Waste Heat Turbocharger

A Major Qualifying Project

Submitted to the Faculty of

Worcester Polytechnic Institute

in partial fulfillment of the requirements for the

Degree in Bachelor of Science

in

Mechanical Engineering

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Date: 4/26/18

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Abstract

Diesel engines are used in a wide range of applications, from passenger vehicles to large scale transportation and shipping. These engines account for roughly 20% of total US fuel consumption, so maximizing their efficiency is paramount. The addition of a turbocharger dramatically increases engine performance and efficiency. Even these devices, however, have their drawbacks. The introduction of a turbine in the exhaust stream creates a restriction that impedes the engine's power. Moreover, there is a delay in developing boost pressure, called turbo lag. Both of these characteristics limit the real-world efficiency and performance. These engines typically have a thermal efficiency of about 40%. This project designed a system that recaptures heat normally wasted through the exhaust and uses it as the heat source for a Rankine cycle that powers the turbocharger. This system aims to improve overall engine efficiency, by reducing the flow restriction in the exhaust stream and the amount of energy wasted as heat through the exhaust.

Acknowledgements

We would like to express our gratitude to certain individuals for their help during the course of this project.

- Our advisor Professor Torbjorn Bergstrom, who gave us helpful advice, feedback and guidance from the beginning of the project until the very end.
- Professor Sullivan, who assisted and guided us through the thermodynamic modelling of our initial system.
- Ken Howe for supplying us with a lot of the necessary materials.
- Carter Reynolds for ECE consultation regarding our Arduino control circuit.
- Aaron Pearl for assisting us with debugging our Arduino code.
- Worcester Polytechnic Institute, in particular Washburn Shops and staff, for allowing us access to various machinery.
- Procon, and specifically Jeff Kulikowski, for donating a water pump to our project.
- Aspen Aerogels, and specifically Brian Cahill, for donating a roll of ultra-high temperature insulation to our project.
- Todd Keiller, Lynda Kelly, and Christopher Lutz for guiding us through the process of filing a preliminary patent application.

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1 Introduction

1.1 Objective

Increase the efficiency of a diesel engine for automotive applications.

1.2 Rationale

Diesel makes up roughly 20% of total U.S. petroleum fuel consumption (Diesel fuel explained, 2017). In 2016, 49.5% of passenger vehicles in the European Union are diesel powered (Share of diesel in new passenger cars). Because of their fuel efficiency and power output, diesel engines are heavily used in both passenger and commodity shipping applications. Though diesel engines are more efficient than conventional gasoline engines, they still contributed to 437 million metric tons of CO2 emissions in 2016 in the U.S. alone (How much carbon dioxide is produced from burning gasoline and diesel fuel?, 2017). Furthermore elevated fine particulate pollution and nitrogen oxide (NOx) emissions associated with diesel engines contribute to additional environmental degradation through increasing smog and acid rain. As the world's finite supply of fossil fuel dwindles, diesel fuel prices are increasing. By increasing the efficiency of diesel engines, greenhouse gas and harmful pollutant emissions will be decreased, and monetary savings through reduced fuel costs can be observed amongst passenger vehicle owners and shipping fleets.

1.3 State of the art

One of the earliest improvements to the diesel engine is the addition of the turbocharger. The turbocharger is device consisting of a compressor in the intake of a diesel engine with a shaft connecting it to a turbine mounted in the exhaust. The turbine is driven by the pressure and velocity of the exhaust gases exiting the engine. The turbocharger greatly improves engine efficiency and performance because the compressor forces more air through the intake into the combustion chamber, allowing a smaller displacement engine to burn more fuel cleaner. This innovation was first conceived in the early 1900's; however it was only implemented on large marine and locomotive engines until bearing and manufacturing technology made it practical for widespread implementation. With development driven by the energy crisis of the 1970's, 1978 saw one of the first large production scale automotive applications of turbochargers in the Mercedes OM617 diesel engine product family (Becker, Jürgens, Köckritz, & Schimpf, 2004). Since then Turbo Diesels have become the standard in production vehicles for both commodity shipping and passenger markets.

Similarly, superchargers accomplish increased engine performance by compressing air as it enters the cylinders. Rather than using the kinetic energy of exhaust gases to power the compressor, a supercharger compressor is directly connected to the engine's crankshaft via a belt or gear. This takes power away from the engine, but increases the power density of air entering the engine, resulting in a net power gain. Cold air intakes are a primitive way in which to boost engine performance by increasing the oxygen content of intake air. Cold air intakes are simply air intakes sourced away from hotter parts of the engine compartment. The reasoning behind it is that cold air is denser and therefore has more oxygen per unit volume; thus the engine can burn more fuel at an increased efficiency.

Like all internal combustion engines, diesel engines have a thermal efficiency associated with them. This means that only a fraction of the energy created by burning fuel is converted to mechanical work that actually drives the vehicle. Energy from the combustion leaves the engine via kinetic and thermal energy of the exhaust gas, frictional losses from bearings, and lastly heat through the radiator and the exhaust. Of these, heat loss accounts for the greatest waste of energy. Because of this, recovering waste heat has become a very attractive way in which automakers are attempting to increase the efficiency of a diesel engine. For example GM's EGHR system uses heat from exhaust gases to preheat coolant (Brzozowski, 2015). This allows the engine to reach its most efficient operating temperature more quickly during startup. In colder regions this has been proven to be particularly effective at saving fuel.

Recently, BMW has been working on a complex heat recovery system that they have dubbed a "Turbosteamer". This concept utilizes a heat exchanger in the exhaust system of the vehicle, combined with the coolant system to capture up to 80% of the engine's waste heat. The heat is used to power a Rankine cycle by evaporating a liquid, creating steam which is driving a turbine or a piston. The mechanical shaft work output of this device is directly coupled to the vehicle's crankshaft and therefore can add additional power and torque to the engine. After exiting the turbine/piston the vapor is condensed into a liquid which is then pressurized to continue the cycle. This combined cycle can increase the efficiency of the vehicle by an estimated 15%. A similar design uses the same concept of capturing waste heat to vaporize a liquid and spin a turbine with the resulting vapor. The shaft output of this turbine is used run a compressor in conjunction with a traditional source, such as the crankshaft or an exhaust-powered turbine.

Another sophisticated design uses coolant and exhaust gases to heat up and boil a working fluid. This working fluid is then used in a Rankine cycle style process to spin a turbine. The turbine output is then used to generate electricity. This process has a myriad of applications. It can be applied to any internal combustion engine and could be used to charge the batteries of a hybrid. It could also be used in large scale industrial electric generators (Recovering wasted heat).

Hyundai has a patent for designs that use the exhaust heat of an internal combustion engine to evaporate the working fluid in a Rankine cycle, the output of which is used to provide supplemental power the vehicle’s turbo compressor, alongside the traditional exhaust gas turbine. This patent covers designs that use the steam-powered turbine as a secondary turbine to power the traditional turbo compressor, or to power a second compressor unit, which can be connected either in series or in parallel to the traditional turbo compressor (Kim, 2014).

The heat of the exhaust gases of an internal combustion engine have been utilized outside of the automotive industry. A General Electric patent outlines the use of the exhaust of an internal combustion engine of a power plant, along with other low-temperature heat sources (such as charge coolers), as the heat sources of an organic Rankine cycle, the output of which is connected to an electricity generator (Freund et al., 2012).

1.4 Approach

The addition of a turbocharger to a diesel engine greatly increases the efficiency and power output of the engine. A turbocharger is a device comprising of a compressor in the intake of the engine mounted on the same shaft as a turbine on the exhaust side of the engine. The hot, pressurized exhaust gases exiting the engine drive the turbine which in turn drives the compressor. The compressor increases the pressure, and therefore the density of the air entering the engine, which means more oxygen is available in the combustion chamber for each power stroke. More oxygen allows more fuel to be burnt without increasing the displacement of an engine. This improves the power to weight ratio of an engine significantly which in turn improves fuel economy. Through this process, the diesel engine efficiency and performance is increased.

Turbos, however, have certain disadvantages. The exhaust side turbine is a restriction in the exhaust gas flow. This creates a high-pressure region between the exhaust valves and the turbine itself, commonly referred to as back pressure. As a result, the pistons must work against this pressure during the exhaust stroke. This effect robs power from the engine and decreases the overall efficiency. For every psi of boost pressure added to the intake, there are approximately 2 psi of back pressure created in the exhaust. If this back pressure could be removed without sacrificing the turbocharger benefits, the efficiency of the engine could be increased further.

Diesel engines are observed to have a thermal efficiency of up to 40%. A large portion of the energy not transferred into usable mechanical work is lost in the form of heat through the engine's coolant and exhaust. If this waste heat energy was recaptured, the efficiency of the diesel engine could be significantly improved. This project aims to increase the efficiency of a diesel engine for automotive applications by recovering waste heat energy and using it to power the turbocharger. To accomplish this, a Rankine cycle was designed to be used to capture waste heat from the exhaust and power the turbine side (exhaust side) of the turbocharger. By removing the turbine from the exhaust gas stream and replacing it with a heat exchanger, the back pressure in the exhaust manifold has the potential to be greatly reduced, further improving the performance of the engine. The heat exchanger will be used to preheat a pressurized working fluid which will go through a final propane boiler to become superheated vapor. The vapor will then pass through the turbine, which is connected to the intake compressor. This system has the potential to increase efficiency by capturing waste heat energy and decreasing back pressure, as well as increase power and engine performance. Specifically, this system could maintain steady levels of boost to completely eliminate turbo lag. These improvements could further improve the power to weight ratio of a diesel engine giving better performance and fuel economy.

2 Methods

2.1 Brainstorming and Preliminary Design

While taking a thermodynamics class at WPI Benjamin Freed came up with the idea of running a turbo charger off a Rankine cycle powered by wasted exhaust heat. After brainstorming with a group a basic design was determined.

2.2 Calculating Compressor Power Requirements

In order to determine whether or not the exhaust gases alone could provide a sufficient quantity of heat to vaporize and superheat a working fluid that powers the turbine, the process was modeled as a simple Rankine cycle. In order to estimate some of the initial conditions, the mass flow rate of the air entering the engine was first determined. This figure together with the intake boost pressure determine the power requirement for the compressor. The mass flow rate also determines how much heat is available in the exhaust gas. In order to calculate this figure, the displacement of the engine and RPM need to be known. For our calculations, we considered a 1.9 L Volkswagen ALH engine, because there was a running engine readily available. Since the engine completes a four-stroke cycle, every two revolutions of the crankshaft a volume equal to the entire displacement of the engine is expelled into the exhaust manifold. Thus the volumetric flow rate can be calculated as one half the RPM multiplied by the displacement. The volumetric flow rate at any given time is therefore proportional to engine RPM. The mass flow rate can then be easily found by multiplying the volumetric flow rate by the density of the air. The density of air is dependent on what boost pressure the compressor of the turbo is generating, as well as the intake air temperature. For any given boost pressure the density of air can be calculated using the ideal gas law.

The work required by the compressor is calculated using the following equations:

Volumetric flow rate through the engine:

$$\dot{\forall}_{air} = \frac{Engine\ Displacement}{2} \times \frac{Revolutions}{minute} \times \frac{1\ minute}{60\ seconds}$$

Power Required to compress intake air from atmospheric pressure to the intake boost pressure:

$$\frac{P_2}{P_1} = \frac{P_{intake,gage} + P_{atm}}{P_{atm}} = \frac{P_{r_2}}{P_{r_1}} \Rightarrow P_{r_2} = P_{r_1} \times \frac{P_{intake,gage} + P_{atm}}{P_{atm}}$$

Isentropic specific Enthalpy, h_{2s}, from Thermo tables (interpolation)

Actual Enthalpy:

$$\frac{h_{2s} - h_1}{h_2 - h_1} = \eta_{comp} \Rightarrow h_2 = \frac{h_{2s} - h_1}{\eta_{comp}} + h_1$$

Temperature:

Use h_2 and thermo tables to get T_2

Air Density: Ideal Gas Law

$$P_2 = \rho_2 R\, T_2 \Rightarrow \rho_2 = \frac{P_2}{R\, T_2}$$

Mass Flow rate of air through the engine:

$$\dot{m}_{air} = \dot{\forall}_{air} \times \rho_{air}$$

Compressor Power:

$$\dot{W}_{comp} = \dot{m}_{air} \times (h_2 - h_1)$$

2.3 Calculating the Power Output of a Rankine Cycle

For these calculations it was assumed that the ambient air temperature was 60°F (15.55 ℃) and had a pressure of 1 atmosphere. The resulting power requirements over a range of boost pressures and RPMs are tabulated in the Results section. The next step was to then use this power requirement to define the working fluid flow rate, pressure, and temperature required to generate the power needed by the compressor. In order to do this, many assumptions were needed for initial conditions and the process was iterated several times. In order to ensure the correct approach, the group consulted Professor Sullivan. The following equations show how the power output of the Rankine cycle was calculated:

Choose RPM, Boost Pressure, Working Fluid High and Low Pressures, Working Fluid Pressure and quality after turbine (state 3), Exhaust Gas Temps:

Use average heat capacity for exhaust gases, based on EGTin and EGTout

Heat Flow from Exhaust to Working Fluid:

$$\dot{Q} = \dot{m}_{air} \times c_p \times (EGT_{in} - EGT_{out})$$

From working fluid low pressure, get enthalpy and specific volume before pump (state 4). Enthalpy after pump (incompressible flow approximation):

$$h_1 = v_4 \times (P_1 - P_4)$$

Based on P_3 (chosen) and x_3 (chosen):

$$h_3 = \left[x_3 \times \left(h_{g3} - h_{f3}\right) + h_{f3}\right]_{P_3}$$

Enthalpy after heat exchanger (state 2):

$$h_2 = \frac{\dot{Q}}{\dot{m}_{wf}} + h_1$$

Turbine Power Output:

$$\dot{W}_{turb} = \dot{m}_{wf} \times (h_2 - h_1)$$

Based on h_2, use thermo tables to find T_2 to eliminate flow rates that result to T_2 > EGTout (impossible case)

- ➔ Run h_2 and turbine power calculation for a range of $\dot{m}_{wf}$ to determine what flow rates are required to match the compressor power requirements

- ➔ Run the calculations for different combinations of engine RPM, Boost, working fluid high pressures, and EGT's, to determine what circumstances result in adequate turbine power output

To check if there are any circumstances where the heat from the exhaust is enough, the maximum theoretical output of the Rankine cycle was calculated by assuming Carnot efficiency and low and high working fluid temperatures of 333K (60 C) and 723K (500C) respectively, using the following equations:

Maximum Rankine Cycle Output assuming Carnot efficiency, with working fluid temperature range 333K (60C) to 723K (500C), for various engine RPM, Boost pressures, EGT drop:

Carnot Efficiency:

$$Carnot\ Efficiency = 1 - \frac{T_{low}}{T_{high}} = 1 - \frac{333K}{723K} = 53.9\%$$

Heat Flow Rate from exhaust to working fluid:

$$\dot{Q} = \dot{m}_{air} \times c_p \times \Delta(EGT)$$

Max Power Output:

$$\dot{W}_{turb,\ max} = \dot{Q} \times (Carnot\ Efficiency)$$

➔ Compare this result to the required power to see for what cases it is absolutely impossible to make the system work, and which cases require efficiencies that are lower than Carnot and potentially feasible

The calculations showed that for the particular motor considered, there are few circumstances that the heat absorbed from the exhaust gases is enough to power the turbo compressor through the Rankine cycle, even when the cycle operates at very high efficiencies. To be able to use the Rankine cycle to power the compressor at all driving conditions, an additional heat source would be needed.

After the heat transfer and capacity calculations for the specific diesel engine used showed that there was not enough heat in the exhaust, the calculations were repeated for larger gas engines. Because gas engines have significantly lower compression ratios compared to diesel engines, they have lower thermal efficiencies which result to higher exhaust gas temperatures. As a result, more heat is available to be extracted from the exhaust. Moreover, because of detonation issues that arise at high compression ratios and high boost pressures, gas engines typically run lower boost pressures, which means that less power is required by the intake compressor.

2.4 Power Loss Due to Exhaust Back Pressure

As previously mentioned, the restriction of the turbine results in back pressure between the cylinders and the turbine side of a conventional turbo. This region of back pressure extends into the cylinders when the exhaust valve open. As a result, the pistons need to push against this pressure during the exhaust stroke, resulting in power loss. By placing a boost gage in the exhaust manifold, and comparing its readings to that of the boost gage in the intake manifold, it was found that there is a 1:2 ratio of boost pressure to back pressure. To calculate the power loss, the following method is followed:

Approximate backpressure as 2*(boost pressure):

$$P_{back} = 2 \times P_{boost}$$

Work needed to push pistons against the back pressure:

Resistance from back pressure only during exhaust stroke. Only one piston at a time:

$$W_{back,\ piston} = F \times Stroke = P_{back} \times A_{piston} \times Stroke = P_{back} \times (Piston\ Displacement)$$

$$= P_{back} \times \frac{Engine\ Displacement}{4\ cylinders}$$

$$W_{back,\ per\,revolution} = W_{back,\ piston} \times 4\ cylinders = P_{back} \times Engine\ Displacement$$

Power Loss because of backpressure:

$$HP_{loss,\ backpressure} = \frac{d}{dt}\left(W_{back,\ per\,revolution}\right) = \frac{d}{dt}(P_{back} \times Engine\ Displacement)$$

$$\Rightarrow HP_{loss,\ backpressure} = P_{back} \times \dot{\forall}_{air}$$

It should be noted that the gauge backpressure should be used, since the bottom side of the pistons are exposed to the air contained in the crankcase, which is at approximately 1 atm.

2.5 Propane Boiler

From initial calculations in Section 2.3, it is shown that there will not be sufficient energy available from the exhaust to drive the turbo. For this reason, an additional propane boiler will serve as a final heater to superheat the working fluid as needed. The assumption which makes this solution viable is that the cost of the propane needed will be less than the savings in diesel fuel. This is dependent on the relative prices of the two fuels. However, looking at it from an environmental standpoint the emissions from propane are far less harmful than that of diesel, and propane generally burns more efficiently than diesel, so the system may still reduce environmental impact even if it is not cheaper to operate. The components for the boiler were made out of 316 Stainless steel because of its ability to withstand high temperatures while resisting corrosion. These parts were all designed using Fusion 360, and modeled after existing parts. For example, the burner tube hole size and spacing were modeled off a propane burner in a grill. Geometric spacing constraints were the primary drivers behind the specific design. The CAM for these parts was also made in Fusion 360, and they were machined on the school's mills and lathes.

2.6 Controls for Propane Boiler

Introducing a separate heat source requires the design of an entirely new control system. This system needs to ignite the propane, monitor the status of the flame, reignite as needed, alert the operator to any malfunction, and turn on/off the propane supply. In order to determine the basic circuitry and logic for the system flow charts were made. An Arduino Uno microcontroller is utilized to manage the various electronic components of the system. Research was conducted, and a design was determined based on current commercial furnaces. This design implements a flame sensor/ignitor unit, solenoid-controlled valve, and LED warning light.

3 Results

3.1 Preliminary Design

Initial brainstorming resulted in the following preliminary design (Figure 1):

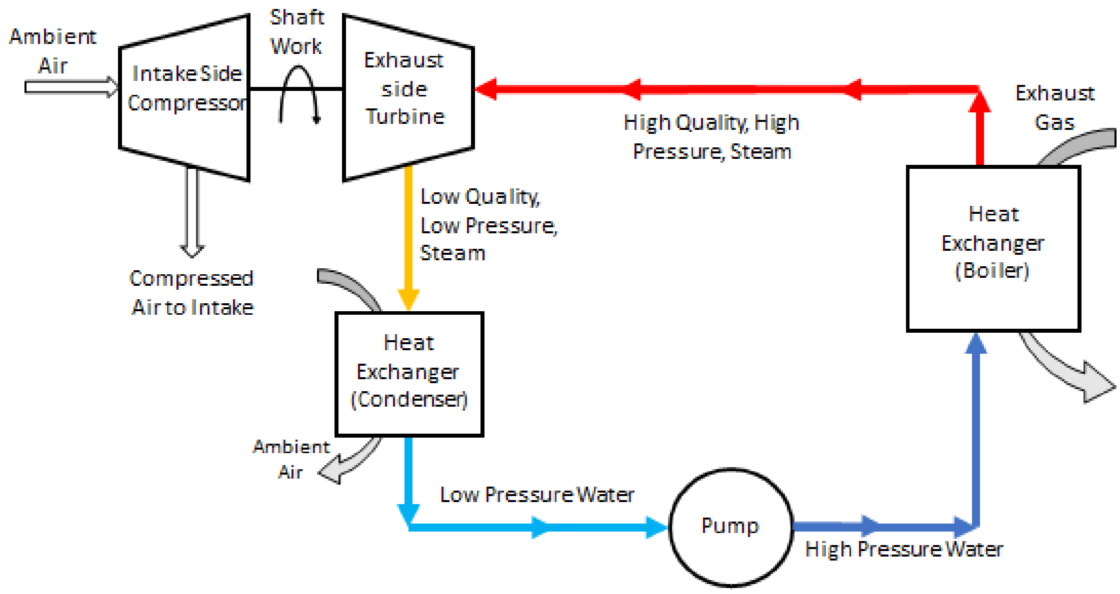

Figure 1. Preliminary System Design

This design utilizes a simple Rankine cycle that runs off wasted heat energy in the exhaust to spin the compressor of a turbo. In lieu of a traditional turbine in the exhaust, there will be a heat exchanger. This heat exchanger allows for greater exhaust gas flow and less back pressure, in addition to waste heat harvesting. By reducing the total back pressure, the amount of work the pistons must do on the exhaust stroke to expel the exhaust gases decreases and therefore the net power loss of the engine decreases.

3.2 Calculating the Power Required by the Compressor

Utilizing the equations outlined in the methods section the following results were tabulated. The first variable calculated was the volumetric flow rate. Table 1 shows the estimated volumetric flow rate over a range of RPMs:

Table 1. Volumetric flow rates of air through the engine, in different units

Volumetric flow rates			
RPM	*(m^3/s)*	*(cfm)*	*(gpm)*
1500	0.0238	50.32	376.4
2000	0.0317	67.10	501.9
2500	0.0396	83.87	627.4
3000	0.0475	100.65	752.9
3500	0.0554	117.42	878.4
4000	0.0633	134.20	1003.9
4500	0.0713	150.97	1129.3

The next quantity calculated was the air mass flow rate, shown in Table 2. This variable changes with RPM and intake boost pressure, because of the density dependence on pressure:

Table 2. Mass flow rate of air through the engine for different intake boost levels

Mass Flow Rates (kg/s)					
RPM	*25 psi*	*20 psi*	*15 psi*	*10 psi*	*5 psi*
1500	0.0527	0.0483	0.0440	0.0391	0.0341
2000	0.0702	0.0644	0.0587	0.0522	0.0454
2500	0.0878	0.0805	0.0734	0.0652	0.0568
3000	0.1054	0.0966	0.0880	0.0783	0.0681
3500	0.1229	0.1127	0.1027	0.0913	0.0795
4000	0.1405	0.1288	0.1174	0.1044	0.0908
4500	0.1580	0.1449	0.1320	0.1174	0.1022

Using the mass flow rates, the amount of power required by the compressor to compress air to the desired boost pressure was then tabulated (Table 3):

Table 3. Power Required to compress the intake air to the target boost pressure

Power Required (kW)					
RPM	*25 psi*	*20 psi*	*15 psi*	*10 psi*	*5 psi*
1500	7.27	5.65	4.12	2.63	1.25
2000	9.69	7.53	5.49	3.51	1.66
2500	12.12	9.41	6.87	4.39	2.08
3000	14.54	11.29	8.24	5.27	2.50
3500	16.97	13.18	9.62	6.15	2.91
4000	19.39	15.06	10.99	7.02	3.33
4500	21.81	16.94	12.36	7.90	3.74

3.3 Calculating the Power Output of a Rankine Cycle

To calculate the Power output of the Rankine cycle many iterative calculations were conducted under different pressure and temperature assumptions. In order to find an adequate mass flow rate, combinations of flow rates were tried over different combinations of pressure and temperatures. The resulting graphs are shown below, in Figure 2. For these calculations, the Rankine cycle parameters were chosen as follows: high pressure = 15 bar (before turbine), low pressure = 0.8 bar (after the turbine), quality after the turbine = 0.95, low temperature = 95 ^{o}C. The calculations were performed for a range of boost pressures between 5 and 25 psi, for engine RPM range of 1500 – 4500, and for two different exhaust gas temperature drops through the heat exchanger, 200 ^{o}C (shown in yellow) and 250 ^{o}C (shown in red).

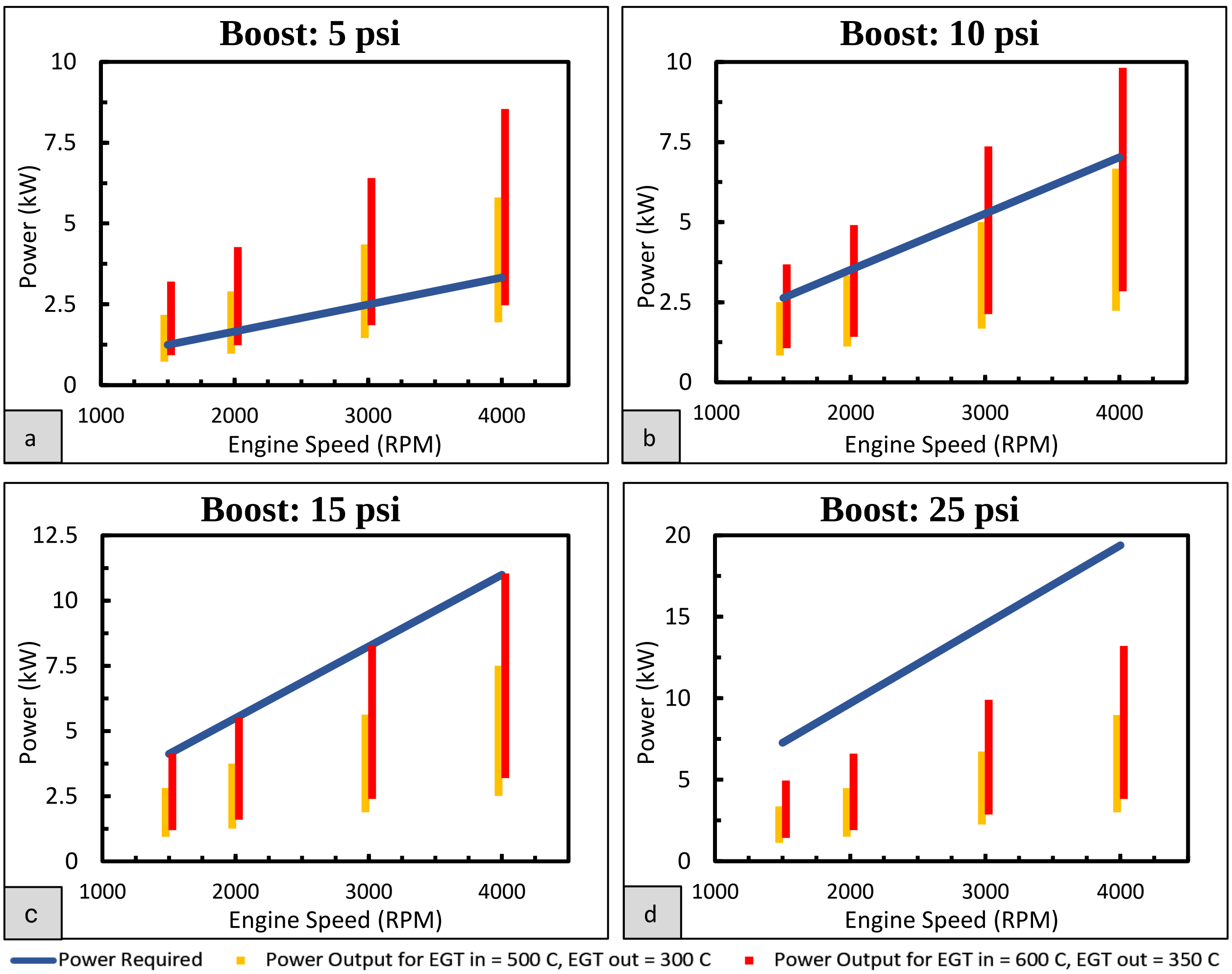

Figure 2. Comparison of the compressor power requirements and the range of possible Rankine cycle power outputs for 2 different EGT ranges, for 4 different boost pressures: a) 5psi, b) 10 psi, c) 15 psi, d) 25 psi. All data for a 1.9L VW ALH diesel engine

The data used to create these figures are tabulated in Appendix A.

It can be observed from the graphs in Figure 2 that the Rankine cycle output cannot match the compressor power requirements for higher boost pressures. Moreover, the Rankine cycle output is higher for the cases with the higher EGT drop (as expected), meaning that with higher EGT drops the system can operate for a wider range of operating conditions. For these reasons, it was decided to run the same analysis for a larger displacement gasoline engine, since gasoline engines typically have higher EGT's and run lower boost pressures.

Larger displacement engine calculations

The thermodynamic calculations presented above were repeated for a larger gasoline engine. For these thermodynamic calculations, a 3.4-liter gasoline engine was chosen (a typical size for a V6 engine). Boost pressures used for the calculations were in the 5-15 psi range. The working fluid conditions used were the same as the ones used in the diesel engine calculations, so that the results could be directly compared. The exhaust gas temperature going into the heat exchanger was chosen to be 700 ^{o}C (a little under 1300 ^{o}F). For every boost level, the calculations were performed for two different exhaust gas temperature drops; a drop of 200 ^{o}C (from 700 to 500), and a drop of 350 ^{o}C (from 700 to 350). To achieve a temperature drop of 350 ^{o}C, the heat exchanger would have to be designed well, but it is still a realistic scenario.

The results of these calculations show that the required compressor power is definitely achievable for the cases where the exhaust temperature drop is 350 degrees. For a temperature drop of 200 degrees, the compressor power requirements can be matched for lower boost pressures. Overall, this analysis shows that larger displacement engines with higher EGT's can take advantage of a system like the one initially proposed here.

The results for the two extremes of the boost pressure range, 5 psi and 15 psi, are presented in Figure 3:

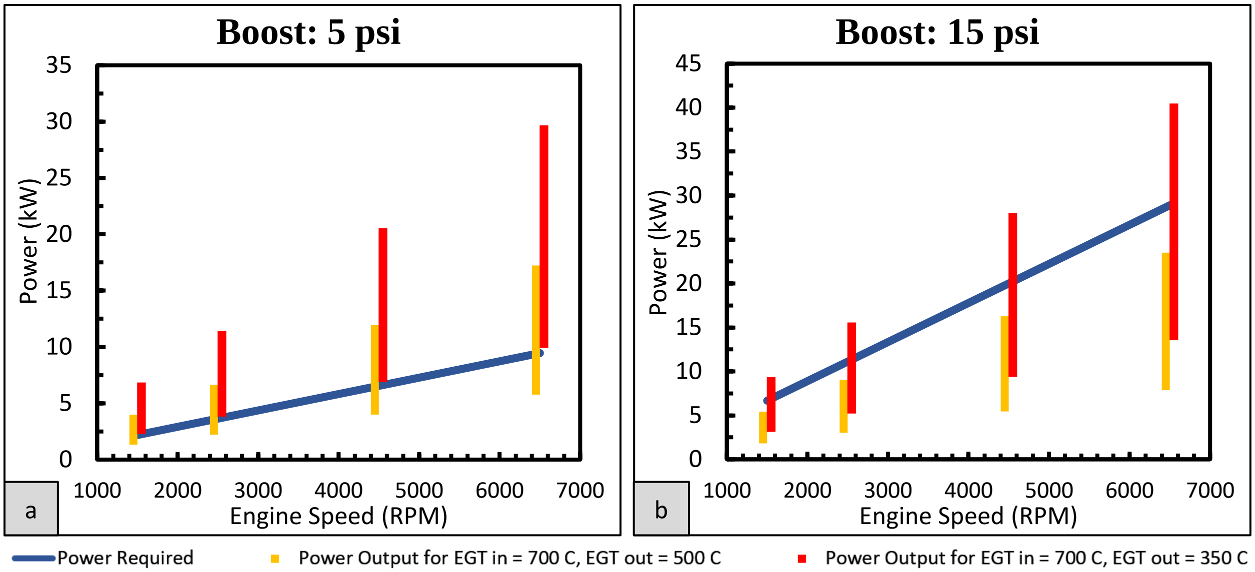

Figure 3. Comparison of the compressor power requirements and the range of possible Rankine cycle power outputs for 2 different EGT ranges, for 2 different boost pressures: a) 5psi, b) 15 psi. All data for a 3.4L gasoline engine.

The data used to create these figures are tabulated in Appendix B.

The graphs in Figure 3 reveal that the larger gasoline engine has enough accessible waste heat energy in the exhaust gases to be able to sustain a Rankine cycle to power the turbo at practically all operating conditions.

3.4 Power Loss due to Exhaust Back Pressure

For the VW diesel engine, the power loss due to back pressure for any given boost pressure between 10 and 25 psi lies within the red-shaded region in Figure 4.

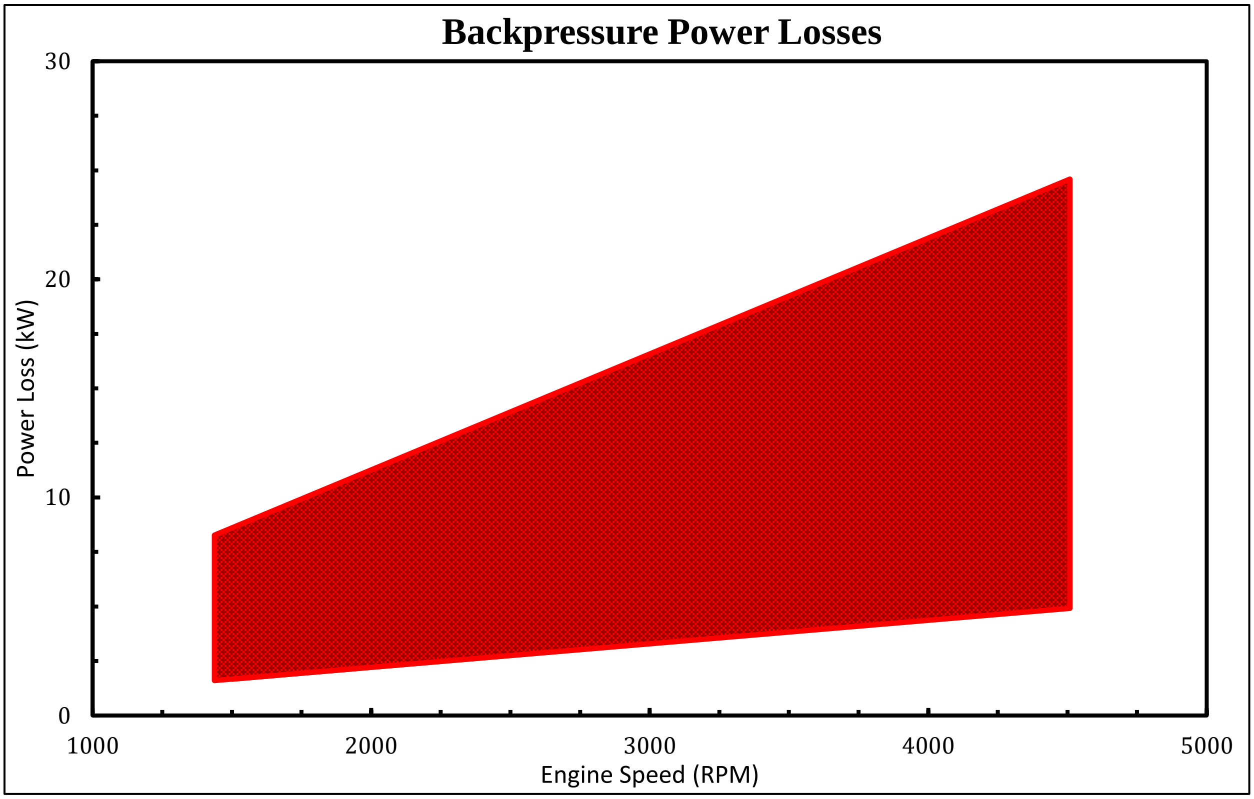

Figure 4. Power loss due to back pressure in the exhaust. When the exhaust valves open in the exhaust stroke, the pistons must push the combustion products out of the cylinder against the back pressure developed in the exhaust manifold, thus lowering power output

In reality, removing the turbine from the exhaust stream will not result in the engine regaining all the calculated power loss, because the exhaust pipes are of finite diameter and therefore do present some restriction to the gas flow, thus developing some back pressure. That back pressure, however, is very small in comparison to a traditional turbo setup.

3.5 Final Propane Boiler

An efficient, reliable, and controllable heat source that would be able to supplement the exhaust heat is propane. Propane burners and propane-based systems are used in a variety of applications, including automotive. This means that propane-based systems can be DOT-approved and used in a car. Moreover, propane is readily available and easy to control.

To build the propane boiler the CAD models and CAM were done in Fusion 360. The team referenced existing designs for home water heaters and commercial boilers. The design uses a central burner similar to that of a gas grill. The burner tube is .375” in diameter and has five rows of holes running axially along it, each 60 degrees apart, with the row at the bottom of the pipe missing since there will be no additional water pipes there. The burner tube is shown in Figure 5:

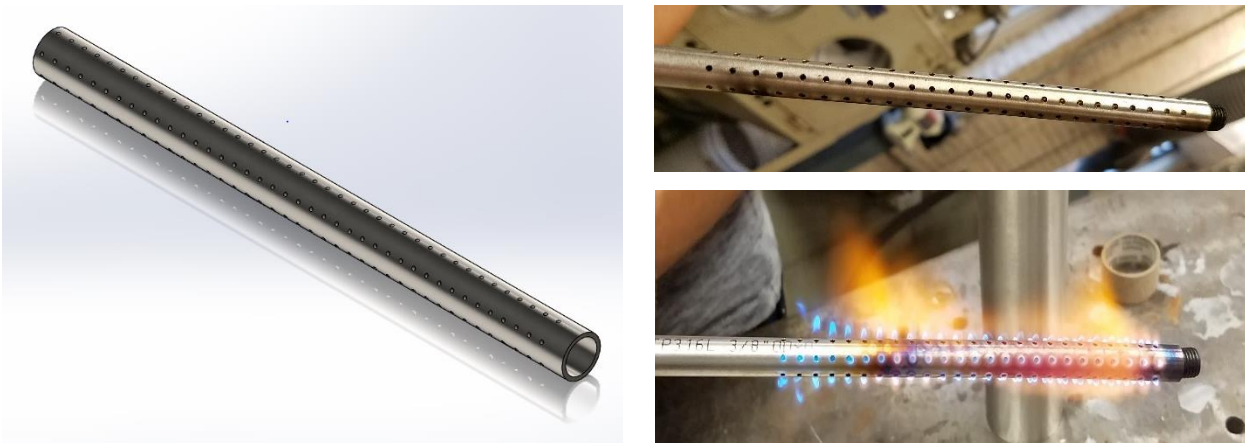

Figure 5. Boiler burner pipe. Shown as a CAD model on the left, and in physical form on the right.

Around the burner tube are five ½" tubes running parallel to the burner tube. In Figure 6, a sectioned view of this setup is shown (2.5 water tubes shown).

Figure 6. Sectioned view of CAD model of the burner and water pipes assembly

Two machined endplates hold the tubes in place and allow the flow of working fluid by connecting the five ½" pipes to the larger feed and exit pipes for the burner. Flat edges had to be added to the end plates for fixturing purposes. The endplates are shown in Figure 7, with the water pipes attached:

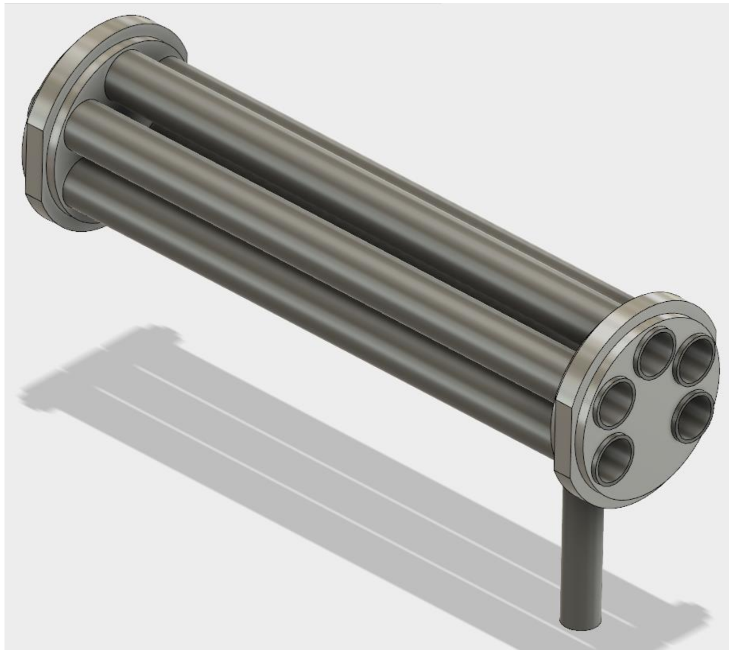

Figure 7. CAD model of the burner pipe, water pipes, and endplates assembly

Around the full heating core of the burner is a shell to hold in heat and contain the burner flames. The shell has openings on the top and bottom to allow airflow into the boiler to provide oxygen to the propane burner, as shown in the sectioned view below (Figure 8). The shell was then wrapped in insulation (not shown in picture).

Figure 8. Sectioned view of the CAD model of the burner pipe, water pipes, endplates, and boiler shell assembly

Finally, machined end cones take the flow of working fluid from the 2” end plates and reduce it down to the 1.5” diameter pipes for feed and exit of the working fluid, shown in Figure 9:

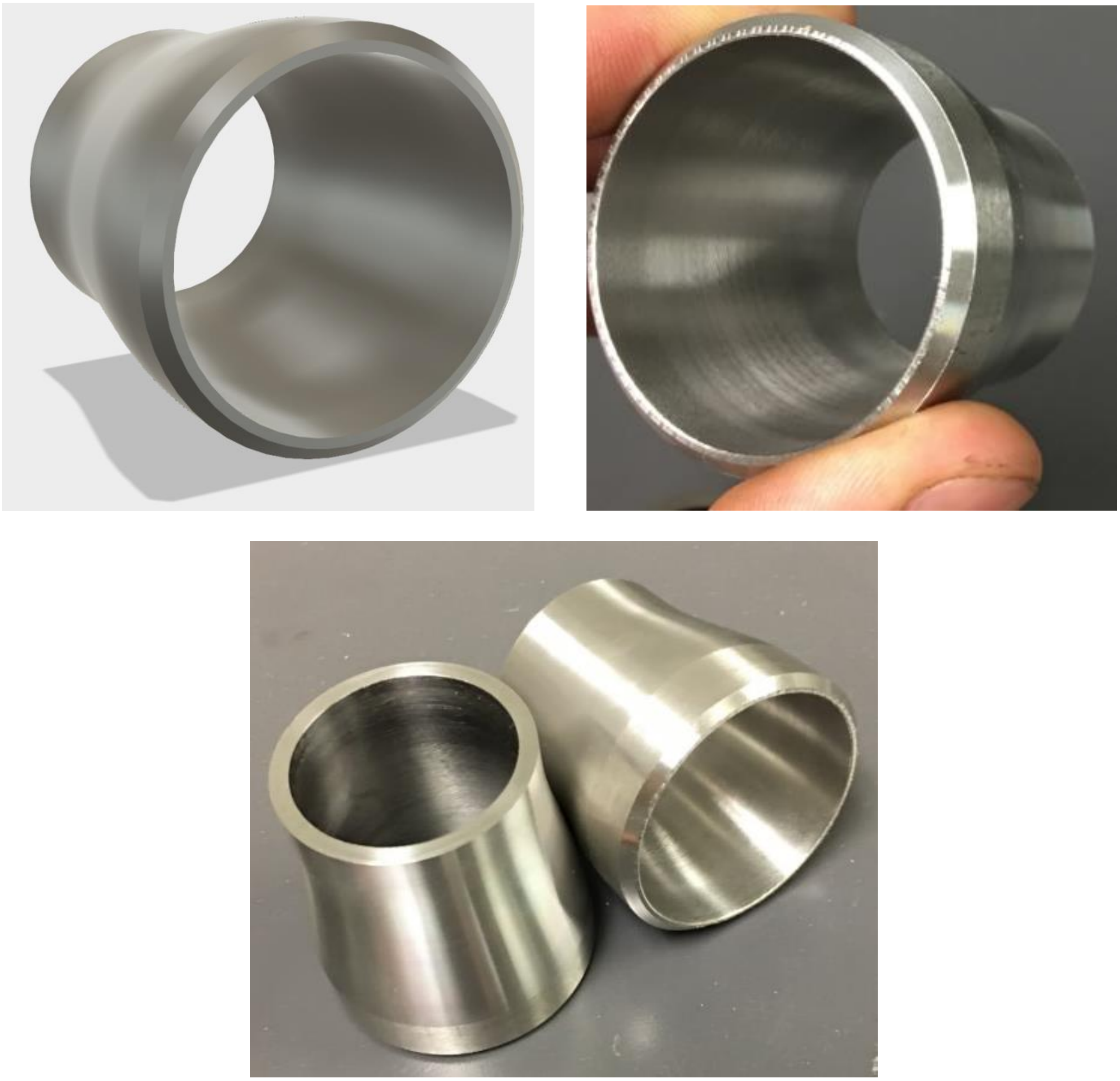

Figure 9. Boiler end cones. CAD model (top left), and machined parts (top right and bottom)

Below is a series of CAD renders of the different stages in the assembly of the boiler (Figure 10):

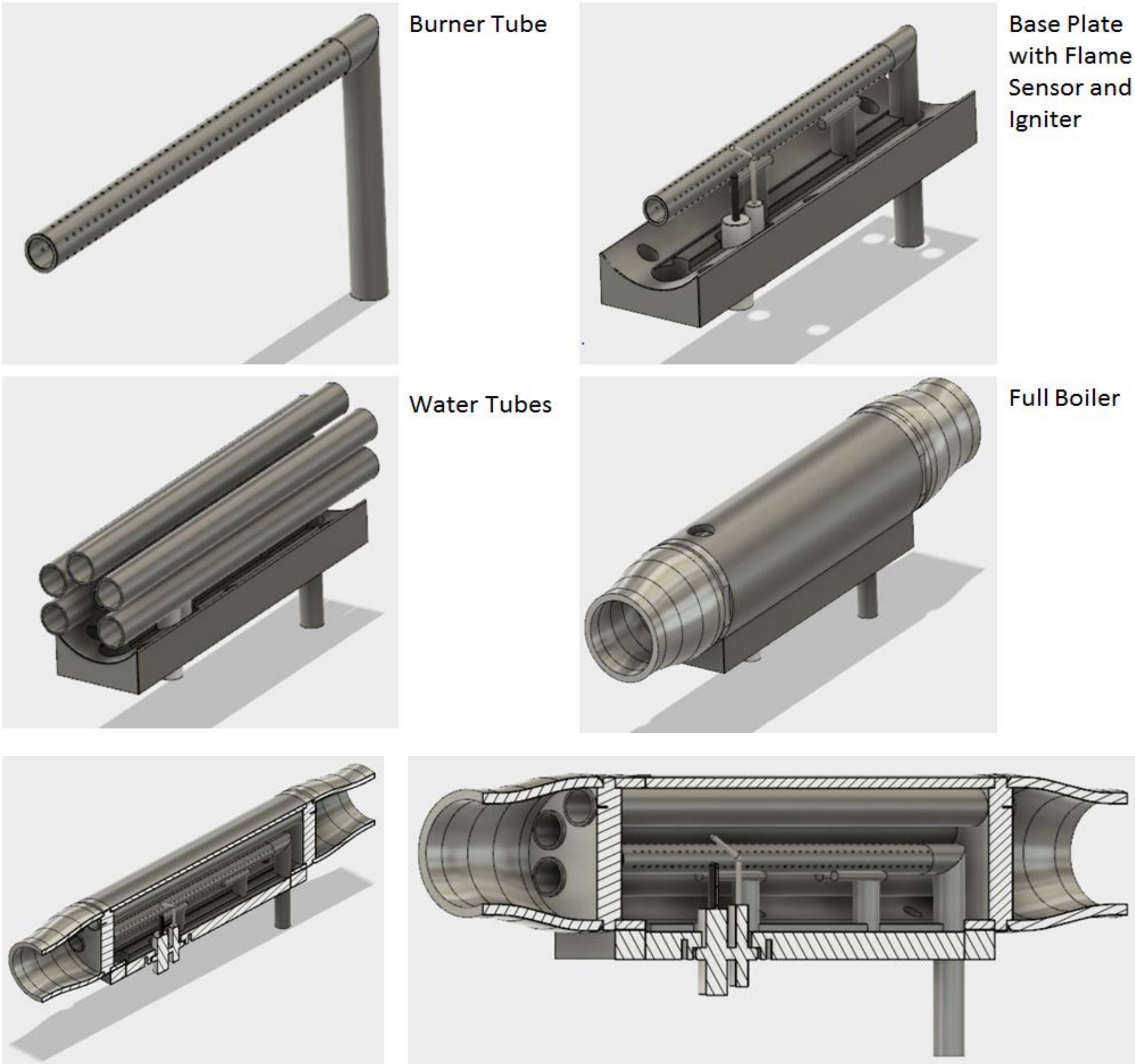

Figure 10. CAD renderings of the boiler at different stages of assembly

3.6 Propane Boiler Controls

Once the propane boiler is assembled there will be no way to access the burner pipe to light the flame. For this reason a flame sensor and igniter unit for a home water boiler will be used to sense the flame and ignite the propane as needed. Since most production boilers are for stationary applications they are designed for use with AC power. In order to run the flame sensor and igniter on DC power instead of 24V AC an Arduino Uno will be used as a control unit. A DC to DC converted is used to convert the 12V supply from the vehicle's battery to 24V for the flame sensor. After the converter the voltage is run through a relay that uses the output from the Arduino to turn on and off the igniter. The output from the flame sensor is run through an op amp to a voltage regulator and then to an input pin on the Arduino. This ensures that the signal into the Arduino is no greater than its maximum allowable input of 5v. If the flame is out then the Arduino outputs a signal to a relay that will send 24 volts to the igniter to relight the flame. If the flame fails to reignite after some quantity of time, the Arduino will send a signal to close an electronic solenoid valve on the propane supply line and indicate to the user this has happened with a warning light in the cab. The burner pipe serves as the ground for the flame sensor, and therefore must be grounded to the Arduino. In order to isolate the burner tube from the surrounding metal which would be grounded to the chassis of the vehicle, a thin wrap of Aerogel Insulation around the burner tube will hold it in place.

3.7 Overall System Design

For our bench test system, we designed our piping and component layout to make the final system be able to reuse as much of the test system as possible. The compact layout designed was then TIG welded together using 316 stainless steel 1.35” O.D. piping and recycled 316 flanges. For the test system a simplified design was used as shown below in Figure 11:

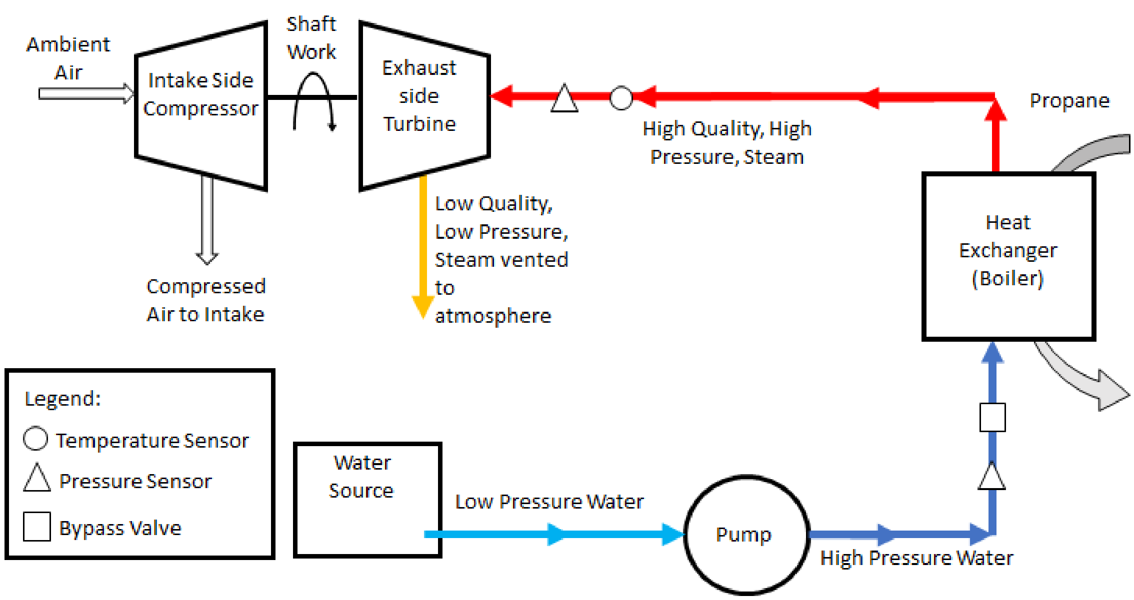

Figure 11. Symbolic representation of the bench test system

By using a large volume water source instead of setting up the system as a closed loop, the low-pressure side of the turbine could just be vented to the atmosphere. This setup simplified the system by eliminating the need for a condenser. This simplified system is sufficient for limited use during testing and allowed quick assembly and modification. Shown below in Figure 12 is the full test system:

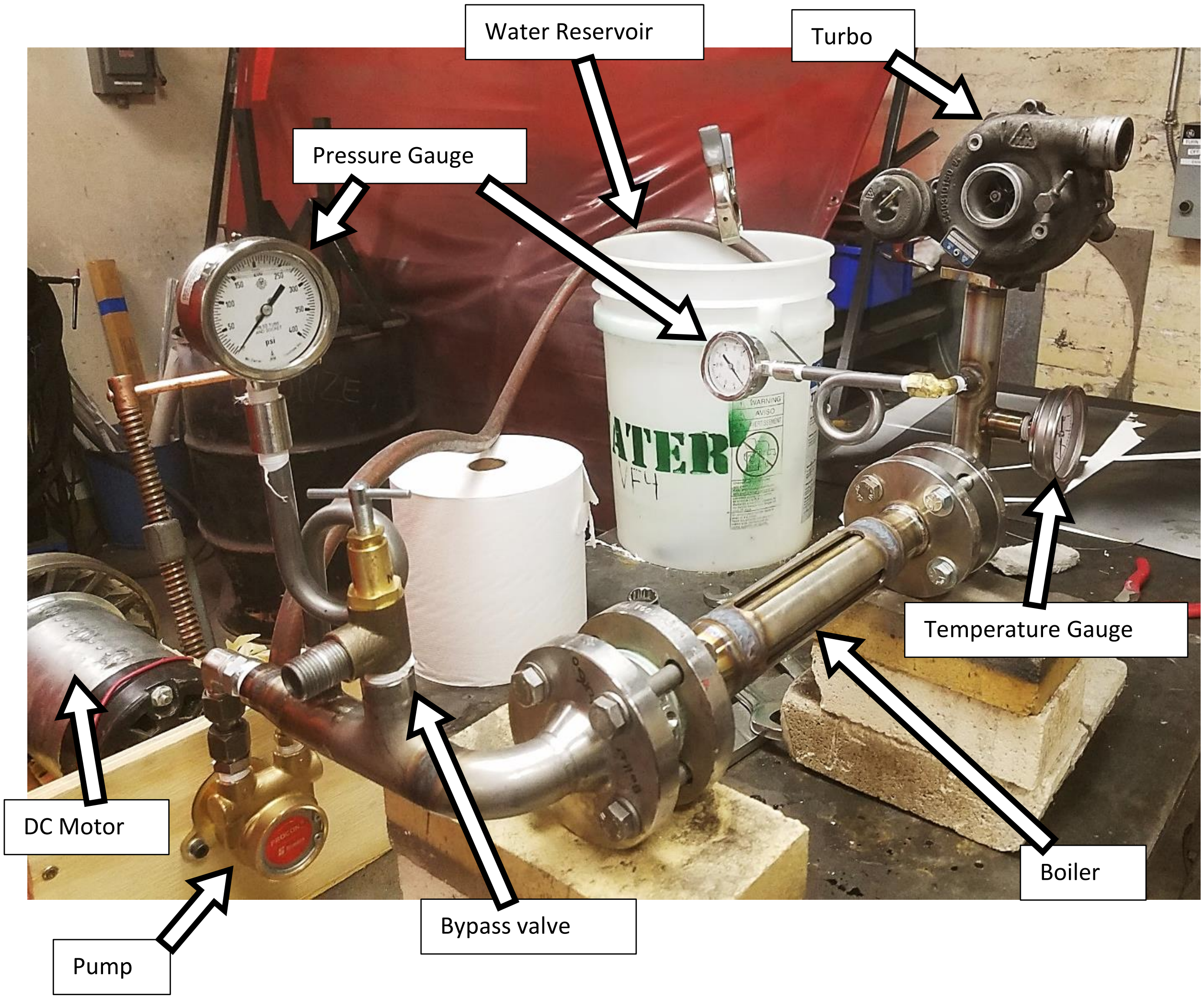

Figure 12. Fully assembled bench test setup

4 Discussion

4.1 Preliminary Heat Capacity Analysis

The mathematical analysis of the initial design for the engine at hand showed that the exhaust gases could realistically provide the Rankine cycle enough heat energy to match the compressor power requirements only in low-boost situations and at limited operational conditions. Therefore, it would be impractical to solely depend on the waste heat of the engine to power the full cycle for the 1.9L engine intended. Other options such as a different working fluid, sequential turbo, and electric motor turbo were considered.

A secondary thermodynamic analysis looking at the ideal Carnot efficiencies of the temperature of the exhaust gas and ambient air temperature showed that regardless of the working fluid, the exhaust gas simply did not have the energy required to power the compressor at a wide range of operating conditions. Stacking multiple turbos in sequence has been accomplished to improve engine performance but adds a lot of complexity even in traditional turbo setups. Electric turbos introduce an array of challenges because of the very high RPM they must spin at and the outright power requirements; the system would need to be significantly higher voltage than the car's battery can provide, and require very complex electrical controls, which are beyond the scope of this project.

After weighing the potential alternatives, the path of the second boiler was chosen as the best solution.

4.2 Parts Sourcing

The majority of the parts used in this project were custom designed and made in WPI's machine shop, located in Washburn Shops. Thanks to the high temperatures and pressures required by the system, all of the piping and flanges were made out of 316 stainless steel. This material is prohibitively expensive to buy new and therefore a local salvage yard was used to source nearly all of the metal. Fortunately, there was a variety of pipe diameters and even some flanges available that could be reused, so it was a matter of cutting the pipes to proper length and TIG welding them back together in the proper orientation with the addition of specific fittings for gages. The pump was a donation from Procon, and the motor was a salvaged 2 HP treadmill motor. This DC motor came with a custom-made motor controller that used pulse width modulation and a household rheostat to control the motor's speed and thereby the flowrate through the system.

4.3 Controls Circuitry for Boiler

The controls circuitry for the boiler utilized an Arduino Uno, two relays, a flame sensor and igniter, an operational amplifier (op-amp), a voltage regulator, breadboards and wires. Debugging the control circuit and Arduino code took significantly longer than expected, partly since we have very limited experience working with electrical control circuits.

The flame sensor works by sending an electrical signal to an electrode that is placed close to the burner tube, which shares a common ground with the control circuit. The flame contains enough charged particles to be able to close the circuit between the sensor electrode and the burner pipe; this way the flame is "sensed". To be able to use the flame sensor, the burner tube had to be electrically insulated from the rest of the boiler. Electrically insulating the burner tube from the rest of the boiler inspired much of the bottom plate design. This plate was designed to be detachable so that the sensor/burner could be serviced. The idea was that the burner tube would be wrapped in a high temperature insulation at the mounting points and tied to the small posts shown in Figure 13 below via wire.

Figure 13. Burner tube, mounted on the boiler bottom plate

When this setup was tested, it was revealed that there were two primary flaws. Firstly, the insulation around the burner was a tight fit to the surrounding water pipes and limited the amount of air that could reach the flame, thus limiting the amount of propane that could be burned inside the boiler. Secondly when there was a flame it was found that due to the proximity of the surrounding water tubes the signal from the flame sensor would switch between the burner pipe and the grounded water tubes, splitting the signal and resulting in very haphazard performance of the flame sensor. In order to combat these issues, the boiler was redesigned to be electrically insulated from the rest of the piping in its entirety. This was achieved by using high-temperature electrically-insulating gaskets at the flanges, along with non-conducting bushings to isolate the bolts.

4.4 Preliminary Testing of the System

After some early testing of individual components, the full boiler assembly was tested. It was found that the designed air inlet and exhaust holes still only provided a limited amount of air to the burner, and it was not enough to burn nearly enough propane. The next thing that was tried was the introduction of a small electric computer fan to actively blow air into the burner. This showed minimal improvement, so a slot was machined along the top of the boiler shell to promote air flow. This, too, did not have the desired effect. Next, air was introduced mixed with propane into the burner via a compressed air supply and an in-line manual air regulator. Though this had the most promising results, it was decided to proceed with a simple test with an oxy-acetylene torch as the heat source. A primary test with acetylene torch was done with a straight section of pipe mounted up instead of the boiler. This test showed that there was not enough heat being transferred to the working fluid to vaporize enough water to spin the turbine. Next, the boiler was swapped in, and the burner tube was directly connected to the oxy-acetylene supply. Even with the acetylene flame as the heat source and this increased surface area the boiler provided, there was still not enough heat to vaporize enough of the working fluid and spin the turbo.

4.5 Boiler Performance

A considerable amount of time was spent designing and constructing the boiler. Yet, its performance in testing was underwhelming. The design used was chosen to minimize the apparatus' dimensions, so that it could be directly transferred into a car's engine bay, without sacrificing heat exchanger surface area, and to bring the water tubes as close to the flames as possible in order to take full advantage of the high temperature of the flame. This resulted in a very compact boiler (a little over a foot long and under 3 inches OD), with very little space around the burner tube, which ultimately did not allow enough air to be mixed in with the propane to produce an efficient burn.

This was best demonstrated when compressed air was fed directly into the propane feed. When no additional air was supplied, there were large flames coming out of the boiler exhaust holes, indicating a very rich fuel mixture in the boiler. As the air allowed into the propane piping was gradually increased, the flames outside the boiler decreased in size, and for a very narrow window of airflow, were fully contained in the boiler. However, a very slight further increase in additional air resulted in the velocity of the air-propane mixture jets coming out of the burner tube to be too high to sustain the flames, thus completely extinguishing the flames.

To proceed with the same engine, the boiler would have to be redesigned from the ground up to be larger, in order to allow for more appropriate air-fuel mixtures to be exploited, and potentially incorporate a larger number of smaller diameter water tubes.

The mathematical modelling of the thermodynamics showed that it can be possible to implement the initially proposed system on a larger engine. Parts of the overall system that were not discussed in depth here, like the condenser sizing, would need to be further refined.

5 Summary

5.1 Conclusion

- Exhaust gas alone for the VW 1.9-L ALH engine does not have enough heat to power a Rankine Cycle turbo for practical applications.
- Just removing the restriction created by the exhaust side turbine could theoretically increase this engine's power by up to 30 HP for certain RPM and boost combinations, due to the decreased backpressure
- A larger engine is a more practical candidate for this type of system; theoretical calculations have shown that the system could work on exhaust gas alone for lower boost pressures for larger displacement engines.
- A larger, more effective water boiler is needed to be used in a bench test model as a standalone heating source.

5.2 Future Work

- Redesign the boiler to promote more airflow and increase surface area.
- Use a larger displacement engine, potentially gasoline-powered.
- Use different working fluids with lower saturation temperatures than water that can fully vaporize with lower heat requirements.
- Design a brake test to simulate engine load on compressor.
- Use any combination of the above to build a bench test prototype
- Use the data obtained from the bench test to refine the system as necessary
- Design mounting brackets to transfer the system into a vehicle
- Tune the system to optimize it for the desired operating conditions

Appendix A

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3	
1500	80	0.95	391.66	2665.8	
RPM	**1500**				
Boost (psig)	**25**				
		cp			
EGT in (oC)	**500**	1.093			
EGT out (oC)	**300**	1.044			
Power Req (kW)	7.2709725				
mdot air (kg/s)	0.0526772				
Qdot (kW)	11.257124				
T1 (oC)	95				
v4 (m^3/kg)	0.001038				
h4 (kJ/kg)	391.66				
h1 (kJ/kg)	393.13396				
h3 (kJ/kg)	2552.093				
mdot (kg/s)	h2 (kJ/kg)	Power out (kW)		h @EGTin	h sat
0.0036	3520.1128	3.484871		3473.1	2792.2
0.003654951	3473.1	3.366235			
0.0038	3355.535	3.053079		m. low	m. high
0.004	3207.4149	2.621288		0.003654951	0.004692
0.0042	3073.4015	2.189496			
0.0044	2951.5712	1.757704		P_avg	2.246444
0.0046	2840.3348	1.325912		ΔP	1.119791
0.004692294	2792.2	1.126653			
0.0047	2788.2667	1.110016			

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3	
1500	80	0.95	391.66	2665.8	
RPM	**2000**				
Boost (psig)	**25**				
		cp			
EGT in (oC)	**500**	1.093			
EGT out (oC)	**300**	1.044			
Power Req (kW)	9.69463				
mdot air (kg/s)	0.0702363				
Qdot (kW)	15.009498				
T1 (oC)	95				
v4 (m^3/kg)	0.001038				
h4 (kJ/kg)	391.66				
h1 (kJ/kg)	393.13396				
h3 (kJ/kg)	2552.093				
mdot (kg/s)	h2 (kJ/kg)	Power out (kW)		h @EGTin	h sat
0.0048	3520.1128	4.646495		3473.1	2792.2
0.004873268	3473.1	4.488313			
0.005	3395.0337	4.214703		m. low	m. high
0.00525	3252.086	3.674963		0.004873	0.006256
0.0055	3122.1337	3.135224			
0.00575	3003.4815	2.595484		P_avg	2.995259
0.006	2894.717	2.055744		ΔP	1.493055
0.00625	2794.6537	1.516004			
0.006256392	2792.2	1.502204			

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3	
1500	80	0.95	391.66	2665.8	
RPM	**3000**				
Boost (psig)	**25**				
		cp			
EGT in (oC)	**500**	1.093			
EGT out (oC)	**300**	1.044			
Power Req (kW)	14.54195				
mdot air (kg/s)	0.105354				
Qdot (kW)	22.51425				
T1 (oC)	95				
v4 (m^3/kg)	0.001038				
h4 (kJ/kg)	391.66				
h1 (kJ/kg)	393.134				
h3 (kJ/kg)	2552.093				
mdot (kg/s)	h2 (kJ/kg)	Power out (kW)		h @EGTin	h sat
0.0072	3520.113	6.969743		3473.1	2792.2
0.007309901	3473.1	6.73247			
0.0076	3355.535	6.106159		m. low	m. high
0.008	3207.415	5.242575		0.00731	0.009385
0.0084	3073.402	4.378992			
0.0088	2951.571	3.515408		P_avg	4.492888
0.0092	2840.335	2.651825		ΔP	2.239582
0.009384589	2792.2	2.253305			
0.0094	2788.267	2.220033			

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3	
1500	80	0.95	391.66	2665.8	
RPM	**4000**				
Boost (psig)	**25**				
		cp			
EGT in (oC)	**500**	1.093			
EGT out (oC)	**300**	1.044			
Power Req (kW)	19.38926				
mdot air (kg/s)	0.140473				
Qdot (kW)	30.019				
T1 (oC)	95				
v4 (m^3/kg)	0.001038				
h4 (kJ/kg)	391.66				
h1 (kJ/kg)	393.134				
h3 (kJ/kg)	2552.093				
mdot (kg/s)	h2 (kJ/kg)	Power out (kW)		h @EGTin	h sat
0.0095	3553.028	9.508886		3473.1	2792.2
0.009746535	3473.1	8.976627			
0.01	3395.034	8.429407		m. low	m. high
0.0105	3252.086	7.349927		0.009747	0.012513
0.011	3122.134	6.270447			
0.0115	3003.482	5.190968		P_avg	5.990517
0.012	2894.717	4.111488		ΔP	2.98611
0.0125	2794.654	3.032009			
0.012512785	2792.2	3.004407			

Similarly, for boost pressures of 20, 15, 10 and 5 psig, and for EGT in and out of 500, 300 ^{o}C and 600, 350 ^{o}C respectively.

Appendix B

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3
1500	80	0.95	391.66	2665.8
RPM	**1500**			
Boost (psig)	**5**			
		cp		
EGT in (oC)	**700**	1.14		
EGT out (oC)	**500**	1.099		
Power Req (kW)	2.181724273			
mdot air (kg/s)	0.059384647			
Qdot (kW)	13.2962225			
T1 (oC)	95			
v4 (m^3/kg)	0.001038			
h4 (kJ/kg)	391.66			
h1 (kJ/kg)	393.13396			
h3 (kJ/kg)	2552.093			
	mdot (kg/s)	h2 (kJ/kg)	Power out (kW)	
@ EGTin	0.004317003	3473.1	3.975989943	
@ Sat	0.005542249	2792.2	1.330732895	
		P_avg	2.653361419	
		ΔP/2	1.322628524	

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3
1500	80	0.95	391.66	2665.8
RPM	**2500**			
Boost (psig)	**5**			
		cp		
EGT in (oC)	**700**	1.14		
EGT out (oC)	**500**	1.099		
Power Req (kW)	3.636207121			
mdot air (kg/s)	0.098974412			
Qdot (kW)	22.16037084			
T1 (oC)	95			
v4 (m^3/kg)	0.001038			
h4 (kJ/kg)	391.66			
h1 (kJ/kg)	393.13396			
h3 (kJ/kg)	2552.093			
	mdot (kg/s)	h2 (kJ/kg)	Power out (kW)	
@ EGTin	0.007195005	3473.1	6.626649905	
@ Sat	0.009237082	2792.2	2.217888158	
		P_avg	4.422269031	
		ΔP/2	2.204380873	

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3
1500	80	0.95	391.66	2665.8
RPM	**4500**			
Boost (psig)	**5**			
		cp		
EGT in (oC)	**700**	1.14		
EGT out (oC)	**500**	1.099		
Power Req (kW)	6.545172818			
mdot air (kg/s)	0.178153942			
Qdot (kW)	39.88866751			
T1 (oC)	95			
v4 (m^3/kg)	0.001038			
h4 (kJ/kg)	391.66			
h1 (kJ/kg)	393.13396			
h3 (kJ/kg)	2552.093			
	mdot (kg/s)	h2 (kJ/kg)	Power out (kW)	
@ EGTin	0.012951009	3473.1	11.92796983	
@ Sat	0.016626748	2792.2	3.992198685	
		P_avg	7.960084256	
		ΔP/2	3.967885572	

P1 (kPa)	P4 (kPa)	x3	h_f3	h_g3
1500	80	0.95	391.66	2665.8
RPM	**6500**			
Boost (psig)	**5**			
		cp		
EGT in (oC)	**700**	1.14		
EGT out (oC)	**500**	1.099		
Power Req (kW)	9.454138515			
mdot air (kg/s)	0.257333471			
Qdot (kW)	57.61696418			
T1 (oC)	95			
v4 (m^3/kg)	0.001038			
h4 (kJ/kg)	391.66			
h1 (kJ/kg)	393.13396			
h3 (kJ/kg)	2552.093			
	mdot (kg/s)	h2 (kJ/kg)	Power out (kW)	
@ EGTin	0.018707013	3473.1	17.22928975	
@ Sat	0.024016414	2792.2	5.766509211	
		P_avg	11.49789948	
		ΔP/2	5.73139027	

Similarly, for boost pressure of 15 psig, and for EGT in and out of 700, 500 ^{o}C and 700, 350 ^{o}C respectively.