

Project Tyr: Printed Jet Engine Technical Document

Q4 2025, Marco Schonert and Alex Boniske

Introduction

In September of 2025, Marco Schonert pitched me on the idea of building a jet engine over the course of the semester. We'd aim to design our own novel engine with mostly printed parts, economical for deployment in a UAV or single-use missile.

In the early 2000s, small engines saw a boom through the adoption of precision casting (Czarnecki, 2025). We based this project on the belief that developments in metal 3D printing could bring a similar expansion through reducing the cost of iteration and raising the ceiling of low-resource engine production. Setting a target cost of \$2,000 after preliminary research, we got to work.

Project Scope and Non-Mathematical Planning

Since our engine would be designed for a UAV/small missile, we decided to build a turbojet: an engine whose only thrust is produced by the exhaust (not additional fans driven by the turbine). This decision is standard for small-scale propulsion (Cumpsty, 2003, p.32). Further, we planned to use an axial turbine (where air flows parallel to the axis of rotation) similar to those designed by Advanced Micro Turbines (AMT, now AMT USA and AMT Netherlands). Historically, axial turbines weren't heavily adopted for small-scale engines due to difficulties with consistent fabrication. But seeing AMT's success with cast axial turbines encouraged us to do the same with our printed one. However for a first version, radial compressor is the way to go due to the high compression ratio achievable with only one stage of radial compressor, rather than many stages of axial compressor. Our compressor would be a radial design.

A quick aside: designing an engine at any scale requires tradeoffs between modeling, testing, and using others' results. There exists a spectrum of abstraction in design. On one end, you could create an engine by purchasing off-the-shelf components and assembling them, reducing the need to model your turbomachinery (higher abstraction); opposite of this end, you could arrive at your own custom parts from first-principle physics, guided through hundreds of prototypes (lower abstraction). Whenever possible, we took the path of low abstraction, only making high abstraction concessions when we wouldn't be able to create prototypes to guide our planning, and instead had to rely on others' historical results.

We planned to use kerosene for fuel due to its high volumetric energy density. Liquid kerosene is generally safer for small aircraft since it can be stored as a hard-to-combust liquid. However, this choice would require us to use propane to start our engine, raising the combustion chamber to a temperature high enough that the kerosene would properly vaporize and combust. Unlike AMT, we wouldn't be using inconel or precision casting since we'd decided to aim for a \$2,000 cost. We'd have less heat resistance and precision using printed stainless steel, so while we could take inspiration, our turbojet would require us to spec almost everything from scratch.

316L is the most standard and widely available printed stainless steel. Compared to inconel or more expensive aerospace materials, it will creep considerably. We were okay with this tradeoff due to our cost

target and intention to design for a short-life UAV application. A similar engine could be designed with precision-cast or printed inconel, but the cost would balloon significantly, and we just didn't care that much about theoretical lifespan.

We chose to shroud our engine's impeller because the efficiency gains are considerable, and this would eliminate the need for a high tolerance small gap that would otherwise exist between the housing and the impeller blades. Moreover a shrouded impeller makes assembly easier. This thinking led to Marco supporting this idea before we did much research of the hobbyist precedent. It turns out, the use of shrouded compressors in hobbyist turbojets was established by Kurt Schreckling over 30 years ago on his "Feuer -Dosen gas canister" series of engines (notably the FD3-64), and his published work and results supported the decision to cover the impeller (Kamps, 2002, p. 24). Given that we were using a radial compressor, we'd need to carefully consider the tip angles of the blades. In theory, radially tipped blades would achieve a higher pressure rise, but in practice, they are much more sensitive to disturbances in pressure and rough surfacing at a small scale. Based on the results of Thomas Kamps, we chose to pursue a compressor with retro-curved tips (backwards angled), meaning that down the line, we'd have to calculate the ideal angle and simulate the airflow in a fluid simulation. For now, we planned to make the impeller our only aluminum part since it would operate at a much lower temperature than the turbine, and aluminium has an excellent strength to weight ratio, which is essential when centrifugal forces from its own weight are trying to rip the impeller apart.

For our rotational assembly, we planned to use unshielded uncaged high rpm hybrid ceramic bearings actively lubricated with oil and cooled by air from the compressor. The turbine-side bearing would be preloaded with a wave spring to accommodate thermal expansion of the shaft and housing, similar to the KJ66 designed by Kurt Schreckling and Jesus Artes in the 1980s.

Modeling and Math Driving the Engine

We designed our engine with a turbine-first approach. This means that calculations began with the material's maximum operating rpm and temperature (in our case stainless steel 316L), deriving all other measurements from there. Alternative approaches exist, notably the compressor-first approach, which is best used for designs based upon an existing (and well-tested) impeller. Thomas Kamps employs this approach in his guide "Model Jet Engines," which we relied on for other design assumptions and features.

We began our math with what we assumed would be the most limiting parts of our engine's physics: our turbine's blades under intense operating conditions. Based on the printing fees at JLC3DP, we identified 316L steel as the most affordable option for the turbine, and also the most machinable. Since the material would be printed, rather than traditionally machined, we needed to find a report of its mechanical properties. A 2020 study from Oak Ridge National Laboratory found that SLM 316L has a steep drop in tensile strength when operating over 600 °C, so we drafted our first calculations with a maximum temperature of 600 °C at the turbine inlet (the hottest point). The study placed printed 316L at a yield strength of 280 MPa, which we would use with a substantial safety factor. Further research suggested that SLM 316L has a nearly identical density to standard 316L, so we calculated using 7980 kg/m³. For our turbine and compressor, we assumed a contemporarily modest 75% efficiency based on Schreckling's 30-year-old recommendation. We also constrained turbine blade sizing and count to Schreckling's results, planning for seventeen blades to extend 1/6 of the total turbine diameter on each side.

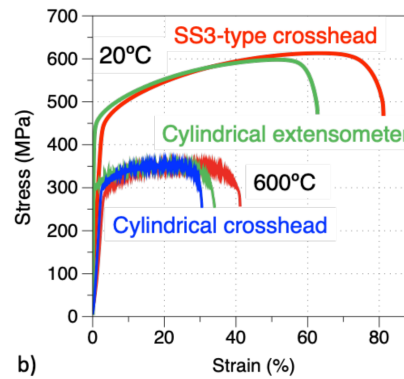
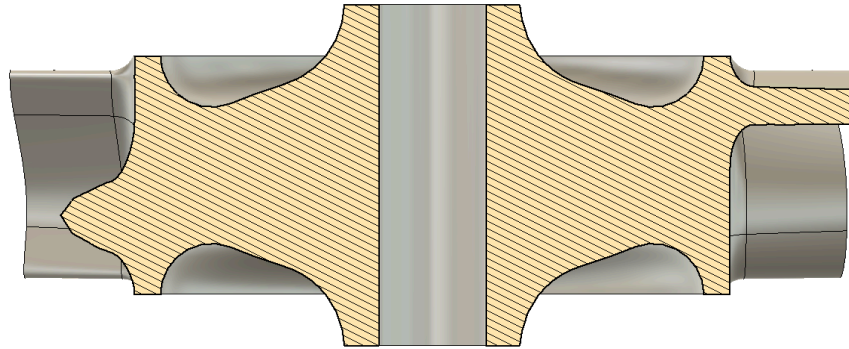


Image 1: Stress-strain of SLM printed 316L steel (Dryepondt et al., 2021)

For preliminary math before ANSYS design, we create a model to understand the maximum non-destructive rpm at various turbine diameters. Substituting the centrifugal force equation into the stress equation, we can derive an equation for peripheral speed at the middle of the blade:



$$\sigma = \frac{F}{A} \mid F = \frac{mv^2}{r} \Rightarrow \sigma = \frac{m \cdot v^2}{A \cdot r} \Rightarrow v = \sqrt{\frac{\sigma \cdot r}{m/A}}$$

Treating r as the middle of the blade (this is relatively accurate since our blades will be a constant thickness without much taper to be best printed in flat layers), m/A as the mass per unit area of the blades, and σ as the 600C yield strength, we can identify the max speed that the middle of the blade can travel at based on our assumptions. The result (once we substitute in for our $\frac{1}{2}$ diameter blade lengths) tells us that the peripheral speed at the radial distance to the center of the blades is the same no matter how the diameter changes:

$$\frac{m}{A} = \rho \cdot L \Rightarrow v = \sqrt{\frac{\sigma \cdot r_{mid}}{\rho \cdot L}} \Rightarrow v = \sqrt{\frac{\sigma \cdot \left(\frac{5}{12} d_{outer}\right)}{\rho \cdot \left(\frac{1}{6} d_{outer}\right)}} \Rightarrow v = \sqrt{2.5 \frac{\sigma}{\rho}}$$

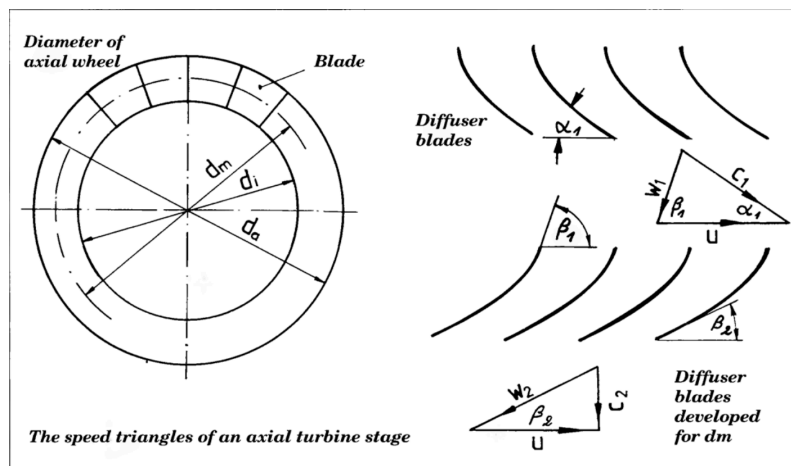
$$= \sqrt{2.5 \cdot \frac{280,000,000}{7980}} = 296.1744389 \text{ m/s}$$

The peripheral speed of the hub is proportionally less than this speed, since it rotates at the same rpm. The stress in it can be higher than the stress at the base of the blades since it is supporting everything beyond it, but it can be made larger to support that mass whereas the blades have a fixed surface area throughout. This is where the iconic turbine shape comes from. The rpm of the shaft and impeller match that of the turbine, which can now be calculated in terms of the diameter:

$$v = \frac{\pi \cdot d \cdot \text{RPM}}{60} = \sqrt{2.5 \frac{\sigma}{\rho}} \Rightarrow \text{RPM} = \frac{60 \cdot \sqrt{2.5 \frac{\sigma}{\rho}}}{\pi \cdot d_{mean}} \Rightarrow \text{RPM} = \frac{72 \cdot \sqrt{2.5 \frac{\sigma}{\rho}}}{\pi \cdot d_{outer}}$$

(This result, while somewhat general across materials, is only valid for modeling the rpm when the blades are 1/6 the length of the diameter.)

To move forward, a value for the target tip speed is needed. Using less than 80% of the max tip speed, we arrived at 220 m/s as our value to base design around. This correlates to a final rpm of ~112,050 for our rotating system, with the diameter being iteratively chosen in order to achieve a set thrust. Returning to Schreckling's approach, we can introduce our most important independent design variable: the turbine blade exit angle. Referred to as B2.



According to Schreckling, "in an ideal design, the shape of the ducts between the diffuser and the turbine blades would be the same, but with opposite camber." We initially adopted this theory and planned for angle α_1 (the exit of the diffuser blades) to match B2. B1 can be calculated based on B2 (however after

many Ansys simulations we ended with different angles). On the speed triangle for the turbine, U is the outer speed of the blade tips ($\text{RPM} \cdot \text{diameter} / 2$), c_2 will be the velocity of air exiting (exhaust speed), and w_2 is the velocity of the air with reference to the blades. C_2 is calculated from the angle of B_2 and the speed of U , with $C_2 = \tan(B_2) \cdot U$.

At this point, the math is not easily walked through as algebra, and a sheet becomes more useful for manipulating variables. We built a spreadsheet model of the engine, effectively forming a very large multivariable system for calculating pressure ratio, thrust, and fuel consumption, then tried different combinations of our input variables until we arrived at something we liked. Using our final values now makes reading our math easier. Almost none of these dimensions were our first “guesses” using the recommendations of hobbyists, and they stand fairly unique to our engine.

For our turbine, we arrived at an inner diameter of 30mm, an outer diameter of 60 mm, blades 15mm long, and an angle B_2 of 40 degrees after multiple (Google Sheets-aided) completions of the following calculations:

We made the following assumptions using our earlier calculations and standard values:

Temperature at compressor inlet (K)	293
efficiency_compressor	0.75
efficiency_turbine	0.75
Cp ambient	1005
cp at turbine inlet	1060
cp at exhaust	1030
temperature at turbine inlet (K)	873
combustor pressure loss (Schreckling estimate)	0.95
standard atm pressure	101325
specific gas constant (dry) air	287.05

To determine our mass flow rate, we first needed to establish the exhaust conditions. The exhaust area of the turbojet was easily calculated (area of the “outer” circle minus the “inner” diameter circle) at 883.57 square millimeters. As mentioned earlier, we averaged the exhaust velocity, c_2 , by assuming it is purely dictated by the peripheral speed at the middle of the blades, U , and our chosen exit angle of $B_2 = 40$ degrees. The absolute velocity of the gas leaving the turbine can be derived from our speed triangles:

$$C_2 = U \cdot \tan(B_2) = 220 \text{ m/s} \cdot \tan(40^\circ) \approx 184.60 \text{ m/s}$$

To find the mass flow rate, we first needed the density of the air at the exhaust. This requires tracing the thermodynamic changes from the compressor to the turbine. We begin by identifying the specific work (w) that our turbine stage can extract. The Euler Turbomachine Equation tells us that the specific work extracted from the air is equivalent to the change in the air’s angular momentum. With our symmetric velocity triangles for the vanes and the blades, and assuming an equal distribution of the stage enthalpy drop between the stator and rotor, the inlet and outlet whirl velocities become equal in magnitude and opposite in direction. This makes the change in whirl velocity equal to twice the peripheral speed, giving a specific work capacity equal to twice the square of the peripheral speed:

$$w = 2 \cdot U^2 = 2 \cdot (220)^2 = 96,800 \text{ J/kg}$$

This work being done by the turbine is then transmitted through the shaft to the compressor. By equating the specific work to the enthalpy rise required by the compressor, we can determine the maximum achievable pressure ratio, π . We assumed an isentropic process but accounted for real-world losses (flow separation, friction, and other heat losses) by introducing a compressor efficiency of 0.75. Using our expected inlet temperature of 293 K and the specific heat capacity of 1060 J/kg K, we use the isentropic flow relation to solve for our pressure ratio (appendix 1 for derivation):

$$\pi = \left[\frac{\eta_c \cdot w}{c_p \cdot T_{in}} + 1 \right]^{\frac{\gamma}{\gamma-1}} = \left[\frac{0.75 \cdot 96,800}{1060 \cdot 293} + 1 \right]^{3.5} = 2.0859$$

From the First Law, the work required to achieve this pressure ratio is reflected in the ideal enthalpy change of the cycle (across the turbine). The required enthalpy rise is as follows:

$$\Delta h = c_p \cdot T_{in} \cdot (\pi^{\frac{\gamma-1}{\gamma}} - 1) \Rightarrow 1060 \cdot 293 \cdot (2.0859^{0.286} - 1) = 176,735.08 \text{ J/kg}$$

Our end goal with the calculations we've done so far is to identify our expected mass flow rate for our turbine dimensions. The turbine exit temperature will be the inlet temperatures (constrained by the 316L as 600C / 873 K) minus the temperature drop = 741.11 K. Using the Ideal Gas Law and assuming expansion to atmospheric pressure, we find the exhaust density, ρ .

$$\rho = \frac{P}{R \cdot T_{exhaust}} \rightarrow \frac{101,325}{287.05 \cdot 741.11} = 0.476 \text{ kg/m}^3$$

Finally, we calculate mass flow rate using the density, turbine exit surface area, and C2, the exhaust velocity from earlier. Mass flow rate is the final part of our puzzle, and yields our engine's shaft power and static thrust; shaft power equals our specific work capacity multiplied by the mass flow, and static thrust is calculated by multiplying the mass flow rate by the exit velocity

$$\dot{m} = \rho \cdot A \cdot c_2 \rightarrow 0.476 \cdot 0.00088357 \cdot 184.60 = 0.0777 \text{ kg/s}$$

$$P_{shaft} = \dot{m} w \rightarrow 0.0777 \times 96,800 = 7,520.24 \text{ W}$$

$$F = \dot{m} \cdot c_2 \rightarrow 0.0777 \cdot 184.60 = 14.34 \text{ N}$$

*Had we calculated our temperature drop correctly, the static thrust would be 13.59 N, not a substantial difference, but underestimating the temperature drop could cause the engine to not function or fail.

To finalize our engine model, we identified the thermal energy of our engine cycle and calculated the fuel required for operation. The thermal power carried by the exhaust can be represented by the overall change in enthalpy of the air from the compressor inlet (at ambient temp/pressure) to the hot turbine outlet. Simplifying by using an average specific heat capacity of 1050 J/kg K, we then modeled the total thermal energy.

$$E_{total} = c_{p,avg} \cdot \dot{m} \cdot (T_{exhaust} - T_{inlet})$$
~~$$= 1050 \cdot 0.0776 \cdot (468.1^\circ\text{C} - 293 \text{ K}) = 14,284.06 \text{ W}$$~~

$$= 1050 \cdot 0.0776 \cdot (741.1 - 293) = 36,506.7 \text{ W}$$

Initially, we made a unit error here that wasn't caught until our first testing of the engine. By mistakenly using the Celsius outlet temperature instead of the Kelvin, we underestimated the total energy by over 50%. Luckily, this didn't carry any error into our design, and had no permanent downstream effects since it only affected our expectations for fuel consumption.

With the energy requirement known, we determined the fuel flow rate needed to sustain our engine at its operating RPM. Kerosene (selected in the project scope section earlier) has a calorific value of 43kJ/g and a density of 0.85 g/mL. To find the volumetric fuel consumption in milliliters per second, we balanced the total thermal energy of the air against the chemical energy provided by the fuel. By dividing the total energy by the volumetric energy density of kerosene (the product of its calorific value and liquid density), we arrived at our required fuel flow rate:

$$\dot{V}_{\text{fuel}} = \frac{E_{\text{total}}}{H_c \cdot \rho_{\text{fuel}}}$$

$$\dot{V}_{\text{fuel}} = \frac{36506.7}{43000 \cdot 0.85} = 0.9988 \text{ mL/s}$$

With that, our preliminary engine math was finished. Here are our summarized specs:

Specification	Value as calculated	In other units
Mass flow rate	.00777 kg/s	-
Exhaust velocity	184.60 m/s	412.9 mph
Static thrust	13.59 N	1.39 kg
Shaft power	7,520.24 W	-
Fuel flow rate	0.9988 mL/s	3.596 L/ hour

Most small UAVs require a thrust-to-weight ratio of 0.3-0.5, suggesting that our engine would have to go into an aircraft with a total weight under 3-4.5kg. This would be a very difficult engineering challenge, and was luckily not our goal. Our design intent was to yield 10-15N with a realistic pressure ratio and high RPM power system.

Simulations

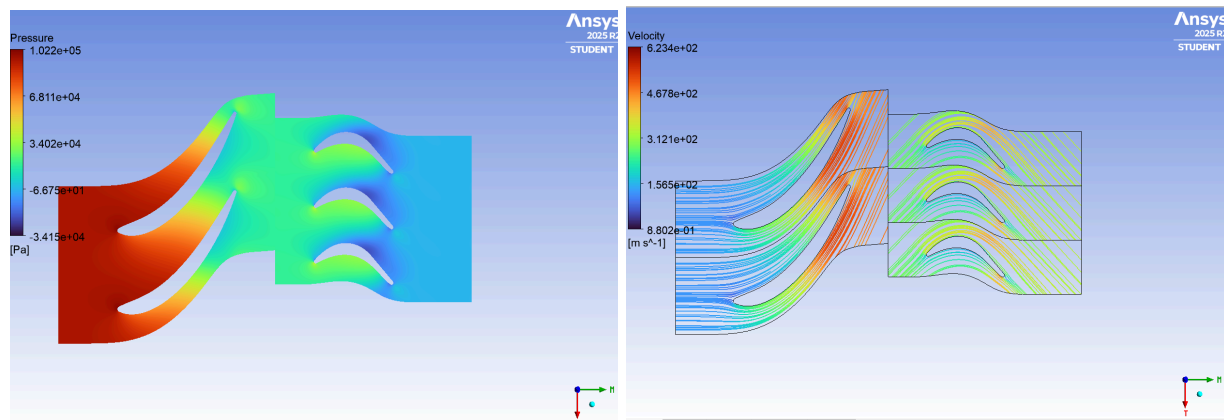
With the preliminary math complete and initial dimensions in hand, we moved into ANSYS to validate and refine our design through CFD. The simulation work followed the same turbine-first logic as our analytical approach.

Turbine Stage

We began by modeling the turbine blades as symmetric airfoils using the geometry recommended in Schreckling's book. Early simulations immediately exposed two problems: excessive residual swirl at the exhaust (our target was approximately zero exit swirl) and sharp flow separation along the blade suction surfaces.

This prompted a systematic exploration of blade geometries. We varied entry and exit angles, then experimented with fundamentally different blade profiles: a cup shape reminiscent of a Pelton wheel bucket, and a cambered airfoil drawn from aircraft wing design. Neither performed adequately. We ultimately converged on the T106C turbine blade profile, a well-characterized low-pressure turbine airfoil from the open literature, which produced the cleanest flow — well-attached streamlines, near-zero exit swirl, and the most favorable pressure distribution across the blade passage.

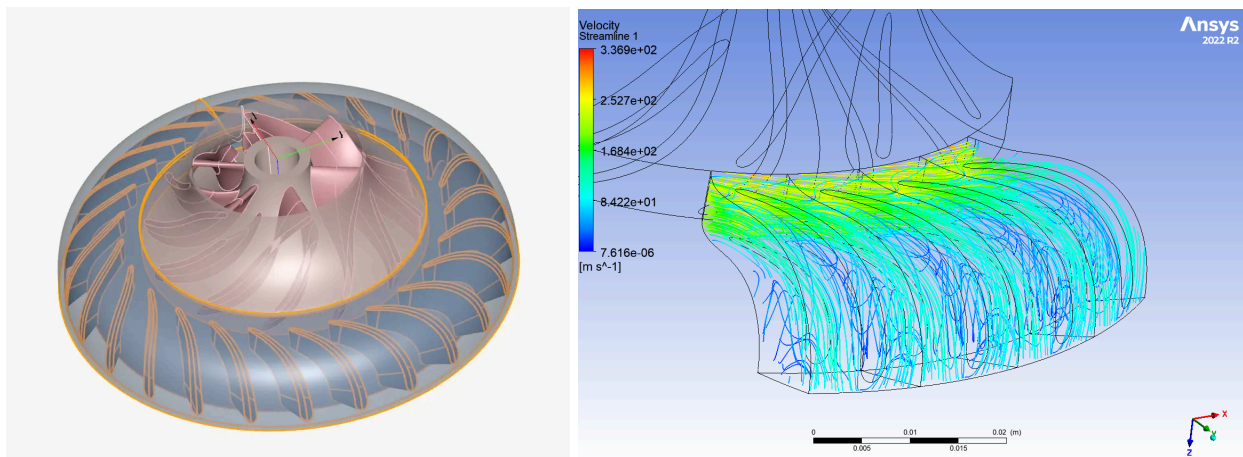
The final turbine dimensions were also revised from the preliminary math: we settled on a 35 mm inner diameter and 50 mm outer diameter. From the converged simulation we extracted updated exhaust velocities, which fed revised thrust and compressor power requirement estimates. The ANSYS CFX performance summary shows a total-to-total isentropic efficiency of 72.0%, consistent with our 75% analytical assumption. Shaft power across all 17 blade rows came out to approximately 8,796 W, slightly above our analytical estimate of 7,520 W.



Compressor Stage

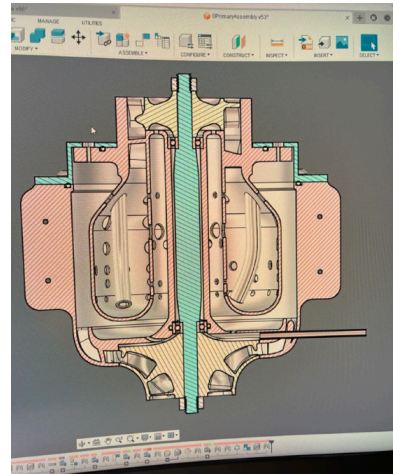
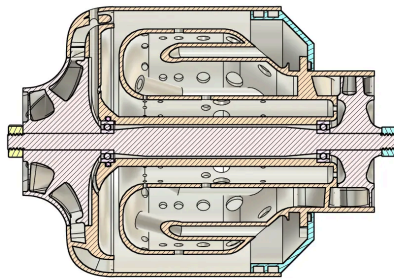
With the turbine characterized, we turned to the compressor. The impeller was designed in CFTurbo, seeded with the inlet and outlet area requirements from our target pressure ratio and mass flow rate, then imported into ANSYS for simulation. The critical constraint driving the diffuser design was bringing the swirling, high-velocity flow out of the impeller down to below roughly 50 m/s before the combustion chamber — exceeding that threshold risks a flameout. This required careful iteration on diffuser vane sweep and vane count to suppress flow separation in the decelerating passages.

One shortcoming worth noting in retrospect: the compressor was optimized almost entirely at the design-point RPM of 112,000. We did not systematically evaluate off-design performance at the intermediate speeds the engine passes through during spool-up. A more thorough treatment would have swept across the operating speed range to assess surge margin and ensure stable compression through the transient — an area for improvement in any future iteration.



Transition to CAD

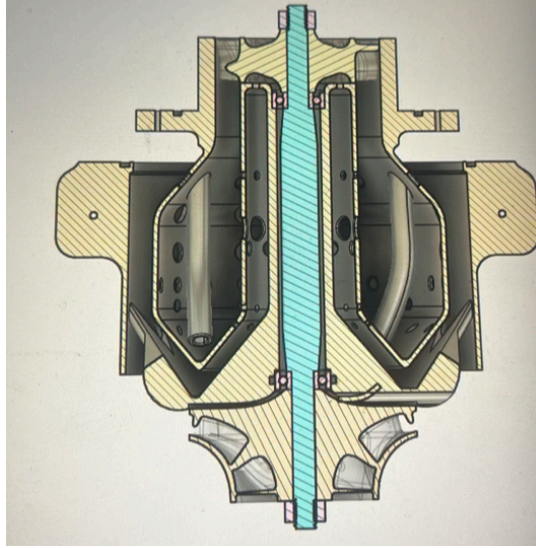
Using the simulated turbine, impeller, and vane models, we transitioned to Fusion. The engine was designed almost entirely in a singular mastersketch which revolved around its central axis. At a high level of abstraction, air was routed to be compressed by the impeller, decelerated as it enters and cools the space outside the combustion chamber, forced into the combustion chamber where it mixes with the kerosene, and finally used to drive the turbine as it exits the control surface.



Our engine, while admittedly small, featured:

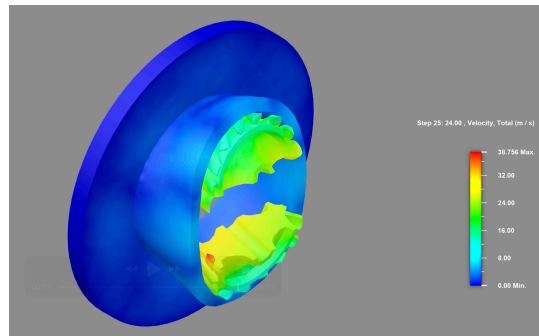
- A shortened central shaft (steel, heat-treated, and ground) to yield the highest resonant frequency possible. This shaft also had left and right-handed threads on opposite ends to retain our compressor and turbine, which were press-fit into place.
- A wave washer spring pre-loads the compressor-side bearing to reduce play and account for thermal expansions.
- An oil lubrication line, allowing oil to enter through a copper tube before lubricating both bearings, passing along the central shaft, then exiting into the combustion chamber to be burnt away.
- Six “walking-stick” style fuel injectors fed with needles to vaporize kerosene into the combustion chamber. (A style of design well-beloved to the hobbyist designs we referenced)
- A combustion chamber with patterned air entrances for ideal combustion
- High-temperature O-rings to seal all mating surfaces.

The first finished model of the engine (above) had a number of issues. Most importantly, it would be nearly impossible to machine. Firstly, the main housing had to be printable. Metal printing is currently very similar to FDM plastic in that it needs supports; however, SLM metal prints need supports to prevent deformation and wick away heat during production. We’d have to slightly edit our original design to better accommodate this. Secondly, our design needed to be post-processable; while we’d print it initially, we’d have to turn down features adjacent to rotation, mill mating surfaces flat, tap a few holes, and remove internal supports.

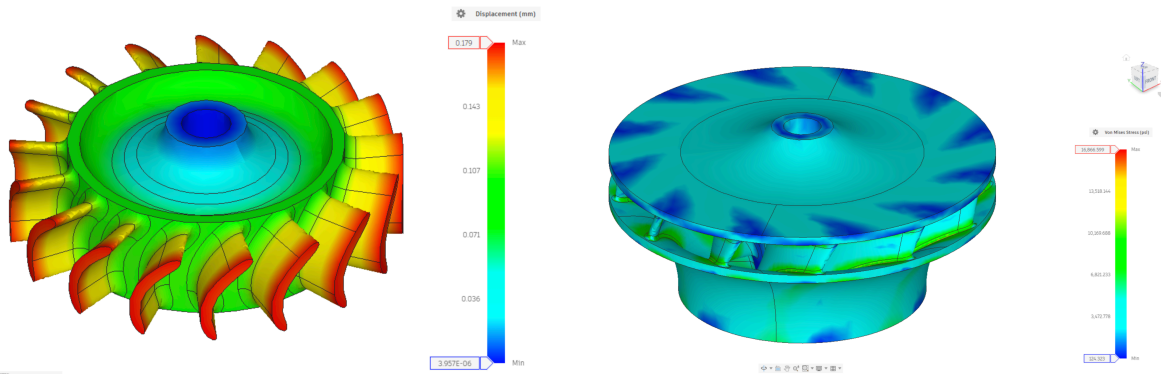


An early ideation for improving support removability and heat dissipation during printing

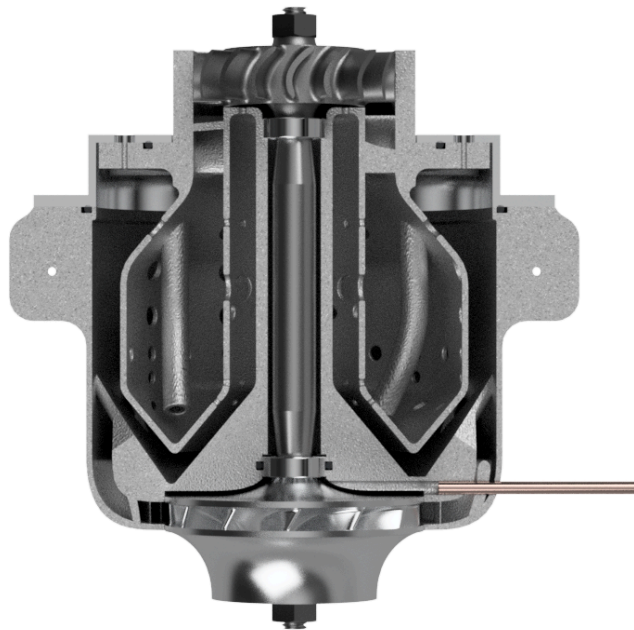
As we increased the fidelity of our design, we moved to safety. Considering that neither of us had designed anything that rotated beyond a few thousand rpm, we consulted a few experts: a turbomachinery engineer at Borgwarner and the staff of the Duke machine shop. After conversations about how unpredictable failure modes could be at high rpm (though we were fairly optimistic that our 316L would creep enough to jam the system engine with friction before anything could fail more catastrophically), we decided to model our housing to ensure it would protect against a turbine burst.



This modeling led us to make our casing substantially thicker surrounding our turbine. Further FEA for the deformation of the impeller and turbine at extreme RPM and operating temperature was also completed, which reinforced our belief that the turbine would fail sooner than the impeller by a significant margin. It should be noted that unexpected resonance or manufacturing error could still cause a burst of the impeller, but we planned to shield the engine from any people during testing to mitigate this risk.



In the end, we arrived at a model that met all of our specs, and would be post-processable using only a manual lathe, manual mill, set of hand taps, acetylene torch, silver solder (needed for the fuel-delivery copper tubing), and an arbor press. This is a far-cry from the precision machining needed for a traditionally produced engine



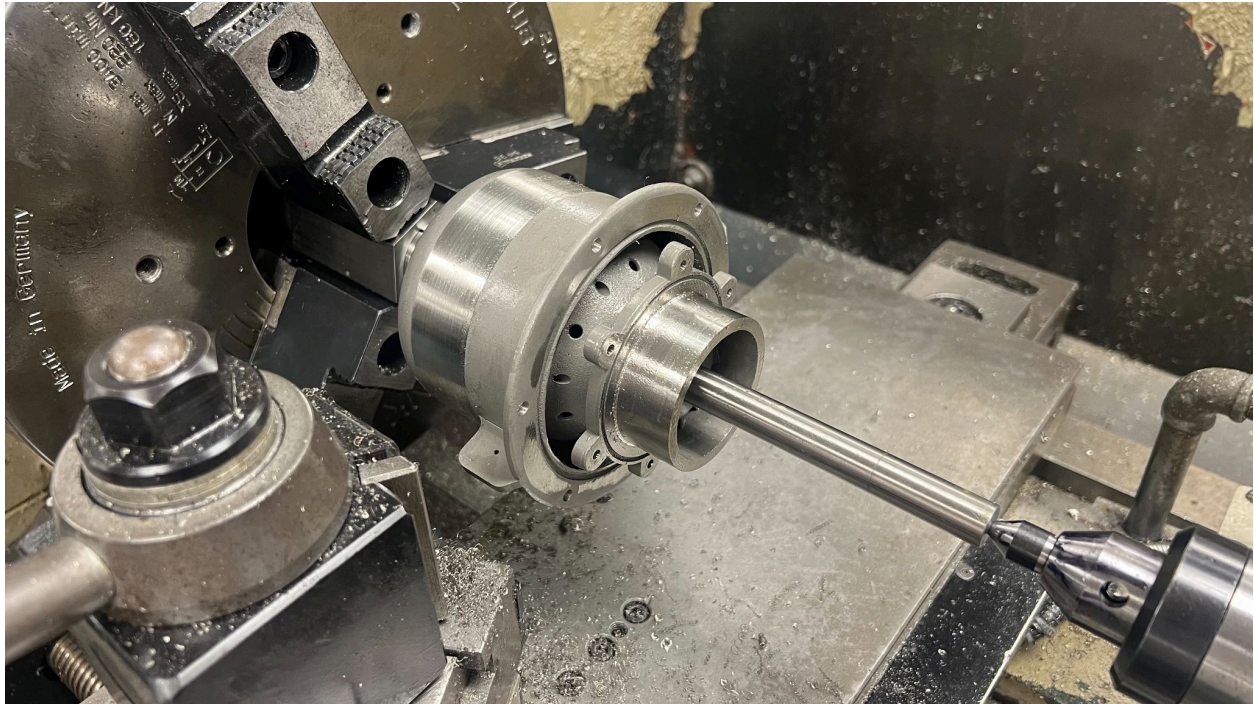
Machining and assembly

While we waited for the engine to arrive, we first machined the shaft. The shaft was rough turned from 4140 annealed steel, and then heat-treated to around 40-45 HRC and finally ground to size to make sure the bearing seatings were tight tolerance. We applied a left-hand tap to one end and a right-hand tap to the other so that neither side would untighten as the shaft spun to speed.

The housing arrived in one beautiful piece. It had a few deformations from the printing process, but all were within the spec we'd added extra material to be machined off for. The post-machining we had to do was very similar to what you'd expect from a cast part.



The housing was reamed inside to allow space for a ground shaft which was then pressed inside the housing. Afterwards the whole piece was held on both sides of the shaft and all outside faces were turned so they would be concentric to each other. From there, the housing was first held on one side and the inner surfaces and bearing seatings were machined down, then on the other side to do the same thing. Theoretically this would have led to perfectly concentric surfaces throughout, in practice I shouldn't have such faith in the chuck.



Housing pressed onto intermediary ground, shaft, with concentric surface being turned

The impeller and turbine were also reamed, turned to be concentric, and then assembled individually onto the shaft with the bearings (outside of the engine). We then set to balancing them; first, we tried to find a balancing machine, but there wasn't one at Duke, and we couldn't find a company to let us use one of theirs. Instead, we first tried the "finger-tip method", as outlined in one of the hobbyist books we'd covered. While humans do have a very hard time measuring force with intuition, we are exceptional at determining vibration and smoothness (most people's fingers could identify a bump a few molecules tall on a perfectly flat surface). The finger-tip method involved the following:

1. Assemble the bearings and one component (either turbine or impeller) onto the shaft
2. Spin the assembly using compressed air, while holding it by both bearings
3. Add a piece of tape at random to the assembly once it stops spinning
4. Spin it again with compressed air
5. If the vibration improved, remove material on the component opposite the tape using a Dremel, then remove the tape; if the vibration instead had worsened, move the tape somewhere else
6. Repeat many times until the system has negligible vibration at high rpm

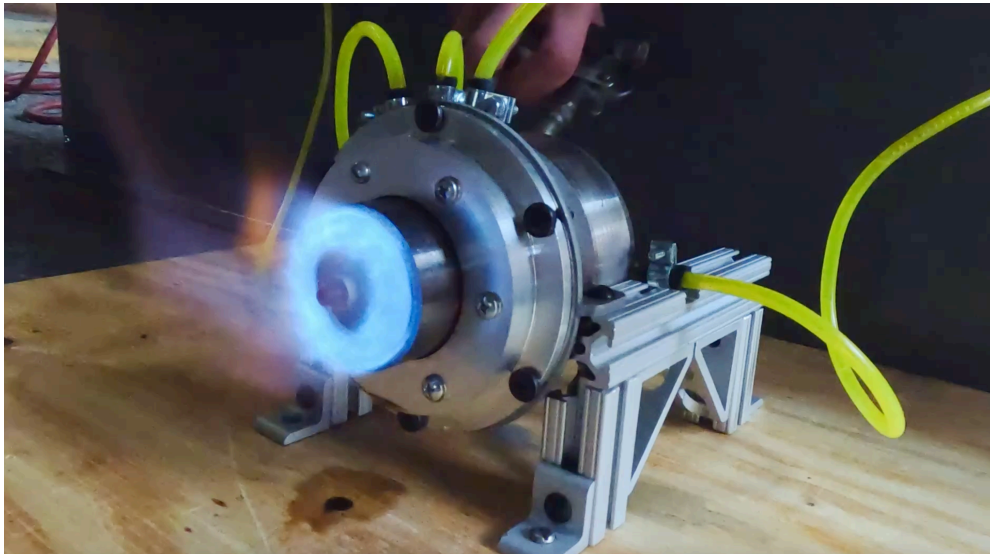
This admittedly became very tedious, and while we were able to make some gains, we instead opted for a less touch-based approach. We turned a piece of aluminum to have an inner diameter slightly larger than our bearings with a concentric, smooth outer surface. Then, we placed the same bearing-shaft-component assembly inside, and rocked it back and forth until one side of the component persistently stayed at the bottom; we then removed material from that side. After many, many hours, our turbine and impeller passed both balancing methods, and we were satisfied to proceed.



Turbine assembled onto shaft with bearings for balancing

The fuel line is made from a bent copper tube in which six holes were drilled and in which small hypodermic needles were bent and then silver soldered.

The final assembly was only a few components: a central shaft inside the housing, with bearings pressed onto both sides; on the compressor/impeller side, the bearing had a wave washer between it and the housing and an o-ring around it; on either side of the shaft we had our compressor and turbine (also pressed on) with nuts retaining them further; the fuel injection tubes were permanently fixed inside the combustion chamber, which was closed with an aluminum cover mounted with an o-ring and 6 stainless steel bolts; the engine was placed atop a simple 80-20 frame on a sheet of plywood, and we were ready for our first spin up.



Appendix

Appendix 1: Derivation of the pressure ratio (π) equation

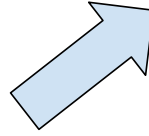
First law

$$w = \Delta h = c_p \cdot (T_{out} - T_{in})$$

$$\eta_c \cdot w = c_p \cdot (T_{out,ideal} - T_{in})$$

Compressor Efficiency (estimated)

$$\eta_c = \frac{c_p \cdot (T_{out,ideal} - T_{in})}{w}$$

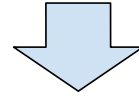


Isentropic Flow Relation

$$\frac{T_{out,ideal}}{T_{in}} = \pi^{\frac{\gamma-1}{\gamma}}$$

$$\frac{\eta_c \cdot w}{c_p \cdot T_{in}} = \frac{T_{out,ideal}}{T_{in}} - 1$$

$$\frac{\eta_c \cdot w}{c_p \cdot T_{in}} = \pi^{\frac{\gamma-1}{\gamma}} - 1$$



$$\pi = \left[\frac{\eta_c \cdot w}{c_p \cdot T_{in}} + 1 \right]^{\frac{\gamma}{\gamma-1}}$$

Citations

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