Design and Performance of a Gas-Turbine Engine from an Automobile Turbocharger

by

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ABSTRACT

The Massachusetts Institute of Technology Department of Mechanical Engineering teaches thermodynamics and fluid mechanics through a pair of classes, Thermal Fluids Engineering I & II. The purpose of this project was to design and fabricate a gas-turbine engine for demonstration use in these two classes. The engine was built from an automobile turbocharger with a combustion chamber connected between its compressor and turbine. Pressure and temperature sensors at different points of the engine cycle allow students to monitor the performance of the individual engine components and the complete engine cycle.

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List of Symbols

Symbols are listed in alphabetical order. Values are listed for symbols denoting constants.

Symbol	Meaning
a	internal diameter of combustor
A_1	area at the fuel hook inlet
A_m	total required cross-sectional area of bolts
A_{m1}	total required cross-sectional area of bolts for operating conditions
A_{m2}	total required cross-sectional area of bolts for gasket seating
A_t	area at the fuel hook outlet
b	diameter of gasket bolt placement
C_d	flow coefficient of propane through the fuel hook holes
c_P	specific heat at constant pressure
c_v	specific heat at constant volume
C_v	flow coefficient through the propane valve
d	outer diameter of combustor flange
f	see Equation 46
F	longitudinal tension force in the cylinder
g	gravitational acceleration
G	mean diameter of gasket contact face
γ	ratio: $\frac{c_P}{c_v}$
h	combustor flange thickness
h	enthalpy
$\bar{h}$	enthalpy of a particular compound at a given T and P
$\bar{h}^{\circ}_{T,P}$	enthalpy of formation
$\bar{h}_{T,P} - \bar{h}_{25^{\circ}\mathrm{C},1\mathrm{atm}}$	enthalpy to raise a compound to temperature T
H	total hydrostatic end force of combustor flange
H_p	total joint contact surface compression load on combustor flange
m	gasket factor
$\dot{m}$	mass flow rate
M_0	bending moment in combustor flange
η	efficiency
$\dot{n}$	molar flow rate
NWR	net work ratio
P	air pressure
$\dot{Q}$	heat transfer
R	gas constant
ρ	density of propane
s	specific entropy
s_2	tangential hoop stress in the flange
s_2'	tangential bending stress in the flange
s_{c1}	longitudinal direct stress in the cylinder

List of Symbols, Continued

Symbol	Meaning
s'_{c1}	longitudinal bending stress in the cylinder
$s_{c,\max}$	maximum longitudinal stress in the cylinder
s_{f1}	radial direct stress in the flange
s'_{f1}	radial bending stress in the flange
S	allowable bolt stress value
t	wall thickness of combustor
T	air temperature
τ_1	see Equation 46
τ_2	see Equation 46
ϑ	velocity
$\dot{V}_{\text{actual}}$	volumetric flow rate of propane
V_0	shear stress in combustor flange
$\dot{W}_{\text{shaft}}$	shaft work transfer
W_{m1}	minimum required bolt load for combustor operating conditions
W_{m2}	minimum required bolt load for combustor gasket seating
$\dot{x}$	molar flow rate of propane
$\dot{y}$	molar flow rate of air
z	vertical position

1 Introduction

The Massachusetts Institute of Technology Department of Mechanical Engineering teaches thermodynamics and fluid mechanics through a pair of classes, Thermal Fluids Engineering I & II. The courses cover the basic principles of thermodynamics, fluid mechanics, and heat transfer. This includes the rate processes involved with heat and work transfer, steady flow components of thermodynamic plants, and energy conversion cycles.

The purpose of this project was to design and fabricate a gas-turbine engine that is suitable for use as a classroom demonstration. The gas turbine designed and built for this project illustrates the operation of the Brayton cycle. The Brayton cycle is a convenient cycle to demonstrate because it involves a combination of standard components, which are used in many other energy conversion applications. The Brayton cycle consists of a compressor, a heat exchanger, a turbine, and another heat exchanger. The gas turbine engine operates on an open version of the Brayton cycle and allows students to measure the temperature and pressure changes associated with each system component. The students can then use these values to calculate the performance of the engine components as well as the overall cycle efficiency.

In order to make the engine suitable for demonstration use, there were several requirements. The engine had to be portable so that it could be easily moved to the desired location. It also needed run on an easily attainable fuel and at a safe maximum temperature. The state of the air flow through the engine also needed to be measurable at each point of the engine cycle so that students could compare the predicted and measured performance of the engine.

The construction of the engine involved the design and selection of each component of the gas turbine and the engine's auxiliary systems. The engine is based around an automobile turbocharger comprised of a compressor and turbine that operate on a common shaft. Between the outlet of the compressor and the turbine inlet is a combustion chamber. The design and fabrication of the combustion chamber represents the bulk of the work for this thesis. In addition to these main system components, the cooling and lubrication system runs oil through the turbocharger, a generator creates a spark to ignite the fuel, and a series of components is used to control the flow rate of fuel to the engine. The completed engine is shown in Figure 1.

This thesis is the continuation of the project begun by Keane Nishimoto for his undergraduate thesis requirement[11]. His thesis included the selection of the turbocharger, the purchase of the oil pump for the lubrication and cooling system, and the initial idea of a concentric shell design for the combustion chamber.

This thesis begins with a description of the basic thermodynamic theory behind gas turbine engines, examining the cycle of the energy conversion system and the individual components involved. Then, there is a brief introduction to automobile turbochargers, which is followed by a discussion of the selection and design of each component of the engine. Section 8 describes the auxiliary systems needed for the engine to run: the lubrication and cooling system, the ignition system, and the temperature and pressure measurement devices used to monitor the engine's performance. Final conclusions and recommendations

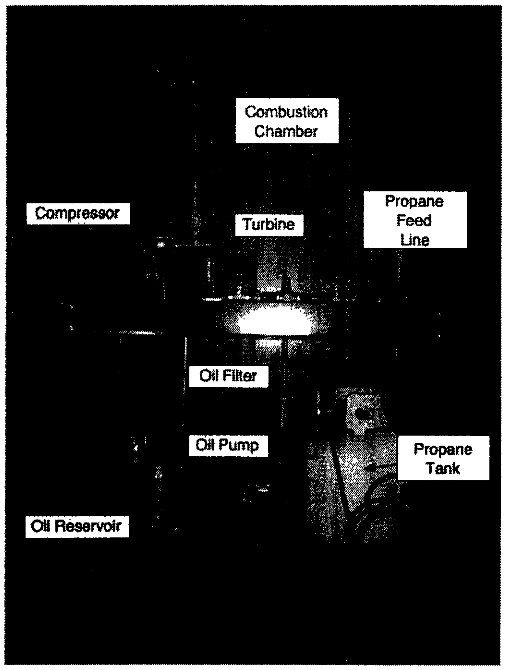

Figure 1: Completed Gas-Turbine Engine

are explained in Section 9.

2 Gas Turbine Power Plants

Gas turbines are thermodynamic systems that use fuel and air to produce a positive work transfer. They convert the chemical potential energy of the fuel to mechanical energy. The gas turbine operates on an open cycle consisting of a compressor, a combustor, and a turbine combined in series (Figure 2). Air from the atmosphere enters the compressor where it is compressed by a negative shaft work transfer. The compressed air is then combined and burned with fuel in the combustion chamber. The combustor increases both the temperature and the specific volume of the air. The hot air is then fed into the turbine where it is expanded. The expansion of the air creates a positive shaft work transfer. The expanded air is then exhausted to the atmosphere. A net positive shaft work transfer is produced because the negative shaft work transfer required to power the compressor is less than the positive work transfer produced by the turbine.

The gas turbine can be modeled as a closed system with air as the working fluid if the following assumptions are made:

- The combustion chamber is modeled as a constant pressure heat transfer device.

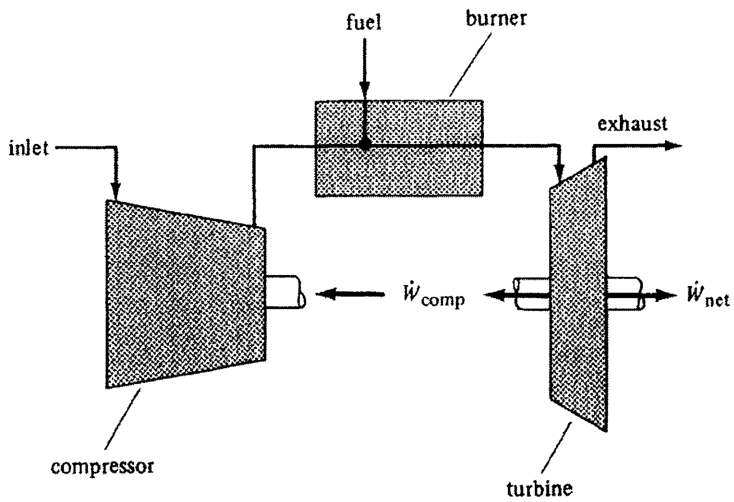

Figure 2: Components of a Basic Gas Turbine Engine [4]

- The heat transfer rate in the combustor is determined by the product of the mass flow rate of the fuel and the heating value of the fuel.

- The increase in the mass flow rate due to the addition of fuel in the combustion chamber is neglected because it is small relative to the flow rate of the air.

In this closed cycle, the exhaust air must be cooled back to the inlet state by a constant pressure heat transfer process. This closed cycle is known as the Brayton cycle.

2.1 The Brayton Cycle

The Brayton cycle consists of two adiabatic work transfers and two constant pressure heat transfer heat processes (Figure 3). From State 1 to State 2 the gas undergoes an isentropic, adiabatic compression. This process increases the temperature, pressure, and density of the gas. From State 2 to State 3, heat is added at constant pressure. For a gas-turbine, heat is added through a combustion process. From State 3 to State 4 the gas passes through an adiabatic isentropic turbine which decreases the temperature and pressure of the gas. For the closed Brayton cycle, heat is removed from the gas between State 4 and State 1 via a heat exchanger.

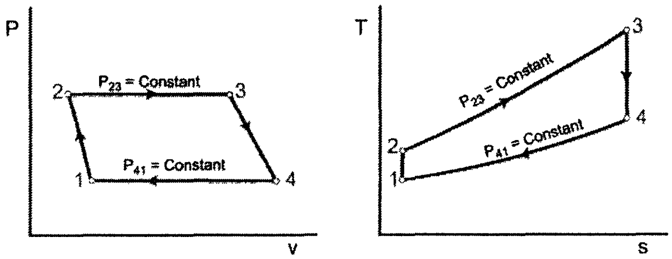

Figure 3: Brayton Cycle Processes [14]

2.2 First Law Analysis of the Brayton Cycle

The first law of thermodynamics for an open system is

$$\frac{dE_{CV}}{dt} = \dot{Q} - \dot{W}_{\text{shaft}} + \sum_{\text{in}} \dot{m} \left(h + \frac{\vartheta^2}{2} + gz \right)_{\text{in}} - \sum_{\text{out}} \dot{m} \left(h + \frac{\vartheta^2}{2} + gz \right)_{\text{out}} . \tag{1}$$

The left hand side of the equation represents the change in energy within the control volume over time. For the steady flow processes of the Brayton cycle and the gas turbine, the system inside the control volume is the same from one point in time to another and so the change in energy is zero. The heat transfer in and out of the system is characterized by $\dot{Q}$, and the shaft work transfer is $\dot{W}_{\text{shaft}}$. The third and fourth terms on the right hand side of the equation represent the energy that is convected out of the control volume by the mass flowing across the control surface. The enthalpy of the flow is represented by h, the velocity term represents the kinetic energy of the flow, and the final term, gz, represents the gravitational potential energy. For the purposes of this analysis, the kinetic and gravitational energy are not significant factors and can be neglected. The first law becomes

$$0 = \dot{Q} - \dot{W}_{\text{shaft}} + \dot{m} \left(h_{\text{in}} - h_{\text{out}} \right) . \tag{2}$$

The first process of the Brayton cycle is an adiabatic compression from P_1 to P_2. Because the compression is adiabatic, there is no heat transfer and the work required to run the compressor is

$$\dot{W}_{1-2} = \dot{m} \left(h_1 - h_2 \right) . \tag{3}$$

The second process is an addition of heat at constant pressure. There is no shaft work transfer so the heat addition is

$$\dot{Q}_{2-3} = \dot{m} \left(h_3 - h_2 \right) . \tag{4}$$

Then the gas is expanded through an adiabatic isentropic turbine. As in the first process, there is no heat transfer. The work transfer is

$$\dot{W}_{3-4} = \dot{m} \left(h_3 - h_4 \right) . \tag{5}$$

Finally heat is rejected using a heat exchanger to return the gas to its inlet state. There is no shaft work transfer so the heat transfer is

$$\dot{Q}_{4-1} = \dot{m}\,(h_1 - h_4) \tag{6}$$

.

3 Gas Turbine Components

3.1 Compressors

Compressors are an example of negative shaft work machines. They increase both the temperature and pressure of the working fluid. Increasing the pressure of the fluid requires a negative shaft work transfer. This work transfer was characterized by Equation 3. Most compressors can be considered adiabatic because the fluid is in the machine for a short time relative to the time necessary for the fluid to reach thermal equilibrium. Therefore there is virtually no heat transfer. Assuming that the compressor operates adiabatically and reversibly, the second law of thermodynamics becomes

$$s_2 - s_1 = c_P \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} = 0. \tag{7}$$

Given an initial temperature and pressure and the pressure ratio of the compressor, the reversible exit temperature, T_{2R}, can be calculated by rearranging Equation 7

$$T_{2R} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} T_1 \tag{8}$$

where γ is the ratio of specific heat at constant pressure, c_P, to specific heat at constant volume, c_v. For air this ratio is 1.4.

Once the reversible outlet temperature is known, the shaft work transfer for the reversible operation of the compressor can be calculated (Equation 3). The compressor efficiency, η_C, is the ratio of the work transfer required for reversible operation, $\dot{W}_{\text{rev}}$, to the actual work transfer, $\dot{W}_{\text{act}}$. For an ideal gas, enthalpy is the product of specific heat at constant pressure, c_P, and temperature. The efficiency becomes

$$\eta_C = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{act}}} = \frac{\dot{m} c_P \,(T_{2R} - T_1)}{\dot{m} c_P \,(T_2 - T_1)} = \frac{(T_{2R} - T_1)}{(T_2 - T_1)}. \tag{9}$$

3.2 Turbines

Turbines are an example of positive shaft work machines. They decrease both the temperature and pressure of the working fluid. This decrease creates a positive work transfer. This work transfer was described by Equation 5, and the turbine can be considered adiabatic for the same reasons as the compressor. Assuming that the turbine operates adiabatically and reversibly, analysis of the second law gives the following equation for the reversible turbine exit temperature

$$T_{4R} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} T_3. \tag{10}$$

Once the reversible outlet temperature is known, the shaft work transfer for the reversible operation of the turbine can be calculated (Equation 5). The turbine efficiency, η_T, is the ratio of the actual work transfer, $\dot{W}_{\text{act}}$, to the work transfer required for reversible operation, $\dot{W}_{\text{rev}}$. For an ideal gas the turbine efficiency becomes

$$\eta_T = \frac{\dot{W}_{\text{act}}}{\dot{W}_{\text{rev}}} = \frac{\dot{m}c_P\left(T_4 - T_3\right)}{\dot{m}c_P\left(T_{4R} - T_3\right)} = \frac{T_4 - T_3}{T_{4R} - T_3}. \tag{11}$$

3.3 Combustion Chambers

The combustion chamber is the component of the gas turbine in which the fuel is combined with the air from the compressor and burned. The combustion chamber functions like a heat exchanger and can be modeled as a constant pressure device. The combustion process raises the temperature of the air in the system by converting the chemical potential energy of the reactants to thermal energy. There is no work transfer involved in the reaction. Therefore, the first law of thermodynamics for steady flow operation of the combustion chamber becomes

$$\dot{Q} = \sum\left(\dot{m}h\right)_{\text{out}} - \sum\left(\dot{m}h\right)_{\text{in}}. \tag{12}$$

However when the fuel is burned inside the chamber the chemical reaction that takes place changes the state of the air and the fuel. It is necessary to analyze the first law for the combustion chamber in terms of the reactants and products of the combustion reaction. The first law for the combustion reaction becomes

$$\dot{Q} = \sum_{\text{prod}}\left(\dot{n}\bar{h}\right) - \sum_{\text{react}}\left(\dot{n}\bar{h}\right) \tag{13}$$

where $\dot{n}$ is the flow rate of the individual reactants and products in mols per second.

The enthalpies of each product and reactant are determined by the sum of the enthalpy of formation at standard temperature and pressure, $\bar{h}^\circ_{T,P}$, and the enthalpy required to raise the compound from the standard temperature of 25°C to the desired temperature, $\left(\bar{h}_{T,P} - \bar{h}_{25^\circ\text{C},1\text{atm}}\right)$.

$$\bar{h} = \bar{h}^\circ_{T,P} + \left(\bar{h}_{T,P} - \bar{h}_{25^\circ\text{C},1\text{atm}}\right) \tag{14}$$

For a gas turbine running on propane, the following reaction occurs in the combustion chamber:

$$C_3H_8 + 5O_2 + 5(3.76)N_2 \Rightarrow 3CO_2 + 4H_2O + 5(3.76)N_2 \tag{15}$$

Propane is mixed with stoichiometric air and burned to form carbon-dioxide and water. The nitrogen does not react with the other compounds, but it does exit the combustion chamber at a higher temperature than it entered. It is important to note that some oxygen may remain unburned and appears on the product side of the reaction. This means that the engine is running lean because there is a low ratio of fuel to air. This will be explained further in Section 7.

3.4 Measuring Cycle Efficiency

The efficiency of the Brayton cycle is generally measured according to two different ratios: the energy conversion efficiency, η_{cycle}, and the net work ratio, NWR. The energy conversion efficiency is characterized by the ratio of net work, $\dot{W}_{\text{net}}$ to the heat added, $\dot{Q}_{2-3}$. Combining Equations 3-5, η_{cycle} becomes

$$\eta_{\text{cycle}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{2-3}} = \frac{(h_1 - h_2) + (h_3 - h_4)}{h_3 - h_2} = 1 - \frac{h_4 - h_1}{h_4 - h_2} \tag{16}$$

$$\eta_{\text{cycle}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \tag{17}$$

Current designs for gas-turbine cycles have efficiencies of about 0.40. The net work ratio is the ratio of net work to the heat that is removed in order to return the turbine exhaust to its initial tempertaure and pressure of State 1.

$$\text{NWR} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{3-4}} = \frac{(h_1 - h_2) + (h_3 - h_4)}{h_3 - h_4} = 1 - \frac{h_2 - h_1}{h_3 - h_4} \tag{18}$$

$$\text{NWR} = 1 - \frac{T_2 - T_1}{T_3 - T_4} \tag{19}$$

Typical industry values for the net work ratio are around 0.25. This shows that three quarters of the work out of the turbine is used by the compressor. Therefore, if there is a 1% loss in the efficiency of the turbine represents, there will be a 4% loss in cycle efficiency.

4 Automobile Turbochargers

Turbochargers are devices commonly used in automobiles to increase power without a significant increase in vehicle weight. A turbocharger is comprised of a compressor and turbine operating on a common shaft. The compressor is located between the car engine's air filter and intake manifold, and it compresses the air flowing into the engine cylinders. This allows more air to be packed into the cylinder and more fuel to be burned. The exhaust air from the cylinders is fed through the turbine blades, spinning the turbine shaft. This in turn causes the compressor blades to spin and compress more air for the engine cylinders. A typical turbocharger is shown in Figure 4[10].

The performance of the turbocharger is heavily dependent on the design of the compressor and turbine housings as well as their blades. Current blade design involves the use of

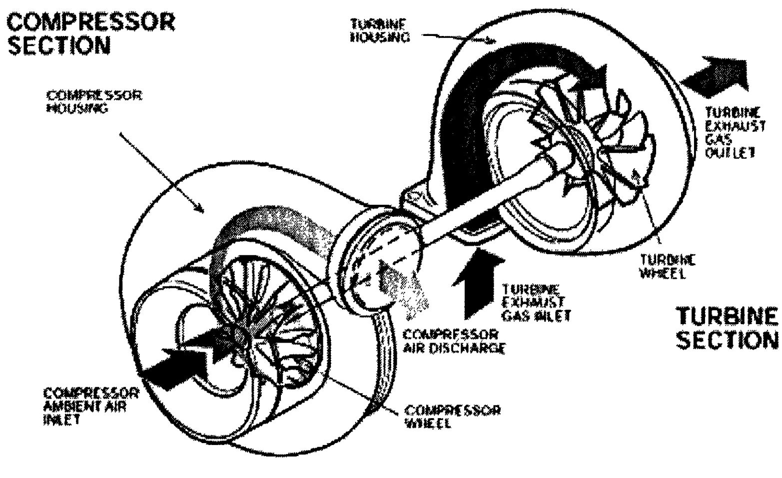

Figure 4: Typical Turbocharger Design[10]

computational fluid dynamics (CFD) to analyze the flow of air. The housings must be carefully designed so that there is a small gap between the rotor and the housing. This allows the rotor to spin freely. However, the gap must be sufficiently small so that the majority of the air flow is directed by the rotor and does not slip between the edge of the rotor and the housing. Bearing design is also important because the shaft can reach speeds of 150,000 rpm. This is almost 30 times faster than most car engines[10].

5 Engine Overview

The selection and design of the components of the gas turbine were limited by the requirement that the completed engine be usable as a classroom demonstration, and fabricated from available materials and tools. The engine needed to be portable so that all of the components fit together on a cart. Also, the engine needed to run on a flow rate of propane that corresponded to an acceptable maximum cycle temperature. The completed engine is shown in Figure 1.

The engine consists of a turbocharger (mounted to the top shelf of the cart) with a combustion chamber between its compressor outlet and its turbine inlet. The turbocharger is mounted so that air enters the compressor horizontally. The air exits the compressor vertically upwards where it is fed through piping to the combustor inlet. Fuel is fed from the propane tank up to the combustor inlet where the air is burned. The flow of propane is monitored by a valve and pressure gauges. From the combustion chamber, the exhaust passes through the turbine and then exits horizontally. Oil for the engine lubrication and cooling is pumped from the bottom of the oil reservoir up through an oil filter to the turbocharger and then back down to the reservoir.

6 Selection of the Turbocharger

An automobile turbocharger was used to operate as the compressor and turbine of the engine. The turbocharger used in the gas turbine was manufactured by Kühnle, Kopp & Kausch (3K), which is currently a division of BorgWarner Turbo Systems. The turbocharger was model number K26 and was originally installed in a 1985 Audi 200 5T. The turbocharger was purchased by Keane Nishimoto from an auto salvage dealership. It was selected based on the good condition of the turbine and compressor blades and the shaft bearings. The performance characteristics of both the compressor and the turbine were obtained from BorgWarner Turbo Systems (Figures 5 & 6).

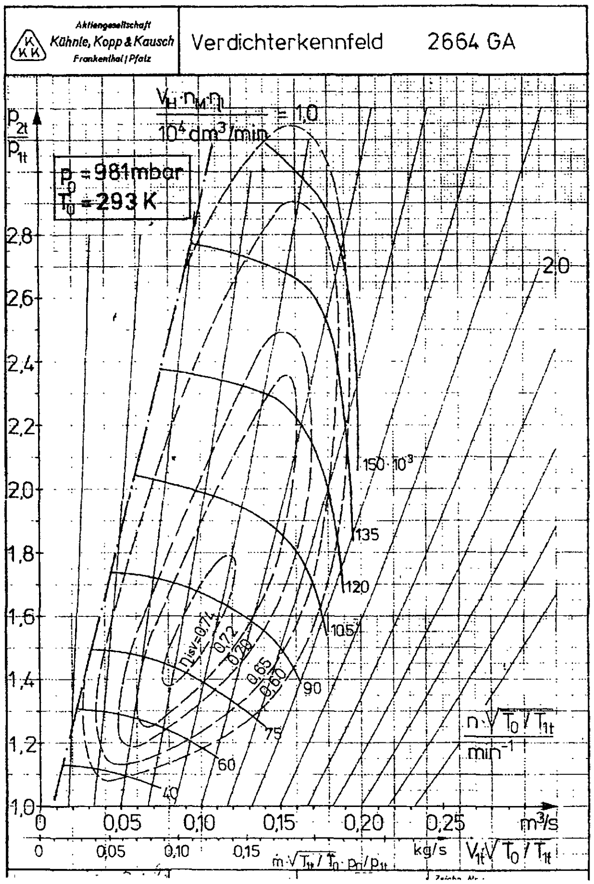

Figure 5: Compressor Map

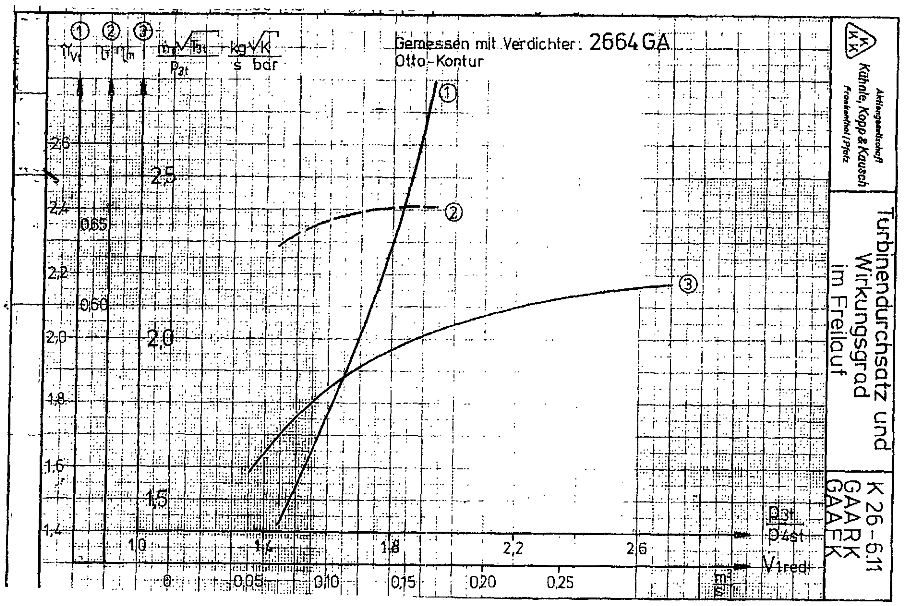

Figure 6: Turbine Map

7 Combustion Chamber Design

The combustion chamber was the most critical part of the design of this gas turbine. The chamber had to be designed so that the flame is self-sustaining and the temperature of the products of combustion is sufficiently below the maximum working temperature of the turbine.

To achieve these goals, the combustion chamber can be divided into two areas: the primary zone and the secondary zone. The primary zone is where the majority of the fuel combustion takes place. The fuel must be mixed with the correct amount of air so that a stoichiometric mixture is present. In the secondary zone, unburned air is mixed with the combustion products to cool the mixture before it enters the turbine (Figure 7).

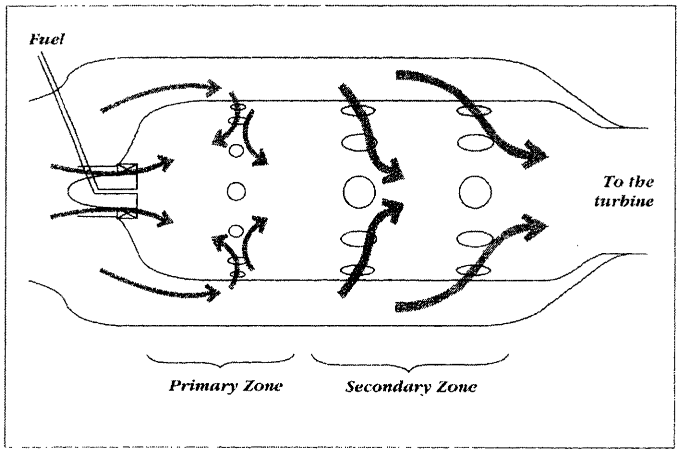

Figure 7: Combustion Zones[7]

The combustion chamber was constructed using a concentric tube design (Figure 8). It was modeled after the design of several different gas-turbines that had been constructed from turbochargers [5, 2]. The outer shell of the combustion chamber is comprised of a 4 in. diameter pipe. The inner shell, the flame tube, is made of 22-gauge perforated steel. Air flow from the compressor is fed through the flame tube as well as around its sides. The air in the flame tube is burned with the propane, while the air on the outside of the flame tube is used to cool the products of combustion before they enter the turbine. To determine the proportion of the air flow from the compressor that needs to be burned in the flame tube, the combustion reaction was analyzed. This analysis is described in the following section.

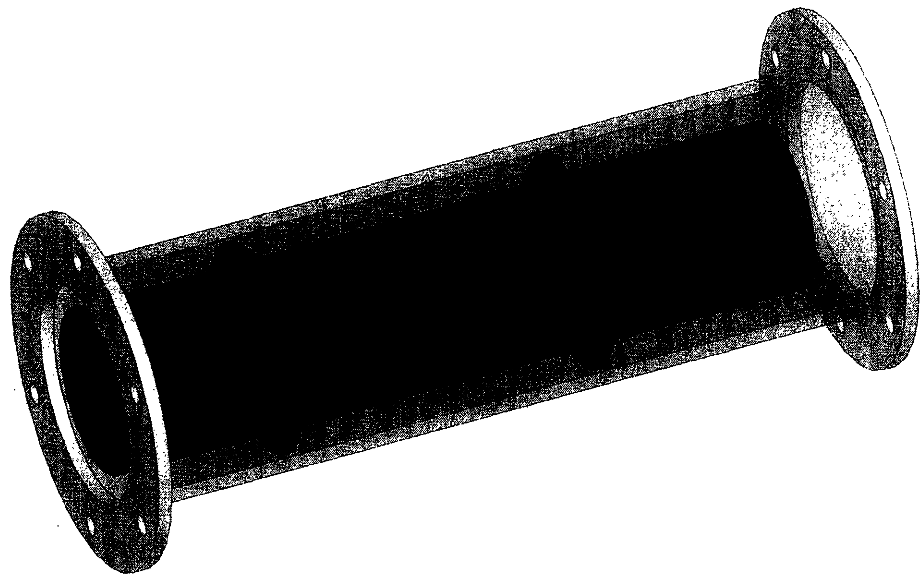

Figure 8: Concentric Shell Design

7.1 Analysis of Combustion Reaction

The amount of air and propane that needs to be burned in the combustion chamber was calculated by analyzing the combustion reaction. First, the air flow through the compressor needed to be estimated. From the BorgWarner performance diagram for the compressor (Figure 5), the operating point of the compressor was taken to be at a corrected mass flow rate of 0.125 kg/sec with a pressure ratio of 2 and an efficiency, η_C =0.72 (Equations 20, 21, and 22).

$$\dot{m}\sqrt{\frac{T_1}{T_0}\frac{P_0}{P_1}} = 0.125\frac{\text{kg}}{\text{sec}} \tag{20}$$

$$\frac{P_2}{P_1} = 2.0 \tag{21}$$

$$\eta_C = 0.72 \tag{22}$$

Atmospheric air enters the compressor at a temperature, T_1 =298 K, and a pressure $P_1 = 10^5$ N/m^2. Combining these values with Equation 20 gives a mass flow rate of air in the compressor, $\dot{m}_{air}$, of 0.918 mol Air/sec.

$$\dot{m}_{air} = \left(\frac{0.126\text{kg}}{\text{sec}}\right)\left(\frac{\text{molAir}}{0.1373\text{kg}}\right) = 0.918\frac{\text{molAir}}{\text{sec}} \tag{23}$$

Modeling the air in the compressor as an ideal gas and assuming that the flow of air through the compressor is reversible, Equation 8 becomes

$$T_{2R} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} T_1 \Longrightarrow T_{2R} = 363.3\text{K}. \tag{24}$$

Using Equation 9, the actual outlet temperature of the compressor, T_2, was estimated to be 388.64 K.

$$\eta_C = \frac{(T_{2R} - T_1)}{(T_2 - T_1)} \longrightarrow T_2 = 388.64\text{K} \tag{25}$$

This temperature represents the temperature of the air at the outlet of the compressor, which is also the temperature of the air on the reactant side of the combustion reaction. The reaction needs to be controlled so that the temperature of the air entering the turbine is within its working range. The maximum operating temperature for the turbine was specified at 1223 K for continuous operation or 1248 K temporarily.

From the performance diagram of the turbine (Figure 6), the desired operating point for the turbine was chosen to be at an outlet temperature, T_4, of 700 K, a pressure ratio of 2, and an efficiency, η_T, of 0.70. From this data the inlet temperature of air to the turbine, T_3, can be calculated by rearranging Equations 10 and 11.

$$\eta_T = \frac{T_4 - T_3}{\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} T_3 - T_3} \tag{26}$$

$$\longrightarrow T_3 = 800.7\text{K} \tag{27}$$

Having determined the desired inlet and outlet temperatures of the combustion chamber and the flow rate of air through the turbine, the flow rate of propane required to reach the turbine inlet temperature can be calculated from the combustion reaction.

As described in Section 3.3 the reaction in the combustion chamber is

$$\mathrm{C_3H_8 + 5O_2 + 5(3.76)N_2 \Rightarrow 3CO_2 + 4H_2O + 5(3.76)N_2} \tag{28}$$

Equation 28 assumes that there is a stoichiometric ratio of air to propane. However, for design purposes some of the air from the compressor is used to cool the products of combustion. Therefore all of the propane burns, but some of the air does not. Letting $\dot{x}$ represent the molar flow rate of propane

$$\dot{x} = \frac{\mathrm{molC_3H_8}}{\mathrm{sec}}, \tag{29}$$

and letting $\dot{y}$ represent the molar flow rate of air

$$\dot{y} = \frac{\mathrm{molAir}}{\mathrm{sec}}, \tag{30}$$

and assuming that there is no heat transfer out of the combustion chamber, the first law for the combustion reaction (Equation 13) becomes

$$\dot{x}\bar{h}_{\mathrm{C_3H_8}} + \dot{y}\bar{h}_{\mathrm{O_2}} + 3.76\dot{y}\bar{h}_{\mathrm{N_2}} = 3\dot{x}\bar{h}_{\mathrm{CO_2}} + 4\dot{x}\bar{h}_{\mathrm{H_2O}} + 3.76\dot{y}\bar{h}^{\star}_{\mathrm{N_2}} + (\dot{y} - 5\dot{x})\,\bar{h}^{\star}_{\mathrm{O_2}}. \tag{31}$$

The assumption that there is no heat transfer out of the combustion chamber will produce the maximum temperature of the products given the temperature of the reactants. When there is no heat transfer all the excess chemical potential energy is used to heat the products of combustion to a higher temperature. Therefore this will overestimate the temperature of the products of combustion.

Solving Equation 31, the molar flow rate of propane is 0.028 $\frac{\mathrm{molC_3H_8}}{\mathrm{sec}}$. This corresponds to a mass flow rate of propane, $\dot{m}_{\mathrm{C_3H_8}}$, of $0.001232\frac{\mathrm{kg}}{\mathrm{sec}}$. To stoichiometrically balance the air with the propane, the mass of burned air per second must be 0.019 $\frac{\mathrm{kg}}{\mathrm{sec}}$. Therefore the percentage of air flow from the compressor that must be burned is

$$\frac{\dot{m}_{\mathrm{burnedair}}}{\dot{m}_{\mathrm{air}}} = 0.153 \tag{32}$$

This means that 15.3% of that air flow should be directed through the flame tube section of the combustion chamber. Figure 9 shows the cross-section of the combustion chamber at the point where the flow from the compressor is divided. At the left edge of the flow plate, the cross-sectional area through the flame tube is 15.3% of the total cross-section that the air is flowing through. The flow plate is the circular steel plate through which the air flows into the flame tube. As the air passes through the center of the flow plate the diameter of the hole expands. This serves two purposes. First, it slows down the velocity of the air. This helps to prevent the flame from going out. Second, the larger diameter directs the flow across the fuel hook so that the air passes over the holes through which the propane is flowing. The fuel hook is described in the following section.

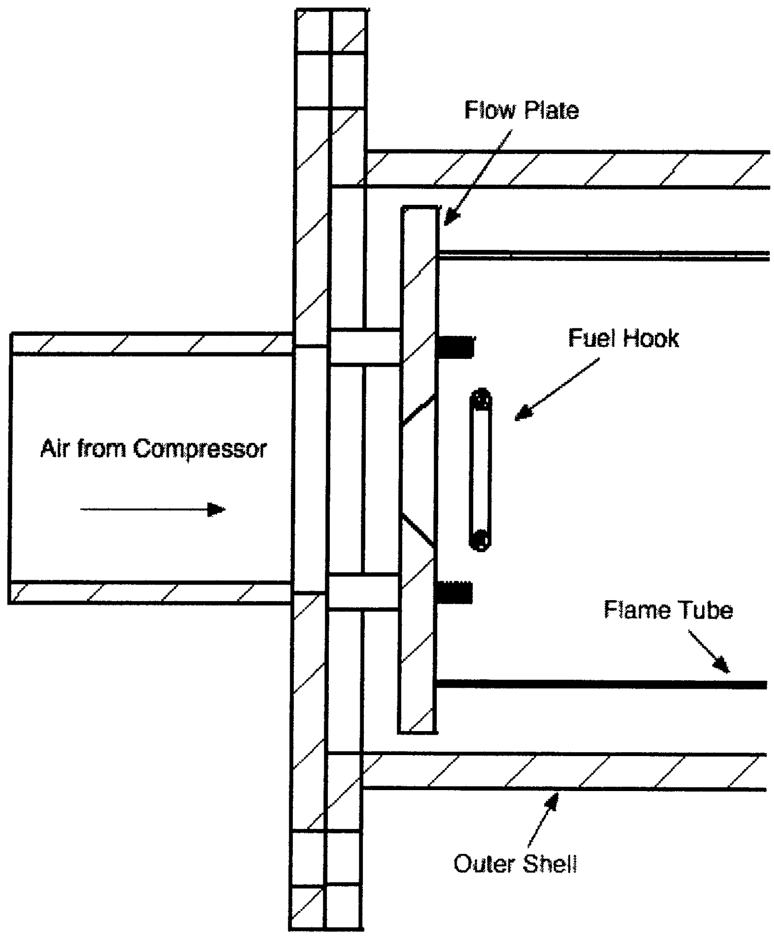

Figure 9: Cross-Section of the Combustor as Air Enters from the Compressor

7.2 Fuel Flow into the Combustion Chamber

In the previous section it was determined that the desired flow of propane through the system was 0.001232 $\frac{\text{kg}}{\text{sec}}$. To monitor this flow rate, the flow of propane from the propane tank to the combustion chamber needed to be regulated. Figure 10 illustrates the components used to

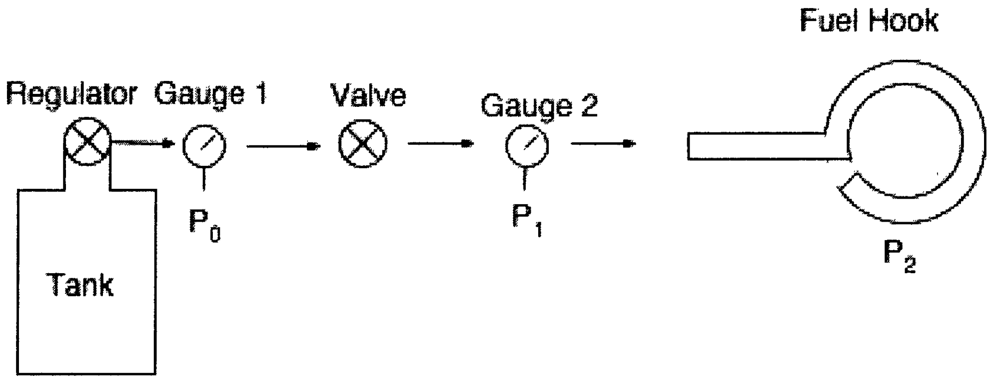

Figure 10: Fuel System Components

control the propane flow. Attached to the propane tank is a propane regulator. The regulator chosen was a TurboTorch Model R-LP Regulator designed to control a propane torch. The regulator can supply propane at pressures ranging from 0 to 60 psi (0 to 4.1×10^5 Pa). The first gauge monitors the pressure of the propane flowing out of the regulator. Downstream of the first gauge, the propane passes through a needle valve which allows the flow to be more delicately controlled. The pressure at the outlet of the valve is measured by the second pressure gauge. From there, the propane flows through the fuel hook. The fuel hook itself is 1/8 in. steel tubing, bent into a hook, with holes drilled around its circumference (Figure 11). The end of the hook is sealed shut. Propane flows out of the holes in the hook to create small flames in a circular pattern. This design was inspired by the design for the standard gas stove burner, as well as a similar gas turbine built in the United Kingdom[2].

The pressure regulator and valve must be adjusted according to the size of the holes in the fuel hook. The flow of the propane through the fuel hook can be treated as an orifice flow plate. For an orifice, the volumetric flow rate is

$$\dot{V}_{\text{actual}} = C_d A_t \sqrt{\frac{2\left(P_{h1} - P_{h2}\right)}{\rho\left[1 - \left(\frac{A_t}{A_{1h}}\right)^2\right]}}. \tag{33}$$

The volumetric flow rate, $\dot{V}_{\text{actual}}$, is calculated from the desired mass flow rate. The pressure at point 2, P_2, is the pressure inside the combustion chamber which is 2×10^5 Pa. C_d is the flow coefficient which is calculated by an appropriate correlation, and A_t is the area at the

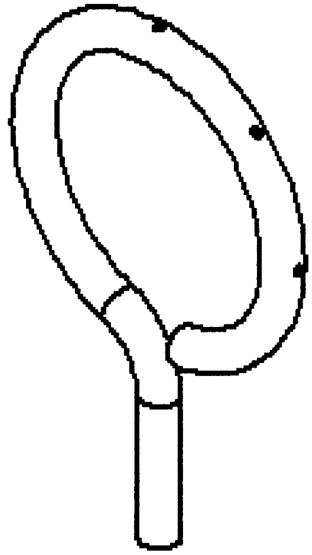

Figure 11: Fuel Hook

outlet of the fuel hook. Because the supply of propane is continuous, Point 1 can be treated as a reservoir. Therefore

$$A_1 \longrightarrow \infty. \tag{34}$$

Equation 33 becomes

$$\dot{V}_{\text{actual}} = C_d A_t \sqrt{\frac{2\,(P_1 - P_2)}{\rho}}. \tag{35}$$

The flow coefficient, C_d, in the case where A_1 is infinite is 0.5961.

Equation 33 can be rearranged to calculate the pressure at the inlet to the fuel hook as a function of outlet area, A_t.

$$P_1 = \left(\frac{\dot{V}_{\text{actual}}}{C_d A_t}\right)^2 \frac{\rho}{2} + P_2 \tag{36}$$

By manipulating the area of the holes and adjusting the pressure regulator, the desired valve characteristics can be determined. The flow through the valve is

$$\dot{V}(\text{GPM}) = C_v \sqrt{\frac{(P_0 - P_1)\,(\text{psi})}{\gamma}}, \tag{37}$$

where C_v is the flow coefficient through the valve, P_0 is the pressure upstream of the valve, and γ is the specific gravity of the working fluid (propane). By varying the value of the pressure controlled by the pressure regulator and the total area of the holes in the fuel hook, it was determined that the necessary values for the valve flow coefficient ranged from 0.02 to 0.18. A valve with a maximum flow coefficient of 0.42 was chosen.

Six holes were drilled around the fuel hook using a size 1/16 in. drill. If the pressure regulator is set to 30 psi, then the desired valve flow coefficient is 0.175, which corresponds to a pressure downstream of the valve of 200,003 Pa. Therefore the desired pressure at Gauge 1 is 30 psi and at Gauge 2 is 200,003 Pa.

7.3 Flange Design

The combustion chamber was connected to the compressor and turbine via flanges on either end. The ASME Pressure Vessel Code provides an analysis to determine the required bolt load for a flange and how many bolts are necessary for a given set of conditions. There are two conditions that the flange bolts must meet. First, the bolt load for normal operating conditions, W_{m1}, must be greater than the sum of the hydrostatic end force, H, exerted from the maximum allowable working pressure and the compression load, H_p, needed to maintain a sufficiently tight joint with the gasket.

$$W_{m1} = H + H_P = 0.785G^2 + (2b \times 3.14GmP) \tag{38}$$

G is the mean diameter of the gasket contact face, b is the effective gasket seating width, m is a function of the gasket material and construction, and P is the internal design pressure. Choosing a conservative gasket factor for the flange on the combustor shell,

$$W_{m1} = 1540\,\text{lb}. \tag{39}$$

The other bolt load requirement is that the gasket must be seated properly by applying a minimum initial load under atmospheric temperature and pressure. This initial load, W_{m2}, is a function of the gasket material and the seating area. The minimum initial bolt load is

$$W_{m2} = 3.14bGy \tag{40}$$

where y is the gasket unit seating load in psi.

The total cross-sectional area of the bolts, A_m, required for the flange can be calculated from the bolt loads and the allowable bolt stress value, S, which can be determined by the bolt material. The total area is the greater of A_{m1} and A_{m2} where

$$A_{m1} = \frac{W_{m1}}{S}, \tag{41}$$

and

$$A_{m2} = \frac{W_{m2}}{S}. \tag{42}$$

In the case of the combustor flange, the operating pressure is relatively low and the initial load is dominant. Therefore A_{m2} is larger than A_{m1}.

$$A_{m1} = 0.06\text{in.}^2, A_{m2} = 0.58\,\text{in.}^2 \tag{43}$$

Six 3/8-in. bolts were sufficient to secure the combustor flange. This previous analysis was carried out for the combustor. The combustor exit is the point in the cycle that exhibits the maximum temperature and pressure. Therefore it will require the greatest required bolt area. This bolt area will be sufficient for all other points in the cycle and was used in the design of other flanges on the engine even though it was greater than the required area.

7.4 Stress Analysis

The stress in the combustion chamber was evaluated to ensure that the pressure inside the combustion chamber would not result in a failure of the engine. As discussed in Section 3.3 the combustion chamber operates as a constant pressure device. The operating pressure of the chamber was estimated to be 2×10^5 Pa. To connect the combustion chamber between the compressor and the turbine of the turbocharger, it was necessary to weld quarter-inch steel plates to either end of the shell to create flanges. Roark's *Formulas for Stress and Strain*[13] gives the following analysis for a flanged and bolted pipe submitted to a uniform internal pressure P psi, and a longitudinal tension F lb.

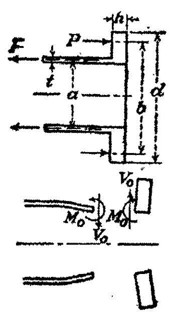

Figure 12: Flanged and bolted pipe with a uniform internal pressure p psi, and a longitudinal tension P lb.[13]

The shear stress, V_0, in the pipe is

$$V_0 = \frac{\left(f^2 - \frac{h^3}{2t}\tau_1\right)(t + 0.2325f\tau_1)P - 2T_2(h + 0.5377f)F}{1.860ft + \tau_1\left[h^2\left(2 + 0.1160\frac{f}{t}\tau_1\right) + 1.6103fh + 0.866f^2\right]} = -505.3\,\text{lb}. \tag{44}$$

and the bending moment, M_0, is

$$M_0 = \frac{(h^2\tau_1 + 1.86ft)V_0 + h\tau_2F - 0.5tP\left(f^2 - \frac{h^3}{2t}\tau_1\right)}{1.5\tau_1h - 3.464t} = 305.7\,\text{lb}. \tag{45}$$

where

$$f = \sqrt{at}; \quad \tau_1 = \frac{t^3(3a^2 + 5d^2)}{h^3(d^2 - a^2)}; \quad \tau_2 = \frac{3.58t^3}{h^3(d^2 - a^2)}\left[\frac{d^2}{3}\log_e\frac{b}{a} + 0.1\left(b^2 - a^2\right)\right]. \tag{46}$$

After determining the shear stress and bending moment, the stresses in the cylinder and flange can be calculated. The longitudinal bending stress in the cylinder, s'_{c1}, is

$$s'_{c1} = \frac{6M_0}{t^2} = 29,250\,\text{psi}. \tag{47}$$

The radial bending stress in the flange, s'_{f1}, is

$$s'_{f1} = \frac{6}{h^2}\left(M_0 - \frac{1}{2}V_0 h\right) = 35,410\,\text{psi}. \tag{48}$$

The longitudinal direct stress in the cylinder, s_{c1}, is

$$s_{c1} = \frac{F + P\pi\left(\frac{1}{2}a - \frac{1}{2}t\right)^2}{\pi a t} = 715.5\,\text{psi}. \tag{49}$$

The radial direct stress in the flange, s_{f1}, is

$$s_{f1} = \frac{V_0}{h} + P = -2000\,\text{psi}. \tag{50}$$

The maximum longitudinal stress in the cylinder, $s_{c,max}$, is the sum of the tension at the outer surface, s'_{c1} and at the junction with the flange, s_{c1}.

$$s_{c,max} = s'_{c1} + s_{c1} = 30,000\,\text{psi}. \tag{51}$$

The tangential bending stress in the flange is

$$\begin{aligned} s'_2 &= s'_{c1} + \frac{0.80}{h^2\left(d^2 - a^2\right)}\left[d^2\left(-15M_0 + 7.5hV_0 + 1.492F\ln\frac{b}{a}\right) + 0.4475F\left(b^2 - a^2\right)\right] && (52) \\ &= -124,400\,\text{psi}. && (53) \\ & && (54) \end{aligned}$$

The tangential hoop stress in the flange is

$$s_2 = \frac{h^2}{4t^3}\tau_1\left(V_0 + hP\right) = -5,500\,\text{psi}. \tag{55}$$

The maximum radial stress in the flange, which is the compression at the outer face at the junction with the cylinder, is

$$s'_{f1} + s_{f1} = 33,410\,\text{psi}. \tag{56}$$

The maximum tangential stress in the flange, which is the tension at the inner face at the junction with the cylinder, is

$$s'_2 + s_2 = -130,000\,\text{psi}. \tag{57}$$

The maximum tensile stress at any point in the combustor is the radial bending stress in the flange, which has a value of 35,000 psi. The yield strength of steel is 40,000 psi. Therefore the stress only reaches 87.5% of the yield stress.

8 Auxiliary Systems

8.1 Lubrication and Cooling System

During normal operation of a turbocharger in an automobile, the turbocharger requires oil to cool the turbine as well as provide lubrication for its drive shaft. The 3K turbocharger requires oil cooling, and the hydrodynamic bearings of the compressor and turbine shaft ride on a thin film of oil. An oil system was developed to ensure the smooth operation of the turbocharger. The oil starts in a two-gallon reservoir where it is drawn through an oil pump, passes through an inline filter, and then through the turbocharger, before returning to the reservoir. The pump used was a Melling Model M-68 which is used on a Ford 302 truck engine (Figure 13). BorgWarner Turbo Systems specifies that oil must be run through

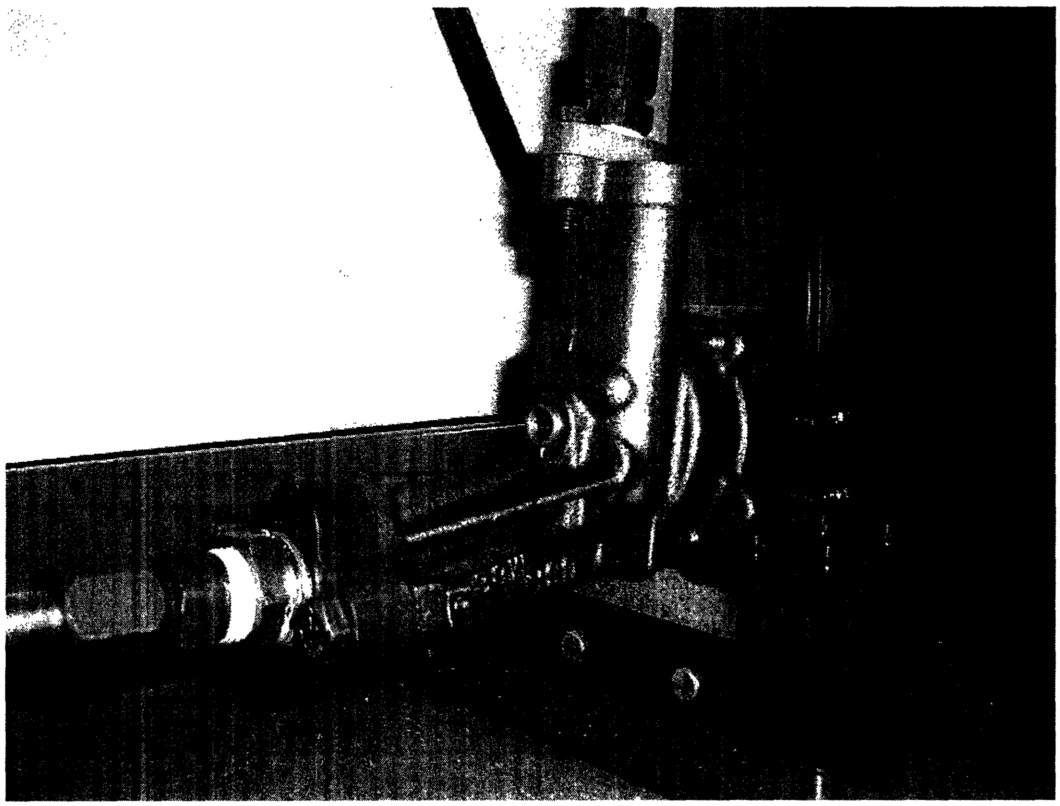

Figure 13: Melling Model M-68 Oil Pump

the turbocharger at a flow rate between 60 and 80 ounces per minute. The normal running pressure should be between 40–50 psi with a low limit of 35 psi and a high limit of 60 psi. The pump is powered by a Black & Decker cordless screwdriver via a 0.25-in. hex shaft. The oil filter used was an Arrow Pneumatics 3/8 in. inline Viton filter, model 9053V. The oil pressure was measured at the inlet to the turbocharger using an Ashcroft 1.5 in. oil gauge with a pressure range from 0–60 psi.

8.2 Ignition System

The ignition system of the gas-turbine was adapted from the push button ignitor of a Char-broil gas barbecue grill. The combustor of the engine is ignited by the push button on the generator (Figure 14). The generator is powered by an internal piezoelectric crystal. When

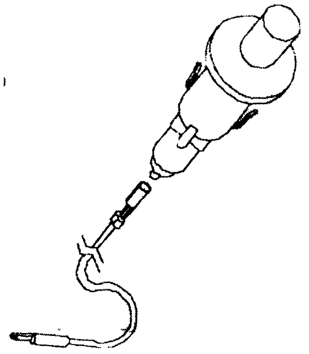

Figure 14: Generator

the button is depressed, pressure is applied to the crystal by a spring-loaded hammer. This creates a large charge separation within the crystal and a high voltage across it[15]. This voltage is carried through the main burner extension wire to the electrode that extends into the combustion chamber. The electrode passes through a ceramic cylinder which insulates it from the steel wall of the combustion chamber. The tip of the electrode is positioned 0.1825 in. from one of the holes in the fuel hook.

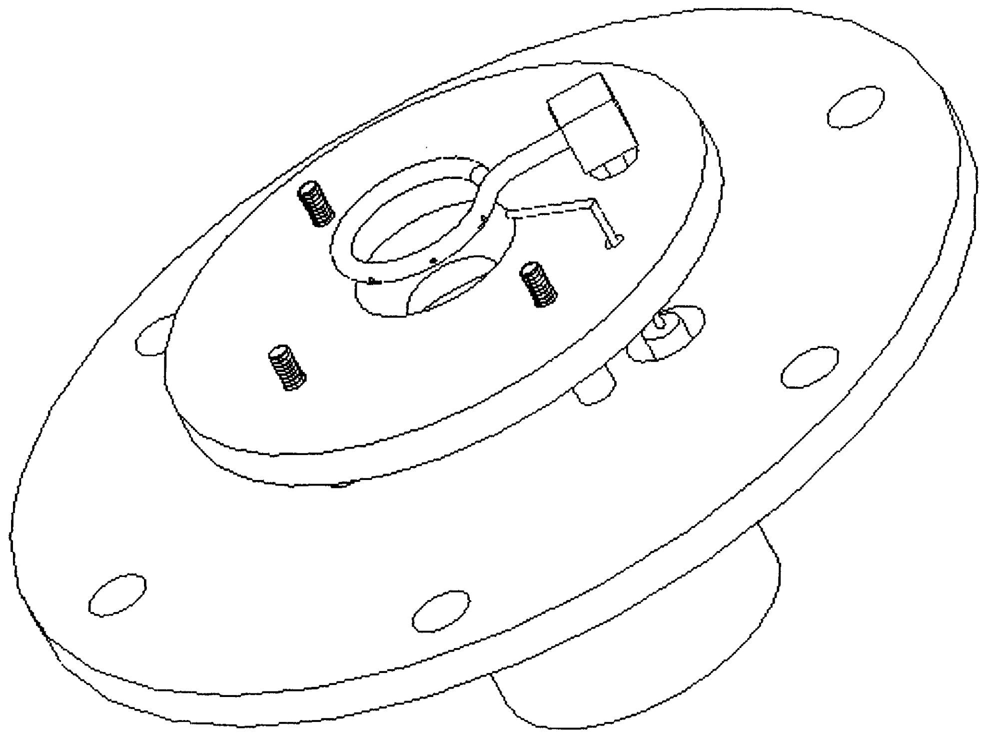

Figure 15: Combustion Chamber Inlet

8.3 Pressure and Temperature Measurement

The gas-turbine was equipped with temperature and pressure sensors to allow students to monitor the state of the air at each point of the cycle. For this purpose three thermocouples were used to monitor the air temperature. The first thermocouple was placed between the compressor outlet and the inlet of the combustion chamber and is used to measure T_2. The second thermocouple was placed between the outlet of the combustion chamber and the inlet of the turbine and is used to measure T_3. The last thermocouple is used to measure T_4 and was placed on the turbine exhaust. The thermocouples were manufactured by Omega. The temperature indicators are model DP460-T. They have a range of -999 to 9,999 °C with a 0.1 °C auto-resolution and an accuracy of 0.5 °C. The thermocouple wires are K type with a maximum temperature of 850°C.

Pressure was measured at three different points in the cycle: the outlet of the compressor, the inlet of the turbine, and the exhaust from the turbine. The gauges have a range of 0–100 psi and display pressure in both psi and Pa. The pressure gauges have a 3/8 in. male NPT connection.

9 Status of the Project & Recommendations

As of now, all components for the combustion chamber have been completed and assembled. The combustion chamber has been mounted between the compressor and the turbine of the turbocharger and connected by flanges. The combustion chamber has been connected to the propane tank with the valve, regulator, and pressure gauges installed. The oil system has also been completed. All of these components have also been mounted to the cart.

Connection of the thermocouples and pressure gauges for monitoring the air as it passes through the gas turbine remains to be completed. The temperature indicators have been purchased but the thermocouple wire needs to be soldered to the correct locations. The pressure gauges have also been purchased and need to be connected. However, the pressure gauges are not rated for the temperature that the air will reach. Therefore, capillary tubing will have to be used to reduce the temperature of the air that the pressure gauge is measuring. A model for this setup must be developed. To complete the construction of the engine, all of the displays for the thermocouples and pressure gauges should be mounted in a display panel so that students can view all of the relevant information in one location. The pushbutton for the igniter could also be mounted to this panel. Once all of these components have been integrated, the gas-turbine will be complete.

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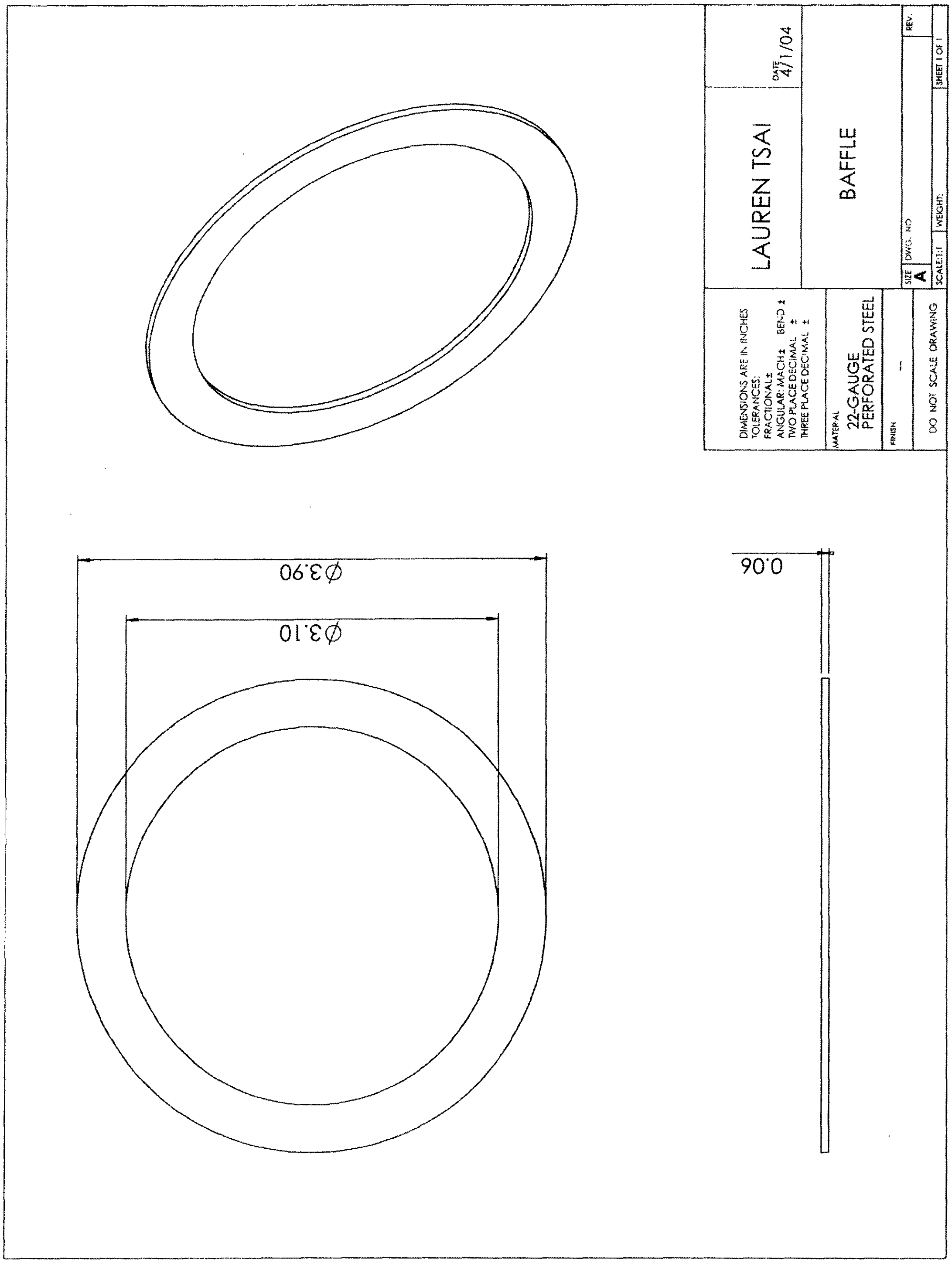

Figure 16: Baffle Part Drawing

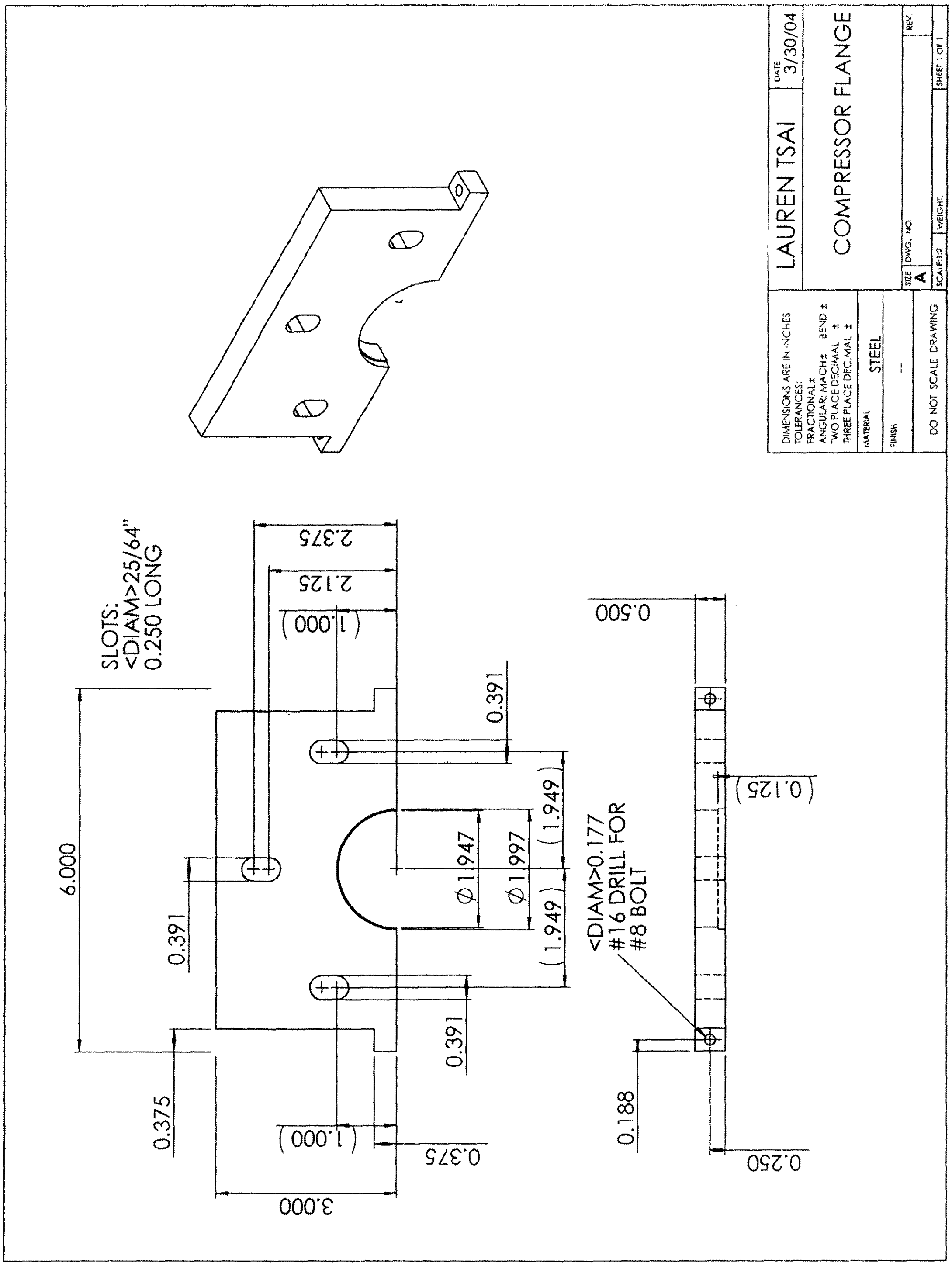

Figure 17: Compressor Flange Part Drawing

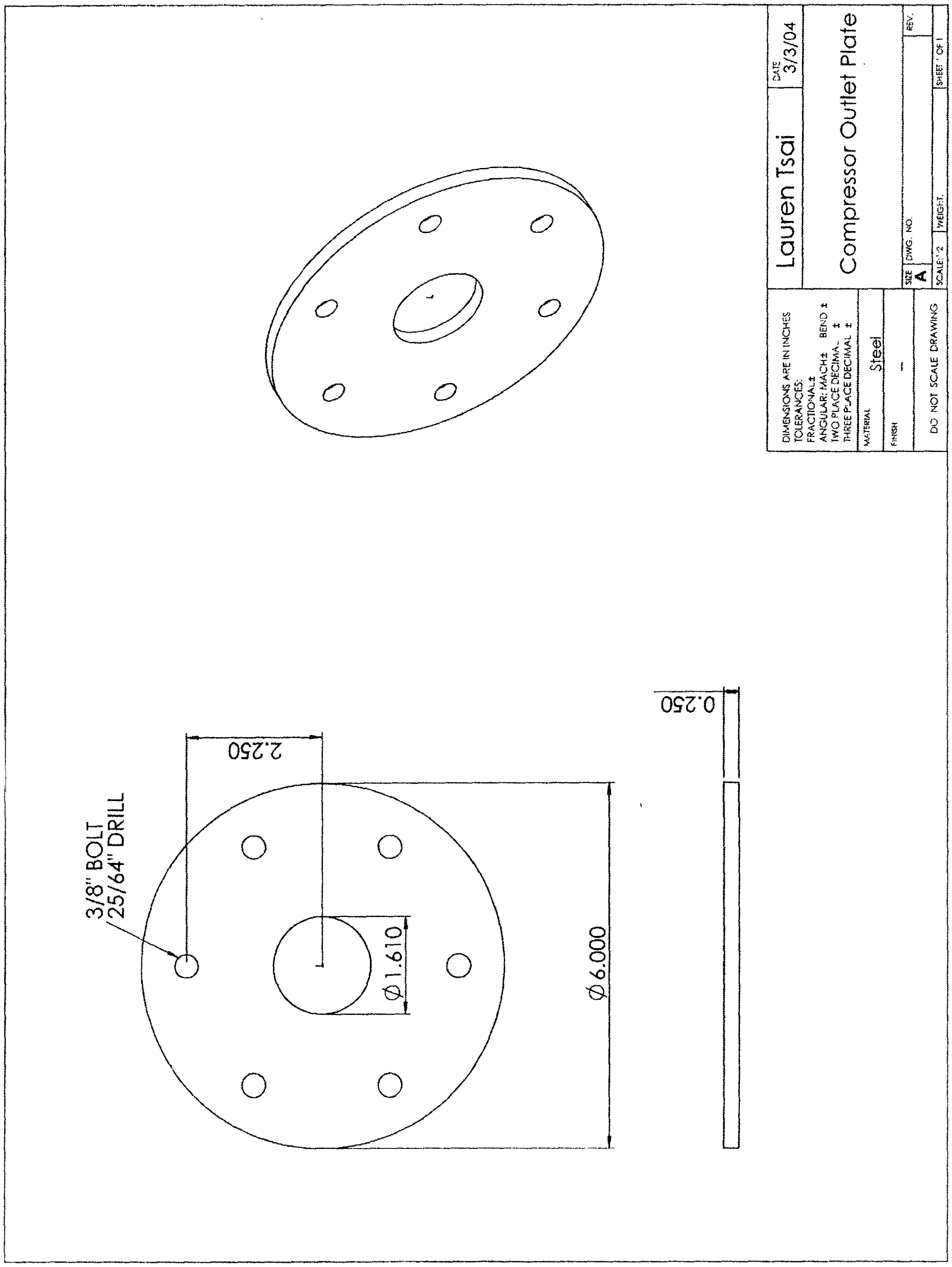

Figure 18: Compressor Outlet Plate Part Drawing

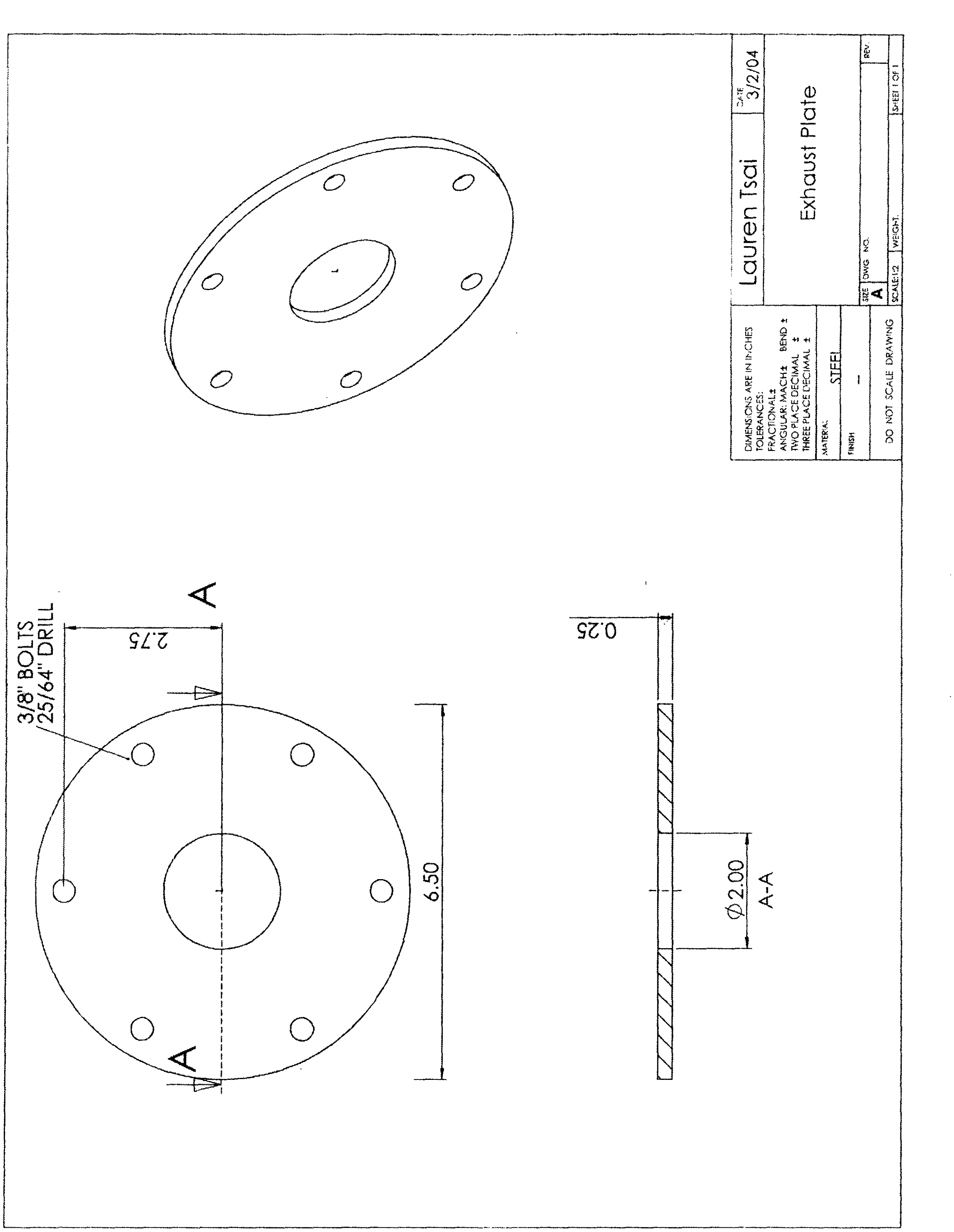

Figure 19: Exhaust Plate Part Drawing

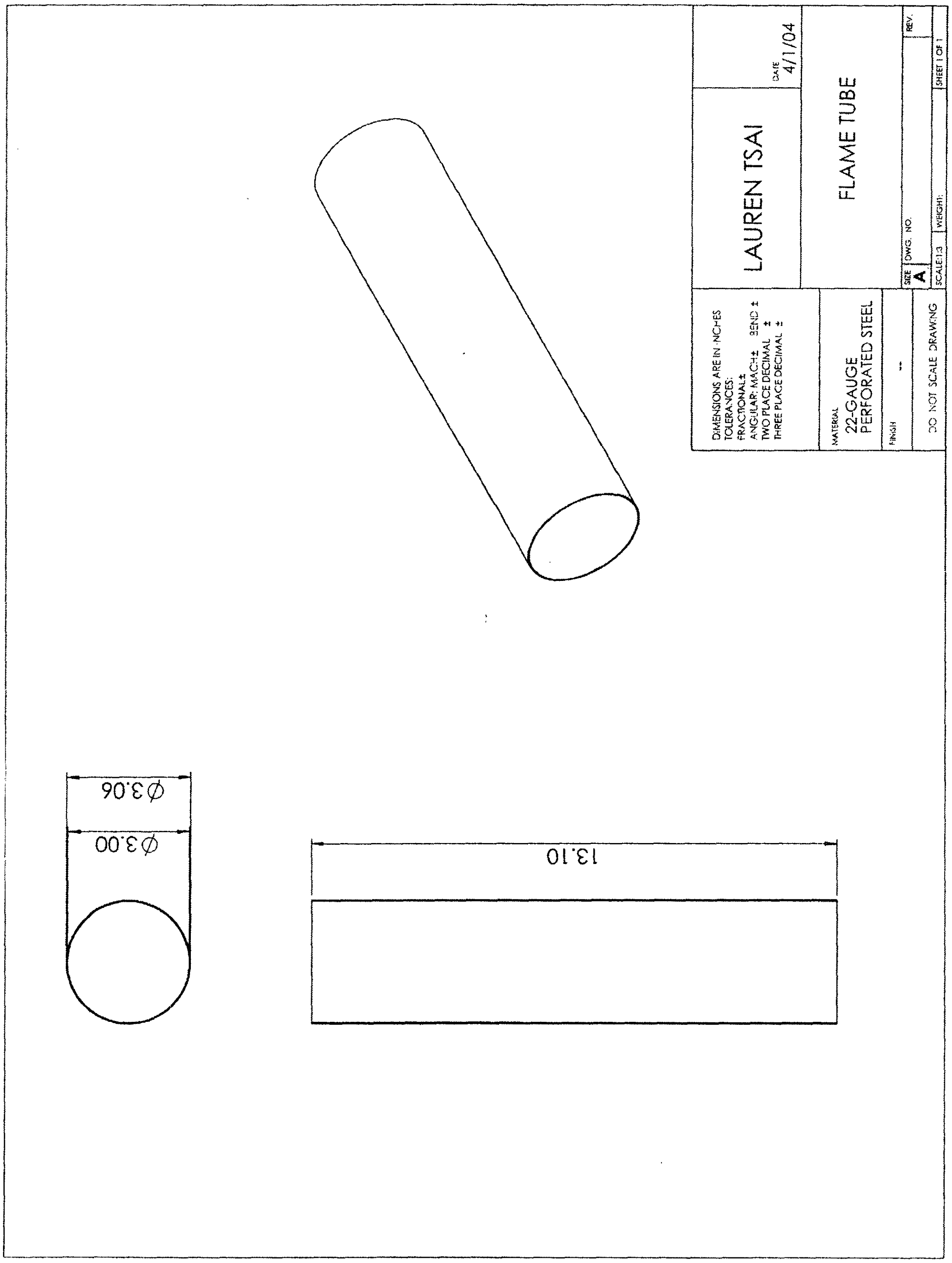

Figure 20: Flame Tube Part Drawing

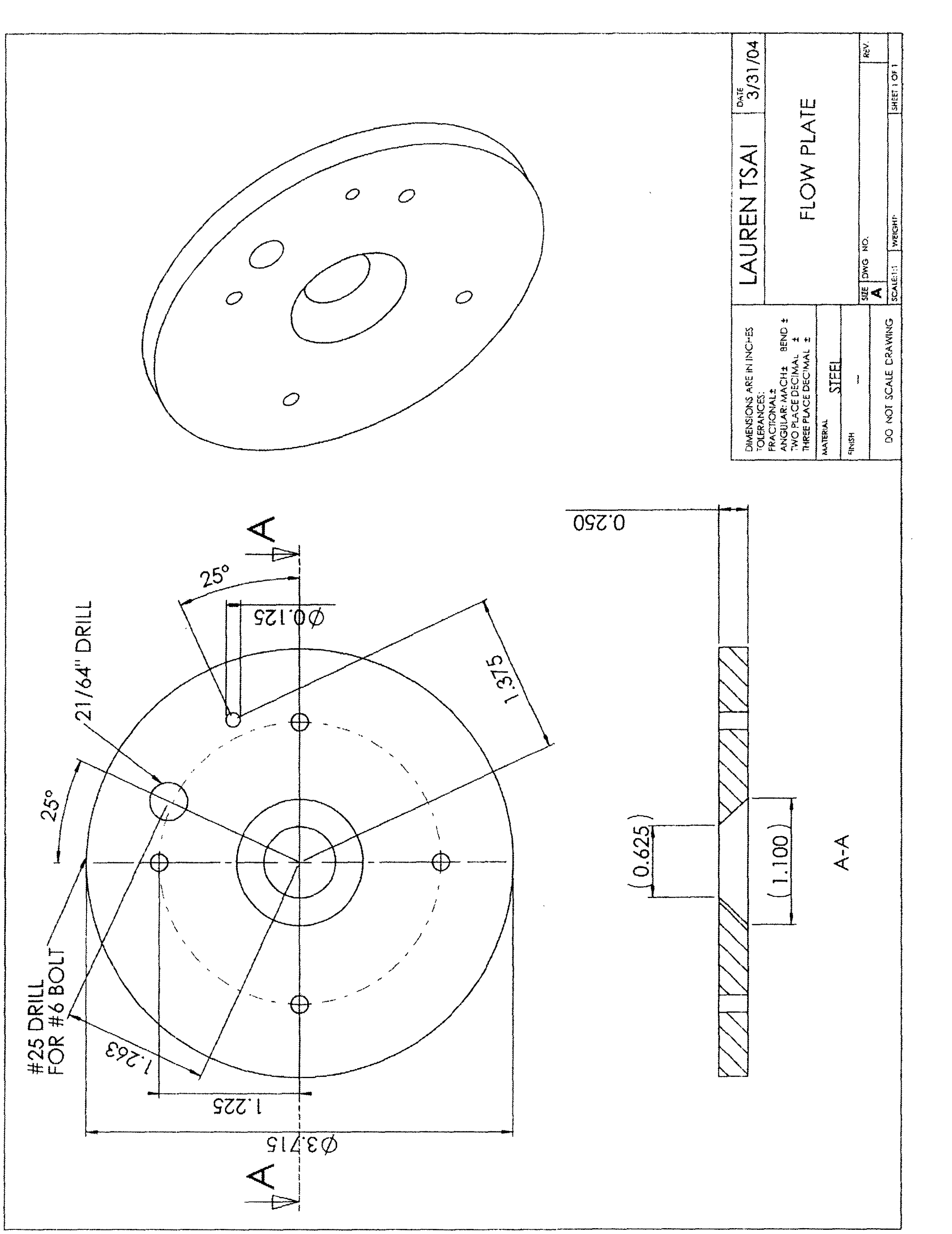

Figure 21: Flow Plate Part Drawing

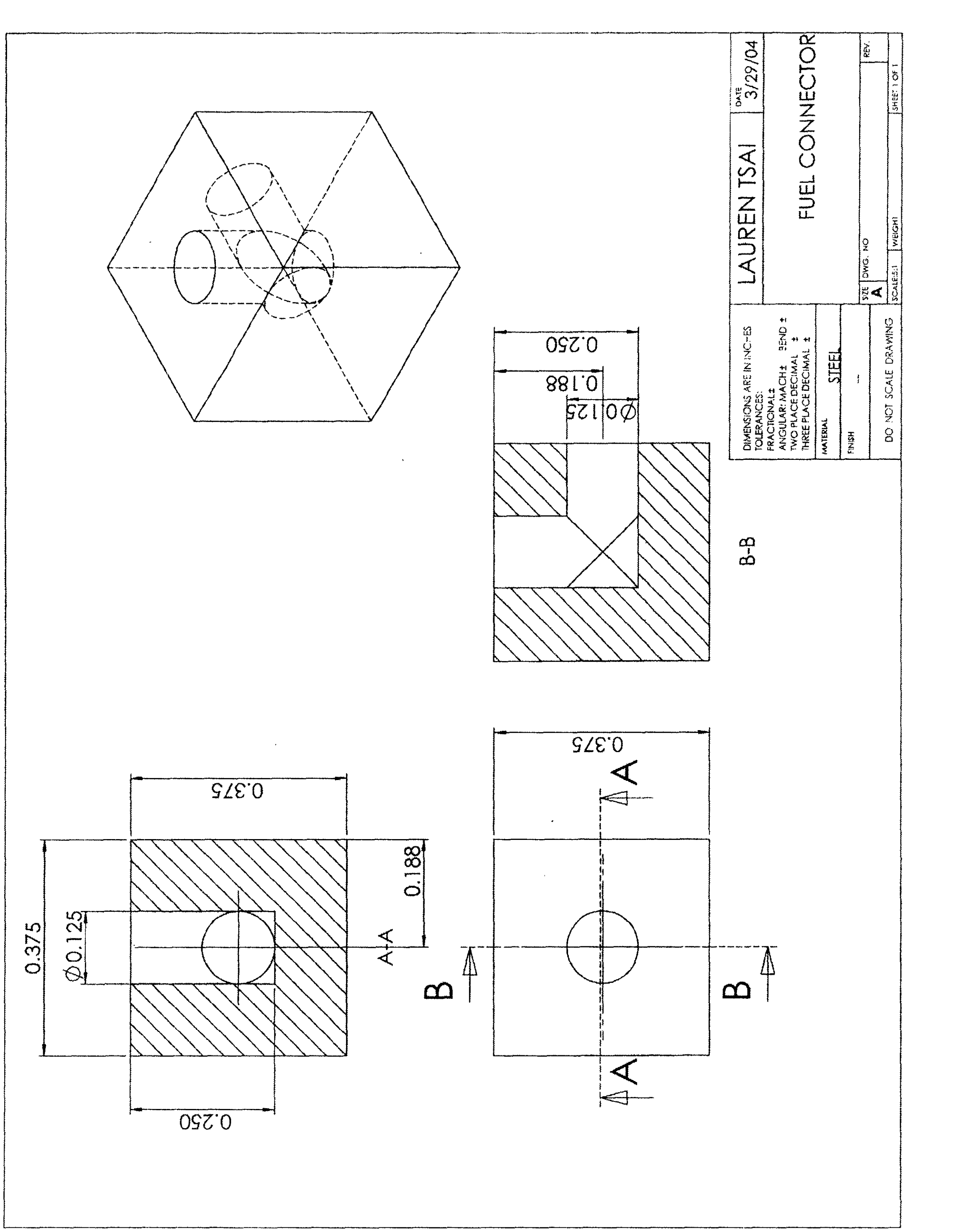

Figure 22: Fuel Connector Part Drawing

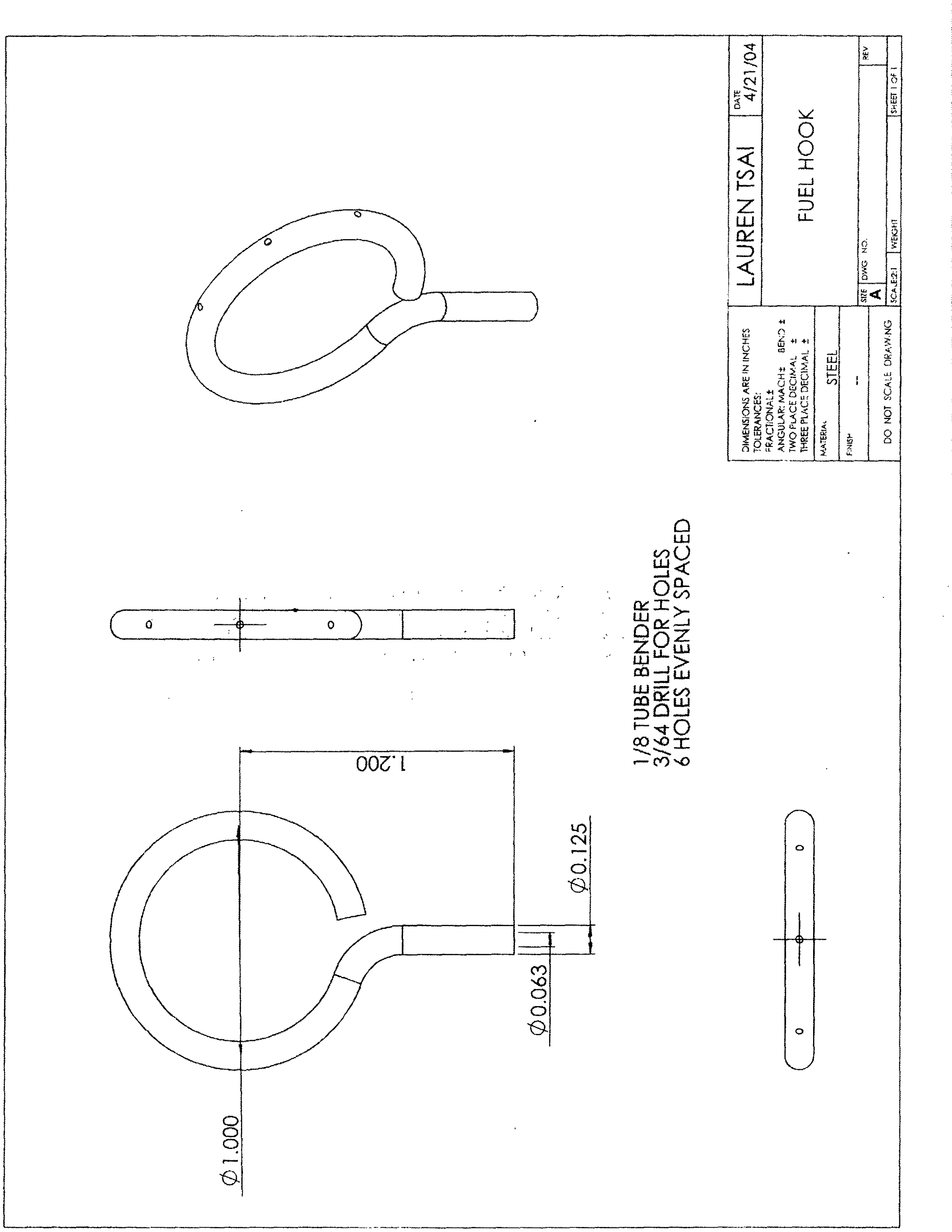

Figure 23: Fuel Hook Part Drawing

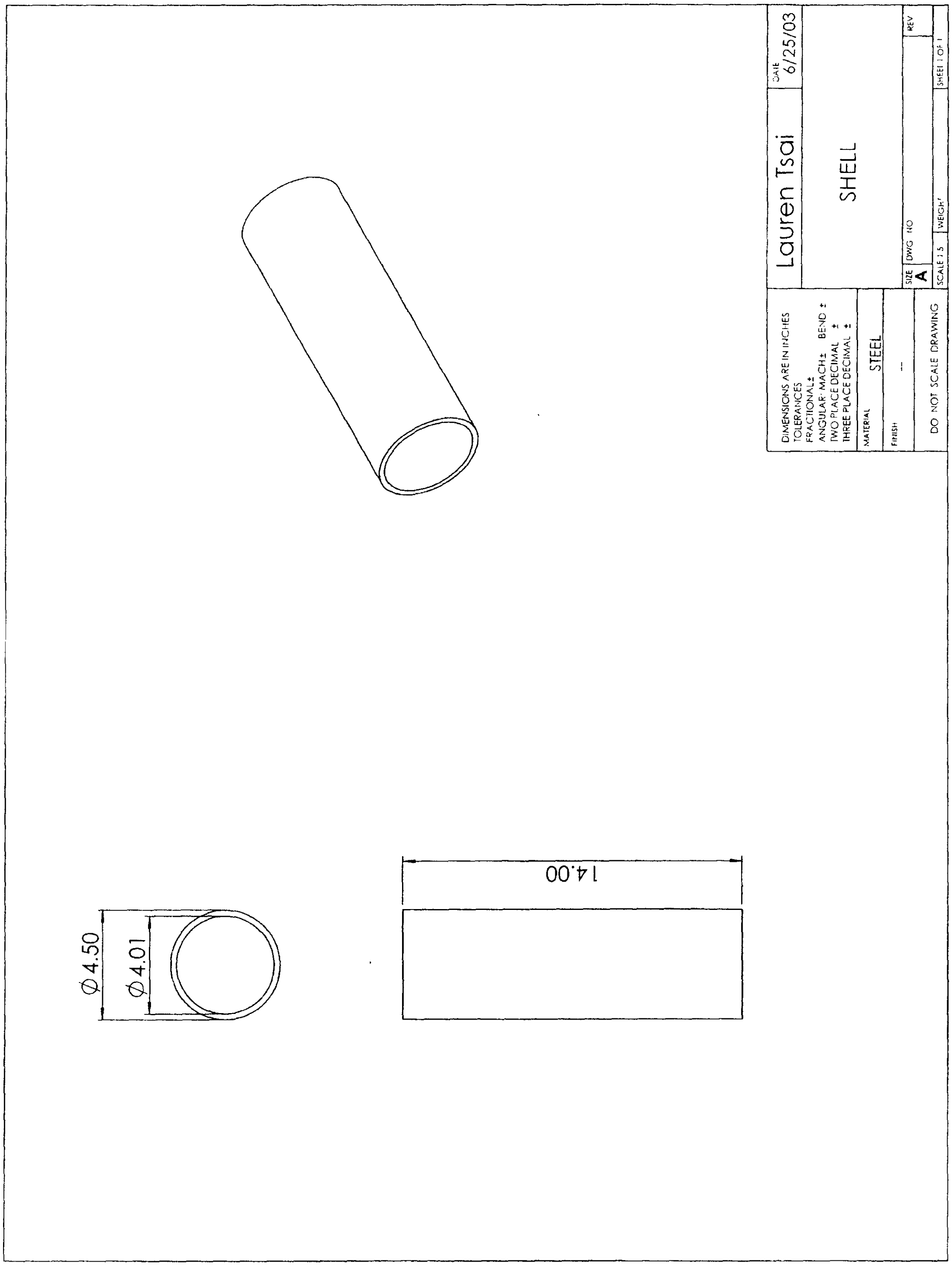

Figure 24: Shell Part Drawing

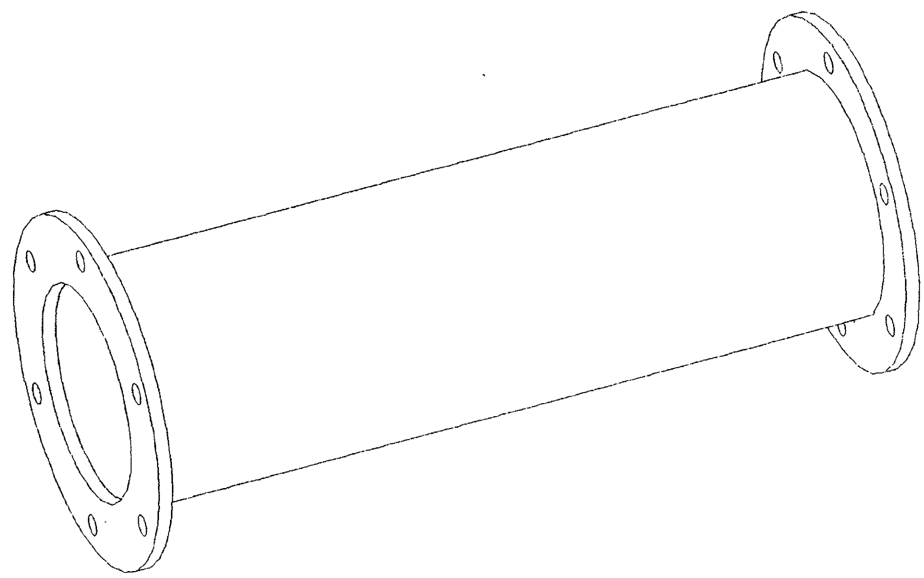

Figure 25: Shell Assembly Drawing

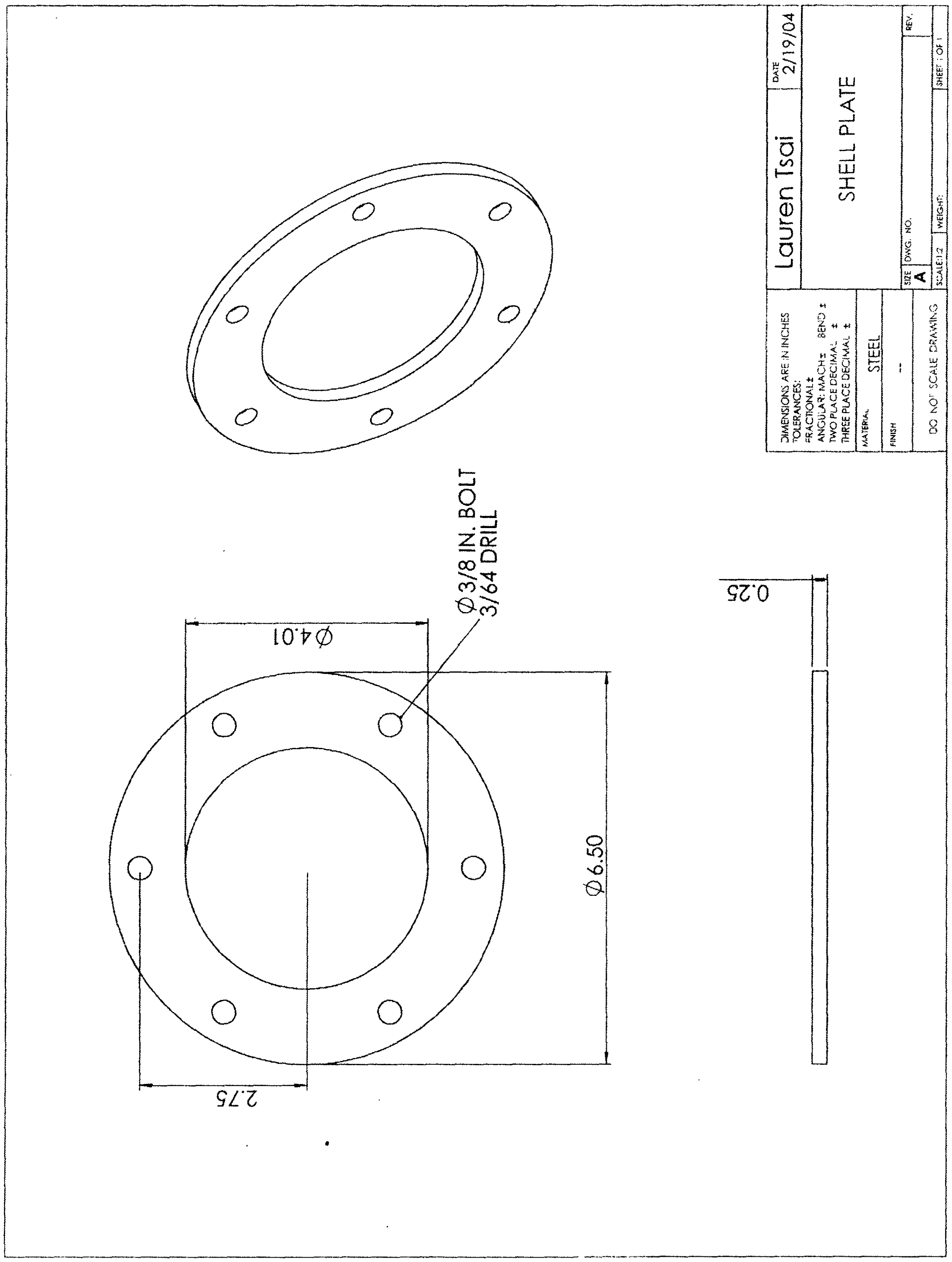

Figure 26: Shell Plate Part Drawing

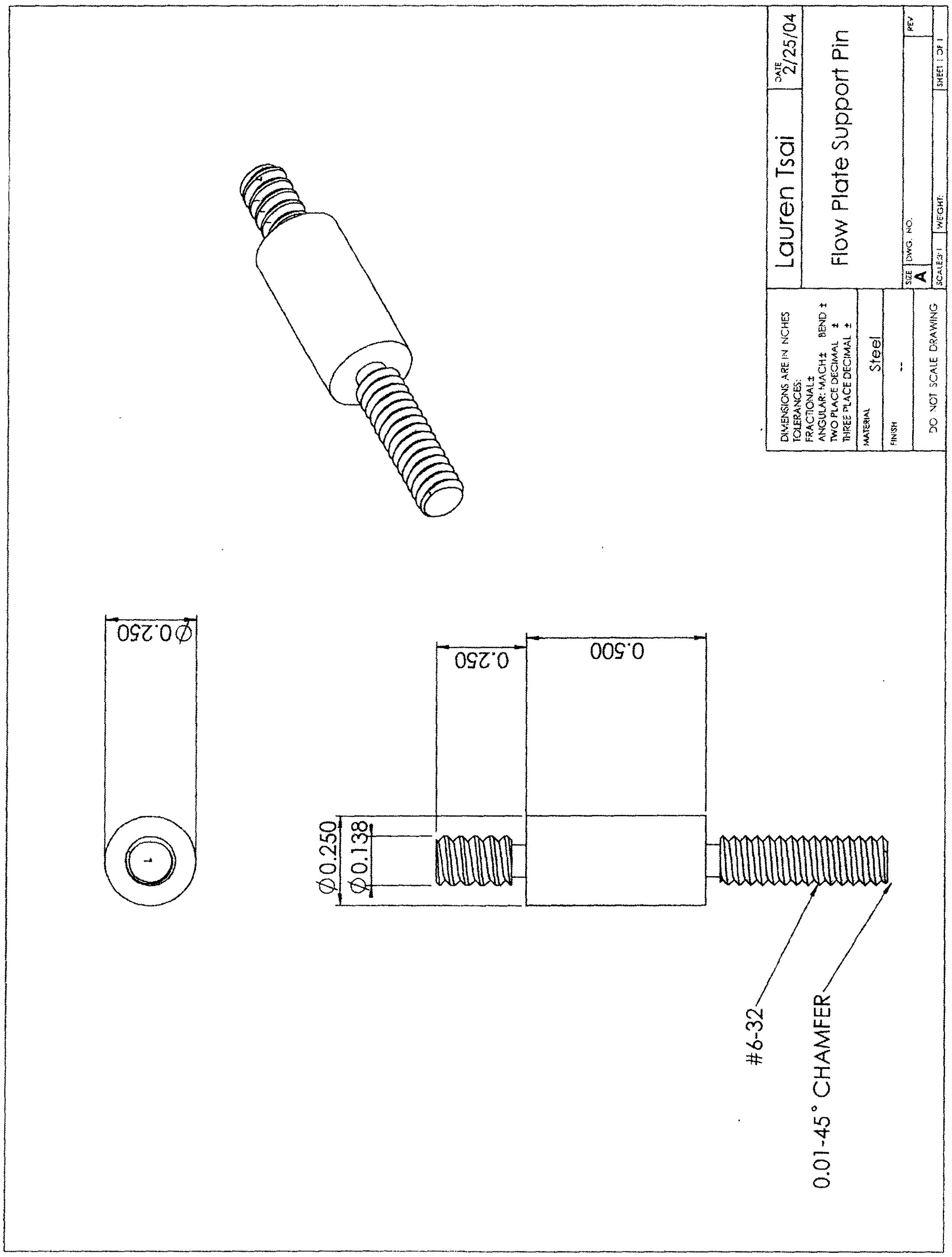

Figure 27: Support Pin Part Drawing

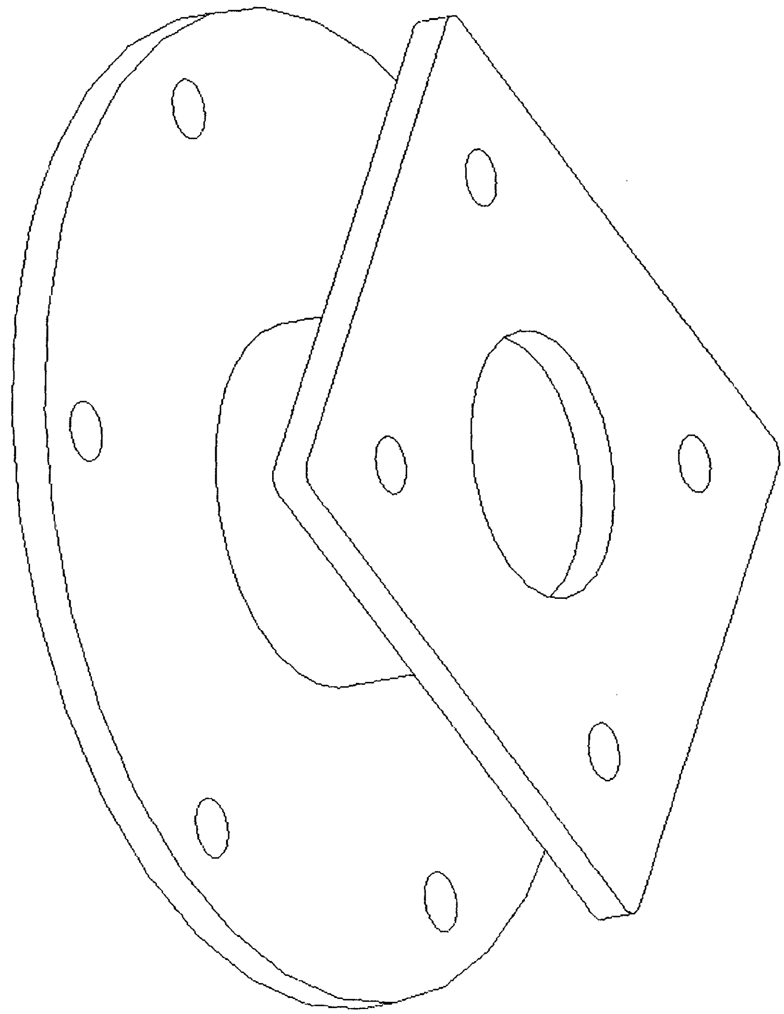

Figure 28: Turbine Inlet Assembly Drawing

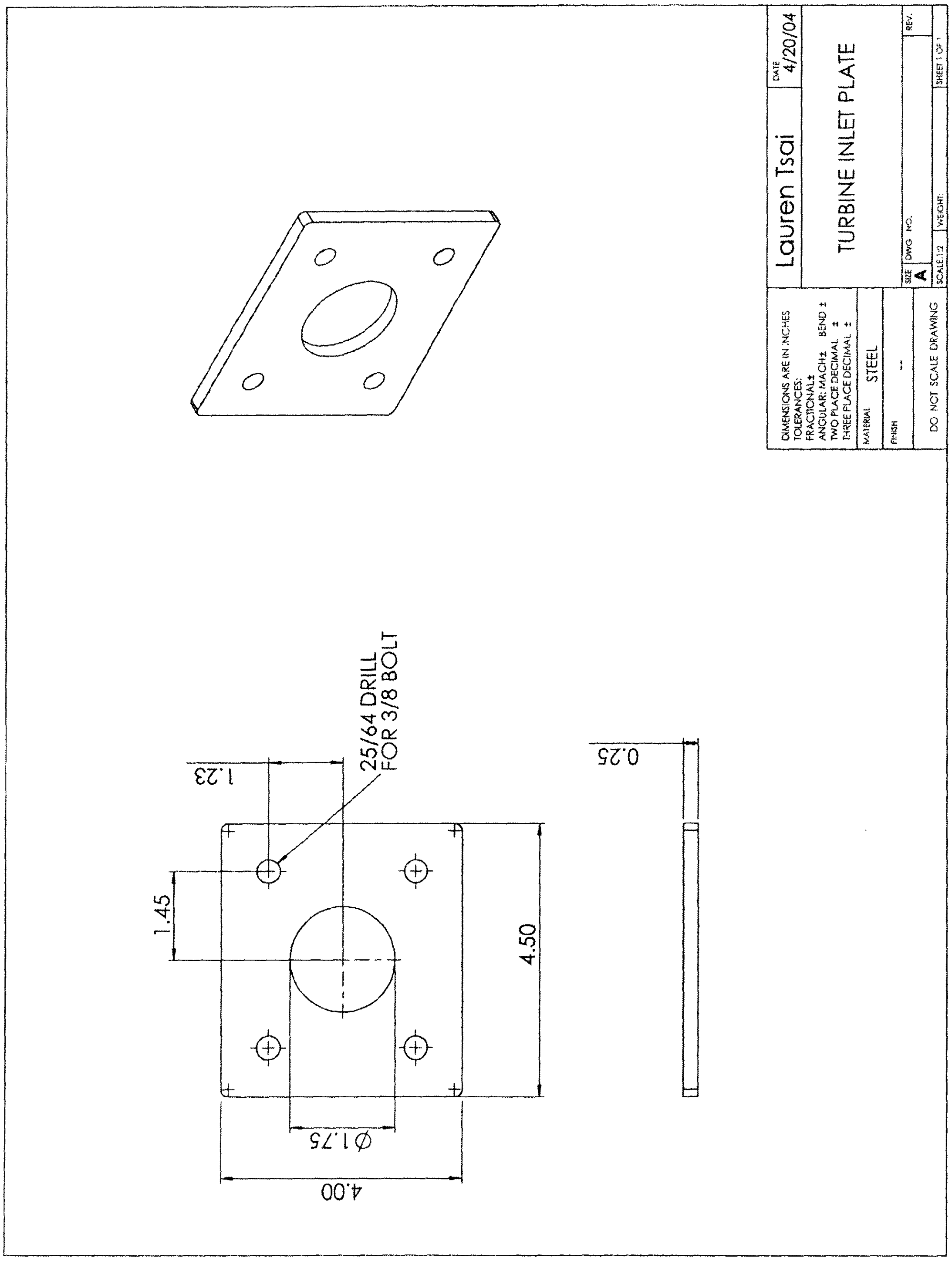

Figure 29: Turbine Inlet Plate Part Drawing