Development of a Resonant Panel Speaker

A Major Qualifying Project
Submitted to the Faculty of
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Submitted by:
Dave Zielinski
Obi Obiora
Drew Sansoucy

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Advisor: Joe Stabile
Abstract:

In a landscape that is producing sleeker, and thinner products, the MQP is looking to apply this idea to the home entertainment system. The goal of this project was to produce a resonant panel that had a first modal frequency of 40 Hz. Because the volume velocity of the air needs to be much greater for such a low frequency, the panel has to cover a much larger area than a traditional speaker. The resonant panel is driven by a moving magnet transducer on its surface. The mass of the magnet and motor stator moving helps to displace the panel more, causing there to be more surface acceleration, and thus a larger volume velocity. Surrounding the panel are other, smaller moving magnet and moving coil speakers, which cover the mid and high frequency ranges. These designs were all analyzed and modeled with Finite Element Analysis techniques and 3D Computer Aided Design methods.
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- Austin McCalmont for allowing us to use his homemade transducer for testing
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Introduction:

This report covers the development of a walled mounted, low-profile speaker system. The speaker system is in the form of a frame that holds together several complementary speaker technologies that cover the whole range of frequencies and are required to maintain a very thin design. Over the years televisions have evolved from thick, bulky appliances to sleek, ultra-thin devices. However, the accompanying audio equipment has, for the most part, not followed this trend. While a small number of speaker manufacturers have created dimensionally impressive models, they are almost all very expensive or have limited frequency ranges, leaving a gap in the sector for more affordable speakers that can achieve an ideal range. The goal of our design was to fill this void and explore technologies not yet widely implemented for this use, creating a unit approximately an inch thick.

Background:

The customer would likely be anyone in the middle to high-end market who would like to be able to have a versatile audio system, capable of playing all types of music, without it taking up an excessive amount of space. Existing subwoofer designs almost all rely on large amounts of volume to produce low frequencies. This volume can be spread out on a “sheet” to make it able be large but thin, a format that could be hung on a wall. It could potentially be used in almost any environment, but most commonly would be found in a living room or elsewhere in a house. Of the constraints placed on the design, thickness was the most imperative. Additional specifications laid out before designing began included the implementation of sound steering, a format that could be hung from a wall in a way similar to a painting, and the capability to hit low notes. Prior
to deciding which technologies to consider to achieve these goals, market research was conducted. This market research yielded a small number of existing products that met our thickness requirement, but not our price target, frequency range, or sound steering ability. The two most comparable products are the Loewe Sound Stand SL and the Martin Logan Motion SLM-XL, which are capable of frequencies down to 150Hz and 100Hz, respectively.

Because the final product was intended to be wall-mounted, additional research was conducted in-order to find technologies that encompassed this idea. Products such as the SoundWall Art Speaker led to the investigation of the distributed mode loudspeaker or DML technology. This technology, developed by now dissolved company NXT, involved placing surface exciters in positions along a panel that correspond with that panel's resonant frequency. The resulting excitation would result in a coupled vibrations of the panel which would then produce sound at that frequency. [Everard, 2008]
Alternate & Final Design:

While we initially explored research and existing designs that take advantage of high-tech solutions like electrostatic films, our design process began with testing a transducer, or exciter, being fixed to a window via a suction cup. This effectively turned the window into a resonant panel by vibrating it at specific frequencies. This technology is reproducible on almost any large glass surface. From there, it was decided that this would be the idea the team would pursue. For the proceeding iterations, it would be necessary to create a system that included a light and portable frame that would emulate the boundary conditions of the window we had used for testing Austin McCalamont’s surface transducer. It was also necessary to include designs to mount the frame to a wall so that it could be as low profile as possible.

Prototype One

The first prototype was composed of sheet of 0.25in acrylic screwed onto the front of a thin pine frame. The heads of the screws were separated from the acrylic panel with rubber washers, while the acrylic was separated from the wooden frame using a strip of caulking cord to isolate vibrations and provide cushioning for the panel. Two different versions of this prototype were produced based on ANSYS simulations resulting in the desired frequency reproduction: one that was 0.5x0.5m and one that was 0.75x0.5m.
Figure 3: Forced Response Analysis showing total deformation of prototype 1 in ANSYS

Figure 4: SolidWorks model of prototype 1 acrylic panel

Figure 5: Prototype 1 with acrylic panel
Prototype Two

The second iteration used a thinner, 3/16 inch polypropylene panel, with smaller dimensions, again arrived at by modeling on ANSYS. In this case the polypropylene panel, measuring 0.5m by 0.5 m, was nailed directly to a frame made of composite wood. It produced some frequencies relatively well, but rattled when others were played. Based on this observation it was apparent that another design had to be developed that included a vibration-absorbing material separating the panel from the frame.

![Figure 6: Forced Response Analysis Showing Total Deformation of Prototype 2 in ANSYS](image)

![Figure 8: SolidWorks Model of Prototype 2 (Acrylic)](image)

![Figure 7: Prototype 2 With Acrylic Panel](image)
Prototype Three

Proceeding with refinement, a third model was fabricated with the same dimensions as prototype two, but a different design. In this iteration we made the frame out of pine boards into which milled grooves in the centers to a depth of about half the width. The panel was inserted in the grooves with Frost King weather stripping rubber foam tape sandwiched in between. This design maintained the high-frequency performance of the previous model and eliminated the rattling, but still didn’t sufficiently produce the low target frequencies. Starting with this model, foam was also used to line the rear of the frame to prevent sound waves from traveling to the front and causing interference.

**Figure 9: Forced Response Analysis Showing Total Deformation of Prototype 2 in ANSYS**

**Figure 10: Prototype 2 Polypropylene Panel**
Prototype Four

The final design was of the same construction as prototype three, but with larger dimension to provide more surface area and therefore more deflection and volume. The first three designs were driven by premade Bose speakers and commercially available Dayton audio “bass shakers”. This model was driven by a moving-magnet transducer, produced by a fellow MQP group, capable of producing more force than the drivers used before. As our final prototype, this assembly was then integrated into a larger unit, housing the components produced by other groups within our MQP: moving magnet-powered passive radiator speakers meant for medium frequencies, standalone moving coil units for high frequencies, capable of being digitally steered, and two physically steered moving-coil speakers.

Prior to each fabrication, ANSYS was used to predict the behavior of the design at different dimensions and Solidworks was used to ensure continuity in understanding of the design between all group members and the professor. The Solidworks models also served as the model to be used for ANSYS.

**Figure 11: Solidworks model of final design with polypropylene panel and moving magnet**
Each speaker will be fed their optimum frequency by a series of circuits, ours receiving frequencies in the range of 40 to 60Hz. The sound steering will be guided by an infrared sensor array (the assembly for which was also designed and fabricated by our group) located at the top of the frame, capable of sensing body heat. This information will then be used to change the output of the steered speakers on an individual basis to create the effect of directed sound. During testing, the prototypes were energized with a battery-powered amplifier, whereas during the actual use the system would be powered with a much more significant amplifier, drastically increasing the deflection and volume.

**Acoustics**

From an acoustic standpoint, the goals that were proposed had been attempted before, but had yet to be perfected for industry use. Previous attempts by dissolved companies, such as
NXT, revealed that using a planar technique to produce low frequencies was possible, but also that it was extremely difficult to produce. In essence this was the crux of the problem we aimed to solve first by finding materials with qualities sufficient enough to reproduce an adequate response at low frequencies. In order to do this we turned to ANSYS, a modeling software used by Bose Corp. among other high profile firms.

The initial goal was to design a panel that would have a fundamental frequency at 40 Hz. In order to understand this concept, further details must be provided. To solve this problem one must have an understanding of the harmonic modes of motion. For an oscillating system, there are different modes of motion which characterize the vibration going through the material. These modes are used to describe dynamical systems in which a time dependent function can describe the motion of the system’s particle in a geometric space (in our case this particle would be a single point on our panel). A mode itself is defined as a standing wave state of excitation that affects all aspects of the vibrating system in a sinusoidal manner. Because of this principle we can think of modes as describing the oscillating behavior of a material in a sinusoidal manner. For reference the figure below are of modes 1, and 2. Mode 1 is simply what the group called a “dome mode” in that the shape of the oscillation we got at this mode was of a dome moving uniformly in and out of the paneling. Mode 2 is similar to that of a “sine wave” traveling north south through the panel. When one peak is reached the other part of the board reaches a trough as depicted in figure 13. Every mode has its own distinct pattern of oscillation, and this fact is the reason that decent bass response is so difficult to achieve for planar speaker. At the lower end of the frequency spectrum it is possible to reproduce adequate bass however encompassing all of the low end frequencies in such a manner as traditional speakers is very difficult. With subtle changes in frequency come more drastic changes in the mode, and not all modes are ideal for use
with planar speaker. Interestingly enough our group found the most beneficial mode for a planar
speak to its fundamental frequency. If we take into consideration figure 4 again, it is easy to see
how our group arrived at this conclusion. Essential one crucial difference between conal and
planar speakers is the way the sound is distributed.

*Figure 13: Comparison of Fundamental Frequency with Second Mode of the Resonant Panel*

In a conical system soundwaves are produced in a conical shape and are propagated at an
angle that is far less than ideal. This is because of the shape of the traditional driver.

*Figure 14: Sound Propagating from a Traditional Speaker [Pintres.com]*
Figure 15: Sound Propagation from a Cylindrical Sound Source [Integracoustics.com]

Figure 5 depicts how the soundwaves from a traditional loudspeaker are dispersed.

This figure serves as a visual representation of the flaws that a planar speaker can fix if allowed to oscillate in mode 1. As the sound waves leave the traditional loudspeaker they become wider and therefore cover more surface area from a listener's perspective.

The issue with this is that at close range there are obvious dead zones where the sound is simply not aimed because of how the waves leaves the loudspeaker. With a planar design oscillating in mode 1, we can change this by dispersing soundwaves a full 180 degrees allowing full coverage and even displacement of the soundwaves as depicted in figure_.

In order to accomplish this, traditional surround systems utilize multiple drivers and placement techniques. However, with a planer speaker the need for this is diminished all together. However, as previously stated, small changes in the frequency being played through the panel cause the panel not to vibrate in a sinusoidal manner as it does in each of its natural modes.
These changes in mode create an uneven frequency response which the group understood as a potential barrier for the design. The initial concept was to first achieve a fundamental frequency at 40 Hz, then use signal filtering to create an even frequency response from 41Hz up to 80Hz with the other drivers in our final design. The final design that was decided upon after multiple simulation efforts was 3/8th inch polypropylene panel with 3/8th inch thick ribs on the back. This initial final design is depicted in figure 17, as well as its dimensions. The polypropylene rib was added to stiffen the overall structure of the panel, thus allowing the fundamental frequency to be brought down to 40 Hz.
Measurements and Testing

A series of tests were conducted with each of our prototypes analyzing both the audio spectrum of the panels as well as the decibel sensation level (dbsl) to see how loud the speaker was. To measure both of these sets of data, a scanning laser vibrometer was used. The instrument works in that the computer sends a single sine wave of a frequency, or a sweep of a desired range to the system being tested. The laser vibrometer, which is housed in a camera like housing, sends a laser beam that is then split into two beams. The first goes to a reference source usually housed within the apparatus itself. The second laser then is directed at the system in question. The vibrations of the object cause the light to be scattered. Some of the light that is scattered is recollected by the vibrometer, and combined with the reference beam, and the frequency of the object’s vibration, its acceleration can then be calculated based on this data. [Lutzman et. al., 2016]
With the scanning laser vibrometer, all of the panel prototypes were tested with a Fast Fourier transform (FFT) test. This test applied a signal from 20Hz to 100Hz to the moving magnet transducers on the panel, causing the panel to vibrate. From this test, the modal frequencies of the panel could then be found. Modal frequency is a term used to describe a case in which the object vibrates at only one frequency throughout the entire body. This is also called the object’s natural frequency. [NTI]

![Figure 18: Example of the results from an FFT scan](image)

The figure above shows the results of an FFT scan measuring frequency against total deformation. Each of the peaks show a frequency at which the panel was moving the most. This scan showed that the panel had a modal frequency at each of the peaks. From this scan, it was determined the frequencies at which a fast scan would be done. A fast scan is function of the scanning laser vibrometer that measures the deformation of a surface by using the laser to scan an array of previously defined points on the surface. This scan is used as a test to determine the shape of the total deformation of the body due to the vibrations that were induced on the surface. The software then compiled the data retrieved from the scan into a color coded, 3D picture and animation of the actual deformation of the panel.
The above figure shows a still frame from the animation of the results of a fast scan on the panel. The legend at the top left show the amount of deformation given by each color on the panel in micrometers. Although the deformations look significant, they are only exaggerated to show deformation.

The two nodes (in red) on the panel show that there is more than one sine wave over the panel. These two maximums are thought of as being “in phase” because they are moving together with respect to the stationary frame around the panel. The middle of the panel (blue), shows a node that is going the other way than the red nodes. This trough is thought of as being “out of phase with the red nodes. Because the nodes are not all moving in the same direction, there will be cancellation in sound pressure level, causing the sound to be quieter.

The purpose of this test was to discover if the panel was achieving its fundamental frequency, or the lowest frequency of a waveform across the whole panel. This is shown by the vibration of the panel in so called “trampoline mode”, or vibrations that showed only one periodic sine wave over the entire panel. This was desired due to the fact that the fundamental frequency is the lowest modal frequency achievable and is often the loudest frequency. Having
more than one wave in the panel would cause interferences in the soundwaves, and thus cause cancellations in sound in front of the panel.

**Figure 20: Constructive Interference of Sound Sources [Fuchs, 2011]**

**Figure 21: Destructive Interference of Sound Sources [Fuchs, 2011]**

From the figure above, constructive and destructive interference is shown from sources that are in phase, or moving in the same direction at the same time, and out of phase, moving in the opposite directions at the same time. Based on addition of the sine waves from the sound sources there is either constructive or destructive interference. When dealing with the panel, having more than one wave on the surface of the panel was treated as two sound sources being out of phase, and thus would cancel each other out. Contrarily, it would be difficult to produce a wave that had two peaks in phase without producing any peaks that did not cancel out the sound.
For this reason, a fundamental frequency mode, or a mode in which there is only one wave, was desired.

Results

After an initial idea and four prototypes, the final product was able to achieve the desired specifications for both producing a modal frequency around 40Hz, and have a single periodic sine wave over the entire panel. This was achieved by etching in a ring on the front of the panel around the center point of the panel. The ring had an average depth of 1.5 mm and an average width of 5.08 cm. Taking this material off caused the polypropylene of the panel to be more flexible, and have more deformation at a lower frequency.

![FFT Scan of the Final Resonant Panel](image)

**Figure 22 FFT Scan of the Final Resonant Panel**

The above figure shows the final FFT scan of the prototype. The cursor on the screenshot shows a peak at 44.53Hz, and a range of high magnitude of displacement between 40Hz and around 60Hz. Because there was a peak at 44.53Hz, there was a fast scan conducted at this frequency.
The above figures shows a still frame from the animation created from the fast scan done by the scanning laser vibrometer. The final design achieved the trampoline mode that was desired. This means that the whole panel resonated as one surface at 44.53Hz. It also showed maximum deformation around the center of the panel, showing that it could move large amounts of air in order to produce the highest sound pressure level.

Below are side by side comparisons of the force analysis and harmonic response based on a theoretical Ansys model against the scans that were done on the panel that was built.
The above figures show the simulations done of the final resonant panel done in Ansys as well the final FFT Scan done in the lab. These graphs differ in their frequency response in that the Ansys shows a gradual decrease in the panel’s velocity until a large dip at 70 Hz, but the scanning laser vibrometer shows a consistent frequency response from 40 Hz to around 60 Hz with a large dip at 80 Hz. This can be attributed to the fact that the ring that was sanded into the panel was not consistent over the entire panel, as it was modeled in Ansys.

**Alternative Solutions:**

When the problem statement was first introduced, there were many different ideas that were conceived to try to solve it. The first idea was that the reverse piezoelectric effect could be
applied to a film to produce sound on a surface that is about as thick as a sheet of paper. The
piezoelectric effect is the ability of some materials to generate an electric charge when a
mechanical force is applied to it. The reverse of this is also true for piezoelectric materials,
meaning that by applying an electric charge to a piezoelectric material, a mechanical force can be
produced. [Yang, 2016] This idea was then found to be far too difficult to reproduce given the
time and access to materials that were available.

Prototype One

The next design that was discussed was to use a sheet of some material as a panel, and
vibrate it using some kind of transducer in its surface. This would cause the panel to act as a
speaker cone would, pushing air and creating a difference in sound pressure level, thus producing
sound. The first prototype was modeled in Ansys with the goal in mind of achieving 40Hz. The
original material for the panel was 0.25” clear acrylic. Acrylic was used because the material
desired needed to be relatively thin and also deflect well with force. The model was refined in
Ansys until a peak was achieved very close to 40Hz. The final design consisted of a frame
measuring 1 m by 0.5m, and a sheet of acrylic of the same dimensions.

Once the analysis was complete, the prototype was constructed according to these
dimensions and tests were done with the scanning laser vibrometer. Because the acrylic was
clear, it had to be painted white in order to get results from the scanning laser tests.
The two figures above, show the frequency response rate of the original prototype over the spectrum from 20 Hz to 200 Hz. The response shows peaks at around 60 Hz and 80 Hz, showing that the panel had the greatest total deformation at these points. Because the group was more concerned with the bass frequencies, only a fast scan at 60 Hz was done. The second figure shows the results of the fast scan at 60 Hz. There is clearly three nodes that exist within the panel, and, as stated before, this is not desired due to the properties of sound cancellation.
Prototype Two

After considering the size and weight of the first prototype along with its performance, aspects of the design were refined more to try to get the fundamental frequency closer to 40 Hz. One thing that changed was the material used for the panel. Polypropylene was substituted for acrylic because the polypropylene is more flexible than acrylic, and would thus deflect more under the same force. The shape was also changed. Because a perfect trampoline mode was desired, a square frame and panel was constructed to give even distance on each side of the center. This would allow the panel to deflect evenly across its surface. Additionally, channels were cut into the frame to allow the panel to sit inside of it, and give more even boundary conditions over the length of each side. Foam was adhered to either side of the panel in order to create a boundary between the polypropylene and wood, to minimize any rattle. Lastly, a mainly bass transducer was used to resonate the panel. Because the final product design called for a modal frequency of 40 Hz, the group decided to focus mainly on the lower frequency using this low frequency transducer.

*Figure 28: Second Prototype with Polypropylene Panel*
After testing the second panel, it was found that the panel needed to be have a larger area to both lower the fundamental frequency of the panel as well as increase the sound pressure level difference. The material decided upon would still be polypropylene due to its flexibility, light weight, and its damp response to high frequencies. The final model is 490mm wide by 600mm tall. These dimensions were tested in Ansys using a foam boundary condition stated above with fixed geometry on the outside of the foam material, and gave a fundamental frequency at 40Hz.
Final Design:

The final design that was used consisted of a polypropylene panel measuring 490mm by 600mm. This panel had a groove sanded into it around on the front with an average radius of 240mm, an average width of 5.08mm, and an average depth of 1.5mm. This groove helped to make the material thinner, thus lowering the fundamental frequency and help reach the trampoline mode stated above.
The panel was secured on its four sides by channels cut into pine wood. The channels cut into the wood measured 0.5” wide but 0.5” deep and ran the length of each of the pieces of wood. The foam on the edges of the panel was force fitted into the channels so as to create a snug but flexible boundary condition. This allowed the edges of the panel to move a little bit, but does not allow contact between the wood and the polypropylene, which would cause some amount of rattle.

The edges of the panel were lined with ⅜ inch Frost King high density rubber foam tape on the front and back sides and fitted into a ½ in by ½ channel on all four sides. The channel was cut into a piece of 1x2 inch piece of pine wood as a frame, this frame was then screwed together to secure the panel on all four sides.

On the back surface of the panel are 3 moving magnet transducers each bonded to a 0.25 inch plastic spacer. The plastic spacer allows the rest of the moving magnet transducers to move
without making contact to the polypropylene. The spacers are then bonded to the polypropylene with plastic adhesive. This bonding ensure that the moving magnets will not break contact with the polypropylene, and ensure that the maximum vibrations are transmitted to the panel.

![Figure 34: Dimensions of Polypropylene Panel and Placement of Moving Magnet Speakers](image)

The figure above shows the overall dimensions of the polypropylene panel, the positioning of the moving magnet speakers, and the groove that was cut into the panel.

**Cost Analysis**

The bill of materials of the polypropylene panel speaker, minus the cost of the moving magnet transducers, includes a 490mm by 600mm sheet of polypropylene that cost $8.34 from McCaster Carr, three 1”x 2”x 8’ pressure treated boards costing $1.84 each, screws at a cost of $9.00 for a pack of 73 (only 8 are needed), and the 3/8” Frost King rubber foam costing $5.00 for one roll. In total, the cost of the resonant panel is itself is $27.86.
Conclusions:

This project attempted to produce a low-profile alternative to conventional subwoofer technology, replacing volume with surface area. It researched many potential technologies to achieve such a task, settling on trying to take advantage of the resonant frequencies of various materials to create sufficient deflection, thereby producing air (sound) waves.

Several iterations of this concept were produce as the general concept was refined. Design 1 didn’t work as well as hoped, potentially due to the lack of flexibility of the quarter-inch Plexiglass that was utilized. Design two, on the other hand, worked decently at frequencies higher than the target range, using an existing speaker design as a makeshift transducer for proof of concept testing. Design 3 originally came close to resonating at our target frequency. The resonant frequency was adjusted by sanding a circular pattern into the panel, effectively making a thinner panel. This design was then placed in a wooden frame which would fasten it to complementary audio technologies.

Now that the panel resonates at the correct frequency, the remaining barrier to an ideal design is the lack of appropriate volume output. This could perhaps be fixed by further adjusting dimensions or material, which a future MQP could take on.

Recommendations:

Going into this project the group’s initial goal was to create a bass panel with a natural resonance at 40Hz and at mode 1. The idea behind this was to disperse this low end frequency at 180 degrees throughout a room. As a group the testing of the final product revealed that the goals
were achieved not only at the 40Hz peak, but at frequencies all the way up to 60Hz. From this perspective, the project itself was successful and the group would give it the “Go”.

Additional recommendations the group had were with regards to the manufacturing process. The use of silicon over foam for the enclosure between the panel and its frame would minimizing vibrations heard in the panel. Furthermore the additional sanding performed that allowed our group to achieve 40Hz would need to be studied more closely. Attempts to create a sustained peak could result in a more ideal planar bass driver and would add to the overall performance and sound quality of the product.
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Appendix:

Ansys Tutorial: Force Analysis and Harmonic Response

Saving your model, the right way!

- Save model as a STEP file (.step) – This way ANSYS will be able to manipulate it

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Importing a STEP File with Ansys

• Open ANSYS Workbench (18.2 in this case)
• The Left hand side of the window you just opened contains the Analysis Systems. Click and drag over the Harmonic Response system into the white space to the right as shown below.
• In the Harmonic Response menu right click Geometry then click import geometry
To import your step file.

Adding your material Properties Pt. 1

• With your step file imported you will be able to manipulate it in ANSYS, however before you do this it is necessary to ensure the correct material properties have been provided.
• Right click Engineering Data in the Harmonic Response Menu and click on edit. This will bring you to the Engineering Data tab
Adding your Material Properties Pt. 2

- You will need to Add the materials you used in your design.
- Click add new material below structural steel here you can type the name of your new material. (Alternatively you can click the Engineering Data sources menu to the left to parse through Ansys’ in built materials list. If your material is there click to add it and ignore the steps below.)
- You’ll need to add the materials Young’s Modulus, Density and Poisson ratio. This typically can be found on google.

*note in order to input these variables you must drag the Density and Isotropic Elasticity Properties from the physical properties menu and drag them to the material properties section.

Conducting Harmonic Analysis Pt. 1

- With the material properties entered return to the main page by clicking the project tab.
- Once there in the Harmonic Response menu, right click model and click edit. The photo to the right is what you should see. The model will be shaded based on the different extrusions.
Conducting Harmonic Analysis Pt. 2

- The first step will be to right click mesh in the menu of the right and hit update. Your model should change accordingly.

Conducting Harmonic Analysis Pt. 3

- Now we will add material properties.
- In the menu on the right click geometry, a list of each body in the model should appear.
- Clicking each individual body will bring up a menu below. In this menu under the materials tab you can change the material to the desired on by right clicking on it.
Conducting Harmonic Analysis Pt. 4

- Right Click Harmonic Analysis, in doing so you should be able to insert a force. As well as set minimum and maximum range frequencies for your simulation.
- To insert a force specify it’s magnitude in the menu below and its direction then select a face for it to interact with on your model using the geometry button. All buttons will be in this menu to the below.

Depending on where you place your force, your model should look like the image below.
The force is the red arrow.

Running the Simulation

- To run the simulation all you have to do is right click on solution and hit solve. If you followed all the steps correctly you should be able to see your data by clicking various tabs in the solution menu.
You’re done!

• At this point you now know how to use ANSYS to do force analysis and harmonic response!