Fatigue Design of Process Equipment

ASME Plant Engineering & Maintenance Technical Chapter

March 12, 2009

Chris Hinnant

chris@paulin.com

Paulin Research Group

Houston, TX

If you would like an electronic version, these presentation slides may be downloaded at:

http://paulin.com/Library/Fatigue_Info/Hinnant_ASME_Plant_Engineering_Presentation.PDF

You may also be interested in the following technical presentations and white papers available at:

http://www.paulin.com/TechnicalArticles.aspx

Experimental Evaluation of the Markl Fatigue Methods and ASME Piping Stress Intensification Factors (July 2008)

This document includes the slides presented at the 2008 PVP Conference in Chicago, Illinois USA. The presentation summarizes the methodology used in experimental fatigue tests that included girth butt welds and unreinforced tees. Also, 600 butt weld data were gathered and compared to the Markl girth mean butt weld fatigue curve. Read more...

Markl, SIFs, and ASME VIII-2 Fatigue Design (January 2008)

This brief article addresses the question of "What are SIFs, where did they come from, and how do they relate to ASME Section VIII-2 fatigue designs?" Read more...

Fatigue Testing of Welded Flat Head Pressure Vessel Joints (July 2007)

Presentation slides of continuing fatigue analysis research conducted at Paulin Research Group. The work addresses effects of thickness on fatigue life and addresses concerns of contained fluid on testing methodology. Read more...

More technical literature available at:

http://www.paulin.com/TechnicalArticles.aspx

Regression Analysis of Markl Girth Butt Weld Data (May 2007)

This article reports the results of regression analysis of data reported by A. R. C. Markl in 1952. Read more...

Items for Discussion Related to the Div 2 Rewrite Master SN Method (February 2007)

Presentation slides of work presented to rewrite of ASME Section VIII, Division 2 fatigue rules. The work concentrates on application of previous work by Neuber on correlation rules. Read more...

Observations Related to the Master SN Method in the ASME Division 2 Rewrite Project (February 2007)

The report addresses specific issues related to the implementation of the Master SN method in the ASME Division 2 Rewrite project. These issues illustrate situations where the implementation can lead to non-conservative solutions. In addition, comparisons against other fatigue methods including European as-welded codes is achieved by using fatigue charts based on specified load instead of stress. These charts show that the proposed Div 2 Rewrite SSM method will produce lower lives for high quality weld joints while allowing greater lives for low quality weld joints in comparison to the existing ASME FSRF method. In addition, Neuber's rule is discussed in terms of Code stress limits and analysis methods to illustrate why it is not required in the Div 2 Rewrite project. Read more...

More technical literature available at:

http://www.paulin.com/TechnicalArticles.aspx

Review of the Master SN Neuber Rule in the ASME Division 2 Rewrite Project (February 2007)

*The goal of this discussion is to illustrate that when fatigue design curves are developed using databases comprised of welded or notched fatigue specimens (not smooth bars), there is no need to apply plasticity correction factors as part of the design process. Such a conclusion is certainly warranted for the pseudo elastic stresses calculated from displacement controlled conditions. ASME primary and secondary stress limits prevent load controlled conditions (primary type loads) from achieving stress ranges where Neuber's would be required (stress ranges greater than 2*Sy) and is therefore not a significant part of this discussion.* Read more...

Update on Fatigue Testing at Paulin Research Group (November 2006)

Presentation slides of continuing research effort at Paulin Research Group. The work focuses effects of thickness on fatigue life in evaluation of the Master Curve method. Testing with air considered to segregate effect of tap water on fatigue life. Read more...

Fatigue Testing and Life Estimates of Welded Flat Head Pressure Vessel Joints (July 2006)

Presentation slides of fatigue analysis research conducted at Paulin Research Group research facility. The testing considered cylindrical shells with flat heads subject to cycling internal pressure. Application of finite element analysis (FEA) methods are also presented with comparisons against fatigue allowables from various pressure vessel design codes. Read more...

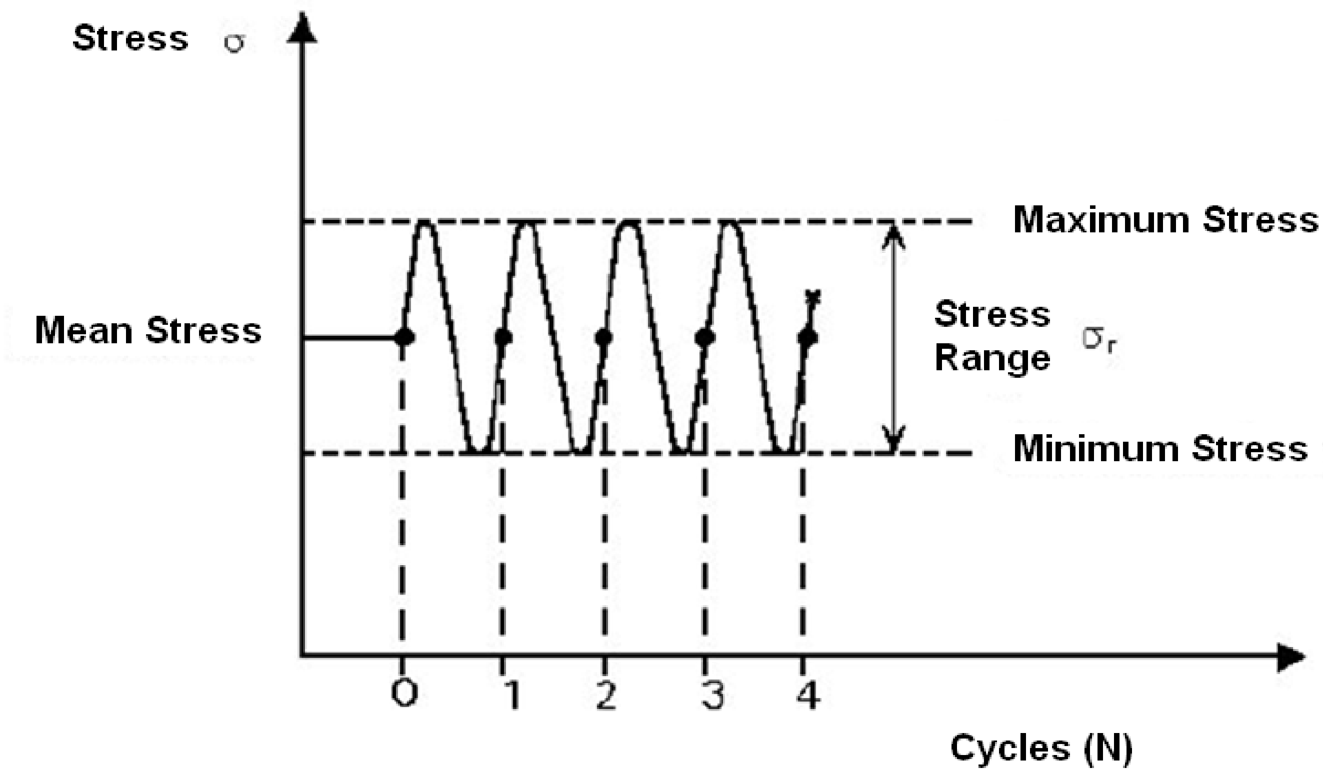
Overview of Presentation

Topics that will be covered:

- **What is Fatigue?**
- **Overview of fatigue design options**
- **Advantages and disadvantages**
- **ASME Division 2 Fatigue Design Methods**
- **Comparison of ASME fatigue methods with test data**
- **Comparison of ASME fatigue methods with other codes**
- **Recommendations for piping analysis in cyclic service**
- **Post-failure activities**
- **Steps to avoid fatigue failures**

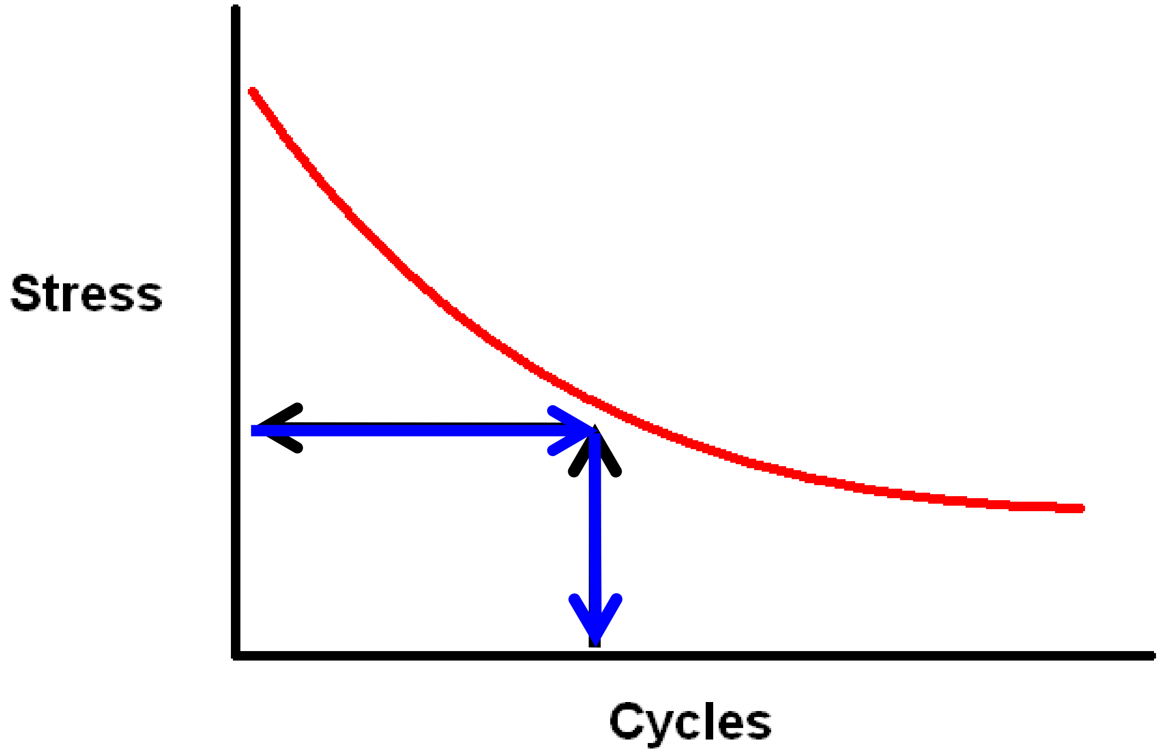
What is Fatigue?

- Accumulation of damage due to oscillating stress/strain
- Begins with crack initiation, progresses with crack growth, and finally fracture



Designing for Fatigue

- Fatigue design charts permit us to relate the stress range (or alternating stress) to the number of permitted cycles.
- Knowing the cycles, we can determine the permitted stress.
- Knowing the required load\stress, we can determine the permitted number of cycles.



What Options Do We Have for Fatigue Design?

- Fatigue methods are often classified by type of stress used.
- Often, the choices ultimately depend on your code of construction.
- Common stress definitions used in the PVP industry include:
  - Nominal Stress – Remote nominal stress (F/A or M/Z)
  - Notch Stress – peak stress at the point of failure
  - Structural Stress – Membrane + Bending Stress
  - Hot Spot Stress – Extrapolated stresses

Every fatigue method \ code has a corresponding stress definition. These are not interchangeable

Code Options for Fatigue Design

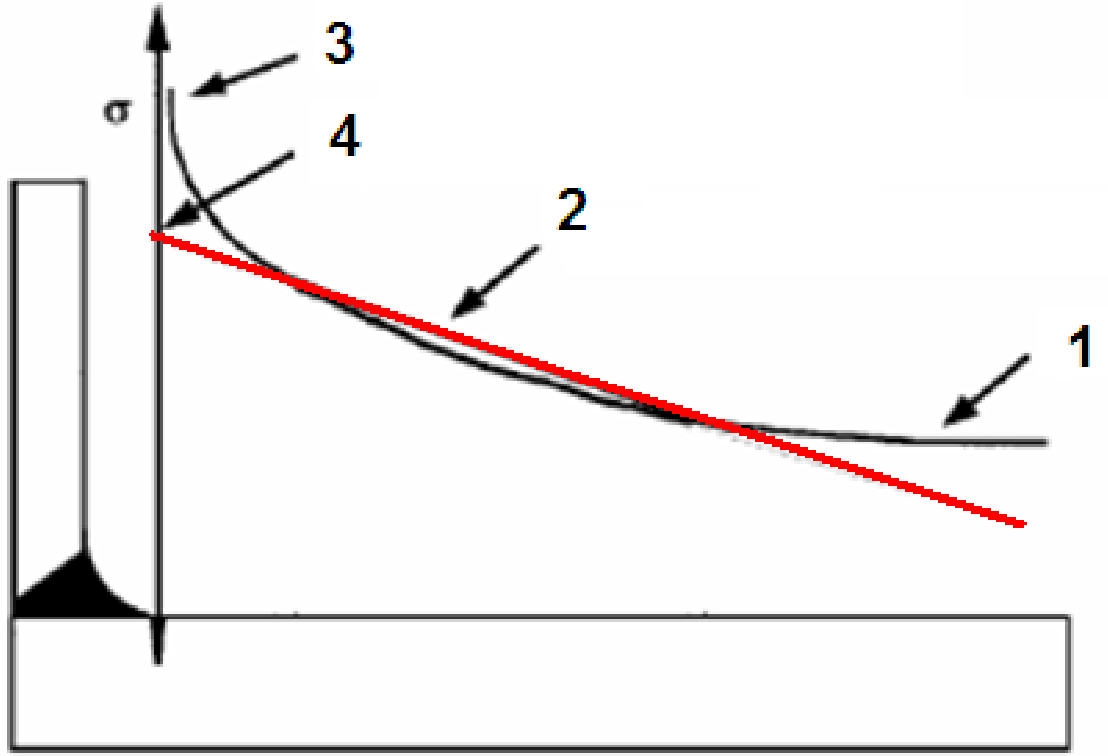
A few of the common fatigue codes used for the PVP industry:

Code	Stress Definitions for Fatigue
ASME VIII-2	Notch Stress (welded, unwelded, & bolts) Structural Stress
BS-5500	Structural Stress Notch Stress (bolts only)
EN-13445	Hot-Spot Stress Notch Stress (unwelded & bolts only)
ASME B31 Piping Codes	Nominal Stress

Other guidance given by IIW, NORSOK, DNV, AWS, AISC, ASME FFS-1, AD-Merkblatter, CODAP, etc, etc.

Next, we will take a look at each stress definition in a little more detail…

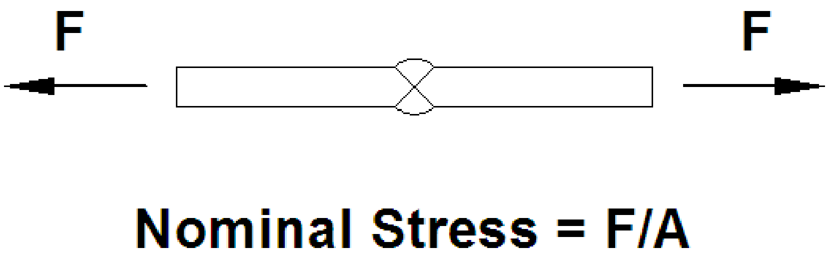
Stress Definitions – A graphical review

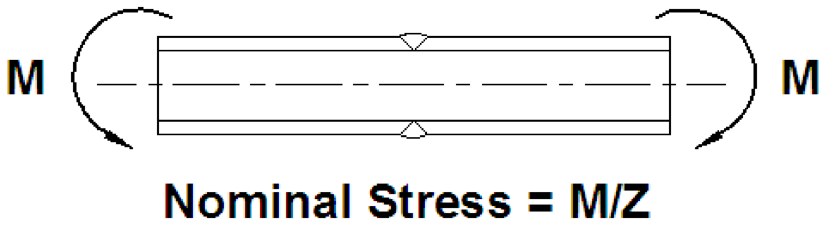


1. Nominal Remote Stress
2. Structural Stress Removed from Toe
3. Notch Stress at Weld Toe
4. Extrapolated Hot Spot Stress

Nominal Stress Methods

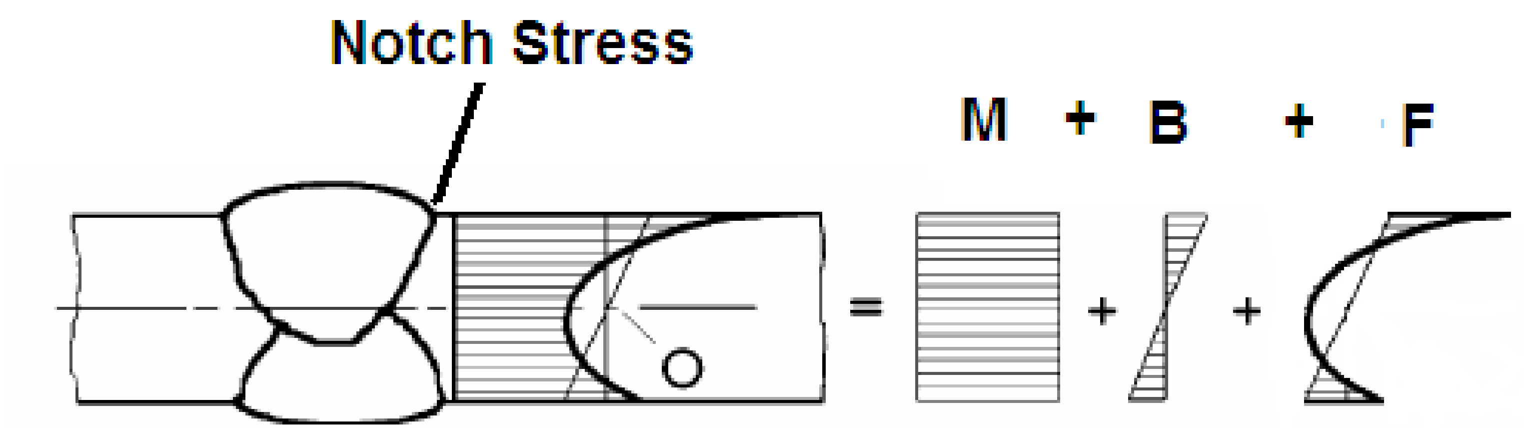
- The nominal stress methods use the stress measured at a point well removed from the anticipated failure site.
- The nominal stress is usually F/A or M/Z – usually a membrane stress.
- Early construction codes (particularly civil and structural) used nominal stress methods for beams with attachments, welds, etc. Many such as AISC and AWS are still used today.
- ASME's piping codes are a notable use of the nominal stress fatigue methods.
- Other "as-welded" methods include attachment details where nominal stress in the main member are used.
- Markl's testing is the basis of the ASME B31 piping codes and relies on the nominal general bending stress.





Notch Stress Methods (Smooth Bar)

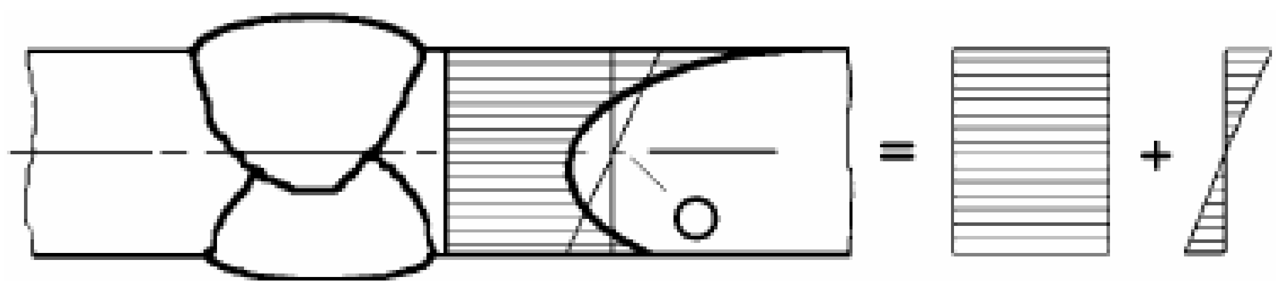
- The notch stress is the total peak stress at the location of interest.
- Peak stress at the notch is used to evaluate the fatigue life. Fatigue life is related back to a smooth bar fatigue curve, possibly derated for welds using an experimentally derived factor.
- Notch stress methods can be used for unwelded and welded locations
- Nearly all PVP Codes provide a notch stress for unwelded regions or threaded bolts.
- ASME VIII-2 and IIW provide explicit rules for evaluation of welded regions using the notch stress approach.



Structural Stress Methods

- **The "Structural Stress" is linearly distributed stress across the section thickness. Essentially the M+B stress through the thickness.**
- **Does not include local peak stresses.**
- **Definitions of the structural stress can vary – for instance the ASME Structural Stress Method uses a specific definition (Equivalent Structural Stress) which is a modified M+B stress.**
- **ASME VIII-2 and PD-5500 are two examples of codes that use a structural stress approaches.**
- **Definition of the structural stress is different in these two codes, but the basic concept of M+B stress is maintained.**

Structural Stress = Membrane + Bending



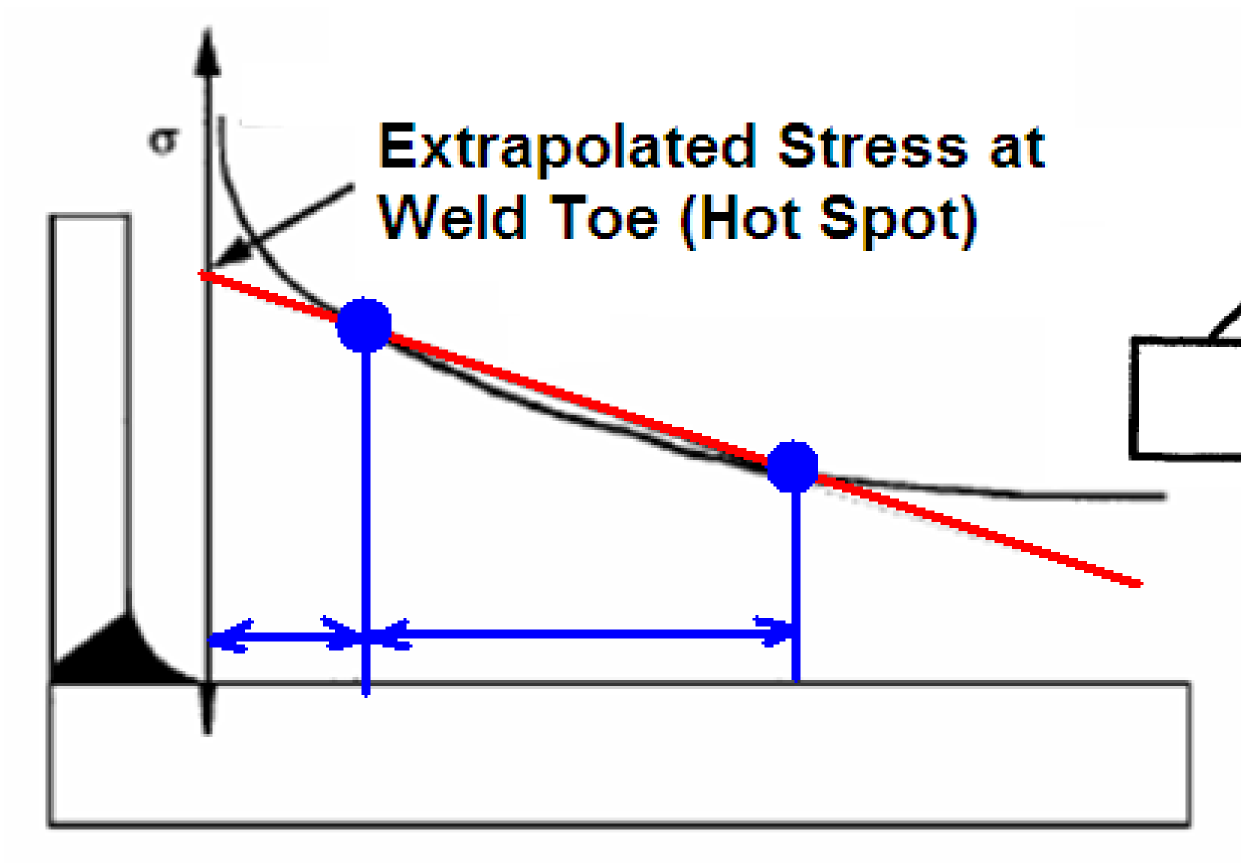
Hot Spot Stress Methods

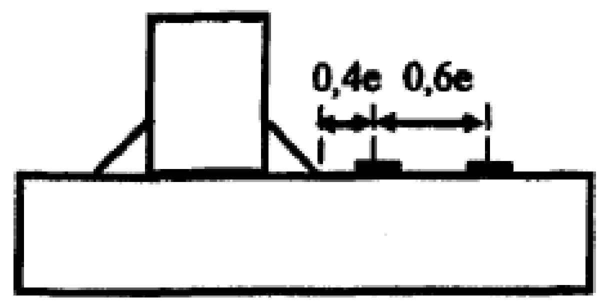
- **Hot spot stress is an extrapolated stress at the failure site. Goal is to capture the structural stress but eliminate the non-linear peak component.**
- **The <u>surface stresses</u> at specified distances from the failure site are extrapolated back to the origin of failure.**
- **Definition is rooted to the testing basis from which the rules are derived and a desire to avoid peak stresses or singularity effects in FEA models.**
- **Where linearization can be used it seems reasonable to use the linearized stress at the point of interest. Of course, this should not be applied in cases where the through-thickness distribution is not expected to be a linear one (ie thermal gradients, nonlinear stress in thick cylinders, etc).**
- **Hobbacher (IIW Doc XIII-1965-03) indicates that hot spot stress can be taken by stress linearization at the weld toe.**

- **EN-13445 and IIW utilize the extrapolation Hot Spot Stress method.**
- **Other codes, such as PD-5500, also reference the extrapolation procedures but it is not the primary stress basis.**

Hot Spot Stress Methods

- **Extrapolation points (locations removed from hot spot) are defined by the applicable code.**
- **Surface stress is extrapolated to the weld toe (hot spot) using linear or quadratic equations (depending on the geometry)**





Hot Spot Stress vs. Structural Stress

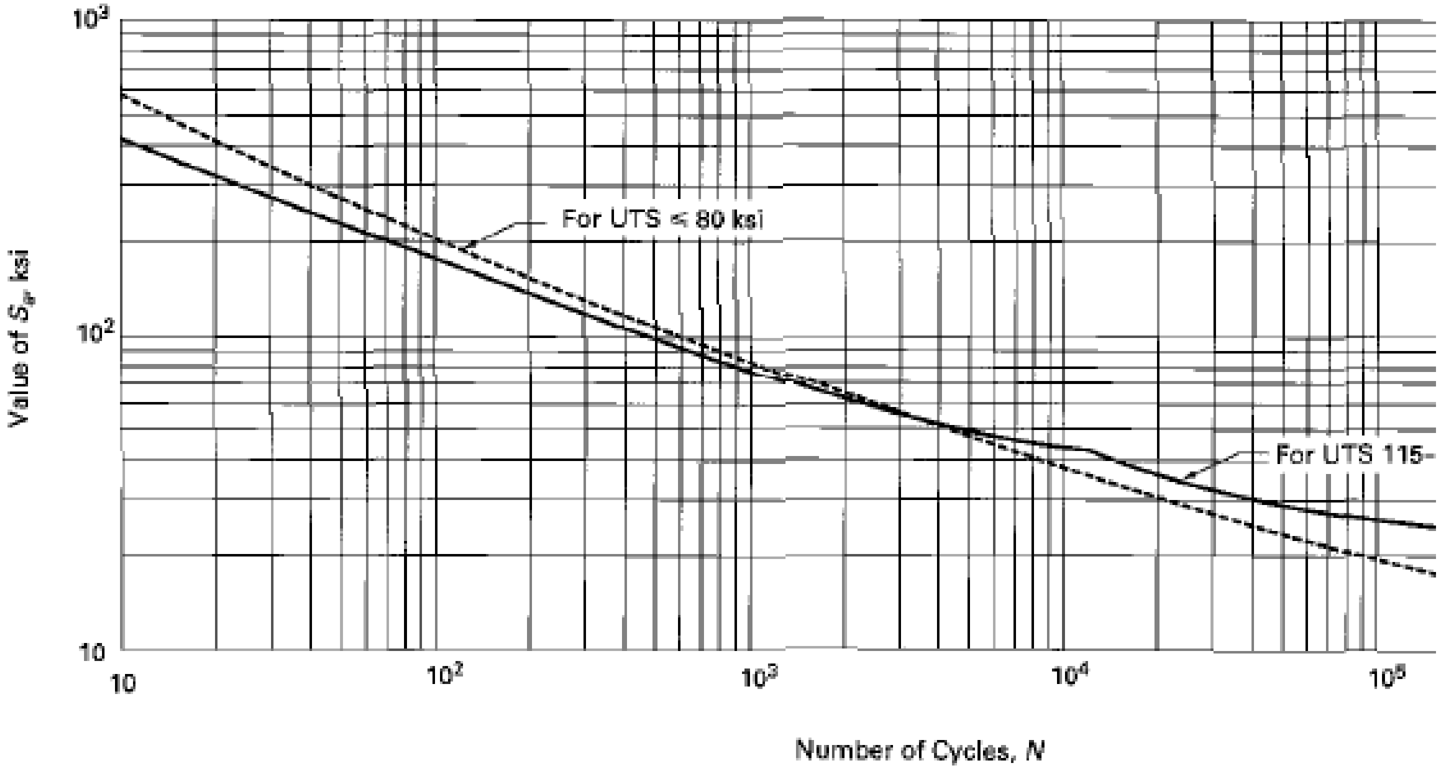
What is the difference between the Hot Spot Stress and Structural Stress?

- These terms are often confused and mistakenly taken to mean the same thing – they are similar, but not always the same.
- For simple geometries where the stress gradient through the thickness is a linear one, there shouldn't be much difference.
- Hot Spot Stress will effectively trap non-linear though thickness distributions as long as they occur on the surface.
- Structural stress seeks to linearize the stress and therefore may not properly predict the "driving force" where the distribution is non-linear.
- Cases where hot-spot stress and structural stress could differ:
    - Non-linear distribution due to thermal gradients
    - Thick walled cylinders where the strain distribution is nonlinear

Now to look at how the stress definitions are implemented into fatigue design charts…

Fatigue Design Curves – Notch Stress Method

- **Fatigue curves are material dependent.**
- **A single curve is given for each material. Can be "shifted" based on the FSRF for welds or other notches.**
- **Design curves are typically a fixed, non-statistical, margin below the mean curve (for example: 2 on stress, 20 on cycles)**



Fatigue Design Curves – Notch Stress Method

- **The following was the original data used to establish the ASME smooth bar carbon steel fatigue curve**

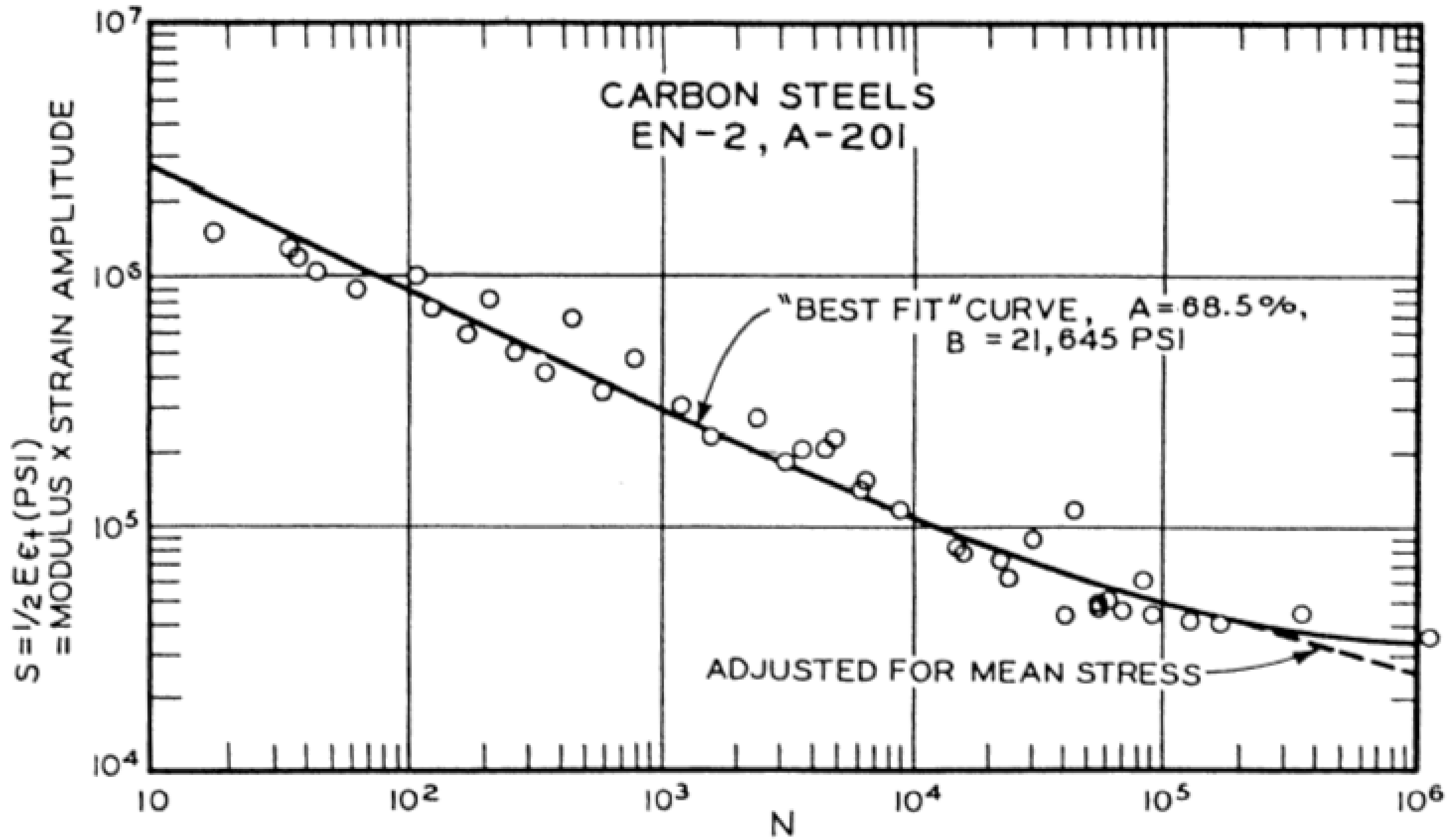
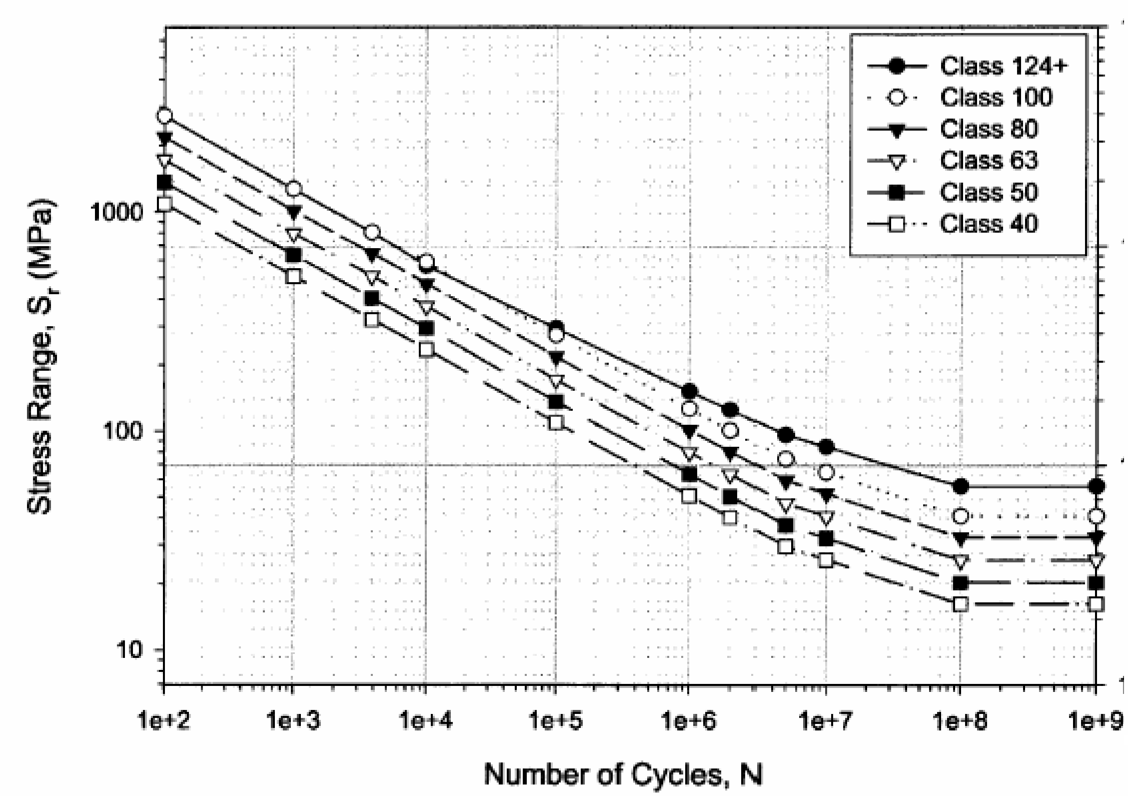
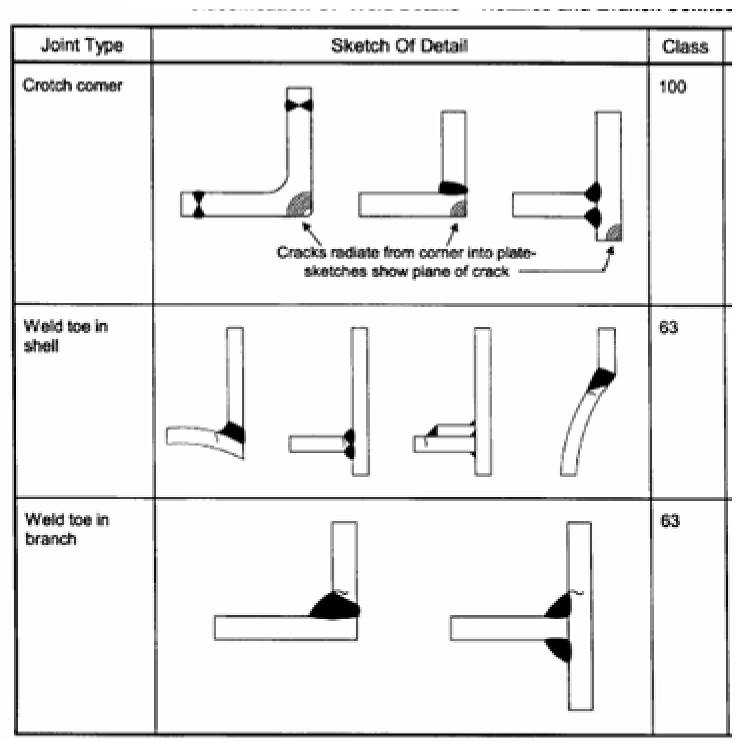


FIG. 9. FATIGUE DATA – CARBON STEELS.

Fatigue Design Curves - Structural or Hot Spot Stress

- Unique curves are given for specific weld details and geometries.
- User selects the closest graphic and associated fatigue design curve.
- Typically, all ferritic steels are designed with a single design curve.
- For the new ASME Structural Stress Method, a single curve is used for all weld types.





Fatigue Design Curves – Nominal Stress

- **ASME B31 piping codes are utilize data from as-welded tests**
- **Basis for Markl girth butt weld shown below (SIF = 1.0)**

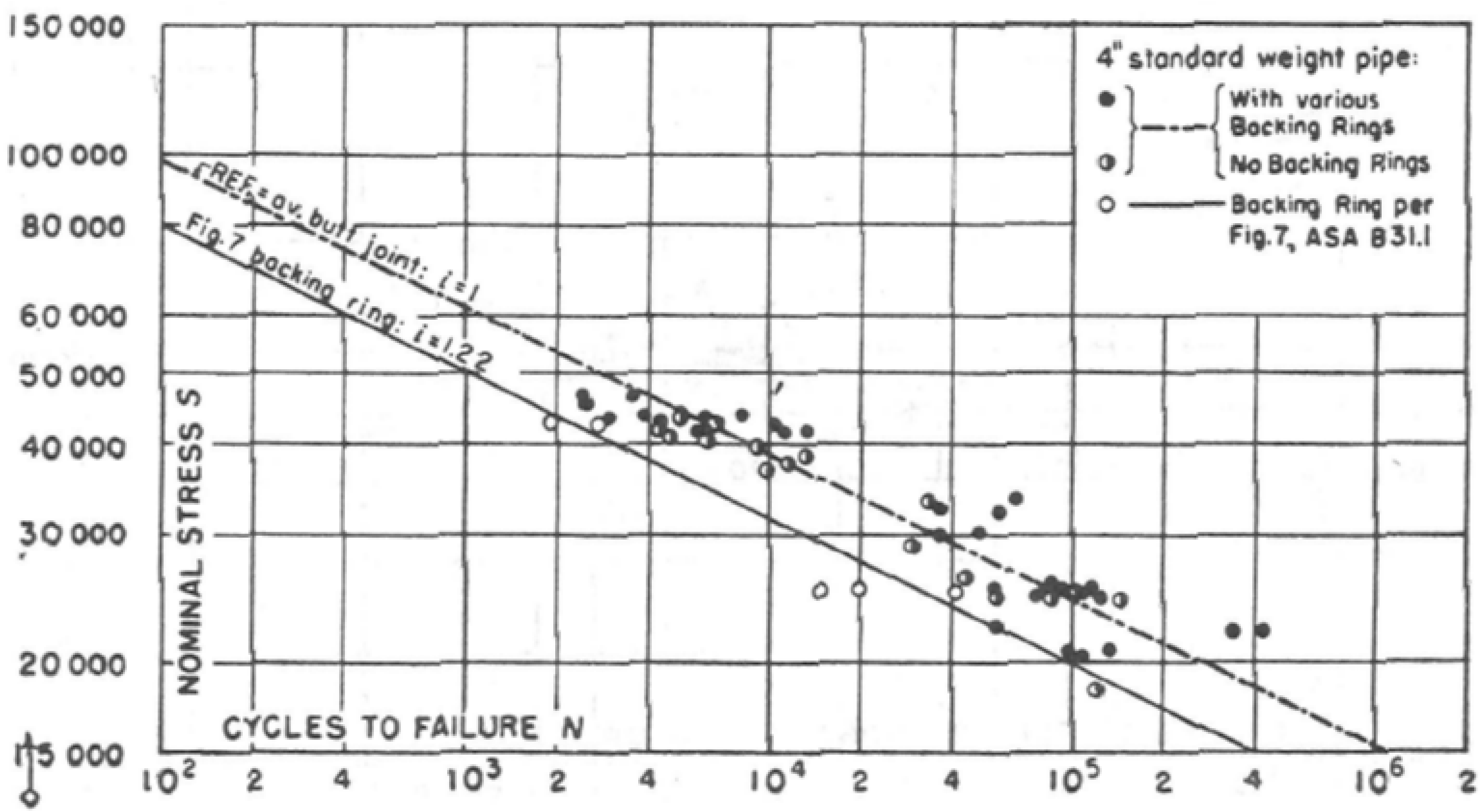


FIG. 5 BUTT-WELDED JOINTS IN STRAIGHT PIPE

Design Margins for Fatigue Methods

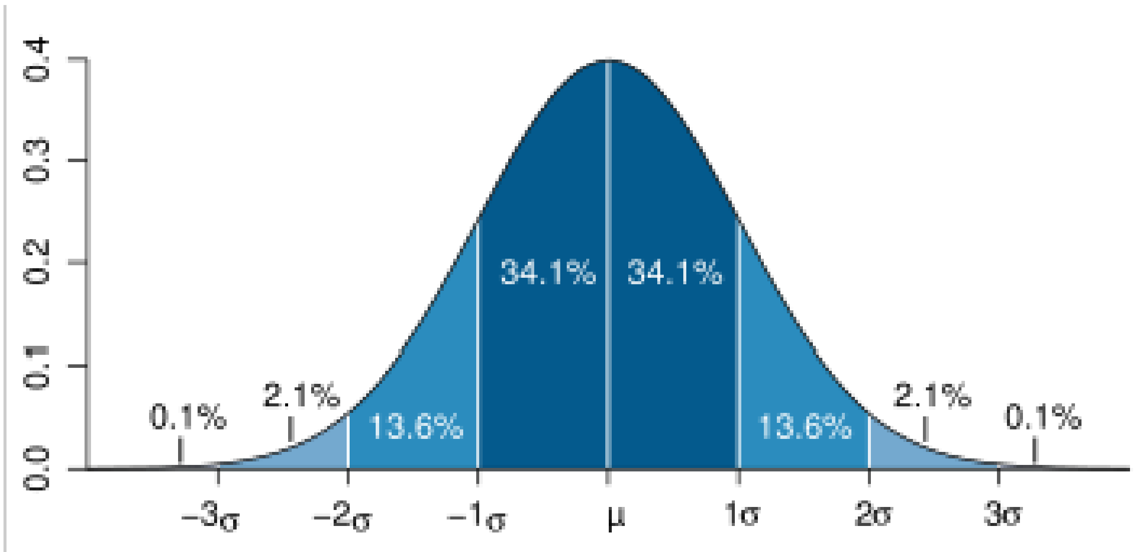
- Fatigue charts are typically derived from experimental data.
- Design curves are derived by applying fixed design margins or using a statistical basis.

Code	Method	Margin
ASME VIII-2	Notch Stress	2 on stress, 20 on cycles
	Structural Stress	User Defined (-3*Std Dev recommended)
BS-5500	Notch Stress	2 on stress, 20 on cycles
	Structural Stress	-2*Std Dev
EN-13445	Notch Stress	1.5 on stress, 10 on cycles
	Hot Spot Stress	-3*Std Dev
B31 Piping	Nominal Stress	Approximately 2 on stress

Design Margins – A Statistical Basis

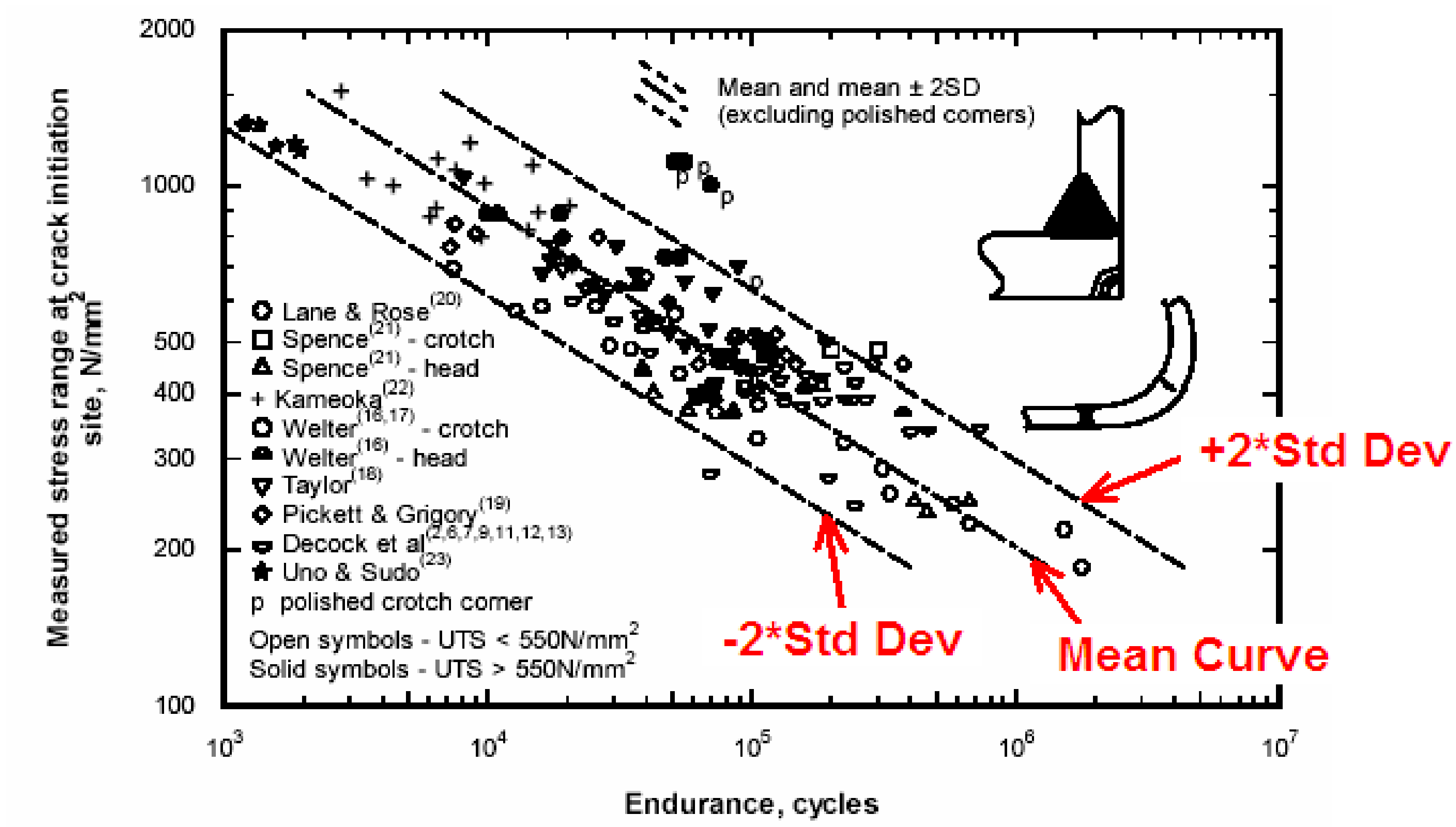
What is "Standard Deviation"?

- **The standard deviation is often reported for a fatigue method…for instance, 0.30, 0.50, 0.60, etc.**
- **The standard deviation of fatigue curves is non-dimensional and reflects the scatter of log(N) – where "N" is the cycles.**
- **Basically, the lower the standard deviation, the better the correlation of the test data to the mean curve.**
- **Fatigue codes typically use either 2 or 3 standard deviations below the mean fatigue curve (~97.7% or ~99.9% probability of survival)**



Design Margins – Statistical Basis

Example of statistical analysis of experimental fatigue data:

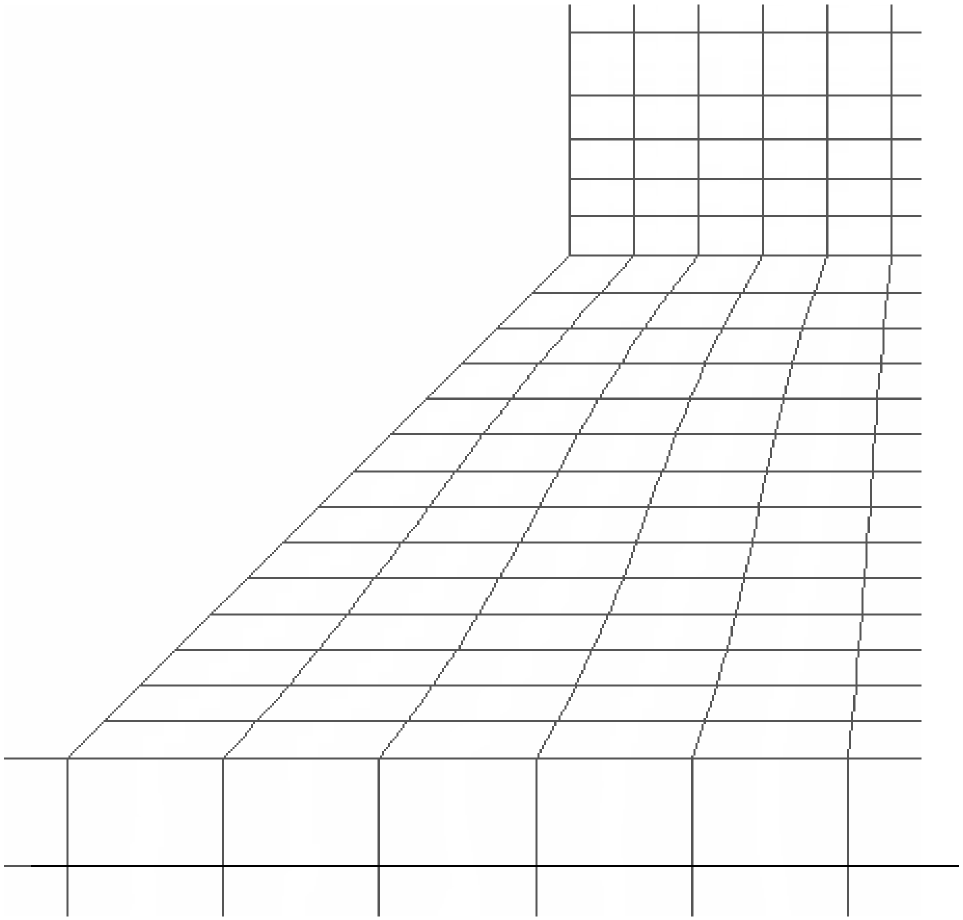


Advantages and Disadvantages

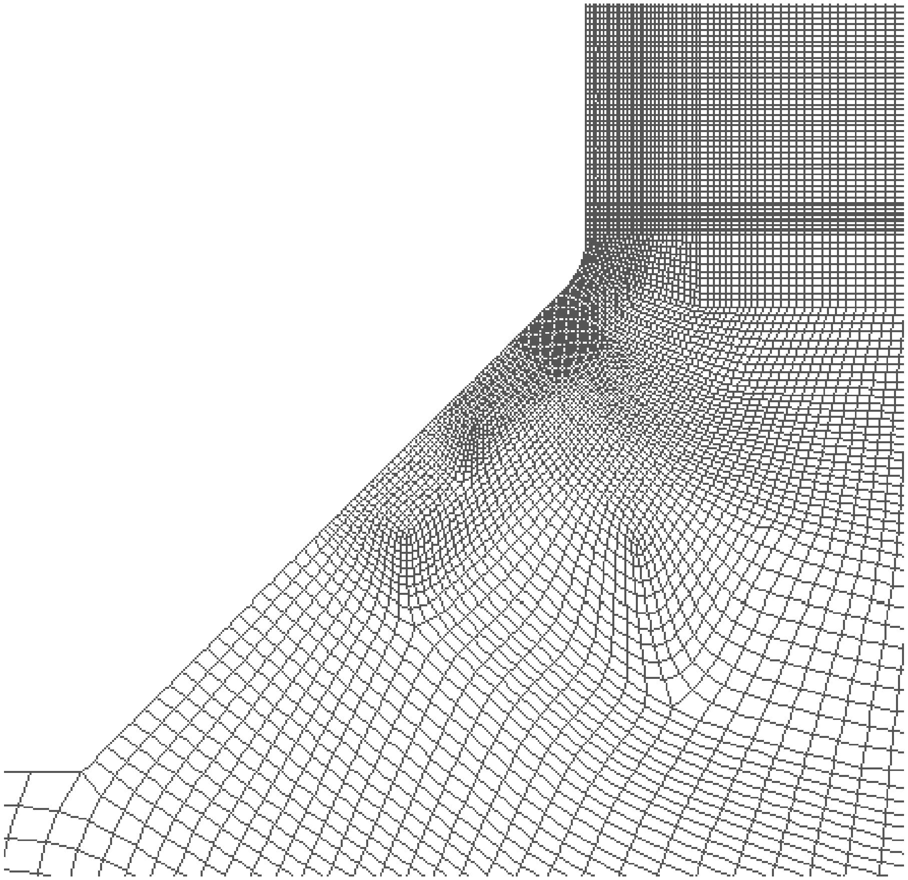
- **Nominal stress methods offer simplicity. However, the stress definition is not advanced enough to characterize complex geometries.**
- **Notch stress methods work well for manufactured notches.**
- **Smooth bar curves can be used for welds provided FSRF’s are available.**
- **Structural Stress based methods are preferred over other methods given the simplicity of the stress calculation using FEA (need only the M+B stress).**
    - **This includes FSRF based notch stress methods.**
- **Structural Stress based methods may not perform as well as the Hot Spot stress method where surface effects dominate the stress state.**
- **Notch stress smooth bar curves better characterize behavior of unwelded metal. Welded metal can show different trends – particularly in the high cycle regime where welds often do not exhibit an endurance limit.**

Advantages and Disadvantages

Mesh Required for Structural Stress (also Notch Stress with FSRF)



Mesh Required for IIW Notch Stress with 0.5mm radius



Example of Fatigue Test at PRG

Video
This was one of the flat head fatigue tests conducted at PRG as part of the Div 2 rewrite project.

2007 ASME VIII-II
Fatigue Design Rules

Important Parts of 2007 Div 2 Related to Fatigue Analysis

The following are some important portions of the 2007 ASME VIII-2 code that are relevant to fatigue design:

- **Annex 3.D – Cyclic stress strain curves for fatigue analysis**
- **Annex 3.F – S-N charts for fatigue analysis**
- **Part 5 – Design by Analysis rules**
  - **Section 5.5 "Protection Against Failure From Cyclic Loading"**
  - **Annex 5.A "Linearization of Stress Results…"**
  - **Annex 5.B "Histogram Development and Cycle Counting…"**
  - **Annex 5.F "Alternative Plasticity Adjustment Factors…"**

Elastic Fatigue Design with 2007 ASME Div 2

The 2007 ASME VIII-2 code provides two S-N chart types.

These S-N chart types allow for three unique elastic fatigue design methods:

1. **S-N charts based on smooth bar specimens**
   - **Peak stress at smooth location or a definable notch**
   - **Peak stress predicted by use of Fatigue Strength Reduction Factors (FSRF's) based on weld joint type and inspection level**
2. **New S-N chart based on welded components**
   - **A single design chart for all weld joint types (butt welds, fillet welds, root failures, etc)**

Due to lack of agreement in the ASME committees, the Structural Stress method may only be used if approved by the purchaser or the owner\operator.

Paragraph 5.5.1.3 – Available S-N Types

- **Tells us that two S-N chart types are available: smooth and welded.**
- **Smooth bar fatigue curves can be used for <u>welded</u> or <u>unwelded</u> locations.**
- **Welded curves (Master S-N method) "<u>shall only be used for welded joints</u>"**
- **When FSRF's are used, the smooth bar method really represents a welded fatigue curve since the FSRF's are developed from tests of welded specimens.**
- **For the structural stress method, be careful with thermal transients since the Master SN database did not contain tests samples under thermal transient loadings.**

5.5.1.3 Fatigue curves are typically presented in two forms: fatigue curves that are based on smooth bar test specimens and fatigue curves that are based on test specimens that include weld details of quality consistent with the fabrication and inspection requirements of this Division.

a) Smooth bar fatigue curves may be used for components with or without welds. The welded joint curves shall only be used for welded joints.

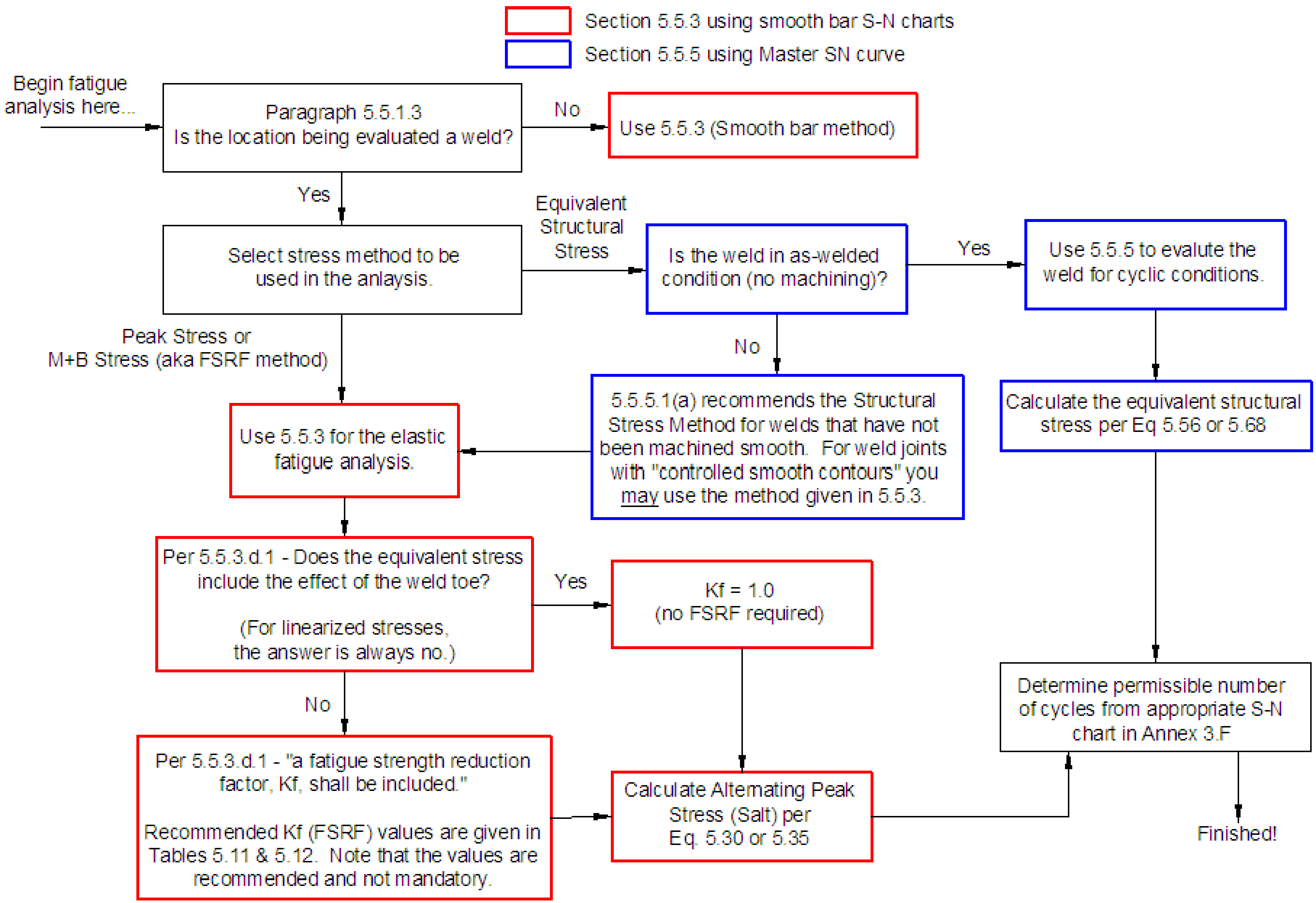
b) The smooth bar fatigue curves are applicable up to the maximum number of cycles given on the curves. The welded joint fatigue curves do not exhibit an endurance limit and are acceptable for all cycles.

c) If welded joint fatigue curves are used in the evaluation, and if thermal transients result in a through-thickness stress difference at any time that is greater than the steady state difference, the number of design cycles shall be determined as the smaller of the number of cycles for the base metal established using either paragraph 5.5.3 or 5.5.4, and for the weld established in accordance with paragraph 5.5.5.

Elastic Fatigue Design : Step-by-Step

1. **Develop a loading histogram (see Annex 5.B)**
2. **Determine if a fatigue analysis is required – you may be exempt.**
   - ***See Section 5.5.2 "Screening Criteria for Fatigue Analysis"***
3. **Pick the fatigue analysis method: smooth bar, smooth bar with FSRF, or new Master SN.**
   - ***Section 5.5.3 "Elastic Stress Analysis and Equivalent Stresses"***
   - ***Section 5.5.5 – Master SN Method***
4. **Calculate stresses for your selected fatigue analysis method.**
   - ***Smooth bar approach requires alternating peak stress.***
   - ***Smooth bar with FSRF requires secondary stress or M+B stress (aka structural stress) which is used to calculate the alternating peak stress***
   - ***Master SN method requires the range of the equivalent structural stress (M+B)***
5. **From Annex 3.F, determine the allowable life for the calculated stress.**

ASME VIII-2 : Elastic Fatigue Design Flow Chart



Elastic Fatigue Design : General Notes

1. **Be careful to account for cases where stress tensors rotate. For instance, if the stress goes from positive to negative during the cycle. A common error is to neglect this when viewing the equivalent stress plots in an FEA model.**
2. **Stress ranges are always determined by first taking the difference between each stress tensor (Sx, Sy, etc) and then determining the equivalent stress range.**
3. **Fatigue methods are only valid for temperatures below the creep regime. ASME Section II, Part D, Table 5A and 5B indicate the temperature at which time dependent properties govern.**

Next, we will discuss fatigue design with the smooth bar S-N charts per Section 5.5.3.

Section 5.5.3 – Fatigue Life using Smooth Bar S-N Charts

- Section 5.5.3 utilizes fatigue curves generated from testing of smooth polished bar samples.
- This is the same method used in ASME VIII-2 since the 1960's.
- Multiple S-N charts are available for various materials in Annex 3.F.
- This method can be used for as-weld joints by introducing experimental based FSRF's. Recommended FSRF's are included in 2007 Div 2.
- Environmental effects, size effects, and mean stress corrections are included in the fatigue charts.
- S-N charts have a minimum design margin of 2 on stress and 20 on cycles.
    - 2.0 for data scatter
    - 2.5 for size effects
    - 4.0 for surface finish and environment
- Includes a simplified elastic plastic correction term "Ke" to correct for under predicted strains and elastic follow-up.

Section 5.5.3 – Fatigue Life using Smooth Bar S-N Charts

- **The smooth bar S-N charts can be used with peak stresses, or secondary (M+B) stresses and an FSRF.**
- **In an FEA model, notch effect of the weld can only be accounted for if there is no singularity. In this case, the FSRF is 1.0 and the peak alternating stress is taken directly from the FEA solution.**
- **At weld toes or other non-definable notches, an FSRF shall be included. In this case, the M+B (or secondary) stress is taken from the FEA model.**

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \left(\Delta S_{P,k} - \Delta S_{LT,k}\right) + K_{v,k} \cdot \Delta S_{LT,k}}{2} \quad (5.30)$$

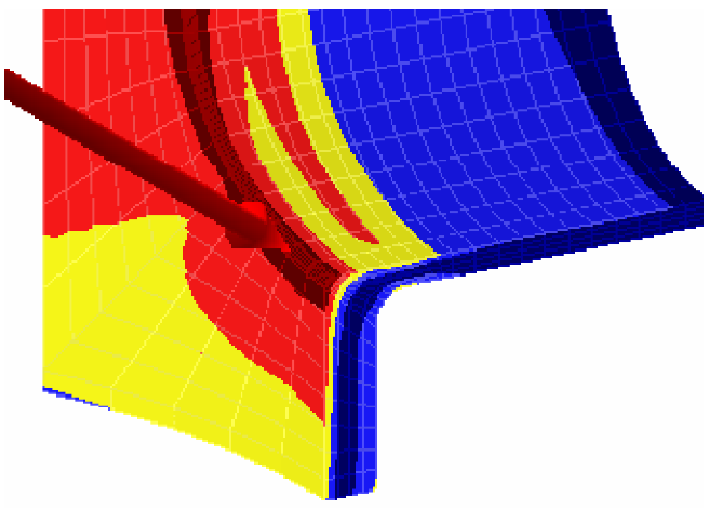
$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2} \quad (5.35)$$

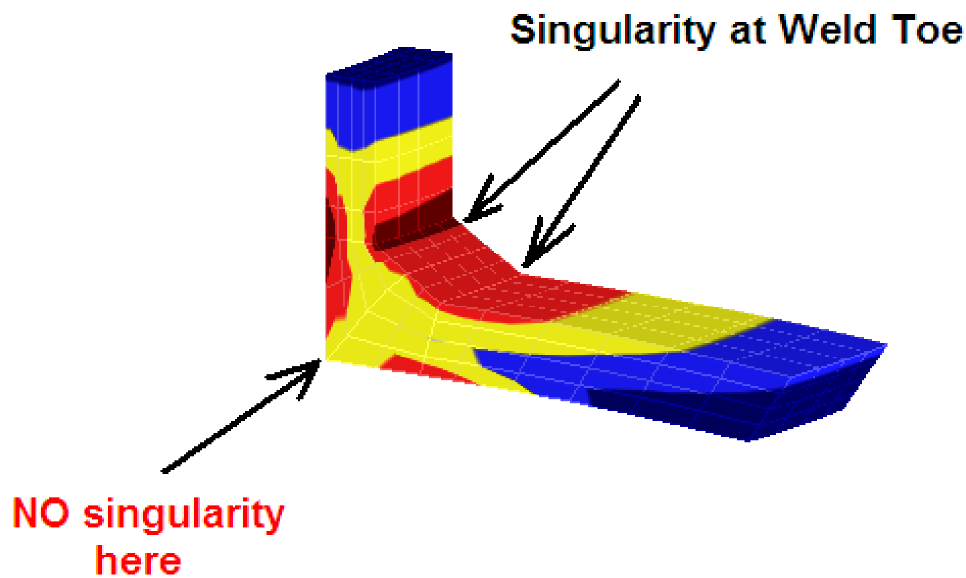
If the local notch or effect of the weld is accounted for in the numerical model, then $K_f = 1.0$ in Equations (5.30) and (5.35). If the local notch or effect of the weld is not accounted for in the numerical model, then a fatigue strength reduction factor, K_f, shall be included. Recommended values for fatigue strength reduction factors for welds are provided in Tables 5.11 and 5.12.

Section 5.5.3 – Fatigue Life using Smooth Bar S-N Charts

- For unwelded regions with manufactured notches (radii, fillets, holes, etc), the alternating peak stress may be taken directly from the FEA solution.
- For as-welded regions, the peak stress can not be taken directly from the FEA solution due to singularities in the model. The peak stress for these areas is calculated using the secondary stress or M+B stress normal to the weld toe:

Pl+Pb+Q+F = (PL+Pb+Q)*FSRF / 2.0 or (M+B)*FSRF / 2.0





Section 5.5.3 – Fatigue Life using Smooth Bar S-N Charts

- **2007 ASME Div 2 now provides FSRF values based on the weld joint type and degree of inspection.**

Table 5.11 – Weld Surface Fatigue-Strength-Reduction Factors

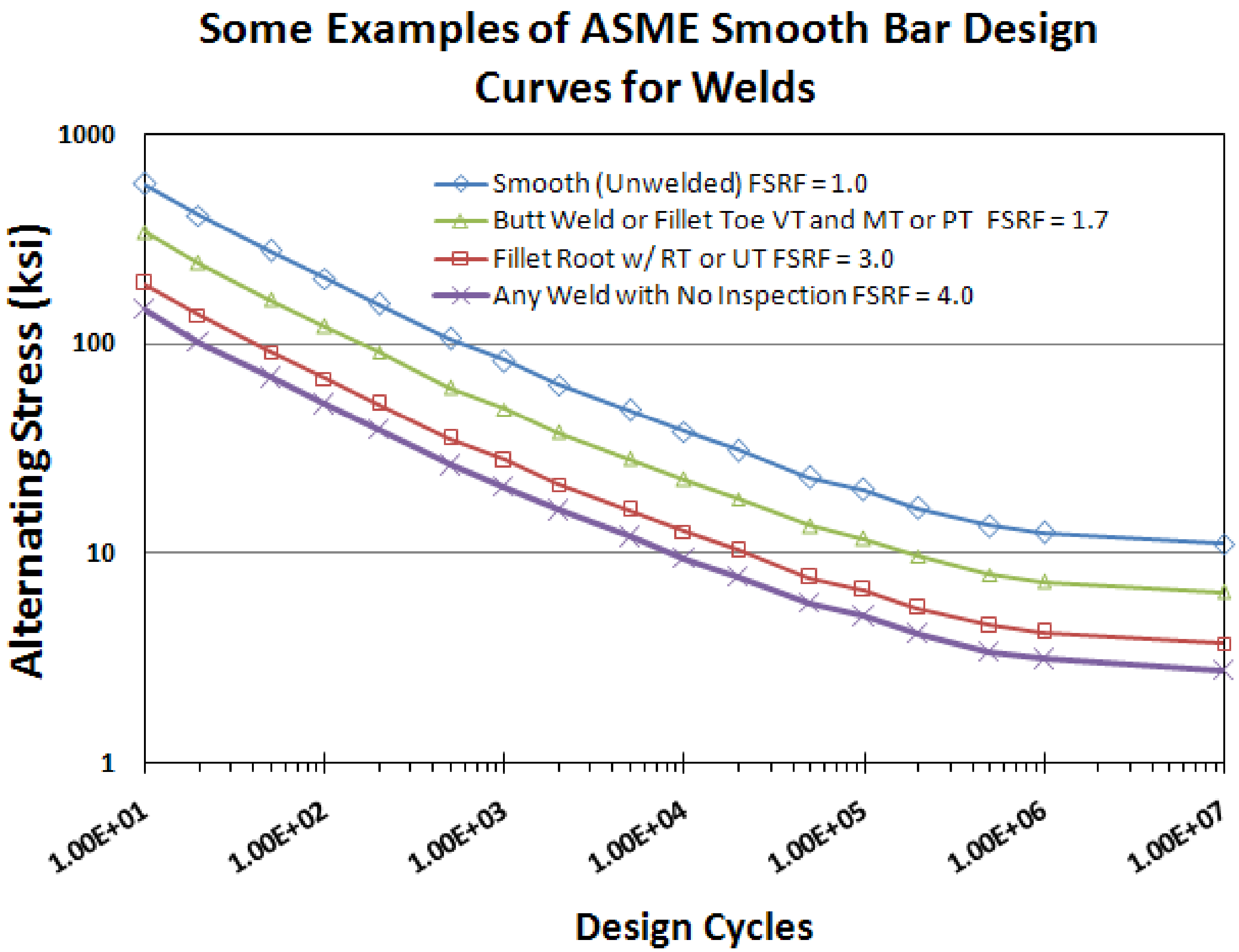
Weld Condition	Surface Condition	Quality Levels (see Table 5.12)						
		1	2	3	4	5	6	7
Full penetration	Machined	1.0	1.5	1.5	2.0	2.5	3.0	4.0
	As-welded	1.2	1.6	1.7	2.0	2.5	3.0	4.0
Partial Penetration	Final Surface Machined	NA	1.5	1.5	2.0	2.5	3.0	4.0
	Final Surface As-welded	NA	1.6	1.7	2.0	2.5	3.0	4.0
	Root	NA	1.5	NA	NA	NA	3.0	4.0
Fillet	Toe machined	NA	NA	1.5	NA	2.5	3.0	4.0
	Toe as-welded	NA	NA	1.7	NA	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	3.0	4.0

Table 5.12 – Weld Surface Fatigue-Strength-Reduction Factors

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MT/PT examination and a VT examination.
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MT/PT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MT/PT examination and VT

Section 5.5.3 – Fatigue Life using Smooth Bar S-N Charts

- **Using the FSRF's produces a series of parallel curves for welds, much like the BS or EN as-welded fatigue curves.**



2007 ASME FSRF / Smooth Bar Method

So, what is the difference between PL+Pb+Q and M+B?

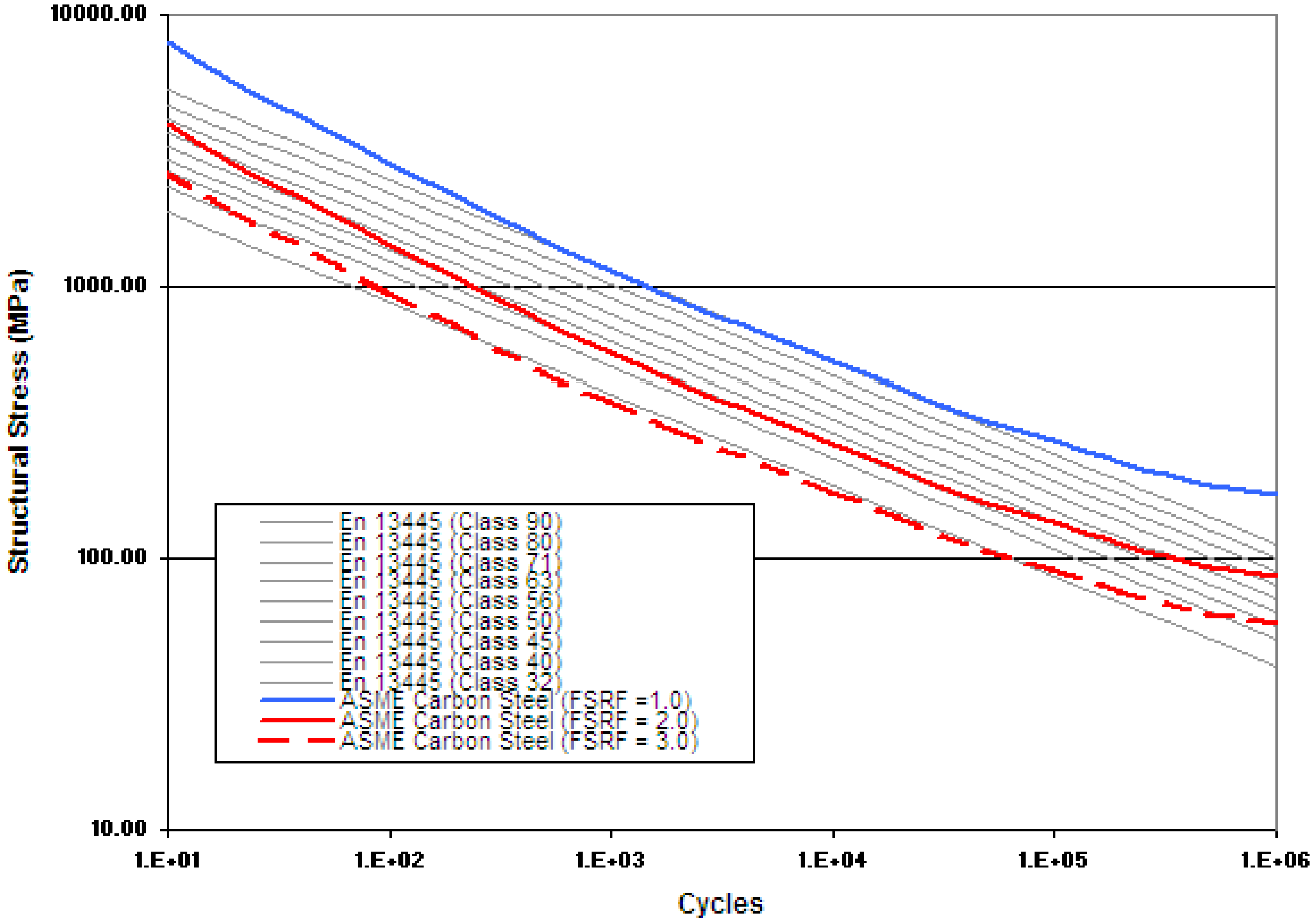
- **PL+Pb+Q is the secondary equivalent stress. It is not necessarily oriented in the direction to propagate a fatigue crack.**
- **PL+Pb+Q is similar to the M+B in that it is often a linear stress distribution.**
- **The M+B stress normal to the fatigue crack is the driving force for crack propagation. It is therefore the best choice for evaluating the fatigue of weldments.**

Which should be used: PL+Pb+Q or M+B?

- **PL+Pb+Q is convenient because this equivalent stress is already calculated for any evaluation in Part 5 (to satisfy secondary stress limits).**
- **M+B normal to the weld toe requires additional evaluation of the stress state.**
- **It is not clear how well the M+B stress normal to the weld toe would address cases with multiaxial loading conditions.**
- **If FSRF's are not intended for specific loading directions, then PL+Pb+Q may be a better choice to ensure conservatism.**
- **Use of PL+Pb+Q offers a conservative and simple choice. If further refinement is required, then use the M+B stress normal to the weld toe.**

FSRFs vs. Welded Curves (EN 13445)

- FSRF's as recommended in Div 2 rewrite provide good match to other as-welded fatigue design codes.
- As shown below, FSRF values envelope EN 13445's welded curves.



Next, we will discuss some aspects of the ASME Master SN Method per Section 5.5.5

Section 5.5.5 – Master SN Method

- **This method may only be used if approved by the purchaser or owner\operator.**
- **Based on the Battelle Master SN Method, but has been modified.**
- **For the analysis of as-welded components.**
- **Not intended for non-welded regions.**
- **Includes explicit correction factors for:**
  - **Thickness effects**
  - **Mean stress effects**
  - **Membrane and bending ratios**
  - **Cyclic plasticity**
- **Design margins are based on statistical confidence intervals.**
- **Recommended interval is 3 standard deviations below the mean life curve (approximately factor of 1.7 on stress and 5.47 on cycles)**
- **An environmental factor of 4.0 is required unless a more appropriate factor is known (see Annex 3.F).**

Section 5.5.5 – Master SN Method

Some features of the Structural Stress Method:

- **A single fatigue curve is used to represent all weld joint types (butt welds, fillet welds, root failures, corner welds, etc)**
- **Utilizes an Equivalent Structural Stress (modified M+B stress) that considers thickness effects, membrane to M+B ratio, mean stress effects, and cyclic plasticity.**
- **No endurance limit exists in the high cycle regime.**
- **Fatigue life is independent of material type – but low cycle predictions are material dependent because of cyclic stress-strain curves.**

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}} \quad (5.56)$$

Section 5.5.5 – Master SN Method

1. **Adjust elastic stresses for plastic action (5.53, 5.54, 5.55)**
   - **Requires iterative solution**
   - **Always increases stress, even for fully elastic stresses (conflicts with ASME Code)**
2. **Calculate bending to M+B ratio (5.62)**
3. **Calculate the Mode I crack growth parameter (5.61)**
4. **Calculate mean stress factor (5.63)**
5. **Calculate the equivalent structural stress (5.56)**

$$\Delta\sigma_k \cdot \Delta\varepsilon_k = \Delta\sigma_k^e \cdot \Delta\varepsilon_k^e \qquad (5.53)$$

$$\Delta\varepsilon_k = \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}} \qquad (5.54)$$

$$\Delta\sigma_k = \left(\frac{E_{ya,k}}{1-\nu^2}\right)\Delta\varepsilon_k \qquad (5.55)$$

$$R_{b,k} = \frac{\left|\Delta\sigma_{b,k}\right|}{\left|\Delta\sigma_{m,k}\right| + \left|\Delta\sigma_{b,k}\right|} \qquad (5.62)$$

$$I^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b,k} - 0.17R_{b,k}^2}{1.007 - 0.306R_{b,k} - 0.178R_{b,k}^2} \qquad (5.61)$$

$$f_{M,k} = \left(1 - R_k\right)^{\frac{1}{m_{ss}}} \qquad (5.63)$$

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}} \qquad (5.56)$$

Master SN : ASME Implementation vs. Battelle Database

- **The ASME implementation of the Battelle Master SN method has a number of alterations:**
    1. **A limit to the thickness adjustment factor is specified. This reduces the fatigue life bonus for "thin" plates.**
    2. **A mean stress correction is applied for high tensile mean stresses within the high cycle regime.**
    3. **The Neuber plasticity correction is required for all loading conditions, not just load controlled low cycle tests beyond the cyclic yield stress.**

- **The Battelle database was formed without thickness correction limits, without mean stress corrections, and without the use of any plasticity correction terms. Therefore, the ASME stress definition for the Master SN method does not match that used in the database curves.**
- **The statistical significance of the design curves is unknown.**

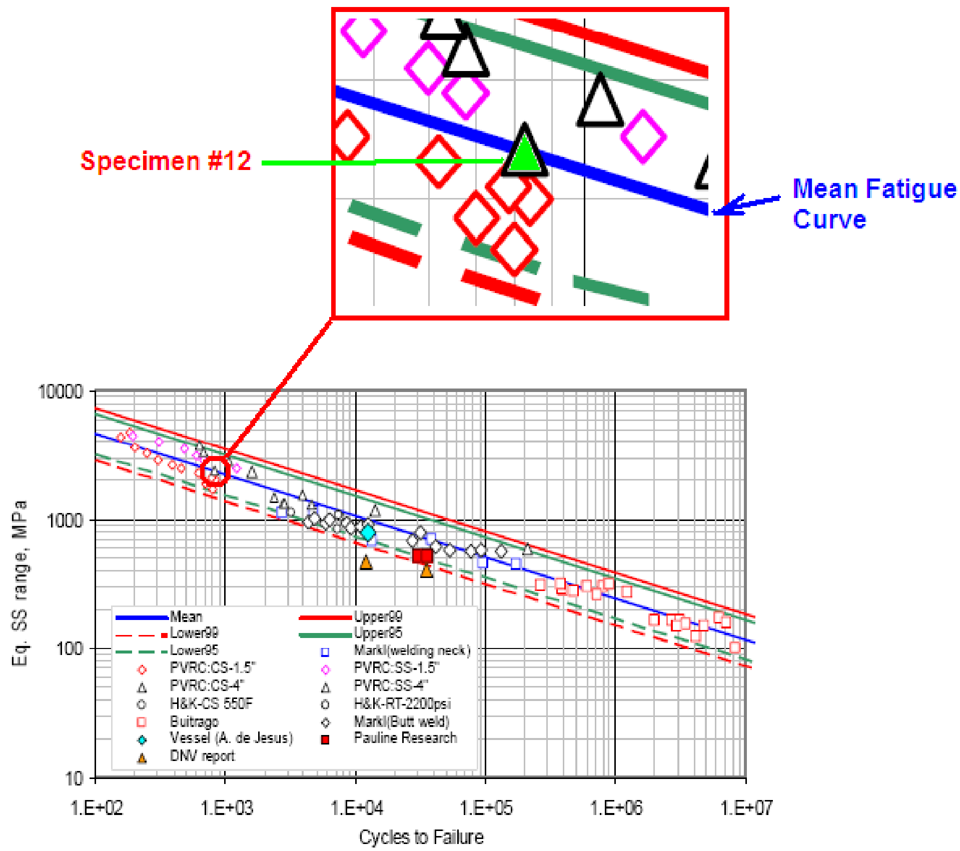
Master SN : ASME Implementation vs. Battelle Database

- **The alterations provide more conservatism, so are they a concern?**
- **These modifications have several implications:**
  1. **Potential for unnecessary design modifications and equipment cost.**
  2. **The stress definition used by ASME does not match that used by Battelle to develop the statistical based design curves. In other words, it is not necessarily clear how the design method relates to the failure database.**
     - **Most statistically based codes (like BS-5500, EN-13445, etc) use the same stress definition used to develop their fatigue curves.**
  3. **The ASME implementation may not provide the same level of accuracy in terms of predicting fatigue lives as the original Battelle method. In other words, a data point that falls on the Battelle mean curve is not accurately predicted by the ASME implementation.**
  4. **If the ASME stress definition was used to re-evaluate the Battelle fatigue database it is likely that different design curves would be derived with a larger standard deviation.**
  5. **Appropriateness in fitness for service evaluations is unclear. FFS evaluations should be done with a firm understanding of the margin against failure.**

Master SN : ASME Implementation vs. Battelle Database

An example of the influence of the altered stress definition:

- **Select a data point that is along the mean curve of the original Battelle database and see if the 2007 ASME Code can predict this test.**
- **The actual life is 222 cycles, nearly on the mean curve of the Battelle database.**
- **The predicted mean life by the 2007 ASME Code is 5 cycles.**



Comparison of Fatigue Calculations with Smooth Bar and Master SN Method

2007 ASME VIII-2 : Differences of Fatigue Methods

- **There can be large differences in the allowable fatigue life of welds depending on the method selected (i.e. FSRF vs Master SN).**
- **Some areas where you will see the most notable differences:**
    - **Stainless Steel Equipment– Master SN Method says stainless and carbon steel have the same fatigue life (since it assumes that all materials are same)**
    - **Fillet root failure – New Master SN Method will give higher fatigue life than the smooth bar method (FSRF=4)**
    - **Butt welds – New Master SN Method will give fewer cycles than smooth bar method (FSRF >= 1.2)**
    - **Fillet weld toe – Life is similar for the FSRF and Master SN methods. A good match is expected since the ASME Master SN method was adjusted to fit PRG test data for fillet toes.**
    - **Design cycle life >1e6 cycles – Smooth bar curves exhibit an endurance limit whereas the Master SN method does not.**

Comparison of 2007 VIII-2 Fatigue Methods

- **For welds, in comparison with FSRF based method (5.5.3), the Master SN method (5.5.5) can result in:**
  - ***Decreased life for "good joints" (for example, butt welds)***
  - ***Increased life for "bad joints" (for example, root of fillet welds)***

A couple of examples for "good" and "bad" weld joints:

- **ASME FSRF method predicts 100,000 cycles for a 1" thick CS girth butt weld and >3e6 for SS…but the Master SN would only allow approximately 12000 cycles for the both.**
- **ASME FSRF method predicts 10,000 cycles for fillet root failure, but the Master SN would allow almost 80,000 cycles for the same joint.**

- **Appears that the ASME Master SN Method is not as sensitive as other Codes to joints where quality and constructability is poor. For instance, there is a wider difference in BS-5500 and EN-13445 for butt welds vs. fillet weld root failures.**

Annex 5.A

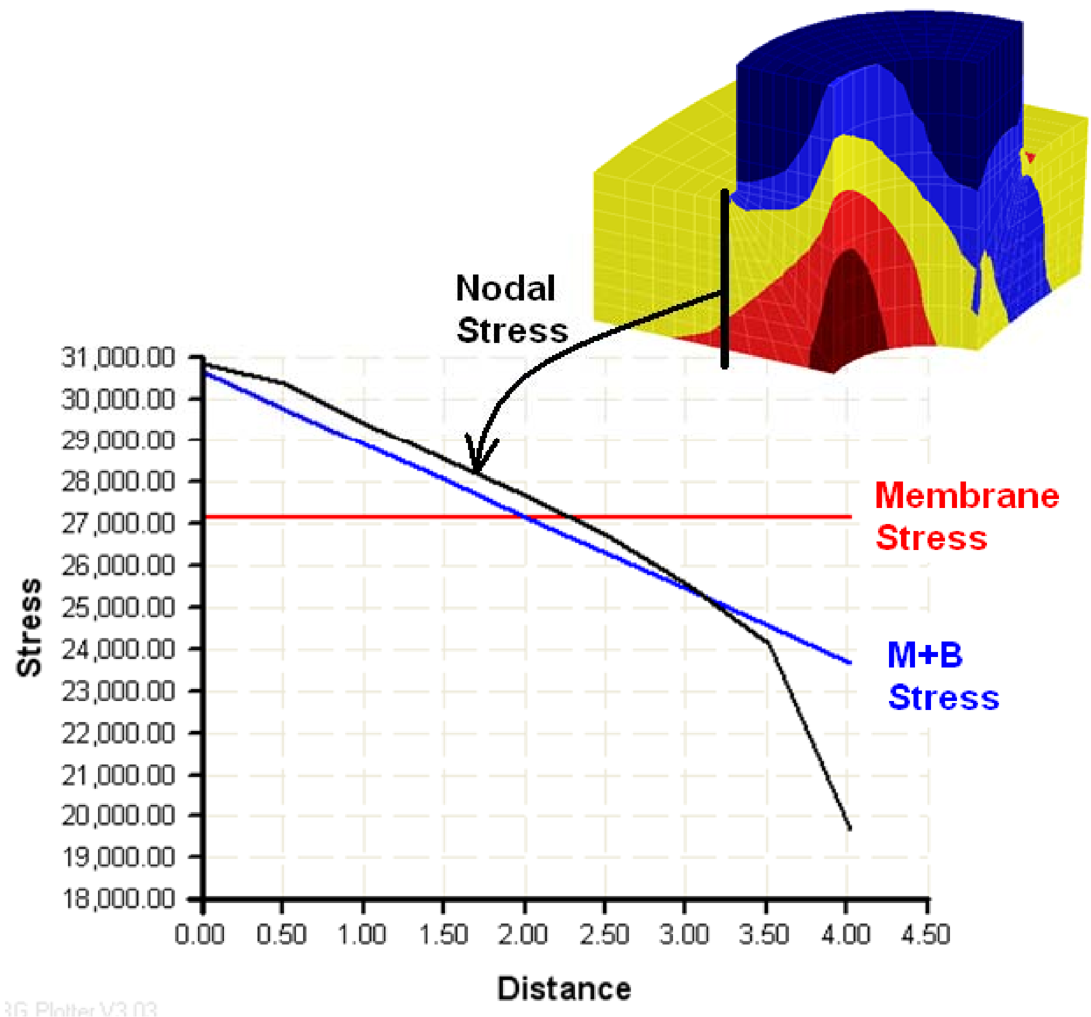
Stress Linearization

Annex 5.A – Linearization of Stress Results….

- **Annex 5.A provides guidelines for performing stress linearization of finite element results.**
- **Stress linearization is required for fatigue analysis anytime “volumetric” elements are used (axisymmetric, brick, etc).**
- **Annex 5.A includes the Nodal Force method <u>patented</u> by Battelle.**
- **Where properly applied, stress linearization and the nodal force method will give very similar results.**
- **It is highly recommended that analysts review the recommendations in WRC 429 for proper application.**
- **Most general use FEA tools do not provide appropriate linearization techniques.**

Annex 5.A – Linearization of Stress Results….

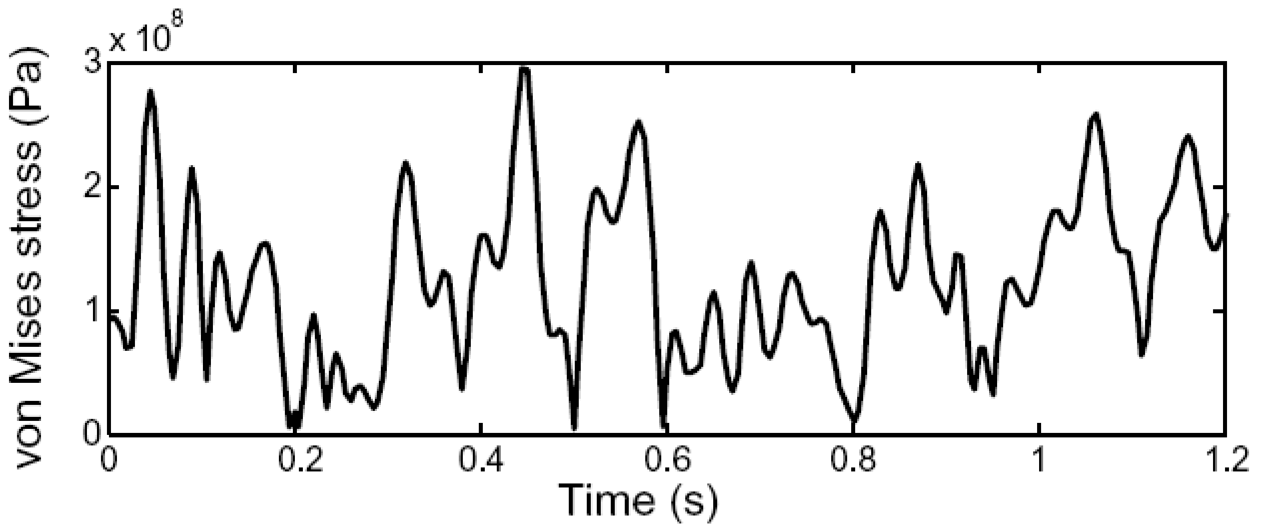
- **Stress linearization extracts an equivalent membrane and bending stress from a through thickness stress distribution.**



Annex 5.B
Histogram Development and Cycle Counting

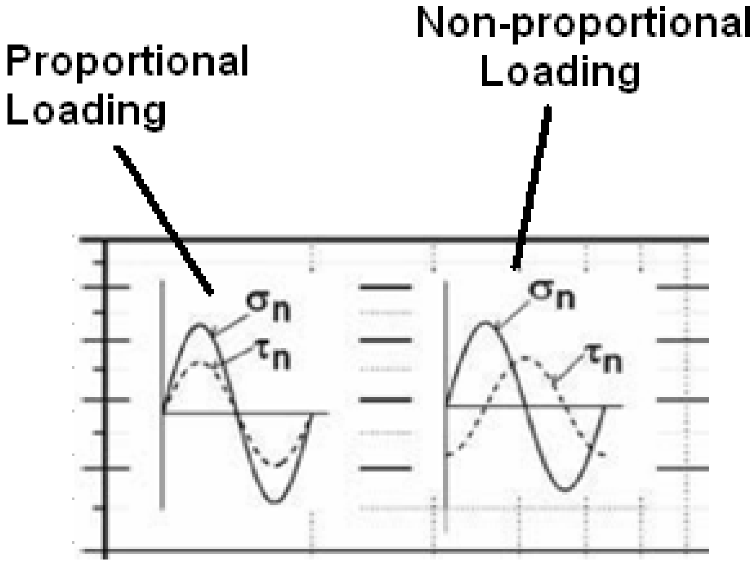
Annex 5.B – Histogram Development and Cycle Counting

- **Annex 5.B includes recommendations for histogram development and cycle counting.**
- **5.B.3 specifies that a histogram for all applicable loads should be developed. Essentially a time history plot for all expected loadings.**
- **What is cycle counting?**
    - Procedures which reduce variable amplitude history to a set of equivalent constant amplitude load or stress ranges.
- **Why is cycle counting required?**
    - Fatigue design methods are normally derived from constant amplitude tests.
    - A method of relating variable amplitude damage to that described by the constant amplitude



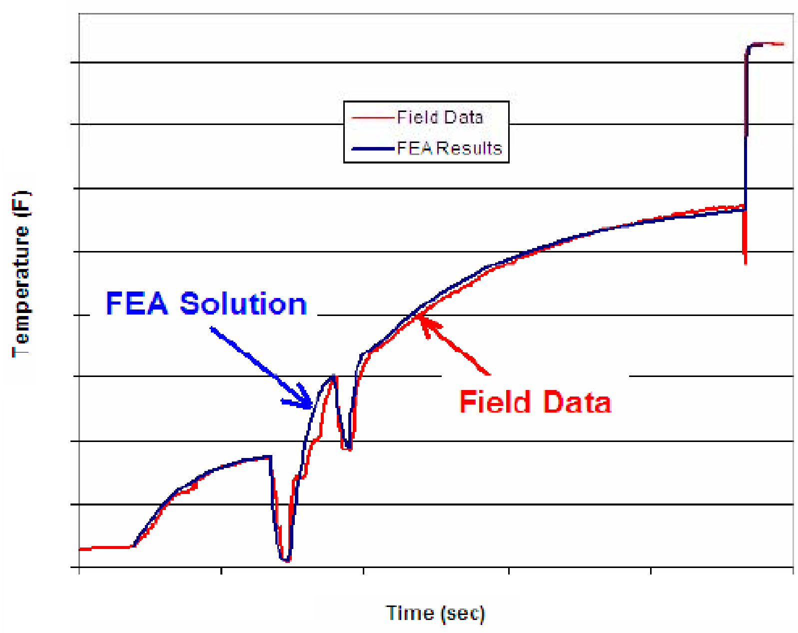
Annex 5.B – Histogram Development and Cycle Counting

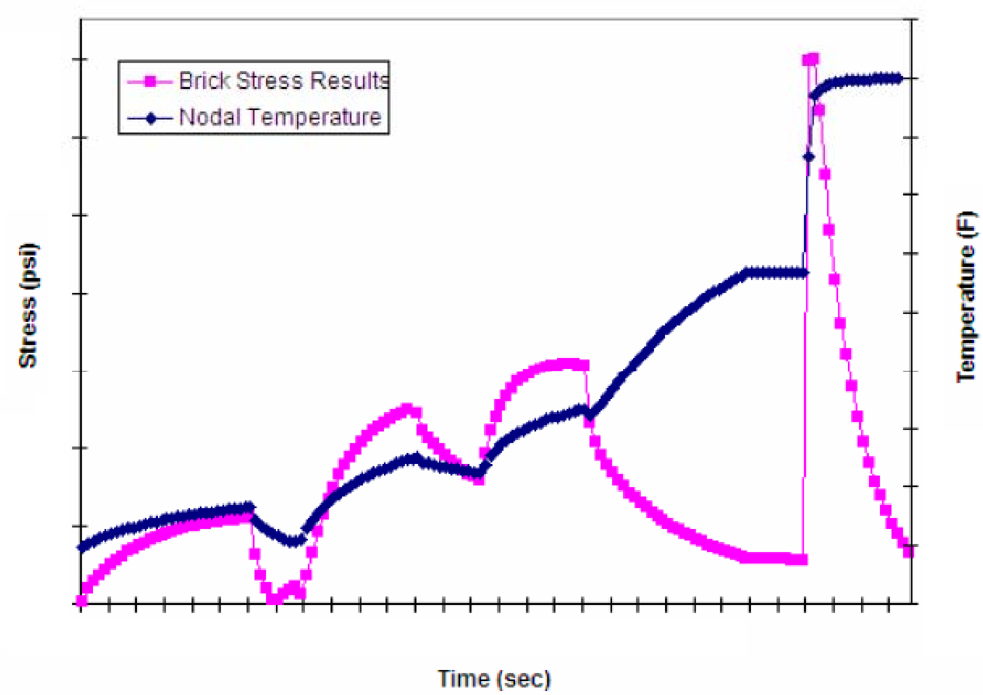
- **Annex 5.B provides two methods for cycle counting:**
  - **Rainflow method**
    - For proportional loads only
  - **Max-min method (aka peak counting)**
    - Proportional or non-proportional loads
- **Proportional loading is the case where all loads can be related by a single parameter that does not vary with time. The ratio remains constant.**
- **In cases with non-proportional loading or multiaxial loading, it is recommended that an expert is consulted.**



Annex 5.B – Histogram Development and Cycle Counting

- **An example of a case where cycle counting is required is shown below for the thermal transient at an inlet nozzle.**





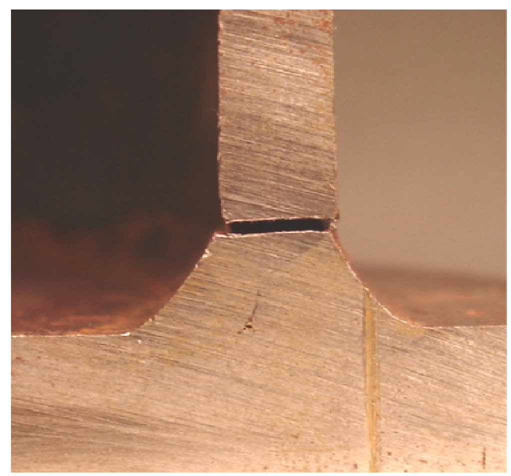
Example of Fatigue Test at PRG

Video
Girth butt weld piping fatigue tests at PRG.

Comparison of ASME Fatigue Methods with Experimental Results

Experimental Validation

- **PRG has tested flat heads attached to cylindrical shells exposed to cyclic internal pressure**
- **These are ASME designs and welds, representative of equipment that might be constructed with the ASME codes.**
- **A total of 43 experimental failures were generated.**
- **ASME FSRF approach closely predicts the experimental failures.**
  - **Standard Deviation = 0.14**
- **ASME Master SN method is under predicts the experimental failures.**
  - **Standard Deviation = 0.585**



Experimental Validation

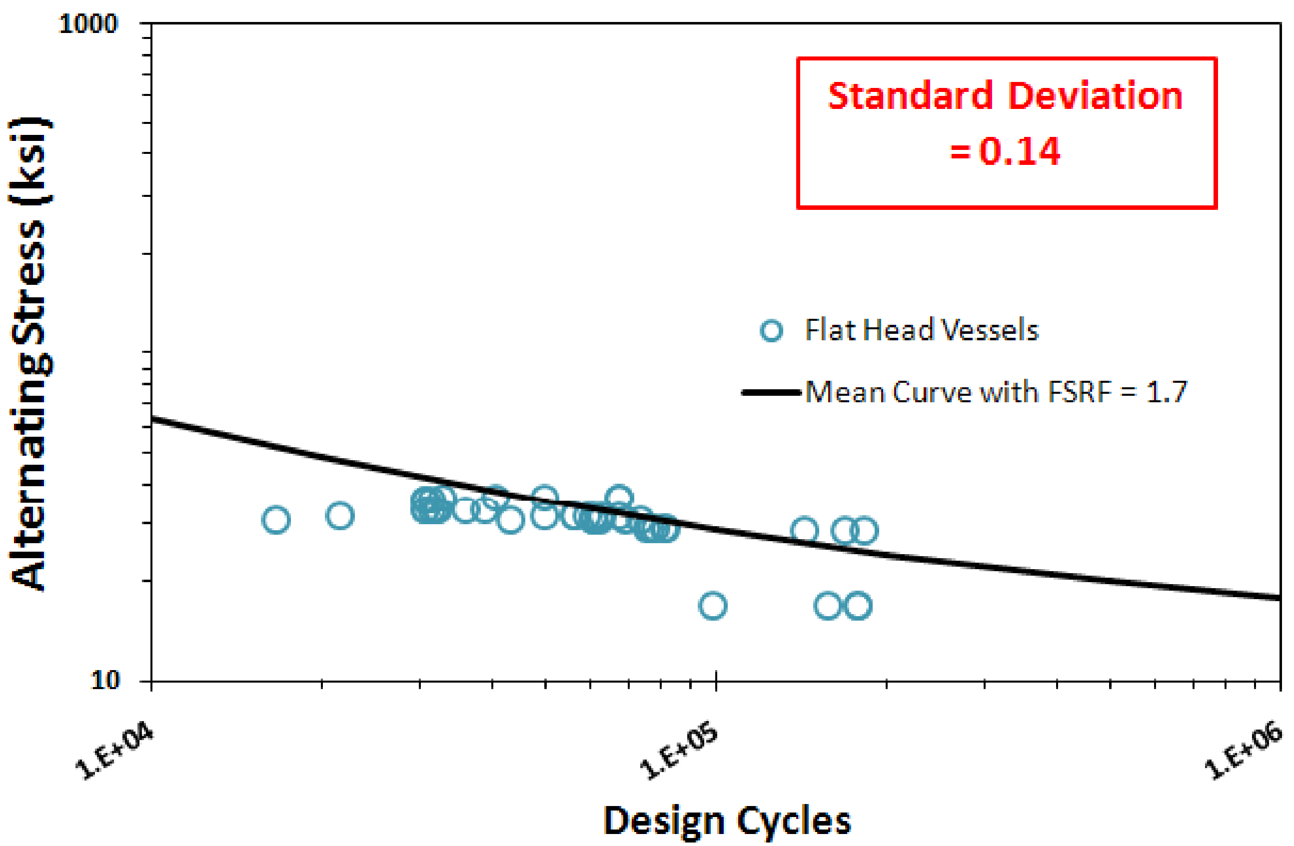
Details of these tests were reported in:

- **PVP2006-ICPVT11-93967 “Fatigue Testing and Life Estimates of Welded Flat Head Pressure Vessels Joints”, C. Hinnant**
- **PVP2007-22662 “Fatigue Testing of Welded Flat Head Pressure Vessel Joints”, C. Hinnant**

- **You can download presentation slides for these PVP publications as well as other presentations to ASME BPVC committees at the PRG website (www.paulin.com)**

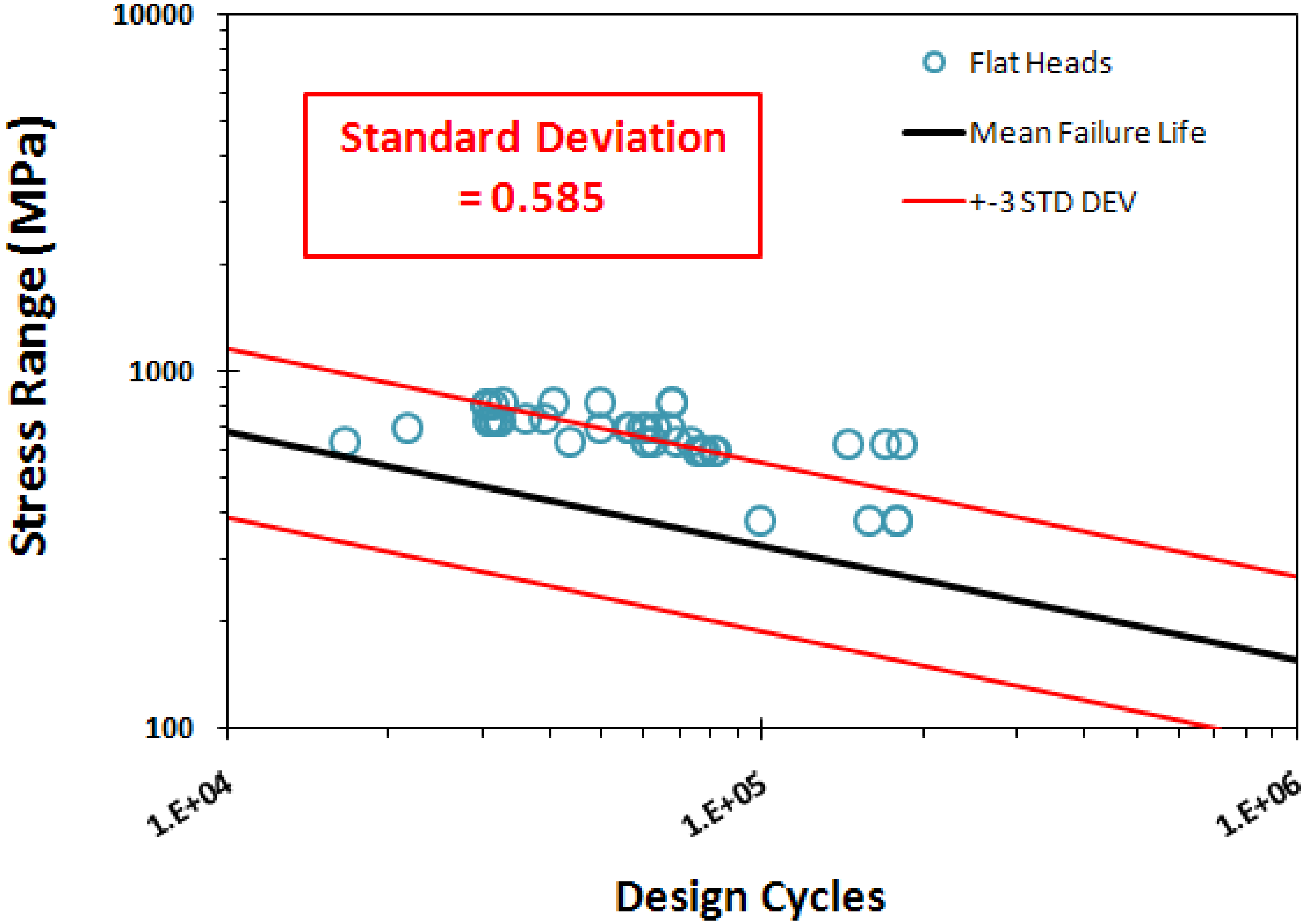
Experimental Validation – FSRF Method

- The following comparison shows the failure prediction using the ASME smooth bar method with an FSRF of 1.7.
- A standard deviation of 0.14 shows a good fit.
- The ASME FSRF of 1.7 is slightly less than the best-fit FSRF of 1.8.



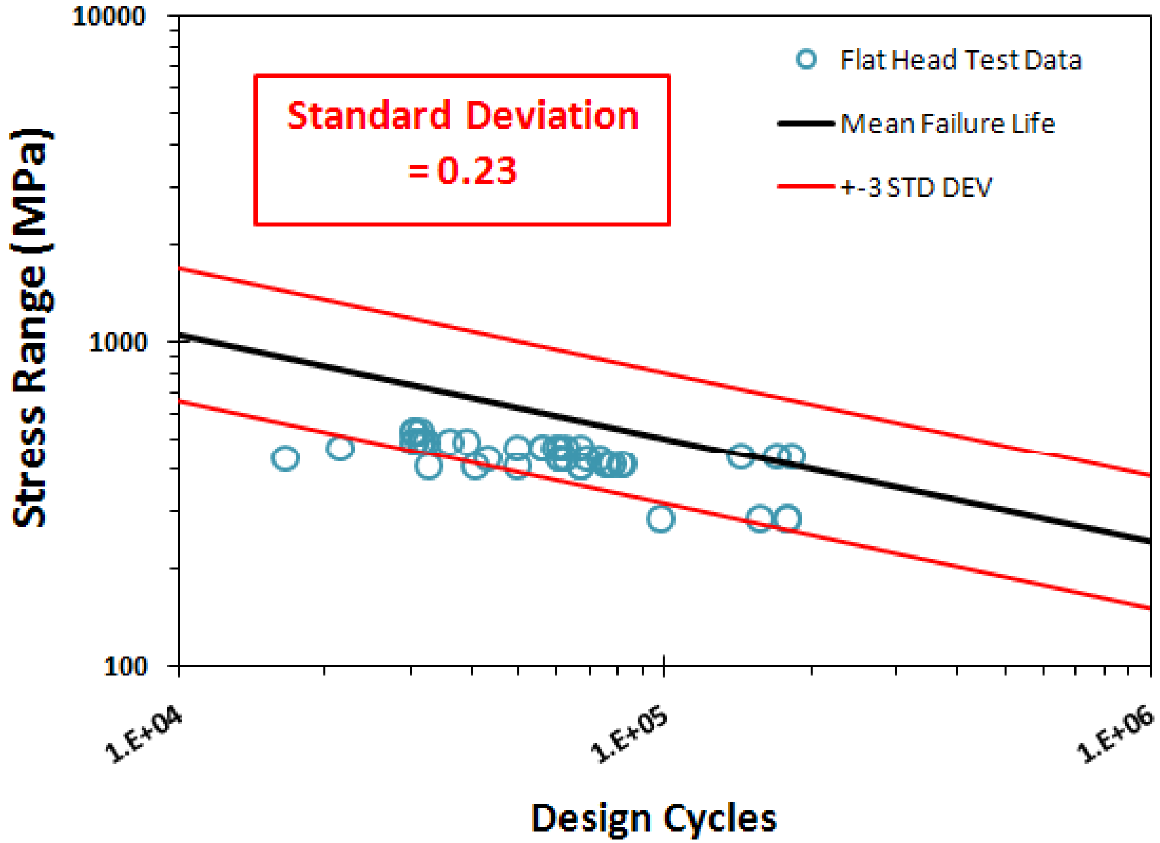
Experimental Validation – New Master SN Method

- **The following comparison shows the failure prediction using the new Master SN Method per 5.5.5.**
- **The standard deviation for the test data is 0.585**



Experimental Validation – Original Battelle Master SN

- **Nearly all PRG data is over predicted by more than one standard deviation. Only stainless data is near mean curve.**
- **From a statistical view, data should be distributed about the mean fatigue curve, not skewed to only a single side.**
- **Standard deviation is slightly greater than the FSRF method**

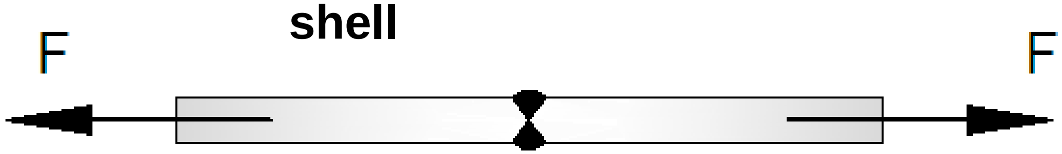


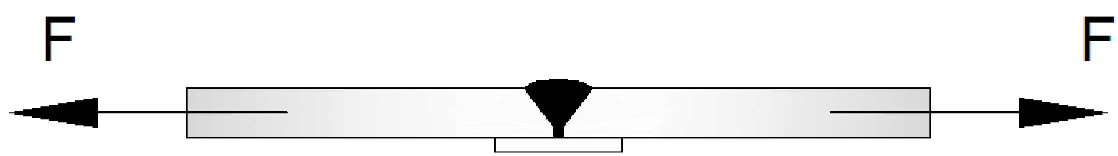
Comparison of Various

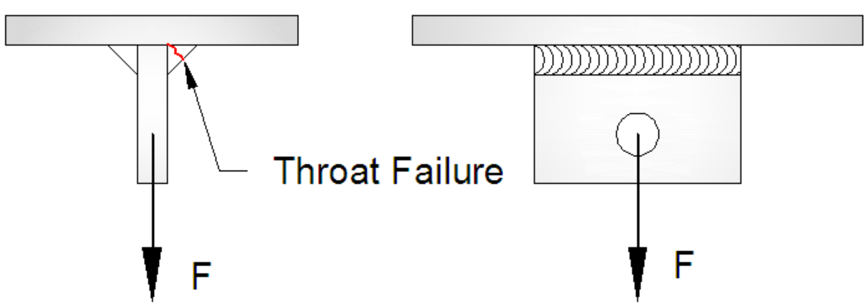
Fatigue Design Methods

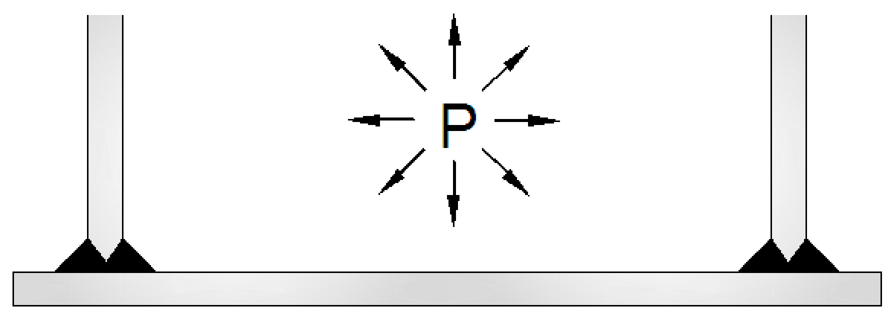
Overview of Weld Joints to be Compared

- The following common weld joints will be compared:
    - Circumferential double-sided butt weld
    - Circumferential single sided butt weld with a backing strip
    - Fillet weld subject to weld throat failure
    - Full penetration weld between a flat head and cylindrical shell







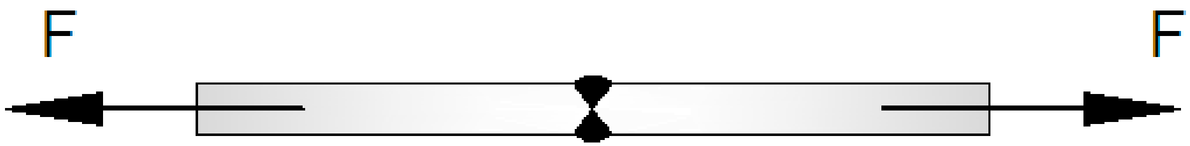


Overview of Fatigue Codes

- **Comparisons will be made using the following fatigue Codes:**
  - **2007 ASME Section VIII-2**
    - **Fatigue Strength Reduction Factor (FSRF) method**
    - **Master SN Method**
  - **PD-5500**
  - **EN-13445**
- **We'll assume carbon steel unless otherwise noted.**
- **An environmental factor of 4.0 on life is used for all Codes. This will ensure a consistent basis with the ASME smooth bar curves that inherently include an environmental factor of 4.0.**
- **It is assumed that the joints are inspected by RT/UT and a surface inspection method (PT, MT, or VT).**

Example 1 – Butt-weld in a cylinder

- **Consider the following example of a circumferential butt-weld in a cylindrical shell subjected to an axial secondary load (restrained thermal).**
- **This is a typical loading condition in fixed tube heat exchangers and all piping and pressure vessel systems.**

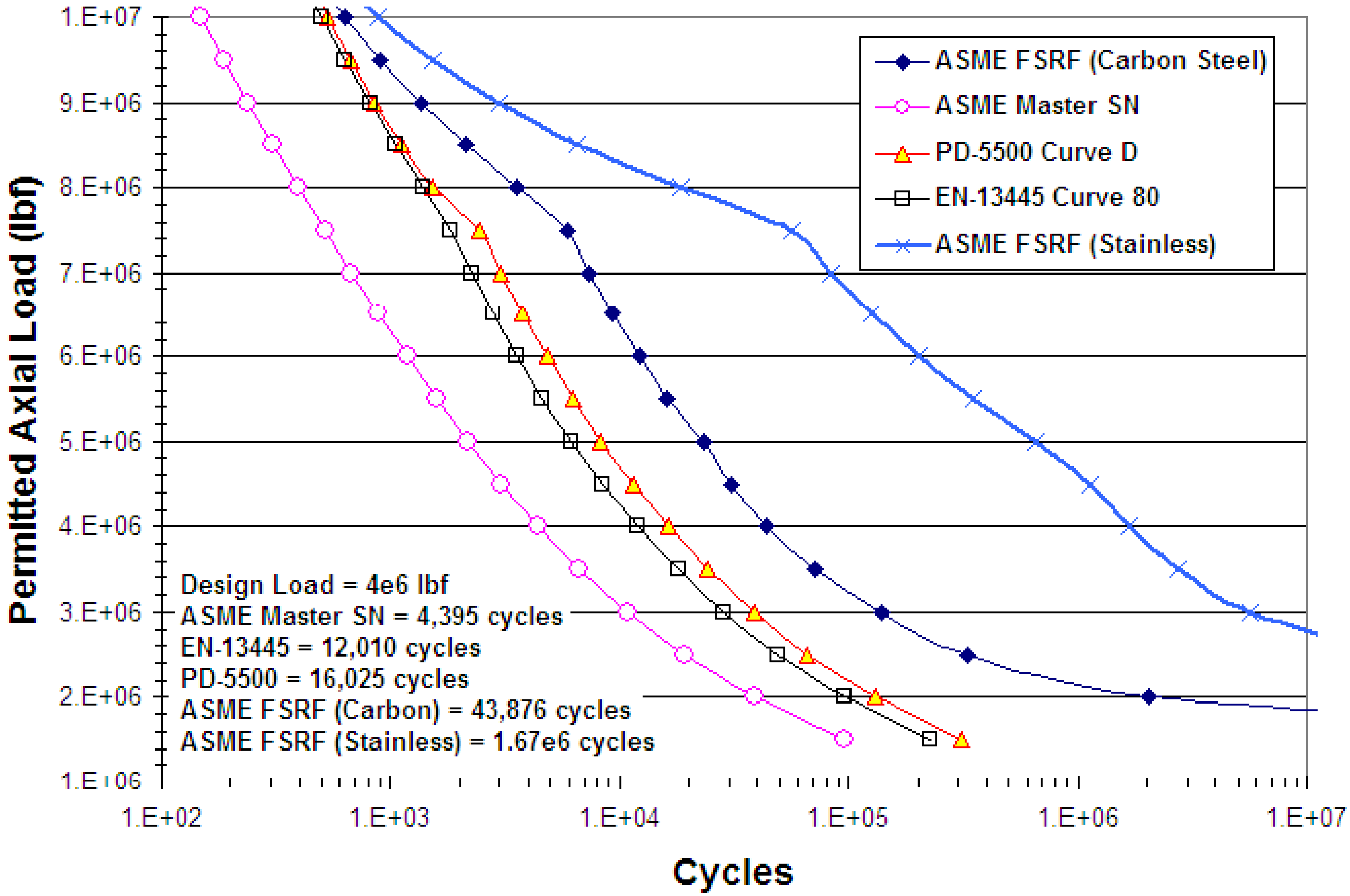


- **Since this is a very simple joint we “analyze” it by hand :**
  - **Nominal stress = Structural Stress = Hot Spot Stress = M+B Stress**
  - **Peak Stress = (Nominal stress x FSRF) / 2.0**
  - **Equivalent Structural Stress begins with nominal stress to determine the equivalent structural stress.**

F = Cyclic Axial Load = 4e6 lbf
A = Shell Cross Sectional Area = 100 inches
Wall thickness = 1.0 inch
Environmental Factor = 4.0 on cycles

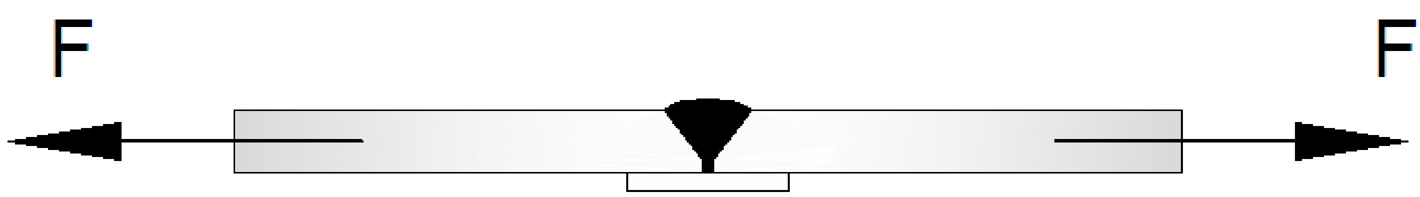
Nominal Stress Range = F/A = 40,000 psi
Alternating Peak Stress = (Nominal x FSRF) / 2.0 = (40,000 x 1.2) / 2.0 = 24,000 psi
EQ Structural Stress = (1.09*24,000) / (1.0*1.22*1.0) = 21,590 psi

Example 1 – Butt Weld SN Comparisons

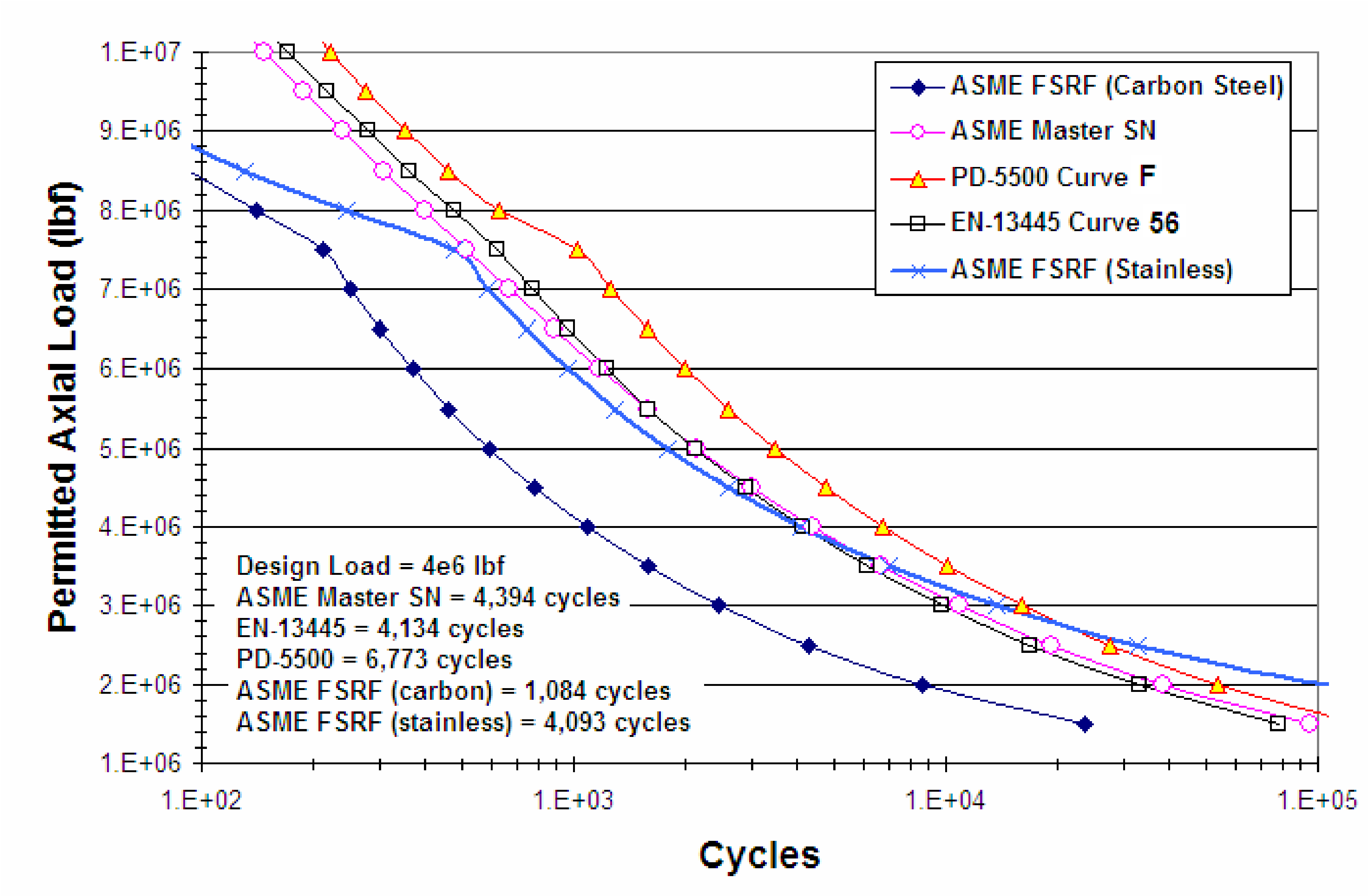


Example 2 – Butt-Weld with Backing Strip

- **Same as Example #1, but this time using a single sided butt weld with a backing strip.**
- **These welds have often been associated with poor fatigue performance, although test data does suggest they may have similar qualities as double sided butt welds (Markl, Maddox, etc) if properly completed.**
- **Stresses are the same as before since the backing strip and weld overfill doesn't significantly alter the local stress distribution.**
- **For this case, the following fatigue design parameters are used:**
    - **ASME Master SN – Same curve for all joints**
    - **ASME FSRF = 4.0 (can't inspect the root at backing strip)**
    - **PD-5500 : Curve F**
    - **EN-13445 : Curve 56**



Example 2 – Butt Weld with Backing Strip

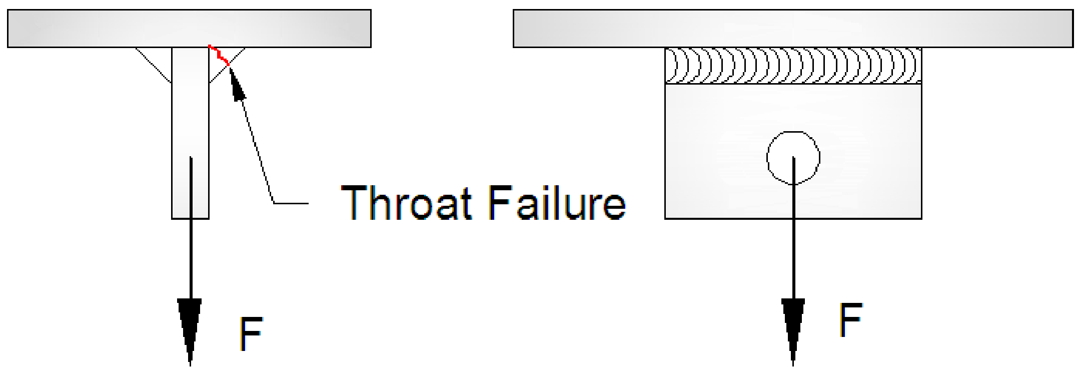


Observations for Butt Welds…

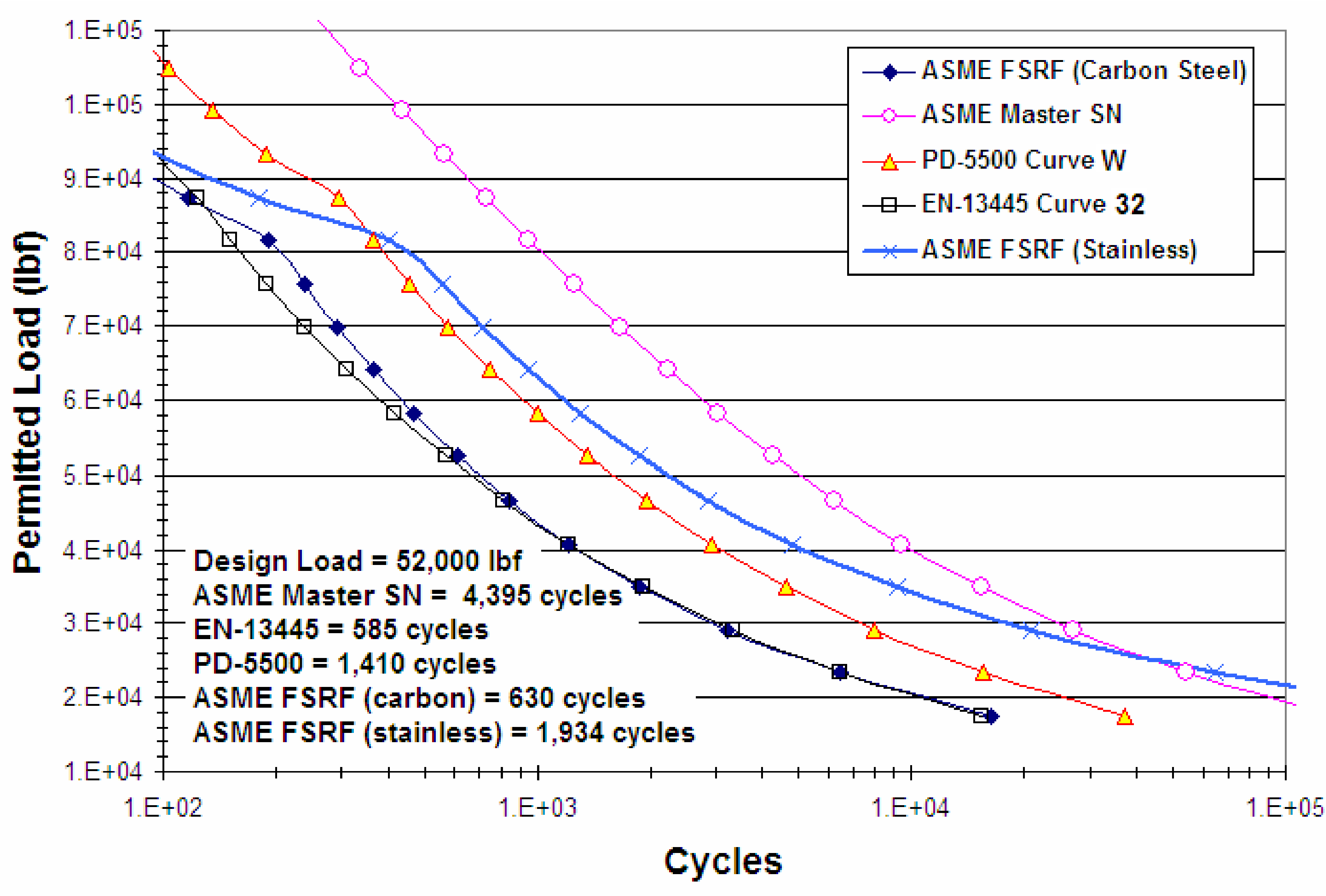
- **For a very simple butt-weld joint there is considerable difference in the allowed cycles for the various Codes. Should we expect better agreement among Codes for such a simple weld?**
- **ASME methods vary by a factor of 40 to 380 for butt welds.**
- **There is better agreement for a butt-weld with a backing strip.**
- **ASME FSRF method with stainless steel SN charts allow for increased design fatigue life. All other methods assume that CS and SS behave equally.**
- **ASME Master SN allows same life for butt-welds with/without backing.**
- **Differences in fatigue life between double sided butt welds and single sided butt welds with backing strip:**
    - **ASME Master SN = no difference**
    - **ASME FSRF (carbon steel) = 40 time decrease on life**
    - **ASME FSRF (stainless) = 408 times decrease on life**
    - **EN-13445 = 2.9 decrease on life**
    - **PD-5500 = 2.4 decrease on life**

Example 3 – Fillet Weld Throat Failure

- **A common pipe support is a flat plate lug that is fillet welded to a vessel, structure, or another pipe.**
- **For this case, the following fatigue design parameters are used:**
  - **ASME FSRF = 4.0 (can't inspect the root of the fillet weld)**
  - **PD-5500 : Curve W**
  - **EN-13445 : Curve 32**
- **For PD-5500, and EN-13445 the average stress on the weld throat is used. (S=F/2a)**
- **For the ASME Master SN method, the equivalent structural stress is calculated based on linearized M+B stress on the weld throat.**



Example 3 – Fillet Weld Throat Failure

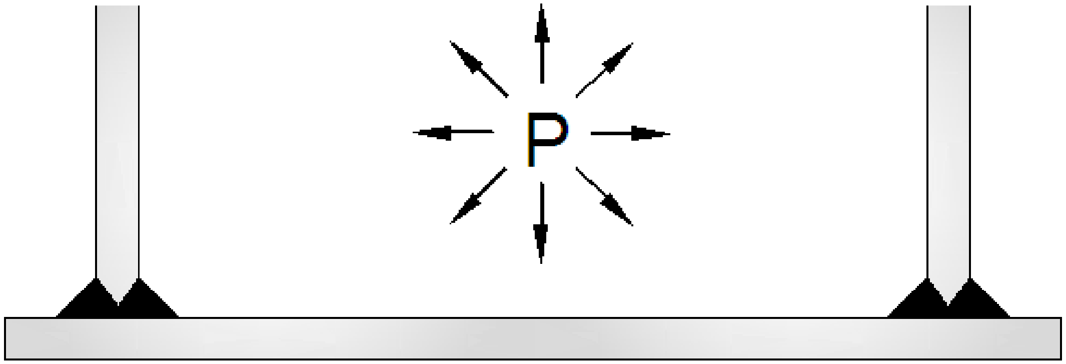


Observations for Fillet Welds…

- **The various fatigue Codes predict a wide range of allowed cycles for the fillet weld throat failure.**
- **ASME FSRF method and EN-13445 agree fairly well.**
- **For the same equivalent structural stress, the ASME Master SN method allows the same cycles as the previous examples (butt weld, butt weld with backing strip).**
- **ASME code rules provide a variation of nearly 7 times on life between the FSRF method and the Master SN method.**
- **In general, you should design the throat thickness such that the maximum stress occurs at the weld toe – not the weld throat.**
- **However, for fatigue service the “rule of thumb” for the fillet weld throat thickness = 0.707*plate thickness may not be sufficient. Recall that the fatigue life is related to the stress range by a power law. Therefore, it might be necessary to increase the weld size by as much as 2.5 times (based on EN-13445 curves 80 & 32).**

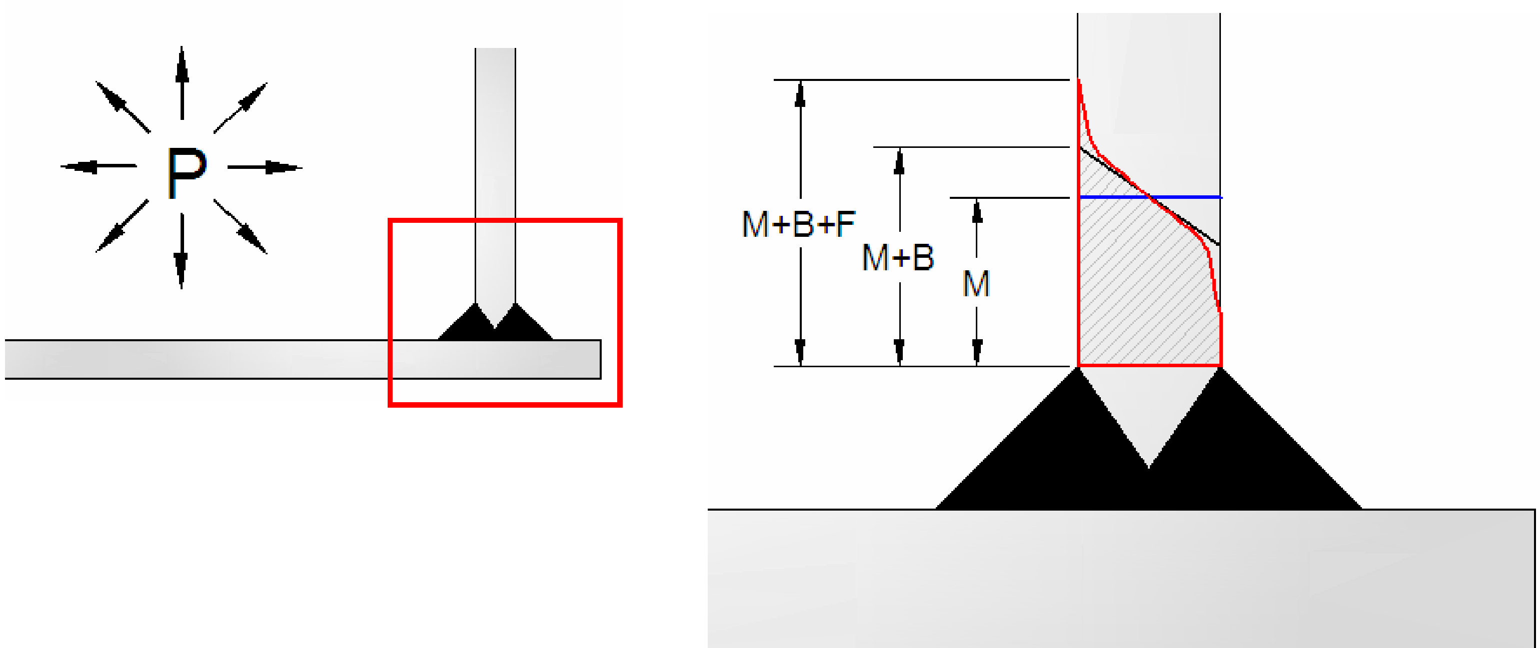
Example 4 – Cylinder to Flat Head Weld Joint

- **The next example is a cylinder attached to a flat head with a full penetration weld and cover fillet welds.**
- **These joints are common welds at heat exchanger tube sheets.**
- **The "applied load" is cyclic internal pressure which causes a bending stress in the attached cylindrical shell.**
- **The following SN curves are used:**
  - **ASME FSRF = 2.0, PD-5500 = Curve F, EN-13445 = Curve 71**
  - **ASME Master SN Method**

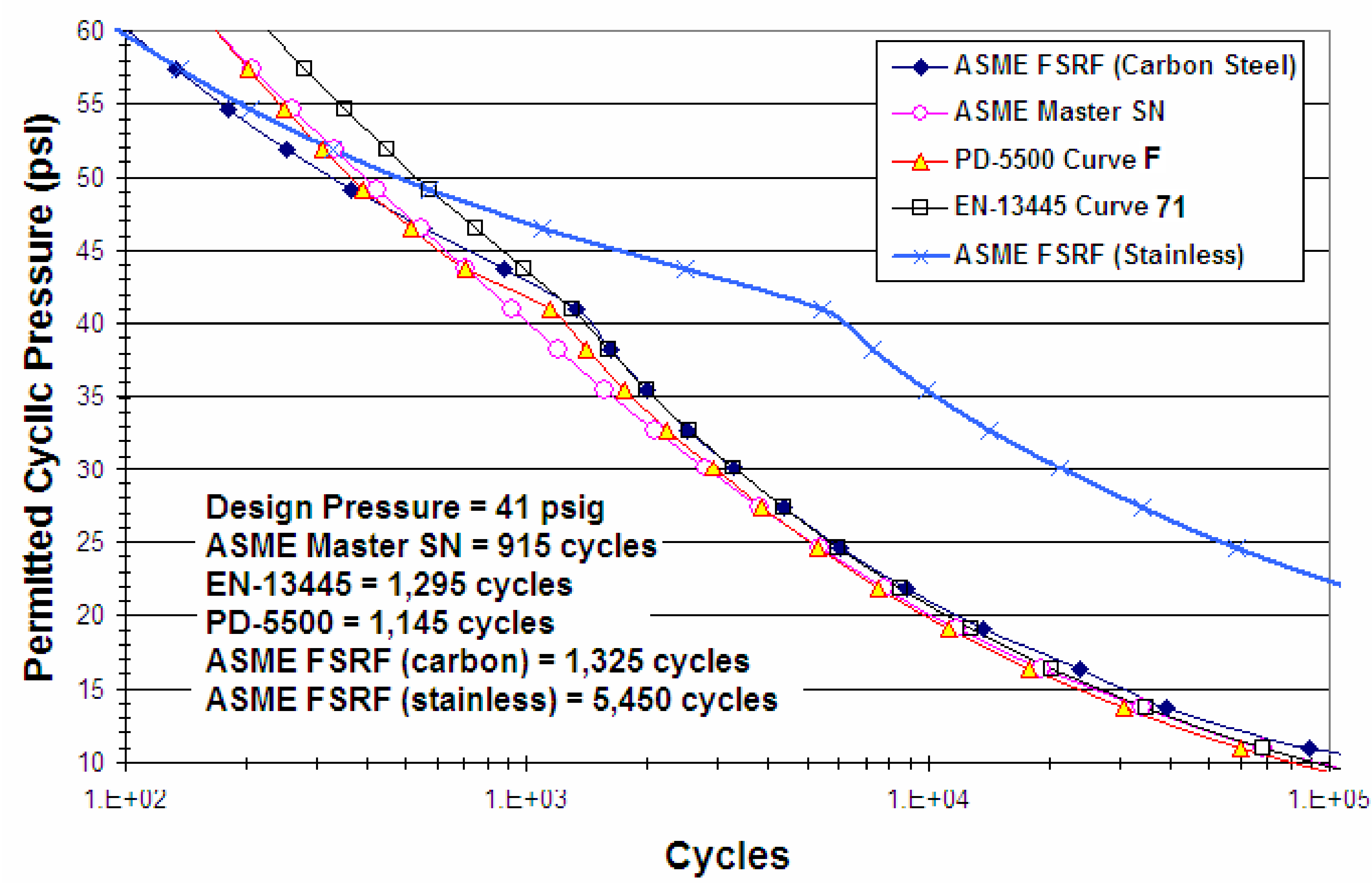


Example 4 – Cylinder to Flat Head Weld Joint

- **For this problem, we need to use stress linearization to determine the M+B stress acting normal to the weld toe.**
- **For ASME FSRF method we can also use the secondary equivalent stress (considers all stress tensors, not just those normal to the weld toe).**
- **In this specific case the EN-13445 Hot-Spot stress will give essentially identical solutions as the stress linearization.**

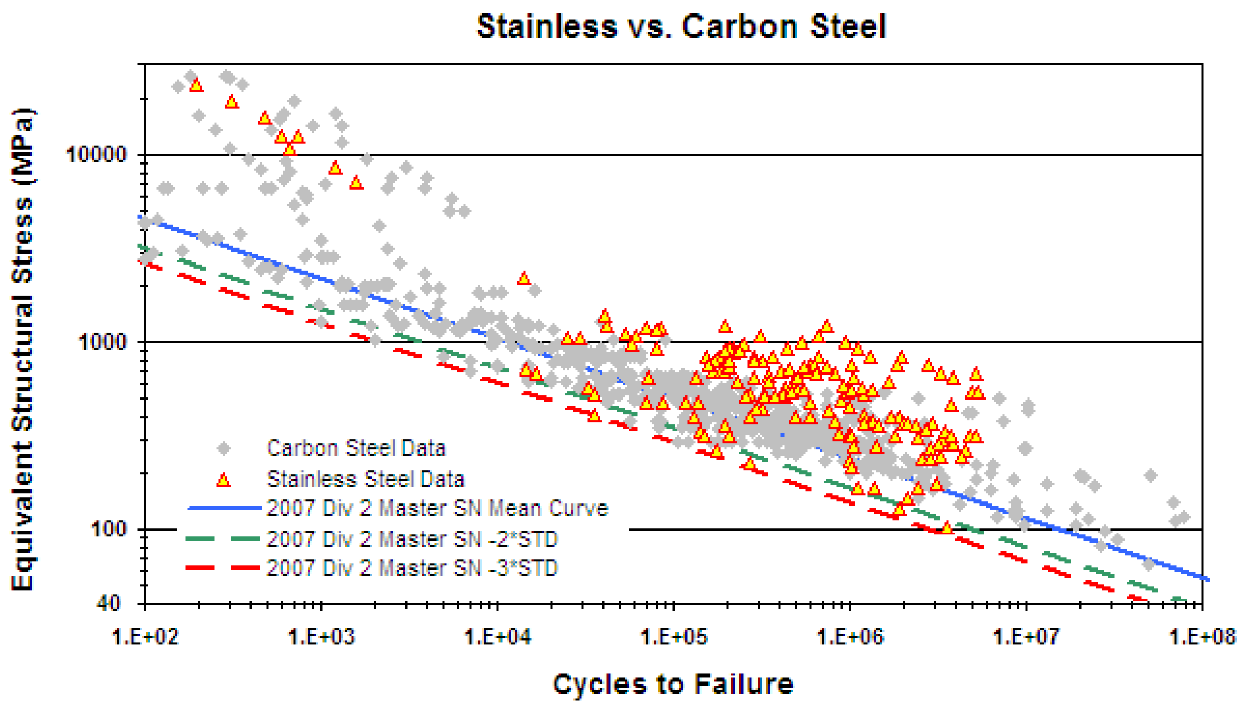


Example 4 – Cylinder to Flat Head Weld Joint



Should ASME Allow a Higher Life for Stainless?

- **As you have seen so far, the ASME FSRF method will allow more cycles for stainless than carbon steel weldments.**
- **Most as-welded fatigue methods do not allow an increased fatigue life for stainless steel construction.**
- **ASME has a long history of allowing an extended operating life for stainless steel equipment in Section III and Section VIII.**
- **Some data suggests an extended life for stainless can be expected:**



Why do the Codes Vary?

Just a few of the reasons fatigue design curves vary:

- **Each Code has a unique background and set of experimental data from which it is derived. Variations in test data and interpretation may produce different design curves.**
- **Design margins vary between the Codes: -2*Std Dev, -3*Std Dev, 2 & 20, etc.**
- **"Codification" of the fatigue method may change the accuracy of the method. Recall in Part II we discussed how the ASME implementation of the Master SN method may shift the data by two standard deviations.**
- **The basis of the methods can be different – for instance ASME smooth bar method vs. as-welded fatigue curves.**
- **Assumptions – the literature is filled with documented assumptions regarding the development of fatigue rules. For instance, in the ASME B31 piping rules, Markl assumed that all welded component curves follow the same slope as unwelded pipe.**

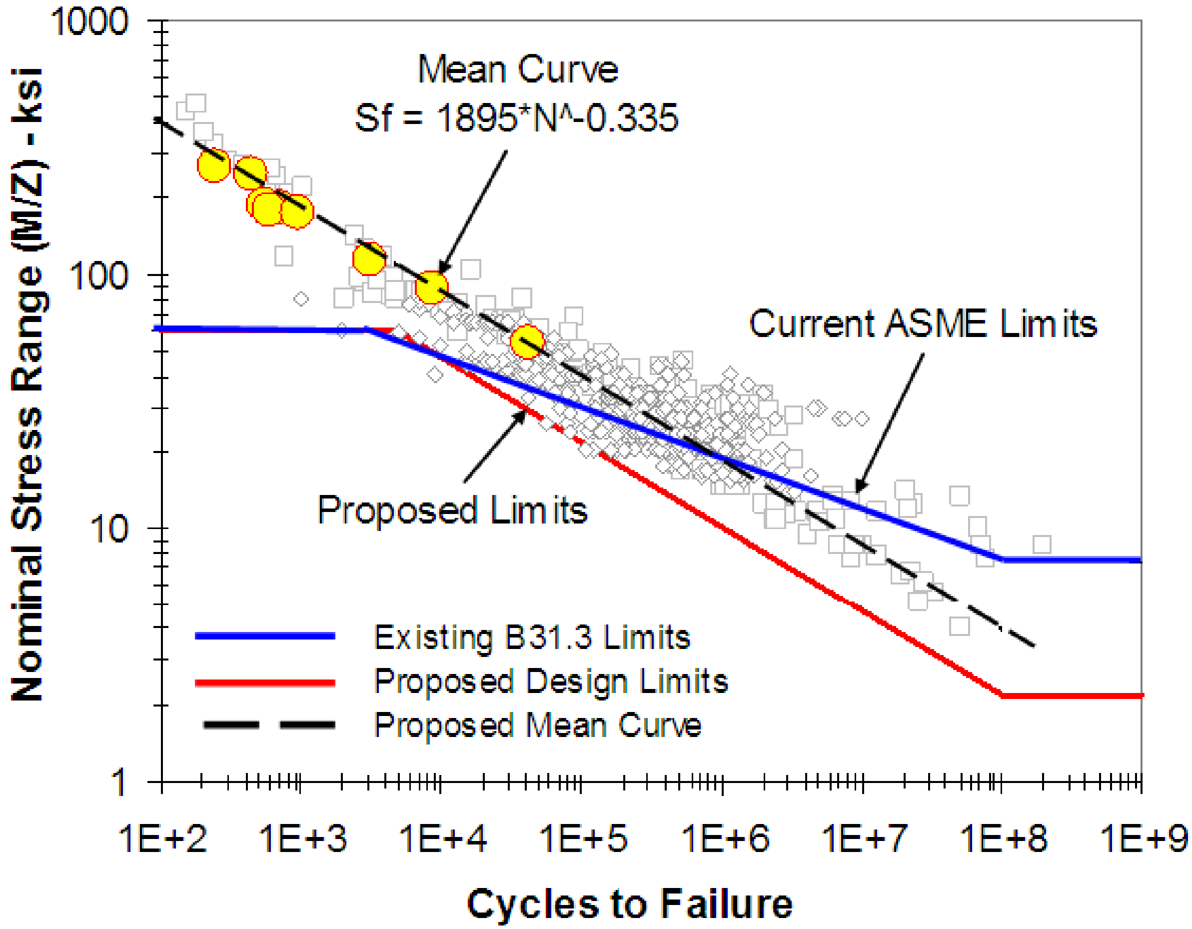
Considerations for ASME B31 Piping Analysis

Piping Fatigue Analysis

- **Be careful in any case where:**
  - **D/T > 80 or d/D > 0.70**
  - **Piping exceeds 6” in diameter**
  - **Rotating equipment**
  - **Any system where friction affects code compliance**
  - **Refractory lined systems**
- **It is recommended that supplemental SIF’s and flexibilities are used in any of the above cases.**
- **SIF’s may be in error on the order of 2.0 to 5.0 times.**
- **Flexibilites may be in error more than 10 times.**
- **In heavily cyclic systems, consider increasing design margins.**
  - **Recent findings indicate that the B31 fatigue curves may not be appropriate for more than 80,000 design cycles.**
  - **The slope of the fatigue curve in B31 piping is 1:5, welds typically show a 1:3 slope**
- **See the following PVP paper for additional details: PVP2008-61871 “Experimental Evaluation of the Markl Fatigue Methods and ASME Piping Stress Intensification Factors”, C. Hinnant and T. Paulin**

Piping Fatigue Analysis

- The following shows the comparison between 800 fatigue data points and the ASME B31.3 design curve.
- The ASME B31.3 design code does fit well with the available data.



Piping Fatigue Analysis – The Next Generation

The next generation of piping analysis technology is now available and includes:

1. **Integrated and automatic FEA analyses of intersections to determine accurate SIF's and stiffnesses.**
2. **Database of FEA results for SIFs and stiffnesses of intersections.**
3. **18 DOF elements to properly analyze flexibile systems where ovalization of the piping near bends, intersections, and supports is important.**
4. **Path dependent friction for realistic loadings.**
5. **Accurate refractory lined piping models with correct stiffness**

Contact Paulin Research Group for details on the piping analysis software "PCL Gold" that incorporates these technologies.

General Considerations for Fatigue Design

Examples of Where Fatigue Failures Occur

- **In general, at any structural discontinuity**
- **For external loads, generally at the toe of the weld.**
- **Inside corner of nozzle openings for cyclic pressure.**
- **Saddle, pipe shoes, stops, lugs, trunnions, staunchions on elbows, etc.**
- **Skirt supports (differential thermal expansion between vessel and skirt)**
- **Agitator nozzles, piping with rotating equipment**
- **Jacketed piping and pressure vessels**
- **General weld flaws:**
  - **Misalignment of plates at butt welds**
  - **Bimetallic welds**
  - **Welds with over\under strength in comparison to base metal. Try to closely match, if possible.**
  - **Lack of fusion**
  - **Cold lap**
  - **Undercut**
  - **Overfill**
  - **Crack-like flaws**
  - **Root of fillet**

Fatigue Design for Section 8, Div 1 Vessels

- **Very often, fatigue design is not considered for Division 1 equipment. This oversight leads to unnecessary failures.**

- **ASME VIII-1 does not include fatigue design rules.**
- **Subgroup Design VIII is working on a simplified fatigue analysis method.**
- **Currently, the user is referred to U-2(g).**
- **U-2(g) allows the user to pick any reasonable method for fatigue analysis. Most calculations use ASME VIII-II Part 5.**
- **Other recognized methods may be used if desired.**

Items Often Overlooked in Fatigue Design

- **Division 1 vessels – often not designed for cyclic service when they should be.**
- **ASME B31 piping with cyclic pressure (not address by the Code)**
- **Accuracy and limitations of B31 piping SIFs and flexibility factors**
- **Thermal shock, thermal transients**
- **Wind vibration**
- **Operating environment**
- **Elevated temperature service – creep/fatigue interaction can significantly reduce fatigue life**
- **Proper selection of fatigue curve or FSRF**
- **<u>Correct</u> inspection at critical welds**
- **Rain bowing – piping can be subject to large bending stresses as a result of rain cooling**
- **Limitations of WRC 107 and WRC 297**
- **Correct assumptions in stress models**
- **Acoustic problems in piping**
- **Alterations or failures of piping support**
- **Changes in operation (decreased cycle time, different heating methods, increased pressure, etc)**

Avoiding Fatigue Failures

1. Be proactive. If you expect more than 500 pressure cycles, or 200 thermal cycles, always specify a fatigue analysis is required.
2. Emphasize inspection. Inspect all highly stressed welds. Emphasis should be on visually appealing welds since internal flaws typically have less influence on fatigue life.
3. Position welds in low stress areas.
4. Improve weld toes. Toe grinding, TIG dressing, peening are a few method used. Make sure the operator is experienced.
5. Use generous radii at the inside corners of nozzle openings.
6. Avoid single sided welds when possible. In cyclic service, use fillet welds as a last option.
7. In an analysis, never use an FSRF less than 1.2, even for smooth flush ground welds.
8. Double check any critical fatigue calculation with an independent method. In the new Div 2, vendors may be tempted to use the most “convenient” method (i.e. the largest design life). Results can easily vary by a factor of 10 on design cycles.
9. Reduce thermal transients by extending the duration of the transient or reducing the temperature differences.
10. Keep the operating environment in mind. Often, metallurgical damage accelerates fatigue damage. Decrease the design cycles where appropriate to consider the environment.
11. Keep an eye on the system. Look for changes in supports, new vibration, etc., especially after a shutdown. Make sure that supports are acting as intended.

After a Fatigue Failure

1. **Gather as much information about the operating history as possible.**
   - Number of cycles to date
   - Environment (process)
   - Operating histogram with pressure, temperature, and other pertinent loads (piping, dead weight, liquid head, etc). DCS dumps are very handy.
   - Any critical events such as fires, extreme upset conditions, wind, earthquake, hurricane, etc.
   - Corrosion surveys
   - Take new measurements where useful (thermocouples, strain gauges, IR imaging, etc).
2. **As-built drawings and any field modifications**
   - Make note of any modified or damaged supports. Also, supports that aren't "supporting".
   - A fresh walk along piping lines is a good idea to check for any abnormalities.
   - Have lines moved?
   - Has there been foundation or support settlement?
   - Movement of equipment at the base.
3. **History of equipment**
   - Original calculations, design code, etc.
   - Other failures
   - Shop and field repairs
   - Post weld heat treatments (shop and field)
   - Any rerates
   - Talk to the "old timers" and those familiar with the operation and equipments
4. **Plan of action**
   - Evaluation and calculations – do you need an expert?
   - Metallurgical evaluation
   - Alterations and repairs – develop a procedure for field work
   - Experimental testing (not as expensive as you may think, or as expensive as another failure).
   - Modify operating conditions (chemistry, batch duration, transient durations)

Thanks!

Thank you for attending this presentation.

If you have any questions, please feel free to contact us:

Chris Hinnant – chris@paulin.com

Paulin Research Group

www.paulin.com

Houston, TX

281-920-9775