

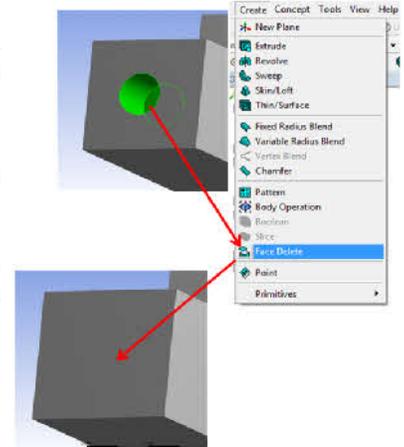
## Face/Off: Finally Having Fun with Slivers

By Doug Oatis



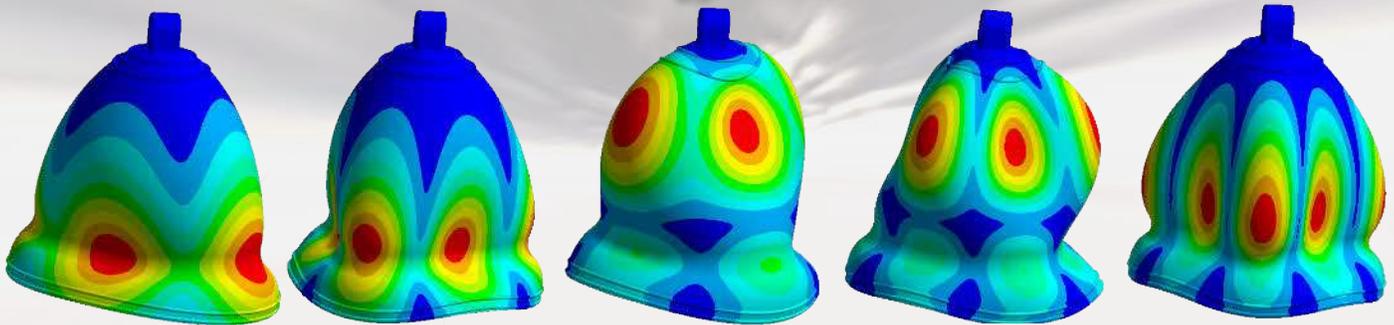
Slivers. That one word can send an analyst into the fetal position. Anyone who's ever worked with problematic geometry, whether it's CAD-program based (not going to name names) or a result of surface lofting/compound fillets/etc, can tell you a large portion of their time is spent fixing the geometry.

In ANSYS Classic, there were really only two fixes. The first involved using MESH200's to seed mesh the offending face to help out the advancing front volume mesher. The second involved some tedious bottom-up modeling that required you to rebuild the lines/areas to eliminate the face. If you had access to ICEM/AI\* Environment, you could use an octree mesher to walk over small details. Now, if there was only an easy way to eliminate the face entirely, and let the CAD engine heal everything back together.

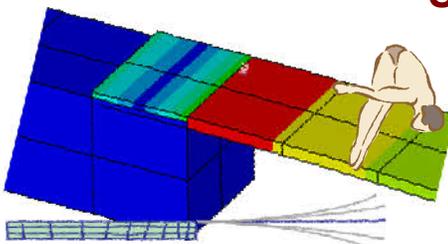


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## Season's Greetings from The Focus!



## Shell-Solid Interface Modeling: Further Studies



By: Dr. K. S. Raghavan, Infotech Enterprises Limited, Hyderabad, INDIA

Editors note: We are very pleased to have a user contribute to this months focus with his observations on connecting Shells and Solids.

### Introduction

This technical note is in continuation of two excellent articles written by Rod Scholl that appeared in August and October issues of FOCUS. Results presented herein will supplement those from the cited articles. The main emphasis of the present note is on configurations wherein a part of the structure can be modeled using only shell elements and in the rest of the structure it is mandatory to deploy 3D solid elements. Typical real life situations include pressure vessel with mountings, and integral bladed disks.

Further, emphasis is on comparative evaluation of two of the three methods that are in vogue. They are constraint equation (CE) method and the painted shell (PS) elements approach. The

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latter is also known as “overlaid shell” and “smeared shell” elements approach. The third method, which is the embedded shell, is not covered in this note as it has limitations of applicability. The main, or perhaps the only, concern of modeling shell solid interface is how to tackle mismatching degrees of freedom between solid 3D and shell elements. In CE approach we seek to impose Kirchhoff bending equation that relates in-plane displacements to bending slope. This is achieved using CERIG command of ANSYS. In PS approach shell elements are painted on the interface surface covering its actual area of coverage. The shell elements, thus added, provide rotational degrees of freedom on solid element nodes thereby ensuring DOF compatibility. In addition, as we will see later, they add unknown stiffness. It is to be said here that CE method has a sound theoretical basis, whereas there is no theoretical basis for the PS approach. Evaluations are made using two case study problems for static analysis and through comparison with full 3D solid FE model results. Applicability for dynamic and nonlinear analysis is also assessed.

**Case Study Problem 1**

This problem concerns a cantilevered plate of two distinct parts. The first part is trapezoidal having variable thickness. The other part is having uniform thickness. Its geometry is as in Figure 1.A which is also the full 3D solid finite element mesh. Figure 1.B is combined shell-solid element mesh.

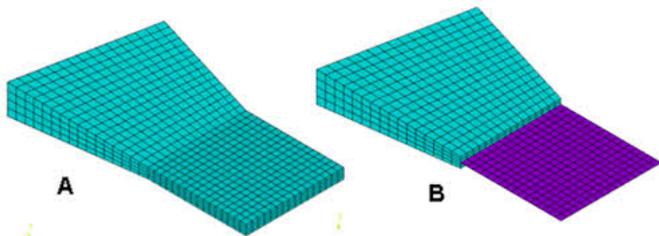


Figure 1: A: Solid Mesh B: Solid Shell Mesh

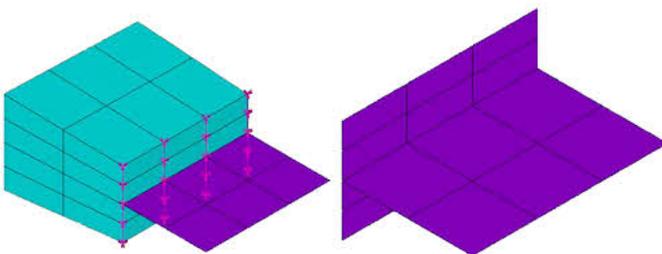


Figure 2: Interface Conditions for CE and PS Models

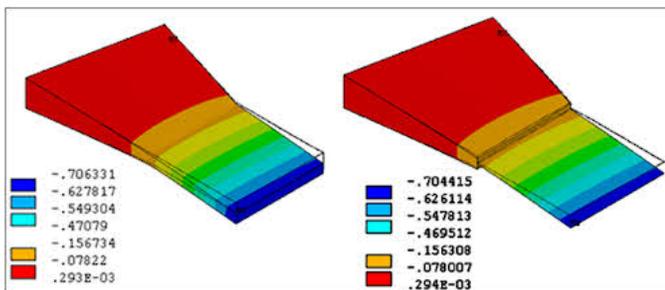


Figure 3: Transverse Displacement Plots

tions are shown for the two shell-solid models – 2.A for the CE model and 2.B for the PS model. Static analysis is carried out for uniform pressure applied on the top surface. Typical deformation patterns are as shown in Fig. 3.

Parametric studies are carried out by varying the thickness of the flat portion with other thickness dimensions varying proportionately. Results of the two shell-solid models are compared with exact results from full 3D solid model. Percentage error values are given in Table 1. For the shell PS model thickness of the painted shell elements is also varied, its ratio (t-ratio) to main plate thickness varying from 0.25 to 4.

As can be expected, CE model gives very accurate results for all the cases. The PS model behavior is very interesting. The overlaid shell elements are relatively ineffective for thickness ratio less than unity. For ratios more than equal to unity the model is giving good results. The deflection is exactly corresponding for thickness ratios of 2.5, 3.0 and 3.5 for the three cases considered. What is remark-

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**Table 1: Tip Deflection - Percentage Errors**

Model	t-ratio	Percent Error		
		t = 10	t=20	t=30
CE	n/a	-0.28	-0.3	-0.28
P.S.	0.25	17.79	35.43	52.83
	0.5	4.16	8.36	12.46
	1	1.17	2.5	3.82
	1.5	0.52	1.23	1.98
	2	0.2	0.64	1.13
	2.5	-0.01	0.25	0.57
	3	-0.17	0	0.28
	3.5	-0.28	-0.21	0
	4	-0.38	-0.38	-0.28

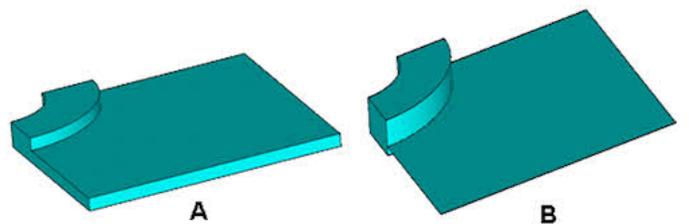


Figure 4: A: Solid Mesh B: Solid Shell Mesh

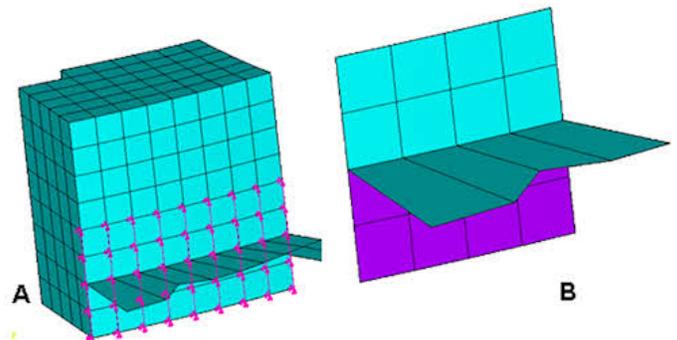


Figure 5: Interface Conditions for CE and PS Models

(Cont. on pg. 3)

(Shell-Solid cont...)

able is the observation that deflection trend changes very abruptly at thickness ratio of around unity.

**Table 2: Maximum Deflection - Percentage Errors**

Model	t - ratio	Percent Error			
		OD=250	OD=300	OD=350	OD=400
CE	n/a	0.9	0.63	0.63	0.73
P.S.	0.25	19	23.81	25.56	25.79
	0.5	6.11	7.76	8.37	8.52
	1	1.58	2.35	2.7	2.94
	1.5	0	0.77	1.17	1.4
	2	-1.13	-0.19	0.27	0.51
	2.5	-2.04	-0.92	-0.45	-0.15
	3	-2.78	-1.53	-0.99	-0.66
	3.5	-3.46	-2.09	-1.53	-1.18
4	-4.07	-2.6	-1.98	-1.62	

**Table 3: Problem 1 Modal Results**

Taper-Flat Plate - Modal Analysis t = 30 , t_ratio = 2					
Mode	3D Solid	CE	PS	%err CE	%err PS
1	154.9	155.1	153.9	0.1	-0.6
2	450.4	450.1	459.7	-0.1	2.1
3	654	654	652.2	0	-0.3
4	858.9	859.3	857.3	0	-0.2
5	1278.3	1277.3	1277.6	-0.1	-0.1
6	1507	1506	1502.4	-0.1	-0.3
7	1913.6	1910.1	1930.2	-0.2	0.9
8	2118.2	2114.5	2130.7	-0.2	0.6
9	2443.4	2445.4	2439.3	0.1	-0.2
10	2580.1	2581.4	2574	0.1	-0.2
11	2801.5	2799	2790.3	-0.1	-0.4
12	2899.7	2895.2	2978.4	-0.2	2.7

**Table 4: Problem 2 Modal Results**

Square plate with circ. Boss, modal , OD = 300 , t_ratio = 2					
Mode	3D Solid	CE	PS	%err CE	%err PS
1	186.3	185.4	185.5	-0.5	-0.5
2	351.5	349.8	349.4	-0.5	-0.6
3	351.5	349.8	349.4	-0.5	-0.6
4	553.5	550.6	555.7	-0.5	0.4
5	678.9	675.6	683.3	-0.5	0.7
6	723.6	719.1	720.9	-0.6	-0.4
7	814	809	810.8	-0.6	-0.4
8	814	809.4	811.1	-0.6	-0.4
9	1036.1	1029.9	1031.4	-0.6	-0.5
10	1036.3	1030.2	1031.8	-0.6	-0.4
11	1121.9	1114.3	1116.9	-0.7	-0.4
12	1173.8	1166.6	1168.4	-0.6	-0.5

**Case Study Problem 2**

For this case study a square clamped plate with circular stiffened cutout is considered. A quarter of the model is shown in Fig. 4. Fig 4.A is the full 3D solid model and 4.B is the Shell – Solid model. Interface modeling details are shown in Fig. 5 for CE and PS models.

Parametric studies similar to those for the earlier case study are carried out. Outer diameter of the stiffener (boss) was varied as parameter. Results of the study are presented in Table 2.

The results are seen to be qualitatively similar to those for the first case study. Results of Tables 1 and 2 are combined and shown plotted in Fig. 6. In this graph X-axis represents the thickness ratios (columns 2 I tables 1 and 2). Y-axis corresponds to percentage error for various parametric variants. From this figure it appears that a thickness ratio anywhere between 1.5 and 2.0 would give acceptable results with PS model. The graph of Fig. 6 is perhaps the main contribution of the present study. However, it should not be overlooked that for both the problems CE model has

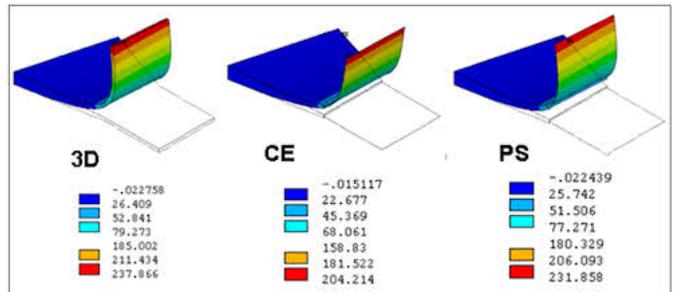


Figure 7: Deformation Plots

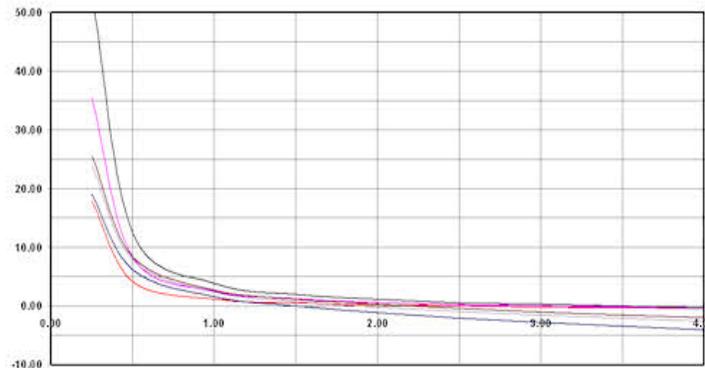


Figure 6: Thickness Ratio vs. Percentage Error

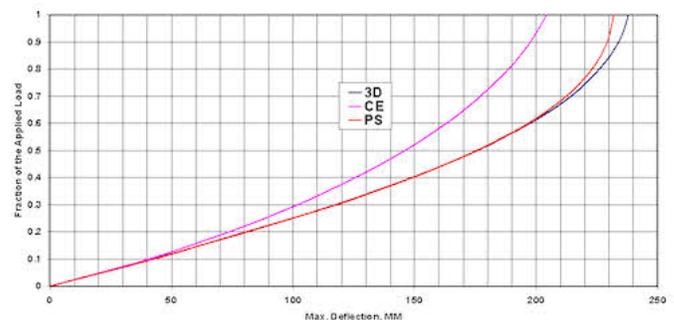


Figure 8: Load Deformation Curve

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(Shell-Solid cont...)

given very accurate results for the seven cases, the maximum error being 0.9 per cent.

### Dynamic Analysis

For both the models modal analyses were carried out and the first twelve natural frequencies were extracted. Results are presented in Tables 3 and 4.

In these studies a thickness ratio of 2.0 is chosen for PS model. The results are seen to be excellent. For CE model all the results are accurate within 1 per cent. The same is true with PS model also, with the exception of two modes (2 and 12) where the error is above 2 per cent. For the overlaid shell elements mass density is set to zero.

### Large Deformation Analysis

By changing the modulus of elasticity in Case Study Problem 1, sensitivities of models to large deformation effects are assessed. This is necessary as CERIG implicitly assumes small deformations. For full load deformed shapes are shown in Fig. 7.

As expected CERIG is stiffening the structure and as a result the deflection predicted is over 14 per cent less than the exact value. PS model, on the other hand, is predicting displacement within about 2.5 per cent. The load deformation curve for the three cases are shown plotted in Fig. 8. We observe that the PS model is accurate up to about 200 mm deflection. For this problem this is 20 times the thickness of the plate.

### Complex Geometries - Tetrahedral Elements

As might have been observed, the case study models are having simple geometries and could be map-meshed. This facilitated easy application of constraint equation using CERIG command with the help of a macro (Fig. 2A and 5A). Life will not be that simple always and it may be quite a challenge to apply CERIG if the geometry is complex or the mesh involves tetrahedron elements. In such case use of overlaid shell elements should become handy.

Figures 9 to 11 presents a case for which it is difficult to apply CERIG. This is solved using SOLID92 and SHELL93 elements. From Fig. 11 it is seen that the deflection predicted by PS model (thickness ratio = 2) is accurate within about 1 per cent.

### Summary

1. Shell – Solid interface modeled with overlaid (or painted) shell elements gives very good results for thickness ratio between 1.5 and 2. The results are comparable to those from CERIG model.

2. For large deformation problems it has been seen that CERIG model is not suitable and PS model is seen to be giving very good results.
3. A major factor in favor of PS model is that it is very easy to implement. Also if the geometry is complex enough to warrant use of free mesh and/or tetrahedrons it is difficult to use CERIG.
4. An important finding of this study is the nature of dependence of the accuracy of painted shell model on the thickness assigned to overlaid shell elements. It is seen that for thickness ratios less than about 1.0 the added shell elements are relatively ineffective. For values higher than about 1 they are effective. For thickness ratio more than about 1.0 the behavior of the models is seen to be almost steady. This interesting, and perhaps curious, behavior needs to be looked into further.

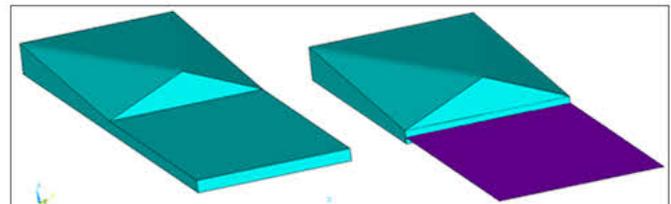


Figure 9: Tetrahedral Model

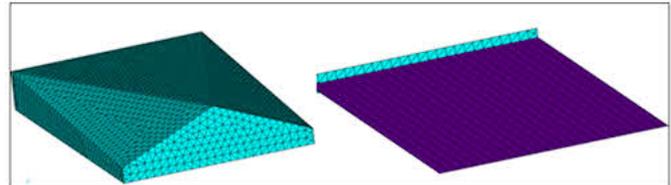


Figure 10: Complex Model

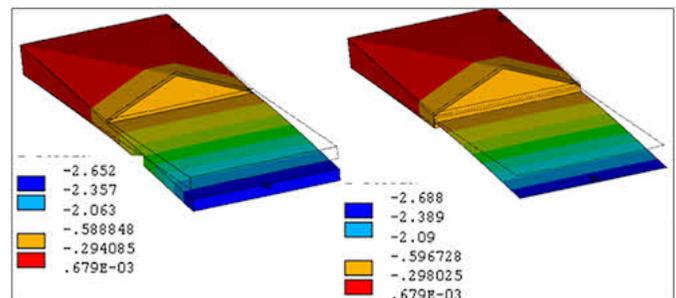


Figure 11: Deformation Plots

## A Further Further Look: Shell-Solid Moment Diagrams

By Rod Scholl

Between the previous article by Dr. K. S. Raghavan, and earlier articles [<link,link,link>](#) we've shown two different geometries and found different results. With the earlier geometry, we show a relatively close match between the approach using constraint equations (CE's) and painted shells. In this latest geometry, the CE's are far better than the painted shell method.

Lest we walk away with the conclusion "it just depends" maybe we can shed some light on which method is better with which geometries.

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(Moment Diagrams cont...)

I hate to do it... but let's pull out some shear force diagrams.

Geometry A, is a simple cantilever – and we find that the through thickness strain varies linearly. Because of this, we can create a rigid region along the red line and not impact the results! Thus we could create CE's at this location.

Geometry B is similar to Geometry A, where again we can identify locations where the creation of a rigid region does not impact the results (red line). The region with the blue line, has a curved strain through the thickness relationship, and we create error by enforcing a rigid region along the blue line. Thus our application of CE's at the interface between the flat to sloped region changes the shape function of the elements at this location, but the results are mostly unchanged.

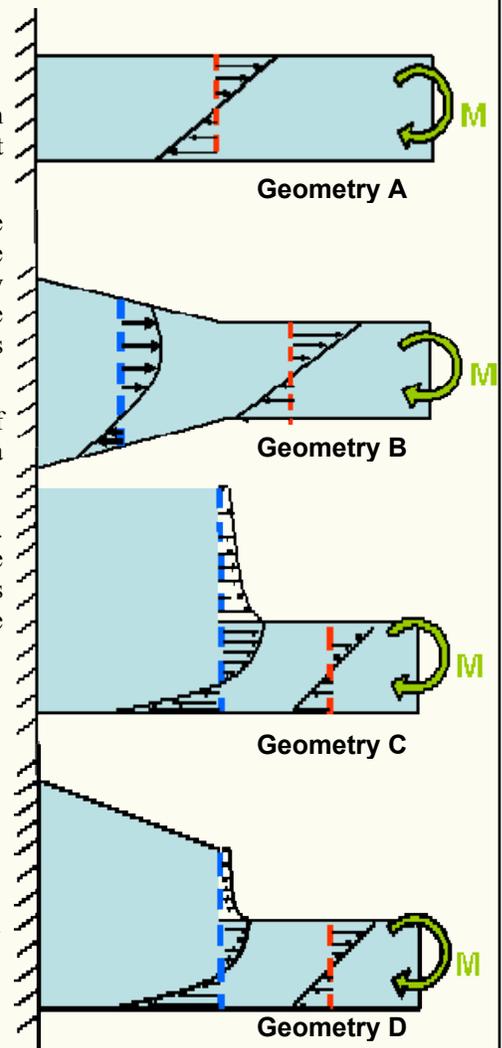
Geometry C, as explored before, presents a step function change in the moments of inertia, and we saw how the painted shells and constraint equations compared, with a slight advantage to the painted shell and contact methods over the CE'd joint.

Geometry D is similar to Geometry B, except we add a step change in the geometry. With this change, using constraint equations to create a rigid region at the blue line creates inaccuracy. In this particular case, the CE's under perform the painted shells as we see in the augmented summary table below. We also note that for choosing the

Geometry	T-ratio	%error t=10
D		
CE	n/a	4
Painted Shell	0.5	12
	0.75	3.9
	1	0.7
	1.5	1.8
	2	2.7

thickness of the painted shells, as used previously, a ratio of 1:1 to the thin region thickness produces the most accurate results.

So, with an appreciation for how CE's create a rigid region, and how some interfaces



(Face/Off cont...)

Fast forward to DesignModeler, and you can now say "I want to take his face...off" and not get looked at weird for quoting Nicolas Cage from "Face/Off". I know no-one can see this, but I'm currently typing in slow-motion, kind of like if John Woo had directed "Office Space".

The Face Delete tool is by far the most used operation I use in DesignModeler. Got a fillet/chamfer that you don't need, just delete it. Got a hole you don't want, no need to fill it, just select the faces and delete them. Got a sliver, you guessed it, just delete it. When you use the Face Delete tool in DesignModeler, you no longer need to maintain an 'analysis' and 'full-feature' CAD model. This means no more headaches troubleshooting CAD models that somehow use a .005-in fillet as a Datum for a future operation. No more rerouting/freezing/etc the CAD model just to simplify it for analysis. Just import the 'full-feature' CAD model into DesignModeler

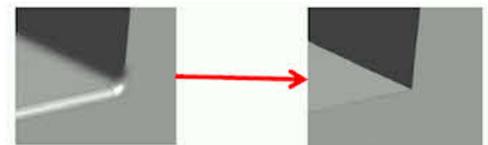
and delete the faces of the offending features. If the designer changes the part/assembly, simply update the DesignModeler session and all of your operations will be performed automatically (assuming no gross topological changes).

Now, when a face delete doesn't work, I've learned to look for the following:

1. Make sure ALL of the faces are selected for the face delete. Really zoom in to make sure there aren't slivers connecting your slivers to the rest of the geometry
2. Don't try to delete too many faces at once. I've run into this problem when deleting compound fillets between multiple surfaces.
3. Sometimes the delete operation is path dependant. Just because it didn't work on the first shot doesn't mean a combination of suggestion#2 and determination won't get it done

4. For very small sliver gaps, sometimes it easiest just to create a dummy extrusion, then delete the excess faces. No need to go cross-eyed trying to create a perfect surface loft (this is FEA...not brain surgery).

So the next time a nightmare assembly comes your way, remember that the Face Delete operation is a quick way to eliminate features the designer spent days on creating. I know you're expecting a closing quote from "Face/Off", but it turns out that all the good quotes are a little too risqué for a technical publication ...but if you like slow-motion gunfights with semi-far-fetched plots (as opposed to sped-up gunfights with plausible plots), I bet I can guess what your next rental's going to be.



# Commercial Tools Every Analyst Should Have

Compiled by Eric Miller

Last month we presented a list of free tools that PADT's engineers like to use to make their analysis job easier. We received so much positive feedback that we thought we would follow up with the next logical compilation - a list of commercial tools that we find useful.

This is not nearly as exciting because, well, it is not a list of free stuff. But most of the tools listed here definitely pay for themselves very quickly. There are not as many selections either, since for most of us the primary commercial tool we use is ANSYS, Workbench, CFX, FLUENT or some other ANSYS, Inc. product and we don't need to add a lot more.

Please take a look and, as with the free list, let us know if you feel we left something out.



## Microsoft Office 2007 Office Software [office.microsoft.com](http://office.microsoft.com)

It is time to get off the fence and take the plunge. Yes, Office 2007 is different. Yes you can't find anything in the menus when you start to use it. Yes, they changed something you have been using for maybe over 20 years. But many of us have been on 2008 for some time and we find it faster, more robust and much more capable. It is worth the pain of learning the new interface.

At \$300 to \$500 dollars a user this may be the best software value in Engineering. It is the dominant report creation tool (Word), the number one engineering calculation tool (Excel) and the de facto standard for creating presentations (PowerPoint). And they all work together and everyone can read what you have done. Quibble about the interface changes or how bill Gates is evil, the bottom line is that Office is the most efficient way to get your work done and where most ANSYS users spend their time when not in ANSYS.



## PaintShop Pro Image Editing [www.corel.com](http://www.corel.com)

Purists may demand Photoshop, and experts may find a difference, but around here we have found that the much less expensive PaintShop Pro works just as well and is a lot more intuitive to use. When you create images in your FEA/CFD program and grab

of the screen or save to disk, PSP has everything we have ever needed to edit, crop, and annotate our plots. It's vector annotation capabilities make jazzing up a results plot simple.



## Mathcad Engineering Math [www.ptc.com](http://www.ptc.com)

We all use Excel for engineering and know we shouldn't. When we get an equation that we no longer understand we grunt that "there has to be a better way" That way is Mathcad. Whether you are creating your own equations or using one of the many libraries that come with, doing "hand calc's" is easy and fast with Mathcad and when you are done you have a readable, printable report of your efforts.



## MATLAB Mathematical Tool Box [www.mathworks.com](http://www.mathworks.com)

It has always been difficult to describe exactly what MATLAB does. Probably the best description is that it is a complete scripting and automation environment for doing calculations, importing data, displaying information graphically and developing algorithms. More than python but less than C++ with a GUI built in. Those that use it, swear by it and use it for everything. It is certainly a fast way to create your own pre or post processing tools for ANSYS.



## Mathematica Symbolic Math [www.wolfram.com](http://www.wolfram.com)

Of all the "three M's" of calculation tools, Mathematica is the one the draws the most cult-like following. It started life as a symbolic math tool, but now it has many of the capabilities found in MATLAB, Mathcad, graphing tools, LaTeX and many other tools, all in one package. Those who know it claim it can do anything, and do it quickly and with intuition. PADT's users swear by it and often take days out of complex simulations by prototyping their approach or reducing their results in Mathematica.



## MoviePlus Video Editing [www.serif.com/movieplus](http://www.serif.com/movieplus)

There are a lot of commercial "easy-to-use" movie making/editing tools out there. We have found that most are made for your

grandmother and lack key functions. Others are too complex. MoviePlus walks that fine line between the two delivering power and ease-of-use. We find it very useful for editing and annotating simulation animations.

## WebEx/iLinc/GoToMeeting Web Conferencing Tools



[www.webex.com](http://www.webex.com)  
[www.ilinc.com](http://www.ilinc.com)  
[www.gotomeeting.com](http://www.gotomeeting.com)

We had to list three solutions here because we are in transition. For many years PADT, and ANSYS, Inc., have used WebEx as a very effective way to share presentations, host conference calls and take over each others desktops for technical support. WebEx works great and we still recommend it. We recently felt that they were gouging us a bit on price so we switched to two alternative tools: iLinc for the high end and GoToMeeting for the low. iLinc is very feature rich and has a lot of nice bells and whistles that make sharing applications and whole desktops easy and powerful. With other engineers we are really feeling that we can do things better and faster with iLinc. But, to be honest, some of our non-technical customers struggled with WebEx and iLinc. We found that GoToMeeting was the best solution for simple sharing of applications over the web. Our non-technical customers have no problem logging into a meeting and using the tools in the program.



## IronCAD Solid Modeler [www.ironcad.com](http://www.ironcad.com)

Most analysts work for a company with a good mid-range CAD or high-end PLM system. And we recommend that analysts use that system for any solid modeling they need to do. They also should have access to ANSYS's DesignModeler. But if you don't have such access, or you need to heavily modify geometry, we recommend IronCAD. There are a lot of good low end solid modeling tools out there, but for fixing/creating/modifying geometry our engineers prefer IronCAD. It allows you the flexibility to tweak things however you want and handle very complex assemblies with ease. It works well as a translator. It

(Cont. on pg. 7)

(tools cont...)

is a good investment for anyone who needs some extra solid modeling firepower.



### ICEM CFD Mesher

[www.ansys.com](http://www.ansys.com)

We have been looking at high-end meshers to supplement what ANSYS, Inc. provides for years. People tell us that this or that mesher is awesome. But once we get deep down into them we keep finding again and again that the ICEM CFD/AI\*Environment meshers from ANSYS, Inc. are the best. So we list it here and will pretend that it is not an ANSYS, Inc. product.



### AISBackup Data Backup Tool

[www.aiscl.co.uk](http://www.aiscl.co.uk)

There are a lot of backup utilities out there, and you company may have a standard. But if they don't we recommend AISBackup. It is affordable, simple and reliable.



### Textpad Text Editor

[www.textpad.com](http://www.textpad.com)

If you do a lot of text editing, and you are not a Vi junky, then you should be using textpad. It is fast, robust and very full featured including spell checking, tools for dealing with columns, and syntax highlighting. This is an awesome tool, especially for APDL.



### KaleidaGraph Scientific Graphing

[www.synergy.com](http://www.synergy.com)

We talked in the free software article about moving beyond Excel's weak scientific graphing by using GnuPlot. If you need professional graphs that are publishing quality, then Kaleidagraph is the right tool. It has everything you want and some, going beyond graphing to powerful statistics and data analysis tools.



### VMWare Virtual Machine Software

[www.vmware.com](http://www.vmware.com)

Want to have your cake and eat it to. Want to use a Mac or Linux box but need to run Windows? Need to run SolidWorks but you are on a Linux Box. You can do that with VMWare. It allows you to create virtual machine running another operating system on any machine. This is the best way to avoid buying a second machine to run Windows applications.



### CADFix Geometry Repair

[www.cadfix.com](http://www.cadfix.com)

We all would like to live in a world without bad geometry. We would all like to take a CAD model and read it into ANSYS or Workbench without any errors. But evil exists in the world and that is why we have CADFix. This is the only tool we have found that is purpose built to fix bad CAD geometry. Many times it is even automatic.

## Accessing Table Array Indices

By Ted Harris

One of the support questions we get asked from time to time here at PADT is, "How do I access my table array indices as parameters?" For those not familiar with table arrays in ANSYS, think of them as lookup tables such that you can interrogate the table via a set of indices and ANSYS will interpolate data within the table relative to those indices. As an example, you might set up a table array of pressure values vs. X and Y coordinates. With a table array, you input the indices as the 0th row and 0th column, as needed, and can then interpolate pressures at any X and Y coordinate values as long as they are within the X and Y range of the table.

Sometimes advanced users want to be able to access the indices of the table for one reason or another, and typically that means to store the indices as parameter values after the table has already been defined. Unlike a regular array in ANSYS, there is no APDL functionality to access the 0th row or 0th column and pull out the index values.

Those who have been in our ANSYS classes at PADT may be familiar with the saying, "With ANSYS, there is always a way." That holds true in this case as well, for there

is at least one way to obtain the indices as parameters. The way I have done it is to write the 0th row or column to a file with \*VWRITE and then read that file back into ANSYS as a regular array with \*VREAD. From the resulting array, one can pull out the indices of that 0th row or column.

A simple example is provided in the following macro:

```
!table_array.mac
!
!ANSYS macro demonstrating a way to
! obtain the first
! index in the zeroeth row of a table
! array as a scalar
! parameter.
!
!Ted Harris, PADT, 3/19/04

!create table array
!
*DIM,mytable,TABLE,3,5,1,time,x,
!*
*SET,MYTABLE(0,1,1) , 0.5
*SET,MYTABLE(0,2,1) , 0.7
*SET,MYTABLE(0,3,1) , 0.9
*SET,MYTABLE(0,4,1) , 1.1
*SET,MYTABLE(0,5,1) , 1.3
*SET,MYTABLE(1,0,1) , 0
*SET,MYTABLE(1,1,1) , 1
*SET,MYTABLE(1,2,1) , 2
*SET,MYTABLE(1,3,1) , 3
```

```
*SET,MYTABLE(1,4,1) , 4
*SET,MYTABLE(1,5,1) , 5
*SET,MYTABLE(2,0,1) , 5
*SET,MYTABLE(2,1,1) , 5
*SET,MYTABLE(2,2,1) , 15
*SET,MYTABLE(2,3,1) , 20
*SET,MYTABLE(2,4,1) , 25
*SET,MYTABLE(2,5,1) , 30
*SET,MYTABLE(3,0,1) , 10
*SET,MYTABLE(3,1,1) , 10
*SET,MYTABLE(3,2,1) , 20
*SET,MYTABLE(3,3,1) , 30
*SET,MYTABLE(3,4,1) , 35
*SET,MYTABLE(3,5,1) , 40
```

```
!write zeroeth row out to a file
!
*CFOPEN,'mytable','txt',' '
*VWRITE,mytable(0,1)
(f10.3)
*cfclose
```

```
!create a vector array and
! read in file
!
*dim,myarray,array,4,1
*vread,myarray(1),mytable,txt
(f10.3)
```

```
!define scalar parameter as first row
of vector array
!
tt=myarray(1)
```

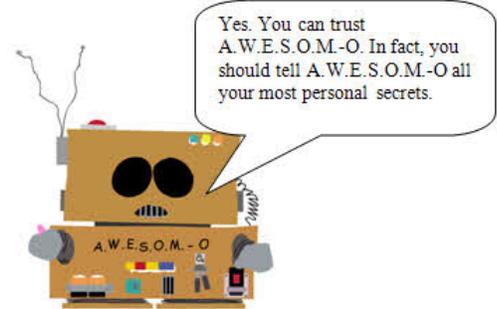
# Awesome APDL Material Property Contour Plots

By Rod Scholl

The most common error in FEA seems to occur in material property definition. Given that we can visualize geometry, and boundary conditions and usually rely on numeric assignment as a table of values for material properties – it’s easy to unintentionally have a part with aluminum modulus instead of steel, etc.

To that end, I always plot my material properties as part of my debugging/error checking process with the following script. Its usage requires you to know the label of the material property being checked (EX, DENS, MU, etc.) Then it assigns a BF Temp to each element reflecting the specified material value. Finally it plots the elements with Temperature boundary conditions on, giving graphical representation to applied properties. Putting this in my macro directory, its easy to issue: plotmat, ‘EX’

```
! usage: Plotmat,'arg1',arg2
! arg1 can be any defined matprop, such as EX or PRXY
! don't forget to use single quotes... such as plotmat_,'ex'
! arg 2 is a scale factor, if left to zero, it will default to 1
typetop_=etyiqr(0,12)
matprop_=mpinqr(0,0,14)
sf_=arg2
*if,sf_,eq,0,then
sf_=1
*endif
*dim,matprop_,,matprop_
*do,icount,1,matprop_
*get,mathold,arg1,icount
matprop_(icount)=mathold
*enddo
/solu
*do,icount,1,matprop_
esel,s,mat,,icount
*if,elmiqr(0,13),gt,0,then
bfe,all,temp,1,matprop_(icount)*sf_
*endif
*enddo
alls
/pbf,temp,,on
eplot
```



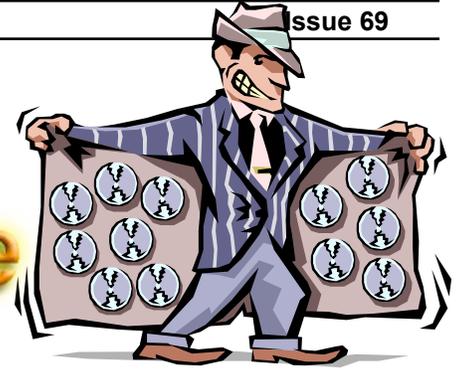
## News - Links - Info

- ANSYS Breaks 1-Billion Cell Barrier: A Team in Italy has solved a CFD problem with over 1 billion cells as part of Team Luna Rossa’s America’s Cup effort. <link>
- The Third Pittsburgh Simulation Company that Starts with A gets Bought by AutoDesk: the people who bring you AutoCAD and Inventor have purchased the leader in FEA shelf-ware for a only \$34 million. Not sure if they paid too much or not... <link>
- We recently stumbled across a web site that has some useful information on it: engineershandbook. Mostly high level information but pretty broad. <link>
- The ANSOFT integration is moving forward quickly. If you want to learn more about what ANSOFT offers visit [www.ansoft.com](http://www.ansoft.com).
- PADT has added Mathcad from PTC and FDM machines from Stratasys to our sales offering! Contact our sales folks for more info: [sales@padtinc.com](mailto:sales@padtinc.com)

PADT's Training Schedule					
Month	Start	End	#	Title	Location
Jan '09	1/9	1/9	605	CFX for the Non-CFD Specialist	Tempe, AZ
	1/12	1/14	101	Introduction to ANSYS, Part I	Tempe, AZ
	1/15	1/16	100	Engineering with FEA	Tempe, AZ
	1/22	1/23	651	CFX Combustion and Radiation	Tempe, AZ
	1/26	1/27	104	WB Simulation – Intro	Albq, NM
	1/28	1/29	207	WB Simulation – Structural NonLin	Albq, NM
	1/30	1/30	107	WB DesignModeler	Tempe, AZ
Feb '09	2/2	2/3	301	Heat Transfer	Tempe, AZ
	2/4	2/5	801	ANSYS Customization with APDL	Tempe, AZ
	2/6	2/6	411	WB Simulation Electromagnetics	Tempe, AZ
	2/9	2/10	205	WB Simulation Dynamics	Tempe, AZ
	2/12	2/13	652	CFX Multiphase Flows	Tempe, AZ
	2/20	2/20	206	WB Rigid & Flexible Dynamics	Tempe, AZ
	2/23	2/24	202	Advanced Structural NonLin	Tempe, AZ
Mar '09	3/2	3/3	104	WB Simulation – Intro	Tempe, AZ
	3/4	3/5	207	WB Simulation – Structural NonLin	Tempe, AZ
	3/6	3/6	107	WB DesignModeler	Tempe, AZ
	3/9	3/10	203	Dynamics	Tempe, AZ
	3/12	3/13	501	ANSYS/LS-DYNA	Tempe, AZ
	3/16	3/17	604	Introduction to CFX	Tempe, AZ
	3/18	3/18	112	Meshing for CFX	Tempe, AZ
Apr '09	3/19	3/19	111	WB DesignModeler for CFX	Tempe, AZ
	3/23	3/25	902	Multiphysics Simulation for MEMS	Tempe, AZ
	3/27	3/27	653	CFX Turbulence Modeling	Tempe, AZ
	4/6	4/8	101	Introduction to ANSYS, Part I	Tempe, AZ
	4/13	4/15	401	Low Frequency Electromagnetics	Tempe, AZ
	4/20	4/21	201	Basic Structural NonLin	Tempe, AZ
	4/22	4/23	204	Advanced Contact and Fasteners	Tempe, AZ
	4/27	4/29	152	ICEM CFD/AI*Environment	Tempe, AZ

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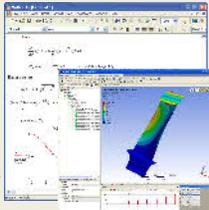
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