 years with a similar initial flaw size, hydrostatic test, and crack growth rate parameters.

Increasing the ratio of test pressure to operating pressure will increase the time to failure and reassessment interval. But the foregoing discussion showed that for gas pipelines the reassessment interval is much longer than the conceivable life of the pipeline. So once a gas pipeline has been hydrostatically tested, it is unnecessary to periodically retest it for the purpose of finding defects that have enlarged by the process of fatigue induced by pressure cycles. This is consistent with the concept embodied in ASME B31.8S that original manufacturing defects lie in the “static” defect category.

It is worth emphasizing that the assertion that a gas pipeline is not susceptible to pressure-cycle induced fatigue in longitudinal seams is entirely dependent on the pipeline having been hydrostatically tested to the usual level above the operating pressure (i.e., 1.25 times MAOP, or greater). If a pipeline has never been hydrostatically tested in this manner, then it is not possible to assume that its longitudinal seam is insensitive to pressure-cycle induced fatigue.

FATIGUE IN OTHER GAS PIPELINE-RELATED SITUATIONS

Fatigue does occasionally occur in gas pipeline-related systems in conjunction with situations other than pressure-cycle effects on longitudinal seams. Fatigue failures from mechanical vibrations, for example, are actually much more likely to occur in a gas compressor station than are seam failures in a gas pipeline due to pressure-cycle effects. Note that fatigue-induced failures still represent a small proportion of reportable incident causes, and most fatigue-related failures occur in facilities such as compressor stations or gas processing plants rather than along the gas pipeline right-of-way, so they generally present a lower risk to the general public.

Pulsation Bottles and Fabricated Assemblies

Reciprocating compressor stations are usually equipped with devices in line at the suction and discharge of compressor units to absorb or dampen gas pulsation effects associated with reciprocating operation. The bottle also functions as a manifold to connect the pipeline to each compressor cylinder. Such devices are often referred to as “pulsation bottles”, and typically resemble a vessel or expanded manifold. Examples of pulsation bottles are shown in Figure 11. Internal construction may be elaborate, often consisting of multiple chambers and communicating passages eventually leading to (or from) a single nozzle attaching it to the pipeline, e.g., as in the bottle being repaired in Figure 12.

Bottles are usually constructed much as vessels with the shell fabricated from vessel plate or from seamless pipe. Internal elements such as baffle plates and choke tubes are typically fillet welded to each other and to the inside wall of the pressure boundary. The internal elements and the shell of the bottle are directly exposed to pressure pulses associated with reciprocating operation, and thus may experience millions of small-amplitude stress cycles in a short period of

time. This type of loading spectrum makes the bottle durability or longevity highly dependent on a number of factors. Such factors include:

- Engineering Design – Some design concepts present greater inherent fatigue susceptibility. For example, placing a stiff structural element such as a baffle plate directly under any one nozzle of a multiple inlet bottle may cause the bottle to become highly loaded at that nozzle due to a fit-up gap at any one flange, leading to chronic fatigue of the nozzle joint as in Figure 13.
- Design Details – The use of T-joints made with fillet-welds, Figure 14(a), instead of beveled or grooved full-penetration welds with a fillet overlay, Figure 14(b), saves fabrication time and cost. However, use of this detail in high-cycle applications (pulsation or vibration) may be counterproductive in the long run because the nonpenetrating or partially-penetrating fillet weld is inferior in fatigue resistance.
- Workmanship Flaws – Seemingly minor workmanship flaws such as weld toe undercut or poor weld bead profile drastically shorten the fatigue resistance of even a properly-specified weld detail.
- Shop Procedures – Failure to perform post-weld heat treatment of welds, or failure to prevent warping of the shell during heat treatment, can lead to premature failure of nozzles due to excessive local stresses. Failure to discard and replace all bolts used as structural attachments of internals following stress-relief heat treatment of the bottle can lead to bolt failure, causing internal components to become inadequately anchored and leading to excess vibration and failure in adjacent welds.
- Installation – Installation of bottles with misaligned nozzles or large nozzle gaps introduces high loads at the affected and adjacent nozzles, increasing stressing in flange welds, outlet welds, and the shell and reducing the threshold for fatigue crack initiation.
- Operating Conditions – Failure to adequately anchor attached piping can lead to failures in girth welds or nozzle fillet welds. Boring out a compressor cylinder can cause previously reliable pulsation bottles to experience chronic fatigue failures due to the increased magnitude of pressure pulsations.

Failures in pulsation bottles usually occur in or originate from the welded connections between internal components, or between the bottle shell and internal or external attachments. Leakage will occur where the cracks propagate into and through the shell. Usually pulsation bottles are sufficiently low-stressed away from the local stress-raising detail that cracks remain leaks, however, leaks can pose a hazard within the compressor building, necessitating a unit shutdown and other actions. This normally does not pose a hazard off the compressor station site such that the public would be affected, but leads to costly downtime and bottle repair. Pulsation analysis programs are available for characterizing the pressure pulse frequencies and amplitudes within the suction and discharge piping of a compressor unit.

Structural Vibration

Structural vibration of pipe produces alternating longitudinal bending stresses across the pipe's circular cross-section, leading to fatigue of pipe girth welds and branch attachments (not to mention non-piping equipment such as instrument lines and compressor components, which are beyond the scope of this discussion). Structural vibration could occur in some situations involving above-ground pipeline segments, as described below.

Pulsation or Reciprocating Excitation

A significant proportion of reportable incidents that occur in compressor stations is associated with the high-cycle fatigue effects of vibration caused by gas pulsation or reciprocating action of compressors. A review of DOT-reportable incidents indicated 104 total reportable incidents occurred in compressor stations between 1985 and 2000. Of these, 13 occurred in the reciprocating compressor units and associated equipment (e.g., pulsation bottles), 2 were failures in piping welds, and 12 were failures in small threaded fittings and tubing, so 26% were almost certainly due to the effects of pulsation or vibration arising from reciprocating excitation.

Vibration effects are sometimes evident from visual observation, or from the occurrence of repeated failures of components or attachments. Fillet-welded assemblies and unbraced masses on cantilevers such as small valves teed off larger pipe runs may be particularly susceptible. No consistent standard or criterion for evaluating the severity of vibrations in piping is currently used in the industry, though criteria for evaluating vibration in piping do exist (e.g., Reference 12 for nuclear power plant piping). Piping affected by mechanical vibration issues can be instrumented to provide information (deflections, accelerations, and pipe stresses) that could be used in an engineering assessment such as the one given by Reference 12. Vibration effects are mitigated by adequately bracing the pipe (though not in a manner that interferes with flexibility required for thermal expansion), or controlling sources of vibration such as ineffective pulsation bottles. A process for the systematic application of this approach could be developed if a particular facility demonstrates the need.

Internal Flow-induced Vibration

The flow of gas inside piping can lead to vibration. Usually flow rates must be quite high (in excess of 40 ft/sec) for turbulent flow within the pipe to cause much vibration. However, even normal rates of flow can occasionally excite structures having low natural

frequencies, such as unbraced cantilevered masses. A typical cantilevered mass would be represented by a small valve teed off a larger header. An example of a vibration-induced failure in a 2-inch extruded outlet in a 30-inch header at the base of just such an arrangement is shown in Figure 15. The source of vibration in this case had to be gas flow, since this incident occurred at a turbine-powered compressor station where pulsation effects could not have been present, and the header was part of a main line valve bypass assembly, most of which was buried and not subjected to mechanical sources of vibration. Vibration effects are mitigated by adequately bracing the pipe (though not in a manner that interferes with flexibility required for thermal expansion), or controlling excessive gas flow rates.

Vortex Shedding

Pipe spans may be exposed to fluid currents flowing crosswise to them. The fluid affecting above-ground pipe spans is air (wind), while the fluid affecting submerged spans exposed in a river or offshore is water. Both wind and water are capable of inducing periodic vibration of exposed spans under specific circumstances, potentially leading to fatigue and failure at girth welds.

Any current flowing across a pipe produces a wake, however, wake conditions that can cause harmful vortex shedding occur only in specific flow regimes defined by the Reynolds Number, a nondimensional parameter. The Reynolds Number is computed as $Re = \rho VD / \mu = VD / \nu$, where ρ is the fluid density, V is the flow velocity, D is the pipe diameter, μ is the fluid dynamic viscosity, and ν is the fluid kinematic viscosity, all in consistent units. Organized vortex shedding occurs if the Reynolds Number falls outside the range 3.5×10^5 to 3.5×10^6 , as shown in Figure 16.[13] A momentary aerodynamic lift force is associated with the breaking-away of the vortex, leading to periodic positive and negative lift forces acting on the span. The magnitude and frequency of the alternating lift force can be estimated from the flow velocity and drag parameters.[14] The frequency of the vortex shedding can be estimated from the expression $f = St(V/D)$, where St is the Strouhal Number, another dimensionless parameter, equal to $St = 0.21 / C_D^{0.75}$, and C_D is the drag coefficient. The variation in C_D and St is also shown in Figure 16. If the frequency of vortex shedding is similar to the fundamental frequency of the pipe span or a multiple thereof, then resonance and steady state vibration or oscillation will ensue.

Vortex shedding in water has led to pipe failures in floods, river crossings, and offshore. It can be mitigated by installing intermediate supports along the spans (increasing their natural

frequency and reducing their dynamic response) or by burying the spans under an overburden that is resistant to scouring. Vortex shedding in high wind has led to pipe failures above ground, usually where the pipe is small in OD and encased in several inches of insulation, and support spacing is regular. It can be mitigated by introducing irregular support spacing (so that no single frequency dominates) or by attachment of passive aerodynamic devices (strakes) or active mechanical vibration absorbers.

Thermal Expansion Loads

Although pressure governs the basic design (wall thickness and specified grade) of pipelines and piping, the design and layout of above-ground piping systems must also consider the effects of thermal expansion or other imposed deflection that produces bending stresses in the pipe through flexure at changes in direction, branch points, and restraint points. Experience and analysis has shown that flexibility associated with piping layout strongly influences the magnitude of terminal reactions and the magnitude and distribution of bending moments throughout the piping system. Many different piping flexibility analysis techniques have been developed to enable the designer to assure that excess reactions at connected equipment, excess stress levels in the pipe, and leakage at flanged joints will not occur. The flexibility analysis estimates forces, moments, and stresses arising from thermal expansion or imposed displacement, and compares them to allowable limits.

Flexibility analysis reveals that bends, branches, and anchors are where the highest bending moments are likely to develop due to thermal expansion of the pipe. The geometry of piping components produces a complex elastic response to bending loads transmitted to the component through adjacent piping and anchors. This response usually results in local through-wall bending stresses and changes in cross-section within the component resulting in greater local flexibility and higher local stresses in the component than would occur in a piece of straight pipe of the same nominal dimensions subjected to the same applied loading. Large or frequent temperature changes associated with start-up and shut-down of processes may initiate fatigue in a piping bend or branch. This risk may be the most important factor in evaluating the suitability of a piping layout design in some types of process piping, independent of considerations for pressure capacity and support of deadweight loadings.

The problem of stresses in fittings and components subject to thermal expansion cycles cannot be successfully addressed without performing a flexibility analysis. The flexibility

analysis, providing a simplified fatigue evaluation of components, constitutes a vital step in the design of above-ground piping found in process and power facilities. Markl's seminal paper[15] explains the rationale for the flexibility analysis and acceptance criteria (allowable stresses) as established in the ASME B31 Code for Pressure Piping. Flexible above-ground piping systems are essential features of pipeline compressor stations, gas dehydration and processing plants, liquid product pump stations, and offshore platforms, and should be evaluated using methods appropriate to such systems, which is to say methods similar to those used in laying out process and power piping. Of course, flexibility is not an issue in a buried, restrained pipeline (with the possible exception of bends in hot oil pipelines).

Structural Discontinuities

It was noted earlier that the fatigue initiation life is shortened if the local stresses are increased by some type of geometric or metallurgical stress-concentrating feature. Stress-concentrating effects associated with common features of pipelines and piping systems include:

- abrupt weld toe geometries or weld bead profiles;
- large differences in metal strength between deposited and base metals leading to localized differences in strain rates under load, which focuses strain in lower-strength grain structures;
- local bending stresses in the pipe wall (referred to as “discontinuity stresses”) due to differences in radial expansion under pressure of adjacent thick and thin shell segments; and
- major structural features such as branch openings.

The vast majority of welds and structural features in common use provide acceptable service life when conventional standards for selection of design details and standards for welding quality are observed. However, occasionally some combinations of design details in combination with welding-related factors, all of which may be acceptable individually, sufficiently enhance local stresses to result in the initiation and propagation of fatigue cracks. Figure 17 shows a pair of fatigue cracks at the base of an unreinforced branch connection having an excessively sharp weld profile in the crotch of the branch. Other structural details that can occasionally give rise to fatigue include reinforcements or appurtenances that are much thicker than the carrier pipe wall and that are attached by fillet welds, including nonintegral reinforcements such as pads or sleeves, and self-reinforcing weld-on branch outlets (e.g., weld-on-let style fittings).

ASME B31.8 specifies that reinforcement pads, saddles, or sleeves that are thicker than the carrier pipe wall are to be tapered at their edges to the nominal pipe wall thickness, with fillet weld leg dimensions about equal to the pipe wall thickness. The reason for this is that tests and analysis have demonstrated that the juncture between drastic changes in thickness, and specifically between thick reinforcements and thin pipe walls, leads to stress concentration and susceptibility to fatigue.[16] While this edge tapering requirement results in a fillet weld throat that is less than the wall thickness, this is not a problem because the fillet weld on nonintegral reinforcement elements serves only as an attachment, not as a structural load-carrying weld.

Self-reinforcing weld-on fittings cannot be readily tapered because doing so would cut into metal required for area replacement of the branch hole in the header. Figure 18 shows several failures due to pressure-cycle fatigue adjacent to heavy-walled self-reinforcing branches attached to thin-walled header pipe. Although these particular failures occurred in liquids pipelines, the risk of fatigue or cycle-enhanced degradation (such as corrosion-fatigue or SCC) at such sites in gas pipelines is possible. Failures in which pressure cycles played a role have occurred at branches in gas pipelines. In some cases, it was known or suspected that other structural loadings on the branch aside from the thrust due to internal pressure were also present.

This is not to suggest that all hot taps or branches pose a high fatigue risk. Observing the good design and fabrication practices (e.g., those contained in the ASME B31 Code) greatly reduces the concern for fatigue. On the other hand, experience has shown that heavy partial- or full-encirclement reinforcements that have not been properly edge-tapered, and weld-on branch fittings that are larger than $\frac{1}{4}$ of the header diameter in headers that operate at greater than 50% of SMYS may be susceptible, particularly if some other adverse loading condition is present, such as vibration or localized settlement.

Wrinkle Bends and Miters

Some older pipelines were constructed using wrinkle bends or miter bends for making changes in direction of the pipeline. A wrinkle bend is a bend formed into a piece of initially straight pipe where the inside of the bend features one or more prominent wrinkles deliberately introduced as a means of foreshortening the inside meridian of the bend. A miter (or mitre) bend is fabricated by cutting straight pipe segments off at an angle and joining one or more such angled joints together to form a polygonal approximation of a bend. Neither wrinkle bends nor

miters are used in modern pipeline construction, although ASME B31.8 still permits them to be used in new construction if the pipeline operates at a hoop stress of less than 30% of SMYS.

All bends embody geometric characteristics that tend to concentrate stresses due to internal pressure or external bending moments, but wrinkle bends and miter bends contain more acute stress concentrating features than an analogous smooth bends. This could in principle make them more susceptible to fatigue from pressure cycles or thermal expansion cycles than smooth bends. Ordinarily, this would still not be expected to raise much of a concern because gas pipeline systems do not experience frequent pressure and temperature cycles. However, both types of bend may contain initial defects that reduce their tolerance for operational cycles: cracks due to excessive strains in the wrinkle bend, and workmanship defects in the miter bend welds. Capacity to tolerate operational cycles may be further reduced if the wrinkle bend or miter bend is capable of flexing in response to thermal expansion cycles as a result of being buried in an incompetent soil.

Most wrinkle bends and miter bends that are operated in low-stress gas pipelines are likely to be reliable, which is fortunate since hydrostatic testing or in-line inspection are unlikely to be effective for singling out those bends that could someday cause a problem. The reason for this is that wrinkle bends are more sensitive to longitudinal stresses imposed by external loadings than the hoop stress due to internal pressure. Older pipelines containing wrinkle bends have experienced failures of the bend, particularly in areas of soil movement. When such bends fail, it is typically due to the effects of external loadings rather than as a result of internal pressure, per se. Fatigue arising from fluctuations in operating pressure and temperature may play a role in the observed sensitivity of some older wrinkle bends to axial loads, as might thermal movement of sections of pipe following excavation. This remains a subject of study in the industry. Where feasible, ILI can be useful in that geometry tools would indicate the location of wrinkle bends and the magnitude of wrinkles.

A way to assess the condition of a given bend is to excavate it and perform nondestructive examination for cracks in the wrinkle or weld defects in the miter. The excavation destroys the consolidation and support for the bend, so any bend that is not cut out and replaced with a modern smooth bend should probably be backfilled with a flowable fill that can fully restrain the bend in place. Therefore, it makes little sense to excavate and mitigate

bends where there is no history of problems and no evidence of unusual external loadings being present. Probably only those bends residing in a pipeline segment that has exhibited problems with similar bends on prior occasions or that is located in a high consequence area would warrant excavation and inspection.

Dents and Mechanical Damage

The term “dent” describes a permanent deformation of a pipe’s circular cross-section caused by external forces. The curvature of the pipe wall within the dent may be reduced, flattened, or reversed. A dent that has no scrapes, gouges, or other stress-concentrating features present in conjunction with it is referred to as a “plain dent”. Dents caused by the installation of a pipeline on rocks in the ditch are usually plain in nature; dents caused by excavating equipment or other machinery striking a pipeline typically are not plain.

A dent that is prevented by the soil from pushing out (rerounding) under the influence of internal pressure is a “constrained dent”. Rock-induced dents are typically constrained (unless the pipeline is excavated). A dent that is free to push out under the influence of internal pressure is unconstrained. Dents caused by excavating equipment typically are unconstrained. They reround as the indenting equipment is withdrawn from the pipe surface. Once a dent is excavated so that it can be examined, it is unconstrained regardless of whether it was constrained or unconstrained prior to excavation.

The term “mechanical damage” refers to features such as gouges, scrapes, or crushed metal introduced by contact from excavating or other mechanical equipment. Mechanical damage typically exhibits one or more of the following features:

- visible scrape, gouge, or smeared metal;
- localized metal loss or reduced wall thickness not due to corrosion;
- cracking within a scrape or gouge; and
- creasing of the pipe wall, or a long narrow indentation.

The features listed above occurring in conjunction with pipe indentations are referred to by 49 CFR 192 as “stress concentrating features”. In most occurrences of mechanical damage, the pipe undergoes indentation simultaneously with gouging of the metal surface. The indentation pushes out (“rerounds”) under the influence of internal pipe pressure as the damaging object withdraws

from the pipe surface. The extent of rerounding depends on the pipe wall thickness, the pressure in the pipe, and the initial dent geometry.

Tests have demonstrated that unrestrained plain dents having residual depths of 2% of the pipe diameter or less, which is about what one would expect for an unrestrained dent after rerounding, exhibited fatigue lives between 10^5 and 10^6 cycles of pressure producing hoop stress levels between 36% and 72% of SMYS. This would likely be equivalent to infinite life for most if not all gas pipelines. In fact, a fatigue failure in a plain shallow dent has never been reported to have occurred in gas pipelines to the authors' knowledge. As shown earlier, liquid pipelines operate with pressure spectra that are far more severe than those of most gas pipelines, so pressure fatigue in a plain dent may be possible in some liquid pipelines. This is supported by service experience where fatigue cracks and leaks have occurred in liquid pipelines affected by plain rock dents a few years after the rock has been removed.

Dents that are restrained, which would be the case for rock dents where the pipe has not been excavated, have at least an order of magnitude greater fatigue life than unrestrained dents of the same size and shape. In one series of tests, constrained dents as deep as 18% of the pipe diameter survived hundreds of thousands of pressure cycles between 36% and 72% of SMYS without failure.[17]

An important exception to the above findings is restrained rock-induced deformations having two closely spaced centers of indentation with a flattened or saddle-shaped area between them. It has been observed in service and in tests[18] that the flattened area between the two dent centers is susceptible to pressure-cycle fatigue, because it is effectively unrestrained and flexes readily in response to pressure cycles. As with other pressure-cycle fatigue situations discussed already, incidents of this nature have been confirmed only in liquid pipelines.

The presence of metal damage caused by excavator teeth and the like changes the picture completely. Such damaged metal would be susceptible to very low-cycle fatigue, which means that a few cycles of high strain will cause a failure. Two mechanisms for the accumulation of high strains is creep-like rerounding at high levels of sustained pressure, and shakedown of the dent to elastic action over several pressure cycles. These mechanisms enable plastic strain, which could be damaging to a strain-sensitive material, to continue to accumulate at the root of the damage even with the steady pressures and infrequent large cycles that characterize gas pipeline operations. This seems to adequately explain how pressure reversals and delayed

failures have occurred in mechanical damage in gas pipelines that do not experience a large number of pressure cycles (e.g., the Edison, NJ incident). What this means is that mechanical damage constitutes a threat to any buried gas pipeline irrespective of considerations for pressure cycles, hence pressure cycles would not constitute a condition or basis for choosing to assess a pipeline for the mechanical damage threat.

Tests conducted for PRCI involving severe dents affecting girth welds showed reduced cycle lives compared to dents in plain pipe, but nearly as good fatigue resistance in modern, high-frequency-welded ERW seams affected by plain dents as similarly dented plain pipe.[19] This outcome seems reasonable considering that high-frequency ERW seams have been shown to have comparable fatigue properties to base metal if the seams are of good initial quality.[10] Although severe dents are detrimental to welds, it appears that plain dents of moderate magnitude affecting girth welds of good quality are not a problem in gas pipelines. Subsequent tests conducted for API [18] demonstrated that dents affecting girth welds, ERW seams, and DSAW seams all exhibited similar fatigue performance as a group. Relative to plain dents in sound pipe, as a group they tended to fail in about a half-order of magnitude fewer cycles. Thus, for shallow dents having a depth of 2% of the OD or less, test results showed fatigue lives between 50,000 and 500,000 cycles of pressure equivalent to nominal hoop stress levels between 36 and 72% of SMYS. This is more than adequate to assure that fatigue failures in shallow dents affecting otherwise sound welds will not occur within the foreseeable life of most gas pipelines. Indefinite fatigue life of large dents affecting welds in gas pipelines, and small or large dents affecting welds in liquid pipelines, cannot be assured however.[20]

It should be noted that all of the tests discussed above were conducted on pipe having nominally sound, ductile welds. The conclusions derived from those tests may not necessarily extend safely to older low-frequency ERW seams or other seam types, acetylene girth welds or severely flawed electric arc girth welds that could potentially be susceptible to brittle fracture.

A simple methodology for assessing the severity of mechanical damage has not yet been developed due to the complexity of the mechanical damage process and interaction of a large number of variables. However, it appears that one factor distinguishing mechanical damage that results in a delayed failure from benign mechanical damage is the tendency to develop fatigue in the damaged material. The groundwork has been laid for performing residual life estimates of mechanical damage based in part on an incremental fatigue crack-growth analysis.[23] Such

analysis is still somewhat experimental since methods for accounting for the initial damage severity, effect of indentation geometry, and other parameters have not been reduced to routine or standardized techniques.

Environmental Cracking

An environmentally-induced crack may become enlarged by fatigue. Stress-corrosion cracking (SCC) is a form of environmentally assisted cracking that occasionally can occur in line pipe steels under known conditions. Moderate to high tensile stresses are required, as is some cycling of the stress in order to reexpose the crack-tip to the adverse environment. When SCC occurs in a liquids pipeline, pressure-cycle fatigue usually can be expected to take over as the mechanism for crack enlargement from the original stress-corrosion crack, after a time. In some rare instances, this appears to have occurred in gas pipelines.

Figure 19 shows a fatigue crack that extended from a small stress-corrosion crack at the weld toe adjacent to a reinforcement saddle for a 16-inch branch on a 26-inch OD natural gas pipeline. The coating around the branch was wax, which can shield the pipe from effective CP if it becomes disbonded from the pipe surface. The pipeline operated at 45% of SMYS and experienced a pressure operation not as severe as what is shown in Figure 4. Although the local stresses in the pipe wall at the “hillside” location adjacent to a reinforcement saddle are enhanced by a stress-concentrating effect (refer to the discussion on structural discontinuities), the saddle assembly would ordinarily have been expected to tolerate on the order of 22,000 cycles based on pressure-fatigue tests of similar configurations and, therefore, would not have been expected to have a service life limited by fatigue.[21]

Figure 20 shows a cross-section of a DSAW longitudinal seam in a 36-inch natural gas pipeline. The seam contains a fatigue crack that extended from a stress-corrosion crack that originated at the toe of the seam. The coating was asphalt and the affected area was positioned under a concrete river weight in a peat bog. The pipeline operated at 69% of SMYS. Note that the outer seam bead was deposited far enough off-center to expose part of the welder tracking groove and produce an acute weld toe. The bending moment couple established by the offset bead, combined with the abrupt toe geometry, significantly concentrated the local stresses compared to a normal DSAW seam profile, enhancing susceptibility to both SCC and the subsequent fatigue.

In both of these cases involving natural gas transmission mainlines, the effects of pressure cycles promoted the occurrence and enlargement of cracking in service. However, as is the case with enlargement of cracks present in mechanical damage, other factors had to first be present for this to occur. Therefore, considerations for pressure cycling would not be the primary basis for either assessing the risk of SCC or for developing a mitigation plan for that risk.

Hydrogen-induced cracking (HIC) is associated with the environment present during welding, under certain conditions. The cracking results from hydrogen produced by the use of a cellulosic (non-low-hydrogen) welding process being trapped in the weld, in conjunction with base materials having sufficient chemistry content to produce a heat-affected zone having a hard, coarse-grained microstructure when exposed to a high weld-cooling rate. Like SCC, HIC can serve as an initiation site for crack growth in service due to fatigue. The application of appropriate production and in-service welding procedures substantially reduces the risk of HIC and subsequent concern for flaw extension in service.

IMPLICATIONS FOR INTEGRITY MANAGEMENT PLANNING

Keeping the Fatigue Threat in Perspective

The foregoing discussion described a few situations in which fatigue can present a threat to the integrity of gas pipelines or related systems, under some circumstances. It is important to keep this threat in perspective with the overall set of threats that could affect gas pipelines. Encroachment damage and corrosion account for a majority of significant gas pipeline incidents. Fatigue does not play a causal role in those incidents in that the root cause was the pipeline becoming damaged, and without which no fatigue would occur. (Low-cycle fatigue in the damaged material may be a key factor distinguishing mechanical damage that causes a delayed failure and benign mechanical damage, however.) The balance of incident causes consists of natural events, incorrect operation, pipe and seam defects, fabrication defects, and non-pipe equipment failures. Fatigue does not play a role in pipe and seam defect incidents in gas pipelines, and it would not be expected to be a factor in operator error. The remaining categories account for not much more than 20% of reportable incidents, and fatigue is not their only potential root cause. Even within compressor stations, which are the gas facilities most

susceptible to fatigue effects, only about one-quarter of reportable incidents appear to be attributable to fatigue. So fatigue is involved in a relatively small proportion of reportable or significant incidents in gas pipelines.

Integrity Management Planning Requirements

Paragraph 192.917(e)(2) states the following:

“(2) Cyclic fatigue. An operator must evaluate whether cyclic fatigue or other loading condition (including ground movement, suspension bridge condition) could lead to a failure of a deformation, including a dent or gouge, or other defect in the covered segment. An evaluation must assume the presence of threats in the covered segment that could be exacerbated by cyclic fatigue. An operator must use the results from the evaluation together with the criteria used to evaluate the significance of this threat to the covered segment to prioritize the baseline assessment.”

The preamble to the original regulation introducing Subpart O and the above paragraph states:

“Particular threats. The rule requires that an operator take specific actions to address particular threats the operator has identified. Those threats, and the required actions, are for third-party damage, cyclic fatigue, manufacturing and construction defects, ERW or lap welded pipe, and corrosion. These threats have been identified for specific action because of their significance to pipeline integrity and because the operational characteristics of gas transmission pipelines dictate that they be treated uniquely. The primary difference in the operation of gas transmission pipeline related to these defects is the absence of significant pressure cycling and the associated absence of the cyclic fatigue driving force for crack growth. The absence of significant cyclic fatigue implies that the failure of pipelines from these threats has unique causes that need to be address in an integrity management program for gas transmission pipelines.”

The authors of the regulation evidently recognized that the operating pressure cycles in gas pipelines do not constitute the degree of threat seen in certain liquid pipelines, particularly with respect to longitudinal seams. Therefore, this clause of the regulation must be concerned about issues other than pressure-cycle effects on long seams. Clues are given by the fact that 192.917(e)(2) refers to “other loading conditions” (although not all examples it cites are cyclical in nature) and refers to deformations or other defects. Figure 21 presents an example of a simple framework for addressing some of the concerns raised in 192.917(e)(2). (The authors and distributors of this report point out that this process may or may not be suitable for all gas pipeline systems. It has not been reviewed or approved by the Office of Pipeline Safety. Incorporation of this process into an Integrity Management Plan does not assure compliance with

regulatory provisions. It has been included herein to illustrate a systematic approach to addressing some of the provisions and threat components listed in 192.917(e)(2).)

The first box establishes criteria for identifying whether pressure cycles could affect a dent. It was established earlier that pressure cycles do not present much of a concern with the majority of dents in gas pipelines. The criterion requires that a pipeline operate at 50% or more of SMYS and experience pressure cycles (peak to valley magnitude) equal to 80% of the MAOP on a daily basis in order for pressure cycles to be a concern with dents. This would exclude the vast majority of gas pipelines, but if the result is positive, then a deformation assessment (by ILI) would be warranted. Obviously, there are likely to be other and better reasons for performing a deformation assessment on many pipelines, so this process would lead to very few assessments for the presence of deformations solely on the basis of pressure cycles during operation, or assessments for deformations that would not be performed anyway.

The process then investigates for susceptibility of buckles or wrinkles, and subsequently girth welds and branch connections, to potentially threatening loading scenarios other than pressure cycles. Some of the loadings are cyclical in nature and some are not. If adverse conditions are present, then an assessment for the presence of such features is warranted. Methods of assessment may be other than the standard methods of hydrostatic testing, in-line inspection, or direct assessment. For example, an assessment for buckles might consist of ILI to detect the presence of a buckle, followed by excavation and nondestructive examination of the buckle to verify an absence of cracks. An assessment of welds on a span across a ditch might consist of an engineering analysis to determine whether the span could be submerged by flood level water flow and for susceptibility to vortex shedding, and if appropriate, radiography of welds not previously inspected at the time of construction.

Situations that apply primarily in reciprocating compressor stations such as pulsation bottles and mechanical vibration were not included because the majority of compressor stations are in relatively sparsely-populated areas, most compressor buildings are equipped to vent gas or detect and combat fires, and most compressor stations have generous buffer areas that would reduce the hazard to the public in the event of a fire in the compressor building. Also, many compressor facilities are designed to accommodate cyclic service, and compressor station piping is limited to operating stresses of 50% of SMYS or less.

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FIGURES

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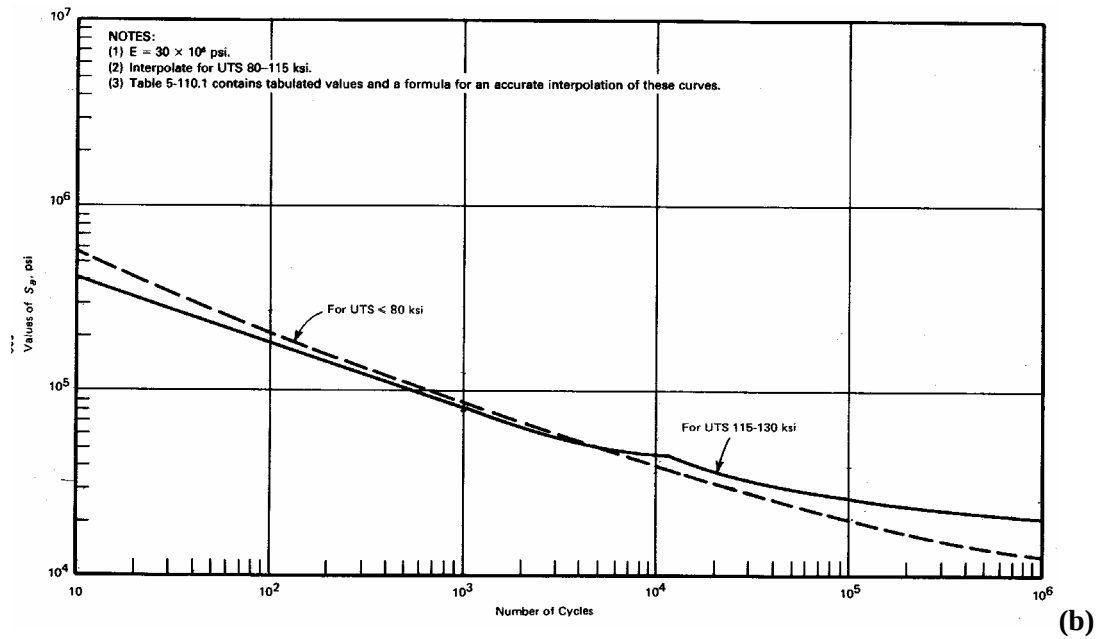
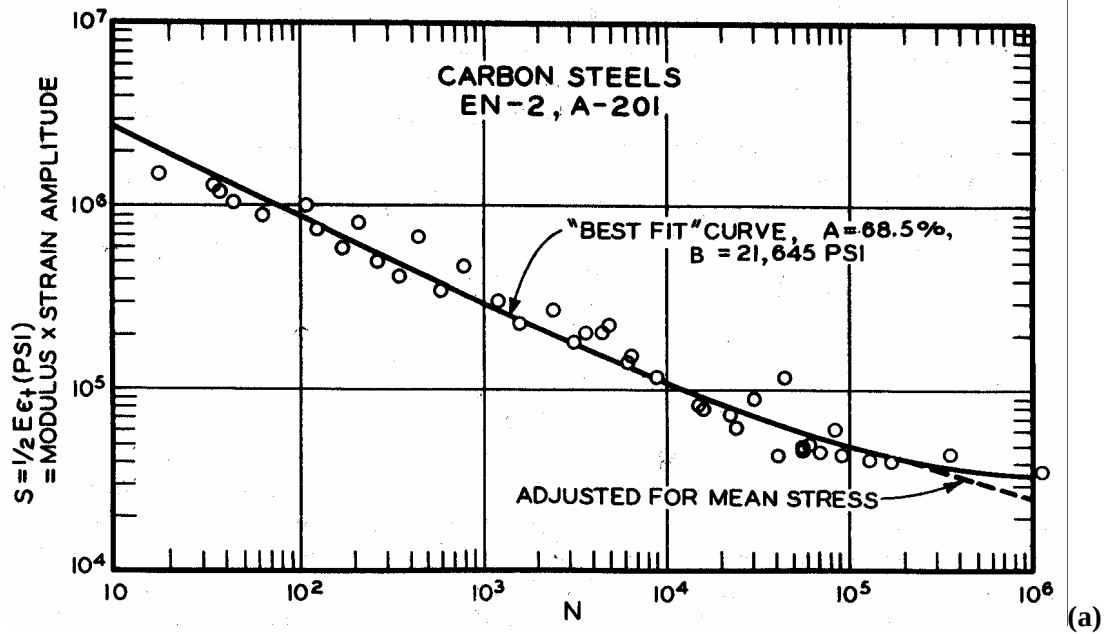


Figure 1 – ASME Fatigue S-N Data and Design Curve

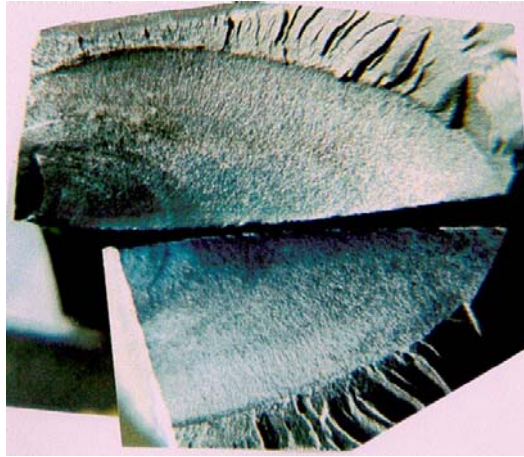


Figure 2 – Typical Fatigue Failure Fracture Surface

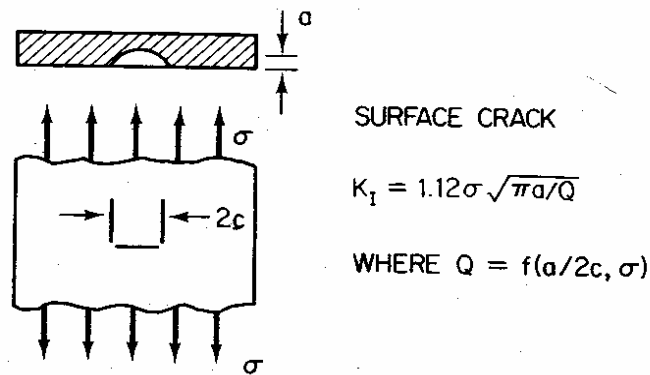


Figure 3 – Simplified Crack Model

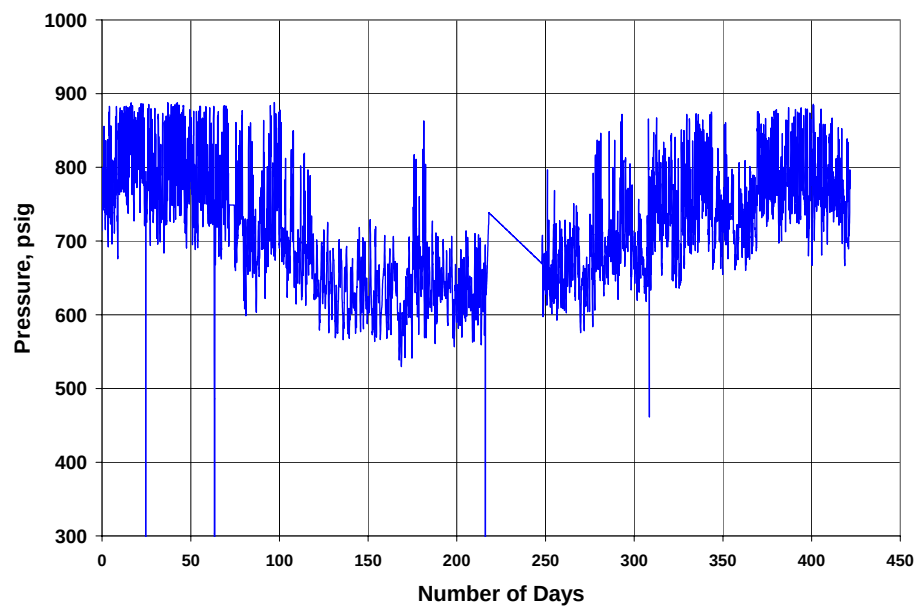


Figure 4 – Aggressive Gas Pipeline Operating Spectrum

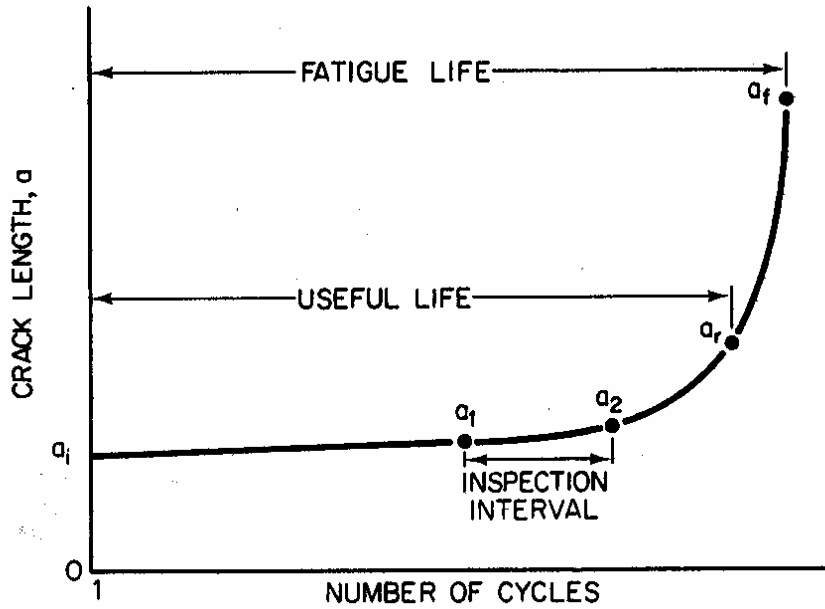


Figure 5 – Crack Propagation in Service

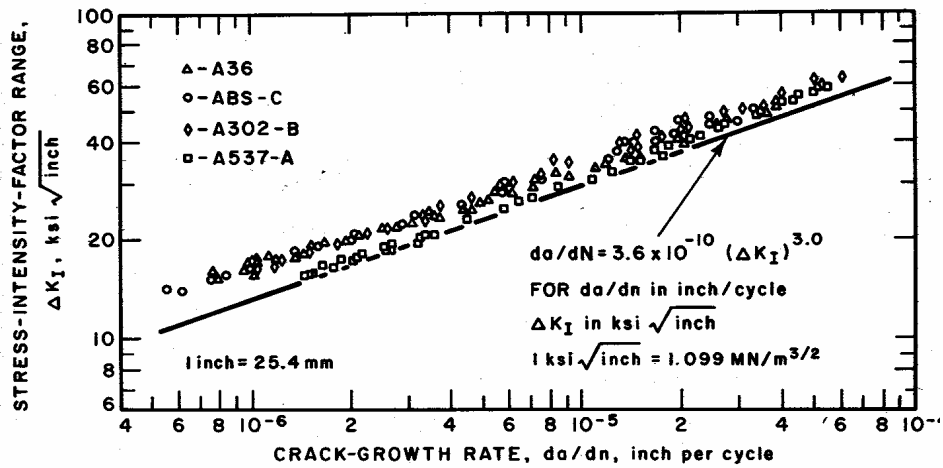
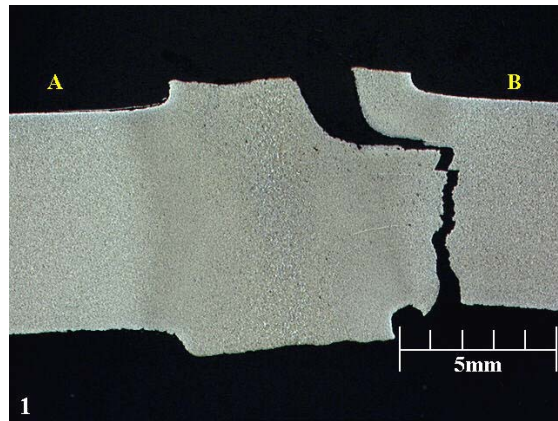


Figure 6 – Some Typical Crack Growth Rate Data for Carbon Steel

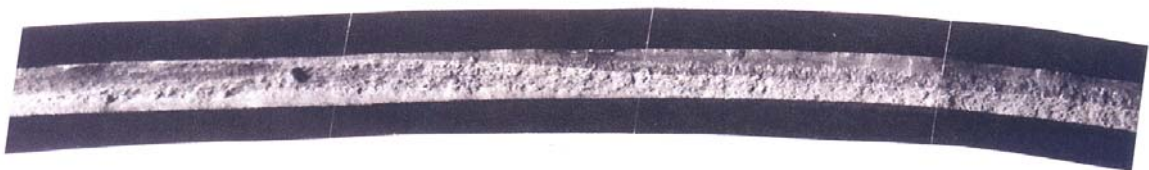


(a)

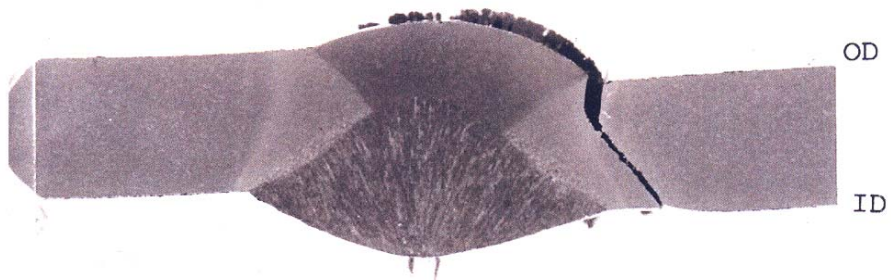


(b)

Figure 7 – Fatigue Crack Originating from ERW Seam Hook Crack



(a)



(b)

Figure 8 – Rail Shipment Fatigue Crack at Toe of DSAW Seam

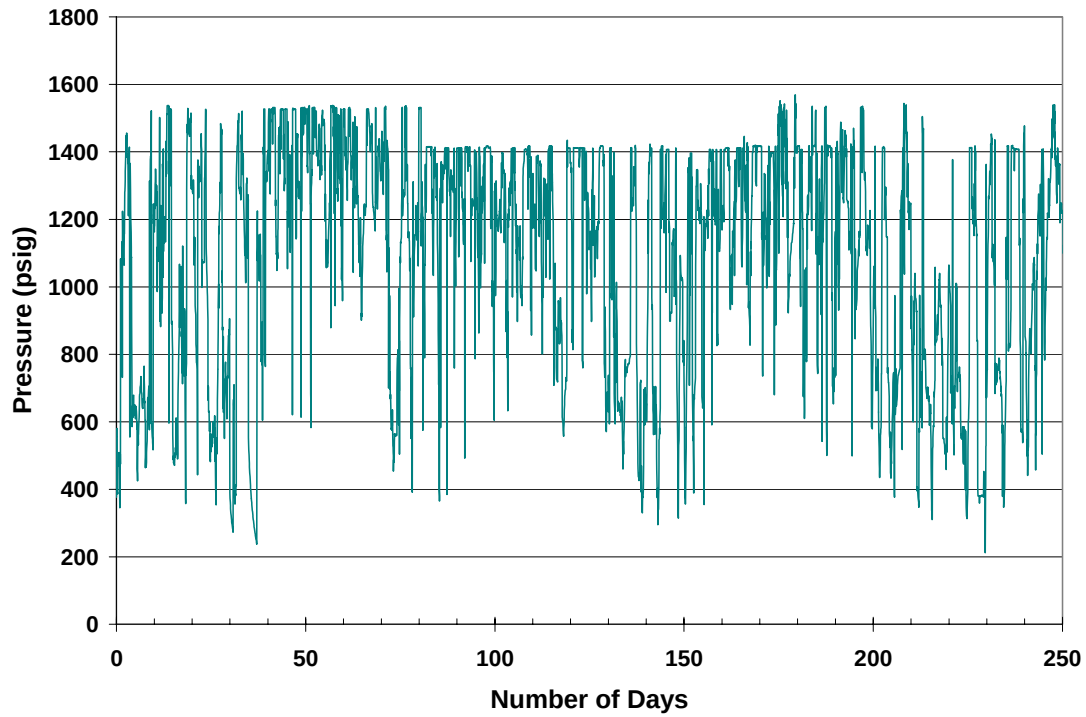


Figure 9 – Aggressive Liquid Pipeline Operating Spectrum

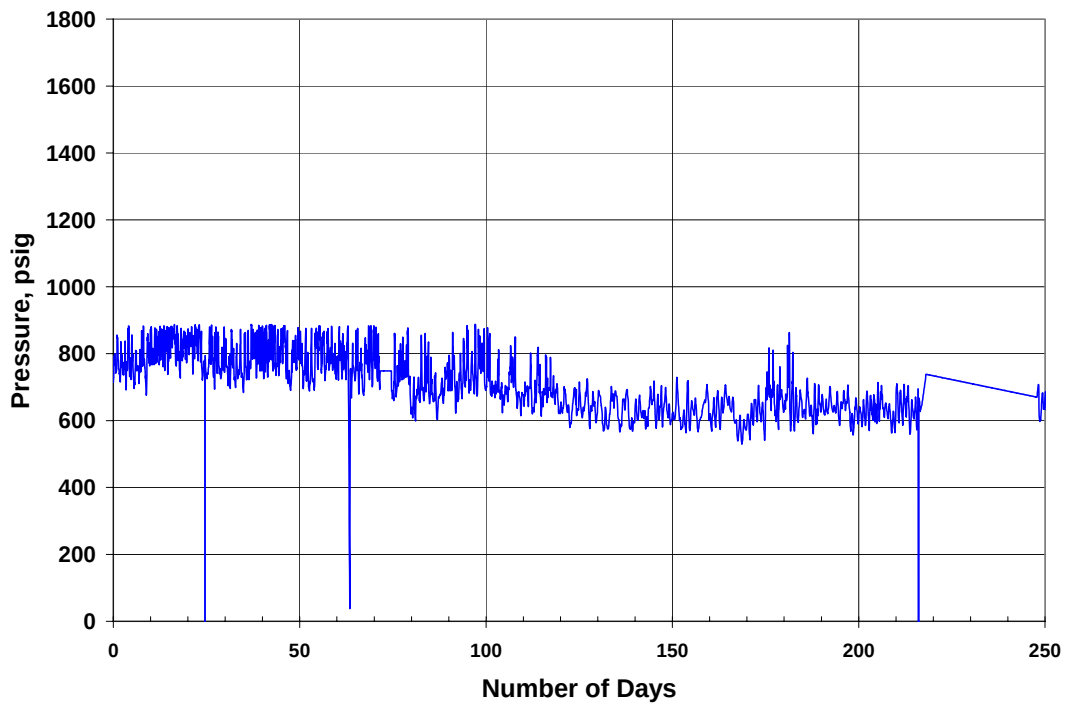


Figure 10 – Gas Pipeline Operating Spectrum (from Fig. 4, Rescaled)

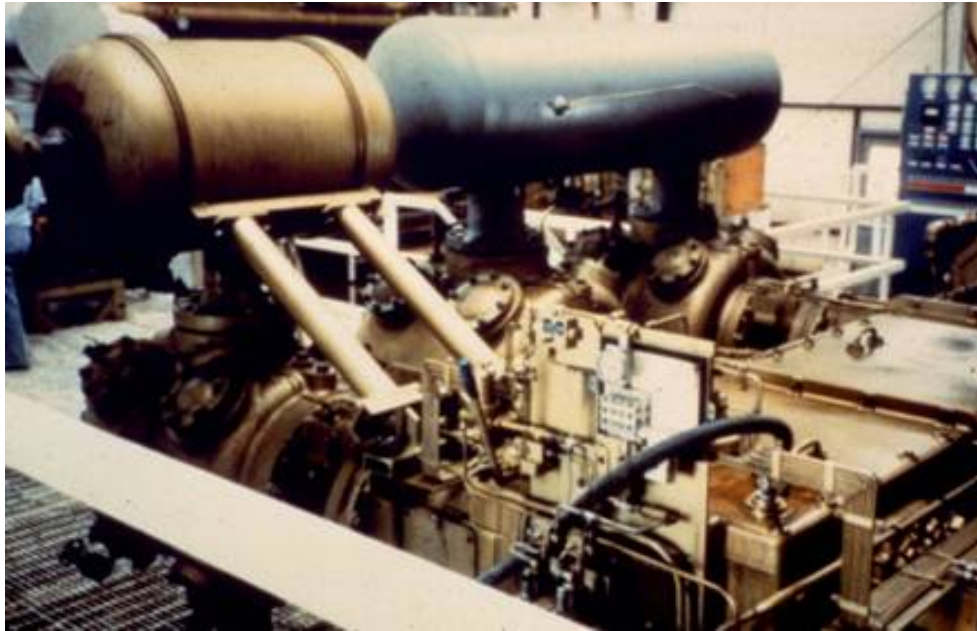


Figure 11 – Pulsation Bottles

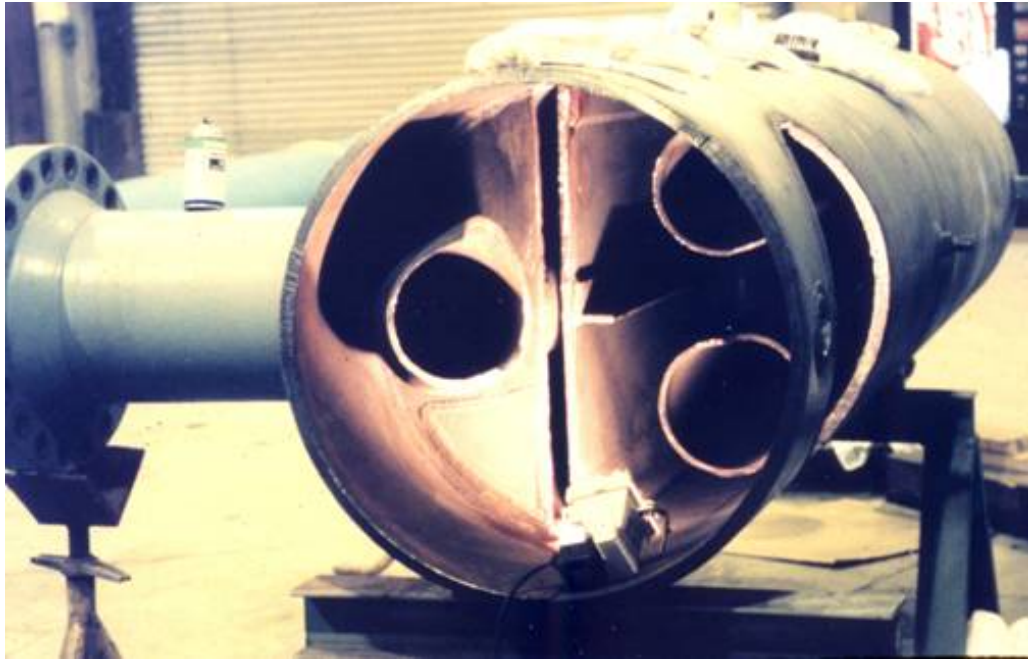


Figure 12 – Pulsation Bottle Internals Undergoing Repair



Figure 13 – Bottle Shell Crack Due to Design Fault

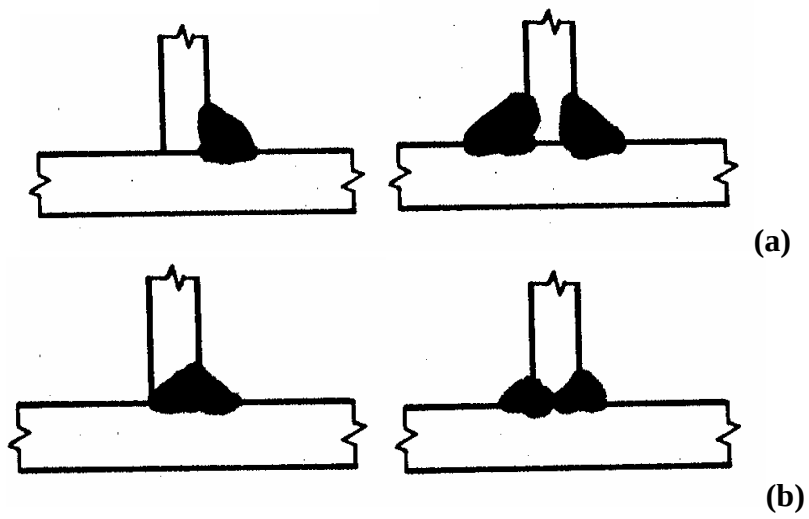


Figure 14 – T-Joint Weld Details

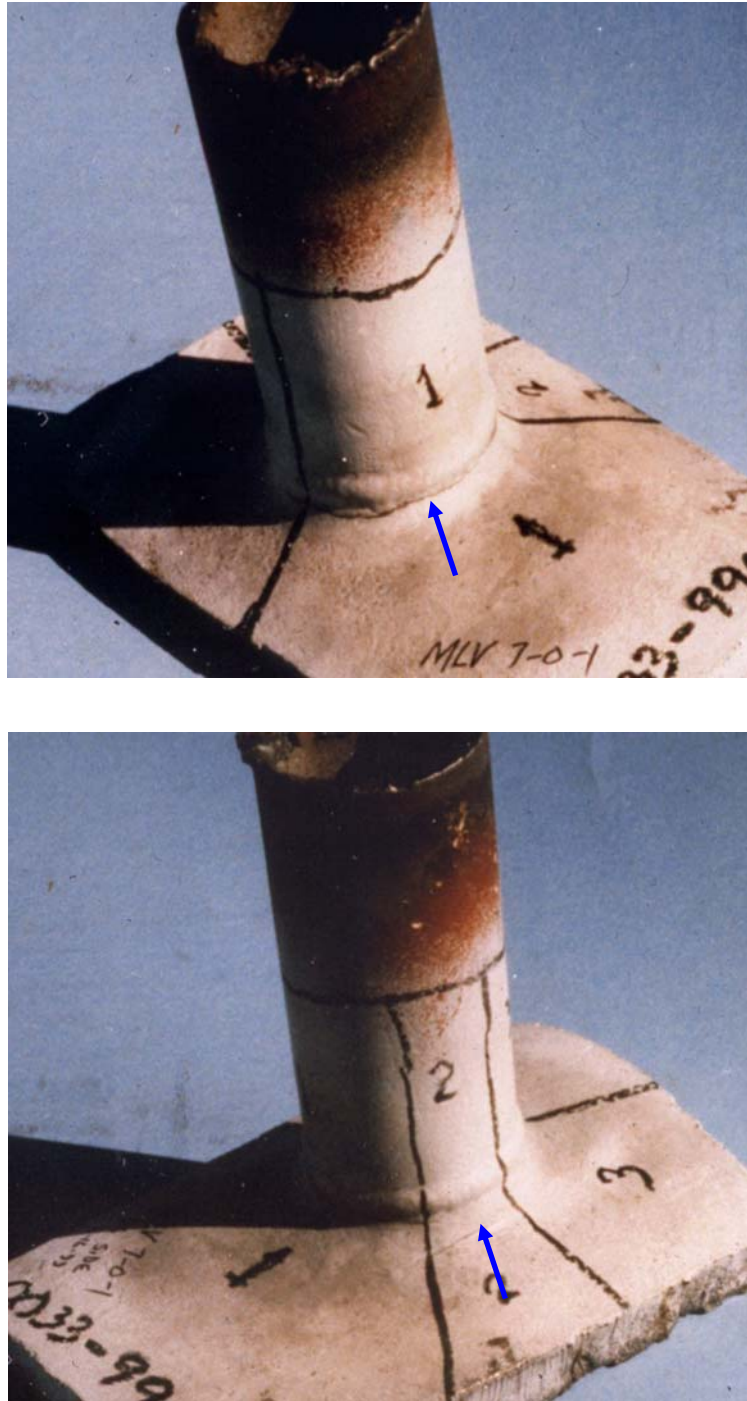


Figure 15 – Structural Vibration Failure in Gas Pipeline Extruded Outlet

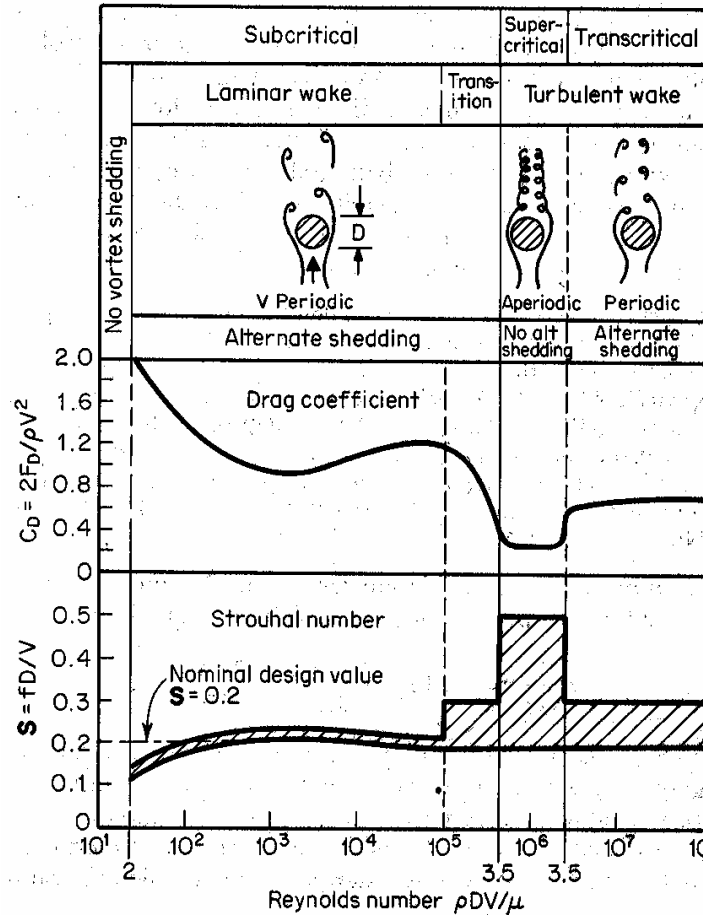


Figure 16 – Flow Regimes Susceptible to Vortex Shedding

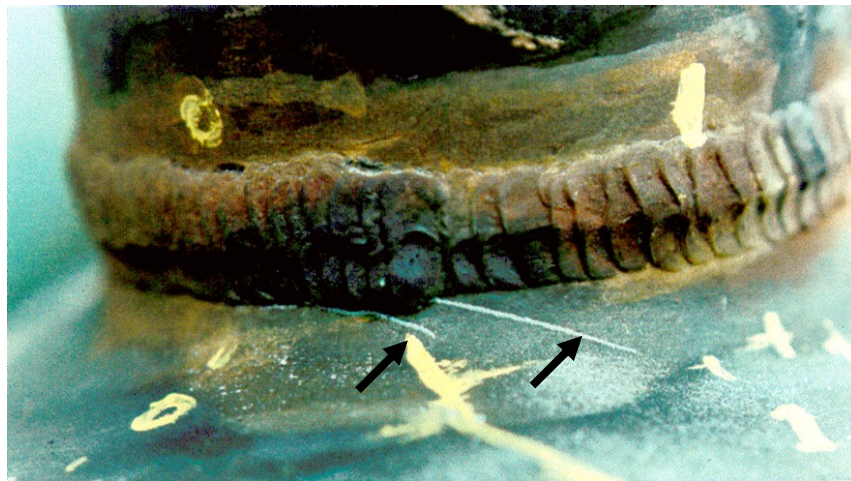


Figure 17 – Fatigue Cracks at Poor Weld Profile on Branch Connection

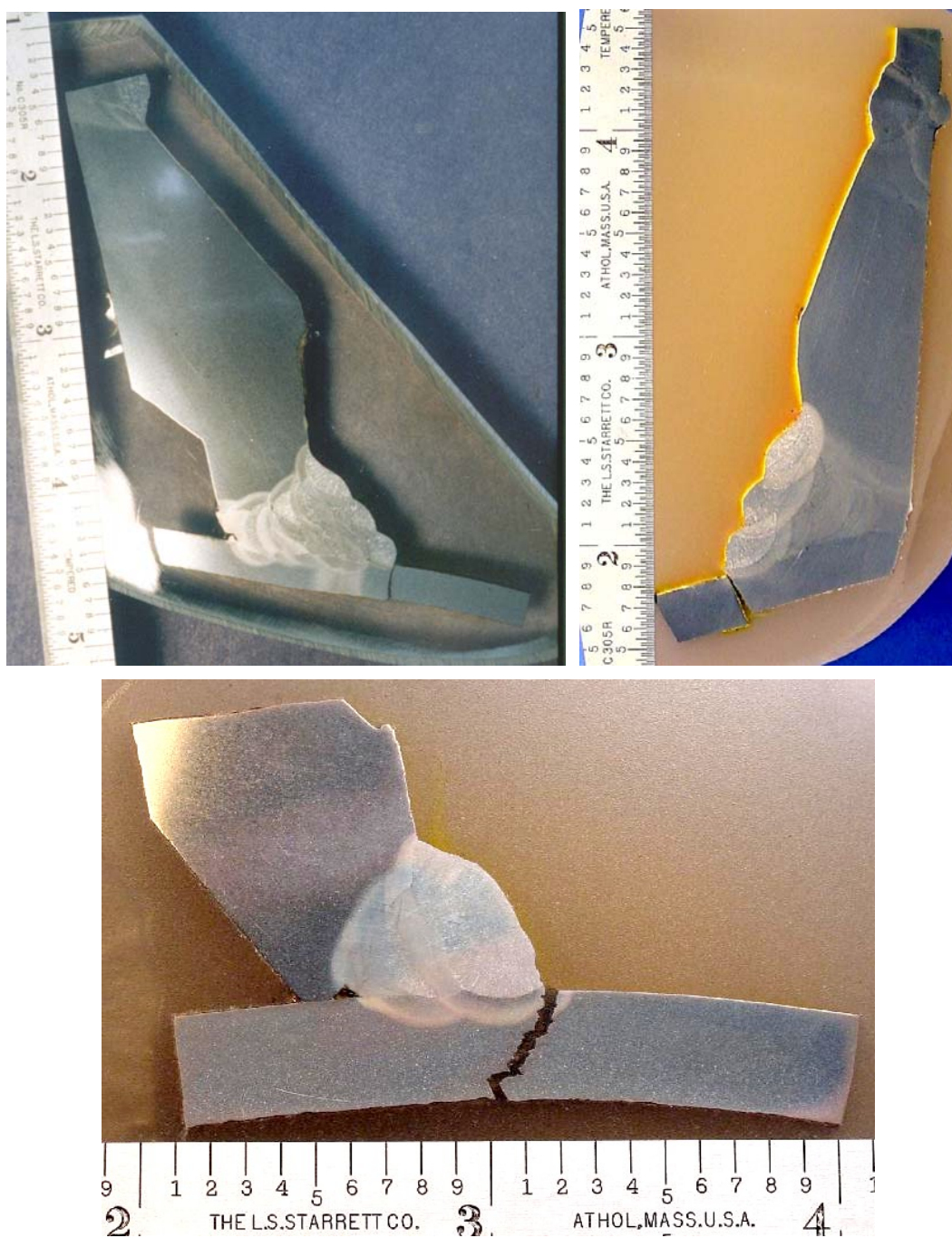


Figure 18 – Example Fatigue Failures Adjacent to Heavy-Walled Attachments

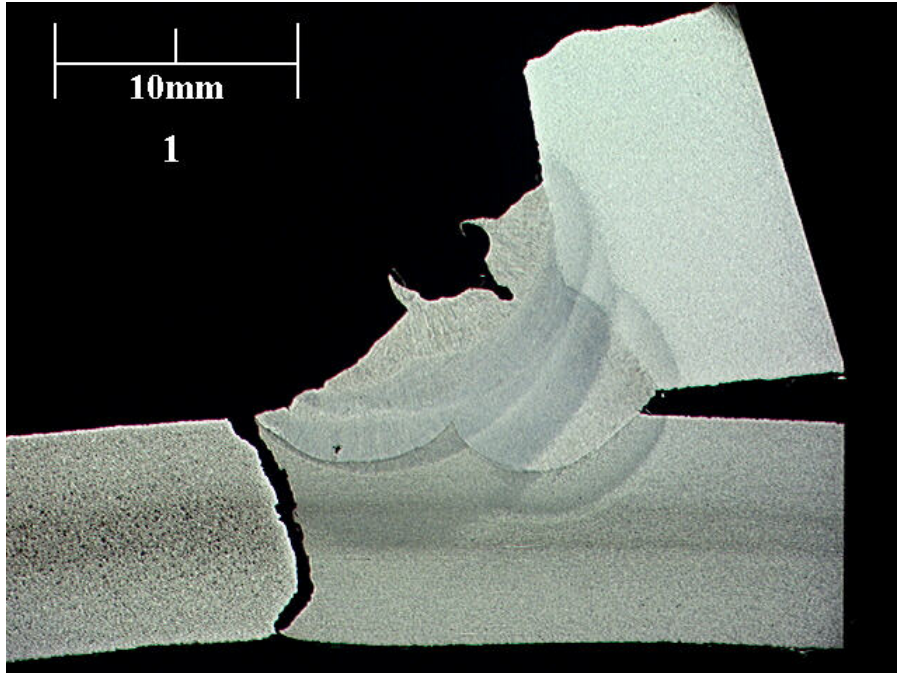


Figure 19 – Fatigue Crack Extending from SCC at Toe of Saddle Weld

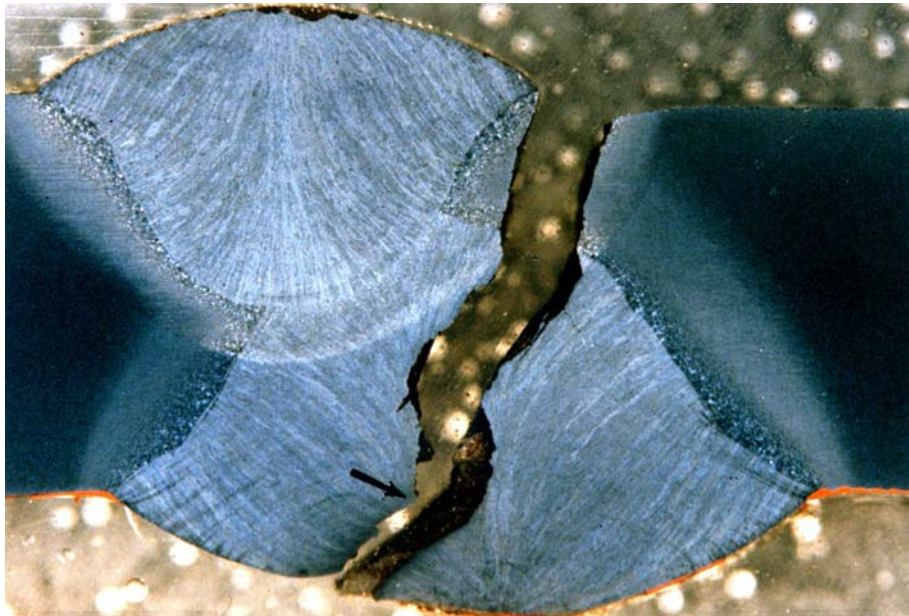


Figure 20 – Fatigue Crack Extending from SCC at Toe of DSAW Seam

Section 192.917(e)(2) "Cyclic Fatigue" Process

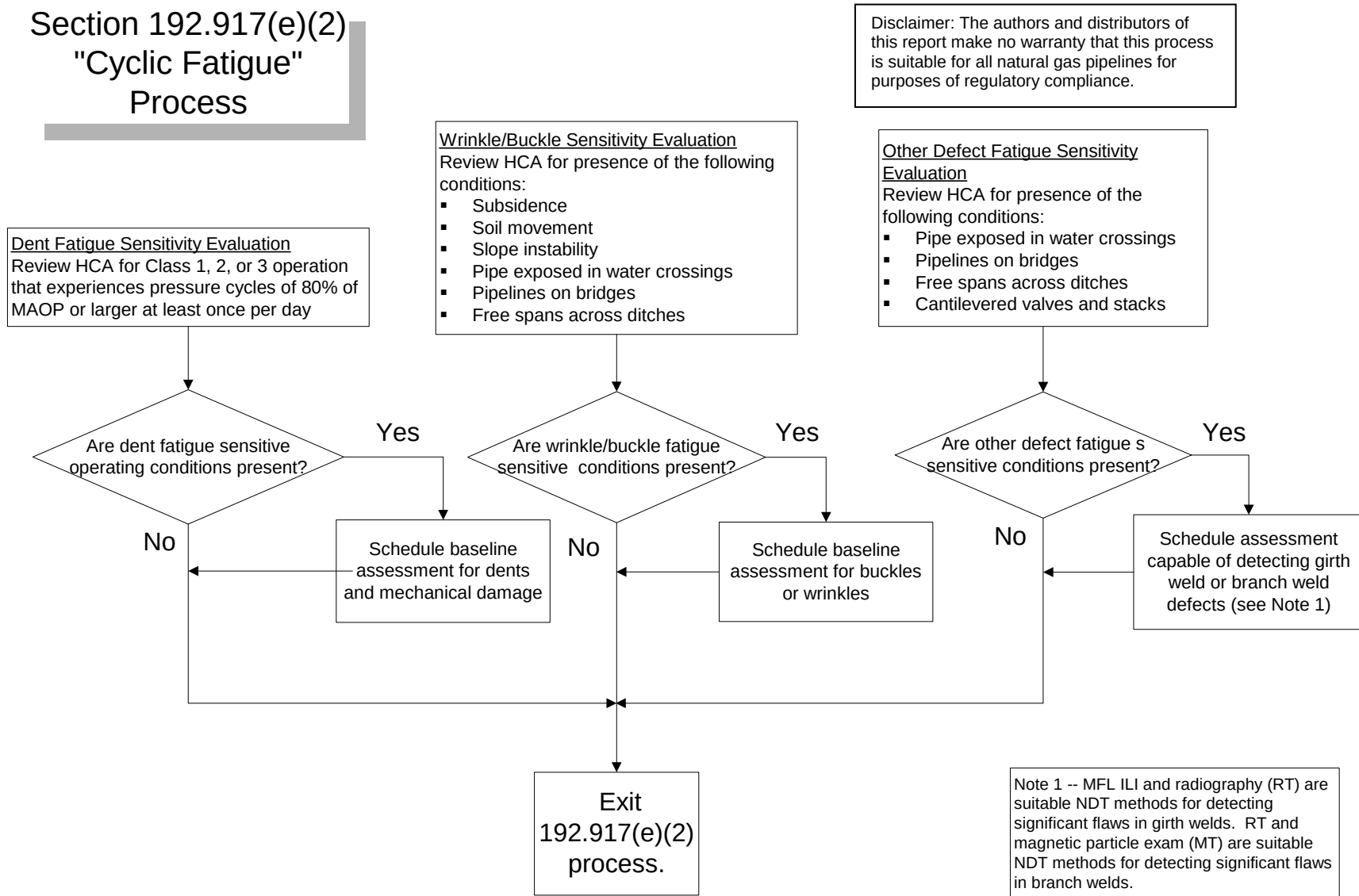


Figure 21 – Example Process for Addressing 192.917(e)(2)

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