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Table 4 Optimum designs from PANDA2 suitable for analysis by STAGS (dimensions in inches)

.	**Case 1** Perfect, no Koiter, ICONSV=1	**Case 2** Imperfect, no Koiter, yes change imperfection amplitude, ICONSV=-1	**Case 3** Imperfect, no Koiter, yes change imperfection amplitude, ICONSV=0	**Case 4** Imperfect, no Koiter, yes change imperfection amplitude, ICONSV=1	**Case 5** Imperfect, yes Koiter, yes change imperfection amplitude, ICONSV=1	**Case 6** As if perfect, no Koiter, Nx=-6000, sbar=120 ksi ICONSV=1	**Case 7** Imperfect, no Koiter, no change in imperfectio n amplitude, ICONSV=1
Variable Name	Optimum Design	Optimum Design	Optimum Design	Optimum Design	Optimum Design	Optimum Design	Optimum Design
B(STR)	0.75519	0.93500	0.93500	0.98170	0.93500	0.93500	1.5708
B2(STR)	0.075519	0.093500	0.093500	0.0981710	0.093500	0.093500	0.15708
H(STR)	0.39795	0.57079	0.58395	0.63651	0.55261	0.55330	0.92254
W(STR)	0.35593	0.38639	0.36056	0.39946	0.29593	0.36761	0.64833
T(1)(SKN)	0.030240	0.033988	0.033795	0.034878	0.039964	0.044110	0.048160
T(2)(STR)	0.019897	0.028540	0.029197	0.031826	0.027631	0.033536	0.046127
T(3)(STR)	0.022209	0.026779	0.029411	0.022835	0.032576	0.024673	0.033702
B(RNG)	6.25	9.3750	8.3333	8.3333	9.3750	8.3333	15.000
B2(RNG)	0.0	0.0	0.0	0.0	0.0	0.0	0.0
H(RNG)	0.52160	0.79425	0.75877	0.79978	0.77659	0.92137	0.86341
W(RNG)	0.17891	0.10000	0.12313	0.24075	0.31922	0.35255	1.0804
T(4)(RNG)	0.026080	0.039713	0.037939	0.040078	0.038830	0.046069	0.043170
T(5)(RNG)	0.021847	0.097842	0.086763	0.029339	0.037873	0.017627	0.054020
WEIGHT	**31.81 lb**	**39.40 lb**	**40.12 lb**	**40.94 lb**	**41.89 lb**	**46.83 lb**	**56.28 lb**
Critical margins from PANDA2, Table 5	**1, 6a,b, 23a,b, 26, 44, 55, 56, 57, see Table 10**	**1, 3, 6a,c,e, 10, 23a, 26, 47, 55, 56, 57, see Table 6.**	**1, 3, 6a,c,e, 10, 23a, 26, 47, 55, 56, 57**	**1, 3, 6a,c,e, 10, 23a,e, 25, 26, 44, 47, 55, 56, 57**	**1, 3, 6a,d, 10, 11, 23a, 44, 47, 55, 56, 57**	**1, 3, 6a,c,e, 10, 11, 23a, 25, 26, 44, 47, 55, 56, 57, 58**	**1, 3, 6a,c,e, 10, 11, 23e, 25, 26, 44, 46, 55, 56, 57, 58**

Almost critical margins from STAGS and mode of elastic collapse	**1, 6a, 44, Collapse was not computed**	**1, 6a, 47, Stringer sidesway and first bay collapse at PA=1.04**	**1, 6a, 47, Stringer sidesway and first bay collapse at PA= 1.05**	**1, 6a, 47, Stringer sidesway and first bay collapse at PA=1.08**	**1, 6a, 47, Stringer sidesway and first bay collapse at PA=1.13**	**1, 6a, 11, 44, 47, Axisym-metric edge collapse at PA=0.970; rv(edge)=0 on 2 curved edges.**	**1, 6a, 11, 47, Stringer sidesway, first,middle and last bay collapse at PA= 1.22(–) PA= 1.15(+)**
Tables & Figures pertaining to the case	**Table 10, Figs. 3, 33-41**	**Figs. 8–32**		**Figs. 1a-c, 2, 4-7, 42-65**	**Table11, Figs. 66-71**	**Figs. 72-74**	**Figs. 75-80**
Comments	**This shell is not practi-cal because no one can fabricate a perfect structure.**	**With this option you MUST check the results via a general-purpose code such as STAGS.**	**With this option you are strongly URGED to check result with use of a general-purpose program.**	**This option may lead to shells with local skin & stringer bending & therefore possibly excessive stresses.**	**This is the best option if you do not plan to check PANDA2 designs. Even so, you SHOULD check them.**	**This widely used option generates a heavy shell. PANDA2 cannot predict axi-symmetric collapse.**	**This option is too con-servative, in my opinion. The imperf-ection can probably be detected easily.**

The optimum designs obtained in this study are listed in Table 4. **Table 4 is the most important item in this paper**. Seven cases are listed in order of increasing weight of half (180 degrees, see Table 1) of each of the optimized cylindrical shells. In Table 4 there appear variables, words, and phrases that must be defined:

1. **"perfect"** and **"imperfect"** have obvious meanings. All of the "imperfect" shells have a general buckling modal imperfection with **initial** amplitude, Wimp = +/- 0.25 inch specified by the PANDA2 user (and possibly changed later by PANDA2 as described in Item 4 below).

2. **"As if perfect"** (Case 6) means that the shell is optimized as if it were perfect (general buckling modal imperfection amplitude, Wimp = 0) and the applied load and maximum allowable stress **are doubled** to compensate for initial imperfections. This doubling is derived from the assumption for this particular configuration that the effect of a maximum allowable general buckling modal imperfection with amplitude Wimp equal to one per cent of the shell diameter (typical ASME allowable: Wimp = 0.25 inch in the other cases explored here) is to cause the shell to fail at half the load that its perfect equivalent would fail at.

3. **"No Koiter"** and **"yes Koiter"** refer to local "postbuckling" (or more generally, local bending perhaps

without any instability) of the panel skin between adjacent stiffeners and local bending of the stringer parts. "No Koiter" means that the "Koiter" branch of PANDA2 [1C, 15] is skipped (Koiter branch is turned **off** in the *.OPT file) and the factor of safety for local buckling is set to unity (actually 0.999 in order to prevent PANDA2 from automatically raising the factor of safety to from 1.0 to 1.1). "Yes Koiter" means that the "Koiter" branch of PANDA2 is entered (Koiter branch is turned **on** in the *.OPT file), that is, local "postbuckling" (bending) states are computed; **the factor of safety for local buckling is set equal to a small number (0.1 in this case, which causes the evolution of the design to be unconstrained by local bifurcation buckling margins);** and the shell is optimized accounting for local bending deformation between adjacent stiffeners and local bending deformation of the stringer parts. **Local buckling (bending) at or below the design load raises the maximum effective stress** because of short-wavelength bending of the skin between stiffeners and of the stringer parts. (See Fig. 7, for example). Local buckling (bending) lowers inter-ring and general buckling load factors because a locally buckled (bent) skin is less stiff in an average sense than an unbuckled (unbent) skin ([24], and see Table 12 in this paper). Corresponding to the optimum design in Case 5 ("yes Koiter") the skin-stringer module (similar to that shown in Fig. 4 but with different cross section dimensions) has not actually buckled locally in the sense that its local buckling load factor is less than the design load, Nx = -3000 lb/in. Indeed, in Case 5 the local buckling load factor is about 1.3 times the design load (approximately Nx = -4000 lb/in). For the Case 4 design with use of the "yes Koiter" option, a very small initial local buckling modal imperfection (equal to five per cent of the shell skin thickness) grows under application of the design load as displayed in Figs. 4 and 5, giving rise to significant local bending stress in the skin and stringer parts at the design load. This local bending stress adds to the overall membrane compression, causing the stress constraints to become critical with "yes Koiter" sooner than with "no Koiter". That is why the "yes Koiter" optimum design (Case 5) is heavier than the "no Koiter" optimum design (Case 4) for this particular configuration, material, and loading. **The additional stress generated by local bending/buckling of the sort depicted in Figs. 4 and 23a, for example, is accounted for with the "yes Koiter" option and ignored with the "no Koiter" option.**

Figures 4 – 7 demonstrate the local bending phenomenon and its effect on stress margins. These figures correspond to the optimum configuration identified as Case 4 in Table 4, the shell optimized with "no Koiter". However, Figs. 4 and 5 are derived from an analysis of the Case 4 configuration with the Koiter branch turned on in the *.OPT file, that is, with the "yes Koiter" option. Figure 4 shows a single skin-stringer discretized module and its local bending deformation at four levels of the applied axial compression Nx (amplitude of bending deformations greatly exaggerated). Figure 5 shows the maximum normal displacement **w** midway between stringers as a function of applied axial compression. Figure 6 demonstrates the effect of omitting (NO KOITER) and including (YES KOITER) the contribution of local bending to the total maximum effective (von Mises) stress in the Case 4 stiffened shell. The optimum design obtained with "no Koiter" (Case 4) is no longer feasible if the Koiter branch of PANDA2 is turned on ("yes Koiter") for this configuration. At the design load, Nx = -3000 lb/in, four of the YES KOITER stress margins are significantly negative. Figure 7 shows a STAGS model of the Case 4 configuration with nodal points concentrated where global bending of the imperfect shell is maximum inward. The effect of local bending of the panel skin in the region of highest nodal mesh density is clearly visible, and the maximum effective stress, sbar(max) = 68.22 ksi, significantly exceeds the maximum allowable effective stress, sbar(allowable) = 60 ksi.

Typically, designs in which local postbuckling is permitted during optimization are lighter than those in which it is not, especially if in-plane shear Nxy is a significant component of the applied loading. Such "typical" designs are more lightly loaded shells in which in-plane shear loading Nxy may be significant and/or shells made of a material with a higher allowable stress than is so for the cases included in this paper. For the cases in this paper (Case 4 and Case 5) the "yes Koiter" option (Case 5) leads to a heavier optimized shell than does the "no Koiter" option (Case 4).

4. **"Yes change imperfection"** means that **Strategy 2** is followed in the PANDA2 main processor. (See Section 15.1 and Table 13 in [1K] for the definition of "Strategy 2"). In **Strategy 2** the **initial (user specified) amplitude** of the general buckling modal imperfection is multiplied by the ratio,

(axial wavelength of **actual** critical general buckling mode)/(**user-specified** length of general buckling mode) **(11.1)**

during all computations. This strategy is derived from the assumption that imperfections of given amplitude with shorter axial wavelengths are easier to detect than those with longer axial wavelengths. They are therefore easier to control during manufacture. Here it is assumed that what is detected in an initially imperfect cylindrical shell is actually the error in **axial slope** of the imperfect generators. For a **given** (specified) allowable error in axial slope of an imperfect generator, that is, the minimum detectable error in axial slope, the **amplitude** of that same initial imperfection shape is proportional to its axial wavelength (inversely proportional to the number of axial halfwaves **m** in the imperfection shape). **With Strategy 2 "turned on" ("yes change imperfection") the shorter the axial wavelength of the initial general buckling modal imperfection the smaller its amplitude**. For axially compressed cylindrical shells the critical general buckling mode usually has several axial halfwaves **m**. If the PANDA2 user chooses the specified axial length of the initial general buckling modal imperfection to be equal to the axial length of the shell, which is in the range of the recommended input in the *.OPT file for the particular cylindrical shells featured in this paper (Table 1), PANDA2 will reduce the user-specified imperfection amplitude by a factor equal to the ratio defined above (Expression 11.1) if Strategy 2 is chosen. **Strategy 2 is based on the assumption that one general buckling modal imperfection is approximately equivalent to another if their maximum axial slopes are equal**.

Notice that the previous paragraph does not mention PANDA2 automatically changing the initial user-specified general buckling modal imperfection amplitude based on possibly easily detectable errors in **circumferential slope** of the wall of the imperfect shell. Only an error in **axial slope** is mentioned. It turns out that, for the particular cases explored in this paper, the error in axial slope is less than that in circumferential slope. For example, if the critical general buckling mode has **(m,n)**critical = (4 axial, 6 circumferential) halfwaves, then the axial halfwavelength is 75.0/4.0 = 18.75 inches and the circumferential halfwavelength is **pi** x **r**/6.0 = **pi** x 25.0/6.0 = 13.09 inches. For a given user-specified amplitude of general buckling modal imperfection, the error in axial slope is 13.09/18.75 = 0.698 or about 70 percent as large as the error in circumferential slope for **(m,n)** = (4,6).

A more systematic way to approach **Strategy 2** (allowing PANDA2 to change the imperfection amplitude based on wavelength of the imperfection) would be to have the PANDA2 user supply **two** new input data: 1. minimum detectable error in axial slope and 2. minimum detectable error in circumferential slope. Then PANDA2 would compute the amplitude of the general buckling modal imperfection. This computed amplitude would depend on **(m,n)**critical. The amplitude would be that which causes the imperfection to be at the threshold of detectability for either axial or circumferential error in slope, whichever barely detectable error in slope leads to the smallest amplitude. The PANDA2 user would supply two new inputs, one for minimum detectable error in axial slope and the other for minimum detectable error in circumferential slope, because the threshold of detectablility of error in axial slope might be different from that of error in circumferential slope. It is logical to assume that an error in axial slope would be easier to detect than the same error in circumferential slope because the axial slope of the perfect cylindrical shell is zero and the quality control engineer is looking for deviations from zero.

The maximum axial slope corresponding to a general buckling modal imperfection of the form, Wimp(x,theta) = A.sin(**m**.pi.x/L)sin(**n**.theta), is A.**m**.pi/L. The minimum **detectable** non-zero axial slope in the cases explored in this paper is given by **0.25.pi/L radians** because the user-specified initial amplitude of the general buckling modal imperfection is 0.25 inch and the user-specified axial halfwavelength is 75 inches, the length of the cylindrical shell, that is, **m** = 1. **The writer does not know what is a reasonable minimum detectable error in axial slope in practice. Hence, the results in this paper labeled "yes change imperfection" in Table 4 are presented for the purpose of demonstration only, not as a guideline for the reader to use in the actual fabrication of shells.**

"No change imperfection" means that **Strategy 1** [1K] is followed; the **user-specified imperfection amplitude is not modified**. Unless the PANDA2 user specifies an initial general buckling modal imperfection with a very small amplitude, optimum designs obtained with the "no change imperfection" option are probably too conservative (too heavy) because the optimum design thus obtained by PANDA2 usually ends up with an imperfection (general buckling mode with several axial halfwaves) with an amplitude that could easily be detected. Therefore such an imperfect shell would have to be repaired or a shell with such an imperfection would have to be discarded. Also, with "no imperfection change" specified in the *.OPT file, design margins often change drastically from design iteration to iteration, a phenomenon described in Section 15.1 and Fig. 20 of [1K] that makes it difficult to find a "global" optimum design.

5. The index, **ICONSV**, is defined above in **Item 676** of Section 9.0. ICONSV = -1 denotes the least conservative model and ICONSV = 1 denotes the most conservative model. See Fig. 99 for an example of how several of the margins in Case 4 vary with ICONSV = -1, 0, and +1.

6. The critical margins from PANDA2, enumerated in the row in Table 4 just below that which lists the shell weights, are defined in **Table 5**. The string, "SANDERS" in Table 5 indicates that Sanders' shell equations [25] are used in the computations. In Table 5 "M" or "m" is the number of axial halfwaves; "N" or "n" is the number of circumferential halfwaves except in Margins 6 and 23 where "n" means nodal point number in a module; "slope" is the slope of the buckling nodal lines as shown in Fig. 9 of [1B]; "FS" is the factor of safety; "STR" = stringer; "SKN" = panel skin; "RNG" = ring; "Dseg" = segment number in the discretized skin-stringer single module model (Fig. 4): Dseg=1 = panel skin in left-hand part of Fig. 4; Dseg=2 = base under the stringer where the stringer web root intersects the panel skin; Dseg=3 = stringer web; Dseg=4 = outstanding stringer flange; Dseg=5 = panel skin in right-hand part of Fig. 4. "z" is the thickness coordinate in a shell wall or stiffener segment wall; "MID" means "midway between rings" (same as Sub-case 1); "RNGS" means "at ring stations" (same as Sub-case 2); "Iseg" means skin-stringer or skin-ring single module segment number (PANDA-type model [1B] , not discretized module [1A]): Iseg=1 = panel skin; Iseg=2 = base under the stiffener (either stringer or ring); Iseg=3 = stiffener web; Iseg=4 = stiffener outstanding flange. "ROOT" means "at web root" (where a stiffener web intersects the panel skin); "allnode" means "at all nodal points in the panel skin"; "C=0" means "slope of buckling nodal lines=0"; "NOPO" means "neglecting local postbuckling effects". "V(i)" is the ith variable. See Table 2 for definitions of the variables, V(i), i = 1 to 13.

7. The quantity "**PA**" that appears in the row in Table 4 pertaining to STAGS predictions is the applied load factor. PA = 1.0 corresponds to the design load, that is, the applied load specified in the PANDA2 input file, *.OPT. The design load is Nx = –3000 lb/in axial compression in all cases except Case 6, for which the design load is Nx = –6000 lb/in.

8. The (-) and (+) that occur in Case 7 immediately following the collapse load factors PA predicted by STAGS, PA= 1.22(-) and PA= 1.15(+), refer to the sign of the amplitude of the buckling modal imperfection, Wimp = - or + 0.25 inch.

9. The string, “stringer sidesway”, means “bending-torsional” buckling or “stringer rolling” [1B] of the type shown in Fig. 20a.

10. All the PANDA2 and STAGS models for which results are listed in Table 4 are based on the assumption that the **material remains elastic**. Therefore, the collapse load factors, PA, listed in Table 4 in the row pertaining to STAGS predictions would be somewhat lower if plastic flow were included. See Section 14 for a discussion of the effects of accounting for elastic-plastic material behavior in STAGS models of some of the shells optimized by PANDA2 (Cases 2, 4, 5, 7).