DESIGN OF BRAKE TEST JIG TO REPRESENT ACTUAL BRAKING CONDITIONS

by

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0480

Dissertation submitted in partial fulfilment of the requirements for the
Bachelor of Engineering (Hons.)
(Mechanical Engineering)

MAY 2004

Universiti Teknologi PETRONAS
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CERTIFICATION OF APPROVAL

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Approved by,

(Ir Masri bin Baharom)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

MAY 2004

CERTIFICATION OF ORIGINALITY

This is to 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.

faizarif

NIK FAIZARIF BIN NIK MOHAMED AFFANDI

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ABSTRACT

Brake is an ingenious invention which has lead to a revolution in the technology development. The design of brakes has evolved since its invention and has been evolving up until today. Every new design of brakes emphasizes on its safety, performance and reliability. This project proposes to design a test jig to represent actual braking condition. Only the brakes would be tested and simulated.

By designing the test jig to simulate braking condition, we would be able to study the performance of the brakes and do research for future improvements and development of brake design. The design of test jig for disc brake system refers to the design of disc brake testing equipment that measures the brake performance of a single disc brake. The equipment should be capable of producing throughout and detail result of the test by taking into account necessary vehicle braking dynamics. Furthermore it can also be used as an education tool.

The design is generated using a Solid Modeling Software and simulated computationally using Computer Simulation Software to determine the outcome of the testing for verification. The result form the computational simulation is compared with actual braking condition by using the same parameters and brake system specifications. By comparing both computational analysis result and actual result, the validity of the result and reliability of the designed test jig can be determined. Hence the project objective would be accomplished.

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CHAPTER 1

INTRODUCTION

1.1 PROJECT OBJECTIVE

This project proposes to design a test jig for simulating actual braking condition on disc brakes to be tested. Thus the main objective of this project is to design a test jig for the purpose of simulating and testing automobile disc brakes. A specific disc brake of a vehicle sold on the market would be selected as the focus of study. The test jig designed should be able to represent actual braking condition as the brake would perform on the road as much as possible. The design of the test jig would be analyzed and validated using engineering software. Gathering data for test jig simulation is also required. The simulation of the test would be conducted using simulation engineering software and the design would be validated by verifying the result of the simulation with benchmark data.

1.2 PROBLEM STATEMENT

A car can be treated as a system which consists of many sub-systems that enables a car to move. Brakes are one of the many sub-systems which are essential in any automobile. Brakes provide the means to stop a car. Obviously, to be in full control of a car, we have to be able to start it and stop it. To control an automobile, we need to be able to start it moving, make it turn, accelerate and decelerate and of major importance, stop it. Brake performance is seldom the important criteria for most drivers. Therefore brake performance is usually not emphasized by car manufacturers. By conducting tests only on the brakes, the performance and life of the brake can be measured and predicted. By this, safety aspect of the vehicle would be increased.

Brakes are rarely used at their maximum capability, but of course they need to work flawlessly in emergencies. The brakes of most cars are rarely tested to ensure they are working at maximum efficiency. Most car manufacturers do not produce their own brakes. Instead the brakes are bought from another company and assembled in the car assembly factory. Vehicle testing is normally conducted on a complete vehicle i.e. Complete Vehicle Testing (CVT). The test cars would be driven on a test track to test its ride, handling and of course braking. The brakes are tested once it is installed on the vehicle. Complete vehicle testing tests the performance of the complete car and not a particular part. Therefore the test result does not portray the actual capability of the brakes.

A test jig to test only the brake is not popular in the industry. This is due to the fact that the cost of testing the brake is very high as compared as testing it on a vehicle. Therefore it is necessary to design a brake test jig at low cost of fabrication, installation, and maintenance. The test jig would be very useful in the research and development purposes as well as education purposes. By having the brake test jig, the brake performance can be monitored and failures from the rotor, caliper or brake pads can be analyzed for design improvements.

1.3 SCOPE OF STUDY

This project would focus mainly on the study of automobile braking system specifically disc brake operations. The project involves static and dynamics which requires a good understanding of the two. Brake operating principles including theories behind it is also under the scope of study.

Engineering CAD software such as AutoCAD, CATIA, ANSYS and ADAMS needs to be learned and mastered in order to be able to design the test jig and simulate the test. Familiarization of the software requires a lot of training and practice.

This project also requires the study of brake testing which is practiced currently. For example, PUSPAKOM has a brake testing facility which tests brakes on the vehicle itself but the wheels of the vehicles are placed onto a friction roller. This type of testing facility would be beneficial information to begin designing a new concept of the brake test jig.

PROTON also conducts brake testing but on a Complete Vehicle Testing (CVT). Multiple sensors are installed on the brake system of the test car prior to testing it on the test track. Data collected from these sensors upon testing determines the performance of the brakes being tested.

How the test is to be conducted, the parameters to be measured and specifications of the test is also part of the project. Much research needs to be done to achieve this. The design needs to undergo simulation to confirm its validity of the design. The simulation requires data and these data needs to be obtained from a third party. The process of obtaining it would be part of the literature research to be done for the whole project duration.

CHAPTER 2
LITERATURE REVIEW AND THEORY

2.1 BRAKE SYSTEM

The braking system provides the means to stop a car which is in motion. Brakes are heat machines. They provide stopping power by generating heat from rubbing of a friction material, the brake lining, against a rotating drum (for drum brakes) and rotor (for disc brakes). The car slows down as friction produced by this rubbing action converts the energy of the moving car into heat. Friction between the stationary brake pads and the rotating disc produces the braking action that slows or stops the wheels. Then friction between the tires and road slows and stops the vehicles. A typical hydraulic brake system is illustrated in Figure 2.1. It has two types of brakes.

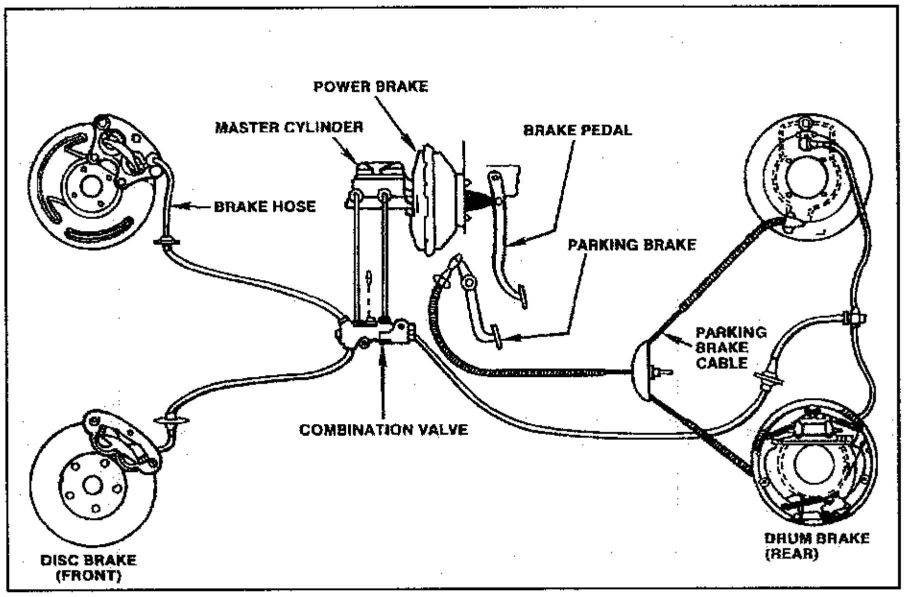


Figure 2.1. Typical automobile hydraulic brake schematic

a) The service brakes, operated by a foot pedal, which slow or stop the vehicle.
b) The parking brakes, operated by a hand lever, which hold the vehicle stationary when applied.

Most automotive services brakes are hydraulic brakes. They operate hydraulically by pressure applied through a liquid. When the brakes are applied by pushing down on the brake pedal, a fluid flows through tubes or brake hose to the brake mechanism at the wheels. The brake mechanism applies force on rotating parts so the wheels are slowed or stopped. There are two types of wheel brake mechanism, drum and disc. In the drum brake, the fluid pressure pushes lined brake shoes against a rotating drum. In the disc brake, the fluid pushes lined brake pads against a rotating disc or rotor.

2.2 DISC BRAKES

A disc brake uses a flat, round disc, or rotor, attached to the wheel hub instead of a drum. The brake shoes, also called pads, are positioned on opposite sides of the rotor and are mounted in the brake caliper. The caliper contains the hydraulic pistons used to apply the shoes and to transmit the braking force from the shoes to the suspension members. Most vehicles are equipped with disc brakes on both the front wheels and some have all four wheels installed. Conventional disc brakes consist of three major parts (refer figure 2.2).

a. Brake pads
b. Caliper
c. Rotor

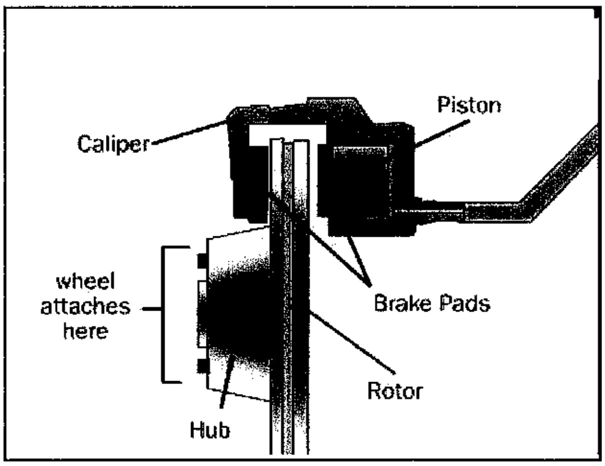


Figure 2.2. Parts of a disc brake

Function of caliper in disk brake operation is to squeeze brake pads on both side of the rotor, this action will result friction between rotor and brake pads. From the friction, the force will eventually slow down rotational of the rotor that is directly attached to tires. Friction between these two surfaces result increase in temperature and in order to overcome the problem most of rotors are designed with vents. The single-piston floatingcaliper disc brake is self-centering and self-adjusting. The caliper is able to slide from side to side so it will move to the center each time the brakes are applied. Also, since there is no spring to pull the pads away from the disc, the pads always stay in light contact with the rotor (the rubber piston seal and any wobble in the rotor may actually pull the pads a small distance away from the rotor).

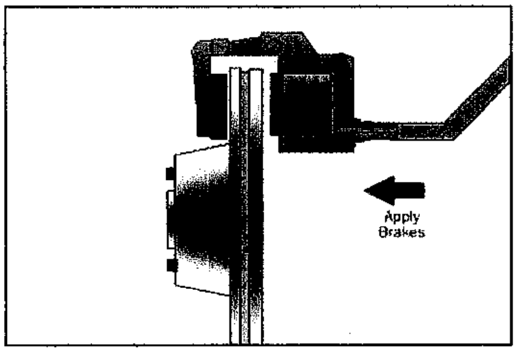


Figure 2.3. Position of brake pads before pedal is pressed

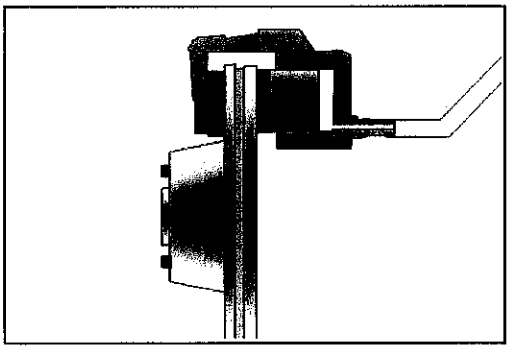


Figure 2.4. Position of brake pads when pedal is pressed

This is important because the pistons in the brakes are much larger in diameter than the ones in the master cylinder. If the brake pistons retracted into their cylinders, it might take several applications of the brake pedal to pump enough fluid into the brake cylinder to engage the brake pads.

2.3 POWER BRAKES / BOOSTERS

Power brakes use a booster to assist the driver in pushing the master cylinder. Normally this consists of a vacuum booster which fits into the linkage between the brake pedal and the master cylinder. The booster can multiply the force of the driver's foot many times. It uses the pressure differential between atmospheric pressure and the intake manifold vacuum against a large diaphragm to obtain this boost (refer Figure 2.5).

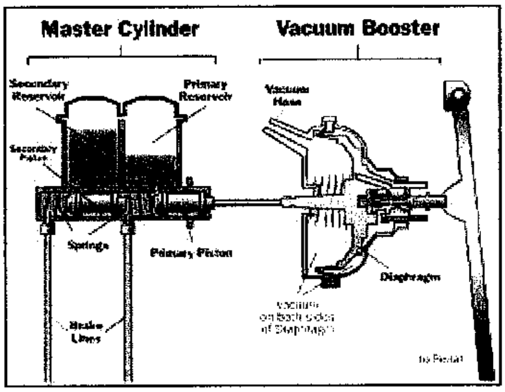


Figure 2.5. Vacuum-booster master cylinder cross section

The assist force, acting on the pushrod which actuates the master cylinder piston, is produced by the difference in pressure across the booster piston or diaphragm with the vacuum or low pressure on the master cylinder side, and the atmospheric or high pressure on the input side. The boost ratio B, is defined as the ratio of the pushrod

$$B = (F_P l_P + F_A) / F_P l_P \,]$$

Where F_A = booster force, N

F_B = pedal force, N

L_P = pedal lever ratio

2.4 COMMON DISC BRAKE PROBLEMS

Disc brake problems can usually be divided into two categories: vibration and noise.

Noises typically indicate brake pad problems;
pedal vibration suggests rotor related problems;
pulling is usually caused by a faulty caliper.

Basically, there are two common types of disc brake problems which are Lateral Run-Out and Judder.

Lateral Runout

Lateral runout is side-to-side motion in the rotor as it turns on the hub. This erratic motion in turn causes uneven wear in the rotor. The rotor hits the pads lightly on each revolution, even without the brakes applied. This extra wear leaves a thin spot on the rotor. Once this thickness variation is worn into the rotor, the brake pedal will vibrate when braking.

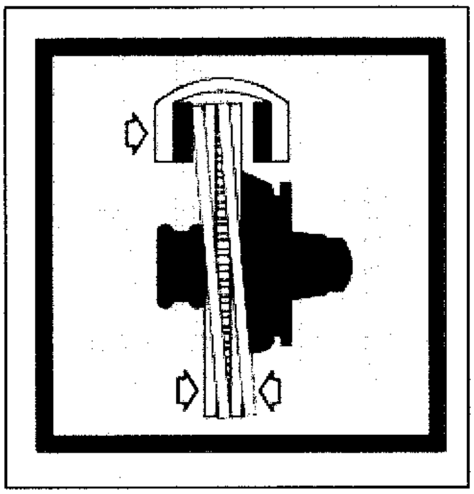


Figure 2.6. Lateral Runout

Judder

Judder is known as the low frequency vibration that is transmitted to a driver through chassis components and the vehicle body during braking. The phenomena occurred due to several factor such as disc thickness variation (DTV) or rotor thickness variation (RTV) and disc Run-Out. Abnormal brake vibration, called pulsation or roughness, is sometimes the result of wheel problems. The vibration is most likely a result of thickness variation in the rotor. As pads squeeze a rotor that is not of uniform thickness, their effectiveness fluctuates — grabbing the thick spots and slipping over the thin. This in turn is felt as a pulsation in the pedal.

Thickness variation is most often caused by lateral runout in the rotor, although rotor dishing or pad material transfer can be contributing factors.

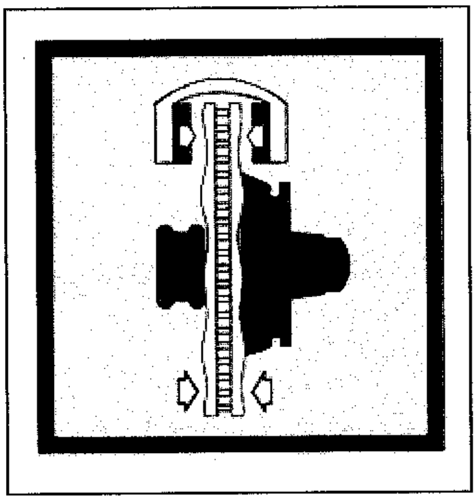


Figure 2.7. Disk Thickness Variation (DTV)

CHAPTER 3

METHODOLOGY/PROJECT WORK

Any project requires proper planning in order for the success of the project and to ensure the project can be completed within the time frame. The suggested schedule for this project is presented in the Gantt chart attached in Appendix 1 and 2. The Gantt chart includes the planning of the project for two semesters. The first semester concentrates on the design stage of the project and second semester concentrates on the simulation and result verification. The project flow process is included in Appendix 3.

The project flow process is a proposed procedure to ensure that the project is running on the right track. The process may be subject to change from time to time depending on the requirements of the project. Much research has been done in the initial stage of the project. This includes reading articles, books, websites and also consulting lecturers and supervisors.

The most important stage is the design stage where the conceptual design of the project is generated. This requires brainstorming and research on testing method available. Then the produced conceptual designs will undergo selection processes whereby the design is evaluated on certain criterion such as reliability and cost.

3.1 GENERATE PRODUCT DESIGN SPECIFICATION (PDS)

Based on the knowledge the writer learned from EMB 4022 (Mechanical Design Technology) taught by Mr Kamarudin Shehabuddeen the total design process is summarized as follows.

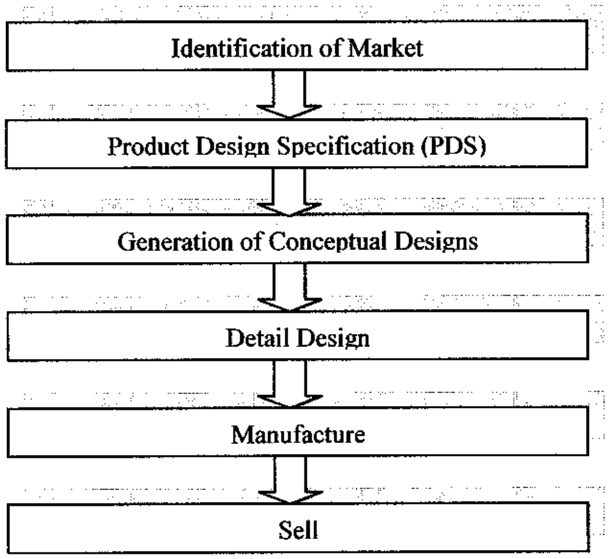


Figure 3.1. Total Design Process

As can be seen from the chart, PDS is an important step in the initial step for the design process. PDS is the requirements or criteria that a product or process must meet in order for it to be successful. PDS represents the specification of what the designer is trying to achieve.

The following is the PDS for this current project.

1) Main Purpose to simulate actual braking condition
2) Able to measure disk's velocity and deceleration as main output
3) Able to be fitted with any type of disc brakes
4) Provides space for brake hydraulic system
5) Disc brake to be rotationally driven
6) Electric motor or rotary actuator as driver

7) Variable rotational speed up to 120 km/h or more
8) Adjustable load to imitate GVW and LVW
9) Cost effective
10) Simple design with easy maintenance
11) Practical size for easy storage
12) Simple operation for educational purposes
13) Service life more than 10 years
14) Extended endurance and durability life for fatigue testing
15) Safe design operation
16) Easily modified/adjustable to add auxiliary features.

3.2 DESIGN OF TEST JIG USING CATIA

The conceptual design is initially drawn onto a graph paper. The design is then redrawn on CATIA to better visualize the design. CATIA is the Computer Aided Drawing software used to generate engineering drawing. CATIA may also be used for structural analysis purposes. For simulation purposes, ADAMS software is used to simulate the brake test based on the design and specification of the test.

Based on the PDS determined, the writer created several jig drawings using CATIA as to choose the best design. Once several designs have been generated, one design should be selected after each has been evaluated to be selected for the simulation of brake testing on ADAMS. So far three conceptual designs have been generated and will be discussed in the discussion section of this report.

3.3 EVALUATION OF CONCEPTUAL DESIGNS

The conceptual design should be evaluated according to certain criteria. The criteria must be based on the design detailed requirements such as Product Design Specification (PDS).

Weightings may be assigned to the criteria and also a datum may be chosen against which all other concepts are to be evaluated. There are two methods of evaluation for conceptual design

- Without using weightings –
  - setting an existing design as datum
  - plus (+) for better than the datum
  - minus (-) for worse than the datum.
  - S (same) for same as datum

- Using weightings –
  - weights are assigned to each of the criterion
  - ratings are given on each criterion by personal judgement.
  - calculate the weighted score of each concept for each criterion
  - calculate the total weighted score of each concept

For this case, the evaluation of conceptual design is done using weightings. Please refer Appendix 6 for the Concept Evaluation Matrix for Design of Jig to Simulate Actual Braking Condition.

3.4 GATHER AND ANALYZE DATA FROM PROTON

In order to better understand the concept of brake testing, visit to PROTON Bhd was conducted. The objective of the visit is to gain exposure on the type of testing and equipment used. Appendix 7 lists the equipments used by Proton Research and Development Department to conduct testing on brake system. Through discussion with the engineer it is found that the project is quite complicated and the cost will be higher than expected.

From the specification of the brake system of Proton Iswara 1.3s (refer Appendix 9), it can be seen that the car uses solid disc brake type for the front brake and drum type for the rear brakes. This project is only interested in the simulation of disc brake at this moment. Thus the dimensions of the disc brake of Proton Iswara 1.3s will be used in the design and simulation. From the specification sheet, the master cylinder is of tandem type and vacuum type brake booster.

The disc is of M-R315 solid disc type with:

1. Disc outer diameter : 23.4 cm
2. Disc inner diameter : 5.1 cm
3. Disc thickness : 1.3 cm
4. Disc mass : 10 kg
5. Brake pad width : 2 cm
6. Brake pad height : 1 cm
7. Brake pad thickness : 0.5 cm

The vehicle weight in study is also included in the data provided. This data provides total weight of the vehicle and also weight distribution in all four sides of the car. This data is important since the project requires the weight of the vehicle to be simulated during braking simulation. For this case, the inertia flywheels will be used to simulate the weight of the vehicle during braking. The weight includes Kerb Weight, Light Vehicle Weight (LVW) and Gross Vehicle Weight (GVW). The writer is only interested in the LVW and GVW and from the data sheet, both LVW and GVW is assumed to be almost the same.

Raw data or test results of brake performance testing for Iswara 1.3s from Proton shall be used for verification purposes in this project. Outcome from simulation of braking vehicle by using ADAMS software shall be compared with the test data. The simulation is expected to produce data that is not much different from the raw data of actual brake performance testing. These raw data is included in appendices 8 – 14. The brake performance testing parameters for Iswara 1.3s is grouped into four as Proton conducts and they are:

- Light Vehicle Weight at 50 km/h
- Light Vehicle Weight at 100 km/h
- Gross Vehicle Weight at 50 km/h
- Gross Vehicle Weight at 100 km/h

As can be seen from the raw data obtained from Proton, the results are plotted in the form of deceleration versus pedal force. The writer is not interested in the pedal force but the braking force. Since the results from ADAMS simulation would yield deceleration versus time, so the writer requires similar data for verification. Therefore several series of calculation has been conducted to convert the raw data into function of time to be applied for the simulation. The calculation is included in the results section of this report.

3.5 STRESS ANALYSIS

Based on the design of the jig, it can be seen that rotor is turn by the shaft which is directly connected to the electric motor and inertia flywheels. It is obvious that the part which experience most stress would be the shaft since it is responsible for the load transfer from the motor and from the inertia flywheels once the motor is stopped. Based on this, the shaft used in the jig should be analyzed using structural analysis tool on CATIA.

3.6 ADAMS SIMULATION

This project requires the use of ADAMS software to simulate the brake test and verify results. After familiarization with the software, the writer then attempts to construct basic models of the disc rotor, pad and shaft in ADAMS.

By using the geometric modeling tool, a cylinder rigid body is selected to generate the disc rotor as well as the shaft which will rotate the rotor. A revolute joint is included in between so the rotor will be able to rotate along with the shaft. The shaft is then applied a torque force at one end in order to simulate the torque generated by the motor which is then transferred to the shaft and the rotor.

A model of a brake pad is included in the rotor to simulate friction between the brake pad and rotor. Figure 3.2 shows the preliminary ADAMS model. This is a simple model to enable the writer to familiarize with the simulation.

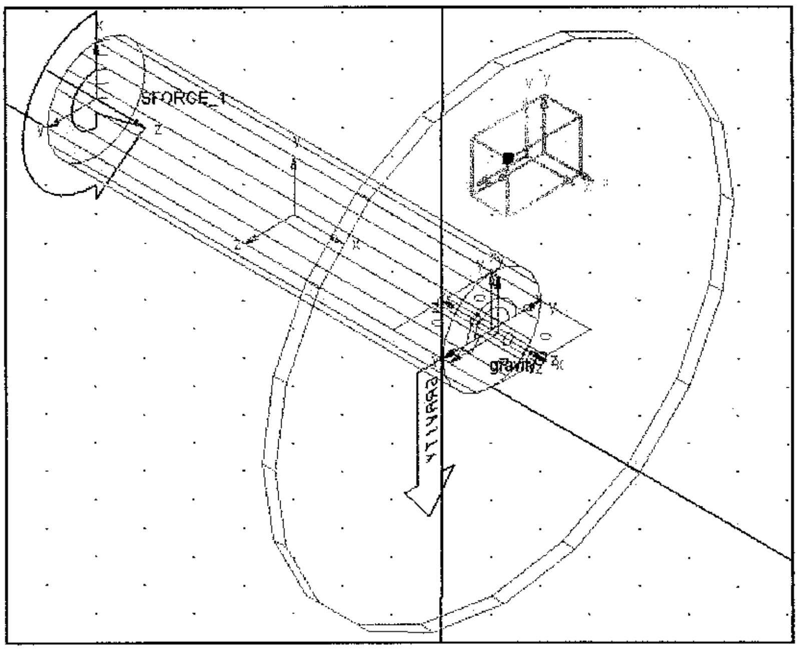


Figure 3.2. Preliminary ADAMS model

Another approach is by importing CATIA drawing of the jig into ADAMS. This can be done by saving the CATIA drawing as Stereolithography (stl) file in CATIA and importing the Stereolithography (stl) file into ADAMS. Each of the part of the jig needs to be imported separately. Figure 3.3 shows the wire mesh of the imported jig drawing from CATIA.

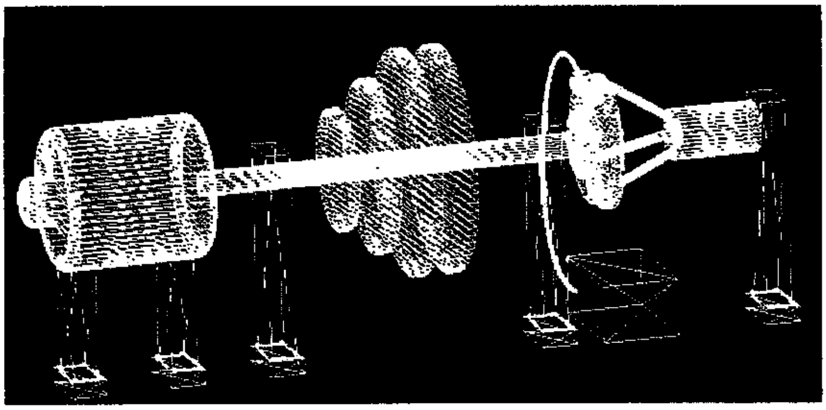


Figure 3.3 Wire mesh of CATIA imported file in ADAMS

To better simulate the jig as of its design, the second approach is used whereby the actual design of the jig is used to simulate braking. The model defines the shaft, flywheel and rotor as separate components however they rotate together as intended in the simulation. The shaft is initially rotated at the test speed and the flywheel will simulate the weight of the vehicle once the brake is pressed. The caliper which is an important component in the model will act to apply force onto the spinning rotor thus slowing and finally stopping it to a halt.

To ensure the simulation is conducted as closest as possible to the actual testing, type of joints needs to be selected carefully as well as applied force and contact force. The simulation is mainly concerned with the rotation of the shaft, flywheel and rotor thus revolute joint is used here. However since the flywheel and rotor spin together with the shaft, fixed joint is applied between flywheel and shaft and between rotor and shaft.

Once shaft begins to spin, its velocity will be at the speed of the desired test speed and will then be slowed down by the friction force between the caliper and the rotor. The joint between caliper and rotor is defined as translational joint. This is due to the fact that the caliper needs to clamp the spinning rotor thus requires translational joint to translate in one direction. However the force applied from the caliper to the rotor is not linear. A function needs to be defined so that the brake force increases with time. In actual, when a driver steps on the brake pedal, brake force is applied accordingly to the effort made by the driver but the brake force starts from initial force to the maximum force applied by the driver and there is a time lag in between thus explains why the brake force increases with time.

Brake function is defined for the force applied from the caliper to the rotor to simulate braking. Since brake force is not instantaneous but increases with time, force to the function of time is defined. The brake pedal force from the raw data is converted to braking force through a series of calculation shown in the results. Different brake test specification yields different brake force therefore different brake force function is defined at the caliper to simulate brake testing according to the test specification.

Contact between solid to solid is set between caliper and rotor to define frictional properties between these two elements. Coefficient of friction between brake pads and tire is set to the same value with actual coefficient of friction between caliper and rotor.

Once ADAMS Simulation is completed, the measures defined in the model will measure the velocity of the rotor. However we are interested in the deceleration of the rotor. Measuring of deceleration is constrained by the fact that there is two component of acceleration for rotational motion namely angular acceleration, a_r and tangential acceleration, a_t. ADAMS is unable to separate these two components, therefore one of the approach used is to differentiate the graph of linear velocity to obtain tangential acceleration since for rotation motion about fixed axis;

$$\frac{dv}{dt} = a_t$$

where

a_t = Tangential acceleration

v = Linear velocity

t = Time

Graph of acceleration versus time for the simulation is then negated to generate the graph of deceleration versus time. ADAMS/Postprocessor tool is used for the task of differentiating and negating of graphs to obtain the deceleration required for verification of results.

The methodology which is followed by the writer for ADAMS simulation is summarized as follows:

1. Import each of the part from Catia as Stereolithography (stl) file separately.
2. Create a unique name for each of the imported part.
3. Define each of the part's material.
4. Create joints for each of the part.
    a. For parts that do not move or fixed to each other *Fixed Joint* is used.
    b. For parts that rotate *Revolute Joint* is used. (e.g. Shaft and Rotor)
    c. For parts that have liner displacement *Translational Joint* is used. (e.g. Caliper)
5. Define *Motion Generator* to simulate motion at the joint. Motion can be either revolution at revolute joint or translational at translational joint.
6. Run trial simulation to simulate rotational motion of the shaft and the rotor.
7. Calculate angular velocity of the shaft corresponding to the test condition.
8. Remove the *Motion Generator* and define the angular velocity calculated at the *Initial Condition* of the *Revolute Joint.*
9. Run trial simulation to simulate rotational motion of the shaft and the rotor and plot the result to verify if the shaft and rotor is rotating at the desired speed.

10. Once verified, the next step is to simulate braking
11. From the raw data, pedal force is converted to braking force because we are only interested in the force applied by the caliper to the rotor.
12. Create *Contact Forces* between the caliper and the rotor
13. Apply *Single Component Force* from the caliper to the rotor. Define brake force function according to brake test specification.
14. Run simulation and plot results.
15. Differentiate velocity plot and negate the acceleration plot to obtain the deceleration plot.
16. Repeat for different test specification
17. Compare results with test data and verify

CHAPTER 4
RESULTS AND DISCUSSION

4.1 JIG DESIGN

During the period of one whole semester which is the first 14 weeks of this project, three conceptual designs have been produced. All three designs are included in Appendix 4. The designs were made after conducting literature research and discussion with supervisor. The design is simple, cost effective and reliable. Selection of the design shall be made according to certain weight such as simplicity, cost, reliability and effectiveness.

The writer has come up with three design concepts and referring to the Product Design Specification (PDS) criteria's determined earlier, each of the criteria are included with weights and are rated for each of the designs. Each of the criteria are evaluated based on the description of the concept (refer Table 4.1 for Morphological Chart) and rated according to the writer's judgment (refer Appendix 6 for Concept Evaluation Matrix). Among the three concepts, Concept Design 03 scored the highest compare to Concept Design 01 and Concept Design 02 and therefore Concept Design 03 will be used as the final design. Final design generated from conceptual design 03 is included in Appendix 5

Major Specifications	Concepts		
	01	02	03
1) Prime mover	Rotary Actuator	AC Motor	AC Motor
2) Weight of vehicle	Weight attach at suspension	Weight attach at tire	Use inertia flywheel
3) Simulation Method	Direct connection to actuator	Two friction rollers	Dynamometer
4) Operation	Rotary actuator rotates disc brake directly	Friction roller connected to motor will rotate tire & disc brake	Motor will rotate shaft connected to flywheels and disc brake
5) Disc brake installation	Disc brake without tire and wheels.	Disc brake with tire and wheels	Disc brake without tire and wheels.

Table 4.1. Morphological Chart

Referring to Conceptual Design 01, the disc brake will be attached to its steering knuckle. The knuckle, in actual is supposed to be attached to the suspension but for simulation purposes, weight should be applied here to simulate vehicle weight. The rotor will be rotated by a rotary actuator and once a specific speed is achieved, the hydraulics will apply pressure to the caliper and stop the rotor.

This design uses the concept of running the test based on the torque of the wheels. This requires the use of Wheel Force Transducers (WFT) which are installed onto the wheels of a vehicle prior testing it on the road to collect torque data. The idea behind using torque to simulate actual braking condition is excellent. However the system would be restricted to only vehicles having the torque data only. Therefore, the design would not be applicable for all models and would not be cost effective.

Conceptual Design 02 uses the concept of friction rollers to simulate wheel rotation. This is a piece of equipment that has a pair of friction rollers. The friction rollers are driven by an electric AC motor. The tire is placed over the rollers and when the motor is started, the tire will rotate up to a specified speed. The brakes are then applied, and the effective power of the brake is measured by the amount of resistance it offers the driving motor.

This design concept is normally used to measure brake power. However the test usually applies only to complete vehicle testing (CVT) which means brake testing using friction rollers is best applied to a Complete Buildup Unit (CBU) because all of the factors considering brake system have been taken into account. Furthermore, weight of the vehicle may not be simulated if the disc brake is detached from the vehicle.

Conceptual design 03 uses the concept of dynamometer bench test. By using the dynamometer concept, we would be able to simulate the weight of the vehicle by replacing with inertia flywheels. Revolution of the disc per minute can also be controlled by controlling the AC motor attached to a tachometer. The velocity can be converted to rotational velocity for testing purposes.

An inertia dynamometer comprises an electric motor capable of controlling a variable speed inertia flywheel and more specifically, the speed of rotation corresponding to the maximum speed of the vehicle for which the brakes are to be tested. It is necessary to communicate a kinetic energy to the inertia masses which is identical to that supplied by the engine to the part of the vehicle affected by the brake being tested.

The inertia masses are the flywheels fixed to the crankshaft. A number of flywheels with different inertia should be used so it is possible to select those that accurately produce the desired total inertia. The disc brake is fitted at the end of the shaft with the caliper installed. A velocity meter and accelerometer is placed along the disk rotor to measure the velocity and deceleration of the rotor during the testing for verification purposes.

A vehicle in motion transmits mechanical energy in the form of kinetic and potential energy. Under stable conditions, namely at a constant speed and on a flat surface, the energy produced by the engine is assumed entirely converted into heat as a result of numerous instances of friction.

When braking, the vehicle's kinetic energy is entirely converted into thermal energy by the interaction of the brake pads and discs. Example calculations are as follows:

If a car weighing 1200 kg and traveling at 120 km/h, and the braking load on the front wheels is 60% to come to a complete halt, the brake must dissipate E_{kin}. [1]

M = vehicle mass = 1200 kg
60 % weight transfer to front wheels = 1200 x 60/100 = 720 kg
Weight on one wheel = 720 /2 = 360 kg
v = velocity = 120 km/h = 33.33 m/s

[1] The conversion of kinetic energy to inertia calculation obtained from Reference [3] BREMBO Brake Disk Manual Handbook

substitute into kinetic energy formula

$$\mathbf{E_{kin} = ½[M\ v^2]}$$

E_{kin} = ½ [360 x 33.33^2]

E_{kin} = 199,960 Joule

kinetic energy formula for rotating parts

$$\mathbf{E_{kin} = ½[I\ \omega^2]}$$

I = mass moment of Inertia

ω = rotational velocity

= v /r = 33.33 / 1 = 33.33 rev / s (assuming disc radius, r = 1 m)

I = 2 E_{kin} / ω^2

= 2(199,960) / 33.33^2

= 360 kg m^2

Bearing in mind that the inertia of a disc rotating around its own axis is :

$$\mathbf{I = ½\ [M\ r^2\ ]}$$

The energy dissipated by the brake is E = 199,960. To store this energy in a dynamometer, will require an inertia equal to I = 360 kg m^2.

If a single inertia flywheel were to be used, it would need for example.

Diameter = 1 m

Mass = 600 kg

If the dynamometer does not have this mass, then a number of flywheels will need to be combined to obtain the nearest possible inertia.

4.2 COMPUTER AIDED DESIGN

One of the important requirements in any design project is graphical presentation of the design. CATIA software is used thoroughly in the design stage of this project. It facilitates the conversion from conceptual design ideas to proper graphical presentation. From the three conceptual designs, only one is chosen to be the final design which is conceptual design 03. Figure 4.2 shows the final design of the jig based on actual dimensioning. For more details of the final design, refer to Appendix 5.

The design is first divided into a number of parts or components namely rotor, caliper, shaft, inertia flywheels, electrical motor, jig stand, torque meter, brake system compartment and brake hose. A complete view of the final design is included in the appendix. All the separate parts are then assembled into one drawing to produce assembly drawing as shown on figure 8.

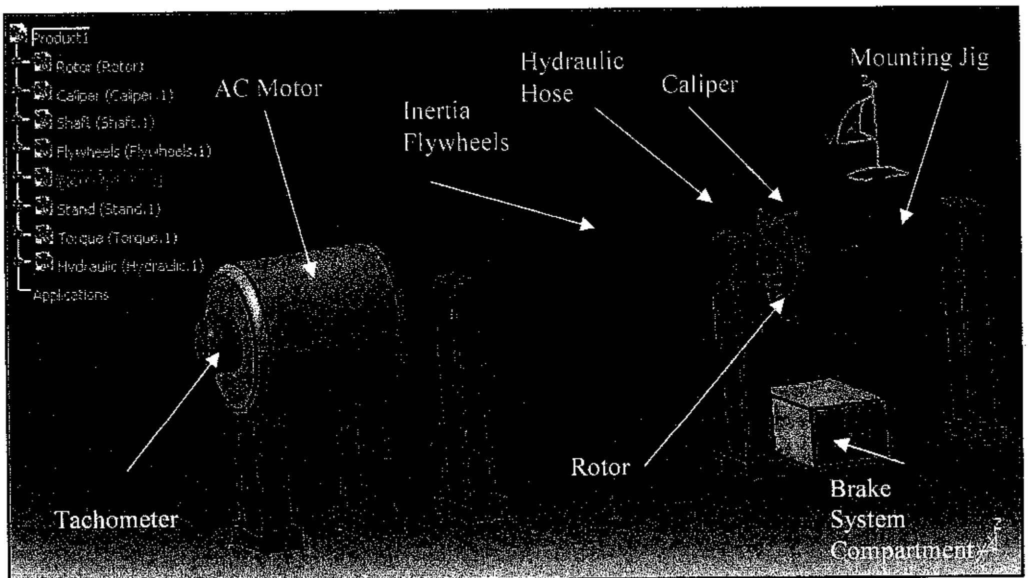


Figure 4.1. Isometric view of the brake test jig

Figure 4.2 shows drafting layout of the test jig on front, right side, top and isometric view. For the first semester, focus is given on design stages, data gathering and familiarization with engineering software such as CATIA and ADAMS. Computational analysis on ADAMS and verification of results is conducted on the second semester.

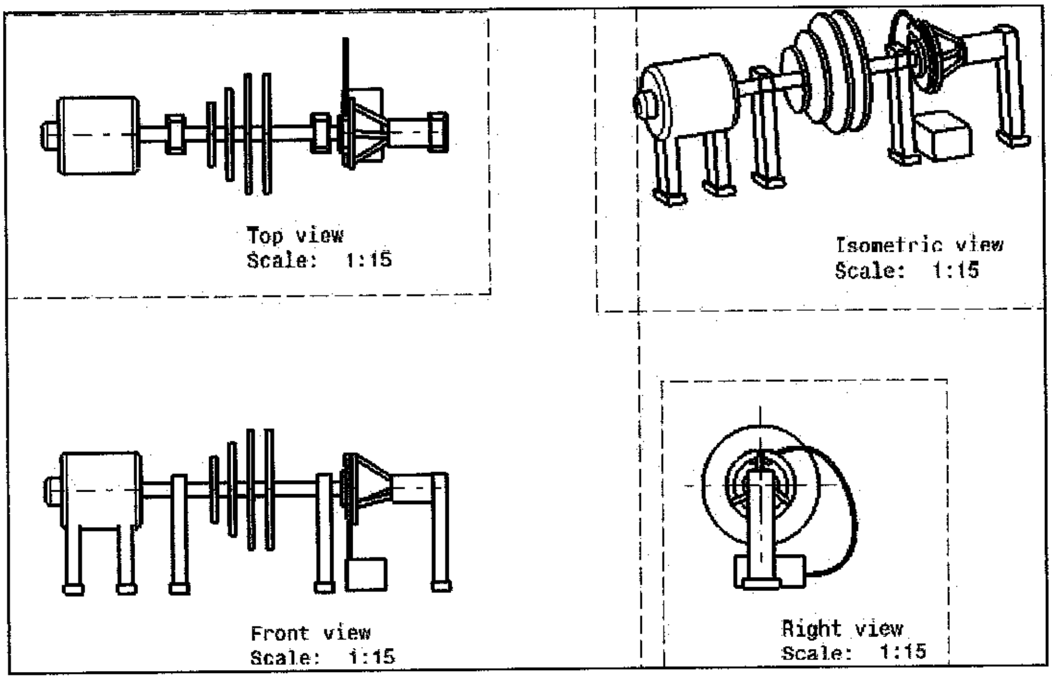


Figure 4.2. Drafting layout of the brake test jig

4.3 STRESS ANALYSIS

Based on the design of the jig, it can be seen that the rotor is turned by the shaft which is directly connected to the electric motor and inertia flywheels. It is obvious that the component which experiences most stress would be the shaft since it is responsible for the load transfer from the motor and from the inertia flywheels once the motor is stopped. Based on this, the shaft used in the jig should be analyzed using structural analysis tool on CATIA.

Referring to PROTON's brake test specification, the brake test is conducted at 50 km/h and 100 km/h at LVW and GVW. Test specification of 100 km/h LVW is chosen to conduct stress analysis. Example calculation as follows:

Calculation of torque on shaft / rotor – (for 100 km/h LVW)[2]

M = vehicle mass = 1137 kg

60 % weight transfer to front wheels = 1137 x 60/100 = 682.2 kg

Weight on one wheel = 682.2 /2 = 341.1 kg

v = velocity = 100 km/h = 27.77 m/s

substitute into kinetic energy formula

$$E_{kin} = ½[M\,v^2]$$

$$E_{kin} = ½\,[341.1 \times 27.77^2]$$

$$E_{kin} = 131{,}523.54 \text{ Joule}$$

kinetic energy formula for rotating parts

$$E_{kin} = ½[I\,\omega^2]$$

I = mass moment of Inertia

ω = rotational velocity

= v /r = 27.77 / 0.117 = 237.35 rad / s (disc radius, r = 0.117 m)

$$I = 2\,E_{kin} / \omega^2$$

$$= 2(131{,}523.54) / 237.35^{\,2}$$

$$= 4.67 \text{ kg m}^2$$

Linear velocity : V = 100 km/h = 27.77 m/s

Vehicle decelerates from 100 - 0 km/h in 3.29 seconds

Thus, acceleration a,

$$a = [(100 - 0) \text{ km/h}] / 3.29 \text{ s}$$
$$= [(27.77 - 0) \text{ m/s}] / 3.29 \text{ s}$$
$$= 8.44 \text{ m/s}^2$$

[2] Based on the conversion of kinetic energy to inertia calculation explained in Section 4.1

Angular acceleration : $\alpha = \frac{a}{r} = \frac{8.44}{0.177} = 47.68 \text{ rad/s}^2$

Torque T, $T = I\,\alpha$

$$= 4.67 \text{ kg.m}^2 (47.68 \text{ rad/s}^2)$$
$$= 222.68 \text{ kg.m}^2 / \text{s}^2$$
$$= 222.68 \text{ N.m}$$

Boundary condition of Moment = 222.68 N. m is defined at one end of the shaft and the other end of the shaft is clamped (to simulate braking). The simulation is then executed and the following results are obtained.

Components	**Applied Forces**	**Reactions**	**Residual**	**Relative Magnitude Error**
F_x (N)	0	$-8.97\text{x}10^{-09}$	$-8.98\text{x}10^{-09}$	$4.3547\text{ x}10^{-13}$
F_y (N)	0	$1.85\text{ x}10^{-11}$	$1.85\text{ x}10^{-11}$	$8.90\text{ x}10^{-15}$
F_z (N)	0	$-8.64\text{ x}10^{-09}$	$-8.65\text{ x}10^{-09}$	$4.20\text{ x}10^{-13}$
M_x (N.m)	0	$-4.59\text{ x}10^{-09}$	$-4.59\text{ x}10^{-09}$	$2.79\text{ x}10^{-13}$
M_y (N.m)	222.68	$-2.23\text{ x}10^{2}$	$1.16\text{ x}10^{-11}$	$7.00\text{ x}10^{-15}$
M_z (N.m)	0	$5.41\text{ x}10^{-09}$	$5.41\text{ x}10^{-09}$	$3.28\text{ x}10^{-13}$

Table 4.2. Boundary Conditions and Reactions Forces

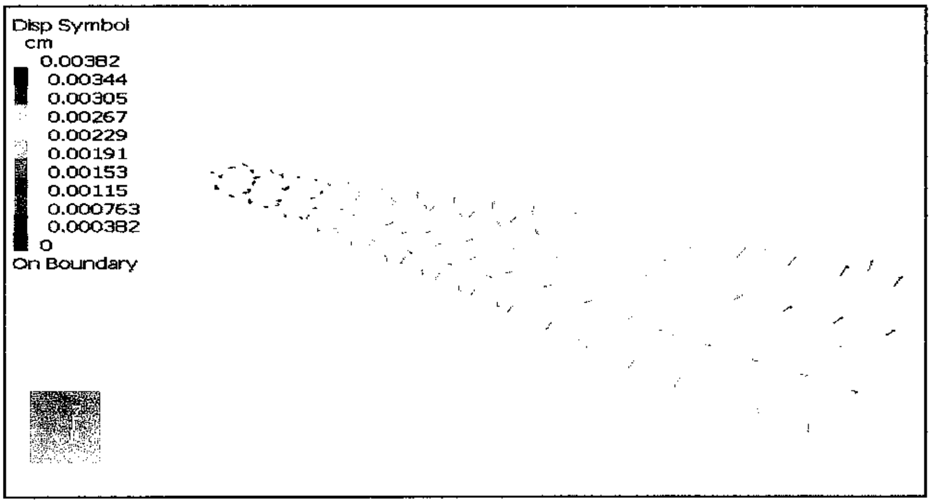


Figure 4.3. Displacement / Deformation of the shaft

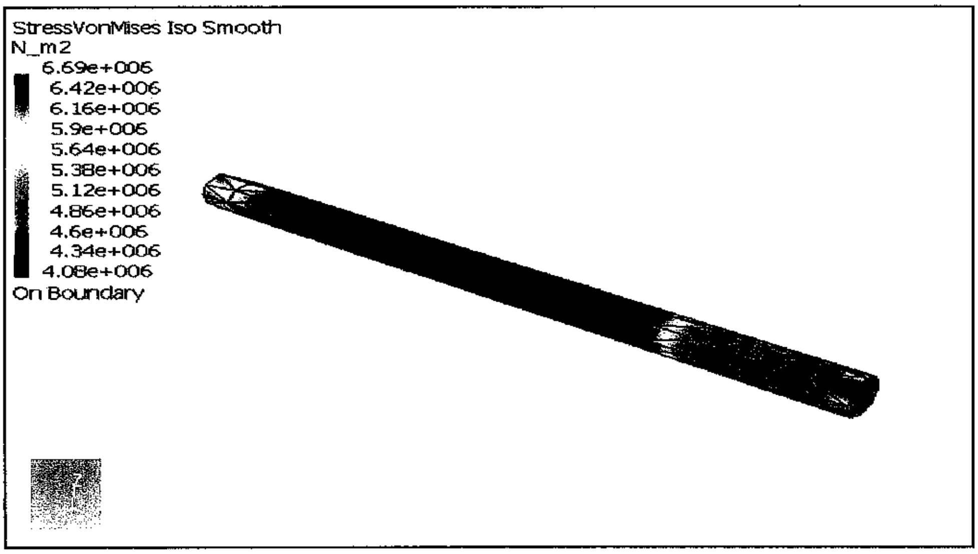


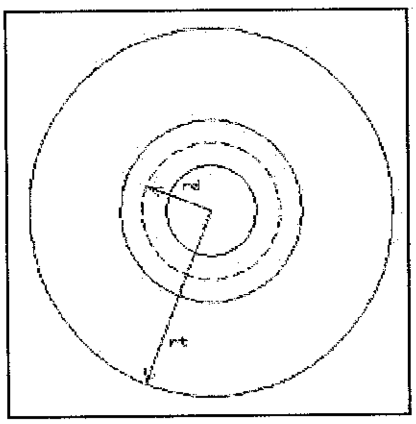
Figure 4.4. Moment Distribution on the shaft

4.4 ADAMS SIMULATION

Once ADAMS model of the test jig has been properly built, its joints and forces well defined, the model may be simulated. The results however need to be compared with actual testing results which are obtained from the test results or raw data from Proton. The raw data is given in the form of deceleration versus pedal force. From the raw data, the value for average deceleration, stopping distance and braking time can be calculated by using Newton's Second law. However the pedal force needs to be converted to braking force first.

Below is the example of calculation used to convert pedal force to braking force for 2.19 kg pedal force (raw data for 50 km/h LVW)[3] :

Inner diameter of hydraulic master pump	= 20.64mm
Diameter of vacuum servo booster	= 180mm
Vacuum reading	= 450mmHg.
Foot pedal leverage ratio	= 4.5:1.
Diameter of cylinder on each wheel	= 51.1mm.
Effective disc diameter, r_d	= 234mm.
Tire diameter, r_t	= 550mm.
Coeff.of friction between the brake pad and the disc, μ_c	= 0.42
Coeff. of friction between the tire and pavement, μ_d	= 0.62



r_d = effective disc diameter = 234mm.

r_t = tire diameter = 550mm

Figure 4.5 Effective disc diameter and tire diameter

[3] Calculation obtained from course Machine Component Design

Solution

Pedal force, $F = 9.81 \text{ x } 2.19\text{kg} = 21.462 \text{ N}$

Force due to pedal lever advantage, $F_1 = 21.462 \times 4.5 = 96.58N$

Area of diaphragm, $A_d = \dfrac{\pi(0.180)^2}{4} = 0.02545m^2$

Force due to power assist by vacuum booster, F_2

$$F_2 = \Delta P \times A_d$$
$$= (760 - 450)mmHg \times \frac{10^5\,Pa}{760mmHg} \times 0.02545$$
$$= 1038.1N$$

Total force exerted to the master cylinder piston

$$= F_1 + F_2$$
$$= 96.58\text{N} + 1038.1\text{N}$$
$$= 1134.68\text{N}$$

Cross sectional area of hydraulic master pump,

$$A_m = \frac{\pi(0.0206)^2}{4} = 3.333 \times 10^{-4}\,m^2$$

Generated brake fluid pressure, $P = \dfrac{F}{A_m}$

$$= \frac{1134.68}{3.333 \times 10^{-4}}$$
$$= 3.4MPa$$

Cross sectional area of slave cylinder,

$$A_s = \frac{\pi(0.051)^2}{4} = 2.043 \times 10^{-3}\,m^2$$

For one cylinder

Braking force on each cylinder,

$$F_g = \text{P x A}$$
$$= 3.4MPa \times 2.043 \times 10^{-3}\,m^2$$
$$= 6955.15N$$

Frictional force between the brake pad and disc

$$
\begin{aligned}
&= F_g \times \mu_c \\
&= 6955.15 \times 0.42 \\
&= 2921.2N
\end{aligned}
$$

Microsoft Excel is used to calculate braking force from pedal force for the rest of pedal forces on raw data. This calculation only provides the value of the braking force at a constant interval. The braking time is required to plot the braking force versus time. The following is a sample calculation to calculate total braking time for 100 km/h LVW test derived from Newton's Second Law.

For 100 km/h LVW (1137 kg)

Average brake force = 8239.18 N[4]

$D_x = F_x = \frac{Fx}{M} = \frac{8239.18}{1137} = 7.2464 \text{ m/s}^2$

Stopping Distance, SD = $\frac{V_o^2}{2D_x} = \frac{(100km/h \times 1000m/km \times 1h/3600s)^2}{2 \times 7.2464 m/s^2} = 53.24$ m

Total Braking Time, $t_s = \frac{V_o}{D_x} = \frac{27.78 m/s}{7.2464 m/s^2} = 3.833$ s

Assuming constant braking rate during brake performance testing, the total braking time is divided into even intervals to obtain period between each reading on test data. Complete calculation is included in Appendix 15. Once the time interval between the braking time is available, graph of brake force to the function of time can be plotted and the plots are included in Appendix 16.

[4] Pedal force is converted to brake force and its average value taken for braking time calculation

To simulate braking in ADAMS using the data from Proton, brake function needs to be generated to vary the force applied by the caliper with the function of time. Based on the raw data, braking force applied is considered as linear as showed on graph of braking force to the function time.

The use of function for application of braking force requires building of syntax derived from the graph of braking force to the function of time. By calculating slope on the graph of braking force to the function of time, syntax can be written as;

$$F = m*TIME + y$$

where

m = Slope of graph

y = Value of force at t = 0

Knowing that braking force applied is presented by the graph of braking force to the function of time, slope, m and value of braking force at t = 0, y can be obtained. The following calculation shows the slope and value of braking force for each parameters based on the graph in Appendix 16. The brake force functions are then applied at the caliper force for the simulation of braking in ADAMS.

For 50 km/h and LVW

$$\textit{Slope, m} = \frac{\Delta\, \textit{Braking force}}{\Delta\, \textit{Time}} = \frac{9501.4 - 6955.1}{1.63 - 0.27} = 1872.28\,\frac{N}{s}$$

$$\textit{Braking force}_{t=0},\ y = 0\ N$$

Therefore $F = m * time + y = 1872.28 * time$

For 100 km/h and LVW

$$Slope,\ m = \frac{\Delta\ Braking\ force}{\Delta\ Time} = \frac{9582.9 - 7012.0}{3.29 - 0.55} = 938.28\ \frac{N}{s}$$

$Braking\ force_{t=0},\ y = 0\ N$

Therefore $F = m * time + y = 938.28 * time$

For 50 km/h and GVW

$$Slope,\ m = \frac{\Delta\ Braking\ force}{\Delta\ Time} = \frac{9707.1 - 6955.1}{1.87 - 0.31} = 1764.1\ \frac{N}{s}$$

$Braking\ force_{t=0},\ y = 0\ N$

Therefore $F = m * time + y = 1764.1 * time$

For 100 km/h and GVW

$$Slope,\ m = \frac{\Delta\ Braking\ force}{\Delta\ Time} = \frac{9707.1 - 6955.1}{3.75 - 0.62} = 879.23\ \frac{N}{s}$$

$Braking\ force_{t=0},\ y = 0\ N$

Therefore $F = m * time + y = 879.23 * time$

The next important parameter required to simulate braking is to simulate the weight of the vehicle. The vehicle weight is simulated by the use of inertia flywheels in ADAMS. There are four test specification therefore four flywheel inertia needs to be calculated. The following is example calculation for LVW of the vehicle at speed of the brake test at 100 km/h.

The weight transfer is assumed to be 60% because weight transfer is analytically calculated using the following formula[5]

$$WT = \frac{DR \times W \times CG}{WB}$$

where WT = weight transfer

DR = deceleration rate (in g value)

W = vehicle weight

CG = Centre of Gravity of vehicle

WB = Wheelbase of vehicle

Wheelbase and CG data of the Proton Iswara 1.3s is not available thus calculation of the weight transfer is not possible. Therefore, the writer has to assume a weight transfer of 60% for this simulation because it is the normal weight transfer value for normal braking. However, different value of weight transfer would be obtained for vehicles with different weight, wheelbase and CG.

Proton Iswara 1.3s weighs 1137 kg at LVW and taking braking speed at 100 km/h, and the braking load on the front wheels is assumed to be 60% to come to a complete halt, the brake must dissipate E_{kin}.

[5] Weight transfer formula obtained from Reference [1] Automotive Braking System by Thomas W. Birch

M = vehicle mass = 1137 kg

60 % weight transfer to front wheels = 1137 x 60/100 = 682.2 kg

Weight on one wheel = 682.2 /2 = 341.1 kg

$$v = \text{velocity} = 100\frac{km}{h} = 100\frac{km}{h}\left(\frac{1h}{3600s}\right)\left(\frac{1000m}{1km}\right) = 27.77\frac{m}{s}$$

substitute into kinetic energy formula

$$E_{kin} = ½[M v^2]$$

E_{kin} = ½ [341.1 x 27.77^2]

E_{kin} = 131,523.54 Joule

kinetic energy formula for rotating parts

$$E_{kin} = ½[I\ \omega^2]$$

I = mass moment of Inertia

ω = rotational velocity

= v /r = 27.77 / 0.117 = 237.35 rad / s (disc radius, r = 0.117 m)

I = 2 E_{kin} / ω^2

= 2(131,523.54) / 237.35^{2}

= 4.67 kg m^2

Bearing in mind that the inertia of a disc rotating around its own axis is :

$$I = ½\ [M r^2]$$

The energy dissipated by the brake is E = 131,523.54. To store this energy in a dynamometer, will require an inertia equal to I = 4.67 kg m^2.

If a single inertia flywheel were to be used, it would need:.

Diameter = 0.25 m

Mass = 149.44 ≈ 150 kg

The purpose of the inertia flywheel is to simulate the inertia of the vehicle when the accelerator is released once brake is pressed. The engine does not provide any energy to move the car once accelerator is released thus the car would move only on its momentum and inertia at the speed of braking.

The following table summarizes the data and parameters which are applied into ADAMS simulation for each of test specification.

Test Specification	Angular velocity (rad/sec)	Inertia (kg . m^2)	Mass of 1 flywheel (∅ = 0.25 m) / kg	Brake force function / N
LVW at 50 km/h	118.72	4.669	150	1872.28*time
LVW at 100 km/h	237.35	4.669	150	938.28*time
GVW at 50 km/h	118.72	5.395	173	1764.1*time
GVW at 100 km/h	237.35	5.395	173	879.23*time

Table 4.3. Parameters applied into ADAMS for different test specification

For the purpose of simulation, the flywheels will be replaced by a single flywheel with a fixed diameter of 0.25 m. For the simulation of the different test specification, the mass of the flywheel will be varied according to the value in Table 4.3 to simulate vehicle's weight during braking. Also instead of using two brake pads / caliper to apply force, only one caliper is used to apply the brake force because it will result in the same output as using two caliper with brake force divided equally between the two. However using one caliper decreases processing time. Figure 4.5 shows the model ready to be simulated in ADAMS.

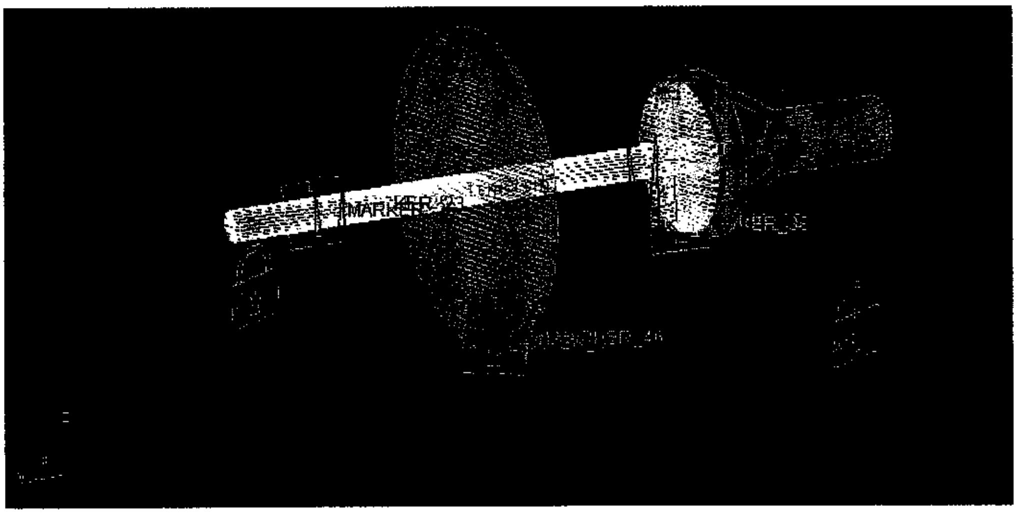


Figure 4.6. ADAMS Model ready for braking simulation

ADAMS braking simulation at 50 km/h LVW

Results for ADAMS Simulation with the following parameters:

Angular velocity = 118.72 rad/sec

Mass of flywheel (∅ = 0.25 m) = 150 kg

Brake force function = 1872.28*time

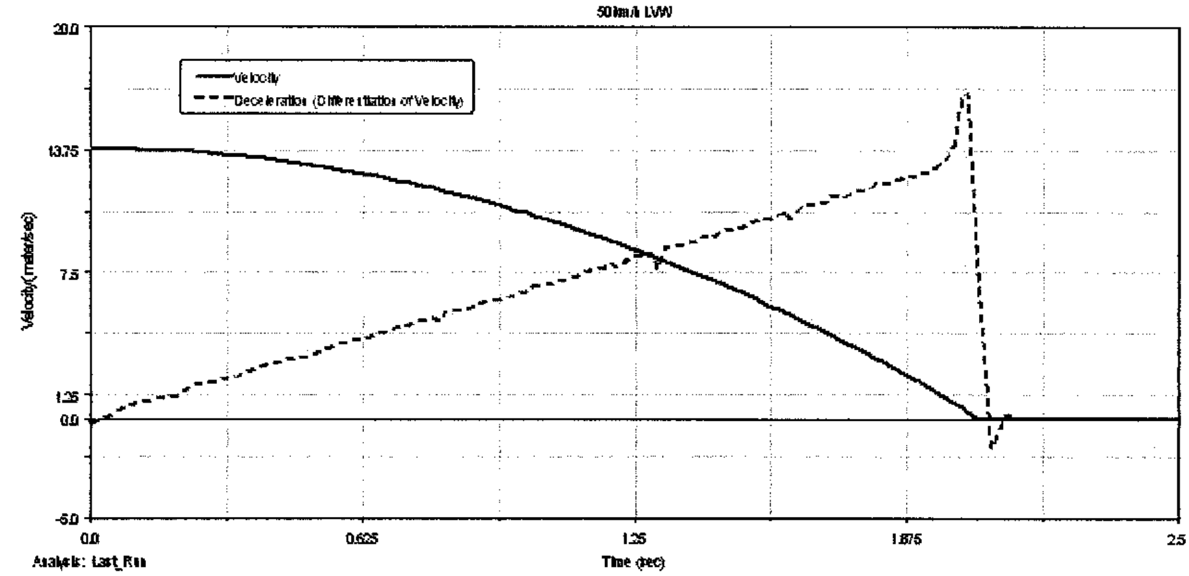


Figure 4.7. Graph of velocity and deceleration for 50 km/h LVW

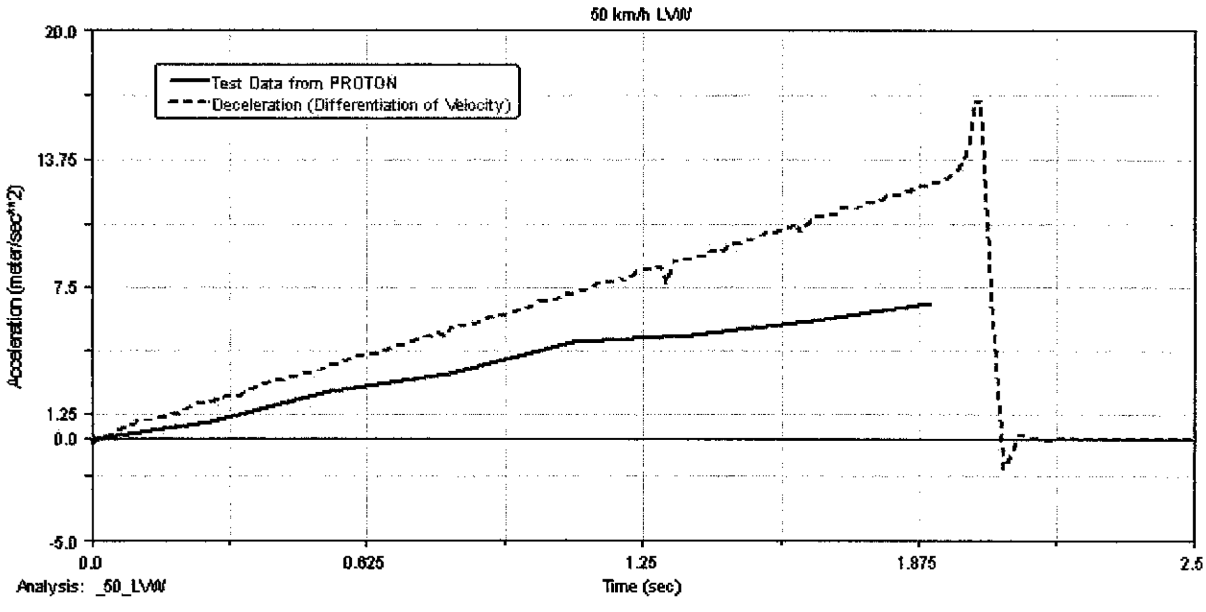


Figure 4.8. Graph of deceleration of ADAMS simulation and PROTON test data

ADAMS braking simulation at 50 km/h GVW

Results for ADAMS Simulation with the following parameters:

Angular velocity = 118.72 rad/sec

Mass of flywheel (∅ = 0.25 m) = 173 kg

Brake force function = 1764.1*time

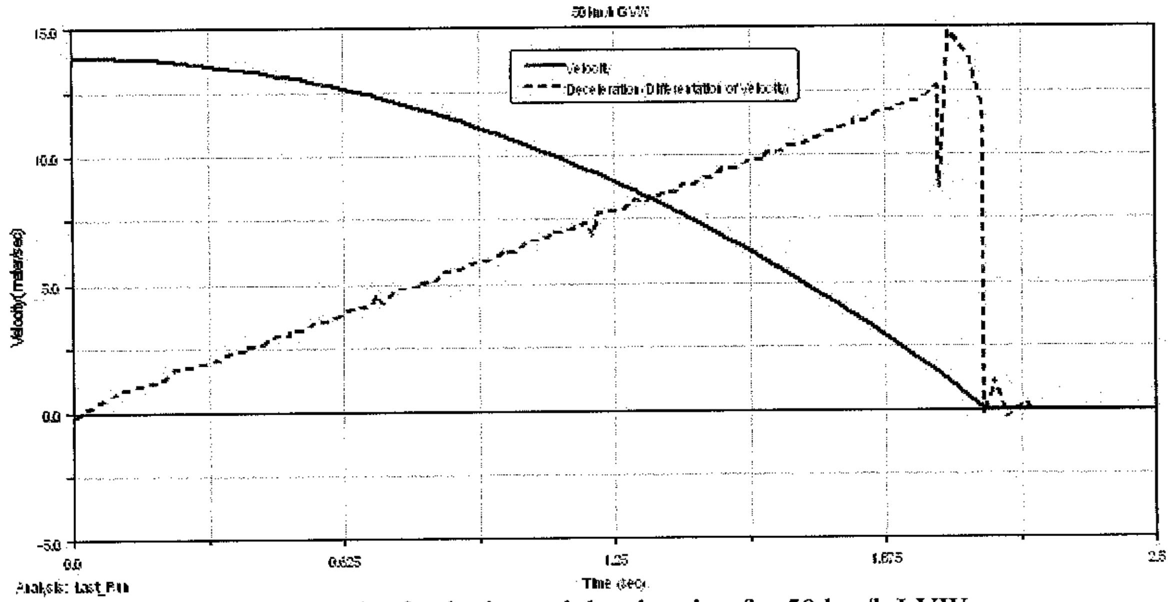


Figure 4.9. Graph of velocity and deceleration for 50 km/h LVW

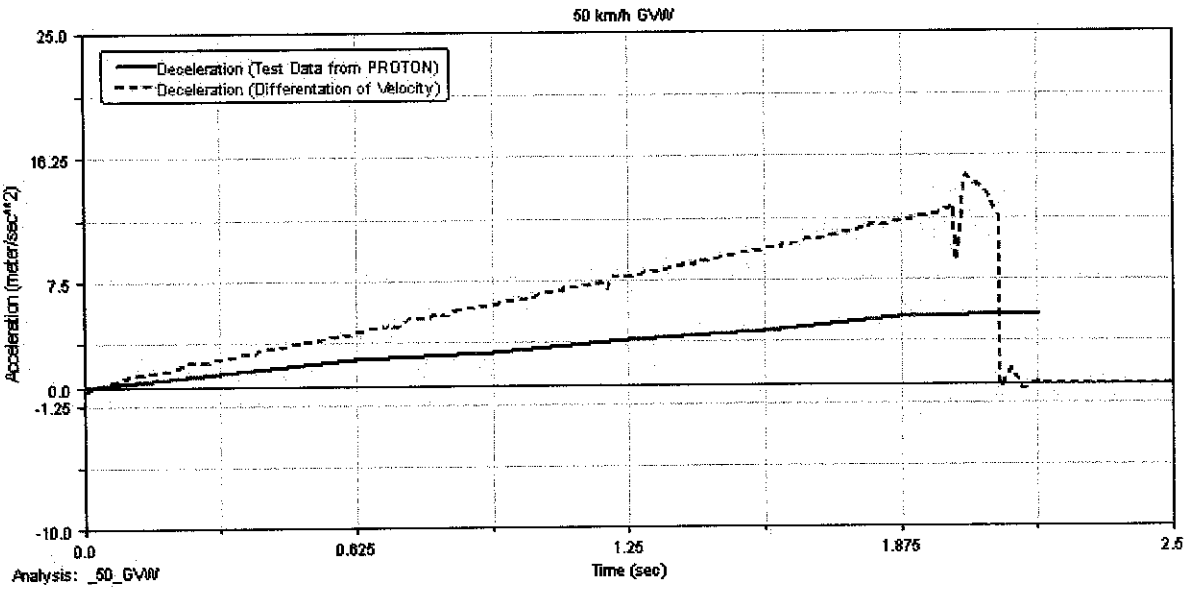


Figure 4.10. Graph of deceleration of ADAMS simulation and PROTON test data

ADAMS braking simulation at 100 km/h LVW

Results for ADAMS Simulation with the following parameters:

Angular velocity = 237.35 rad/sec

Mass of flywheel (∅ = 0.25 m) = 150 kg

Brake force function = 938.28*time

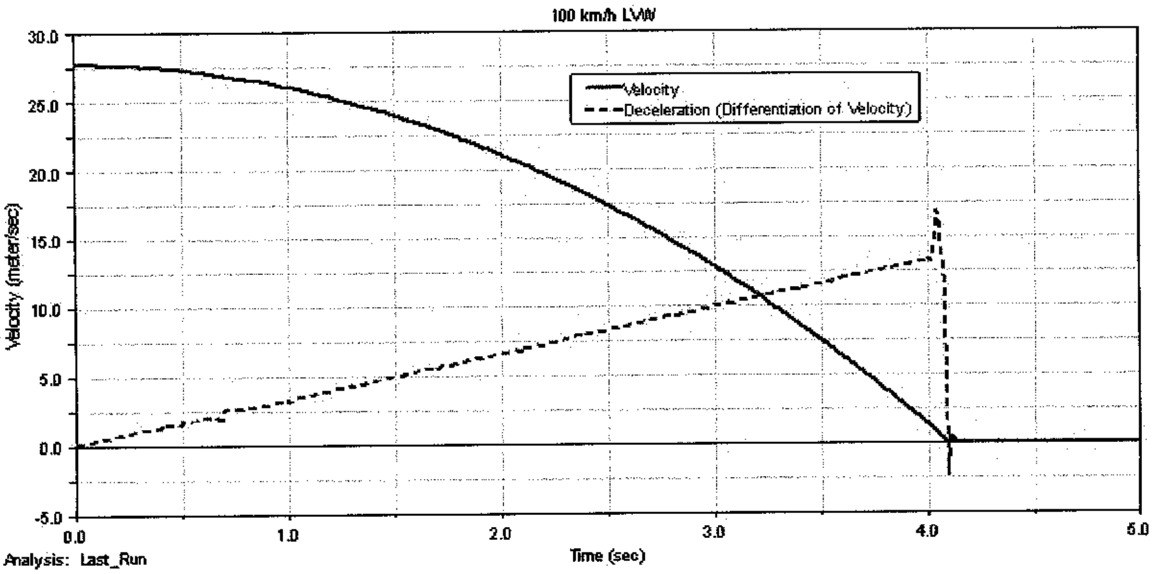


Figure 4.11. Graph of velocity and deceleration for 100 km/h LVW

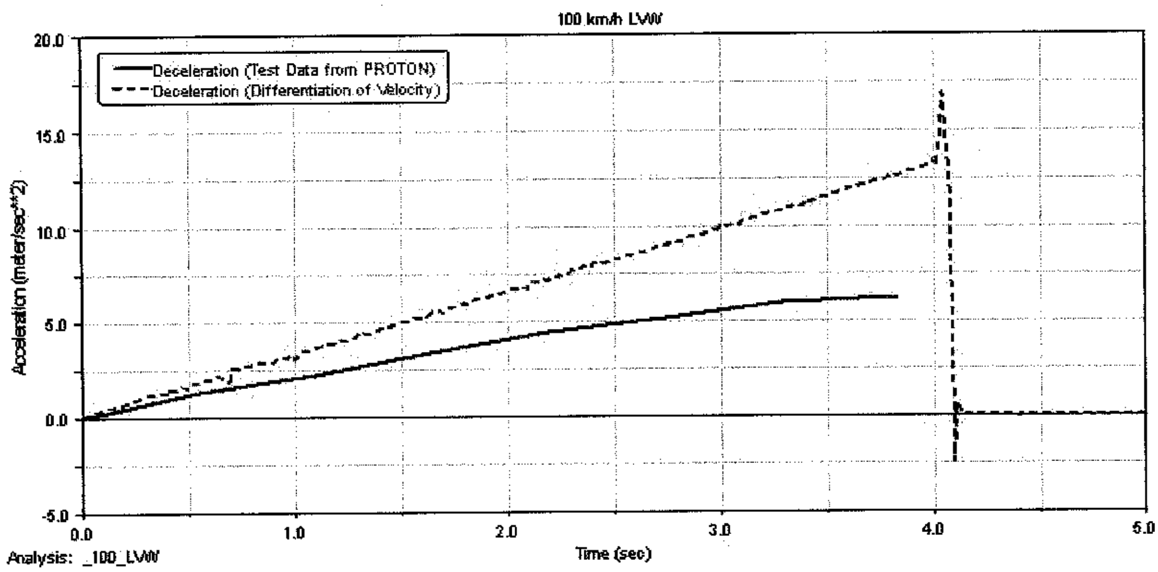


Figure 4.12. Graph of deceleration of ADAMS simulation and PROTON test data

ADAMS braking simulation at 100 km/h GVW

Results for ADAMS Simulation with the following parameters:

Angular velocity = 237.35 rad/sec

Mass of flywheel (∅ = 0.25 m) = 173 kg

Brake force function = 879.23*time

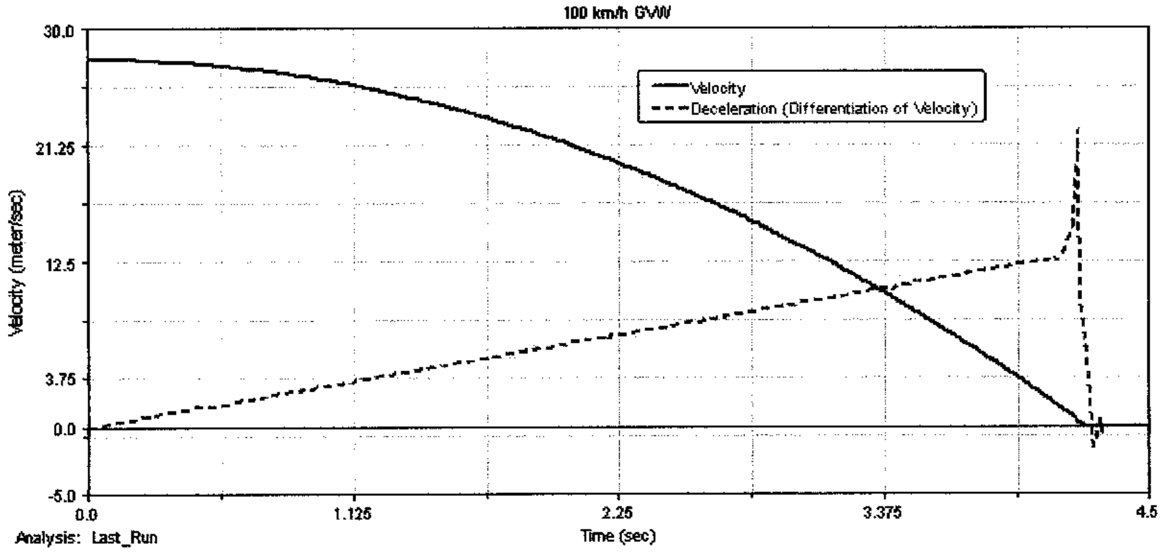


Figure 4.13. Graph of velocity and deceleration for 100 km/h GVW

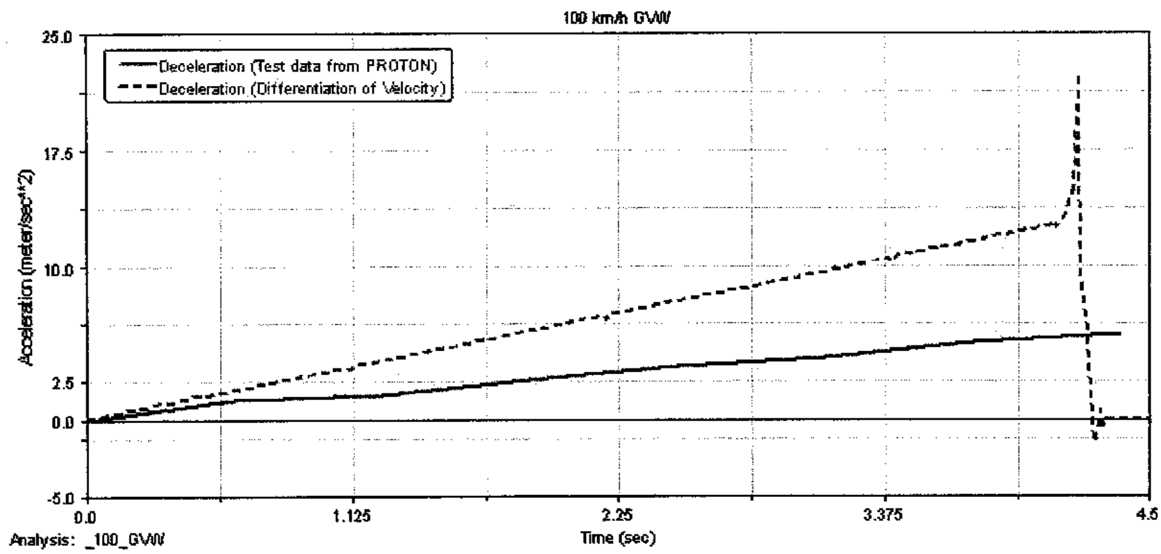


Figure 4.14. Graph of deceleration of ADAMS simulation and PROTON test data

4.5 RESULT ANALYSIS

This brake test simulation assumes that the simulation begins as soon as the velocity of the vehicle reaches the desired test velocity which is 50 km/h and 100 km/h respectively. Analysis of the result shows that the rotor rotates initially at the braking speed and gradually decreases as a result from the braking force from the caliper.

To obtain the deceleration of the rotor, the velocity plot is differentiated and negated using ADAMS Postprocessor Tool. The deceleration obtained from simulation is then compared with the deceleration data from actual braking test done by PROTON. From the results it can be seen that the objective has been accomplished whereby the simulation is able to simulate braking producing results as expected.

The brake force function which is calculated from the test data works to apply the brake force to the rotor accordingly as in actual braking condition. During simulation, braking force which is represented by single line force at caliper start to increase from the value of 0 N at t = 0 (initial condition where no pedal force is applied yet). The braking force will increase according to its brake force function which has been defined. Thus the velocity of the rotor decreases with time and results of the braking time is compared between simulated data and test data in Table 4.4.

Test Specification	Simulated braking time / sec	Test Data stopping time / sec	Deviation / sec	% Deviation
LVW at 50 km/h	2.0	1.9	0.1	5
LVW at 100 km/h	4.1	3.8	0.3	7.9
GVW at 50 km/h	2.1	2.2	0.1	4.5
GVW at 100 km/h	4.2	4.4	0.2	4.5

Table 4.4. Braking time results

Referring to the results, it can be seen as expected, braking time for test speed at 100 km/h is longer as compared to 50 km/h. This is due to the fact that the faster the vehicle, the more momentum and kinetic energy it carries thus more time is required for the brake to convert the rotor's kinetic energy to heat before going to rest. Since brake force applied is a function of time, therefore more braking force is required to stop a vehicle at 100 km/h than a vehicle at 50 km/h.

Weight of the vehicle also contributes to the braking time of the vehicle. The test data comprises of two set of weight which are Light Vehicle Weight (LVW) and Gross Vehicle Weight (GVW). However both LVW and GVW are not much different from each other thus the effects of both weights does not contribute much difference to the results. However, from the results, it can be seen that more braking time is required for braking with a heavier car. This is because the heavier the car, the longer it takes for the vehicle to come to a complete halt because it carries more momentum for a vehicle at the same speed. Therefore more braking force is required to stop a vehicle with GVW than a vehicle with LVW.

The objective of this project is to simulate actual braking condition and the only way to ensure this is by verifying the result with a benchmark data. The test data which is obtained from actual brake test on Proton Iswara 1.3s is used as benchmark to verify the results from ADAMS simulation. Each of the deceleration result from each test specification is compared with deceleration data from Proton test data. From each of the result, it can be seen that the ADAMS model can successfully simulate actual braking condition based on the parameters of actual brake testing conducted by Proton.

From the results, ADAMS simulation exhibits actual braking condition whereby longer braking time is required for a higher vehicle speed and weight. However the writer was unable to produce deceleration results without any deviation or error as compared to the test data from Proton because of the assumptions used in the calculation of the parameters in the simulation. Some of the reasons behind this deviation in result are explained as follows.

In any simulation, we are only rarely able to reproduce the exact environment within which the item tested must perform. The vehicle brings stresses into play that are almost impossible to reproduce in the simulation. Among these are weight transfer and efficiency between the front and rear brakes, the environment (temperature) mechanical stresses, deformity caused by contact with the ground and the effect of vibrations that are not produced by the brake itself.

As mentioned in the result, several data which are required for the calculation of the vehicle's weight transfer is not available thus the writers assumes a weight transfer of 60% for all test specification. The weight of the vehicle is simulated through the use of inertia flywheels to simulate the inertia exerted by the vehicle's mass once brake is applied and no power from the engine. The calculation used to measure this inertia is based on an assumption that all the kinetic energy from the rotor is converted to heat. This is not true in reality where many other forces should be taken into account such as lateral forces, ground forces, vibration forces etc. The simulation test also assumes that the braking takes place in a straight road or straight line without any cornering involved. In actual when braking to a complete halt, there is a tendency to steer to one side. Therefore weight transfer should be recalculated and lateral forces should be taken into account.

The calculation of the brake force function is calculated based on the knowledge of the brake pedal force from the test data. This lengthy calculation is shown in the result section. The calculation used to convert brake pedal force to brake force function is very much reliable. However, in actual braking, there will definitely be some power loss as a result of many contributing factors such as booster vacuum pressure variation, pressure loss in vacuum line, power loss from engine, hydraulic line pressure loss etc. Therefore, even with the usage of such reliable calculation, it is almost impossible to simulate actual braking force simply through mathematical calculation.

Also in reality, the friction between caliper and the rotor produces a massive amount of heat and this heat will affect the brake performance in many ways. Heat may change the property of the hydraulic fluid supplying the brake force to the caliper and also change the property of the brake pads and rotor. Overheating of the brake pads and rotor may result in brake failure. But in actual, the heat is dissipated to the environment with the help of winds which result from the vehicle's movement that helps to dissipate the heat faster. But in ADAMS simulation, no such heat is produced and no such wind is simulated. Therefore some error should occur from this factor.

When a vehicle is motion, the vehicle would be subjected to drag forces. Drag force is the force opposing the motion of the vehicle. Drag can only be reduced but never eliminated therefore the drag actually help the vehicle to slow down during braking. Even though the effect of drag is more relevant on moving vehicle rather than braking but it still contributes some braking effect. ADAMS simulation does not take into account drag forces. The braking in ADAMS is applied only by the brake force function and contact forces which are defined in the simulation thus results in some deviation in the deceleration as compared to the test data.

There are also several restrictions to the model that cannot be simulated. Since the model does not take into account the friction between the wheels and the road, therefore it cannot simulate a condition called 'slip' where the brakes lock the wheels even but the vehicle is still moving. When slip occurs the wheel would be locked and has zero velocity but the vehicle would still be moving. This simulation assumes that no slip can occur during the testing.

However, having mentioned all the assumptions and errors in the ADAMS simulation, actual simulation using the jig would result in much accurate results because the simulation would consider actual braking condition and actual brake parts. Hydraulic pressure will be used to apply brake force to the caliper. The wind may be simulated by placing a fan towards the rotor to dissipate the heat generated. ADAMS Simulation has verified that this jig design is applicable to be used as a measure to simulate actual braking condition.

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

The objective of this project which is to design a brake test jig to simulate actual braking condition has been accomplished successfully. Three conceptual designs have been generated and from the three, one design which is based on dynamometer concept was chosen to be the final jig design to be simulated in ADAMS. The final design is generated with the actual dimension of disc brake from Proton Iswara 1.3s. The shaft which is a critical part of the jig has been structurally analyzed using CATIA's analysis tool and the results are within acceptable limits.

The jig which has been chosen to simulate braking in ADAMS has been successfully simulated based on the parameters calculated from the raw data obtained from PROTON. The raw data provided by PROTON is converted to data which is applicable to the simulation through a series of calculation. Simulation in ADAMS is conducted exactly as the writer intends to which is based on inertial dynamometer concept.

The ADAMS simulation has generated results for all four test specifications of braking which PROTON conducts for brake performance test of Proton Iswara 1.3s. The results of the deceleration in ADAMS simulation is verified by comparing it with the test data from PROTON and it can be seen that both results shows a very similar pattern with small deviation. This deviation is caused by the assumptions made in the simulation. Therefore the writer concludes that the verification of the simulated result and test data confirms that the design of the jig is feasible and practical thus further steps to fabricate the jig and conduct lab simulation of braking may be conducted.

5.2 RECOMMENDATIONS

The brake test jig design is successfully able to simulate braking condition based on the parameters of brake performance test conducted by Proton. However, there are many rooms for improvement of the design to be able to simulate and produce much more accurate results.

A complete data of the vehicle in study is required to be able to calculate all the required parameters needed to simulate braking as close as possible to actual braking. Assumptions should be reduced as much as possible in the calculations.

The development of the design should include the fabrication of the jig to enable the jig to simulate braking in a much more realistic environment. Sensors such as thermocouple, accelerometer, noise microphone, pressure, force and torque sensor, etc may be included in the fabricated jig to better monitor the performance of the disk brake being tested. The brake test jig may be implemented for the use of automotive industries to reduce their cost of brake testing.

The brake test jig may also be implemented as an educational tool for universities, colleges or schools as it could serve an excellent method of teaching. Students can understand how brakes function better by observing the operation of the brake test jig. Furthermore, the ease of installation and operation enables students to study any type of disk brakes that may be used for research or educational purposes.

Last but not least, the brake test jig should be developed to be able to conduct braking simulation not only for disk brakes but also drum brakes. Drum brakes are still used in the automotive industry thus there should be the need to test their performance and safety.

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APPENDICES

	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Selection of Project Topic																
	-Propose Topic																
	-Project Award																
2	Preliminary Research Work																
	-Introduction																
	-Objective																
	-List of references/literature																
	-Project planning																
3	Submission of Preliminary Report				15/8												
4	Project Work																
	-Literature Review & Research																
	-CAD Software Training																
	-Conceptual Design																
5	Submission of Progress Report									22/9							
6	Project work continue																
	-Literature Research																
	-Design Stage 1																
7	Submission of Interim Report Final Draft													20/10			
8	Oral Presentation															28/10	
9	Submission of Interim Report																21/11

SW Study Week
EW Exam Week

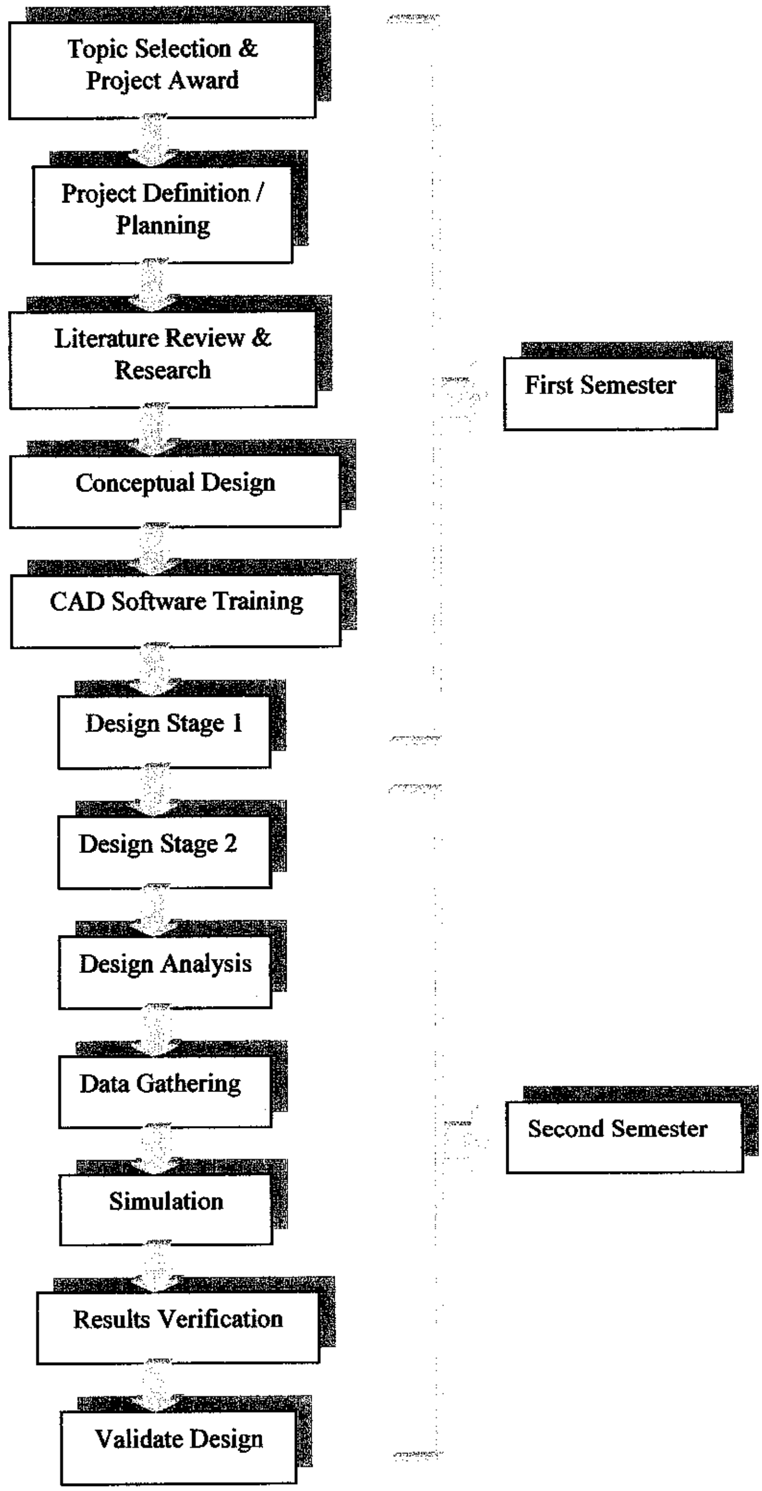
Completed
On – Going
Milestone

No.	Detail/ Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	SW	EW
1	Project Work Continue																
	-Design Stage 2																
	-Design Analysis																
2	Submission of Progress Report 1				16/2												
3	Project Work Continue																
	- Data Gathering for Simulation																
	- Design Simulation																
4	Submission of Progress Report 2									22/3							
5	Project work continue																
	- Simulation Results																
	- Design Improvements																
6	Poster Submission												12/4				
7	Submission of Dissertation Final Draft													20/4			
8	Oral Presentation															26/4	
9	Submission of Project Dissertation																31/5

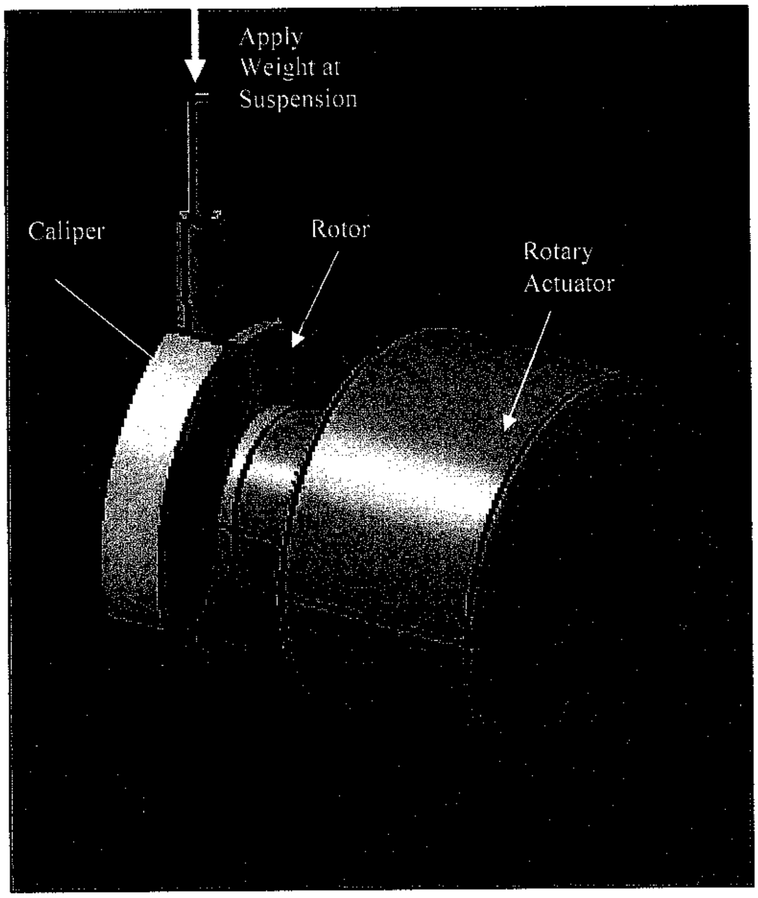
SW Study Week
EW Exam Week

Completed
On – Going
Milestone

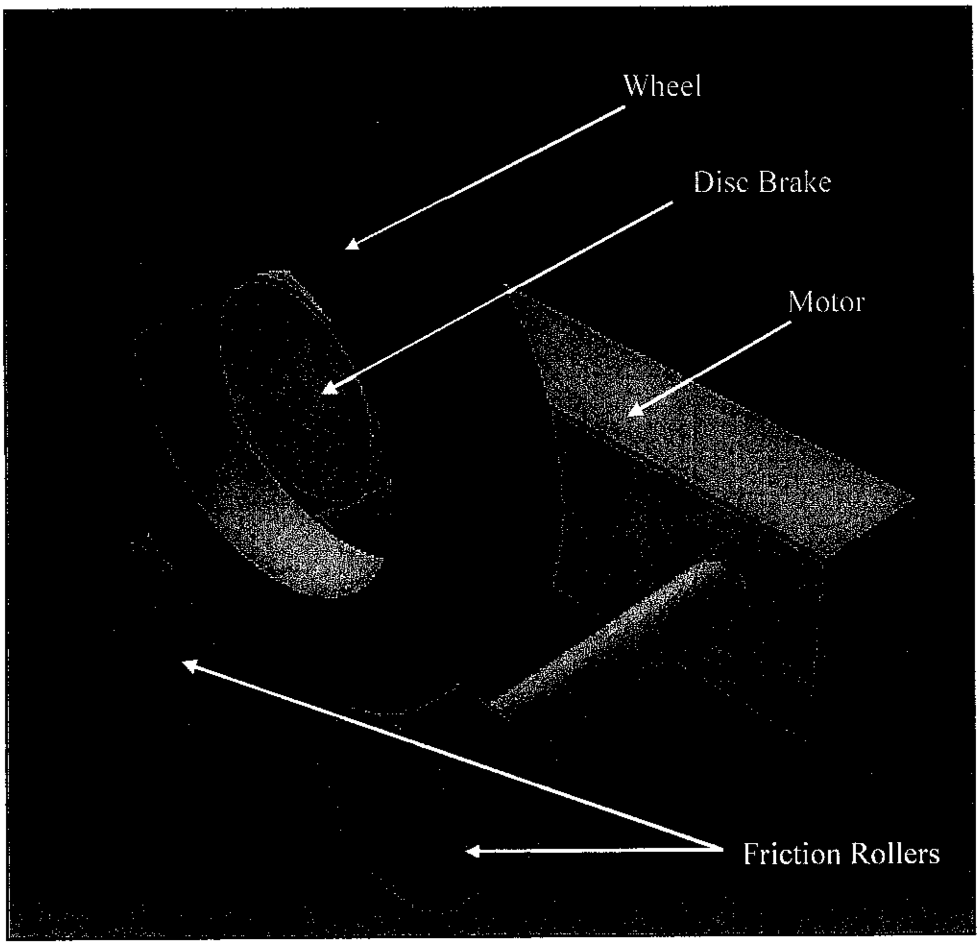
APPENDIX 3 – PROJECT FLOW PROCESS



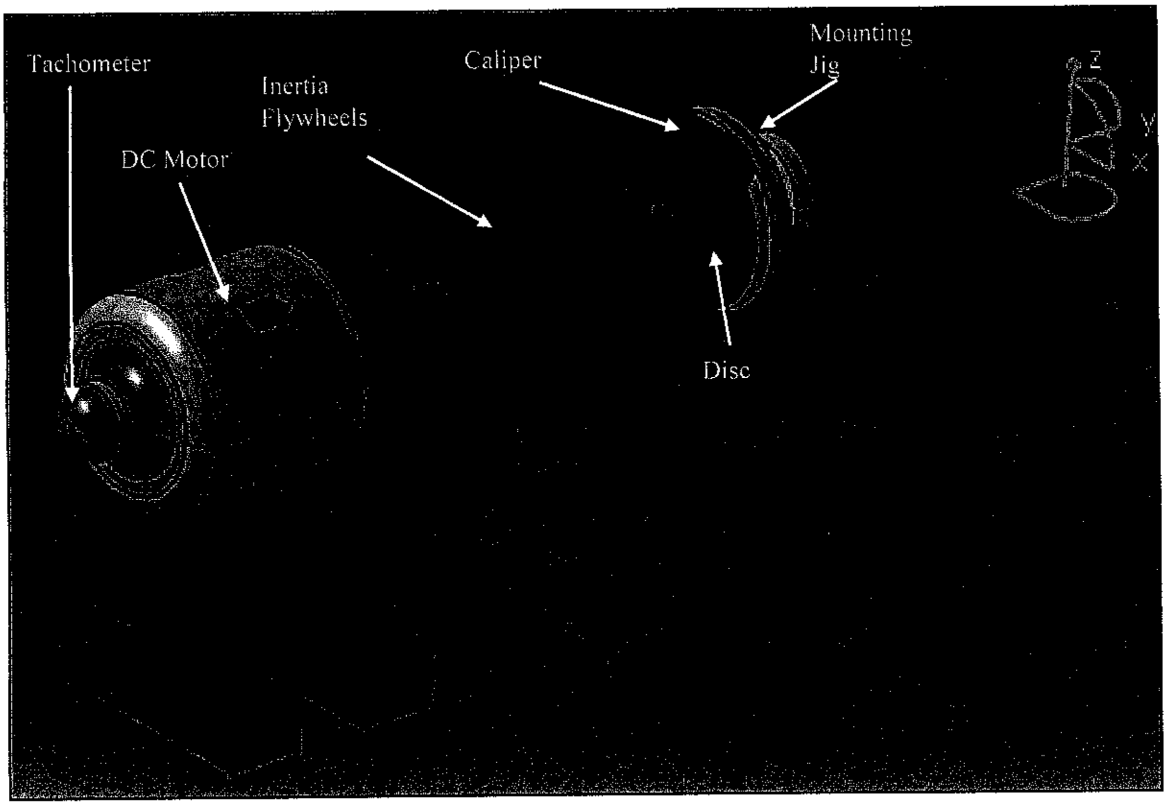
APPENDIX 4



Design Concept 01

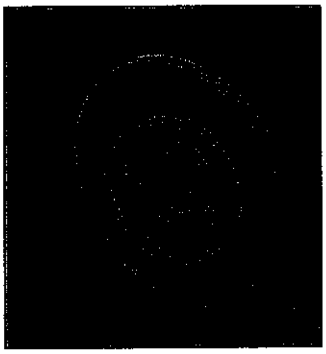


Design Concept 02

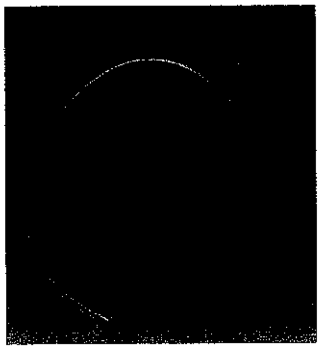


Design Concept 03

APPENDIX 5:
Parts Final Design

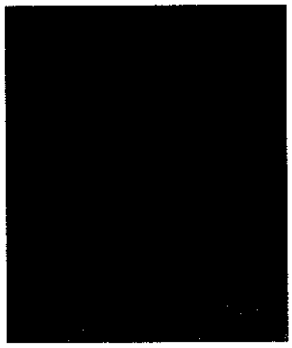


Rotor

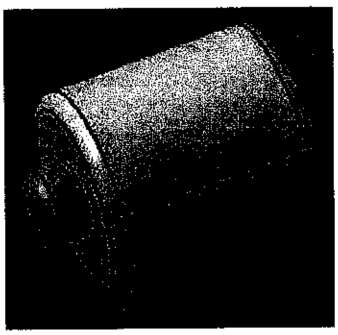


Caliper with
Brake Hose

Shaft



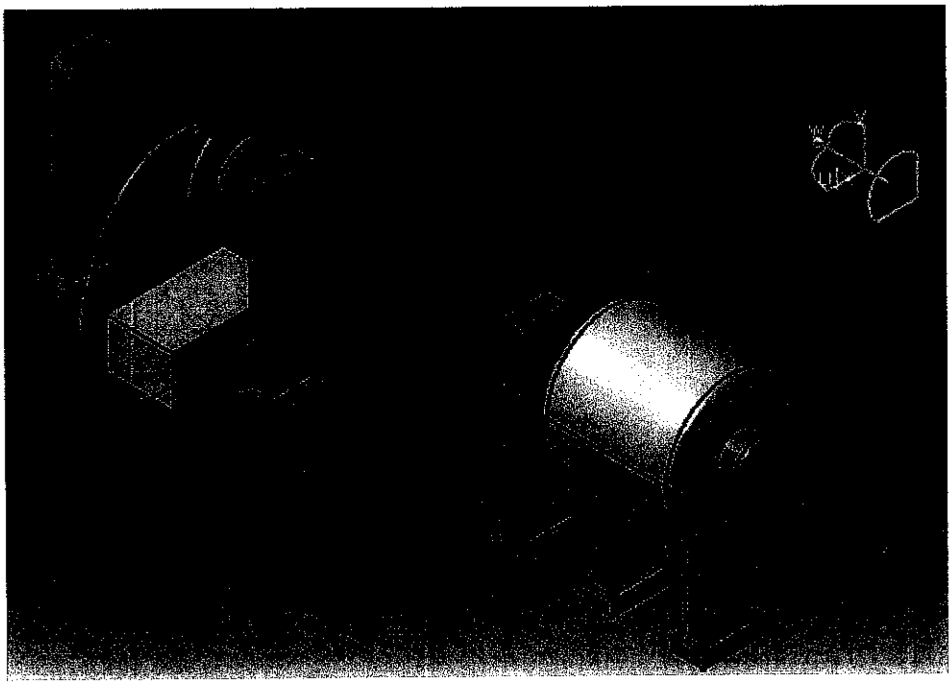
Inertia Flywheels



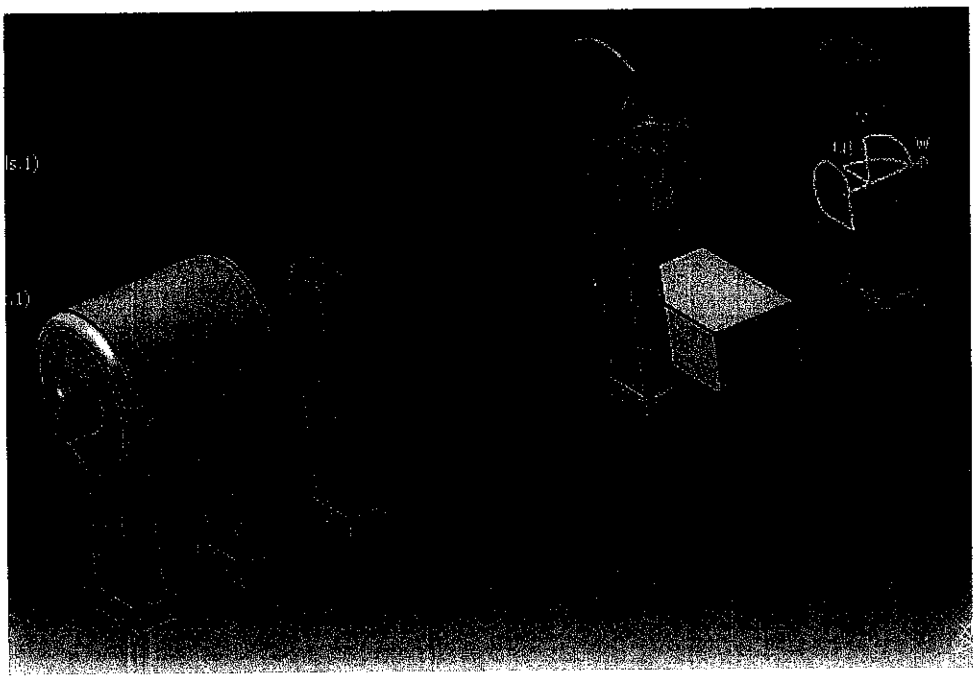
AC Motor

Mounting Jig

Assembled Final Design



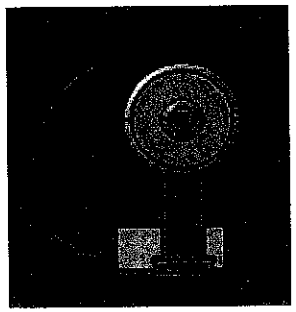
Isometric View 1



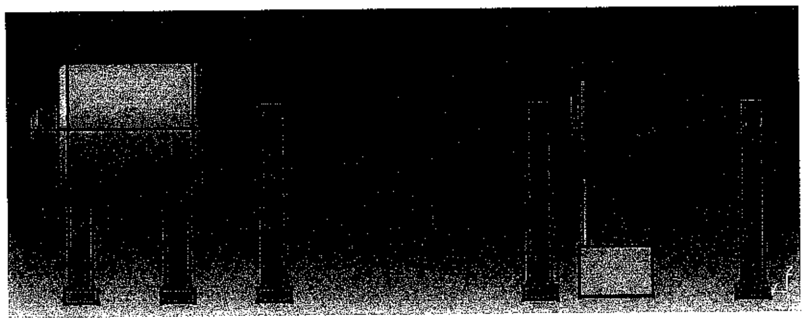
Isometric View 2

Assembled Final Design

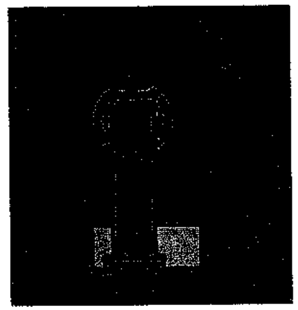
Top View



Front View



Side View

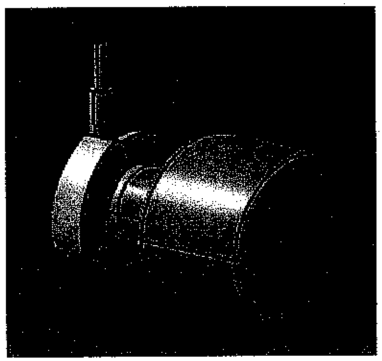


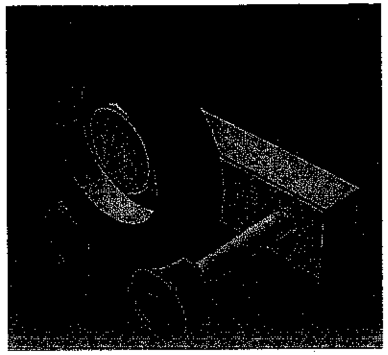
Rear View

Rating Scale

1 ⟷ 10

poor good





Criteria	Weighting	Rating	Weighted score	Rating	Weighted score	Rating	Weighted score
Simulate actual braking	10	9	90	6	60	10	100
Output measurement	10	6	60	4	40	9	90
Flexible	8	8	64	8	64	9	72
Simulation method	8	5	40	8	64	9	72
Primer mover	8	8	64	8	64	9	72
Variable speed	10	4	40	4	40	10	100
Simulate weight	6	5	30	6	36	8	48
Space for hydraulic sys	8	2	16	4	32	6	48
Cost	4	6	24	4	16	8	32
Size & Weight	5	6	30	5	25	7	35
Maintenance	6	4	24	6	36	6	36
Durability	8	5	40	6	48	8	64
Adjustability	8	4	32	4	32	8	64
Total weight			**554**		**557**		**833**

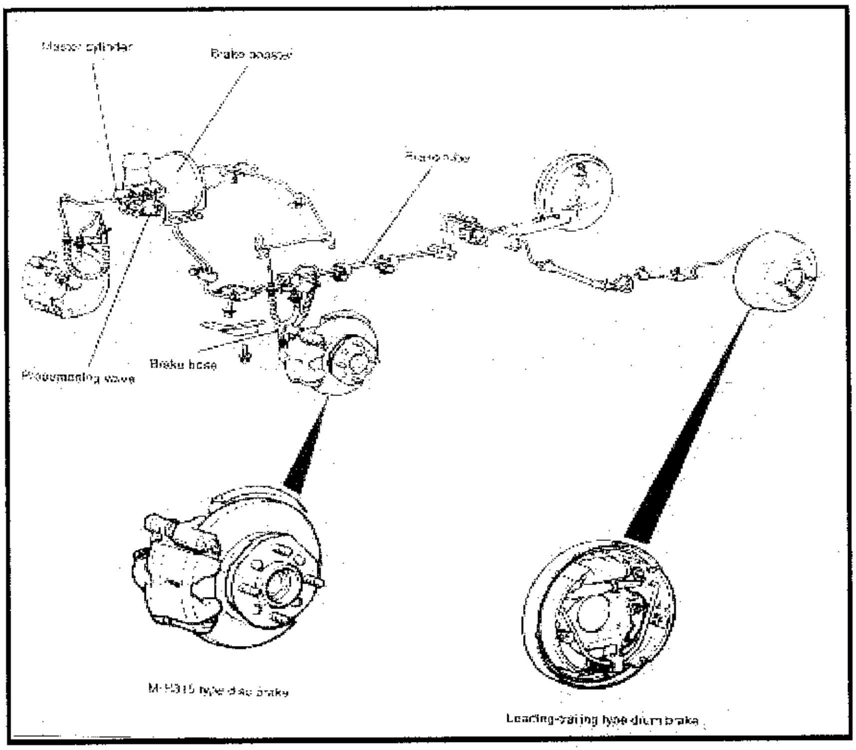
APPENDIX 6: Conceptual Design Evaluation Matrix

APPENDIX 7:

List of equipment used by Proton Research and Development Department to conduct testing on brake system

1. DC 100
2. Indicator
3. Pedal Force Sensor
4. Parking Brake Force Lever Sensor
5. Non Contact Speedometer
6. Vehicle Speed Sensor
7. Thermocouple
8. Pressure Sensor
9. Digital Pressure Meter
10. Digital Dial Gauge
11. Digital Mini Processor
12. Torque Meter
13. Data Acquisition System
14. Wheel Speed Encoder
15. Terminal Display
16. LCD Monitor

APPENDIX 8:



Brake System Layout for Proton Saga / Iswara 1.3

APPENDIX 9:

1.4 SPECIFICATIONS

Master cylinder	
Type	Tandem type
I.D. mm	20.64
Brake booster	
Type	Vacuum type
Effective dia. of power cylinder mm	180
Boosting ratio (Brake pedal depressing force)	3.5 (at 21 kg)
Proportioning valve	
Split point kg/cm²	27
Decompression ratio	0.3
Front brakes	
Type	M-R215 solid disc
Disc O.D. mm	234
Pad thickness mm	10.0
Disc thickness mm	13
Cylinder I.D. mm	[illegible]
Clearance adjustment	Automatic
Rear brakes	
Type	Leading-trailing, drum
Drum I.D. mm	180
Brake lining thickness mm	4.3
Cylinder I.D. mm	19.0
Clearance adjustment	Automatic
Parking brake	
Type	Mechanical brake acting on rear wheels
Brake lever type	Lever type
Cable arrangement	V-type

Proton Iswara / Saga Brake Specification

APPENDIX 10:

proton

Perusahaan Otomobil Nasional Berhad

CHASSIS DEVELOPMENT,
HOMOLOGATION & TESTING DEPT.,
RESEARCH & DEVELOPMENT DIVISION.

VEHICLE WEIGHT

Test Car	lm GC4	*Master Cylinder*			*Servo Ratio*	
Tester	saidon/rashidi	*Booster Type*			*Tyre Press*	
Date		*Fr. Brake*	*Type*		*Colour*	
Max. Torque			*W/Cylin*		*T/Mission*	
Max. Power			*Eff. Rad*		*ABS/Non ABS*	Non ABS
Ambient Temp.		*Rr. Brake*	*Type*		*Engine*	1.3l
W/Speed			*W/Cylin*			
Humidity			*Eff. Rad*			

KERB WEIGHT

FR. LEFT	FR. RIGHT	TOTAL FRONT
272.5 kg	272.5 kg	545 kg
RR. LEFT	**RR. RIGHT**	**TOTAL REAR**
187.5 kg	187.5 kg	375 kg
SPEC	TOTAL	920 kg

L.V.W. (MID LADEN)

FR. LEFT	FR. RIGHT	TOTAL FRONT
321.4 kg	320.4 kg	641.8 kg
RR. LEFT	**RR. RIGHT**	**TOTAL REAR**
240.6 kg	254.6 kg	495.2 kg
SPEC	TOTAL	1137 kg

L.V.W. (MID LADEN)

FR. LEFT	FR. RIGHT	TOTAL FRONT
334.4 kg	328.8 kg	663.2 kg
RR. LEFT	**RR. RIGHT**	**TOTAL REAR**
312.4 kg	338 kg	650.4 kg
SPEC	TOTAL	1313.6 kg

Proton Saga / Iswara Weight Data

APPENDIX 11

BRAKE PERFORMANCE TEST

DYNAMIC BRAKE TEST	LIGHTLY LADEN WEIGHT EFFECTIVENESS

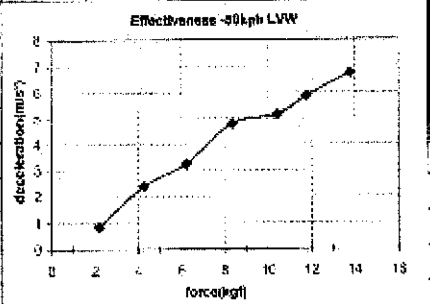
Test Car	LM 41	*Master Cylinder*			*Servo Ratio*	
Tester		*Booster Type*			*Tyre Press*	
Date		*Fr. Brake*	*Type*		*Colour*	
Max. Torque			*W/Cylin*		*T/Mission*	
Max. Power			*Eff. Rad*		*Brake Type*	
Ambient Temp.		*Rr. Brake*	*Type*	drum	*Engine*	1.3
W/Speed			*W/Cylin*		*Test Weight*	
Humidity			*Eff. Rad*			

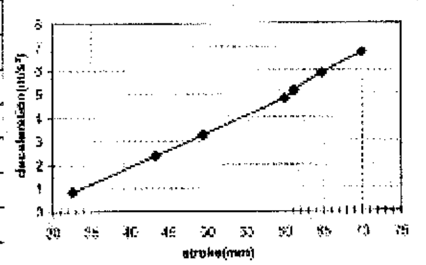
(LIGHTLY LADEN WEIGHT EFFECTIVENESS = kg

Braking Speed : 50 0Km/hr

Gear Position : Braking Interval: ± 5 ° C

	Pedal Force (kg)	Stopping Distance (m)	Time (sec)	Average fr press (bar)	Average rear press (bar)	Pedal Stroke (mm)	Decel (m/s²)	Remarks
1	2.19						0.83	
2	4.28						2.38	
3	6.25						3.23	
4	8.37						4.78	(5.5kgf , 3m/s²)
5	10.45						5.13	(*2kg* , 6m/s²)
6	11.6						5.98	
7	13.6						6.74	
8								
9								
10								
11								
12								
13								
14								
15								
16								
17								
18								





Lock G		Lock Wheel	

Raw data for LVW at 50 km/h

APPENDIX 12

BRAKE PERFORMANCE TEST	
DYNAMIC BRAKE TEST	LIGHTLY LADEN WEIGHT EFFECTIVENESS

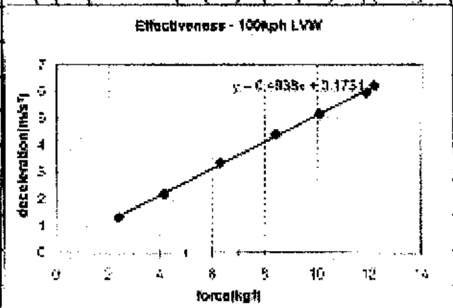
Test Car		*Master Cylinder*			*Servo Ratio*	
Tester		*Booster Type*			*Tyre Press*	
Date		*Fr. Brake*	*Type*		*Colour*	
Max. Torque			*W/Cylin*		*T/Mission*	
Max. Power			*Eff. Rad*		*Brake Type*	
Ambient Temp.		*Rr. Brake*	*Type*		*Engine*	
W/Speed			*W/Cylin*		*Test Weight*	
Humidity			*Eff. Rad*			

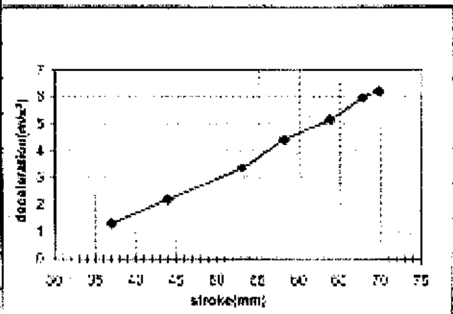
LIGHTLY LADEN WEIGHT EFFECTIVENESS = kg

Braking Speed :	100	0Km/hr		
Gear Position :		Braking Interval:		± 5 ° C

	Pedal Force (kg)	Stopping dist (m)	Time (sec)	Average fr press (bar)	Average rear press (bar)	Pedal Stroke (mm)	Decel (G)	Remarks
1	2.4						1.31	
2	4.16						2.2	
3	6.3						3.36	
4	8.42						4.41	(5.7kgf , 3m/s²)
5	10.1						5.15	(12kgf , 6m/s²)
6	11.9						5.96	
7	12.2						6.22	
8								
9								
10								
11								
12								
13								
14								
15								
16								
17								
18								

Lock G		Lock Wheel	





Raw data for LVW at 100 km/h

APPENDIX 13

BRAKE PERFORMANCE TEST	
DYNAMIC BRAKE TEST	GROSS VEHICLE WEIGHT EFFECTIVENESS

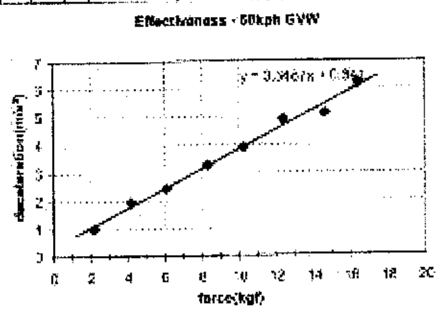
Test Car		*Master Cylinder*			*Servo Ratio*	
Tester		*Booster Type*			*Tyre Press*	
Date		*Fr. Brake*	*Type*		*Colour*	
Max. Torque			*W/Cylin*		*T/Mission*	
Max. Power			*Eff. Rad*		*Brake Type*	
Ambient Temp.		*Rr. Brake*	*Type*		*Engine*	
W/Speed			*W/Cylin*		*Test Weight*	
Humidity			*Eff. Rad*			

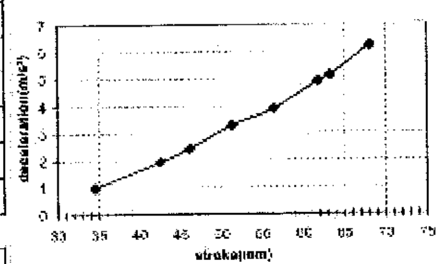
(GROSS VEHICLE WEIGHT EFFECTIVENESS = kg

Braking Speed : 50 0Km/hr

Gear Position : Braking Interval: ± 5 ° C

	Pedal Force (kg)	Frt. Whl I & A (°C)	Rr. Whl I & A (°C)	Average fr press (bar)	Average rear press (bar)	Pedal Stroke (mm)	Decel (G) 50kph	Remarks 100kph
1	2.19						0.99	
2	4.18						1.93	
3	6.09						2.43	
4	8.3						3.27	(7.6kgf , 3m/s²)
5	10.25						3.86	(16.1kgf , 6m/s²)
6	12.36						4.86	
7	14.6						5.09	
8	16.4						6.19	
9								
10								
11								
12								
13								
14								
15								
16								
17								
18								





Lock G		Lock Wheel	

Raw data for GVW at 50 km/h

APPENDIX 14

BRAKE PERFORMANCE TEST

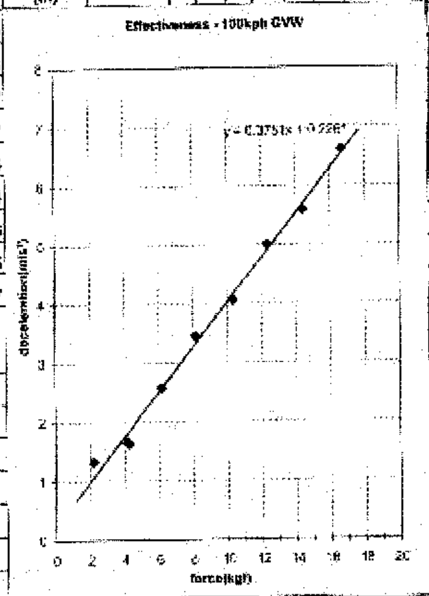
DYNAMIC BRAKE TEST	GROSS VEHICLE WEIGHT EFFECTIVENESS

Test Car		*Master Cylinder*			*Servo Ratio*	
Tester		*Booster Type*			*Tyre Press*	
Date		*Fr. Brake*	*Type*		*Colour*	
Max. Torque			*W/Cylin*		*T/Mission*	
Max. Power			*Eff. Rad*		*Brake Type*	
Ambient Temp.		*Rr. Brake*	*Type*		*Engine*	
W/Speed			*W/Cylin*		*Test Weight*	
Humidity			*Eff. Rad*			

(GROSS VEHICLE WEIGHT EFFECTIVENESS = kg

Braking Speed :	100	0Km/hr		
Gear Position :		Braking Interval:		± 5 ° C

	Pedal Force (kg)	Stopping distance (m)	Time (sec)	Average fr press (bar)	Average rear press (bar)	Pedal Stroke (mm)	Decel (G)	Remarks
1	2.19						1.32	
2	4.26						1.62	
3	4.17						1.65	
4	6.17						2.57	
5	8.16						3.44	(7.2kgf , 3m/s²)
6	10.4						4.05	(15.2kgf , 6m/s²)
7	12.36						5	
8	14.45						5.59	
9	16.7						6.61	
10								
11								
12								
13								
14								
15								
16								
17								
18								



Lock G		Lock Wheel	

Raw data for GVW at 100 km/h

APPENDIX 15

Derivation of Braking Time

For 50 km/h LVW (1137 kg)

Average brake force = 8288 N

$$D_x = F_x = \frac{Fx}{M} = \frac{8288}{1137} = 7.289 \text{ m/s}^2$$

$$\text{Stopping Distance, SD} = \frac{V_o^2}{2D_x} = \frac{(50km/h \times 1000m/km \times 1h/3600s)^2}{2 \times 7.289m/s^2} = 13.23 \text{ m}$$

$$\text{Total Braking Time, } t_s = \frac{V_o}{D_x} = \frac{13.88m/s}{7.289m/s^2} = 1.9 \text{ s}$$

For 100 km/h LVW (1137 kg)

Average brake force = 8239.18 N

$$D_x = F_x = \frac{Fx}{M} = \frac{8239.18}{1137} = 7.2464 \text{ m/s}^2$$

$$\text{Stopping Distance, SD} = \frac{V_o^2}{2D_x} = \frac{(100km/h \times 1000m/km \times 1h/3600s)^2}{2 \times 7.2464m/s^2} = 53.24 \text{ m}$$

$$\text{Total Braking Time, } t_s = \frac{V_o}{D_x} = \frac{27.78m/s}{7.2464m/s^2} = 3.833 \text{ s}$$

For 50 km/h GVW (1310 kg)

Average brake force = 8323.3 N

$D_x = F_x = \frac{Fx}{M} = \frac{8323.3}{1310} = 6.354 \text{ m/s}^2$

Stopping Distance, SD = $\frac{V_o^2}{2D_x} = \frac{(50km/h \times 1000m/km \times 1h/3600s)^2}{2 \times 6.354m/s^2} = 15.18 \text{ m}$

Total Braking Time, $t_s = \frac{V_o}{D_x} = \frac{13.89m/s}{6.354m/s^2} = 2.186 \text{ s}$

For 100 km/h GVW (1310 kg)

Average brake force = 8320.9 N

$D_x = F_x = \frac{Fx}{M} = \frac{8320.9}{1310} = 6.352 \text{ m/s}^2$

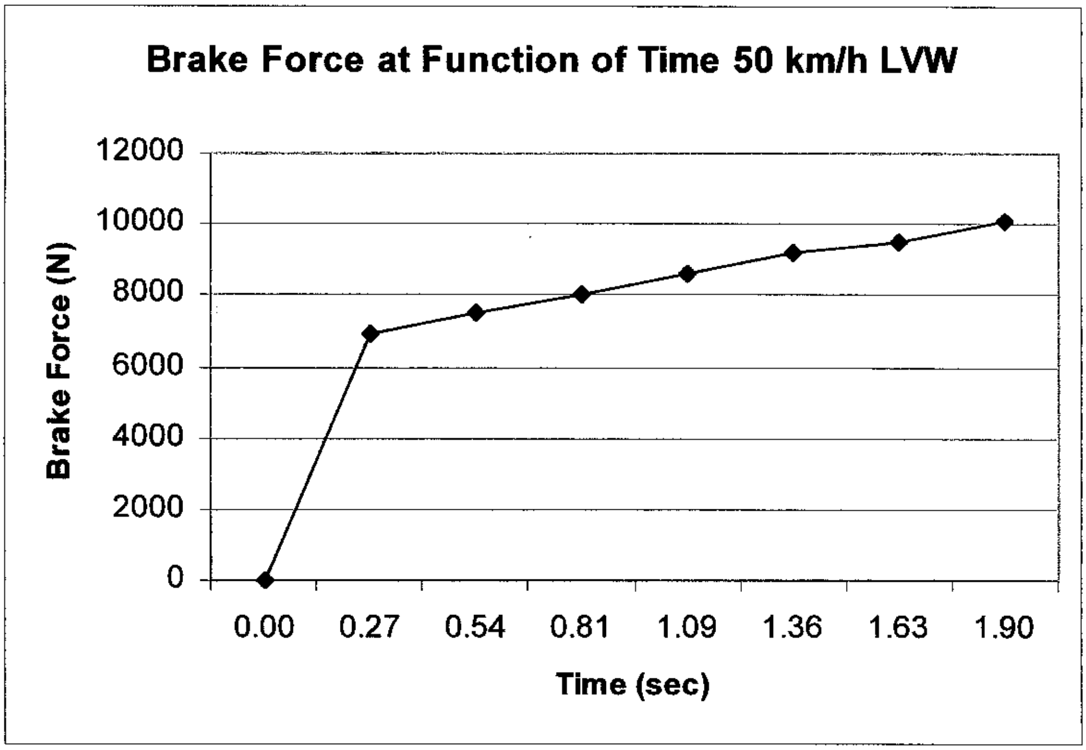
Stopping Distance, SD = $\frac{V_o^2}{2D_x} = \frac{(100km/h \times 1000m/km \times 1h/3600s)^2}{2 \times 6.352m/s^2} = 60.74 \text{ m}$

Total Braking Time, $t_s = \frac{V_o}{D_x} = \frac{27.78m/s}{6.352m/s^2} = 4.373 \text{ s}$

APPENDIX 16

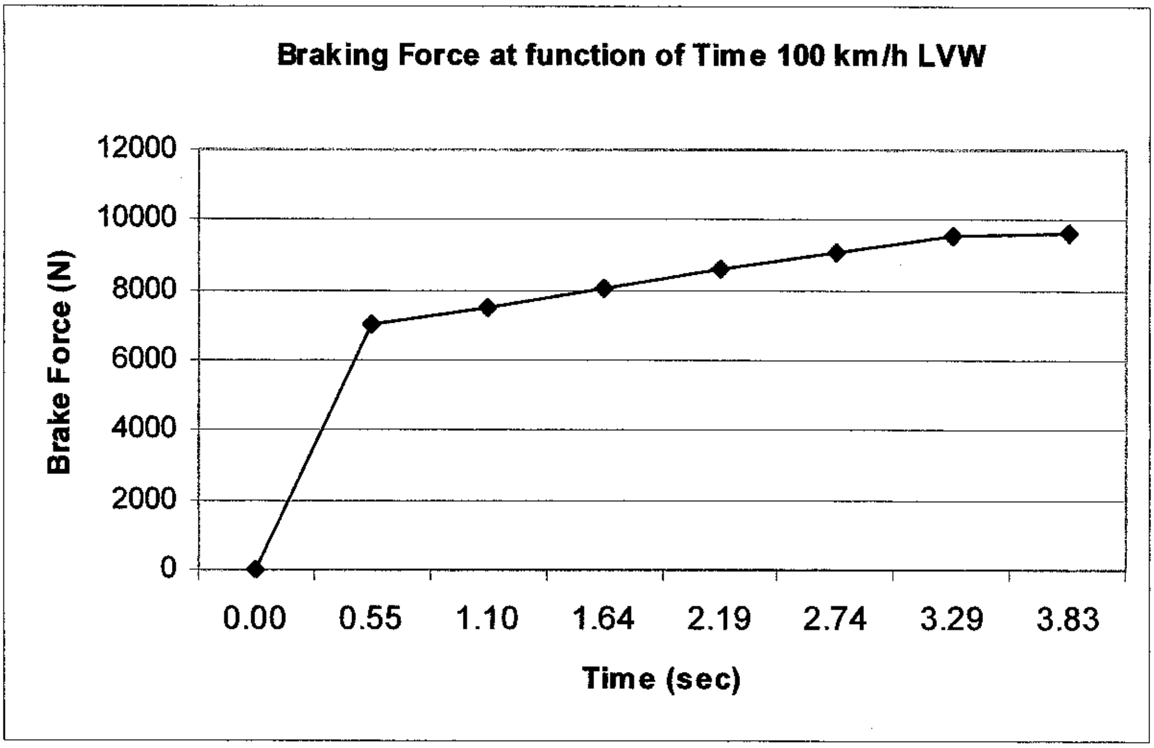
Brake force to the function of time graphs

Time (sec)	Braking force on each cylinder (N)
0.00	0
0.27	6955.1
0.54	7520.7
0.81	8048.3
1.09	8627.4
1.36	9190.2
1.63	9501.4
1.90	10096.7



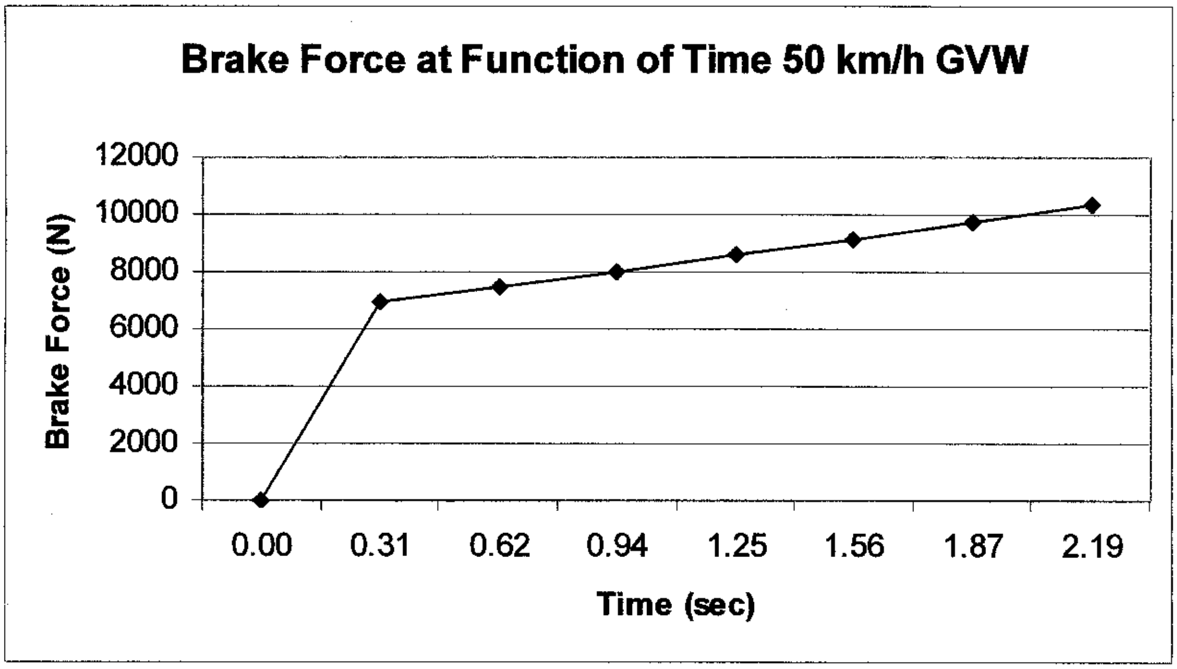
Braking force at function of time for 50 km/h LVW

Time (sec)	Braking force on each cylinder (N)
0.00	0
0.55	7012.0
1.10	7488.2
1.64	8067.3
2.19	8640.9
2.74	9095.5
3.29	9582.9
3.83	9663.8



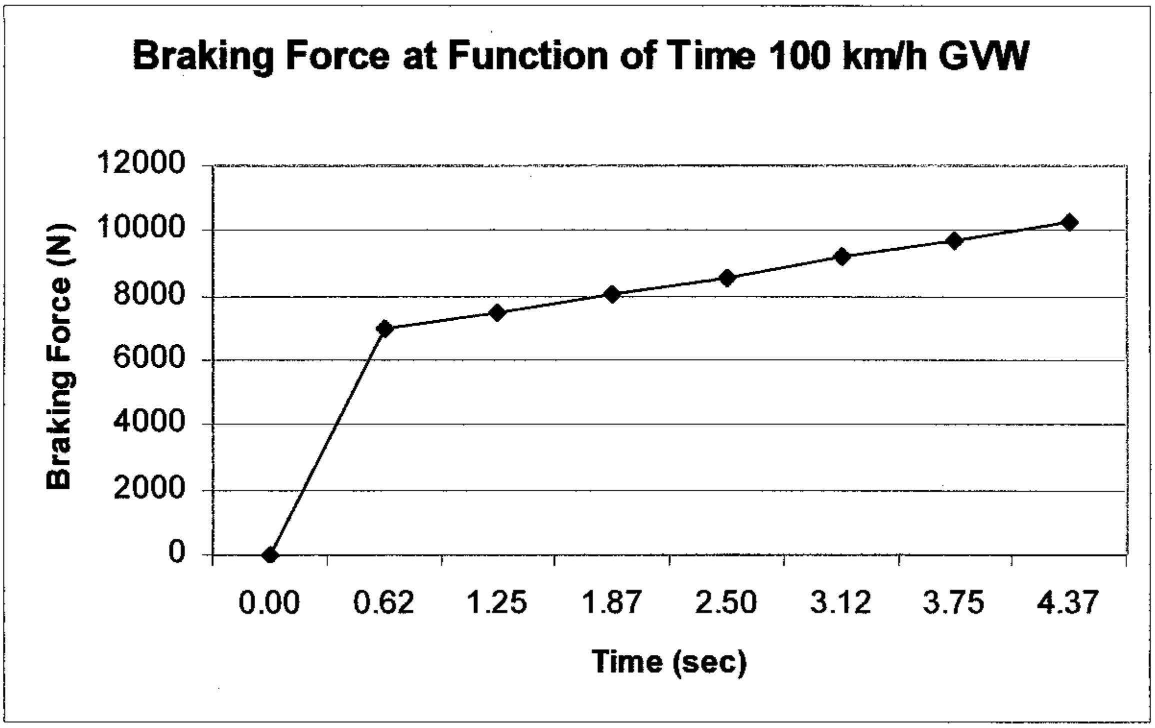
Braking force at function of time for 100 km/h LVW

Time (sec)	Braking force on each cylinder (N)
0.00	0
0.31	6955.1
0.62	7493.6
0.94	8010.4
1.25	8608.5
1.56	9136.1
1.87	9707.1
2.19	10313.2



Braking force at function of time for 50 km/h GVW

Time (sec)	Braking force on each cylinder (N)
0.00	0
0.62	6955.1
1.25	7490.9
1.87	8032.1
2.50	8570.6
3.12	9176.7
3.75	9707.1
4.37	10272.6



Braking force at function of time for 100 km/h GVW