ON THE ALLOWABLE LOADS OF NUCLEAR COMPONENTS USING DIFFERENT STRESS ASSESSMENT AND STRESS CLASSIFICATION

Luciano Mendes Bezerra*, Carlos Alexandre de J. Miranda**, and Miguel Mattar Neto**

*Universidade de Brasília - UnB
Departamento de Engenharia Civil
70910-900 Brasília, DF - Brasil
e-mail: lmbz@guarany.cpd.unb.br

**Comissão Nacional de Energia Nuclear - IPEN/SP
Travessa R, 400 (Cidade Universitária)
05508-900 São Paulo, SP - Brasil
e-mail: mmattar@net.ipen.br

ABSTRACT

The "design by analysis" is essentially based on elastic stress analysis. The elastically calculated stresses are broken down into primary (P), secondary (Q), and peak stresses (F). Sometimes, it is difficult to categorize a stresses into P or Q. The intended limits on P and Q consider perfect plasticity and limiting theory as a floor below which a component is viewed as safe [1]. This paper is about the influence of stress categorization and stress assessment on the determination of *allowable loads* of nuclear components. In this work, the stresses are estimated by simplified hand calculations and also by sophisticated finite element elastic and elastic-plastic analyses. The *allowable load* are determined according to the requirements of ASME Code [2]. In this paper, it will be possible to observe how different methods of stress assessment and stress classification may deeply influence the design of nuclear components. The examples show situations where the stress analyst could be taken to practically no safety margins.

INTRODUCTION

In the early sixties, the "design by rules" which led the designers to overconservatism was replaced by the "design by analysis" [1]. The reason for that was the recognition that undetailed stress analysis accepted by the "design by rules" - limiting only the hoop stress to tabulated allowable values - was "reflecting lack of knowledge" [1]. A special committee was established to review Code stress basis and investigated what "changes in the Code design philosophy might permit use of higher *allowable stresses* without reduction in safety. It soon became clear that one approach would be to make better use of modern methods of stress analysis" [1]. This position reflected the current knowledge of the 60's and introduced the "design by analysis." Today, modern stress analysis are substantially based on Finite Element Analysis (FEA). FEA is a well known and trustful technology. The impact of FEA and computer technologies in stress calculation, specially in the nuclear industry, is not an illusion. Sophisticated FE programs (e.g. ANSYS [3]) offer the opportunity to calculate not only elastic stress but also precise *limit loads*. Using FE programs one could determine *allowable loads* and investigate the safety margins in designs guided by the Code rules.

The intended safety margins of the Code, for the determination of *allowable stresses*, are based on perfect plasticity and limiting theory as a lower bound below which a component may be considered safe [1]. Despite the progresses in structural analysis, the Code rules for the calculation of *allowable loads* are still strongly grounded on elasticity and "simple beam" reasoning [1,2]. The elastically calculated stresses are categorized into primary (P), secondary (Q), and peak stresses (F). P and Q have to be linearized and to such linearized stresses different limits are prescribed [2]. The choice for P and Q limits "was accomplished by the application of limit design theory tempered by some engineering judgment and some conservative simplification" [1]. Moreover, the rules governing linearization and categorization of stresses into P and Q are not very clear specially when finite element method is used [4,5,6]. The difficult in categorizing stresses is also recognized in document [1]: "many cases arises in which it is not obvious which category a stress should be placed in." To standardize the classification of stresses, a table was prepared covering most of the situations [1,2]." The practice has shown that such a table is incomplete and in many occasions, it seems very difficult to partition a stress into P and/or Q [4,8].

The influence of stress assessment and stress categorization on the design of nuclear components is the main concern of this paper. Considering the Code definition of limit load analysis as the intended base for design - and using elastic and elastic-plastic FEA and also simple hand calculation - this paper determines and discusses the *allowable loads* of typical nuclear components.

We strongly emphasize that the goal of the present work is <u>not to arrive to any decisive conclusion</u> about the Code rules. We are simply reporting numerical results that, in our personal view, seem at least questionable in front of the Code safety philosophy. The cases here studied show that there are situations where the stress analyst could be taken to designs with no safety margins or at least to very reduced safety margins. The authors acknowledge with great emphasis that B&PV ASME Code produced an outstanding record for safety since its first publication in 1915. However, we think that there are critical points (see also references [4,5,6,7,8,9,10], among others) that must be addressed to strengthen the use of the Code to modern methods of stress analysis and computational techniques [9].

MOTIVATION

Among other types of structures, cone-cylinder junctures are suitable structures to study because of the types of stresses the cone can induce. The cone induces hoop and axial bending stresses in the cylinder and in the cone itself. In practical situations, cone-cylinder junctures are found in support skirts of pressure vessels. Changing the attachment angle α between the cone and the cylinder permits us to investigate the effect of the cone-induced bending stresses on the collapse load.

In 1995, Hollinger and Hechmer [4] presented in the Pressure Vessel and Piping Conference held in Minneapolis a typical support skirt configuration made of a cone attached to a cylinder. They studied this support skirt configuration using different stress assessment (including finite element analysis) and showed interesting relationships between the collapse load and the loads allowed by the Code [2] stress limits. The skirt they examined had a 18° cone-cylinder attachment angle and an arbitrary end-load of 1122*psi* applied on the bottom cross section of the support cylindrical part. They have assessed the stress state in the cone-cylinder juncture using simple hand calculations (which they called "simple beam" approach; $\sigma = F/A \pm Mc/I$) and FEA. In [4], performing a FE elastic-plastic analysis, the end-load was gradually incremented and a lower bound *limit load* was determined.

Taking the ASME Code [2] definition of *limit load* analysis as the intended basis for design, Hollinger and Hechmer [4] concluded that: (a) *allowable load* based on the general primary stress limit (using the stresses obtained from simple "beam approach") was within a 10% margin with respect to the *limit load* results, (b) the membrane and/or bending stress was related to collapse mode, and (c) the "beam approach" gave a more conservative *allowable* end-load than the one obtained with elastic FE stresses.

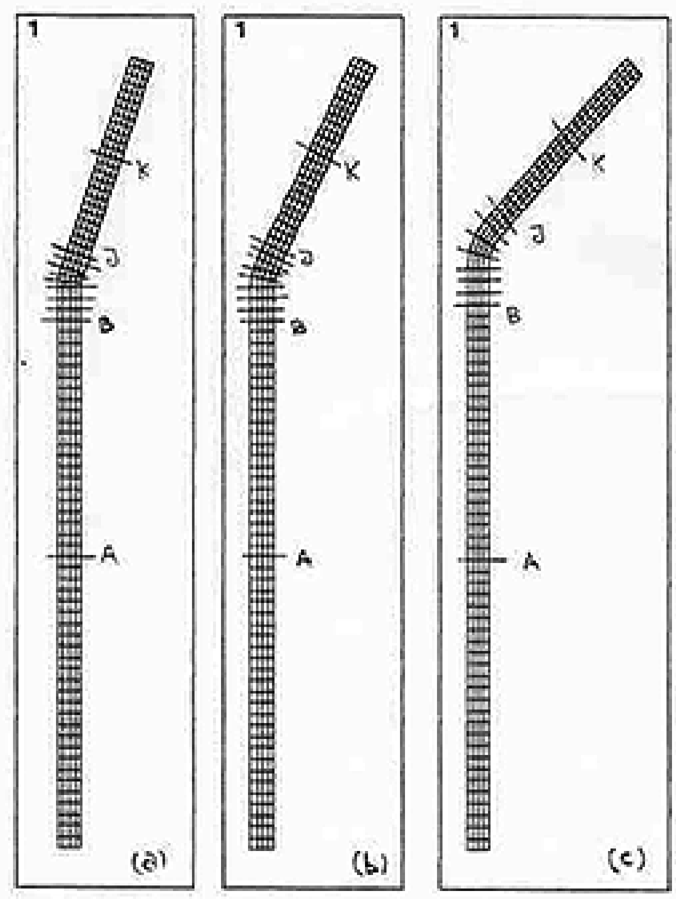

Figure-1 - (a) Skirt with α=18°, (b) Skirt with α=25°, and (c) Skirt with α=45°

To see the extent of the conclusions arrived by Hollinger and Hechmer [4], this paper analyzes three different skirts (with different attachment angle) and a torispherical pressure vessel head.

Using the same strategy employed in reference [4], the *limit loads* of each structure will be obtained. Following the recommendations of Subsection NB in conjunction with different stress assessment and stress classification, we will determine the *allowable loads* of the support skirts and vessel head. It will be possible to observe the influence of the support skirt attachment angle (between cone and cylinder) on the *allowable loads* since such attachment is responsible for the appearance of bending stress. The influence of the bending stress on the failure mode will be observed.

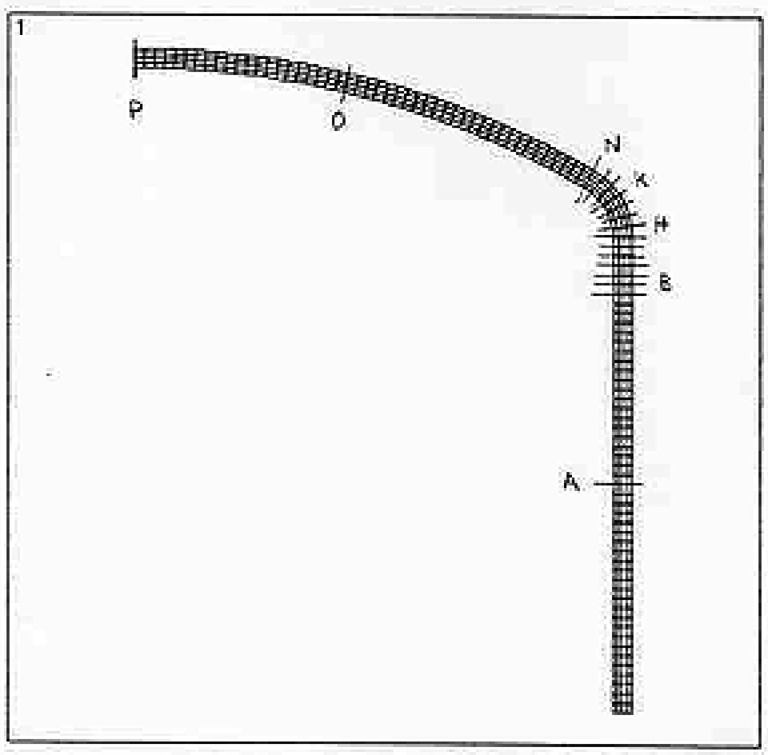

Figure 2. Pressure Vessel and Torispherical Head

SKIRTS AND HEAD GEOMETRIES

This paper analyzes three different skirts and a torispherical head of a pressure vessel. The skirts have three different angles of attachment (α) between the cylinder connected to the cone. The angle α is responsible for the bending stress inherent in the cone-cylinder juncture area. Different α are used so that the impact of increasing bending values on the *limit loads* could be observed. Figure 1 shows the FE meshes of the support skirts.

The internal radius of the cylindrical part of the skirts is R_i=48.64*in* and the thickness t=2*in*. At the end of the cylindrical part an axis-symmetric stress of F=1122*psi* (acting on the inferior cross section of the cylinder) was applied [4]. To simulate a rigid foundation for the support skirt, the cone upper part is completely fixed (actually, in Fig. 1, the cone-cylinder is upside-down). The nodes at the inferior section of the cylindrical part - where the end-load is applied - are coupled to simulate zero rotation.

Due to the geometric transition between cone and cylinder and the presence of an applied stress at the bottom of the cylinder, an attenuation length of $5\sqrt{Rt}$ is considered in our FE model. A blend radius of 2t in the juncture notch cylinder-cone is used so that stress singularity in the notch could be prevented.

For the vessel with the torispherical head, the same internal radius and thickness of the skirt cylinder were used. In this case, however, an arbitrary internal pressure of p=450*psi* is adopted. 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 2 shows the FE mesh of the vessel and its torispherical head.

The material SA533-Gr.B is used for the torispherical head and for the support skirts. It has the following proprieties: Young's Modulus of elasticity $E=29.5\times10^6$*psi*, Poisson's ratio ν=0.3, allowable stress S_m=26,700*psi*, and yielding stress S_y= 40,000*psi* [11].

ELASTIC & LIMIT ANALYSES

The support skirts and the pressure vessel head described in the last section are analyzed using simple hand calculation and also the FEM. The FE program used for such analyses was ANSYS [3]. The end-load of 1122*psi* and the pressure of 450*psi* produce only elastic stresses respectively in the support skirts and in the torispherical head. Figure 1 and 2 also shows the stress classification lines SCL, where the elastic stresses could be linearized.

For the verification of the *allowable loads*, the ASME Code admits the use of elastic analysis and also plastic analysis. The limit analysis is a special case of plastic analysis. To perform the limit analysis, the material is taken 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* (see Figure 3).

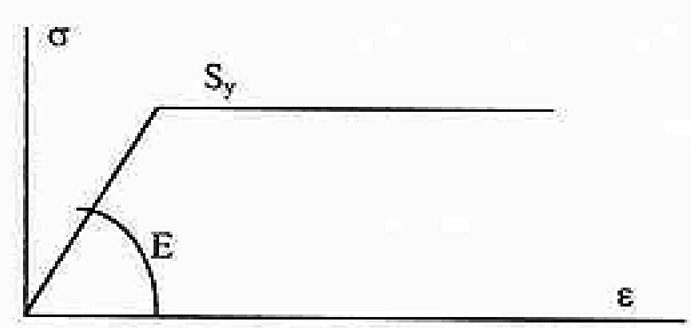

Figure 3. Stress-Strain Curve for Perfect Plasticity

According to ASME Code requirements, the *allowable load* based on limit analysis is defined as two-thirds of the lower bound collapse load. In the ANSYS inelastic analysis, we have adopted the classical bilinear curve and the Von-Mises yield surface flow law. The load steps employed during the plastic analysis of the structures were very small. An end-load increment of 100*psi* was applied for the skirts. For the torispherical vessel, an internal pressure increment of 50*psi* was used. The convergence criterion to stop the inelastic analyses was the default option of the ANSYS program.

A summary of the collapse loads, obtained with the FE models, is presented in Table-A. In that table, one can observe the high influence of the attachment angle α (responsible for the bending stress) on the final collapse load. For the support skirts, the bigger the attachment angle α becomes, the smaller the collapse load is. We observe that the ASME Code [2] considers, in areas of geometric discontinuity, membrane as local and bending as secondary. Taking the induced bending as secondary means that, in the Code, this bending is not related to collapse. As we know, secondary stress must satisfy an imposed strain pattern rather than being able to equilibrate an external load. However, in Table A the bending stress in the cylinder-cone transition is somehow related to the equilibrium since it influences the lower bound *limit load*. Therefore, at least some portion of the membrane + bending stress component, in the juncture area, could be treated as primary [4].

TABLE-A - Collapse loads for skirts and vessel head

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

STRESS ASSESSMENT & ALLOWABLE LOADS

The adequate safety conceded by the limits imposed to general primary membrane ($\leq S_y/1.5$) and primary membrane + bending stress ($\leq S_y$) in the Code is strongly based on the "simple beam" approach. Using simple argumentation based on "beam-approach," the collapse takes place when the axial membrane stress reaches the yielding stress value S_y=40,000*psi*. Considering the safety factor of 1.5, the *allowable* end-loads for the three skirt configurations are then obtained after dividing S_y per 1.5. Thus, for all the skirts, the *allowable* end-load based on simple stress assessment ("beam approach") is 26,700*psi*. For the vessel, the membrane stress in the cylinder is σ=pR/t and in the spherical crown is σ=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×40000÷R, or p=1611*psi*. Applying again the safety factor of 1.5 for the general membrane stress, the *allowable* internal pressure for the "beam-approach" of the vessel is 1074*psi*.

A more precise value for the collapse load can be achieved with FE techniques. The collapse loads are reported in Table A. We notice that our FE model is longer than the model presented in [4] and that we have coupled the vertical displacement of the nodes where the end-load is applied, forcing the rotation of that cross section to be zero.

For the skirt with σ=18°, our FE results confirmed the values reported by Hollinger and Hechmer in reference [4]. For example, in the vicinity of the cylinder-cone juncture, the maximum membrane stress intensity is in SCL-J (see Fig. 1). The stress intensity, there, is dominated by the meridional stress and reaches 1153*psi* which totally agrees with reference [4]. The Code classifies as local (P_L) the membrane stress in the bending region (cylinder-cone junction), and limits local stress to $P_L \leq 1.5S_m$=40,000*psi*. Thus, if the end-load of 1122*psi* causes such a stress value of 1153*psi*, the maximum *allowable* end-load is then 40000×1122÷1153=38924*psi*. For the support skirt with α=25°, the maximum P_L stress intensity is reached at SCL-E (Fig. 1) with P_L=1562*psi* and the maximum *allowable* end-load is 40000×1122÷1562=28732*psi*. Similarly, for the other skirt with α=45° the maximum FE 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 the ASME Code local membrane limit, is 40000×1122÷3066=14637*psi*.

For the pressure vessel, the maximum FE P_L stress intensity is reached at SCL-I (Fig. 2) and the value of P_L=17130*psi* gives an *allowable* internal pressure equal to 40000×450÷17130=1050*psi*.

Continuing with our finite element results, the maximum membrane + bending (P_L+P_B) stress intensities are achieved in SCL-F, SCL-F again, and SCL-G (Fig. 1). That is; the maxima of (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 $P_L+P_B \leq 1.5S_m$=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=486*psi*. However, the ASME Code considers the bending stress in the cylinder-cone transition as secondary. Categorizing the membrane + bending as primary + secondary, the Code limits $P_L+P_B \leq 3S_m$=80000*psi*. Therefore, in such a case the *allowable* end-loads are the double of the values obtained when limiting P_L+P_B to $1.5S_m$, i.e.; 31616*psi*, 25964*psi*, and 16780*psi* for the skirts (respectively with α=18°, 25°, and 45°), and 972*psi* for the pressure vessel. It should be noted that if P_L+P_B exceeds $3S_m$=80000*psi* appropriate penalty factors should be applied for the fatigue evaluation, and appropriate verification should be performed to prevent ratcheting.

TABLE - B: Allowable loads for the skirts with different attachment angles.

	End - Loads PSI - Cross Section					
	Cone with 18°		Cone with 25°		Cone with 45°	
	At Limit	Allowable	At Limit	Allowable	At Limit	Allowable
"Beam approach"	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: Allowable loads for the skirts and the sections for stress verification.

		End-Load								
		Cone with 18°			Cone with 25°			Cone with 45°		
		At Limit	Allowable	SCL	At Limit	Allowable	SCL	At Limit	Allowable	SCL
"Beam approach"		40000	**26700**	----	40000	**26700**	----	40000	**26700**	----
FEA	M as PM	38425	25616	A	37809	25206	A	36019	24013	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	38425	**25616**	A	37809	**25206**	A	36019	24013	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**	----

TABLE - D: Allowable loads for the vessel and the sections for verification.

		Internal Pressure		
		At Limit	Allowable	SCL
"Beam Approach"	Cylinder	1611	**1074**	------
	Head	3223	2148	------
FEA	M as PM	1510	1008	A - Cylinder
	M as PL	1050	1050	I - Knuckle
	M+B as PL+PB	486	**486**	I - Knuckle
	M as PM	1597	1065	O - Crown
	M as PL	1205	1205	N - Crown
	M+B as PL+PB	790	790	N - Crown
FEA	M as PM	1008	**1008**	A - Cylinder
	M as PL	1050	1050	I - Knuckle
	M+B as P + Q	972	---	I - Knuckle
	M as PM	1597	1066	O - Crown
	M as PL	1205	1205	N - Crown
	M+B as PL+PB	790	790	N - Crown
FEA, Limit Load		1200	**800**	-----

DISCUSSIONS & SOME CONCLUSIONS

The discussions and conclusions that follows are based on the SCL's analyzed in the last section and drawn in Figure 1 and 2. Those SCL's were selected on the basis of the authors' expertise. More SCL's could be studied but those are sufficient because they are located in strategic areas. In general, it is advisable to consider as much SCL's as possible and also (before breaking up the stresses and making the stress classification) take into account the recommendations and the practical suggestions given in reference [1,5,11].

The results of the last section with more details are reported in Tables B and C for the skirts and in Table D for the pressure vessel. Observing those tables one can concluded that the "beam approach," the elastic FEA, and the lower bound FE approaches give very distinct *allowable loads*. It is clear from the calculations that the stress classifications and the stress assessment methods influence the determination of the *allowable loads*.

If we consider the Code definition of *limit load* analysis as the basis for design, and compare the *allowable load* obtained from the *limit load* FE analysis with the other *allowable loads*, it can be concluded that: (a) the simple "beam approach" gave always greater *allowable loads*; (b) with FEA, categorizing the membrane near structural discontinuity as P_L also resulted in greater *allowable loads*; (c) taking membrane + bending FE stresses, near structural discontinuity, as primary produced smaller *allowable loads*; (d) considering FE membrane + bending at structural discontinuity as primary + secondary resulted in greater *allowable loads*. Notice that P_M, P_L, and P_B limits are related to plastic collapse, while P+Q limit is related to fatigue and incremental plastic deformation.

Observing the values in Tables B and C, some more conclusions can be reached: (a) for the skirt with α=18°, our results confirm the observations and the conclusions reached by Hollinger and Hechmer [4]. But we observe that in this case, both the "beam approach" and the elastic FE results (away from the cone-cylinder juncture) approximate the limit collapse end-load controlled by the higher stresses at the juncture. In view of the collapse mechanism and place, it is unlikely that stresses computed in SCL-A, away from juncture, control the stresses allowed at the juncture. (b) for the skirt with α=45°, the stress assessment based on elastic FE (and M as P_L) shows an *allowable load* 17% greater than the FE *limit load* (14637*psi* against 12467*psi*). The *allowable load* based on the "beam approach" is 114% greater than the FE *limit load* (26700*psi* against 12467*psi*). In this case, the elastic FE stress at SCL-E approximates the plastic collapse in the skirt much better than the "beam approach." (c) for the skirt with α=25°, neither the "beam approach" nor the elastic FEA gave *allowable loads* close to the FE *limit load*: 26700*psi* (30% greater) and 25206*psi* (23% greater) compared to 20466*psi*.

For the pressure vessel, similar results are observed in Table-D: (a) categorizing membrane + bending as primary is too conservative. (b) the *allowable load* obtained with the simple hoop stress in the cylinder (1074*psi*) and the elastic FEA (1008*psi*) gave comparable results between them but greater (34% and 26%, respectively) than the FE *limit load* (800*psi*). (c) surprisingly, classifying M+B as primary in the transition knuckle-crown region gave an *allowable loads* (790*psi*) closed to the FE *limit load* (790*psi*).

In front of the varieties of values obtained for the *allowable loads* (specially in the skirt with α=25°) it is recommended that the cases here studied should be looked at with more concern. The *limit loads* found with the FE models were based on the assumption of perfect plasticity as suggested by the Code and were taken as the intended base for design (strain-hardening assumption would give larger *limit loads*). The *limit loads* represented lower bounds bellow which the structures were considered safe. The cases here studied showed that depending on the type stress assessment (hand calculations, elastic FEA or FE limit load) and on the stress categorization (primary or secondary), the designer could be taken to non-conservative designs or at least to design with very small safety margins. Needless to say that it is very costly to perform limit load analysis every time we want to be sure about an *allowable load* of a nuclear equipment. Although it seems to be the safest and cheapest way other than experiment. On the other hand, it appears questionable to compute *allowable loads* based on "beam approach" reasoning, unclear stress categorization, and simple elastic stress assessments.

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