DESIGN OF TWO STROKE SI LINEAR ENGINE WITH SPRING MECHANISM

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DESIGN OF TWO STROKE SI LINEAR ENGINE WITH SPRING MECHANISM

MOHD FAIZAL BIN MOHD PAUZI

A report submitted in partial fulfillment of the
requirement for the award of the
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
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NOVEMBER 2008

SUPERVISOR DECLARATION

I hereby declare that I have read this project report and in my opinion this project report is sufficient in terms of scope and quality for the award of Bachelor in Mechanical Engineering with Automotive Engineering.

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DECLARATION

I declare this thesis that was entitled “Design of Two Stroke SI Linear Engine with Spring Mechanism” is the result of my own research except as cited in the references. The thesis has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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DEDICATION

To my beloved parents, Mr. Mohd Pauzi Bin Mohd Tahir and Mdm. Normah Binti Awaludin, other siblings, family and friends, without whom his/her efforts in encouraging and supporting my dream to continue my study in the higher education of Mechanical Engineering field. And all the staffs of Faculty Mechanical Engineering from Universiti Malaysia Pahang especially my supervisor Prof Madya Dr Rosli Bin Abu Bakar and my co-supervisor Mr. Aguk Zuhdi Muhammad Fathallah for giving me priceless knowledges in order to accomplish this project.

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ABSTRACT

The objectives of this research project is to design the spring mechanism of single cylinder two stroke SI linear engine, to analyze the stress of spring mechanism of linear engine and draw the single cylinder two stroke SI linear engine. The problem arise when the modern control technology that available today is expensive to be used with small linear engine and spring is proposed to solve the related problem of piston motion control for this project. The design of the linear engine with spring mechanism for this project is based on conventional two-stroke engine. The modification is made for the crankcase, crankshaft, and connecting rod of conventional two-stroke engine with new crankcase, spring and new connecting rod of linear engine. The SI linear engine with spring mechanism is modeled by using Solid works software and the spring software is used to design the spring for this linear engine for this project.

ABSTRAK

Projek Sarjana Muda ini bertujuan mereka satu enjin dua lejang yang menggunakan spring bagi menggantikan “Crankshaft” enjin dua lejang yang asal. Masalah timbul apabila teknologi sistem kawalan terbaru untuk mengawal pergerakan piston yang terdapat di pasaran sekarang adalah terlalu mahal apabila teknologi ini hendak digunakan untuk menggantikan “Crankshaft” bagi enjin dua lejang dalam projek ini. Reka bentuk untuk enjin dua lejang bagi projek ini adalah direka berdasarkan enjin dua lejang yang asal. Pengubahsuaian dibuat ke atas enjin dua lejang yang asal adalah pada bahagian “Crankcase”, “Crankshaft” dan juga “Connecting Rod” digantikan dengan “Crankcase”, “Crankshaft” dan juga “Connecting Rod “ enjin yang baru. Enjin bagi projek sarjana muda ini di reka menggunakan perisian Solid work manakala perisian mereka spring digunakan dalam proses untuk mereka spring bagi enjin untuk projek ini.

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LIST OF SYMBOLS

d	Cylinder Bore
A	Piston Area
L	Stroke
Vs	Displacement or Swept Volume
N	Number of cylinders in Engine
Vc	Clearance Volume
r	Compression Ratio
V_T	Cylinder Volume When the Piston is at the Bottom Dead Centre
T	Temperature
n_C	Number of Spring End
n_G	Ground Coils of Spring
S_s	Desired level of safety of a spring exposed to static loading
S_f	Desired level of safety of a spring exposed to fatigue loading
G	Modulus of elasticity at operational temperature
P	Density
S_u	Ultimate tensile strength
τ_A	Permissible torsional stress
τ_e	Ultimate fatigue strength in shear
τ_f	Fatigue strength by finite life

LIST OF ABBREVIATION

TDC	Top Dead Centre
IDC	Inner Dead Centre
BDC	Bottom Dead Centre
ODC	Outer Dead Centre
IMEP	Indicative Mean Effective Pressure

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CHAPTER 1

INTRODUCTION

1.1 Project Background

The conventional two stroke engine is mounted with the crankshaft. Crankshaft is a device, which converts the up and down movement of the piston into rotary motion. This shaft is presented at the bottom of an engine and its main function is to rotate the pistons in a circular motion. In order to make this engine become linear engine which is a engine that do not use the crankshaft to control the piston motion but it is a result of the interaction of forces from the combustion cylinder gases, a rebound device and a load device, the crankshaft is substituted with the new kind of the connecting rod and also coupled with spring. The new connecting rod and spring in this project are defined as spring mechanism. This new kind of connecting rod and spring are used in the linear engine which functioning same as the crankshaft of the conventional two stroke engine except that the movement of the new connecting rod and spring is linear compare to crankshaft moves in rotation. So, this project focus more on the design of the spring at early stage and after the spring design is finished, the best design of the spring is selected, then the project continue to the stage of linear engine design. The linear engine design is the same as the conventional of the two stroke engine except that the modification is made at the crankcase, crankshaft of the conventional engine. The spring of linear engine is designed by using Mitcalc- Helical Compression Spring Version 1.12. The input parameters used in the designed of the spring based on the performance of the linear engine in the form of force which is subjected to the spring. The performance of the linear engine is obtained from GT-Power software. The engine which is simulated in

GT-Power environment is actually conventional two stroke engine but the conventional two stroke engine can be considered as linear engine because the modification has been made at the friction factor of the conventional two stroke engine. The usage of spring for SI linear engine on this project makes the working principle of the linear engine simple. Apart from that, if the design process success the cost to fabricate this linear engine is cheaper than the other linear engine. It is because this SI linear engine uses spring that work as rebound device for this engine that no need the usage of electricity compare to the other linear engine such as single piston hydraulic free piston engine. The rebound device for this linear engine type requires supply of electric power to make it function. This linear engine latter will be used with linear electric generator for producing the electric.

1.2 Problem Statement

The main challenge of free piston engine is the piston motion control as the engine does not have a crankshaft to limit the dead centre of the piston motion, other means of control must be introduced in order to avoid excessive in cylinder gas pressures, the piston hitting the cylinder head, while at the same time ensure a sufficiently high compression ratio for fuel spark ignition and efficient combustion. The control challenges that associated with the concept can be treated by using modern control technology such as hydraulic cylinder and a gas filled bounce chamber. However, the problem arises when that modern control technology is very expensive to be used with the small linear engine and other mean of piston motion control need to be figured out. The spring is used as piston motion control for this project because it is simple, cheap and can also solve the related problem with free piston engine if thorough study is conducted.

1.3 Objectives

The objectives of this project are, design the spring for the linear engine, determine the best spring design for the linear engine and lastly, design the single cylinder of two stroke SI linear engine with spring mechanism.

1.4 Project Scopes

The scopes for this project are, obtain data of linear engine performance and measure the dimension design the spring for the linear engine for the first project stage. After that, design the spring for the linear engine and determine the best spring design by comparing the designed springs based on parameter of interest for the second stage of this project. Lastly, design the 3D model of linear engine where the third stage of this project.

CHAPTER 2

THEORITICAL BACKGROUND AND LITERATURE RIVIEW

2.1 Definition of Engine

An engine is a device which transforms one form of energy into another form. However, the efficiency of conversion plays an important role while transferring the energy from one form to another. Most of the engines convert thermal energy to the mechanical work which is another term called heat engines. Heat engine is a device which transforms chemical energy of fuel into thermal energy. This thermal energy is utilized to perform useful work. Thus, thermal energy is converted to mechanical energy in heat engine. Heat engines can be divided into the following two categories [7]:

i. Internal Combustion Engines (IC Engines)
ii. External Combustion Engines (EC Engines)

Engines whether Internal Combustion or External Combustion are of two types. The engines can be classified into the following types [7]:

i. Rotary engines
ii. Reciprocating engines

A detailed classification of heat engines is shown in Figure 2.1. The most widely used ones of heat engines are the reciprocating internal combustion engine, the gas turbine and the steam turbine. The steam engine is rarely used nowadays. The reciprocating internal combustion engine has some advantages over the steam turbine because of the absence of heat exchangers in the passage of the working fluid (boilers and condensers in steam turbine plant). This results in the mechanical simplicity and improve power efficiency of the internal combustion engine. Another advantage of reciprocating internal combustion engine over the other type of engines is that very high working fluid temperature in the cycle of reciprocating engine can be employed resulting in higher thermal efficiency. Furthermore, in internal combustion engines, higher thermal efficiency can be obtained with moderate maximum working pressure of the fluid in the cycle, and therefore, the weight to power ratio is less than that of the steam turbine plant. Also it is possible to develop reciprocating internal combustion engines of very small output with very reasonable thermal efficiency and cost [7][19].

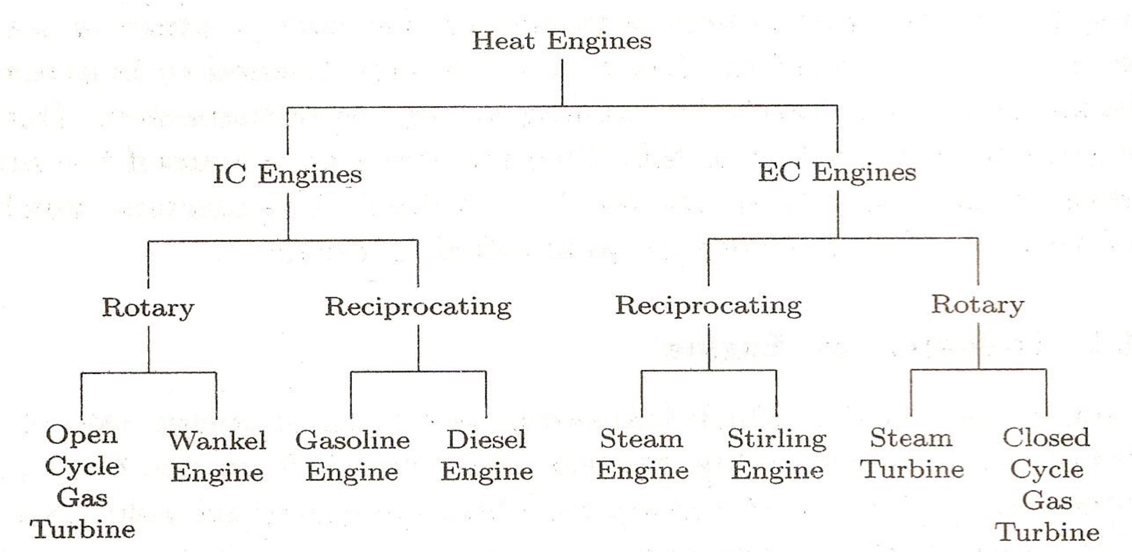


Figure 2.1 Detailed Classifications of Heat Engines [7]

The main disadvantage of this type of engine is the problem of vibration caused by the reciprocating components. Also, only certain liquid or gaseous fuels of given specification can be efficiently used. This fuel is relatively more expensive nowadays because the source of the fuel is depleting due to increasingly usage of this fuel in the whole world. The reciprocating internal combustion engines have been found suitable for use in automobiles, motorcycles and scooters, power boats, ships, slow speed aircraft, locomotives and power units of relatively small output [7].

2.2 Basic Engine Components and Nomenclature

Even though the reciprocating internal combustion engines look quite simple, they are highly complex machines. There are hundreds of components which have to perform their functions satisfactorily to produce output power. So this chapter will go through the important engine components and nomenclature associated with the spark ignition engines [7].

2.2.1 Engine Components

Figure 2.2 shows a cross section of a single cylinder spark-ignition engine with over-head valves. The major components of the engine and their functions are briefly described below [7].

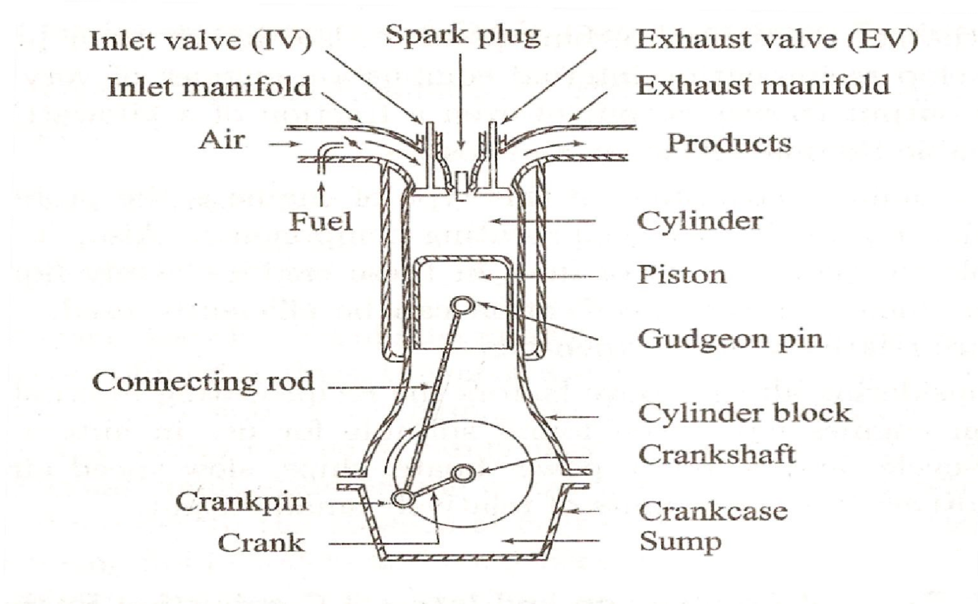


Figure 2.2 Cross Section of a Single Cylinder Spark-Ignition Engine [7]

Cylinder Block: The cylinder block is the main supporting structure for the various components. The cylinder head is mounted on the cylinder block. The cylinder head and cylinder block are provided with water jackets in the case of water cooling or with cooling fins in the case of air cooling. Cylinder head gasket is incorporated between the cylinder block and the cylinder head. The cylinder head is held tight to the cylinder block by numbers of bolts or studs. The bottom portion of the cylinder block is called crankcase. A cover called crankcase which becomes a sump for lubricating oil is fastened to the bottom of the crankcase. The inner surface of the cylinder block which is machined and finished accurately to cylindrical shape is called bore [7][19].

Cylinder: It is cylindrical vessel or space in which the piston makes a reciprocating motion. The varying volume created in the cylinder during the operation of the engine is filled with the working fluid and subjected to different thermodynamic processes[7][19].

Piston: It is cylindrical component fitted into the cylinder forming the moving boundary of the combustion system. It fits perfectly into the cylinder providing a gas tight space with the piston rings and the lubricant. It forms the first link in transmitting the gas forces to the output shaft [7][19].

Combustion Chamber: The space enclosed in the upper part of the cylinder, by the cylinder head and the piston top during the combustion process. The combustion of fuel and the consequent release of thermal energy results in building up of pressure in this part of the cylinder [7][19].

Inlet Manifold: The pipe which connects the intake system to the inlet valve of the engine and through which air or air-fuel mixture is drawn into the cylinder [7][19].

Exhaust Manifold: The pipe which connects the exhaust system to the exhaust valve of the engine and through which the products of combustion escape into the atmosphere.

Spark Plug: It is a component to initiate the combustion process in Spark Plug engines and usually located on the cylinder head[7][19].

Connecting Rod: It interconnects piston and the crankshaft and transmits the gas forces from the piston to the crankshaft. The two ends of the connecting rod are called as small end and the big end and it is shown in the Figure 2.3. Small end is connected to the piston by gudgeon pin and the big end is connected to the crankshaft by crankpin [7].

Crankshaft: It converts the reciprocating motion of the piston into useful rotary motion of the output shaft. In the crankshaft of a single cylinder engine there is pair of crank arms and balance weights. The balance weights are provided for static and dynamic balancing of the rotating system. The crankshaft is enclosed in a crankcase [7].

Piston Rings: Piston rings, fitted into the slots around the piston, provide tight seal between the piston and the cylinder wall thus preventing leakage of combustion gases. It is shown in the Figure 2.3. [7]

Gudgeon Pin: It forms the link between the small end of the connecting rod and the piston. [7]

Camshaft: The camshaft and its associated parts control the opening and closing of the two valves. The associated parts are push rods, rocker arms, valve springs and tappets. This shaft also provides the drive to the ignition system. The camshaft is driven by the crankshaft through timing gears. [7]

Cams: These are made as integral parts of the camshaft and are designed in such a way to open the valves at the correct timing and to keep them open for the necessary duration.[7]

Flywheel: The net torque imparted to the crankshaft during one complete cycle of operation of the engine fluctuates causing a change in the angular velocity of the shaft.

In order to achieve a uniform torque an inertia mass in the form of a wheel is attached to the output shaft and this wheel is called the flywheel. [7]

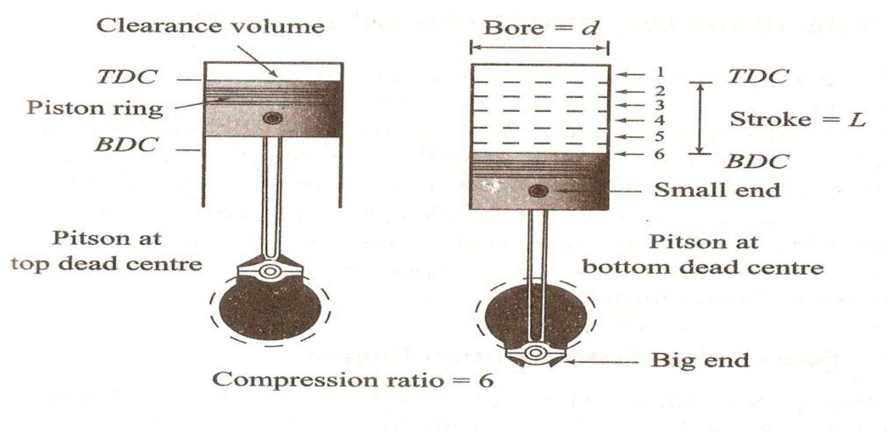


Figure 2.3 Top and Bottom Dead Centre [7]

2.2.2 Nomenclature

Cylinder Bore (*d*)**:** The nominal inner diameter of the working cylinder is called the cylinder bore and is designated by the letter *d* and is usually expressed in millimeter (mm). [7]

Piston Area (*A*)**:** The area of a circle of diameter equal to the cylinder bore is called the piston area and is designated by the letter *A* and usually expressed in square centimeter (cm^2).[7]

Stroke (*L*)**:** The nominal distance through which a working piston moves between two successive reversals of its direction of motion is called the stroke and is designated by the letter *L* and is expressed usually in millimeter (mm).[7]

Stroke to Bore Ratio: *L/d* ratio is an important parameter in classifying the size of the engine.

If $d<L$, it is called under-square engine. If $d=L$, it is called square engine. If $d>L$, it is called over-square engine. An over-square engine can operate at higher speeds because of larger bore and shorter stroke. [7]

Dead Centre: The position of the working piston and the moving parts which are mechanically connected to it, at the moment when the direction of the piston motion is reversed at either end of the stroke is called the dead centre. There are two dead centers in the engines and indicated in Figure 2.3. They are: [7]

i. Top Dead Centre (TDC): It is the dead centre when the piston is farthest from the crankshaft. It is designated as TDC for vertical engines and Inner Dead Centre (IDC) for horizontal engines.
ii. Bottom Dead Centre (BDC): It is the dead centre when the piston is nearest to the crankshaft. It is designated as BDC for vertical engines and Outer Dead Centre (ODC) for horizontal engines.

Displacement or Swept Volume (*Vs*): The nominal volume swept by the working piston when travelling from one dead centre to the other dead. It is expressed in terms of cubic centimeter (cc) and given by: [7]

$$Vs = A \text{ x } L = \Pi/4\, d^2L \tag{2.1}$$

Cubic Capacity or Engine Capacity: The displacement volume of a cylinder multiplied by number of cylinders in an engine will give the engine capacity. For example if there are *N* cylinders in an engine, then [7]

$$\text{Cubic Capacity} = Vs \text{ x } K \tag{2.2}$$

Clearance Volume (*Vc*): The nominal volume of the combustion chamber above the piston when it is at the top dead centre is the clearance volume. It is designated as *Vc* and expressed in cubic centimeter (cc). [7]

Compression Ratio (*r*): It is the ratio of the total cylinder volume when the piston is at the bottom dead centre, $V_{T,}$ to the clearance volume, *Vc.* It is designated by the letter *r.* [7]

$$r = V_T / Vc = Vc + Vs / Vc = 1 + Vs / Vc \tag{2.3}$$

2.3 Working Principle of Two Stroke Engine

In two stroke engine, there is one important process that known as scavenging. Various different scavenging systems have been designed. Today, three main categories are generally accepted which: loop scavenging, cross scavenging and uniflow scavenging [1] [2] [3]. A two stroke engine is one which completes its cycle of operation in one revolution of the crankshaft or in two strokes of the piston [4] [5]. In this engine the functions of the intake and exhaust processes of the four-stroke engine are taken care of by the incoming fresh charge which is compressed either in the crankcase or by means of a separate blower while the engine piston is near the bottom dead center [6] [7]. The engine piston needs only to compress the fresh charge and expand the product of combustion. Since a two-stroke engine will have twice as many cycles per minute as a four-stroke engine, the power output of this engine also depend upon the number of kilograms of air per minute available for combustion [6] [8].

For the crankcase scavenged engine, as the piston moves down, it first uncovers the exhaust port, and the cylinder pressure drops to atmospheric level as the combustion product escape through these ports [9] [6] [7]. Further, downward motions of the piston uncover the transfer port, permitting slightly compressed mixture or air (depending upon the type of the engine) in the crankcase to enter the engine cylinder [10] [11] [12] [13].

The top of the piston and the port are usually shaped in such a way that the fresh air is directed towards the top of the cylinder before showing towards the exhaust ports. This is for the purpose of scavenging the upper part of the cylinder of the combustion products and also to minimize the flow of the fresh fuel-air mixture directly through the exhaust ports [7] [14] [15]. The projection on the piston is called the deflector [2].

As the piston returns from bottom center, the transfer ports and then the exhaust ports are closed and compression of the charge begins. When the exhaust slot is uncovered near the end of the power stroke, immediately followed with an intake process of compressed air or air-fuel mixture [16]. Motion of the piston during compression lowers the pressure in the crankcase so that the fresh mixture or air is drawn into the crankcase through the inlet reed valve [7] [17] [18]. Ignition and expansion take place in the usual way, and the cycle is repeated. Due to the flow restriction in the inlet reed valve and the transfer ports the engine gets charged with less than one cylinder displacement volume [19] [7] [10] [20].

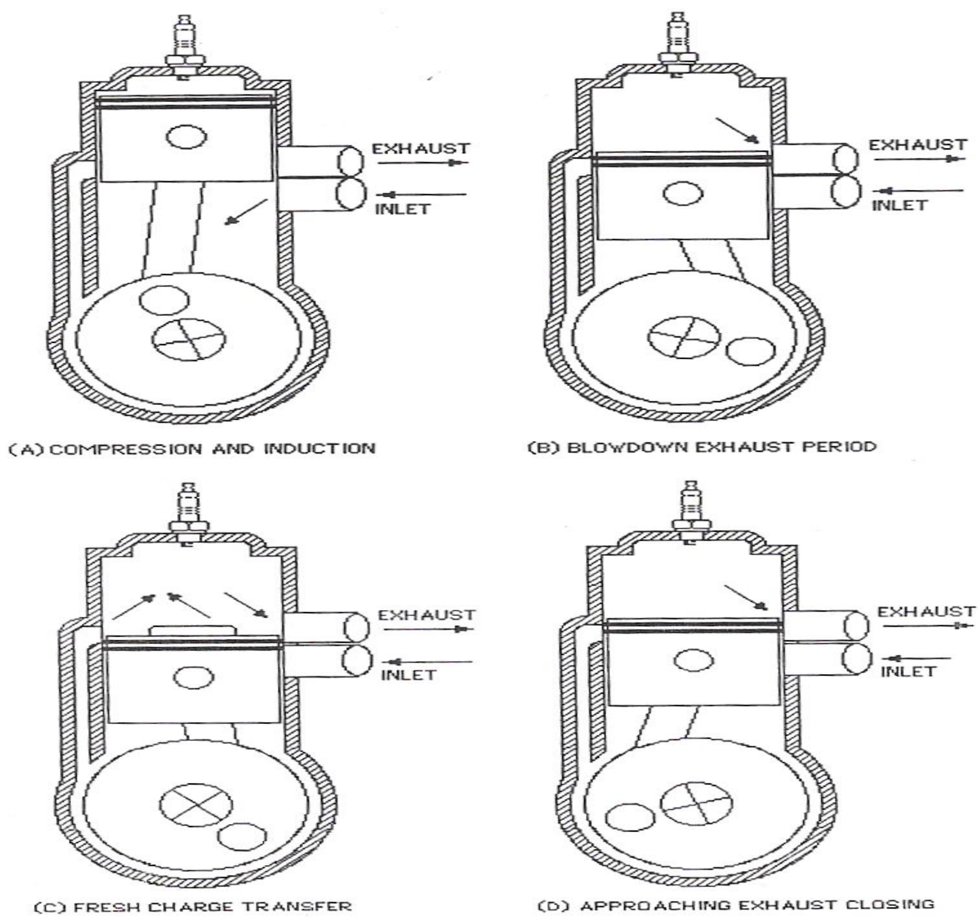


Figure 2.4: Working Principle of Two-Stroke Engine [21]

2.4 Free Piston Engine History

The free-piston engine concept was first presented by Pescara [20] in 1928, and since then a number of free-piston designs have been proposed. Common for these is that the piston motion is not restricted by the motion of a rotating crankshaft, as known from conventional engines, but that the piston is free to move between its endpoints, only influenced by the gas and load forces acting upon it. This gives the free-piston engine some distinct characteristics, most importantly variable stroke length and high control requirements. The original Pescara patent describes a single piston spark ignited air compressor. Pescara started his work on free-piston engines around 1922 and he developed prototypes with both spark ignition (1925) and diesel combustion (1928). The latter led to the development of the Pescara free-piston air compressor [23]. Pescara continued his work on free-piston machinery and also patented a multi-stage free-piston air compressor engine in 1941 [24].

Only a handful of successful free-piston engines been reported. Free-piston air compressors developed by the German company Junkers were used by the German navy during World War 2, supplying compressed air for launching torpedoes [21]. In the 1940s, free-piston engines found use as gas generators, feeding hot gas to a power turbine. This concept was employed in marine and stationary power plants, most successful being a model developed by Société Industrielle Générale de Mécanique Appliquée (SIGMA) in France [22]. Both Ford and General Motors developed prototype vehicles with small-scale free-piston gas generators as prime movers, but none of these made it past the prototype stage [23] [24]. As the technology of both conventional engines and gas turbines matured, the interest in the free-piston engine vanished in early 1960s.

With the introduction of modern control methods, the free-piston engine has again caught interest among present-day engineers seeking to reduce engine emissions and increase efficiency. Most successful linear engine is the hydraulic free-piston engine, where among others the Dutch company Innas BV has reported performance advantages over conventional technology [25]. A number of other research groups are also working on this type of engine, which is used for off-highway vehicles such as forklift trucks and earth-moving machines. The coupling of a free-piston engine to a linear electric generator is also being investigated, amongst others by researchers at University of West Virginia [26]. The main applications of the units are hybrid-electric vehicles.

2.5 Free Piston Engine Concept

The term free piston engine or linear engine comes from the purely linear piston motion that is not restricted by a crank mechanism. It means that free piston engine is crank less internal combustion engine which the piston motion of the free piston engine is not controlled by the crankshaft like the conventional engine. The numbers of variations of free piston engine configuration such as single piston free piston engine and many others configuration exist but the main parts of the engine are a combustion cylinder, a bounce chamber cylinder and a linear electric machine. Figure 2.5 and show the design of free piston engine basic concept [27].

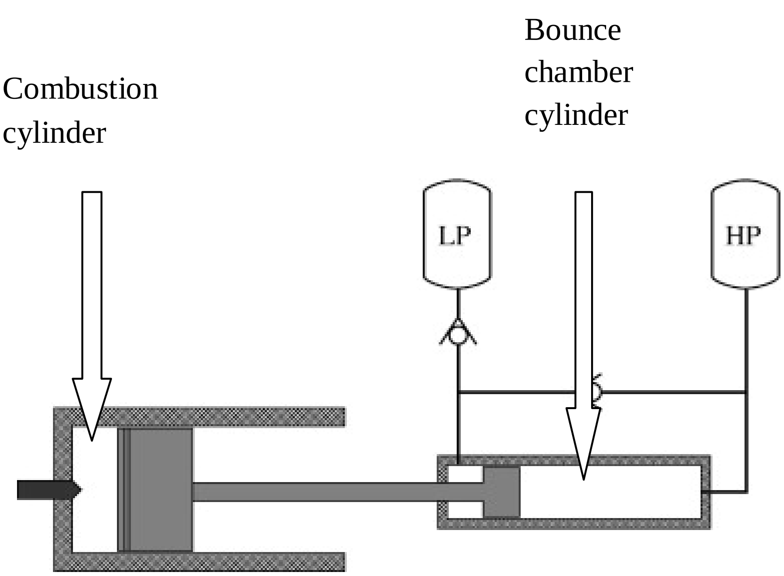


Figure 2.5 Basic Concept of Free Piston Engine [28]

2.6 Free Piston Engine Configuration

Free-piston engines are usually divided into three categories based on the piston/cylinder configuration. A fourth category, free-piston gas generators, identifies engines where the load is extracted purely from an exhaust turbine and not from a load device mechanically coupled to the engine.

2.6.1 Single Piston

A single piston free-piston engine is shown in Figure 2.6. This engine essentially consists of three parts: a combustion cylinder, a load device, and a rebound device to store the energy required to compress the next cylinder charge. In the engine shown in the figure the hydraulic cylinder serves as both load and rebound device, whereas in other designs these may be two individual devices, for example an electric generator and a gas filled bounce chamber.

A simple design with high controllability is the main strength of the single piston design compared to the other free-piston engine configurations. The rebound device may give the opportunity to accurately control the amount of energy put into the compression process and thereby regulating the compression ratio and stroke length [28].

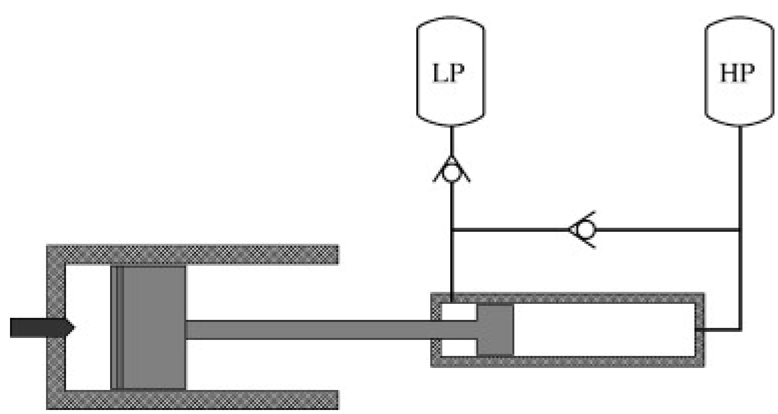


Figure 2.6 A Single Piston Hydraulic Free Piston Engine [28]

2.7 Free Piston Engine Application

Since the free-piston engine was first developed around 1930 a number of different designs of free piston engine have been proposed using the free-piston concept. The majority of these were, however, not commercially successful. This section gives an overview of known free-piston engine developments, with an emphasis on engines where experimental results or operational performance data have been reported. It should be noted that in addition to these a high number of patents describing free-piston machinery exist, where the actual development of the engines has not been reported.

2.7.1 Free Piston Air Compressors

The original free-piston configuration proposed by Pescara was an air compressor, and these machines proved to possess some very attractive features. Despite the large research efforts on the free-piston gas generator during 1940–1960, the air compressor is by many considered to be the only really successful free-piston engine concept. The excellent performance of the air compressors was a strong contributor to the later significant research efforts put into the free-piston gas generator.

These engines were of the opposed piston type, making them vibration-free. Farmer [23] discusses how the interaction between the bounce chamber and the compressor cylinders controls the compression energy delivered to the combustion cylinder and makes the engine essentially self-regulating.

Despite its apparent good performance, the free-piston air compressor did not gain widespread commercial success. No reports of serious lacks or flaws in the concept explaining this can be found, except that the free-piston air compressor had a narrow output range. Most reports are, in fact, of the opposite opinion, such as McMullen and Payne who state that the free-piston air compressor has proved ‘reliable and efficient under all conditions of service’.

Beachley and Fronczak [28] evaluate the lack of success of the free-piston air compressor and present some possible factors, including that

i. Stationary installations tended to use cheaper electric motors to drive compressors.
ii. Demand for variable power output disfavored the narrow-output free-piston air compressor for portable applications.
iii. Low fuel prices and a limited market for such applications discouraged the development of such unconventional design.

2.7.2 Hydraulic Free Piston Engine

Many of the modern approaches in free-piston engine technology are hydraulic engines, in which the combustion piston is directly coupled to a hydraulic pump cylinder. A number of projects are ongoing, both within academia and in industry. Most of these units are aimed at off-highway vehicles such as forklift trucks and earth-moving machinery and, consequently, most developments are of small size (typically 30–50 kW). Such vehicles typically have high hydraulic loads from vehicle accessories and propulsion, and they are commonly powered by a conventional diesel engine coupled to a hydraulic pump.

Hydraulic free-piston engines may apply a hydraulically driven rebound device, using part of the produced hydraulic energy to return the piston, or a bounce chamber. A number of prototypes have been developed in recent years and experimental results from these are currently being reported. The reports show generally good fuel economy and very good performance at part load. [27] [28]

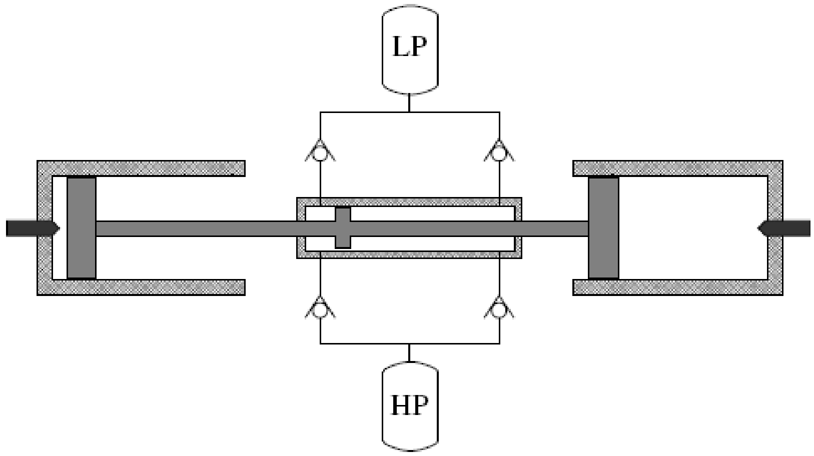


Figure 2.7 Dual Pistons Hydraulic Engine [28]

2.7.3 Free Piston Engine Generators

The free-piston engine generator consists of a free-piston engine coupled to a linear electric machine. Such technology is currently being explored by a number of research groups worldwide. The high efficiencies of electrical machinery, along with flexibility and controllability, make this an interesting concept. A driving force behind the interest in free-piston engine generators is the automotive industry's increasing interest in hybrid-electric vehicle technology.

Free-piston engine generator designs of both single piston and dual piston types have been reported. A bounce chamber may be applied in the single piston engine, but the use of the electric machine as rebound device has also been proposed. Implementation of appropriate power electronics may allow the use of the electric machine in motoring mode to aid engine control and for starting [27] [28].

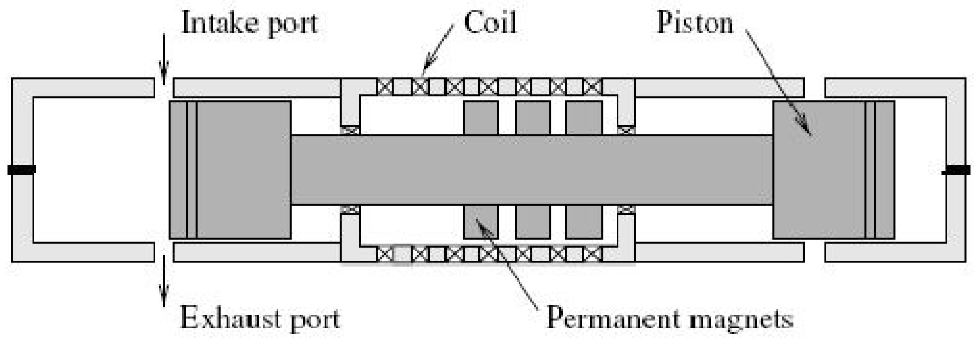


Figure 2.8 Illustration of the Free Piston Engine Generator [28]

2.8 Spring

Springs are constructional elements designed to retain and accumulate mechanical energy, working on the principle of flexible deformation of material. Springs belong to the most loaded machine components and are usually used as [35]:

i. Energy absorbers for drives and reciprocating devices
ii. Interceptors of static and dynamic forces
iii. Elements to create force joints
iv. Shock absorbers in anti-vibration protection
v. Devices for controlling and measuring of forces

Spring function is evaluated according to the course and extent of its deformation depending on its load. Figure 2.9 shows spring characteristic curve [35].

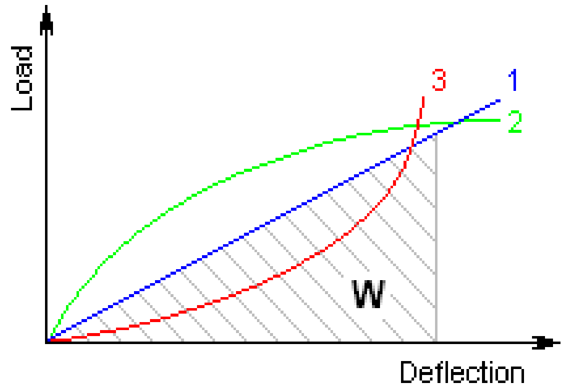


Figure 2.9 Spring Characteristic Curve [35]

Based on the deformation pattern, springs can be divided into the following three types [35]:

1. springs with linear characteristics
2. springs with digressive characteristics
3. springs with progressive characteristics

The *W* area under the spring characteristic curve represents the deformation work (energy) of a spring performed by the spring during its loading. Deformation energy of springs subjected to compression is specified by the formula [35]:

$$W = \int_{o}^{s} F.ds \qquad (2.4)$$

Where:

W is work in unit of (J, ft.lb)

The basic quantity specifying the spring functionality is its stiffness (spring constant). Spring constant **k** specifies the intensity of load (force) which causes unit deformation (shift) of the spring [35].

$$k = d\,F/\,Ds \tag{2.5}$$

Where:

K is spring constant in unit (N/mm, lb/in)

The spring with linear characteristics have invariable spring constant; other springs have variable spring constant. Springs are mounted with initial stress, i.e. in the state when the spring is subjected to the minimum working load. In view of spring function, there are four basic states of springs as shown in Table 2.1. The mentioned indexes are used to specify individual parameter of the spring related to the given state of the spring as shown in the Figure 2.10[35].

Table 2.1 Four Basic States of Springs [35]

State of the spring	**Description of states of a spring**	**Index**
Free	The spring is not loaded	0
Preloaded	The spring is exposed to minimum operational loading	1
Fully loaded	The spring is exposed to maximum operational loading	8
Limiting	The spring is exposed to the limit load – given by the material strength or design limitations (e.g. compression of the coil spring to bring all coils into contact).	9

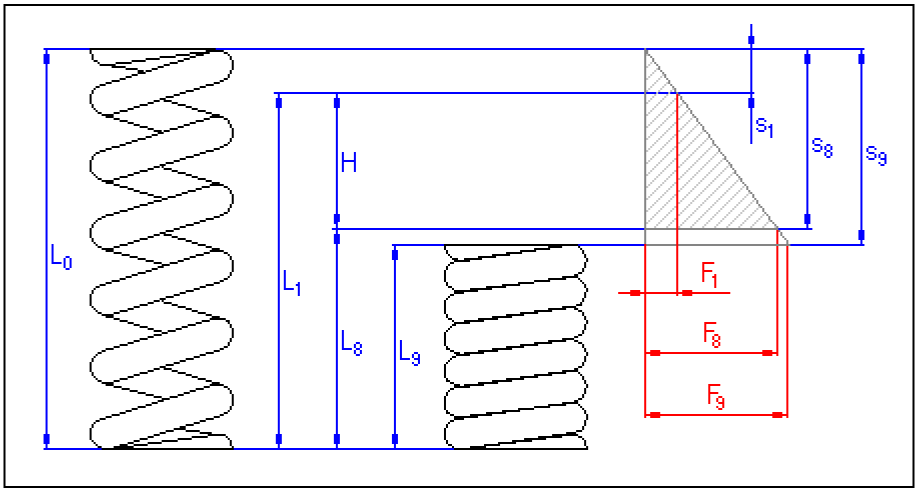


Figure 2.10 The States of the Spring According to Index of Basic State of Springs [35]

2.8.1 Helical Cylindrical Compression Spring

Springs of cylindrical shape made of helically coiled wires, with constant clearance between the active coils, able to absorb external counter-acting forces applied against each other in their axis. Springs with wire diameter up to approx. 16 mm are usually cold wound. Hot forming shall be used for the production of heavily loaded springs of greater sizes with a diameter of the over 10 mm. Compression springs are usually made of wires and rods of round section. Figure 2.11 shows the picture of helical cylindrical compression spring. Specific properties of the spring are described below [35]:

i. Suitable for low and medium load forces
ii. Linear working characteristics
iii. Relatively low spring constant
iv. Easy mounting and dismantling
v. Low production costs

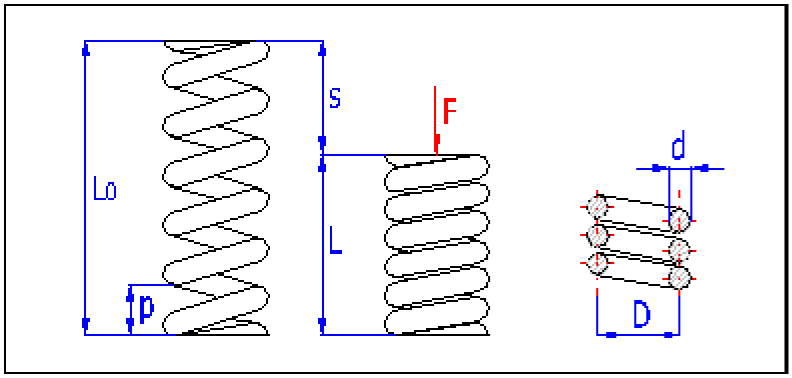


Figure 2.11 Helical Cylindrical Compression Spring [35]

2.8.2 Basic Equation for Spring of Round Wire

The basic equation for spring of round wire is shown below [35].

$$S = 8.\ F.\ n\ .\ D^3\ /\ G\ .\ d^4 \tag{2.6}$$

$$\tau = K_s\ .\ 8.\ F.\ D\ /\ \Pi.\ d^3 \tag{2.7}$$

$$k = G.\ d^4\ /\ 8.\ n.\ D^3 \tag{2.8}$$

$$K_s = f\ (D/\ d) \tag{2.9}$$

Where:

c is spring index (c=D/d; c=D/b)

b is wire width (mm, in)

d is wire diameter (mm, in)

D is mean spring diameter (mm, in)

F is loading of spring (N, lb)

G is modulus of elasticity in shear (MPa, psi)

h is wire height (mm, in)

k is spring constant (N/mm, lb/in)

K_s is curvature correction factor

L_0 is free spring length (mm, in)

L_S is solid length (mm, in)

n is number of active coils

p is pitch between coils (mm, in)

s is spring deflection (mm, in)

t is torsional stress of the spring material (MPa, psi)

2.8.3 Curvature Correction Factor

The coil bending causes additional bending stresses in coil springs. Therefore the calculation uses the correction coefficient to correct the tension. The shear correction factor is used for static loading only where is shown in Equation 2.6. Wahl factor which is shear and curvature corrections used for fatigue loading shown in Equation 2.7 [35].

$$Ks = 1 + 0.5/ C \tag{2.6}$$

$$Kw = (4C – 1/ 4C – 4) + (0.615/ C) \tag{2.7}$$

2.8.4 Design of Spring End

In case of compression springs, several various designs of spring ends are used. These differ in numbers of ends and machined coils and designs of supporting surfaces of the springs. Figure 2.12 shows the common types of spring ends designs [35].

i. End coils are edge coils of the spring, co-axial with the active coils, whose angle pitch does not change during functional deformation of the spring. End coils create a supporting surface for the spring and with compression springs one, end coil is usually used at both ends of the spring.

ii. Ground coils are edge coils of the spring, machined to a flat surface perpendicular to the spring axis. Usually machined from three-fourths of half of the end coil up to its free end. Machined coils are commonly used only with springs with diameters of wires d > 1 mm.

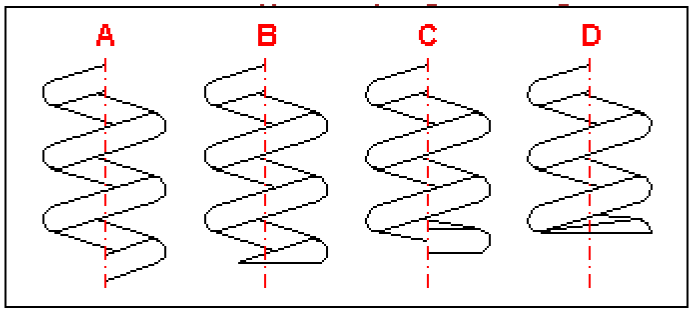


Figure 2.12 Common Types of Spring End Designs [35]

A. Open ends not ground: the edge coil is not bent to the next one, the supporting surface is unmachined
B. Open ends ground: the edge coil is not bent to the next one, the supporting surface is machined to a flat end perpendicular to the spring axis
C. Closed ends not ground: the edge coil is bent to the next one (it usually adjoins its free end), the supporting surface is unmachined
D. Closed ends ground: the edge coil is bent to the next one, the supporting surface of the spring is machined

2.8.5 Check of Buckling

In case of compression springs, it is always necessary to check its protection against side deflection. The check is performed by comparison of the maximum working deformation of the spring with the permitted deformation. The value of the permitted deformation is determined empirically for the given slenderness ratio of the spring L_0/D and the type of seating of the spring. Generally, the risk of possible side deflection increases with an increasing value of the slenderness ratio and increasing value of the working compression of the spring. The manner of seating of the spring has a significant effect on its possible side deflection. A spring which cannot be designed as secured against side deflection is usually installed on a pin or inside a sleeve. If there is a danger of damage of the spring due to friction, the spring can be divided into several shorter springs arranged in series. Figure 2.13 shows the seating of the spring whereas Figure 2.14 shows the curves of permitted deformation according to the type of seating of the spring [35].

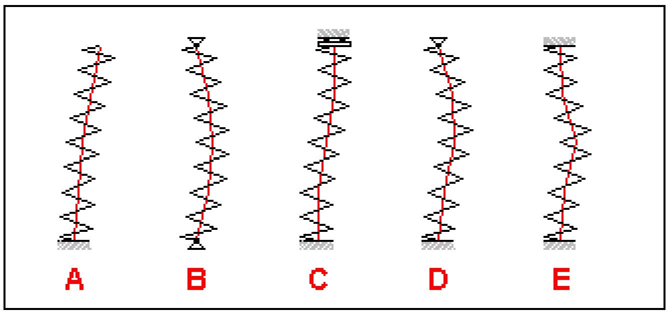


Figure 2.13 Seating Types of the Springs [35]

A. Fixed - free ends
B. Pinned - pinned ends
C. Clamped - clamped ends with lateral restraint
D. Clamped - pinned ends
E. Clamped - clamped ends without lateral restraint

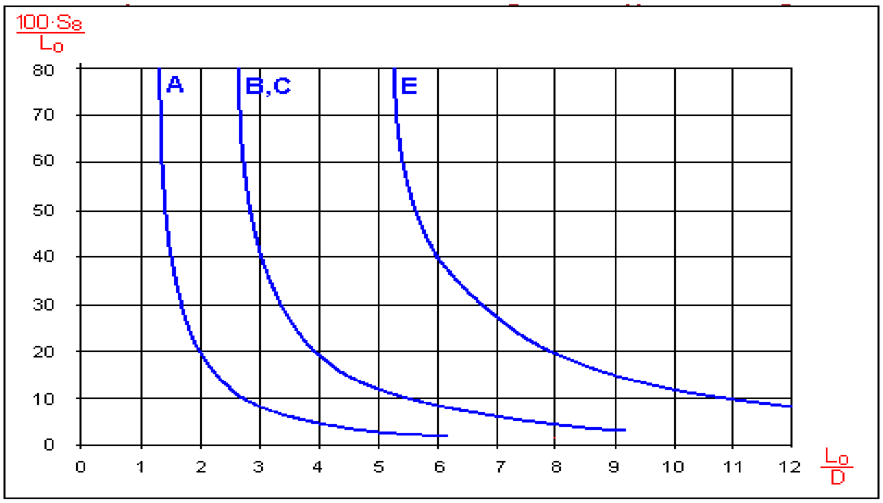


Figure 2.14 Curves of Permitted Deformation According to the Type of Seating of the Spring [35]

CHAPTER 3

METHODOLOGY

The methodology of this project is briefly shown in the flowchart of Figure 3.1. The work of this project is done systematically according to the flowchart start from the beginning of the project until the end. The works involve in this project include title confirmation of the final year project with supervisor, do literature study to understand more about information that related with this final year project, determine the required equipment and software that is needed to accomplish this project, collect the required data that is needed in the spring design process, using Mitcalc- compression spring software 1.12 in the spring design process, select the best design of the spring for the linear engine, design the linear engine using Solidworks 2005 software based on the spring design that have been selected and prepare the final report of the works that have been carried out from the beginning of the project until the end for the purpose of the documentation and future reference.

The methodology applied in the implementation of this project was as follows:

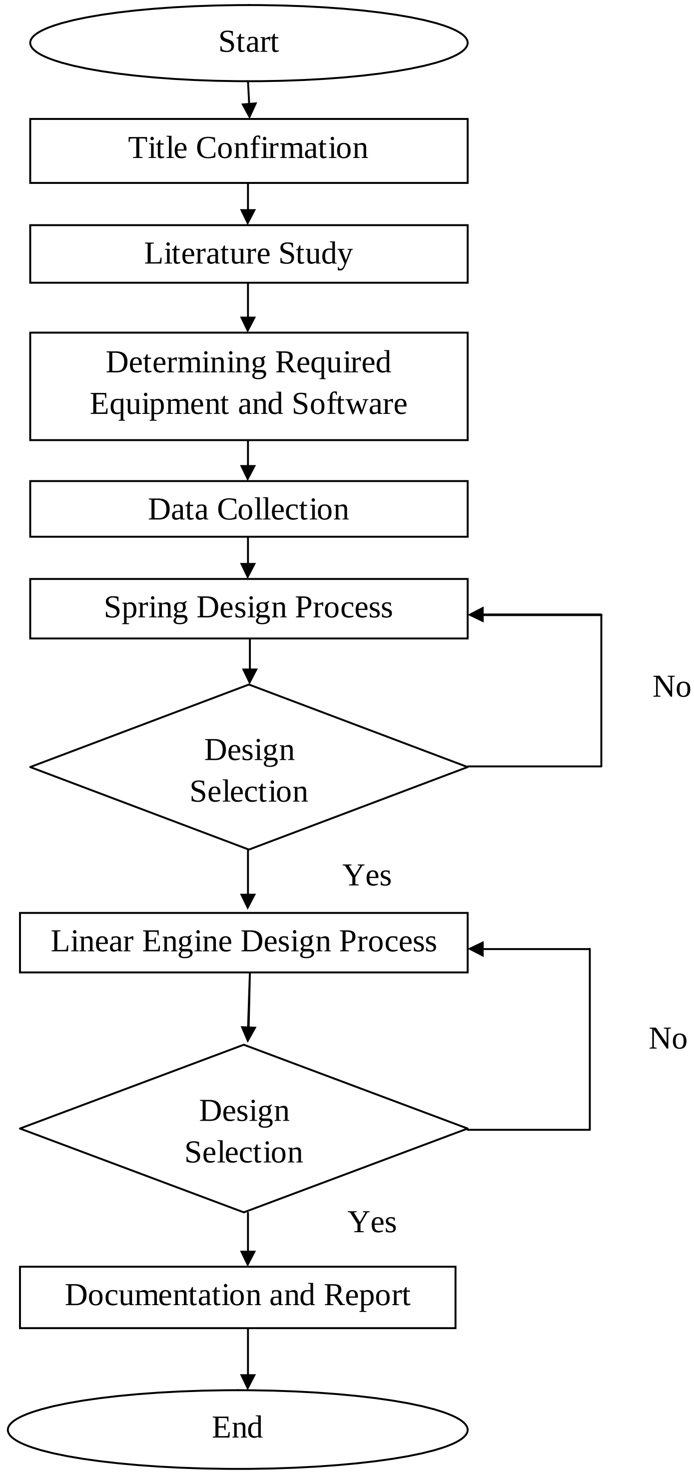


Figure 3.1 Flowchart of Project Implementation

3.1 Title Confirmation

This project begins with receive and confirm the final year project title which is entitled 'Design of Two Stroke SI Linear Engine with Spring Mechanism' with the co-supervisor. Then co-supervisor gives the explanation about the project background, project objectives, project scopes, overview the related research of linear engine. This information is very useful and important to provide the understanding about the works that need to be done from the beginning until the end of the project. The duration of the final year project is about 2 semesters which is equivalent to 1 year. The Gantt chart is constructed to plan the work that need to be done throughout the whole 2 semesters. This very important to ensure that the project is done and accomplished within the required time limit. The works that have been done in this project for the whole 2 semester which is final year project 1 for the semester 1 and final year project 2 for semester 2 is shown in the Gantt chart at the appendix section.

3.2 Literature Study

During this step, the study is carried out about the two stroke engine, linear engine and process of the spring design through other thesis, technical paper, web site, journal, and books. The information about the two stroke engine, linear engine and the process of spring design that related to this project is collected within the 1st semester which is called the final year project 1 whereas the 2nd semester all the works is focused more onto the process of the spring design and linear engine design. The review of the other researcher works, technical paper and journal is important to provide the understanding and information about the linear engine that have been produced in the world today. Besides, there are several books that provide the information about the process of spring design. These books will give the understanding in term of spring design process.

3.3 Determining Required Equipment and Software

Before the process of spring design and linear engine design, the required equipment and software are determined. The right equipment and software are used to facilitate the student in order to accomplish the final year project. The equipment used in this final year project is vernier caliper. The vernier caliper is used to measure the input parameters that are needed in the process of the linear engine design. This final year project does not require a lot of equipments because all the processes of the spring design and linear engine design is done by using computer and this project also do not involved the fabrication of the linear engine. The spring is designed with the help of spring design software and linear engine is designed by using CAD software. Figure 3.2 below shows the vernier caliper that is used in order to measure the parameters that is needed in the process of the spring and linear engine design.

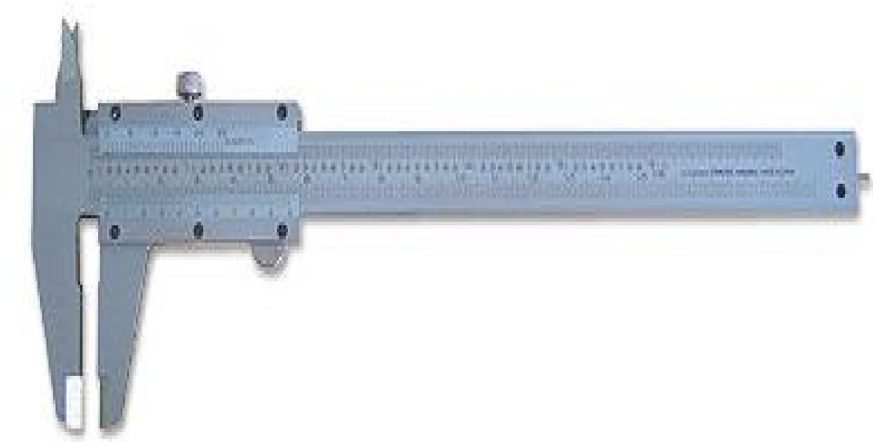


Figure 3.2 The Vernier Caliper

The Mitcalc- helical compression spring software 1.12 is used in the process of the spring design. The process of the spring design actually is a very long process of calculations. There are many springs that have been design can be used with the linear engine but only one spring is required in this project. The best spring design is selected based on the specific design requirement of this project. The software used in this project is very helpful because it shorten the time required in the process of the spring design. The Figure 3.3 shows the Mitcalc- helical compression spring software 1.12.

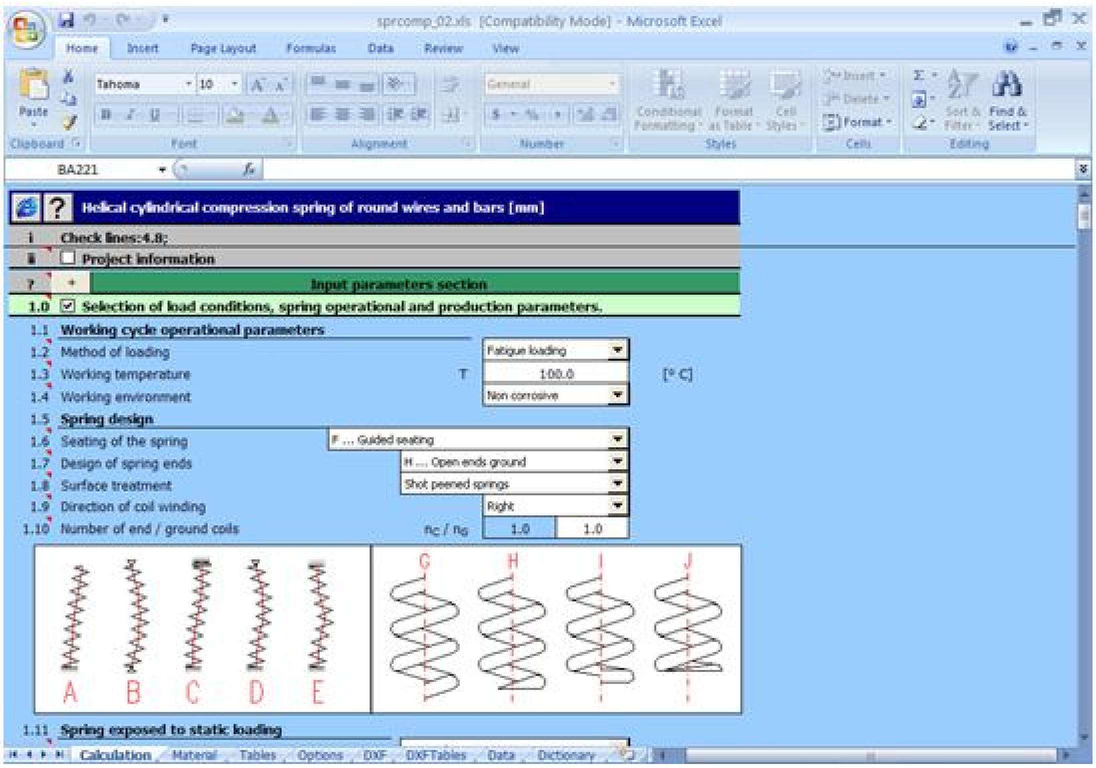


Figure 3.3 Mitcalc- Helical Compression Spring Software Versions 1.12

After the process of the spring design finished, the best spring design for the linear engine is selected and the next step is the process of linear engine design based on the dimension of spring that has been designed. During this process, the Solidworks software 2005 is used to aid the process of the linear engine design. Figure 3.4 shows the Solidworks software 2005 that have been used in the process of the linear engine design.

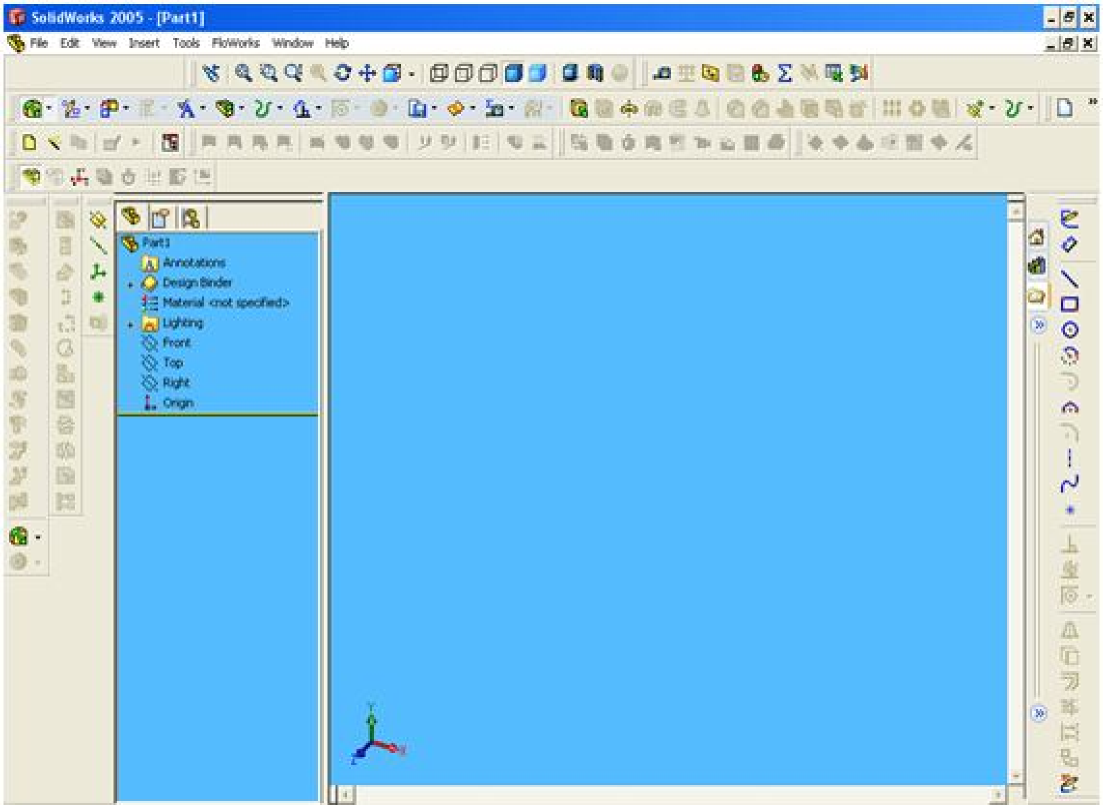


Figure 3.4 Solidworks Software Version 2005

3.4 Data Collection

The data for this project is required in the process of the spring design and linear engine design. The IMEP data of the linear engine is required in the process of the spring design which is obtained from the GT-Power software. The engine dimension data of Back Pack Brush Cutter (BG-328) engine is needed in the process of the linear engine design. The dimension of the linear engine design is measured by using vernier caliper. This project is entitled 'Design of Two Stroke SI Linear Engine with Spring Mechanism. The design of the linear engine of this project is actually modified from the small two stroke engine of the conventional one and because of that the engine dimensions is required in the process of the linear engine design. The two stroke engine that is used in this project is small two stroke engine of Back Pack Brush Cutter (BG-328). The engine is a two stroke engine, single cylinder, forced air-cooled, used

Gasoline fuel. Figure 3.5 shows the picture of the engine. Table 3.1 shows the engine specifications of the Back Pack Brush Cutter (BG-328).

Figure 3.5 The Engine Picture of Back Pack Brush Cutter (BG-328)

Table 3.1 Engine Specifications of Back Pack Brush Cutter (BG-328)

Model	BG-328
Type	2 cycles, Single cylinder, Forced air-cooled, Gasoline engine.
Displacement	30.5 cc
Max. Output	0.81k W/ 6000rpm
Carburetor	Float type
Ignition system	IC ignition (Solid state)
Ignition plug	BM-7A or CHAMPION CJ6
Fuel	Mixed fuel of Gasoline and 2 cycle oil at 25:1

Fuel tank capacity	1.2 liters

The parameter used in the process of the spring design is IMEP of the linear engine which is obtained from the GT-Power software. This work actually is done by my colleague of final year project, Mr. Mohd Nordin B. Zazalli. He has done a thesis entitled 'Prediction Studies for the Performance of a Single Cylinder High Speed SI Linear Engine' [36].

3.5 Spring Design Process

The spring design process is a very crucial process in this project. It is because the design of the linear engine is derived from this spring design. As mentioned before, the linear engine design of this project is modified from the conventional two stroke engine. The modification is made at the crankcase part of the conventional two stroke engine so as to accommodate a place for the spring when the spring coupled with the engine later. There are a lot of calculations involved in the process of the spring design. So, the process of the spring design is done with the aid of the Mitcalc- helical compression spring software version 1.12 to avoid mistake during the calculations. The procedures and the equations of spring design calculation used in the software are same as many spring design books available in the world today. The task of spring design cannot be solved directly and allows considerable freedom in options of the design, dimensions or loading of the spring. There are many springs of various designs and dimensions may meet requirements of the desired input parameters of the task in this project. Therefore, it is necessary to proceed iteratively and successively evaluate individual designs of the spring by creating a database of spring design based on the calculations that have been done before in the form of table. After that, from the table the spring of suitable dimension with the linear engine is determined. The design procedure of spring by using the software is given in the following items and this items need to be filled up with the required parameters of spring that need to be produced.

i. Set operational parameters of the working cycle (manner of loading, temperature and aggressivity of the working environment).
ii. Select production and installation properties of the spring.
iii. Select the corresponding mode of loading and set the desired level of safety.
iv. In case of fatigue loading of the spring, set the mode of fatigue loading, desired service life and level of safety.
v. Choose an adequate processing of the spring.
vi. According to the recommended scope of use, select the material of the spring.
vii. Enter the desired parameters of the working cycle (loading, length and stroke of the spring).
viii. Set the necessary filters and marginal conditions of the spring design.
ix. Select the manner of classification of results and press the button for initiation of the design calculation.
x. Select a suitable solution from the table of optimum design.
xi. Check parameters of the designed spring result in chapter 4 of this software.

Actually, there are 2 stages of spring design process in this project. For stage 1 where the process of the spring design is called preliminary spring design process, the spring is designed so as the spring can fairly deflect when subjected with different forces that produced from operating linear engine. The spring fairly deflect mean that the spring is not become too stiff when subjected with force of low magnitude and the spring do not easily deflect when subjected with force of high magnitude. The springs that have been designed from the preliminary spring design process is then go to the next step where the spring will be tested with forces that comes from operating linear engine.

3.5.1 Selection of Load Conditions, Spring Operational and Production Parameters

The selection of load conditions, spring operational and production parameters is the section 1 of calculation sheet in the software. In this stage, enter basic input parameters characterizing the manner and mode of loading, design and method of seating the spring and parameters of the working environment to the section 1 of the software. The load conditions, operational and production parameters for the process of the spring design is shown in the Table 3.2.

Table 3.2 The Load Conditions, Operational and Production Parameters for the Process of the Spring Design

Method of loading	Fatigue loading
Working temperature, T (^oC)	100
Working environment	Non-corrosive
Seating of the spring	Guided seating
Design of spring ends	Open ends ground
Surface treatment of the spring	Shot peened springs
Direction of coil winding	Right
Number of end/ ground coils, n_C/n_G	n_C =1, n_G=1
Operational mode of a spring exposed to static loading	Heavy duty service
Desired level of safety of a spring exposed to static loading, S_s	S_s = 1.25
Method of stress curvature correction for spring exposed to static loading	Correction by Wahl
Operational loading mode of a spring exposed to fatigue loading	Continuous loading
Desired spring service life in thousands of	N= infinite life

cycles, N	
Desired level of safety of a spring exposed to fatigue loading, S_f	S_f= 1.05
Method of stress curvature correction for spring exposed to fatigue loading	Correction by Wahl

3.5.2 Option of Spring Material

The option of spring material is section 2 of calculation sheet in the software. This paragraph can be used for selection of the spring material. Immediately after selection of material in the list, all information necessary for the design and calculation of the spring is displayed in the table of this software. If more detailed information on the selected material, define or modify the material for the spring, switch over to the material sheet "Material". The information of the field of use, mechanical and physical properties, and strength characteristics of material for the spring design is shown in the Table 3.3.

Table 3.3 The Information Related to the Material Used in the Spring Design

Production method	Hot formed springs
Spring material	Chrome-vanadium alloy steel wire SAE 6150
Field of use of the selected material	
Suitability for fatigue load	Excellent
Relative strength	High
Corrosion resistance	Good

Maximum operational temperature, (^{o}C)	220
Mechanical and physical properties of the material	
Modulus of elasticity in shear, G_{20}	78500
Modulus of elasticity at operational temperature, G	76538
Density, ρ	7850
Strength characteristics of the material	
Ultimate tensile strength, S_u	1350
Permissible torsional stress, τ_A	954
Ultimate fatigue strength in shear, τ_e	477
Fatigue strength by finite life, τ_f	477

3.5.3 Spring Design Parameter

The spring design parameter is section 3 of calculation sheet in the software. At this stage, all the important input parameters to design a spring are needed to be filled up at the section of “Spring Design” in this software environment. The included input parameters are maximum and minimum working loading, fully loaded spring length and required spring working stroke. At this stage also, it is necessary to specify various filters and marginal conditions of the design calculation. Their setting may significantly affect the course of the spring design and determined the speed, accuracy and quality of the design, the scope and number of suitable solutions and a qualitative standard for evaluation of the best designs. Furthermore at this stage, the spring design is sorted in the optimum design table based on the option of solution that has been chosen. Table 3.4 shows the input parameters of spring design. Table 3.5 shows the filters of the designed solution and options of solutions of spring design calculations. Figure 3.6 shows the picture of the spring that has been applied input parameters.

Table 3.4 Input Parameters of the Spring Design

Maximum working loading, F_8 (N)	1570
Minimum working loading, F_1 (N)	965.4
Required spring working stroke, H (mm)	30

Table 3.5 The Filters of the Designed Solution

Filters of the designed solution	
Permissible division of the number of active coils	1/4
Permissible exceeding of spring limit dimensions	0.0%
Perform check of buckling	Yes

Perform check of limit working length	Yes
Keep to the required level of safety with the strength check	Yes
Quality criterion	Deviation from desired dimensions
Number of design iteration	medium
Options of solutions	
Sort design result by	Qualities of solutions

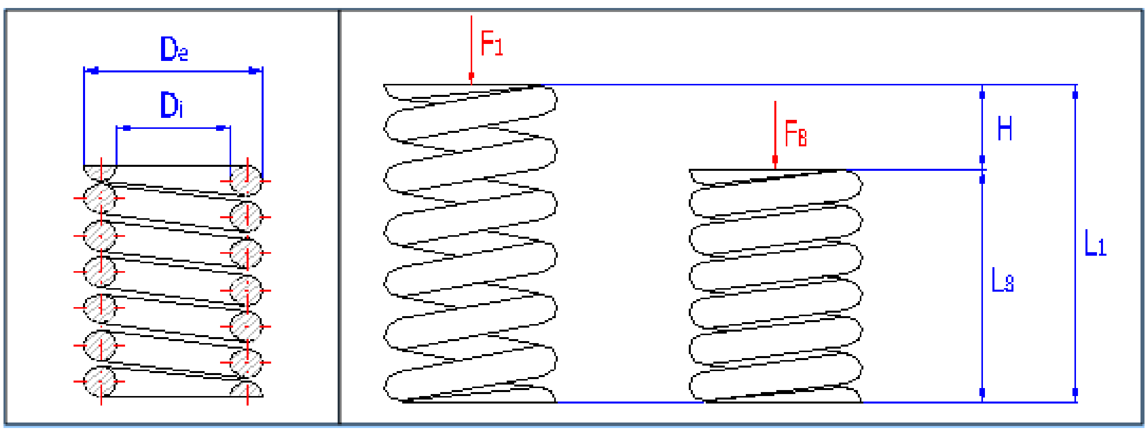


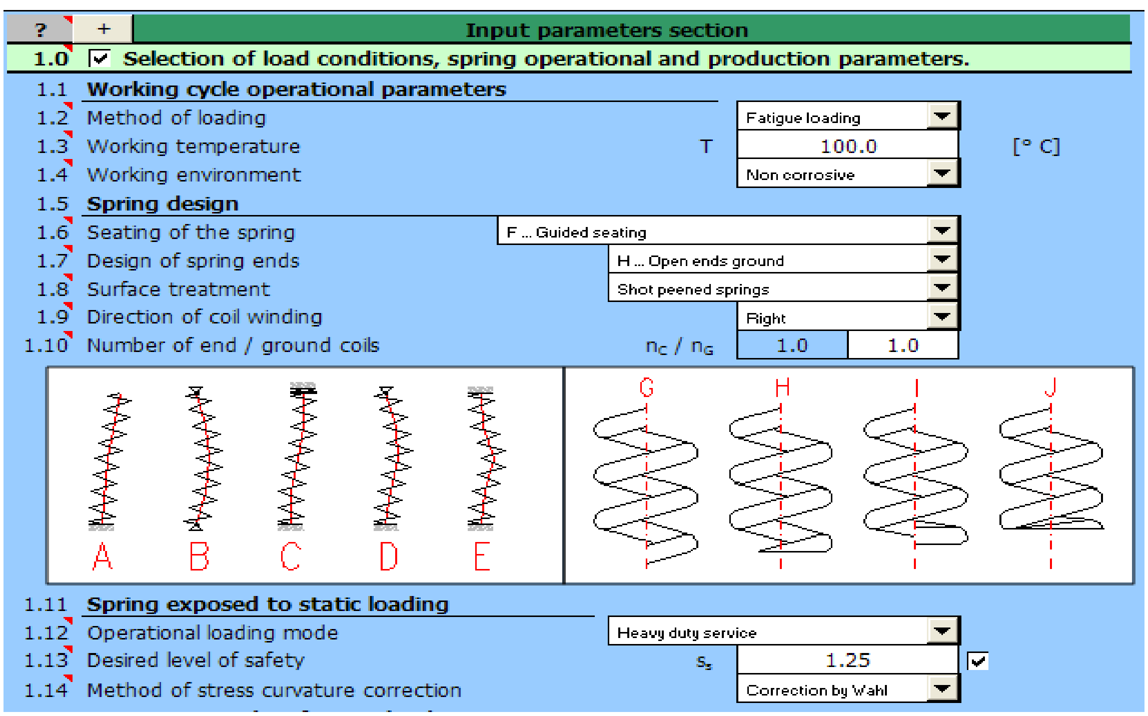
Figure 3.6 Input Parameters Applied to the Spring [35]

3.5.4 Summarized List of Designed Spring Parameter

The summarized list of designed spring parameter is section 4 of calculation sheet in the software. At this stage, all necessary parameters describing the designed spring are shown in this paragraph for the given loading and dimensions of the spring. This section actually displays all the parameters related to the spring that have been designed based on the input data that have been entered. The process of spring design can be done directly at this section by substituting the values of spring parameters that need to be designed. The process of spring design in this project is done by using this section. The process of spring design in this project used the technique called trial and error where the values of spring parameters that need to be designed is substituted in this section over and over again until a optimum spring design is obtained. The optimum spring design is a spring that meets the requirement of spring design for this project as stated in section 3.6.

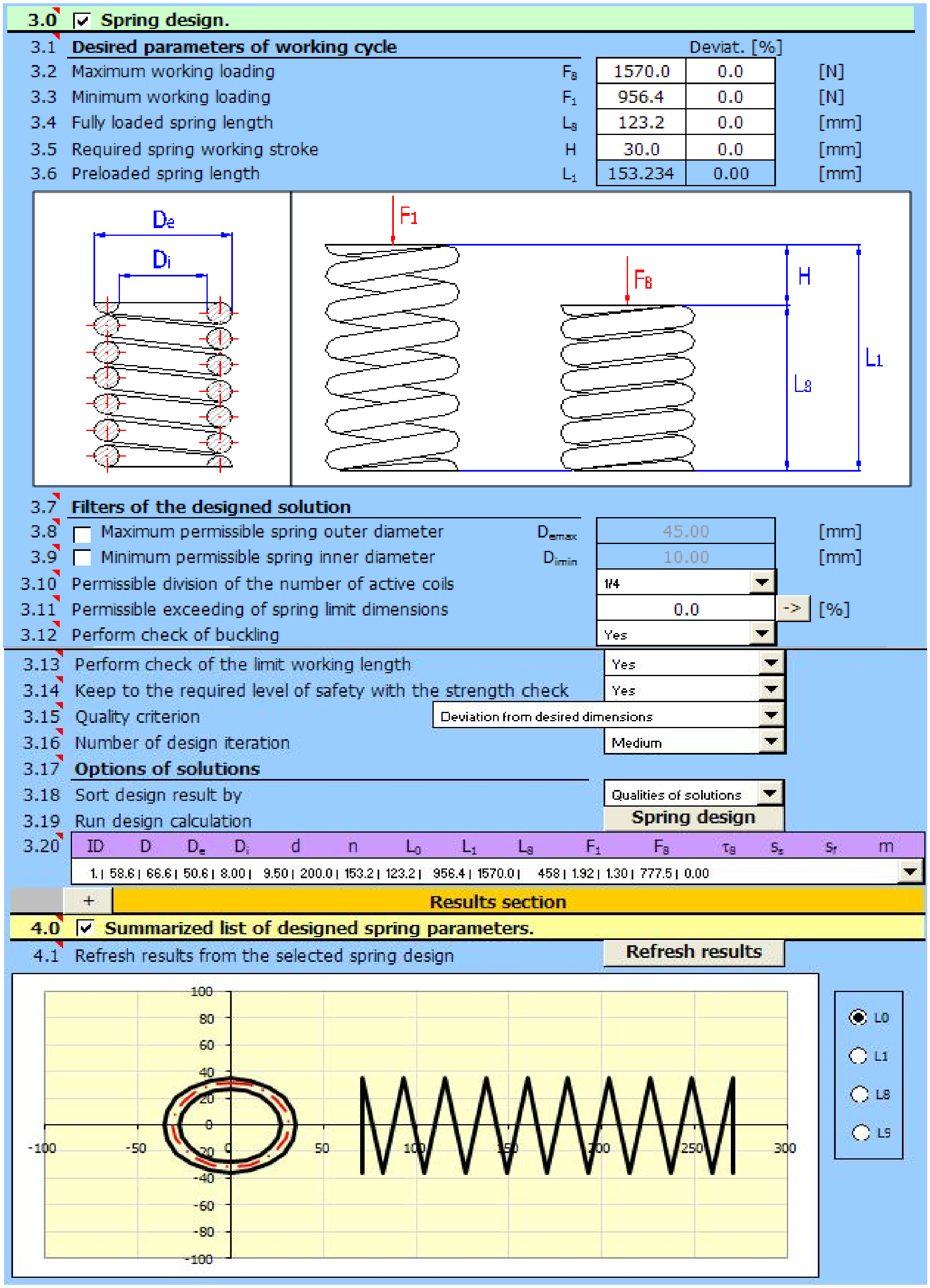
3.5.5 The Calculation Sheet of the Mitcalc – Helical Compression Spring Software Version 1.12

The calculation sheet of the software is shown in the Figure 3.7. This gives a detailed description of what have been discussed in the section 3.6.1, 3.6.2, 3.6.3 and lastly 3.6.4 of this methodology. The Figure 3.7 gives a clear description of where the location for entering input parameters of designed spring of interest that have been discussed in the section 3.6.1, 3.6.2 and 3.6.3 in this methodology.



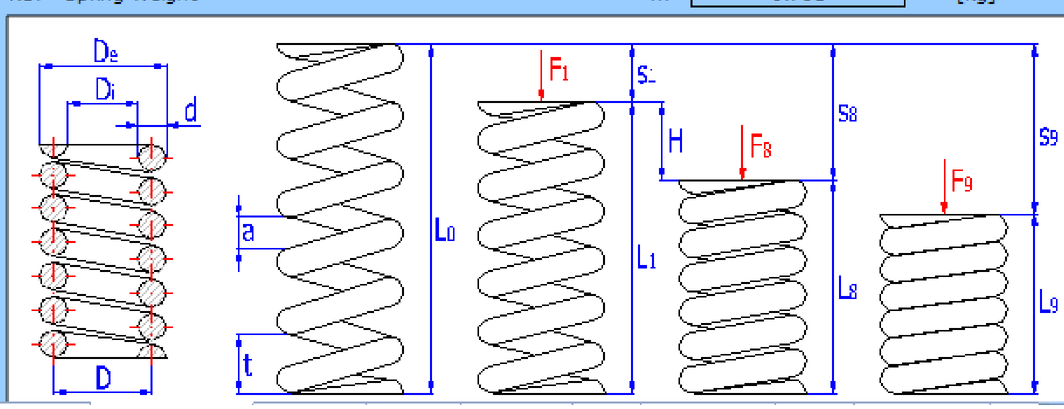
1.15	**Spring exposed to fatigue loading**			
1.16	Operational loading mode		Continuous loading	
1.17	Desired spring service life in thousands of cycles	N	Infinite life	
1.18	Desired level of safety	s_f	1.05	☑
1.19	Method of stress curvature correction		Correction by Wahl	
2.0	☑ **Options of spring material.**			
2.1	Production method :		Hot formed springs	
2.2	Spring material :		Chrome-vanadium alloy steel wire SAE 6150	
2.3	**Field of use of the selected material**			
2.4	Suitability for fatigue load		Excellent	
2.5	Relative strength		High	
2.6	Corrosion resistance		Good	
2.7	Max. operational temperature		220	[° C]
2.8	Delivered wire diameters		8 - 80	[mm]
2.9	**Mechanical and physical properties of the material**			
2.10	Modulus of elasticity in shear	G_{20}	78500	[MPa]
2.11	Modulus of elasticity at operational temperature	G	76538	[MPa]
2.12	Density	ρ	7850	[kg/m³]
2.13	**Strength characteristics of the material**			
2.14	Ultimate tensile strength	S_u	1350	[MPa]
2.15	Permissible torsional stress	τ_A	940	[MPa]
2.16	Ultimate fatigue strength in shear	τ_e	470	[MPa]
2.17	Fatigue strength by finite life	τ_f	470	[Mpa]

Figure 3.7 The Calculation Sheet for Spring Design Process in the Mitcalc - Helical Compression Spring Version 1.12 Software. [35]



(Continue of Figure 3.7)

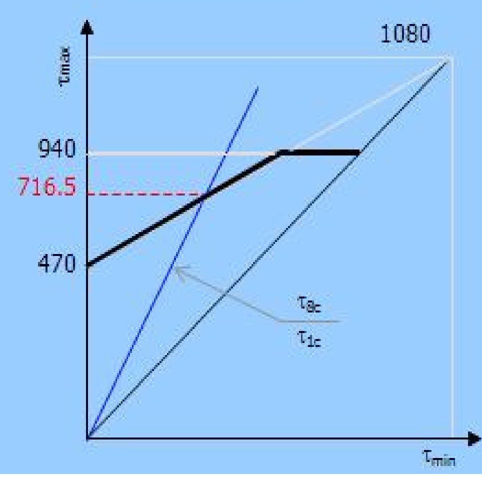
4.2	**Spring loading**				
4.3	Minimum working loading	F_1	956.40		[N]
4.4	Maximum working loading	F_8	1570.00		[N]
4.5	**Spring dimensions**				
4.6	Mean spring diameter	D	63		[mm]
4.7	Recommended limits of wire diameter	d_{min} / d_{max}	8.00	21.00	[mm]
4.8	Wire diameter	d	8		[mm]
4.9	Outer / inner spring diameter	D_e / D_i	71	55	[mm]
4.10	Spring index	c	7.88		
4.11	Number of active coils	n	9		
4.12	Recommended limits of free spring length	L_{0min} / L_{0max}	178.10	319.85	[mm]
4.13	Free spring length	L_0	199.99		[mm]
4.14	Recommended pitch limits	t_{min} / t_{max}	18.90	34.65	[mm]
4.15	Space / pitch between coils of free spring	a / t	13.33	21.33	[mm]
4.16	**Parameters of preloaded spring**				
4.17	Spring deflection	s_1	54.92		[mm]
4.18	Spring length	L_1	145.07		[mm]
4.19	Spring stress	τ_1	299.67		[MPa]
4.20	**Parameters of fully loaded spring**				
4.21	Spring deflection	s_8	90.16		[mm]
4.22	Spring length	L_8	109.83		[mm]
4.23	Spring working stroke	H	35.24		[mm]
4.24	Spring stress	τ_8	491.94		[MPa]
4.25	**Parameters of spring limit state**				
4.26	Theoretic spring limit loading	F_9	2089.43		[N]
4.27	Theoretic spring deflection / length	s_9 / L_9	119.99	80.00	[mm]
4.28	Teoretic stress	τ_9	654.69		[MPa]
4.29	Sum of min. permissible spaces between active coil	s_{amin}	25.560		[mm]
4.30	Minimum spring limit length	L_{minF}	105.56		[mm]
4.31	**Spring mechanical and physical properties**				
4.32	Spring constant	k	17.41		[N/mm]
4.33	Spring deformation energy	W_8	70.78		[J]
4.34	Critical spring speed	v_k	4.70		[m/s]
4.35	Natural spring frequency	f	78.70		[Hz]
4.36	Developed wire length	l	2016		[mm]
4.37	Spring weight	m	0.795		[kg]



(Continue of Figure 3.7)

4.38	**Spring strength check**			
4.39	Curvature correction factor	K_s	1.0635	
4.40	Corrected stress of fully loaded spring	τ_{8C}	523.17	[MPa]
4.41	Permissible torsional stress	τ_A	940	[MPa]
4.42	Level of safety		1.797	
4.43	**Check of buckling**			
4.44	Permissible / actual max. working compression of spring		100 / 45.08	[%]
4.45	**Strength check of a spring exposed to fatigue loading**			
4.46	Curvature correction factor	K_f	1.1872	
4.47	Corrected stress of fully loaded spring	τ_{8C}	584.02	[MPa]
4.48	Fatigue strength for the given loading	τ_{max}	717	[MPa]
4.49	Level of safety		1.227	

6.0	☑ **Check of loading capacity of a spring exposed to fatigue loading.**			
6.1	Curvature correction factor	K_f	1.1872	
6.2	Corrected stress of preloaded spring	τ_{1c}	355.77	[MPa]
6.3	Corrected stress of fully loaded spring	τ_{8c}	584.02	[MPa]
6.4	Ultimate shear strength	S_{us}	1080	[MPa]
6.5	Permissible torsional stress	τ_A	940	[MPa]
6.6	Ultimate fatigue strength in shear	τ_e	470	[MPa]
6.7	Fatigue strength by finite life	τ_f	470	[Mpa]
6.8	Fatigue strength for the given loading	τ_{max}	716.5	[Mpa]
6.9	Level of safety		1.227	



(Continue of Figure 3.7)

3.5.6 Preliminary Spring Design Process

The spring design process at this stage is done by substituting directly the input parameter value of designed spring of interest at the section 4 of calculation sheet in the Mitcalc- Helical Compression Spring 1.12 software. The required value for the section 1,2 and 3 for this calculation sheet in the software is the same as the value that have been stated in the section 3.6.1, 3.6.2 and 3.6.3 of this methodology. The spring parameters that have been entered to the section 4 of the calculation sheet in the software are described in the Table 3.5.

Table 3.6 Input Parameters for Preliminary Spring Design

Minimum Working Loading, F_1 (N)	956.40
Maximum Working Loading, F_8 (N)	1570.00
Mean Spring Design, D (mm)	Integer of 63 to 38
Wire Diameter, d (mm)	8,7.5,7,6.5,6
Number of active coils	≥ 3

The linear engine that needs to be designed in this project is shown in the Figure 3.8 below. There is a force which is called minimum force when the spring is mounted to the linear engine at the time when the linear engine is not operating. The calculation for this minimum force is shown in the result and discussion section of this report. The maximum force for this spring design process is the summation of minimum force and forces taken at operating linear engine. The forces at operating linear engine for this design stage is taken from the mean pressure value of IMEP versus mean piston speed of linear engine. The equation of graph line is constructed first if to find the mean pressure. The equation of graph line is found by using polynomial interpolation. The polynomial interpolation is used instead quadratic interpolation because the equation derived from this technique has lower percent relative error compare to the quadratic interpolation. Figure 3.9 and 3.10 show the quadratic interpolation and polynomial interpolation and from the figures the line of polynomial interpolation nearly fit with the true graph line compare to the quadratic interpolation. The calculations for mean pressure and forces at operating linear engine are shown at section 4.2 of result and discussion in this report.

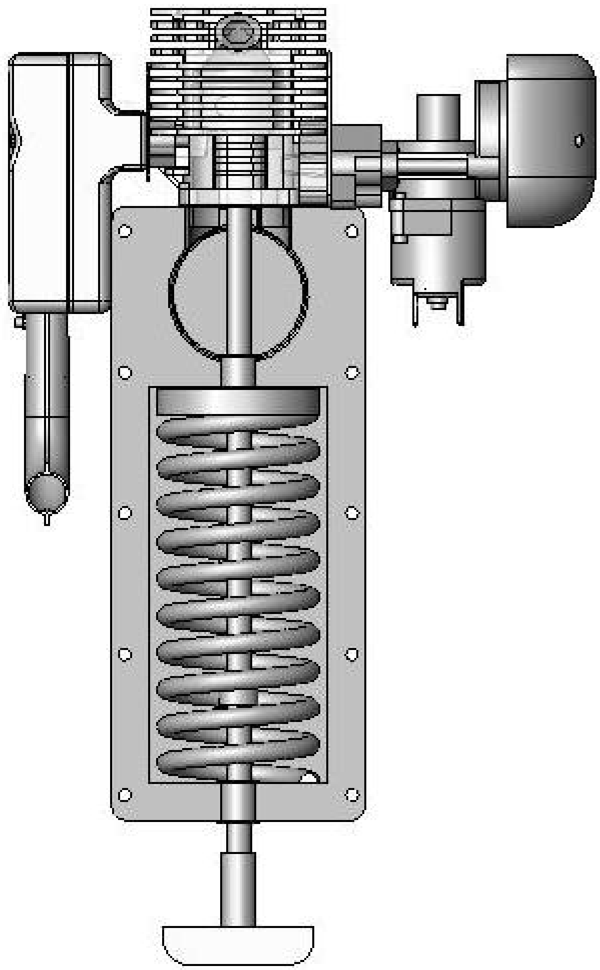


Figure 3.8 Designed Linear Engine

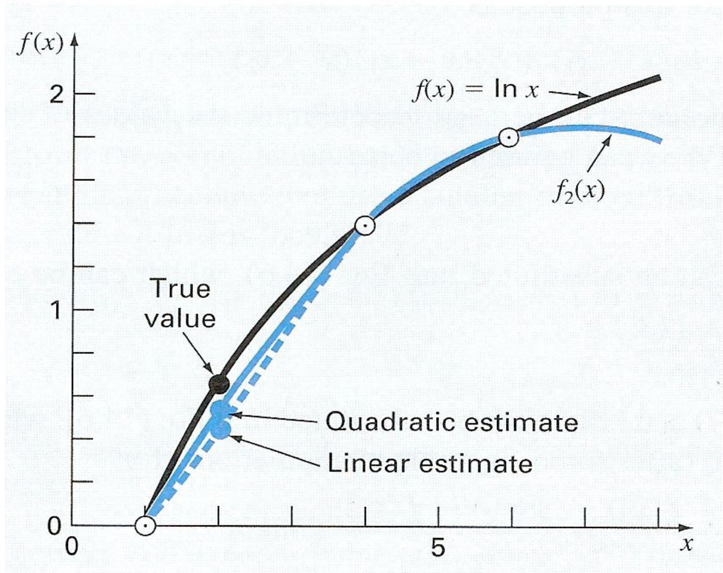


Figure 3.9 Quadratic Interpolation [34]

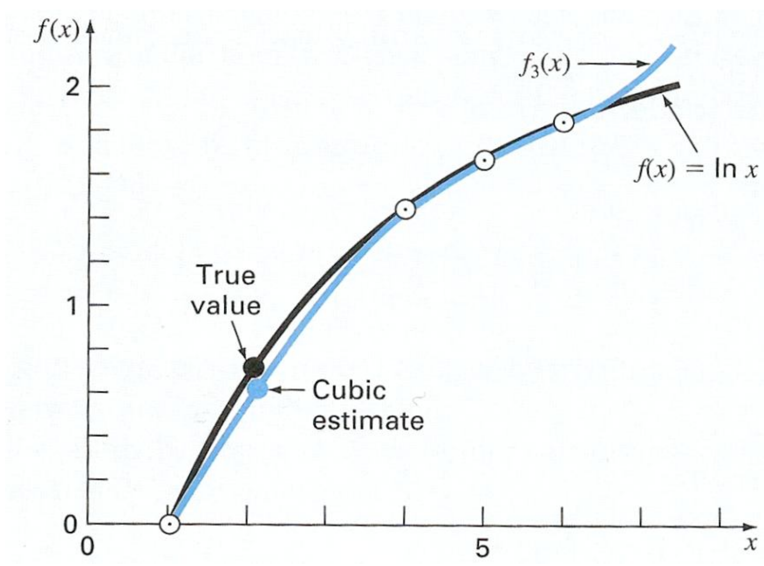


Figure 3.10 Polynomial Interpolation [34]

3.5.7 Spring Design at Operating Linear Engine

The springs that have been designed in the preliminary process is tested with forces of linear engine while the engine is operating. The input parameter is the same as stated in Table 3.6 except that the values maximum force is substituted by values of maximum force at operating linear engine. The maximum force at this stage is summation of minimum force and forces of linear engine at the operating condition. The force at the operating linear engine is taken from the value of pressure at the mean piston speed of the linear engine ranging from 0.5m/s to 6 m/s. The pressure values are taken from the graph of IMEP versus mean piston speed of linear engine. The data for the maximum forces at the operating linear engine and it sample of calculation is shown at the section 4.3 of result and discussion. The value of maximum forces is shown in the Table 4.2 of section 4.3.

3.6 Spring Selection Process

The spring that has been evaluated at the stage of operating linear engine is then go to the next step which is the spring selection process. There are many springs that have been designed are suitable in using with the linear engine this linear engine only need one design of spring. The spring that will couple with the engine later must go through the screening or selection process in order to get the best designed spring. The selection of best designed spring is based on following weights stated below.

i. Spring deflection ≥ 30mm
ii. Dynamic safety level ≥ 1.05
iii. Static safety level ≥ 1.25
iv. The spring must light in weight
v. Critical speed is lower than working speed to avoid resonance.

3.7 Linear Engine Design Process

After the best designed spring that is suitable with the linear engine, the next process is the process of linear engine design. As mentioned before, the linear engine design of this project is designed based on the conventional two stroke engine. The linear engine design of this project is made by modifying the crankshaft, crankcase of the conventional engine. The linear engine of this project use new connecting rod and spring which functioned similar with the crankshaft of the original engine. The name of linear engine is derived from the linear motion of the spring compare to rotational motion of crankshaft of conventional two stroke engine when subjected with force come the combustion process of air – fuel mixture in the combustion chamber. The linear engine use new crankcase in order to accommodate the place for the spring when the spring is mounted to the engine later. The new crankcase of linear engine is somewhat bigger than the crankcase of the conventional linear engine. The linear engine of this project has the same components as the original engine except the crankshaft and crankcase. The list of components of the original engine which is Back Pack Brush Cutter (BG – 328) engine is shown at the appendix section of this report. So, the components of the linear engine such as the piston and cylinder head have the same dimension like piston and cylinder head of original engine. The vernier caliper is used to measure the dimension of the components such as piston and others. After that, the dimension of the components that have been measured is used to design the linear engine in the Solid work software version 2005. The new crankcase or the correct term is the new casing of the linear engine designed based on the dimension of the spring that has been designed.

CHAPTER 4

RESULT AND DISCUSSION

4.1 Data of the Linear Engine Performance

Data of the linear engine performance is obtained form the GT-Power software as mentioned before. The data of linear engine performance here is IMEP, Indicated Mean Effective Pressure of linear engine taken at operating linear engine ranging from 0.5 m/s to 6 m/s. Figure 4.1 shows the graph of IMEP versus mean piston speed of linear engine at operating condition. Table 4.1 shows the value of force at the operating linear engine ranging from 0.5m/s to 6 m/s.

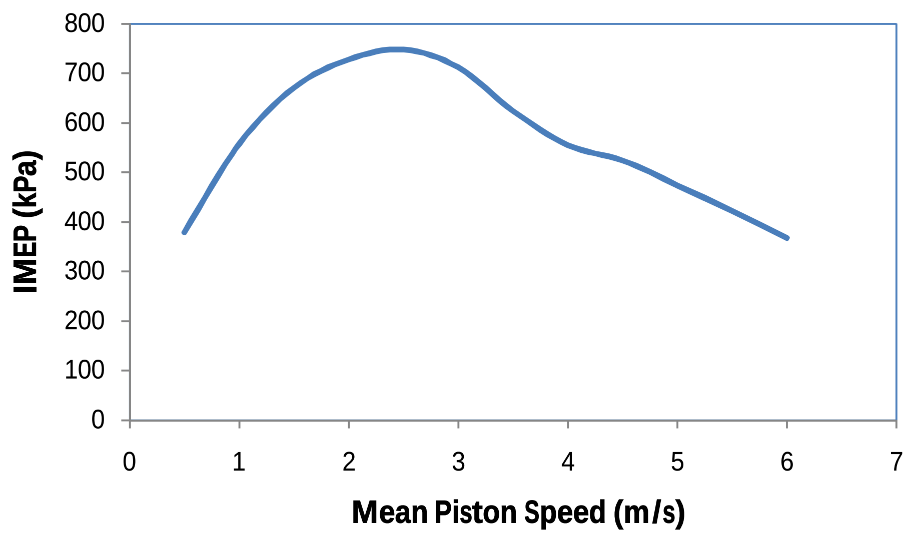


Figure 4.1 IMEP versus Mean Piston Speed of Linear Engine

Table 4.1 The Value of forces at the Operating Linear Engine

Mean Piston Speed, (m/s)	**IMEP, (kPa)**	**Force, (N)**
0.5	379	398.746
1.0	557	586.02
1.5	671	705.959
2.0	728	765.929
2.5	748	786.971
3.0	712	749.095
3.5	624	656.51
4.0	555	583.916
4.5	524	551.3
5.0	474	498.695
5.5	422	443.986
6.0	367	386.121

4.1.1 Sample calculation of the Forces at the Mean Piston Speed of Linear Engine

The value of forces in the Table 4.1 is calculated by using Equation 4.1. The sample calculation for the forces is shown below.

$$F = PA \tag{4.1}$$

Where:

F is Force

P is Pressure

A is Area

From the equation above, the value of pressure is IMEP of linear engine whereas the value of area is the piston crown surface area. The value of IMEP is get from the graph of IMEP versus mean piston speed of linear engine which is from the graph of Figure 4.1. The piston crown surface area is calculated by using following Equation 4.2 and the calculation is shown below.

$$A = \Pi/4\ (d^2) \tag{4.2}$$

Where:

d is piston diameter of linear engine.

So, the piston diameter of linear engine in this project is 36.6 mm.

$A = \Pi/4 \text{ x } (36.6\text{x}10\text{^-}3)$

$= 1.0521\text{x}10\text{^-}3 \text{ m}^2$

So, the value of force at the IMEP value of 379kPa is:

$F = (379\text{x}10\text{^}3 \text{ Pa}) \text{ x } (1.0521\text{x}10\text{^-}3 \text{ m}^2)$

$= 398.746 \text{ N}$

4.2 Data for the Preliminary Spring Design Process

The input data for preliminary spring design process is minimum and maximum forces. The sample calculations for the minimum and maximum forces are shown below.

i. Calculation for minimum force is calculated by using the Equation 4.1. The value for the area is the piston crown surface area of linear engine and the pressure value is calculated from the compression ratio of the linear engine. The typical pressure at the top dead centre (TDC) is 9 atm. So, the pressure value of 9 atm is equivalent to 9.09 x10^5 Pa. Then the calculation for the minimum force is shown below.

$F\ min$ = (9.09 x10^5 Pa) x (1.0521x 10^-3 m^2)

= 956.36 N

ii. The calculation for the maximum force is the summation of minimum force with the force value that derived from mean pressure of the IMEP versus mean piston speed graph. The calculation of maximum value is calculated by using Equation 4.1. The value of area is piston crown surface area and the value of pressure is mean pressure of IMEP versus mean piston speed graph. The calculation for mean pressure is shown below.

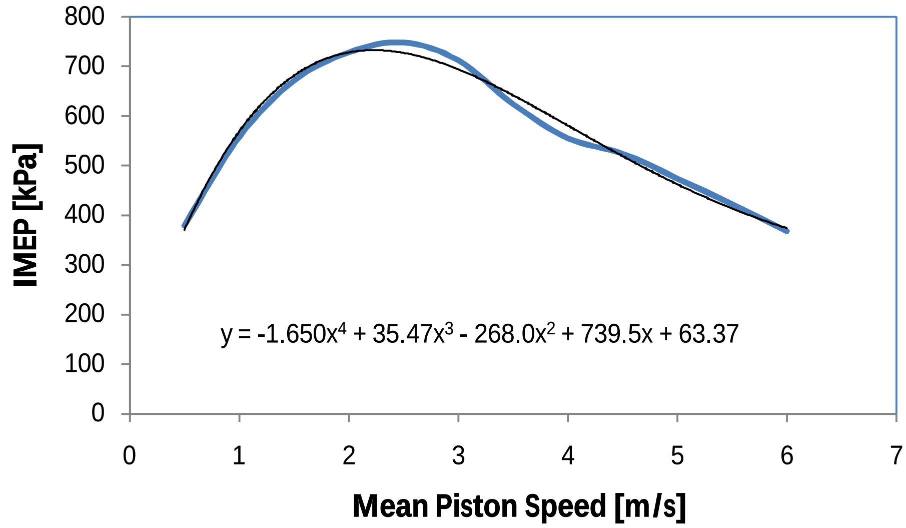


Figure 4.2 Polynomial Interpolation of IMEP versus Mean Piston Speed

The equation for the Polynomial Interpolation line of graph IMEP versus mean piston speed of operating linear engine is shown below.

$$y = -1.650x^4 + 35.47x^3 - 268.0x^2 + 739.5x + 63.37 \tag{4.3}$$

The equation to find the mean pressure is shown below.

$$P\ mean = \int_a^b f(x)\,dx / (b-a) \tag{4.4}$$

Where:

P mean is mean pressure of IMEP versus mean piston speed graph of linear engine

a is initial value of mean piston speed

b is final value of mean piston speed

So the mean pressure is.

$$\text{Mean pressure} = [-1.650X^5/5]_{0.5}^{6} + [35.47X^4/4]_{0.5}^{6} + [-268X^3/3]_{0.5}^{6} + [739.5X^2/2]_{0.5}^{6} + [63.37X]_{0.5}^{6}$$

= 3207.920262/ (6-0.5)

= 583.26 kpa

Then the value for maximum force for the preliminary spring design process is.

$F\ max$= 613.6 + 956.36

$F\ max$= 1569.96N

4.3 Data for the Spring Design at Operating Linear Engine

The input data for spring where is working at the operating linear engine are minimum and maximum force. The value for minimum force has been calculated in the section 4.2 which the value is 956.36 N. The value for maximum force is taken from the forces at the different mean piston speed ranging from 0.5m/s to 6 m/s. The maximum force for the spring design at the operating linear engine is the summation of minimum force with forces at the operating linear engine of mean piston speed from 0.5 m/s to 6 m/s. The maximum forces at the different mean piston speed is briefly stated in the Table 4.2.

Table 4.2 Maximum Forces for Spring Design at Operating Linear Engine

Mean Piston Speed, (m/s)	**Force at Operating Linear Engine , (N)**	**F max of Spring Design at Operating Linear Engine, (N)**
0.5	398.746	1355.106
1.0	586.020	1542.380
1.5	705.959	1662.319
2.0	765.929	1722.289
2.5	786.971	1743.331
3.0	749.095	1705.455
3.5	656.510	1612.870
4.0	583.916	1540.276
4.5	551.300	1507.660
5.0	498.695	1455.055
5.5	443.986	1400.346
6.0	386.121	1342.481

The sample calculation of forces that is used in spring design at the operating linear engine is shown below. The equation 4.4 is used to calculate the forces at the operating linear engine.

$$F\ max = F\ min + F\ opt \tag{4.5}$$

Where:

F max is maximum force

F_{min} is minimum force

F_{opt} is force at operating linear engine

F_{max} = 398.746 + 956.36

= 1753.852 N

4.4 Preliminary Spring Design Analysis

This paragraph discuss about how the selection process of the best springs that have been designed at the preliminary spring design process is made. The parameters of the best designed springs at the preliminary design process are shown in the Table 4.3. The springs shown in the table below is the best designed springs of wire diameter 6mm, 6.5mm, 7mm, 7.5mm and 8mm. The best spring is selected based on criteria of dynamic safety level and static safety level, the deflection of the spring, the spring must light in weight, the critical speed of the spring is lower than working speed of linear engine. The criteria of selection process of spring design stated in the section 3.7 under the chapter of methodology of this report.

Table 4.3 The Best Designed Springs of 6mm, 6.5mm, 7mm, 7.5mm and 8mm

Wire diameter, d (mm)	6	6.5	7	7.5	8
Number of active coils, n	14	15	11	11	9
Mean spring diameter, D (mm)	35	37	48	50	61
Outer spring diameter, Do	41	43.5	55	57.5	69

(mm)					
Inner spring diameter, Di (mm)	29	30.5	41	42.5	53
Free spring length, Lf (mm)	200	200	200	200	200
Preloaded installed length, Li (mm)	153.7	157.45	149.35	156.56	150.13
Fully loaded operating length, Lo (mm)	124	130.15	116.85	128.69	118.15
Theoretic spring limit length, Ls (mm)	90	104	84	90	80
Minimum working loading, F min (N)	956.4	956.4	956.4	956.4	956.4
Maximum working loading, F max (N)	1570	1570	1570	1570	1570
Theoretic spring limit loading, Fs (N)	2272.23	2157.82	2190.39	2421.71	2301.76
Critical spring speed, Vk (m/s)	8.36	5.82	6.38	7.42	6.4
Natural spring frequency, fn (Hz)	122.94	111.23	97.06	95.84	83.95
Developed wire length, l (mm)	1680	1894	1843	1920	1952
Spring constant, k (N/mm)	20.66	22.48	18.88	22.02	19.18
Spring weight, m (kg)	0.373	0.493	0.557	0.666	0.77
Permissible torsional stress, t_A (Mpa)	967	961	954	947	940
Static Level of safety	1.375	1.64	1.589	1.859	1.852
Fatigue strength for the given	728	726	723	719	717

loading, t_{max}					
Dynamic Level of safety	0.892	1.062	1.061	1.24	1.26
Spring deflection (mm)	29.7	27.3	32.5	27.87	31.99

From the Table 4.3, all spring will be evaluated based on the best spring criteria that stated in section 3.7 of chapter 3. The criteria of best designed spring are stated below:

vi. Spring deflection ≥ 30mm
vii. Dynamic safety level ≥ 1.05
viii. Static safety level ≥ 1.25
ix. The spring must light in weight

For the 1st criteria, there are 2 springs which deflect more than 30 mm. The springs are 8mm and 7mm of wire diameter. For the 2nd criteria, all the springs have dynamic level of safety more than desired dynamic level of safety, Sf = 1.05 except that the spring with wire diameter of 6mm which the value for the dynamic level of safety is 0.892. For the 3rd criteria, all the designed springs has static level of safety more than the value of desired static level of safety, Ss = 1.25. The spring with wire diameter of 6mm has lowest weight compare to the other springs whereas the weight is increase as the wire diameter increase from 6mm to 8mm. For this stage, there are two springs that meet the criteria of best designed spring. The springs that meet the criteria as stated above are the springs with wire diameter of 7mm and 8mm. The best designed spring at this stage is spring with wire diameter of 7mm. It is because the spring has lower weight compare to the spring with wire diameter 8mm.

4.5 Spring Design at Operating Linear Engine Analysis

All the designed springs from the preliminary design stage is further tested with the condition of operating linear engine. This paragraph objective is to find the best designed spring that can withstand the operating condition of linear engine. The best designed spring criteria for this stage is stated below with the additional criteria which the spring critical speed must lower than working speed of linear engine to avoid resonance .

i. Spring deflection ≥ 30mm
ii. Dynamic safety level ≥ 1.05
iii. Static safety level ≥ 1.25
iv. The spring must light in weight
v. Critical speed is lower than working speed of linear engine to avoid resonance.

The graph is constructed based on the best designed spring criteria as shown above to facilitate the selection process of best spring that mounted in the linear engine while the engine is operating. The best spring is the spring that can meet all the requirement of best spring criteria that have been stated above.

4.5.1 Spring Deflection

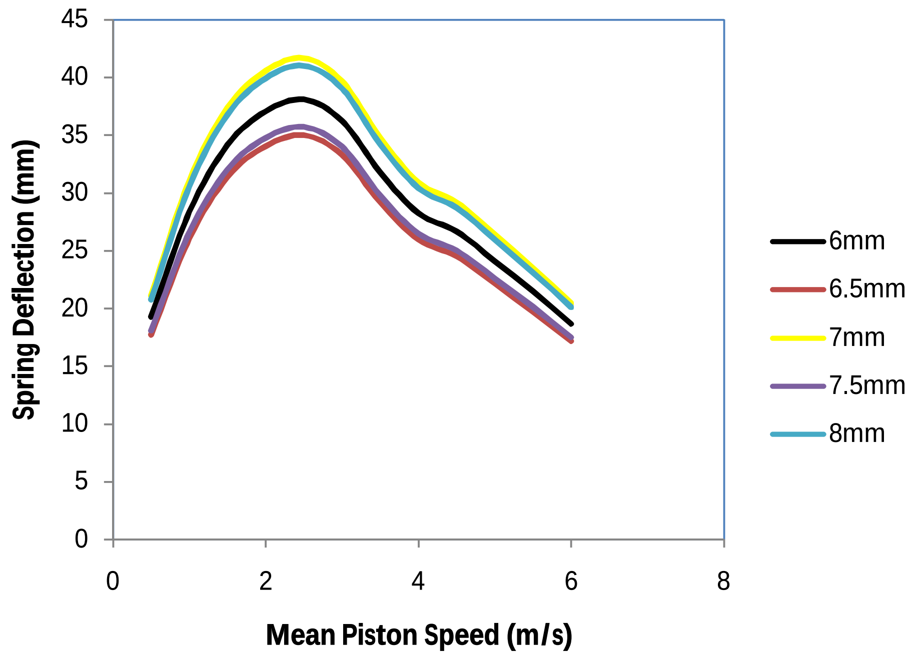


Figure 4.3 Spring Deflection versus Mean Piston Speed

From the Figure 4.3 of spring deflection of different wire diameter versus mean piston speed the spring with wire diameter of 7mm and 8mm are the springs that meet the criteria of spring that can deflect ≥ 30mm. It is because these two springs with wire diameter of 7mm and 8mm can deflect over 30 mm more compare to other spring. The spring that can deflect over 30mm almost at every mean piston speed is the best spring which later the spring can be used in the design of linear engine. The spring that can deflect over 30mm at every mean piston speed is the important of best spring criteria. It is because the intake port of the linear engine fully open when the spring can deflect over 30mm. The intake port that fully open provide the linear engine with sufficient amount of air that help the process of linear engine combustion.

4.5.2 Dynamic Level of Safety

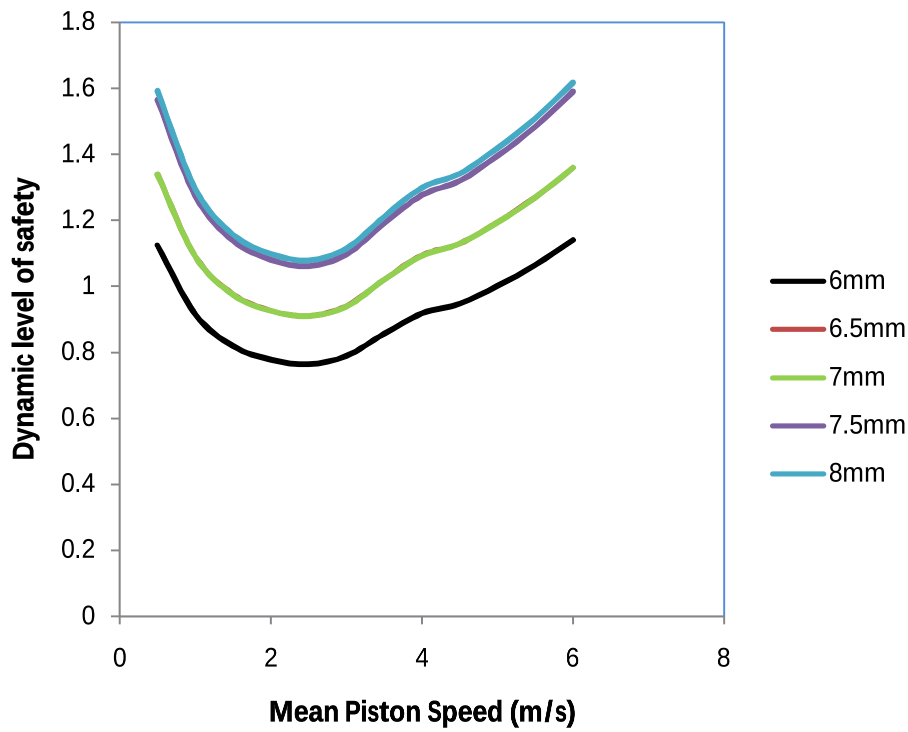


Figure 4.4 Dynamic Level of Safety versus Mean Piston Speed

From the Figure 4.4 of dynamic level of safety of spring with different wire diameter versus mean piston speed, the springs of wire diameter of 7.5mm and 8mm meet the best designed spring criteria of dynamic level of safety of spring should more than the desired dynamic level of safety for designed spring, Sf = 1.05. The springs with wire diameter of 7.5mm and 8mm are the springs that meet the criteria mentioned before. The curve line for spring with wire diameter 6.5mm coincide with the curve line of spring with wire diameter 7mm. It is because the value of dynamic level of safety for these two springs at operating linear engine not having much different. The springs that do not meet this requirement, the life of spring is not long when the spring working at the dynamic condition of linear engine.

4.5.3 Static Level of Safety

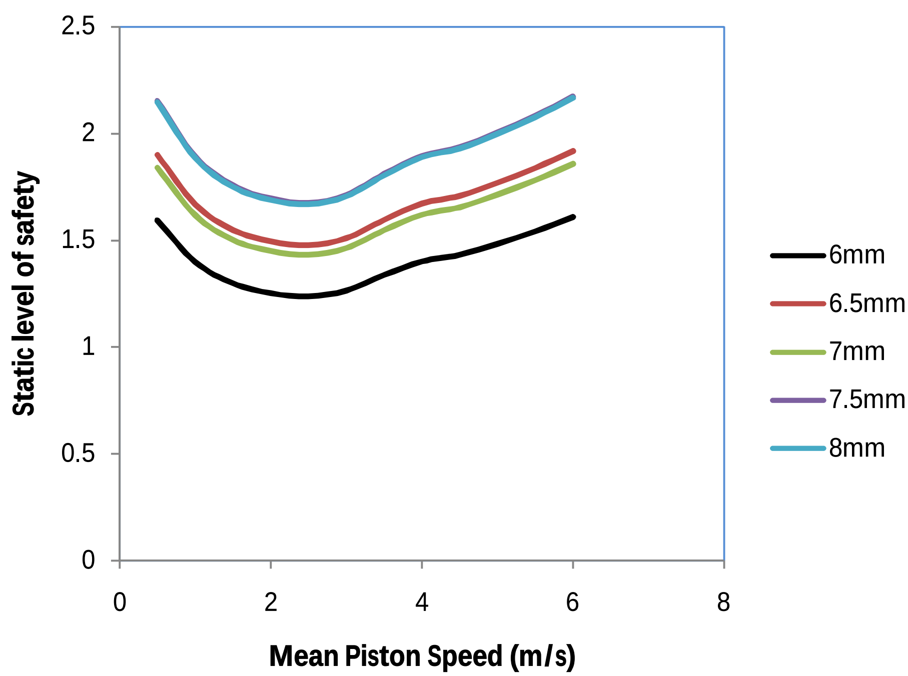


Figure 4.5 Static Level of Safety versus Mean Piston Speed

From the Figure 4.5, all the springs do not have problem with the static level of safety except the spring with the spring with wire diameter of 6mm. actually, this criteria is not a big concern when designing the spring. It is because the spring is subjected with fatigue loading due to variable mean piston speed of linear engine. So, the dynamic level of safety criteria is very important compare to this static level of safety. The curve line of spring wire diameter 7.5mm cannot be seen on graph because the curve line of spring with wire diameter 8mm coincide with it. It is because the value of static level of safety for these two springs not having much different.

4.5.4 Critical Spring Speed

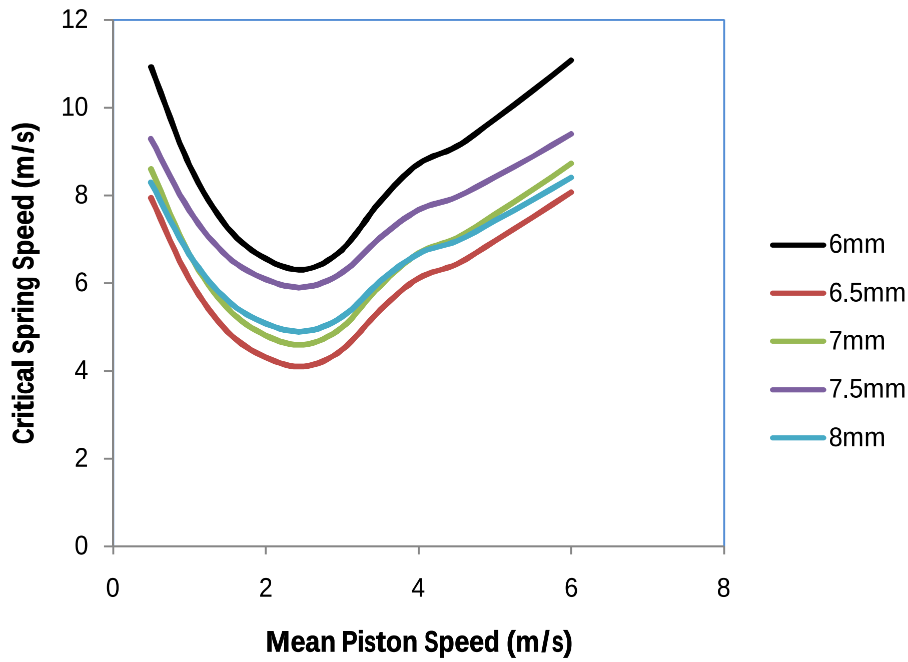


Figure 4.6 Critical Spring Speed versus Mean Piston Speed

From the Figure 4.6, all the springs do not have problem with resonance or in the spring industry it is called spring surge. The resonance or spring surge occurs when the value of critical spring speed is same as the value of the mean piston speed of operating linear engine.

4.6 Linear Engine Design Configuration

The linear engine design of this project is designed based on the best designed spring. The linear engine that have been assembled is shown if Figure 4.7, 4.8, 4.9 and 4.10. The figures of components for this linear engine are shown in the appendix section. The figure of linear engine mounted on the bed also is shown in the appendix section.

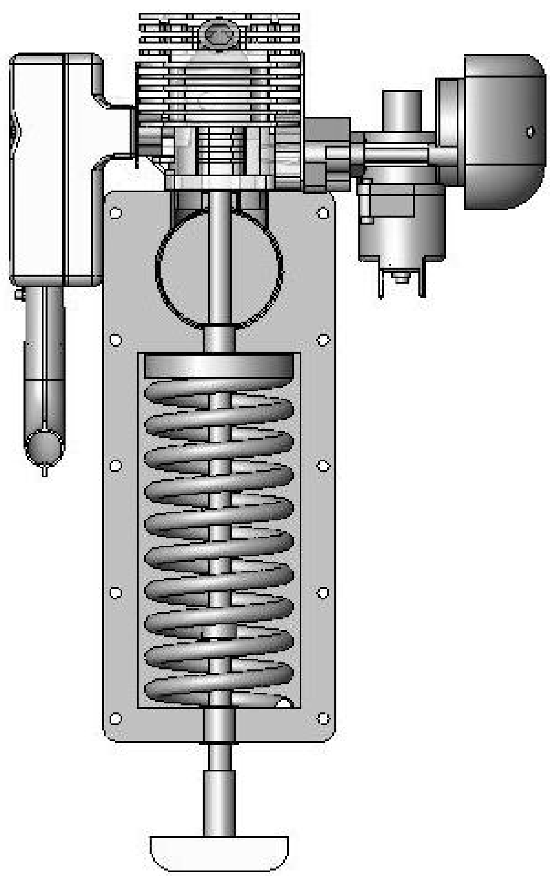


Figure 4.7 Front View of Linear Engine

From the Figure 4.7 the diametral clearance of the spring with the engine casing wall was 0.1D, where D is mean spring diameter of spring. The clearance gives the spring a considerable space to allow it expands and this can avoid the friction and wear on the spring from rubbing on the engine casing wall. The 0.1D clearance is commonly recommended diametral clearance suggested by spring design book.

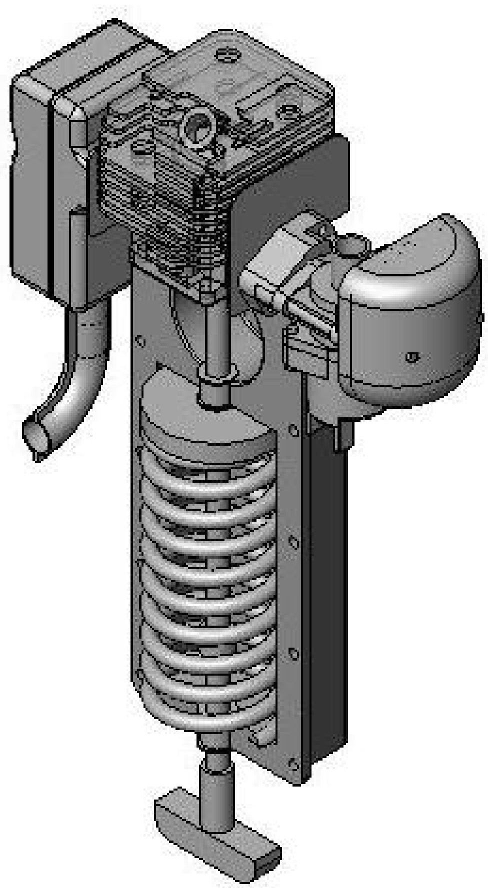


Figure 4.8 Isometric View of Linear Engine

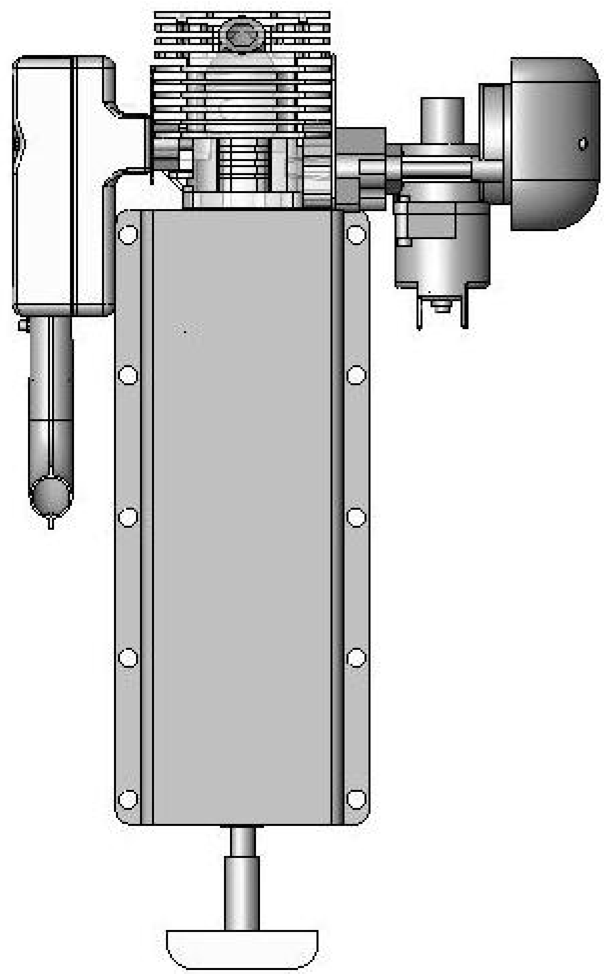


Figure 4.9 Front View of Full Assembly Linear Engine

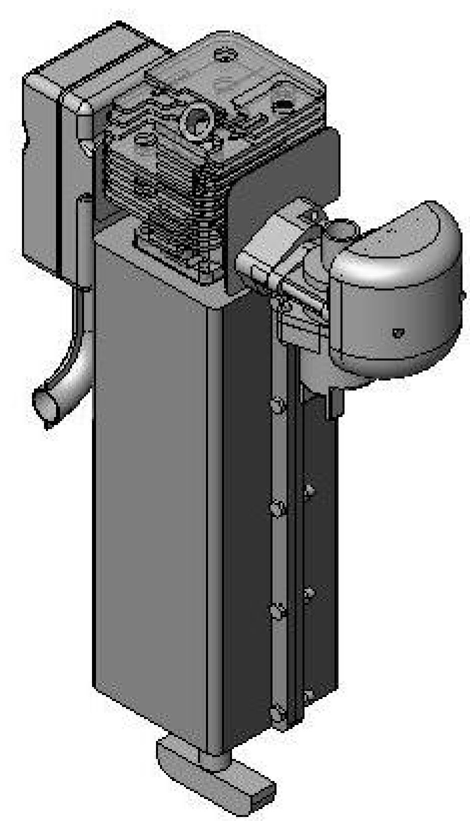


Figure 4.10 Isometric View of Full Assembly Linear Engine

From the analysis of spring, there is a case spring will deflect more than 30mm because of the action of forces that acting to the spring. Most of these cases happen when spring mounted to the linear engine at the operating linear engine. Spring that deflect more than 30mm will cause the piston collide with dead centre of linear engine. So, to avoid this happen the connecting rod of the piston is specially designed with a stopper. This stopper will limit the movement of the piston and ensure that the collision of piston with dead centre of linear engine not happened. The stopper is shown in red in color of Figure 4.11.

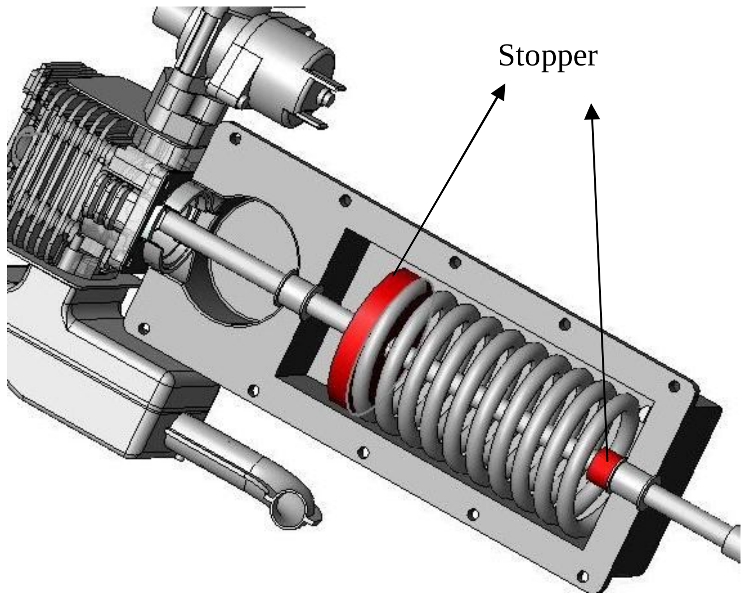


Figure 4.11 Location of Stopper in Linear engine

Figure 4.12 shows the linear engine mounted with spring at preloaded condition where the spring is firstly mounted in linear engine Figure 4.13 shows the linear engine mounted with spring that deflect ≥ 30mm. The spring is mounted in operating linear engine.

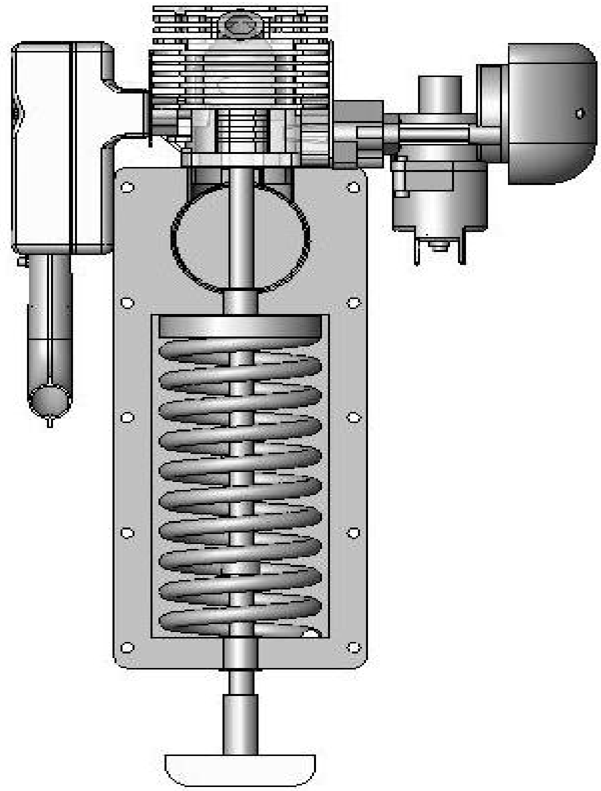


Figure 4.12 Spring at Preloaded Condition

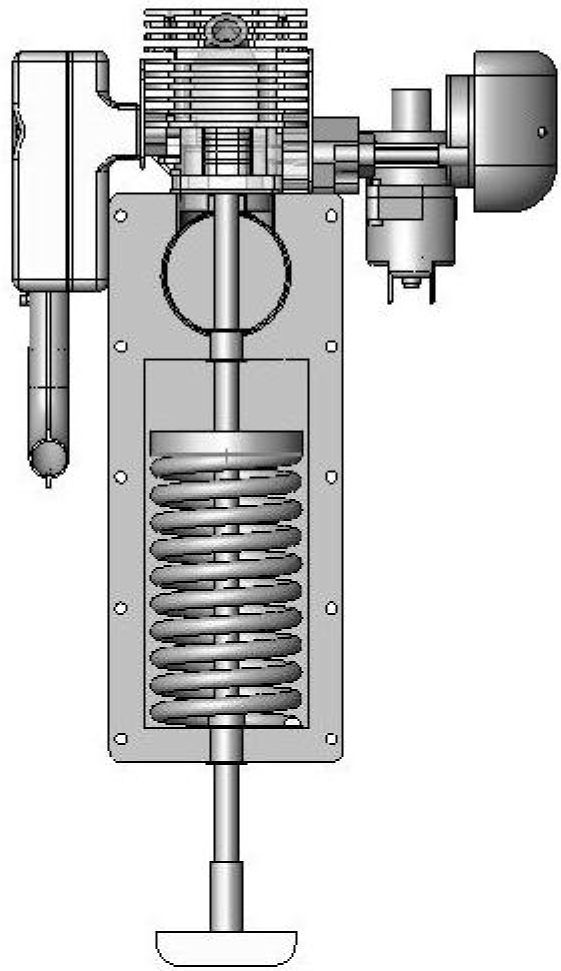


Figure 4.13 Spring at Fully Loaded Condition

CHAPTER 5

CONCLUSION

5.1 Conclusion

The process of spring design for linear engine in this project is based on the values of minimum and maximum forces. These forces are subjected to the spring. There are two stages of spring design process. The preliminary process of spring design is the first stage of spring design while the process of spring design at operating linear engine is the second stage of spring design in this project. For the preliminary process of design spring, the minimum force is obtained from the value of compression ration of the engine which is 9 atm. Furthermore, the maximum value of force for the preliminary process of design spring is obtained from the summation of minimum force with mean force subjected to the spring while the linear engine is operating. The mean force is obtained from the value of mean pressure from the IMEP data of operating linear engine. For the second stage process of the spring design where the spring is designed at operating linear engine the spring also is designed based on the value of minimum and maximum force as mentioned in the first stage of spring design except that the value of maximum force is obtained from the summation of minimum force and pressures subjected to the spring while the linear engine is operating.

The best designed spring for this project is spring with wire diameter 8mm. It is because the spring has met the requirement of best spring criteria of this project. The criteria of best designed spring for this project are shown in the section 4.5 chapter 4 of this report. For the 1st criteria, this spring deflect over 30mm for about almost all mean piston speed of linear engine. The spring also has higher dynamic level of safety more than desired dynamic level of safety for the designed spring of this project, which Sf = 1.05. Furthermore, the spring also has static level of safety more than desired static level of safety for the designed spring of this project, where the desired static level of safety for this project is 1.25. The value of critical speed of this spring is higher compare to the mean piston speed of the linear engine. So, the resonance or spring surge for this spring will not occur and therefore the spring with wire diameter of 8mm has most life span compare to the other springs.

The linear engine design for this project is designed based on the dimension of best designed where the best spring is spring with wire diameter of 8mm. The linear engine in this project is designed based on the conventional two stroke engine except that the modification is made at the crankshaft and crankcase of the conventional two stroke engine as mentioned in methodology section. The crankshaft of conventional two stroke engine is substituted with spring and new connecting rod of linear engine. This spring and new kind of connecting rod for the piston are called spring mechanism in this project. The linear engine is designed so as to give a diametral clearance of 0.1D where this is to anticipate the action of deflected spring where the spring will expand in it diameter when the spring deflected. The new kind of connecting rod coupled to the linear engine is designed with stopper. The function of stopper is to limit the movement of piston and ensure that the collision between the spring and dead centre of linear engine is not happen when the spring deflect more than 30 mm. The linear engine of this project is very simple because using spring mechanism as rebound device compare to the linear engine exist in the world today where use the complicated kind of rebound device. This linear engine is very light because the usage of spring mechanism as rebound device and this linear engine also easy to fabricate due there is no complex geometrical shape is used in this linear engine design.

5.2 Recommendation for the Future Work

For recommendation for the future work, the extensive research needs to be done in term of ignition system, lubrication system, and starter method for the linear engine in order to make this linear engine work. Actually the purpose for this linear engine is to generate the electricity when couple with the generator and because of that the research also need to done in how to couple this linear engine with the generator. Apart from that, this linear engine can be used as laboratory equipment to conduct an experiment to see the performance of the linear engine compare to the performance of conventional two stroke engine.

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Appendix A
Project Gantt chart

Gantt chart for FYP 1

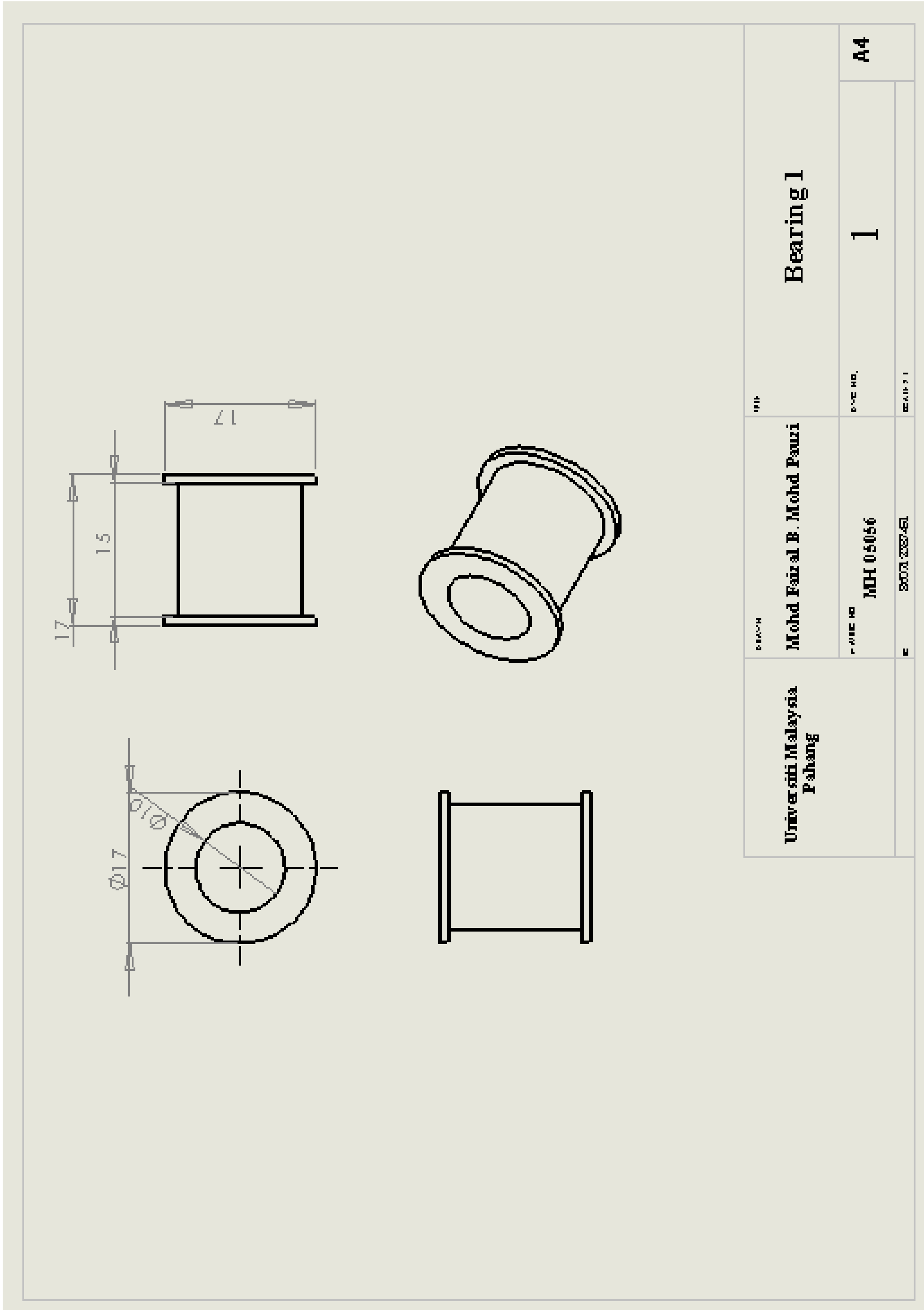
Activities/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Title of project briefing by supervisor																
Verify the project title, objectives and scopes																
Write down the background, abstract, objective and scope																
Literature Study																
Study of spring design process																
Study of linear engine																
Study spring software																
Determined the input parameter for the spring designed process																
Determined the input parameter for linear engine design process																
Designed the linear engine																
Submit proposal and draft of report																
Presentation of Proposal																

Gantt chart for FYP 2

Activities/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Literature Study																
Data collection																
Spring design process																
Analysis of data and results																
Designed spring selection																
Designed the linear engine																
Conclusion of the project																
Final Presentation																
Preparing and submit report																

Appendix B

Drawing in Solid Works



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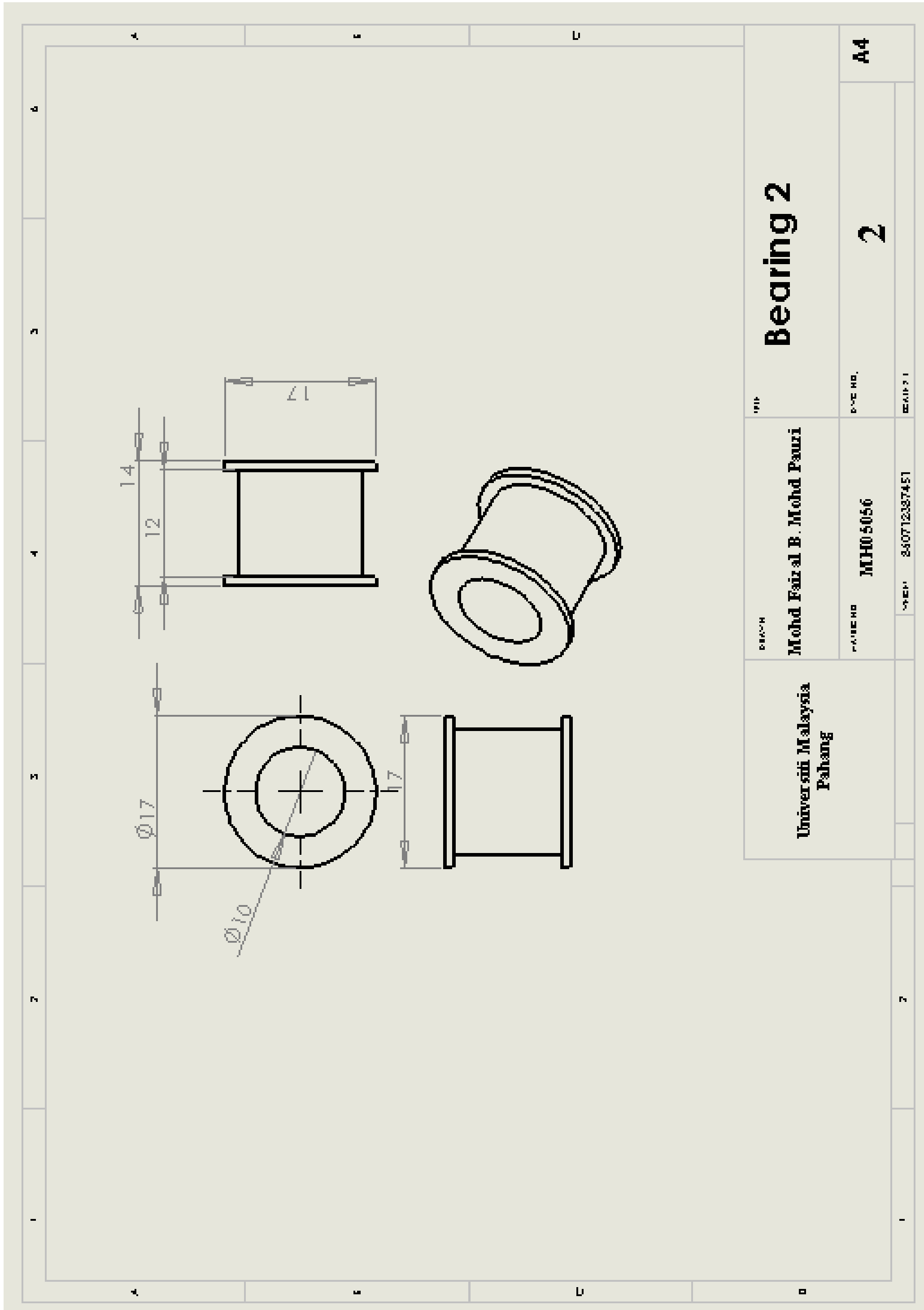
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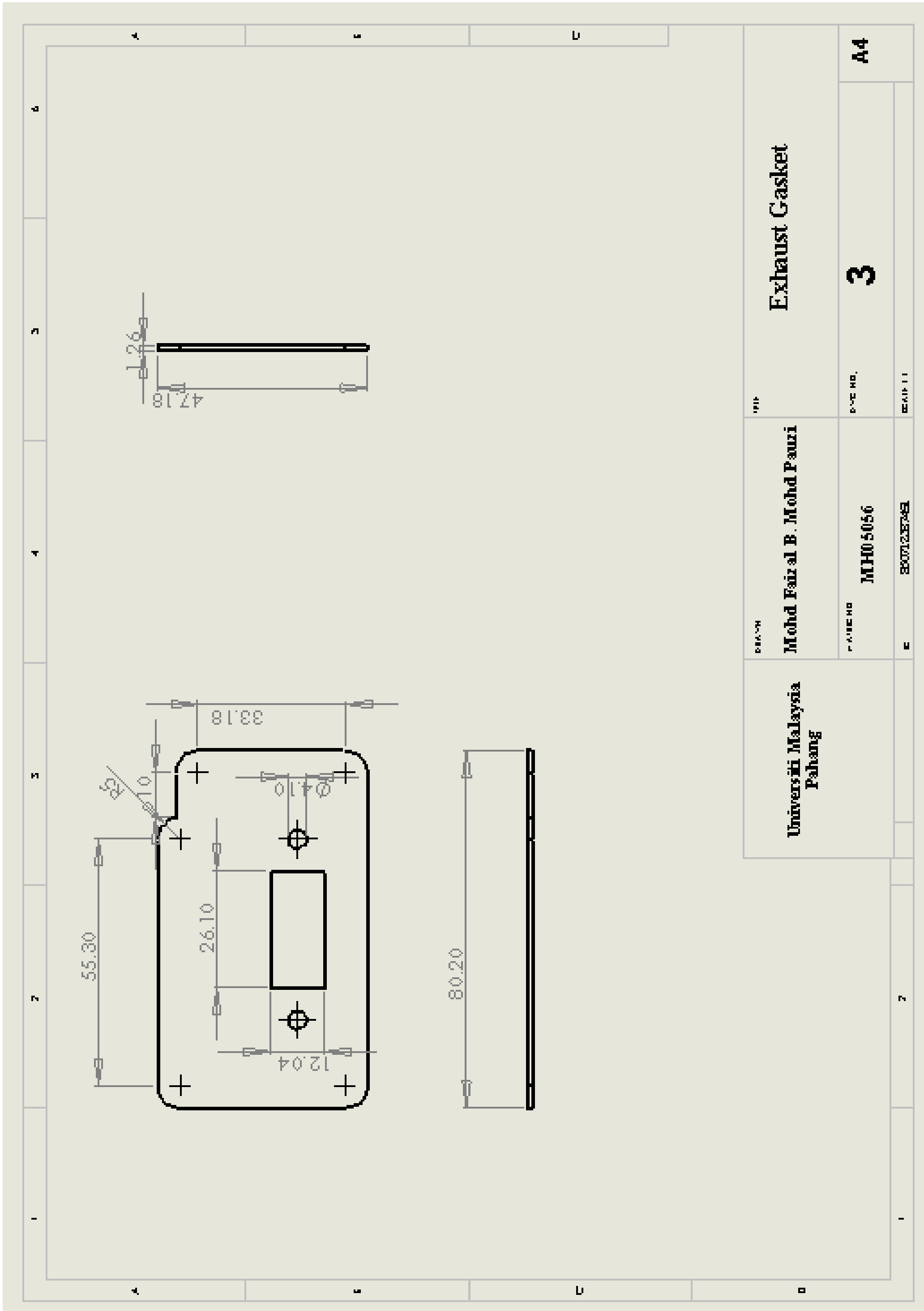
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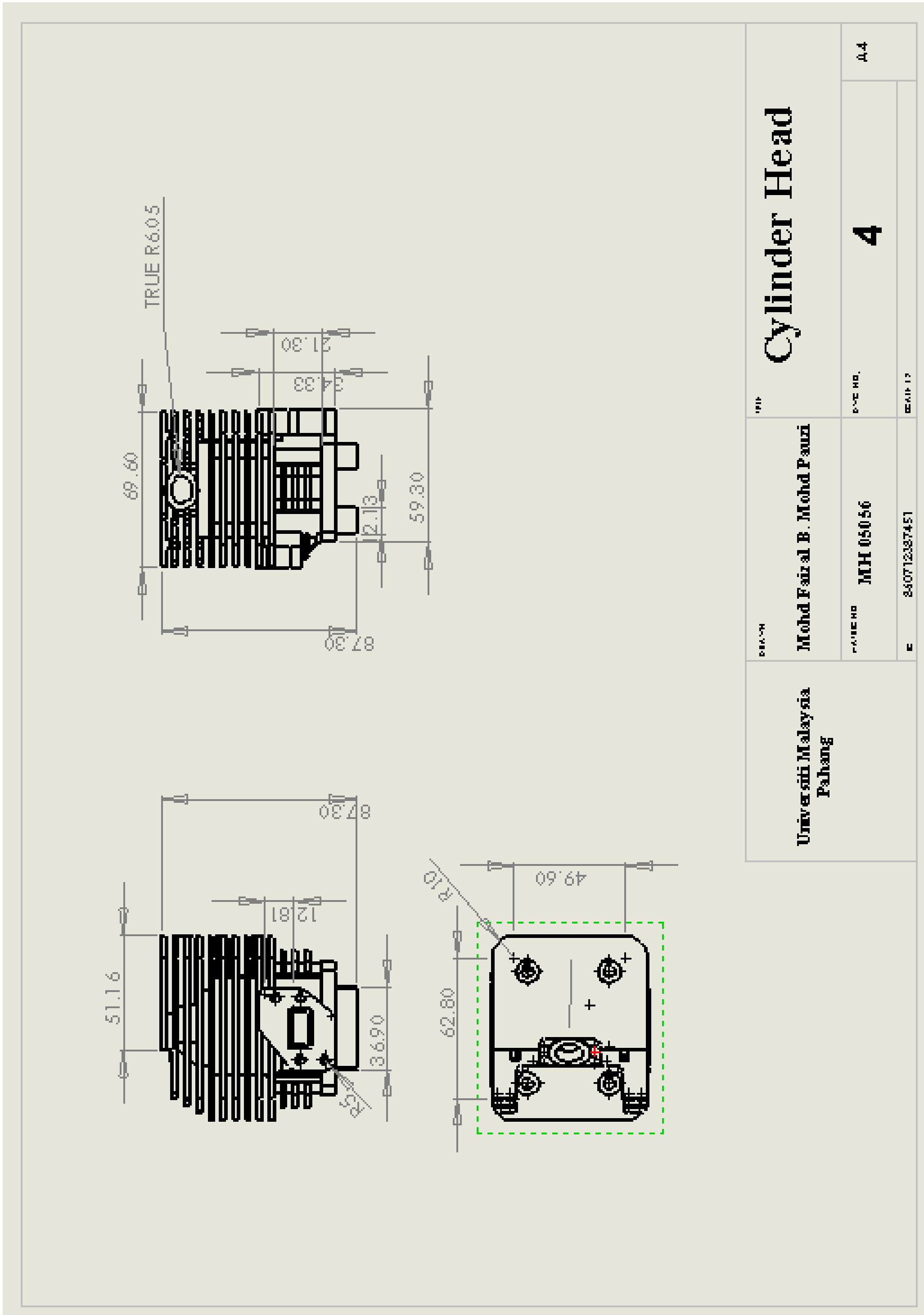
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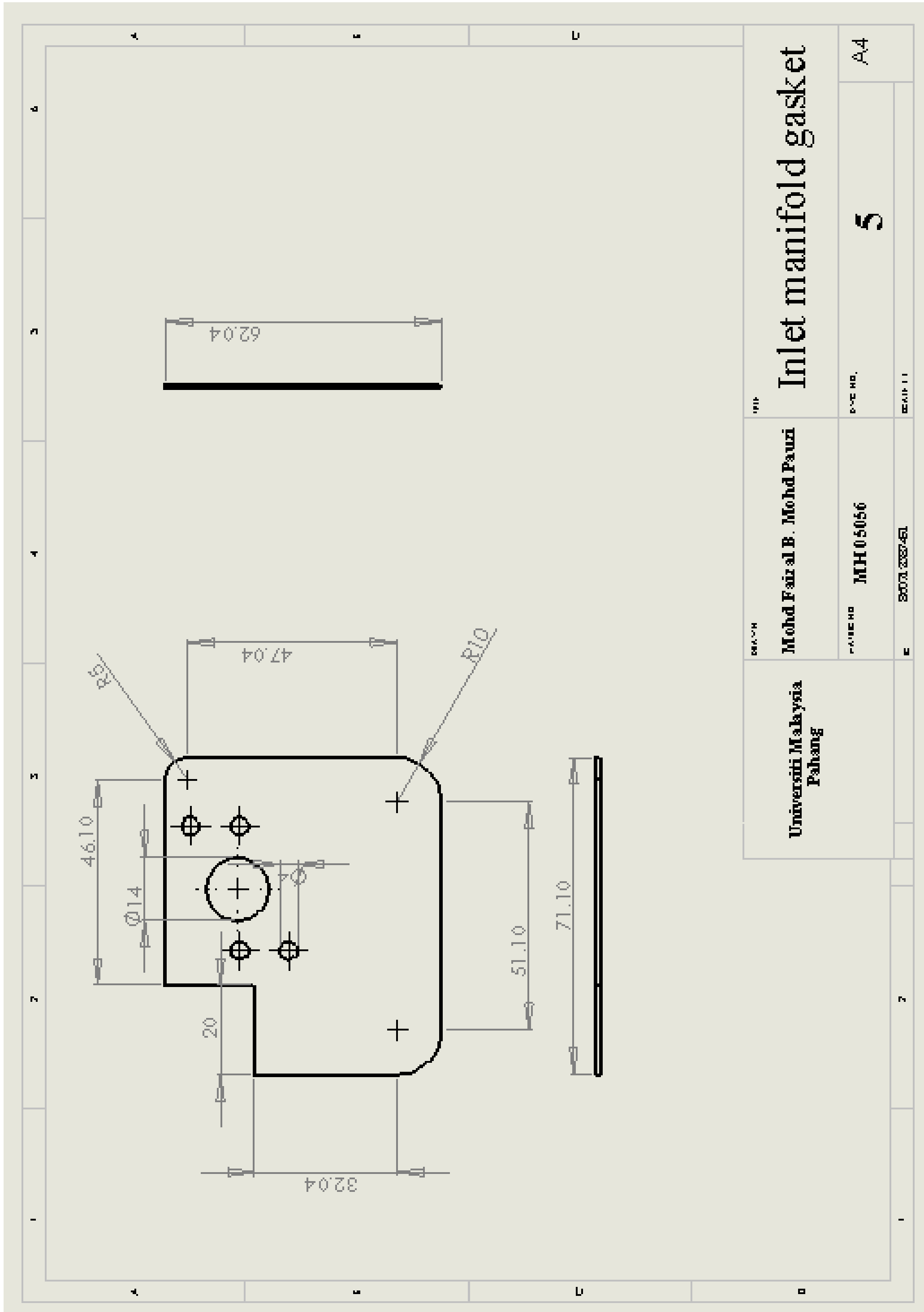
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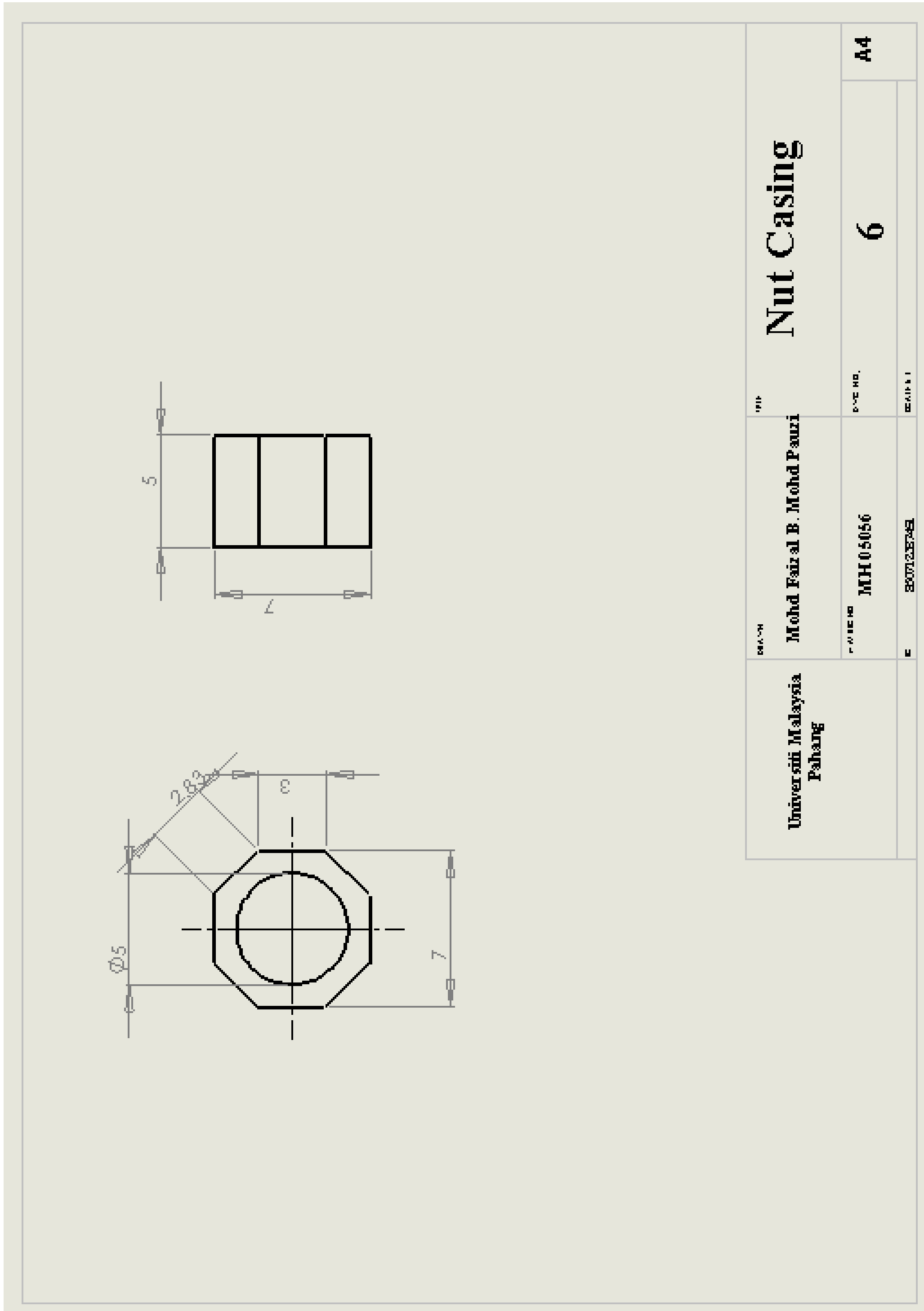
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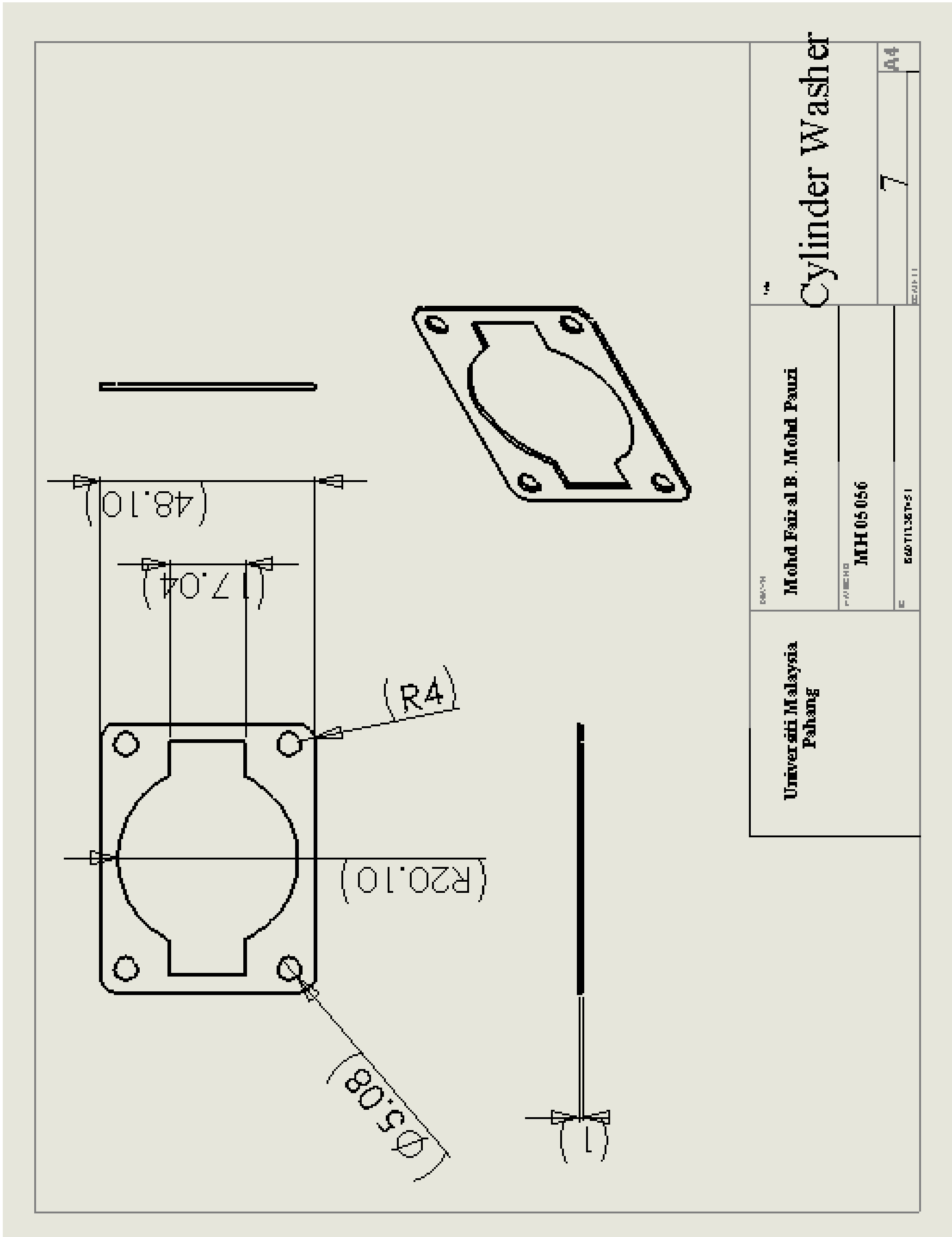

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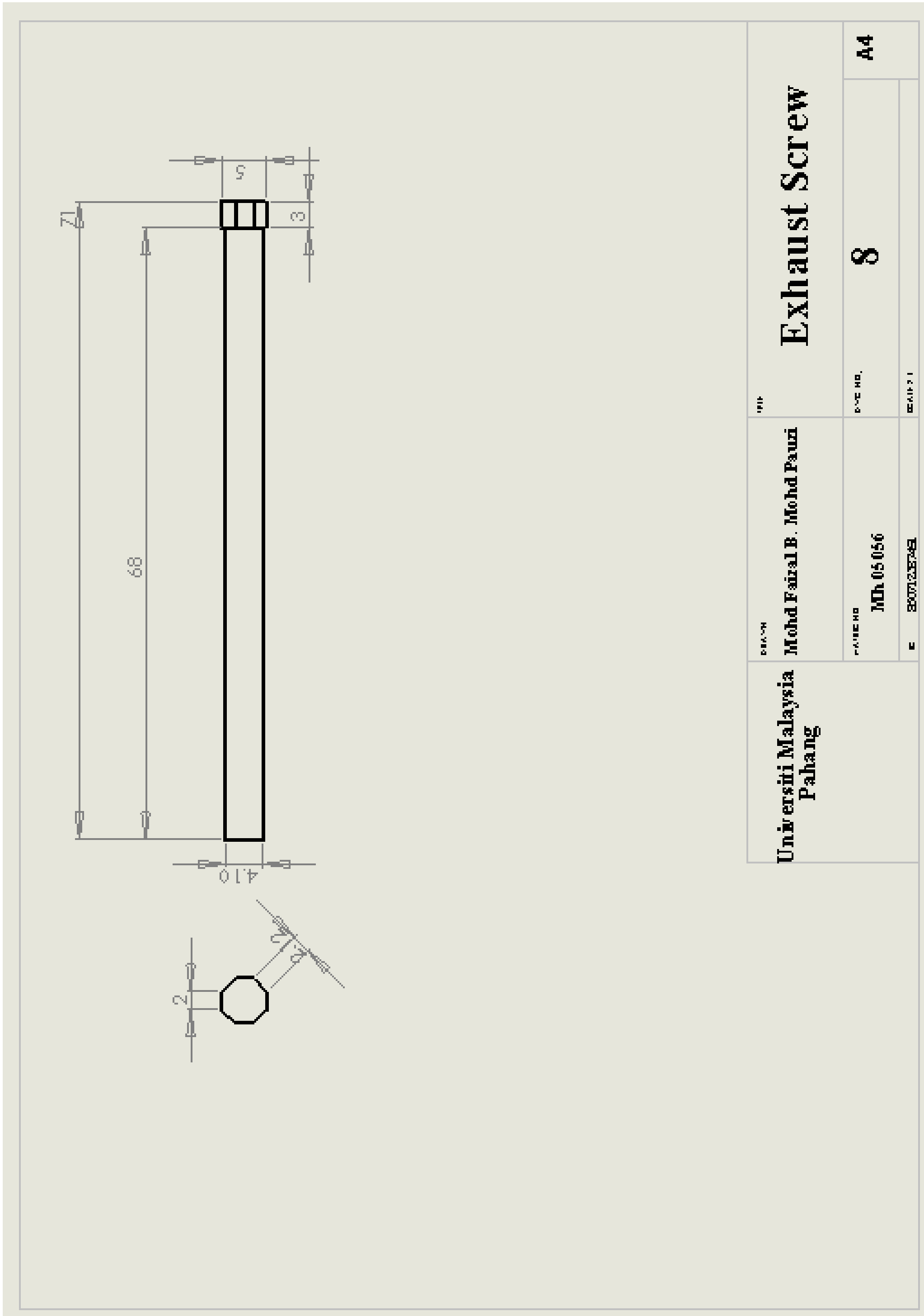
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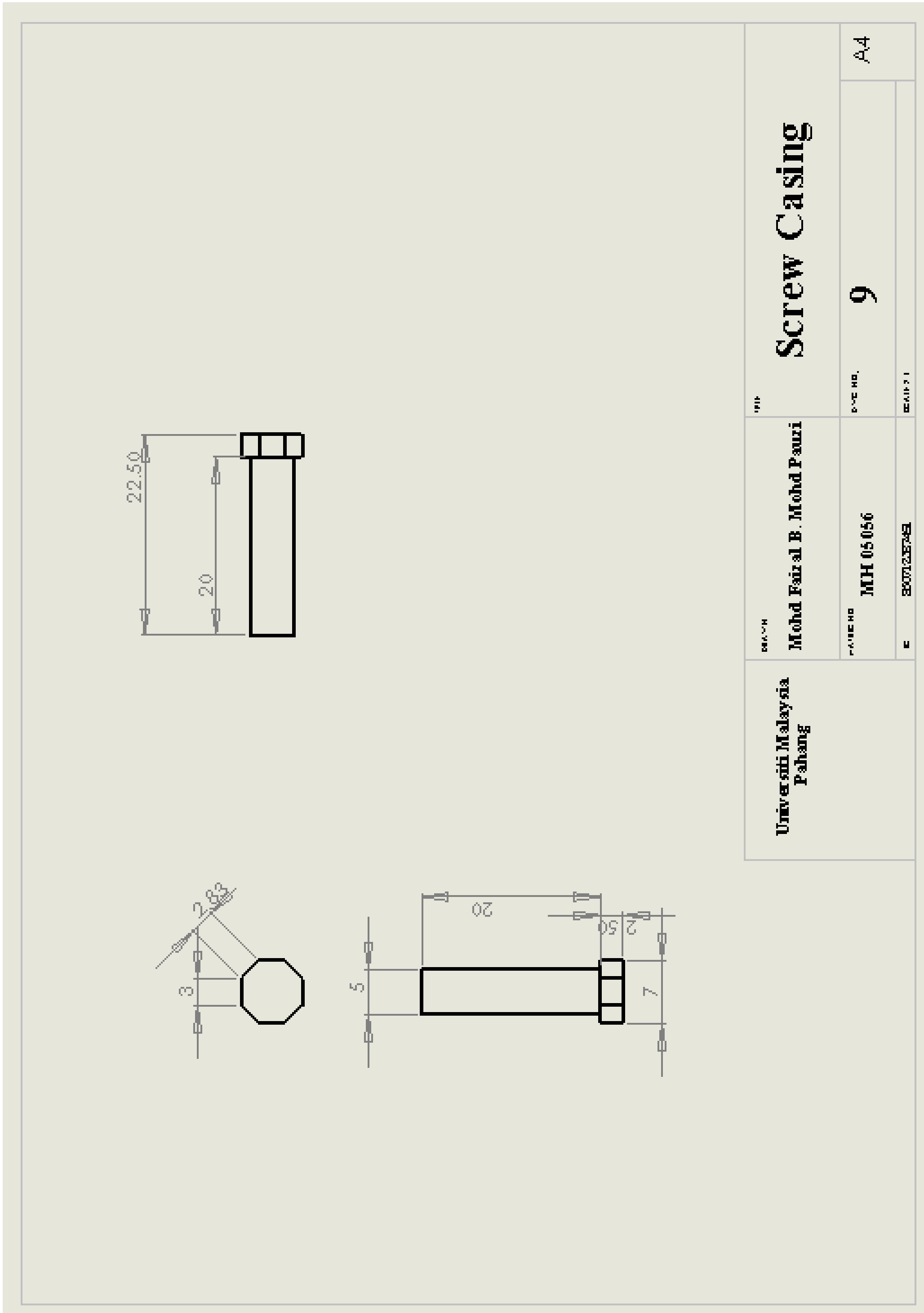

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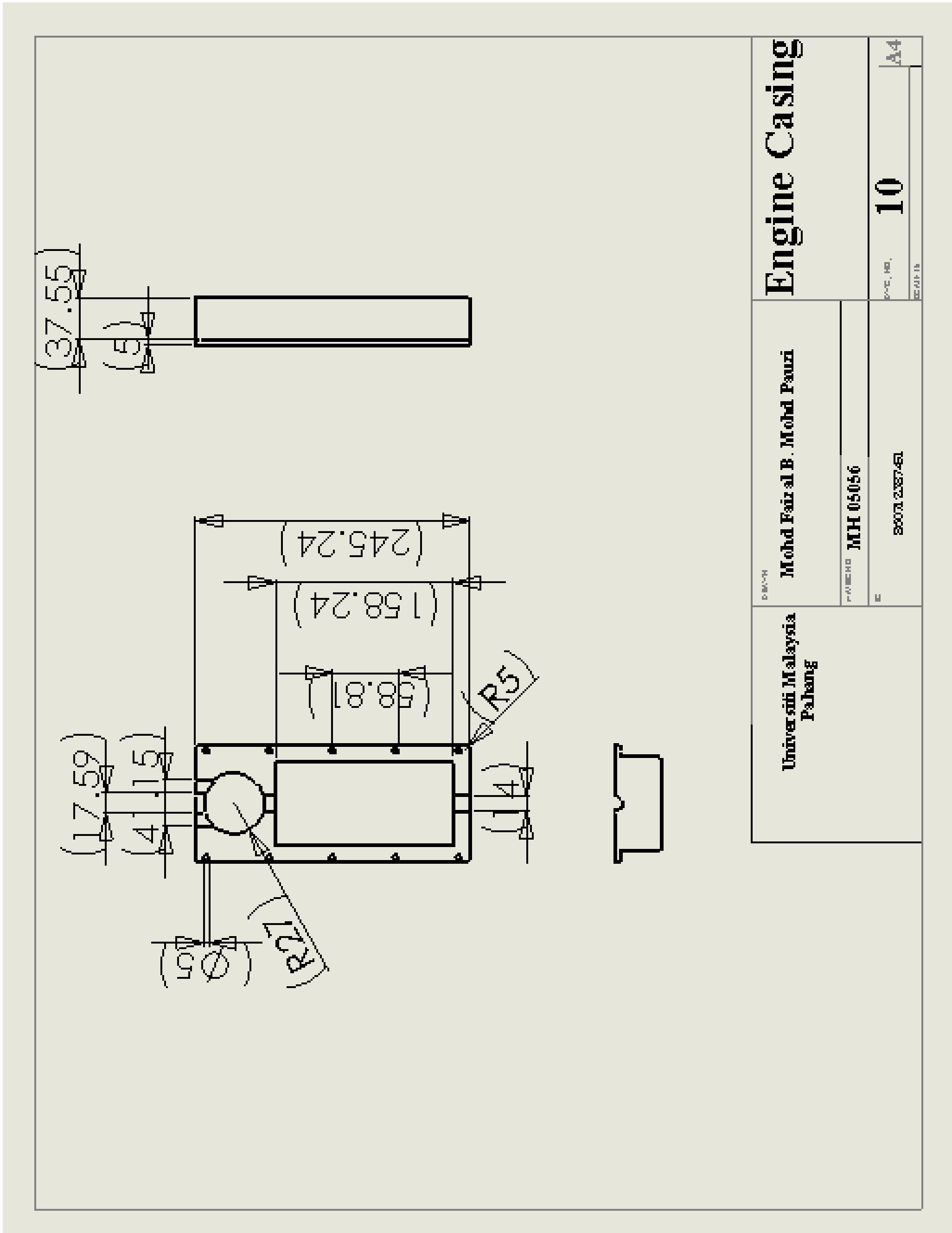
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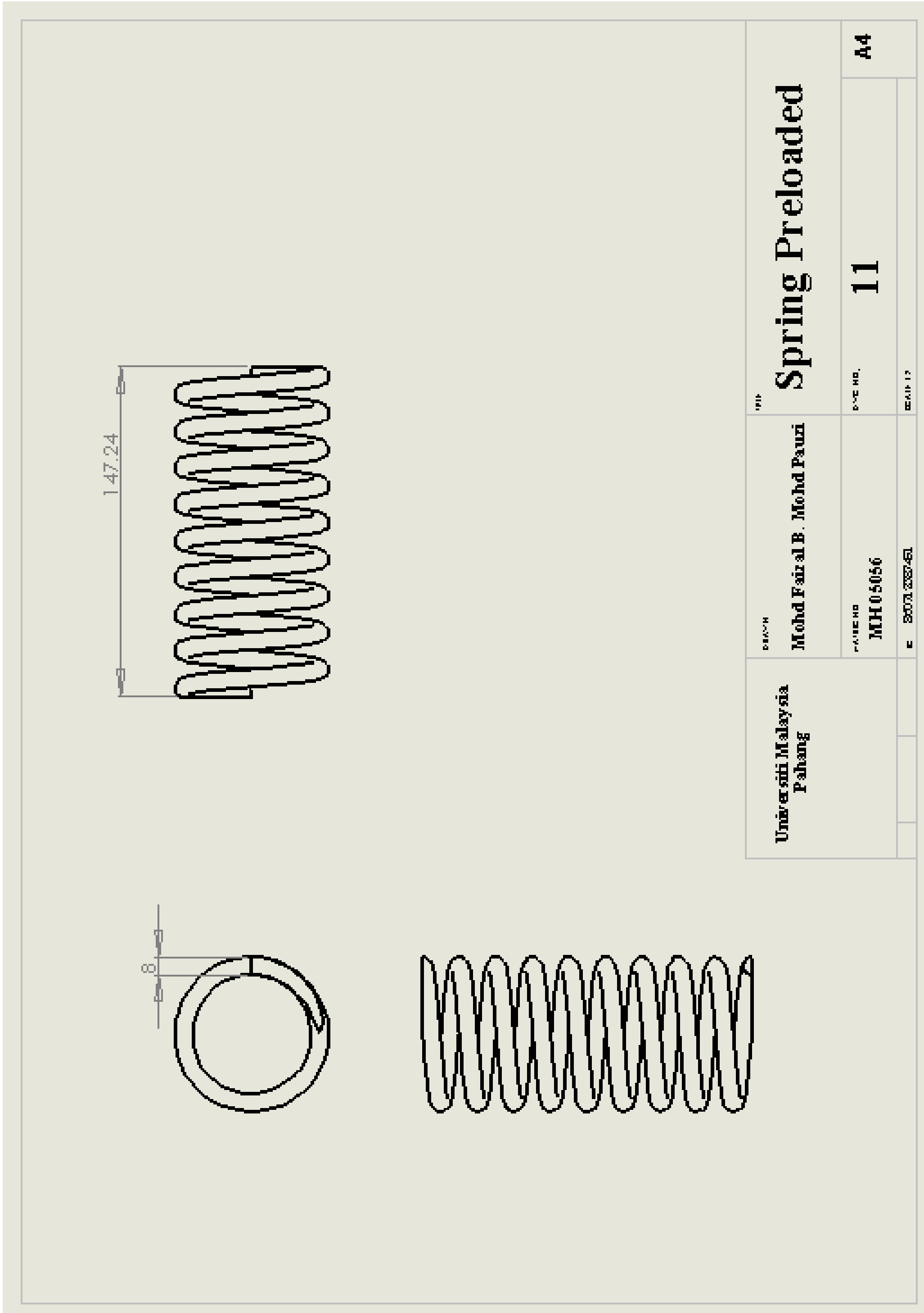
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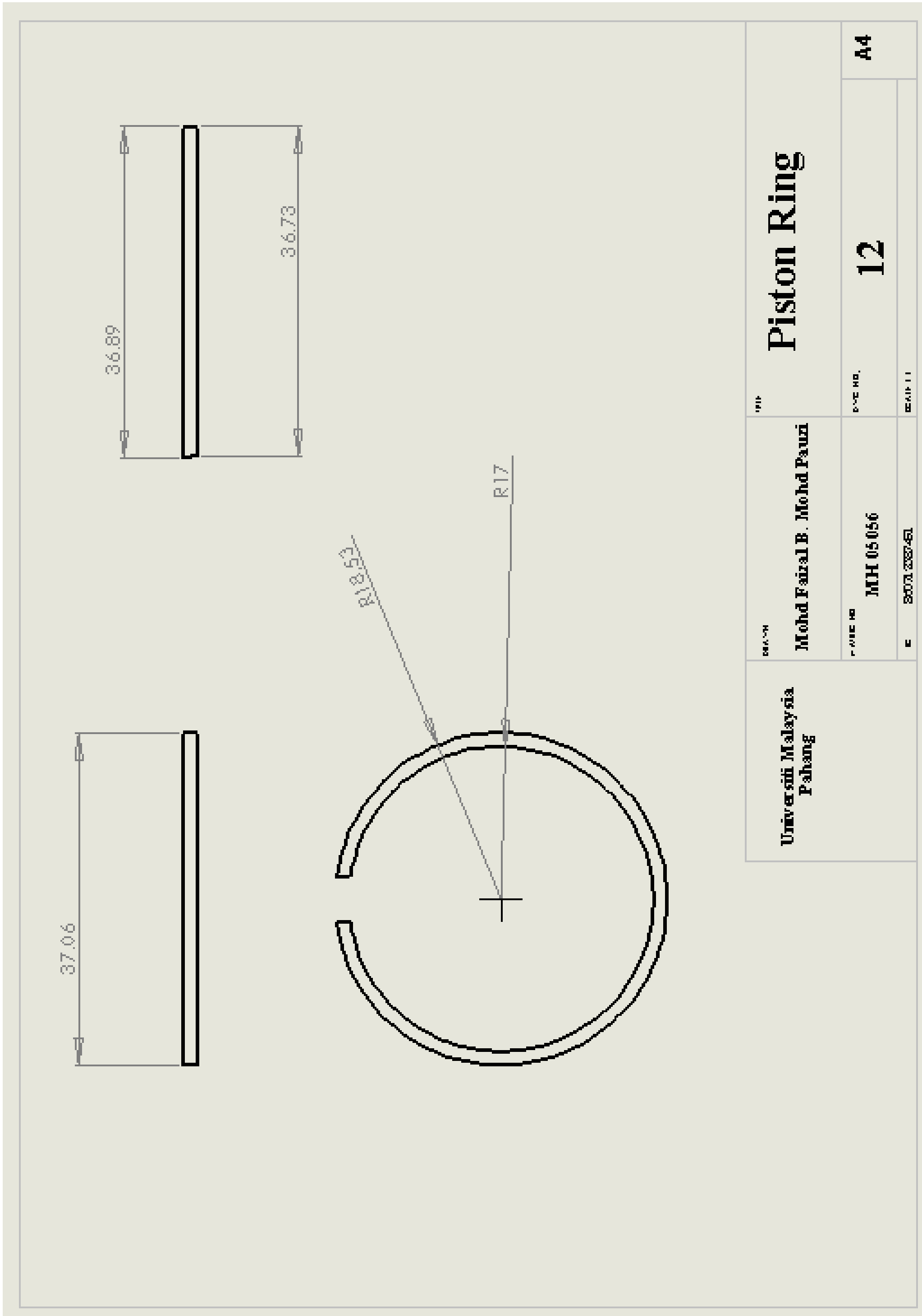
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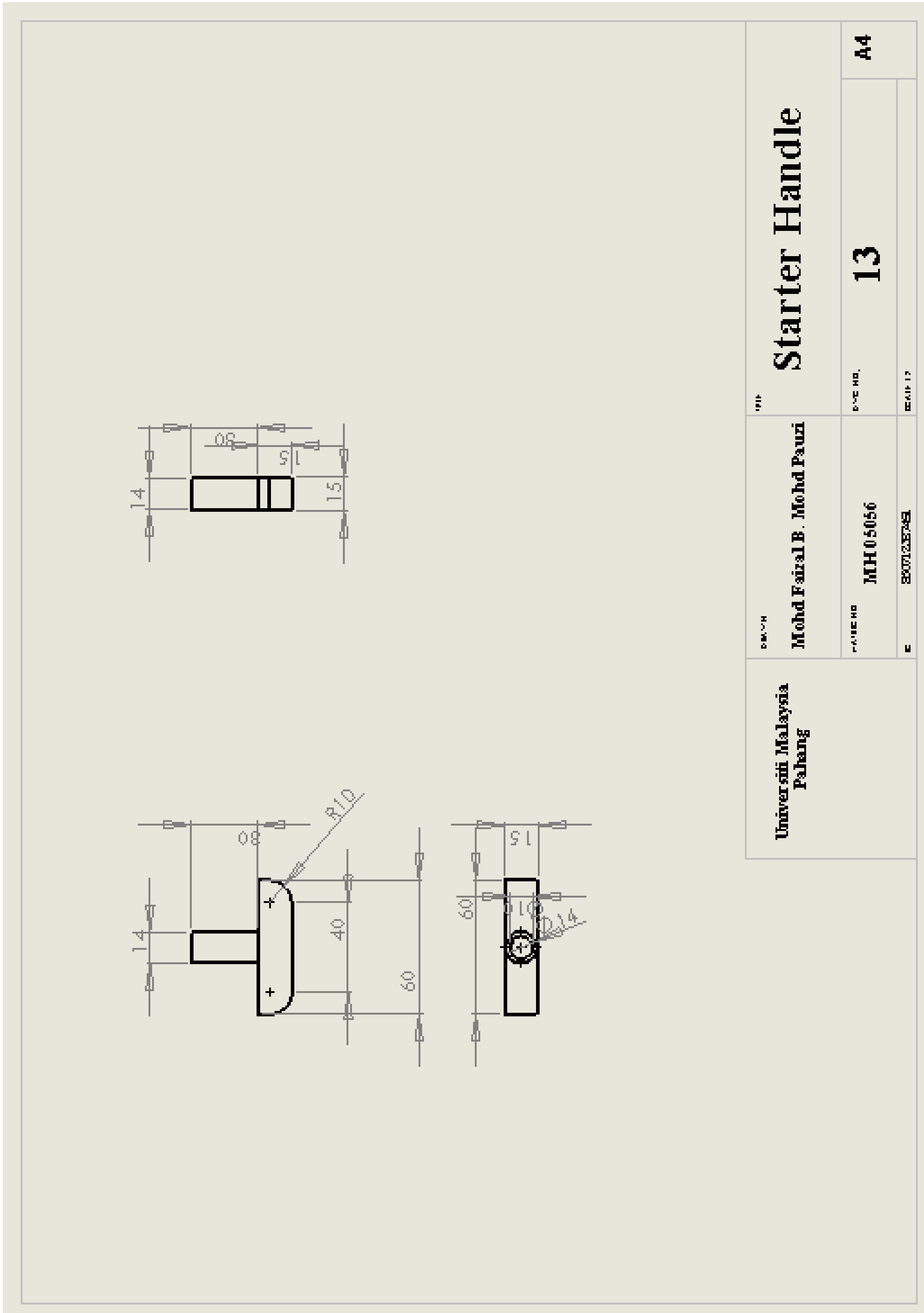

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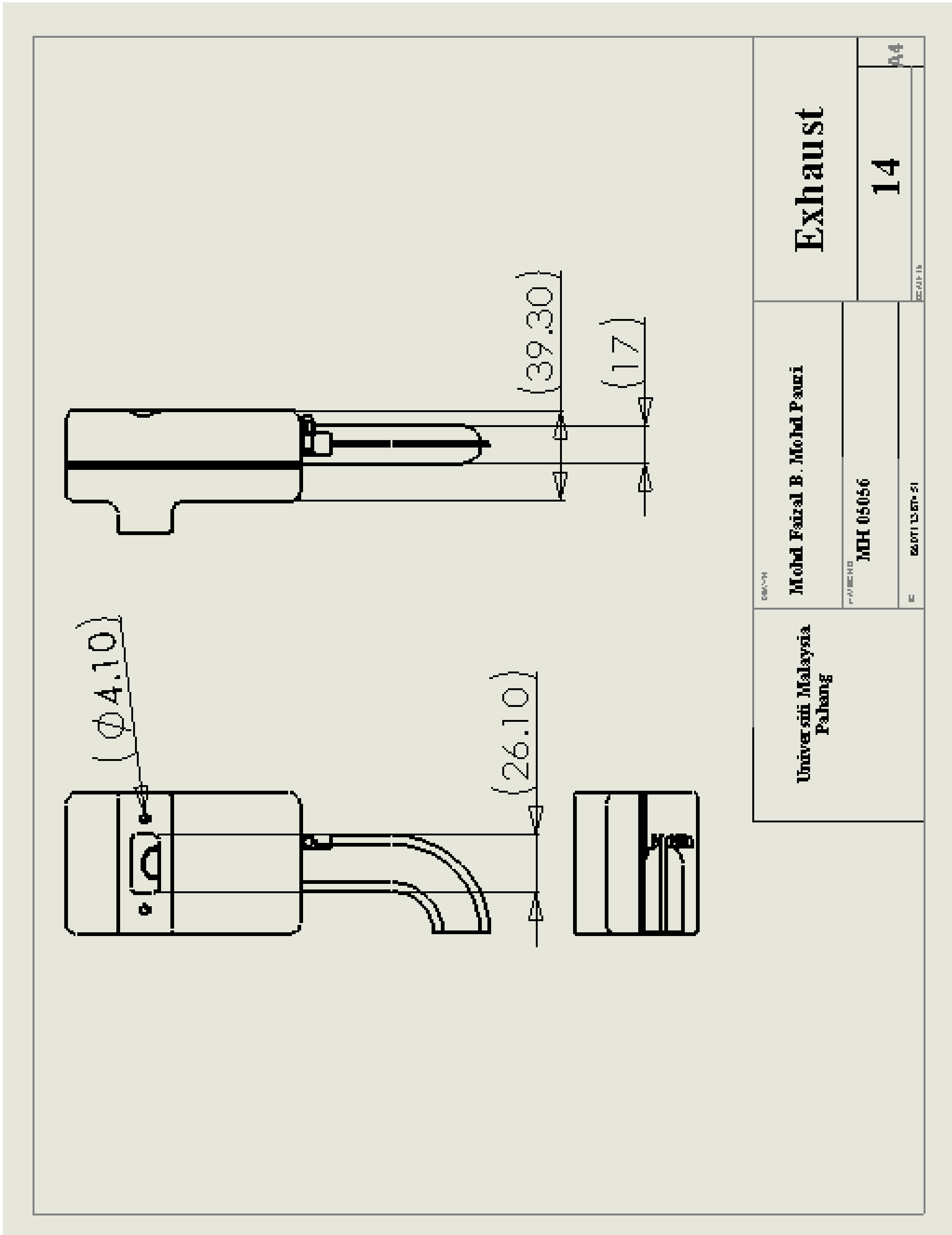
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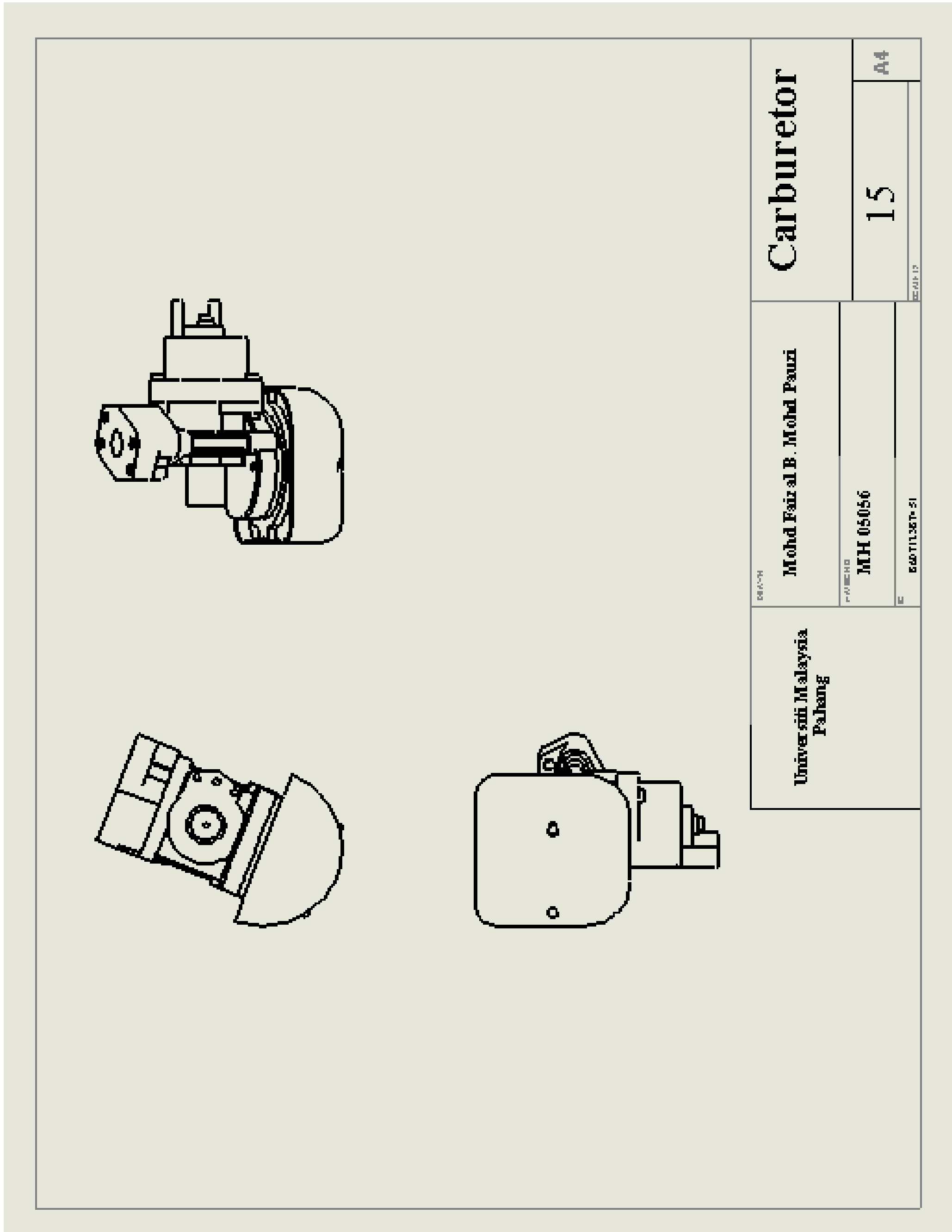
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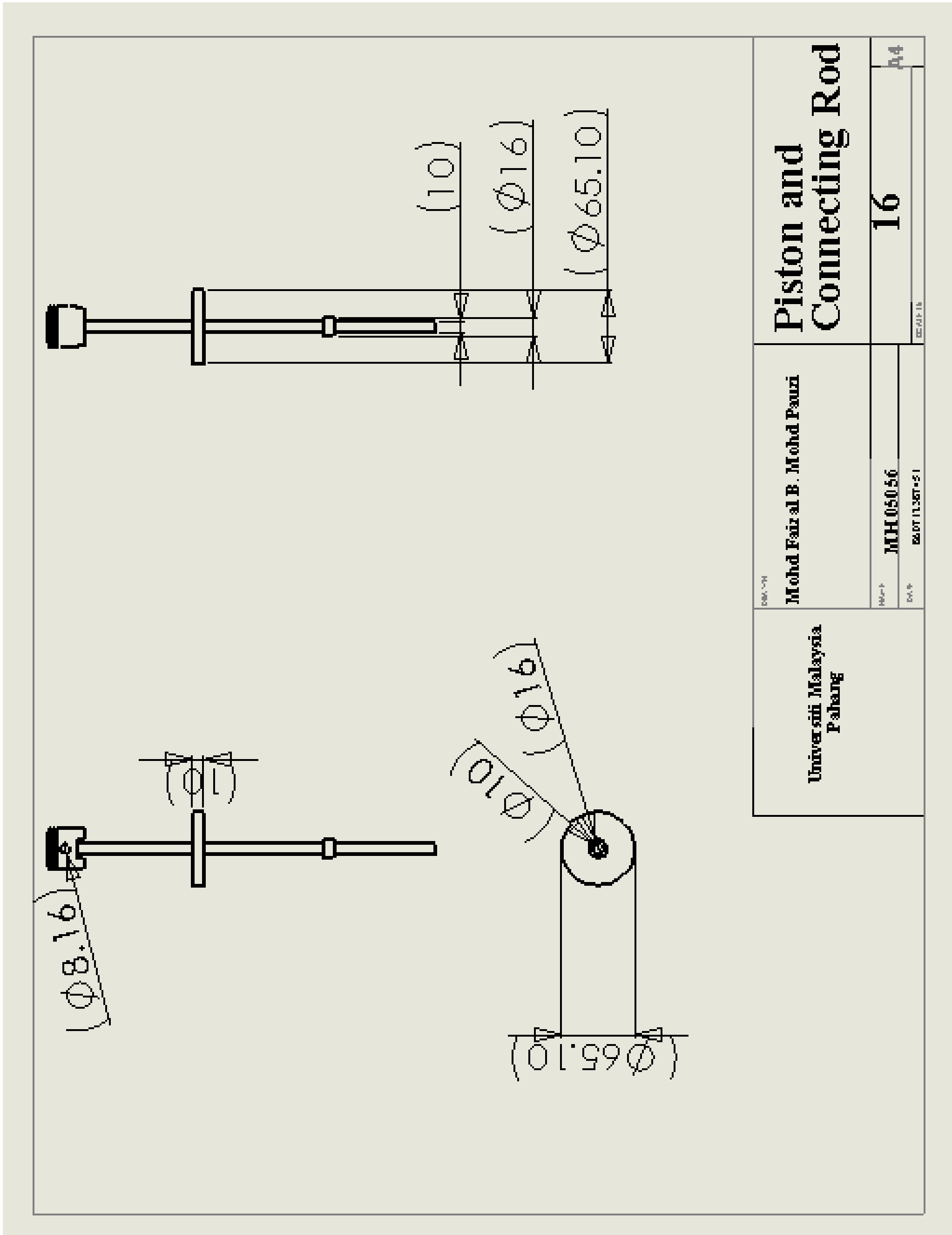
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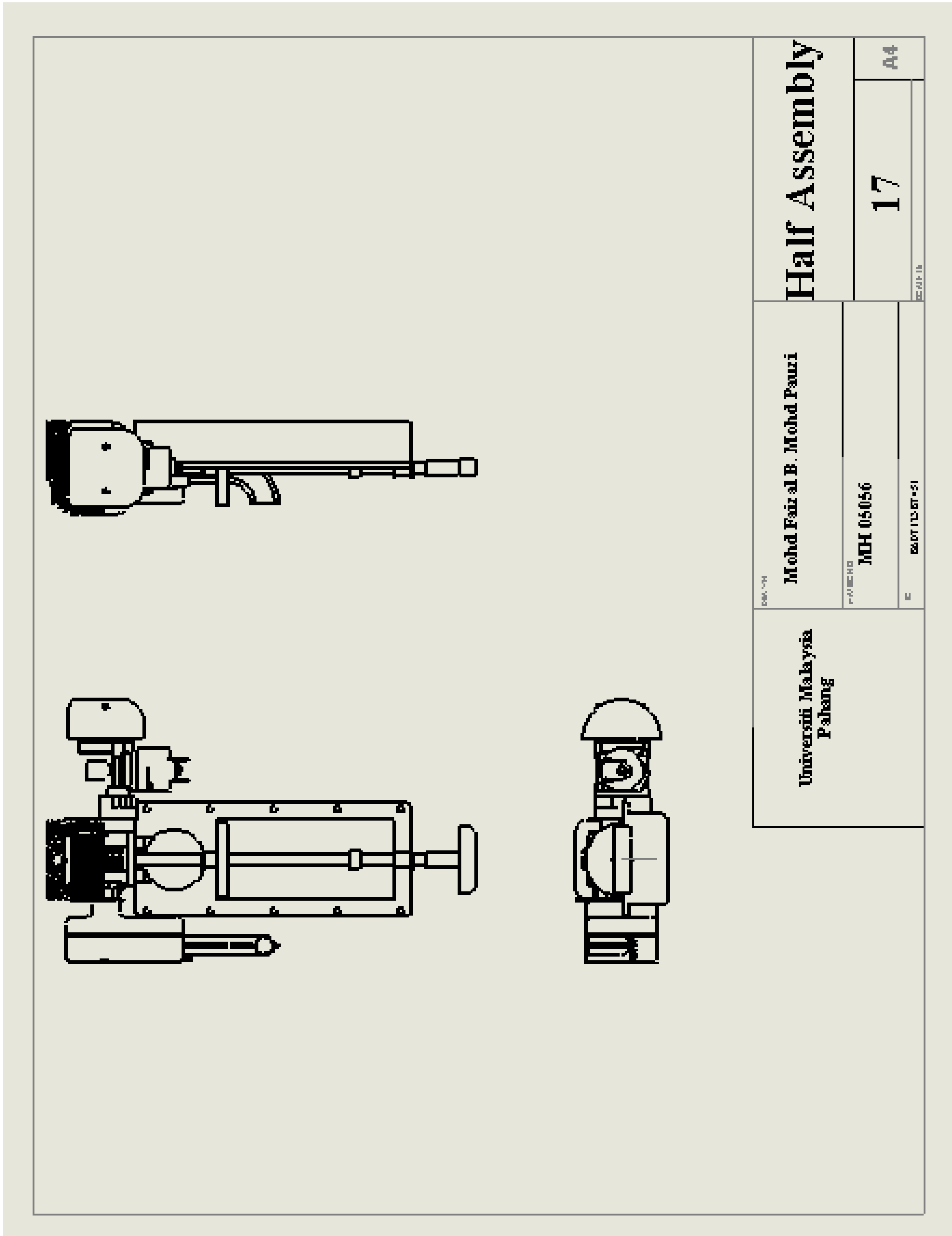
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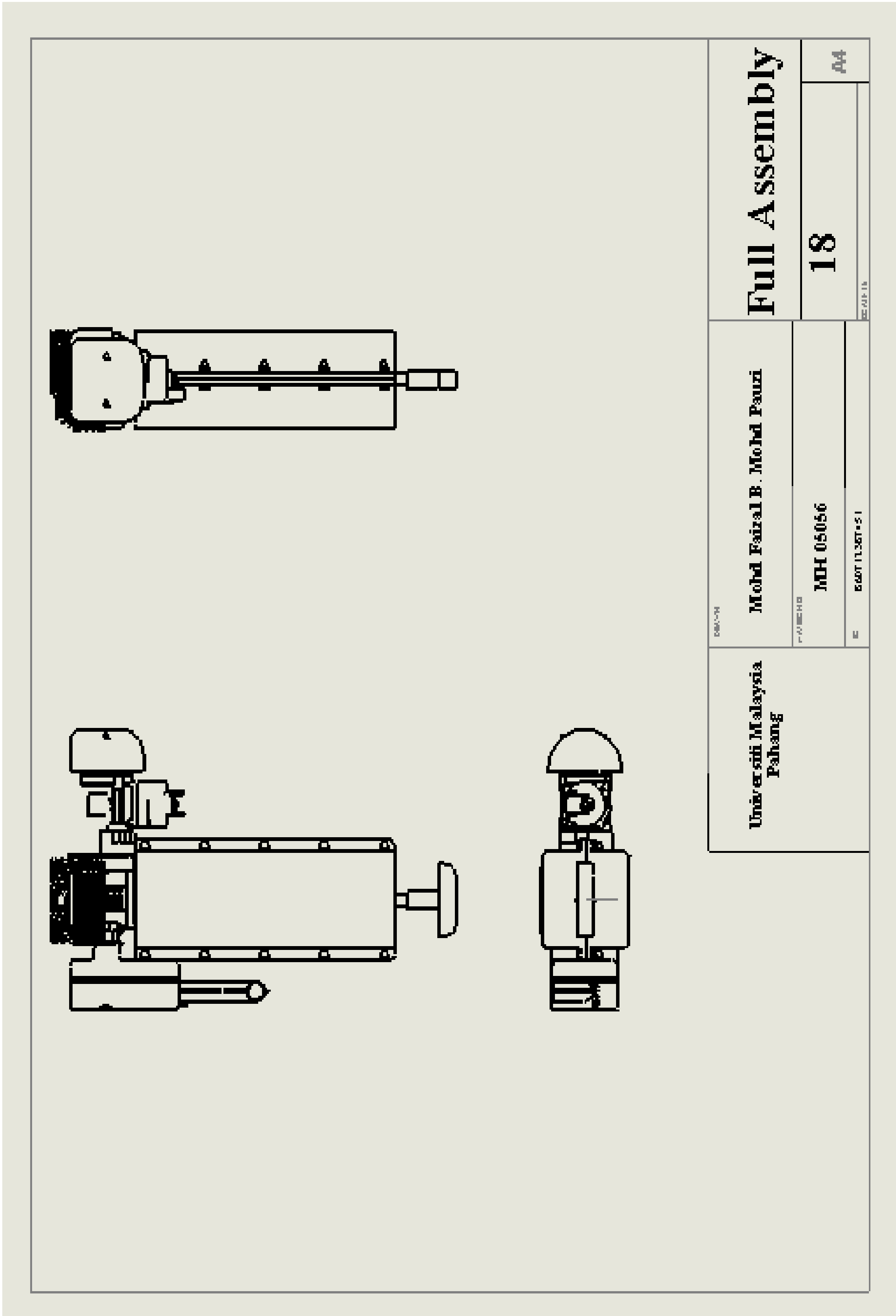
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