Understanding Piping Code Stress Evaluation Paradoxes And ASME B31.3 Appendix P

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August, 2013

Abstract

A piping system is designed based on the piping code created for each individual industry. There are several different piping codes in U.S and the World to cater to the different natures of the industries. To ensure the structural integrity of the piping, each code will start out with allowable materials and their basic allowable stresses and then will figure out what stresses need to be calculated and how to calculate them. Finally, a set of allowable stresses, comprising the basic allowable stresses, is set to validate the structural integrity of the piping system. At each code, the allowable stresses are consistent only with the stresses calculated using the method of the individual code. Therefore, it is obvious that the allowable of one code should not be used for the stress calculated with another code. In fact, all the codes have warned that each code must be applied in its entirety.

The different stresses calculated by each code together with different allowable values permitted by each code have created considerable paradox in the piping community. Engineers are often confused about what to do when alternative methods to the code may be required to deal with special cases. This paradox has recently crept into ASME B31.3 Appendix-P. This paper will present the stress criteria background and explain why the Appendix P is formulated based on confusing logic that may very well lead to unsafe design of the piping system.

1. Introduction

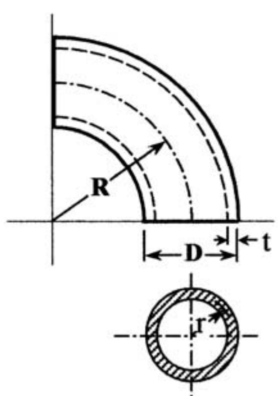
Piping engineers are often confused by the many different codes and different stress evaluation criteria given on a piping system. The availability of different codes catered to different industries is required to achieve an effective, economical and social utilization of goods and capital. These codes each used in its entirety would not create any problem. A problem occurs when we try to cross use the different codes or criteria. This is even more troublesome when we try to mingle different design philosophies based on the seemingly supreme stress criteria.

Throughout the history of piping system design, there have been two distinctly different design approaches and philosophy.

In the beginning, due to the lack of accurate knowledge and calculation tools, the piping was designed with minimum calculations, but based on a lot of rules and experiments. This approach, though not exact, has designed a lot of successful piping systems for many decades. Therefore, it is valued and used extensively even up to this date and will continue to be used for many more decades to come. This is the design approach adopted by ASME B31 Code for Pressure Piping and many other piping codes in the world.

As the technology progressed and the piping got more critical, such as in the case of nuclear piping that involves the welfare of society, engineers started looking for something more concrete and exact. How to be more exact? First, of course, is to calculate all the stresses that exist in the piping accurately. Then, set up the stress criteria for the calculated stresses to protect all conceivable modes of failure. This new approach is called "Design by Analysis," in contrast to the "Design by Rules" adopted by B31 codes.

The "Design by Analysis" approach is more theoretical oriented and has theoretical basis to back it up. For this reason, engineers may be tempted to use the criteria set up for this approach for the stress calculated by B31. This mixed-up use of the codes is, of course, not permitted, but somehow considered valid by some engineers thinking the stress criteria for the "Design by Analysis" must be universally applicable. This is the confusion that has crept into Appendix P of ASME B31.3 [1]. Leaving it as-is, the Appendix P may lead to unsafe design of the piping system.



10" Std Long Radius Elbow:

$D = 10.75"$
$t = 0.365"$
$R = 15"$
$r = 5.1925"$
$I = 161\ in^4$
$Z = 29.9\ in^3$

Figure 1, Example Component – 10" Std Long Radius Elbow

Throughout this paper, we will use a typical 10" standard wall thickness long radius elbow, as shown in Figure 1, as the example component to exam the differences and validities of the two approaches. The use of this example elbow provides some comparison numbers instead of just symbols.

Stress Category	PRIMARY			SECONDARY		PEAK
	General Membrane	Local Membrane	Bending	Expansion	Membrane Plus Bending	
DESCRIPTION	Average primary stress across solid section. Excludes effects of discontinuities and concentrations. Produced by pressure and mechanical loads	Average stress across any solid section. Considers effects of discontinuities but not concentrations. Produced by pressure and mechanical loads, including inertia earthquake effects	Component of primary stress proportional to distance from centroid of section, excludes effects of discontinuities and concentrations. Produced by pressure and mechanical loads, including inertia earthquake effects	Stresses which result from constraint of "free end displacement" and the effect of anchor point motions resulting from earthquakes. Considers effects of discontinuities but not local stress concentration	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by pressure, mechanical loads, or by differential thermal expansion. Excludes local stress concentrations	(1) Increment added to primary or secondary stress by a concentration (notch). (2) Certain thermal stresses which may cause fatigue but not distortion.
SYMBOL	P_m	P_L	P_b	P_e	Q	F

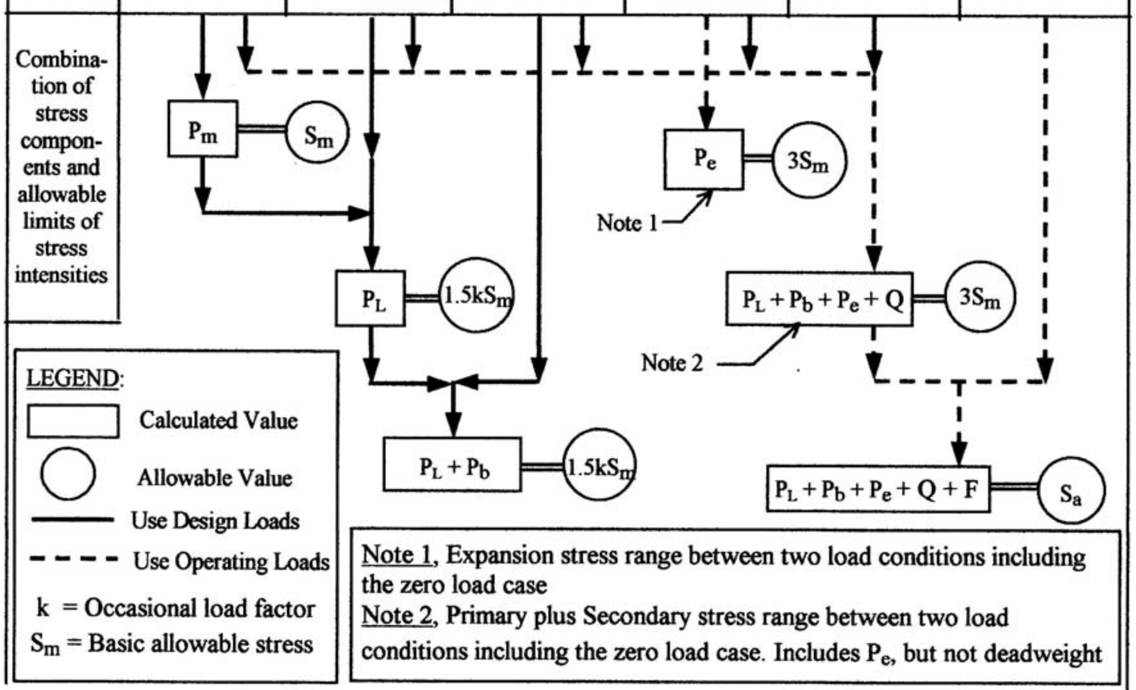


Table 1, General Stress Criteria for Design by Analysis Approach

2. Design by Analysis

The Design by Analysis approach is adopted for critical piping to weed out all the potential uncertainties to positively ensure the safety of the piping system. Since it involves a lot of more calculations that might substantially increase the cost and schedule of a project, this approach is currently adopted only by ASME B&PV Section III [2] Class 1 Components and Section VIII Division 2 [3] Alternative Rules for Pressure Vessels in design of piping and pressure vessels. We will use Class 1 nuclear piping as the guide to see what the stresses are calculated and what are the allowable values.

In formulating the design requirements of this Design by Analysis approach, a special ASME review committee was created to establish the Stress Criteria [4] for protecting the piping from potential modes of failures based on calculated stresses. These stress criteria are summarized in Table 1. From the table, it is clear that the criteria provide the piping with membrane protection and fatigue protection. The loads are classified as primary (or sustained) and secondary (or self-limiting), two main categories. The definitions are also mostly included in Table 1. The primary loads are also called non-self limiting loads. All sustained loading (non self-limiting) at design conditions are evaluated for membrane protection, while all repeating primary as well as secondary load ranges are all included in fatigue evaluation. Dead weight is not included in primary plus secondary stress evaluation as it is non cyclic, but live weight such as carrying fluid is included.

2.1 Membrane Protection

Besides the standard pressure design that is comparable to all codes, the membrane protection is checked by the following equation: (Table 1, P_L which includes $P_m + P_b$)

$$B_1 \frac{PD}{2t} + B_2 \frac{D}{2I} M_{i,L} \le 1.5 S_m \qquad \text{(A)}$$

Where,
P = Design pressure
B_1 = Primary stress index for pressure (= 0.5 for bend) (1983 Edition)
h = Flexibility characteristic ($= tR/r^2$ for bend) (= 0.203 for example elbow)
B_2 = Primary stress index for moment ($= 1.30/h^{2/3}$ for bend) (= 3.76 for example elbow)
2I/D = Z (Section modulus)
S_m = Allowable stress intensity at design temperature
(= 2/3 $S_{y,h}$ or less. Use 2/3 $S_{y,h}$ for simplifying comparison)
$S_{y,h}$ = Yield strength of the material at design temperature
$M_{i,L}$ = Resultant moment due to combination of design mechanical loads including inertia earthquake loads. The resultant moment is calculated from moment components in each direction combining all included loads in a conservative way.
(See Figure 1 for dimensions)

For the example elbow component, Equation (A) can be rewritten as follows:

$$\frac{PD}{4t} + 3.76\frac{M_{i,L}}{Z} \leq 1.0S_{y,h} \qquad \text{(for 12" Std L.R. Elbow)} \qquad \text{(AA)}$$

This will later be used to compare with the evaluation by B31 codes

2.2 Fatigue Protection

The fatigue protection is provided by evaluating all pairs of primary plus secondary stress ranges following the dot line flows in Table 1. Since the main concern is fatigue, the stresses calculated are ranges between two operating conditions. The zero load condition is also considered as one loading condition. From Table 1, there are three items that need to be evaluated. One is the primary plus secondary stress intensity range, the second is the thermal expansion stress intensity range, and the third is peak stress intensity range, which is used to obtain the allowable number of operating cycles from applicable fatigue curve for each operating pair. In the process, it needs to be concerned about potential plastic hinge that might invalidate the elastic analysis. Additional considerations and handlings are also needed when plastic cycling or ratcheting may be present in any operating pair or pairs.

2.2.1 Check for Plastic Hinge – Gross Ratcheting

This is applicable only to piping systems, but not to vessels. Thermal expansion of a piping system may involve a fairly large movement. A few inches of movement is not uncommon in a power or process plant. This movement has the potential of behaving like a sustained load when there is an especially weak location in the system. This is the reason when designing the vessel nozzle connection, the piping loads are often considered as sustained or primary load.

Within the piping system, if the thermal expansion stress intensity range at a certain location is greater than twice of the yield strength, then the location may become a plastic hinge or produce strain follow-up. If the system has one or more plastic hinges, the commonly used elastic analysis is invalid. Therefore, for secondary stresses, the first thing to be checked is the following: (Table 1, $P_e < 3S_m$)

$$S_e = C_2\frac{D}{2I}M^*_{i,E} \leq 3S_m \qquad \text{(B)}$$

Where,
Se = Thermal expansion stress
C_2 = Secondary stress index for moment (= $1.95/h^{2/3}$ for bend) (= 5.65 for example elbow)
$M_{i,E}$* = Resultant range of moment due to thermal expansion and thermal anchor movement only.

S_m = Allowable stress intensity of the material at highest temperature throughout the operation cycle involved. (To simplify comparison, use 2/3 $S_{y,h}$)

For the example elbow, Equation (B) can be written as follows:

$$S_e = 5.65\frac{M^*_{i,E}}{Z} \leq 2S_{y,h} \qquad \text{(for 12" Std L.R.. Elbow)} \qquad \text{(BB)}$$

It is important to note that once Equation (B) is not satisfied, all the other calculations are simply meaningless. Equation (BB) will be used later to compare with the evaluation by B31 codes

2.2.2 Check for Thermal Ratcheting

If the total primary plus secondary stress intensity range, including effects of discontinuity but not local stress concentration, exceeds twice the yield strength, a small amount of plastic deformation accumulates during each cycle of operation. This phenomenon is called ratcheting. The primary plus secondary stress intensity range for each operating pair is calculated by (Table1, $P_L + P_b + P_e + Q$)

$$S_n = C_1\frac{P_o D}{2t} + C_2\frac{D}{2I}M_{i,T} + C_3 E_{ab}[\alpha_a T_a - \alpha_b T_b] \leq 3S_m \qquad \text{(C)}$$

Where,
C_1 = Secondary stress index for pressure (= (2R –r)/2(R – r) for bend)
(=24.81/19.613=1.26 for example elbow)
P_o = Range of service pressure
C_2 = Secondary stress index for moment (= 5.65 for example elbow)
$M_{i,T}$ = Resultant range of moment from all thermal expansion, mechanical loads, live weight, wind and/or earthquake inertia loads, earthquake and/or wind anchor displacement, etc. Dead weight is not included as it is not cyclic.
C_3 = 1.0 for bend
$E_{ab}[\alpha_a T_a - \alpha_b T_b]$ = Gross temperature discontinuity stress. E and α are based on room temperature.

For the example elbow, Equation (C) can be written as

$$S_n = 1.26\frac{P_o D}{2t} + 5.65\frac{M_{i,T}}{Z} + E_{ab}[\alpha_a T_a - \alpha_b T_b] \leq 2S_{y,h} \qquad \text{(for 12" Std L.R. Elbow)} \qquad \text{(CC)}$$

It is important to note that the participation of pressure in this equation is the maximum hoop stress, not the longitudinal stress as in Equation (AA). This is due to the fact that the maximum local stress is generally in the circumferential direction,

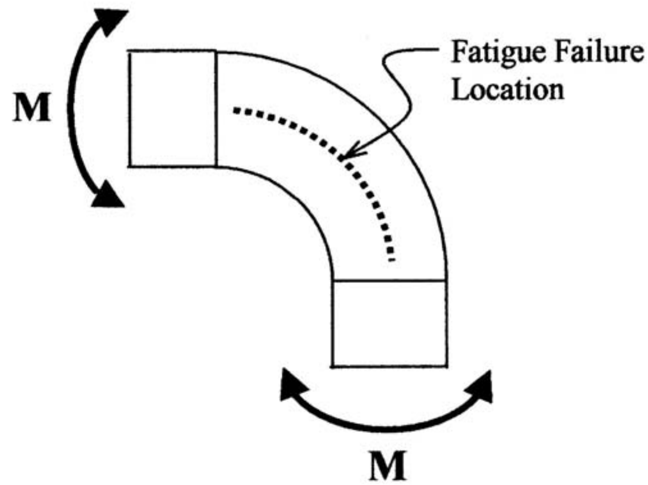


Figure 2, Fatigue Failure at Bend due to Moment Loading

which is in the same direction as the pressure hoop stress. See Figure 2 for the fatigue failure location of the bend due to moment loading.

The piping system may still be acceptable even if Equation (C) is not satisfied. However when $S_n > 2S_y$ the system will have a ratcheting effect, accumulating some small plastic strain through each cycle of operation. In this case, the evaluation will have to go through elastic-plastic process. To simplify our discussion, we will consider only the cases when Equation (C) is satisfied.

The satisfaction of Equation (C) does not necessarily mean the fatigue protection is fulfilled. It all depends on S_n stress level and number of operating cycles of all pairs of loading conditions. To satisfy the fatigue protection, the following peak stress intensity range and alternating stress intensity for each pair of loading have to be calculated and evaluated.

2.2.3 Peak Stress Intensity Range and Alternating Stress Intensity

The peak stress range including local notch stress concentration factor is calculated by (Table 1, $P_L + P_b + P_e + Q + F$)

$$S_p = K_1 C_1 \frac{P_o D}{2t} + K_2 C_2 \frac{D}{2I} M_{i,T} + \frac{1}{2(1-\mu)} K_3 E\alpha\,|\,\Delta T_1\,| + K_3 C_3 E_{ab}\,[\,\alpha_a T_a - \alpha_b T_b\,] + \frac{1}{1-\mu} E\alpha|\Delta T_2| \qquad \text{(D)}$$

Where,
K_1, K_2, K_3 are stress concentration factor of the components. All are equal to1.0 for the elbow component away from a weld.
ΔT_1 = Effective linear temperature difference between outside and inside walls across the thickness. (See Ref .2 for details)

ΔT_2 = Nonlinear temperature gradient across the wall thickness that is not included in the linear ΔT_1. (See Ref.2 for details)
μ = Poisson ratio
α = Thermal expansion rate

When Equation (C) is satisfied, the alternative stress intensity is calculated as one-half of the peak stress. That is

$$S_{alt} = \frac{S_p}{2} \qquad (\text{When } S_n \leq 3S_m) \qquad \text{(E)}$$

This S_{alt} for each operation load pair is checked with the allowable fatigue curve to obtain the allowable number of operating cycles, N_{iA}. This is then converted to the usage factor of the load pair as $U_i = N_i / N_{iA}$. N_i is the number of the operating cycles for the i-th load pair. The total usage factor or cumulative damage for all load pairs is the sum of all the individual load pair usage factors. The total usage factor shall be less than 1.0.

The peak stress intensity range is not directly comparable to any stress calculated by B31. Therefore, no comparison will be made in this paper. It should be noted, however, that C_2K_2 is closely related to twice of the stress intensification factor of B31 codes [5].

3. B31 Approach – Design by Rules

In the earlier days when knowledge was not sufficient to precisely look into many stress details, piping systems were designed with rough calculations on basic items and a lot of rules and experiments. The rules include details on junction shapes, design specifications, standard support details, limit on support spacings, operating procedures, etc. Experiments from previous operations on the finished plants are eventually all put into design specifications and/or codes. Local stress behaviors on piping components are tested with real components. Stress range concept on secondary stress and definite thermal expansion stress evaluation were eventually adopted in 1955 edition of B31 code. Allowable stresses are set lower to accommodate uncertainties.

While the Design by Analysis approach was first adopted by nuclear vessels in 1963 and nuclear piping in 1969, the non-nuclear industries would like to get away as far as possible from it. The main reason is cost and availability of manpower. The cost would be unheard of in non-nuclear industries to calculate all those additional stresses in local notch, weld detail, thermal discontinuity and thermal gradient during transient and steady state operations, dynamic earthquake, fluid transient load, etc. in precise and reliable manner. The added calculations, documentations, checking, and independent review would increase the required man-hours by 20-fold and would extend the project schedule by three times. Therefore, the old B31 approach, although not a hundred percent theoretically defendable, is the mainstay of traditional industries.

B31 code does not exactly follow the stress criteria given in Table 1. Nevertheless, it does the same types of protections as given in Table 1. That is, the membrane protection and fatigue protection are properly addressed. Unless otherwise noted, the following discussions follow B31.1[6] for simplicity. Also to compare with the Design by Analysis, the items applicable to Design by Analysis are termed as the "Stress Criteria."

3.1 B31 Membrane Protection

After the pressure design, which is compatible to all codes, the membrane protection is checked with the following equation:

$$S_L = \frac{PD}{4t} + 0.75i\frac{M_A}{Z} \le 1.0S_h \qquad \text{(a)}$$

Where,
P = Design pressure
h = Flexibility characteristic (= tR/r^2 for bend) (= 0.203 for example elbow)
i = Stress intensification factor (= $0.9/h^{2/3}$ for bend) (= 2.61 for example elbow)
M_A = Resultant moment due to weight and mechanical load
S_h = Allowable stress at hot (design) condition = 2/3 $S_{y,h}$ or smaller (use 2/3 $S_{y,h}$ for simplifying comparison)

For the example elbow component, Equation (a) can be written as follows:

$$S_L = \frac{PD}{4t} + 1.96\frac{M_A}{Z} \le 0.67S_{y,h} \quad \text{(for 12" Std L.R. Elbow)} \qquad \text{(aa)}$$

The above requirement is not directly comparable with the Stress Criteria requirement given in Equation (AA) without some manipulation. Since both requirements has PD/4t longitudinal pressure stress term (hoop stress does not need to be included, see Ref. [7]), which can be set as 0.5 S_h = $0.333S_{y,h}$. After substituting PD/4t = $0.333S_{y,h}$, the requirements become

$$1.96\frac{M_A}{Z} \le 0.34S_{y,h} \qquad \text{(aa - 1)} \quad \text{for B31.1}$$

$$3.76\frac{M_{i,L}}{Z} \le 0.67S_{y,h} \qquad \text{(AA - 1)} \quad \text{for Stress Criteria}$$

These can be rearranged to obtain the allowable moment for the example 12" Std L.R. elbow as

$$M_A \le 0.173S_{y,h}\,Z \qquad \text{(aa - 2)} \quad \text{for B31}$$

$$M_{i,L} \le 0.178S_{y,h}\,Z \qquad \text{(AA - 2)} \quad \text{for Stress Citeria}$$

From the above, it is clear that for membrane protection requirement, B31 and Stress Criteria are almost identical, except that the moment $M_{i,L}$ in Stress Criteria includes also operational earthquake inertia load. B31 considers the earthquake (and/or wind) as occasional load and has an increased allowable of 1.2 S_h = 0.8 $S_{y,h}$. By doing the same deduction as above, the comparable allowable occasional load moment ($M_A + M_B$) would be equal to 0.24 $S_{y,h}$ Z as compared to 0.178 $S_{y,h}$ Z of the Stress Criteria. M_B is the resultant moment of occasional loads, such as operational earthquake or/and wind, relief valve discharge force, turbine trip loads, etc. For earthquake and wind loads, several design levels may be provided depending damage level to be tolerated. At operational level, B31 is somewhat shy of the Stress Criteria requirement.

It is important to note that the sustained stress calculated in B31 for moment loads is only about one-half of the theoretical or actual stress implied by the Stress Criteria.

3.2 B31 Fatigue Protection

B31 fatigue protection deviates considerably from the Stress Criteria. Only thermal expansion and anchor/support displacement are included. Gross thermal discontinuity and thermal gradient are not considered. Local notches are covered by adjusting stress basis and by testing actual components. The current evaluation approaches were developed mostly by Markl and George [8] and Markl [9-10] and were officially adopted by B31 code in 1955 edition. Very little modifications have been made since then.

Since thermal expansion stress is self-limiting. Its mode of failure is fatigue due to repeated operating cycles. A self-limiting stress does not cause gross structural deformation when the yield strength is exceeded. By allowing higher than yield, the thermal stresses can yield or relax at hot operating condition, resulting in stress reversal throughout the operating cycle. Therefore, the stress range throughout the operating cycle should be used for the design evaluation. Cold spring that affects only the initial stress level is not credited for improving fatigue strength.

As all local stresses affect the fatigue damage, quantitative evaluation of local expansion stresses was introduced through stress intensification factors, which are derived and obtained mainly through tests. However, since there are considerable reliable theoretical stress relationships available on bend components, the theoretical bend formulas are used as guides for establishing test data correlations and code formulas.

Through strain controlled fatigue tests of piping components, Markl and his coworkers found that a pipe with an as-weld girth weld had a stress intensification of about two as compared with polished rods. In order to save the effort of identifying all the welds, and also other minor notches and clamping locations, they chose to use the pipe with girth weld as basis to establish the stress intensification factors of all components. This, in essence, cut the calculated stress in half when stress intensification factor is involved.

It is important to note that the thermal expansion and anchor displacement stress calculated by B31 is actually only one-half of the real and theoretical stress.

For fatigue protection, B31 requires that the following evaluation be met:

$$S_E = \frac{iM_C}{Z} \le f\,(1.25S_c + 1.25S_h - S_L) \qquad \text{(b)}$$

Where,
S_E = Thermal expansion stress
h = Flexibility characteristic (= tR/r^2 for bends) (=0.203 for example elbow)
i = Stress intensification factor (= $0.9/h^{2/3}$ for bends) (= 2.61 for example elbow)
M_C = Resultant moment due to thermal expansion and anchor displacement
M_C = $M_{i,E}$* of Equation (BB)
f = Fatigue strength reduction factor, =1 for 7000 cycles or less
S_c = Basic allowable stress at ambient (cold) condition (2/3 $S_{y,c}$ or less. Use 2/3 $S_{y,c}$)
S_h = Basic allowable stress at operation (hot) condition (2/3 $S_{y.h}$ or less. Use 2/3 $S_{y,h}$)
S_L = Sustained stress from Equation (a)

For the example elbow, Equation (b) can be rewritten for f = 1.0 as

$$S_E = 2.61\frac{M_C}{Z} \le (0.833S_{y,c} + 0.833S_{y,h} - S_L) \qquad \text{(bb)}$$

To compare with the Stress Criteria, we assume that S_L take up a stress of S_h. With S_L = S_h = 2/3 $Sy_{,h}$, we have

$$S_E = 2.61\frac{M_C}{Z} \le (0.833S_{y,c} + 0.167S_{y,h}) \quad \text{for example elbow} \qquad \text{(bb - 1)}$$

This M_C can be compared with $M_{i,E}$* of the Stress Criteria. For the example elbow, the protection for plastic hinge by the Stress Criteria is

$$S_e = 5.65\frac{M_{i,E}^*}{Z} \le (S_{y,c} + S_{y,h}) \quad \text{for example elbow} \qquad \text{(BB} - 1)$$

The original Stress Criteria and Section III nuclear code applies only for 700°F and below for ferrite steel and 800°F and below for austenitic steel. With this temperature limitation the difference between $S_{y,c}$ and $S_{y,h}$ is not very great, so $S_{y,c}$ is not used in the nuclear piping code. To be comparable to B31, the original $2S_y$ is separated into $S_{y,c}$ + $S_{y,h}$. By rearranging equations (bb-1) and (BB-1), we have the maximum allowable moment loading for the example elbow as

$$M_C \le (0.319S_{y,c} + 0.064S_{y,h})Z \quad \text{(B31)} \qquad \text{(bb} - 2)$$

$$M_i^* \le (0.177S_{y,c} + 0.177S_{y,h})Z \quad \text{(Stress Criteria)} \quad \text{(BB} - 2)$$

From the above allowable moment comparison, current B31 thermal expansion allowable is comparable to the Stress Criteria for plastic hinge protection at lower temperature range, but somewhat short of the Stress Criteria requirement at high temperature range. This shortcoming is somewhat compensated by applying the highest stress intensification factor to all in-plane, out-plane, and torsion moments in B31.1. (At bends, B31.3 uses smaller SIF for out-of-plane bending and no SIF for torsion)

As for actual fatigue damage evaluation, B31 relies on fatigue tests of actual components. Although gross thermal discontinuity and thermal gradients are not included, the hoop pressure stress is considered in the allowable via S_L, which has a maximum of S_h the same as for hoop pressure stress. With the allowable stress limit as established in B31, Markl found that based on tests, the implied safety factors are in the order of 2 in terms of stress, and in the order of 30 ($\sim 2^5$) in terms of cyclic life. The very least factor available, considering the 25% spread encountered between individual test data, might be estimated as 1.25 in terms of stress and 3 in terms of cyclic life. **However, these safety factors were based on the allowable using the basic allowable stress as 5/8 of the yield strength. The basic allowable stress has later been increased to 2/3 of the yield strength; based on new basic allowable stresses, the adjusted safety factor in terms of stress has to be reduced by a factor of 0.94, and in terms of life reduced by a factor of 0.75.** Therefore, an accurate calculation of the stress and a conservative estimate of the number of operation cycles are important.

4. B31.3 Appendix P [1]

In the 2004 edition of ASME B31.3, an Appendix P was added to provide “Alternative Rules for Evaluating Stress Range.” This Appendix has later been substantially revised in the 2010 edition. It uses Edwards’ [11] paper as background to layout some rules and equations to evaluate the stress range in a way quite different from the existing code.

Edwards’ main theoretical basis is the Stress Criteria [4] (shown in Table 1) developed for alternative rules for pressure vessel code [3] and code for nuclear components [2]. However, it appears that the stress bases of B31.3 and the Stress Criteria have been confused, leading to a set of incorrect stress calculations and unsafe stress limits.

Besides the main concern of stress range evaluation, the Appendix P also concerns the stress due to axial forces. Though axial forces are routinely considered in sustained stress calculation and are important thermally for situations such as a straight run in between two restraints and at jacketed piping system, they are generally ignored in thermal expansion stress calculations. The shear forces are also generally ignored. They can be included, of course, if needed, and will not be discussed here. In fact, most piping stress computer programs do calculate Tresca stress that includes not only the axial force but also shear force and pressure hoop stress. It is just not used by the code stress evaluation.

The main concerns here are the Equations (P1a) and (P1b) given in Appendix P. Equation (P1b), which is more closely related to the Stress Criteria, will be discussed first.

(a) Checking for Plastic hinge or Gross Ratcheting

Appendix P stipulates that the allowable thermal expansion stress range shall be

$$S_E \le S_{EA} = 1.25\,f\,(S_c + S_h) \qquad \text{(P1b)}$$

From Table 1 Stress Criteria, the thermal expansion stress range is used to check the potential plastic hinge or gross ratcheting. The allowable is given as $2S_y$ as shown in Equation (B) and in Equation (BB) for the example elbow component. In order to compare with Stress Criteria, we have to know how the expansion range is calculated. Disregarding the axial and shear forces, the expansion stress is calculated in B31.3 as

$$S_E = \frac{1}{Z}\sqrt{(i_i M_i)^2 + (i_o M_o)^2 + (i_t M_t)^2} \qquad \text{(1b)}$$

Where,
S_E = Expansion stress range, including anchor displacement, but no sustained load
i_i = SIF for in-plane bending moment (= $0.9/h^{2/3}$ for bends) (= 2.61 for example elbow)
i_o = SIF for out-of-plane bending moment (= $0.75/h^{2/3}$ for bends) (= 2.17 for example elbow)
i_t = SIF for torsion moment (= 1.0 for all components)
M_i = In-plane bending moment
M_o = Out-of-plane bending moment
M_t = Torsion moment

As discussed previously, this S_E is not a real stress. It is just a reference stress about one-half of the theoretical real stress. In order to compare with the Stress Criteria, we will use i_i value also for i_o and i_t. This will soften the difference between B31.3 and the Stress Criteria and greatly simplify the comparison. By doing so, Equation (1b) becomes

$$S_E = i_i \frac{M_R}{Z} \qquad \text{(1b - 1)}$$

This is the same form as Equation (b) used by B31.1, except the resultant moment range M_R is called M_C in B31.1. For the example elbow, Equation (1b-1) becomes

$$S_E = 2.61\frac{M_R}{Z} \qquad \text{(1b - 2)}$$

The allowable for S_E is S_{EA} given by Equation (P1b). Since the protection is against plastic hinge or gross ratcheting, the number of allowable cycle is theoretically zero. That is f = 1.0. Also for simplifying the comparison, S_c and S_h are assumed to be governed by the yield strength. Therefore, Equation (P1b) can be written as

$$S_{EA} = 1.25(0.667S_{y,c} + 0.667S_{y,h}) = (0.833S_{y,c} + 0.833S_{y,h}) \qquad \text{(1b - 3)}$$

Making $S_E < S_{EA}$, we have Appendix P requirement as

$$2.61\frac{M_R}{Z} \le (0.833S_{y,c} + 0.833S_{y,h}) \quad \text{(for example elbow)} \qquad (1b-4)$$

We will then compare Equations (1b-4) against Equation (BB) to see if Appendix P meets the Stress Criteria. The two equations cannot be compared directly. One way to compare is to look at the magnitude of the moments allowed in each case. By converting Equations (1b-4) and (BB) to allowable moments for the example elbow, we have

$$M_R \le (0.319S_{y,c} + 0.319S_{y,h})Z \qquad \text{(Appendix P)} \qquad (1b-5)$$

$$M_i^* \le (0.178S_{y,c} + 0.178S_{y,h})Z \qquad \text{(Stress Criteria)} \qquad (BB-2)$$

M_R and $M_{i,E}$* are exactly the same and cover only the thermal expansion and anchor and/or support displacement ranges. From the above, we know Appendix P is roughly 80% over the Stress Criteria. In other words, by using Appendix P, an 80% overstressed component would still be considered acceptable.

(b) Primary Plus Secondary Stress Intensity Range

As Equation (P1b) already shows that Appendix P is a poorly conceived, unsafe alternative rules to the main code, there is no need to investigate Appendix P further. However to be complete, we will also take a look at the so-called operating stress limit as given by the following Equation (P1a).

$$S_O \le S_{OA} = 1.5(S_c + S_h) \qquad (P1a)$$

From Table 1 Stress Criteria and Equation (C), the allowable for primary plus secondary ($P_L+P_b+P_e+Q$) stress is two times the yield strength. This matches the allowable of Equation (P1a). The problem is the way and the number S_O is calculated.

First, we already know (explained many times previously in this paper) S_O calculated by B31.3 is just one-half of the actual real stress. From this point only, the allowable stress in (P1a) should be cut in half to something like 0.75 (S_c + S_h).

Second, the mode of failure of secondary stress is fatigue. All fatigue evaluations have to use stress range rather than a one-shot operation stress. This is clearly given in Table 1 as dotted line flow process.

Third, only cyclic loadings or stresses cause a fatigue failure. Although Stress Criteria calls for primary plus secondary stress intensity, the dead weight is not included. On the other hand, the pressure cycling form zero to full pressure is included. For the pressure, the participation stress is the hoop stress rather than the longitudinal pressure stress as in most of so-called operation stress calculations.

Fourth, there are gross thermal discontinuity and thermal gradient stresses that need to be included, but is not done in B31.3. Equation (C) also includes operation base earthquake inertia together with anchor displacement and other fluid transient loads, which are not generally included in operational stress calculations.

Fifth, when so many different load types are involved, a reliable conservative combination or superimposing approach is required. For instance, the yielding and relaxation nature of expansion stress, cold spring effect, dual directional effects of earthquake, etc. have to be considered.

From the above, it should be clear that Equation (P1a) itself does not have a problem. The problem is that the S_O stress is not properly calculated in B31.3. It is also roughly 100% deficient, assuming the stress calculation method is correct.

5. Conclusion

When dealing with local stress for fatigue evaluation, B31 use the Stress Intensification Factor (SIF) to modify the nominal stress to a reference stress. SIF is mainly obtained from fatigue tests [5] of actual piping components guided by available known theories on components such as bends. In determining the SIF, a commercial pipe with a girth weld is used as basis. Since a pipe with girth weld has a theoretical fatigue SIF of about 2.0, the B31 SIF is actually only one-half of the theoretical SIF. In other words, the secondary stress calculated by B31 is just one-half of the actual theoretical stress. This does not cause any problem if the allowable amount is adjusted accordingly.

B31.3 Appendix P uses B31.3 calculated stress to compare with theoretical stress implied by the Stress Criteria [4] as allowable. This makes the allowable stress twice as large as it should be. This will no doubt lead to unsafe design of the piping system. Furthermore, the calculation approach of the so-called operation stress is not consistent with Stress Criteria and actual load behaviors.

With the above presentation, it should be clear enough that Appendix P and related application on Appendix S should be withdrawn from the B31.3 code. The fatigue strength reduction factor, f, should also be limited to equal to or less than 1.0.

The problem of Appendix P is actually very simple. It all just comes down to the difference of the stress bases. A simple problem is often the most difficult to explain. If 16 pages of explanation are still not clear, the following joke might be of help.

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Just a Joke

An American businessman in Japan needs a pump, so he asks an American company to give him a quote. The American representative says he can sell him one for $2,000. The businessman says, “Okay, it is a deal.” He then sends the American company 2,000 yen for the pump. The American representative objects, “No, it is 2,000 US dollars.” The businessman says, “Well, the U.S. dollar is not user friendly here, so I sent you 2,000 yen as an alternative.”

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