USC Viterbi School of Engineering

USC University of Southern California

AME 486 Bonus Project

[Replace Final]

Henry Glover

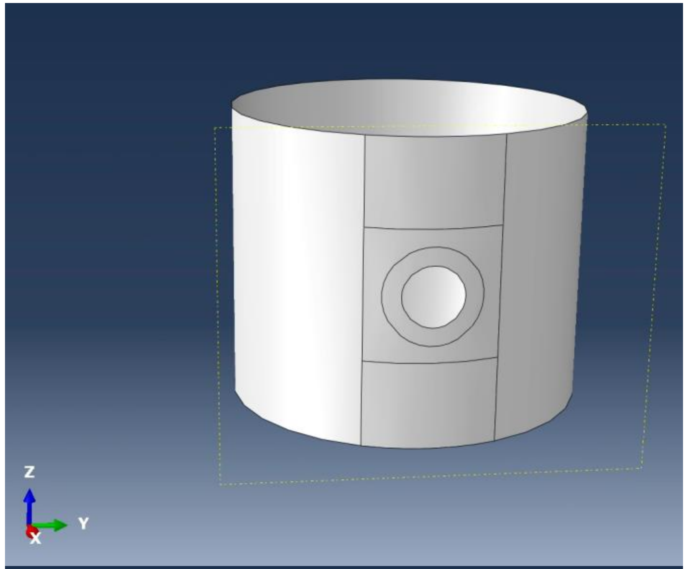
Question 1

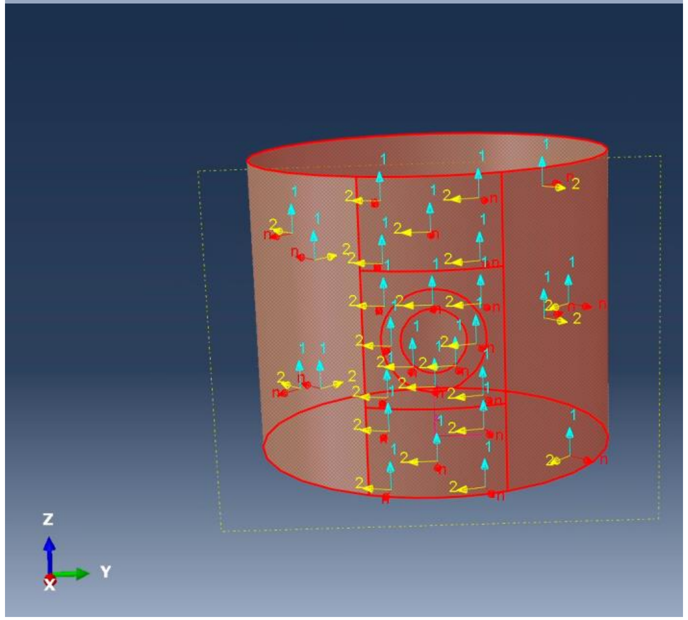
- **Cylinder modelling**
  - Height = 50in
  - Radius = 30in
- **Access hole**
  - Diameter = 10in
  - Distance of center from end = 25in
    - Hole is in the middle of the cylinder
- **Shell material**
  - Laminate = [0 /+30/-30/45/-45/+30/-30/0]
  - Ply thickness = 0.008 in
  - 0 degrees = parallel to cylinder axis
  - Physical properties
    - E11 = 12Msi = 12 e6 psi
    - E22 = 12Msi = 12 e6 psi
    - V12 = 0.05
    - G12 = 0.7Msi = 0.7 e6 psi
    - G13 = 0.5Msi = 0.5 e6 psi
    - G23 = 0.5Msi = 0.5 e6 psi
- **Modelling setup**
  - Element = S4
- **Boundary condition**
  - Clamped-clamped, allow movement in cylinder axial direction on one end
  - Characterized by 2 BCs
    - BC1: pin one end edge of cylinder
    - BC2: other end edge of the cylinder is only allowed movement in cylinder axis direction (z-direction)
- **Unit definition**
  - Length as inch
  - Mass as Ib.
  - Time as seconds
- **Derived units**
  - Pressure as psi
  - Load as Ib.

https://www.youtube.com/watch?v=Ko8CfcbyDq0&t=1188s

Question 1

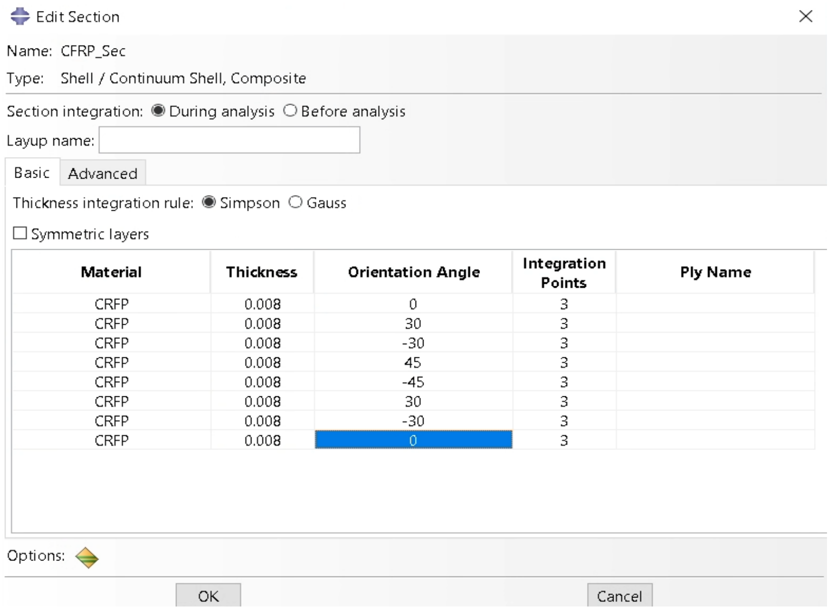
- **Model construction**
  - 3D shell using extrusion
  - Datum plane on selected principal plane
  - Cut extrude set to go through one side of cylinder
  - Partitioned one side of the cylinder with hole to refine mesh control in that region
    - Leads to better results
- **Composite ply angle global orientation**
  - The ply rotation angle is defined relative to the reference axis (ref1)
  - Ref1 needs to align along the cylinder's axis to correctly define fiber orientations
  - To fix the misalignment
    - The normal axis needed to be adjusted
    - Axis 1 was set parallel to the cylinder's length
    - We see this in the bottom image to the right, where the blue arrow is pointing up





Question 1

- **Material properties**
  - Mechanical – Elastic – Lamina properties set
  - Based on given values from the problem statement
- **Composite ply angle's**
  - 0-deg ply is along the axis of the cylinder
  - 8 total ply's, NOT SYMMETRIC
  - Material orientation was assigned to the cylinder
  - Assign composite layup so I don't need to spend extra computing time on calculating over every single ply



Edit Section

Name: CFRP_Sec

Type: Shell / Continuum Shell, Composite

Section integration: During analysis / Before analysis

Layup name:

Basic | Advanced

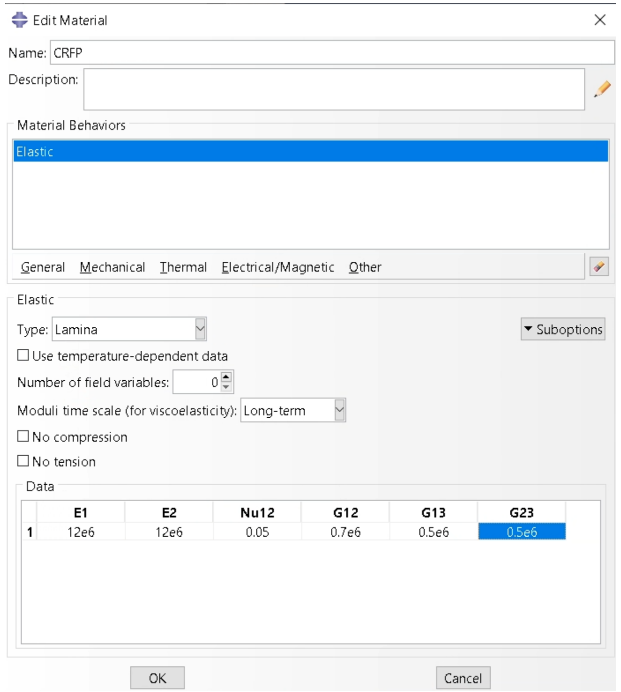
Thickness integration rule: Simpson / Gauss

Symmetric layers

Material	Thickness	Orientation Angle	Integration Points	Ply Name
CRFP	0.008	0	3	
CRFP	0.008	30	3	
CRFP	0.008	-30	3	
CRFP	0.008	45	3	
CRFP	0.008	-45	3	
CRFP	0.008	30	3	
CRFP	0.008	-30	3	
CRFP	0.008	0	3	

Options:

OK Cancel



Edit Material

Name: CRFP

Description:

Material Behaviors

Elastic

General Mechanical Thermal Electrical/Magnetic Other

Elastic

Type: Lamina

Suboptions

Use temperature-dependent data

Number of field variables: 0

Moduli time scale (for viscoelasticity): Long-term

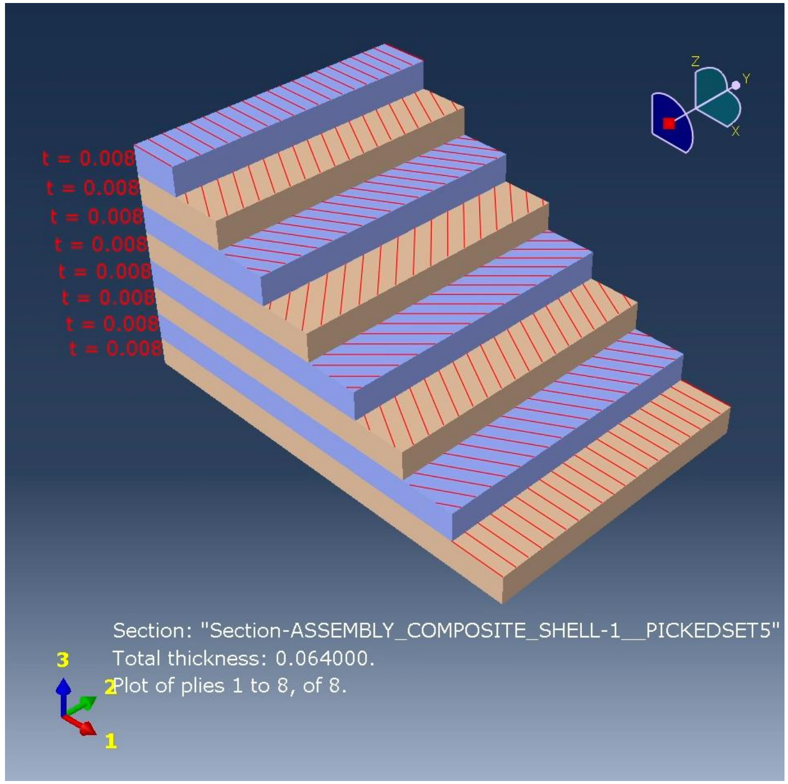
No compression

No tension

Data

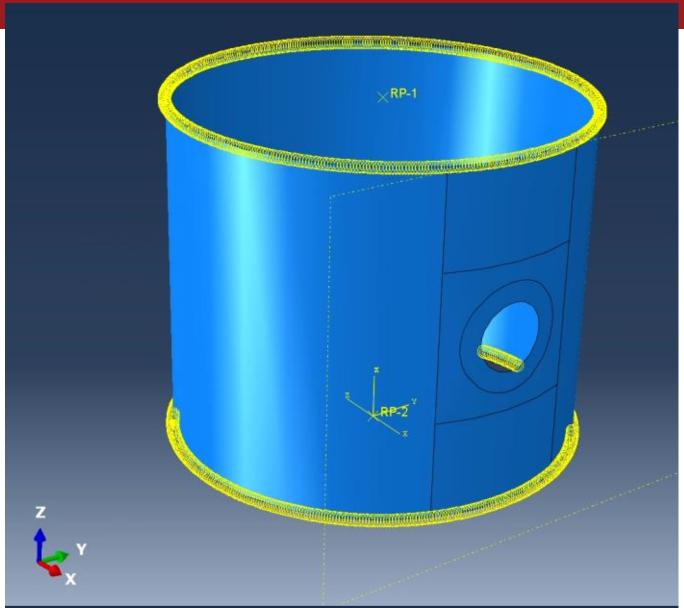
	E1	E2	Nu12	G12	G13	G23
1	12e6	12e6	0.05	0.7e6	0.5e6	0.5e6

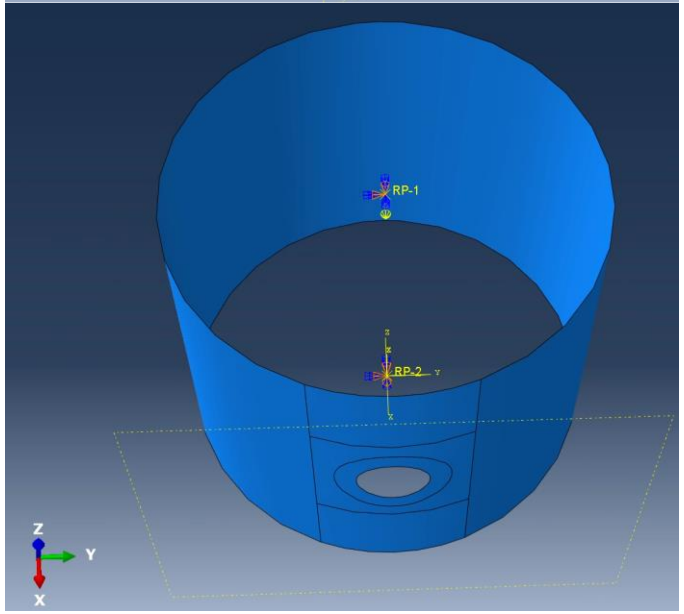
OK Cancel



Question 1

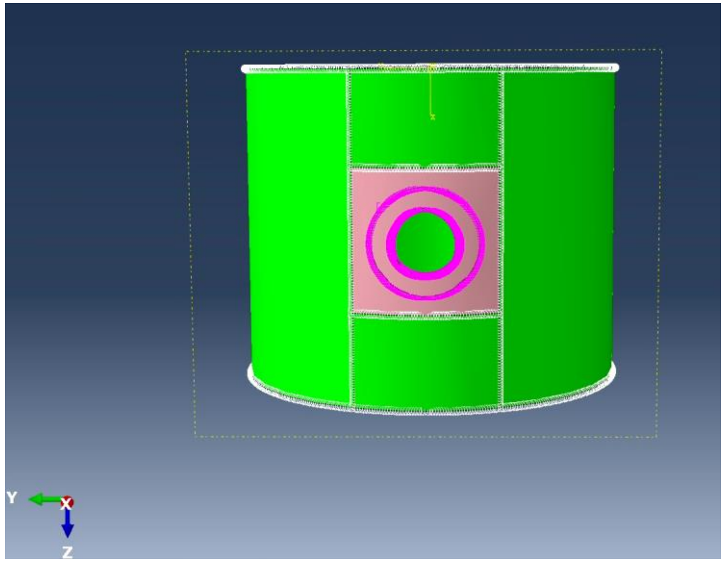
- **Boundary conditions and load**
  - The top and bottom edges are rigid bodies
    - Sets are made to represent the edges (easier to select)
  - Datum system is set to the bottom of the cylinder
  - Reference points for top (RP-1) and bottom (RP-2) are set
  - Boundary conditions
    - Bottom = fixed (Encastre)
    - Top = allows for displacement and rotation but, we only move in the cylinder axis (z-direction)
    - From problem statement, assuming clamped-clamped
    - Select everything except U3 (z-direction), only allowing axial movement
  - Load
    - Applied at top using RP-1
    - CF3 = z-direction, + and – follow axis directing as shown
    - Because compression, CF3 = -1
      - The load points into the cylinder (compression)
  - -1 is arbitrary number to get eigenvalue (critical load)

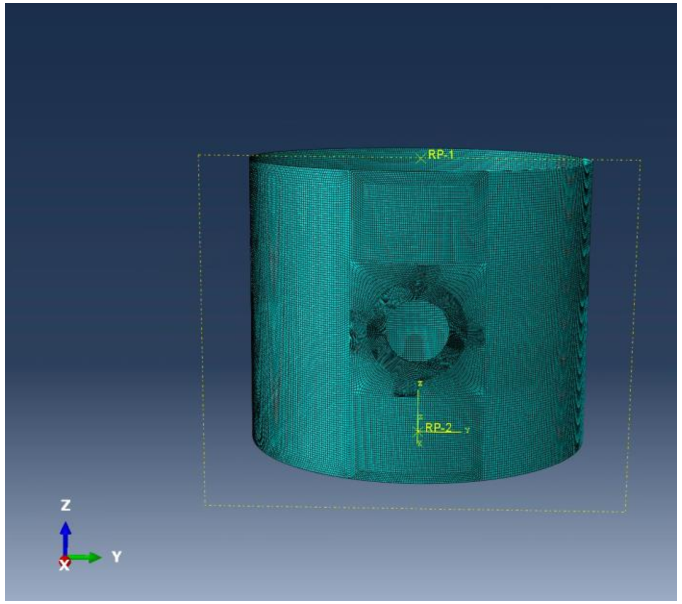


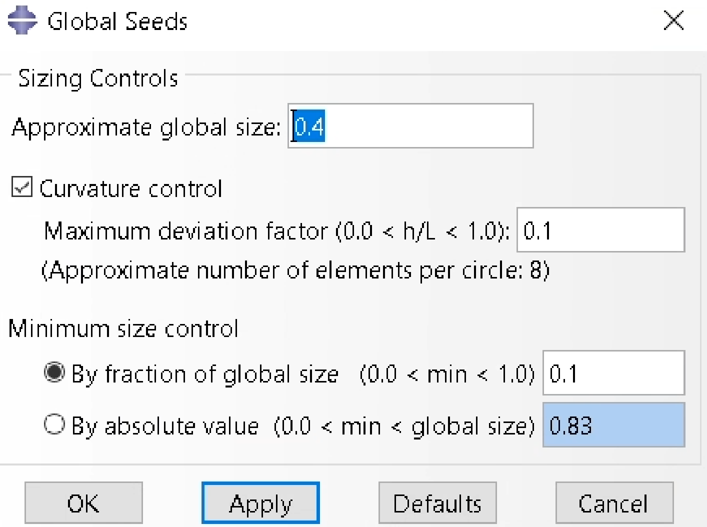


Question 1

- **Mesh**
  - Elements is set as S4 by turning off reduced integration
  - Majority of the cylinder (green) is set to "Structured" – makes mesh more reasonable
  - Hole and close surrounding area within the partition lines are set to "Free" with medial axis turned on – makes it more symmetric
  - Global seed is set to 0.4 – shown in the image
  - The two circles are meshed independently to make it finer, local seeds are set "by number" and set to 300 element numbers

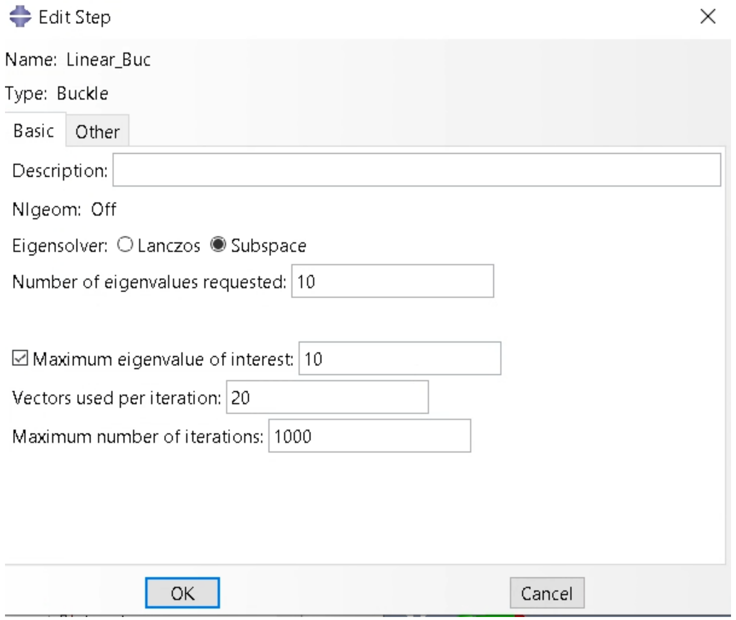


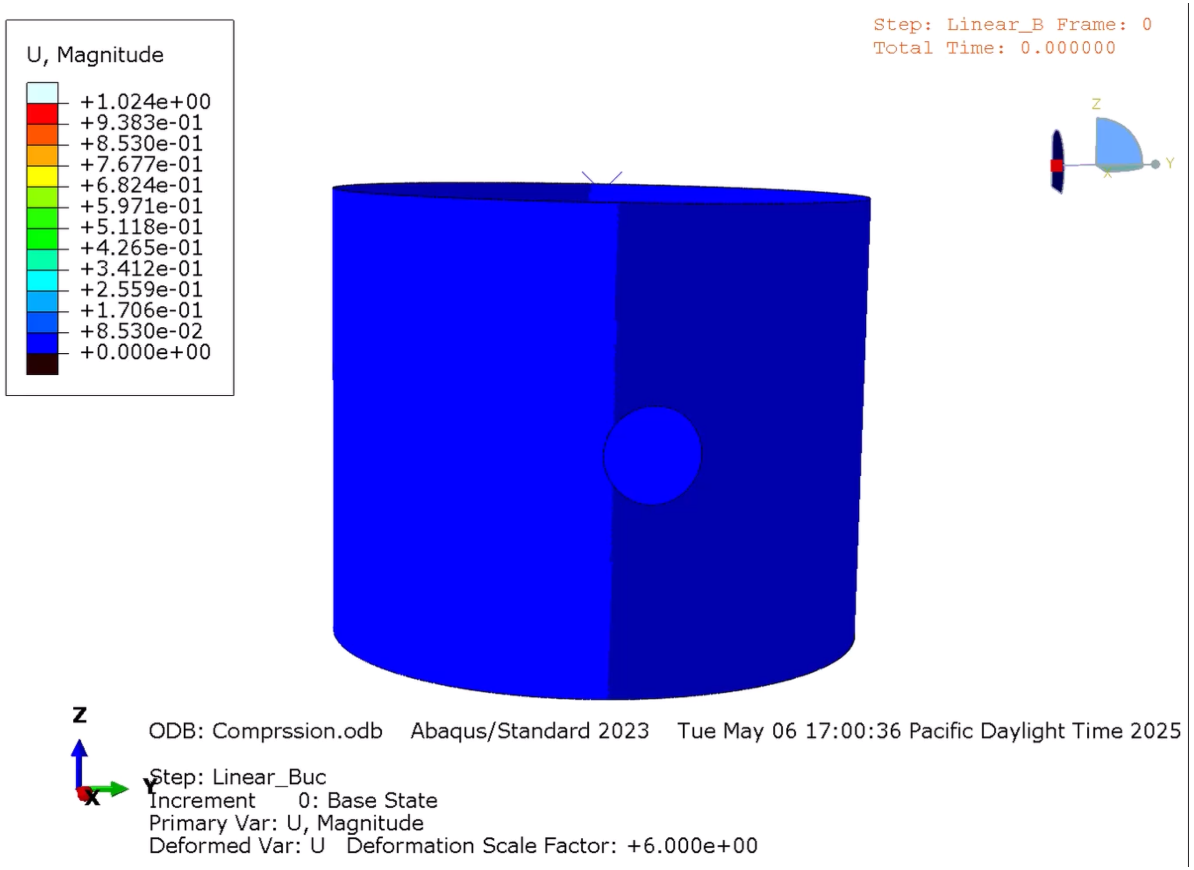




Question 1

- **Results**
  - Linear perturbation "Buckle" step is selected
  - "Subspace" eigen solver is appropriate for smaller models
  - We want 10 buckling modes (eigenvalues)
    - Only the first mode is relevant
  - Set maximum number of iterations to about twice the number of eignevalues
  - GIF video of all eigenvalues/buckling modes (PRESS PLAY) →



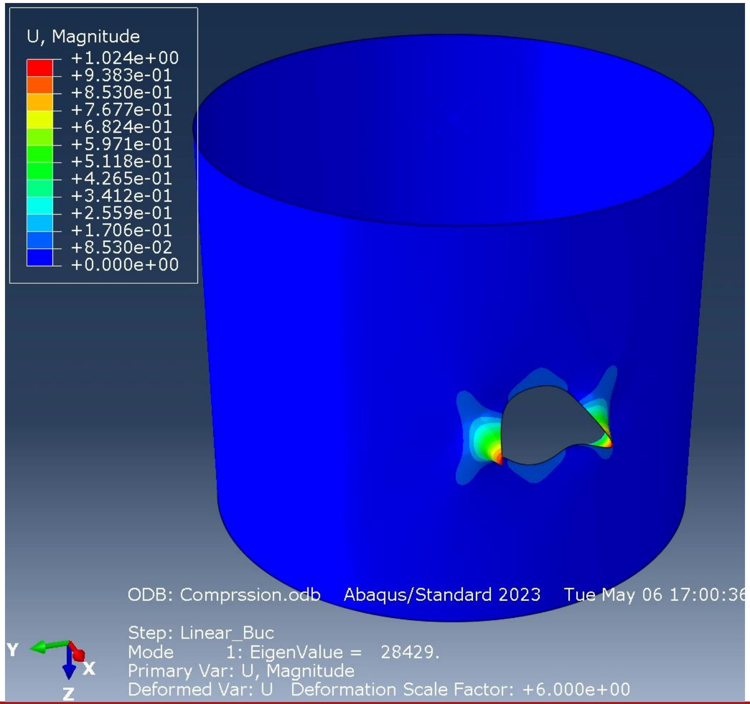


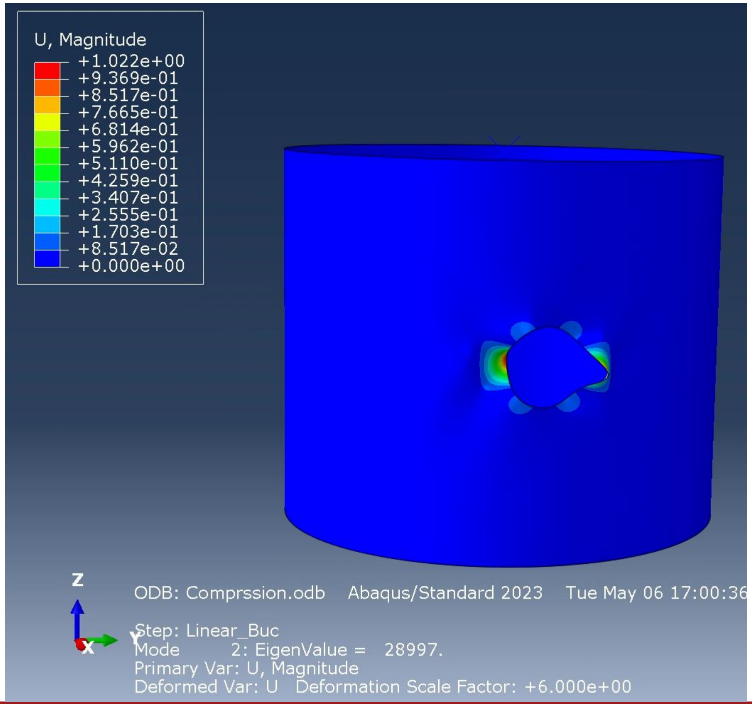
Play button

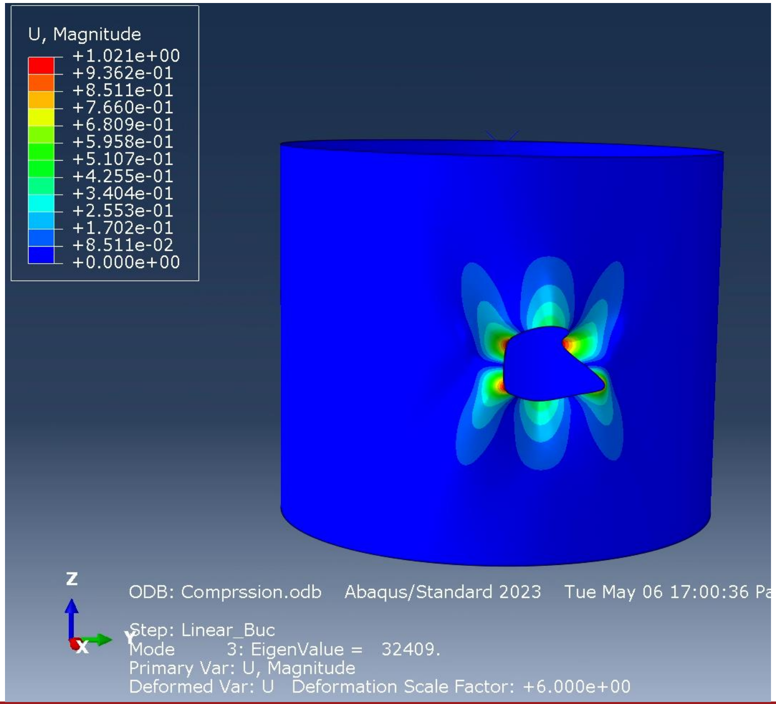
Question 1

- **Linear model results**
  - Eigenvalues (First 3 modes)
    - 28429, 28997, 32409
    - Our reference load was -1, so we can use these values as the critical load when the shell buckles
  - Knockdown factor = 0.5
  - Minimum load where buckling observed (Mode 1) = 28429 lb.
  - Compression design load allowable = 0.5 * 28429 = 14214.5 lb.
  - Mode shape of the buckle is shown in the image (leftmost image below = mode 1)
    - Mode 1 is the lowest eigenvalue which is the most critical failure mode

- Mode 2 and Mode 3 results are just shown for context
- Mode 1 is the most important

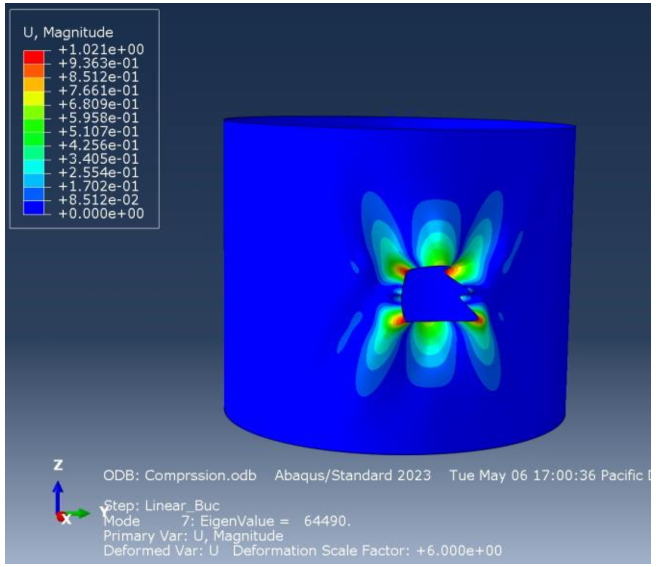


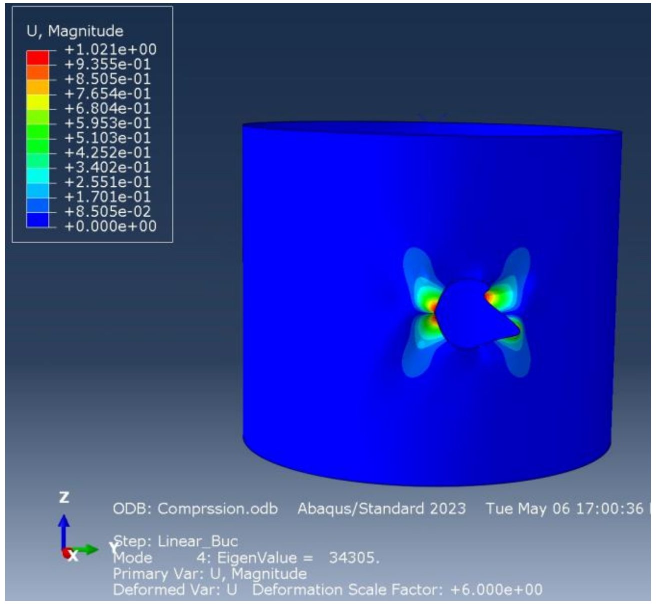


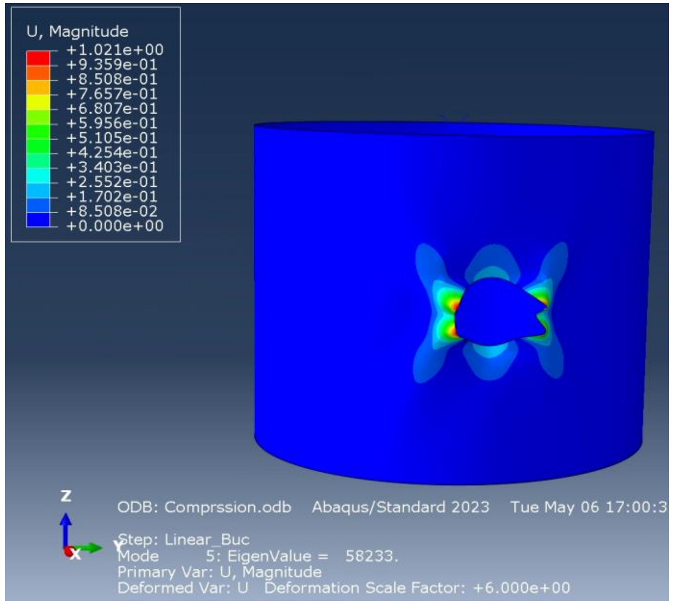


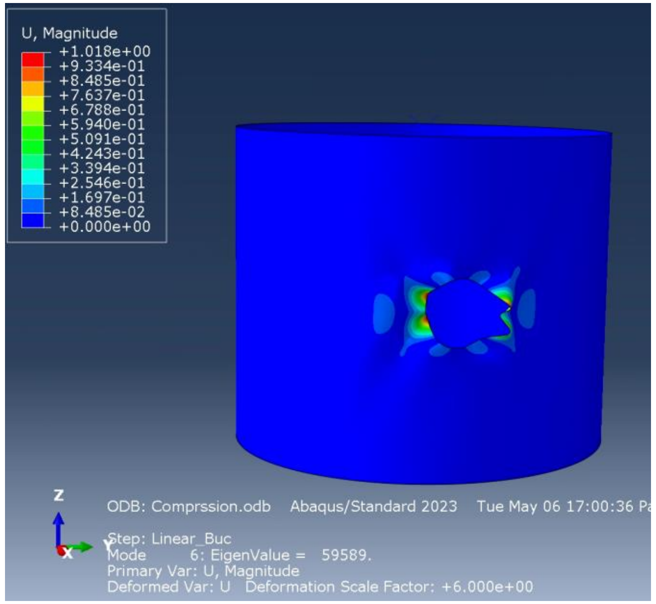
Question 1

- **Results**
  - Modes 4,5,6,7
  - Just shown for context to show I did FEA correctly



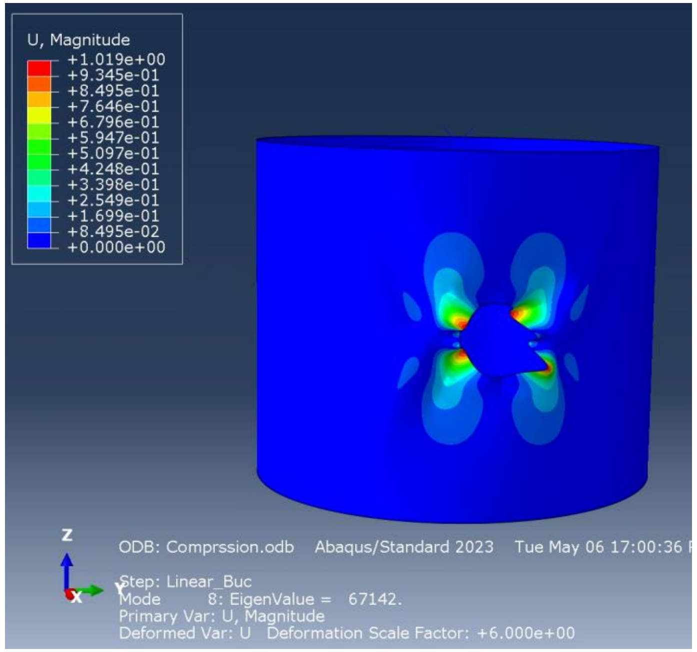


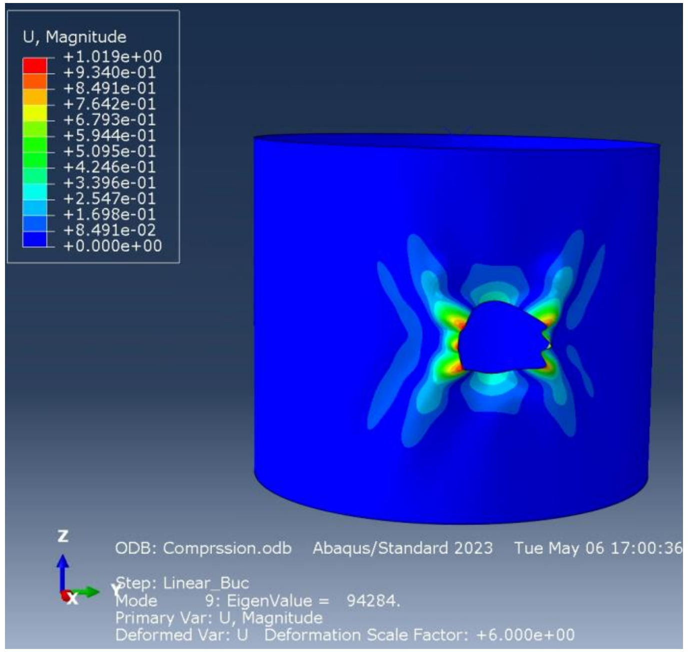


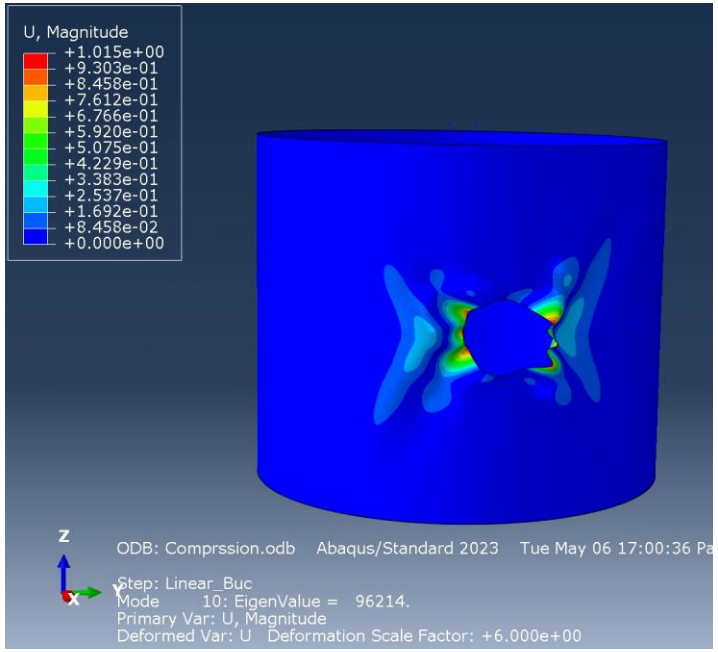


Question 1

- **Results**
  - Modes 8, 9, 10
  - Just shown for context to show I did FEA correctly

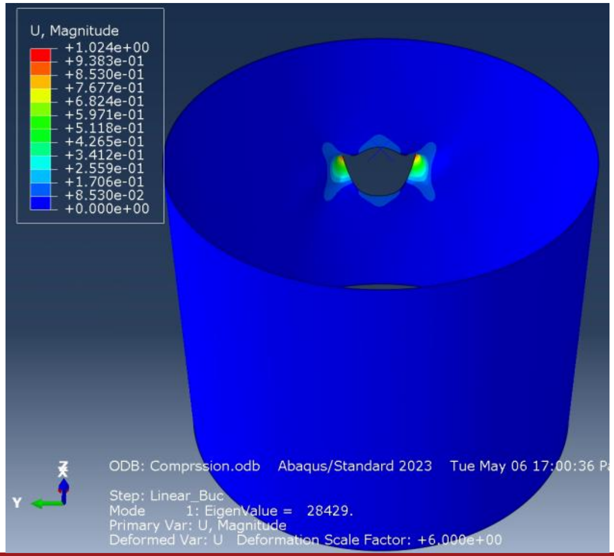


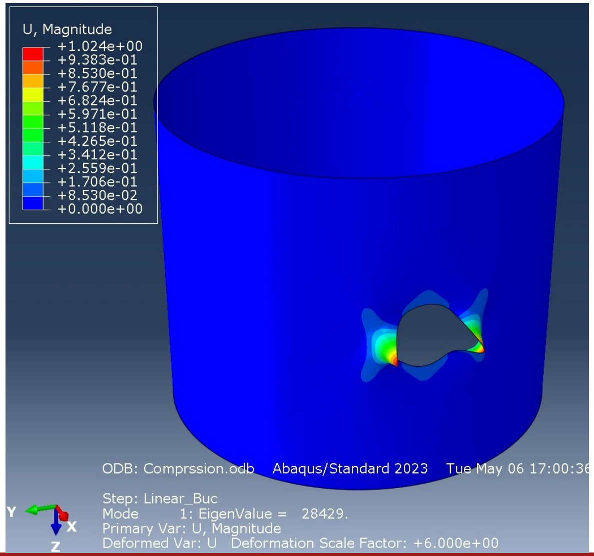


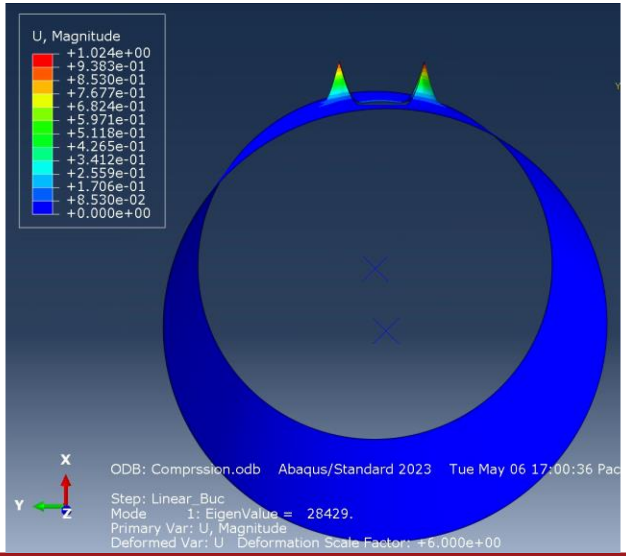


Question 1

- **Conclusion**
  - No nonlinear analysis was required for this Bonus Project but from 585 experience, we do know it is more accurate/less conservative for buckling modeling
    - The linear model underpredicts buckling capabilities
  - Composite cylinder shells buckle differently than metal structures with cutouts
    - Buckling response is dominated by nonlinear effects
  - To determine structure buckling loads, it is ideal to run physical testing and compare those results to analytical buckling loads found in FEA
  - In Mode 1, our minimum load where buckling was observed = 28429 lb.
  - Compression design load allowable accounting for knockdown factor
    - = 0.5 * 28429 = 14214.5 lb.
  - Mode shape of the buckling shown below in images (Mode 1)

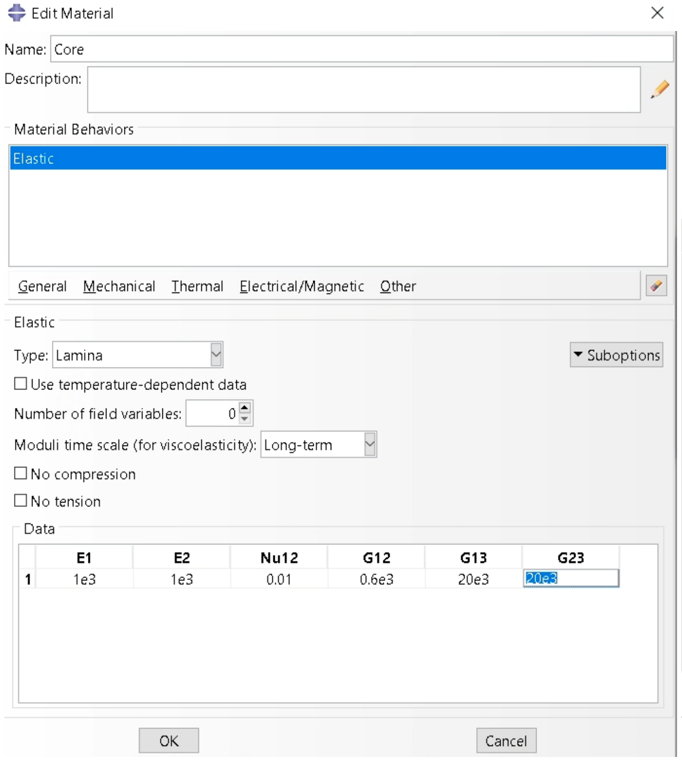


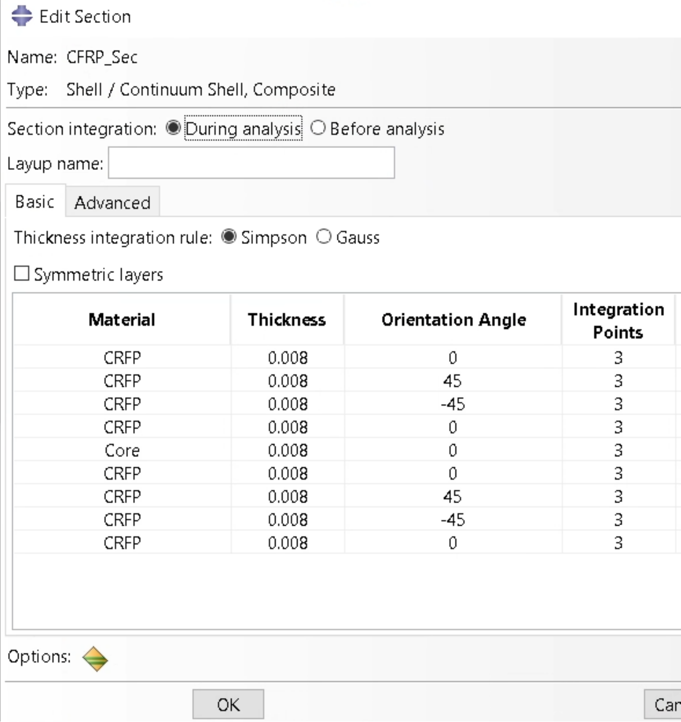




Question 2

- **Setup Changes**
  - Same setup as Question 1 for modeling
  - Only thing that changes is laminate stacking sequence and new core material properties
  - Core material properties are inputted
    - E11 = 1 ksi
    - E22 = 1 ksi
    - v12 = 0.01
    - G12 = 0.6 ksi
    - G13 = 20 ksi
    - G23 = 20 ksi
  - Plys are changed to match new stacking sequence
    - [0 /+45/-45/0/Core/0/+45/-45/0]
    - Core = 0 degrees
    - Not symmetric
  - Started with 0.008 in. thick core to establish baseline test results
    - Thickness of core was iterated from here

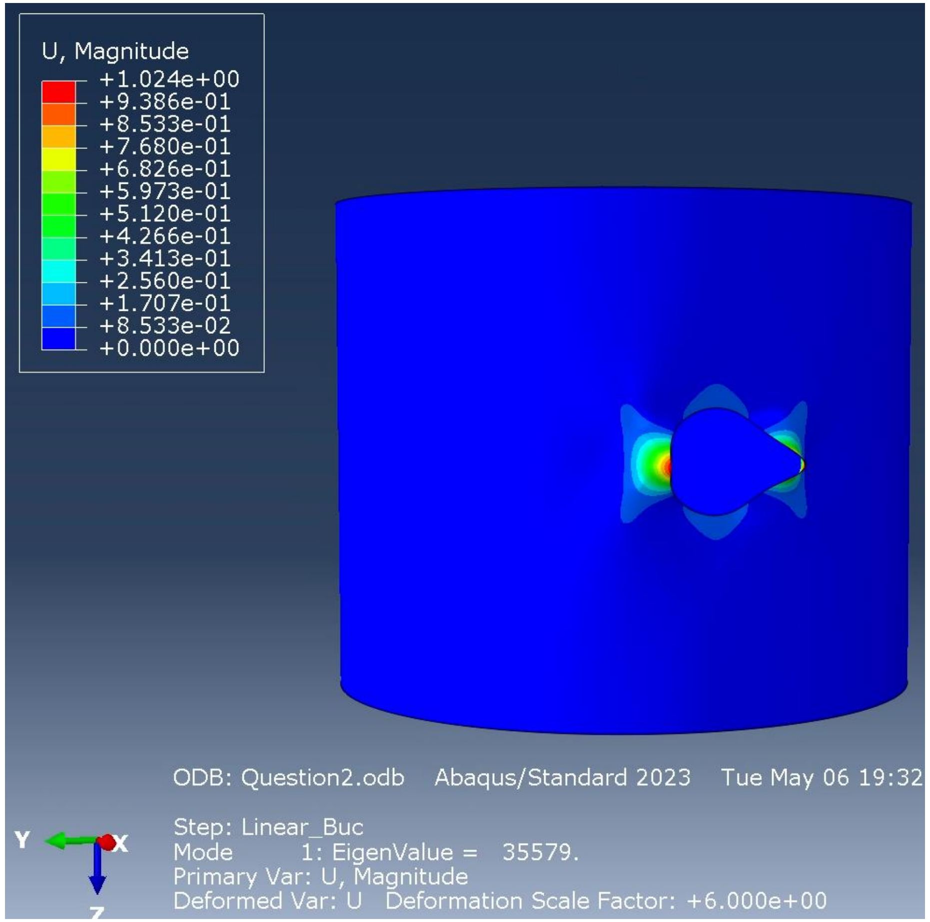




Question 2

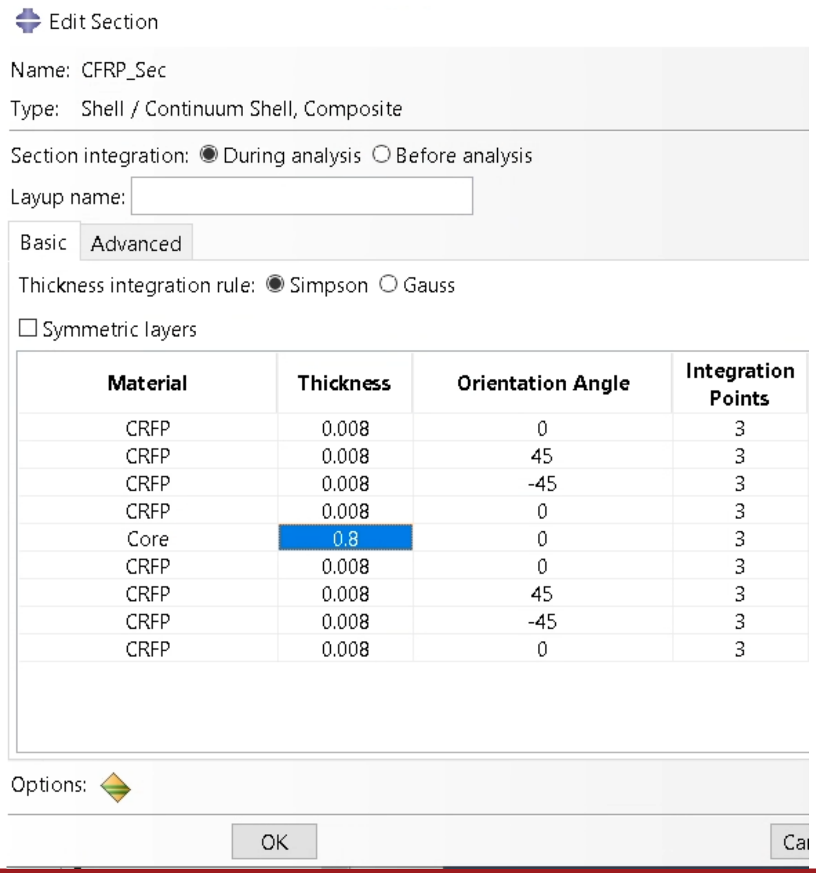
- **Problem statement**
  - Problem statement says we need to determine core thickness to survive a max compression load of 2,000,000 lbs.
  - Need to incorporate a knockdown factor for material imperfection and a F.O.S.
    - Knockdown = 0.5
    - F.O.S = 1.4 for human spaceflight
  - We need the buckling loads to exceed:
    - (20000000) x (1.4) /(0.5) = 5,600,000 lbs.
  - So, buckling load in the FEA needs to exceed 5,600,000 lbs. while still being as light as possible
  - Mode 1 is the lowest eigenvalue which is the most critical failure mode (looking at these values)

- **First Run**
  - First trial was with 0.008 in. core thickness
  - Mode 1 Eigenvalue = 35,579 lbs.
  - Far too low, but good test for a baseline



Question 2

- **Second Run**
  - Core thickness set to 0.8 in.
  - Mode 1 Eigenvalue = 1,165,310 lbs.
  - Still buckling load was still too little but, getting close to value


Edit Section

Name: CFRP_Sec

Type: Shell / Continuum Shell, Composite

Section integration: During analysis Before analysis

Layup name:

Basic Advanced

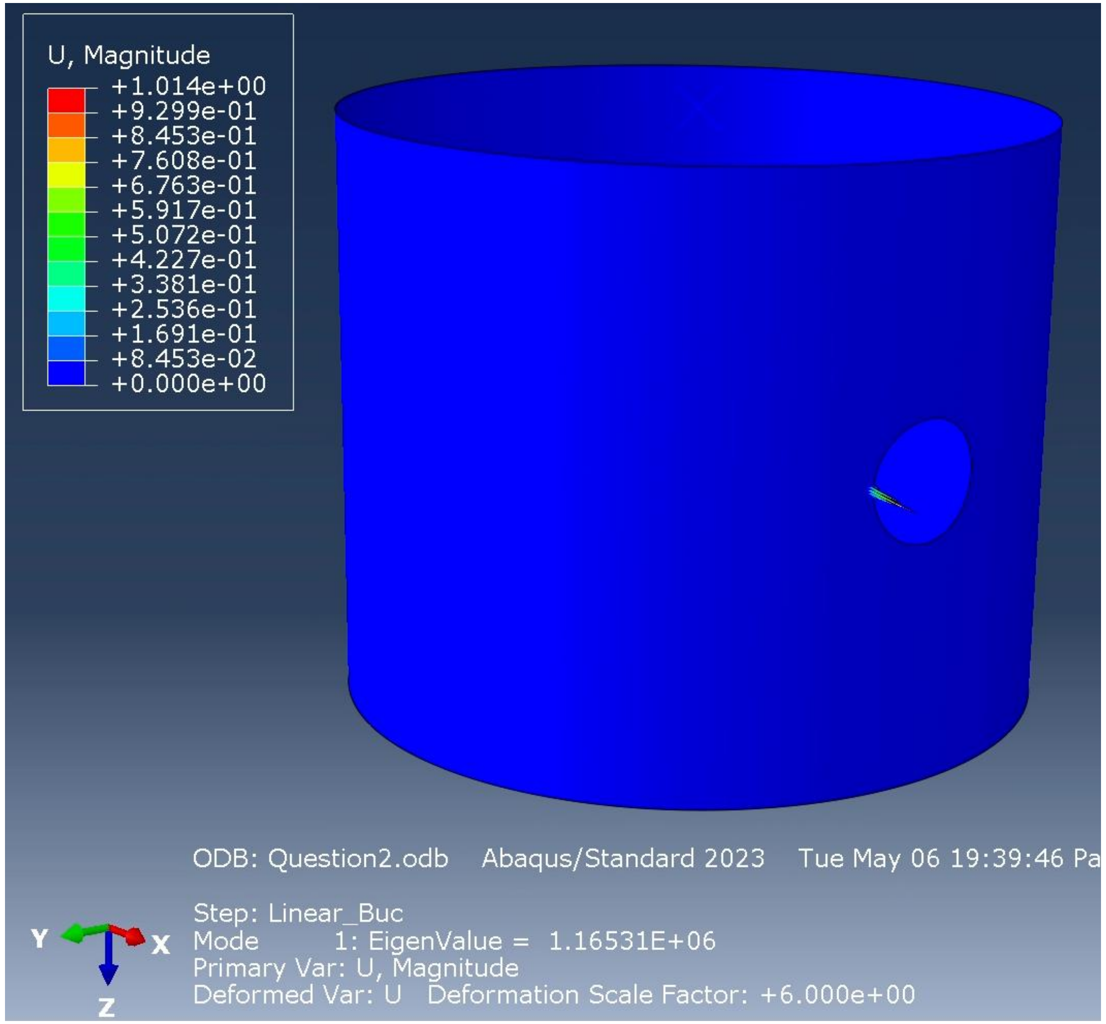
Thickness integration rule: Simpson Gauss

Symmetric layers

Material	Thickness	Orientation Angle	Integration Points
CRFP	0.008	0	3
CRFP	0.008	45	3
CRFP	0.008	-45	3
CRFP	0.008	0	3
Core	0.8	0	3
CRFP	0.008	0	3
CRFP	0.008	45	3
CRFP	0.008	-45	3
CRFP	0.008	0	3

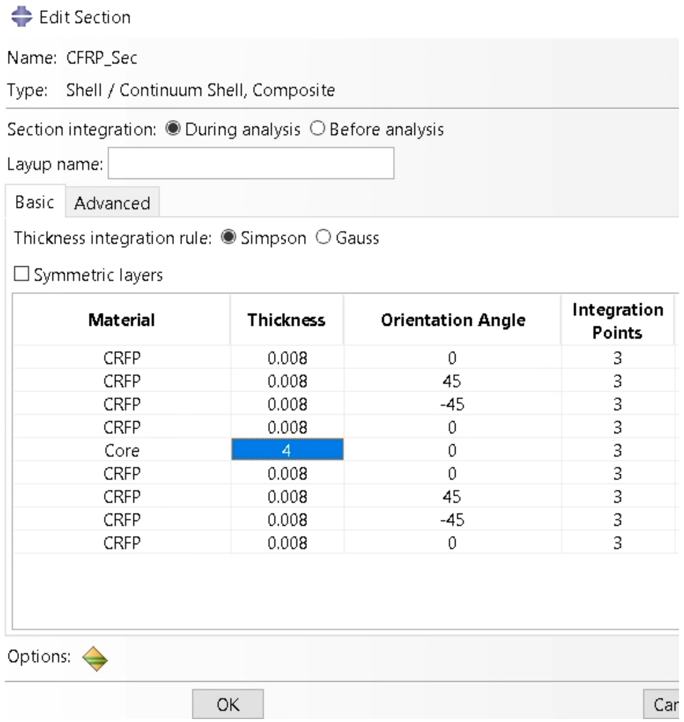
Options:

OK Ca

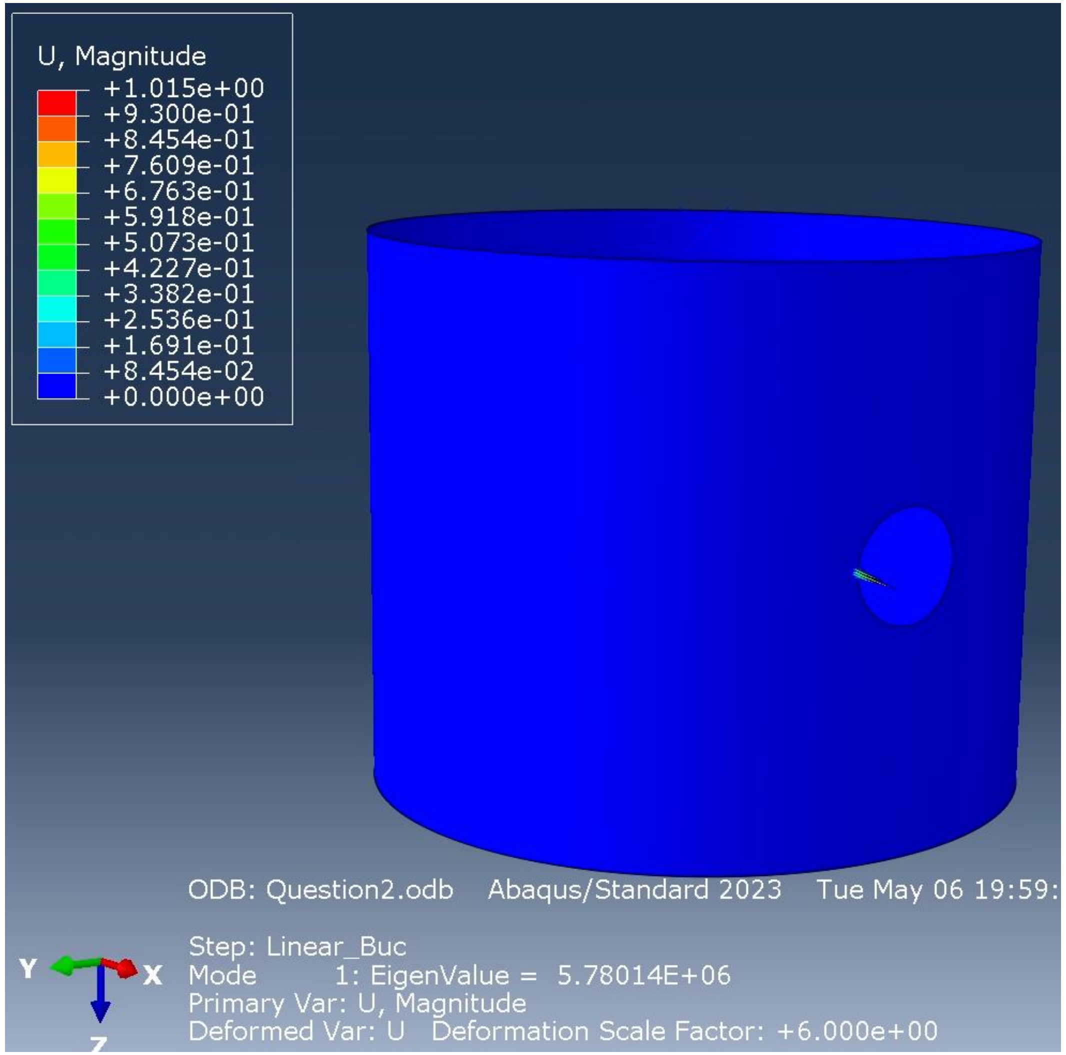


Question 2

- **Third Run**
  - Core thickness set to 4 in.
  - Mode 1 Eigenvalue = 5,780,140 lbs.
  - Buckling load is greater than our expected load when including knockdown factor and F.O.S
    - 5,780,140 > 5,600,000 lbs.
  - We can do more iterations; we want the lightest available core, and we can slightly reduce the core thickness to achieve that

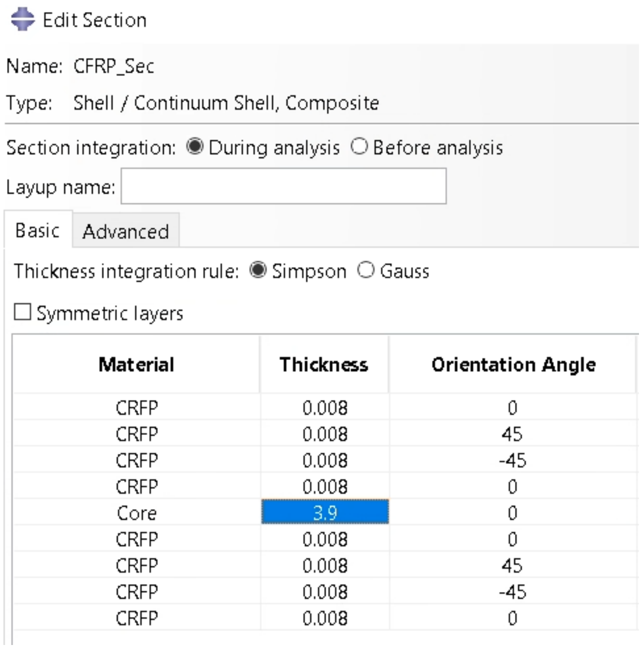


Material	Thickness	Orientation Angle	Integration Points
CRFP	0.008	0	3
CRFP	0.008	45	3
CRFP	0.008	-45	3
CRFP	0.008	0	3
Core	4	0	3
CRFP	0.008	0	3
CRFP	0.008	45	3
CRFP	0.008	-45	3
CRFP	0.008	0	3



Question 2

- **Last Run**
  - Core thickness set to 3.9 in.
  - Mode 1 Eigenvalue = 5,632,670 lbs.
  - Buckling load was greater than our expected load when including knockdown factor and F.O.S
    - 5,632,670 > 5,600,000 lbs.
  - This eigenvalue is perfect, we keep the weight of the core as light as possible and still exceed our necessary load requirements


Edit Section

Name: CFRP_Sec

Type: Shell / Continuum Shell, Composite

Section integration: ◉ During analysis ○ Before analysis

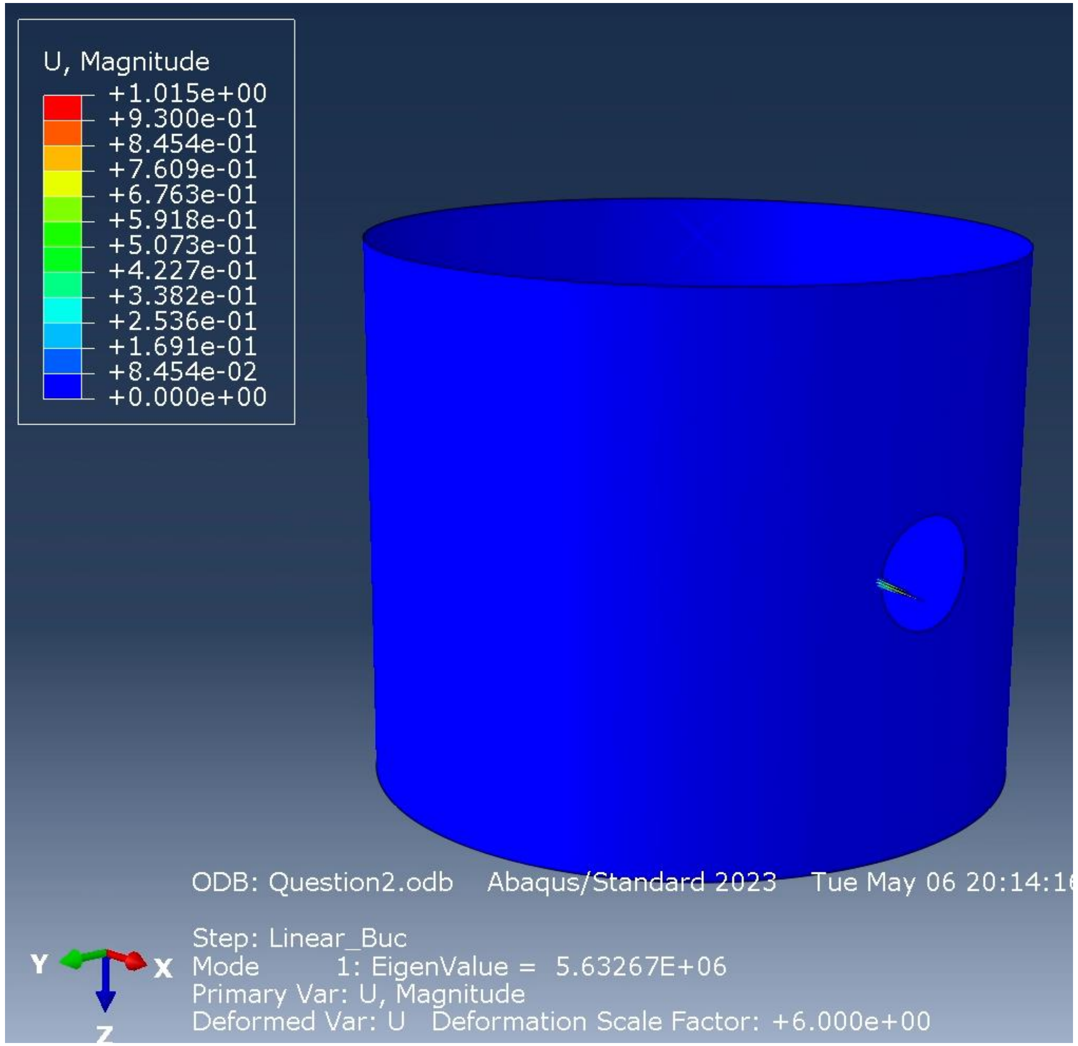
Layup name:

Basic | Advanced

Thickness integration rule: ◉ Simpson ○ Gauss

☐ Symmetric layers

Material	Thickness	Orientation Angle
CRFP	0.008	0
CRFP	0.008	45
CRFP	0.008	-45
CRFP	0.008	0
Core	3.9	0
CRFP	0.008	0
CRFP	0.008	45
CRFP	0.008	-45
CRFP	0.008	0



Question 2

- **Conclusion**
  - The buckling load from linear buckling analysis with a 3.9 in. core thickness = 5,632,670 lbs.
    - No need to reduce more
  - This is slightly greater than our required minimum buckling load (after applying knockdown and F.O.S)
  - This minimizes weight of the core as much as possible, of course I could reduce the core thickness more by removing a 100^{th} of an inch but, I deemed it was unnecessary and hard to manufacture in real life
  - I also like the “pocket margin” with the 3.9-inch-thick core
  - We also know that linear analysis errs very conservatively, so I am happy with a 3.9 in. core and even happier with a 4.0-inch core if weight requirements allow it
    - 4.0 in. is an easy even number that could be easier to manufacture in real application. (just a suggestion/commentary)
  - The sandwich design of the core significantly improves buckling resistance with minimal weight penalty

  - Thank you, Professor Goyal and Aniket!