Static Stress Analysis on the Large Capacity Thin-Wall UF_6 Storage Cylinder Intended To Be Used in Transport*

C.K. Chung, J.L. Frazier, D.K. Kelley
Lockheed Martin Energy Systems, Inc.

INTRODUCTION

Large diameter (48-inch) cylinders are used for the storage and transport of depleted uranium hexafluoride (DUF_6). This DUF_6 is the residual material from uranium enrichment operations from the three gaseous diffusion plants in the United States. Gaseous diffusion operations since the mid 1940s have produced an inventory of approximately 50,000 of these cylinders with nominal content of 14 tons DUF_6.

The large diameter cylinders were originally designed and fabricated for temporary storage of DUF_6. The cylinders were fabricated from pressure vessel grade steels according to the provisions of the *ASME Boiler and Pressure Vessel Codes* (Section VIII, Div. 1, 1992). These cylinders were constructed in thick wall (5/8-inch wall thickness) and thin-wall (5/16-inch wall thickness) versions.

These cylinders are currently stored in outdoor arrays in a two tier assembly with the contacts being shell to stiffener rings. The lower tier cylinders are supported by chocks. The cylinders are subjected to general and accelerated atmospheric corrosion and to possible damage from stacking and other handling operations.

Finite element stress analysis and fracture mechanics methodology have been utilized to assess cylinder structural integrity during handling operations and degradation of the cylinder shell resulting from corrosion or other damage mechanisms.

ASSUMPTIONS

The mass of the DUF_6 content is assumed in a line element, which coincides with the centerline of the cylinder. The centerline element distributes the lumped weight of the content on the selected nodal points of the shell by connection with weightless line

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elements. Each node on the centerline element connects 13 nodes on the shell which are in the plane of the lower half portion of the cylinder separated by 15 degrees.

The shell to stiffener ring contacts are each modeled with nine rigid links. They form an area of 1 inch (ring thickness) in the axial direction by 3/8 inch on the cylinder circumference.

The contact elements between the lifting lug and the stiffening ring are treated as a continuous body.

The cylinder pressure vessel grade steels of construction are assumed nominal values of 30×10^6 lbs/in^2 and 0.3 for the modulus of elasticity and a Poisson's ratio, respectively.

MODELING OF THE FINITE ELEMENT PROBLEMS

The cylinder assemblies are fabricated of 5/16-inch thick plates for the shell, three 2-1/2 × 1-inch thick stiffener rings, and two 2:1 ellipsoidal heads. Since the geometry and loadings are symmetrical with respect to a plane containing the centerline of the middle stiffener ring, a half of the cylinder is modeled for the analysis.

The right hand coordinate system of the computer model defines the direction of gravitational acceleration as negative in the Y-axis, and the centerline of the cylinder axis is in the X-axis. The static analyses are performed with the consistent mass model by the Engineering Mechanics Research Corporation (EMRC), Display III of the NISA II program (1994).

The model has three-dimensional solid, shell, and beam elements. The centerline elements of the cylinder, beam elements, carry the entire content weight of the cylinder. The center line elements distribute the weight on the nodes of the shell elements by connecting bar elements from nodes on the shell elements to the centerline elements.

The shell and stiffener ring contacts are modeled with rigid links that express the deflection at one end as kinematically equivalent deflections of the other end. The wooden chocks are modeled both as rigid supports and as flexible supports.

DISTRIBUTION OF THE CONTENT WEIGHT ON THE CYLINDER SHELL

The method of calculation of the density for the content, d_{bar}, and for the shell, d_{shell} follows. Maximum volume of the content, V, for the type 48-G or 48-OM cylinder is 139.72 ft^3. The equivalent length, L_e, of the cylinder with an inside diameter of 48 inches to contain a volume, V, is

$$V = 139.72 * 1728 \text{ in.}^3 = 241{,}436.16 \text{ in}^3 = \pi * (r_i)^2 * L_e,$$

therefore,

$$L_e = 241{,}436.16/(3.14159 * 24^2) = 133.42288 \text{ in., and}$$
$$d_{shell} = 490/(1728 * 32.174 * 12) = 7.34456 * 10^{-4} \text{ lbf-s}^2/\text{in.}^4.$$

The models have bar elements with cross sectional areas of 1.000 square inch to represent the UF_6 content weight.

d_{bar} = 28,000 lbf/(cross sectional area * L_e) = 28,000/(1.000 * 133.42288)
= 0.54355 lbf-sec^2/in^{-4}.

STRESS ANALYSIS

Location of the Wooden Support Chocks Versus Maximum Stress for Single Tier Storage Cylinders. The stresses have been calculated for the shell thickness of 0.250-inch cylinders at different support locations of the wooden chocks. One wooden chock on the half cylinder has been modeled with rigid restraints and with flexible restraints with the spring rate of 1.0×10^6 lbs/in. The summary of computations with the maximum von Mises-Hencky (M-H) stress versus the location of the chock is provided on Table 1 and plotted on Figure 1. It can be concluded that the location of the chock is better outside and close to the stiffener ring.

For the three computations at the top of Table 1, FS250H, FWS250, and FWR250, the chock locations are the same, but the type of support or fillet weld with the stiffener rings are different. The flexible support on the fillet-welded model has higher M-H stress, and the rigid supported model has lower M-H stress than that of the flexible one.

Table 1. Location of Chocks versus von Mises-Hencky Stress for Single-Tier Cylinders

File Name	Distance between Center Lines of the Ring and Chock[1] (inch)	Wooden Chock Model (R for Rigid Support, F for Flexible Support)	Fillet Weld is Modeled at the Plate and Ring Contact.	Maximum von Mises-Hencky Stress (ksi)
FS250H	-5.500	F	No	6.36
FWS250	-5.500	F	Yes	6.72
FWR250	-5.500	R	Yes	5.71
FWR25a	-6.500	R	Yes	6.40
FWR25b	-7.500	R	Yes	6.68
FWR25d	-9.500	R	Yes	7.25
FWR25g	-12.500	R	Yes	8.96
FWR-S0	5.500	R	Yes	9.49
FWR-Sa	6.500	R	Yes	11.99
FWR-Sb	7.500	R	Yes	11.66
FWR-Sc	8.500	R	Yes	10.65
FWR-Sd	9.500	R	Yes	9.70

[1] The cylinder is supported by wooden chocks (7-inch width by 8.5 inches high). The negative sign denotes chock locations outside the ring; the positive sign indicates distances inside the ring (toward the center of the cylinder).

Maximum Stress Versus Location of the Stacked Cylinder. The two-tier stacking array forms an isosceles triangle with two 52.5 degree angles. Figure 2 represents the maximum M-H contact stress for the wall thickness of 0.250 inch for horizontal offset

Table B

Position of the Stacked Cylinder versus Stress

File Name	Offset distance between two cylinders(1) [RtR] (inch)	Location of the wooden chock(2) (inch)	Fillet weld is modeled at the plate and ring contact	Maximum Hencky von Mise's Stress (ksi)
S6L188FW	-6.000	-4.000	YES	28.74
S6L250FW *	-6.000	-4.000	YES	21.31
S6L312FW	-6.000	-4.000	YES	16.46
S6La25FW	-6.000	4.000	YES	21.80
S6Lb25FW	-6.000	7.500	YES	22.00
S6L250	-6.000	-4.000	NO	23.21
S4L250FW *	-4.000	-4.000	YES	20.41
S2L250FW *	-2.000	-4.000	YES	17.53
S25L25FW *	-1.250	-4.000	YES	16.03
S25L25	-1.250	-4.000	NO	43.05

(1) The horizontal offset distance between bottom cylinder and the top cylinder. The negative distance denotes a negative x-direction between the top and bottom cylinder.

(2) The bottom cylinder is supported by the wooden chocks (7" width by 8.5" high). It is assumed that the supporting region of the chock is 5.000" width saddle symmetrically with respect to the centerline of the width. A negative sign denotes that the centerline of the chock is outside (cylinder head side) from the centerline of the ring. A positive sign denotes the direction toward the center of the cylinder.

(*) These points were used for the Figure 2.

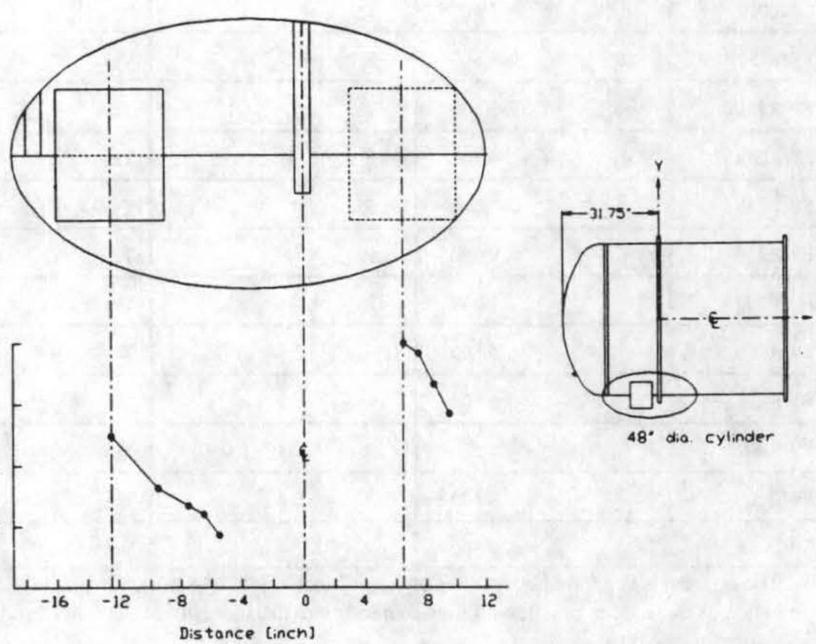

Figure 1 Location of Chocks versus von Mises-Hencky Stresses for Single Cylinders

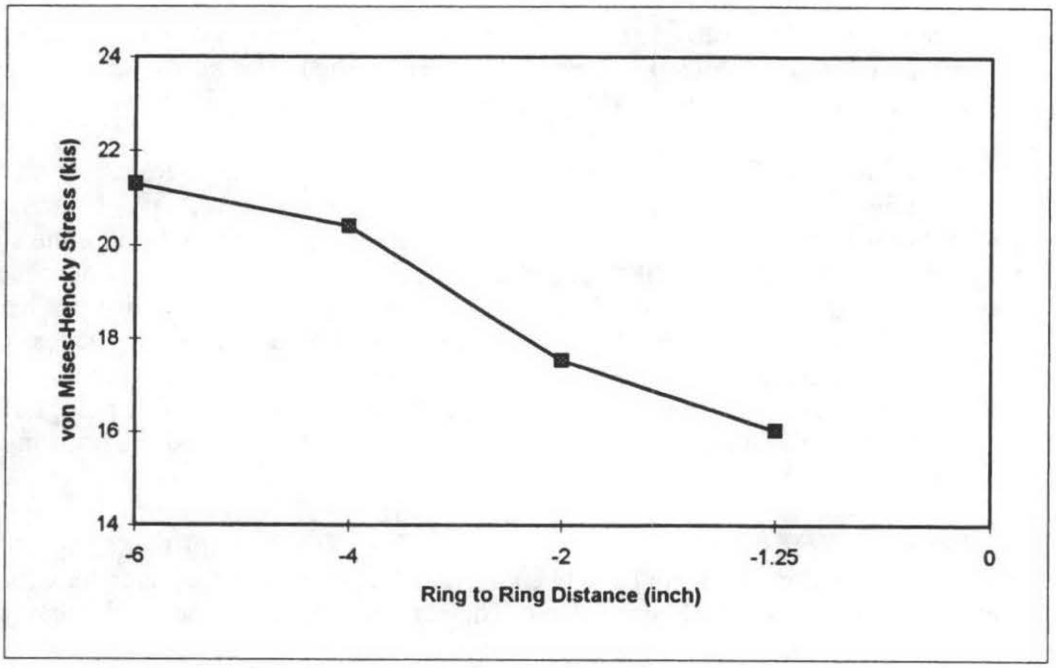

Figure 2 UF6 Cylinder Stacking Stress

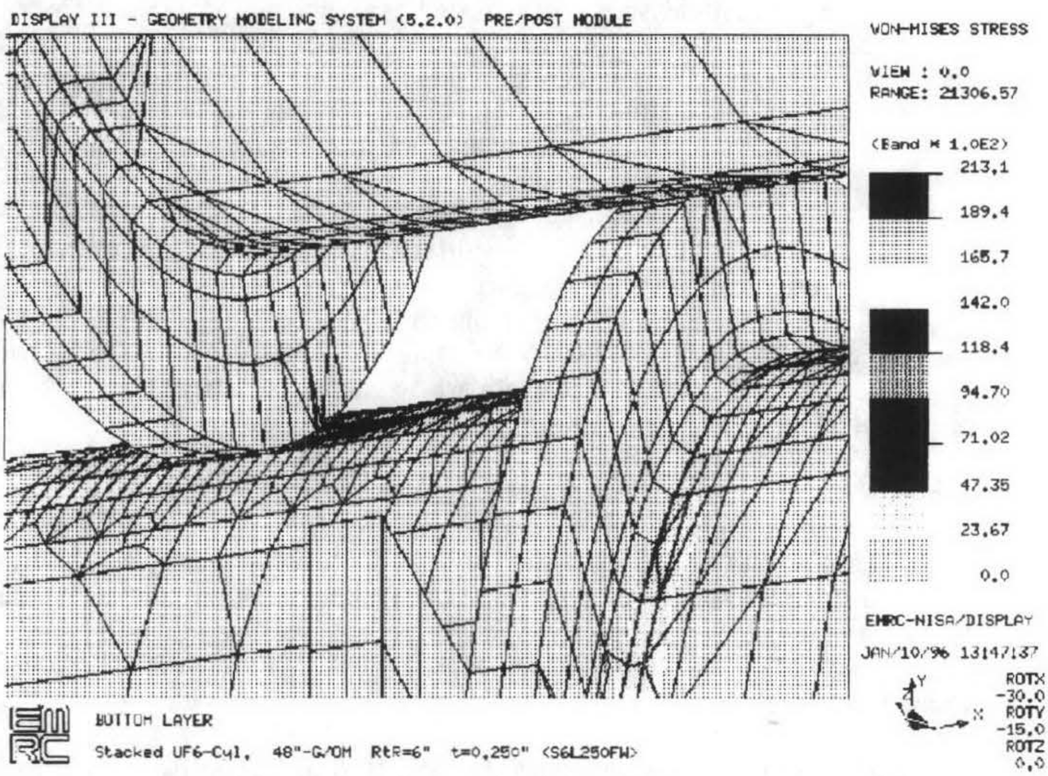

Figure 3 von Mises-Hencky Stresses for Stacked Cylinders distance between the bottom and the top cylinder. The maximum stress occurs at the contact point, and it is localized because of the thin shell. The maximum stress becomes higher if the offset distance is greater in the negative direction from the bottom cylinder.

The model with the fillet weld yields relatively lower contact stresses as the loading point is moved closer to the stiffener. The highest stress is at the contact points, and it is inversely proportional to the shell thickness. As an example, from Table 2 the stresses of the files S6L250 and S6L250FW (Figure 3) are 23.21 ksi and 21.31 ksi, respectively; those of S25L25 and S25L25FW are 43.05 ksi and 16.03 ksi, respectively. The file names with "FW" denote that the model has a fillet weld of 0.250 inch on both sides of the stiffener. The file names with "S6L" and "S25L" represent distances of 6.000 and 1.250 inches, respectively, between the center lines of the bottom ring and top ring. The model S25L has a face to face distance of 0.250 inch between the top and the bottom cylinder stiffener.

Lifting of Breached Cylinders. Stresses have been analyzed for the lifting of four breached UF_6 cylinders. Lifts have been analyzed using lifting lugs, a crane with straps, and the Raygo Wagner cylinder carrier. The summary of load cases and stresses are provided in Table 3.

Cylinder stresses were computed by FEM. A uniform shell thickness of 0.250 inch was used for the calculations except for the thinned regions adjacent to the breached areas of the cylinders. The thinned region was modeled as a border around the breach having a thickness of the thinnest measurement reported as shown by "Min. Shell Thickness Measured" on Table 3. The maximum calculated stresses occur in the cylinder shell adjacent to the breached areas and are presented in Table 3. The load cases and computed stresses in Table 3 provide either a "recommended" or "not recommended" for a particular cylinder and lift condition. The maximum acceptable stress for a recommended lift condition is below material yield strength.

Inspection Criteria for 48-inch UF_6 Cylinders. A standard for UF_6 cylinders has been established by American National Standard Institute (ANSI). This standard, ANSI N14.1 (1990) includes only those cylinders meeting all of the acceptance criteria for UF_6 handling. The defect criteria of ORO-651 distinguishes between accepTable 1nd unacceptable classes of damage, and these are classed for severity. This study considered unacceptable damage such as gouges or cuts and visible cracks.

The significance of gouges or cuts and cracks can be evaluated in terms of the localized effects on the stress fields resulting from internal pressurization and from mechanical loadings. These effects have been treated both by finite element analysis and by fracture mechanics techniques as shown in Appendix A. This study shows that such defects, which do not violate the 0.250-inch minimum wall thickness specified in ANSI N14.1 and ORO-651, do not result in local stresses which exceed the ASME Section VIII, Division 1 allowable values.

For cylinders with 0.250-inch wall thickness, if a 1/16-inch deep flaw or gauge is not adjacent to the stacking contact points the cylinder stress meets ASME Section VIII, Division 2 (1992) requirements.

Table 3. Summary of the Cylinder Load Cases and Stresses

Cylinder ID (Type)	File Name	Lifting Device	Minimum Shell Thickness Measured[2] (inch)	Maximum Stress[3] (psi)	Position of the Lifting Devices on the Cylinder[1]	Recommendation of the Lifting Device
C1 (48-OM)	R14951	Lugs	0.120	25,398	Lugs	Not Recommended
	A14951	Straps [5]	0.120	6,752	x_1= (-) 9.0" to (-) 11"	Recommended Use 6" × 30' straps, (3.4°)[4], (See note 7)
	S14951	Straps[5,8]	0.120	6,717	x_1= 17.0", and x_2 = 82.0" to 88.0"	Recommended (Use the same straps as above)
	B14951	Raygo Wagner	0.120	6,314	-	Not Recommended (See note 6)
C2 (48-OM)	101244	Raygo Wagner, or Straps [5]	0.050	22,960	For straps, x_1 = 75.0" Set the Raygo Wagner centered on the cylinder	Recommended for three lifting devices. Use 6" × 10' straps, (2.7°) or 6" × 15' straps, (1.8°).
	1L1244	Lugs	0.050	22,820	Lugs	
C3 (48-T)	7953A	Lugs	0.109	33,520	Lugs	Not Recommended
	7953S	Straps [5]	0.109	23,460	x_1 = 1.0",	Recommended Use 6" × 15' straps, (3.3°)
	7953RW	Raygo Wagner	0.109	23,490	Set the Raygo Wagner centered on the cylinder	Recommended
C4 (48-OM)	None	Lugs, or Straps [5]	0.112	Stresses are lower than those of cylinder C1	x_1 = 83" to 89" for straps	Recommend for Straps and Lugs. Use 6" × 10' straps, (2.7°)

[1] Dimension x_1 or x_2 is measured from the origin of the coordinates (see Figure 1) to the close edge of the strap.
[2] Measurements were obtained from the UF_6-Cylinder Wall Thickness Report.
[3] A calculated equivalent uniaxial tensile stress adjacent to the breached area—Hencky von Mises stress.
[4] Estimated angle between the vertical axis and the strap/chain in the plane of the cylinder axis and vertical axis.
[5] Straps are placed symmetrically with respect to center of gravity except where x_1 is specified.
[6] Recommends Raygo Wagner for rotating, if necessary but not for carrying the cylinder.
[7] For the alternate straps, if needed, 4-inch wide or 2-foot long straps can be used.
[8] This condition is to lift the plug end only to relocate the chock. It is to be used, if necessary, to achieve a vertical lift for the case C1.

CONCLUSION

Eighty-eight percent of the thin-wall UF_6 cylinders, types 48OM and 48G at the Lockheed Martin Energy Systems plants had been designed and constructed to ASME Section VIII, Division 1 requirements. In the case of the normal stacking condition in the cylinder yards, the stacked cylinders under the dead weight create stresses, generally bending, especially at the regions where the high localized stresses occur. For a cylinder wall thickness of 0.250 inch, unless the defects, such as a gouge or cut of up to 1/16-inch depth are in the neighborhood of the stacking contacts, the cylinder meets Division 2 stress requirements.

The typical distance between the center of the wooden chock and the stiffener ring at the K-25 and Paducah sites is not more than 8 inches (4 inches from the surface of the ring to the closest point of the chock). The computations presented in Table 1 show that, the distance between the chock and the stiffner ring needs to be minimized and the chocks are better located on the head sides of the rings.

For the storage of cylinders by two tier stacking, Table 2 can be referenced. The distance between the upper and lower stiffener ring needs to be minimized. The lifting lugs should be carefully positioned to avoid contact with the cylinder shell. From computations with the file names S6L188FW, S6L250FW, and S6L312FW from Table 2, the localized stresses at the contact points are basically the bending stresses, and they are inversely proportion to the wall thickness considered.

For the lifting of breached cylinders, even though stresses are high at the region of the breach, cylinders can be lifted with the lifting lugs, straps, and Raygo Wagner if the remaining wall thickness is great or equal to 0.250 inch.

REFERENCES

American National Standards Institute, *Uranium Hexafluoride—Packaging for Transport*, ANSI Standard N14.1 (1990).

American Society of Mechanical Engineers, *Boiler and Pressure Vessel Code*, Section VIII, Division I, and Section XI (1992).

Engineering Mechanics Research Corporation, Display III, NISA II (Version 91.0), Troy, Michigan, (1994).

Suresh, S., et al., "Environmentally Affected Near Threshold Fatigue Crack Growth in teels," Proceedings of the Fracture Mechanics 14th Symposium, Vol. 1, *Theory and Analysis* (1981).

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APPENDIX A. THE SURFACE FLAW OR GAUGE ON THE UF_6 CYLINDER

Method for Stress Intensity Factor (K_I) Determination. The stress intensity factor for the flaw model can be calculated from the membrane and bending stresses, as provided in Article A-3000, Appendix A, ASME Section XI, at the flaw location using the following equation.

$$K_I = S_m M_m\, (\pi a/Q)^{1/2} + S_b M_b\, (\pi a/Q)^{1/2}$$

where
S_m = membrane stress (psi),
S_b = bending stress (psi),
a = minor half-diameter (inches) of embedded flaw or flaw depth for surface flaw,
Q = flaw shape parameter,
M_m = correction factor for membrane stress, and
M_b = correction factor for bending stress.

The crack extension resistance under conditions of crack tip plane strain, or plane strain fracture toughness, K_{IC}, is based on the lower bound of static initiation of critical K_I values measured as a function of temperature. K_{IC} of A-516, Gr.70 at the room temperature is obtained as 212.030 ksi$\sqrt{}_{in}$ from Suresh et al. (1981). Since K_{IC} for the material A-516 Gr.55 is not available, that for A-516 Gr.70 was used.

Circumferential Stresses (S_θ) for the Surface Flaw. A sample problem of <S6L250FW> is selected from Table B to calculate K_I.

$S_m = S_\theta^P$ = -1,400 psi (@ 10 torr), and $S_b = S_\theta^{DW}$ = 12,616 psi.

The major half diameter of an elliptical flaw, ℓ = 1.000", S_y <@ 100°F> = 30,000 psi, Q = 0.933, M_m = 1.33, and M_b = 1.00.

$K_I = (S_m M_m + S_b M_b)\, (\pi a/Q)^{1/2} = (-1{,}400 * 1.33 - 12{,}616 * 1.00)\,(3.14 * 0.0625 * 0.933)^{0.5} = -6{,}197$ psi $\sqrt{\text{in}}$ = $<< K_{IC}$ (212,030 psi $\sqrt{\text{in}}$)
Therefore, the flaw is not expected to propagate.

Longitudinal Stresses. $S_m = S_l^P$ = -700 psi ((@ 10 torr) and $S_b = S_l^{DW}$ = -24,607 psi
$K_I = (S_m M_m + S_b M_b)\, (\pi a/Q)^{1/2} = (-700 * 1.33 - 24{,}607 * 1.00)$
$(3.14 * 0.0625 * 0.933)^{0.5}$ = 10,931 psi $\sqrt{\text{in}}$) $<< K_{IC}$ (212,030 psi $\sqrt{\text{in}}$).

The flaw is not expected to propagate.

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