AME 486

Project 9(HW10) – Sandwiches (Individual)

Henry Glover +5 Points
4/14/2025

#1

- Kevlar 49/Epoxy material
  - [0/30/90/45/0]s
  - Thickness = 0.009 in.
    - Laminate total = 0.009 (10) = 0.09 in.
  - Subjected to pressure
  - Radius = 10 in.
  - 1.5%, 1.4%, 1.6%, 1.45%, 1.6%, 1.55%
  - Factor of safety = 1.5
- A-basis allowable = 99% in 95% confidence window (1 in 100 specimens will fail with 95% confidence level)
  - Using the “Composite Material Handbook. Volume 1. Polymer matrix Composites” found online I am able to find the statistical methods for A-Basis allowables
  - https://apps.dtic.mil/sti/tr/pdf/ADA426516.pdf (A-basis CHARTS - composites) (pg. 565)
  - For a sample size $n = 6$, $K_A = 5.062$
- Calculate A-basis allowable
  - $\varepsilon_A = u - K_A \cdot \sigma$ = 0.01517 – 5.062 (0.000745) = 0.011399
- Apply F.S. to A-basis allowable
  - $\varepsilon_{allowable} = \dfrac{0.11399}{1.5} = 0.007599$
- Apply Margin of safety on top
  - $MS = \dfrac{\varepsilon_{allowable}}{\varepsilon_{actual}} - 1 = 0.15$
  - $\varepsilon_{actual} = \dfrac{\varepsilon_{allowable}}{1.15} = \dfrac{0.007599}{1.15} = 0.006607$

Standered Deviation	Mean
0.015	0.015
0.014	0.014
0.016	0.016
0.0145	0.0145
0.016	0.016
0.0155	0.0155
0.000745356	0.015167

Volume 1, Chapter 8 Statistical Methods

MIL-HDBK-17-1F

TABLE 8.5.11 *One-sided A-basis tolerance limit factors, k_A, for the normal distribution, Reference 8.5.11), continued on next page.*

n	k_A	n	k_A	n	k_A	n	k_A
2	37.094	36	2.983	70	2.765	104	2.676
3	10.553	37	2.972	71	2.762	105	2.674
4	7.042	38	2.961	72	2.758	106	2.672
5	5.741	39	2.951	73	2.755	107	2.671
6	5.062	40	2.941	74	2.751	108	2.669
7	4.642	41	2.932	75	2.748	109	2.667
8	4.354	42	2.923	76	2.745	110	2.665
9	4.143	43	2.914	77	2.742	111	2.663
10	3.981	44	2.906	78	2.739	112	2.662
11	3.852	45	2.898	79	2.736	113	2.660
12	3.747	46	2.890	80	2.733	114	2.658
13	3.659	47	2.883	81	2.730	115	2.657
14	3.585	48	2.876	82	2.727	116	2.655
15	3.520	49	2.869	83	2.724	117	2.654
16	3.464	50	2.862	84	2.721	118	2.652
17	3.414	51	2.856	85	2.719	119	2.651
18	3.370	52	2.850	86	2.716	120	2.649
19	3.331	53	2.844	87	2.714	121	2.648
20	3.295	54	2.838	88	2.711	122	2.646
21	3.263	55	2.833	89	2.709	123	2.645
22	3.233	56	2.827	90	2.706	124	2.643
23	3.206	57	2.822	91	2.704	125	2.642
24	3.181	58	2.817	92	2.701	126	2.640
25	3.158	59	2.812	93	2.699	127	2.639
26	3.136	60	2.807	94	2.697	128	2.638
27	3.116	61	2.802	95	2.695	129	2.636
28	3.098	62	2.798	96	2.692	130	2.635
29	3.080	63	2.793	97	2.690	131	2.634
30	3.064	64	2.789	98	2.688	132	2.632
31	3.048	65	2.785	99	2.686	133	2.631
32	3.034	66	2.781	100	2.684	134	2.630
33	3.020	67	2.777	101	2.682	135	2.628
34	3.007	68	2.773	102	2.680	136	2.627
35	2.995	69	2.769	103	2.678	137	2.626

#1 Cont.

- Kevlar 49
  - Tensile modulus 18E6 Psi (shown in table data)
  - Max allowable pressure
- $\sigma_{hoop} = \frac{pr}{t}$
- $p = \frac{t \cdot E \cdot \varepsilon}{R} = \frac{0.09 \cdot 18E06 \cdot 0.006607}{10} =$ 1,070.334 psi
- Maximum pressure the vessel can handle with margin of safety accounted for MS ≥ 15%
  - 10,704 Psi
- The most critical ply angle are the 90 degrees plies, where the 0-degree plies would be aligned. This is where max strain is located

https://www.dupont.com/content/dam/dupont/amer/us/en/safety/public/documents/en/Kevlar_Technical_Guide_0319.pdf

SECTION II: PROPERTIES OF DUPONT™ KEVLAR®

This section lists and describes the typical properties of Kevlar®. The data reported are those most often observed, and are representative of the particular denier and type indicated. The properties are reported in both U.S. and S.I. units.

For information on safety and health, refer to the Kevlar® Material Safety Data Sheet.

TYPICAL AND COMPARATIVE PROPERTIES OF KEVLAR®

Table II-1 lists the typical yarn, tensile and thermal properties of Kevlar® 29 and Kevlar® 49 yarns.

Additional products in the Kevlar® family of fibers are available with different combinations of properties to meet your engineering design needs.

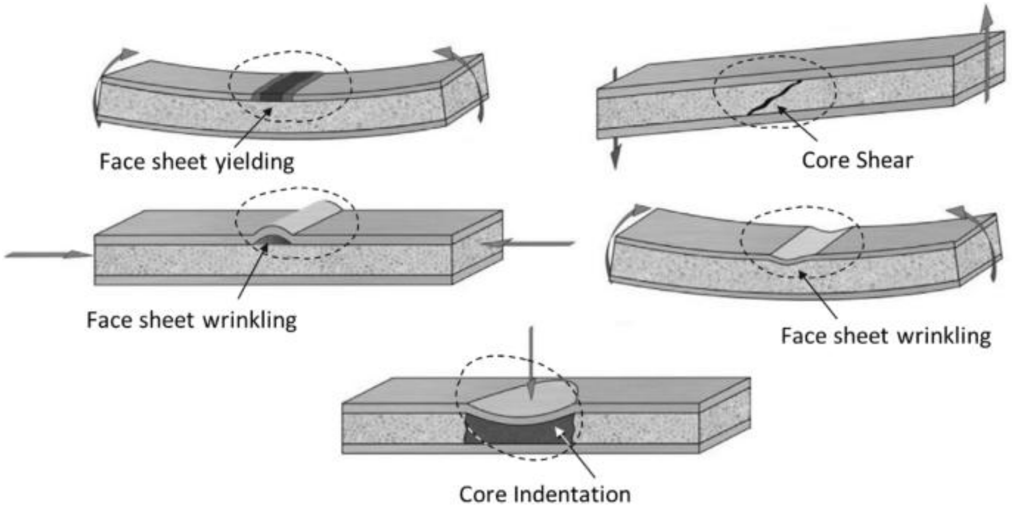
Please contact your DuPont Representative or call 1-800-931-3456 to discuss your specific application and determine the optimum Kevlar® fiber for you.

Table II-1 Typical Properties of DuPont™ Kevlar® 29 and Kevlar® 49 yarns

Property	Unit	Kevlar® 29	Kevlar® 49
Yarn			
Type	denier (dtex) # of filaments*	1,500 (1,670) 1,000	1,140 (1,270) 768
Density	$lb/in.^3$ (g/cm^3)	0.052 (1.44)	0.052 (1.44)
Moisture Levels			
As Shipped**	%	7.0	3.5
Equilibrium from Bone-Dry Yarn***	%	4.5	3.5
Tensile Properties			
Straight Test on Conditioned Yarns†			
Breaking Strength	lb (N)	76.0 (338)	59.3 (264)
Breaking Tenacity	g/d (cN/tex)	23.0 (203)	23.6 (208)
	psi (MPa)	424,000 (2,920)	435,000 (3,000)
Tensile Modulus	g/d (cN/tex)	555 (4,900)	885 (7,810)
	psi (MPa)	10.2×10^6 (70,500)	6.3×10^6 (112,400)
Elongation at Break	%	3.6	2.4
Resin Impregnated Strands††			
Tensile Strength	psi (MPa)	525,000 (3,600)	525,000 (3,600)
Tensile Modulus	psi (MPa)	12.0×10^6 (83,000)	18.0×10^6 (124,000)

#2

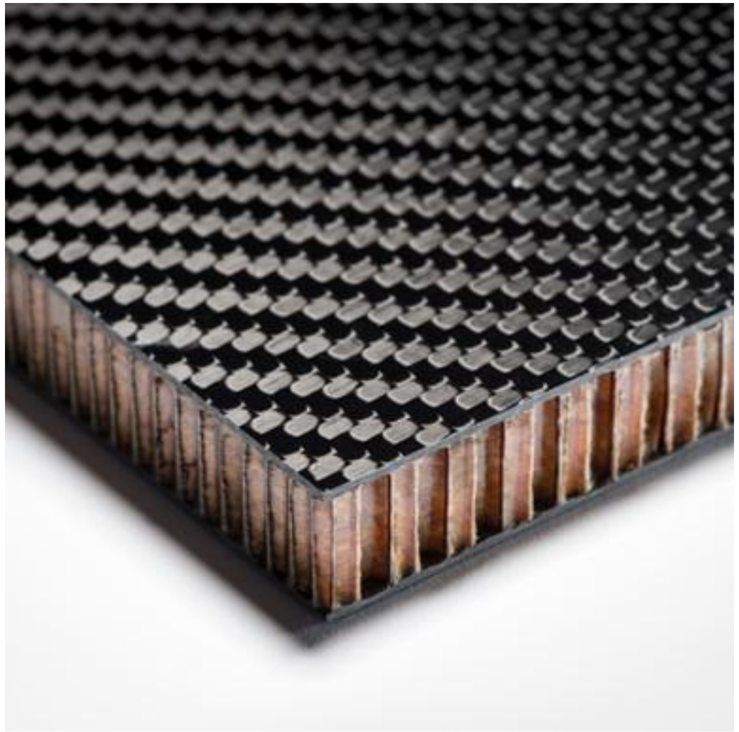
- Failure modes for sandwich structures
  - Buckling and face wrinkling
    - Overall elastic instability
    - Intracell buckling – the core cell size has to be small enough to precent intra cell buckling
    - Skin wrinkling – compressive modulus of the facing skin and the core compression strength need to be high enough to prevent skin wrinkling
  - Core shear failure
    - Core carries majority of transverse shear loads
  - Facesheet-core debonding
  - Facesheet failure
    - Facesheet intralaminar and interlaminar failure
      - In-plane compression load due to bending or axial loads
  - Facesheet Dimpling (intercellular buckling)
  - Core crushing (core indentation)
    - Out of plane concentrated loads
  - Shear crimping
    - Shear instability failure
    - Core thickness and shear modulus must be adequate to prevent the core from premature failure in shear under end compression loads



https://www.nature.com/articles/s41598-024-64198-y

#3

- Sandwich structures (cores)
  - Honeycomb – High stiffness to weight ratio, high compressive and shear strength, good thermal and acoustic insulation
    - **Nomex and Korex** – Aramid paper that is flame resistant, has good insulating properties, low dielectric properties, and good formability
    - **Thermoplastics** – Good insulation, energy absorption, moisture and chemical resistance, clean, and low cost
    - **Aluminum** – High strength to weight ratio, good energy absorption, good heat transfer and EMI shielding, low cost
    - **Steel** – Strong heat resistance, good thermal and electromagnetic properties, heavier
    - **Specialty metals (Titanium)** – Expensive, high strength to weight ratio, excellent heat and chemical resistance
    - **Fiberglass** – formable, low dielectric, good insulation, can be manipulated for good shear properties
    - **Carbon fiber** – high stiffness, low thermal expansion, stable at extreme temperatures, but expensive
    - **Ceramics** – expensive, delicate, high temperature resistance, good insulation, small cell sizes



https://enhancedcomposites.com.au/shop/aramid-honeycomb-nomex-core-carbon-fibre-plate-7mm-x-1900mm-x-800mm/

Nomex core (example)

#3 Cont.

- Sandwich structures (cores)
  - Foam Cores
    - Fiber based / soft core – hybrid soft-hard composites, low density, flexible, fatigue resistant, not rigid, NOT used as structural sandwich core
      - Dyneema Core
    - Polyetherimide – Closed cell, high performance, excellent strength, hear and solvent resistant, dimensional stability, expensive
      - Rohacell
    - Polyvinyl chloride – Closed cell, medium to high density foam, strong, durable, fire resistant, formable with head, and resin compatible
      - Divinycell
      - Airex
    - Polypropylene – adhesive and epoxy compatible, dissolves in fuel and solvents, can be cut
    - Polyurethane- Inexpensive, flame resistant, easily shaped
      - Last-A-Foam
    - Phenolic – low mechanical strength, lightweight, fire resistant
    - Polystyrene – Closed cell, aircraft grade, high compressive strength, water resistant, easily shaped
      - Depron

DragonPlate Carbon Fiber Composites

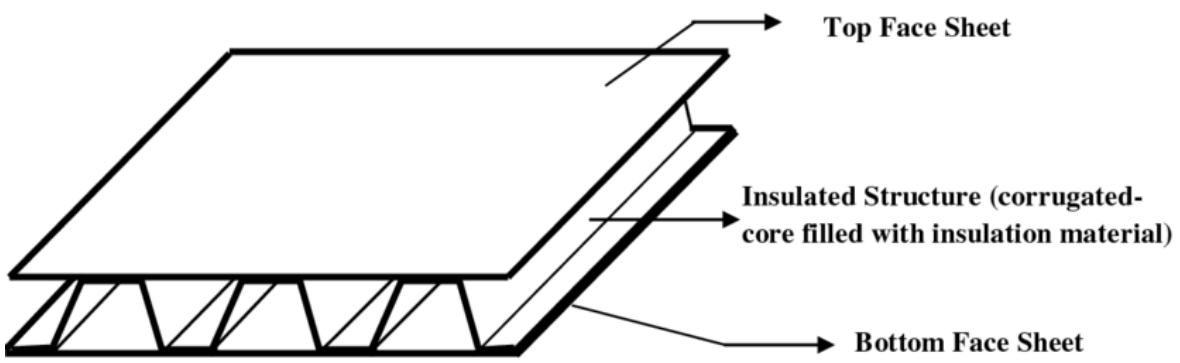
	Comparison Criteria				
Sandwich Products	Stiffness to Weight	Toughness	Crushability	Moisture Resistance	Sound Absorbency
Birch Core	BETTER	BEST	BEST	GOOD	POOR
Balsa Core	BETTER	GOOD	BETTER	POOR	GOOD
Nomex Honeycomb Core	BEST	GOOD	BETTER	BETTER	BEST
Depron Foam Core	BETTER	POOR	POOR	BETTER	BETTER
Airex Foam Core	BEST	GOOD	GOOD	BETTER	BETTER
Divinycell Foam Core	BETTER	BETTER	BETTER	BETTER	GOOD
Last-A-Foam Core	BETTER	BETTER	BETTER	BETTER	BETTER
Dyneema® Core	GOOD	BEST	BEST	BEST	POOR
Kevlar Core	GOOD	BEST	BEST	GOOD	POOR

https://dragonplate.com/carbon-fiber-sheets-with-core#/orderby=5&pagenumber=3

Comparison of different Foam, wood, and honeycomb structures

#3 Cont.

- Sandwich structures (cores)
  - Corrugated – Core is embraced of a corrugation sheet between two thin surface sheets (can be trapezoidal or sinusoidal)
    - High strength-to-weight ratio
    - Enables structure to act as single thick plate
    - Opposes bending, twisting, and vertical shear
    - Resist vertical deformations
  - Balsa wood and birch core (wood)
    - Sustainable and biodegradable
    - High stiffness to weight ratio
    - Good fatigue properties and bonding characteristics
    - High compressive and shear strength
  - Lattice/truss core
    - Load distributed over a large area
    - High strength to weight ratio
    - High energy absorption
    - Mechanical properties can be changed
    - Used in additive manufacturing



https://www.researchgate.net/figure/A-corrugated-core-sandwich-structure-concept-for-Integrated-Thermal-Protection-System_fig11_268475275

Corrugated core

#4

- Key formulas for failure modes
  - Face wrinkling instability
  - Dr. Scott pecks version
    - Face wrinkling load (The edge load applied to the entire sandwich panel)
      - $N_x = 4\sqrt{\beta D_{11}^f}$
    - Where B is the foundational modulus
      - $\beta = \frac{G_c}{h_c}$
    - And D is the bending stiffness of a single faceskin
      - $D_{11}^f = \frac{E_f \cdot t_f^3}{12\left(1-\nu_f^2\right)}$
    - Another more accurate example of face wrinkling instability can be derived – shown below
      - $\sigma_{fw} = \frac{3}{(12(3-\nu_{cxz})^2(1+\nu_{cxz})^2)^{-1/3}} E_{fx}^{1/3} E_3^{2/3}$
      - Where E_{fx} = The modulus of elasticity of the facesheet in the x-direction
      - Where E_3 = Modulus of elasticity of the core material in the z - direction (thickness)
      - Where v_{cxz} = The Poisson's ratio
        - c = Core
        - xz = Coupling between the x and z directions

- Best, most conservative approach in general
  - Skin stress at failure
    - $0.76(E_f E_c E_C)^{\frac{1}{3}}$

#4 Cont.

- Key formulas for failure modes
  - Face dimpling Intercellular buckling
  - Dr. Scott Peck's version
    - Generalized approximation
    - Buckling load of the face sheet over a single honeycomb cell, expressed in terms of the overall load applied to the entire sandwich structure
      - $\widetilde{N_x} = \frac{8}{d^2}\left(D_{11}^f + 2D_{12}^f + D_{22}^f\right)$
        - Where N_x = The stress resultant
        - Where d = The characteristic dimension
        - Where $D_{11}^f\ D_{12}^f\ D_{22}^f$ = Bending stiffens in the x, y, and z directions
  - Intercellular buckling
    - Findingf the interfacial facesheet compressive stress
      - $\sigma_{fi} = \frac{2E_{fx}}{1-\nu_{fxy}^2}\left(\frac{2t}{\alpha}\right)^2$
        - Where E_{fx} = The modulus of elasticity of the facesheet
        - Where v_{fxy} = The Poisson's ratio of the facesheet
        - Where t = The thickness of the facesheet
        - Where α = The cell size

- Overall elastic instability
  - Find the critical in-plane loading
    - $\widetilde{N_x} = \frac{\pi^2 D_{11}}{a^2 + \frac{\pi^2 D_{11}}{hG_{13}}}$
      - Where D_{11} = The bending stiffness of the facesheet in the x – direction
      - Where a = The length of the buckling mode
      - Where h = The thickness of the core
      - Where G_{13} = The shear modulus of the core
- Shear Crimping
  - $\widetilde{N_x} = \frac{4\pi^2 D_{11}}{b^2 + \frac{4\pi^2 D_{11}}{hG_{13}}}$
    - Where b = The length of the buckling mode
    - We know all the other variables as shown above
  - The approximation is as follows
  - $\boxed{\widetilde{N_x} = hG_{13}}$
    - Where h = the core thickness
    - Where G_{13} = The shear modulus of the core

#5

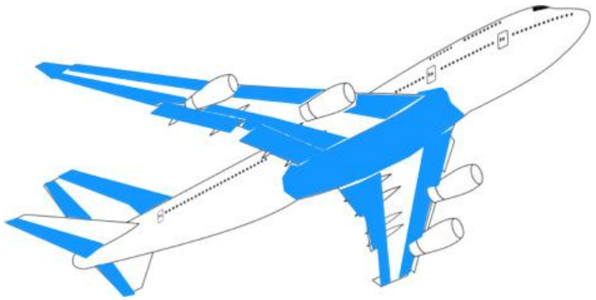
- Benefits of sandwich structure compared to laminates
  - For similar weight, bending stiffness is higher
  - Greater resistance to buckling
  - Under bending moment there is less facesheet stress
- The comparison between a laminate and a sandwich structure can be shown using the following equation

$$\frac{\sigma_{lam}}{\sigma_{sand}} = \frac{D_{sand}}{D_{lam}} \cdot \frac{1}{1+(h/f)} = \frac{1+\frac{E^c}{E^f}\left(\frac{h}{f}\right)^3 + 3\left(\frac{h}{f}\right) + 3\left(\frac{h}{f}\right)^2}{1+(h/f)}$$

- The further apart the facesheet, the greater the bending stiffness
- In class we did an example:
- If Ef = 10 Msi, Ec = 1 ksi, h = 1 in., and f =0.05 inch
  - We get a 60x greater strength on the sandwich than we do on the laminate structure

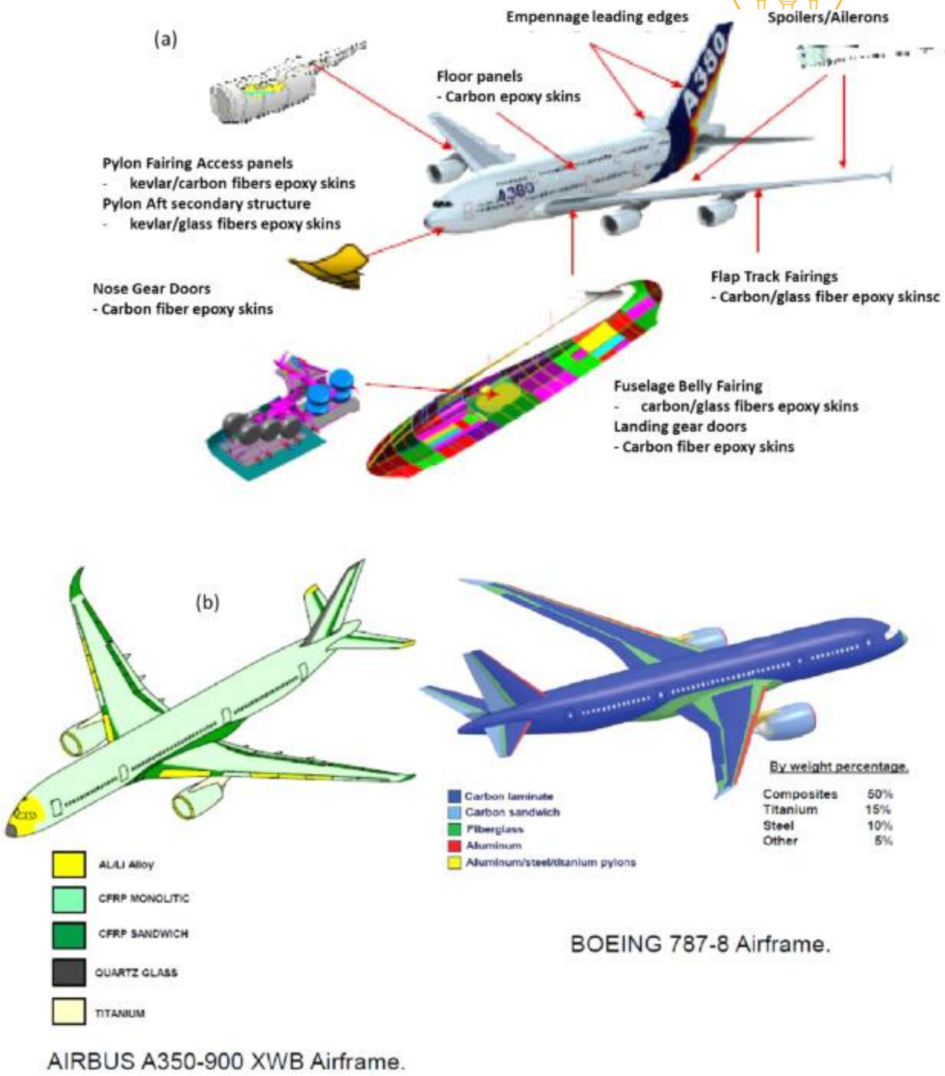
#6

- 5 applications where sandwich structures are used in industry
  - Aircraft
    - Boeing 747 – half the surface of the wing, leading and trailing edges made out of glass fiber and Nomex, belly fairing
      - Flaps are made with sandwich structure but have aluminum honeycomb cores



    - Airbus A320, A330, A340
      - belly fairing, the nacelles, the front landing gear doors, some ailerons and rudder all made out of sandwich structures

- https://www.sciencedirect.com/science/article/pii/S2666682020300049



#6 Cont.

- 5 applications where sandwich structures are used in industry
  - Construction
    - Paneling on walls, roofs, floors
    - I-Beams with honeycomb core
  - Marine structures
    - Decks
    - Hull panels and bulkheads
    - Interior walls/reinforcement areas
  - Automotive industry
    - Used in motorsport for light weight, strong mechanical properties
    - Hood - Combining thermoset and thermoplastic polymers with reinforcing fibers, has great impact resistance, good strength to weight ratio
    - Chassis – syntactic foam used as a common core material
    - Bumpers, doors, roofs, trunk, leaf springs
  - Defense/military
    - Armor systems, vehicle hulls, blast protections
    - Lightweight ballistic protection systems
    - Military is a combination of many industries

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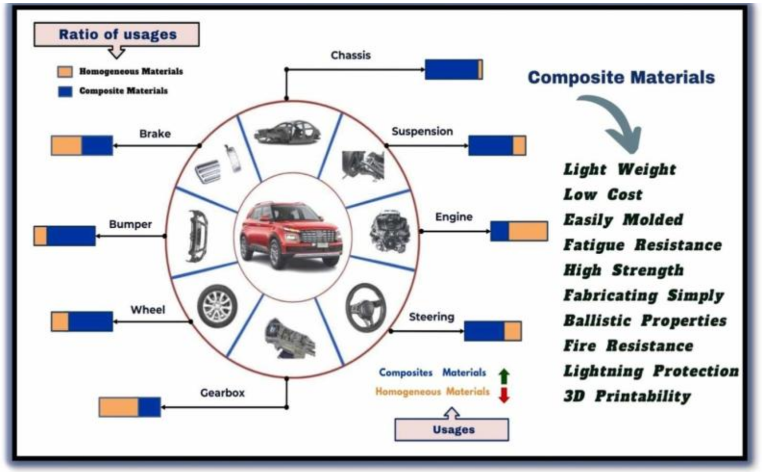
https://www.sciencedirect.com/science/article/pii/S2307187724000440

Types of composites	Advantages	Usage in Automobile Components	Ref.
Carbon Fibre Reinforced Polypropylene	Tensile and Flexural Strength, Hardness, Impact Resistance	Chassis especially Bonnet	[182]
Glass Fibre Reinforced Thermoplastic	Light Weight, High Strength and Rigidity, Durability	Front End, Seat Frame, Engine Noise Shield, Bumper, Instrument Panel Bracket	[183]
Ceramic Matrix Composite	Elevated Hardness, Wear Resistance	CAM	[184]
Epoxy SMC	Dimensional Stability, Low Thermal Expansion	Tire Frame, Outer surface	[185]
Aluminium Matrix Composite	High Quality to Weight Proportion, Malleability, Modulus, Low Warm Coefficient, Wear and Consumption Obstruction	Exhausts, Liners, Cylinder, Brake Plate, Brake Drum, Drive Shaft	[186]
Natural Fibre Composite	High Strength, Stiffness and Ductility, Moisture Absorbability	Seatbacks, Door Panel, Brake Panel	[187]
Synthetic Fibre Composite	Good Flexural Strength, and Interlaminar Shear Strength	Seat Cover, Door Panel	[188]



#7

- Using excel calculator compared to problem 1
- Excel calculator – Aluminum core 5056
  - \+ 1 in. thick
  - Plate 5x5 in
  - Subject to bending loads M_{xx} = 1000 lbs. and M_{yy} = -2000 lbs.
  - Moment applied M_{xy} = 300 lbs.
  - Weight comparison –
    - The sandwich structure is significantly heavier
    - The laminate weighs 0.1136 lb. and the sandwich weighs 2.6273 lb.
    - Weight increase in the sandwich by 23.12%
  - Strength comparison –
    - The sandwich structure shows much lower fiber stress and strains, due to bending stiffness from the thick core
    - The curvatures for kx and ky are lower for the sandwich = less deformation
    - The laminate will fail and cannot carry these loads
  - The load distribution –
    - The core carries almost – no in-plane stress but has a large moment
    - Stresses are taken up by the outer composite facesheets

$$\frac{\sigma_{lam}}{\sigma_{sand}} = \frac{D_{sand}}{D_{lam}} \cdot \frac{1}{1+(h/f)} = \frac{1+\frac{E^c}{E^f}\left(\frac{h}{f}\right)^3 + 3\left(\frac{h}{f}\right) + 3\left(\frac{h}{f}\right)^2}{1+(h/f)}$$

Moment T	Value (lb-in/in)
Mxx	1000
Myy	-2000
Mxy	300

Project 10
1.160E+07
8.000E+05
0.34
3.100E+05

Thickness	Angle (°)
0.009	0
0.009	30
0.009	90
0.009	45
0.009	0
0.009	0
0.009	45
0.009	90
0.009	30
0.009	0

#7

- Using excel calculator compared to problem 1
- Excel calculator – Aluminum core 5056

Laminate: Mechanical Strains and Stresses (Locally Layer by Layer)

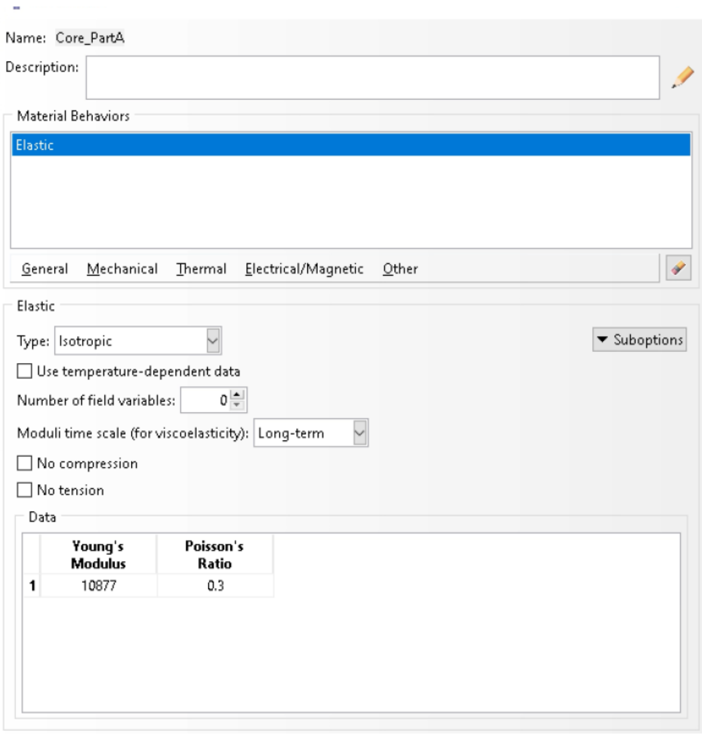
eps_1	10.161%	6.585%	-31.711%	-2.002%	1.129%	-1.129%	2.002%	31.711%	-6.585%	-10.161%
eps_2	-57.079%	-43.077%	5.645%	-13.638%	-6.342%	6.342%	13.638%	-5.645%	43.077%	57.079%
gamma_12	34.907%	-31.717%	-19.393%	-22.414%	3.879%	-3.879%	22.414%	19.393%	31.717%	-34.907%
sig_1	1.032E+06	6.519E+05	-3.693E+06	-2.715E+05	1.146E+05	-1.146E+05	2.715E+05	3.693E+06	-6.519E+05	-1.032E+06
sig_2	-4.324E+05	-3.293E+05	-4.142E+04	-1.155E+05	-4.805E+04	4.805E+04	1.155E+05	4.142E+04	3.293E+05	4.324E+05
tau_12	1.082E+05	-9.832E+04	-6.012E+04	-6.948E+04	1.202E+04	-1.202E+04	6.948E+04	6.012E+04	9.832E+04	-1.082E+05

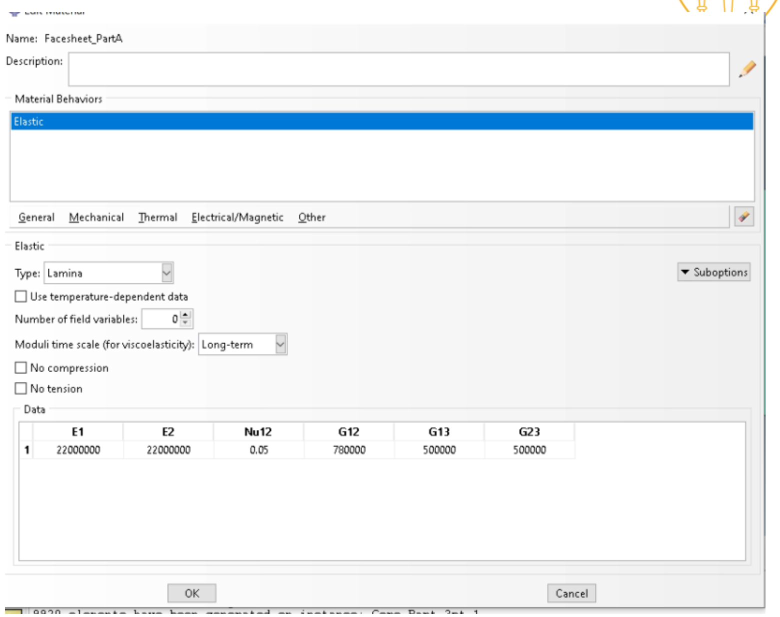
Sandwhich Structure: Mechanical Strains and Stresses (Locally Layer by Layer)

eps_1	0.385%	0.375%	-1.510%	-0.030%	0.360%	0.000%	-0.360%	0.030%	1.510%	-0.375%	-0.385%
eps_2	-1.562%	-1.532%	0.373%	-1.088%	-1.458%	0.000%	1.458%	1.088%	-0.373%	1.532%	1.562%
gamma_12	1.114%	-1.111%	-1.077%	-1.850%	1.040%	0.000%	-1.040%	1.850%	1.077%	1.111%	-1.114%
sig_1	4.079E+04	3.962E+04	-1.756E+05	-6.447E+03	3.808E+04	1.010E-28	-3.808E+04	6.447E+03	1.756E+05	-3.962E+04	-4.079E+04
sig_2	-1.154E+04	-1.132E+04	-1.135E+03	-8.857E+03	-1.077E+04	0.000E+00	1.077E+04	8.857E+03	1.135E+03	1.132E+04	1.154E+04
tau_12	3.454E+03	-3.443E+03	-3.339E+03	-5.736E+03	3.224E+03	0.000E+00	-3.224E+03	5.736E+03	3.339E+03	3.443E+03	-3.454E+03

#8

- Abaqus Part A – Setup
  - [0/45/45/0] woven
  - Thickness of core = 0.75
  - Length = 13 in.
  - P length = 5 in.
  - Width = 2 in.
  - E11 = E22 = 22 Msi
  - V12 = 0.05
  - G12 = 0.78 Msi
  - G13 = 0.5 Msi
  - G23 = 0.5 Msi
  - Ec = 10877
  - V= 0.3
  - Tensile strength = 200 psi

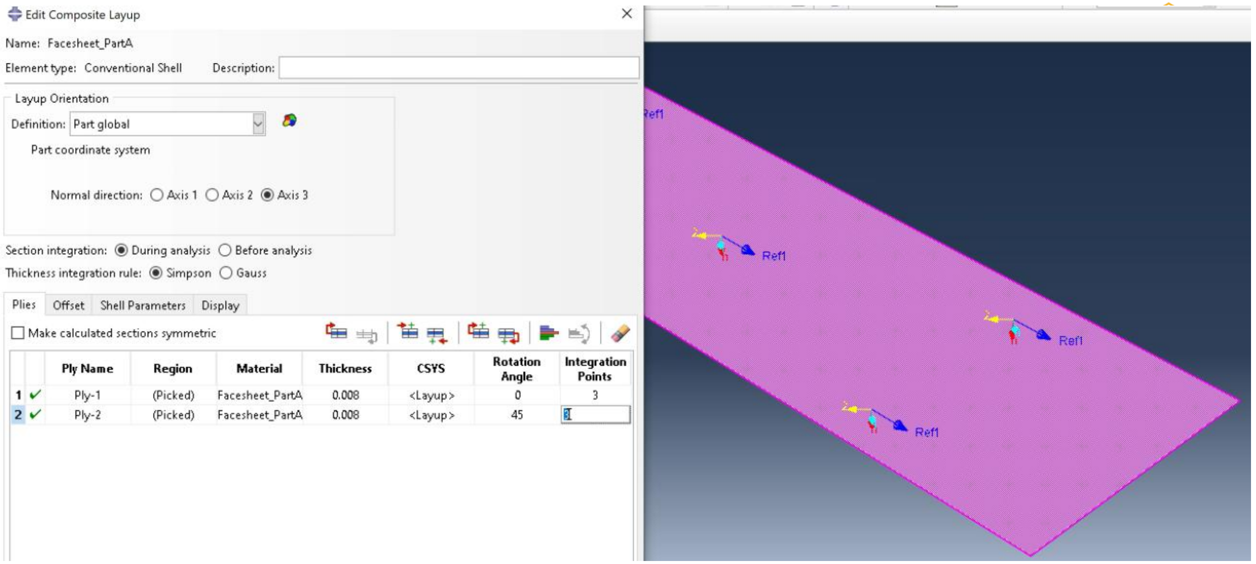


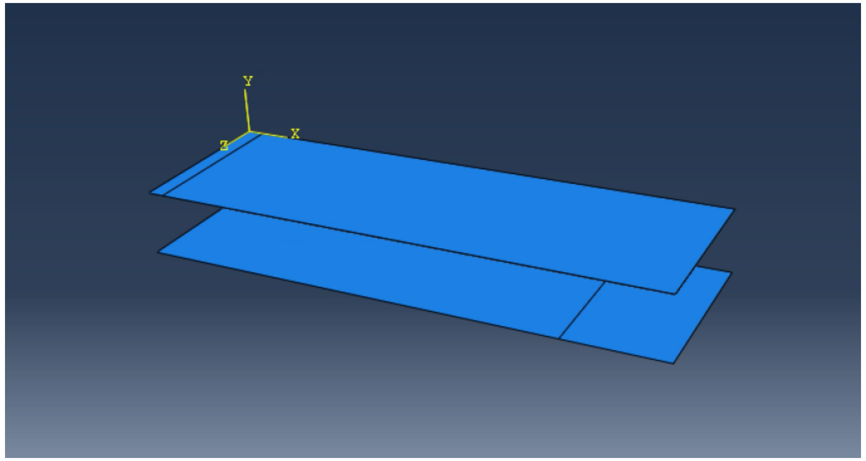


- Material properties are set for both the Core and the Facesheet following the directions of the problems
- The values are different than the shown video

#8

- Abaqus Part A – Setup
    - [0/45/45/0] woven
    - Thickness of core = 0.75
    - Length = 13 in.
    - P length = 5 in.
    - Width = 2 in.
    - E11 = E22 = 22 Msi
    - V12 = 0.05
    - G12 = 0.78 Msi
    - G13 = 0.5 Msi
    - G23 = 0.5 Msi
    - Ec = 10877
    - V= 0.3
    - Tensile strength = 200 psi

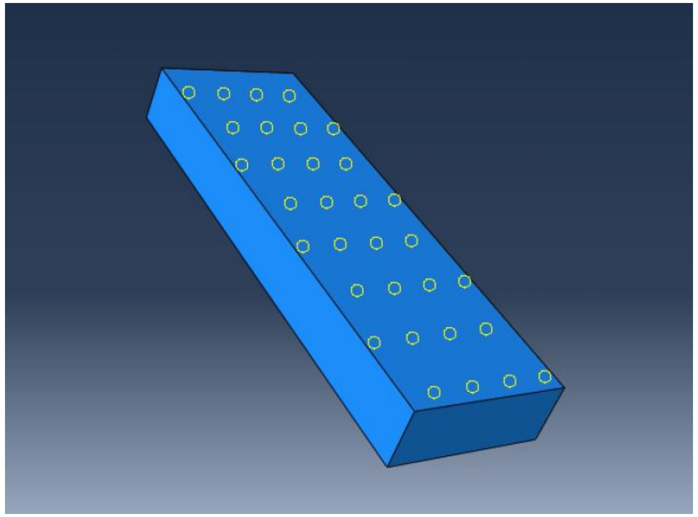


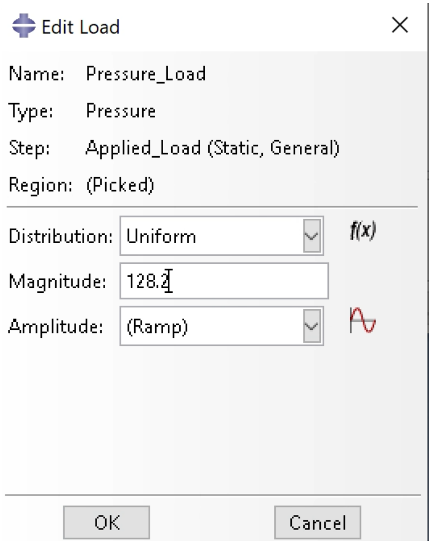


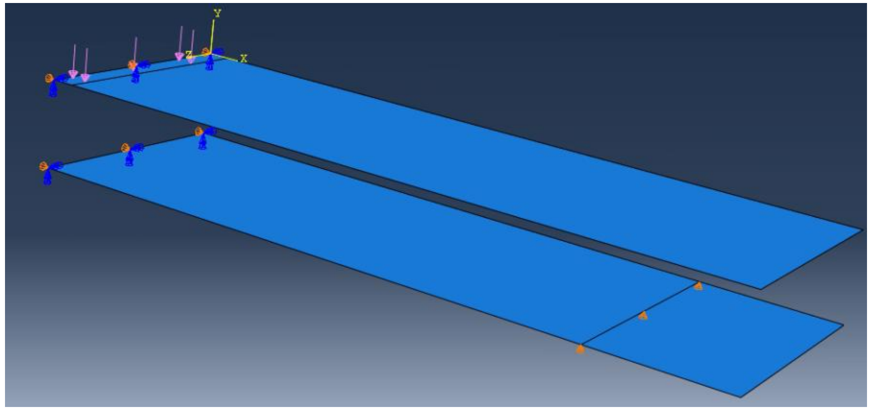
- Material properties for the shell material (Facesheet) is applied in the composite layup section
    - The ply orientation, thickness (0.008 in), and integration points are set
    - The ply is made symmetric = 4 total plies
- The facesheets are partitioned to simulate the 3-pt bending test
    - Shown with the two lines on the lower image

#8

- Abaqus Part A – Setup
  - The material interaction is set on the bottom and top of the core
  - Boundary conditions on the left side of the facesheet are set to symmetric (Encastre)
  - There is a rotating boundary condition on the right-side partition
  - The load is then set to 128.2 as calculated in the video and placed on the left most edge – in-between the partition and the edge
  - This loading condition was a mistake, the proper applied load should be P=F/A = 100/2 x 0.2 = 250 psi

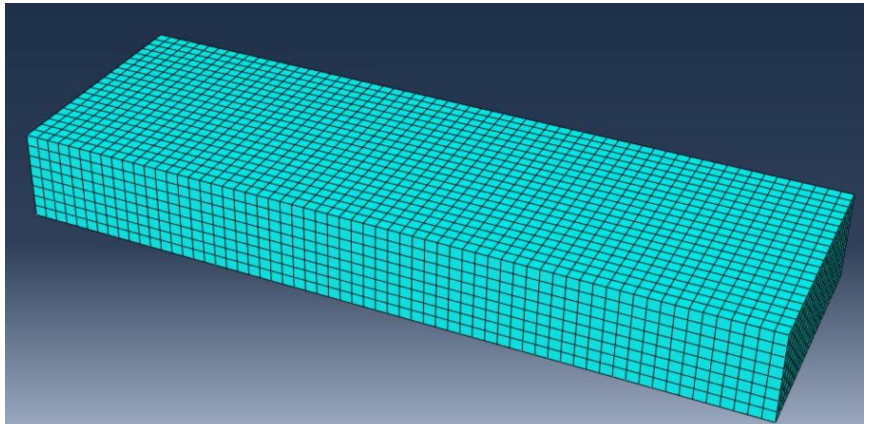


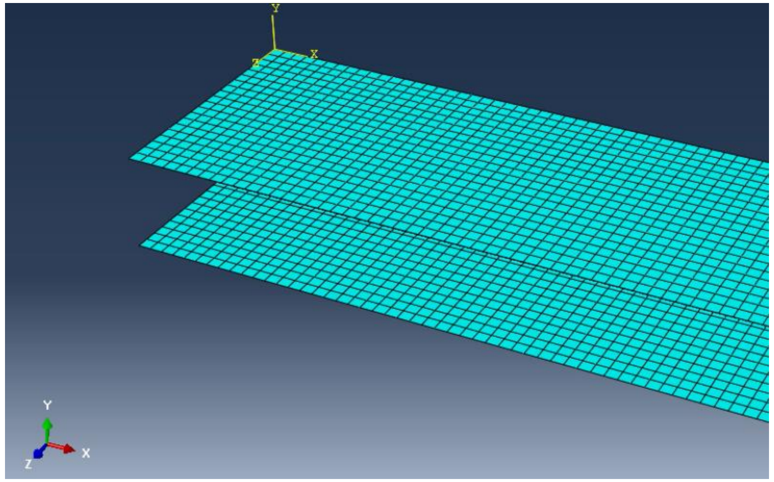




#8

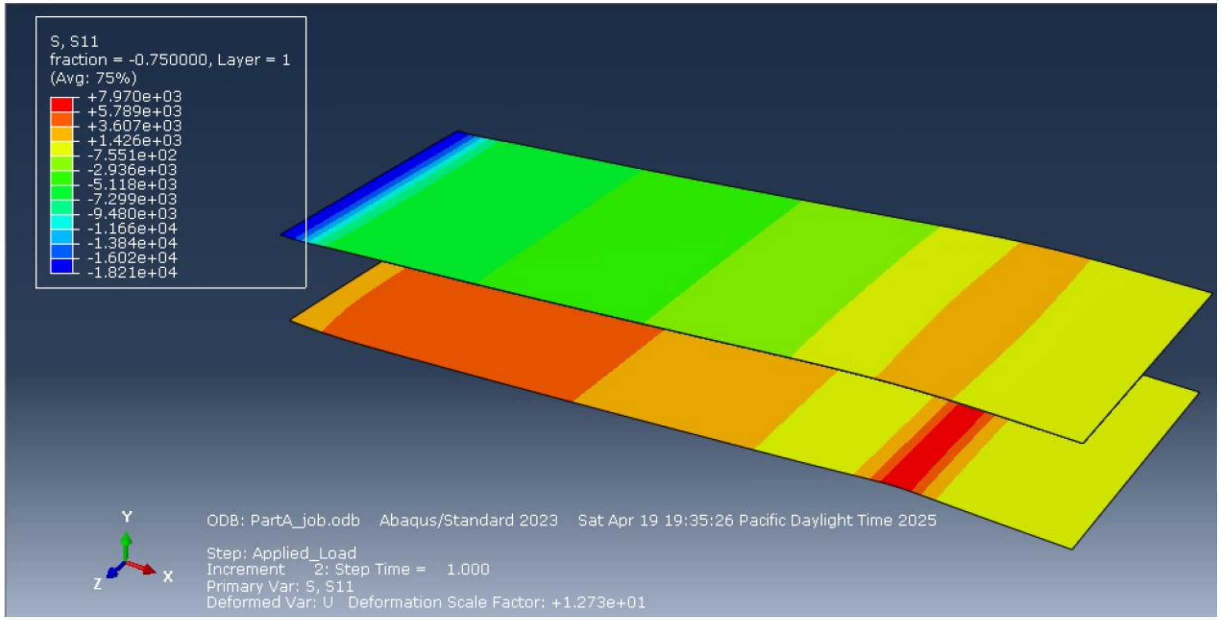
- Abaqus Part A – Setup
  - The mesh setup is shown above
  - Quad-structured mesh with seeded edges = 0.1

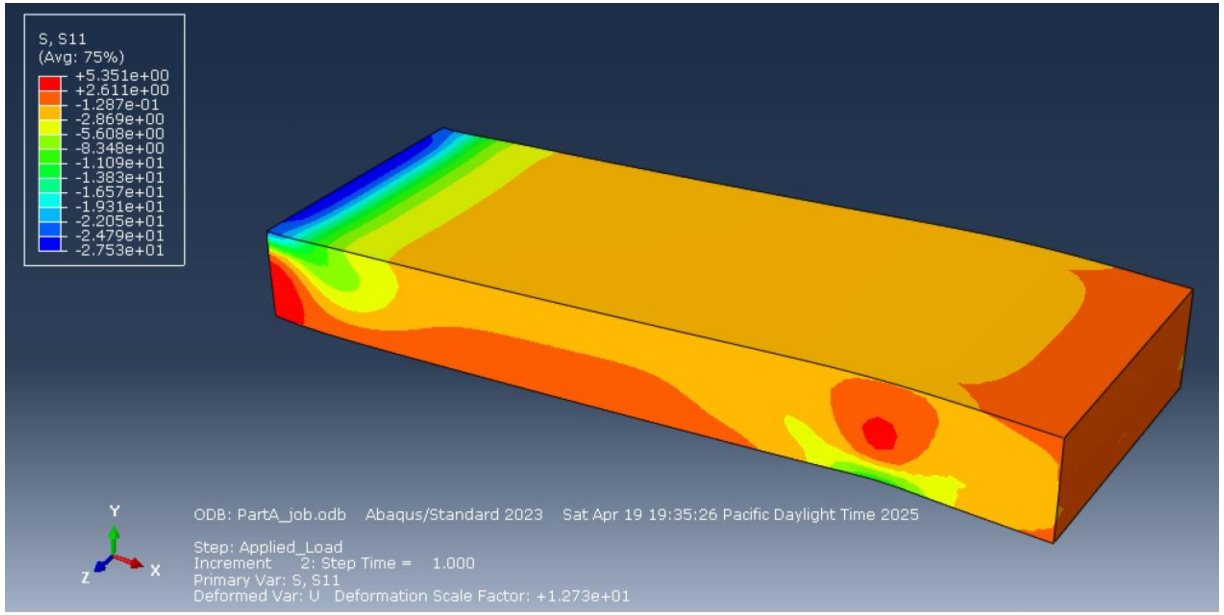




#8

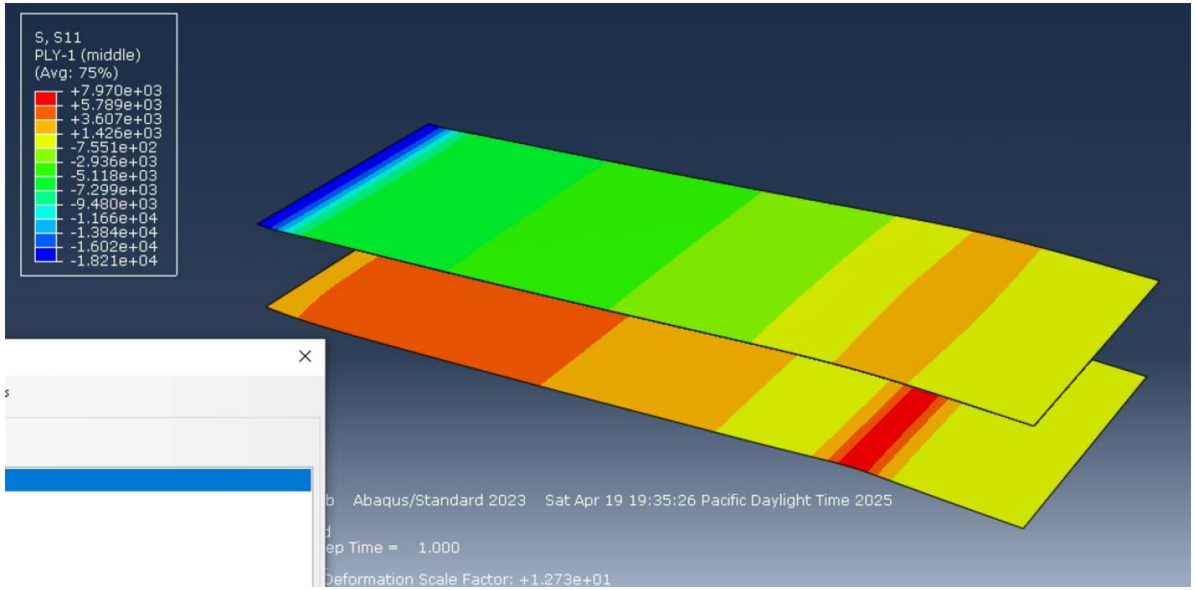
- Abaqus Part A- Results
  - The contour plot represents the S11 strains for both the core and the facesheet – indicating where the material is experiencing the highest stress
  - Half the composite is mirrored to simulate the full bending test. This is achieved using the symmetrical and simple support boundary conditions

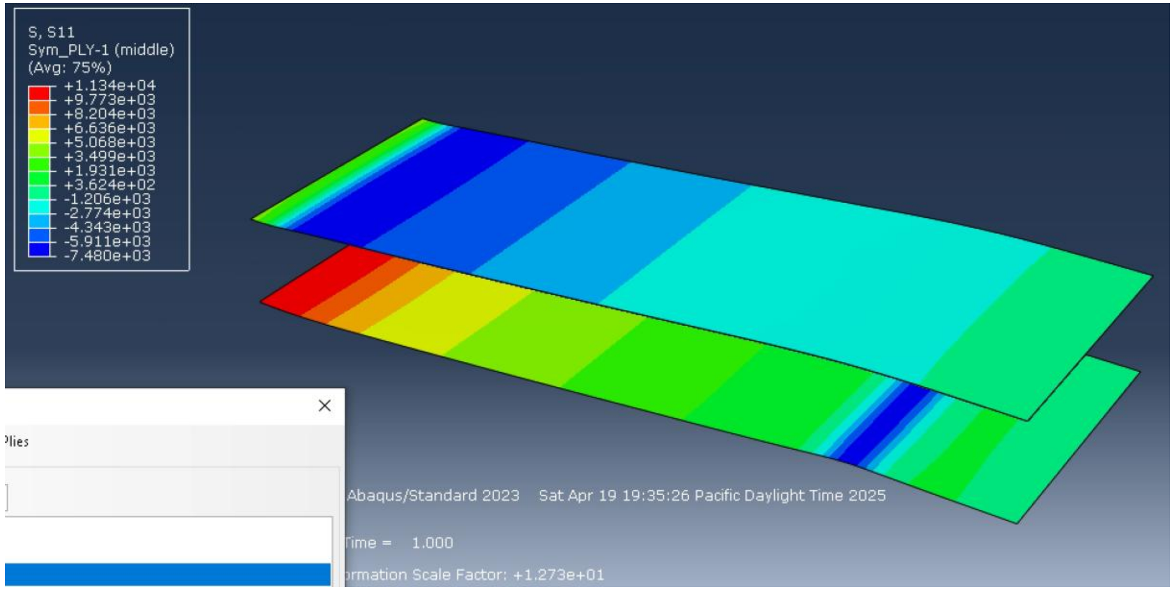




#8

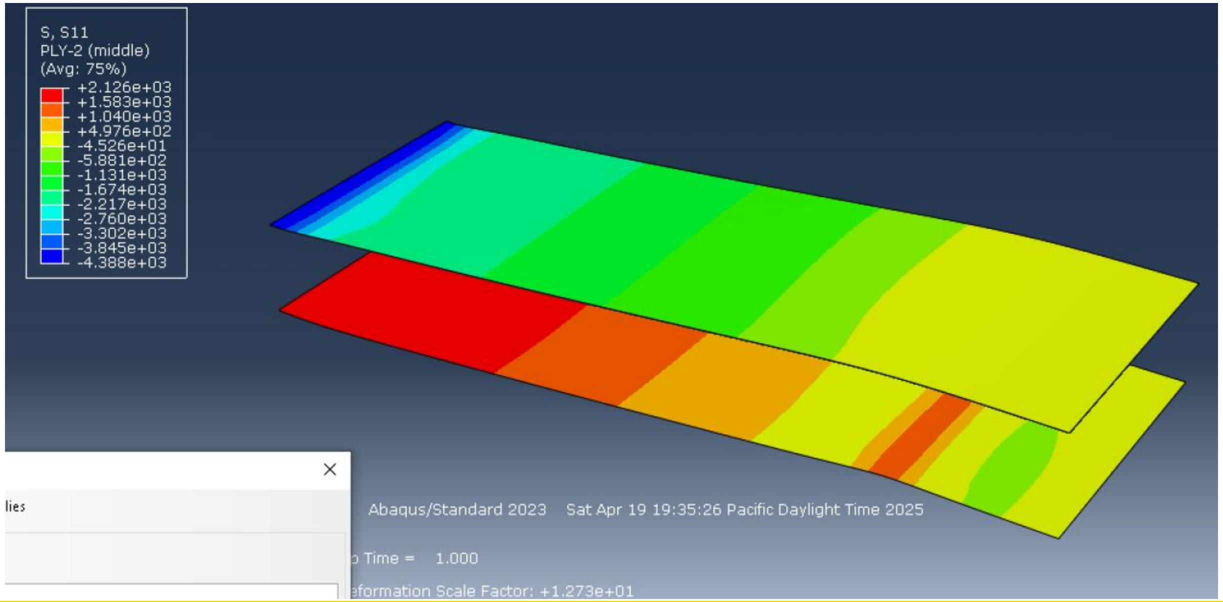
- Abaqus Part A- Results
  - S11 for ply 1 and the following symmetric ply contour plots are shown below
  - For S11 we see a max load of 7970 psi in ply 1
  - We know that S22 results have a great stress concentration on the pressure load coming down. S11 is greater for pressure loads going up
  - The middle plies (like ply 2 and its symmetric pair) are experiencing much lower axial stress, reinforcing that bending is the dominant loading mode, not pure tension – shown in the next slide

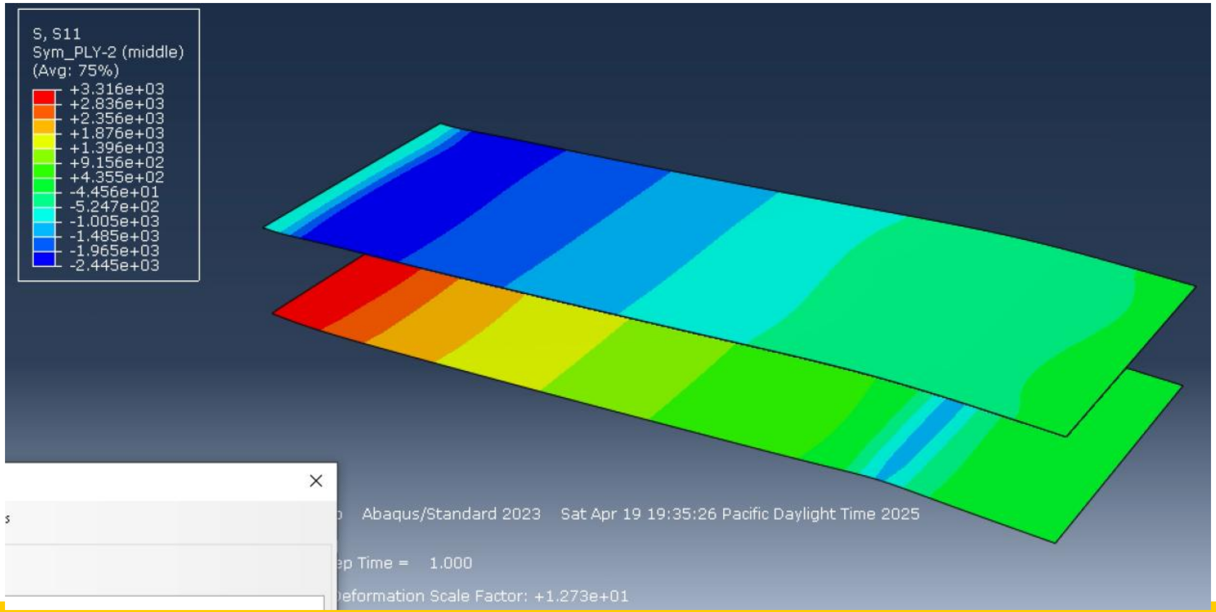




#8

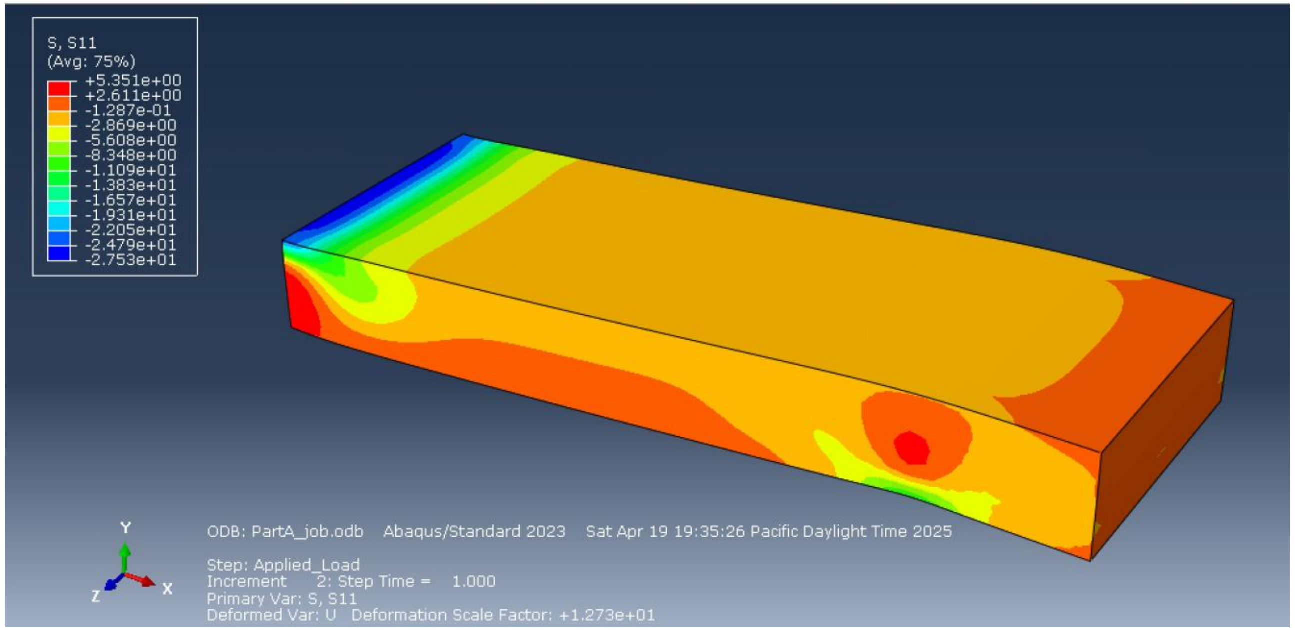
- Abaqus Part A- Results
  - S11 for ply 2 and the following symmetric ply contour plots are shown below
  - These are much lower than ply 1 which makes sense
  - S11 in ply 2 has a max stress concentration of 2126 psi
    - More than half of what the S11 max stress was
  - Also, note that in the symmetric plies, the max stress is lower in the second ply by more than half as well
  - Outer ply angles and orientations have a more critical role in determining strength

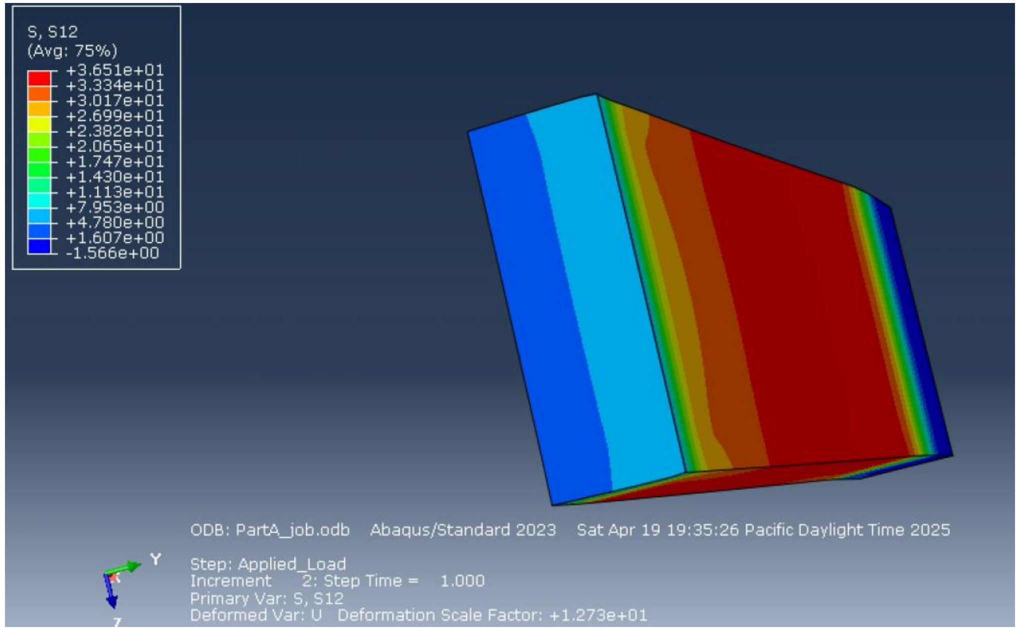




#8

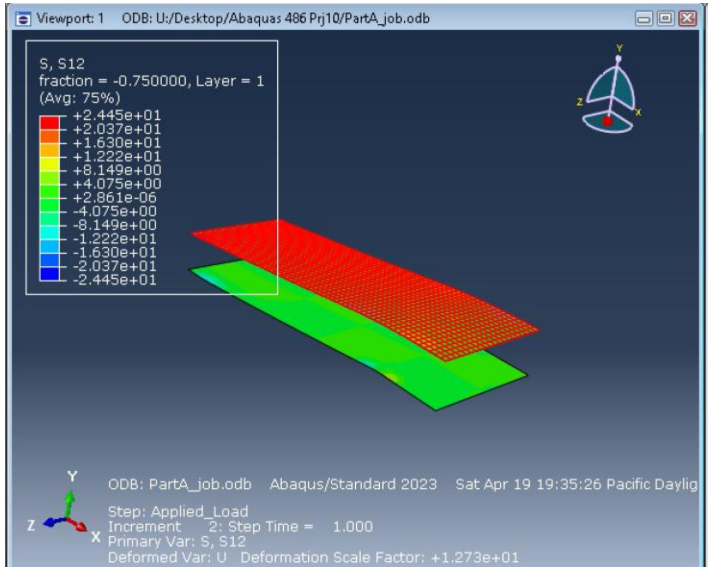
- Abaqus Part A – Results
  - The S12 contour plot visualizes the stress concentration for the core of the composite structure
    - The image on the right shows the uniform stress concentrations in light blue and blue colors
- The maximum principal stress is 36.5 psi (on the right)
- The max S11 of the core is 53.5 psi

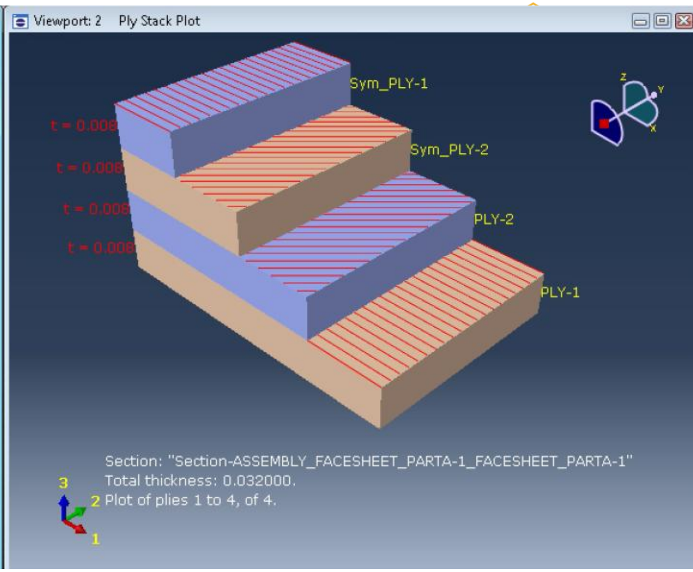


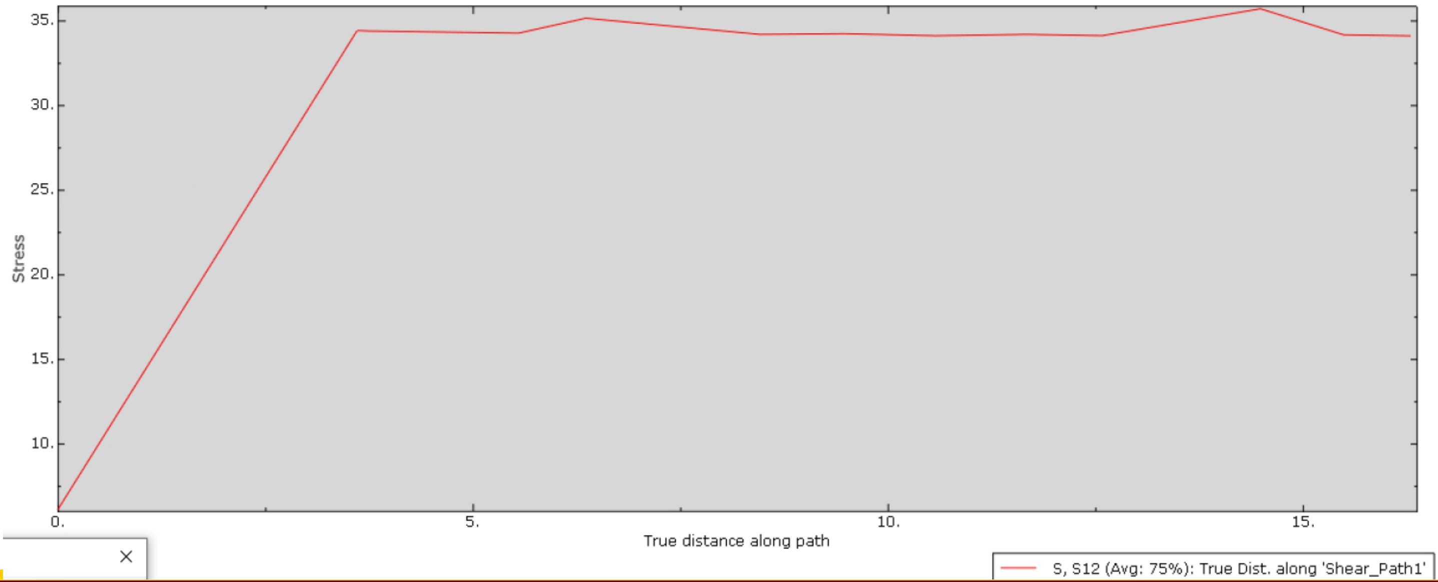


#8

- Abaqus Part A- Results
  - The S12 maximum principal stress is 36.5 psi
  - This is plotted in the image below with section points by taking the cross section of the core
  - These values match the contour plot figures
  - An image of the Ply angles and ply number (4) with there orientation is shown in the above image
  - Our tensile strength is 200 Psi thus, our core can take this pressure load easily with plenty of margin

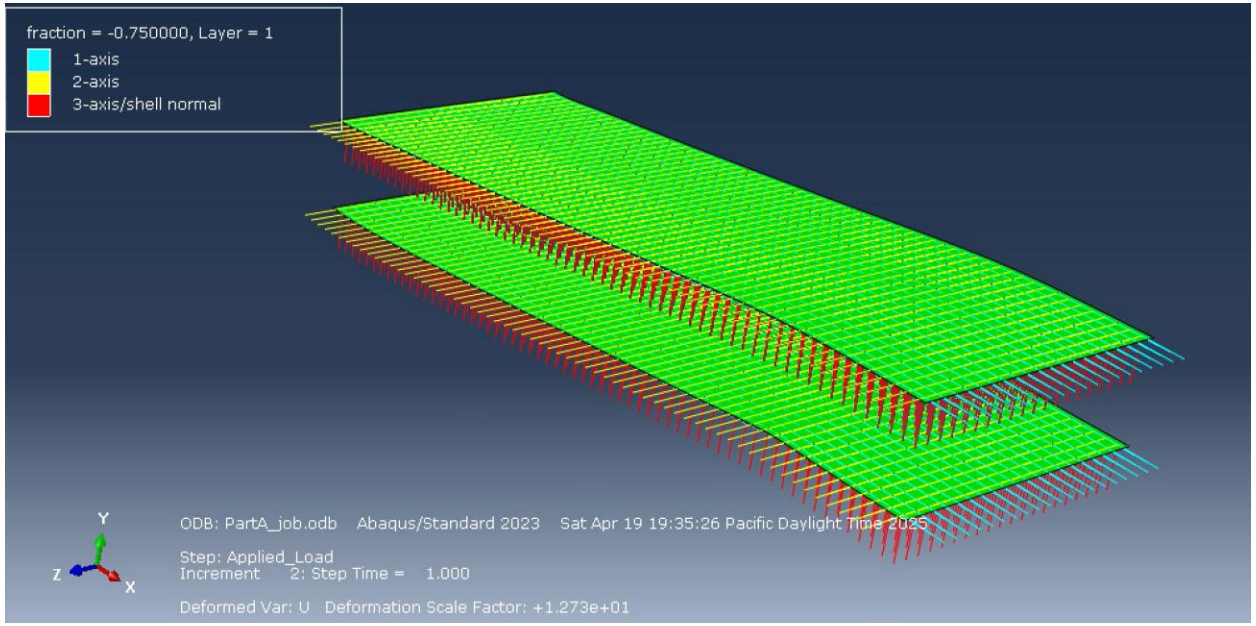


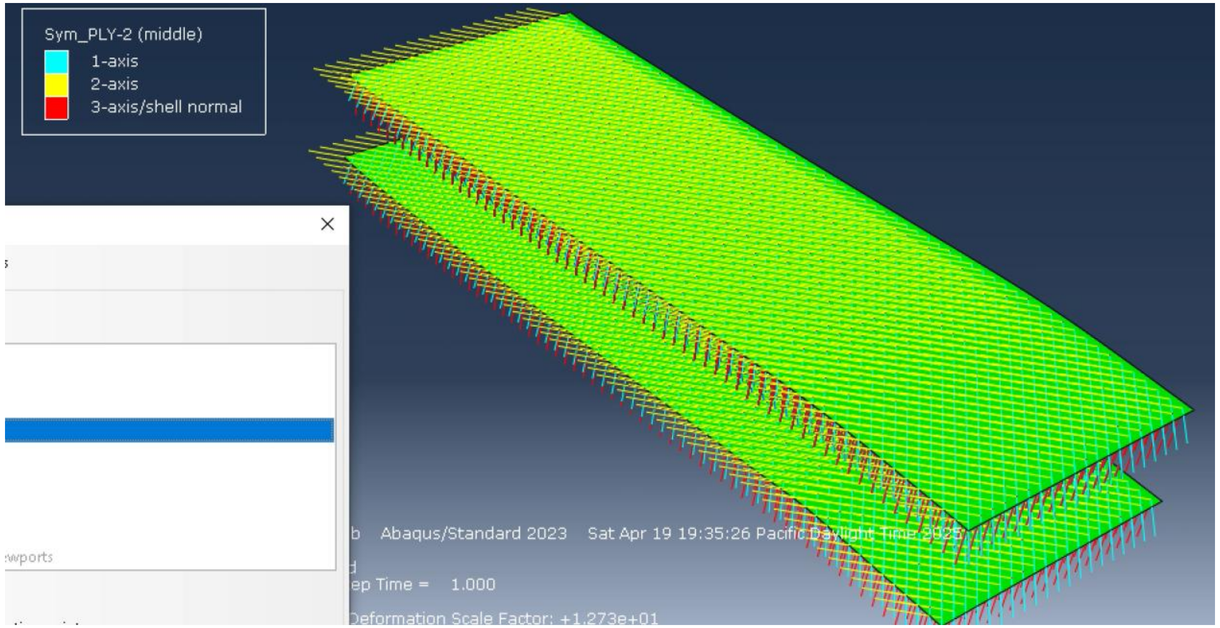




#8

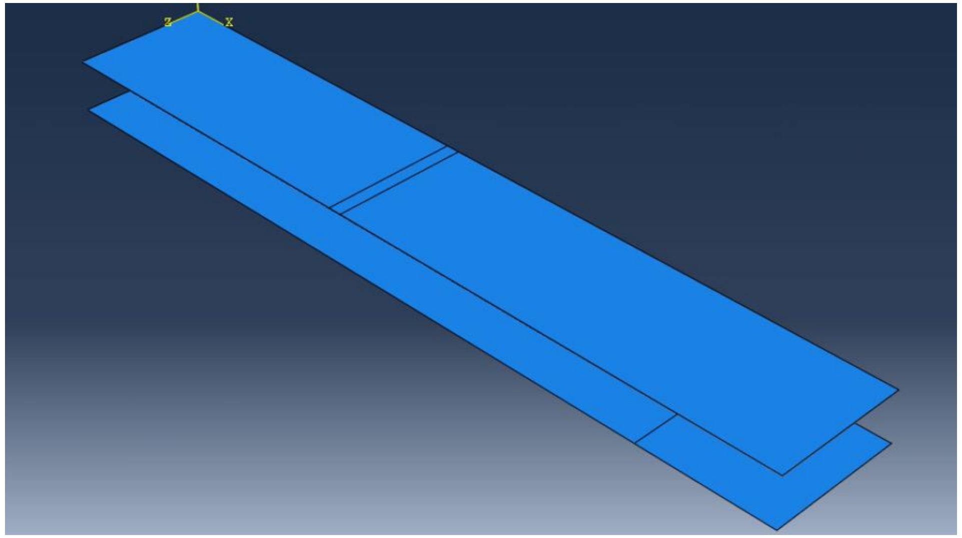
- Abaqus Part A – Results
  - Ply orientation visualization
  - This is not needed in the problem but a good example of what the software can do

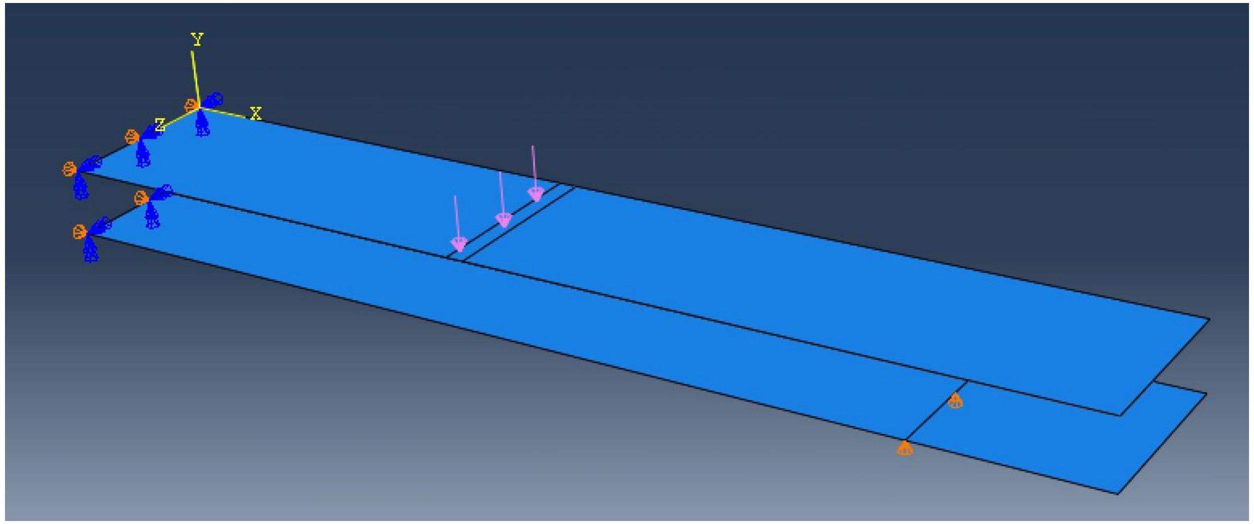




#8

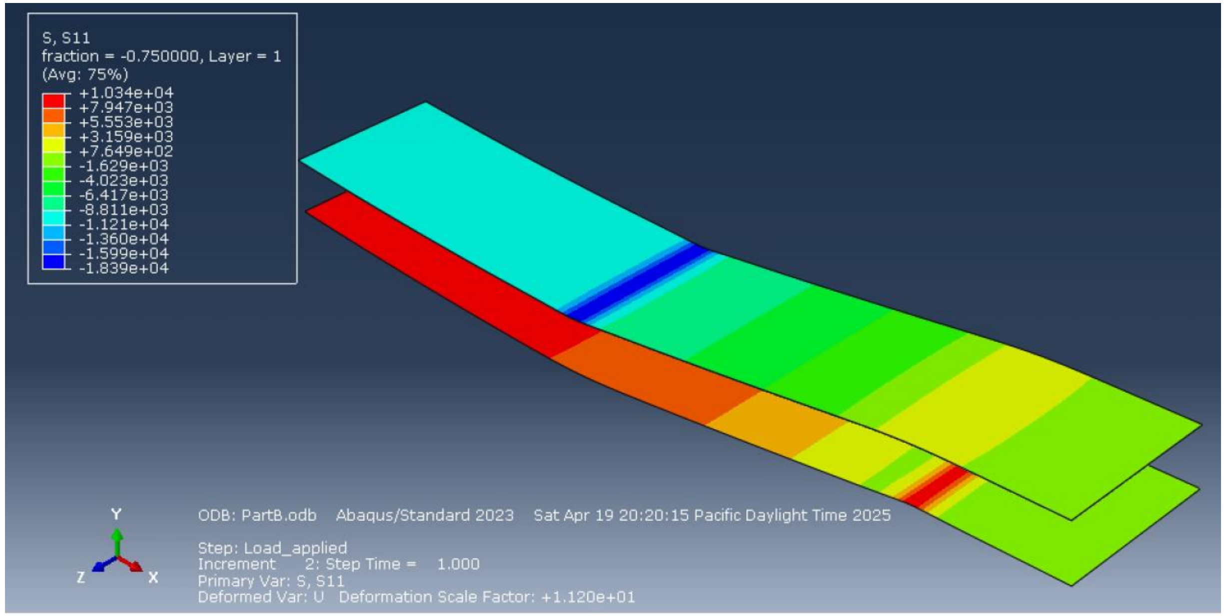
- Abaqus Part B – Setup
  - The setup for Part A and Part B are the same except for
  - The governing dimensions of the part
  - The 4pt bending setup partitions on the facesheet show below

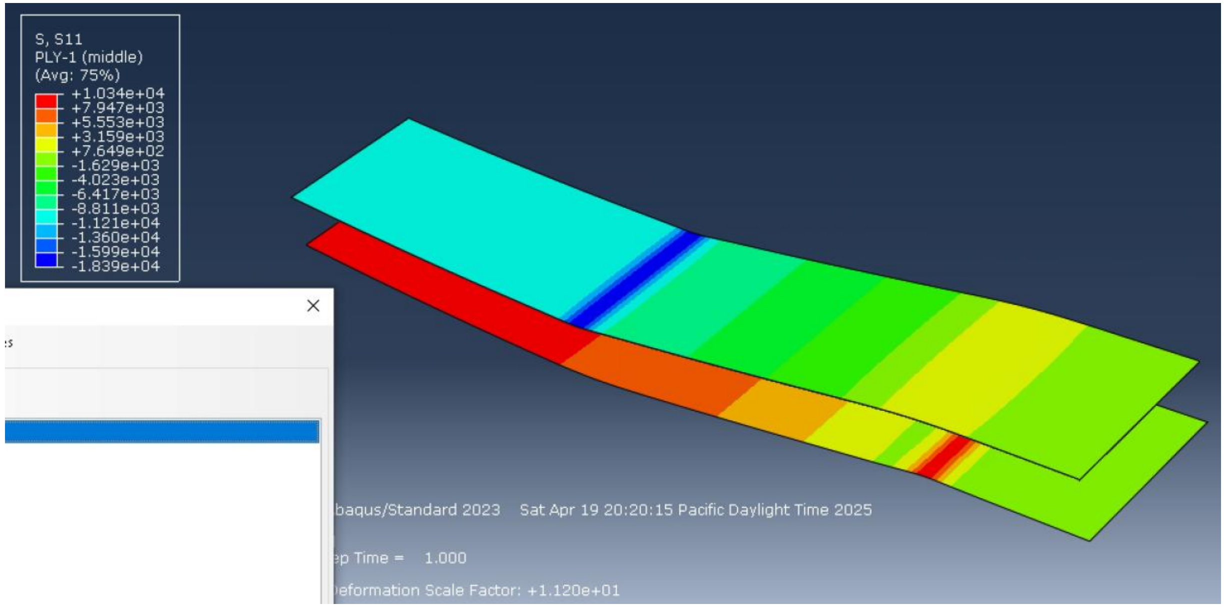




#8

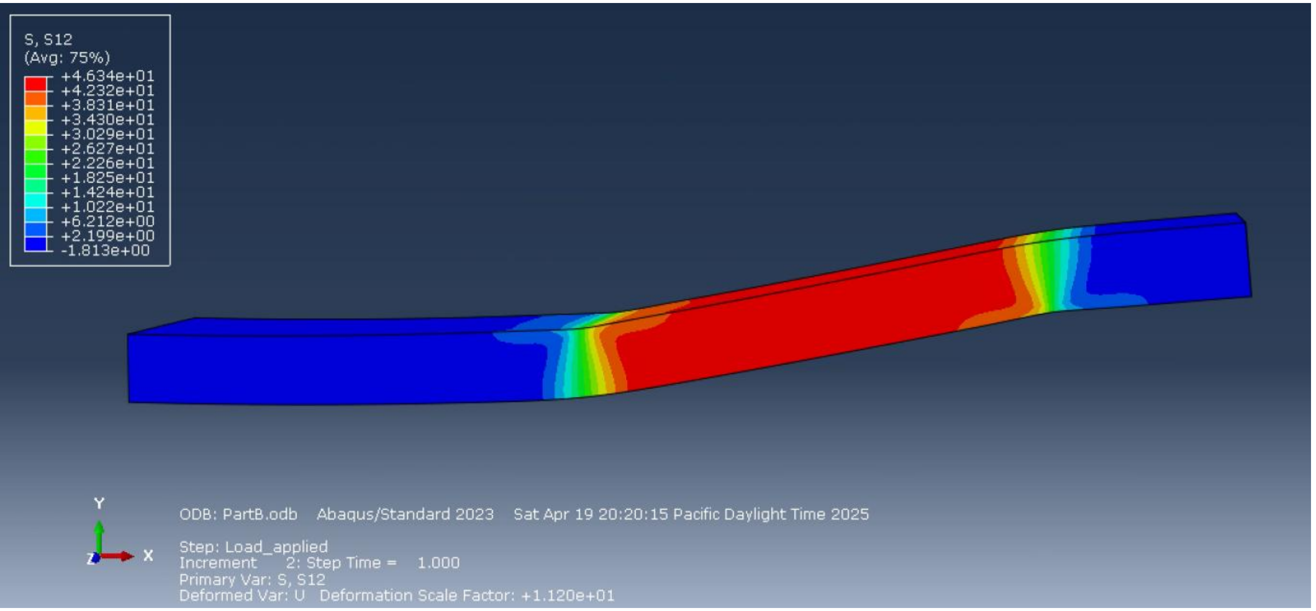
- Abaqus Part B – Results
  - Keep in mind we used the incorrect pressure of 266 psi as used in the video; I was supposed to calculate F = P/A
  - The S11 stress concentration is shown below, where the highest stress concentration is on the bottom facesheet at 10,340 psi
  - There is both compressive stress and tensile stress
  - The stress concentration is much higher on the bottom facesheet due to a high localized stress either from the small area distribution, or the resistance to the applied load

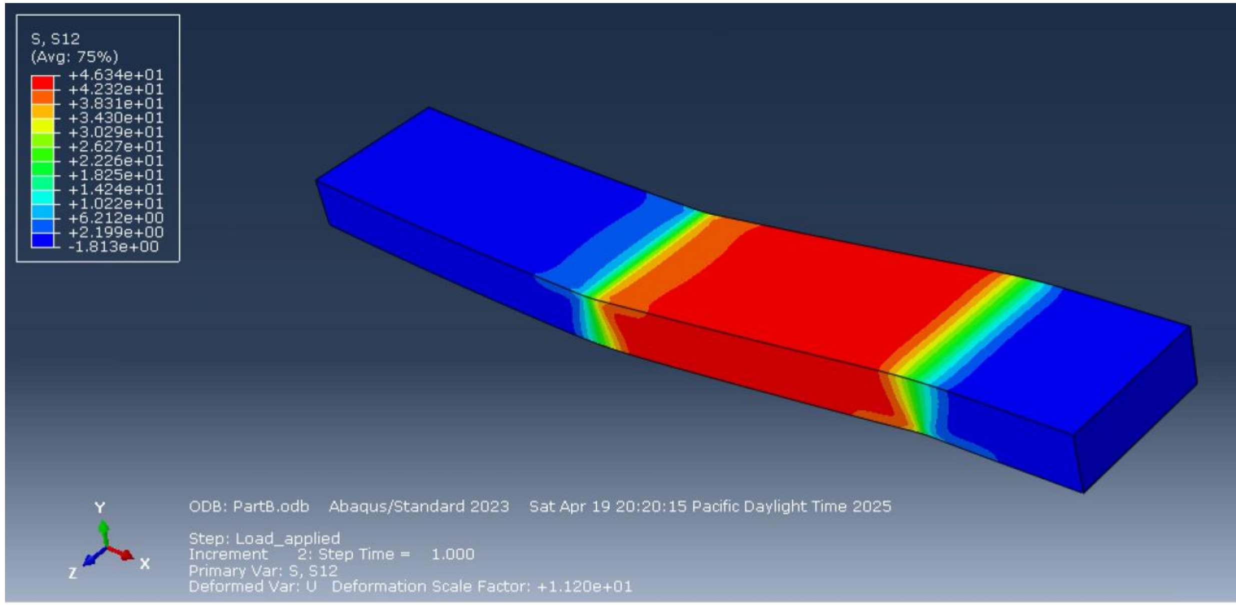




#8

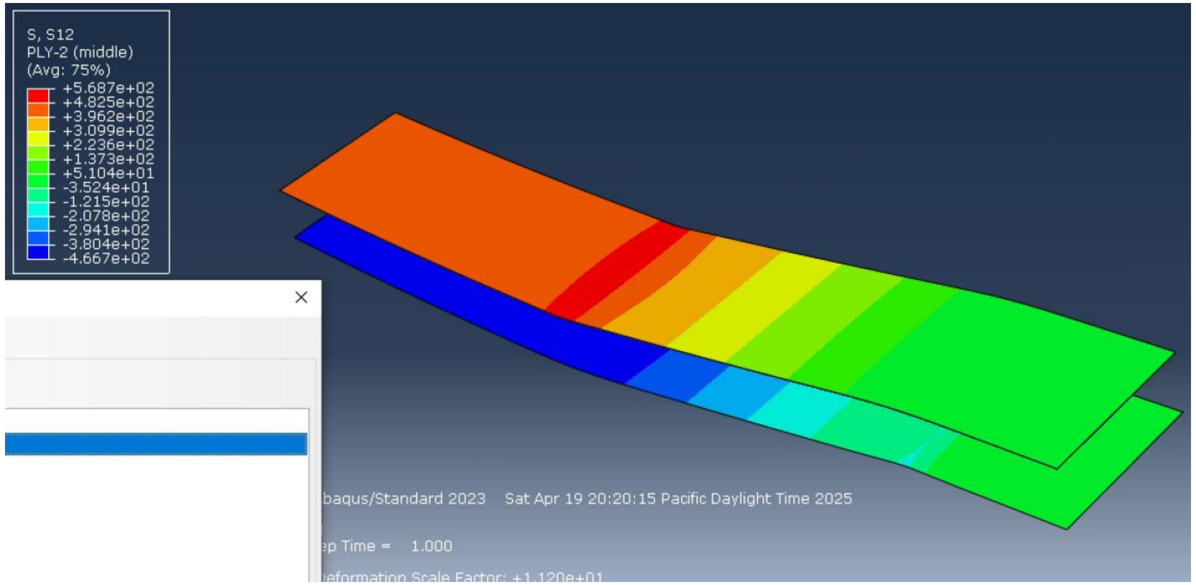
- Abaqus Part B – Results
  - The stress concentration for the S12 core are shown below
  - The max stress concentration in the S12 direction is 46.34 psi
  - There is a lot of visual deformation shown in the side profile image
  - The structure is in bending, the top sheet is compressing down, and the bottom facesheet stretches in tension
    - S12 is highest in the middle shown in red

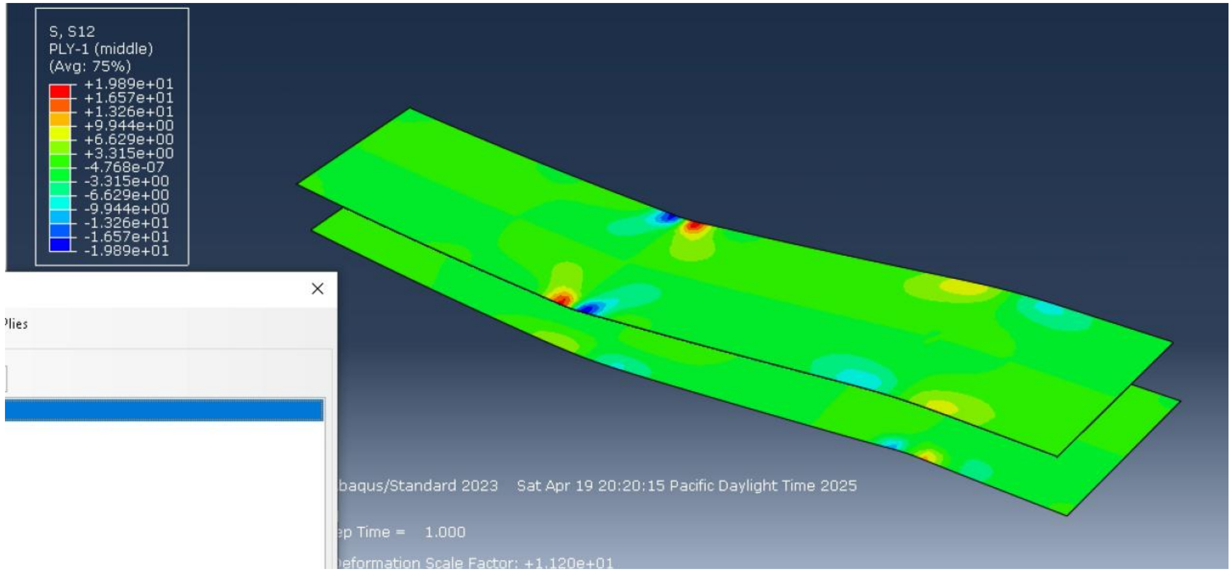




#8

- Abaqus Part B – Results
  - The Ply 1 and Ply 2 layers are plotted below
  - There is significantly more stress in the second ply – max stress = 568.7 psi
  - There is less stress in the ply 1 direction - 19.89 psi
  - Ply orientation matter in shear (S12)
    - Shear stress is highest in off-axis plies, this makes sense





#8

- Abaqus Part B – Results
  - The cross-section of the core is measured and plotted on the figure below
  - There is less uniformity and more discrepancies in the cross-section load distribution
  - This could be caused by non-uniform load distribution due to material inconsistencies, imperfections in the geometry, uneven load transfer, or poor measurement technique by me
  - To see when the compression strain exceeds 1.2%, I combined the strain and reaction force data into a plot
    - I was unable to see anything of value
  - Referring to the contour plot we see S11 minimum of -18,390 psi
  - $E = 18{,}390/\ 18{,}000{,}000 = -0.0001022$
  - $P \max = 1.0 \cdot 0.012/0.0001022 \approx 1174 / \text{Area}$
  - P max = 48 Psi
  - To not exceed a compression strain of 1.2% you can only apply roughly around 48 psi
  - Note: this value is a little off due to using the wrong applied load of 266 psi
  - Generally estimating, around 38 psi is what we can apply without exceeding the 1.2%