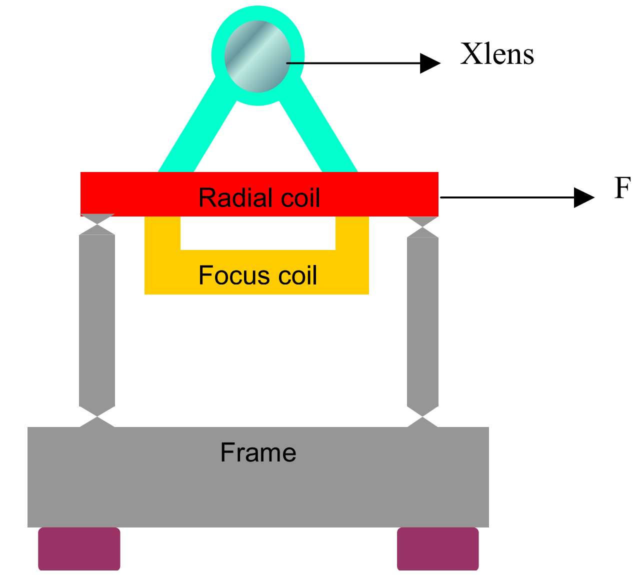
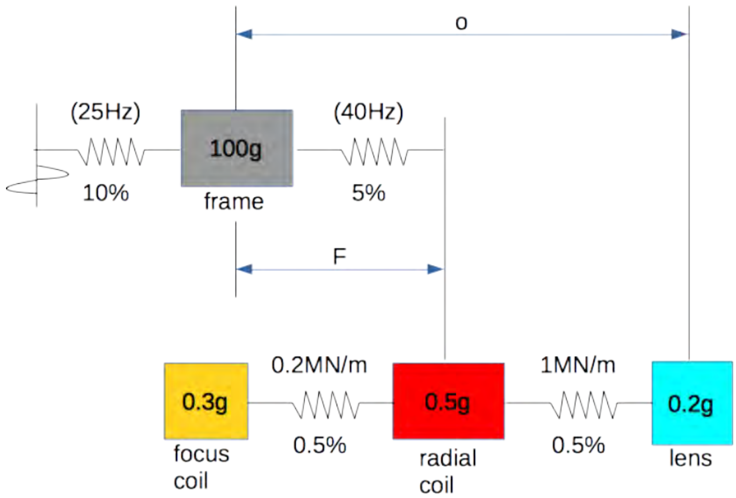
Assignment: CD-head internal dynamics

1 The effect of internal dynamics: PART 1

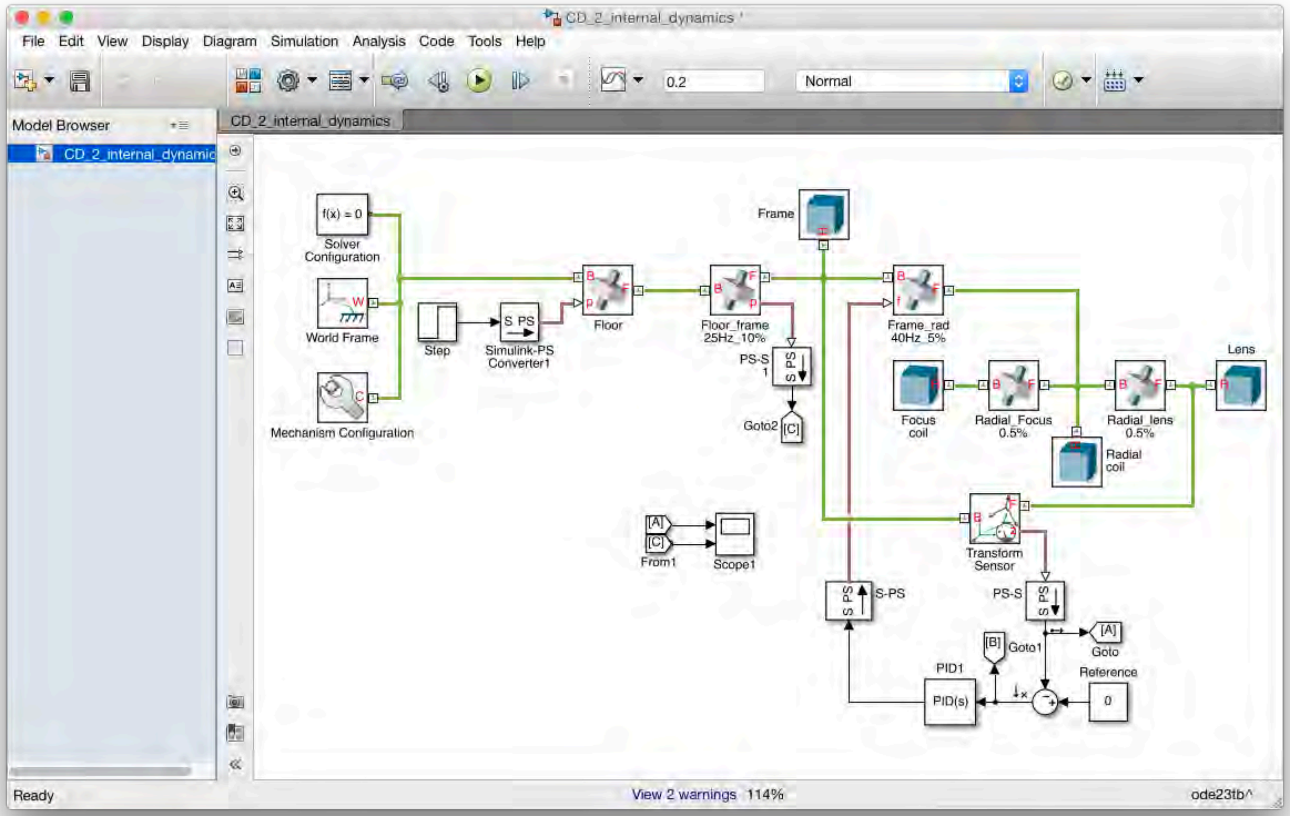
We have now seen aspects of control relating to the accuracy of the CD actuator. The controller designed allows us to achieve the required position accuracy when disturbances are present.
The dynamic model of the mechanical structure is, however, extremely simple. In reality the internal dynamics of the actuator play an important role. To make a simple model for the radial direction we can look at the top-view of the actuator.



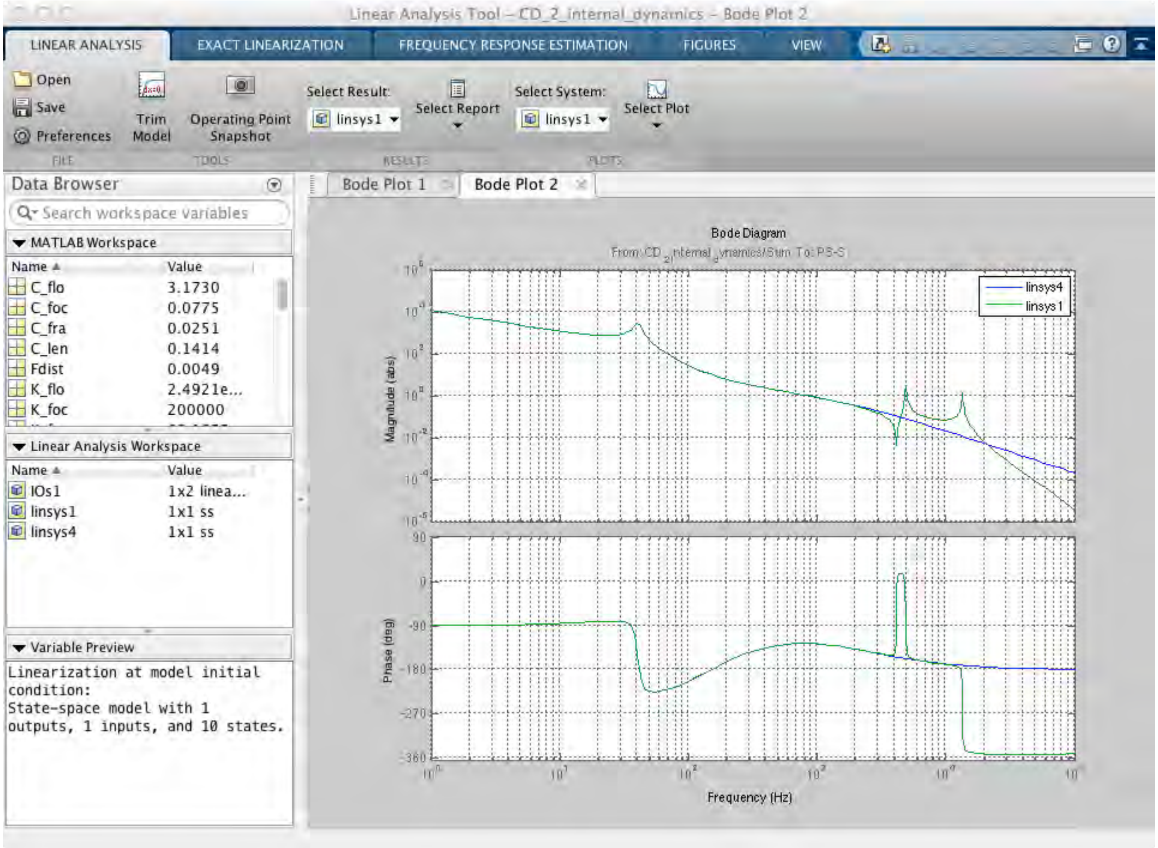
The hinges of the suspension support the total mass at a frequency of about 40 Hz. The radial force is applied at the radial coil. The radial displacement of the lens is measured.
The design has a certain resemblance to a set of book-shelves. The lens, the radial coil and the focus coil can be seen as rigid in radial direction and are connected by two upright stands.
For the radial dynamics a first approximation with 3 masses and connecting springs can be used.
The mass, stiffness and damping of the different connections and the sensor (o) and actuator (F) are indicated in the figure. Assume an actuator BW of 5 kHz, and a controller delay of 1μs.



1. Starting with the model developed for assignment 1, CD-tracking, develop a model for the CD-head including these internal dynamics. Set the PID controller to a 800 Hz BW using the values obtained using the PID rules of thumb. The actuator BW and controller delay are not considered in this model. The resulting model should resemble this figure.

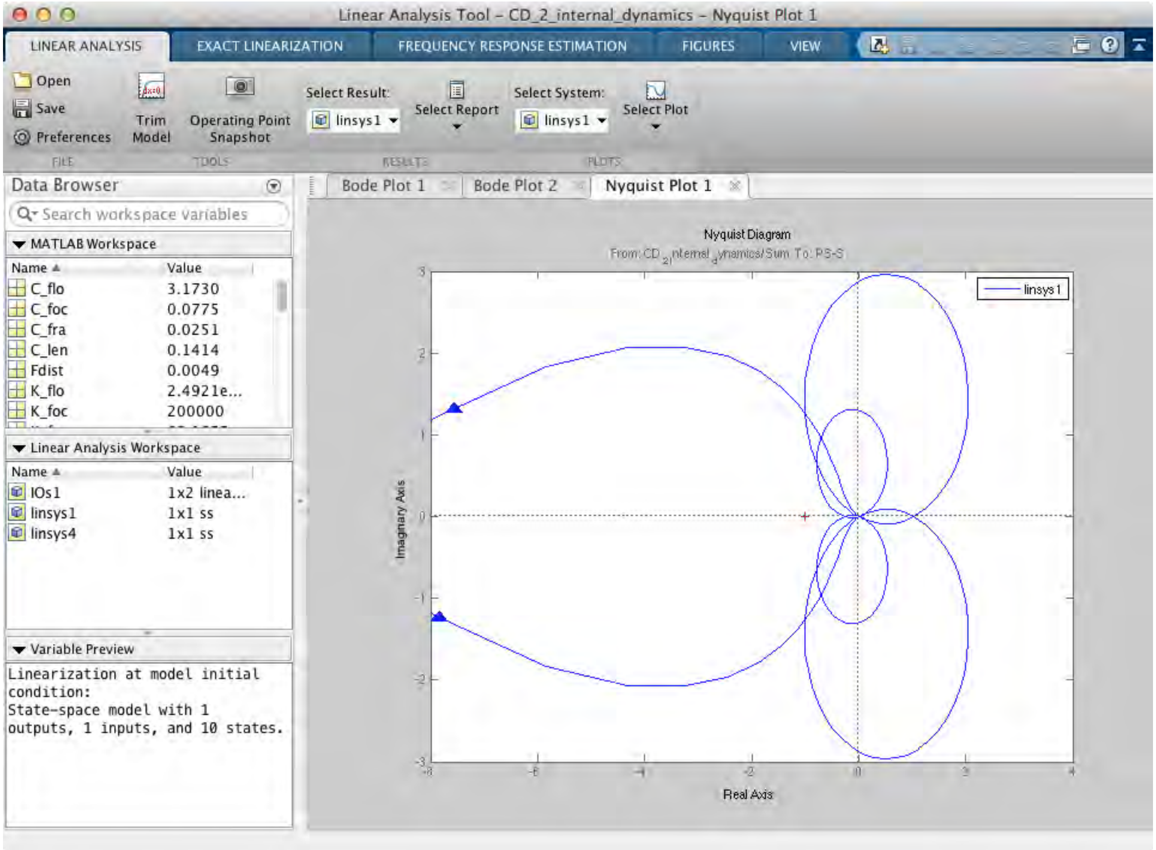


2. Set the connecting stiffness between focus coil, radial coil and lens to very high values and check that the response from force F to lens-frame displacement x is similar to the one calculated in assignment 1. (Notable difference is the 40Hz resonance of the frame lens-assembly connection.)
3. Now set the stiffness back to the original values as shown in the figure. Separating the lens-assembly into different masses will have considerable effects on the control of the actuator. Check that the above model leads to the open loop transfer shown below.

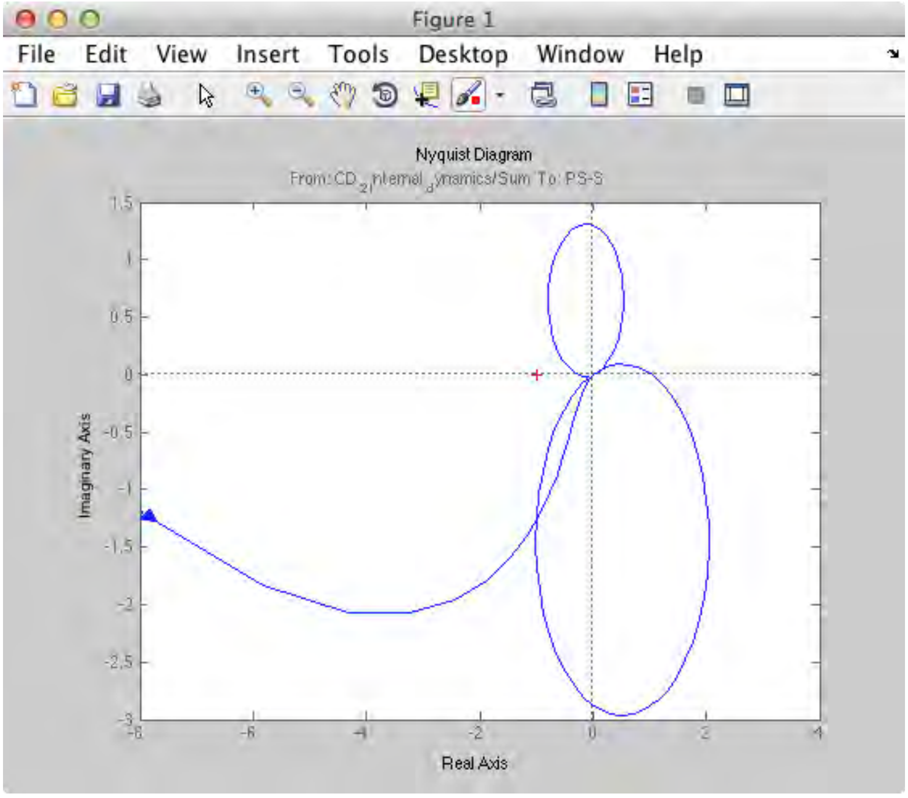


At 4 kHz and 5 kHz we find an anti-resonance and resonance peak. There is a positive phase shift associated with this combination. The system is stable.

4. Check the open loop transfer function stability using Nyquist.



5. From this picture it is clear that the ‘feature’ of Matlab to add negative frequencies to the Nyquist plot is not very helpful. If you want to plot Nyquist without negative frequencies, move the linear system variable to the Matlab workspace and plot the Nyquist plot from there:
   >> h = nyquistplot(linsys1);
   >> setoptions(h, ‘ShowFullContour’, ’off’);



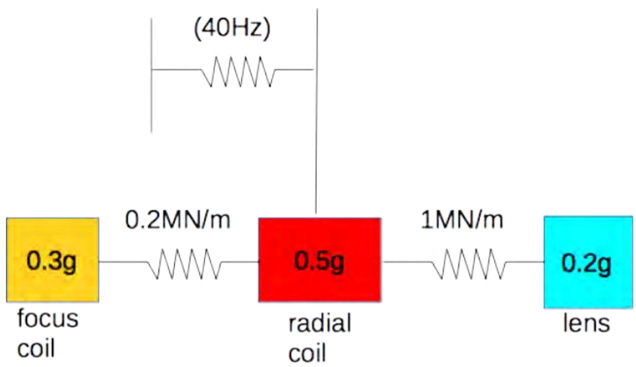
6. Vary the lens-radial coil stiffness from 1000 kN/m, to 150 kN/m and 100 kN/m and note the effect on the dynamic behavior in Bode-plot and Nyquist. By carefully selecting the value of this stiffness a system with a particularly good dynamic behavior can be obtained. Find the most suitable value for the lens-body stiffness by testing different values on that range and observing the Bode and Nyquist plots. Try to obtain the best mechanical system. With the current controller this system is unfortunately no longer stable and we will address that issue in the next question.
7. From the Nyquist diagram it is clear that the modified system is not stable. So now we have an improved mechanical design that unfortunately in combination with the chosen PID-controller results in an unstable system. However it is possible improve the controller: From the Nyquist diagram it is clear that the system will be stable again if we are able to 'rotate the phase'.
   Add a low-pass 1^{st}-order filter to the controller with an optimal BW-frequency, and observe the effect in the Nyquist-diagram and Bode-plot. The system is stable again! The optimal frequency BW-frequency for this low-pass filter is close to 10 f_{BW}.

2 The effect of internal dynamics: PART 2

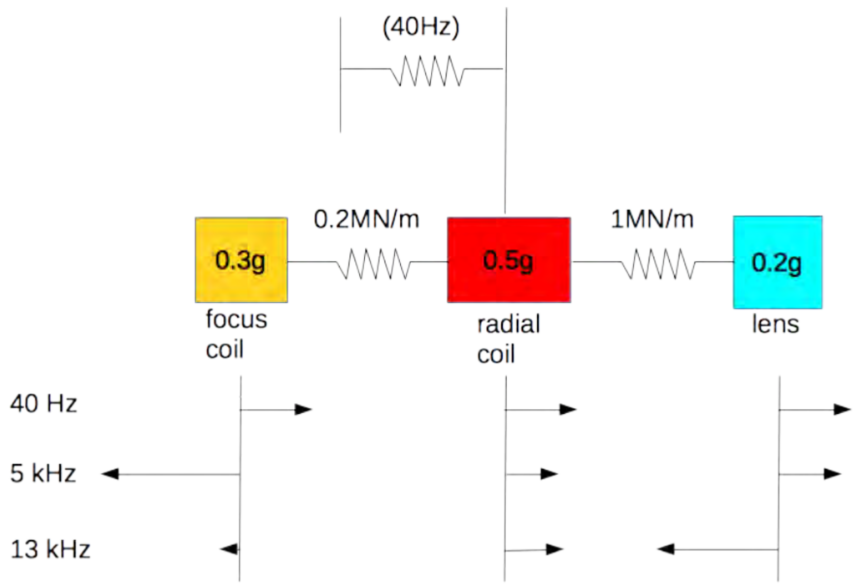
In question 3 we observed a dynamic behavior around 4-5 kHz and above that might compromise performance or controllability. In this case it is theoretically even possible to increase the gain to get a unity-gain crossover frequency above the first resonance frequency. However, the second resonance is in reality just a sign for many others that follow with their accompanying negative phase, which makes it wise to stay away from this frequency range.

In the creation of high-accuracy positioning devices one will attempt to increase controller gains as much as possible. Thus in most cases one will have to deal with the dynamics of the mechanical structure. We have learned that it is advantageous to obtain a better understanding of these dynamics. This improved understanding builds upon a description of the system dynamics using “modal decomposition”.

To use this approach we will consider the un-damped mechanical model. Because the frame is relatively heavy this will be approximated by a fixed constraint for the actuator modes. For the resulting model we can calculate the three resonance frequencies from the equations of motion. Associated with these frequencies we can calculate the shape found in the system at that resonance frequency.



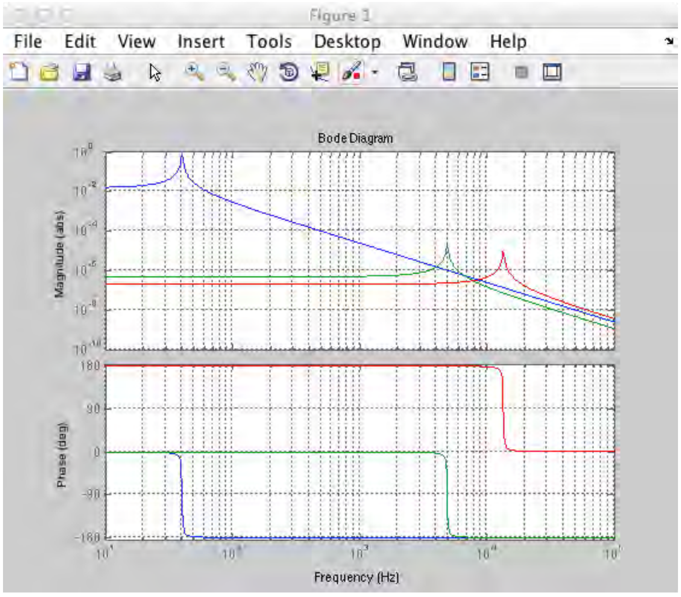
8. Study the matlab file ‘CD_head_modal’ and use it to calculate the eigenfrequencies and corresponding mode shapes for the CD-head assembly. Verify that in a graphical presentation the mode shapes look like:

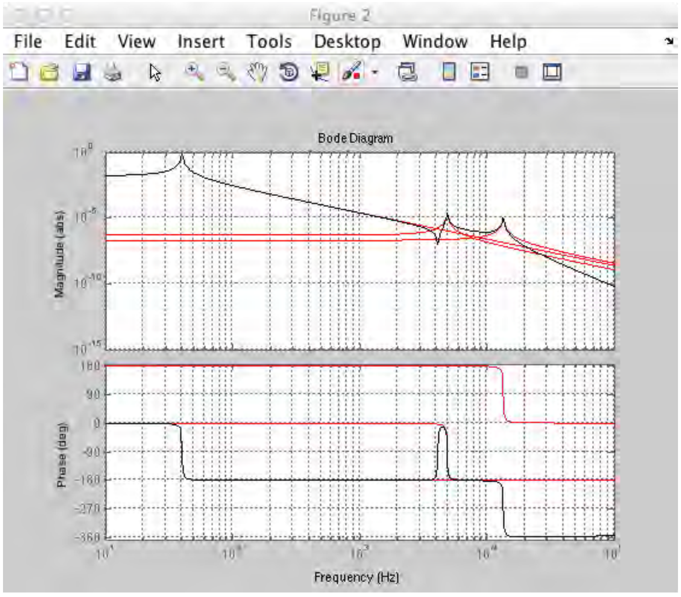


The first resonance occurs at 40 Hz and describes the desired nominal motion of the actuator in the radial direction.
The second shape indicates that the focus coil moves in counter phase with the lens and the radial coil. In the third shape the lens moves in counter phase with the radial coil.

In mathematical terms the three mode shapes can be used together with three modal coordinates to describe the system behavior. Basically this means that the system behavior can be described with a linear combination of the movements in the three modal-coordinates.

To obtain the total transfer function we need to sum the three modal contributions. Each modal contribution consists of a one-mass-spring system. In the following figure the three separate Bode diagrams are shown together with the total response. Note the difference in the phase-curve for the third mode, due to the opposite motion in the mode shape of the actuation, on the radial coil, and the sensor on the lens.





9. Study the simulink file ‘CD_head_modal_ws’.
   In this file the basic concept of modal decomposition, or composition in this case, is expressed. The displacement *x* at a position *i*, x_i, as a function of a force *F* at position *k*, F_k, is equal to the sum of the modal contributions with modal mass, stiffness, damping and shape resp. M_i, K_i, C_i, and $\boldsymbol{\Phi}_i$:
   $$\frac{x_l}{F_k} = \sum_{i=1}^{n} \frac{\Phi_{il}\Phi_{ik}}{M_i s^2 + C_i s + K_i}$$
   Check the simulink model and verify that this relation is expressed in the model, and plot the transfer function from a force on the radial coil to the displacement of the lens.
10. From the insights obtained using this modal analysis a direct method to find the optimal stiffness between lens and radial coil can be derived:
    For a certain stiffness the mode shape for mode 2 will show a node at the radial coil. This means that this mode will not be excited by vibrations entering via the frame. The mode will be unactuated.
    To find the optimal stiffness, note that in case the focus coil mass-spring-damper system has the same eigenfrequency as the lens mass-spring-damper system the radial coil remains stationary between these two (dynamic balancing). Calculate this optimal stiffness and check in the matlab workspace that your calculation is correct.
11. Enter this stiffness in the SimMechanics simulation and confirm your results.