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PRG Webinar 043

External Loads on Nozzles and Pipe Intersections

(Part 1)

Paulin Research Group

Tony Paulin

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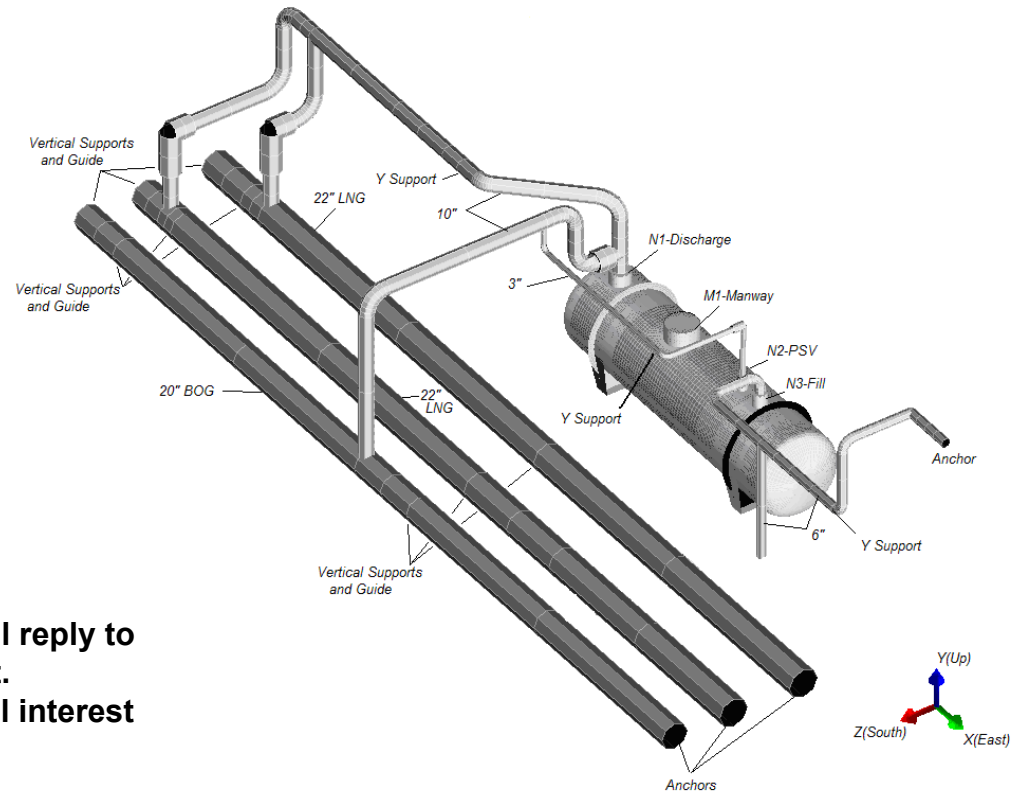
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Assumptions Control Loads on Nozzles & Pipe Intersections

- 1) Load Assumptions (how accurate are the loads)
- 2) Stress Assumptions (how accurate are the stresses)
- 3) Allowable Assumptions (how accurate are the allowables)

Governing Equation:

$$(\text{Stress Factor})(\text{Load})(\text{Geometry}) < \text{Allowable}$$

BEAM: $(i)(M/Z) < \text{Allowable}$

SHELL: $(\text{FSRF})(P_i + P_b + Q) < \text{Allowable}$

$$(\text{FSRF})(\text{Shell FEA or WRC Result}) < \text{Allowable}$$

$$(\text{FSRF})(C_2)(M/Z) < \text{Allowable}$$

BRICK (NO WELD): $\text{Stress (FEA)} < \text{Allowable}$

BRICK (WELD): $\sigma(\text{SCL}) < \text{Allowable}$

How accurate do we need the stress, load and allowable?

How accurate do we need the stress, load and allowable?

It depends on what you want to do:

If you want to perform a design according to the Code, then you usually need to be sure you're making conservative assumptions, and that you're far enough away from failure or economic impracticality.

If you are performing a failure analysis, then generally we're looking for accuracy in all key variables.

$$\begin{aligned}w_R &= (dR/dS w_S)^2 + (dR/dL w_L)^2]^{1/2} \\&= [(L w_S)^2 + (S w_L)^2]^{1/2} \\&= \{[(LS)(\%err_S)]^2 + [(LS)(\%err_L)]^2 \}^{1/2} \\&= (LS)[(\%err_S)^2 + (\%err_L)^2]^{1/2}\end{aligned}$$

w_i – possible error in property “i”
 w_R – possible error in property “R”
 dR/dS – variation in R as a function of variation in S
 dR/dL – variation in R as a function of variation in L
S – Stress or Reaction function
L – Load Function
 $\%err_S$ – percentage error in S
 $\%err_L$ – percentage error in L

$$\begin{aligned}LS + w_R &< \text{Allowable} \\LS + w_R &< \text{Failure}\end{aligned}$$

... PRG Piping Checklist
can help evaluate %err.

The Stress, Load, Allowable Assumptions ...

Load Assumptions:

- 1) Weights are correct (valves & actuators)
- 2) Frictional effects are considered if necessary.
- 3) Thermal expansion coefficients are correct
- 4) Temperature distribution is correct (thru-wall, thermal transients)
- 5) Structural steel supports are rigid or accurately modeled
- 6) Local flexibilities are entered properly. Differences between pad reinforced, unreinforced, laterals, hillsides, etc. are properly considered. *(Are point springs enough in these cases? For hillsides and laterals – possibly not – need 6x6.)*
- 7) Ovalization effects on stiffness are considered where needed. *(Where is the flange or stiffener with respect to the elbow, nozzle or pipe junction?)*
- 8) Settlement is considered if present
- 9) Thermal expansion of attached components is properly evaluated, i.e. the thermal expansion from the top of the skirt to the nozzle elevation.
- 10) A sufficient amount of the piping system is included in the model. *(If you don't have all the model – you don't have all the loads.)*

Stress Assumptions:

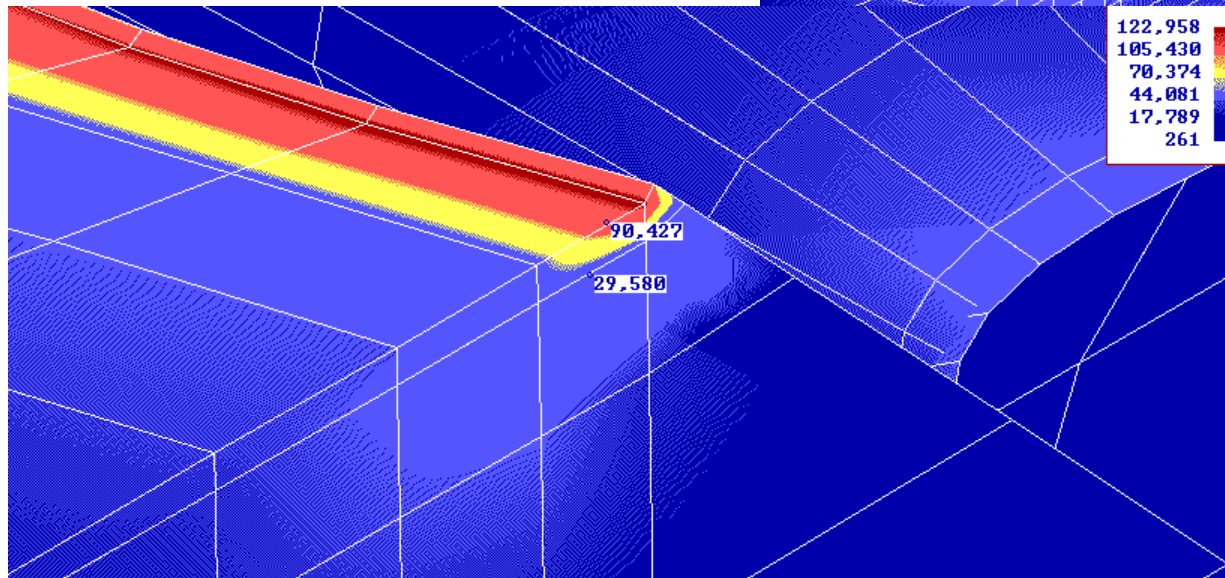
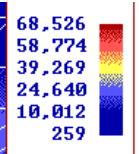
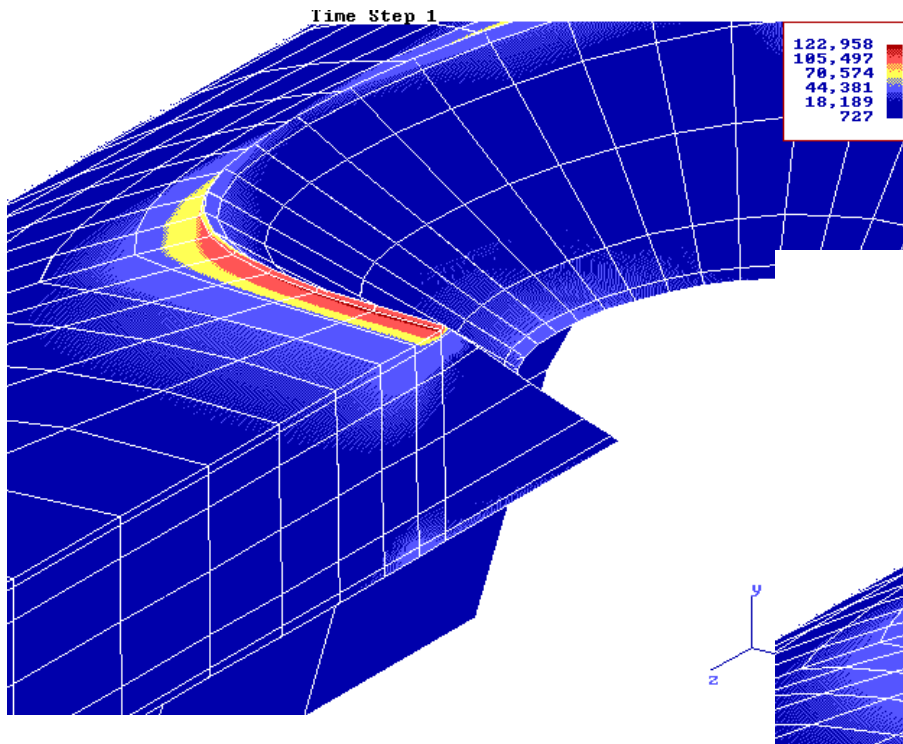
- 1) Moments through the run/branch are independent (*for B31*)
- 2) Combination of moments on any single component are made by SRSS.
- 3) Is the d/D significant enough so that the loads in the vessel shell effect the nozzle/branch connection stresses? (*$d/D > 0.4$ interaction may increase*)
- 4) Does pressure design “use-up” all of the primary load “allowed stress?”
- 5) Does the Code pressure stress design match the FEA design 100%? *Can a Code pressure design show to be overstressed in FEA? (Yes – just like many B16.5 flanges are shown to be overstressed when ASME Section VIII Div 1 App 2 rules are used.)*
- 6) Stress factors are accurate
 - a) Ovalization (how close is the flange?) *Effect on pressure stresses at intersections, (nonlinear effect)*
 - b) Geometric parameters (*i.e. $D/T < 100$ for B31 Appendix D*)
 - c) Geometry is known (*olets, forgings, size-on-size, hand ground radii*)
 - d) FEA singularities are not included in the solution or are minimized so that they do not affect the results.
 - e) Is the stress along the SCL determined correctly? (*From singularity need small area if going to use stresses. Usually 6 or more elements through the thickness, or use forces.*)

Allowable Assumptions

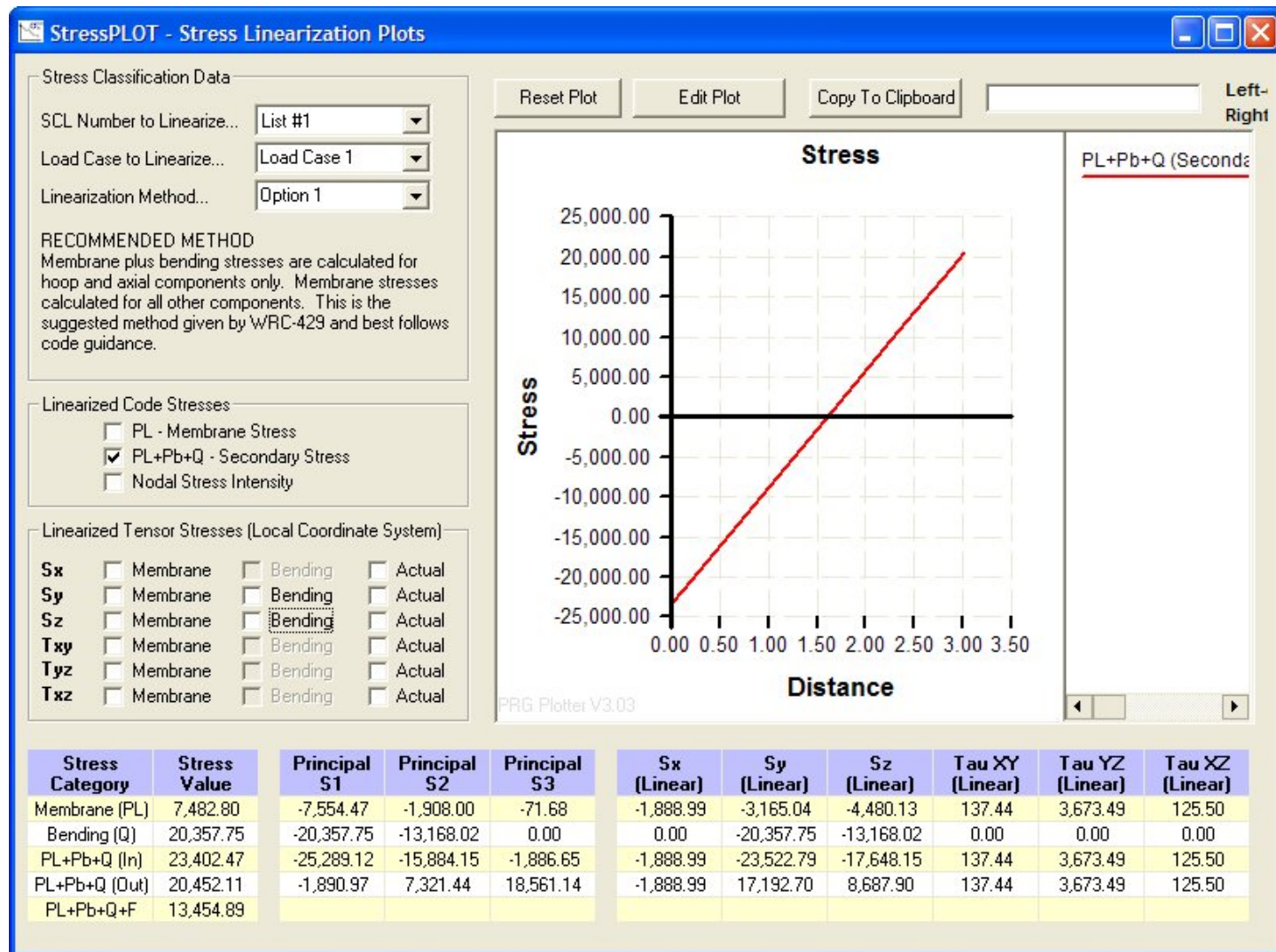
- 1) Pressure safety usually 2.4 to 3.0, and for intersections – a little higher.
- 2) External primary loads – usually 2 to 5.
- 3) Fatigue – $SF > 4$ when cycles < 300 ? And $SF = 2$ when cycles > 3125 , although probably dealing with a 97.6% probability of survival at 100% of the allowable stress.

Example items that determine accuracy of the assumptions ...

FEA Singularities



Proper use of stress linearization, or elemental forces (instead of stresses) minimizes the effect of FEA singularities at welds.





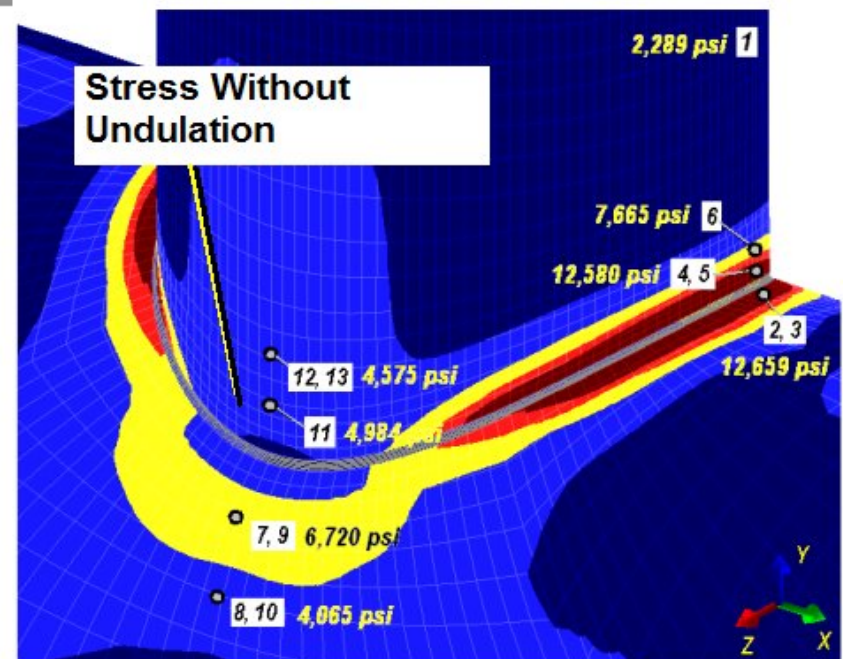
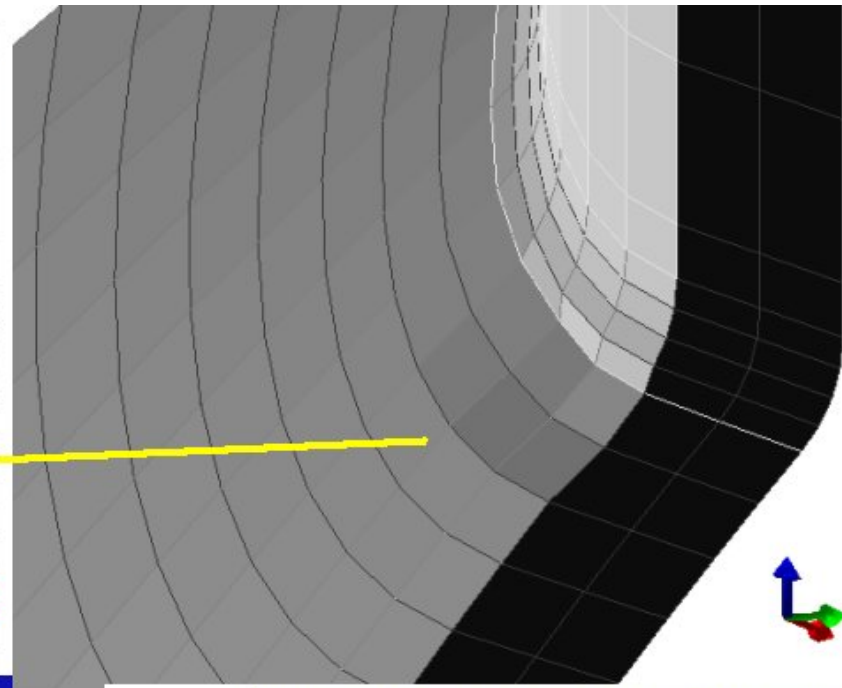
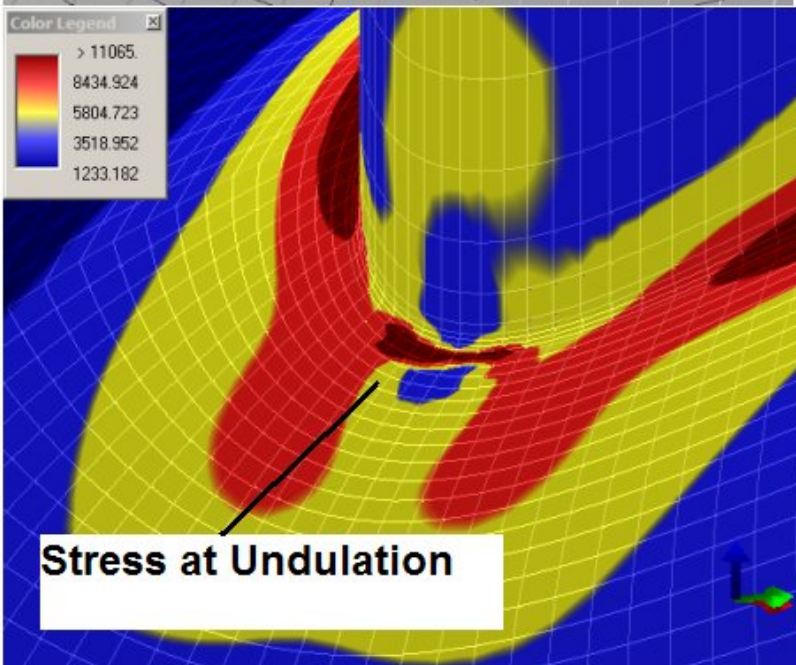
Actual Geometry Effects on Stress ...

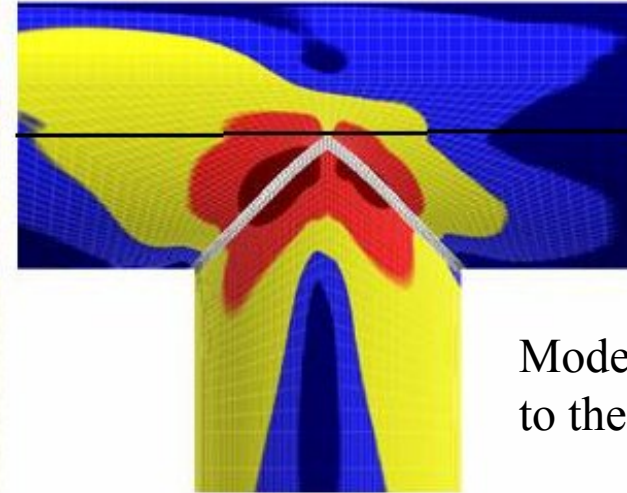
Hand-ground radii



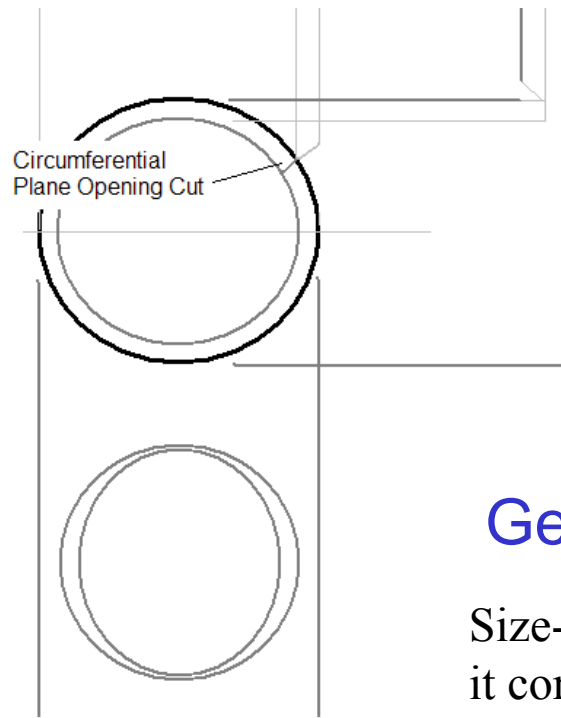
Geometry Effects... radii

Variations in Stress due to radii undulations

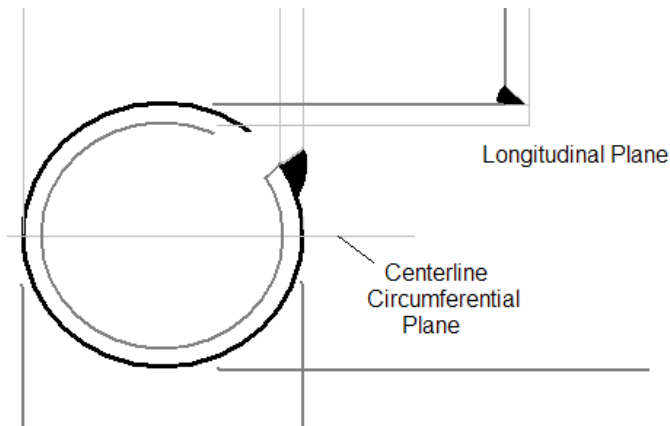




Model: Exactly
to the centerline



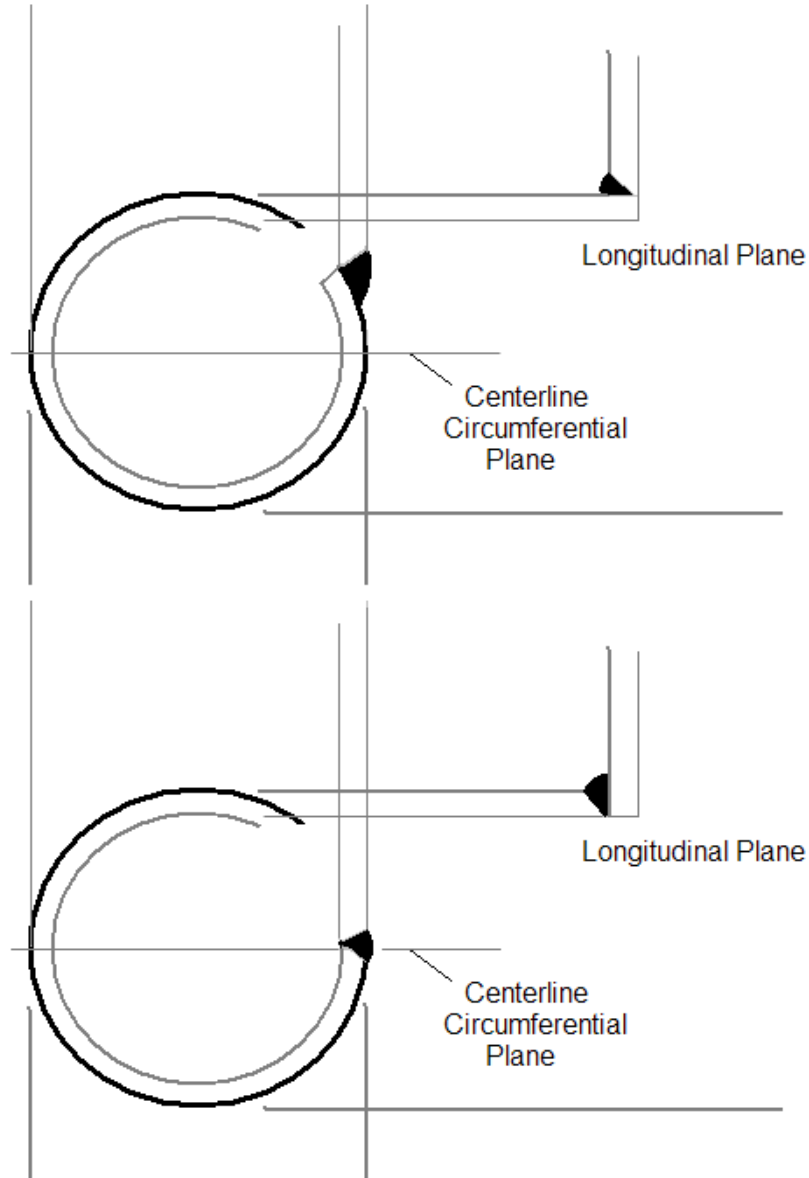
Top View "Cat's" Eye Opening



Geometry Effects - Fabrication

Size-on-Size "Cat's Eye" Effect – How can we analyze it correctly if we don't know how it's fabricated?

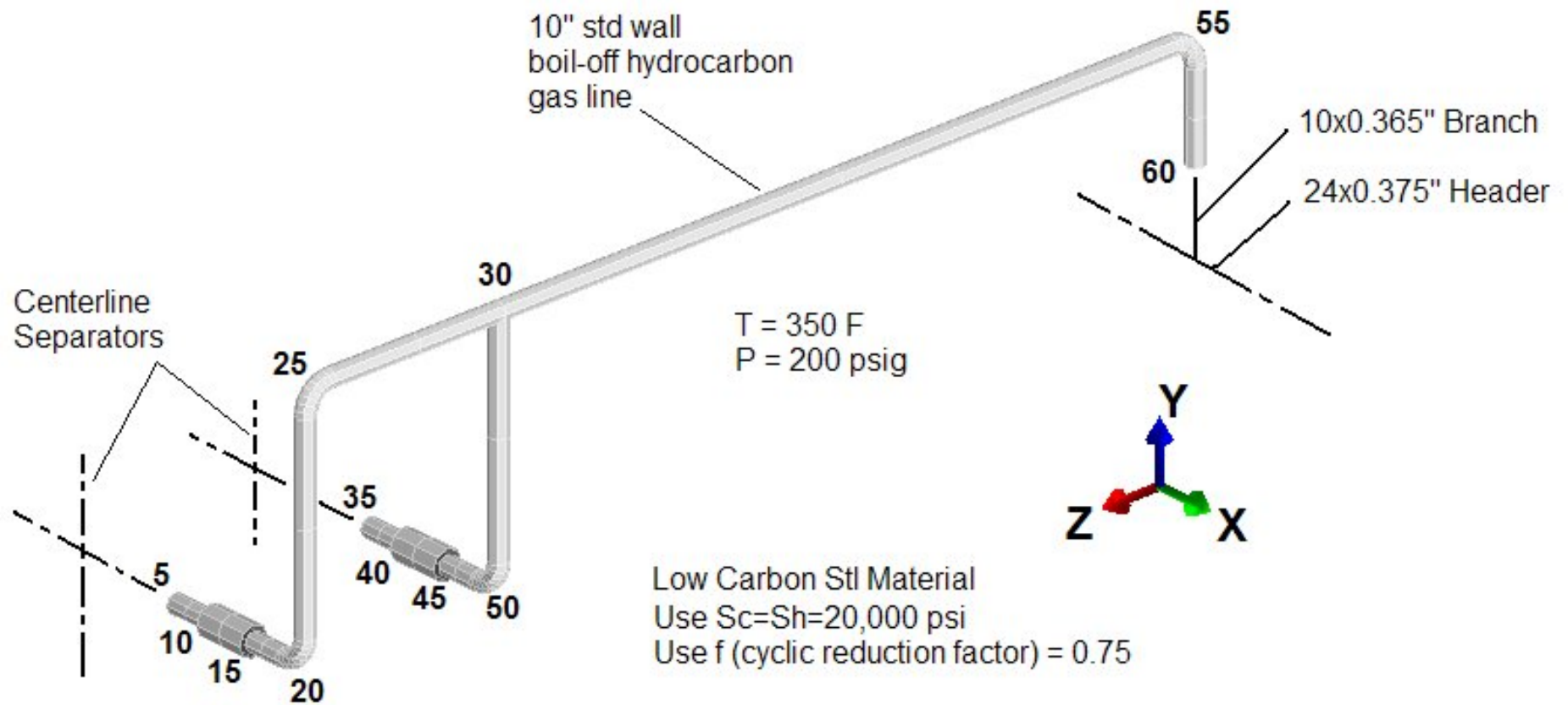
Typical Size-on-Size Fabrication Details



What you actually get is a function of:

- 1) automation of the fabrication facility,
- 2) established practice at the automation facility, and
- 3) how your inspector feels the days when he's walking through the shop.

Example External Load Analysis



Expansion Stress Allowable Calculation

Cyclic Reduction factor = 0.75 ... *specified by client ?*

$$6N^{-0.2} = \text{cyclic reduction factor} = 0.75$$

$$N = (0.75/6)^{-5} = 32,768 \text{ cycles } (\text{what client is telling us})$$

Assume: $32,768 / 40 / 365 =$ a little more than 2 cycles per day

$$S_A = f(1.25(S_c + S_h) - S_l) = (0.75)(1.25)(20,000 + 20,000)$$

(assume S_l is small)

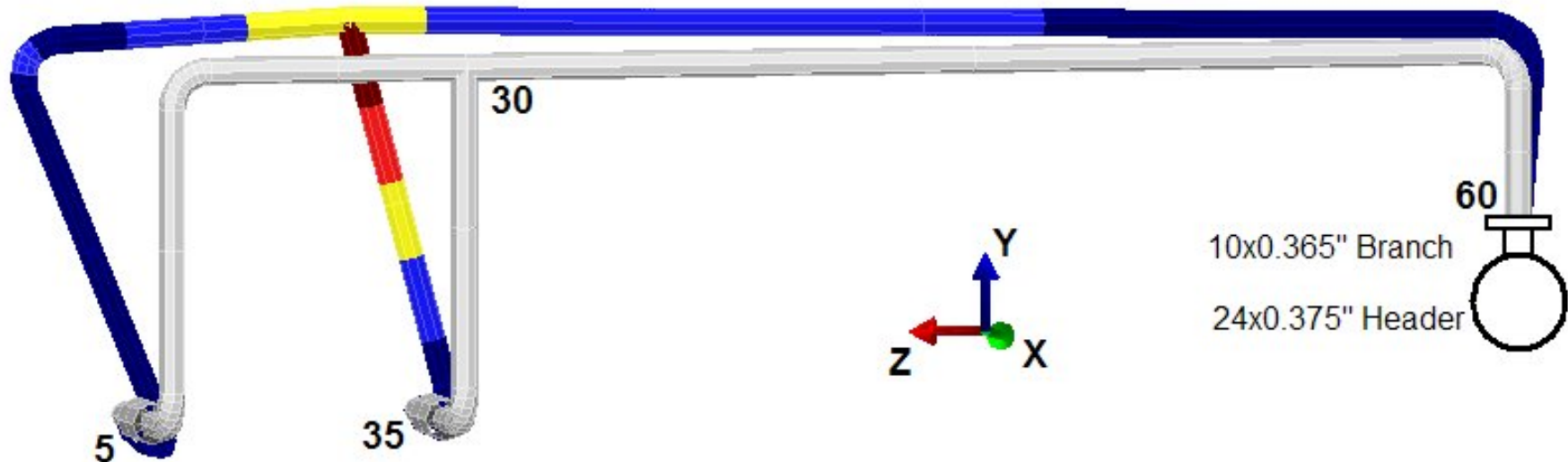
$$\mathbf{S_A = 37,500 \text{ psi}} = \text{allowable expansion stress (fatigue)}$$

Typical First Analysis Results



$52,134 > 37,500$ psi

(high stress at node 30)



Will flexibility at node 60 help?

Finite Element Analysis Using General or Specific Tool

Compute Surface Stiffness at Node 60:

Geometry Input | Loads | Optional | Materials | Analysis Results |

Parent Data

Parent Type: **Cylinder**

Outside Diameter [in] D = 24

Wall Thickness [in] T = 0.375

Cylinder Length [in] L = Optional

Branch Data

Branch Type: **Straight Nozzle**

Branch OD [in] d = 10.75

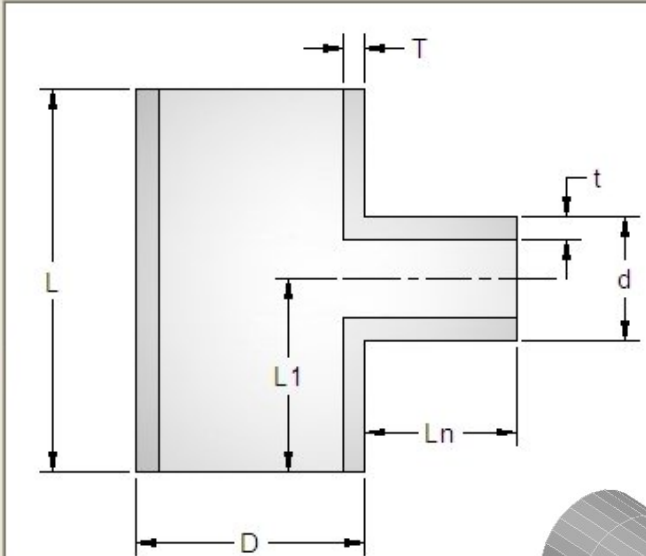
Branch Thickness [in] t = 0.365

Branch Length [in] Ln = Optional

Nozzle Location [in] L1 = Optional

Lateral or Hillside? **None**

Fillet Size at Nozzle [in] = Optional

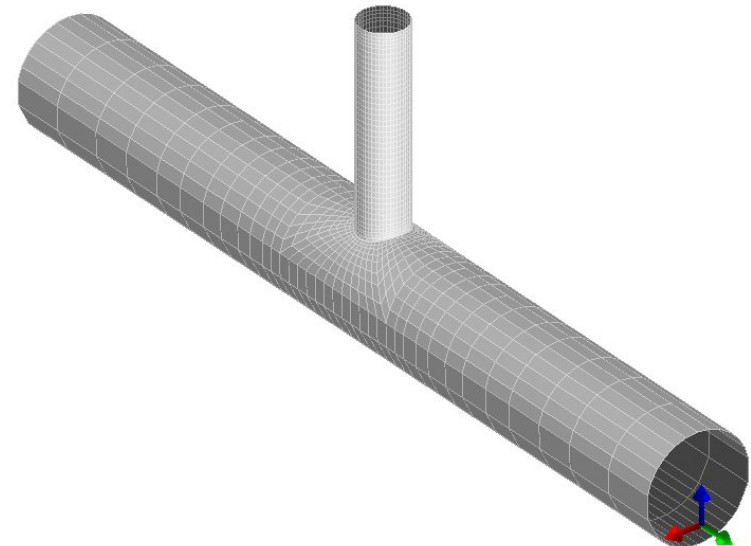


FEA Stress | Allowables | **Flexibilities** | FEA vs. WRC | WRC 107 | WRC 107

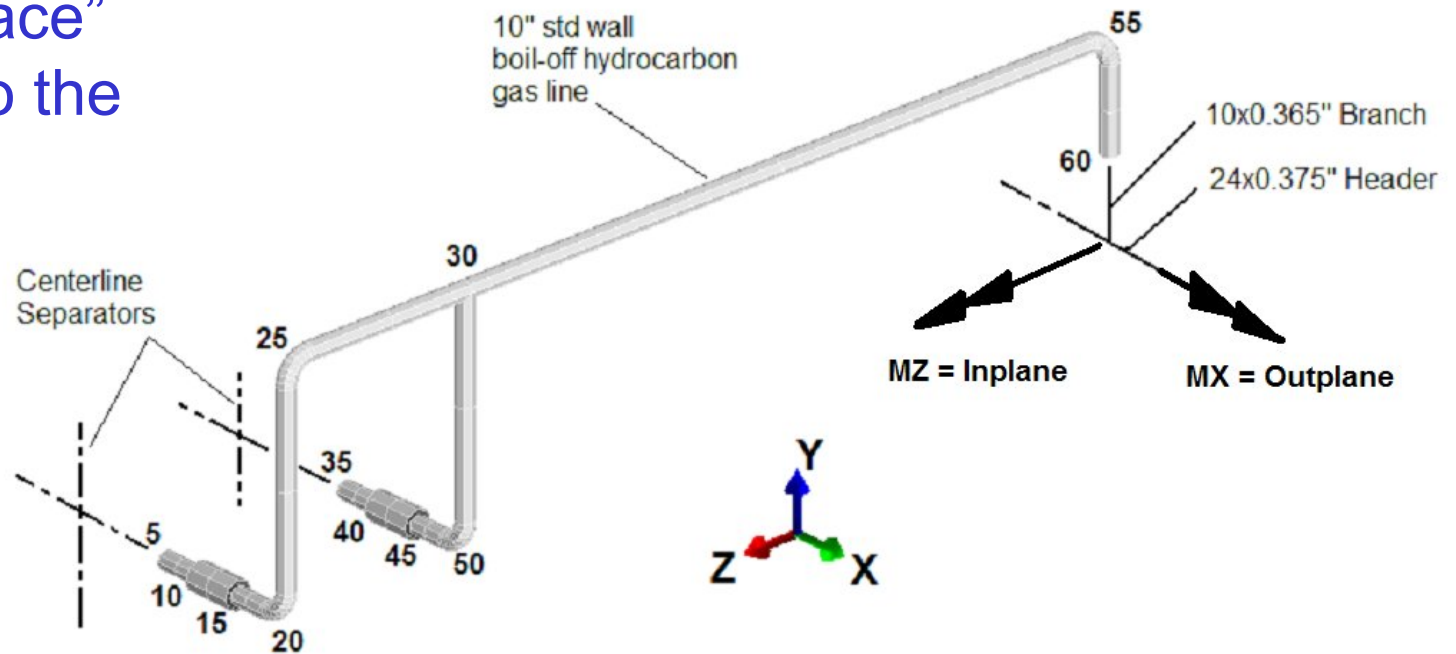
The following are the nozzle flexibility results generated by the Finite Element Method. These values are used in a piping beam analysis for a more accurate model of the actual nozzle intersection stiffnesses:

Axial Stiffness [lb/in.]	269774
In-Plane Bending Stiffness [in.lb./deg]	895134
Out-of-Plane Bending Stiffness [in.lb./deg]	145080
Torsional Stiffness [in.lb./deg]	1.33E+07

Example using FE107 or FESIF, can get same result from any FEA program.



Insert "Surface" Stiffness into the Model



Axial Stiffness [lb/in.]	269774
In-Plane Bending Stiffness [in.lb./deg]	895134
Out-of-Plane Bending Stiffness [in.lb./deg]	145080
Torsional Stiffness [in.lb./deg]	1.33E+07

Calculated Stiffness

Node List
60

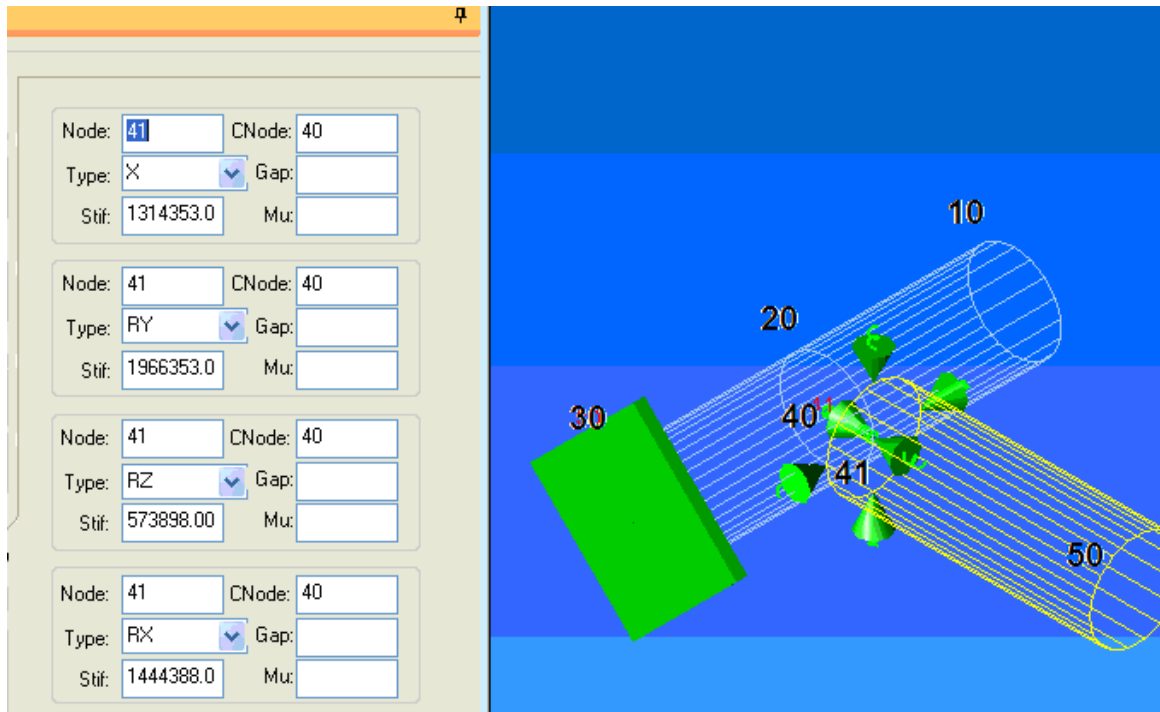
Restraint Data
Directions (see ?-help) **Anchor**

Base Node (Opt.)

Stiffnesses
lb. <or> in. lb. per deg.

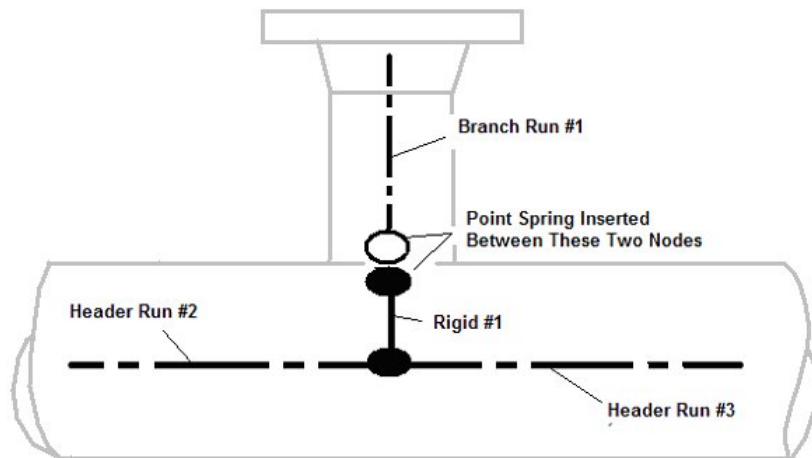
X	Y	Z	MX	MY	MZ
rigid	rigid	rigid	145080	rigid	895134
			Outplane		Inplane

Stiffness Inserted into
Program...

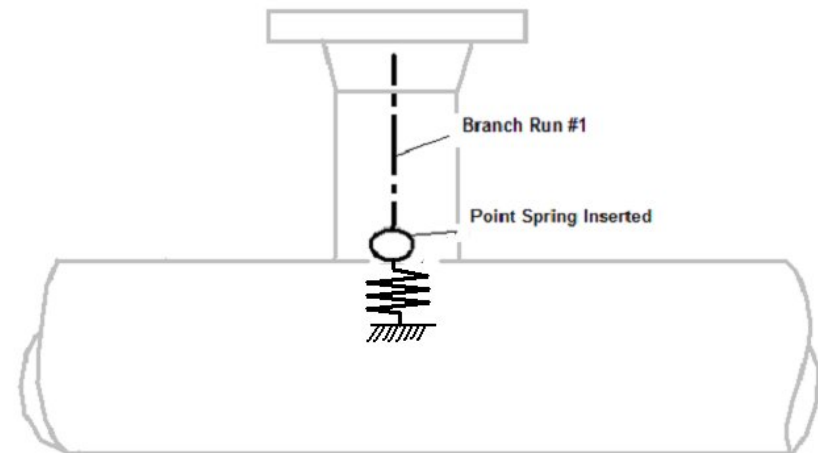


Examples of
how Stiffness
is entered in
CAESAR II.

“Surface” stiffness shown on
right, “thru-system” stiffness
shown on left.



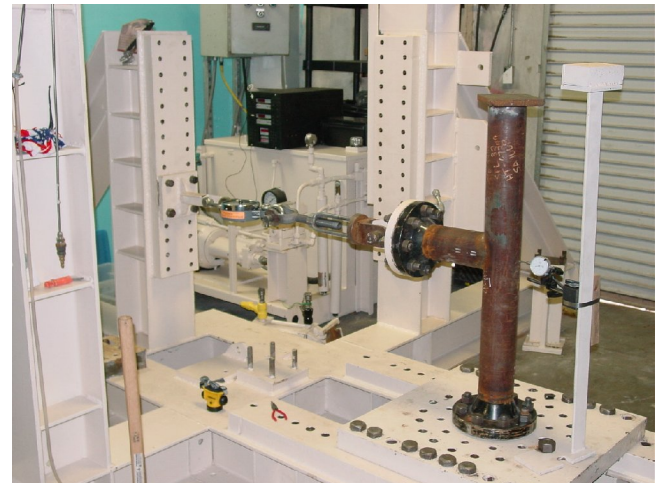
Nozzle/Branch Flexibility at a “thru-
node” in the piping system



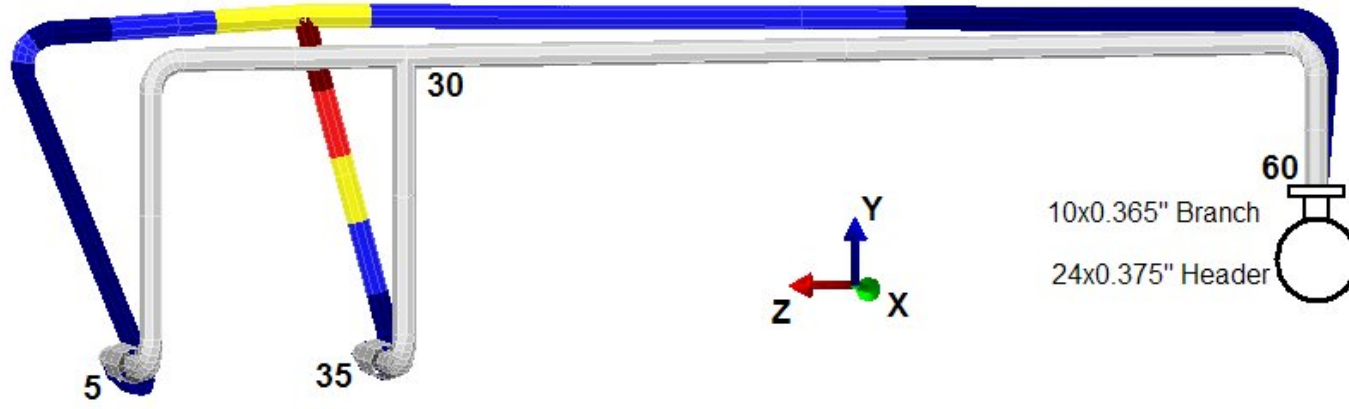
Nozzle/Branch Flexibility at a piping
system boundary condition

Three Different Intersection Stiffness Approaches

- 1) Ignore intersection stiffness (rigid)
- 2) Assume completely flexible (envelope solution)
(stiffness = 0)
- 3) Enter accurate stiffness from FEA or other source
(test) or reliable document.

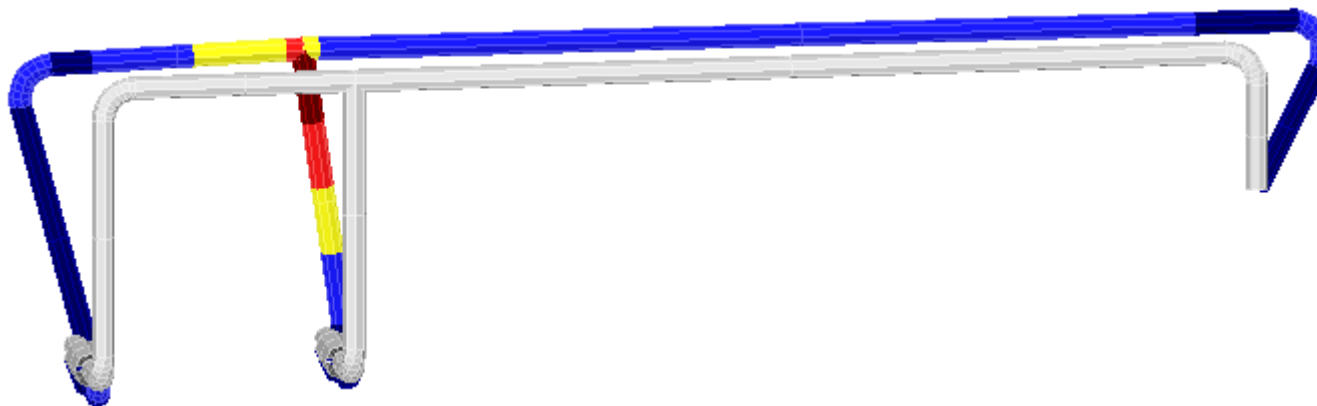


Different Stiffness Methods



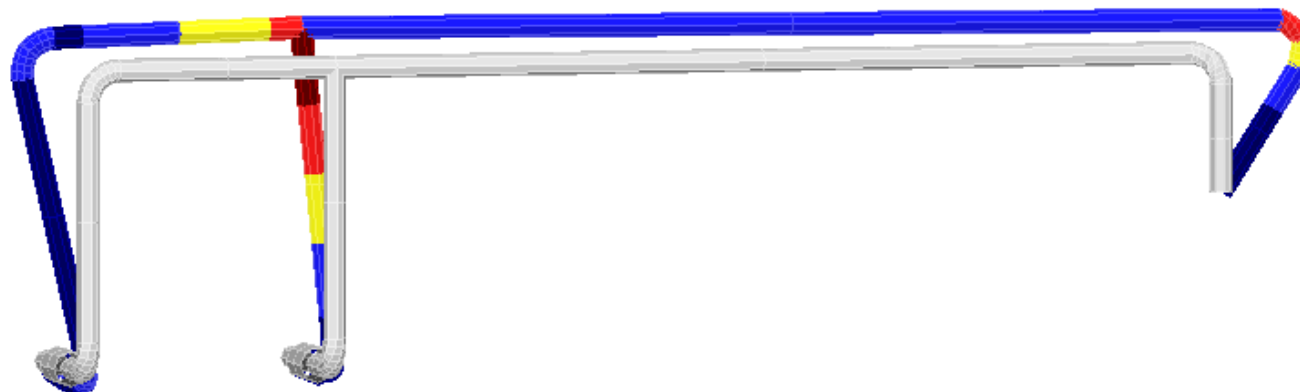
Stress at 30 =
52,134 psi
MX at 60 = 285,826

Rigid



Stress at 30 =
25,944
MX at 60 = 54,179
in.lb.

FEA Stiffness

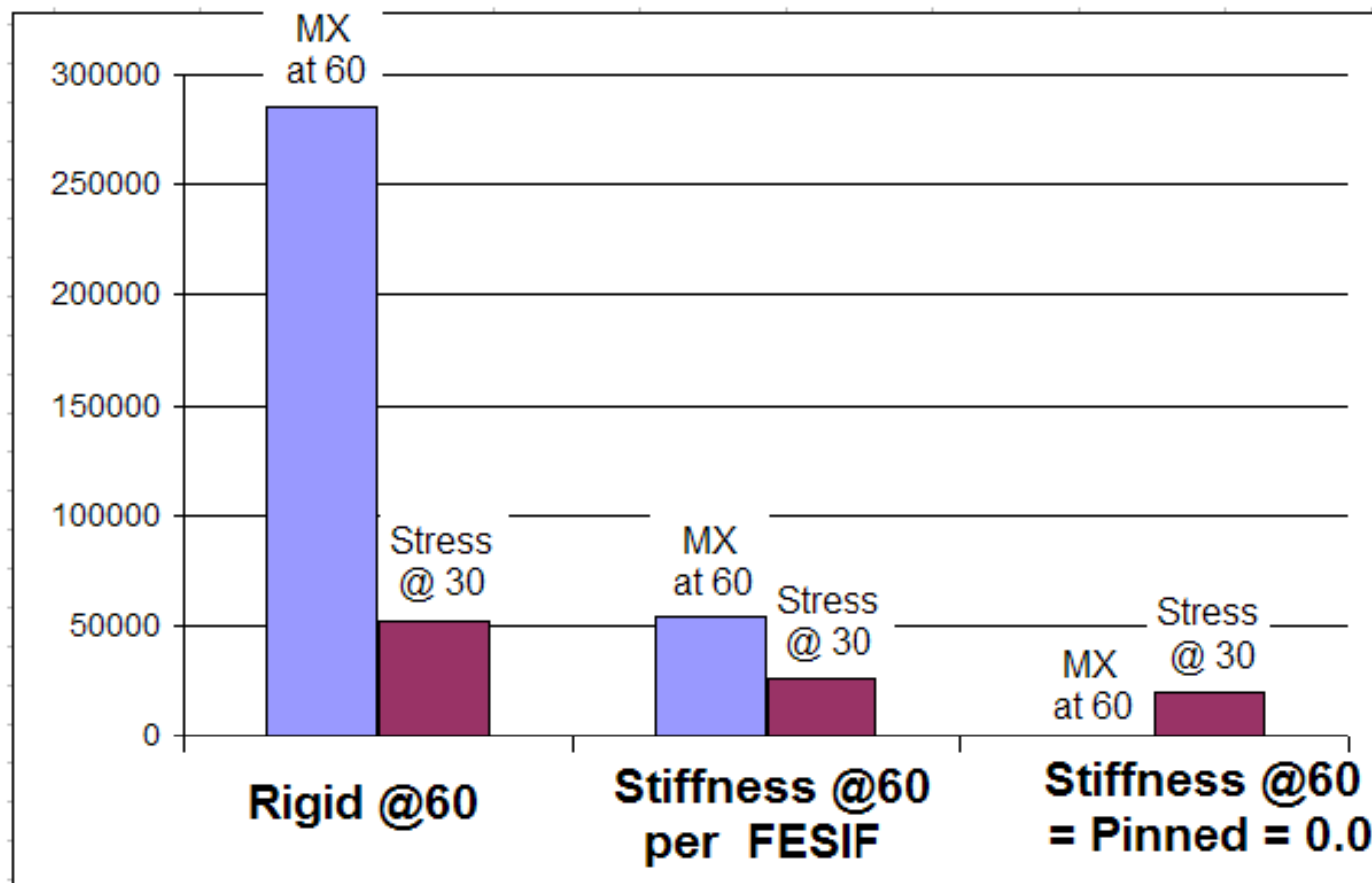


Stress at 30 =
19,792
MX at 60 = 0

Zero Stiffness

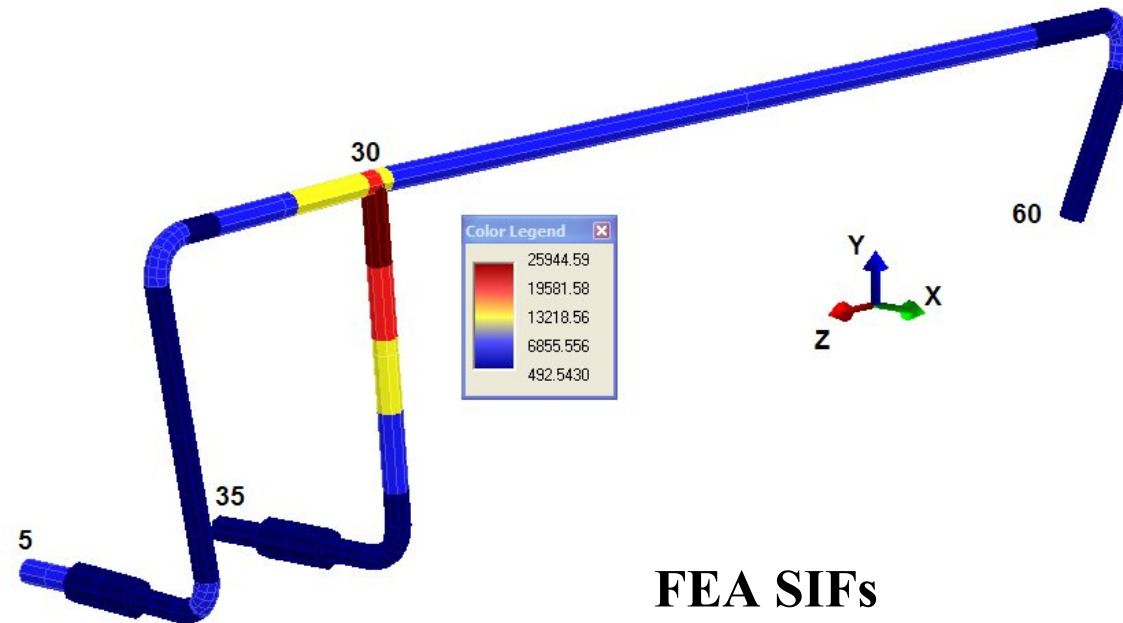
Results of Different Intersection Stiffness Models

Rigid and pinned certainly envelope the stress at node 30 for this problem. (We'd run the pinned end solution, see that it helps the stress at node 30, and then would buy FESIF, NozzlePRO or FE/Pipe and run the FEA solution, <or> would ask the FEA group to provide the stiffnesses.)



Flexibilities (stiffnesses) Solved Stress Problem at 30

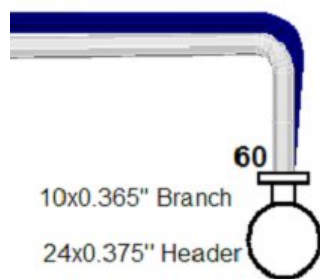
What about at Node 60?



Expansion Stress
Allowable

$$25,994 < 37,500 \text{ psi}$$

The Over-Stress Problem at
Node 30 is SOLVED



Source	Case	Pressure	Axial	In-Plane	Out-of Plane	Torsion
FEA	Peak	0.00	20.24	4.06	12.91	2.79
WRC 107	Peak		23.21	4.11	13.08	1.19
WRC 297	Peak		34.83	7.33	19.35	1.22
FEA	Secondary	0.00	29.98	6.02	19.12	4.14
WRC 107	Secondary		34.38	6.10	19.38	1.76
WRC 297	Secondary		51.60	10.86	28.66	1.81
FEA	Primary	0.00	7.67	2.93	4.76	3.33
WRC 107	Primary		8.15	2.64	4.41	1.76
WRC 297	Primary		10.20	2.94	2.42	1.81

Extra Straight Element Data

Node 60

Type

Pad <or> Saddle Thickness

Junction Crotch Radius

Outside Fitting Radius

Data For User Entered Sif's

Element Nodes 55 60

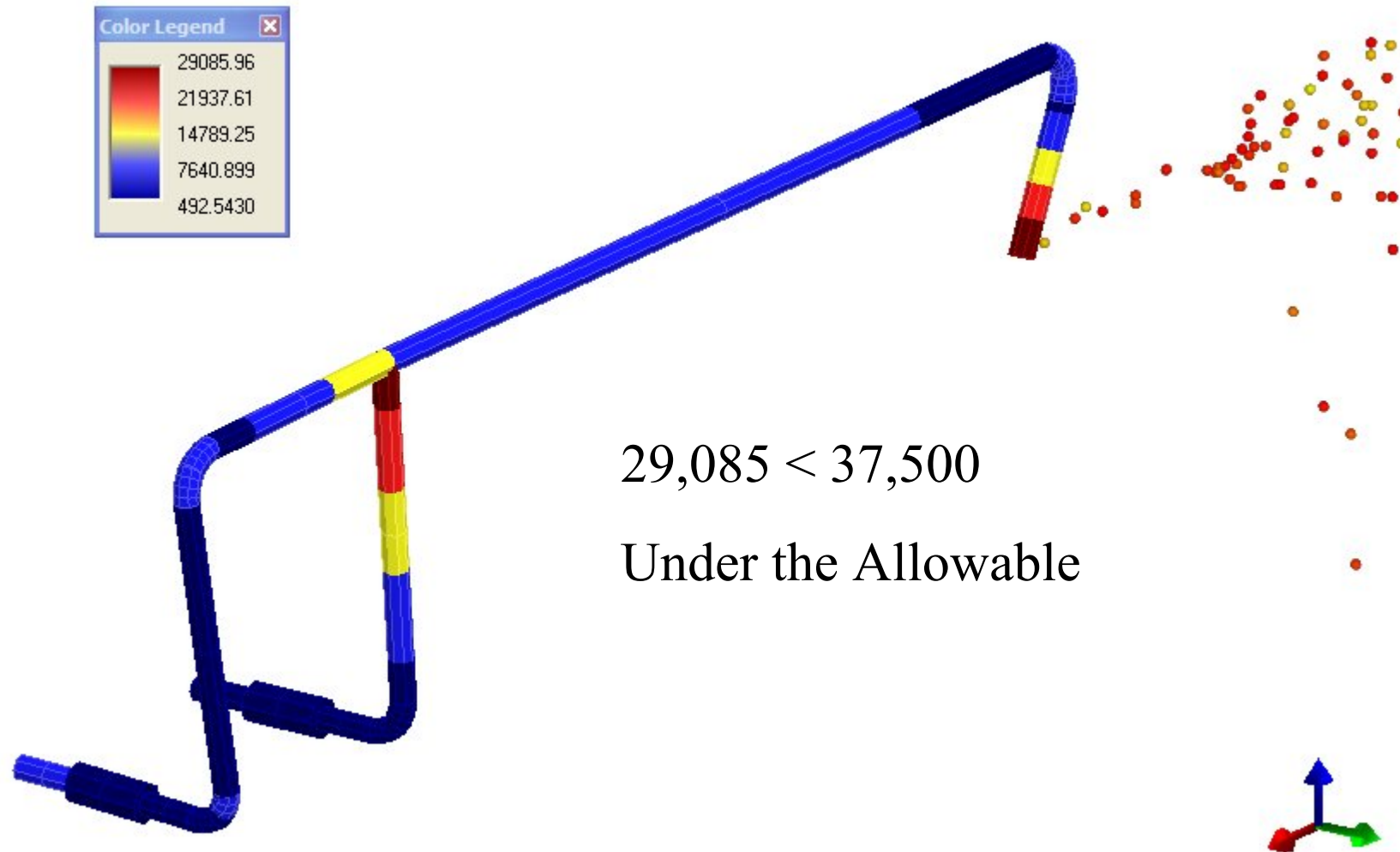
Inplane Direction Vector 0 0 1

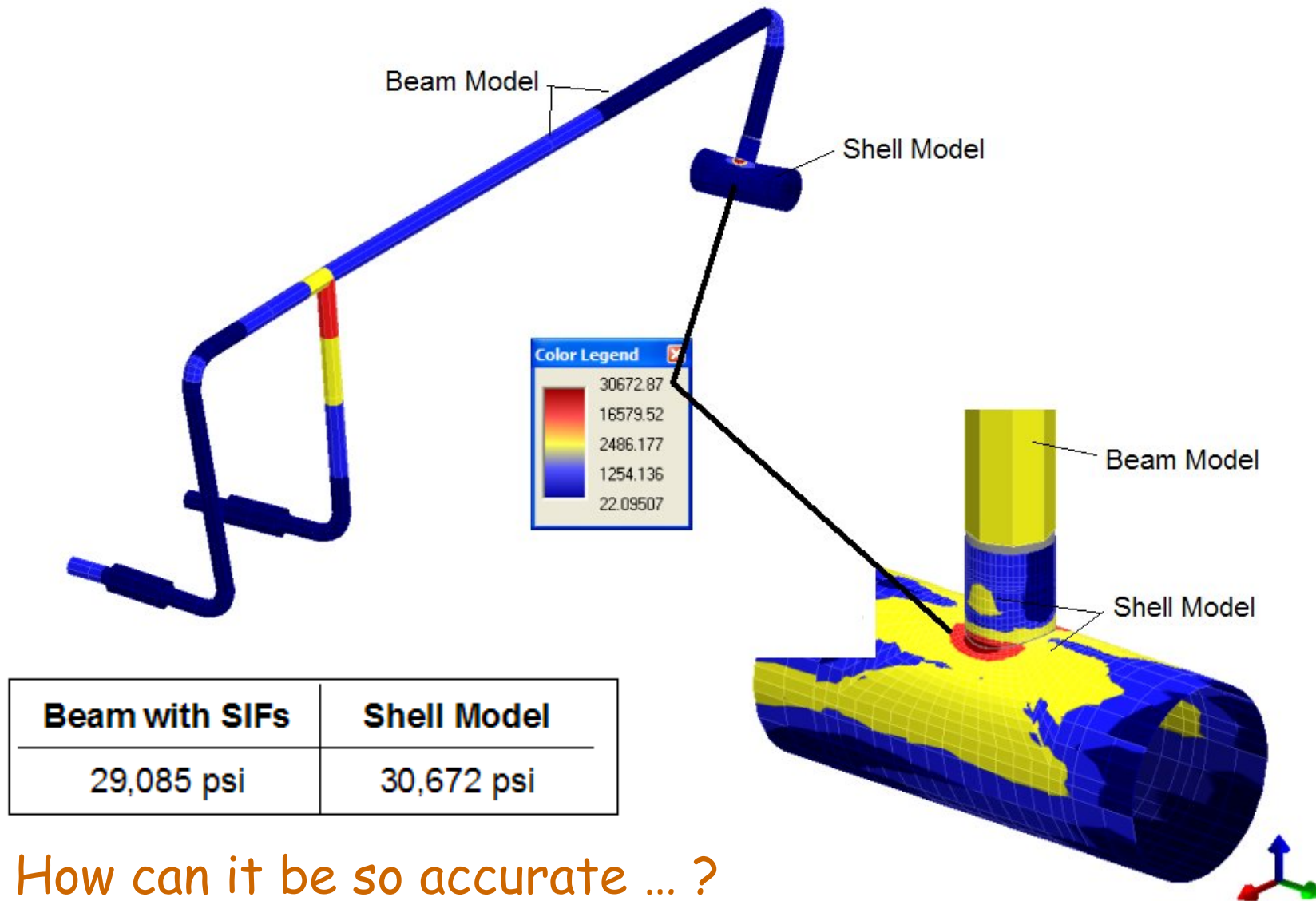
ii, io, it, iax 4.1, 12.9, 2.7, 20

i_i i_o i_t i_{ax}

Results from FESIF, NozzlePRO, FE/Pipe or other FEA tool produce
stiffnesses (flexibilities) AND SIFs. (Note FEA and 107/297 Comparison)

Stress went up at Node 60 due to SIF, but within Allowable





How can it be so accurate ... ?

(SIF is from the FEA model and problem is single directional – it should be so accurate.)

Effect of Combining Moments on Branches or Nozzles

Item	Model 1		Model2		Model3	
D	100		100		100	
d	70		70		70	
T	0.375		1.25		3	
t	1.5		1.5		3	
d/D	0.7		0.7		0.7	
D/T	266		80		33	
<u>Outplane</u>	17538	36.6%	1643	22.7%	15079	56.6%
<u>Torsional</u>	24,283	50.7%	5100	70.4%	15064	56.5%
<u>Inplane</u>	15,230	31.8%	2206	30.4%	11442	42.9%
SRSS	33,603	70.29%	5794	80%	24,191	90.8%
ABS	57,051	119%	8948	123%	41585	156%
FEA Combination	47,804	100%	7235	100%	26,617	100%

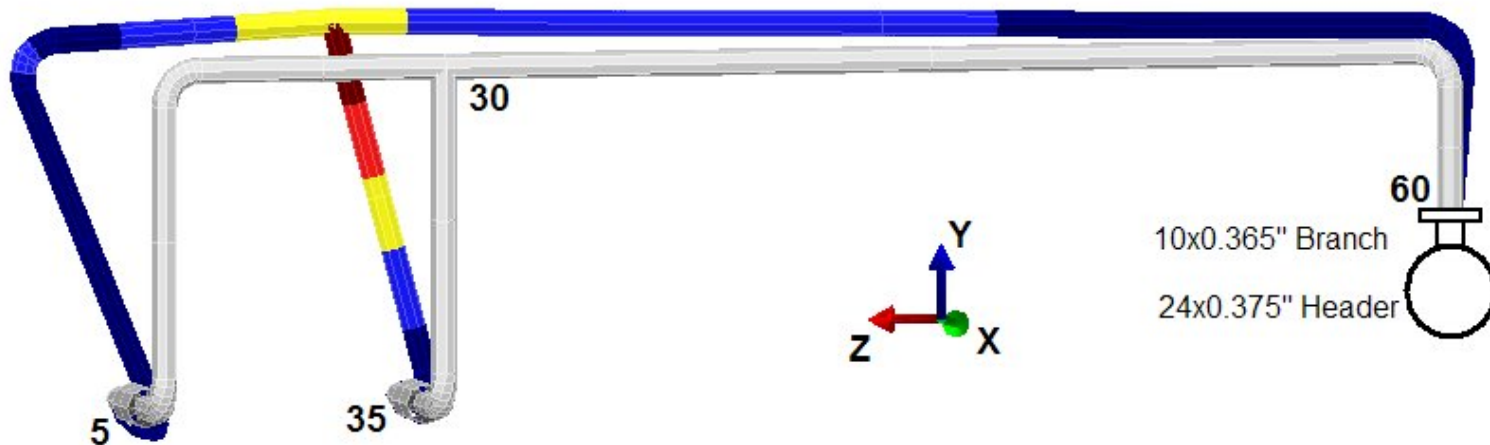
B31.3 Moment Combination on Branches:

$$S_b = \{ [(i_i M_i)^2 + (i_o M_o)^2]/Z_e + (M_t/Z)^2 \}^{1/2}$$

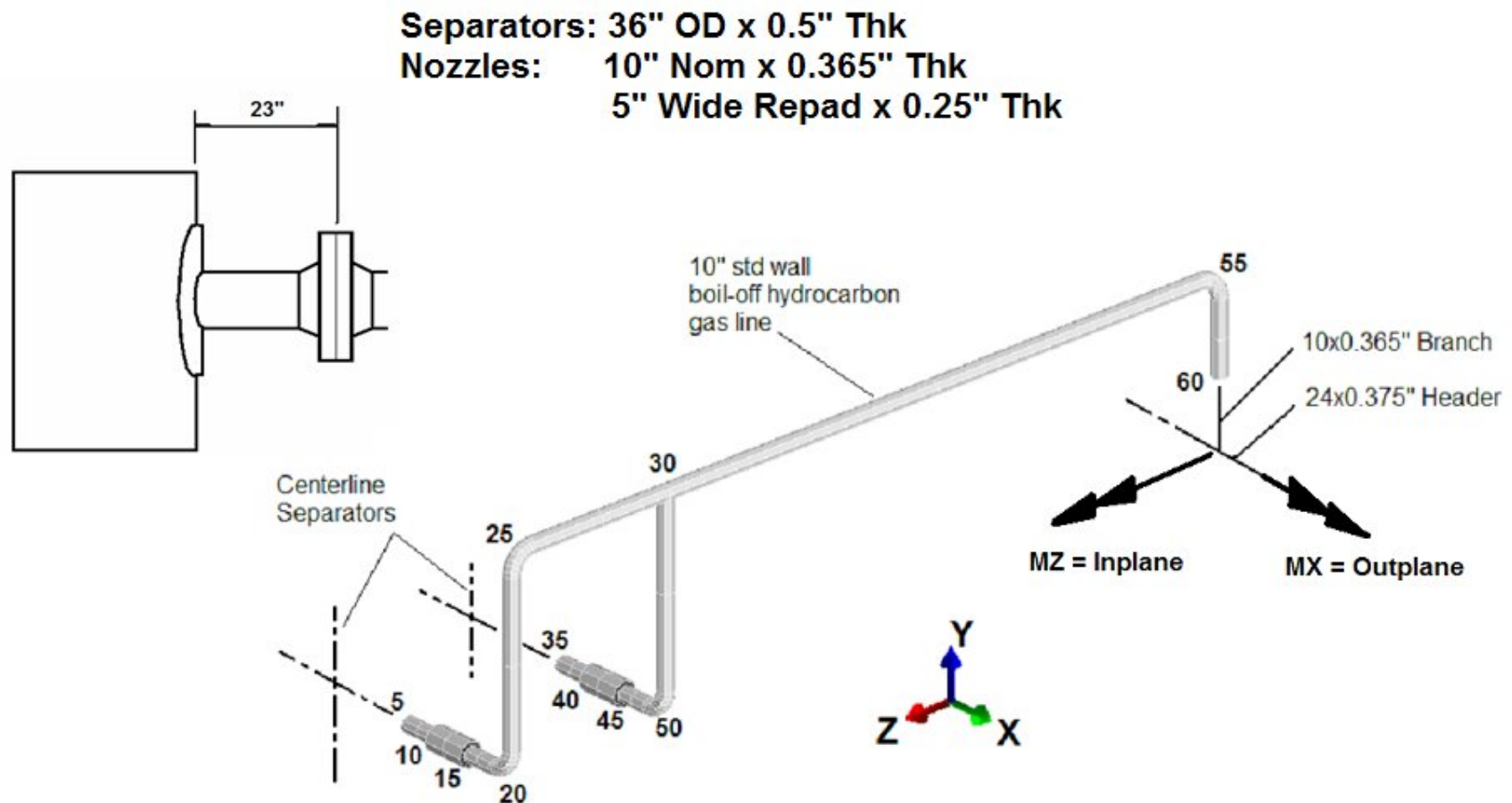
B31.3 Current Method for Combining Loads

With a significant number of cycles (32,768) – the stresses at the separator (nodes 5 & 35) are subject to torsion and bending.

$$S_b = \{ [(i_i M_i)^2 + (i_o M_o)^2]/Z_e + (M_t/Z)^2 \}^{1/2}$$



Shouldn't stop when we have a “good” answer – should stop when we have the right answer – The flexibilities and stresses at the boundary conditions in this problem – clearly have an effect.



Enter SIFs and Flexibilities for BOTH Separators at 5 and 35.

Geometry Input | Loads | Optional | Materials | Analysis Results

Parent Data

Parent Type: **Cylinder**

Outside Diameter [in] D = **36**

Wall Thickness [in] T = **0.5**

Cylinder Length [in] L = **Optional**

Branch Data

Branch Type: **Pad Reinforced**

Branch OD [in] d = **10.75**

Branch Thickness [in] t = **0.365**

Pad Width [in] W = **5**

Pad Thickness [in] Tp = **0.25**

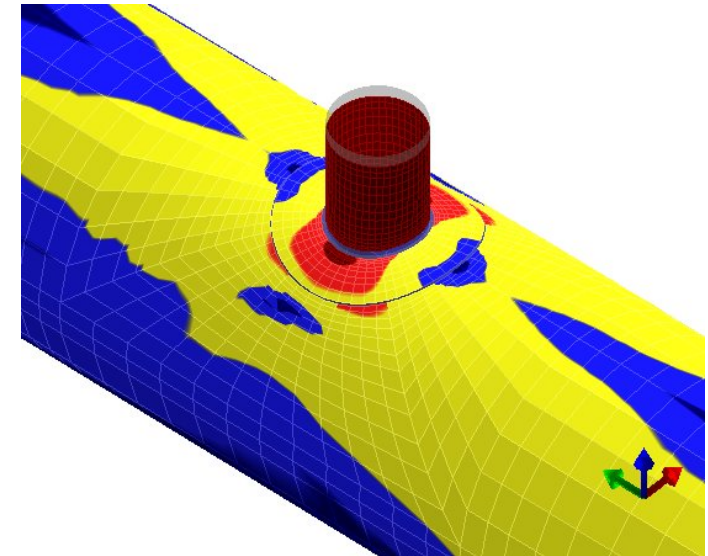
Branch Length [in] Ln = **13**

Nozzle Location [in] L1 = **Optional**

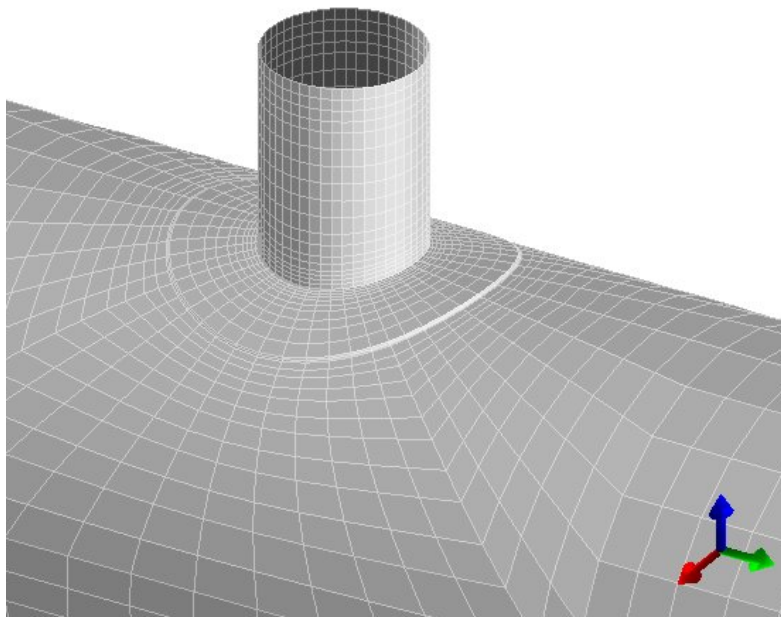
Lateral or Hillside? **None**

Fillet Size at Nozzle [in] = **0.25**

Fillet Size at Pad Edge [in] = **0.25**



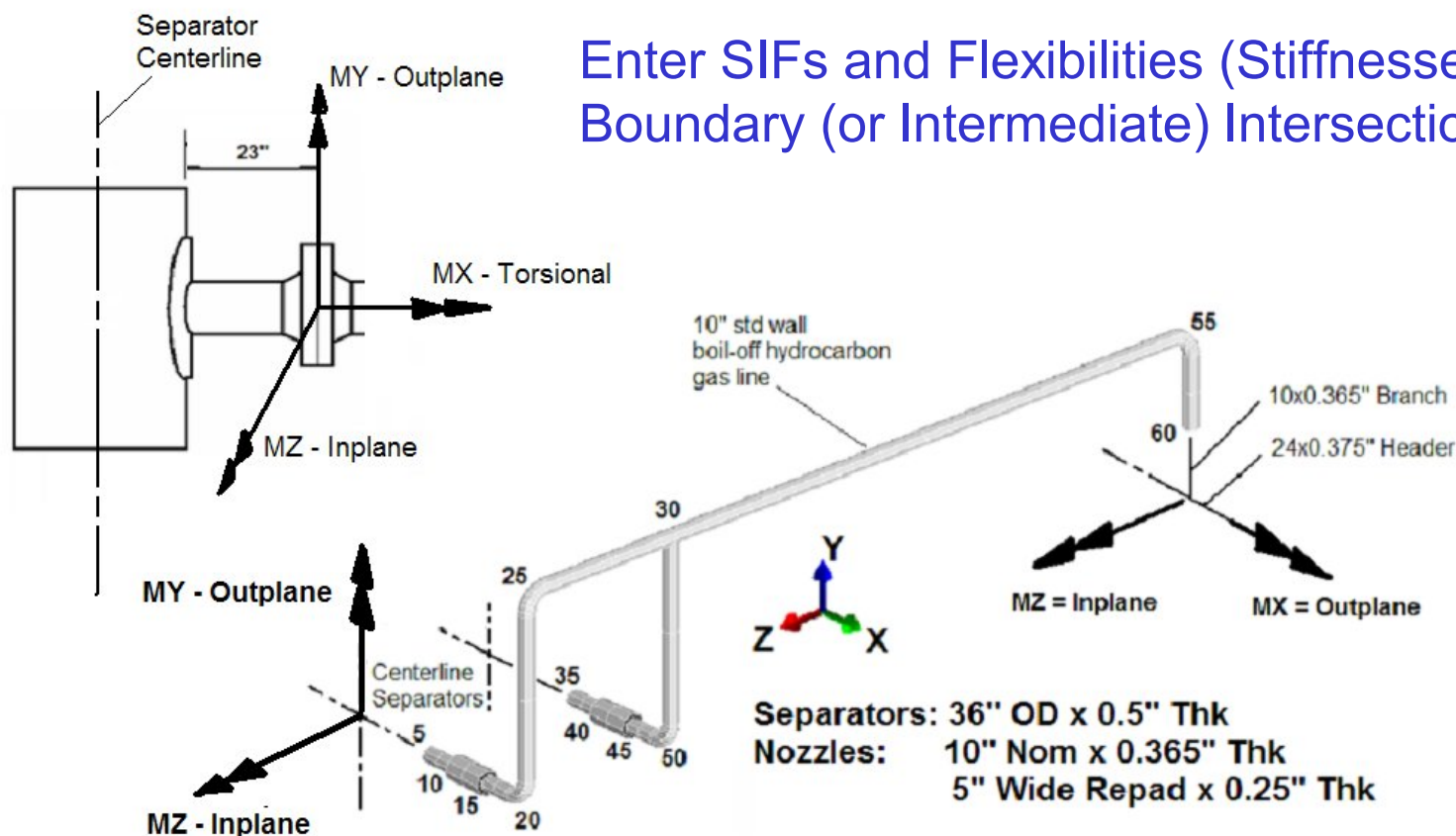
FEA Stress	Allowables	Flexibilities	FEA vs. WRC
Axial Stiffness [lb/in.]			332994
In-Plane Bending Stiffness [in.lb./deg]			1.52E+06
Out-of-Plane Bending Stiffness [in.lb./deg]			316086
Torsional Stiffness [in.lb./deg]			3.52E+08



Source	Case	Pressure	Axial	In-Plane	Out-of-Plane	Torsion
FEA	Peak	0.00	18.25	3.32	8.08	1.49
WRC 107	Peak		12.72	2.15	3.96	0.59
WRC 297	Peak		14.26	6.46	11.28	1.22

t/T ratio makes 107 suspect and 297 overconservative!
(especially with the pad influence on t/T effect)

Enter SIFs and Flexibilities (Stiffnesses) at Boundary (or Intermediate) Intersections of Interest



Node List	
5 35	

Restraint Data	
Directions (see ?-help)	Anchor
Base Node (Opt.)	
Stiffnesses lb. <or> in. lb. per deg.	rigid, rigid, rigid, rigid, 316086, 1.52e6
	X Y Z RX RY RZ

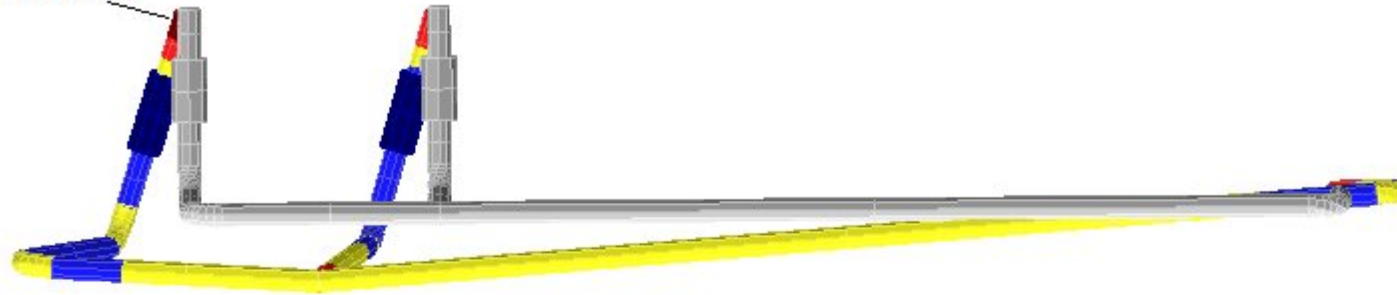
Typical Flexibility (stiffness) Input

Extra Straight Element Data	
Node	5
Type	
Pad <or> Saddle Thickness	
Junction Crotch Radius	
Outside Fitting Radius	

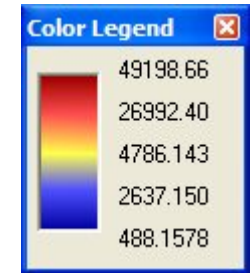
Data For User Entered Sif's	
Element Nodes	5 10
Inplane Direction Vector	0 0 1
ii, io, it, iax	3.32, 8.1, 1.5, 18

Typical SIF Input

High Stress



Top View



Side View

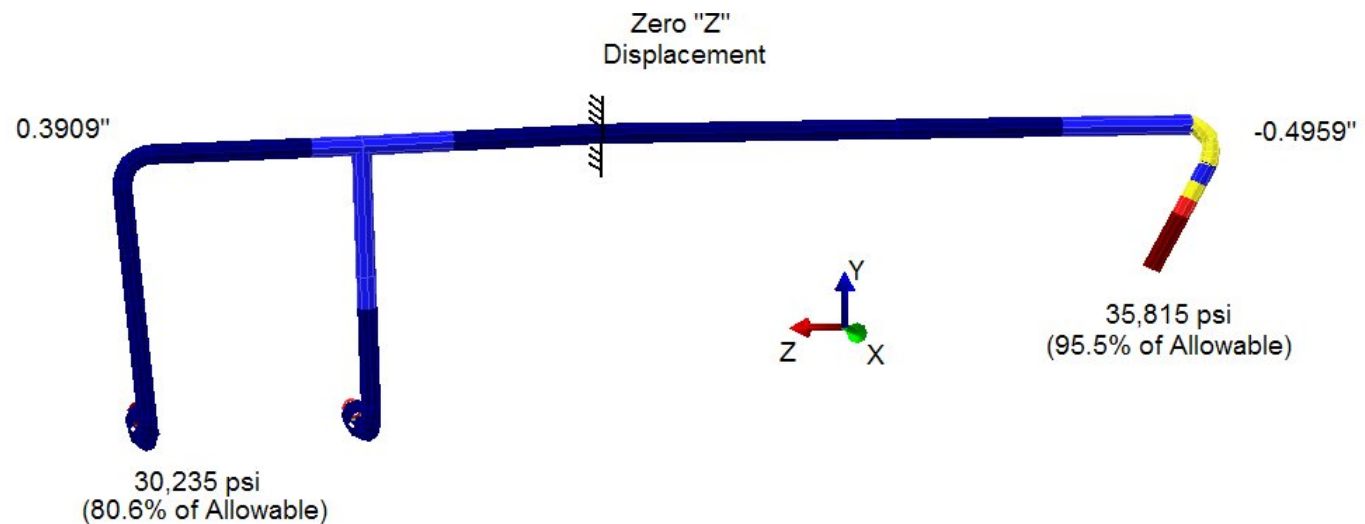
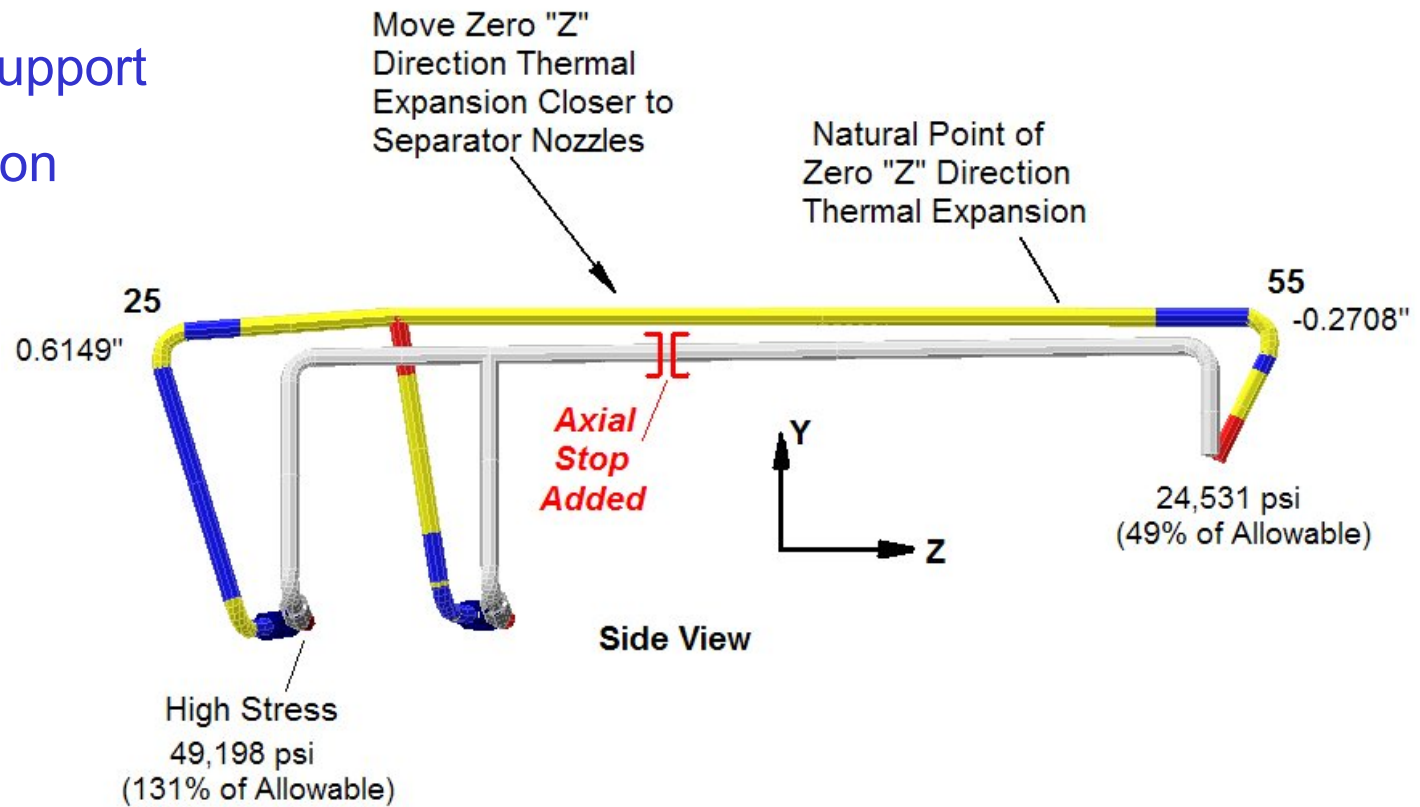
High Stress

$$49,198 > 37,500$$

Now we're overstressed

We believe we have a good SIF/Flexibility solution. Now we have to check the loads, recheck the model, and perhaps reroute the pipe or change the supporting if there's no other mitigating effect? (*Will there really be 32,000+ cycles?*)

ADD-A-Support Solution



Inspection Solution

FSRF's (or SCF's) based on examination

Table 5.12 – Weld Surface Fatigue-Strength-Reduction Factors

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MTPT examination and a VT examination.
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MTPT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MTPT examination and VT examination
1.6	2	As-welded weld that receives a partial volumetric examination, and a surface that receives MTPT and VT examination
1.5	3	Machined or ground weld surface that receives MTPT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
1.7	3	As-welded or ground weld surface that receives MTPT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
2.0	4	Weld has received a partial or full volumetric examination, and the surface has received VT examination, but no MTPT examination
2.5	5	VT examination only of the surface; no volumetric examination nor MTPT examination.
3.0	6	Volumetric examination only
4.0	7	Weld backsides that are non-definable and/or receive no examination.

Notes:

1. Volumetric examination is RT or UT in accordance with Part 7.
2. MTPT examination is magnetic particle or liquid penetrant examination in accordance with Part 7
3. VT examination is visual examination in accordance with Part 7.
4. See WRC Bulletin 432 for further information.

Approach: Actual Flexibilities and SIFs First
Reroute (or other change to the system) Next

Geometry Input | **Loads** | Optional | Materials | Analysis Results

Parent Data

Parent Type: Cylinder

Outside Diameter [in] D = 24

Wall Thickness [in] T = 0.375

Cylinder Length [in] L = Optional

Branch Data

Branch Type: Straight Nozzle

Branch OD [in] d = 10.75

Branch Thickness [in] t = 0.365

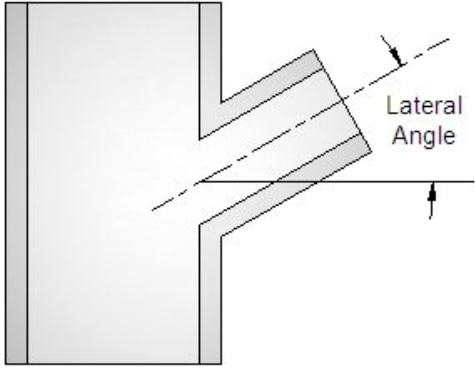
Branch Length [in] Ln = Optional

Nozzle Location [in] L1 = Optional

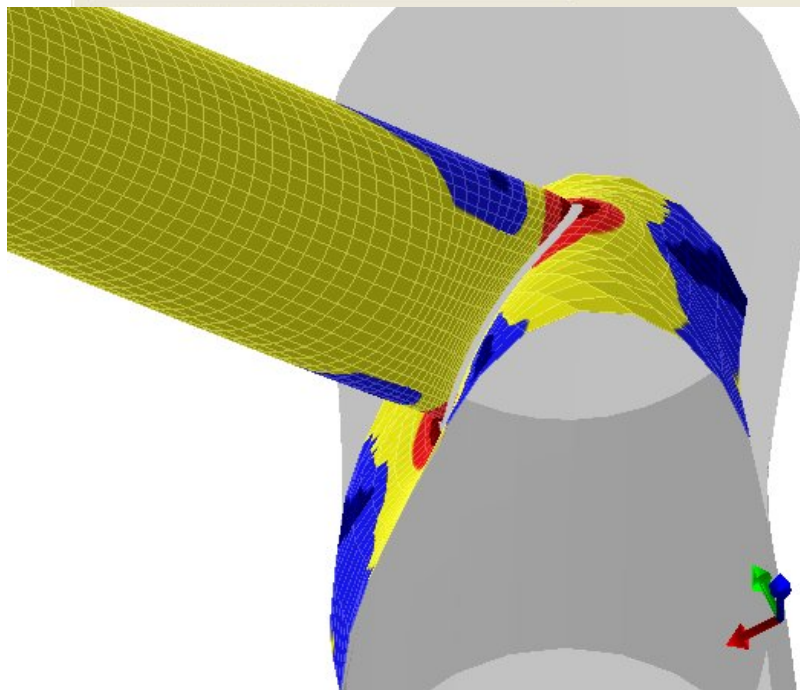
Lateral or Hillside? Lateral

Lateral Angle [deg] A = 45

Fillet Size at Nozzle [in] = Optional



Laterals



Lateral

Source	Case	Axial	In-Plane	Out-of Plane	Torsion
FEA	Peak	12.65	3.73	8.43	8.47
WRC 107	Peak	23.21	4.11	13.08	1.19
WRC 297	Peak	34.83	7.33	19.35	1.22

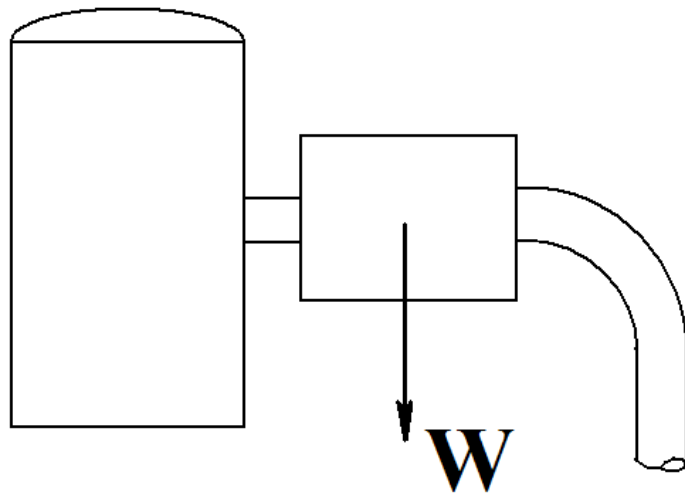
90° Branch (Not Lateral)

Source	Case	Axial	In-Plane	Out-of Plane	Torsion
FEA	Peak	20.24	4.06	12.91	1.40
WRC 107	Peak	23.21	4.11	13.08	1.19
WRC 297	Peak	34.83	7.33	19.35	1.22

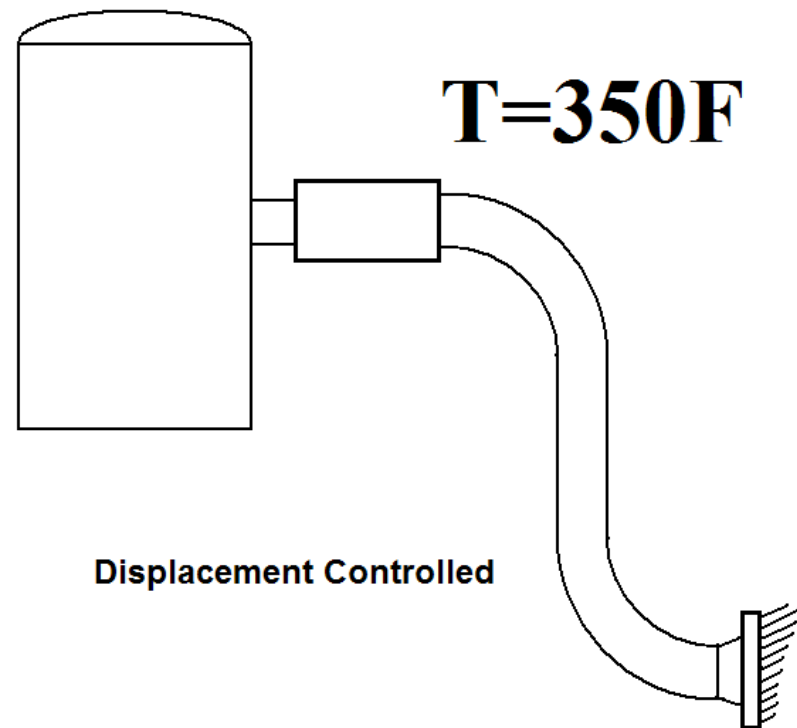
Looking Ahead ... Part 2:

Load controlled – primary allowables

Displacement controlled – secondary and fatigue allowables

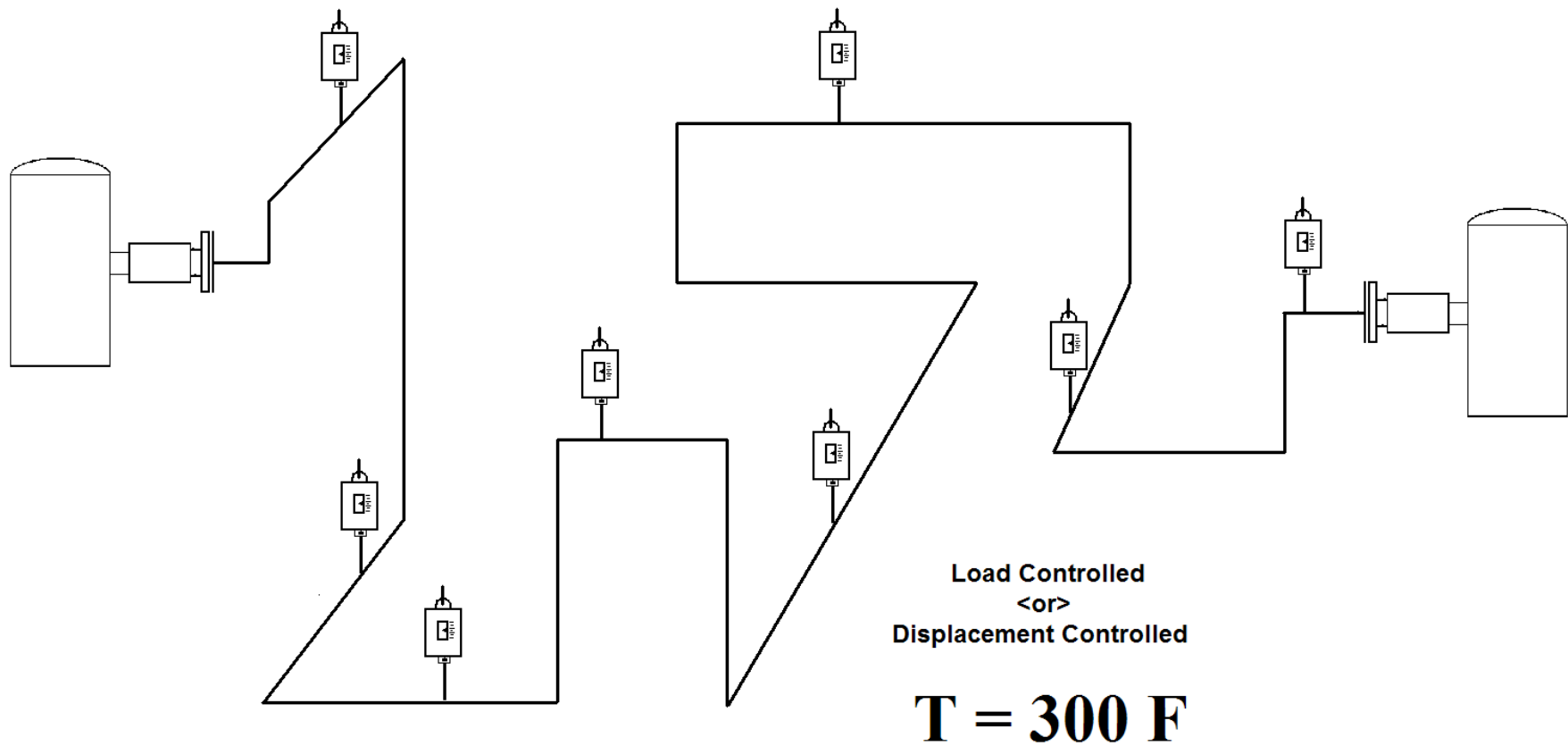


Load Controlled



Displacement Controlled

Is this system displacement controlled or load controlled ?



Part 2 – Tuesday April 15

- 1) Discuss Pressure stresses and interaction with external loads.
- 2) Discuss volumetric fea – look at examples and comparisons with shells.
- 3) Take a closer look at load combinations.
- 4) *Load controlled or displacement controlled?*
- 5) Discuss through-wall thermal effects
- 6) Discuss load combinations, rainflow cycle counting and specific application of 2007 Div 2. (AISC load combinations)
- 7) Plastic analysis.

Conclusions:

- 1) Evaluate the effects of assumptions. Welded geometries may be problematic. *(See Woods, Rodabaugh WRC “Fatigue Life of Integrally Reinforced Weld On Fittings” i.e. tees.)*
- 2) Don't believe your calculation is accurate. Hope it is close. *(there are exceptions – Nuclear for example.)*
- 3) Be very comfortable with the calculation and use of SIFs and flexibilities. *(The rigid and pinned approaches require little expertise to envelope the solution.)*
- 4) Inspection may be used to improve fatigue life.
- 5) Be careful with singularities in volumetric FEA solutions.
- 6) Watch Part 2 next week.

Remaining April PRG Webinar Schedule

Tuesday April 15 - External Loads on Nozzles and Branch Pipe Intersections (Part 2)

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” (Part 1)

Email web043@paulin.com to ask questions, or to receive a copy of this presentation and a 1 hour PDH certificate. Your email address must match your webinar registration email.

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