PERFORMANCE ANALYSIS OF A COMPRESSION IGNITION INTERNAL COMBUSTION ENGINE USING SUPERHEATED ETHANOL VAPOR

A THESIS SUBMITTED TO
THE GRADUATE SCHOOL OF INSTITUTE OF NATURAL SCIENCES
OF
MIDDLE EAST TECHNICAL UNIVERSITY

BY

ÇAĞDAŞ AKSU

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR
THE DEGREE OF MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

MAY 2011

Approval of the thesis:

PERFORMANCE ANALYSIS OF A COMPRESSION IGNITION INTERNAL COMBUSTION ENGINE USING SUPERHEATED ETHANOL VAPOR

submitted by **ÇAĞDAŞ AKSU** in partial fulfillment of the requirements for the degree of **Master of Science in Mechanical Engineering Department, Middle East Technical University** by,

Prof. Dr. Canan Özgen
Dean, Graduate School of **Natural and Applied Sciences** ________________

Prof. Dr. Süha Oral
Head of Department, **Mechanical Engineering** ________________

Prof. Dr. A. Demir Bayka
Supervisor, **Mechanical Engineering,METU** ________________

Examining Committee Members:

Prof. Dr. Kahraman ALBAYRAK
Mechanical Engineering, METU ________________

Prof. Dr. A. Demir BAYKA
Mechanical Engineering, METU ________________

Prof. Dr. Tuna BALKAN
Mechanical Engineering, METU ________________

Assoc. Prof. Dr. Cemil YAMALI
Mechanical Engineering, METU ________________

Dr. Anıl KAREL
STM A.Ş. ________________

Date: ________________

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Name, Last Name: ÇAĞDAŞ AKSU

Signature :

ABSTRACT

PERFORMANCE ANALYSIS OF A COMPRESSION IGNITION INTERNAL COMBUSTION ENGINE USING SUPERHEATED ETHANOL VAPOR

Aksu, Çağdaş
M.S., Department of Mechanical Engineering
Supervisor : Prof. Dr. A. Demir Bayka

May 2011, 152 pages

The aim of this study is to experimentally measure performance characteristics of a compression ignition (CI) internal combustion engine using superheated fuel vapor. The engine is a 1.3L inline 4 cylinder turbocharged and intercooled direct injection (DI) compression ignition (CI) engine. The engine will be fed with superheated ethanol as homogeneous fuel-air mixture through intake manifold as the secondary fuel, and with diesel fuel through the injectors as the primary fuel. Ethanol will be superheated using a new patented double heat exchanger which has been manufactured by Prof. Dr. Demir Bayka(METU), Dr. Anıl Karel (STM A.Ş) and Deniz Çakar (TAI INC). The results will indicate if the suggested concept can be applicable.

Keywords: Diesel, Dual Fuel, Ethanol Fuel, Fumigation

ÖZ

KIZGIN ETANOL BUHARIYLA ÇALIŞAN BİR SIKIŞTIRMA İLE YANMALI MOTORUN PERFORMANS ANALİZİ

Aksu, Çağdaş
Yüksek Lisans, Makine Mühendisliği Bölümü
Tez Yöneticisi : Prof. Dr. A. Demir Bayka

Mayıs 2011, 152 sayfa

Bu çalışmanın amacı, kızgın yakıt buharı ile beslenen bir sıkıştırma ile patlamalı içten yanmalı motorun performans değerlerini elde etmektir. Çalışmada kullanılacak motor 1.3l hacimli, sıralı 4 silindirli turbo beslemeli ve intercooler'lı, doğrudan püskürtmeli, sıkıştırma ile ateşlemeli bir motordur. Motor, emme manifoldundan ikinci yakıt olarak homojen hava-etanol buharı karışımı ile beslenirken, temel yakıt olan dizel yakıtı motora enjektörlerden verilecek. Etanolü buharlaştırırken, Prof. Dr. Demir Bayka(ODTÜ), Dr. Anıl Karel (STM A.Ş) ve Deniz Çakar (TAI INC) tarafından üretilen ve adlarına patentli çift eşanjör kullanılacak. Elde edilecek sonuçlar, önerilen sistemin ugulanabilirliğini ölçecek.

Anahtar Kelimeler: Dizel, Çift Yakıtlı Dizel, Etanol, Tütsüleme

For all true friends

ACKNOWLEDGMENTS

The author expresses his deepest gratitudes to Prof. Dr. A.Demir Bayka for his instructions and endless support throughout the study.

The author thanks to Reşat Yasin Özçelebi, İlter İmran Bodur, Nevzat Çakmak and Ali çağrı Batıhan for their personal supports as friends. They willingly helped out with the smaller and bigger problems in both during this study and in personal life. The author is also grateful to the technical personnel of mechanical engineering department for their support.

The author also thank to TÜBİTAK for the financial support during the study.

Finally, the author is thankful to everyone who supported him both technically and in personal affairs throughout the study.

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CHAPTER 1

INTRODUCTION

Energy is one of the fundamental components that directly influence economy, since every manufacturing facility and service demand on its supplies [1]. It is a well known fact that there is a global growth in economies which yields an increase in overall energy demand, thus an increase in overall supply and use of energy sources. In a closer look, coal, natural gas, oil, hydroelectricity and nuclear energy are the major energy sources for the world. Further categorizing, coal, natural gas and oil are defined as fossil fuels and these three types are limited in reserves. In contrast, fossil fuels provide 80,3% of overall energy demand on the world, where oil has the largest share with 35.4% [2]. Detailed information about the fact is provided in figure 1.1.

Energy use in transportation sector is focused on consumption of oil and a large sum of oil production is utilized for transportation [3]. In contrast to improvements on vehicles and the increase in their efficiencies, the number of vehicles is increasing, hence the consumption of oil. It is already stated that depletion of oil reserves is an inevitable result, where the depletion time is estimated to be around 35 years [4]. But this fact is not the only critical issue about oil market. According to the statistics, large oil reserves are concentrated on several regions of the world acting as the main supplies [5]. Smaller reserves still serve in local scale, but are not satisfactory regarding the consumption values. This situation is the most obvious for European countries, where local oil reserves can provide for around 10% of the demand [5]. The remaining fraction is filled in by exporting. Supplying from local sources or exporting, energy sources should be reliable. However oil market is unreliable. When all the negative sides of oil are considered, it becomes clear that proper alternatives that can meet reliability, cost and sustainability requirements.

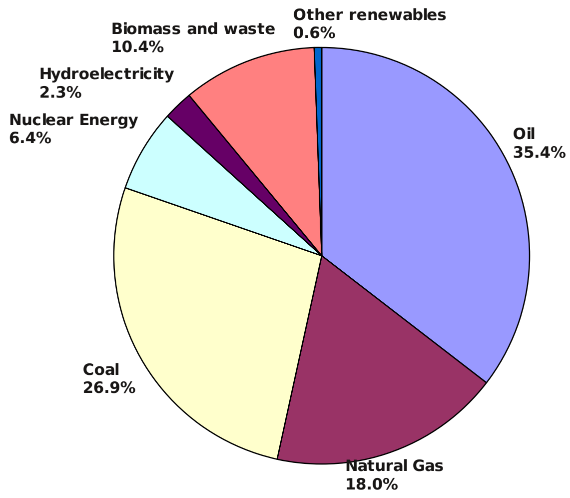


Figure 1.1: World Energy Consumption apportioned according to energy sources [1]

Petroleum is a raw source, and it is used after decomposing into fuels and other derivatives. While other derivatives like asphalt, wax or alkalenes are used as raw materials in chemistry or plastics industry, fuels serve as energy sources. The percentage of petroleum refined into fuels is 90,6% [6]. Further details of petroleum products and their fractions are given in figure 1.2. Physical and chemical properties of fuels are well known, and current machinery and equipment used on applications are purpose-built considering the characteristics of the fuels. Therefore, a proper substitute for the listed fuels is also expected to be compatible with the application.

Fuels are utilized through combustion process. Any type of fuel that undergoes combustion has end products, namely emissions. Emissions are pollutant gases that cause air pollution, hence affecting human health. Within the pollutant gas sources, road vehicles have a large share. According to European Union emission inventory report, 34% of CO and 41% of NOx emissions are resulting from road vehicles [7]. In order to keep the emissions under control, vehicle manufacturers are asked to meet defined standards. While these standards may differ

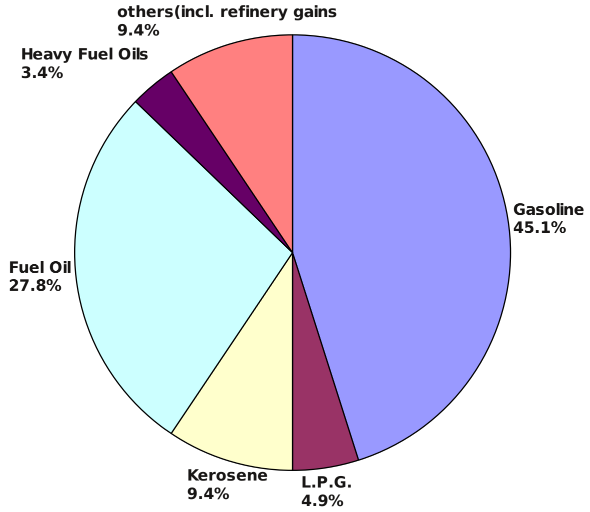


Figure 1.2: A breakdown of the products made from a typical barrel of US oil [6]

due to local implementations, there are also widely accepted regulations such as EPA regulations in U.S.A. or EURO emission regulations in European Union countries. Manufacturers can meet these requirements for fossil fuels, so the alternatives are expected to be at least as good, while improvements are favorable.

1.1 Alternative Fuels

Alternative fuels, or unconventional fuels are defined as substances that can be used as fuel, and not derivatives or products of fossil fuels. The most important difference between conventional and non-conventional fuels is sustainability. While fossil fuels are extracted from limited reserves, alternative fuels are obtained from sustainable sources like biomass, vegetables, waste, etc. Bioethanol, biodiesel, vegetable oils, biomass, hydrogen and non-fossil gases are well known examples for alternative fuels.

Common internal combustion engine types are SI and CI engines. Within fossil fuels, SI engines utilize gasoline, while diesel engine favors either diesel fuel. There are two important parameters that define these fuels: cetane number and octane rating. Cetane number is a term that is used for diesel fuel, defining ignition delay of the light distillate diesel fuel. The other term, octane rating is used for gasoline, which is resistance of the fuel to engine knocking. These parameters are also applicable to alternative fuels, which further identifies field of use of the fuel in discussion. Based on their characteristics, ethanol, methanol, propane, butane and hydrogen are proper gasoline substitutes. For diesel engines, dimethyl ether (DME), biodiesel, compressed natural gas (CNG) and vegetable oils are well known examples. Most of these fuels can be used in the engines without any adjustments or with minor hardware changes [8]. A very common application of such application is running diesel engines using compressed natural gas as the secondary fuel, which can provide clean burning without any performance loss [20] [22]. Public transportation buses equipped with simple additional equipment are one of the best examples for actual use of alternative fuels [21];

The most common type of alternative fuels is biofuels, including bioethanol, biodiesel and biogases. Current biofuels that are available at refill stations are known as first generation biofuels, and are manufactured from sugar, starch, vegetable oil or animal fat [9]. This is a concern for governments, since allocation of food supplies as energy source is not favorable. Regarding this concern, biofuels that are manufactured from non-food sources are proposed and are known as advanced biofuels. Second generation biofuels are also within this category and are manufactured from biomass like non-food crops and biowastes [10]. These sources are known as lingo-cellulosic materials, which trap sugar content within lignin and cellulose layers [11]. While manufacturing methods for second generation biofuels continue, the concept of third generation biofuels is defined. Raw material for third generation is offered as microbial sources or algae [12].

Raw material for biofuels can be generalized as agricultural products and their wastes, which can be produced on lands with agricultural potential. Bioethanol is the most popular alternative fuel worldwide. First generation bioethanol is manufactured through fermentation, while biochemical and thermochemical processes are offered for production of second generation biofuels [13].

Bioethanol is known to be a proper substitute for gasoline since gasoline engines can be operated with ethanol only, or its blends in various compositions. However, this case is not the same for diesel engines. Since properties of diesel fuel and ethanol is considerably different, both engine and fuel require modifications based on the type of the application. Still, both as a renewable fuel and for lower emission levels, ethanol is a promising alternative fuel. That's why it is important to utilize it in diesel engines and this motivates researchers to solve current problems for this application and to offer it for commercial use. Following sections of this chapter will discuss on historical background of ethanol, its applications on diesel engines and their limitations and problems are discussed. The last section will explain the details of this research.

1.2 Diesel Engines

Diesel engines, also known as CI engines are invented by Rudolph Diesel. The theory behind this engine is burning fuel by raising air temperature inside the cylinder up to self ignition temperature of the fuel by compression and then introducing the fuel inside the cylinder. In February 1897, the first self working prototype of Diesel Engine is run for the first time, and later turned into commercial product [14].

This engine has found a large area of application from passenger cars to heavy duty diesel engines that are used in power ships and power generators. Diesel engines are famous for their durability, better torque characteristics, higher energy conversion efficiencies, reliabilities and safety of handling their fuels. However, their lower specific power outputs, higher noise levels, narrower power band and maintenance costs are major problems of diesel engines. In terms of exhaust emissions, diesel engines have stronger and weaker points compared to gasoline engines: they tend to emit less CO but higher NOx and soot.

The most noteworthy property of diesel engines is their adaptability for variety of fuel types, including the fuels that are suitable for SI engines. Gasoline and its alternatives are normally unsuitable for regular diesel operation, but they can be burned successfully if they are fed through intake port. Ignition in this system is provided by self ignition of diesel fuel. This application is named as dual fuel diesel engine, and is in use for a long time. The most widely known application is use of CNG in stationary diesel engines in power generators. As another example, public busses of Ankara are operated with CNG. This system is preferable due to reduced stresses on injection system, enhance reliability, lower cost of secondary fuels and improvements on NOx emissions [21] [30].

1.3 Ethanol

Ethanol is a volatile, colorless and flammable matter, and is in liquid phase at STP. This material is manufactured by various methods. Current manufacturing methods for ethanol are hydration of ethylene from petrochemical sources, and fermentation of sugar by yeast from biological sources [15]. While advanced methods that use lignocellulosic matters are in research, use of petrochemical sources become unreliable due to the status of oil market.

Ethanol and its blends are commercially available products as pure ethanol fuel or as fuel blends. However, these fuels have some issues. Ethanol fuel, also known as E100; contains 96,4% of ethanol and the remaining fraction as water, which is the highest purity level that can be obtained by distillation. For fuel blends, suppliers prefer using anhydrous ethanol in various fractions. The most popular fuel blend throughout the world is E10 that consists of 10% of anhydrous ethanol by volume. Use of biological materials gradually replaces petrochemical materials in ethanol production, and biological production methods are further improved. As a result, ethanol becomes a promising alternative fuel for road transportation. While biodiesels and vegetable oils dominate the market as diesel alternatives, proper ways of ethanol use in diesel engines is further researched.

1.4 Ethanol Use in Diesel Engines and Their Drawbacks

Diesel engines are famous for their adaptability to any fuel type. Since combustion occurs due to self ignition of the fuel, it can be told that any combustible fluid can be used in diesel operation as long as required conditions can be satisfied for a particular fuel. However, this statement is not as easily applied. Current engines are designed for light distillate fuel oil, namely diesel fuel, which means that combustion chamber pressure and temperature reaches or barely exceeds the values that is enough for self-ignition of diesel fuel. Focusing on ethanol, it can be seen that self-ignition temperature of this fuel is higher than that of diesel fuel.

Various approaches that can solve the stated problem are offered by both academic researches and companies. These approaches depend on modifications on the engine, fuel or both. While they can be evaluated as proper solutions, each of them has their own drawbacks.

1.4.1 Diesel-Ethanol Blends and Emulsions

Ethanol can form homogenous mixtures with diesel if the temperature is higher than 10C, but the mixture dissociates below this limit [16]. This property is undesirable for cold weathers since weather temperatures drop below the mentioned level in winder throughout the world. In addition, water content is also a parameter that affects solubility of ethanol in diesel fuel [16].Homogeneity can be ensured by using co-solvents or bioemulsifiers, which are commercially available.

Compatibility of ethanol and diesel engines is an important issue, especially for fuel delivery system. First of all, viscosity of ethanol is lower than diesel fuel and alters fuel spray characteristics. Second and the most critical issue is corrosive nature of ethanol. Polarity of ethanol molecule, water content and impurities cause corrosion on metal parts and have side effects on rubber parts of the engine [17]. Manufacturers report that majority of the engines can run with ethanol blends without any modifications, if ethanol content is kept below 10% by volume. For blends or emulsions of higher ethanol concentration, fuel line modifications are necessary.

1.4.2 Direct or Indirect Injection of Ethanol Fuel

As mentioned, ethanol is known to be harmful for engine components and has side effects on fuel spray characteristics. Compatibility of engine is an important factor, which directly influences the reliability and life of an engine. In addition, low cetane number and higher self ignition point are the most important issues regarding used of ethanol. There are several suggestions that are reported to be proper suggestions during modifications and design activities exclusive for ethanol use. A basic approach is using separate injectors, fuel lines and tanks for ethanol and diesel fuels. This system is known to replace up to 90% of diesel [18]. In contrast, recent studies focus on using ethanol as a total replacement of diesel fuel; hence suggest proper ways and approaches to modify the engines. Increasing compression ratio, installing ignition assistance devices and applying thermal insulation to the combustion chamber are the suggested as proper methods [27]. Another approach is suggested as addition of ignition improvers to the fuel. A recent example for such fuels is ED95, a product is offered by SEKAB and suggested for use in modified diesel engines [27]. This fuel is used for public transportation busses in Sweden.

1.4.3 Ethanol Fumigation

The most common dual fuel diesel applications use diesel fuel for igniting the fuel that is introduced through intake port. Diesel engines that consume compressed natural gas make the best example with their wide range of use from transportation to power generators. Other than CNG, most of the liquid and gaseous fuels can be used in this method including ethanol. The method in which ethanol is fed through the intake by carburetion, injection or vaporization is named as ethanol fumigation [18]. This method requires minimal modifications on the engines. However, ethanol fumigation has limitations. The most significant one is engine knock that occurs as the amount of ethanol replacing diesel fuel reaches certain values that depend on the properties of the test engines and ethanol. The highest ratio of diesel replacement is reported in the literature is 50% [18] [24].

1.5 Literature Survey

Ethanol fumigation is a promising method because of its ease of application compared to other methods. Because of this fact, researchers studied on different aspects of this method. The most significant studies focus on observing the maximum ratio of ethanol that can substitute diesel fuel, the optimum ratio of ethanol that can improve thermal efficiency or the variation of exhaust emissions.

Broukhiyan and Lestz (1981) studied on effects of ethanol and methanol fumigation on diesel engines using a turbocharged indirect injection diesel engine for 1/4, 1/2, 3/4, and full rack settings at 1500, 1720 and 2100 rpm. Fumigation was provided by pressurized nitrogen Results showed that ethanol can provide 50% of the energy input by fumigation. This percentage is limited by occurrence of knock; where severe knock did not occur for rack settings of 1/4 and 1/2. Increase in thermal efficiency is observed on these test points. [26]

Hayes et. al. (1988) studied ethanol fumigation with 100, 125, 150, 175 and 200 proof ethanol on a turbocharged diesel engine. Ethanol was injected into the intake ports with separate injectors for each cylinder. Use of ethanol in any proof led to reduction of NOx emissions at lower loads, but resulted as increase in HC emissions for all loads. [27] Ottikutti et. al. performed experiments on a turbocharged diesel engine and examined effects of ethanol fumigation on emissions. Transient loading conditions for heavy duty diesel engines was used throughout the experiment. Ethanol replaced between 10 to 30 percent of total energy input to the engine. Results showed that increasing ethanol fraction decreased NOx emissions and increased HC and CO emissions. [28]

Ajav et. al. (1998) researched ethanol fumigation on a single cylinder stationery diesel engine and focused on replacement of diesel fuel by ethanol and its effects on engine performance. During the experiment, air-ethanol mixture is used in both preheated and un-preheated form. Results of this study indicate that maximum ratio of ethanol replacement is not affected by preheating. The highest replacement ratio is observed to be 33.6% throughout the experiment, where only 15% of replacement was possible for full load conditions. [29]

Abu-Quadis et. al. (2000) performed experiments in order to observe effects of ethanol fumigation in terms of engine performance and emissions. A single cylinder naturally aspirated DI engine was tested at 2000 rpm. Ratios of diesel and ethanol were adjusted for fixed energy input. Compressed ethanol is injected into intake port through a spray nozzle. The results showed 7.5% increase in break thermal efficiency, 55% increase in CO emissions, 51% reduction in soot emissions and 36% increase in HC emissions for 20% ethanol. The same experiments were performed for 20% ethanol blend, and fumigation is observed to give better results in comparison. [18]

Cheng et. al. (2010) studied on reduction of exhaust emissions by using fumigated ethanol as oxidation catalyst. Experiments were performed on a four cylinder direct injection diesel engine at 1800rpm for break mean effective pressures of 0.08, 0.19, 0.39, 0.58 and 0.70 MPa. Ethanol replacement ratios were selected as 5, 10, 15 and 20% throughout the experiment. Results indicate 7% increase in break thermal efficiency for 10% replacement and 9% of improvement for 20% ethanol replacement. In terms of exhaust emissions, CO, HC and NO2 emission values increase as ethanol replacement ratio is increased. For 10 and 20% of replacement, CO emissions tend to increase 1.8 and 2.7 times and HC emissions show 1.6 and 3.3 times increment respectively. On the other hand 6.6% and 14.7% of reduction in NOx is observed for the denoted replacement ratios at 0.37MPa, while these values drop to 0 and 3.9% at 0.70MPa. In terms of smoke and particulate emissions, 56% of reduction in opacity occurs at 0.58MPa , which drops down to 19% at full load. [25]

1.6 Scope of The Study

The purpose of this study is to operate the diesel engine using superheated ethanol, which is fed through the intake port as air/fuel mixture and replacing diesel fuel, and to compare the performance characteristics of the engine in dual fuel mode to the results that are obtained from the engine operating with diesel fuel. Exhaust gases provide heat for superheating ethanol. Ethanol will be superheated using a new patented double heat exchanger has been manufactured by Prof. Dr. Demir Bayka, Dr. Anıl Karel and Deniz Çakar. After superheating, air/ethanol mixture is prepared in a gas mixer that is installed on air intake line after intercooler. Ethanol used in this experiment is anhydrous ethanol that contains 99.7% of its volume as alcohol and the remaining fraction as water and other impurities. The implemented type of ethanol is selected in order to research the probability of use of a low cost alcohol as a replacement for diesel fuel. The results of the experiment will indicate if the suggested system is applicable. The hypotheses that motivate for this research are as following:

- Injection in diesel engines provides better ignition than a spark plug, by which misfiring is mostly eliminated and combustion characteristics are improved. This application is known as dual fuel diesel engine application and is successfully applied for various gaseous and liquid fuels.

- Exhaust gases contain rejected heat that can be used for evaporating and superheating ethanol. Utilizing rejected heat is expected to improve thermal efficiency of the engine.

Tests will be performed at 100%, 75% and 50% load conditions. and at engine speeds between 4000 and 2000 rpm with 250 rpm intervals. Torque characteristics of the engine in diesel mode will identify the torque values that will be met during dual fuel mode. Performing the tests and collecting data at the mentioned conditions, characteristics of the engine in diesel mode and dual fuel mode will be used for comparison during evaluation.

In dual fuel experiments, engine will run on diesel fuel until superheated ethanol is generated and introduced to the engine. Once ethanol reaches the utilizable state, throttle pedal position and needle valve position will be adjusted synchronously in order to achieve minimum rate of diesel fuel consumption at the tested point. The maximum amount of ethanol flowrate and minimum rate of diesel flowrate will be decided based on engine knock.

CHAPTER 2

EXPERIMENTAL SETUP

In this chapter, the test setup, measurement and control components, and the software will be explained in detail. The setup consists of three main parts :

1. Engine performance test system : The system that will measure torque, speed, fuel consumption and air intake.
2. Superheated fuel vapor generation and control system : The system that generates superheated ethanol and ensures safety of the test environment.
3. Software, Data Acquisition System and Other Measurement and Control peripherals: These devices control every electronic equipment that actuates mechanical elements.

A simple schematic of the system can be viewed in figure 2.1, and each component installed on the system will be explained in details throughout the chapter.

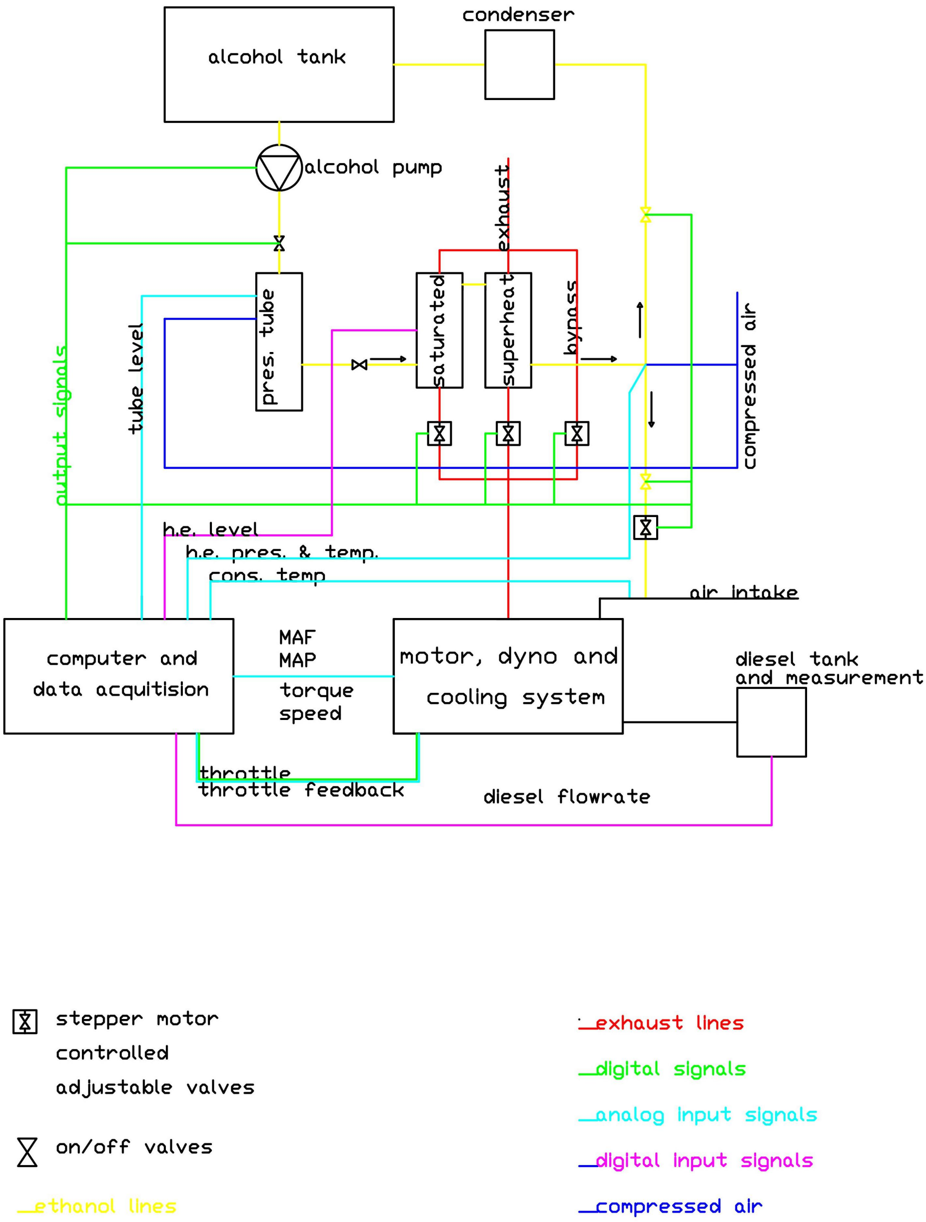


Figure 2.1: Schematic of the Test Setup

2.1 Engine Performance Testing System

Engine performance test system is build up of the test engine, dynamometer, air intake measurement system, fuel flowrate measurement system including diesel fuel and ethanol measurement devices, and the cooling system. All the measurement devices in this system generate their outputs as electric signals, and these data are collected by the data acquisition system, and the controls of these equipment are again performed over the software through control hardware. Figure 2.2 provides the view of engine performance testing system.

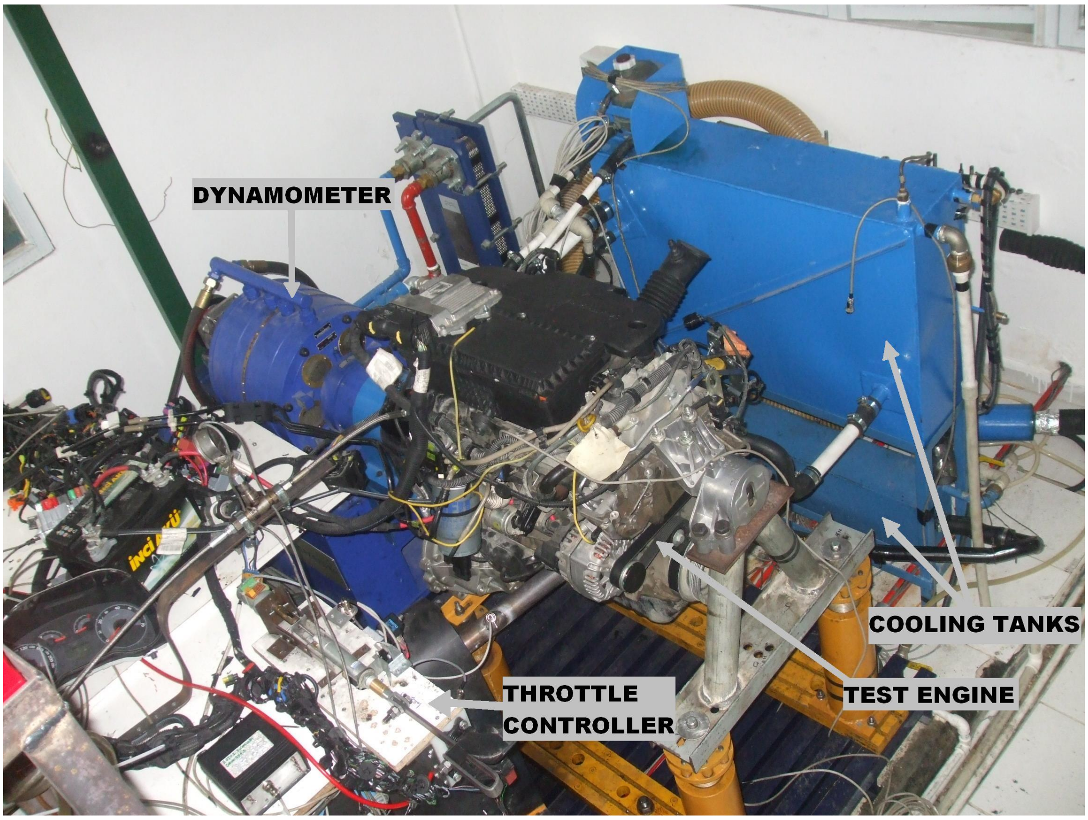


Figure 2.2: Engine Performance Testing System

2.1.1 Test Engine

Experiments are performed on a 4 stroke 1248 cc turbocharged intercooled DI Diesel engine manufactured by FIAT. The specifications of the engine are given in table 2.1.

This engine was formerly used on a test vehicle by the manufacturer, so it arrived with the

Table 2.1: Specifications of the Engine

Manufacturer	FIAT
Engine Type	4 stroke Diesel, 4 valves per cylinder
Displacement	1248 cc
Bore and Stroke	69.6 mm x 82 mm
Compression Ratio	17.6:1
Fuel delivery	Common rail type
Aspiration	Turbocharged and Intercooled
Maximum Torque	200 Nm at 1750 rpm
Maximum Power	90 HP at 4000 rpm

harness that is uninstalled from the vehicle. There were no electrical, mechanical or software modifications done on the engine, harness and engine control unit; only a gas mixer is installed on intake line and signal measurement cables are soldered to signal and ground channels of manifold absolute pressure sensor and mass air flowmeter. Adding these cables did not alter operation of the components.

The engine is controlled by its engine control unit, and adjust the operating conditions if necessary. Such as, if temperature of the engine rises to dangerous levels, it will adjust diesel fuel injection rate to a smaller value to keep the engine safe. In addition, it also detects any malfunctioning or unconnected component, it will restrict power output in order to prevent from any undesired condition that can put the vehicle into a dangerous situation.

The engine may not be mounted on the vehicle in the laboratory, but these self control properties are beneficial, since it can be known that engine is running properly if its power output is not restricted.

2.1.2 Dynamometer

Dynamometer is a device that is used for measuring force, moment of force or power of prime movers. Power of the prime mover can be calculated from these values since power is function of rotary speed and torque. These devices measure torque output of the coupled prime mover in two ways. In the first method, a known amount of breaking force is applied to the prime mover. On the second method, work output of a pump, fan or an electric generator coupled to the prime mover is measured. In this experiment, AVL DynoPerform 160 eddy current dynamometer is utilized. This dynamometer was available at the laboratory, and is the component that the test setup is built around this device due to the availability of its peripherals. Installation location on the test system can be viewed in figure 2.2.

The dynamometer can be used for the engines up to 160kW of power output and can operate up to engine speeds of up to 10000 rpm. The tested engine can run at maximum engine speed of 5000 rpm and has maximum power output of 90 HP (54 kW), which lies within the measurement range of the dynamometer.

The dynamometer has loading unit control unit as its components. Loading unit applies breaking force on the engine and measures its speed simultaneously. Breaking force is generated by applying magnetic field on the rotating discs that are parts of the shaft. A magnetic sensor generates engine speed data as frequency. Control unit measures and processes the engine speed data and sends electric current to the loading unit. This unit is also the user interface of the system, where desired input are given by either its adjustment knobs or from a computer as input signals. During the experiment, dynamometer is controlled from the software.

2.1.3 Throttle Controller

This engine utilizes a throttle pedal equipped with a potentiometer for reading throttle input data and adjusts engine parameters based on its output. In this system, structure of throttle pedal is not modified, but a system is built for manipulating its position. The system that serves for this purpose is named as throttle controller. Throttle controller is a combination of a solenoid valve, a pneumatic piston, a DC motor and mechanical components.

DC motor rotates a bronze threaded cylindrical piece and this action drives a M5 powerscrew. This motion pulls the pneumatic piston when throttle position is required to increase, and pushes the piston back for the opposite effect. This motor can rotate at 72 rpm if operated at 24 V of supply voltage. Motor is controlled by the software like all the other electrical control elements.

Pneumatic piston serves for quick release of throttle when the physical properties of ethanol vapor meet the required conditions for consumption. In such cases, most of the diesel fuel consumption is expected to be replaced by ethanol, so an instantaneous operation is essential. As an additional benefit, this piston serves as a safety equipment in case of an emergency.

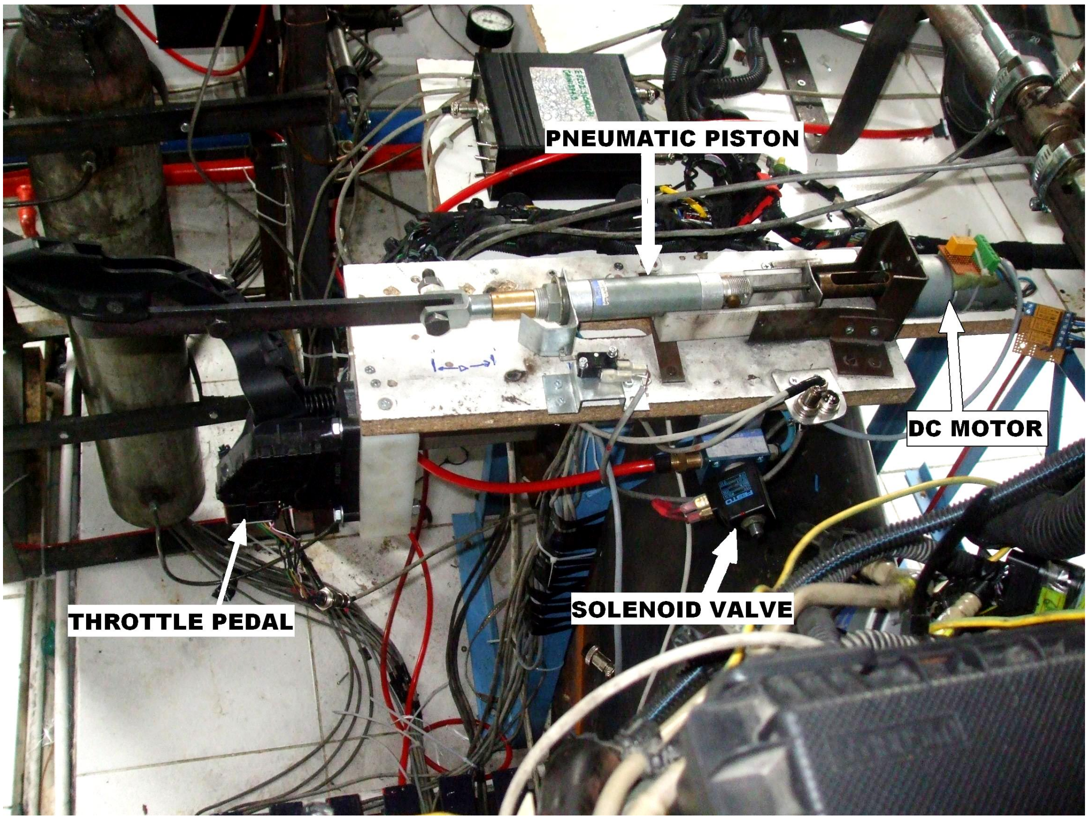


Figure 2.3: Throttle Controller System

Since throttle pedal is equipped with springs and can return to its release position on its own, piston is used only when pedal is pulled. Furthermore, stroke of this kind of pistons are fixed but the required movement for the piston is depends on the throttle position from where it will be released. As an example, pushing the pedal in a distance that is equivalent to 100% throttle position in a condition where 50% is required will inevitably break the pedal.In order to provide for both activation and safety of pedal, piston is actuated from only one side to pull the pedal. This movement is controlled by a solenoid valve that is triggered by the computer.

Throttle position is measured from the throttle pedal potentiometer by connecting the signal outputs of this component to the data acquisition system. Output signal from throttle potentiometer is analog voltage that ranges between 1.0 V to 4.18 V, which changes linearly with the throttle position. Measuring throttle position from this component did not cause any instabilities or changes in characteristics of the engine.

Finally, a safety switch is installed in order to define a range where piston can be re-activated after a release. This switch sends a signal to the computer if the piston is the safe range. This range corresponds to the range of 0 to 45% throttle positions when this piston is activated.

The remaining mechanical components serve as a chassis that holds the explained components as a single unit. Picture of the system can be seen in figures 2.3 and the technical drawings of the assembly and components are available in Appendix C.

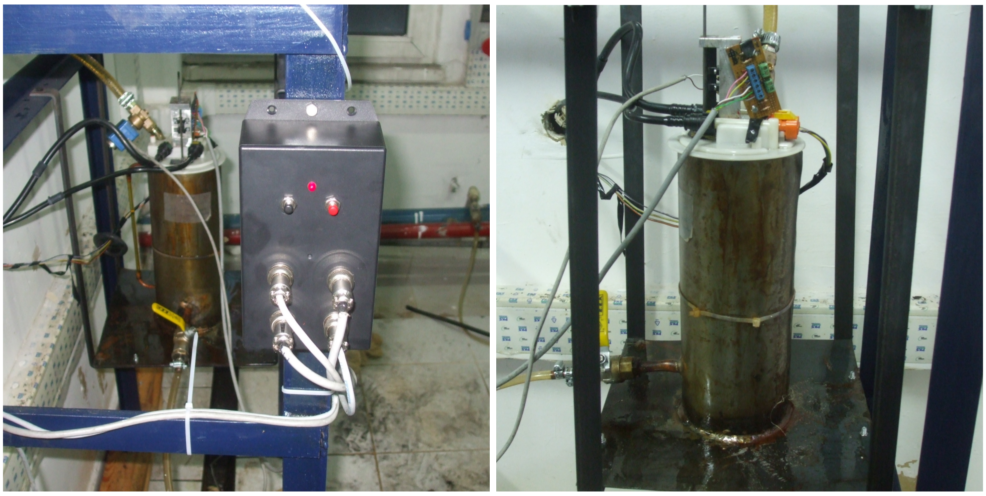
2.1.4 Diesel Fuel Measurement System

The fuel measurement system used during the tests is specially designed and manufactured in the laboratory for this engine. This system combines manufactured parts of the measurement system, an electronic card and original fuel pump of the engine.

Measurement system utilizes two optical sensors mounted on the fuel pump and a slotted flag that is attached on a floater assembly. Level of the floater changes as the fuel is consumed or refilled. Those two operations drive the floater, hence the slotted flag. Slotted flag provides signals to a pair of H21A1 infrared sensors which indicate top or bottom position of the calibrated level. This flag is designed in a way that it can trigger either upper or lower sensor, but not both at the same time. Sensors are protected against ambient light so that unexpected activation of signals is prevented. In addition, sensors are carried by an aluminium piece that is tightened onto the fuel pump with screw connections. Finally, fuel pump is mounted on a fuel container that contains the measured amount of fuel and some surplus in order to provide sufficient fuel for the pump to work correctly.

More examples of this device were used in the laboratory in former experiments, and this particular device is designed using prior examples as reference.

Electronic card collects the signals from the optical sensors, then drives the refill solenoid valve. If the triggered sensor is the lower one, refill solenoid is opened, and the operation is reversed for the opposite case. Refilling operation is several times faster than the maximum fuel consumption rate of the engine, so that the engine will not run out of fuel and resume its operation without any fuel related interruptions. In the same time, these signals are sent to the computer and processed in a timer operation. Since the distance between the sensors are fixed and the filled volume is constant, the amount of fuel filled and consumed is equal and constant in every cycle. This card is secured inside a plastic circuit box and necessary



(a) Diesel Fuel Measurement System (b) Container, Fuel Pump and Sensors

Figure 2.4: Diesel Fuel Measurement System

connections are provided with MICE sockets.

During the tests, computer collects the aforementioned signals and processes them in order to compute the fuel consumption rate. Processing occurs in the same manner as the electronic card does. What differs is, computer holds track of consumption time via the test timer, and divides the volume of the used fuel amount by it. The result is the fuel consumption rate in grams per second.

This system is calibrated by collecting the defined amount of fuel from the system into a flask and refilling it again. This operation is repeated for at least 30 times where both repeatability of the system is checked and average of the measured volume is calculated. Collection of fuel is performed by another solenoid valve, which is opened as the flag reaches to the upper position, and closed as the lower sensor is triggered.

The components mentioned in this section are shown in figure 2.4(a) and 2.4(b).

2.1.5 Turbinemeter

This study focuses on running the engine with superheated ethanol. For this reason, it is essential to measure the rate at which engine consumes ethanol vapor. Formerly used in a similar experiment in the laboratory, this device is known to have suitable characteristics for this purpose [33].

Ethanol flowrate is measured by Omega FTB 931 turbine meter. This device has frequency output, which is converted to analog voltage for connecting to data acquisition system. This conversion is done by a circuit that utilizes LM2907 frequency to voltage converter. This device is calibrated by comparing voltage readings to the values read from a rotameter at standard temperature and pressure. During the experiment, This component is a convenient choice for the system, since it can measure actual flow rate. It will be mentioned later that ethanol hoses are well insulated, so the temperature drop of ethanol vapor is expected to be negligible. In addition, turbine meter is installed close to intake manifold in order to take manifold pressure reading as the reference pressure for calculating density of ethanol vapor at observed conditions.

Catalog information about the component is given in Appendix B.2.

2.1.6 Cooling System

The vehicles that use this type of engine is equipped with two radiators; one for coolant water and one as intercooler. These radiators are cooled by oncoming air and fans on the vehicle, but the engine is stationary at laboratory conditions. In order to provide sufficient cooling, these radiators are submerged into separate tanks where cool water flows around the radiators. This application is beneficial since adequate cooling and this operation does not affect ambient temperature. Once coolant water collects heat load from the radiator, it's pumped back to cooling tower. In addition, a thermostatic valve controls water flow into the tank where coolant water radiator is submerged; so that engine can be kept in optimum temperature range without overheating or cooling below favorable range.

Engine emits heat to its surroundings during operation. This increases temperature in the test room. In order to maintain constant ambient temperature, an electric fan is used. This fan blows air into the test room at ambient temperature and hot air is blown out of the room. Fan is located in such a position that resulting air current is directed onto the engine. This is an important point, since there are no special equipment used for cooling engine oil, hence sump needs to be cooled down.

This system is located as close to the engine as possible in order to meet the conditions as in the vehicle. But still, due to the space restrictions of the test environment, intercooler had to be located relatively distant in comparison to the original configuration. As a result, intake air charge has to travel a longer route, which will cause a higher pressure drop than the original configuration.

Location of the coolant tanks can be viewed in figure 2.2.

2.2 Ethanol vapor Generator and Flowrate Control System

One of the fundamental concepts that motivates this study is to generate superheated ethanol vapor using the available heat sources on a vehicle; that is rejected heat in the exhaust gases in this case. That's why the system is equipped with both flow rate control components and safety components. This section will describe these components and their uses in details.

2.2.1 Double Heat Exchanger

The only feasible way to transfer heat between fluids without mixing them is using heat exchangers. The heat exchanger used in this study is a new patented double heat exchanger that has been manufactured by Prof. Dr. A. Demir Bayka, Dr. Anıl Karel and Deniz Çakar, and used for superheating gasoline [33]. This heat exchanger operates with ethanol and exhaust gases as working fluids where exhaust gases provide the heat input for evaporation and superheating of ethanol. This heat exchanger has three main components:evaporator and superheater heat exchangers, and bypass line.

Ethanol enters vapor generator at evaporator heat exchanger. Evaporator is a shell and tube heat exchanger. Exhaust gas flow inside the tube bundle and transfers heat to the ethanol that is kept at the shell side. Ethanol level is monitored by a switch that is triggered by a floater. As a result the signal sent to data acquisition system informs about the condition of the ethanol level as full or empty. Once the level is considered as empty, ethanol is pumped into evaporator by pressurized tube. System is refilled rather frequently due to the nature of floater driven switch, but this is convenient for an uninterrupted ethanol supply to the engine.

After evaporation, ethanol is sent to superheater. Superheater is also a shell-and-tube heat exchanger with helical coil. Exhaust gases flow around the helical coil and heat saturated ethanol vapor into superheated state. Temperature and pressure of ethanol vapor is measured by the thermocouple and pressure sensor respectively. These components will be discussed at the last section of this chapter. Thermocouple and and pressure sensor are located at the exit of superheater heat exchanger. At this location, ethanol can either be sent to the system for consumption or back to ethanol tank in case pressure inside the generator rises above permissible level. An additional pipe supplies pressurized air to the system, and is operated at the end of an experiment in order to drive ethanol back to the pressurized tube. All these operations are decided by the software and are performed by control valves.

Flow rate of exhaust gases into the heat exchangers is adjusted by butterfly valves that are driven by stepper motors. This control is essential in order to balance vapor generation and consumption rates and to keep temperature of the vapor within desired values. In such cases, some portion of exhaust is directed to the bypass route. Pictures of the system is available in figure 2.5.

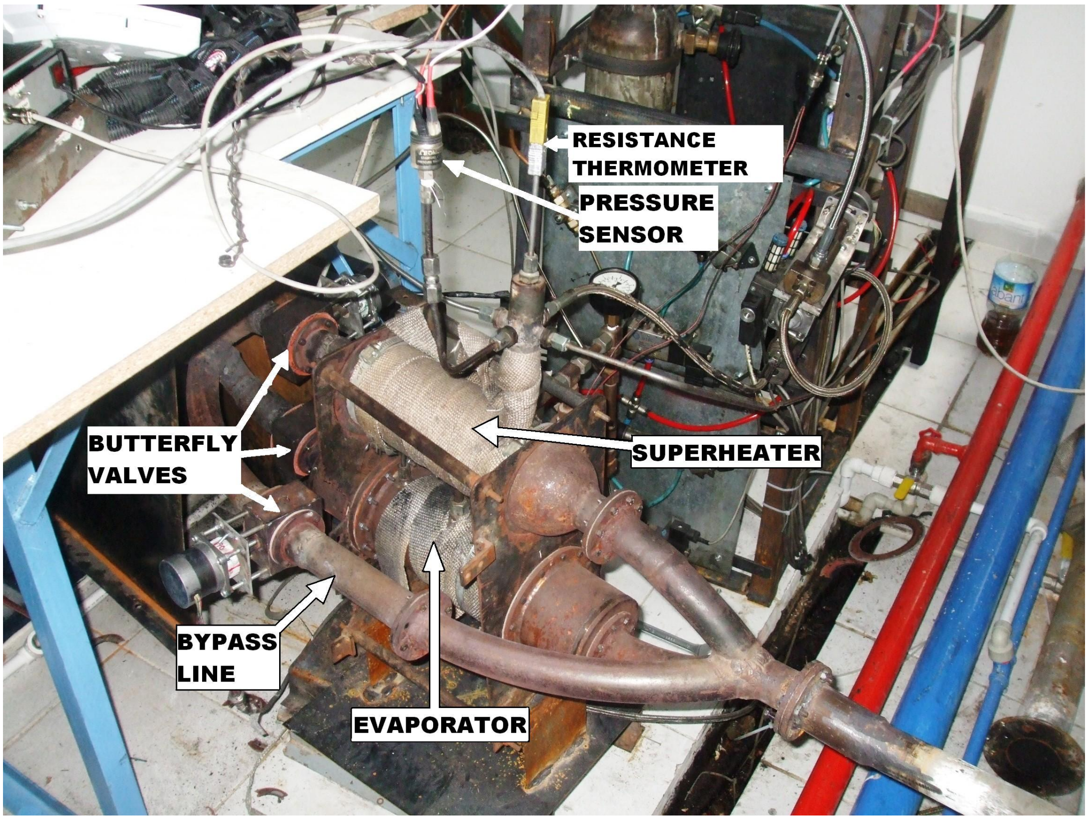


Figure 2.5: Double Heat Exchanger

2.2.2 Pressurized Tube

Pressure inside the heat exchanger is several times higher than atmospheric conditions, therefore it is necessary to force ethanol into the evaporator. This is accomplished by the pressurized tube. As a result, pressurized tube should not be considered as storage for ethanol, but as a pumping device that replaces a fuel pump.

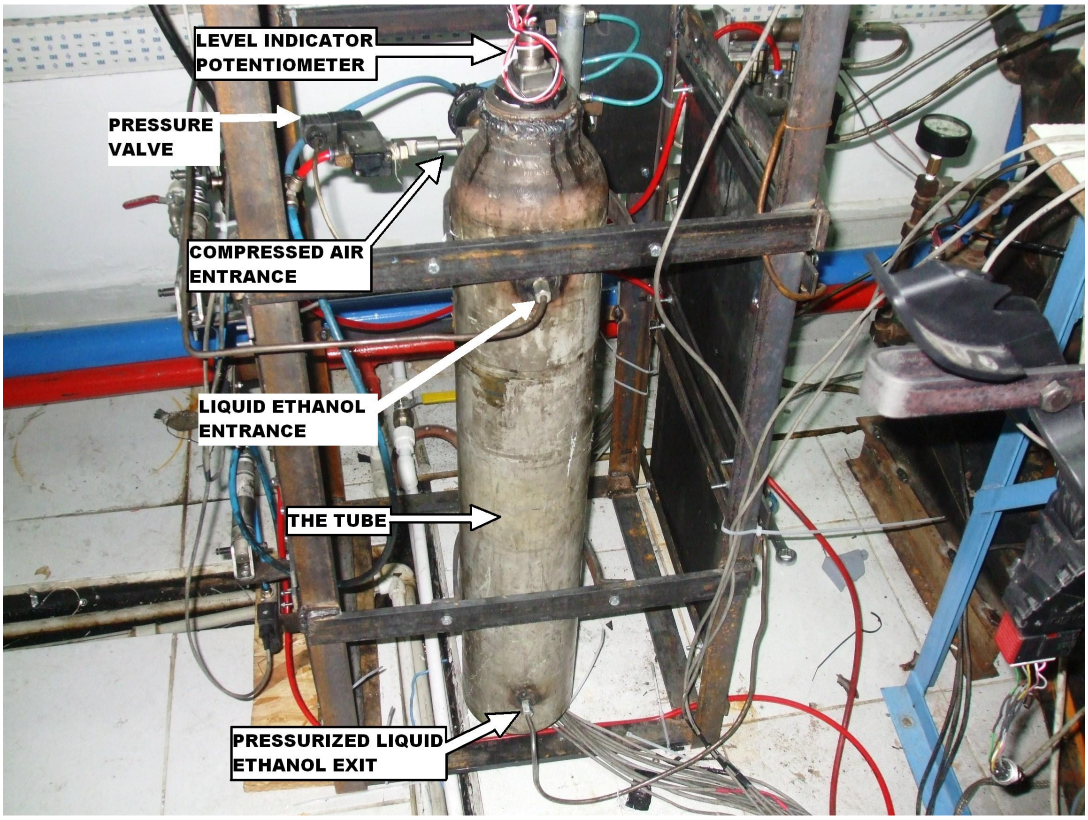


Figure 2.6: Pressurized Tube

In the pressurized tube, pumping effect is obtained by increasing the pressure of the tank by introducing compressed air. If ethanol level inside the heat exchanger drops, pressure valve and heat exchanger feed valve is triggered simultaneously and ethanol is forced into the heat exchanger. After a couple of heat exchanger refills, liquid level inside the pressurized tube drops below predefined lower limit. In this situation, fuel pump and the tube refill control valve are triggered in order to refill the tube. Control strategy of the whole system is designed in a way that heat exchanger refill and tube refill can never coincide.

Three gas fittings located around the tube. Locations of the fittings can be seen on figure 2.6 .

The upper fitting is used for mounting pressurizer valve that either supplies compressed air or releases it from the tube depending on the described situations. The second fitting is located below that of pressurizer valve, where ethanol is fed into the tube if ethanol level inside drops below the allowable lower limit. The last fitting is where ethanol exits the tube, and is located at the lower part of the tube. As the last element¡of the pressurized tube, a floater driven potentiometer is used for monitoring tube level.

2.2.3 Valves And Piping

2.2.3.1 On/Off Control Valves

In this setup, on/off control valves are used for directing ethanol flow through the required routes both in liquid or vapor phase. Ethanol has corrosive effects on most of the metals and their alloys, especially in moving parts. This factor is the reason that the valves used in the system are made of stainless steel, which is resistant against corrosive effects of ethanol in both phases. The only exception for control valves is the solenoid valve that enables or disables flow of compressed air into the pressurized tube.

The valves used in the system are stainless steel spherical valves. Since these components are designed for manual operation, pneumatic pistons are mounted for enabling computer actuation. Each of the pistons are driven by two position five way solenoid valves. The spherical valve, pneumatic piston and solenoid valve form into a single valve unit and are mounted on a steel chassis. The only exception is the vapor consumption valve, actuator of which provides 90°rotary motion and is equipped with electrical parts. Actuation of these units are done by the software over data acquisition board. Including tube pressure solenoid valve. Figure 2.7(a) displays a single unit. There are six control valves installed, and they can be listed as:

1. Tube refill valve: This valve is opened when ethanol pump is activated for filling the tube, and closed as the operation is complete. While refilling does not occur by gravitational force, it is still necessary to prevent from any air leakage when tube is pressurized, so this component is important as a precaution.

2. Heat Exchanger refill valve: This component is used in a similar purpose as the tube

refill valve. This component is kept at closed state, except when the heat exchanger is refilled.

3. Vapor consumption valve: As the computer decides to whether ethanol vapor is in required conditions, vapor consumption valve is opened and ethanol is introduced into the needle valve for adjusting flow rate. While needle valve can stop ethanol flow completely, it operates rather slowly and can not close the system instantly. This is the case when ethanol consumption valve becomes important.

4. Superheated Vapor Return Valve: Pressure inside the heat exchanger rises when evaporation rate is higher than consumption rate. In case of an unexpected response from the system, pressure may rise above safe levels. This valve is used in such cases as a precaution. In such a case, ethanol vapor escapes through the valve into a helical copper pipe into the main ethanol container.

5. Heat Excanger Drain Valve: At the end of an experiment, ethanol is sent to the pressurized tube by the aid of compressed air. Drain valve is used for controlling this process.

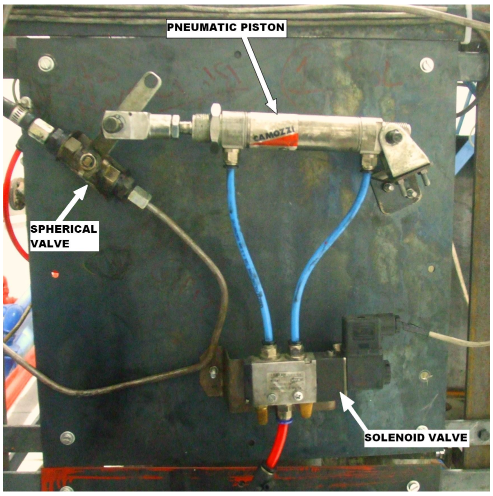
6. Tube Pressure Control Valve: This component is a two position five way solenoid valve that controls flow of compressed air into and out of the pressurized tube. This component is installed onto the pressurized tube, where ethanol is in liquid form and can not rise up to this installation level.

7. Turbinemeter calibration valve: This component is not used during experiments, but is utilized during ethanol vapor flow rate calibration of the turbine meter. This valve assembly with its pneumatic controllers is the only valve that is built for this study, while other valves were available from former studies. The valve manipulated by the pneumatic piston is a three way valve. One end of the valve sends condensed ethanol to a large container. The other end, which is opened during calibration for 66 seconds, sends ethanol to measurement bottles.

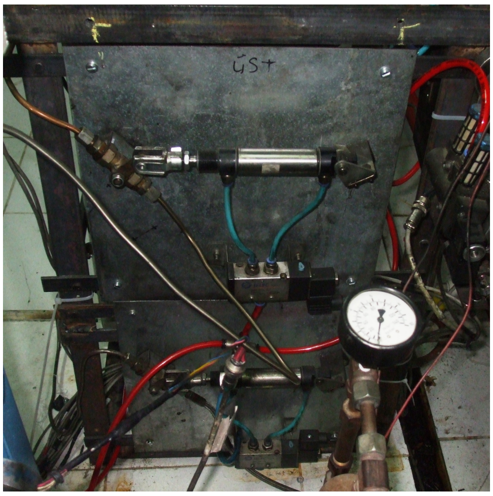
Installation locations of these components can be viewed at figures 2.7(c), 2.7(b) and 2.7(d).

2.2.3.2 Needle Valve

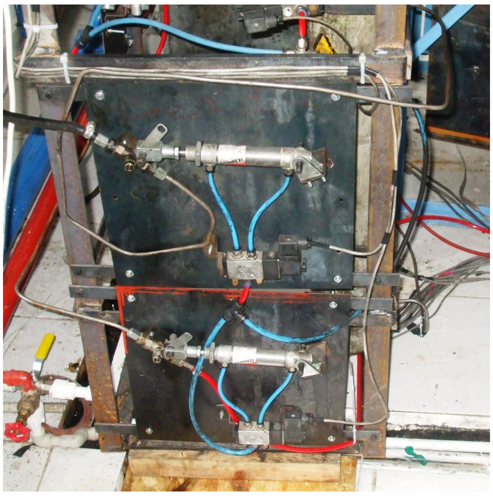
Needle valve is the component that is used for adjusting flow rate of ethanol vapor. Driven by a stepper motor, this component is installed at the location where ethanol exits vapor consumption valve. As an assembly, this component consists of two main parts: the needle valve and a stepper motor. Stepper motor is used in order to be able to control the valve through the computer. Since this specific unipolar stepper motor has an angular sensitivity of 1.8°/steps, flow rate adjustments are done at adequate precision. Picture of the needle valve assembly is available on figure 2.7(d).



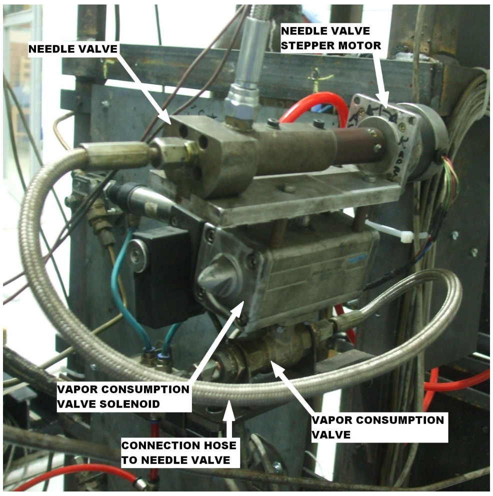
(a) Single Valve Unit



(b) Heat Exchanger Refill and Superheated Vapor Return Valves



(c) Tube Refill and Heat Exchanger Drain Valves



(d) Needle Valve and Vapor Consumption Valve

Figure 2.7: Valves of the System

2.2.3.3 Piping

Ethanol requires care when handling in vapor form because it can cause corrosion on metal parts or can deform derivatives of plastic materials, result of which will be failure in components, hence leakage will occur. For this reason, piping is done using stainless steel pipes and hoses are selected as Teflon. These materials are resistant to effects of ethanol. In addition, piping components in the system can endure high temperatures and pressures. Stainless steel pipes are used for fixed connections where vibration is minimal. On the other hand, Teflon hoses are flexible parts and can serve properly in existence of vibration and can reduce its effects. In addition, thermal conductivity of Teflon hoses are low enough to act against temperature drop through the line. In order to improve thermal insulation, exhaust wrap material is winded around the hoses. It is noteworthy to mention that Teflon hoses are used in wherever superheated ethanol flow is directed to consumption line, regardless of existence of vibration.

2.3 Ethanol Vapor Condenser

Ethanol vapor condenser is utilized for two purposes: calibration of turbinemeter and pressure relief collecting ethanol from the heat exchanger in case pressure rises above consumption conditions.

This device is a shell-and-tube type heat exchanger with copper helical coil, and is built in the laboratory, essentially for calibration of turbinemeter. Superheated ethanol flows through the tube side, and goes through phase change, while water acts as the coolant and flows on the shell side.

2.4 Software, Data Acquisition System and Other Measurement and Control peripherals

A complex system is prone to operational failures where operation of the system depends on human control. In this system, existence of ethanol vapor at high pressure and temperature is a source of hazard itself if not well handled. Control software and hardware holds utmost importance for this reason.

Software and control hardware components serve for the following purposes:

- Measurement of data collection and control signals : In addition to engine performance data including ethanol consumption rate, pressure and temperature of the ethanol vapor inside double heat exchanger are monitored in every test cycle. A test cycle lasts 100 ms.

- Ethanol level inside the heat exchanger and pressurized tube are also critical measurements. In case of overflow, ethanol may spill inside the test room and will cause hazardous situations. On the opposite condition, engine will not be fed with ethanol vapor when needed, and this will cause time loss or may even need restarting the session.

- Command outputs of the system are excelled in sensitivity in comparison to human operation. As an example, software can set the dynamometer speed that will both be safe for the engine and precise for the quality of the collected data. As another example, stepper motors can control the related valves with more precision than a human.

- Last and the most important issue is, software will never miss reading a data or controlling a device. As a result, the system can be maintained within the defined ranges. In case an external interruption causes improper conditions, software can sense it and will take the proper cautions. In extreme conditions, system will stop the experiment and supply the safest conditions in the test environment. This section will describe each of the components that contributes the control system in details.

2.4.1 Data Acquisition Card and Terminal Boards

It has been mentioned before that data are collected by the computer. Data acquisition card is the component which actually reads the data as voltage, digital signal or frequency, and converts it to data that computer can process with. As the the data is processed and output values are decided, signals are again sent by this component except for the stepper motor signals. This card can send digital signals of 5V or analog signals in the range of -10V to 10 V.

Advantech PCI 1716 is used during the experiments. This device has 16 analog and 16 digital input channels and a frequency read channel for data collection, and 16 digital output and two analog output channels. Cable connection to the card is done over Advantech PLCD 8710 terminal board. While this terminal board provides connectors for digital input and output channels, the operations related to these channels are accomplished over another terminal board.

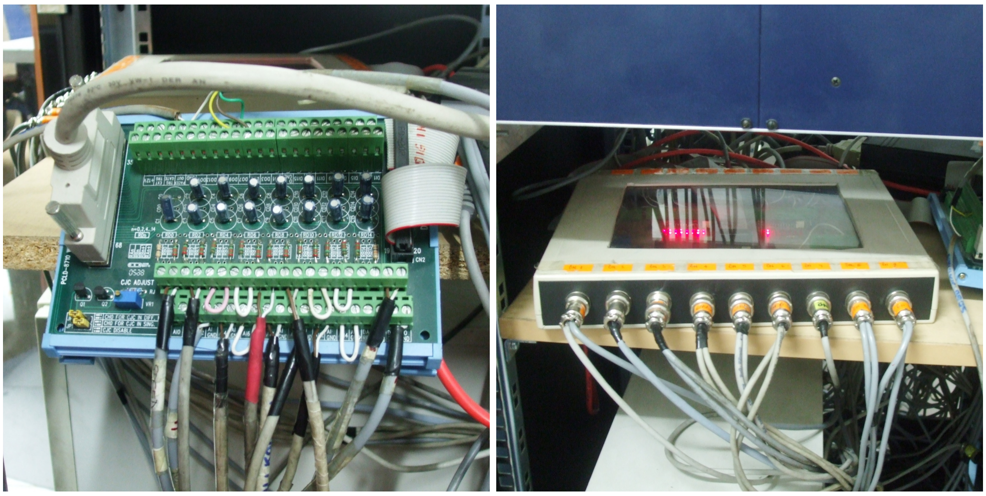
Analog input channels of data acquisition card can read values between -10 to 10V. If a signal source is known to be in a smaller scale, e.g. ranging between 0.5 to 4.5 V, the analog input channel corresponding to that connection can be programmed to measure between a smaller range; which is 0 to 5V for the given example. This is possible since the card is equipped with a scaler. This property is extensively used during the experiment, where all these channels are set to read voltages between 0 to 10 V. Analog input channels can further be programmed to work as differential inputs, but this property is not used in the experiment. For improving the quality of the analog signals, inputs of each channel is filtered by a 1 Hz high pass RC filter utilizing a 10μF capacitor and a 10 kohm resistor. Terminal board provides soldering locations for these components for both single input and differential input modes.

Analog output function of the card is utilized for sending speed setting information to the dynamometer. Working with analog voltages can hold occasional risks, so the signal is buffered by an amplifier circuit. This amplifier circuit has a gain of 1. This card has only one frequency read channel, and is used for mass air flowmeter input. manufacturer reports that this channel can read frequencies from 0 up to 10 MHz. The output range of the flowmeter ranges between 2 kHz at idle to 5.6 kHz at the maximum intake flowrate.

Wiring of analog input and output voltages, and frequency read channel is done over wiring terminal. This board incorporates wire connectors for external connections, a connector for the cable between the data acquisition card, and separate 20 pin connectors for digital input and output channels. These 20 pin connectors are where the connection to the secondary terminal card is done.

Secondary terminal board is used for digital input channels because the setup requires additional features that the card does not provide as a package; such as pull-up resistors for digital input channels or relay connections for digital outputs. Digital input signals may be active or passive low, depending on the source of the signal. If the signal source is passive, it is highly probable that input signals will not be read properly. Adding a pull-up or pull-down resistor are easy and effective solutions and this is one of the reasons that makes use of the secondary terminal board necessary. The second reason is, digital output channels of data acquisition card provide TTL level voltages at low current which is not satisfactory for driving a component that requires high voltage and/or current levels. Such a task can be simply accomplished when a relay is used. Since most of the relays require at least 12V for operation, signals of digital output channels are amplified using transistors. And the last reason is compactness of the system. This component may seem bigger in size in comparison, but most of the electrical connections for driving elements are completed within this component using internal terminals or external sockets, so electrical connections related to digital channels are protected against shorting or failure.

Connection of the elements to the terminal cards and corresponding channels are listed in Appendix A and both terminal boards can be viewed at figure 2.8(a) and 2.8(b).



(a) PLCD 8710 Terminal Board (b) Secondary Terminal Board

Figure 2.8: Terminal Boards

2.4.2 Measurement Devices and Signal Conditioners

2.4.2.1 Thermocouples

Thermocouples are temperature measurement elements which are suitable for measuring high temperatures. These elements are made up of a pair of metal wires of well defined alloys. If these wires are connected at one ends, the resulting combination generates voltage, and measuring it will inform about the measured temperature.

K type thermocouples are used during this experiment. K type thermocouples are made of a chromel-alumel wire pair and this type can measure between -270°C and 1370°C . Since exhaust temperatures can reach up or exceed 600°C, it is necessary to use a compatible thermocouple. K type thermocouples are both widely used and durable at high temperatures, so they are utilized throughout the experiments.

K type thermocouples have a characteristic voltage output of 41μV / °C [32]. Further information about voltage outputs is given at Appendix B.1 . Most of the voltage measurement devices are not capable of sensing such low signal levels, so it is usually necessary to amplify output of thermocouples. In this experiment, thermocouple outputs are connected to the amplifiers that are built in the laboratory.

2.4.2.2 Pressure Sensor

Pressure sensor is utilized for measuring pressure of the ethanol vapor inside the heat exchanger. This pressure sensor is manufactured by Omega and its product code is PX120-200GV. This particular device is offered for use in corrosive media.

This device measures gage pressure. Its output changes linearly in 0 to 50mV range for 0 to 200 psi (13.8 bar) gage pressures. For better measurements, its output is connected to an amplifier with 500x gain. In addition, measured values are calibrated by using a dead weight tester. Another issue about the device is its allowable temperature range for operation. Manufacturer denotes the allowable range as 0°C to 80°C, which exceeds even the lowest temperature inside the heat exchanger during operation. In order to solve this issue, a steel pipe with inner diameter of 6 mm is installed for extending the distance between the sensor and measurement location. Inner diameter of the pipe is small enough to eliminate any fluid movement, so vapor can not reach to the sensor, and heating effects related to temperature of the heat exchanger casing is eliminated by carrying the sensor to a position that it can be exposed to cooling air stream.

Further information about the pressure sensor can be viewed at Appendix B.1.

2.4.2.3 Mass Air Flowmeter (MAF)

Mass Air Flowmeter is a device that measures the mass flowrate of intake air charge. Test engine is equipped with a Bosch HFM 6 type mass air flowmeter, and its output is connected to the data acquisition system. This device provides air flowrate data as frequency that varies between 2 kHz and 10 kHz, where the engine uses its range up to 6 kHz. In this range, frequency output is a linear function of air flowrate.

This device is used by the engine itself, so there were no modifications made. On the other hand, neither manufacturer of the mass air flowmeter nor the engine manufacturer provides any catalog information or calibration data. That's why calibration of output frequency vs. flow rate is done by Go-Power Systems M-5000 air-fuel meter. The results of calibration provided the function of mass flowrate of air vs. frequency output of the flowmeter.

Picture of this component is available in figure 2.9(a).

(a) HFM 6 Mass Air Meter (b) Manifold Absolute Pressure Sensor

Figure 2.9: Original Equipment Sensors of the Engine Used During Experimental Measurements

2.4.2.4 Manifold Absolute Pressure Sensor (MAP)

MAP is the device that measures pressure of intake charge. Especially in turbocharged engines, it is necessary to measure the pressure at the intake manifold. This component is manufactured by Bosch, and its technical details can be viewed at Appendix B.3. Its output is analog voltage ranging from 0.4V to 4.65V. These values correspond to 20 kPa and 300 kPa absolute pressures, and change in the output voltage versus pressure is linear.

Like the mass air flowmeter, pressure sensor is installed on the engine and used by engine control unit(ECU). That's why, there were no modifications done on the sensor or the cabling. The only addition is measurement cables that connect data acquisition card and this pressure sensor. Picture of the component is given in figure 2.9(b).

2.4.2.5 Stepper Motor Controller

In the system, exhaust valves and needle valve is operated through stepper motor controller. This device is designed exclusively for unipolar stepper motors and built in the laboratory. It has four channels, which is sufficient considering the number of channels needed. Core of this device is PIC 16F84A microcontroller which accepts command from the computer through serial port and adjusts the positions of stepper motors. Circuit works with 5V supply.

2.4.2.6 Amplifiers

Amplifiers are used for altering signal levels and properties. In this study, these devices are used for amplifying signal levels of measurement devices, namely thermocouples and pressure sensor, to meet the measurement range of data acquisition card. These devices are designed by Prof. Dr. A. Demir Bayka, and manufactured in the laboratory.

These amplifiers are calibrated by adjusting pre-amplification zero, amplification zero and gain potentiometers. It is important to adjust pre-amplification voltage in order to ensure that the device does not generate any additional voltage to the amplified signal. Once this condition is ensured, amplification zero is adjusted. The second step is necessary to adjust the output of the device to 0V when the measured element is not excited. The final step is adjustment of gain level, and this step defines the amplification ratio. Once the adjustment is complete, user can select operation mode as either thermocouple or normal mode. While no additional voltage manipulation is done at normal mode, thermocouple mode sends the signal to an additional operational amplifier in order to add the voltage that corresponds to the ambient temperature to the measured one, and then the signal is amplified. Once data acquisition card reads the input voltage, corresponding temperature is found by the software based on K type thermocouple properties table. This table can be found at Appendix B.1. Thermocouple mode is used exclusively for amplifying thermocouple inputs.

This device is supplied with ± 15 V, ± 5 V and a ground voltages from amplifier power supply.

This power supply is purpose built for amplifiers since no commercial products are available with this configuration. This unit is equipped with a 90 W transformer to convert 220 V 50 Hz residential mains to 18 V alternating current. After this conversion, the current is rectified and voltages are adjusted by the related regulator and sent to the connected amplifiers.

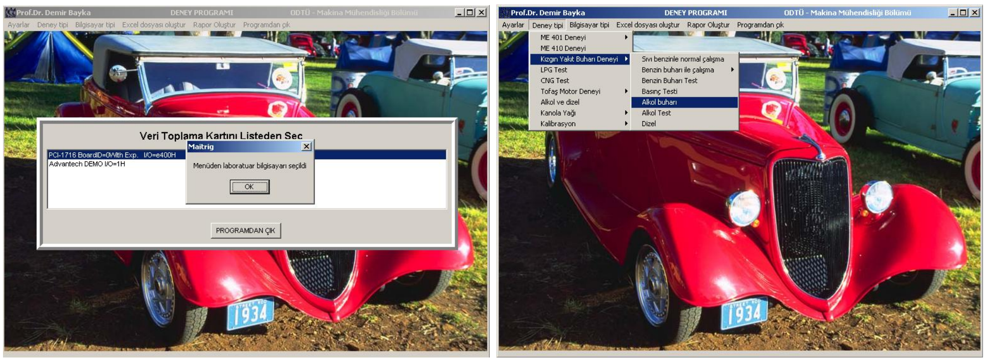
Figure 2.10 shows appearance of the component with thermocouple, signal and power connection cables.

Figure 2.10: Amplifier and Thermocouple

2.4.3 Software

All the components listed until now are controlled by the software. This software is coded in Delphi 4 and every single unit that is used in the laboratory is supported by the software.

After the execution of the software, initialization of the peripherals begin. At the beginning, user is asked to select the data acquisition device. There are two options as the card itself and its demo.Demo option is used in the computers that are not equipped with the card itself, but are used for editing the codes, such as personal computers or office computers that are not used in the experiment. On the other hand, user needs to select the card instead of demo for running the actual experiment.



(a) Card Selection Screen (b) The Main Page and Experiment Type Selection

Figure 2.11: Card Type Selection Interface and the Main Page

After the selection, the main page appears where the researcher can give commands to or read data from any connected device, select the test type, convert collected data into an excel file, generate a report based on a previous results file or can use specialized calibration tools provided for several measuring devices. In the user selects a test type, referring interface is initiated from this page. In the experiments, the option ”fuel vapor” is selected from the menu. Both card selection screen and main page are shown in figure 2.11.

Once the fuel vapor page is selected, the actual experiment procedure begins and related files are activated. On this page, user is provided with all the necessary tools in addition to the experiment interface. The experiment interface accepts throttle position, dynamometer set speed, needle valve position and atmospheric conditions, and data record inputs from the user. The rest of the visual elements are added for monitoring the system with ease. Figure 2.12 shows the screen where these visual elements are displayed and they can be listed as:

- on/off states of the solenoid valves: The system operates without any faults during the experiment, but it is suggested to keep the system in observation. The software is prepared for the foreseen conditions and possible failures. For this reason, operator of the setup should be well informed about the condition of solenoid valves, since these elements are the major components that make the system.

- Engine speed, torque, air intake, manifold pressure and fuel consumption measurements: These are the basic parameters for evaluating performance of an engine.

- position of the stepper motors, throttle, dynamometer setting and needle valve: These elements are adjustable components of the system. While the stepper motors are automated in default settings, user can switch to manual control mode if necessary. In order to adjust these elements, user should be well informed before deciding on the level and direction of the adjustments.

- calculated values of measurements and control parameters: These values are the actual ones that will be saved when the user decides to collect the data.

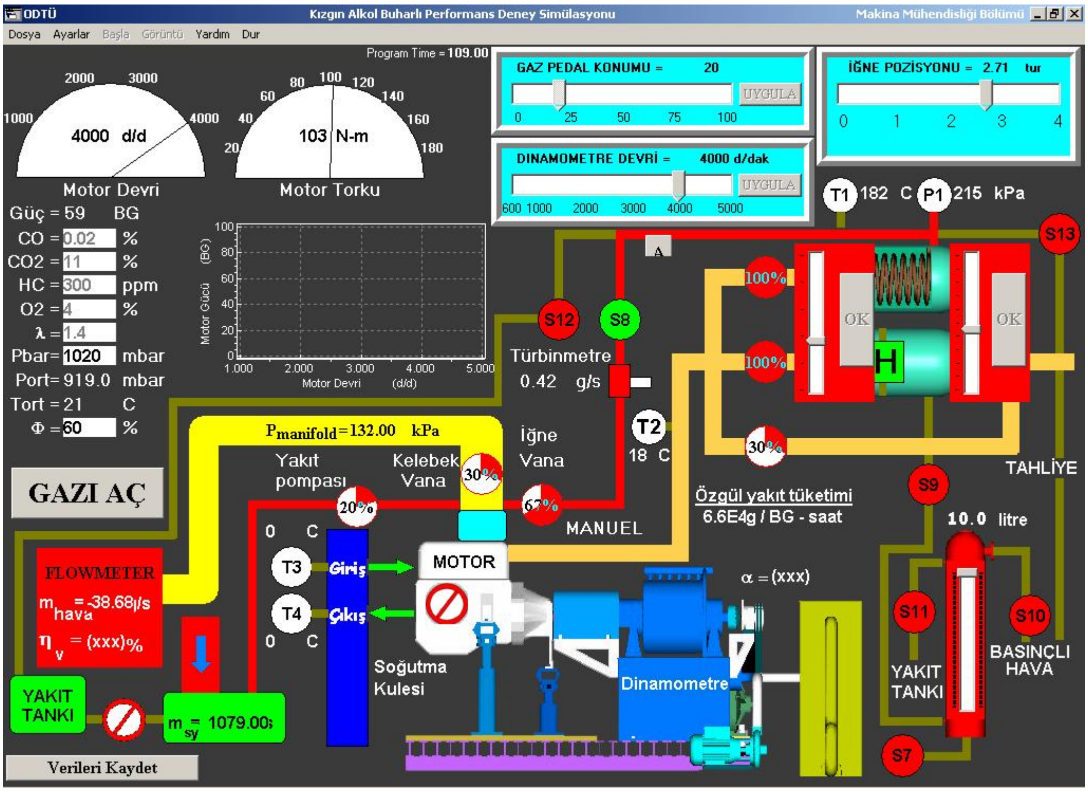


Figure 2.12: Card Selection Screen

As the user is informed about the decisions and measurements, automated part of the software works for keeping the system in the state where physical properties of ethanol are suitable for consumption. This control procedure begins with adjusting the exhaust valves for superheated ethanol test.The next step is filling the pressurized tube and heat exchanger respectively, and these two operations are given priority over all the other controls at any time they are invoked. Once these components are filled, the software begins running in the test timer.

Test timer is the software procedure that follows the control routine in every 100 ms, and that interval is the length of a single cycle. During a single cycle, data is collected from every measurement device and processed. Depending on the values calculated, the program decides to execute a suitable one from the following conditions:

- Fuel level inside the pressurized tube is measured. If the level is below the minimum allowable value, the timer that controls tube filling loop is initiated. If the level is acceptable, other parameters are checked.
- Heat exchanger floater switch is checked, and fuel is pumped if necessary.
- Temperature and pressure inside the heat exchanger is measured. based on these readings, stepper motors are driven. The basic idea behind the controls is to maintain the vapor properties within a reasonable range by adjusting flow of exhaust gases into the related heat exchangers. The priority is on evaporator heat exchanger, since the pressure of the system depends on evaporation rate of ethanol. If the measurements indicate pressure drop, the butter fly valve at the evaporator entrance is opened to let more exhaust gas enter, or the opposite is applied for high pressures. the second parameter is the temperature, which is affected by the amount of exhaust gas that enters superheater, and position of the exhaust valve is decided based on this reading. If some portion of exhaust gases are considered unnecessary, that portion is directed to bypass route and the adjustment is done over bypass valve.
- If ethanol condition is suitable for feeding the engine, then consumption valve is actuated and vapor is allowed to flow through. While turning on or off is decided by the software, flowrate is controlled by the user only.
- If the temperature or pressure inside the heat exchanger rises over safety limits, release valves are opened and ethanol vapor is sent to the container.
- At every cycle, program displays the measurement values based on voltage readings. These values are recorded into a text file if user presses record button.

This software is a part of the experimental software used in internal combustion engines laboratory, and is used during former experiments within the laboratory. Core functions of the software and the user interface was available since the beginning of the study. Still some additional functions and procedures were required in order to perform the experiment and collect the data, hence some modifications and additions were made on the software. The modifications and additions can be listed as following:

- The functions that are used for adjusting needle valves are optimized for exhaust gas temperature, ethanol pressure and temperature inside the tube.
- Valve control procedure was coded for releasing throttle pedal when pressure and temperature of ethanol is suitable for consumption. This detail is removed from the code.
- Calibration data of the measurement devices are arranged, and that of newly added components are introduced.
- Frequency reading procedures were available in the software, but has not been used until this study. Related codes are debugged and utilized for mass air flowmeter.
- Calibration procedures and user interface elements of turbine meter and mass air flowmeter are added to the software.

During this chapter, all the details of the experimental setup are provided, and the next chapter will explain how the test is performed step by step.

CHAPTER 3

EXPERIMENTAL METHOD AND TEST PROCEDURE

Experiment procedure is planned in order to obtain a data set that will inform about the performance of the engine throughout its operating range. This approach covers speed ranges between 4000 rpm to 1500 rpm, which is suggested by the manufacturer, and load conditions between full load and 25% of the full load. This chapter will inform about properties of the test fuels and test routine in details.

3.1 Properties of Test Fuels

Test fuels of the experiment are diesel fuel and ethanol. The manufacturer strictly suggests that this engine is designed for ultra-low sulphur diesel. This type of diesel fuel is known as eurodiesel in European market, and is the one that fuelled the engine in this study. The second fuel is anhydrous ethanol that has purity level of 99.7% by volume. The remaining fraction of 0.3%, either water or residual chemicals after distillation and other treatments, is unknown. Since the residual fraction is negligible, this fuel is treated as pure ethanol throughout the calculations. Properties of the fuels are given in table 3.1.

The properties given in the table indicate the differences between these two fuels. It is known that fuels with less carbon content and more oxygen content in its molecular structure are favorable regarding exhaust emissions, and this is a strong point for ethanol. On the other hand, properties of ethanol are not compatible with diesel engines due to its low cetane number. Cetane number indicates ignition delay, and fuels with higher cetane number yield to better performance. In addition, self ignition temperature of ethanol is higher than that of diesel fuel. Finally, heating value of ethanol is lower than diesel, whih is another reason for

Table 3.1: Properties of Test Fuels

		Diesel	Ethanol(anhydrous)
Chemical Formula		$C_{10.8}H_{18.7}$	C_2H_5OH
Carbon Content	(wt%)	84	52
Hydrogen Content	(wt%)	14	13
Oxygen Content	(wt%)	N/A	35
Cetane Number		55	7
Lower Heating Value	kJ/kg	42600	28800
Density	g/cc (20°C)	0.82	0.79
Viscosity	cS (20°C)	4,50	1,52
Flash Point	°C	58	9 - 11
Auto-Ignition Point	°C	177-329	422

incompatibility issue. However, ethanol is not injected into the combustion chamber as diesel fuel. It is rather fed through the intake manifold in dual fuel mode, and these problems are expected to be solved since cetane number and self ignition related delays are expected not to occur. In addition, difference in viscosities will not affect engine performance since ethanol is not introduced to fuel lines or injectors.

3.2 Data Points and Important Test Parameters

Scope of the study is to evaluate applicability of the proposed system, so it is essential to collect the data that can be used for comparing the performance characteristics of the engine for both modes. In order to evaluate the applicability, the maximum amount of ethanol that the engine can consume is decided as the fundamental parameter. During the preliminary studies, it has been observed that the system can't be kept in steady state below 2000 rpm at 50% of load conditions due to exhaust gas flowrate and exhaust gas temperatures; that's why the mentioned point will be the lower limit during the experiments. 100% load condition is the natural higher limit for the engine, thus the data points. In addition, manufacturer's suggests the engine to be operated below 4000 rpm. Considering these limitations,table 3.2 can be obtained.

The order of the data points and data set will be strictly followed throughout the experiments. In the first phase of the experiments, all the performance data of the engine for diesel fuel will be collected. Within these data, torque values hold the utmost importance, because they define

Table 3.2: Test Points

data #	Diesel			Dual Fuel		
	100%	75%	50%	100%	75%	50%
1	4000 rpm	4000 rpm	4000 rpm	4000 rpm	4000 rpm	4000 rpm
2	3750 rpm	3750 rpm	3750 rpm	3750 rpm	3750 rpm	3750 rpm
3	3500 rpm	3500 rpm	3500 rpm	3500 rpm	3500 rpm	3500 rpm
4	3250 rpm	3250 rpm	3250 rpm	3250 rpm	3250 rpm	3250 rpm
5	3000 rpm	3000 rpm	3000 rpm	3000 rpm	3000 rpm	3000 rpm
6	2750 rpm	2750 rpm	2750 rpm	2750 rpm	2750 rpm	2750 rpm
7	2500 rpm	2500 rpm	2500 rpm	2500 rpm	2500 rpm	2500 rpm
8	2250 rpm	2250 rpm	2250 rpm	2250 rpm	2250 rpm	2250 rpm
9	2000 rpm	2000 rpm	2000 rpm	2000 rpm	2000 rpm	2000 rpm

the load condition. As an example, engine produces 138 Nm of torque at 4000 rpm at full throttle condition, which is 100% load condition for that engine speed. Further proportioning these values, corresponding torque values can be obtained at this speed. Once tests for diesel fuel are completed, the same load conditions will be met in dual fuel mode at the same speeds and torque values.

During dual fuel tests, the percentage of diesel fuel replaced by ethanol will be the subject of interest rather than maximum torque output. As aforementioned, engine knock is expected during the tests, which is the major limiting factor and the deciding parameter for applicability. This will also decide the maximum rate of ethanol consumption at a given test point. In order to achieve this, consumption of diesel fuel will be reduced while more ethanol is introduced until knocking starts. The point when the knocking starts will denote maximum ethanol consumption rate.

In addition to torque values, exhaust gas temperature will also be considered as an important parameter during dual fuel tests. Due to the design of diesel fuel measurement system, diesel fuel flowrate is not updated in every cycle. In case proper equivalence ratio is not met, it will yield to either an increase of combustion temperature or incomplete combustion. While the second case will cause deviation in ethanol flowrate and affect results, the first case may cause mechanical damage to the components of the engine due to overheating. In order to avoid such undesired results, the recorded exhaust gas temperature values during diesel tests will be used as upper limits of this parameter in dual fuel tests.

3.3 Test Procedure

Engine performance is tested at engine speed range 4000 rpm and 2000 rpm with 250 rpm intervals for 100%, 75% and 50% conditions. Before introducing ethanol , performance characteristics of the engine is recorded when the engine ran with diesel fuel only. On the next step, ethanol is generated and introduced to the engine and a new set of four performance data are collected. All the controls of the experimental setup is aided by the software during test sessions. These procedures are prepared in order not to exclude a single detail, so the researcher can successfully operate the system without skipping any important detail.

3.3.1 Constant Load Variable Speed Performance Tests in Diesel Mode

1. Check the level of ethanol inside the ethanol tank.
2. Check the level of diesel fuel inside the diesel tank
3. Check the water level of the engine coolant water radiator.
4. Switch on the pump that supplies coolant water to the engine radiators and dynamometer from the cooling tower
5. Switch on the cut-out switch that enables electric lines of the devices.
6. Turn on the switches that control electric supply to the controller and coolant feed pump of the dynamometer.
7. Check all the power supply units.
8. Start the computer.
9. Once the computer is logged in and ready to use, execute the experimental software.
10. On the first screen, select the card type as ”PCI 1716”.
11. Check digital output channels 1 and 2. These outputs trigger external control options of the dynamometer. Repeat all the steps beginning from step 4 if these outputs are not activated.
12. Check the positions of heat exchanger stepper motors. Stepper motors should be at the following positions

- Bypass Valve : 100 %
- Evaporator Valve : 0%
- Superheater Heat Exchanger Valve : 0%

13. Correct the positions of stepper motors if any of them is not in the correct position. This step can be skipped if the positions are correct.

14. Run the experiment program.

15. Type atmospheric pressure and relative humidity values to the relevant boxes.

16. Create a new file and type information about the engine. These information can be found at "Test Engine" section of Chapter 2

17. Press start button to initiate test routine. Program will first check ethanol level inside the pressurized tank and heat exchangers, and will not allow any input until they are filled.

18. Program will adjust heat exchanger butterfly valves for ethanol vapor generation, but their positions are needed to be as defined in step 12. Set the control of exhaust valves to manual mode and adjust to the denoted positions from the software.

19. Start the engine.

20. Change the throttle position to 25% and wait until the engine reaches at steady state.

21. Wait until the engine reaches at working temperatures.

22. Switch on the fan to blow air onto the engine at ambient temperature.

23. Set the dynamometer to 2000 rpm.

24. Change the throttle position to 50% and wait until the engine reaches at steady state.

25. Set the dynamometer to 3000 rpm.

26. Change the throttle position to 75% and wait until the engine reaches at steady state. Once the engine stabilizes, set the throttle to 100%.

27. Set the dynamometer to 4000 rpm.

28. If experiment is done in partial load conditions, decrease throttle position until the required torque value is observed.

29. Wait until the system reaches at steady state

30. Press record data button.

31. Decrease dynamometer speed by 250 rpm

32. Repeat steps 29 and 31 until data of 2000 rpm is collected.

33. If all the data are recorded successfully, reduce the throttle position to 0%.

34. Wait until the engine cools down to safe temperatures.

35. Press stop button

3.3.2 Constant Load Variable Speed Performance Tests with Superheated Ethanol

1. Do the steps between 1 and 27 of previous section. The only step that should be skipped is step 18 since ethanol is used in this mode.

2. Wait until ethanol is superheated and ethanol consumption valve is opened.

3. Once desired vapor conditions are met, reduce throttle position until engine torque drops below the torque value that will be measured.

4. increase needle valve opening by small increments until required engine torque is met. If engine knocking is observed, stop increasing needle valve.

5. If further torque increase is required, make the adjustment using throttle.

6. repeat steps 3 and 5 until the desired engine torque is observed. The torque value that should be observed is already obtained during diesel mode sessions.

7. Decrease dynamometer speed by 250.

8. Remove throttle pedal spacers

9. Repeat steps 2 to 8 until data is recorded at 2000 rpm.

10. If all the data are collected, reduce needle valve position to 0 turns and throttle pedal position to 0%.

11. Wait until ethanol inside the heat exchanger cools down until ethanol consumption valve is closed. This will ensure that there will be no ethanol left inside the intake manifold.

12. Once the conditions of the previous step are met, press stop button.

CHAPTER 4

CALCULATION PROCEDURE

Experiments are performed by following the procedure that was explained in the previous chapter. During the experiments, al the data that identifies the characteristics of the engine is collected. The collected data is processed by the software, and recorded into a text file. This chapter explains how data is processed into actual parameters. Each parameter and their variables are explained in the related section.

4.1 Calculation of Atmospheric Conditions

Atmospheric pressure, relative humidity (RH) and ambient temperature are recorded during tests and used in calculations. Atmospheric pressure is obtained in mbar, corrected for elevation of test location Pressure of dry air and several correction factors are then calculated using these variables. In addition, pressure inside the manifold and turbocharger pressure ratio are common variables throughout the calculations. These parameters appear frequently throughout this chapter, so they are introduced as basis. These variables are calculated by the following formulae:

$$P_{atm} = \frac{P_{mbar} - 101}{10} \tag{4.1}$$

$$P_d = P_{atm} - \frac{RH \times P_v}{100} \tag{4.2}$$

$$P_d = P_{atm} - \frac{RH \times P_v}{100} \tag{4.3}$$

$$P_v = \left(\frac{101.325}{760}\right)\left(6.10708 \times 10^{\frac{7.5\times T_{amb}}{237.3+T_{amb}}}\right) \times 0.75 \tag{4.4}$$

P_{mbar} : Atmospheric pressure measured in mbar
101 : Atmospheric pressure correction factor for Ankara
P_d : Pressure of dry air, kPa
P_{atm} : atmospheric pressure, kPa
P_v : Vapor pressure in air, kPa
RH : Relative humidity, in percent (%)
T_{amb} : Measured temperature in °C

4.2 Calculation of Parameters from Input Data

It is stated in Chapter 2 that measurements are done by the computer. Once data is collected as either analog voltages or as frequency, it is processed into actual data. In this section, the formulae used for processing these inputs into actual parameters are explained. The order of formulae is the same as the connection channel order of the elements. Connection list can be found in appendix A.

Ambient temperature in the test media is measured using the temperature sensor that is integrated onto the primary terminal board, and the value is calculated by the following relation:

$$T_{amb} = (V_{T,amb} - 273) \times 100 \tag{4.5}$$

T_{amb} : Ambient temperature in the test media
$V_{T,amb}$:Voltage value read from the temperature sensor
100 : Factor for converting voltage value into Kelvin (K)

Engine speed is obtained from the controller of the dynamometer as analog signal and calculated as :

$$\theta = V_\theta \times C_\theta \tag{4.6}$$

$$C_\theta = 1000\left(\frac{rpm}{V}\right) \tag{4.7}$$

θ : Engine speed in rpm

V_θ: Voltage input from the dynamometer

C_θ : Conversion factor from volts to rpm; such as 1000 rpm corresponds to 1V

Engine torque is also obtained from the controller of the dynamometer as analog signal and calculated as :

$$T_{meas} = C_{torque} \times V_{torque} \tag{4.8}$$

$$C_{torque} = 40\left(\frac{Nm}{V}\right) \tag{4.9}$$

T_{meas} : Measured Torque in Nm

V_{torque} : Measured value of torque as DC voltage.

C_{torque} : Conversion factor from volts to Nm, 1V corresponds to 40 Nm.

Manifold absolute pressure is obtained from the MAP sensor of the engine. Software relies on the calibration curve that manufacturer of manifold absolute pressure (MAP) sensor provides during calculations. Using the given data, the following relation is obtained:

$$P_{manifold} = 65.8823 \times V_{MAP} - 6.3529 \tag{4.10}$$

$$r = \frac{P_{manifold}}{P_{atm}} \tag{4.11}$$

$P_{manifold}$: Manifold Pressure, in kPa

V_{MAP} : Manifold absolute pressure sensor output, in Volts

r : Pressure ratio between charge cooler outlet and ambient pressure

Exhaust gas temperature is obtained from the amplified voltage of thermocouple output. After reading this value, it is divided by the gain factor and the actual voltage at thermocouple wire ends is obtained. Finally, this value is compared to the standard output values listed in appendix B.1. The results of these calculations give $T_{exh.tc}$ value. The following formula explains how this data is processed into actual thermocouple voltage.

$$V_{exhtc} = \frac{V_{exhaust}}{G_{exhaust}} \tag{4.12}$$

$V_{exh.tc}$: Actual voltage of exhaust thermocouple, in V

$V_{exhaust}$: Voltage reading from the amplifier

$G_{exhaust}$: Gain value of the connected amplifier, 274.5x

Throttle position may not be a value that is included in calculations, but it is an important control parameter for the user. This value is obtained from the original equipment throttle pedal of the engine. Manufacturer supplies throttle position vs. voltage output data, and it was used for obtaining the following relation:

$$TP = 1000 \times \frac{V_{throttle} - V_{throttle_min}}{V_{throttle_max} - V_{throttle_min}} \tag{4.13}$$

The given relation is a linear interpolation, and the result is calculated as percentage.

TP : Throttle position as percent(%)

$V_{throttle}$: Measured throttle voltage, in V

$V_{throttle_min}$: Reference throttle voltage for maximum position, 4V

$V_{throttle_max}$: Reference throttle voltage for minimum position, 1V

As in case of throttle pedal, tube level is a control parameter and is measured by a floater driven potentiometer. The level of the tube is calculated as:

$$TubeLevel = C_{tube} \times V_{tube} \tag{4.14}$$

The given relation is a linear interpolation, and the result is calculated as percentage.

$TubeLevel$: Volume of ethanol inside the tube in measurement range, liters

$V_{throttle}$: Measured throttle voltage, in V

C_{tube} : Coefficient for converting voltage input into liters, l/V

Flowrate of ethanol is measured using turbinemeter, which was explained in the previous chapter. In addition, pressure inside the turbinemeter is accepted as manifold pressure, and the temperature as ethanol temperature inside the heat exchanger. Basis of these assumptions are explained in Chapter 2. The formulae used for obtaining mass flowrate of ethanol is as follows:

$$\dot{m}_{vapor} = (0.4705 \times V_{turbine} + 0.0041) \times \rho_{vapor} \tag{4.15}$$

$$\rho_{vapor} = \frac{P_{manifold} \times M_{ethanol}}{(T_{cons} + 273) \times R_u} \tag{4.16}$$

$\dot{m}_{vapor}$: mass flow rate of ethanol vapor, in g/s

$V_{turbine}$: Voltage reading from turbinemeter

ρ_{vapor} : Density of ethanol vapor calculated from ideal gas equation

$P_{manifold}$: Manifold pressure in kPa

T_{cons} : Temperature of ethanol vapor at gas mixer inlet

Temperature of superheated ethanol vapor is obtained using the resistance thermometer. This device is connected to an amplifier. The amplifier used for this device is operated at signal amplification mode and its gain is adjusted to 100x. Temperature versus output voltage is calibrated using hot and cold waters of known temperature. This temperature can be calculated using the following relation:

$$T_{vapor} = \left(V_{T,vapor} - 0.0372\right) \times 56.875 \tag{4.17}$$

$V_{T,vapor}$: Resistance thermometer amplifier output voltage, in V

T_{vapor} : Ethanol Vapor Temperature, in Kelvin (K)

0.0372 and 56.875 are constants that are obtained from calibration

Pressure of superheated ethanol vapor is obtained from the pressure sensor. This device is connected to an amplifier in order to widen the output range from 50mV to 10V. Read values are then calibrated using a dead weight tester. The result of this calibration procedure yields to the following relation:

$$P_{vapor} = 126.45 \times V_{vapor} + 24.806 \tag{4.18}$$

P_{vapor} : Vapor Pressure, in kPa

V_{vapor} : Amplified voltage of the pressure sensor, in V

During turbinemeter calibration, pressure and temperature of superheated ethanol at turbinemeter exit is recorded. Pressure data is collected using Omega PX03K1-015G5T. The highest pressure range is selected. Voltage output of the device changes linearly with pressure between 0-5 V, so the following linear calibration curve can be obtained :

$$
\begin{aligned}
P_{cal} &= V_{vapor} \times \frac{101.325kPa}{14.7psi} \times \frac{15psi}{5V} \\
&= 3 \times V_{cal} \times \frac{101.325}{14.7}
\end{aligned}
\tag{4.19}
$$

P_{cal} : Vapor Pressure, in kPa

V_{vapor} : Amplified voltage of the pressure sensor, in V

Temperature of superheated ethanol at gas mixer inlet is measured using a thermocouple, and is calculated just like exhaust gas temperature. The only difference is gain settings of the amplifiers. This device is amplified 500x, and thermocouple voltage is obtained by:

$$V_{cons,tc} = \frac{V_{cons}}{G_{cons}} \tag{4.20}$$

$V_{cons,tc}$: Actual voltage of exhaust thermocouple, in V

V_{cons} : Voltage reading from the amplifier

G_{cons} : Gain value of the connected amplifier, 500x

Finally, charge air mass flowrate is measured using the mass air flowmeter. It is observed that this device generates stead 2000 Hz output voltage when engine is not running and there is no air flow. Unfortunately, there was not calibration data available for this device, so it is calibrated as explained in Chapter 2. After the calibration, the following relation is obtained:

$$\dot{m}_{air,actual} = 0.0167 \times F_{MAF} - 38.901 \tag{4.21}$$

Parameters of the equation are :

$\dot{m}_{air,actual}$: Actual air flowrate in g/s

F_{MAF} : Frequency input from mass air flowmeter, in Hz

4.3 Engine Torque

Performance characteristics of an engine is identified based on its performance at sea level and dry air. Since the actual conditions do not meet the required condition, these values are corrected based on several standards. During the calculations, ISO/DIS 1585.2 is used in this scope. The following relations explain how torque value is corrected.

$$T_{corr} = \alpha_c \times T_{meas} \tag{4.22}$$

Parameters of the equation are:

T_{corr} : Corrected torque

α_c : Torque Correction Factor for Compression Ignition Engines

T_{ref} : Corrected Torque for sea level elevation and dry air

$$\alpha_c = (f_a)^{f_m} \tag{4.23}$$

Where

f_a : Atmoshperic Factor

f_m : Characteristic parameter for engine type

These parameters are calculated as follows:

$$f_a = \left(\frac{P_{d,ref}}{P_d}\right)^{0.7} \times \left(\frac{T_{amb} + 273}{T_{ref}}\right)^{0.7} \tag{4.24}$$

$$f_m = 0.036 \times q_c - 1.14 \tag{4.25}$$

$$q_c = \frac{q}{r} \tag{4.26}$$

$$q = \frac{Z \times \dot{m}_{fuel}}{V \times \theta} \tag{4.27}$$

$P_{d,ref}$: Reference dry air pressure, 99 kPa

T_{ref} : Reference temperature, 298 K

P_d : Measured dry air pressure in kPa

T_{amb} : Measured temperature in °C

q : Fuel delivery parameter

r : ratio between absolute static pressure at the end of charge cooler and ambient pressure

$\dot{m}_{fuel}$: fuel consumption rate in g/s

V : engine displacement in liters

Z : 120,000 for 4 stroke engines

4.4 Engine Power

Engine power is a function of engine speed and torque. Using these two parameters, power is calculated as:

$$P_{corr} = T_{corr} \times \theta \times 1.36 \times \frac{\pi}{30000} \tag{4.28}$$

In this equation, the parameters are:

T_{corr} : Corrected torque

θ : Engine speed in rpm

1.36 : conversion factor from kW to hp

$\pi/30000$: conversion factor from W-rev/min to kW-rad/s

4.5 Fuel consumption

Diesel fuel and ethanol are the test fuels, and their flow rates are calculated separately since they are not mixed and they are measured in separate devices. Consumption rate of diesel fuel is calculated using a fixed volume container and fto sensors that indicate upper and lower levels of the container.Flow rate of diesel fuel is obtained using the following formula:

$$\dot{m}_{diesel} = \frac{V_{container} \times \rho_{diesel}}{t_{consumption}} \tag{4.29}$$

Parameters of the formula are: $\dot{m}_{diesel}$: Mass flowrate of diesel fuel in g/s

ρ_{diesel} : Density of eurodiesel fuel, 0.827 g/cc

$V_{container}$: Volume of measurement container, 142.24 cc

$t_{consumption}$: consumption time of the fuel inside the container

Mass flowrate of ethanol vapor was explained in the first section. Using both fuel mass flowrate data, total consumption rate is calculated as:

$$\dot{m}_{fuel} = \dot{m}_{vapor} + \dot{m}_{diesel} \tag{4.30}$$

$m\dot{_{fuel}}$: total fuel consumption rate, in g/s

4.6 Volumetric Efficiency

Theoretical air flowrate and volumetric efficiency are found by the following formulae:

$$\dot{m}_{air,theoretical} = \frac{V_{swept} \times \theta \times \rho_{atm} \times r}{Z \times 60} \tag{4.31}$$

$$\eta_V = 100 \times \frac{\dot{m}_{air,actual}}{\dot{m}_{air,theoretical}} \tag{4.32}$$

$\dot{m}_{air,theoretical}$: Theoretical air flowrate of the engine in g/s

V_{swept} : Displacement of the engine, in m^3

ρ_{atm} : Density of air at ambient conditions, calculated from ideal gas equation

60 : Constant for converting engine speed from rpm to rev/s

Z : Stroke parameter for internal combustion engines, 2 for 4 stroke engines

η_V : Volumetric efficiency

4.7 Brake Specific Fuel Consumption

Brake specific fuel consumption (BSFC) is a measure of fuel efficiency; it represents fuel consumption of an engine for one unit power output per unit time. In this study, break specific fuel consumption is calculated in g/HP-h. The formula for calculating this term is as follows:

$$BSFC = 3600 \times \frac{\dot{m}_{fuel}}{P_{corr}} \tag{4.33}$$

$BSFC$: Break specific fuel consumption (in g/HP-h)

4.8 Brake Specific Energy Consumption

Break specific fuel consumption is not a proper method to compare efficiencies of different fuel types, because difference in heating value of fuels is not taken into consideration. On the other hand, break specific energy consumption covers this detail, so it is a better indicator of fuel efficiency. Considering the fuel types in the experiment, Break specific energy consumption (BSEC) is calculated using the following formula.

$$BSEC = 3600 \times \frac{\dot{m}_{diesel} \times Q_{diesel} + \dot{m}_{vapor} \times Q_{ethanol}}{P_{corr}} \tag{4.34}$$

$BSEC$: Break specific energy consumption (in kJ/Hp-h)

Q_{diesel} : Heating value of diesel fuel, 42600 kJ/kg

$Q_{ethanol}$: Heating value of diesel fuel, 27000 kJ/kg

4.9 Air / Fuel Ratio

Air/fuel ratio is calculated through the following equation:

$$\left(\frac{A}{F}\right)_{actual} = \frac{\dot{m}_{fuel}}{\dot{m}_{air,actual}} \tag{4.35}$$

4.10 Excess Air Coefficient

Theoretical air/ fuel (A/F) ratio is calculated using the stoichiometric equation. In order to calculate this parameter, ratio of measured ethanol and diesel flowrates are accepted as basis, since it identifies the maximum amount of ethanol vapor that can be introduced to the engine before knocking begins. The theoretical air/fuel ratio is calculated through the following formula:

$$\lambda = \frac{\left(\frac{A}{F}\right)_{actual}}{\left(\frac{A}{F}\right)_{stoic}} \tag{4.36}$$

Calculation of stoichiometric air/fuel ratios for diesel mode and dual fuel mode are given in appendices .1 and .2 respectively.

Another common term defining the relation between stoichiometric and actual air/fuel ratios is called equivalence ratio, and is reciprocal of excess air coefficient. This term is defined as follows:

$$\phi = \frac{1}{\lambda} \tag{4.37}$$

4.11 Thermal Efficiency

For an internal combustion engine, thermal efficiency identifies how efficiently an engine converts heat from the fuel to work output. The following equations identify how this term is calculated for diesel and for dual fuel modes:

$$\eta_{th,diesel} = \frac{P_{corr} \times 3600}{\dot{m}_{diesel} \times Q_{L,diesel} \times 1.341} \tag{4.38}$$

$$\eta_{th,dual} = \frac{P_{corr} * 3600}{\left(\dot{m}_{vapor} \times Q_{L,ehanol} + \dot{m}_{diesel} \times Q_{L,diesel}\right) \times 1.341} \tag{4.39}$$

$\eta_{th,diesel}$: Thermal efficiency of the system in diesel mode

$\eta_{th,dual}$: Thermal efficiency of the system in dual fuel mode

$\dot{m}_{vapor}$ and $\dot{m}_{diesel}$ are in kg/h

CHAPTER 5

TEST RESULTS

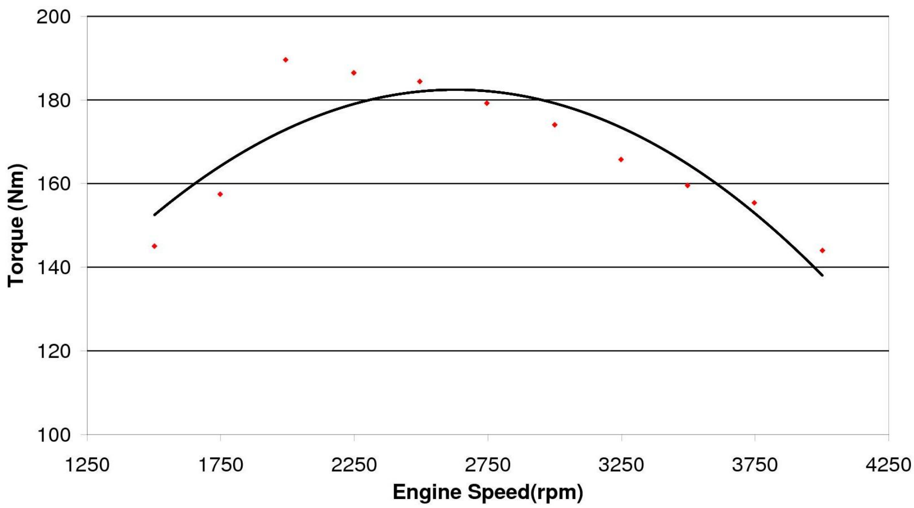
In this chapter, results of the experiment will be presented, plotted in graphs and compared. Second degree curve fit polynomials are used for displaying characteristics of each data set for each property. Details of each data set can be viewed at Appendix 6. While system is observed to work as expected, no performance data could be obtained in dual fuel mode.

5.1 Torque and Power

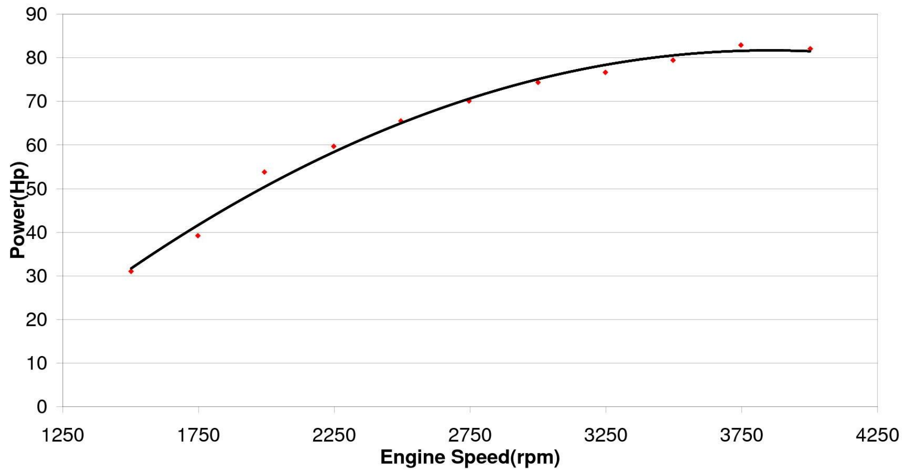
Full range torque and power characteristics of the engine are shown in figures 5.1(a) and 5.1(b). A second degree polynomial is fitted to the test points in order to display characteristics of each parameter. These curves are obtained from a single data set during preliminary tests, and are used for understanding the characteristics of the engine.

Manufacturer of the engine states that the peak torque of the engine is viewed at 1750 rpm, but the results showed loss of torque and manifold pressure as the engine speed drops below 2000 rpm. Furthermore, maximum value that the manufacturer stated could not be observed during the test sessions.

The reason for this result is expected as exhaust line modifications. The original exhaust pipe incorporates only several bends in order to meet space restrictions on the vehicle. However, test setup includes butterfly valves, joints and heat exchangers, all of which is a source of exhaust back pressure increase. As a result, turbo pressure drops as engine speed drops below 2000 rpm and the engine can't produce specified torque values. The mentioned sudden torque drop can be viewed at figure 5.1(a)



(a) Torque characteristics



(b) Power characteristics

Figure 5.1: Power and Torque Characteristics of the Engine

On the other hand, power curve is observed to be smooth in comparison to torque characteristics. Engine power rises until engine speed reaches at 4000 rpm.The values past 4000 rpm are not collected because the manufacturer does not suggest operation at that range. During the preliminary tests, it is observed that torque drops rapidly past 4000 rpm, which signs the initiation of speed limiter.

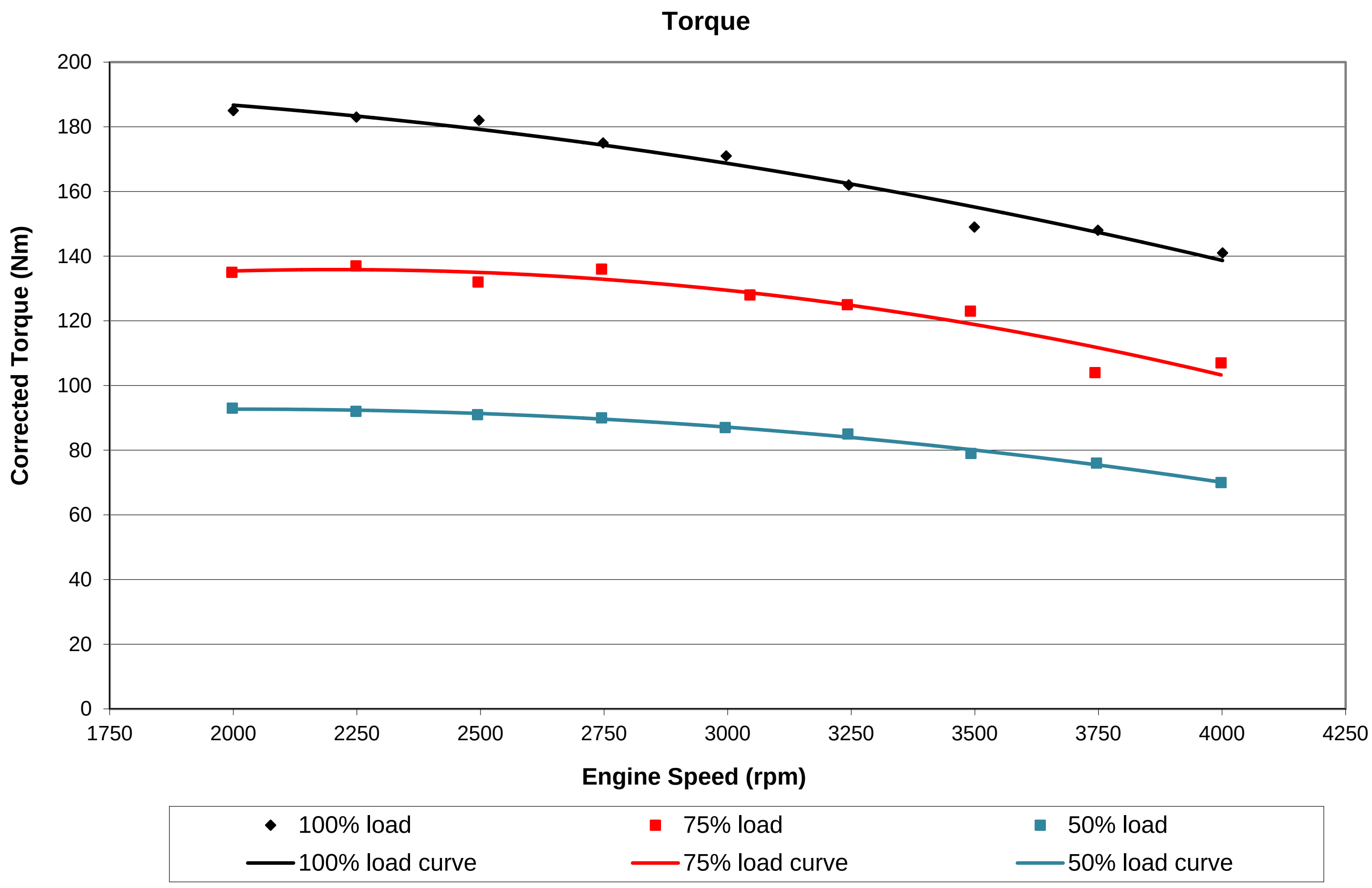


Figure 5.2: Torque Curves of the Engine at Tested Load Conditions

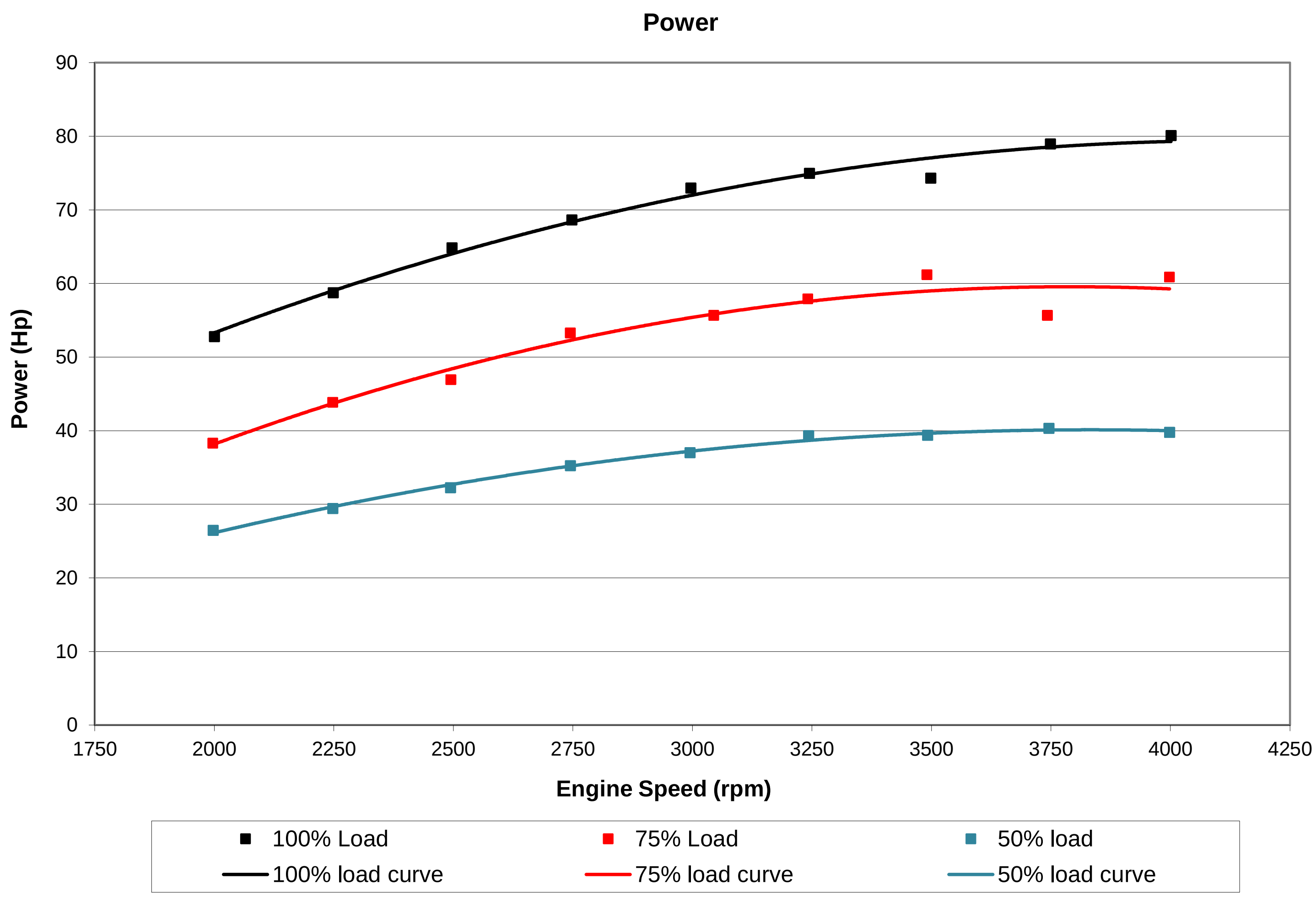


Figure 5.3: Power Curves of the Engine at Tested Load Conditions

Three more data sets are collected for diesel fuel operation, and are used for comparison between engine performance in dual fuel mode and diesel mode. Based on the information gathered about the engine during preliminary tests, it is found out that superheated ethanol generation rate can not be kept at steady state below 2000 rpm in the tested load conditions. That's why any data point below 2000 rpm are omitted. Power and torque output of the engine in 100%, 75% and 50% load conditions are displayed on figures 5.2 and 5.3.

During these sessions, it is observed that engine control unit interferes whenever exhaust gas temperature rises above a level, and reduces power output of the engine. This temperature is not known precisely, since it is a parameter recorded in the software and can't be accessed or modified. As a result of ECU interference, torque output is observed to drop around 10 Nm for full load condition during actual tests.

Using torque data of full load condition, the torque output to be provided during the remaining tests are identified. During partial load tests, it was aimed to adjust the conditions to read torque within 5% of the aimed value. This deviation is close to 3 Nm for this engine for 4000 rpm and 50% load, where the measured value will be minimum.

5.2 Volumetric Efficiency

Volumetric efficiency values of the engine varies between 62% and 83%. Analysing the figure, it can be seen that volumetric efficiency of the engine comparably higher for full load condition and below 2750 rpm than the rest, but diesel mode over this speed at 75% load shows better results for this parameter.

If volumetric efficiency values the engine in diesel and dual fuel modes are compared, it can be seen that dual fuel mode displays less efficient results. The difference of volumetric efficiencies is observed to be the maximum at 2000 rpm and full load. Considering the average of local differences, the following figures can be calculated: 6.33% drop at full load, 5,19% drop at 75% load, and 2.24% drop at 50% load. Intake air flowrate is measured using mass air flowmeter. Using properties of the environment, theoretical air flowrate is calculated. The ratio of these two variables is known as volumetric efficiency. Volumetric efficiency characteristic of the engine is given in figure 5.4.

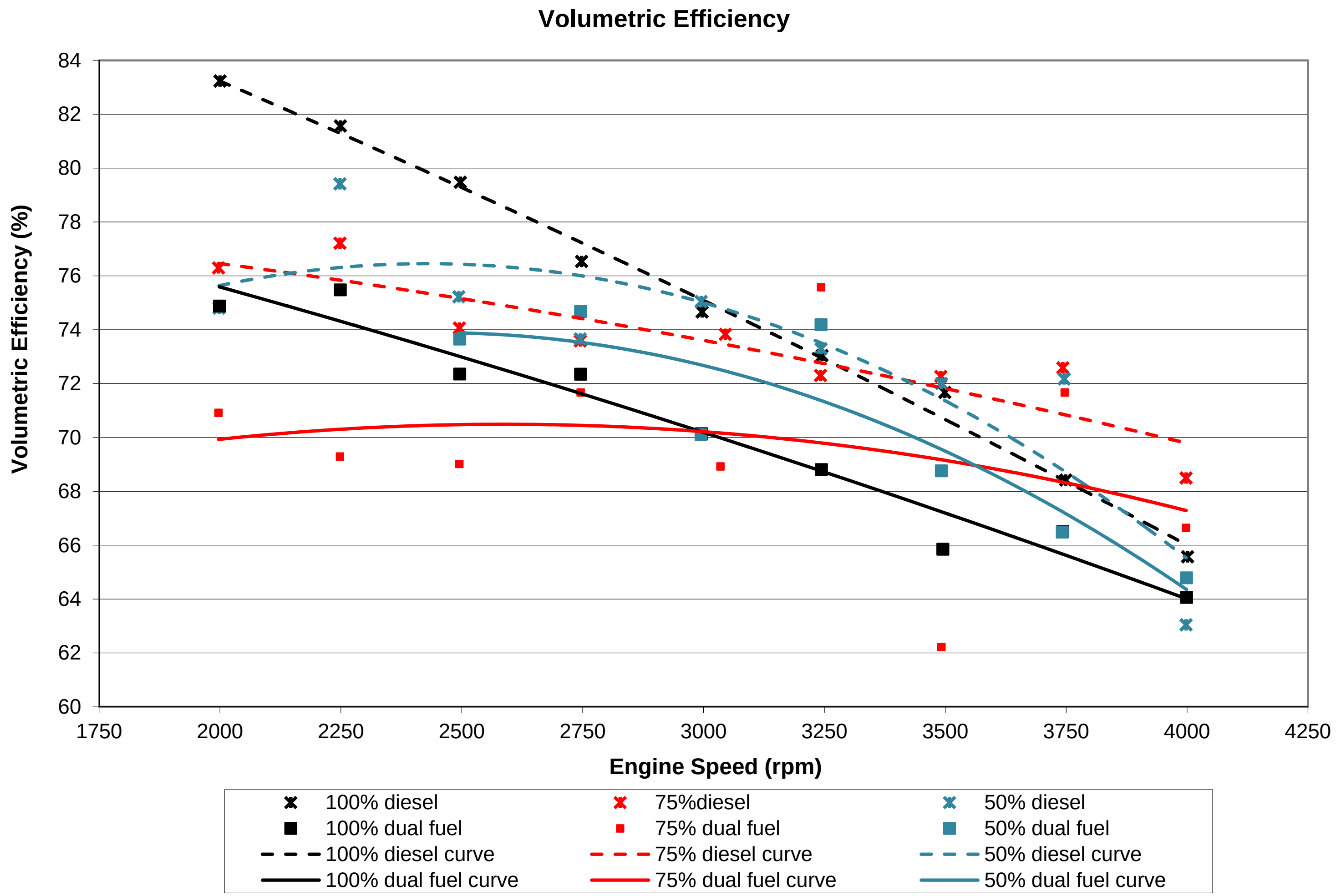


Figure 5.4: Volumetric Efficiency Characteristics of the Test Engine

The mentioned reduction is an expected result. It is known that, the increase in the temperature of intake charge has negative effects on volumetric efficiency. In addition, fuel is not introduced to intake manifold in diesel operation, and any additional material is expected to reduce the rate which air is forced into the cylinder. Both of such cases appear in dual fuel modes, and result as reduction in volumetric efficiency.

Finally, it is important to notice that, this engine utilizes exhaust gas recirculation (EGR) system. The operational parameters of EGR and its effects to volumetric efficiency are unknown, since flowrate or temperature of recirculated exhaust gases are not measured.

5.3 Excess Air Coefficient(λ)

Air/Fuel coefficient expresses the ratio of mass of air that is forced into the cylinder and the mass consumption rate of fuel by the engine, and calculated using the measured mass flowrates. Stoichiometric air/fuel ratio is another parameter that expresses this mass ratio for stoichiometric conditions. The ratio of actual to stoichiometric air/fuel ratios result as excess air coefficient, which explains whether engine uses fuel rich or air rich combustion media.If this parameter is over one, the media is said to be air rich. Considering that latest trend is to ensure air rich conditions, it is important to check and evaluate this parameter. Table 5.1 lists the air/fuel ratios and excess air coefficients measured during the tests.

Table 5.1: Air/Fuel Ratio and Excess Air Coefficients

engine speed	load	diesel		dual fuel	
		A/F	λ	A/F	λ
4000	100%	13.289	0.931	10.914	0.881
2250	100%	13.689	0.959	10.941	0.886
2000	100%	14.611	1.023	10.701	0.861
2250	100%	14.798	1.036	11.320	0.909
2000	100%	14.675	1.028	11.074	0.896
2250	100%	14.732	1.032	10.720	0.857
2000	100%	14.786	1.035	10.835	0.866
2250	100%	15.033	1.053	10.521	0.840

Table 5.1Air/Fuel Ratio and Excess Air Coefficients – continued

engine speed	load	diesel		dual fuel	
		A/F	λ	A/F	λ
2000	100%	15.085	1.056	9.954	0.796
4000	75%	19.202	1.345	12.404	1.031
3750	75%	21.308	1.492	12.496	1.045
3500	75%	18.433	1.291	12.862	1.051
3250	75%	18.906	1.324	14.490	1.185
3000	75%	19.194	1.344	14.002	1.150
2750	75%	18.076	1.266	14.300	1.174
2500	75%	18.374	1.287	12.109	1.011
2250	75%	17.821	1.248	11.038	0.925
2000	75%	16.926	1.185	10.260	0.881
4000	75%	19.765	1.384	18.951	1.730
3750	75%	23.199	1.625	19.285	1.784
3500	75%	21.541	1.509	19.114	1.794
3250	75%	22.949	1.607	19.629	1.769
3000	75%	22.452	1.572	16.514	1.480
2750	75%	21.248	1.488	17.304	1.528
2500	75%	19.945	1.397	17.332	1.468
2250	75%	19.126	1.339	N/A	N/A
2000	75%	16.837	1.179	N/A	N/A

Comparing the results, it can be seen that air/fuel ratios and excess air coefficients are lower in dual fuel mode tests than diesel mode tests. Recalling the combustion characteristics of premixed charges, it can be seen that this is not an unexpected result. However, it is observed that engine operated at considerably low excess air coefficients, which are below unity, the value that is observed in stoichiometric condition. As a result of this fact, it is expected that some of the fuel will be wasted as unburned fuel. If exhaust emission tests were performed, it would be seen that unburned hydrocarbon emissions would rise in dual fuel tests.

The fact that excess air coefficient values drop below unity explains can bring an explanation to the increase in exhaust gas temperatures. One of the two known reasons that can explain the increase of exhaust gas temperatures is continuation of the chemical reaction through the exhaust pipes.

5.4 Brake Specific Fuel Consumption

Brake specific fuel consumption is a measure of fuel efficiency of an engine that explains efficiency in terms of amount of fuel used per unit power. Figure 5.5shows brake specific fuel consumption characteristic of the test engine. Second degree polynomials are fitted for each set of data in order to have a better understanding about BSFC of the engine.

Brake specific fuel consumption values in diesel mode tests are lower than that of dual fuel mode tests, as it can be seen in the graph. This situation is an expected result of difference of heating values of the fuels; that is the engine consumes more fuel in dual fuel mode in order to produce the same amount of power that diesel modes decided.

Focusing on the results of diesel mode, it can be seen that the most fuel efficient speed range of the engine in diesel mode is the interval that lies between 2500 and 3000 rpm, and this range shifts to 2750 and 3250 rpm in dual fuel mode. However, it is not possible to define a range for fuel efficiency in 50% load conditions. This is a result of engine behavior on data points where 50% load corresponds. The torque values that needed to be met lie on a region where turbo pressure shows abrupt changes. A probable reason for such a behavior can be injection of excess fuel in order to generate higher exhaust gas temperature. Comparably higher exhaust gas temperatures at 4000 and 2500 rpm, and comparably cooler temperatures at 3750 and 3500 rpm in dual fuel mode are positive clues about this provision. Considering that fuel maps of the engines are coded for better drivability of the engine, such deviations can be expected.

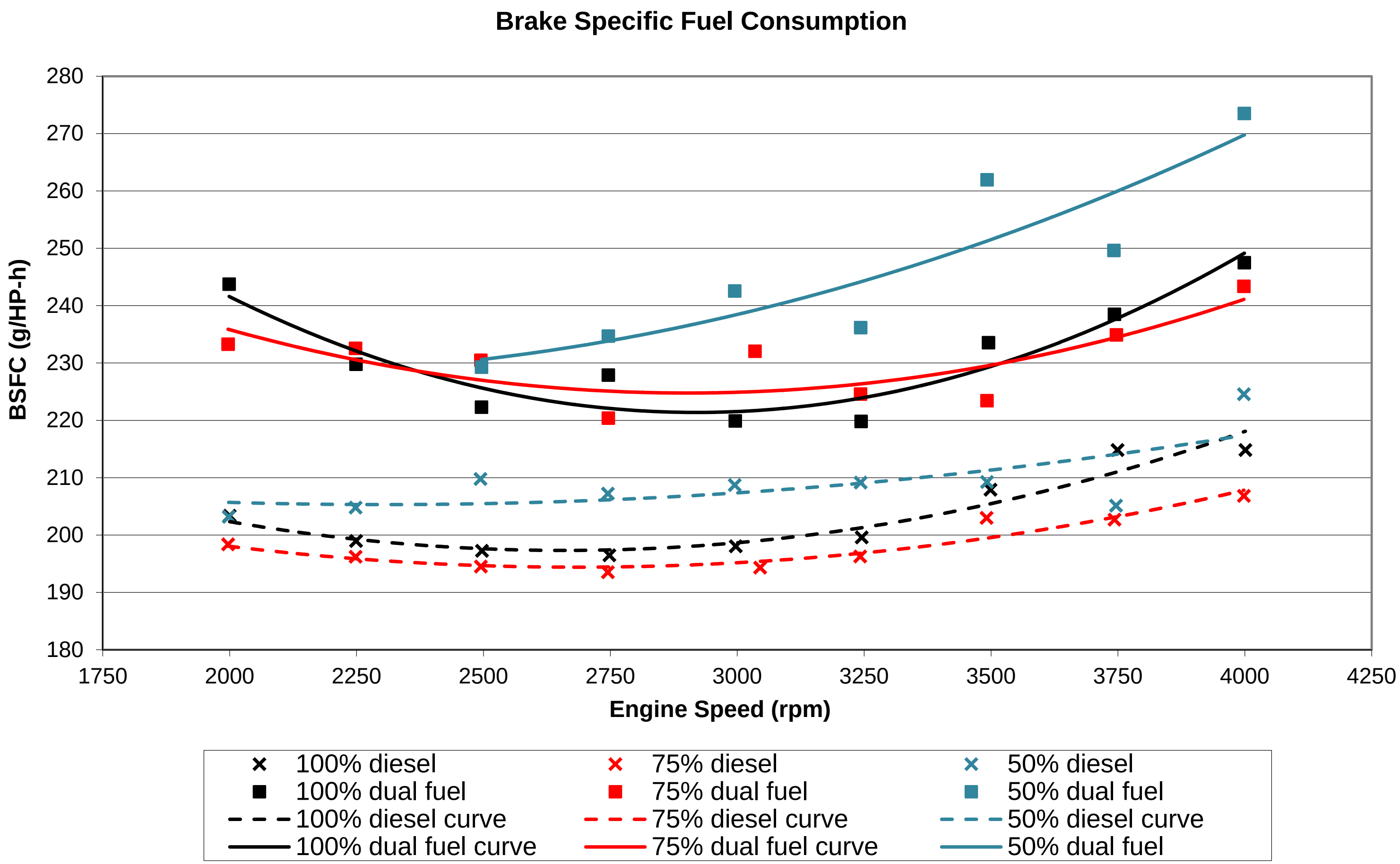


Figure 5.5: Break Specific Fuel Consumption Characteristics of the Test Engine

Examining dual fuel mode results, the facts about diesel mode results are still valid, but the ranges are different. In dual fuel mode, the most efficient range for both 100% and 75% load is the range between 2750 and 3250 rpm. In addition, results of 50% load showed the same behavior in both modes. However, deviations are observed to be larger in dual fuel mode for all load conditions. There can be two reasons for this situation. First, air/fuel ratio was not controlled during the tests. It is possible that some fraction of ethanol fed into the combustion chamber was unburned, and process continued in exhaust pipe. Second, it is possible that responses of engine control unit is not compatible with the conditions. At whatever load conditions the engine operates, the amount of diesel fuel used will always be less than programmed value. As a consequence, it is possible that these data points lay on an unstable region.

due to the energy content of exhaust gases and consequent unsteadiness of ethanol vapor generation system ar 50% dual fuel mode below 2500 rpm, it was not possible to collect data that defines maximum diesel replacement rate. Below 2500 rpm, the pressure inside the heat exchanger keeps a dropping trend, so that pressure falls below design limits and ethanol consumption valve closes.

It is necessary to note that, BSFC, BSEC and thermal efficiency are related to each other since fuel consumption rates are involved in all. That's why the fluctuating trend will affect all three properties.

5.5 Brake Specific Energy Consumption

Brake specific energy consumption is a measure of efficiency for an internal combustion engine. It identifies the amount of energy required in order to produce 1 unit of power. This property is a more accurate comparison tool than BSFC when two or more types of fuels are utilized. Figure 5.6 displays BSEC characteristics of the test engine in both modes and all load conditions.

Brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) are related to each other by fuel consumption rates, as it is explained in chapter 4. As a result, the similarities of these curves is expected. In addition, the deviations in BSFC that are associated with interference of ECU and turbo activation are also valid for BSEC values.

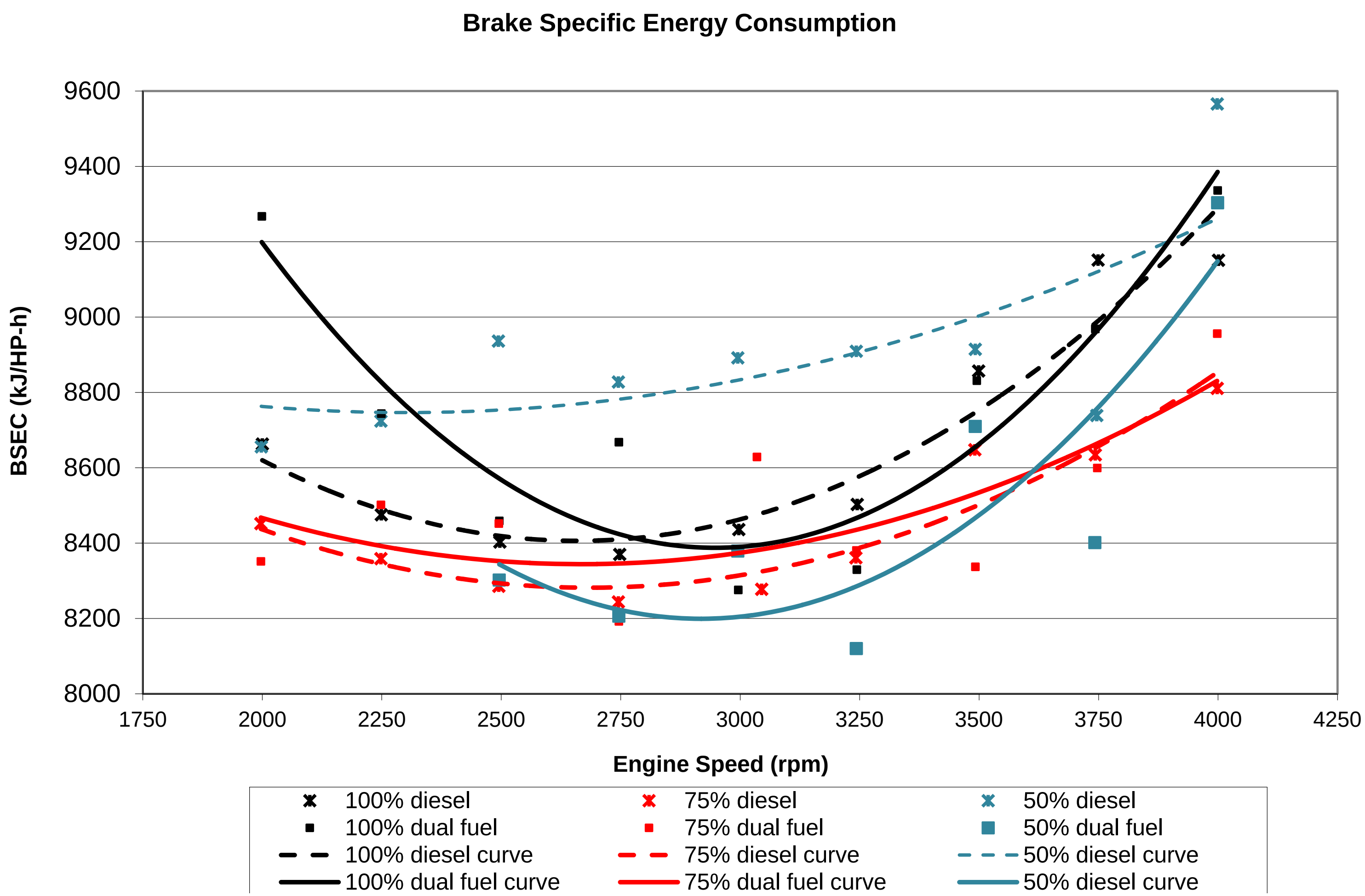


Figure 5.6: Break Specific Energy Consumption Characteristics of the Test Engine

Analysing figure 5.6, it can be seen that the only notable improvement is observed at 50% load. The point of maximum improvement is 3250 rpm with 8.85%, and the average of improvement is 5.38%. In addition, it is observed that improvement of BSEC below 3500 rpm are almost two times of the rest. Focusing on 75% load, it can be said that use of ethanol had a slight inefficiency throughout the speed range. Since the deviations from the characteristic curve are comparably high, it will not be reasonable to comment on every single data. But the overall result show a 0.56% increment in BSEC values for this load condition, which indicated loss of efficiency. Finally, efficiency loss increases for 100% load and the average increase in BSEC is 1.13%.

5.6 Thermal Efficiency

Thermal efficiency is the most basic concept that defines efficiency of a heat engine. Using the collected data, this property of the engine is obtained and plotted. The plot is available in figure 5.7.

The range between 27% and 33% covers all the thermal efficiency values of the engine, for both diesel and dual fuel modes and in all load conditions. In diesel mode, peak values for full load and 75% load are observed at 2750 rpm as 31.6% and 32.1% respectively. In 50% load, the peak value appears at 2000 rpm as 30.6%. In terms of overall thermal efficiency figures, 50% load is noticed to be the least efficient by great margin.

Analyzing dual fuel mode results, it can be seen that thermal efficiency curves are close to their diesel mode equivalents. Though, the most noteworthy data set is obtained for 50% load, where thermal efficiency values are observed to be the highest among all the data sets in contrast to the lowest diesel mode figures obtained in diesel mode for this load.

Improvement on thermal efficiency is expected to be a result of higher ethanol vapor temperatures at the inlet of gas mixer. During 100% and 75% load tests, temperature of ethanol vapor were slightly higher than saturation temperatures in most of the data points. This fact can be evaluated as a sign of condensation in intake manifold. It is known that utilization of wasted heat in exhaust gases is a way to improve thermal efficiency of a heat engine, and effects of the application are observed in 50% load case. In addition, higher ethanol vapor temperature is also a probable factor that has positive effects on thermal efficiency.

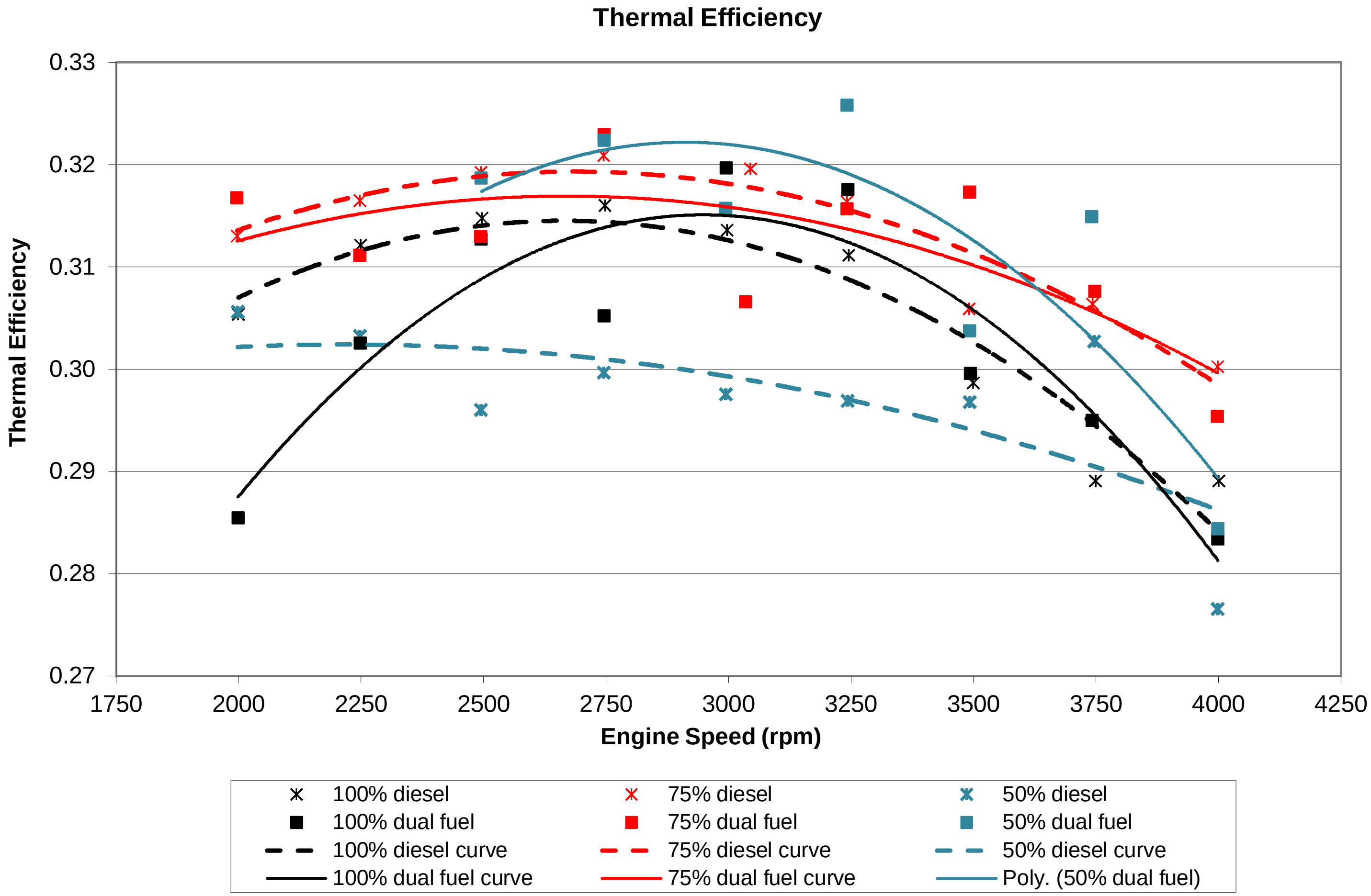


Figure 5.7: Thermal Efficiency Characteristics of the Test Engine

5.7 Ratio of Diesel Replacement by Superheated Ethanol

As the fundamental concern of the study, the ratio of diesel usage replaced by superheated ethanol is discussed in this section. In other words, it expresses the ratio of power generated from use of ethanol. The figures can be viewed in table 5.2.

Table 5.2: Diesel Fuel Replacement Ratios

engine speed	load	diesel		dual fuel		percent
		$\dot{m}_{diesel}$	$\dot{m}_{ethanol}$	$\dot{m}_{diesel}$	$\dot{m}_{ethanol}$	substitution
4000	100%	4.781	0	3.530	1.928	26.18
3750	100%	4.712	0	3.391	1.923	28.05
3500	100%	4.291	0	3.391	1.799	20.99
3250	100%	4.155	0	3.198	1.653	23.02
3000	100%	4.013	0	3.023	1.697	24.68
2750	100%	3.747	0	3.054	1.508	18.49
2500	100%	3.554	0	2.750	1.351	22.63
2250	100%	3.247	0	2.574	1.264	20.71
2000	100%	2.983	0	2.441	1.212	18.16
4000	75%	3.496	0	2.381	1.726	31.89
3750	75%	3.134	0	2.320	1.779	25.96
3500	75%	3.450	0	2.380	1.476	31.02
3250	75%	3.156	0	2.411	1.496	23.60
3000	75%	3.004	0	2.256	1.457	24.88
2750	75%	2.865	0	1.978	1.283	30.94
2500	75%	2.534	0	1.811	1.364	28.53
2250	75%	2.391	0	1.634	1.272	31.66
2000	75%	2.111	0	1.342	1.302	36.42
4000	50%	2.481	0	1.116	1.837	55.02
3750	50%	2.297	0	0.977	1.800	57.47
3500	50%	2.289	0	0.908	1.907	60.33
3250	50%	2.285	0	1.010	1.485	55.82
3000	50%	2.146	0	1.051	1.473	51.04

Table 5.2Diesel Fuel Replacement Ratios – continued

engine speed	load	diesel		dual fuel		percent
		$\dot{m}_{diesel}$	$\dot{m}_{ethanol}$	$\dot{m}_{diesel}$	$\dot{m}_{ethanol}$	substitution
2750	50%	2.029	0	0.996	1.233	50.89
2500	50%	1.880	0	1.057	0.911	43.79
2250	50%	1.674	0	N/A	N/A	N/A
2000	50%	1.493	0	N/A	N/A	N/A

During the studies, the maximum figures are obtained at 50% load. The most significant value is observed at 3500 rpm as 60.33%. The highest value for 75% load is 36.42% at 2000 rpm, and for full load is 28.05% at 3750 rpm. Evaluating the results for full and half load conditions, it can be said that diesel replacement ratio is higher at faster engine speeds. On the other hand, 75% load results do not comply with such a behavior and higher values are obtained in higher and lower ranges of speed interval.

The minimum figures are observed as 18.16% at 2000 rpm for full load, 23.60% for 75% load and 43.79% at 2500 rpm for half load. Results indicate reduction in diesel replacement ratios as the engine speed drops at full and half load. As mentioned before, the behavior of the engine is the opposite for 75% load condition, where the minimum values are obtained at higher and lower ranges of the test speed interval.

The average replacement ratio values appear as 22.54% at full load, 29.43% at 75% load and 53.48% at half load.

As for other parameters, interference of engine control unit has been a decisive factor throughout the experiments. Diesel fuel feed rate is controlled by the engine itself through the engine map, so it was not possible for some points to adjust diesel consumption rate. While it may not be reasonable to suggest higher diesel replacement ratios, it is clear that the data collected would be closer to the actual limits and there would be less fluctuations overall, given that diesel fuel flowrate could be adjusted with more precision.

CHAPTER 6

CONCLUSION AND DISCUSSIONS

In this study, performance analysis of a turbocharged and intercooled compression ignition internal combustion engine is carried out for superheated ethanol vapor. Ethanol vapor is superheated by the ethanol vapor generation system and supplied to the engine through the intake manifold using a gas mixer. Using this test setup, engine performance data is collected for both diesel mode and dual fuel mode. In this final chapter, the results will be evaluated, and use of superheated ethanol in compression ignition engines will be discussed.

During the tests, power and torque characteristics of the engine in diesel mode were collected to form a basis during dual fuel tests. Since the torque values of diesel and dual fuel tests vary within 5% range, there will be no performance gain or loss.

Volumetric efficiency of the engine is higher in dual fuel mode. Volumetric efficiency drop up to 10% is observed throughout the tests. The curve fitted to the results indicate a uniform volumetric efficiency drop throughout the range.

Air/fuel ratio and excess air coefficient values display a significant difference between two modes. In diesel mode, engine operates at an air/fuel ratio that is close to stoichiometric conditions at full load condition, between 1.2 and 1.5 at 75% load with average value of 1.309, and between 1.2 and 1.625 at half load with an average value of 1.456. During dual fuel tests, these values are observed to be lower. The most significant undesirable condition occurs at full load, where excess air coefficient drops down to 0.796 and never rises over 0.909. In 75% load tests, this figure is observed to be slightly over or below stoichiometric condition for the majority of data points. The only load condition where this coefficient is observed to display improvement is half load condition at higher speeds. Intake manifold pressure is observed to be higher compared to diesel mode results.

Brake specific fuel consumption is higher in dual fuel operation. This is also an expected result, since heating values of ethanol and diesel are different. Energy content of ethanol is roughly 2/3rd of diesel. As a result the compensation of energy loss will be the increase of fuel consumption rate, and will have an inverse proportion to the energy contents.

Brake specific energy consumption and thermal efficiency values are observed to be higher in diesel mode for higher load conditions by small margins. The average drop in thermal efficiency at higher load conditions are 0.4% and 1.03% for full and 75% load respectively. However, 5.74% average improvement in thermal efficiency is calculated at half load from the recorded data. As a variation of thermal efficiency, brake specific energy consumption displayed the same characteristics with almost similar magnitudes.

diesel replacement ratios in higher loads appear as 22.54% and 29.43% full load and 75% load respectively. In contrast, the average of 50% load data appears as 53.48% with a peak value of 60.33%. Looking at the ethanol temperature records at gas mixer entrance, it can be seen that the data points where higher temperatures are recorded are also the points where higher replacement ratio figures are observed. While these temperatures are slightly over saturation temperature, it is highly probable that condensation occurs when introduced to intake air.

The most significant problem that influenced the results was poor control over throttle. Diesel fuel delivery is controlled by ECU, and the only input that user supplies is throttle pedal position. There are several throttle pedal positions throughout the operation range where engine torque displays abrupt changes, and some of these positions coincide with data points, especially at lower ranges. Analysing torque curves, it will become obvious that the mentioned deviations exist even in diesel tests. Furthermore, responses of throttle pedal also vary on the readings from the sensors that the engine use.

Ethanol temperature at gas mixer connection is considerably lower compared to heat exchanger outlet conditions. There are two probable reasons that can cause such a drop. The first and the most effective reason is expansion of superheated ethanol as it exits needle valve. This valve is slightly opened at higher heat exchanger pressures. As a result, this valve acts similar to an expansion valve. Second, the piping route from the heat exchanger to gas mixer is long, which is a reason for heat loss, hence temperature drop. The hoses on this route are made of Teflon,a material with low heat conduction properties. In addition, this route, turbine meter and needle valve are covered with exhaust wrap material for improving heat insulation, but still some heat loss and resulting temperature drop is expected. Combining these two, it is clear that an alternative method that replaces use of needle valve is necessary for further studies on this system, while shortening ethanol vapor delivery line will also improve retaining ethanol temperature.

Another major issue about superheated ethanol generation system is characteristics of heat exchanger. This component was designed for use in spark ignition engines where exhaust temperatures are higher. In this experiment, one of the major problems was associated with vapor generation rate at lower loads and engine speeds. The system could not be operated at its limits below 2000 rpm in any of the load conditions, while this speed is observed as 2500 rpm at half load condition. In order to utilize a speed range which covers operation range of the engine, a purpose built heat exchanger is necessary.

Current approach for using ethanol in diesel engines is forming both fuels in to diesel-ethanol blends that consists of 10% ethanol, and mostly pure ethanol is used for forming them. In this study, the smallest diesel replacement ratio is recorded as 18.16 at full load, while average for this load condition is 22.54%. Using this minimum value for calculating the volume fractions, it corresponds to 34.2%, and rises up to 68.7% at the highest diesel replacement condition. If the In addition, the problems related to diesel/ethanol blends do not affect this approach and there is no power loss, as long as the required amount of fuel is supplied.

Mechanical components of superheated ethanol generation system are designed or selected for versatility during the experiments since most of the parameters were unknown before the experiments. Operation of the system is handled via four stepper motors and six two way valves excluding calibration equipment. If this system is applied to a road going vehicle, the amount of controllers can be reduced or precise devices can be switched to simpler ones. For example, two piston operated valves and a solenoid valve are used just for pumping ethanol into the heat exchanger, since the limiting parameters like heat exchanger pressure were not known beforehand. If this system will be mounted on a vehicle as a commercial product, a high pressure fuel pump and a one way valve can easily solve the task. This one and similar modifications will further simplify the system.

During the experiments, it is observed that the system can reach at steady state, and this process occurs very quickly. Responses of exhaust butterfly valves are quick, and its effects can be observed with a small delay time that doesn't affect power output significantly. Torque drop that is a result of pressure fluctuations is less than 10 Nm, and can be recovered in 30 seconds. These fluctuations can be minimized if the software is programmed for sharper exhaust butterfly valve actuation. Heat exchanger pressure is not recorded in real time, so accurate pressure versus time records are not available.

The final issue to be discussed about the system is size and form of the heat exchanger. This device occupies a large space, which can be roughly defined as half the space test engine requires. However, a complete redesign of heat exchanger can solve this problem, and this system can fit into smaller passenger vehicles.

To sum up, the first tests of the system show the system's success. First of all, the amount of power generated by ethanol varies between 18% and 60%, and this ratio is higher than that of blends. In addition, ethanol is not introduced to fuel delivery lines, so corrosive effects will not cause any problems in long term use, and will not require any modifications on fuel delivery system. Second, power losses related to diesel-ethanol blends are not present in this method. Most of the late model diesel engines control fuel flow based on fuel map records and deliver fuels based on the records, so the required increase in fuel flow is not taken into consideration. Delivery of ethanol in this system is independent of fuel map and power loss can be compensated by delivering right amount of ethanol. Third, ethanol exists in liquid form in all climate conditions, which is a plus considering transportation. The current trend is to utilize compressed natural gas, transportation of which relies on compressed tanks. Use of ethanol eliminate the risks related to handling gaseous fuels and will provide safer operation. Finally, the equipment used for generating and delivering superheated ethanol vapor performed successfully in the test engine, and the same situation is expected for other diesel engines.

The results of the tests and their evaluations are given. Based on them, the following items are suggested for future studies:

- During the tests, data are collected for discovering engine limits just for evaluating compliance of ethanol with diesel engines. A parametric study should be performed in order to investigate the effects of ethanol temperature and pressure. In addition, delivering ethanol by separate injectors or pipes just before intake valve may have positive or negative effects. These cases should be investigated.

- In this study, ethanol purity was very close to absolute ethanol. taking production costs into account, azeotropic ethanol should be used in further studies. While ethanol of higher purity will increase production costs, lower purity samples will be further distilled in the heat exchanger and water will be collected after some operation.

- Test engine is a four stroke turbocharged engine, so it is not possible to generalize these results for all diesel engines. Two stroke diesel engines are rarely used, but naturally aspirated engines are still as common as turbocharged or supercharged engines.

- Compliance of ethanol with diesel engines is still unknown. Tests should be performed in order to check whether ethanol has side side effects on the engine in long term uses.

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EXPERIMENTAL DATA Experimental data collected during the experiments will be given in this chapter. The first section will inform about the results of preliminary diesel test that is performed for full load condition and in speed range of 1500 to 4000 rpm. Later sections will display the data of performance comparison tests.

.1 DIESEL MODE FULL SPEED RANGE TEST

Table .1: Results of 100% Load Full Speed Range Diesel Test

data	unit	1	2	3	4	5	6	7	8	9	10	11
T_{amb}	°C	22.2	22.1	22.1	22.0	22.0	22.1	21.8	22.0	22.0	22.0	21.7
P_{amb}	kPa	920.4	920.4	920.4	920.4	920.4	920.4	920.4	920.4	920.4	920.4	920.4
RH	%	61.0	61.0	61.0	61.0	61.0	61.0	61.0	61.0	61.0	61.0	61.0
speed	rpm	4002	3747	3497	3248	3000	2746	2495	2248	1993	1748	1502
throttle	%	100	100	100	100	100	100	100	100	100	100	100
torque	Nm	139	150	154	160	168	173	178	180	183	152	140

Table .1: Results of 100% Load Full Speed Range Diesel Test – continued

data	unit	1	2	3	4	5	6	7	8	9	10	11
T_{corr}	Nm	144.0	155.4	159.5	165.8	174.1	179.2	184.4	186.5	189.6	157.5	145.0
Power	Hp	82.08	82.93	79.46	76.68	74.36	70.09	65.53	59.70	53.81	39.20	31.03
$\dot{m}_{air}$	g/s	62.56	62.57	60.31	59.70	57.43	54.81	51.48	47.79	44.35	34.54	24.75
$P_{manifold}$	kPa	196.7	202.3	205.8	210.7	211.6	213.9	214.4	216.1	218.8	193.3	166.5
η_V	%	64.63	67.13	68.15	70.95	73.58	75.89	78.27	80.01	82.71	83.14	80.49
$\dot{m}_{fuel}$	g/s	4.06	4.01	3.90	3.57	3.50	3.31	3.06	2.71	2.60	1.88	1.62
$\dot{m}_{vapor}$	g/s	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
BSFC	g/Hp-H	178.0	174.2	176.6	167.8	169.7	170.0	167.8	163.4	174.2	172.6	188.0
BSEC	kJ/Hp-H	7584	7421	7521	7146	7224	7239	7150	6958	7421	7354	8007
A/F		15.41	15.59	15.48	16.71	16.39	16.56	16.85	17.64	17.03	18.37	15.28
λ		1.08	1.09	1.08	1.17	1.15	1.16	1.18	1.24	1.19	1.29	1.07
η_{th}	%	0.354	0.362	0.357	0.376	0.372	0.371	0.375	0.386	0.362	0.365	
T_{exh}	°C	634	655	640	621	617	599	570	540	543	497	463
V0	V	2.952	2.951	2.951	2.950	2.950	2.951	2.948	2.950	2.950	2.950	2.947
V1	V	4.002	3.747	3.497	3.248	3.000	2.746	2.495	2.248	1.993	1.748	1.502
V2	V	3.475	3.750	3.850	4.000	4.200	4.325	4.450	4.500	4.575	3.800	3.500
V3	V	3.082	3.167	3.220	3.295	3.308	3.343	3.351	3.377	3.418	3.030	2.624
V4	V	7.232	7.476	7.302	7.034	6.836	6.486	6.135	6.170	6.135	5.631	5.234
V5	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V6	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V7	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V8	V	4.000	4.000	4.000	4.000	4.000	4.000	4.000	4.000	4.000	4.000	4.000
V9	V	8.310	8.310	8.310	8.310	8.310	8.310	8.310	8.310	8.310	8.310	8.310
V10	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V11	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V12	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V13	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V14	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V15	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
freq	Hz	5419	5420	5299	5266	5145	5005	4826	4629	4445	3919	3395

.2 RESULTS OF 100% LOAD DIESEL TEST

Table .2: Results of 100% Load Diesel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	21.0	20.8	24.1	21.9	21.9	21.2	21.1	20.9	20.3
P_{amb}	mBar	925	925	927	925	925	925	925	925	925
RH	%	60	60	60	60	60	60	60	60	60
rpm		4001	3749	3499	3245	2997	2748	2497	2249	2000
throttle	%	100	100	100	100	100	100	100	100	100
torque	Nm	137	144	145	158	166	171	177	178	180
T_{corr}	Nm	141	148	149	162	171	175	182	183	185
Power	Hp	80.13	78.97	74.31	74.94	72.96	68.66	64.86	58.75	52.81
$\dot{m}_{air}$	g/s	63.54	64.51	62.70	61.48	58.89	55.20	52.55	48.81	45.00
P_{man}	kPa	196.5	203.9	205.0	211.1	214.2	213.1	214.9	215.8	218.8
η_V	%	65.57	68.42	71.67	73.04	74.66	76.53	79.48	81.56	83.23
$\dot{m}_{fuel}$	g/s	4.781	4.712	4.291	4.155	4.013	3.747	3.554	3.247	2.983
$\dot{m}_{gas}$	g/s	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
BSFC	g/Hp-h	214.8	214.8	207.9	199.6	198.0	196.5	197.3	198.9	203.4
BSEC	kJ/Hp-h	9151	9151	8857	8502	8436	8370	8403	8475	8663
A/F		13.289	13.689	14.611	14.798	14.675	14.732	14.786	15.033	15.085
λ		0.931	0.959	1.023	1.036	1.028	1.032	1.035	1.053	1.056
η_{th}	%	0.289	0.289	0.299	0.311	0.314	0.316	0.315	0.312	0.305
T_{exh}	°C	675	672	611	608	618	610	602	579	578
P_{gas}	kPa	25	25	25	25	25	25	25	25	25
T_{gas}	°C	17	20	22	22	23	23	24	24	25
T_{cons}	°C	18	20	29	30	31	31	31	31	33
freq	Hz	6134	6192	6084	6011	5856	5635	5476	5252	5024
V_0	V	2.940	2.938	2.971	2.949	2.949	2.942	2.941	2.939	2.933
V_1	V	4.001	3.749	3.499	3.245	2.997	2.748	2.497	2.249	2.000
V_2	V	3.434	3.610	3.633	3.956	4.160	4.266	4.423	4.445	4.491
V_3	V	3.080	3.192	3.207	3.301	3.348	3.331	3.359	3.372	3.417
V_4	V	7.713	7.670	6.958	6.929	7.042	6.945	6.855	6.586	6.570

Table .2: Results of 100% Load Diesel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.004	0.004	0.004	0.004	0.005	0.004	0.004	0.004	0.004
V_6	V	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
V_7	V	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
V_8	V	4.003	4.001	4.006	3.997	3.997	3.997	3.996	3.996	3.995
V_9	V	0.07	0.074	0.079	0.07	0.069	0.07	0.054	0.07	0.078
V_{10}	V	8.350	8.350	8.065	8.350	8.350	8.350	8.350	8.350	8.350
V_{11}	V	0.34	0.396	0.42	0.432	0.437	0.449	0.46	0.46	0.479
V_{12}	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V_{13}	V	0.078	0.081	0.076	0.088	0.089	0.089	0.089	0.089	0.089
V_{14}	V	0.366	0.417	0.596	0.612	0.645	0.642	0.627	0.63	0.667
V_{15}	V	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004

.3 RESULTS OF 75% LOAD DIESEL TEST

Table .3: Results of 75% Load Diesel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	20.6	20.8	21.0	21.1	21.2	21.3	21.5	21.7	21.8
P_{amb}	mBar	928	928	928	928	928	928	928	928	928
RH	%	49	49	49	49	49	49	49	49	49
rpm		3998	3743	3491	3242	3045	2745	2495	2248	1997
throttle	%	79	74	75	72	69	62	58	53	55
torque	Nm	106	104	122	124	127	135	131	135	133
T_{corr}	Nm	107	104	123	125	128	136	132	137	135
Power	Hp	60.85	55.67	61.18	57.89	55.65	53.29	46.91	43.86	38.31
$\dot{m}_{air}$	g/s	67.12	66.78	63.59	59.67	57.65	51.78	46.57	42.60	35.73
P_{man}	kPa	198.6	199.3	204.5	206.6	208.2	208.2	204.8	199.6	190.8
η_V	%	68.50	72.59	72.27	72.30	73.83	73.58	74.07	77.21	76.29
$\dot{m}_{fuel}$	g/s	3.496	3.134	3.450	3.156	3.004	2.865	2.534	2.391	2.111
$\dot{m}_{gas}$	g/s	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
BSFC	g/Hp-h	206.8	202.7	203.0	196.3	194.3	193.5	194.5	196.2	198.4
BSEC	kJ/Hp-h	8810	8634	8648	8361	8277	8244	8286	8359	8451
A/F		19.202	21.308	18.433	18.906	19.194	18.076	18.374	17.821	16.926
λ		1.345	1.492	1.291	1.324	1.344	1.266	1.287	1.248	1.185
η_{th}	%	0.300	0.306	0.306	0.316	0.320	0.321	0.319	0.316	0.313
T_{exh}	°C	475	432	483	475	463	468	452	464	474
P_{gas}	kPa	25	25	25	25	25	25	25	25	25
T_{gas}	°C	27	33	34	34	34	35	35	36	36
T_{cons}	°C	23	30	41	46	48	47	48	48	47
freq	Hz	6349	6328	6137	5902	5782	5430	5118	4880	4469
V_0	V	2.936	2.938	2.940	2.941	2.942	2.943	2.945	2.947	2.948
V_1	V	3.998	3.743	3.491	3.242	3.045	2.745	2.495	2.248	1.997
V_2	V	2.649	2.588	3.049	3.106	3.179	3.376	3.270	3.381	3.315
V_3	V	3.110	3.122	3.201	3.232	3.256	3.257	3.205	3.126	2.992
V_4	V	5.372	4.873	5.464	5.368	5.232	5.291	5.107	5.248	5.358

Table .3: Results of 75% Load Diesel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_6	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_7	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_8	V	3.368	3.214	3.241	3.158	3.074	2.869	2.744	2.578	2.649
V_9	V	0.087	0.089	0.093	0.083	0.074	0.079	0.073	0.073	0.078
V_{10}	V	8.351	8.351	8.351	8.351	8.351	8.350	8.351	8.350	8.350
V_{11}	V	0.508	0.616	0.637	0.637	0.634	0.644	0.657	0.666	0.669
V_{12}	V	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V_{13}	V	0.072	0.079	0.084	0.088	0.088	0.088	0.088	0.088	0.088
V_{14}	V	0.465	0.61	0.845	0.94	0.976	0.97	0.973	0.973	0.957
V_{15}	V	0.005	0.005	0.005	0.005	0.004	0.005	0.005	0.005	0.005

.4 RESULTS OF 50% LOAD DIESEL TEST

Table .4: Results of 50% Load Diesel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	22.0	22.0	22.1	22.1	22.1	22.2	22.3	22.3	22.3
P_{amb}	mBar	928	928	928	928	928	928	928	928	928
RH	%	49	49	49	49	49	49	49	49	49
rpm		3998	3746	3492	3243	2995	2745	2494	2248	1998
throttle	%	68	67	65	66	65	59	56	54	49
torque	Nm	69	75	78	84	86	89	90	91	92
T_{corr}	Nm	70	76	79	85	87	90	91	92	93
Power	Hp	39.78	40.32	39.38	39.34	37.02	35.25	32.27	29.42	26.46
$\dot{m}_{air}$	g/s	49.04	53.30	49.31	52.45	48.19	43.11	37.50	32.01	25.14
P_{man}	kPa	158.4	160.5	159.7	179.7	174.6	173.7	162.9	146.1	137.1
η_V	%	63.04	72.17	72.00	73.29	75.05	73.66	75.22	79.42	74.80
$\dot{m}_{fuel}$	g/s	2.481	2.297	2.289	2.285	2.146	2.029	1.880	1.674	1.493
$\dot{m}_{gas}$	g/s	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
BSFC	g/Hp-h	224.5	205.1	209.3	209.1	208.7	207.2	209.8	204.8	203.2
BSEC	kJ/Hp-h	9566	8739	8914	8909	8891	8828	8936	8724	8655
A/F		19.765	23.199	21.541	22.949	22.452	21.248	19.945	19.126	16.837
λ		1.384	1.625	1.509	1.607	1.572	1.488	1.397	1.339	1.179
η_{th}	%	0.277	0.303	0.297	0.297	0.297	0.300	0.296	0.303	0.306
T_{exh}	°C	452	430	437	428	449	459	474	480	473
P_{gas}	kPa	25	25	25	25	25	25	25	25	25
T_{gas}	°C	36	35	35	34	34	35	35	36	36
T_{cons}	°C	49	51	52	51	52	55	54	54	52
freq	Hz	5266	5521	5282	5470	5215	4911	4575	4246	3835
V_0	V	2.950	2.950	2.951	2.951	2.951	2.952	2.953	2.953	2.953
V_1	V	3.998	3.746	3.492	3.243	2.995	2.745	2.494	2.248	1.998
V_2	V	1.730	1.871	1.961	2.109	2.148	2.232	2.249	2.274	2.290
V_3	V	2.561	2.532	2.520	2.824	2.747	2.733	2.569	2.313	2.178
V_4	V	5.109	4.848	4.936	4.823	5.069	5.185	5.361	5.429	5.346

Table .4: Results of 50% Load Diesel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_6	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_7	V	0.005	0.005	0.005	0.005	0.004	0.005	0.005	0.005	0.005
V_8	V	3.045	3.006	2.963	2.966	2.965	2.784	2.688	2.624	2.471
V_9		0.071	0.057	0.084	0.094	0.096	0.088	0.079	0.063	0.076
V_{10}		8.350	8.350	8.350	8.350	8.350	8.350	8.350	8.350	8.350
V_{11}		0.664	0.644	0.647	0.635	0.641	0.644	0.653	0.664	0.671
V_{12}		0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
V_{13}		0.089	0.09	0.09	0.09	0.091	0.091	0.091	0.091	0.09
V_{14}		0.996	1.036	1.061	1.043	1.063	1.113	1.111	1.094	1.053
V_{15}		0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.004

.5 RESULTS OF 100% LOAD DUAL FUEL TEST

Table .5: Results of 100% Load Dual Fuel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	25.3	25.4	25.4	25.4	25.5	25.5	25.7	25.7	25.6
P_{amb}	mBar	919	919	919	919	919	919	919	919	919
RH	%	18	18	18	18	18	18	18	18	18
rpm		3999	3743	3495	3244	2996	2746	2496	2249	1999
throttle	%	76	75	78	76	73	68	63	57	60
torque	Nm	133	142	152	163	171	172	175	175	177
T_{corr}	Nm	139	150	161	172	181	184	187	188	190
Power	Hp	79.39	80.21	80	79.46	77.28	72.08	66.41	60.14	53.96
$\dot{m}_{air}$	g/s	59.56	58.14	55.53	54.92	52.27	48.91	44.43	40.38	36.37
P_{man}	kPa	191.4	192.3	198.7	202.6	204.9	202.8	202.8	196.1	200.2
η_V	%	64.06	66.51	65.85	68.81	70.14	72.35	72.35	75.48	74.88
$\dot{m}_{fuel}$	g/s	3.530	3.391	3.391	3.198	3.023	3.054	2.750	2.574	2.441
$\dot{m}_{gas}$	g/s	1.928	1.923	1.799	1.653	1.697	1.508	1.351	1.264	1.212
BSFC	g/Hp-h	247.5	238.5	233.5	219.8	219.9	227.9	222.3	229.8	243.7
BSEC	kJ/Hp-h	9336	8968	8831	8330	8276	8668	8459	8744	9267
A/F		10.914	10.941	10.701	11.320	11.074	10.720	10.835	10.521	9.954
λ		0.881	0.886	0.861	0.909	0.896	0.857	0.866	0.840	0.796
η_{th}	%	0.283	0.295	0.300	0.318	0.320	0.305	0.313	0.303	0.285
T_{exh}	°C	679	682	672	693	658	673	657	669	620
P_{gas}	kPa	312	323	309	319	303	312	277	277	261
T_{gas}	°C	157	196	192	206	209	239	248	261	261
T_{cons}	°C	87	87	91	90	93	92	100	97	100
freq	Hz	5613	5511	5325	5281	5092	4852	4532	4243	3956
V_0	V	2.983	2.984	2.984	2.984	2.985	2.985	2.987	2.987	2.986
V_1	V	3.999	3.743	3.495	3.244	2.996	2.746	2.496	2.249	1.999
V_2	V	3.315	3.562	3.800	4.074	4.266	4.308	4.371	4.377	4.419
V_3	V	3.001	3.015	3.113	3.172	3.206	3.174	3.175	3.073	3.135
V_4	V	7.749	7.794	7.678	7.912	7.514	7.689	7.503	7.636	7.071

Table .5: Results of 100% Load Dual Fuel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.006	0.005	0.005	0.005	0.006	0.005	0.005	0.005	0.005
V_6	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_7	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
V_8	V	3.268	3.250	3.337	3.287	3.200	3.049	2.892	2.700	2.794
V_9		1.383	1.374	1.256	1.129	1.158	1.035	0.945	0.907	0.859
V_{10}		5.980	3.895	7.823	6.772	4.664	3.610	7.298	6.244	5.459
V_{11}		2.804	3.486	3.421	3.659	3.72	4.232	4.39	4.634	4.629
V_{12}		2.275	2.361	2.251	2.323	2.203	2.271	1.995	1.993	1.870
V_{13}		0.087	0.09	0.091	0.092	0.093	0.093	0.092	0.092	0.092
V_{14}		1.761	1.767	1.844	1.831	1.9	1.881	2.026	1.971	2.033
V_{15}		0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005

.6 RESULTS OF 75% LOAD DUAL FUEL TEST

Table .6: Results of 75% Load Dual Fuel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	24.0	24.1	24.5	24.6	24.7	24.9	25.1	25.3	25.3
P_{amb}	mBar	921	921	921	921	921	921	921	921	921
RH	%	50	50	50	50	50	50	50	50	50
rpm		3998	3747	3492	3243	3035	2746	2495	2248	1997
throttle	%	65	66	67	63	59	47	59	57	50
torque	Nm	102	112	121	130	129	131	132	131	133
T_{corr}	Nm	107	118	125	136	133	136	140	140	143
Power	Hp	60.75	62.83	62.13	62.63	57.61	53.27	49.6	44.98	40.81
$\dot{m}_{air}$	g/s	50.94	51.23	49.59	56.61	52.00	46.63	38.45	32.07	27.13
P_{man}	kPa	156.7	156.4	187.4	189.7	204.2	194.8	183.7	169.5	157.7
η_V	%	66.65	71.67	62.21	75.58	68.92	71.67	69.01	69.29	70.91
$\dot{m}_{fuel}$	g/s	2.381	2.320	2.380	2.411	2.256	1.978	1.811	1.634	1.342
$\dot{m}_{gas}$	g/s	1.726	1.779	1.476	1.496	1.457	1.283	1.364	1.272	1.302
BSFC	g/Hp-h	243.4	234.9	223.4	224.6	232.0	220.4	230.5	232.6	233.3
BSEC	kJ/Hp-h	8956	8600	8337	8381	8629	8192	8452	8502	8352
A/F		12.404	12.496	12.862	14.490	14.002	14.300	12.109	11.038	10.260
λ		1.031	1.045	1.051	1.185	1.150	1.174	1.011	0.925	0.881
η_{th}	%	0.295	0.308	0.317	0.316	0.307	0.323	0.313	0.311	0.317
T_{exh}	°C	609	661	553	560	511	532	571	589	617
P_{gas}	kPa	288	299	257	291	269	251	248	225	176
T_{gas}	°C	154	176	196	174	177	178	187	193	186
T_{cons}	°C	80	97	106	103	90	109	84	80	105
freq	Hz	5370	5387	5289	5710	5433	5112	4622	4240	3944
V_0	V	2.970	2.971	2.975	2.976	2.977	2.979	2.981	2.983	2.983
V_1	V	3.998	3.747	3.492	3.243	3.035	2.746	2.495	2.248	1.997
V_2	V	2.552	2.793	3.022	3.254	3.217	3.280	3.308	3.284	3.322
V_3	V	2.474	2.470	2.941	2.976	3.196	3.054	2.885	2.670	2.490
V_4	V	6.934	7.550	6.282	6.361	5.793	6.036	6.493	6.707	7.029

Table .6: Results of 75% Load Dual Fuel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.005	0.005	0.005	0.005	0.006	0.006	0.006	0.005	0.005
V_6	V	0.005	0.005	0.005	0.006	0.005	0.006	0.005	0.005	0.005
V_7	V	0.005	0.005	0.005	0.005	0.005	0.006	0.005	0.005	0.005
V_8	V	2.954	2.985	3.019	2.897	2.769	2.398	2.777	2.709	2.512
V_9		1.487	1.608	1.139	1.132	0.987	0.959	1.009	1.009	1.191
V_{10}		4.938	3.373	3.895	7.296	5.191	7.954	7.824	5.717	3.895
V_{11}		2.75	3.135	3.477	3.099	3.144	3.173	3.318	3.424	3.315
V_{12}		2.082	2.167	1.833	2.104	1.934	1.786	1.766	1.584	1.197
V_{13}		0.085	0.087	0.09	0.09	0.091	0.091	0.093	0.093	0.093
V_{14}		1.636	1.971	2.16	2.092	1.832	2.22	1.702	1.629	2.137
V_{15}		0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005

.7 RESULTS OF 50% LOAD DUAL FUEL TEST

Table .7: Results of 50% Load Dual Fuel Test

Data	#	1	2	3	4	5	6	7	8	9
T_{in}	°C	22.6	22.6	22.6	22.9	22.9	22.1	22.3	N/A	N/A
P_{amb}	mBar	927	927	927	927	927	927	927	N/A	N/A
RH	%	33	33	33	33	33	33	33	N/A	N/A
rpm		3999	3742	3492	3243	2995	2746	2496	N/A	N/A
throttle	%	54	52	53	55	55	45	47	N/A	N/A
torque	Nm	68	74	77	81	86	86	86	N/A	N/A
T_{corr}	Nm	68	75	78	82	88	87	87	N/A	N/A
Power	Hp	38.87	40.05	38.69	38.03	37.46	34.2	30.9	N/A	N/A
$\dot{m}_{air}$	g/s	55.96	53.55	53.81	48.97	41.68	38.58	34.11	N/A	N/A
P_{man}	kPa	176.2	175.6	182.8	166.2	162.1	153.2	151.2	N/A	N/A
η_V	%	64.79	66.48	68.76	74.19	70.10	74.68	73.65	N/A	N/A
$\dot{m}_{fuel}$	g/s	1.116	0.977	0.908	1.010	1.051	0.996	1.057	N/A	N/A
$\dot{m}_{gas}$	g/s	1.837	1.800	1.907	1.485	1.473	1.233	0.911	N/A	N/A
BSFC	g/Hp-h	273.5	249.6	261.9	236.2	242.5	234.7	229.3	N/A	N/A
BSEC	kJ/Hp-h	9303	8401	8710	8120	8379	8206	8302	N/A	N/A
A/F		18.951	19.285	19.114	19.629	16.514	17.304	17.332	N/A	N/A
λ		1.730	1.784	1.794	1.769	1.480	1.528	1.468	N/A	N/A
η_{th}	%	0.284	0.315	0.304	0.326	0.316	0.322	0.319	N/A	N/A
T_{exh}	°C	415	407	391	463	497	498	533	N/A	N/A
P_{gas}	kPa	248	242	224	220	245	204	192	N/A	N/A
T_{gas}	°C	200	166	164	200	189	142	194	N/A	N/A
T_{cons}	°C	103	109	107	100	98	98	93	N/A	N/A
freq	Hz	5613	5471	5486	5199	4768	4584	4320	N/A	N/A
V_0	V	2.956	2.956	2.956	2.959	2.959	2.951	2.953	N/A	N/A
V_1	V	3.999	3.742	3.492	3.243	2.995	2.746	2.496	N/A	N/A
V_2	V	1.690	1.861	1.927	2.034	2.153	2.149	2.153	N/A	N/A
V_3	V	2.771	2.761	2.871	2.620	2.557	2.502	2.373	N/A	N/A
V_4	V	4.675	4.583	4.401	5.229	5.629	5.637	6.047	N/A	N/A

Table .7: Results of 50% Load Dual Fuel Test – continued

Data	#	1	2	3	4	5	6	7	8	9
V_5	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	N/A	N/A
V_6	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	N/A	N/A
V_7	V	0.005	0.005	0.005	0.005	0.005	0.005	0.005	N/A	N/A
V_8	V	2.625	2.552	2.590	2.654	2.653	2.352	2.395	N/A	N/A
V_9		1.5	1.494	1.513	1.271	1.286	1.000	0.846	N/A	N/A
V_{10}		6.768	4.137	4.662	7.296	5.715	6.242	5.191	N/A	N/A
V_{11}		3.56	2.956	2.919	3.556	3.365	3.000	3.000	N/A	N/A
V_{12}		1.763	1.716	1.572	1.540	1.738	1.421	1.319	N/A	N/A
V_{13}		0.089	0.09	0.09	0.09	0.09	0.083	0.084	N/A	N/A
V_{14}		2.105	2.207	2.171	2.027	1.988	2.000	2.000	N/A	N/A
V_{15}		0.005	0.005	0.005	0.005	0.005	0.005	0.005	N/A	N/A

STOICHIOMETRIC EQUATIONS AND AIR/FUEL RATIO CALCULATIONS

.1 STOICHIOMETRIC EQUATION FOR DIESEL MODEL

$$C_{10.8}H_{18.7} + 15.475\,(O_2 + 3,76N_2) \longrightarrow 10.8CO_2 + \frac{18.7}{2}H_2O + 15.475\,(3.76N_2) \tag{.1}$$

The relation between mass flowrate and molar flowrate is as follows:

$$\dot{m_{diesel}} = \dot{N_{diesel}} \times MW_{diesel} \tag{.2}$$

$\dot{m_{diesel}}$: flowrate of diesel fuel in kg/s

$\dot{N_{diesel}}$: Flowrate of diesel fuel in kmol/s

MW_{diesel} : Molecular weight of diesel fuel, 148.7 kg/kmol

The following relation can be obtained using equations .1 and .2, and defines stoichiometric air/fuel ratio for diesel fuel:

$$
\begin{aligned}
\left(\frac{A}{F}\right)_{stoic} &= \frac{\dot{m_{air}}}{\dot{m_{diesel}}} \\
&= \frac{\dot{N_{air}} \times MW_{air}}{\dot{N_{diesel}} \times MW_{diesel}} \\
&= \frac{15.475 \times (32 + 3.76 \times 28)}{148.7} \\
&= 14.273
\end{aligned} \tag{.3}
$$

$\left(\frac{A}{F}\right)_{stoic}$: Stoichiometric air/fuel ratio

.2 STOICHIOMETRIC EQUATION FOR DUAL FUEL MODEL

Ethanol and diesel fuel are utilized in dual fuel mode. A proper approach to calculate stoichiometric air/fuel ratio is to find the amount of air required for the measured fuel consumption rate. In the following equations, mass flowrates are converted to molar flowrates:

$$\dot{m}_{diesel} = \dot{N}_{diesel} \times MW_{diesel} \tag{.4}$$

$$\dot{m}_{vapor} = \dot{N}_{vapor} \times MW_{vapor} \tag{.5}$$

In the equation, $\dot{N}$ denotes molar flowrate of fuels, and MW denotes molecular weights of species. Molecular weight of diesel is 148.3 kg/kmol and ethanol is 46 kg/kmol.

Collecting these terms in the stoichiometric equation:

$$\begin{aligned} N_{diesel}C_{10.8}H_{18.7} &+ N_{ethanol}C_2H_5OH + N_{air}\left(\frac{O_2}{4.76} + \frac{3,76N_2}{4.76}\right) \longrightarrow \\ & N_{CO_2}CO_2 + N_{H_2O}H_2O + N_{air}\left(\frac{3,76N_2}{4.76}\right) \end{aligned} \tag{.6}$$

The following relations identify mole numbers of products

$$N_{CO_2} = 10.8N_{diesel} + 2N_{ethanol} \tag{.7}$$

$$N_{H_2O} = \frac{1}{2}(18.7N_{diesel} + 6N_{ethanol}) \tag{.8}$$

$$N_{N_2} = \frac{3,76N_{air}}{4.76} \tag{.9}$$

N_{air} can be found by using the balance of O_2 molecules. In the reactants, ethanol, water and air has oxygen content. Subtracting oxygen content of these reactants from that of products will result as oxgen content of intake air.

$$\begin{aligned} N_{O_2,air} &= N_{CO_2,prod} + \frac{1}{2}N_{H_2O,prod} - \frac{1}{2}N_{ethanol} \\ &= 10.8N_{diesel} + 2N_{ethanol} + \frac{1}{4}(18.7N_{diesel} + 6N_{ethanol}) + \frac{1}{2}N_{ethanol} \\ &= 15.475N_{diesel} + 3N_{ethanol} \end{aligned} \tag{.10}$$

Including masses of fuels in the equation, mole number of O_2 becomes:

$$\begin{aligned} N_{O_2,air} &= 15.475\frac{\dot{m}_{diesel}}{MW_{diesel}} + 3\frac{\dot{m}_{ethanol}}{MW_{ethanol}} \\ &= 15.475\frac{\dot{m}_{diesel}}{10.8 \times 12 + 18.7\ times 1} + 3\frac{\dot{m}_{ethanol}}{2 \times 12 + 6 \times 1 + 16} \\ &= 15.475\frac{\dot{m}_{diesel}}{148.3} + 3\frac{\dot{m}_{ethanol}}{46} \end{aligned} \tag{.11}$$

Integrating equation .11 into definition of air/fuel ratio gives :

$$\begin{aligned}\left(\frac{A}{F}\right)_{stoic} &= \frac{N_{O_2,air}\times 32 + 3.76\times N_{O_2,air}\times 28}{\dot{m}_{diesel}+\dot{m}_{ethanol}} \\ &= \frac{\left(15.475\frac{\dot{m}_{diesel}}{148.3}+3\frac{\dot{m}_{ethanol}}{46}\right)(MW_{O_2}+3.76MW_{N_2})}{\dot{m}_{diesel}+\dot{m}_{ethanol}} \\ &= \frac{\left(15.475\frac{\dot{m}_{diesel}}{148.3}+3\frac{\dot{m}_{ethanol}}{46}\right)137.28}{\dot{m}_{diesel}+\dot{m}_{ethanol}} \\ &= \frac{14.28\dot{m}_{diesel}+8.95\dot{m}_{vapor}}{\dot{m}_{diesel}+\dot{m}_{vapor}} \qquad (.12)\end{aligned}$$

SAMPLE CALCULATION Chapter 4 explained the methodology for calculating the parameters. A sample calculation based on a data point will be performed in this chapter of the appendices following the explanation order of chapter 4.

The following values are actual records of a data point that is measured during dual fuel mode test at 3500 rpm and 75% load, and will be used to demonstrate a sample calculation.

R.H. : 50%

Atmospheric Pressure : 1022 mbar

V_0 : 2.975V

V_1 :3.492V

V_2 : 3.022

V_3 : 2.941V

V_4 : 6.282V

V_5 : 0.005V

V_6 : 0.005V

V_7 : 0.005V

V_8 : 3.019V

V_9 : 1.139V

V_{10} : 3.895V

V_{11} : 3.477V

V_{12} : 1.833V

V_{13} : 0.09V

V_{14} : 2.16V

V_{15} : 0.005V

$\dot{m}_{diesel}$: 2.380 g/s

Ambient Temperature :

$$\begin{aligned} T_{amb} &= (V_{Tamb} - 2.73) \times 100 \\ &= (2.975 - 2.73) \times 100 \\ &= 24.5^{\circ}C \end{aligned}$$

Atmospheric Pressure :

$$\begin{aligned} P_{atm} &= \frac{P_{mbar} - 101}{10} \\ &= \frac{1022 - 101}{10} \\ &= 92.1\ kPa \end{aligned}$$

Vapor Pressure :

$$\begin{aligned} P_v &= \left(6.10708 \times 10^{\frac{7.5 \times T_{amb}}{237.3 + T_{amb}}}\right) \\ &= \left(6.10708 \times 10^{\frac{7.5 \times 21.3}{237.3 + 21.3}}\right) \\ &= 30.74\ mbar = 3.704\ kPa \end{aligned}$$

Pressure of dry air:

$$\begin{aligned} P_d &= P_{atm} - \frac{R.H. \times P_v}{100} \\ &= 92.1 - \frac{50 \times 3.704}{100} \\ &= 90.563\ kPa \end{aligned}$$

Engine speed:

$$\begin{aligned} \theta &= V_{\theta} \times C_{\theta} \\ &= 3.492 \times 1000 \\ &= 3492\ rpm \end{aligned}$$

Measured torque:

$$\begin{aligned} T_{meas} &= C_{torque} \times V_{torque} \\ &= 40 \times 3.022 \\ &= 120.88\ Nm \end{aligned}$$

Manifold absolute pressure :

$$\begin{aligned} P_{manifold} &= 65.8823 \times V_{MAP} - 6.3529 \\ &= 187.41\ kPa \end{aligned}$$

Turbo pressure coefficient

$$\begin{aligned} r &= \frac{P_{manifold}}{P_{atm}} \\ &= \frac{187.41}{90.563} \\ &= 2.069 \end{aligned}$$

Exhaust gas temperature :

$$\begin{aligned} V_{exhtc} &= \frac{V_{exhaust}}{G_{exhaust}} \\ &= \frac{6.282}{274.5} \\ &= 22.89\ mV \end{aligned}$$

This output voltage corresponds to 553°C in thermocouple properties table (table B.1).

Throttle Position:

$$\begin{aligned} TP &= 100 \times \frac{V_{throttle} - V_{throttle_min}}{V_{throttle_max} - V_{throttle_min}} \\ &= 100 \times \frac{3.019 - 1.05}{4.10 - 1.05} \\ &= 65.1\% \end{aligned}$$

Tube level :

$$\begin{aligned} TubeLevel &= C_{tube} \times V_{tube} \\ &= 1 \times 3.895 \\ &= 3.90\ l \end{aligned}$$

Ethanol flowrate :

$$\begin{aligned} \rho_{ethanol} &= \frac{P_{manifold} \times \dot{m}_{ethanol}}{(T_{cons} + 273) \times R_u} \\ &= \frac{187.41 \times 46}{(T_{cons} + 273) \times 8.314} \\ &= 2.735\ g/cc \end{aligned}$$

$$
\begin{aligned}
\dot{m}_{ethanol} &= (0.4705 \times V_{turbine} + 0.0041) \times \rho_{ethanol} \\
&= (0.4705 \times 1.139 + 0.0041) \times \rho_{ethanol} \\
&= 1.477\ g/s
\end{aligned}
$$

Vapor temperature and pressure in heat exchanger:

$$
\begin{aligned}
T_{ethanol,he} &= (V_{ethanol} - 0.0372) \times 56.875 \\
&= (3.477 - 0.0372) \times 56.875 \\
&= 195.63°C
\end{aligned}
$$

$$
\begin{aligned}
P_{ethanol} &= 126.45 \times V_{ethanol} + 24.806 \\
&= 126.45 \times 1.833 + 24.806 \\
&= 256.59\ kPa
\end{aligned}
$$

Ethanol Consumption Temperature:

$$
\begin{aligned}
V_{cons} &= \frac{V_{cons}}{G_{cons}} \\
&= \frac{2.16}{500} \\
&= 4.32\ mV
\end{aligned}
$$

This value corresponds to 105.41°C.

Air flowrate :

$$
\begin{aligned}
m_{air,\dot{a}ctual} &= 0.0167 \times F_{MAF} - 38.901 \\
&= 0.0167 \times F_{MAF} - 38.901 \\
&= 49.43\ g/s
\end{aligned}
$$

Calculation of power correction coefficient:

$$
\begin{aligned}
f_a &= \left(\frac{P_{d,ref}}{P_d}\right)^{0.7} \times \left(\frac{T_{amb} + 273}{T_{ref}}\right)^{0.7} \\
&= \left(\frac{99}{90.56}\right)^{0.7} \times \left(\frac{24.5 + 273}{298}\right)^{0.7} \\
&= 1.063
\end{aligned}
$$

$$
\begin{aligned}
q &= \frac{Z \times \dot{m}_{fuel}}{V \times \theta} \\
&= \frac{120000 \times (1.477 + 2.380)}{1.248 \times 3497} \\
&= 106.204
\end{aligned}
$$

$$
\begin{aligned}
q_c &= \frac{q}{r} \\
&= \frac{106.204}{2.069} \\
&= 51.331
\end{aligned}
$$

$$
\begin{aligned}
f_m &= 0.036 \times q_c - 1.14 \\
&= 0.036 \times 51.331 - 1.14 \\
&= 0.707
\end{aligned}
$$

$$
\begin{aligned}
\alpha_c &= (f_a)^{f_m} \\
&= (1.058)^{0.584} \\
&= 1.044
\end{aligned}
$$

Corrected torque :

$$
\begin{aligned}
T_{corr} &= \alpha_c \times T_{meas} \\
&= 1.044 \times 120.88 \\
&= 126.20\ Nm
\end{aligned}
\tag{.13}
$$

Engine power :

$$
\begin{aligned}
P_{corr} &= T_{corr} \times \theta \times 1.36 \times \frac{\pi}{30000} \\
&= 126.20 \times 3492 \times 1.36 \times \frac{\pi}{30000} \\
&= 62.76\ Hp
\end{aligned}
$$

Theoretical air flowrate :

$$\rho_{atm} = \frac{P_{amb}}{R_{air} \times T_{amb}}$$
$$\rho_{atm} = \frac{92.1}{0.287 \times 297.5}$$
$$\rho_{atm} = 1.079 kg/m^3 \ (1.079 g/l)$$

$$\begin{aligned} \dot{m}_{air,theoretical} &= \frac{V_{swept} \times \theta \times \rho_{atm} \times r}{Z \times 60} \\ &= \frac{1.248 \times 3492 \times 1.079 \times 2.069}{2 \times 60} \\ &= 81.075 \ g/s \end{aligned} \tag{.14}$$

Volumetric efficiency:

$$\begin{aligned} \eta_V &= 100 \times \frac{\dot{m}_{air,actual}}{\dot{m}_{air,theoretical}} \\ &= 100 \times \frac{49.43}{81.075} \\ &= 61.0\% \end{aligned}$$

Brake Specific Fuel Consumption (BSFC):

$$\begin{aligned} BSFC &= 3600 \times frac\dot{m}_{fuel}P_{corr} \\ &= 3600 \times frac1.477 + 2.38062.76 \\ &= 221.24 \ g/HP-h \end{aligned}$$

Brake Specific Energy Consumption:

$$\begin{aligned} BSEC &= 3600 \times \frac{\dot{m}_{diesel} \times Q_{diesel} + \dot{m}_{ethanol} \times Q_{ethanol}}{P_{corr}} \\ &= 3600 \times \frac{2.38 \times 10^{-3} \times 42600 + 1.477 \times 28800}{62.76} \\ &= 8255.77 \ kJ/HP-h \end{aligned}$$

Actual air / fuel ratio :

$$\begin{aligned} \left(\frac{A}{F}\right)_{actual} &= \frac{\dot{m}_{air,actual}}{\dot{m}_{fuel}} \\ &= \frac{49.43}{2.380 + 1.477} \\ &= 12.816 \end{aligned}$$

Theoretical air / fuel ratio

$$
\begin{aligned}
\lambda &= \frac{14.28 \times \dot{m}_{diesel} + 8.95 \times \dot{m}_{ethanol}}{\dot{m}_{diesel} + \dot{m}_{ethanol}} \\
\lambda &= \frac{14.28 \times 2.380 + 8.95 \times 1.477}{2.380 + 1.477} \\
\lambda &= 12.238
\end{aligned}
$$

Excess Air Coefficient :

$$
\begin{aligned}
\lambda &= \frac{\left(\frac{A}{F}\right)_{actual}}{\left(\frac{A}{F}\right)_{stoic}} \\
\lambda &= \frac{12.816}{12.238} \\
\lambda &= 1.047
\end{aligned}
$$

Equivalence ratio :

$$
\begin{aligned}
\phi &= \frac{1}{\lambda} \\
\phi &= \frac{1}{1.0047} \\
\phi &= 0.955
\end{aligned}
$$

Thermal efficiency :

$$
\begin{aligned}
\dot{m}_{diesel} &= 2.380 g/s \times \frac{3600 s}{1 hour} \times \frac{1 kg}{1000 g} \\
&= 8.568 kg/h \\
\dot{m}_{ethanol} &= 3.90 g/s \times \frac{3600 s}{1 hour} \times \frac{1 kg}{1000 g} \\
&= 5.317 kg/h \\
\eta_{th} &= \frac{P_{corr} \times 3600}{(\dot{m}_{diesel} \times Q_{L,diesel} + \dot{m}_{eth} \times Q_{L,eth})\, 1.341} \\
&= \frac{62.76 \times 3600}{(8.568 \times 42600 + 5.317 \times 28800) \times 1.341} \\
&= 0.317\, (31.7\%)
\end{aligned}
$$

APPENDIX A

WIRING LOCATIONS OF MEASUREMENT AND CONTOL CABLES

A.1 CONNECTION OF STEPPER MOTORS

The valves mentioned on the following list are controlled using stepper motors, and wired to the stepper controller. Stepper controller connection is not related to data acquisition card, and is done over serial port of the computer. Connections to the card are listed below:

1. Channel 1 : Bypass route butterfly valve
2. Channel 2 : Evaporator Heat Exchanger butterfly valve
3. Channel 3 : Superheater Heat Exchanger butterfly valve
4. Channel 4 : Needle valve

A.2 ANALOG INPUT CHANNELS

Measurement devices with analog outputs are wired to data acquisition card over terminal card. Channel are enumerated from 0 to 15, making a total of 16 analog input channels. Connection channels of the elements are listed below:

1. V0 : Ambient Temperature sensor, integrated to the terminal card
2. V1 : Engine speed, signal is sourced from dynamometer
3. V2 : Engine torque, signal is sourced from dynamometer
4. V3 : Manifold Pressure, obtained from manifold absolute pressure sensor
5. V4 : Exhaust temperature, collected from amplified voltage of exhaust thermocouple
6. V5 : Not used, shunted
7. V6 : Not used, shunted
8. V7 : Not used, shunted
9. V8 : Throttle position potentiometer, collected from the throttle pedal of the engine
10. V9 : Ethanol flowrate, collected from FTB 930 series turbine meter after frequency-voltage conversion
11. V10 : Pressurized tube level, collected from floater driven potentiometer
12. V11: Superheated ethanol temperature, collected from amplifier of the related thermocouple
13. V12 : Superheated ethanol pressure, collected from amplifier of pressure sensor
14. V13 : Not used, shunted
15. V14 : Not used, shunted
16. V15 : Not used, shunted

A.3 DIGITAL INPUT CHANNELS

Measurement devices with digital outputs are wired to data acquisition card over secondary terminal card. Enumeration of channels are the same as analog inputs. Connection channels of the elements are listed below:

1. DI0 : Global channel for checking connection of secondary terminal card
2. DI1 : Global channel for checking status of the dynamometer
3. DI2 : Low level indicator optical sensor connection for diesel fuel measurement device
4. DI3 : High level indicator optical sensor connection for diesel fuel measurement device
5. DI4 : Throttle controller safety microswitch, used with pull-up resistor
6. DI5 : Not used
7. DI6 : Ethanol level indicator switch of heat exchanger, used with pull-up resistor
8. DI7 : Not used
9. DI8 : Not used
10. DI9 : Not used
11. DI10 : Not used
12. DI11 : Not used
13. DI12 : Not used
14. DI13 : Not used
15. DI14 : Not used
16. DI15 : Not used

A.4 DIGITAL OUTPUT CHANNELS

Control devices that require relay connections are wired over secondary terminal card, and others are wired over the main terminal card. Connection channels of the elements are as follows:

1. DO0 : Driver channel for engine key on/off relays,12V
2. DO1 : Dynamometer control signal, 12V
3. DO2 : Dynamometer control signal, 12V
4. DO3 : Not used
5. DO4 : Flush solenoid for diesel fuel measurement chamber calibration 12V
6. DO5 : Throttle controller pneumatic piston controller, 24V
7. DO6 : Ethanol pump connection, 12V
8. DO7 : Fuel consumption on/off valve, 24V
9. DO8 : Heat exchanger refill valve, 24V
10. DO9 : Compressed air solenoid of pressurized tube, 24V
11. DO10 : Pressurized tube refill valve solenoid, 24V
12. DO11 : Superheated vapor return valve solenoid, 24V
13. DO12 : Heat exchanger pressurizer valve solenoid, 24V
14. DO13 : Not used
15. DO14 : 24V supply relay for throttle pedal controller motor, 24V
16. DO15 : Direction control relay for throttle pedal controller motor, 12V

APPENDIX B

SPECIFICATION OF EXPERIMENTAL DEVICES

B.1 K TYPE THEMOCOUPLE PROPERTIES TABLE

Table B.1: ITS-90 Table for Type K Thermocouples

°C	0	-1	-2	-3	-4	-5	-6	-7	-8	-9	-10
-270	-6.458										
-260	-6.441	-6.444	-6.446	-6.448	-6.450	-6.452	-6.453	-6.455	-6.456	-6.457	-6.458
-250	-6.404	-6.408	-6.413	-6.417	-6.421	-6.425	-6.429	-6.432	-6.435	-6.438	-6.441
-240	-6.344	-6.351	-6.358	-6.364	-6.370	-6.377	-6.382	-6.388	-6.393	-6.399	-6.404
-230	-6.262	-6.271	-6.280	-6.289	-6.297	-6.306	-6.314	-6.322	-6.329	-6.337	-6.344
-220	-6.158	-6.170	-6.181	-6.192	-6.202	-6.213	-6.223	-6.233	-6.243	-6.252	-6.262
-210	-6.035	-6.048	-6.061	-6.074	-6.087	-6.099	-6.111	-6.123	-6.135	-6.147	-6.158
-200	-5.891	-5.907	-5.922	-5.936	-5.951	-5.965	-5.980	-5.994	-6.007	-6.021	-6.035
-190	-5.730	-5.747	-5.763	-5.780	-5.797	-5.813	-5.829	-5.845	-5.861	-5.876	-5.891
-180	-5.550	-5.569	-5.588	-5.606	-5.624	-5.642	-5.660	-5.678	-5.695	-5.713	-5.730
-170	-5.354	-5.374	-5.395	-5.415	-5.435	-5.454	-5.474	-5.493	-5.512	-5.531	-5.550
-160	-5.141	-5.163	-5.185	-5.207	-5.228	-5.250	-5.271	-5.292	-5.313	-5.333	-5.354
-150	-4.913	-4.936	-4.960	-4.983	-5.006	-5.029	-5.052	-5.074	-5.097	-5.119	-5.141
-140	-4.669	-4.694	-4.719	-4.744	-4.768	-4.793	-4.817	-4.841	-4.865	-4.889	-4.913
-130	-4.411	-4.437	-4.463	-4.490	-4.516	-4.542	-4.567	-4.593	-4.618	-4.644	-4.669
-120	-4.138	-4.166	-4.194	-4.221	-4.249	-4.276	-4.303	-4.330	-4.357	-4.384	-4.411
-110	-3.852	-3.882	-3.911	-3.939	-3.968	-3.997	-4.025	-4.054	-4.082	-4.110	-4.138
-100	-3.554	-3.584	-3.614	-3.645	-3.675	-3.705	-3.734	-3.764	-3.794	-3.823	-3.852

Table B.1ITS-90 Table for Type K Thermocouple – continued

°C	**0**	**-1**	**-2**	**-3**	**-4**	**-5**	**-6**	**-7**	**-8**	**-9**	**-10**
-90	-3.243	-3.274	-3.306	-3.337	-3.368	-3.400	-3.431	-3.462	-3.492	-3.523	-3.554
-80	-2.920	-2.953	-2.986	-3.018	-3.050	-3.083	-3.115	-3.147	-3.179	-3.211	-3.243
-70	-2.587	-2.620	-2.654	-2.688	-2.721	-2.755	-2.788	-2.821	-2.854	-2.887	-2.920
-60	-2.243	-2.278	-2.312	-2.347	-2.382	-2.416	-2.450	-2.485	-2.519	-2.553	-2.587
-50	-1.889	-1.925	-1.961	-1.996	-2.032	-2.067	-2.103	-2.138	-2.173	-2.208	-2.243
-40	-1.527	-1.564	-1.600	-1.637	-1.673	-1.709	-1.745	-1.782	-1.818	-1.854	-1.889
-30	-1.156	-1.194	-1.231	-1.268	-1.305	-1.343	-1.380	-1.417	-1.453	-1.490	-1.527
-20	-0.778	-0.816	-0.854	-0.892	-0.930	-0.968	-1.006	-1.043	-1.081	-1.119	-1.156
-10	-0.392	-0.431	-0.470	-0.508	-0.547	-0.586	-0.624	-0.663	-0.701	-0.739	-0.778
0	0.000	-0.039	-0.079	-0.118	-0.157	-0.197	-0.236	-0.275	-0.314	-0.353	-0.392
°C	0	-1	-2	-3	-4	-5	-6	-7	-8	-9	-10

°C	0	1	2	3	4	5	6	7	8	9	10
0	0.000	0.039	0.079	0.119	0.158	0.198	0.238	0.277	0.317	0.357	0.397
10	0.397	0.437	0.477	0.517	0.557	0.597	0.637	0.677	0.718	0.758	0.798
20	0.798	0.838	0.879	0.919	0.960	1.000	1.041	1.081	1.122	1.163	1.203
30	1.203	1.244	1.285	1.326	1.366	1.407	1.448	1.489	1.530	1.571	1.612
40	1.612	1.653	1.694	1.735	1.776	1.817	1.858	1.899	1.941	1.982	2.023
50	2.023	2.064	2.106	2.147	2.188	2.230	2.271	2.312	2.354	2.395	2.436
60	2.436	2.478	2.519	2.561	2.602	2.644	2.685	2.727	2.768	2.810	2.851
70	2.851	2.893	2.934	2.976	3.017	3.059	3.100	3.142	3.184	3.225	3.267
80	3.267	3.308	3.350	3.391	3.433	3.474	3.516	3.557	3.599	3.640	3.682
90	3.682	3.723	3.765	3.806	3.848	3.889	3.931	3.972	4.013	4.055	4.096
100	4.096	4.138	4.179	4.220	4.262	4.303	4.344	4.385	4.427	4.468	4.509
110	4.509	4.550	4.591	4.633	4.674	4.715	4.756	4.797	4.838	4.879	4.920
120	4.920	4.961	5.002	5.043	5.084	5.124	5.165	5.206	5.247	5.288	5.328
130	5.328	5.369	5.410	5.450	5.491	5.532	5.572	5.613	5.653	5.694	5.735
140	5.735	5.775	5.815	5.856	5.896	5.937	5.977	6.017	6.058	6.098	6.138
150	6.138	6.179	6.219	6.259	6.299	6.339	6.380	6.420	6.460	6.500	6.540
160	6.540	6.580	6.620	6.660	6.701	6.741	6.781	6.821	6.861	6.901	6.941
170	6.941	6.981	7.021	7.060	7.100	7.140	7.180	7.220	7.260	7.300	7.340

Table B.1ITS-90 Table for Type K Thermocouple – continued

°C	**0**	**-1**	**-2**	**-3**	**-4**	**-5**	**-6**	**-7**	**-8**	**-9**	**-10**
180	7.340	7.380	7.420	7.460	7.500	7.540	7.579	7.619	7.659	7.699	7.739
190	7.739	7.779	7.819	7.859	7.899	7.939	7.979	8.019	8.059	8.099	8.138
200	8.138	8.178	8.218	8.258	8.298	8.338	8.378	8.418	8.458	8.499	8.539
210	8.539	8.579	8.619	8.659	8.699	8.739	8.779	8.819	8.860	8.900	8.940
220	8.940	8.980	9.020	9.061	9.101	9.141	9.181	9.222	9.262	9.302	9.343
230	9.343	9.383	9.423	9.464	9.504	9.545	9.585	9.626	9.666	9.707	9.747
240	9.747	9.788	9.828	9.869	9.909	9.950	9.991	10.031	10.072	10.113	10.153
250	10.153	10.194	10.235	10.276	10.316	10.357	10.398	10.439	10.480	10.520	10.561
260	10.561	10.602	10.643	10.684	10.725	10.766	10.807	10.848	10.889	10.930	10.971
270	10.971	11.012	11.053	11.094	11.135	11.176	11.217	11.259	11.300	11.341	11.382
280	11.382	11.423	11.465	11.506	11.547	11.588	11.630	11.671	11.712	11.753	11.795
290	11.795	11.836	11.877	11.919	11.960	12.001	12.043	12.084	12.126	12.167	12.209
300	12.209	12.250	12.291	12.333	12.374	12.416	12.457	12.499	12.540	12.582	12.624
310	12.624	12.665	12.707	12.748	12.790	12.831	12.873	12.915	12.956	12.998	13.040
320	13.040	13.081	13.123	13.165	13.206	13.248	13.290	13.331	13.373	13.415	13.457
330	13.457	13.498	13.540	13.582	13.624	13.665	13.707	13.749	13.791	13.833	13.874
340	13.874	13.916	13.958	14.000	14.042	14.084	14.126	14.167	14.209	14.251	14.293
350	14.293	14.335	14.377	14.419	14.461	14.503	14.545	14.587	14.629	14.671	14.713
360	14.713	14.755	14.797	14.839	14.881	14.923	14.965	15.007	15.049	15.091	15.133
370	15.133	15.175	15.217	15.259	15.301	15.343	15.385	15.427	15.469	15.511	15.554
380	15.554	15.596	15.638	15.680	15.722	15.764	15.806	15.849	15.891	15.933	15.975
390	15.975	16.017	16.059	16.102	16.144	16.186	16.228	16.270	16.313	16.355	16.397
400	16.397	16.439	16.482	16.524	16.566	16.608	16.651	16.693	16.735	16.778	16.820
410	16.820	16.862	16.904	16.947	16.989	17.031	17.074	17.116	17.158	17.201	17.243
420	17.243	17.285	17.328	17.370	17.413	17.455	17.497	17.540	17.582	17.624	17.667
430	17.667	17.709	17.752	17.794	17.837	17.879	17.921	17.964	18.006	18.049	18.091
440	18.091	18.134	18.176	18.218	18.261	18.303	18.346	18.388	18.431	18.473	18.516
450	18.516	18.558	18.601	18.643	18.686	18.728	18.771	18.813	18.856	18.898	18.941
460	18.941	18.983	19.026	19.068	19.111	19.154	19.196	19.239	19.281	19.324	19.366
470	19.366	19.409	19.451	19.494	19.537	19.579	19.622	19.664	19.707	19.750	19.792

Table B.1ITS-90 Table for Type K Thermocouple – continued

°C	**0**	**-1**	**-2**	**-3**	**-4**	**-5**	**-6**	**-7**	**-8**	**-9**	**-10**
480	19.792	19.835	19.877	19.920	19.962	20.005	20.048	20.090	20.133	20.175	20.218
490	20.218	20.261	20.303	20.346	20.389	20.431	20.474	20.516	20.559	20.602	20.644
510	20.644	20.687	20.730	20.772	20.815	20.857	20.900	20.943	20.985	21.028	21.071
510	21.071	21.113	21.156	21.199	21.241	21.284	21.326	21.369	21.412	21.454	21.497
520	21.497	21.540	21.582	21.625	21.668	21.710	21.753	21.796	21.838	21.881	21.924
530	21.924	21.966	22.009	22.052	22.094	22.137	22.179	22.222	22.265	22.307	22.350
540	22.350	22.393	22.435	22.478	22.521	22.563	22.606	22.649	22.691	22.734	22.776
550	22.776	22.819	22.862	22.904	22.947	22.990	23.032	23.075	23.117	23.160	23.203
560	23.203	23.245	23.288	23.331	23.373	23.416	23.458	23.501	23.544	23.586	23.629
570	23.629	23.671	23.714	23.757	23.799	23.842	23.884	23.927	23.970	24.012	24.055
580	24.055	24.097	24.140	24.182	24.225	24.267	24.310	24.353	24.395	24.438	24.480
590	24.480	24.523	24.565	24.608	24.650	24.693	24.735	24.778	24.820	24.863	24.905
600	24.905	24.948	24.990	25.033	25.075	25.118	25.160	25.203	25.245	25.288	25.330
610	25.330	25.373	25.415	25.458	25.500	25.543	25.585	25.627	25.670	25.712	25.755
620	25.755	25.797	25.840	25.882	25.924	25.967	26.009	26.052	26.094	26.136	26.179
630	26.179	26.221	26.263	26.306	26.348	26.390	26.433	26.475	26.517	26.560	26.602
640	26.602	26.644	26.687	26.729	26.771	26.814	26.856	26.898	26.940	26.983	27.025
650	27.025	27.067	27.109	27.152	27.194	27.236	27.278	27.320	27.363	27.405	27.447
660	27.447	27.489	27.531	27.574	27.616	27.658	27.700	27.742	27.784	27.826	27.869
670	27.869	27.911	27.953	27.995	28.037	28.079	28.121	28.163	28.205	28.247	28.289
680	28.289	28.332	28.374	28.416	28.458	28.500	28.542	28.584	28.626	28.668	28.710
690	28.710	28.752	28.794	28.835	28.877	28.919	28.961	29.003	29.045	29.087	29.129
700	29.129	29.171	29.213	29.255	29.297	29.338	29.380	29.422	29.464	29.506	29.548
710	29.548	29.589	29.631	29.673	29.715	29.757	29.798	29.840	29.882	29.924	29.965
720	29.965	30.007	30.049	30.090	30.132	30.174	30.216	30.257	30.299	30.341	30.382
730	30.382	30.424	30.466	30.507	30.549	30.590	30.632	30.674	30.715	30.757	30.798
740	30.798	30.840	30.881	30.923	30.964	31.006	31.047	31.089	31.130	31.172	31.213
750	31.213	31.255	31.296	31.338	31.379	31.421	31.462	31.504	31.545	31.586	31.628
760	31.628	31.669	31.710	31.752	31.793	31.834	31.876	31.917	31.958	32.000	32.041
770	32.041	32.082	32.124	32.165	32.206	32.247	32.289	32.330	32.371	32.412	32.453

Table B.1ITS-90 Table for Type K Thermocouple – continued

°C	0	-1	-2	-3	-4	-5	-6	-7	-8	-9	-10
780	32.453	32.495	32.536	32.577	32.618	32.659	32.700	32.742	32.783	32.824	32.865
790	32.865	32.906	32.947	32.988	33.029	33.070	33.111	33.152	33.193	33.234	33.275
800	33.275	33.316	33.357	33.398	33.439	33.480	33.521	33.562	33.603	33.644	33.685
810	33.685	33.726	33.767	33.808	33.848	33.889	33.930	33.971	34.012	34.053	34.093
820	34.093	34.134	34.175	34.216	34.257	34.297	34.338	34.379	34.420	34.460	34.501
830	34.501	34.542	34.582	34.623	34.664	34.704	34.745	34.786	34.826	34.867	34.908
840	34.908	34.948	34.989	35.029	35.070	35.110	35.151	35.192	35.232	35.273	35.313
850	35.313	35.354	35.394	35.435	35.475	35.516	35.556	35.596	35.637	35.677	35.718
860	35.718	35.758	35.798	35.839	35.879	35.920	35.960	36.000	36.041	36.081	36.121
870	36.121	36.162	36.202	36.242	36.282	36.323	36.363	36.403	36.443	36.484	36.524
880	36.524	36.564	36.604	36.644	36.685	36.725	36.765	36.805	36.845	36.885	36.925
890	36.925	36.965	37.006	37.046	37.086	37.126	37.166	37.206	37.246	37.286	37.326
900	37.326	37.366	37.406	37.446	37.486	37.526	37.566	37.606	37.646	37.686	37.725
910	37.725	37.765	37.805	37.845	37.885	37.925	37.965	38.005	38.044	38.084	38.124
920	38.124	38.164	38.204	38.243	38.283	38.323	38.363	38.402	38.442	38.482	38.522
930	38.522	38.561	38.601	38.641	38.680	38.720	38.760	38.799	38.839	38.878	38.918
940	38.918	38.958	38.997	39.037	39.076	39.116	39.155	39.195	39.235	39.274	39.314
950	39.314	39.353	39.393	39.432	39.471	39.511	39.550	39.590	39.629	39.669	39.708
960	39.708	39.747	39.787	39.826	39.866	39.905	39.944	39.984	40.023	40.062	40.101
970	40.101	40.141	40.180	40.219	40.259	40.298	40.337	40.376	40.415	40.455	40.494
980	40.494	40.533	40.572	40.611	40.651	40.690	40.729	40.768	40.807	40.846	40.885
990	40.885	40.924	40.963	41.002	41.042	41.081	41.120	41.159	41.198	41.237	41.276
1000	41.276	41.315	41.354	41.393	41.431	41.470	41.509	41.548	41.587	41.626	41.665
1010	41.665	41.704	41.743	41.781	41.820	41.859	41.898	41.937	41.976	42.014	42.053
1020	42.053	42.092	42.131	42.169	42.208	42.247	42.286	42.324	42.363	42.402	42.440
1030	42.440	42.479	42.518	42.556	42.595	42.633	42.672	42.711	42.749	42.788	42.826
1040	42.826	42.865	42.903	42.942	42.980	43.019	43.057	43.096	43.134	43.173	43.211
1050	43.211	43.250	43.288	43.327	43.365	43.403	43.442	43.480	43.518	43.557	43.595
1060	43.595	43.633	43.672	43.710	43.748	43.787	43.825	43.863	43.901	43.940	43.978
1070	43.978	44.016	44.054	44.092	44.130	44.169	44.207	44.245	44.283	44.321	44.359

Table B.1ITS-90 Table for Type K Thermocouple – continued

°C	0	-1	-2	-3	-4	-5	-6	-7	-8	-9	-10
1080	44.359	44.397	44.435	44.473	44.512	44.550	44.588	44.626	44.664	44.702	44.740
1090	44.740	44.778	44.816	44.853	44.891	44.929	44.967	45.005	45.043	45.081	45.119
1100	45.119	45.157	45.194	45.232	45.270	45.308	45.346	45.383	45.421	45.459	45.497
1110	45.497	45.534	45.572	45.610	45.647	45.685	45.723	45.760	45.798	45.836	45.873
1120	45.873	45.911	45.948	45.986	46.024	46.061	46.099	46.136	46.174	46.211	46.249
1130	46.249	46.286	46.324	46.361	46.398	46.436	46.473	46.511	46.548	46.585	46.623
1140	46.623	46.660	46.697	46.735	46.772	46.809	46.847	46.884	46.921	46.958	46.995
1150	46.995	47.033	47.070	47.107	47.144	47.181	47.218	47.256	47.293	47.330	47.367
1160	47.367	47.404	47.441	47.478	47.515	47.552	47.589	47.626	47.663	47.700	47.737
1170	47.737	47.774	47.811	47.848	47.884	47.921	47.958	47.995	48.032	48.069	48.105
1180	48.105	48.142	48.179	48.216	48.252	48.289	48.326	48.363	48.399	48.436	48.473
1190	48.473	48.509	48.546	48.582	48.619	48.656	48.692	48.729	48.765	48.802	48.838
1200	48.838	48.875	48.911	48.948	48.984	49.021	49.057	49.093	49.130	49.166	49.202
1210	49.202	49.239	49.275	49.311	49.348	49.384	49.420	49.456	49.493	49.529	49.565
1220	49.565	49.601	49.637	49.674	49.710	49.746	49.782	49.818	49.854	49.890	49.926
1230	49.926	49.962	49.998	50.034	50.070	50.106	50.142	50.178	50.214	50.250	50.286
1240	50.286	50.322	50.358	50.393	50.429	50.465	50.501	50.537	50.572	50.608	50.644
1250	50.644	50.680	50.715	50.751	50.787	50.822	50.858	50.894	50.929	50.965	51.000
1260	51.000	51.036	51.071	51.107	51.142	51.178	51.213	51.249	51.284	51.320	51.355
1270	51.355	51.391	51.426	51.461	51.497	51.532	51.567	51.603	51.638	51.673	51.708
1280	51.708	51.744	51.779	51.814	51.849	51.885	51.920	51.955	51.990	52.025	52.060
1290	52.060	52.095	52.130	52.165	52.200	52.235	52.270	52.305	52.340	52.375	52.410
1300	52.410	52.445	52.480	52.515	52.550	52.585	52.620	52.654	52.689	52.724	52.759
1310	52.759	52.794	52.828	52.863	52.898	52.932	52.967	53.002	53.037	53.071	53.106
1320	53.106	53.140	53.175	53.210	53.244	53.279	53.313	53.348	53.382	53.417	53.451
1330	53.451	53.486	53.520	53.555	53.589	53.623	53.658	53.692	53.727	53.761	53.795
1340	53.795	53.830	53.864	53.898	53.932	53.967	54.001	54.035	54.069	54.104	54.138
1350	54.138	54.172	54.206	54.240	54.274	54.308	54.343	54.377	54.411	54.445	54.479
1360	54.479	54.513	54.547	54.581	54.615	54.649	54.683	54.717	54.751	54.785	54.819
1370	54.819	54.852	54.886								

B.2 OMEGA PX120 PRESSURE SENSOR

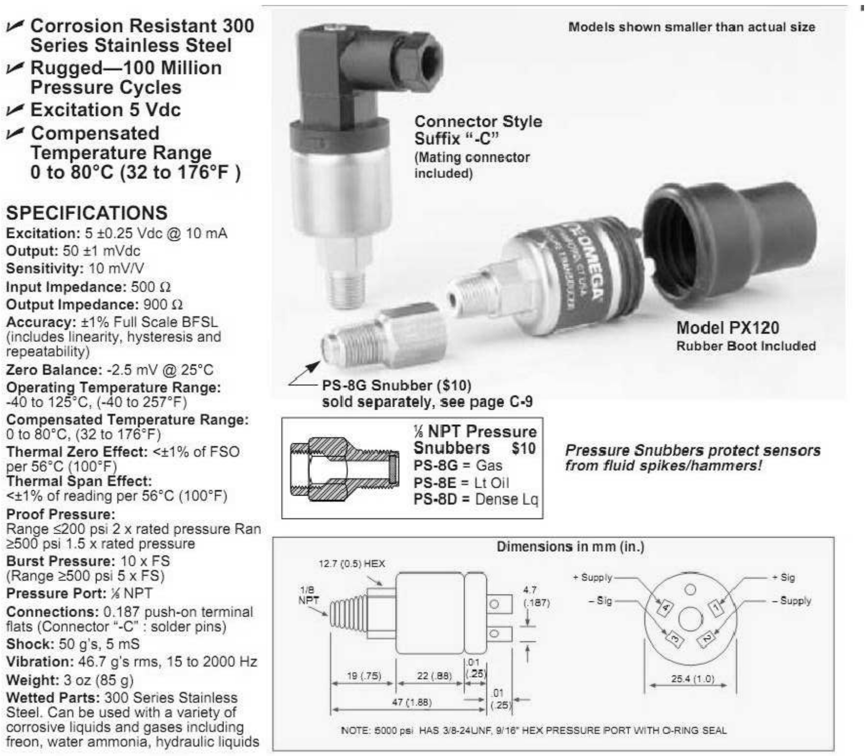
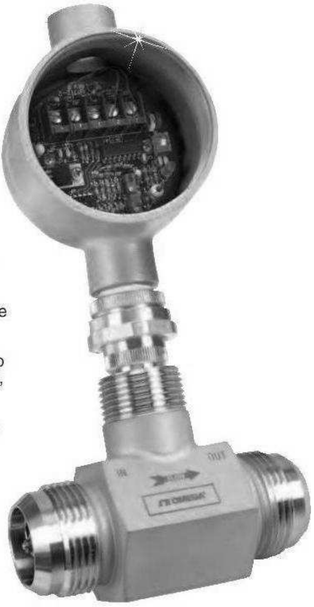


Figure B.1: Properties of Pressure Sensor

B.3 OMEGA FTB 931 TURBINEMETER

GAS TURBINE FLOWMETERS
with Male NPT Fittings

MADE IN USA | 1 YEAR WARRANTY | NIST Standard | RoHS



- ±1% of Reading Accuracy
- Ball Bearing Design for Economy
- 440C Bearing Retainers for Durability
- Ball Bearings Field Replaceable Without Loss of Calibration
- Deflector Cones Stabilize Low-Mass Rotor for Increased Bearing Life
- NIST Certificate Supplied Standard

The OMEGA® FTB-930 and FTB-940 Series turbine meters have male NPT end fittings for easy connections. They are suitable for use with gases with a minimum density of 0.025 lb/ft³ (air at STP is 0.0752 lb/ft³). These units come supplied with a mating 2-wire connector and can be supplied with integrally mounted FLSC-60 Series signal conditioners to provide 4 to 20 mA, 0 to 5V, or amplified pulse outputs. These turbine gas meters are intended for clean service only; where there is any doubt about the fluid, strainers are recommended.

OMEGA® precision turbine gas flowmeters are designed to measure actual cubic feet or actual volume passing through the meter. Before sizing a flowmeter, it is necessary to convert flow units (*i.e.*, SCFM, LPM, etc.) to actual units. To convert to actual measured volume (ACFM) from standard volume (SCFM), use the following formula:

ACFM = SCFM x 14.7/Pa x Ta/530

where:

ACFM = Actual cubic feet per minute of measured gas flow
SCFM = Standard cubic feet per minute gas flow
Pa = Operating pressure (psia) in = (psig + 14.7)
Ta = Temperature in degrees Rankine = (°F + 460)

SPECIFICATIONS

Accuracy: ±1.0% of reading
Repeatability: ±0.25%
Materials of Construction:
Body: 304/316 stainless steel
Rotor: 17-4 PH SS
Bearings: 440C stainless steel
Output: 30 mV p-p sinewave min.
Electrical: 2-wire connector included
Max Temp. Range: -267 to 232°C (-450 to 450°F)
Maximum Pressure Drop: Gas density in lb/ft³ x 4 (psid)

Compatible Meters: DPF701 (page M-5), DPF402 (page M-11), DPF70 with FLSC-AMP (page M-22), DPF10 (page M-23), FC21, 22 (page M-27)

MOST POPULAR MODELS HIGHLIGHTED!

To Order *(Specify Model Number)*

Model No.	Price	Maximum Linear Range (ACFM)	Nominal K-Factor (p/ACF)	Operating Pressure (psi)	Length mm (in)	End Connections MNPT (in)	Approx. Weight. kg (lb)
FTB-931B	$1060	0.35 to 3.5	24,200	5000	76 (3)	½	1 (2)
FTB-932	1060	0.75 to 5	14,890	5000	76 (3)	½	1 (2)
FTB-933	1060	1 to 10	14,580	5000	76 (3)	½	1 (2)
FTB-934	1060	2 to 20	8790	5000	76 (3)	½	1 (2)
FTB-935	1060	2.5 to 28	3660	4000	83 (3.25)	¾	2 (4)
FTB-936	1065	4 to 60	880	3000	89 (3.50)	1	2.5 (5)
FTB-937	1096	6 to 100	720	2500	98 (3.88)	1¼	3 (7)
FTB-938	1205	8 to 130	430	2000	111 (4.38)	1½	3.5 (8)
FTB-939	1335	15 to 250	230	1500	121 (4.75)	2	6 (13)
FTB-940	1550	25 to 450	135	1500	154 (6.06)	2½	8 (18)
FTB-941	1997	40 to 650	74	1250	254 (10)	3	8.5 (19)

Figure B.2: Catalog Information of the Turbinemeter

B.4 MASS AIR PRESSURE SENSOR

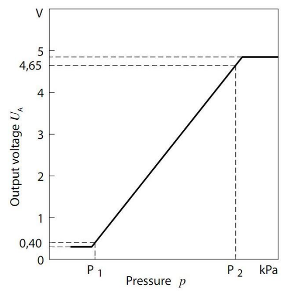
Part number **0 281 002 487**

Technical data

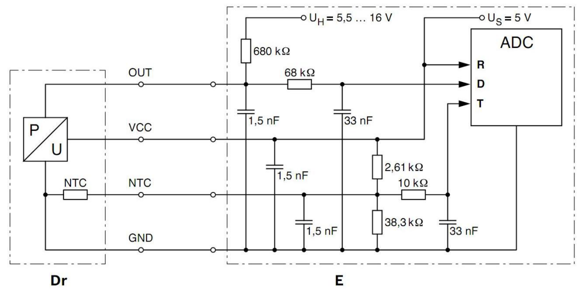
Parameter			min	type	max
Pressure range kPa (p_1...p_2)			20		250
Operating temperature	ϑ_B	°C	-40		+130
Supply voltage (1 min)	U_V	V	4,5	5	5,5
Load resistance to U_V or ground	$R_{pull\text{-}up}$	kΩ	5	680	
Load resistance to U_V or ground	$R_{pull\text{-}down}$	kΩ	10	100	
Response time	$\tau_{10/90}$	m		1	

Illustration

Characteristic curve

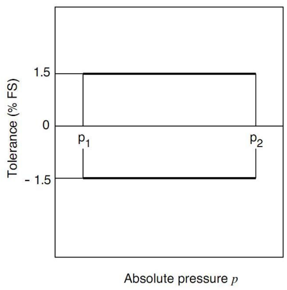


Recommendation for signal evaluation

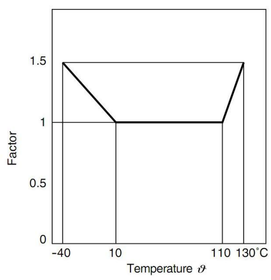


R Reference
D Pressure signal
T Temperature signal
Dr Pressure sensor
E Electronic control unit

Characteristic-curve tolerance



Tolerance extension factor

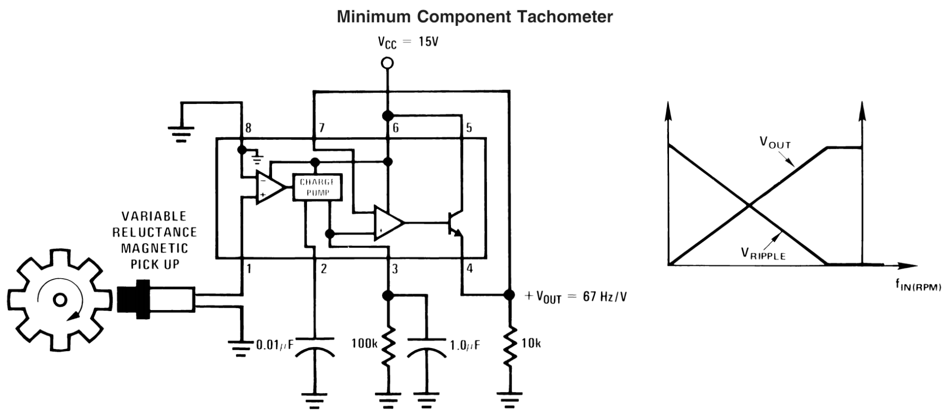


Accessories		Part number
Connector housing	Quantity required: 1 x	1 928 403 966
Contact pins	Quantity required: 3 x; Contents: 100 x	1 928 498 060
Individual seals	Quantity required: 3 x; Contents: 10 x	1 928 300 599

Accessories are not included in the scope of delivery of the sensor and are therefore to be ordered separately as required.

Figure B.3: Properties of Manifold Absolute Pressure Sensor

B.5 FREQUENCY TO VOLTAGE CONVERTER



(a) Circuit Schematic of Frequency to Voltage Converter

V_{CC} = 12 V_{DC}, T_A = 25˚C, see test circuit

Symbol	Parameter	Conditions	Min	Typ	Max	Units
TACHOMETER						
	Input Thresholds	V_{IN} = 250 mVp-p @ 1 kHz (Note 2)	±10	±25	±40	mV
	Hysteresis	V_{IN} = 250 mVp-p @ 1 kHz (Note 2)		30		mV
	Offset Voltage	V_{IN} = 250 mVp-p @ 1 kHz (Note 2)				
	LM2907/LM2917			3.5	10	mV
	LM2907-8/LM2917-8			5	15	mV
	Input Bias Current	V_{IN} = ±50 mV_{DC}		0.1	1	µA
V_{OH}	Pin 2	V_{IN} = +125 mV_{DC} (Note 3)		8.3		V
V_{OL}	Pin 2	V_{IN} = −125 mV_{DC} (Note 3)		2.3		V
I_2, I_3	Output Current	V2 = V3 = 6.0V (Note 4)	140	180	240	µA
I_3	Leakage Current	I2 = 0, V3 = 0			0.1	µA
K	Gain Constant	(Note 3)	0.9	1.0	1.1	
	Linearity	f_{IN} = 1 kHz, 5 kHz, 10 kHz (Note 5)	−1.0	0.3	+1.0	%

(b) Electrical Characteristics of Frequency to Voltage Converter

Figure B.4: Properties of LM2907 Frequency to Voltage Converter

APPENDIX C

TECHNICAL DRAWINGS

This chapter of appendices will display technical drawings of purpose built devices and their components. All the necessary information are avalable on the drafts.

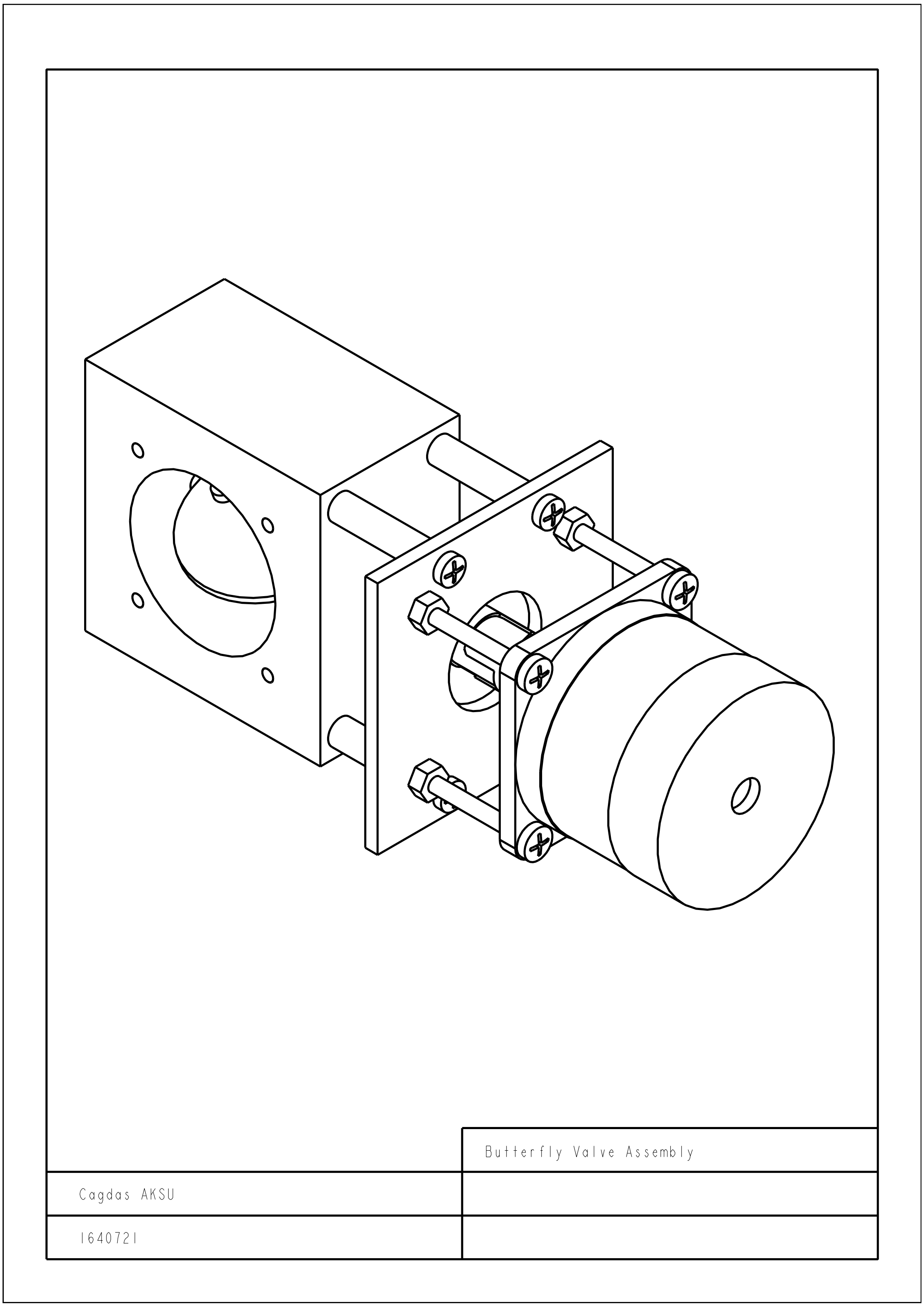


Figure C.1: Isometric View of Butterfly Valve Assembly

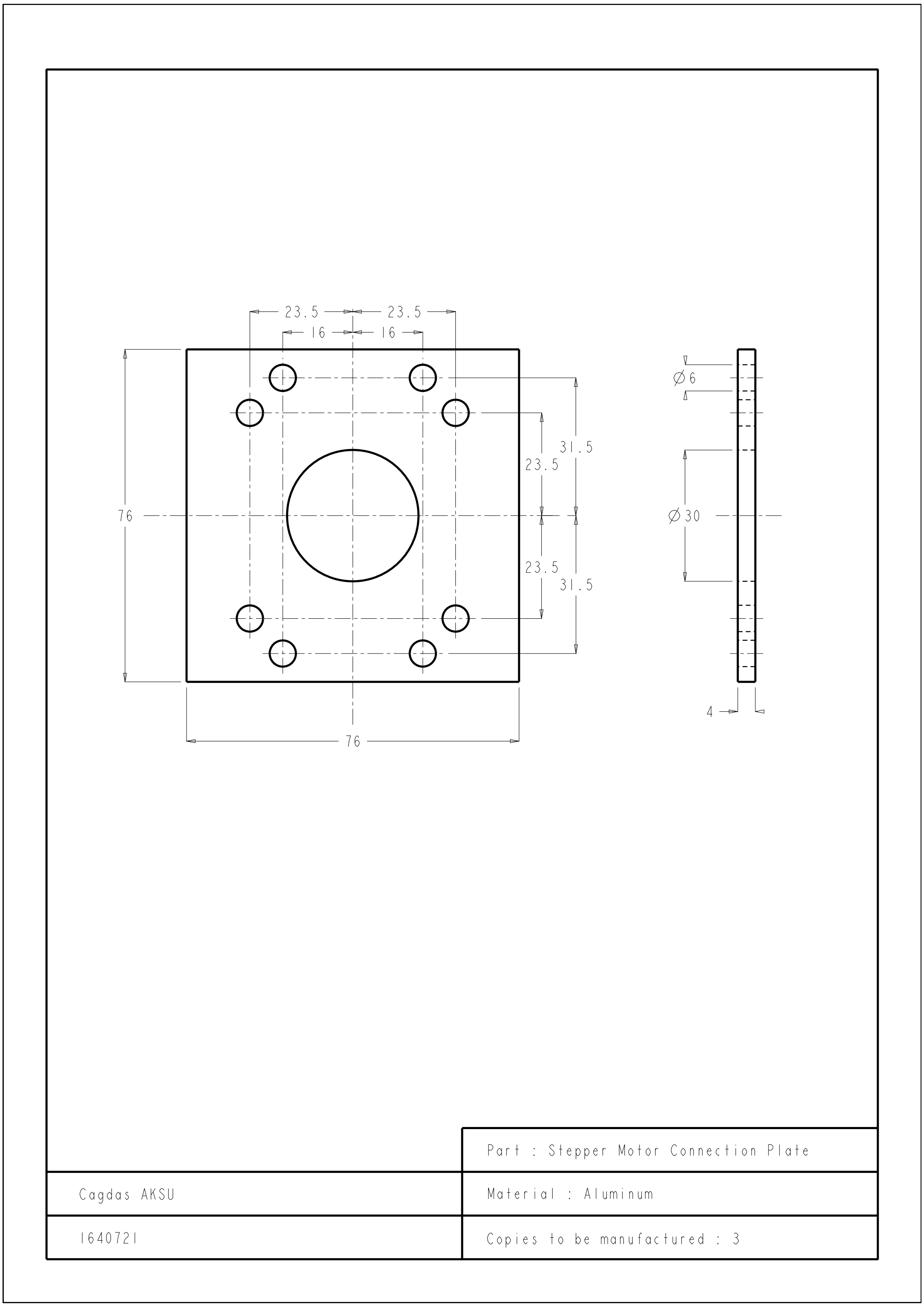


Figure C.2: Technical Drawing of Stepper Motor Connection Plate

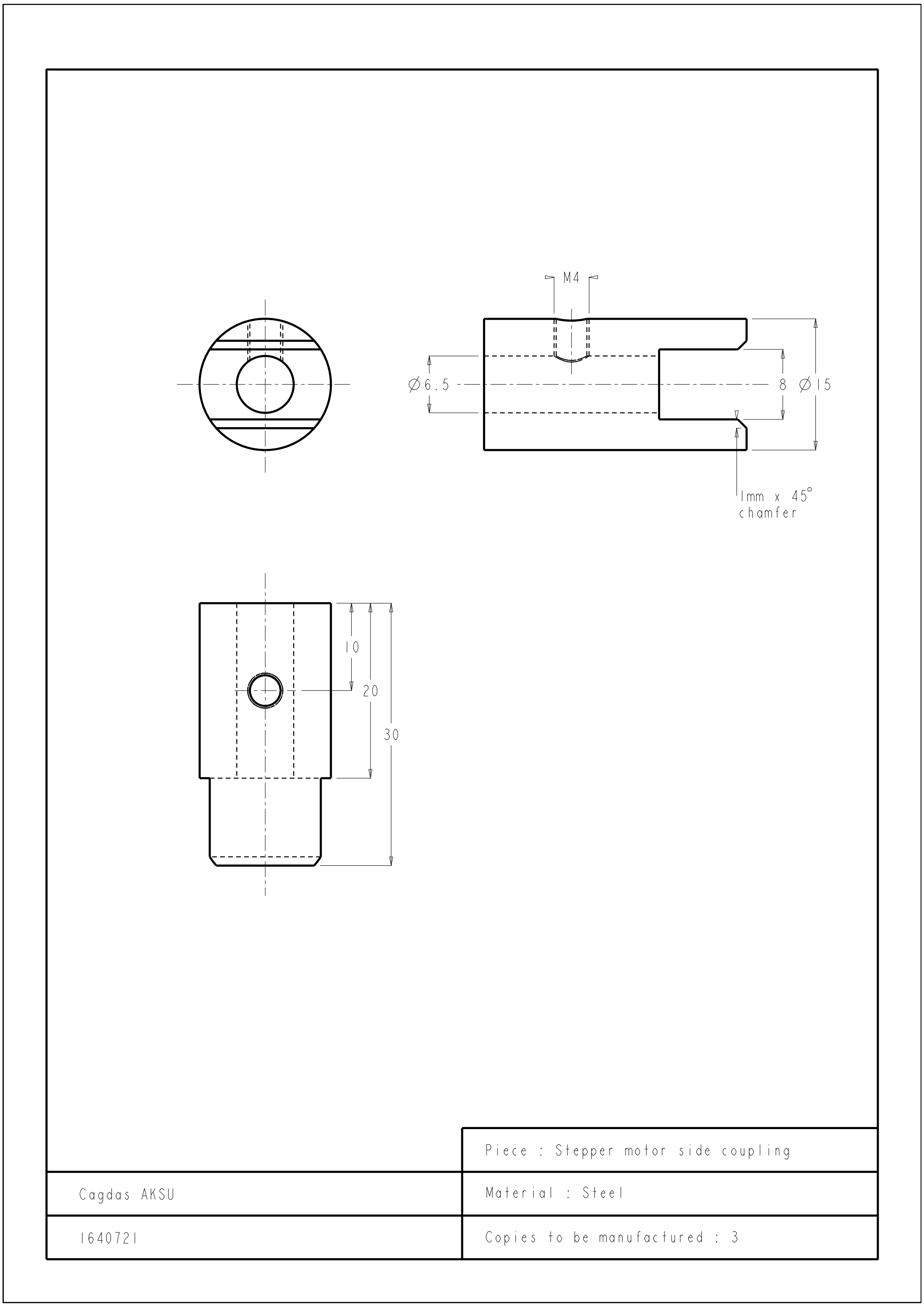


Figure C.3: Technical Drawing of Coupling Piece at Stepper Motor Side

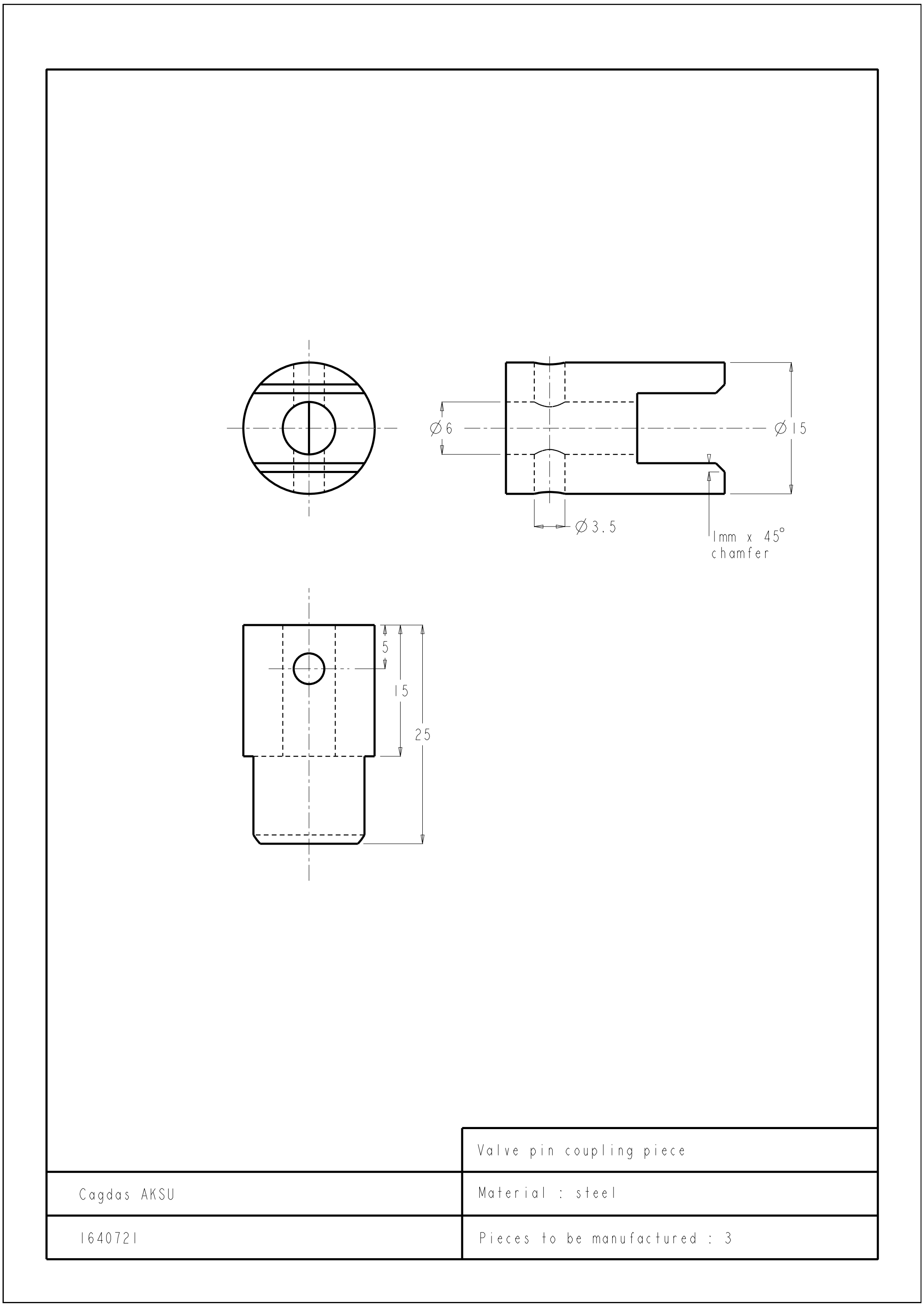


Figure C.4: Technical Drawing of Coupling Piece at Butterfly Valve Side

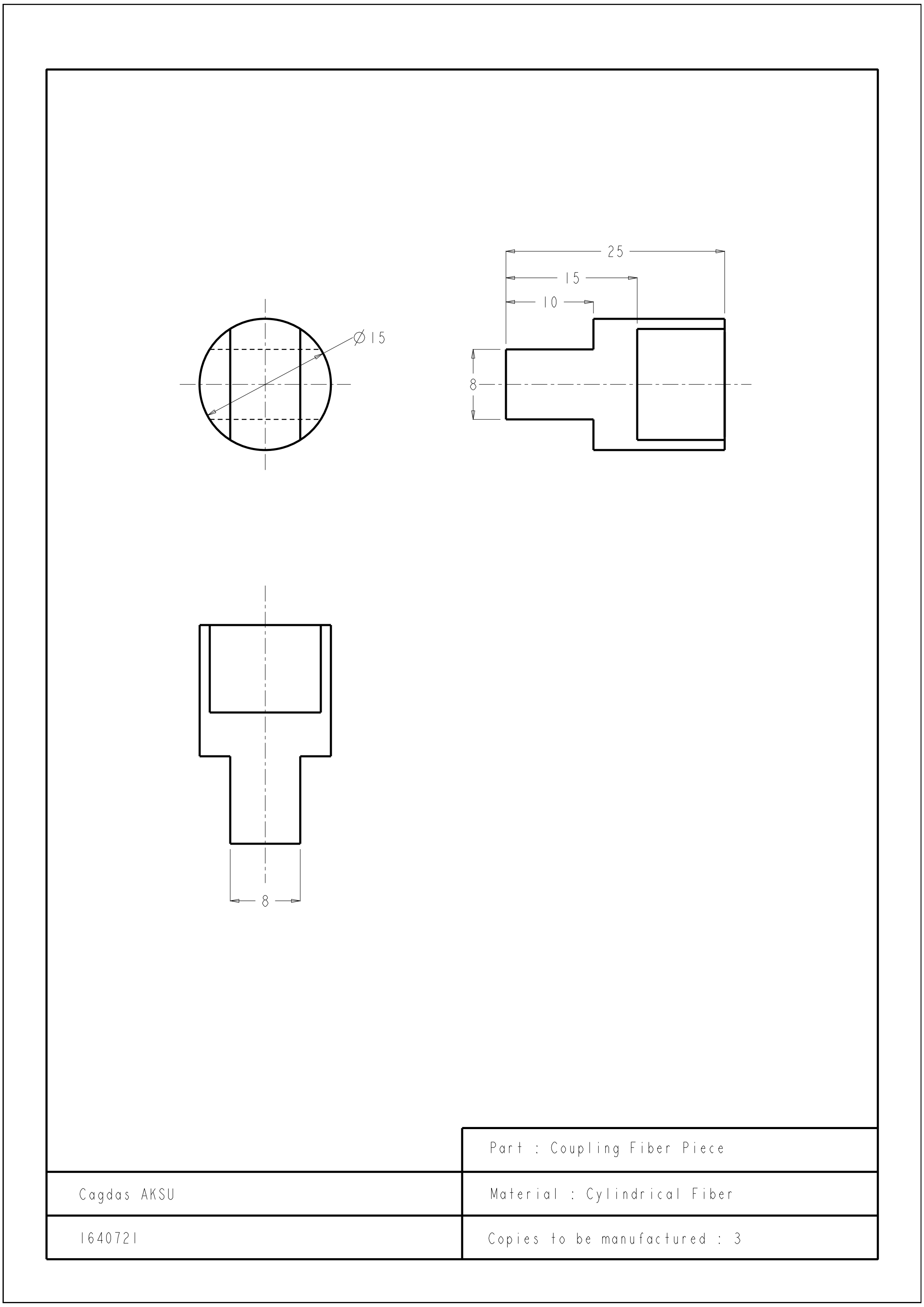


Figure C.5: Technical Drawing of Fiber Coupling Piece

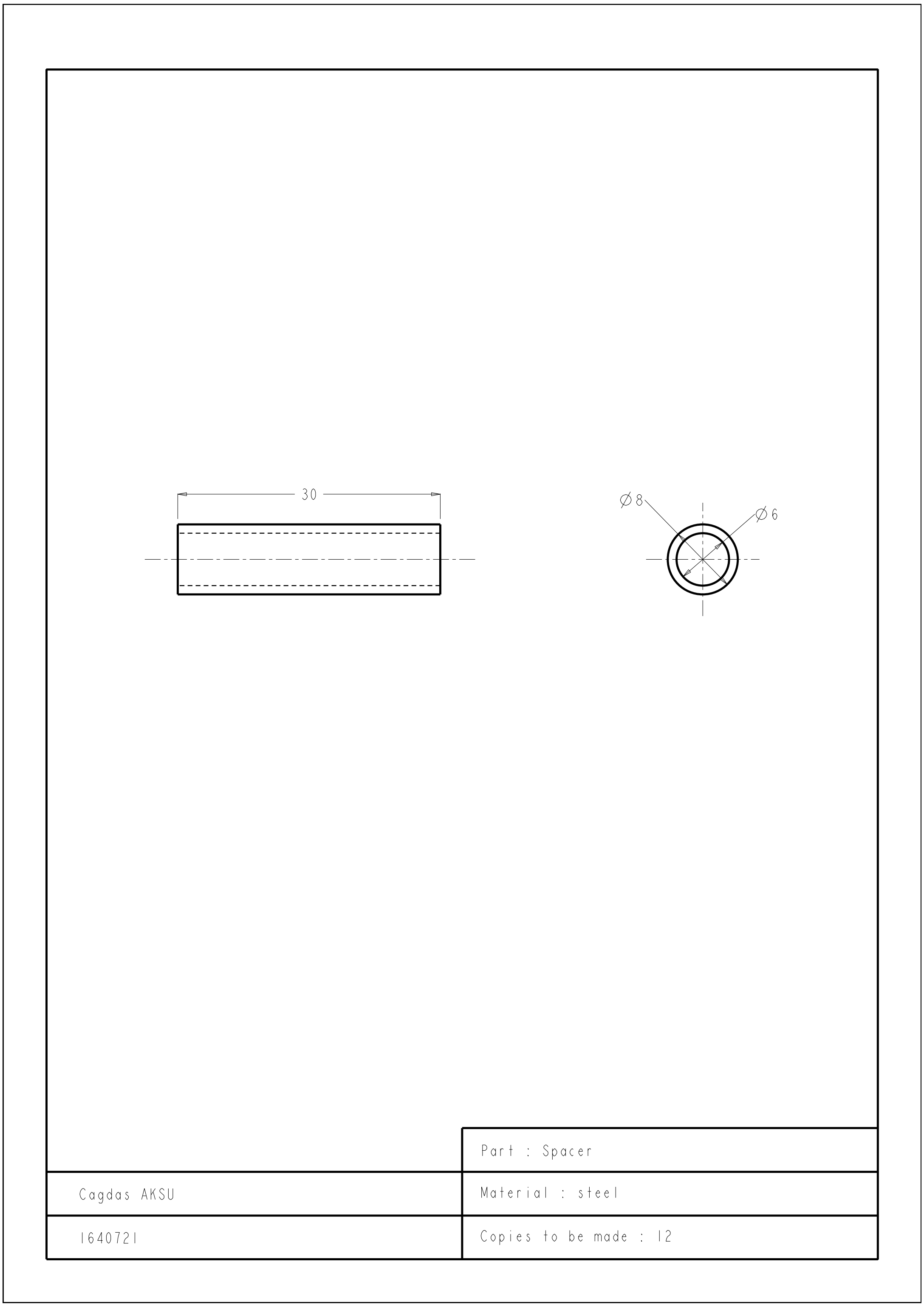


Figure C.6: Technical Drawing of Connection Plate Spacer

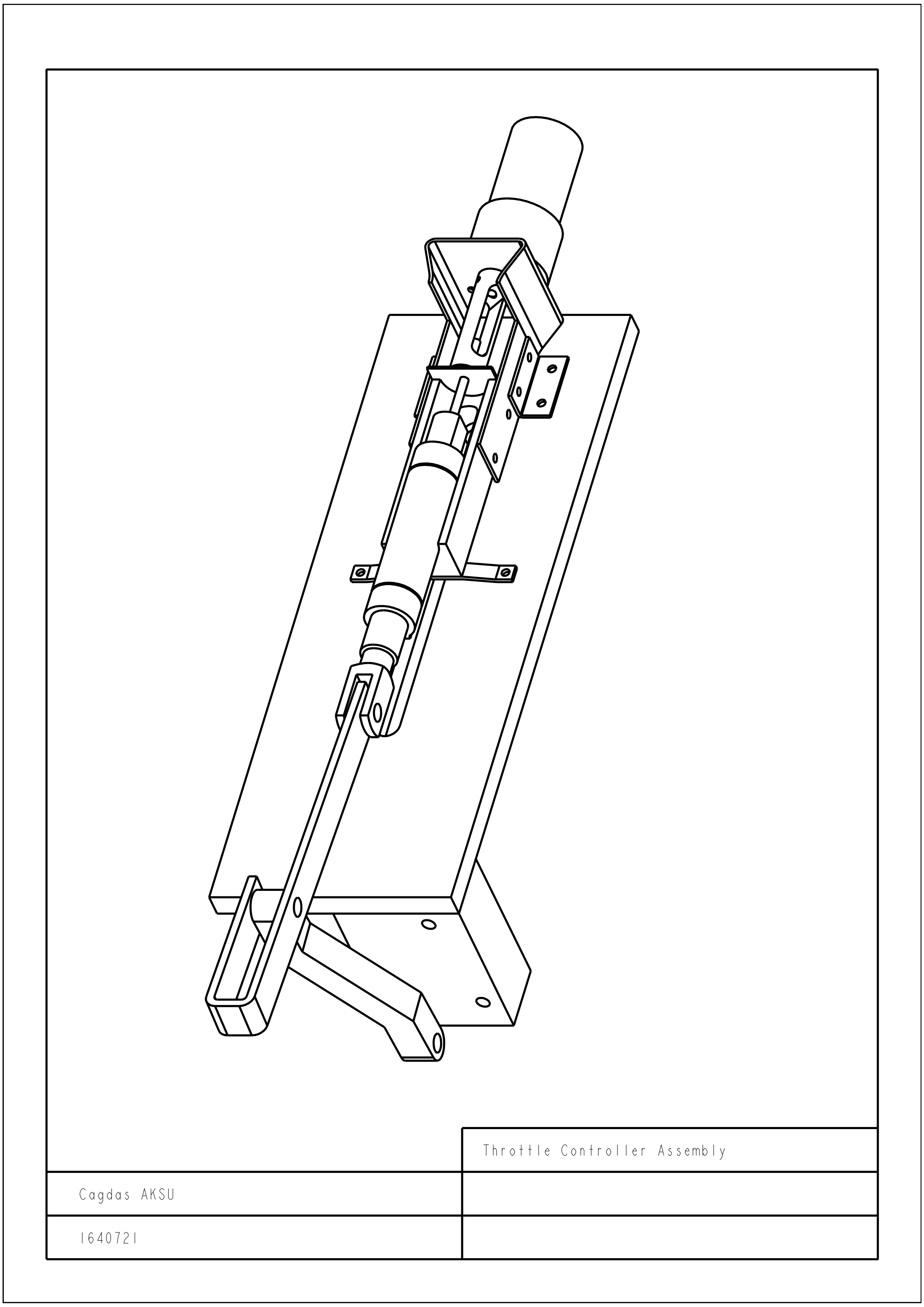


Figure C.7: Isometric view of Throttle Controller Assembly

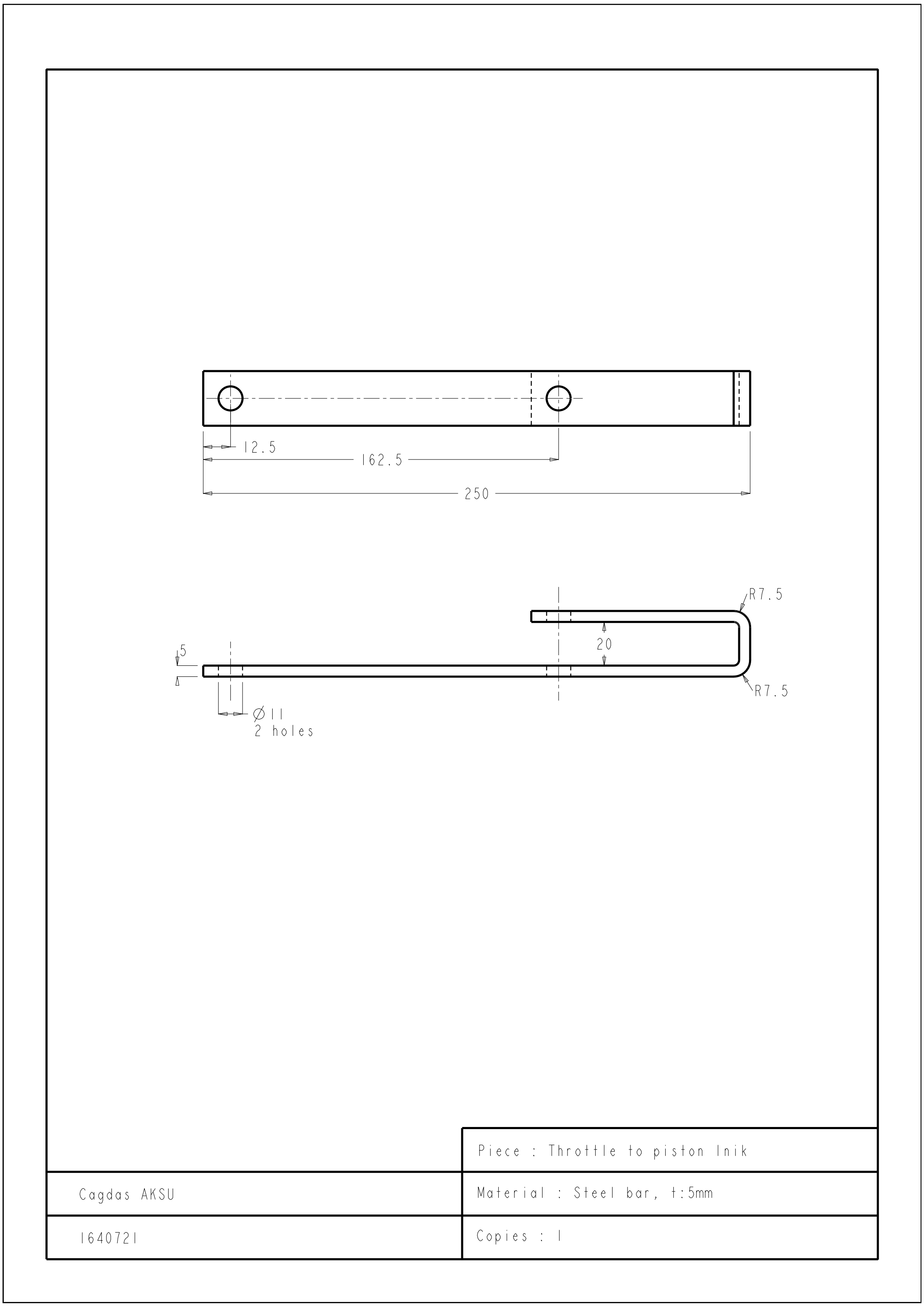


Figure C.8: Technical Drawing of Throttle to Piston Link

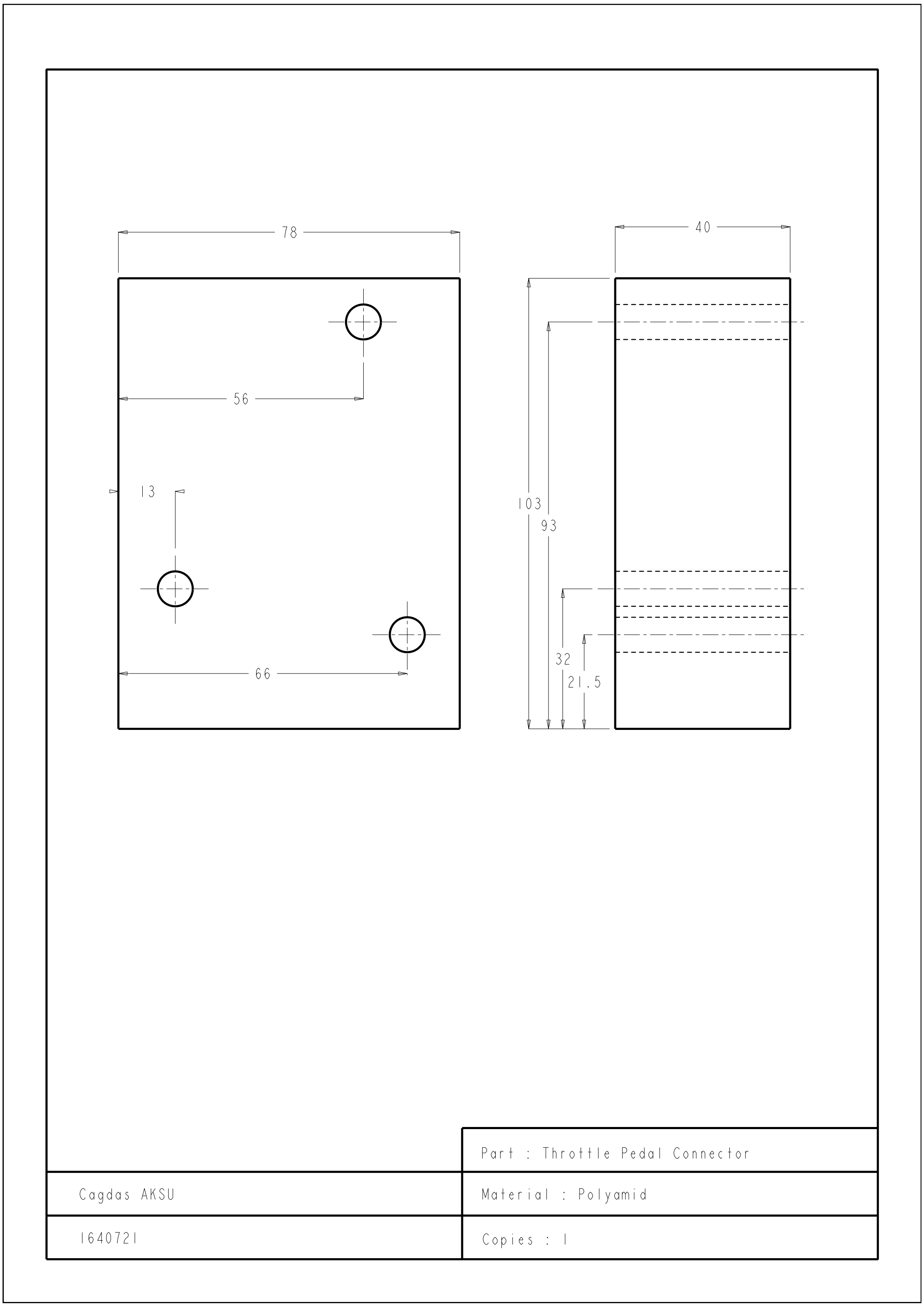


Figure C.9: Technical Drawing of Throttle Pedal Connector

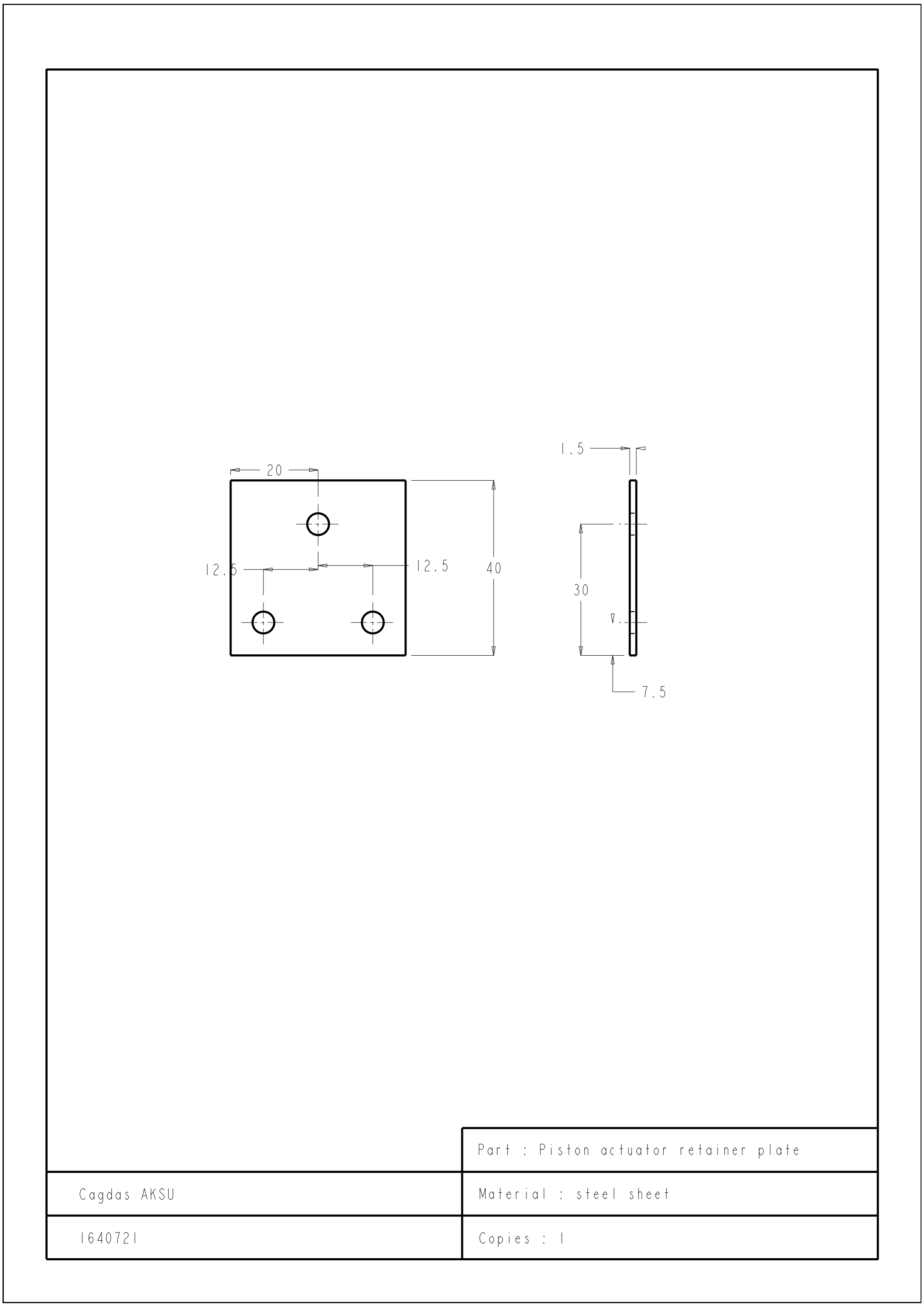


Figure C.10: Technical Drawing of Piston Actuator Retainer Plate

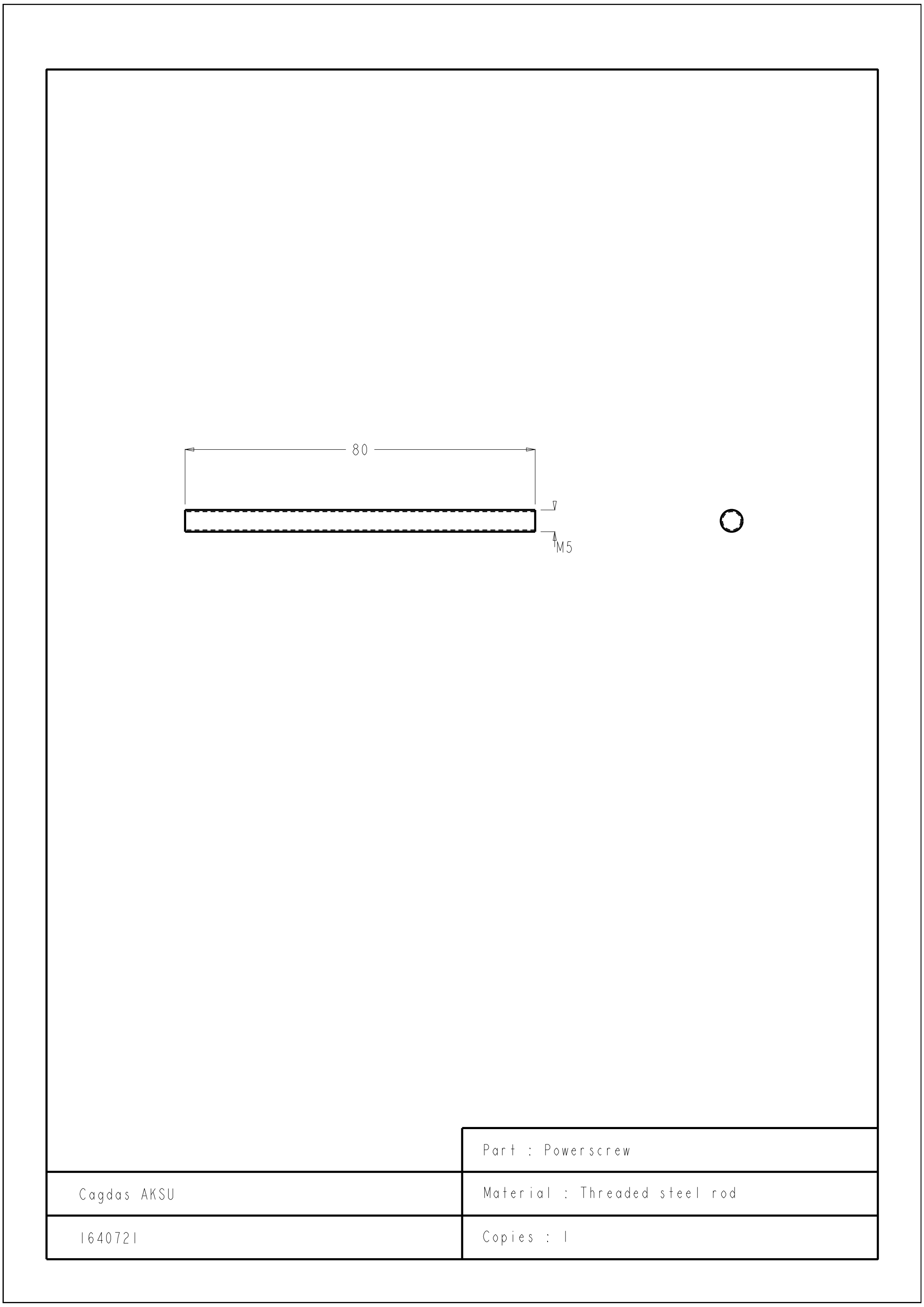


Figure C.11: Technical Drawing of Power Screw

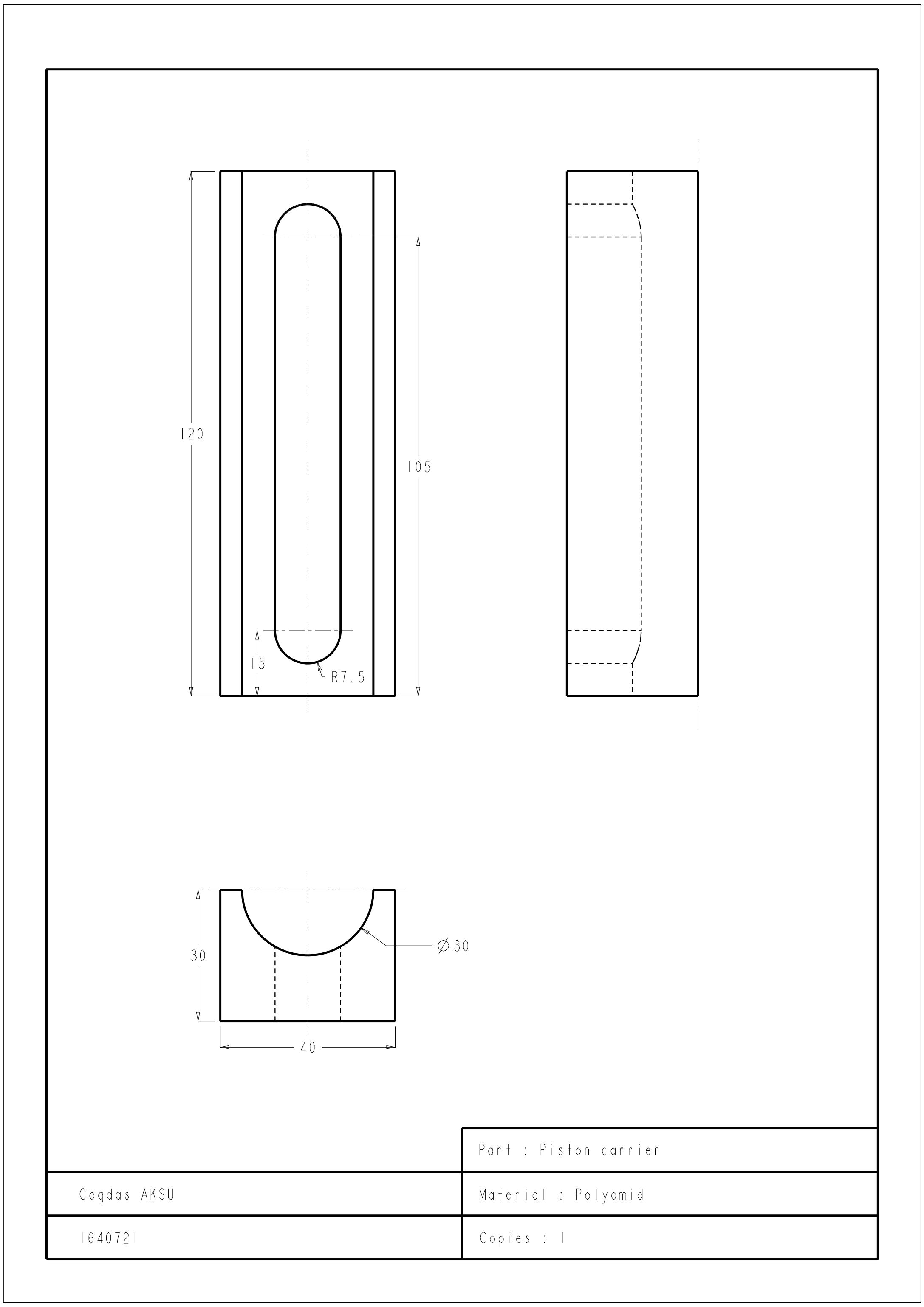


Figure C.12: Technical Drawing of Piston Carrier

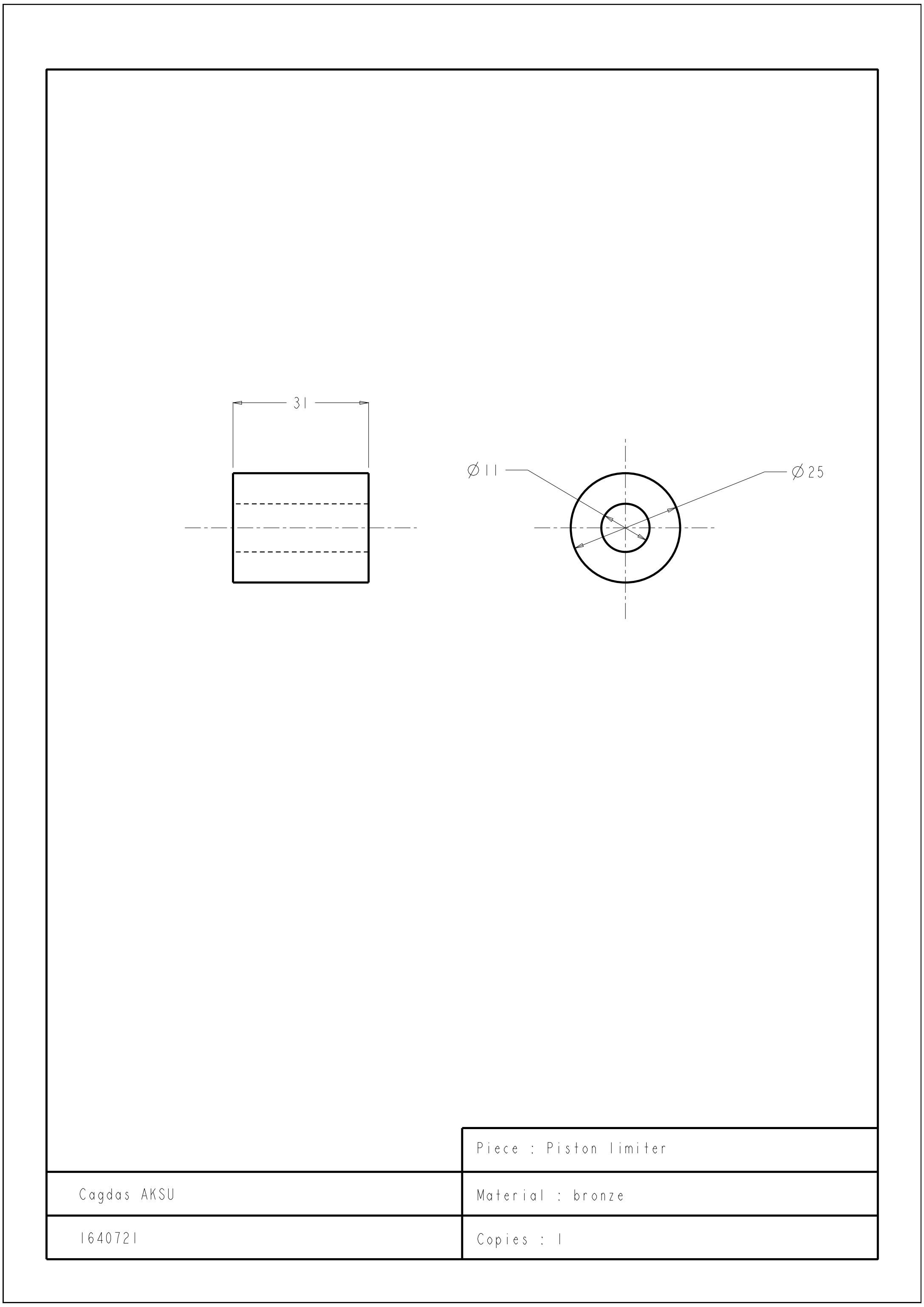


Figure C.13: Technical Drawing of Piston Limiter

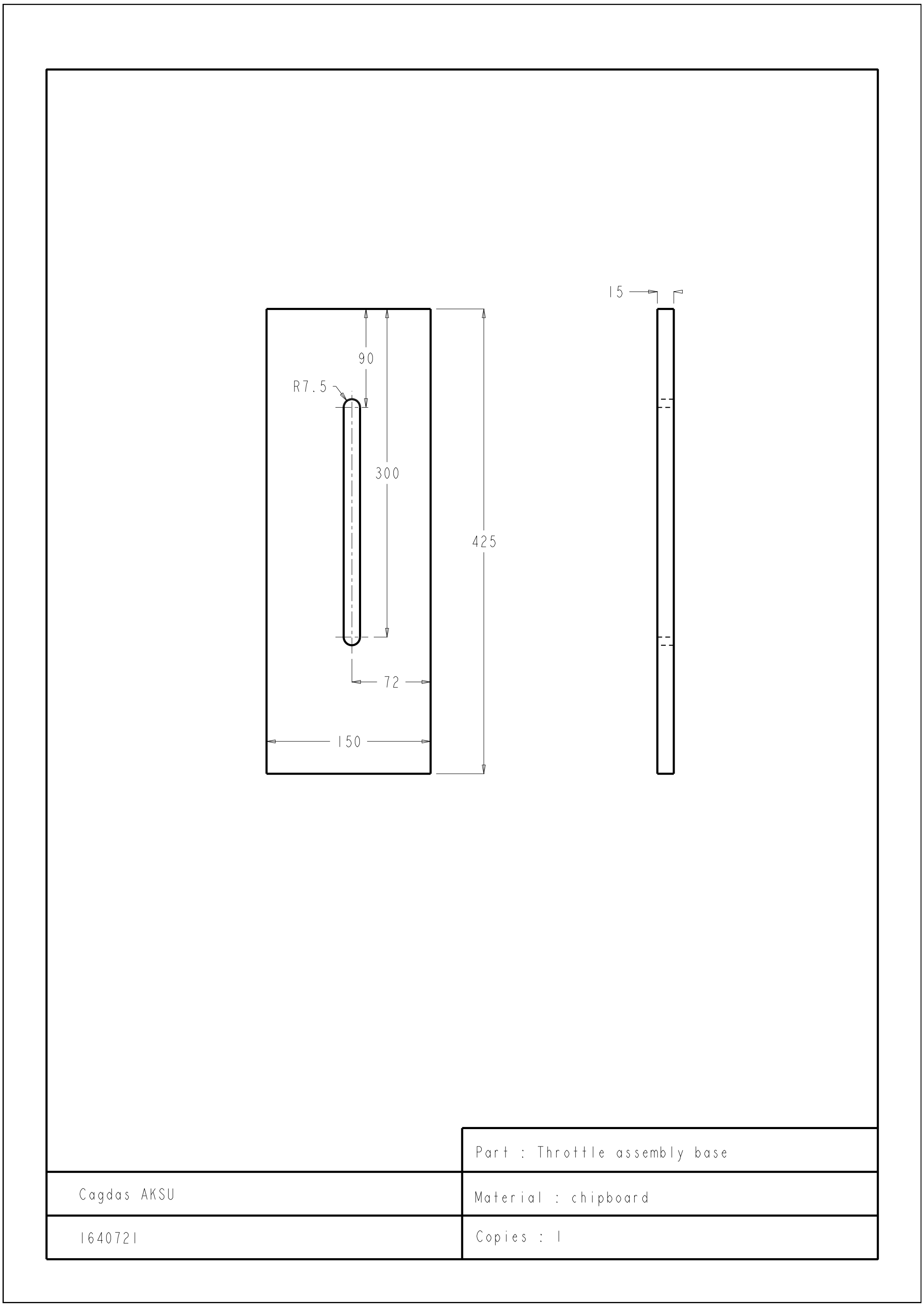


Figure C.14: Technical Drawing of Throttle Assembly Base

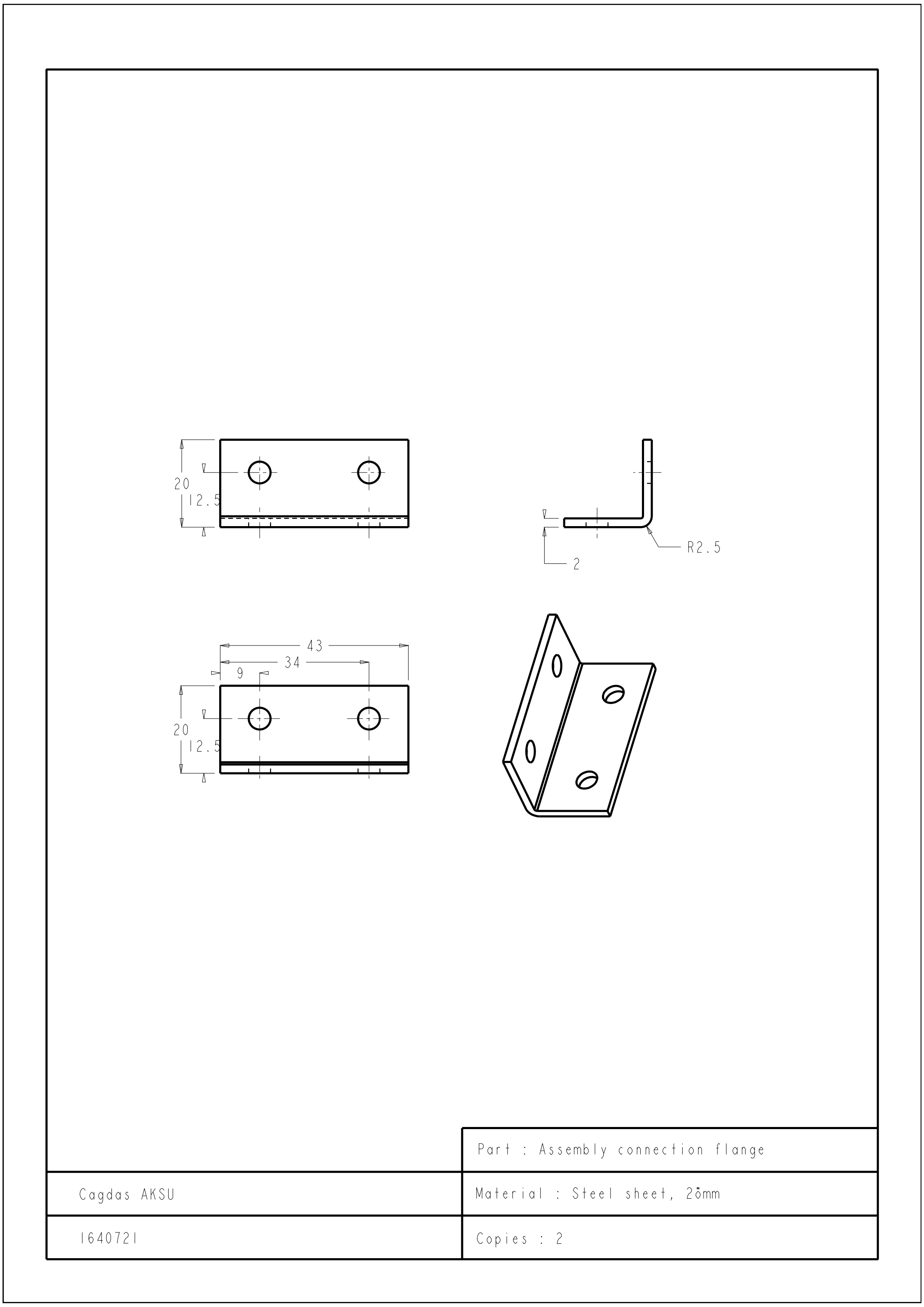


Figure C.15: Technical Drawing of Assembly Connection Flange

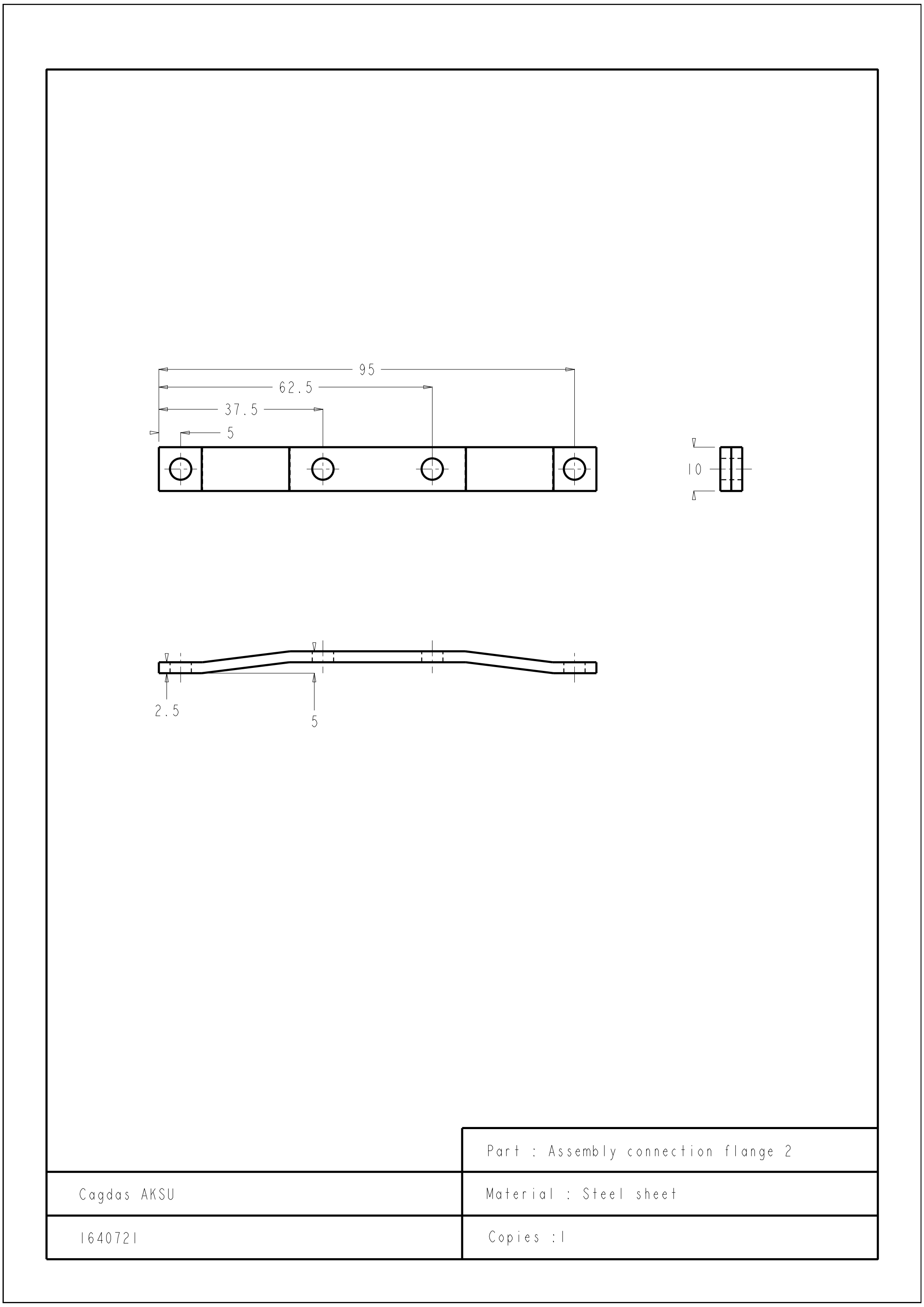


Figure C.16: Technical Drawing of Assembly Connection Flange 2

APPENDIX D

ALGORITHMS OF EXHAUST VALVE CONTROL FUNCTIONS

In this section, the algoritms of exhaust butterfly valve control codes will be given. The actual codes, algorithms of which are given, are repeated in every data recording cycle. Algorithm of each valve will be explained in its own section

D.1 CONTROL ALGORITHM OF SATURATED VAPOR BUTTERFLY VALVE

1. ”i” will be used for checking in which interval heat exchanger pressure lies.
2. VP is a variable used for assigning a temporary value before final decision on butterfly valve position.
3. Read heat exchanger pressure.
4. Round up heat exchanger pressure to the closest integer value and assign to i.
5. check the value of i.
    - If i is smaller than 150, assign 100 to VP.
    - If i is between 151 and 180, assign 90 to VP.
    - If i is between 181 and 210, assign 80 to VP.
    - If i is between 211 and 230, assign 70 to VP.
    - If i is between 231 and 250, assign 40 to VP.
    - If i is between 251 and 270, assign 10 to VP.
    - If i is between 271 and 500, assign 0 to VP.

6. Assign VP to saturated vapor heat exchanger butterfly valve position.
7. If saturated vapor heat exchanger butterfly valve position is changed, send activation signal.

D.2 CONTROL ALGORITHM OF SUPERHEATED VAPOR BUTTERFLY VALVE

Control procedure of superheated vapor heat exchanger butterfly valve is more complex compared to saturated vapor side, the latter has greater priority in utilization of heat of exhaust gases. In general, there are two decisive variables defined, which are AddT and SubP. AddT contains positive values while SubP has negative ones. Difference of AddT frmo SubP results as valve position actuation value.

1. Define variable ”i” as integer. It will be used for checking in which interval heat exchanger temperature lies.
2. Define variable”j” as integer. It will be used for checking in which interval heat exchanger pressure lies.
3. Define variable AddT as integer.is a variable used for assigning a temporary value before final decision on butterfly valve position .
4. Define variable SubP as integer.is a variable used for assigning a temporary value before final decision on butterfly valve position .
5. Round up heat exchanger pressure to the closest integer value and assign to j.
6. Round up heat exchanger temperature to the closest integer value and assign to i.
7. check the value of j.
   - If i is smaller than 150, assign 100 to SubP.
   - If i is between 151 and 200, assign 30 to SubP.
   - If i is between 201 and 500, assign 0 to SubP.
8. check the value of j.
   - If i is smaller than 60, assign 0 to AddT.
   - If i is between 61 and 180, assign 100 to AddT.
   - If i is between 181 and 200, assign 60 to AddT.
   - If i is between 201 and 210, assign 30 to AddT.

- If i is between 211 and 500, assign 0 to AddT.

9. Subtract SubP from AddT, and assign the result as superheated vapor valve position.

10. If superheated valve position is negative, re-assign its value as zero.

11. If superheated vapor valve position is changed, send activation signal.

D.3 CONTROL ALGORITHM OF BYPASS BUTTERFLY VALVE

1. Define variable ”i” as integer. It will be used for checking in which interval heat exchanger temperature lies.
2. Define variable”j” as integer. It will be used for checking in which interval heat exchanger pressure lies.
3. Define variable AddT as integer.is a variable used for assigning a temporary value before final decision on butterfly valve position .
4. Define variable SubP as integer.is a variable used for assigning a temporary value before final decision on butterfly valve position .
5. Round up heat exchanger pressure to the closest integer value and assign to i.
6. Round up heat exchanger temperature to the closest integer value and assign to j.
7. check the value of i.
   - If i is smaller than 170, assign 0 to bpp1.
   - If i is between 171 and 190, assign 10 to bpp1.
   - If i is between 191 and 230, assign 40 to bpp1.
   - If i is between 231 and 500, assign 100 to bpp1.
8. check the value of j.
   - If i is smaller than 50, assign -100 to bpp2.
   - If i is between 51 and 80, assign -50 to bpp2.
   - If i is between 81 and 150, assign -20 to bpp2.
   - If i is between 151 and 180, assign 0 to bpp2.
   - If i is between 181 and 200, assign 30 to bpp2.
   - If i is between 201 and 230, assign 40 to bpp2.
   - If i is between 231 and 500, assign 50 to bpp2.
9. sum up bpp1 and bpp2, and assign the result as bypass valve position.
10. If bypass valve position is negative, re-assign its value as zero.

11. If bypass valve position is greater than 100, re-assign its value as 100.

12. If bypass valve position is changed, send activation signal.

APPENDIX E

CALIBRATION CURVES

In this section, calibration procedures of experimental devices will be explained and calibration curves will be submitted.

E.1 CALIBRATION CURVE OF MASS AIR FLOWMETER

Mass air flowmeter is calibrated by comparing recorded frequency values to pressure difference records of Go-Power Systems M5000 measurement device. This device is supplied with pressure difference vs. air flowrate curves for various nozzle sizes. The following table informs about calibration data points

Table E.1 shows the data obtained for mass air flowmeter calibration.

Table E.1: Calibration Data of Mass Air flowmeter

frequency Hz	barrel record lb/hour	flowrate g/s	corrected g/s
6000	527.938	66.519	62.461
5770	507.296	63.918	60.019
4790	337.628	42.540	39.945
4500	332.830	41.936	39.378
5445	443.814	55.920	52.509
3950	219.649	27.675	25.987
3590	201.137	25.343	23.797
2600	40.217	5.067	4.758

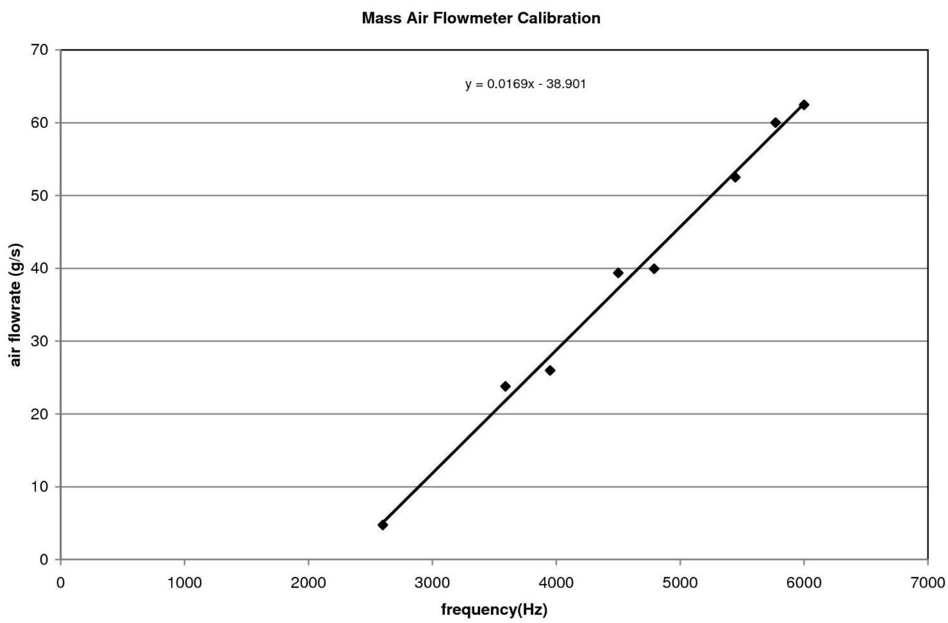


Figure E.1: Calibration Curve of Mass Air Flowmeter

E.2 CALIBRATION CURVE OF TURBINE METER

Turbinemeter is calibrated using ethanol vapor at operating conditions in order to eliminate possible effects of viscosity change. During the calibration, mass flowrate of ethanol, output voltage from frequency-to-voltage converter, turbinemeter exit pressure and temperature are recorded for every calibration data. Pressure, temperature and mass flowrate values are used in ideal gas equation in order to obtain volumetric flowrate.

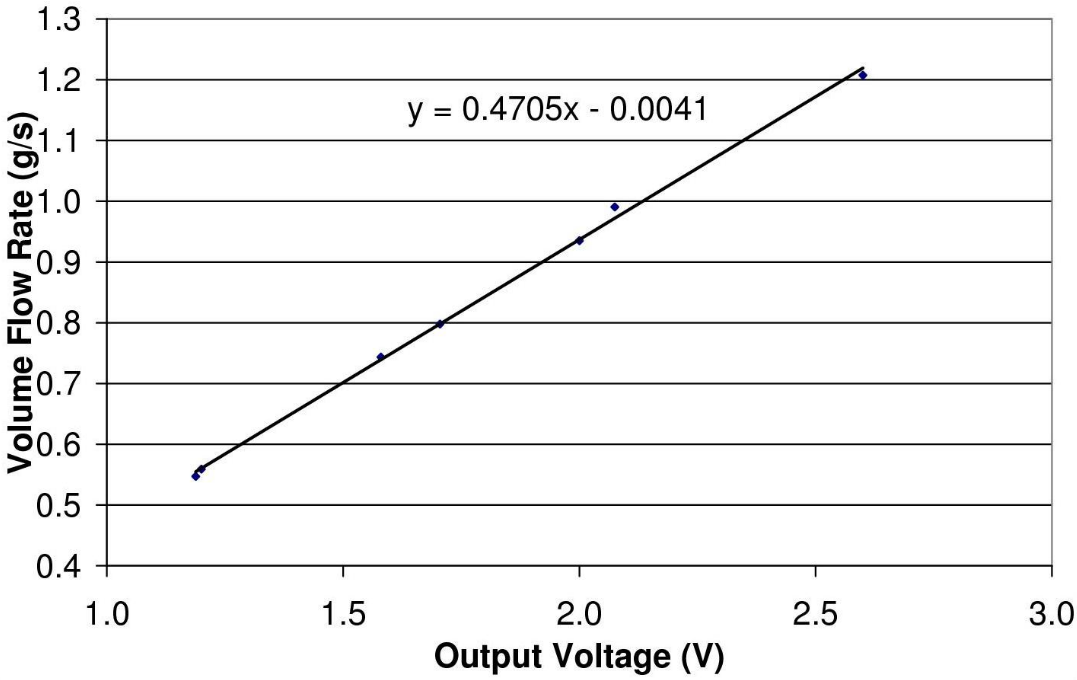


Figure E.2: Calibration Curve of Turbine Meter

Mass flowrate is obtained by directing superheated ethanol stream through condenser heat exchanger during 20 seconds (200 test cycles). Condensed ethanol is collected in flasks, dry masses of which are measured beforehand. The difference between two weight measurements give net amount of condensed ethanol. Figure E.2 displays calibration curve of the device.

E.3 CALIBRATION CURVE OF RESISTANCE THERMOMETER

Resistance themometer is calibrated by comparing amplified output voltage of the component to the values read by a thermometer. Water is used during calibration. While higher temperatures are provided using hot water, saturated water-ice mixture is used for at lowest and room temperature at the second lowest point. The resulting curve appears as E.3.

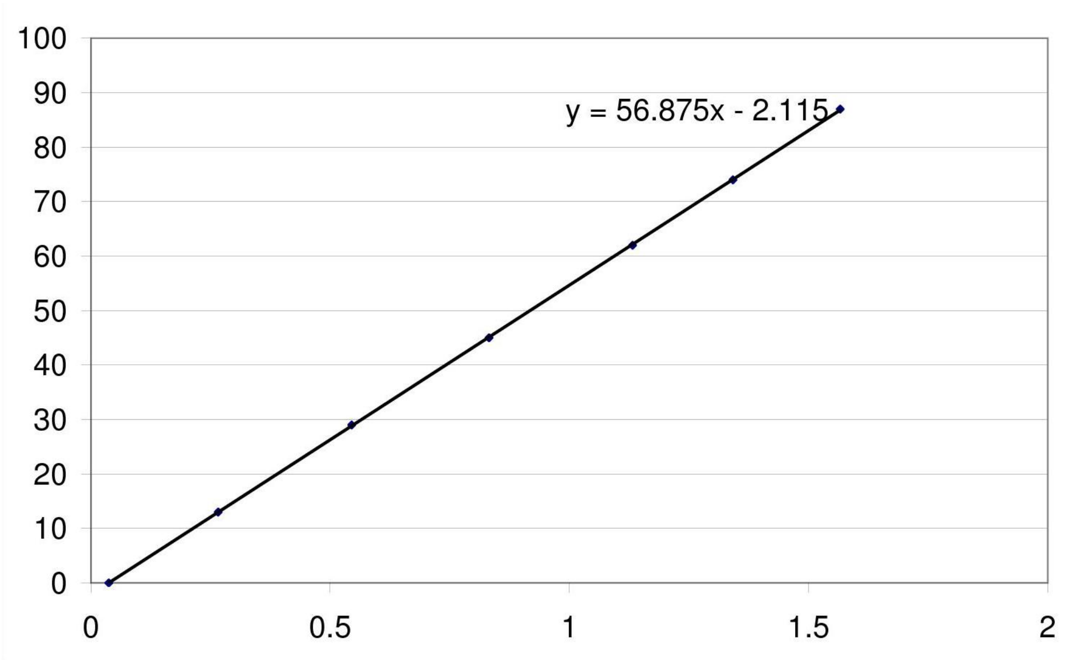


Figure E.3: Calibration Curve of Resistance Thermometer

E.4 CALIBRATION CURVE OF PRESSURE TRANSDUCER

Amplified output voltage of pressure transducer is calibrated using a dead weight tester. Sead weight tester applies 100kPa of pressure for every 1 kg of load. Recorded values are available in table E.2 and the resulting calibration curve can be viewed in figure E.4.

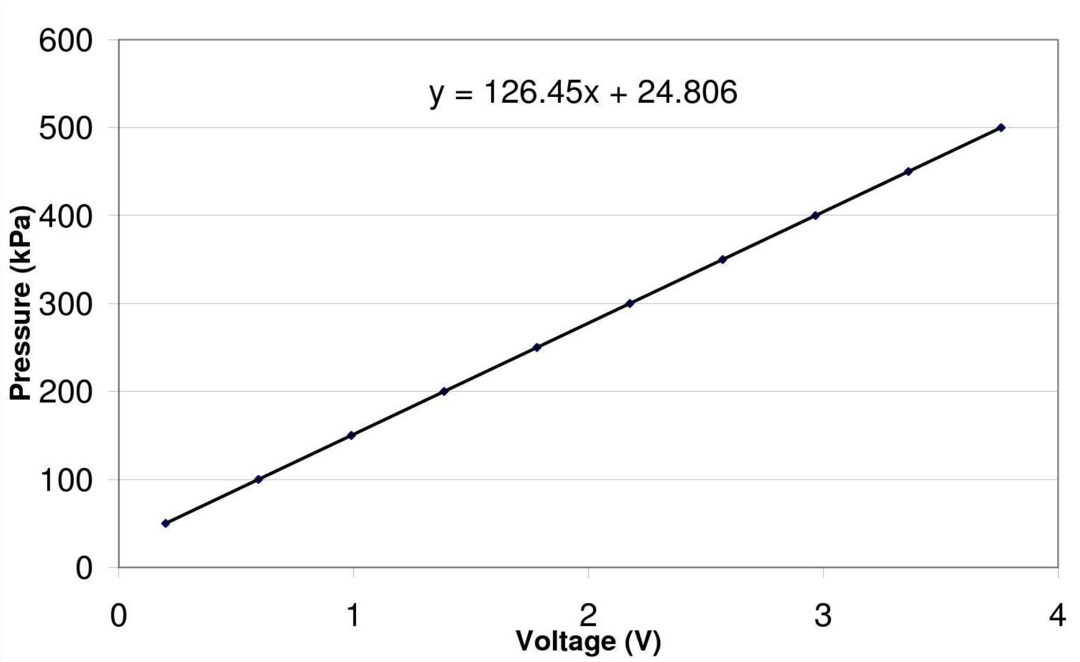


Figure E.4: Calibration Curve of Pressure Transducer

Table E.2: Calibration Data of Pressure Transducer

Voltage V	Mass kg	Pressure kPa
3.758	5.0	500
3.363	4.5	450
2.967	4.0	400
2.572	3.5	350
2.176	3.0	300
1.781	2.5	250
1.385	2.0	200
0.990	1.5	150
0.595	1.0	100
0.199	0.5	50

APPENDIX F

ERROR ANALYSIS

In this chapter of appendices, error analysis of the formulae and data conversion that influence performance evaluation results will be performed using act experimental data.

This analysis will be performed using 3500 rpm 75% load duel fuel test data. Assumptions of the calculation are as following: Error in ambient pressure ($\delta_{p,read}$): 5 mbar
Error in relative humidity (δ_{RH}):: 0.5%
Error in measured voltages (δ_{analog}): 0.005 V
Error in measured frequency (δ_{freq}): 50 kHz
Error in software loop time (δ_{loop}): 1 ms
Error in fuel measurement time (δ_{diesel}): 1 cycle

Data set is as follows: R.H. : 50%

Atmospheric Pressure : 1022 mBar

V_0 : 2.975V

V_1 :3.492V

V_2 : 3.022

V_3 : 2.941V

V_4 : 6.282V

V_5 : 0.005V

V_6 : 0.005V

V_7 : 0.005V

V_8 : 3.019V

V_9 : 1.139V

V_{10} : 3.895V

V_{11} : 3.477V

V_{12} : 1.833V

V_{13} : 0.09V

V_{14} : 2.16V

V_{15} : 0.005V

t_{diesel} : 69.40 s = 649 cycles

Formulae for calculating error is :

$$f = f(x_1, x_2, x_3, ..., x_n) \tag{F.1}$$

$$\sigma_f^2 = \sum \left(\frac{\partial f}{\partial x_i}\right)^2 \sigma_{x_i}^2 \tag{F.2}$$

For linear equations, this is simplified to:

$$\sigma_f = \sum \left(\frac{\partial f}{\partial x_i}\right) \sigma_{x_i} \tag{F.3}$$

Error in ambient temperature:

$$\begin{aligned}
\delta_{T,amb} &= \frac{\partial}{\partial V_{Tamb}}\left((V_{Tamb} - 2.73) \times 100\right) \times \delta_{V,Tamb} \\
&= 100 \times 0.005 \\
&= 0.5^{\circ}C \; at \; 24.5
\end{aligned}$$

Error in atmospheric pressure :

$$
\begin{aligned}
\delta_{P,atm} &= \frac{\partial}{\partial P_{mbar}}\left(\frac{P_{mbar}-101}{10}\right)\delta_{P,mbar} \\
&= \frac{5}{10} \\
&= 0.5\ kPa\ at\ 92.1\ kPa\ ambient\ pressure
\end{aligned}
$$

Error in eth pressure :

$$
\begin{aligned}
\delta P,eth &= \frac{\partial}{\partial T_{amb}}\left(6.10708\times 10^{\frac{7.5\times T_{amb}}{237.3+T_{amb}}}\right)\delta_{T,amb} \\
&= \frac{7.5T_{amb}}{237.3+T_{amb}}\left(6.10708\times 10^{\frac{6.5T_{amb}-237.3}{237.3+T_{amb}}}\right)\left(\frac{1779.75}{(237.3+T_{amb})^2}\right)\delta_{T,amb} \\
&= \frac{7.5\times 24.5}{237.3+24.5}\left(6.10708\times 10^{\frac{6.5\times 24.5-237.3}{237.3+24.5}}\right)\left(\frac{1779.75}{(237.3+24.5)^2}\right)0.5 \\
&= 0.04\ mbar = 0.004\ kPa
\end{aligned}
$$

Error in pressure of dry air:

$$
\begin{aligned}
\delta P,dry &= \sqrt{\left(\frac{\partial}{\partial P_{atm}}\delta^2_{P,atm}+\frac{\partial}{\partial RH}\delta^2_{RH}+\frac{\partial}{\partial P_{eth}}\delta^2_{P,eth}\right)(P_{atm}-RH\times P_{eth})^2} \\
&= \sqrt{\left(\delta^2_{P,atm}+P^2_{eth}\delta^2_{RH}+RH^2\times\delta^2_{P,eth}\right)} \\
&= 0.524\ kPa\ =\ 0.579\%
\end{aligned}
$$

Error in engine speed:

$$
\begin{aligned}
\delta V,\theta &= \frac{\partial}{\partial_{V,theta}}(V_\theta\times C_\theta)\,\delta\theta \\
&= C_\theta\times\delta\theta \\
&= 0.005\times 1000 \\
&= 5\ rpm\ =\ 0.143\%\ at\ 3492\ rpm
\end{aligned}
$$

Error in Measured torque:

$$
\begin{aligned}
\delta T,meas &= \frac{\partial}{\partial_{V,theta}}\left(C_{torque}\times V_{torque}\right)\delta V,torque \\
&= (C_{torque}\times\delta V,torque \\
&= 40\times 0.005 \\
&= 0.2\ Nm\ =\ 0.165\%\ at\ 120.88\ Nm
\end{aligned}
$$

Error in Manifold absolute pressure:

$$\begin{aligned}
\delta P, MAP &= \frac{\partial}{\partial_{V,theta}}((65.8823 \times VMAP) - 6.3529)\,\delta V, MAP \\
&= 65.8823 \times \delta V, MAP \\
&= 65.8823 \times 0.005 \\
&= 0.329\ kPa\ =\ 0.175\%\ at\ 187.41\ kPa
\end{aligned}$$

Error in Turbo pressure coefficient:

$$\begin{aligned}
\delta r &= \sqrt{\left(\frac{\partial}{\partial P_{atm}}\left(\frac{P_{MAP}}{P_{atm}}\right)\right)^2 \delta^2_{P,atm} + \left(\frac{\partial}{\partial P_{MAP}}\left(\frac{P_{MAP}}{P_{atm}}\right)\right)^2 \delta^2_{P,MAP}} \\
&= \sqrt{\left(-\frac{P_{MAP}}{P^2_{atm}}\right)^2 \delta^2_{P,atm} + \left(\frac{1}{P^2_{atm}}\right)^2 \delta^2_{P,MAP}} \\
&= \sqrt{\left(\frac{187.41}{90.56^2}\right)^2 0.5 + \left(\frac{1}{90.56^2}\right)^2 0.329} \\
&= 0.0077\ =\ 0.374\%
\end{aligned}$$

Error in Ethanol Consumption Temperature:

$$\begin{aligned}
V_{cons} &= \frac{\partial}{\partial V_{cons}}\left(\frac{V_{cons}}{G_{cons}}\right)\delta_{V,cons} \\
&= \frac{\delta_{V,cons}}{500} \\
&= 1 \times 10^{-5}\ mV
\end{aligned}$$

Thermocouples generate 41 μV/°C, and the error is 0.244 °C at this temperature. This corresponds to 2.31% relative error.

Error in Ethanol flowrate :

$$\begin{aligned}
\delta\rho, v &= \sqrt{\left(\left(\delta_{P,MAP}\frac{\partial}{\partial P_{MAP}}\right)^2 + \left(\delta_{T,cons}\frac{\partial}{\partial T_{cons}}\right)^2\right)\left(\frac{P_{MAP} \times MW_{eth}}{(T_{cons} + 273) \times R_u}\right)^2} \\
&= \sqrt{\left(\frac{M_{eth} \times \delta_{P,MAP}}{(T_{cons} + 273) \times R_u}\right)^2 + \left(\frac{P_{MAP} \times M_{eth} \times \delta_{T,cons}}{((T_{cons} + 273) \times R_u)^2}\right)^2} \\
&= \sqrt{\left(\frac{46 \times 0.329}{(105.41 + 273) \times 8.314}\right)^2 + \left(\frac{187.41 \times 46 \times 0.244}{((105.41 + 273) \times 8.314)^2}\right)^2} \\
&= 0.0048\ g/cc\ =\ 0.17\%\ at\ 2.733\ g/cc
\end{aligned}$$

$$
\begin{aligned}
\delta \dot{m}, v &= \sqrt{\left(\frac{\partial}{\partial V_t}\left(0.4705V_t + 0.0041\right)\rho_v\right)^2 \delta^2_{V,t} + \left(\frac{\partial}{\partial \rho_v}\left(0.4705V_t + 0.0041\right)\rho_v\right)^2 \delta^2_{\rho,v}} \\
&= \sqrt{\left(0.4705\rho_v\delta_{V,t}\right)^2 + \left(\left(0.4705V_t + 0.0041\right)\delta_{\rho,v}\right)^2} \\
&= \sqrt{(0.4705 \times 2.733 \times 0.005)^2 + ((0.4705 \times 1.139 + 0.0041)\,0.0048)^2} \\
&= 0.0069\ g/s\ =\ 0.46\%\ at\ 1.477\ g/s
\end{aligned}
$$

Error in Air flowrate :

$$
\begin{aligned}
\delta_{\dot{m},act} &= \frac{\partial}{\partial V_t}\left(0.0167 \times F_{MAF} - 38.901\right)\delta_{F,MAP} \\
&= 0.0167\delta_{F,MAP} \\
&= 0.0167 \times 50 g/s \\
&= 0.835\ g/s\ =\ 1.69\%\ at\ 49.43\ g/s
\end{aligned}
$$

Error in Diesel Flowrate

$$
\begin{aligned}
time &= cycle \times cycle\ time \\
\delta_t &= \sqrt{\left(\frac{\partial}{\partial cycle}\left(cycle \times cycle\ time\right)\right)^2 \delta^2_{cycle} + \left(\frac{\partial}{\partial cycle\ time}\left(cycle \times cycle\ time\right)\right)^2 \delta^2_{cycle\ time}} \\
&= \sqrt{cycle\ time^2 \delta^2_{cycle} + cycle^2 \delta^2_{cycle\ time}} \\
&= \sqrt{0.1^2 \times 1^2 + 694^2 \times 0.001^2} \\
&= 0.701\ seconds \\
\dot{m}_{diesel} &= \frac{167.41g}{time} \\
\delta_{\dot{m},diesel} &= \frac{\partial}{\partial time}\left(\frac{167.41}{time}\right)\delta_{time} \\
&= \frac{167.41}{time^2}\delta_{time} \\
&= \frac{167.41}{67.4} 0.701 \\
&= 0.02\ g/s\ =\ 1.04\%
\end{aligned}
$$

Error in Total Fuel Flowrate:

$$
\begin{aligned}
\dot{m}_{total} &= \dot{m}_{diesel} + \dot{m}_{eth} \\
\delta_{\dot{m},total} &= \sqrt{\left(\left(\frac{\partial}{\partial \dot{m}_{diesel}}\right)\left(\dot{m}_{diesel} + \dot{m}_{eth}\right)\right)^2 \delta^2_{diesel} + \left(\left(\frac{\partial}{\partial \dot{m}_{eth}}\right)\left(\dot{m}_{diesel} + \dot{m}_{eth}\right)\right)^2 \delta^2_{eth}} \\
&= \sqrt{\dot{m}^2_{eth}\delta^2_{diesel} + \dot{m}^2_{diesel}\delta^2_{eth}} \\
&= 0.03\ g/s\ =\ 0.87\%
\end{aligned}
$$

Error in Power Correction Coefficient:

$$\begin{aligned}
\delta_{f_a} &= \sqrt{\left(\left(\frac{\partial}{\partial P_d}\right)\left(\left(\frac{P_{d,ref}}{P_d}\right)^{0.7}\left(\frac{T_{amb}+273}{T_{ref}}\right)^{0.7}\right)\right)^2 \delta_{P_d} + \left(\left(\frac{\partial}{\partial T_{amb}}\right)\left(\left(\frac{P_{d,ref}}{P_d}\right)^{0.7}\right)\right)^2 \delta_{T_{amb}}} \\
&= \sqrt{\left(-0.7\delta_{P_d}\left(\frac{T_{amb}+273}{T_{ref}}\right)^{0.7}\left(\frac{P_{d,ref}^{0.7}}{P_d^{1.7}}\right)\right)^2 + \left(\frac{0.7\delta_{T_{ref}}}{T_{ref}^{0.7}\,(T_{amb}+273)^{0.3}}\left(\frac{P_{d,ref}}{P_d}\right)^{0.7}\right)^2} \\
&= 0.0044
\end{aligned}$$

$$\begin{aligned}
\delta_q &= \sqrt{\frac{\partial}{\partial \dot{m}_{fuel}}\left(\frac{Z\times \dot{m}_{fuel}}{V\times\theta}\right)^2 \delta^2_{\dot{m},fuel} + \frac{\partial}{\partial\theta}\left(\frac{Z\times \dot{m}_{fuel}}{V\times\theta}\right)^2 \delta^2_{\theta}} \\
&= \sqrt{\left(\frac{Z\delta_{\dot{m},fuel}}{V\theta}\right)^2 + \left(-\frac{Z\dot{m}_{fuel}\delta_\theta}{V\theta^2}\right)^2} \\
&= 0.840
\end{aligned}$$

$$\begin{aligned}
\delta_{q_c} &= \sqrt{\left(\frac{\partial}{\partial q}\left(\frac{q}{r}\right)\right)^2 \delta_q^2 + \left(\frac{\partial}{\partial r}\left(\frac{q}{r}\right)\right)^2 \delta_r^2} \\
&= \sqrt{\frac{\delta_q^2}{r^2} + \frac{\delta_r^2 q^2}{r^4}} \\
&= 0.449
\end{aligned}$$

$$\begin{aligned}
f_m &= \frac{\partial}{\partial q_c}\,(0.036\times q_c - 1.14)\,\delta_{q_c} \\
&= 0.036\delta_{q_c} \\
&= 0.0162
\end{aligned}$$

$$\begin{aligned}
\delta_{\alpha_c} &= \sqrt{\left(\frac{\partial}{\partial f_a}\left((f_a)^{f_m}\right)\right)^2 \delta_{f_a}^2 + \left(\frac{\partial}{\partial f_m}\left((f_a)^{f_m}\right)\right)^2 \delta_{f_m}^2} \\
&= \sqrt{\left(f_m f_a^{f_m-1}\right)^2 \delta_{f_a}^2 + \left(f_a^{f_m}\; ln f_a\right)^2 \delta_{f_m}^2} \\
&= 0.00322 \;=\; 0.309\% \; for\; 1.044
\end{aligned}$$

Error in Corrected Torque :

$$\begin{aligned}
\delta_{T,corr} &= \sqrt{\left(\frac{\partial}{\partial \alpha_c}\,(\alpha_c T_{meas})\right)^2 \delta_{\alpha_c}^2 + \left(\frac{\partial}{\partial \alpha_c}\,(\alpha_c T_{meas})\right)^2 \delta_{T,meas}^2} \\
&= \sqrt{T_{meas}^2\delta_{\alpha_c}^2 + \alpha_c^2\delta_{T,meas}^2} \\
&= 0.44\; Nm \;=\; 0.35\% \; for\; 126.20\; Nm
\end{aligned}$$

Error in Engine Power :

$$
\begin{aligned}
\delta_{P,corr} &= \sqrt{\left(\frac{\partial}{\partial T_{corr}}\left(T_{corr}\theta 1.36\frac{\pi}{30000}\right)\right)^2 \delta^2_{T,corr} + \left(\frac{\partial}{\partial\theta}\left(T_{corr}\theta 1.36 s\frac{\pi}{30000}\right)\right)^2 \delta^2_{\theta}} \\
&= \sqrt{\left(\theta\delta_{T,corr}1.36\frac{\pi}{30000}\right)^2 + \left(T_{corr}\delta_{\theta}1.36\frac{\pi}{30000}\right)^2} \\
&= 0.237\ HP\ =\ 0.38\%\ at\ 62.76\ HP
\end{aligned}
$$

Error in Theoretical Air Flowrate :

$$
\begin{aligned}
\delta_{\rho,atm} &= \sqrt{\left(\frac{\partial}{\partial P_{amb}}\left(\frac{P_{amb}}{R_{air}T_{amb}}\right)\right)^2 \delta^2_{P,amb} + \left(\frac{\partial}{\partial T_{amb}}\left(\frac{P_{amb}}{R_{air}T_{amb}}\right)\right)^2 \delta^2_{T,amb}} \\
&= \sqrt{\left(\frac{\delta_{P,amb}}{R_{air}T_{amb}}\right)^2 + \left(-\frac{P_{amb}\delta_{T,amb}}{R_{air}T^2_{amb}}\right)^2} \\
&= 0.00586\ kg/m^3\ (0.00586 g/l) \\
&= 0.543\%\ for\ 1.079\ kg/m^3 \\
\delta_{\dot{m},th} &= \sqrt{\left(\frac{\partial}{\partial\theta}\left(\frac{V_{swept}\theta\rho_{atm}r}{60Z}\right)\right)^2 \delta^2_{\theta} + \left(\frac{\partial}{\partial\rho_{atm}}\left(\frac{V_{swept}\theta\rho_{atm}r}{60Z}\right)\right)^2 \delta^2_{\rho,atm} + \left(\frac{\partial}{\partial r}\left(\frac{V_{swept}\theta\rho_{atm}r}{60Z}\right)\right)^2 \delta^2_{r}} \\
&= \sqrt{\left(\frac{V_{swept}\delta_{\theta}\rho_{atm}r}{60Z}\right)^2 + \left(\frac{V_{swept}\delta_{\rho,atm}\theta r}{60Z}\right)^2 + \left(\frac{V_{swept}\delta_{r}\theta\rho_{atm}}{60Z}\right)^2} \\
&= 0.546\ g/s\ =\ 0.674\%\ for\ 81.075\ g/s
\end{aligned}
$$

Error in Volumetric Efficiency:

$$
\begin{aligned}
\delta_{\eta,V} &= \sqrt{\left(\frac{\partial}{\partial\dot{m}_{th}}\left(100\frac{\dot{m}_{air,act}}{\dot{m}_{air,th}}\right)\right)^2 \delta^2_{\dot{m},th} + \left(\frac{\partial}{\partial\dot{m}_{act}}\left(100\frac{\dot{m}_{air,act}}{\dot{m}_{air,th}}\right)\right)^2 \delta^2_{\dot{m}_{act}}} \\
&= \sqrt{\left(100\frac{\dot{m}_{air,act}\delta_{\dot{m}_{th}}}{\dot{m}_{th}{}^2}\right)^2 + \left(100\frac{\delta_{\dot{m}_{act}}}{\dot{m}_{th}}\right)^2} \\
&= 1.108\ =\ 1.82\%\ error\ for\ 61\%\ \eta_V
\end{aligned}
$$

Error in Brake Specific Fuel Consumption (BSFC):

$$
\begin{aligned}
\delta_{BSFC} &= \sqrt{\left(\frac{\partial}{\partial\dot{m}_{total}}\left(3600\frac{\dot{m}_{total}}{P_{corr}}\right)\right)^2 \delta^2_{\dot{m},total} + \left(\frac{\partial}{\partial P_{corr}}\left(3600\frac{\dot{m}_{total}}{P_{corr}}\right)\right)^2 \delta^2_{P,corr}} \\
&= \sqrt{\left(\frac{3600\delta_{\dot{m},total}}{P_{corr}}\right)^2 + \left(\frac{3600\dot{m}_{total}\delta_{P,corr}}{P^2_{corr}}\right)^2} \\
&= 1.913\ g/HP-h\ =\ 0.865\%\ at\ 221.24\ g/HP-h
\end{aligned}
$$

Brake Specific Energy Consumption:

$$\begin{aligned}
\delta_{BSEC}^{2} &= \left(\frac{\partial}{\partial \dot{m}_{eth}}\left(3600\frac{\dot{m}_{dies}Q_{dies}+\dot{m}_{eth}Q_{eth}}{P_{corr}}\right)\right)^{2}\delta_{\dot{m}_{eth}}^{2} \\
&+\left(\frac{\partial}{\partial \dot{m}_{dies}}\left(3600\frac{\dot{m}_{dies}Q_{dies}+\dot{m}_{eth}Q_{eth}}{P_{corr}}\right)\right)^{2}\delta_{\dot{m}_{dies}}^{2} \\
&+\left(\frac{\partial}{\partial P_{corr}}\left(3600\frac{\dot{m}_{dies}Q_{dies}+\dot{m}_{eth}Q_{eth}}{P_{corr}}\right)\right)^{2}\delta_{\dot{m}_{P_{corr}}}^{2} nonumber \qquad \text{(F.4)} \\
&= 3600^{2}\left[\left(\frac{Q_{eth}\delta_{\dot{m}_{eth}}}{P_{corr}}\right)^{2}+\left(\frac{Q_{dies}\delta_{\dot{m}_{dies}}}{P_{corr}}\right)^{2}+\left(\frac{(\dot{m}_{dies}Q_{dies}+\dot{m}_{eth}Q_{eth})\,\delta_{P_{corr}}}{P_{corr}^{2}}\right)^{2}\right] \\
&= 58.93\ kJ/HP-h\ =\ 0.71\%\ for\ 8255.77\ kJ/HP-h
\end{aligned}$$

Thermal efficiency :

$$\begin{aligned}
\delta_{m,\dot{diesel}} &= 0.02g/s\times frac3600s1hour\times\frac{1kg}{1000g} \\
&= 0.072kg/h \\
\delta_{m,\dot{ethl}} &= 0.0069g/s\times frac3600s1hour\times\frac{1kg}{1000g} \\
&= 0.02484kg/h \\
\delta_{\eta_{th}}^{2} &= \left(\frac{\partial}{\partial \dot{m}_{dies}}\left(\frac{P_{corr}\times 3600}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})\,1.341}\right)\right)^{2}\delta_{m,\dot{dies}}+ \\
&\left(\frac{\partial}{\partial \dot{m}_{eth}}\left(\frac{P_{corr}\times 3600}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})\,1.341}\right)\right)^{2}\delta_{m,\dot{eth}}+ \\
&\left(\frac{\partial}{\partial P_{corr}}\left(\frac{P_{corr}\times 3600}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})\,1.341}\right)\right)^{2}\delta_{P_{corr}} \\
&= \left(\frac{3600P_{corr}Q_{L,diesel}\delta_{m,\dot{dies}}}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})^{2}\,1.341}\right)^{2}+ \\
&\left(\frac{3600P_{corr}Q_{L,eth}\delta_{m,\dot{eth}}}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})^{2}\,1.341}\right)^{2}+ \\
&\left(\frac{3600\delta_{P,corr}}{(\dot{m}_{dies}\times Q_{L,diesel}+\dot{m}_{eth}\times Q_{L,eth})\,1.341}\right)^{2}+ \\
\delta_{\eta_{th}} &= 0.00236\%\ = 0.747\%\ error\ at\ 31.7\%\ \eta_{th}
\end{aligned}$$