DESIGN AND DEVELOPMENT OF TRANSMISSION FOR TWO-SEATED URBAN CAR

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FACULTY OF MECHANICAL ENGINEERING

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MR.IDRIS MAT SAHAT

Examiner Signature

DESIGN AND DEVELOPMENT OF TRANSMISSION FOR TWO-SEATED URBAN CAR

MAARUF BIN MUHAMAD

Thesis submitted in fulfilment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
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DESEMBER 2010

SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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Date : 6 December 2010

DEDICATION

Dedicated to my parents

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ABSTRACT

This thesis is aim on designing and customizes the transmission from 4 stroke Modenas Kriss 110cc engine to be used in two seated urban car. Design requirement followed by the rules of the Eco-challenge marathon race. The suitable gear is to be identified and selected because of the transmission is decided to use only single gear ratio that suitable and the power produce must be approximate nearly to the power required to avoid losses. It's important because the design of two seated urban car required the participant to design and development two seated urban car that have low fuel consumption. The design will carry out to produce power that approximate enough to the required power and also have a suitable design for two seated urban car that will be design. In this thesis the result to use a single gear ratio is chosen. This is because of the result from the analysis by calculation showed that power produce is near to the power required to move the car. Then finally this condition can make the car running as efficient as possible.

ABSTRAK

Thesis in memfokuskan dalam merekabentuk dan mengubah suai kotak roda gigi daripada engine 4 lejang Modenas Kriss berkapasiti 110cc untuk disesuaikan dengan kereta moden yag mempunyai dua tempat duduk. Kehendak rekaan berpandukan kepada perlumbaan Eco-marathon. Nisbah roda gigi yang sesuai adalah dikehendaki kerana rekaan kotak roda gigi telah diputuskan untuk menggunakan hanya satu nisbah roda gigi yag sesuai dan mampu menghasilkn kuasa yang menghampiri kuasa sesaran pada enjin untuk mengelakkn pembaziran tenaga. Ini adalah penting kerana rekaan kereta moden dua tempat duduk memerlukan peserta untuk merekabentuk dan membangunkan kereta moden dua tempat duduk yang jimat dalam penggunan bahan bakar. Rekaan akan akan menghasilkan kuasa yang menghampiri kuasa sesaran enjin dan juga mempunyai rekabentuk yang sesuai dan padan dengan kereta moden dua tempat duduk yang akan dibangunkan. Di dalam thesis ini penggunaan satu nisbah roda gigi telah dipilih. Ini kerana kajian daripada pengiraan telah mendapati kuasa yang dihasilkan adalah menghampiri dengan kuasa yang diperlukan untuk menggerakkan kereta. Dan akhir sekali keadaan ini dapat membolehkan kereta menggunakan kuasa dengan berkesan.

TABLE OF CONTENTS

LIST OF TABLE

LIST OF FIGURES

CHAPTER 1

INTRODUCTION

1.1 Introduction

This project is related to Eco-marathon races that need participant to design and fabricate two seated urban car. This race is the first UMP participation and the first car (two seated urban car) for UMP. The purpose of the two seated urban car design is to train student to show their creativity and also apply knowledge that acquired in the class.

The criteria of two-seated urban car is it must look like a real car that have front and tail lamp, body shape, side mirror and most important part is the car consume low fuel. The car also should be able to travel long distance and have basic ability like the real car on the road.

For this project, one of the methods to achieve low fuel consumption is to redesign the transmission. Transmission that going to be redesign must be able to increase the efficiency of the engine power output.

1.2 Problem statement

The transmission to be used in this car is from the 110cc MODENAS engine that has four ratio of gear. This gear ratio is changed by using clutch mechanism (centrifugal and wet multiple disc). From common theory, multiple gear ratios will increase the load of the engine because of the weight from the transmission. Engine producing power to move the car by transmitting it to the transmission before goes to tire. This load will occur from the increasing of weight and also the using of clutch mechanism that can create the heat loss from the engine. This loss can decrease the efficiencies of the engine.

This point has been used by lot of inventors to improve the fuel consumption by developing the transmission system. The more efficient transmission system, the more it can decrease fuel consumption by transfer the torque higher to the rotating device with high efficiency. Result can see from many type of transmission have been developed today such as manual transmission, automatic transmission, semi-automatic transmission, CVT transmission etc. This all development and improvement is about to decreased fuel consumption by increasing the efficiency of transmission.

1.3 Objectives

The purpose of this project is to determine the correct gear ratio and also to minimize the loss that occur from the transmission that can affect the efficiency of the engine that will be used by two seated urban car that wanted it less in fuel consumption. Beside that this project purpose is also to obtain the data that can be used for further development of transmission for two seated urban car that has been used in Shell Eco-challenge races by UMP team in future.

1.4 Scopes of study

In order to achieve the stated objective, the following scopes have been identified:

- The transmission that will be used in this project is from MODENAS 110cc engine.
- The maximum velocity of the car is estimated from 40km/h to 50km/h.
- The other scope is based on Shell Eco-challenge races rules.

1.5 Flow chart

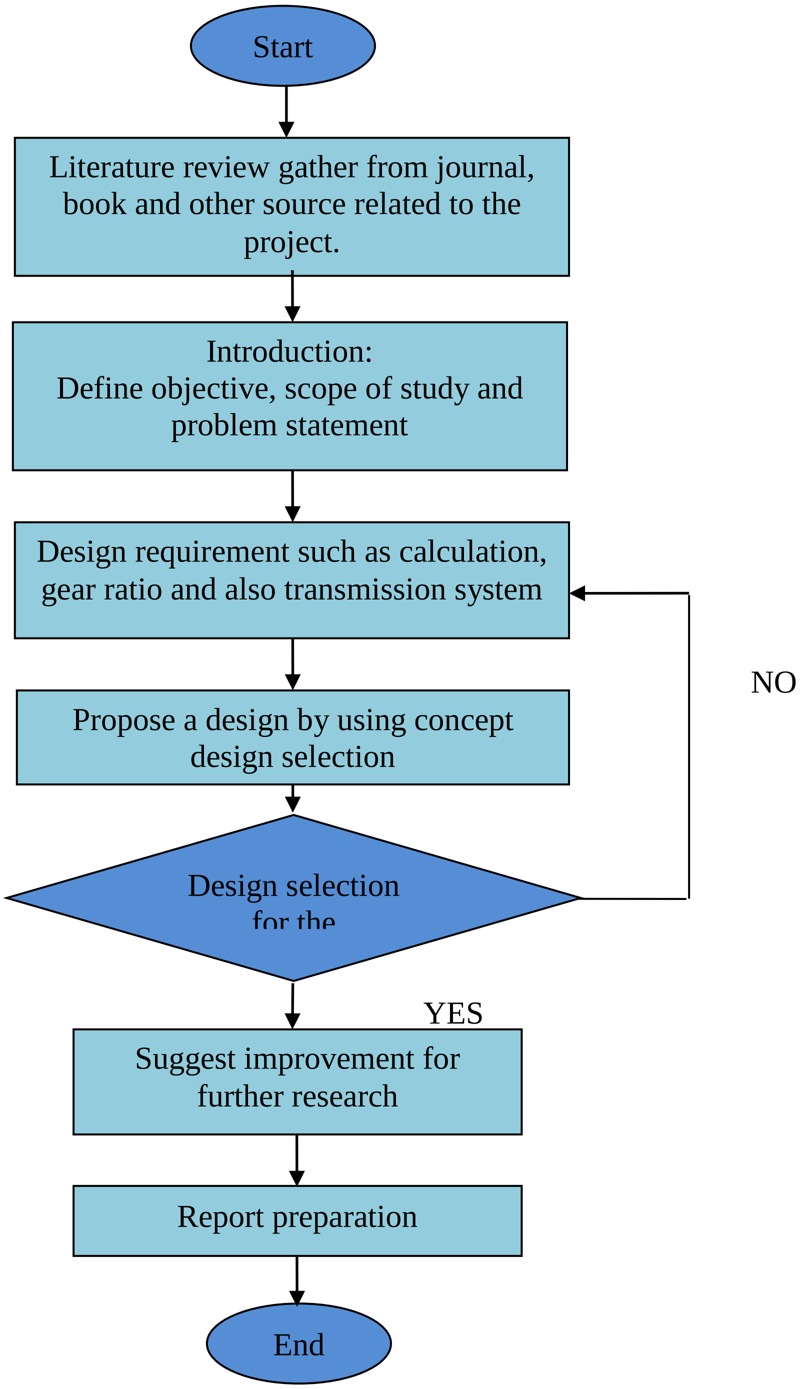


Figure 1.1: Flow chart for the project

From the flow chart project started with proposal of title for final year project from supervisor and also making a schedule for the meeting. After that, make the literature review by gather information about design and development transmission for two seated urban car. Title is related to the Eco-challenge marathon races that need participant to produce two seated urban car that need to be low fuel consumption. The rules and specification of the car must be following from Eco-challenge marathon races.

Then make an introduction for the project that included the objective, scope and problem statement. To choose the better gear ratio and produce efficient, torque calculation must be from the related formula. Then it followed by finding and defining design requirement for transmission by calculation to choose the gear ratio that suitable for the transmission.

Suitable gear ratio is important to the transmission for efficiency. Design is proposed by concept design selection for select the transmission system design and also basic concept before goes for the next step. Then make selection from design concept selection for transmission design. At this state, the transmission design must be choose correctly and made modification to make the transmission is suitable for the system. Then make a detail drawing for the transmission design for finalize.

After that make conclusion and suggest improvement for the transmission design. Finally make a report for the final year project to complete.

CHAPTER 2

LITERATURE REVIEW

2.1 Transmission

The purpose of the transmission or transaxle is to provide neutral, forward gear speeds or ranges and reverse. They must be able to provide a gear ratio that is low enough, when multiplied with the final drive ratio, to increase the engine's torque sufficiently to accelerate the vehicle at the desired rate. The highest gear ratio should allow the vehicle to cruise at an engine speed that is low enough to save fuel and decrease noise level. There also need to intermediate ratio spaced so that the engine will not overrev before a shift or lug after shift. Reverse might be roughly the same ratio as first since vehicle will be starting from a stop in both cases (Birch T. and Rockwood C. (2007)).

A transmission is including a first transmission mechanism and a second transmission mechanism. When a clutch is in an engaged state, the first transmission mechanism transmits a driving force from the engine to the axle shaft. When a clutch is in disengaged state, the second transmission mechanism transmits a driving force from the engine to the axle shaft.

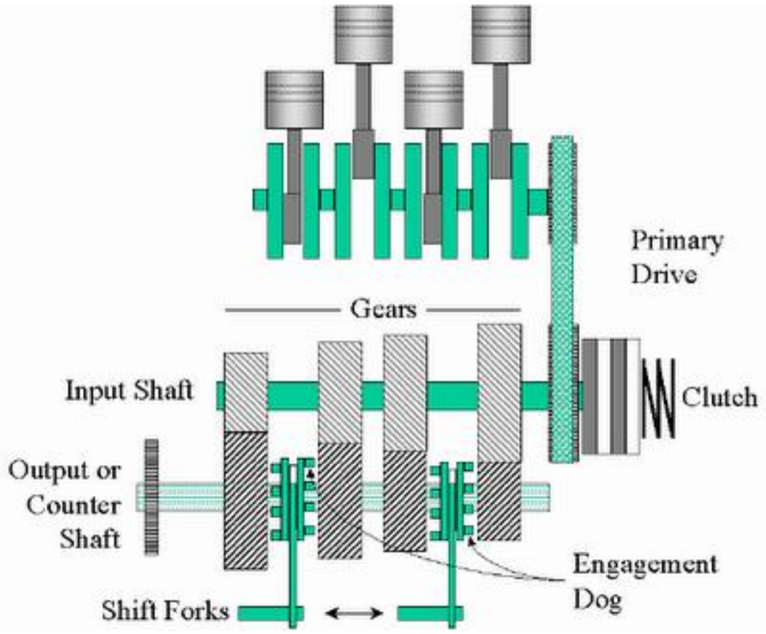


Figure 2.1: Transmission system for motorcycle.

Source: Birch T. and Rockwood C. (2007)

For any given vehicle, transmission gear ratio is selected to satisfy performance requirements of gradability, fuel economy, acceleration, and ease of operation. Fuel economy considerations are essential to the selection of gear ratios. Basically too high a gear ratio causes the engine to run too fast and thus, failing to operate at optimum fuel efficiency. Otherwise, an extremely low numerical gear ratio will affect vehicle performance, such as acceleration. Hence, acceleration is also important concern in gear ratio selection. To maximize acceleration, gear ratios should be selected such that upshifts occur only when the next higher gear will provide more torque to the drive wheels (Razzacki S.T., Troy, and MI (US) (2004)).

Providing maximum torque to the drive wheels in each gear requires consideration of the ratio steps. Wide ratio steps in the lower gears provide for more fuel efficiency due to less frequent shifting (Razzacki S.T., Troy, and MI (US) (2004)). However, shifting through wide ratio steps will requires skilful technique otherwise it can cause a loss in fuel efficiency.

A transmission consist a first transmission mechanism and a second transmission mechanism. When a clutch is in an engaged state, the first transmission mechanism transmits a driving force from the engine to the excel shaft. When a clutch is in disengage state, the second transmission mechanism transmits a driving force from the engine to the axle shaft (Yasui Y. and Saitama (JP) (2006)).

Usually, a transmission will have multiple gear ratios or simply gears, with the ability to switch between them as speed varies. This switching may be done manually by the operator, or automatically. Directional forward and reverse control may also be provided. Single ratio transmissions also exist, which simply change the speed and torque and sometimes direction of motor output. In motor vehicle applications, the transmission will generally be connected to the crankshaft of the engine. The output of the transmission is transmitted via driveshaft to one or more differentials, which in turn drive the wheels. While a differential may also provide gear reduction, its primary purpose is to change the direction of rotation.

There are two types of gear sets used in transmission sliding gears that are in constant mesh. Both types are used in modern transmissions. The forward gear ranges are all in constant mesh, with reverse typically being a sliding gear.

2.2 The Necessity for a Transmission

Power from a petrol or diesel reciprocating engine transfers its power in the form of torque and angular speed to the wheels of the vehicle to move it. The transmission is used to enable the engine’s turning effect and its rotational speed output to be adjusted by choosing a range of under and overdrive gear ratios so that the vehicle responds to the driver’s requirements within the limits of the various road conditions. An insight of the forces opposing vehicle motion and engine performance characteristics which provide the background to the need for a wide range of transmission designs used for different vehicle application will now be considered.

2.3 Resistance to Vehicle Motion

To keep the vehicle moving, the engine has to develop sufficient power to overcome the road resistance power, and to move vehicle from a standstill or to accelerate a reverse of power in addition to that absorbed by the road resistance must be available when required.

Road resistance is expressed as tractive resistance (kN). The propelling thrust at the tyre to road interface needed to overcome this resistance is known as tractive effect (kN) (Fig.2.2). For matching engine power output capacity to the opposing road resistance it is sometimes more easy to express the opposing resistance to motion in terms of road resistance power.

The road resistance opposing the motion of the vehicle is made up of three components as follows:

- Rolling resistance
- Air resistance
- Gradient resistance

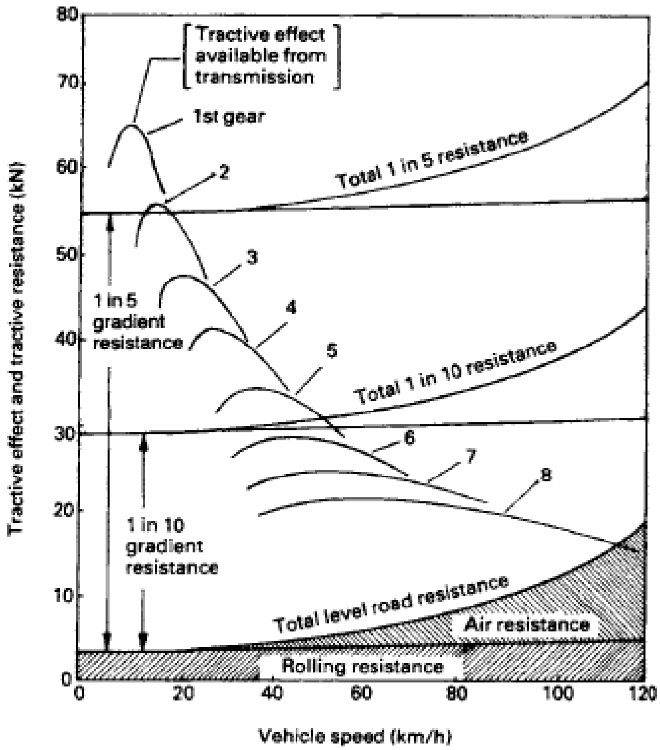


Figure 2.2: Vehicle tractive resistance and effort performance chart.

Source: Heisler H. (2004).

2.4 Ratio Span

Another major consideration when selecting gear ratios is deciding the steepest gradient the vehicle is expected to climb and the maximum level road speed the vehicle is expected to reach in top gear with small surplus of about 0.2% gradibility.

The two extreme operating conditions just described set the highest and the lowest gear ratios. To fix this conditions, the ratio of road speed in highest gear to road speed in lowest gear at a given engine speed should be known (Heisler H. (2004)). This quantity is referred to as the ratio span.

$$\text{Ratio span} = \frac{\text{Road speed in highest gear}}{\text{Road speed in lowest gear}} \qquad (2.1)$$

2.5 Engine Torque Rise and Speed Operating

Commercial vehicle engines used to pull large loads are normally is designed to have a positive torque rise curve, that is from maximum speed to peak torque with reducing engine speed the available torque increases (Fig 2.3). The amount of engine torque is normally expressed as a percentage of the peak torque from maximum speed back to peak torque.

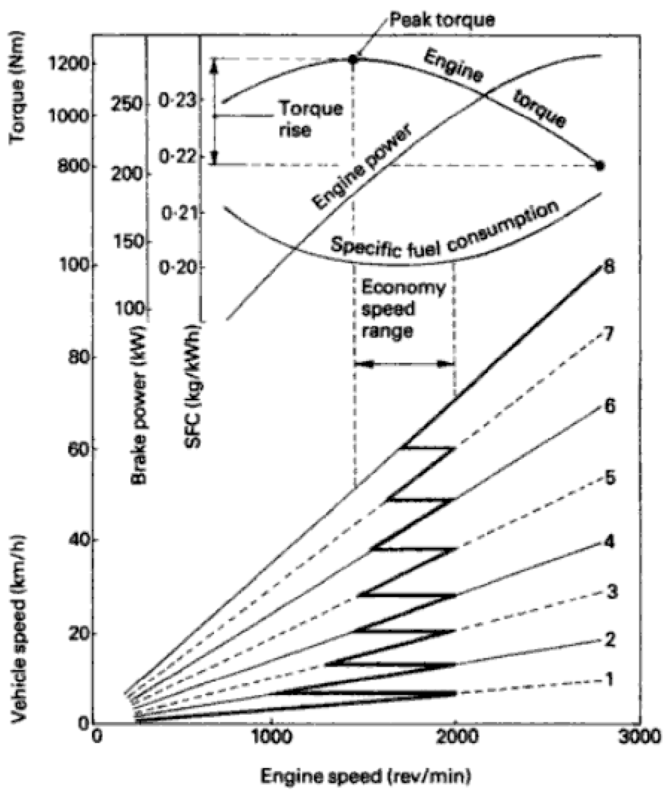


Figure 2.3: Engine performance and gear split chart for an eight speed gearbox.

Source: Heisler H. (2004).

The torque rise can be shaped depending upon engine design and taking into account such features as naturally aspirated, resonant induction tuned, turbocharged, turbocharged with intercooling and so forth.

A large torque rise characteristic rises the engine's operating ability to overcome increased load if the engine's is pulled down caused by changes in the road conditions. If the torque is rise is small it cannot help as a buffer to supplement the high torque demands and the engine speed will rapidly fade (Heisler H. (2004)). Frequent gear changes therefore become necessary compared to engines operating with high torque rise characteristics. Once the engine speed falls below peak torque, the torque rise becomes negative and the pulling ability of the engine drops off very quickly.

Vehicle driving technique should be such the engines are continuously driving between the speed range of peak torque and governed speed. the driver can either choose to operate the engine's speed in range varying just below the maximum rate power to achieve maximum performance and journey speed or, to improve fuel economy, wear and noise, within a speed range of between 200 to 400 rev/min on the positive torque rise side of the engine torque curve that is in a narrow speed band just beyond peak torque (Heisler H. (2004)). Fig 2.3 showed that the economy speed range operates with the specific fuel consumption at its minimum and that the engine speed band is in the most effective pulling zone.

2.6 Gear Ratio

In all gear speeds but one, the power flow from the main drive gear (input) to the cluster gear and then from the cluster gear to the mainshaft (output). The power passes through two gear sets. The exception is a 1:1 ratio, where the power flows directly from the main drive gear to the mainshaft. All the forward gears are normal in constant mesh so they always rotate at their design speed relative to the engine speed. The gears of the cluster gear rotate as an assembly. The output gears usually are mounted to the main shaft so they float or rotate freely. These gears are called speed gears, they completed the ratio for each gear speed when they become coupled to the mainshaft. The mainshaft includes synchronizer assemblies for each pair gear speed and can lock the individual speed gears to the mainshaft. This is done for each shift (Birch T. and Rockwood C. (2007)).

When choosing the lowest and higher gear ratio, the most important factor to consider is not just the available engine power but also the weight of the vehicle and any load it is expected to propel. Another major consideration when selecting gear ratios is deciding upon the steepest gradient the vehicle is expected to climb and the maximum level road speed the vehicle is expected to reach in top gear with a small surplus of about 0.2% grade ability (Heisler H. (2004)).

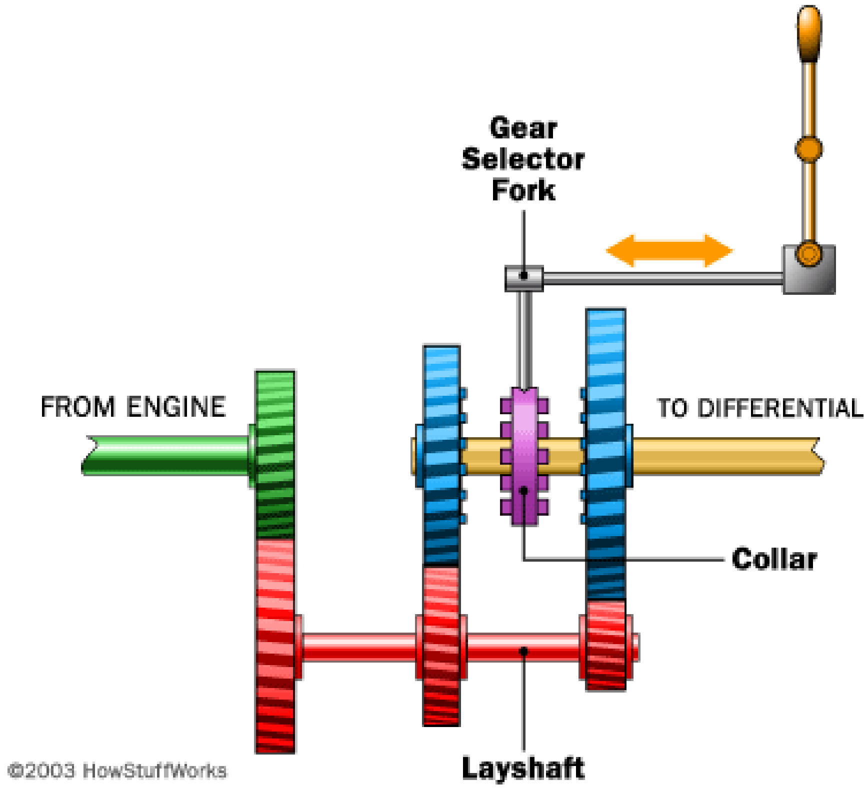


Figure 2.4: Simple gear system in transmission.

Source: Brain M. (2007).

The method includes selecting gear ratios for a specific application includes calculating a low gear ratio, and a high gear ratio based upon vehicle parameters and performance requirements. The total ratio spread is determined by dividing the low gear ratio by the high gear ratio. Using the total ratio spread, a geometric sequence is created with a plurality of terms, such that each of the terms respectively represents the ratio steps between the gears. Lastly, each gear ratio is divided by its respective ratio steps plus one to fine the gear ratio for the next gear. This method provides and objective method for selecting gear ratios, such that the steps between each of the ratios are uniformly progressive (Razzacki S.T., Troy, and MI (US) (2004)).

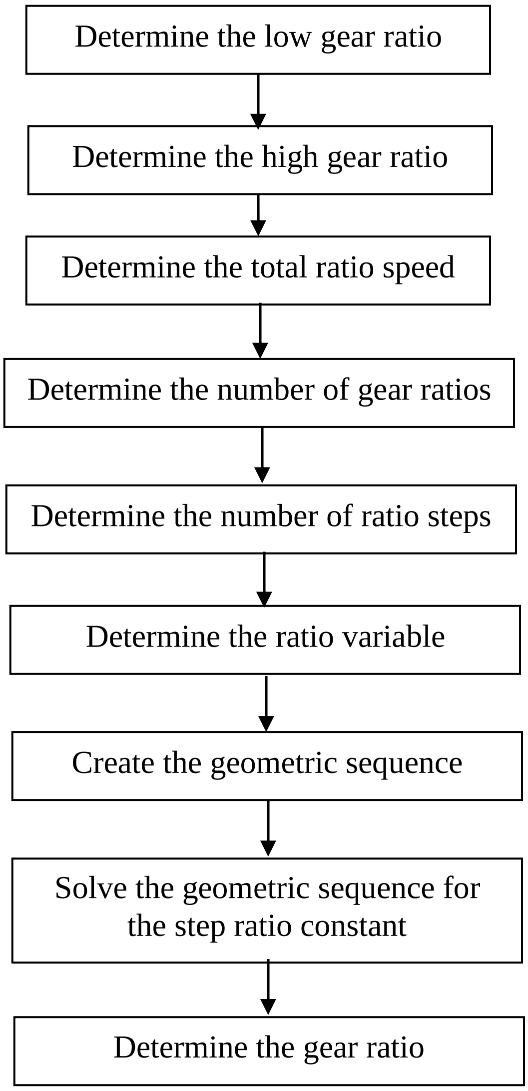


Figure 2.5: Method for selecting gear ratios.

Source: Razzacki S.T., Troy, and MI (US) (2004).

2.7 Gear Synchronization and Engagement

The gearbox basically consists of an input shaft driven by the engine crankshaft by way of the clutch and an output shaft coupled indirectly either through the propeller shaft or intermediate gears to the final drive (Razzacki S.T., Troy, and MI (US) (2004)). Between these two shafts are pairs of gear wheels of different size meshed together.

If the gearbox is in neutral, only one of these pairs of gears is actually attached to one of these shafts while the other is free to rotate on the second shaft at some speed determined by the existing speeds of the input and output drive shafts (Razzacki S.T., Troy, and MI (US) (2004)).

To engage any gear ratio the input shaft has to be disengaged from the engine crankshaft via the clutch to release the input shaft drive. It is then only the angular momentum of the input shaft, clutch drive plate and gear wheels which keeps them revolving. The technique of good gear changing is to be able to judge the speeds at which the dog teeth of both the gear wheel selected and output shaft are rotating at a uniform speed, at which point in time the dog clutch sleeve is pushed over so that both sets of teeth engage and mesh gently without grating (Razzacki S.T., Troy, and MI (US) (2004)).

Because it is difficult to know exactly the time to make the gear change a device known as the synchromesh is utilized. Its function is to apply a friction clutch braking action between the engaging gear and drive hub of the output shaft so that their speeds will be unified before permitting the dog teeth of both members to contact (Razzacki S.T., Troy, and MI (US) (2004)).

Synchromesh devices use a multiplate clutch or a conical clutch to equalise the input and output rotating members of the gearbox when the process of gear changing is taking place. Except for special applications, such as in some splitter and range change auxiliary gearboxes, the conical clutch method of synchronization is generally employed (Razzacki S.T., Troy, and MI (US) (2004)).

With the conical clutch method of producing silent gear change, the male and female cone members are brought together to produce a synchronizing frictional torque of sufficient magnitude so that one or both of the input and output members' rotational speed or speeds adjust automatically until they revolve as one. Once this speed uniformity has been achieved, the end thrust applied to the dog clutch sleeve is permitted to nudge the chamfered dog teeth of both members into alignment, thereby enabling the two sets of teeth to slide quietly into engagement (Razzacki S.T., Troy, and MI (US) (2004)).

2.8 Motorcycle Transmission

A motorcycle gearbox is generally controlled by a gear-shift pedal which enables the gears to be selected only in a rising or descending sequence (Wenger U., Jenni H.R., and Wuthrich A. (1996)). The motorcyclist is not being able freely to select a given gear directly from the gear in which, at a given moment, the gearbox is placed.

These particular characteristics of motorcycle gearboxes make them not suited to equip motor vehicle. Since their lightness and simplicity of design are properties which would be required to be able to exploit in motor vehicles (Wenger U., Jenni H.R., and Wuthrich A. (1996)). Especially in those which are of a small size and intended for town use. The principle aim to the present invention thus consists of providing a gearbox designed according to the principles of a motorcycle gearbox, but especially adapted for use in a motor vehicle, in particular in a whose driving motor has its maximal torque at relatively high rate (Wenger U., Jenni H.R., and Wuthrich A. (1996)).

A motorcycle engine can create an enormous amount of power, which must be delivered to the wheels of the vehicle in a controllable way. The power delivered to the rear wheel through a series of structures that include the gear set, the clutch and the drive system. In order to make the gearbox as light as possible it is also construct it without a synchronizing device (Wenger U., Jenni H.R., and Wuthrich A. (1996)).

2.8.1 Gears

A gear set enable a rider to move from a complete stop to a cruising speed. Transmissions on motorcycles usually have four to six gears, although small bikes may have as few as two. The gears are engaged by shifting a lever, which moves shifting forks inside the transmission (Wenger U., Jenni H.R., and Wuthrich A. (1996)).

2.8.2 Clutch

The job of a clutch is to engage and disengage power from the engine crankshaft to the transmission. Without the clutch, the only way to stop the wheels from turning would be to turn off the engine an impractical solution in any kind of motorized vehicle. The clutch is a series of spring-loaded plates that, when pressed together connect the transmission to the crankshaft. When a rider wants to shift gears, he uses the clutch to disconnect the transmission from the crankshaft. Once the new gear is selected, he uses the clutch to reestablish the connection (Brain M. (2007)).

2.8.3 Drive System

There are three basic ways to transmit engine power to the rear wheel of a motorcycle: chain, belt or shaft. Chain final-drive systems are by far the most common. In this system, a sprocket mounted to the output shaft is connected to a sprocket attached to the rear wheel of the motorcycle by a metal chain. When the transmission turns the smaller front sprocket, power is transmitted along the chain to the larger rear sprocket, which then turns the rear wheel. This type of system must be lubricated and adjusted, and the chain stretches and the sprockets wear, requiring periodic replacements.

Belt drives are an alternative to chain drives. Early motorcycles often used leather belts, which could be tensioned to give traction using a spring-loaded pulley and hand lever. Leather belts often slipped, especially in wet weather, so they were abandoned for other materials and designs. By the 1980s, advances in materials made belt final-drive systems viable again. Today's belts are made of cogged rubber and operate much the same way as metal chains (Brain M. (2007)). Unlike metal chains, they don't require lubrication or cleaning solvents.

Shaft final-drives are sometimes used. This system transmits power to the rear wheel via a drive shaft. Shaft drives are popular because they are convenient and don't require as much maintenance as chain-based systems. However, shaft drives are heavier and sometimes cause unwanted motion, called shaft jacking, in the rear of the motorcycle. The other components that make a motorcycle a motorcycle are part of the chassis (Wenger U., Jenni H.R., and Wuthrich A. (1996)).

2.9 Shell Eco-marathon 2010

The Shell Eco-marathon challenges high school and college student teams from around the world to design, build and test energy efficient vehicles. With annual events in the Americas, Europe and Asia, the winner is the team that goes the furthest distance using the least amount of energy (Shell (2010)). This event also affords an outstanding engagement opportunity for current and future leaders who are passionate about finding sustainable solutions to the world's energy challenge.

Shell organizes energy-economy competitions on a real motor racing circuit in both the Americas and Europe. Known as the Shell Eco-marathon, this competition is governed by the Official Rules presented herein (Shell (2010)).

Participants can design vehicles for the 'Prototype' or the 'Urban Concept' Group. The Prototype Group encourages a maximum of technical creativity, imposing only minimal restriction on critical automotive design aspect. The Urban Concept Group is intended to be closer to road going vehicles in appearance and technology, addressing current transportation aspects (Shell (2010)).

2.9.1 Urban Concept

The Urban Concept category was introduced in 2003 with the aim of broadening the scope of the Shell Eco-marathon challenge. It challenged students to develop fuel-efficient that could be applied to the cars for today.

Unlike Prototype vehicles, all Urban Concept cars must meet a series of roadworthiness criteria found in modern passenger vehicles (such as having four wheels, a steering wheel, head and tail lights, a brake pedal, doors, etc.) (Shell (2010)).Participant whom entries into the Urban Concept category are not intended for commercialization and may not be driven on public roads.

2.9.2 Energy

The rules for this urban concept car allowed using hybrid technology .Hybrid technology means the combined use of internal combustion engine and electric motors in vehicles supported by an electric power accumulation system. Solar panels are not allowed for hybrid vehicles however regenerative energy braking systems can be used (Shell (2010)).

The rules allowed vehicles using one of the following fuel or energy types:

- Shell Unleaded 95 (EU) / Shell Plus 89 (US) Petrol/Gasoline
- Shell Diesel
- Liquefied Petroleum Gas (LPG)
- Shell Gas To Liquid (100% GTL)
- Fatty Acid Methyl Ester (100% FAME)
- Ethanol E100 (100% Ethanol)
- Hydrogen
- Solar

2.9.3 Vehicle Dimensions

- The total vehicle height must be between 100 cm and 130 cm.
- The total vehicle width must be between 120 cm and 130 cm.
- The total vehicle length must be between 220 cm and 350 cm.
- The track width must be at least 100 cm for the front axle and 80 cm for the rear axle.
- The wheelbase must be at least 120 cm.
- The driver's compartment must have a minimum height of 88 cm and a minimum width of 70 cm at the driver's shoulders.
- The ground clearance must be at least 10 cm.
- The maximum vehicle weight (excluding the driver) must be 160 kg.

Figure 2.6: Urban Car design for Eco-marathon.

Source: Shell (2010).

2.10 Summary

From this chapter, summaries of the efficiency of the engine can be made by redesign the transmission in other word by choosing the correct gear ratio to be used on urban car. The result is to overcome the power output is as near as possible to the power available produce by the transmission. The design of transmission can be made by considered all resistance and specification for the car that stated on Shell Eco-marathon rules. The transmission that to be redesign will be used current transmission from Modenas Kriss 110cc.

CHAPTER 3

METHODOLOGY

3.1 Introduction

This chapter will describe about the development of transmission for two seated urban car. The development process is conducted by using calculation analysis to define the gear ratio that suitable in any condition and road structure. Transmission used to develop is from Modenas Kriss 110cc motorcycle engine.

The model to be developed in this chapter is from the current research. This model is limited to the engine model and followed by rules of Eco-marathon races.

3.2 Model Specification

There is specification that followed from Eco-marathon rules and the scope of study that will be used in this project. The specification of the engine will affect the transmission ratios that will be used.

The specification of the vehicle is also important to the calculation analysis such as the weight, dimension and also the vehicle condition. The transmission for this project is from the multiple current gear ratios from Modenas Kriss 110 that will be redesign for the two seated urban car. Transmission will be design either using single or multiple gear ratio is defined by analysis and calculation process which focusing on engine power requirement and power produce from gear ratio.

3.3 Setting Gear Ratio

For matching the engine's performance characteristics to suit the vehicle's operating requirements is provided by choosing a final gear reduction. Then select a range of gear ratio for maximum performance. This range is including of the ability to climb gradients, achievement of good acceleration through the gears and ability to reach some predetermined requirement speed on a level road.

3.4 Design Concept

In this project, transmission from Modenas Kriss 110 engine is designed for two seated urban car. The design starts from the calculation to determine whether the gear ratio is suitable and can reduce the losses that occur from the transmission. A calculation was including the new body and requirement for two seated urban car that different from the old body of Modenas Kriss 110.

Power produced from various gear ratios is defined and compared with the power required to accelerate the two seated urban car. To reduce the losses, the power that produces from gear ratio should be approximately same as the power required. Result can be achieved by using various calculation methods that need to be compared with each other.

The design concept transmission used in this project is same as the current design for transmission that used for Modenas Kriss 110. The modification provided after the calculation analysis to search the suitable gear ratio that can be applied to the car to move on flat and gradient road condition.

Table 3.1: Gear ratio for Modenas Kriss 110

Source: Modenas, (1995-2008).

Gear	**Ratio**
1^{st}	2.83
2^{nd}	1.71
3^{rd}	1.24
4^{th}	0.96

3.5 Vehicle Resistance and Road Disturbance

To determine the maximum vehicle speed, the engine brake power curve is superimposed onto the power requirement curve which can be plotted from the sum of both the rolling (R_r) and air (R_a) resistance covering the entire vehicle's speed range.

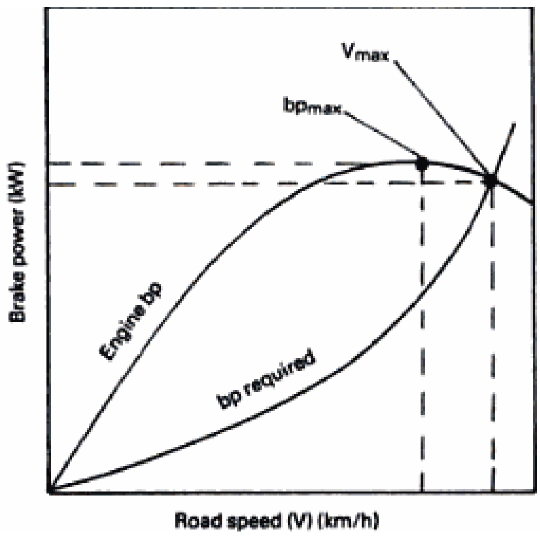


Figure 3.1: Relationship of power developed and road power required over the vehicle's speed range.

Source: Heisler H. (2004).

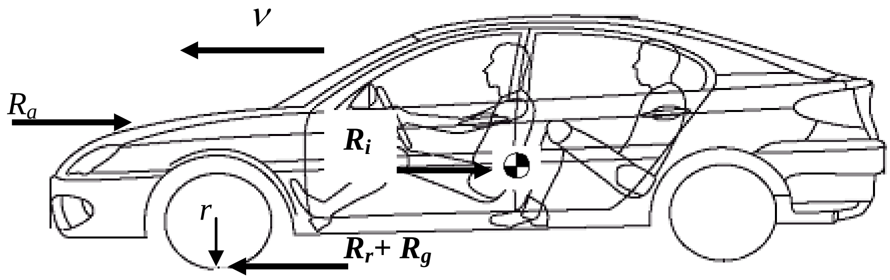


Figure 3.2: Force acting on a vehicle.

Source: Ariyono S. (2008).

By applying Newton's second law in the longitudinal direction, the friction force (R_w) is described by reference (Gillespie T.D. (1992)):

$$R_w = R_a + R_r + R_g + R_i \tag{3.1}$$

where,

R_a : the aerodynamic drag resistance

R_r : rolling resistance

R_g : gravitational resistance

R_i : acceleration resistance,

The external torque could be modelled as:

$$T_w = R_w \times r \tag{3.2}$$

where,

r : the radius of the tyre (m)

3.5.1 Aerodynamics Resistance

Power is needed to counteract the tractive resistance created by the vehicle moving through the air. This is caused by air being pushed aside and the formation of turbulence over the contour of the vehicle's body. Thus at very low vehicle speeds aerodynamics resistance is insignificant, but it becomes predominant in the upper speed range.

The aerodynamic forces of a vehicle arise from two source-forms, drag and viscous friction. For aerodynamic drag coefficient, C_d it assumed to be 0.39. Since airflow over a vehicle is so complex, it is necessary to use semi-empirical model to represent the effect (Gillespie T.D. (1992)). Thus the aerodynamic drag, R_a, is characterised by equation:

$$R_a = \tfrac{1}{2} \rho C_d \, A v^2 \tag{3.3}$$

where,

ρ : the air density (kg/m^3)

C_d : the aerodynamic drag coefficient

A : the maximum vehicle cross-section area (m^2)

v : vehicle speed (m/sec).

3.5.2 Rolling Resistance

A moving wheeled vehicle will gradually slow down due to rolling resistance including that of the bearing. The coefficient of rolling resistance is generally much smaller for tires or balls than the coefficient of sliding friction.

It has been found that the flattening distortion of the tyre casing at the road surface interface consumes more energy as the wheel speed increases and therefore the rolling resistance will also rise slightly.

Aerodynamic resistance becomes equal to the rolling resistance only at the speed of 80 – 100 km/h (Gillespie T.D. (1992)). There are several factors affecting rolling resistance such as tyre temperature, tire inflation pressure, vehicle velocity, and type of tyre. The coefficient is proportional to the speed but the effect is small at moderate and low speeds and is often assumed to be constant.

$$R_r = (c_a + c_b \, v) mg \tag{3.4}$$

where,

c_a : constant (0.013) (Gillespie T.D. (1992)).

c_b : constant (0.00004) [11]

m : mass of vehicle (kg)

g : gravitational force (m/sec^2).

3.5.3 Gravitational Resistance

All objects on earth experience a force of gravity which is directed downward towards the centre of the earth. The force of gravity on earth is always equal to the weight of the object.

The gravitational resistance opposing motion, and therefore the tractive effect or power needed to drive the vehicle forward, is directly proportional to the laden weight of the vehicle and the magnitude of gradient.

When a vehicle climbs an inclined road, force due to gravity needs to be considered. This extra force due to inclined road has to be overcome by the engine. When the road slope is declining the engine should be able to provide engine break by increasing its transmission ratio.

Gravitational force, R_g, is approximated by:

$$R_g = mg\sin\theta \tag{3.5}$$

where,

θ : Road slope (%)

3.5.4 Acceleration Resistance

Mass is assume to be fixed and, so that the acceleration resistance (R_i) is equal to

$$R_i = m\frac{dv}{dt} = m\dot{v} \tag{3.6}$$

Where'

$\dot{v}$: Acceleration in the direction of motion of the vehicle.

Table 3.2: Parameter for two seated urban car concept.

Items	**Value**	**Unit**
Vehicle mass (m), include driver.	160	Kg
Gravity (g)	9.81	m/s^2
Tyre radius (r)	0.2159	m
Frontal area (A)	1.69	m^2
Coefficient of drag (C_d)	0.39	
Rolling resistance (C_a) [11]	0.013	
Coefficient of tire (C_b) [11]	0.00004	
Engine efficiency	80%	
Air density (ρ) (sea level)	1.226	kg/m^3
Final gear ratio (Gfd)	3.0	

3.6 Engine Parameters

Engine to be used is from MODENAS model and the velocity for the vehicle is taking from the rules that required vehicle to running at 40km/h to 50km/h. The engine parameters state as:

- MODENAS Kriss 110cc.
- Power maximum, P_{max} is 6.6kW at 8500rpm.
- Torque maximum, T_{max} is 9.3Nm at 4000rpm.

Engine power has to be higher than the required power. The power required is a function of vehicle speed as follow:

$$P_{req} = \frac{T_w \omega_w}{\eta} = \frac{R_w v}{\eta} \tag{3.7}$$

and

$$v = \frac{2}{60} \frac{N_e \pi r}{G} \tag{3.8}$$

For engine power:

$$Pe = \frac{TeGv}{r} \tag{3.9}$$

where,

G the overall transmission ratio (differential gear time’s transmission ratio)

3.7 Summary

According to this chapter the resistance is considered from the road condition and also the car condition. For the road resistance it will be used the 2.0% of the road slope define from the previous race at Sepang International circuit. Method that will be used to define the suitable gear ratio for the car design is by the calculation that showed in this chapter. This calculation will be used for the next chapter to prove the theory that has been review.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Introduction

This chapter is aim to explain the analysis to come out the result for the project. The analysis will show the result from the calculation data that include engine performance, power required to accelerate the vehicle and also the important of the project is gear ratio that can minimize the loss that occur from the transmission.

Full result of the project will show in this chapter that include figure, graph and also the related table data.

4.2 Engine Performance

The design of manual shift transmission gearboxes depend considerably on performance prediction calculation. This starts only from the maximum power and torque of the engine speed. Performance graph result will showed the engine performance and also prove the engine performance from the calculation and compare to the manufacture and review.

From figure 4.1 showed that the power/torque versus speed characteristic approximately at the peak values. This condition is calculated from the requirement for the vehicle to run about 40km/h to 50km/h.

Result showed in figure 4.1 is nearly same as the manufacture and the review about the engine performance. Result power and torque from calculation crossing each other after plotted on the graph. For more consumption, engine should be operated at speed range from the torque peak value to the crossing point for torque and power.

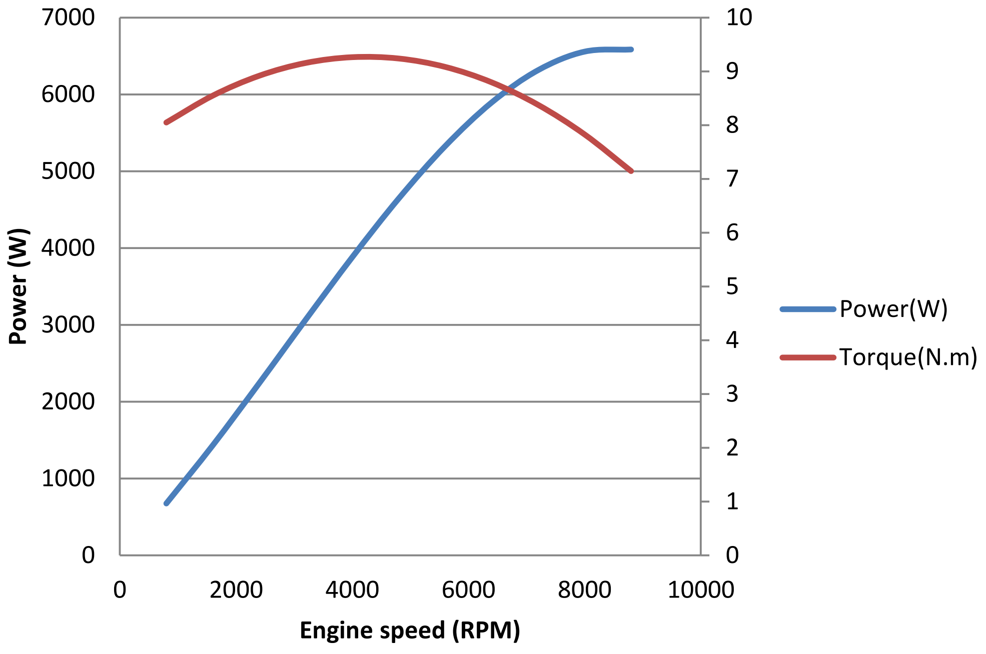


Figure 4.1: Graph engine performance torque and power versus engine speed.

From the table data that calculated is power engine and torque engine nearly to the engine specification that state the engine power maximum is at 6.6kW at 8500rpm and the torque maximum is at 9.3N.m at 4000rpm (Modenas, (1995-2008)).

Table 4.1: Engine performance data table.

RPM	Te(N.m)	Pe(W)
800	8.0469	674.1306
1600	8.5477	1432.182
2400	8.9172	2241.132
3200	9.1553	3067.973
4000	9.262	3879.654
4800	9.2374	4643.227
5600	9.0814	5325.609
6400	8.7941	5893.867
7200	8.3755	6314.977
8000	7.8254	6555.797
8800	7.144	6583.438

4.2.1 Example Calculation of Engine Power

From equation (3.9) engine power is calculate:

$$Pe = \frac{TeGv}{r}$$

Te = 8.0469

G = 2.89

v = 6.2585

r = 0.2159

$$Pe = \frac{(80.469)(2.89)(6.2585)}{0.2159}$$

= 674.1306

4.3 Power Required Result for The Engine

Power required graph is important to define power for the engine to accelerate or move the vehicle. There are two road situations, flat and gradient must be consider for make sure the ratio that will be used is suitable and efficient enough to move the car.

For this project, power required is determined from the vehicle speed that needs to maintain 40km/h to 5km/h, and then road gradient is 2.0% taken from the Sepang International Circuit condition. Figure 4.2 showing the result for flat and gradient road. For flat road condition, at 40km/h to 50km/h the power required is 985.8643W to 1722.4753W and for the gradient road is 1530.864W to 2403.726W.

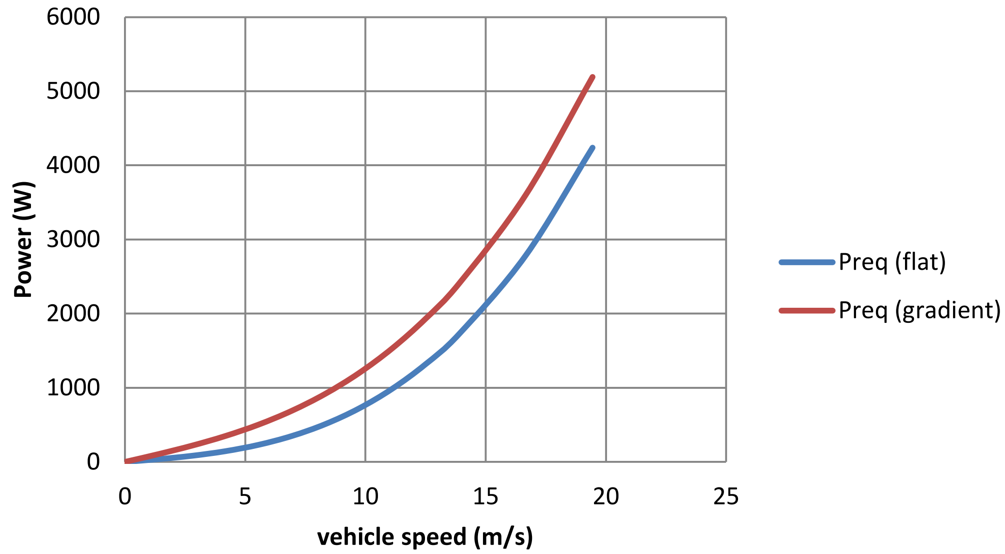


Figure 4.2: Graph power required for flat road and gradient road condition.

Table 4.2: Power required for flat and gradient road condition data.

V (m/s)	Preq flat (W)	Preq gradient (W)
1.3889	36.9298	105.0554
2.7778	82.2812	218.5323
4.1667	144.1725	348.5491
5.5556	230.7227	503.2249
6.9444	350.0403	690.6631
8.3333	510.2615	919.0099
9.7222	719.4963	1196.37
11.1111	985.8643	1530.864
12.5000	1317.4844	1930.609
13.8889	1722.4753	2403.726
16.6667	2785.0354	3602.543
19.4444	4238.4635	5192.202

4.3.1 Sample Calculation for Power Required

From equation (3.7) power required is calculate:

$$Preq = \frac{Rwv}{\eta}$$

Rw = 21.2714

v = 1.3889

η = 0.8

$$Preq = \frac{(21.2714)(1.3889)}{0.8}$$

= 36.9298

4.4 Vehicle Speed and Engine Speed Result

Figure 4.3 showed the result for the maximum engine speed for various gear ratios. In theoretically engine should works at its maximum power to have the best performance. Then engine should works at its maximum torque for more fuel consumption. Graph result on figure 4.3 can be used to determine the condition to control the speed of the vehicle for the best performance. As current transmission design that using multiple gear ratio it must changing the gear after the engine speed reach at the point either it in performance or fuel consumption. From the figure 4.3 gears should be changing after reach at 4000rpm engine speed because. This situation is because of fuel consumption engine should running at maximum torque (figure 4.1).

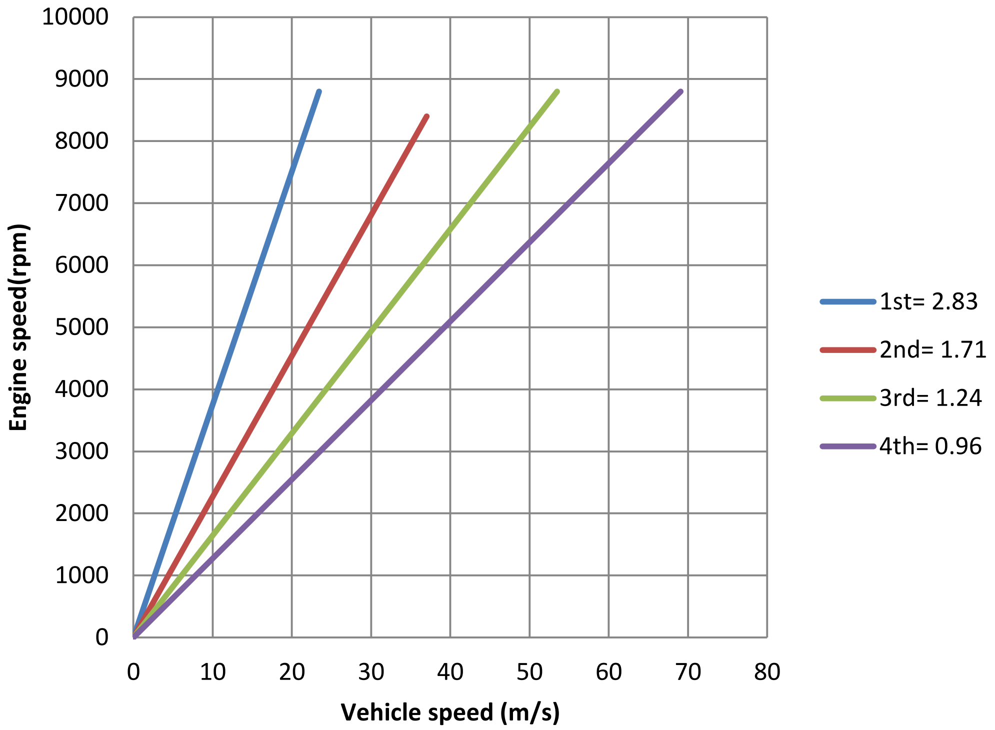


Figure 4.3: Graph engine speed versus vehicle speed.

Table 4.3: Vehicle speed and engine speed for various gear ratios.

	1st gear	**2nd gear**	**3rd gear**	**4th gear**
RPM	**V(m/s)**	**V(m/s)**	**V(m/s)**	**V(m/s)**
400	1.0652	1.7629	2.4311	3.1401
800	2.1304	3.5258	4.8621	6.2803
1200	3.1956	5.2887	7.2932	9.4204
1600	4.2608	7.0515	9.7243	12.5606
2000	5.326	8.8144	12.1554	15.7007
2400	6.3912	10.5773	14.5864	18.8408
2800	7.4564	12.3402	17.0175	21.9809
3200	8.5216	14.1031	19.4486	25.1211
3600	9.5869	15.8659	21.8797	28.2612
4000	10.6521	17.6288	24.3107	31.4014
4400	11.7173	19.3917	26.7418	34.5415
4800	12.7825	21.1546	29.1729	37.6817
5200	13.8477	22.9175	31.6039	40.8218
5600	14.9129	24.6804	34.035	43.9619
6000	15.9781	26.4433	36.4661	47.1021
6400	17.0433	28.2062	38.8972	50.2422
6800	18.1085	29.969	41.3283	53.3823
7200	19.1737	31.7319	43.7593	56.5225
7600	20.2389	33.4948	46.1904	59.6626
8000	21.3041	35.2577	48.6215	62.8028
8400	22.3693	37.0206	51.0526	65.9429
8800	23.4345	38.7835	53.4836	69.083

4.5 Tractive Efforts Result

From figure 4.4 showed the relationship between the tractive efforts at the point of contact with the road surface by the driven wheel and the vehicle speed for each gear ratio. Curve from the graph showed that the tractive effort decrease as the transmission shifted to the higher gear. As for the project need the car to run at speed 40km/h to 50km/h figure 4.4 showed only the 0.96 and 1.24 ratios can achieve the range of the speed needed.

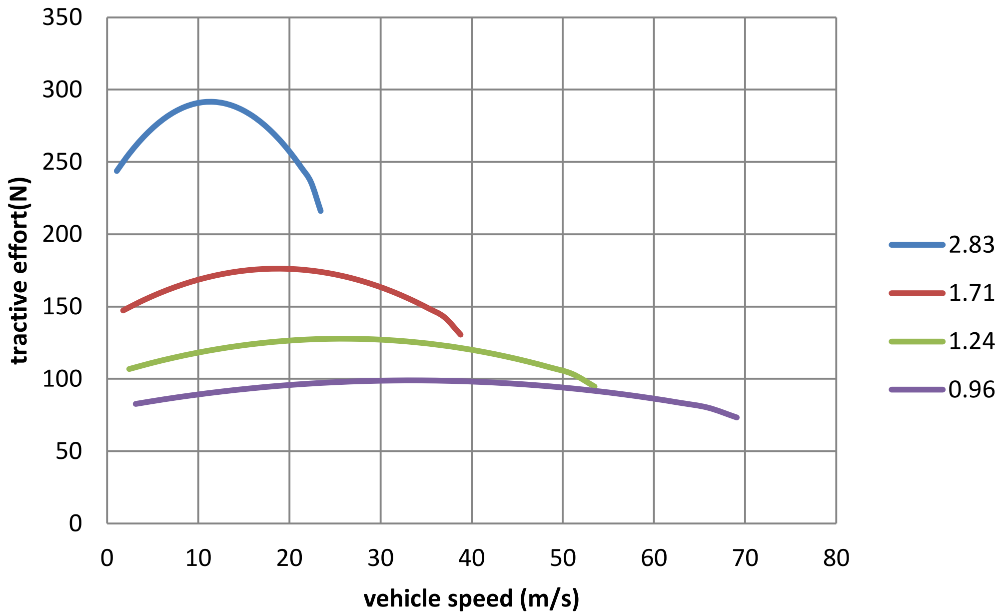


Figure 4.4: Graph tractive effort for gear ratio.

4.5.1 Example Calculation of Tractive Effort

$TFw = \frac{TeG\eta}{r}$

(4.1)

Te = 7.742

G = 8.49

$\eta = 0.8$

r = 0.2159

$TFw = \frac{(7.742)(8.49)(0.8)}{0.2159}$

= 243.7192

Table 4.4: Tractive effort data.

1st gear		2nd gear		3rd gear		4th gear	
V(m/s)	**TFw**	**V(m/s)**	**TFw**	**V(m/s)**	**TFw**	**V(m/s)**	**TFw**
1.065	243.719	1.762	147.265	2.431	106.788	3.140	82.675
2.130	253.147	3.525	152.961	4.862	110.919	6.280	85.873
3.195	261.540	5.288	158.033	7.293	114.597	9.420	88.720
4.260	268.902	7.051	162.481	9.724	117.822	12.560	91.217
5.326	275.228	8.814	166.304	12.155	120.594	15.700	93.363
6.391	280.526	10.577	169.505	14.586	122.916	18.840	95.160
7.456	284.785	12.340	172.079	17.017	124.782	21.980	96.605
8.521	288.016	14.103	174.031	19.448	126.198	25.121	97.701
9.586	290.212	15.865	175.358	21.879	127.160	28.261	98.446
10.651	291.373	17.628	176.059	24.310	127.668	31.401	98.840
11.717	291.502	19.391	176.137	26.741	127.725	34.541	98.884
12.782	290.599	21.154	175.591	29.172	127.329	37.681	98.577
13.847	288.661	22.917	174.421	31.603	126.480	40.821	97.920
14.912	285.691	24.680	172.626	34.035	125.179	43.961	96.913
15.978	281.690	26.443	170.208	36.466	123.426	47.102	95.555
17.043	276.653	28.206	167.165	38.897	121.219	50.242	93.847
18.108	270.585	29.969	163.498	41.328	118.560	53.382	91.788
19.173	263.484	31.731	159.208	43.759	115.449	56.522	89.380
20.238	255.349	33.494	154.292	46.190	111.884	59.662	86.620
21.304	246.179	35.257	148.751	48.621	107.866	62.802	83.509
22.369	235.977	37.020	142.586	51.052	103.396	65.942	80.048
23.434	216.063	38.783	130.554	53.483	94.6709	69.083	73.293

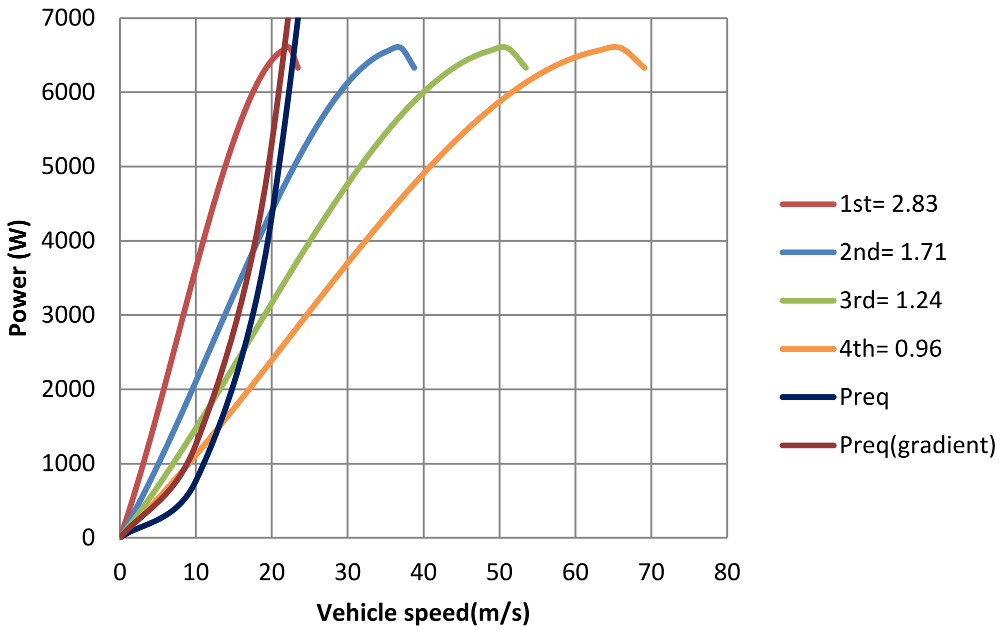
4.6 Transmission Performance Result

For efficient use of power from the engine to the vehicle, the power available curve must be as near as possible to the power required for decrease the losses.

From figure 4.5 showed the result of power required curve and power available curve from every gear ratio. First gear producing powers that far to the power required curve for both flat and gradient road condition and transmission ratio for third and fourth is nearly enough to the both power required. This condition is defined from the specification that need the car to be maintain vehicle speed 40km/h to 50km/h.

For 4^{th} gear ratio in gradient condition, the power available is as low as 1044.729W to 1832.345W for power required on gradient 1258.443W to 5312.795W showed in table 4.5. This power will not support the power required to move the car at gradient 2.0% of road vehicle.

For the 3^{rd} gear ratio showed in table 4.5 the power available is 1432.181W to 2241.129W at 40km/h to 50km/h is enough to support the power required for vehicle in gradient 2.0% of road vehicle.



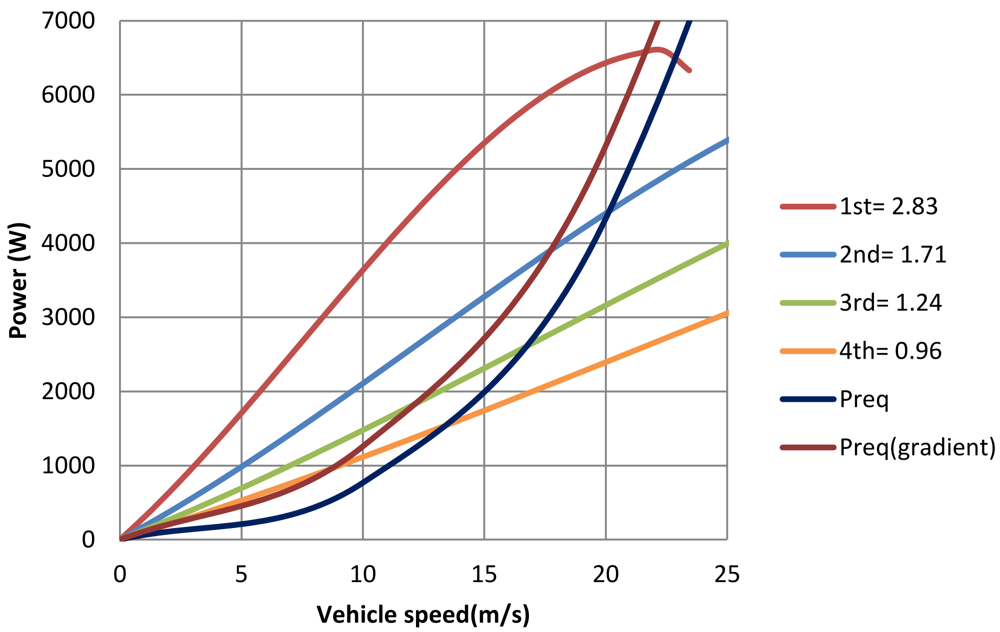


Figure 4.5: Power required and power available to move the car.

Table 4.5: Engine power for various gear ratios.

1st gear		2nd gear		3rd gear		4th gear	
V(m/s)	**Pe**	**V(m/s)**	**Pe**	**V(m/s)**	**Pe**	**V(m/s)**	**Pe**
1.0652	324.512	1.7629	324.516	2.4311	324.517	3.1401	324.51
2.1304	674.131	3.5258	674.141	4.8621	674.128	6.2803	674.138
3.1956	1044.72	5.2887	1044.73	7.2932	1044.72	9.4204	1044.72
4.2608	1432.17	7.0515	1432.17	9.7243	1432.18	12.560	1432.18
5.326	1832.33	8.8144	1832.34	12.155	1832.34	15.700	1832.34
6.3912	2241.12	10.577	2241.13	14.586	2241.12	18.840	2241.13
7.4564	2654.34	12.340	2654.36	17.017	2654.35	21.980	2654.35
8.5216	3067.95	14.103	3067.97	19.448	3067.97	25.121	3067.97
9.5869	3477.79	15.865	3477.76	21.879	3477.78	28.261	3477.77
10.652	3879.67	17.628	3879.64	24.310	3879.65	31.401	3879.66
11.717	4269.52	19.391	4269.50	26.741	4269.50	34.541	4269.51
12.782	4643.23	21.154	4643.22	29.172	4643.22	37.681	4643.22
13.847	4996.62	22.917	4996.61	31.603	4996.60	40.821	4996.61
14.912	5325.61	24.680	5325.61	34.035	5325.60	43.961	5325.60
15.978	5626.09	26.443	5626.09	36.466	5626.08	47.102	5626.09
17.043	5893.86	28.206	5893.87	38.897	5893.86	50.242	5893.86
18.108	6124.86	29.969	6124.85	41.328	6124.87	53.382	6124.86
19.173	6314.97	31.731	6314.97	43.759	6314.97	56.522	6314.98
20.238	6459.99	33.494	6459.99	46.190	6459.99	59.662	6459.99
21.304	6555.78	35.257	6555.79	48.621	6555.79	62.802	6555.79
22.369	6598.30	37.020	6598.31	51.052	6598.31	65.942	6598.31
23.434	6329.17	38.783	6329.19	53.483	6329.18	69.083	6329.18

4.7 Transmission Design

From the result on figure 4.5, 3rd gear is chosen for the ratio that will be used in transmission design for this project. The ratio of the gear is 1.24 which can support power required from the vehicle.

Figure 4.6 showed the design of transmission for the result in gear ratio selection. The design is same as the current transmission design that using the synchronizer to mesh the gear to transmitted power to the wheel. The different is it applies simple single gear ratio that can reduce the weight of the engine compartment.

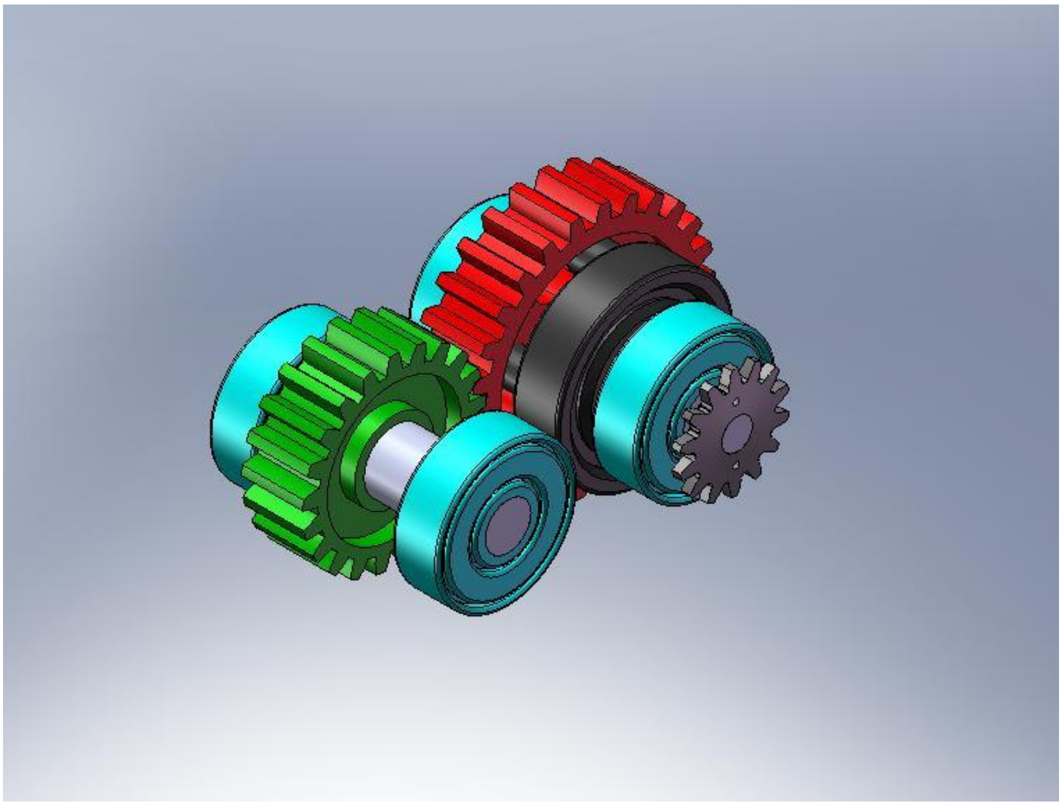


Figure 4.6: Transmission design for two seated urban car.

The specification result for two seated urban car showed on table 4.6. This specification is made from the scope of the project to create result from the calculation analysis. The results will not the same as the specification change.

Table 4.6: Specification for two seated urban car.

Items	**Value**	**Unit**
Vehicle mass (*m*), include driver.	160	Kg
Vehicle speed	40 - 50	km/h
Tyre radius (*r*)	0.2159	m
Frontal area (*A*)	1.69	m^2
Gear ratio	1.24	
Final gear ratio (*Gfd*)	3.0	

4.8 Summary

From this chapter it showed that the result to make the power available nearly to the power required can be found on the 3^{rd} gear that has a ratio 1.24. This result is proved from the calculation analysis that applied in this chapter by consider a factor for the resistance. From the ratio result defined, this project can further overcome the transmission design that will be used for two-seated urban car. The transmission design will be used only single gear ratio to make it more efficient in order can reduce the weight of the transmission by reducing the gear part.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

This project outline the third gear 1.24 is the correct ratio to move the car with the power available from the transmission as near to the power required. By analysis and redesign the transmissions from multiple gear ratios to single gear ratio it can increase the transmission efficiency by remove the entire ratios that produce power far to the power required for the car to move. The result is showed from the power curves that produce from various transmission ratios.

To achieve the result for the single 1.24 ratio, the car must be in the specifications that have been set. The specification for the car should have a weight maximum of 160kg, the velocity maximum to 40km/h to 50km/h, tyre radius is 0.2159m, the frontal area is 1.69m^2 and the final gear ratio is 3.0. The result for the power curve is change due to the change of the specification that has been set. This specification can be used for the further development of this project.

5.2 Recommendation

The improvement need to be done is the frontal areas that need to decrease and also the weight of the car. This condition will affect the performance in term of efficient of the engine. It will reduce the resistant that cause the losses to the power efficiency.

There are several recommendations would like to express for the future final year project. The title of the project must be given on the field of the course taken. The important is the equipment for run the project must be prepared for the study.

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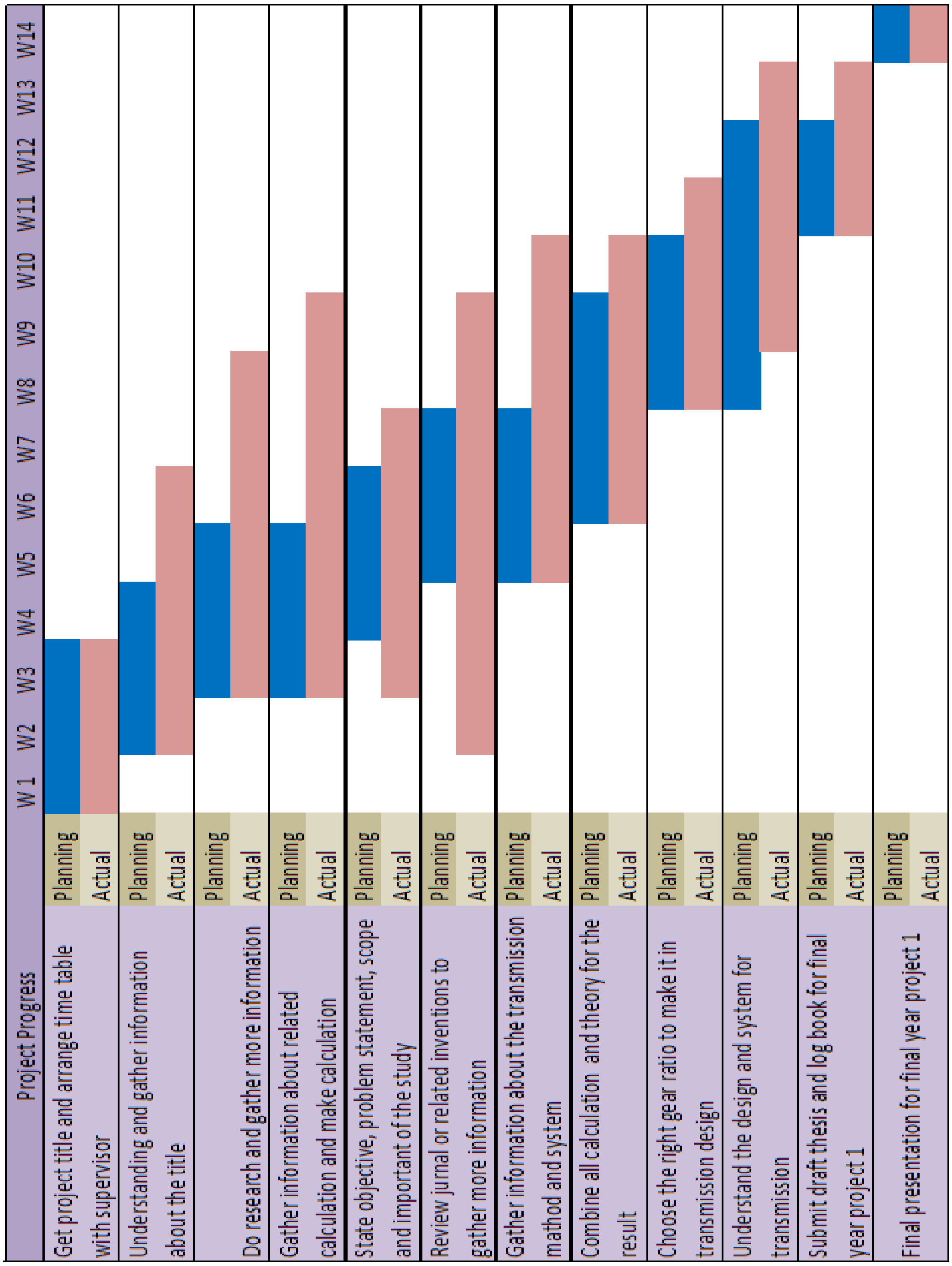
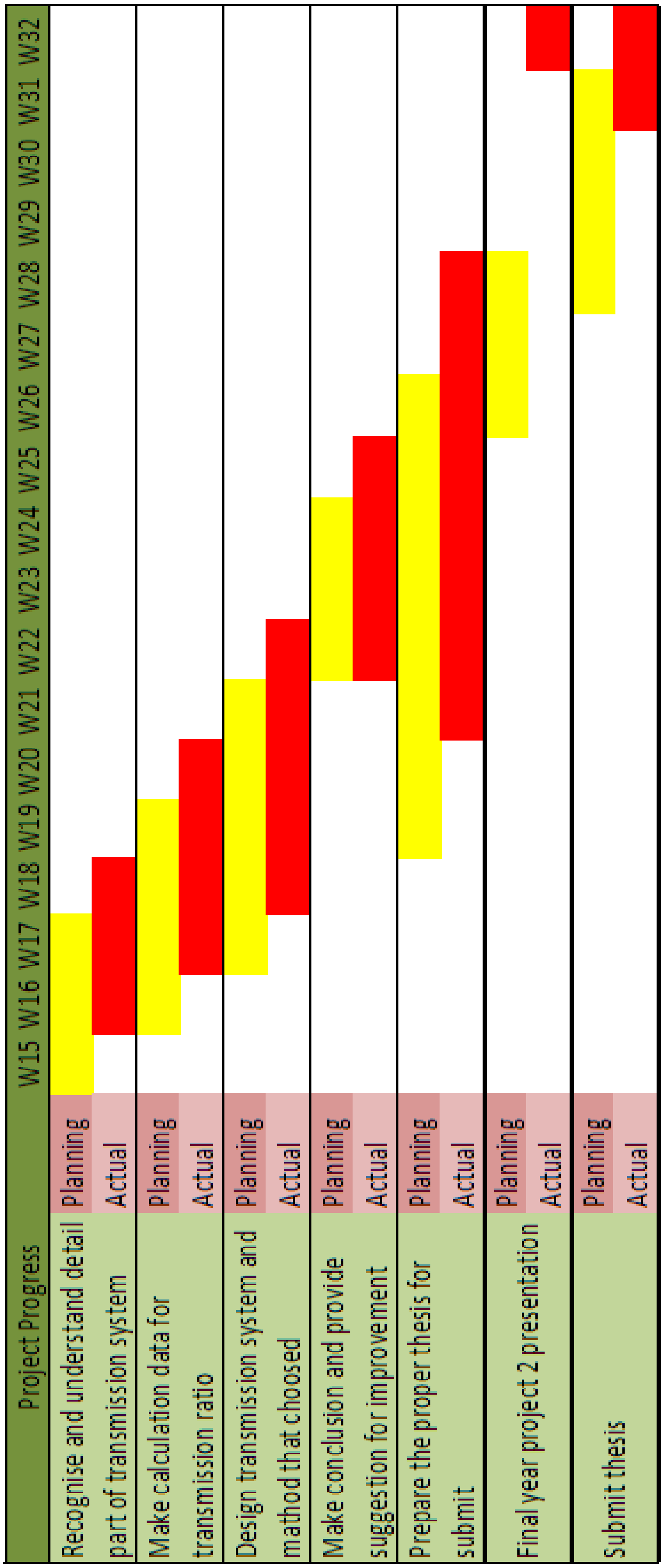
APPENDICES

APPENDIX A

Gantt chart of the project

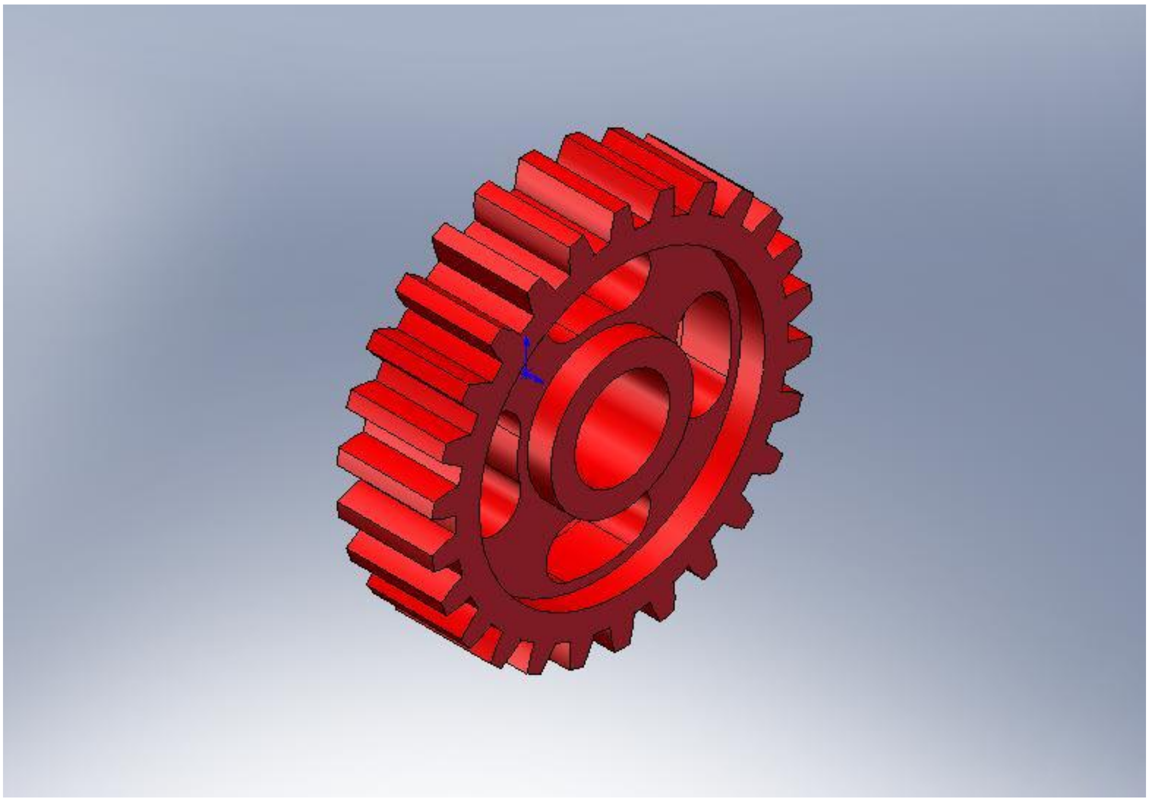




Project Progress
W15 W16 W17 W18 W19 W20 W21 W22 W23 W24 W25 W26 W27 W28 W29 W30 W31 W32
Recognise and understand detail part of transmission system
Make calculation data for transmission ratio
Design transmission system and mathod that choosed
Make conclusion and provide suggestion for improvement
Prepare the proper thesis for submit
Final year project 2 presentation
Submit thesis
Planning
Actual

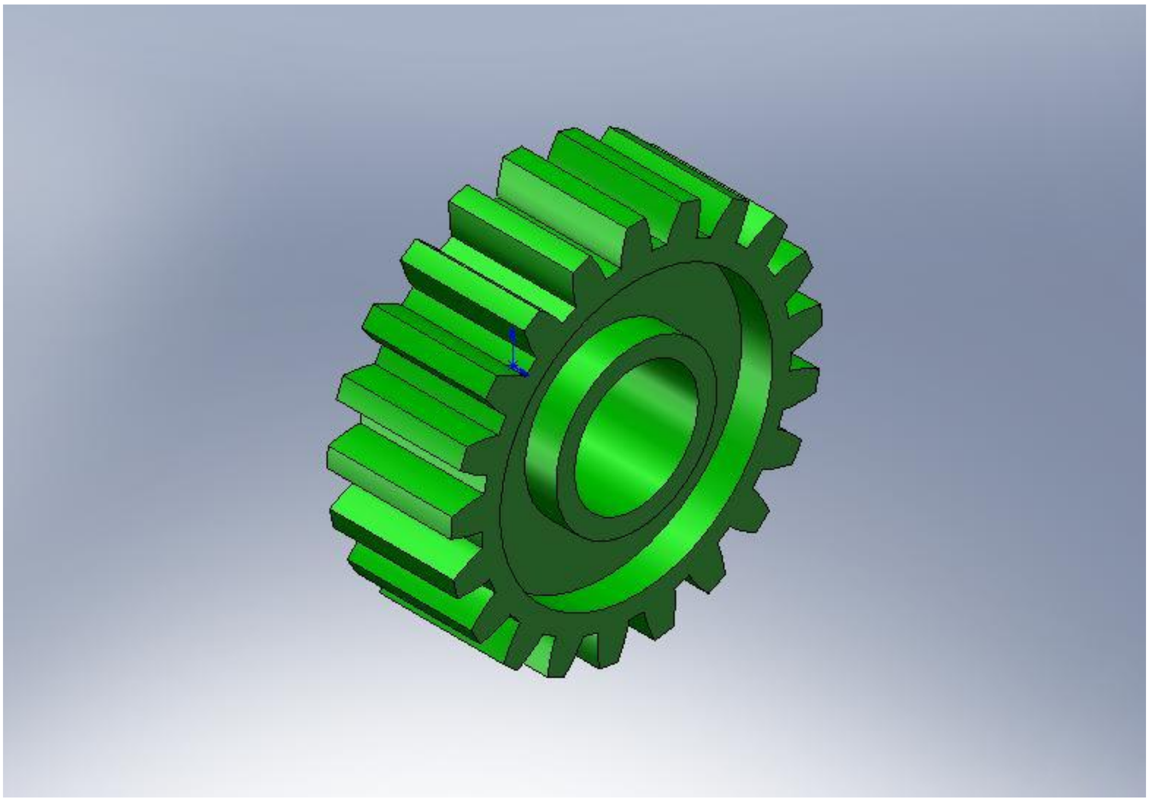
APPENDIX B

Secondary gear 26 teeth diagram



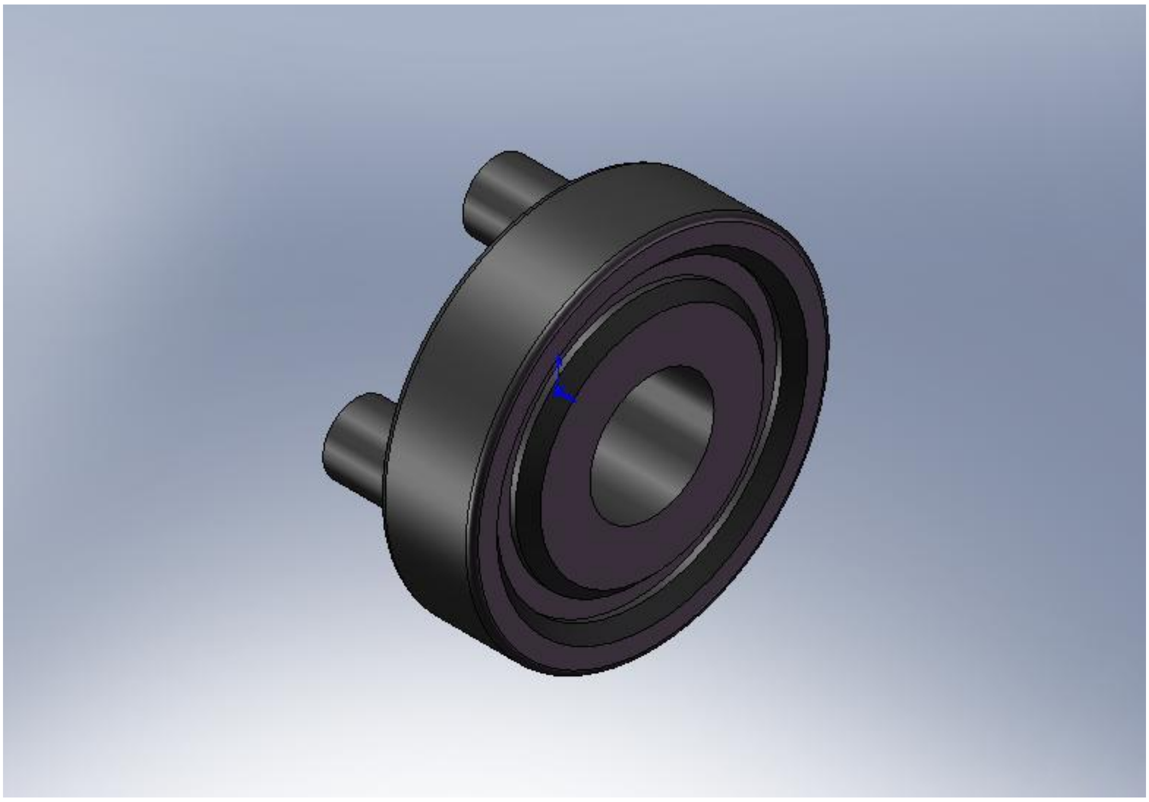
APPENDIX C

Primary gear 21 teeth diagram



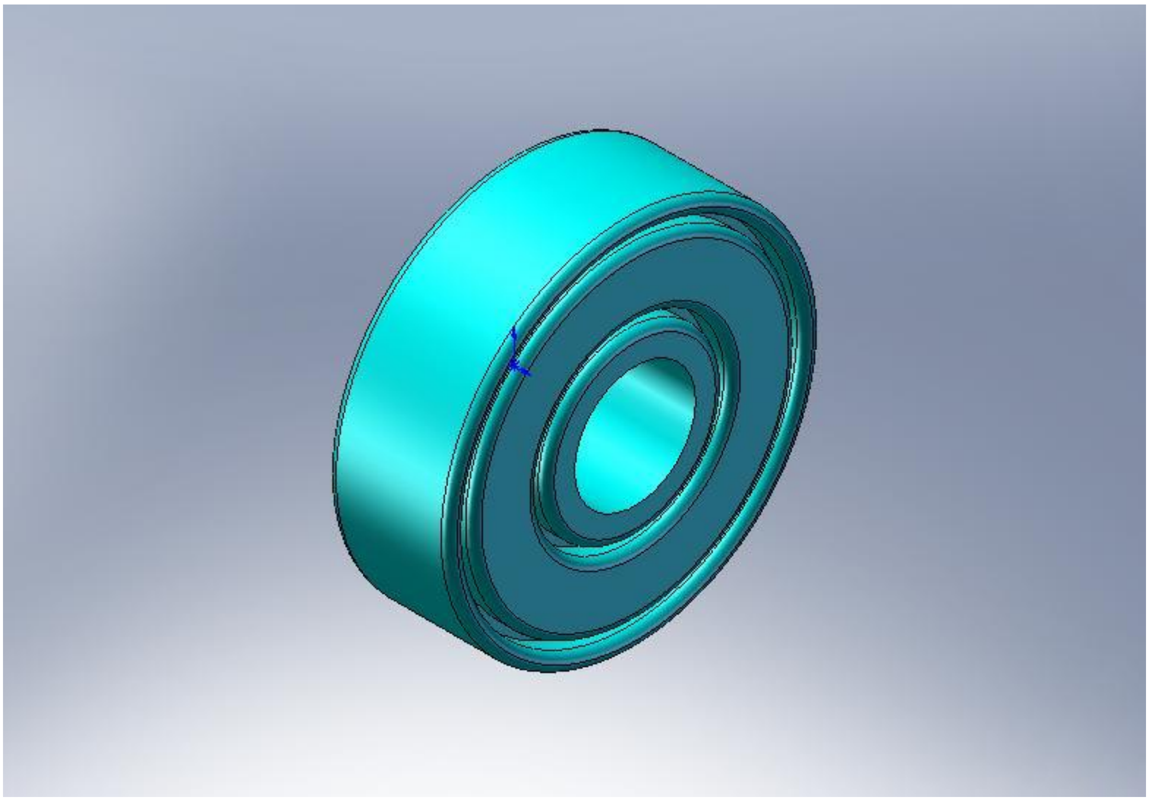
APPENDIX D

Synchronizer for engage gear diagram



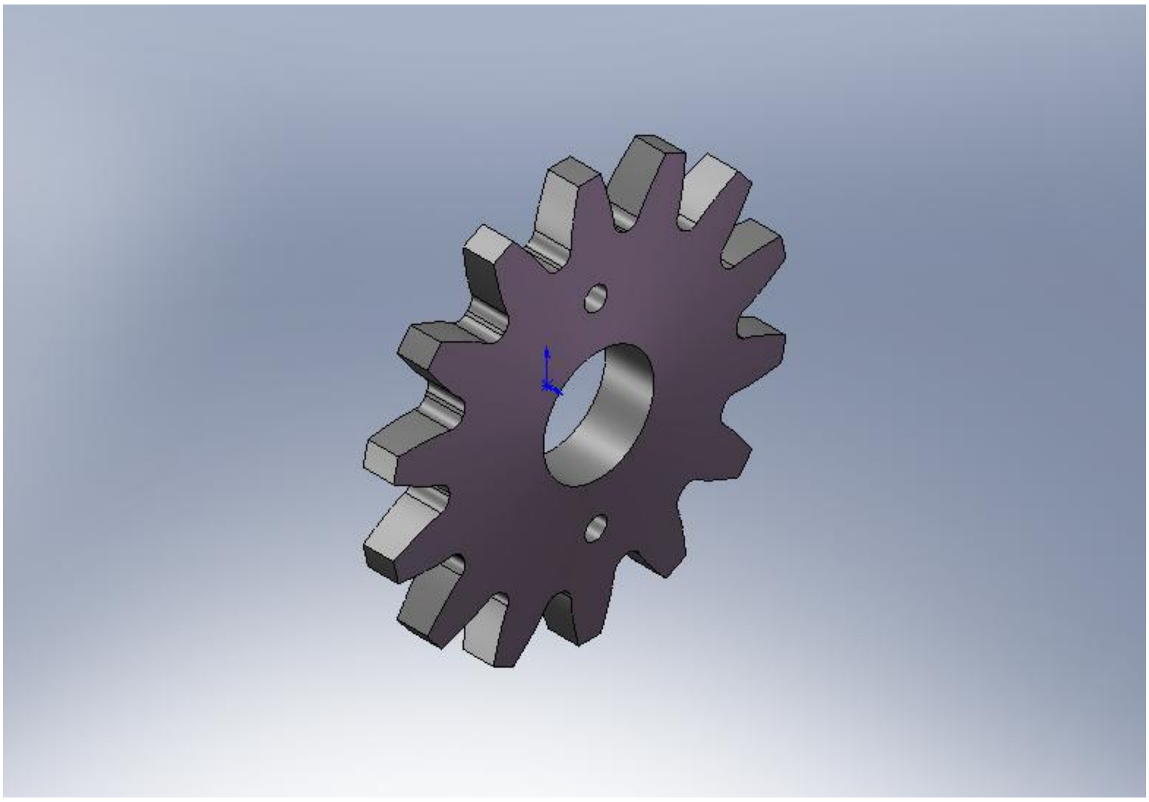
APPENDIX E

Bearing diagram



APPENDIX F

Driver sprocket 14 teeth



APPENDIX G

Sample calculation for engine torque

Pmax = 6.6kW @ 8500rpm

Tmax = 9.3kW @ 4000rpm

$$\mathrm{P_1} = \frac{\mathrm{P_m}}{\mathrm{W_m}} = \frac{6.6\mathrm{kW}}{8500\mathrm{rpm}} = 7.4147$$

$$\mathrm{P_2} = \frac{\mathrm{P_m}}{\mathrm{W_m}^2} = \frac{6.6\mathrm{kW}}{(8500\mathrm{rpm})^2} = 8.33 \times 10^{-3}$$

$$\mathrm{P_3} = -\frac{\mathrm{P_m}}{\mathrm{W_m}^3} = \frac{6.6\mathrm{kW}}{(8500\mathrm{rpm})^3} = -9.33 \times 10^{-6}$$

Engine torque, Te:

$$\mathrm{T_e} = \frac{\mathrm{P_e}}{\mathrm{W_e}}$$

$$\mathrm{T_e} = \mathrm{P_1} + \mathrm{P_2 W_e} + \mathrm{P_3 W_e}^2$$

$$T_e = 7.4147 + 8.33 \times 10^{-3} W_e - 9.36 \times 10^{-6} {W_e}^2$$