Modeling Victaulic Couplings in CAESAR II

By: David Diehl

Introduction

Over the years I fielded many modeling questions on the Victaulic Coupling. In most cases, since I had no direct knowledge of the joint, I asked the **CAESAR II** user what it was they were trying to do, what was important to them and what they knew about its characteristics. In most cases, once the basic modeling concept was conveyed, the user could move on to complete the input without further assistance. It wasn't until last year at a trade show – the Power Gen Show in Dallas – that I got the chance to see the joint, talk to the people in the Victaulic booth and generate my own opinions about modeling this joint in **CAESAR II**. After reviewing their catalog and speaking to their engineering group in Pennsylvania, I decided to write this article. Even if you do not use these joints, the concepts covered here might improve your **CAESAR II** models.

The Victaulic booth at the show featured the coupling itself. They were running a contest to see who could assemble the joint the fastest. That's the major feature of this pipe joining assembly – it's quick and easy. The pipe or pipe component has a circumferential groove near the connection end. A unique gasket sits over and across the gap between the two ends to be joined. The final piece of this assembly, the housing, holds the gasket in place and engages the two components with a ridge seated in each of the two grooves. See Figure 1. Bolts hold the housing together to complete the assembly. Certainly not as rugged as many other joining methods but a Victaulic coupling is a feasible alternative for its range of temperatures and pressures.

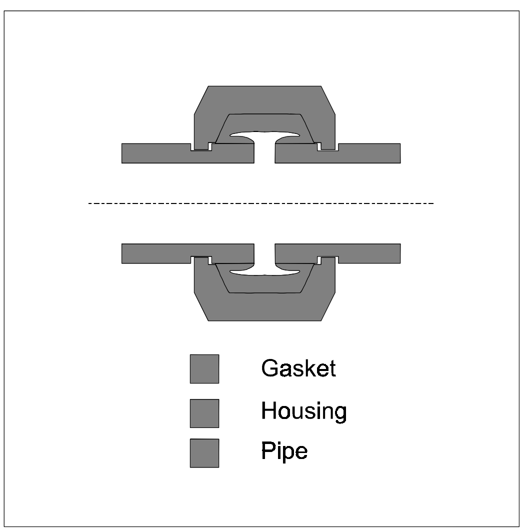

Figure 1

Again, this joint allows direct contact between the gasket and the pipe contents through the gap between the connected pieces. It's the pressure in the concave cross section in the gasket (the "C" shaped Victaulic gasket) that seals the joint. This gasket sets the limits on the applicable pressures and temperatures for the joint. Typically the working temperature range is -30 to 350 ºF and the maximum pressure of 1000 psi for pipe up to 6 inches but this rating drops to 250 psi for 24 inch pipe. (There is a safety factor of 3 in these pressure limits.) Many low temperature systems using these joints probably would not require formal analysis. But engineers have cause to evaluate these systems if they are connected to sensitive equipment or equipment that's subject to vibration. The reason is that Victaulic couplings can absorb thermal growth and vibration.

Victaulic produces two types of these couplings – the rigid system and the flexible system. There is no play in the rigid system but, as the name implies, the flexible system has an inherent "looseness". The ridges at the ends of the housing do not fill the full gap in the pipe grooves. The gasket seals the joint but there is a sloppiness in the coupling. This play can be very useful in many piping designs. Thermal strain, settlement and vibration can be absorbed in these joints thereby eliminating the need for added piping flexibility. The Victaulic catalog even shows an expansion joint composed of several of these couplings in series over a 30-inch length. **CAESAR II** users wishing to take advantage of this flexibility want to model it accurately.

The important characteristics of the coupling

The connected pieces can separate up to ¼ inch by using the space in these grooves. The pipe size and groove manufacture control this coupling performance. Grooves can be rolled or cut. (Rolled grooves have half the "freedom" of the cut grooves.) When designing systems with these couplings, Victaulic recommends reducing these "gaps" by 25 to 50% depending on pipe size. These grooves also allow the pipe to deflect off its axis. The table (for rolled grooves) shows 3.4 degrees for ¾-inch pipe down to 0.3 degrees for 24-inch pipe. These numbers may sound small but experienced **CAESAR II** users recognize the value of this flexibility in reducing pipe bending loads and stresses. How do we model this in **CAESAR II**?

The key in CAESAR II

Everyone wants to model the coupling as a special element in **CAESAR II** – perhaps an expansion joint or just a flimsy piece of pipe from one groove to the other. But that won't provide the play or gap in the joint. Gaps are a nonlinear effect. **CAESAR II** does not have nonlinear elements but it does have nonlinear restraints. This restraint definition can be used to model these couplings. Just as the pipe running between two nodes supplies the stiffness relationship between the nodes, so too, restraints can be defined in all six degrees of freedom (X, Y, Z, RX, RY &

RZ) to serve a similar purpose. The key is the restraint CNODE in **CAESAR II**. The NODE/CNODE restraint pair serves in place of the element FROM/TO pair. The stiffness fields for each of the six restraints are left blank to use the default (rigid) stiffness. The play in the coupling is modeled by specifying a GAP for the appropriate restraint(s). Since it's either axial deflection or off-axis angulation, pick the degree(s) of freedom that you believe will control the joint.

An example will help here – see Figure 2. A pipe runs from node 10 to node 20 and then another from node 20 to node 30. This piping runs in the X direction. Let's put a Victaulic coupling at node 20. Simply change the FROM node on the second pipe to 21 and define a set of six rigid restraints (X, Y, Z, RX, RY & RZ) at NODE 20 with a CNODE 21. As it stands now, this definition works just like the 10-20 and 20-30 model. What makes it a Victaulic Coupling is the additional GAP definition for the X restraint to model the axial play in the joint. (A GAP on the RY and RZ restraints would model the angulation instead.)

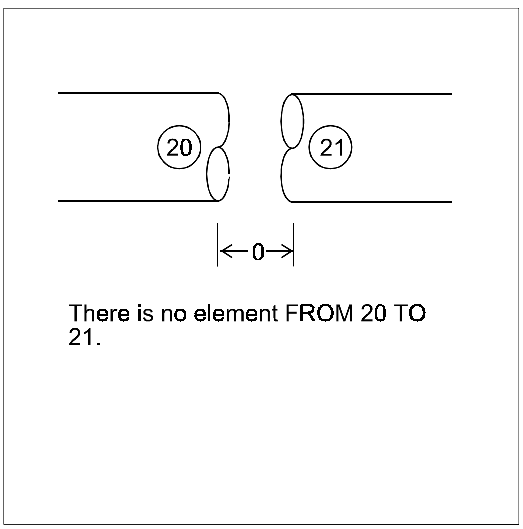

Figure 2

Several issues should be mentioned when using this simple approach. First of all, don't spend your valuable time entering data for each and every coupling. On the first pass code through the couplings and enter these NODE/CNODE restraints only at those locations where you think they may be "important". You can always go back and add more where you need them. Oftentimes we have (new) users call up about a job that will not converge* on a solution only to discover that they completely modeled every nonlinear effect available – one way supports, gaps, rod models and friction. Instead of moving to convergence, the iterative technique gets trapped in a loop, repeating a sequence of restraint changes until you abort the whole process. We tell them that a more complicated model does not necessarily produce a "better" model. You will finish your work faster by starting with a simpler model and improving it when and where the system results dictate.

Except for non-convergence, **CAESAR II** will always produce results. They may not be sensible but numbers will always be generated. You must check your modeling assumptions especially when nonlinear conditions (here, gaps) are present. Make sure the pipe moves in the right direction. Check the loads at those points where the gap was ignored, if they're high, re-run the job with the gap added. If you put in an axial gap that doesn't close and there's a large bending moment, you may want to replace the axial gap with a bending "gap" and vice versa. This means that when you first view results you are checking your modeling assumptions. If your assumptions prove to be correct, then you can check the system results. If your simplifying assumptions prove wrong, then take the time to adjust your model. This strategy will save time in the long run.

Remember, too, that the **CAESAR II** gap is both positive and negative – a 1/8 inch gap on an X restraint between nodes 20 and 21 means that node 20 can move up to 1/8 inch in X either towards or away from 21. If you wish to get fancier, break the X restraint into a +X restraint and a –X restraint and put the appropriate gap on one or both. Don't forget that the Victaulic gap table (on page 9 of the catalog) lists gaps for rolled grooves; cut grooves have double the movement. Also, Victaulic recommends reducing these movements (by 25 to 50%) when designing your systems. They, too, recognize that more accurate data for these little bits does not necessarily produce a better analysis overall.

One benefit to this NODE/CNODE model is that the loads across the coupling show up in the restraint report. In the Restraint Summary, all six restraints at the node show up on a single line. These NODE/CNODE combinations are quite useful in other applications as well. If you model through a pump and wish to itemize the nozzle loads, just specify a single anchor restraint between a NODE/CNODE pair at the flange. Be sure to list the node on the pump side as the CNODE so that the output report shows the proper signs – the piping loads on the pump and not the pump loads on the piping.

A finer model

The modeling technique described to this point will do a good job for a majority of piping systems but there is always a better model. Other considerations were addressed in my discussion with the Victaulic representative. First of all, the gasket will add no stiffness to the connection so no load is required to compress or open the gasket. A simple free-or-fixed gap model is sufficient (**CAESAR II**'s bilinear restraint stiffness is not necessary). The joint may not have gasket stiffness but the joint will require some load to start it moving. On a good connection, a line pressure of 15 psi will pull the pipe out to the limits of the coupling. You could say that pressure thrust must overcome the joint friction to start moving through the gap†. This pressure thrust in the joint is not included in **CAESAR II** models. (Except for expansion joint models and the explicit inclusion of bourdon effects, pipe deflection due to pressure is not considered in **CAESAR II**.) If pressure "pops" the joint, then the thermal expansion will have the full gap available for expansion. To take advantage of this extra gap, you can include the pressure thrust by adding a force on either end of the joint pointing away from the joint. See Figure 3. The magnitude of the thrust force is simply the pressure times the inside area of the pipe. Friction, as such, is not included in this model but there may be reasons to account for it. If at least one gap is not closed along a straight run, there will be no axial load calculated along that run, not even the inherent friction load. If this run connects to a piece of sensitive equipment, the friction load necessary to close these joints should be included "by hand", that is, add a force directly to the nozzle. If the pipe is out of round, these friction loads may be higher.

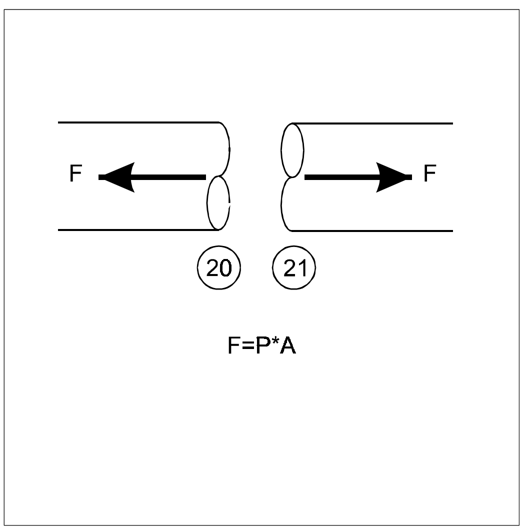

Figure 3

Let's review one additional model adjustment. The coupling model to this point isolates the axial gap from the bending gap. I suggest using one or the other. But in fact, these two terms, axial separation and off-axis angulation are not independent. At the limits, a joint that fully extends axially will be rigid in bending and a joint that takes all the bending cannot extend. "But what about the middle ground?" I asked the Victaulic rep. He said if I wanted to combine both terms I should be sure to keep the sum of the actual to allowable ratios less than 1 (axial displacement / axial gap + bending / bending gap < 1). So, if you run the simple model for axial movement but you have a relatively large bending moment, you can include a rotational gap set to the pro-rated limit. As you might imagine, several iterations may be required to settle the model down. Such iteration may improve the analytical model but I am unsure of its value in the overall analysis.

Other considerations

Keep in mind that the groove in the pipe may have a reduced wall thickness. While the roll groove removes no metal from the pipe (the entire cross section follows the groove), the cut groove removes metal. This reduction in wall thickness should be considered as an allowance in the required wall thickness. It's not a deep groove though; in fact it's less than the depth required in threaded connections. Check this in thin wall and high-pressure piping.

One concerned user called to ask about flexibility and stress implications of the joint, particularly on bends. Appendix D of the B31.3 piping code shows adjustments for bends with flanges. The flanges addressed by the code serve as stiffening rings for the bend and prevent the bend from ovalizing under a bending load. Bend ovalization reduces the moment of inertia of the cross section and this, in turn, reduces the element stiffness while at the same time the stress increases. The flanges therefore decrease the bend flexibility and reduce the stress intensification factors. Do these Victaulic couplings stiffen the bend? I think not. Does the groove alone call for an increase in the stress intensification factor (SIF)? B31.3 shows an SIF of 2.3 for threaded pipe joints; but how similar is the groove to a threaded joint? It's probably better to focus on the overall connection to answer this question. A threaded connection is a rigid, load-bearing joint; the Victaulic coupling is not. I don't think the coupling should have an SIF of 2.3 but it probably wouldn't hurt to specify it as such. If the point is "overstressed" (on paper) it's probably overloaded. The high stress in the output would serve as a reminder to take a closer look. If additional consideration indicates that the joint is OK, rerun without the SIF so the final report doesn't cause others to ask the question you just answered.

Get the big picture

The main point of this review is to properly model a Victaulic coupling. On the way, though, we also touched on several aspects of general modeling and CAESAR II specifics. Hopefully this information will be useful even if you do not use these fittings. Keep in mind that CAESAR II is a system analysis tool not a local analysis package. Yes, we can check shell stresses, attached equipment and flanges but the focus is on the system loads and deflections and the pipe stresses that result. You would be fooling yourself to push for the accuracy found in today's finite element analysis but, then again, it would be a waste of modeling time and computing resources to use finite element software for **system analysis.**

With this big picture in mind, you can see why I suggest a simple model for the initial pass and save the fancy modeling for situations that deserve it. That implies (and rightly so) that the results should be checked not for only high stress and high load but also for the soundness of the input. Clearly those stress and load results are invalid if the model is incorrect. So check those results to confirm the model first and refine the model where and when it's significant to your analysis. Fine tuning a model, where unnecessary, is a waste of time. You use these tools to save time.

* **CAESAR II** uses an iterative method to determine whether or not a nonlinear restraint is active. If the restraint is active, its stiffness is included in the analysis; if it's not active, the stiffness is not included. After running the load case, the program will test each nonlinear restraint to see if the linear assumption was correct. All incorrect "guesses" are updated and the load case is re-analyzed. Iteration continues until convergence, until all these restraint assumptions are met.

† If you estimate the friction force as 15 psi times the inside area of the pipe, you can compare the axial load at nodes where the couplings were not modeled to this value. If the piping loads are much higher than this friction force and you need the flexibility, you would benefit by including this coupling in the model.

Genesis of PDMS to CAESAR II Transfer Tool

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Synopsis

The stress analysis group of SHEDDEN UHDE Pty. Ltd. has achieved a simple method of electronically transferring piping configuration data from PDMS to **CAESAR II**. This process allows large gains in productivity, elimination of modeling errors and improved understanding between piping design and stress analysis engineers.

Introduction

In past years, piping design has been divided between the layout designers and stress analysis engineers. With the proliferation of new generation software, these two groups can be more closely interrelated, resulting in a dramatic improvement in overall design efficiency. A solid understanding of the preferences and limitations of other engineering disciplines, and to a large extent improved bi-directional communication, enhances this improvement.

One of the new generation of software packages that allows this possibility to happen is **CADWorx/PIPE**. It is the first CAD software to feature fully bi-directional interfacing capabilities with the analytical package **CAESAR II**. Although **CADWorx/PIPE** has wide applicability, it has been employed by SHEDDEN UHDE in the past 12 months with the primary aim of :

- Creating 3D isometric piping models for stress analysis purposes,
- Generating CAD drawings that include stress analysis results information from **CAESAR II** – Stress ISO and Multiple ISO features,
- Automatic production of piping fabrication isometric drawings – Auto Isometric feature.

After an extensive "on the job" testing period, **CADWorx/PIPE** has become an important modeling and report-generating tool. The significant benefits it brought have been appreciated not only by stress analysis engineers, but also by piping layout designers and engineers from other disciplines.