

The COADE Mechanical Engineering News Bulletin is published periodically from the COADE offices in Houston, Texas. The Bulletin is intended to provide information about software applications and development for Mechanical Engineers serving the power, petrochemical, and related industries. Additionally, the Bulletin will serve as the official notification vehicle for software errors discovered in those Mechanical Engineering programs offered by COADE. (Please note, this bulletin is published only two to three times per year.)

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PC Hardware & Systems for the Engineering User (Part 17)

ESL News

It is a little known fact that computers of the same make and model, purchased from the same vendor, on the same day, can be very different internally. These differences range from a variety of chips on boards, to a variety of boards, all

(supposedly) performing the same function. Recently an ESL (External Software Lock) access problem involving IBM PS/2 Model 77 machines has been solved. For all users of COADE software running on IBM PS/2 Model 77 machines, there is a new ESL available which should resolve all access problems. The revision necessary to the ESL was made possible through the assistance of M. K. Ferguson, Inc., which lent a Model 77 to the ESL manufacturer, Software Security, Inc., for testing purposes. (The loaner machine was necessary because both COADE and SSI had obtained Model 77s directly from IBM, and they accessed the ESL correctly.) It was only through testing on a "problem" machine that the issue could be resolved. Users experiencing problems with IBM PS/2 Model 77 machines should contact COADE to arrange an ESL upgrade.

Another parameter to check on PS/2 models is the *Arbitration Level of the Parallel Port*. The following eight steps should be used to insure the arbitration level is "disabled":

1. Press [Ctrl] - [Alt] - [Del] simultaneously to reboot the machine.
2. When the cursor moves to the upper right corner of the screen, press [Ctrl] - [Alt] - [Ins] simultaneously. This will display a menu.
3. From this Main Menu, select Item 3, Set Configuration. This will produce another menu.
4. From this menu, select Item 2, Change Configuration. This produces a configuration screen.
5. From the configuration screen, choose *Parallel Port Arbitration Level*.
6. Press [F5] as many times as necessary until the setting "Disabled" is reached.
7. Press [F10] to save the configuration.
8. Exit this setup program by pressing: [Enter], [F3], [F3], and [Enter].

While this is an involved process, some users have been able to access their ESL once the above steps have been implemented.

At this time, the only other machine with known ESL access problems is the DELL 486. Unfortunately, DELL has been less than cooperative, so this problem will take some time to resolve. However, for those users with DELL 486s, the following command, placed in the AUTOEXEC.BAT file, may help the situation:

```
SET SSI_ACT=40,40,40
```

Additionally, the updated ESL for the PS/2 problem has worked on several problem DELL machines. DELL users who want to swap their ESLs should also contact COADE.

Users attempting to connect notebook computers to networks are typically using adapters from Xircom. Unfortunately, this adapter connects to, and takes over, the parallel port - where the ESL is attached. This behavior by the Xircom adapter effectively hides the ESL from the software, rendering it inoperable.

For users with a "Xircom 1" adapter, it is possible to utilize an ESL, once three requirements are met. First, a "parallel port multiplexer" must be obtained from Xircom. This device essentially provides two ports, one for the network and one for the ESL. Second, the driver PPX.COM, also from Xircom, must be loaded. Third, revised software must be obtained from COADE. COADE was informed of this software change in early November, after the **CAESAR II** Version 3.20 master disks were generated. New versions of **CAESAR II** and **PROVESSEL** will be available in December to address this software change to address PPX.COM. A revised **CodeCalc** will be available in January.

For users with a "Xircom 2" adapter, there is no current work around. COADE has been informed by Software Security that Xircom informed all hardware lock vendors that they (Xircom) will no longer make allowances for, or support, hardware locks.

Network News

For those users running COADE software on networks, several configuration items need to be addressed. The first item for consideration is the actual network in use. Novell 3.11 is implemented at COADE - all testing and network development is targeted toward a Novell network.

The second item to consider is the setup of the workstation to properly address the printer. This requires the use of the

Novell "capture" command, which can be placed in the login script. An example "capture" command is illustrated below:

```
#CAPTURE L=3 Q=que_name J=special TI=15 NB
```

Each item in the above command is explained as follows:

Required to execute a DOS program from inside a login script. If this command is manually entered from the keyboard, the "#" should not be used.

L=3 Specifies that LPT3 should be captured.

Q=que_name Assigns LPT3 to the queue whose name is que_name. Queue names are system dependent.

J=jcon_name Applies a queue "job configuration" file whose name is jcon_name to each print job. This may be necessary for postscript printers. Job configuration files are usually obtained from the printer manufacturer.

TI=15 Indicates that the print queue should close a print file after 15 seconds of inactivity. By default the Novell print queues do not close a print job/file until the application (.EXE) is exited. This may cause a problem with some print jobs, especially if they contain graphics. The "time out" parameter is used to force the queue file to close. Note, this was required at COADE for both an HP and a postscript printer. This parameter was not necessary for a dot matrix printer. Note also that this option should only be used if absolutely necessary, otherwise text reports could print sooner than expected.

NB Specifies that the printouts are not preceded by a banner page. For large networks where many users print jobs, banner pages may be desired to separate jobs and indicate ownership.

The third item to consider is the size of the hard disk on the file server. The current file manager contained in all COADE software packages has a limit of 250 files in any given directory, and a limit of 155 directories on the disk partition. The current file manager was not designed for 1.2 Gbyte disks configured as a single partition. Additionally, the list of valid disk drives assumes they are contiguous.

CAESAR II Version 3.2 Features

Documentation: One of the major components of the 3.2 release is the rewrite of the documentation. The new documentation is presented in three manuals to aid users in finding the desired information. Each manual (User's Manual, Applications Guide, Technical Reference Manual) contains an introduction stating the purpose and intended use of the remaining chapters.

Solvers: Version 3.2 provides users with static solvers (both in-core and out-of-core) converted to 32 bit operations running in extended memory. Most of the jobs that previously required the out-of-core solver will now run in-core. (The out-of-core solver is provided for situations where a large job must be analyzed on a machine with insufficient extended memory.) The static output processor has been similarly converted. These changes mean that many of the past memory allocation problems have been eliminated. *Additionally, solution times have been accelerated by an order of magnitude.*

Time History: Modal time history analysis capabilities have been added to the dynamic solution and animation modules. This provides users with another method of evaluating piping system performance under impulse loads.

Static Load Cases: The internal storage area for the static load case data has been significantly expanded. Users will no longer receive errors when attempting to define many complex load cases.

Printing: The direct printer access by **CAESAR II** has been significantly altered for Version 3.2. Users can now configure the program to direct printed output (text and graphics) to any parallel port. Additionally, the user may define printer control strings, thereby allowing **CAESAR II** to change the number of characters per line, and the number of lines per page.

As an example of the use of this file, a *beta* user has submitted his preferred configuration string, reproduced below:

```
27 40 49 48 U 27 40 s p 49 50 h 49 48 v s b T 27 & a 49 48 L
```

This string sets the laser jet printer to 12 characters per inch and adjusts the left margin to the right by 10 characters to allow for binding holes.

Another *beta* user submitted the following string:

```
27 E 27 & 1 49 o 53 46 56 c 56 E 27 & a 49 56 L 27 & k 50 S
```

This string sets the laser jet printer to Landscape mode, 58 lines per page, 16.7 characters per inch.

Output Processor: The output processor, in addition to operating in 32 bit protected mode, is now able to handle much larger models and more load cases. This output processor also provides a "Table of Contents" of generated reports, and incorporates our standard file manager for directing output reports to disk.

Tank Nozzles: API-650 Nozzle Flexibilities can be incorporated into a piping model, similar to WRC-297 nozzles.

Data Bases: Version 3.2 provides the Australian Structural Steel tables, and spring hanger tables from China (SINOPEC), India (BHEL), and Italy (Flexider).

Using the New CAESAR II Documentation

The new release of **CAESAR II** includes three new manuals. The **User's Manual**, the **Applications Guide**, and the **Technical Reference Manual**. These three manuals represent a much more logical organization of the **CAESAR II** documentation. Each guide or manual serves a specific purpose, so the user will be able to access the relevant material without having to sift through all of the available documentation. Hopefully this will allow for more direct searches for examples, explanations, etc. without having to strain to lift the old manual off of the shelf. The preliminary work on the new documentation dealt mainly with developing the logical organization that will allow for expansion of the material, without simply cluttering chapters. For this reason, now that the groundwork has been laid, we hope to improve the documentation further by including more examples, new tutorials, extra troubleshooting tips, and overall, more quality material.

The **User's Manual** is the starting point, providing general knowledge of all of the different capabilities of **CAESAR II**, beginning with the installation program. If the concern is simply what button to push, or what procedure to use, the **User's Manual** provides this general information. The questions answered by this manual are more "What do I do?" rather than "Why am I doing this?". To help the first time user to get his/her feet wet, a Quick Start chapter has been

added to this manual. This chapter gives a brief overview of a static analysis, beginning with the input screens. The manual also provides general model creation information, as well as descriptions of each of the available menu items. Additionally, a Troubleshooting chapter has been added at the end of the document. It is our hope that the Troubleshooting chapter will continue to expand to accommodate as many commonly asked questions as possible.

The **Applications Guide** should be accessed for specific modeling questions. This manual answers questions of the nature of "How do I do this?". To answer this question, the manual provides single component modeling examples, as well as entire example problems. In the component examples, specific elements such as mitered bends, flexible nozzles, tied expansion joints, etc. are addressed. In the examples chapter, complete analyses, such as earthquake, relief loads, water hammer, etc. are performed. In future releases, additional examples will be added, as well as several tutorials. Preliminarily planned tutorials include those on static analysis, dynamic analysis, buried pipe, and one on the various equipment and components analyzed by **CAESAR II**.

The third manual, the **Technical Reference Manual**, contains technical aspects of both the software itself, and the software applications. The software aspects discussed are those which we felt would needlessly complicate the discussions in the **User's Manual**. To begin with, there is a complete discussion of each of the elements that make up the **CAESAR II** configuration file. The **User's Manual** addresses the configuration file very briefly, whereas the discussion in the **Technical Reference Manual** is much more thorough. Also, a complete **CAESAR II** update history, and a complete listing of all of the installed files can be found in this manual. The **Technical Reference Manual** should also be accessed for more in-depth discussions of specific topics. Where the **Application Guide** attempts to teach by example, this manual attempts to provide the significant background material.

In addition to the above three manuals, a **Quick Reference Guide** will also be distributed. This guide provides quick access to commonly needed information, such as intersection types, material lists, and code stress equations. The **Quick Reference Guide** also provides a complete index listing of each of the three manuals. The complete index listing is probably the most effective way to begin a search for specific information needed for the job at hand.

Network Versions of CAESAR II and CodeCalc in Testing

In August, the first network versions of **CAESAR II** and **CodeCalc** were completed, for in-house testing. These versions are designed to run on Novell's Netware Version 3.11, and utilize a special network ESL. These versions are currently being evaluated at COADE and must still undergo a "beta test" phase at several client locations before they are generally available.

Users contemplating installing either **CAESAR II** or **CodeCalc** on their Novell network should consider the following points:

- The network ESL must be attached to the serial port of the file server. During the startup of the file server, an NLM is loaded to control communications with this ESL, and monitor the number of available licenses remaining.
- The operation of the software is somewhat slower on the network than on a stand alone PC, due to the communication protocols of the network. Additionally, the network's serial ESL is slower than the PC's parallel ESL.
- It is recommended that 1 out of every 5 licenses be provided as a local ESL. This way the program can be used in the field or other off site locations, which don't have access to the network.
- There will be no "limited" network version of **CAESAR II**. All network versions will be either full purchases or monthly leases. Pricing on the network versions has not been finalized, however, network licenses will be no less than the stand alone licenses. There will be a conversion fee for users wishing to switch from stand alone licenses to network licenses.

COADE Seminar Schedules

The following piping seminars have been scheduled for 1994 in the COADE Houston offices.

January 19-21	Introduction to Pipe Stress
January 24-28	Statics & Dynamics
March 7-11	Statics & Dynamics
May 2-6	Statics & Dynamics
September 21-23	Introduction to Pipe Stress
September 26-30	Statics & Dynamics
November 7-11	Statics & Dynamics

The following pressure vessel seminars have been scheduled for 1994 in the COADE Houston offices.

February 7-9	ASME Sect VIII
October 17-19	ASME Sect VIII

Mechanical Engineering News Article Index

At the request of many COADE clients, we have compiled the following index of articles from all past issues of *Mechanical Engineering News*. This index is intended to aid clients in finding reference articles quickly. (Articles on Software Development, Seminars, and General News have been omitted for brevity.)

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API-650 Nozzle Flexibilities

Version 3.2 of **CAESAR II** incorporates nozzle flexibilities according to API-650. API-650 nozzles are sometimes referred to in the literature as “*low tank*” nozzles. These nozzles are typically located in the bottom shell course of large storage tanks. The API-650 computations performed by **CAESAR II** provide flexibilities at pipe terminations similar to the WRC-297 bulletin. These computations are performed in accordance with the 9th edition of API-650, dated July 1993. (The 9th edition differs only slightly from the 1988, 8th edition. Appendix P, which deals with nozzle flexibilities, corrects all of the mistakes in the 8th edition, and modifies the equation for nozzle rotation.)

The main objective of incorporating API-650 nozzle flexibilities into a piping analysis is to ensure that the proper loads can be computed at this point, and that these loads will not exceed the maximum allowed loads - also determined by API-650. Although there are three global forces and three global moments which can be applied to the tank opening, the radial force, the longitudinal moment, and the circumferential moment are the only three normally considered significant enough to cause shell deformation. The stiffnesses associated with these three forces and moments are computed and automatically inserted into the **CAESAR II** analysis when the user specifies an API-650 nozzle. The figure below shows the orientation of these loads.

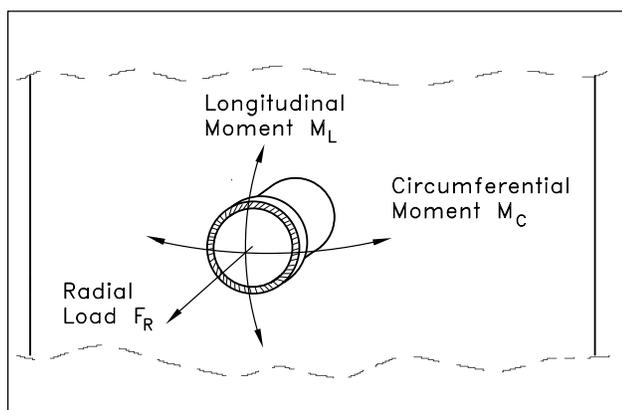


Figure 1

The API-650 Appendix P also provides equations to compute the radial growth and longitudinal rotation of the shell at the nozzle location. These values are computed by **CAESAR II** and provided as additional information. **These values are not automatically applied as pre-defined displacements.** If these values are desired as pre-defined displacements, the user must manually specify them on the piping input spreadsheet.

Appendix P also provides equations to compute the maximum allowed piping loads. These values are also computed by **CAESAR II** and provided to the user as additional information. Once the **CAESAR II** run has been made, the restraint loads computed at the nozzle should be compared to these limits to assure compliance. (Note, the maximum piping loads computed according to API-650 assume only a single load is active, i.e. when evaluating the radial force, API-650 assumes the two moments are zero.)

The **CAESAR II** implementation of API-650 Appendix P is as follows:

1. The data base containing the digitized curves from Appendix P is located and made available to the program.
2. The stiffness coefficients, based on the user's nozzle and tank parameters, necessary to access these curves are determined.
3. Using these stiffness coefficients and the digitized curves, the dimensionless values are obtained. Based on these values, the modulus of elasticity and the nozzle radius, the desired stiffnesses are obtained. These stiffnesses are then applied by **CAESAR II** to the nozzle node as restraint stiffnesses. (Users should note that these curves from Appendix P are log-log curves. Not only must logarithmic interpolation be used, but a slight variation in the curve values can produce stiffness variations in excess of 50%.)
4. The unrestrained deflection and rotation of the nozzle are determined, based on the product head and thermal expansion. As previously noted, these values may be used as pre-defined displacements if the user so desires. **CAESAR II** will not automatically apply these values as nodal displacements.
5. Finally, the limiting piping loads (forces and moments) are determined. These values are also obtained through the use of interpolation of the digitized API curves.

The input screen necessary to describe an API-650 nozzle is shown in Figure 2 below. This data is very similar to that required for a WRC-297 nozzle.

Nozzle Node Number (API-650)	110.000
Tank Node Number	
Nozzle Outside Diameter (in.)	24.000
Nozzle Wall Thickness (in.)	.375
Tank Outside Diameter (in.)	3120.000
Tank Wall Thickness (in.)	1.330
Reinforcing on (1) Shell or (2) Nozzle	2.000
Nozzle height (in.)	24.750
Fluid height (in.)	768.000
Fluid Specific Gravity	1.000
Tank Expansion Coefficient (in./in./deg)	.667000E-05
Delta Temperature (F)	130.000
Tank Modulus of Elasticity (lb./sq.in.)	.288000E+08

Figure 2

The results of the API-650 computations are shown in Figure 3. Note that the only quantities used by CAESAR II in the flexibility analysis are the three stiffness values. All other data is presented for information only.

API-650 NOZZLE DATA FOR NODE 110		
INPUT PARAMETERS:		
Tank Diameter	(in.) =	3120.000
Nozzle Height above bottom plate	(in.) =	24.750
Nozzle Outer Diameter	(in.) =	24.000
Shell Course Thickness	(in.) =	1.330
Max Fluid Height above bottom plate	(in.) =	768.000
delta Temperature	(F) =	130.000
Modulus of Elasticity	(lb./sq.in.) =	.288000E+08
Expansion Coefficient	(in./in./deg) =	.667000E-05
Fluid Specific Gravity	=	1.000
Reinforcement on (S)hell or (N)ozzle	=	N
COMPUTED VALUES:		
AXIAL TRANSLATIONAL STIFFNESS	(lb./in.) =	114152.
LONGITUDINAL BENDING STIFFNESS	(in.lb./deg) =	1382592.
CIRCUMFERENTIAL BENDING STIFFNESS	(in.lb./deg) =	2497816.
RADIAL DEFLECTION DUE TO FLUID & TEMP	(in.) =	2.38
LONGITUDINAL ROTATION DUE TO FLUID & TEMP	(deg) =	-1.86
MAX ALLOWED RADIAL FORCE	(lb.) =	63476.
MAX ALLOWED LONGITUDINAL MOMENT	(ft.lb.) =	130649.
MAX ALLOWED CIRCUMFERENTIAL MOMENT	(ft.lb.) =	429899.
<C> TO CONTINUE <P> TO PRINT		

Figure 3

CodeCalc: Hillside and Off-Angle Nozzle Angles

There are two main categories of off-angle vessel nozzles: those which are off-angle in the longitudinal plane of the cylinder (Y-angle nozzles, Figure 4), and those which are off angle in the circumferential plane of a cylinder, or in a head (Hillside nozzles, Figure 5). In order to analyze these kinds

of nozzles, CodeCalc asks the user for the angle between the nozzle centerline and a tangent to the vessel mean radius, as also shown in Figures 4 and 5. However, for hillside nozzles, the determination of this angle can be difficult. The purpose of this article is to provide a few simple equations that can help the user determine this nozzle angle.

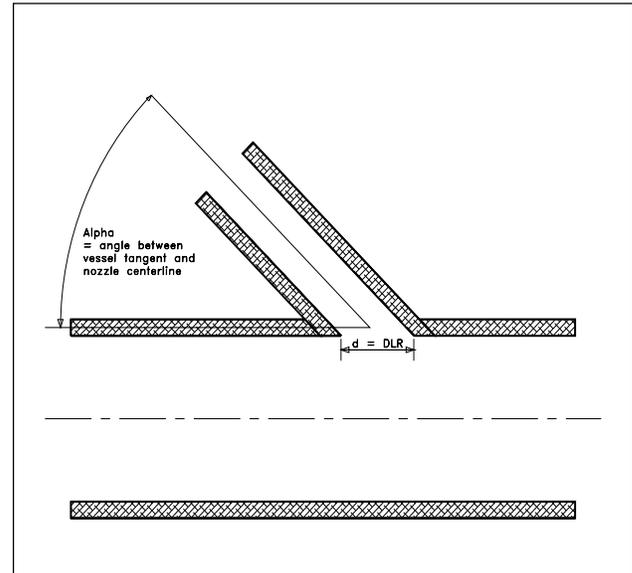


Figure 4 - Y-angle Nozzle: Angled in longitudinal plane of cylinder

The overall goal of this calculation is simple: find an angle for which the calculated diameter of the hole will match the actual diameter of the hole. The finished diameter is the dimension *d* in Figures 4 and 5, which is called *DLR* in the CodeCalc print-outs. If we had the nozzle in front of us and could measure *d*, then we could calculate the input angle very simply using the following equation:

$$\sin \alpha = \frac{d_n}{d}$$

where: d_n = inside diameter of nozzle

d = DLR = finished diameter of hole

$\sin \alpha$ = sine of angle between nozzle and vessel

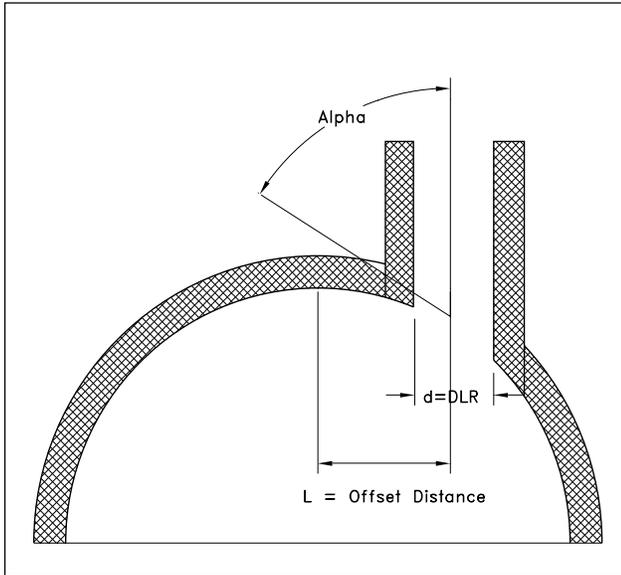


Figure 5 - Hillside Nozzle

When we analyze Y-angle nozzles, the angle is known and the result is exact — this is really all the information we need.

$$d = DLR = \frac{d_n}{\sin \alpha}$$

However, when we analyze hillside nozzles, as shown in Figure 5, the angle is usually not known. Instead, we may know the offset distance for the nozzle. This distance, *L*, is the distance between the centerline of the cylinder or head, and the centerline of the nozzle. A first approximation to the angle would take the cosine of the angle as *L*/*r_m*, where *r_m* is the mean cylinder or head radius at the point of attachment. However, this approximation turns out to be too inaccurate for normal use.

The ASME Code has a sample problem, L-7.7, which shows what their preferred method is. (They do not explicitly address this off-angle problem in the body of the Code.) Figure 6, taken from ASME, Section VIII, Division 1, Addenda A92, page 512, shows this sample problem. The key to their approach is the calculation of two angles, α_1 and α_2 , and then the calculation of the finished diameter from the difference between these two angles. You can follow their calculation on page 512 and 513 of the Code. For our purposes, we do not need to carry the calculation that far. The angle we are looking for is just the average of the two Code angles, calculated as:

$$\alpha = \arccos \frac{L - r_n}{r_m}$$

$$\alpha = \arccos \frac{L + r_n}{r_m}$$

$$\alpha = \frac{\alpha + \alpha}{2}$$

where: *L* = offset distance cylinder/head centerline

r_n = inside nozzle radius

r_m = mean vessel radius

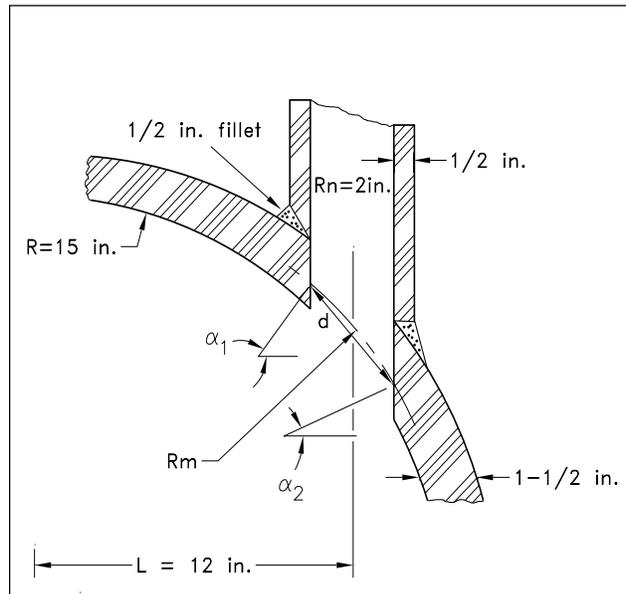


Figure 6 - ASME Code Figure L-7.7

These three equations can be used without any further information for any hillside nozzle in a cylinder. However, you need to apply them carefully to hillside nozzles in heads. When a hillside nozzle is in an elliptical or torispherical head, the nozzle may be located in the spherical portion of the head, the toroidal portion of the head, or it may straddle the two portions. This is shown in Figure 7. Each of these cases requires a slightly different *L* and *r_m* to be used in the equations.

When the nozzle lies entirely within the spherical portion of the head (Figure 7(a)), *L* is simply the offset from the head centerline, and *r_m* is the spherical radius of the head. For spherical or torispherical heads this should be a known

radius (Code dimension L in Figure 1-4 of Appendix 1). For elliptical heads, the spherical portion is taken to be a circle drawn on the head with a diameter of 80 percent of the head diameter. The radius of the spherical portion is taken to be 0.90 times the head diameter. The nozzle offset from the vessel centerline should be known from the vessel drawings.

The nozzle can also lie entirely in the knuckle portion of the head (Figure 7(c)). The mean radius (r_m) is the mean knuckle radius, and the offset (L) is the distance from the origin of the knuckle radius to the centerline of the nozzle. Note that for an elliptical head, the knuckle is defined as anything outside a circle drawn on the head with a diameter of 80 percent of the head diameter. The knuckle radius is 0.17 times the vessel diameter.

Finally, the nozzle may be located so that part of the nozzle is in the spherical portion, and part in the knuckle (Figure 7(b)). In this case, the angle at the part of the nozzle in the spherical portion should be calculated as described for Figure 7(a), and the angle at the part in the knuckle portion should be calculated as described for Figure 7(c). That is, calculate the inside angle using the spherical radius of the head and offset from the centerline. Calculate the outside angle using the mean radius of the knuckle and the offset from the knuckle origin.

Use of these equations should yield correct nozzle angles for almost all off-angle nozzle configurations.

An Introduction to Time History Analysis

With the release of **CAESAR II** Version 3.20, users can now perform linear modal time history analysis of dynamic events imposed on piping systems. (This analysis technique provides a true simulation of the dynamic system response as a function of time due to the imposed load, and is usually more accurate than the corresponding response spectrum method.) This article will introduce this analysis technique through the use of a cantilever beam model, as well as demonstrate its accuracy in simulating dynamic response. This cantilever beam is described in Figure 8 below.

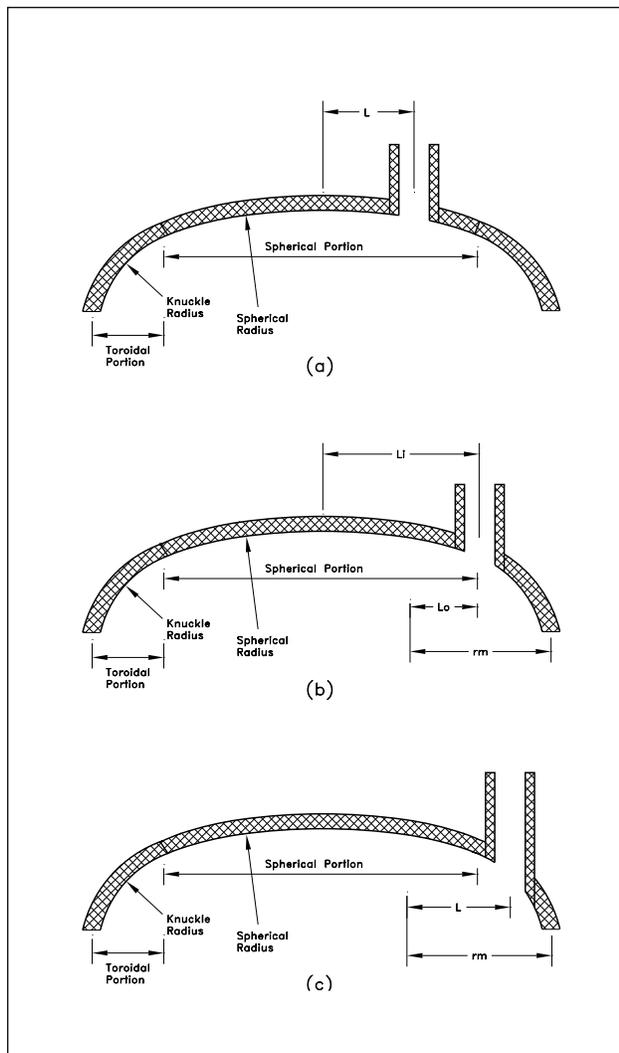


Figure 7 - Hillside nozzles in heads

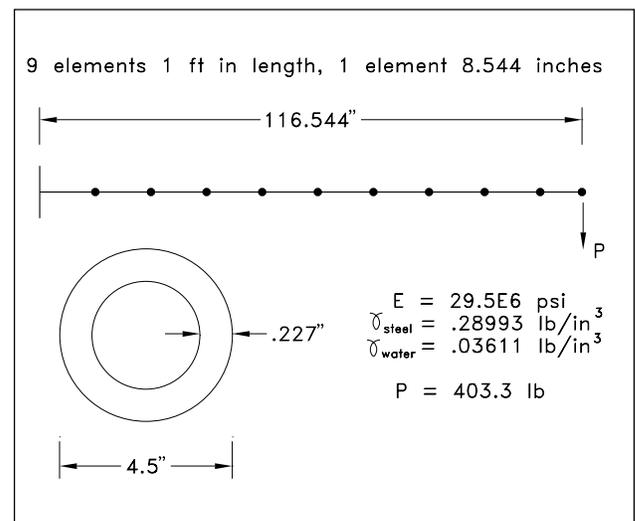


Figure 8

This cantilever beam is a 4 inch standard wall, low carbon steel pipe. The diameter, wall thickness, density, elastic modulus, and fluid density are shown in the figure. Notice that the length is 116.544 inches and a point load of 403.3 pounds is applied at the free end. These two values are

somewhat contrived so that the results and behavior of the beam are predictable. The length was obtained such that the first natural frequency of the beam is exactly 10 Hz, which corresponds to a period of 0.1 seconds. The first natural frequency of a cantilever beam modeled as a series of lumped masses can be calculated as:

$$f = \frac{1}{2\pi} \sqrt{\frac{Eg}{\sum W_i X_i^3}}$$

where: g = acceleration of gravity = 386.088 in/sec²
 W_i = weight of the pipe and fluid lumped at node (i)
 E = Young's modulus of pipe material
 I = moment of inertial of pipe
 X_i = distance of node (i) from fixed end

Alternatively, one could turn to a reference, such as *Marks' Standard Handbook for Mechanical Engineers*, 9th Edition. Page 5-74 provides the following equation for the first five natural frequencies of a cantilever beam:

$$f_n = c_n * \sqrt{\frac{Eg}{wL^4}}$$

where: w = weight per unit length of the beam
 L = length of the beam
 c_n = coefficients defined in the Table below

Frequency:	1	2	3	4	5
c _n value :	0.56	3.57	9.82	19.2	31.8

Once the length of the beam is known, the load required to produce a static deflection of 1 inch was determined from:

$$P = (1.0) * 3 * E * I / L^3$$

where "P" is the tip load, and all other terms are defined above. The value of "P" is computed as 403.3 pounds.

The objective of the analysis is to determine if the dynamic response of the beam under the time history matches the expected results (i.e. as predicted by dynamic theory). For example:

- A load applied instantaneously to an un-damped system should yield twice the displacement of the same load applied statically. Additionally the system should cycle between plus and minus this displacement forever.
- The same load applied to an under-damped system should show close to the same initial displacement, but further cycling decays the displacement magnitude.

- The same load applied to a critically-damped system should produce a response which goes through one cycle before stabilizing at the static displacement.
- The same load applied to an over-damped system should produce a response which does not cycle at all. Rather the displacement slowly attains the static magnitude.

Entering the **CAESAR II** dynamic input module and selecting "Time History" as the analysis type reveals that data in the following input forms must be specified (all other input forms are optional for this type of analysis).

- 3 Spectrum/Time History Definitions
- 7 Spectrum/Time History Force Sets
- 9 Spectrum/Time History Load Cases
- B Control Parameters

(For the purposes of this example, since only motion in the vertical plane is of interest, the "1 - Lumped Mass" option was used to zero the masses in the X and Z directions.)

For the "3 - Time History Definition", the impulse is defined as:

CANT TIME FORCE LINEAR LINEAR

which states that the impulse name is "CANT", it is a time verse force curve, and linear interpolation should be used for both axes. Only two data points are necessary to describe this impulse; point 1 is at 0.0, 1.0, and point 2 is at 3000.0, 1.0. This defines a straight line (instantaneous) impulse from 0.0 to 3000 milliseconds with a multiplier of 1.0. (Alternatively, the force of 403.3 pounds could be entered here.) This impulse is shown in Figure 9:

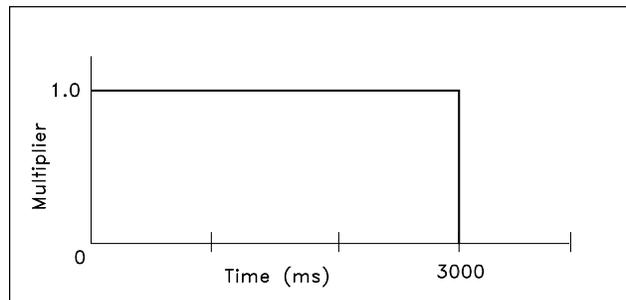


Figure 9

For the "7 - Time History Force Set", the magnitude and direction of the impulse are specified as:

$$-403.3 \text{ Y } 1100 \text{ 1}$$

which states that a load of -403.3 pounds is applied in the Y

direction at node 1100 and is associated with force set number 1.

For the “9 - Time History Load Cases”, the force set is associated with the impulse as follows:

CANT 1 Y 1

which states that the impulse curve CANT is included in the load case with a scale factor of 1.0, in the Y direction, and it acts on the load from force set 1. The total load applied to the system is the product of the force multiplier times the force set load times the load case scale factor.

For the “B - Control Parameters” screen, the following items must be specified:

```

TIME <-- Analysis Type (HARMONIC/SPECTRUM/MODES/RANGE/TIMEHIST)

5 <-- Max. No. of Eigenvalues calculated (0-Not used)
1.0 <-- Closely Spaced Mode Criteria/Time History Time Step (ms)
.2 <-- Load Duration (Time History or DSRSS method) (sec.)
0.03 <-- Damping (Time History or DSRSS) (ratio of critical)
40 <-- ZPA (Reg. Guide 1.60 - g's)/# Time History Output Cases

```

The analysis type should be set to “TIME”. The analysis is configured to extract and use the first five natural frequencies. The value for the time step is set such that the time step multiplied by the maximum frequency is less than 0.1, which yields a value of approximately .001 seconds. (Actually the lowest frequency was used here, since it dominates the system displacement response by a wide margin, as can be seen by a review of the Participation Factor report.) The duration is recommended to be at least as long as one full cycle, based on the lowest frequency (highest period). The fundamental period of this beam model is 0.1 seconds, so a duration of 0.2 seconds should produce two full cycles. The number of requested reporting cases was set at 40, providing reports (displacements, forces, stresses) every 5 milliseconds.

The value for the damping ratio was varied to show not only the effects of damping, but to provide data to visualize the response of the beam. (The damping ratio is the ratio of the actual damping to the critical damping. Values less than 1.0 are indicative of under-damped systems, while values over 1.0 are indicative of over-damped systems, and a value of exactly 1.0 indicates a critically damped system.) Five runs were made with the damping ratio set to 0.0, 0.03, 0.5, 1.0, and 2.0. For each run, the “Y” displacement at the tip of the cantilever (node 1100) as a function of time was stored in a data file. A plot of this data (generated using the 2-D plotting module of **CAESAR II**) is shown in Figure 10.

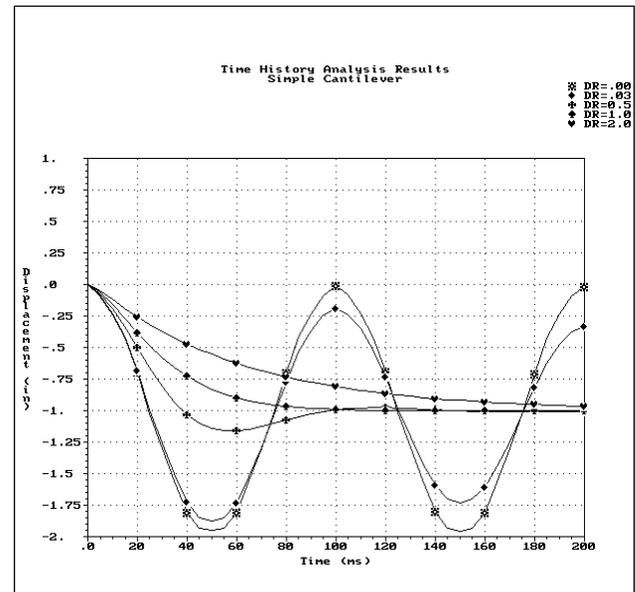


Figure 10

The curve for a damping ratio of zero (DR=0.0) represents the response of an un-damped system. The curve shows that the maximum response peaks at just less than -2 inches, which is the expected result. Additionally, the curve shows that a full cycle of response is attained every 100 milliseconds, as expected. This curve also shows that the vibration of an un-damped system does not diminish with time, i.e. the amplitude of the response for each cycle remains constant. Also notice that the motion is cycling about the magnitude of the static displacement of -1.0 inches. All of these characteristics of the “DR=0.0” curve are as expected.

The curve (with the diamond symbols) for a damping ratio of 0.03 (DR=.03) represents the response of a typical piping system. The maximum and minimum response values occur at multiples of the first natural period, as in the un-damped case. However, as can be seen in the figure, the amplitude of the response continuously decays with time. This behavior can be shown further by extending the analysis for a duration of 2 seconds. The resulting plot of the tip displacement in the

“Y” direction is shown in Figure 11.

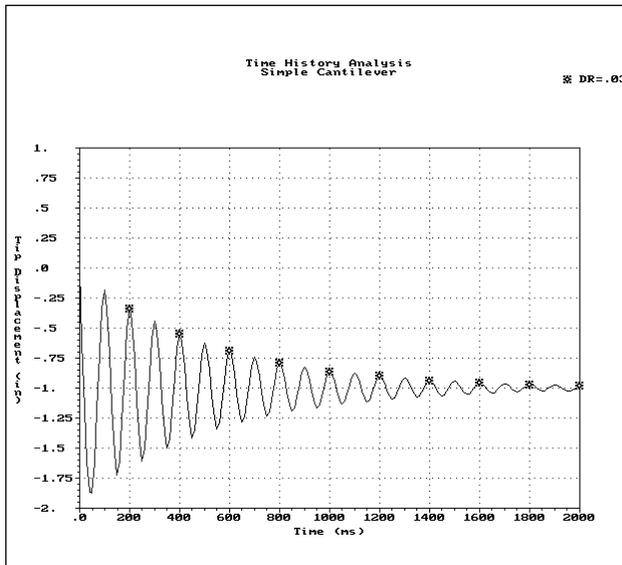


Figure 11

In Figure 10, the curve (with the clover symbols) for a damping ratio of 0.5 (DR=0.5) represents a highly damped, though still under-damped system. The oscillations after the first cycle are barely noticeable, and in fact die out after 1.5 cycles. The curve (with the spade symbols) for a damping ratio of 1.0 (DR=1.0) represents a critically damped system. The response for this run shows that the static displacement is attained in one cycle, after which there are no further oscillations. The curve (with the heart symbols) for a damping ratio of 2.0 (DR=2.0) represents an over-damped system. As the figure shows, the response for an over-damped system is not much more than a slow static response.

These results and a study of Figure 10 show that the behavior of a cantilever beam in a CAESAR II time history analysis is as predicted by theory. This article also illustrates the ease with which a time history analysis can be implemented using CAESAR II.

Selecting and Evaluating an Expansion Joint Assembly

The basics of the CAESAR II Expansion Joint Modeler appeared in the last issue of Mechanical Engineering News (The CAESAR II Expansion Joint Modeler - Volume 16). This article reviewed important model characteristics of the joint and discussed how the program modeled a simple expansion joint. The article did not indicate how a particular assembly was selected. For example, why would one recommend a simple, untied expansion joint instead of

universal, tied expansion joint. Reviewing the input matrix for the CAESAR II expansion joint modeler, one finds there are at least six assemblies available (see Fig. 12). Each assembly must have its advantages, so what are they? This article will examine these and other assemblies to highlight their individual uses.

Pressure	15	25	50	150	300	
Style	Untied	Tied	Hinged	Gimbal	U-UNI	T-UNI
Conv Matl	304SS	316SS	600Inc	625Inc		
<Arrow Keys> - Change Highlight Options <Enter> - Use Current Selections <Escape> - Ignore Request <DATA BASE: PATHWAY.DAT >						

Figure 12

Pipe is quite rugged. High loads and stresses in the piping do not justify the installation of an expansion joint. The equipment to which the piping is attached is another story. In many cases the piping attached to rotating equipment may be loaded to only 5% of its allowable so that the pump, compressor, or turbine loads do not exceed their allowable limits. This low load limit must be handled in both the cold and hot piping positions. Adding or adjusting supports should reduce the cold loads on the equipment, but the change between the hot and cold loads is a function of the thermal loads and the piping flexibility — adding flexibility is the purpose of the expansion joint. How much room is available for the joint and what sort of load must be decreased determines what expansion joint configuration should be used. Clearly identifying the force or moment to be reduced is the first step in specifying an expansion joint assembly.

What type of joint arrangement should be used? To quickly review the different expansion joint assemblies, just page through any expansion joint manufacturer’s catalog. On a more generic level, one would refer to A Practical Guide to Expansion Joints by the Expansion Joint Manufacturers Association, Inc. (25 North Broadway, Tarrytown, NY 10591). This booklet describes the parts of an expansion joint, how to design a system containing expansion joints, and recommendations on proper installation and handling. Closer to home, the CAESAR II expansion joint modeler offers the following:

- UNTIED (Single Expansion Joint) — The simplest assembly, a single joint has no restrictions on its motion; but, except for hoop stress, it also has no pressure containing capabilities. In exchange for providing the most freedom, this joint requires special care in overall design. The piping around the joint must be well guided to prevent any squirm in the joint that would be further aggravated by pressure thrust on the joint. Simple expansion joints also require axial stops or anchors

somewhere up and down the line to absorb the pressure thrust load. *So with all the freedom in this joint, the pressure containment requirements reduce its flexibility (in most applications) to only the axial direction.* A Double Expansion Joint is a related configuration. This is an assembly of two single joints separated by a short run of pipe. The pipe separating the two joints is usually restrained from motion by an anchor. The use of a double joint is the same as a single joint but it can share the total axial deflection between the two joints.

- **TIED (Single Tied Expansion Joint)** — Pressure thrust on the joint may be contained without adding anchors and guides to the attached piping. Instead, the axial thrust is checked by the addition of tie rods around the joint. These tie rods drastically alter the nature of the joint. When untied, the joint provides flexibility in the axial direction. Now tied, the joint is essentially rigid in the axial direction, but lateral flexibility is available to the designer and no additional thrust blocks or guides are required. *A simple, tied expansion joint, then, is installed perpendicular to the plane of required flexibility.*
- **HINGED (Hinged Expansion Joint)** — Tie bars on either side of a single expansion joint may also be hinged. With a single joint, though, the only allowed motion is angulation, not axial or transverse deflection, and angulation about the one hinge axis only. Hinged joints are quite compact and they easily contain the pressure thrust loads. In many cases, two or even three hinged joints work together to provide needed flexibility. Hinged joints often require guides to drive the piping into the flexible direction.
- **GIMBAL (Gimbal Expansion Joint)** — Gimballed joints combine two, perpendicular, hinges across an expansion joint. These bars, from either end of the joint, hinge off a ring at the center of the joint. This articulated joint allows bending about both axes perpendicular to the axis of the joint. Gimbal joints are usually used with other gimbal joints or hinged joints and with pipe guides. Guides are used to force motion in a line perpendicular to the hinge axes of the joints.
- **U-UNIV (Universal Expansion Joint)** — A universal expansion joint is a double joint without an anchor on the center spool piece. The lack of restraint on the center piece turns it into a linkage between the two joints. This linkage assembly converts the joints' bending flexibility into large transverse displacements. A longer center piece produces greater transverse offsets with the same bending on each joint. However, the lack of restraints or other pressure containing elements limits the application of this joint to low pressure lines.

- **T-UNIV (Universal Tied Expansion Joint)** — As the name suggests, these are assemblies that have a universal expansion joint configuration with a set of tie rods running over both joints to contain the pressure thrust in the line. The tie rods would also have some means of attachment to the center spool piece to stabilize the entire unit. Here again, the tie rods eliminate any axial flexibility but permit a great range of transverse movement through the bending of the two joints. The greater the length of the center pipe, the greater the transverse deflection with the same amount of joint bending.

Other configurations of note:

- **Swing Expansion Joint** — Some piping configurations require transverse flexibility in one direction but not the other. In these situations, a swing expansion joint is recommended. A swing joint is similar to a tied universal joint in that it has a pair of bellows and a center spool piece. Instead of tie rods, the swing joint will have hinged bars restraining the pressure thrust. The parallel hinges at either end of the assembly allow bending about only one axis rather than two. These joints therefore direct the transverse deflection of the joint along a defined vector perpendicular to the axis of the expansion joint assembly.
- **Pressure-balanced Expansion Joint** — Utilization of axial flexibility in an expansion joint usually requires the joint to be untied and heavily guided. Tie rods, while containing the pressure thrust forces, eliminate axial flexibility of an expansion joint. Another way of keeping axial flexibility without adding extra guides and thrust-resisting anchors is by using pressure-balanced expansion joints. A way of understanding this axial flexibility is to examine the center spool piece of a universal tied expansion joint. This center piece is relatively free to move axially. It is not affected by the pressure thrust as that load is carried across the entire joint through the tie rods. The center spool piece is resisted only by the axial stiffness of the attached expansion joints. If the piping ran out through the center spool piece (by placing a tee in the spool piece) and the pipe after the tie rods is capped, then the axial (or lateral) flexibility remains.

Additional hardware is needed for the proper installation on several types of expansion joint assemblies. Guides are required to drive the piping in a specific direction so that the joint deflects in a controlled and safe manner. Pressurized joints without tie rods or hinges require anchors to hold the pressure thrust load. In most instances, this pressure thrust load is contained by the pipe wall. The flexibility of the joint cannot limit the axial deflection of the pipe due to pressure and so this thrust must be held elsewhere upstream and

downstream from the untied joint. Any supports on the line must be examined to confirm their ability to withstand the pressure thrust load. *CAESAR II* (as most any other pipe stress analysis program) does not automatically incorporate structural analysis of pressure loading. It is up to the analyst to confirm that the load is of correct magnitude and applied at an acceptable location on the pipe. *CAESAR II* will apply the pressure thrust load on either end of an untied joint but this is only a good approximation. The properly located thrust load may be determined by imagining a position inside the joint; the pipe wall seen upstream and downstream from the joint is the proper point to apply the pressure thrust load. This point may be beyond a support that was assumed to contain the pressure thrust.

Expansion joint assemblies add flexibility to a piping system; that is certainly understood. What may not be so clear is that these assemblies cannot supply flexibility in all directions. The requirement to restrain the pressure thrust load must still be satisfied and this requirement will eliminate one or more flexible degrees or freedom from the configuration. One way, then, of categorizing the various assemblies is to examine the available flexibilities of these joints:

Type	Freedom	Notes
UNTIED	axial	lateral & bending with low pressures, reqs. guides and thrust supports
TIED	lateral	compact and stable
HINGED	bending	(one axis) used in combinations for lateral flexibility, guides recommended
GIMBAL	bending	(both axes) see above
U-UNIV	lateral	stable only for low pressure applications, total offset determined by length of center piece
T-UNIV	lateral	total allowable offset is a function of center piece length
Press. Bal.	“axial”	used with a bend or tee on center piece so it may be considered lateral, provides axial flexibility without additional pipe supports

Expansion joint assemblies, then, can be selected based on their ability to provide flexibility in specific directions, their

space requirements, and their support requirements. This article presents a sensible way to address the variety of configurations available to the piping system designer. The next issue of Mechanical Engineering News will review individual joint selection and evaluation.

Estimation of Nozzle Loads Using CAESAR II Software

Results of pipe stress analysis (or, for that matter, any structural analysis) can vary depending upon the assumptions made when modeling the problem. For example, the best means of defining boundary conditions, loading combinations, and material/element behavior is rarely clearly defined. For this reason, modeling conventions often develop in order to simplify the decision making procedure — or, so the engineer need not reinvent the wheel for every new problem.

In the pipe stress field, each piping code normally has modeling conventions associated with it, which have been found to produce the best estimate of pipe stresses under loadings normally associated with the industry to which the particular code applies. For example, the B31.1 Power Piping Code, used in an industry where pressures may not get high enough to significantly stiffen pipe, does not consider pressure stiffening during the calculation of flexibility factors of elbows. Likewise, pressure strain effects are usually included only on very long runs of highly pressurized pipes, such as might be found on cross-country gas transmission pipes, covered by the B31.8 Gas Transmission Piping Code. Also, the different piping codes have their individual conventions for calculating the stress intensification factors of local components.

The true behavior of the piping system under load is much more complex than that of any linear system solved by a pipe stress program. Evidence that the best assumptions for modeling piping systems are not hard and fast is indicated by the diversity of the various code requirements. One thing that is certain, however, is that the means of calculating pipe stresses for each code (which vary for every code) has been proven to be adequate when compared to the means of calculating the allowable stresses for the same code (which also vary from code to code).

However, the conventions chosen as best for stress calculation purposes may not provide the best means of maximizing the accuracy of other results — displacements, reactions, etc. — which are not addressed in as much detail by the codes. Nozzle loadings also vary by code (or more accurately, vary by modeling conventions, which may vary by code),

whereas the allowables do not, since they are set by the equipment manufacturers or according to standards of third parties, such as the American Petroleum Institute. Therefore it is important to get as accurate an estimate as possible of nozzle loads, independent of the particular code's modeling conventions. A single analysis, conducted for code compliance purposes, may show that the nozzle loads are well within the equipment's nozzle allowables. However, altering the modeling conventions may cause the loadings to vary significantly—in fact, far more than do the maximum system stresses. In some cases, it may be advisable to do several analyses, with different modeling assumptions, in order to determine the potential range of results.

Consider the system of which a portion is shown in Figure 13. This system represents a 42"-diameter, 1-1/2" wall, gas transmission pipe, with an internal pressure of 1000 psig. The pipeline originates at an anchor (not shown), and runs approximately one mile, with two expansion loops in the vertical plane, before turning horizontally as it enters a compressor station. The system temperature is expected to vary from an ambient of 70°F to a maximum of 100°F. The line is connected (restrained in six degrees-of-freedom) to a compressor nozzle at node point 360.

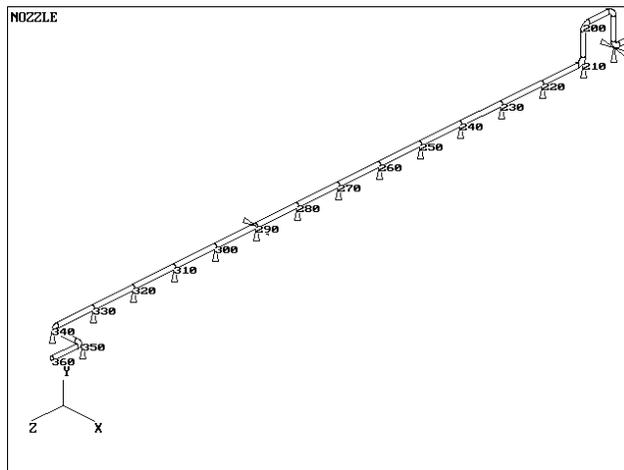


Figure 13

The initial analysis, done under the B31.8 Code using CAESAR II's defaults (i.e., pressure stiffening of elbows, absence of pressure strain effects, default rigid stiffnesses for all restraints and the anchor to the compressor, etc.) yielded the results shown for the first case (labeled **NOZZLE**) in Table 1. The maximum system Operating, Sustained, and Expansion stresses of 9538 psi, 7679 psi, and 3090 psi, respectively (with the maxima occurring at various points along the run), are acceptable against the code allowables. The nozzle loads on the compressor (for the Operating load

case), of 1185 lb (FX), -7745 lb (FY), 15025 lb (FZ), -30279 ft-lb (MX), 220575 ft-lb (MY), and 2697 ft-lb (MZ), which should be checked against the manufacturer's allowables, may be acceptable as well.

As noted above, the stress calculation is perfectly acceptable here, since the B31.8 allowable stresses take into consideration the modeling conventions normally associated with this code. The nozzle load calculations may be a different matter; the analyst should be more interested in finding a less code-dependent result. This can be done by re-analyzing the system with a few different modeling conventions—the true nozzle loads probably fall somewhere within the range of the results. The results for a number of modeling conditions are shown in Table 1. The analyses shown in this table were generated by making the following changes to the original model:

NOZZL1 — This model is identical to the original, except that pressure stiffening of the elbows is neglected. This matches the treatment of elbow flexibilities as endorsed by the B31.1 and other piping codes. Pressure stiffening reduces the flexibility of elbows by dividing the flexibility factor by:

$$[1 + 6 (P/E) (r_2/T)^{7/3} (R_1/r_2)^{1/3}]$$

Where:

P = internal pressure, psi

E = modulus of elasticity of pipe material, psi

r_2 = mean radius of pipe, in

T = wall thickness of pipe, in

R_1 = bend radius of elbow, in

For this application, the flexibility factor is reduced by a factor of 1.128.

Pressure stiffening of elbows can be activated, de-activated, or used as per the specific requirements of the code in use by setting a parameter in the CAESAR II configuration file (Option 9 of the Main Menu).

NOZZL2 — This model is identical to the original, except that pressure stiffening of straight pipes is included. This is the phenomenon that occurs when the sag of a guy-wire is reduced as it is tightened—likewise, a beam highly stressed in tension will deflect less under a lateral load than would the same beam subject to compressive stresses. The effect of tensile stresses on bending of a beam can be determined by

examining the element shown in Figure 14:

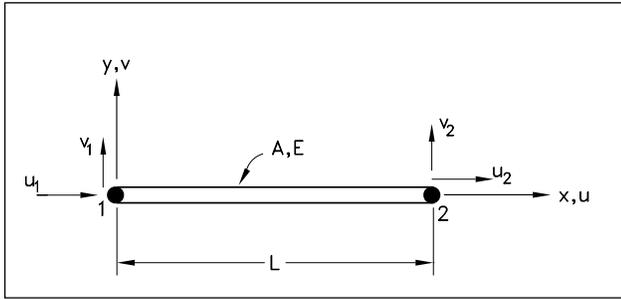


Figure 14

Initially assuming the beam in the figure is infinitely stiff in bending (so it is straight in any configuration), but can deform axially, the total axial strain (for small lateral displacements) is constant over the cross-section, and is computed as:

$$\epsilon_x = \epsilon_u + \epsilon_v$$

Where:

$$\begin{aligned} \epsilon_u &= \text{axial strain due to axial load} \\ &= (u_2 - u_1) / L \end{aligned}$$

$$u_2 = \text{axial displacement at node 2}$$

$$u_1 = \text{axial displacement at node 1}$$

$$L = \text{length of beam}$$

$$\begin{aligned} \epsilon_v &= \text{axial strain due to rotation of the bar through a} \\ &\quad \text{small angle } \theta \text{ while holding x-motion constant} \\ &= (1/\cos \theta) - 1 = (\sec \theta - 1) = \theta^2/2 \\ &\approx [(v_2 - v_1)/L]^2/2 \end{aligned}$$

$$v_2 = \text{lateral displacement at node 2}$$

$$v_1 = \text{lateral displacement at node 1}$$

If the beam is capable of bending, then the axial strain is no longer constant over the cross-section, since bending puts the cross-section in varying levels of compression and tension, based upon the distance to the neutral axis. For this case, the axial strain is a function of the distance from the neutral axis. An additional term must be added representing the rotation of the element face, and the ϵ_v term must be modified to take into consideration the fact that it is no longer linear with

respect to x:

$$\epsilon_x = u' - y \theta' + \theta^2 / 2$$

Where:

u' = first derivative of axial strain (with respect to x)

y = distance of point of interest from neutral axis

θ' = first derivative of angle of curvature (with respect to x)

Total elemental strain energy U (considered over the entire length and the entire cross-section) is calculated as:

$$U = \int_0^L \int_A \frac{1}{2} E \epsilon_x^2 dA dx$$

$$U = \int_0^L \int_A \frac{1}{2} E [u' - y \theta' / 2]^2 dA dx$$

Where:

E = material modulus of elasticity

A = cross-sectional area of element

Noting that (for a prismatic member):

$$\int_A dA = A$$

$$\int_A y dA = 0$$

$$\int_A y^2 dA = I$$

$$\int_A y u' dA = P$$

Where:

I = moment of inertia of element

P = axial force acting on element cross-section

The elemental strain energy can be written as:

$$U = \int_0^L \frac{1}{2} EA u'^2 dx + \int_0^L \frac{1}{2} P \theta' dx + \int_0^L \frac{1}{2} I \theta'^2 dx$$

The first integral in the expression above provides the standard stiffness matrix for a truss element (or the axial terms of the stiffness matrix for a beam element). The second

integral provides the lateral and rotational stiffness matrix for a beam element. These are the only stiffness terms that are included in **CAESAR II**'s default stiffness matrix formulation (as well as that of virtually all other stiffness method programs).

The third integral however, represents the work done when differential elements (of size dx) are stretched an amount of $\theta^2 dx/2$ by an axial force P. From this integral, then, the stress stiffening matrix $[K_s]$, which should be added to the default element stiffness matrix, can be derived. Equating strain energies:

$$\frac{1}{2} \{d\}^T [K_s] \{d\} = \int_0^L \frac{P}{2} \theta^2 dx = \frac{1}{2} \{d\}^T [G]^T P [G] \{d\}$$

Since:

$$\theta = [N]\{d\}, \text{ and}$$

$$\theta' = [G]\{d\}$$

Where:

$\{d\}$ = vector of elemental displacements

$[G]$ = first derivative (with respect to x) of elemental shape function matrix $[N]$, with on diagonal entries of:

$$\begin{aligned} G_1 &= 0 \\ G_2 &= -6x/L^2 + 6x^2/L^3 \\ G_3 &= 1 - 4x/L + 3x^2/L^2 \\ G_4 &= 0 \\ G_5 &= 6x/L^2 - 6x^2/L^3 \\ G_6 &= -2x/L + 3x^2/L^2 \end{aligned}$$

$[N]$ = elemental shape matrix, with on diagonal entries of:

$$\begin{aligned} N_1 &= 0 \\ N_2 &= 1 - 3x^2/L^2 + 2x^3/L^3 \\ N_3 &= x - 2x^2/L + x^3/L^2 \\ N_4 &= 0 \\ N_5 &= 3x^2/L^2 - 2x^3/L^3 \\ N_6 &= -x^2/L + x^3/L^2 \end{aligned}$$

Carrying through the matrix multiplication and the integration, the additional stiffness due to axial force is found to be:

$$[K_s] = \frac{P}{30L} \begin{pmatrix} 36 & & & & & \\ 3L & 4L^2 & & & & \\ 0 & 0 & 0 & & & \\ -36 & -3L & 0 & 36 & & \\ 3L & -L^2 & 0 & -3L & 4L^2 & \\ & & & & & \end{pmatrix}$$

Note that the upper triangular portion of the matrix is symmetric about the main diagonal.

Activating pressure stiffening of straight pipes in **CAESAR II** simply applies the stress stiffening matrix to the elemental stiffness matrices (of straight pipes only), using an axial force P equal to the internal pressure (user selectable P1 or P2) times the internal area of the pipe. Note that other internal forces (due to thermal or imposed mechanical loads) are not included in the P force — this is not a non-linear effect. This option is activated by entering 1 (to use pressure value P1) or 2 (for pressure value P2) on the **Use Stress Stiffening due to Pressure** entry under **Special Execution Parameters** of the **Kaux** menu.

NOZZL3 — This model is identical to the original, except that pressure strain effects (often called the Bourdon effect) have been included. This effect considers the elongation of pipes under pressure tensile stresses as a loading condition. For example, the piping codes generally require that pressure be considered as having not a strain component, but only a stress component, which should be simply added to the calculated sustained stresses:

$$S_{sus} = PD_i^2 / (D_o^2 - D_i^2) + iM/Z$$

Where:

S_{sus} = total sustained stresses

P = internal pressure

D_i = internal diameter of pipe

D_o = external diameter of pipe

i = stress intensification factor

M = moment on cross-section due to non-pressure sustained loads

Z = section modulus of pipe

For straight pipes, the pressure strain can be calculated in a straight-forward manner. Longitudinal pressure stresses cause longitudinal strains (approximately equal to $Pr_i/2tE$), which are reduced by the Poisson contraction due to the hoop and radial stress (approximately equal to $-2\nu Pr_i/2tE$), for a net longitudinal strain of:

$$\epsilon = (1 - 2\nu) Pr_i/2tE$$

Where:

- ϵ = longitudinal pipe strain due to pressure
- ν = poisson's ratio
- P = internal pressure
- r_i = internal radius of pipe
- t = pipe wall thickness
- E = modulus of elasticity of pipe material

For elbows, the calculation of the end displacements is more complex. Assuming that the pressure strain is constant throughout the length of the bend, the bend end distortion can be modeled similarly to that for uniform thermal growth. From Roark and Young's Formulas for Stress and Strain (5th Edition), the end displacement and rotation of a curved beam under uniform strain is given by:

$$d = 2\epsilon R \sin(\theta/2) = (1 - 2\nu) Pr_i R \sin(\theta/2)/tE$$

$$\phi = 0$$

Where:

- d = displacement of bend (directionally along line through ends)
- R = bend radius
- θ = included angle of bend, radians
- ϕ = change in angle of bend

These pressure strains can be incorporated into the analysis (and were, for analysis NOZZL3) by entering 1 on the **Activate Bourdon Pressure Effects** entry under **Special Execution Parameters** of the **Kaux** menu.

NOZZL4 — This model is similar to case **NOZZL3** described above, except that the pressure strain effects have been applied in a slightly different way — using the “true” Bourdon effect. A number of sources, such as MEC-21 (the

original Mare Island piping flexibility program written in 1959) and Crocker & King's Piping Handbook, 5th Edition (pages 4-37 through 4-38) alternatively view pipe bends as behaving similarly to Bourdon tubes — hence the popular name Bourdon effect for pressure elongation (note that pressure strain of straight pipes is calculated the same under both methods).

Bourdon tubes are flattened tubes of steel bent into a circle, which tend to straighten out when pressurized. This introduces an angular change in the tube (or, accordingly, the pipe bend) as the tube opens up under pressure. (It is COADE's contention that the Bourdon tube straightens as a consequence of the flattened tube trying to develop a round cross-section, so this effect should not really apply to a forged pipe elbow, which already has a truly round cross-section. It may have some validity when discussing bends which have been bent from straight pipe.)

MEC-21's derivation of the end rotation and displacement strains of a pipe elbow under pressure are shown below:

$$\phi = \frac{C P r_i^4 [1 - 2\nu + a - 1.5 \nu t^2 / R^2]}{2 E I}$$

$$a = \frac{b R}{\theta \sin \theta}$$

$$c = \frac{b R}{\theta \cos \theta} - 1$$

Where:

- a = displacement in direction a (as per Figure 15)
- c = displacement in direction c (as per Figure 15)

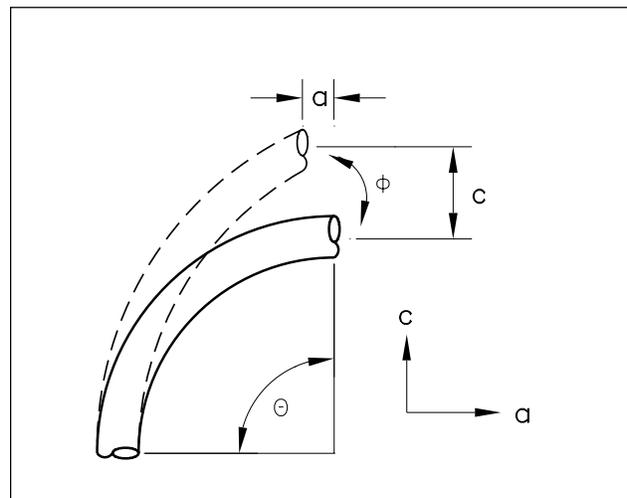


Figure 15

The full Bourdon effect, considering the rotational effect of the elbows as per MEC-21 was incorporated into this case by entering 2 on the **Activate Bourdon Pressure Effects** entry under **Special Execution Parameters** of the **Kaux** menu.

NOZZL5 — This model contained more than one of the effects described above — pressure stiffening of elbows, pressure stiffening of straight pipes, and Bourdon effect number 2. This would probably be a better model of reality in that these effects would not occur in isolation.

NOZZL6 — This model was identical to the original case, except that stiffnesses of 1E6 lb/in and 1E6 in-lb/deg were used for the translational and rotational stiffness respectively at the compressor connection. This is not necessarily a representative value; however, it is certain that the compressor connection is not infinitely rigid, and this case can be used to give an indication of how much the results may be affected by modeling the compressor stiffness. The true measure of compressor rigidity may be estimated through finite element analysis, test, manufacturer's documentation, or other means.

RESULTS AND CONCLUSION: The maximum and minimum values were tabulated for the six components of the compressor nozzle load, as well as the maximum Operating, Sustained, and Expansion stress of any point in the system for the seven versions of the model. These results are shown in Table 1.

Case	Compressor Nozzle Loads (OPE Case)						Max System Stress		
	FX	FY	FZ	MX	MY	MZ	OPE	SUS	EXP
	(lb)			(ft-lb)			(psi)		
NOZZLE	1185	-7745	-15025	-30279	220575	-2697	9538	7679	3090
NOZZL1	1107	-7777	13876	-30519	201406	2683	9750	7900	3251
NOZZL2	1415	-7923	16725	-29702	194005	2483	9376	7662	2964
NOZZL3	1734	-7751	21974	-30320	322520	2687	11029	8050	3090
NOZZL4	2028	-7753	25834	-30340	380473	2683	11873	8878	3090
NOZZL5	2398	-7927	28396	-29734	335800	2475	11605	8746	2964
MAX MAG	2398	-7927	28396	-30519	380473	2697	11873	8878	3251
MIN MAG	1107	-7745	13876	-29702	194005	2475	9378	7662	2964
% VAR	117%	2%	105%	3%	96%	9%	27%	16%	10%
NOZZL6	1859	-6424	12094	-45	14394	100	9366	7718	2835
% VAR	32%	23%	141%	67820%	2711%	2724%	27%	15%	15%

Table 1

Considering only models NOZZLE through NOZZL5, some nozzle load components were seen to vary (from the minimum to the maximum) by such large numbers as 117% (FX), 105% (FZ), and 96% (MY). Unexpectedly, the ranges between the minimum and maximum code stresses were much smaller, with differences of only 27% (Operating), 16% (Sustained), and 10% (Expansion).

In case NOZZL6, where the nozzle flexibilities are introduced, the results are even more striking. Selected components of the maximum tabulated nozzle loads were higher by 141% (FZ), 67,820% (MX), 2711% (MY), and 2724% (MZ) vs. the results with this model. Meanwhile, the stresses vary by only 27%, 15%, and 15% for the Operating, Sustained, and Expansion cases respectively. Even though this may not be a fully realistic case, this effectively represents the impact that modeling of flexibilities may have on nozzle loads, without necessarily providing a corresponding effect on the stress calculations.

Therefore it is clear that nozzle load calculations, which should be independent of piping code used (and its associated modeling conventions), may be more sensitive to modeling variations than code stress calculations would be. For this reason, the engineer may be interested in performing more than one analysis in order to determine the extremes of loading which the equipment may be expected to endure.

CAESAR II Specifications

Listed below are those bugs/errors/omissions in the **CAESAR II** program that have been identified since the last newsletter. These items are listed in two classes. Class 1 errors are problems or anomalies that might lead to the generation of erroneous results. Class 2 errors are general problems that may result in confusion or an abort condition, but do not cause erroneous results.

Class 1

- 1) **Piping Element Generator:** A restraint direction cosine tolerance error has been discovered in Version 3.19. *This error only affects skewed restraints with direction cosines greater than 0.9998 (angles less than 1.146 degrees).*

The original tolerances were set at 0.999 (2.56 degrees), and at user requests were changed in Version 3.19 to 0.99999 (0.256 degrees). The limits in the Element Generator (of 0.9998) were overlooked. This error has been corrected in Version 3.2.

- 2) Piping Error Checker: An error has been discovered in the routine which combines models for the "large job include". This error caused some of the allowable stress data, *on the first element of the included job*, to be lost. This error has been corrected in Version 3.2.

This error only affects the first element of the included job if allowable stresses, different from the preceding job, were entered.

Class 2

- 1) Static Output Module: A labeling error was discovered with regard to the Basic Engineering spring hangers. Figure BE400 was incorrectly labeled BE404. This error has been corrected for Version 3.2.
- 2) Static Output Module: Version 3.19 provided access to the intermediate data generated by the error checker (flexibility factors, expansion coefficients, minimum wall thickness, etc.) when an input echo was requested. This data was intended to be available when the output was directed to either the printer or a disk file. Unfortunately access to the printer for this data was omitted from this version. This oversight has been corrected in Version 3.2.
- 3) Batch Stream Processor: The Batch Stream processor provided in Version 3.19 inadvertently deleted an intermediate job file upon completion. This file (the _1 file) is necessary for both static and dynamic animation graphics. This problem has been corrected in Version 3.2.
- 4) In **CAESAR II** Version 3.20 an error has been discovered in the presentation of the Spring Hanger Data spreadsheet. The text data for this spreadsheet is acquired from the file SCREEN.TXT, located in the installation directory. For this particular spreadsheet, the units labels were actually part of the text, instead of variable. **THIS IS A TEXT ERROR ONLY, IT DOES NOT AFFECT THE OPERATION OF THE PROGRAM.**

Instructions for correcting this problem accompanied the distribution of the disks. Alternatively, users can down-load a correction file (F320A.ZIP) from the COADE bulletin board.

- 5) In **CAESAR II** Version 3.20 an error has been discovered in the Equipment program for both the API-610 and API-617 modules. This error could cause a program abort, and return the user to the main menu.

Users can obtain a corrected version of the Equipment program by down loading the file F320A.ZIP from the COADE bulletin board.

- 6) Main Menu Module: A problem has been discovered in the COADE file manager when attempting to copy files. If the "destination path" is left blank (i.e. not specified) the program will abort when the copy process is initiated. Future versions of the file manager will trap this error condition and report it to the user.

CAESAR II Technical Note

CAESAR II Version 3.20 includes a change to the equation for computing the B31.8 Sustained Allowable Stress. Prior to Version 3.20, **CAESAR II** computed this allowable as $0.75 * S * F * T$, in accordance with codes prior to the 1990 addendum. Version 3.20 utilizes the new code equation $0.75 * S$. This new equation effectively provides an increase in the allowable of at least 28%.