FINITE ELEMENT ANALYSIS

TITLE: Finite Element Analysis of a Closet Rod Under Loads
PREPARED BY: Kyle Jackson
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1. OVERVIEW & OBJECTIVE

This project models a residential closet rod loaded with a realistic set of clothes, including hoodies, winter coats, and fishing waders. The clothing was represented using multiple discrete downward point loads applied along the span of the rod. The objective of the analysis was to determine the maximum deflection of the rod and the reaction forces at the supports. Results were checked using mesh refinement and a full hand calculation based on classical beam theory.

2. MODELING APPROACH

A linear static structural analysis was completed in ANSYS Mechanical. The closet rod was modeled as a one-dimensional beam using quadratic beam elements. A beam model is appropriate because the rod is slender relative to its span, and the primary response is bending. The beam centerline was aligned with the global X axis. All applied garment loads were vertical and applied in the global Z direction. Large deflection effects were not enabled. The final maximum deflection was small compared to the span length, so linear assumptions were appropriate for this analysis.

3. GEOMETRY

The geometric model consisted of a 45 inch line body representing the centerline of the closet rod. The cross section is an oval hollow section, and it was defined using a user-defined cross section in ANSYS to match the physical rod dimensions.

Outer dimensions (in):

- Wall-to-wall width: 0.58
- Arc-to-arc height: 1.18
- Wall thickness: 0.04

No defeaturing was required because the model was already simplified to a beam centerline.

Figure 1: Closet rod beam geometry and span length (45 in).

Figure 2: User-defined oval hollow cross section used for the beam model (dimensions in inches).

4. MATERIAL PROPERTIES

The rod was assigned Structural Steel in ANSYS and modeled as a linear elastic material, meaning stress is proportional to strain and the material returns to its original shape after unloading. This assumption is appropriate because the applied loading is moderate and the rod is not expected to approach yielding. The material properties used in the model were a Young's modulus of $E = 29{,}000{,}000$ psi and a Poisson's ratio: $\nu = 0.30$

5. MESH

The rod was meshed using quadratic beam elements, which are well-suited for capturing bending behavior because they include mid-nodes and allow for a quadratic displacement field along each element. An edge sizing control was applied to ensure a consistent and uniform element distribution along the 45-inch length. The mesh consisted of 30 beam elements and 91 nodes, corresponding to an average element length of 1.5 inches. This provided adequate resolution to capture deflection along the rod while maintaining low computational cost.

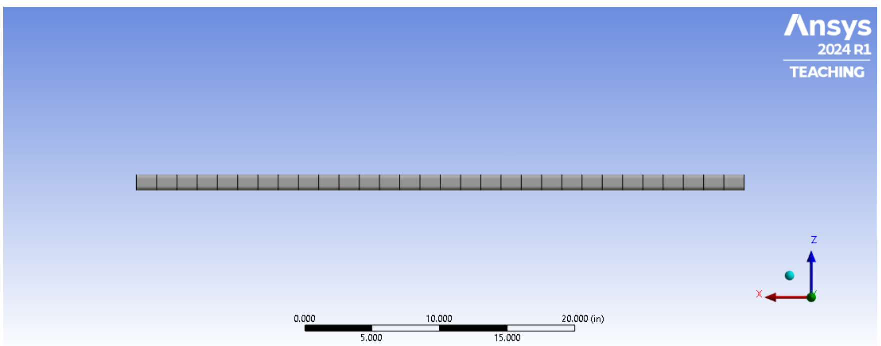

Figure 3: Beam element mesh with edge sizing of 1.5 inches along the 45-inch rod.

6. LOADS & CONSTRAINTS

The rod was modeled using remote displacement boundary conditions, with the global vertical direction defined as Z. At $X = 0$ in, all translational degrees of freedom were constrained ($U_X = 0$, $U_Y = 0$, $U_Z = 0$) and rotation about the X-axis was fixed ($\theta_X = 0$), while the remaining rotations were free. At $X = 45$ in, axial displacement was left free (U_X free), while transverse displacements were constrained ($U_Y = 0, U_Z = 0$), and all rotations were free. This support configuration prevents rigid body motion while allowing bending rotations at the supports; however, axial expansion is restrained at the left end due to the $U_X = 0$ constraint. This configuration was selected to ensure numerical stability by eliminating rigid body motion while still representing realistic pinned-type supports under bending loads.

Fourteen point loads were applied downward in the Z direction to represent hoodies, coats, and fishing waders of varying weight. The loads were applied at evenly spaced nodes along the rod (every 3 inches from $X = 3$ in through $X = 42$ in). The applied loads (lbf) were:

$$3,\ 5,\ 4,\ 6,\ 7,\ 4,\ 6,\ 5,\ 3,\ 6,\ 4,\ 5,\ 6,\ 4$$

The total applied load was:

$$\sum P = 68 \text{ lbf}$$

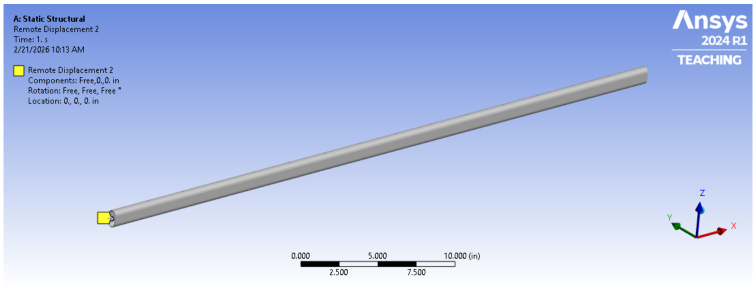

Figure 4: Remote displacement boundary condition at $X = 0$ in.

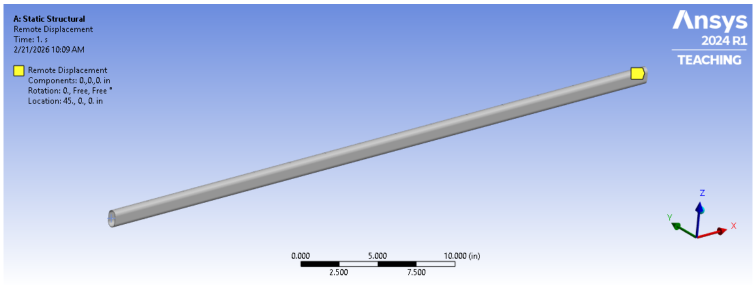

Figure 5: Remote displacement boundary condition at $X = 45$ in.

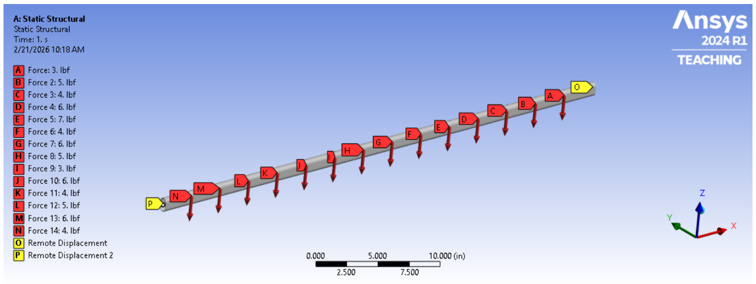

Figure 6: Applied loading configuration along the rod.

7. SOLUTION

The solution produced a maximum total deformation of:

$$\delta_{\text{max,FEA}} = 0.17164 \text{ in}$$

The deformation contour and beam shape shown by the graphics annimation matched what would be expected in real life for a simply supported rod loaded downward by hanging garments. The maximum deflection occurred near midspan, and the beam curvature was smooth with no unexpected kinks or rigid body motion.

Support reaction forces reported by ANSYS were:

$$R_{X=45} = 34.333 \text{ lbf}, \qquad R_{X=0} = 33.667 \text{ lbf}$$

These reactions sum to the total applied load, which confirms static equilibrium.

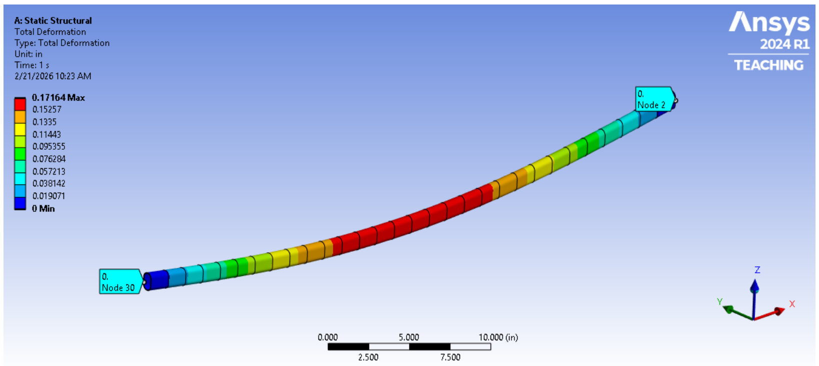

Figure 7: Total deformation (side view). Maximum deformation is 0.17164 in.

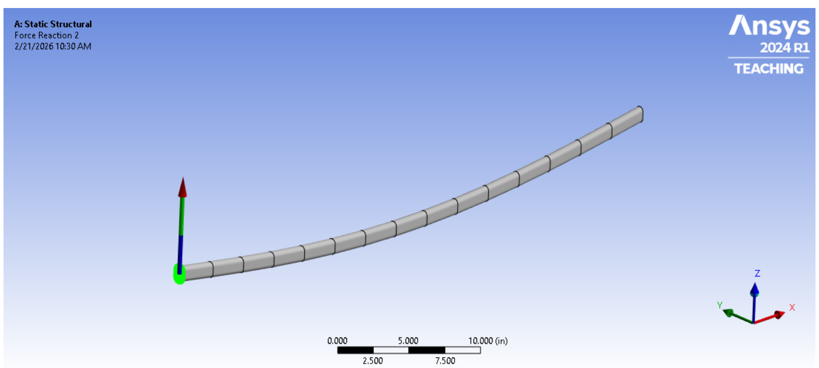

Figure 8: Reaction force at the $X = 0$ in support.

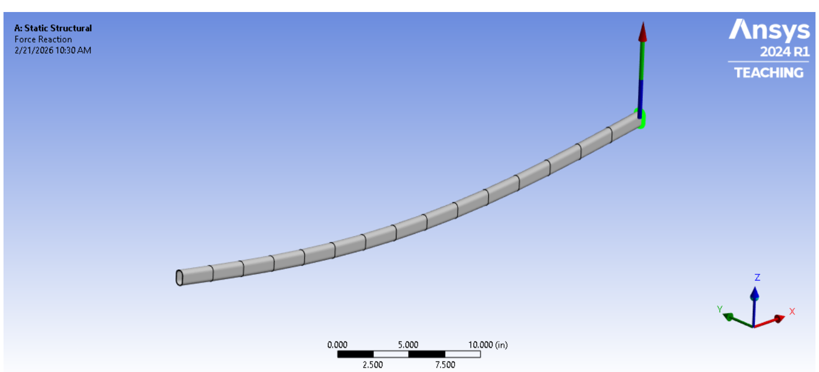

Figure 9: Reaction force at the $X = 45$ in support.

8. VERIFICATION OF RESULTS

Verification was performed using mesh refinement, statics, and classical beam theory.

1. Mesh Independence

Element Size (in)	Elements	Max Deflection (in)
3.0	15	0.17164
1.5	30	0.17164
0.5	90	0.17164

Table 1: Mesh refinement study.

Maximum deflection remained unchanged to five decimal places, confirming the solution is mesh independent.

2. Geometry, Material, and Loading

Beam length:

$$L = 45 \text{ in}$$

Material and section properties:

$$E = 29{,}000{,}000 \text{ psi}, \qquad I_y = 0.01783 \text{ in}^4$$

Point loads were applied at

$$x = 3, 6, 9, \ldots, 42 \text{ in}$$

Total applied load:

$$\sum P = 68 \text{ lbf}$$

3. Support Reactions (Statics)

Moment equilibrium about $x = 0$:

$$R_L = \frac{\sum(P_i x_i)}{L} = \frac{1545}{45} = 34.333 \text{ lbf}$$

Force equilibrium:

$$R_0 = 68 - 34.333 = 33.667 \text{ lbf}$$

These reactions match the FEA results.

Node	x (in)	P (lbf)	Px (lbf·in)
2	3	3	9
3	6	5	30
4	9	4	36
5	12	6	72
6	15	7	105
7	18	4	72
8	21	6	126
9	24	5	120
10	27	3	81
11	30	6	180
12	33	4	132
13	36	5	180
14	39	6	234
15	42	4	168
		Totals:	1545

Table 2: Applied loads and corresponding moments about $x = 0$.

4. Deflection by Superposition

Maximum deflection from FEA occurs near:

$$x^* = 22.4 \text{ in}$$

For a simply supported beam with a point load P at a:

For $x \leq a$:

$$\delta(x) = \frac{Pbx}{6LEI}\left(L^2 - b^2 - x^2\right)$$

For $x \geq a$:

$$\delta(x) = \frac{Pa(L-x)}{6LEI}\left(L^2 - a^2 - (L-x)^2\right)$$

Each load contribution was evaluated at x^* and summed:

$$\delta_{\text{max,hand}} = 0.17130 \text{ in}$$

5. Comparison with FEA

$$\delta_{\text{max,FEA}} = 0.17164 \text{ in}$$

Node	a (in)	P (lbf)	$\delta_i(x^*)$ (in)
2	3	3	0.002193
3	6	5	0.007180
4	9	4	0.008353
5	12	6	0.015972
6	15	7	0.021916
7	18	4	0.013874
8	21	6	0.021893
9	24	5	0.018234
10	27	3	0.010391
11	30	6	0.018749
12	33	4	0.010623
13	36	5	0.010414
14	39	6	0.008591
15	42	4	0.002916
		Sum:	0.17130

Table 3: Deflection contributions at $x^* = 22.4$ in.

$$\% \text{ Difference} = \frac{|0.17164 - 0.17130|}{0.17164} \times 100 = 0.20\%$$

The close agreement confirms that the supports, loads, and beam stiffness are modeled correctly.