DYNAPAC MOTION **Design, Manufacture, and Installation of Motion Products**

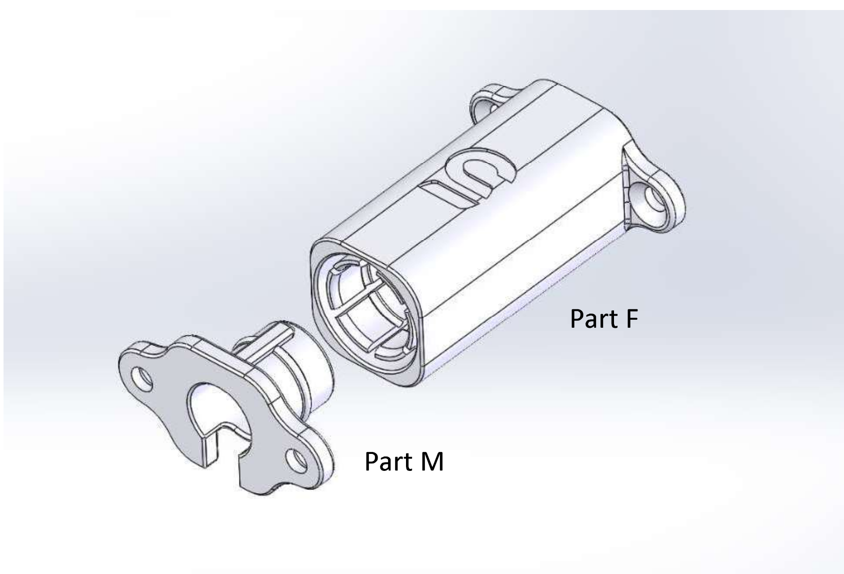
Analysis and Rating of the Five Inc. Snap Assembly

Prepared and Stamped by Scott Tew, PE

5/8/2025

Introduction

This report provides an overview of analysis performed on the Five Inc. Snap Assembly according to the designs provided to Dynapac. The snapping interface was analyzed to certify a maximum weight rating of the assembly in tension (with a factor of safety to account for jarring loads), then this weight was used to analyze the parts for static failure modes in shear and bending.



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Method of Analysis

Dynapac was provided with a Solidworks 3D model of the two part system shown in the photo above. Part M has an embossed ring that pushes the “teeth” of Part F open until it seats in a groove part way down the teeth. Part F is designed in two different lengths, but only the longer one was analyzed as it will be more prone to failure. The system is designed to be in a horizontal position during normal use but may need to bear a load under tension.

The analysis assumed that all parts are made from ABS Plastic, and that failure would occur if the plastic yields at 1.89 ksi (matweb.com).

The purpose of this analysis was to 1) determine a safe maximum load rating that will not cause the system to decouple under incidental tension and 2) assure that neither part will fail under this loading during normal use.

The first part of the analysis (page 1-2 of Appendix A) looked at a simplified version of a single tooth on part F. The combination of these teeth is what holds Part M from decoupling. This tooth was modeled as a simple cantilevered beam with a uniform rectangular cross section. The ‘decouple’ load was calculated using the known distance that the tooth would have to deflect to release part M. This load was multiplied by 6 to account for 6 teeth holding the

REGISTERED PROFESSIONAL ENGINEER SCOTT M. TEW M 42354 EXP. 3/31/2027 MECHANICAL STATE OF CALIFORNIA part. Then, a safety factor of 4 was applied to account for potential jolting and other unknown loading factors. This safety factor of 4 is a standard used at Dynapac for any load that is suddenly arrested from a falling condition.

The second part of the analysis looked at part M and F separately (see pages 3-4 in appendix A). Each part was modeled as a simple cantilevered beam with a uniform cross section, loaded at the end with the maximum rated load calculated in part 1. These beams were analyzed for Shear and Bending stresses at the cantilever point and these stresses were compared to the shear and axial yield stress of the material respectively.

Results

1) Maximum rated load in tension

Decouple Load	With n=4 factor of safety
14.2 lb	3.5 lb

2) Shear and Bending analyses

	Part M	Part F
Shear factor of safety	17.8	95
Bending factor of safety	7.4	22

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Conclusion

The certified weight rating for a single snap assembly determined by Dynapac is 3.5 lbs. All components of the assembly were determined to bear this weight without failure in tension, shear, and bending. Refer to appendix A for the calculations.

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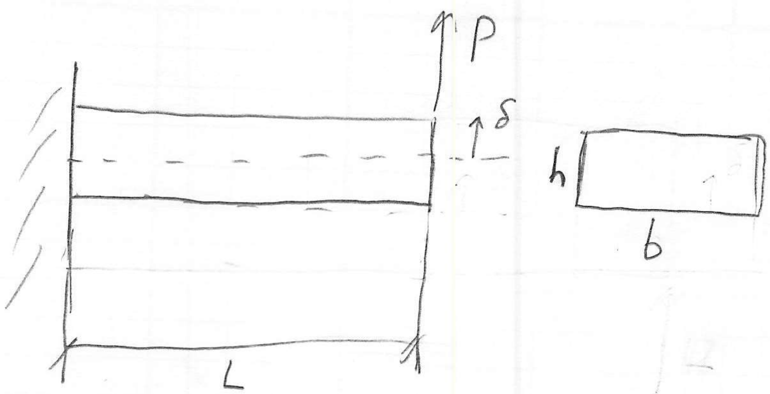
Pull-out resistance

Assumptions: Model a single tooth as a cantilevered beam with uniform rectangular cross section



Solve for P that causes δ to release retaining ring from notch.

$$\delta = \frac{PL^3}{3EI} \qquad (1)$$

From design geometry:

$$L = 0.204\ in$$

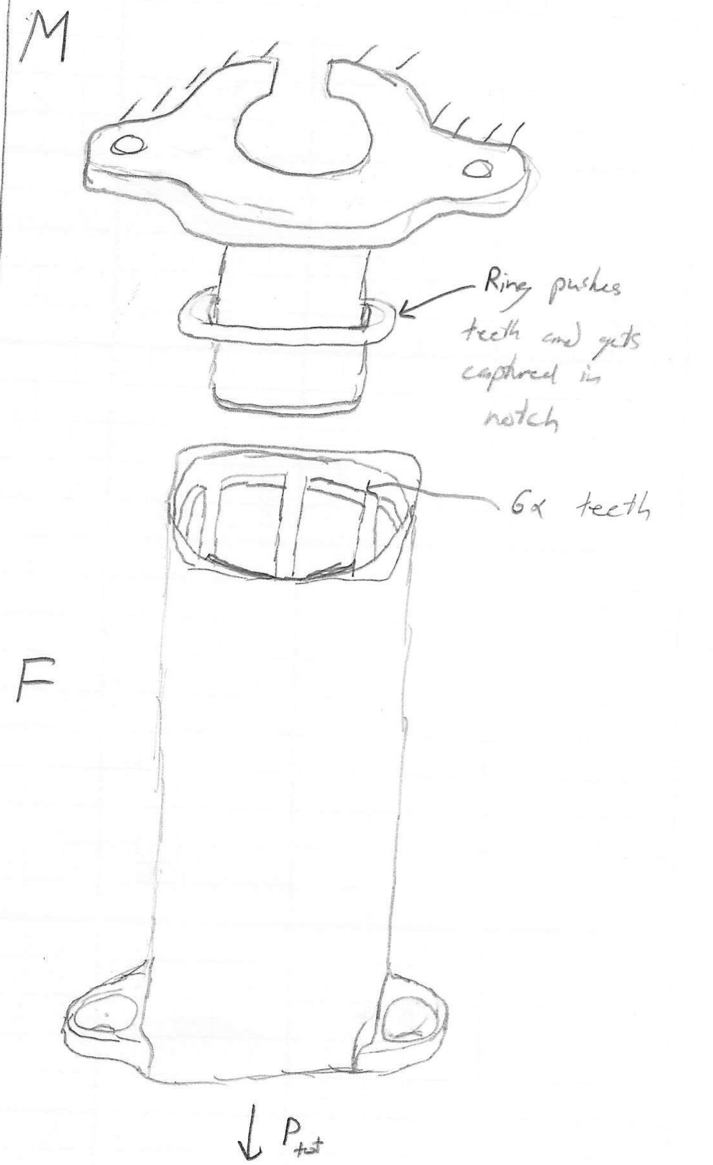
$$I = \frac{bh^3}{12} = \frac{(0.05)^3(0.2215)}{12}$$

$$= 2.31 \times 10^{-6}\ in^4$$

$$\delta_{crit} = 0.02\ in$$

$$P_{crit} = \frac{\delta 3EI}{L^3} = \frac{3(0.02\ in)(145{,}000\ psi)(2.31\times10^{-6}\ in^4)}{(0.204)^3}$$

$$= \boxed{2.37\ lbs}$$



Material: ABS

Yield Strength: 1.89 ksi
Elastic Modulus: 145 ksi
} Matweb.com

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P_{cut} was found for 1 teeth, so for 6 teeth

$$P_{tot} = 6 P_{cut} = 6(2.37 \text{ lbs}) = 14.20 \text{ lbs}$$

For a jolting load of 4x,

$$P_{rate} = \eta P_{tot} = \frac{1}{4}(14.20 \text{ lbs}) = \boxed{3.55 \text{ lbs}}$$

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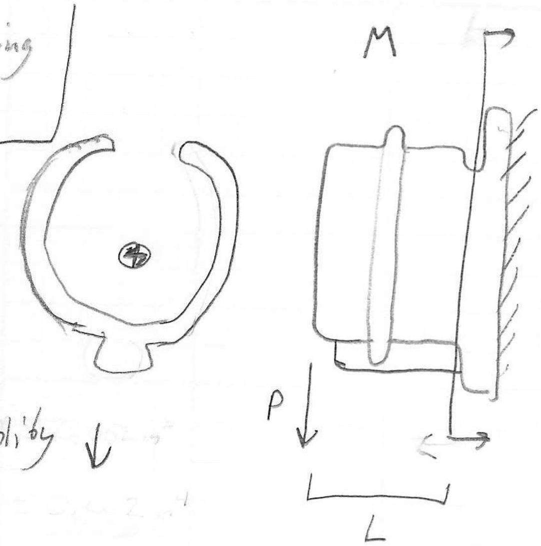
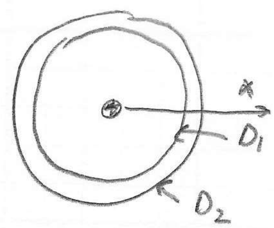
Test Part M for Shear + Bending at rated load



Simplify ↓



Shear:

$$V = \frac{P}{A} = \frac{3.55 \text{ lb}}{6.71\times10^{-2}\text{ in}^2} = 52.9 \text{ psi}$$

$$V_{crit} = 0.5\,\sigma_{crit} = 0.5 \times 1.89 \text{ ksi} = 945 \text{ psi}$$

$$\eta = \frac{V_{crit}}{V} = \frac{945}{52.9} = \boxed{17.8 \gg 1}$$

Part M will not fail in shear at the rated load.

Bending:

$$\sigma_B = \frac{My}{I} = \frac{PLD_2}{2I} = \frac{(3.55)(.565)(.545)}{2(2.13\times10^{-3})}$$

$$= 256 \text{ psi}$$

$$\sigma_{crit} = 1890 \text{ psi}$$

$$\eta = \frac{\sigma_{crit}}{\sigma_B} = \frac{1890}{256} = \boxed{7.4 > 1}$$

Part M will not fail in bending at the rated load.

$P = 3.55$ lbs

$L = 0.565$ in

$D_1 = 0.460$ in

$D_2 = 0.545$ in

$$A = \frac{\pi}{4}(D_2^2 - D_1^2) = 6.71\times10^{-2}\text{ in}^2$$

$$I = \frac{\pi}{64}(D_2^4 - D_1^4) = 2.13\times10^{-3}\text{ in}^4$$

$\sigma_{crit} = 1.89$ ksi = Yield Strength

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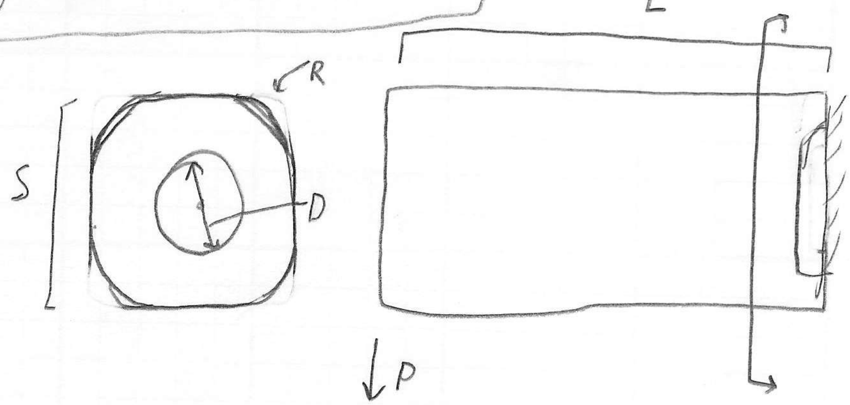
Test Part F for shear and bending at the rated load



Shear:

$$V = \frac{P}{A} = \frac{3.55 \text{ lbs}}{0.36 \text{ in}^2} = 9.86 \text{ psi}$$

$$V_{crit} = .5\sigma_{crit} = .5(1.89 \text{ ksi}) = 945 \text{ psi}$$

$$\eta = \frac{V_{crit}}{V} = \frac{945}{9.86} = 95 >> 1$$

Part F will not fail in shear under the rated load.

Bending:

$$\sigma_B = \frac{My}{I} = \frac{PLS}{2I} = \frac{(3.55)(1.875)(.85)}{2(3.39 \times 10^{-2})}$$

$$= 83 \text{ psi}$$

$$\eta = \frac{\sigma_{crit}}{\sigma_B} = \frac{1890}{83} = 22 >> 1$$

Part F will not fail in bending under the rated load.

$P = 3.55$ lbs

$L = 1.875$ in

$S = 0.85$ in

$R = 0.25$

$D = 0.46$ in

$$A = S^2 - \pi R^2 - \frac{\pi}{4} D^2 = (0.85 \text{ in})^2 - \pi(0.25)^2 - \frac{\pi}{4}(0.46)^2$$

$$= 0.360 \text{ in}^2$$

$$I = I_{square} - I_{circle}$$

$I_{square} = 3.61 \times 10^{-2} \text{ in}^4$ → Online Calculator

$$I_{circle} = \frac{\pi}{64} D^4 = 2.20 \times 10^{-3} \text{ in}^4$$

$$I = 3.61 \times 10^{-2} - 2.20 \times 10^{-3} = 0.0339 \text{ in}^4$$

σ_{crit} = Yield Strength = 1.89 ksi

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