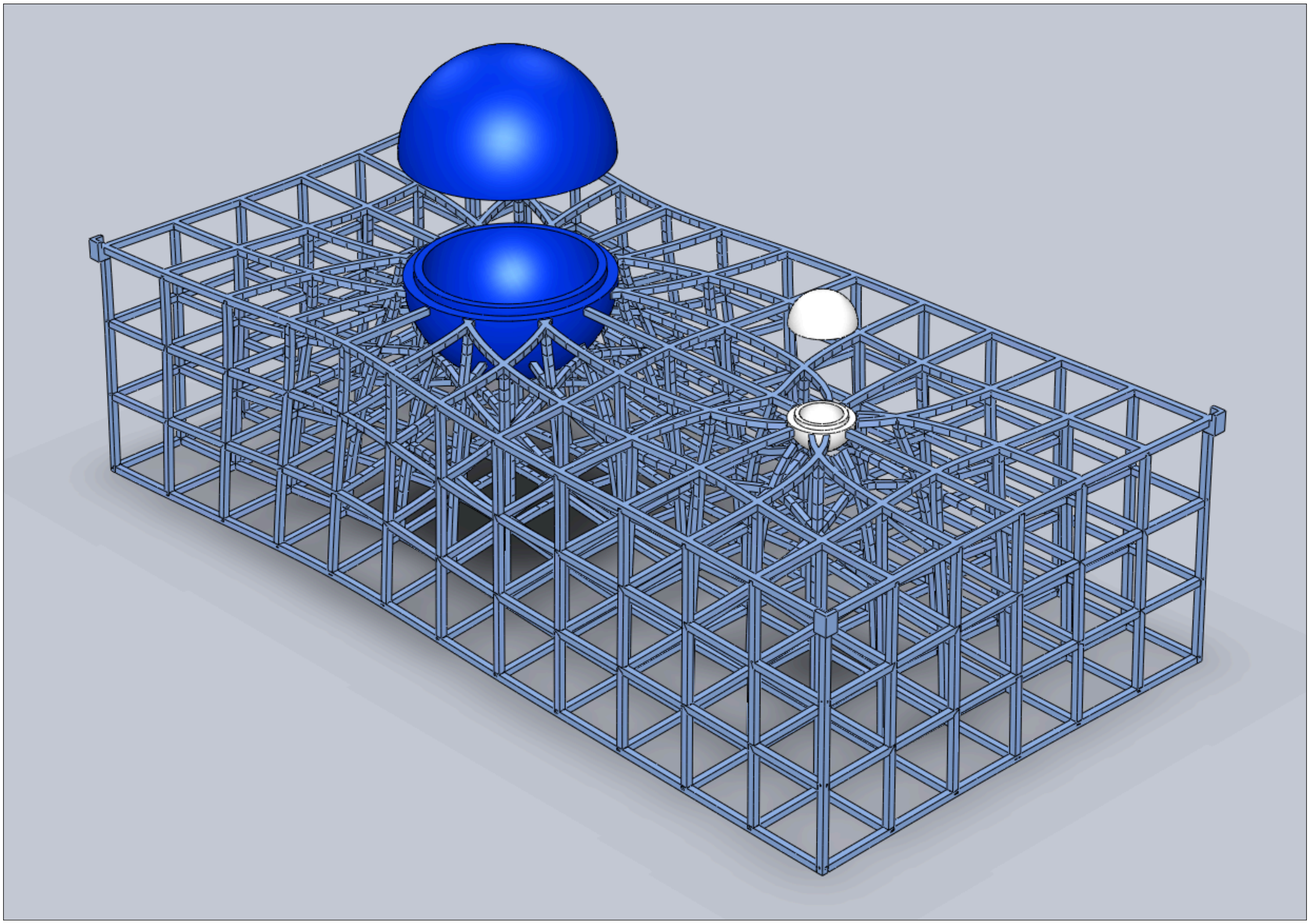
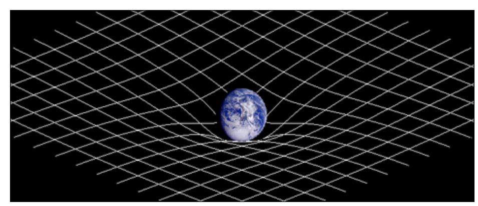
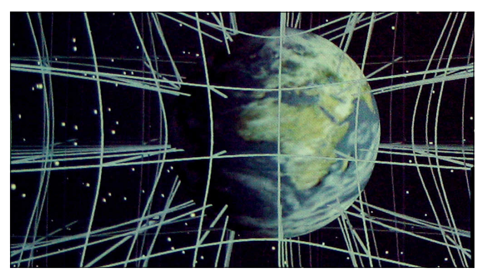


Design and Fabrication of an Educational Model to Illustrate the Three-Dimensional Curvature of Spacetime

In which an interactive model with complex geometry is designed with programming-based design tools and built using Selective Laser Sintering.





The "rubber sheet" model (top) is a useful analogy to illustrate how the fabric of spacetime is warped by large bodies like the Earth. However, because the model is only two-dimensional, it cannot capture the actual shape of the warping, which is better depicted in the second illustration.

Additive manufacturing technology provides inspiring opportunities to design products and structures whose geometries would be impossible or infeasible to create using traditional manufacturing methods. Coding and mathematics can become tools to enable new possibilities that are both beautiful and functional.

Improving the rubber sheet model

Einstein's theory of general relativity holds that large bodies — like the Earth and Moon — warp the fabric of the space surrounding them, much like how a heavy bowling ball stretches a tight rubber sheet when it is placed upon it. Conceptually, this model is a popular and useful analogy; however, in describing the visual reality of the distortion, it falls short, as it is fundamentally limited to describing space in only two dimensions.

Using a combination of programming and traditional CAD modeling, a complex interactive model of planetary gravity was designed and fabricated in Selective Laser Sintering (SLS), creating a tangible visualization of the warping of spacetime, with which students can interact and develop their understanding of a three-dimensional concept that is difficult to visualize in two dimensions.

Requirements and constraints

The model was to be created with a SinterStation SLS machine, using Nylon-11 powder as the build material.

Designing the model

The design began with the familiar concept of a three-dimensional grid with regular, equally-spaced divisions in its undistorted state. The outer dimensions were chosen at 12″ × 6″ × 6″, sized to fit conveniently on a bookshelf or a classroom table but also respecting the constraints imposed by the SinterStation build chamber.

Two bodies were chosen to be placed within the lattice: a larger body to represent the Earth and a smaller body to represent the Moon. The mass difference between these two bodies (the Earth having four times the mass of the Moon) allows the model to depict different degrees of distortion due to masses of different sizes.

Generating the lattice

Starting with these initial concepts, a custom desktop application was coded to help establish the specific geometry of the grid. Using Einstein's field equations to calculate the displacement of points within the lattice, a simple user interface allowed several variables to be interactively manipulated:

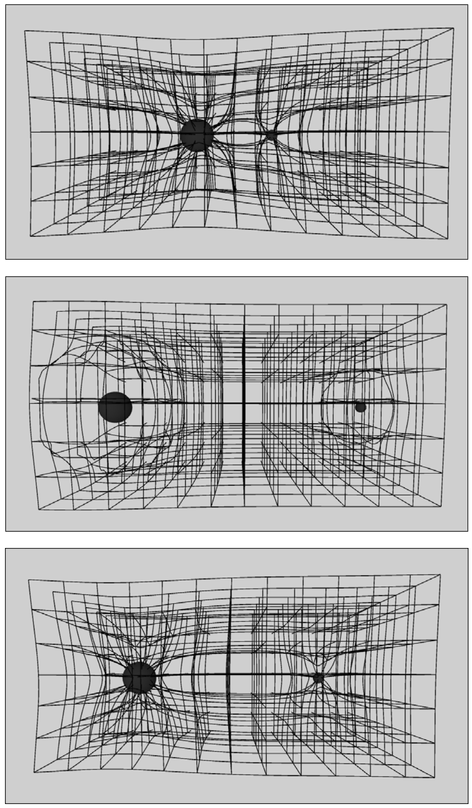
- the bodies' spacing within the grid,
- the bodies' masses (but keeping the 4:1 ratio constant), and
- the strength of the gravitational constant.

Once these design parameters were established, the program generated a series of 3D coordinates representing the 231 lines of the grid. These coordinates were grouped and formatted into line definitions in a DXF file, which was imported into SolidWorks. The lines were then used as guide paths for sweeps to build the grid's three-dimensional form.

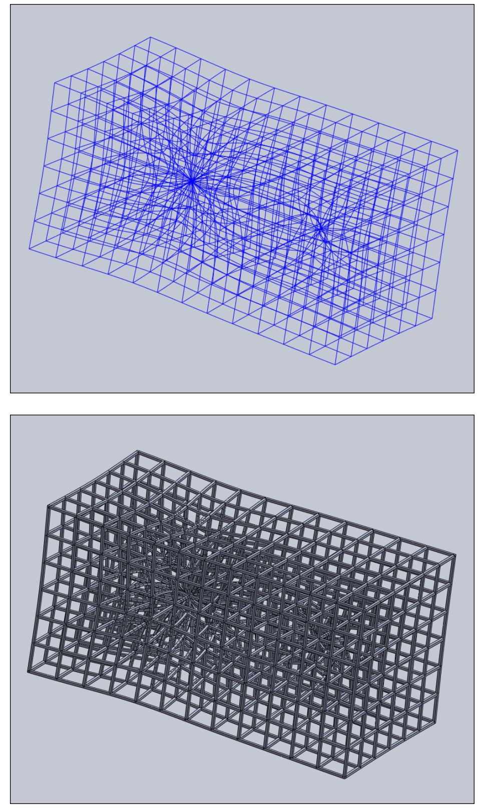
Refining the design

After building the model in SolidWorks, several parameters were modified to improve the clarity of the design:

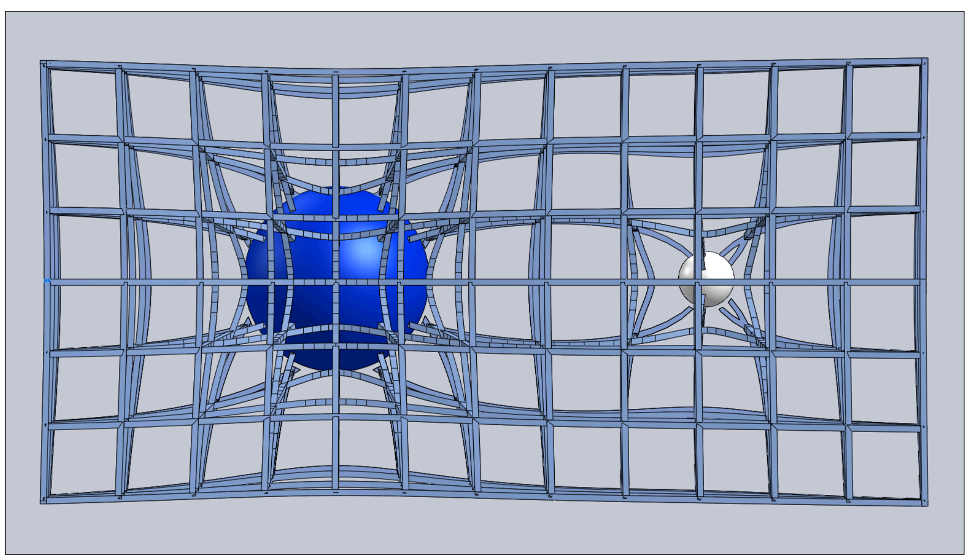
- The thickness of the grid lines was reduced from ⅛″ to 1⁄12″.
- The diameters of the planets were increased to enhance their visibility within the grid.



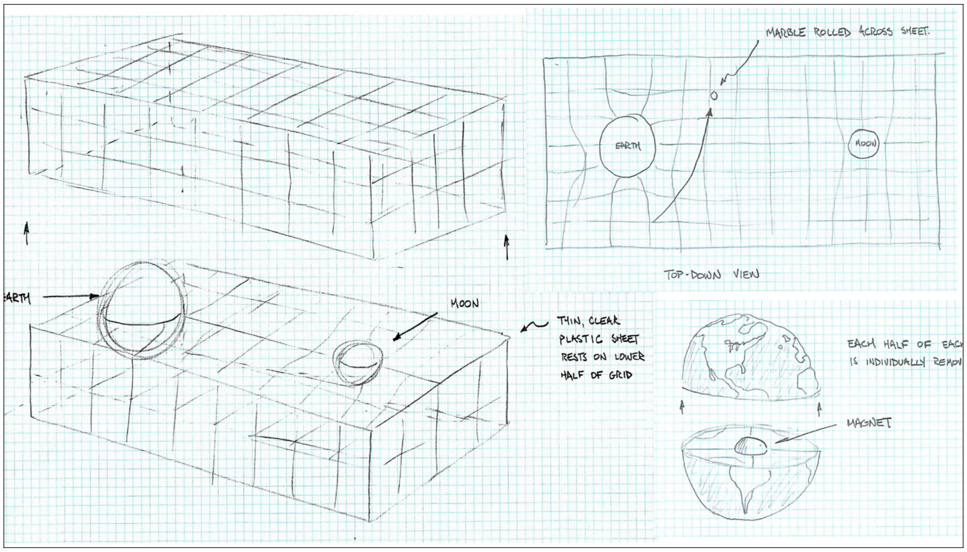
A custom desktop application was developed to establish the most important visual design parameters, such as the spacing between the planets and the magnitude of the distortion due to gravity. These values could be experimented with and visually tested before committing to the more labor-intensive task of building the grid in SolidWorks.



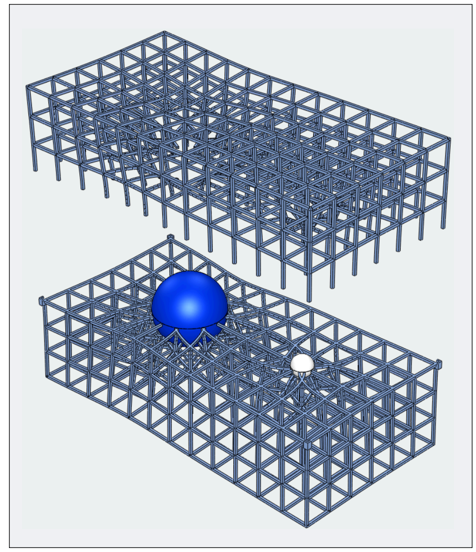
Coordinates from the gravitation application were exported into SolidWorks as a series of lines, which became the "scaffolding" on which to build the realized three-dimensional structure, using sweeps and other CAD primitives.



After the initial grid was realized in a three-dimensional form in SolidWorks, it was possible to make further refinements to the design while preserving its basic structure.



Concept sketches illustrate how the model simulates the planets' gravitational fields with embedded magnets. The top-down view (upper right) illustrates how the Earth's magnetic field curves a magnetic marble's path as it rolls across the lower half of the grid (left).



The final form of the model is depicted with the upper half of the grid raised above the lower half.

Making the model interactive

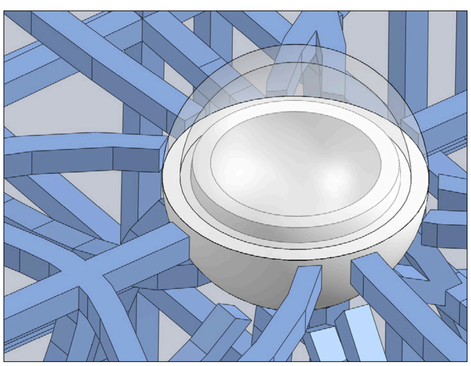
A key idea improved the model beyond its original conception as a static visual piece.

It was recognized that the relationship describing the strength of the gravitational field between two bodies — with force proportional to $1 / \text{distance}^2$ — also governs magnetism. A design was developed to capitalize on this insight, simulating the planets' gravitational fields and creating an element of interactivity.

First, each of the planets was made hollow. After fabrication, magnets were embedded inside the Earth and Moon, in proportions approximating the relative strengths of the planets' gravitational fields.

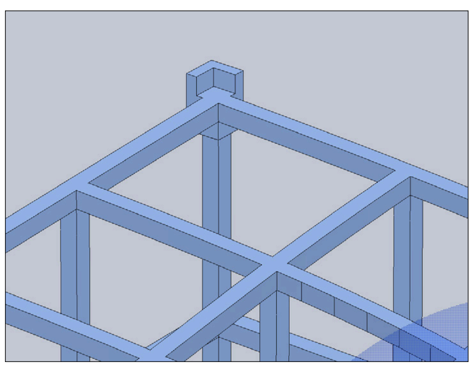
The grid was divided just above its central horizontal plane, allowing the upper half of the grid to be lifted off and separated from its lower half. An acrylic plastic sheet was machined to fit on top of the lower grid, giving the lower half of the grid a transparent horizontal surface.

As a user rolls a magnetic marble across this acrylic sheet, its path is curved as it traverses the planets' magnetic fields. By experimenting with the marble's speed and direction, students can develop an understanding of how gravitational fields act in space, actively investigating and learning from the feedback provided by the model.



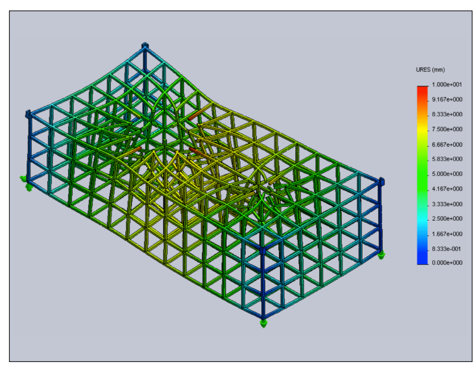
DFM: Curling and tolerancing

As with traditional manufacturing methods, DFM concerns must be addressed in Selective Laser Sintering. Temperature variations in the powder bed can cause deformations ("curling") during the sintering process, especially at the boundaries of the parts' geometry. Tolerances must be built into the design to allow for the parts' expansion.



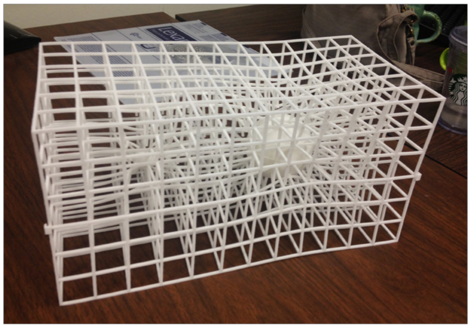
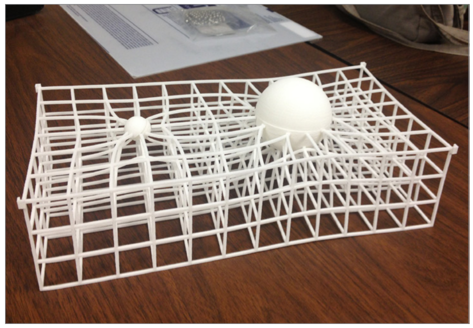
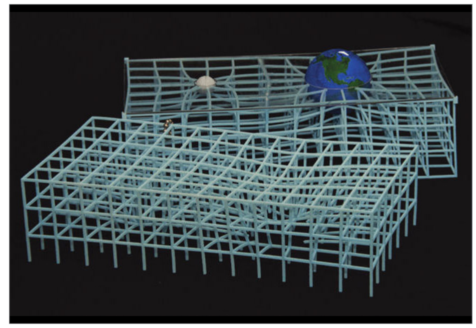
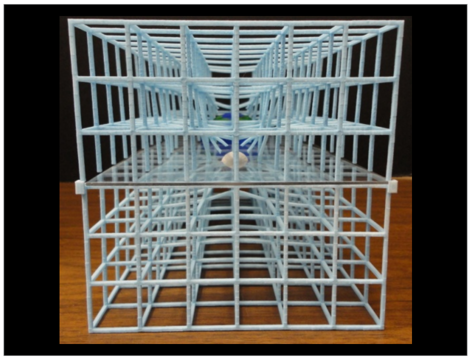
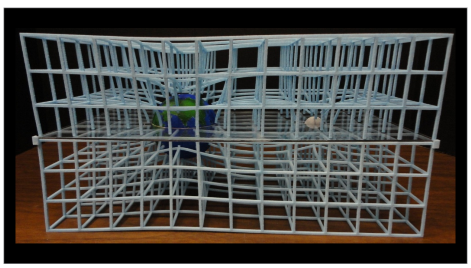
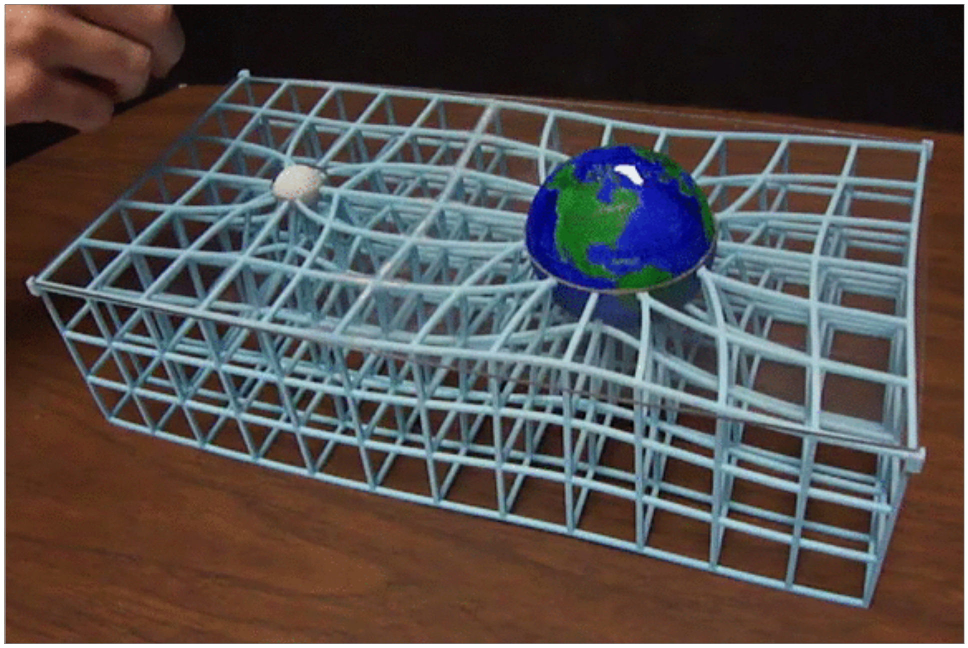
Corner brackets

Corner brackets were designed to be aesthetically minimal but structurally adequate to keep the grid halves together and the acrylic sheet in place while in use.



Failure mode effects and analysis (FMEA)

Under all reasonable usage scenarios, the model should not expect to experience any significant stresses. However, several failure scenarios were considered, including accidental droppage and weights (such as books) being placed on the model. Finite element analysis (FEA) was performed to determine the maximum allowable loads that the model could sustain.


Post-processing and finishing

After the SLS build was complete, several post-processing steps were performed to finish the model.

- The grid was dyed.
- A 1/16″ acrylic sheet was machined to fit the upper plane of the lower half of the grid.
- The planets were hand-painted to resemble the Earth and the Moon.
- Magnets were placed into each of the planets, and user testing was performed to adjust the relative strengths of the magnetic fields.

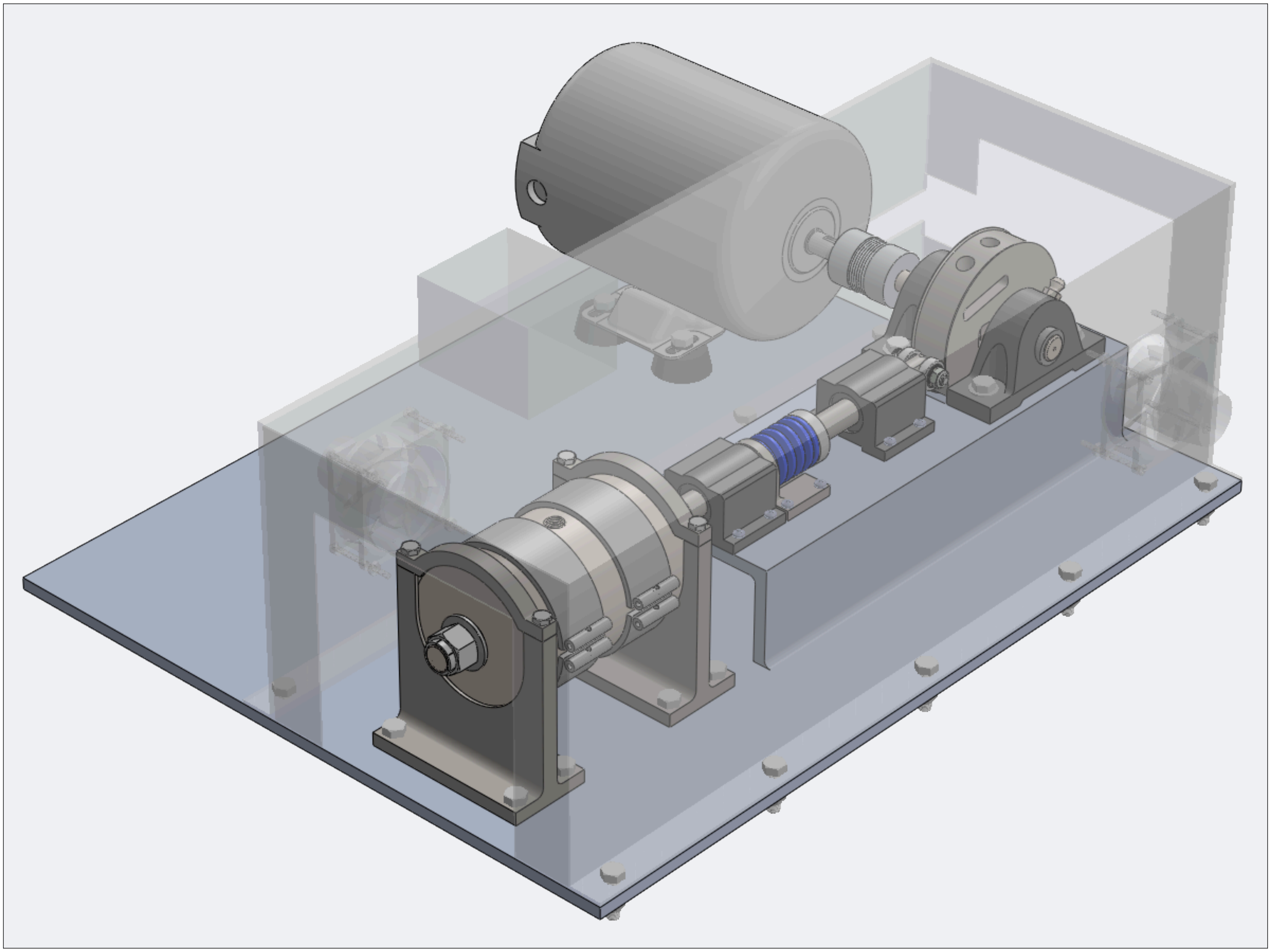
Recognition

- Featured in a TED Talk by Dr. Carolyn Seepersad at TEDx Jamaica 2013.
- Featured in the academic journal article “Challenges and Opportunities in Design for Additive Manufacturing,” *3D Printing and Additive Manufacturing*, March 2014.

Above (smaller photos): Photographs show the model from various angles and in varying states of finish.

Middle (larger photo): A user rolls a magnetic marble across the acrylic sheet.

Below: Dr. Carolyn Seepersad features the spacetime model as an example of the complex geometric designs possible with computer-based tools and additive manufacturing, at TEDx Jamaica 2013.



Design of a Fixture to Test Semi-Dynamic O-Ring Seals in a High-Temperature, High-Pressure, Abrasive Environment

To protect the trade secrets and intellectual property of the Client who commissioned this project, sensitive identifying details have been deliberately obscured or altered. The essential character of the problem, the design approach and solutions have been preserved.

O-ring performance is critical

In a specific industrial setting, the Client operates production equipment in extreme environmental conditions. A high-pressure, high-temperature, water-based fluid must be kept from entering the interior of the production machinery, where sensitive electronic components must remain at conditions near standard atmospheric pressure.

To keep this fluid out, a set of semi-flexible, elastomeric O-ring seals is installed on a vibrating piston, which interfaces with a cylinder at the machine's interior-exterior boundary. The O-rings compress at this interface, sealing the gap between the piston and cylinder and keeping the fluid out.

Failure costs millions

When these O-ring seals fail, production must be halted, equipment must be repaired or replaced, and millions of dollars are usually lost in personnel costs and downtime. However, if seals are replaced more often than needed, production time is also lost unnecessarily.

A set of small, elastomeric O-ring seals in the Client's production equipment protects critical components from the high-pressure, high-temperature fluid present in the external environment.

Extreme conditions limit O-ring life

The O-rings' effective lifetime is known to be impacted by several factors:

- temperature and pressure;
- friction and abrasion due to particulate matter in the fluid; and
- the frequency and amplitude of the vibration experienced at the piston-cylinder interface.

Predicting time-to-failure is difficult

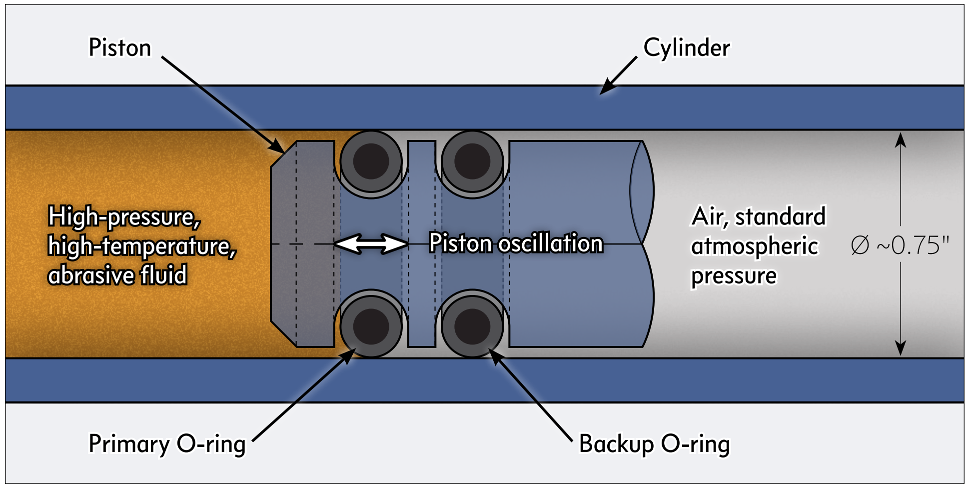
While the general effects of pressure, temperature, friction, and vibration are known, they are not quantified in enough detail to make useful predictions of the O-rings' time-to-failure over the full range of conditions experienced in production. Furthermore, interaction effects — if any — are not well-documented or understood. Analytical and computational methods such as Finite Element Analysis (FEA) have not yet been able to predict time-to-failure with acceptable accuracy.

Accurate prediction is valuable

If accurate predictions of the O-rings' time-to-failure can be made, production time can be improved, and downtime and failure can be reduced, providing the Client millions of dollars of value. However, production decisions must be supported by a relevant body of data corresponding to the environmental and vibrational conditions in production.

Realistic simulation yields valid data

To develop a valid body of data on the O-rings' performance, a test fixture was designed to reproduce the combined effects of pressure, temperature, friction, and vibration over the full range of conditions experienced in the production environment. This test data will enable the Client to make more accurate predictions of the O-rings' time-to-failure and maximize production time.



The schematic diagram above illustrates the piston-cylinder geometry as replicated in the test fixture. The O-rings installed on the piston must keep the high-pressure, high-temperature fluid from entering the interior cylinder chamber, which must remain at atmospheric pressure and cannot be exposed to the external environment. The piston oscillates at a fixed frequency and amplitude continuously for up to 150 hours.

Fixture requirements and constraints

The Client required the fixture to meet the following minimum performance standards, as well as outlining target values to be attained, if possible:

Requirement	Minimum	*Desired*
Pressure	0–30 ksi	*45 ksi*
Temperature	20–200 °C	–
Frequency	0–20 Hz	*40 Hz*
Amplitude *	0.00–0.05″	*0.10″*
Test duration	150 hours	–

* Amplitude is peak-to-peak.

Additionally, several qualitative guidelines were specified.

- The *geometry and surface finish* of the piston-cylinder interface must be replicated exactly as it exists in production.
- The fixture must *interface with the Client's existing test facilities*, accommodating the autoclave equipment that supplies the pressurized fluid.
- Appropriate *safety factors* must be researched, documented, and incorporated into the design.
- The design must be *realistically achievable*; however, cost and power consumption are not specifically constrained.

Design approach

The complete target range of performance requirements was ultimately accomplished in a design having two distinct subsystems:

- a *piston-cylinder system* and
- a *drive system*.

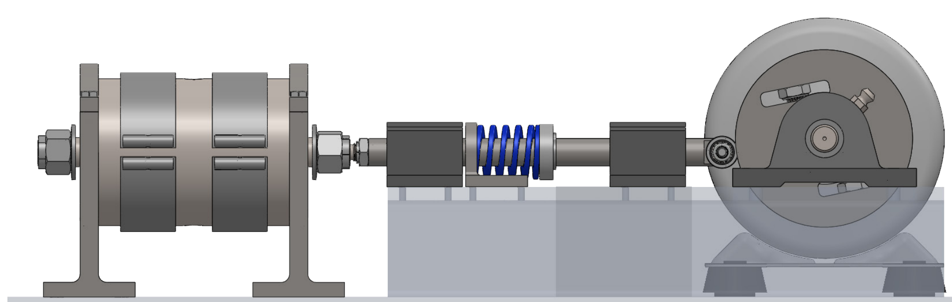
To maximize development resources, much of the detailed design work for each subsystem was performed independently, assuming a rigid coupling between the two subsystems at the interface between the piston and drive mechanism.

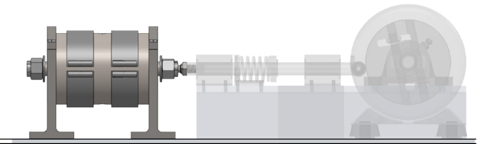
The piston-cylinder system:

- replicates the production geometry,
- accommodates the high-pressure testing fluid, and
- generates and regulates the system temperature.

The drive system:

- applies input force to push and pull the piston in a linear sinusoidal cycle,
- over the required range of frequencies
- and amplitudes.





Piston-cylinder system

Design of the piston-cylinder system followed from the primary requirement to replicate the geometry of the production equipment at the O-ring interface, with high-pressure fluid (up to 45,000 psi) on one side of the seals and air at atmospheric pressure on the other.

Pressure forces and displacement challenges

In the simplest configuration satisfying the geometric requirements (*left diagram*), the piston abuts against a static pocket containing the pressurized liquid. At maximum pressure, this pocket exerts nearly 20,000 pounds of force against the end face of the piston, which the drive system would have to overcome in order to produce any motion. With a single test requiring up to 40 strokes per second, continuously for 150 hours, the mechanical and power consequences of this configuration are problematic.

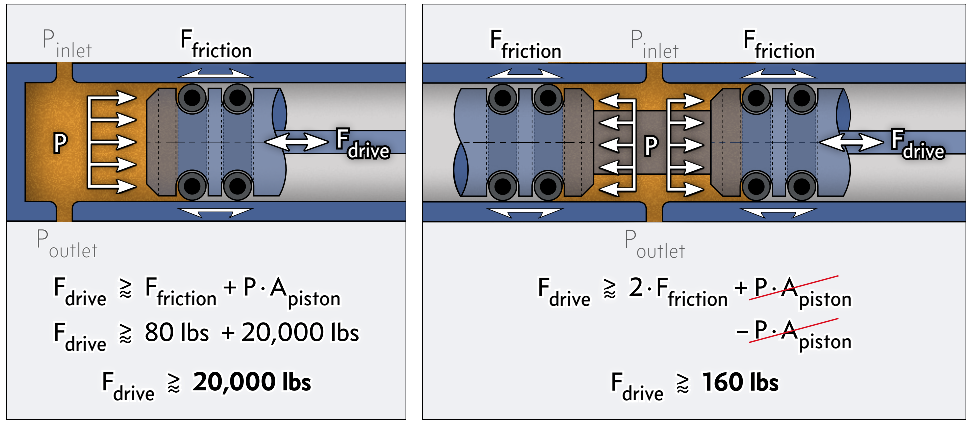
Additionally, the test fluid is a liquid and essentially incompressible. For the piston to move against it at all, a system would have to be devised to allow the shape of the fluid pocket to change dynamically, accommodating the piston's movement while maintaining its volume and pressure.

Solution

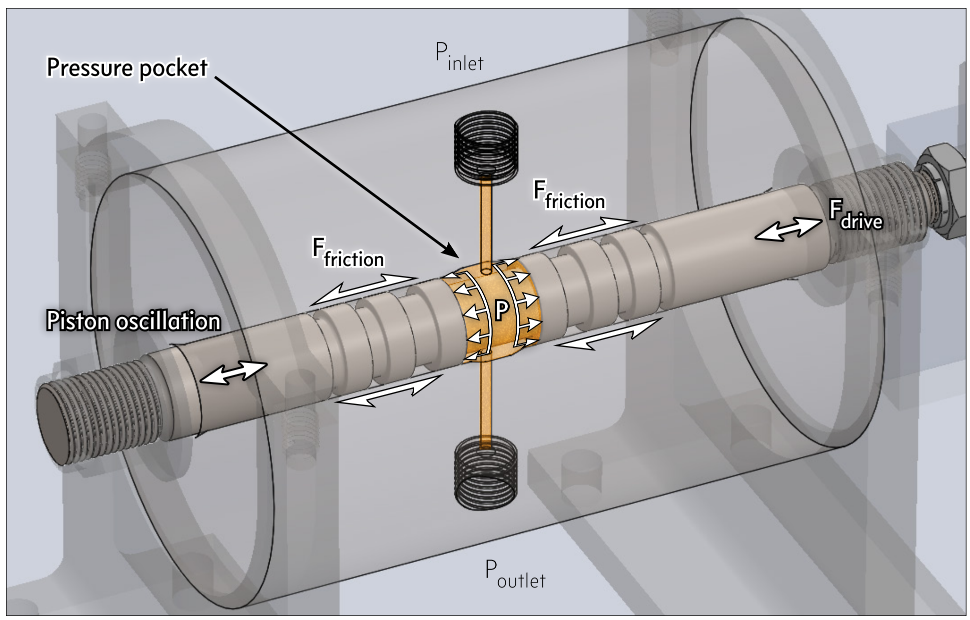
A geometric solution overcomes both of these challenges. By mirroring the piston geometry (*right diagram*), an internal "pressure pocket" is created, containing the pressurized fluid and moving with the piston as it is pushed and pulled back and forth.

This geometry balances the forces exerted by the pressurized fluid against the piston's internal surfaces. Thus, the required drive force is reduced by a factor of 100, only needing to overcome the O-ring friction (estimated at 80 pounds per stroke, per set, factoring in squeeze and the pressure differential).

This arrangement has the added benefit of testing two sets of O-rings per test. Should one set fail before a test runs to completion, the Client will still have valid data for the second set through the moment that the first set fails.



In the simplest geometric configuration (left), the force required to drive the piston approaches 20,000 pounds per stroke. However, by mirroring the piston geometry (right), a pressurized pocket is created that moves with the piston and balances the pressure forces against the piston's internal surfaces. This configuration reduces the required drive force by over 100×.

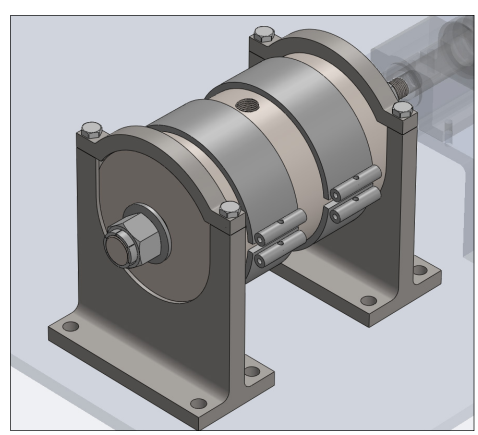


The realized piston-cylinder design is illustrated and annotated with the relevant forces.

Temperature generation and control

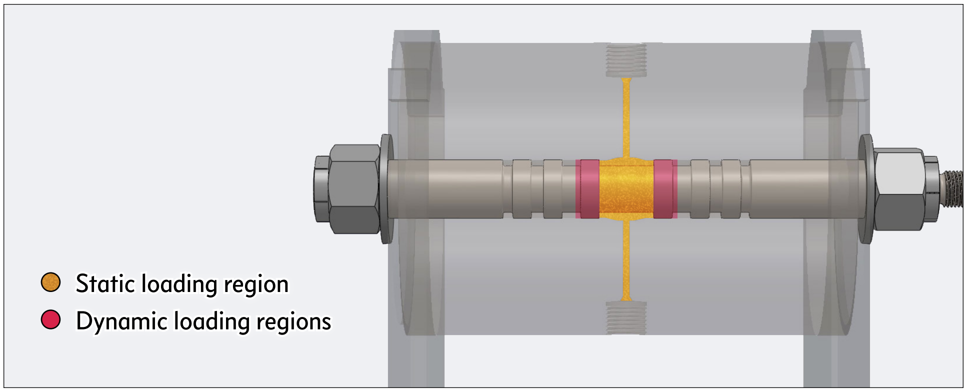
Constant-temperature electric heating bands, capable of producing temperatures over the required range, were sourced and incorporated into the design (*right*).

Tribological analysis determined that friction independently generates as much as 70 Watts of heat during operation. To prevent the possibility of runaway heating, two fans were sized and designed into the fixture enclosure to ensure adequate surface convection to offset this heat input.



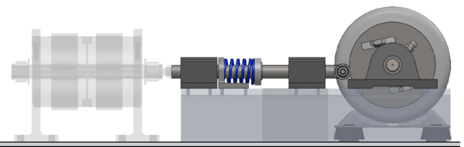
Failure mode and effects analysis (FMEA)

- *Static failure*: The final cylinder dimensions were established from two constraints: 1) the ASME-required safety factor for high-pressure cylindrical pressure vessels, and 2) the selection of commercially-available solid blanks having sufficient yield strength and diameter to manufacture the cylinder. Dimensions were chosen to guarantee the minimum required wall thickness at the cylinder's vertical center plane, where the wall is thinnest. Additionally, all material choices were vetted for the possibility of yield strength anomalies due to coring, which can occur during manufacture of the solid cylindrical ingots.



- *Fatigue*: The oscillation of the pressure pocket produces two regions of reversible loading and unloading in the cylinder walls, for up to 21M cycles per test (*above diagram*). Goodman fatigue analysis validated that stresses were sufficiently below the material's endurance limit and confirmed the infinite-life assumption for steel.
- *Catastrophic failure*: The ends of the piston were fitted with safety washers and nuts, sized to withstand the impact in case of a blowout and preventing the piston from becoming a projectile. A basic enclosure was designed to deflect the small volume of spray should any pressurized fluid escape the cylinder.

Drive system



The drive system design began with the choice of a DC motor for input. While linear solenoids and hydraulic actuators were also considered, the DC motor was chosen for its simplicity, long expected life, and ability to produce constant rotation throughout a 150-hour test.

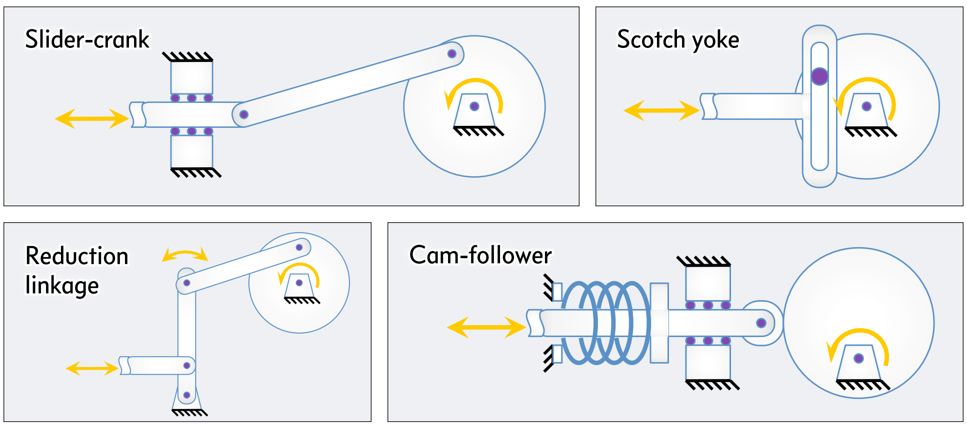
Drive mechanism design

To convert the motor's rotational input into linear, sinusoidal output, several mechanism designs were investigated in detail.

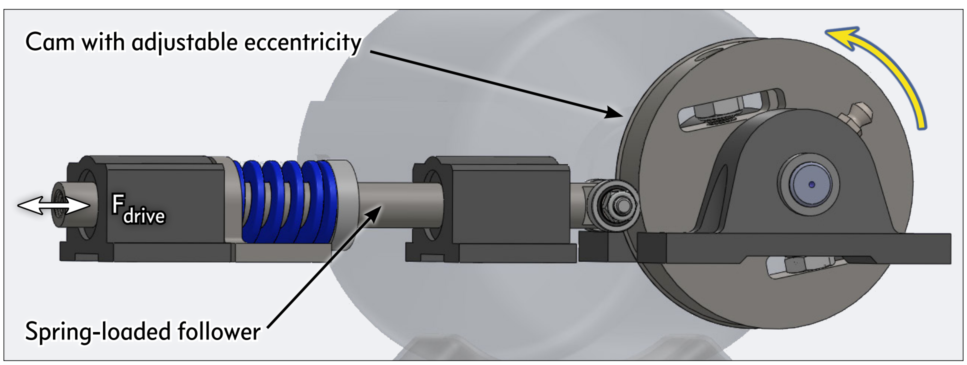
Geometric analysis showed that the slider-crank and reduction linkage mechanisms yielded output that deviated significantly from ideal sinusoidal motion. However, the Scotch yoke and cam-follower mechanisms performed exceptionally well, producing output within 0.25% of ideal sinusoidal translation.

The cam follower was then chosen for two clear advantages:

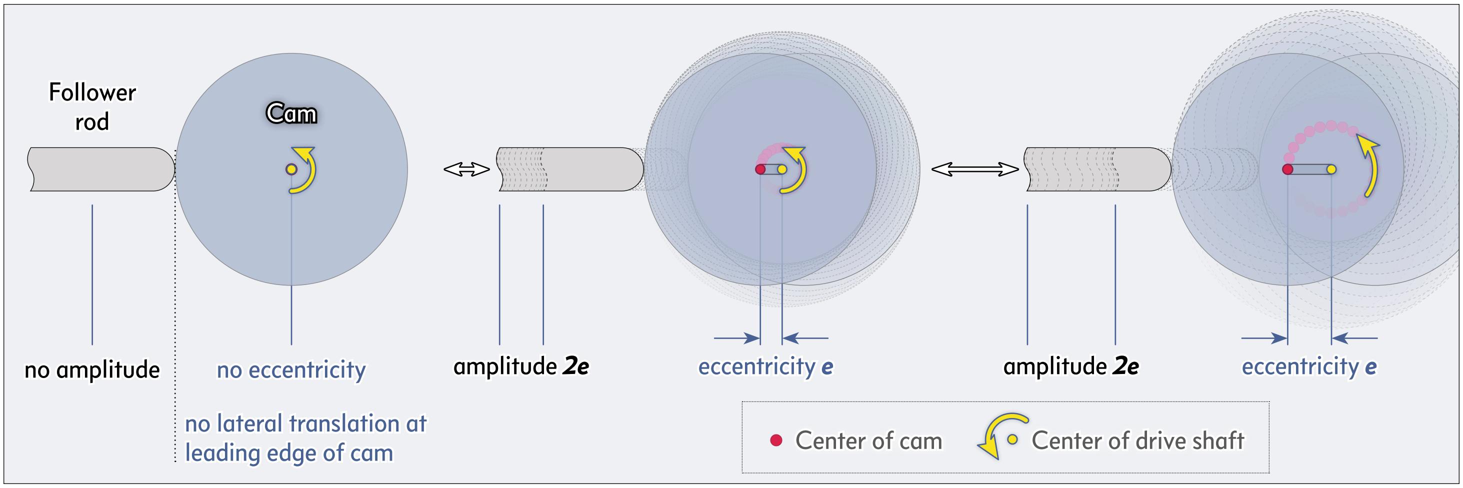
- fewer mechanism contact points, leading to less wear, less error due to tolerance stacking, and fewer points of potential failure; and
- the simplicity of producing output over the required amplitude range (from 0 to 1⁄10 of an inch).



To convert the motor's rotational input into linear sinusoidal output, several candidate mechanism designs were evaluated in detail.



A cam-follower system was chosen for its high output accuracy, low potential for error and failure, and the simplicity of producing the output amplitude within the required range.

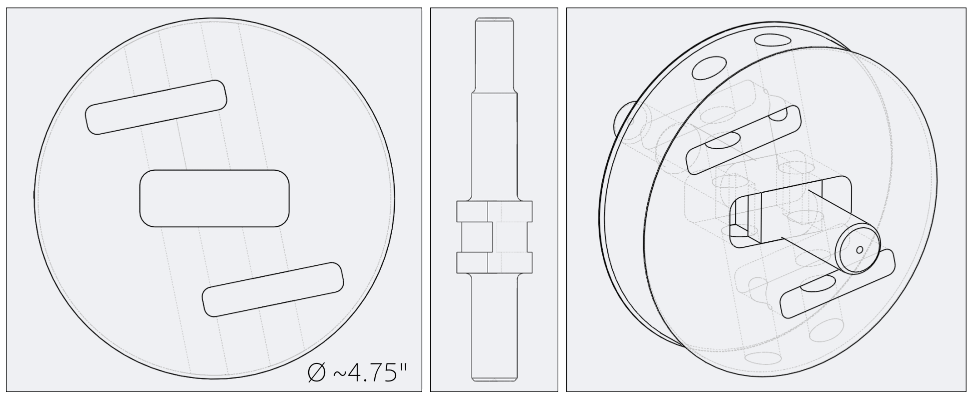


In the cam-follower system, the output amplitude is controlled by the eccentricity of the cam. When the cam is concentric with the drive shaft's rotational axis (left), the cam's leading edge does not translate, producing no output amplitude. However, when the center of the cam is offset from the drive shaft's axis (center and right), its leading edge oscillates back and forth with each rotation, generating a sinusoidal pattern with a peak-to-peak output amplitude exactly twice the offset eccentricity.

Generating the output amplitude

The drive system ouput is determined by the position of the follower rod, which translates horizontally, following the lateral displacement of the cam's leading edge as it rotates. If the cam is aligned so that it is concentric with the rotational axis of the drive shaft, its leading edge experiences no displacement; thus, the follower does not move, and the output amplitude is zero.

However, if the cam is positioned off-center from the drive shaft axis, the cam rotates eccentrically, creating a pattern of linear sinusoidal translation at its leading edge. This drives the follower rod with a peak-to-peak amplitude of exactly twice the magnitude of the eccentricity.



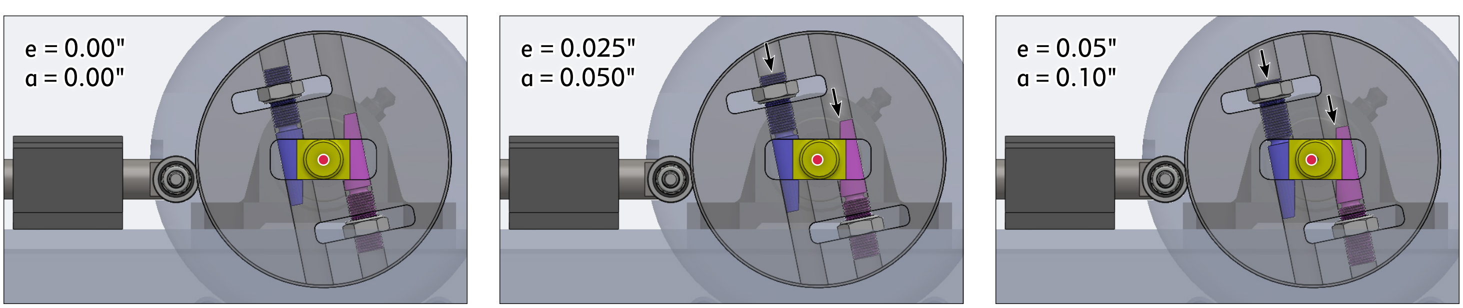
The squared-off section of the drive shaft interfaces with a horizontal slot in the cam, allowing the cam to be adjusted forward or backward relative to the drive shaft's rotational axis.

Making eccentricity adjustments

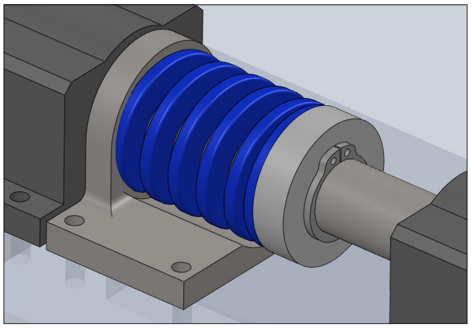
The drive shaft and cam are designed with a unique interface. The drive shaft is machined with a rectangular cross-section, whose horizontal faces make a coincident mate with the horizontal surfaces of a slot passing through the center of the cam.

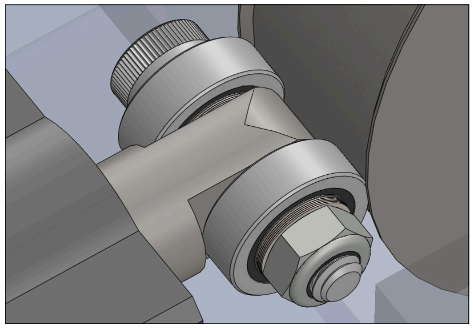
Embedded inside the cam are two symmetrical wedges, which fit tightly against the drive shaft's vertical faces and secure the cam to the shaft. The eccentricity of the cam is changed by adjusting the position of these wedges, which govern where the cam mates with the shaft.

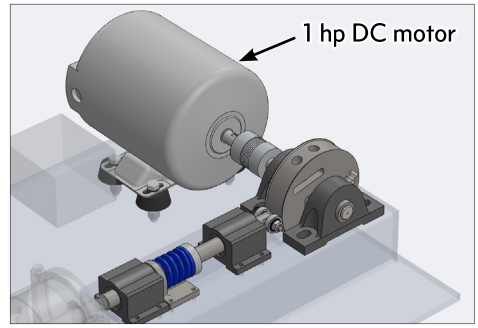
Very fine adjustments are possible with this configuration. Three full turns of the wedge nuts adjusts the eccentricity by just 1/40 of an inch, yielding an output amplitude of 0.05 inches. Any arbitrary amplitude within the specified range may be set *in situ* and confirmed with caliper measurements before a test begins.



Scale renderings of the adjustable cam system illustrate how the wedges' positions are adjusted to produce very small horizontal eccentricities.







Failure mode and effects analysis (FMEA)

- *Loss of contact between cam and follower*: If the follower spring is not sized with an adequate spring constant and preload force, the follower may lose contact with the cam, greatly disrupting operation of the drive system. Dynamic system analysis yielded functions relating the follower displacement to that of the cam, with the spring constant and preload force as input variables. Iterating using springs sourced from vendors, a specific spring and preload force were found to ensure enough spring load to keep the follower in contact with the cam.

- *Resonance/shaft whirl*: Analysis of the cam and drive shaft geometry revealed a minimum natural frequency of approximately 103 Hertz, where the cam's eccentricity would be undesirably amplified due to resonance effects. However, the motor's maximum speed (3000 rpm, equivalent to 50 Hertz), limits the rotational velocity to less than half of the natural frequency; thus, resonance is not a concern.

- *Drive shaft fatigue*: As with the dynamic region of the cylinder, the drive shaft experiences up to 21M cycles of reversible bending per test. Goodman fatigue analysis was performed at the point in the shaft most susceptible to failure — the stress concentration at the interface between the cylinder and rectangular cross sections. This analysis validated that the shaft stresses never approach the endurance limit and confirmed the infinite-life assumption for steel.

- *Spring fatigue*: Similar fatigue analysis yielded an acceptable safety factor for fatigue of the cam spring, which experiences a mean shear stress due to preload and alternating stress due to the cyclical displacement.

Motor sizing

With the force requirements of the drive system fully established, a DC motor was sourced having a torque-speed curve with the required minimum torque over the full range of rotational velocity (0–2400 rpm, equivalent to 0–40 Hertz). A control system was sourced to operate the motor with constant rotation. Vibration-absorbing mounts and a flexible motor coupling were chosen to diminish vibration in the drive mechanism and the base plate.

Details and part sourcing

The fixture design was refined in successive iterations as commercial parts were sourced, fasteners were chosen, and material choices were finalized. Machine shop personnel were consulted to verify manufacturability of custom parts and to estimate labor costs and machining methods.

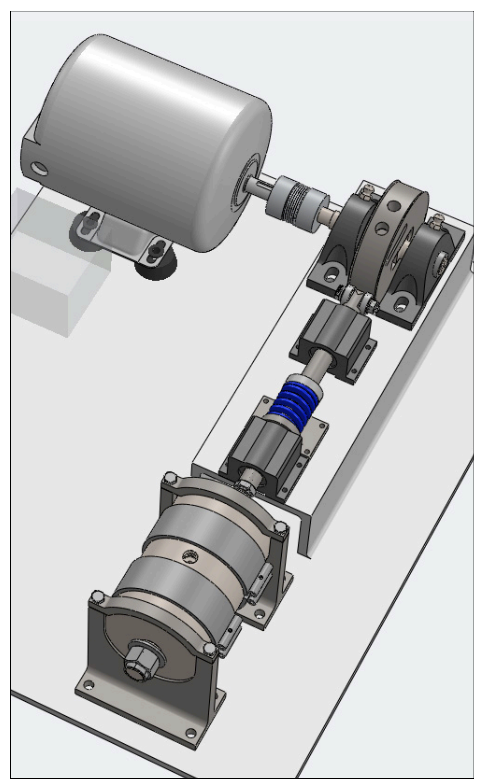
Several minor refinements were made to the fixture geometry as specific parts were sourced and incorporated into the design. For example, the diameters of the follower rod and drive shaft were modified to conform to the dimensions of the commercially-sourced bearings.

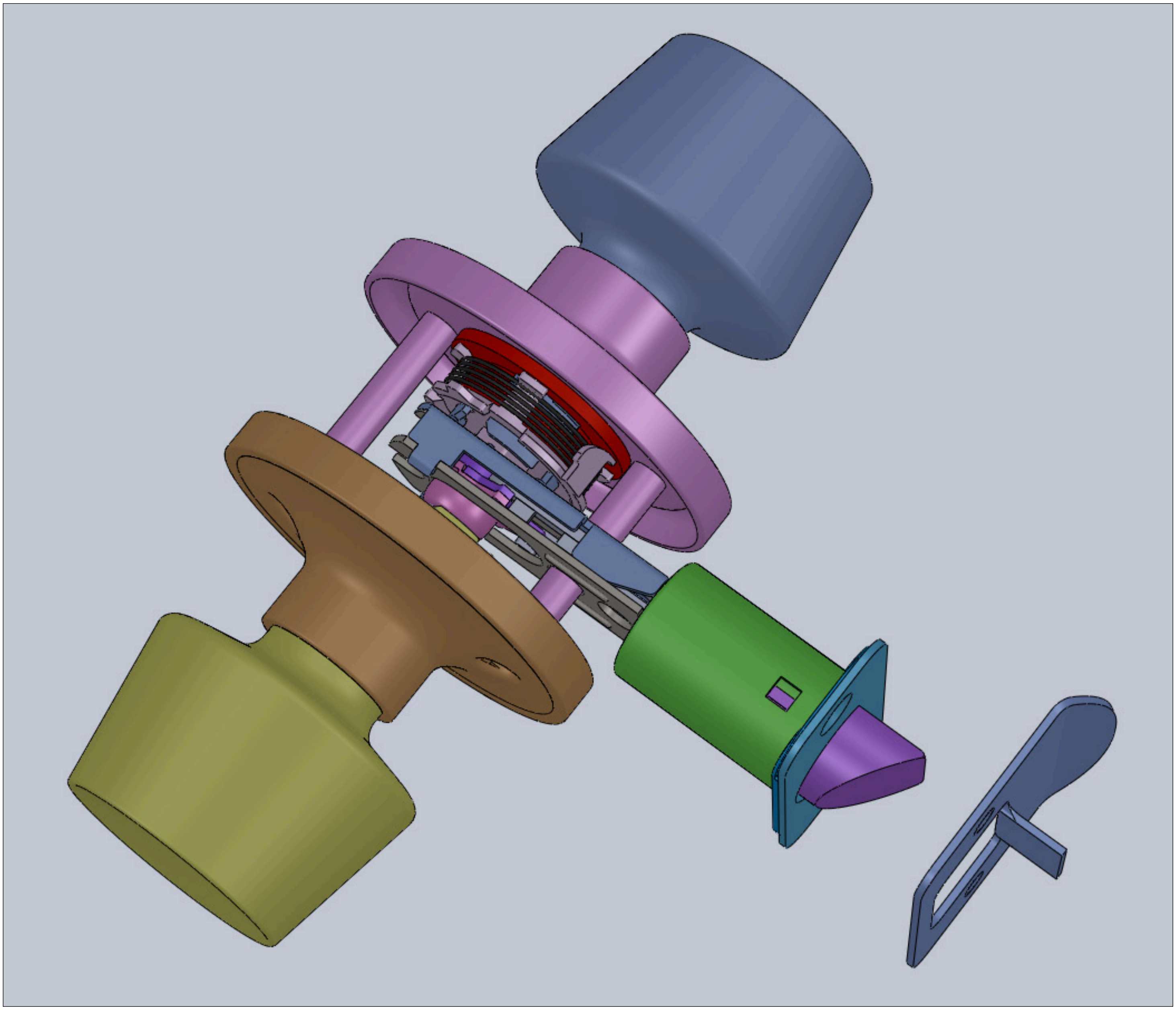
After each successive iteration, analysis was re-performed to account for changes in geometry and materials, validating that all performance objectives were still achieved and all safety factors were still respected.

Deliverables

The project team delivered to the Client a technical data package describing the fixture design, including:

- *CAD model files and engineering drawings* of all parts and assemblies with tolerances and critical dimensions specified;
- a *bill of materials* including vendor and ordering information for all commercial off-the-shelf parts and raw material sourcing details for 11 custom parts;
- *calculations, models, and analysis* documenting the fixture's design and performance capabilities; and
- a *user's manual* describing setup, teardown, and operation of the fixture.



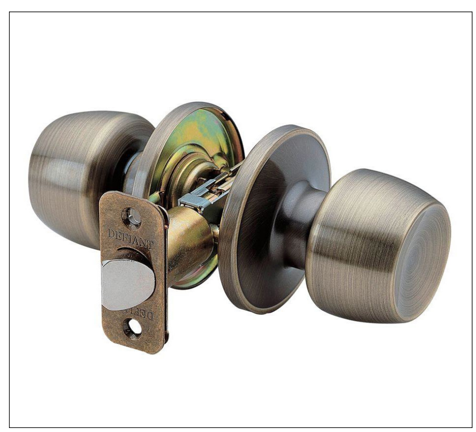


Reverse-Engineering, Modeling and Prototyping of a Commercial Consumer Doorknob

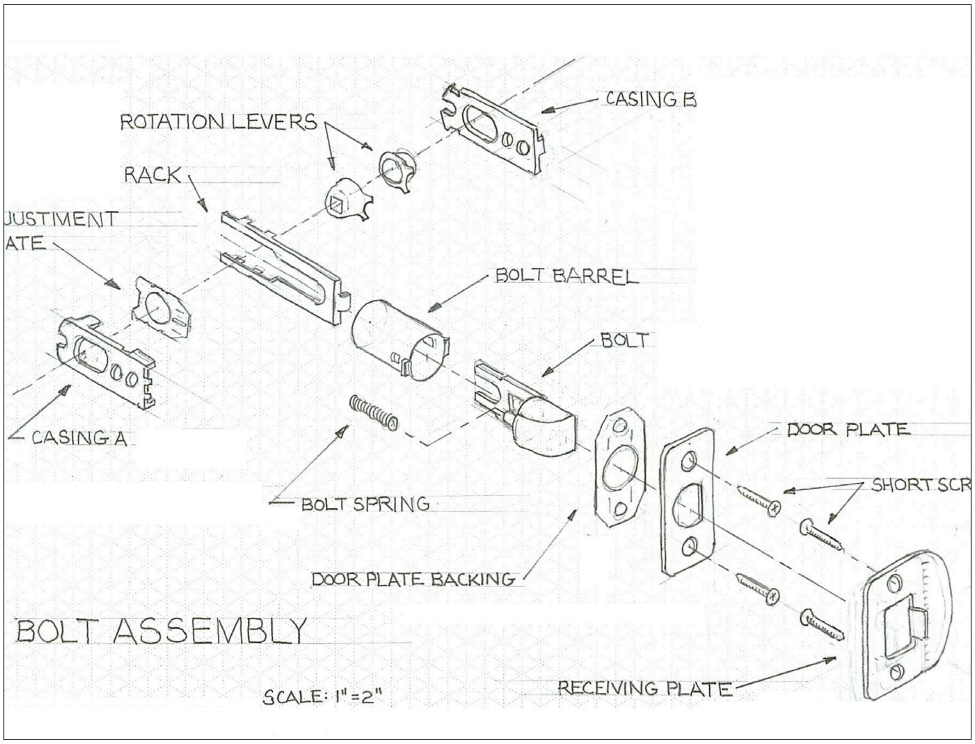
In which a consumer product is analyzed and modeled in CAD, and a replica prototype is fabricated using a Fused Deposition Modeling (FDM) 3D printer.

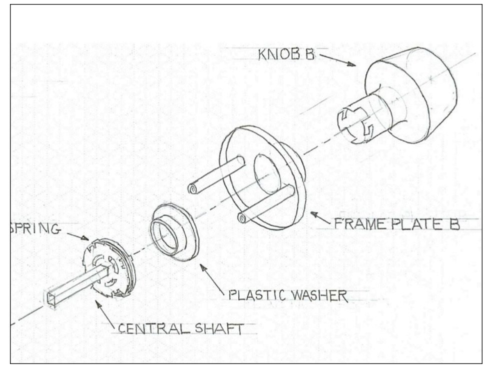
In this simple reverse-engineering project, a doorknob was disassembled and replicated, first as a SolidWorks model and then as a 3D-printed prototype. Prototyping methods like these can be valuable during the design process, to validate and improve mechanical functionality, to investigate DFM and DFA concerns, and to evaluate human factors and ergonomics. A physical model also serves as a useful, tangible tool for communicating with Clients and other members of a design team.

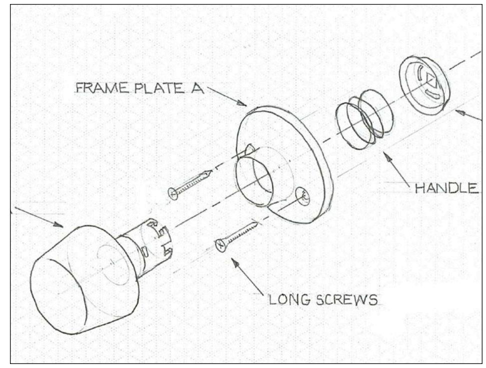
The project's main objective was to replicate the doorknob and its mechanical functionality as closely as possible. The final prototype mechanism reproduced the doorknob's functionality almost completely, except for the bolt's "spring-back" action, which could not be achieved due to the significantly reduced shear strength of the 3D-printed parts. This shortcoming shows that while FDM prototyping can be valuable during the design process, it has limitations that still must be considered.



A Defiant hall and closet doorknob, like the one pictured above, was analyzed, modeled, and replicated in ABS plastic.

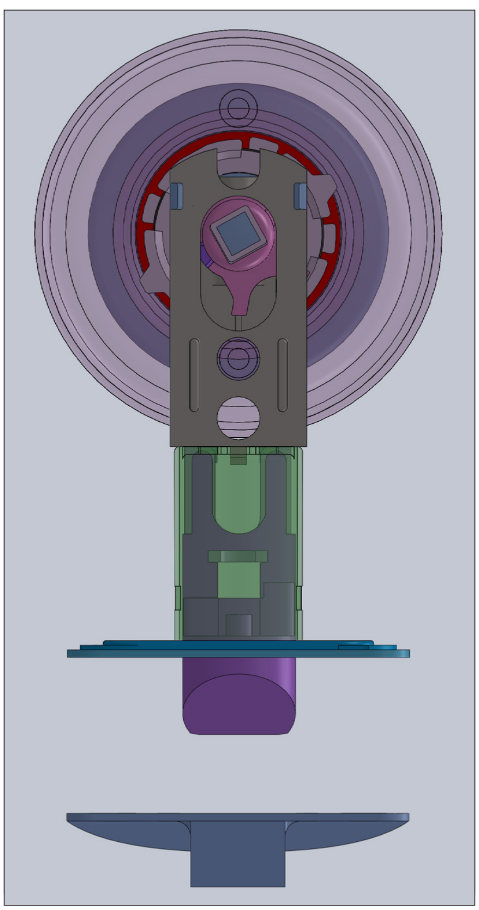


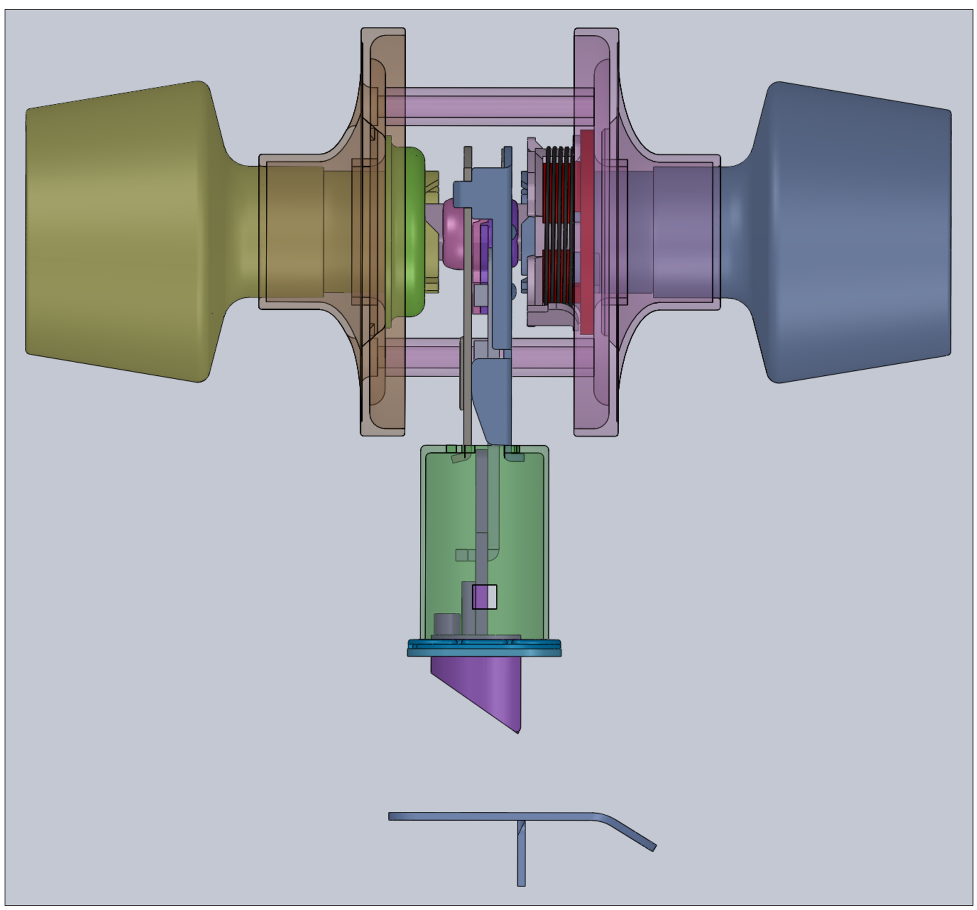


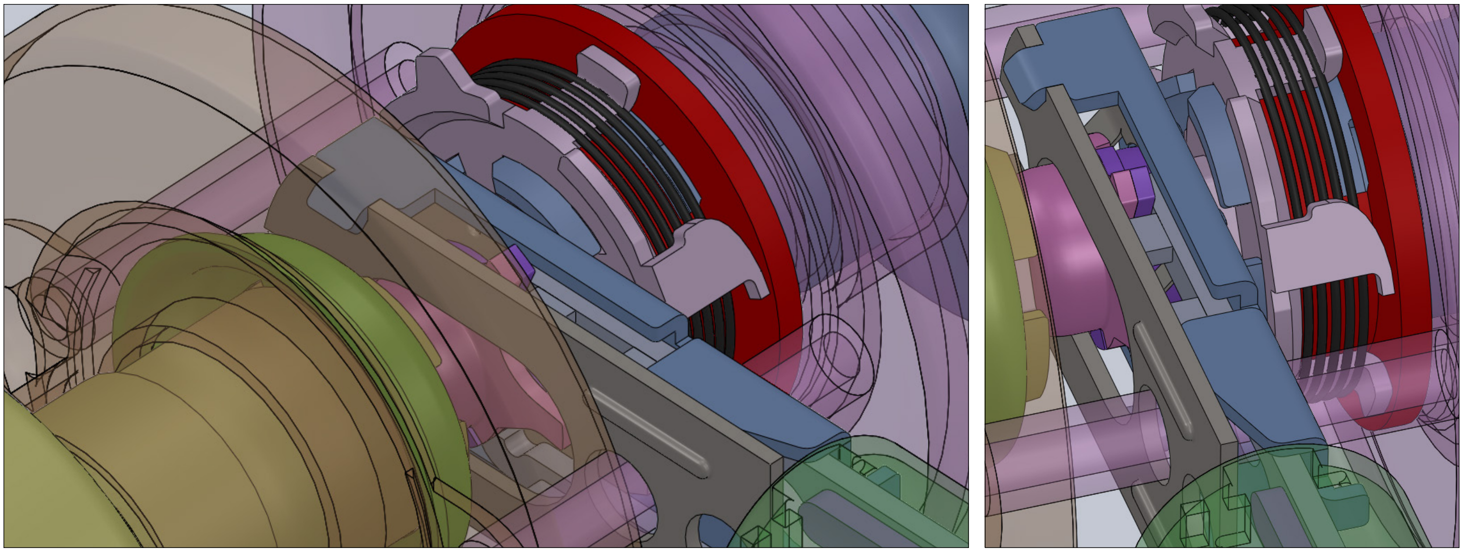


Exploded sketches of the doorknob's subassemblies illustrate the individual parts in the doorknob's bolt-withdrawal mechanism and their relationships to each other. The mechanism's functionality can be understood in terms of the roles of its subassemblies: the handles transmit torque input to the bolt subassembly's rotation levers, which convert the rotation into linear force, withdrawing the bolt.

Properly-defined part geometry and mating relationships yield a computer model that accurately represents the original product and its functionality. Non-functional aesthetic details, such as the contours of the handles, were approximated due to project time constraints.







The computer model of the full mechanism assembly is illustrated in detail.

Several of the replicated parts, in green ABS plastic, are pictured next to their corresponding originals. The assembled prototype (center) successfully reproduces most of the doorknob's functionality, except for the spring-back action, which returns the bolt to its extended position after a handle is released. The coiled steel handle spring imposes stresses too great for the plastic central shaft (lower right) to endure.