ETME 3100 Junior Design Practicum

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Subject:	Junior Design Practicum Final Design Package
Date	May 7th, 2024

Throughout this project, the team was tasked to design, fabricate, and test an Elmer's Twin Wobbler engine. An altered model of the provided Wobbler air engine design was created using Solidworks. The design was scaled up, making the engine twice as large as the provided schematic. Modifications were made to the flywheel, column, and cylinders, and implementing ball bearings. The largest change was to the port timing on the cylinders and column. A document of EES code was written to calculate the theoretical power and efficiency of the engine. The team then fabricated each part using the detailed drawings created in Solidworks. Once all parts were machined, the team assembled and broke in the air engine using a powerful cordless drill, following a 3-minute break-in regimen at different RPMs.

Once the engine was assembled, the team found excessive air loss between the piston and cylinder mate. After trying to test once, the engine did not perform under a load - prompting the team to quickly remake the cylinders and shave down the pistons to create a tighter fit. The pistons' fit improved, but the engine could still not support a load once attached to the dynamometer. The team tested the air engine without time for further modifications and still could not produce any power data. The engine speed was reported to be 655 RPM at 40 psi, with a minimum operating pressure of 12 psi. These performance values were significantly lower than the expected values.

The team believes that this lack of power originates from a complex combination of design flaws. The first one was the port timing and a possibly improper understanding of how the air was to flow into the cylinder chamber. The second flaw was due to the end of the exhaust outlet being significantly larger than the entrance diameter, resulting in a drop in back pressure in the engine as it vented to the atmosphere. The last design flaw was the chamfered ports resulting in a higher-than-expected air loss between the column and cylinders. These factors were not reflected in the EES code, and are likely the reason behind the lack of power.

The team does not recommend changing the port timing in future iterations of this engine. It is also suggested that the ports should not be chamfered, as this resulted in increased air loss. For the fabrication of the piston and cylinders, the team found a boring rod to be imprecise. Instead of a boring rod, a drill bit and reamer should be used to drill the piston hole into the cylinder, and the pistons should be lathed and sanded for a low-tolerance fit. The team also recommends using two separate air systems, providing each cylinder with optimal exhaust and intake cycles. This will help mitigate interference between the cylinder/piston cycles and improve the engine's performance.

ETME 3100 – Team Heat – Final Design Package

Date	Revision	Author	Comments
2024-05-07	A	Team Heat	Original Document

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1 Project Objective

This project's ultimate goal was to design, build, optimize, and test an Elmer's Wobbler air engine that utilizes compressed air to cycle the engine. The group's model was first created in Solidworks, followed by an analysis in Engineering Equation Solver; findings will be presented professionally in the following design report. The team's determined goals were to have the air engine run below 2 PSI, produce up to 35 W at 40 PSI, and have an operating efficiency of 25%.

2 Design Narrative

The team was tasked with creating Elmer's Twin Wobbler Engine as a class project in Junior Design to combine the knowledge of fluid mechanics, general physics, and hands-on machining skills. Original plans were provided by Professor Hewlin that were to be scaled up two times to match the needs of the project; for example, the flywheel diameter, piston stroke, and bore diameter were all to be enlarged. As detailed in the Design Project Description, "a Wobbler steam engine is a valveless oscillating engine with the connecting rod and piston formed as one rigid piece" [1], where the cylinders rock back and forth to act as the method of air transfer by aligning corresponding ports on the column. Once the team understood the project goal, the first Solidworks model was assembled over a week to create a working prototype of the air engine as prescribed. By creating a working 3D model of the engine, the team analyzed the motion of the pistons through the cylinders & original ports using Solidworks' wireframe view.

2.1 Innovations & Theory

With a better grasp of how Elmer envisioned his Twin Wobbler to work, the team set off to discuss what portions of the engine could benefit from a closer look. Through many meetings, long discussions, and research, the focus was shifted toward optimizing the original design to achieve lofty RPMs and high electrical output. Initial innovations included the addition of sealed ball bearings, experimental port timing, and overall attempted improvement of the original Elmer design [1]. Bearings and reduction of friction offered bright prospects, as utilizing these components would assist the engine in achieving both goals of high performance. Port timing appeared to be the secret to breaking the arbitrary 1300RPM barrier that would require the most development and precision. In theory, these small optimizations would add up over time as the second engine prototype began to materialize - cumulative minor gains that would eventually lead the team to reach the ultimate goals set at the start of the semester.

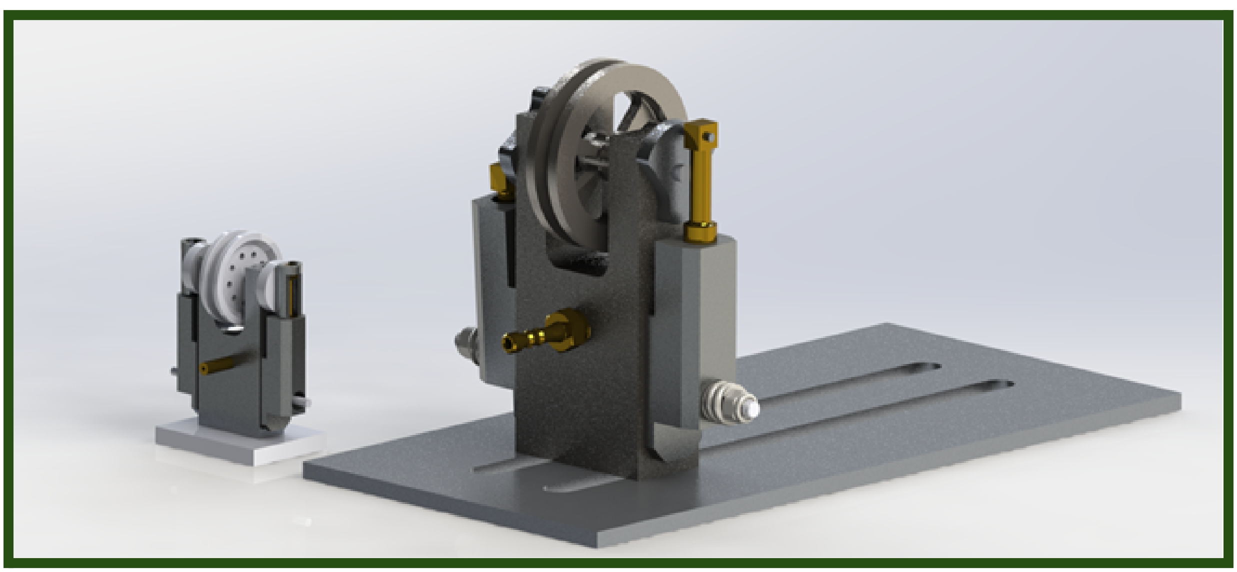


Figure 1: First prototype engine (left) compared to the third prototype (right).

2.1.1 Friction Mitigation

Unsealed ball bearings, ⅝ inch in diameter, were proposed to the team by Professor Barringer to aid in the smooth movements of any rotating part against a through-hole. Taking this idea into consideration, the team set off to design the second iteration of the engine. The new design would include pockets for bearings to sit in along the path of the driveshaft first to allow free rotation of the shaft, therefore also reducing friction experienced between the shaft itself and the "ears" of the column (*Appendix C6*). In tandem, each cylinder would also feature a recessed ball bearing on its outer face to further decrease the friction of the cylinder against the pivot shaft. In pursuit of further lowering any form of rough component contact, each mating face was also to be given a smooth surface finish by way of a grinding wheel & high-grit sandpaper. This process polished the contact portions of the cylinder against the column to promote the smooth oscillation movement created by the engine. This innovation was deemed to make up a relatively small portion of the overall design, but its ability to mitigate friction (and therefore premature wear) was of great use in moving towards the team's goals.

In continuation with attempting to reduce friction, the team hypothesized that the piston must be long enough to stay in the cylinder at its furthest point, but also reduce the contact area to maximize air volume within the cylinder to force the piston through its cycles. When trying to determine how long the piston needed to be, a sketch of a stroke diagram (*Figure 2*) was created that included four points of rotation at the top dead center (TDC), bottom dead center (BDC), and then two points at 90° and 180° of rotation. The team found that the piston doesn't make optimal use of the available cylinder bore depth. At both the 90° and 180° mark, the cylinder would be approximately 67% occupied while the BDC would be fully occupied. To determine the stroke length, all that was needed was to subtract the TDC measurement from the BDC depth. Approximately, taking the difference in values (1.630in at TDC to 0.380in at BDC) from the piston head to the bottom of the bore left the team with a total stroke of 1.25in. From this, the team determined that the piston length should be ⅛ inch shorter than the original Elmer plans [1].

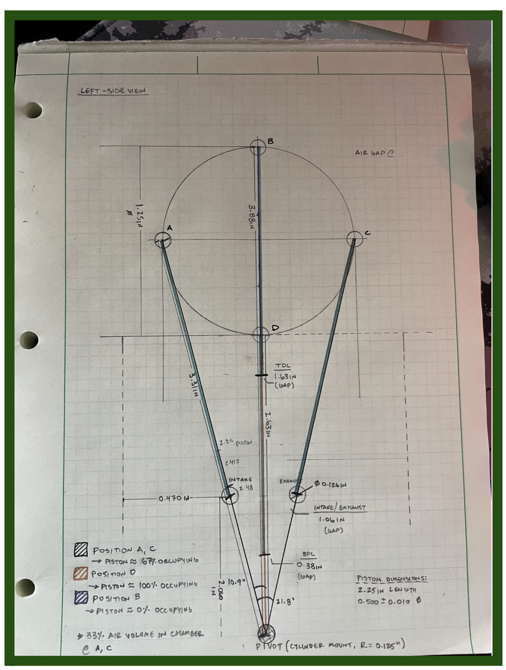


Figure 2: Hand-drawn stroke diagram.

2.1.2 Port Timing Analysis

The possibility of adding port timing to the engine only came about when analyzing the motion of the first Solidworks prototype model using the wireframe set. At this point, every required part had been scaled appropriately with modifications, but the size & placement of the airports on the column & cylinder remained the same as Elmer's design [1]. It appeared that Elmer's use of two side-by-side ports on the column and a single port on the cylinder did not scale to size the way the team had hoped. By analyzing the wireframe view of the piston/cylinder assembly, the ports were only aligned for a short period during the "power" strokes of the engine (90° rotation from BDC, clockwise) and did not appear to be an efficient use of the available air pressure. From this, the air column was modified to have two offset ports at different heights that would correspond to two vertically stacked ports on the cylinder. Set at specific distances, the column ports would allow the cylinder to maximize its time spent aligned with the intake port on the cylinder and therefore allow more air to enter the cylinder (Shown in *Figure 3*).

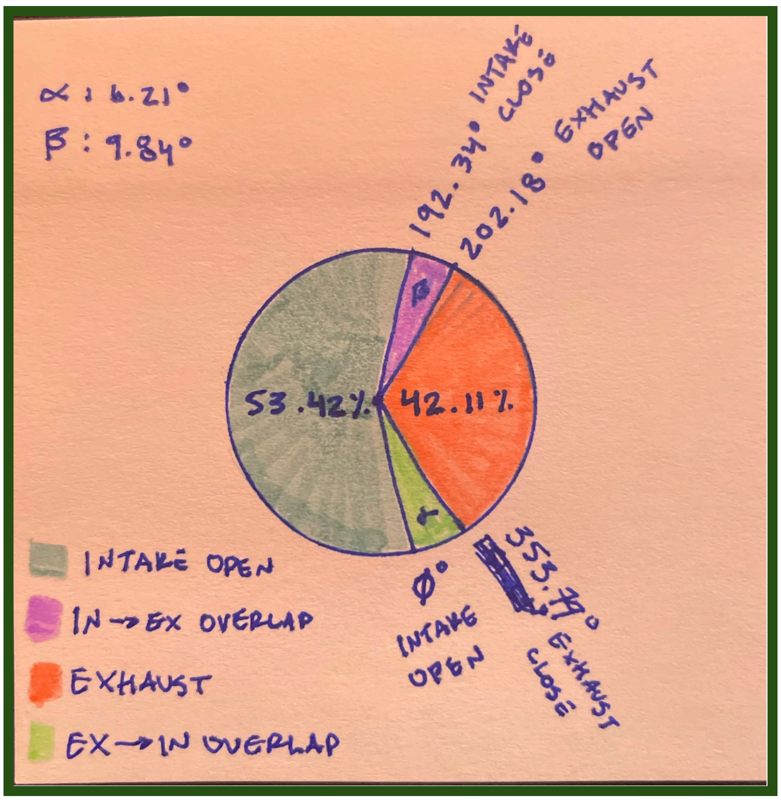


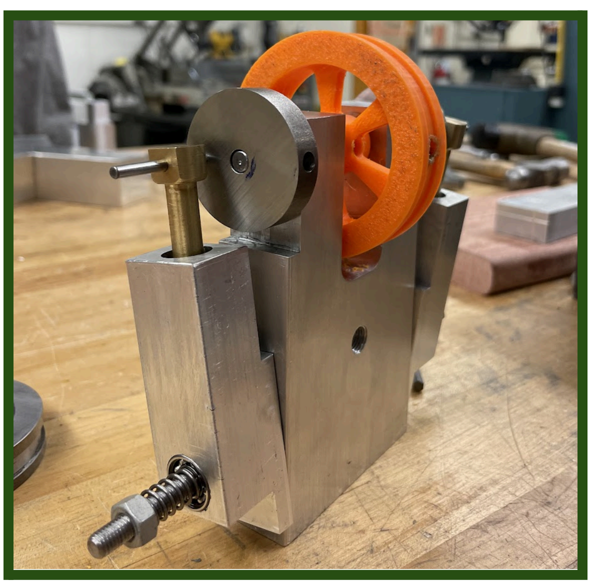
Figure 3: A diagram of the new port-timing idea, showing two points of wasted rotation (α, β).

In addition to increasing the duration of airflow into the chamber, the cylinder's intake would also be a larger diameter than Elmer's original design. To ensure that pressure would be able to escape similarly, the exhaust port was designed to dump out the air earlier in the piston's cycle of rotation and let momentum carry the cylinder to its next intake sequence, similar to other steam engines [2][3]. A small chamfer was added to each port to slightly increase the open surface area and facilitate the direction of the air as it was transferred from column to cylinder. By increasing the diameter of the ports, shaping those ports, and increasing airflow, the team had a great theoretical path to secure impressive results.

2.1.3 Experimental Flywheel Development

When designing the overall shape & look for the flywheel component of the air engine, it was decided that the team could reap the most benefits by using a denser material to conserve momentum. Utilizing AISI 1030 steel, the team members responsible for the flywheel took it upon themselves to develop the G-code required to use the CNC machine for cutting down the round stock into its finished form (*Appendix D2*). As this process was being undertaken, the question of engine torque and overweight concerns arose. Yes, the flywheel does incorporate ingenious pendulum-shaped spokes and kept most of its mass on the outer diameter, but the team was concerned that the pistons would struggle to overcome the weight of the flywheel and lead to worsening losses over time as the engine received higher air pressures for testing. With this in mind, an emergency contingency plan was initiated if the engine would be unable to handle the rotating mass of an approximately 1 lb steel flywheel.

To remedy this situation, the team decided to make use of another option - since the team had already used the CNC machine to create one part, the project guidelines stated that there was also the option of 3D printing a component if there was such a need for one (*Figures 4, 5*). Given that the flywheel files had already been completed and polished, it was a simple matter of converting the .sldprt file to a .stl for any generic 3D printer to produce in a matter of hours. A team member had direct access to a printer that would perfectly fit the team's needs, and since only one type of filament was available at the time, material selection was quite easy. A spool of polylactic acid (PLA) was chosen as the plastic to print the flywheel out of as it was readily accessible and cheap. Over a few hours, the team had a secondary emergency flywheel that weighed significantly less than its steel counterpart, fulfilling the objective of reducing the amount of torque required to complete a piston cycle. This PLA flywheel did not feature a set screw or threading, as the resolution of the 3D printer used was too inaccurate to directly print a 0.250" through-hole - this small oversight led to the emergency flywheel being used as a press-fit to stay in place on the driveshaft.



Figures 4, 5: Depiction of the PLA flywheel in the engine, and an isometric view of the PLA flywheel on its own.

In pre-testing, the PLA flywheel performed exactly how the team had expected it to - the pistons were able to fully cycle at a lower PSI as they had to push against much less weight. Consequently, the overall top speed was hindered as a result of the loss of momentum from an overall less dense flywheel. Despite each flywheel being made of a vastly different material, the engine performed roughly the same no matter which flywheel was fitted to it. There was no noticeable difference in the maximum RPM achieved at 40psi and continued to perform the same up to a speculated 60psi. With the designs being the same (with minor inaccuracies offered by the printer's limited resolution), this proved that the flywheel itself had potentially negligible changes to the performance of the engine. As a result, the team opted to continue developing the engine with the steel flywheel (*Appendix D2*) fitted which would ultimately be used for final testing.

2.2 Component Design

The pistons (*Appendix C1*) were to be made out of brass that would be cut from rod stock; the reason that brass was selected was for easy machinability when cutting the material. Another reason the team selected brass was so that when it was tested and run, the material would be different from the aluminum cylinders. This was so the two materials wouldn't mold together during testing, known as "galling". The team also designed the pistons to be rounded on the corners and have a slightly bigger head. This was so that it would slide easier down the cylinder walls with rounded corners, and create a tighter seal with a bigger head.

The crankshaft (*Appendix C3, C4*) was designed to be more similar to those found in car engines. If it were to be produced how it was initially designed the crank would follow a more T shape. This T shape is to improve momentum allowing in theory for the engine to run at a lower PSI. With advice from a skilled machinist it was determined that the crank cutoffs were not needed as the momentum added would not be sufficient enough to keep the engine running at a lower PSI.

The flywheel (*Appendix C2*) was designed out of AISI 1030 leaded steel. AISI 1030 was chosen for its weight and ease of machinability. The design of the flywheel allows for a higher amount of momentum to be conserved while trying to remove as much heavy material to reduce the overall weight. This allows for the flywheel to help the pistons move as much as possible while having a relatively low minimum pressure needed. A concentric cut on both sides was made to reduce the weight in the middle of the circle while aiming to keep some weight along the rim of the flywheel. A spoked pattern was then cut into the flywheel to further reduce weight. This pattern is feasible to make with the use of a CNC machine. The flywheel was designed to be machined using standard bits and cutting pieces.

The column (*Appendix C5, C6*) was designed to be made out of aluminum 6061-T6 and it was cut out of a 3x1 inch rectangular box. This was supposed to be a 2.5 inch by 1.25 inch block but in the machine shop, 3x1 bar stock was the closest it had to that dimension. With all that accounted for some dimensions did have to be adjusted. These included the clevis cut as well as the depth of the exhaust holes and the diameter of the bottom pivot shaft through-hole. These changes were small but were effective when making the assembly and every component meshed together well. The outtake and intake holes were offset to maximize the airflow. This was making it more like an air pump rather than an engine, to maximize airflow. The idea was to avoid the suction and vacuum forces experienced by the pistons while increasing speed. This did not happen as the engine was plateauing at a certain revolutions per minute, as it did at about 655 rpm after 40 psi was reached.

The twin-cylinder components (*Appendix C7, C8*) of the engine were created with the idea of port timing in mind. Each cylinder matched the other in size, length, and most importantly - hole placement. Drilling the through-hole for the pivot shaft to pass through had to be as close to the center as possible to prevent any undue torsion while the engine operated. In addition, the ports placed on the back of the cylinder had to be aligned vertically down the center to ensure that the holes would match up on time with each stroke. Featuring a ball bearing that allows the pivot shaft to twist freely, chamfered intake and exhaust ports, and a 0.002" tolerance with its accompanying piston, the cylinders were an important component of the air engine that dealt with the majority of friction experienced. The goal was to use the port timing on the column to increase overall performance and speed but it produced a risk of complete failure if the theory proved to be incorrect.

Minor hardware utilized in the engine's construction included the pivot shaft, driveshaft, nuts, springs, washers, and bearings (*Figure 6)*. All of these components were vital to the operation of the air engine and were already premade for the team to use. Both shafts for the engine were simply ¼ inch round stock AISI 1030, cut down to size. In the case of the pivot shaft, it required approximately an inch and a half of UNC ¼ - 20 threading on both ends to accommodate the nuts used to keep tension on the springs against the cylinders. To promote smooth action, a washer was placed over the bearings to keep the spring from interfering with the mechanism. This simple system allowed the team to adjust how tightly the cylinder was pressed against the column and have more control over the fine-tuning of the engine while in operation. About Elmer's original plans, this was the method used in the given diagram to perform the same task the team needed it to.

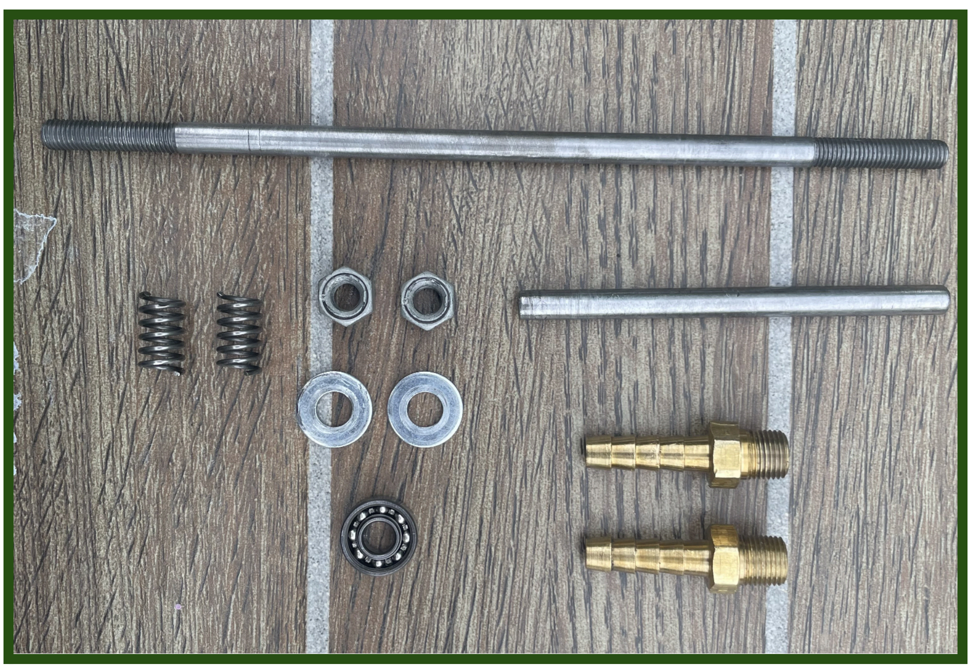


Figure 6: Examples of minor hardware used, including both component shafts.

3 Fabrication

The pistons (*Appendix D1)* were fully fabricated from 2 cylinders of brass. Starting with one long cylinder of brass, the overall length was cut down to 4 inches. The pistons were then both cut on the lathe to form the head and the necks for both of them. However, the pistons were still attached end-to-end and the team thought it was best to just shave the material from the pistons going backwards from each end and working the way towards the head of each piston. Once the necks were shaved off with plunge cuts and then surfaced finished to be smooth. After that, the pistons were removed from the lathe and taken to the band saw to be cut into two separate pistons measuring roughly 2 and ¼ inches in length. Next, they were brought to the milling machine to square off both of the bottoms so that they would fit smoothly up against the face of each crankshaft. Lastly, a ⅛ hole was then drilled horizontally through each squared-off bottom piece so that they would fit onto the crankshaft pins. Finishing touches were sanding off sharp corners of each piston on the tops and bottom to add a fillet so there were no sharp edges when it would be placed inside the cylinder.

The flywheel (*Appendix D2)* was fabricated out of AISI 1030 leaded steel. A 0.700-inch piece was cut off of the stock cylinder using the shop's band saw. The material was then cut down to its final thickness of 0.625 inches on the lathe. A series of plunge cuts on the lathe were then used to remove 0.213 inches for the weight reduction on the flywheel face. A center drill was then used, followed by a 15/64th drill bit and quarter-inch reamer to create the through hole for the shaft. The flywheel was then bolted to a sacrificial piece of scrap to cut the 0.050 in belt grove in the flywheel. The spoke pattern was then cut out using a CNC machine and G-code. Once the spokes were cut the through hole and threaded hole for the set screw could be drilled using a milling machine. The flywheel was then deburred using a file and sanded using a belt sander.

The crankshafts (*Appendix D3)* were cut from 1 and ¼" diameter ASTM 304 stainless steel. The two discs were originally meant to have a thickness of ¼ " but after guidance from an experienced machinist, it was ground down to have a smooth finish. With the finished discs now having a thickness of 0.336". The change in crank thickness did not affect the overall assembly even with the column's shoulder cutouts. With the overall shape finished, the next step for the crank was to drill the ¼" through-hole for the flywheel pin. After the hole had been drilled it was reamed to 0.250" to accommodate the driveshaft. The next step was to drill the crank pinhole. The second to final step for the crankshafts was creating the set screw-tapped hole. With all previous steps completed the final step was press fitting the crank pins. These pins had to be sanded down slightly to allow them to be press-fitted into the crank face itself.

The column (*Appendix D4)* underwent significant changes from the original design, including the addition of two bearings to address errors that became apparent when drilling the through-hole for the pivot shaft. Initially planned to be 2.5 inches wide and 1.25 inches in length, the team had to adjust to 3x1 inches due to the sizing of stock available in the machine shop. This made the drilling of holes difficult to place as all references were different, so new drawings had to be made. The holes were pretty much in the same place, just had more space in width and less space in length. With this the holes were drilled concerning the new drawing with some modifications of dimensions like the threaded holes on the bottom had to be in the center of each half of the section to make the support even while attached to the base plate. All holes for the intake, outtake, and rod support holes were in the same place as the original drawing. The first problem that was encountered was the bottom through hole for the rod that holds the cylinders was diagonal and not straight through. The drill bit walked through the material considerably from where it needed to be. This issue was corrected by creating bearing holes and making a bigger hole that was machined out with a 9/32 end mill that would make this new hole straight for the rod to pass through. The second problem was the flywheel hole did not sit flush with the bearing holes on the top so more clearance had to be made for it. Attempting this without zeroing out the scale was not a good idea as the event turned into a disaster because each side was not even, but was later fixed by expanding out the area of clearance so that the flywheel could line up with the hole and spin freely on the rod. Next, the cylinders and cranks did not fit flush like how they were drawn up, so shoulders were made to account for that and it worked nicely as they were only 3/16 off each of the top sides of the columns. Lastly, the screw holes were too short due to not knowing how deep to drill them, and again their depth was based on an educated guess. They were fixed by shortening the screws to account for the shortened holes. All these issues were fixed and noted, but the main one that affected the function of the engine was the port timing. This was the offset port holes with them usually being right in the center of the length side but the group decided to offset them both just a little bit from the center to the left and right based on previous analysis & development. The exhaust port is 0.531 inches from the right and the intake is 0.469 inches from the left. This was thought to work in the group's favor but it was ultimately the engine's demise. Chamfering was attempted to enhance airflow, yet plugging the outtake holes resulted in unintended acceleration due to air recycling This was discovered after initially testing it, and realizing that there was a problem. With all the design changes that the column had it was not enough to make the engine run properly and the port timing is to blame - this was a test to see if offsetting ports could make for a faster, more efficient engine and it did not work unfortunately. Ultimately, these adjustments were insufficient to optimize engine performance, highlighting the importance of port timing. While the concept of offset ports showed promise, the team's experiment did not yield the desired outcome.

Figure 7: Side-by-side comparison of port alignment.

Figure 8: Side view of port offset.

Finally, the twin cylinders shown in *Appendix D5* were machined as one of the last parts to be made. Due to the nature of the cylinders and how the manufacturing timeline was laid out, the pistons and air column were determined to be made before the cylinders were. To begin the machining process for the cylinders, each piece was made to completion before the next was begun. Starting with a 4-inch long, 1x1" square 6061-T6 aluminum bar cut to length using the bandsaw, the cylinder "blank" was then faced off using the turret mill for accurate measurements. Once faced, the blank was then cut down to a total length of 3.300" as detailed in the Solidworks drawings, and a portion of the material was removed - creating portions known as the "backstrap" and "front face". The backstrap was to rub against the column wall and required a smooth surface finish, whereas the front face contained the bearing and was oriented towards the outside of the engine. Once the material had been removed, the next step in the process was to drill out the bore for the piston to operate within, as well as a 0.625" diameter, 0.200" deep bearing pocket. Initially, the bore was drilled using a ½" HSS drill bit, which was the "rough cut" to begin the boring process for the first set of cylinders.

In doing this, the team had opted to use a boring bar to get precise cuts down to 0.001" at a time, which was against the recommendations offered by Professor Barringer. When consulting with Professor Pritchard, there seemed to be little reason to not utilize the boring bar to precisely hone the diameter for a tight piston fit. As the boring process commenced, it became apparent that the tool itself was inaccurate to a certain degree - cutting far too little on one pass and then cutting far too much on the next. This, of course, was an issue. The boring bar put the first set of cylinders out-of-round and resulted in serious fitment issues with the existing pistons - despite the cylinders matching to 0.005" tolerance with the piston heads, the uneven hone on the bore caused binding issues on test runs. Without knowing the true extent of the issue, the inlet & exhaust holes were then drilled at exact locations for the ideal match-up at the column interface.

Following the completion of the first set of cylinders, it became apparent that the interference issues with the pistons on the pre-test run were causing serious issues to the engine's performance. Due to the break-in procedure, the air gap had grown considerably between the piston/cylinder interface and resulted in serious air leakage. This necessitated a second set of cylinders, which were made in the same fashion as the first set with a boring bar to try and machine the bores as straight as possible. It wasn't until the team decided to use the 0.501" reamer on the third set of cylinders (made in 4 hours before final testing) that the pistons and cylinders worked flawlessly together. On the final set of cylinders, all decorative portions of the component were scrapped - just the bare minimum was machined to achieve some kind of successful performance for the final testing session. It's worth noting that each port drilled through the cylinder backstrap and into the bore was perfect for all 3 sets - the failures stemmed primarily from the bore being out-of-round and misalignment of the piston/crank whilst the engine was operating. Whenever the piston would reach its highest point in rotation, the head would scrape down the side and once more against the opposing wall on its way up, causing undue friction and premature wear that damaged the team's desired vacuum-like seal.

4 Test Report

During trial runs and testing, the air engine was attached to a supplied baseplate and dyno assembly (*Figure 13*). During fully assembled test runs, the engine would turn the dyno at a slightly lower-than-expected rate. Upon testing a failure of the dyno was detected, with the dyno applying no resistance to the belt. The team replaced the dyno and reassembled the engine. With the new dyno, the engine was unable to turn the dyno without a load applied. This lack of power was believed to come from excess air loss between the piston and cylinder. Due to this noticeable air loss, the team decided to re-make the cylinders and shave down the pistons to ensure a better fit. It was also noted that once exhaust was restricted, a higher RPM would be achieved. This indicates that back pressure may be needed to improve the efficiency of the engine. The team did not have time to remake the column. There was discussion about adding drops of superglue into the exhaust ports as a last-minute fix but ultimately chose not to risk damaging the engine.

Once the cylinders and pistons were machined to have a better fit, the team retested the air engine. Upon retesting, the team was unable to collect detailed power test data for the designed wobbler air engine, as the engine was unable to support a load at 40 PSI. The team used a maximum RPM of 1370 to run the EES calculations. The engine only achieved a maximum of 655 RPM at 40 PSI without a load applied, and the minimum pressure the air engine would run at is 12 PSI. These lackluster results are due to excess air loss between the cylinder and column, and improper port timing. Since no test data was collected, the team will analyze their EES code results.

Using EES the team calculated the engine would produce a maximum of 12.18 watts of electrical power at 1370 RPM seen in *Figure 9*. With the engine running at only 655 RPM, the EES model calculates an electrical wattage of about 5.2 Watts. The team calculated a maximum mechanical power of 22.56 watts at 1370 RPM as seen in *Figure 10*. At 655 RPM the theoretical mechanical power is about 9.2 watts. With the engine not running with any applied load, it is safe to assume these new theoretical results are still higher than the actual results. This lack of power is likely due to air loss between the cylinders and the column.

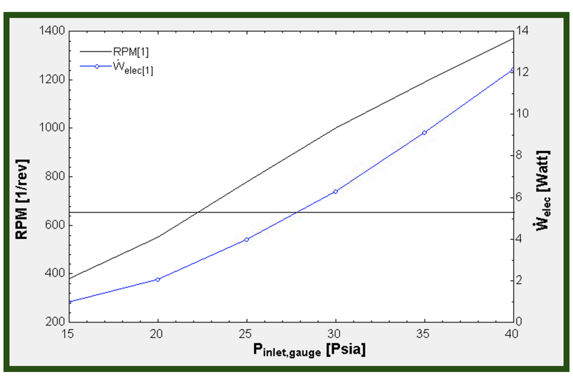


Figure 9: Theoretical Electrical Power vs. Pressure Plot

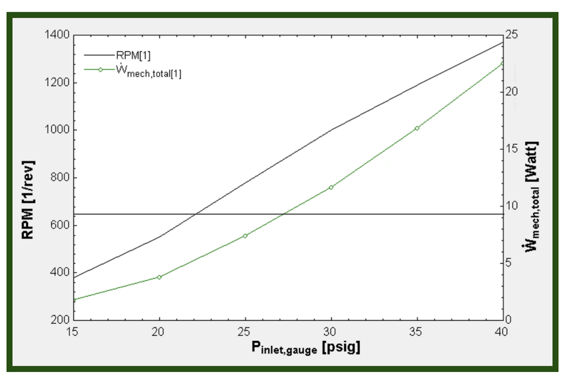


Figure 10: Theoretical Mech. Power vs. Pressure Plot

With the theoretical electrical power and mechanical power, if the design was perfect it should perform as expected. However, the compressor efficiency is relatively low, having an efficiency of 13.4 % at 1370 RPM seen in *Figure 11*. This is also reflected in the theoretical electrical efficiency, with the efficiency being 7.2% at 1370 RPM seen in *Appendix B1*. This is likely due to increases in air loss and an increase in turbulent flow inside the ports. It is interesting to note that this is not reflected in the mechanical efficiency viewed in *Figure 12*. However, it is important to note that the EES code predicts an efficiency of 120%. With this number being unrealistic, it is believed that there is an error in either the piston force calculations or the total mechanical work calculation. The team could not find a specific reason as to why these values are not within a realistic range. This error is isolated to the parametric table, as running the code by itself returns an efficiency of 48.4% at 40 PSI seen in *Appendix B5*.

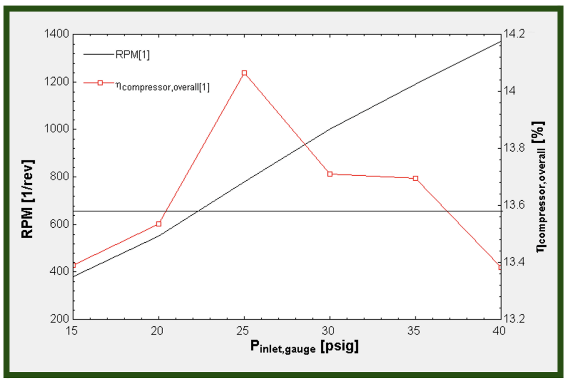


Figure 11: Theoretical Overall Compressor Efficiency vs. Pressure Plot

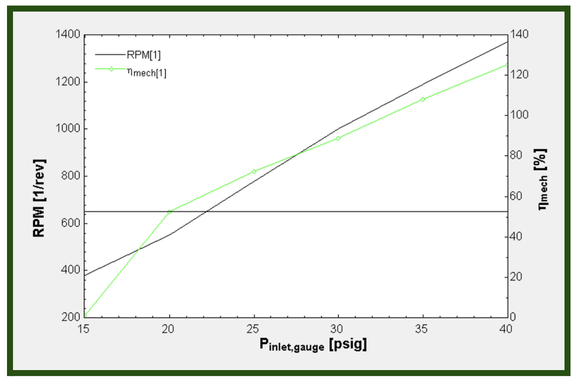


Figure 12: Theoretical Mech. Efficiency vs. Pressure Plot

All of the EES critical work and efficiency values can be viewed below in *Table 1*. If the engine had performed at 1370 RPM at 40 PSI, the engine would create 22.56 horsepower with a mechanical efficiency of 48.4%. According to this data, the machine should have functioned relatively well and should have been able to function with an applied load. However, with 175 cfm of air leakage, the engine is only using 14.65% of the provided air. This higher air loss was likely compounded by the imperfections of machining, meaning a small amount of air was being used to generate power. Furthermore, with the machine only running at 655 RPM, there are likely some port timing issues as well as excessive air loss.

Table 1: EES Array of Calculated Values

RPM (1/min)	Inlet Pressure (psig)	Electric Work (W)	Mechanical work (hp)	Mechanical Losses (hp)
1370	40	12.18	22.56	24.05
Air Leakage (cfm)	Mechanical Efficiency	Thermal Efficiency	Compressor Efficiency	Electrical Efficiency
175	48.4%	32.72%	13.38%	7.36%

5 Evaluation of Prototype as Compared to Project Performance and Requirements Document

Upon the team's initial test day, the air engine ran at about **700 RPM** and did **not** support any load. The team deemed this to be due to excess air loss between the piston and cylinder interface. The piston hole in the cylinder was initially made using a boring rod, allowing for the hole to be the exact size of the piston. However, this technique introduced some inaccuracies, with the bore being difficult to precisely measure. The team decided to remake the cylinder and sand down the pistons to fit a new 0.501-inch reamed hole in the new cylinders. This resulted in a much tighter tolerance between the piston and cylinder - approximately 0.002”. This helped improve the air loss coming from the cylinders, however the engine still did achieve the expected speed. Upon retesting a maximum speed of 655 RPM was achieved at 40 PSI.

5.1 Points of Failure

Upon disassembly and analysis of the engine, the team discovered several potential flaws that resulted in the engine not performing as expected. The first issue discussed was the port timing of the column and cylinders. The team also found the machine to perform better with the exhaust ports restricted, causing more back pressure. Lastly, the team wonders how the chamfers on the intake and exhaust port may have affected the performance of the engine. Each flaw will be discussed in more detail below.

5.1.1 Port Timing: Design Flaws

The port timing ultimately was the demise of this air engine design. Despite the forethought that went into the development of the port timing innovation, from wireframe model analysis to in-lab testing, there was an evident plateau that was being reached at nearly the same speeds repeatedly. As the ports were evaluated, the main goal was to cycle air through the engine as if it were more of a pump. By moving the ports to different locations along the cylinder’s oscillation, the team was able to roughly split the intake & exhaust strokes to compose nearly the entirety of the “power stroke”. By viewing *Figure 3*, the intake stroke was receiving air for 53.42% of its rotation by way of port alignment with the column. On the exhaust stroke, the team could only achieve a port alignment of 42.11% - where 151.5° of the 360° rotation was solely for the piston to push out the air. The “wasted” degrees of rotation, where the cylinder didn’t align with any port at all, was reduced to only 16%, therefore leading the team to believe that the new design would be infinitely more efficient at cycling air at high speeds.

Unfortunately, this ended up not being the situation for the team. The port timing contributed to several issues, specifically everything related to the cycling of air in the engine. The original port design from Elmer’s drawings [1] kept both ports directly next to each other, aligned horizontally. This is now understood as the best method to make use of the air’s fluid motion as it passes through the engine. As the compressed air enters, the “pulse” of air simply shoves the piston high enough to rotate with the help of momentum and its corresponding piston on the other side of the column. As the team understood, it was hoped that the air would “flow” - pushing the piston up continuously through the intake cycle, and immediately be dumped out the rear of the engine. For this reason, the new port design hindered the engine’s performance. The flow of air simply blew past the piston for too long and escaped between the piston/cylinder interface, resulting in a momentary stall in momentum as the piston struggled to make the downstroke. Given that Elmer had also placed the ports much closer together in the rotation, the distance between strokes in the team's port design was far too large for momentum to make up the gap. As seen in *Figure 7* and *Figure 8*, the ports are diagonally drilled concerning each other. Despite placing two corresponding holes on the cylinder to match those column ports, the gap in rotation was simply too long and the distance too great. Overall, had the ports been kept side-by-side as Elmer envisioned, utilizing the pulses of air would've been far more efficient for the motion of the engine's rotation.

5.1.2 Discovery of Back Pressure Complications

With the machine running at any pressure, it was noted that when the exhaust was restricted, the engine would maintain a noticeably higher RPM. The team suspects that this may indicate a smaller or more restrictive exhaust system will improve the performance. It is possible that having one intake port allows for the back pressure of the system to be significantly lower than the EES calculated. This would be due to the connected geometry of the ports and their opposing cycles. The EES calculations would likely be more accurate if each cylinder intake and exhaust acted as independent open systems. With the intake system being connected, there was likely an interface between the two intake cycles that EES did not account for.

5.1.3 Chamfer Inaccuracies

The team ended up adding chamfers on all of the intake and exhaust ports. This was done to account for any potential inaccuracies with the placement of the ports between the column and the cylinders. The team initially thought this would act as a diffuser and a nozzle in series, canceling the effect of both out. However, this likely resulted in a higher air loss between the column and the cylinder. The team could not figure out how to simulate this port geometry and thought any effects of it would be negligible, therefore it is not reflected in the EES code. This likely resulted in the engine having a significant decrease in power.

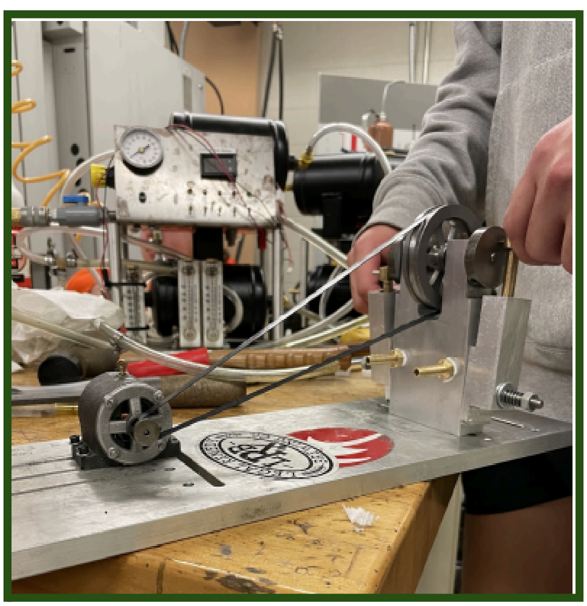


Figure 13: Air engine on dynamometer test bed for final testing.

6 Recommendations for Further Development

For this section, the team will suggest what should be improved with this particular air engine design. This will be an in-depth analysis concerning three specific issues that were found. All three of the following design evaluations should be improved for this design to run properly and without fail. Again, the engine developed by the team did not run as anticipated, leaving much to learn from this failure. Failure is a harsh word but unfortunately is the best description of the cumulative semester's work to develop this engine. The recommendations that are mentioned in this section are the most prevalent to the failure of The Heat's air engine, and a strong word of warning for those who wish to create their own port timing design. The following information is the result of hours of analysis, data interpretation, and failed tests - that being said, it is an excellent starting point for those who might wish to imitate this design.

6.1 Further Analysis of Port Timing Design

The first recommendation for further development would be the revaluation of the port timing design. The team attempted to innovate in a way not previously done and took a large risk in doing so. For this particular design, the portholes drilled into the column were expected to assist in the quick & efficient cycling of air through the engine. Unfortunately, these exact deviations from Elmer's original design are the chief reason that the engine did not run in the fashion the team had designed it to. Excessive air loss was observed between the interface of the cylinders & column, missing its intended target and resulting in a poor air seal that matched properly but failed to achieve the results the team had expected. These ports were a good idea at the time and theoretically, it could work, but it did not in this case. Despite analysis of diagrams and attempts to maximize power gains, the ports ultimately failed to deliver air in the same theoretical method that the team had hoped for. This shortcoming can be explained best by how compressed air moves through the engine; not as a flow of pressure, but rather as a short pulse that forces the piston upward briefly. The addition of the ports, in summation, was a solid & well-thought-out innovation that was intended to maximize performance, yet fell short of expectations due to an improper understanding of the working fluid contained within the engine.

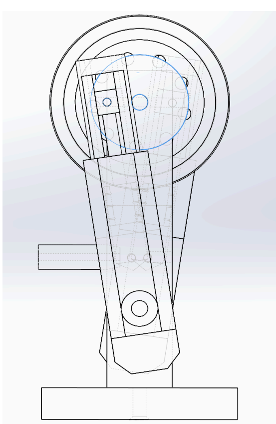


Figure 14: Wireframe of Rev.1 Port Timing.

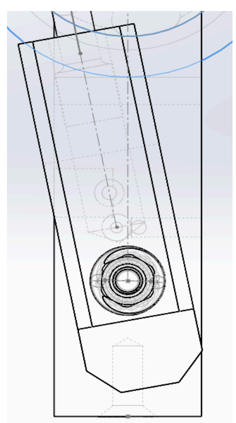


Figure 15: Close-up Wireframe of Rev.3 Port Timing.

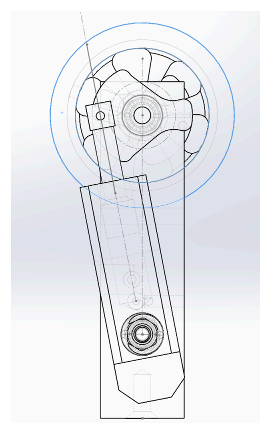


Figure 16: Expanded Wireframe view of Rev.3.

6.2 Specific Methodology for Machining Practices

When machining the cylinders and pistons, they must have a minimal amount of play once they are assembled. Play is the amount of wiggle room between the piston and the cylinder at the piston's lowest point. This is very important because the air loss between the piston and the cylinder can be the difference in how effectively that piston-cylinder assembly operates. If air can escape from the cylinder, the engine will lose a significant amount of power. This play came from using a bore on the milling machine to create the piston hole in the cylinders seen in *Figure 18*. This was one of the largest issues that resulted in the team having to make a brand-new cylinder with minimal time to spare. The team recommends that when machining the cylinder, drill and ream the hole for the piston in the cylinder first seen in *Figure 17*. Once the hole is reamed, cut down the piston to a slightly larger diameter than the reamed hole, and sand down the piston, checking the fit frequently to ensure a tight tolerance fit. This method gave the team a much tighter fit between the cylinder and column, with minimal play and air loss, and is recommended over the use of a bore.

Figure 17: 0.501" oversize reamer.

Figure 18: Boring bar & accessories.

6.3 Port Diameter, Chamfering

When machining the intake and exhaust ports, it is recommended that the ports are not chamfered. The team chamfered the intake and exhaust ports to account for potential inaccuracies as the cylinder follows its oscillatory path against the side of the air column. Instead, the chamfers contributed to more air loss between the column and cylinders and were difficult to calculate the effects of. To ensure minimum air loss, the column and cylinder ports need to align perfectly and remain the same size between the two parts. When designing the exhaust ports, it is important to consider what port size will suit the engine the most. The team believes the engine's exhaust ports were too large, as restriction of the exhaust would result in better performance. The idea of an adjustable exhaust may allow for finer tuning of the intake and exhaust cycle. This could be achieved by the addition of a valve allowing for fine-tuning of the amount of air restriction that occurs in the exhaust ports.

7 Conclusions

As the semester-long project comes to a close, several valuable lessons have emerged from this experience. The collaborative process, characterized by weekly meetings and constant communication, proved effective in navigating challenges and sharing ideas. Team meetings were held often and sometimes multiple times a week between team members responsible for unique components. Documenting the process as the semester continued was vital to writing reports and intermediate design reviews, made most useful by maintaining a server collectively in Discord. All images taken during the development process were kept in this Discord, allowing a much more accessible trove of documentation for future use. In addition to documenting the team's communication, a log of all meeting notes was kept for reference between class periods and group voice calls. In the area of engine development, the team learned the importance of allowing ample time for testing and adjustments, avoiding last-minute rushes to accommodate unforeseen machining errors with critical engine components. The time-management portion of the project was neither amazing nor awful - more simply put, the team was consistently swamped with work from other courses that frequently hindered progression. Critical meetings had to be shortened, hours spent in the machine shop were interrupted, and an overall emphasis on the importance of the project was lost in the flurry of chaos that was junior year of college.

While the ultimate outcome didn't align with the projected expectations, the risk taken was a worthwhile attempt to explore a novel idea. With the initial hope being that the team's engine would perform flawlessly and the eventual dwindling of that expectation over time, the consensus is that the team's time spent was still well worth the effort. Exploring the possibilities offered by such an interesting project, the chance to flex a problem-solving mindset, and an opportunity to practice vital machining skills ultimately comprise the team's feelings about the project as a whole. Despite the disappointment, the failure of the team's interpretation of Elmer's Twin Wobbler Engine provided valuable insights, which The Heat hopes will benefit future teams.

8 References

[1]
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[2]
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[5]

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9 Appendices

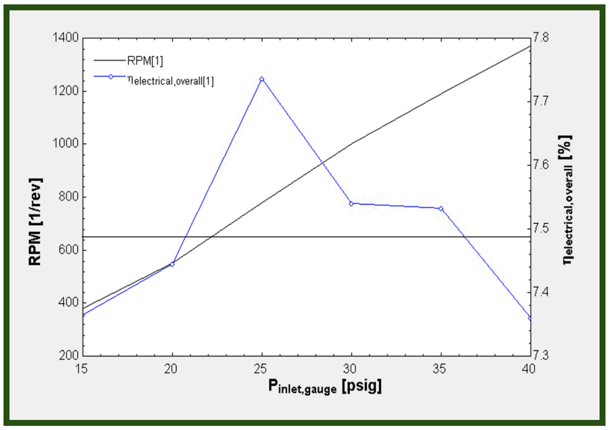
Appendix A: Division of Duties

ETME 3100						
	Max Petersen	Andrew Karls	Dylan DiQuarto	Ethan Lentz	Curtis Miller	Total* (should equal 100%)
PDR Presentation	20%	20%	20%	20%	20%	100%
Air motor drawing Set	60%	20%	5%	5%	5%	100%
Dyno test bed drawing set	60%	20%	5%	5%	5%	100%
Instrument panel drawing set	60%	20%	5%	5%	5%	100%
Project plan	10%	60%	5%	20%	5%	100%
*EES Model**	100%	100%	33%	33%	33%	300%
Progress Reports	20%	20%	20%	20%	20%	100%
Performance Specs	20%	20%	20%	20%	20%	100%
PSR Presentation	20%	20%	20%	20%	20%	100%
Final Project Presentation	20%	20%	20%	20%	20%	100%
Final Design Package	25%	25%	25%	25%	25%	100%
Project plan	10%	60%	5%	20%	5%	100%
*EES Model**	35%	50%	5%	5%	5%	100%

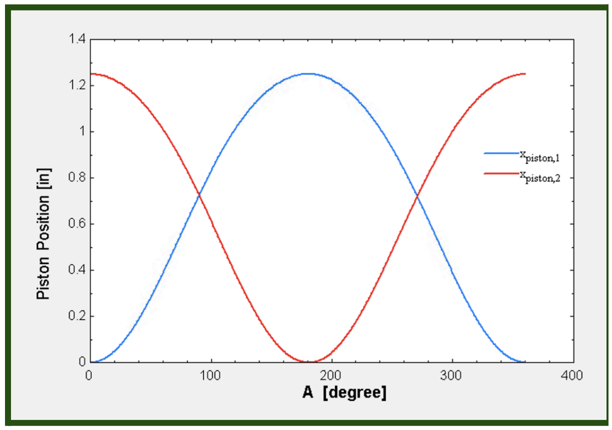
**Total for EES assignment will exceed 100%*

ETME3100L						
	Max Petersen	Andrew Karls	Dylan DiQuarto	Ethan Lentz	Curtis Miller	Total (should equal 100%)
Air motor fabrication	20%	20%	20%	20%	20%	100%
Dyno test bed fabrication	0%	75%	0%	0%	25%	100%
Instrument panel fabrication	0%	0%	0%	0%	0%	100%

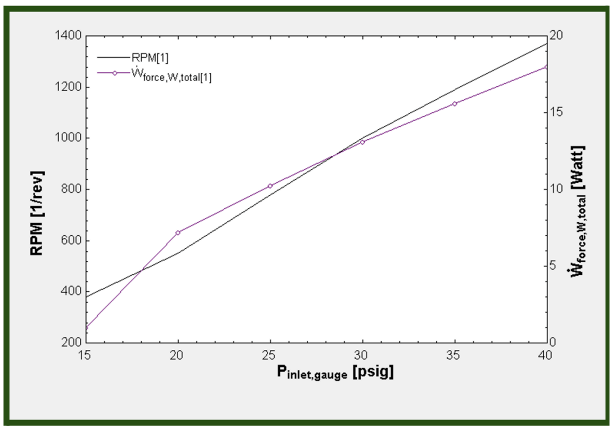
Appendix B: EES Graphs and Calculated Values



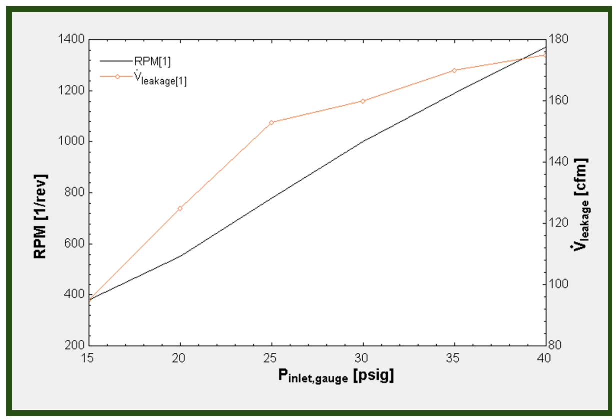
Appendix B1: Theoretical Electrical Efficiency vs Pressure Graph



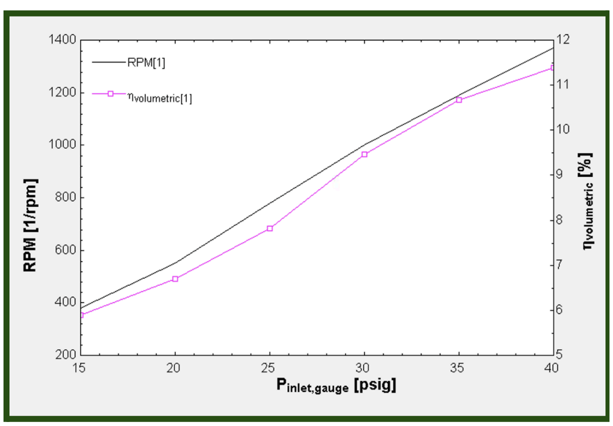
Appendix B2: Piston Position graph



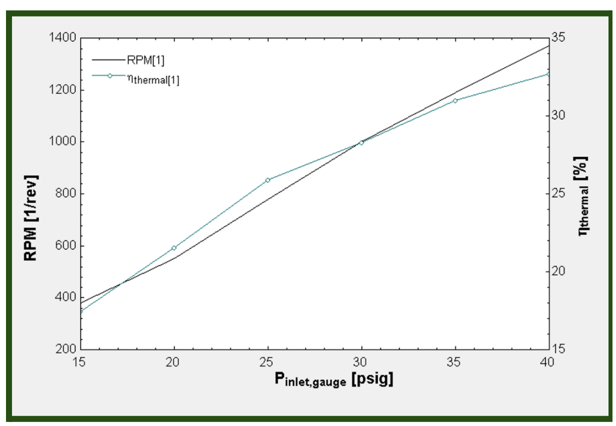
Appendix B3: Theoretical Power produced



Appendix B4: Theoretical volumetric flow leakage



Appendix B5: Theoretical volumetric efficiency



Appendix B6: Theoretical thermal efficiency

Unit Settings: Eng F psia mass deg

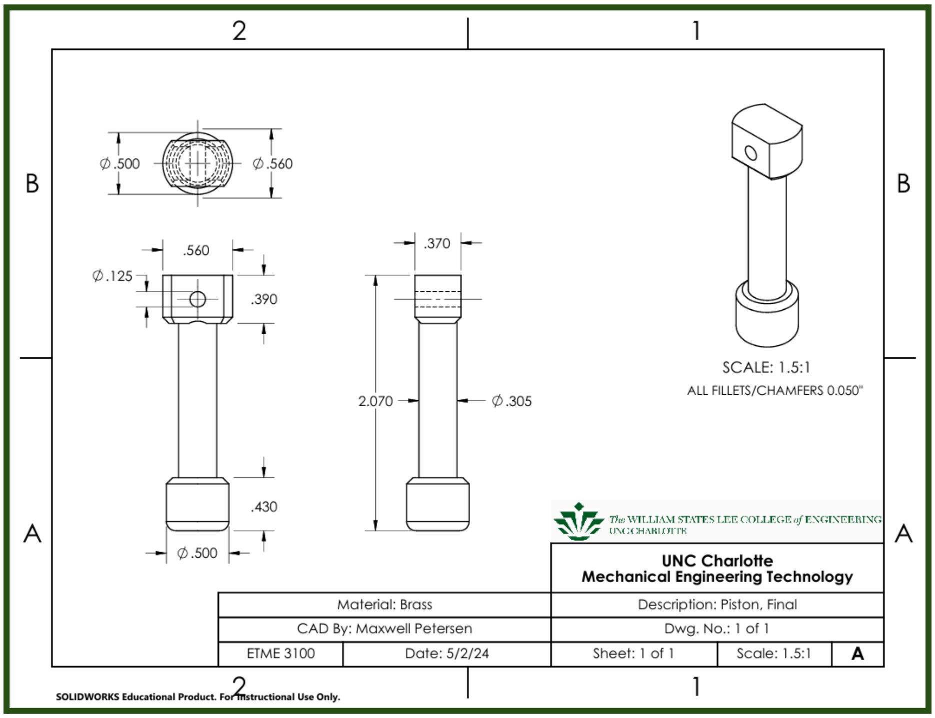
A = 180 [degree]	$Area_{exit}$ = 0.01227 [in^2]	$Area_{inlet}$ = 0.01227 [in^2]
C = 180 [degree]	$cylinder_{port,size}$ = 0.125 [in]	D = 360 [degree]
$\delta_{P,exit}$ = 0.9403 [psia]	$\delta_{P,inlet}$ = 0.2625 [psia]	Dia_{piston} = 0.5 [in]
$\eta_{compressor,overall}$ = 13.38 [%]	η_{elec} = 0.6 [-]	$\eta_{electrical,overall}$ = 7.36 [%]
η_{mech} = 48.4 [%]	$\eta_{thermal}$ = 32.72 [%]	$\eta_{volumetric}$ = 11.39 [%]
f_{exit} = 0.07407 [-]	f_{inlet} = 0.07407 [-]	g = 386.4 [in/s^2]
$h_{L,bend,exit}$ = 15512 [in]	$h_{L,bend,inlet}$ = 1120 [in]	$h_{L,contraction,exit}$ = 3284 [in]
$h_{L,expansion,inlet}$ = 237.2 [in]	$h_{L,inlet}$ = 268.6 [in]	$h_{L,inlet,total}$ = 1626 [in]
K_{bend} = 4.152 [-]	$K_{contraction}$ = 0.8789 [-]	$K_{expansion}$ = 0.8789 [-]
L_{exit} = 0.64 [in]	$L_{exit,total}$ = 1.3 [in]	L_{inlet} = 1.02 [in]
μ_{air} = 0.000001025 [lbm/in-s]	$\dot{m}$ = 0.000904 [lbm/s]	$\dot{m}_{total}$ = 0.001808 [lbm/s]
$P_{cylinder,1}$ = 54.44 [psia]	$P_{cylinder,2}$ = 15.64 [psia]	$P_{cylinder,high,abs}$ = 54.44 [psia]
$P_{inlet,gauge}$ = 40 [psig]	rel_{rough} = 0.048 [-]	Re_{inlet} = 8983 [-]
ρ_{inlet} = 0.0001613 [lbm/in^3]	RPM = 1370 [1/min]	stroke = 1.25 [in]
$Torque_2$ = 1.659E-07 [lbf-in]	$Torque_{avg,1}$ = 1.421 [lbf-in]	$Torque_{avg,2}$ = 1.454 [lbf-in]
$Volume_{cylinder}$ = 0.2454 [in^3]	$\dot{V}_{cylinder}$ = 5.604 [in^3/s]	$\dot{V}_{cylinder,cfm}$ = 0.1946 [cfm]
$\dot{V}_{outlet}$ = 30 [cfm]	$\dot{V}_{total}$ = 0.3892 [cfm]	$\dot{V}_{total,cfh}$ = 23.35 [ft^3/hr]
$\dot{W}_{elec}$ = 12.18 [W]	$\dot{W}_{force,1,hp}$ = 0.03089 [hp]	$\dot{W}_{force,1,W}$ = 23.04 [W]
$\dot{W}_{force,hp,total}$ = 0.06249 [hp]	$\dot{W}_{force,W,total}$ = 46.6 [W]	$\dot{W}_{hydrostatic}$ = 68.94 [W]
$x_{piston,1}$ = 1.25 [in]	$x_{piston,2}$ = 3.267E-09 [in]	$x_{p,1}$ = 1.375 [in]

Appendix B7: Table of Theoretical EES values, calculated at 40 psi and 1370 RPM

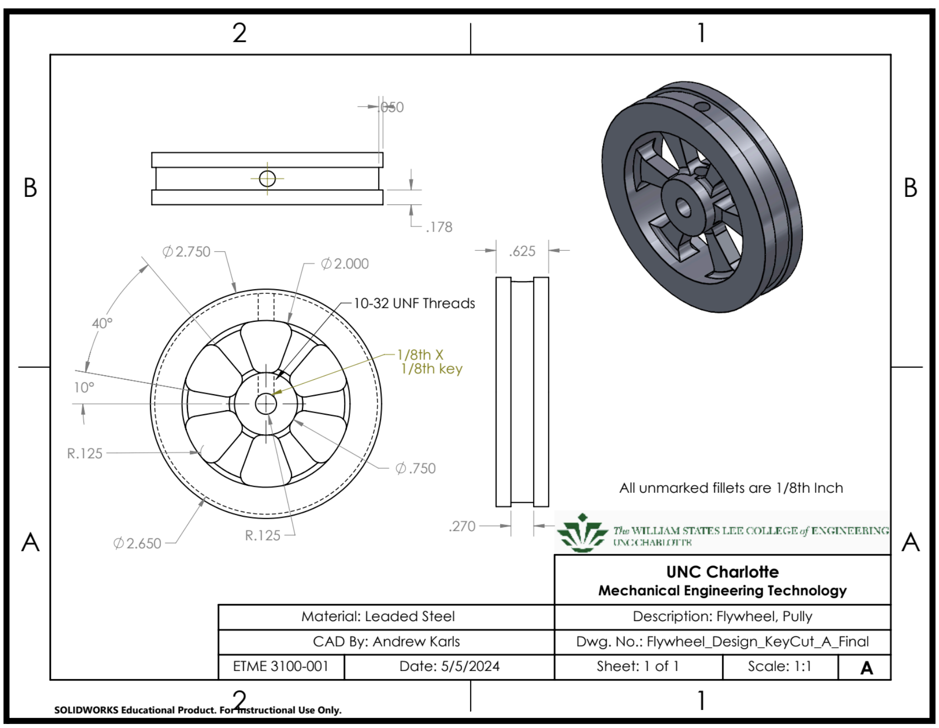
$Area_{piston}$ = 0.1963 [in^2]	B = 0.05023 [degree]
$Data_{row}$ = 6 [-]	$Data_{table\$}$ = 'Sample Data'
ε = 0.0005 [ft]	η_{belt} = 0.9 [-]
$\eta_{hydrostatic,overall}$ = 17.67 [%]	η_{iso} = 0.55 [-]
$Exhaust_{valve,closes}$ = 353.8 [degree]	$Exhaust_{valve,opens}$ = 202.2 [degree]
$\gamma_{P,exit}$ = 0.00004338 [lbf/in^3]	$\gamma_{P,inlet}$ = 0.0001614 [lbf/in^3]
$h_{L,exit}$ = 2878 [in]	$h_{L,exit,total}$ = 21675 [in]
$Intake_{valve,closes}$ = 192.3 [degree]	$Intake_{valve,opens}$ = 0 [degree]
L_{barb} = 0.5 [in]	$L_{cylinder}$ = 0.16 [in]
$L_{inlet,total}$ = 1.68 [in]	L_{rod} = 2 [in]
$number_{of,cylinders}$ = 2 [-]	P_{atm} = 14.7 [psia]
$P_{cylinder,low,abs}$ = 15.64 [psia]	$P_{inlet,abs}$ = 54.7 [psia]
Re_{outlet} = 8983 [-]	ρ_{exit} = 0.00004335 [lbm/in^3]
T = 70 [F]	$Torque_1$ = -0.005856 [lbf-in]
vel_{exit} = 1699 [in/s]	vel_{inlet} = 456.7 [in/s]
$\dot{V}_{inlet}$ = 205 [cfm]	$\dot{V}_{leakage}$ = 175 [cfm]
$\dot{W}_{compressed,air}$ = 91.02 [W]	$\dot{W}_{compressor,elec}$ = 165.5 [W]
$\dot{W}_{force,2,hp}$ = 0.0316 [hp]	$\dot{W}_{force,2,W}$ = 23.57 [W]
$\dot{W}_{mech,loss}$ = 24.05 [W]	$\dot{W}_{mech,total}$ = 22.56 [w]
$x_{p,2}$ = 2.625 [in]	

Appendix B8: EES Calculated Values at 40 psi and 1370 RPM, continued

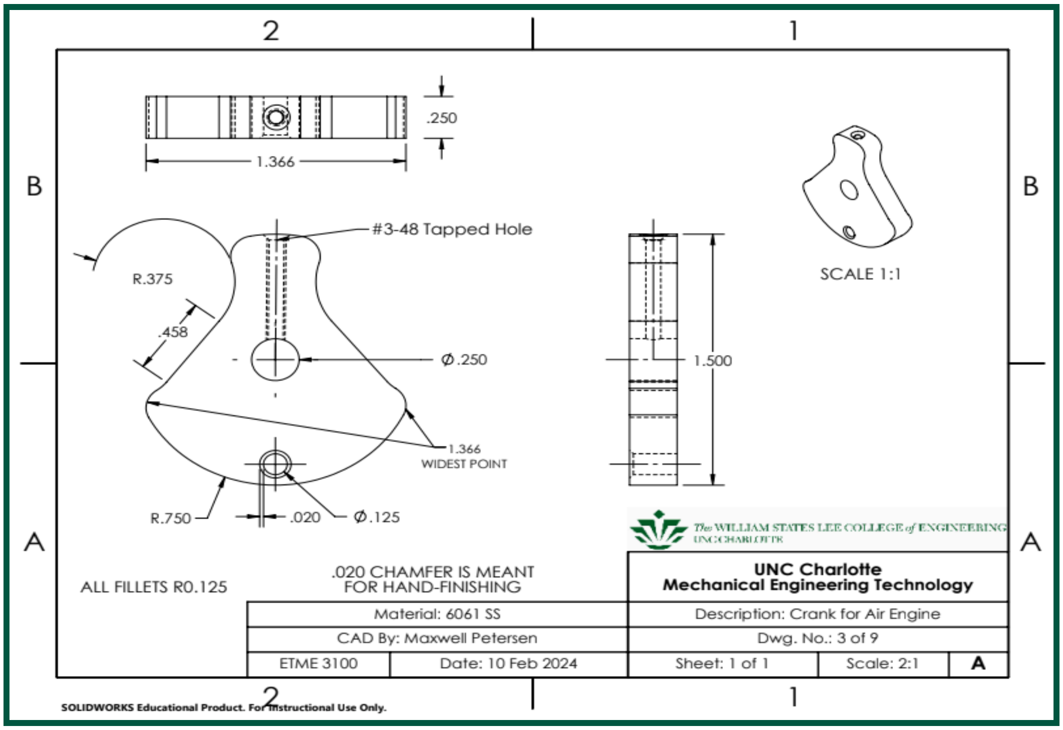
Appendix C: Drawings



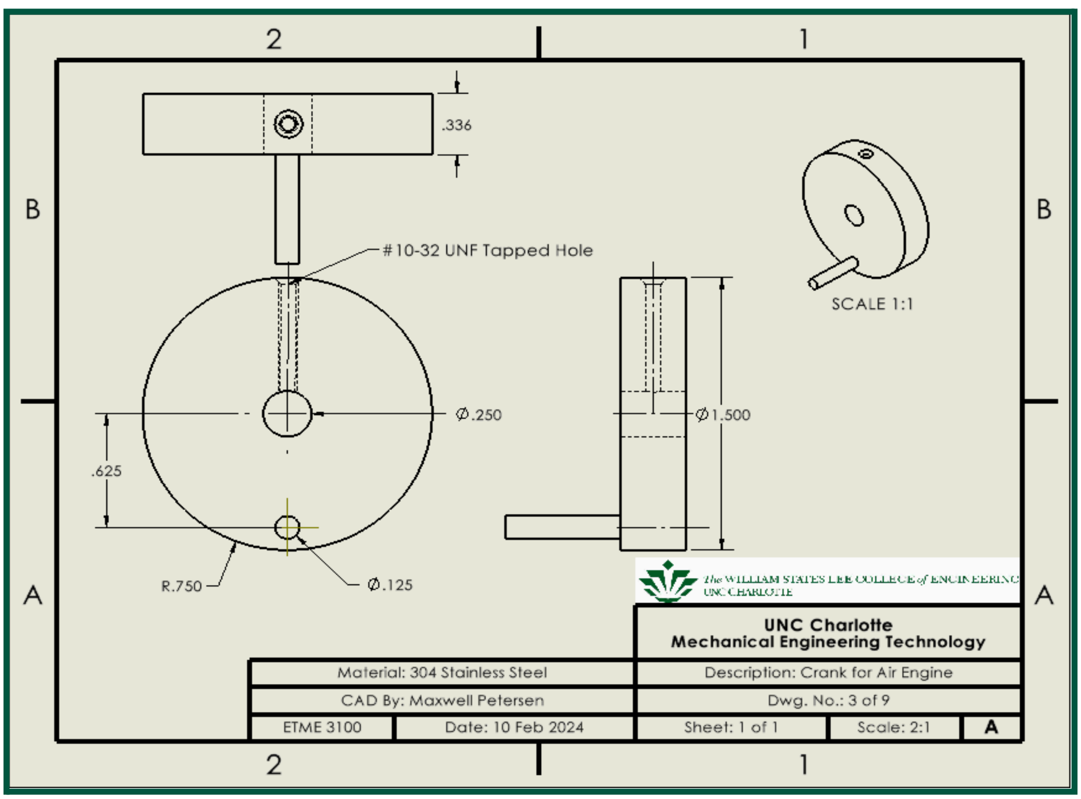
Appendix C1: Piston Part Drawing



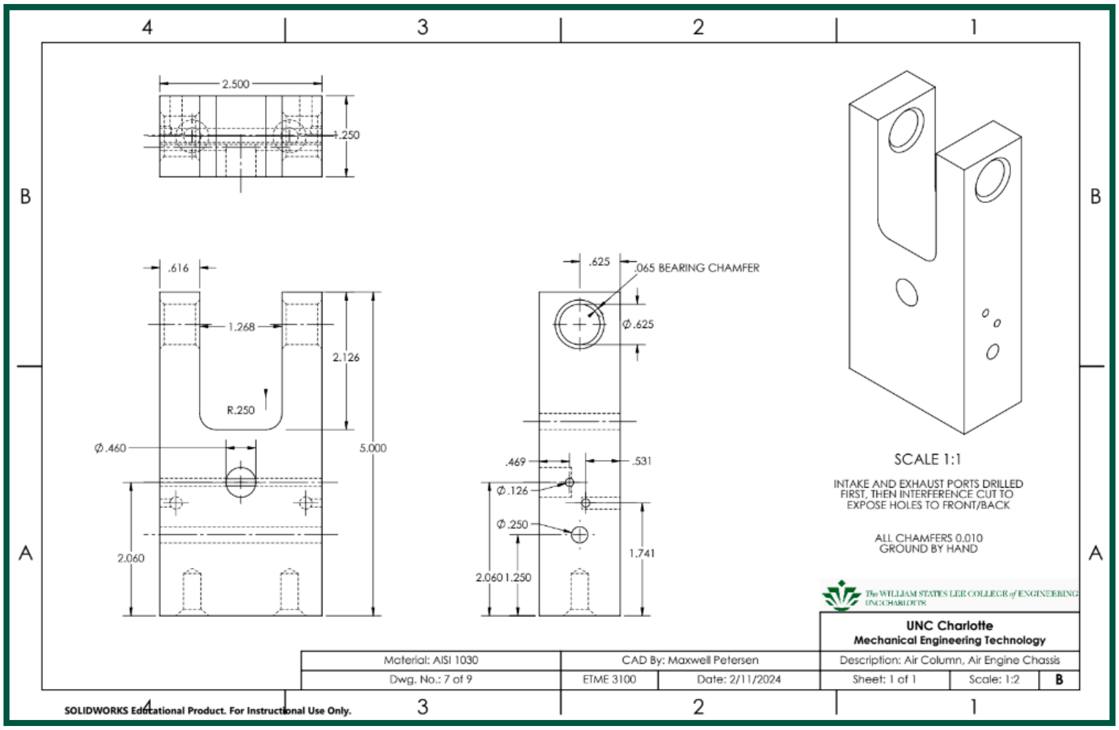
Appendix C2: Flywheel Part Drawing



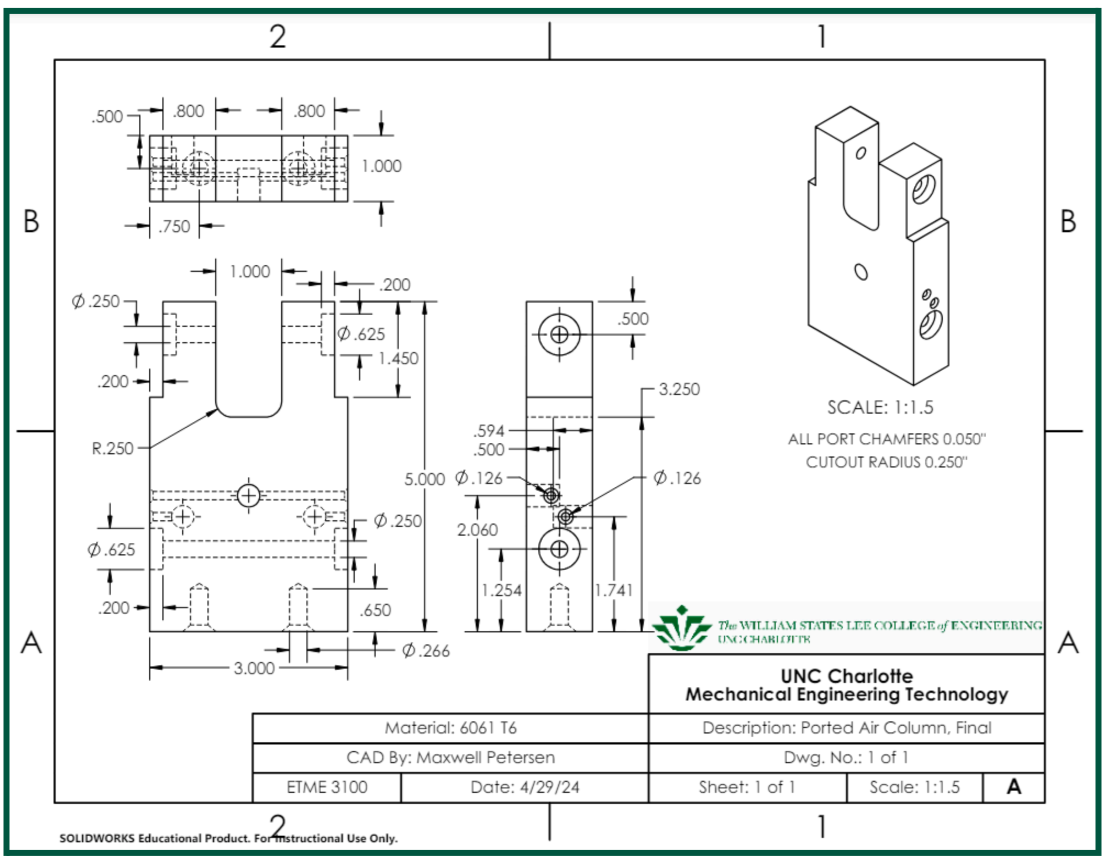
Appendix C3: Rev.1 Crank Design Drawing



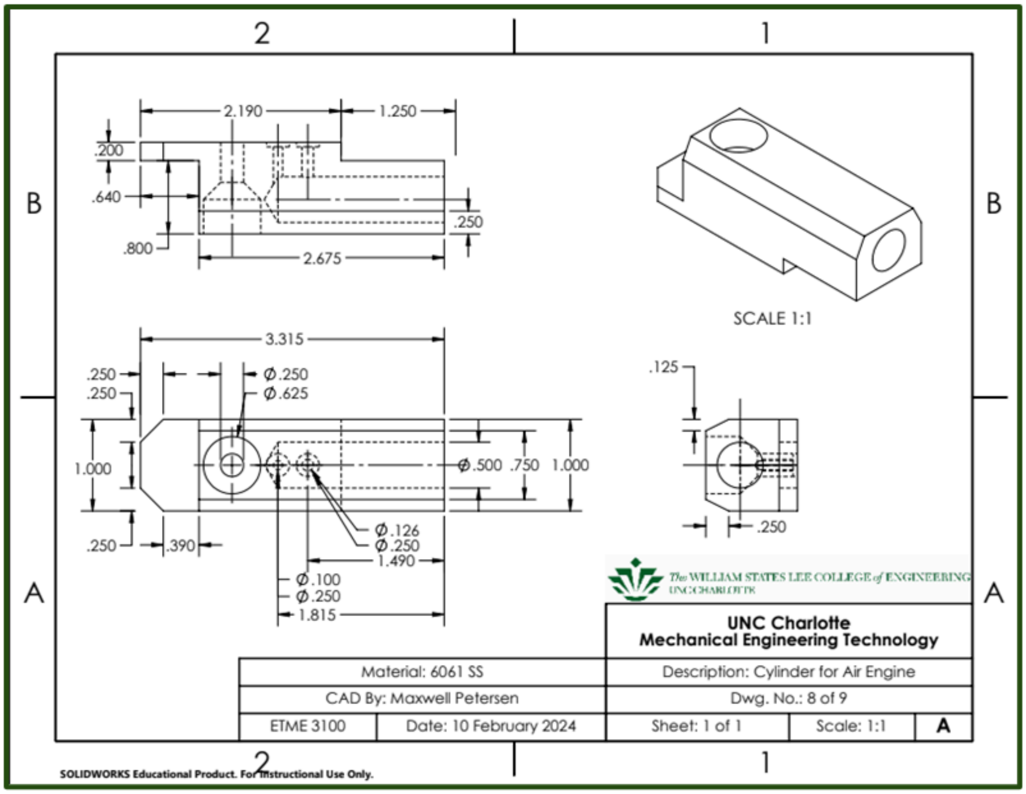
Appendix C4: Rev.3 Crank Design Drawing



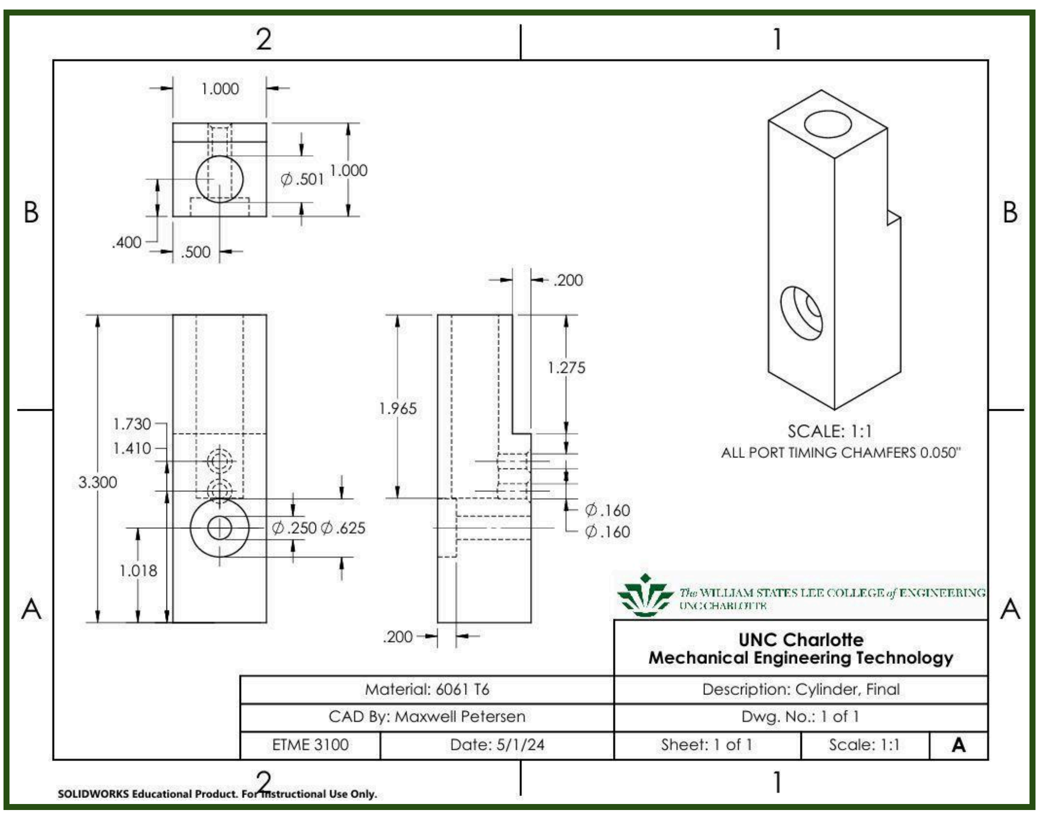
Appendix C5: Rev.2 of the Column Drawing.



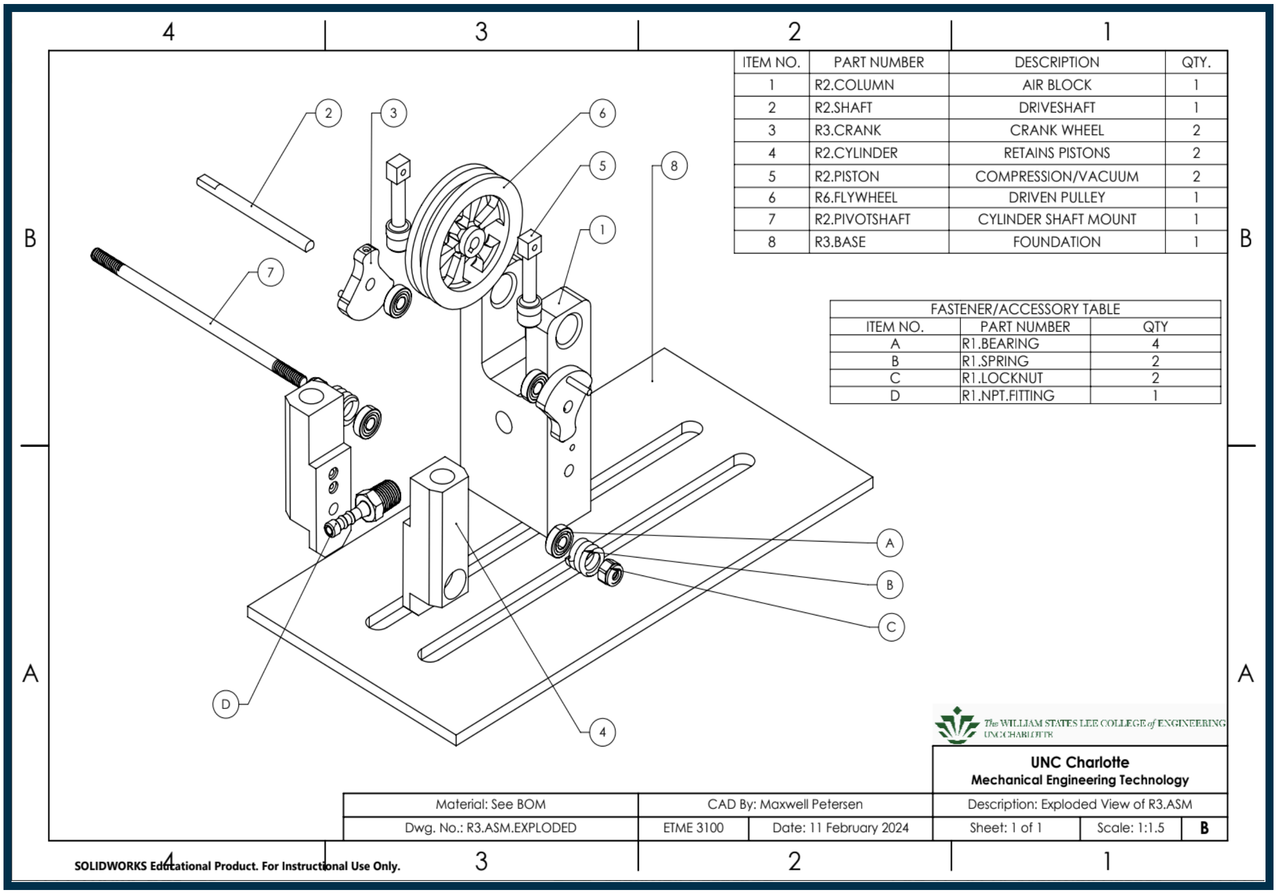
Appendix C6: Rev.4 of Column Drawing.



Appendix C7:Original Cylinder Design



Appendix C8: Final Cylinder Drawing.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	R2.COLUMN	AIR BLOCK	1
2	R2.SHAFT	DRIVESHAFT	1
3	R3.CRANK	CRANK WHEEL	2
4	R2.CYLINDER	RETAINS PISTONS	2
5	R2.PISTON	COMPRESSION/VACUUM	2
6	R6.FLYWHEEL	DRIVEN PULLEY	1
7	R2.PIVOTSHAFT	CYLINDER SHAFT MOUNT	1
8	R3.BASE	FOUNDATION	1

FASTENER/ACCESSORY TABLE

ITEM NO.	PART NUMBER	QTY
A	R1.BEARING	4
B	R1.SPRING	2
C	R1.LOCKNUT	2
D	R1.NPT.FITTING	1

Appendix C9: Final Assembly Drawing

Appendix D: Finished Parts

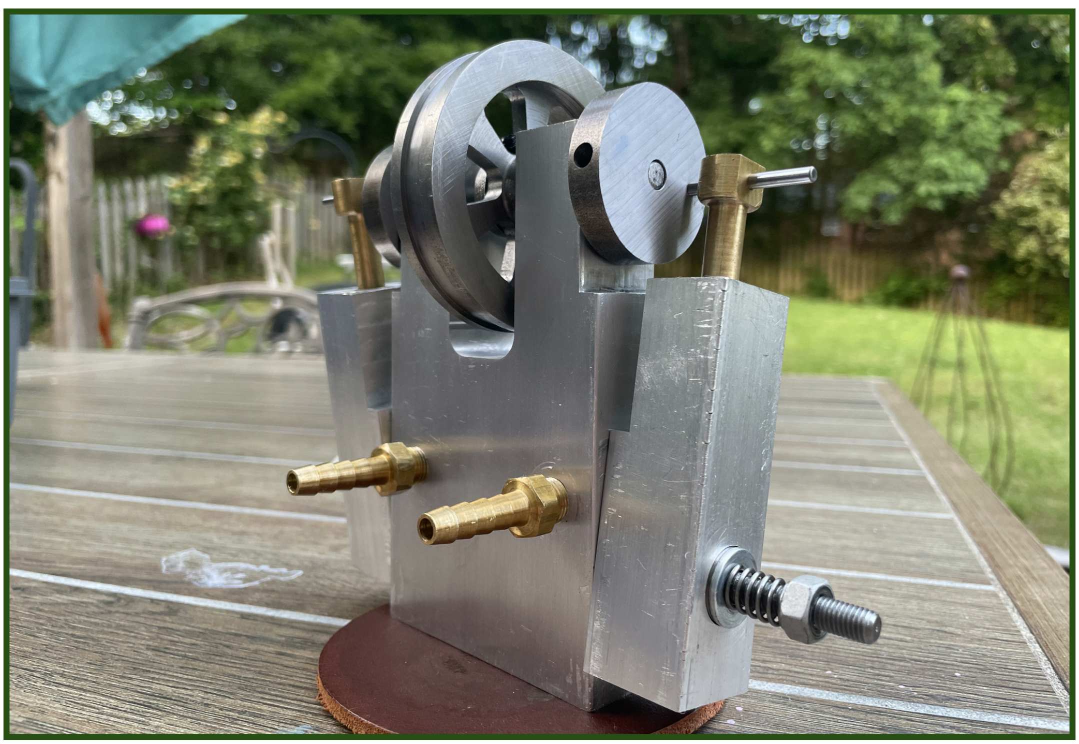
Appendix D1:Finished Machined Pistons, Rev.2.

Appendix D2: Finished Machined Flywheel, Rev.2.

Appendix D3:Finished Machined Crankshafts.

Appendix D4: Finished Machined Column, Rev.4.

Appendix D5: Completed Machined Cylinders, Rev.3.



Appendix D6: Completed Twin Wobbler Air Engine.

Appendix E: Budget and Hours of Labor

Category	Quantity	Cost ($)	Sub. Total ($)	Total ($)
Tool Box	2	$109.46	X	$218.92
BOM Items	26	$249.44	X	$249.44
Labor				$36,250.00
Students	325	$100.00	$32,500.00	
Professor	25	$150.00	$3,750.00	
			Overall Total ($)	$36,471.41

Appendix E1:Team Heat Budget

Appendix F: Hours for Each Member

Teammate	Hours
Maxwell Petersen	134.5
Andrew Karls	96
Dylan DiQuarto	93
Curtis Miller	94.5

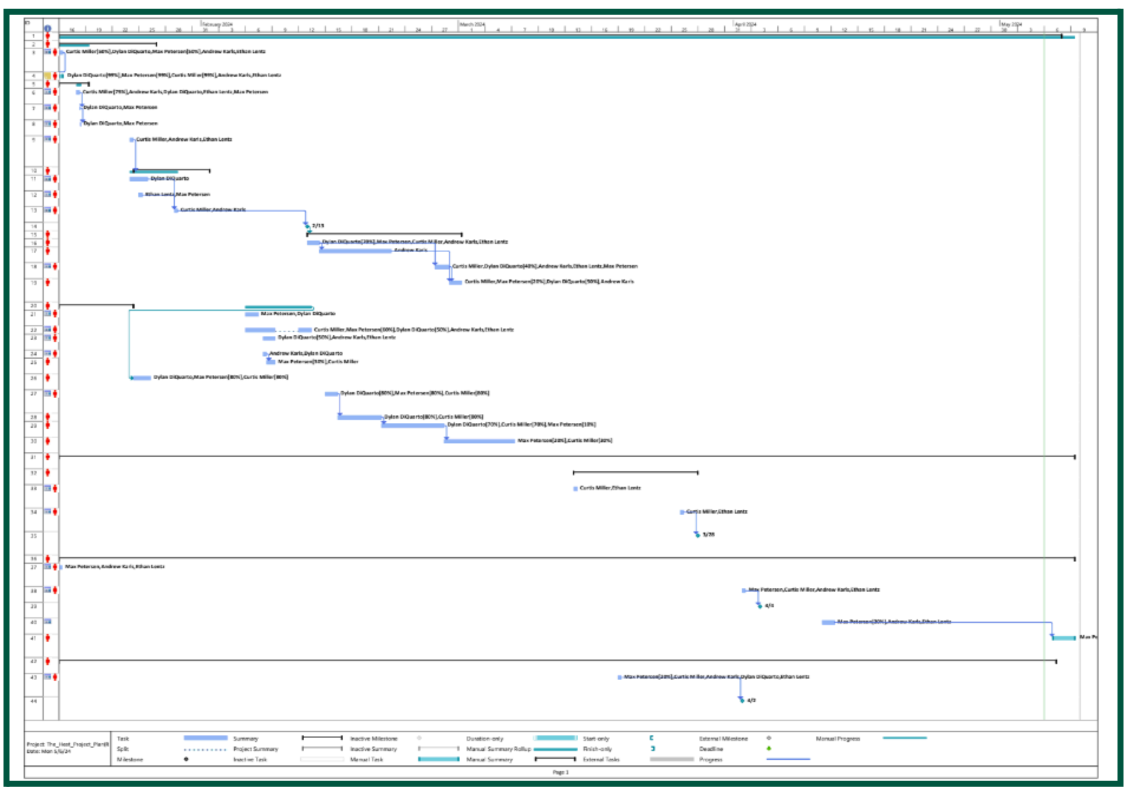
Ethan Lentz	102.5
Total:	520.5

Appendix F: Teams Total Logged Hours.

Appendix G: Bill of Materials and Gantt Chart

Quantity	Vendor PN	Vendor	Part	Material Description	We Have	We Need	Cost Per ($)	Total ($)
2	F4181	Metals Depot	Cylinders	6061 SS		X	$7.16	$7.16
2	F4182	Metals Depot	Crank	6061 SS		X	$7.16	$7.16
1	F4183	Metals Depot	Shaft	6061 SS		X	$7.16	$7.16
1	SQ112	Metals Depot	Column	AISI 1030		X	$5.08	$5.08
1	R13	Metals Depot	Pivot Shaft	AISI 1030		X	$73.75	$73.75
1	R13	Metals Depot	Flywheel	AISI 1030		X	$73.75	$73.75
1	N/A	N/A	Base Plate	Aluminum Housing	X		N/A	N/A
1	N/A	N/A	Engine Dynamometer	Aluminum Housing	X		N/A	N/A
2	95615A101	Mcmaster	Locknuts	Black-Oxide Steel Locknut		X	$0.07	$7.20
2	1526N11	Mcmaster	Pistons	Brass		X	$3.11	$6.20
1	5346K191	Mcmaster	NPT Fitting	Brass		X	$23.92	$23.92
1	N/A	N/A	Rubber Belt	Standard Rubber Belt	X		N/A	N/A
1	92313A842	Mcmaster	Set Screw 10-31	Stainless Steel		X	$0.45	$4.50
2	92311A263	Mcmaster	Set Screw 3-48	Stainless Steel		X	$0.14	$6.98
4	2342K183	Mcmaster	Ball Bearings	Steel Housing	X		$3.54	$14.15
1	5133A11	Mcmaster	Clamp	Steel Housing		X	$7.50	$7.50
2	965757K316	Mcmaster	Springs	Zinc-Plated Music-WIRE STEEL		X	$1.64	$4.93
							Total	$249.44

Appendix G1: Final Bill of Materials



Appendix G2: Gantt Chart

Appendix H: Change Form

UNC CHARLOTTE
The WILLIAM STATES LEE COLLEGE of ENGINEERING
Industrial Solutions Laboratory
9201 University City Boulevard, Charlotte, NC 28223-0001

ETME 3100 Engineering Change Request

Use this form to change the requirements, statement of work or deliverables after the Conceptual Design Review has been completed.

Date: 4/10/2024	Project Name: The Heat Junior Design	Supporter:

Description of Change Summary: The addition of shoulders to the column to accommodate the cranks and cylinders.
Reason for Change: Crank and cylinder were not lining up correctly and there was a gap between.
Cost Implications: Two hours more of machining so 200$.

Attach copies of documents changed when submitting this form for approval, e.g. Requirements document old and new.

APPROVALS/ DATES

Project Lead: Max Peterson	Project Mentor:	Supporter Technical Rep:
Grading Instructor: Dr. Hewlin		

Engineering Change Request　　　　Revision: A

Appendix H1: Column's Engineering Change Form Request.