ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

School of Mechanical Chemical and Materials Engineering

Design, Modeling and Simulation of Crank Gear and Valve Gear Mechanism of IC Engine for SUV Vehicle

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Study on Design, Modeling and Simulation of Crank gear mechanism and Valve gear mechanism of IC Engine for SUV Vehicle

by

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DECLARATION

I, the undersigned, declare that the thesis comprises my own work carried out from November 2017 to September 2018 under the supervision of Melesse Haile (Ass. Professor). In compliance with internationally accepted practices, I have acknowledged and refereed all materials used in this work. I understand that non-adherence to the principles of academic honesty and integrity, misrepresentation/ fabrication of any idea, data, fact and source will constitute sufficient ground for disciplinary action by the University and can also evoke penal action from the sources which have not been properly cited or acknowledged.

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LIST OF ABBREVIATIONS AND SYMBOLS

SUV	Sport Utility Vehicle
BAI	Bishoftu Automotive Industry
bsfc	Brake specific fuel consumption
bmep	Brake mean effective pressure
NVH	Noise, Vibration, and Harshness
R425DOHC	Double Overhead Cam
DRP	Disaster Recovery Planning
MANAG	Maschinenfabric Augsburg-Nuerenberg
CAE	Computer-aided Engineering
FEA	Finite Element Analysis
CNT's	Carbon Nanotubes
NTA	National Testing Agency
FEM	Finite Element Method
CMM	Coordinate-Measuring Machine
ALMMC	Aluminum Metal Matrix Composite
HDPE	High Density Polyethlene
CNG	Compressed Natural Gas
SEM	Scanning Electron Microscope
FGM	Functionally Graded Material
GSI	Geological Survey of India
fmep	Friction Mean Effective Pressure
imep	Indicate Mean Effective Pressure
η_m	Mechanical Efficiency

η_v	Volumetric Efficiency
Φ_{max}	Maximum Grade Ability (degree)
a_{max}	Maximum Vehicle Acceleration
FC_{min}	Minimum Fuel Consumption
P_{w-max}	Maximum Power Available at the Driving Wheels
R_{t-max}	Maximum Total Road Resistance
F_{tr-max}	Maximum Vehicle Tractive Force
T_{w-max}	Maximum Torque Available at the Driving Wheel
ω_w	Wheel Angular Velocity
θ	Angle of Inclination of the Crank from Top dead center, Twist angle
GR_i	Initial Gear Ratio
GR_F	Final Gear Ratio
N_{w-max}	Maximum Wheel Speed (rpm)
r_p	Pressure Ratio
r_c	Cut-off Ratio
β	Cam Angle for Maximum Follower rise (degree)
Z	Section Modulus, Mach index
k	Spring Stiffness
C	Spring index

ABSTRACT

Ethiopia automotive industries imported engine from different foreign countries to assembling for domestic market. Now a days the country is one of the fastest growing economy from agricultural lead to industries lead agriculture transformation as a result the government policy is highly encourage the industries to manufacture all parts of car engine made in Ethiopia. With the soon to be inaugurated engine factory they will be able to assemble and manufacture some of the models as 100% made in Ethiopia, but still lack of indigenous skilled man power to design engine affecting the industries as well as the country's economy. In case of this to transfer the technology to Ethiopia, crank and valve gear mechanism is designed. If the industries starts manufactured the price of the car will reduced.The crank gear mechanism and valve gear mechanism will be designed with C++, modeled using CAD and simulated using 3DS MAX. The aim of this thesis is to design, simulate and model a real engine crank gear mechanism and valve gear mechanism, having into account all necessary calculations concerning with kinematics, dynamics and strength calculation of basic details of compression ignition working Internal Combustion Engine. Studies have found no alternative type that well designed parts promises to have significant advantage in fuel economy or pollution control than the existing IC Engines parts. This work can be the base for future detailed designing work of IC Engine along with stress, kinematics analysis and simulation. This paper comes up with design aspects includes components like, piston, piston rings, cylinder, connecting rod, crank and crank shaft, cam and cam shaft along with valve and valve gear mechanism. The results of the program (C++) as well as data generated in the analysis indicated there was a significant difference between the existing engine part and the newly designed engine in diameter almost all parts and bearing pressures. The newly designed engine parts totally prevent failerity piston deformation and increasing strength. In size compare to analysis, the diameter of the newly designed engine and the existing diameter is different. These indicate that diameter the new design was almost have to greater or the material must be changed. The existing design was not safe by bearing pressure is out of allowable stress. All parameters that were calculated from dual cycle process that produces cylinder gas pressure taken the maximum cylinder gas pressure of 107.8 bar.

Keywords: Internal combustion engine, engine kinematics, engine kinetics, piston, piston ring, connecting rod mechanism, crank shaft, valve gear mechanism.

CHAPTER ONE

1 INTRODUCTION

Lissan megazine [1], In Ethiopia history the first car was imported by emperor menelik To Menelik in a Motor-Car known as "jemimas" by which name Bentley had learnt in his youth to designate the spring-side boots dear to old ladies. So, Ethiopian emperor Menelik II was the first African to drive a car, in 1907. Emperor Menelik laughs at the simplicity of the drip feed lubricator. British and Russian Ministers in back of car. They walked in procession to the porch of the Palace, and there Bentley gave a short lecture as to the working of a motor-car.

Muluken yewndosen [2] Ethiopia's first car assembly is Holland car. Holland Car has unveiled a new vehicle assembly plant 11 km west of Addis Ababa at a place known of Tatek, at a cost of 25 million birr. The company is finalizing the assembling infrastructure for the 'Cassiopeia' [a Greek word for prince of Ethiopia] factory which rests on a 30,000 m^2 plot. When Cassiopeia begins operating at full capacity in May 2009, it will assemble 1500 pickup vehicles per year. Cassiopeia is a sister company of the first vehicle assembler in Ethiopia, Holland Cars, known for its Abay [Amharic for the Nile River] model.

Zelalem girma [3] Ethiopia currently produces over 8,000 commercial and private vehicles a year for the local market, however it is below the country's potential. Currently, there are about 6 car assembly plants in Ethiopia, in which most of them are Chinese companies. On the other hand, the cost of importing cars from abroad is very expensive in this country. According to the Ethiopian Customs and Revenue Authority, both commercial and private vehicles imported into the country can be subjected to five different types of taxes. Hence, in order to encourage people to buy locally made cars, the government has to give incentives and tax exemptions to foreign car manufacturers. Since they are encouraged to setting up and assembling new vehicles in this country, the country will minimize spending more foreign currency that may be used for imported cars.

In few years time the facility would be producing all materials for their products within the country! Although BAI assemble and manufacture in house 60% of most of the eight models today, they still import chassis and engines from different countries. However with the soon to be inaugurated engine factory they will be able to assemble and manufacture some of the models as 100% made in Ethiopia. So, the most challeng to design and manufuacture from the vehicle part is IC engine. In this chapter we focus IC engine background; i.e, performance curve and design condition of IC engines, principal parts, mechanisms and statement of the problem, objectives, significance of the project and scope.

1.1 Background

The internal combustion engine was conceived and developed in the late 1800s. It has had a significant impact on society, and is considered one of the most significant inventions of the last century. The internal combustion engine has been the foundation for the successful development of many commercial technologies.

Internal combustion engines can deliver power in the range from 0.01 kW to 20 x 10^2 kW, depending on their displacement. The major applications are in the vehicle (automobile and truck), railroad, marine, aircraft, home use and stationary areas. The vast majority of internal combustion engines are produced for vehicular applications, requiring a power output on the order of 10^2 kW.

The adoption and continued use of the internal combustion engine in different application areas has resulted from its relatively low cost, favorable power to weight ratio, high efficiency, and relatively simple and robust operating characteristics.

The internal combustion engines are spark-ignition engines (sometimes called Otto engines, or gasoline or petrol engines, though other fuels can be used) and compression-ignition or diesel engines. Because of their simplicity, ruggedness and high power/weight ratio, these two types of engine have found wide application in transportation (land, sea, and air) and power generation. It is the fact that combustion takes place inside the work producing part of these engines that makes their design and operating characteristics fundamentally different from those of other types of engine.

Practical heat engines have served mankind for over two and a half centuries. For the first 150 years, water, raised to steam, was interposed between the combustion gases produced by burning the fuel and the work-producing piston in-cylinder expander. It was not until the 1860s that the internal combustion engine became a practical reality [4, 5]. The early engines developed for commercial use burned coal-gas air mixtures at atmospheric pressure there was no compression before combustion. J. J. E. Lenoir (1822-1900) developed the first mar-' ketable engine of this type. Some 5000 of these engines were built between 1860 and 1865 in sizes up to six horsepower. Efficiency was at best about 5 percent.

New materials now becoming available offer the possibilities of reduced engine weight, cost, and heat losses, and of different and more efficient internal combustion engine systems. Alternative types of internal combustion engines, such as the stratified charge (which combines characteristics normally associated with either the spark-ignition or diesel) with its wider fuel tolerance, may become sufficiently attractive to reach large-scale production. The engine development opportunities of the future are substantial. While they present a formidable challenge to automotive engineers, they will be made possible in large part by the enormous expansion of our knowledge of engine processes which the last twenty years has witnessed.

1.2 Statement of the problem

Ethiopia automotive industries have imported engine from different foreign countries to assemble for domestic market. However, most of the time bishoftu automotive industry imports huanghai that specifies in the scope have a big maintenance problem. In other words the engine problem did not solved simply by maintenance rather it requires a series design problem. In 2009, 30 engines of SUV were maintained. Among this 7 of 30 engine came back for maintenance without usage. But the technitians did not know the right solution even if they know the symptom such as cold starting problem (after four crank), deformation of the piston and rings, wearing, noise, sound and seize (third). Now a days the country is one of the fastest growing economy from agricultural lead to industries lead agriculture transformation as a result the government policy is highly encouraging the industries to manufacture all parts of car engine made in Ethiopia. To be inaugurated engine factory they will be able to assemble and manufacture some of the models as 100% made in Ethiopia, but still lack of indigenous skilled man power to design engine affecting the industries as well as the country's economy. In case of this to transfer the technology to Ethiopia and to solve the problems SUV engine crank and valve gear mechanism is designed, the industries starts manufacturing and the price of the car will reduce.

1.3 Objectives

1.3.1 General Objective

The general objective of this thesis is to design, model and simulate engine crank gear and valve gear mechanism of CI Engine.

1.3.2 Specific Objectives

The specific objective of the proposed project are as follows:

- To collect data related to Sport Utility Vehicle (SUV)
- To design crank gear mechanism of compression ignition engine
- To design valve gear mechanism of compression ignition engine

- To simulate the desined mechanism by using C++ software programing
- To geometrically model each parts and assemble of crank gear and valve gear mechanism by using CAD software
- To prepare prototype from available materials

1.4 Significance of the project

The significance of the project are as follows:

- It helps in technology transfer.
- It initiates Ethiopia auto industry.
- It acts as an input for starting engine manufacturing in Ethiopia.
- The price of the car reduces.
- It creates job opportunities.

1.5 Beneficiaries

- Ethiopia car assembler and manufacturer industry. For example, Bishoftu Automotive Industry
- Car buyer as a whole will benefit due to reduction of vehicle cost.
- Ethiopia government and people due to creation of job.

1.6 Delimitation (scope)

This work is delimited to designing, simulation and modelling of a crank gear mechanism and valve gear mechanism of compression ignition internal combustion engine. Design, simulation and model of the mechanism is done by using C++, Excel and CAD respectively. The work includes designing, modeling, and simulation of main parts of the mechanism for Sport utility vehicle (SUV), Car model of DD6470C, Engine model of R425DOHC, Four cylinders in line, four stroke, and water cooled, inter cooler electric control, common rail diesel engine per cycle with four valves, Displacement (L) of 2499CC, Max. Power (kw/r/min) of 105/4000, Max. Torque (Nm/r/min) of 340/2000, Vehicle weight of 1860kg, Vehicle + load + passenger + driver of 2510kg, Max. Speed of 130km/hr., Fuel tank capacity of 90L, Emission of Euro III. The porotype of the engine is subjected to the availability facilities and resources will be done for demonstration.

CHAPTER TWO

2 LITERATURE REVIEW

2.1 History and Development of Engine

The paper on this design focuses IC diesel engine that imports from china to Ethiopia bishoftu automotive industry. So to know about diesel engine Rudolf Diesel is the first inovetor.

2.1.1 The Importance of Rudolf Diesel

Rudolf Diesel designed many heat engines, including a solar-powered air engine. In 1893, he published a paper describing an engine with combustion within a cylinder, the internal combustion engine. In 1894, he filed for a patent for his new invention, dubbed the diesel engine. Rudolf Diesel was almost killed by his engine when it exploded. However, his engine was the first that proved that fuel could be ignited without a spark. He operated his first successful engine in 1897.In 1898, Rudolf Diesel was granted patent #608,845 for an "internal combustion engine" the Diesel engine. The diesel engines of today are refined and improved versions of Rudolf Diesel's original concept. They are often used in submarines, ships, locomotives, and large trucks and in electric generating plants. Rudolf Diesel's inventions have three points in common: They relate to heat transference by natural physical processes or laws; they involve markedly creative mechanical design; and they were initially motivated by the inventor's concept of sociological needs. Rudolf Diesel originally conceived the diesel engine to enable independent craftsmen and artisans to compete with large industry. At Augsburg, on August 10, 1893, Rudolf Diesel's prime model, a single 10-foot iron cylinder with a flywheel at its base, ran on its own power for the first time. Rudolf Diesel spent two more years making improvements and in 1896 demonstrated another model with the theoretical efficiency of 75 percent, in contrast to the ten percent efficiency of the steam engine. His engines were used to power pipelines, electric and water plants, automobiles and trucks, and marine craft, and soon after were used in mines, oil fields, factories, and transoceanic shipping.

He set up a laboratory in Paris in 1885, and took out his first patent in 1892. In August 1893 he went to Augsburg, Germany, where he showed the forerunner of Maschinenfabrik Augsburg-Nuerenberg (MAN AG) a three-meter-long iron cylinder with a piston driving a flywheel. It was an economic thermodynamic engine to replace the steam engine. Diesel called it an atmospheric gas engine, but the name didn't stick. He worked on. On New Year's Eve 1896 he proudly displayed an engine that had a theoretical efficiency of 75.6 percent. Of course, this theoretical efficiency could not be attained, but there was nothing to equal it - and there is nothing to equal it to this day - in thermodynamic engines.

The self-igniting engine was a sensation of the outgoing century, though Rudolf Diesel's dream of enabling the small craftsmen to withstand the power of big industry did not ripen. Instead, big industry quickly took up his idea, and Diesel became very rich with his royalties. From all over the world money flowed to him as his engines became the standard to power ships, electric plants, pumps and oil drills. In 1908 Diesel and the Swiss mechanical firm of Saurer created a faster-running engine that turned at 800 rpm, but the automotive industry was slower to adopt Diesel's engine.

MAN was the first, and in 1924, a MAN truck became the first vehicle to use a direct-injection diesel engine. At the same time Benz & Cie in Germany also presented a diesel truck, but Benz used the mixing chamber that Daimler-Benz kept into the 1990s. The first diesel Mercedes-Benz hit the road in 1936. But Rudolph Diesel didn't get to see his inventions' victorious march through the automotive world. He drowned in 1913 in the English Channel.

The modern diesel engine came about as the result of the internal combustion principles first proposed by Sadi Camot in the 19th century. Dr. Rudolf Diesel applied Sadi Camot’s principles in to a patented engine cycle or method of combustion that has become as the diesel cycle. His patented engine operated when the heat generated during the compression of the air fuel charge caused ignition of the mixture, which then expanded at a constant pressure during the full power stroke of the engine.

Dr. Diesel’s first engine run on coal dust and a compression pressure of 1500 psi to increase its theoretical efficiency. Also, his first engine did not have provisions for any type of cooling system. Consequently, between the extreme pressure and the lack of cooling, the engine exploded and almost killed its inventor. After recovering from his injuries, Diesel tried again using oil as the fuel, adding a cooling water jacket around the cylinder, and lowering the compression pressure to approximately 550 psi. This combination eventually proved successful. Production rights to the engine were sold to Adolphus Bush, who built the first diesel engines for commercial use, installing them in his St. Louis Brewery to drive various pumps.

2.2 Performance Curves and Design Conditions of IC Engines

During engine design, one of the most important tasks is to obtain the maximum cylinder gas pressure (P_{g-max}) during the whole range of engine speed (from N_{min} up toN_{max}). This can be determined from the engine Performance Curves. During engine design, the required major Performance Curves are: the brake power (P_b), brake torque (T_b), brake specific fuel consumption ($bsfc$), brake mean effective pressure ($bmep$), mechanical efficiency (η_m) and volumetric efficiency (η_v) plotted as a function of engine speed (N). Such curves can be obtained using Analytical Relations or Empirical Relations.

Analytical_Relations can be formulated from a tedious mathematical analysis of engine cycles processes with a detail mathematical analysis of combustion processes. Hence, such a task is not only very difficult, but also time consuming. Whereas, Empirical_Relations can be formulated from experimental (engine performance test) data using Curve Fitting methods. Hence, such a task is relatively easier. So in this paper were used both empirical relation for initial design that will be discussed in chapter four and analytical relation for checking the existing engine size determination that will discussed in chapter five.

Changes in the basic loads acting on the engine parts are dependent on the operating conditions of the engine. Generally, the engine parts are designed for the most sever conditions.

For CI (Turbocharged) engines, the basic design conditions include:

i) Maximum brake Torque (T_{b-max}): which occurs at $N_{@Tb-max} \approx$ (0.5to0.8) x $N_{@Pb-max}$, when the gas pressure reaches nearly its maximum value (P_{g-max}) while the inertia forces are comparatively lower. Even though the maximum cycle gas pressure (P_{g-max}) in the case of CI engines is attained at the point of P_{b-max} or at $N_{@Pb-max}$, the point of T_{b-max} can be taken as the first design point (DP-1).

ii) Maximum brake (Rated) Power (P_{b-max}): which occurs at $N_{@Pb-max}$, when the gas pressure reaches its maximum value (P_{g-max}) and also inertia loads are comparatively higher. This point can be taken as the second design point (DP-2).

iii) Maximum engine speed (N_{max}): which occurs at $N_{max} \approx N_{@Pb-max}$, when the inertia forces reach their maximum value while the gas pressure is comparatively lower. This point can be taken as the third design point (DP-3).

While designing an engine for a motor vehicle (automobile), the most important thing to be taken into considerations are: the maximum vehicle velocity (V_{max}), the maximum gradeability (Φ_{max}), the maximum vehicle acceleration (a_{max}) and the minimum Fuel Consumption (FC_{min}).

- The maximum vehicle velocity (V_{max}) is limited by the maximum Power available at the driving wheels (P_{w-max}). The maximum engine (brake) Power (P_{b-max}) depends on the maximum Power available at the driving wheels (P_{w-max}). The maximum Power available at the driving wheels (P_{w-max}), in turn, depend so the maximum total road resistance (R_{t-max}) and the maximum vehicle velocity (V_{max}).
- The maximum grade ability (Φ_{max}) and vehicle acceleration (a_{max}) depends mainly on the maximum vehicle tractive force (F_{tr-max}). The maximum vehicle tractive force (F_{tr-max}), in turn, depends on the maximum Torque available at the driving wheels (T_{w-max}). The maximum

Torque available at the driving wheels (T_{w-max}), in turn, depends on the Power and angular speed available at the driving wheels (P_w and ω_w).

- The minimum Fuel Consumption (FC_{min}) is limited by the brake Power (P_b) and the brake specific fuel consumption (bsfc) of the engine.

2.3 Principal parts of an I.C Engine

The principal parts of an engine are; 1. Cylinder and cylinder liner, 2. Piston, piston rings and piston pin or gudgeon pin, 3. Connecting rod with small and big end bearing, 4. Crank, crankshaft and crank pin, and 5. Valve gear mechanism.

The design of the above mentioned principal parts are discussed in detail in chapter five. But those principal parts can be categorized in to two main mechanisms; crank and valve gear mechanism. The third mechanism is used for the connection of the two mechanism. So, in this paper neglected timing gear mechanism for detail design.

2.3.1 IC Engine Mechanisms

The mechanisms of an IC engine are: Crank Gear and Valve Gear.

1. **The Crank Gear Mechanism**

The main function of the crank gear mechanism of a reciprocating IC engine is, to convert the Indicated Power (p_i) available on the piston head (a portion of Thermal or Heat Energy) into Brake Power (p_b) at the flywheel (Mechanical Energy). It consists of the Cylinder, Cylinder Head, Piston, Piston Rings (compression and oil control rings), piston pin, connecting rod, crankshaft, bearings (journal and thrust), flywheel, and vibration damper.

Even if the crank gear mechanism have those the above mentioned parts; this paper designed only cylinder liner, piston, Piston Rings (Compression and oil control rings), piston pin, connecting rod, and crankshaft as discussed in chapter five.

The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod, piston, piston rings, and piston pin. Combustion and inertia forces are acting on the crankshaft.

2. **The Valve Gear Mechanism**

The main function of the Valve Gear Mechanism of a reciprocating IC Engine is, to actuate (open and close) the inlet and exhaust valves at the required time with respect to the position of the crankshaft (θ). It consists of the camshaft, cam, rocker assembly, valve assembly.

In the valve gear mechanism the paper designed camshaft, cam, rocker arm, valve and valve spring.

Internal combustion engine valves are precision engine components. The valve train system is one of the major parts of internal combustion engine, which controls the amount of air-fuel mixture to be drawn into the cylinder and exhaust gas to be discharged. The fresh charge (air -fuel mixture in Spark Ignition Engines and air alone in Compression Ignition Engines) is induced through inlet valves and the products of combustion get discharged to atmosphere through exhaust valves. This seals the working space inside the cylinder against the manifolds [4, 5 and 6]. So design of valve lift profiles and valve train components is most important for the engine performance, valve train durability, and noise, vibration and harshness (NVH). Therefore; valve train system should be optimally designed so as to avoid an abnormal valve movement, such as valve jumping or bounce up to the maximum engine speed. There are different types of valves used by the manufactures; some common types of valves being poppet valves, slide valves, rotary valves and sleeve valveas shown in figure 2.1, bellow.

The two mechanisms can be connected by means of a Timing Gear or a Timing Chain or a Timing Belt drive as shown in figure 2.1, below. In doing so, power is transmitted from the crankshaft to the camshaft to actuate (open and close) the valves.

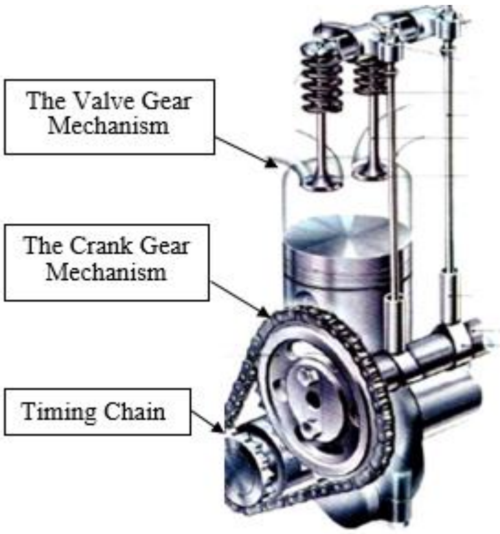


Figure 2. 1 Crank and Valve Gear Mechanisms of a Reciprocating IC Engine [37]

2.4 Design Aspects of Cylinder and Cylinder Liner

The cylinder liner is typically plateau honed resulting in a smooth load bearing area of the surface with deep valleys, called honing grooves.An image of a typical cylinder liner surface can be seen in Figure 3.2. The honing grooves can act as a reservoir for the lubricant and together with the plateau sections they can also act as micro bearings in the contact [7]. The micro bearing effect can generate more hydrodynamic pressure [8] thus increasing the oil film thickness resulting in less friction and risk of wear in the contact. Plateau honed cylinder liner typically leads to decreased oil consumption and risk of scuffing and wear on both cylinder liner and piston ring [9].

Wet liners have water in direct contact and are used in engine bore size 130 mm plus. Wet liners reduces the foundry problem of casting the cylinder and cylinder is not subjected to thermal stress due to water in it, but it is difficult to replace wet liner from the cylinder as chances of water leakage in the combustion space and crank case. Dry liners are fitted in the cylinder wall and can be easily replaced as there is no chance of water leakage from jacket. The upper part of liner is cooled better with the use of dry liners, the casting of cylinder becomes complicated and heat flow through the composite wall of liner and cylinder is reduced [10].

2.5 Design Aspects of Piston and Piston Rings

This topic shows review on design analysis of piston on the basis of improving strength according to the material properties. Ch. Venkata et al. [11], focused on Design analysis and optimization of piston using CATIA and ANSYS. He had optimized with all parameters are within consideration. Target of optimization was to reach a mass reduction of piston. In this analysis, a ceramic coating on crown is made. In an optimization of piston, the length is constant because heat flow is not affected the length, diameter is also made constant due to same reason. The volume varied after applying temperature and pressure loads over piston as volume is not only depending on length and diameter but also on thickness which is more affected. The material is removed to reduce the weight of the piston with reduced material. The results obtained by this analysis shows that, by reducing the volume of the piston, thickness of barrel and width of other ring lands, Von mises stress is increased by and deflection is increased after optimization. But all the parameters are with in design consideration S. Chougule et al. [12], focused on the main objective of this paper is to investigate and analyze the stress distribution of piston at actual engine condition during combustion process the parameters used for simulation is operating gas pressure and material properties of piston. She concluded that there is a scope for reduction in a scope for reduction in thickness of piston and therefore 0ptimization of piston is done with mass reduction by 24.319% than non-optimized piston. The static and dynamic analysis is carried out which are well below the permissible stress value. The study of L. Singh et al. [13], is related to the material for the piston is aluminum-silicon composites. The high temperature at piston head, due to direct contact with gas, thermal boundary conditions is applied and for maximum pressure mechanical boundary conditions are applied. After all these analysis all values obtained by the analysis is less than permissible value so the design is safe under applied loading condition. The study of R. C. Singh et al. [14], discussed about failure of piston in I.C. engines, after all the review, it was found that the function coefficient increases with increasing surface roughness of liner surface and thermal performance of the piston increases. The stress values obtained from FEA during analysis is compared with material properties of the piston like aluminum alloy zirconium material. If those value obtained are less than allowable stress value of material then the design is safe.

2.6 Design Aspects of Connecting Rod Assembly

K. Sudershan et al. [15], this paper describes modelling and analysis of connecting rod. In this project connecting rod is replaced by Aluminium reinforced with Boron carbide for Suzuki GS150R motorbike. A 2D drawing is drafted from the calculations. A parametric model of connecting rod is modelled using PRO-E 4.0 software. Analysis is carried out by using ANSYS software**.** B. Anusha et al. [16] the modelled connecting rod imported to the analysis software i.e. ANSYS. Static analysis is done to determine von-misses stresses, strain, shear stress and total deformation for the given loading conditions using analysis software i.e. ANSYS. In this analysis two materials are selected and analyzed. The software results of two materials are compared and utilized for designing the connecting rod. Mr. H. B. Ramani et al. [17], in this study, detailed load analysis was performed on connecting rod, followed by finite element method in Ansys-13 medium. In this regard, In order to calculate stress in Different part of connecting rod, the total forces exerted connecting rod were calculated and then it was modelled, meshed and loaded in ANSYS software. The maximum stresses in different parts of connecting rod were determined by analysis. The maximum pressure stress was between pin end and rod linkages and between bearing cup and connecting rod linkage. The maximum tensile stress was obtained in lower half of pin end and between pin end and rod linkage. It is suggested that the results obtained can be useful to bring about modification in Design of connecting rod.

A. Gautam [18], performed static stress analysis of connecting rod made up of SS 304 material used in Cummins NTA 885 BC engine they developed model in CATIA V5 software and imported to ANSYS WORKBENCH and presented the results of the material and reported that the area close to root of the smaller end of connecting rod is very prone to failure, may be due to higher crushing load due to gudgeon pin assembly.

2.7 Design Aspects of Crank Shaft Assembly

Jian et al. [19], analyzed crankshaft model and crank throw were created by Pro/ENGINEER software and then imported to ANSYS software. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal, crankpin and crank cheeks. Gu et al. [20], researched a three-dimensional model of a diesel engine crankshaft was established by using PRO/E software. Using ANSYS analysis tool, it shows that the high stress region mainly concentrates in the knuckles of the crank arm & the main journal and the crank arm & connecting rod journal, which is the area most easily broken. Farzin et al. [21], investigated first dynamic load analysis of the crankshaft. Results from the FE model are then presented which includes identification of the critically stressed location, variation of stresses over an entire cycle, and a discussion of the effects of engine speed as well as torsion load on stresses. Deshbhratar et al. [22], analyzed 4- cylinder crankshaft and model of the crankshaft were created by using Pro/E Software and then imported to ANSYS software. The maximum deformation appears at the centre of crankshaft surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So this area prone to appear the bending fatigue crack. Abhishek et al. [23], have analyzed crankshaft model and 3-dimentional model of the crankshaft were created by SOLID WORKS Software and imported to ANSYS software. The crankshaft maximum deformation appears at the centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journals and crank cheeks and near the central point journal. The edge of main journal is high stress area.

The crank shaft is subjected to both twisting moment and bending moment hence it must have sufficient strength to withstand the forces.

2.8 Design Aspects of Camshaft

S.G.Thorat et al. [24] the goal of the project is to design cam shaft analytically, its modelling and analysis under FEM. In FEM, 3heavier of cam shaft is obtained by heavier the collective 3heavier of the elements to make the cam shaft robust at all possible load cases. This analysis is an important step for fixing an optimum size of acamshaft and knowing the dynamic behaviours of the camshaft. Initially the model is created by the basic needs of an engine with the available background data such as power to be transmitted, forces acting over the camshaft by means of valve train while running at maximum speed.

M. Shobha [25], in this project, a cam shaft will be designed for a 150cc engine and modelled through pro/engineer. Present used material for camshaft is cast iron. In this work, the camshaft material will be replaced with steel and aluminium alloy. Structural analysis and model analysis will be done on cam shaft using cast iron, steel and aluminium alloy. Comparison will be done for the three materials to verify the better material for camshaft. Modelling will be done using pro/Engineer software and analysis will be done using ANSYS.

Zeyaullah et al. [26], In the present work composed Automobile camshaft by Numerical Calculations there after it is planned by utilizing Modelling software PRO-E and CAE Analysis is done in ANSYS by differing material AL Metal Matrix Composite (ALMMC) to research the deformation, stress and strain developed on camshaft. The examination will give the most ideal approach to think us for the further future work of camshaft. Keywords: Design, Analysis, Pro-E, Ansys, IC Engines.

2.9 Design Aspects of Rocker Arm

Jafar et al. [27], they found the various stresses under extreme load condition. For this they were modelled the arm using design software and the stressed regions are found out using ANSYS software. Here in this thesis they were observed that by changing different materials how the stresses are varied in the rocker arm under extreme load condition. After comparing results they are proposed best suitable material for the rocker arm under extreme load conditions. The usual materials used for such purpose are Steel, Aluminum, and Forged steel to Stainless steel, alloys and composites. They investigated the possibility creating a light weight rocker arm that could provide a friction reducing fulcrum using needle bearings and a roller tip for reduced friction between the rocker and the valve stem but still be less expensive than steel lies in the development of composite rocker arms.

Syed et al. [28], worked on Design of Rocker Arm, they were discussed about Rocker arm of Tata Sumo victa that was designed and analysed to find the critical regions. They were created CAD models of Rocker Arm using Pro/E and ANSYS software was used for analysis of rocker arm. They found that the CAD model was inputted in ANSYS Workbench and Equivalent Stress and Maximum Shear Stress. The obtained results provided by ANSYS Workbench are compared to the results obtained by manual calculation. They found the shear stress at the pin of a rocker arm was evaluated by calculation and ANSYS software. The values of both shear stress and critical shear stress was nearly same. Thus they concluded that pin of rocker arm is under shear stress.

2.10 Design aspects of valve

Goli et al. [29], proposed the Failure Analysis of Internal Combustion Engine Valves by using ANSYS. The present study is focused on checking the size of IC engine valves by calculation. Naresh Kr. Raghuwanshi et.al [30], had given a review on the failure analysis of IC engine valves. From the study, it is quite evident that a common cause of valve fracture is fatigue. J.G. et al. [31], presented an optimal design of the exhaust system layout to suppress the discharge noise from an idling engine. Oh et al. [32], investigated the valve stem failure of exhaust valve from a Waukesha P9390 GSI gas engine. He concluded that the valve failed as a result of overheating. From the literature review it is seen that most of the studies on exhaust valve design had focused on fatigue behavior, wear behavior, deformation mechanisms in metallic materials. However, very less research is done in design of exhaust valve based on optimization of its parameter i.e. fillet radius, valve stem diameter, valve head diameter. Therefore this study is conducted to design the exhaust valve by optimizing the fillet radius and recommending the best alternative material for valve through experimentation and validation in order to increase the working life of exhaust valve.

2.11 Design Aspects of vavle spring

P. H. Jain et al. [33], performed static and fatigue life analysis on the valve spring made up of EN 47 steel. Fatigue study is performed to find life of valve spring in terms of number of cycles. The fatigue study was performed by using msc-fatigue software. The above analysis is performed for different values of the pitch. Satbeer [34], designed the spring based on static load, buckling and natural frequency. The static analysis was performed was two different spring materials namely steel and epoxy composites. Gajanan [35], performed fatigue analysis of valve spring and suggested that in order to improve the fatigue life the stress induced in the spring must be lower. Shot peening can also improve the fatigue life. Syed [36], calculated the forces acting on the rocker arm of the engine and performed the static analysis of rocker arm to verify the maximum shear stress.

CHAPTER THREE

3 MATERIALS AND METHODS

3.1 Materials

The material requirement for this project are as follows:

Table 3. 1 Material Requirements

Equipment name	Number
Manual for model R425DOHC	1
Micro meter	1
Deep guage	1
Verner caliper	2
C++ software	1
CAD software	1
Solid work and ansys softwares	2
Machnes (cutting, drilling, welding, leath and)	5

3.2 Methods

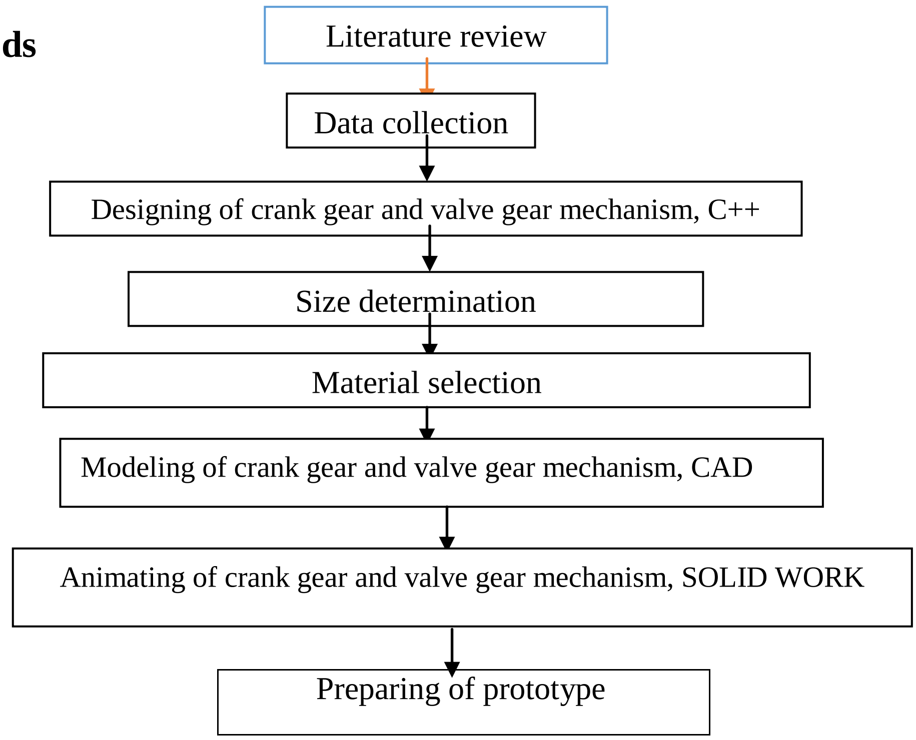


Figure 3. 1 Flow Chart Diagram

3.2.1 Data collection

Table 3. 2 Main Technical Parameter Specification of the Vehicle

Vehicle model: DD6470C **Engine model:** R425DOHC

Model of diesel engine	Direct-array, water cooling, four-stroke, dual overhead camshaft
Model of combustion chamber	Direct-injection
Number of cylinders-bore×stroke	4-92*94
Piston displacement (L)	2.499
Cylinder sleeve	Wet cast cylinder sleeve
Compression ratio	17.5
Overall dimension: length×width×height (mm)	710×630×765(without a fan)
Rated power/speed (kW/r/min)	105/4000
Lowest fuel consumption under full road (g/ kW.hr.)	210
Maximal torque/speed (Nm/r/min)	340/2000
Maximum no load governed speed (r/min)	4800
Minimum stable idle speed (r/min)	800
Weight (Kg)	278(including accessories such as AC, etc.)
Cylinder working order	1-3-4-2
Method of adjusting valve lash	Hydraulic lash adjuster
Inlet valve open	15.6°(before top dead center)
Inlet valve closed	64.4°(after bottom dead center)
Exhaust valve open	66°(before bottom dead center)
Exhaust valve closed	32°(After top dead center)
Thermostat	Wax-bead type
Direction of rotation of the engine (from the fly wheel end)	Counter-clockwise
Fuel-injection system	High-pressure common-rail system
Lubricating method	Combined pressure cycling and splashing
Lubricating oil capacity(L)	5.5-6.5(with an oil filter)
Cooling method	Closed pressure cycle
Generator(voltage—capacity)	12V—50A
Starter(voltage—output power)	12V—2.2KW
Exhaust-temperature (℃)	≤650
Outlet temperature of cooling water (℃)	85±5
Oil pressure (MPa) rated speed/idle speed	0.38-0.42/0.1

Transmission

Items	Specification	
transmission models	JC538T3, JC538T8, JC538T9 constant mesh, 5-speed	
Number of gears	Gear ratio	Efficiency
1st Gear	4.016	
2nd Gear	2.318	
3rd Gear	1.401	
4th Gear	1.000	
5th Gear	0.778	
Reverse gear	3.549	
Final drive	6.776	
Lubricating oil specification	GL-4, 75W/90	
Fill content of lubricating oil	1.80L	
Rating torque	380N.m	
Clutch control	Hydraulic pressure	
Lubricating oil filling method	Inject to the necessary lubricating oil level from the side tap hole: the lower level of the tap hole.	
weight of transmission	50 kg (no oil)	
Vehicle Dimension		
Length	4620 mm	
Width	1860 mm	
Height	1830 mm	
Wheelbase	2730 mm	
Max. speed	130 km/hr.	
Approaching angle/departure angle	26/28	
Seating capacity	5 persons	
Average fuel consumption under limited conditions	7.8L/100 km	

Table 3. 3 Design Data for CI Engines

	Noc	R	D (mm)	S/D	N_{max} (rpm)	$bmeP_{max}$ (bar)	p_{b-max}/E_c (kW/liter)	W_e/p_{b-max} (kg/kW)	$bsfc_{@pb-max}$ (g/kWh)
Motorcycles	4s	17 - 23	75 - 10	0.9 - 1.	4000 – 5000	5 - 7.5	18 - 22	2.5 – 5	250
Passenger Ca	4s	16 - 22	100 - 1	0.8 - 1.	2100 – 4000	6 – 9	15 - 22	4 -7	210
Trucks (NA)	4s	14 - 20	90 - 13	0.8 - 1.	2100 – 4000	12 – 18	18 - 26	3.5 – 7	200

(Source: J.B. Heywood: Internal Combustion Engines Fundamentals).

Figure 3. 2 Photo of SUV Vehicle

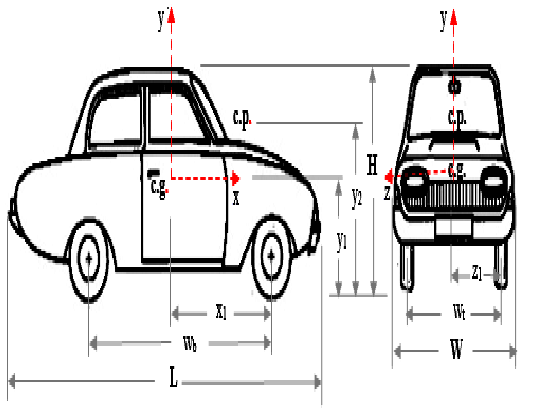


Figure 3. 3Vehicle Drawing and Dimensioning [37]

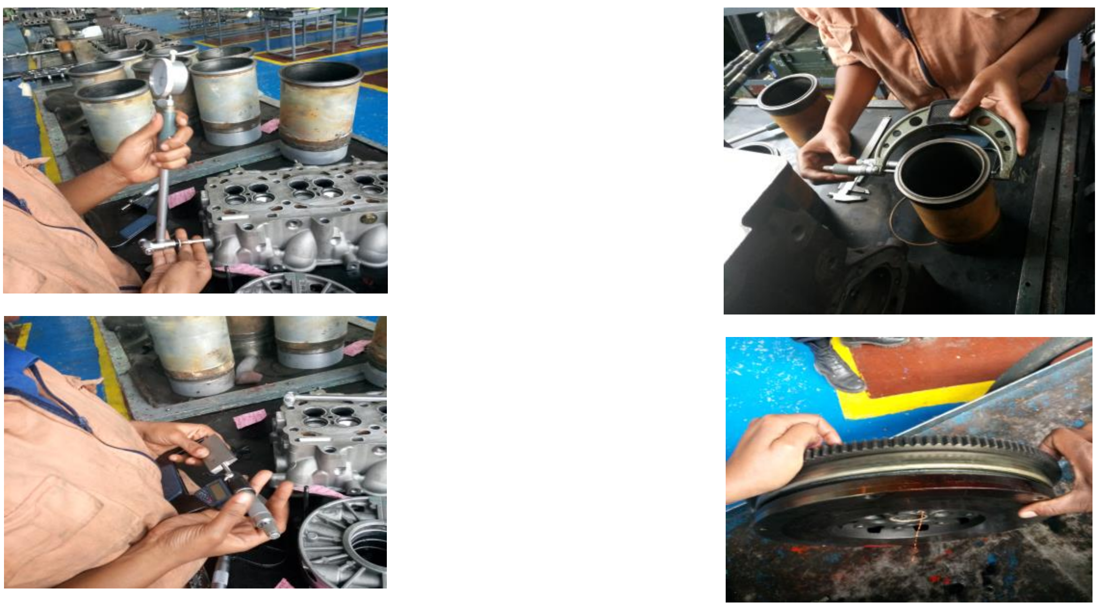


Figure 3. 4 Direct Measurement of Engine Parts

3.3 Designing of crank gear and valve gear mechanism

Detail design is the part of a design process which completes the embodiment of technical products with final instructions about the shapes, forms, dimensions and surface properties of all individual components, the definitive selection of materials, and a final scrutiny of the production methods, operating procedures and costs. Hence the three dimension sketch is presented in Figure 3.5.

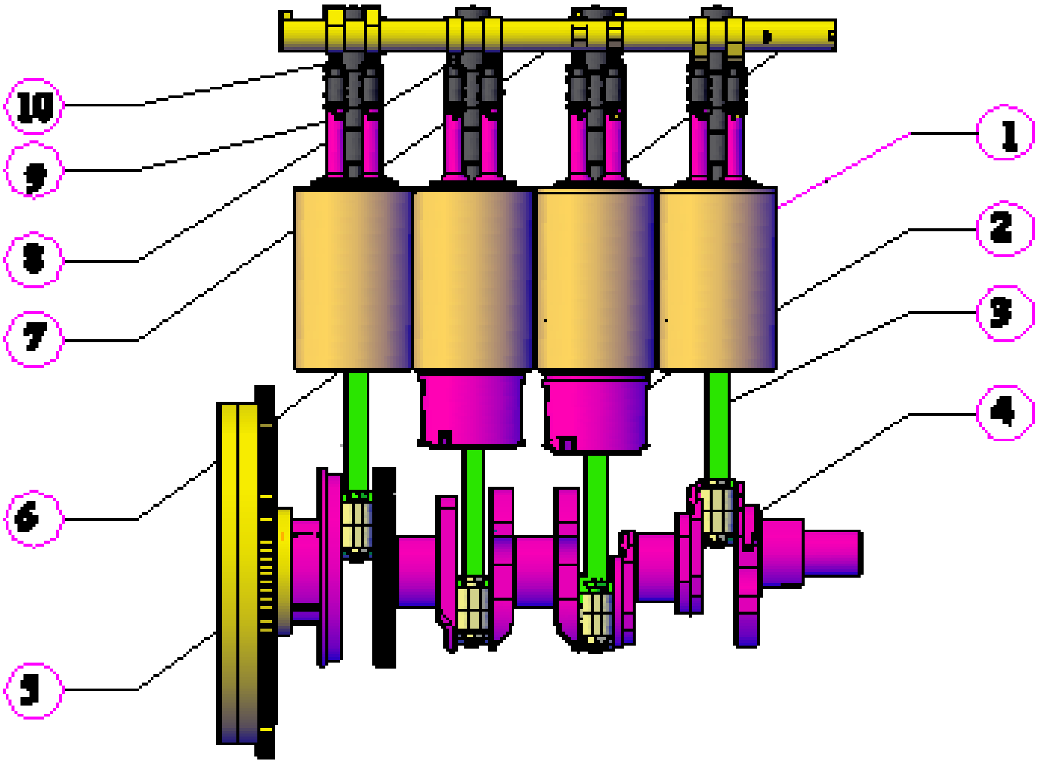


Figure 3. 5 Three Dimensional Sketch of the Crank and Valve Gear Mechanism: (1) Cylinder Liners, (2) Pistons, (3) Connecting rod, (4) Crankshaft, (5) Flywheel, (6) Camshafts, (7) Cams, (8) Rocker Arms, (9) Valves, (10) Valve Springs

Assumptions for the design analysis

- ❖ Since the vehicle is modern it uses dual cycle process
- ❖ The minimum vehicle gradient (level road), $\phi = 1°$
- ❖ The maximum vehicle gradient, $\phi = 12°$
- ❖ The optimum gear ratio 1.
- ❖ The head wind velocity according Ethiopia weather context is 0 m/s.

3.3.1 Design of the Crank Gear Mechanism

The crank gear mechanism of an IC engine convert chemical energy of fuel into heat and then to mechanical energy. Piston, piston pin, connecting rod, crankshaft and fly wheel are parts in crank gear mechanism. Even if there are many many crank gear mechanism arrangement; based on the selected vehicle that are exist in BAI with the manual of technical parameter of engine and vehicle dimension is shown in table 3.2. By considering the manual, direct measurment and the above data the cylinder pressure, engine performance, kinematics and kinetics is calculated in this chapter and each part of the mechanism size (by analysis and direct measurement) and material determined in chapter five. Measurement of each part were conducted by a chief inspection professional as shown in figure 3.4 and the mechanical property of the material is presented in appendex table C.

To determine the effective radius of a tire (r_w)

Based on observation that are written on the tire we can read Aluminum alloy rim 17”235/65 tire. So as shown in figure 3.6 below, from these we can calculate the effective tire radius using the following relation.

$h_T = 235x65\% = 152.75\ mm = 6\ in$

$r_w = \frac{2(h_T+14)}{2} = \frac{2(6+17)}{2} = 23\ in = 0.5842\ m$

Angular velocity of wheel (ω_w)

According many books and literature, the vehicle velocity (V) or the wheel velocity (V_w) and the angular velocity of the wheel (ω_w), as shown in Figure 4.4, can be related as:

$V = V_w = \omega_w\ x\ r_w$ ……………………………………………………………………….……. (1)

$\omega_w = \frac{V}{r_w}$

$\omega_w = \frac{36.11}{0.5842} = 61.81$ rad/s

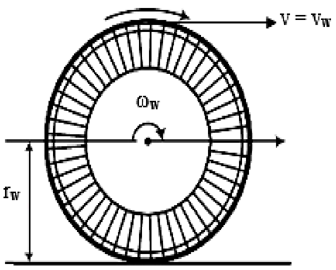


Figure 3. 6 Relation between V or V_w and ω_w at the driving wheels [37]

3.3.1.1 Maximum vehicle speed

In order to Determine the final drive Gear Ratio (GR_f), considering the gear ratio for the Top gear (GR_t) is 0.778:1 from table 3.2 while the engine is running at speed of ($N_{@Pb-max}$) 4000 rpm from table 4-1 in the Operating range and ckecked with the maximum vehicle speed.

$\frac{N}{V} = 2.65x\frac{GR_i xGR_f}{r_W}$ ………………………………………………………………..…….. (2)

$\Longrightarrow GR_f = \frac{N_{@Pb-max}\ x\ r_W}{2.65\ x\ V_{T-max}\ x\ GR_t}$

$$\Longrightarrow V_{max} = \frac{2\pi}{60} x r_w x \frac{3.6 N_{@Pbmax}}{GR_t}$$

$$\Longrightarrow V_{max} = \frac{2\pi}{60} x0.5842\ mx \frac{3.6x4000rpm}{0.778\ x\ 5.21838}$$

V_{max} = 130 km/h………. this is the max vehicle speed @ top gear

The changes of the wheel angular velocity (ω_w) depend on the values of the gear ratio of the gearbox (GR_i), the gear ratio of the final drive (GR_F). Hence,

$$\omega_w = \frac{\omega}{GR_i\ x\ GR_{fd}} \quad \text{……………………………………………………………………………… (3)}$$

Where; $\omega = \frac{2\ \pi\ x\ N}{60}$ = 418.879 rad/s

Maximum wheel speed (N_{w-max})

$$N_{w-max} = \frac{N_{w@Pb-max}}{GR_t} \quad \text{……………………………………………………………………… (4)}$$

$N_{w-max} = \frac{4000}{6.7765356}$ = 590.272 rpm

Maximum engine speed at maximum brake power ($N_{max@Pb-max}$)

$$N_{max@Pb-max} = N_{w-max}\ x\ GR_t \quad \text{………………………………………………………… (5)}$$

$N_{max@Pb-max}$ = 590.272 x 6.7765356 = 4000 rpm

Maximum engine speed (N_{max}): which occurs at $N_{max} \approx 1.2N_{@pb-max}$, when the inertia forces reach their maximum value while the gas pressure is comparatively lower. This point can be taken as the third design point (dp-3).

$$N_{max} = 1.2\ N_{max@Pb-max} \quad \text{……………………………………………………………………… (6)}$$

N_{max} = 1.2 x 4000 = 4800 rpm

The maximum vehicle velocity of a motor vehicle (v_{max}) is limited mainly by the maximum power available at the driving wheels(p_{w-max}). But in order to know about maximum power available at the driving wheels, first we must determine resistances to the vehicle motion.

3.3.1.2 The total resistances to vehicle motion

The major resistances to be considered in the design of a motor vehicle and/or an engine are:

1. Air Resistance (R_a), 2. Rolling Resistance (R_{rl}), 3. Gradient (Grade or Slope) Resistance (R_g).

1. **Air Resistance (R_a) for a Front Wheel Drive (FWD)**

Based on lecture courses of automotive engineering, the vehicle motion is resisted by air resistance. The air resistance can be calculated as:

$$R_a = \frac{1}{2} \rho_a C_D A_{PF} (V)^2 \text{…………………………………………………………………… (7)}$$

$R_a = \frac{1}{2} x1.05x0.3x0.8 \times 1.86x1.83x(36.11)^2$ = 559.2288 N

Where; ρ_a = local air density ($\frac{P_a}{Z \times R_a \times T_a}$ ≈1.05 kg/m^3)

C_D = Drag Coefficient (≈ 0.3)

V =. Vehicle Speed (36.11 m/s)

A_{PF} = Projected frontal area ($SF \times W \times H$)

Where; SF = Shape factor (≈ 0.8)

W = Vehicle width (m)

H = Vehicle height (m)

2. **Rolling Resistance (R_{rl}) for a Front Wheel Drive (FWD)**

The rolling resistance obsiously exists between the vehicle tire and the road, and can be calculated as:

$$R_r = \mu \times F_N = (0.015 + 0.00016 \times V) x\, W_{max} \quad \ldots\ldots\ldots (8)$$

R_r = (0.015+0.00016 x 130) x (2510 x 9.81) = 881.50698 N

Where; $F_N = 2F_{zl}$ is the normal force at the driving wheels (N)

μ = Friction Coefficient of $(0.015 + 0.00016 \times V)$

V = Vehicle speed (130 km/h)

W_{max} = Total vehicle weight (2510 kg)

3. **Gradient (Grade or Slope) Resistance (R_g) for a Front Wheel Drive (FWD)**

When the vehicle is going up to some inclination it gets resistance to motion is called dradient resistance. But in this research to attain the maximum vehicle speed, the vehicle must be front wheel drive. So, based on the above assumption I had taken the minimum gradient and can be calculated as:

$$R_g = W_{max} \times \sin\phi \quad \ldots\ldots\ldots (9)$$

R_g = 2510 x 9.81 x sin1.72 = 740 N

Where; ϕ = Minimum grade or slope (1°)

Therefore, The total vehicle resistance (R_t) When vehicle moves up a gradient is the sum of air, rolling and gradient resistances.

$$R_{t-max} = (R_a + R_r + R_{g-max}) \quad \ldots\ldots\ldots (10)$$

R_{t-max} = 2180.836 N

3.3.1.3 Resistance Power (P_{rq})

As shown in figure 3.7, the vehicle is running on the level road (i.e, ϕ ≈ 0-1) can attain the maximum resistance power using the following formula.

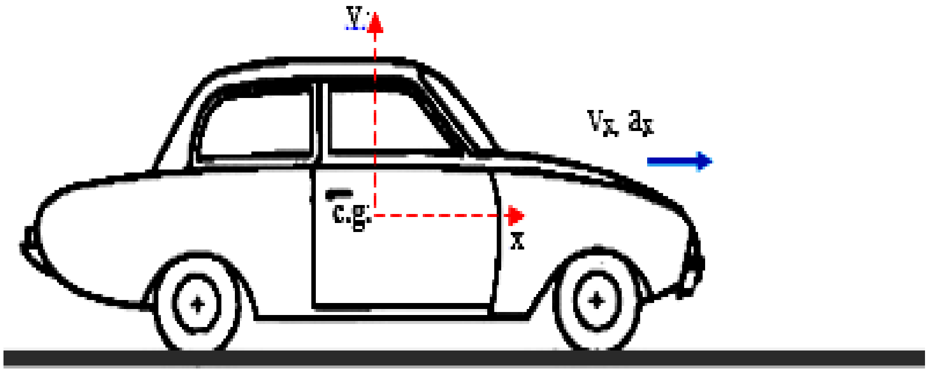


Figure 3. 7 The vehicle running on level road [37]

$P_{rq} = R_{t-max}$ x V_{max} .... (11)

P_{rq} = 2180.836 x 36.11 m/s = 78.75 kW

Where; V = vehicle speed (36.11 m/s).

The expected resistance Power (P_{rq}) of a FWD motor vehicle, accelerating from rest ($V = 0$) to a maximum velocity ($V = V_{max}$) on a level road (ϕ ≈ 0-1) in Top Gear (GR_t), can be plotted the graph using a computer Programming Language (C++) that were written below and/or Application Software's (MS-Excel) by table 4-3 as shown in Figure 4.7, below.

Table 3. 4 Speed vesus Resistance Power resulted from C++ # 1 Program.

V (km/h)	Pr (kW)
0	0
7.47	2.41
14.95	4.97
22.42	7.71
29.9	10.66
37.37	13.83
44.85	17.25
52.32	20.95
59.8	24.96
67.27	29.3
74.75	34
82.22	39.09
89.7	44.58
97.17	50.51
104.65	56.89
112.12	63.77
119.6	71.16
127.07	79.09
134.55	87.58
142.02	96.66
149.5	106.36

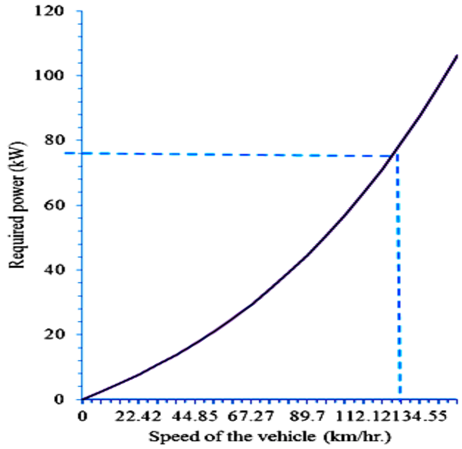


Figure 3. 8 Power required (P_{rq}) by a FWD motor vehicle on a level road.

The curve shows, the amount of power required (P_{rq}) to propel a Front Wheel Drive (FWD) motor vehicle on a level road. In order to design an engine for a maximum vehicle velocity (v_{max}) of, say, 130 km/h (as shown in the figure 3.8), and the maximum power available at the driving wheels (P_{w-max}) should be 78.75 kW.

On the other hand different books have shown, the maximum Power available at the driving wheels (P_{w-max}) also depends on the maximum engine brake power (P_{b-max}), the efficiencies of the gearbox (η_{Gi}) and the final drive (η_{fd}) is given by,

$$P_{w-max} = P_{b-max} \text{ x } \eta_{Gi} \text{ x } \eta_{fd} \quad \text{(12)}$$

But we have no data to know the efficiencies of gearbox. The product of gearbox efficiencies and final drive efficiency gives total efficiency.

Total efficiency

In principle (theoretically), the engine brake power (P_b) is expected to be equal to the power available at the driving wheels (P_w). But, due to the inefficiencies in the power train (driveline) due to problems associated with manufacturing processes, quality of materials, lubrication, (particularly, at the Gearbox and the Final drive). As a result, P_w is always less than P_b and both can be related by the total efficiency (η_t) as:

$$\eta_t = \eta_{Gi} \text{ x } \eta_{fd} = \frac{P_w}{P_b} \quad \text{(13)}$$

$$\eta_t = \frac{P_{w-max}}{P_{b-max}} = \frac{78.75 \text{ kw}}{105\text{kw}} = 75 \text{ \%}$$

3.3.1.4 Performance of IC Engines (Empirical Relations) For CI Engines with Open Combustion Chamber

As Meless H. [37], lecture notes the following Empirical Relations are obtained from experimental data of engine performance test using a curve fitting method used for analyzed brake power,frictional mean effective pressure, brake torque, brake mean effective pressure, indicated mean effective pressure, mechanical efficiency, brake specific fuel consumption, volumetric efficiency, brake thermal efficiency, indicated power and friction power.

Brake Power (P_b)

Maximum brake (Rated) Power (P_{b-max}): which occurs at $N_{@pb-max}$, when the gas pressure reaches its maximum value (P_{g-max}) and also inertia loads are comparatively higher. This point can be taken as the second design point (DP-1).

$$P_b = P_{b-max} \text{ x } \frac{N}{N_{@Pb-max}} \text{ x} \left[0.87 + \left(1.13 \text{ x} \frac{N}{N_{@Pb-max}}\right) - \left(\frac{N}{N_{@Pb-max}}\right)^2\right] \quad (14)$$

Let N = 1000 rpm

$$P_b = 105\text{x } \frac{1000}{4000} \text{x} \left[0.87 + \left(1.13 \text{ x} \frac{1000}{4000}\right) - \left(\frac{1000}{4000}\right)^2\right] = 28.6 \text{ kW}$$

Friction mean effective pressure (fmep)

$$fmep = [0.089 + (0.0118 \text{ x } V_{Pm})] \text{ x } 10 \quad (15)$$

$$fmep = [0.089 + (0.0118\text{x}3.13)]\text{x}10 = 1.259 \text{ bar}$$

Where; $V_{Pm} = \frac{2 \text{ x S x N}}{60} = \frac{2 \text{ x } 0.094 \text{ x } 1000}{60} = 3.13$ m/s

Brake Torque (T_b)

Maximum brake Torque (T_{b-max}): which occurs at $N_{@Tb-max} \approx$ (0.4 to 0.6) x $N_{@pb-max}$ and $N_{@Tb-max} \approx$ (0.5 to 0.8) x $N_{@pb-max}$, Diesel and Turbocharged respectively when the gas pressure reaches nearly its maximum value (P_{g-max}) while the inertia forces are comparatively lower. Even though the maximum cycle gas pressure (P_{g-max}) in the case of CI engines is attained at the point of P_{b-max} at $N_{@pb-max}$, the point of T_{b-max} or $bmep_{max}$ at $N_{@Tb-max}$ can be taken as a design point (DP-2) by considering a proper Factor of Safety (FS).

$$T_b = \frac{60 \text{ x } P_b}{2 \pi \text{ x N}} \text{ x } 1000 \quad (16)$$

$$T_b = \frac{60 \text{ x } 28.6 \text{ kW}}{2 \pi \text{ x } 1000 \text{ rpm}} \text{x}1000 = 273.1 \text{ Nm}$$

Brake mean effective pressure (bmep)

$$bmep = \frac{4\pi \text{ x } T_b}{E_c \text{ x } 10^5} \quad (17)$$

$$\text{bmep} = \frac{4\pi \text{ x } 273.1\text{Nm}}{0.002499\text{m}^3 \text{ x } 10^5} = 12.62 \text{ bar}$$

Indicated mean effective pressure (imep)

$$\text{imep} = \text{bmep} + \text{fmep} \quad \ldots \quad (18)$$

$$\text{imep} = 12.62 \text{ bar} + 1.259 \text{ bar} = 13.879 \text{ bar}$$

Mechanical Efficiency (η_m)

$$\eta_m = \frac{\text{bmep}}{\text{imep}} \text{x}100 \quad \ldots \quad (19)$$

$$\eta_m = \frac{12.62 \text{ bar}}{14.\text{bar}} \text{x}100 = 90 \ \%$$

Brake specific fuel consumption (bsfc)

$$\text{bsfc} = \text{bsfc}_{@Pb-max} \text{x} \left[1.55 - \left(1.55 \text{ x} \frac{N}{N_{@Pb-max}}\right) + \left(\frac{N}{N_{@Pb-max}}\right)^2\right] \quad \ldots \quad (20)$$

$$\text{bsfc} = 210\text{x}\left[1.55 - \left(1.55 \text{ x} \frac{1000}{4000}\right) + \left(\frac{1000}{4000}\right)^2\right] = 231 \text{ g/kWh}$$

Volumetric Efficiency (η_v)

$$\eta_v = \frac{m_a}{(\rho_i \text{ x } v_s \text{ x ncps})} \quad \ldots \quad (21)$$

$$\eta_v = \frac{0.1133125}{(0.010896 \text{ x } 0.00062475 \text{ x } 2)} = 83.23 \ \%$$

Where; $m_a = m_f \text{ x AF} = 0.1133125\text{kg/s}$

$$\rho_a = \frac{P_a}{(Z_a \text{ x R x } T_a)} = 0.010896\text{kg/m}^3$$

$$v_s = 0.00062475 \text{ m}^3$$

Brake thermal Efficiency ($\eta_{b.th}$)

$$\eta_{b.th} = \frac{P_b}{\dot{m}_f \text{ x } Q_{LHV}} \text{x } 100 \quad \ldots \quad (22)$$

$$\eta_{b.th} = 11 \ \%$$

Where; $m_f = \text{bsfc x } P_b = 0.231 \text{ x } 28.6 = 6.6$

Indicated Power (P_i)

$$P_i = \frac{P_b}{\eta_m} \quad \ldots \quad (23)$$

$$P_i = \frac{28.6 \text{ kW}}{0.90} = 31.727 \text{ kW}$$

Friction Power (P_f)

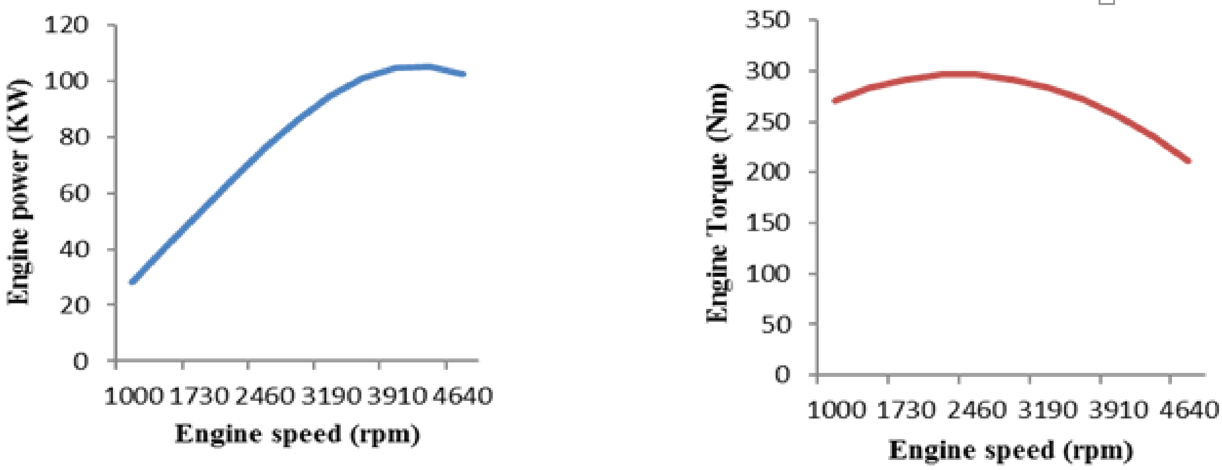
$$P_f = P_i - P_b \quad \ldots \quad (24)$$

$$P_f = 31.727 - 28.6 = 3.1274 \text{ kw}$$

Using a computer Programming Language (C++) and Application Software's (MS-Excel), the expected Engine Performance Curves (Characteristic curve) of R425DOHC diesel engine can be plotted from table 3.5 as shown in Figure 3.9.

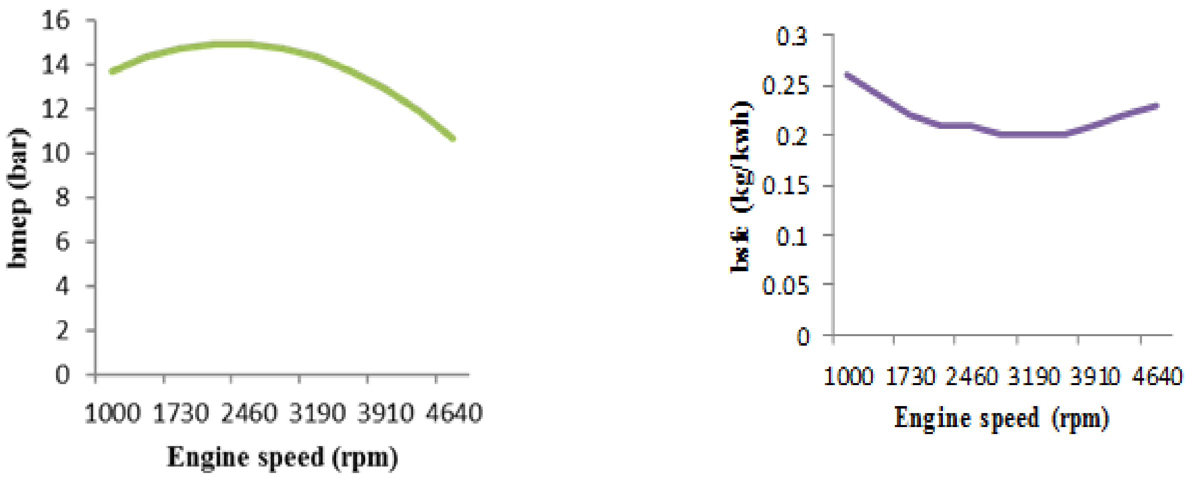
Table 3. 5 Characteristic value of Engine Performance at Different Engine Speed resulted from C++ # 1.

Pb (kW)	Tb (Nm)	bmep (bar)	bsfc (kg/kWh)	ETA_m (%)	N (rpm)
28.6	273.22	13.73	0.26	92.15	1000
40.63	285.41	14.35	0.24	91.89	1.36E+03
52.86	293.53	14.75	0.22	91.47	1.72E+03
64.81	297.6	14.96	0.21	90.89	2.08E+03
76.03	297.6	14.96	0.21	90.15	2.44E+03
86.06	293.55	14.75	0.2	89.22	2.80E+03
94.44	285.42	14.35	0.2	88.07	3.16E+03
100.71	273.24	13.73	0.2	86.64	3.52E+03
104.42	257	12.92	0.21	84.85	3.88E+03
105.09	236.69	11.9	0.22	82.6	4.24E+03
102.28	212.32	10.67	0.23	79.69	4.60E+03



a)

b)



c) d)

Figure 3. 9 Performance Curves a) brake power, b) torque power, c) break mean effective pressure and d) break specific fuel consumption of IC Engines at Full Load.

Engine Size from C++ program #1

$E_C = 2498.799$ cc	$V_t = 662.56$ cc	$r_{cr} = 48.14$ mm
$V_s = 624.7$ cc	$D = 94.23$ mm	$l_{con} = 192.55$ mm
$V_{cl} = 37.86$ cc	$S = 96.27$ mm	

3.3.1.5 Calculation of basic Geometrical Properties of the Engine

As shown in figure 4.9, basic geometrical properties of the engine such as swept volume (**V_s**), Bore (D), Stroke (S), Crank radius (r_{cr}) and Connecting rod length (l_{con}) are calculated.

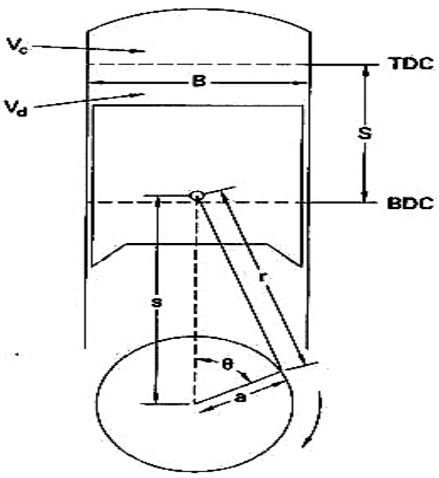


Figure 3. 10 Geometrical Properties of IC Engines [37]

Table 3. 6 Engine Size

From technical specification	
Engine Cap (E_c)	2.499 lit
Bore	92 mm
stroke	94 mm

Under square engine (D<S): If stroke length (S) is greater than the bore diameter (D). Smaller diameter cylinder with shorter flame travel distance; Combustion chamber surface area will be smaller resulting less heat loss per cycle.

Swept Volume (V_s),

$$E_c = V_s \ x \, Z \ldots\ldots (25)$$

$$\Longrightarrow V_s = \frac{E_c}{z} = \frac{2499 \ cc}{4} = 624.75cc = 0.00062475m^3$$

Bore (D) and Stroke (S),

$$V_s = \frac{\pi D^2}{4} \ x \, S \ldots\ldots (26)$$

$$\Longrightarrow D = \left(\frac{4 \ x \ V_s}{\pi \ x \ ratio_{SD}}\right)^{1/3} = \left(\frac{4 \ x \ 624.75 \ mm}{\pi \ x \ 1.0217391}\right)^{1/3} = 91.955 \approx 92 \text{ mm}$$

$$\Longrightarrow s = ratio_{SD} \ x \, D = 1.0217391 \text{ x } 92 \text{ mm} = 93.9 \approx 94 \text{ mm}$$

Crank radius (*r_{cr}*),

$$r_{cr} = \frac{S}{2} = \frac{94 \ mm}{2} = 47 \text{ mm}$$

Connecting rod length (l_{con}),

$$l_{con} = \frac{r_{cr}}{\lambda} \ldots\ldots (27)$$

$$l_{con} = r_{cr} \ x \ Z_c = 47mm \ x \ 4 = 188 \text{ mm.}$$

Where; λ = crank radius (r_{cr}) to con-rod length (l_{con}) ratio

3.3.1.6 Kinematics of IC Engines

Figure 4.10 shows Piston displacement x_p in m, piston velocity v_p in m/s and piston acceleration a_p in m/s^2 against or versus crank angle θ for engine speeds of$N_{@Tb-max}$, $N_{@Pb-max}$, and $N_{@max}$ are calculated using C++ progrming by inserting inserting equation 27, 28 and 30 in the program respectively.

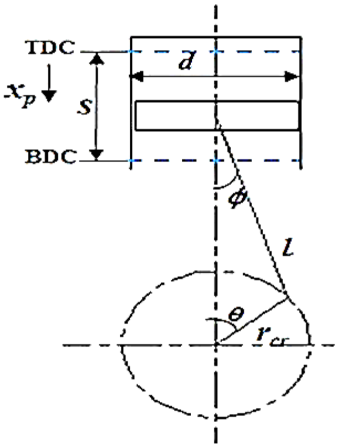


Figure 3. 11 Engine Kinematics

Where; ϕ = angle between the center line of the cylinder and con-rod.

θ = Crank angle.

Piston displacement (x_p)

$$x_p = r_{cr}\left((1-\cos\theta) + \frac{\lambda \sin^2\theta}{2}\right) \quad (28)$$

Where; x_p, r_{cr} Are in m

Piston velocity (v_p)

$$v_p = \omega\, r_{cr}\left(\sin\theta + \frac{\lambda \sin 2\theta}{2}\right) \quad (29)$$

Where; v_p in m/s.

The design (maximum) mean piston velocity (V_{pm-max}) can be calculated as,

$$V_{pm-max} = \frac{2 \times S \times N_{max}}{60} \approx \frac{V_{pmax}}{1.6} \quad (30)$$

$$V_{pm-max} = \frac{2 \times 0.094 \times 4800}{60} = 15.04 \text{ m/s}$$

From this relation the design (maximum) mean piston velocity (V_{pm-max}) can be determined. The expected design (maximum) mean piston velocity will be in the range of 5 to 16 m/s. The main reasons why engines should operate in this range are:

- due to limitation in material strength,
- Due to drop in engine performance as a result of drop in volumetric efficiency at higher engine speed operation. The drop in volumetric efficiency causes poor charging and discharging processes and also poor combustion.
- Due to continuous increment of friction power (P_{fr}) with engine speed. This may cause a considerable drop in mechanical efficiency.

It is mainly on these reasons that the maximum engine speed of IC engines are limited to in the range of $\approx$ 2100 to 4000 rpm from design data book as shown in table 4-1.

Piston acceleration (α_p)

$$\alpha_p = \omega^2 r_{cr}(\cos\theta + \lambda \cos 2\theta) \quad \text{.......} \quad (31)$$

Where; α_p in m/s^2.

Engine Kinematics from the C++ program ## the value engine displacement, velocity and acceleration are: xp = 0.91 mm, vp = 4.36 m/s and ap = 10300 m/s*s and the graghs are shown in figure 4.11.

Table 3. 7 Xp, Vp and αp results against crank angle from C++# 2.

theta	Xp	Vp	αp
Degree	(m)	(m/s)	(m/s*s)
0	0	0	51.489
10	0.18	4.256	50.243
20	0.7	8.307	46.6
30	1.55	11.964	40.832
40	2.68	15.065	33.359
50	4.04	17.491	24.712
60	5.58	19.167	15.475
70	7.22	20.07	6.232
80	8.9	20.22	-2.49
90	10.57	19.681	-10.265
100	12.16	18.544	-16.8
110	13.64	16.92	-21.952
120	14.97	14.924	-25.726
130	16.12	12.667	-28.252
140	17.08	10.244	-29.758
150	17.83	7.728	-30.52
160	18.37	5.17	-30.817
170	18.69	2.598	-30.888
180	18.8	0.024	-30.893

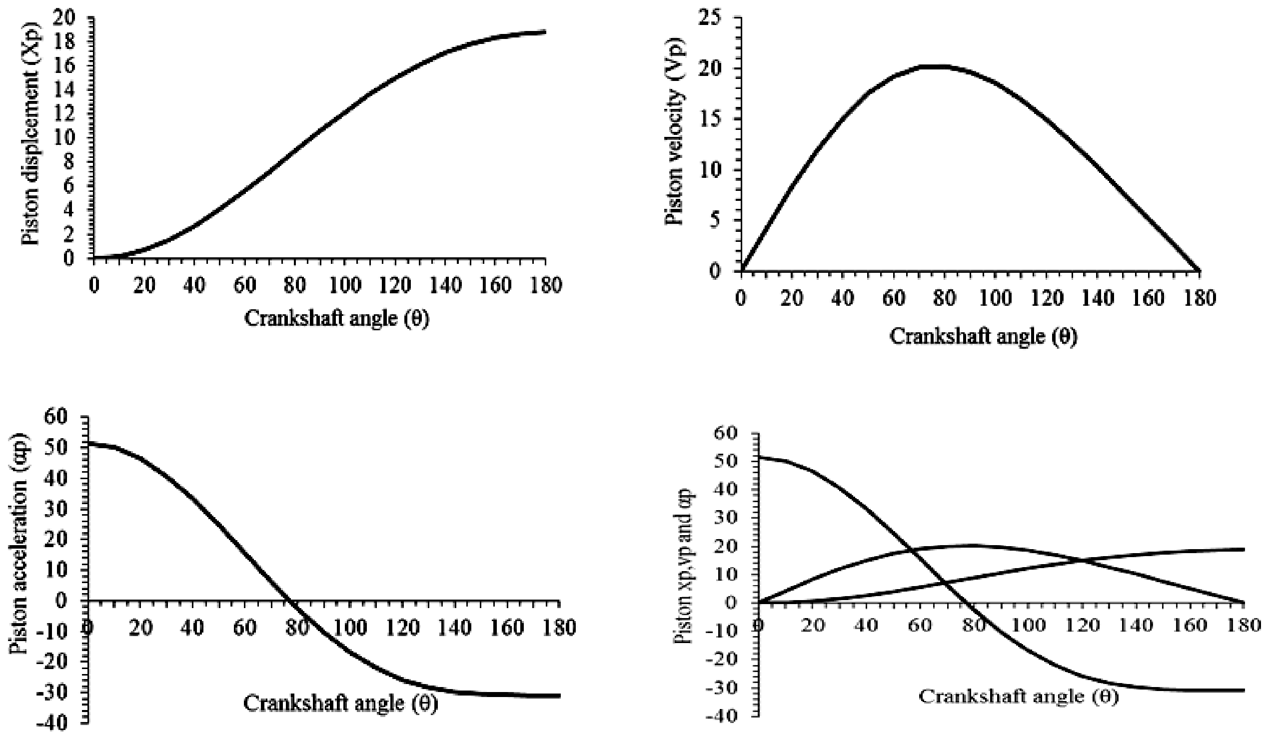


Figure 3. 12 Engine kinematics of crank gear mechanism curve.

3.3.1.7 Engine Dynamics (Kinetics)

The various forces acting on the component parts of a crank gear mechanism as shown in figure 3.13 are: 1. Force due to gas pressure (F_g)

2. Force due to inertia of reciprocating parts (F_i)
3. Force due to the weight of reciprocating parts (W_r)
4. Force due to friction (F_f)
5. Torque or Turning Moment (T).

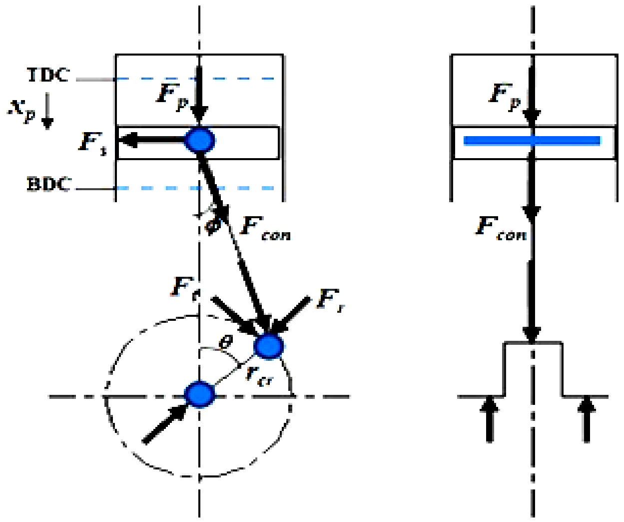


Figure 3. 13 Free Body Diagram (FBD-1) of Crank Gear Mechanism during Power Stroke.

Piston head gas Force (F_{g-max}),

$$F_{g-max} = P_{g-max} \text{ x } A_p \quad \text{.......... (32)}$$

$F_{g-max} = \frac{\pi}{4} \text{ x } (0.092)^2 \text{ x } 107.8 \text{ x } 10^5 = 71.661$ kN

Where; A_p = Piston head area

P_{g-max} = Maximum cylinder gas pressure during the power stroke

Reciprocating parts inertia force (F_i),

$$F_i = m_r \text{ x } a_p \quad \text{.......... (33)}$$

$F_i = 1.5 \text{ x } 10308.25349 = 15.4623$ kN

Where; m_r = reciprocating parts mass (≈ 1.5 kg). This includes the masses of piston, Piston rings, piston pin and 1/3rd mass of connecting rod.

a_p = Piston acceleration

$$a_p = \left(\frac{2\pi N}{60}\right)^2 \text{ x } r_{cr} \left(\cos\theta + \lambda\cos 2\theta\right) = 10308.25349 \text{ m/s}^2$$

Reciprocating parts weight (W_r),

$$W_r = m_r \text{ x } g \quad \text{.......... (34)}$$

$W_r = 1.5 \text{ kg x } 9.81 \text{m/s}^2 = 14.715$ N

Where; g = gravitational acceleration

As shown on the 'Free Body Diagram (FBD-1)' of CGM in Figure 4.10, the net force acting on the piston (F_p) during the 'Power Stroke' for a vertical, in-line engine is given by:

$$F_p = F_g \pm F_i + W_r - F_f \quad \text{.......... (35)}$$

Neglecting the effects of W_r and F_f, the following relations can be obtained:

Piston force (F_p),

$$F_p = F_g \pm F_i \quad \text{.......... (36)}$$

$F_p = 56.1987$ kN or 59.6977 kN

Side thrust(F_s),

$$F_s = F_p \text{ x } \tan\phi \quad \text{.......... (37)}$$

$F_s = 56.1987 \text{ kN x } \tan 2.488 = 2.44$ kN

Frictional force (F_f),

$$F_f = \mu_f \text{ x } F_s \quad \text{.......... (38)}$$

$F_f = 0.1 \text{ x } 2.44 = 0.244$ kN

Where; μ_f = friction coefficient (0.1)

F_s = Side (trust) force

Force along the connecting rod axis(F_{con}),

$$F_{con} = \frac{F_p}{\cos\phi} \quad \ldots\ldots (39)$$

$$F_{con} = \frac{56.1987}{\cos(2.488)} = 56.25 \text{ kN}$$

Radial force acting along the crank web (F_r)

$$F_r = \frac{F_p \text{ x } \cos(\theta+\phi)}{\cos\phi} \quad \ldots\ldots (40)$$

$$F_r = \frac{56.1987 \text{ x } \cos(10+2.488)}{\cos(2.488)} = 54.869 \text{ kN}$$

Tangential force acting on the crank pin(F_t),

$$F_t = \frac{F_p \text{ x } \sin(\theta+\phi)}{\cos\phi} \quad \ldots\ldots (41)$$

$$F_t = \frac{56.1987 \text{ x } \sin(10+2.488)}{\cos(2.488)} = 12.1636 \text{ kN}$$

Turning moment (torque) (T),

$$T = F_t \text{ x } r_{cr} \quad \ldots\ldots (42)$$

$$T = 12.1636 \text{ x } 0.047 = 571.6892 \text{ Nm}$$

P_g Can be solved from the indicated mean effective pressure equation of dual cycle

$$\text{imep} = \frac{p1[r^{\gamma} r_p \gamma(r_c-1)+r^{\gamma}(r_p-1)-r^{1-\gamma}(r_p r^2{}_c-1)]}{(\gamma-1)(r-1)} \quad \ldots\ldots (43)$$

Where; r_p = pressure ratio

r_c = Cut-off ratio

But the equation 1.1 contains two unknown, so to solve this equation I had used C++ programing. From A3-CI C++ program # 3 the data below shows are the result of pressure ratio, cut-off ratio and indicate mean effective pressure (r_p = 1.4, r_c = 2.4 and imep = 14.01 bar)

Table 3. 8 Cylinder volume, Cylinder pressure and Temperature value from C++ # 3.

Theta (deg.)	V (cc)	P (bar)	T (K)
40	133.7	0.9	313
50	182.34	0.9	313
60	237.1	0.9	313
70	295.69	0.9	313

80	355.83	0.9	313
90	415.36	0.9	313
100	472.37	0.9	313
110	525.22	0.9	313
120	572.66	0.9	313
130	613.72	0.9	313
140	647.8	0.9	313
150	674.5	0.9	313
160	693.64	0.9	313
170	705.14	0.9	313
180	708.97	0.9	313
190	705.14	0.91	313.59
200	693.64	0.93	315.4
210	674.5	0.96	318.51
220	647.8	1.02	323.04
230	613.72	1.09	329.21
240	572.65	1.2	337.29
250	525.22	1.34	346.15
260	472.36	1.54	357.34
270	415.36	1.82	371.4
280	355.83	2.23	389.04
290	295.69	2.84	411.26
300	237.1	3.78	439.43
310	182.34	5.32	475.45
320	133.7	7.96	521.84
330	93.3	12.7	581.3
340	63	21.16	653.97
350	44.23	33.52	727.23
360	37.86	41.02	761.91
360	37.86	107.8	2.00E+03
370	44.23	107.8	2.34E+03
380	63.01	68.05	2.10E+03

390	93.31	40.84	1.87E+03
400	133.7	25.59	1.68E+03
410	182.35	17.09	1.53E+03
420	237.1	12.15	1.41E+03
430	295.7	9.12	1.32E+03
440	355.84	7.17	1.25E+03
450	415.37	5.86	1.19E+03
460	472.37	4.96	1.15E+03
470	525.23	4.32	1.11E+03
480	572.66	3.86	1.08E+03
490	613.73	3.53	1.06E+03
500	647.8	3.29	1.05E+03
510	674.51	3.12	1.03E+03
520	693.65	3.01	1.02E+03
530	705.14	2.95	1019.11
540	708.97	2.93	1017.46

From table 3.8, we can observe the maximum cylinder gas pressure is occurred at crank angle between from 360 to 370 degree. i.e, P_{g-max} = 107.8 bar. The graph of each parameter are plotted as shown in figure 4.13, below.

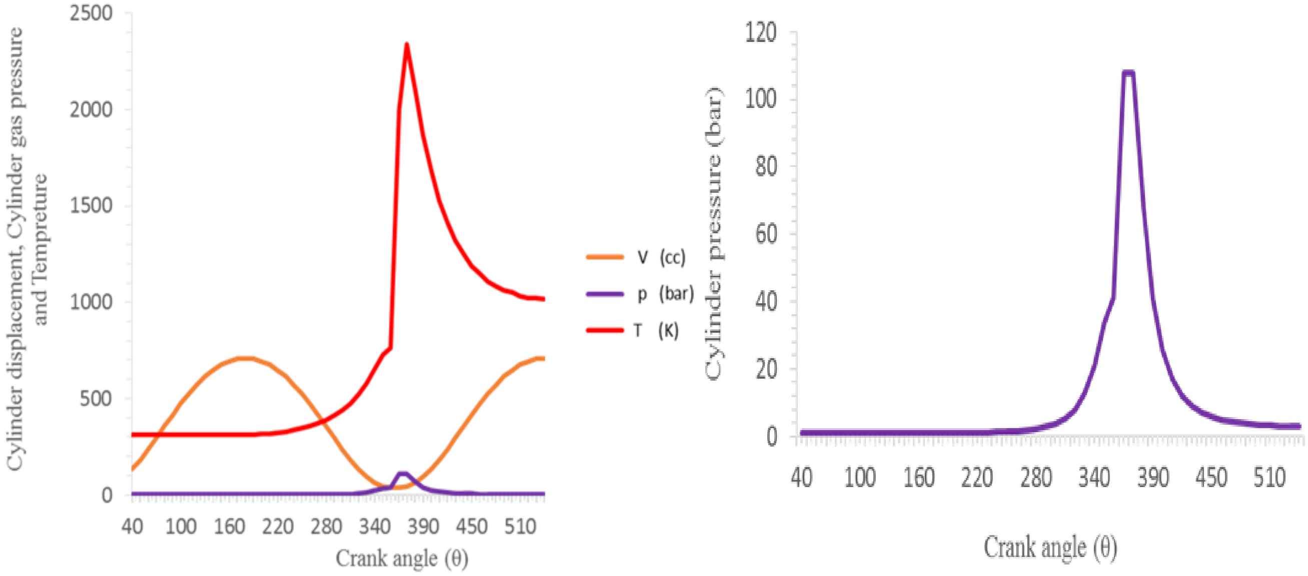


Figure 3. 14 Cylinder displacement, gas pressure and temperature versus crank angle curve.

The following C++ programing have prepared for kinetics of engine crank gear mechanism.

From A4-CI C++ program #### The data below shows Engine Dynamics (Kinetics) or the various forces acting on the component parts of a crank gear mechanism as a function of crank angle θ.

Engine Kinetics

Fg = 75.16 kN	Fs = -0.14 kN	Ft = 10.23 kN
Fi = 15.45 kN	Fcr = 59.71 kN	T = 492.27 Nm
Fp = 59.71 kN	Fr = 58.82 kN	

3.3.2 Design of Valve Gear Mechanism

Before designing the component parts of the valve gear mechanism, the following have to be determined:

- Direction of Engine Rotation (CW or CCW when viewed from the front side of the engine).
- Type of drive (Gear, Chain, Toothed Belt).
- Numbering of Cylinders.
- Firing Order.
- Valve Timing Diagram.

While designing the valve gear mechanism, two important requirements must be fulfilled:

i) Obtaining maximum passage to provide good filling and cleaning of the engine cylinder.
ii) Minimizing the mass of rotating and reciprocating parts to reduce inertia stresses.

The main function of the valve gear mechanism of a reciprocating I.C. Engine is, to actuate (open and close) the inlet and exhaust valves at the required time with respect to the position of the crankshaft (θ) or the position. Air alone for CI engines is admitted to the engine cylinder through the inlet port (inlet valve) and the burned gases are escaped through the exhaust port (exhaust valve). In vertical IC engines, the cam (on the camshaft) opens the valve by pushing through a follower, a push rod, a tappet and a rocker arm. The valve is closed by the valve spring. In doing so, power is transmitted from the crankshaft to the camshaft by means of a gear or a chain or a toothed belt drive.

The fuel is admitted to the engine by the inlet valve and the burnt gases are escaped through the exhaust valve. In vertical engines, the cam moving on the rotating camshaft pushes the cam follower and push rod upwards, thereby transmitting the cam action to rocker arm. The camshaft is rotated by the toothed belt from the crankshaft. The rocker arm is pivoted at its center by a fulcrum pin. When one end of the rocker arm is pushed up by the push rod, the other end moves downward. This pushes down the valve stem causing the valve to move down, thereby opening the port. When the cam follower moves over the circular portion of the cam, the pushing action of the rocker arm on the valve is released and the valve returns to its seat and closes it by the action of the valve spring.

For a Simple Harmonic Motion (SHM), the cam displacement (y), velocity (y'), acceleration (y'') and jerk (y'') can be derived as in the following formulas and the graph is shown in figure 3.15 below.

$$y = \frac{h_f}{2}\left(1 - \cos\frac{\pi\theta}{\beta}\right) \quad (44)$$

$$y' = \frac{h_f\pi}{2\beta}\left(\sin\frac{\pi\theta}{\beta}\right) \quad (45)$$

$$y'' = \frac{h_f\pi^2}{2\beta^2}\left(\cos\frac{\pi\theta}{\beta}\right) \quad (46)$$

$$y''' = \frac{h_f\pi^3}{2\beta^3}\left(\sin\frac{\pi\theta}{\beta}\right) \quad (47)$$

Where; $\theta = \theta_{dvo}$ cam angle

$\beta =$ Cam angle for maximum follower rise (h_f)

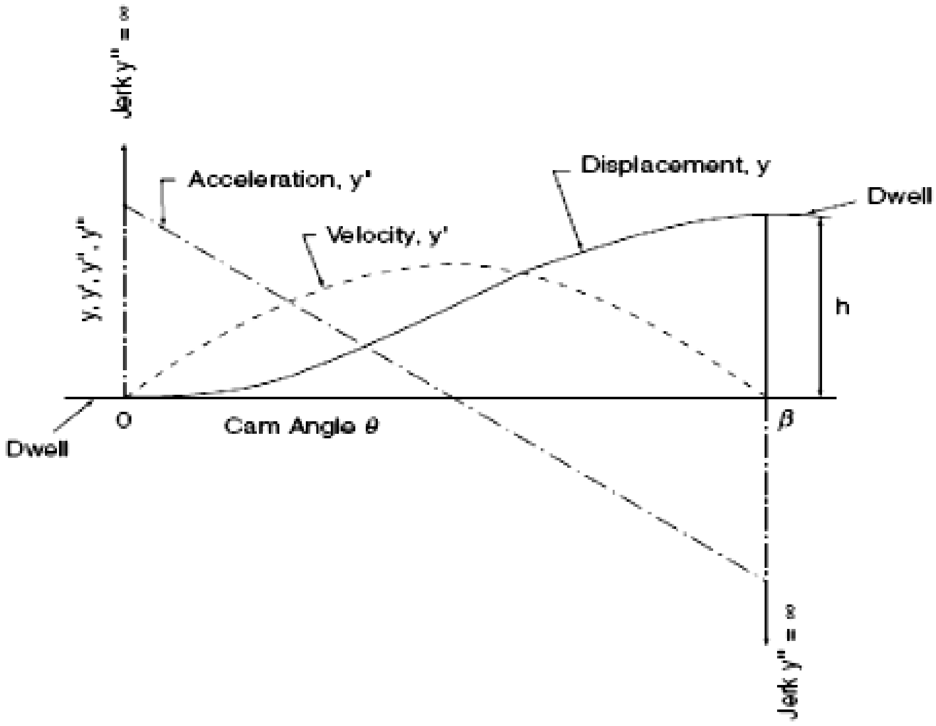


Figure 3. 15 The displacement, velocity, acceleration and jerk curves a Cam of a SHM [37]

3.3.3 Material selection and size determination

I determine the size of the engine selected parts for to be designed is by conducting measurement that are shown in figure 4.4 from actual SUV engine and by calculating from the relation of diesel engine baesd on maximum cylinder pressure that was generated from C++ program # 3 (A3-CI). Finally comparation were discussed in chapter six within allowable range.

3.3.3.1 Cylinder and Cylinder Liner

Function: The function of a cylinder is to retain the working fluid (fuel-air mixture) and to guide the piston.

Construction: Since the cylinder has to cylinder withstand high temperature due to the combustion of fuel, therefore, some arrangement must be provided to cool the cylinder. The multi-cylinder engines (such as of cars) are provided with water jackets around the cylinders to cool it. In smaller engines the cylinder, water jacket and the frame are made as one piece, but for all the larger engines, these parts are manufactured separately. The cylinders are provided with cylinder liners so that in case of wear, they can be easily replaced and with lower cost. Cylinder liners are of two types: Dry liner and Wet liner. The wet cylinder liner is shown in Figure 3.16, below.

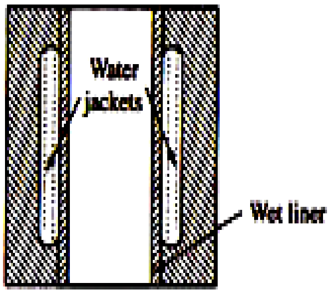


Figure 3. 16 Wet liners

Design consideration for cylinder liner: Since the cylinder has to withstand high combustion temperature, pressure (gas force) and side thrust, the design has to consider thermal and mechanical loads. In order to meet these requirements, the cylinder (cylinder block) has to possess:

- Adequate strength to withstand mechanical loads (F_{g-max} andF_s).
- Proper coefficient of thermal expansion and thermal conductivity so as to avoid seizure of the piston and to transfer the required amount of heat to the coolant in the water jacket faster.

Material and Manufacturing processes: The cylinders are usually made of cast iron or cast steel. The cylinder liners are made from good quality close grained cast iron (i.e. pearlitic cast iron), nickel cast iron, and nickel chromium cast iron. The inner surface of the liner should be properly heat treated in order to obtain a hard surface to reduce wear. The types of manufacturing processes are: casting, boring, honing, drilling, grinding, etc.

Size determination of cylinder liner

Thickness of the Cylinder Wall (t): The design of cylinder mainly takes into consideration the maximum thermal and mechanical loads due to maximum combustion temperature and pressure. The piston side thrust (F_s) also has to be considered as it exerts a resisting force (friction) on the wall of the cylinder and causes wear. The maximum gas force (F_{g-max}), as shown in figure 3.17, below which imposes mechanical load on the cylinder, produces the following two types of stresses as shown in figure 3.18 (a) and 3.19(b) respectively.

a) Longitudinal stress (σ_l), and
b) Circumferential or Hoop stress (σ_c).

Since these two stresses act at right angle to each other, the net stress in each direction may reduce. The longitudinal stress (σ_l), using a thin cylindrical wall formula, is given by:

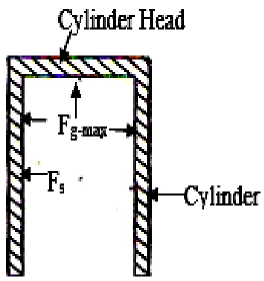


Figure 3. 17 Forces or Loads acting on the Cylinder and Cylinder Head

$$\sigma_l = \frac{\text{Force}}{\text{Area}} = \frac{(\pi/4) \text{ x } D^2 \text{ x } P_{g-max}}{\frac{\pi}{4}[(Do)^2 - D^2]} = \frac{D^2 \text{ x } P_{g-max}}{(Do)^2 - D^2} \quad \text{.......... (48)}$$

$$\sigma_l = \frac{(\pi/4) \text{ x } (92 \text{ mm})^2 \text{ x } 107.8 \text{ x } 10^5 \text{ N/mm}^2}{\pi/4[(102 \text{ mm})^2 - (92 \text{ mm})^2]} = \frac{(92 \text{ mm})^2 \text{x } 107.8 \text{ x}10^5 \text{ N/mm}^2}{(102 \text{ mm})^2 - (92 \text{ mm})^2} = 47 \text{ MPa}$$

Where; D = cylinder bore diameter (mm)

Do = Outer cylinder bore diameter (mm)

P_{g-max} = Maximum cylinder gas pressure (N/mm^2)

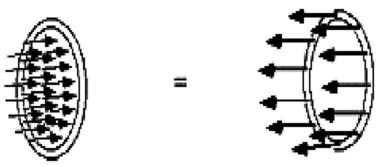


Figure 3. 18 Longitudinal Stress (σ_l) [37]

The circumferential or hoop stress (σ_c), using a thin cylindrical wall formula, is given by:

$$\sigma_c = \frac{\text{Force}}{\text{Area}} = \frac{D \text{ x } l \text{ x } P_{g-max}}{2t \text{ x } L} = \frac{D \text{ x } P_{g-max}}{2t} \quad \text{.......... (49)}$$

$$\sigma_c = \frac{92 \text{ mm x } 107.8 \text{ x } 10^5 \text{ N/mm}^2}{2 \text{ x } 10.224 \text{ mm}} = 48.5 \text{ MPa}$$

Where; L = Cylinder length

t = Cylinder wall thickness

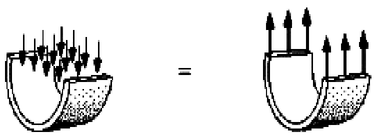


Figure 3. 19 Circumferential Stress (σ_c) [37]

Net longitudinal stress $= \sigma_l - \frac{\sigma_c}{m} = 47 - \frac{48.5}{4} = 34.875$ MPa

Net circumferential stress $= \sigma_c - \frac{\sigma_l}{m} = 48.5 - \frac{47}{4} = 36.75$ MPa

From the above two relations, $\sigma_c = 2\sigma_l$. As the circumferential or hoop stress (σ_c) has a higher value than the longitudinal stress (σ_l). Hence, the thickness of the cylinder wall (t) should be determined using a thin cylindrical wall formula derived for a circumferential stress.

$$t = \frac{P_{g-max} \text{ x } D}{2\,\sigma_c} + C \quad \text{...... (50)}$$

$$t = \frac{107.8 \text{ x } 10^5 \text{ N/mm}^2 \text{ x } 92 \text{ mm}}{2 \text{ x } 48.5 \text{ MPa}} + 2.112 = 12.336 \text{ mm}$$

Where; $\sigma_c =$ Permissible circumferential for the cylinder material (35 MPa to 100 MPa)

C = Allowance for re-boring and by interpolation it's calculated value (2.112 mm)

Table 3. 9 Table Allowance for Re-boring for I.C Engine Cylinders

D (mm)	75	100	150	200	250	300	350	400	450	500
C (mm)	1.5	2.4	4.0	6.3	8.0	9.5	11.0	12.5	12.5	12.5

Bore and length of the cylinder: the bore (i.e. inner diameter) and length of the cylinder may be determined as the power produced inside the engine cylinder, i.e. indicated power,

$$P_i = \frac{\text{imep x S x A x n}}{60} \text{ (Watts)} \quad \text{...... (51)}$$

$$P_i = \frac{P_{b-max}}{\eta m} \quad \text{...... (52)}$$

$$P_i = \frac{105\text{kW}}{0.90} = 116.66 \text{ kW}$$

Where; P_{b-max} = maximum brake power (kW)

η_m = Mechanical efficiency

imep = Indicated mean effective pressure (N/mm^2)

S = stroke length (m)

A = cylinder Cross-sectional area (mm^2)

n = Number of working strokes per min (N/2)

Where; N = Engine speed (rpm)

From the above equations (51 and 52), $S \times A = 2.499857 \times 10^{-7} mm^3$.

And from this expression, the bore (D) and length of stroke (S) can be determined. Since there is a clearance on both sides of the cylinder, therefore length of the cylinder is taken as 15 percent greater than the length of stroke. In other words, Length of the cylinder, L = 0.15 x Length of stroke = 0.15 x 94 = 108.1 mm

Cylinder flange and studs: The cylinders are cast integral with the upper half of the crankcase or they are attached to the crankcase by means of a flange with studs or bolts and nuts. The cylinder flange is integral with the cylinder and should be made thicker than the cylinder wall. The flange thickness (t_f) should be taken as, t_f = 1.2t to 1.4t = 14.8 mm to 17.27 mm, take t_f = 16 mm

Where; t = cylinder wall thickness (12.336 mm)

The diameter of the studs or bolts may be obtained by equating the gas load due to the maximum pressure in the cylinder to the resisting force offered by all the studs or bolts. Mathematically,

$$\frac{\pi}{4} \times D^2 \times P_{g-max} = n_s \times \frac{\pi}{4} (d_c)^2 \sigma_t \quad \ldots\ldots (53)$$

$$\Rightarrow \sigma_t = \frac{\frac{\pi}{4} \times (92)^2 \times 107.8 \times 10^5}{6 \times \frac{\pi}{4} (15)^2} = 67.585 \text{ MPa}$$

Where; n_s = Number of studs (0.01 D + 4 to 0.02 D + 4)

σ_t = Allowable tensile stress (35 to 70 MPa)

d_c = Core or minor diameter (0.84d mm)

Where; d = stud or bolt nominal or major diameter (d = 0.75 t_f to t_f = 12 to 16 mm), Take d =16 mm

Where; t_f = the thickness of flange.

In no case a stud or bolt less than 16 mm diameter should be used. The distance of the flange from the center of the hole for the stud or bolt should not be less than d + 6 mm (21 mm) and not more than1.5 d (22.5 mm). In order to make a leak proof joint, the pitch of the studs or bolts should lie between $19\sqrt{d}$ = 73.6 mm to $28.5\sqrt{d}$ = 110.38 mm,

Cylinder head: Usually a separate cylinder head or cover is provided with most of the engines. It is usually made of box type section of considerable depth to accommodate ports for air and gas passages, inlet valve, exhaust valve and atomize at the center of the cover (in case of diesel engines). The cylinder head may be approximately taken as a flat circular plate whose thickness (t_h) may be determined from the following relation:

$$t_h = D \text{ x } \sqrt{\frac{C \text{ x } P_{g-max}}{\sigma_c}} \quad \text{……………………………………………………………….………………} \quad (54)$$

$$t_h = 92\text{mm x } \sqrt{\frac{0.1 \text{ x } 107.8 \text{ x } 10^5 \text{ N/mm}^2}{48.5 \text{ MPa}}} = 13.7 \text{ mm}$$

Where; t_h = cylinder head thickness (mm)

C = Constant, its value is taken as 0.1.

σ_c = Allowable circumferential stress (30 to 50 MPa)

The studs or bolts are screwed up tightly along with a metal gasket or asbestos packing to provide a leak proof joint between the cylinder and cylinder head. The tightness of the joint also depends upon the pitch of the bolts or studs, which should lie between $19\sqrt{d}$ (73.6 mm) to $28.5\sqrt{d}$ (110.38 mm). The pitch circle diameter (D_P) is usually taken as, D_p = D + 3d = 137 mm.

Other dimensions can be determined using the following empirical relations.

Thickness of the water jacket wall (t_{wjw}), t_{wjw} = 0.032D + 1.6 mm = 4.544 mm

Width of the water jacket (t_{wj}), t_{wj} = 0.08D + 6.5 mm = 13.86 mm

Total length of the cylinder (l_{cyl}), l_{cyl} = v_{cl} +S + 35 mm = 166.86 mm

Where, v_{cl} = clearance volume length (37.863636 mm)

The size of clearance volume has to be determined by considering a permissible compression ratio (r). After anlysis the size, model was drawn by solidwork as shown in 3.20 below.

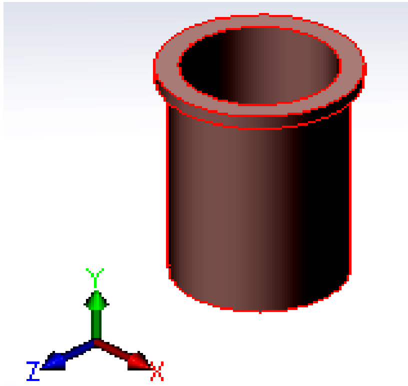


Figure 3. 20 Three Dimensional Sketch of Cylinder Liner

3.3.3.2 Piston

Function: The piston is a disc which reciprocates within a cylinder. It is either moved by the fluid or it moves the fluid which enters the cylinder. The main function of the piston of an internal combustion engine is to receive the thermal energy (gas pressure) from the expanding gas and transmit this energy (force and motion) to the crankshaft through the connecting rod. In addition, it also transmits a small portion of thermal energy (heat) to the cylinder wall.

Construction: The piston is constructed to possess piston head, ring grooves, piston pin hole, boss and skirt. The ring grooves houses the piston rings (compression and oil control rings); the piston pin hole with boss supports the piston pin; the skirt act as a bearing for the side thrust of the connecting rod on the walls of cylinder. The piston head or crown may be of flat, convex or concave depending upon the design of combustion chamber. In this design the piston is made concave shape. Each construction of piston is shown in figure 3.21 below clearly.

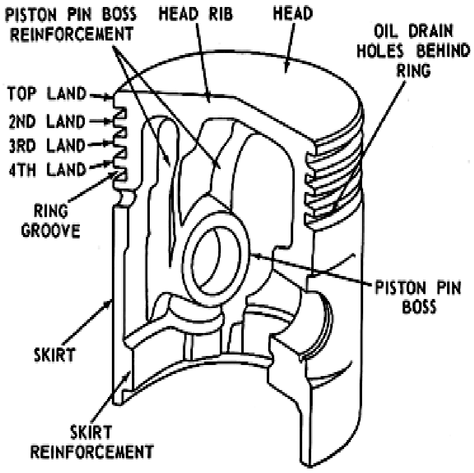


Figure 3. 21 Construction of Piston [37]

Design consideration for a piston: the design has to consider the following:

- It should have enormous strength to withstand the high gas pressure and inertia forces and thermal load.
- It should have minimum mass to minimize the inertia forces.
- It should form an effective gas and oil sealing of the cylinder.
- It should provide sufficient bearing area to prevent undue wear and noise.
- It should disperse the heat of combustion quickly to the cylinder walls.
- It should have high speed reciprocation without noise.
- It should be of sufficient rigid construction to withstand thermal and mechanical distortion.
- It should have sufficient support for the piston pin.

Piston head or Crown

The piston head or crown is designed keeping in view the following two main considerations,

1. It should have adequate strength to withstand the straining action due to pressure of explosion inside the engine cylinder, and
2. It should dissipate the heat of combustion to the cylinder walls as quickly as possible.

On the basis of first consideration of straining action, the thickness of the piston head is determined by treating it as a flat circular plate of uniform thickness, fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure over the entire cross-section.

Material and Manufacturing processes: Pistons are usually made from cast iron or aluminum alloy by casting or forging, drilling, grinding, etc. Since the coefficient of thermal expansion for aluminum ($0.24x10^{-6}$ m/^{0}C) is about 2.5 times that of cast iron ($0.1x10^{-6}$ m/^{0}C), Since the heat conductivity of aluminum alloy (175 W/m/^{0}C) is about 4 times that of cast iron (47 W/m/^{0}C), aluminum pistons must ensure a higher rate of heat transfer. This keeps down the maximum temperature difference between the center and edges of the piston head. Under full load conditions,

- The temperature at the center of the piston head for aluminum alloy is about 260^{0}C to 290^{0}C.
- The temperature at the edges of the piston head for aluminum alloy is about 185^{0}C to 215^{0}C.
- Aluminum alloy is about 3 times lighter than that of cast iron.

Size determination of piston

As shown in figure 3.22 below the size of piston such as piston diameter, piston head thickness, lands, grooves, piston ring and piston skirt etc… were determined analytically.

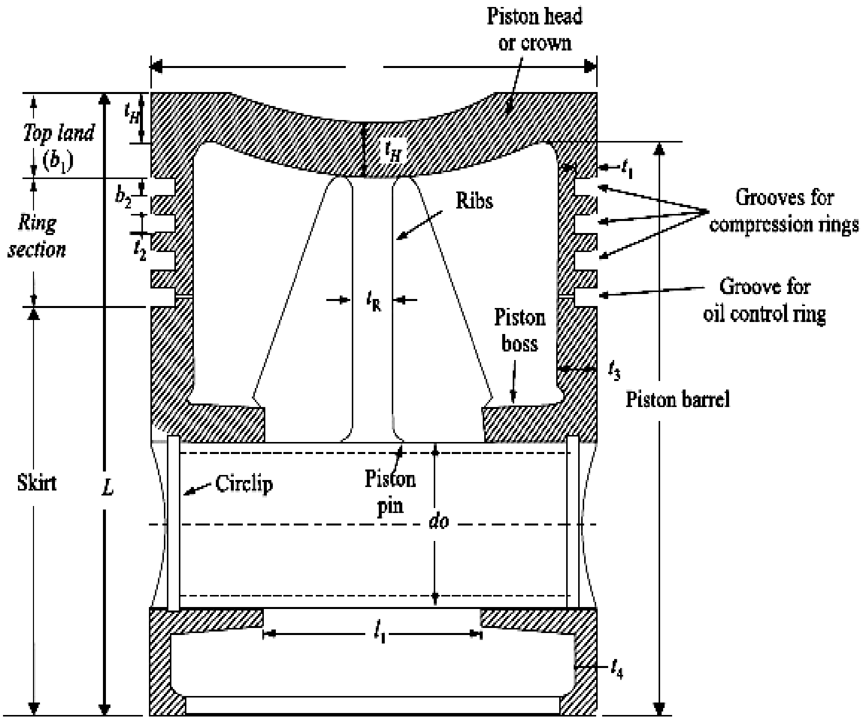


Figure 3. 22 Forces or Loads on a Piston and Piston Dimensions [37]

Piston diameter (d_p): The piston diameter (d_p) is given by, $d_p = D - c_d = 91.955$ mm

Where; c_d = diametral clearance (≈ 0.045 mm)

Piston head thickness (t_H): On the basis of strength, The thickness of the piston head (t_H) can be determined by treating it as a flat circular plate of uniform thickness fixed at the outer edges and subjected to a uniformly distributed load due to the gas pressure. From Grashoff's formula, it is given by,

$$t_H = \sqrt{\frac{3\ P_{g-max}\ x\ D^2}{16\ \sigma_t}} \qquad (55)$$

$$t_H = \sqrt{\frac{3\ x\ 107.8\ x\ 10^5\ \ x\ 92^2}{16\ x\ 88}} = 10.4 \text{ mm}$$

Where; σ_t = Permissible bending (tensile) stress (50 to 90 MPa for nickel cast iron and aluminum alloy)

On the basis of heat transfer, the heat flowing through the piston head (H) may be determined by the following expression,

$H = C \times HCV \times m \times P_{b-max}$ .... (56)

$H = 0.05 \times 45 \times 10^3$ kJ/kg x 6.125×10^{-3} kg/s x 26.25 kW = 361.7578 kW

$$Q_{\psi} = \frac{C \times Q_{LHV} \times bsfc \times P_b}{3.6} \quad \text{.. (57)}$$

$$Q_{\psi} = \frac{0.05 \times 42500 \times 210 \times 105}{3.6} = 13015625 \text{ W}$$

Where; H, Q_{ψ} = piston head heat flow (kW and W) respectively

C = constant of the heat supplied to the engine which is absorbed by the piston. (0.05)

HCV = Fuel higher calorific value (45 x 10^3kJ/kg for diesel)

m = Fuel mass (6.125×10^{-3}kg/s kg/s)

Q_{LHV} = Fuel lower Heating Value (≈ 42.5MJ/kg for Diesel)

$bsfc@T_{b-max}$ = The mass of fuel consumed (g/kWh)

$P_b@T_{b-max}$ = The brake power of the engine per cylinder (kW)

The thickness of the piston head (t_H) should also be designed in such a way that the heat absorbed by the piston due to the combustion has to be quickly transferred to the cylinder wall. Considering the piston head as a flat circular plate, its thickness is given by,

$$t_H = \frac{H}{12.56 \times K\,(T_C - T_E)} \quad \text{.. (58)}$$

$$t_H = \frac{361757.8 \text{ W}}{12.56 \times 174.75 \text{ W/m/}^{\circ}\text{C} \,(75^{\circ}\text{C})} = 2.2 \text{ mm}$$

Where; H = Heat flowing through the piston head (KJ/S or Watts)

K = Heat conductivity factor (174.75 W/m/℃ for aluminum alloys)

T_C = piston head at the center Temperature (℃)

T_E = piston head at the edges Temperature (℃)

(T_C – T_E) = 75^0C for aluminum.

1. By using equations (55) and (58), larger of the two values obtained should be adopted, t_H = 10.4 mm

2.When t_H is greater than 6 mm, then a suitable number of ribs at the center line of the boss extending around the skirts should be provided to distribute the side thrust from the connecting rod and thus to prevent distortion of the skirt. The thickness of the ribs (t_R) may be takes as,

$t_R = t_H/3$ to $t_H/2$ = 3.467 to 5.2 mm, Take t_R = 5 mm

3. Since the ratio of S/D = 1.0217, therefore a cup in the top of the piston head with a radius equal to 0.7D = 64.4 mm is provided. This is done to provide a space for combustion chamber.

4. The thickness of the piston head (t_H) should be compared with the value obtained from the following empirical relations, $t_H = 0.075D = 6.9$ mm

3.3.3.3 Piston Rings

Function: The main functions of piston rings are to seal the clearance between the piston and the cylinder wall (compression rings) and to control and prevent the flow of oil from entering into the combustion chamber (oil-control rings). In addition, piston rings are also used to transfer heat from the piston to the cylinder wall and absorb part of the piston fluctuation due to the side thrust. Oil-control rings provide proper lubrication to the liner by allowing sufficient oil to move up during upward stroke and at the same time scraps the lubricating oil from the surface of the liner in order to minimize the flow of the oil to the combustion chamber.

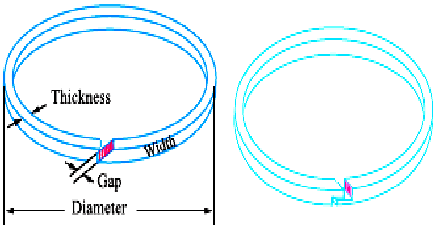
Construction: The piston rings are of the following two types:

1. Compression rings or pressure rings,

The compression rings or pressure rings are inserted in the grooves at the top portion of the piston and may be three to seven in number. The compression rings or pressure rings are constructed to possess end gap for ease of assembly and disassembly with various types of end joints or cuts (diagonal, butt, step, etc.). The compression rings are usually made of rectangular cross-section and the diameter of the ring is slightly larger than the cylinder bore.

2. Oil control rings or oil scraper.

The oil control rings or oil scrapers are provided below the compression rings. The oil control rings or oil scrapers are constructed to possess top rail, bottom rail and expander spacer. A part of the ring is cut-off in order to permit it to go into the cylinder against the liner wall. The diagonal cut or step cut ends as shown in figure 3.23 may be used. The gap between the ends should be sufficiently larger when the ring is put cold so that even at the highest temperature, the ends do not touch each other when the ring expands, otherwise there might be buckling of the ring.



(a) Diagonal cut (b) Step cut

Figure 3. 23 Construction of Piston Rings [37]

Design consideration for piston ring: As the piston rings have to withstand the high combustion temperature, pressure and side thrust force, the design has to consider thermal and mechanical loads. The end gap should be determined precisely, otherwise at higher temperature the ends may touch each other due to thermal expansion and this causes buckling and then breaking of the rings.

Material and Manufacturing processes: Piston rings are usually made from grey cast iron or alloy cast iron due to their good wearing and spring properties and also they retain spring characteristics even at high temperatures. The types of manufacturing processes are: centrifugal casting, grinding, etc.

Size determination of piston ring:

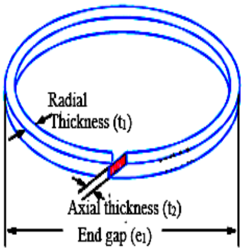


Figure 3. 24 Piston Ring Dimensions [37]

Radial thickness of the Compression Ring (t_r): as shown in the above figure 3.24, the radial thickness of the compression ring can be obtained by considering a radial pressure between the cylinder wall and the ring. From bending stress consideration, For the Ring Groove, $t_1 \approx t_r + (0.3 \text{ to } 0.4 \text{ mm}) = 3.44$ to 3.54 mm the radial thickness is given by,

$$t_r = D \text{ x } \sqrt{\frac{3\,P_w}{\sigma_t}} \quad \text{.......} \quad (59)$$

$$t_r = 92 \text{ x } \sqrt{\frac{3 \text{ x } 0.035}{90}} = 3.14 \text{ mm}$$

Where; P_w = cylinder wall gas pressure (0.035 N/mm^2)

σ_t = Allowable bending (tensile) stress (90 MPa For cast iron rings)

Axial thickness of the Compression Ring (t_a): The axial thickness of the compression ring for the Ring Groove, $t_2 \approx t_a (0.03 \text{ to } 0.06$ mm) = 2.7 to 2.73 mm may be taken as,

$t_a = 0.7\, t_r$ to t_r = 2.199 to 3.14 mm, take $t_a = 2.67$ mm

The minimum axial thickness (t_a): may also be obtained from the following empirical relation:

$$t_a = \frac{D}{20 \text{ x } n_R} = 1.533 \text{ mm}$$

Where; n_R = the number of compression rings (3)

Axial thickness of the Oil-control Ring (t_{ao}): Axial thickness of oil-control ring for the Ring Groove, $t_{2o} = t_{ao} + (0.02 \text{ to } 0.2$mm) = 5.36 to 5.54mm can be determined as, $t_{ao} = 2 \text{ x } t_a$ = 5.34mm

End gap of the compression ring (e_1): The end gap of the compression ring when it is put inside the cylinder can be determined as, e_1 = 0.002D to 0.004D = 0.184 to 0.368 mm.

Free end gap of the compression ring (e_2): The free end gap of the compression ring when it is outside of the cylinder can be determined as, e_2 = 3.5t_r to 4t_r= 10.99 to 12.56 mm.

The width of the top land (b_1): The distance from the top of the piston to the first compression ring groove is made to be a bit larger than the thickness of the piston head, so as to protect the top ring from the high combustion temperature. Hence, it can be taken as, $b_1 = t_H$ to 1.2 t_H = 10.4 to 12.48 mm.

Take b_1=11.44 mm

The width of other ring lands (b_2): The distance between the ring grooves in the piston may be made equal to or slightly less than the axial thickness of the compression ring (t_a). Hence it can be taken as, $b_2 \approx t_2 + (0.5 \text{ to } 1$ mm) = 3.215 to 3.715 mm or b_2 = 0.75 t_a to t_a = 2 to 2.67 mm.

Take b_2 = 2.33625 mm

The depth of the ring grooves should be more than the depth of the ring so that the ring does not take any piston side thrust.

Piston Barrel Thickness (t_3): It is a cylinder portion of the piston. The maximum thickness (t_3) of the piston barrel may be obtained from the following empirical relation:

t_3 = 0.03D + b + 4.5 mm = 0.03 x 92 mm + 3.54 mm + 4.5 mm = 10.8 mm

Where; b = Radial depth of piston ring groove (t_r + 0.4 mm = 3.14 mm + 0.4 = 3.54 mm)

The above relation may be written as,

t_3 = 0.03 D + t_r + 4.9 mm = 0.03 x 92 mm +3.14 mm + 4.9 mm = 10.8 mm

Piston Wall Thickness at the Bottom (t_4): towards the open end the piston wall thickness is decreased and should be taken as, $t_4 = 0.5t_3$ to t_3 or t_4 = 0.25t_3 to 0.35t_3 = 2.7 to 3.78 mm

Take t_4 = 3.24 mm

3.3.3.4 Piston Skirt

The portion of the piston below the ring section is known as piston skirt. In acts as a bearing for the side thrust of the connecting rod. The length of the piston skirt should be such that the bearing pressure on the piston barrel due to the side thrust does not exceed 0.5 N/mm^2 of the projected area for high speed engines. It may be noted that the maximum thrust will be during the expansion stroke. The side thrust (R) on the cylinder liner is usually taken as 1/10 of the maximum gas load on the piston. We know that maximum gas load on the piston,

$$F_{g-max} = P_{g-max} \text{ x} \frac{\pi D^2}{4} \quad \text{......} (60)$$

F_{g-max} = 107.8 x10^5 N/mm^2 x $\frac{\pi (92mm)^2}{4}$ = 7.1661 x 10^7 kN

Maximum side thrust on the cylinder,

$$F_{s-max} = F_{g-max}/10 = 0.1 \text{ x } P_{g-max} \text{ x } \frac{\pi D^2}{4} \quad \text{......} (61)$$

F_{s-max} = 7.1661 x 10^7/10 = 0.1 x 107.8 x 10^5 x $\frac{\pi D^2}{4}$ = 7.166 x 10^6 kN

The side thrust (F_s) is also given by,

$$F_s = P_{bp} \text{ x D x } l_s \quad \text{......} (62)$$

$\Rightarrow P_{bp} = \frac{7.166 \text{ x } 10^6}{92 \text{ x } 65}$ = 1198.3484 kN/mm^2

Where; P_{bp} = Bearing pressure (N/mm^2)

l_s = Piston skirt length

From equations (61) and (62), the length of the piston skirt (l_s) is determined. In actual practice, the length of the piston skirt is taken as, l_s = 0.65D to 0.8D = 59.8 to 73.6 mm. take l_s = 65 mm

Total Length of the Piston (l_p): Total length of the piston is given by,

l_p = Length of skirt + Length of ring section + Top land

$= l_s + (n_R t_a + 2b_2) + t_{2a} + b_1$

= 65 + (3 x 2.67 + 2 x 2.33625) + 5.34 + 11.44 = 94.46 mm

In actual practice, the length of the piston usually varies between, D and 1.5 D = 92 to 138 mm.

It may be noted that a longer piston provides better bearing surface for quiet running of the engine, but it should not be made unnecessarily long as it will increase its own mass and thus the inertia forces. Solid work model of piston is shown in figure 3.25 below.

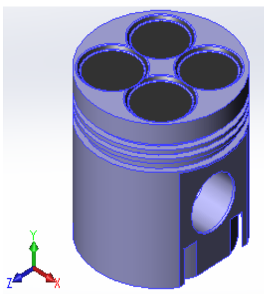


Figure 3. 25 Three Dimensional Sketch of the Piston

3.3.3.5 Piston Pin

Function: The piston pin (also called gudgeon pin or wrist pin) is used to connect the piston and the connecting rod.

Construction: It is usually made hollow and tapered on the inside, the smallest inside diameter being at the center of the pin, as shown in figure 3.26. The piston pin passes through the bosses provided on the inside of the piston skirt and the bush of the small end of the connecting rod. The center of piston pin should be 0.02 D to 0.04 D above the center of the skirt, in order to off-set the turning effect of the friction and to obtain uniform distribution of pressure between the piston and the cylinder liner.

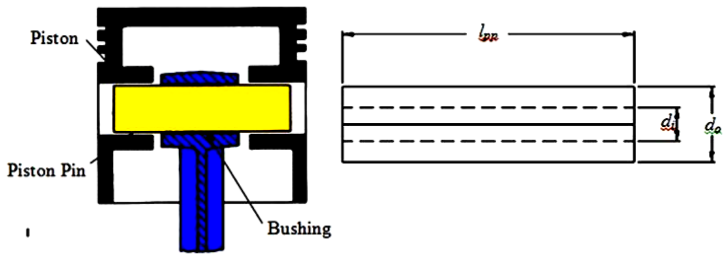


Figure 3. 26 Piston Pin Construction and Dimension

Design consideration for a piston pin: As the piston pin is mainly subjected to maximum gas and bearing pressure, the design has to consider mechanical loads.

Material and Manufacturing processes: The material used for the piston pin is usually case hardened steel alloy containing nickel, chromium, molybdenum or vanadium having tensile strength from 710 MPa to 910 MPa by turning, drilling, etc.

Size determination of piston pin

Length of the Piston Pin (l_{pp}): The total length of the piston pin will be about 0.85 of the piston diameter (d_p). $l_{pp} \approx 0.85 d_p = 78$ mm

Length of the Piston Pin (l_1): The length of the pin inside the connecting rod bushing will be about 0.45 of the piston diameter allowing for the end clearance of the pin.

$l_1 = 0.45 d_p = 0.45 \times 91.955$ mm $= 41.37975$ mm

Outer Diameter of the Piston Pin (d_0): The piston pin should be designed for the maximum gas force. The outside diameter of the piston pin can be determined by equating the load on the piston due to the maximum gas force and the load on the piston pin due to bearing pressure (p_{bp-pp}) at the small end of the connecting rod or piston pin bushing.

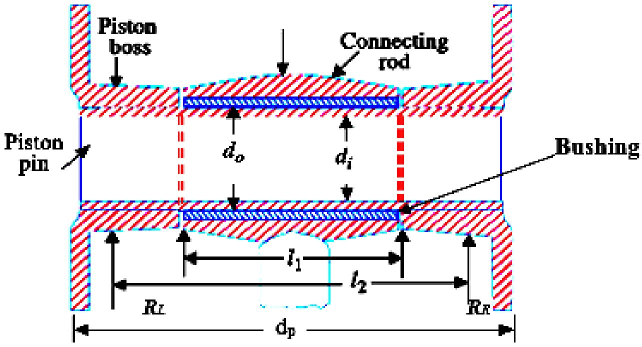


Figure 3. 27 Piston pin Load and Dimension [37]

The piston pin bushing is usually made from bronze of about 3 mm thick with allowable bearing pressure on the piston pin (p_{bp-pp}) of about 10.5 to 15 MPa. From this, the outside diameter of the piston pin can be determined as, F_{g-max} = bearing pressure x projected area

$$F_{g-max} = p_{bp-pp} \text{ x } d_0 \text{ x } l_1 \quad \ldots\ldots (63)$$

$$\Rightarrow d_0 = \frac{F_{g-max}}{p_{bp-pp} \text{ x } l_1} = 32 \text{ mm}$$

p_{bp-pp} Should be >55 MPa

The outside diameter of the piston pin may also be determined by considering the bending stress and assuming a uniformly distributed maximum gas load or force as shown in figure 3.28 below.

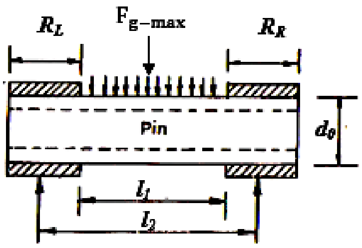


Figure 3. 28 Bending load on a Piston Pin [37]

The maximum bending moment (M_{max}) is given by,

$$M_{max} = \left(F_{g-max} \text{ x } \frac{l_2}{2}\right) - (R_R \text{ x } l_2) \quad \ldots\ldots (64)$$

$$M_{max} = \left(7.1661 \text{ x } 10^7 \text{ x } \frac{66.67}{2}\right) - (3.5883 \text{ x } 10^7 \text{ x } 66.67) = 0 \text{ N-mm}$$

Where; $l_2 = l_1 + \frac{d_p - l_1}{2} = \frac{d_p + l_1}{2} = 66.67$ mm

$$R_R = R_L = \frac{F_{g-max}}{2} = 3.5883 \text{ x } 10^7 \text{ kN}$$

The bending stress at the outer surface of the piston pin is given by,

$\sigma_{bp} = \frac{M_{max}}{I} \text{ x } \frac{d_0}{2}$ .... (65)

$\sigma_{bp} = \frac{0}{44801} \text{ x } \frac{32}{2} =$ 0 KN-mm

Where; σ_{bp} = allowable bending stress for the piston pin material (≈75 MPa)

$I = \frac{\pi}{64} \text{ x } (d_0{}^4 - d_i{}^4)$

Note: The two outside diameters have to be compared and the larger value will be taken.

Inner Diameter of the Piston Pin (d_i): the inner diameter of the piston pin will be about 0.6 of the outer diameter of the piston pin. $d_i = 0.6d_0 = 19.2$ mm

Load on the piston due to gas pressure or gas load $= \frac{\pi D^2}{4}$ x P = 7.152 x 10^7 kN..................... (66)

Load on the piston pin due to bearing pressure or bearing load $= P_{b1} \text{ x } d_0 \text{ x } l_1 = 33.040$ kN........ (67)

P_{b1} = Bearing pressure at the small end of the connecting rod bushing (25 N/mm^2 for bronze bushing).

From equations (65) and (66), the outside diameter of the piston pin may be obtained.

The piston pin may be checked in bending by assuming the gas load to be uniformly distributed over the length l_1 with supports at the center of the bosses at the two ends. Now maximum bending moment at the center of the pin,

$M_{max} = \frac{F_{g-max} \text{ x } D}{8}$ .... (68)

$M_{max} = \frac{7.1661 \text{ x } 10^7 \text{ x } 92}{8} = 8.24 \text{ x } 10^8$ kN-mm

We have already discussed that the piston pin is made hollow. Let d_0 and d_i be the outside and inside diameters of the piston pin as shown in the above figure 5.12. We know that maximum bending moment,

$M_{max} = Z \text{ x } \sigma_b = \frac{\pi}{32} \left[\frac{(d_0)^4 - (d_i)^4}{d_0}\right] \sigma_b$ .... (69)

$M_{max} = \frac{\pi}{32} \left[\frac{(32)^4 - (19.2)^4}{32}\right] 84 = 235.205$ kN-mm

Where; Z = section modulus

σ_b = Allowable bending stress for the material of the piston pin

84 MPa for case hardened carbon steel and 140 MPa for heat treated alloy steel. Assuming $d_i = 0.6d_o$, the induced bending stress in the piston pin may be checked using equation 68 and 69. $d_o = 32$ mm and $d_i = 19.2$ mm. the solidwork model is shown in figure 3.29 below.

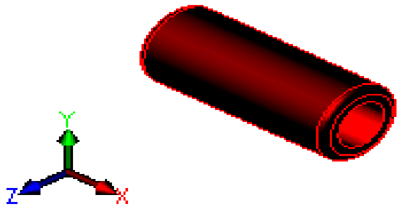


Figure 3. 29 Three Dimensional Sketch of Piston pin

3.3.3.6 Connecting Rod with Small end (piston pin) and Big end (Crank pin)

Function: The connecting rod is the intermediate member between the piston and the crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crankpin and thus convert the reciprocating motion of the piston into the rotary motion of the crank.

Construction: It consists of a long shank, a small end and a big end with bearings and cap. The cross-section of the shank may be rectangular, circular, tubular, I-section or H-section. Generally circular section is used for low speed engines while I-section is preferred for high speed engines. The small end of the connecting rod is usually made in the form of an eye and is provided with a bush of phosphor bronze. It is connected to the piston by means of a piston pin. The big end of the connecting rod is usually made split (in two halves) as shown in figure 3.30 below so that it can be mounted easily on the crankpin bearing shells. The split cap is fastened to the big end with two cap bolts.

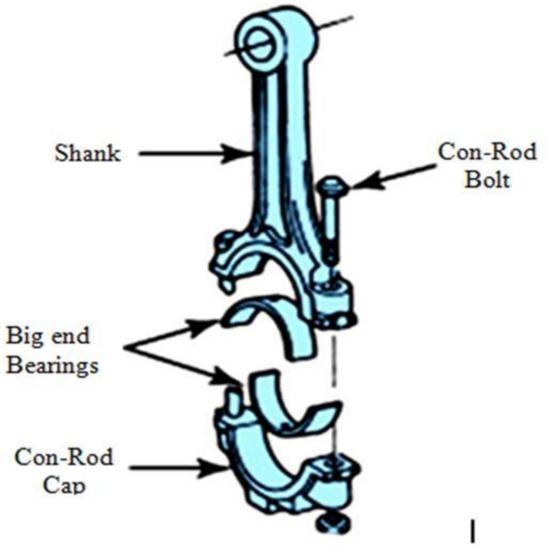


Figure 3. 30 Construction of Connecting Road [37]

Design consideration for connecting rod: As the connecting rod is mainly subjected to: axial (compressive and tensile), twisting and fatigue (fluctuating or cyclic) types of loads, the design has to consider the combined effect of these types of loads.

Material and Manufacturing processes: The material mostly used for connecting rods varies from mild carbon steels (having 0.35 to 0.45 percent carbon) to alloy steels (chrome-nickel or chrome-molybdenum steels). The alloy steels have an ultimate tensile strength of about 1050 MPa and are used for connecting rods of aero engines and automobile engines. The bearing shells of the big end are made of steel, brass or bronze with a thin lining (about 0.75 mm) of white metal or babbit metal. The wear of the big end bearing is allowed for by inserting thin metallic strips (known as shims) about 0.04 mm thick between the cap and the fixed half of the connecting rod. As the wear takes place, one or more strips are removed and the bearing is trued up.

Since connecting rod is manufactured by forging, therefore the sharp corner of I-section are rounded off as shown in figure 3.32 (b) below for easy removal of the section from dies.

Size determination of connecting rod

As shown in figure 3.31 below the length of the connecting rod (l_{con} or l) depends upon the ratio of (l_{con}/r_{cr}), where r_{cr} is the radius of crank. It may be noted that the smallest length will decrease the ratio (l_{con}/r_{cr}). This increases the angularity of the connecting rod which increases the side thrust of the piston against the cylinder liner which in turn increases the wear of the liner. The larger length of the connecting rod will increase the ratio (l_{con}/r_{cr}). This decreases the angularity of the connecting rod and thus decreases the side thrust and the resulting wear of the cylinder. But the larger length of the connecting rod increases the overall height of the engine. Hence, a compromise is made and the ratio (l_{con}/r_{cr}) is generally kept as 4 to 5.

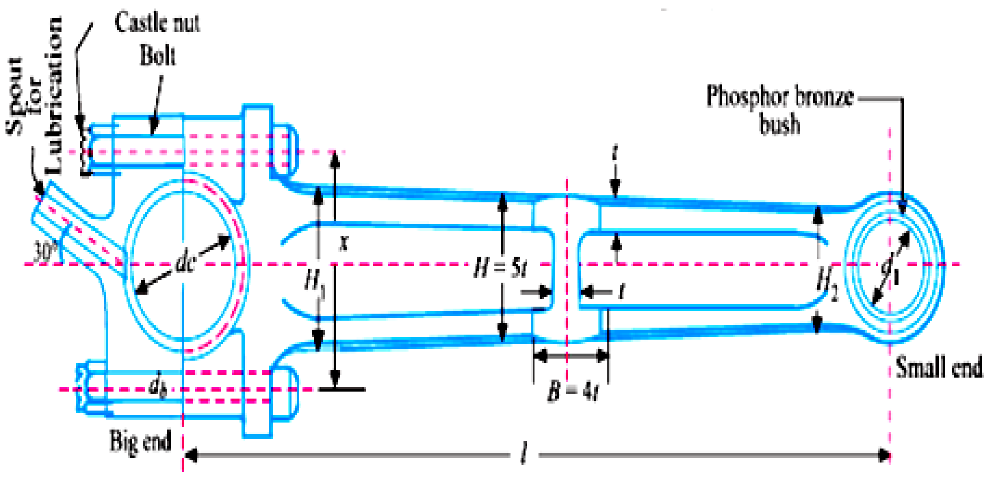


Figure 3. 31 Connecting rod Dimension [37]

In designing a connecting rod, the following dimensions are required to be determined:

1. Dimension of cross-section of the connecting rod,
2. Dimension of the crankpin at the big end and the piston pin at the small end,
3. Size of bolts for securing the big end cap, and
4. Thickness of the big end cap.

The procedure adopted in determining the above mentioned dimensions is discussed as below:

1. Dimension of cross-section of the connecting rod

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, therefore, the cross-section of the connecting rod is designed as a strut and the Rankine's formula is used.

A connecting rod, is subjected to an axial load W may buckle with X-axis as neutral axis (i.e. in the plane of motion of the connecting rod) or Y-axis as neutral axis (i.e. in the plane perpendicular to the plane of motion). The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. A connecting rod should be equally strong in buckling about the axes. The most suitable section for the connecting rod is I- section with the proportions as shown in figure 3.32 (a) below.

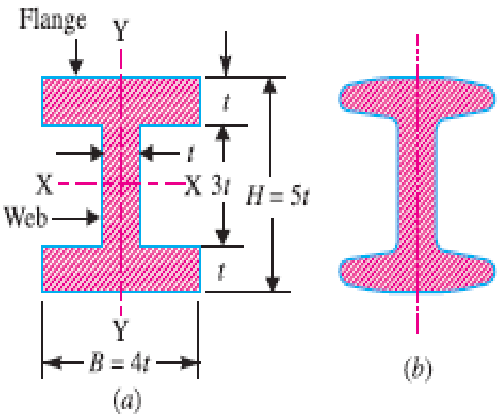


Figure 3. 32 I-section of Connecting rod [37]

Let thickness of the flange and web of the section = t = 5 mm ≈ 5 mm

Width of the section, B = 4t = 20 mm and depth or height of the section, H = 5t = 25 mm

From figure 3.32 (a), we find that area of the section, A = 2(4t x t) + 3t x t = $11t^2 = 275mm^2$

Moment of inertia of the section about X-axis, $I_{xx} = \frac{1}{12}[4t(5t)^3 - 3t(3t)^3] = \frac{419}{12}t^4 = 21822.92\ mm^4$

And moment of inertia of the section about Y-axis,

$$I_{yy} = [2 \text{ x } \frac{1}{12} t \text{ x } (4t)^3 + \frac{1}{12}(3t)\, t^3] = \frac{131}{12} t^4 = 6822.92 \text{ mm}^4$$

Therefore, $\frac{I_{xx}}{I_{yy}} = \frac{419}{12} \text{ x } \frac{12}{131} = 3.2$

Since the value of $\frac{I_{xx}}{I_{yy}}$ lies between 3 and 3.5, therefore, I-section chosen is quite satisfactory.

Let I_{xx} And I_{yy} is Moment of inertia of the section about X-axis and Y-axis respectively, and k_{xx} and k_{yy} = Radius of gyration of the section about X-axis and Y-axis respectively.

According to Rankine's formula, the buckling load will be: For both ends hinged About X-axis (L = l)

$$W_B = \frac{\sigma_c A}{1 + \alpha\left(\frac{L}{k_{xx}}\right)^2} = \frac{\sigma_c A}{1 + \alpha\left(\frac{l}{k_{xx}}\right)^2} \quad \text{..........} (70)$$

$$W_B = \frac{1517.945 \text{ x } 275}{1{+}1.33 \text{ x } 10^{-4}\left(\frac{188}{8.9}\right)^2} = 3.94 \text{ x} 10^8 \text{ KN}$$

For both ends fixed About Y-axis ($L = \frac{l}{2}$)

$$W_B = \frac{\sigma_c A}{1 + \alpha\left(\frac{L}{k_{yy}}\right)^2} = \frac{\sigma_c A}{1 + \alpha\left(\frac{l}{2k_{yy}}\right)^2} \quad \text{..........} (71)$$

$$W_B = \frac{1517 \text{ x } 275}{1{+}1.33 \text{ x } 10^{-4}\left(\frac{188}{2 \text{ x } 4.98}\right)^2} = 3.98 \text{ x } 10^8 \text{ KN}$$

Where; σ_c = Compressive yield stress (1517.945 MPa)

A = connecting rod Cross-sectional area ($11t^2 = 275\text{mm}^2$)

α = Constant (1/7500 for mild steel; 1/9000 for wrought iron; 1/1600 for cast iron)

L = connecting rod Equivalent length (188 mm)

l = Connecting rod length (mm)

k = Radius of gyration ($I = A \text{ x } k^2$)

In order to have a connecting rod equally strong in buckling about both the axes, the buckling loads must be equal, i.e. $\frac{\sigma_c.A}{1+\alpha\left(\frac{l}{k_{xx}}\right)^2} = \frac{\sigma_c.A}{1+\alpha\left(\frac{l}{2k_{yy}}\right)^2}$ or $\left(\frac{l}{k_{xx}}\right)^2 = \left(\frac{l}{2k_{yy}}\right)^2$

Therefore, $k_{xx}{}^2 = 4k_{yy}{}^2$ or $I_{xx} = 4I_{yy}$ and $I = A.k^2$

This shows that the connecting rod is four times strong in buckling about Y-axis than about X-axis. If $I_{xx} > 4I_{yy}$, then buckling will occur about Y-axis and if $I_{xx} < 4I_{yy}$ buckling will occur about X- axis. In actual practice, I_{xx} is kept slightly less than $4I_{yy}$. It is usually taken between 3 and 3.5 and the connecting rod is designed for buckling about X-axis. The design will always be satisfactory for buckling about Y-axis.

After deciding the proportions for I-section of the connecting rod, its dimensions are determined by considering the buckling of the rod about X-axis (assuming both ends hinged) and applying the Rankine's formula. We know that buckling load,

$$W_B = \frac{\sigma_c . A}{1+\alpha\left(\frac{L}{k_{xx}}\right)^2} = \frac{1517 \text{ x } 275}{1+1.33 \text{ x } 10^{-4}\left(\frac{188}{8.9}\right)^2} = 3.94 \text{ x } 10^8 \text{ kN}$$

The buckling load (W_B) may be calculated by using the following relation, W_B = Max. Gas force x Factor of safety.

$$W_B = F_{g-max} \text{ x } FS \quad \ldots\ldots (73)$$

W_B = 7.1661 X 10^7 kN x 5.5 = 3.94 x 10^8 kN

The factor of safety may be taken as 5 to 6. Let factor of safety = 5.5

The dimensions B = 4t and H = 5t, as obtained above by applying the Rankine's formula, are at the middle of the connecting rod. The width of the section (B) is kept constant throughout the length of the connecting rod, but the depth or height varies. The depth near the small end (or piston end) is taken as H_2 = 0.75H to 0.9H = 18.75 to 22.5 mm and the depth near the big end (or crank end) is taken H_1 = 1.1H to 1.25H = 27.5 to 31.25 mm.

2. Dimensions of the crankpin at the big end and the piston pin at the small end

Since the dimensions of the crankpin at the big end and the piston pin (also known as gudgeon pin or wrist pin) at the small end are limited, therefore, fairly high bearing pressures have to be allowed at the bearings of these two pins.

The crankpin at the big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1 mm or less) of bearing metal (such as tin, lead, babbit, copper, lead) on the inner surface of the shell. The allowable bearing pressure on the crankpin depends upon many factors such as material of the bearing, viscosity of the lubrication and the space limitations. The value of bearing pressure may be taken as 7 N/mm^2 to 12.5 N/mm^2 depending upon the material and method of lubrication used.

The piston pin bearing is usually a phosphor bronze bush of about 3 mm thickness and the allowable bearing pressure may be taken as 10.5 N/mm^2 to 15 N/mm^2.

Since the maximum load to be carried by the crankpin and piston pin bearings is the maximum force in the connecting rod (F_{con}), therefore the dimensions for these two pins are determined for the maximum force in the connecting rod (F_{con}) which is taken equal to the maximum force on the piston due to gas pressure (F_g) neglecting the inertia forces. We know that maximum gas force,

$$F_{g-max} = \frac{\pi D^2}{4} x P_{g-max} \quad \text{..........} (74)$$

$F_{g-max} = \frac{\pi (92)^2}{4} x\ 107.8\ x\ 10^5 = 7.1661\ x\ 10^7$ kN

Now the dimensions of the crankpin and piston pin are determined as discussed below:

We know that load on the crank pin $= d_{cp}\ x\ l_{cp}\ x\ p_{bc}$ .... (75)

Where; d_{cp} = Crank pin Diameter (mm)

l_{cp} = Crank pin Length ($1.25d_{cp}$ to $1.5d_{cp}$)

p_{bc} = Allowable bearing pressure ($9.75 N/mm^2$)

Similarly, load on the piston pin $= d_p\ x\ l_{pp}\ x\ p_{bp}$ .... (76)

Where; d_p = Piston pin diameter (mm)

l_{pp} = Piston pin length ($l_{pp} = 1.5d_p$ to $2d_p$)

p_{bp} = Allowable bearing pressure ($26.05 N/mm^2$)

Equating equation (74) and (75), we have $F_L = d_{cp}\ x\ l_{cp}\ x\ p_{bc} = 7.1661\ x\ 10^7$ kN

Taking $l_{cp} = 1.25d_{cp}$ to $1.5d_{cp}$, the value of d_{cp} and l_{cp} are determined from the above expression.

Again, equating equations (74) and (76), we have $F_L = d_p\ x\ l_{pp}\ x\ p_{bp} = 7.1661\ x\ 10^7$ kN

Taking $l_{pp} = 1.5d_p$ to $2d_p$, the value of d_p and l_{pp} are determined from the above expression.

3. Size of bolts for securing the big end cap

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead center on the exhaust stroke. We know that inertia force of the reciprocating parts.

$$F_i = m_R\ x\ \omega^2\ x\ r_{cr}\left(\cos\theta + \frac{\cos 2\theta}{l/r_{cr}}\right) \quad \text{..........} (77)$$

At the top dead center, the angle of inclination of the crank with the line of stroke, $\theta = 0$

$F_i = m_R\ x\ \omega^2\ x\ r_{cr}\left(1 + \frac{r_{cr}}{l}\right) = 1.5\ x\ (418.879)^2\ x\ 0.047\left(1 + \frac{0.047}{0.188}\right) = 15.46238$ kN

Where; m_R = reciprocating parts mass (1.5 kg)

ω = Angular engine speed (rad/s),

r_{cr} = Crank radius (m)

θ = Angle of inclination of the crank from top dead center (deg.)

l = Connecting rod length (m)

The bolts may be made of high carbon steel or nickel alloy steel. Since the bolts are under repeated stresses but not alternating stresses, therefore, a factor of safety may be taken as 6. Equating the inertia force to the force on the bolts, we have

$$F_i = \frac{\pi}{4}(d_{cb})^2 \text{ x } \sigma_t \text{ x } n_b \quad \text{......} (78)$$

$$\Rightarrow d_{cb} = \sqrt{\frac{F_i}{\frac{\pi}{4} \text{ x } 3 \text{ x } \sigma_t \text{ x } n_b}} = \sqrt{\frac{15.46238}{\frac{\pi}{4} \text{ x } 3 \text{ x } 38.4636 \text{ x } 2}} = 16 \text{ mm}$$

Where; d_{cb} = Bolt core diameter (mm)

σ_t = Allowable tensile stress for the material of bolts (MPa), and

n_b = Number of bolts

From this expression, d_{cb} is obtained. The nominal or major diameter (d_b) of the bolts is given by

$$d_b = \frac{d_{cb}}{0.84} = 19.0476 \text{ mm}$$

4. Thickness of the big end cap

The thickness of the big end cap (t_c) may be determined by treating the cap as a beam freely supported at the cap bolt centers and loaded by the inertia force at the top dead center on the exhaust stroke (i.e. F_i when $\theta = 0$). This load is assumed to act in between the uniformly distributed load and the centrally concentrated load. Therefore, the maximum bending moment acting on the cap will be taken as,

$$M_C = \frac{F_i \text{ x } X}{6} \quad \text{......} (79)$$

$$M_C = \frac{15.46238 \text{ x } 87}{6} = 0.22420451 \text{ kN-mm}$$

Where; X = Distance between the bolt centers.

= d_c + 2 x Thickness of bearing liner (3 mm) + Clearance (3 mm).

Section modulus for the cap,

$$Z_C = \frac{b_C (t_C)^2}{6} \quad \text{......} (80)$$

Where; b_c = Cap width (mm)

t_c = Crank pin thickness or Big end cap thickness (mm)

Bending stress,

$$\sigma_b = \frac{M_C}{Z_C} = \frac{F_i \text{ x } X}{6} \text{ X } \frac{6}{b_C \ (t_C)^2} = \frac{F_i \text{ x } X}{b_C \ (t_C)^2} \quad \text{......} (81)$$

σ_b = Allowable bending stress for the material of the cap (MPa)

From this expression, the value of t_c is obtained.

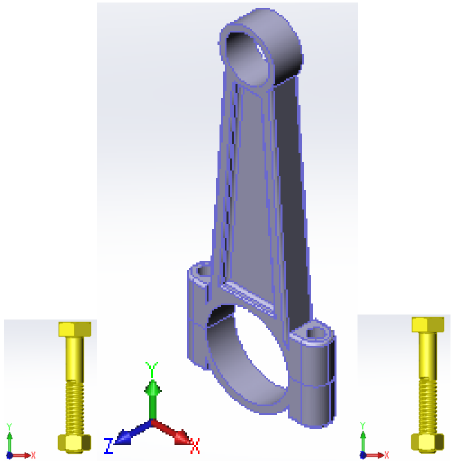


Figure 3. 33 Three Dimensional Sketch of Connecting rod

3.3.3.7 Crankshaft (Crank Pin, Crank Web, Main Journal)

Function: The main function of the crankshaft is to convert the reciprocating motion of the piston into rotary motion or vice versa.

Construction: as shown in figure 3.34 below, crankshaft is constructed to possess: Crank Pins, Crank Webs, and Main Journals, Balance or Counter Weights, Flange, Oil Passages, etc. The crankshaft consists of the shaft parts which revolve in the main bearings, the crankpins to which the big ends of the connecting rod are connected, the crank arms or webs (also called cheeks) which connect the crankpins and the shaft parts.

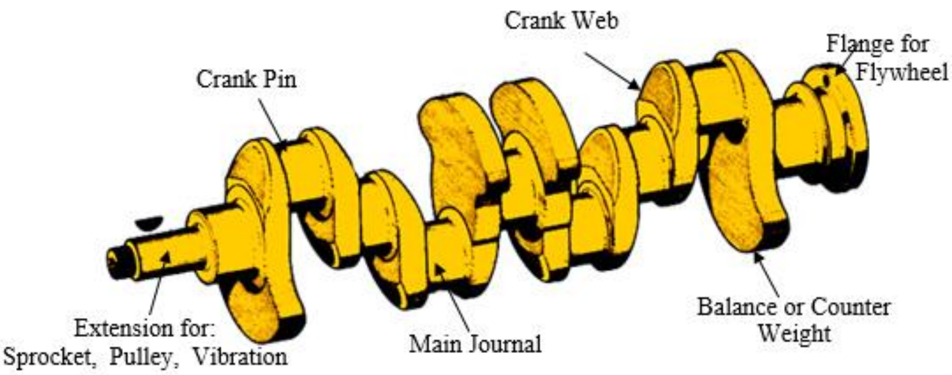


Figure 3. 34 Main Parts of a Crankshaft [37]

Design consideration for crankshaft: The crankshaft is mainly subjected to: bending, twisting, axial (tensile and compressive), cyclic or fatigue types of stresses with torsional vibration due to gas and inertia forces and their couples. In addition, the bearing pressure acting on the crank pin and on the main journal has to be taken into consideration as it depends on: the maximum gas pressure, engine speed, amount and method of lubrication. Hence, the design should consider the combined effects of all types of stresses, control of vibration and accurate balancing. A high factor of safety (FS) about 3 to 4 may be used due to the endurance limit.

Bearing Pressures and Stresses in Crankshaft

The bearing pressures are very important in the design of crankshafts. The maximum permissible bearing pressure depends upon the maximum gas pressure, journal velocity, amount and method of lubrication and change of direction of bearing pressure. The following two types of stresses are induced in the crankshaft.

1. Bending stress; and 2. Shear stress due to torsional moment on the shaft

Most crankshaft failures are caused by a progressive fracture due to repeated bending or reversed torsional stresses. Thus the crankshaft is under fatigue loading and, therefore, its design should be based upon endurance limit. Since the failure of a crankshaft is likely to cause a serious engine destruction and neither all the forces nor all the stresses acting on the crankshaft can be determined accurately, therefore a high factor of safety from 3 to 4, based on the endurance limit is used.

Material and Manufacturing processes: The crankshaft material has to have adequate strength, toughness, hardness, wear and fatigue resistance. As a result, crankshafts of IC engines are usually made from a high or medium grade steel (for medium and high speed engines) by drop forging. The surface of the crankpin is hardened by case carburizing, nitriding or induction hardening.

Size determination

Case i): as shown in figure 3.35 below the crank is at the maximum bending moment position ($\theta \approx 15^0$ @ TDC). When the gas pressure attains its maximum value ($\theta \approx 15^\circ$@ TDC), the bending moment on the crank pin (crankshaft) becomes maximum and the twisting moment is negligible. At this position of crank angle, the connecting rod force (F_{con}) has two components (F_t and F_r). Neglecting the effects ofF_t.

$$F_r = F_{con} = \frac{F_p}{\cos\phi} \quad \ldots\ldots\ldots\ldots (82)$$

$F_r = F_{con} = \frac{7.1661 \times 10^7}{\cos(15)}$ = 7.4188926 x 10^7 kN

Where; $F_p = F_{g-max}$, neglecting the effects of inertial force (F_i).

F_r is already calculated at engine kinetics part. Points 1 and 2 in figure represent the center of the main journals bearings.

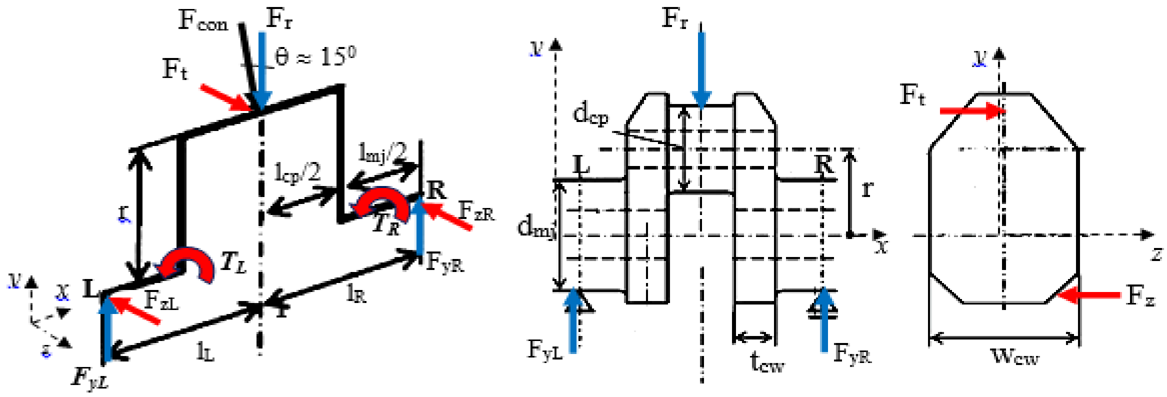


Figure 3. 35 Loads on the Crankshaft at F_{p_max}.

Note: Points L and R represent the center of Left and Right main journals.

Length of the Crank Pin (l_{cp}): The length of the crank pin for CI engines can be taken as,

0.5 D to 0.7 D = 46 to 64.4 mm

Thickness of the Crank Web (t_{cw}): The thickness of the crank web for CI engies can be taken as,

0.24 D to 0.27 D = 22.08 to 24.84 mm

Distance between the centers of the two Main Journals (l_1+l_2): The distance between the centers of the two main journals for CI engines can be taken as,

1.25 D to 1.30 D = 115 to 119.6 mm

Length of the Main Journal (l_{mj}): The length of the main journal can be determined as,

$l_{mj} = (l_1 + l_2) - l_{cp} - (2 \text{ x } t_{cw}) = 27.3$ mm

0.4267D = 39.256 mm ≈ 37.90mm

The vertical reaction forces acting at the center of the main journals (at points 1 and 2) can be determined as,

$R_{z1} = \frac{F_r \text{ x } l_2}{(l_1+l_2)} = 3.709 \text{ x } 10^7$ kN And

$R_{z2} = \frac{F_r \text{ x } l_1}{(l_1+l_2)} = 3.709 \text{ x } 10^7$ kN

Where; $l_2 = 57.5$ mm

$l_1 + l_2 = 115$ mm

The maximum bending moment at the center of crank pin (M_{max-cp}) as shown in figure 3.36 below is given by,

$M_{max-cp} = R_{z1} x l_1 = 3.709 \text{ x } 10^7 \text{ x } 57.5 = 2.132931632 \text{ x } 10^9$ kN-mm

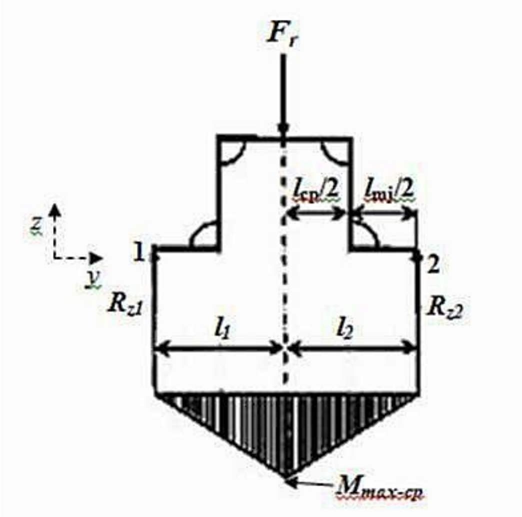


Figure 3. 36 Bending Moment Diagram (BMD) of the Crankshaft on a y-z plane

Diameter of the Crank Pin (d_{cp}): The diameter of the crank pin for CI engines can be taken as,

0.65 D to 0.75 D = 59.8 to 69 mm

The maximum stress on the Crank Pin at the critical section and at the corresponding critical point can be determined as,

i) Based on Combined Stresses using:

- The Maximum Normal (Principal) Stress Theory (for Brittle materials such as: Cast Iron)

$$\sigma_{max} = \frac{\sigma_x+\sigma_y}{2} + \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2{}_{xy}} \leq \sigma_{all} = \frac{\sigma_{yl}}{FS} \quad \ldots\ldots (83)$$

$$\sigma_{min} = \frac{\sigma_x+\sigma_y}{2} - \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2{}_{xy}} \leq \sigma_{all} - \frac{\sigma_{yl}}{FS} \quad \ldots\ldots (84)$$

- The Maximum Shear Stress Theory (for Ductile materials such as: Mild Steel)

$$\tau_{max} = \frac{\sigma_{max}-\sigma_{min}}{2} = \frac{1}{2}\sqrt{(\sigma_x - \sigma_y)^2 + 4\tau^2{}_{xy}} \leq \tau_{all} = \frac{\tau_{yl}}{FS} = \frac{\sigma_{yl}}{(2 \text{ x } FS)} \quad \ldots\ldots (85)$$

- The bending stress on the Crank Pin (σ_{yb}) is,

$$\sigma_{yb} = \frac{M_{max-cp} \text{ x } z}{I_{xx}} \leq \sigma_{all} \quad \ldots\ldots (86)$$

$$\sigma_{yb} = \frac{2.132931632 \text{ x } 10^9 \text{ x } 29.95}{627732.5295} = 101.765 \text{ MPa}$$

Where; $M_{max-cp} = R_{z1} x l_1 = 2.132931632 \text{ x } 10^9$ kN-mm

$z = \frac{d_{cp}}{2} = 29.95$ mm and

$I_{xx} = \frac{\pi \text{ x } d^4{}_{cp}}{64} = 627732.5295 \text{ mm}^4$

Since it is less than the allowable stress (241.1 MPa) the design is safe.

Hence, $\sigma_x = 0$; $\sigma_y = \sigma_{yb}$; $\tau_{xy} = 0$

ii) Based on Bearing Pressure (P_{bp-cp}), F_r = Bearing pressure x projected area

$$F_r = p_{bp-cp} \text{ x } d_{cp} \text{ x } l_{cp} \quad \text{...} (87)$$

$$\Rightarrow p_{bp-cp} = \frac{F_r}{d_{cp} \text{ x } l_{cp}} \leq p_{bp-m}$$

$$\Rightarrow P_{bp-cp} = \frac{7.4188926 \text{ x } 10^7}{\cos(15)} = 26.9699 \text{ MPa}$$

The mean bearing pressure on the crank pin (P_{bp-m}) for CI Engines is 8 to 16 MPa.

Width of the Crank Web (w_{cw}): The width of the crank web for CI engine can be taken as,

1.05 D to 1.3 D = 96.6 to 119.6 mm

The maximum stress on the Crank Web at the critical section and at the corresponding critical point can be determined using:

i) Based on Combined Stresses using:

- The axial stress (Compressive) on the left side Crank Web (σ_{xa}) is,

$$\sigma_{za} = \frac{R_{z1}}{w_{cw} \text{ x } t_{cw}} \quad \text{...} (88)$$

$$\sigma_{za} = \frac{3.59 \text{ x } 10^7}{96.6 \text{ x } 22.08} = 16.831 \text{ MPa}$$

- The bending stress (Compressive) on the left side Crank Web (σ_{zb}) is,

$$\sigma_{zb} = \frac{M_{max-cw} \text{ x } y}{I_{xx}} = \frac{M_{max-cw}}{z_{xx}} \quad \text{...} (89)$$

$$\sigma_{zb} = \frac{1.23855 \text{ x } 10^9 \text{ x } 11.04}{86654.89244} = \frac{1.23855 \text{ x } 10^9}{7849} = 157.79 \text{ MPa}$$

$$\sigma_x = \sigma_{max} = \sigma_{za} + \sigma_{zb} = 16.831 + 157.79 = 174.621 \text{ MPa.}$$

Where; $M_{max-cw} = R_{z1} \text{x} \left(l_1 - \frac{l_{cp}}{2}\right) = 3.59 \text{ x } 10^7 \text{ x} \left(57.5 - \frac{46}{2}\right) = 1.23855 \text{ x } 10^9$ kN-mm

$$y = \frac{t_{cw}}{2} = \frac{22.08}{2} = 11.04 \text{ mm;}$$

$$I_{xx} = \frac{w_{cw} \text{ x } t^3{}_{cw}}{12} = \frac{96.6 \text{ x } 22.08^3}{12} = 86654.89244 \text{mm}^4;$$

$$z_{xx} = \frac{I_{xx}}{y} = \frac{w_{cw} \text{ x } t^3{}_{cw}}{6} = \frac{96.6 \text{ x } 22.08^2}{6} = 7849 \text{ mm}^3$$

Since it is less than the allowable stress (241.1 MPa) the design is safe.

Hence, $\sigma_x = \sigma_{max} = \sigma_{za} + \sigma_{zb}$; $\sigma_y = 0$; $\tau_{xy} = 0$

Note: The stresses on the right side Crank Web can be determined in a similar way or the same size as the left side Crank Web can be taken, if both crank webs are equally loaded considering Engine Balancing.

Diameter of the Main Journal (d_{mj}): The diameter of the main journal for CI Engines can be taken as,

0.7 D to 0.9 D = 64.4 to 82.8 mm

0.667D = 61.364 mm ≈ 62.99 mm

The maximum stress on the Main Journal at the critical section and at the corresponding critical point can be determined as,

i) Based on Combined Stresses using:

- The bending stress on the Main Journal (σ_{yb}) is,

$$\sigma_{yb} = \frac{M_{max-mj} \text{ x } z}{I_{xx}} \leq \sigma_{all} \quad \text{.......(90)}$$

$$\sigma_{yb} = \frac{8.422 \text{ x } 10^8 \text{ x } 32.2}{844332.23} = 38 \text{ MPa}$$

Where; $M_{max-mj} = R_{z1} x(l_1 - \frac{l_{cp}}{2} - \frac{t_{cw}}{2}) = 3.59 \text{ x } 10^7 x\left(57.5 - \frac{46}{2} - \frac{22.08}{2}\right) = 8.422 \text{ x } 10^8$ kN-mm

$$z = \frac{d_{mj}}{2} = \frac{64.4}{2} = 32.2 \text{ mm}$$

$$I_{xx} = \frac{\pi \text{ x } d^4{}_{mj}}{64} = 844332.23 \text{ mm}^4$$

Since it is less than the allowable stress (241.1 MPa) the design is safe.

Hence, $\sigma_x = 0$; $\sigma_y = \sigma_{yb}$; $\tau_{xy} = 0$

ii) Based on Bearing Pressure (p_{bp-mj}), R_{zl} = Bearing pressure x projected area

$$R_{zl} = p_{bp-mj} \text{ x } d_{mj} \text{ x } l_{mj} \quad \text{.......(91)}$$

$$\Rightarrow p_{bp-mj} = \frac{R_{z1}}{d_{mj} \text{ x } l_{mj}} \leq p_{bp-m}$$

$$\Rightarrow p_{bp-mj} = \frac{3.59 \text{ x } 10^7 \text{ kN}}{64.4 \text{ x } 39.256} = 14.2 \text{ MPa}$$

The mean bearing pressure on the main journal (p_{bp-m}) for CI Engines is 3 to 7 MPa.

Case (ii): When the Crank is at the Maximum Twisting Moment position (θ ≈ 35^0 at TDC). The twisting moment on the crank pin (crankshaft) becomes maximum, when the tangential force (F_t) attains its maximum value. The crank angle (θ) corresponding to the maximum tangential force has to be determines by plotting the curves of F_t versusθ. The approximate value is 35° at TDC. At this position of crank angle, the connecting rod force (F_{con}), acting at the center of the crank pin (as shown in Figure 5.22), has two components (F_t and F_r). F_t is the tangential force, whereas F_r is a radial force.

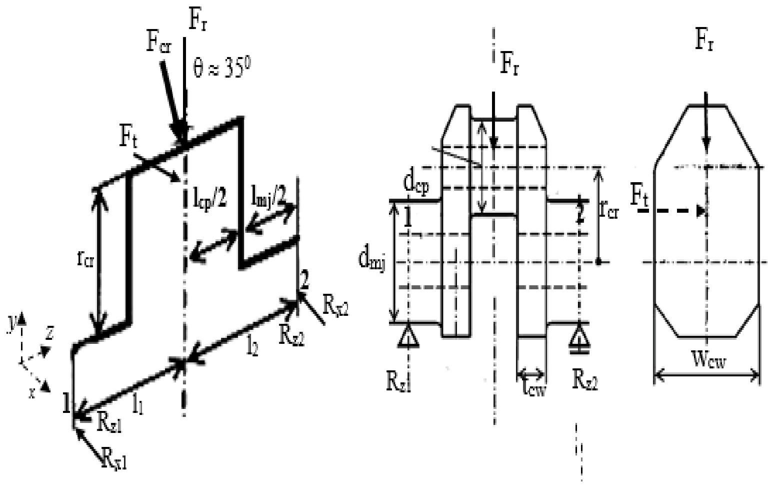


Figure 3. 37 Forces acting on the Crankshaft when the Crank is at the Maximum Twisting Moment position ($\theta \approx 35^0$ at TDC) and main Dimensions

The connecting rod force (F_{con}) at this position ($\theta \approx 35^0$ at TDC) is given by,

$$F_{con} = \frac{F_p}{\cos\phi} \quad \ldots\ldots (92)$$

$$F_{con} = \frac{5.619862 \text{ x } 10^7}{\cos(35)} = 6.8605847 \text{ x } 10^7 \text{ kN}$$

Where; $F_p = F_g \pm F_i = 7.166 \text{ x } 10^7 - 1.546238 \text{ x } 10^7 = 5.619862 \text{ x } 10^7$ kN

As shown figure 5.23 below Forces acting on the Crankshaft and the resulting Bending and Twisting Moment diagrams.

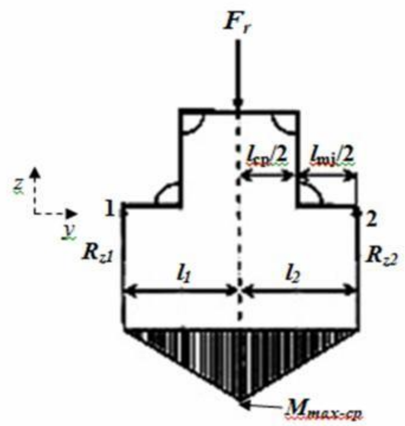


Figure 3. 38 Bending Moment Diagram (BMD) and Twisting Moment Diagram (TMD) Of the Crankshaft on x-y and x-z planes

The vertical reaction forces acting at the center of the main journals (at points 1 and 2) can be determined as,

$R_{z1} = \frac{F_r \, x \, l_2}{(l_1 + l_2)} = \frac{4.4509 \, x10^7 \, x \, 57.5}{(115)} = 2.22545 \, x10^7$ kN

$R_{z2} = \frac{F_r \, x \, l_1}{(l_1 + l_2)} = \frac{4.4509 \, x10^7 \, x \, 57.5}{(115)} = 2.22545 \, x10^7$kN

Where, $F_r = F_{con} \, x\cos(\theta + \phi) = 4.4509 \, x10^7$ kN

The horizontal reaction forces acting at the center of the main journals (at points 1 and 2) can be determined as,

$R_{x1} = \frac{F_t \, x \, l_2}{(l_1 + l_2)} = \frac{4.186 \, x10^7 \, x \, 57.5}{(115)} = 2.093 \text{ x } 10^7$ kN

$R_{x2} = \frac{F_t \, x \, l_1}{(l_1 + l_2)} R_{x2} = \frac{4.186 \, x10^7 \, x \, 57.5}{(115)} = 2.093 \text{ x } 10^7$ kN

Where; $F_t = F_{con} \, x\sin(\theta + \phi) = 4.186 \, x10^7$ kN

The maximum bending moment at the center of crank pin (M_{max-cp}) is given by,

$M_{max-cp} = R_1 \, x \, l_1 = 3.85 \, x10^{14} \text{ x } 57.5 = 2.2 \, x10^{16}$ kN-mm

Where; $R_1 = \sqrt{R^2{}_{x1} + R^2{}_{z1} - 2R_{x1}R_{z1}\cos(180 - 2\theta)}$

$R_1 = \sqrt{(2.09 \text{ x } 10^7)^2 + (2.225 \, x10^7)^2 - 2(2.09 \text{ x } 10^7 \, x2.225 \, x10^7)\cos(180 - 2\theta)}$

$= 3.85 \, x10^{14}$ kN

The maximum stress on the Crank Pin at the critical section and at the corresponding critical point can be determined as,

i) Based on Combined Stresses using:

- The bending stress on the Crank Pin (σ_{yb}) is,

$$\sigma_{yb} = \frac{M_{max-cp} \, x \, z}{I_{xx}} \leq \sigma_{all} \quad \text{.......... (92)}$$

$\sigma_{yb} = \frac{2.2 \, x10^{16} \, x \, 29.95}{627732.5} = 1.06$ MPa

Where; $M_{max-cp} = R_1 x l_1 = 2.2 \, x10^{16}$ kN-mm;

$z = \frac{d_{cp}}{2} = 29.9$ mm;

$I_{xx} = \frac{\pi \, x \, d^4{}_{cp}}{64} = 627732.5 \text{ mm}^4$

Since it is less than the allowable stress (241.1 MPa) the design is safe.

Hence, $\sigma_x = 0$; $\sigma_y = \sigma_{yb}$; $\tau_{xy} = 0$

- The shear stress on the Crank Pin (τ_{xy}) is,

$$\tau_{xy} = \frac{T_{max-cp} \times r_{cp}}{J} \leq \tau_{all} \quad \ldots\ldots (93)$$

$$\tau_{xy} = \frac{6.2685 \times 10^8 \times 29.9}{1255465} = 14.92 \text{ MPa}$$

Where; $T_{max-cp} = R_{x1} x r_{cr} = 2.093 \times 10^7 x\, 29.95 = 9.837 \times 10^8$ KN-mm

$r_{cp} = \frac{d_{cp}}{2} = 29.9$ mm;

$J = \frac{\pi \times d^4{}_{cp}}{32} = 1255465$ mm

Hence, $\sigma_x = 0$; $\sigma_y = \sigma_{yb}$; $\tau_{xy} = \tau_{xy}$

ii) Based on Bearing Pressure (p_{bp-cp})

$$F_r = p_{bp-cp} \times d_{cp} \times l_{cp} \quad \ldots\ldots (94)$$

$$\Rightarrow p_{bp-cp} = \frac{F_r}{d_{cp} \times l_{cp}} \leq p_{bp-m}$$

$$\Rightarrow p_{bp-cp} = \frac{4.4509 \times 10^7}{59.8 \times 46} = 16.18 \text{ MPa}$$

The mean bearing pressure on the crank pin (p_{bp-m}) for CI Engines is 8 to 16 MPa.

The maximum stress on the Crank Web at the critical section and at the corresponding critical point can be determined using:

i) Based on Combined Stresses using:

- The axial stress (Compressive) on the left side Crank Web (σ_{za}) is,

$$\sigma_{za} = \frac{R_{z1}}{w_{cw} \times t_{cw}} \quad \ldots\ldots (95)$$

$$\sigma_{za} = \frac{2.22545 \times 10^7}{96.6 \times 22.08} = 10.4338 \text{ MPa}$$

- The bending stresses on the left side Crank Web are, $\sigma_x = \sigma_{za} + \sigma_{zb}$ but σ_{zb} has two components σ_{zb-t} (Compressive) and σ_{zb-r} (Compressive).

$$\sigma_{zb-t} = \frac{M_{cw-t} \times X}{I_{yy}} = \frac{M_{cw-t}}{z_{yy}} \quad \ldots\ldots (96)$$

$$\sigma_{zb-t} = \frac{9.837 \times 10^8 \times 48.3}{1658628.8} = \frac{9.837 \times 10^8}{34340} = 28.645894 \text{ MPa}$$

Where; $M_{cw-t} = R_{x1} \times r_{cr} = 2.093 \times 10^7 x\;\, 47 = 9.837 \times 10^8$ kN-mm

$X = \frac{w_{cw}}{2} = 48.3$ mm;

$I_{yy} = \frac{t_{cw} \times w^3{}_{cw}}{12} = \frac{22.08 \times (96.6)^3}{12} = 1658628.8 \text{mm}^4$;

$$z_{yy} = \frac{I_{yy}}{x} = \frac{t_{cw}\ x\ w^2{}_{cw}}{6} = \frac{75120.898}{48.3} = \frac{22.08\ x\ (96.6)^2}{6} = 34340;$$

$$\sigma_{zb-r} = \frac{M_{cw-r}\ x\ y}{I_{xx}} = \frac{M_{cw-r}}{z_{xx}} \quad \text{...} \quad (97)$$

$$\sigma_{zb-r} = \frac{7.67625\ x10^8\ x\ 11.04}{86654.89} = \frac{7.67625\ x10^8}{7849} = 97.799 \text{ MPa}$$

Where; $M_{cw-r} = R_{z1}x(l_1 - \frac{l_{cp}}{2}) = 2.225\ x10^7 x\left(57.5 - \frac{46}{2}\right) = 7.67625\ x10^8$ KN-mm

$$y = \frac{t_{cw}}{2} = 11.04 \text{ mm};$$

$$I_{xx} = \frac{w_{cw}\ x\ t^3{}_{cw}}{12} = \frac{96.6\ x\ (22.08)^3}{12} = 86654.89 \text{ mm};$$

$$z_{xx} = \frac{I_{xx}}{y} = \frac{w_{cw}\ x\ t^2{}_{cw}}{6} = \frac{86654.89}{11.04} = \frac{96.6\ x\ (22.08)^2}{6} = 7849;$$

Therefore; $\sigma_x = \sigma_{za} + \sigma_{xb-t} + \sigma_{xb-r} = 10.4338 + 28.645894 + 97.799 = 136.878694$ MPa

Hence, $\sigma_x = \sigma_{za} + \sigma_{xb-t} + \sigma_{xb-r};\ \sigma_y = 0;\ \tau_{xy} = \tau_{xy} = 0$

- The shear stress on the left side Crank Web (τ_{xy}) is,

$$\tau_{xy} = \frac{T_{max-cw}\ x\ r_{cw}}{J} \leq \tau_{all} \quad \text{...} \quad (98)$$

$$\tau_{xy} = \frac{7.22\ x\ 10^8\ x\ 48.3}{1745283.69} = 19.99 \text{ MPa}$$

Where; $T_{max-cw} = R_{x1}x\left(l_1 - \frac{l_{cp}}{2}\right) = 2.093\ x\ 10^7\ x\left(57.5 - \frac{46}{2}\right) = 7.22\ x\ 10^8$ kN-mm

$$r_{cw} \approx \frac{w_{cw}}{2} = 48.3 \text{ mm};$$

$$J = I_{xx} + I_{yy} = 86654.89 + 1658628.8 = 1745283.69 \text{ mm}^4$$

- The stresses on the right side Crank Web can be determined in a similar way or the same size as the left side Crank Web can be taken, if both crank webs are equally loaded considering Engine Balancing. The maximum stress on the Main Journal at the critical section and at the corresponding critical point can be determined as,

i) Based on Combined Stresses using:

- The bending stress on the Main Journal (σ_{yb}) is,

$$\sigma_{yb} = \frac{M_{max-mj}\ x\ z}{I_{xx}} \leq \sigma_{all} \quad \text{...} \quad (99)$$

$$\sigma_{yb} = \frac{5.22\ x10^8\ x\ 32.2}{I_{xx}} = 19.9 \text{ MPa}$$

Where; $M_{max-mj} = R_{z1}x\left(l_1 - \frac{l_{cp}}{2} - \frac{t_{cw}}{2}\right) = 2.23\ x10^7 x\left(57.5 - \frac{46}{2} - \frac{22.08}{2}\right) = 5.22x10^8$ kN-mm

$$z = \frac{d_{mj}}{2} = \frac{64.4}{2} = 32.2 \text{ mm};$$

$$I_{xx} = \frac{\pi \times d^4{}_{mj}}{64} = 844332.2312 \text{ mm}^4$$

Since it is less than the allowable stress (241.1 MPa) the design is safe.

Hence, $\sigma_x = 0$; $\sigma_y = \sigma_{yb}$; $\tau_{xy} = \tau_{xy}$

- The shear stress on the Main Journal (τ_{xy}) is,

$$\tau_{xy} = \frac{T_{max-mj} \times r_{mj}}{J} \leq \tau_{all} \quad \text{.......... (100)}$$

Where; $T_{max-mj} = \frac{F_t}{2} \times r_{cr} = \frac{4.186 \times 10^7}{2} \times 47 = 9.8371 \times 10^8$ kN-mm

$$r_{mj} = \frac{d_{mj}}{2} = 32.2 \text{ mm}$$

$$J = I_{xx} + I_{yy}$$

ii) Based on Bearing Pressure (P_{bp-cp})

$$R_{z1} = P_{bp-cp} \times d_{mj} \times l_{mj} \quad \text{.......... (101)}$$

$$\Rightarrow P_{bp-cp} = \frac{R_{z1}}{d_{mj} \times l_{mj}} \leq P_{bp-m}$$

$$\Rightarrow P_{bp-cp} = \frac{2.225 \times 10^7}{64.4 \times 39.256} = 8.8 \text{ MPa}$$

The mean bearing pressure on the main journal (P_{bp-m}) for CI Engines is 3 to 7 MPa.

Solid work model is shown in figure 3.39 below.

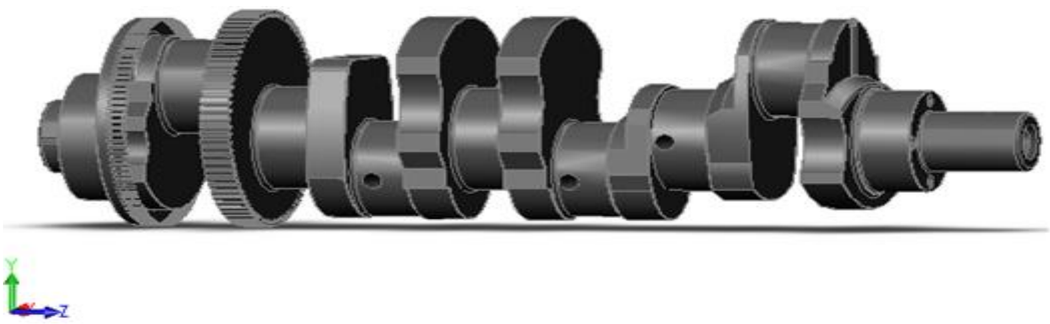


Figure 3. 39 Three Dimensional Sketch of Crankshaft

3.3.3.8 Camshaft

Function: The main functions of a Camshaft is to open and close the Intake and Exhaust Valves by means of Cams according to the engine Valve Timing and Firing Order sequences. In addition, a Camshaft can also be used to drive some component parts of engine systems such as: Ignition Distributor, Oil Pump, Mechanical Fuel Pump, etc. It receives power from the Crankshaft by means of a Gear or a Chain or a Timing or Toothed Belt drives.

Construction: A Camshaft is constructed to possess: Journals (Bearings), Cams, Oil Holes and Grooves, Keyway, Drive Gear (for Ignition Distributor and Oil Pump), Eccentric (for Mechanical Fuel Pump) as shown in figure 3.40 below.

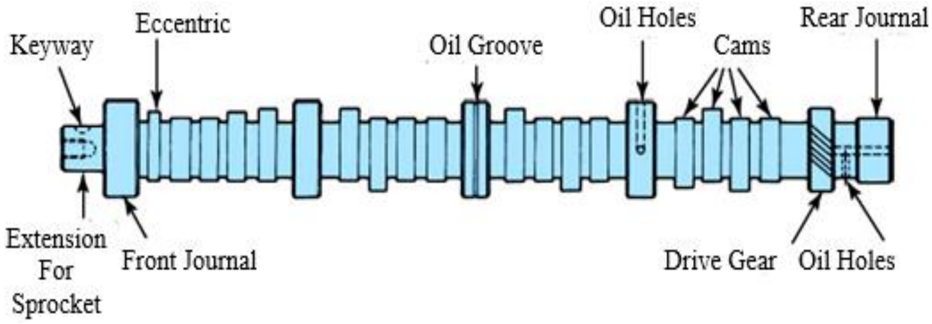


Figure 3. 40 Construction of a Camshaft [37]

Design consideration for camshaft: The Camshaft is mainly subjected to a combined; Bending, Twisting (Torsional), Bearing, fatigue (Cyclic) types of stresses due to Inertia Forces of the valve gear component parts, Gas Force, Various Forces from the Drives and Rotational motion. Hence, the design has to consider the effect of all types of stresses.

Note: The Camshaft is designed to rotate at half the Crankshaft speed (N).

Material and Manufacturing Processes: The choice of material for the Camshaft should mainly consider: Mechanical Properties (such as: Strength, Torsional and Lateral Rigidity, Fatigue and Wear resistance, etc.), Cost and Ease of Manufacturing Processes. Accordingly Camshafts for In IC Engines are usually made from Alloy Steel by Forging or Cast Iron by Casting.

Size determination camshaft

This maximum force (F_e) exerted on the cam by the exhaust valve at the beginning of valve opening is as shown in Figure 3.41 below.

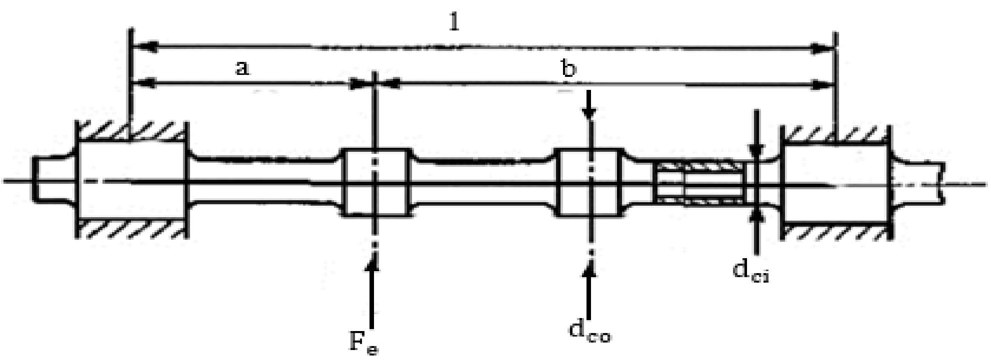


Figure 3. 41 Maximum Force acting on the Camshaft [37]

Camshaft Outer Diameter (d_{co})

i) Based on Rigidity

a) Torsional Rigidity: The permissible angle of twist (θ) should not exceed 0.3 deg/m.

For a hollow circular shaft.

$$\theta = \frac{584 \times T \times l}{G \times (d^4{}_{co} - d^4{}_{ci})} \quad \text{......} \quad (102)$$

For a solid circular shaft.

$$\theta = \frac{584 \times T \times l}{G \times d^4} \quad \text{......} \quad (103)$$

Where; $\theta =$ Twist angle (deg.)

$T =$ Twisting (Torsional) Moment (Nm)

$l =$ Camshaft length (0.489 m)

$G =$ Rigidity Modulus (N/m^2)

d_{co} = Camshaft outer diameter (m)

d_{ci} = Camshaft inner diameter (m)

b) Lateral Rigidity: The permissible deflection (y) should not exceed 0.02 to 0.05 mm.

$$y = 0.8 \times F_e \times \frac{a^2 \times b^2}{E \times l \times (d^4{}_{co} - d^4{}_{ci})} \quad \text{......} \quad (104)$$

Where; F_e = Maximum force exerted on the cam by exhaust valve

E = Elasticity Modulus (N/mm^2)

a and b = space available ($a \approx 25$ mm and $b = 70$ mm)

$d_{ci} \approx 0.3 d_{co}$ = 8.226 mm

Solid work model is shown in figure 3.42 below.

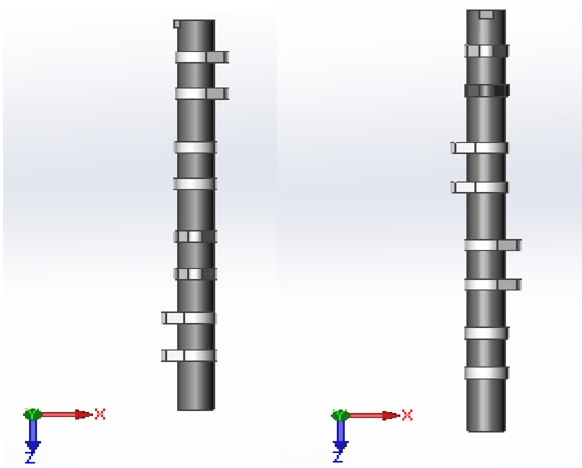


Figure 3. 42 Three Dimensional Sketch of Camshaft

3.3.3.9 Cam

Function: The main function of a Cam is to convert the rotary motion of the camshaft into reciprocating motion. In doing so, it opens and closes the valves (together with the Valve Assembly) according to the engine Valve Timing and Firing Order sequences. It also limits the maximum valve lift (h), duration and also the opening and closing points of valves.

Construction: A Cam is constructed to possess: a Base Circle, Heel or Foot, Timing Points, Trailing and Leading Flanks, Lobe or Nose, Lift, Width (W_{cam}), Duration or Cam angle (θ_{dvo}) as shown in figure 3.43 below.

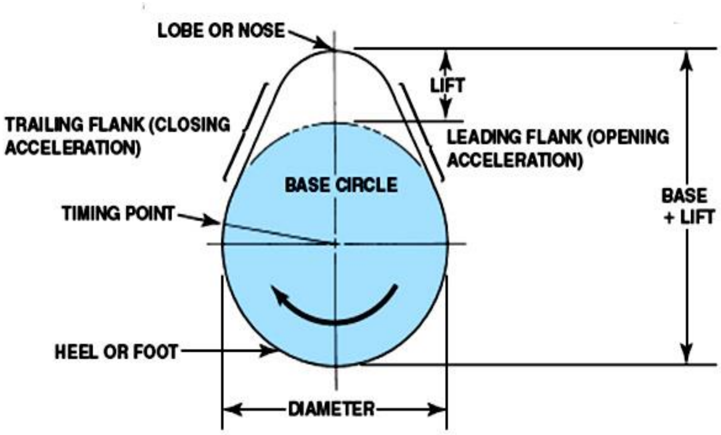


Figure 3. 43 Construction of a Cam [37]

Design consideration for cam: The Cam for a Conventional Camshaft has to be designed to provide a maximum volumetric efficiency at the point of maximum brake mean effective pressure ($bmep_{max}$) or at the point of maximum brake torque (T_{b-max}) on the Engine Performance Curves. It is designed to rotate at half the Crankshaft speed (N). During engine operation, the surface (width) of the Cam is subjected to bearing pressure which tends to cause surface wear.

Material and Manufacturing Processes: In IC engines, a Cam is manufactured as one piece or an integral part of a Camshaft. It is manufactured from Alloy Steel by Forging.

Size determination

As the rocker arm for exhaust valve is most heavily loaded than for the inlet valve, it is preferable to design for the exhaust valve first and then the same dimension can be taken for the intake considering the cost and ease of manufacturing processes.

The camshaft diameter (D_{cs}) can be determined empirically as, $D_{cs} = 0.16D + 12.7 = 27.42$ mm

The base circle diameter (D_{cam}) can be taken about 3 mm greater than the camshaft diameter.

$D_{cam} = D_{cs} + 3 = 30.42$ mm

The cam width (W_{cam}) can also be determined empirically as, $W_{cam} = 0.09D + 6 = 14.28$ mm

Assuming the rocker arm as a Flat Follower, the maximum lift of the follower (h_f) is given by,

$$h_f = h \times \frac{l_f}{l_v + l_f} \quad \text{(105)}$$

$$h_f = 28.9877 \times \frac{1.3}{1 + 1.3} = 16.38435 \text{ mm}$$

Where; h = maximum valve lift (28.9877 mm).

Cam Width (W_{cam}): The Cam Width can be taken as about 20 to 25 mm, but it has to be checked for Bearing Pressure. The Allowable Bearing Pressure can be 400 to 1200 MPa.

3.3.3.10 Rocker Arm

Function: The main function of the rocker arm is a kind of lever which is used to transmit force and motion from the follower or the cam to the valve or vice versa.

Construction: A typical rocker arm for operating the exhaust valve is shown in figure 3.44. The lever ratio a/b is generally decided by considering the space available for rocker arm. For high speed engines, the ratio a/b is taken as 1/1.3. The various force acting on the rocker arm of exhaust valve are the gas load, spring force and force due to valve acceleration.

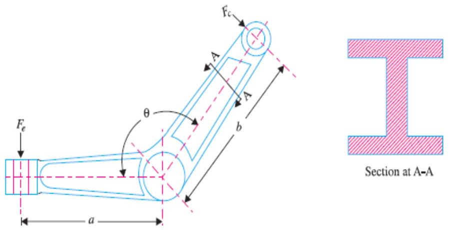


Figure 3. 44 Rocker Arm for Exhaust Valve [37]

Design consideration for rocker arm: In order to reduce inertia of the rocker of the rocker arm, an I-section is used for the high speed engines and it may be rectangular section for low speed engines. In four stroke engines, the rocker arms for the exhaust valve is the most heavily loaded. Though the force required to operate the inlet valve is relatively small, yet it is usual practice to make the rocker arm for the inlet valve of the same dimensions as that for exhaust valve. A rocker arm is mainly subjected to inertia and bending stresses. In order to get higher strength or to reduce the combined stresses, an I-section is used for high and medium speed engines.

Material and Manufacturing Processes: A rocker arm is made from: cast iron, cast steel or malleable iron by casting, forging, etc.

Size determination

As the rocker arm for exhaust valve is most heavily loaded than for the inlet valve, it is preferable to design for the exhaust valve first and then the same dimension can be taken for the inlet valve considering the cost and ease of manufacturing processes (mass production).

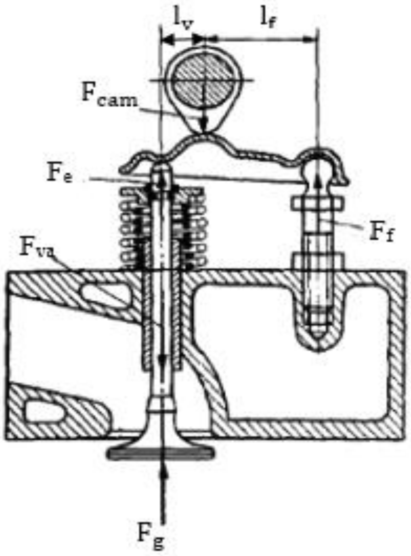


Figure 3. 45 Valve gear Mechanisms of OHC Engines [37]

Maximum load on the rocker arm for exhaust valve,

$$F_e = F_g + F_s + F_{va} \quad \ldots \quad (109)$$

$F_e = 5.5 \times 10^5 + 4.87 \times 10^6 + 1.9 \times 10^6 = 7.32 \times 10^6$ kN

Maximum load on the rocker arm for inlet valve is

$$F_i = F_s + F_{va} \quad \ldots \quad (110)$$

$F_i = 5.5 \times 10^5 + 1.9 \times 10^6 = 2.45 \times 10^6$ kN

Diameter of the Fulcrum Pin (d_1), the resultant reaction force acting on the fulcrum pin (R_p) as,

$$R_p = \sqrt{F^2{}_e + F^2{}_f - 2F_eF_f\cos(180-\theta)} \quad \ldots \quad (111)$$

$$= \sqrt{(7.32 \times 10^6)^2 + (5.63 \times 10^6)^2 - 2 \times 7.32 \times 10^6 \times 5.63 \times 10^6 \cos(80)} = 8.425\times10^6 \text{ kN}$$

Where; $F_e = F_s + F_g + F_{va}$

$$F_f = F_e \times \frac{l_v}{l_f} = 5.63 \times 10^6 \text{ kN}$$

The bearing pressure on the fulcrum pin (p_{bp-p}) is given by,

$$p_{bp-p} = \frac{R_p}{l_1 \times d_1} \quad \text{.......... (112)}$$

$$\Rightarrow d_1 = \frac{R_p}{l_1 \times d_{bp-p}} = 11.61 \text{ mm}$$

Where; p_{bp-p} = bearing pressure ($\approx$ 4 to 6 MPa)

l_1 = Fulcrum pin Length ($\approx 1.25 d_1$)

d_1 = Fulcrum pin diameter (mm)

- Check for the shear stress on the fulcrum pin (τ_p).

Note: the pin is in double shear.

$$\tau_p = \frac{R_p}{2A_s} \quad \text{.......... (113)}$$

$$\tau_p = \frac{8.425 \times 10^6}{2 \times 1.05865} = 3.979 \times 10^6 \frac{\text{kN}}{\text{mm}^2}$$

Where; $A_s = \frac{\pi}{4} \times d_1{}^2 = 105.865 \text{ mm}^2$

External Diameter of the Boss (D_1), $D_1 = 2d_1$ = 23.22 mm

Internal Diameter of the Boss (d_h) as shown in figure 5.31 below, Assuming a phosphor bronze bush of 3 mm thick, the internal diameter of the boss is, $d_h = d_1 + (2 \times 3)$ = 17.61 mm

- Check boss section for bending moment,

$$\sigma_b = \frac{My}{I} \quad \text{.......... (114)}$$

Where; $M = F_e x l_v =$

$y = \frac{D_1}{2} = 11.61$ mm

$I = \frac{1}{12} x l_1 x [D_1{}^3 - d_h{}^3] = 8536.272 \text{ mm}^4$

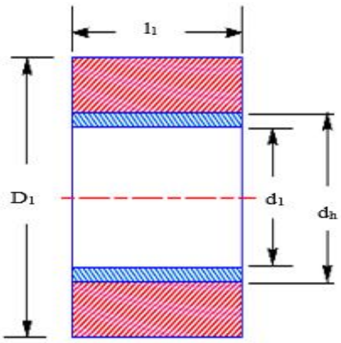


Figure 3. 46 Cross-section of the Boss [37]

Rocker Arm Cross-section and Dimension

The type of rocker arm cross-section for a medium and high speed engines is I-section as shown in figure 3.47. Assuming an allowable bending stress for the rocker are material (σ_{b-arm}) about 50 to 70 MPa, the thickness (t) and other dimensions of the cross-section can be determined as,

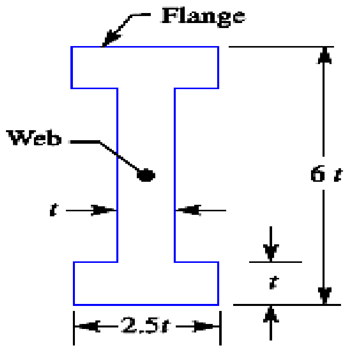


Figure 3. 47 Cross-section of the Rocker Arm [37]

$$\sigma_{b-arm} = \frac{My}{I} \quad \text{......} \quad (115)$$

Where; $M = F_e x \left(l_v - \frac{D_1}{2}\right)$;

$y = \frac{6t}{2} = 9$ mm

$I = \frac{1}{12}[2.5t\,x(6t)^3 - 1.5t\,x\,(4t)^3] = 35964\ mm^4$

Since the maximum load on the rocker arm for exhaust valve is more than that of inlet valve, therefore, the rocker arm must be designed on the basis of maximum load on the rocker arm for exhaust valve, as discussed below:

1. Design for fulcrum pin. The load acting on the fulcrum pin is the total reaction (R_F) at the fulcrum point.

$$R_F = d_1.l_1.p_b \quad \text{......} \quad (116)$$

$R_F = 1.161\ x\ 1.45125\ x\ 6 = 10.109$ N

Where; d_1 = Fulcrum pin diameter (mm) and

l_1 = Fulcrum pin length (mm)

The ratio of l_1/d_1 is taken as 1.25 and the bearing pressure (p_b) for ordinary lubrication is taken from 3.5 to 6 N/mm^2 and it may go up to 10.5 N/mm^2 for forced lubrication. The pin should be checked for the induced shear stress. The thickness of the phosphor bronze bush may be taken from 2 to 4 mm. the outside diameter of the boss at the fulcrum is usually taken twice the diameter of the fulcrum pin.

2. Design for forked end. The forked end of the rocker arm carries a roller by means of a pin. For uniform wear, the roller should revolve in the eyes. The load acting on the roller pin is F_c.

$F_c = d_2 . l_2 . p_b$ ………………………………………………………………………...…. (117)

Where; d_2 = Roller pin diameter (mm) and

l_2 = Roller pin length (mm)

The ratio of l_2/d_2 may be taken as 1.25. The roller pin should be checked for induced shear stress. The roller pin is fixed in eye and the thickness of each eye is taken as half the length of the roller pin. Therefore, thickness of each eye = $l_2/2$

The radial thickness of eye (t_3) is taken as $d_1/2$ = 0.5805 mm. Therefore, overall diameter of the eye,

$D_1 = 2d_1$ = 2.322 mm

The outer diameter of the roller is taken slightly larger (at least 3 mm more) than the outer diameter of the eye. A clearance of 1.5 mm between the roller and the fork on either side of the roller is provided.

3. Design for rocker arm cross-section. The rocker arm may be treated as a simply supported beam and loaded at the fulcrum point. We have already discussed that the rocker arm is generally of I-section. Due to the load on the valve, the rocker arm is subjected to bending moment.

M = σ_b x l …………………………………………………………………...…………. (118)

Where; σ_b = Permissible bending stress, and

l = Each rocker arm effective length

Bending moment,

M = σ_b x Z ………………………………………….…………...……………….. (119)

Where; Z = Section modulus.

From equations (118) and (119), the value of Z is obtained and thus the dimensions of the section are determined.

4. Design for tappet. The tappet end of the rocker arm is made circular to receive the tappet which is a stud with a lock nut. The compressive load acting on the tappet is the maximum load on the rocker arm for the exhaust valve (Fe).

$F_e = \frac{\pi}{4}(d_c)^2\sigma_c$ ………………………………………………………………..………….. (120)

$\Rightarrow d_c$ = 4.31 mm

Where; d_c = Tappet core diameter (mm), and

σ_c = Permissible compressive stress (50 MPa for mild steel)

From this expression, the core diameter of the tappet is determined. The outer or nominal diameter of the tappet (d_n) is given as $d_n = d_c/0.84$ = 5.13 mm

The diameter of the circular end of the rocker arm (D_3) and its depth (t_4) is taken as twice the nominal diameter of the tappet (d_n), i.e. $D_3 = 2d_n$ and $t_4 = 2d_n$ = 10.27959 mm

5. Design for valve spring. The valve spring is used to provide sufficient force during the valve lifting process in order to overcome the inertia of valve gear and to keep it with the cam without bouncing. The spring is generally made from plain carbon spring steel. The total load for which the spring is designed is equal to the sum of initial load and load at full lift.

$$W = W_1 + W_2 \quad \text{.......... (121)}$$

Where; W_1 = Spring initial load (force on the valve tending to draw it in to the cylinder on suction)

W_2 = Load at full lift (Full lift x spring stiffness)

Solid work model is shown in figure 3.48 below.

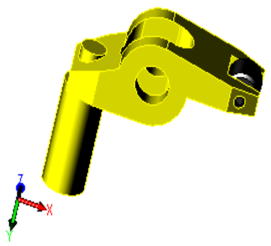


Figure 3. 48 Three Dimensional Sketch of Rocker Arm

3.3.3.11 Valve

Function: The main function of the valve is to let the fresh charge into the engine cylinder (Intake Valve) and to let the burned gases out of the engine cylinder (Exhaust Valve).

Construction: The head and face of the valve are separated by a small margin (≈ 1.5 to 2.0 mm thick) to avoid sharp edge of the valve and also to perform regrinding of the face. The valve face angle is chosen for the best possible compromise between valve opening and valve sealing. Valve opening is maximum when the face angle is zero. This increases the volumetric efficiency, but on the other hand, this causes poor valve sealing which results valve burning. In order to compromise these two opposing effects, valves are designed with a valve seat angle of 30^0 or 45^0. Furthermore, there will be about 1^0 angle difference (interference angle) between the valves seat and face. This provides

- An effective seat or increase the sealing effect,
- A knife-edge contact between the valve face and valve seat so as to minimize carbon deposit formation around the valve seat.

A Poppet Valve is widely used in IC engines, due to its reasonable weight, good strength and good heat transfer characteristics. It is constructed to possess: Head, Margin, Face Angle (Tapped to ≈ 30^0 to 45^0), Stem, Locks Groove and Tip as shown in figure 3.49 below.

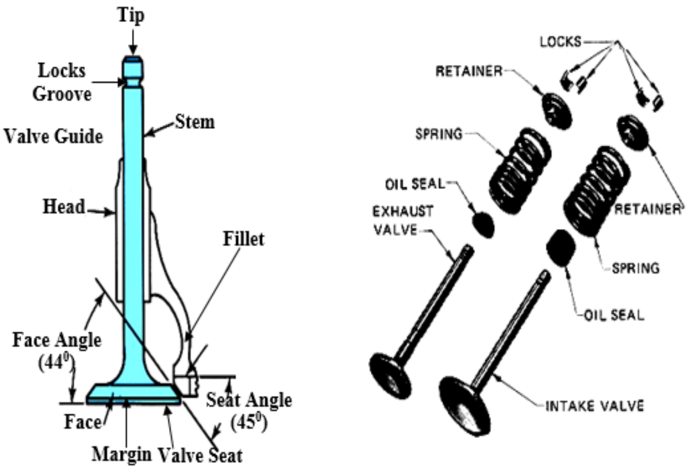


Figure 3. 49 Construction and Assembly of a Poppet Valve [37]

Design consideration for valve: both the inlet and exhaust valves are subjected to thermal and mechanical loads. In addition, these loads tend to cause fatigue or cyclic type of stresses. It is, therefore, necessary that the material of the valves should withstand these combined stresses.

Material and manufacturing processes: Since both the inlet and exhaust valves are subjected to high temperatures of 1930℃ to 2200℃ during the power stroke, therefore, it is necessary that the material of the valves should withstand these temperatures. Thus the valve material should possess: good heat conductivity, heat resistant, corrosion resistant, wear resistant, shock resistant, etc. in order to attain such physical and mechanical properties, Nickel Chromium Alloy is used for inlet valve and a Special Alloy of Silicon and Chromium (Silichrome) is used for exhaust valve as it is subjected to more thermal load. But this paper uses alloy steel. The type of manufacturing processes to be used is forging, casting, grinding, welding, etc.

Size determination

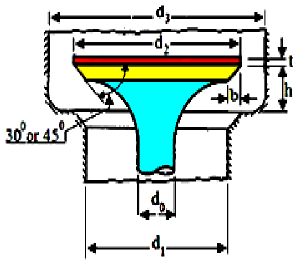


Figure 3. 50 Geometry of Poppet or Mushroom Valve [37]

In designing a valve assembly, it is required to determine the following dimensions such as piston area, valve port diameter, the velocity of gas through the port, the mean piston velocity, valve face angle thickness of valve disc maximum valve lift, and etc… as shown in the above figure 5.35.

Piston area (α_p),

$$\alpha_p = \frac{\pi D^2}{4} \quad \text{.....................} \quad (122)$$

$$\alpha_p = \frac{\pi \times 92^2}{4} = 6641\text{mm}^2$$

Valve Port Diameter (d_{vp} or d1),

$$d_{vp} = \sqrt{\frac{4 \times \alpha_p}{\pi}} \quad \text{.....................} \quad (123)$$

$$d_{vp} = \sqrt{\frac{4 \times 6641}{\pi}} = 91.99 \text{ mm}$$

The mean velocity of the fluid (gas) through the port (v_1) is limited by the local Sonic velocity (α_s).

$$\alpha_s = \sqrt{\gamma RT} \quad \text{.....................} \quad (124)$$

$$\alpha_s = \sqrt{1.35 \times 286 \times 323} = 353 \text{ m/s}$$

Where; $\gamma = \frac{C_p}{C_v}$

$R = C_p - C_v$ and,

$T =$ Local gas temperature (K)

The velocity of gas through the port (v_1),

$$v_1 = Z \times \alpha_s \quad \text{.....................} \quad (125)$$

$$v_1 = 0.55 \times 353 = 85 \text{ m/s}$$

Where; $Z =$ Mach index

The mean velocity of the gas (v_1) taken from table 3.10.

Table 3. 10 Mean Velocity of the Gas (v_1)

Type of engine	Mean velocity of the gas (v_1) m/s	
	Inlet valve	Exhaust valve
Low speed	33 – 40	40 – 50
High speed	80 – 90	90 – 100

Source: The mean velocity of gas (v_1) should be checked using **Design Handbooks**

Based on experimental data, the graph of Volumetric Efficiency (η_v) as a function of Mach Index (Z) is as shown in Figure 3.51. The graph shows that for various valve lift (h) and port diameter (d_{vp}), there is a particular value of Z to obtain a maximum volumetric efficiency, after which the volumetric efficiency (η_v) starts to drop. This value of Mach Index Z during engine design will be 0.5 to 0.6.

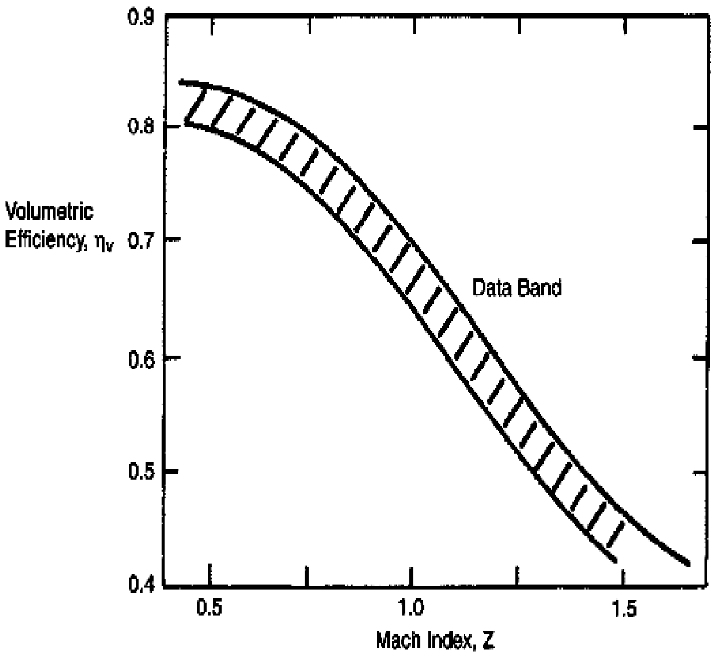


Figure 3. 51 Volumetric Efficiency (η_v) as a Function of Mach Index (Z) [37]

The mean piston velocity (v_{mp}),

$$v_{mp} = \frac{2 \times S \times N_{max}}{60} \quad \text{.....} \quad (126)$$

$$v_{mp} = \frac{2x0.094x4000}{60} = 12.53 m/s$$

Neglecting losses and assuming the fluid (gas) is incompressible, the volume flow rate of the fluid passing through the valve port is equal to the fluid (gas) passing through the cylinder bore. Hence, from continuity equation (fluid mechanics), valve port area (α_1)

$$\alpha_1 = \frac{\alpha_p \cdot v_{mp}}{C_i v_1} \quad \longrightarrow \rho = \text{constant} \quad \text{.....} \quad (127)$$

Where; C_i = the average flow coefficient (0.15 to 0.35).

Area of inlet valve port

$$\alpha_1 = \frac{\alpha_p \cdot v_{mp}}{C_i v_1} = \frac{6641x12.53}{0.25x85} = 3915.85 mm^2$$

Area of exhaust valve port

$$\alpha_1 = \frac{\alpha_p \cdot v_{mp}}{C_i v_1} = \frac{6641x12.53}{0.25x95} = 3503.65\text{mm}^2$$

Note: The Inlet Port should be made 20 to 40 % larger than the Exhaust Port for better cylinder filling.

Valve Face Angle (a_v), the valve face angle can be taken as 30^0 or 45^0 . I have taken the average of the two. i.e. 37.5^0.

Thickness of the valve disc (t), may be determined empirically from the following relation,

$$t = k\, d_{vp} \sqrt{\frac{p_{g-max}}{\sigma_b}} \quad \text{.......} (128)$$

$$t = 0.42 \text{ x } 70.60 \sqrt{\frac{107.8 \text{ x } 10^5}{120}} = 8.887 \text{ mm}$$

Where; k = Constant (0.42 for steel, 0.54 for cast iron)

d_{vp} = Valve port diameter (mm)

σ_b = Permissible bending stress (100 to 120 MPa for alloy steel)

Maximum Valve Lift (h_{max}), In order to reduce the pressure drop and air restriction across the valve and port, the total area of the valve opening should be at least equal to that of the throat area considering an allowance for valve stem. The lift of the valve may be obtained by equating the area across the valve seat to the area of the port. For a conical valve as,

$$\pi\, d_{vp}.h \cos a_v = \frac{\pi}{4}(d_{vp})^2 \quad \text{.......} (129)$$

$$\Rightarrow h = \frac{d_{vp}}{4 \cos a_v} = \frac{91.99}{4 \cos 37.5} = 28.9877 \text{ mm}$$

Where; a_v = valve seat tapered angle (30^0 to 45^0.deg)

In case of flat heated valve, the lift of valve is given by (In this case, $\alpha = 0^0$)

$$h = \frac{d_{vp}}{4} = 22.99 \text{ mm}$$

The valve seats usually have the same angle as the valve seating surface. But it is preferable to make the angle of valve seat $1/2^0$ to 1^0 larger than the valve angle as shown in figure 3.52. This result in more effective seat.

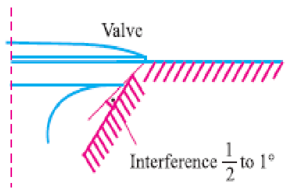


Figure 3. 52 Valve Interference Angle [37]

a) Margin Thickness (t), The Margin thickness (t) will be 1.5 mm to 2.0 mm. I have taken 2 mm.

d) Valve Face Width (b), the width of the valve face can be determined using the following relation,

$$b = \left(\frac{k \times d_{vp}}{\tan a_v}\right) \times \sqrt{\frac{P_{g-max}}{\sigma_{all}}} \quad \ldots\ldots (130)$$

$$b = \left(\frac{0.42 \times 91.99}{\tan 37.5}\right) \times \sqrt{\frac{107.8 \times 10^5}{103}} = 16.2888 \text{ mm}$$

Where; σ_{all} = allowable (bending) stress (103 MPa for high grade steel).

e) Valve Head Diameter (d_v or d2), the valve head diameter is given by,

$$d_v = d_{vp} + 2b \quad \ldots\ldots (131)$$

d_v = 91.99 + 2 x 16.2888 = 124.5676 mm

f) Valve Stem Diameter (d_s or d0), the valve stem diameter is given by,

$$d_s = \frac{d_{vp}}{8} + 6.35 \text{ to } \frac{d_{vp}}{8} + 11 \quad \ldots\ldots (132)$$

$d_s = \frac{124.5676}{8} + 6.35 \text{ to } \frac{124.5676}{8} + 11$ mm = 21.92 to 26.57 mm

The valve is subjected to spring force which is taken as concentrated load at the center. Due to this spring force (F_s), the stress in the valve (σ_t) is given by,

$$\sigma_t = \frac{1.4\, F_s}{t^2}\left(1 - \frac{2 d_s}{3 d_p}\right) \quad \ldots\ldots (133)$$

$$\sigma_t = \frac{1.4 \times 5.5 \times 10^5}{2^2}\left(1 - \frac{2 \times 24}{3 \times 91.92}\right) = 0.2 \text{ MPa}$$

The average volume flow rate ($\dot{V}_{ave}$),

$$\dot{V}_{ave} = \frac{\frac{\pi}{4} \times D^2 \times S}{t_{dvo}} = \frac{360^0 \times \pi \times D^2 \times h \times N_{@Tb-max}}{4 \times 60 \times \theta_{dvo}} \quad \ldots\ldots (134)$$

$$\dot{V}_{ave} = \frac{\frac{\pi}{4} \times 92^2 \times 94}{0.0068} = \frac{360^0 \times \pi \times 92^2 \times 28.9877 \times 2000}{4 \times 60 \times 21061} = 109794.74 \text{ mm}^3/\text{s}$$

The flow area (a_1),

$$a_1 = C_i \times \pi \times d_{vp} \times h \times z_v \quad \ldots\ldots (135)$$

$a_1 = 0.25 \times \pi \times 91.99 \times 28.9877 \times 2 = 4188.65$ mm^2

The velocity of gas through the port (V_1) is given by,

$$V_1 = \frac{\dot{V}_{ave}}{a_1} \quad \ldots\ldots (136)$$

$$V_1 = \frac{109794.74 \text{ mm}^3/\text{s}}{4188.65 \text{ mm}^2} = 26.2 \text{ mm/s}$$

The duration of valve opening (θ_{dvo}),

$$\theta_{dvo} = \frac{1.5 \text{ x } D^2 \text{ x } S \text{ x } N_{@Tb-max}}{C_i \text{ x } V_1 \text{ x } d_{vp} \text{ x } h \text{ x } z_v} \quad \text{.......... (137)}$$

$$\theta_{dvo} = \frac{1.5 \text{ x } 92^2 \text{ x } 94 \text{ x } 2000}{0.25 \text{ x } 85 \text{ x } 91.99 \text{ x } 28.9877 \text{ x } 2} = 21061 \text{ rad/s}$$

Where; z_v = the number of intake or exhaust valves per cylinder.

The valve opening and closing angles before and after TDC or BDC (θ_{vb} andθ_{va}) can be determined as,

$$\theta_{dvo} = \theta_{vb} + 180^0 + \theta_{va} \quad \text{.......... (138)}$$

$$\theta_{dvo} = 81.6 + 180^0 + 96.4 = 358^{\circ}$$

Duration of Valve Opening *(θ_{dvo})*, the total duration of valve opening in second (t_{dvo}) can be expressed as,

$$t_{dvo} = \frac{60 \text{ x } \theta_{dvo}}{N_{@Tb-max} \text{ x } 360^0} \quad \text{.......... (139)}$$

$$t_{dvo} = \frac{60 \text{ x } 358}{2000 \text{ x } 360^0} = 0.0068 \text{ second}$$

Solidwork model is shown in figure 3.53 below.

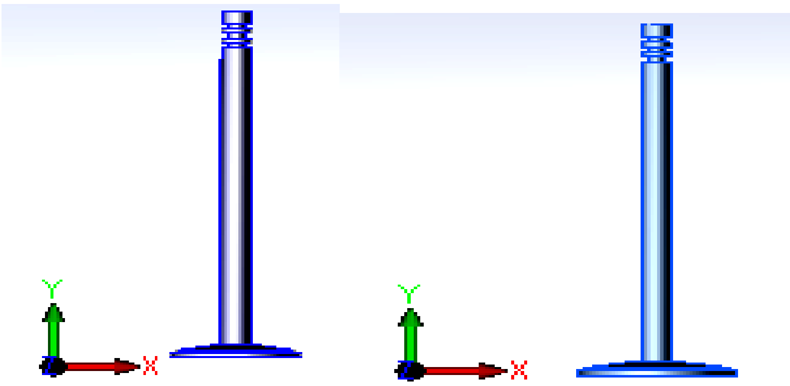


Figure 3. 53 Three Dimensional Sketch of Intake and Exuast Valve

3.3.3.12 Valve Spring

Function: The main function of the valve spring is to close the valve and also to provide a good contact between the tappet and the cam. Since the spring is subject to compressive loads, it is preferred to be ground flat at each end to ensure an even pressure distribution.

Construction: The types of valve springs commonly used in IC engines are as shown in figure 3.54 below.

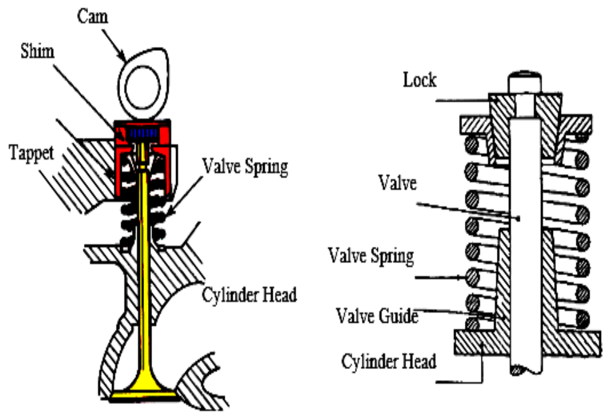


Figure 3. 54 Assembly and Construction of a Valve Spring [37]

Design consideration for valve spring: The valve spring must provide:

i) a gas tight seal between the valve and its seat,

ii) a continuous kinematic contact among the component parts (Valve, Tappet, Cam, etc.).

The exhaust valve is to be designed first, as it is more heavily loaded. The inlet valve can also be designed separately or a proportional dimension of the exhaust valve can be considered.

Material and Manufacturing Processes: The valve spring is generally made from plain carbon steel.

Size determination

The various forces or loads on the exhaust valve are initial spring force (F_s), gas force (F_g) (when the exhaust valve is opened), and inertia force due to valve acceleration (F_{va}). And the various forces or loads on the inlet valve are initial force (F_s), and inertia force due to valve acceleration (F_{va}).

Initial spring force (F_s),

$$F_s = \frac{\pi}{4} \times d_v{}^2 \times p_s + w_v \quad \text{............} (140)$$

$F_s = \frac{\pi}{4}(124.5676)^2 \times 0.45 \times 10^5 + 0.2 \times 9.81 = 5.5 \times 10^5$ kN

Where; p_s = Suction pressure (≈ 0.3 to 0.6 bar)

w_v = Valve and its associated parts weight ($m_v \times g$)

Where; m_v = Valve and its associated parts mass (0.1 to 0.3 kg)

Gas force (F_g) (when the exhaust valve is opened),

$$F_g = \frac{\pi}{4}(d_v)^2 p_c - w_v \quad \text{............} (141)$$

$F_g = \frac{\pi}{4}(124.5676)^2 \times 4 \times 10^5 - 0.2 \times 9.81 = 4.87 \times 10^6$ kN

Where; p_c = Cylinder gas pressure as the exhaust valve is opened (≈ 4 bar)

The valve acceleration,

$$a_v = \omega^2 x\, r_h = \left(\frac{2\pi}{t_{cam}}\right)^2 x\, r_h \qquad (142)$$

$$a_v = \omega^2 x\, r_h = \left(\frac{2\pi}{0.00024376}\right)^2 x\ 14.49385 = 9.6\ x\ 10^6\ m/s^2$$

Where; $r_h = \frac{h}{2}$ = 14.49385 mm and,

$$t_{cam} = \frac{\text{Duration of valve Opening}(\theta_{dvo})}{\text{Angle Turned by the Camshaft}(\theta_{cam})} = \frac{\theta_{dvo}}{\frac{N\ x\ 360^\circ}{2\ x\ 60}} = 0.00024376\ s$$

Time taken by the camshaft for a uniform or constant valve acceleration (a_v) (for a four-stroke engine).

Inertia force due to valve acceleration (F_{va}),

$$F_{va} = m_v\ x\ a_v \qquad (143)$$

$$F_{va} = 0.2\ x\ 9.6\ x\ 10^6 = 1.9\ x\ 10^6\ kN$$

Hence, the total force or load on the exhaust valve (F_e) is given by,

$$F_e = F_s + F_g + F_{va} \qquad (144)$$

$$F_e = 5.5\ x\ 10^5 + 4.87\ x\ 10^6 + 1.9\ x\ 10^6 = 7.32\ x\ 10^6\ kN$$

$$F_e = k\ x\ \delta_{max} \qquad (145)$$

Where; k = Spring stiffness (≈ 8 to 12 N/mm)

$\delta_{max} \approx h_{max}$ = Spring maximum compression (22.06 mm)

Similarly, The total load on the inlet valve (F_i) is given by,

$$F_i = F_s + F_{va} \qquad (146)$$

$$F_i = 5.5\ x\ 10^5 + 1.9\ x\ 10^6 = 2.45\ x\ 10^6\ kN$$

$$F_i = k\ x\ \delta_{max} \qquad (147)$$

Spring Wire Diameter (d), The diameter of a spring wire (d), as shown in Figure 3.55, can be calculated using the following relations,

$$\tau_{sp} = k\ x \frac{8\ xF_e\ x\ C}{\pi\ x\ d^2} \approx 420\ MPa \qquad (148)$$

Where; $k = \frac{4\,C-1}{4\,C-4} + \frac{0.615}{C} = \frac{F_e}{\delta}$ and

C = Spring index ($\frac{D}{d} \approx 4.24$ (5 to 10))

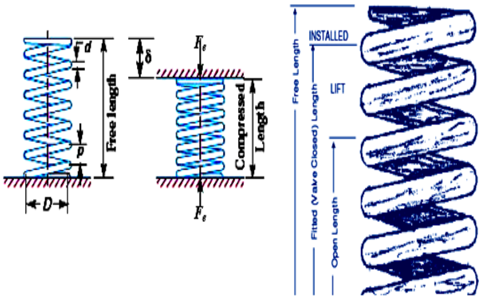


Figure 3. 55 Spring Dimension

Spring Coil Mean Diameter (D_m), $D_m = C \times d =$ 14 mm

Spring Coil Outer Diameter (D_0), $D_0 = D + d =$ 17.3 mm

Spring Coil Number of Turns (n), from axial deflection equation,

$$\delta = \frac{8 \times F_e \times C^3 \times n}{G \times d} \quad (149)$$

$$\Longrightarrow n = \frac{\delta}{F_e} \times \frac{G \times d}{8 \times C^3} = 9$$

Where; G (≈ 84 GPa)

For square and ground end, the total number of turns (n'), $n' = n + 2 =$ 11

Spring Coil Free Length (L_F), $L_F = n' \times d + \delta + 0.15 \times \delta =$ 45.26 mm

Pitch of the Coil (p), $p = \frac{L_F}{n' - 1} =$ 4.526 mm

Solid work is shown in figure 3.56 below.

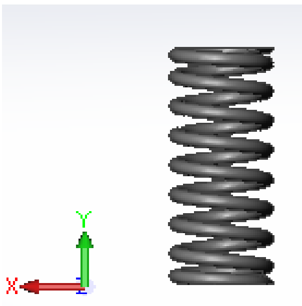


Figure 3. 56 Three Dimensional Sketch of Valve Spring

3.3.4 Development of the Prototype

Finally, after calculation and modeling making of the prototype is very essential because any machine which is designed and modeled must be made prototype before going out for real task. Therefore, this machine is made and recorded its mechanism and operation.

3.3.5 Cost analysis

The cost involved to design the implement had been evaluated in terms of raw materials and production (machine and labour) costs only. Materials, wastage and overhead costs are estimated from raw material and production cost. As shown in Table 3.11 the cost of implement and the detail procedure is given in Appendix C Table D.1, D.2.

Table 3. 11 Summarized Cost of the Implement

No.	1	2	3	4	5	6	Total cost
Variable	Raw materials	Material wastage	Production	overhead	profit	Sell tax	
Cost (ETB)	2050		1230				**3280**

CHAPTER FOUR

4 RESULTS AND DISCUSSION

The diesel engine for SUV is designed, modeled and simulated at ASTU. The design is analyzed with imprical relation and simulated with C++ programing by considering the shape and using SUV engine manual. Besides, in depth analysis of the crank and valve gear mechanism of SUV engine parts application, construction, design consideration, and material were considered to determine the size.

4.1 Collection of data related vehicle

In order to collect the data I had used SUV engine manual and design data for CI engines that presented in table 3.2 and 3.3 respectively. As indicated in table 3.2 main technical parameters was employed to prioritize the effective tire radius, angular wheel velocity, maximum wheel speed, final gear ratio and the engine size. As indicated in the table 3.3 design data for CI engine used for to know the range of the calculated values from both imprical relation and C++ programing results.

4.2 Selection of diesel engine mechanism to design

Among three mechanisms two mechanisms (crank and valve gear) were designed in this research. Based on its effective radius of a tire (0.5842 m), angular velocity of wheel (61.81 rad/s), maximum vehicle speed (130 km/h), maximum wheel speed (590.272 rpm), maximum engine speed at maximum brake power (4000 rpm), maximum engine speed (4800 rpm) the total resistances to vehicle motion, resistance power is calculated analatically. The expected resistance power at different velocity ($V = 0$ to $V = V_{max}$) on a level road could be plotted using a computer programming (C++) program # 1. Design of IC Engine (A1-CI) and application software's (MS-Excel) as shown in figure 3.8. from both analaytically calculated and resulted C++ program that have seen in table 3.4 and figure 3.8 curve the value of maximum resistance power on level road is equal (78.75 kw). Based on maximum resistance power and maximum brake power mechanical efficiency were calculated (75 %) in equation 13.

4.3 Evaluation of Performance Using Empirical Relation and Developing C++ Programing

In order to design all the selected engine parts, important design points such as brake power brake torque, brake mean effective pressure, mechanical efficiency, brake specific fuel consumption and etc. were analyzed empirically at 1000 rpm engine speed (equation 14 to 24). As indicated in table 3.3 those parameters were generated from C++ program #1 at different engine speed (1000 to 4600 rpm). From the table each parameter also plotted characteristic performance curve using application software (MS-Excel). Based on the result, the maximum brake power, brake torque, brake mean effective pressure, brake specific fuel consumption and mechanical efficiency were known. Engine size were compared from C++ program #1 with the given technical specification that are mentioned in table 3.2.

Table 4. 1 Comparison of Engine Size

Engine size	**From specification**	**From C++ program # 1**
Displacement	2499 cc	2498.799 cc
Bore diameter	92 mm	94.23 mm
Stroke length	94 mm	96.27 mm

From the above table shown, it is clear that the design has the problem on the engine size compatibility and it might be preferred designed on results from C++ program # 1.

4.4 Analysis of Design Parameters of engine design

Analysis of engine design as a whole is the part of a design process which helps the embodiment of part designs with initial instruction about the displacement, velocity, acceleration, forces, whieght, and torque that were acted on the individual components. So, those parameters were engine kinetics and kinematics.

4.5 Engine kinetics and kinematics

Since the difficult of solving equations (28 to 31) manually, inserted it in C++ program # 2 and piston displacement, piston velocity and piston acceleration were generated as shown in table 3.7 against crank angle (0 – 180). As indicated figure 3.9 engine kinematics of crank gear mechanism curves were plotted using table 3.7 results by MS-Excel.

The other most important design parameter to design engine crank gear mechanism were engine dynamics (kinetics) that various forces apply on it. As we saw from equation 32 to 42, piston head gas force, reciprocating parts inertia forces, reciprocating parts weight, piston force, side thrust, frictional force, force along the connecting rod axis, radial force acting along the crank web, tangential force acting on the crank pin and turning moment (torque) were calculated analaytically. Those equations inserted in C++ program # 4 (A4-CI.dat), the engine dynamics (kinetics) values have some slight difference from that of the analytical value.

Table 4. 2 Engine Dynamics (kinetics) on Crank gear mechanism parts as function of Crank Angle

Parameters	Kinetics resulted from analytical	Kinetics generated from C++ program # 4
Piston head gas Force (F_{g-max})	71.661 kN	75.16 kN
Reciprocating parts inertia force (F_i)	15.4623 kN	15.45 kN
Reciprocating parts weight (W_r)	14.715 kN	
Piston force (F_p)	56.1987or59.6977 kN	59.71 kN
Side thrust (F_s)	2.44 Kn	-0.14 kN
Frictional force (F_f)	0.244 KN	
Force along the connecting rod axis (F_{con})	56.25 KN	59.71 kN

Radial force acting along the crank web (F_r)	54.869 kN	58.82 kN
Tangential force acting on the crank pin (F_t)	12.1636 KN	10.23 kN
Turning moment (torque) (T)	571.6892 KN	492.27 kN

The results shown in the above table 4.2 came from an assumption that since SUV is modern light duty vehicle so that it used dual cycle process. After determined the cycle in order to solve gas pressure indicated mean effective pressure equation (43) of dual cycle was used. To know two unknown variables r_p = pressure ratio (1.4) and r_c = Cut-off (2.4) ratio as well as cylinder volume, cylinder pressure and temperature values were generated using C++ program # 3 and the result saw in table 3.8, from crank angle 40 to 540 degree. So, from table 3.8 the maximum cylinder gas pressure was occurred at crank angle between from 360 to 370 degree. i.e, P_{g-max} = 107.8 bar.

4.6 Analysis of Detail Design of engine parts

Detail design is the final design process which completes the embodiment of technical products about the function, construction, design consideration, material selection and size determination of all the selected diesel engine mechanism parts.

Measurement each parts for comparison with analytical calculation were applied by professional inspection employee. The measurement parameters includes length, thickness, width and diameter using Verner caliber, micrometer and depth gauge. From the specification data it was designed each parts of the implement including material selection of the part, force analysis, stress analysis of the implement. In order to designe each part the paper was started from the purpose, construction, design consideration, material selection and manufacturing process in a few detail. But size determination of each part were more focused in order to solve diameter, force, stress based on the selected material that is shown in table 4.3 below.

Table 4. 3 Critical Parts of Theoretical Stress Analysis with Material Stress

No	Part name	Material name	Material yield stress (MPa)	Theoretical result of stress (MPa)

1	Cylinder liner	Cast iron	100	47, 48.5
2	Piston	Aluminum alloy	90	88
3	Piston pin	Alloy steel	140	84
4	Connecting rod	Alloy steel	1518	38
5	Crankshaft	High grade steel	240	175
6	Camshaft	Alloy steel	1200	400
7	Rocker arm	Cast steel	70	50
8	Valve	High grade steel	120	103
9	Valve spring	Plain cast steel	400	420

Before the force and stress analyzed, comparison is made between measured and calculated value of size such as diameter, length and width or detail dimension of each part in the following table.

Table 4. 4 Comparison of Detail Dimension Between Measured and Calculated Value.

parts	Dimension	Measured value (mm)	Calculated value (mm)
Cylinder liner	Cylinder wall thickness (t)	9.5	12.336
	Cylinder bore and length (D, L)	103.53	92, 108.1
	Cylinder flange and studs (t_f, d)	10.3, 15.95	14.8 to 17.27, 12 to 16 (16)
	Cylinder head thickness (t_h)	9	13.7
	Cylinder total length (l_{cyl})		166.86
piston	Piston diameter (d_p)	91.91–91.928	91.955
	Piston head thickness (t_H)	5	10.4
Piston ring	Compression ring radial thickness (t_r)		3.14 (3.44-3.54)
	Compression ring axial thickness (t_a)		2.67 (2.7-2.73)

	Oil-control ring axial thickness (t_{ao})		5.34 (5.36-5.54)
	Compression ring end gap (e_1)		0.184 to 0.368
	Compression ring free end ($\mathbf{e_2}$)		10.99 to 12.56
	Top land width ($\mathbf{b_1}$)		10.4 to 12.48 (11.44)
	Ring grooves between distance (b_2)		2 to 2.67 (2.33625)
	Piston barrel thickness (t_3)		10.8
	Piston wall thickness at the bottom (t_4)		2.7 to 3.78 (3.24)
Piston skirt	Piston skirt length (l_s)	53	65
	Piston total length ($\mathbf{l_p}$)	85	92 to 138 (94.46)
Piston pin	Piston pin length ($\mathbf{l_{pp}}$)	70.90	78
	Inside piston pin length ($\mathbf{l_1}$)		41.37975
	Piston pin outer diameter ($\mathbf{d_0}$)	31.992-31.996	32
	Piston pin inner diameter (d_i)	16.80	19.2
Connecting rod	Flange and web thickness (t)	4.8	5
	Section width (B)	16.35	20
	Section height (H)	32.2	25
	Small end depth (H_2)	28.80	18.75 to 22.5
	Big end depth (H_1)	35.30	27.5 to 31.25
	Connecting rod length (l)	158.95	188
Crankshaft	Crank pin length ($\mathbf{l_{cp}}$)	31.50	46 to 64.4
	Crank web thickness ($\mathbf{t_{cw}}$)	29.50	22.08 to 24.84
	Two main journals distance ($\mathbf{l_1+l_2}$)	110.4	115 to 119.6
	Main journal length ($\mathbf{l_{mj}}$)	37.90	39.256

	Crank pin diameter ($\mathbf{d_{cp}}$)	53.93	59.8 to 69
	Crank web width ($\mathbf{w_{cw}}$)	80.60	96.6 to 119.6
	Main journal diameter ($\mathbf{d_{mj}}$)	62.99	64.4 to 82.8
Camshaft and cam	Camshaft outer diameter (d_{co})		
	Camshaft diameter (D_{cs})	25.3, 27.75	27.42
	Base circle diameter (D_{cam})	28	30.42
	Cam width (W_{cam})	12	20 to 25 (14.28)
Rocker arm	Fulcrum pin diameter ($\mathbf{d_1}$)	11.4	11.61
	Fulcrum pin length ($\mathbf{l_1}$)	34.2	14.5125
	Boss external diameter ($\mathbf{D_1}$)	16	23.22
	Boss internal diameter ($\mathbf{d_h}$)	12	17.61
Valve	Valve port diameter (d_{vp})		91.99
	Valve face angle ($\mathbf{a_v}$)	45.25-45.35	37.5^0
	Valve disc thickness **(t)**	7.3-7.73,7.5-7.55	8.887
	Valve maximum lift ($\mathbf{h_{max}}$)		28.9877, 22.99 (0)
	Margin thickness **(t)**	2.31-2.47, 2.41-2.57	1.5 to 2 (2)
	Valve face width **(b)**	3.5, 3.42	16.2888
	Valve head diameter ($\mathbf{d_v}$)	34.2-34.2, 32.4-32.4	124.5676
	Valve stem diameter ($\mathbf{d_s}$)	5.952 - 5.972, 5.942 - 5.96	21.92 to 26.57
Valve	Spring wire diameter **(d)**	3.30	3.30

spring	Spring coil mean diameter (D_m)		14
	Spring coil outer diameter (D_0)	14	17.3
	Spring coil number of turn **(n)**	9	9
	Spring coil free length (L_F)	45.26	45.26
	Coil pitch (p)	5	4.526

4.7 Checking forces and stresses on each part within the range

The final report on this paper was checked the validity of each part according its ability due the force that were discussed in chapter four applied on it and the induced stress within the limit. The stresses, longitudinal stress (σ_l) and circumferential stress (σ_c) on the cylinder liner and cylinder head; allowable tensile stress (σ_t) on the cylinder flange and studs or bolts; permissible bending (tensile) stress (σ_t) on the piston head, compression ring; allowable bearing pressure and allowable bending stress (σ_b) on the piston pin, connecting rod, crank pin, crank web, main journal, fulcrum pin, rocker arm and on the valve were used.

The cylinder wall thickness (t) = 12.336 mm as equation (50) and cylinder head thickness (t_h) = 13.7 mm as equation (54) were designed by circumferential or hoop stress (σ_c) = 48.5 MPa. Piston head thickness (t_H) = 10.4 mm on the basis of strength and 2.2 mm on the basis of heat transfer as equations (55) and (58) respectively were calculated by permissible bending (tensile) stress σ_t = 88 MPa. But the larger value was adopted. Radial thickness of the compression ring t_r = 3.14 mm as equation (59) by allowable bending (tensile) stress σ_t = 90 MPa. Piston skirt length l_s = 65 mm as equation (62) by side trust F_s = 7.166 x 10^6 kN and bearing pressure P_{bp} = 1198.3484 kN/mm^2. Outer diameter of piston pin d_0 = 32 mm as equation (63) by maximum gas force F_{g-max} = 107.8 bar and bearing pressure p_{bp-pp} = 55 MPa and by allowable bending stress σ_b = 84 MPa as equation (69). Connecting rod length l = 188 mm as equation (70) by buckling load W_B = 3.94 x 10^8 kN and compressive yield stress σ_c = 1517.945 MPa. Crank pin diameter d_{cp} = 59.8 mm and piston pin diameter d_p = 32 as equations (75) and (76) by allowable bearing pressure p_{bc} = 9.75N/mm^2 and p_{bp} = 26.05 N/mm^2 respectively. Bolt core diameter d_{cb} = 16 mm as equation (78) by allowable tensile stress σ_t = 38.4636 MPa. Thickness of the big end cap (t_c = as equation (81) by allowable bending stress σ_b = MPa. Based on combined stresses crank pin diameter d_{cp} = 59.8 mm as equation (86) by bending stress σ_{yb} = 101.765 MPa, width of the crank web $\mathbf{w_{cw}}$ = 96.6 mm as equation (88) by axial (compressive) stress σ_{xa} = 16.831 MPa, as equation (89) by bending (compressive) stress σ_{zb} = 157.79 MPa on the left side, diameter of the main journal $\mathbf{d_{mj}}$ = 64.4 mm as equation (90) by bending stress σ_{yb} = 38 MPa; and based on bearing pressure as equation (87) by crank pin bearing pressure P_{bp-cp} = 26.9699 MPa, as equation (91) by main journal bearing pressure p_{bp-mj} = 14.2 MPa were analyzed at the maximum bending moment position ($\theta \approx 15^0$ @ TDC). In a similar way Based on combined stresses crank pin diameter d_{cp} = 59.8 mm as equation (92) by bending stress σ_{yb} = 1.06 MPa and as equation (93) by shear stress τ_{xy} = 14.92 MPa, width of the crank web $\mathbf{w_{cw}}$ = 96.6 mm as equation (95) by axial (compressive) stress σ_{za} = 10.4338 MPa, as equation (96) and (97) by bending (compressive) stress σ_{zb} = 28.645894 MPa and 97.799 MPa respectively and as equation (98) by shear stress τ_{xy} = 19.99 MPa on the left side, diameter of the main journal $\mathbf{d_{mj}}$ = 64.4 mm as equation (99) by bending stress σ_{yb} = 19.9 MPa and as equation (100) by shear stress τ_{xy} = ; and based on bearing pressure as equation (94) by crank pin bearing pressure P_{bp-cp} = 16.18 MPa, as equation (101) by main journal bearing pressure p_{bp-mj} = 8.8 MPa were analyzed at the maximum twisting moment position ($\theta \approx 35^0$ @ TDC) of the crankshaft. Diameter of the fulcrum pin $\mathbf{d_1}$ = 11.61 mm as equation (112) by fulcrum pin bearing pressure p_{bp-p} = 6 MPa. Rocker arm effective length l = mm as equation (118) by permissible bending stress σ_b = MPa. Tappet core diameter d_c = mm as equation (120) by permissible compressive stress σ_c = 50 MPa. Thickness of the valve disc t = 8.887 mm as equation (128) by permissible bending stress σ_b = 120 MPa. Valve face width b = 16.2888 mm as equation (130) by allowable bending stress σ_{all} = 103 MPa. Valve stem diameter $\mathbf{d_s}$ = 21.92 to 26.57 mm as equation (133) by tensile stress σ_t = 0.2 MPa. The other dimension of each parts were calculated by their empirical relations.

In order to clarify witch part was safe witch part is not safe according the given limit of its stress and pressure the following table 4.5 shows the result.

Table 4. 5 Validity of the Part within the Range

Parts designed	Allowable stress limit (MPa)	Calculated value (MPa)	Bearing pressure limit (MPa)	Calculated value (MPa)	Remark
Cylinder wall thickness (t)	35 to 100	48.5	+	+	Safe
Stud diameter (d)	35 to 70	67.585	+	+	Safe

Cylinder head thickness (t_h)	30 to 50	48.5	+	+	Safe
Piston head thickness (t_H)	50 to 90	88	+	+	Safe
Compression ring radial thickness ($\mathbf{t_r}$)		90	+	0.035	
Piston skirt length (l_s)			0.5	1198300	
Piston pin outer diameter ($\mathbf{d_0}$)		84	10.5 to 15	> 55	Not safe
Connecting rod length (l)		1517.945	+	+	
Crank pin Diameter (d_{cp})			+	9.75	
Piston pin diameter (d_p)			+	26.05	
Bolt core diameter (d_{cb})		38.463	+	+	
Crank pin Diameter (d_{cp})	241.1	101.765, 1.06	8 to 16	26.9699, 16.18	Not safe
Crank web width ($\mathbf{w_{cw}}$)	241.1	174.621, 136.87869	+	19.99	
Mainjournal diameter($\mathbf{d_{mj}}$)	241.1	38,19.9	3 to 7	14.2, 8.8	Not safe
Cam width ($\mathbf{W_{cam}}$)	400 to 1200		+		
Fulcrum pin diameter ($\mathbf{d_1}$)			4 to 6		
Rocker arm thickness (t)	50 to 70		+	+	
Valve disc thickness (t)	100 to 120	120	+	+	
Valve face width (b)	103	103	+	+	
Initial spring force (F_s)			0.3 to 0.6 bar		
Spring wire diameter (d)		420	+	+	

Note: + sign indicates those values did not considered.

The result shown that material selection of new design was not safe operate because of calculation results of stress especially bearing stress greater than the limit of allowable stress. So the diameter or the material the exsisting design should be changed and it needs a serious test of material selection.

4.8 Making of the Prototype at Addis Ababa institute of technology (AAiT)

These newly diesel engine prototype for SUV were made in Addis Ababa institute of technology at workshop, which gives enough for demonstration. The actual model that drawn in solid work was two camshaft but the prototype was done in one camshaft and the shape, size, etc… did not exactly same as the actual as shown in figure 4.1. Based on Ethiopia automotive industry context, the implement was not simple to manufacture it easily because of lack of everything. The study was initiated to investigate the working principle and identify each parts while making each part.

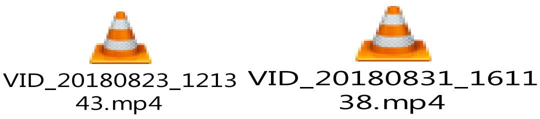


Figure 4. 1 Photo of the Assemble and Video for its Operation of the practical model

CHAPTER FIVE

5 CONCLUSIONS AND RECOMMENDATION

5.1 Conclusions

Internal Combustion engine is one of the most important inventions of the last century. It has been developed in the late 1800s and from there on it has had a significant impact on our society. It has been and will remain for foreseeable future a vital and active area of engineer research.

The aim of this project is to design a four-cylinder internal combustion engine taking into consideration all necessary calculations concerning its basic components. The project begins with short description of the history of engines and how they have developed through the years, because despite of the fact that everyday new and new engines are invented, the main components piston, block, crankshaft, valves and connecting rod have remained basically unchanged. And initially to make a simulation of the design engine of crankgear mechanism and valve gear mechanism. First step of the assignment is required power – determination of power available at the driving wheels, which shows the changes of values depending on the vehicle speed. Second step of the assignment is engine performance curve – determination of brake power, torque power, brake specific fuel consumption and mechanical efficiency, which shows the changes of values depending on the engine speed in rpm; and determination of displesment, gas pressure and temperature and creating graphs, which show the changes of values depending on the crank angle. Next step of the assignment is kinematics of the engine - determination of piston motion, acceleration and velocity and creating graphs, which show the changes of values depending on the angle between crank and connecting rod. And the last step of the assignment is kinetics – determination of forces on crank gear mechanism, which show the changes of values depending on the gas pressure on C++ programming and Excell softwares.

In addition the most proper materials which have to be used have been determined. It has been taken into chech that the chosen materials must resist on the maximum forces, moments and stresses that occur when the engine is operating. After that gas and inertia forces acting on the connecting rod are determined. Next to that, the project continues with an explanation of the functions of these parts and the used materials for their production. The next point of consideration is size determination of engine parts and what their major stresses are. Next step made is the calculation of tensile stress and temperature of the cylinder. Another thing that is taken into consideration is the forces acting on strength stud bolts. This is important because strength stud bolts provide density between the cylinder and cylinder head at all mode of operation. The project continues with calculation of piston group. These calculations are made very precisely, because the piston is the main reciprocating part of an engine and its movement creates an imbalance. To transmit the energy of the piston to the crank, the piston is connected to a connecting rod. Calculation of the connecting rod is of vital importance as it should be done very precisely because when the engine is running the rod is under varying in size and direction gas and inertia forces. For this reason it has been made by alloy steel with high resistance to fatigue. Next step that is taken is calculation of crankshaft mechanism, considering that it is subject to the action of gas forces, inertia forces and moments which are periodical functional angle of knee. In designing the crankshaft, parameters of already existing engines are being used.

After making all mentioned calculation the necessary part dimensions are achieved and the project continues with drawings on AUTOCAD. First of all, 3D drawings of all calculate parts have been made. In addition 2D drawings based on 3D are made. After that ready 3D parts are connected in a product. Having a whole product is just the beginning of the simulation.

Having calculated all forces, moments and stresses in an allowable range and the main design question have been answered and the objectives of the project achieved.

5.2 Recommendations

Based on the findings of this study, the following future works are made

- It is recommended to have an additional design by fabricating the prototype for technology dissemination in similar areas of the study.
- The designed was mechanical only in force, material and size of mechanism parts thus further research should be done on thermal design and lubrication process.
- It need to farther research more than 15 % mechanical load and thermal effects of in this types of diesel engine.
- The real prototype must be prepare and test will be implement to find the desired performance
- It is recommended to have design by new material and validate by ANSYS software

5.3 Future work

The future work of the work is

Design and simulation of crank and valve gear mechanism by using ANSYS from light weight materials.

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APPENDIX

Appendix – A. C++ Programs for Solving Mathemtical Models

Program # 1

```
// Design of CI Engine (A1-CI).
#include<iostream.h>
#include<iomanip.h>
#include<conio.h>
#include<math.h>
#include<fstream.h>
 void main()
  {
 Start:
   clrscr();
    char ch1;
   float L,W,H,wb,rw,Wv,v_max,PHI_max,PHI_min,WD_FWper,bmep_Pbmax,bsfc_Pbmax;
   float Zs,Zc,N_min,FN_TbPb,GR_P,GR_B,GR_T,GR_F,ETA_Gtper,Cd,SF,OptG,RHOa;
   float g,Factv,RN_TbPb,ETA_Gt,WD_FW,FzF,FzR,Apf,radsin_PHImin,Pr_vmax;
```

```
   float radsin_PHImax,v_min,v_Pbmax,v_maxL,delv,v_mps,v_mpsPbmax,Ra,Rg,RrlF;
   float Rtr,Pr,Pw,RrlR,Pw_max,Pb_max,v,v_Pbmaxmps,v_maxmps,v_maxmpsL;
   float OMEGAw_Pbmax,OMEGAw_max,Nw_Pbmax,N_Pbmax,OMEGAw_maxL,Nw_maxL;
   float Nw_max,N_max,N_Tbmax,OMEGA_Tbmax,xii,Pb_Tbmax,Tb_max,Zrpc,N,bmep_kPa;
   float ncps_Pbmax,ncps_Tbmax,Ec,Ec_cc,delN,x,Pb,OMEGA,Tb,ncps,bmep,bsfc;
   float Ec_liter,Vs,Vcl,Vt,Vs_cc,Vcl_cc,Vt_cc,powd,d,d_mm,s,s_mm,fmepb;
   float r,r_mm,lcon,lcon_mm,Ap,vpm,fmepa,fmep,ETA_m,ETA_mper,Rc,Rsd,Ratiorl;
// Engine Performance:
   float ETA_v,ETA_vper,ETA_vMmax,ETA_vMmin,ZMmin,ZMmax,ZM,Rrl,AF,RHOim;
   float cpim,cvim,Rim,nim,nAim,RpimpA,npTAim,Tima,Tim,Zim,vim;
   float n_cps,imep,pim,pA,TA,Z_rpc,Tb2,bmep2,bsfc2,mA_kgph,mA_kgps;
   float mF_kgph,mF_kgps,xR,xA,xF,mm_kgps;
     mA_kgps = xA*(1.0-xR)*mm_kgps;
   const float PI = 3.14;
 ofstream outfile ("A1-CI.dat");
 clrscr();
  cout<<"\n\n\t\t Adama Science and Technology University (ASTU)";
  cout<<"\n\t   Department of Mechanical and Vehicle Engineering";
  cout<<"\n\t\t         Amsalu Fentahun(MSc.Student)";
  cout<<"\n\t\t\t       ****************** ";
  cout<<"\n\n\t\t Design of Compression-Ignition(CI)Engine(A1-CI)";
  cout<<"\n -----------------------------------------------------";
  cout<<"\n This 'Computer Program (in C++)'is used to determine:";
  cout<<"\n Vehicle Performance,Engine Performance and Engine capacity(Ec) of:";
  cout<<"\n a Two-Wheel,a Three-Wheel and a Four-Wheel Motor Vehicles.\n";
  cout <<"\n -----------------------------------------------------";
  outfile<<"\n\n\t\t Adama Science and Technology University(ASTU) ";
  outfile <<"\n\t Department of Mechanical and Vehicle Engineering";
  outfile <<"\n\t\t         Amsalu Fentahun(MSc.Student)";
  outfile <<"\n\t\t\t       ****************** ";
  outfile <<"\n\n\t\t Design of Compression-Ignition (CI) Engine (A1-CI)";
  outfile <<"\n -------------------------------------------------";
  outfile<<"\n This'Computer Program(in C++)'is used to determine:";
  outfile<<"\n Vehicle Performance, Engine Performance and Engine capacity (Ec)
of:";
  outfile<<"\n a Two-Wheel,a Three-Wheel and a Four-Wheel Motor Vehicles.\n";
  outfile <<"\n -------------------------------------------------";
  cout <<"\n  Enter the 'vehicle Length (L)' in 'm':";
  cin >> L;
  cout <<"\n  Enter the 'vehicle Width (W)' in 'm':";
  cin >> W;
```

```
cout <<"\n  Enter the 'vehicle Height (H)' in 'm':";
cin >> H;
cout <<"\n  Enter the 'length of wheelbase (wb)' in 'm':";
cin >> wb;
cout <<"\n  Enter the 'effective tyre radius (rw)' in 'm':";
cin >> rw;
cout <<"\n  Enter the 'vehicle maximum Weight (Wv)' in 'N':";
cin >> Wv;
cout <<"\n Enter the 'Weight Distribution on the Front Wheels'(30% to 70%):";
cin >> WD_FWper;
cout <<"\n  Enter the 'maximum vehicle velocity (v_max)' in 'km/h':";
cin >> v_max;
cout<<"\n  Enter the 'maximum gradient(PHI_max)'(5 to 15 deg.):";
cin >> PHI_max;
cout <<"\n  Enter the 'minimum gradient(PHI_min)'(0 to 2 deg.):";
cin >> PHI_min;
cout <<"\n Enter the 'brake mean effective pressure(bmep_Pbmax)'(2 to 12 bar):";
cin >> bmep_Pbmax;
cout<<"\n Enter the 'fuel consumption(bsfc_Pbmax)'(0.2 to 0.5 kg/kWh):";
cin >> bsfc_Pbmax;
cout <<"\n  Enter the 'Air-Fuel Ratio (AF)':";
cin >> AF;
cout <<"\n  Enter the 'number of strokes (Zs)':";
cin >> Zs;
cout <<"\n  Enter the 'number of engine cylinders (Zc)':";
cin >> Zc;
cout <<"\n  Enter the 'Ratio of stroke-to-bore (Rsd)':";
cin >> Rsd;
cout <<"\n  Enter the 'Ratio of crank-to-conrod length (Rrl)':";
cin >> Ratiorl;
cout <<"\n  Enter the 'compression Ratio (Rc)':";
cin >> Rc;
cout <<"\n Enter the 'minimum or idling engine speed (N_min)'(500 to 1000 rpm):";
cin >> N_min;
cout <<"\n  Enter the 'engine speed at Pb_max (N_Pbmax) =";
cin >> N_Pbmax;
cout <<"\n  Enter the 'engine speed at Tb_max (N_Tbmax) =";
cin >> N_Tbmax;
cout <<"\n Enter the 'ratio of Bottom-Gear (GR_B)'(2.5 to 4.5):";
cin >> GR_B;
cout <<"\n  Enter the 'ratio of Top-Gear (GR_T)' (0.6 to 1.2):";
```

```
  cin >> GR_T;
  cout <<"\n  Enter the 'ratio of Final-Gear (GR_F)' (2.5 to 9):";
  cin >> GR_F;
  cout<<"\n Enter the 'total transmission efficiency (ETA_Gt)'(70 to 90 %):";
  cin >> ETA_Gtper;
  cout<<"\n  Enter the 'drag Coefficient (Cd) of the vehicle'(0.3 to 0.8):";
  cin >> Cd;
  cout <<"\n Enter the'Shape Factor(SF)for the vehicle's frontal area'(0.7 to
0.9):";
  cin >> SF;
  cout <<"\n Enter your choice 'UG or OptG or OG' (0.8 to 1.2):";
  cin >> OptG;
   g = 9.81; Factv = 1.15; ETA_Gt = ETA_Gtper/100.0; WD_FW = WD_FWper/100.0;
   FzF = WD_FW*Wv; FzR = (1.0-WD_FW)*Wv; Apf = W*H*SF; RHOa = 1.18;
   RN_TbPb = N_Tbmax/N_Pbmax;
   radsin_PHImin = PHI_min*(PI/180.0); radsin_PHImax = PHI_max*(PI/180.0);
  clrscr();
  cout <<"\n\n\t\t Adama Science and Technology University (ASTU)";
  cout <<"\n\t   Department of Mechanical and Vehicle Engineering";
  cout <<"\n\t\t         Amsalu Fentahun(MSc.Student)";
  cout <<"\n\t\t\t       ****************** ";
  cout <<"\n\n\t\t Design of Compression-Ignition(CI)Engine(A1-CI)";
  cout <<"\n -----------------------------------------------------";
  cout <<"\n\n  'Vehicle Resistance'";
  cout <<"\n ________________________ ";
  outfile <<"\n\n  'Vehicle Resistance'";
  outfile <<"\n ________________________";
  cout <<"\n\n Select the 'type of Drive' for the Vehicle,\n";
  cout <<"\n   1. Front Wheel Drive (FWD)";
  cout <<"\n   2. Rear Wheel Drive (RWD)";
  cout <<"\n   3. Four Wheel Drive (4WD)";
  cout <<"\n\n Enter your choice (1 or 2 or 3):";
  cin >> ch1;
   v_min = 0.0; v_Pbmax = v_max*OptG; v_maxL = Factv*v_max;
   delv = (v_maxL-v_min)/20.0;
     if ( ch1 == '1' )
      goto Perform1;
      else if ( ch1 == '2' )
      goto Perform2;
      else if ( ch1 == '3' )
      goto Perform3;
```

```
cout <<"\n\n  Press 'Enter' twice, to 'proceed'....";
getch();
getch();
clrscr();
cout<<"\n\n\t\t Adama Science and Technology University (ASTU)";
cout<<"\n\t     Department of Mechanical and Vehicle Engineering";
cout<<"\n\t\t          Amsalu Fentahun(MSc.Student)";
cout<<"\n\t\t\t       ******************";
cout<<"\n\n\t\t Design of Compression-Ignition(CI)Engine(A1-CI)";
cout<<"\n -----------------------------------------------------";
Perform1:
cout <<"\n\n 'Pr' versus 'v'";
cout <<"\n __________________";
cout <<"\n\n\t  v\t  Pr ";
cout <<"\n\t(km/h)\t (kW) \n";
outfile <<"\n\n\n\t 'Pr' versus 'v'";
outfile <<"\n\t ____________________";
outfile <<"\n\n\t\t  v\t  Pr ";
outfile <<"\n\t\t(km/h)\t (kW) \n";
for(v = v_min; v <= v_maxL; v = v+delv)
 {
   v_mps = v/3.6; v_mpsPbmax = v_Pbmax/3.6;
   Ra = 0.5*RHOa*Cd*Apf*(v_mps*v_mps); Rg = Wv*radsin_PHImin;
   Rrl = (0.015+(0.00016*v))*Wv; Rtr = Rrl+Ra+Rg;
   Pr = (Rtr*v_mps)*1.0e-3;
cout << setprecision(2)<<"\n\t "<<v;
cout << setprecision(2)<<"\t "<<Pr;
outfile << setprecision(2)<<"\n\t\t "<<v;
outfile << setprecision(2)<<"\t "<<Pr;
getch();
 }
goto Perform4;
Perform2:
cout << "\n\n\n 'Pr' versus 'v'";
cout << "\n ____________________";
cout << "\n\n\t  v\t  Pr ";
cout << "\n\t(km/h)\t (kW) \n";
outfile << "\n\n\n\t 'Pr' versus 'v'";
outfile << "\n\t ____________________";
outfile << "\n\n\t\t  v\t  Pr ";
outfile << "\n\t\t(km/h)\t (kW) \n";
```

```
 for(v=v_min; v<=v_maxL; v=v+delv)
  {
    v_mps = v/3.6; v_mpsPbmax = v_Pbmax/3.6;
    Ra = 0.5*RHOa*Cd*Apf*(v_mps*v_mps); Rg = Wv*radsin_PHImin;
    Rrl = (0.015+(0.00016*v))*Wv; Rtr = Rrl+Ra+Rg;
    Pr = (Rtr*v_mps)*1.0e-3;
 cout << setprecision(2)<<"\n\t "<<v;
 cout << setprecision(2)<<"\t "<<Pr;
 outfile << setprecision(2)<<"\n\t\t "<<v;
 outfile << setprecision(2)<<"\t "<<Pr;
 getch();
  }
 goto Perform4;
 Perform3:
 cout << "\n\n\n 'Pr' versus 'v'";
 cout << "\n ____________________";
 cout << "\n\n\t  v\t  Pr ";
 cout << "\n\t(km/h)\t (kW) \n";
 outfile << "\n\n\n\t 'Pr' versus 'v'";
 outfile << "\n\t ____________________";
 outfile << "\n\n\t\t  v\t  Pr ";
 outfile << "\n\t\t(km/h)\t (kW) \n";
 for(v=v_min; v<=v_maxL; v=v+delv)
  {
    v_mps = v/3.6; v_mpsPbmax = v_Pbmax/3.6;
    Ra = 0.5*RHOa*Cd*Apf*(v_mps*v_mps); Rg = Wv*radsin_PHImin;
    Rrl = (0.015+(0.00016*v))*Wv;
    Rtr = Rrl+Ra+Rg;
    Pr = (Rtr*v_mps)*1.0e-3;
 cout << setprecision(2)<<"\n\t "<<v;
 cout << setprecision(2)<<"\t "<<Pr;
 outfile << setprecision(2)<<"\n\t\t "<<v;
 outfile << setprecision(2)<<"\t "<<Pr;
 getch();
  }
 goto Perform4;
Perform4:
  cout<<"\n\n Enter the resistance Power at: v="<<v_Pbmax<<"km/h:";
  cin >> Pr_vmax;
    Pw_max = Pr_vmax; Pb_max = Pw_max/ETA_Gt;
    v_Pbmaxmps = v_Pbmax/3.6; v_maxmps = v_max/3.6; v_maxmpsL = v_maxL/3.6;
```

```
    OMEGAw_Pbmax = v_Pbmaxmps/rw; OMEGAw_max = v_maxmps/rw;
    Nw_Pbmax = (60.0*OMEGAw_Pbmax)/(2.0*PI); xii = RN_TbPb;
    N_Pbmax = Nw_Pbmax*(GR_T*GR_F); OMEGAw_maxL = v_maxmpsL/rw;
    Nw_maxL = (60.0*OMEGAw_maxL)/(2.0*PI);
    Nw_max = (60.0*OMEGAw_max)/(2.0*PI); N_max = Nw_maxL*(GR_T*GR_F);
    N_Tbmax = N_Pbmax*RN_TbPb; OMEGA_Tbmax = (2.0*PI*N_Tbmax)/(60.0);
    Pb_Tbmax = Pb_max*(-(xii*xii*xii)+(1.13*xii*xii)+(0.87*xii));
    Tb_max = (Pb_Tbmax*1.0e3)/(OMEGA_Tbmax); Zrpc = Zs/2.0;
    ncps_Pbmax = N_Pbmax/(Zrpc*60.0); ncps_Tbmax = N_Tbmax/(Zrpc*60.0);
    Ec = (Pb_max*1.0e3)/(bmep_Pbmax*1.0e5*ncps_Pbmax); Ec_cc = Ec*1.0e6;
    Vs = Ec/Zc; Vcl = Vs/(Rc-1.0); Vt = Vcl+Vs;
    Vs_cc = Vs*1.0e6; Vcl_cc = Vcl*1.0e6; Vt_cc = Vt*1.0e6;
    powd = (4.0*Vs)/(PI*Rsd); d = pow(powd,0.33); d_mm = d*1.0e3; s = d*Rsd;
    s_mm = s*1.0e3; r = s/2.0; r_mm = r*1.0e3; lcon = r/Ratiorl;
    lcon_mm = lcon*1.0e3; Ap = (PI*d*d)/4.0;
cout << "\n\n  Press 'Enter' twice, to 'proceed'.... ";
getch();
getch();
clrscr();
cout<<"\n\n\t\t  Adama Science and Technology University (ASTU)";
cout<<"\n\t   Department of Mechanical and Vehicle Engineering";
cout<<"\n\t\t         Amsalu Fentahun(MSc.Student)";
cout<<"\n\t\t\t       ****************** ";
cout<<"\n\n\t\t Design of Compression-Ignition(CI)Engine(A1-CI)";
cout<<"\n ---------------------------------------------------------";
cout<<"\n  'Engine Performance'";
cout<<"\n ______________________";
cout<<"\n\n\t  Pb\t  Tb\t  bmep\t  bsfc\tETA_m\t  N ";
cout <<"\n\t (kW)\t (Nm)\t (bar)\t(kg/kWh) (%)\t(rpm) \n";
outfile <<"\n  'Engine Performance'";
outfile <<"\n ______________________";
outfile <<"\n\n\t  Pb\t  Tb\t  bmep\t  bsfc\tETA_m\t   N ";
outfile <<"\n\t (kW)\t (Nm)\t (bar)\t(kg/kWh) (%)\t  (rpm) \n";
     delN = (N_max-N_min)/10.0;
for(N=N_min; N<=N_max; N=N+delN)
 {
    v_mps = v/3.6; x = N/N_Pbmax; OMEGA = (2.0*PI*N)/60.0;
    Pb = Pb_max*(-(x*x*x)+(1.13*x*x)+(0.87*x)); // open cc
    Tb = (Pb*1.0e3)/OMEGA; Zrpc = Zs/2.0; ncps = N/(Zrpc*60.0);
    bmep = ((Pb*1.0e3)/(Ec*ncps))*1.0e-5;
    bsfc = bsfc_Pbmax*((x*x)-(1.55*x)+1.55);
```

```
      fmepa = 0.001*N; fmepb = (0.001*N)*(0.001*N);
      fmep = 0.97+(0.15*fmepa)+(0.05*fmepb); imep = bmep+fmep;
      ETA_m = bmep/imep; ETA_mper = ETA_m*100.0; RHOim = 1.25;
      mF_kgph = bsfc*Pb; mF_kgps = mF_kgph/3600.0; xR = 0.05;
      xA = AF/(AF+1.0); xF = 1.0/(AF+1.0); mm_kgps = mF_kgps/(xF*(1.0-xR));
      mA_kgps = xA*(1.0-xR)*mm_kgps;
      ETA_v = mA_kgps/(RHOim*Ec*ncps); ETA_vper = ETA_v*100.0;
       cout << setprecision(2)<<"\n\t "<<Pb;
       cout << setprecision(2)<<"\t "<<Tb;
       cout << setprecision(2)<<"\t "<<bmep;
       cout << setprecision(2)<<"\t "<<bsfc;
       cout << setprecision(2)<<"\t"<<ETA_mper;
       cout << setprecision(2)<<"\t"<<N;
       outfile << setprecision(2)<<"\n\t "<<Pb;
       outfile << setprecision(2)<<"\t "<<Tb;
       outfile << setprecision(2)<<"\t "<<bmep;
       outfile << setprecision(2)<<"\t  "<<bsfc;
       outfile << setprecision(2)<<"\t"<<ETA_mper;
       outfile << setprecision(2)<<"\t"<<N;
       getch();
        }
     cout <<"\n\n\t  * Pb_max = "<<setprecision(2)<<Pb_max<<" kW  at  N =
"<<N_Pbmax<<" rpm ";
     cout <<"\n\t  * Tb_max = "<<setprecision(2)<<Tb_max<<" Nm  at  N =
"<<N_Tbmax<<" rpm ";
     cout <<"\n\t  * bsfc_Pbmax = "<<setprecision(2)<<bsfc_Pbmax<<" kg/kWh  at  N =
"<<N_Pbmax<<" rpm ";
   cout << "\n\n  Press 'Enter' twice, to 'proceed'.... ";
   getch();
   getch();
   clrscr();
   cout<<"\n\n\t\t Adama Science and Technology University (ASTU)";
   cout<<"\n\t   Department of Mechanical and Vehicle Engineering";
   cout<<"\n\t\t         Amsalu Fentahun(MSc.Student)";
   cout<<"\n\t\t\t       ****************** ";
   cout<<"\n\n\t\tDesign of Compression-Ignition (CI) Engine (A1-CI) " ;
   cout<<"\n ---------------------------------------------------- ";
     cout<<"\n\n  Engine Size";
     outfile<<"\n\n  Engine Size";
     cout<<"\n -------------- ";
     outfile<<"\n ------------- ";
```

```
    cout <<setprecision(3)<<"\n  Ec = "<<Ec_cc<<" cc";
    cout <<setprecision(2)<<"\n  Vs = "<<Vs_cc<<" cc";
    cout <<setprecision(2)<<"\n  Vcl = "<<Vcl_cc<<" cc";
    cout <<setprecision(2)<<"\n  Vt = "<<Vt_cc<<" cc";
    cout <<setprecision(2)<<"\n\n  d = "<<d_mm<<" mm";
    cout <<setprecision(2)<<"\n  s = "<<s_mm<<" mm";
    cout <<setprecision(2)<<"\n  r = "<<r_mm<<" mm";
    cout <<setprecision(2)<<"\n  lcon = "<<lcon_mm<<" mm";
    outfile <<setprecision(3)<<"\n  Ec = "<<Ec_cc<<" cc";
    outfile <<setprecision(2)<<"\n  Vs = "<<Vs_cc<<" cc";
    outfile <<setprecision(2)<<"\n  Vcl = "<<Vcl_cc<<" cc";
    outfile <<setprecision(2)<<"\n  Vt = "<<Vt_cc<<" cc";
    outfile <<setprecision(2)<<"\n\n  d = "<<d_mm<<" mm";
    outfile <<setprecision(2)<<"\n  s = "<<s_mm<<" mm";
    outfile <<setprecision(2)<<"\n  r = "<<r_mm<<" mm";
    outfile <<setprecision(2)<<"\n  lcon = "<<lcon_mm<<" mm";
  End:
  cout << "\n\n  Press 'Enter' twice, to 'Exit' !";
  getch();
  getch();
  getche();
   }
```

Program # 2

```
#include<iostream.h>
#include<conio.h>
#include<math.h>
#include<iomanip.h>
#include<fstream.h>
 void main()
  {
    float r,rlratio,omega,rad,rpm,rcr,rcrl,theta;
    const float  pi = 3.14;
    float xp,vp,ap;
  ofstream outfile ("AMSALU-CI.dat");
  clrscr();
  cout << "\n\n\t\t  Adama Science and Technology University(ASTU)";
  cout << "\n\t        Mechanical Systems and Vehicle Engineering";
  cout << "\n\t\t         Amsalu Fentahun (MSc.Student)";
  cout << "\n-----------------------------------------------------";
  cout << "\n\t\t Design of Compression-Ignition (CI) Engine";
  cout << "\n AMSALU-CI: A computer programm in C++ to plot:'xp','vp' and 'ap' vrs.'theta'";
```

```
  cout << "\n................................................";
   cout <<"\n Enter the maximum engine speed(in rpm) = ";
   cin >> rpm;
   cout <<"\n Enter crank radius (in mm) =";
   cin >> rcr;
   cout<<"\n Enter the ratio of crank to con-rod length(0.2to0.3)=";
   cin >> rcrl;
   r = rcr/1000.0;
   omega = rpm*(pi/30.0);
     cout << "\n\n\ttheta" <<"\t  xp"<<"\t  vp"<<"\t   ap";
     outfile <<"\n\n\ttheta" <<"\t  xp"<<"\t  vp"<<"\t   ap";
     cout <<"\n\t(deg.)"<<" (m) "<<" \t (m/s) " << "\t (m/s*s)\n";
     outfile <<"\n\t(deg.)"<<"(m)"<<" \t (m/s)"<< "\t (m/s*s) \n";
      for (theta = 0.0; theta <= 180.0; theta = theta + 10.0 )
       {
        rad = (theta*pi)/180.0;
        xp = (r*((1.0-cos(rad))+(rcrl*(sin(rad)*sin(rad))/2.0)))*200.0;
        vp = omega*r*(sin(rad)+(rcrl*sin(2.0*rad))/2.0);
        ap = (omega*omega*r*(cos(rad)+(rcrl*cos(2.0*rad))))/200.0;
         cout <<"\n\t"<<theta <<"\t"<<setprecision(2)<< xp<<"\t"<<setprecision(3)<<vp<<"\t
"<<setprecision(3)<<ap;
         outfile <<"\n\t"<<theta <<"\t"<<setprecision(2)<<
xp<<"\t"<<setprecision(3)<<vp<<"\t "<<setprecision(3)<<ap;
         getch();
          }
   cout <<"\n\n\t\t Press any key twice to return !";
   getch();
   getche();
     }
```

Program # 3

```
// Actual Engine Cycle - CI Engine (A3-CI).
#include<iostream.h>
#include<iomanip.h>
#include<conio.h>
#include<math.h>
#include<fstream.h>
 void main()
   {
     float Ec_cc,Zc,Rsd,Rrl,a,b,RHO_kgplit,QLHV,ZC,ZH,ZO,RHO_Alkgplit,QLHV_Al;
     float xAl_per,xR_per,LAMBDA,pim,p1,T1,Rc,N,ETA_mper,theta_ivobTDC;
     float theta_ivcaBDC,theta_scbTDC,theta_dc,theta_evobBDC,theta_evcaTDC;
     float Zs,Z_rpc,ETA_m,xR,theta_a,theta_b,theta_ivc,theta_1,theta_sc;
     float theta_2,theta_ec,theta_3 = theta_ec,theta_evo,theta_4,theta_5;
     float theta_6,theta_0,Ec_liter,Ec,Vs,Vcl,Vt,Vs_cc,Vcl_cc,Vt_cc,d,d_mm;
     float powd,s,s_mm,r,r_mm,lcon,lcon_mm,Ap,g,Ru,mA,mF,mres,Tim,theta_c;
```

```
    float a1O2r,a2O2r,a3O2r,a4O2r,cpO2,cvO2,a1N2r,a2N2r,a3N2r,a4N2r,cpN2;
    float cvN2,cpA6b,cvA6b,MmA,cp6b,cv6b,R6b,n6b,RHO6b,v6b,Z6b,Z1,V1,R1;
    float mm,MmO,MmN,MmC,MmH,yO2,yN2,MmO2,MmN2,xAl,xG,xF,MmG,MmAl,QLHV_G;
    float nAl,nG,nAlG,yAl,yG,yF,yAl_per,MmF,n1th,n2th,atha,ath,n3th,Pb;
    float n10,n3,n1,n2,nt,y1,y2,y3,y10,MmAth,MmFth,nO2,nN2,nA,AFth,AF,QLHV_F;
    float RHO_Fkgplit,RHO_F,ETA_cmax,ETA_c,ETA_cper,T6bi,RHO_Gkgplit;
    float Rm,Zm,Vm,n4,n5,y4,y5,yAL,Mm1,Mm2,Mm3,Mm4,Mm5,nF,mF3;
    const float PI = 3.1416; g = 9.81; Ru = 8.314;
// process 0-b (constant pressure intake process).
    float V0bi,p0bi,T0bi,m0bi,R0bi,rad,Z0b,xp0b,Vs0b,V0b,p0b,T0b,R0b,m0b;
    float Vb,pb,Tb,mb,thetab,deltheta,theta,mb1i;
// process b-1 (polytropic compression):
    float Vb1i,pb1i,Tb1i,Rb1i,xpb1,Vsb1,Vb1,delmb1,VbVb1,nb1,nb1T,Rb1;
    float Zb1,pb1,Tb1,mb1,theta1;
// process 1-2 (polytropic effective compression):
    float m12i,V12i,p12i,T12i,xp12,Vs12,V12,V1V12,n12,n12T,R12,Z12;
    float p12,T12,m12,V2,p2,T2,theta2,W12i;
// process 2-3 (polytropic combustion):
    float V23i,p23i,T23i,T23,W23i,xp23,Vs23,V23,m23,theta2dc,rad2dc;
    float xb,Qin,Q1,Q23,cpO2o2,R23,Wx3,Qx3;
    float cpN2o2,cpFo2,cpr2,cvr2,Mmr2,cp2,cv2,R2,m2,RHO2;
    float v2,hfoF,hfoO2,hfoN2,delhFo2,delhO2o2,delhN2o2,UrFo2,UrO2o2,UrN2o2;
    float Ur2,Ur,a1CO2p,a2CO2p,a3CO2p,a4CO2p,cpCO2,cvCO2,cp123,a1H2Op,a2H2Op;
    float a3H2Op,a4H2Op,cpH2O,cvH2O,cp223,a1N2p,a2N2p,a3N2p,a4N2p;
    float cp323,a1O2p,a2O2p,a3O2p,a4O2p,cp423,a1COp,a2COp,a3COp;
    float a4COp,cpCO,cvCO,cp523,cp23_kmol,cv23_kmol,Mm23,n23,cp23,cv23;
    float delh1o23,delh2o23,delh3o23,delh5o23,hfo1,hfo2,hfo3,hfo4,hfo5,U123;
    float U223,U323,U423,U523,U23,Up,Qin23,mF23,nF23,delUpr,error23,T3;
    float p23,delV23,delW23,W23,V3,p3,m3,theta3,To,Qin1,QW23,p33,RSE23,LSE23;
    float error23per,T33,cv123,cv223,cv323,cv423,cv523,rad33;
    float Rpx3i,Rp2x,n2x,n2xT,theta33,xp33,Vs33,V33,Rcf,Pi,Wi2x,imep2x;
    float RSE2x,LSE2xa,LSE2x,Rp2xi,error2x,error2xper,Rp,px,Tx,Vx,Vx3i,px3i;
    float Tx3i,theta_x,xpx3,Vsx3,Vx3,mx3,px3,Tx3,W2x;
// process 3-4 (polytropic expansion):
    float V34i,p34i,T34i,m34i,xp34,Vs34,V34,V3V34,n34,n34T,R34,Z34,p34,T34;
    float delm34,m34,V4,p4,T4,theta4,W34i;
// process 4-5 (polytropic exhaust blowdown):
    float V45i,p45i,T45i,m45i,xp45,Vs45,V45,V4V45,n45,n45T,R45,Z45,p45,T45;
    float delm45,m45,V5,p5,T5,theta5,W45i;
// engine parameters
    float Wnet,ETA_ith,ETA_ithper,imep_bar,W12,W2,W3,Q3,W4,W34,W45,W5,Pi1;
    float Wi,n_cps,Pb_kW,mF_kgps,mF_kgph,bsfc,fmepa,fmepb,fmep,bmep,imep;
 clrscr ();
 ofstream outfile ("A3-CI.dat");
 cout << "\n\n\t\t  Adama Science and Technology University (ASTU)";
```

```
outfile <<"\n\n\t\t Adama Science and Technology University(ASTU)";
cout << " \n\t   Department of Mechanical and Vehicle Engineering";
outfile <<" \n\t Department of Mechanical and Vehicle Engineering";
cout << " \n\t\t         Amsalu Fentahun(MSc.Student)";
outfile << " \n\t\t      Amsalu Fentahun(MSc.Student)";
cout <<"\n...................................................\n";
outfile << "\n..............................................\n";
cout <<"\n Enter the 'Engine capacity (Ec_cc)' in 'cc' = ";
cin >> Ec_cc;
cout <<"\n Enter the 'Number of cyliders (Zc)' =";
cin >> Zc;
cout <<"\n Enter the 'Ratio of stroke-to-bore (Rsd)' =";
cin >> Rsd;
cout <<"\n Enter the 'Ratio of crank-to-con rod legnth (Rrl)' =";
cin >> Rrl;
cout <<"\n Enter the 'number of Carbon atoms in Diesel (a)' =";
cin >> a;
cout <<"\n Enter the 'number of Hydrogen atoms in Diesel (b)' =";
cin >> b;
cout <<"\n Enter the 'density of Diesel (RHO_G)' in 'kg/liter' =";
cin >> RHO_Gkgplit;
cout <<"\n Enter the 'Lower Heating Value of Diesel (QLHV_G)' in 'MJ/kg' = ";
cin >> QLHV_G;
cout <<"\n Enter the 'number of Carbon atoms in Alcohol (ZC)' =";
cin >> ZC;
cout <<"\n Enter the 'number of Hydrogen atoms in Alcohol (ZH)'=";
cin >> ZH;
cout <<"\n Enter the 'number of Oxygen atoms in Alcohol (ZO)' =";
cin >> ZO;
cout <<"\n Enter the 'density of Alcohol (RHO_Al)'in 'kg/liter'=";
cin >> RHO_Alkgplit;
cout <<"\n Enter the 'Lower Heating Value of Alcohol(QLHV_AL)'in 'MJ/kg' = ";
cin >> QLHV_Al;
cout <<"\n Enter the 'percentage of Alcohol by mass' = ";
cin >> xAl_per;
cout <<"\n Enter the 'residual mass fraction (xres) in '%' = ";
cin >> xR_per;
cout <<"\n Enter 'Air-Factor' (LAMBDA)' = ";
cin >> LAMBDA;
cout <<"\n Enter the 'engine Load (pim)' in 'bar' = ";
cin >> pim;
cout <<"\n Enter the 'pressure (p1)' in 'bar' = ";
cin >> p1;
cout <<"\n Enter the 'Temperature (T1)' in 'K' = ";
cin >> T1;
```

```
cout <<"\n Enter the 'compression Ratio (Rc)' = ";
cin >> Rc;
cout <<"\n Enter the 'required engine speed (N)' in 'rpm' = ";
cin >> N;
cout <<"\n Enter the 'brake Power (Pb)' in 'kW' = ";
cin >> Pb;
cout <<"\n Enter 'bmep' in 'bar' = ";
cin >> bmep;
cout <<"\n Enter 'bsfc' in 'kg/kWh' = ";
cin >> bsfc;
cout <<"\n Enter the 'mechanical Efficiency (ETA_m)' in '%' = ";
cin >> ETA_mper;
cout <<"\n Enter 'theta_ivobTDC' in 'deg.' = ";
cin >> theta_ivobTDC;
cout <<"\n Enter 'theta_ivcaBDC' in 'deg.' = ";
cin >> theta_ivcaBDC;
cout <<"\n Enter 'theta_scbTDC' in 'deg.' = ";
cin >> theta_scbTDC;
cout <<"\n Enter 'theta_dc' in 'deg.' = ";
cin >> theta_dc;
cout <<"\n Enter 'theta_evobBDC' in 'deg.' = ";
cin >> theta_evobBDC;
cout <<"\n Enter 'theta_evcaTDC' in 'deg.' = ";
cin >> theta_evcaTDC;
  Zs = 4.0; Z_rpc = Zs/2.0; n_cps = N/(Z_rpc*60.0);
  Tim = T1-10.0; xR = xR_per/100.0;
  theta_a = 0.0; theta_0 = theta_evcaTDC; theta_b = 180.0;
  theta_ivc = 180.0+theta_ivcaBDC; theta_1 = theta_ivc;
  theta_sc = 360.0-theta_scbTDC; theta_2 = theta_sc; theta_c = 360.0;
  theta_ec = (theta_dc-theta_scbTDC)+360.0; theta_3 = theta_ec;
  theta_evo = 540.0-theta_evobBDC; theta_4 = theta_evo; theta_5 = 540.0;
  theta_6 = 720.0-theta_ivobTDC;
  Ec_liter = Ec_cc*1.0e-3; Ec = Ec_liter*1.0e-3;
  Vs = Ec/Zc; Vcl = Vs/(Rc-1.0); Vt = Vcl+Vs;
  Vs_cc = Vs*1.0e6; Vcl_cc = Vcl*1.0e6; Vt_cc = Vt*1.0e6;
  powd = (4.0*Vs)/(PI*Rsd); d = pow(powd,0.33); d_mm = d*1.0e3; s = d*Rsd;
  s_mm = s*1.0e3; r = s/2.0; r_mm = r*1.0e-3; lcon = r/Rrl;
  lcon_mm = lcon*1.0e3; Ap = (PI*d*d)/4.0;
cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
getch();
getch();
clrscr ();
cout << "\n\n\t\t  Adama Science and Technology University (ASTU)";
outfile <<"\n\n\t\t Adama Science and Technology University(ASTU)";
cout << " \n\t\t         Amsalu Fentahun(MSc.Student)";
```

```
outfile << " \n\t\t          Amsalu Fentahun(MSc.Student)";
cout << "\n...............................................................\n";
outfile << "\n...............................................................\n";
cout << "\n  Engine Size";
outfile << "\n  Engine Size";
cout << "\n -------------- ";
outfile << "\n ------------- ";
  cout <<setprecision(3)<<"\n  Ec = "<<Ec_cc<<" cc";
  cout <<setprecision(2)<<"\n  Vs = "<<Vs_cc<<" cc ";
  cout <<setprecision(2)<<"\n  Vcl = "<<Vcl_cc<<" cc ";
  cout <<setprecision(2)<<"\n  Vt = "<<Vt_cc<<" cc ";
cout << "\n\n\t\t Press 'enter' twice to proceed ...";
getch();
getch();
      MmO = 16.0; MmN = 14.0; MmC = 12.0; MmH = 1.0;
      yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
      xAl = xAl_per/100.0; xG = 1.0-xAl; xF = xAl+xG;
      MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
      nAl = xAl/MmAl; nG = xG/MmG; nAlG = nAl+nG;
      yAl = nAl/nAlG; yG = 1.0-yAl; yF = yAl+yG; yAl_per = yAl*100.0;
      MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yG*MmG)+(yAl*MmAl);
      n1th = (ZC*yAl)+(a*yG); n2th = ((ZH*yAl)+(b*yG))/2.0;
      atha = (2.0*n1th)+n2th-(ZO*yAl); ath = atha/2.0;
      AFth = (ath*4.76*MmA)/MmF; QLHV_F = (QLHV_Al*xAl)+(QLHV_G*xG);
      RHO_Fkgplit = (RHO_Alkgplit*yAl)+(RHO_Gkgplit*yG);
      RHO_F = RHO_Fkgplit;
 if (LAMBDA == 1.0)
  {
      n1 = (ZC*yAl)+(a*yG); n2 = ((ZH*yAl)+(b*yG))/2.0;
      n3 = 2.0*3.76*LAMBDA*ath; n4 = 0.0; n5 = 0.0; nt = n1+n2+n3+n4+n5;
      y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5/nt;
      AF = LAMBDA*AFth;
      }
 if (LAMBDA < 1.0)
  {
      n2 = ((ZH*yAl)+(b*yG))/2.0; n3 = (2.0*3.76*LAMBDA*ath)/2.0;
      n1 = yAL+(2.0*LAMBDA*ath)-(ZC*yAL)-(a*yG); n4 = 0.0;
      n5 = (ZC*yAL)+(a*yG)-n1; nt = n1+n2+n3+n4+n5;
      y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5/nt;
      AF = LAMBDA*AFth;
      }
 if (LAMBDA > 1.0)
  {
      n1 = (ZC*yAl)+(a*yG); n2 = ((ZH*yAl)+(b*yG))/2.0;
      n3 = 2.0*3.76*LAMBDA*ath; n4 = 0.0; n5 = 0.0; nt = n1+n2+n3+n4+n5;
```

```
        y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5/nt;
        AF = LAMBDA*AFth;
        }
        ETA_cmax = 0.95;
        ETA_c = ETA_cmax*(-1.6+(4.65*LAMBDA)-(2.07*LAMBDA*LAMBDA));
        ETA_cper = ETA_c*100.0;
        R1 = 0.287; Z1 = 0.99; V1 = Vt;
        mm = (p1*1.0e5*V1)/(Z1*R1*1.0e3*T1);
        mA = (AF/(AF+1.0))*(1.0-xR)*mm;
        mF = (1.0/(AF+1.0))*(1.0-xR)*mm; mres = xR*mm;
        ETA_m = ETA_mper/100.0;
    cout <<setprecision(2)<<"\n\n  ath = "<<ath<<"  ;   AF = "<<AF;
    cout <<setprecision(2)<<" ;  ETA_c = "<<ETA_cper<<" % ";
    cout <<setprecision(2)<<" ;  yAl_per = "<<yAl_per<<" % ";
    cout <<setprecision(6)<<"\n\n  mm = "<<mm<<" kg ";
    cout <<setprecision(6)<<"\n  mA = "<<mA<<" kg ";
    cout <<setprecision(6)<<"\n  mF = "<<mF<<" kg ";
    cout <<setprecision(6)<<"\n  mres = "<<mres<<" kg ";
    cout <<setprecision(2)<<"\n\n  MmF = "<<MmF<<" kg/Kmol ";
    cout <<setprecision(2)<<";  RHO_F = "<<RHO_F<<" kg/liter ";
    cout <<setprecision(2)<<";  QLHV_F = "<<QLHV_F<<" MJ/kg ";
    cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
    getch();
    getch();
  clrscr();
  cout << "\n\n\t\t  Adama Science and Technology University (ASTU)";
  cout << " \n\t  Department of Mechanical and Vehicle Engineering";
  cout << " \n\t\t         Amsalu Fentahun(MSc.Student) ";
  cout << "\n.........................................................\n";
  cout << "\n\n   'V', 'p' and 'T' as a function of 'theta' ";
  outfile << "\n\n  'V', 'p' and 'T' as a function of 'theta' ";
  cout << "\n -------------------------------------------\n";
  outfile << "\n\t------------------------------------------";
  cout <<"\n\ttheta\t  V\t  p\t  T ";
  cout <<"\n\t(deg.)\t (cc)\t (bar)\t (K) \n";
  outfile <<"\n\t   theta\t   V\t   p\t  T ";
  outfile <<"\n\t   (deg.)\t  (cc)\t  (bar)\t  (K) \n ";
       deltheta = 10.0;
   for (theta = 0.0; theta <= 720.0; theta = theta+deltheta)
    {
       V0bi = Vcl; p0bi = pim; T0bi = Tim; m0bi = mres; R0bi = R6b;
     if (theta >= theta_0 && theta <= theta_b)
      {
// process a-7 (constant pressure intake process).
        rad = theta*(PI/180.0); Z0b = 0.99;
```

```
        xp0b = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vs0b = Ap*xp0b; V0b = Vcl+Vs0b; p0b = p0bi; T0b = T0bi; R0b = 0.287;
        m0b = m0bi+((p0b*1.0e5*V0b)/(Z0b*R0b*1.0e3*T0b));
        Vb = V0b*1.0e6; pb = p0b; Tb = T0b; mb = m0b; thetab = theta;
          cout << setprecision(2)<<"\n\t"<<thetab;
          outfile << setprecision(2)<<"\n  " <<thetab;
          cout << setprecision(2)<<"\t "<<Vb;
          outfile << setprecision(2)<<"\t"<< Vb;
          cout <<setprecision(2)<<"\t "<<pb;
          outfile <<setprecision(2)<<"\t " << pb;
          cout <<setprecision(2)<<"\t"<<Tb;
          outfile <<setprecision(2)<<"\t " << Tb;
          getch();
           }
       Vb1i = Vb*1.0e-6; pb1i = pb; Tb1i = Tb; mb1i = mb;
     if (theta > theta_b && theta <= theta_1)
      {
// process b-1 (polytropic compression):
        rad = theta*(PI/180.0);
        xpb1 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vsb1 = Ap*xpb1; Vb1 = Vcl+Vsb1; VbVb1 = Vb1i/Vb1;
        nb1 = 1.35; nb1T = nb1-1.0; Rb1 = 0.287; Zb1 = 0.99;
        pb1 = pb1i*pow(VbVb1,nb1); Tb1 = Tb1i*pow(VbVb1,nb1T);
        delmb1 = mb1i-((pb1*1.0e5*Vb1)/(Zb1*Tb1*Rb1*1.0e3));
        mb1 = mb1i-delmb1;
        V1 = Vb1*1.0e6; p1 = pb1; T1 = Tb1; theta1 = theta;
          cout << setprecision(2)<<"\n\t"<<theta1;
          outfile << setprecision(2)<<"\n  " <<theta1;
          cout << setprecision(2)<<"\t "<<V1;
          outfile << setprecision(2)<<"\t"<< V1;
          cout <<setprecision(2)<<"\t "<<p1;
          outfile <<setprecision(2)<<"\t " << p1;
          cout <<setprecision(2)<<"\t"<<T1;
          outfile <<setprecision(2)<<"\t " << T1;
          getch();
           }
        m12i = mb1;
        V12i = V1*1.0e-6; p12i = p1; T12i = T1; m12i = mb1;
     if (theta > theta_1 && theta <= theta_c)
      {
// process 1-2 (polytropic effective compression):
        rad = theta*(PI/180.0);
        xp12 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vs12 = Ap*xp12; V12 = Vcl+Vs12; V1V12 = V12i/V12;
        n12 = 1.3; n12T = n12-1.0; R12 = 0.287; Z12 = 0.99; m12 = m12i;
```

```
   p12 = p12i*pow(V1V12,n12); T12 = T12i*pow(V1V12,n12T);
   W12 = (m12*R12*(T12-T12i))/(1.0-n12); W2 = W12;
   V2 = V12*1.0e6; p2 = p12; T2 = T12; theta2 = theta; W12i = W12;
      cout << setprecision(2)<<"\n\t"<<theta2;
      outfile << setprecision(2)<<"\n  " <<theta2;
      cout << setprecision(2)<<"\t "<<V2;
      outfile << setprecision(2)<<"\t"<< V2;
      cout <<setprecision(2)<<"\t "<<p2;
      outfile <<setprecision(2)<<"\t " << p2;
      cout <<setprecision(2)<<"\t"<<T2;
      outfile <<setprecision(2)<<"\t " << T2;
      getch();
       }
      W12 = W2; V23i = V2; p23i = p2; T23i = T2;
      Rp2xi = 1.1;
      do
      {
       Rp2x = Rp2xi; n2x = 1.3; n2xT = 1.0-n2x; theta33 = theta_3;
       rad33 = theta33*(PI/180.0); Rp = Rp2x;
       xp33 = r*((1.0-cos(rad33))+((Rrl*sin(rad33)*sin(rad33))/2.0));
       Vs33 = Ap*xp33; V33 = Vcl+Vs33; Rcf = V33/Vcl;
       Pi = Pb/ETA_m; Pi1 = Pi/Zc; Wi2x = Pi1/n_cps;
       imep2x = ((Wi2x*1000.0)/Vs)*1.0e-5;
       RSE2x = (imep2x*(n2x-1.0)*(Rc-1.0))/(p1*pow(Rc,n2x));
       LSE2xa = Rp*(pow(Rcf,n2x)-1.0);
       LSE2x = Rp*(Rcf-1.0)+(Rp-1.0)-(pow(Rc,n2xT)*LSE2xa);
       error2x = (RSE2x-LSE2x)/RSE2x; error2xper = abs(error2x);
       Rp2xi = Rp2x+0.1;
        }
       while (error2xper <= 0.01);
   Wx3 = Wi2x; Rp = Rp2x*0.73; px = p23i*Rp;
   Tx = T23i*(px/p23i); Vx = V23i;
   Vx3i = Vx*1.0e-6; px3i = px; Tx3i = Tx; theta_x = theta_c;
if (theta >= theta_x && theta <= theta_3)
 {
   rad = theta*(PI/180.0);
   xpx3 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
   Vsx3 = Ap*xpx3; Vx3 = Vcl+Vsx3; mx3 = m12i;
   px3 = px3i; Tx3 = ((px3/px3i)*(Vx3/Vx3i))*Tx3i;
   V3 = Vx3*1.0e6; p3 = px3; T3 = Tx3; m3 = mx3; theta3 = theta;
      cout << setprecision(2)<<"\n\t"<<theta3;
      outfile << setprecision(2)<<"\n  " <<theta3;
      cout << setprecision(2)<<"\t "<<V3;
      outfile << setprecision(2)<<"\t"<< V3;
      cout <<setprecision(2)<<"\t "<<p3;
```

```
            outfile <<setprecision(2)<<"\t " << p3;
            cout <<setprecision(2)<<"\t"<<T3;
            outfile <<setprecision(2)<<"\t " << T3;
            getch();
             }
         W23 = W3; Q23 = Q3; mF3 = mF23; mF23 = mF3;
         V34i = V3*1.0e-6; p34i = p3; T34i = T3; m34i = m12i;
      if (theta > theta_3 && theta <= theta_4)
       {
// process 3-4 (polytropic expansion):
         rad = theta*(PI/180.0);
         xp34 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
         Vs34 = Ap*xp34; V34 = Vcl+Vs34; V3V34 = V34i/V34; m34 = m12i;
         n34 = 1.3; n34T = n34-1.0; R34 = 0.287; Z34 = 0.99;
         p34 = p34i*pow(V3V34,n34); T34 = T34i*pow(V3V34,n34T);
         W34 = (m34*R34*(T34-T34i))/(1.0-n34); W4 = W34;
         V4 = V34*1.0e6; p4 = p34; T4 = T34; theta4 = theta;
            cout << setprecision(2)<<"\n\t"<<theta4;
            outfile << setprecision(2)<<"\n  " <<theta4;
            cout << setprecision(2)<<"\t "<<V4;
            outfile << setprecision(2)<<"\t"<< V4;
            cout <<setprecision(2)<<"\t "<<p4;
            outfile <<setprecision(2)<<"\t " << p4;
            cout <<setprecision(2)<<"\t"<<T4;
            outfile <<setprecision(2)<<"\t " << T4;
            getch();
             }
         W34 = W4;
         V45i = V4*1.0e-6; p45i = p4; T45i = T4; m45i = m12i;
      if (theta > theta_4 && theta <= theta_5)
       {
// process 4-5 (polytropic exhaust blowdown):
         rad = theta*(PI/180.0);
         xp45 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
         Vs45 = Ap*xp45; V45 = Vcl+Vs45; V4V45 = V45i/V45; m45 = m12i;
         n45 = 1.3; n45T = n45-1.0; R45 = 0.287; Z45 = 0.99;
         p45 = p45i*pow(V4V45,n45); T45 = T45i*pow(V4V45,n45T);
         delm45 = m45i-((p45*1.0e5*V45)/(Z45*T45*R45*1.0e3));
         m45 = m45i-delm45; W45 =(m45*R45*(T45-T45i))/(1.0-n45); W5 = W45;
         V5 = V45*1.0e6; p5 = p45; T5 = T45; theta5 = theta;
            cout << setprecision(2)<<"\n\t"<<theta5;
            outfile << setprecision(2)<<"\n  " <<theta5;
            cout << setprecision(2)<<"\t "<<V5;
            outfile << setprecision(2)<<"\t"<< V5;
            cout <<setprecision(2)<<"\t "<<p5;
```

```
         outfile <<setprecision(2)<<"\t " << p5;
         cout <<setprecision(2)<<"\t"<<T5;
         outfile <<setprecision(2)<<"\t " << T5;
         getch();
          }
         W45 = W5;
           }
         mF_kgph = bsfc*Pb; mF_kgps = mF_kgph/3600.0;
         mF = mF_kgps/(Zc*n_cps);
         Qx3 = (mF*QLHV_F*1.0e3)*ETA_c; Qin = Qx3; Wnet = Wx3; // W12+W23+W34;
         ETA_ith = Wnet/Qin; ETA_ithper = ETA_ith*100.0;
         imep_bar = ((Wnet*1.0e3)/Vs)*1.0e-5; imep = imep_bar;
         Wi = Wnet*Zc; Pi = Wi*n_cps; Pb_kW = Pi*ETA_m;
   cout <<setprecision(2)<<"\n\n  Rp = "<<Rp<<"  ";
   cout <<setprecision(2)<<";  Rcf = "<<Rcf<<"  ";
   cout <<setprecision(2)<<";  ETA_ith = "<<ETA_ithper<<" % ";
   cout <<setprecision(2)<<";  ETA_m = "<<ETA_mper<<" % ";
   cout <<setprecision(2)<<"\n  imep = "<<imep_bar<<" bar ";
   cout <<setprecision(2)<<";  Pb = "<<Pb_kW<<" kW ";
   cout <<setprecision(2)<<";  bsfc = "<<bsfc<<" kg/kWh ";
   cout << "\n\n\n\t\t Press 'enter' twice to exit ! ";
   getch();
   getch();
   getche();
      }
```

Program # 4

```
#include<iostream.h>
#include<conio.h>
#include<math.h>
#include<fstream.h>
#include<iomanip.h>
  void main()
   {
      int ap;
    float Ec_liter,Vs,Vs_cc,Vc_cc,Zc,rcr,d,dmm,s,smm,l,sd_ratio,rcrl_ratio;
    float
rcrmm,Ap,omega,theta_pgmax,rad_pgmax,Nmax,xp,xpmm,vp,pg_max,Ec_cc;
    float Fg_max,Fi,Fp,Fs,Fcr,Fr,Ft,Tmax,rad3,r,PHI_pgmax,OMEGA_pgmax,N;
    float mass_r,THETA_pgmax,N_Tbmax,rad_THETA,rad_PHI,bmep_max,Rsd,Rrl,Rc;
    const float PI = 3.1416;
 clrscr();
```

```
 ofstream outfile ("A4-CI.dat");
 clrscr();
  cout <<"\n\n\t\t  Adama Science and Technology University (ASTU)";
  cout <<"\n\t    Department of Mechanical and Vehicle Engineering";
  cout <<"\n\t\t         Amsalu Fentahun(MSc.Student)";
  cout <<"\n\t..........................................................";
  outfile <<"\n\t................................................";
 cout <<"\n\n Enter the 'engine capacity (Ec)' in 'cc' =";
 cin >> Ec_cc;
cout <<"\n Enter the 'stroke-to-bore Ratio (Rsd)' =";
 cin >> Rsd;
 cout <<"\n Enter the 'crank radius to con-rod ratio (Rrl)' =";
 cin  >> Rrl;
 cout <<"\n Enter the 'number of cylinders (Zc)' =";
 cin >> Zc;
 cout <<"\n Enter the 'maximum gas pressure (pg_max)' in 'bar' =";
 cin >> pg_max;
 cout <<"\n Enter the 'crank angle at the maximum gas pressure'(10 to 20
deg. aTDC) =";
 cin >> THETA_pgmax;
 cout <<"\n Enter the 'mass of reciprocating parts'(1.5 to 2.5kg)=";
 cin >> mass_r;
 cout <<"\n Enter the 'engine speed (N)' in 'rpm' =";
 cin >> N;
  sd_ratio = Rsd; rcrl_ratio = Rrl; N_Tbmax = N; // r = Rc;
     Vs_cc = Ec_cc/Zc; Vs = Vs_cc*1.0e-6;
       d = pow(((4.0*Vs)/(PI*sd_ratio)),0.33);
        dmm = d*1000.0;
        smm = dmm*sd_ratio;
     rcrmm = smm/2.0;
        rcr = rcrmm*1.0e-3;
       OMEGA_pgmax = N_Tbmax*(2.0*PI)/60.0;
        rad_THETA = (PI/180.0)*THETA_pgmax;
 xp = rcr*((1.0-
cos(rad_THETA))+(rcrl_ratio*(sin(rad_THETA)*sin(rad_THETA))/2.0));
```

```
 vp=OMEGA_pgmax*rcr*(sin(rad_THETA)+(rcrl_ratio*sin(2.0*rad_THETA)/2.0));
ap=OMEGA_pgmax*OMEGA_pgmax*rcr*(cos(rad_THETA)+(rcrl_ratio*cos(2.0*rad_THET
A)));
      PHI_pgmax = asin(rcrl_ratio*sin(THETA_pgmax));
      rad_PHI = (PI/180.0)*PHI_pgmax;
      rad3 = ((THETA_pgmax+PHI_pgmax)*PI)/180.0; xpmm = xp*1000.0;
       Ap = (PI*d*d)/4.0;
      Fg_max = (pg_max*1.0e5*Ap)*1.0e-3;
       Fi = (mass_r*ap)*1.0e-3;
       Fp = (Fg_max-Fi);
       Fs = Fp*tan(rad_PHI);
      Fcr = Fp/cos(rad_PHI);
       Fr = Fcr*cos(rad3);
       Ft = Fcr*sin(rad3);
     Tmax = (Ft*1000.0)*rcr;
  clrscr();
  cout <<"\n\n\t\t  Adama Science and Technology University (ASTU)";
  cout <<"\n\t    Department of Mechanical and Vehicle Engineering";
  cout <<"\n\t\t          Amsalu Fentahun(MSc.Student)";
  cout <<"\n\t...................................................";
  outfile <<"\n\t................................................";
     cout <<"\n\n   Engine Kinematics";
     cout <<"\n  -------------------";
     cout <<"\n\t xp = "<<setprecision(2)<<xpmm<<" mm";
     cout <<"\n\t vp = "<<setprecision(2)<<vp<<" m/s";
     cout <<"\n\t ap = "<<setprecision(2)<<ap<<" m/s*s";
     outfile <<"\n\n\n   Engine Kinematics ";
     outfile <<"\n  -------------------";
     outfile <<"\n\n\t xp = "<<setprecision(2)<<xpmm<<" mm";
     outfile <<"\n\n\t vp = "<<setprecision(2)<<vp<<" m/s";
     outfile <<"\n\n\t ap = "<<setprecision(2)<<ap<<" m/s*s";
     cout <<"\n\n   Engine Kinetics ";
     cout <<"\n  -----------------";
     cout <<"\n\t Fg = "<<setprecision(2)<<Fg_max<<" kN";
     cout <<"\n\t Fi = "<<setprecision(2)<<Fi<<" kN";
     cout <<"\n\t Fp = "<<setprecision(2)<<Fp<<" kN";
     cout <<"\n\t Fs = "<<setprecision(2)<<Fs<<" kN";
```

```
    cout <<"\n\t Fcr = "<<setprecision(2)<<Fcr<<" kN";
    cout <<"\n\t Fr = "<<setprecision(2)<<Fr<<" kN";
    cout <<"\n\t Ft = "<<setprecision(2)<<Ft<<" kN";
    cout <<"\n\t T = "<<setprecision(2)<<Tmax<<" Nm";
    outfile <<"\n\n\n   Engine Kinetics ";
    outfile <<"\n  -----------------";
    outfile <<"\n\n\t Fg = "<<setprecision(2)<<Fg_max<<" kN";
    outfile <<"\n\n\t Fi = "<<setprecision(2)<<Fi<<" kN";
    outfile <<"\n\n\t Fp = "<<setprecision(2)<<Fp<<" kN";
    outfile <<"\n\n\t Fs = "<<setprecision(2)<<Fs<<" kN";
    outfile <<"\n\n\t Fcr = "<<setprecision(2)<<Fcr<<" kN";
    outfile <<"\n\n\t Fr = "<<setprecision(2)<<Fr<<" kN";
    outfile <<"\n\n\t Ft = "<<setprecision(2)<<Ft<<" kN";
    outfile <<"\n\n\t T = "<<setprecision(2)<<Tmax<<" Nm";
  cout <<"\n\n\t\t Press 'Enter' twice to exit ! ";
  getch();
  getch();
 getche();
   }
```

Appendix – B. Part and Assemble Drawing

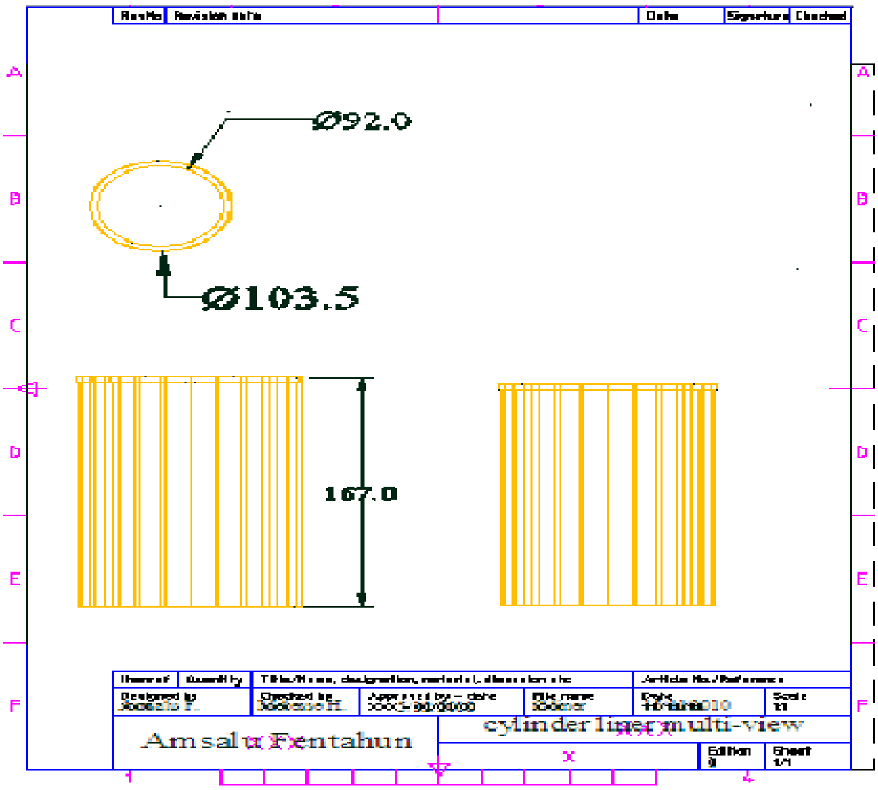


Figure B. 1 Two Dimensional Sketch of Cylinder Liner

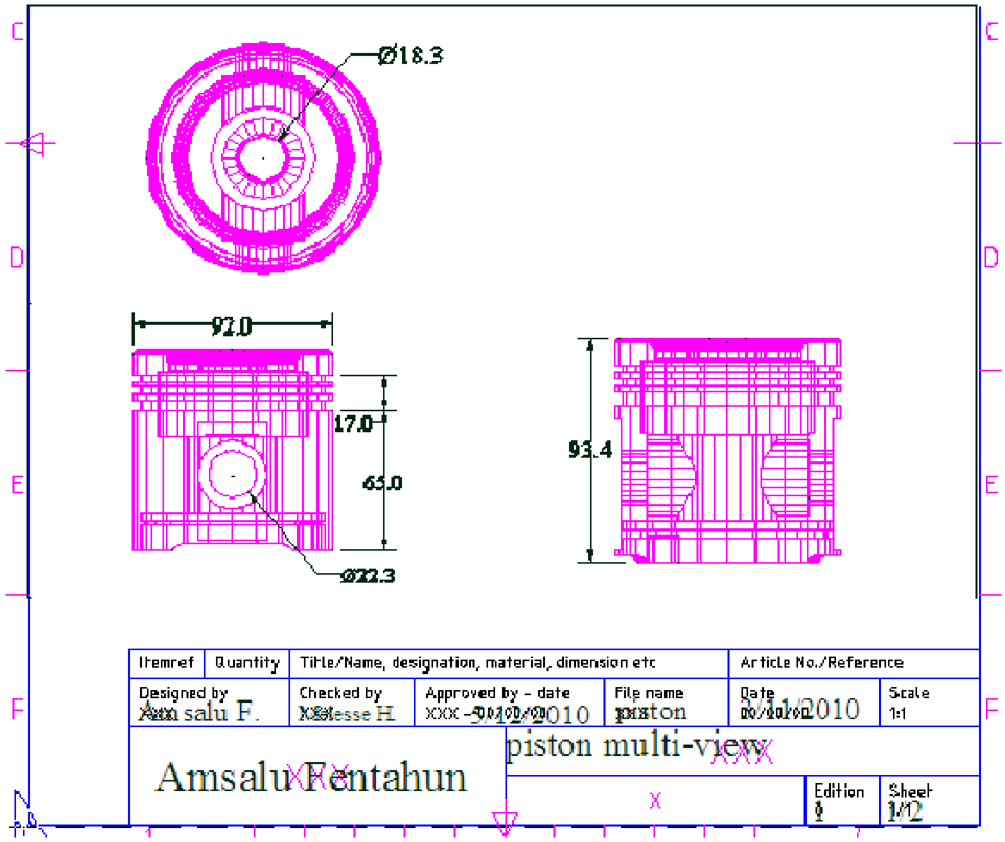


Figure B. 2 Two Dimensional Sketch of Piston

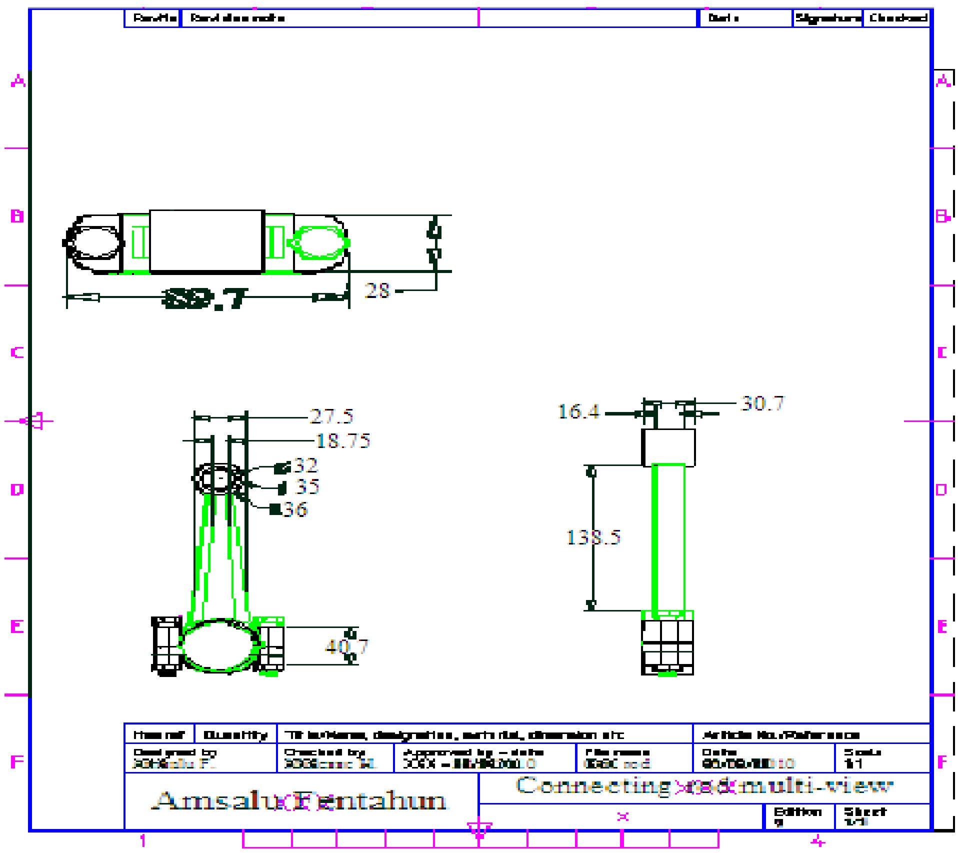


Figure B. 3 Two Dimensional Sketch of Connecting rod

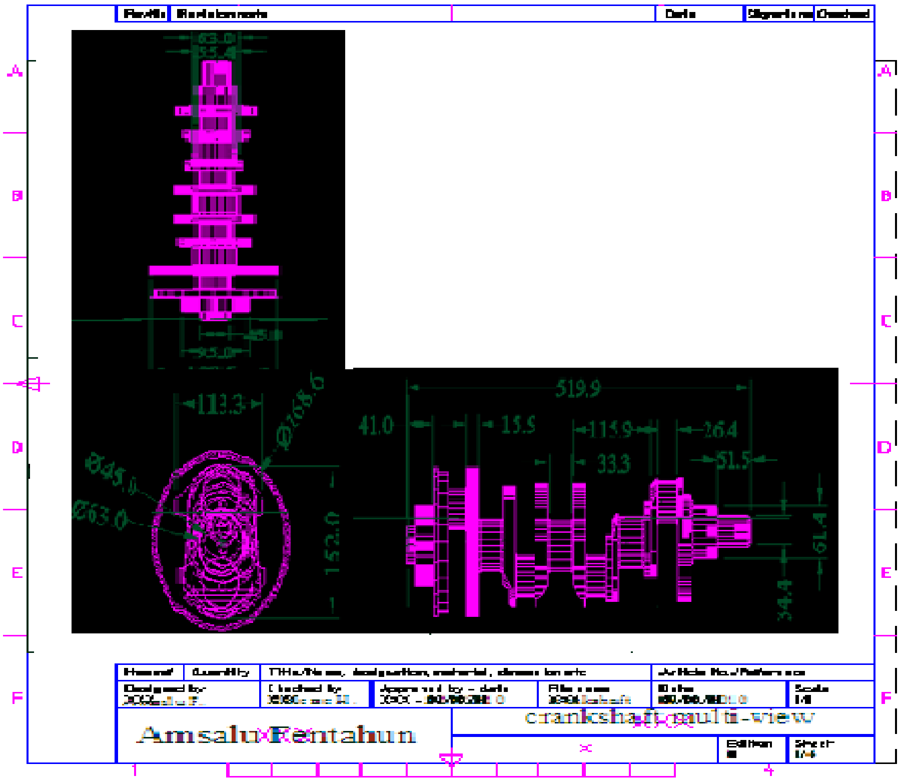


Figure B. 4 Two dimensioal Sketch of Cranckshaft

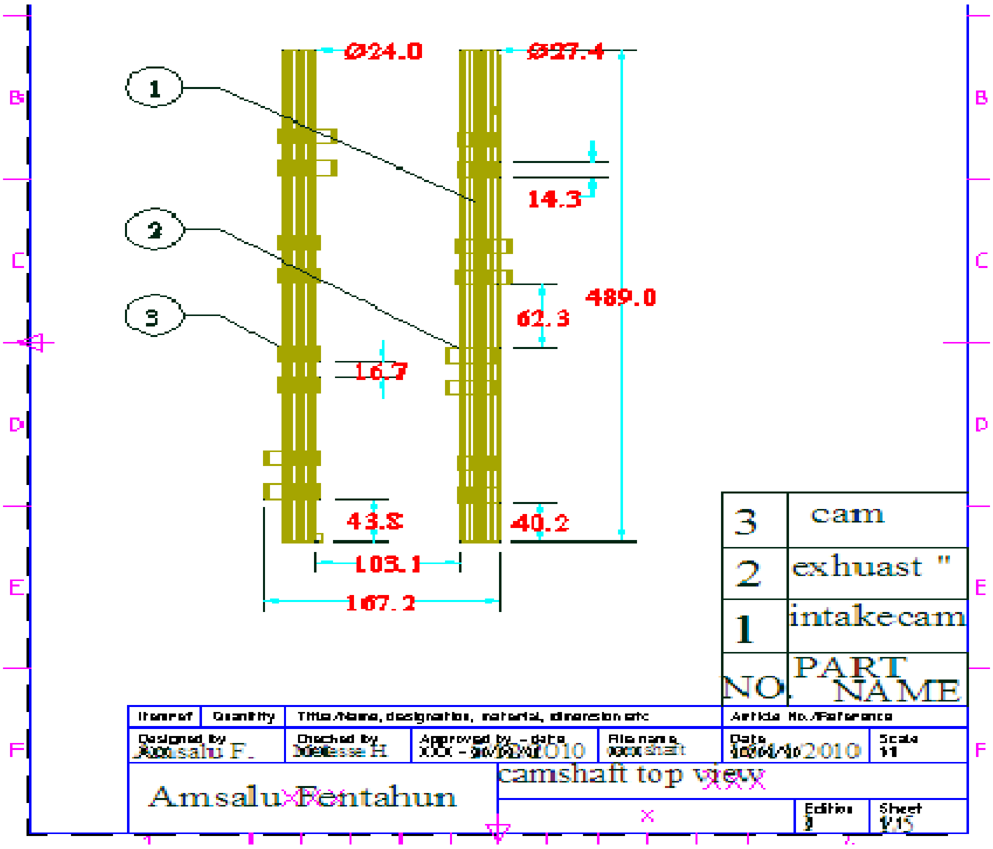


Figure B. 5 Two Dimensional Sketch of Camshaft

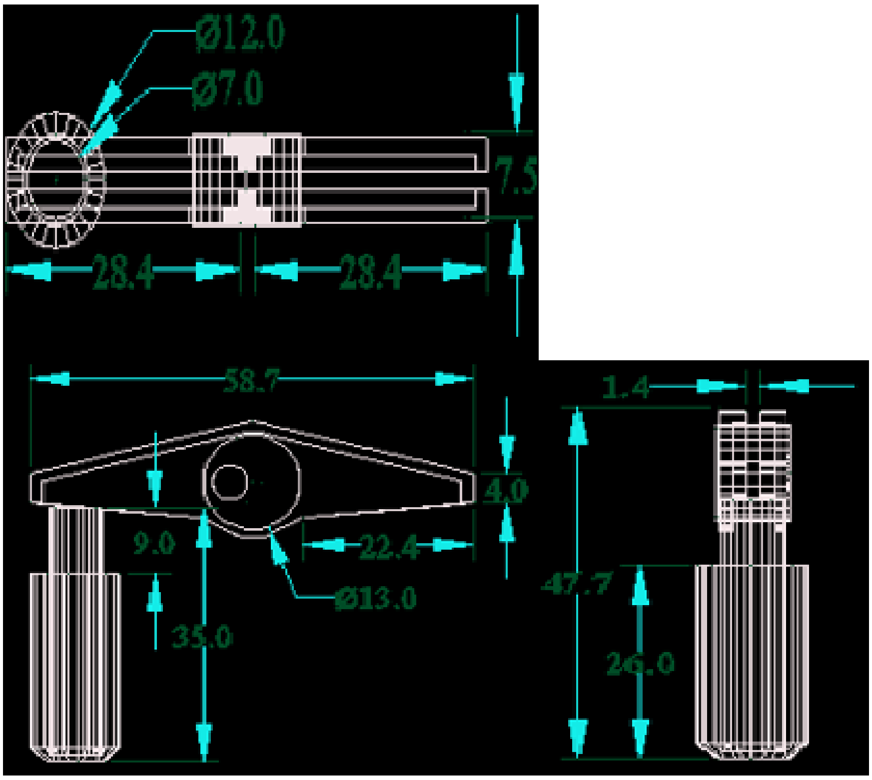


Figure B. 6 Two dimensional Sketch of Rocker Arm

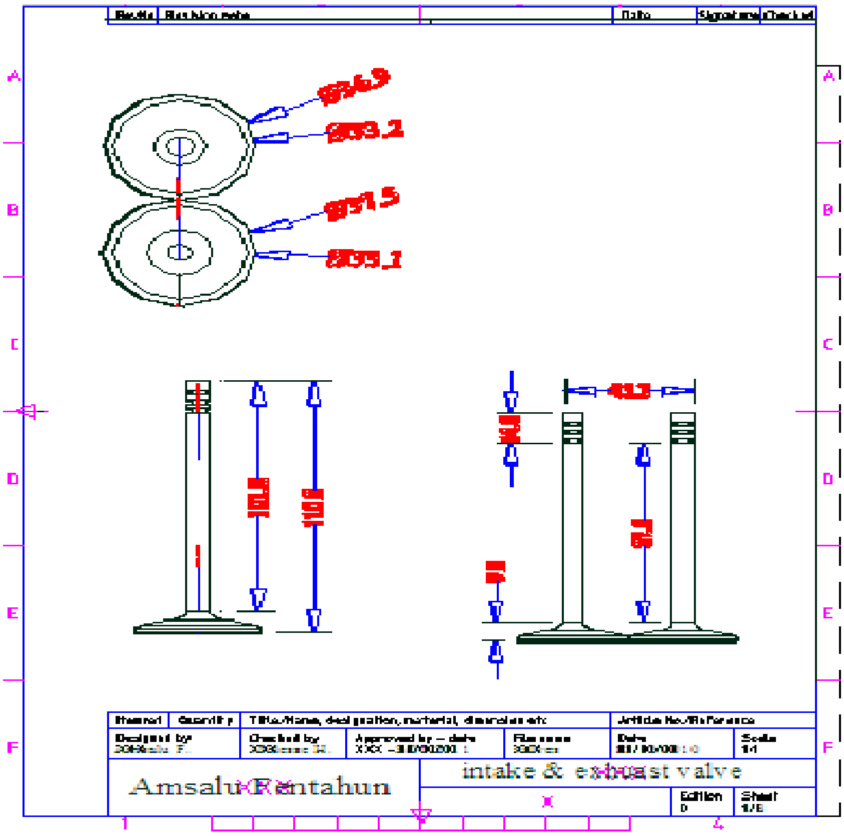


Figure B. 7 Two Dimensional Sketch of Intake and Exuast Valve

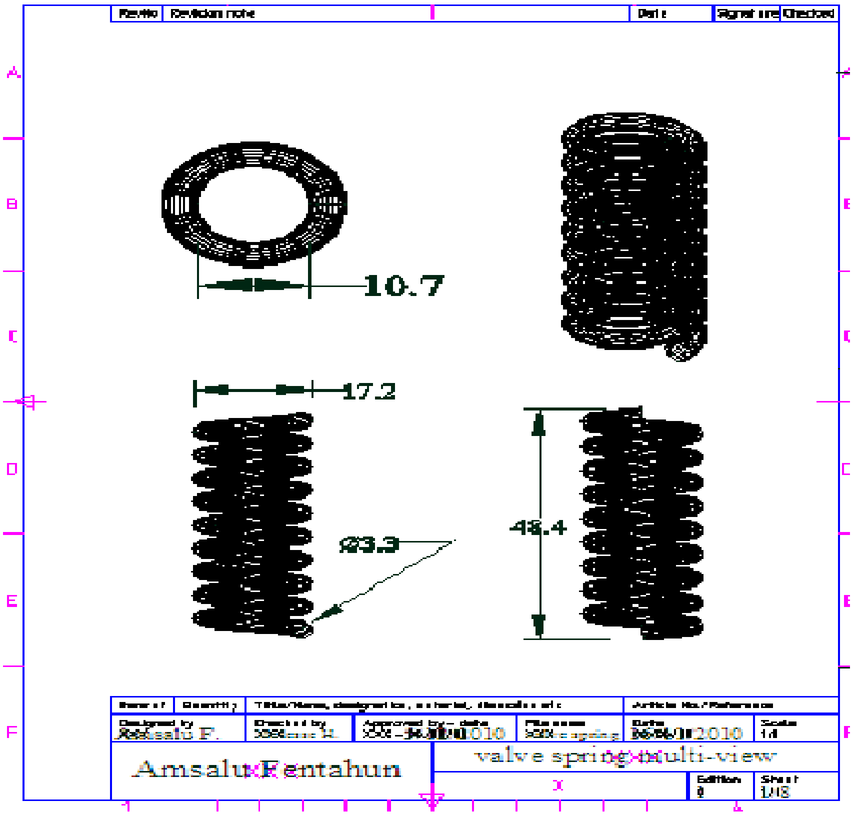


Figure B. 8 Two Dimensional Sketch of Valve Spring

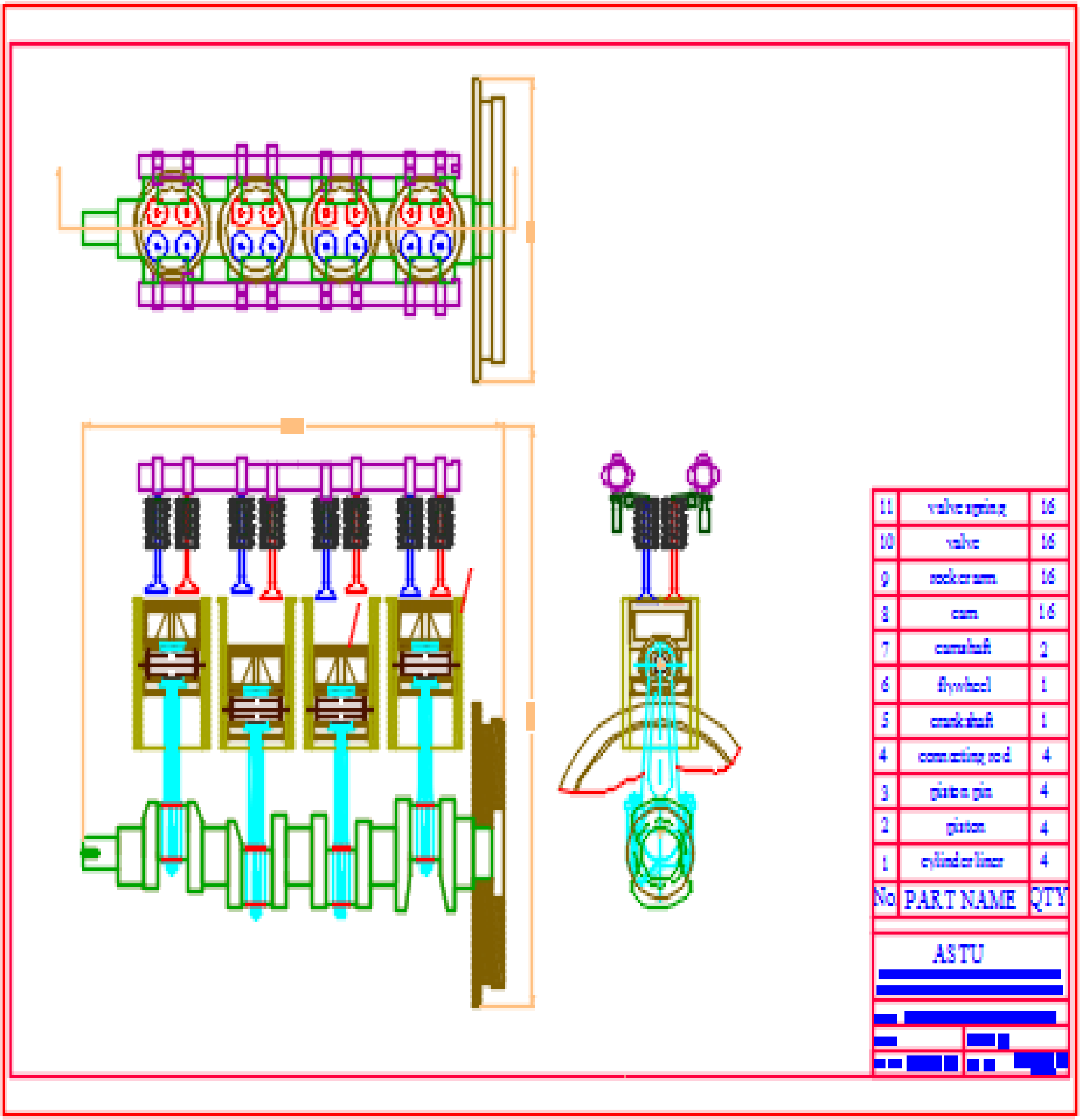


Figure B. 9 Two dimensional Sketch of Engine Assembly

Appendix – C. Mechanical Property of Selection Materials

Table B. 1 Physical Properties of some Metals.

Metal	Density (kg/m^3)	Melting point (°C)	Thermal conductivity ($W/m°C$)	Coefficient of linear expansion at 20°C ($\mu m/m°C$)
Aluminium	2700	660	220	23.0
Brass	8450	950	130	16.7
Bronze	8730	1040	67	17.3
Cast iron	7250	1300	54.5	9.0
Copper	8900	1083	393.5	16.7
Lead	11400	327	33.5	29.1
Monel metal	8600	1350	25.2	14.0
Nickel	8900	1453	63.2	12.8
Silver	10500	960	420	18.9
Steel	7850	1510	50.2	11.1
Tin	7400	232	67	21.4
Tungsten	19300	3410	201	4.5
Zinc	7200	419	113	33.0
Cobalt	8850	1490	69.2	12.4
Molybdenum	10200	2650	13	4.8
Vanadium	6000	1750	_	7.75

Source: R.S.Khurmi, A Text Book of Machine Design.

Table B. 2 Dimensions of Metric (M) Bolts and Nuts

Designation	Pitch Mm	Major or nominal diameter Nut and Bolt (d = D) mm	Effective or pitch diameter Nut and Bolt (dp) mm	Minor or core diameter (dc) mm		Depth of thread (bolt) mm	Stress area mm^2
				Bolt	Nut		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Coarse series							
M 12	1.75	12.000	10.863	9.858	10.106	1.074	84.0
M 14	2	14.000	12.701	11.546	11.835	1.227	115
M 16	2	16.000	14.701	13.546	13.835	1.227	157
M 18	2.5	18.000	16.376	14.933	15.294	1.534	192
M 20	2.5	20.000	18.376	16.933	17.294	1.534	245

Table B. 3 Design Data for Bearings.

Machinery	Bearing	Maximum bearing pressure (p) in N/mm^2	Operating values			
			Absolute viscosity (Z) in $\frac{kg}{m} - s$	ZN/Z Z in $\frac{kg}{m} - s$ P in N/mm^2	$\frac{c}{d}$	$\frac{l}{d}$
Automotive and air-craft engines	Main	5.6- 12	0.007	2.1	-	0.8-1.8
	crank pin	10.5-24.5	0.008	1.4		0.7-1.4
	Wrist pin	16-35	0.008	1.12		1.5-2.2
Four stroke-gas and oil engines	Main	5-8.5	0.02	2.8	0.001	0.6-2
	crank pin	9.8-12.6	0.04	1.4		0.6-1.5
	Wrist pin	12.6-15.4	0.065	0.7		1.5-2
Two stroke-gas and oil engines	Main	3.5-5.6	0.02	3.5	0.001	0.6-2
	crank pin	7-10.5	0.04	1.8		0.6-1.5
	Wrist pin	8.4-12.6	0.065	1.4		1.5-2
Marine steam engines	Main	3.5	0.03	2.8	0.001	0.7-1.5
	crank pin	4.2	0.04	2.1		0.7-1.2
	Wrist pin	10.5	0.05	1.4		1.2-1.7
Stationary, slow speed steam engines	Main	2.8	0.06	2.8	0.001	1-2
	crank pin	10.5	0.08	0.84		0.9-1.3
	Wrist pin	12.6	0.06	0.7		1.2-1.5
Stationary, high speed steam engines	Main	1.75	0.015	3.5	0.001	1.5-3
	crank pin	4.2	0.030	0.84		0.9-1.5
	Wrist pin	12.6	0.025	0.7		1.3-1.7
Reciprocating pumps and compressors	Main	1.75	0.03	4.2	0.001	1-2.2
	crank pin	4.2	0.05	2.8		0.9-1.7
	Wrist pin	7.0	0.08	1.4		1.5-2.0
Steam locomotives	Main	3.85	0.10	4.2	0.001	1.6-1.8
	crank pin	14	0.04	0.7		0.7-1.1
	Wrist pin	28	0.03	0.7		0.8-1.3

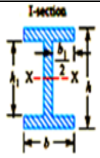
Table B. 4 Mechanical Properties of some Mild Steels.

Indian standard designation	Ultimate tensile strength, MPa	Yield strength, MPa
40 C 8	560 – 670	320
45 C 8	610 – 700	350
50 C 4	640 – 760	370
50 C 12	700 Min	390

Source: R.S.Khurmi, A Text Book of Machine Design.

Note: These materials are widely used in: Crankshafts, Camshafts, Connecting rods, Push rods, etc.

Table B. 5 Properties of some Commonly used Cross-Sections.

section	Area (A)	Moment of inertia (I)	Distance from the neutral axis to the extreme fiber (y)	Section of modulus $\left[Z = \frac{I}{y}\right]$	Radius of gyration $\left[k = \sqrt{\frac{I}{y}}\right]$
	$bh = b_1h_1$	$I_{xx} = \frac{bh^3 - b_1h_1{}^3}{12}$	$\frac{h}{2}$	$Z_{xx} = \frac{bh^3 - b_1h_1{}^3}{6h}$	$K_{xx} = 0.289\sqrt{\frac{bh^3 - b_1h_1{}^3}{bh - b_1h_1}}$

Source: R.S.Khurmi, A Text Book of Machine Design.

Note: The Polar Moment of Inertia (J) is given by, $J = I_{xx} + I_{yy}$

Table B. 6 Allowable Bending and Shear Stresses.

Material	Endurance limit in MPa		Allowable stress in MPa	
	Bending	Shear	Bending	Shear
Chrome nickel	525	290	130 to 175	72.5 to 97
Carbon steel and cast steel	225	124	56 to 75	31 to 42
Alloy cast iron	140	140	35 to 47	35 to 47

Table B. 7 Mechanical Properties of some Mild Steels (SAE), Malleable and Ductile Cast Irons (ASTM). American Standards (SAE and ASTM)

SAE designation	Tensile Strength(σ_u) (MPa)	Yield Strength (σ_{yl}) (MPa)	Modulus of Elasticity (E) (GPa)
SAE 1010	365	305	200
SAE 1020	420	205	200
SAE 1030	525	440	200
SAE 1045	625	530	201
ASTM A220 (Malleable Cast Iron)	586	483	172
ASTM A536 (Ductile Cast Iron)	496	345	172

Source: http://www.substech.com/dokuwiki/doku.php.

Note: Malleable Cast Irons are widely used in: bearing caps, parts of power train of vehicles, steering gear housings, etc. while Ductile Cast Irons and Mild Steels are widely used in: crankshafts, camshafts, heavy duty gears, etc.

Appendix – D. Cost Analysis of Prototype of Diesel Engine Mechanism Parts

Table C. 1 Raw Materials Cost

No.	Items description	Unit	quantity	Unit price (ETB)	Total price (ETB)
1	Pipe for liner (diameter 50 mm, length 90 mm)	Pieces	4	50	200
2	wood for piston (diameter 40mm, length 50 mm)	Pieces	4	5	20
3	Sheet metal for stand with support (width 25, total length 27x4+120 mm, and thickness 2 mm)	Pieces	1	60	60
4	mild steel bar for crankshaft (diameter 30, length 100mm)	Pieces	1	250	250
5	Bearing with housing	No.	2	100	200
6	Chain length 82 mm	No.	4	70	280
7	mild steel bar for cam(diameter 40 mm,	No.	1	220	220
		Pieces	1	400	400

	length 450 mm)				
8	round bar (diameter 20 mm, length 60 mm)	pieces	8	25	200
9	compression Spring	No. 8	8	10	80
10	sprocate (D=100 mm, d= 50 mm)	No. 2	2	120,50	240, 100

Table C. 2 Production Cost

No.	Types of machine	Machine cost/hr.	Labor cost/hr.	Working hour	Cost (ETB)
1	Welding machine	180	250	12 hours	430
2	Grinding machine	30	50	6 hours	80
3	Rolling machine	70	150	6 hours	220
4	Metal cutting	120	180	3 hours	300
5	Universal drill machine	50	150	1 hours	200
Sub total		**350**	**780**	**28 hours**	**1230**

Table C. 3 Cost Summary

No.	1	2	3	4	5	6	Total cost
Variable	Raw materials	Material wastage	production	overhead	profit	Sell tax	
Cost (ETB)	2050	-	1230	-	-		**3280**