- *The ability to evaluate the safety of welding tees with varying crotch thicknesses.*
- *A method to know when thick walled intersections will experience high stress cracking due to thermal transients.*
- *A method to know when beam-type piping programs will produce questionable results.*
- *An understanding of Piping/Vessel Heat Transfer Analysis Basics.*
- *The ability to turn temperature gradients into stresses, and compare them to the Code allowables.*
- *An understanding of the relationship between the ASME and B31 piping codes. (Makes it much easier to render interpretations to the Codes intent when you understand how a problem is looked at in several ways.)*

No prior finite element experience is required to attend. **Fe/Pipe** and **ANSYS** are used in the course. There is no "matrix algebra", college physics, or theoretical derivations presented. All explanations are kept practical, to-the-point, and useful. The course instructor is Tony Paulin, original author of the **CAESAR II** pipe stress program, co-author of **Fe/Pipe** and lecturer on pipe stress methods to over 1000 engineers and designers around the world. Optional night sessions continue the computer training and example presentations.

The finite element seminar schedule is as follows:

May 13-15	Frankfurt, Germany
June 17-19	San Francisco, California
June 23-25	Houston, Texas

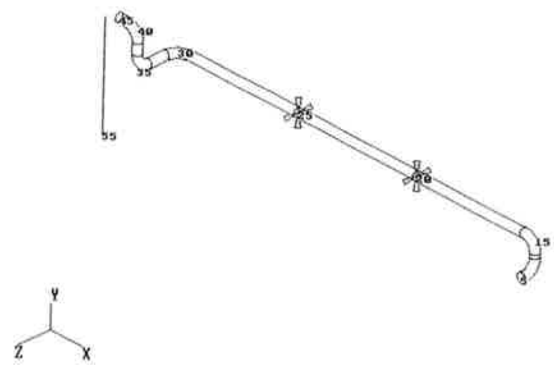
In Germany the contact for arrangements is W.Fuchs 06172-34424 (Tel) 06172-303861 (Fax), In San Francisco the contact is Synergy Engineering, Inc. 408-253-1466 (Tel), 408-253-0544 (Fax), and in Houston, the contact is COADE Research Services, Inc. 713-251-8084 (Tel), 713-251-1830 (Fax).

Fe/Pipe - CAESAR II TRANSFER LINE STUDY

A 14 inch transfer line from a furnace to a tower has been analyzed on **CAESAR II**.

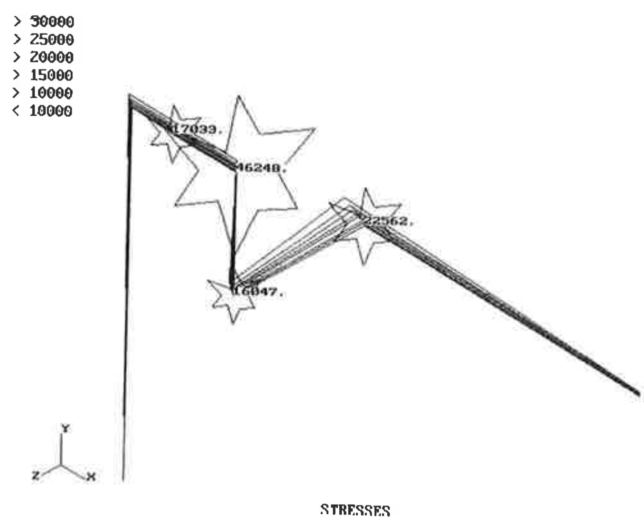
The input for the line configuration is shown below.

Temperature	= 550 deg. F
Pressure	= 150 psi
Sc	= 17,000 psi
Sh	= 15,500 psi
Vessel OD	= 48.0 in.
Vessel Thickness	= 0.5 in.
Pad Thickness	= 0.5 in.
Pad Width	= 6.0 in.



CAESAR II INPUT PLOT

From the output plot shown below it can be seen that the elbow adjacent to the intersection is overstressed. The **CAESAR II** calculated expansion stress is 45,858 psi, and the allowable 38,962 psi.



CAESAR II OUTPUT PLOT

The **CAESAR II** operating restraint loads, and expansion forces, moments and stresses are printed in the following reports.

CAESAR II RESTRAINT REPORT FILE:TRNFER
CASE 3 (OPE) W+T1+P1+FOR DATE:FEB 23,1992

NODE	FX	FY	FZ	MX	MY	MZ	TYPE
	—Forces(lb.)—			—Moments(ft.lb.)—			
5	21642.	-2384.	783.	1572.	-26004.	-56041.	Rigid ANC
20	0.	0.	0.	0.	0.	0.	Rigid +Y
20	0.	0.	1331.	0.	0.	0.	Rigid Z
25	0.	-2954.	0.	0.	0.	0.	Rigid +Y
25	0.	0.	-7822.	0.	0.	0.	Rigid Z
46	21642.	-1705.	-5708.	22068.	-44844.	56348.	Rigid ANC
55	-21642.	1705.	5708.	63557.	33428.	271695.	Rigid ANC
35	0.	-700.	0.	0.	0.	0.	Prog Design VSH

CAESAR II FORCE/STRESS REPORT FILE:TRNFER
CASE 5 (EXP) D5(EXP)=D3-D4 DATE:FEB 23,1992

DATA POINT	FX	FY	FZ	MX	MY	MZ	SIFI	SIFO	CODE	ALLOW.
	—Forces(lb.)—			—Moments(ft.lb.)—				(lb./sq.in.)		
5	-21558	1607	-729	-3334	26155	56150	1.00	1.00	13978	38941
10	21558	-1607	729	5273	16962	-18422	2.80	2.33	10927	39064
10	-21558	1607	-729	-5273	-16962	18422	1.00	1.00	5766	39258
15	21558	-1607	729	3633	15686	27271	2.80	2.33	19090	39092
15	-21558	1607	-729	-3633	-15686	-27271	1.00	1.00	7136	39248
20	21558	-1607	729	3633	3838	1145	1.00	1.00	1218	38984
20	-21558	593	-2193	-3633	-3838	-1145	1.00	1.00	1218	38984
25	21558	-593	2193	3633	-35651	-9536	1.00	1.00	8356	38381
25	-21558	2023	5758	-3633	35651	9536	1.00	1.00	8356	38381
30	21558	-2023	-5758	91	30270	-45963	2.80	2.33	21719	39167
30	-21558	2023	5758	-91	-30270	45963	1.00	1.00	12402	39273
35	21558	-2023	-5758	1568	-61354	-8235	2.80	2.33	14521	38052
35	-21558	1967	5758	-1568	61354	8235	1.00	1.00	13954	38895
40	21558	-1967	-5758	20282	-51277	58387	2.80	2.33	45858	38962
40	-21558	1967	5758	-20282	51277	-58387	1.00	1.00	18097	39180
45	21558	-1967	-5758	20282	-44079	55927	1.00	1.00	16685	39145

The stress report for the node 45, at the transfer line connection to the tower shows a stress of 16,685 psi. But we notice that the stress intensification factor (SIF) used for this connection is 1.0. However, we know that the SIF is almost never 1.0. With an allowable of 39,145 psi, a SIF of 39,145/16,685 = 2.34 would put the nozzle right at the allowable stress. (And a SIF of 2.34 does not seem that unreasonable.) We also note that the axial forces in the system as a whole are high. (We know that the B31 Code equations do not consider axial forces in their expansion stress calculations, and we want to make sure that we don't miss anything here because of the stress due to this high axial load.) The connection at the vessel is definitely thin walled, (D/T = 48/0.5 = 96), so the standard Code equations are probably not going to give us much guidance at the tower junction.

This is a perfect application for **Fe/Pipe**. The input for the problem takes about (2) minutes. The problem ran in 43 minutes on a 33 Mhz 486. (The stiffness matrix was saved so that later runs will only take about 20 minutes.) The output that we are primarily interested in from the first run of **Fe/Pipe** is shown below:

```
TRNFER                     Fe/Pipe  Version 2.0
FEB 29,1992                COADE RESEARCH SERVICES
6:17pm                     Released Jan. 15, 1992

Computed Stress Intensification Factors

Pad/Header at Junction

Peak Stress Sif ....     3.679 (Inner) Axial
                         3.937 (Outer) Axial
                         2.064 (Inner) Inplane
                         1.414 (Outer) Inplane
                         2.635 (Inner) Outplane
                         2.386 (Outer) Outplane
                          .682 (Inner) Torsional
                          .454 (Outer) Torsional

Branch at Junction

Peak Stress Sif ....     8.334 (Inner) Axial
                         9.328 (Outer) Axial
                         2.399 (Inner) Inplane
                         2.582 (Outer) Inplane
                         4.698 (Inner) Outplane
                         5.690 (Outer) Outplane
                         1.082 (Inner) Torsional
                         1.118 (Outer) Torsional

Pad Outer Edge Weld

Peak Stress Sif ....     3.369 (Inner) Axial
                         2.507 (Outer) Axial
                         1.341 (Inner) Inplane
                          .639 (Outer) Inplane
                         2.275 (Inner) Outplane
                         1.947 (Outer) Outplane
                          .365 (Inner) Torsional
                          .471 (Outer) Torsional

B31.3
Peak Stress Sif ....      .000         Axial
                         3.566         Inplane
                         4.504         Outplane
                         1.000         Torsional
B31.1
Peak Stress Sif ....      .000         Axial
                         4.504         Inplane
                         4.504         Outplane
                         4.504         Torsional

Flexibilities

 The following stiffnesses should be used in a piping,
 "beam-type" analysis of the intersection.  The stiff-
 nesses should be inserted at the surface of the
 branch/header or nozzle/vessel junction.  The general
 characteristics used for the branch pipe should be:

 Outside Diameter =   14.000  in.
 Wall Thickness   =     .375  in.

 Axial Transverse Stiffness       =       701864.  lb./in.
 Inplane Rotational Stiffness     =      2111914.  in.lb./deg
 Outplane Rotational Stiffness    =       572928.  in.lb./deg
 Torsional Rotational Stiffness   =     60526980.  in.lb./deg

 FLEXIBILITY LOAD REDUCTION:

 The precentages given below show how much the loads would
 be reduced if a flexible model of the intersection was
 included in the piping analysis. This calculation is only
 valid when strain limited loads, i.e. thermal, make up the
 majority of the operating load, and when a rigid model of
 the junction was used to compute the original loads entered
 into Fe/Pipe.

 Axial Load Reduction              =   0. %
 Inplane Moment Load Reduction     =  18. %
 Outplane Moment Load Reduction    =  45. %
 Torsion Moment Load Reduction     =   0. %
```

The only input required to generate the above listing was the basic geometry of the intersection. From the above reports we can draw the following conclusions about the intersection.

The highest computed SIF for the intersection was 5.69 for an out-of-plane moment, and 2.58 for an inplane moment. From the **CAESAR II** force/moment report above we can see that the outplane moment is 44,079 ft.lb, and the inplane moment is 55,927 ft.lb. Using the **Fe/Pipe** computed SIF's and **CAESAR II** loads the stress at the intersection would be: (the 12 is to get from ft.lb. to in.lb.)

Stress $= (12/Z) ([M_i*i_i]^2 + [M_o*i_o]^2)^{1/2}$
$=(12/53.25) ([55927*2.58]^2 + [44079*5.69]^2)^{1/2}$
$= 65,206$ psi

The allowable is 39,145 psi. We also know that there will be some contribution to the stress due to the axial load that we have not accounted for yet. The axial load on the nozzle shows to be 21,558 lb. From the **Fe/Pipe** SIF reports above we can see that the SIF for this load is 9.328. It is not unusual for axial SIF's to be this high. (Fortunately most axial loads in piping systems are small.) **This axial load would, however, be completely ignored if a strictly code calculation was performed.**

The load printed from **CAESAR II** in the element "force/moment" report is a "structural" type of load on the nozzle. It **does not** include the axial pressure forces that exist in the system. (Pressurize a straight pipe in **CAESAR II** and see that no forces show up in the force/moment report.) Since the large axial load shown in the **CAESAR II** reports is compressive, the (P times A) pressure load will counteract this force. The net external load on the nozzle will probably be somewhere around:

Net Axial Load $= 21558 - (P*A)$
$= 21558 - (150*137.9)$
$= 873$ lb.

As long as pressure acts with temperature the axial load on the nozzle will be approximately balanced. If the system ever comes to temperature at a lower pressure, the unbalanced load of 21,558 lb. will act on the nozzle junction. To check the stress that would result from this unbalanced axial load, divide the axial load by the cross-sectional area of the pipe to get the nominal axial stress, and then multiply by the **Fe/Pipe** SIF:

Stress $= F/A * i$
$= 21558/16.05 * 9.328$
$= 12,529$ psi

To follow the Codes simplification for computing the maximum shear stress intensity, the axial stress would be added to the bending stress:

Total Stress = Axial Stress + Bending/Torsion Stress
$= 12,529 + 65,206$
$= 77,735$ psi.

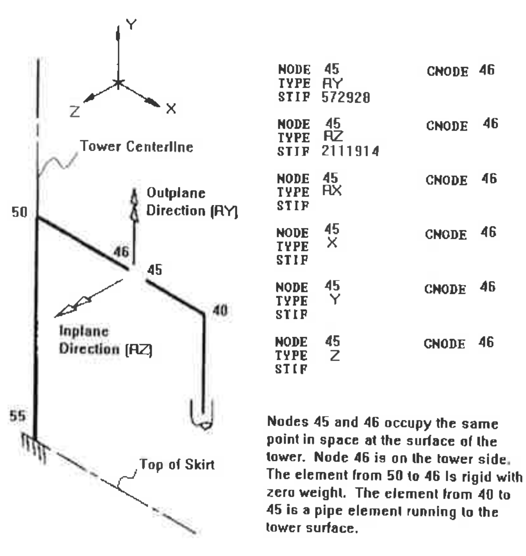
In any event, the moment loads on the nozzle must be reduced because the nozzle is overstressed, and the loads in the piping system must be reduced because the elbow is overstressed.

The load reduction report printed from **Fe/Pipe** (the last few lines in the previous listing) gives us an idea of the magnitude of the load reduction on the intersection we can expect if the local stiffnesses of the vessel are inserted back into our piping analysis. According to this report the out-of-plane moment will be reduced by 45%, and the inplane moment will be reduced by 18%. These percentage reductions fit directly into the bending stress equation above as shown below:

Stress $= (12/Z) ([M_i*i_i*\%_i]^2 + [M_o*i_o*\%_o]^2)^{1/2}$
$= (12/53.25) ([55927*2.58*0.82]^2 + [44079*5.69*0.55]^2)^{1/2}$
$= 40,954$ psi

This would put the calculated stress in the junction right at the allowable of 39,145 psi. The inclusion of these flexibilities in the piping system will also redistribute other loads in the vicinity of the intersection and its adjacent supports. This almost certainly includes the first elbow that is overstressed. Unless it is a simple matter to reroute the pipe, including the **Fe/Pipe** local stiffness of the vessel is the most practical next step to take. Because the axial and torsional stiffnesses of the vessel shell will have no effect on the load reduction (see the **Fe/Pipe** Load Reduction Report) they will not be put back into **CAESAR II**.

A sketch of the portion of the **CAESAR II** model including the **Fe/Pipe** local stiffnesses is shown below. Note that all six degrees of freedom for the nozzle connection must be defined. There are the two flexible directions for inplane and outplane rotational restraint, RZ and RY respectively, and four rigid supports for the other degrees of freedom: RX, X, Y and Z. The local stiffness values: 572,928 in.lb./deg and 2,111,914 in.lb./deg. come directly from the **Fe/Pipe** flexibility report shown at the front of this article. Note how the RY direction corresponds to the "**outplane**" axis on the vessel, and the RZ direction corresponds to the "**inplane**" axis on the vessel.



LOCAL STIFFNESSES IN CAESAR II

At this point the **CAESAR II** job would be rerun and the new loads and stresses in the piping system computed. Output from this run is shown below:

```
CAESAR II    RESTRAINT REPORT     FILE:TRNFR2
CASE 3 (OPE) W+T1+P1+FOR          DATE:FEB 25,1992
```

NODE	FX	FY	FZ	MX	MY	MZ	TYPE
	—Forces(lb.)—			—Moments(ft.lb.)—			
5	18581.	-2113.	574.	645.	-22228.	-48246.	Rigid ANC
20	0.	0.	0.	0.	0.	0.	Rigid +Y
20	0.	0.	1908.	0.	0.	0.	Rigid Z
25	0.	-3406.	0.	0.	0.	0.	Rigid +Y
25	0.	0.	-8558.	0.	0.	0.	Rigid Z
46	18581.	0.	0.	0.	0.	0.	Rigid X
46	0.	0.	0.	0.	0.	42455.	Flex RZ
46	0.	0.	0.	0.	-23981.	0.	Flex RY
46	0.	0.	0.	22183.	0.	0.	Rigid RX
55	-18581.	1918.	6076.	68950.	11830.	240093.	Rigid ANC
46	0.	-1918.	0.	0.	0.	0.	Rigid Y
46	0.	0.	-6076.	0.	0.	0.	Rigid Z
35	0.	-732.	0.	0.	0.	0.	Prog Design VSH

```
CAESAR II  FORCE/STRESS REPORT  FILE:TRNFR2
CASE 5 (EXP) D5(EXP)=D3-D4      DATE:FEB 25,1992
```

DATA POINT	FX	FY	FZ	MX	MY	MZ	SIFI	SIFO	CODE	ALLOW.
	—Forces(lb.)—			—Moments(ft.lb.)—					(lb./sq.in.)	
5	-18538	1340	-523	-2430	22434	48459	1.00	1.00	12046	38934
10	18538	-1340	523	4195	14642	-16017	2.80	2.33	9420	39041
10	-18538	1340	-523	-4195	-14642	16017	1.00	1.00	4980	39249
15	18538	-1340	523	3017	13726	23348	2.80	2.33	16411	39064
15	-18538	1340	-523	-3017	-13726	-23348	1.00	1.00	6141	39237
20	18538	-1340	523	3017	5223	1569	1.00	1.00	1404	38982
20	-18538	325	-2555	-3017	-5223	-1569	1.00	1.00	1404	38982
25	18538	-325	2555	3017	-40767	-4284	1.00	1.00	9262	38369
25	-18538	2205	6116	-3017	40767	4284	1.00	1.00	9262	38369
30	18538	-2205	-6116	-842	36880	-43984	2.80	2.33	25286	39250
30	-18538	2205	6116	842	-36880	43984	1.00	1.00	12936	39303
35	18538	-2205	-6116	487	-41906	-11542	2.80	2.33	11228	38051
35	-18538	2166	6116	-487	41906	11542	1.00	1.00	9795	38895
40	18538	-2166	-6116	20365	-31203	44914	2.80	2.33	33050	39143
40	-18538	2166	6116	-20365	31203	-44914	1.00	1.00	13150	39255
45	18538	-2166	-6116	20365	-23558	42205	1.00	1.00	11819	39228

From the expansion stress report for the elbow at 40, we can see that including the vessel local stiffnesses dropped the moments on the elbow, so that the stresses went from 45,858 psi, to 33,050 psi., a drop of 28%. The allowable is 39,143 psi. **The elbow is not overstressed.**

Fe/Pipe's load reduction calculations predicted 45% and 18% reductions in the outplane and inplane loads on the junction respectively. The results are summarized below:

Inplane: 42,455/56,348 = 25% reduction

Outplane: 23,981/44,844 = 46.5% reduction

Because our original reduction in stress calculation used the **Fe/Pipe** estimate of 18% instead of the 25%, we can recompute the stress calculation for the intersection and see what the actual reduction in load will do for the stresses.

$$\text{Stress} = (12/Z)\left([M_i * i_i * \%_i]^2 + [M_o * i_o * \%_o]^2 \right)^{1/2}$$
$$= (12/53.25)\left([55927*2.58*0.75]^2 + [44079*5.69*0.55]^2 \right)^{1/2}$$
$$= 39{,}510 \text{ psi}$$

This value is still just slightly over the allowable of 39,145 psi.

If this system was to undergo only a small number of total load cycles during its lifetime, (say less than 4000), then any further analysis is probably unwarranted. There is enough extra safety factor built into the B31 piping codes for systems cycling under 7000 cycles. If this system was to undergo a significant number of thermal loading cycles, or the thermal loading cycles are to be superimposed onto a high occasional loading cycle, (to compute life fractions), then a further analysis is certainly warranted.

The further analysis in this case with **Fe/Pipe** is simple. The loads from **CAESAR II** are entered back into **Fe/Pipe** and a re-analysis made using the old stiffness matrix. This run takes about 20 minutes on a 33Mhz 486. The pertinent results from this analysis are described below.

Quick Look at Fe/Pipe Results:

The B31 Expansion Stress report provides the quickest, "piping-type" summary of the results. (Vessel Engineers would probably prefer going directly to the ASME "Overstressed Areas" report.)

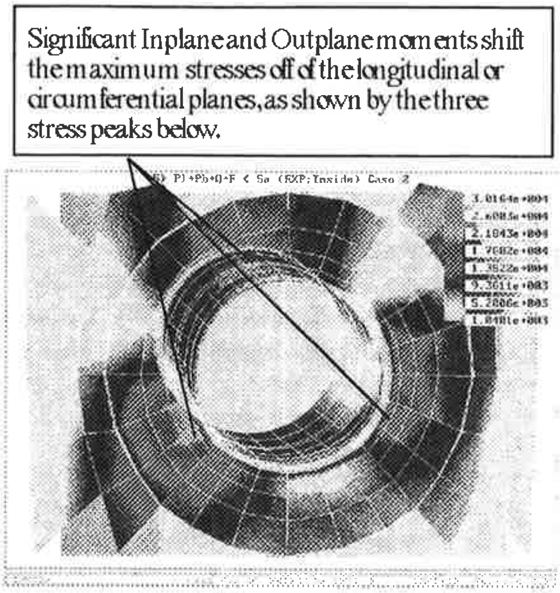
```
TRNFER
JAN  5,1991              Fe/Pipe  Version 2.0
0:02am          COADE RESEARCH SERVICES
                         Released Jan. 15, 1992

B31 Expansion Stresses
```

Expansion Stress psi	B31 Allowable psi	ASME Allowable psi	Markl Allowable psi	Regions / Notes
24383.	40625.	39828.	41701.	Pad/Header at Junction Load Case 2, Inner, Plot 5
17667.	40625.	39854.	41701.	Pad Outer Edge Weld Load Case 2, Inner, Plot 5
9462.	40625.	39854.	41701.	Header Outside Pad Area Load Case 2, Inner, Plot 5
35601.	40625.	20746.	41701.	Branch at Junction Load Case 2, Outer, Plot 6
11279.	40625.	39854.	41701.	Branch removed from Junction Load Case 2, Inner, Plot 5

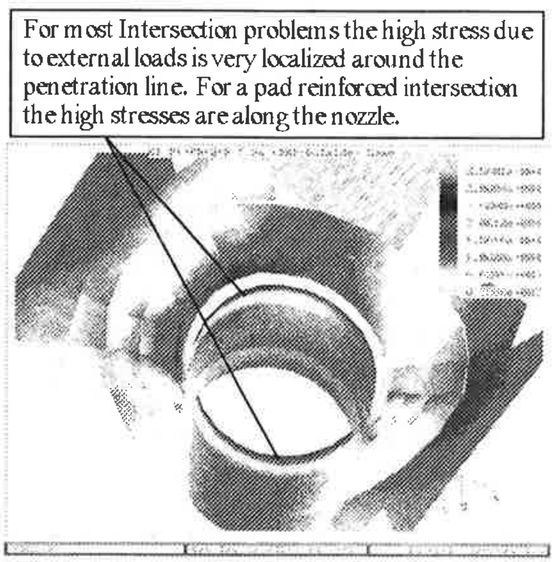
What we see from this report is a reconfirmation of the previous discussion and the hand calculations. The highest computed expansion stress in the report is 35,601 psi. This value is directly from the finite element stress calculations for the intersection. Using the SIF's from **Fe/Pipe** (shown above), and including the load reduction, the hand calculated stress was 39,510 psi. The fact that the moment acts skewed to either the inplane or outplane axis can account for this 10% difference. This particular 10% drop puts the calculated stress 10% under the allowable. This is a much more comfortable position to defend.

It is interesting to note that the ASME allowable for the "Branch at Junction" region is about half of the allowable from either B31 or Markl. This is because of an overconservatism in the ASME codes. (This overconservatism is discussed in detail below.)

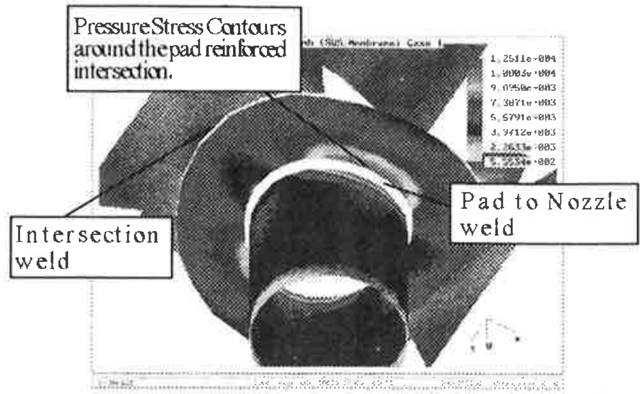
Fe/Pipe plots of the stresses for this intersection are typical of most intersection problems.



Stress Orientation Around Nozzle



Localized Nature of Stress



Pressure Stress Distribution

Detailed Look at the Fe/Pipe Report:

The fact that the ASME allowable is half of the other allowables, (as seen above), is a little bit disconcerting. This is a weakness that has been previously pointed out in the NUREG/ORNL document: *"Comparisons of ASME Code Fatigue Evaluation Methods for Nuclear Class 1 Piping with Class 2 or 3 Piping"*, by E.C. Rodabaugh. The nature of this problem will be discussed below so that **Fe/Pipe** and **CAESAR II** users will know how to apply good engineering judgement when addressing this disparity between equally safe, (although not equally conservative codes).

The ASME Section VIII, Div. 2 and ASME Section III NB-3200 Codes, (hereinafter ASME Codes), have the following basic requirements:

- Primary
- Secondary (Shakedown)
- Fatigue

The B31 piping codes have only the following basic requirements:

- Primary
- Fatigue

The shakedown allowable in the ASME Code is designed, among other things, to permit a material to undergo some amount of plastic deformation due to thermal-type loads during the initial startup loading cycles. After the first few cycles the system has "cold-sprung" itself very locally, and further cycling only produces elastic deformations, i.e. the system has "shaken-down" to purely elastic deformation. If elastic shakedown does not occur there will be some amount of plastic deformation during each load cycle. The ASME codes permit some plastic deformation to exist providing the strain concentration due to the continued plastic deformation will not cause a fatigue crack to initiate in the highly strained part of the material. When the shakedown limit is exceeded the ASME Codes require that a strain concentration factor (Ke) be calculated. The allowable fatigue stress must be divided by the strain concentration factor. (Ke is greater than 1.0.) If the fatigue stress is still less than the allowable divided by the strain concentration factor then the part is still judged to be O.K. Unfortuntately, the Ke calculation is too conservative in the situation where the notch, or peak stress effect is very small, (i.e. like on the surface of a bend). *This is a limitation of the ASME codes that should be recognized.* Fortunately this produces conservative results. Unfortunately, the results are too conservative, and the conservatism is not applied uniformly. This is the reason in the above B31 Expansion stress report, that the ASME allowable for the "Branch at Junction" stress is so low. "Ke", or the "strain concentration factor" is given in the detailed stress reports (shown on the following pages). The user can easily see when this value is greater than 1.0.

The Ke value may be too conservative when:

1) The "stress" concentration factor is 1.3 or less. This value is input by the user, and determines the amount of notch, or peak stress effect that will occur in a particular region. (The current default is 1.0, i.e. the weld is ground or otherwise dressed.) ***The current Fe/Pipe default for this value exposes the user to this possible overconservatism on the part of the ASME codes.***

2) The number of design cycles is less than about 10,000. It is only in the low cycle range that repeated plastic deformation is permitted.

3) The secondary, or "shakedown" limit is exceeded. In ASME Code terminology: when Pl+Pb+Q > 3Sm.

Fortunately this does not cause too much inconvenience. The ASME codes (with respect to the piping codes), give the designer more freedom when the number of cycles is lower than 7000, and this tends to compensate for the overconservatism, (at least it makes it not so difficult to work with). Often times in a refinery or fossil power application, the strain concentration factor may be too high, but the system will still show to be fine for 2000 cycles or so. If the actual expected number of cycles is less than 2000 then the designer knows that he is being overly conservative, and safe. For the example problem, the allowable for 7000 cycles is 20,746 psi., but the allowed number of cycles for the 35,601 psi stress is 1381 cycles. For many systems this is in excess of the actual number of cycles expected.

It is more important for the Section VIII Div.2 , or Section III NB-3200 user to know the actual number of cycles his system is to undergo. The allowables vary more significantly in the low cycle range. For B31 users the Codes don't change the allowables once the number of cycles drops below 7000. Most B31 users would consider Section VIII Div. 2, as satisfying the B31 code rules, especially if the B31 allowables are used for Sc and Sh. It is demonstrable that the B31 rules for flexibility stresses and allowables suffer a number of weaknesses that are not found in Section VIII Div. 2 Appendix 4 & 5 approaches. For this reason Section VIII Div. 2 is considered a "more rigorous analysis", as the B31 Codes set only "minimum requirements".

The following report for the transfer line example problem shows the ASME overstressed areas in the model. The first two values in this table are Pl+Pb+Q stresses. These are shakedown, or secondary stresses. Their allowable is 3Sm, which is intended to reflect (for the most part), two times the average material yield strength. The fact that the shakedown stresses exceed 3Sm, means that a strain concentration factor needs to be computed for the fatigue allowable calculation, and that there will be some plastic deformation during each cycle of the loading. **Exceeding 3Sm does not mean that a code failure has occurred.**

The Pl+Pb+Q+F stresses are the fatigue, or peak stresses. These are the stresses that are directly comparable to the stresses computed in a pipe stress program. Note that in the first case where the fatigue stress exceeds its allowable by 111%, the number of permitted cycles for the calculated fatigue stress level is 5050, and in the second case where the fatigue stress exceeds its allowable by 171%, the number of permitted cycles is 1381. The strain concentration factor (Ke) for this last stress is 1.9. Providing the number of cycles the transfer line was actually to undergo is less than 1381, the system is certainly safe. It has the ASME Codes intended safety factor plus some additional safety factor because of the extra conservatism in Ke.

```
TRNFER                         Fe/Pipe  Version 2.0
JAN  5,1991                    COADE RESEARCH SERVICES
0:02am                         Released Jan. 15, 1992

ASME Overstressed Areas

Branch at Junction

  Pl+Pb+Q      3(Smavg)     Primary+Secondary (Inner) Load Case 2
   60,327        48,750     Plot Reference:
      psi           psi     3) Pl+Pb+Q < 3(Smavg) (OPE,Inside) Case 2

          123%

  Pl+Pb+Q      3(Smavg)     Primary+Secondary (Outer) Load Case 2
   71,201        48,750     Plot Reference:
      psi           psi     4) Pl+Pb+Q < 3(Smavg) (OPE,Outside) Case 2

          146%

Pl+Pb+Q+F            Sa     Primary+Secondary+Peak (Inner) Load Case 2
   30,164        27,020     Stress Concentration Factor = 1.0
      psi           psi     Strain Concentration Factor = 1.5
                            Cycles Allowed for this Stress = 5050.1
          111%              "B31" Fatigue Stress Allowable = 40625.0
                            Markl Fatigue Stress Allowable = 41701.0
                            Plot Reference:
                            5) Pl+Pb+Q+F < Sa (EXP,Inside) Case 2

Pl+Pb+Q+F            Sa     Primary+Secondary+Peak (Outer) Load Case 2
   35,601        20,746     Stress Concentration Factor = 1.0
      psi           psi     Strain Concentration Factor = 1.9
                            Cycles Allowed for this Stress = 1380.9
          171%              "B31" Fatigue Stress Allowable = 40625.0
                            Markl Fatigue Stress Allowable = 41701.0
                            Plot Reference:
                            6) Pl+Pb+Q+F < Sa (EXP,Outside) Case 2
```

The only **Fe/Pipe** input required for this example is the geometry of the intersection, pad, and weld, and the material properties. (The total input time for this problem was about two minutes.) All of the **Fe/Pipe** input fields are self explanatory except for the "Attached Pipe Length " fields shown below:

```
Input Data Echo

**************************************************************************
                    General
**************************************************************************

    YES     <— Compute Inplane, Outplane, Axial and Torsional
                   Sif's and Flexibilities

  26*12     <— Inplane attached pipe length  (in.)
  27*12     <— Outplane attached pipe length (in.)
  21*12     <— Axial attached pipe length (in.)
```

When the "Compute Sif's and Flexibilities" field is "YES", and "Attached" Lengths are entered, **Fe/Pipe** makes the intersection load reduction calculation. The directional attached lengths characterize the piping system bending or translation that is possible in a given direction. For example, the inplane attached length is the sum of the lengths of all of the pipe from the first effective support to the intersection whose axes are along the axial or outplane direction. It is these pipe runs that will "bend" when an inplane moment is applied to the intersection. The ?-Help text for these inputs should be read carefully. As can be seen from the example, the load reduction estimation from **Fe/Pipe** can be a very useful tool in doing a piping component evaluation. The computed results are worth the effort it takes to find the data. Special thanks for help with this problem are extended to Ahmad Maskeen. Mr. Maskeen can be reached in Saudi Arabia at 85-62-324.

COADE OFFICE EXPANSION

As some users know, COADE has recently undergone an office expansion. The existing COADE office (at 12777 Jones Road) will continue to develop and support **CAESAR II**, **PROVESSEL**, and **CodeCalc**. The new office (at 15207 Jones Road) will develop and support **Fe/Pipe**.

Users requesting support and/or sales and production information are urged to contact the proper office for more efficient service. The addresses, telephone, and fax numbers are listed below.

COADE Engineering Software	COADE Research Services
Piping & Pressure Vessels	Finite Element Applications
12777 Jones Road, Suite 480 Houston, Texas 77070	15207 Jones Road Houston, Texas 77070
Ph: 713-890-4566 Fax: 713-890-3301	Ph: 713-251-8084 Fax: 713-251-1830

SOFTWARE STATUS

CAESAR II In December of 1991, **CAESAR II** Version 3.16 shipped to all users current on their updates. The most notable features of this version are: the Stoomwezen piping code, modification of the modulii of elasticity in conformance with the 1990 code updates, and a configuration program to manipulate the setup file.

The next **CAESAR II** release will be Version 3.17 and has entered the QA procedures. Version 3.17 will include: support of the DOS environment, user control of text colors, on-line error processing, access to all ancillary programs via the utilities menu, input/output associations, improvements to the "Flange stress/leakage" module, and the correction of those errors and omissions discussed under "**CAESAR II** Specifications".

The most significant feature of Version 3.17 is the support of the DOS environment, which allows the software to be run from various subdirectories, in addition to the installation subdirectory. This enables the user to separate job files based on project or client, to aid in disk organization and data archiving. In order to utilize this feature of **CAESAR II**, two changes must be made to the system start up file "AUTOEXEC.BAT". These changes are the modification of the "PATH" statement and the addition of a single "SET" command. Users unfamiliar with this topic are urged to consult their DOS manual (or other references) before installing Version 3.17. (See also *PC Hardware for the Engineering User* earlier in this issue.)

Fe/Pipe Version 2.0 On January 15, 1992 development of Version 2.0 of **Fe/Pipe** was completed. Documenation has just been finished. The full package of software and documentation is being shipped to all users the week of March 9. There were a number of significant enhancements made to the **Fe/Pipe** program in version 2.0 that make it a more practical tool for the piping and vessel engineer.

- Direct computation of intersection stiffnesses for input into a "beam-type" pipe stress program. Piping designers no longer have to wastefully overdesign vessel and pipe connections. Accurate stiffnesses are automatically generated that can significantly reduce loads in the piping/vessel system.

- Direct computation of stress intensification factors for the piping intersection or for a pipe/vessel nozzle junction. These SIF's can be put back into a "beam-type" piping program to generate more accurate stresses at the junction. Comparisons are made to the B31 SIF equations so that the user can know if his B31 analysis