The main objections to this tutorial are the inefficiencies. The easiest to see is that the parts have at least half symmetry about the horizontal plane (if gravity is neglected). Such symmetry drastically reduces the size of the non-linear by eliminating half the volume of nodes. The symmetric displacement boundary conditions eliminate a few more unknowns.

A less obvious inefficiency is the definition of the no penetration contact surface areas. Every node on the first contact surface is compared to every node of the second contact surface to determine if it penetrates the second surface. Thus, it drastically speeds the trial and error non-linear solution when engineering judgement is used to limit the size of both contact surfaces. Instead, in this tutorial the sizes of both surfaces were maximized by picking the total of the internal surfaces of the holder and the total of the external surface of the pipe segment, as shown in the contact visualization tool in Figure 2. To efficiently define contact surfaces the parts have to be modified with split lines to define small surface regions, and engineering judgement must be used to estimate the range of motion where contact is possible.

Figure 3 shows the initial and final contact (tangent) points where an imposed displacement (fixture) of 2.15 inches will assure a valid contact analysis. That figure also shows how much of the two surfaces need to be selected by using split
lines. For the pipe the contact area is the circumferential part seen in Figure 3 in conjunction with the width of the holder. The new split area of the pipe is given in Figure 4, along with the shorter portion of the split inner holder surface. Using half symmetry and reducing the surface search area led to a much faster non-linear run time. Two images of the true scale deformed shapes are given in Figure 5, and they show that the model is incomplete. The pipe end sections have been treated as free surfaces; yet they are supported by the omitted portions of the pipe. In other words, the deformed end of the pipe is missing the significant structural support provided by the actual pipe length that has been omitted. The extended pipe mesh can have large elements since the stiffness there is important, but the stresses are not.

Another misleading aspect of this tutorial is the presentation of a single plot of von Mises equivalent stress repeated in Figure 6. The assembly has two materials with significantly different materials which always calls for at least two stress plots with each scaled to a failure level for each material. The stress plot, with the mesh displayed, shows that the
mesh was not acceptable for accurate stresses because the pipe section had only a single element through the wall thickness and one element cannot correctly model the bending stresses. For elastic bending stresses at least three elements should be present through the thickness in regions of high bending (high changes in curvature).

Therefore, a new quarter symmetry model was formed by removing material in the axial direction of the pipe from the center of the contact area. In addition, the pipe was extended one-and-a-half diameters away from the original model as shown in Figure 7. In addition, the pipe enforced displacements was changed. The line condition implied that there was a rigid object attached to and moving the pipe. However, that prevented any possible bending in the x-direction. The new enforced displacement was applied at the extended end, like the pipe being moved by hand.

Figure 8 shows the vertical (y-) displacement contours that verify that the extended far end of the pipe is remaining circular during the process, which is better that the results in Figure 5 that were obtained with a too short pipe.
Figure 7 Quarter symmetry model

Figure 8 Vertical pipe displacements (at step 5)