A Comparison of The Stress Results from Several Commercial Finite Element Codes with ASME Section VIII, Division 2 Requirements

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ABSTRACT

The interpretation of 3-D stresses computed using Finite Element (FE) techniques has been the focus of an ongoing PVRC study "3D Stress Criteria: Guidelines for Application" (Hechmer and Hollinger, 1995). This paper proposes an FE stress evaluation procedure for plate element models in the spirit of that recommended in the PVRC guideline. A sample model is analyzed, using five commercially available FE codes. The results are compared to illustrate the variability in the FE codes. Additionally, the practical difficulties in implementing the PVRC recommended procedure in the various FE codes is discussed.

DESCRIPTION OF PROBLEM

In a previous paper (Porter *et al*., 1995), it was shown that the use of the conventional type of analysis for closely coupled equipment systems could result in non-conservative estimates of the stress in the system. In particular, it was shown that applying the forces and moments derived from a piping program analysis to a WRC 107 analysis could under-report the stress in the nozzles when compared to an FE analysis.

The use of FE for stress evaluation of pressure vessel and piping components, however, gives rise to concerns about the modeling suitability and comparability of results using commercially available FE codes. The greatly increased usage of FE analysis by all segments of the pressure containment industry has provided the design engineer with more detailed information about stress, strain and displacement than in the past. This information should allow the engineer to provide the most cost effective-design solutions. But, as design engineers review the FE results, how can they be assured that the modeling techniques and results are satisfactory for the mandatory ASME analysis. This is particularly true when more than one engineering group or FE code is involved in the design and analysis of an item.

There is not adequate reference information in the literature to assure that there are consistent results from analyses using the various commercial FE codes. Primm and Stoneking (1989) gave some preliminary guidelines for the construction of a pressure vessel/nozzle FE model, but presented only data obtained from one FE code. In order to assess the variability among FE codes, it was decided to use a similar pressure vessel/nozzle configuration to compare several FE codes and to assess the deviation of the results. This comparison was expected to help develop confidence in the use of FE modeling techniques and in the consistency of the results.

A review of the referenced literature did not reveal a clear procedure for the comparison of the FE code-developed stress data with the ASME Section VIII Div I and Div II allowable design stresses. Although many papers have been published concerning the assessment of brick element FE models (Kroenke, 1973; Kroenke, *et al*, 1975; Heckmer and Hollinger, 1987), there seems to have been little information published concerning the interpretation of the plate element models commonly used for thin walled vessels. In fact, the Draft 3D Stress Criteria Guidelines (Hechmer and Hollinger, 1995) specifically refer to 3D solid element analysis and not to the use of plate elements.

In the Draft Guidelines (Hechmer and Hollinger, 1995), the need to linearize the stresses through the thickness of the model is discussed at some length, along with discussions on which stresses are to be used and how they are to be combined during the linearization process. All of these discussions are for brick (3D Solid) element models. Unfortunately, while the linearization process is discussed and the results illustrated in the Draft Guidelines, the actual numerical procedures for linearization are not presented. Kroenhe, *et al*. (1975) and Hechmer and Hollinger (1987, 1991a, 1991b) have discussed this process at some length. Hsu and McKinley (1990) published a description of a computer program for computing the linearized stresses, and at least one of the FE codes currently available has a linaerization routine built into the package. A clear description of this process (in terms that a typical vessel engineer could use to repeat the process) however, was not found in the literature.

With plate elements, the assumption that the stress is linear through the element is implicit in the element formulation. Thus, no further linearization is necessary. There is, however, a need for a means of interpretation of the plate element FE stresses as they relate to the stress criteria contained in Div II Appendix 4 - Mandatory Design Based on Stress Analysis. The authors determined that it would be necessary to propose a basis for comparison of the various FE code stresses to the Basis Stress Intensity Limits defined in Paragraph 4-131 in Appendix 4.

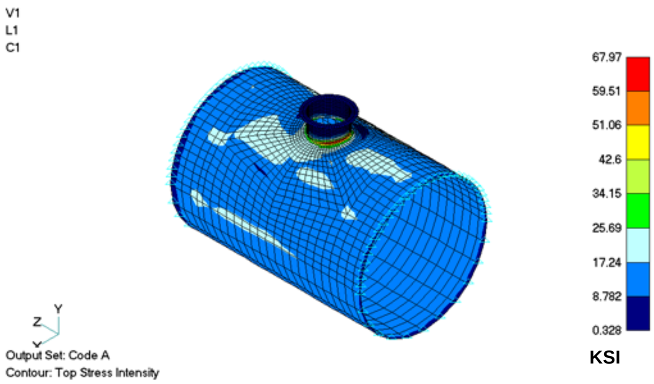

Figure 1 - Basic Model Geometry and Stresses

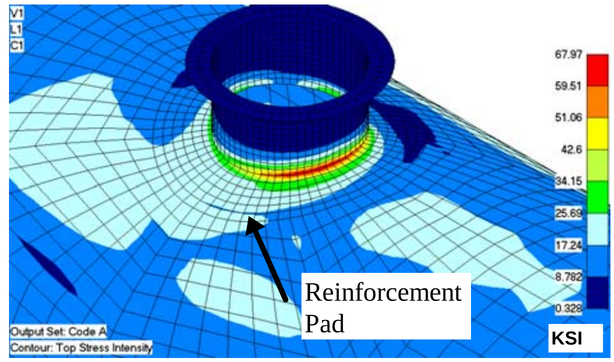

Figure 2 - Closeup View of Nozzle

FE Model

The model used for comparison in this paper consists of a 96" diameter, ½" thick vessel with a 24" diameter, ½" thick nozzle attached perpendicular to the centerline of the vessel. The vessel was reinforced at the nozzle intersection with a 42" diameter, ½" thick pad. The material is SA516-70 at 500 °F. For this material and temperature, the ASME Section VII, Division 2 properties are: Sm = 20.5 Ksi and Sy = 30.7 Ksi . The basic geometry of the model is illustrated in Figures 1 and 2.

Since the vessel and nozzle may both be considered "thin" (r/t = 24, R/T = 48 in the pad area, 96 in the vessel in general), plate elements were used to model the vessel and nozzle. The effort required to model this intersection with brick elements is greater than would normally be justified in a typical design project. The pad surrounding the nozzle was modeled by simply increasing the thickness of the elements. While this simplification does reduce the accuracy of the computed stresses, the work by Chen and Chao (1993) would seem to indicate that the magnitude of the errors introduced would be comparatively minor. Of somewhat greater concern is the lack of a means for modeling the fillet weld typically used in such nozzles. Although the radius of this weld is usually considered a factor in the peak stress levels, this geometry is not accounted for when using plate elements.

The mesh employed consisted of 64 elements around the periphery of the nozzle with the other dimension of the elements sized to maintain an aspect ratio of 2:1 or less. This mesh size was selected based upon previous work with a similar model which indicated that reasonable convergence of the results could be expected and seems to be consistent with the work by Primm and Stoneking (1989).

The model was loaded as follows:

Internal Pressure:	165 PSI
Force Y:	-6,480 LB
Moment X:	33,160 FT-LB
Moment Y:	38,250 FT-LB
Moment Z:	25,500 FT-LB

Note that the actual forces applied to the nozzle included a 74,644 LB load in the Y direction due to the pressure thrust on the nozzle. This thrust load must be added to the -6,480 LB load that was reported by the piping analysis. However, the thrust load is not reported by the piping analysis programs and is an often overlooked load, especially in WRC 107 analyses.

The model geometry and loading were created parametrically using the program described by Martens, *et al*. (1996). This data was then converted to input files for 5 general purpose, P.C. based, FE codes using the Finite Element Modeling and Post-Processing program "FEMAP" as a conversion tool. No additional changes were made to the models prior to running the various processors.

COMPUTED STRESSES

The stress intensities (top surface) in the model as computed by FE code A are illustrated in Figures 1 and 2. These plots are representative of the plots obtained from all of the FE codes. As may be seen, the highest stress is indicated at the intersection of the nozzle and vessel/pad. Due to the application of the moments to the nozzle, the highest stress is indicated at a point approximately 28 degrees off the centerline of the vessel.

As may be observed in Figures 1 and 2, the indicated Stress Intensities range from near zero to nearly 68 Ksi depending on the location in the model. The obvious question is: How do these indicated stresses relate to the ASME Code? The highest stresses occur in only a very small region. Are these “Peak” stresses or “Secondary” stresses? There is no obvious, straightforward guidance for the engineer to evaluate these stresses.

In order to better characterize the stresses, a section of the model containing the highest stresses was identified. This section is illustrated in Figure 3. For purposes of characterizing the stress, the directions **y** and **r** are defined and the Stresses Intensities along nodal lines A and B were plotted as a function of the dimensionless parameter δ. This parameter is computed by: δ = y/t on the nozzle side and δ = r/T on the shell side, where t and T are the nozzle and vessel plus pad thicknesses respectively. The values computed along lines A and B were then averaged on the top (outside) and bottom (inside) faces of the elements.

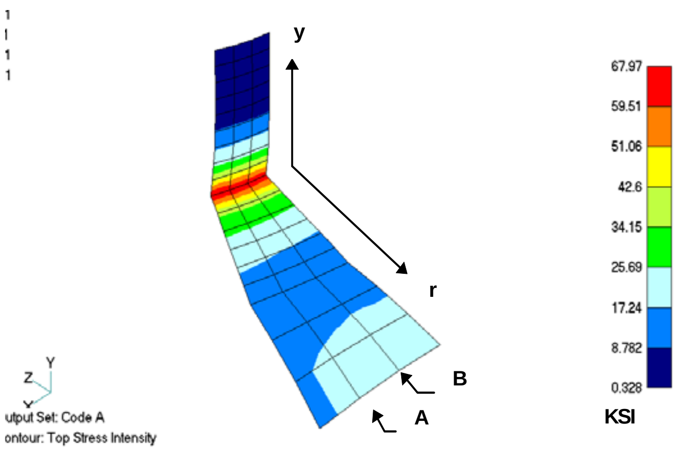

Figure 3 - Stress Section

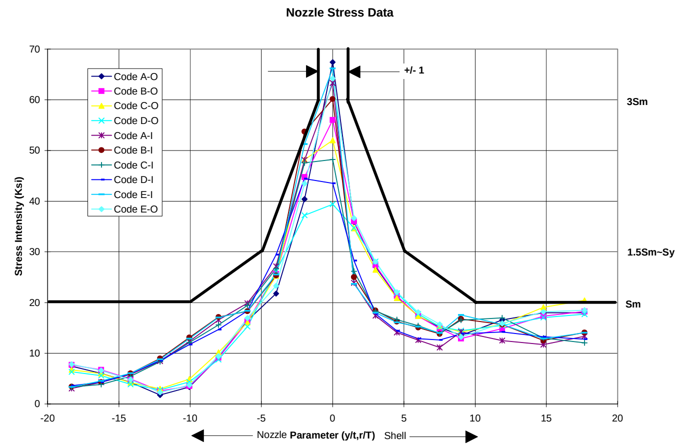

Figure 4 - Stress Intensities - All Data

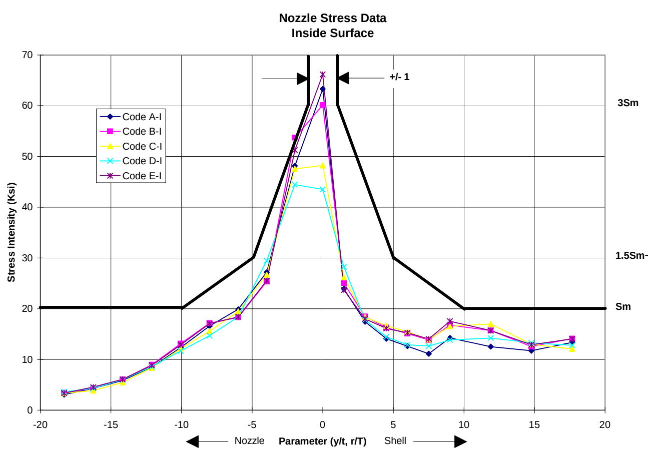

Figure 5 - Inside Surface Stress Intensity

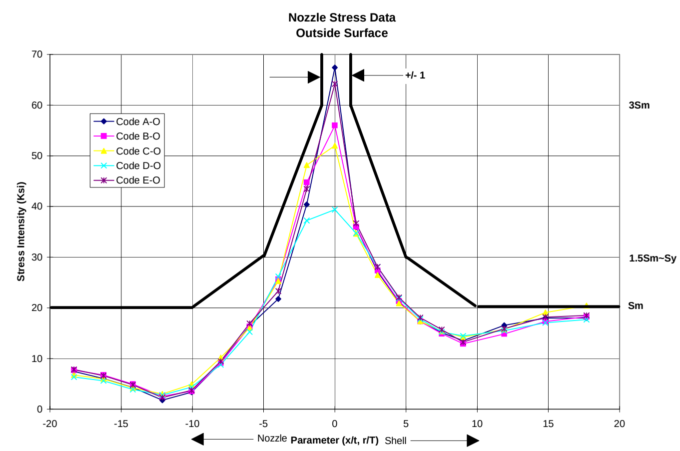

Figure 6 - Outside Surface Stress Intensity

Figure 4 illustrates the resulting Stress Intensities as discussed above for all five FE codes. Although there is considerable scatter in the data, some regions of converging values are evident. If we separate the inside and outside plots, these areas of agreement become even more evident. Figures 5 and 6 illustrate the indicated Stress Intensities on the inside and outside surfaces respectively for all five FE codes. With the exception of the intersection point and the first point on the nozzle (ρ = 0 to 1), there is very little divergence among the FE codes. On the inside surface at the intersection and first node on the nozzle, the FE codes seem to break into two groups, with the same grouping occurring in the pad area. This divergence of reported stress values seems to correlate well with the element formulations used by the FE codes. FE codes A, B and E use a standard "thin" plate element which discounts shear deflection; FE codes C and D default to more of a “thick” shell element that takes shear deflection into account. Thus, we would expect that the stresses in the elements that take shear deflection into account would be lower, as seems to be the case. Since all indicated stress values near the intersection are above the material yield, the results of a linear analysis cannot be considered quantitatively accurate.

The magnitude of the difference in the reported values at the intersection (42-62 Ksi) is somewhat disturbing. Even more disturbing is the spread in reported Stress Intensities of 38 to 67 Ksi on the outside surface. Further effort to resolve these differences would seem appropriate.

Once away from the intersection, however, the reported values from all of the FE codes seem to be quite consistent. All of the reported data fall within +/- 10% except at the intersection and at the first point (ρ = 1) on the nozzle. In this region, all of the indicated stresses are above the material yield. Since these are linear analyses, the magnitude of the indicated values above yield have questionable meaning. The elements which take shear deflection into account, however, are more likely to approximate the actual stress in the nozzle.

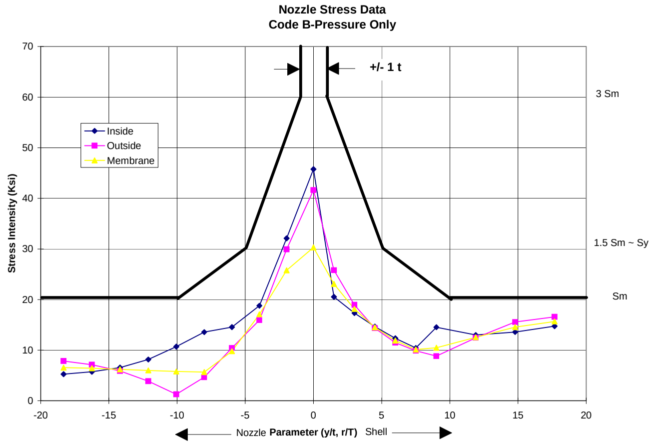

Figure 7 - Bending and Membrane - Pressure only

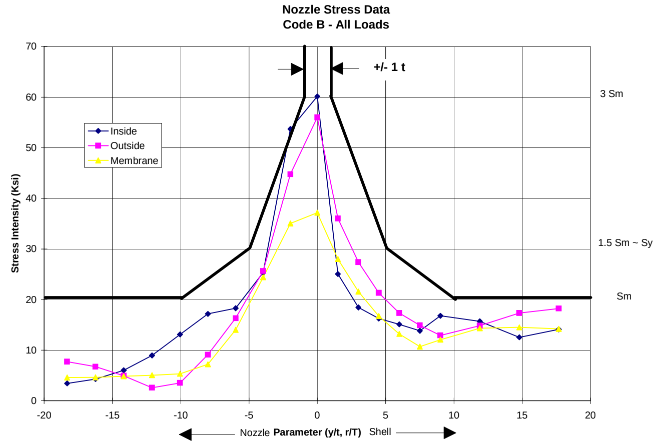

Figure 8 - Bending and Membrane - All Loads

INTERPRETATION OF RESULTS

We may summarize Appendix 4 of ASME Section VIII, Div II as follows:

- Using the design load, the General Membrane stress (P_M), which excludes "discontinuities and concentrations," should be limited to kSm. If the factor k is unity, then the General Membrane stress is limited to the allowable for the material and temperature involved.
- Using the design load, the Primary stress (P_M + P_L + P_B), which is due solely to mechanical loads and excludes stress concentrations, is limited to 1.5 kSm. Again, if k is unity, then the limit is 1.5 times the allowable. Note that the local membrane portion of the Primary stress (P_L) "Considers discontinuities" while the bending portion (P_B) excludes discontinuities.
- Using the operating load, the combination of Primary plus Secondary Membrane plus Bending (P_L + P_B + Q) is limited to 3Sm. Although this stress may be caused by mechanical or any other load, "local stress concentrations" are to be excluded.
- Finally, the "Peak" (F) stress (that due to a stress concentration or "Certain thermal stresses which may cause fatigue but not distortion of vessel shape" due to operating loads) is limited by Sa, which is obtained from the fatigue curves, Figs. 5-110.1, 5-110.2 and 5-110.3.

The interpretation of these requirements has been the subject of numerous technical papers and, within the design community, certainly countless discussions. In order to compare the results of the FE runs with the Code requirements, several defining assumptions, specific to the nozzle configuration, have been made:

The General Membrane (P_M) stress in this analysis is the stress well away from the intersection of the nozzle and vessel. "Well away" will be defined as a $\mathscr{d}$ >10. That is, more than ten times the thickness of the vessel (and pad) from the juncture on the vessel side and ten times the thickness of the nozzle on the nozzle side.

The Primary (P_L + P_B) stress, that due to mechanical loads but not including "concentrations," will occur in the region where $\mathscr{d}$ is between 5 and 10.

The Secondary Membrane plus Bending (PL + PB + Q) stress, which "excludes local stress concentrations" will be the stress which occurs in the region where the $\mathscr{d}$ is between 1 and 5.

Finally, the stress levels in the region where the $\mathscr{d}$ is within 1 of the intersection will be assumed to be Peak (F) stresses. Since the fillet has not been taken into account, the actual stress values are likely to be lower than these indicated stresses. The magnitude of this adjustment would not be expected to lower the indicated values to less than yield.

Historically, it has been customary to separate the membrane and bending components of stress in the analysis. If we look at the Code assessment guideline in Appendix 4, however, this is not generally required. In fact, if we can establish that P_B due to the primary load is negligible except at stress concentrations, then separating the values is unnecessary and Stress Intensity due to all loads may be used.

Figure 7 illustrates the stress in the model as computed with FE code B, with only the pressure portion of the load. Figure 8 illustrates the results for FE code B with all loads imposed. In addition to the Stress Intensities on the inside and outside surfaces, we have plotted the Stress Intensity that is computed using only the membrane component of the stress. Away from the intersection, the membrane stress typically falls between the values for the inside and outside surfaces. Nearer the intersection, the membrane stress tends to be only part of the total. That is, it is near the intersection, (even for pressure only) that bending adds significantly to the total stress. Thus by limiting the stress intensity at $\mathscr{d}$ <= 5 away from the intersection to 1.5 Sm, we should meet the intent of the Appendix 4 guidelines for P_L.

Hechmer and Hollinger (1991) state "There are numerous discontinuities where PL stresses exist, but need not be evaluated, because the design is established by Code rules. Most notable is the nozzle-shell juncture, where reinforcing rules ensure that the P_L limit (or its intent) is met." The selection of the $\mathscr{d}$ <= 5 point for the evaluation of the compliance with 1.5 Sm is, therefore, likely to be conservative.

Based upon the shape of the stress vs. $\mathscr{d}$ plots, it seems reasonable that there should be a smooth transition of the allowable stresses between the points defined at d = 1, 5 and 10. Such an envelope has been indicated on Figures 4-7. As may be seen, all of the FE data points, save one, fall within the envelope. Since this nozzle meets the material replacement requirement of the Code for pressure as well as the meeting the WRC 107 criteria for both pressure and external loads (see Appendix A), we would expect the FE analysis to show compliance. Thus, the use of plate elements and the proposed criteria curve, for the example problem at least, would seem to offer a reasonable means of compliance assessment.

CONCLUSIONS AND RECOMMENDATIONS

Away from the intersection by 2-4 thicknesses, the indicated stress values from the various FE codes examined were in very good agreement. At and near the intersection, a wide range of values was reported, which seemed to be a function of the of element type used by the specific FE code (thick shell to include shear deflection or thin plate). In any case, for this example, the same conclusion regarding compliance would have been reached with any of the FE codes employed.

The proposed criteria curves, which are a function of Sm and d provide a simple means of evaluating FE stress analyses as they relate to the ASME Code. While they would seem to work for the geometry analyzed in this paper, the applicability to other geometries and loadings remains to be demonstrated.

The five commercially available FE codes tested give reasonably consistent results for the interpretation of the stresses in this model configuration. The choice of FE codes would not seem to be a large factor in the evaluation of stresses in similar models.

The use of thin plate elements for modeling relatively thin vessels (r/t > 10) seems to yield reasonable results. One point of consideration is the calculation of peak stress within the d = 1 range. The Code calculation of peak stress seems to have been developed based on linear, thin plate and beam analyses. The FE codes that take shear deflection into account (thick shell), appear to give results that may not be consistent with the intent of the ASME Code definition of Peak stress. Thus, the levels reported by thick shell elements and the corresponding allowable values presented in the Code may not be comparable. In order to use this type of element for comparison with the Code Peak Stress values (F), it may be necessary to apply a stress concentration factor to the reported stress values.

Where the nozzle and shell intersect, typically referred to as the “ring” area, a linear analysis indicates that a considerable volume of the material is above yield. In reality, this volume will strain until an energy balance is achieved and some larger volume is at the yield stress. The 3 Sm criteria has been used to define this allowable strain, based upon a linear analysis. The analysis in this paper and that in the Code criteria do not really address the volume of material that is involved. Since we now have the tools to assess this volume of material, perhaps an additional criterium needs to be developed.

At this point, the evaluation procedure used has not been validated for other nozzle geometries. If validated, however, it would offer the design engineer a relatively simple means of evaluating FE stress results. To this end, we would strongly recommend that the PVRC consider the development of such a procedure for FE analyses employing plate elements. In this development procedure, some means of equating the volume of material indicated to be above yield by the linear analysis to the actual volume of material that would be at yield in a nonlinear analysis needs to be developed.

ACKNOWLEDGMENTS

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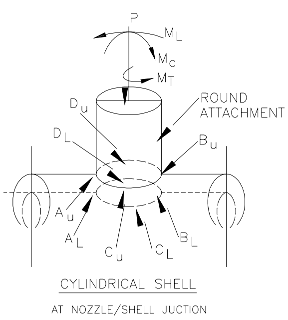
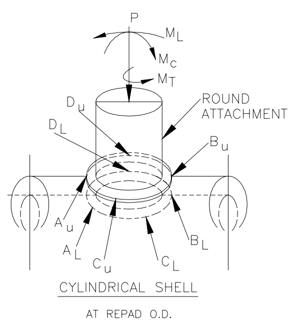
Appendix A - WRC 107 Calculations





WRC 107 Input Data

	Loads		**Moments**	
	Rad. P - LB	**Long. M_L - Ft-LB**	**Circ. M_C - FT-LB**	**Tors. M_T - FT-LB**
Pressure Only	-74,644	0	0	0
All Loads	-68,164	33,160	25,500	38,250

WRC 107 Solution Data

Pressure only Maximum Combined Stress Intensity - PSI

	A_U	**A_L**	**B_U**	**B_L**	**C_U**	**C_L**	**D_U**	**D_L**
Nozzle/Shell	48,746	31,244	48,746	31,244	54,824	15,143	54,824	15,143
Repad O.D.	50,273	44,810	50,273	44,810	41,266	28,018	41,226	28,018

All Loads Maximum Combined Stress Intensity - PSI

	A_U	**A_L**	**B_U**	**B_L**	**C_U**	**C_L**	**D_U**	**D_L**
Nozzle/Shell	33,940	6,449	56,649	11,239	34,315	11,296	69,597	17,639
Repad O.D.	39,706	22,704	54,915	10,524	10,019	32,403	68,615	1,443