KAUNAS UNIVERSITY OF TECHNOLOGY
FACULTY OF MECHANICAL ENGINEERING AND DESIGN

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DESIGN AND ANALYSIS OF AXIAL INTERNAL COMBUSTION ENGINE

Master‘s Degree Final project

Supervisor
Assoc. Prof. Dr. Sigitas Kilikevicus

KAUNAS, 2016

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KAUNAS, 2016

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Design & Analysis of Axial Internal Combustion Engine

DECLARATION OF ACADEMIC INTEGRITY

2016 June 09

Kaunas

I confirm that the final project of mine, **Mohamed Riyazdheen Shahul Hameed**, on the subject "Design & Analysis of Axial IC-Engine with piston axis parallel to drive shaft" is written completely by myself; all the provided data and research results are correct and have been obtained honestly. None of the parts of this thesis have been plagiarized from any printed, Internet-based or otherwise recorded sources. All direct and indirect quotations from external resources are indicated in the list of references. No monetary funds (unless required by law) have been paid to anyone for any contribution to this thesis.

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(name and surname filled in by hand) *(signature)*

Mohamed Riyazdheen Shahul Hameed. DESIGN AND ANALYSIS OF AXIAL INTERNAL COMBUSTION ENGINE. *Master's* Final Project / supervisor assoc. prof. Dr. Sigitas Kilikevicus; Faculty of Mechanical Engineering & Design, Kaunas University of Technology.

Research field and area

Keywords: *Barrel engine, Finite Element Methods, Connecting rod, Fatigue failure.*

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SUMMARY

The work deals with design of six-cylinder piston engine with piston axis parallel to drive shaft and to run analysis studies using Solid Works. This paper presents a unique design of barrel engine specific mechanisms to convert reciprocating motion of the piston to rotary motion of the crank shaft. Unlike normal IC engines this engine has six separate crank shafts and a unique cam design.

The Engine was designed using Solid Works computer aided design software. This unique design reduces and replaces components in the engine. The analysis study is about analysing the maximum torque and reaction forces generated by this design using SolidWorks Motion. The reaction forces calculated for the connecting rod by analytically and using SolidWorks Motion is used in further study. In an engine a connecting rod is always subjected to complex loading of cyclic loads due to high compressive load and tensile loads which lead to fatigue failure. So further study is conducted by detailed load analysis using Finite Element Methods; and followed by fatigue analysis to predict the life of the connecting rod using SolidWorks. The study is done using both computer simulation and analytical calculation using MATLAB. The research indicates introducing a new four stroke engine, the differences in connecting rod life for different loading ratios and to estimate maximum torque developed by the engine design.

Mohamed Riyazdheen Shahul Hameed AŠINIO VIDAUS DEGIMO VARIKLIO PROJEKTAVIMAS IR TYRIMAS. Magistro baigiamasis projektas / vadovas doc. dr. Sigitas Kilikevičius; Kauno technologijos universitetas.

Raktiniai žodžiai: ašinis variklis, baigtinių elementų metodas, švaistiklis, nuovargis

Kaunas, 2016. 51 psl.

SANTRAUKA

Darbe atliktas ašinio vidaus degimo variklio, kuriame stūmoklių ašis lygiagreti varomajam velenui, projektavimas ir tyrimas naudojant Solid Works. Pateikiamas unikalus projektinis sprendimas slenkamąjį stūmoklio judesį pakeisti alkūninio veleno sukamuoju judesiu. Skirtingai nei įprastame vidaus degimo variklyje, šis variklis turi šešis atskirus alkūninius velenus ir unikalią kumštelių konfigūraciją. Variklis buvo sukurtas naudojant Solid Works kompiuterinio projektavimo programinę įrangą. Šis unikalus projektinis sprendimas leidžia sumažinti variklio komponentų skaičių. Judesio analizės programinė įranga Solid Works Motion panaudota analizuojant didžiausią sukimo momentą ir reakcijos jėgas ir gauti tyrimo rezultatai palyginami su analitiniu metodo sprendiniu, gautu panaudojant MATLAB programinę įrangą. Atsižvelgiant į tai, kad variklio švaistiklis veikiamas intensyvių tempimo ir gniuždymo ciklinių apkrovų, dėl ko gali atsirasti nuovarginis lūžis, baigtinių elementų metodu atliktas švaistiklio nuovargio tyrimas naudojant Solid Works Simulation. Išanalizavus rezultatus parinkta tinkama švaistiklio medžiaga ir geometrinė forma. Darbe atlikti tyrimai parodė, kad pasiūlytas naujas keturtaktis variklis atitinka varikliams keliamus ilgaamžiškumo ir galios reikalavimus.

KAUNAS UN IVERSITY OF TECHNOLOGY
FACULTY OF MECHANICAL ENGINEERING AND DESIGN

Approved:
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(Name, Surname)

MASTER STUDIES FINAL PROJECT TASK ASSIGNMENT
Study programme MECHANICAL ENGINERING - 621H30001

Approved by the Dean's Order No.V25-11-7 of May 3rd , 2016 y

Assigned to the student Mohamed Riyazdheen Shahul Hameed
(Name, Surname)

1. Title of the Project

Design and Analysis of Axial Internal Combustion Engine

2. Aim of the project

The aim of this thesis is to introduce a new design/model of an internal combustion axial engine (barrel engine); and investigate its characteristics and operation.

3. Tasks of the project

Summary, Introduction, Literature review (selection of engine type and methodology of predicting fatigue life), An illustration on the design of axial IC-engine (3D model created with precise design calculation and selection of material for connecting rod for fatigue study), Fatigue model life prediction (Kinematic analysis to plot reaction forces, numerical calculation for reaction forces using Mat Lab, Fatigue analysis, numerical calculations of life, 3-D assembly drawing.

4. Specific Requirements

- This engine is designed to produce high torque and power with reduction in size and number of parts.
- A connecting rod is subjected to a cyclic load over cycles; So the part is to check and redesign the connecting rod to have an infinity fatigue life

5. This task assignment is an integral part of the final project

6. Project submission deadline: 2016 May 20th.

Task Assignment received Mohamed Riyazdheen Shahul Hameed _______________
(*Name, Surname of the Student*) (*Signature, date*)

Supervisor Assoc. prof. Dr. Sigitas Kilikevičius _______________
(*Position, Name, Surname*) (*Signature, date*)

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INTRODUCTION

All of us almost take internal combustion engines for granted. All we do is buy a vehicle, hop in and drive around. The very first engines that had been conceived wasn't even like the engine we know today. The IC-engine was conceived and developed in the late 1800s. It had a significant impact on the society and is considered one of the most significant inventions of the last century. The IC-engine is the foundation for all the successful development of many commercial technologies. For example, consider how this type of engines has transformed the transportation industry that allowed the invention of automobiles, trucks, airplanes and trains to a next level.

An IC-engine is defined as an engine in which energy of the fuel is released inside the engine and used directly to do mechanical work, opposed to an external combustion engine in where a separate combustor is used to burn the fuel. Internal combustion engines can deliver power in the range from 0.01 kW to 20×10^3 kw, depending on their displacement. The components of a reciprocating IC- engine such as blocks, pistons, valves, crankshaft and connecting rod have remained unchanged since the late 1800s. The deference between today's engines and ones built centuries ago are the thermal efficiency, emission level and their size. For many years IC-engine research was aimed at improving thermal efficiency, weight and size reduction, and also reducing noise and vibration. Currently emission control requirements and weight reduction is one of major factors in the design and performance of internal combustion engines. [1]

The aim of this thesis is to introduce a new design/model of an internal combustion axial engine *(barrel engine)*; and investigate its characteristics and operation. The major tasks raised to reach the aims of project are:

1. literature on the topic has to be reviewed for the selection of engine type, and the investigation method used to estimate the fatigue life of connecting rod;
2. to design an engine model, having all the necessary calculations concerning with kinematics, dynamics and strength for the materials selected for basic design;
3. to conduct a motion study to estimate the maximum torque developed and reaction forces acting on the connecting rod for the analysis and also make comparison with calculated reaction force. The largest force is used to design the rod for infinity life.
4. to conduct a static FEA for the second part of the study using the reaction force for two different materials AISI4340 and C70 steel.
5. to conduct a fatigue study using the stress results obtained by static FEA, the fatigue analysis with two different loading ratios is carried out for the chosen materials AISI4340 and C70 steel.

6. to create 3D model and animation of internal combustion axial engine using SolidWorks_2015 and key shot. And also a 2-D assembly drawing.

Therefore, the overview is to design an axial engine, estimate the reaction force acting on connecting rod; and study the causes and area of failure with help of FEA and redesigned the connecting rod for infinity life cycles.

1. LITERATURE SURVEY ON AXIAL ENGINE

1.1. Engine:

An engine is a machine that can convert energy (fuel) into useful mechanical work (power or motion). If the engine produces kinetic energy (motion) from fuel source, it is called a prime mover; if it produces kinetic energy from a pre-processed fuel like electricity, hydraulic fluid is called as motor. A main device that powers an automobile is called engine. A locomotive is also referred to as an engine.

Originally at earlier an engine is a mechanical device which were made to converted force into some reaction (motion). Military devices such as battering rams, catapults, and trebuchets were mentioned as *siege engines*. The term gin is from word cotton gin is identified as a short form of the Old French word *engine*, in turn derived from the Latin word called *ingenium*, which is in turn related to *ingenious.* Most devices used in Industrial revolution were referred as engines, and this is the how the steam engine received its name.

In modern term *engine* is describe to devices that perform mechanical work, follows on the original steam. In many cases, the work is distributed by exerting a torque, which is used to operate different machinery to generate electricity, pump water/to compress gas. In the case of propulsion systems, an air-breathing engine makes use of atmospheric air to oxidize the fuel, instead of carrying an oxidizer, as in rocket.

Engine in antiquity simple machines, such as club, oar (examples of the lever) are prehistoric. More complex engines used human power, animal power, water power, wind power and also steam power. [2]

1.2. Internal Combustion Engine:

An internal combustion engine is where the ignition of fuel occurs inside the combustion chamber. IC-engines are mostly used in transportation and also other several uses like portable situation where you need a non-electrical motor, application like an IC-engine driving an electric generator. The main advantage is the portability. It is more convenient to use this type of engine in vehicles instead of electricity. In case of hybrid vehicles, internal combustion engine is used to charge the battery. The major disadvantage is pollution.

These engine uses the fuel known as diesel, gasoline and petroleum as an energy source to generate power. Most internal combustion engines are designed for multi-fuel without major modifications except for fuel delivery components. Bio fuels like soy bean oil can also be used; and some can run on hydrogen.

All internal combustion engines have a method for achieving ignition in their cylinders for producing combustion. Engines use either an electrical method or a compression ignition system. These engines use either air cooling system or a liquid cooling system. [3-7]

1.3. Classifications of engines on basis of cylinder arrangement:

1. Line arrangement
2. V-engine
3. Radial engine
4. Opposed cylinder engine
5. Axial engine

1.4. Axial engines:

Axial engines (sometimes known as barrel engines) are a type of reciprocating engine with piston arranged around the output shaft with their axes parallel to the shaft. Barrel refers to the cylindrical shape. The pistons are arranged and spaced evenly around the central output shaft and aligned perpendicular or parallel to crankshaft axis based on the mechanism used. So far axial engines designed using cam/wobble plates/swash plates/Z-cranks for power transmission to the centre shaft. The Z-crank refer to the shape of the crankshaft.

The main advantage is, it's a very compact engine. It has a variation in the compression ratio of the engine while running. In a swash plate engine, the piston rods stay parallel with the shaft. An alternate design, the rand cam engine, replace the plate with a sine shaped cam. It can also be said as cam engine/swash plate/wobble plate engine. A wobble plate is similar to a swash plate, in that the piston press down on the plate in sequence, forcing a nutation around its centre. This motion can be simulated by placing a small disc (compact disc) on a ball bearing at centre that forces a downward motion around the circumference of the engine. The difference is that a wobble plate nutates and the swash plate rotates. [8]

1.5. Evolutions of axial engines:

1.5.1. Small-bone engine (year-1906):

The idea was developed by Harry Eales Small-bone and it is said to be the first axial IC engine. It was designed to run on town gas, not on gasoline or petrol. It is a water cooled 4-cylinder wobble plate engine or even known as static barrel engine. It is now not currently known; since if it was ever developed or it was successful. [9]

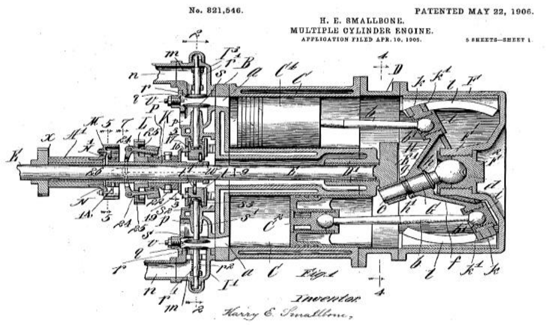


Figure 1-1 Small-bone engine 1906 [8]

1.5.2. Macomber engine (year-1911):

In year 1911 the company called Macomber in Los Angles introduced the market a rotary engine made of seven cylinders with variable compression ratio. Wobble plate is used for power transmission, but it was said to be rotary engine; since the whole engine rotated apart from the casings at each end. [10] [11]

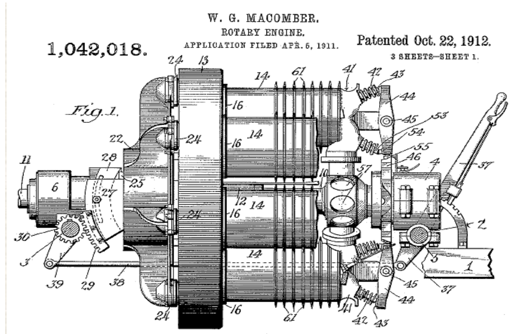


Figure 1-2 Macomber engine 1911 [8]

1.5.3. Trebert engine (year-1913):

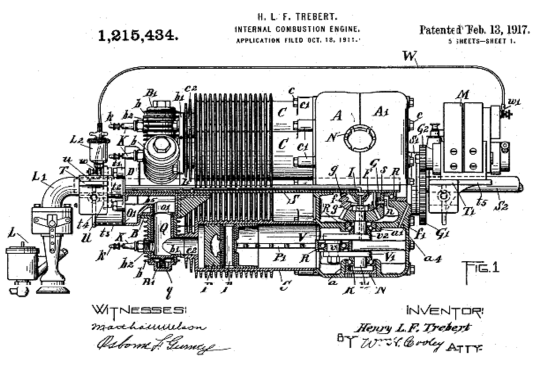


Figure 1-3 Terbert engine 1913 [8]

Henry L.F.Trebert developed a remarkable rotary axial aeroplane engine in year 1913.The engine is a rotary type with the cylinders and crank case arranged in cylindrical axis, with the central shaft remains in the centre. This is the only axial engine that does not use either a wobble plate or a swash plate; instead each cylinder is connected with a small crankshaft which is in turn connected to a bevel gear at the inner end of each shaft. With bevel gears at the inner end, each crankshaft is engaged with a large gear on the centre output shaft to receive power. So that propeller speed was cut into one half of the crankshaft speed. [12]

1.5.4. Statax engine (year-1913).

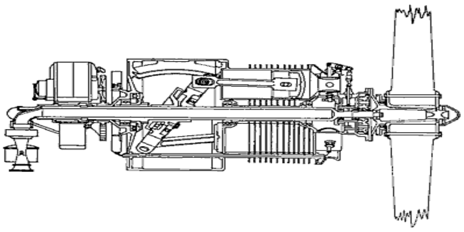


Figure 1-4 Statax engine 1913 [8]

Statax engine is said to be the first IC axial engine in Europe. A prototype was developed by Statax motors, Zurich, Switzerland in the year 1913. The model was designed by Dr. F.J.M. Hansen who worked with German air force in World War one. [8]

1.5.5. Salmson engine (year-1913):

The Salmson axial engine development was initiated in the year 1913; the engine was demonstrated at Paris air salon, in the Grand Palais. Even though it had fins, it was water cooled. The propeller shaft is at the right, and at the left end of the engine there with cooling fan for the radiator. Just near to the fan towards the right, there is a series of a levers which is a dynamo, it is driven from the periphery of the cooling fan. [8]

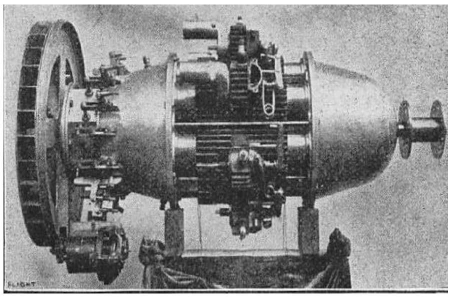


Figure 1-5 Salmson engine 1913. [8]

1.5.6. Almen A-4 engine (year-1921):

The A-4 Almen was the 4th experimental engine developed by John O Almen in Washington. The engine is said to be a water cooled barrel type with a wobble plate in the centre with nine sets of opposed piston arrangement. The engine was tested at McCook Field, Ohio. The project began in 1921 and by the mid-1920s Almen A-4 had passed its acceptable tests. Even though the test was successful the Almen engine never produced in mass production, because of a growing emphasis by the US Air force. It was rated at 425hp, and weighed only 749 pounds which was a significant achievement for the time. [13] [14]

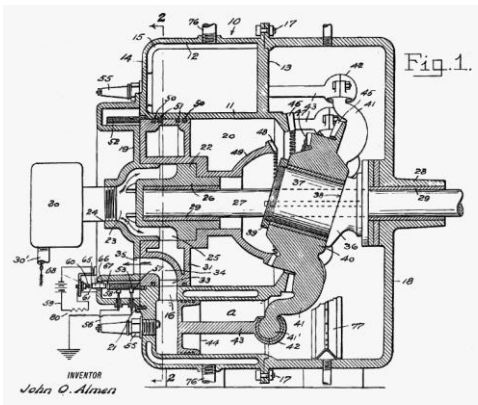


Figure 1-6 Almen A-4 engine 1921. [8]

1.5.7. ALI outboard engine (year-1922):

The ALI engine was made by Arvid Lind in Sweden. It is a two stroke water cooled petrol engine with four working cylinders above the wobble plate and four scavenging cylinder pumps below. The engine was designed to be used as an outboard engine for boating purpose with magneto ignition. [8]

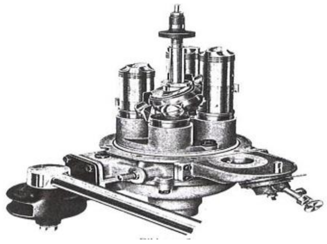


Figure 1-7 ALI outboard engine 1922. [8]

1.5.8. Rolls Royce engine (year-1923):

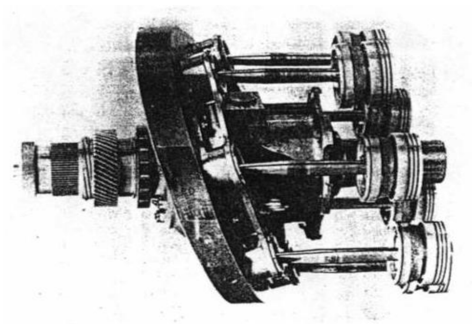


Figure 1-8 Rolls Royce engine 1923 [8]

In 1921 Rolls Royce was interested in developing an axial engine and in following year, Air Ministry contract allowed them to develop one experimental engine. Engine was designed with seven-cylinder arrangement with a cast iron cylinder block with detachable cylinder heads. The bore was 3inches, but the capacity was currently unknown. Push-rod operated overhead valves were used with a car type carburettor. Some development was done, by the year 1925 interest was waned and the project was passed onto Napier’s. [8]

1.5.9. Wishon engine (year 1923):

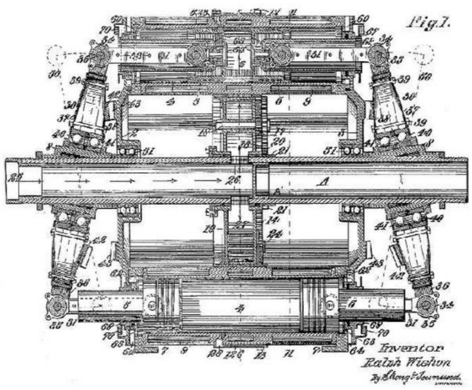


Figure 1-9 Wishon engine 1923[8]

The Wishon engine is a wobble plate design, developed by Ralph Wishon during 1923. The design was unusual, as it combines the axial engine principle by using opposed pistons with rotatory valves. There were nine cylinders bore with 18 pistons faced opposed to each other. [8]

1.5.10. Sparost engine (1930's):

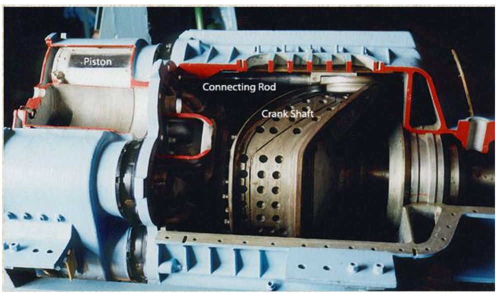


Figure 1-10 Sparost cam engine 1930's.

The Sparost axial engine was developed during 1930's. It has six cylinders driving a cam roller that fitted in between two sinusoidal ridges. The engine was rated at 600HP and was expected reach 1200HP by late 1930's. [8]

1.5.11. Hulsebos engine (year -1938):

Wichert Hulsebos designed a wobble plate engine and took 14-Europen patents from 1925's to 1938's on features of the engine. These five-cylinder wobble plate engine works with low compression combustion system. It is a hybrid between spark ignition and diesel operation. High density oil was sprayed into the cylinder at the rate of $103N/cm^2$ during the compression stroke. However, petrol had to be used for starting. [8]

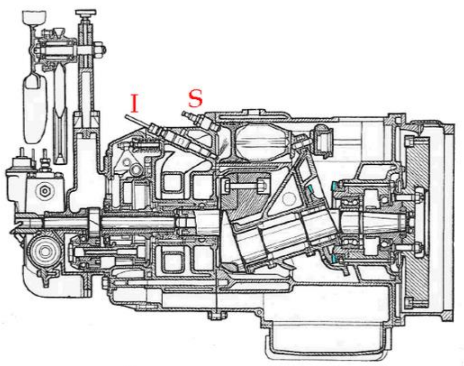


Figure 1-11. Hulsebos Hesselman 1938. [8]

1.5.12. Sterling engine (year -1938):

The sterling engines are two stroke 8-cylinder engine. Inclined discs or wobble plates are used for power transmission. The pumps like piston attached to 4-piston act as compressors by forcing air into the ports and combustion chamber. This swashplate engine was designed to run on diesel. [8]

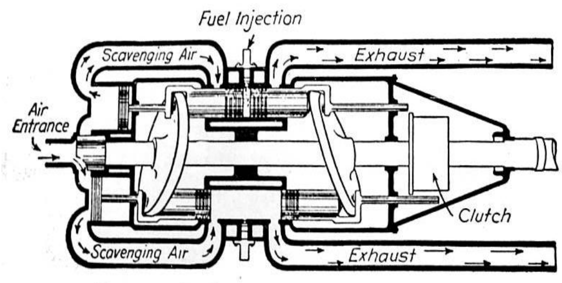


Figure 1-12 Sterling engine 1938. [8]

1.1.13. Dyna Cam engine (year-1941):

The Dyna-cam engine is a twelve-piston engine with 6-bores and twelve combustion chambers. All the cylinders are ignited in one single revolution of the drive shaft. The sinusoidal cam is used to operate the inlet and exhaust ports. In this engine the special cam design which replaces the timing gears and belt drives. [15] [16]

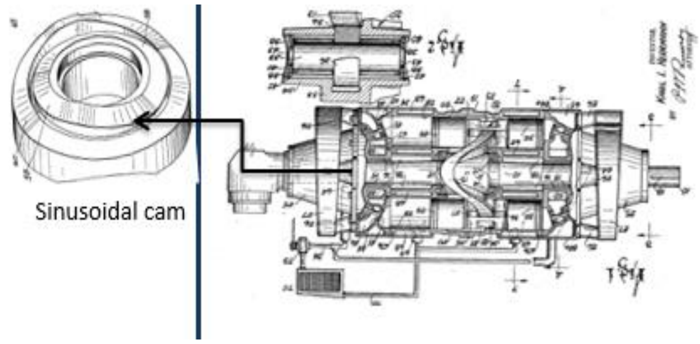


Figure 1-13 Dyna-cam engine 1941. [8]

1.1.14. Duke's Engine (year-2011):

Duke's company developed a valve less five-cylinder engine with three injectors. It is an internal combustion axial engine that claims to have zone first order vibration with significant reduction of size and weight. It has very high power density and ability to run on multiple fuels including bio fuels. The engine was tested in Australasia and Europe. [17]

Figure 1-14 Duke's Engine 2011. [8]

The first part of literature survey about the axial engines discussed above shows the evolutions of the engines from late 1900’s to the latest models in market and their types. Since this project is about axial engine the brief history of the axial engines and the design changes from late 1900’s to the latest Duke engine shows that these engines are mostly designed with swash plate or wobble plate or cam types. Except one model Trebert 2-stroke engine designed and manufactured at 1913. This thesis makes use of this design and is aimed to converts it to a 4-stroke engine with less number of parts.

1.6. Simulation and Fatigue analysis

1.6.1. Motion Study:

There are two types of motion study, kinematic and dynamic.

- Kinematic analysis studies show how the design moves due to force and motion drivers applied to the assembly. The key results of the interest are the assembly range of motion and determining parts displacement, velocity and acceleration.
- Dynamic motion analysis evaluates the forces generated by movement, and as well as the movement itself.

Motion analysis can be solved using two different solution paradigms, time based motion, event based motion:

- In a time based analysis, external action occurs at a pre-set time irrespective of the assembly motion.
- In event based motion analysis, the motion of the assembly triggers the external action.

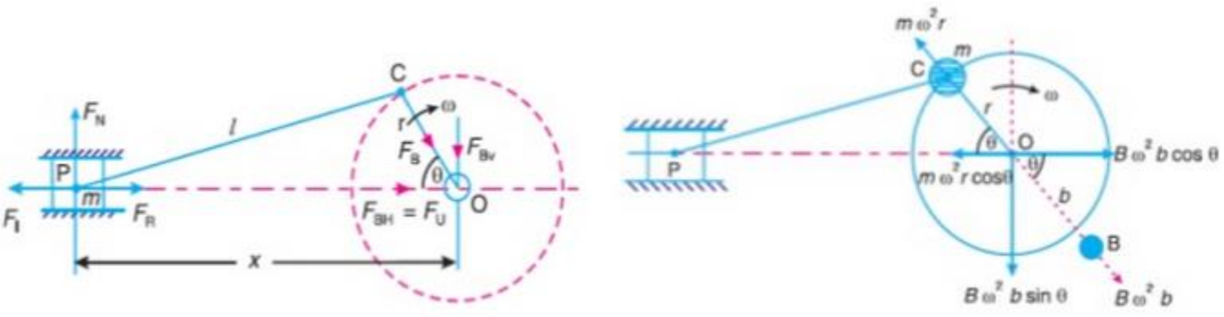
1.6.2. Engine kinematics:

The motion of a piston is 1-DOF system represented by a slider crank mechanism. Since the piston has lateral motion (sideways in bore), as the rotational Cartesian Z-axis lies in origin it moves in Y-axis alone. The maximum height it will reach is called top dead centre (TDC), which is sum of crankshaft, length of connecting rod and half of the stroke. The lowest point will reach is called bottom dead centre (BDC), which is the rod length minus the crank throw. The TDC corresponds to rotation of crank begin zero and BDC occurs at 180 degree of crank rotation. [18]

Velocity and acceleration of the piston is given by:

$$\alpha = r\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n'}\right) \tag{1.1}$$

Were θ is measure from inner dead centre position and the negative sign indicates the acceleration is towards the crank. Thus, if M is the mass of the reciprocating parts,



(a) (b)

Figure 1-155 Forces acting on connecting rod [19].

Inertia force

$$F_i = Mr\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n'}\right) \tag{1.2}$$

It may be noted that the inertia force of reciprocating parts opposes the force on the piston when it moves from (TDC-BDC). On other hand inertia forces also helps the force on the piston when it moves from (BDC-TDC).

The net force acting on the piston or piston pin (gudgeon pin/wristpin),

i.e.

$$P = pA \pm Mr\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n'}\right) \tag{1.3}$$

Were pA is the force of the fuel explosion on the piston towards the crankshaft.

The negative sign is used when piston moves from TDC to BDC and positive sign is used when the piston moves from BDC to TDC.

When mass of the reciprocating parts $WR = mR \times g$ (gravity) is to taken in consideration, then [19] [20]

$$P = pA \pm Mr\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n'}\right) \pm WR \tag{1.4}$$

1.6.3. Fatigue analysis:

It is observed that repeated loading and unloading weakness objects over time even when the induced stress is considerably less than allowable stress limit. This phenomenon is called fatigue; each cycle of stress fluctuation weakens the object to some extent; after a certain number of cycles, the object becomes so weak that it fails. Fatigue is the prime cause to failure of many objects, especially those made of metals. Examples of failure sue to fatigue include, rotating machinery, bolts, flight wings, consumer products, offshore platforms, ships, vehicle axles, bridges and bones.

Linear and non-linear structural studies do not predict fatigue failure, they calculate the response of a design subjected to a specified environmental restraints and loads. If the analysis assumptions are observed and the calculated stress are within the allowable limits, they conclude that the design in this environment regardless of how many times the load is applied.

1.6.4. Stages of failure due to fatigue:

Fatigue occurs in three stages.

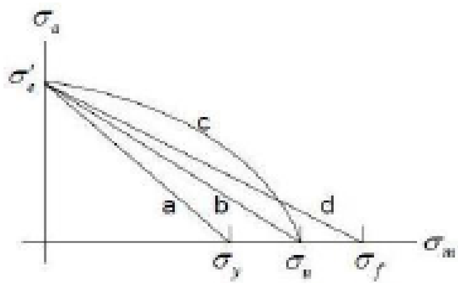
- **Stage1.** One or more cracks developed in the material. Crack can develop anywhere in the material but it usually occurs on the boundary face due to higher stress fluctuations. Cracks can develop due to many reasons. Imperfect microscopic structure of the materials and surface scratches caused by tooling or handling are some of them.
- **Stage 2.** The ability of the design to withstand the applied loads continues to deteriorate until failure occurs.
- **Stage 3.** Some of the cracks grow as a result of continuous loading.

Fatigue cracks starts on the surface of the material. Strengthening the surface of the model increases the life of the model under fatigue events.

1.6.5. Fatigue life evaluation:

Prediction of failure from fluctuating load is given by four major theories.

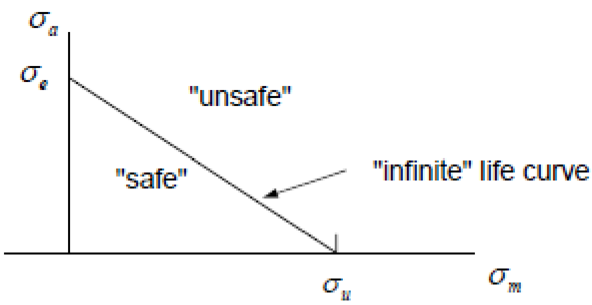
1. Soderberg (1930) $\frac{\sigma_a}{\sigma'_e} + \frac{\sigma_m}{\sigma_y} = 1$ (1.5)
2. Goodman (1898) $\frac{\sigma_a}{\sigma'_e} + \frac{\sigma_m}{\sigma_u} = 1$ (1.6)
3. Geber (1874) $\frac{\sigma_a}{\sigma'_e} + \left(\frac{\sigma_m}{\sigma_u}\right)^2 = 1$ (1.7)
4. Morrow (1960) $\frac{\sigma_a}{\sigma'_e} + \frac{\sigma_m}{\sigma_f} = 1$ (1.8)



σ_a- Alternating stress, σ_m- Mean stress, σ_y- Yield Stress, σ_u- Ultimate Stress, σ_f- Ture Fracture Stress, σ'_e- Effective alternating stress at failure for a life time of N_f cycles.

1.6.6. Use of empirical curves for an infinite life design:

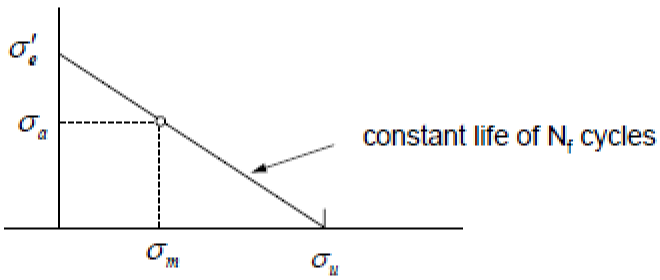
Use a particular curve such as the Goodman with $\sigma'_e = \sigma_e$ the endurance limit, N_f is infinity.



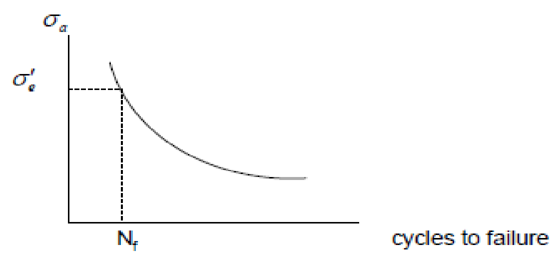
Any combination of mean and alternating stress that are to the left of the curve are deemed safe, those to the right are not.

1.6.7. Use of empirical curves and S-N data for a finite life design:

1. A given combination of mean and alternating stress is taken to lie on a constant life curve such as the Goodman line.



2. That curve is then used to solve for the effective purely alternating stress σ'_e that will cause failure at this same life time.
3. Using this effective alternating stress, determines the lifetime for this stress (and the corresponding original alternating and mean stress) from the S-N diagram for the given material. [21-27]

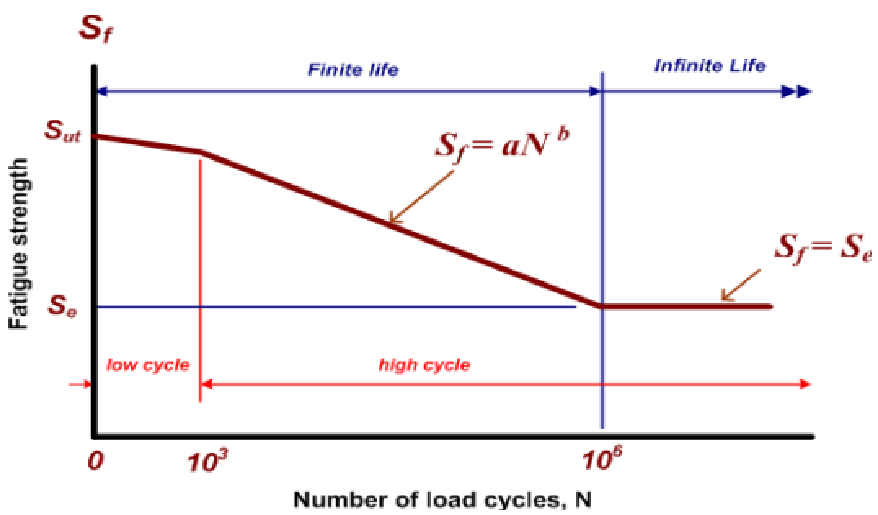


1.2.8. Design for infinite life:

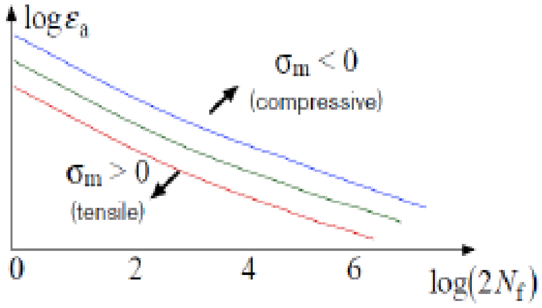
When the component to be designed for infinite life, the endurance limit becomes the criteria of failure. The amplitude stress induced in components should be maintained lower than the endurance limit in order to withstand the infinite number of cycles. Such a component is designed with the help of following equation.

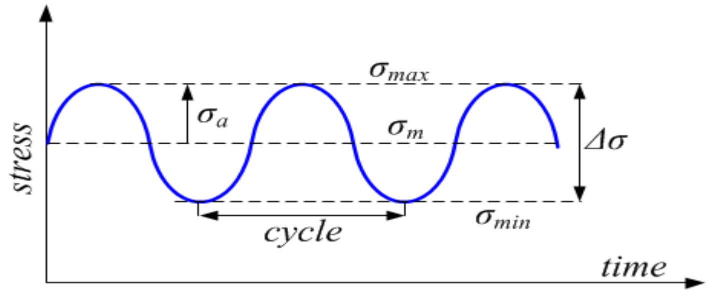
$$\sigma_a = S_f/factor\ of safty\ (\mathrm{f.o.s}) \tag{1.9}$$

The S-N curve shown below. The curve is valid for steels.



A connecting rod is always subjected to a full reversed loading, which means $\sigma_m = 0$.





Cyclic loading

$$\sigma_m = \sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{1.10}$$

$$\sigma_m = \sigma_{alternating} = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{1.11}$$

$$R = \frac{\sigma_{min}}{\sigma_{max}} = -1[fully\ reversed\ loading] \tag{1.12}$$

High fatigue life is achieved (N > 1000cycles) and infinity life if (N >10^6).

Equation if a line $y = ax + c$:

$$\log S_f = a(\log N_f) + c$$

$$\log S'_l = a(\log 10^3) + c = 3a + c$$

$$\log S'_e = a(\log 10^6) + c = 6a + c$$

$$a = -\frac{1}{3}\log\frac{S'_l}{S'_e} \tag{1.13}$$

$$c = \log\frac{(S'_l)^2}{S'_e} \tag{1.14}$$

$$\therefore S_f = 10^c(N_f)^a or\ N_f = (S_f 10^{-c})^{1/a} \tag{1.15}$$

$$S_f - Fatigue\ Strength$$

$$S'_u = Endurance\ limit = 0.75S_u(Axial\ loading) \tag{1.16}$$

$$S'_u = 0.5S_u;\ S_u \le 200ksi\ \approx 1400MPa \tag{1.17}$$

N_f - gives the number of cycles to failure, crack development. [28]

The second part of the literature survey is about designing a component for infinity life, "here we are analysing the infinity life of connecting rod and redesigning for infinity life". The above discussion shows why fatigue is important and the details discussion on the theories about fatigue failure and also how prediction the fatigue life cycles which is carried out in this project.

2. AN ILLUSTRATION ON THE DESIGN OF AXIAL IC-ENGINE

2.1. 3D Model of new axial (*barrel*) engine:

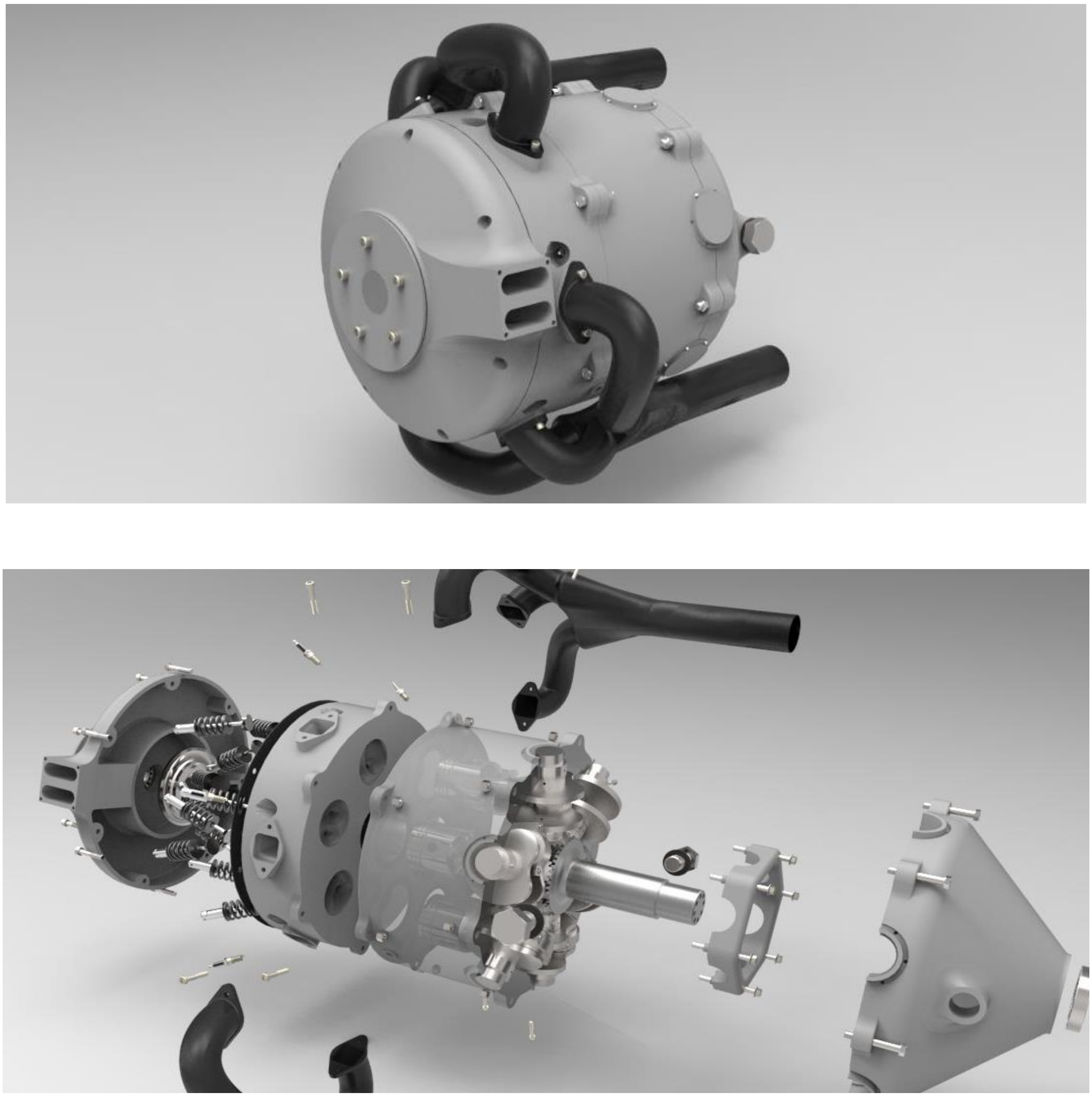


Figure 2-1 3D Model of new axial (barrel) engine (designed using SolidWorks 2015)

The design is a combination of two existing 2-stroke [Trebert axial engine] and 4-stroke [Dyna-cam engine]. In this design the body of the engine is divided into 4 segments [engine head cover, engine head, bore block and crank case]. The engine has a single cam component as like one Dyna-cam engine. Cam lift is 360/6(no of cylinders) degree angle division with a lift of 1.32in. Intake valve and exhaust valve is separated at an angle of 180^{o} angle. The power transmission is like Trebert axial engine, were there are six individual crank shafts which is to perform four-stroke operation. The crank shaft is in-turn connected to helical bevel gears; helical bevel gears are used to avoid slip and friction between the gear teeth; also for better transmission, the gear design ratio is like 2(driver):1(driven) ratio. Roller followers are used in this design to reduce the friction between the cam and the follower. This design was concentrated to reduce size and parts like (timing gears, rocker arms, push rods, timing gears and any balancing shafts needed) but to produce high torque.

2.2 Engine parameters:

As explained in the introduction, the primary goal is to find put the inertia forces developed for this design, and the secondary goad of the proposed tool is the fatigue on the connecting rod used material AISI 4340 STEEL and C70 for a six cylinder (*barrel*) axial engine. The engine base parameters are summarised in table 2.1, and the material properties are summarised in table 2.2.

Table 2.1 Engine parameters

Bore diameter	100 mm
Stroke length	80 mm
Connecting rod length	160 mm
Assembled piston mass with piston pin	312.55 g
Total weight of the engine	82.33 kg

Table 2.2. Material properties AISI 4340 STEEL and AISI 1070 (C70)

Material	AISI 4340	AISI 1070
Elastic Modulus	205000 MPa	211500 MPa
Poisson’s Ratio	0.32	0.3
Shear Modulus	80000 MPa	81350 MPa
Mass Density	7850 kg/m^3	7850 kg/m^3
Tensile Strength	1110 MPa	965.8 MPa
Compressive Strength	710 MPa	573.11
Yield Strength	710 MPa	573.11
Specific Heat	475 J/(kg-K)	440 J/(kg-K)

2.3. Calculation of piston groups

2.3.1 Piston:

The piston is which reciprocates within a cylinder. It is ether moved by the fluid or it moves the fluid which enters the cylinder. The function of the piston in an internal combustion engine is to receive the impulse from the exploded gas to transmit energy to the crankshaft through connecting rod. The piston must also diffuse a large amount of heat from combustion chamber to cylinder walls. In the piston design, it is used parameters of existing engines as elements are pre-calculated strength, without taking into account different methods of loads.

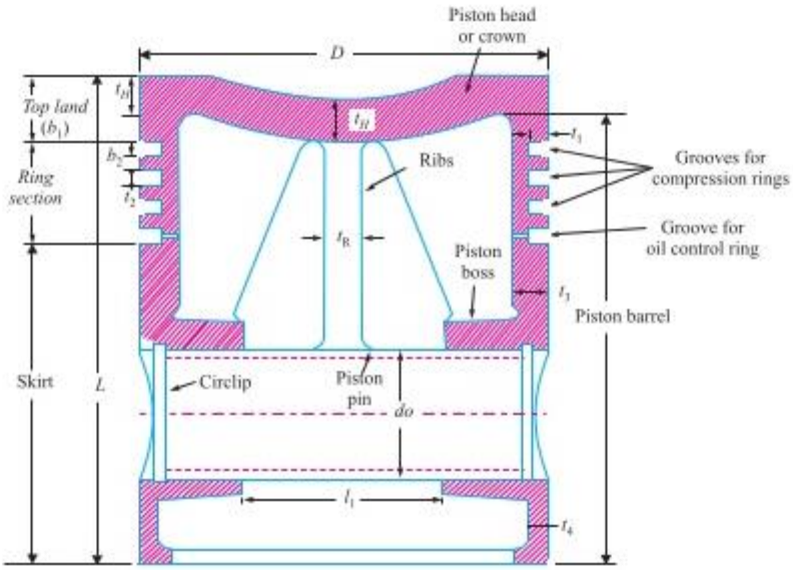


Figure 2-2 Piston for I.C. engines (Trunk type). [29]

Thickness of the piston crown.

$$t_H = (0.05 - 0.12)D = 8.64\ mm \tag{2.1}$$

Height of the piston.

$$L = (0.8 - 1.3)D = 92.6\ mm \tag{2.2}$$

Distance from the piston head to the axis of the piston pin

$$h_1 = (0.45 - 0.47)D = 46.9\ mm\ \approx 47\ mm. \tag{2.3}$$

Diameter of the piston boss.

$$d = (0.3 - 0.5)D = 46.91\ mm \approx 47\ mm. \tag{2.4}$$

Size

$$l_1 = 40.74\ mm \tag{2.5}$$

Wall thickness of leading part

$$t_3 = (1.5 - 4.5) = 3.7\, mm. \tag{2.6}$$

Thickness of sealing part

$$t_4 = (0.03 - 0.8)D = 3.7\, mm. \tag{2.7}$$

Top land

$$b_1 = (0.06 - 0.12)D = 8.64\, mm. \tag{2.8}$$

Groove thickness for the compression and oil ring

$$t_1 = (0.03 - 0.05)D = 4.3\, mm. \tag{2.9}$$

Number of oil holes

$$n_m = (6 - 12) = 10$$

Diameter holes for oil

$$d_M = (0.03 - 0.05)d_0 = 26\, mm. \tag{2.10}$$

Height of the piston

$$L = (0.6 - 0.8)D = 68\, mm. \tag{2.11}$$

2.3.2. Piston pin:

Piston pin or Gudgeon pin are subjected of varying shape and size loading and cause bending, shearing and surface tension pressure, so we select steel as a material that they have been made. [1] Structural dimensions of the piston can be determined using existing engines:

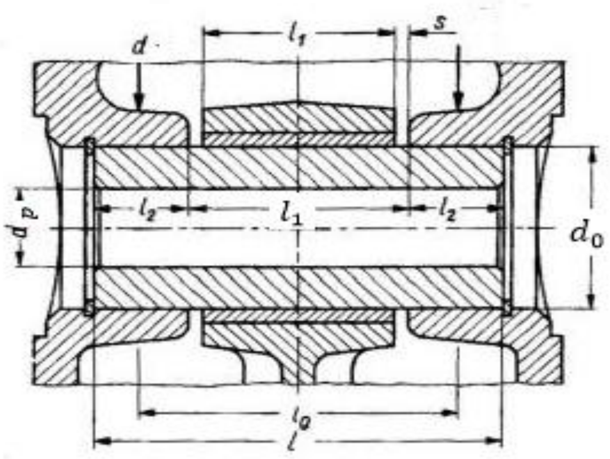


Figure 2-3 Piston pin [1].

Outer diameter of the bolt

$$d_0 = (0.22 - 0.28)D = 26\, mm \tag{2.12}$$

Inner diameter of the bolt

$$d_p = (0.65 - 0.75)d_0 = 18\ mm \tag{2.13}$$

Length of the bolt

$$l = (0.88 - 0.93)D = 90\ mm \tag{2.14}$$

Length of the upper head of the rod

$$l_f = (0.28 - 0.38)D = 39\ mm \tag{2.15}$$

2.3.3. Piston rings:

Piston rings provide tight pressure on the cylinders. They work at high temperature, so they must have high elasticity, strength, durability and law coefficient of the friction with the cylinder wall. Materials for their manufacturing are iron with chromium, copper, nickel, titanium and others materials. [1]

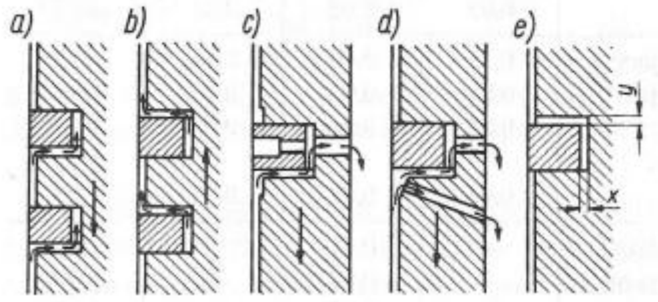


Figure 2-4 Examples and design of piston rings [30].

a), b) Pumping oil through the rings, c), d) Scraping and oil flow towards the piston,
e) Clearance between the piston ring and the piston itself.

The structural dimensions of the piston rings can be adopted according to existing engines:

Radial thickness of the grummet

$$t_y = (0.039 - 0.045)D = 3.5\ mm \tag{2.16}$$

Radial thickness of the oil collecting ring

$$t_M = (0.038 - 0.043)D = 3.5\ mm \tag{2.17}$$

Height of the ring

$$a = (2 - 4) = 3\ mm \tag{2.18}$$

Radial clearance of the oil collecting ring

$$\Delta t_M = (0.50 - 0.95) = 0.8\, mm \tag{2.19}$$

Radial clearance of the grummet

$$\Delta t_y = (0.50 - 0.95) = 0.8\, mm \tag{2.20}$$

Axial clearance of the ring

$$\Delta a = (0.04 - 0.08) = 0.06mm \tag{2.21}$$

Clearance in the slot of the ring at Free State

$$A_0 = 12\, mm$$

2.4. Calculation of the connecting rod

A connecting rod is a machine part which is always subjected to alternating direct compressive and tensile force. Since the compressive force are much higher than the tensile force, therefore the cross-section of the connecting rod is designed as a strut, and the Rankine's derivation is used. A connecting rod subjected to an axial P may buckle with x-axis as neutral axis to the plane of motion of connecting rod, or y-axis is a neutral axis. The connecting rod is considered like both ends hinged for buckling about x-axis or both ends fixed for buckling about y-axis. A connecting rod should be equally in buckling about either axis. [31] [32] [33]

According to Rankine formulae

About x-axis P_{cr}

$$= [\sigma_c \times A] + a[LK_{xx}]^2 = [\sigma_c \times A]1 + a[l^2 K_{xx}]^2\ [\therefore for\ both\ ends\ hinged\ L = l] \tag{2.22}$$

About y-axis P_{cr}

$$=[\sigma_c \times A]1 + a[LK_{xx}]^2 = [\sigma_c \times A]1 + a\left[l^2 K_{yy}\right]^2 [\therefore for\ both\ ends\ fixed\ L = l/2] \tag{2.23}$$

In order to have a connecting rod equally strong in buckling about both axis, the buckling load must be equal. i.e.

$$= [\sigma_c \times A]1 + a[lK_{xx}]^2 = [\sigma_c \times A]1 + a\left[l^2 K_{yy}\right]^2 [or][lK_{xx}]^2 = \left[l^2 K_{yy}\right]^2$$

$${K_{xx}}^2 = 4{K_{yy}}^2 [or] I_{xx} = 4I_{yy} [\therefore I = A \times K^2] \tag{2.24}$$

This shows 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 $I_{xx} < 4I_{yy}$, then buckling will occur about x-axis. In actual practice I_{xx} is kept slightly less than $4I_{yy}$. It is usually taken between 3-3.5 and the connecting rod is designed for buckling about x-axis. The design will a be always satisfactory for buckling about y-axis.

The most suitable for the area of the cross section.

$$.= [4t \times t] + 3t \times t = 11t^2 \quad (2.25)$$

Moment of inertia about x-axis

$$I_{xx} = 112[4t(5t)^3 - 3t(3t)^3] = 41912[t^4]$$

Moment of inertia about y-axis

$$I_{yy} = 2 \times 112 \times t \times (4t)^3 + 112(3t)t^3 = 13112(t^4)$$

$$\frac{I_{xx}}{I_{yy}} = [419/2] \times [12/131] = 3.2 \quad (2.26)$$

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

2.4.1 Pressure calculation for 3800cc 6-cylinder axial (barrel) engine:

For one-cylinder engine specification 630cc

Engine type 4-stroke

Bore (B) × Stroke (S) = $100\ mm \times 80\ mm$

The formula for displacement is

$$D = \frac{\pi}{4}B^2SN \text{ (N = no of cylinders, N=1)} \quad (2.27)$$

Displacement $D = 628.32cc \approx 630cc$

Pressure Produced inside the cylinder of a petrol engine maximum is 50 bar. [30]

2.4.2. Design calculation for connecting rod:

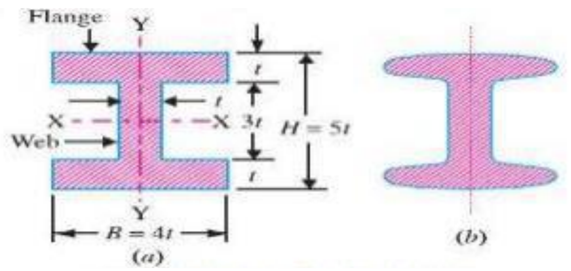


Figure 2-5 I-section [31]

Thickness of flange and web of the connecting rod cross section = t

Width of section B = 4t (2.28)

The standard dimension of I-section.

Height of section H = 5t (2.29)

Area of section $A = 11t^2$

$.K_{xx} = {}^{I_{xx}}\!/_{I_{yy}} = 3.2$ [$Since\ K^2 = {}^{I}\!/_{A}$] (2.26)

$.K = \sqrt{\frac{I}{A}} = 1.78t$

Length of connecting rod

$$(\boldsymbol{L_M}) = 160 \text{ mm (Twice the stroke length)} \tag{2.27}$$

Crank radius

$$(\boldsymbol{r}) = \frac{Stroke\ length}{2} = {}^{80}\!/_{2} = 40\ mm \tag{2.28}$$

$$n' = \frac{length\ of\ the\ connrcting\ rod}{crank\ radius} = {}^{160}\!/_{40} = 4 \tag{2.29}$$

Angular speed ω [Assuming N = 3000 rpm]

$$\omega = {}^{2\pi N}\!/_{60} = 314\ {}^{rad}\!/_{sec}\ [or]\ 12.56\ m/s \tag{2.30}$$

Inertia force of reciprocating part (F_i)

$$F_i = Mr\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n'}\right) \tag{1.2}$$

M = Mass of the piston + Mass of the Piston rings + Mass of the Piston pin (gudgeon pin) + 1/3rd Mass of the connecting rod

M = 388.09 g

.r – radius of the crank

$F_i = 3061.11$ N

$pA = \frac{\pi D^2}{4} \times 5$ (Maximum force on the piston due to the explosion 50 Bar) (2.31)

$pA = 39269.91$ N

$Reaction\ force\ P = Force\ due\ to\ gas\ pressure\ \pm Inertia\ force$

$$P = pA \pm Mr\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n}\right) \tag{1.3}$$

$P = 42331\ N(expansion\ stroke)$

$\sigma_c = 710\ MPa$ [Compressive yield stress]

$a = {}^{\sigma_c}\!/_{\pi^2 E}$; $a = \frac{1}{7500}$ (According to Rankine equation) (2.32)

Critical force F_c = Maximum gas force $pA \times fos$ [taking factor of safety =6]

$$F_c = \left(\frac{\sigma_c \times A}{1 + a\left(\frac{L}{K_{xx}}\right)^2}\right) \tag{2.33}$$

By substituting all the values;

t = 5.6 mm – 6 mm

Width of section B = 4t

B = 20 mm – 22.4 mm

Height of section H = 5t

H = 26.29 mm – 32 mm

Connecting rod inner diameter $d = d_0 = 26\ mm$ (2.27)

Connecting rod length

$$L_M = 160\ mm\ (twice\ the\ stroke\ length) \tag{2.12}$$

Minimal radial thickness of upper head

$$h_r = (0.5 - 0.55)l_f = 21.91\ mm \tag{2.34}$$

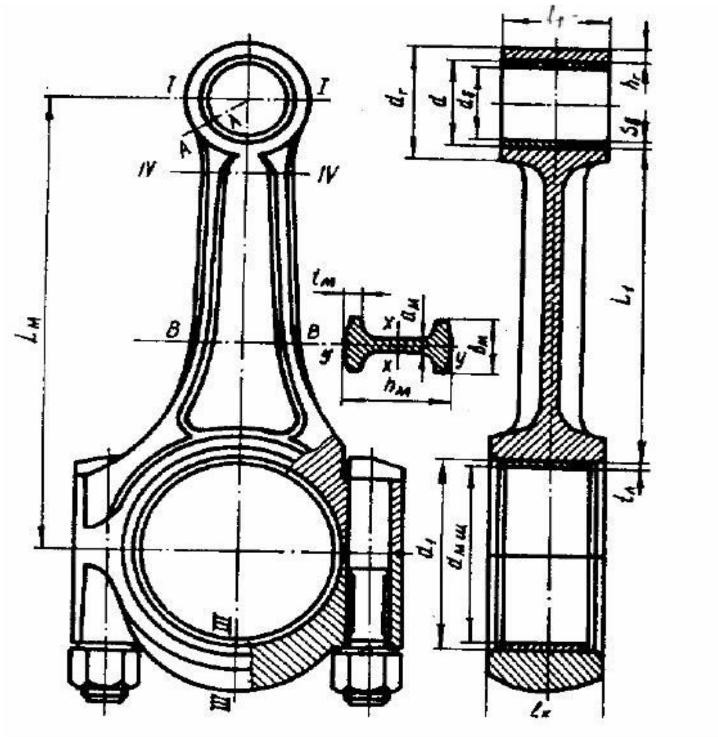


Figure 2-6 Connecting rod [1]

Diameter of big end inner neck

$$d_{M\varphi} = (0.03 - 0.05)D = 60\ mm\ [56\ mm \sim 75\ mm] \tag{2.35}$$

Thickness of the bearing shell

$$T_\pi = (0.03 - 0.05)d_{M\varphi} = 3\ mm \tag{2.36}$$

Distance between connecting rod bolts

$$C = (1.30 - 1.75)d_{M\varphi} = 84\ mm \tag{2.37}$$

Minimal radial thickness of rod at big/lower head

$$l_x = (0.45 - 0.95)d_{M\varphi} = 45\ mm \tag{2.38}$$

Let the diameter of big end of bearing = d_c; and length of big end = l_c

$$d_c/l_c = (1.25\ mm\ to\ 1.5\ mm) \tag{2.39}$$

Thickness of the bush

$$[t_b] = 2\ mm\ to\ 6\ mm$$

Marginal thickness

$$[t_m] = 5\ mm\ to\ 15\ mm$$

Nominal diameter of bolt

$$[d_b] = 1.2 \times root\ diameter\ of\ the\ bolt = 1.2 \times 5 = 6\ mm] \tag{2.40}$$

2.5. Calculation of crank shaft

Crankshaft of the engine is subjected to the action of gas force, inertia force and momentum, which are periodical functional angle of the knee. These force and moments induced torsion stress, tension and compression. Furthermore, periodically change in moments cause twisting and bending vibrations, which create additional tensions. [1]

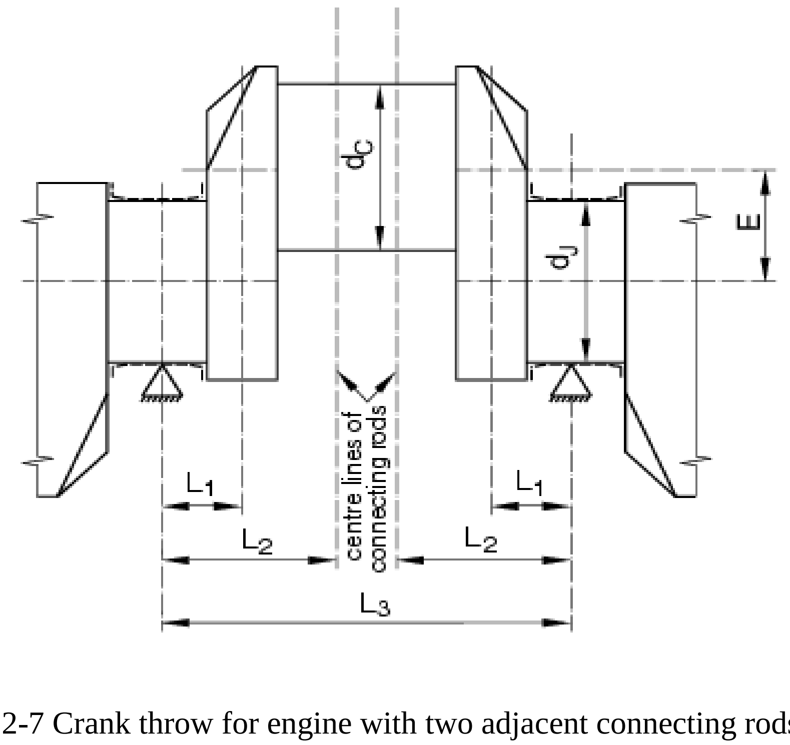


Figure 2-7 Crank throw for engine with two adjacent connecting rods [34]

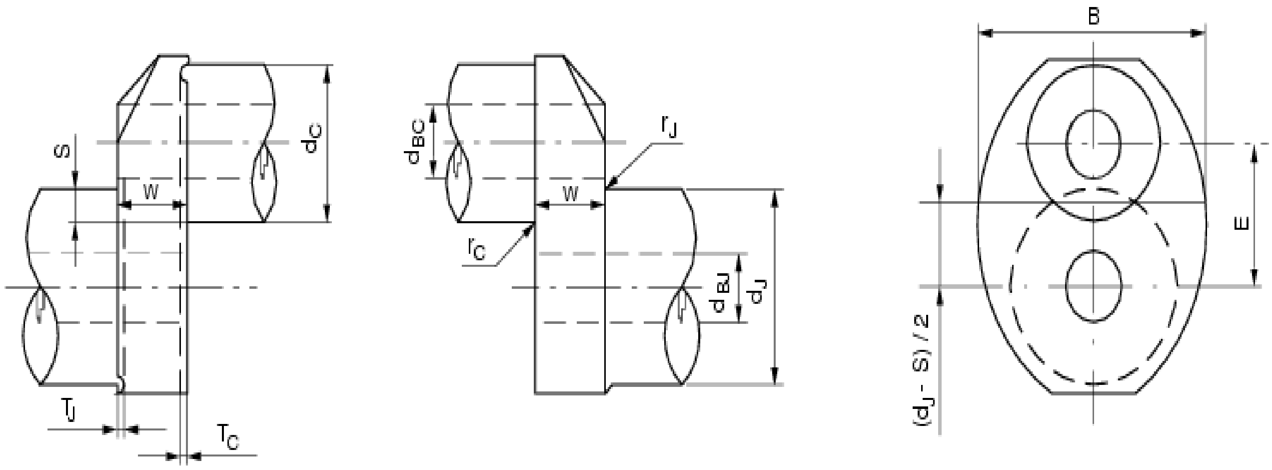


Figure 2-8 Crank throw of semi build crank [34]

In designing crank shaft, we can use parameters of already existing engines:

Distance between gear and main journal

$$l_c = (1.10 - 1.25) = 123.5\, mm\ \pm 0.05\, mm \tag{2.41}$$

Diameter of the main journal

$$d_{ow} = (0.5 - 0.8)D = 70\, mm \tag{2.42}$$

Length of the main journal

$$l_{ow} = (0.5 - 0.6)d_{ow} = 54\, mm \tag{2.43}$$

Diameter of the rod journal

$$d_j = (0.5 - 0.7)D = 70\, mm \tag{2.44}$$

Length of the rod journal

$$l_{mw} = (0.45 - 0.65)d_j = 48\, mm \tag{2.45}$$

Thickness of the crank

$$W = (0.15 - 0.35)d_j = 12.5\, mm \tag{2.46}$$

Width of the crank

$$B = (1.7 - 2.9)d_j = 167.3\, mm \tag{2.47}$$

Thickness of the rounded

$$r_j = (0.06 - 0.1)d_{mw} = 5\, mm \tag{2.48}$$

3. FATIGUE MODEL LIFE PREDICTION

3.1. Kinematic analysis:

The Kinematic Study is conducted using SolidWorks Motion 2015, the piston components in motion study are introduced to gas pressure for different with crank angles 120° , and the force is active up to 180° crank rotation to find the Maximum reaction forces and torque developed with all the components. To check the difference between the mathematical calculation and motion simulation. The higher force obtained from both simulation and mathematical calculation is used to conduct a fatigue study.

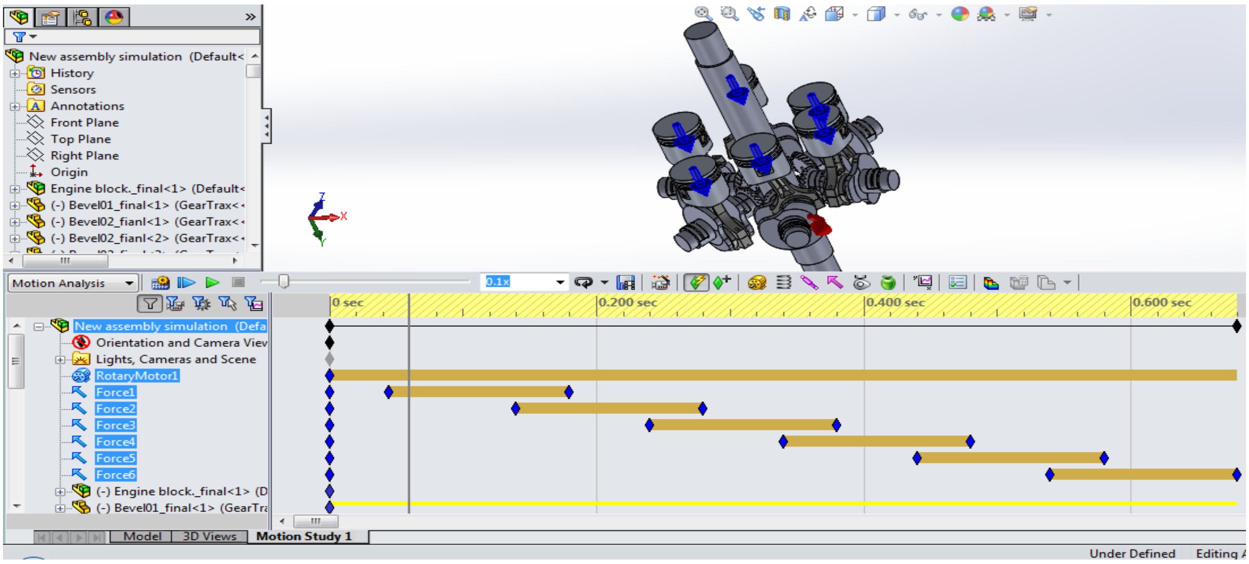


Figure 3-1 Kinematic study (using SolidWorks 2015)

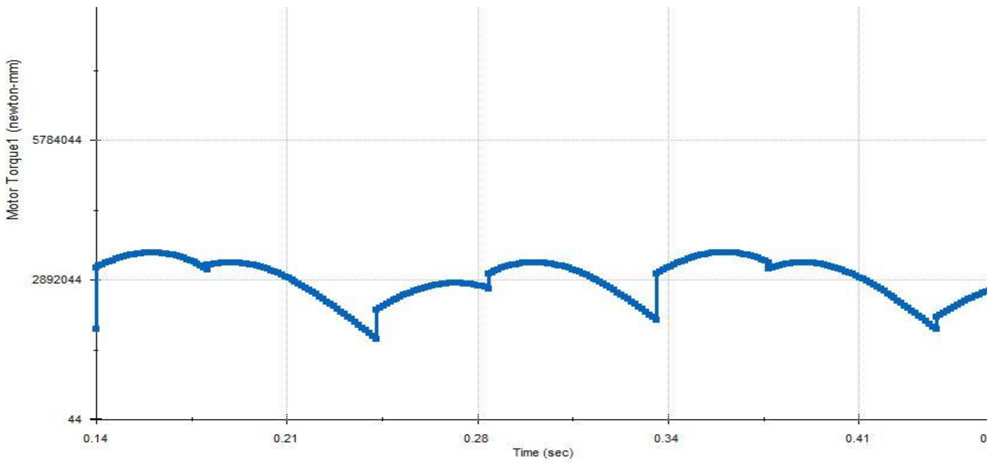


Figure 3-2 Motor torque (using SolidWorks Motion 2015)

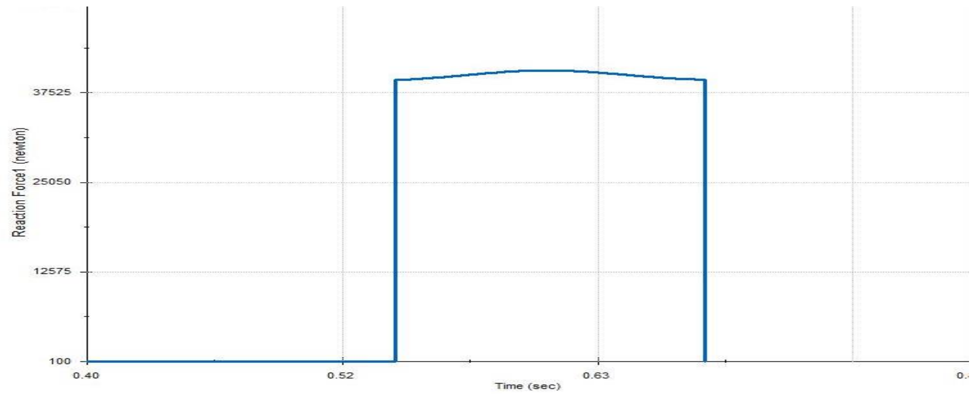


Figure 3-3 Reaction force (using SolidWorks Motion 2015).

Table 3.1. Reaction force plots from SolidWorks Motion

Time (sec)	Reaction Force (newton)	Time (sec)	Reaction Force (newton)	Time (sec)	Reaction Force (newton)	Time (sec)	Reaction Force (newton)
0.54	43.38708	0.575	40010.41	0.61	40566.25	0.64625	39896.81
0.54	39256.31	0.57625	40044.74	0.61125	40562.69	0.6475	39863.67
0.54125	39267.29	0.5775	40078.72	0.6125	40557.28	0.64875	39830.76
0.5425	39279.89	0.57875	40112.23	0.61375	40550.04	0.65	39798.2
0.54375	39294.09	0.58	40145.19	0.615	40541	0.65125	39766.06
0.545	39309.87	0.58	40145.19	0.61625	40530.18	0.6525	39734.43
0.54625	39327.17	0.58125	40177.5	0.6175	40517.62	0.65375	39703.39
0.5475	39345.96	0.5825	40209.07	0.61875	40503.36	0.655	39673.03
0.54875	39366.2	0.58375	40239.82	0.62	40487.46	0.65625	39643.42
0.55	39387.83	0.585	40269.64	0.62125	40469.96	0.6575	39614.65
0.55125	39410.81	0.58625	40298.47	0.6225	40450.91	0.65875	39586.78
0.5525	39435.07	0.5875	40326.2	0.62375	40430.39	0.66	39559.89
0.55375	39460.56	0.58875	40352.77	0.625	40408.44	0.66125	39534.04
0.555	39487.22	0.59	40378.09	0.62625	40385.16	0.6625	39509.31
0.55625	39514.98	0.59125	40402.09	0.6275	40360.59	0.66375	39485.75
0.5575	39543.77	0.5925	40424.7	0.62875	40334.84	0.665	39463.42
0.55875	39573.53	0.59375	40445.85	0.63	40307.96	0.66625	39442.39
0.56	39604.16	0.595	40465.48	0.63125	40280.05	0.6675	39422.69
0.56125	39635.6	0.59625	40483.52	0.6325	40251.18	0.66875	39404.38
0.5625	39667.77	0.5975	40499.94	0.63375	40221.45	0.67	39387.5
0.56375	39700.58	0.59875	40514.67	0.635	40190.95	0.67125	39372.1
0.565	39733.97	0.6	40527.68	0.63625	40159.77	0.6725	39358.21
0.56625	39767.86	0.60125	40538.93	0.6375	40127.99	0.67375	39345.87
0.5675	39802.07	0.6025	40548.38	0.63875	40095.71	0.675	39335.1
0.56875	39836.59	0.60375	40556.01	0.64	40063.03	0.67625	39325.93
0.57	39871.31	0.605	40561.79	0.64125	40030.03	0.6775	39318.38
0.57125	39906.15	0.60625	40565.71	0.6425	39996.82	0.67875	39312.48
0.5725	39941.01	0.6075	40567.76	0.64375	39963.48	0.68	39308.23
0.57375	39975.8	**0.60875**	**40567.94**	0.645	39930.11	0.68	34.82374

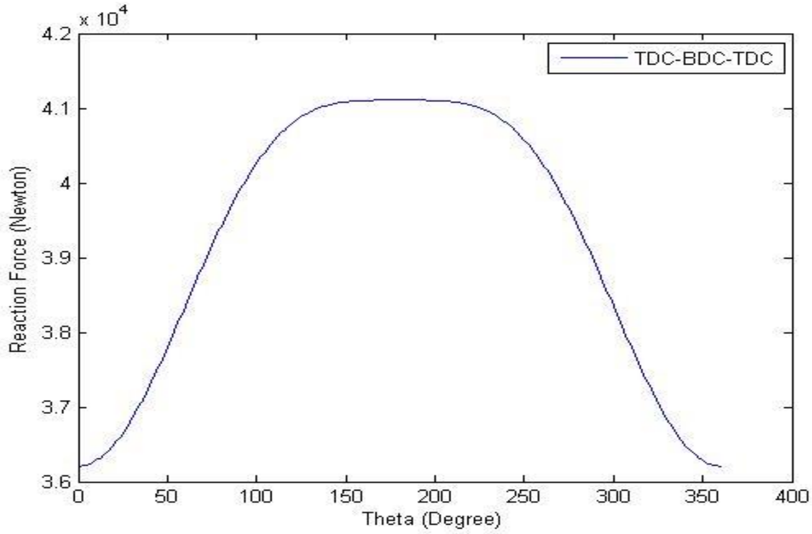


Figure 3-4 Reaction force calculated using Mat lab

The maximum torque developed in the engine is about 31031209.03N-mm (648.10 lb-ft.) at 3000rpm from (figure 3-2). Which means that the engine has the ability to produce power about 371hp@3000rpm. Power of an engine is given by the equation $Power(hp) = \frac{(torque(lb-ft)\times rpm)}{5252}$. And the maximum reaction force obtained through the motion study is 40,567.94 N (from table 3.1). Were as in the analytical calculation (using Mat lab) maximum reaction force obtained is 42,331 N. Using the maximum reaction forces the connecting rod is designed to have an infinity life whatever might be the situation in real-time use. The reaction forces calculated are without any mass properties.

3.2. Finite element analysis (Static analysis):

The objective of FEA is to investigate stresses, displacement and hotspots experienced by the connecting rod. The stresses obtained can be used to predict the fatigue life and determine the expected failure regions. Linear elastic analysis was used since the connecting rod is designed for longer life were stress are mainly elastic. The stress distribution helps us to find out the loading ratio for the reaction force acting on the connecting rod.

In the static study, the connecting rod big end is fixed and small end is distributed with reaction forces (bearing pressure) in axial considered for all analyses, since it is the primary service loading. Totally two study is conducted using two loading ratio R = -1 (fully reversed) and R = 1.25. [35]

Since,

$$R = \frac{\sigma_{min}}{\sigma_{max}} \tag{1.12}$$

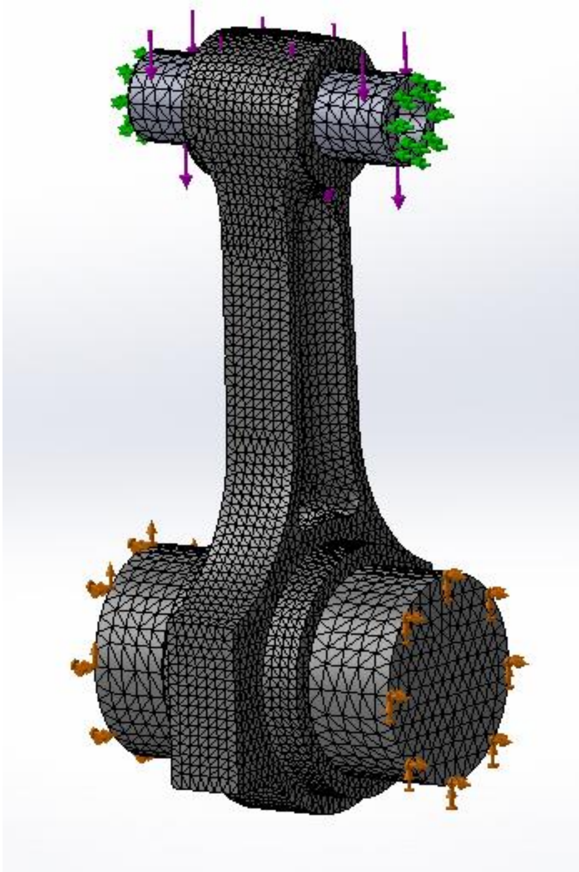


Figure 3-5 Connecting rod mesh and loadings

The connecting rod parameters calculated above is used to design the connecting rod for the analysis study. The model was designed without any bolts and crank and pin bearings, as these details are not expected to have any significant influence on obtained results at critical regions and their removal allowed simplification of the model.

The connecting rod was assembling with a crank and piston pin as shown in Figure 3.5. In the static study the connecting rod pin holes' surfaces and pins outer surfaces were provided with contact set with no penetration condition; and a global contact bounded for the entire assembly. The two surface of the crank pin is fixed and the piston pin is resisted movement in x and z direction; and load was applied on the piston pin.

A parabolic tetrahedron element type with four Jacobin points was used for the solid mesh, as this was the default option by SolidWorks for any 3-D analysis. Figure 3.5 shows the mesh type used for analysis. The element size of 3 mm was used with ratio 1.5 for the piston pin and the crank pin. A high quality mesh with total nodes 107455 and element type 69483.

R-ratio is considered for only the longitudinal direction for all the regions, since. R-ratio is considered for the critical regions. Due to the bearing pressure on the region I from figure 3.6 and 3.7 under axial load conditions, the mean stress at critical region is not large. Since the value of σ_{min} at the critical transition region are small as compared to yield strength of AISI 4340 and C70 steel. At the region near the pin end the stresses are close to yield strength; these regions are at or near the loading and constraints. Therefore, stresses at these locations may not be accurate. Note that, the nodes with largest alternating stress do not necessarily have the largest σ_{max} (i.e. the difference between static and cyclic stress analysis).

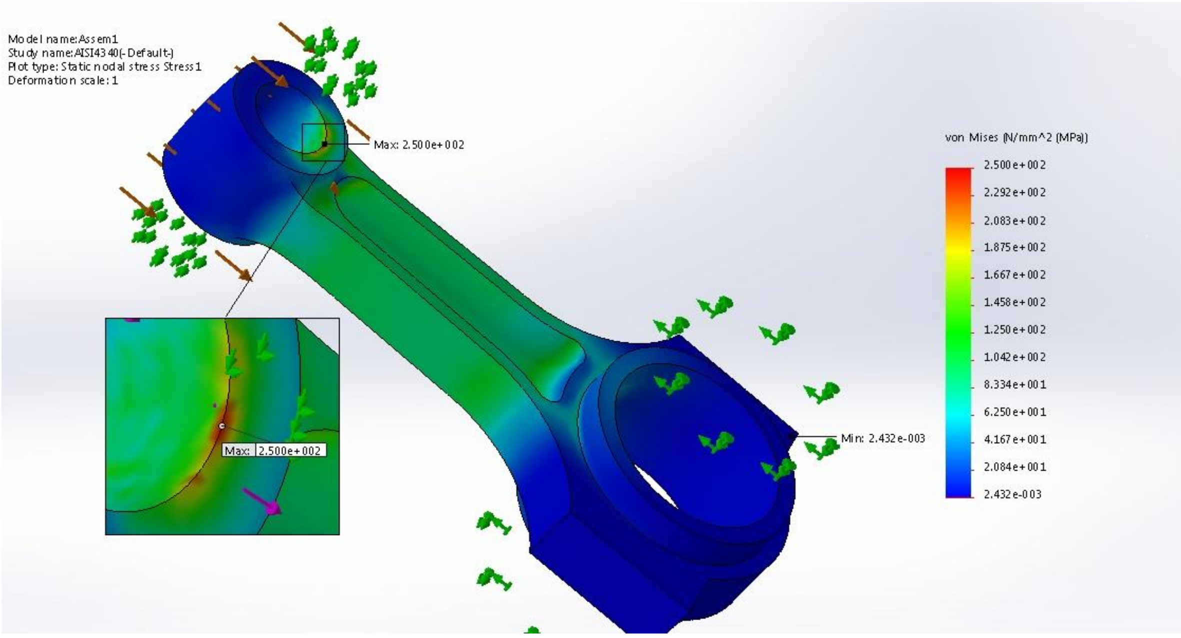


Figure 3-6 Von-Mises stress for 4340, for bearing load 42,331 N.

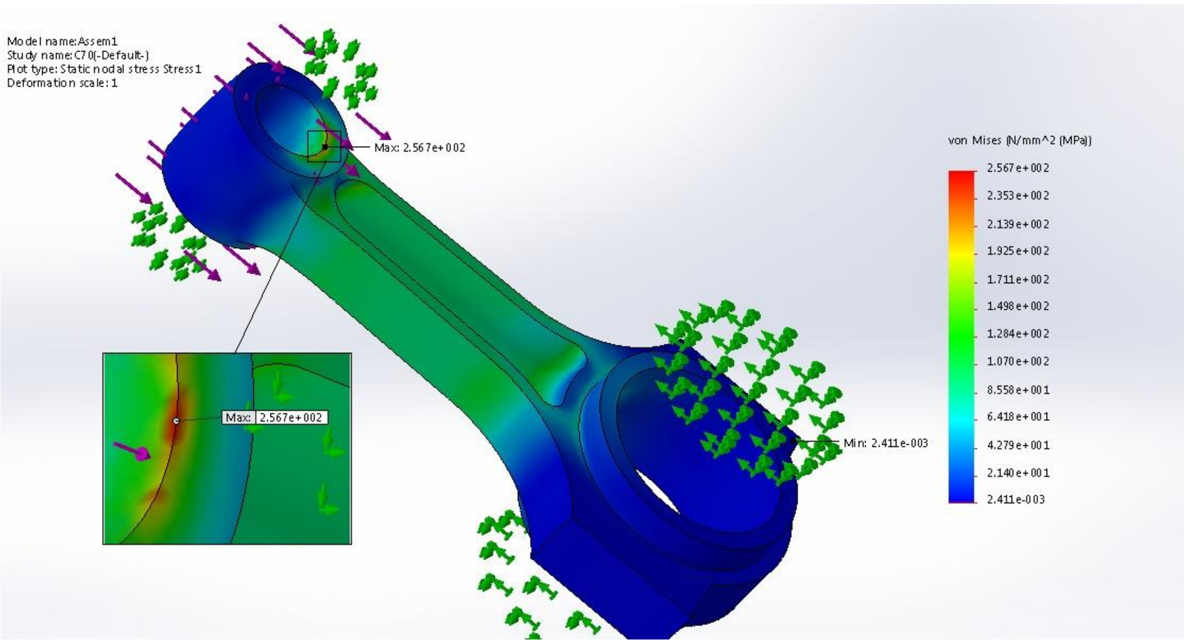


Figure 3-7 Von-Mises stress for AISI 1070 for bearing load 42,331 N.

3.3. Fatigue life of the connecting rod

3.3.1 Calculation of fatigue life for connecting rod 4340 for R = -1 (*for 42,331N*)

From static study (fig. 3-6)

Maximum von stress in compression loading

$$\sigma_{max} = 250\ MPa$$

Loading ratio R = -1

$$\frac{\sigma_{min}}{\sigma_{max}} = -1 \tag{1.12}$$

$$\sigma_{min} = -\sigma_{max}$$

Mean stress

$$\sigma_{mean} = \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{1.10}$$

$$\sigma_m = 0\ MPa$$

Alternative stress

$$\sigma_{alternative} = \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{1.11}$$

$$\sigma_a = 250\ MPa$$

Factor of safety

$$\frac{1}{f.o.s} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_e} \tag{3.1}$$

$\sigma_y = 710\ MPa$ – Yield stress; $\sigma_e\ (or)\ S_e = 555\ MPa$ – Endurance stress

$$f.o.s = 2.22$$

Endurance limit

$$S'_l = 0.75S_u(axial\ loading) \tag{1.16}$$

$$S_e = 0.5S_u;\ S_u \leq 200\ ksi \approx 1400\ MPa \tag{1.17}$$

$$S_u = 1110\ MPa$$

$S_u(or)\sigma_u$ = Ultimate tensile stress

$$S'_l = 832.5\ MPa$$

$$S'_e = S_e \times k_a \times k_{sr} \times k_{sz} \tag{3.2}$$

$k_a -$ Load correction factor for reversed axial load = 0.8

$k_{sr} -$ Surface finish factor = 1.2

$k_{sz} -$ Size factor = 1

$$S'_e = 582.21\ MPa$$

Endurance limit is where the stress level below which the failure will never occur.

$$c = \log\frac{(S'_l)^2}{S'_e} = 3.3 \tag{1.13}$$

$$a = -\frac{1}{3}\log\frac{S'_l}{S'_e} = -0.0646 \tag{1.14}$$

Number of cycles to failure

$$N_f = \left(S_f 10^{-c}\right)^{\frac{1}{a}} \tag{1.15}$$

Fatigue strength

$$S_f = \frac{f.o.s \times \sigma_a}{1-\left(\frac{f.o.s \times \sigma_m}{\sigma_u}\right)}$$

$$S_f = 555\ MPa$$

$S_f\,(Fatigue\ strength)$- Stress level at which a corresponding number of cycles N_f will lead to failure (crack initiation).

$$N_f = 4.001988193 \times 10^8\ number\ of\ cycles\ to\ failure$$

3.3.2. Calculation of fatigue life for connecting rod 4340 for R=-1.25 (*for 42,331N*)

From static study (fig. 3-6)

Maximum von stress

$$\sigma_{max} = 250\ MPa$$

For loading ratio R = -1.25 [35]

$$\sigma_{max} = -312.5\ MPa \tag{1.12}$$

Mean stress

$$\sigma_m = 31.25\ MPa \tag{1.10}$$

Alternating stress

$$\sigma_a = 281.25\ MPa \tag{1.11}$$

Factor of safety

$$f.o.s = 1.82 \tag{3.1}$$

Fatigue strength

$$S_f = 539.52\ MPa$$

Number of cycles

$$N_f = 6.20087259 \times 10^8 cycles \tag{1.15}$$

3.3.3. Calculation of fatigue life for connecting rod 1070 for R = -1 (*for 42,331N*)

From static study (fig. 3-7)

Maximum von stress

$$\sigma_{max} = 256.7\ MPa$$

For loading ratio R = -1

$$\sigma_{min} = 256.7\ MPa \tag{1.12}$$

Mean stress

$$\sigma_{mean} = \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{1.10}$$

$$\sigma_m = 0\ MPa$$

Alternating stress

$$\sigma_{alternative} = \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{1.11}$$

$$\sigma_a = 256.7\ MPa$$

Factor of safety

$$\frac{1}{f.o.s} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_e} \tag{3.1}$$

$$Yield\ strength\ \sigma_y = 573.11\ MPa; Ultimate\ tensile\ strength\ S_{ut}(or)\sigma_u = 965.8MPa$$

$$\sigma_e = 0.5 \times \sigma_u = 482.9\ MPa$$

$$f.o.s = 1.87$$

Endurance limit

$$S'_l = 724.35MPa \tag{1.16}$$

$$S'_e = 482.9 \times 0.8 \times 1.2 \times 1 \tag{3.2}$$

$$S'_e = 463.58\ MPa$$

$$c = \log\frac{(S'_l)^2}{S'_e} = 3.05 \tag{1.13}$$

$$a = -\frac{1}{3}\log\frac{S'_l}{S'_e} = -0.0578 \tag{1.14}$$

Fatigue strength

$$S_f = \frac{f.o.s \times \sigma_a}{1 - \left(\frac{f.o.s \times \sigma_m}{\sigma_u}\right)}$$

$$S_f = 465.24\ MPa$$

Number of cycles to failure

$$N_f = \left(S_f 10^{-c}\right)^{\frac{1}{a}} \tag{1.15}$$

$$N_f = 2.184650873 \times 10^7\ Cycles$$

3.3.4. Calculation of fatigue life for connecting rod 1070 for R = 1.25 (for 42,331N)

From static study (fig.3-7)

Maximum von stress

$$\sigma_{max} = 256.7\ MPa$$

For loading ratio R = -1.25

$$\sigma_{min} = 320.9\ MPa$$

(1.12)

Mean stress

$$\sigma_{mean} = \sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{1.10}$$

$$\sigma_m = 30.06\ MPa$$

Alternating stress

$$\sigma_{alternative} = \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

(1.11)

$$\sigma_a = 288.8\ MPa$$

Factor of safety

$$\frac{1}{f.o.s} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a}{\sigma_e} \tag{3.1}$$

$$f.o.s = 1.53$$

Fatigue strength

$$S_f = \frac{f.o.s \times \sigma_a}{1 - \left(\frac{f.o.s \times \sigma_m}{\sigma_u}\right)}$$

$$S_f = 465.49\ MPa$$

Number of cycles to failure

$$N_f = \left(S_f 10^{-c}\right)^{\frac{1}{a}} \tag{1.15}$$

$$N_f = 4.078941543 \times 10^7\ cycles$$

Table 3.2 Fatigue

S.no.	**Material**	**Loading ratio R**	**Mean Force $\sigma_m (MPa)$**	**Alternating Force $\sigma_a\ (MPa)$**	$f.o.s$	**Number of cycles N_f**	**Probability of occurrence**
1.	AISI 4340	-1	0	250	2.22	4.001988193×10^8 cycles	1
		-1.25	31.25	281.25	1.82	6.20087259×10^8 cycles	1
2.	AISI 1070	-1	0	256.7	1.87	2.184650873×10^7 cycles	1
		-1.25	32.06	288.79	1.53	4.078941543×10^7 cycles	1

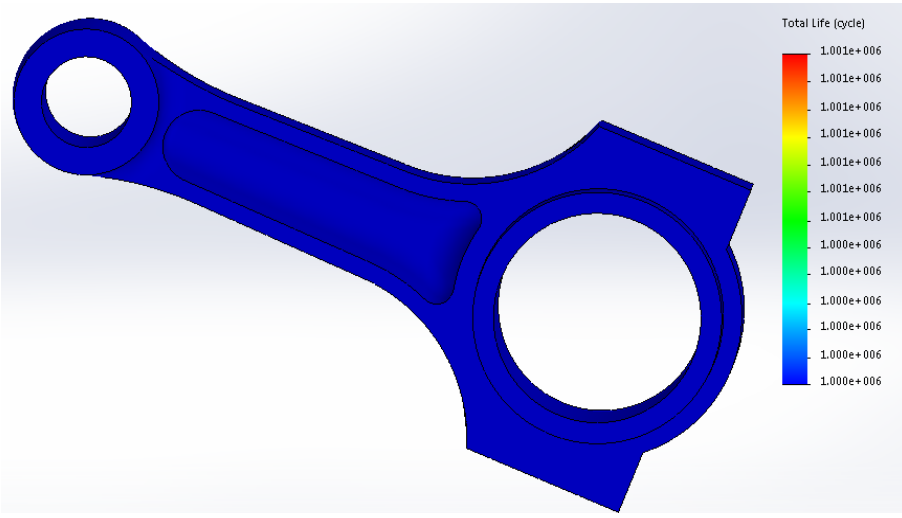


Figure 3-8 Fatigue study showing infinity life (using SolidWorks Fatigue 2015)

The static analysis from (figure 3-6; 3-7) shows how the stress are distributed. The maximum stress concentration is at neck below the gudgeon pin hole. The (table 3.2) shows the variation in alternating stress based on loading ratio and materials used, and how it affecting the fatigue life cycle (calculated results). The AISI4340 steel responded with better life cycle than the AISI 1070 steel material. The fatigue analysis for both materials for different loading ratio from (figure 3-8) shows the stresses acting on the connecting rod are maintained below the endurances stress, since the stress acting on the rod is below the endurance stress there is no crack initiation below 10^6. As it is said a component with beyond 10^6 cycles is said to have infinity life.

CONCLUSION

- The conclusion that could be drawn from the literature survey, is that the design is not much different from all the other existing axial engine. This design is the upgraded form a 2-stroke axial engine (Trebert axial engine 1912) to a 4-stroke axial engine with reduction of size and more parts like (timing gears, rocker arms, push rods, timing gears and any balancing shafts needed).
- The engine is designed having all the necessary calculations, and specifically the connecting rod is designed for maximum engine speed and gas pressure.
- The kinematic study using SolidWorks motion shows that the engine is estimated to produce high torque and power, and the maximum reaction force obtained from the motion study is 4.25% less compared to analytical calculation. Since the motion study makes use of all the mass properties of all components in engine assembly, but for the analytical calculation is done for single piston assembly. So taking that difference in reaction force into consideration the connecting rod is designed for the maximum reaction force to avoid failure in real-time application.
- The static analysis showed the maximum stress concentration was at the neck, and the crack was initiated at 10^5cycles. So with redesigning there was a little mass removal near the gudgeon pin hole and slight change in I section the connecting rod stress distribution was reduced and end result the rod was able to withstand up to $10^7 to 10^8 cycles$ (infinity life).
- On comparison with the life cycle predicted for the connecting rod for materials AISI 4340 steel and AISI 1070 steel. AISI steel is much more reliable than AISI 1070 steel for longer life. AISI 4340 material is with carbon percentage 0.37-0.43% in austenitic region in Iron-Carbon phase diagram were as the AISI 1070 which also lies in the austenitic region has carbon percentage 0.65-0.75% which makes the component low ductile compared to AISI 4340.
- Create 3D model was animated using SolidWorks with realistic rendering using key shot to show the engine operation. And also a 2-D drafting of engine assembly was created with all standards using solid works.
- This project gave me a platform to understand the how to design a component for a required life in detail, also scope for further studies on advance research on connecting rod.

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ANNEXURES

A-1. Mat Lab program

```
clear all;
close all;
clc;
l=160;%length of connecting rod
r=40;%crank  radius
v=12.56;% angular velocity for 3000rpm
n=l/r;%ratio of length of the connecting rod
T=[0:360];%theta
TT=0;%theta(at maximum)
D=100;%bore diameter
P=5*(pi/4)*D^2%explosion force acting on piston
%------------------------IF<inertia force>----------------------
IF=(0.38809*v^2*r)*(cos(T*pi/180)+(cos(2*(T*pi/180))/n));
%------------------------R<Reaction force>---------------------
R=P-IF;
IF1=(0.38809*v^2*r)*(cos(TT*pi/180)+(cos(2*TT*pi/180))/n))
Rt=P+IF;%maximum reaction force
plot(T,R,'b')%graph plot reaction force from 0-360degree.
xlable('Theta (Degree)');
ylable('Reaction force (Newton)');
legend('TDC-BDC-TDC)
```

FORMAT	ZONE	NO.	DESIGNATION	NAME	QUANTITY	NOTES
			Documentation			
A1			EN-00.00.000 AD	Assembly drawing	2	A1
			Assembly Units			
A3		1	EN-00.01.000	Engine Cylinder assembly	1	
A3		2	EN-00.02.000	Engine Head assembly	1	
			Parts			
A3		3	EN-00.00.003	Engine cylinder	1	
A4		4	EN-00.00.004	Crankshaft with driving gear	6	
A4		5	EN-00.00.005	Transmision shaft with driven gear	1	
A4		6	EN-00.00.006	Crankshaft bearing cap	1	
A4		7	EN-00.00.007	Crankshaft bearing set	24	
A4		8	EN-00.00.008	Piston head	6	
A4		9	EN-00.00.009	Pressure ring / scavenging rings	12	
A4		10	EN-00.00.010	Oil ring assembly	6	
A4		10	EN-00.00.010	Gudgeon pin	6	
A4		11	EN-00.00.011	Connecting rod	6	
A4		12	EN-00.00.012	Connecting rod cap	6	
A4		13	EN-00.00.013	Connecting rod bearing set	6	

Resp. department	Technical reference	Document type	Document status
MED		Parts list	Training

Legal owner	Created by	Title, Supplementary title	EN-00.00.000			
KTU	Mohamed Riyazdheen Shahul Hameed	AXIAL IC-ENGINE	Rev.	Date	Lang.	Sheet
	Approved by: Sigitas Kilikevičius		A	2016-05-14	En	1/3

FORMAT	ZONE	NO.	DESIGNATION	NAME	QUANTITY	NOTES
A3		14	EN-00.00.014	Cylinder head	1	
A4		15	EN-00.00.015	Intake valve	6	
A4		16	EN-00.00.016	Exhaust valve	6	
A4		17	EN-00.00.017	Retainer sleeve	12	
A4		18	EN-00.00.018	Valve seal	12	
A4		19	EN-00.00.019	Retainer sleeve lock	12	
A4		20	EN-00.00.020	Valve springs	12	
A4		21	EN-00.00.021	Gasket cylinder head	1	
A4		22	EN-00.00.022	Overhead cam inlet+exhaust	1	
A4		23	EN-00.00.023	Exhaust manifold	2	
A4		24	EN-00.00.024	Gasket exhaust manifold	6	
A4		25	EN-00.00.025	Engine intake manifold	1	
A4		26	EN-00.00.026	Gasket intake manifold	1	
A4		27	EN-00.00.027	Intake manifold cap	1	
A4		28	EN-00.00.028	Gasket intake manifold cap	1	
A4		29	EN-00.00.029	Oil sump	1	
A4		30	EN-00.00.030	Gasket oil sump	1	
A4		31	EN-00.00.031	Oil sump cap	1	
A4		32	EN-00.00.032	Spark plug	6	

Resp. department	Technical reference	Document type	Document status
MED		Parts list	Training

Legal owner: KTU

Created by: Mohamed Riyazdheen Shahul Hameed

Approved by: Sigitas Kilikevičius

Title, Supplementary title: **AXIAL IC-ENGINE**

EN-00.00.000

Rev.	Date	Lang.	Sheet
A	2016-05-14	En	2/3

FORMAT	ZONE	NO.	DESIGNATION	NAME	QUANTITY	NOTES
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				Bolts		
				AS 1420 - M6 x 45-S	12	
				AS 1420 - M8 x 50-S	12	
				AS 1110.2 - M36 x 70 -S	2	
				AS 1112.3 C- M12-N	12	
				"IS 15582 - M10 x 100 x 26-C"	6	
				"B18.3.6M - M6 x 1.0 x 20 Spline Socket Type I Cup Pt. SS --S"	36	
				"B18.3.1M - 10 x 1.5 x 50 Hex SHCS -- 32SHX"	7	
				"B18.2.3.2M - Formed hex screw, M12 x 1.75	12	
				Bearings		
				"AFBMA 20.1 - 10-70 - l,DE,NC,Full"	1	
				"AFBMA 20.1 - 10-70 - 2,DE,NC,22"	1	
				IS 6458 DC - 49-50-D	6	
				washer		
				AS 1237.1 S - 36	2	
				"B18.3.1M - 12 x 1.75 x 25 Hex SHCS -- 25NHX"	5	
				Wrist pins		
				B27.7M - 3BM1-26	12	

Resp. department	Technical reference	Document type	Document status
MED		Parts list	Training

Legal owner	Created by / Approved by	Title, Supplementary title	EN-00.00.000			
KTU	Created by: Mohamed Riyazdheen Shahul Hameed	**AXIAL IC-ENGINE**	Rev.	Date	Lang.	Sheet
	Approved by: Sigitas Kilikevičius		A	2016-05-14	En	3/3


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(812)
(135)
(247)
(256)
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Φ 70 L0/k6
A2 (2 : 3)
A3 (2 : 2.5)
Φ 26H7/g6
A3 (2 : 3)
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Φ 110H7/l0
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A6 (2 : 2.5)
B-B (1:2.5)
A7 (2 : 2.5)
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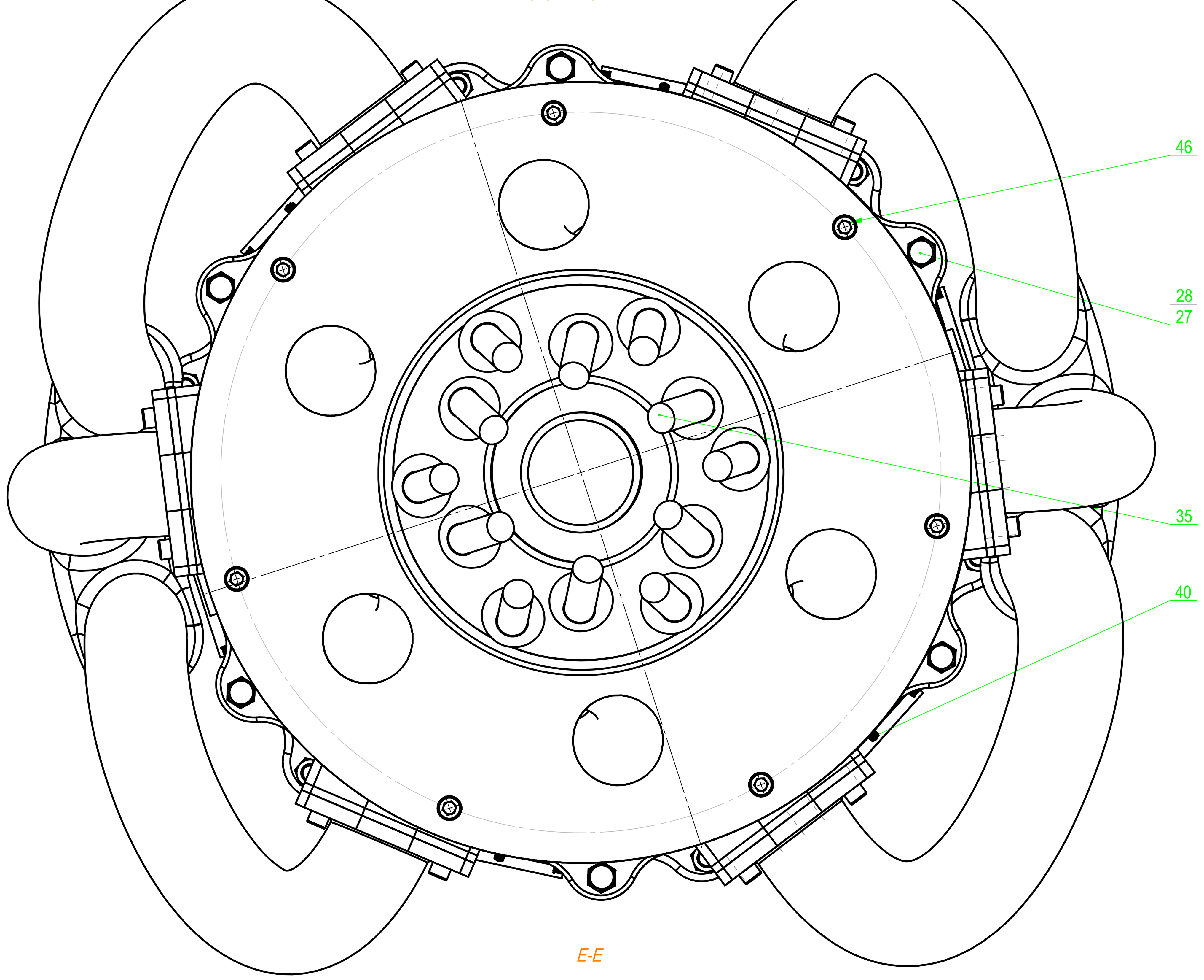
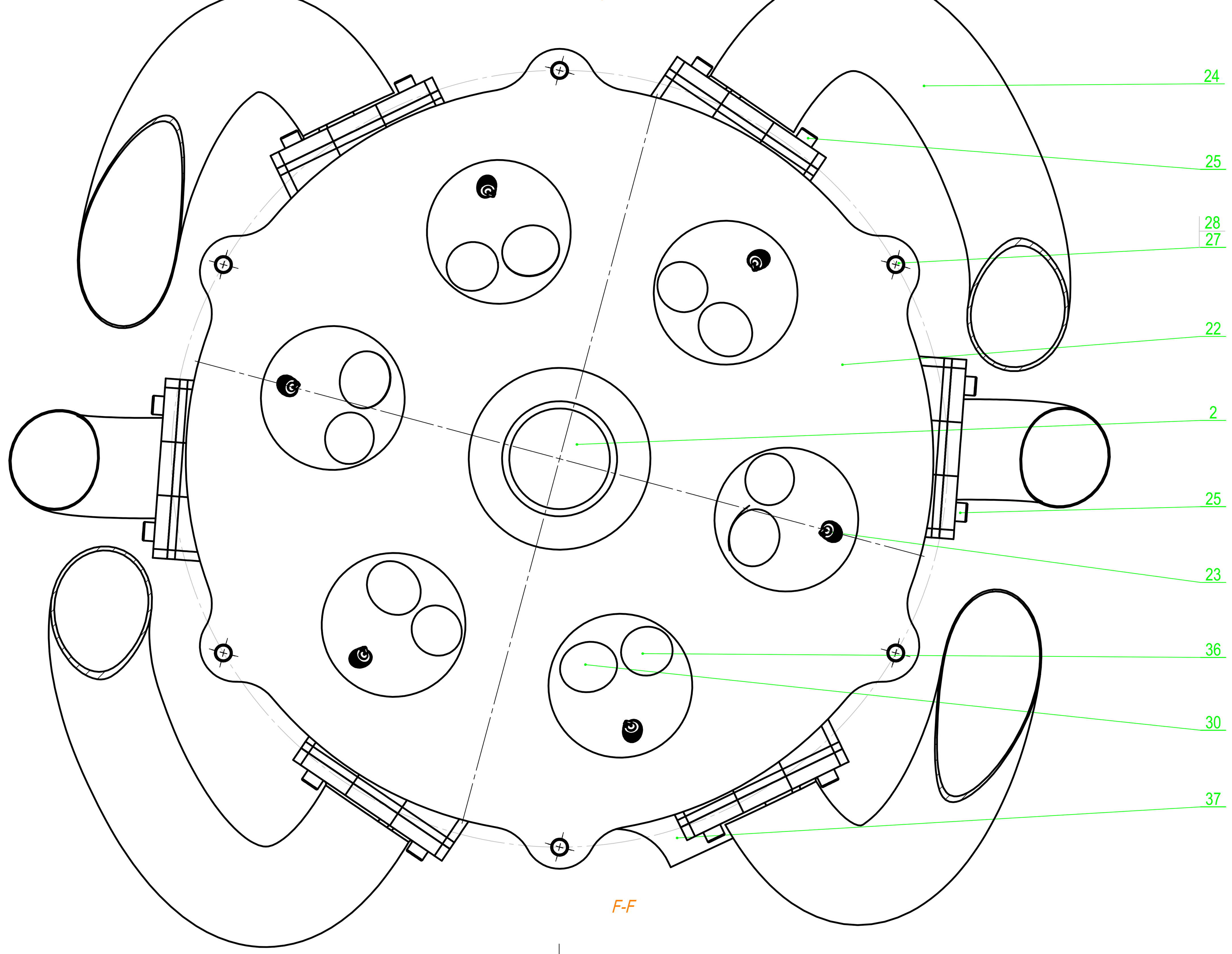
6xΦ 50 L0/k6
G-G (1:2)
(Φ 520)
(Φ 250)
(Φ 480)
(Φ 540)
Unspecified tolerance limit according LST EN 22768.
File name
Additional information
Material
Scale
1:3
Resp. department
MED
Technical reference
Document type
Assembly drawing
Document status
Training
Legal owner
KTU
Created by
Mohamed Riyazdheen Shahul Hameed
Approved by
Sigitas Kilikevičius
Title, Supplementary title
AXIAL IC-ENGINE
EN-00.00.000 AD
Rev.
A
Date
18-05-2016
Lang.
En
Sheet
1/2

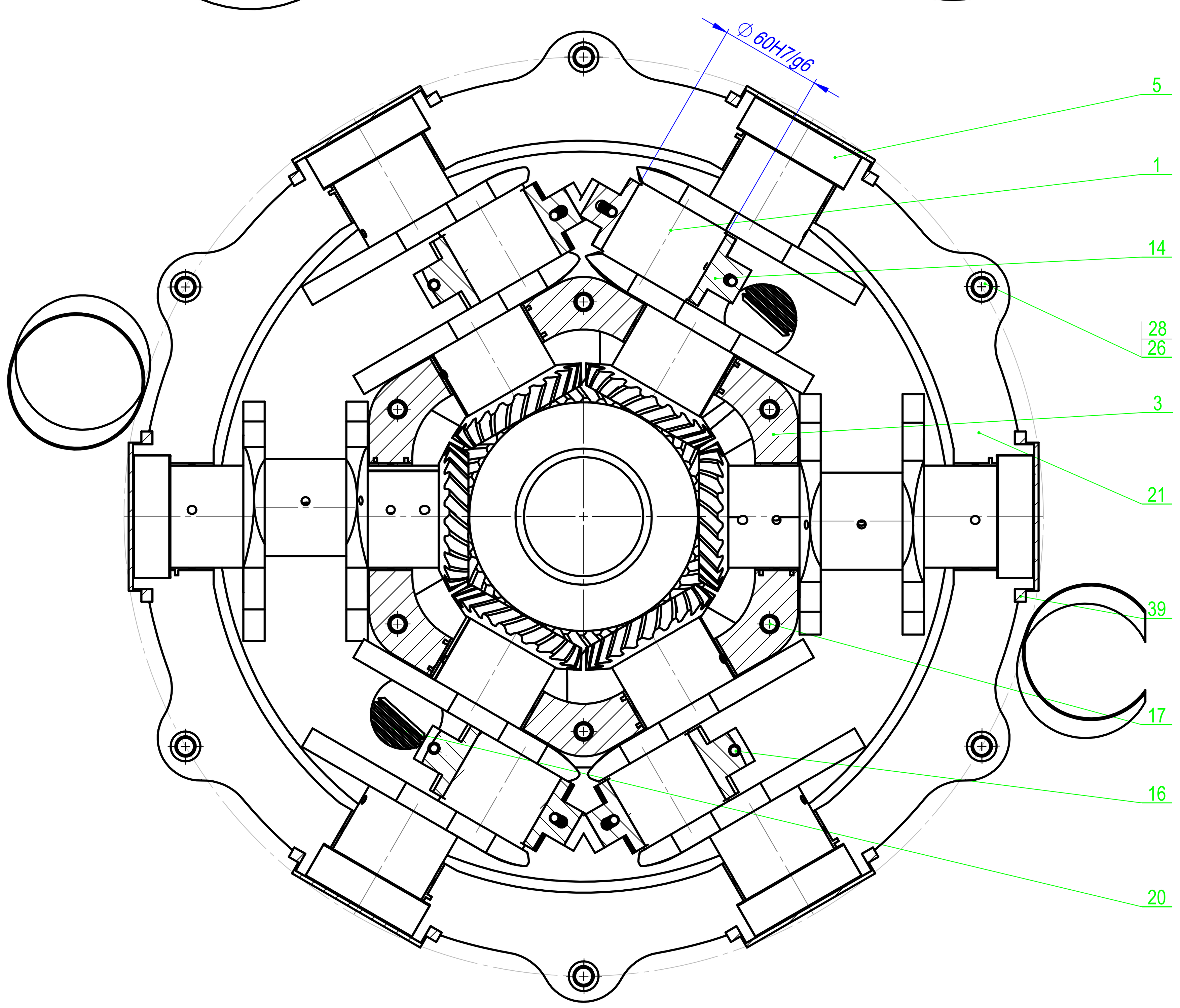
C-C ⌒ 18°



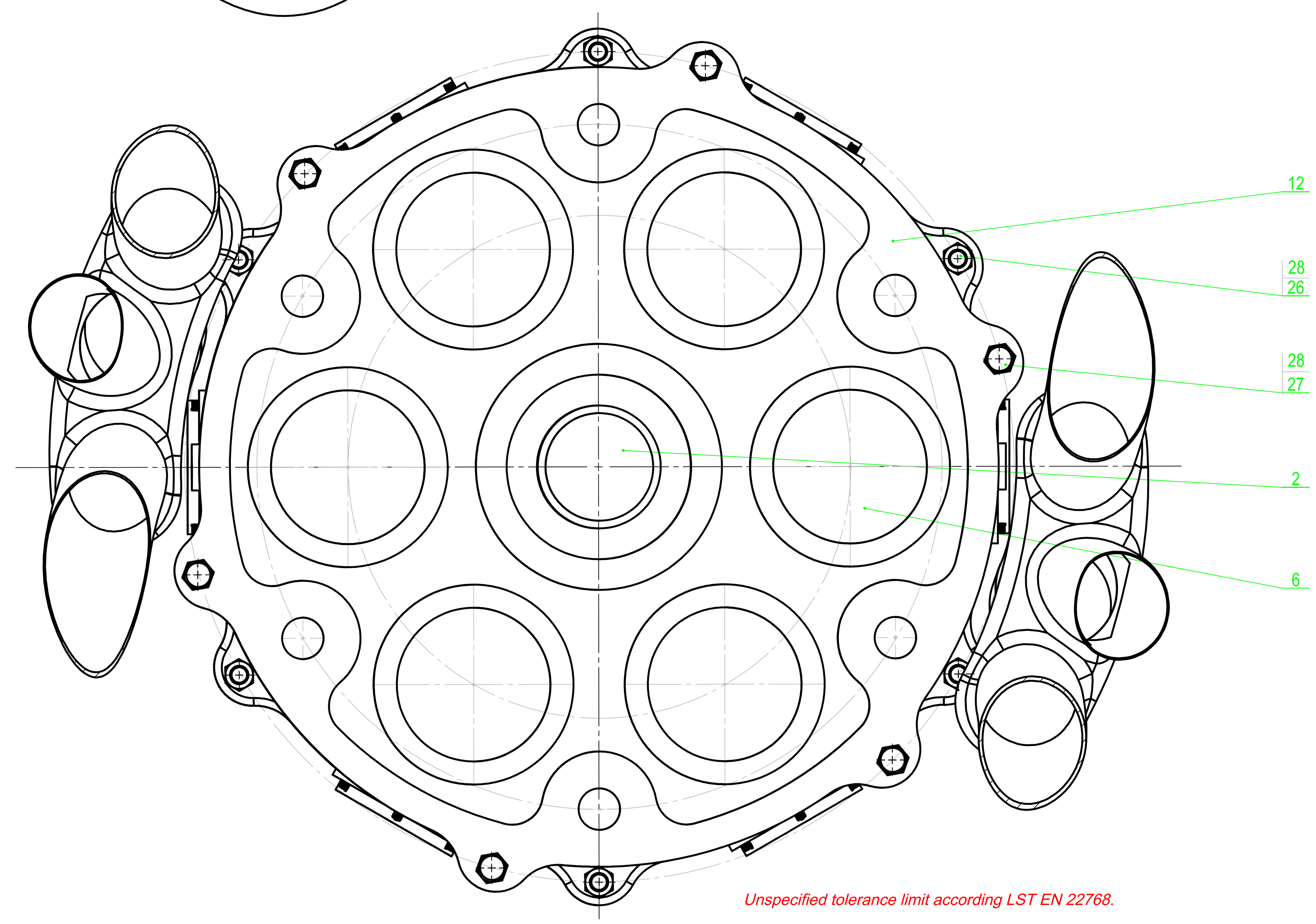
D-D ⌒ 15°



E-E



F-F



Unspecified tolerance limit according LST EN 22768.

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Resp. department MED	Technical reference	Document type Assembly drawing	Document status Training	
Legal owner KTU	Created by Mohamed Riyazdheen Shahul Hameed	Title, Supplementary title AXIAL IC-ENGINE	EN-00.00.000 AD	
	Approved by Sigitas Kilikevičius		Rev. A / Date 18-05-2016 / Lang. En / Sheet 2/2	