CERTIFICATION OF APPROVAL

Cylinder Deactivation of PERODUA Viva 1.0L Engine for Better Fuel Consumption

by

Mohd Bariq Bin Md Yazid

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Approved by,

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(IR. DR. MASRI BAHAROM)

UNIVERSITI TEKNOLOGI PETRONAS

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CERTIFICATION OF ORIGINALITY

This is certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHD BARIQ BIN MD YAZID

ABSTRACT

Fuel consumption is one of the parameter that measures the economical of a particular engine. The PERODUA Viva 1.0L engine has insufficient mileage-contribution for the purpose of competition. Plus, the current engine has unbalance power-to-weight ratio, unnecessary power for acceleration and some parasitic losses that affect the fuel consumption. Therefore cylinder deactivation was introduced to the engine to provide more mileage of the car. The technology of cylinder deactivation is not new and it is currently being employed in many modern cars. Though the technology has been proven by some car manufacturer, it is worth to try it in the other ways. The project is to implement the cylinder deactivation on the engine by totally removing the piston assembly and connecting rod in one of the target cylinder. Then, it will be replaced by a balancer to reduce the problem of engine vibration. The scope of study restricted on the current engine, modeling on some components, running some simple analysis on the proposed system and experiment on the real engine. The process of study began with the understanding of the engine behavior towards the fuel consumption and reviewed some available literatures such are current method of cylinder deactivation, engine balancing theory, parasitic losses in the engine and power-to-weight ratio. A quick calculation on power-to-weight ratio was done to enforce the need of cylinder deactivation. It showed that by performing cylinder deactivation, the car gained back its original value of power-to-weight ratio rather than unnecessary over-power. The involved engine components were crankshaft, connecting rod, piston and piston pin were modeled by CATIA for the purpose of analysis and designing the balancer. In analysis by using ADAMS, only the lateral force develops on the middle crankshaft was investigated. Optimization of the balancer weight was performed to find the best weight to cater the developed lateral force. The balancer has mass of 460g. Then, the balancer was fabricated and assembled on the engine crankshaft. Other modifications had been done on other engine components, namely the intake and exhaust camshaft, ignition system and intake system. This is to ensure the engine is working as expected and normal. From the completed test, the result indicated that cylinder deactivation managed to improve the car mileage about 11% from the original car mileage.

ACKNOWLEDGEMENTS

I would like to express my gratitude to Allah S.W.T. for his permission to let me finish my Final Year Project (FYP) in UTP. First, I would like to appreciate and thank Ir. Dr. Masri Baharom, my FYP supervisor who had guide and assist me in this project throughout 2 semesters. The knowledge and skills that I gained will be fully utilized in my job future. Moreover, the idea of cylinder deactivation that will be discussed throughout this report was contributed from him.

Second, I would like to thank again Ir. Dr. Masri Baharom and Dr. Zainal Ambri Abdul Karim. They were supervisor of PERODUA Eco-Challenge (PEC) 2010. Not to forget, all the technicians especially Mr. M Hazri A Shah Pin and M Fahmi Ahmad Razali, the team members of PEC 2010 for their guidance, support, efforts and contributions to carry out the project together with me till we won the competition.

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ABBREVIATIONS AND NOMENCLATURES

PERODUA	Perusahaan Otomobil Kedua (Malaysia)
SI	Spark ignition
Con-rod	Connecting Rod
CMM	Coordinate-Measuring Machine
WW2	World War 2
GM	General Motors
TDC	Top dead center
BDC	Bottom dead center
WOT	Wide-open throttle
FEAD	Front end accessory drive
P-V	Pressure-volume

CHAPTER 1

INTRODUCTION

1.1 PROJECT BACKGROUND

The project idea was generated from the PERODUA Eco-Challenge 2010. The competition is about to modify the given car so that it reaches a very minimum fuel consumption and travel the longest distance. Among the modifications is to deactivate one of the engine cylinder or called cylinder deactivation.

Cylinder deactivation in its history was used to cater the problem of a large displacement SI engine becomes very inefficient when low output power is required *(Pulkrabek, 2004)*. Fuel consumption can be improved 10% - 20% and hydrocarbon emissions are also reduced *(Murphy, 1998)*. The method was used in various car manufacturers all over the world. Different manufacturers usually give different technologies to implement the cylinder deactivation. The preferred approach is to allow the intake event to occur to bring in air and fuel and to combust it and leave it (uncombusted exhaust) in the cylinder *(Murphy, 1998)*. All of these were implemented on a very large engine capacity such are 6 and 8 number of cylinders.

While the engineers are creating any other technologies for better fuel consumption, it is worth to see the different approach of having cylinder deactivation and the implementation on relative small engine. The method is totally removed the targeted piston assembly in a cylinder and replaced with a balancer. A study and experiment of cylinder deactivation on the PERODUA Viva 1.0L engine will be carried out for the purpose of the competition. Table 1 shows the PERODUA Viva 1.0L original engine specification *(PERODUA, 2010)*, *(Wikipedia, 2009)*:

Table 1 below shows the original engine specification

Engine Specification	Description
Engine type	EJ-VE (Daihatsu)
Valve mechanism	DOHC, 12V (4 valves per cylinder) with DVVT
Total displacement	989cc
Bore x stroke	72mm x 81mm
No. of cylinders	3 (in-line)
Compression ratio	10
Max. output (DIN)	45kW @ 6000rpm
Max. torque (DIN)	90Nm @ 3600rpm
Fuel system	EFI
Fuel tank capacity	36L
Air-Fuel Ratio	14.7 (Tested in UTP)
Fuel consumption	15.1km/L (The car)

1.2 PROBLEM STATEMENTS

The current engine of PERODUA Viva 1.0L has an efficient fuel management system. The Viva car was awarded the most fuel-saving car among Malaysian's car. However, the car mileage is not sufficient to compete in the competition. The reasons are:

1. The high power-to-weight ratio. The car has surplus of power after a process of major weight reduction which makes the original car weight of 800kg reduces to 650kg (assuming the minimum allowable weight in the competition). In simple term, more power means more fuel consumption.
2. Friction loss is still a major contribution to engine parasitic losses.

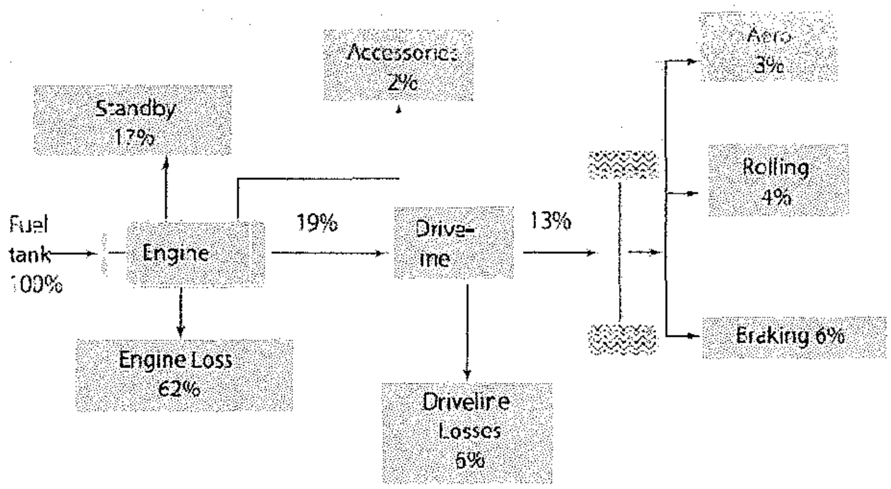


Figure 1 shows the losses of energy percentage for typical car in urban style driving *(Board, 2006)*.

According to figure above, engine contributes higher percentage in power loss for a typical car. Furthermore, unbalance of engine operation after deactivating a cylinder (if simply remove the piston assembly) may cause internal damage and affects the fuel consumption. By using the proposed technique of cylinder deactivation, it is expected to reduce the engine power output (therefore, lower fuel consumption), balance the power-to-weight ratio and reduce the friction losses inside the engine as well as maintain the engine vibration level close to normal.

1.3 OBJECTIVE AND SCOPE

The objectives of this project are:

1. To reduce the engine fuel consumption (improve the car mileage) by proposed cylinder deactivation method.
2. To study the engine crankshaft balancing and reduce the engine vibration after deactivating a cylinder.

The scope of the project is only the proposed cylinder deactivation system in PERODUA Viva 1.0L engine. The system involves the engine crankshaft, connecting rods, piston assemblies and the balancer. No extensive studies on these components to be done.

All the necessary components are modeled (by reverse engineering). The balancer will be designed according to an analysis on crank-motion in that engine. Given a proper 3D axis on the system that consist of crankshaft, connecting rod, piston pin and piston where all the significant forces and joints are imposed, the component forces in lateral direction on the main journal is the main interest and will be analyzed. The investigation will be done before and after removal of the targeted connecting rod and piston assembly as well as with the balancer. The crankshaft balancing theory will be applied. The balancer will be fabricated and installed on the real PERODUA Viva 1.0 engine for experimental purpose. Finally, the car mileage will be tested while running with the cylinder deactivation engine. Engine vibration is observed at this point.

The power-to-weight ratio merely involves simple mathematical equation. The inputs are based on the available data from PERODUA and some approximations. The parasitic losses are discussed without any mathematical calculations and simulations.

CHAPTER 2

LITERATURE REVIEW

2.1 CYLINDER DEACTIVATION

Cylinder deactivation is defined as shut down one or more cylinders in engine so that it will not operate during the engine operation. As the consequence, the fuel can be consumed less. Other name of this technology is variable displacement. This is not a new technology in automobile world. The first attempt on the engine was done by General Motors (GM) Company on Cadillac engine *(Peter, 2005)* during WW2. The company used multi-cylinder engine where the 8 cylinders can be operated on 6- cylinder or 4-cylinder depending on the driving requirement. However, the attempt ended with unpredictable failure due to lack of technology at that time.

Then, Mitsubishi developed its own variable displacement engine which proved that this kind of technology is working *(Mitsubishi, 2003)*. They attempted on their 4-cylinder engine where it will run on 2-cylinders under certain conditions. At a time later, they developed a technology called MIVEC-MD (Mitsubishi Innovative Valve timing Electronic Control system – Modulated Displacement). It was an improved technology that enables the switch from 4 to 2 cylinders to be made almost imperceptibly. In MD mode, the MIVEC engine utilizes only two of its four cylinders, which reduces significantly the energy wasted due to pumping losses. In addition, power loss due to engine friction is also reduced. Depending on conditions, the MIVEC-MD system can reduce fuel consumption by 10–20 percent; although some of this gain is from the variable valve timing system, not from the variable displacement feature. Modulated Displacement was dropped around 1996.

Until now, many other companies introduced their own variable displacements technology. Honda came out with their Variable Cylinder management and GM with their Active Fuel Management. Typically, cylinder deactivation is simply keeping the intake and exhaust valves closed through all cycles for a particular set of cylinders in the engine. Depending on the design of the engine, valve actuation is controlled by one of two common methods *(Gable & Christine, 2010)*

1. By pushrod designs:

   When cylinder deactivation is called for—the hydraulic valve lifters are collapsed by using solenoids to alter the oil pressure delivered to the lifters. In their collapsed state, the lifters are unable to elevate their companion pushrods under the valve rocker arms, resulting in valves that cannot be actuated and remain closed.

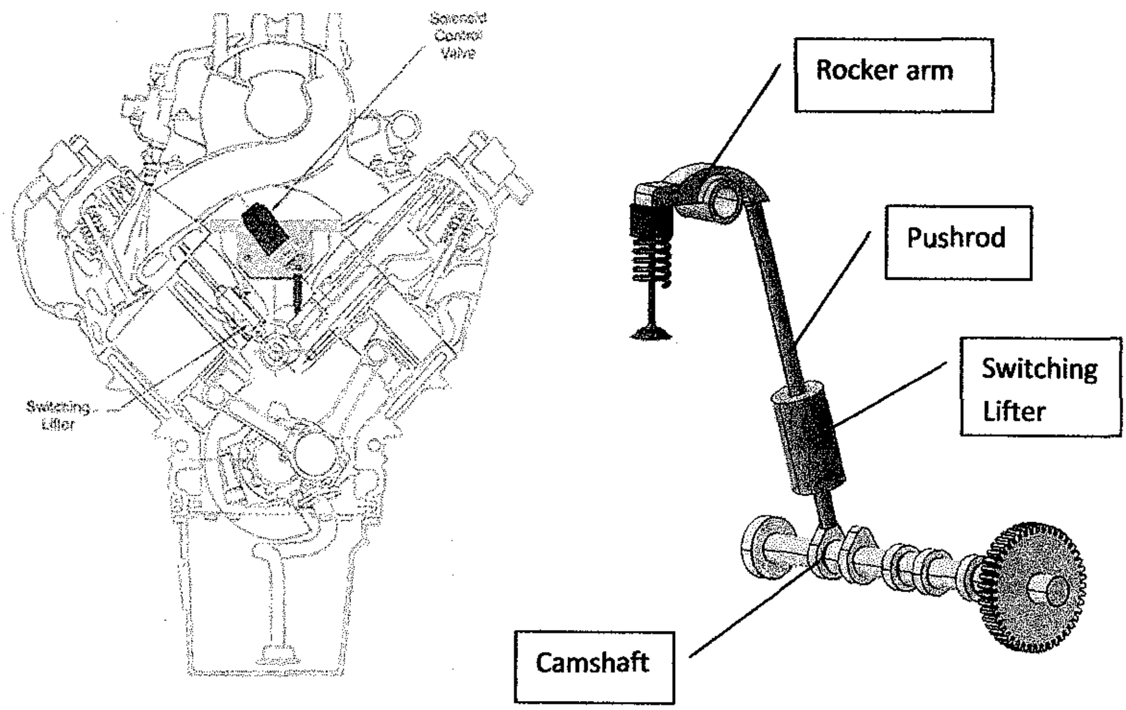


Figure 2 shows pushrod design (left) *(J.D. Power and Associates, 2010)* and an illustration by author (right)

2. Overhead cam design

Generally a pair of locked-together rocker arms is employed for each valve. One rocker follows the cam profile while the other actuates the valve. When a cylinder is deactivated, solenoid controlled oil pressure releases a locking pin between the two rocker arms. While one arm still follows the camshaft, the unlocked arm remains motionless and unable to activate the valve.

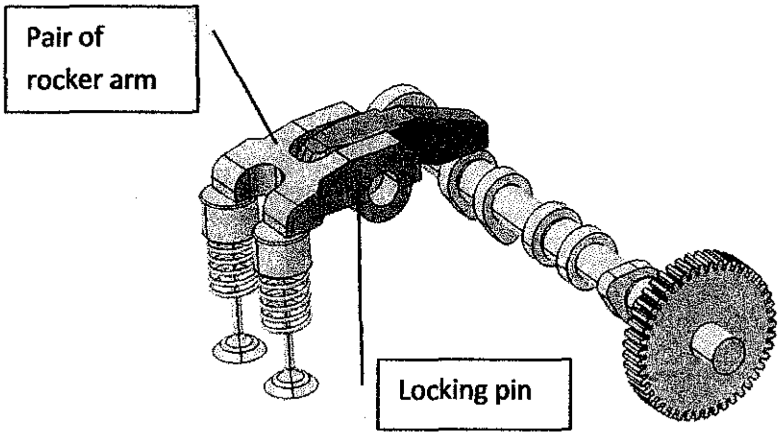


Figure 3 shows an illustration by author of overhead cam design as to deactivate cylinder

Deactivating cylinders at light load forces the throttle valve be opened more fully to create constant power, and allows the engine to breathe easier. Better airflow reduces drag on the pistons and the associated pumping losses. The result is improved combustion chamber pressure as the piston approaches top dead center (TDC) and the spark plug is about to fire. Better combustion chamber pressure means a more potential and efficient charge of power is unleashed on the pistons as they thrust downward and rotate the crankshaft *(Christine&ScottGable).*

2.2 POWER-TO-WEIGHT RATIO

Power-to-weight ratio is a measurement of actual performance of any engine or power sources. It is also used as a measurement of performance of a vehicle as a whole, with the engine's power output being divided by the weight (or mass) of the vehicle, to give a metric that is independent of the vehicle's size *(Wikipedia, 2010)*.

2.3 PARASITIC LOSSES

Theoretically, the power loss by the piston movement can be through both ways:

1. Pumping loss *(Mechadyne International, 2010)*:

   When operating at part load the throttle restricts the airflow into the engine, reducing the volumetric efficiency, and as a result the air pressure in the intake manifold falls significantly below atmospheric pressure. In order to draw air from the manifold into the cylinder, the piston is required to do work against the manifold depression. The work done by the piston is a result of the pressure differential between that of the manifold and the crankcase. The losses contribute about 10% at $\lambda=1$ *(Dirk Hofmann, 2006)*.

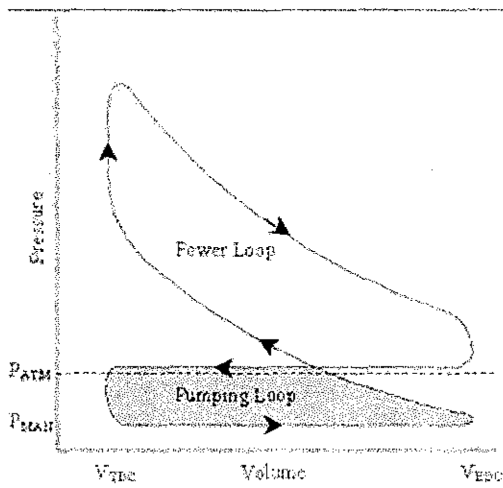


Figure 4 shows PV diagram for a throttled SI engine. Throttled SI engines to exhibit very poor efficiency under part load conditions compared to their efficiency under full load operation.

2. Friction between piston and cylinder wall(liner)

A literature shows that about 10-20 % of the indicated power of internal combustion engines is lost due to mechanical factors *(Anderson, 2005)*. The mechanical efficiency, for example the ratio of the brake power to the indicated power, of internal combustion engines usually varies between 0.4 and 0.9. The frictional losses at the sub-mechanisms of an internal combustion engine are responsible for a significant proportion of the mechanical power losses of the engine. Friction losses arise mainly at the pistons, piston rings, bearings and the valve mechanism. Almost 50% of the losses are related to the piston assembly.

Deactivating cylinders at light load forces the throttle valve be opened more fully to create constant power, and allows the engine to breathe easier. Better airflow reduces drag on the pistons and the associated pumping losses. The result is improved combustion chamber pressure as the piston approaches top dead center (TDC) and the spark plug is about to fire. Better combustion chamber pressure means a more potential and efficient charge of power is unleashed on the pistons as they thrust downward and rotate the crankshaft *(Christine&ScottGable)*.

After disable one of the cylinders, the effect on the engine should be considered. The consequence such as unbalance force distribution of crankshaft will lead to severe engine vibration. The PERODUA Viva 1.0L engine has 3 cylinders (in-line) which well-known has poor-balancing configuration (unless the engine equipped with balancer shaft). Thus, disabling one of the cylinders sounds like to have severe unbalance force-distribution. Therefore, the project serves to reduce this problem. A study will be conducted to figure out the way to balance the crankshaft by the balancer.

CHAPTER 3

METHODOLOGY / PROJECT WORK

3.1 PROCEDURE

The project involves the procedure as follow:

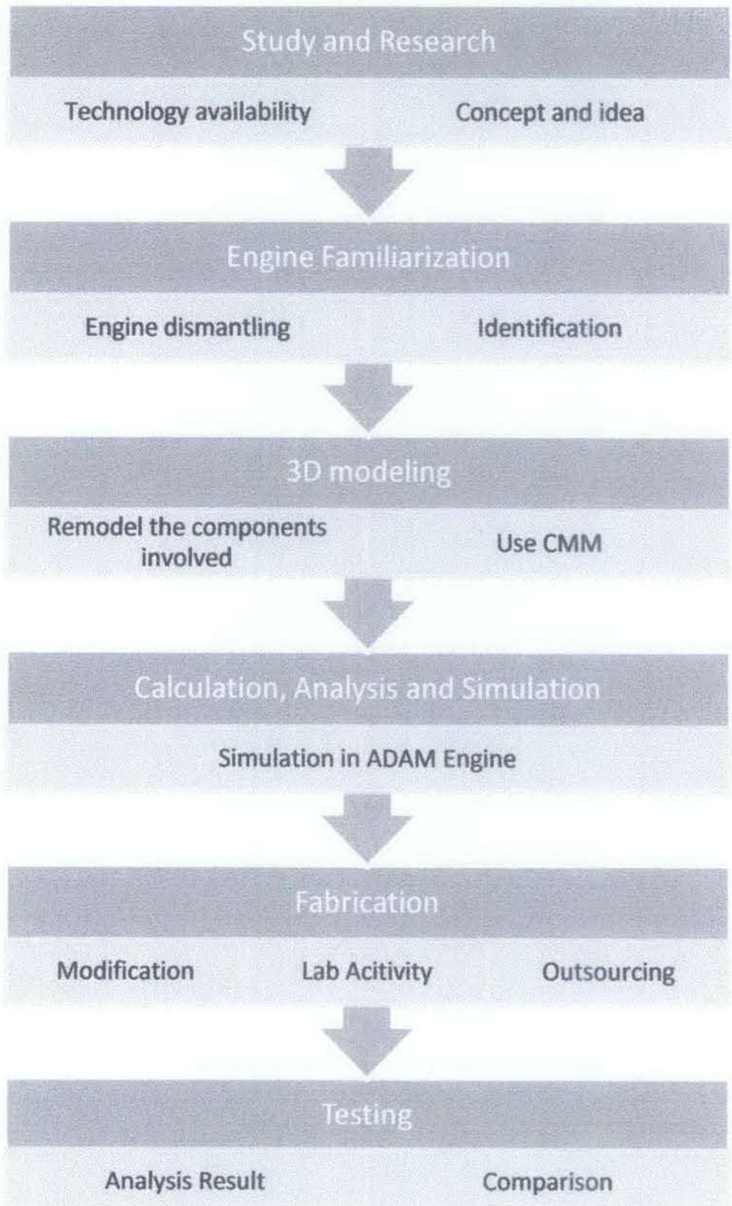


Figure 5 shows the procedures in modification of the engine

3.2 THE CRANKSHAFT

The engine had been dismantled for internal parts identification and study. The following figure shows a crankshaft of the engine.

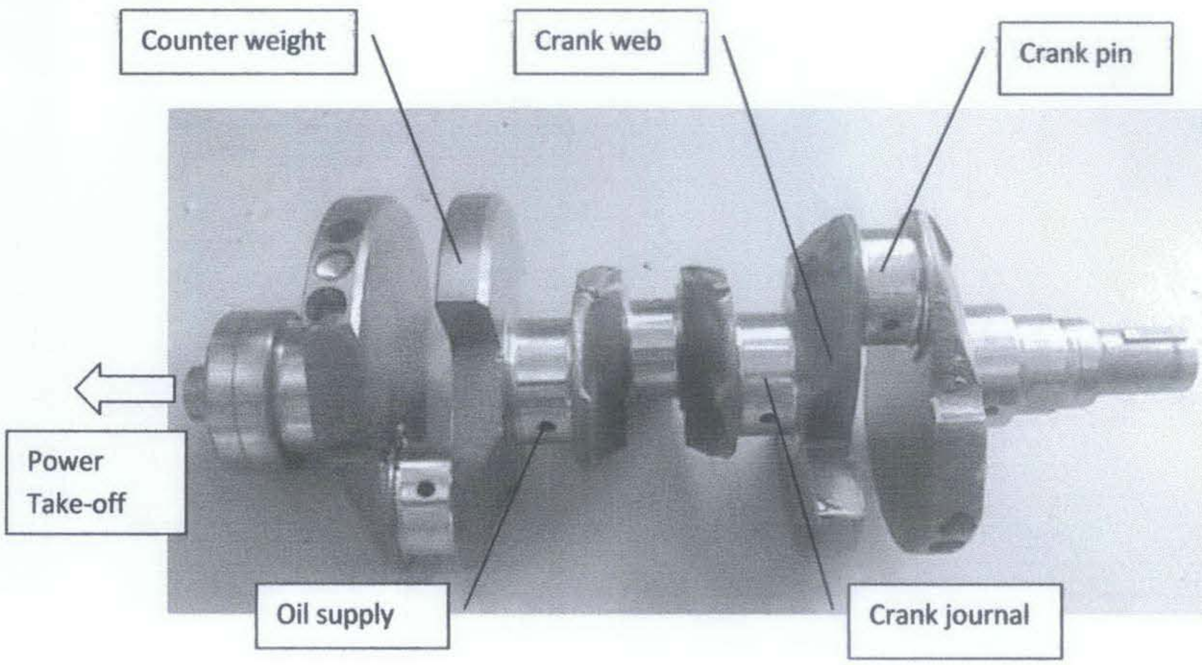


Figure 6 shows the crankshaft of PERODUA 1.0L engine

3.3 POWER-TO-WEIGHT RATIO

The calculation of power-to-weight ratio is discussed here. The method is very clear. A sample power output based on both 2 and 3 cylinders engine will be compared. The sample powers are taken from a thermodynamic analysis. The analysis merely focuses on a single cylinder Otto cycle. However, the engine power can be attained by multiplication of the number of cylinders. The power here is referred to brake power output. The use of brake power output because it represents the power output measured directly from the crankshaft.

Several assumptions are made to ease the calculation:

1. Engine operates at WOT
2. Running at 6000rpm
3. Mechanical efficiency of 86%
4. Type of fuel is gasoline with heating value of 43000kJ/kg
5. Combustion efficiency of 95%
6. No residual exhaust gasses
7. Specific heat ratio of k=1.35
8. Constant volume specific heat, c_v=0.821kJ/kg-K
9. Initial temperature, T_1=30°/303K
10. Initial Pressure, P_1=100kPa

Refer to appendices for complete calculation procedure. The results are processed with vehicle weight input as to compute the power-to-weight ratio for both conditions. To further discuss on the matter of engine power, the minimum power required for the car is computed as well. This power represents the ideal minimum power needed by the car to travel during the competition with the following assumptions:

1. Drag coefficient, $C_D = 0.36$
2. Frontal area, $A = 2.26m^2$
3. Air density, $\rho = 1.2$ kg/m^3
4. Vehicle speed, V = 40km/h=11.1m/s
5. Rolling resistance coefficient, $C_{rr} = 0.01$
6. Constant losses at drivetrain
7. Any other forces exerted on the car are neglected (except drag and rolling resistance forces)

The calculation procedure may refer to appendices. These values are considered to suit the competition condition.

3.3 ENGINE BALANCING

Experiments have shown that approximately two-thirds of the connecting-rod mass contribute to the rotating mass component of the rods, whereas only one-third of the rod's mass effectively reciprocates, thus:

(rotating con-rod mass = two-thirds of the con-rod mass)

$m_c = 2/3C$

where m_c = rotating con-rod (kg)

C = con-rod mass (kg)

Therefore, the total rotating inertia components acting on the crankshaft are:

(total rotating mass = crankpin mass + big-end mass)

$m = m_j + m_c$

$= \rho(\pi/4)d^2L + 2/3C$

Based on this formula, it is understood that the balancer that is going to be made has reference weight of two-third of con-rod. By referring this, it eases to find and optimize the balancer weight later.

3.4 3D MODELING

The important parts such as the crankshaft, piston, piston pin and connecting rod are modeled. During the measurement process, Coordinate-Measuring Machine (CMM) is used for accuracy as well as manual measuring method by using Vernier caliper. The use of CMM only is only focusing on the complex shapes and profiles on the crankshaft. For example, the counterweight profile is quite complex to determine by manual measuring method.

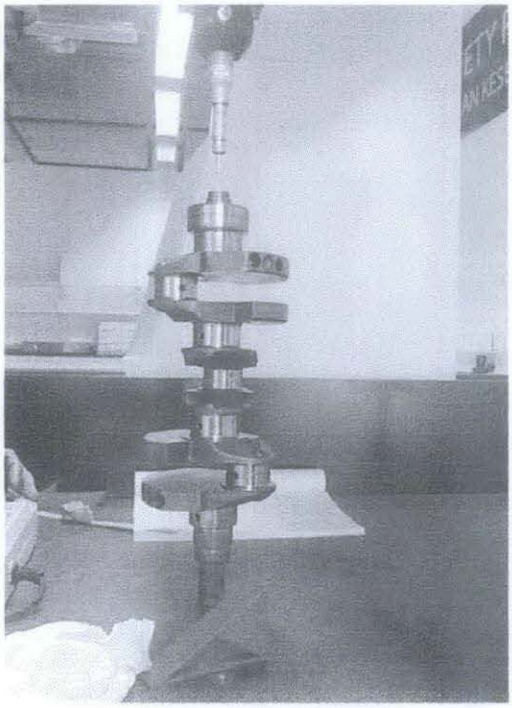


Figure 7 shows crankshaft being measured by CMM

The crankshaft is the most crucial component that has to be controlled carefully. The counterweight width, thickness, angle and so on must be as exact as possible. The component will undergoes some analysis that needs the 3D model to be exact to resemble the real object. Mass of the crankshaft is another factor to be considered. The mass in 3D model should almost similar with the real one. However, it is expected that there will be some variation of the value between 3D model and real object as well as between software.

The connecting rod, piston pin and piston are considered as auxiliary model which to assist the determination of unbalance force in crankshaft. Therefore, the components were modeled with not high-precision. However, their masses were carefully controlled. All 3D models were made in CATIA V5R16.

3.5 SIMULATION AND ANALYSIS

The simulation and analysis are performed in ADAMS/View. The 3D models are imported from CATIA. The crankshaft, connecting rods, piston pins and pistons are assembled as a completed assembly in and 3 cylinders engine. On the crankshaft itself, piston locates in the origin (towards the FEAD system) is designated as number 1 and followed by 2 and 3.

For kinematic inspection, different components are linked with joints and motion generator is applied on the crankshaft. The piston assemblies will reciprocate properly while crankshaft is rotating.

Once checked, the force is applied on top of the piston to represent the combustion force. The forces are inspected on each of the crankshaft main journal. The only main focus is the forces occur in lateral direction due to rotational of crankshaft.

To simulate the cylinder deactivation condition, the middle piston assemblies on the crankshaft is selected to be removed. While maintaining the applied force on top of the piston, the lateral force of the middle crankshaft main journal is inspected. This is the easiest way to capture the condition on the crankshaft when one set of piston assembly is removed. To further investigation, the balancer is placed on the middle crankpin. Again, the middle main journal is inspected to compare the previous condition. The result helps to improve the mass and design of the balancer.

3.6 FABRICATION AND ASSEMBLY

The final design of balancer had been completed. The design was ready to be fabricated. Due to some constraints, the fabrication was outsourced. At the same time, the engine had been dismantled to install the balancer inside the engine. Figure 8 shows the engine being dismantled by outsource mechanic.

Figure 8 shows 2 outsource mechanics dismantle the engine

The cylinder deactivation would not work by simply deactivating a cylinder. The other components and system must take into account. By considering the operation of combustion in a cylinder, the intake and exhaust port have to be ceased as well as the ignition system. Otherwise, complication might occur on the engine operation such are:

1. Fuel will be wasted as it is still being injected into the target cylinder.
2. The possibility of engine oil from crankcase spill out through the opening of intake and exhaust valve due to piston absent.
3. The possibility of engine oil burning due to active ignition system in the target cylinder.
4. The exhaust product gets inside the engine due to back pressure.

Hence, intake port, exhaust port and ignition system at the target cylinder must be deactivated as well.

3.7 TESTING

The method to test the engine is simply by a driving course. The car is driven on a track with unknown length. However, the mileage is determined directly from the car odometer. The car is filled with 1 liter of fuel (Primax RON95 as stated in the competition rules) and run till the car is out of fuel. The test is conducted several times for verification. The result is compared with the latest previous test on the car mileage before the cylinder deactivation is applied.

Figure 9 shows the car was under testing

CHAPTER 4

RESULTS AND DISCUSSION

4.1 POWER-TO-WEIGHT RATIO

The lesser number of cylinders, the lower engine power output. For the sake of understanding, how much the power can be reduced is most interested. Besides, this preliminary calculation may prove the cylinder deactivation provides effective reduction in engine fuel consumption.

The result reveals that engine power reduced about 33% of original power when one of the cylinders is shutting down. The following table shows the comparison between 2 and 3 cylinders of power output. The speed of 6000rpm is selected according to maximum power output produced by the actual engine available in the engine specification table. (Refer appendix)

Table 2 shows the power output comparison of 2- and 3-cylinders

	3- Cylinders	2-Cylinders
Power output computed at 6000rpm (refer appendices)	76.7kW	51.1kW
Actual power output at 6000rpm	45kW	*30.2kW
Computed Power-to-weight ratio based 800kg (original vehicle weight)	0.056	0.038
Computed Power-to-weight ratio based 650kg (modified vehicle weight)	0.069	0.046

*This value is hypothetical value which is generated based on actual power output

However, the calculated power does not match with the actual power. The different is due to many reasons such are assumption made during computation, the use of Otto cycle model which is not exactly represents the actual cycle and others. The most significant point here is the amount of reduction in power after deactivating one cylinder. Based on the actual power output (45kW), it is expected to have a reduction by about 33% as well. Thus, the estimation value for 2-cylinders actual power output would be 30.2kW.

According to the table, the car already has a balance power-to-weight ratio at 0.056 and this will be a reference. In the case where none of cylinders is deactivated, the power-to-weight ratio is increased, about 0.069 due to massive weight reduction on the car. This increment is unnecessary as mentioned earlier in objectives that the car is ran not against the speed or time but the distance.

When cylinder deactivation is introduced, the power-to-weight ratio comes to balance back, about 0.046 but a bit lower. In other words, the car will run at normal condition with lower power requirement and thus, save the fuel.

On the other hand, the computed minimum power required by the car to travel during the competition reveals as low 1.4kW. This value included the drag force and rolling resistance force exerted on the moving vehicle. Only these types of forces are being considered. Theoretically, the vehicle does need much power produced by the engine. It only needs the engine which produces slightly more than the value computed above. However, the ideal condition is not always true for the real condition. The value might be greater than computed.

From the computed power of 2 cylinders earlier, the engine still has surplus of power, much higher than computed power required. Bear in mind, the power of 2 cylinders is based on WOT condition. In real driving condition, the engine do not operates at WOT but partially. Thus, both values are not really accurate to compare directly. The emphasized point here is the sense of difference between these values. Hence, there is potential to further reduce number of cylinder to reduce more engine power output. However, the project is limited to remove one cylinder.

4.2 3D MODELING

The output from CMM was coordinate values of particular profiles. Figure below shows the printed screen of CMM interface for crankshaft counterweight profile. The sample coordinates from CMM may refer to appendix.

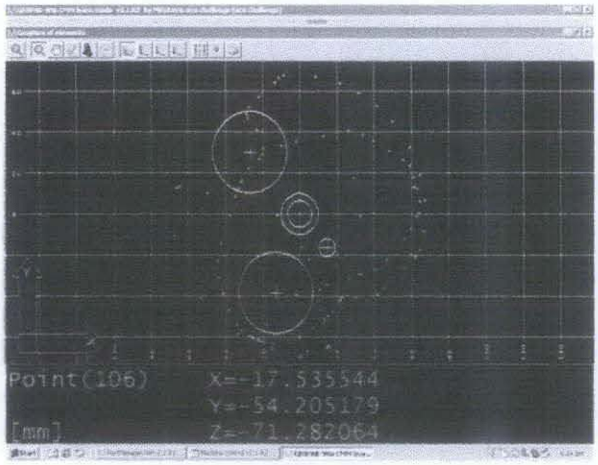
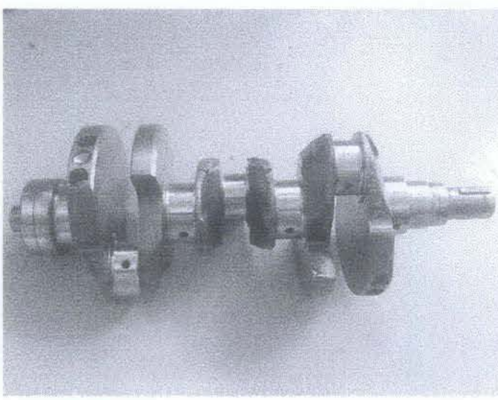
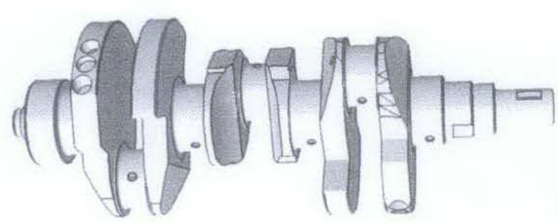


Figure 10 shows the interface of CMM and position of coordinates for crankshaft

Based on these coordinate data and the manual-measuring method, the crankshaft was modeled in 3D by using CATIA. The following figures show the comparison between actual crankshaft and 3D model in CATIA.





Actual Crankshaft
Mass = 8.5kg
Material = Cast Iron

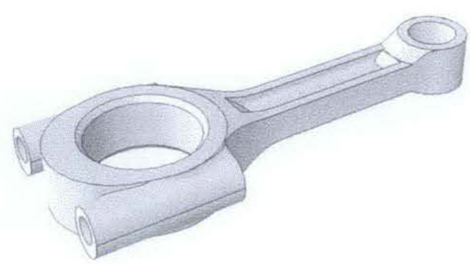
Modeled Crankshaft
Mass = 8.6kg
Material = Cast Iron

Figure 11 shows the actual and 3D model of crankshaft

The actual crankshaft has a total mass of 8.5kg while the measured mass in CATIA was 8.6kg. The deviation of mass value from the actual is about 1% which still acceptable in the analysis.

The difference was caused mainly by some dimensions were took manually. Besides, the following figures show the piston, piston pin and connecting rod model in CATIA.

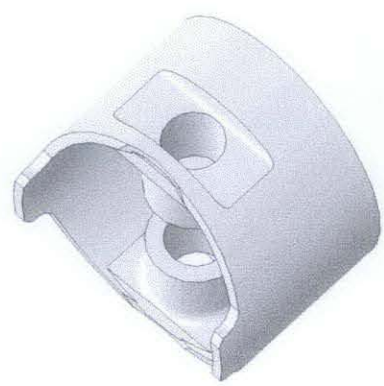
Actual Connecting Rod
Mass = 425g
Material = Cast Iron



Modeled Connecting Rod
Mass = 425g
Material = Cast Iron

Figure 12 shows the actual and 3D model of connecting rod

Actual Piston
Mass = 195g
Material = Aluminum



Modeled Piston
Mass = 195g
Material = Aluminum

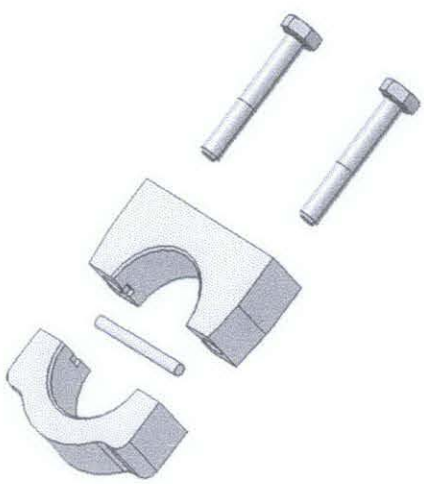
Figure 13 shows the actual and 3D model of piston

Actual Piston Pin
Mass = 65g
Material = Steel

Modeled Piston Pin
Mass = 66g
Material = Steel

Figure 14 shows the actual and 3D model of piston pin

The final design of the balancer had been completed. The design is almost similar with the big end of connecting rod so that it will fit properly on the crankpin. Figure below shows the complete design of balancer.



The balancer consists of 4 components:

1. Upper half-ring
2. Lower half-ring
3. 2 bolts
4. Retainer rod

Figure 15 shows the balancer

Safety is the most crucial aspect which had been taken into account. One of the considerations was the dynamic clearance. The dynamic clearance is the space between the moving parts and static parts. For example, the clearance between rotations of the balancer inside the engine block wall was inspected. Unwanted clash will occur if the clearance is not much. However, there was no specific definition or limit considered for setting this clearance. It was provided just enough to make the balancer run freely.

Besides, another feature is retainer rod. The use of this retainer is to lock off the balancer from rotating about the crankpin by itself. It was designed to fit tightly inside the oil passage on the crankpin. This small safety feature avoids major possible damage if the balancer loose. Figure below shows how the retainer locks the balancer on the crankpin.

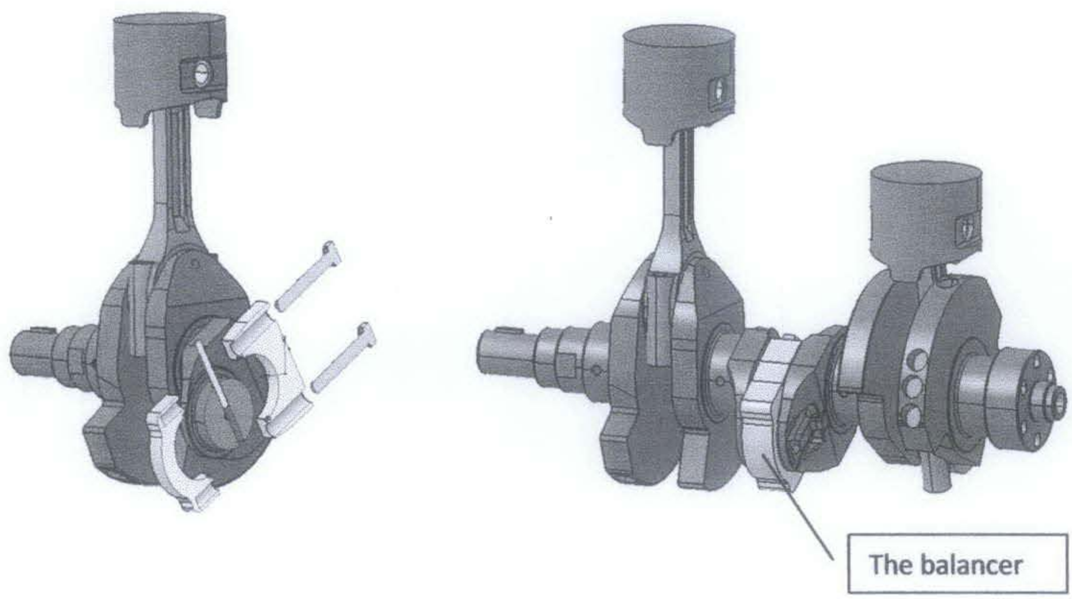


Figure 16 shows the function of retainer

The selected material was mild steel. The cost and availability are the upmost factors being considered for material selection. Instead of easy for manufacturing, the mild steel is relatively heavier. Thus, smaller size of balancer can be fabricated out of mild steel. Refer appendix for complete balancer drawing specification.

4.3 SIMULATION AND ANALYSIS

The imported models from CATIA to ADAMS/View were assembled as shown in next figure. The joints applied between each component.

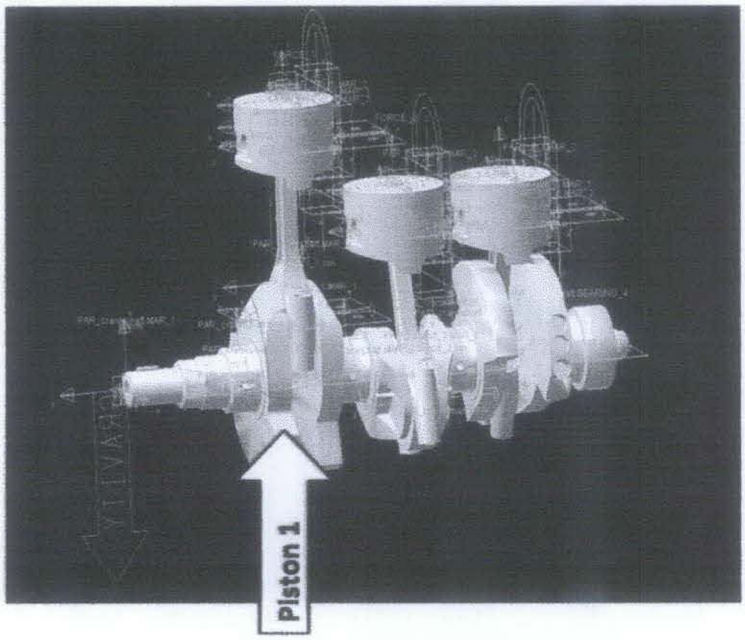


Figure 17 shows the assembled model in ADAMS/View

The force was applied on top of piston 1. To justify on this conduct, the simulation model in ADAMS/view is done rigid body model instead of flexible body model. It means that some behaviors on the crankshaft such as elasticity of body could not be recorded. As the result, only a single force sufficient to represent in this case. The force represented a combustion force which derived from P-V relation equation of Otto cycle. However, this force was an initial force which assuming that the piston was moved by compressed air trapped in the chamber. In other words, the piston was decompressed and compressed the air. The derivation of force from the P-V diagram is shown in appendix

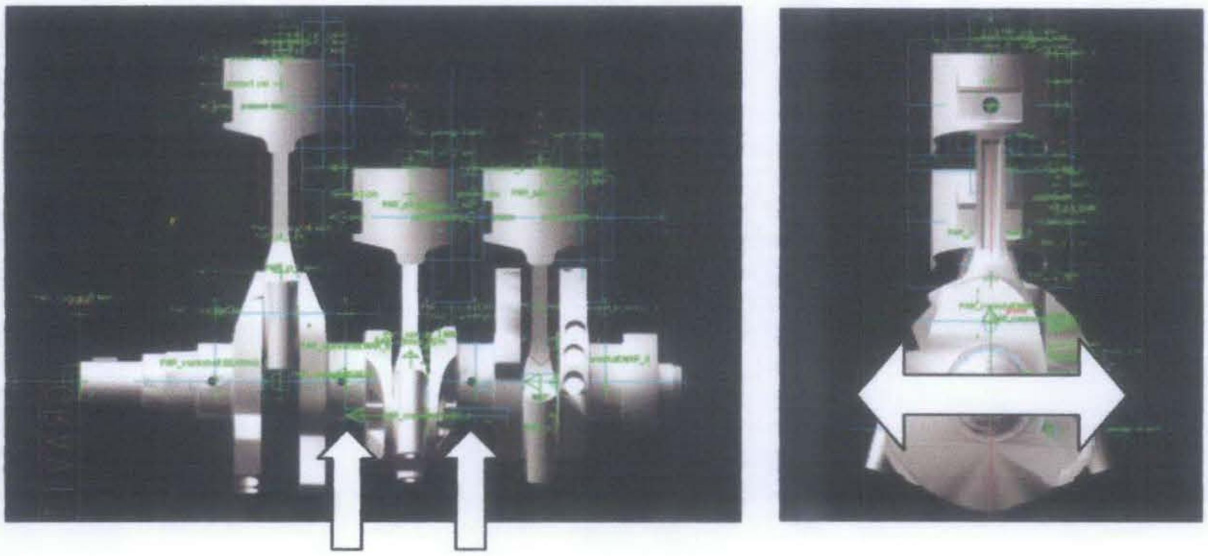
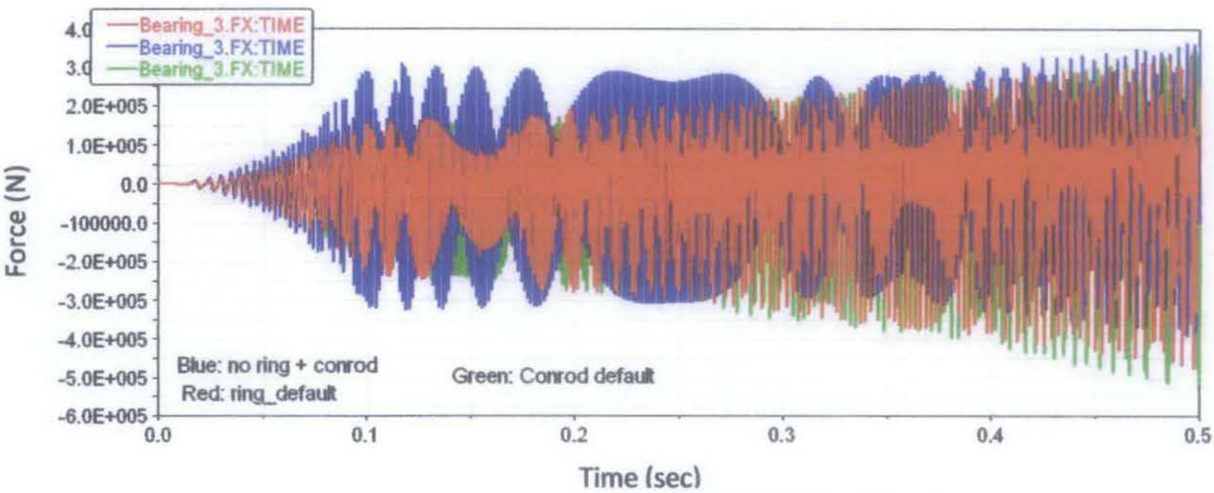


Figure 18 shows where the lateral forces that will be inspected

The pressure applies on the piston generates force on the crank journal. The lateral forces were most concerned and investigated. The result shows in the following figure.

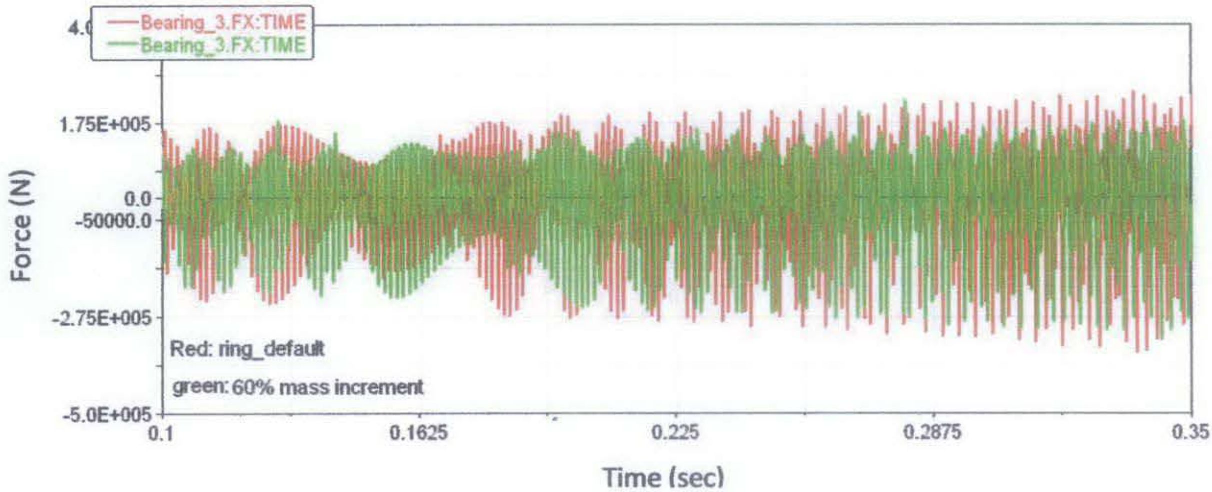


Legend:
Green : Crankshaft with normal condition
Blue : Crankshaft with no con-rod and balancer
Red : Crankshaft with balancer

Figure 19 shows the comparison between 3 conditions of crankshaft

According to the graph, the crankshaft ran with high lateral forces when no con-rod or balancer was attached to it. However, these forces were reduced when the balancer was attached. Despite the amount of force almost similar with the one with con-rod, the idea was very clear.

The balancer helps to reduce the lateral forces. The unstable condition at the end of the graph (both figure 19 and 20) was assumed by the author due to developed forces (inertial and lateral forces) during the simulation. The fundamental point here is the obvious different in amplitude of forces. Figure below shows the optimization result of balancer weight.



Legend:
Green : Crankshaft with 60% of balancer weight increment
Red : Crankshaft with 2/3 of con-rod mass

Figure 20 shows the final result of balancer weight optimization

After optimization of balancer weight, both increase and decrease, the final result shows in figure above. At 60% of weight increment, the lateral forces seem to be reduced. This gave the weight of balancer is about 460g. The value was referred back to the design stage for modification on the balancer model.

4.4 FABRICATION AND ASSEMBLY

The balancer fabrication was outsourced. A fabricated balancer is shown in figure below.

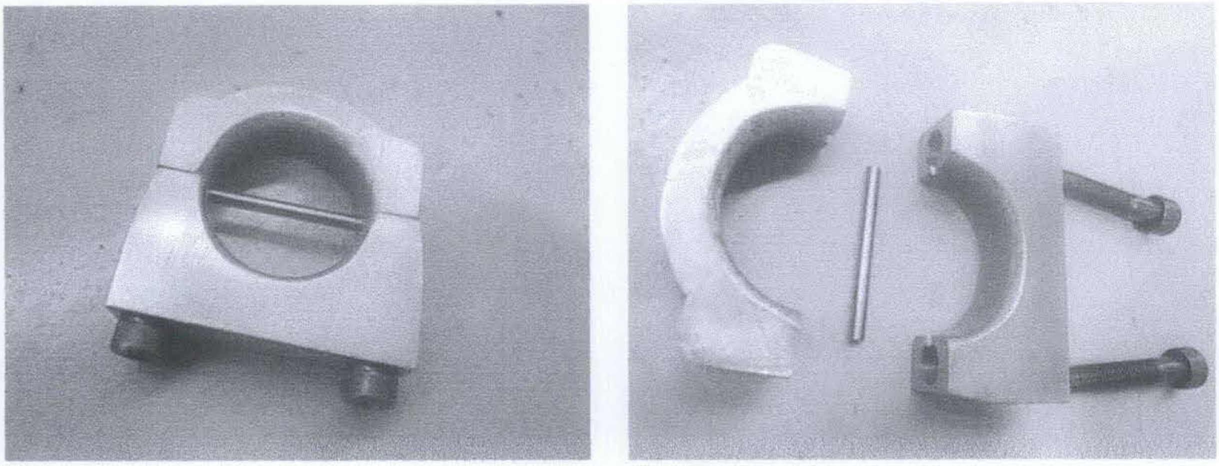


Figure 21 below shows the fabricated balancer

There was an issue during the assembly. The retainer rod was fabricated slightly loose. According to the design requirement, the rod should fit the oil passage on that particular crankpin. This is important because:

1. The retainer rod should fit firmly so that it can hold the balancer neatly
2. Despite to fit, it blocks the engine oil coming from the engine block. The oil originally is used to lubricate the con-rod bearing. Otherwise, the engine oil experiences pressure drop due to leak.

It was solved by putting high-temperature sealant together with the retainer rod. The car now has been ready for a test on the road.

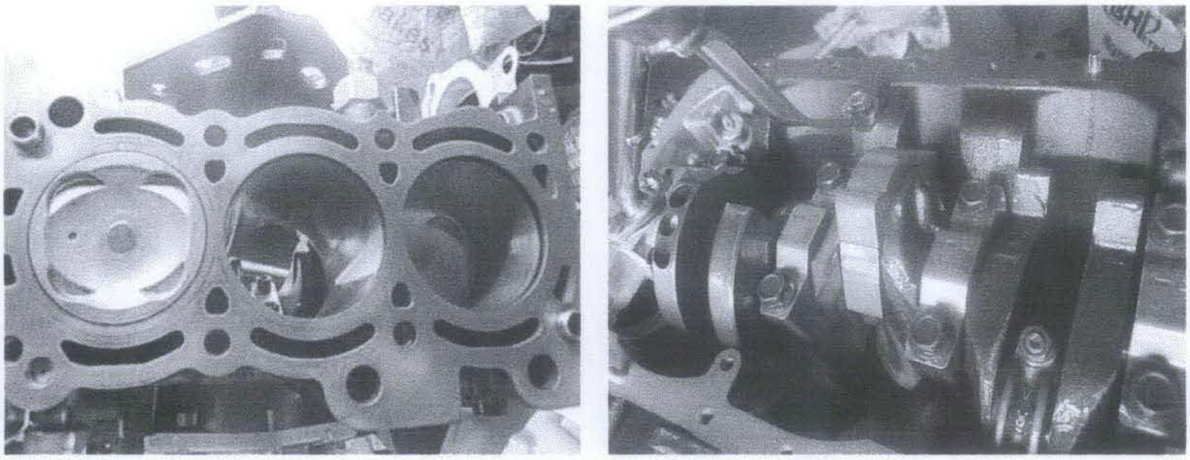


Figure 22 shows the balancer on the crankshaft and removed piston assembly in the middle

Apart from balancing the crankshaft, other systems which are intake, exhaust, ignition and fuel-supply system were modified. This to ensure the deactivated engine is working properly. The modification on ignition system was just simply plug off the wire socket to spark plug. Let the spark plug inactively remains on the head to seal the combustion chamber. For intake and exhaust port, the cam lobe at target cylinder was grinded (to roundness instead of having oval shape) for both intake and exhaust camshaft. This is to ensure the camshaft does not push the valves during engine operation. Hence, both intake and exhaust valves remains close. The following figure shows the rounded cam lobe for both intake and exhaust camshaft.

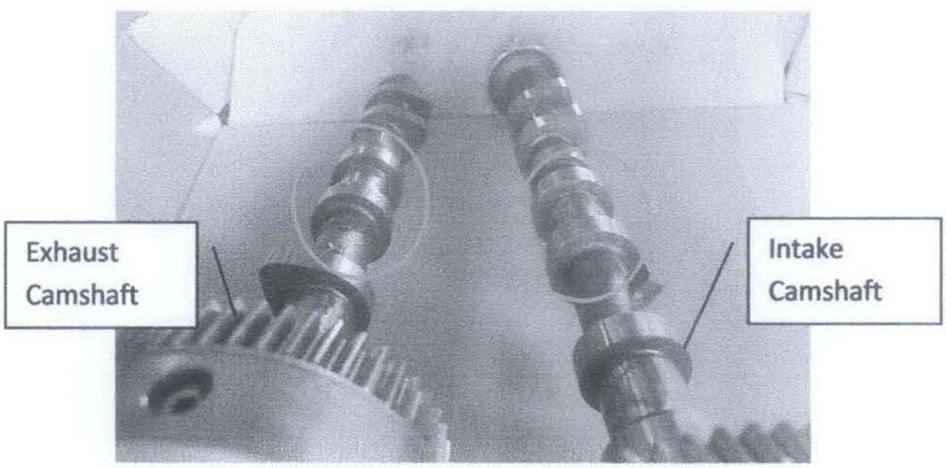


Figure 23 shows the modified exhaust and intake crankshaft

Besides, the injector also had been ceased by simply plug off the wire socket to injector. Thus, the injector would not receive the input from engine control unit (ECU) and remains inactively seal the intake runner. In summary, the modification on PERODUA Viva 1.0L engine is depicted in the following figure.

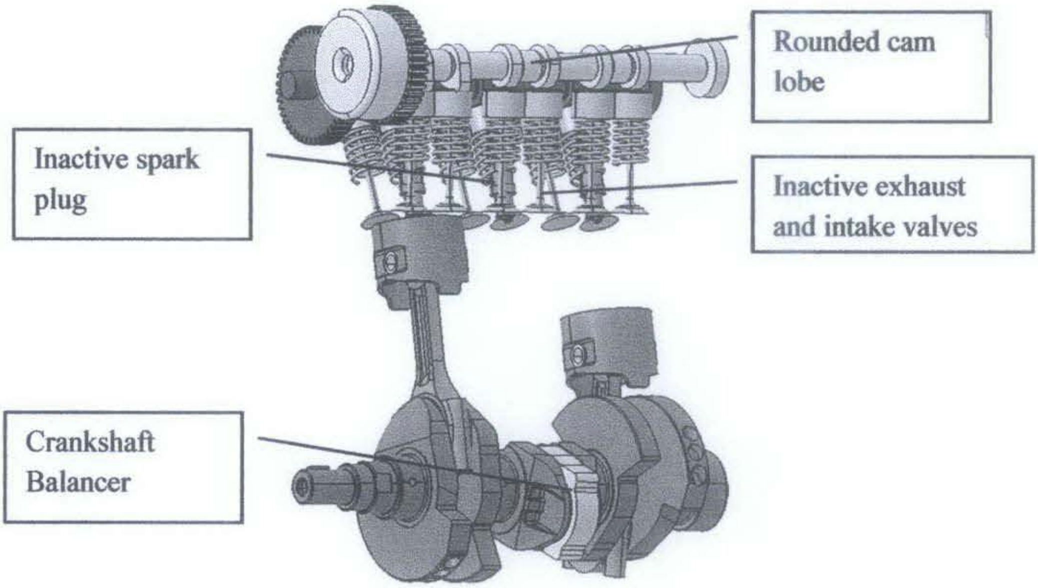


Figure 24 shows modification summary on PERODUA 1.0L engine for cylinder deactivation

4.5 TESTING

The following table shows the mileage improvement of the car after having cylinder deactivation.

Table 3 shows the car mileage improvement

	Without Cylinder Deactivation	With Cylinder Deactivation
Mileage	26km	28km

Based on the manufacturer specification, the claimed fuel consumption is 15.1km/L. In order to maintain the consistency of result, a benchmark test had been conducted. The result depicted the car could go 18.1km/L on original. This will be the baseline reference. The car underwent continuous modification before cylinder deactivation which made its fuel consumption improved up to 26km/L. According to the table, cylinder deactivation gives another improvement about 8% (2km/L). If the improvement is computed based on the original car fuel consumption (18.1km/L), the 2km/L of margin represents about 11% of improvement. This value proved the literature study of cylinder deactivation before which mentioned that the cylinder deactivation alone capable to reduce the fuel consumption in the range of 10 to 20%.

In term of engine vibration, the car had run more than 200km on road and never shows any failure. On naked eyes, engine operation looks normal even runs with 2 cylinders. However, further investigation on vibration level was unable to carry on due to some constraints.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

The proposed method of cylinder deactivation was totally different with the available technology nowadays. Based on the study and literature review, the PERODUA Viva 1.0L engine has potential of deactivating of one of its cylinder. Even though the engine is relatively small to employ this technology, the preliminary computation showed that quite a big amount of power reduction can be attained by deactivating a cylinder. Theoretically, the power reduction by 30% could save the fuel up to 20% based on the literature review made. After the modification was done on the engine, the car managed to travel for another 2 km after deactivating its cylinder. The improvement represents 11% of fuel saving. Thus, the objective of reducing the car fuel consumption was achieved successfully.

However, the vibration level is still not being investigated completely. Since the first engine fire up after deactivating its cylinder, it never encounters any problem during operation. By naked eye, the engine vibration seems the same as original.

For recommendation, there are several matters could be discussed for future improvements, which are:

1. Further study on effect of vibration thoroughly to increase lifespan of the engine.
2. The possibility of reducing more cylinders to further reduce fuel consumption.

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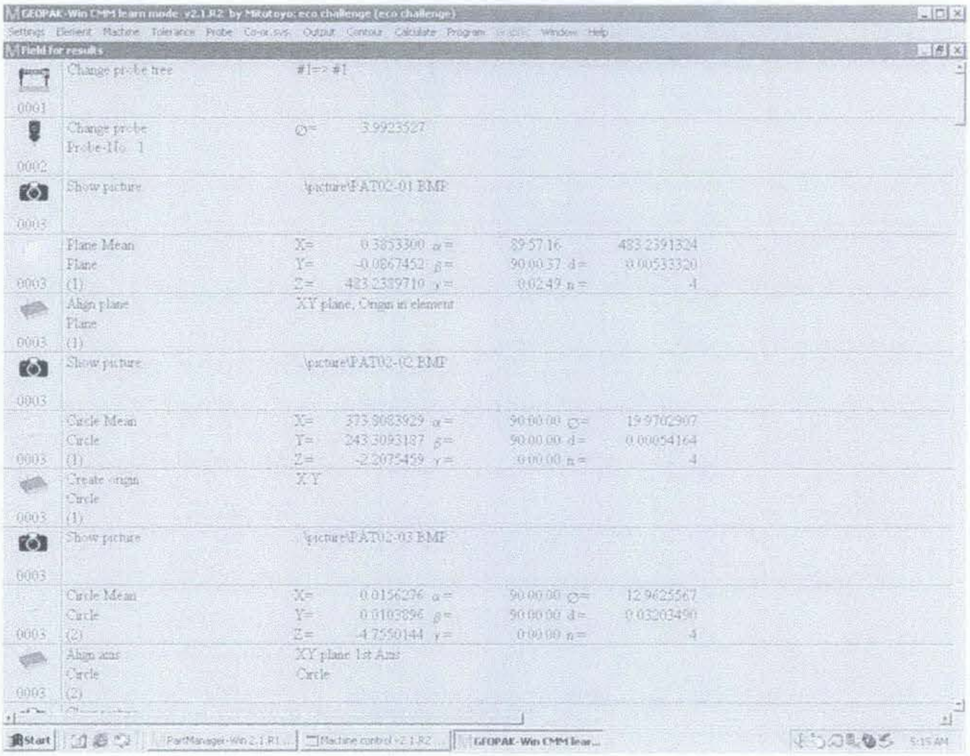
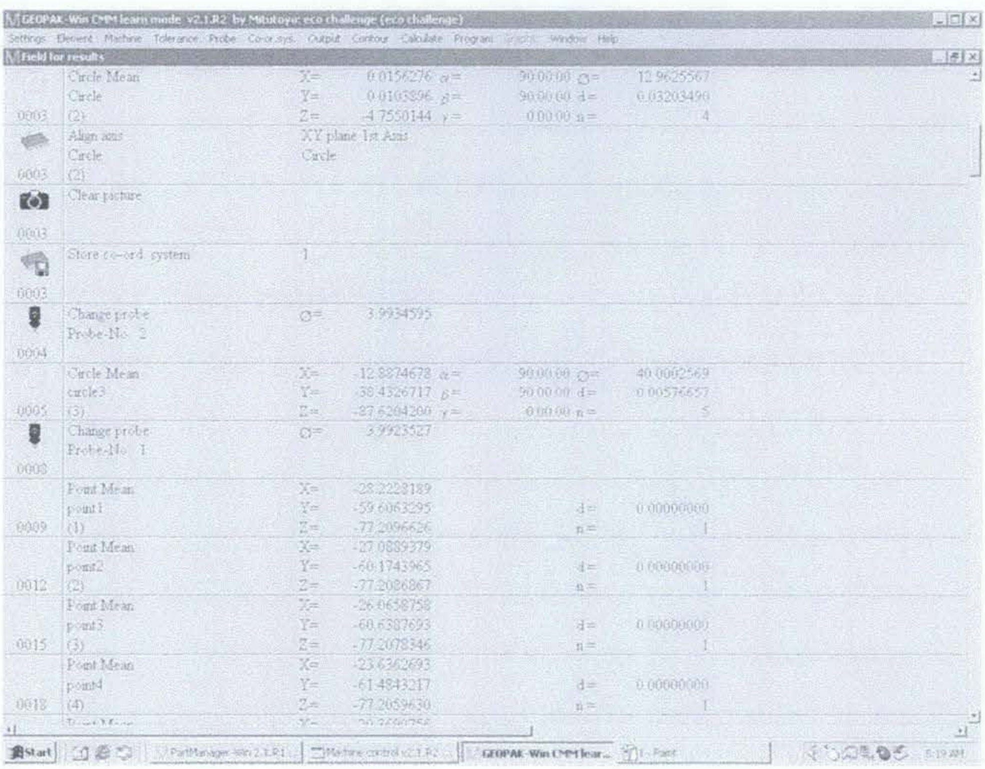
APPENDICES

1. SAMPLE COORDINATES FOR CRANKSHAFT FROM CMM





2. POWER-TO-WEIGHT RATIO COMPUTATION

Thermodynamic analysis on a single cylinder:

Displacement volume,

V_d = 0.989L = 0.329L = 0.000329m^3

Clearance volume,

$$r_c = \frac{V_1}{V_2} = \frac{V_c + V_d}{V_c} = 10.0$$

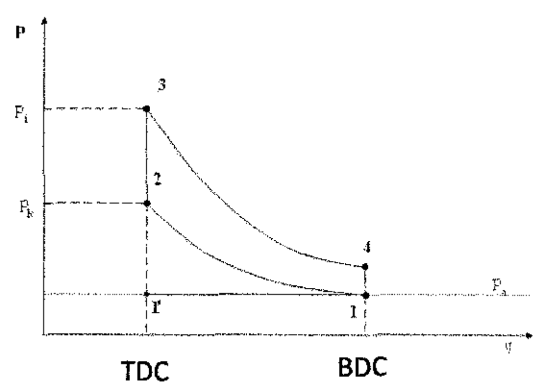
Therefore:

$$V_c + V_d = 10.0V_c$$

$$V_d = 9V_c$$

$$V_c = \frac{0.000329}{9}$$

$$= 3.65 \times 10^{-5} m^{-3}$$



Otto Cycle

<u>At State 1:</u>

T_1=30^{o}C=303K

$$P_1=100kPa$$

$$V_1=V_d+V_c=0.000329+3.65\text{x}10^{-5}=3.655\text{x}10^{-4}\ m^3$$

By Ideal Gas Law, mass of the mixture:

$$\begin{aligned} m_m &= \frac{P_1V_1}{RT_1} \\ &= \frac{(100)(3.655\times10^{-4})}{(0.287)(303)} \\ &= 4.203\times10^{-4}\,kg \end{aligned}$$

At State 2:

$$\begin{aligned} \frac{P_2}{P_1} &= \left(\frac{V_1}{V_2}\right)^k \\ P_2 &= (100)(10.0)^{1.35} \\ &= 2238.7kPa \end{aligned} \qquad \begin{aligned} \frac{T_2}{T_1} &= \left(\frac{V_1}{V_2}\right)^{k-1} \\ T_2 &= (303)(10.0)^{1.35-1} \\ &= 678.3K \end{aligned}$$

$$\begin{aligned} V_2 &= \frac{mRT_2}{P_2} \\ &= \frac{(4.203\times10^{-4})(0.287)(678.3)}{2238.7} \\ &= 3.654\times10^{-5}\,m^3 \end{aligned}$$

The mixture composition:

$$\begin{aligned} m_m &= m_f + m_a \\ m_a &= \left(\frac{14.7}{15.7}\right)(4.203\times10^{-4}) \\ &= 3.94\times10^{-4}\,kg \\ m_f &= \left(\frac{1}{15.7}\right)(4.203\times10^{-4}) \\ &= 2.63\times10^{-5}\,kg \end{aligned}$$

At State 3:

The heat added during one cycle:

$$Q_{in} = m_f Q_{HV} \eta_c = m_m c_v (T_3 - T_2)$$

$$(2.63\times10^{-5})(43000)(0.95) = (4.203\times10^{-4})(0.821)(T_3 - 678.3)$$

$$T_3 = 3791.7K$$

Since, $V_3 = V_2 = 3.654\text{x}10^{-5}\ m^3$

By Ideal Gas Law, the pressure at 3:

$$\frac{P_3}{P_2} = \frac{T_3}{T_4}$$

$$P_3 = (2238.7)\left(\frac{3791.7}{678.3}\right)$$

$$= 12514.3kPa$$

At State 4:

$$\frac{T_4}{T_3} = \left(\frac{1}{r_c}\right)^{k-1}$$

$$T_4 = (3791.7)\left(\frac{1}{10.0}\right)^{1.35-1}$$

$$= 1693.7K$$

$$\frac{P_4}{P_3} = \left(\frac{1}{r_c}\right)^{k}$$

$$P_4 = (12514.3)\left(\frac{1}{10.0}\right)^{1.35}$$

$$= 559kPa$$

Since, $V_4 = V_1 = 3.655\text{x}10^{-4}\ m^3$

Work produced in the isentropic power stroke for one cylinder in one cycle:

$$W_{3-4} = \frac{mR(T_4 - T_3)}{1-k}$$
$$= \frac{(4.203\times10^{-4})(0.287)(1693.7-3791.7)}{1-1.35}$$
$$= 0.723kJ$$

Work absorbed during the isentropic compression stroke for one cylinder in one cycle:

$$W_{1-2} = \frac{mR(T_2 - T_1)}{1-k}$$
$$= \frac{(4.203\times10^{-4})(0.287)(678.3-303)}{1-1.35}$$
$$= -0.129kJ$$

Net Work produced for one cylinder:

$$W_{net} = W_{1-2} + W_{3-4}$$
$$= (-0.129) + (0.723)$$
$$= 0.594kJ$$

Brake work produced for one cylinder:

$$W_b = \eta_m W_{net}$$
$$= (0.86)(0.594)$$
$$= 0.511kJ$$

Brake power produced at 6000rpm:

1. 3 Cylinders:

$$W_b = \left(6000\frac{rev}{\min}\frac{\min}{60\sec}\right)\left(0.5\frac{cycle}{rev}\right)\left(0.511\frac{kJ}{Cyl-cycle}\right)(3Cyl)$$
$$= 76.7kW$$

2. 2 Cylinders

$$W_b = \left(6000 \frac{rev}{\min} \frac{\min}{60 \sec}\right)\left(0.5 \frac{cycle}{rev}\right)\left(0.511 \frac{kJ}{Cyl - cycle}\right)(2Cyl)$$
$$= 51.1 kW$$

3. MNIMUM POWER REQUIRED COMPUTATION

The power required is computed from:

$$P_{\mathrm{Req}} = F_{Total} V$$

F_{total} is the summation of force that exerted on the car movement. In this case, the only forces being considered are the drag force and the rolling resistance force.

The drag force:

$$F_D = \frac{1}{2} \rho V^2 C_d A$$
$$= \frac{1}{2}(1.2)(11.1)^2(0.36)(2.26)$$
$$= 60.1N$$

The rolling resistance force:

$$F_{Rolling} = C_{rr} N_f$$
$$= (0.01)(6867)$$
$$= 68.67N$$

Where:

C_{rr} = 0.01~0.0006 for smooth tyre contact

N_f = Normal force exerted on wheels

= (650kg + 50kg)(9.81m/s^2)

= 6867N

Vehicle mass = 650kg

Driver mass = 50kg

The formula is not adjusted for velocity for simplicity

Hence, total force:

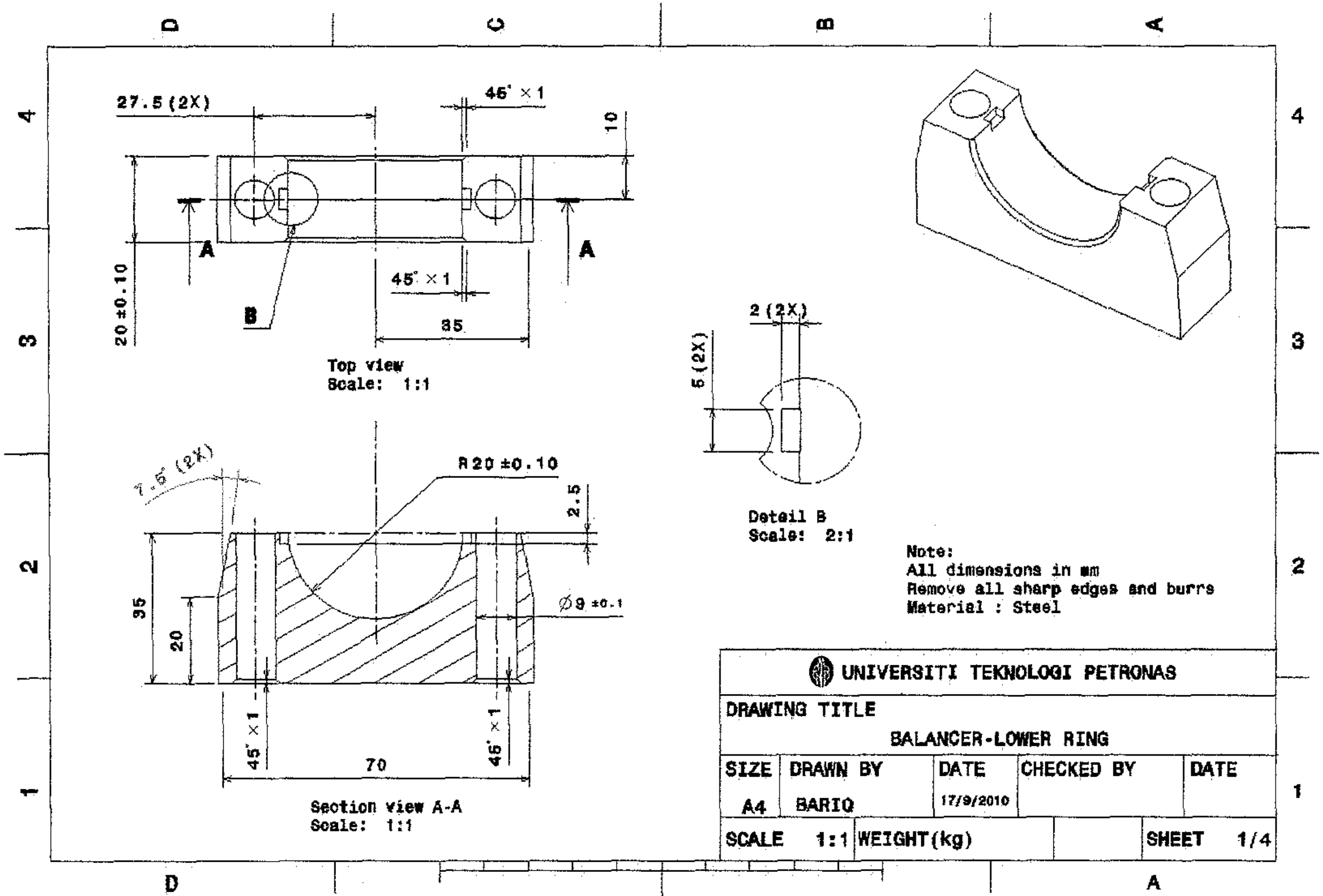
$$F_{total} = 60.1N + 68.67N$$
$$= 128.7N$$

Power required:

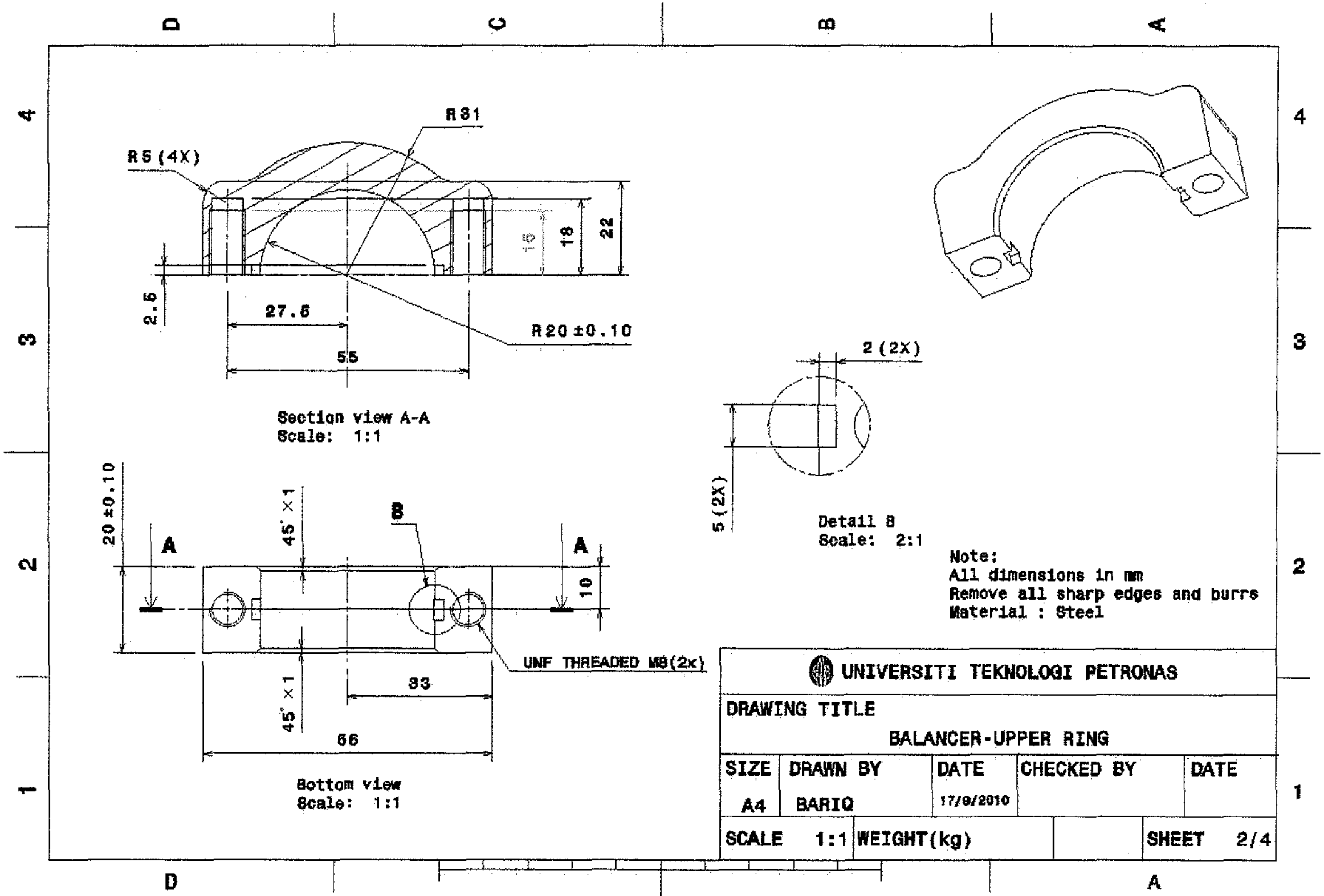
$$P_{\mathrm{Req}} = (128.7)(11.1)$$
$$= 1429W$$
$$= 1.4kW$$

4. DRAWING SPECIFICATION FOR BALANCER

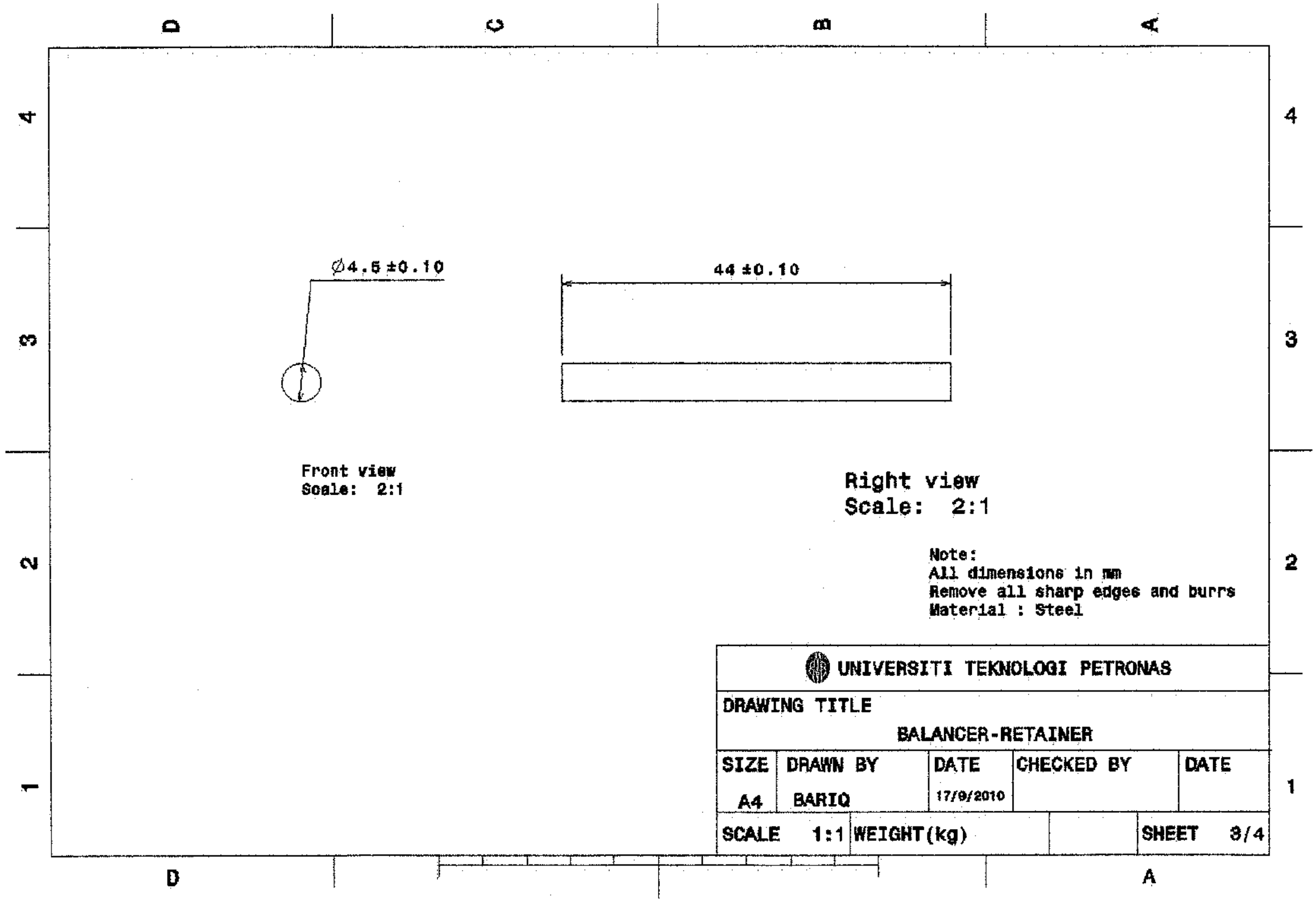
Part name: Lower half-ring



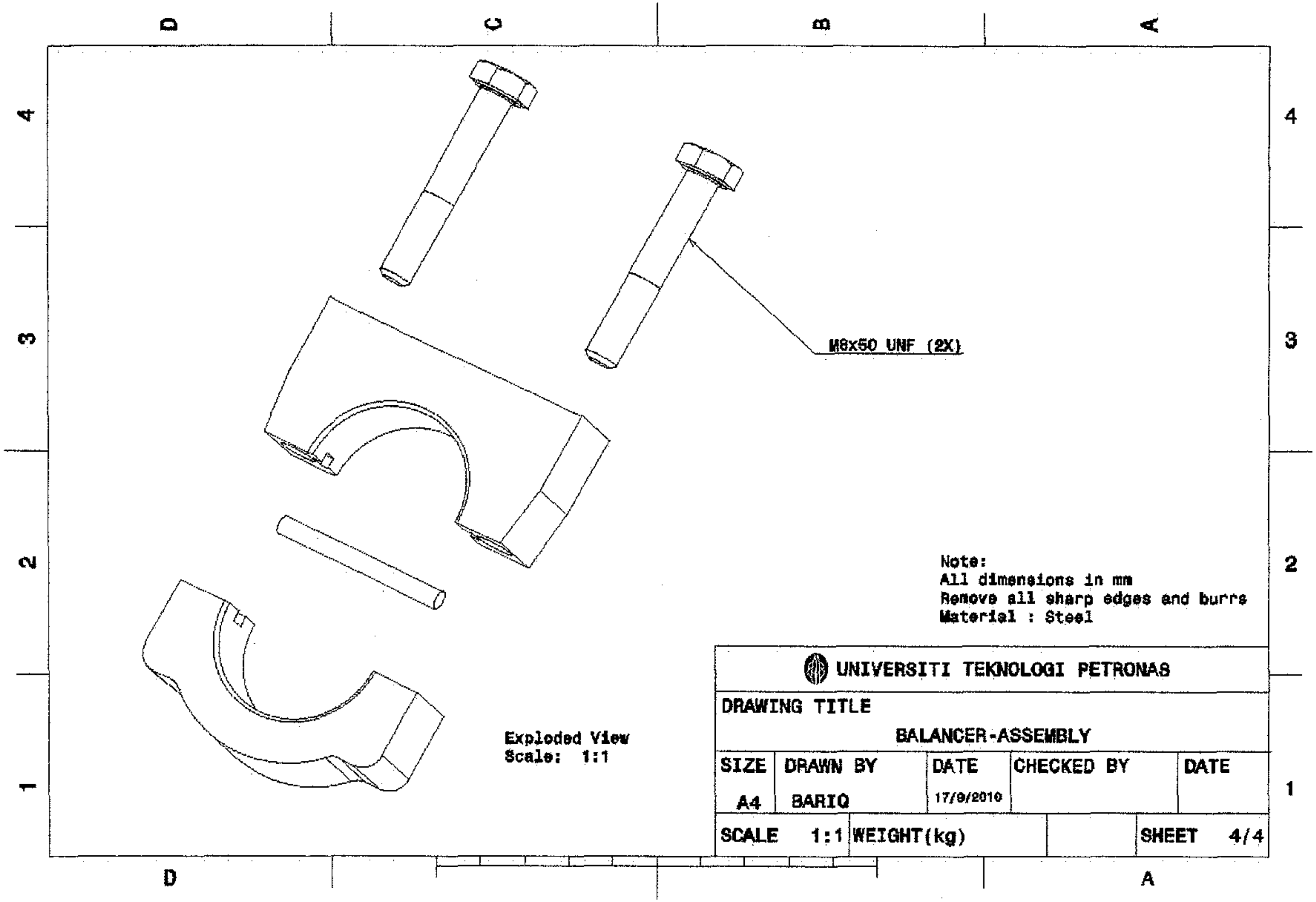
Part name: Upper half-ring



Part name: Retainer rod

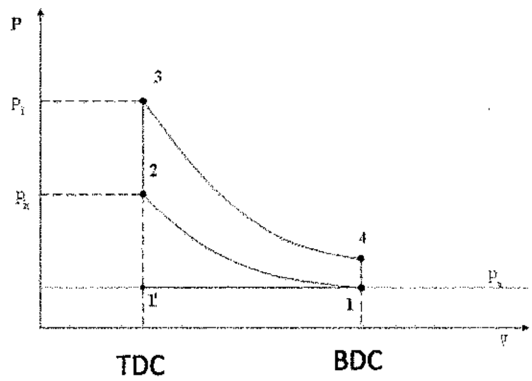


Part name: Balancer assembly



5. DERIVATION OF FORCE FROM P-V DIAGRAM

P-V diagram of Otto Cycle



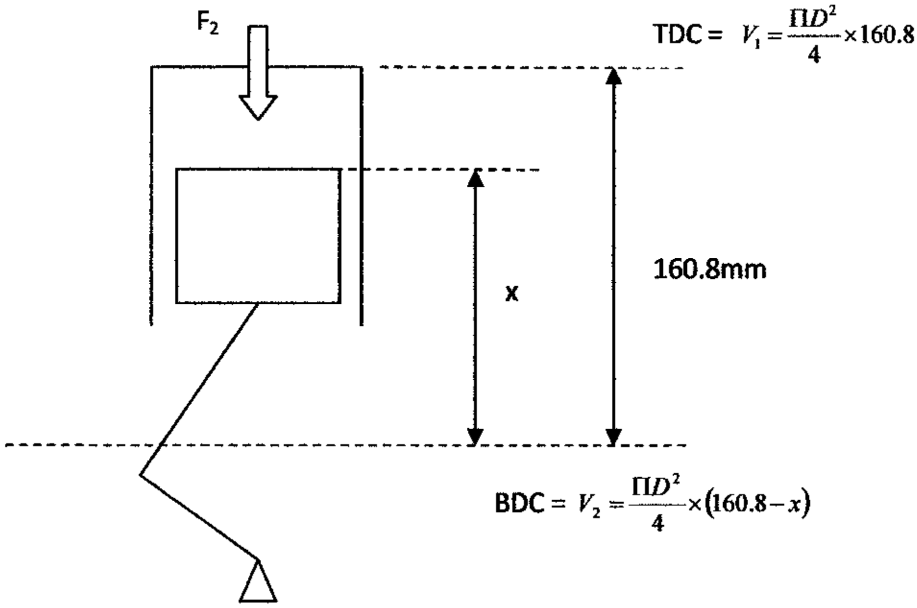
$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^K$$

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^K \qquad \therefore P_1 = P_{atm}$$

$$F_2 = AP_1\left(\frac{V_1}{V_2}\right)^K$$

$$F_2 = \left(\frac{\Pi(72)^2}{4} mm^2\right)\left(1.0135 N / mm^2\right)\left(\frac{V_1}{V_2}\right)^{1.35}$$

Figure below represents the sketch model of a single piston and derivation of force



$$\therefore F_2 = \left(\frac{\Pi (72)^2}{4} mm^2 \right) \left(1.0135 N/mm^2 \right) \left(\frac{160.8}{160.8 - x} \right)^{1.35}$$