Transactions of the **13th International Conference on Structural Mechanics in Reactor Technology** (SMiRT 13), Escola de Engenharia - Universidade Federal do Rio Grande do Sul, Porto Alegre, Brazil, August 13-18, 1995

ASME limits and contradictions in the finite element analysis of pressure vessels

Bezerra, L.M., Cruz, J.R.B, Miranda, C.A.J., Mattar Neto, M.
Comissão Nacional de Energia Nuclear/SP-IPEN, São Paulo, SP, Brazil

ABSTRACT: There is a legitimate concern in the engineering community to interpret Finite Element (FE) stress results on the light of the ASME rules and Philosophy. The fact is that the ASME simple beam approach is not completely adequate to interpret all stress results. There are many papers in the literature offering guidelines on how to deal with FE results in front of the requirements of stress classification and linearization required by the ASME Code. This paper is not one more guideline but an independent contribution focusing stress classification and failure mechanism in a FE framework. This work tries to complement the appealing work presented in the PVP-94 by Hollinger and Hechmer (1993). These authors examined a typical support skirt and showed interesting relations between the collapse load obtained by finite element analysis and the loads allowed from the ASME stress limits. The scope of our paper is then to complement Hollinger and Hechmer (1993) with more numerical data. Here, different skirt geometry configurations - with different angles of attachments between cylinder and cone parts - and a pressure vessel head are investigated. These configurations permitted us to observe the influence of larger bending stress in the collapse load. They also showed the bending stress relationship to the allowable loads when calculated using the ASME limits. Using elastic and limit load FE analyses, the present paper determines the collapse loads of the skirts and pressure vessel head. We also establish the relationships between the calculated collapse loads and limits dictated by the ASME Code Subsection NB. Based on the NB rules, it will be possible to observe how different methods of stress assessment, classification, and limits may influence in the design of pressure vessel parts.

1 INTRODUCTION

Today, it is a fact that computer technology and finite element analyses have had an overwhelming technological advance and invaded our engineering offices. We live a completely different situation compared to the sixties when the "design by analysis" was written for the B&PV Code. The impact of computer technology and Finite Element Analysis (FEA) in engineering judgment is a real fact specially in sophisticated industries like the nuclear industry. Nowadays, computers and FEA are essential tools for the designers and stress analysts of Nuclear Power Plant (NPP). Before interpreting FE results, the B&PV designer must understand the principles of the Code (in this paper ASME Code, Subsection NB). However, some concepts in the Code may lead the designer to unnecessary expenses or even "unexpected" collapses generating higher costs and risks. These facts can affect the prospects for a return to the building of new NPP and a threat to human being and the environment. Today, the temporary halt observed in the construction of new NPP turned the nuclear industry emphasis from the design and construction of new plants to the reliability, life-time extension and maintenance of operating units. Therefore, this is a convenient time for revisions and changes in the Code. It is wise to work out the controversial matters of the Code before the nuclear industry picks steam up again.

This work is a modest and independent contribution to PVRC working groups on the stress classification and the failure mechanism. This paper tries to call attention to some critical points of ASME Subsection NB that need more attention. Such points are stress assessment and stress classification. The use of modern FE programs, in the design of nuclear components, offers to the stress analyst the opportunity to calculate elastic stress and also limit-loads. Using FE results and NB rules, it is possible to obtain allowable loads. The designer can observe that these allowable loads may be quite different and therefore can question the safety philosophy behind the simple ASME rules. The Code rules are strongly grounded on simple beam theory and, in certain situations, allowable ASME loads based on usual elastic stress results may lead to non-conservative designs. The present paper shows that there are situations that the stress analyst can be taken to designs with no safety margins. Using finite element elastic and plastic analyses, this paper shows that some non-conservative situations may arise when allowable loads (based in different stress assessment and stress classification) are calculated. However, we emphasize that the goal of the present article is <u>not</u> to conclude that the Code rules are inadequate and useless. It is well acknowledged by the authors that since 1915 - when B&PV Code was first published out - an outstanding record for safety was produced. However, there are critical problems in the Code that must be addressed. An interesting survey about the weak links in the Code can be found in (Hollinger and Hechmer 1986, 1993). The unclear concepts of the Code must be removed with responsibility and the same traditional ASME safety philosophy to strengthen and adequate the utility of the Code to the new and wide spread computer technologies and numerical modeling of today.

2 A TYPICAL SUPPORT SKIRT

Hollinger and Hechmer (1993) selected a typical support skirt configuration to study. Such skirt has a 18° angle of attachment between cone-cylinder. An end-load of 1122*psi* was applied on the bottom cross section of the support cylindrical part. They have assessed the stress in the juncture using hand calculations from simple beam theory (σ=F/A $\pm$ Mc/I) and FEA. They have incremented the end-load and performed a FE elastic plastic analysis and determined a lower bound limit load. Taking the Code (ASME, 1989) definition of limit-load analysis as the intended basis for design, they have then inferred the following: 1) the general primary stress, calculated by beam theory, gave an allowable load within 10% margin with respect to the limit-load results, 2) the membrane and/or bending stress was related to collapse mode, 3) beam theory gave a more conservative end-load than the end-loads based on elastic FEA. In this paper, the work by Hollinger and Hechmer (1993) is extended a little bit more. Our contribution is to analyze three support skirts and also a torispherical pressure vessel adopting the same strategy used by Hollinger and Hechmer (1993). The limit-loads of each configuration will also be determined and compared with the diversity of loads allowed by the Code when different stress classification approach is used. It will be possible to observe the influence of the support skirt attachment angle between cone-cylinder on the allowable loads. Such attachment is responsible for the bending stress. The influence of the bending stress classification on the failure mode is also observed.

3 GEOMETRIC CONFIGURATIONS STUDIED

Following the same type of geometric configuration studied by Hollinger and Hechmer (1993), we have analyzed three different skirts and a torispherical head. The skirts chosen for the analyses were formed by putting together a cylinder connected to a cone with three different angles of attachment. At the end of the cylindrical part an axial load is applied. This end-load consists of an arbitrary axis-symmetric stress of F=1122*psi* acting on the inferior cross section of the cylinder. The material properties used in the analyses were $E=29.5\times10^6$*psi*, ν=0.3, From the ASME Appendices [5], the allowable stress (S_m) is 26,700*psi* and the yielding stress (S_y) is 40,000*psi*. The internal radius of cylindrical part of skirts is R_i=48.64*in* and the thickness t=2*in*. The cone part is attached to the cylindrical part making an angle α with respect to the central axis. This angle is responsible for the bending stress in the cone-cylinder juncture area. Different α angles are used in this work so that the impact of increasing bending values on the limit loads could be observed. Figure 1 shows the FE meshes of the skirts. To simulates a rigid foundation for the support skirt, the cone upper part is completely fixed. In addition, the nodes at the inferior section of the cylindrical part - where the end-load is applied - are coupled to simulate zero rotation. Taking into account the geometric change cone-cylinder and the locally applied stress at the bottom of the cylinder, an attenuation length of $5\sqrt{Rt}$ is considered. A blend radius of 2t in the juncture notch cylinder-cone is used so that the stress singularity in the notch can be prevented. In addition to the support skirts we also analyzed a pressure vessel with the same internal radius and thickness of the skirt cylinder but under an arbitrary internal pressure of p=450*psi*. The material of the vessel is the same used for the support skirts. The tangential internal radius of curvature between the knuckle region and the spherical (crown) segment is L_i=90*in*. The internal radius of the knuckle is r_i=6*in*. The transitional angle between cylinder and crown is ϕ=60° with respect to the horizontal axis. Figure 1 (d) shows the FE mesh of the vessel.

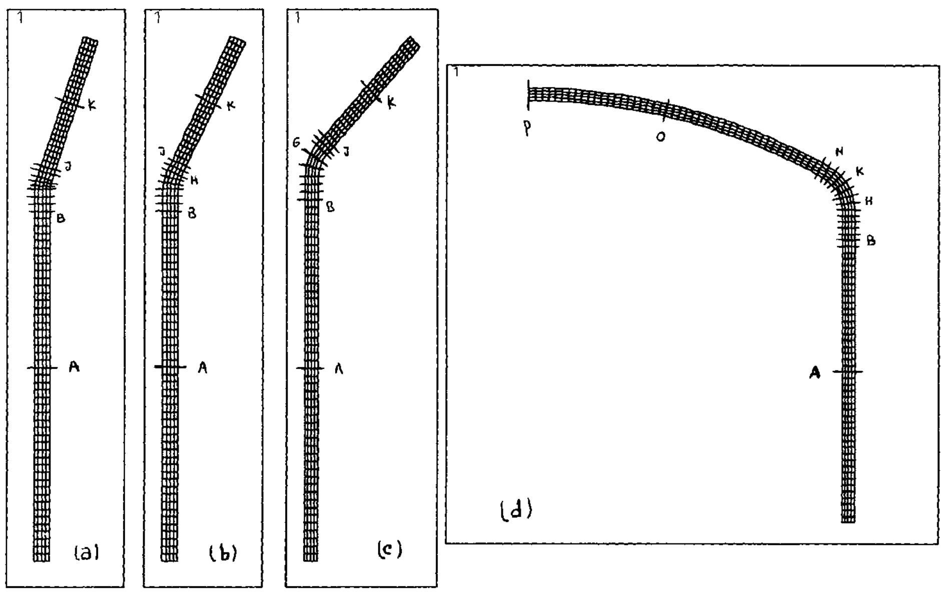

Figure-1 - (a) Skirt with α=18°, (b) Skirt with α=25°, (c) Skirt with α=45°, and (d) Pressure Vessel

4 ELASTIC & LIMIT ANALYSES

The geometric configurations previously described are analyzed using the FEM available in the general purpose ANSYS program. The arbitrary end-load of 1122*psi* in the support skirts, and the pressure of 450*psi* inside the torispherical vessel causes only elastic stresses in the respective component. Figure 1 shows the FE models. Also in that figure, the stress classification lines SCL, where the elastic stresses are linearized, are represented. Other than verifications with elastic analysis the ASME Code also admits the use of the plastic analysis. The limit analysis is a special case of plastic analysis taking the material as ideally plastic with no strain hardening. Considering a structure made of such material, a lower bound collapse load can be defined as the maximum load that such a structure can carry without an unbound increase of deformations. In this paper, for the knowledge of the limit load of each configuration, the lower bound collapse approach is used. For the SA533-Gr.B, the collapse load can be achieved assuming that the stress-strain curve has an initial slope of $E=29.5\times10^6 psi$, and a stress plateau at Sy=40.000*psi*. Accepting two-thirds of the lower bound collapse load is compatible with the ASME Code requirements to find the allowable load based on limit analysis. The classical bilinear option from ANSYS with the Von-Mises yield surface flow law was employed in the inelastic FE model. Very small load steps are adopted during the plastic analysis of the structures. For the skirts, an end-load increment of 100*psi* is applied whereas in the case of the torispherical vessel, pressure increments of 50*psi* are utilized. For the convergence criteria, the ANSYS default convergence criterion is put into action during the typical 10 iterations used for each load step increment. The résumé presented in Table-A shows the collapse loads obtained for each configuration looked at in this paper. From Table-A, the high influence that the bending stress has on the final collapse load is clear. For the support skirts, it is straightforward the that the bigger the attachment angle α becomes, the smaller the collapse loads are. Thus, it should be reasonable to treat at least some share of the membrane + bending stress as primary stress (Hollinger and Hechmer 1993). The Code (ASME, 1989) considers, in areas of geometric discontinuity, membrane as local and bending as secondary.

TABLE-A

Component analyzed	Collapse Load
Skirt angle α=18°	36300 psi - end load
Skirt angle α=25°	30700 psi - end load
Skirt angle α=45°	18700 psi - end load
Pressure vessel	1200 psi - internal pressure

5 STRESS ASSESSMENT & ALLOWABLE LOADS

The philosophy of "adequate safety" that is granted by the limits imposed to general primary membrane ($\leq S_y/1.5$) and primary membrane-plus-bending stress (S_y) in the ASME Code is based on the simple beam theory. In this paper, following the same steps presented in (Hollinger and Hechmer 1993), the beam-approach collapse loads of the configurations here analyzed are set when the axial membrane stress, hypothetically, reaches the yielding stress value S_y=40,000*psi*. Considering the safety factor of 1.5, the allowable end-loads for all the skirt configurations are then obtained after dividing S_y per 1.5. The allowable skirt end-load from the beam approach is then 26,700*psi*. For the vessel, the membrane stress in the cylinder is $\sigma=pR/t$ and in the spherical crown is $\sigma=pR/(2t)$. In the knuckle region the bending stress is predominant. Therefore, limiting the general membrane stress σ to 40,000*psi*, the maximum internal pressure can reach $t\times40000\div R$, or p=1611*psi*. Applying the safety factor 1.5 for the general membrane stress, the allowable internal pressure is 1074*psi*.

Using the FE techniques one can establish a more precise value for the collapse load and, therefore, verify the robustness of the limits set by the ASME rules. Such approach was endeavored for the support skirt configurations and the pressure vessel described in the last paragraph. For the skirt with α=18°, the results reported hereon confirm the values given in (Hollinger and Hechmer 1993). For example, in the vicinity of the cylinder-cone juncture, the maximum membrane stress intensity is in SCL-J. The stress intensity, there, is dominated by the meridional stress and reaches 1153*psi*. It can be mentioned that our model is longer than the model of (Hollinger and Hechmer 1993) and also because the nodes where the end-load was applied were coupled to zero the rotation of that cross section. In the bending region cylinder-cone, the membrane stress can be classified as a local stress (P_L). The Code rule limits local stress $P_L\leq1.5S_m$=40,000*psi*. Thus, if the end-load of 1122 causes such a stress value, the maximum allowable end-load is then $40000\times1122\div1153=38924$*psi*. For the support skirt with α=25°, the maximum P_L stress intensity is reached at SCL-E and the value of P_L=1562 *psi*. In this case the maximum allowable end-load is $40000\times1122\div1562=28732$*psi*. Similarly, for the other skirt with α=45° the maximum P_L is observed at SCL-E and gets to P_L=3066 *psi*. In such a case, the maximum allowable end-load, based on ASME local membrane limit, is $40000\times1122\div3066=14637$*psi*. For the pressure vessel, the maximum P_L stress intensity is reached at SCL-I and the value of P_L=17130*psi* which gives an allowable internal pressure equal to $40000\times450\div17130=1050$*psi*.

From the finite element results, the maximum membrane + bending (P_L+P_B) stress intensities are achieved in SCL-F, SCL-F again, and SCL-G, respectively. The maxima (P_L+P_B) are: 2839*psi* in SCL-F for the skirt with α=18°, 3457*psi* in SCL-F for α=25°, and 5349*psi* in SCL-F for α=45°. For the pressure vessel, P_L+P_B=36980*psi*. Considering membrane + bending as primary stresses the Code limits P_L+P_B≤1.5S_m=40000*psi*. Taking such limiting value, the allowable end-loads for the skirts are: a) 40000×1122÷2839=15808*psi* for the skirt with α=18°, b) 40000 ×1122÷3457=12982*psi* for the skirt with α=25°, and c) 40000×1122÷5349=8390*psi* for the skirt with α=45°. For the vessel the allowable internal pressure is 40000×450 ÷36980*psi*. The ASME Code considers the bending stress in the cylinder-cone transition as secondary. Categorizing the membrane + bending as primary + secondary stress, the Code limits P_L+P_B≤3S_m=80000*psi*. Therefore, in such a case the allowable end-loads are the double of the values obtained before, i.e., 31616*psi*, 25964*psi*, and 16780*psi*, for the skirts with α=18°, 25°, and 45°, respectively. In the case of the pressure vessel, the allowable internal pressure is 972*psi*. It should be noted that if P_L+P_B>3S_m=80000*psi* appropriate penalty factors should be applied for the fatigue evaluation, and also that appropriate verification should be performed to prevent ratcheting.

6 CONCLUSIONS

Observing Tables B, C, and D, one can concluded that the simple beam theory, the elastic FEA, and the lower bound FE approaches gave very different results. It is obvious from the calculations that the stress classifications and the stress assessment methods influence the determination of the allowable loads. Examining Table B, C, and D and considering the Code definition of Limit-Load Analysis as the basis for design, and comparing such allowable load with the other results, it can be concluded that: (a) The beam approach gave greater allowable loads; (b) categorizing the membrane near structural discontinuity as P_L also resulted in greater allowable loads; (c) taking membrane + bending near structural discontinuity as primary, produced smaller allowable loads; (d) considering membrane + bending at structural discontinuity as primary + secondary (P+Q) resulted in greater allowable loads. Notice that P_M, P_L, and P_B limits are related to plastic collapse, while P+Q is related to fatigue and incremental plastic deformation.

TABLE-B

	End - Loads PSI - Cross Section					
	Cone at 18°		Cone at 25°		Cone at 45°	
	At Limit	Allowable	At Limit	Allowable	At Limit	Allowable
Beam - simple theory	40000	26700	40000	26700	40000	26700
FEA, M as PL	38924	38924	28732	28732	14637	14637
FEA, M + B as PL + PB	15808	15808	12982	12982	8390	8390
FEA, M + B as P + Q	31616	---	25964	---	16780	---
FEA, Limit Load	36300	**24200**	30700	**20466**	18700	**12467**

TABLE - C

		End-Load								
		Cone with 18°			Cone with 25°			Cone with 45°		
		At Limit	Allowable	SCL	At Limit	Allowable	SCL	At Limit	Allowable	SCL
Beam Theory		40000	**26700**	----	40000	**26700**	----	40000	**26700**	----
FEA	M as PM	25648	25648	A	25238	25238	A	24043	24043	A
	M as PL	38924	38924	J	28732	28732	E	14637	14637	E
	M+B as PL+PB	15808	**15808**	F	12982	**12982**	F	8390	**8390**	G
FEA	M as PM	25648	**25648**	A	25238	**25238**	A	24043	24043	A
	M as PL	38924	38924	J	28732	28732	E	14637	**14637**	E
	M+B as P+Q	31616	----	F	25964	----	F	16780	----	G
FEA, Limit Load		36300	**24200**	----	30700	**20466**	----	18700	**12467**	----

The conclusions taken so far were based on the SCL's drawn in Figure 1. Those SCL's were selected on the basis of the designer's expertise. Before breaking up and classifying the stresses, it is wise to consider all the SCL's in conjunction with the Code recommendations (ASME 1968, 1989) and also take into account the suggestions given in (Hollinger and Hechmer 1986). On the light of these recommendations and suggestions, some more conclusions can be made: (a) For the skirt with $\alpha=18°$, see Table-B & C, the results confirm the observations and the conclusions given in (Hollinger and Hechmer 1993). However, we observe that in this case, both the beam theory and the elastic FEA results (away from the cone-cylinder juncture) approximate the plastic collapse end-load which is controlled by the higher stresses at the juncture. (b) For the skirt with $\alpha=45°$, see Tables B & C, the stress assessment based on elastic FEA (M as P_L) shows an allowable load slightly greater than the FE limit-load analysis (14637*psi* against 12467*psi*). In this case, the allowable load based on the beam theory is much greater than the FE limit-load analysis (26700*psi* against 12467*psi*). The elastic FEA (SCL-E) approximates the plastic collapse in the skirt much better than the simple beam theory. (c) For the skirt with $\alpha=25°$, see Tables B & C, neither the simple beam theory nor the elastic FEA gave allowable loads close to the FE limit-load analysis (26700*psi* and 25238*psi* against 20466*psi*). (d) For the pressure vessel, similar results are observed in Table-D. Categorizing membrane + bending as primary is too conservative but the simple hoop stress in the cylinder and the elastic FEA gave similar results, although greater (≈20-25%) than the FE limit-load analysis. It is recommended that the pressure vessel case and the skirt with $\alpha=25°$ case, among other cases, should be looked at with great care and responsibility. The results obtained may lead the stress analyst to non-conservative designs putting in risk the environment or, elevating the costs.

TABLE- D

		Internal Pressure		
		At Limit	Allowable	SCL
Simple Theory	Cylinder	1611	**1074**	------
	Head	3223	2148	------
FEA	M as PM	1008	1008	A
	M as PL	1050	1050	I
	M+B as PL+PB	486	**486**	I
	M+B as PL+PB	1066	1066	O
FEA	M as PM	1008	**1008**	A
	M as PL	1050	1050	I
	M+B as P + Q	972	---	I
	M as PM	1066	1066	O
FEA, Limit Load		1200	**800**	-----

REFERENCES

Hollinger , G. L. and Hechmer, J. L., 1993, Three Dimensional Stress Evaluation Guidelines Progress Report, PVP-Vol. 277.

Hollinger , G. L. and Hechmer, J. L., 1986, Three Dimensional Stress Criteria - A Weak Link in Vessel Design and Analysis, PVP-Vol. 109.

ASME, 1968, Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Section III and VIII, Divisiòn 2, The American Society of Mechanical Engineering, New York, NY.

SME, 1989, Section III Rules for Construction of Nuclear Power Plant Components, Division 1 - Subsection NB, Class 1 Components. The American Society of Mechanical Engineers, New York, NY.

ASME, 1989, Section III Rules for Construction of Nuclear Power Plant Components, Division1-Appendices. The American Society Mechanical Engineers, New York, NY.

Hollinger, G. L. and Hechmer, J. L., 1991, Three Dimensional Stress Criteria. PVP-Vol. 210-2