Review of the Master SN Neuber Rule
in the
ASME Division 2 Rewrite Project

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1.0 Introduction

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.

The idea that plasticity corrections are not required for welded fatigue curves is supported by current design rules in ASME where fatigue curves are based on welded fatigue tests (B31 codes) and no plasticity correction factors are employed. As Rodabaugh points out in NUREG 3243, there is no need to apply plasticity correction factors for welded fatigue curves based pseudo elastic stresses. The primary reason is that the testing methods using linear elastic extrapolations are analogous to the linear elastic analysis methods used for design work with displacement controlled conditions.

> The test method is consistent with an elastic analysis of a piping system, even though calculated stresses may be above the material yield strength and some plastic deformation may occur. Accordingly, an adjustment analogous to the K_e used in Code 1 is not needed.

Pingsha Dong, developer of the Master SN method, also provides support for the idea that the Neuber correction is not required when displacement controlled conditions are analyzed. The following statement is taken from the special ASME Fatigue Forum in Columbus, OH (Dec. 2006):

- The Neuber-based procedure for estimating pseudo elastic structural stress in low-cycle regime is only needed when performing linear FEA when pseudo elastic load or stress is not known

As will be shown, applying Neuber's rule to displacement controlled analyses will lead to error in the fatigue calculations and is not required as part of the fatigue design process. Several examples and illustrations will be used to support the conclusion that a Neuber's adjustment is unnecessary and inappropriate:

1. A simple flat plate with a circular hole will be examined. This example will show that the Neuber adjustment will lead to significantly reduced fatigue life predictions for fatigue curves based on pseudo elastic stress and notched fatigue specimens.
2. Using fatigue test data from WRC 433, the Neuber's approach is illustrated to grossly under predict the mean fatigue life for a fatigue point which originally falls nearly directly on the Master SN mean curve (the predicted life is significantly less than the actual failure life).
3. Supporting evidence in terms of how welded fatigue data is used in existing ASME fatigue codes and literature including commentary by Rodabaugh, Pingsha Dong, and original work on the application of Neuber's rule by R.M. Wetzel.

2.0 Description of Example

The goal of this section is to describe the fatigue testing procedures which would be used to produce a fatigue curve where the stress ordinate is the pseudo elastic nominal stress. The final result will be a fatigue curve which relates the pseudo elastic nominal stress to the mean cycles to failure. Nominal stress is used for simplicity but these concepts are equally applicable to the structural stresses at the hole.

A simple case will be discussed here to illustrate the process by which fatigue curves based on displacement controlled test data and pseudo elastic nominal stress should be used. The example used is a flat bar with a circular hole in the center of the plate as reported by R.M. Wetzel (Ref. Figure 1). Fatigue test results for this exact geometry by R.M. Wetzel are also included to help illustrate the concepts.

Before proceeding into the details of this example, there is a very subtle, yet critical point to this discussion:

> *Fatigue curves for weldments or notched specimens which are based on pseudo elastic nominal or structural stress inherently contain plasticity effects and therefore do not require any plasticity adjustments when used with displacement controlled loadings in an analysis. This is not the same as when smooth bar curves are used for fatigue design in which case a Neuber's correction must be used no matter the loading type.*

The flat plate specimen discussed here is a standard detail widely studied in the application of Neuber's rule in the prediction of local strains for low cycle fatigue using smooth bar fatigue curves. Typically, Neuber's rule is used when predicting the fatigue lives of notched specimens with smooth bar fatigue curves. However, in this application, we'll illustrate how the pseudo elastic stress fatigue curve is generated and why applying the Neuber curve to a notched fatigue curve using pseudo elastic stress is inappropriate.

Although this not an example of a weldment, the concepts illustrated for the flat plate specimen are representative of any component with a "notch", welded or machined. Therefore, the procedures and discussion are equally applicable to the fatigue testing and design of weldments subjected to displacement controlled conditions.

Note that the pseudo elastic nominal stress described here is the nominal stress across the net section at the center of the hole [Se = Fe / (Thk * (Plate Width – Hole Diameter))]. Therefore, the stresses to be used for this fatigue curve are the pseudo elastic nominal stress determined from applied displacement conditions. It is very important to be clear that the design fatigue curve is based on the pseudo elastic nominal stress derived from a linear extrapolation of the elastic system stiffness curve and not smooth bar curves.

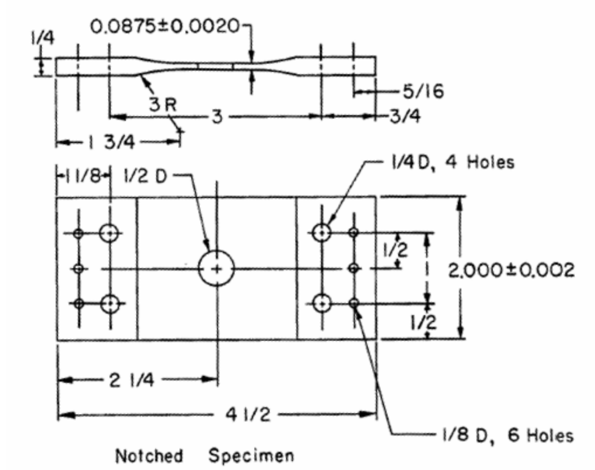

Figure 1 – Illustrative example.

Before the fatigue curves can be established, the fatigue database would need to be generated for a sufficient number of test specimens. The basic process for generating the fatigue database would involve the following steps, the most important point being that displacement controlled conditions are utilized:

1. Generate a load-deflection curve for the test specimen. (Ref. Figure 2).
2. Using linear elastic portion of the load-deflection curve, calculate the linear system stiffness "K". (Ref. Figure 3).
3. Select a displacement to be cyclically applied during the fatigue test, "Da".
4. Perform the fatigue test using the controlled displacement "Da" and record the cycles to failure "Nf".
5. Calculate the pseudo elastic force "Fe" using the relationship Fe = K*Da.
6. Calculate the pseudo elastic stress "Se" using a linear elastic model and the pseudo elastic force "Fe". In this case, "Se" is the nominal stress across the net section at the hole.
7. Plot the fatigue test result into the S-N chart using the cycles to failure "Nf" and the pseudo elastic stress "Se". (Ref. Figure 4)
8. Repeat steps #1 thru #7 for all specimens. (Ref. Figure 4 for all specimens in SN curve)

Now that a hypothetical fatigue design curve has been established, the next section will focus on how the design curve should be used.

Table #1 – Summary of fatigue test results for flat plate specimen by Wetzel.

Specimen #	Cycles to Failure	Displacement (inches)	Applied Load (lbf)	Pseudo Elastic Load (lbf)	Pseudo Elastic (ksi)
	Nf	**Da***	**Fm**	**Fe**	**Se**
1	90	7.67E-02	7941	31000	236.2
2	2,400	3.59E-02	6956	14515	110.6
3	4,500	1.77E-02	5959	7157	54.5
4	15,000	1.25E-02	4974	5046	38.4
5	32,000	9.79E-03	3977	3955	30.1
6	36,000	9.79E-03	3977	3955	30.1
7	130,000	7.28E-03	2979	2942	22.4
8	400,000	5.35E-03	2192	2160	16.5

* Estimated by elastic plastic analysis using applied loads given by Wetzel.

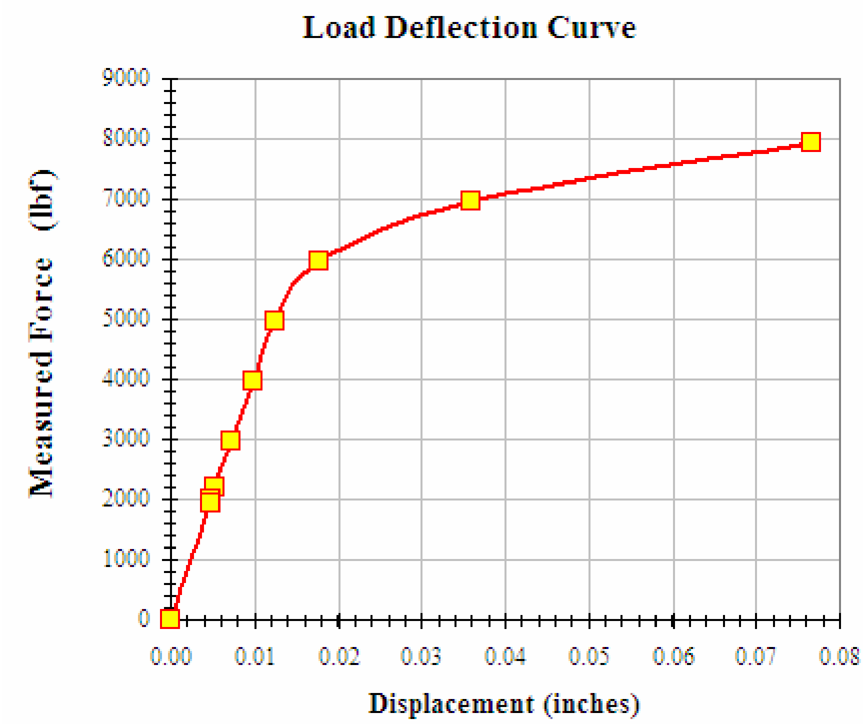

Figure 2 – Estimated load-deflection curve for Wetzel's notched plate specimens.

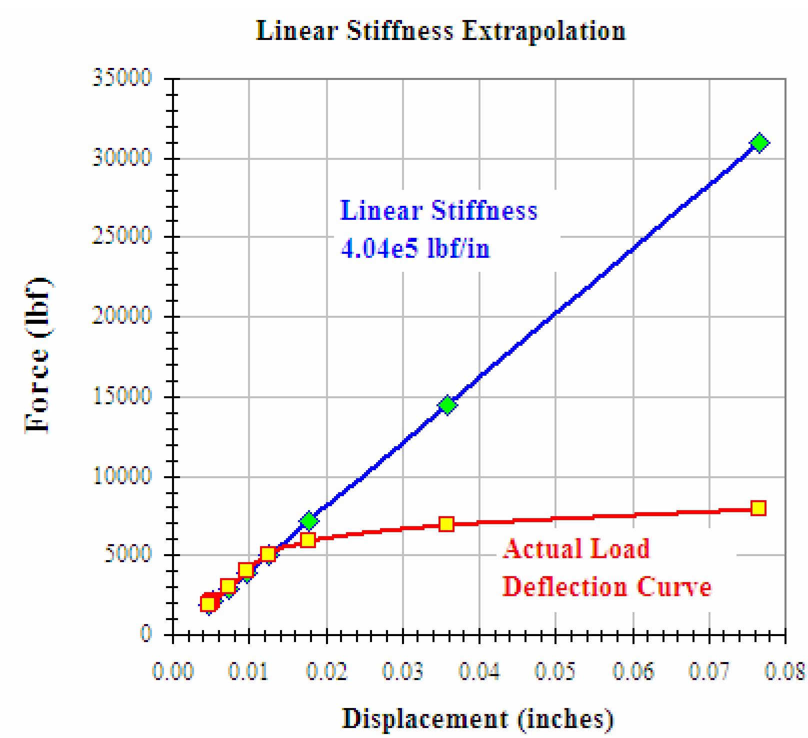

Figure 3 – Linear extrapolation of elastic stiffness from actual load-deflection estimate.

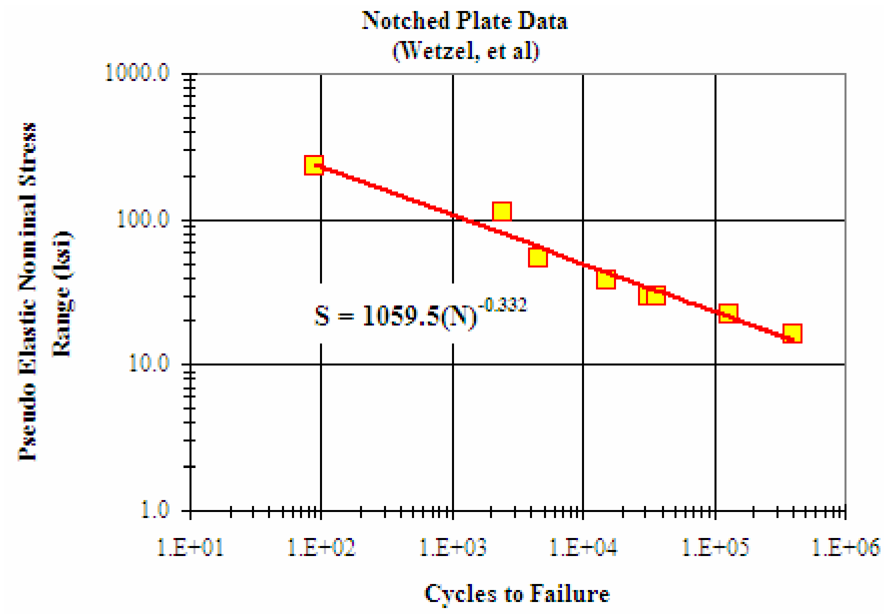

Figure 4 – Pseudo elastic stress fatigue curve.

3.0 Fatigue Design Procedure

In the previous section, a fatigue curve was generated using pseudo elastic nominal stress. As will be shown below, the correct usage of this fatigue curve does not involve the Neuber's correction. In fact, using a Neuber's adjustment with the pseudo elastic force will lead to an error on the order of 1125%. Table 2 provides a summary of the calculations for this discussion which are addressed in detail below.

Table 2 – Summary of hypothetical design calculation results.

Method	Calculated Pseudo Elastic Nominal Stress	Calculated Life	Actual Life	Actual / Calculated
Without Neuber's Rule	236.1 ksi	92 cycles	90 cycles	98%
With Neuber's Rule K = 91,600 psi n = 0.110	530.2 ksi	8 cycles	90 cycles	**1,125%**

To illustrate the use of the fatigue design curve generated by the previous procedures, consider a structure which utilizes the exact components described in the previous procedures (flat plate with a hole). The structure is subjected to thermal loading such that the flat plate detail experiences displacement controlled conditions. For this case, the applied thermal conditions result in plastic strains well beyond the yield point of the material. However, since a linear elastic material model is assumed in analysis, the resulting forces in the analysis are the pseudo elastic forces. That is, the calculated forces are a direct result of the linear extrapolation of the elastic material response.

Furthermore, in this hypothetical design case, the calculated pseudo elastic force happens to exactly match one of the pseudo elastic forces used in the fatigue tests described within the previous section. In other words, the design case contains thermal loading which produce displacements with a resulting pseudo elastic force which exactly match the pseudo elastic force "Fe" applied to one of the fatigue test samples. We'll refer to this calculated pseudo elastic force as "F1". For this example, we'll apply thermal conditions which produce a pseudo elastic force of F1 = Fe_1 = 31,000 lbf, which corresponds to the first fatigue test point presented in the previous section (Ref. Specimen #1 in Table #1).

The hypothetical design procedure could be similar to the following:

1. Develop a linear elastic model of the structure which includes the flat plate with a circular hole. This could be an FE model or in the present case, a simple "hand" calculation.
2. Apply the specified thermal conditions and determine the resultant pseudo elastic force.
    a. *In our example, the resultant pseudo elastic force is F1 = Fe_1 = 31,000 lbf.*
3. Calculate the stress "S1" caused by the applied displacement. S1 = (F1)/(Thk*(Plate width – Hole Diameter)) = (31,000)/(0.0875(2.0-0.50)) = 236.2 ksi.
    a. *The stresses calculated are pseudo elastic stresses since the applied displacements would result in strain beyond yield.*
    b. *Here again, in the hypothetical design case, the thermal conditions result in a pseudo elastic force "F1" which corresponds exactly to a pseudo elastic force range "Fe_1" in the fatigue test database. In this case F1 = Fe_1 given in Table 1.*
    c. *Since "F1" = "Fe_1", the calculated stress "S1" is exactly the pseudo elastic stress "Se_1" determined during the fatigue test and plotted into the fatigue curve.*
4. Without any plasticity correction, enter the fatigue design curve with the calculated pseudo elastic stress "S1" and determine the permitted cycles.
    a. *The mean fatigue curve is given in Figure 4, with equation S = 1059.5(N^-0.332).*
    b. *The calculated fatigue life is N = 92 cycles (which closely agrees with the test result).*

It should now be clear that if the calculated stress "S1" was adjusted for plasticity in Step #4, "S1" would no longer be equivalent to expected value of "Se_1". In fact, the Neuber's adjustment will increase S1 from 236.1 ksi to 530.2 ksi. Correspondingly, the calculated fatigue cycles for the Neuber corrected stress is 8 cycles vs. the expected value of 90 cycles. This represents an error in the solution of 1125%.

The conclusion is that Neuber's rule should not be used with the pseudo elastic force calculated from applied displacement conditions when using fatigue curves based on pseudo elastic nominal or structural stresses. If Neuber's rule is used with pseudo elastic forces, then significant errors can be generated resulting in undue conservatism for fatigue design.

4.0 Illustrative Example for a PVP Geometry

The following is an illustrative example which shows that when the Neuber's rule is applied to the pseudo elastic load, the actual fatigue life is grossly under predicted (result is less than the actual failure cycles). This example is unique in that the fatigue test result plots directly on the Master SN's mean fatigue curve (Ref Figure 5). However, applying the Neuber correction to the pseudo elastic stress calculated from displacement controlled conditions will result in a predicted mean life 40 times less than the actual value. The conclusion here is that the Neuber adjustment is not necessary and leads to inaccurate solutions.

The hypothetical case of the flat plate specimen considered before provides a reasonable illustration that plasticity correction factors are not required for stresses which are calculated from displacement controlled conditions or strain limited stresses and the fatigue curve is based on pseudo elastic stresses from the notched specimens. However, more concrete proof is available by considering actual test results and how the proposed Neuber adjustment would apply. For this example, Specimen #12 from Scavuzzo's girth butt welded piping tests will be evaluated. Specimen #12, constructed from 1-1/2" Carbon Steel pipe, was thoroughly documented within WRC 433 (see Table 3).

As mentioned before, Specimen #12 was chosen because the combination of the applied equivalent structural stress and cycles to failure happen to fall almost directly on the mean curve of the Master SN curve (Ref. Figure 5). The actual cycles to failure was 202 cycles while the mean curve for the applied elastic stress predicts 201 cycles. Therefore, the mean curve coefficients will be used as the focus of discussion. As such, it is expected that the proposed calculations should predict the cycles to failure nearly exactly (given that the test specimen's stress-cycle ordinate lies directly on the mean curve).

As shown in Table #4, the design procedure, <u>without using a low cycle fatigue correction</u>, accurately predicts the cycles to failure. The predicted cycles to failure is 222 cycles while the actual cycles to failure is 202 cycles. Recall that this result is expected since the fatigue failure data point is nearly exactly on the mean fatigue curve of the Master SN database.

However, as shown by the calculations in Table #5, the low cycle fatigue correction significantly increases the equivalent structural stress when the Neuber procedure is applied. As a result, the predicted cycles to failure falls from 222 cycles to only 5 cycles. In comparison to the actual cycles to failure, the mean curve under predicts the fatigue life by 40 times. This is significant given the fact that the actual failure data point exists on the mean fatigue curve at 201 cycles.

As illustrated by the presented fatigue data, applying Neuber's rule to the stresses calculated from the applied displacement range is unnecessary and inappropriate. This has been demonstrated by the fact that the Neuber adjusted stress no longer accurately predicts the failure of a fatigue data point which is nearly on the mean curve of the database. The primary reason the Neuber's correction is not required is that the fatigue curves are generated from displacement controlled results and the calculated design stresses are based on displacement controlled conditions.

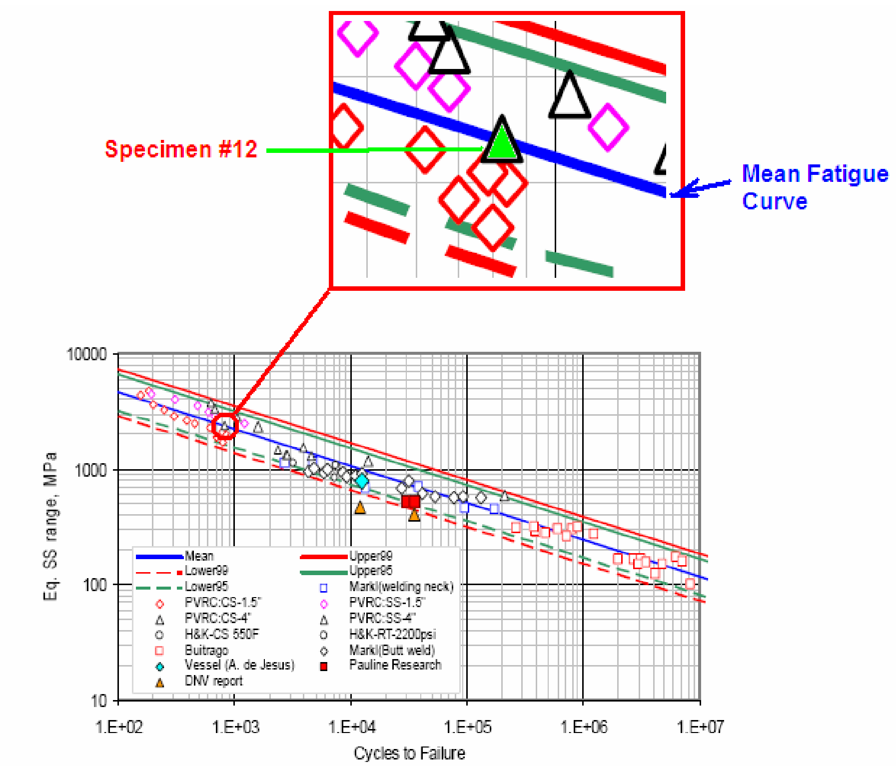

Figure 5 – Location of Specimen #12 on Master SN fatigue chart.
(original S-N diagram taken from Pingsha Dong's PVP 2006 paper)

5.0 Non-Conservatism of the Neuber Adjustment for Applied Loads

As described in Pingsha Dong's PVP 2006 paper, the Neuber's adjustment is intended to predict the pseudo elastic stress using applied loads when the pseudo elastic loads are unknown. One may validate this approach by using the applied or measured load in a fatigue test to calculate the applied stress. Next, the applied structural stress is modified by the Neuber's adjustment to predict the pseudo elastic stress.

However, when significant plasticity occurs, the Neuber's rule is no longer valid and can lead to non-conservative design lives. As shown in Figure 6 and Table 6, the proposed Div 2 Rewrite rules with the lower third standard deviation would lead to design margins in excess of 7 times the actual cycles to failure for Scavuzzo's girth butt weld pipe tests. In other words, the design rules are non-conservative since they over predict the actual failure life.

Note that in Table #6, design margins less than 1.0 imply that failure occurs at cycles less than the design cycles.

As shown in Figure 7, in cases of gross plasticity, the applied loading can not be used to accurately predict the plastic strains or pseudo elastic loads using a Neuber's adjustment. The reason for this is that once the onset of gross plasticity begins, very little additional load needs to generate significant strains. Therefore, the load used to enter the Neuber's hyperbola solution will never be able to predict the actual plastic strain range nor the corresponding pseudo elastic load. There is simply too great of a separation between the applied load and the pseudo elastic load based on the extrapolated plastic displacements. These statements are supported by Wetzel's conclusions that the Neuber's correction does not apply for gross plasticity.

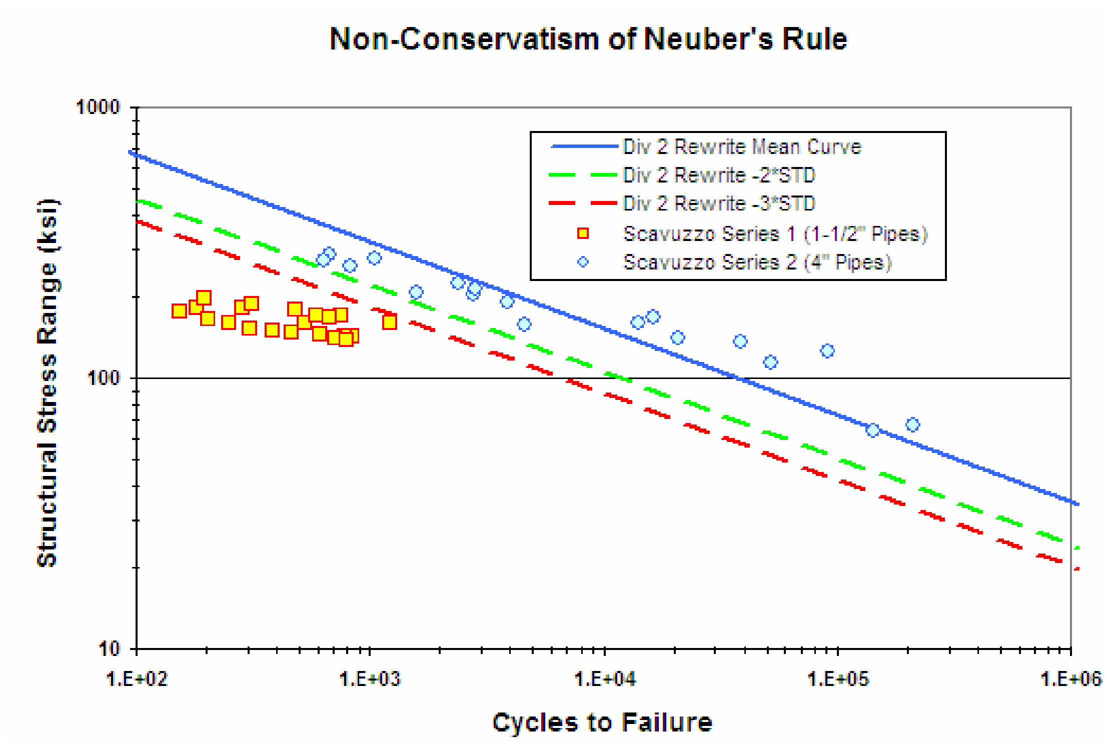

Figure 6 –Fatigue life calculations using applied loads and Neuber adjustment.

Table 6– Non-Conservatism of Div 2 Rewrite Rules for Neuber's Rule

Specimen	Description	Applied Force (lbf)	Applied Stress Range (psi)	Cycles to Failure	Allowed Design Cycles -3*STD	Design Margin < 1 implies failure at less than allowed design cycles.
1	CS – No Weld	2468	90165	523	1620	**0.32284**
2	CS – No Weld	2710	99005	285	1068	**0.266854**
3	CS	2468	90165	249	1620	**0.153704**
4	CS	2407	87955	304	1802	**0.168701**
5	CS	2316	84641	615	2120	**0.290094**
6	CS	2649	96795	155	1184	**0.130912**
7	CS	2271	82983	836	2301	**0.36332**
8	CS	2347	85746	461	2008	**0.229582**
9	CS	2286	83536	786	2239	**0.35105**
10	CS	2377	86850	386	1902	**0.202944**
11	CS	2710	99005	183	1068	**0.171348**
12	CS	2528	92375	202	1457	**0.138641**
13	CS	2256	82431	708	2365	**0.299366**
14	CS	2226	81326	791	2499	**0.316527**
15	SS – No Weld	2607	95272	751	1271	**0.590873**
16	SS	2607	95272	595	1271	**0.468135**
17	SS	2878	105152	195	809	**0.241038**
18	SS	2481	90661	1224	1582	**0.773704**
19	SS	2698	98565	478	1090	**0.438532**
20	SS	2788	101859	311	938	**0.331557**

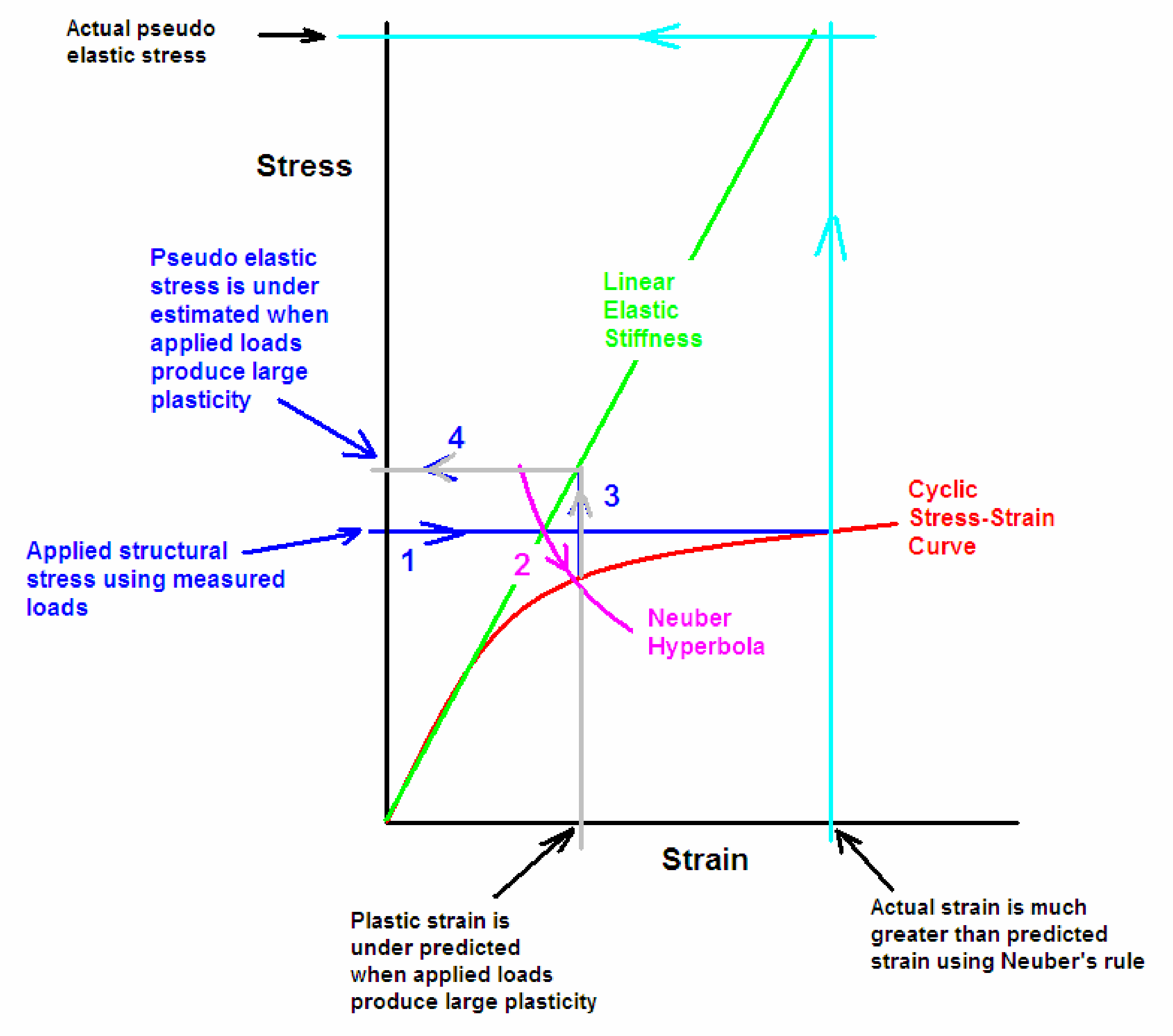

Figure 7 – Illustration of how applied loads can lead to under prediction of the actual pseudo elastic stresses.

6.0 Supporting References and Current Applications

The concept of not using plasticity correction factors when fatigue design curves are based on weldments and displacement controlled conditions is not a new one. Since their inception, most of the ASME piping codes have used a fatigue design approach based on displacement controlled testing of weldments. These codes are primarily based on the fatigue testing of welded piping components conducted by A.R.C Markl. In piping analysis, a primary concern is the stress range due to applied thermal conditions. Since the calculated stresses are based on a linear elastic analysis, these calculated stresses match the definition of the elastic stress taken from the linear extrapolations in the fatigue curves and no plasticity correction is required.

Rodabaugh notes this fact in his evaluation of the Markl based piping codes in NUREG 3243. As shown in the excerpt below, Rodabaugh clearly states that no plasticity correction is required for fatigue design methods based on welded components, unlike the fatigue design methods such as that used in Class 1 nuclear piping ("Code 1") which are similar to the current ASME Division 2 smooth bar methods.

Additionally, in the draft copy of ASME B31.J, a similar statement is made which illustrates that even though plasticity may occur, the linear elastic analyses typically used for piping analysis matches the linear elastic extrapolation procedures used in fatigue testing of weldments.

Another example which illustrates plasticity correction factors are not required is provided in Pingsha Dong's PVP 2006 paper "Low-Cycle Fatigue Evaluation Using the Weld Master S-N Curve". Dong states that the stress to be used in the Master SN design curves is the pseudo elastic structural stress "as shown in Fig. 6c". In "Fig 6c", the pseudo elastic stress is the stress resulting from an extrapolation of the test specimen's linear elastic stiffness curve. Therefore, so long as strain controlled stresses are calculated when using the Master SN curve, no plasticity adjustment is required. This is true since the linear elastic FE analysis provides stresses calculated from applied strains and the elastic modulus (which is analogous to the testing extrapolation procedures described herein for a given displacement or strain).

E.C. Rodabaugh – NUREG 3243 "Comparisons of ASME Code Fatigue Evaluation Methods for Nuclear Class 1 Piping with Class 2 or 3 Piping"

The results were reported as points on S_f vs N_f plots. N_f is the number of cycles to failure (crack through the wall). The corresponding nominal stress was computed by the ordinary beam formula, $S_f = WL/Z$. The load range W was taken from the load-deflection calibration, or for loads causing plastic deformation, from straight-line extrapolation of the elastic portion of the load-deflection calibration. The lever arm L was measured from the point of load application to the point of initial failure.

The test method is consistent with an elastic analysis of a piping system, even though calculated stresses may be above the material yield strength and some plastic deformation may occur. Accordingly, an adjustment analogous to the K_e used in Code 1 is not needed.

B31.J-2006 (Draft) "Standard Method for the Determination of Stress Intensification Factors (i-Factors) for Piping Components by Test"

Markl's tests were based on linear elastic equivalent moments, i.e. a constant displacement or rotation was applied and the moment at the failure location was based on extrapolation of the M-θ (or F-δ) elastic curve. This allows agreement with the way linear elastic thermal expansion analyses are used, even though predicted stresses may be above yield.

Pingsha Dong – "Low-Cycle Fatigue Evaluation Using the Weld Master S-N Curve"

~~reaching beyond the cyclic yield strength of the material.~~ To be consistent with the master S-N curve definition in the low-cycle regime, the structural stress range should be converted to an elastic pseudo structural stress range as shown in Fig. 6c before the master S-N curve can be consistently used. ~~Otherwise, the~~

- The Neuber-based procedure for estimating pseudo elastic structural stress in low-cycle regime is only needed when performing linear FEA when pseudo elastic load or stress is not known

7.0 Further Discussion

It is important to note that a plasticity correction is not necessary so long as the conditions applied in the design analysis are displacement controlled conditions or strain limited. Examples of such displacement controlled results include the stresses due to thru thickness temperature gradients, restrained thermal expansion of components such as piping, and secondary stresses (strain limited) due to primary loads such as the bending stresses at a nozzle junction due to internal pressure.

Secondary stresses due to primary loads are similar to displacement controlled conditions since the overall load-deflection curve will remain elastic while a small local region of material will experience strain beyond yield. Since the material surrounding the highly stressed zone remains elastic, the highly stressed material is essentially exposed to strain controlled conditions. In such cases, the load-deflection curve remains linear elastic while the local material experiences plastic strain. Of course, this concept does not apply if the primary loads produce gross plasticity and result in a non-linear load-deflection curve.

If the calculated stresses used to enter the fatigue curve are based on an applied load and the stresses are not strain limited, then the calculated stresses would require adjustment using a Neuber correction or similar technique. This is necessary since the applied loads will under predict the actual displacements (strain) when a linear elastic analysis technique is used.

However, given the ASME limits on load controlled conditions (primary type loads such as pressure), the use of a Neuber correction would never be necessary. Structural stresses due primary loads would be classified to secondary stress category and limited to the range of 2*Sy. Within this range, the structural stress should shake down to elastic action and alternating plasticity will not occur. If there is no plasticity, then there is no need to apply plasticity correction factors.

8.0 Conclusions

The following conclusions can be drawn based on the examples and references presented here:

1. Neuber's rule need not be applied in ASME codes designs.
2. For displacement controlled conditions such as thermal loadings or applied displacements, the resulting stresses in a linear elastic analysis are already the pseudo elastic structural stresses.
3. Applying a Neuber's adjustment to displacement controlled analyses will lead to undue conservatism and gross errors in life predictions.
4. As illustrated by test results, if significant plasticity occurs and measured or applied loads are used in an elastic analysis, Neuber's rule can not effectively predict the pseudo elastic stresses and non-conservative design lives could be produced.
5. ASME Code stress limits on stresses due to applied loads (primary and secondary stresses due to sustained loadings) ensure that the structural strain ranges will shake down to elastic action. Therefore, Neuber's rule is not required for applied loadings in the ASME Code.

Table 3 – Fatigue Test Results and Calculations for Specimen #12

Description	Value	Reference
Cycles to Failure “Nf”	202 cycles	WRC 433
Displacement Range “Dr”	3.8” (96.52mm)	WRC 433 Table 1
Pseudo Elastic Nominal Stress Range “Sn”	2,965 MPa	WRC 433 Table 1
Pseudo Elastic Structural Stress “Se”	3,110 MPa	WRC 474
Equivalent Structural Stress “Ss”	3,649 MPa	PVP 2006 Coeff’s.
Predicted mean cycles for Ss = 3,649 MPa	201 cycles	PVP 2006 Master SN Curve

Table 4 – Fatigue Design of Specimen #12 without LCF Correction

Step	Description	Result
1	Calculate the linear elastic force using the specified displacement range of 3.8” (96.52 mm). Here, the entire system stiffness as given by WRC 433 is used in lieu of the stiffness of the pipe only. This seems to be more reasonable since the test was a system of beams, not a single beam.	F = 6300*Dr = 6300*3.8 = 23,940 lbf
2	Calculate the linear elastic bending moment for the applied displacement range.	M = F*a/2 = (23,940*15/2) = 179,550 in-lb
3	Calculate the pseudo elastic nominal stress range	Sn = M*D*0.5/I = (179,550*1.9*0.50 / 0.39) = 437,365 psi (3,015 MPa)
4	Calculate the structural stress. A structural stress concentration factor of 1.05 will be used as was applied in WRC 474.	Se = 513,730 * 1.05 = 437,365 psi (3,015 MPa)
5	Calculate the equivalent structural stress. Note that no plasticity correction factor is applied here since the stresses are strain limited (calculated from the applied displacement). No mean stress adjustment is included either.	Ss = 512,998 psi (3,537 MPa)
6	Determine the cycles from the fatigue curve. Note that in this case, the design curve has been taken as the mean curve since the original data point is directly on the mean curve and the goal is to attempt to replicate the test data.	N = 222 cycles

Table 5 – Fatigue Design of Specimen #12 with LCF Correction

Step	Description	Result
1	Calculate the linear elastic force using the specified displacement range of 3.8" (96.52 mm). Here, the entire system stiffness as given by WRC 433 is used in lieu of the stiffness of the pipe only. This seems to be more reasonable since the test was a system of beams, not a single beam.	F = 6300*Dr = 6300*3.8 = 23,940 lbf
2	Calculate the linear elastic bending moment for the applied displacement range.	M = F*a/2 = (23,940*15/2) = 179,550 in-lb
3	Calculate the pseudo elastic nominal stress range	Sn = M*D*0.5/I = (179,550*1.9*0.50 / 0.39) = 437,365 psi (3,015 MPa)
4	Calculate the structural stress. A structural stress concentration factor of 1.05 will be used as was applied in WRC 474.	Se = 513,730 * 1.05 = 437,365 psi (3,015 MPa)
5	Calculate the pseudo elastic structural stress using the proposed Neuber's low cycle fatigue correction term.	Sess = 1,488,812 psi (10,265 MPa)
56	Calculate the equivalent structural stress using the pseudo elastic structural stress based on Neuber's rule. No mean stress adjustment is included either.	Ss = 1,746,979 psi (12,045 MPa)
6	Determine the cycles from the mean fatigue curve using the equivalent structural stress based on Neuber's rule.	N = 5 cycles