Simulating and Testing the Damping of Targets in Accelerator Based Experiments

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Abstract

This research will focus on the effect of stress on a target when a high-energy proton beam causes the target to deform and vibrate. The target will be modelled and simulated in its most basic state, a cylindrical rod, in ANSYS to determine the damping ratio. From there, more complexities to make the target model as accurate as possible will be added as will certain modifications to other aspects in an effort to perform a sensitivity study. The results of this sensitivity study will inform us about what increases, decreases or has little to no effect on the damping of the target. We will also create a basic design to test it experimentally so we can benchmark the simulation results. The complexities and modifications will be added physically as well so the results can be compared and observed. At the moment, the project has not yet progressed into the later stages of running any sensitivity studies or assembling the concept design. This particular paper will focus on the initial steps as the simulation is setup and tested to make sure it is running properly.

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Introduction

All accelerator based experiments in use at Fermilab use some form of a target, therefore, making it necessary to understand the targets reactions and how it is effected as it is used. Targets are used specifically in these accelerator based experiments to convert high energy proton beams into new particles. A basic diagram of the target and its connections to the rest of the device is shown below in figure 1. The long gray extrusion is the actual target, while the extraneous parts are simply detailing how the target is connected to the horn and the rest of the device.

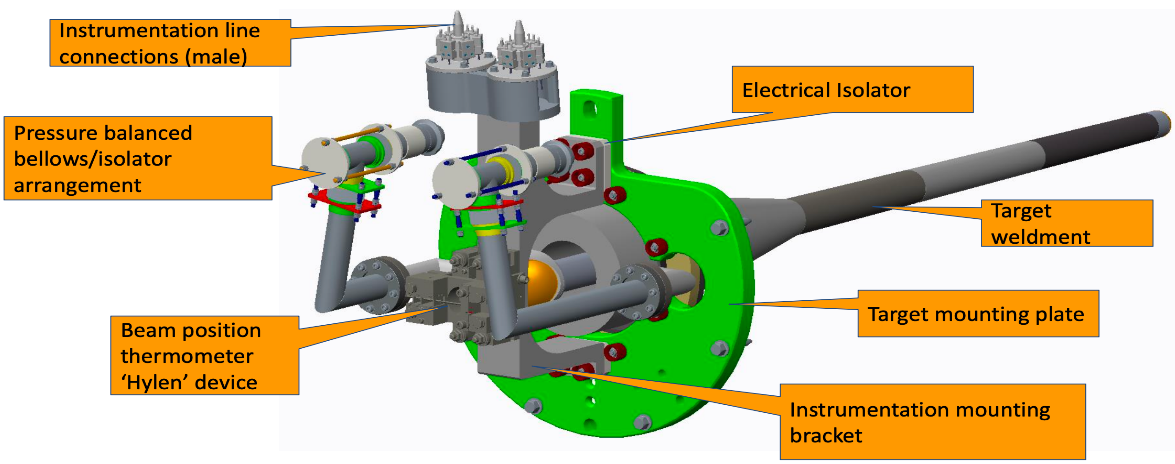

Figure 1. Target for LBNF/Dune Project

The process of converting high energy proton beams into new particles can subject the target to a great deal of stresses, high temperatures, and corrosion, all of which can affect the longevity of the part. Instead of the only stress coming from the pulse itself as it goes through the target, the pulse sets off aftershocks that run through the target. This results in vibrations and oscillations prolonging the target's motion. These aftershocks, therefore, apply further stresses to the target and can reduce its overall lifespan if they are not accounted for beforehand. For instance, if the aftershocks are not taken into account when the targets lifespan is determined, the lifespan may be overestimated, therefore, skewing the target's fatigue measurements, which can be a detrimental error when applying the targets in practice. Ideally, in actual application, the target will interact with the pulse symmetrically, so that as the pulse travels through the target, it simply expands instead of oscillates. Any asymmetries, such as up and down oscillations, that do occur are most likely due to some kind of offset in the target's loading.

These aftershock oscillations, however, can be reduced by increasing damping, the gradual dissipation of energy over a period of time. By increasing damping, the life span of the target can be increased by minimizing the amount of oscillations it is subjected to, therefore, reducing the stress applied to the target. There are various ways that damping can be modified. For instance, changing the material used can alter the damping, as can modifying the mounting design of the target, the target dimensions, adding material, or using a composite structure that will in turn increase the internal friction. Damping can also be modified by implementing a gap element instead of simply a solid rod, or applying internal fluids with a high viscosity to absorb some of the vibrations.

Research Objectives

In an effort to observe the damping of the target and to, hopefully, increase the damping to extend the life of the target, the goal of this project is to measure the damping ratio of the target. This is done by modelling a basic cylindrical rod to mimic the actual target and then proceeding to simulate the oscillations and vibrations it endures when it is deflected a certain amount and then released. By observing the release, we can measure the frequency of the movements that occur post deflection and then calculate the ratio. We would then run sensitivity studies by adding complexities gradually to the simulation and observing how they affect the damping. Following the sensitivity studies, the last step would be to design and assemble an experimental design to physically test the simulated setup. By performing these tests, we can benchmark the analytical results and ensure the accuracy. Overall, the tests will enable us to determine how to effectively increase damping.

Methods

By modelling the cantilever beam in its most basic state, a simple rod, we are able to observe the damping caused by internal friction of the target, and determine if it is enough to damp out the oscillations before the next beam pulse occurs. In order to model the target in its basic state, we used ANSYS. The basic model can be seen below in figure 2.

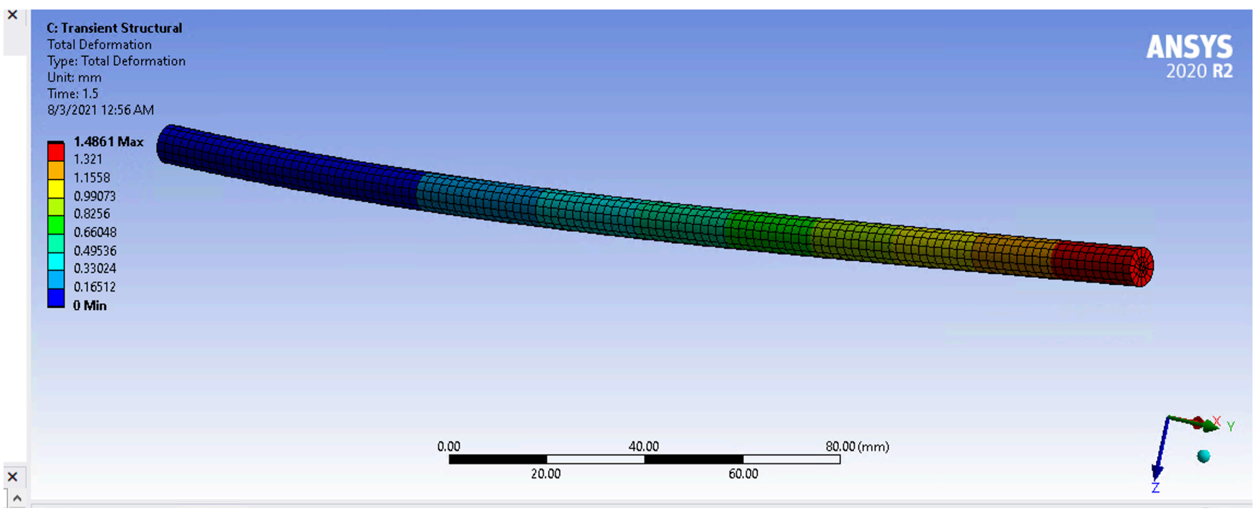

Figure 2. Basic ANSYS Cantilever Rod Design Used to Mimic the Target

It was modelled using titanium, specifically Ti-6Al-4V, and the overall dimensions were collectively scaled down for future ease of application. Reducing the rod dimensions by a factor of 6, we changed the length from 1.5 meters to 250 millimeters and the diameter from 0.48 meters to 8 millimeters. This was done primarily to reduce necessary steps when the experimental design starts being implemented. By decreasing the overall target size in ANSYS, the physical testing can be performed on a smaller scale using less materials, and it will also relate to the simulated results more accurately without any reason for conversion due to size variations.

Once the rod had been modeled accurately, we ran a basic modal analysis to determine the natural frequencies of the cantilever rod. We determined the natural frequencies both analytically and experimentally through the simulation. In order to calculate it analytically, we used the equation for the moment of inertia, as seen below, which utilizes diameter, d, measured in meters.

$$I = \frac{\pi d^4}{64}$$

After the moment of inertia was determined, that value was used to calculate the frequencies, ω, of the first three modes using the equation below. The c value, as seen below, is a constant coefficient that is modified from one mode to the next. The calculations can be seen in the Appendix. The equation also utilizes the modulus of elasticity, E, the previously calculated moment of inertia, I, density, ρ, area, A, and the rod's length, L.

$$\omega = c^2 \sqrt{\frac{EI}{\rho A L^4}}$$

By using the equations listed above, we were able to calculate and compare the analytical results to the experimental results from our modal analysis in ANSYS. Acting as a checkpoint, we want the analytical and experimental results to be fairly similar. After calculating the analytical results, we need to able to check those numbers against the experimental ones so they can tell us if something in the simulation or in the calculations are off. Once the natural frequencies have been determined and the simulation proved to be fairly accurate when compared to the hand-calculations, we can move on to the next step and run a static analysis.

Going on to the static analysis, the main modifications that need to be made to the ANSYS design from the modal analysis is the addition of a fixture at one end of the rod to mimic the clamping method of the target, and a force applied at the opposite end to deflect it. In order to run the static analysis, however, we need to apply a starting force or deflection to set off the oscillations. For our purposes, we simply made an educated guess and determined a reasonable deflection that the rod might experience during its use. Therefore, we decided to start with a deflection of 1/16 of an inch and observe the resultant oscillations. In order to apply that deflection in ANSYS, we needed to convert it to a force. This was done using Roark and Young's Equations, listed below, for determining the deflection of a cantilever beam fixed at one end and deflected on the other, as seen below in figure 3.

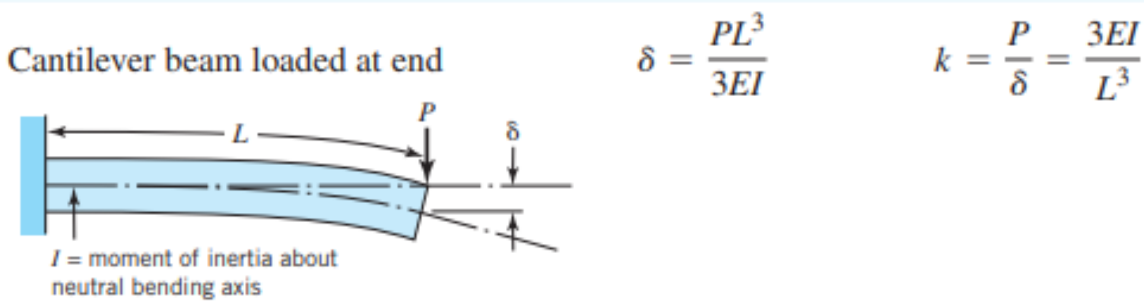

Figure 3. Roark and Young's Equation for Deflection

This equation utilizes the predetermined deflection, δ, the beam's length, L, the modulus of elasticity, E, and the rod's moment of inertia, I, all while solving for the force, F. By calculating this, we determined that a 7-newton force would be ideal to start with.

$$\delta = \frac{FL^3}{3EI}$$

By running a static analysis, we are able to apply the force and observe the initial deflection to ensure the simulation results will still hold constant to the analytical results (the pre-determined deflection). Once that has been verified, a transient analysis, used to determine the rods time dependent response so the damping ratio can be calculated, can be run and compared to the static analysis. This is a similar situation to the previously compared analytical modal analysis results and the experimental results. We need the initial simulated deflection, which was garnered from the static analysis, to be fairly similar to the initial deflection determined by the transient analysis to ensure, once again, the accuracy and consistency from one analysis to the other.

The next steps would be to observe the resultant graphs and to calculate the damping ratio from that. This will provide the basis damping ratio that our latter simulations will be compared to once complexities have been added.

Results

Calculating the natural frequencies for modes 1 through 3, the following values, displayed in figure 4, were calculated and compared to the values determined using the modal analysis in ANSYS.

	Frequency (Hz)	
Mode	**Analytical**	**Experimental**
1	90.83	88.311
2	569.25	551.51
3	1594.08	1535.7

Figure 4. Natural Frequency Comparison

Figure 4 shows that the simulated results are fairly similar to the calculated results. The largest margin of error is in mode 3, with an error of 3.6%. Due to the low percent error value, we were able to move on and run the static analysis.

Comparatively, the actual natural frequencies when the cylindrical rod is not scaled down are shown below in figure 5.

	Frequency (Hz)	
Mode	**Analytical**	**Experimental**
1	544.98	529.866
2	3415.5	3309.06
3	9564.48	9214.2

Figure 5. Natural Frequency Comparison (Actual Measurements)

The predetermined deflection was 1/16 of an inch, or 1.5875 mm, and the static analysis initial deflection was 1.693mm, resulting in an error of 6.645%. After running the transient analysis, it was observed that the static analysis' and transient analysis' initial deflections were constant. Therefore, continuing on, we applied the force to the end of the rod and set the time steps so the force would be applied gradually, then released instantaneously, therefore, triggering the vibrations. As seen below in figure 6 and 7, the force was applied gradually to the rod at 0.5 seconds. The force was then removed after another 0.5 seconds before being released at 1 second. Once it was released, the oscillations were recorded for the next 0.5 seconds.

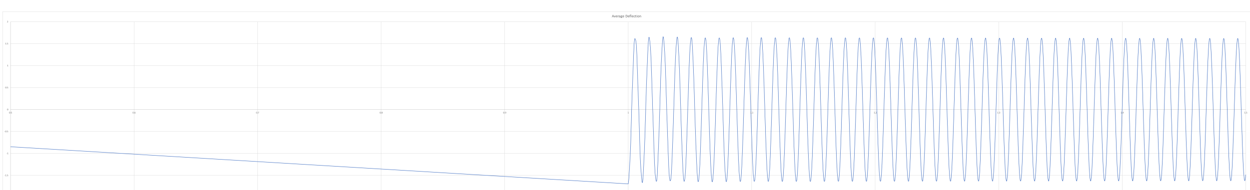

Figure 6. Deflection of Target without Material Damping

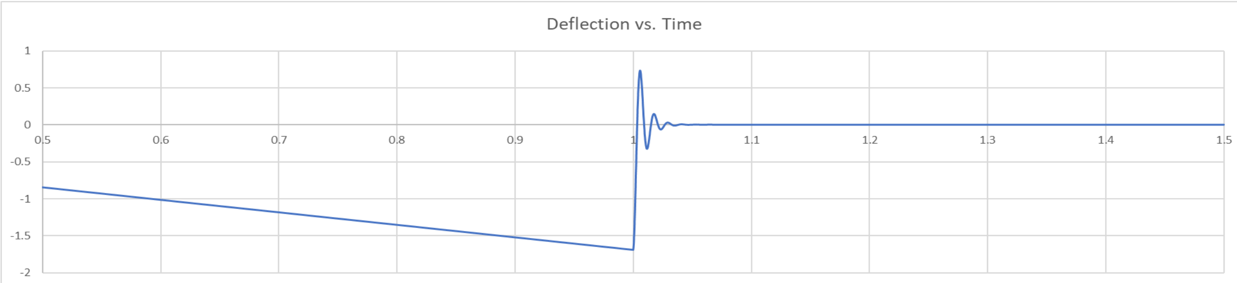

Figure 7. Deflection of Target with Damping

There were a few delays as we tried to determine how to accurately model the application of pressure, the clean release and the oscillations without any interference.

As of right now, no damping is visible. We believe this is currently due to the fact that we have to apply the damping coefficient value to the material manually, so it can account for that as well, instead of acting as if no damping is produced by the material itself. It is for this reason that in figure 6, following the rods release, the amplitude of the wavelengths remain fairly constant. Continuing work on this project, the next steps would be to apply the material damping coefficient, so the effects can be observed more accurately. In figure 7, a damping coefficient of 0.001 was applied, enabling us to capture a vague example of how it would affect the damping of the rod. A more accurate coefficient for titanium, specifically, still needs to be applied, but this illustrates a general idea of what it will most likely look like once the material's damping is being taken into account during the simulation.

Concept Design

The second part of the project was to design an experimental set-up for future benchmarking purposes. As is shown below in figure 8, a basic concept has been designed.

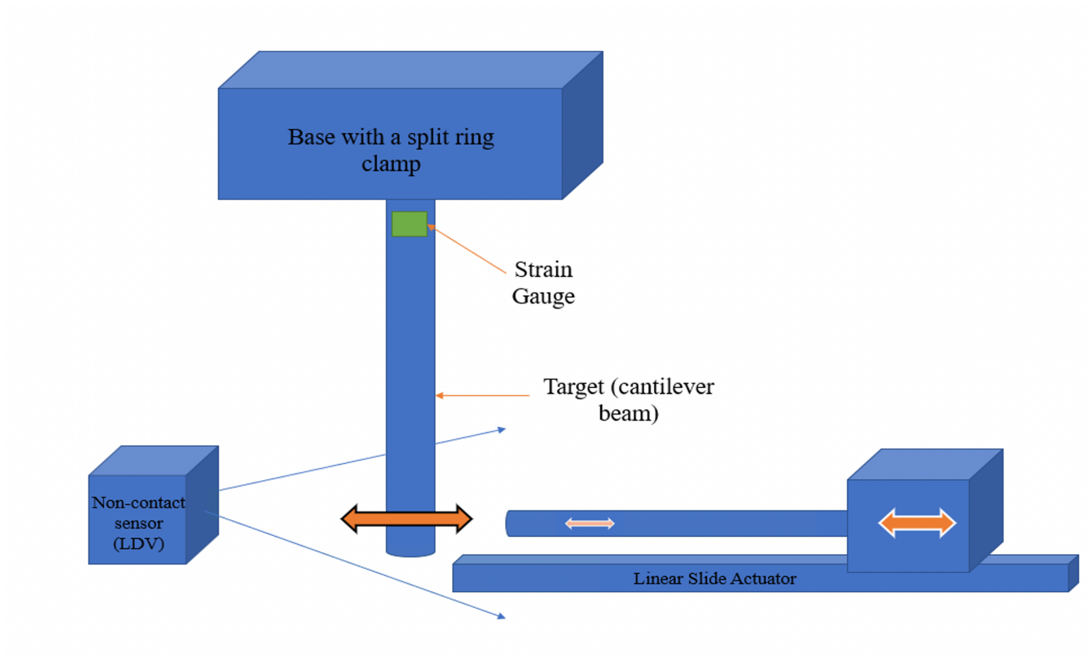

Figure 8. Basic Concept Design of Physical Experiment

It will utilize a base suspended with a split ring clamp. The cantilever beam will be able to slide into the clamp while also allowing for the clamp size to be modified as modifications are made to the beam. As was mentioned before, one of the alterations that may be incorporated is the alteration of the rod's dimensions to observe how it affects damping. Therefore, a clamping method that can be modified is necessary. The base and clamp will hold the rod in a vertical position so the testing can be performed without the effects of gravity. This way, we can separate the effects caused by the deflection and the effects that may be a product of both the deflection and gravity deforming it.

One of the factors that could result in possible asymmetries is gravity. This is a major concern when testing in the horizontal direction instead of the vertical. For instance, if the rod was tested in the horizontal direction with the deflection applied so the rod will oscillate side to side, due to gravity, the rod will start with a slight offset. Due to its horizontal orientation, the rod is already being pulled downwards. When the side to side to side deflection is applied, this initial offset can cause the oscillations to move along more than one axes, and will result in a circular motion instead of motion along a singular plane. Therefore, we decided to orient it in the vertical direction.

The deflection will be applied to the end of the rod using a linear slide actuator, set so it will move the same amount each time it is utilized, therefore, resulting in a constant deflection from one trial to another. In order to observe and measure the results, two different sensors will be used. One will use a contact-centric method as a way to measure the deflection, while the other will use a contactless method. At the top of the rod, closest to the base and the split ring clamp, a strain gauge will be placed. This is the method that will utilize contact to observe the results. Therefore, the gauge needs to be placed where the strain will be greatest, and, according to the equation for moment ($M = Fd$), the greatest strain will be applied at the point furthest away from the applied force. In this case, that would be closest to the base. The contactless method will be measured using a sensor such as a laser doppler vibrometer, which will be placed in the direct line of motion of the rod to record the most accurate results.

Future Steps

Continuing on, the next steps would be to modify the simulation so that it will account for the damping from the material, because, as of right now, it is not taking that into account and the graphs are being skewed as a result. Using the modified simulation, the actual amplitudes can be determined from one period to another and the damping ratio can be calculated.

The concept design will also need to be physically constructed so experimental testing can begin. It will be used to run the experiment and to benchmark the simulation results. From there, complexities will be added to the simulation cantilever beam model to create a more accurate beam design and to add varying aspects that could modify the current damping of the target. The basic steps detailed in the previous sections will be repeated in an effort to run a sensitivity study. Each time a complexity is added, we can observe which complexities increase, decrease or have no effect on the damping of the target. Therefore, by performing sensitivity studies, we will be able to determine the best way to increase the damping in the targets and how to extend their lifespans.

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Appendix

I. Material Values (Ti-6Al-4V)

Density (ρ) = 4430 kg/m^3

Modulus of Elasticity (E) = 1.14*10^{11}N/m^2

II. Moment of Inertia Calculation

$$I = \frac{\pi d^4}{64}$$

$$I = \frac{\pi(.008)^4}{64} = 2.0106 * 10^{-10} m^4$$

III. Area of the Cylindrical Rod

$$A = \pi r^2$$

$$A = \pi(.004)^2 = 5.02654 * 10^{-5} m^2$$

IV. Natural Frequency Calculations

$$\omega = c^2 \sqrt{\frac{EI}{\rho A L^4}}$$

Mode 1

$$\omega_1 = (1.875)^2 \sqrt{\frac{EI}{\rho A L^4}}$$

$$\omega_1 = (1.875)^2 \sqrt{\frac{(1.14*10^{11})(2.0106*10^{-10})}{(4430)(5.02654*10^{-5})(0.25)^4}} = 570.6914381 \frac{rad}{sec}$$

$$\omega_1 = 570.6914381 \frac{rad}{sec} = 90.83Hz$$

Mode 2

$$\omega_2 = (4.694)^2 \sqrt{\frac{EI}{\rho A L^4}}$$

$$\omega_2 = (4.694)^2 \sqrt{\frac{(1.14*10^{11})(2.0106*10^{-10})}{(4430)(5.02654*10^{-5})(0.25)^4}} = 3576.720331 \frac{rad}{sec}$$

$$\omega_2 = 3576.720331 \frac{rad}{sec} = 569.25Hz$$

Mode 3

$$\omega_3 = (7.855)^2 \sqrt{\frac{EI}{\rho A L^4}}$$

$$\omega_3 = (7.855)^2 \sqrt{\frac{(1.14*10^{11})(2.0106*10^{-10})}{(4430)(5.02654*10^{-5})(0.25)^4}} = 10015.92795 \frac{rad}{sec}$$

$$\omega_3 = 10015.92795 \frac{rad}{sec} = 1594.08Hz$$

	Frequency (Hz)	
Mode	**Analytical**	**Experimental**
1	$90.83 * 6 = \mathbf{544.98}$	$88.311 * 6 = \mathbf{529.866}$
2	$569.25 * 6 = \mathbf{3415.5}$	$551.51 * 6 = \mathbf{3309.06}$
3	$1594.08 * 6 = \mathbf{9564.48}$	$1535.7 * 6 = \mathbf{9214.2}$

Natural Frequency Comparison Calculations (Actual Measurements)

V. Percent Error Calculation for Frequency

$$Percent\ Error = \left|\frac{Experimental - Theoretical}{Theoretcial}\right| * 100$$

$$Percent\ Error\ for\ mode\ 1(\omega_1) = \left|\frac{88.311 - 90.83}{90.83}\right| * 100 = 2.77\%$$

$$Percent\ Error\ for\ mode\ 2(\omega_2) = \left|\frac{551.51 - 569.25}{569.25}\right| * 100 = 3.12\%$$

$$Percent\ Error\ for\ mode\ 3(\omega_3) = \left|\frac{1535.7-1594.08}{1594.08}\right| * 100 = 3.66\%$$

VI. Percent Error for Deflection (Static Analysis vs. Analytical)

$$Percent\ Error\ for\ Deflection = \left|\frac{1.693-1.5875}{1.5875}\right| * 100 = 6.645\%$$

VII. Force Calculations

$$\delta = \frac{FL^3}{3EI}$$

$$F = \frac{3\delta EI}{L^3}$$

$$F = \frac{3(1.5875)(1.14*10^{11})(2.0106*10^{-10})}{(0.25)^3} = 6.986272$$

$$F \approx 7N$$