Effects of Heat Addition After the Exhaust Valve
on a Small Turbocharged Diesel Engine

By

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Abstract

Designers of engines have always looked for ways to improve the power to weight ratio of mobile internal combustion engines. This was especially true in aircraft engine design and engines for various forms of racing. Today designers are looking for ways to make everything from cars to road tractors to farm tractors lighter and thereby more efficient. In addition, in many cases these vehicles only need the maximum power that an engine can produce for a small amount of time. What is needed is a small, lightweight engine with the ability to produce a large amount of power for a short duration.

The work here describes one possible method for constructing just such a type of engine. By adding a combustion chamber in the exhaust flow between the engine exhaust valve and the turbine inlet on a turbocharged diesel engine, it should be possible to increase the turbine temperature. This will in turn allow the turbine to deliver more power to the compressor and create a higher inlet pressure and allow the engine to create more power. This paper describes both a computer simulation and an engine with this combustion chamber installed. There were however, problems with both the simulation as well as the test engine. While no quantitative data was obtained from the test engine, some valuable observations were made. The computer simulation yielded results and from these results and observations made while testing the engine with the combustion chamber installed it was determined that this design shows promise of creating an engine with higher specific power.

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A	Cylinder area
AFR_{diesel}	Air fuel ratio of diesel fuel
B	Bore diameter
CN	Centane number of fuel
cp_{air}	Specific heat of air
$cp_{exhaust}$	Specific heat of the exhaust
c_v	Specific heat
CV	Clearance volume
Dis	Engine Displacement
eff_{comp}	Compressor efficiency
eff_{turb}	Turbine efficiency
FR	Percentage of fuel consumed by combustion
$fuel_{\text{int}erp}$	fuel flow rate interpolated from engine speed and throttle position data
h_c	Average heat flux
IVC	Crankshaft angle at which the intake valve closes
m	Mass
$mair_{cyl}$	Mass of the air in the cylinder
m_{BD}	Mass of exhaust that exits during the blowdown event
$m_{beforeBD}$	Mass of combustion products in the cylinder before blowdown occurs
mc	Percentage of mixing controled fuel consumed by combustion
$mdot_{air}$	Mass flow rate of air
$mdot_{diesel}$	Mass flow rate of diesel fuel
$mdot_{exhaust}$	Mass flow rate of diesel fuel
m_e	Mass of the exhaust gasses
$m_{exhaust}$	Mass of the exhaust gasses
m_{Intake}	Mass of the air in cylinder

$mfuel_{cyl}$	Mass of fuel injected into the cylinder per engine cycle
$m_{remainingincyl}$	Mass of combustion products in the cylinder after blowdown occurs
N	Engine speed (Revolutions per minute)
N_{comp}	Corrected compressor speed
N_{turb}	Corrected turbine speed
N_{turbo}	Turbocharger speed
P	Pressure
P_1	Pressure at compressor inlet
P_2	Pressure at compressor exit
P_3	Pressure after intake valve
P_4	Engine exhaust pressure, turbine inlet pressure
P_5	Turbine exhaust pressure
$P_{beforeBD}$	Cylinder pressure immediately before the exhaust valve opens
pm	Percentage of pre-mixed fuel consumed by combustion
P_m	Motored engine pressure
PR	Pressure ratio
$PR_{turbine}$	Pressure ratio across the turbine
P_{tdc}	Pressure in the cylinder with the piston at top dead center
$P_{turbineinlet}$	Pressure at the turbine inlet
Q	Heat transfer
Q_{out}	Energy lost due to heat transfer to the cylinder wall
r	Connecting rod length
R	Universal gas constant
R_{air}	Gas constant for air
R_o	Universal gas constant
S	Stroke
T	Temperature

T_1	Temperature at compressor inlet
T_2	Temperature at compressor exit
T_4	Turbine inlet temperature
T_5	Turbine exhaust temperature
T_{BD}	Temperature of exhaust that exits during the blowdown event
$T_{beforeBD}$	Temperature of combustion products in the cylinder before blowdown occurs
T_e	Temperature of the exhaust gasses
$T_{exhaust}$	Temperature of the exhaust gasses
t_{id}	Ignition delay (crankshaft angle)
T_m	Motored engine temperature
t_{prime}	The crankshaft angle at which combustion begins, subtracted from the present crankshaft angle divided by the total crankshaft angle allowed for combustion
$T_{remainincyl}$	Temperature of combustion products in the cylinder after blowdown occurs
T_{tdc}	Temperature in the cylinder with the piston at top dead center
T_{wall}	Average temperature of the cylinder wall
V	Cylinder volume
V_{loss}	Percentage of pressure loss across the intake valve
V_p	Average piston velocity
w	Local average gas velocity in cylinder
W_{comp}	Corrected mass flow rate for the compressor
$Work_{compressor}$	Work required to drive the compressor
W_{turb}	Corrected turbine flow rate
β	Phase proportional factor (Percentage of fuel consumed by pre-mixed combustion)
θ	Crankshaft Angle
ρ_3	Density after intake valve
τ_{id}	Ignition delay (time)

γ_{air}	Ratio of specific heats for air
$\gamma_{exhaust}$	Ratio of specific heats for exhaust gasses
ϕ_{diesel}	Equivalence ratio of diesel fuel
x	Piston position

1. Introduction

1.1. *Background*

Since the design of the first internal combustion engine, engineers have been trying to increase the specific power, defined as power per mass. In the first days of aircraft design, Orville and Wilbur Wright had difficulties finding a power plant for their airplane that was both light and powerful enough to achieve flight. In the present time, car manufacturers go to much trouble to make cars light in order to increase fuel economy. Another recent example of a need for an engine with a high specific power output is the advent of diesel engines for use in light aircraft, an area once reserved for engines burning high octane gasoline. These are a few examples of many reasons why engineers seek to increase specific power output.

There are only two ways to achieve a high specific power – making the engine lighter or creating more power with a given engine. Reduction of engine weight is limited because engine parts must be strong enough to withstand the high stresses engines endure. At present, some gasoline engine components, such as cylinder heads and cylinder blocks, are made of aluminum to reduce weight. Switching to aluminum makes the engine lighter; therefore increasing specific power. Reducing rotating mass can also increase power. Lower mass allows the engine to operate at a higher speed by lessening the forces caused by acceleration. By producing the same amount of torque at a higher speed, the engine produces more power. While materials that are both light and strong, such as titanium, carbon composites and boron composites are beginning to be used in some racing applications, they are prohibitively expensive for most applications.

Therefore, the engine designer is left determining how to create more power from an engine of a given size. Many ways have been developed to do this including supercharging, tuning of the intake and exhaust, and computer modeling to optimize combustion chamber design. This thesis will address a system having the ability to increase the power of diesel engines. This system is a deviation from a normal turbocharged system, shown in Figure 1.1, to a modified system, shown in Figure 1.2, where a combustion chamber is added between the exhaust manifold and the turbocharger turbine inlet. The proposed system has the ability to produce more inlet pressure and thereby more power than a normal turbocharged diesel engine.

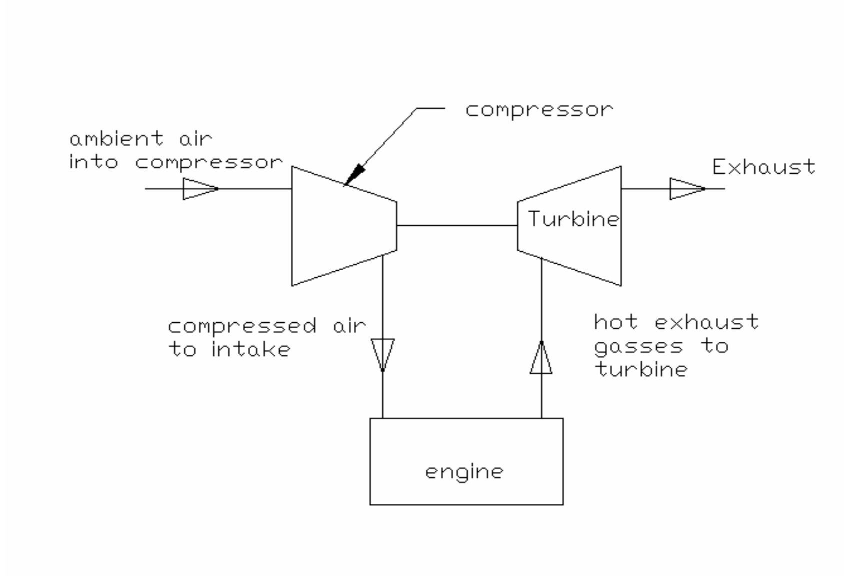


Figure 1.1 Airflow Path for a Normal Turbocharged Engine

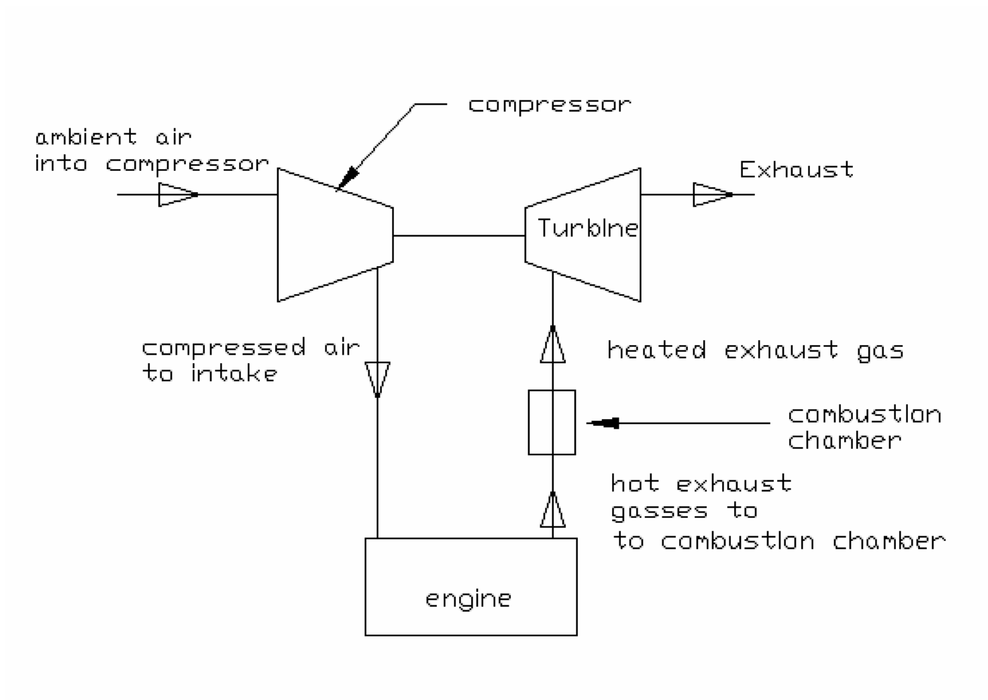


Figure 1.2 Airflow Path for an Turbocharged Engine Using the Proposed System

1.2. Diesel Engine Theory

The diesel engine, ideal cycle considered here, is a reciprocating, four-stroke, compression ignition engine that ideally has a constant pressure heat addition. Reciprocating describes the action of the pistons in the engine. Four-stroke describes the number of strokes by the piston required to produce one power stroke. Assuming the piston starts at the top of the cylinder bore, the first stroke is down drawing air into the cylinder. When the piston reaches the bottom of the cylinder, the intake valve that was allowing air to be drawn in is closed and the piston is driven upwards, compressing the air in the cylinder. Once the piston reaches the top of the cylinder, fuel is injected in a fine spray and the high pressure and temperature caused by the air compression ignite it. This combustion takes place in such a way that the pressure in the cylinder remains constant during part of the stroke as the piston is driven downwards, after the fuel is cut off the high temperature high pressure gases continue to expand until the piston reaches the bottom of the stroke. This is the power stroke. After the piston reaches the bottom of the cylinder the exhaust valve opens and the piston moves upward forcing the exhaust gases out of the cylinder so that the cycle can start over.

The cycle as explained above is the ideal cycle. There are differences between this ideal cycle and an actual diesel engine. In a diesel engine, the valves opening and closing do not occur exactly when the piston reaches the top or the bottom of the stroke. The fuel is injected just prior to the piston reaching the top of the stroke to account for delay in the combustion. And lastly the combustion does not take place at constant pressure.

1.3. Turbocharger Cycle

A turbocharger contains a turbine that drives a compressor. For use on small displacement engines of less than approximately 2000 cubic inches, most turbochargers have both radial flow compressors and turbines. The compressor compresses the air by accelerating the flow in the rotor, creating a rise in the total pressure and decelerating the flow into a scroll that slows the air and causes a static pressure rise. The radial inflow turbine withdraws energy from the engine’s hot exhaust flow in the opposite manner. Because the compressor and the turbine are on the same shaft, the speeds of both must be the same. The power produced by the turbine must equal the power absorbed by the compressor plus power lost due to friction.

Turbocharging an engine increases the power output of the engine by increasing the mass flow of air into the engine. The power produced by the engine increases since an increase in the mass flow of air also increases the mass flow of burnable fuel. The pressure rise across the compressor of the turbocharger increases the density of the air thereby increasing the mass flow through the engine. This suggests that engine designers would run as high an inlet pressure as possible in order to maximize engine power output. However, there are many drawbacks. Higher pressure requires more power from the turbine, which can increase the exhaust pressure needed to drive the turbine. Increased exhaust pressure has a negative effect on engine performance because it reduces scavenging. By increasing the inlet pressure the intake temperature also rises. This in turn produces higher cylinder and exhaust temperatures that can damage the engine. In particular, the exhaust valve is at risk of damage because there is little that can be done to cool it. Most of the heat transfer takes place from the valve stem to the valve guide. This is some distance from the valve face which has little cooling due to the small amount of surface area of the valve face in contact with the cylinder head. If engineers could design a system to operate primarily as a turbocharger and produce more boost only when needed they could have the benefits of a high-pressure turbocharger without any of the drawbacks. Even though the proposed system will develop high cylinder temperature and pressure as well as higher turbine inlet temperature the short duration of use should cause only a minor decrease in engine life.

1.4.Proposed turbocharger system

The test system consists of a combustion chamber placed between the exhaust manifold and the turbocharger turbine, shown in Figure 1.2. By increasing the temperature going into the turbine, the turbine can produce more work without the drawback of increasing the back pressure on the engine. This in turn can lead to higher compressor pressure ratios, a greater mass flow rate and more output power. The system does have disadvantages. Specifically, the system has high fuel consumption and, due to the extremely high turbine inlet temperatures, it can only be used intermittently. The higher levels of intake pressure will create a high intake temperature, especially on an engine with no aftercooler and with a compressor sized to operate at a lower pressure ratio. Because the system is only meant to operate when the application needs more power than the engine can normally provide these drawbacks are not significant issue. The proposed system should have little, if any, effect on engine operation under normal operation. In applications where the load varies greatly the need for full throttle power is infrequent and even then it is usually only for a short duration. Such applications include use in automobiles, heavy trucks and tractors and aircraft at takeoff. The test system would fit well this type of application.

While a search did not reveal any previous research on such a system, there has been work completed with similar intent, to investigate the affects of increased turbine inlet temperature. Rakopoulos et al [1] simulated the unsteady gas flow in a turbocharged, four-stroke, six-cylinder diesel engine using a computer simulation and then ran tests on the actual engine to compare to their simulation. Their program itself is quite complex, solving partial differential equations for gas flow dynamics simultaneously with the application of First Law of Thermodynamics to the cylinder to model the engine. They used this simulation to determine the influence of insulating the exhaust system prior to the turbocharger.

Of particular interest are the results reported for the tests on the insulated exhaust system. While Rakopoulos et al do not deal with heat addition as in my proposed system, their experiment gave similar results to those expected; an increase in the temperature at the turbocharger inlet due to the added insulation. They only performed a Second Law of Thermodynamics evaluation of the system and gave no data dealing with engine power or torque. They stated that insulating the exhaust manifold prior to the turbocharger “yields significant turbine mean-availability increases.” This indicates that adding heat will have a similar, yet greater, effect on the performance of the turbocharger, which will affect the overall engine performance.

1.5. Scope of Project

This purpose of this project is to test the feasibility of the proposed system. That is, would it work in both practice and theory? Several questions arise concerning the design and building of this system. These questions include the difficulty of initiating combustion in the exhaust stream, because of the small amount of oxygen present, and obtaining a steady flame in a pulsating exhaust flow with varying frequency and oxygen content. Also in question is what maximum temperature rise can be achieved before there is no more oxygen left in the exhaust stream. Of more practical concern is whether this temperature rise is enough to produce a marked increase in power that can be produced by the turbine. This project aims to answer these questions. The proposed system is not meant to be an optimal solution due to the complexities of designing a combustion chamber for use over varying mass flow rates, oxygen content and density. Instead, the goal of this research is to see if the idea is feasible, and to be a catalyst for further possible research.

The scope of the project initially was to model the system and then, based on the results of this model, build a working system and test it. Because of the problems with the fuel pump, which are discussed later, I was unable to complete testing of the system. This thesis deals mainly with the computer model and the predicted results.

1.6. Approach

The project is divided into three distinct parts: computer simulation, design and construction, and testing. There are two reasons for computer simulation. First, the computer simulation will determine if the system is feasible and will provide an indication of the expected results. Second, simulation determines the temperatures, pressures and flow rates needed to design the combustion chamber. The design and construction portion was the least time consuming. The combustion chamber was sized using data from the computer simulation, and it was constructed using stainless steel sheet metal. Testing was the final and most time-consuming portion.

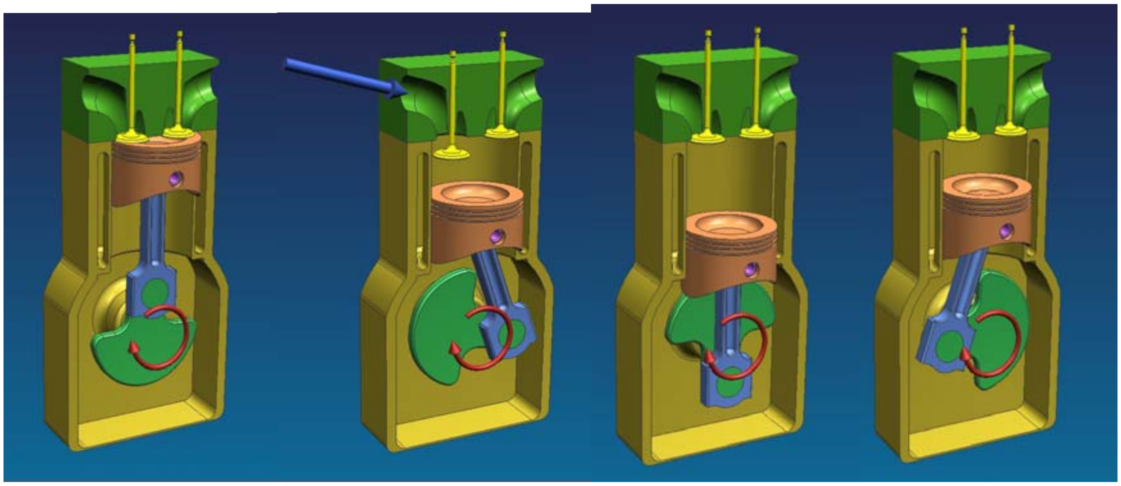
The intent was to fit the engine with only the turbocharger system and collect data at several operating speeds and fuel flow rates. Using this data to solve for the constants in the computer simulation, the simulation closely modeled the actual cycle. Once this was completed, the simulation could predict the performance of the proposed system. The engine was then to be tested using the proposed system at multiple engine speeds with various fuel flow rates of diesel fuel to the engine and propane to the pre-turbine combustion chamber.

2. Simulation

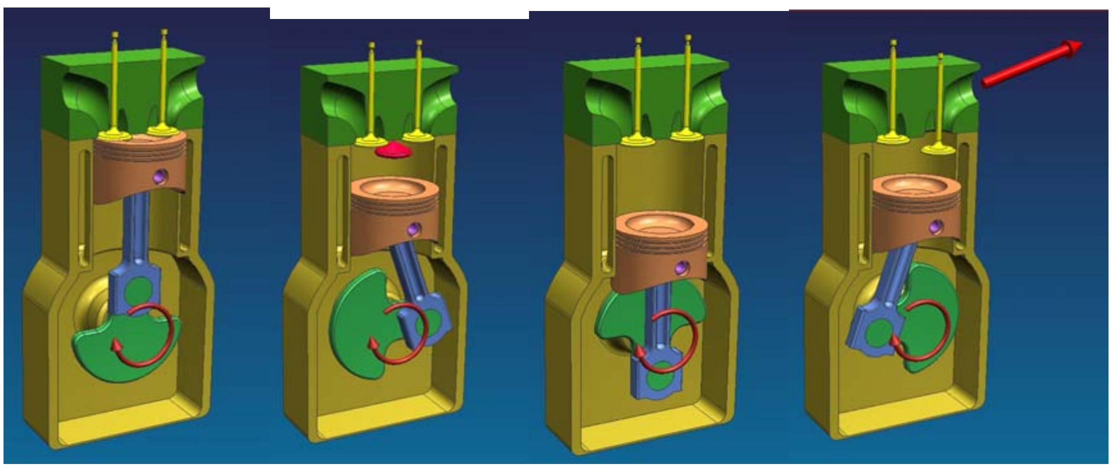
2.1. Overview of Program

Before designing the combustion chamber, a computer model of the entire engine cycle was produced. There were two reasons for doing this. First, the computer model could ensure that the idea was thermodynamically possible. Second, there were several variables, such as the turbine inlet temperature and the mass flow rate of air, which needed to be determined before the design of the combustion chamber could take place. Using MathCAD 2000™ (MathCAD), a computer program was written to simulate the process, see Appendix A. It needs to be noted that there are some small differences in the presentations of the equations in this paper and the equations in the program. These differences deal with the way in which MathCAD uses units and the process utilized to keep the units correct in the equations.

The operating cycle of the internal combustion engine can be separated into five events: intake, compression, combustion and expansion, and exhaust as shown in figure 2.1. These events were modeled as a function of the angle of the crankshaft because this position determines the position of every other engine component. By basing calculations on the crankshaft position, calculations are independent of time and engine speed. Because this is a turbocharged engine, equations were also developed for the turbocharger's compressor and turbine. The equations were created by curve fitting the compressor and turbine maps supplied by the manufacturer of the turbocharger, Garrett Boosting Systems, Inc, Torrance, CA. To summarize, a turbocharged engine can be split into two cycles. There is the turbocharger, which can be modeled as a Brayton cycle, and then there is the internal combustion engine cycle, which can be simulated by modeling the five events of the cycle list above.



0 degrees (TDC) | Intake | 180 degrees (BDC) | Compression



360 degrees | Combustion | Expansion | Exhaust

Figure 2.1: Engine Cycle

While modeling the systems separately is a straightforward task, combining the two cycles is somewhat more difficult. Conditions at the inlet of the engine must match those at the outlet of the compressor. Furthermore, the conditions at the exit of the exhaust manifold must match those at the entrance of the turbine. The conditions at the outlet of the compressor and the inlet conditions to the engine were matched by assuming a compressor outlet pressure. By knowing the pressure, the mass flow rate of air and the compressor exit temperature could be calculated. Once these values were known, and the diesel fuel addition was solved for, the propane addition was calculated. Subsequently, the mass flow rates of the fuels were added to the mass flow rate of the air to find the mass flow rate of the exhaust for use in the turbine equations. The turbine was then modeled to find the turbine inlet temperature and pressure. This portion of the turbocharger simulation had to be done before completing the engine model because the turbine inlet pressure (which must equal the exhaust pressure) is needed to model the exhaust blow-down process. After solving the turbine equations the engine model was completed. The combustion chamber was modeled using a mass and energy balance to arrive at an exhaust temperature, which was compared to the required turbine inlet temperature. The compressor exit temperature was then iteratively changed until the engine exhaust temperature matched the turbine inlet temperature and the simulation was complete for that operating point. An overview of the program cycle is shown on the following page in figure 2.2.

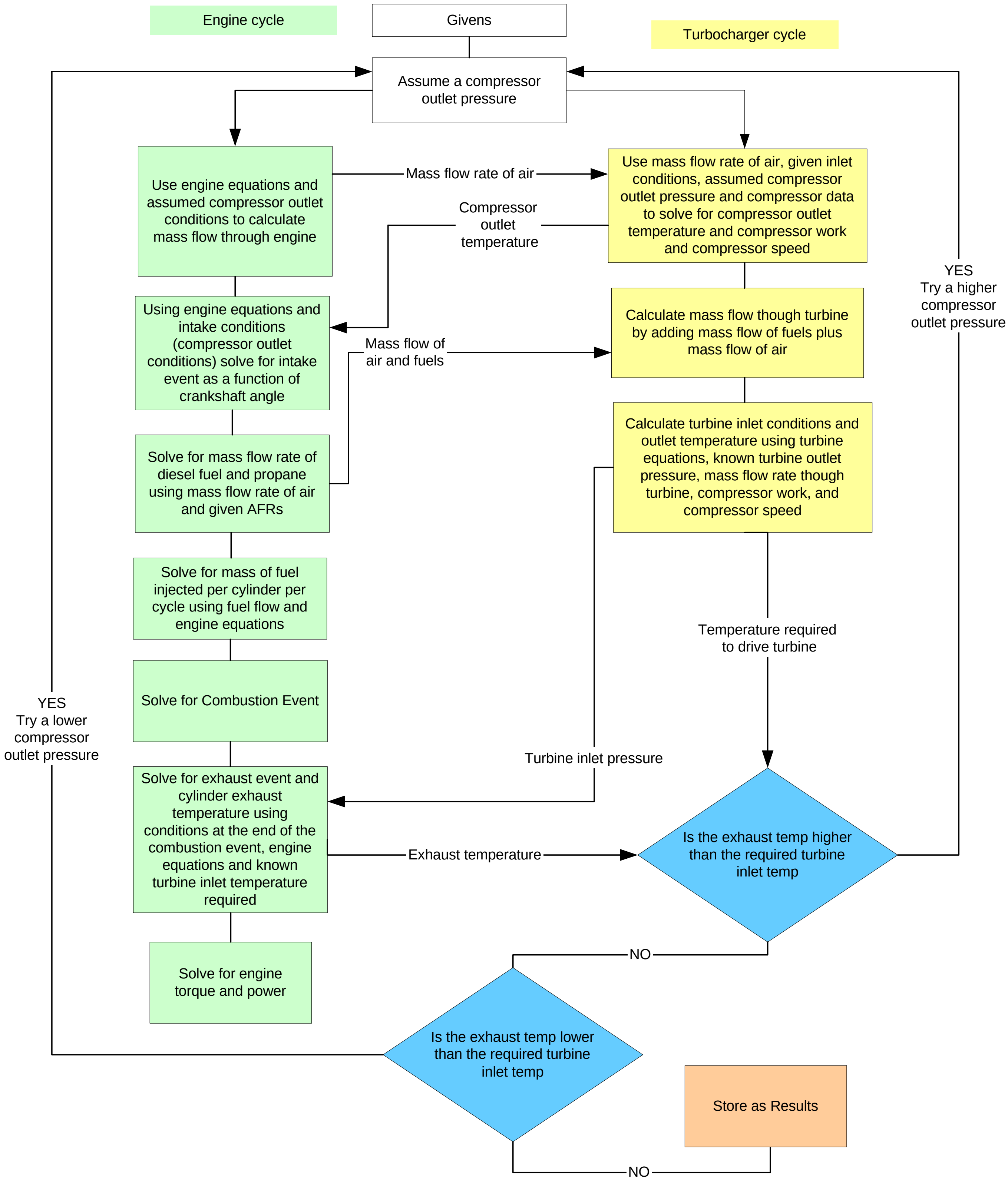


Figure 2.2: Simplified program flowchart showing interaction between engine and turbocharger cycles.

Many assumptions were made while writing the computer simulation to simplify the complexities inherent to an internal combustion engine. The working fluids (air and exhaust gases) were assumed to be ideal gases with constant specific heats. Other assumptions made will be explained as they occur throughout the description of the model. The program is described in the sequence in which it was written and not the order in which the air flows through the cycle.

2.2. Overview of the two cycles involved

2.2.1. Engine Cycle

As previously stated, the engine cycle can be broken down into five different events. The equations that model the engine cycle are solved step-wise as a function of crankshaft angle. The program models each event separately and solves for pressure as a function of crankshaft angle for each event. The mass added by the fuel is neglected in this cycle. This calculated pressure function is used along with the volume function to solve for indicated mean effective pressure (IMEP), which is the work produced by the working fluid per unit volume of displacement. The engine power is the IMEP multiplied by the displacement rate and the mechanical efficiency of the engine.

2.2.2. Turbocharger Cycle

The turbocharger cycle resembles a thermodynamic Brayton Cycle, which consists of a compressor and turbine connected by a shaft and separated by a combustion chamber. There are two differences between the Brayton Cycle and the turbocharger cycle. First, in the case of the turbocharger, the turbine inlet pressure is not necessarily, nor usually equal to the compressor exit pressure. Second, an internal combustion engine replaces the combustion chamber as was shown in Figure 1.1. The hot exhaust gases from the engine expand through the turbine to produce work, which in turn drives the compressor to supply dense air into the intake of the engine. Unlike the engine simulation, the turbocharger simulation is solved as though there is a steady flowrate through the system.

The various locations in the flow path are shown in both the physical view of the engine, figure 2.3, as well as the block diagram figure 2.4. These numbers and descriptions coincide with the subscripts in the program that relate to the turbocharger cycle. The simulation begins with air at ambient conditions (subscript “amb”). Air is drawn through the air filter into the compressor inlet (subscript “1”). The air then exits the compressor scroll and continues to the upstream side of the intake valve (subscript “2”). The air is then forced through the valve into the cylinder (subscript “3”) This is where the engine model begins. After completing the engine cycle the working fluid, now exhaust gasses, exits the exhaust manifold at an average condition and enters the combustion chamber. The exhaust exits the combustion chamber and enters the turbine (subscript “4”). The gasses exit the turbine scroll and enter the exhaust pipe (subscript “5”) and the return to the ambient environment.

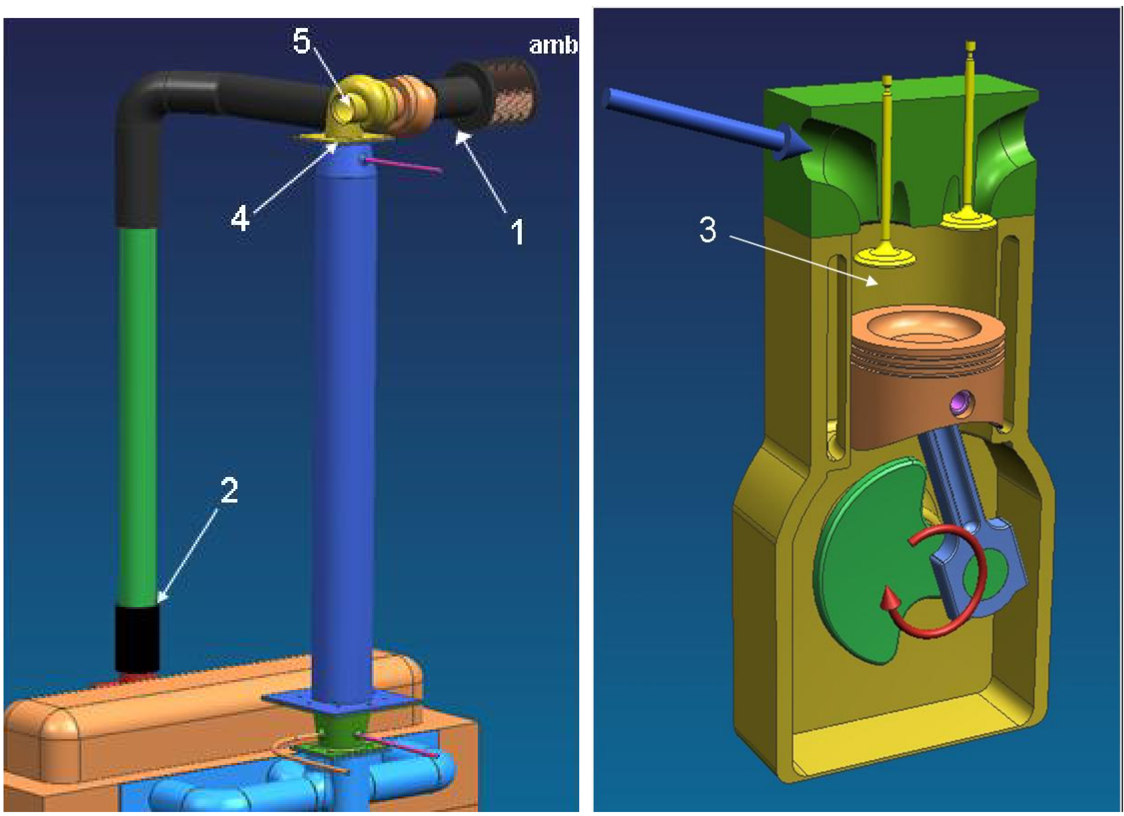


Figure 2.3: Physical view of the turbocharger cycle flow path

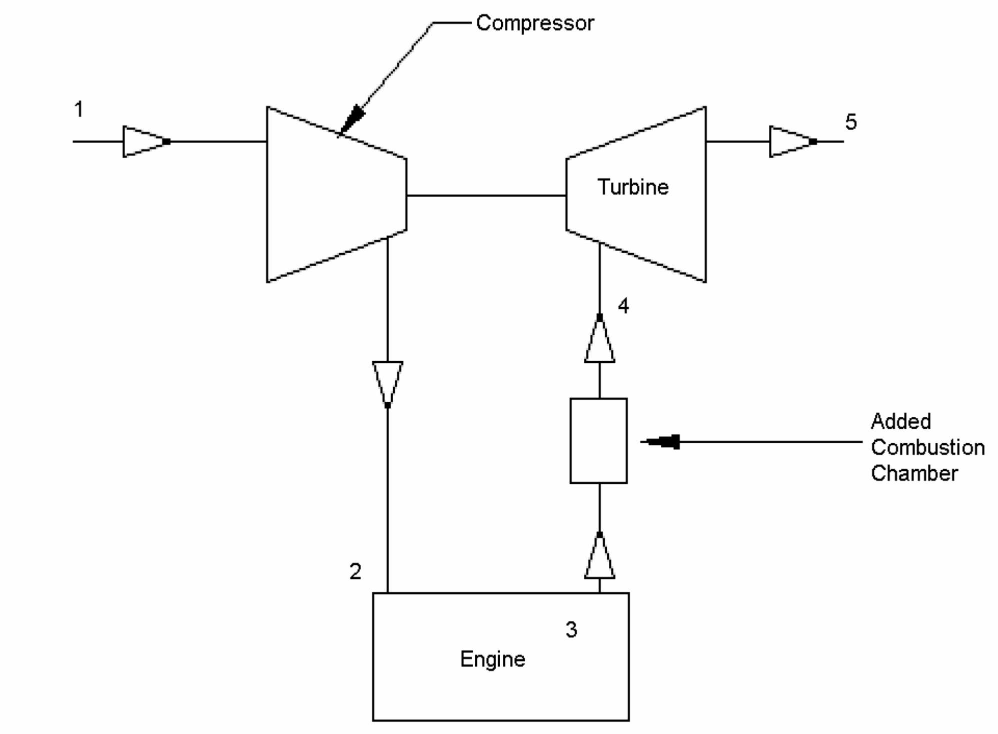


Figure 2.4: Block diagram of the turbocharger cycle flow path

2.3.Simulation Program

2.3.1. Basic Equations from Geometry

From a mechanical standpoint, an engine is an uncomplicated machine. It is not difficult to understand how the components relate to one another because each component has a direct relationship with the crankshaft that is only a function of crankshaft angle. From a thermodynamic and fluid flow standpoint, the process of air entering the engine and exhaust gases leaving the engine is very complex. Therefore, I will begin with the geometry. The first step was to enter all of the known values including, dimensions, engine geometry, and testing conditions. Next, the range of the crankshaft angles for which the program will solve must be established. Because this is a four-stroke engine requiring intake, compression, expansion, and exhaust strokes to complete one power cycle, the range for the crankshaft angles over which the program solves was set to 720 degrees. To ensure good resolution, the step size used was one eighth of a degree. While this resolution is only truly needed during the combustion phase of the model, with current technology and the speed of computers little is gained by using a variable step size. Equation 2.1 shows the equations for the motion of the piston, as a function of crankshaft angle.

$$x(\theta) = \left(\frac{S}{2} + r\right) - \left[\frac{S}{2} \times \cos(\theta) + \sqrt{r^2 - \left(\frac{S}{2} \times \sin(\theta)\right)^2}\right]$$

Equation 2.1

From Equation 2.1, it is simple to solve for cylinder volume as a function of crankshaft angle, as shown in Equation 2.2, and cylinder surface area as a function of crankshaft angle, as shown in Equation 2.3. Surface area is calculated assuming a flat head and flat piston. While the head is flat, the piston is dished which increases the area for heat transfer. This area is neglected because it is both small and unknown. The derivatives of both Equations 2.2 and 2.3 are:

$$V(\theta) = CV + x(\theta) \times \frac{B^2 \times \pi}{4}$$

Equation 2.2

$$A(\theta) = \left(\frac{B}{2}\right)^2 \times \pi + B \times \pi \times x(\theta)$$

Equation 2.3

2.3.2. Compressor and Intake Simulation

The fluid flow through the intake valve is greatly simplified. It is assumed that the intake valve will open instantaneously at top dead center (TDC), and that this will occur simultaneously with the closing of the exhaust valve. These two assumptions make the possibility of flow reversion, exhaust flow out of the intake valve and intake of exhaust through the exhaust valve, impossible. The intake valve is also assumed to close at bottom dead center (BDC). A restriction is caused by the intake valve which incurs a pressure loss such that the air entering the cylinder is at a lower pressure than at the exit of the compressor. A pressure loss across the intake valve is assumed. This number was arrived at by assuming a volumetric efficiency that was reasonable for a naturally aspirated, low swirl, diesel engine and applying the loss to the turbocharged engine.

In a further effort to simplify the intake event modeling, the program does not account for the heat transfer in this process even though the coefficient is high. The heat transfer coefficient is high because of the high velocity of the intake air. Nevertheless, because the temperature difference between the incoming air and the cylinder wall temperature is small, the heat transfer is slight. By assuming that no heat transfer takes place, the program gives a solution for horsepower that is marginally higher than the engine produces when the intake temperature is below the cylinder wall temperature. If the cylinder wall temperature is lower than the intake temperature, the program's solution for power will be lower than the actual power.

After determining how to model the intake process, the inlet conditions must be determined. To establish these, the mass flow through the compressor must be equal to the mass flow through the engine. The mass flow through the engine is equal to

$$mdot_{air} = \frac{N}{2} \times Dis \times \rho_3$$

Equation 2.4

where

$$\rho_3 = \frac{P_3}{R_{air} \times T_2}$$

Equation 2.5

where

$$P_3 = P_2 \times \left(1 - V_{loss}\right)$$

Equation 2.6

And

$$T_2 = T_1 \times \left[1 + \frac{\left[PR^{\frac{(\gamma_{air} - 1)}{\gamma_{air}}} - 1 \right]}{eff_{comp}} \right].$$

Equation 2.7

To solve for the compressor efficiency, the compressor map curve fits were used. This data was imported into MathCAD where it was first curve fitted and then interpolated using a function in MathCAD. This data is not included because it is proprietary information of Garret Boosting Systems Inc. The data is the compressor efficiency as a function of corrected mass flow rate and compressor pressure ratio, and corrected compressor speed as a function of corrected mass flow rate and compressor pressure ratio. Therefore, two necessary variables for the curve fit are the pressure ratio across the compressor, which is known,

$$PR = \frac{P_2}{P_1}$$

Equation 2.8

and the corrected mass flow rate

$$W_{comp} = \frac{mdot_{air} \times \sqrt{\frac{T_1}{545°R}}}{P_1}.$$

Equation 2.9

However, to find the corrected mass flow rate, the mass flow rate of air is needed, which is the variable that was being solved for. Therefore, these equations were solved simultaneously to find the conditions at the intake of the engine. After solving these equations, the turbocharger shaft speed is solved for, as it will be needed later to solve for the turbine conditions. The shaft speed is determined by first using the equation that was fit to the compressor maps for pressure ratio as a function of corrected mass flow and corrected compressor speed. (Neither this map, nor the turbine maps which are mentioned later are included in the appendix because they are proprietary data from the manufacture.) This gives the corrected speed, which can then be used to find the turbocharger shaft speed by using

$$N_{turbo} = \sqrt{\frac{T_1}{545°R}} \times N_{comp} .$$

Equation 2.10

The solution of the turbine equations requires that the power required to drive the compressor be known. Equation 2.11 was used to solve for this power:

$$Work_{compressor} = mdot_{air} \times cp_{air} \left(T_2 - T_1\right)$$

Equation 2.11

After knowing the inlet conditions, the modeling of the actual intake event is straightforward. The mass of the air in the cylinder as a function of the crankshaft angle can be solved for using Equation 2.12.

$$m_{Intake} = V \times \rho_3$$

Equation 2.12

This equation can be used to solve for the mass of air from the time the intake valve opens until the time the exhaust valve opens. For all crankshaft angles less than the point at which the intake valve closes the mass is the volume of the cylinder at that point times the density. For times after the intake valve closes the mass in the cylinder remains constant until the exhaust valve opens. The event of the exhaust valve opening is more complex and is explained later.

2.3.3. Fuel Addition

After solving for the mass flow rate of air, the mass flow rate of diesel fuel and propane can be found. Testing of the engine was to provide data on the fuel flow rate as a function of throttle position and engine speed. This data was then to be interpolated to yield the fuel flow in the units of cubic centimeters per minute. Because of the problems detailed in the later sections, this data was not available. Therefore, a diesel equivalence ratio was assumed and the fuel flow rate was calculated from this using the following equation.

$$mdot_{diesel} = \frac{\phi_{diesel} \times mdot_{air}}{AFR_{sdiesel}}$$

Equation 2.13

The mass of fuel injected per cylinder per injection for the 4 cycle 3 cylinder engine is then solved using equation 2.14.

$$mfuel_{cyl} = \frac{mdot_{diesel}}{N \times \frac{3}{2}}$$

Equation 2.14

The program can now solve for the amount of propane to add. First, the amount of air left after combustion is calculated by subtracting the air that was used for combustion from the total mass flow of air, as shown in Equation 2.15.

$$mdot_{airremaining} = mdot_{air} - mdot_{diesel} \times AFR_{sdiesel}$$

Equation 2.15

This assumes that the all of the fuel injected is burned with a stoichiometric amount of air. This is an approximation, as the amount of air left in the exhaust flow will most likely be more, especially at high diesel fuel equivalence ratios. Multiplying the desired propane equivalence ratio by the stoichiometric ratio by the mass flow rate of air left in the exhaust gave the amount of propane that can be added. The next step is to solve the turbine model.

2.3.4. Turbine Simulation

To solve the turbine model, five equations and two interpolations of curve fits must be solved. The data which was fitted to the curve is the turbine efficiency as a function of corrected mass flow rate and turbine pressure ratio, and corrected turbine speed as a function of corrected mass flow rate and turbine pressure ratio. This data was imported into MathCAD where it was fit to a curve, and then interpolated. To solve for the turbine efficiency and the corrected mass flow, the turbine pressure ratio and the corrected speed must be known. As with the compressor maps, this data is not included due to being proprietary data obtained from the manufacturer.

The corrected turbine speed and corrected mass flow can be calculated using the reference conditions given by Garrett Boosting Systems Inc. and the equations for corrected turbine speed and corrected mass flow of

$$N_{turb} = \frac{N_{turbo}}{\sqrt{\frac{T_4}{519°R}}}$$

Equation 2.16

and

$$W_{turb} = \frac{\frac{mdot_{exhaust}}{\frac{lb}{\min}} \times \sqrt{\frac{T_4}{519°R}}}{\left(\frac{P_4}{29.92 \cdot inHG}\right)}.$$

Equation 2.17

In order to use these equations the inlet temperature and pressure to the turbine must be known. Equation 2.18 solves for the inlet temperature by balancing the energy between the turbine and the compressor.

$$mdot_{air} \times cp_{air} \times (T_2 - T_1) = mdot_{exhaust} \times cp_{exhaust}(T_4 - T_5)$$

Equation 2.18

Equation 2.19 solves for the required inlet pressure to the turbine using the definition of the pressure ratio for a turbine.

$$PR_{turbine} = \frac{P_4}{P_5}$$

Equation 2.19

To solve for the turbine outlet temperature in equation 2.18 the definition of the turbine efficiency is substituted into a form of the ideal gas law to form

$$T_5 = T_4 \left[1 - eff_{turb} \times \left[1 - \left(\frac{1}{PR_{turbine}} \right)^{\frac{\gamma_{exhaust} - 1}{\gamma_{exhaust}}} \right] \right]$$

Equation 2.20

Equations 2.19 and 2.20 both contain the variable $PR_{turbine}$, which is the variable that was being solved for. Consequently, because there are seven variables and seven unknowns the equations to model the turbine are solved simultaneously. Now that inlet pressure to the turbine is known, which is equal to the pressure after the exhaust blowdown, the remainder of the engine cycle can be modeled.

2.3.5. Compression Event (180 degrees until ignition)

The compression event takes place after the intake valve shuts. The model has this happen instantaneously at bottom dead center (BDC), and before the ignition of the air fuel mixture that happens at approximately 350 degrees. However, the point at which ignition takes place is a function of engine speed, equivalency ratio, intake charge temperature and pressure, and the centane rating of the fuel. Solving for the point at which ignition takes place calls for solving many other equations that affect ignition delay, explained in the following section. For now, it will suffice to state that the compression ends when ignition begins.

As with the intake event, heat transfer is neglected. In this case the heat transfer coefficient is low because the velocity is low. The difference between the air temperature and the cylinder wall temperature increases as the air is compressed, but not significantly enough to make up for the low heat transfer coefficient.

For this event, the air is modeled as an ideal gas with no heat transfer. The two governing equations for this process are therefore the equation of state for an ideal gas, see Equation 2.21, and the first law of thermodynamics for an ideal gas with a constant specific heat, Equation 2.22.

$$PV = mRT$$

Equation 2.21

$$m \times c_v \times \frac{d}{d\theta} T = \frac{d}{d\theta} Q - P \times \frac{d}{d\theta} V$$

Equation 2.22

As shown by Ferguson [2], a convenient way to solve for the pressure for in the cylinder are to take the natural logarithm of both sides of the ideal gas law (equation 2.21) and then differentiate them with respect to crank angle which yields

$$\frac{1}{P} \times \frac{d}{d\theta} P + \frac{1}{V} \times \frac{d}{d\theta} V = \frac{1}{T} \times \frac{d}{d\theta} T \ .$$

Equation 2.23

Next, the right hand side of first law of thermodynamics for an ideal gas, see Equation 2.22, is divided by mRT and the right hand by PV. Rearranging gives Equation 2.24.

$$\frac{1}{T} \times \frac{d}{d\theta} T = (\gamma - 1) \times \left(\frac{1}{PV} \times \frac{d}{d\theta} Q - \frac{1}{V} \times \frac{d}{d\theta} V \right).$$

Equation 2.24

Combining this equation with equation 2.22 and setting the heat transfer equal to zero yields

$$\frac{d}{d\theta} P = -\gamma \times \frac{P}{V} \times \frac{d}{d\theta} V \ .$$

Equation 2.25

This equation was chosen because it was already being used later in the program in the program to model the combustion event by including the heat transfer.

The cylinder pressure (P) for the intake event can then be solved using the ordinary differential equation solver in MathCAD 2000TM. Once this is solved, the cylinder pressure is set equal to the intake pressure for crankshaft angles less than 180 degrees, when the intake closes, and equal to the motored engine pressure after the intake valve closes. The motored temperature as a function of the crankshaft angle can then be calculated. For angles less than 180 degrees, the temperature is equal to the compressor outlet temperature. Equation 2.26, the ideal gas law, is used for solving for angles greater than 180 degrees during the compression event.

$$T_m = \frac{P_m \times V}{mair_{cyl} \times R_{air}}$$

Equation 2.26

2.3.6. Combustion Event (time of ignition until all fuel is burned)

The first step in evaluating the combustion event is to calculate when the compression ends and ignition begins. There is a delay from the time the fuel is injected into the cylinder until the time combustion begins. This time is known as ignition delay. While the correlation to calculate ignition delay developed by Watson [3] is widely used, Heywood [4] recommends the correlation that was developed by Hardenburg and Hase [5], which is

$$t_{id} = (.36 + .22 \times V_p) \times \exp\left[E_a \times \left(\frac{1}{R_o \times T_{tdc}} - \frac{1}{17190}\right) \times \left(\frac{21.2}{P_{tdc} - 12.4}\right)^{.63}\right]$$

Equation 2.27

where

$$E_a = \frac{618840}{(CN + 25)}.$$

Equation 2.28

The reason Heywood suggests this correlation instead of Watson’s is because it takes into account the dependence on engine speed and fuel type. Furthermore, it only requires calculating pressure and temperature at one point, which is TDC [1]. The pressure and temperature for this point were calculated previously in equations 2.25 and equation 2.26 respectively.

After solving for ignition delay, the point at which combustion begins, can be solved for by subtracting the ignition delay from the manufacturer's dynamic ignition advance, a value given by the engine manufacturer, in this case 13 degrees before top dead center (or 347 crankshaft degrees). This gives the point at which ignition begins. The model that was chosen to describe the heat release was developed by Watson [6]. It was chosen because it is simple to implement, and the primary focus of the model was the effects of the combustion chamber on the output of the engine; the combustion was of modest concern. The formulae used for this are empirically based and therefore not as accurate as a model that simulates fuel spray and its related vaporization and combustion. The simulation used contains a formula for pre-mixed combustion

$$pm = 1 - \left(1 - t_{prime}{}^{K_1}\right)^{K_2},$$

Equation 2.29

and a formula for mixing controlled combustion

$$mc = 1 - \exp\left(-K_3 \times t_{prime}{}^{K_4}\right).$$

Equation 2.30

The variable t_{prime} is defined as the crankshaft angle at which combustion begins, subtracted from the present crankshaft angle divided by the total crankshaft angle allowed for combustion. Heywood suggests this to be 125 degrees [4]. The coefficients in these equations are obtained empirically, and the ones used for this simulation were taken from Heywood's *Internal Combustion Engine Fundamentals*, and are as follows:

$$K_1 = 2 + 1.25 \times 10^{-8} \times \left[\tau_{id} \times RPM\right]^{2.4}$$

$$K_2 = 5000$$

$$K_3 = \frac{14.2}{\phi_{diesel}{}^{.644}}$$

$$K_4 = .79 \times K_3{}^{.25}$$

Where τ_{id} is the amount of time the ignition delay takes. This time was found by dividing the ignition delay angle by 360 degrees and then by the engine speed. Equations 2.29 and 2.30, are related by a phase proportionality factor (β) in the following equation which gives ratio of fuel burned at time a given time.

$$FR = \beta \times pm + (1 - \beta) \times mc$$

Equation 2.31

Where the phase proportional factor, the percentage of the fuel burned in the premixed combustion is given by

$$\beta = 1 - \frac{a \times \phi_{diesel}{}^{b}}{\tau_{id}{}^{c}}$$

Equation 2.32

The ranges for the constants for a turbocharged engine are given by Watson [6] are

$$.8 < a < .95$$
$$.25 < b < .45$$
$$.25 < c < .5$$

Equation 2.32 proved difficult to implement because of the very small ignition delays calculated from equation 2.27. The problem has to do with the form of the equation. If the ignition delay is small and the equivalency ratio is high, then β can become negative as can be seen in Figure 2.5. This cannot be the case because, on inspection of equation 2.31, the heat released by the premixed fuel addition would become negative and the heat released by the mixing controlled combustion would be greater that the amount of energy available by the combustion of the fuel. This problem was overcome by first solving β using equation 2.32 for a range of ignition delays and equivalency ratios common to diesel engines. Points at zero were then added and curves were fit to this new data using MathCAD. This was done so that when the ignition delay equaled zero, β equaled zero. This insures that β will not become negative because the ignition delay cannot become zero. The results of this can be seen in figure 2.5

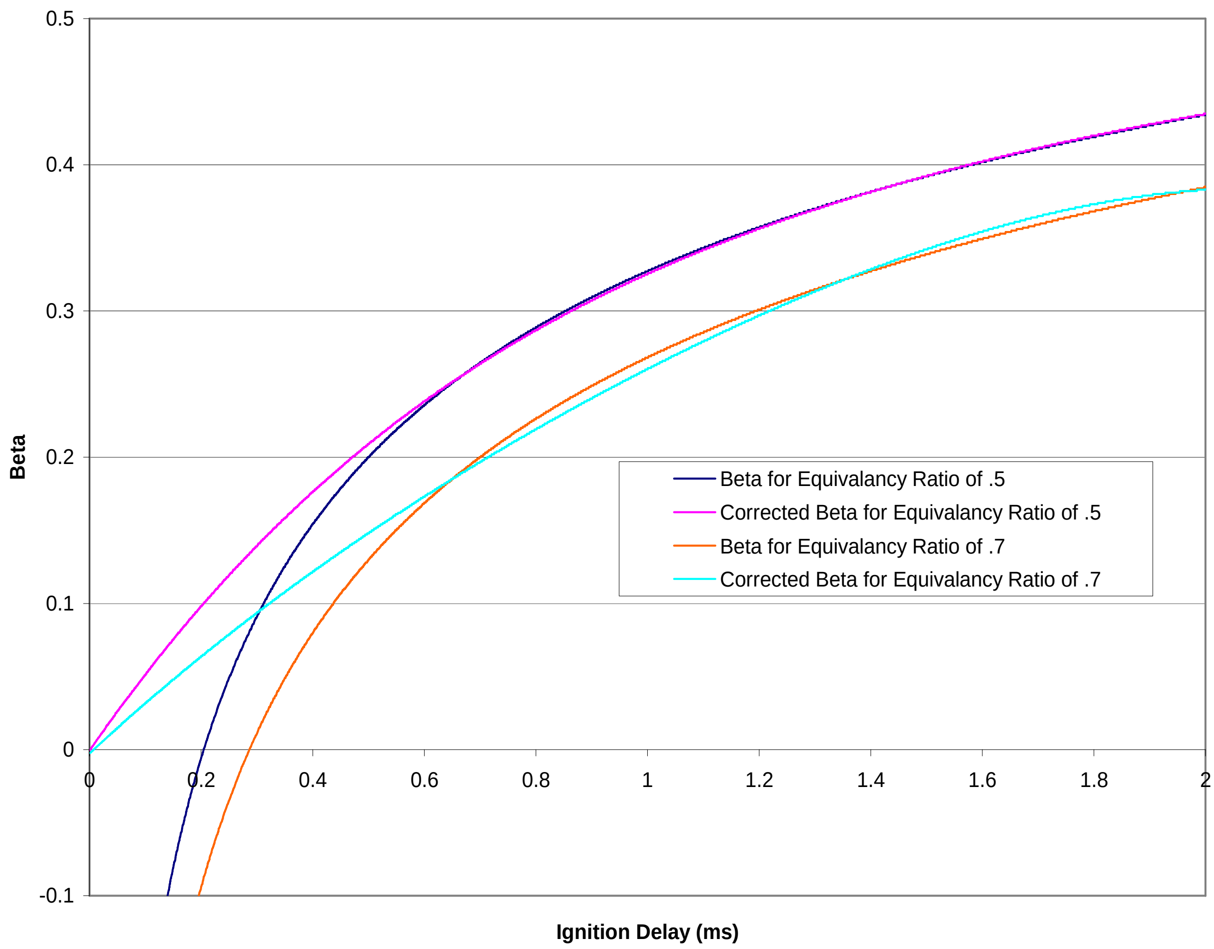


Figure 2.5: An example showing beta becoming negative for instances of very small ignition delay periods calculated from equation 2.32 as well as corrected values of beta. Constants a=.8 b=.25 and c=.25.

Now that β is known it can be substituted into equation 2.31 to solve for the ratio of the mass of fuel burned to the total mass of fuel injected as a function of crankshaft angle. Figure 2.6 shows the premixed burning function, the mixing controlled function and the function of mass fraction burned. Each is shown as a function of crankshaft angle. The heat released as a function of crankshaft angle can be solved by multiplying the previous equation by the lower heating value of diesel fuel; then by the mass of fuel injected per cylinder per cycle. The derivative of this equation is taken with respect to crankshaft angle to solve for the heat release function. For use in the simulation it is assumed that heat release takes place between the time injection starts and when equation 2.31 equals .99999. In Figure 2.7 the shape of the heat release function shows the premixed combustion and the mixing controlled combustion. The more fuel added to the model, the more heat is released. Even if perfect mixing and combustion could be achieved, a limit of how much energy could be added would be reached at a stoichiometric air fuel ratio. Fuel added beyond this ratio is wasted because it cannot burn. In an actual engine, because the fuel and air mixture does not mix nor burn perfectly, the limit of how much fuel can be added is reached before the equivalency ratio equals one. Because of this, the computer simulation should predict that more power can be produced at higher equivalency ratios that is possible.

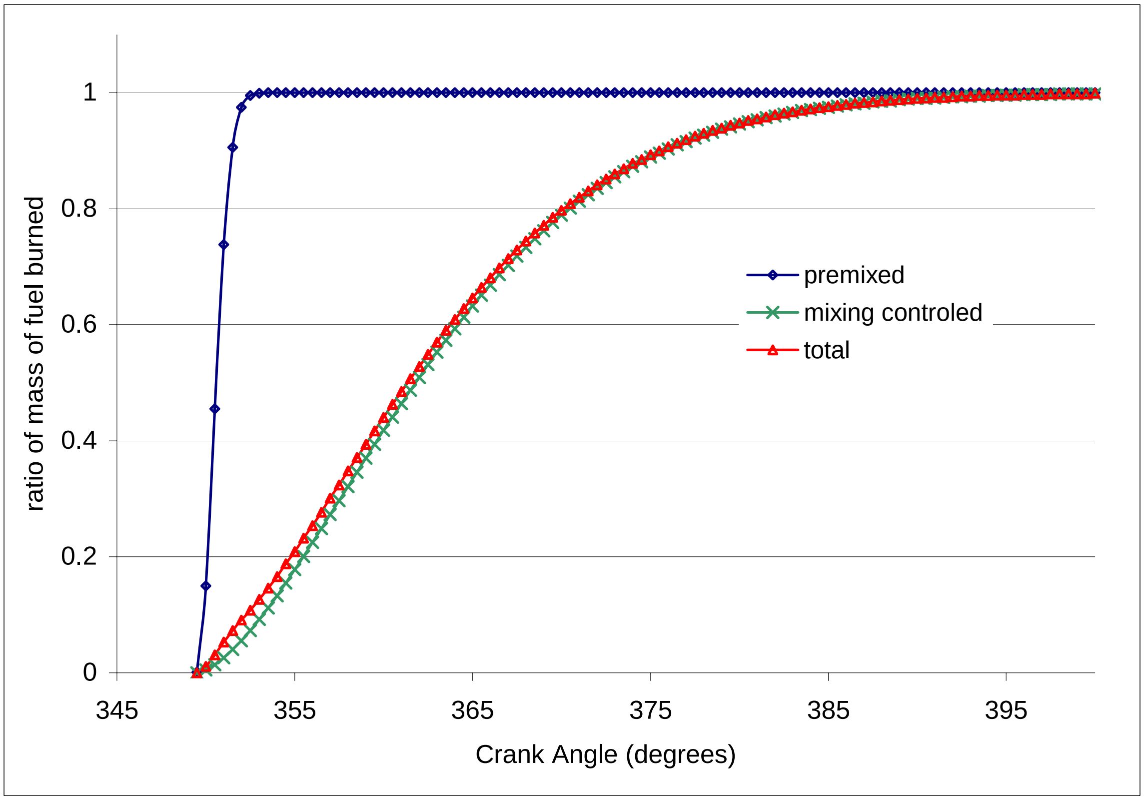


Figure 2.6: An example of the rates at which the fuel is burned for each type of combustion

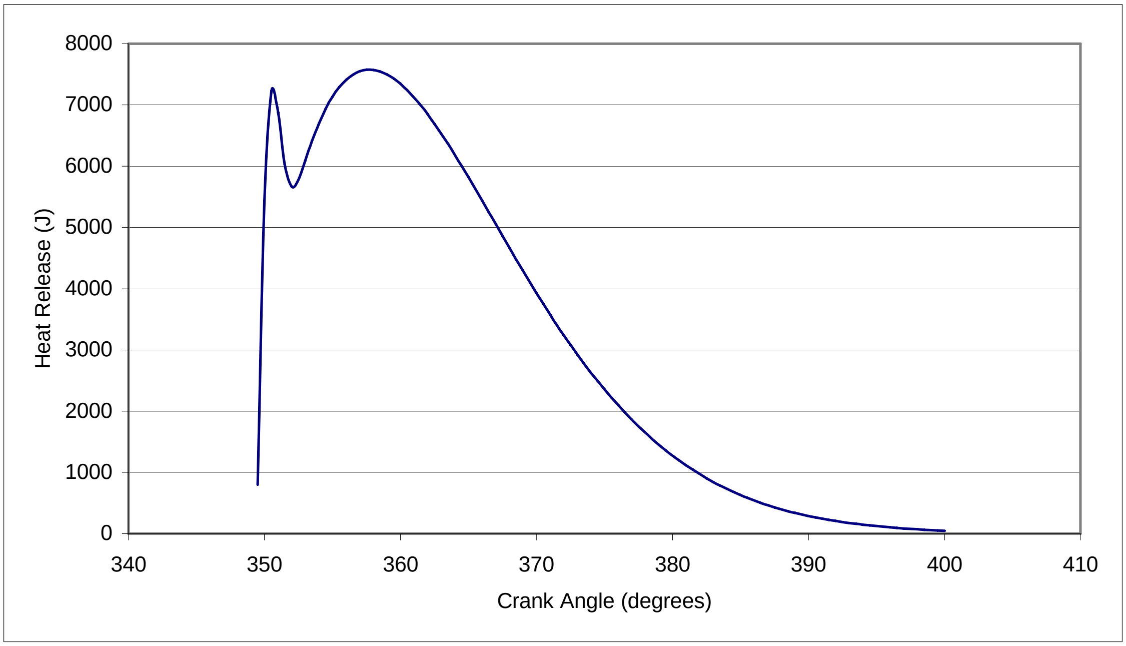


Figure 2.7: Typical shape of heat release function. The effects of the two types of combustion that take place can be seen

Since the heat addition is known, the cylinder pressure during combustion can be calculated . The pressure in the cylinder during the combustion process is modeled using the same equations that were used for the compression event combining, Equation 2.22 with Equation 2.24. The difference is that now they include the heat transfer term yielding

$$\frac{d}{d\theta}P = -\gamma \times \frac{P}{V} \times \frac{d}{d\theta}V + (\gamma - 1) \times \frac{1}{V} \times \frac{d}{d\theta}Q$$

Equation 2.33

Q is the heat added to the system. However, during the combustion process heat transfer must be accounted for because of the high heat transfer coefficient and high temperature differential between the wall and the combustion products. Therefore, the total heat transfer, shown in Equation 2.34, is the heat added by the combustion of the fuel minus the heat loss due heat transfer through the cylinder walls.

$$\frac{d}{d\theta}Q_{out} = h_c \cdot A \cdot (T - T_{wall})$$

Equation 2.34

where *A* and *T* are the area of the cylinder wall and the average temperature in the cylinder, respectively. To solve for the heat loss of the two unknowns, the heat transfer coefficient and the cylinder wall temperature must be found. Woschni's [7] correlation, which represents the spatially averaged heat flux, was used to solve for the heat transfer coefficient. This correlation (with the correct units) is expressed as:

$$h_c\left(\frac{W}{m^2 \times K}\right) = 3.26 \times B(m)^{-.2} \times P(kPa)^{.08} \times T(K)^{-.55} \times w\left(\frac{m}{s}\right)^{.8}$$

Equation 2.35

To solve for *w,* local average gas velocity required in equation 2.35, another of Woschni's equations, derived experimentally, was used. For the combustion and expansion period of a direct injection compression ignition engine with low swirl the equation is:

$$w = V_p \times 2.28 + 3.24 \times 10^{-3} \times \frac{V(IVC) \times T(IVC)}{P_m(IVC) \times V(IVC)} \times (P - P_m)$$

Equation 2.36

IVC in the previous equation indicates that the value for the variable in question is taken at the point at which the intake valve closes. In solving for these equations, Woschni assumed the correlation was of the form Nu=0.035 Re^m. However, as shown Annand [8], this correlation can be quite different from some of the data points used to arrive at the correlation. Therefore, the heat transfer term, which comes from this correlation, must be fitted to the actual engine being simulated. Wall temperature is the last unknown needed to solve for heat transfer and it is assumed that the wall temperature is equal to the coolant temperature. This is only an approximation as the cylinder wall temperature, in practice, will be somewhat higher than the coolant temperature, leading to a slightly lower heat transfer. The local average gas velocity is substituted into equation 2.35, which, along with the heat addition from the combustion of the fuel, is then substituted into equation 2.34 to arrive to Equation 2.37.

$$\frac{d}{d\theta} P = -\gamma \times \frac{P}{V} \times \frac{d}{d\theta} V + (\gamma - 1) \times \frac{1}{V} \times \frac{d}{d\theta} \Big[Q_{in} - 3.26^{-2} \times P^{.8} \times T^{-.55} \times$$

$$\left[V_p \times 2.28 + 3.24 \times 10^{-3} \times \frac{V(IVC) \times T(IVC)}{P_m(IVC) \times V(IVC)} \times (P - P_m) \right]^{.8} \times A \times (T - T_{wall}) \Big]$$

Equation 2.37

Equation 2.37 describes the pressure during combustion as a function of crank angle. This equation is solved using the ordinary differential equation solver in MathCAD. The results of the pressure during combustion can be combined with the pressure of the motored engine pressure to solve for the pressure in the cylinder for 0 degrees until the end of combustion. This statement sets the pressure equal to the result of Equation 2.25 for all crankshaft angles less than the point at which ignition begins, and equal to the results of equation 2.37 at all other points. The temperature as a function of crankshaft angle can now be solved using the ideal gas law, knowing the mass of air in the cylinder as a function of crank angle and the pressure in the cylinder as a function of the crankshaft angle.

2.3.7. Exhaust Event

The exhaust stroke is the last event of the engine cycle in need of modeling. During this process, the exhaust valve opens and allows the hot, high-pressure exhaust gases out of the cylinder. The gases leaving the cylinder can be broken down into two distinct events. One event, known as exhaust blowdown, occurs just as the valve opens and the high-pressure gases in the cylinder rapidly escape to the low pressure in the exhaust manifold. This, in turn, causes the temperature to drop. Gases continue to move into the manifold until the pressure in the cylinder equals the pressure in the exhaust manifold. At this point, the second part of the exhaust process takes place as the upward motion of the piston forces the remaining combustion products out of the cylinder, bringing the four stoke engine cycle to completion.

Stone, in his book *Introduction to Internal Combustion Engines,* states that the blowdown process can be assumed to take place instantaneously [9]. Because of this, the piston is assumed to be stationary, and at BDC, at which point the exhaust valve opens.

This is not the case though, because the exhaust gases leaving the cylinder take some finite amount of time during which the piston is moving. In addition, the exhaust valve does not open at bottom dead center but at some point before bottom dead center (40 degrees before bottom dead center, in the case of the test engine). In this simulation, it was found that a closer approximation to the test data was obtained by assuming the blowdown process occurred at the volume, pressure and temperature at which the exhaust valve opens while still assuming that the process takes place at bottom dead center.

If the point at which the exhaust valve actually opens is used and the piston is assumed to be stationary and the blowdown process takes place instantaneously, then as the piston continues to move down afterwards, the exhaust gases are drawn back into the cylinder. While this may occur in an engine at low speeds, it should not occur when an engine is operating in its designed operating speed range.

First, the known values were established. These include the temperature, pressure and mass of the exhaust gasses in the cylinder prior to the opening of the exhaust valve. The pressure in the exhaust manifold, which is the pressure required by the turbine, is also known. With these variables the state of the gases remaining in the cylinder can be calculated by solving for the temperature. Knowing that the volume remains constant and the pressure drops to the pressure required by the turbine, and that this process is isentropic allows the temperature remaining in the cylinder to be solved using

$$T_{remaining} = T_{beforeBD} \times \left(\frac{P_{turbineinlet}}{P_{beforeBD}} \right)^{\frac{\gamma - 1}{\gamma}}$$

Equation 2.38

After solving for the temperature, the mass can be solved using the ideal gas law such that

$$m_{remainingincy1} = \frac{P_{turbineinlet} \times V}{R_{air} \times T_{remaingincy1}}$$

Equation 2.39

To solve for the exhaust temperature Stone suggests an energy balance between the exhaust gas masses, as previously broken down, minus the mass that leaves the cylinder during the blowdown and the mass that is forced out of the cylinder by the upward motion of the piston [9]. The work done on the gas by the pistons upward motion is included by solving for enthalpy instead of the stored energy and piston work separately. The specific heat cancels out and yields

$$m_e \times T_e = T_{BD} \times m_{BD} + T_{remainincycl} \times m_{remainincyl}$$

Equation 2.40

To solve for the temperature of the gas that leaves during the blowdown process Stone uses the internal energy of the gas to arrive at

$$T_{BD} = \frac{\left(m_{beforeBD} \times T_{beforeBD} - m_{remainincyl} \times T_{remainincyl}\right)}{\gamma \times \left(m_{beforeBD} - m_{remainincyl}\right)}$$

Equation 2.41

Stone then goes on to solve the equation for the exhaust gas temperature; however, the equation he gives is incorrect [9]. The correct equation is arrived at by first substituting the right side of Equation 2.41 into Equation 2.40. Then, a substitution for m_{BD} can be made.

$$m_{BD} = m_{exhaust} - m_{remainingincyl}$$

Equation 2.42

After rearranging the equation, it was found that

$$T_{exhaust} = \frac{T_{beforeBD}}{\gamma} + \left(\frac{m_{remainincyl}}{m_{beforeBD}}\right) \times T_{remainincyl} \times \left(1 - \frac{1}{\gamma}\right)$$

Equation 2.43

Knowing the exhaust temperature for the engine, the temperature increase due to the propane addition, can be calculated. This is done by multiplying the lower heating value of propane by the mass flow rate of propane to solve for the heat released by the combustion of the propane. The mass flow rate of propane is solved by multiplying the given propane equivalency ratio by the amount of air remaining in the exhaust stream. To find the temperature increase of the exhaust gas, the energy added by the propane combustion is divided by the specific heat of the exhaust gas, and then by the mass flow rate of the exhaust. This is added to the temperature of the exhaust to find the temperature entering the turbine. If this temperature is less than the temperature required to drive the turbine the guess for the compressor exit temperature is lowered and the program is run again. If the temperature entering the turbine is higher than the temperature required to drive the turbine the guess for the compressor exit temperature is raised and the program is run again. If the temperature entering the turbine is equal to the temperature required to drive the turbine the program proceeds.

A cycle diagram showing pressure as a function of crank angle, Figure 2.8, can be plotted, as well as, a diagram of pressure as a function of cylinder volume, Figure 2.9. The area enclosed by the graph is the work done by the working fluid, and the work done by the working fluid per unit of swept volume can be calculated. This is known as indicated mean effective pressure (imep).

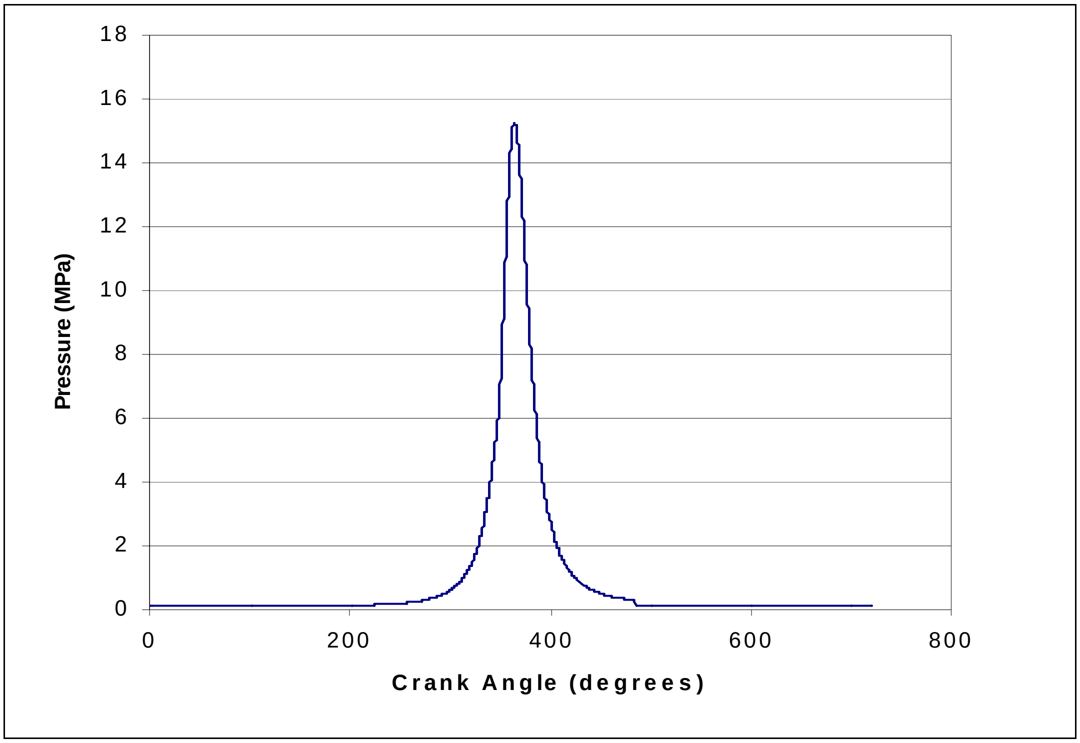


Figure 2.8: Cylinder Pressure as a Function of Crank Angle

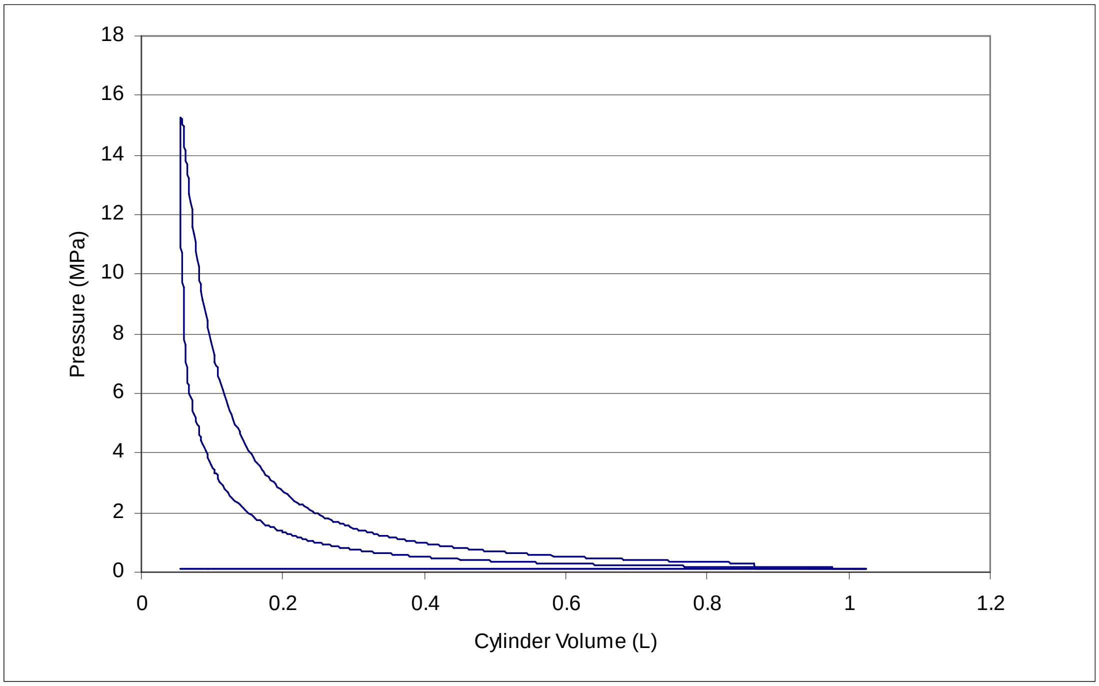


Figure 2.9: Cylinder pressure as a function of cylinder volume

Now that the pressure is known as a function of crankshaft angle and thus as a function of volume the work produced by the engine can be solved for. By definition, work done by a gas is pressure times the change in volume. Because the program the program solves the engine in stepwise manner, the work then becomes the sum of all the steps from 0 degrees to 720 degrees of the pressure times volume change times the step size as shown in Equation 2.44.

$$Work = \sum_{\theta}\left[\frac{d}{d\theta}V \times P \times d\theta\right]$$

Equation 2.44

The imep can then be calculated by dividing the work by the cylinder volume. The torque produced by the working fluid can then be calculated by multiplying the imep by the engine displacement and dividing by 4 times Pi. This torque, however is not the torque produced at the crankshaft. Some of the torque is absorbed by the friction. The manufacture states that the frictional losses are 15kW at 2500 rpm. Because engine friction losses are difficult to calculate directly, Stone [9] suggests four different correlations, and that most of the friction losses are caused by the sliding friction between the cylinder walls and the piston rings, it was assumed that frictional losses are linear with speed. The equation to solve for power lost to friction is in units of kilowatts is therefore:

$$P_{friction\_loss} = .006 \times \frac{kW}{RPM}$$

Equation 2.45

Torque lost can then be calculated by dividing the power lost by the engine speed in radians per second. This can then be subtracted from the torque produced by the working fluid to solve for the torque output of the engine. This equation yields:

$$Torque : \frac{(imep \times Dis)}{4\pi} - \frac{P_{friction\_loss}}{2\pi}$$

Equation 2.46

The torque was then multiplied by the engine speed in radians per second to find he power produced by the engine. The results of the simulation were then exported to a file for later review.

2.4. Program Results

2.4.1. Overview

As stated above, the program uses an assumed compressor outlet pressure to calculate both an engine manifold exhaust temperature as well as a required turbine inlet temperature sufficient to drive the compressor. When these two temperatures are equal, the cycle is at equilibrium and an operating point is recorded. However, results are recorded from each run of the simulation regardless of whether or not it is found to be an operating point of the engine. Recording these results makes it easier to identify trends and problems.

The starting value for the compressor outlet pressure was 100 kPa for each operation point listed above. The pressure was then incremented by 1 kPa up to 160kPa. It was found that this method worked better than iterating to find an operation point. The problem with iterating was due to fitting the compressor and turbine maps. If the operating point for either the compressor or the turbine is outside of the range of data given then the operating point of the engine cannot be determined even though the program fits the given data.

2.4.2. Simulation conditions

The following variables were held constant:

1. Atmospheric Temperature: 298 °K
2. Atmospheric Pressure: 1 atm
3. Intake restriction: 3 kPa
4. Coolant Temperature: 363 °K

The program was run with the following conditions:

1. Engine speeds: 1400, 1600, 1800, 2000, 2200,2400 and 2500rpm
2. Diesel Equivalency Ratios of: 50%, 55%, 60%, 65%, and 70%
3. Propane Equivalency Ratios of: 0%, 25%, 50%, and 75%

2.4.3. Expected Results

It has been said that before running a simulation the answer should be known. This is to say that one should know what to expect for data trends and the range the data is expected to be in. In this case this is true. The engine torque output as a function of engine speed was known although it was with an aftercooler installed. The aftercooler was removed for the purpose of simplifying the engine simulation, therefore it was expected that the torque curve produced by the simulation would be below the one given by the manufacturer.

As for the required turbine inlet temperature and the engine exhaust temperature as a function of compressor outlet pressure certain data and data trends were expected. First, it was expected that the maximum exhaust temperature without adding propane should be below 1200°K. The engine manifold exhaust temperature is expected to increase relatively slowly in relationship to the compressor outlet pressures. This increase is due to the increased compressor outlet temperature which increases with the pressure ratio across the turbine. The required turbine inlet temperature is expected to start below the manifold exhaust temperature for low compressor outlet pressures and then rise at a faster rate than the manifold exhaust temperature and cross at some operating point. This should be true for all data sets except those with low equivalence ratios where the exhaust temperature is simply not high enough to drive the compressor. The exact shape of this curve is unknown due the complexities of the system, namely the compressor map, but it should be smooth since nothing changes dramatically with compressor outlet pressure. It would be expected that as the operating points of the compressor and turbine move away from the design points that the turbine required inlet temperature would rise faster due to the decreasing efficiency but where this occurs is not known. The last thing that should be noted is the expected effect of the combustion chamber. It was expected that the addition of the propane would have little effect on the required turbine inlet temperature as a function of compressor outlet pressure because the only thing changing is the mass flow through the turbine and this changes only marginally. The addition of the propane would of course increase the engine manifold exhaust temperature and shift the curve up and thereby shift the intersection point of the required turbine inlet temperature curve an the engine manifold exhaust curve to the right. This of course is the goal, to increase the compressor outlet pressure and in turn make more power. As more propane is added the engine manifold exhaust temperature curve should continue to shift upwards. This trend will continue until the engine exhaust temperature becomes too hot to safely drive the turbine.

2.4.4. Simulation Results

The results of the computer simulation were used to first create graphs of engine manifold exhaust temperature and the required turbine inlet temperature as a function of compressor outlet pressure. These graphs shown in Appendix B, figures B1 through B20 were then used to determine the operating points of the engine.

Reviewing the engine manifold exhaust temperature shows that the pattern follows what was expected, exhaust temperature rises slowly with increased compressor outlet pressure. With the addition of propane the graphs of this temperature keep relatively the same shape, only translate up the Y-axis of the graph as expected. The temperature is however, somewhat lower than expected. This could have been influenced by correlation used to solve for the heat transfer to the cylinder wall.

What is of more interest is the required turbine inlet temperature. As can be seen from the figures, while there is a definite trend, there are also a large number of data points that were omitted because they were well outside of the trend. These tend to occur at engine speeds over 2000 rpm. Arriving at the cause of this erratic output is not a straightforward task due to the complexities involved in the simulation of the compressor and turbine. The first step was to check the operating points on both the turbine and compressor maps.

Several points that did not follow the trend were examined to find the cause. It was determined that there were two reasons the points did not follow the trend. The first reason discovered was for the data at low turbine pressure ratios and low corrected turbine mass flow. The efficiency curves on the turbine map in this location are almost vertical and were not well fitted by the software. To fix this, the equations could be changed to hold the turbine efficiency constant, but this would lead to other errors.

The source of the second problem was more difficult to locate but after some time it was discovered that for some reason, and only for certain data points, the simulation gave different results dependent on whether it was run for a single point or solving many points at once in a batch run. This software vendor was informed of this problem but never arrived at a cause nor a solution.

The engine operating points do not span all of the operating points simulated for a few reasons. Most of the reasons are stated above but there is also the case where the temperature leaving the combustion chamber is greater than the maximum turbine inlet temperature. This is the case for almost all the data for 75% propane equivalence ratios. This can be seen in Figures B16 through B20. If the graph continued for higher pressure ratios the two graphs would intersect but the data would be meaningless because the temperature would be well above the allowable turbine inlet temperature.

For data points where the two curves would not intersect (see Figure B1) the operating point was assumed to be at a pressure ratio of one. In actuality, the pressure ratio would be at some value less than one because a pressure loss would be incurred by the compressor.

Taking all of the above assumptions and problems into account, the data of interest, engine torque as a function of speed was plotted and is included in Appendix C Figures C1 through C5. In all cases, the engine torque is greater with the added propane than without. With a 25% propane equivalence ratio the engine torque was increased on average by 19.8%. With a 50% propane equivalence ratio the engine torque was increased on average by 23.5%. This 21.7% average gain does not reflect the large increase in torque output because of the limited number of data points, all of which are at low engine speeds where increases are smaller as shown by the data at a 25% propane equivalence ratio. This can be seen in all figures in

Appendix C. In some cases the output is substantially greater, such as the operating point at 2400 rpm with a 50% diesel equivalence ratio and a 25% propane equivalence ratio. This is shown in Figure C1. By adding the propane, the engine torque is increased by 46%.

3. Test Design and Set Up

3.1. Combustion Chamber Design

In designing a combustion chamber, the primary goal is to stabilize the flame front. To do this with laminar flow one would have to match the exhaust velocity through the combustion chamber with the flame speed for propane (approximately 3.3 m/s). The exhaust would have to be slowed substantially in the combustion chamber and then accelerated prior to the turbine. This would incur a pressure loss, which is very undesirable. In addition, such a combustion chamber would work for one velocity only. An increase in exhaust velocity would cause the flame to be blown out of the combustion chamber and would lead to flame out. In addition, if the exhaust velocity decreased, it would lead to erratic combustion, as the flame would propagate in the opposite direction of the exhaust flow. Because the engine will be tested over a wide engine speed and hence a large range of exhaust velocities such a combustion chamber is not practical.

The solution is to induce turbulence into the exhaust flow so that the flame front mixes with the exhaust stream and has more surface area with which to react. A device known as a flameholder is employed to achieve this. Many types of combustion chambers use this device in one form or another, from home oil furnaces to gas turbines. In its simplest form, a set of vanes induces toroidal flow reversal to stabilize the flame. To support fast mixing of the fuel and exhaust, the fuel is injected radially into the exhaust stream just downstream of the flameholder. The ignition source is located where the flame is expected to stabilize, just downstream from the flameholder.

The combustion chamber design was based on my previous undergraduate work building a gas turbine [10]. As the intention of this research was not to design a combustion chamber, but rather to determine if the addition of one would significantly improve engine power, little time was spent optimizing the previous design. Because the turbine had approximately the same mass flow rate as the one in my previous research the same design was reused. For more information on the combustion chamber design, see Lefebre [11].

3.2. Combustion Chamber Fabrication

A two-piece design was chosen to allow easy access to the flameholder, fuel nozzle, and ignition system. The lower section is a weldment fabricated from 316 series stainless steel. It consists of a two flanges joined by a divergent section. The flanges are 1/8 inch sheet and the conical section was formed from .030 inch sheet. The lower flange mates with the exhaust manifold, and the upper flange mates with the upper half of the combustion chamber. This lower section of the combustion chamber also contains the fuel injection nozzle, a thermocouple to measure exhaust temperature, and swirling vanes that are the flameholder. The flameholder is constructed using the same .030 inch 316 material. The fuel nozzle and thermocouple holder were both made from ¼ inch dia 304 series stainless steel tubing. The fuel nozzle contained four holes, which were angled to inject fuel radially into the exhaust stream ½ inch past the swirling vanes. This can be seen in figure 3.1 as well as in figure 3.2.

Figure 3.1: Lower combustion chamber after test

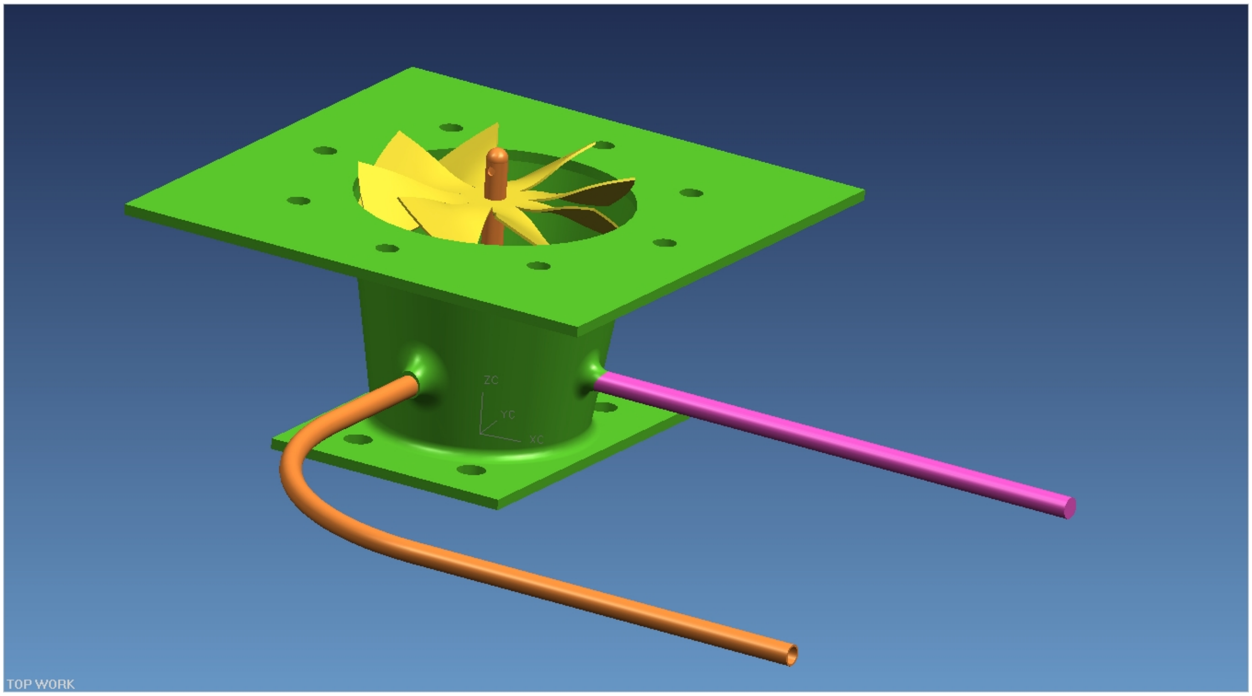


Figure 3.2: Lower combustion chamber as designed

The picture in figure 3.1 was taken after the testing and shows two of the blades missing from the flameholder and one present but broken. These blades were most likely damaged due to the high exhaust temperature combined with the vibrations induced by pulsations in the exhaust flow causing the metal to fatigue. Fortunately, no further damage was caused, as these blades were too large to fit into the turbine inlet.

The upper half of the combustor chamber shown if figure 3.3 was constructed using the same materials as the lower half.

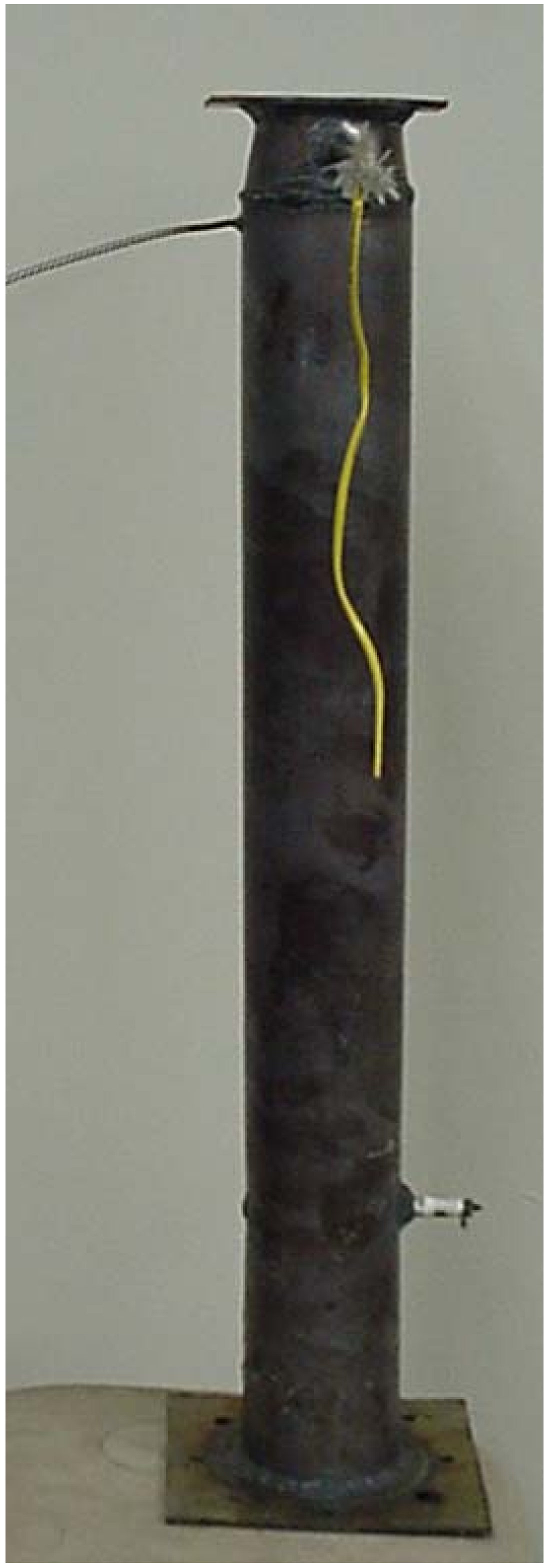


Figure 3.3: Upper combustion chamber as section

As with the lower half, the upper section was TIG welded and then pressure tested to ensure against air leaks. The upper section is where the combustion takes place. It was made longer than required to insure flames were kept out of the turbine. If there is insufficient mixing of the exhaust flow prior to the turbine, the flames can cause hot spots and damage to the turbine may occur.

A modified automotive spark plug was the igniter used because of its availability and its ability to withstand high temperatures. This was mounted three inches downstream of the swirling vanes, as can also be seen in figure 3.3. A furnace coil was used as the electrical source for the ignition system. The reason for choosing this as an ignition source was twofold. First it was easy to obtain and relatively inexpensive, but most importantly it could be operated continuously. Because the combustion chamber design was not optimal there was a good chance of flameout. If this occurred and there was no spark present to re-ignite the fuel there would be the possibility for fuel to build up in the exhaust system of both the engine and of the test cell. When the ignition system was turned back on all of the very volatile exhaust gasses could ignite causing a large and potentially damaging explosion.

3.3. Engine Setup

The test engine, donated by the Biological Systems Engineering department for this experiment, is a John Deere™ series 300 model 3179T. The engine is a three cylinder, 2.9-liter turbo charged and aftercooled diesel. The engine is rated for 79 peak horsepower at 2500 rpm and peak torque at 196 foot-pounds at 1500 rpm in stock form. Previously, the engine was a test engine and a hole that was drilled into the combustion chamber required plugging before the engine could be used. This was accomplished by turning a threaded plug on a lathe to the required size and shape. The only other modification to the engine was to remove the aftercooler from the engine. This was done to remove variables from the simulation and testing.

The engine was mounted in the test cell in the basement of Randolph Hall at Virginia Tech. It was affixed to the test stand using fabricated mounts, and adapters were fabricated to connect the flywheel with the input shaft of the water brake dynometer. The cell was equipped with a water-to-water heat exchanger for engine cooling which was used for this testing.

3.4. Instrumentation

After the engine and combustion chamber were mounted the controls and instrumentation were installed. Dyne Systems DTC-1 digital throttle controller was used so that the fueling of the engine at a given speed would be repeatable. The controller operated a servo, which positioned the throttle on the injection pump. Two rotometers were used to measure the fuel flow. One of the rotometers was fitted after the electric fuel pump, prior to the fuel filter, in order to measure flow into the fuel pump. The other was mounted in the return line to measure return flow. Figure 3.4 shows: 1. The servo that operated the throttle, 2. The supply rotometer, 3. The return rotometer, 4. The fuel injection pump and the 5. electric fuel pump. The fuel filter is on the far side of the engine and is not visible.

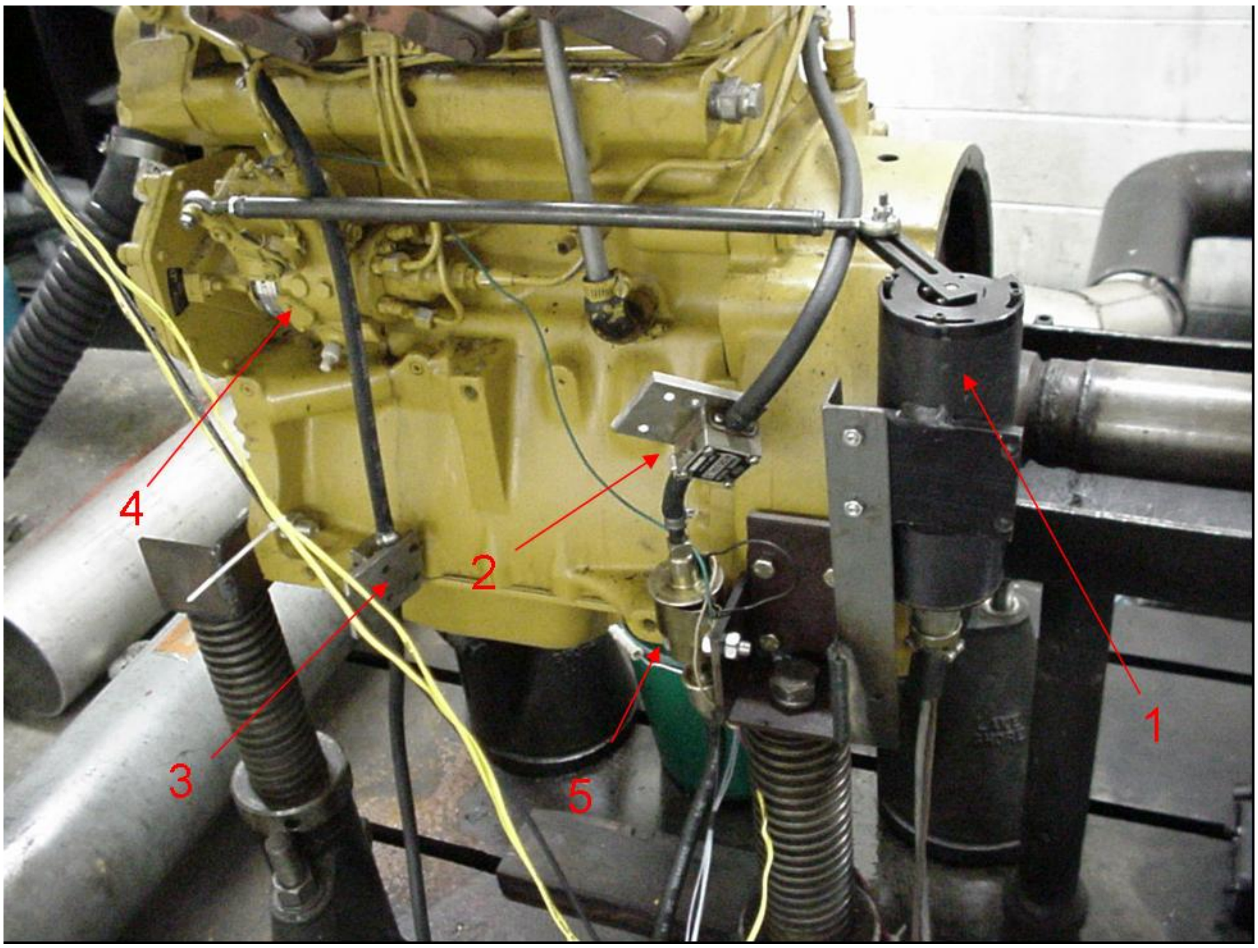


Figure 3.4: Engine setup showing various components of the fuel system.

To measure compressor outlet pressure and turbine inlet pressure, GM™ 2 bar MAP sensors, part number 08869124, were used because of their availability. They were calibrated prior to use. A five-volt computer power supply regulated the voltage needed by the MAP sensors. Tubes were installed just prior to the inlet manifold measured the total inlet pressure and were installed 2 inches upstream of the turbine inlet to measure total turbine inlet pressure. Type-K thermocouples with ceramic insulation and stainless steel holders were constructed for use after the compressor exit, prior to the combustion chamber, and after the combustion chamber prior to the turbine. Fluke digital multi-meters displayed the thermocouple voltages. A 20 pound propane tank provided the propane supply. From the tank, the fuel flowed into an adjustable pressure regulator and then into an electric actuated valve turned the propane flow on and off. A needle valve used to set the propane mass flow rate was located downstream from the electric valve. After some testing it was determined that the manually adjustable pressure should be replaced with a pressure regulator commonly known as a rising rate regulator. The purpose of this regulator is to reference the pressure differential across the regulator to the turbocharger outlet pressure. By doing this the pressure differential and thus the mass flow rate of propane stays constant as the inlet pressure rise. In the initial setup, the flow rate of fuel would decrease as turbocharger outlet pressure increased.

3.5. Calibration

Calibrating the dynamometer was the first step in the testing procedure. The dynamometer was calibrated to 180 foot-pounds to match the engines expected output. The engine was started and run for a half hour at varying speeds to check for water and oil leaks, and any other possible problems. The bolts in the drive shaft were then re-torqued for safety.

Next, the fuel delivery curves for the fuel pump needed to be established. The idea was to produce a function of fuel flow as a function of both throttle position and engine speed. The first step to create this function was to program the dynamometer to hold the engine at a certain speed. The throttle position would then be varied from the minimum required to hold that engine speed to full throttle in some increment. For each throttle setting at each engine speed fuel flow rate would be measured. This was to be done for engine speeds starting with 1800 rpm and ending at 2500 rpm with incremental steps of 100 rpm.

At this point I made two discoveries. First, the governor would not allow the engine to reach the maximum operating speed of 2500 rpm as stated by John Deere™. This was not a major problem because testing could continue at lower speeds. Second, the fuel pump did not deliver fuel at lower throttle settings consistently, causing a major problem. After examining the data, it was found that the equivalency ratios at which the pump would operate, and effectively control fuel flow, began at approximately 0.75 and continued to above 1. The problem with this is two-fold. First, the heat release function in the computer simulation does not work well with values above 0.75 because the empirical data used to create this function was taken at lower equivalency ratios. This is due to the function that determines how much of the heat addition is caused by the pre-mixed burning combustion and mixing control combustion. Second, the combustion chamber was designed to operate at lower equivalency ratios and there would not be enough oxygen left in the exhaust flows at higher equivalency ratios to allow combustion in the combustion chamber.

After consulting Al Clary, an ASE certified diesel mechanic, it was discovered that the fuel delivery problem was most likely due to a rusted governor spring and that it is a very common problem for diesel engines that have not been operational for long periods. This would also explain the reason the engine would not reach the governed maximum speed of 2500 rpm. Unfortunately, by the time the problem was finally diagnosed the test setup had been dismantled, and the engine returned to the BSE department.

Had the fuel pump been functional the plan was to first establish a baseline for the engine performance at various speeds and equivalency ratios and to compare this to the engines performance when operating with different flow rates of propane into the external combustion chamber. The results of interest would have been engine torque, turbine inlet temperature and pressure, and compressor outlet pressure and temperature

3.6. Results

The engine was tested at operating points where the equivalency ratio was above 0.75. This data proved to be of little use because the computer model could not be matched to it due to problems with the heat release equations. The combustion chamber was also tested at these high equivalency ratios and found not to work as expected due to the low oxygen content in the exhaust.

However, the combustion chamber was operational at lower equivalency ratios and a large increase in the temperature of the gases entering the turbine could be achieved. At an engine speed of 2000 rpm, with no load from the dynamometer, the exhaust temperature entering the turbine increased from 753°K to 1335°K, and the intake pressure increased from 99 kPa to 162 kPa. The dynamometer would not lock because the fuel pump was not regulating the fuel correctly and the engine could not be held at a constant speed. Because of these problems, no useful data was obtained from the test engine.

4. Conclusion and Recommendations

While the experiment did not yield any quantitative results and the computer simulations had problems as well, there are some deductions to be made from both the simulation and from observations made during the testing of the engine. While these observations do show promise, they are not the definitive results that were anticipated.

Reviewing the graphs of engine exhaust temperature and required turbine inlet temperature as a function of compressor outlet temperature indicates several things. Only a low equivalency ratio of propane is required to reach the maximum turbine inlet temperature. This is a positive outcome because achieving combustion in an exhaust flow with little oxygen left is easier at lower equivalency ratios. The ability to reach the maximum turbine inlet temperature with little propane addition is also promising because the maximum temperature is only the maximum for the turbine used. If the factory turbine was replaced with a turbine designed for high temperature use, more propane could be added to achieve higher temperatures.

Turbines for high temperature use have modifications such as water cooled bearings, and ceramic scrolls and rotors. While the water-cooling is common practice on larger turbines, the ceramic components are still in experimental use only and have proven to be somewhat fragile.

The observation made during testing was that the combustion chamber did function as expected. This case was at a low engine speed and a low diesel equivalence ratio and the dynamometer would not lock, however it proved that the added combustion chamber was operational. The turbine inlet temperature increased as expected and in turn, the compressor outlet pressure increased accordingly. This shows promise, but future work would need to be conducted to find the limits of the design and the possible performance gains that can be obtained.

If future work is conducted, there are several things to be considered. Any work, whether a simulation or engine testing, should be concentrated at lower propane equivalence ratios than were used for this research. High propane equivalence ratios yield no data because the temperatures are above the maximum turbine inlet temperatures.

If testing is to be done, a functioning engine should be obtained. Mass flow meters should be implemented to measure incoming airflow so that mass flow rate is not calculated from compressor maps, temperature, and pressure ratio data. Electronic flow meters should also be used to measure both fuel supply flow as well as fuel return. By doing two things the diesel equivalence ratio can be obtained directly. Adding an electrical flow meter to the propane flow will allow the propane equivalence ratio to be calculated directly. All of this data along with the data obtained from the instrumentation implemented in the previous test should be recorded using a data acquisition card and computer. This will allow much more data to be recorded and do so more accurately.

If the computer simulation is to be run there are also changes to be made. The program should be written in some lower level language that can be more directly controlled. A few examples are C, C++, Matlab, or FORTRAN. This would allow more automated approach as well as eliminate the problem associated with using MathCAD.

While the research did not yield all the data that was hoped for, it did yield encouraging results. Initial results showed average torque increases of approximately 20% and maximum increases of 43%. These results are very promising. By adding heat prior to the turbine significant gains in engine torque and power are possible as hypothesized.

5. References and further reading

5.1. References

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[7] Woschni G., "A Universally Applicable Equation For the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine," SAE paper 670931, SAE Trans., vol. 76, 1967.

[8] Annand, W. J. D. "Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines," Proc. Instn Mech. Engrs, vol. 177, no 36, pp. 973-990, 1963.

[9] Stone, Richard, Introduction to Internal Combustion Engines, 3rd edition, SAE, 1999.

[10] Brandon, S. J., Douglas, J.W., and Sexton, M.R., "Gas Turbine Engine: A Senior Design Project" Proceedings of the ASEE Annual Conference (St. Louis, MO June 2000)

[11] Lefebre, Arthur H., editor. Gas Turbine Combustor Design Problems. New York: Hemisphere Publishing Corp.; 1980.

5.2. Further Reading

Sexton, Michael R., "Turbomachinery Notes," Virginia Military Institute, 1999, on file with author.

Taylor, Charles Fayette, The Internal Combustion Engine in Theory and Practice, Volume 1: Thermodynamics, Fluid Flow, Performance, 2nd Revised Edition, M.I.T. Press, 1990.

Jones, J.B. and Dugan, R.E., Engineering Thermodynamics, Prentice Hall, Englewood Cliffs, New Jersey, 1996.

Appendix A: COMPUTER PROGRAM

Establishing Units not defined by MathCAD

$g := \frac{1}{1000} \cdot kg$ $\quad kJ := 1000 \cdot J$ $\quad kPa := 1000 \cdot Pa$ $\quad MPa := 10^6 \cdot Pa$ $\quad mBar := Pa \cdot 100$

Variables

$RPM := 2500 \cdot \frac{1}{min}$ $\quad T_1 := 298 \cdot K$ $\quad P_{atm} := 1 \cdot atm$ $\quad \phi_{diesel} := .5$ $\quad V_{loss} := .1$

$HX := .6$ $\quad \phi_{propane} := .25$ $\quad T_{coolant} := 363 \cdot K$ $\quad T_{wall} := T_{coolant}$ $\quad Step\_size := .125 \cdot deg$

$Intakererstriction := 3 \cdot kPa$ $\quad P_1 := P_{atm} - Intakererstriction$

$Backpressure := 7.5 \cdot kPa$

The Compressor outlet pressure is adjusted until the temperature after exhaust blowdown; and the required turbine inlet temperature are the same.

$P_2 := 100 \cdot kPa$ $\quad PR := \frac{P_2}{P_1}$ $\quad PR = 1.017$

Known Values

$n := 3$ $\quad B := 106 \cdot mm$ $\quad CR := 17.8$ $\quad S := 110 \cdot mm$ $\quad \tau_i := 13 \cdot deg$ $\quad r := 180 \cdot mm$

$AFR_{sdiesel} := 15.015$ $\quad EVO := 485 \cdot deg$ $\quad IVC := 180 \cdot deg$

$AFR_{spropane} := 15$ $\quad R_{air} := .287 \cdot \left(\frac{kJ}{kg \cdot K}\right)$ $\quad hv_{diesel} := 18250 \cdot \frac{BTU}{lb}$

$cp_{air} := 1.009 \cdot \frac{kJ}{kg \cdot K}$ $\quad hv_{propane} := 19916 \cdot \frac{BTU}{lb}$ $\quad \gamma_{air} := 1.4$

$CN := 40$ $\quad cp_{exhaust} := 1.131 \cdot \frac{kJ}{kg \cdot K}$ $\quad \gamma_{exhaust} := 1.3$

$\theta := 0, Step\_size .. 720 \cdot deg$

Data from compressor maps

:=

I:\..\Compspeed.xls

Separating the large matrix in two so that MathCAD can fit them

$$comp := \text{submatrix}(Compspeed, 0, \text{rows}(Compspeed) - 1, 1, 2)$$

$$N_c := \text{submatrix}(Compspeed, 0, \text{rows}(Compspeed) - 1, 0, 0)$$

:=

I:\..\Compeff.xls

$$comp_{eff} := \text{submatrix}(Comp\eta, 0, \text{rows}(Comp\eta) - 1, 1, 2)$$

$$eff_c := \text{submatrix}(Comp\eta, 0, \text{rows}(Comp\eta) - 1, 0, 0)$$

Data from turbine maps

:=

I:\..\Turbinemap.xls

$$turb := \text{submatrix}(Turbinemap, 0, \text{rows}(Turbinemap) - 1, 0, 1)$$

$$PR_{turb} := \text{submatrix}(Turbinemap, 0, \text{rows}(Turbinemap) - 1, 2, 2)$$

$$eff_t := \text{submatrix}(Turbinemap, 0, \text{rows}(Turbinemap) - 1, 3, 3)$$

Equations from Geometry

$$CylV := S \cdot \frac{B^2 \cdot \pi}{4} \qquad CylV = 0.971L$$

$Dis := n \cdot CylV \qquad Dis = 2.912L$

$CV := \frac{CylV}{CR} \qquad CV = 54.535cm^3$

$V_p := 2 \cdot S \cdot RPM \qquad V_p = 9.167\frac{m}{s}$

$$x(\theta) := \left(\frac{S}{2} + r\right) - \left[\frac{S}{2} \cdot \cos(\theta) + \sqrt{r^2 - \left(\frac{S}{2} \cdot \sin(\theta)\right)^2}\right]$$

$$dxd\theta(\theta) := \frac{d}{d\theta}x(\theta)$$

$$V(\theta) := CV + x(\theta) \cdot \frac{B^2 \cdot \pi}{4}$$

$$dVd\theta(\theta) := \frac{d}{d\theta}V(\theta)$$

$$A(\theta) := \left(\frac{B}{2}\right)^2 \cdot \pi + B \cdot \pi \cdot x(\theta)$$

Compressor Equations

Guesses

$eff_{comp} := .72 \qquad T_2 := T_1 \cdot \left[1 + \frac{\left[PR^{\frac{(\gamma_{air}-1)}{\gamma_{air}}} - 1\right]}{eff_{comp}}\right] \qquad T_2 = 300.002K$

$\rho_2 := \frac{P_2}{R_{air} \cdot T_2} \qquad \rho_2 = 1.161\frac{kg}{m^3} \qquad mdot_{air} := \frac{RPM}{2} \cdot Dis \cdot \rho_2 \qquad mdot_{air} = 0.07\frac{kg}{s}$

$W_{comp} := \frac{mdot_{air} \cdot \sqrt{\frac{T_1}{545 \cdot R}}}{\frac{P_1}{28.4 \cdot in\_Hg}} \qquad W_{comp} = 0.068\frac{kg}{s}$

$N_{cregress} := regress(comp, N_c, 4)$

$$N_{comp} := \text{interp}\left[N_{cregress}, comp, N_c, \begin{pmatrix} \dfrac{W_{comp}}{\frac{lb}{min}} \\ PR \end{pmatrix}\right] \qquad N_{comp} = 56347$$

$$eff_{cregress} := \text{regress}\left(comp_{eff}, eff_c, 4\right)$$

$$eff_{comp} := \text{interp}\left[eff_{cregress}, comp_{eff}, eff_c, \begin{pmatrix} \dfrac{W_{comp}}{\frac{lb}{min}} \\ PR \end{pmatrix}\right] \qquad eff_{comp} = 0.444$$

$$W_{comp} := \frac{W_{comp}}{\frac{lb}{min}} \qquad W_{comp} = 9.019$$

Given

$$W_{comp} = \frac{\dfrac{RPM \cdot min}{2} \cdot \dfrac{Dis}{ft^3} \cdot \dfrac{P_2 \cdot \left(1 - V_{loss}\right) \cdot s^2 \cdot \frac{ft}{lb}}{R_{air} \cdot \dfrac{s^2}{ft^2} \cdot R \cdot \left[\dfrac{T_1 \cdot \left[1 + \dfrac{\left[PR^{\frac{\left(\gamma_{air}-1\right)}{\gamma_{air}}} - 1\right]}{eff_{comp}}\right]}{R}\right]} \cdot \sqrt{\dfrac{\frac{T_1}{R}}{545}}}{\dfrac{\frac{P_1}{in\_Hg}}{28.4}}$$

$$N_{comp} = \text{interp}\left[N_{cregress}, comp, N_c, \begin{pmatrix} W_{comp} \\ PR \end{pmatrix}\right]$$

$$eff_{comp} = \text{interp}\left[eff_{cregress}, comp_{eff}, eff_c, \begin{pmatrix} W_{comp} \\ PR \end{pmatrix}\right]$$

Find

$$\begin{pmatrix} W_{comp} \\ N_{comp} \\ eff_{comp} \end{pmatrix} := \text{Find}\left(W_{comp}, N_{comp}, eff_{comp}\right)$$

$$W_{comp} := W_{comp} \cdot \frac{lb}{min} \qquad W_{comp} = 0.0612\frac{kg}{s}$$

$eff_{comp} = 0.47164$

$N_{comp} = 52229$

$$mdot_{air} := \frac{W_{comp}}{\dfrac{\sqrt{\dfrac{T_1}{545 \cdot R}}}{\dfrac{P_1}{28.4 \cdot in\_Hg}}} \qquad mdot_{air} = 0.063\frac{kg}{s}$$

$$\rho_2 := \frac{mdot_{air}}{\dfrac{RPM}{2} \cdot Dis} \qquad \rho_2 = 1.042\frac{kg}{m^3}$$

$$T_2 := T_1 \cdot \left[1 + \frac{\left[PR^{\frac{(\gamma_{air}-1)}{\gamma_{air}}} - 1\right]}{eff_{comp}}\right] \qquad T_2 = 301.057K$$

$$\rho_2 := \frac{P_2 \cdot (1 - V_{loss})}{R_{air} \cdot T_2}$$

Intake Event

$P_3 := P_2 \cdot (1 - V_{loss})$

$\rho_3 := \rho_2$

Mass of air in cylinder for the intake stroke

$m_{intakeair}(\theta) := V(\theta) \cdot \rho_3$

Mass of air in cylinder for Theta =0 to 540 degrees

$mair_{cyl}(\theta) := if(\theta < 180deg, m_{intakeair}(\theta), if(\theta \le EVO, m_{intakeair}(180 \cdot deg), 0))$

Heat transfer

Heat flux is essentially 0 for cycle except for the combustion period (Heywood 668)

Solving for Motored Engine Pressure

Given

$$\frac{d}{d\theta}p_m(\theta) + \gamma_{air} \cdot p_m(\theta) \cdot \frac{dVd\theta(\theta)}{V(\theta)} - (\gamma_{air} - 1) \cdot dxd\theta(\theta) = 0$$

$$p_m(\pi) = P_3$$

$$p_m := \text{Odesolve}(\theta, 3\pi) \cdot \text{Pa}$$

$$P_m(\theta) := \text{if}(\theta \le \text{IVC}, P_2, \text{if}(\theta > \text{EVO}, p_m(540 \cdot \text{deg}), p_m(\theta)))$$

Solving for fuel addition

$$mdot_{diesel} := \frac{mdot_{air}}{1} \cdot \frac{\phi_{diesel}}{AFR_{sdiesel}}$$

$$mdot_{diesel} = 2.104\frac{g}{s}$$

mass of fuel injected per cylinder per injection

$$mfuel_{cyl} := \frac{mdot_{diesel}}{RPM \cdot \frac{3}{2}}$$

$$mfuel_{cyl} = 3.367\times 10^{-5}\,kg$$

$$mdot_{airdieselcombustion} := mdot_{diesel} \cdot AFR_{sdiesel}$$

$$mdot_{airdieselcombustion} = 0.032\frac{kg}{s}$$

$$mdot_{airremaining} := mdot_{air} - mdot_{airdieselcombustion}$$

$$mdot_{airremaining} = 0.032\frac{kg}{s}$$

$$mdot_{propane} := \frac{mdot_{airremaining}}{AFR_{spropane}} \cdot \phi_{propane}$$

$$mdot_{propane} = 5.2662962 \times 10^{-4}\,\frac{kg}{s}$$

$$Q_{propane} := mdot_{propane} \cdot hv_{propane}$$

$$Q_{propane} = 2.44 \times 10^{4}\,\frac{J}{s}$$

Solving Turbine Equations

Known

$$N_{turbo} := \sqrt{\frac{T_1}{545 \cdot R}} \cdot N_{comp}$$

$$N_{turbo} = 51816$$

$$Work_{compressor} := mdot_{air} \cdot cp_{air} \cdot \left(T_2 - T_1\right)$$

$$Work_{compressor} = 0.261hp$$

$$P_5 := P_{atm} + Backpressure$$

Guesses

$$T_4 := 1000 \cdot K$$

$$P_4 := 1.3 \cdot atm$$

$$PR_{turbine} := \frac{P_4}{P_5}$$

$$eff_{turb} := .7$$

$$T_5 := T_4 \cdot \left[1 - eff_{turb} \cdot \left[1 - \left(\frac{1}{PR_{turbine}}\right)^{\frac{\gamma_{exhaust}-1}{\gamma_{exhaust}}}\right]\right]$$

$$W_{turb} := \frac{mdot_{air} \cdot \sqrt{\frac{T_4}{519 \cdot R}}}{\frac{P_4}{29.92 \cdot in\_Hg}}$$

$$N_{turb} := \frac{N_{turbo}}{\sqrt{\dfrac{T_4}{519 \cdot R}}}$$

$$PR_{tregress} := \mathrm{regress}\left(turb, PR_{turb}, 5\right)$$

$$PR_{turbine} := \mathrm{interp}\left[PR_{tregress}, turb, PR_{turb}, \begin{pmatrix} N_{turb} \\ \dfrac{W_{turb}}{\frac{lb}{min}} \end{pmatrix}\right]$$

$$eff_{tregress} := \mathrm{regress}\left(turb, eff_t, 7\right)$$

$$eff_{turb} := \mathrm{interp}\left[eff_{tregress}, turb, eff_t, \begin{pmatrix} N_{turb} \\ \dfrac{W_{turb}}{\frac{lb}{min}} \end{pmatrix}\right]$$

$$mdot_{exhaust} := mdot_{air} + mdot_{diesel}$$

Making variables unit less for use in solver

$$T_5 := \frac{T_5}{R}$$

$$T_4 := \frac{T_4}{R}$$

$$T_1 := \frac{T_1}{R}$$

$$T_2 := \frac{T_2}{R}$$

$$P_4 := \frac{P_4}{in\_Hg}$$

$$W_{turb} := \frac{W_{turb}}{\frac{lb}{min}}$$

Given

$$PR_{turbine} = interp\left[PR_{tregress}, turb, PR_{turb}, \begin{pmatrix} N_{turb} \\ W_{turb} \end{pmatrix}\right]$$

$$eff_{turb} = interp\left[eff_{tregress}, turb, eff_{t}, \begin{pmatrix} N_{turb} \\ W_{turb} \end{pmatrix}\right]$$

$$T_5 = T_4 \cdot \left[1 - eff_{turb} \cdot \left[1 - \left(\frac{1}{PR_{turbine}}\right)^{\frac{\gamma_{exhaust}-1}{\gamma_{exhaust}}}\right]\right]$$

$$mdot_{air} \cdot cp_{air} \cdot \left(T_2 - T_1\right) = mdot_{exhaust} \cdot cp_{exhaust} \cdot \left(T_4 - T_5\right)$$

$$PR_{turbine} = \frac{P_4}{\frac{P_5}{in\_Hg}}$$

$$N_{turb} = \frac{N_{turbo}}{\sqrt{\frac{T_4}{519}}}$$

$$W_{turb} = \frac{\frac{mdot_{exhaust}}{\frac{lb}{min}} \cdot \sqrt{\frac{T_4}{519}}}{\left(\frac{P_4}{29.92}\right)}$$

$$\begin{pmatrix} PR_{turbine} \\ eff_{turb} \\ P_4 \\ N_{turb} \\ W_{turb} \\ T_5 \\ T_4 \end{pmatrix} := Find\left(PR_{turbine}, eff_{turb}, P_4, N_{turb}, W_{turb}, T_5, T_4\right)$$

Returning units to Variables after use in solver

$T_1 := T_1 \cdot R$

$T_{turbineinlet} := T_4 \cdot R$

$P_{turbineinlet} := P_4 \cdot in\_Hg$

$T_2 := T_2 \cdot R$

$T_5 := T_5 \cdot R$

$W_{turb} := W_{turb} \cdot \frac{lb}{min}$

$T_{turbineinlet} = 1.47 \times 10^3 K$

Solving Engine Variables

Equations for heat release

$R_o := 8.31441 \cdot \left(\frac{J}{mol \cdot K}\right)$

$\Delta t_{comb} := 90 \cdot deg$

Maximum length of combustion process (degrees crank angle)

Ignition Delay
(Stone 426)

$E_a := \frac{618840}{(CN + 25)}$

$E_a = 9.521 \times 10^3$

Solving for Conditions at TDC

$P_{tdc} := P_m(360 \cdot deg)$

$P_{tdc} = 5.471 \times 10^6 Pa$

$$T_{tdc} := T_2 \cdot \left(\frac{P_{tdc}}{P_2}\right)^{\frac{(\gamma_{air}-1)}{\gamma_{air}}}$$

$$T_{tdc} = 944.578K$$

$$t_{id} := \left[.36 + .22 \cdot V_p \cdot \left(\frac{s}{m}\right)\right] \cdot \exp\left[E_a \cdot \left[\frac{1}{R_o \cdot T_{tdc} \cdot \left(\frac{mol}{J}\right)} - \frac{1}{17190}\right] \cdot \left[\frac{21.2}{P_{tdc} \cdot \left(\frac{1}{Pa \cdot 100000}\right) - 12.4}\right]^{.63}\right] \cdot deg$$

$$t_{id} = 3.639deg$$

Ignition delay (degrees crank angle)

Relating Crankshaft Degrees to Time

$$spd := \frac{1}{360 \cdot RPM}$$

$$spd = 6.6666667 \times 10^{-5} s$$

Seconds per degree of crankshaft revolution

$$\tau_{id} := \frac{t_{id}}{deg} \cdot spd$$

$$\tau_{id} = 2.426 \times 10^{-4} s$$

Ignition delay (time)

Heat release (premixed and mixing controled)
(Heywood 779)

$$t_{ign} := 360 \cdot deg - \tau_i + t_{id}$$

$$t_{ign} = 350.639deg$$

Point at which combustion begins (crankshaft degrees)

$$t_{end} := t_{ign} + \Delta t_{comb}$$

$$t_{prime}(\theta) := \frac{(\theta - t_{ign})}{\Delta t_{comb}}$$

Non-dimensional time for combustion

Constants

$a := .8$

$.8 < a < .95$

$a = 0.8$

$b := .25$

$.25 < b < .45$

$b = 0.25$

$c := .25$

$.25 < c < .5$

$c = 0.25$

$$K_1 := 2 + 1.25 \cdot 10^{-8} \cdot \left[\tau_{id} \cdot \left(\frac{1000}{s}\right) \cdot RPM \cdot (min)\right]^{2.4}$$

$K_1 = 2.0596648$

$K_2 := 5000$

$K_2 = 5 \times 10^3$

$$K_3 := \frac{14.2}{\phi_{diesel}{}^{.644}}$$

$K_3 = 22.19$

$K_4 := .79 \cdot K_3{}^{.25}$

$K_4 = 1.715$

Creating the equivalence ratio matrix to be used to create an equation for Beta

$\text{equiv}_{.3} := 0, 4 .. 135$

$\text{equiv}_{\text{equiv}_{.3}} := .3$

$\text{equiv}_{.4} := 1, 5 .. 135$

$\text{equiv}_{\text{equiv}_{.4}} := .4$

$\text{equiv}_{.5} := 2, 6 .. 135$

$\text{equiv}_{\text{equiv}_{.5}} := .5$

$\text{equiv}_{.6} := 3, 7 .. 135$

$\text{equiv}_{\text{equiv}_{.6}} := .6$

Creating the ignition delay matrix to be used to create an equation for Beta

$\text{tid}_i := 2 .. 117$

$$\text{Tid}_{\text{tid}_i + 18} := \text{round}\left(\frac{\text{tid}_i}{4}, 0\right) \cdot .06 + .44$$

Solving for Beta using Watsons equation to be used to create an equation for Beta which is always positive as long as Tid>0

$i := 20 .. 135$

$$\text{Beta}_i := 1 - \frac{a \cdot \left(\text{equiv}_i\right)^b}{\left(\text{Tid}_i\right)^c}$$

$\text{Betadata} := \text{augment}(\text{equiv}, \text{Tid}, \text{Beta})$

$\phi Tid_{tofit} := \text{submatrix}(Betadata, 0, \text{rows}(Betadata) - 1, 0, 1)$

$Beta_{tofit} := \text{submatrix}(Betadata, 0, \text{rows}(Betadata) - 1, 2, 2)$

$Beta_{regress} := \text{regress}\left(\phi Tid_{tofit}, Beta_{tofit}, 4\right)$

$$\beta := \text{interp}\left[Beta_{regress}, \phi Tid_{tofit}, Beta_{tofit}, \begin{pmatrix} \phi_{diesel} \\ \dfrac{\tau_{id} \cdot 1000}{s} \end{pmatrix}\right]$$

$f_1(\theta) := 1 - \left(1 - t_{prime}(\theta)^{K_1}\right)^{K_2}$

$f_2(\theta) := 1 - \exp\left(-K_3 \cdot t_{prime}(\theta)^{K_4}\right)$

$FR(\theta) := \beta \cdot f_1(\theta) + (1 - \beta) \cdot f_2(\theta)$

this is the ratio of fuel burned to unburned fuel at time tprime

$HR(\theta) := FR(\theta) \cdot hv_{diesel} \cdot mfuel_{cyl}$

$f(\theta) := \frac{d}{d\theta} HR(\theta)$

$Q_c(\theta) := \text{if}\left[\left(\theta > t_{ign}\right) \cdot \left(f_2(\theta) < .9999\right), f(\theta), 0\right]$

Equation for Pressure in cyl for compression event

$P_{comp}(\theta) := P_m(\theta)$

Equation for Pressure in cyl for combustion event

Given

$$\frac{d}{d\theta}P_{comb}(\theta) + \gamma_{exhaust} \cdot P_{comb}(\theta) \cdot \frac{dVd\theta(\theta)}{V(\theta)} - \frac{(\gamma_{exhaust} - 1)}{V(\theta)} \cdot \left[Q_c(\theta) - \left[\frac{3.26 \cdot HX \cdot \left(\frac{B}{m}\right)^{-.2} \cdot \left(\frac{P_{comb}(\theta)}{kPa}\right)^{.8} \cdot \left(\frac{\frac{P_{comb}(\theta) \cdot V(\theta)}{mair_{cyl}(IVC) \cdot R_{air}}}{K}\right)^{-.55} \cdot \left[\frac{V_p \cdot 2.28 + 3.24 \cdot 10^{-3} \cdot \frac{V(IVC) \cdot T_2}{P_m(IVC) \cdot V(IVC)} \cdot (P_{comb}(\theta) - P_m(\theta)) \cdot \left(\frac{\frac{m}{s}}{K}\right)}{\frac{m}{s}}\right]^{.8}}{RPM} \cdot \frac{W}{m^2 \cdot K} \cdot A(\theta) \cdot \left(\frac{P_{comb}(\theta) \cdot V(\theta)}{mair_{cyl}(IVC) \cdot R_{air}} - T_{wall}\right) \right] \right] = 0$$

$$P_{comb}(t_{ign}) = P_m(t_{ign})$$

$$P_{comb} := \text{Odesolve}(\theta, 3\pi) \cdot Pa$$

$$P(\theta) := \text{if}(\theta < t_{ign}, P_m(\theta), \text{if}(\theta \le EVO, P_{comb}(\theta), P_{turbineinlet}))$$

Solving for Temperature as a function of crank angle from 0-540 degrees

$$T(\theta) := \text{if}\left(\theta < IVC, T_2, \frac{P(\theta) \cdot V(\theta)}{mair_{cyl}(IVC) \cdot R_{air}}\right)$$

Solving for temperature after exhaust blowdown

$$T_{remaingincyl} := T(EVO) \cdot \left(\frac{P_{turbineinlet}}{P(EVO)}\right)^{\frac{\gamma_{exhaust}-1}{\gamma_{exhaust}}}$$

$$T_{remaingincyl} = 626.653\,K$$

$$mair_{remainingincyl} := \frac{P_{turbineinlet} \cdot V(EVO)}{R_{air} \cdot T_{remaingincyl}}$$

$$mair_{remainingincyl} = 4.495 \times 10^{-4}\,kg$$

$$T_{afterBD} := \frac{mair_{cyl}(EVO) \cdot T(EVO) + mair_{remainingincyl} \cdot T_{remaingincyl} \cdot \left(\gamma_{exhaust} - 1\right)}{\gamma_{exhaust} \cdot mair_{cyl}(EVO)}$$

Solving for Temperature as a function of crankshaft angle 0-720 degrees

$$T(\theta) := \mathrm{if}\left(\theta \le IVC, T_2, \mathrm{if}\left(\theta < EVO, \frac{P(\theta) \cdot V(\theta)}{mair_{cyl}(IVC) \cdot R_{air}}, T_{afterBD}\right)\right)$$

$$T_{exhaust} := \frac{Q_{propane}}{\left(mdot_{air} \cdot cp_{exhaust}\right)} + T_{afterBD}$$

$$T_{exhaust} = 1.027 \times 10^{3}\,K$$

Solving for Torque and Power

$$Work := \sum_{\theta} \left[\left(\frac{d}{d\theta}V(\theta)\right) \cdot P(\theta) \cdot Step\_size\right]$$

$$Work = 654.135\,N \cdot m$$

$$imep := \frac{Work}{CylV}$$

$$Torque_{imep} := \frac{imep \cdot Dis}{4 \cdot \pi}$$

Equation to estimate motored engine losses (given that the friction losses are 15 kW at 2500 rpm)

$$Torque := Torque_{imep} - \frac{RPM \cdot kW \cdot s^2}{231.2\pi}$$

$$Power := Torque \cdot RPM \cdot 2\pi$$

$$output := \left(\frac{RPM}{\frac{1}{min}} \quad \frac{Power}{kW} \quad \frac{Torque}{N \cdot m} \quad \phi_{diesel} \quad \phi_{propane} \quad \frac{P_{atm}}{kPa} \quad \frac{P_1}{kPa} \quad \frac{P_2}{kPa} \quad \frac{P_3}{kPa} \quad \frac{P_{turbineinlet}}{kPa} \quad \frac{T_1}{K} \quad \frac{T_2}{K} \right.$$

$$\frac{P_{atm}}{kPa} \quad \frac{P_1}{kPa} \quad \frac{P_2}{kPa} \quad \frac{P_3}{kPa} \quad \frac{P_{turbineinlet}}{kPa} \quad \frac{T_1}{K} \quad \frac{T_2}{K} \quad \frac{T(EVO \cdot deg)}{K} \quad \frac{T(EVO - .1 \cdot deg)}{K} \quad \frac{T_{exhaust}}{K} \quad \frac{T_{turbineinlet}}{K} \quad \beta$$

Appends results to file

APPENDPRN("I:\work\thesis\mathcad\results.prn") := output

Appendix B: SIMULATION RESULTS (Engine Exhaust Temperature and Turbine Inlet Temperature as a function of Compressor Outlet Pressure)

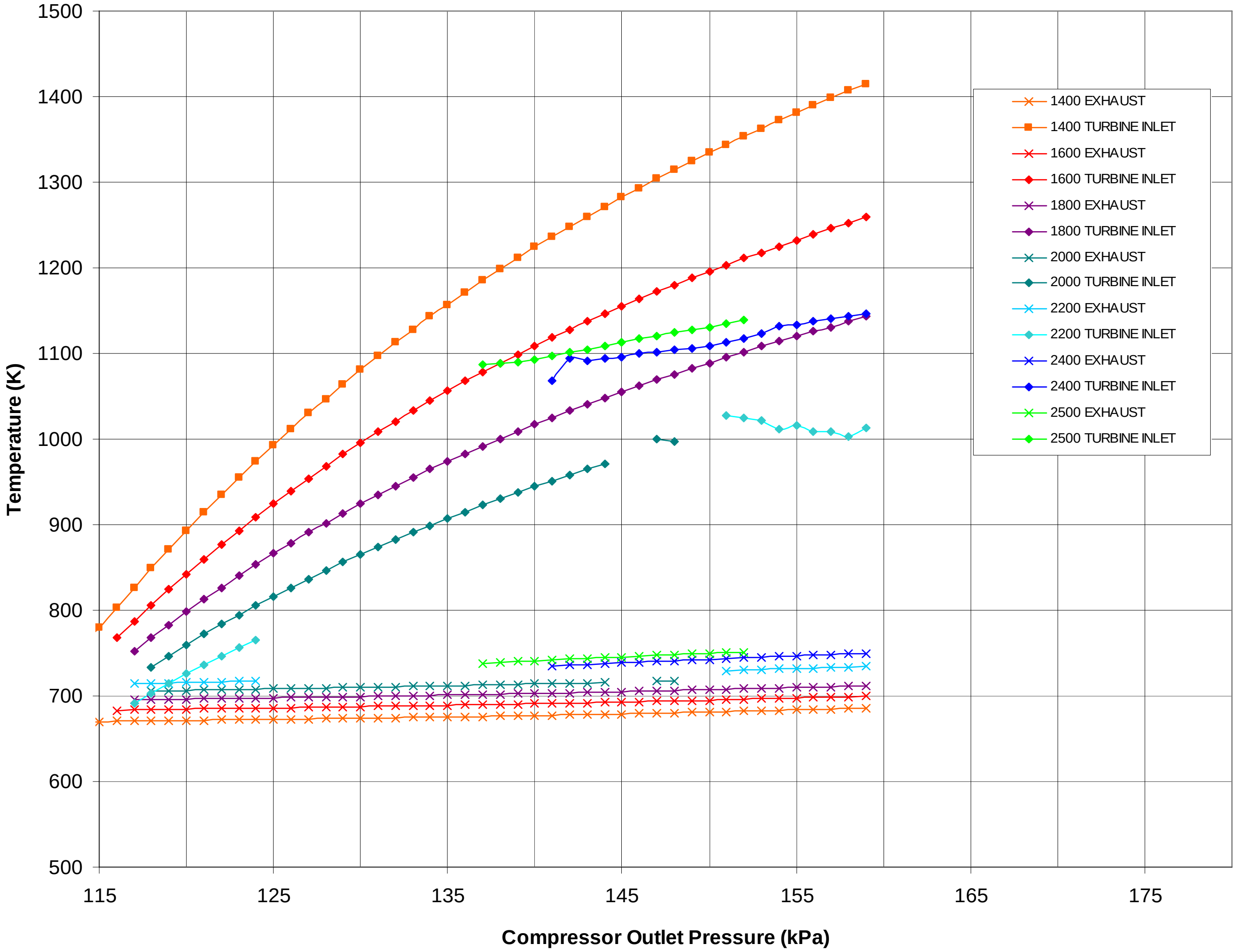

Figure B1: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 50 percent diesel equivalency ratio and no added propane.

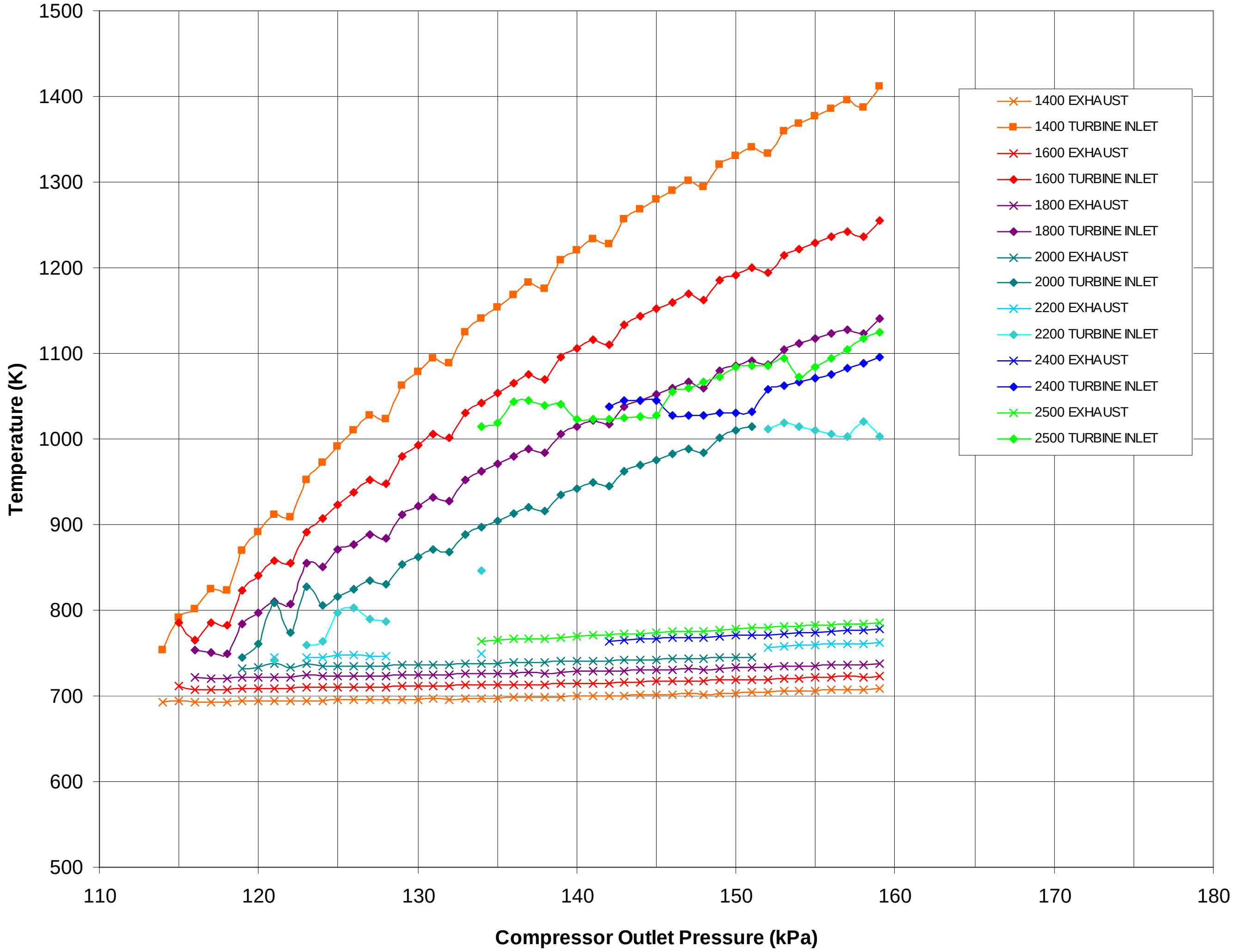

Figure B2: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 55 percent diesel equivalency ratio and no added propane.

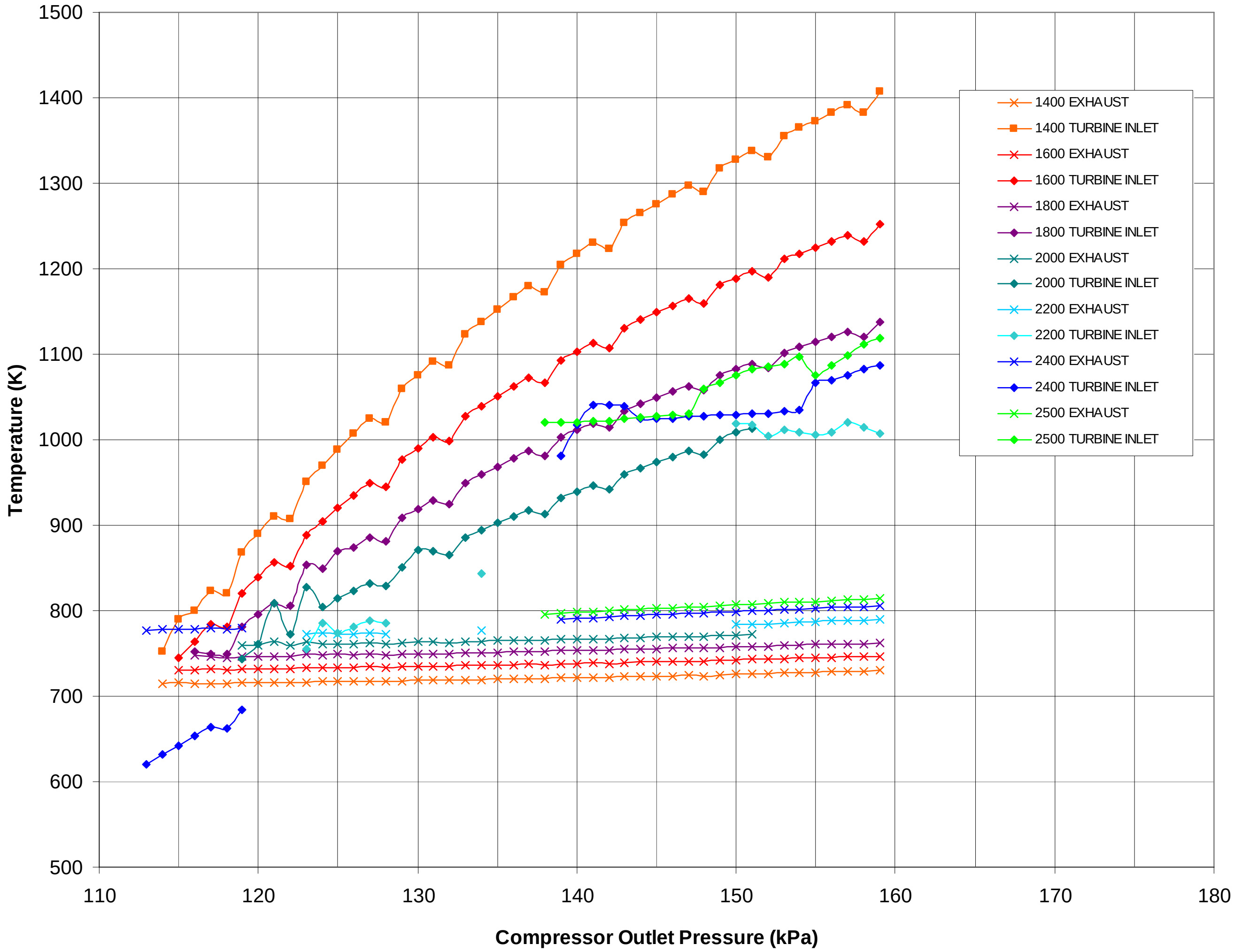

Figure B3: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 60 percent diesel equivalency ratio and no added propane.

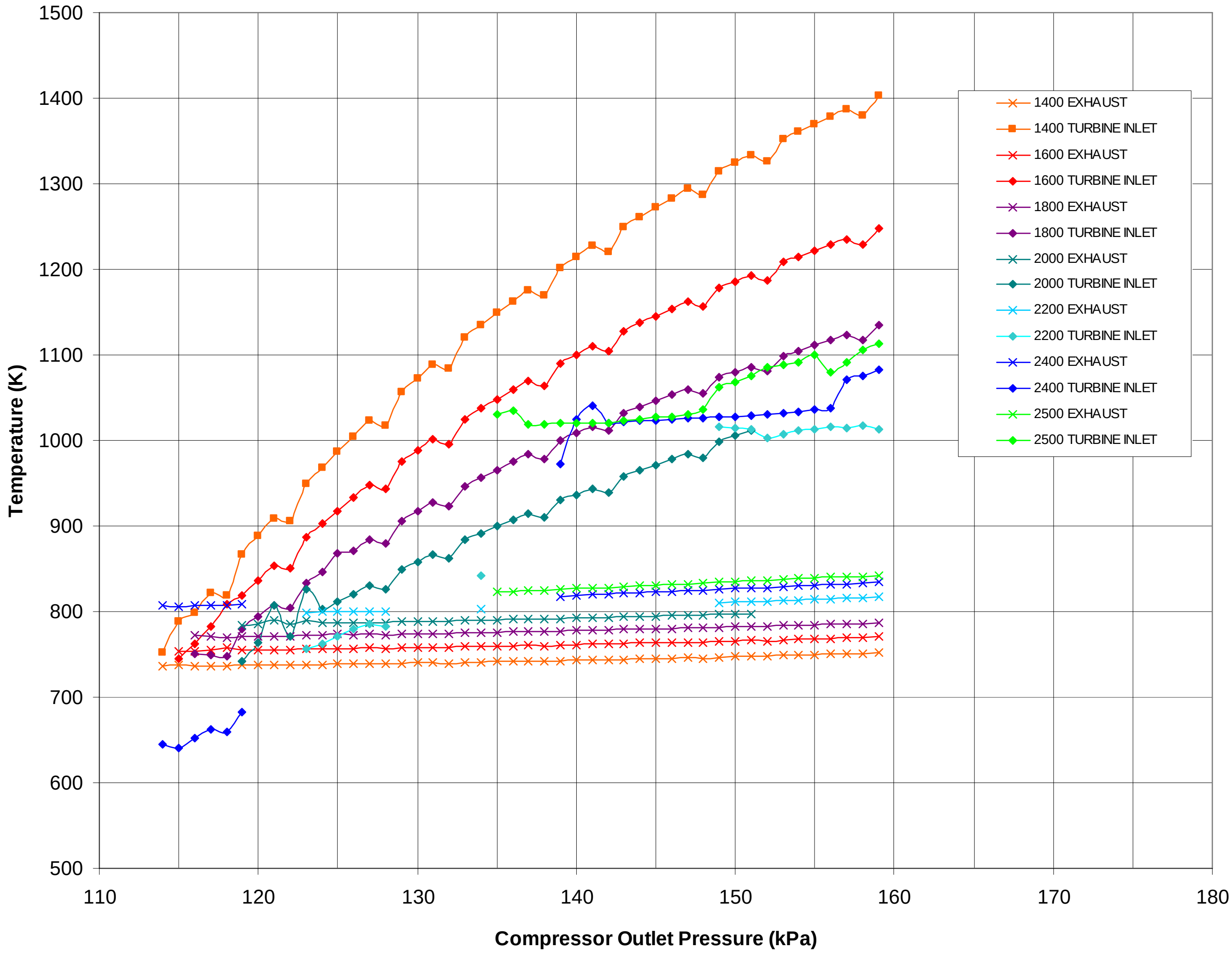

Figure B4: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 60 percent diesel equivalency ratio and no added propane.

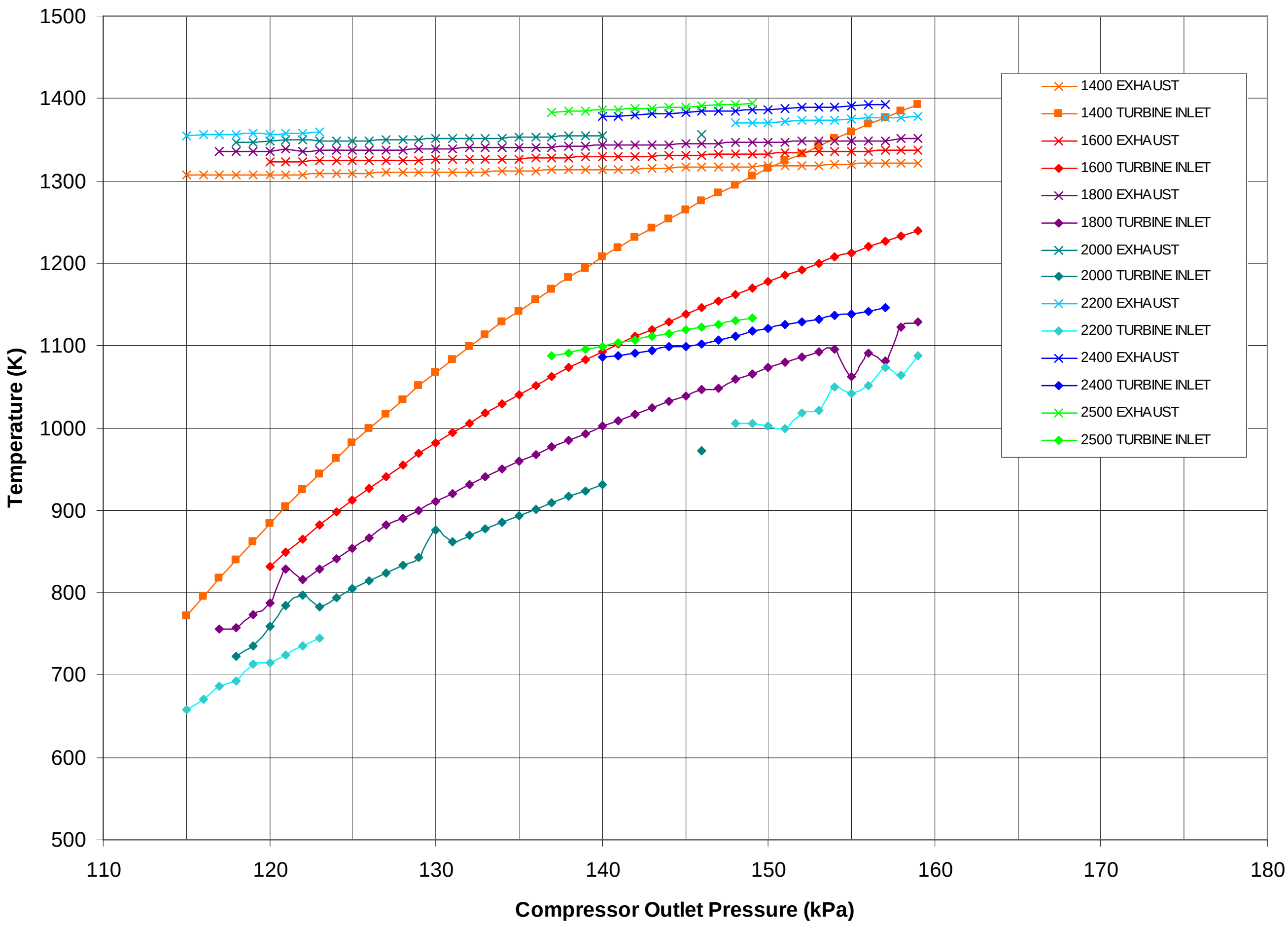

Figure B5: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 70 percent diesel equivalency ratio and no added propane.

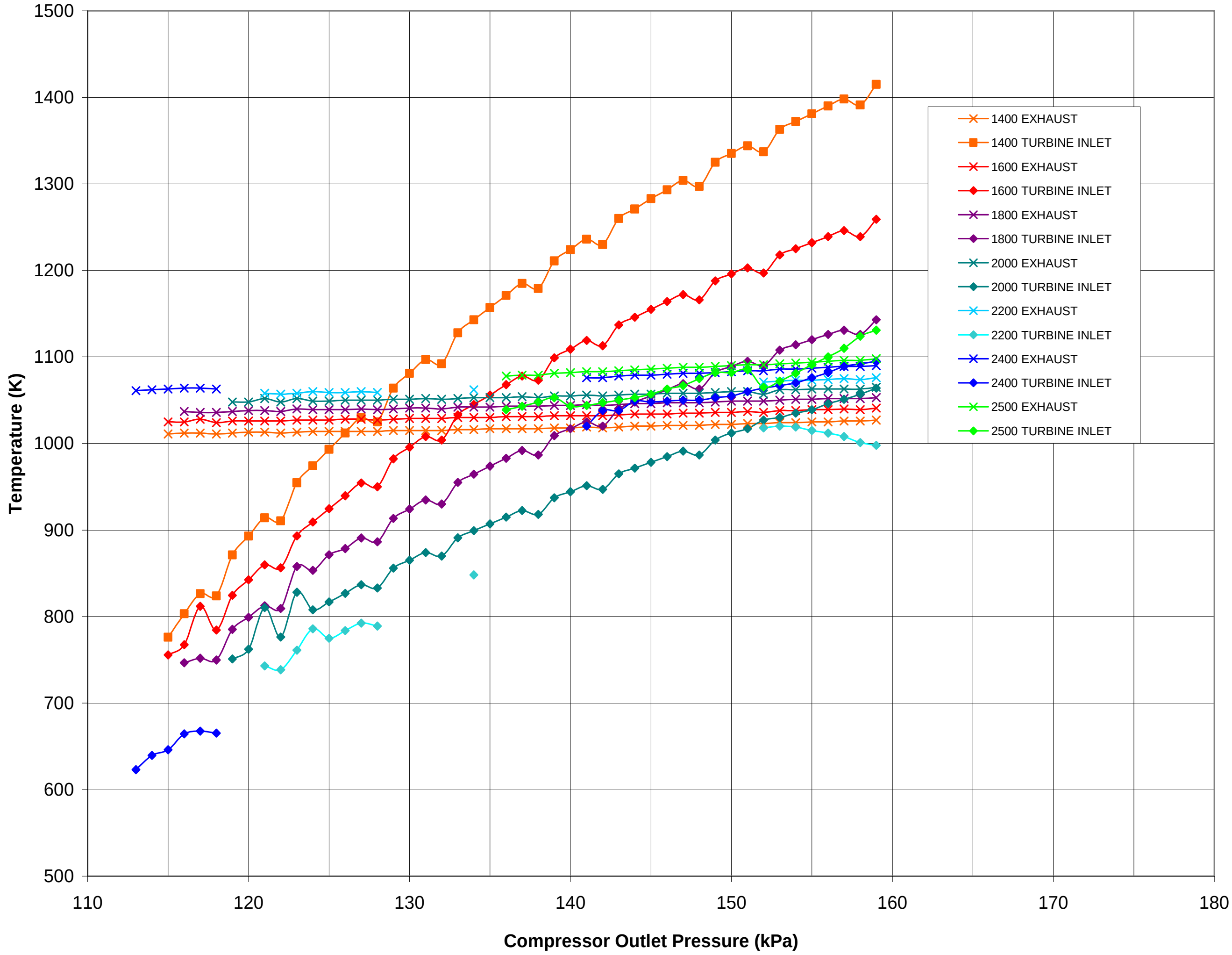

Figure B6: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 50 percent diesel equivalency ratio and 25 percent propane equivalency added propane.

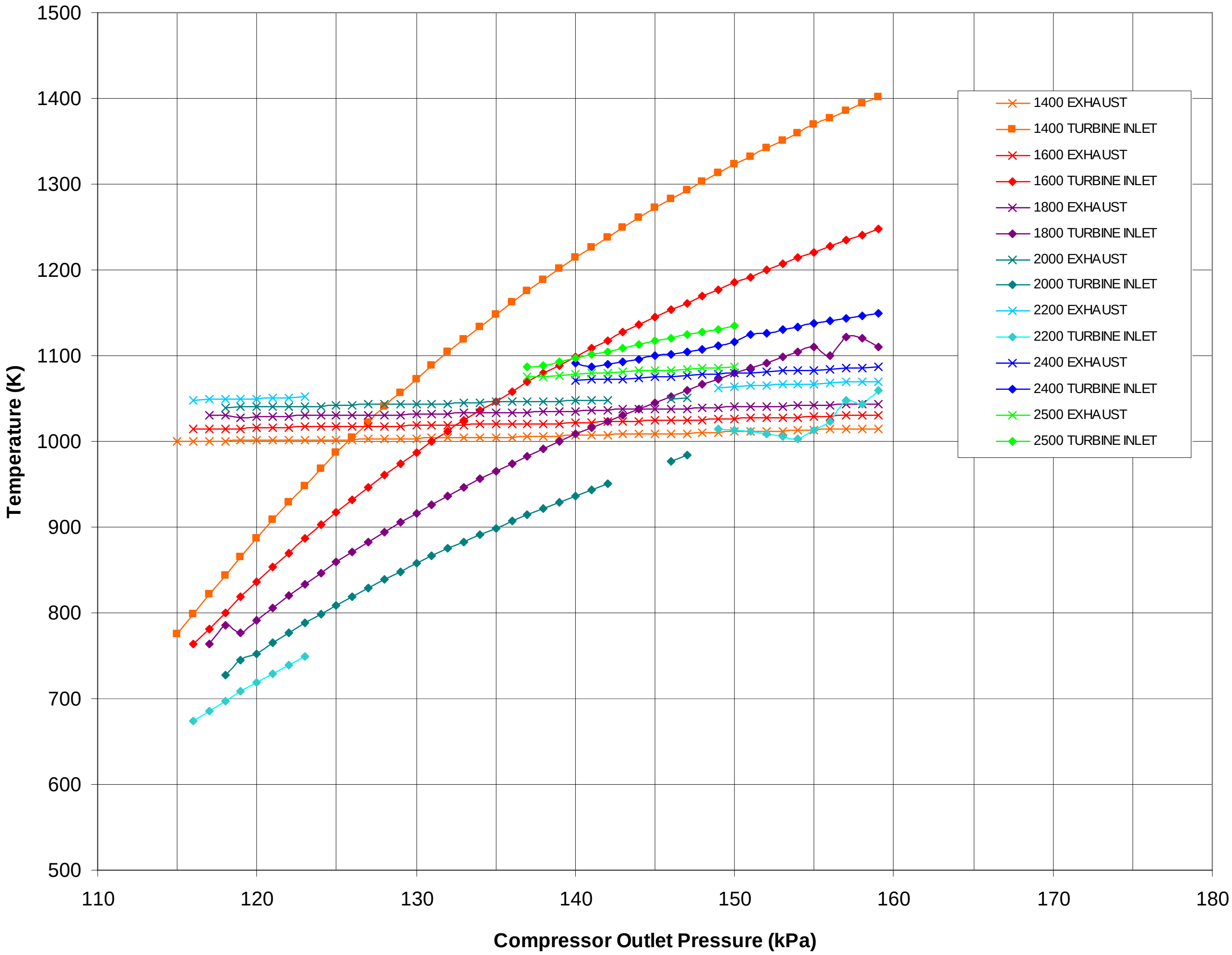

Figure B7: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 55 percent diesel equivalency ratio and 25 percent propane equivalency added propane.**

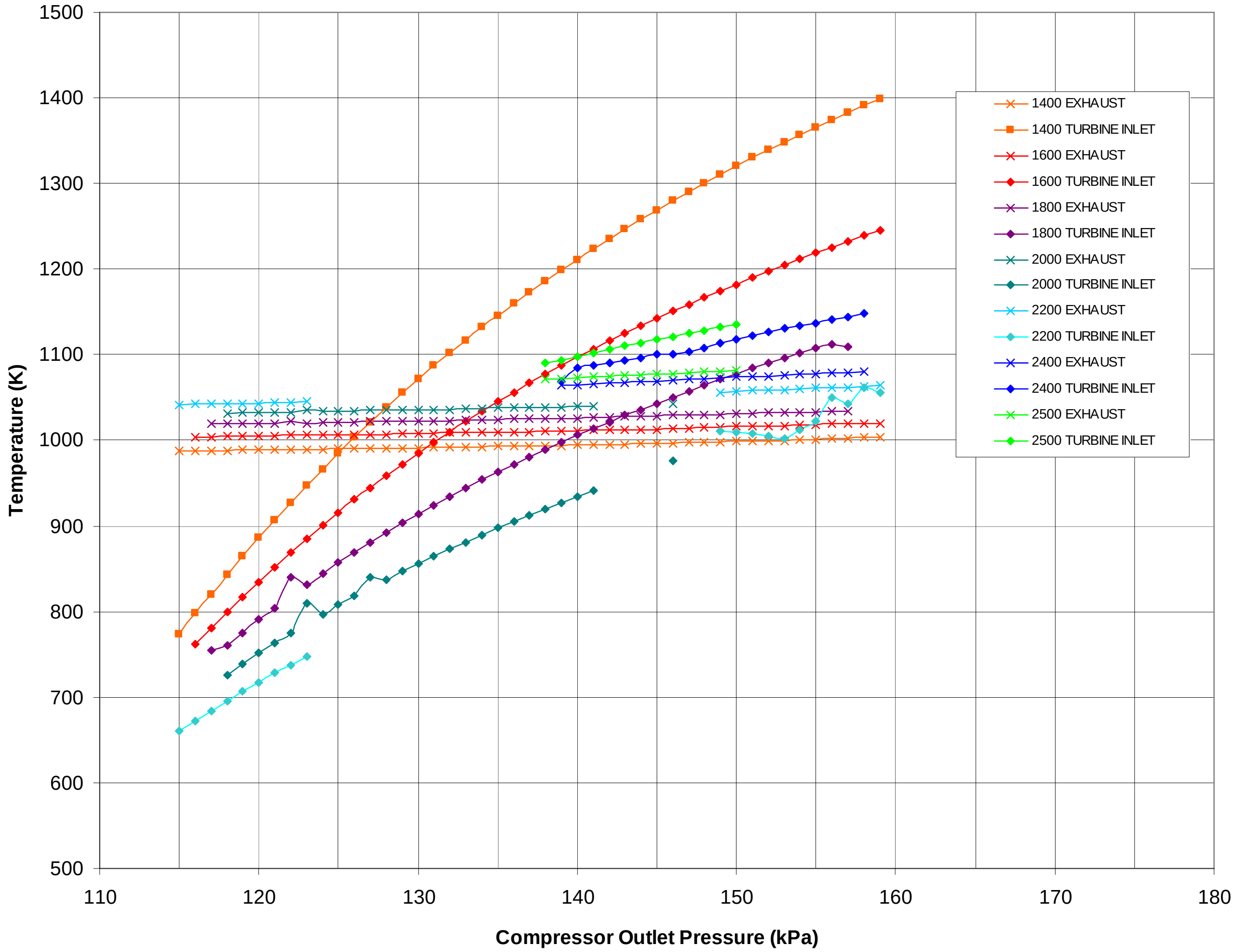

Figure B8: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 60 percent diesel equivalency ratio and 25 percent propane equivalency added propane.**

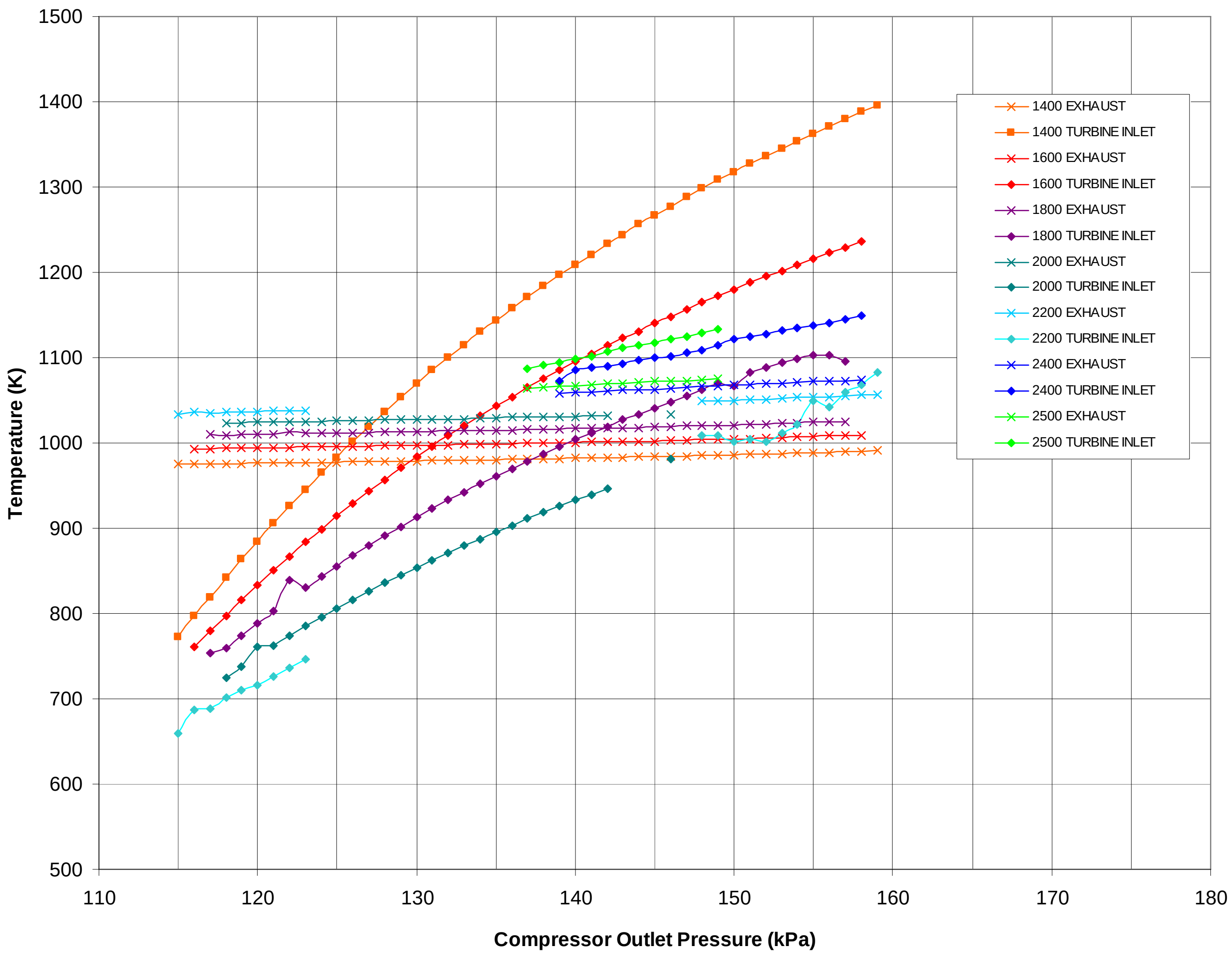

Figure B9: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 65 percent diesel equivalency ratio and 25 percent propane equivalency added propane.

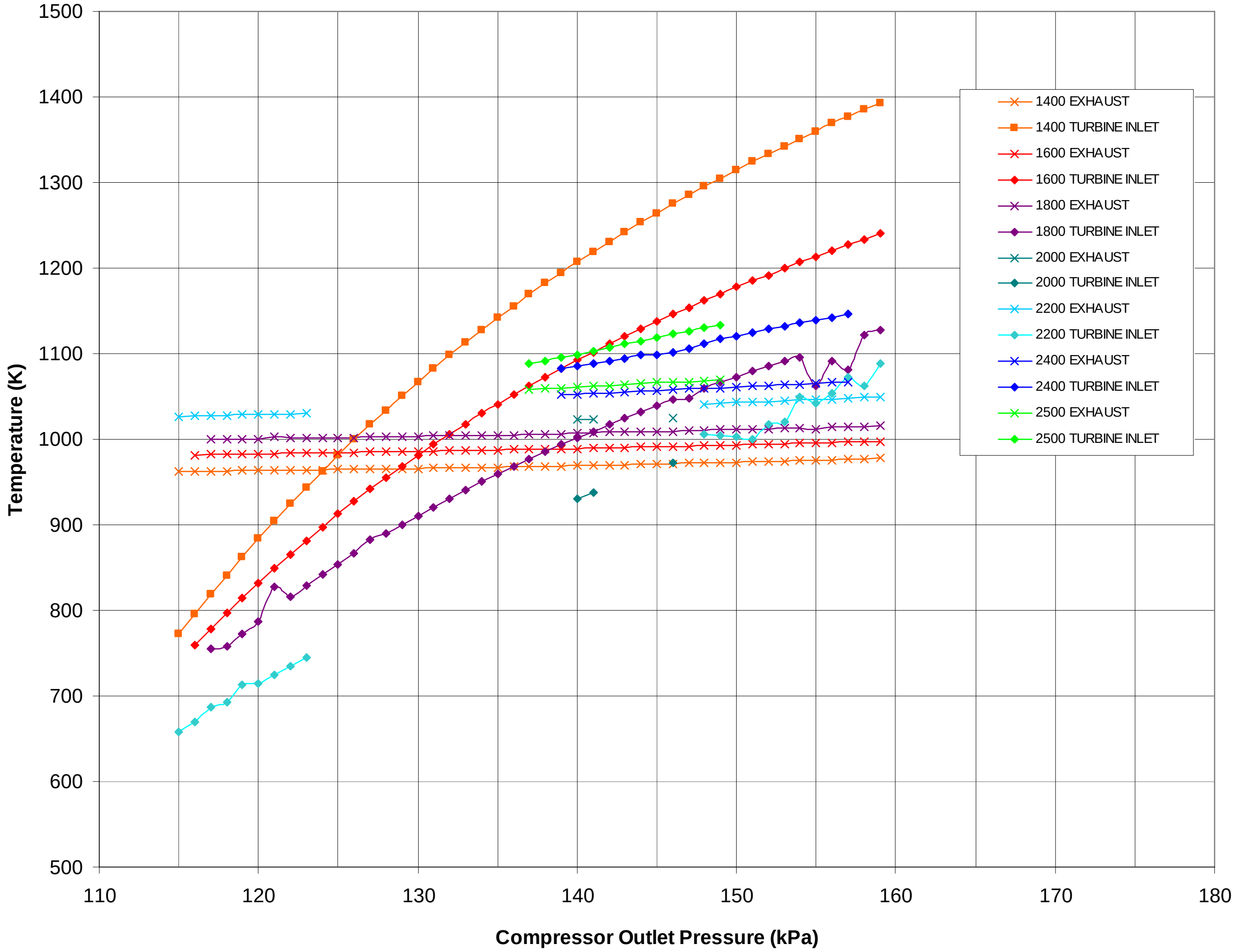

Figure B10: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 70 percent diesel equivalency ratio and 25 percent propane equivalency added propane.**

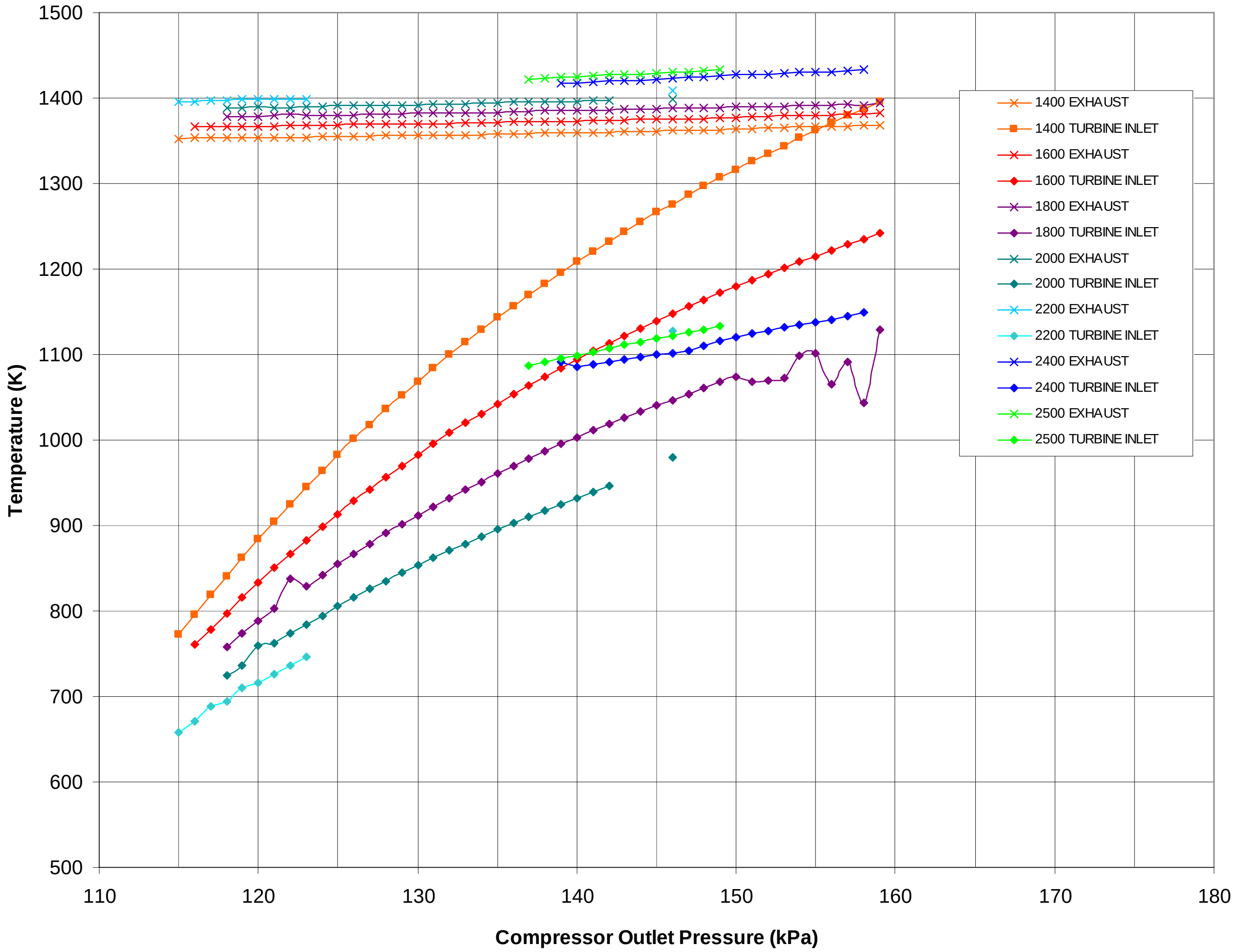

Figure B11: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 50 percent diesel equivalency ratio and 50 percent propane equivalency added propane.**

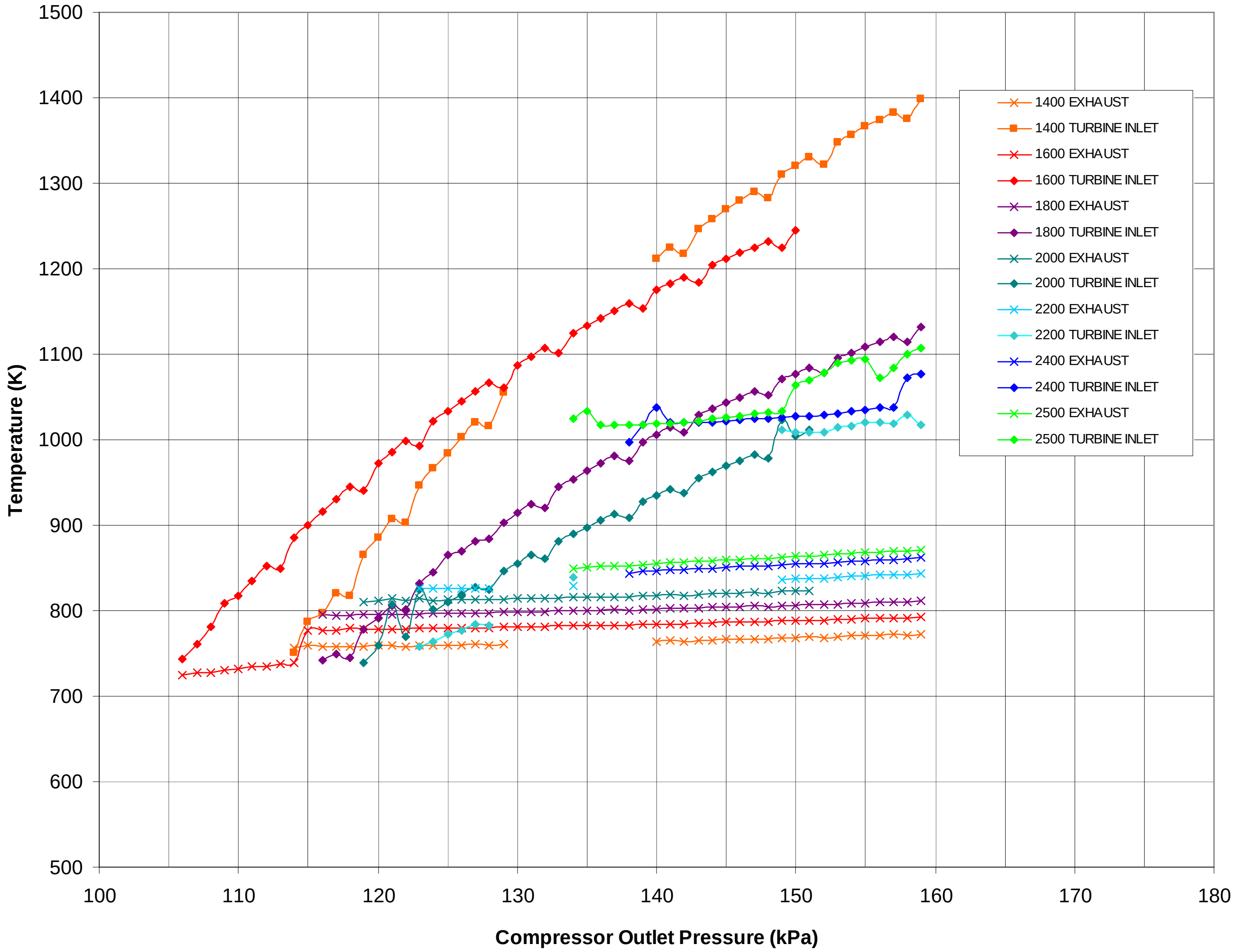

Figure B12: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 55 percent diesel equivalency ratio and 50 percent propane equivalency added propane.**

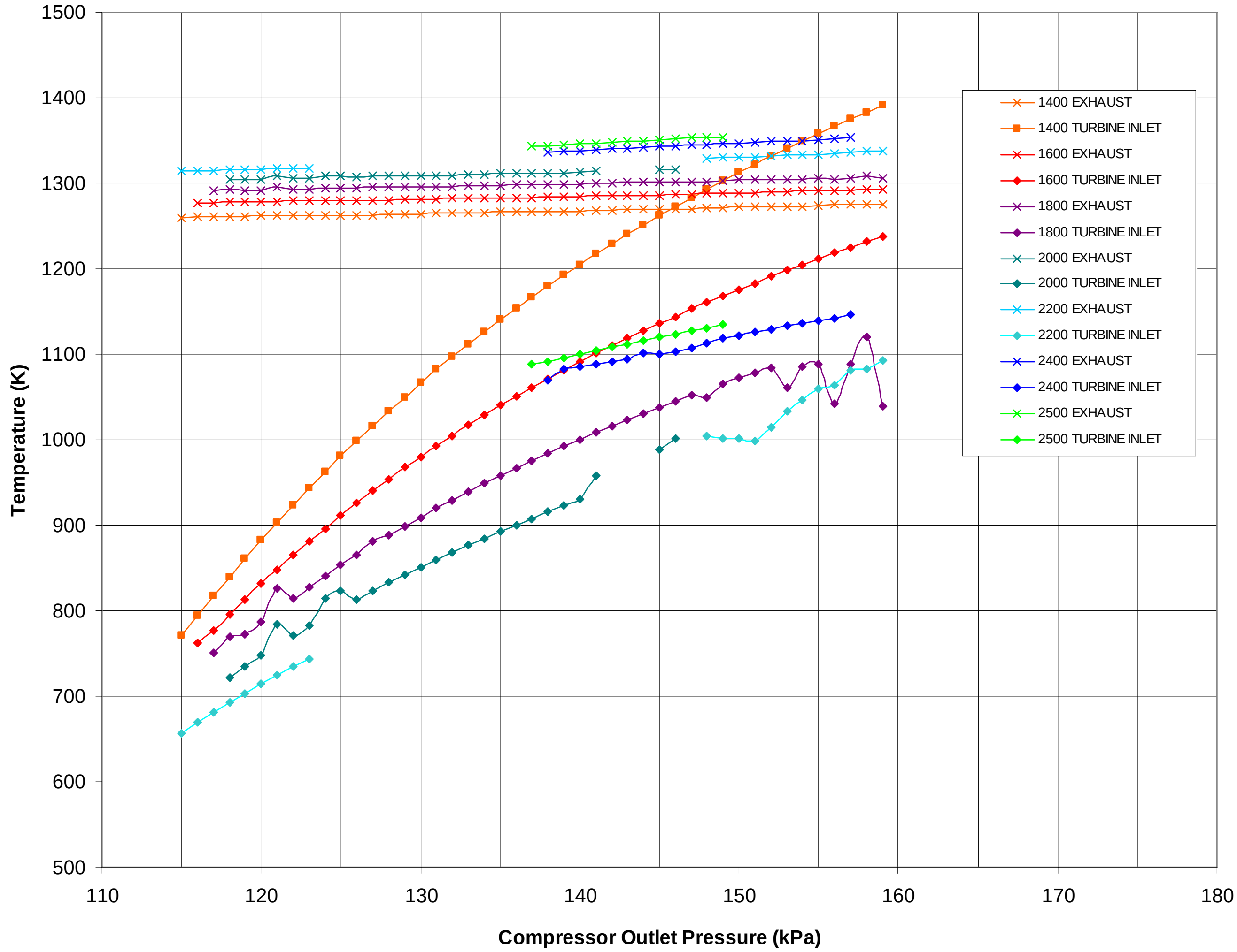

Figure B13: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 60 percent diesel equivalency ratio and 50 percent propane equivalency added propane.**

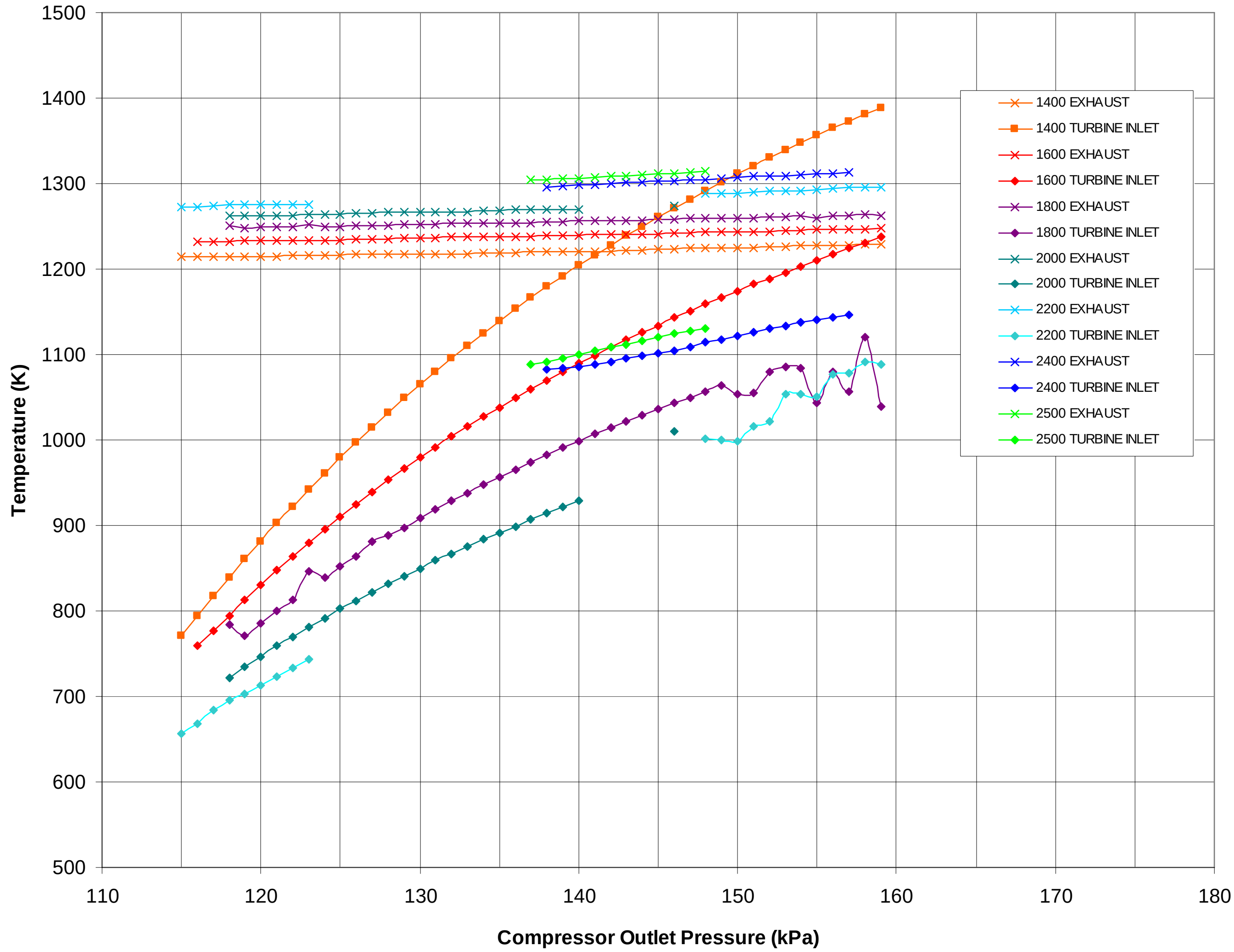

Figure B14**: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 65 percent diesel equivalency ratio and 50 percent propane equivalency added propane.**

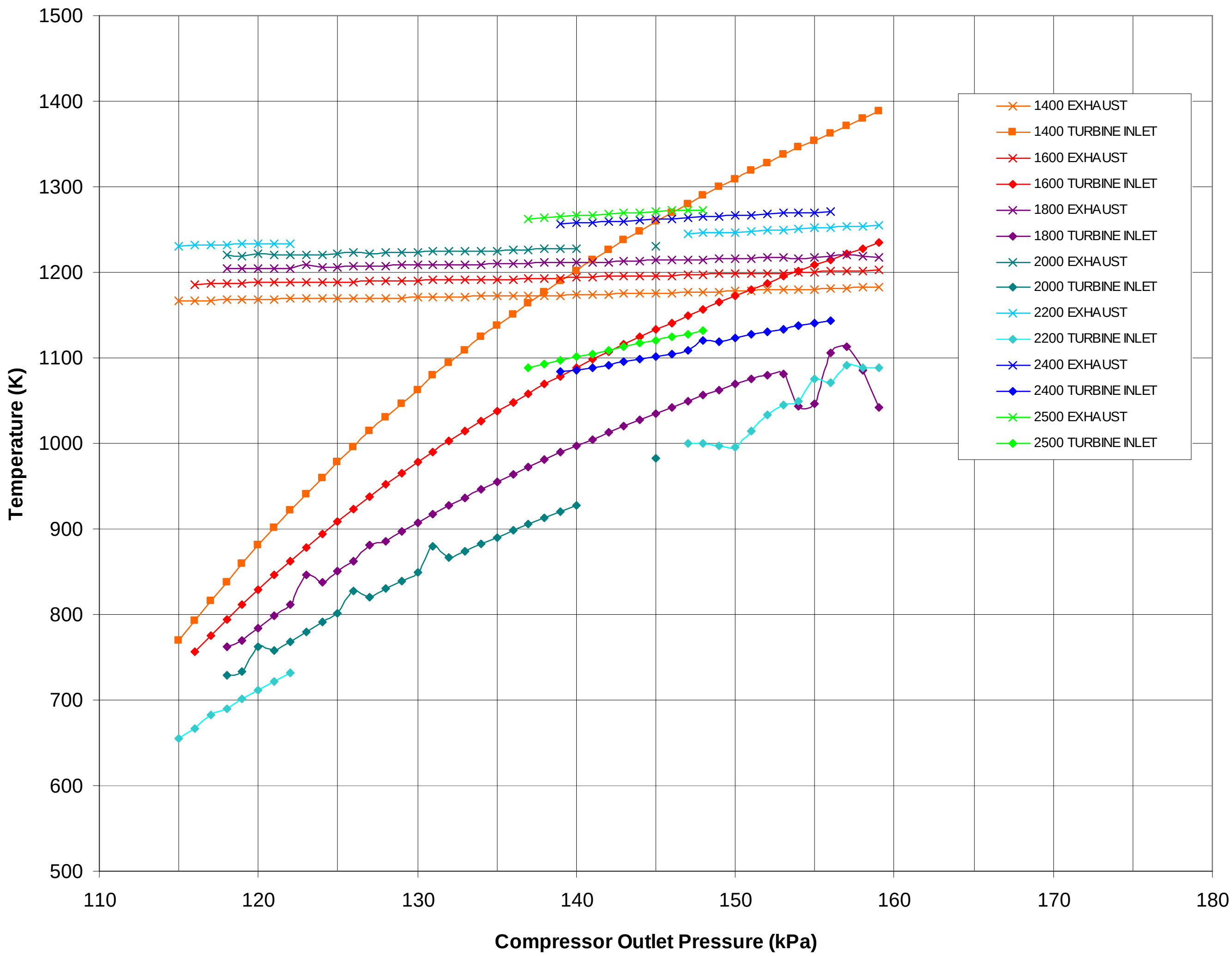

Figure B15: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 70 percent diesel equivalency ratio and 50 percent propane equivalency added propane.**

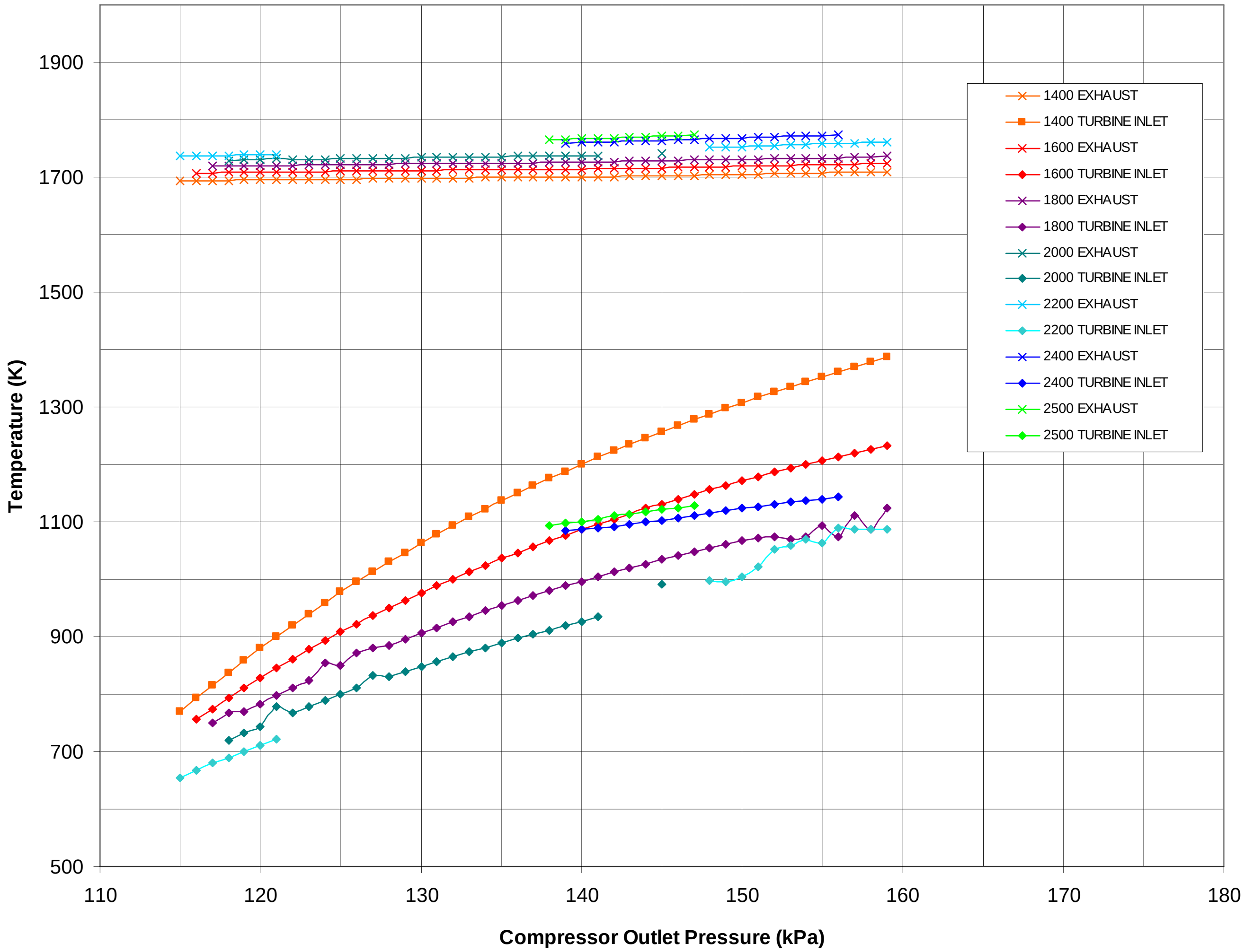

Figure B16: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 50 percent diesel equivalency ratio and 75 percent propane equivalency added propane.

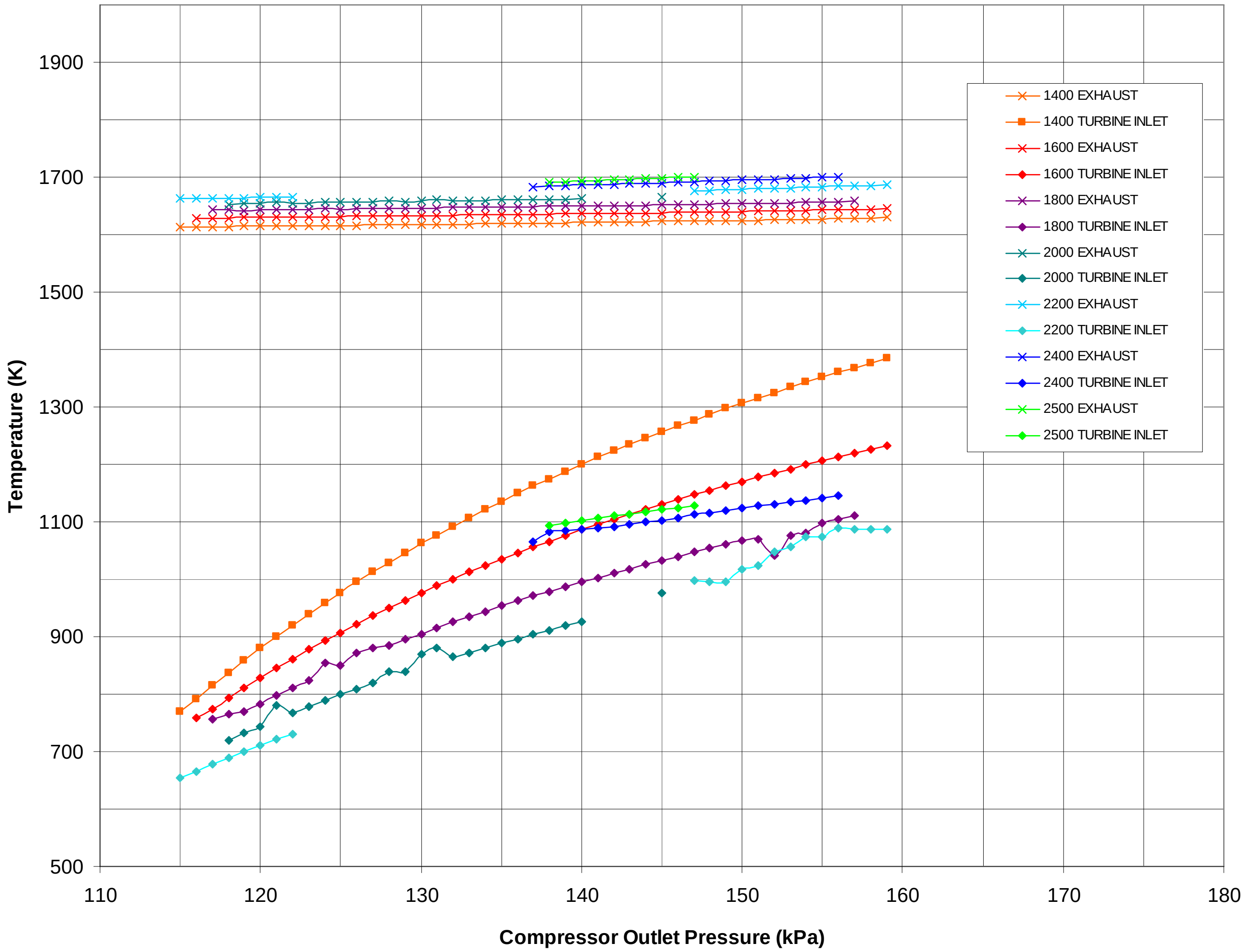

Figure B17: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 55 percent diesel equivalency ratio and 75 percent propane equivalency added propane.**

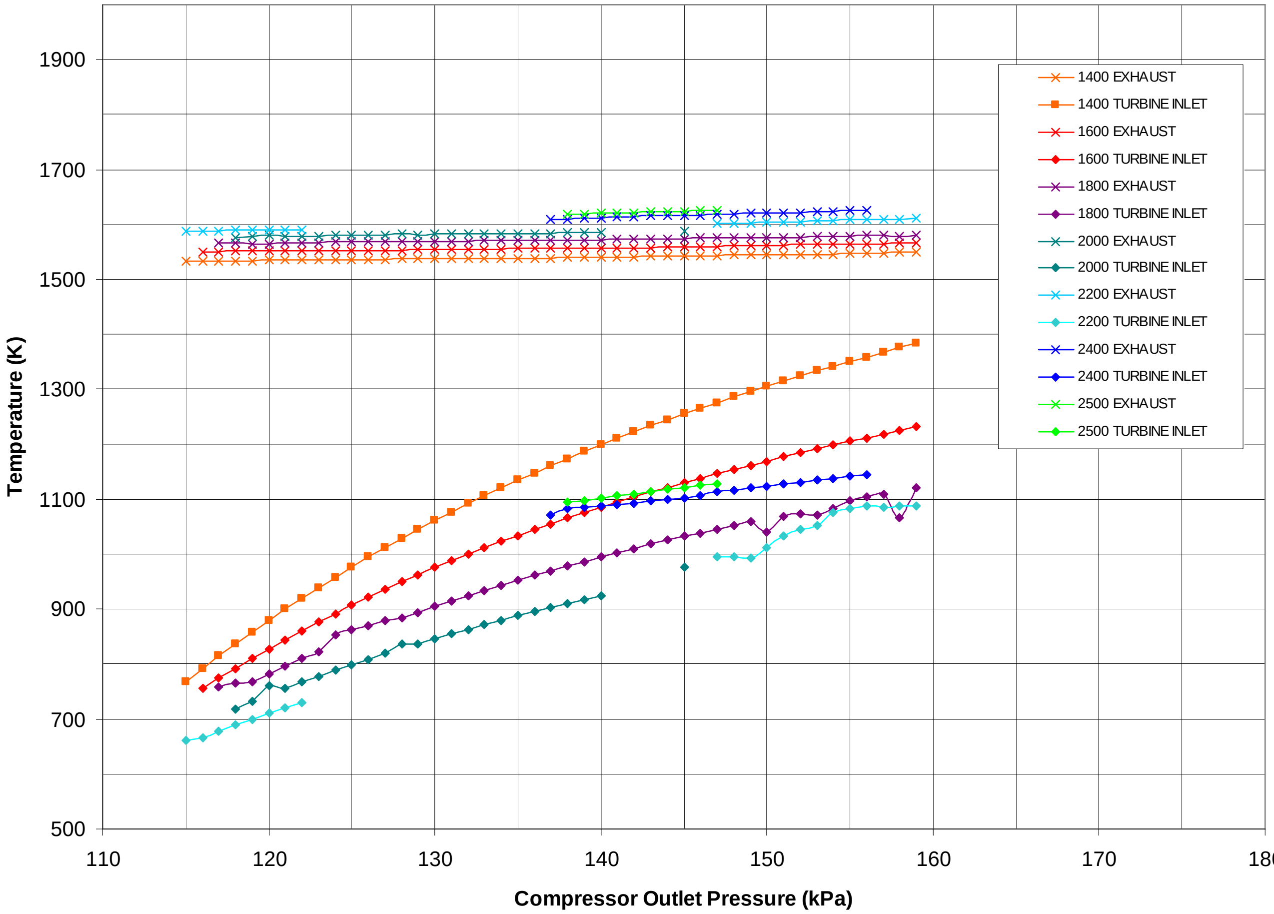

Figure B18: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 60 percent diesel equivalency ratio and 75 percent propane equivalency added propane.

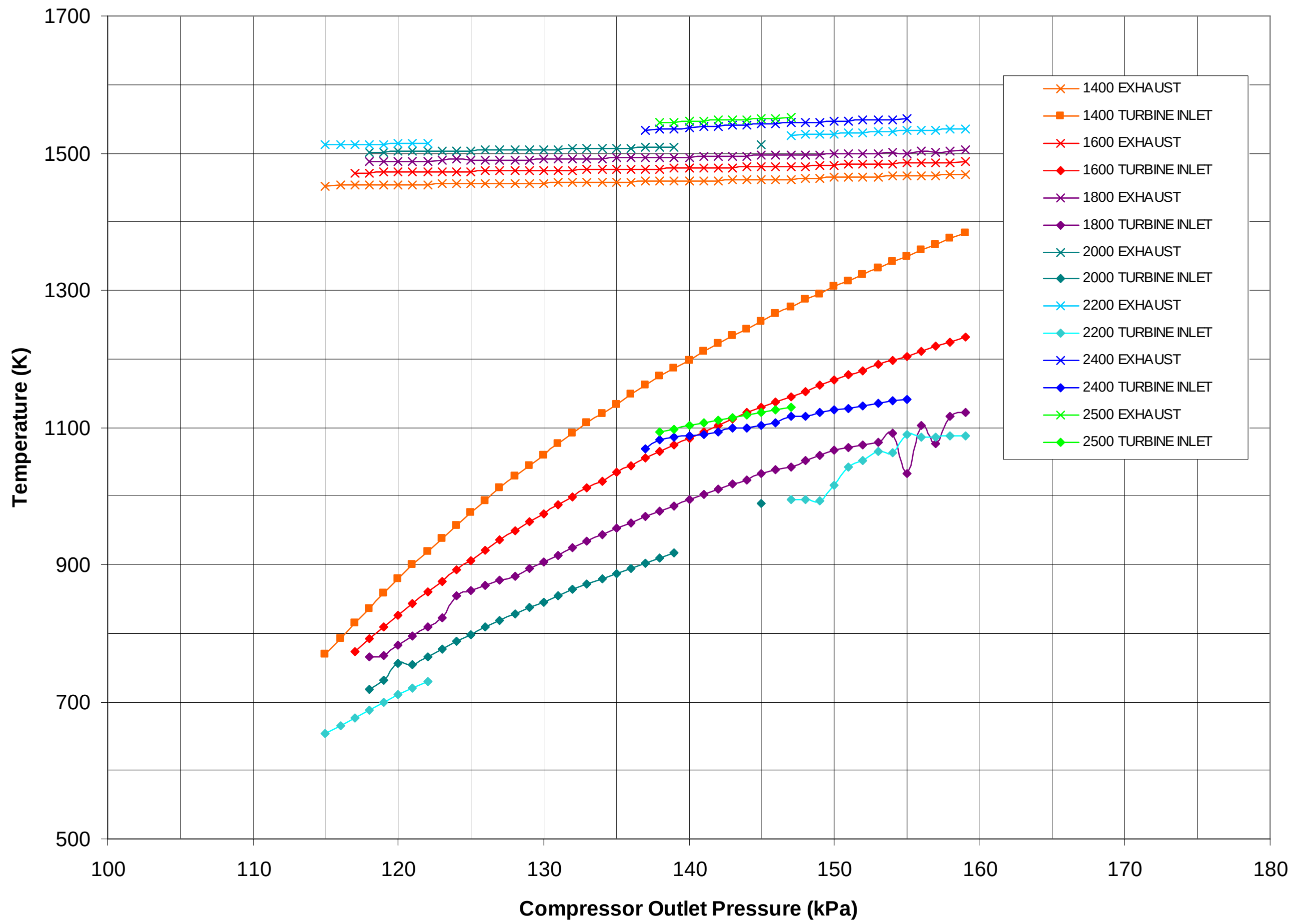

Figure B19: Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 65 percent diesel equivalency ratio and 75 percent propane equivalency added propane.

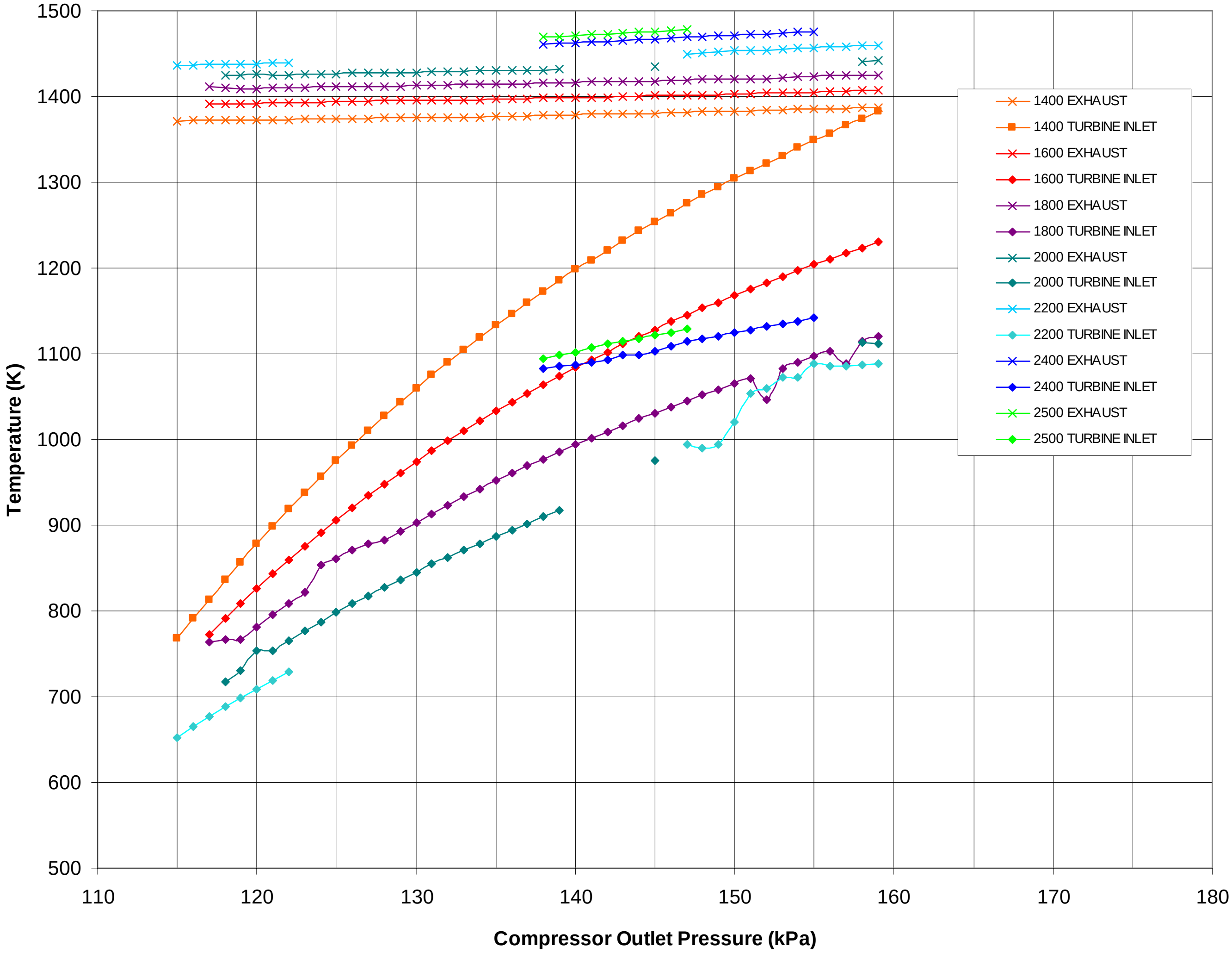

Figure B20: **Required turbine inlet temperature and combustion chamber exhaust temperature as a function of compressor outlet pressure for 70 percent diesel equivalency ratio and 75 percent propane equivalency added propane.**

Appendix C: SIMULATION RESULTS (Engine Torque as a function of Engine Speed)

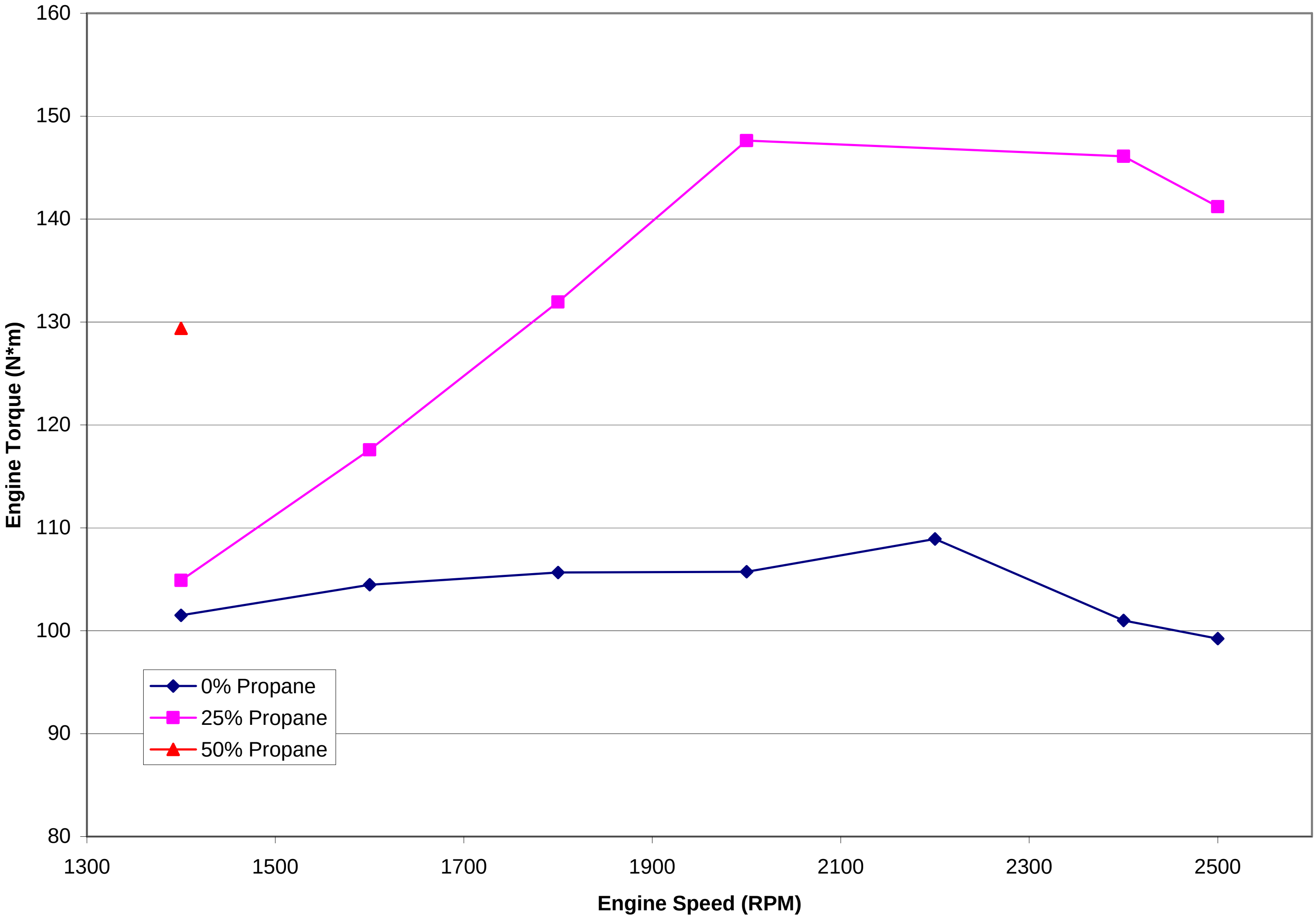

Figure C1: Engine torque as a function of engine speed for 50% diesel equivalence ratio and three propane equivalence ratios.

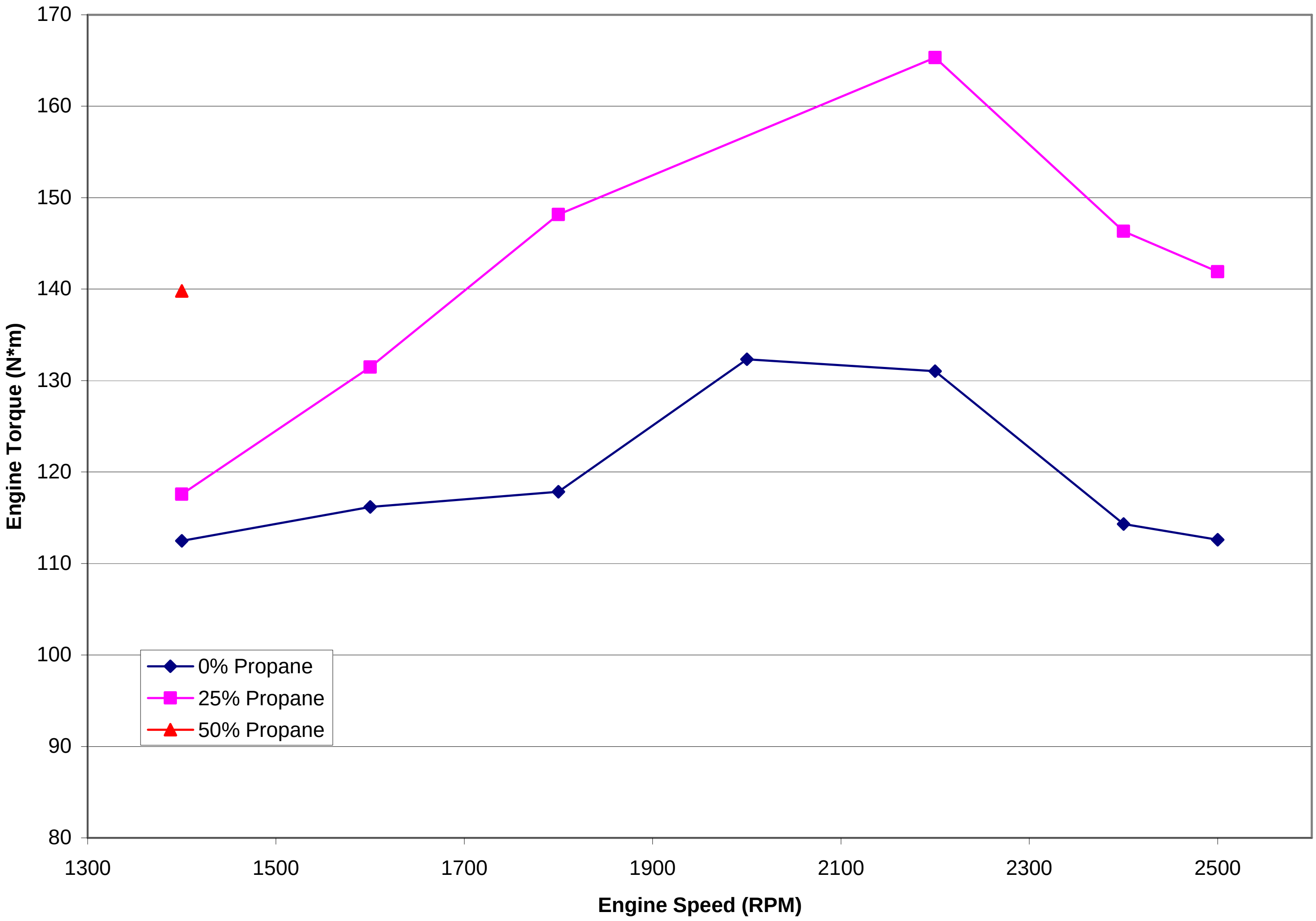

Figure C2: Engine torque as a function of engine speed for 55% diesel equivalence ratio and three propane equivalence ratios.

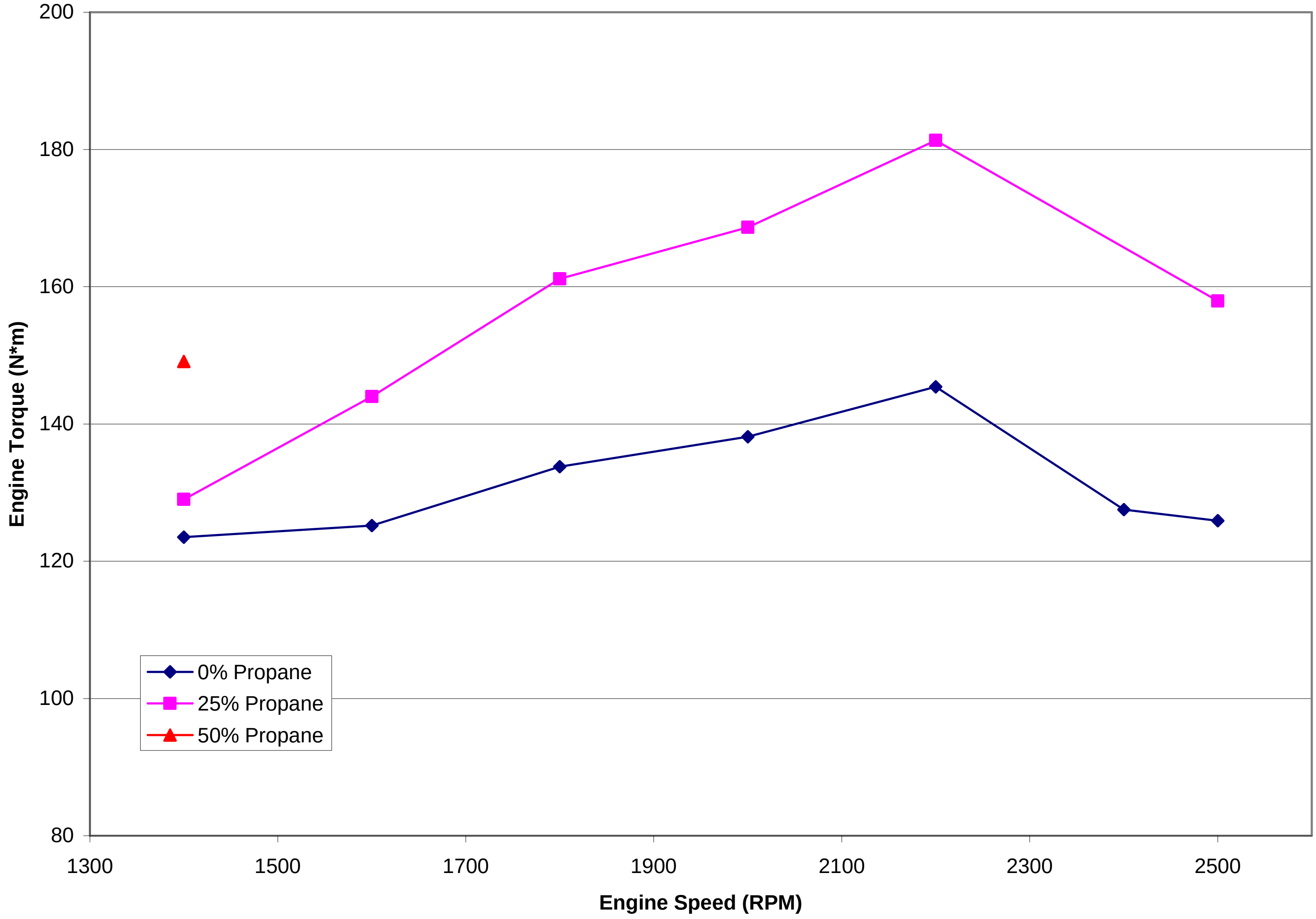

Figure C3: Engine torque as a function of engine speed for 60% diesel equivalence ratio and three propane equivalence ratios.

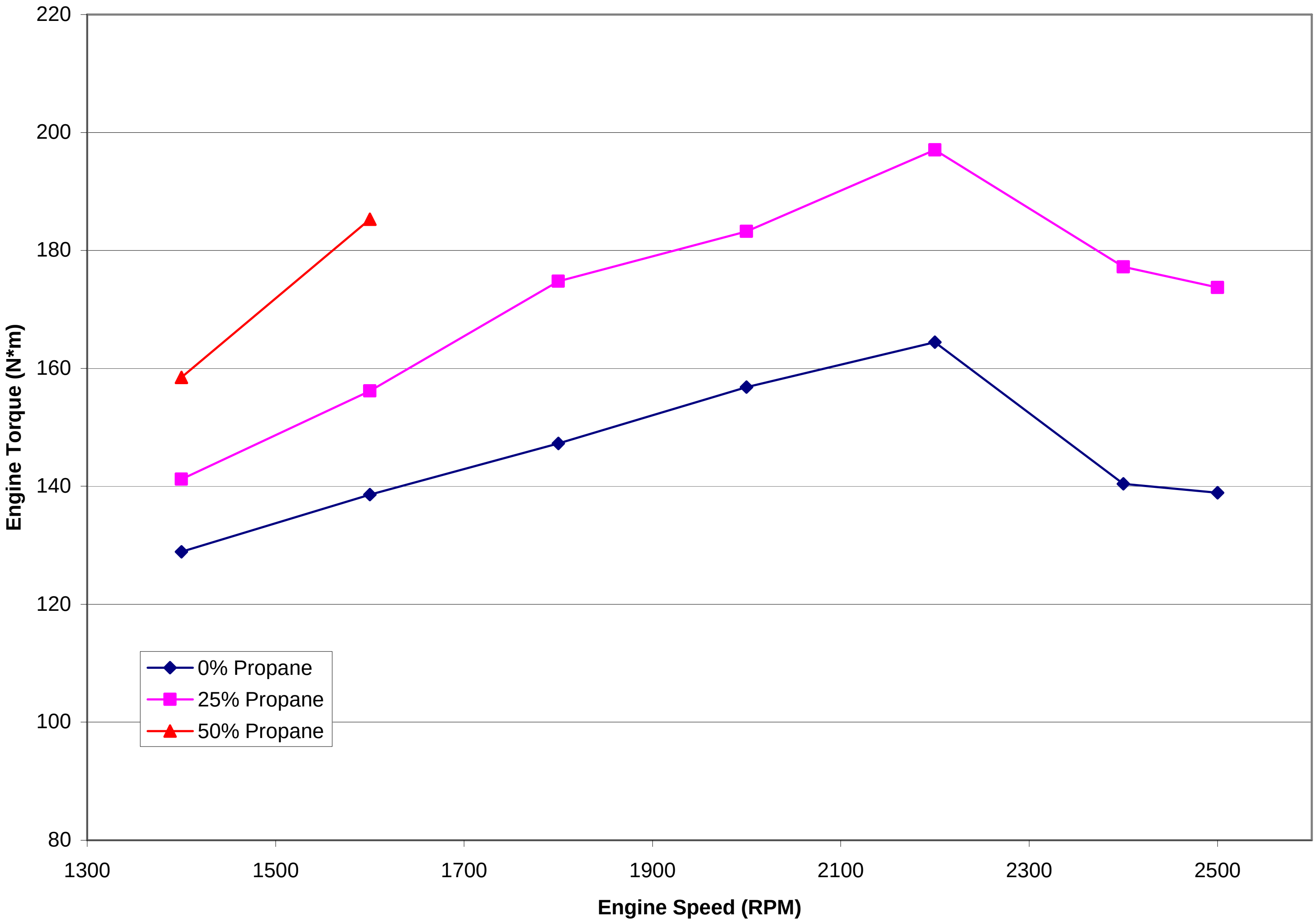

Figure C4: Engine torque as a function of engine speed for 65% diesel equivalence ratio and three propane equivalence ratios.

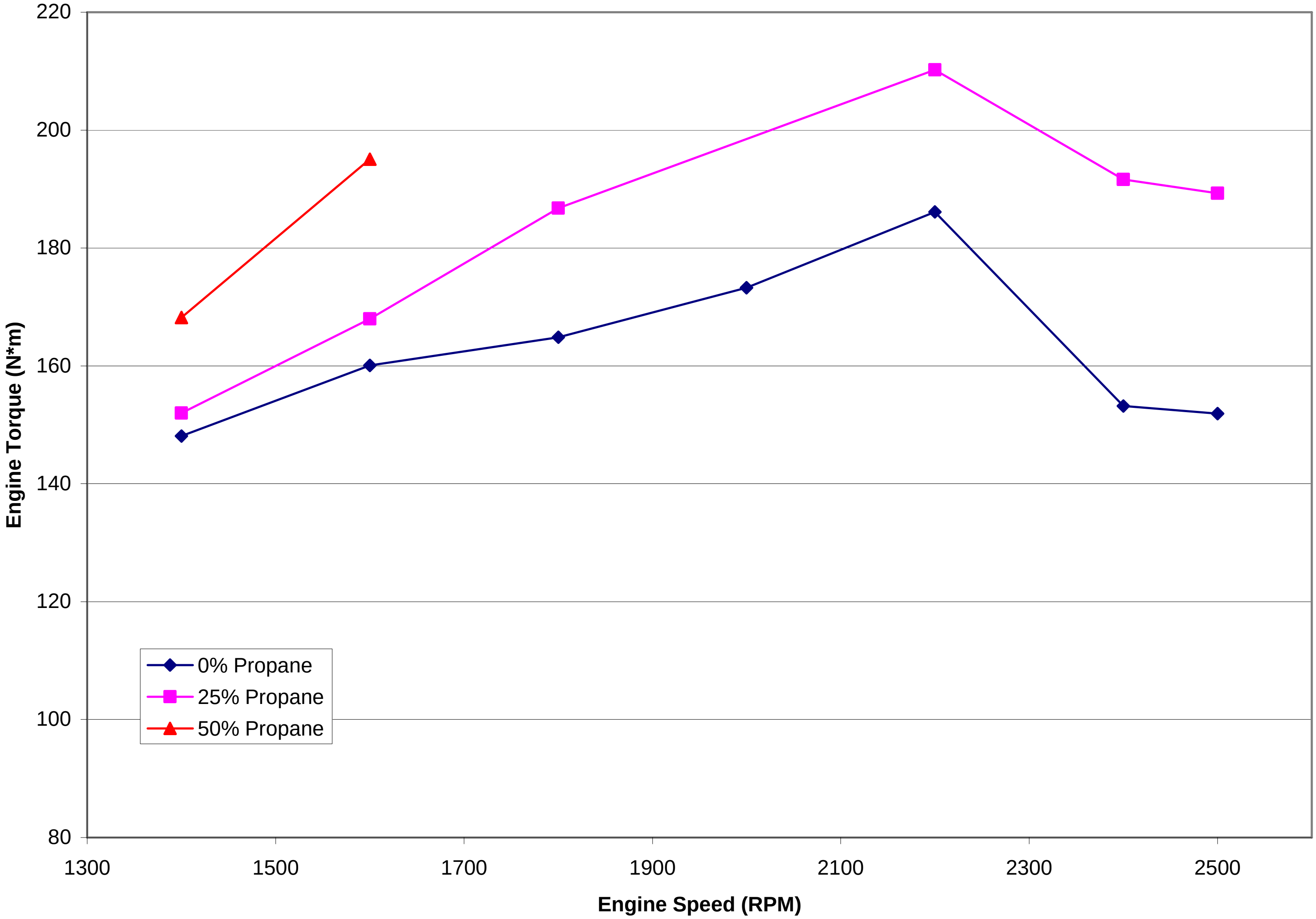

Figure C5: Engine torque as a function of engine speed for 70% diesel equivalence ratio and three propane equivalence ratios.