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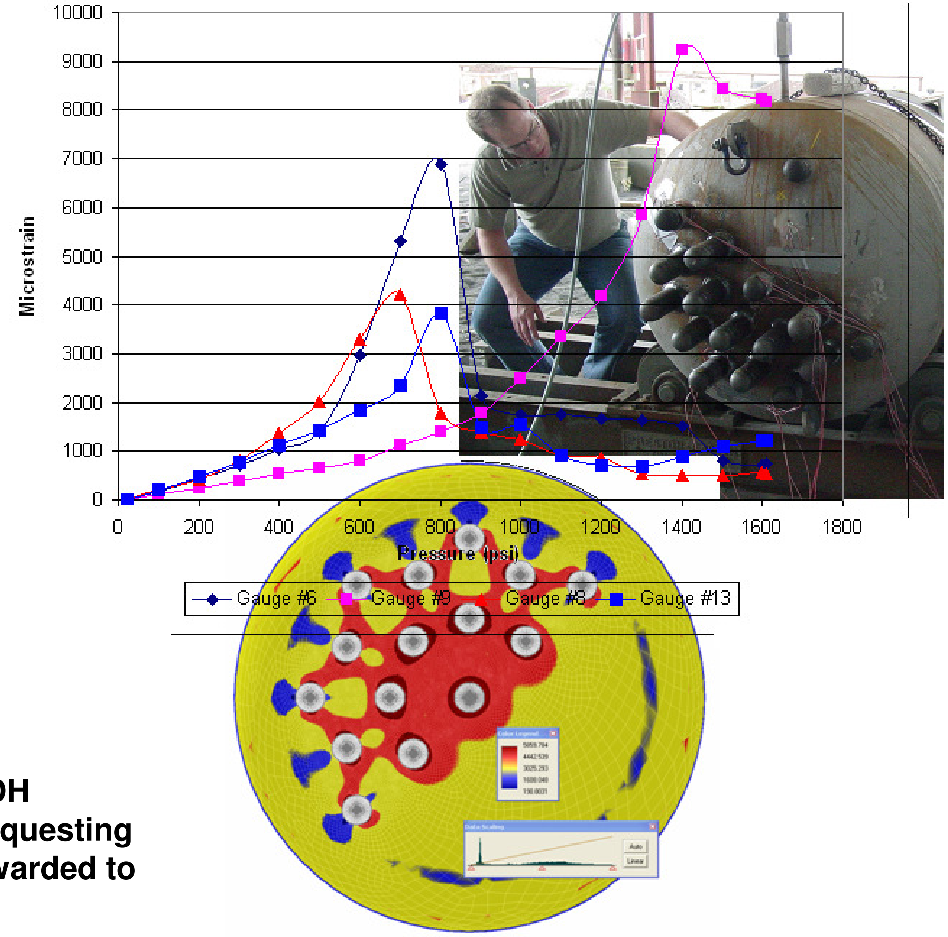
PRG Webinar 044

External Loads on Nozzles and Pipe Intersections

(Part 2)

Paulin Research Group

Tony Paulin and Chris Hinnant



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Pipe and Vessels can have a lot in common.

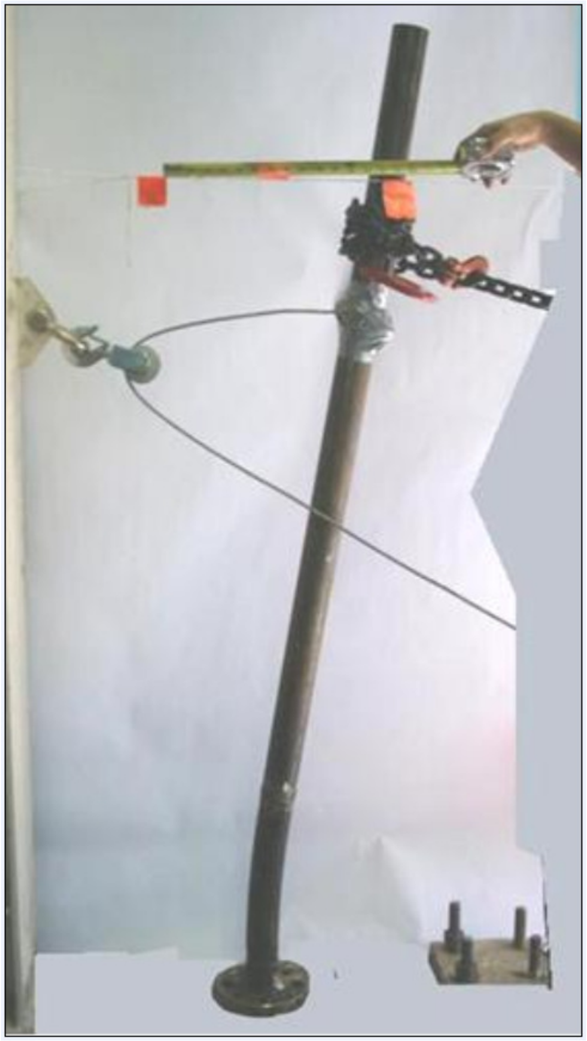
The basic rules of mechanics apply equally to pipes and pressure vessels – for pressure AND external loads.

Note:

Several of the topics contained in this webinar are interpretive and remain the responsibility of the designer to determine application.

In particular the interpretation voiced during the webinar regarding the 2004 ASME Section VIII Div 2 Art 4-138 and the 2007 ASME Section VIII Div 2 Art 5.6 has been modified in these notes.

Further work by PRG is planned to help clarify the differences and similarities between these separate Code year sections which address the same topic, i.e. stress classification for nozzle necks.



The pipes (or vessels) are going to bend too much, or cycle too much.

For Piping:

Calculated Stress from Piping Code < Allowable

For Vessels:

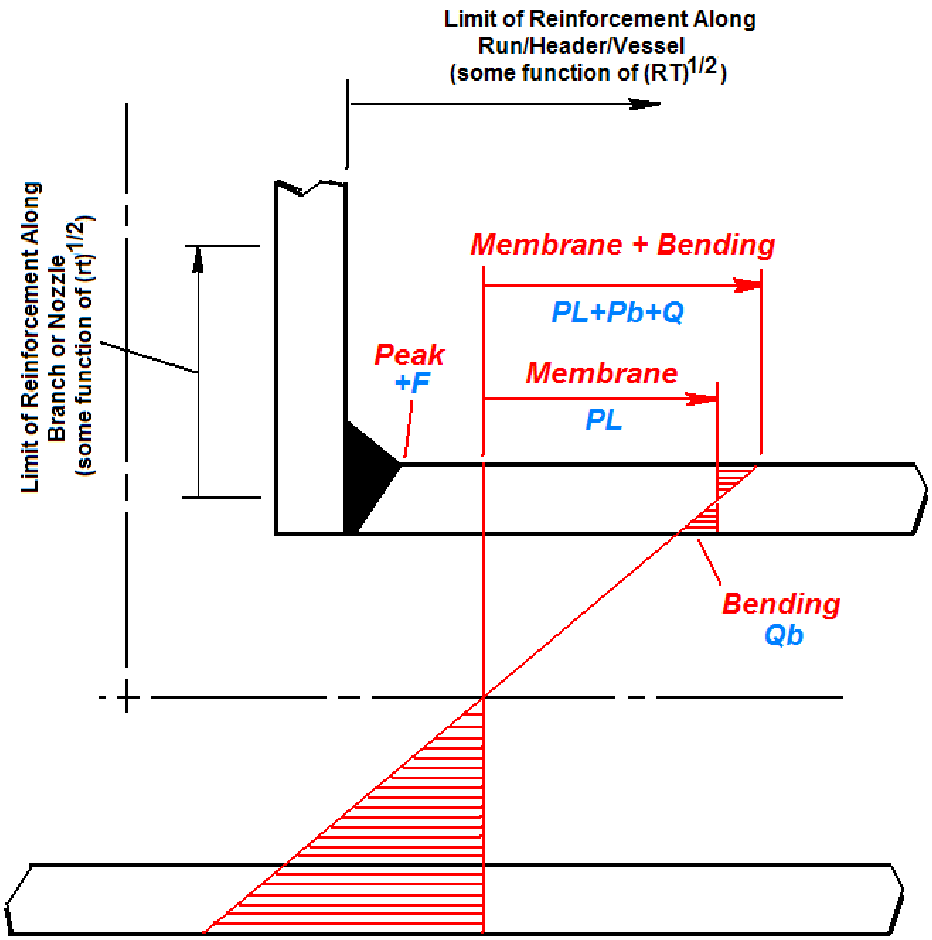
Determine Stress

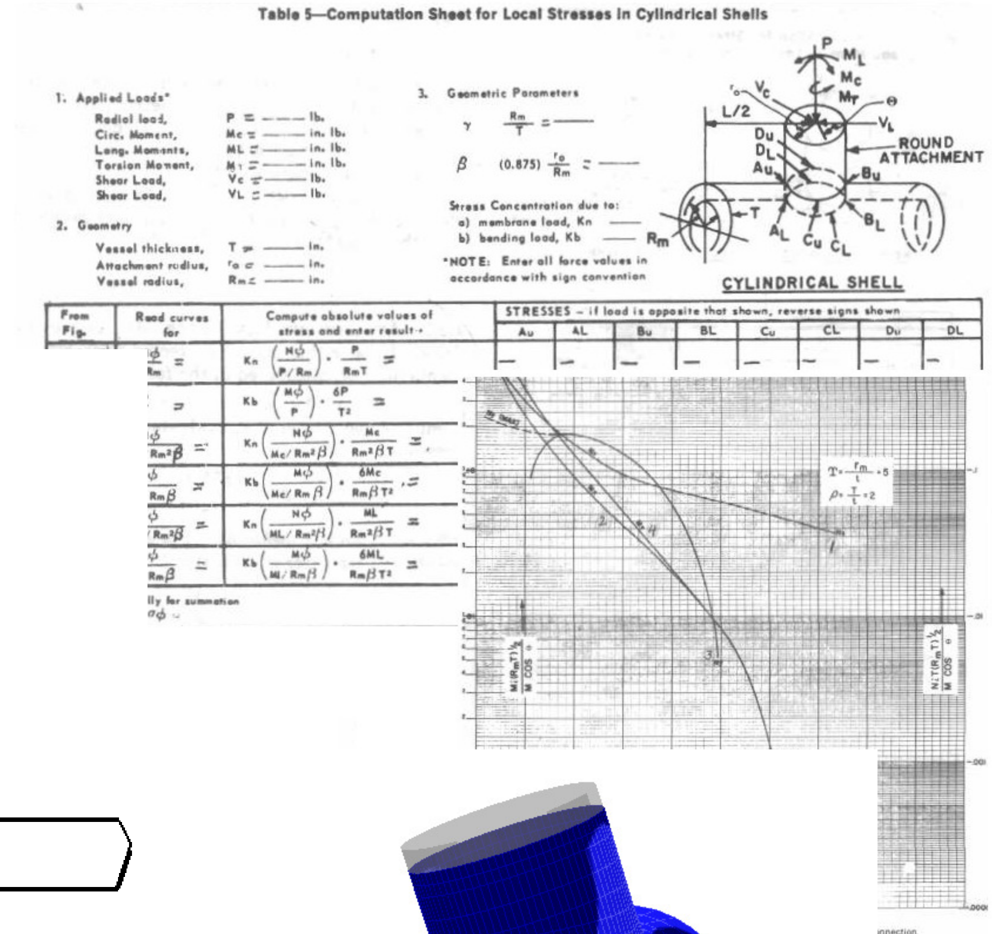
Find Allowable

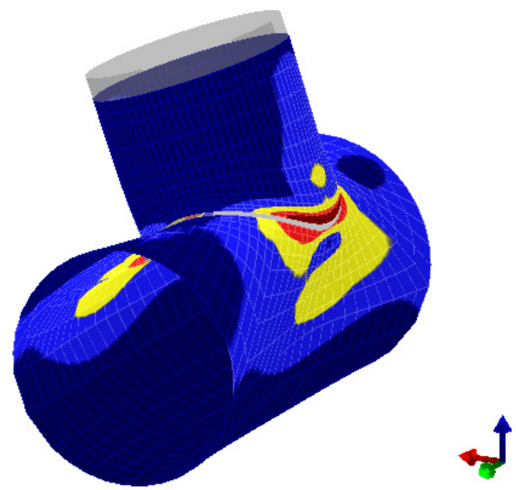
Determined Stress < Allowable

For “vessels” the external load evaluation is a little more difficult…

Calculate membrane and bending stresses from WRC 107/297 or FEA …







"Simplified" Analysis For Piping:

$Pd/4t + iM/Z < Sh$ …. for weight and pressure moments "M"

$iM/Z < 6N^{-0.2} [(1.25) (Sc+Sh)]$ …for thermal moments "M"

"Simplified" Analysis For Vessels:

$Pl < 1.5S_{mh}$ …. Pl = **primary local membrane stress**

$Pl+Pb+Q < 3.0S_{mavg}$ … Pl+Pb+Q = secondary stress

$Pl+Pb+Q+F < S_a$ … Pl+Pb+Q+F = peak stress *(cyclic, fatigue)*

How do stresses due to thermal expansion of the piping system end up in the Pl < 1.5 Sm category?

To Simplify the Analysis For Vessels:

$Pl < 1.5S_{mh}$ …. Pl = primary local membrane stress

$Pl+Pb+Q < 3.0S_{mavg}$ … Pl+Pb+Q = secondary stress

$Pl+Pb+Q+F < S_a$ … Pl+Pb+Q+F = peak stress

To Simplify the Analysis For Vessels:

$Pl < 1.5S_{mh} = Sy$ YIELD…. Often(?) W+P only **(no thermal?)**

$Pl+Pb+Q < 3.0S_{mavg} = 2Sy$ … Shakedown/Ratcheting **(yes – thermal)**

$Pl+Pb+Q+F < S_a$ … Pl+Pb+Q+F = peak stress **(yes – thermal)**

thermal = restrained free end displacements of attached pipe

5.6 Supplemental Requirements for Stress Classification in Nozzle Necks

The following classification of stresses shall be used for stress in a nozzle neck. The classification of stress in the shell shall be in accordance with paragraph 5.2.2.2.

a) Within the limits of reinforcement given by paragraph 4.5, whether or not nozzle reinforcement is provided, the following classification shall be applied.

~(RT)$^{1/2}$

1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as stresses, other than discontinuity stresses, due to external loads and moments including those attributable to restrained free end displacements of the attached pipe.

2) A P_L classification shall be applied to local primary membrane equivalent stresses derived from discontinuity effects plus primary bending equivalent stresses due to combined pressure and external loads and moments including those attributable to restrained free end displacements of the attached pipe.

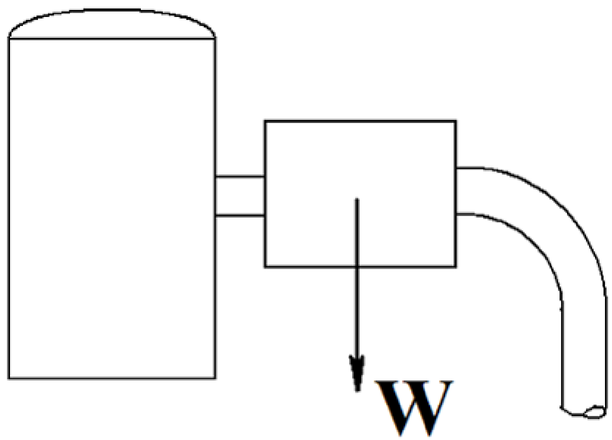
3) A $P_L + P_b + Q$ classification (see paragraph 5.5.2) shall apply to primary plus secondary equivalent stresses resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe.

b) Outside of the limits of reinforcement given in paragraph 4.5, the following classification shall be applied.

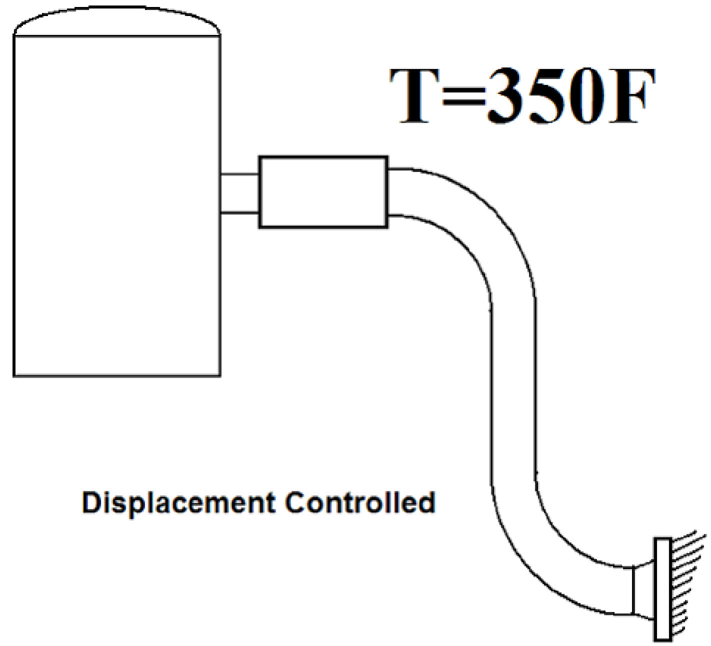
1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as the average stress across the nozzle thickness due to externally applied nozzle axial, shear, and torsional loads other than those attributable to restrained free end displacement of the attached pipe.

2) A $P_L + P_b$ classification is applicable to the equivalent stresses resulting from adding those stresses classified as P_m to those due to externally applied bending moments except those attributable to restrained free end displacement of the pipe.

3) A $P_L + P_b + Q$ classification (see paragraph 5.5.2) is applicable to equivalent stresses resulting from all pressure, temperature, and external loads and moments, including those attributable to restrained free end displacements of the attached pipe.



Load Controlled



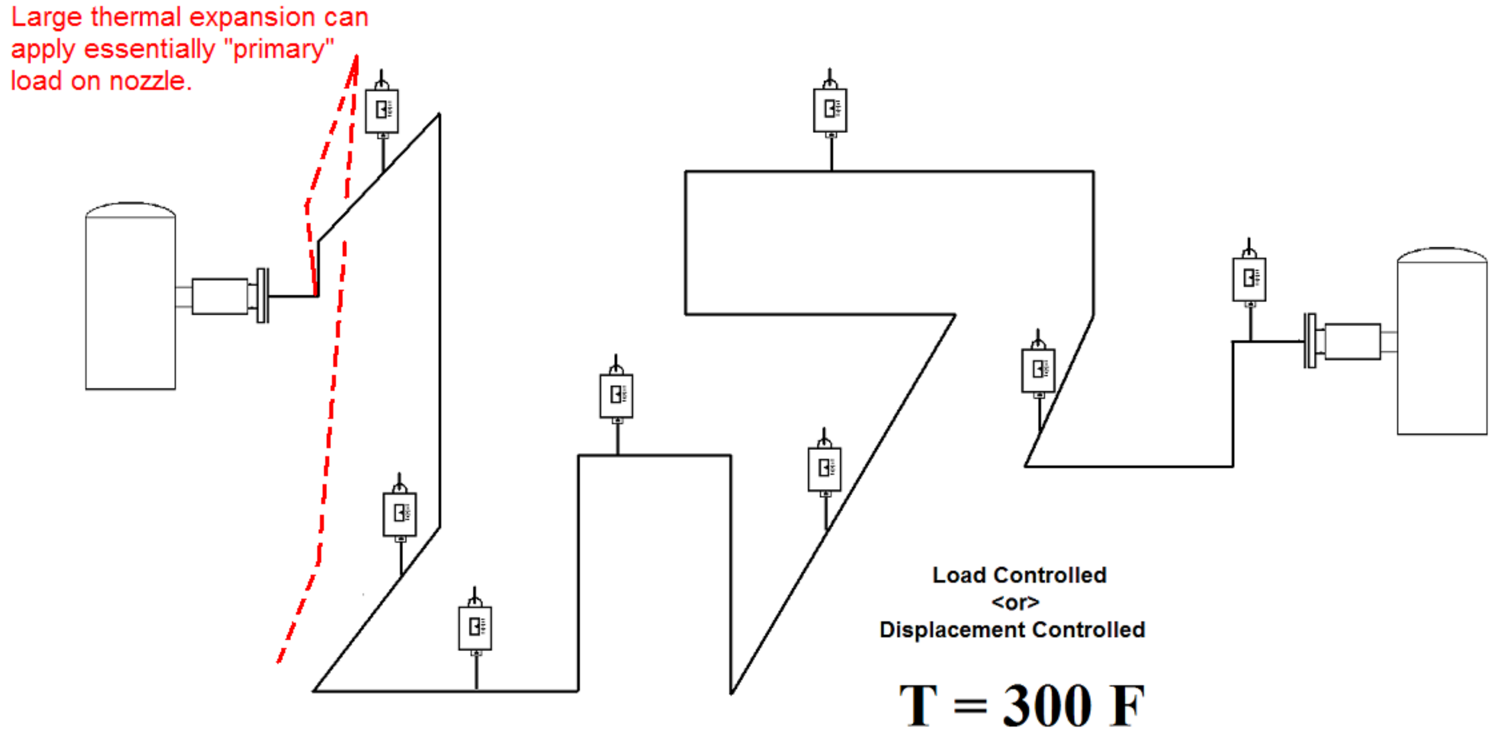
Typically: $PL < 1.5Sm$

Typically: $PL+Pb+Q < 3Sm$

$PL+Pb+Q+F < Sa$

Pipe is relatively short, thermal strains will be secondary in nature and DO NOT need to be included in the local membrane stress due to pressure and weight.

The above statement in red is an interpretation made by the Paulin Research Group. The user must make his own determination if thermal loads due to restrained free end displacements are primary or secondary in nature. An example where they may be primary in nature is shown on the following slide. Most common piping analysis programs evaluate nozzle junctions regularly and do not include any component of thermal loads in the primary local stress evaluation whose allowable is Sm (the hot allowable stress). There has been no known (to PRG) failure or distortion due to this effect, although (as shown on the following slide), it can be hypothesized to exist. Where long axial runs frame into nozzles with short offsets, an increased straining can occur, although this is an elastic followup effect of limited strain extent and has not been shown to produce ratcheting or distortion in piping systems.



Typically: $P_L < 1.5S_m$

(include loads due to thermal expansion of pipe in primary stress evaluation.)

Typically: $P_L+P_b+Q < 3S_m$

$P_L+P_b+Q < S_a$

Thermal strains in systems of this type may not be limited and might behave in a primary manner with respect to distortion of the nozzle neck. Due to strain hardening, redistribution of plastic strains (see later slides), and extensive experience with a vast number of piping systems that have been analyzed since the 1960's it is thought that there is no or little contribution to local primary membrane stresses due to restrained free end displacements of the attached pipe. The Div 2 Code permits the designer to include these thermally induced membrane stresses into the nozzle evaluation for local primary membrane stresses. It is thought that this will be an excessively conservative evaluation when nozzles or pipe branch connections are optimized for pressure.

2007 ASME Section VIII Division 2 "Code" Designed Nozzle Neck:

1. Check M/Z + PD/4T in the nozzle neck *(within reinforcement)* and ensure it is less than **1.0S**. *The moment "M" should include all external loads due to free end displacements (thermal).* *(No "i" factor, no discontinuity component)*

2. Check the local membrane stresses caused by M and P and ensure they are less than **1.5S**. "M" should include all external loads due to free end displacements (thermal) **that induce primary local stresses**. (It is believed that most free end displacements do NOT produce primary local stresse.)

3. Check the local M**+B** stresses caused by M, P, and T and ensure they are less than 2Sps. Within the nozzle neck there is no reference to 5.5.6.2 which permits 2Sps to be exceeded. Outside of the reinforcement distance there is explicit direction to permit M+B stresses (PL+Pb+Q) to exceed 2Sps (3Sm, 2Sy). It is generally not good design guidance to use 5.5.6.2 to exceed 2Sy within nozzle reinforcement. For post-construction operating evaluations, the user must consider all factors and make a reasonable engineering assessment.

In the 50+ years of B31 Code useage, thermal stresses on branch connections (nozzles) have never been analyzed as primary loadings. Do vessel engineers have to do so? Could this create an artificially low allowable nozzle load?

To Simplify the Analysis For Piping:

Can be small if system is hot

$Pd/4t + iM/Z < Sh$ …. for weight and pressure moments "M"

$iM/Z < \underbrace{6N^{-0.2}}_{\text{1.0 or 1.2 (usually)}} [(1.25) (\underbrace{Sc}_{\text{Sc does not get smaller with temperature.}}+Sh)]$ …for thermal moments "M"

Thermal Moments are "strain limited" (See 319.2.3)

Is this what 2007-VIII Div 2 wants?

$Pd/4t + iM_T/Z < Sh$ …. for weight and thermal? and pressure moments "M"

The following slides show that including thermal loads in the local primary membrane stress will likely not govern any analysis when the pressure and weight induced primary membrane stresses inside the nozzle reinforcement zone are small.

When pressure and weight induced primary membrane stresses inside the nozzle reforcement zone are large, and when the hot allowable stress is considerably lower than the cold allowable stress the effect may be significant.

General guidance from PRG based on many years of piping experience with similar nozzle junctions indicates that there is no (or little) primary character to the moments induced on nozzles due to the restrained free end displacements of the attached pipe.

5.6 Supplemental Requirements for Stress Classification in Nozzle Necks

1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as stresses, other than discontinuity stresses, due to external loads and moments including those attributable to restrained free end displacements of the attached pipe.

2) A P_L classification shall be applied to local primary membrane equivalent stresses derived from discontinuity effects plus primary bending equivalent stresses due to combined pressure and external loads and moments including those attributable to restrained free end displacements of the attached pipe.

3) A $P_L + P_b + Q$ classification (see paragraph 5.5.2) shall apply to primary plus secondary equivalent stresses resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe.

1) M/Z < Sh ... note that there's no "i" factor.

2) PL < 1.5 Sh ... membrane stress (+thermal+pressure+weight)

3) $iM/Z = < 2Sy(avg) \sim \underbrace{(1.2)(1.25)}_{=1.5=3/2}(Sc+Sh)$

B31.3:

$(i)(M/Z) < (f)(1.25)(Sc+Sh)$

(M/Z) = (f)(1.25)(Sc+Sh) / (i) = max allowed nominal

$(i) = (0.9)/(h^{2/3})(t/T)$ <or> $(i_b) = 1.5(R/T)^{2/3}(d/D)^{1/2}(t/T)$

WRC 497 *(Koves, Mokhtarian, Rodabaugh, Widera)*

$P_L/(M/Z) = f(d/D,D/T,t/T)$ … f() from FEA correlations

$P_L < 1.5Sm$

$P_L = f(d/D,D/T,t/T)(M/Z) = 1.5Sm$

(M/Z) = 1.5Sm / [f(d/D,D/T,t/T)] = max allowed nominal

WRC 497 Limits

$0.333 \le d/D < 1.0$

$20 \le D/T < 250$

$d/D < t/T < 3.0$

WRC 497 *(Koves, Mokhtarian, Rodabaugh, Widera)*

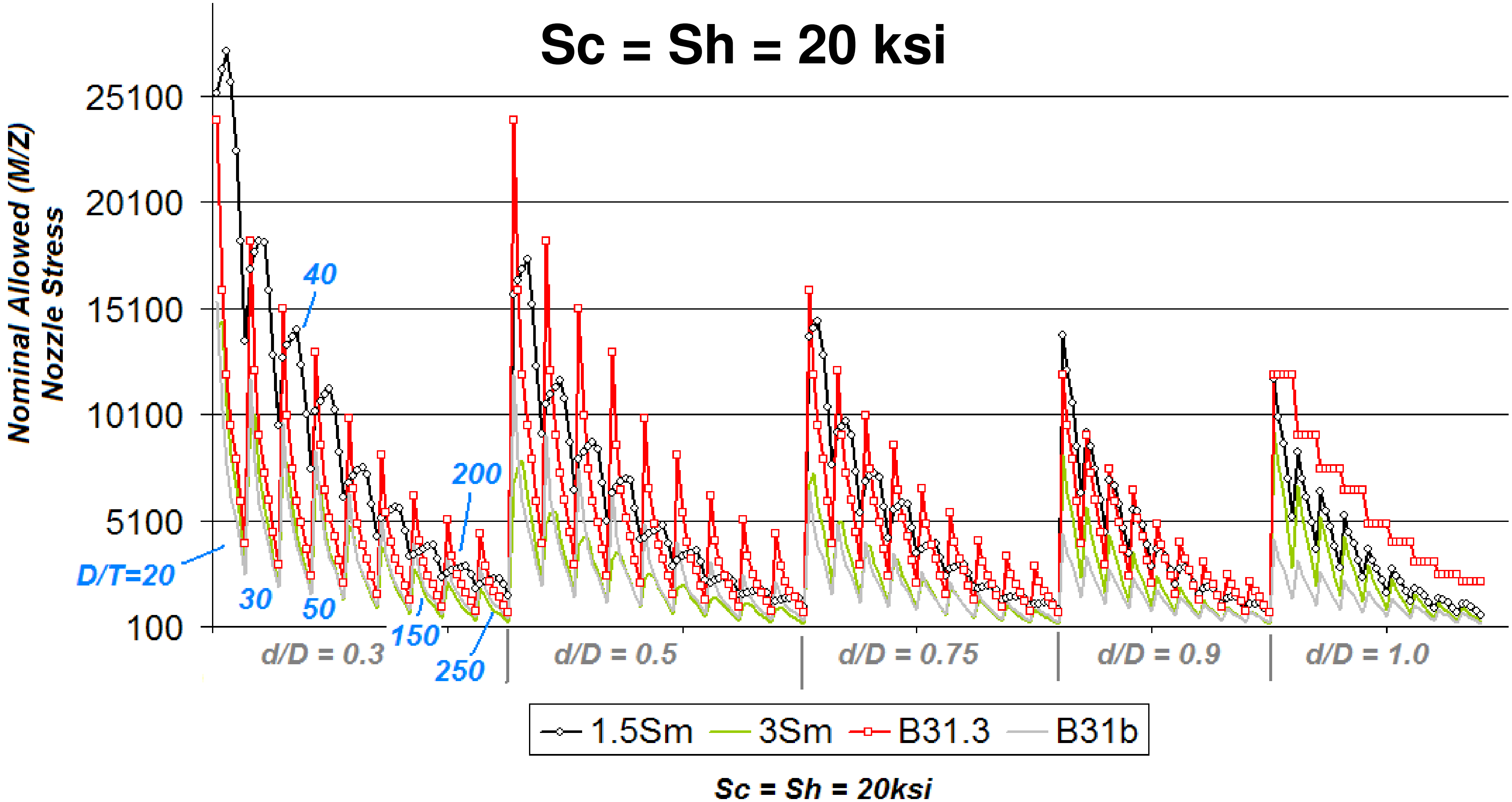
$P_L+Pb+Q/(M/Z) = f_o(d/D,D/T,t/T)$ … $f_o()$ from FEA correlations

$P_L+Pb+Q < 3.0Sm$

$P_L+Pb+Q= f_o(d/D,D/T,t/T)(M/Z) = 3.0Sm$

(M/Z) = 3.0 Sm / [f_o(d/D,D/T,t/T)] = max allowed nominal

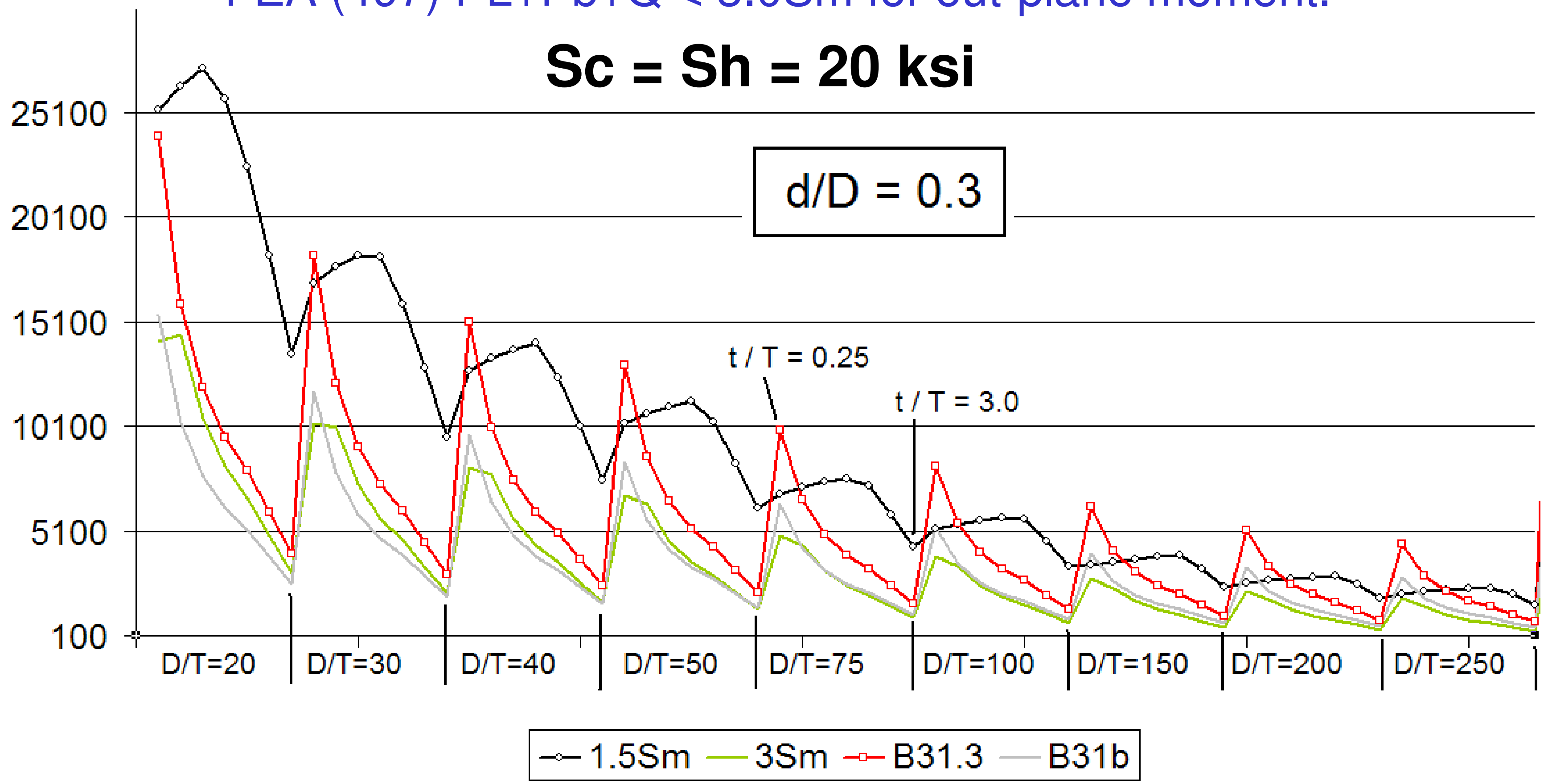
Maximum Allowed Nominal Stress from B31.3 Appendix D and from from reduced branch equation (B31b), and from FEA (497) PL<1.5Sm, and from FEA (497) PL+Pb+Q < 3.0Sm for out-plane moment.



D/T = 20, 30, 40, 50, 75, 100, 150, 200, 250

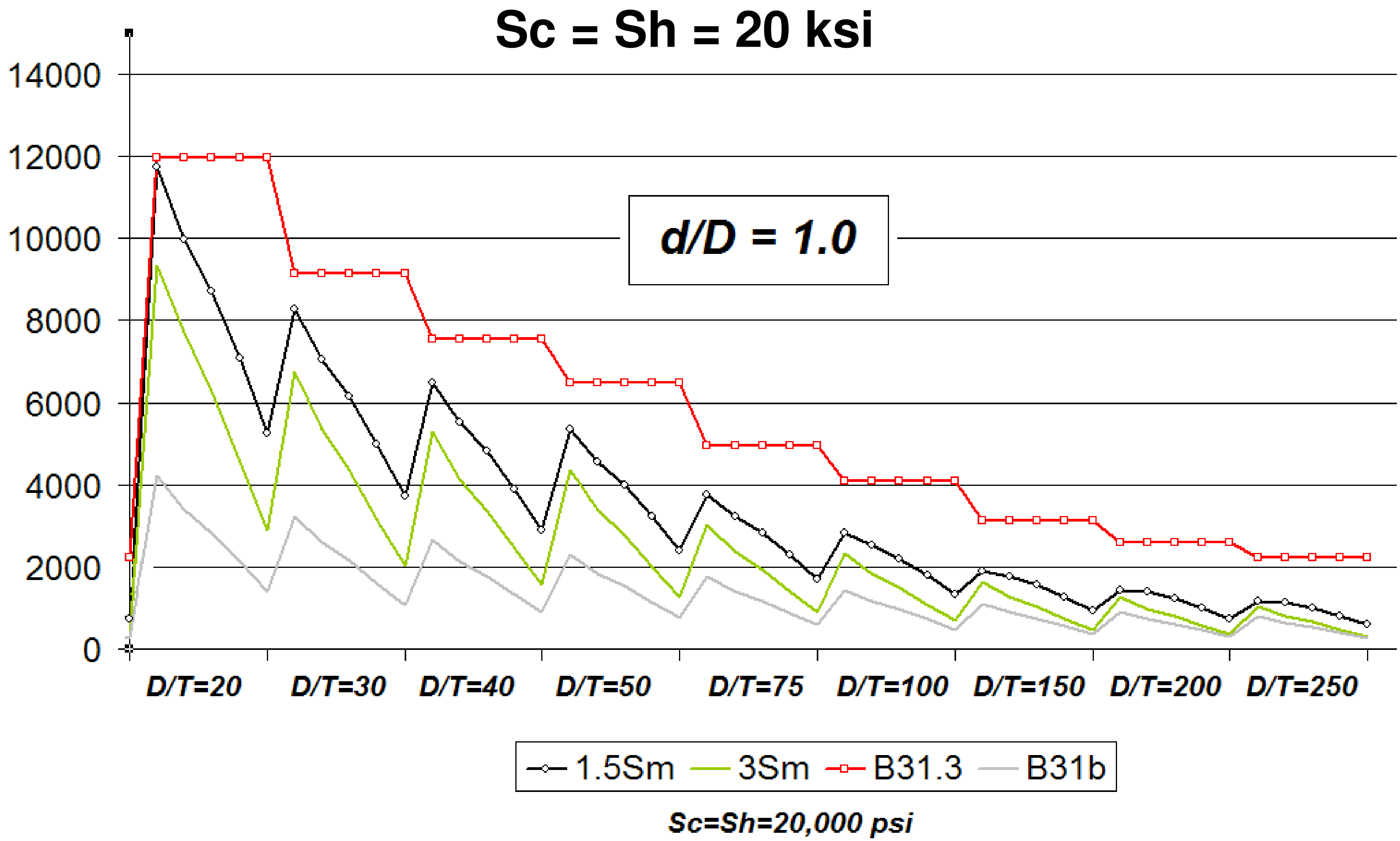
t /T = 0.25, 0.5, 0.75, 1.0, 1.25, 1.5, 2.0, 3.0

Maximum Allowed Nominal Stress from B31.3 Appendix D and from from reduced branch equation (B31b), and from FEA (497) PL<1.5Sm, and from FEA (497) PL+Pb+Q < 3.0Sm for out-plane moment.



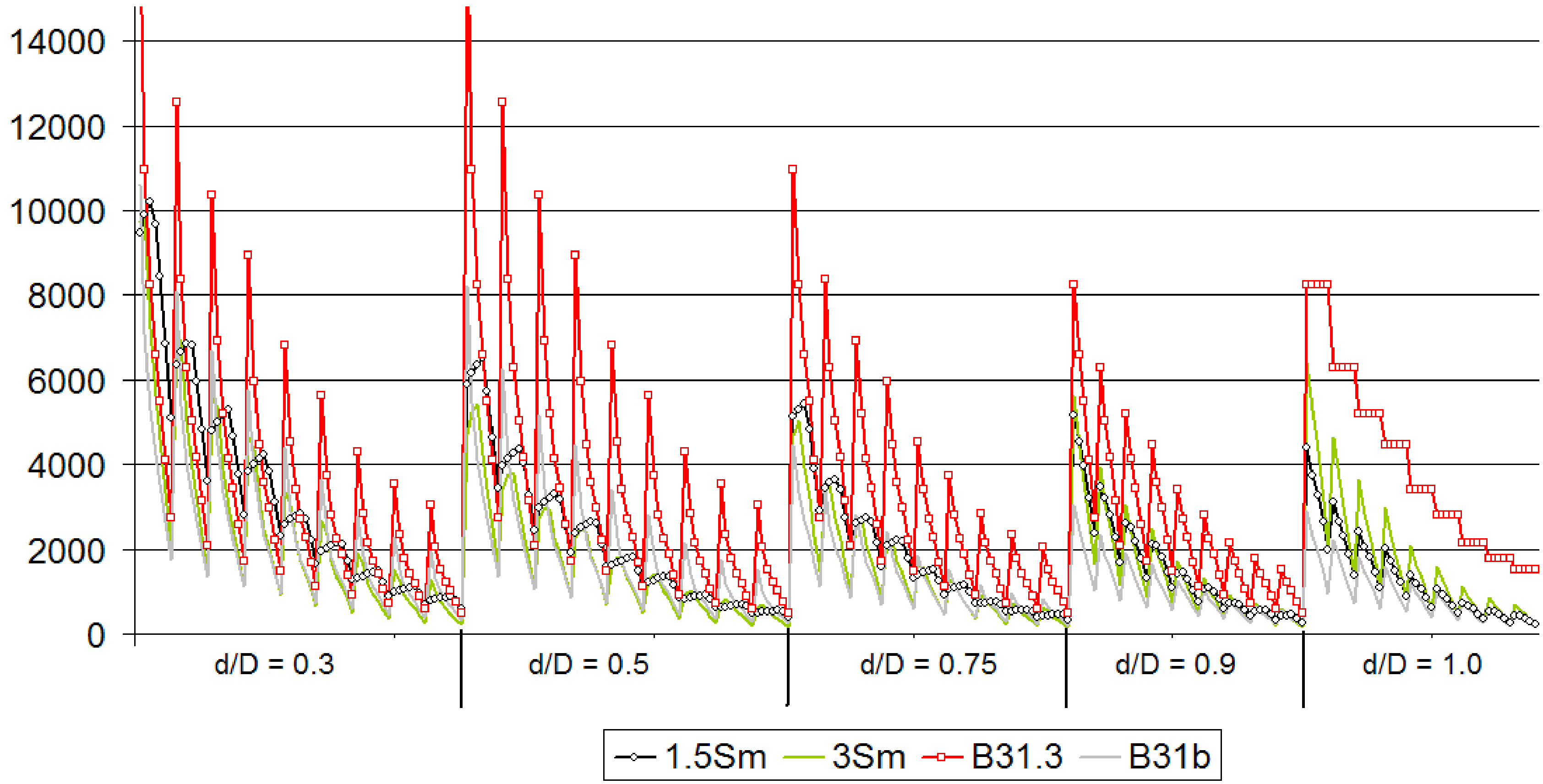
Notes: Membrane limit and B31.3 App D "i" factor are highest. Secondary Stress limit and B31 branch factor are lowest and similar.

Maximum Allowed Nominal Stress from B31.3 Appendix D and from from reduced branch equation (B31b), and from FEA (497) PL<1.5Sm, and from FEA (497) PL+Pb+Q < 3.0Sm for out-plane moment.



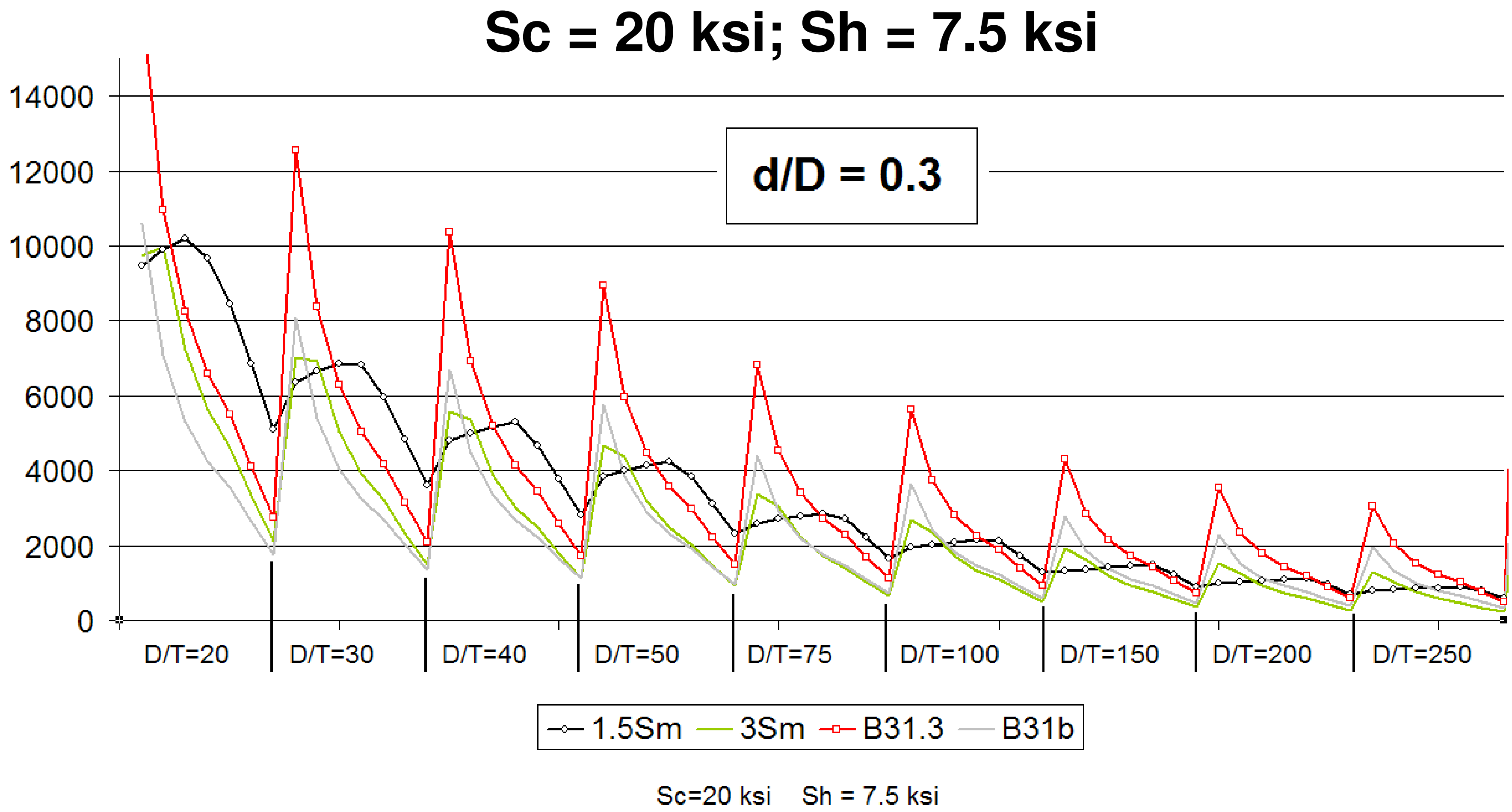
Maximum Allowed Nominal Stress from B31.3 Appendix D and from from reduced branch equation (B31b), and from FEA (497) PL<1.5Sm, and from FEA (497) PL+Pb+Q < 3.0Sm for out-plane moment.

Sc = 20 ksi; Sh = 7.5 ksi



Will membrane stress requirement take over when the hot allowable is small?

Maximum Allowed Nominal Stress from B31.3 Appendix D and from from reduced branch equation (B31b), and from FEA (497) PL<1.5Sm, and from FEA (497) PL+Pb+Q < 3.0Sm for out-plane moment.



How does the pressure stress effect each result?

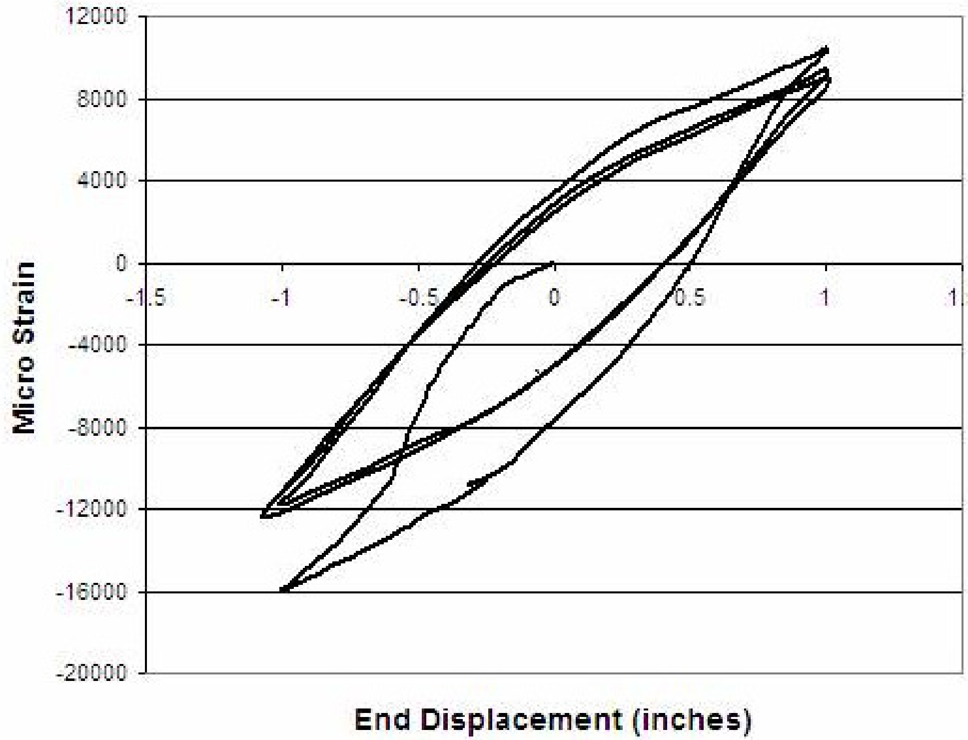
Conclusion:

1) There are small areas where the local membrane stress may control stress due to external moments, but the difference between the oft-used piping allowables and the controlling Pl+Pb+Q < 3.0Sm for secondary loading (including thermal) can be 2.0 or greater, (i.e. successful piping intersections in some cases may have more than twice the load than corresponding vessel intersections – more of a concern when the system is heavily cyclic.)

2) ***Effect of pressure interacting with external moments may produce artificially low allowable external loads. (P+M interaction not evaluated in WRC 497.) In most cases the restrained free end displacements of attached pipe do not have a local primary character inside the reinforcements of nozzles.***

3) ***Run elastic FEA, include free end displacements in membrane analysis. If stresses are excessive – run elastic/plastic calculation with large rotation to show that they are not. (MOST CONSERVATIVE – BUT SOMEWHAT UNREALISTIC APPROACH.)***

Get a "feel" for for external loads?



Strain Range = 16000 + 11000 = 27,000 microstrain
= 2.7% strain

Displacement Controlled Test
Cycles to Failure = 240

Failure by crack and thru-wall leak.

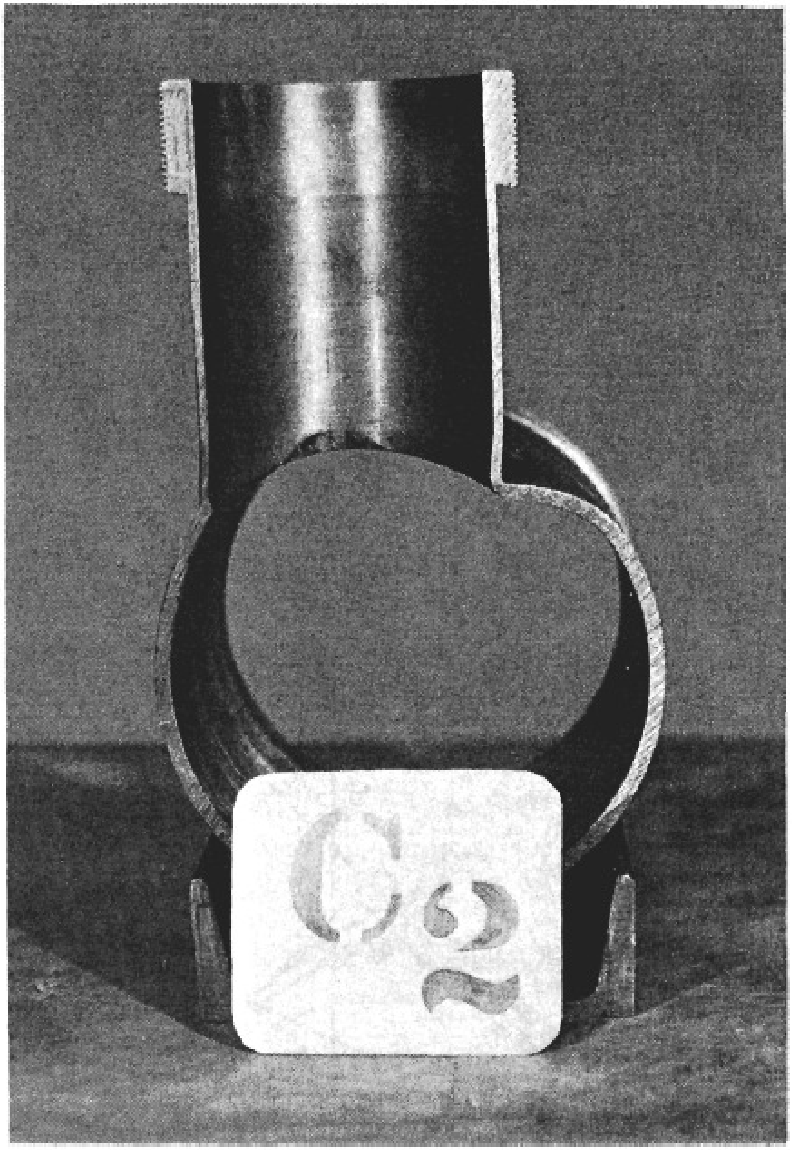
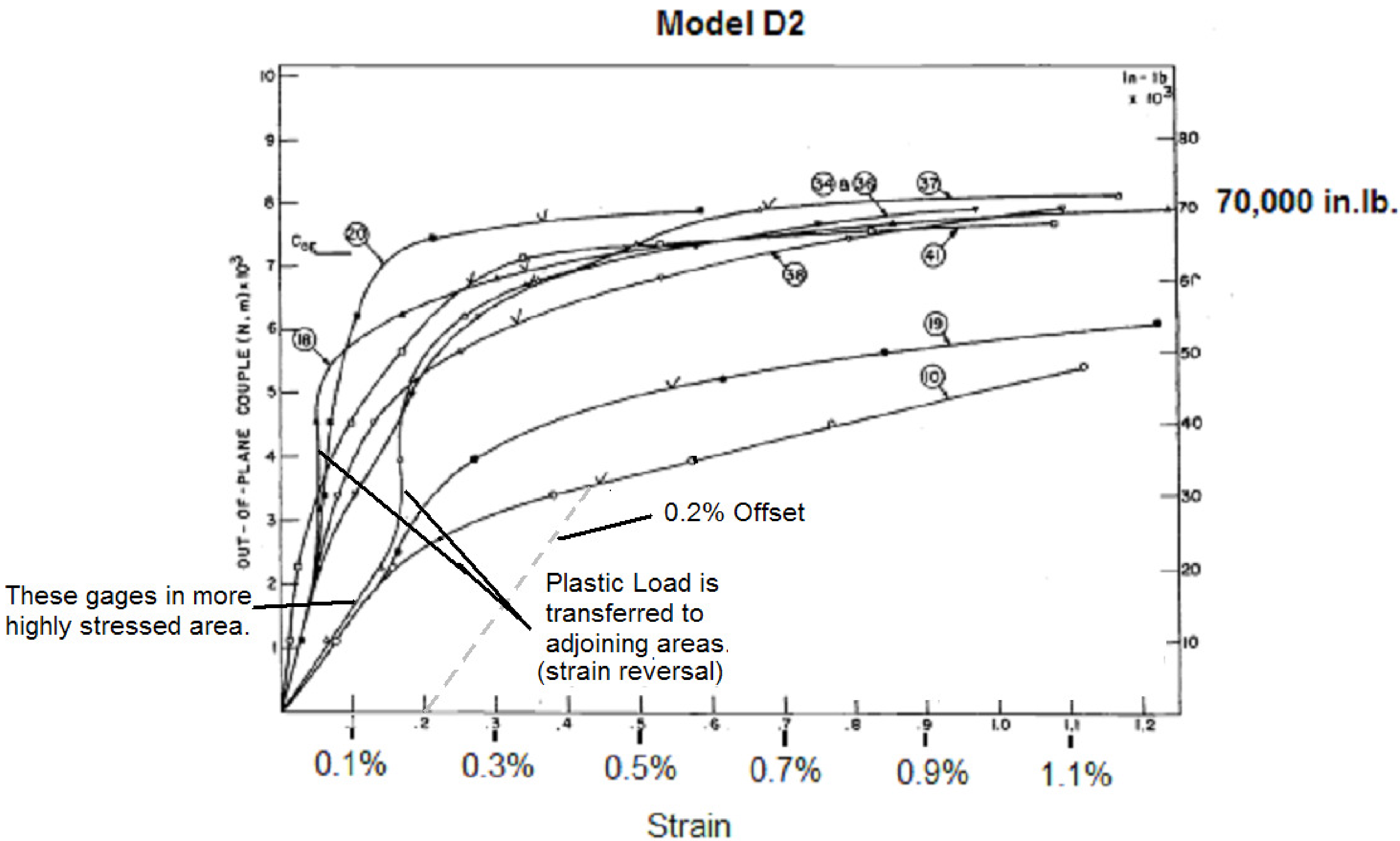


Fig. 32—A photograph of plastically deformed model C_2, subjected to 3730 N.m (33 in.-kip) out-of-plane couple, and sectioned along the transversal plane

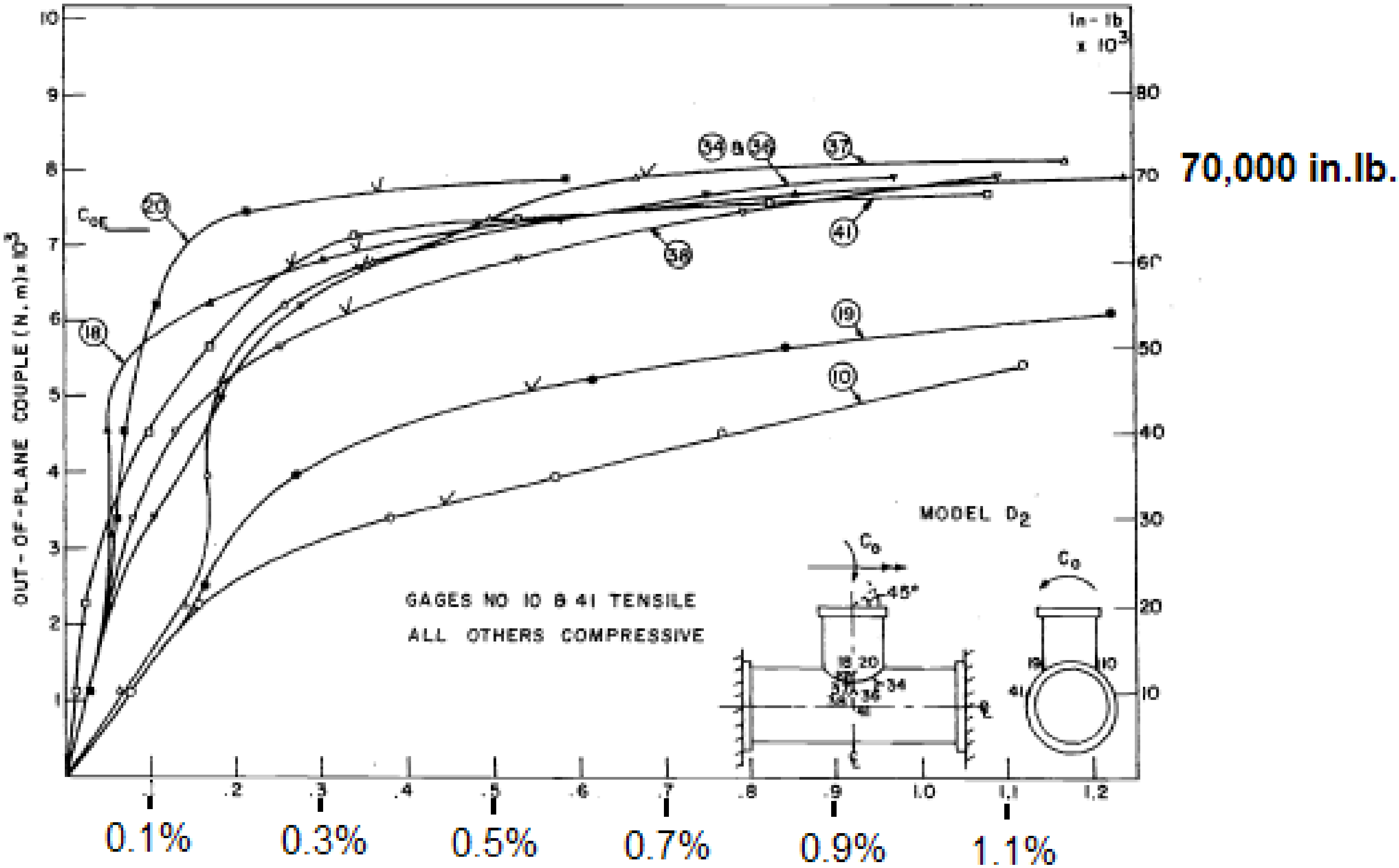
WRC 230 Model C2 and D2 deformation after loading are almost identical …

Cylindrical intersection was machined. Strain gages were applied, and then the specimen loaded in the out-plane direction.

Strain Gage Results on Out-Plane Loaded Model D2

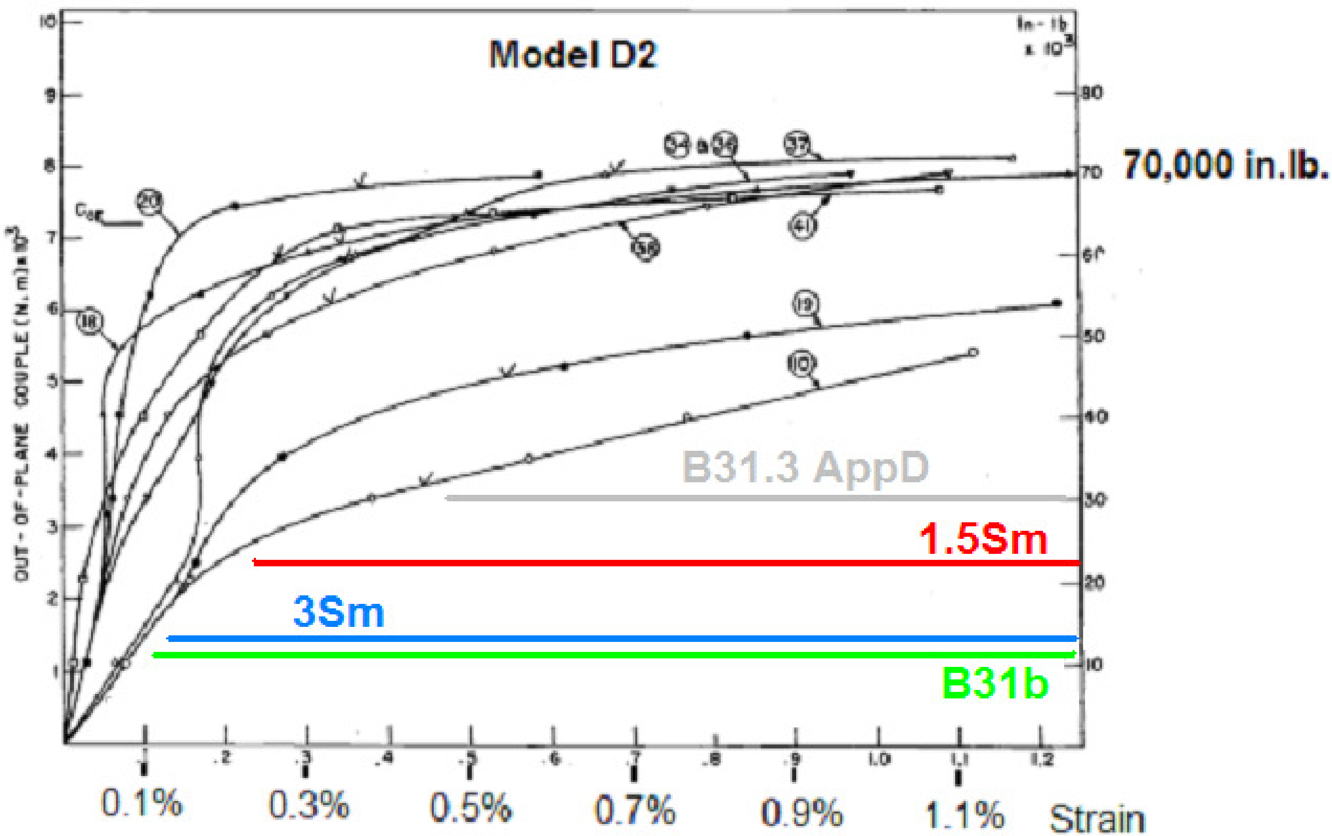


Model D2



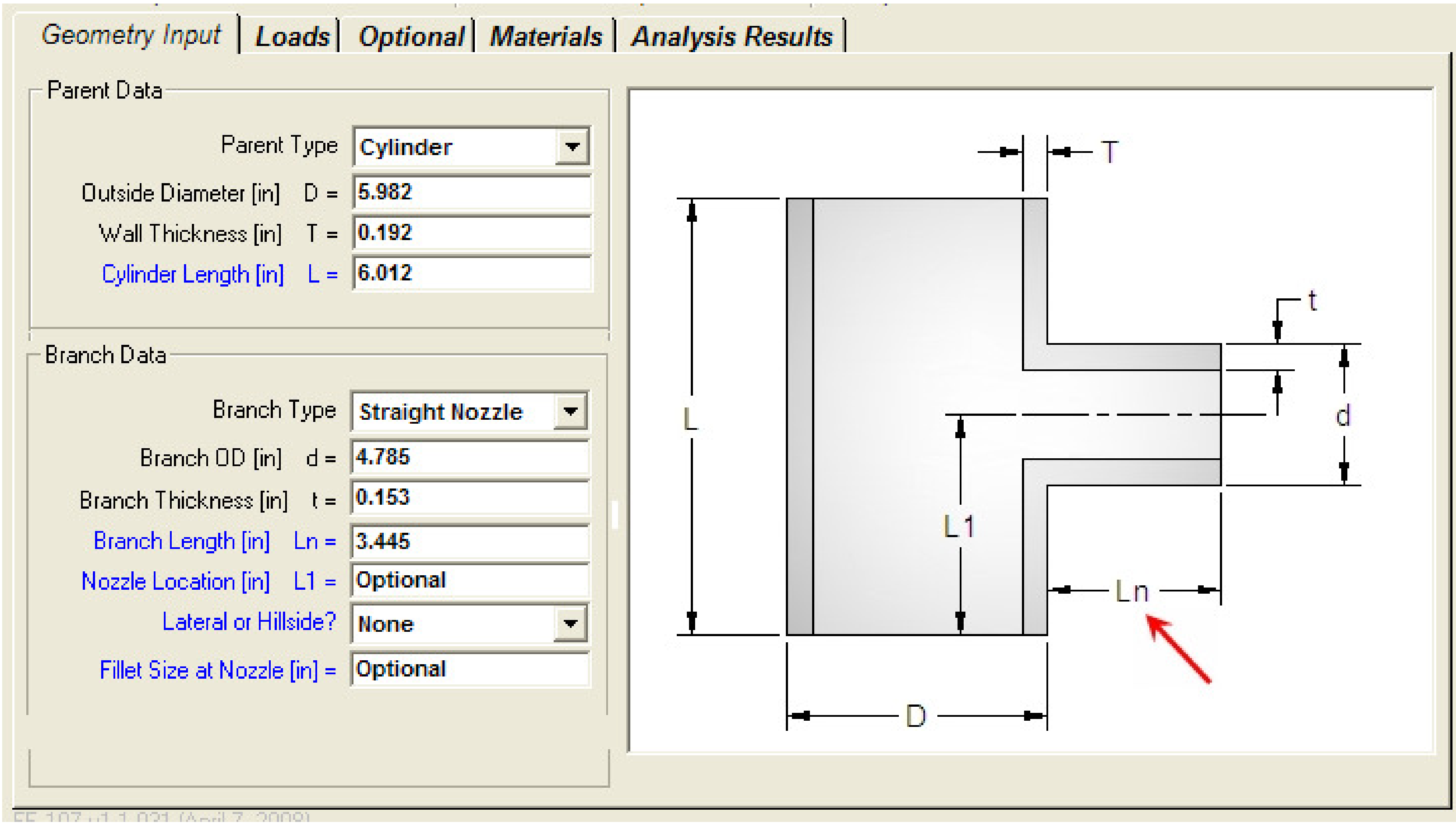
d/D	D/T	t/T	1.5Sm M/Z(mem)	3Sm M/Z(Sec)	B31-D	B31b	
0.79	29.70	0.79	9262.	5117.	11699.	4646.	
Z = 2.576 cu.in.			23,803	13,181	30,059	11,968	in.lb.

Allowed Nominal Loads



d/D	D/T	t/T	M/Z(mem)	M/Z(Sec)	B31-D	B31b	
0.79	29.70	0.79	9262.	5117.	11699.	4646.	
Z = 2.576 cu.in.			23,803	13,181	30,059	11,968	in.lb.
			PL<1.5Sm	Pl+Pb+Q<3Sm	iM/Z< 1.25(Sc+Sh)	i_bM/Z< 1.25(Sc+Sh)	

Evaluate 70,000 in.lb. external out-plane moment…

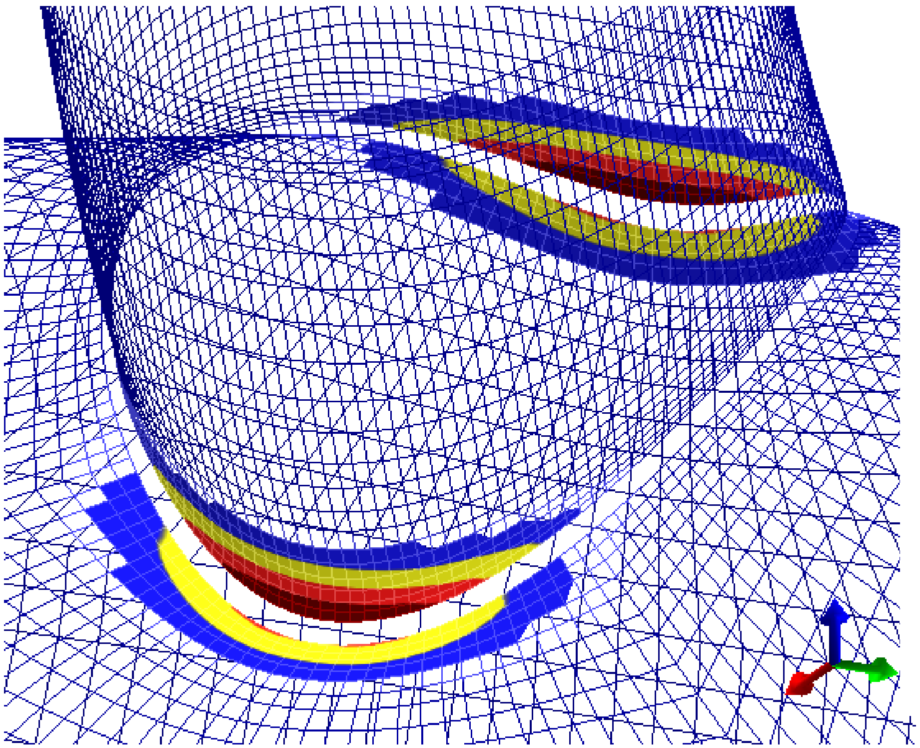


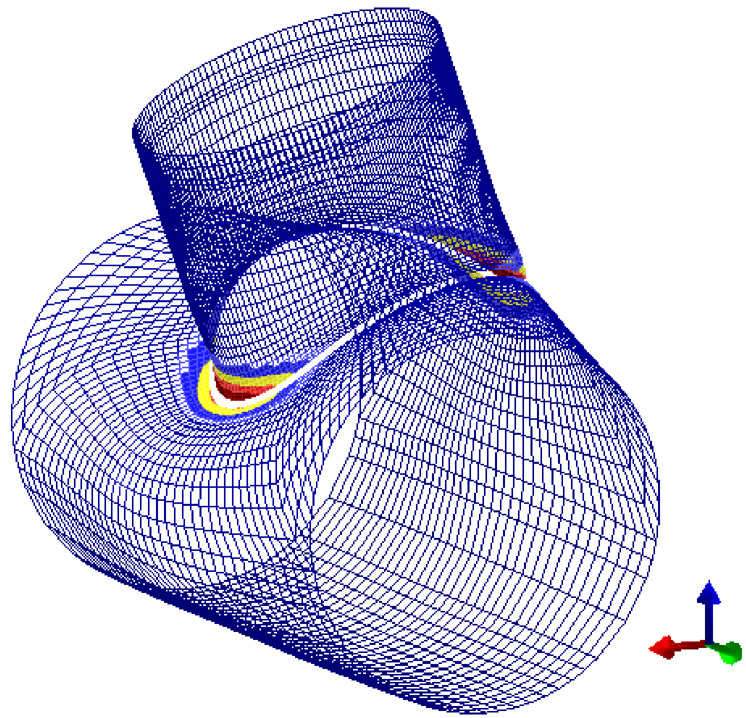
Shell models and WRC 429 intersection model – economic

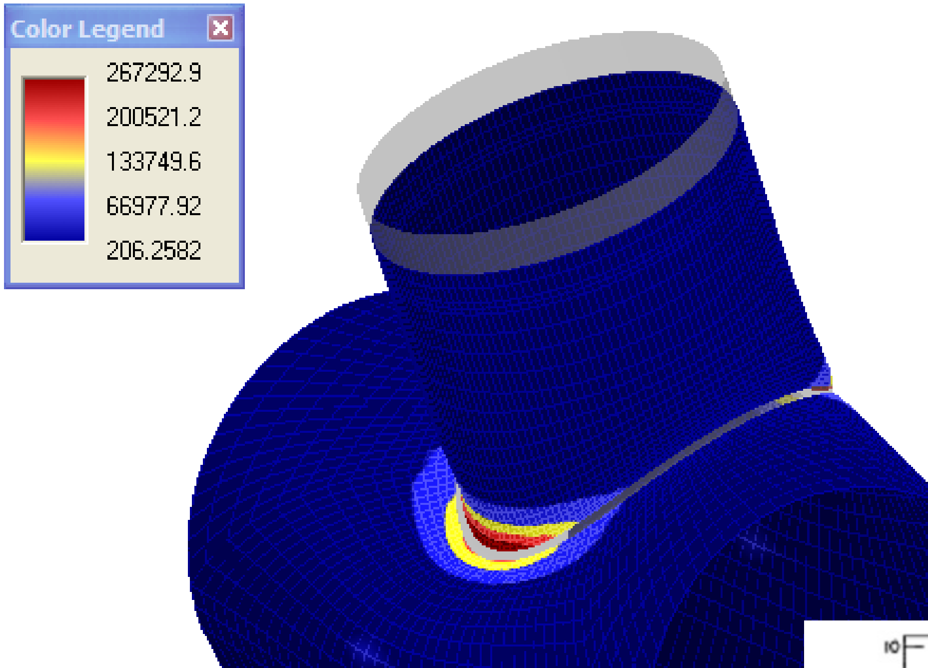
(easily run with Ansys, NozzlePRO, FE107, … FE107 input shown above.)

Source	Case	Pressure	Axial	In-Plane	Out-of Plane	Torsion
FEA	Peak	0.00	8.80	3.93	9.54	1.48
WRC 107	Peak		23.65	5.63	14.00	1.45
WRC 297	Peak		43.22	10.20	24.10	1.82
FEA	Secondary	0.00	8.80	3.93	9.54	1.48
WRC 107	Secondary		23.65	5.63	14.00	1.45
WRC 297	Secondary		43.22	10.20	24.10	1.82

All stresses greater than 80ksi = 2Sy are plotted.

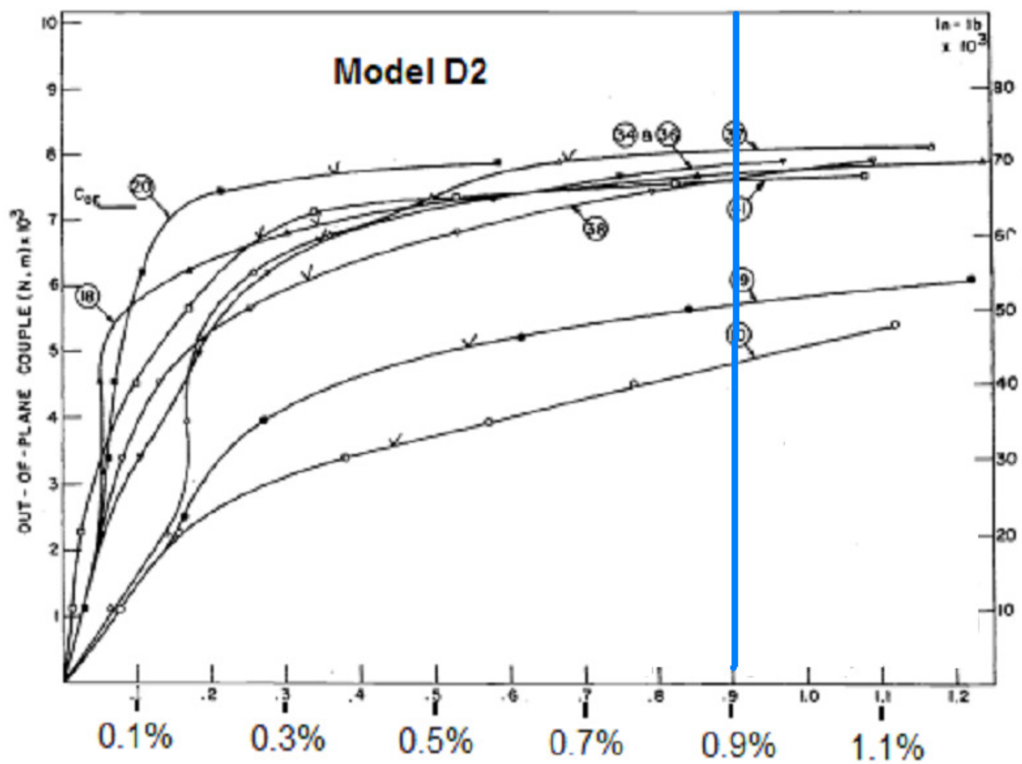


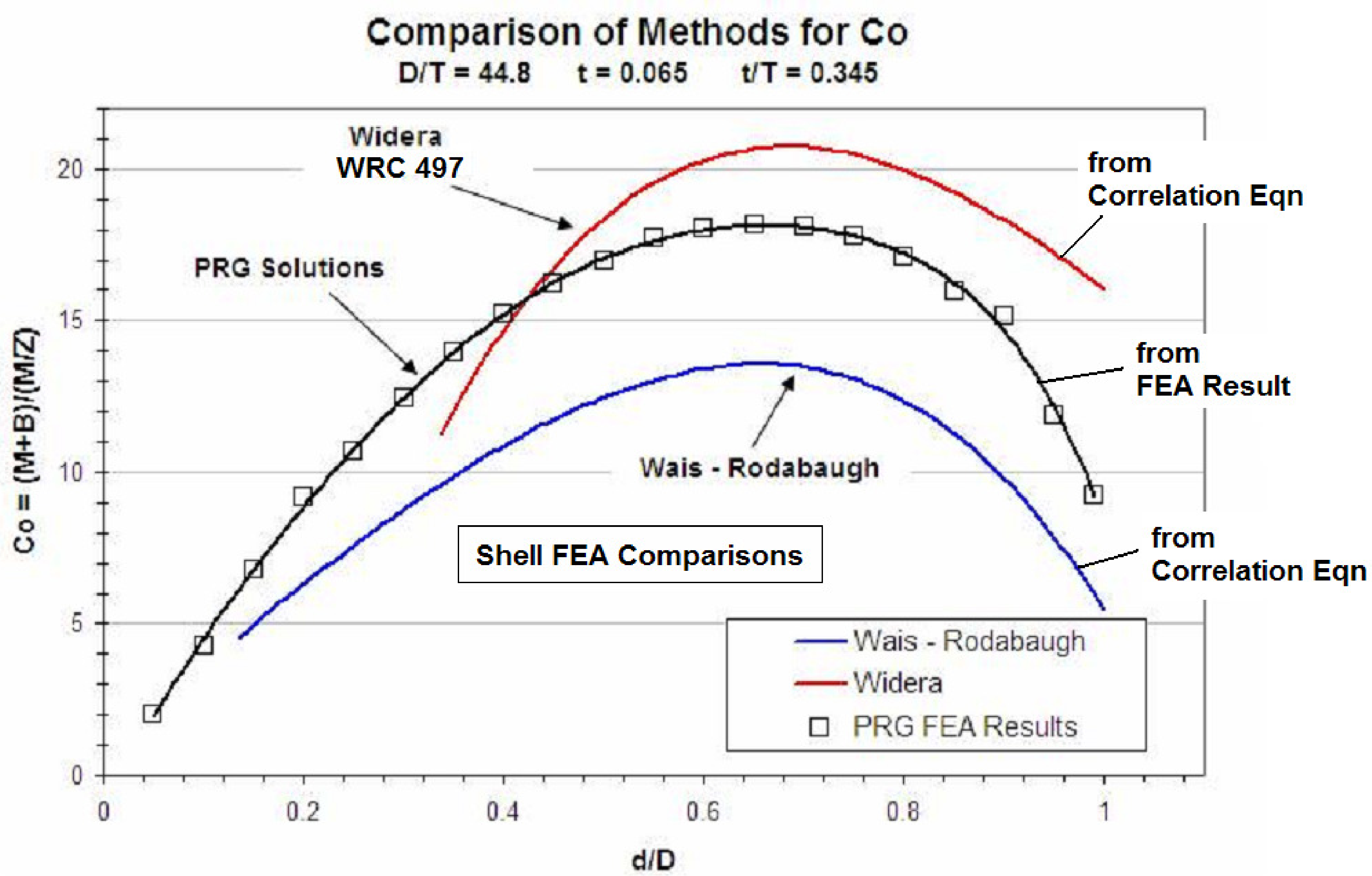




PL+PB+Q Outside

267,292/29e6 = 0.009





How accurate are the individual load component correlations from FEA that have been performed so far?

Elastic-Plastic Analysis Methods

Two inelastic options are available in the 2007 ASME VIII-2 for design of nozzles.

- **Lower Bound Limit Analysis**
- **Elastic-Plastic Analysis**
- **Lower bound limit analysis is for primary loads only.**
  - **A good alternative to nozzle reinforcement design and can offer substantial savings with little analysis effort. Individual components may be sized while the remainder of the vessel is designed by common rules.**
- **Elastic-plastic analysis can be used for any loading state.**
  - **Can be used to highly optimize a design. Significant reductions in minimum wall thickness can be achieved. 25% or more saving on material cost is routine.**
- **Even if material savings are not important, you may be able to use inelastic design to permit greater loadings. This is common when a customer revises the loads and fabrication is already underway. Helps avoid unnecessary changes.**

Lower Bound Limit Analysis

- **Represents an idealized lower bound estimate of the actual load to cause plastic collapse in the structure.**
- **Ensures that unrestrained plastic deformation does not occur (i.e. a plastic collapse state is not reached)**
- **A simple example of a lower bound limit analysis is a bar with an axial tensile load applied. The lower bound collapse load is the load at which F/A = Sy.**
- **The yield stress for the elastic-perfectly plastic material model is approximated by using 1.5S as the yield strength.**
    - **Using 1.5S ensures that the limit of 2/3 on yield is achieved but also considers the safety factor of 2.4 on UTS to ensure that high yield-to-tensile ratio materials are safely employed in designs.**
- **ASME requires a margin of 2/3 against this lower bound limit. This limit is achieved by using the specified load case combinations in Div 2 (essentially multiply the expected loads by 1.5).**

Lower Bound Limit Analysis

Requirements for lower bound limit analysis:

- **An elastic-perfectly plastic material model must be used.**
- **Analysis must not use geometric nonlinearity (small displacement theory must be used).**
- **Applies to primary loading only – for instance, pressure.**
- **Thermal loads are not valid and should not be analyzed as they are strain limited and the lower bound limit analysis approach is invalid.**
- **Should not be used in cases where the geometry may become unstable or experience geometric weakening under the applied loadings:**
  - **Compressive loads (external pressure, axial loads, etc)**
  - **Closing moments on elbows and bends**
  - **Out-of-plane loadings on nozzles or intersections**
  - **Any time geometrical weakening is anticipated**

Lower Bound Limit Analysis

Often, lower bound limit analysis is one step in a multi-part code compliance analysis:

- **Can only be used to address primary loads, such as pressure or piping sustained/weight loads.**
- **Since it isn't applicable to thermal loads, any nozzles with restrained thermal piping loads will require an additional analysis.**
- **Often, nozzles openings are design for internal pressure using lower bound limit analysis and then the secondary piping loads.**
- **Alternatively, these nozzles can be designed in a single step using an elastic-plastic analysis….**

Elastic-Plastic Analysis

- Elastic-plastic analysis attempts to predict the actual collapse load of the structure by taking into account the true stress-strain behavior of the structure.
- Is more complex than a lower bound limit analysis but does offer numerous advantages:
    - Primary and secondary type loading can be analyzed.
    - Large displacement theory must be considered.
    - Includes the effects of strain hardening – allows increased allowable design loads.
- In the 2007 ASME Div 2, a safety factor of 2.4 against collapse is required. This limit is consistent with the margin on UTS in the design-by-rules section (Part 4).

Why Use a Elastic-Plastic Analysis Method?

- Simplifies the code compliance post-processing since the results are a “go” or “no-go” answer.
- Stress categorization need not be performed. Limits on PL and PL+Pb+Q need not be satisfied.
    - This is really beneficial in complex 3D shapes where stress linearization may produce ambiguous results.
- SAVE MONEY!
    - These more accurate analysis methods allow you to reduce the required thickness and carry larger loads while maintaining a consistent safety margin.
- Fitness-for-service : reduce excess conservatism associated with elastic analysis methods and allow existing equipment to be used longer, in more severe operating conditions, and sometimes eliminate replacement costs.

Nonlinear Methods – Some final considerations

- **You must still satisfy requirements for ratcheting, fatigue analysis, and local strain limits (prevent necking –** not included **in most FEA material formulations used for PVP).**
- **Any design margins that are used should meet the intent of the governing design code. For instance, if you are designing parts of a Div 1 vessel, you should use a limit of 3.5 against plastic collapse, not 2.4 as used in the new Div 2.**

Conclusions:

1) Relatively straight-forward shell FEA can answer a considerable number of questions about combinations of loads, i.e. are SRSS methods good enough when pressure is a principle load? (A boundary condition of one end free provides the most conservative stress assumptions.)

3) For small d/D, Appendix D of B31.3 may be non-conservative, although including the reduced branch connection SIF equation can bring the allowables in line with ASME Section VIII requirements. *(CAESAR, for example has this as an option.)* *(See the high red peaks in the allowed nominal stress comparison plots.)*

4) Very high local elastic (FEA) type stresses may still be very far from failure or collapse and are mostly objectionable from a fatigue crack point-of-view.

5) Whether membrane limits are included or not – does not make much difference for out-of-plane loads investigated unless pressure stresses are high. (EN-13445 approach does not include P+M for membrane stresses.)

6) Likely problems at nozzles or pipe connections occur because of incorrect stiffness or friction modeling, or unstable friction supporting (i.e. line can walk around.) or because of high temperature.

7) Including free end displacements in local membrane solutions will require more FEA that is probably NOT warranted (PRG opinion).

Remaining April PRG Webinar Schedule

Tuesday April 22 – Buckling of Piping Components (Part 1a). Buckling of Pressure Vessel Components (Part 1b).

Tuesday April 29 – FRP Pipe Failures and Lessons to Be Learned." (Part 1) *(Guest Lecturer – Dr. Hans Bos)*

Thursday May 1 – Pipe & Pressure Vessel Ethics – Conditions of Disagreement

Tuesday May 6 – Piping Problems to Avoid – Examples (part 1)

Tuesday May 13 – Piping & Vessel Problems to Avoid – Examples (part 2)

Tuesday May 20 – Piping & Pressure Vessels – When to Worry – Examples

Tuesday May 27 – CAD and Pipestress

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