Final Design Report:

Vibration Table for CVBT- VT4 Design Team

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Executive Summary:

CVBT (Center for Vocational Building Technology) teaches villagers in Northeastern Thailand how to make various concrete products. Many of these products require the use of a vibration table in order to remove air and water trapped within the concrete. Removal of these voids improves the outside surface of the molded concrete, and also allows the use of a lower water to cement ratio, allowing a much stronger finished product. The currently used vibration table (VT3) has several disadvantages associated with it, which the new table (VT4) eliminates. Compared with the VT3, the VT4 has a more rigid tabletop and improved suspension system in order to vibrate the concrete within the wooden mold frames and maintain the integrity of concrete molds. Additionally, with the use of a single Damping Mass of 10 kg, the VT4 outputs the required amplitude of 0.3 to 0.4 mm and frequency of 3900±200 RPM to correctly consolidate concrete ranging from 2 to 40 kg. Furthermore, as desired, the VT4 uses the same motor and support frame as the VT3. See Figure 1 for a drawing of the final VT4 design.
**Introduction:**

The overall project is to design an adjustable vibration table with a rigid tabletop and suspension system for the Center for Vocational Building Technology (CVBT). This will be used in Northeastern Thailand to create employment for the villagers. The table will be used in an important step in producing concrete paving bricks and roof tiles from molds. Vibrating the concrete reduces the number of voids in the mixture, which in turn reduces the water to cement ratio (w/c). This allows for a stronger finished product and maximizes the use of materials. Vibration also improves outside surface finish, which allows the product to look better and last longer by reducing the chances of cracks forming from foreign contamination.

CVBT’s own developed vibration table does not currently meet the necessary requirements that they desire. This current design includes a flexing tabletop that resembles the head of a drum. This provides uneven vibration which may result in a non-uniform finished product. The current vibration table is also incompatible with the current wooden tray design. Therefore, the molds must be placed directly on the vibration table, which reduces the mold life span considerably due to the mold not being supported by a more rigid wooden tray. Additionally, lips of the plastic molds occasionally crack under the stress of the concrete while transferring from the table to the cooling stack.

**Background:**

Our initial background research consisted of finding existing vibration tables that might perform the necessary task. From this search, several different categories of table design emerged. For suspension, three different types were present. These categories are springs, rubber mounts and air cushions. The springs were used in two different configurations, including hanging springs (in tension) and vertically aligned compressed springs (see Figure 22 and Figure 24 in Appendix C). The designs that used springs in tension generally allowed vibrations in the horizontal plane, while the compressed springs allowed for vertical vibrations. The ASTM standard for concrete vibration states that the tabletop vibrators must operate in the vertical direction. This standard limits the spring suspension design to vertically compressed springs. Solid rubber mounts in compression (as used in VT3) were less widely used, as they did not allow as much movement as the other two. However they are generally the least expensive, and may be used if low amplitude vibrations are desired. Rubber mounts can also be mounted in such a way as to only experience shearing forces. The effective spring rate can be calculated from the dimensions of the rubber mount and the shear modulus of the rubber. This configuration is also good for low amplitudes of vibration, and when a moderate amount of damping is required. Rubber mounts used in shear can be seen on the Syntron Vibratory Equipment vibration table (see Figure 25 in Appendix C). In a few instances, rubber mounts were also used as a centering pivot point for inverted pendulum based vibration tables (see Figure 23 in Appendix C). This configuration allowed for horizontal movement only and therefore cannot be used in our design. The last suspension category, air cushion, was the most common (see Figure 10). Air cushion suspension was seen in most of the higher end vibration tables. All the designs seen using this type of suspension were configured with a rigid table mounted with air cushions in the four corners. Air cushions are generally more expensive and less
readily available than the other suspension types listed. However, these air cushions are very adjustable, as air can be easily added or removed from the system. The change in air pressure will change both spring rate and damping, which is an important quality to have in our design since such a wide range of loads will be used on the tabletop.

As well as researching the different types of suspension, many different sources of vibration were found. However, our design requirement of using a 12 volt DC motor limited the list of sources to a few viable options. Sources of vibration were limited to using a rotating shaft with an eccentricity or using cams. Both options allow for adjustable vibration amplitudes. The rotation of an eccentric mass was the most common source of vibration seen. This design is easy to fabricate, has minimal wear, and can be relatively easy to adjust. Alternatively, cams will provide constant amplitude of vibration, regardless of load (unlike a design using an eccentricity). However, cams will be difficult to machine because according to the ASTM handbook for concrete vibration table design, vibration amplitudes should range from 0.3 to 0.4 mm at a frequency of 3900±200 RPM. Also, our low torque motor will be unable to lift the heavy loads required of it. Therefore, the most viable source of vibration is the rotating eccentricity as used in the VT3.

Steel was almost exclusively used in the construction of vibration tables. We feel that steel is the best option for our design as well, due to its durability, relatively low cost, rigidity, and high availability. Steel is also a very forgiving metal to weld, unlike aluminum, which usually requires a heat treatment process after welding.

In addition to researching existing products, we contacted various local concrete production facilities, which provided us with valuable information from the industry. We also toured Midstate Concrete, a precast concrete plant, and spoke to concrete specialist Rob Vander Veen. We found that vibration tables used in industry are designed for a specific concrete mixture. This is because desired amplitudes, frequencies, and vibration times change with specific mixtures and are all taken into consideration when choosing table components. Therefore, we found that it is important to keep the mixture constant between batches in order to have a uniform product. One method to control consistency is the use of a slump cone and associated slump test. According to Rob Vander Veen, concrete strength is determined almost entirely by the components in the mixture. In his facility, vibration of the concrete is only used to improve the surface finish, or cosmetics of the product. We were also told that the ‘throw’ of the tabletop is a very important design consideration. The term throw refers to the ability of the table to disperse the aggregate in a desired direction within the concrete mixture. Improper throw can cause non-uniform distribution of aggregate. For this reason, most concrete vibration tables have two opposing eccentric masses, in order to cancel out any horizontal force seen by the tabletop. This will be considered in our design, but may not end up being cost effective. There are also other ways to induce uniform throw over the entire concrete mold, such as having the user rotate the mold manually during the vibration process. Knowing the throw of the table is important because it determines how the aggregate is displaced within the mold after vibration. A simple way to check the throw of the table is to put sand on the table and see how it is displaced when the table is vibrating. If there is no extreme tendency for the sand to shift in a preferential direction the aggregate within the mold should also be fairly evenly dispersed.
Besides using traditional vibration, many concrete production facilities are transitioning to using self-consolidating concrete (SCC) exclusively. SCC contains a mix additive which replaces most of the water normally used in the mix. This additive reduces the w/c ratio, increases the strength, and (according to the manufacturer) eliminates the need to use vibration. In order to keep the same strength (between SCC and standard concrete), less cement is needed for the given application. Since the SCC additive is less expensive than cement and the operation costs are minimized, the price of the product is reduced. We feel SCC should be considered in the future for a CVBT concrete production alternative.

**Objectives:**

The goal of this project is to create a rigid vibration tabletop and suspension system with adjustable amplitude, which mounts to the existing CVBT VT3 frame. This upgraded design will be used for increasing the lifespan of the molds, reducing the w/c ratio in concrete, and improving the cosmetics and therefore the life span of the finished product. This vibration allows for an overall better finished product. The vibration table must have the following characteristics:

**Design Goals:**

- Be able to vibrate between 2 to 40 kg of concrete
- Maintain the amplitude of vibration between 0.3 and 0.4 mm for all mold sizes
- Use the Tokushu Denso TD2570-333 motor
- Design must use the same frame as the VT3
- Must vibrate the whole tabletop, not just the center like the VT3
- Tabletop must not flex more than 0.2 mm under load
- Compatible with wooden crates and molds ranging in size from 45 x 45 cm to 30 x 30 cm
- Must operate within the frequency range of 3900±200 RPM
- Be able to work in an unclean environment, including concrete, moisture and dirt
- Durable; must last without major repair for 2-5 years
- Must be easily built and repaired using parts available from nearby department store and tools that are readily accessible (lathe, drill press, arc welder, and various hand power tools)
- Must run for 3-5 work days on one fully charged 12 volt battery
- Final product must cost less than 8500 baht ($253) to build
- Prototype must cost less than $700 to build, provided by Cal Poly
Deliverables:

- Metric drawings updated with new design on CVBT title block. Files will be in .pdf, AutoDesk Inventor formats. Part numbers assigned according to CVBT standards.
- vibrator and tabletop designed, fabricated, and tested.
- Specified manufacturing processes, jigs, tools, and process plans. Any special jigs or tools designed and fabricated.
- Cost list of assembly and parts.
- Table frame fabricated according to existing design.
- 2 wooden mold frames fabricated according to existing designs.
- English Language and Thai Language Instruction sheet.
- Maintenance instruction and record sheets

Table 1: Design Parameter Summary Table.

<table>
<thead>
<tr>
<th>Spec. #</th>
<th>Parameter Description</th>
<th>Requirement</th>
<th>Tolerance</th>
<th>Risk</th>
<th>Compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Total Project Cost</td>
<td>$253 Max</td>
<td>H</td>
<td>A</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Tabletop Amplitude Range</td>
<td>0.3 to 0.4 mm Min, Max</td>
<td>M</td>
<td>A, T, S, I</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Load Range</td>
<td>2 to 40 kg Min, Max</td>
<td>H</td>
<td>A, T</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Tabletop Frequency Range</td>
<td>50 to 70 Hz Min, Max</td>
<td>M</td>
<td>A, T, S, I</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Noise</td>
<td>&lt;60 db Max</td>
<td>L</td>
<td>T</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Durability</td>
<td>3-5 Years Min</td>
<td>H</td>
<td>A, T, I</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Power Source</td>
<td>12 Volts</td>
<td>L</td>
<td>A</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Mold Accommodation</td>
<td>45 x 45 cm Max</td>
<td>L</td>
<td>A, T</td>
<td></td>
</tr>
</tbody>
</table>

Abbr. | Significance
---|----------------------
H  | High
M  | Medium
L  | Low
A  | Analysis
T  | Test
S  | Similarity to existing designs
I  | Inspection
Management Plan:

As our design team consists of two members, working together has shown, in many project tasks, to have the highest efficiency and create the highest quality product. However there have been several areas of the design process that dividing the work load has worked best if. These areas include the CAD drawings, initial calculations for our respective design concepts, in-depth calculations for subsystems of our selected design, and construction of the prototype. The Gantt chart in Appendix A shows our time management plan and project flow path in an easy to read format. This Gantt chart displays the order in which the various tasks must be completed, which tasks are related to each other, important due dates, and projected amount of time allocated for individual tasks.

Design Development:

From initial brainstorming, a few designs emerged that we felt would work well for the VT4. The first of these designs was a rigid tabletop that had a cam directly attached to the motor shaft. This design would have the benefit of keeping constant amplitude of vibration for any load size. After refining our background research we found that the amplitudes of the tabletop should be between 0.3 and 0.4 mm. From this we determined that machining a cam for that range of amplitudes would be extremely difficult and no longer a viable design solution with the given tools available. The next two brainstorming ideas involved the tabletop sliding horizontally on glide bearings positioned on the top of the frame. These designs have the advantage of taking the preload from the concrete off of the springs. The first design consisted of one vertically aligned car spring in tension. A positive aspect of this design was that the amplitude could be change easily. This spring would act in the transverse direction, so a transverse spring constant equation, based on the physical dimensions of the spring, was calculated from Castigliano’s Theorem. This equation was performed in Matlab and can be seen in Appendix F. The second design with glide bearings was a four corner spring design that had the spring mounted in a horizontal position. Using a Simulink model of a spring with an eccentric mass we checked to make sure that the correct amplitudes were possible from these systems. The Simulink model is described in more detail in the following sections of the report and can be seen in Appendix E. A calculation was done in order to show that the glide bearings would not wear out too quickly. This calculation can be seen in Appendix D. Design sketches for both horizontal motion designs can be seen in Figure 2.

Figure 2: Horizontal motion design sketches, showing glide bearing locations and spring alignment for axial and transverse deflection.
From our calculations both horizontal design ideas showed to be viable design solutions. Further background research of ASTM standards for tabletop concrete vibrators ruled out both of the designs, stating that tabletop vibrations are required to act in the horizontal direction. From this analysis we acquired a useful tool for calculating the horizontal displacement of our tabletop with a vertically aligned suspension.

![Diagram of tabletop reinforced with angle iron and rubber mounts replaced with springs, rubber shocks, or air cushions/shocks.](image)

**Figure 3: Design concept layout for various suspension systems.**

After preliminary calculations and more specific background research, the concepts for our prototype design were narrowed down to three viable solutions. All three designs have a rigid tabletop reinforced with angle iron. These designs share a common four point contact vertical displacement tabletop design, but differ in suspension type. Figure 3 shows how the various suspension types will replace the rubber mounts of the existing VT3 design in their exact location. The first design uses four vertically aligned springs in compression, the second design uses four rubber mounts acting in shear, and the third design uses air cushions or air cylinders. We have also proposed using a single bike inner tube in place of four air cushions. This has the advantage of a lower initial cost, increasing contact area (less relative tabletop deflection), and keeping uniform pressure distribution. With the single inner tube suspension design the tabletop will not have to have angle iron reinforcement because of the increased contact area. Tabletop deflection analysis for both the reinforced and not reinforced configurations can be seen in Figure 4 and Figure 16 respectively.

**Tabletop:**

Although angle iron reinforcements were not implemented into the final design choice for the VT4, the following section summarizes the analysis performed on the reinforced tabletop design that is common between the three four corner suspension type design options. The tabletop for our design has to be very rigid because the amplitudes of vibration are between 0.3 and 0.4 mm. Since the amplitudes are so small, the flex of table due to the forces from the eccentricity coupled with the weight of concrete and acceleration forces must be limited to less than 0.2 mm. This is to ensure that the tabletop does not vibrate only the center of the table, as with the VT3. To accomplish this required rigidity and
to minimize cost, we decided to keep the existing tabletop thickness of 4 mm, while adding angle iron structural supports in an optimized location. This optimized location can be seen in Figure 4.

![Figure 4: Deflection analysis of tabletop design, performed in Solidworks](image)

Figure 4 shows a 4 mm thick steel plate with four sections of angle iron welded to the underside. The steel angle iron has dimensions of 60x60x5 mm and is also made of steel. The simulation is representative of when the tabletop is at its lowest point, in which the springs are at their maximum compression. From our mass spring damper Simulink model described in the following section on spring suspension, we expect that this location will cause the maximum bending of the tabletop. At this location, spring forces and the eccentricity force are acting on the bottom of the tabletop, while the weight of the concrete and the acceleration forces are acting on the top of the tabletop. This simulation assumes a distributed load of 392.4 N (40 kg of concrete) over an area of 45x45 cm, which is the maximum load required of the vibration table. The simulation also accounts for the forces exerted by eccentricity distributed across its contact area (as seen by the small rectangle located in the bottom center of the tabletop). This force was calculated using an m*e (mass times eccentric radius) of .00346 kg*m, which is approximately 2.5 times greater than the eccentricity used on the VT3, and assumes a rotation speed of 3800 RPM. This force is caused by the angular acceleration force of the mass imbalance and amounts to 548 N. The dimensions of the steel plate are the same as the larger plate used on the VT3 with major dimensions of 600x550 mm. The maximum displacement seen in the tabletop for the forces provided is 0.069 mm, located in the corners of the tabletop. This displacement is well below the required maximum deflections of 0.2 mm, and only appears in non-critical areas of the
If the angle iron reinforced tabletop was chosen for our final design, further analysis would be done to minimize the size of the angle iron in order to reduce weight and overall cost.

![Stress analysis of tabletop design performed in Solidworks](image)

Figure 5: Stress analysis of tabletop design performed in Solidworks

Figure 5 shows the stresses associated with the same system described in Figure 4. As can be seen, the stresses are negligible in the design of the tabletop. Figure 5 shows that at the point of maximum stress, the design had a minimum factor of safety of 9.34. This is because the tabletop is designed to be so rigid. Overall the deflections of the tabletop are what will drive the design. As of now, we feel that the stress and deflection analysis have a few notable differences between the model and the actual system. Our model assumes a perfect joint between the angle iron and the tabletop, while the actual model will most likely use spot welds. Also, the spring contact points are modeled as stationary pivot points, when in actuality the springs/rubber mounts will be dynamic. However, in order to make our static model more representative of the actual system, we added a distributed acceleration force of 1248 N to the top of the table. The calculations for the forces seen on the tabletop can be seen in Appendix D.

**Spring Suspension System:**

The spring concept design will feature four springs mounted in the contact points shown in Figure 1. A Simulink model (see Appendix E) was created in order to predict the required spring rate needed to match the frequency (3900±200 RPM) and amplitudes (0.3 to 0.4 mm) desired for our VT4 system design. In this model, spring constant, damping, load, frequency of rotation, and m*e are all variables, and uses the equation of motion developed in Appendix D. Initially, we inputted the same eccentricity found in the VT3 into our Simulink model. Additionally, we assumed a very low damping value to model a system with springs only. By using a load of 60 kg (concrete plus tabletop mass), and a frequency of 3800 RPM, we determined that the system response changes dramatically with the change in spring constant value. At low spring rate values, k, the natural frequency of the system is greatly
reduced from the 3800±300 RPM requirement. For instance, with the motor input frequency of 3800 RPM the tabletop may only vibrate at a fraction of that frequency. This output can be seen in Figure 6.

![Simulink Plot of input force and tabletop displacement over time.](image)

**Figure 6:** Simulink Plot of input force and tabletop displacement over time. The plot shows the reduced tabletop frequency with low k values and minimal damping. \( k=20 \text{ N/mm}, C=0.001 \text{ N*sec/m}, \text{ and } m_e=0.00146 \text{ Kg*m} \)

On the other hand, with larger values of k, the natural frequency matches the input frequency of the motor. In this situation the beat frequency emerges, which is a secondary vibration frequency. This beat frequency is undesirable because the amplitude of vibration varies with time in a periodic manner. This beat frequency can be seen in Figure 7.
Figure 7: Simulink Plot of input force and tabletop displacement over time. The plot shows the beat frequency apparent with high $k$ values and minimal damping. $k=2000 \text{ N/mm}$, $C=0.001 \text{ N*sec/m}$, and $me=0.00146 \text{ Kg*m}$

In order to obtain the desired output of the system, a damper must be present. With a damper the beat frequency disappears and the system has constant amplitude with a frequency equal to that of the drive shaft. However, the damper does directly affect the amplitude of vibration and therefore, for the same amplitude, a different eccentricity is required. This is shown in Figure 8.
Figure 8: Simulink Plot of input force and tabletop displacement over time. The plot shows the suppressed beat frequency apparent when moderate damping is used with high $k$ values. $k=2000 \text{ N/mm}$, $C=10 \text{ N*sec/m}$, and $m_e=0.00146 \text{ Kg*m}$

Overall, this Simulink simulation is a simplified model of the actual system, as it has a few key assumptions that may not be negligible. First, the model treats the tabletop as perfectly rigid member, although Figure 4 shows that this is not the case. Additionally, the simulation assumes that the motions of the springs are restricted to the vertical axis, and that all four springs vibrate synchronously. In actuality, the rotating mass will put a changing horizontal force on the springs and therefore induce a horizontal vibration. This horizontal force will translate to a horizontal deflection of the tabletop. The magnitude of the horizontal displacement is calculated using the transverse spring rate of the springs. The transverse spring rate is calculated in our Matlab program (see Appendix F) derived from Castiglano’s Theorem. When sizing the spring suspension this transverse deflection will be taken into consideration. Finally, the simulation does not take into account the damping produced by the concrete or the operator. Since the concrete is not cured yet, it is essentially a fluid damper sitting on the tabletop itself. The effective damping of the concrete is extremely difficult to model and will drastically change with each mold size. Additionally, operators are instructed to push down on the concrete in the molds while vibrating. This operator input is also an effective damper and will vary from operator to operator. For his reason the response of the system will have to be verified through testing.
Rubber Mount Suspension System:

In our rubber mount suspension concept for the VT4, four rubber mounts will be attached at the same contact points as shown in Figure 3. The mounts may resemble the one shown in Figure 9, or they may also be configured in a circular housing (not shown), like a motor mount.

![Shear rubber mount configuration](image)

The rubber mount suspension design and analysis is very similar to that of the spring suspension, but there are two key differences. First, unlike the springs, the rubber mounts will have an inherent damping associated with them. This is a positive aspect of this design since, as previously described, damping is desired in our system. Unfortunately, once a mount is selected, the amount of damping cannot be easily changed to account for differences in mold sizes. Secondly, with the rubber mount configuration, vibrations can be limited to the vertical direction. This is also beneficial for our system and makes our Simulink model (same Model used to analysis spring suspension) more representative of the actual system. From the force applied and the deflection angle $\gamma$ desired, the spring rate of the mount can be calculated based on the shear modulus and the dimensions ($a$, $b$, and $c$) of the rubber. By varying the dimensions, the effective spring rate of the mounts can be optimized. A sample calculation of the effective spring rate can be seen in Appendix D.

Air Cushion Suspension System:

As with the other two suspension types, the air cushion design concept for the VT4 consists of four air cushions/shocks mounted in the same location shown in Figure 3 or a single inner tube suspension seen in Figure 15. This air suspension is modeled with the same simulation used by the rubber and spring suspension systems. Air cushions/shocks also have an effective spring rate associated with them. If the spring rate is unknown it can be calculated for a cylindrical shock based on the dimensions and pressure, but for an air cushion of irregular shape the spring rate may not be linear and would therefore have to be determined experimentally. The benefits to an air suspension system is that the damping/spring rate can be easily changed by inflating or deflating the suspension. This would be a simple way to control the vibration amplitudes for various mold sizes. Additionally, in some instances, depending on the shock/cushion, the vibration can be limited to the vertical direction. Figure 10 shows an example of a vibration table with an air cushion type suspension.
Eccentricity Assembly:

The eccentricity assembly in our concept design is similar to the one used in the VT3. However, the input shaft has increased to 0.5 in diameter so as to allow a wider selection of bearings. This is necessary because the nearby department store located in Thailand has a limited selection of bearings and shafts. In order to prevent unnecessary bearing impact loads, the input shaft is press fit with both of the bearings and the bearings are press fit into their seats. Further more, the eccentricity itself will be changed slightly from the eccentricity used in the VT3, in order to allow for the amplitudes of vibration to remain within specifications. Two primary eccentricity options were considered for the VT4.

The first option is based on the idea of having an eccentricity that is easily adjustable and will function as the primary variable in tuning the amplitudes of vibration for different mold sizes. While the VT3 required the user to disassemble the vibration box assembly in order to remove and then reposition the cylindrical eccentricity, the first option (See Figure 11) allows the user to change both the mass and radius of mass to center without disassembly. A ratchet coming into the assembly through the bottom can thread additional nuts or plates on to the U-Bolt, thus changing the forces exerted by the spinning mass. This change in force results in a change in tabletop amplitude and frequency (due to the motor curve seen in Appendix G, Figure 26). The calculations used to predict change in force due to changes in eccentricity are found in Appendix D. In order to prevent loosening of nuts on the U-bolt, either a thread locker (LocTite) or stacking (two nuts torqued together in order to cause a high pre-load) can be used.
Figure 11: Eccentricity assembly, with input shaft, housing, bearings, and eccentric mass. The eccentricity itself is a U-bolt which allows for an easy way to add mass and or change eccentric radius. This is accomplished by adding bolts to the free end of the U-bolt, and then using the threads to change the distance to the center.

The second option (chosen for the final design) uses the same eccentricity as the VT3 but with slightly different dimensions. This chosen design can be seen in Figure 12. This option is not adjustable but is fine tuned to work with the other chosen design variables (tube height, shaft speed, or Damping Mass). Although the adjustable eccentricity was not used in the final VT4 design, it was used as a tool to find the appropriate eccentricity dimension needed for the final design. Instead having to fabricate multiple eccentricities like Figure 12 with varying dimensions, the U-bolt eccentricity was adjusted until the correct amplitudes and frequencies of vibration were met. Even though the adjustable U-bolt eccentricity assembly was proven to work, the second option was chosen for a few key factors described later in the final design section.

Figure 12: Eccentricity assembly, with input shaft, housing, bearings, and eccentric mass, showing the non-adjustable eccentricity chosen for the final VT4 design.
Rheostat:

Another available option which will change the amplitude of the tabletop is a rheostat. A rheostat (variable resistor) can be connected in series with the motor in order to provide a different voltage drop through the line. This voltage drop means the motor itself will receive less than 12 V. Therefore, even though the battery is outputting a constant voltage, the motor will only be able to use a certain percentage of it, due to the voltage drop across the rheostat. By manually changing the resistance across the rheostat, the motor will be able to change from a rated speed of 4500 RPM to about 500 RPM (based upon the max resistance available on the rheostat). By changing the speed, the amplitude of vibration of the tabletop will also change. See both Figure 13 and Figure 14 to see the impact of a change in motor speed on the amplitude of the tabletop.

Figure 13: This plot shows the displacement of the table and input force from eccentricity vs. time. The spring constant = $2\times10^6$ N/m, $c = 10$ N*sec/m, $m*e = .00248$ kg*m, and input speed = 3800 RPM.
Figure 14: This plot shows the displacement of the table and input force from eccentricity vs. time. The spring constant $k = 2 \times 10^6$ N/m, $c = 10$ N·sec/m, $m \cdot e = 0.00248$ kg·m, and input speed = 3700 RPM. Notice the large difference in displacement from the previous graph, due to a small change in input speed.

However, there are some problems in using a rheostat in our system. During use, and depending on the resistance setting, the rheostat may get very hot. The rheostat would have to be placed in such a location that the heat will not affect the user. Also, because the rheostat is removing energy from the system by way of non-usable heat, the efficiency of the system drops depending on the rheostat resistance setting. Output torque will also drop as the rheostat’s resistance increases, and may reach a point where the motor can no longer rotate the eccentricity. See Appendix G for the motor curves. Additionally, as seen in our Simulink model, small variations in shaft speed can have large effects on the amplitude. The exact change in efficiency, speed, and torque for a change in resistance cannot yet be determined, as the armature resistance and other motor characteristics are not yet known. According to Ali Shaban, a professor in the EE department at Cal Poly, testing is the best (and in some cases the only) way to find how a rheostat will affect the motor and system characteristics.

**Damping Mass:**

Additional mass can also be used to change the amplitudes of the tabletop. By varying the size of the mass hanging from the bottom of the tabletop depending on the mold size, the effective mass that the tabletop sees can be maintained to a relatively constant value. For instance if the tabletop were designed to reach the desired amplitude and frequency for a 40 kg mold, the operator would add 15 kg to the tabletop when a 25 kg mold is vibrated. Therefore, for each case the vibrator will respond similarly. For each case, small amplitude variations will result from differences in weight distribution. Testing will determine the minimum number of masses and their respective sizes needed in order to maintain the amplitudes within specification for all mold sizes.
Development Test Plan:

Since the three suspension types (spring, rubber mounts, and air cushions) were all viable options for the final design of the VT4, we decided that a prototype must be fabricated in order to progress our design process in the most efficient way possible. Although it would have been beneficial to build and test each design and then compare results, because of time limitations and cost considerations, we decided to start with the air cushion design since it has the most desirable characteristics. These characteristics include damping, maximum weight distribution, low cost, and the ability to change the tube height, shaft speed, and size of added mass to optimize performance with different mold sizes.

Figure 15: This figure shows a side view of the bike inner tube concept.

For our air cushion prototype we decided to start simple and then add components such as the rheostat, stabilizers, and the adjustable eccentricity as needed. A model of the prototype can be seen in Figure 15. As seen in this figure, we chose to use a 406.4 mm (16 in) diameter thorn proof kids’ bike inner tube for the air cushion suspension. We selected this inner tube because it is a readily available size and thorn proof. The thorn proof model is desirable because it has thicker walls, therefore allowing a longer life in terms of wear. It will also fit inside the outside edges of the tabletop, leave enough room for the stabilizers, and leave enough space between the inside tube walls and the vibration box.
The single inner tube design was chosen because of two benefits achieved over the more common four cushion suspension design. The first benefit is that the pressure will be evenly distributed throughout the inner tube and can be changed quickly and easily through one valve using a bike pump. The inner tube can be replaced by detaching the motor to shaft coupling and then inserting the tube through the hole located in the bottom plate, as seen in Figure 16. Additionally, this design is simple because neither the inner tube cushion nor the bike pump will have to be modified in order for this suspension system to be adjustable. The second benefit is that the inner tube actually helps to minimize bending deflections in the top plate. This is due to the location and the size of the contact area between the tube and the plates. With this added support, the angle iron bracing is no longer needed to minimize deflection of the tabletop. Figure 17 shows that the deflection of the tabletop with the inner tube air suspension is well below the maximum acceptable bending deflection.
Figure 17: This figure shows the tabletop displacement of the bike inner tube concept when the concrete mass is 40kg and the eccentric and acceleration forces are assuming an eccentric mass of 0.00248 kg·m spinning at 3800 rpm.

An important consideration of the inner tube suspension system is that the suspension itself provides little support in the horizontal direction. This means that the without guidance the tabletop may not have been limited to the vertical direction as desired. Stabilizers were fabricated to minimize horizontal translations and rotations. This helps to minimize unwanted movement and reduce loads on the motor shaft. For the design of the stabilizers, we chose to use 6.35 mm (0.25 in) reinforced rubber sheets bent in C shaped configurations between each adjacent corner of the top and bottom plates. These rubber stabilizers will be mounted to the steel plates by “sandwiching” the rubber between the steel plates and a metal strip. The stabilizer configuration is seen in Figure 15. There are additional benefits to the stabilizer design besides those mentioned above. First, the stabilizers add damping in the corners of the tabletop. This is desirable since the corners see the largest deflections without damping, as seen in Figure 17. Damping from the stabilizers reduces corner deflections and keeps the amplitude of the tabletop more uniform. Secondly, the stabilizers stop the top plate from tipping over if too much force is placed on one edge of the table. Without these stabilizers the top plate could possibly slide off and injure the operator. Finally, the bolts that make up the stabilizer assembly are arranged as a failsafe mechanism incase the inner tube fails. If the inner tube deflates or pops the top plate bolts limit the displacement of the tabletop by coming in contact with the bottom plate. This in turn limits the misalignment of the eccentricity shaft and the motor shaft which reduces max loads seen by the motor and the possibility of motor damage. This failsafe mechanism of the stabilizers can be seen in Figure 18. These dual uses are a large benefit to the stabilizer design, as without them additional materials and components would be needed to safely secure the top plate.
**Testing and Results:**

Extensive testing was performed on the VT4 prototype as much of the final design parameters relied on the testing results for crucial information on how to properly tune the table and find the most effective and easiest mode of adjustment. Initial testing was performed on the Cal Poly Syntron tabletop concrete vibrator located in the Civil Engineer department. Amplitudes and frequencies of the tabletop were recorded using an accelerometer and an oscilloscope. These measurements were taken to provide another reference point, on top of the ASTM standard, for how the VT4 should ideally perform. The oscilloscope and accelerometer setup displayed a waveform of the tabletops acceleration with respect to time. Frequency was calculated by dividing the numbers of cycles in the waveform by the change in time. To calculate the amplitude that the tabletop displaces, the acceleration waveform was approximated by a sinusoidal wave and then integrated twice. The displacement waveform and max displacement can be seen in equations 1 and 2 respectively, where A is the change in voltage read by the oscilloscope multiplied by the accelerometer calibration constant of \((9.81 \, \text{m/s}^2)/100.2 \, \text{mV})\) and \(f\) is the frequency in cycles per second. The Syntron's maximum amplitude ranged from 0.13 mm to 0.21 mm at a frequency of 3600 RPM, which is just outside of the ASTM standard. Higher amplitudes may have been possible by increasing the source voltage to the Syntron, but we stopped increasing the voltage because the tabletop sounded like it was bottoming out on the frame.

\[
\delta = \left(\frac{A}{(2\pi f)^2}\right) \sin 2\pi ft \quad (1)
\]

\[
\delta_{max} = \left(\frac{A}{(2\pi f)^2}\right) \quad (2)
\]
After measurements were recorded from the Syntron vibrator, with the help of Civil Engineering Masters student Benjamin Turner, we made 6 small concrete blocks using the 2 kg mold size. Each mold was vibrated for a different amount of time ranging from 0 to 60 seconds. Various vibration times were tested in order to get a better understanding of how vibration time affects the surface finish of the concrete product. Although the block that vibrated for 10 seconds was the best, all blocks that were vibrated showed to have a better surface quality than the block that was not vibrated. There was not enough evidence to determine what the optimum vibration time for concrete is, because various mold sizes where not tested to the same procedure. In general, a better understanding of concrete production and the vibration process was obtained, along with 6 good concrete block reference point made on a commercial vibrator. These 6 concrete blocks were later visually compared to blocks made on the VT4.

**VT4 Prototype Vibration Testing:**

As mentioned in the Development Test Plan section, a baseline VT4 prototype with an inner tube suspension system was fabricated. This baseline prototype included the adjustable eccentricity and vertical slots instead of holes in the motor mount plate in order easily adjust the motor shaft and eccentricity shaft alignment. These adjustable components were initially needed because the desired eccentricity size and tube height were determined through testing. Having adjustable components made testing many VT4 variables a relatively quick process, as without them new fabrications would have had to be done for each test. Initial testing performed on the VT4 prototype were done without any form of stabilizers in place. While running the tabletop, displacements remained in the vertical direction with minimal rotations and horizontal translations seen. Although the vibration process was unhindered, complications were evident while flipping the table upside down to change the eccentricity. Without restraints the tabletop was held in place by friction with the inner tube. When the vibrator was tipped on its side, friction was lost and undesirable loads were placed on the motor shaft, eccentricity shaft, and bearings due to the related misalignment of the rubber shaft coupling. Furthermore, large displacements could be seen in the corners of the tabletop as minimal damping was provided in those locations. Care was taken when changing the eccentricity and extensive vibration testing was performed since the vibration process showed to perform well without stabilizers.

All tests were performed with 6 different masses on the tabletop and the amplitudes and frequencies were calculated from oscilloscope acceleration readouts as performed on the Cal Poly Syntron vibration table. The masses used were no load, 2, 5, 15, 30 and 40 kg. For each test the motor mount plate was adjusted to allow for shaft alignment when 15 kg was placed on the table. This procedure was done to control the tests by only changing one variable at a time, eccentricity size or tube height. Table 2 shows a summary of experimentally found data from initial vibration testing performed on the VT4 prototype. Initially, we tested the effects of having an eccentricity that produced only 10 or 20% of the acceleration forces seen by the original eccentricity used on the current VT3 model. Relatively small eccentricities were capable of generating amplitudes higher and lower than the required values, but the frequencies generated were above the desired range. Additionally, during initial testing a rheostat was connected in series with the motor to test how a lower voltage drop across the motor would affect the amplitudes and frequencies of vibration. It was quickly apparent that a rheostat
adjustment would be a poor mode of adjustment for our table because too much variation occurred when different masses were placed on the table. Furthermore it was difficult to reproduce the same effect for a giving voltage drop because the motor tried to compensate by either slowing down or pulsing. It should also be noted that for these initial tests no added force or damping was added from the operator. It was later found out that CVBT operators in Thailand are instructed to push down on the molds with a trowel to increase the concrete products surface finish and aid the vibration process. Once we actively damped the system by pressing on the molds while vibrating amplitude values greatly reduced from those seen in Table 2. With this added input it was evident that larger eccentricities should be tested in order to decrease the frequency and increase the forces on the tabletop.

Table 2: Minimum and maximum frequency and amplitude recorded from the VT4 prototype for various eccentricity sizes and load tube heights with shaft alignment at 15 kg.

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<tr>
<th>Eccentricity (kg-m)</th>
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<th>Frequency (RPM)</th>
<th>Amplitude (mm)</th>
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Since larger forces were required to overcome the added feed back from the operator, three new eccentricity masses were fabricated out of a ¼ in steel strip to bolt onto the adjustable eccentricity mount. These new eccentricity configurations had eccentric values of 0.001286, 0.001521, and 0.001742 kg-m. These values ranged on either side of the eccentricity used on the VT3, which had an eccentric mass value of 0.00141 kg-m. Table 3 shows a summary of the experimental data from the new heavier eccentricities. For all cases, a noticeable reduction in the frequencies was seen. In most cases the frequencies fell within the proper range of the requirement. The heaviest eccentricity was tested first at a no load tube height of 46.8 mm. This test showed relatively huge amplitudes of up to 1.15 mm and the size was determined to be too large for the VT4. The medium size eccentricity was tested next and a similar outcome was seen, with maximum amplitudes well outside of the amplitude requirement range. The smallest new eccentricity was then tested at tube height of 30.5, 42.5, and 46.8 mm. From these tests it was determined that the inner tube at maximum height of 46.8 mm with the eccentricity of 0.001286 kg-m was the best combination of design variables tested. This choice was made because the frequency and amplitude had the tightest range and were the closest to the design requirements.
Table 3: Minimum and maximum frequency and amplitude recorded from the VT4 prototype for various eccentricity sizes and load tube heights with shaft alignment at 15 kg.

<table>
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<tr>
<th>Eccentricity (kg-m)</th>
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Once the optimum tube height and eccentricity were chosen, we looked to further reduce the maximum amplitudes seen in order to get the whole amplitude range within the requirement. Since the maximum amplitudes were seen when the smallest mass was on the table, we realigned the motor shaft 2 mm below the previous alignment. This new alignment causes the motor to see a higher torque for small masses and a lower torque for larger masses. Both of these effects are desirable as the shaft speed slows down for masses with maximum amplitudes therefore reducing their amplitude and speeds up for masses with low amplitudes. The range of amplitudes for the new shaft alignment was reduced to 0.30 to 0.50 mm.

As mentioned earlier, larger displacements were seen in the corners of the tabletop because minimal damping was applied in locations far from the inner tube contact area. To lower the corner displacements, fiber-reinforced rubber stabilizers were fabricated and installed between the corners of the top and the bottom plate as seen in Figure 18. The stabilizers were designed to both provide damping in the corners and to restrict in-plane rotations and horizontal translations of the top plate. Once installed, the top plate motion was completely confined to the vertical direction and the instantaneous amplitudes across the tabletop were more inform. With the stabilizers in place and all previous variables held constant, the amplitude range tightened to 0.32 (for larger concrete loads) to 0.44 mm (for smaller concrete loads) and an average frequency of approximately 4000 RPM. It should be noted that this new amplitude range falls very close to the requirement of 0.3 to 0.4 mm with no additionally tuning or variable changes necessary for different mold sizes.
In order to completely satisfy the amplitude requirement, the final variability option mentioned in the Design Development section was tested. This final variability option was the Damping Mass. Since amplitudes that fell outside of the requirement range occurred in test with mold sizes less than 30 kg, we determined that a 10 kg addition mass should be added to the table when 20 kg or less was being vibrated. To keep the tabletop surface free and to make sure the added mass is balanced, we designed the additional mass to hang from the bottom of the vibration box assembly. This location also reduces work for the operator as the mass can be stored under the table when not in use and a minimal lifting distance is required. Figure 19 shows an image of how the mass is hung from the vibration box. The results from testing showed the final amplitude range to fall within the design requirement with a minimum of 0.32 mm and a maximum of 0.39 mm along with a frequency range between 3860 and 4020 RPM. Figure 20 shows the amplitude and frequency for each mold tested with all final design parameters set.
Endurance Testing:

With all design variables chosen for the final design, endurance testing was performed on VT4. For the endurance testing, 20 kg was strapped on the tabletop with the additional 10 kg Damping Mass hanging from the vibration box. This setup was chosen because it is the maximum total design mass in which the Damping Mass is used. We wanted to include the Damping Mass in the endurance test so that wear on the inner tube and wear on the cord holding the kit could both be inspected. Additionally, prior to vibration testing the diameters of the inside bearing race and the eccentricity shaft were measured so that wear between the two surfaces could be quantified. Measurements for the inner race and shaft diameters at each bearing location were 12.70 mm (0.500 in) and 12.675 to 12.70 mm (0.499 to 0.500 in) respectively. Even though both bearings had to be pressed on, it should be noted that the 12.70 mm end of the shaft fell within the bearing manufactures recommended tolerance of 12.70±0.004 mm, while the 12.675 mm end did not. Furthermore, prior to testing, all fasteners were installed without the use of Loc-tite. This was done to get an understanding of which fasteners were in jeopardy of loosening overtime due to vibrations generated by the tabletop.

The motor was cycled on and off during endurance testing to make sure that the motor did not overheat and burn out. To come up with an appropriate cycle, the vibrator was turned on and motor temperature was monitored using a thermocouple. After approximately 15 minutes the motor reached a steady state temperature of 118°F. Even though this steady state temperature did not seem high enough to damage the motor, we chose to cycle the vibrator on and off every 15 minutes as a safety precaution. With the endurance cycle chosen, the endurance testing proceeded for 48 hours total. Therefore, the VT4 was actually vibrating for a total of 24 hours.

Once endurance testing was completed all fasteners were checked for tightness. None of the fasteners showed significant signs of loosing, but Loc-tite or lock washers are still recommended on all fasteners for the safety of the operators and the longevity of the vibrator. The VT4 was then disassembled and wear was checked on the eccentricity shaft, inner bearing race, Damping Mass cord, and inner tube contact surfaces. The diameter of the inner races of both bearings and the shaft end within tolerance were the same as pre-endurance testing, but the shaft end that started out of tolerance showed significant wear with a new diameter measurement of 12.62 mm (0.497 in). This wear on the eccentricity was due to the shaft breaking free from the press fit and spinning inside of the harder inner race of the bearing throughout testing. Press fits into the bearing seats were both within manufactures tolerances and remained intact throughout the entire endurance testing. Endurance testing confirmed that the manufactures tolerances for both the bearing seats and the shaft diameter are the proper tolerances for the VT4. The Damping Mass cord was than visually inspected for wear and no damage was seen. Next the rubber stabilizers were disassembled and inspected. No damage to the fiber-reinforce rubber was found. Finally, the inner tube was visually inspected for wear and only superficial surface scratches were seen on contact surfaces. If significant wear were seen the inner tube would have been cut open and the thickness of the tube would be measured at the contact surfaces and compared to the thickness of the tube in places were no contact was seen. Overall, endurance tests results showed that the VT4 is capable of operating for extended periods of time without significant damage. Since endurance testing was only performed for 24 hours of vibration time, it is recommended
that operators follow the maintenance log sheet provided in Appendix I in order to maximize the safety of the operator and the life span of the VT4.

**Final Design:**

![Figure 21: VT4 Final Design.](image)

The final design for the VT4 can be seen in Figure 21. As shown in Figure 21 the final design includes an inner tube suspension system with fiber-reinforced rubber stabilizers and a Damping Mass to control amplitudes for mold sizes less than 20 kg. All design requirements were achieved. As desired, the VT4 uses the existing frame and motor from the VT3 and has incorporated a full suspension system. The suspension system allows concrete to be vibrated within the wooden frames and therefore maintains the integrity of the molds. Additionally, the total cost was $153 which is $100 dollars below budget. A cost list of assemblies and parts can be found Appendix J. Finally, with a single Damping Mass of 10 kg, the VT4 outputs amplitudes between 0.3 and 0.4 mm and frequencies between 3850 and 4050 RPM for all mold sizes between 2 and 40 kg. Part drawings for all components were made using AutoDesk Inventor and can be found in Appendix K. The Operator’s Manual which details the assembly, safety concerns, and recommended operating procedure of the VT4 is located in Appendix L.
Appendices:

Appendix A: Gantt Chart

Appendix B: QFD House of Quality

Appendix C: Existing Products

Appendix D: Hand Calculations

Appendix E: Four Spring Simulink Model

Appendix F: Translational Spring Constant Matlab script

Appendix G: Tokushu Denso Motor Curves

Appendix H: Prototype Testing Pictures

Appendix I: Maintenance Log Sheet

Appendix J: Bill of Materials

Appendix K: VT4 Part Drawings

Appendix L: Operator’s Manual
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**Gantt Chart**
QFD House of Quality

See .pdf Attachment
Existing Products

Figure 22: Martin Electric Vertical Spring Suspension Vibration Table
(www.shake-it.com/compaction-tables.shtml Model VT 4/7)

Figure 23: Patent Using Solid Rubber Mount as Pivot Point (www.freepatentsonline.com/4483621.pdf)

Figure 24: CWC Hanging Tabletop With Horizontal Motion (www.cwc.org/gl_bp/gbp3-0403.htm)

Figure 25: Syntron Vibration Table with rubber shear mounts (located at Cal Poly)
Hand Calculations:
See .pdf Attachment
Four Spring Simulink Model

Thomas Bevan and Matthew Laurino

clear;
clc;

options = odeset('RelTol',1e-10);

%constants
k=410.4*1000;  %spring constant (N/m)
c=5;           %damper constant (N*sec/m)
Mt=31/4;       %total mass of table (kg)
M=.23;        %mass of rotating object including eccentricity (kg)
m=.001;       %mass of eccentricity (kg)
e=.01;         %radius of eccentricity (m)
me=.00346/4;  %eccentricity * mass (kg*m)

w=1800*2*pi()/60;  %shaft speed (rad/sec)

%Simulink Model Run
sim('fourCorner');

%Plot
% subplot (3, 1, 1);
% plot(tout, Displacement,'LineWidth', 2);
% grid on;
% title('Position of table');
% xlabel('Time (sec)');
% ylabel('Displacement (m)');
% axis([0 1 -.0015 .0015]);
% subplot (3, 1, 2);
% plot(tout, Velocity,'LineWidth', 2);
% grid on;
% title('Velocity of table');
% xlabel('Time (sec)');
% ylabel('Velocity (m/s)');
% axis([0 1 -1 1]);
% subplot (3, 1, 3);
% plot(tout, Acceleration,'LineWidth', 2);
% grid on;
% title('Acceleration of table');
% xlabel('Time (sec)');
% ylabel('Acceleration (m/s^2)');
% axis([0 1 -500 500]);
% Plot 2
plotyy(tout,input,tout,Displacement)
grid on;
%title('Position of table');
%ylabel1('Input Force (N)')
%ylabel2('Table Displacement (m)')
xlabel('Time (sec)');
ylabel('Input Force (N)');
Translational Spring Constant Matlab script

clc

clear all

syms Q theta gg
load=60*9.81;
pi=3.1415;
%gamma=pi/180*15
d=.312*.0254; %wire diameter (m)
A=pi*d^2/4;
R=1.312/2*.0254; %spring radius (m)
E=200*10^9; %modulus of elasticity (pa)
v=.3; %poisson's ratio
G=E/(2*(1+v)); %shear modulus ()
Ix=pi*d^4/64;
J=pi*d^4/32;
%Q=255.4/4; %transverse force on spring (N)
%theta=2*pi;

Lo=2.5*.0254; %free length (m)
Ls=1.95*.0254; %solid length (m)
Nt=Ls/d; %total number of coils
Na=Nt-2; %active number of coils

p=(Lo-2*d)/Na; %pitch

gamma=atan(p/2/(2*R));

Fx1=-Q*cos(theta); %radial shear force
Fk11=Q*sin(theta)*cos(gamma); %normal (axial) force
Fn=-Q*sin(theta)*sin(gamma); %shear force

Mx1=Q*R*theta*sin(theta)*tan(gamma); %bending moment about x1
Mn=Q*R*sin(gamma)*(theta*cos(theta)-sin(theta)); %torsional moment
Mk11=Q*R*theta*cos(theta)*tan(gamma)*sin(gamma)-Q*R*sin(theta)*cos(gamma); %bending moment about k

%Partials of forces and moments with respect to Q

dFndQ=diff(Fn,Q);
dFk11dQ=diff(Fk11,Q);
dFx1dQ=diff(Fx1,Q);
dMndQ=diff(Mn,Q);
dMex1dQ=diff(Mx1,Q);
dMk11dQ=diff(Mk11,Q);

dAxial=Fn*dFndQ*R/cos(gamma);
dShear1=Fk11*dFk11dQ*R/cos(gamma);
dShear2=Fx1*dFx1dQ*R/cos(gamma);
dTorsion=Mn*dMndQ*R/cos(gamma);
dBendl=Mx1*dMex1dQ*R/cos(gamma);
dBend2=Mk11*dMk11dQ*R/cos(gamma);

%Transverse deflection (m)

delta=2/(A*E)*int(dAxial,theta,0,pi)+12/(5*A*G)*int(dShear1,theta,0,pi)... +12/(5*A*G)*int(dShear2,theta,0,pi)+2/(J*G)*int(dTorsion,theta,0,pi)... +2/(E*Ix)*int(dBendl,theta,0,pi)+2/(E*Ix)*int(dBend2,theta,0,pi);
\%subs(delta,\{Q,\theta\},(255.4/4,2\pi*N));

K=Q/delta; \hspace{1cm} \text{\textit{Stiffness for single coil}}
Keff=K/Na; \hspace{1cm} \text{\textit{Effective Stiffness for N coils (N/m)}}
Ktransverse=(subs(Keff,\{Q,\theta\},(255.4/4,2\pi))/1000)
\%\text{\textit{simplify (Keff)}}
\%Q=255.4/4; \hspace{1cm} \text{\textit{transverse force on spring (N)}}
xdeflect=Q/Keff;
Xdeflect=(subs(xdeflect,\{Q,\theta\},(255.4/4,2\pi)*1000) \hspace{1cm} \text{\textit{transverse deflection (mm)}}

Kaxial=d^4*G/(8*(2*R)^3*Na)/1000 \hspace{1cm} \text{\textit{axial spring constant (N/mm)}}
ydeflect=(load)/Kaxial;
Ydeflect=(subs(ydeflect,\{Q,\theta\},(255.4/4,2\pi))) \hspace{1cm} \text{\textit{preload + eccentricity}}
axial spring deflection (mm)

MaxAllowDeflect=(Lo-Ls)*1000 \hspace{1cm} \text{\textit{Max allowable axial spring deflection (mm)}}

\%\text{\textit{Critical Spring Frequency}}
spw=76.5*1000; \hspace{1cm} \text{\textit{specific weight (N/m^3)}}
W=pi^2*d^2*2*R*Na*spw/4; \hspace{1cm} \text{\textit{weight (kg)}}
freq=.5*(Kaxial*1000*9.81/W)^.5*60 \hspace{1cm} \text{\textit{critical frequency (RPM)}}
Tokushu Denso Motor Curves

Figure 26: Tokushu Denso 2570-333 motor curves.
Prototype Testing Pictures
## Maintenance Log Sheet

<table>
<thead>
<tr>
<th>Date</th>
<th>Inner Tube (Every 2 weeks): Check for excessive wear at contact surfaces and tube height is 40 to 45mm with only unloaded tabletop</th>
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<tr>
<td></td>
<td>Hose Connector (Every 2 weeks): Ensure clamps are tight</td>
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<td>Bearing-Motor Side (Every 2 weeks): Ensure inner race rotates with shaft, and bearing does not rotate in seat</td>
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<td>Bearing-Far Side (Every 2 weeks): Ensure inner race rotates with shaft, and bearing does not rotate in seat</td>
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<td>Motor (Every 2 weeks): Look/listen for irregularities with motor running, and with motor off, check shaft rotates freely</td>
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<td>Eccentricity Assembly (Every 2 weeks): Make sure weight is centered and perpendicular to shaft and nuts are tight</td>
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<td>Rubber Stabilizers (Every 2 months): Check for large stress cracks. Ensure horizontal tabletop movement is minimal</td>
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<td>Electric Wiring (Every 2 months): Ensure insulation and connections are intact. Check that wire is properly restrained</td>
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<td>Fasteners (Every 2 weeks): Ensure motor mount, stabilizers, and vibration box fasteners are tight</td>
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### Key:
- Good
- Bad
- Fixed

### Remarks/Measurements:
- Cost:
<table>
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<tr>
<th>Part No.</th>
<th>Description</th>
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<td>Encoder</td>
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<td>Motor</td>
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<td>Controller</td>
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**Bill of Materials**
VT4 Part Drawings
See Attachment
Operator’s Manual
See Attachment