Drive mechanism for Personal Transporter (PT)

by

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Dissertation submitted in partial fulfilment of

the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

JUN 2010

Universiti Teknologi PETRONAS

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

____

(Mr. Azman Bin Zainuddin)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

JUN 2010

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.

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(Syahril Izzat Bin Azmi)

ABSTRACT

Dependency on fossil-fuel transportation contributes to excessive emission of carbon dioxide gas in atmosphere. This leads to a serious environmental problem which is called global warming. Serious action has to be taken immediately by all in order to stop the problem from becoming worst. One of the effective ways to doing so is to stop depending on fossil-fuel vehicle. Thus, Personal Transporter provides an answer for this problem. There are companies such as Segway Corp., Honda and Toyota have taken advance step by producing their own transporter. As it is new in the market, the cost is still expensive. Thus, there is a need to make it affordable for all. One of the crucial elements in Personal Transporter is its drive mechanism. Reduction of cost in drive mechanism will significantly reduces the price of the transporter. To design such mechanism, designer has to consider all criteria needed for the system and come up with another cheaper solution. This report provides parts needed for the drive mechanism, the arrangement of the system and also the unique function of the drive mechanism.

ACKNOWLEDGEMENT

This project has taken about a year of planning and working processes in order to fulfill the requirements of Final Year Project subject. In a year duration, many people had being involved either directly or indirectly whose contribute in assorted ways to this project in order to make it a success. Thus, it is a pleasure for me to convey my heartfelt gratitude to them all in this acknowledgement.

Greatest gratitude to God Almighty, because of His bless I am able to complete my final year yroject. I would like to express my full appreciation to Mr. Azman Bin Zainuddin. He was not only served as my supervisor but also encouraged and challenged me throughout his constructive comments, critics and valuable suggestions. Besides, his full supports and advices had actually inspired me and develop my skills as a researcher and engineer to be.

It is also a pleasure to pay tribute to the lecturers from automotive division in Mechanical Engineering Department, Dr. Ir Masri Bin Baharom and Dr. Vu Trieu Minh for their endless support and guidance throughout this project. Not to forget, my parents and family members who have given a never-ending support in the quest to gain valuable knowledge and experience in this university throughout the completion of this project. They deserve a special mention for their inseparable supports and prayers.

Certainly on the list of peoples to be thank of are all friends from all around and also to those people that not be mentioned here.

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Chapter 1
INTRODUCTION

1.1 Background of study

Nowadays, personal transporter (PT) starts to get attention mostly in developed country. As awareness about the environment increases align with level of knowledge among the new century citizen, world starts to look at environment friendly means of transportation. Now, designer starts to consider preserving environment in their design. Personal transporter (PT) is very ideal to help reducing carbon dioxide footprint in atmosphere. This is because the biggest contribution of carbon dioxide comes from transportation.

Back in 2001, world's first self-balancing personal transporter has been introduced to the public. This new concept of personal transportation gives more advance and mobility to all users all around the world with zero emission. Giant car companies are also join the race. Companies like Honda and Toyota have produced their own personal transporter.

The most crucial aspect in personal transporter is its drive mechanism. Generally, drive mechanism consist of electric motor, battery and controller. It is important for designer come up with a reliable drive mechanism with low cost so that it will cut down the production cost and make it affordable to everyone.

1.2 Problem statement

The conventional vehicle which is combustion engine vehicle has several disadvantages. Therefore, there is a need to improve the weaknesses so that it will benefit us even more. The problems/weaknesses of the conventional vehicle are as below:

1.2.1 <u>Dangerous gas releases by conventional vehicle</u>

Exhaust gas from engine combustion produced Carbon Monoxide which is poisonous for human. Besides that, it also releases Carbon Dioxide which is commonly known as Green House Gas that can causes global warming. It is clear that world need to stop depending on oil and gas in order to have better life. The energy generated to power the Electrical Vehicle (EV) and the energy to move the vehicle is 97 percent cleaner in term of noxious pollutants compared to combustion engine vehicle. [1]

1.2.2 Low efficiency of energy conversion in combustion engine

There are only 20% of the chemical energy in gasoline gets converted into useful work at the wheels of an internal combustion vehicle, 75% or more of the energy from battery reaches its wheels [1].

1.2.3 Needs to stop depending on non-retrievable energy source (Oil and Gas)

Looking on the trend of technology, now most of the big car manufacture companies started to consider Electrical-Powered vehicle in their latest model. Take Honda for example, this year they had lunched the new Honda Hybrid Car which is also uses Electrical Power as secondary power system for driving the car. It is predicted Electrical Powered Vehicle will be the chosen type of transportation in the future as awareness of the need of clean energy increases among people nowadays.

1.2.4 The needs of practical and safe transportation for short distance travel

In 2005, study shows there are 1.7 person/car for Malaysia Population [12]. In sense of short distance travel, it is not practical to travel with car as compared to motorcycle/scooter especially for single traveler. But Motorcycle/Scooter also have their own disadvantage which is its accident rates are really high as compared to cars. The safest way of short distance travel is just by walking where they rarely share the same path with vehicles.

1.2.5 The available transporter in market is too expensive

One manufacturing company called Segway Corp take an advanced step by producing a Human Transporter which is suitable to be used on walking path as well as conventional road. The scooter solves most of the short-distance-travel's problem but it is too costly for middle income group of people. Therefore, it is crucial for inventors to come out with alternative to compete with Segway-PT will cost around $8000 which is about RM29000 and for consumer purposes will cost around $3000 which is RM11000.

1.3 Objective and scope of study

The objective of this project is to design drive mechanism system for personal transporter (PT) which emphasis on reduction of cost by focusing on simple design**.**

The project only covers aspects of drive system which are the electrical motor selection, gearing design, and battery selection. Other components that affect the drive mechanism system such as steering, overall design, structure of the transporter, and etc will be assumed.

1.4 Significance of project

A good drive mechanism system will provide useful info for the PT designers. It somehow will accelerate the adoption of the technology to the society. With low production cost of drive mechanism, the total cost of the new PT will also reduces. Therefore, more people can afford to have the transporter which also leads to low carbon dioxide emission.

Life in congested traffic area would also be better, as personal transporter might be a better alternative to avoid traffic congestion and at the same time economically better for not using fuel as power source. Indirectly, less pollution will be made by human. Not just air pollution but sound pollution also will be better from time to time.

Chapter 2
LITERATURE REVIEW

In this chapter, four designs of the existing personal transporter (PT) in operation will be reviewed. There are Segway-PT, Honda UX-3, Toyota Winglet, and also Yikebike.

2.5 Segway-PT

In Segway PT, two different DC motors use to drive the transporter. Both motors are connected to two distinguish gearboxes before it is connected to tires. In figure 1, the picture shows the location where the motor is placed inside the Segway-PT.

Figure 2.1: Drive motor is placed at the side of Segway-PT main body

2.5.1 Gear box

A two-stage reduction system provides a 24:1 reduction. Each gear is cut to a helical profile, which creates a spiral engagement to minimize noise and increase the load capability of the gears. The number of teeth on each gear is chosen to produce noninteger gear ratios. This means that the teeth will mesh in a different location each revolution, maximizing the life of the gearbox. The gearbox is pre-assembled and lubricated, and is designed to require no maintenance over the life of the segway-PT [3]. Figure 2 schematic of the gear assembly inside the Segway-PT's gearbox and figure 3 show the real picture of where the Segway-PT's gearbox is located.

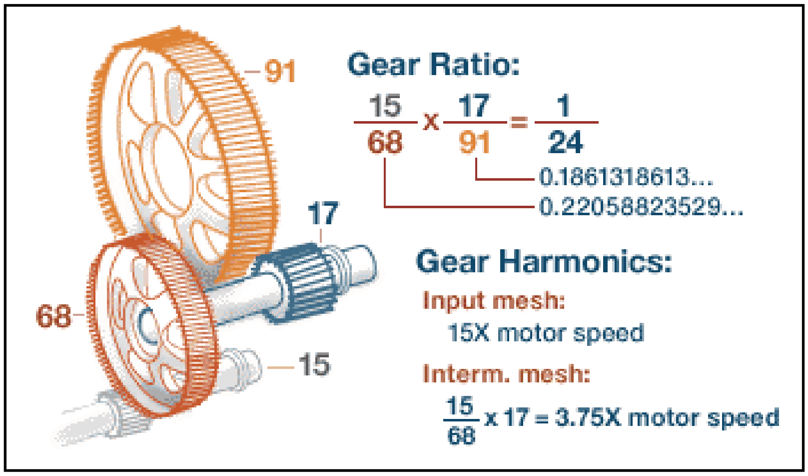

Figure 2.2: Sketch of gear system assembly inside Segway-PT's gearbox[19]

Figure 2.3: Workers install two gearboxes at both sides of Segway-PT main body.

2.5.2 Suspension

Segway-PT tyres have been uniquely designed by Michelin to act as the suspension for the device. Therefore there is no suspension or any other shock absorber mounted inside Segway PT body. This somehow simplifies the design of the transporter itself [3].

2.6 Honda U3-X personal transport device

Honda U3-X is said to be the newest revolutionary personal transport device. Weighing just less than 10kg, roughly one meter tall, 30cm long and 15cm wide – this self-righting unicycle is so small that it can fit into the car door of Honda's new concept electric car the EV-N [4].

Figure 2.4: Honda U3-X transporter and also how it fit into EV-N door

The U3-X boasts about an hour of battery life and it has top speed of just under 10kmp/h. The key concepts behind the design are simply to make a personal transport device that is easily stored, non-obtrusive if taken on larger public transportation like trains or buses, easy to operate and fun. Table below shows key specification of the model.

Table 1: Key specification of the U3-X personal transporter

Length × Width × Height(mm)	315 × 160 × 650
Weight	less than 10kg
Battery Type	Lithium ion battery
Operation time (with fully charged battery)	1 hour

Honda developed the world's first wheel structure which enables movement in all directions including forward, backward, side-to-side and diagonally. Multiple small-diameter motor-controlled wheels were connected in-line to form one large-diameter wheel. By moving the large-diameter wheel, the device moves forward and backward, and by moving small-diameter wheels, the device moves side-to-side. By combining these movements the device moves diagonally [5]. Figure 5 below shows illustration of small-diameter and large-diameter wheel for the transporter.

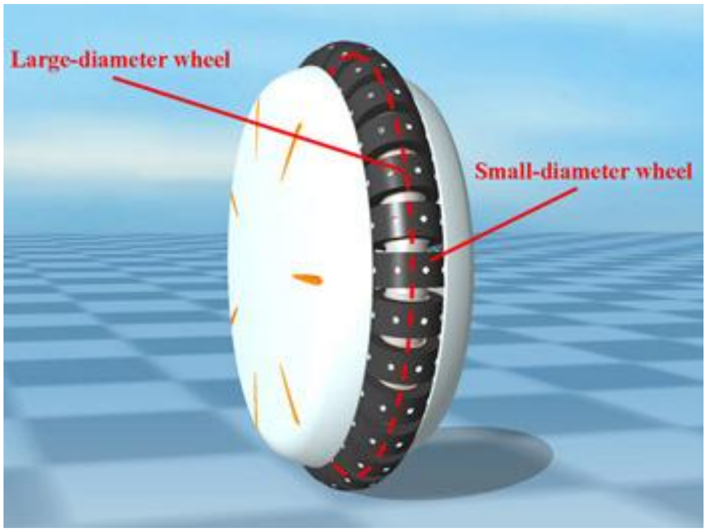

Figure 2.5: Small and large diameter wheel for the transporter

2.7 Winglet Personal Transporter

The Winglet, Toyota's personal transporter in the style of Segway, will start initial production next year, 2009. Aiming for corporate customers, Toyota will deliver the first batch of 10 Winglets to the Central Japan International Airport in Aichi Prefecture

Figure 2.6: Winglet PT by Toyota

The two machines (Segway PT and Winglet PT) look similar and employ similar technologies- self-balancing technology. One noticeable difference is the size: the 10kg Winglet is just one-third the weight of the Segway PT, which looks more robust. The Winglet's top speed is 6km/h compared to 12.5km/h for the Segway PT [6]. With Toyota's focus on portability, the small size and light weight is a step in the right direction. However, expect to see more rugged versions of the Winglet in the future [6]. The Winglet range consists of the L, M and S versions, the latter two of which don't have handlebars and instead are gripped by the calves. L version has maximum range of 5 km/single charge while M and S versions can go to 10 km/single charge [6].

A bigger difference is the price. Segway-PT sells for around 11,000 USD while the projected price for a mass produced Winglet is just around 3,300 USD, according to Toyota [7]. The Winglet PT has a body the size of an A3 sheet of paper that houses an electric motor, two wheels, and internal sensors that constantly monitor the rider's position [8].

2.8 Yikebike

The 'yikebike' by inventor grant ryan and engineer peter higgins of new zealand, is a mini-farthing bike designed to battle the increasing urban congestion of today. It uses carbon fiber frame and weighs less than 10kg [9]. Yikebike's electronic can travel at speeds up to 20 km/h and having range of 10 km.

Figure 2.7: Folded and non-folded Yikebike

The transporter cost around 5,500 USD [9]. It uses Electric brushless DC motor which generates 1 kW of power.

Built from carbon fiber and weighing in at 22 pounds, the Yike Bike is powered by a custom 1kW motor, a better power to weight ratio than many sports cars, and can be fully recharged in under 30 minutes. Weight limit for the usage is about 100 kg. In terms of form factor, the Yike Bike operates using an electric chainless drive on its front 20′ hubless wheel.

Chapter 3
METHODOLOGY

There are several processes involve in order to design a proper drive mechanism that meets all proposed requirements. The steps cover from determination of design specification until the detail design.

The first step is Preliminary design. In this step, writer develops design concept for the new transporter. In this stage also, criteria and technical specification of the transporter will be determine align with the objective of the project.

After the preliminary design stage done, next step is force determination. For this stage, writer considers the highest load for the design should be carried. The drive mechanism should be able to achieve its targeted speed with its full load.

Then, after the maximum performance is known, the next step is to select the suitable electric motor and battery that available in the market. The selected parts should be able to deliver performance as been mention in the objective and also in the stage before. The project will not include electric motor and battery design. Writer just uses parts that available in the market.

The next step is to determine gear ratio. This step only can be done after specification of electric motor is known. In this stage, writer uses try and error method. Motor torque and speed is set as constant variable while gear ratio is manipulated variable. These two variables will results in tyre speed and torque. Then, the most suitable gear ratio will be use for transporter

Detail design is performed after gear ratio is confirmed. In this stage, detail about gears, arrangement of parts and other relevant issue will be determine. Lastly, after all matters that been discuss previously is confirmed, technical drawing will be done. Finished technical drawing indicates the accomplishment of the project.

3.1 Process flowchart

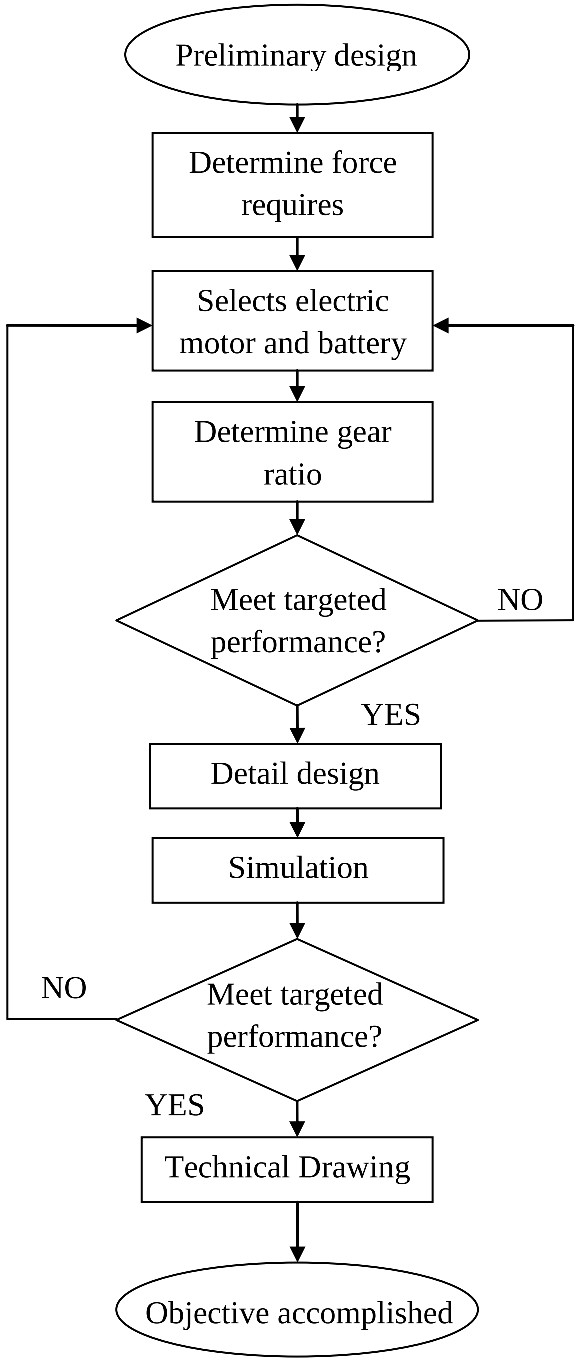

Figure 3.1: Process flowchart for designing drive mechanism

3.2 Gantt Chart

Activities		FYP 1	29-Jan	5-Feb	12-Feb	19-Feb	26-Feb	5-Mar	12-Mar	19-Mar	26-Mar	2-Apr	9-Apr	16-Apr	23-Apr	30-Apr
Planned	**Descriptions**	Sem 7	Week 1	Week 2	Week 3	Week 4	Week 5	Week 6	Week 7	Week 8	Week 9	Week 10	Week 11	Week 12	Week 13	Week 14
Priliminary design	Development of design concept															
	Design criteria															
	Selection of design															
Determine force requires	Load requirement calculation															
	Gear ratio determination															
Select electrical parts	Browse suitable items from supplier															
	Determine the suitability of the items															
Detail Design	Design Gear															
	Design arrangement of parts															
Detail Drawing	Produce detail drawing using CATIA															
Simulation	Using MATLAB															

Figure 3.2: Milestone of the project

3.3 Software tools

CATIA

The software is used to produce drawing. The main purpose of using CATIA instead of AutoCAD is because the finished drawing can be saved into IGS file format where then the file can be run using ADAMS for simulation purposes. Besides that, types of material can be specified in CATIA which add the parameter such as density to the part. This will increase the value of the analysis.

MICROSOFT EXCEL

The software is used to do simple iteration. Even though the function quite limited as compared to MATLAB, Microsoft Excel is practical because it easy to used. In this project, this software been used as estimation tools where try and error method is performed by using the software.

MATLAB

There is some iteration that cannot be performing by using Microsoft Excel. For that purposes, MATLAB is used as replacement for the task.

Chapter 4
RESULTS AND DISCUSSIONS

To start designing drive mechanism, first step taken by writer is estimates the torque requires. In the stage, the transporter is assumed to work in its highest operation load which is 165 kg and the road condition is zero inclination. Then, the next step is focusing on design selection which the basic design of the transporter is been set. Next step is selection of electrical parts and continued with parts arrangement.

4.1 Torque-required Estimation

The drive system should be able to provide sufficient amount of traction force to counter these opposition forces in order to make the transporter move. The force requires is formulated as:

$$\mathbf{F_{Total} = F_{Traction} = D = F_{Air} + F_{Roll} + F_{Slope} + F_{Accel}}$$

Where;

Component	Unit	Equation
Air resistance	N	$\mathbf{F_{Air} = \frac{1}{2} \rho v^2 A C_D}$ Where: ρ = Air density (kg/m^3) v = Velocity of the transporter (m/s) A = Frontal Area of Transporter + Rider (m^2) C_D = Drag Coefficient
Body surface area	m^2	$\mathbf{BSA = \sqrt{(h * m / 3600)}}$ Where: h = Rider height (cm) m = Rider mass (kg)
Frontal surface area	m^2	**FSA = A = 2/5 BSA** Where: BSA = Body surface area ($m^{2)}$
Rolling Force	N	$\mathbf{F_{Roll} = C_R\, m\, g\ ,\ C_R = 0.0136 + 0.04 * 10^{-6} * (v * 3.6)^2}$ Where: C_R = Rolling Coefficient V = Velocity of the transporter (m/s)
Inclination Force	N	$\mathbf{F_{Slope} = s\, m\, g}$ Where:s = Slope inclination measured from surface (°)

		m = Mass of transporter + Rider (Kg) g = Gravity (m/s$^{2)}$
Acceleration Force	N	$\mathbf{F_{Accel} = m\ a}$ Where: m = Mass of transporter + Rider (kg) a = Transporter acceleration (m/s^2)
Angular Velocity	Rad/sec	$\boldsymbol{\omega = v/r}$ Where: v = Velocity of the transporter (m/s) r = Wheel radius (m)
Rotational Speed	Rev/min	**RPM = (ω × 60)/2Π** Where:ω = angular velocity (rad/s)
Torque	Nm	**Ʈ = F×r** Where :F = Force exerted (N) r = wheel radius (m)
Power	W	**P = (2Π ×Ʈ × RPM)/60** Where:Ʈ= Torque (N.m)

4.1.1 Area of air resistance

For transporter, the frontal area is assumed same as Segway Human Transporter, while for rider fontal area is assumed 2/5 of body surface area where Mosteller Formula is applied for calculating body surface area [10].

Segway-PT area: $A_{transporter} = 0.2\ m^2$

Human body area: BSA = √ (h * m / 3600) = √ (167.7cm * 75kg / 3600) = 3.49

Where: h = 167.7cm [3] and m = 75kg [3]

FSA = 2/5 BSA = = 2/5 (3.49) = 1.40 m^2

Total air resistance area: A = 1.40 + 0.2 = 1.60 m^2

With; Air density: ρ = 1.225 kg/m^3

Drag Coefficient, C_D = 1.5 [2]

4.1.2 Roll resistance

Rolling resistance has non-linear relationship with the velocity of the transporter. The equation is based on type of tyre. For calculation purposes, tyre of the transporter is assumed to be Radial-ply tyre [10]. The coefficient equation is as below:

Rolling Coefficient

$$C_R = 0.0136 + 0.04 * 10^{-6} * (v * 3.6)^2$$

Where: v = velocity of the transporter

4.1.3 Inclination force

The inclination force is the y-component force as the transporter at non horizontal terrain. The transporter will be designed to be capable of carrying passenger at terrain up to 20º of slope at it full load.

4.1.4 Acceleration force

Acceleration of the transporter is assumed to be linearly reducing as it is moving. The maximum acceleration will be at the starting point where the transporter just started to move and it will reach zero acceleration at its top speed at 8.89m/s. The maximum acceleration is taken as $0.94 m/s^2$ [10]

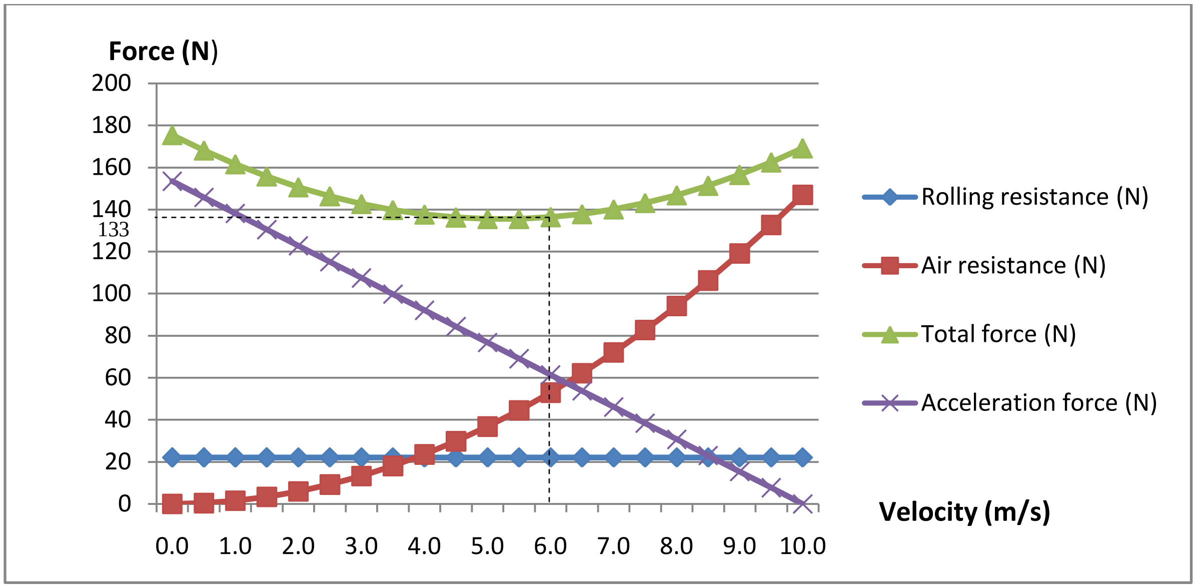

Figure 4.1: Estimated force element in the transporter by referring to Segway-PT (refer to appendix 1)

Traction force needed in must be equal to total force in figure 4.1 in order to make the transporter move with acceleration $0.94 m/s^2$.

4.2 Design selection

The design for the new transporter comes with many possibilities of outcomes. For this project, the writer has narrowed down the things to be considered into four criterions which are;

a) Either two motor been used or only one motor been used.

b) Either the new transporter is two wheels or three wheels

c) either the transporter is rear wheel drive (RWD) or Front wheel Drive (FWD) – for three wheel only

d) Either two tyres in front or one at the back or vice versa – for three wheel only

Each criterion above has its own values that need to be evaluated in order to decide the best option. To select the best design for the transporter, the writer uses Weighted Decision Matrix of selection method. Weighted decision matrix is a method of evaluating completing concepts by ranking the design criteria with weighting factors and scoring the degree to which each design concept meets the criterion [5]. To determine the score, a 5-

point scale is used as the criteria for evaluation is not very detailed [5]. Higher points will be given to the parts that meet the criteria the most.

The weight factor value is obtained by multiplying the weight of each values and the weight of parts itself. Figure 7 below shows the hierarchical objective tree.

4.2.1 Hierarchical Objective Tree

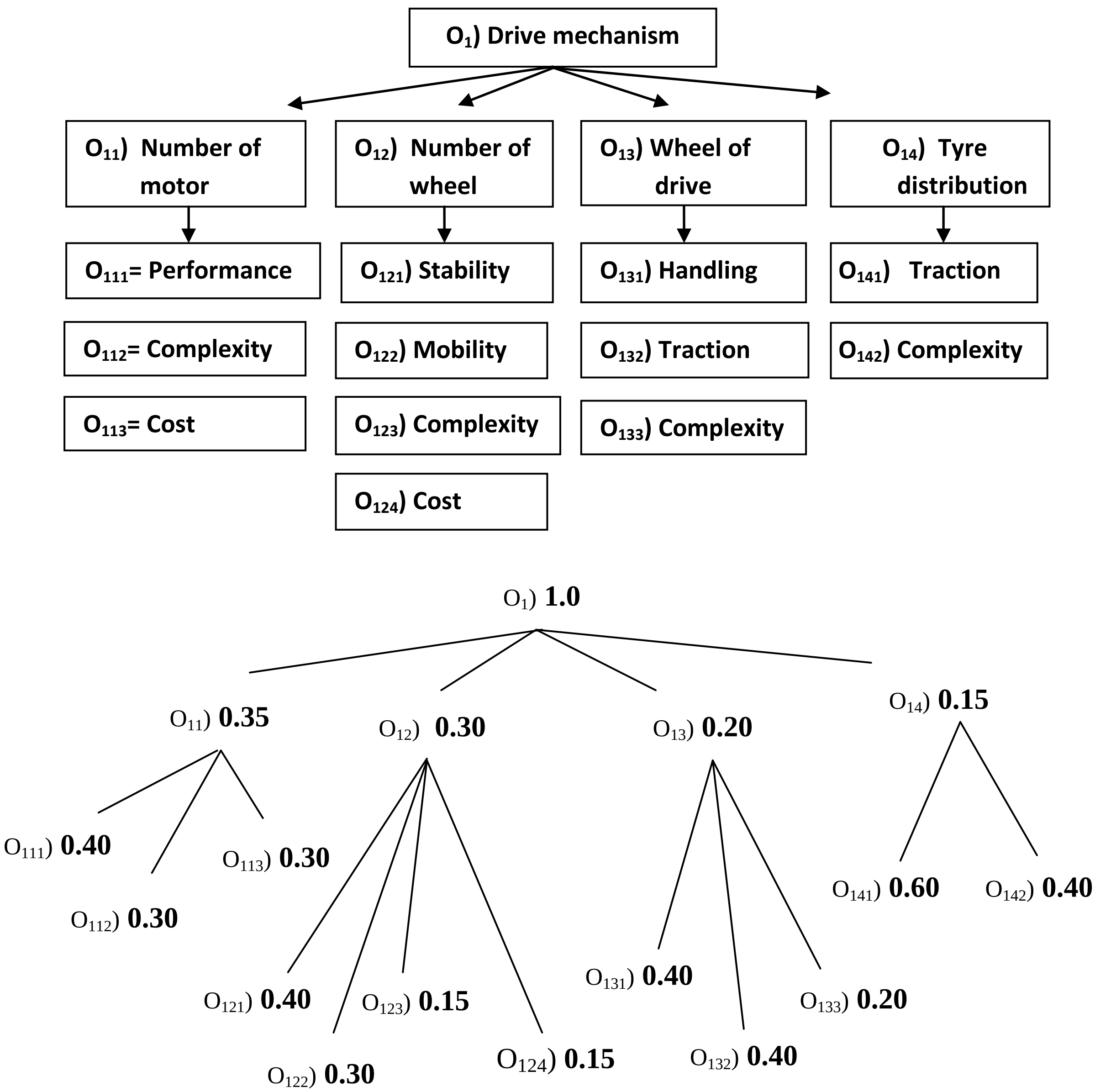

Figure 4.2: Hierarchical objective tree for calculating the weight factor

4.2.2 Number or motor used (one or two)

Based on figure 7, performance, complexity and cost weighted 0.5, 0.25 and 0.25 respectively. Performance weighted the most due to its criticality as it is the main key evaluation of the transporter. Low complexity also plays an important element as it will affect the level of difficulty during manufacturing and also in maintenance job.

- <u>Better performance</u>

  Two motors will exert extra torque compared to a single motor. In this evaluation, writer is comparing the same type of motor. Two motors exert twist power than a single motor; therefore the score for two motors will be twist as compared to a single motor

- <u>Low complexity</u>

  The design with low complexity will score higher as compared to high complexity. Writer put higher score to a single motor because it involves simple gearing system as compared to two motors.

- <u>Low cost</u>

  Even though single motor requires extra parts as been discussed earlier, the main cost normally comes from the motor itself. Therefore, in rating this criterion single motor should be better as compared to two motors.

Table 1 shows the results of above evaluation. Based on the result, two motors will be used as it scores higher compared to a single motor.

Table 4.1: Show the results for number of motor used

Consideration	Weight	Weight Factor	2 motors		Single motor	
			Score	Rating	Score	Rating
Better performance	0.4	0.14	5	0.7	4	0.56
Low complexity	0.3	0.105	2	0.21	4	0.42
Low cost	0.3	0.105	3	0.315	4	0.42
Total	**1**	**3.5**		**1.225**		**1.4**

4.2.3 Number of wheels (two or three)

In this evaluation, stability, complexity, mobility and cost weighted as 0.4, 0.3, 0.15 and 0.15 respectively. Stability scores the most due to safety reason. The transporter should have high stability in order to prevent the rider from falling. Meanwhile, complexity and cost score weighted the lowest as it is not a critical issue as compared to stability and mobility.

- High stability

  3 wheel transporter has higher stability compared to 2 wheel vehicle. Two wheels transporter has a possibility to stumble in direction perpendicular to its tires alignment. Meanwhile, for 3 wheels transporter it has no tendency to stumble in any direction as the tires support the vehicle in triangle-shaped base

- High Mobility

  High mobility is an important factor for any transporter. High Mobility is a level of how easy something to be in motion. In the transporter sense, it means the level of easiness for the transporter to tackle any terrain for the transporter to move. Three tyres will impose extra friction as compared to two tyre transporter. This somehow will exert extra load to the vehicle.

- Low complexity

  Transporter with two tyres is more complex than transporter with three tyres. This is because, two tyres transporter need a balancing system in order to allow rider to use it smoothly. By using three tyres, we can ignore balancing system and it will simplify the design a lot.

- Low cost

  Three tyres will add extra cost to the transporter as the parts needed are increased. One of the targets for in this project is to minimize the cost of manufacture. Therefore, extra score will be given to the design that has a lower cost. In this case, two wheels scores 3 and three wheels scores 2.

Table 2 shows the number of tyres evaluation's result. In this analysis, three wheels transporter is better as compared to two wheels transporter.

Table 4.2: Results for number of wheel used analysis

Consideration	Weight	Weight Factor	2 wheel		3 wheel	
			Score	Rating	Score	Rating
High stability	0.4	0.12	2	0.24	4	0.48
High mobility	0.3	0.09	4	0.36	3	0.27
Low complexity	0.15	0.045	1	0.045	3	0.135
Low cost	0.15	0.045	3	0.135	2	0.09
Total	**1**	**0.3**		**0.78**		**0.975**

4.2.4 Wheel of drive (FWD or RWD)

There are three types of drive system in conventional vehicle; front wheel drive (FWD), rear wheel drive (RWD) and also four wheel drive (4WD). For the transporter, writer only considers FWD and RWD. This is because the transporter does not require 4WD system as it is not designed to be used on hard terrain such as off-road path. To evaluate this, there are three criteria need to be evaluated which are handling, traction and complexity which respectively weighted 0.4, 0.4 and 0.2. Handling and traction score the highest as it is the most important thing to be considered when choosing the drive system.

•Better handling

Front wheel drive has better handling than a Rear Wheel Drive can deliver — especially in rain and snow. The front wheels pull the car instead of the rear wheels pushing it. Pulling force delivered by front wheel will direct the transporter easily as compared to pushing force from behind. This will lead to better control of the transporter.

•<u>Better traction</u>

Rear and front wheel drive do not contribute much for traction differences since the transporter is designed for light weight purposes (small vehicle). In car, weight of the engine/transaxle sits on top of the (front) drive wheels, which further helps the car get a grip. So it is advantage to front wheel drive.

•<u>Low complexity</u>

Cost normally increases with increment of complexity. In conventional vehicle, complexity will be higher when the wheel has multiple functions such as deliver the drive force and also act as steering mechanism. The project is only focusing on drive mechanism, therefore the complexity of rear and front drive mechanism are considered the same.

Table 3 shows the result of evaluation for wheel of drive. Front wheel drive (FWD) is chosen as it scores the highest in the evaluation compared to rear wheel drive (RWD)

Table 4.3: Results for type of drive system used

Consideration	Weight	Weight Factor	RWD		FWD	
			Score	Rating	Score	Rating
Better handling	0.4	0.08	3	0.24	5	0.4
Better traction	0.4	0.08	4	0.32	4	0.32
Low complexity	0.2	0.04	2	0.08	2	0.08
Total	**1**	**0.2**		**0.64**		**0.8**

4.2.5 Tyres distribution (One tyres in front or two tyres in front)

As we are using three tyres for the transporter, it is crucial to decide on the location of the tyres. To decide on this matter, the evaluation looks into traction and also complexity. The weights for both criteria are 0.6 for traction and 0.4 for complexity.

•<u>Better traction</u>

As per discussed, the transporter will be using Front Wheel Drive (FWD) which locates two tyres in front as it will exert less traction. But it is not happened light weight vehicles which the different in performance is not noticeable.

•<u>Low complexity</u>

By having two tyres in front and FWD system, the complexity with one tyre in front is lower than if two tyres at the in front. This is because; we only need a single part of drive mechanism for driving one tyre as compared to two parts if it is double. Example of the part is the gearbox.

Table 4 below shows the result of the above evaluation, two tyres in front of the transporter is chosen as it scores higher than the other option.

Table 4.4: Results tyres distribution

Consideration	Weight	Weight Factor	1 tyre infornt		2 tyre infront	
			Score	Rating	Score	Rating
Better traction	0.6	0.09	2	0.18	5	0.45
Low complexity	0.4	0.06	4	0.24	2	0.12
Total	**1**	**0.15**		**0.42**		**0.57**

Based on analysis above, the new transporter will be using two motors. It will have three wheels which are two in the front and another one wheel behind. The transporter also is front wheel drive (FWD).

4.3 Electrical Parts

Drive mechanism consist two major electrical parts- DC motor and Batteries. Writer has browsed few suppliers that provide the parts. In Malaysia, writer found no supplier that provides these parts for public usage. Supplier in Malaysia prefers to serve in high quantity which normally is for industrial usage. Therefore it is not economical to consider Malaysia supplier for this project. Most of the suitable suppliers come from oversea, specifically from United State. The project is feasible as there is not impossible to obtain parts from oversea nowadays. Tables and figures below show technical specification of the considered parts.

Table 4.5: Specification of selected electric motor for the transporter

Electric motor	
Power (W)	1000
Voltage (V)	36
Max. Current (Amp)	35.6
Free Rev (Rad/sec)	314.16
Free Torque (N-m)	3.183
Stall torque (N-m)	35
Dimension (cm)	Diameter: 10.795 , Length: 15.24 (17.78 including shaft)
Mounting Bracket Dimensions (cm)	12.065 x 7.303
Weight (kg)	5.0
Price (USD)	129.99
Quantity	1
Website	http://www.monsterscooterparts.com/36voltmotors.html

Figure 4.3: Selected electrical motor for the new drive mechanism

Table 4.6: Specifications of electric motor that will be used for the transporter.

Battery	
Voltage, (V)	12
Amp hour, (Ah)	18
Weight, (kg)	5.94
Dimension, (cm)	7.62 × 16.83 × 18.10
Price, (USD)	54.95
Quantity	3
Website	http://www.electricscooterparts.com/batteries.html

Figure 4.4: Selected battery for the new drive mechanism

4.3.1 Battery Performance

The maximum battery performance is calculated based on assumption that the transporter is operated in its top speed with maximum load on zero inclination of road surface. To estimate how long the battery will last, below is the calculation involve.

At maximum speed, motor requires 35.6 amp of current. At this rate, 18 Ah batteries will last for t_{mo} where;

$$\begin{aligned} t_{mo} &= 18Ah/35.6A \\ &= 0.506\ hr \\ &= 30\ min\ 20\ sec = \mathbf{1820\ sec} \qquad (4.3.1) \end{aligned}$$

4.3.2 Motor Performance

The maximum power of the motor is 1000 W at free rpm of 3000 RPM (314 rad/sec). To calculate torque delivered by the motor at this speed, an equation applicable to permanent DC motor is used.

Permanent magnet DC motors, there is a linear relationship between torque and rpm for a given voltage [13]. The maximum torque occurs at 0 rpm, and is called *stall torque*. The minimum torque (zero) occurs at maximum rpm, reached when the motor is not under a load, and is thus called *free rpm*. The formula for torque at any given rpm is:

$$\mathbf{T = T_s - (N\ T_s \div N_f)}$$

T is the torque at the given rpm **N**, $\mathbf{T_s}$ is the stall torque, and $\mathbf{N_f}$ is the free rpm.

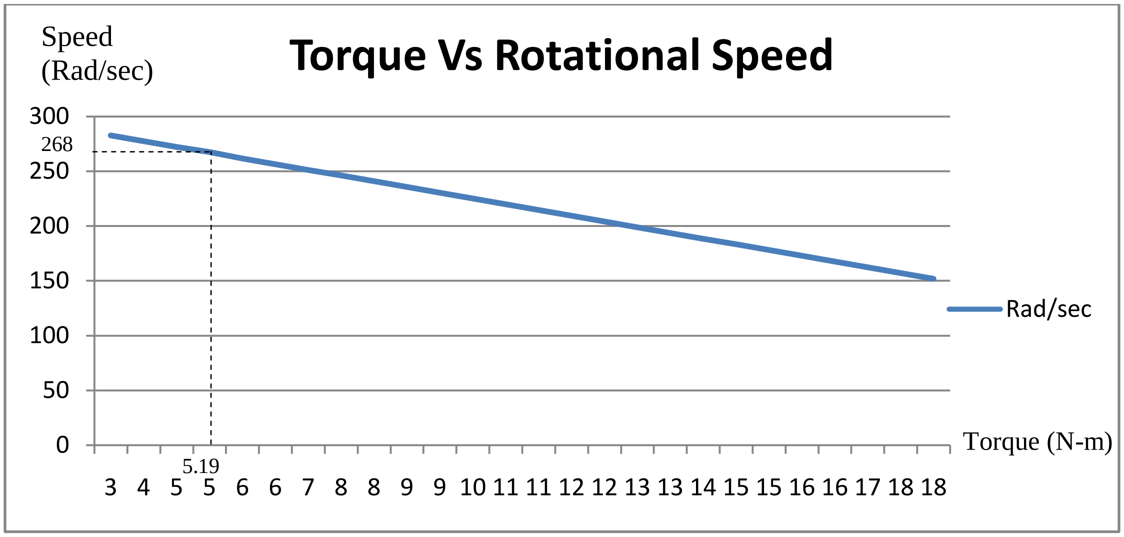

Figure 4.5: Relationship between speed and Torque delivered by selected electric motor

4.4 Parts arrangement

To design gear trains that will results in 1:8.6 gear ratio. The space available must be specified. The criteria that the design should be in line with are as below;

1 Safety
   - The design should not have uneven surface for the rider to stand. Uneven surface expose the rider to danger of losing balance.

2 Should be able to accommodate 1 electric motor, three 12V batteries, gear terrain

3 Good functionality
   - The design should be functioning well
   - The motor should be easily taken out to serve the drive mechanism second function

4 Minimum space as possible

5 Maintainability
   - Easy access for parts replacement

6 Cost

Battery has thickness of 3 inch (7.62cm). With thin thickness it is suitable to be placed in the floor of the transporter with laying position.

Meanwhile, for gearbox, it has 18:1 gear ratio system. Therefore it is expected to contain big diameter of gear (> 3 inch) as compared to the thickness of the battery. Unless the gears is positioned with it exist pointing upward, the gear box is not suitable to be located at the floor of the transporter. Other option for location of the gear box is around of rider's footstep.

For electric motor, it has diameter or 10.795 cm which is relatively big to be placed under rider's footstep. But, if it located between the footsteps, there should be no problem regarding to the issue.

As the project is focusing drive mechanism and not on the overall design of the PT, writer only go briefly on this section. The purposes of the drawings in this section are just for illustration purposes and it is not being drawn in the proper scaling.

4.4.1 First design layout

Three batteries arranged as figure 4.6 which provides an even surface for rider to stand. As electric motor mounted at the front, it makes the motor easily to be taken out easily. The design uses only one gearbox which will cut down cost to manufacture. Figure 11 below shows the layout of design 1.

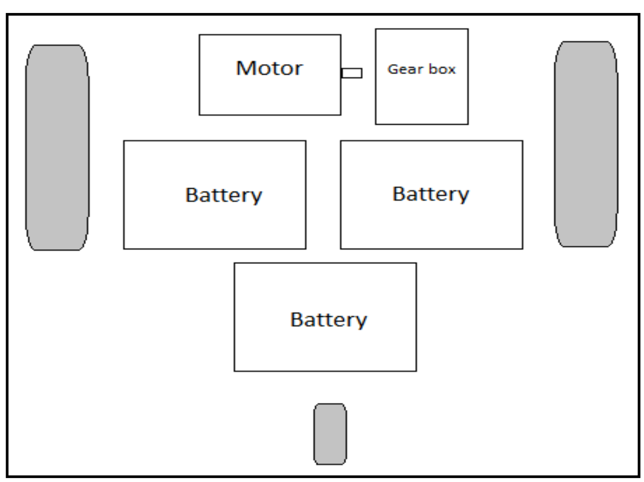

Figure 4.6: Plan view of the first design layout

4.4.2 Second design layout

Design 2 has gearbox at both sides. This will contribute to extra cost compared to design 1. As all three batteries is arrange is laying position, the foot step area is flat. Motor located in front make it easy to be unattached which is good for the drive system second function. Figure 12 below shows the illustration of design 2.

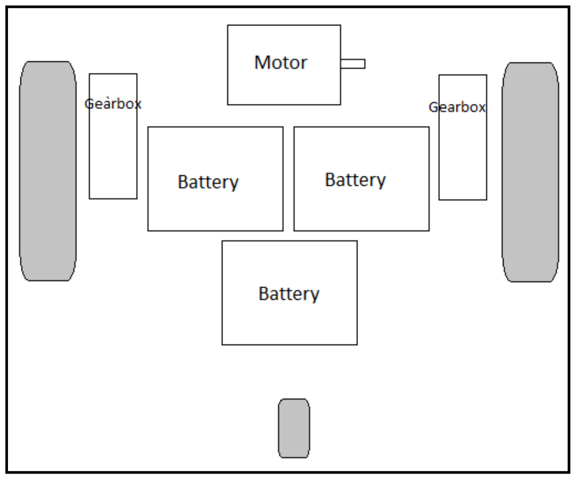

Figure 4.7: Plan view of second design layout

4.4.3 Third design layout

Design 3 has two gearboxes which are located beside each tyre. For this design, writer put electric motor at the center of the transporter body. The design is more compact than design 1 and design 3.

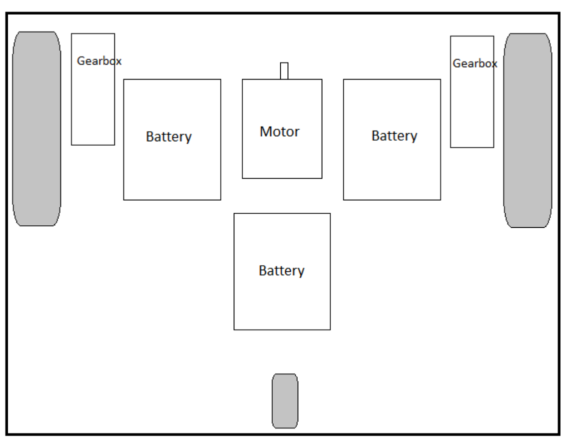

Figure 4.8: Plan view of third design layout

4.4.4 Design Selection

As discussed above, the criteria of the design are safety, cost functionality and maintainability. The sequences of the criteria are arranged in priority manners- safety is the most important and maintainability is the least important.

Safety is the most crucial element because writer does not want the design of the transporter expose hazard to rider. Thus, writer weighted safety as 0.35 for evaluation purposes. Meanwhile for cost and maintainability are weighted as 0.35, 0.30 and 0.15 respectively. Figure below shows the hierarchical objective tree for design selection

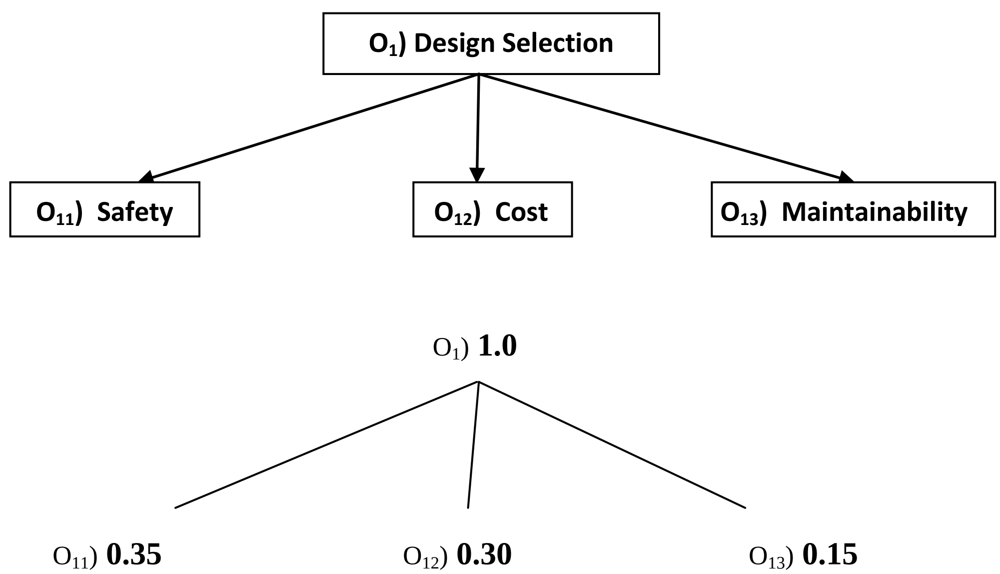

Figure 4.9: Hierarchical objective tree for design selection

4.4.5 Design Score

Weighted decision matrix is a method of evaluating completing concepts by ranking the design criteria with weighting factors and scoring the degree to which each design concept meets the criterion [17]. To determine the score, a 5-point scale is used as the criteria for evaluation is not very detailed [17].

4.4.5.1 Safety

For safety, writer evaluates the score for Design 1 and Design 2 is 5 which are the highest as the surface of the transporter is flat. For design 3, writer rates 3 for the score as it has a bum in the middle of the floor.

4.4.5.2 Cost

Design 1 score 5 for the cost and Design 2 and Design 3 score 4. This is because, Design 2 and Design 3 have two gearboxes compared to design 1 which only has one.

4.4.5.3 Functionality

Writer expected all design to function well. Therefore, all design score 5 for this criteria.

4.4.5.4 Maintainability

Design 1 scores 5 for this criteria as it only has one gear box and the electric motor is mounted in front which provide easy excess for mechanics. As the batteries is arrange close to each other, it also contribute to easy maintenance purposes. Meanwhile, Design 2 scores 4 for the criteria. The only different for Design 2 compared to design 1 is it contains two gear boxes. Lastly, Design 3 scores 3 as it has two gear boxes and the electric motor is in the center of the transporter main body which makes it a bit hard to excess as compared to the first two designs.

Table 4.7: Summary of design evaluation

Consideration	Weight	Design 1st		Design 2nd		Design 3rd	
		Score	Rating	Score	Rating	Score	Rating
Safety	0.35	5	1.75	5	1.75	3	1.05
Cost	0.30	5	1.50	4	1.20	4	1.20
Maintainability	0.15	5	0.75	4	0.6	3	0.45
Total			4.00		3.55		2.70

Form the evaluation Design 1 Score the highest compared to the other two designs. Therefore, design 1 is selected to continue the next design process.

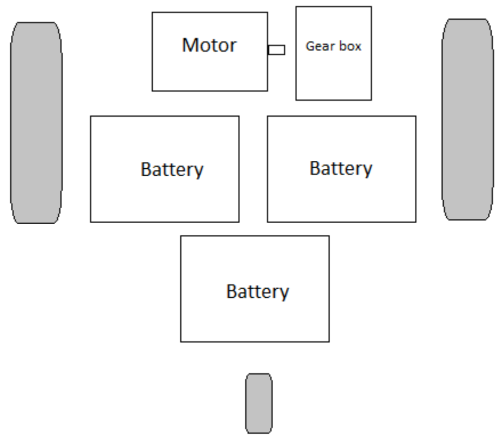

Figure 4.10: Selected design of parts arrangement for the new transporter

4.5 Free-Body Diagram

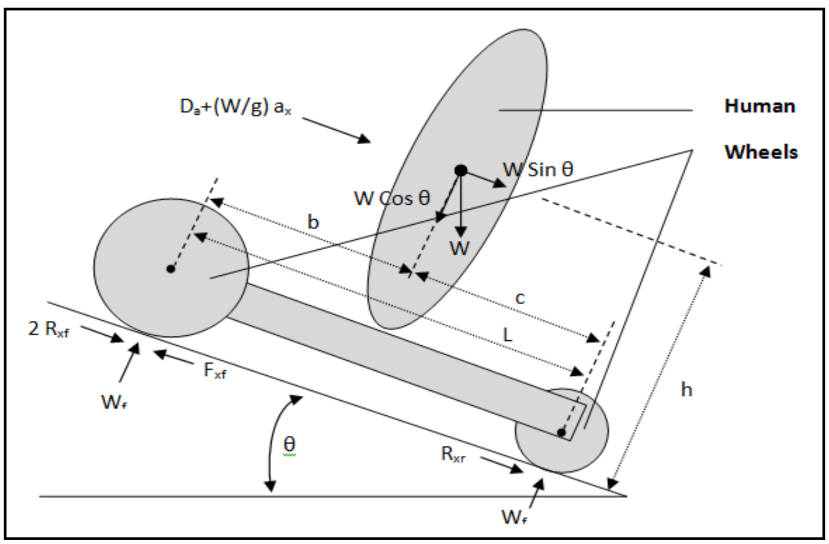

Figure 4.11: Free-body diagram of the transporter from side view

Acting force; Sum. F = 0;

At front wheel, $W_f = [W\ c\ \cos\theta - (Wa_x/g + D_a)\ h - W\ h\ \sin\theta]/L$ (eq. 4.5.1)

At rear wheel, $W_r = [W\ b\ \cos\theta + (Wa_x/g + D_a)\ h + W\ h\ \sin\theta]/L$ (eq. 4.5.2)

4.5.1 Center of gravity for the system

Assumptions: Weight of gear system and transporter body is negligible

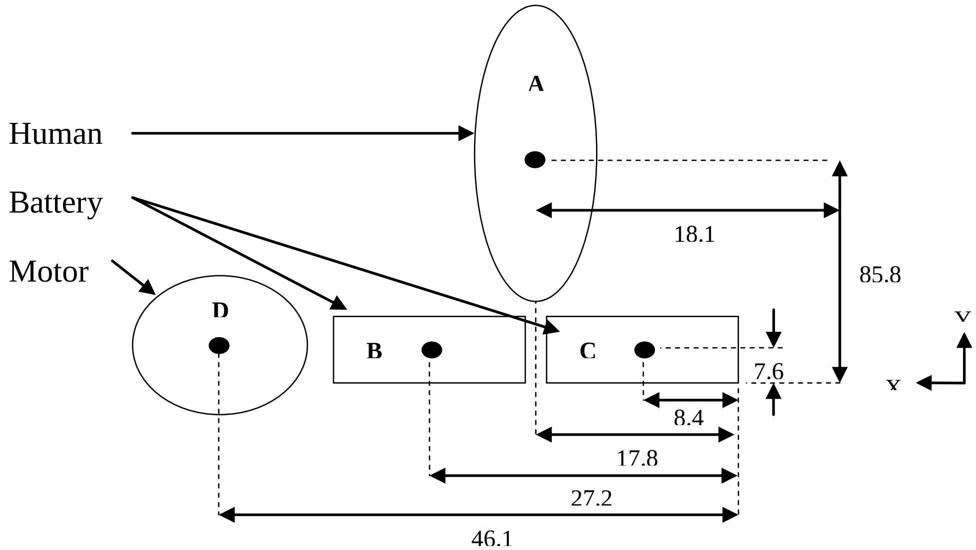

Figure 4.12: Dimension of parts inside the transport (from side view) in centimeter

A: Human (Height 180cm, weight 113kg)

B: two batteries arrange side by side (Weight 5.9kg each)

C: Single battery (Weight 5.9kg)

D: Electric Motor (5.0kg)

Mass = 135 kg

Weight = 1324.35 N

To obtain center of gravity acting the composite body;

$$X = \Sigma\, xW / \Sigma\, W \tag{4.5.3}$$

$$Y = \Sigma\, yW / \Sigma\, W \tag{4.5.4}$$

Substitutes all values into equation (4.5.3) and (4.5.4);

$$X = [(8.4\times5.9)+(17.8\times113)+(27.2\times11.8)+(46.1\times5)] / (5.9+113+5.9+5)$$

$$= 18.1 \text{ cm} \tag{4.5.5}$$

$$Y = [3.8(5.9+5.9+5.9+5) + 113(98)] / (5.9+5.9+5.9+5)$$

$$= 82.2 \tag{4.5.6}$$

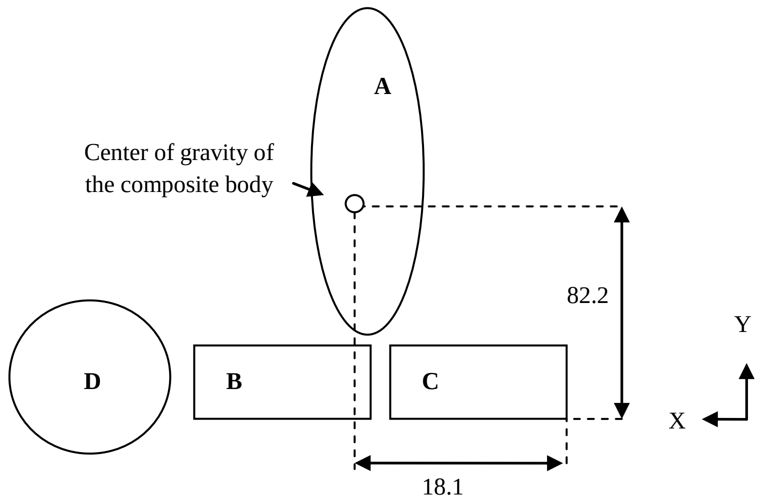

Figure 4.13: Shows location of center of gravity in centimeter

Substitute value in (4.5.5), (4.5.6) and all information in (eq. 4.5.1) and (4.5.2) with θ=0;

Assumption: 1) The transporter is about to move, (Drag force = 0 N)

2) Drag force is acting at center of gravity

At front wheel, $W_f = 293.7$ N

At rear wheel, $W_r = 814.7$ N

4.6 Gear System Design

Gear design is the main parts in designing drive mechanism. The set of gear which is termed as transmission which alters speed and torque delivered by torque-source (electric motor) until the torque reaches the road. The first step in designing gear system is to determine the gear ratio.

4.6.1 Gear ratio

To calculate gear ratio, writer is considering the highest speed that the transporter is designed for. The target is to achieve at least 6.223 m/s which are 70% of Segway PT maximum speed and for distance of 10.08 km above. From Interim of FYP 1, to move 6.223 requires 137 N of traction force at tyre [8]. To calculate torque and speed to be produced with different gear ratio, writer uses equation (3), (4) and (5).

Gearbox efficiency is a much discussed subject, but accurate values are very difficult to determine. Analytical estimates must be confirmed by testing to gain a degree of confidence in the procedure. With good design and manufacturing practice, efficiencies of 99% per mesh and better are possible [9]. For this project, writer makes an assumption that transmission efficiency is 100%.

Torque at tyre (T_{tyre}) = Motor Torque (T_{motor}) × Gear Ratio (GR) (eq. 4.6.1)

Tyre Speed (ω_{tyre}) = Motor Speed (ω_{motor}) / Gear Ratio (GR) (eq. 4.6.2)

Torque (T) = Force (F) × Tyre radius (r) (eq. 4.6.3)

Torque requires at tyre is varies with tyre's size. Small-diameter tyre requires less torque as compared to large-diameter tyre if the product force is constant. To choose suitable gear ratio that leads to targeted speed, iteration of torque, speed and tyre size are summarized in Appendix 1 and appendix 2.Below shows calculation of speed produced by the transporter.

Take tyre radius as 0.2 m, thus based on appendix 2, torque requires at tyre is 27.40 N-m. Then, by referring to appendix 2 we will get gear ratio and also speed produced. Table 4 below shows a fraction from appendix 2 and appendix 3.

Table 4.8: Relationship between Gear ratio, Angular speed and torque outcomes

Gear ratio	**Angular Speed Outcome (Rad/sec)**	**Torque out comes (N.m)**
9.0	34.907	28.647
8.5	36.960	27.0555

By interpolation,

$(28.647 – 27.056)/ (27.40– 27.056) = (34.907-36.960) / (\omega_{T=27.40} – 36.960)$

$$\omega_{T=27.40} = 36.521 \text{ Rad/sec} \qquad (4.6.4)$$

$(9.0-8.5)/ (GR_{T=57.2} – 8.5) = (28.647-27.056) / (27.40-27.056)$

$$GR_{T=57.2}= 8.6 \qquad (4.6.5)$$

Using try and error method, the most suitable combination of gear teeth for 2 stage transmission is;

$$N_{Gear\ A}/ N_{Gear\ B} \times N_{Gear\ C}/ N_{Gear\ D} = (15/45) \times (15/43) = 1/8.6 \qquad (4.6.6)$$

Therefore, GR of 8.6 is needed to produce angular velocity of 36.521 Rad/sec with 0.2m of tyre radius.

Therefore,

Speed of the transporter, v = 0.2 m × 36.521 rad/sec

= **7.30 m/s (>6.223 m/s)** (4.6.7)

From equation (1), battery will last for 1820 sec, thus,

Operation distance, D = 7.30 m/s × 1820 sec

= 13286 m

= **13.3 km (>10.08 km)** (4.6.8)

As mention earlier the required maximum speed is **6.223 m/s** with at least **10.08 km** of operation distance**.** Thus, this proven that the new transporter capable of achieving required performance.

4.6.2 Gear design

Table 4.9: Information of four gears that being used inside the gearbox

Parameter	**Gear A**	**Gear B**	**Gear C**	**Gear D**
RPM, n	3000.000	1000.000	1000.000	348.837
Diametral pitch, P	6.000	6.000	6.000	6.000
Number of teeth, N (eq. 4.6.6)	15.000	45.000	15.000	43.000
Pitch Diameter, d (in)	3.000	9.000	3.000	8.600
Module, m	0.200	0.200	0.200	0.200
Circular pitch, p	0.628	0.628	0.628	0.628
Addendum, a	0.167	0.167	0.167	0.167
Dedendum, b	0.225	0.225	0.225	0.225
Material: Ductile Iron Grade 60, Density; 7.10g/cm^3 (Refer to appendix 4)[14]				

Face width for spur gear should have a face width from 3 to 5 times the circular pitch [13].

($3p \le F \le 5p$)

Take F= 3p = 3(0.628) = 1.884

Lewis form factor, Y (Appendix 6)   Geometry Factor, J(Appendix 7)

$Y_{GEAR\ A} = 0.290$ $\qquad$ $J_{GEAR\ A} = 0.250$

$Y_{GEAR\ B} = 0.400$ $\qquad$ $J_{GEAR\ B} = 0.388$

$Y_{GEAR\ C} = 0.290$ $\qquad$ $J_{GEAR\ C} = 0.250$

$Y_{GEAR\ D} = 0.397$ $\qquad$ $J_{GEAR\ D} = 0.380$

Pitch-line velocity

Equation: $V = (\pi\ d_P\ n_p)/12)$

	Gear A- Gear B	Gear C- Gear D
Pitch-line velocity, V (ft/min)	2356.193	785.398

Transmitted Load

Equation: $W^t = (33000 \times H)/V$, where; H = Horsepower , V = Pitch-line velocity

	Gear A- Gear B	Gear C- Gear D
Transmitted load, W^t (kN)	18.782	56.345

Dynamic Factor

Equation: $K_V = [(A+\sqrt{V})/A]^B$,

Where; A = 50 + 56(1-B), B= $0.25(12-Q_v)^{2/3}$

Quality Number, Q_v = 6 (assumption)

	Gear A- Gear B	Gear C- Gear D
Dynamic Factor, K_V	1.634	1.374

Reliability Factor

Equation: $K_R = 0.5 - 0.109\ln(1-R)$, Where: $0.99 \le R \le 0.9999$

$= 1.002$

Stress cycle factor (Appendix 8)

Bending Stress

Equation: $Y_N = 1.3558\ N^{-0.0178}$

	Gear A	Gear B	Gear C	Gear D
Stress cycle factor, Y_N	0.938	0.956	0.956	0.974

Pitting resistance stress

Equation: $Z_N = 1.4488\ N^{-0.023}$

	Gear A	Gear B	Gear C	Gear D
Stress cycle factor, Z_N	0.900	0.923	0.923	0.945

Size Factor

Equation: $K_s = 1/k_b = 1.192\ (F\sqrt{Y}/P)^{0.0535}$

	Gear A	Gear B	Gear C	Gear D
Size Factor, Ks	1.084	1.093	1.084	1.093

Load -Distribution factor

$K_M = C_{mf} = 1+C_{mc}(C_{pf}\, C_{pm} + C_{ma}\, C_e)$

Where; C_{mc} = 0.8 (crowned teeth)

C_{pf} = (F/10d) – 0.0375+0.0125F (1<F<17in)

(For F/10d <0.05, use F/10d =0.05)

C_{pm} = 1 (Straddle-mounted pinion with $S_1/S<1$)

C_{ma} = $A+BF+CF^2$ (refer to Appendix 8) (eq.4.6.2)

C_e = 1 (Assumed)

	Gear A	Gear B	Gear C	Gear D
C_{pf}	0.049	0.036	0.049	0.036
C_{mc}	0.800	0.800	0.800	0.800
C_{pm}	1.000	1.000	1.000	1.000
C_{ma}	0.157	0.157	0.157	0.157
C_e	1.000	1.000	1.000	1.000
Load -Distribution factor, K_m	**1.164**	**1.154**	**1.164**	**1.154**

Fatigue

Min face width to counter bending fatique,

$F_{bend} = n_d W^t K_o K_v K_s P_d (K_m K_B/J) (K_T K_R/S_t Y_N)$

Min face width to resist wear fatigue,

$F_{wear} = (C_p C_N/S_c K_T K_R)^2 n_d W^t K_o K_v K_s (K_m C_f/d_P I)$

Where;

Where;	n_d	= Design Factor = 1.5
	W^t	= Load transmitted
	K_o	= Overload factor = 1
	K_B	= Rim Thickness Factor = 1
	K_T	= Temperature Factor =1
	K_S	= Size Factor = 1.192 $(F\sqrt{Y}/P)^{0.0535}$
	P_d	= Diametral Pitch
	K_m	= Load-Distribution Factor
	J	= Geometry Factor
	K_R	= Reliability Factor = 0.5 – 0.109ln (1-R) ; $0.99 \leq R \leq 0.9999$
	S_t	= 77.3 H_B + 12800 = 37536psi (Refer to Appendix 10)
	S_c =	(Refer to Appendix 11)
	C_p	= Elastic coefficient = $[1/\pi[(1-v_P^2)/E_P + (1-v_G^2)/E_G]]^{1/2}$ (Refer to Appendix 12)

	Gear A	Gear B	Gear C	Gear D
Transmitted load, kN	18.782	18.782	56.345	56.345
Dynamic Factor, K_V	1.634	1.634	1.374	1.374
Reliability Factor, K_R	1.002	1.002	1.002	1.002
Stress cycle factor, Y_N	0.938	0.956	0.956	0.974
Stress cycle factor, Z_N	0.900	0.923	0.923	0.945
Size Factor, K_S	1.084	1.093	1.084	1.093
Temp. Factor, Kt	1.000	1.000	1.000	1.000
Elastic factor, C_P	2300	2300	2300	2300
F_{bend} (in)	**0.0673**	**0.1016**	**0.5484**	**0.2637**
F_{wear} (in)	**0.0752**	**0.0257**	**0.1924**	**0.0688**

From result for F_{bend} and F_{wear} in table above, the minimum face width to counter bending stress and wear is far smaller than 3p of face width. The widest face from the result above is 0.5485 inch. Thus, to avoid over design, writer make a face width correction to 0.55 inch.

Correct K_s and $K_{m;}$

K_s $= 1.192\ (0.55\sqrt{0.290}/6)^{0.0535}$ $= 1.015$

$F/10d_P = 0.55/10\times6$ $= 0.0917$

C_{pf} $= 0.0917 - 0.0375 + 0.0125\ (0.55)$ $= 0.0473$

K_m $= 1 + 0.8(0.0473\times1 + 1\times0.157)$ $= 1.163$

Bending stress by W^t and AGMA Factor of Safety

Equation: $\sigma = W^t K_o K_v K_s (P_d/F) (K_m K_B/J)$

$S_F = (S_t Y_N / K_T K_R) / \sigma$

	Gear A	Gear B	Gear C	Gear D
Bending Stress, σ (psi)	1676.20	1080.77	4228.28	2783.63
AGMA factor of safety, S_F	20.95	33.14	8.47	13.11

Wear contact stress and AGMA Factor of Safety

Equation: $\sigma_c = C_P \sqrt{[W^t K_o K_v K_s (K_m/d_P F)(C_f/I)]}$

$S_H = (S_c Z_N / K_T K_R) / \sigma_c$

	Gear A	Gear B	Gear C	Gear D
Wear stress, σ_c (psi)	31965.13	18461.42	51063.96	30169.74
Safety factor, S_H	5.06	8.98	3.25	5.63

Rim Thickness

Rim thickness = m_B (addendum + dedendum), where, $m_B \geq 1.2$

= 1.2 (0.167+0.225) = 0.4704 inch

Therefore, rim thickness of 0.5 is used.

4.7 Performance

To evaluate the performance of the drive system, simulation with Matlab is performed. Below is the steps taken to develop the Matlab's program.

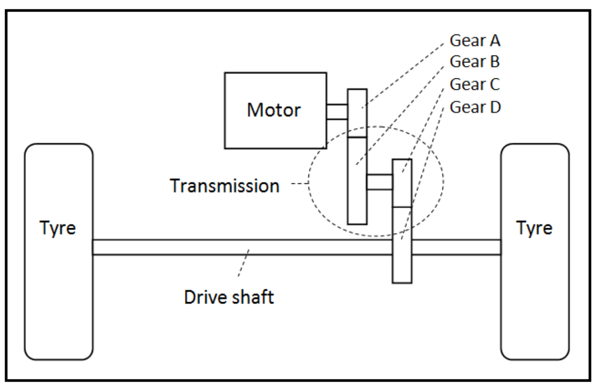

Figure 4.14: Arrangement of gears in the drive mechanism

Where;

$Ŧ_m$	= Motor torque
$Ŧ_t$	= Transmission torque
$Ŧ_{w\&ds}$	= Wheel and driveshaft torque
R	= Radius of respective element
M_w	= Wheel and driveshaft mass
$I_{t,\ A,\ B,C\&D}$	= Moment of inertia for transmission A, B,C,D , Wheel and driveshaft respectively
α_t, α_w	= Angular acceleration of transmission and tyre respectively
N_t	= Gear ratio
t	= Thickness of the gear

Torque required, $\tau_{w\&ds} = \tau_t - (I_w\ \alpha_w) = F_x\, r_w$ (4.7.1)

Where; $\tau_t = \tau_m N_t - (I_t\ \alpha_t)$ (4.7.2)

$I_w = \frac{1}{2}\, m_w^{\ 2}\, r_w$ (4.7.2)

Given, Acceleration of transporter a = 0.94m/s = $\alpha_{\ tyre}$ = 4.7 rad/sec^2

Gear $_{Volume}$ = Thickness × total volume

$t[\pi R_1^{\ 2} - 3(1/3\pi R_2^{\ 2} + 1/3\pi R_3^{\ 2}) + 3lh] = t[\pi(R_1^{\ 2} - R_2^{\ 2} + R_3^{\ 2}) + 3lh]$

<u>Gear A and C</u>

Mass = Density × Gear $_{Volume}$ = 7100 kg/m^3 × [$\pi 0.0381^2 \times 0.01397$-] = 0.4523 kg

Radius, $r_{A\&C}$ =0.038m

<u>Gear B</u>

Mass, m =7100 kg/m^3 × 0.01397{π [0.1143^2- 0.0987^2 + 0.0318^2] + 3(0.0667×0.0254)}
= 1.831 kg

Radius, r_B =0.1145m

<u>Gear D</u>

Mass, m =7100kg/m3×0.01397{π [0.1092^2- 0.0937^2 + 0.0318^2] + 3(0.0619×0.0254)}
= 1.75 kg

Radius, r_D =0.109m

<u>Tyres</u>

Mass, m=1.5kg/each = 3kg , Radius, r= 0.2m

<u>Shaft</u>

Mass, m =1.5kg , Radius, r =0.01m

4.7.1 Matlab Program

Below is the program used to simulate the performance of the drive system. In the program, mass of small shaft between gear B and gear C is assumed to be zero.

```
%1,2,3,4,5,6 are tyre, shaft, GearD, GearC, GearB and GearA
%respectively
%mass of elements (kg)
m1=3,m2=1.5,m3=1.75,m4=0.4523,m5=1.831,m6=0.4523
%radius of elements(m)
r1=0.2,r2=0.01,r3=0.109,r4=0.038,r5=0.1145,r6=0.038;
%moment of inertia of tyre
I1=0.5*(m1^2)*r1;
%moment of inertia of shaft
I2=0.5*(m2^2)*r2;
%moment of inertia of gear D
I3=0.5*(m3^2)*r3;
%moment of inertia of gear A and gear C
I4=0.5*(m4^2)*r4;
%moment of inertia of Gear B
I5=0.5*(m5^2)*r5;
I6=I4;
%Angular acceleration(rad/sec2)
GR1=15/43,GR2=15/45
a1=4.7
a2=a1;
a3=a2;
a4=a3/GR1;
a5=a4;
a6=a5/GR2;

%torque
Tm=3.183:31.83
T1=Tm*8.6-(I6*a6)-(I5*a5)-(I4*a4)-(I3*a3)-(I2*a2)-(I1*a1);

%speed V=transporter velocity(m/s), v=angular velocity of the
tyre(rad/sec)
vm=1:314.06;
V=(vm/8.6)*0.2

%Plot the graph
plot(vm,V)
plot(Tm,T1)
```

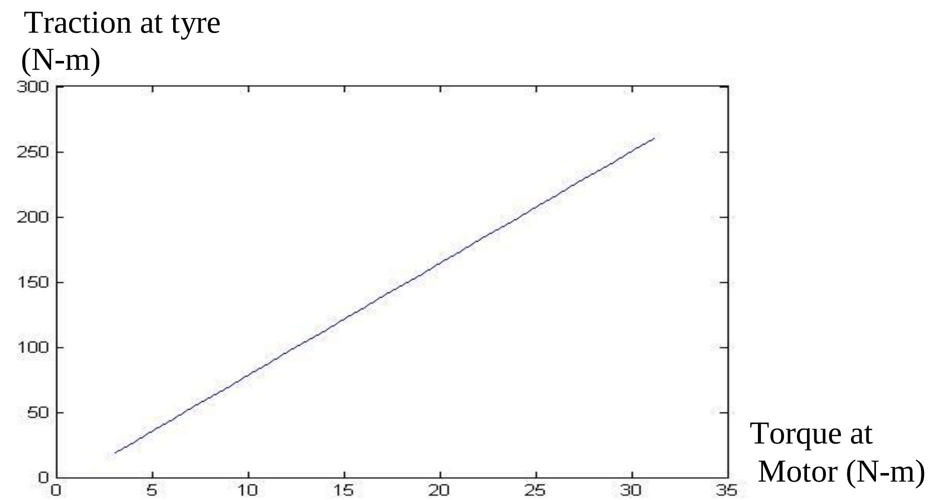

Figure 4.15: Motor Torque versus Tyre torque

Traction at tyres is directly proportional with torque produced by motor. As discussed earlier in section (4.3.2), motor torque in other hands, is inversely proportional with motor angular speed. The highest torque is stall torque which is 31.83.Refering to figure 4.15, at 30 N-m, torque produced is 250 N-m which higher than the requirement torque for the transporter to move (42 N-m, refer to appendix 1). This means, the actual acceleration for the transporter is higher 0.94m/s.

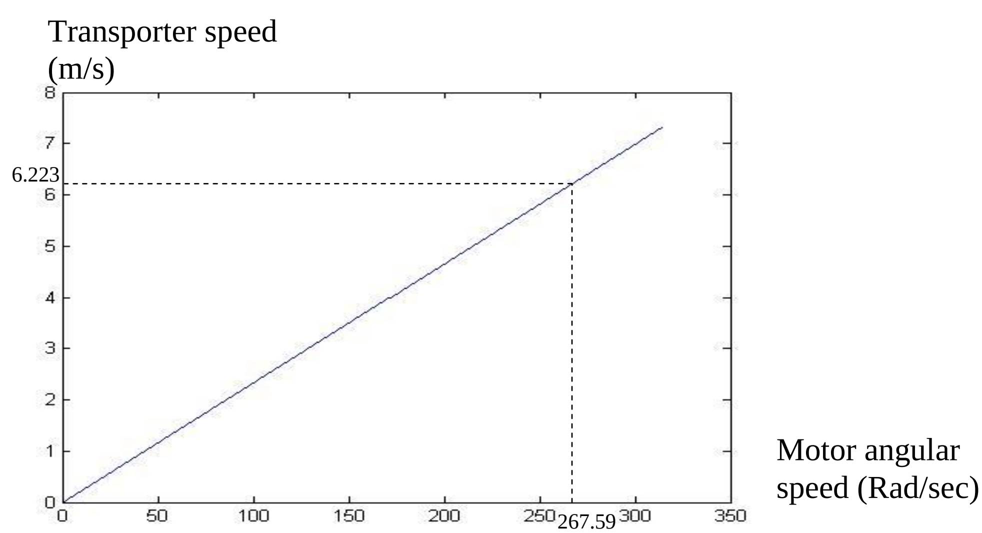

Figure 4.16: Motor Torque versus Motor angular speed

Transporter's speed is directly proportional with the motor's angular speed. The maximum speed produced by motor is at its free rotational which is 314.16 rad/sec. To achieve 6.223m/s the motor must rotates at 267.589 rad/sec. By referring to figure 4.5, at 267.589 rad/sec, torque delivered is 5.19 N-m which produces 34.30 N-m traction at tyres. From equation 4.7.1, traction force is 171.5 N, much higher than required force which is 133N (refer to fig. 4.1)

4.7.2 Reliability

The material is tested at10^7 cycles with 0.99 reliability (appendix 11)

(10^7)/2555rpm = 39,134.35 min = 2 348 061 sec

With speed of 6.223 m/s, Distance travel equal to;

= 6.223 x 2000000 = 14 611 983 m

= **14 612 km (predicted distance travelled before gear failure)**

4.8 Technical Drawing

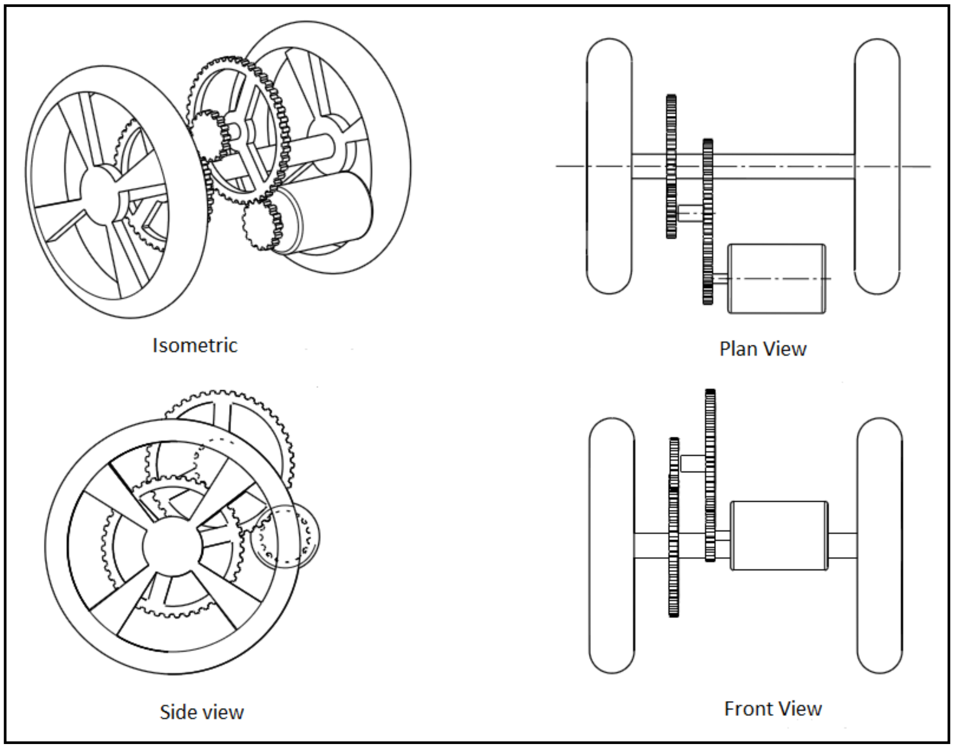

Figure 4.17: Isometric view of assembled parts in the transporter

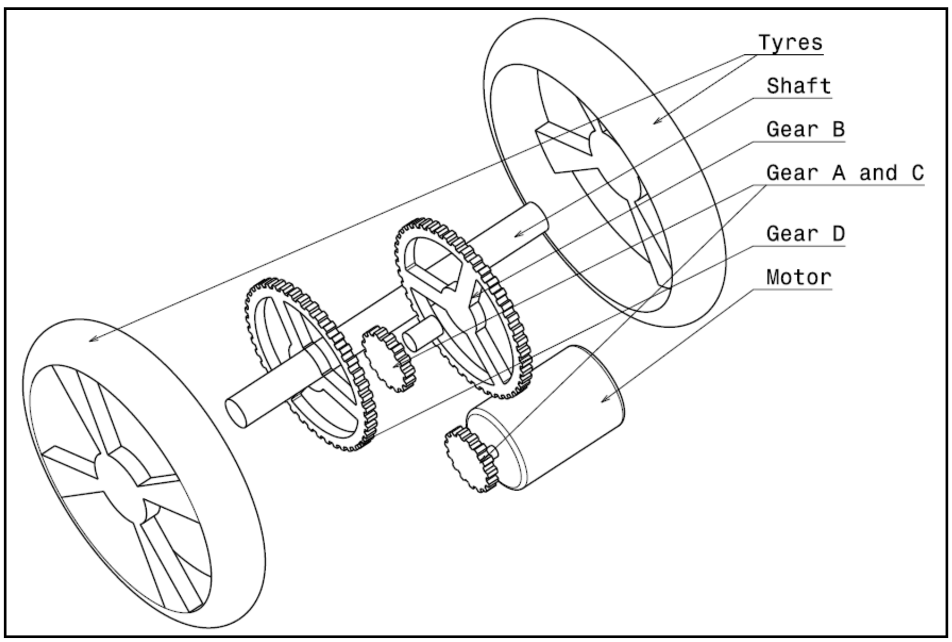

Figure 4.18: Boom drawing of the transporter

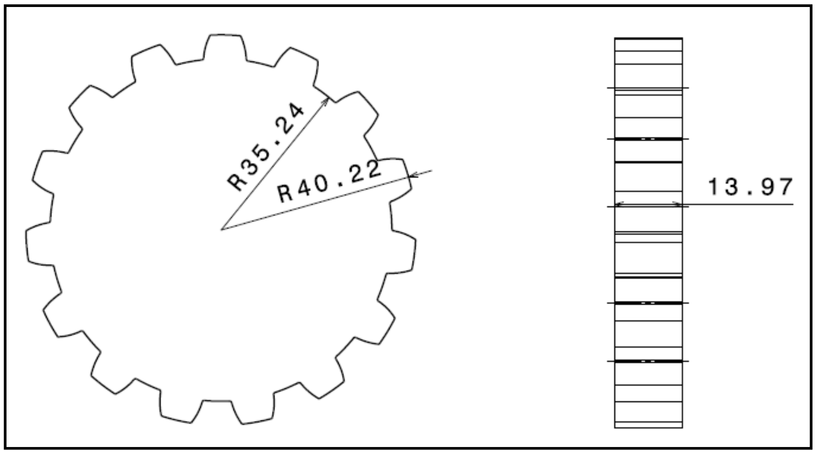

Figure 4.18: Gear A and C with dimension in millimeter

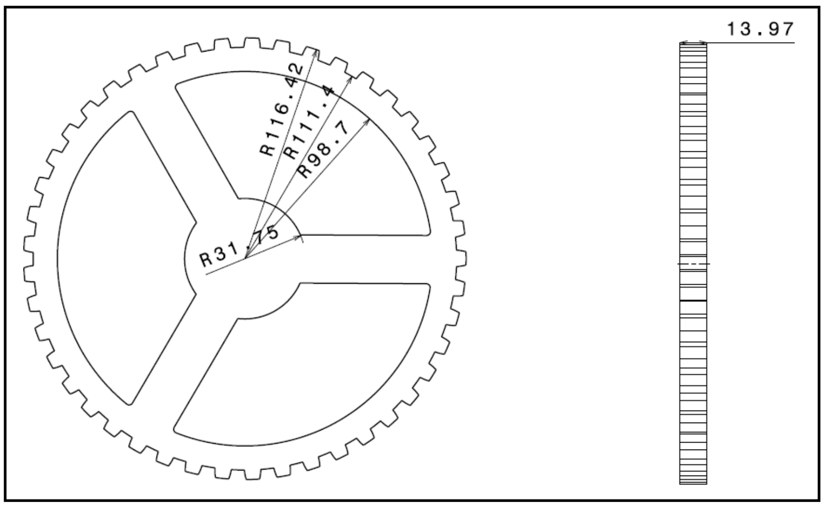

Figure 4.19: Gear B with dimension in millimeter

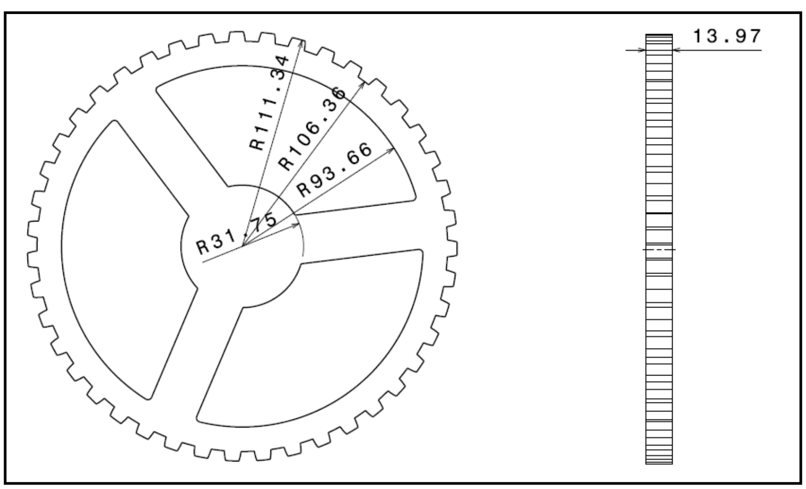

Figure 4.20: Gear D with dimension in millimeter

4.9 Extra function- driven conventional bicycle

In order to be competitive, the new transporter has to be unique. Therefore, the writer improvises the drive mechanism in such way that it also can be used to drive conventional bicycle. The concept is still in early stage of development. Writer confident about its potential as there are abundant of bicycle user in this country. Below is brief schematic of the attachment of the drive mechanism parts on conventional bicycle.

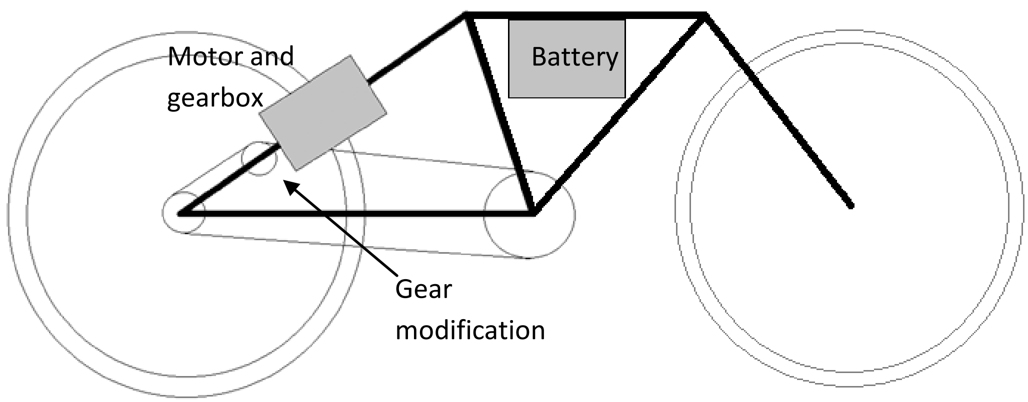

Figure 4.21: Suggested unique function for the design concept of the transporter

The concept is the drive mechanism is designed in such way that it also can be attached to a bicycle with little modification. The advantages of this design are;

1) User can enjoy both PT and electric-power bicycle in one product.
2) It promotes more people to use electric vehicle and cut down carbon emission.
3) Safe cost.

Chapter 5
RECOMMENDATION & CONCLUSION

5.1 Recommendation

1) Do an experiment

To get the real performance for the transporter, several experiments need to be performed. Result from computerize and analytical analysis may not product the exact or actual performance of the transporter. By doing an experiment, the uncertainty that not being considered in this project may emerge. The suggested experiments by writer are;

a) Performance of battery and electric motor
   - The purpose of the experiment is to verify the real performance of the electrical system. The experiment may also include difference type of battery and electric motor as variables in order to increase quality or the study.

b) Cyclic loading on moving parts (especially gears)
   - Failure due to cyclic loading can be predicted on paper. In order to verify the predicted result, the best method is to do an experiment. UTP has few types of equipment in material lab that suitable for this test. The test may be destructive or non-destructive test.

2) Perform material selection

Weight of the parts affects the transporter performance. Lighter material is more preferable as it may increase the efficiency of power transmission from electric motor to tyres. The selected material must meet the requirements of it function for example; material of gear must be able to withstand cyclic loading and high stress. In this project, writer does not performed material analysis. There might be more suitable material that can be selected in order to perform any specific task presented in this project. This is one important area for next researcher to look at in order to make the transporter's performance as efficient as possible.

3) Manufacturing planning

One of the main targets in this project is to reduce cost of producing the transporter. In that sense, the right selection on manufacturing procedures may results in different production cost. The cost can be amplified if it is a mass production. Therefore, this is also one of the areas that should be taking into account for continuity of this project.

4) Details study on battery

The performance of drive system is closely related to the durability of its energy source (battery). Good battery makes the transporter's operation longer which is desired criteria for this machine. The batteries also, consume space the most as compared to other parts in the transporter. Therefore, study on battery is crucial in this project. Next researcher on this topic should look more detail on battery selection or perhaps battery design.

5) Perform reverse engineering in existing transporter

Most of the transporter's manufacturer does not reveal their technology to public. In other word they are using "black box" technology. As a new comer in the industry we have so much more to learn. This process can be fasten is we can learn from others experience. To do so, in the sense of producing drive mechanism, writer suggests reverse engineer from existing transporter is performed. By doing this, we can have a bigger picture on how personal transporter manufacturer design their transporter which might be useful for us to design ours.

6) Add additional value of to the transporter

To be competitive, our product must have unique factor (X-factor) form others. This is important because rule of thumb states that consumer/user only prefer the best option for their usage. In this project, writer had given a brief idea on how X-factor can be adopted in the design which the parts in the drive mechanism can serve a bicycle with little modification. Therefore, next designer/researcher should come with another unique factor in their design which makes them different with others.

5.2 Conclusion

Based on current issue and people life styles nowadays, PERSONAL TRANPORTER is predicted having a high prospect in the future. Any early step taken by our local designer in order to produce the transporter is considered a good step. There are many possible outcomes in designing the transporter. The outcomes only can be determine after the whole design process is been performed. The methodology used in producing drive mechanism in the project is more toward analytical analysis which must be continue with testing before a real drive mechanism can be produced.

Gear ratio is an important variable in designing drive mechanism. In this project gear ratio is taken as 5:43 or 1:8.6. Other than gears, another important element in drive mechanism is its electric motor. The specification of electric motor is closely related to the gear ratio selection. Besides that, arrangement of parts inside the transporter is also an important issue to be looking at. Good arrangement will leads to better functionality, maintainability, reduce cost and also consume less space. In order to produce prototype of the drive mechanism, lots of engineering works need to be done e.g.: material selection, manufacturing method, durability analysis and so on.

The drive mechanism needs to undergo performance experiment as the analytical calculation before is not considering efficiency. There are no exact values for efficiency that can be referred to in any engineering book [9]. To obtain the real performance of the drive mechanism, it requires experiment. The X-factor provided in the design may increase the value of the transporter itself. Most of the space in the transporter main frame consists of batteries. If the three batteries can be united to be one, writer expected the consumption of space can be reduce significantly. Therefore, writer suggests the next work can also focus on producing a proper battery for the transporter.

Overall, the project achieves its objective which is producing the design of Drive Mechanism for Personal Transporter. Writer satisfies with the achievement of the project even though the project only covers a part of drive mechanism design. There is still few area of improvement for writer to improve personally as well as the project management itself.

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APPENDIXES

Velocity, v (m/s)	Angular velocity (Rad/s)	RPM	Air density, ρ (kg/m3)	Gravity, G (m/s2)	Drag Coefficient, CD	Acceleration	Body + Segway area (m)	Total Mass (kg)	Human Mass (kg)	Rolling Coefficient, CR	Inclination Degree(°)	Inclination Radian	Air resistance (N)	Rolling resistance (N)	Inclination Force (N)	Acceleration force (N)	Total force (N)	Tyre torque (N.m)	Power (W)
0.0	0	0	1.225	9.81	1.5	0.93	1.6	165	113	0.01360	0	0.0000	0	22	0	153	175	42	0
0.5	2	20	1.225	9.81	1.5	0.88	1.6	165	113	0.01360	1	0.0175	0	22	28	146	168	40	84
1.0	4	40	1.225	9.81	1.5	0.84	1.6	165	113	0.01360	2	0.0349	1	22	56	138	162	39	162
1.5	6	60	1.225	9.81	1.5	0.79	1.6	165	113	0.01360	3	0.0524	3	22	85	130	156	37	234
2.0	8	80	1.225	9.81	1.5	0.74	1.6	165	113	0.01360	4	0.0698	6	22	113	123	151	36	301
2.5	10	99	1.225	9.81	1.5	0.70	1.6	165	113	0.01360	5	0.0873	9	22	141	115	146	35	366
3.0	13	119	1.225	9.81	1.5	0.65	1.6	165	113	0.01360	6	0.1047	13	22	169	107	143	34	428
3.5	15	139	1.225	9.81	1.5	0.60	1.6	165	113	0.01361	7	0.1222	18	22	197	100	140	34	489
4.0	17	159	1.225	9.81	1.5	0.56	1.6	165	113	0.01361	8	0.1396	24	22	225	92	138	33	550
4.5	19	179	1.225	9.81	1.5	0.51	1.6	165	113	0.01361	9	0.1571	30	22	253	84	136	33	613
5.0	21	199	1.225	9.81	1.5	0.46	1.6	165	113	0.01361	10	0.1745	37	22	281	77	136	33	678
5.5	23	219	1.225	9.81	1.5	0.42	1.6	165	113	0.01362	11	0.1920	44	22	309	69	136	33	746
6.0	25	239	1.225	9.81	1.5	0.37	1.6	165	113	0.01362	12	0.2094	53	22	337	61	136	33	818
6.5	27	259	1.225	9.81	1.5	0.33	1.6	165	113	0.01362	13	0.2269	62	22	364	54	138	33	896
7.0	29	279	1.225	9.81	1.5	0.28	1.6	165	113	0.01363	14	0.2443	72	22	392	46	140	34	981
7.5	31	298	1.225	9.81	1.5	0.23	1.6	165	113	0.01363	15	0.2618	83	22	419	38	143	34	1073
8.0	33	318	1.225	9.81	1.5	0.19	1.6	165	113	0.01363	16	0.2793	94	22	446	31	147	35	1175
8.5	35	338	1.225	9.81	1.5	0.14	1.6	165	113	0.01364	17	0.2967	106	22	473	23	151	36	1286
9.0	38	358	1.225	9.81	1.5	0.09	1.6	165	113	0.01364	18	0.3142	119	22	500	15	156	38	1408
9.5	40	378	1.225	9.81	1.5	0.05	1.6	165	113	0.01365	19	0.3316	133	22	527	8	162	39	1543
10.0	42	398	1.225	9.81	1.5	0.00	1.6	165	113	0.01365	20	0.3491	147	22	554	0	169	41	1691

Appendix 1: Force variation in difference transporter velocity

Tyre Radius (m)	Torque (N-m)
0.05	6.85
0.10	13.70
0.15	20.55
0.20	27.40
0.25	34.25
0.30	41.10
0.35	47.95
0.40	54.80
0.45	61.65
0.50	68.50

Appendix 2: Traction torque need for specific tyre size

Gear ratio	Angular Speed Outcome (Rad/sec)	Torque out comes (N.m)
16.0	19.635	50.928
15.5	20.268	49.3365
15.0	20.944	47.745
14.5	21.666	46.1535
14.0	22.440	44.562
13.5	23.271	42.9705
13.0	24.166	41.379
12.5	25.133	39.7875
12.0	26.180	38.196
11.5	27.318	36.6045
11.0	28.560	35.013
10.5	29.920	33.4215
10.0	31.416	31.83
9.5	33.069	30.2385
9.0	34.907	28.647
8.5	36.960	27.0555
8.0	39.270	25.464
7.5	41.888	23.8725
7.0	44.880	22.281
6.5	48.332	20.6895
6.0	52.360	19.098
5.5	57.120	17.5065
5.0	62.832	15.915

Appendix 3: Relation between torque and gear ratio needed to produce the required torque

Table 14–4

Repeatedly Applied Bending Strength S_t for Iron and Bronze Gears at 10^7 Cycles and 0.99 Reliability

Source: ANSI/AGMA 2001-D04.

Material	Material Designation[1]	Heat Treatment	Typical Minimum Surface Hardness[2]	Allowable Bending Stress Number, S_t,[3] psi
ASTM A48 gray cast iron	Class 20	As cast	—	5000
	Class 30	As cast	174 HB	8500
	Class 40	As cast	201 HB	13 000
ASTM A536 ductile (nodular) Iron	Grade 60–40–18	Annealed	140 HB	22 000–33 000
	Grade 80–55–06	Quenched and tempered	179 HB	22 000–33 000
	Grade 100–70–03	Quenched and tempered	229 HB	27 000–40 000
	Grade 120–90–02	Quenched and tempered	269 HB	31 000–44 000
Bronze		Sand cast	Minimum tensile strength 40 000 psi	5700
	ASTM B–148 Alloy 954	Heat treated	Minimum tensile strength 90 000 psi	23 600

Appendix 4: Repeatedly Applied Banding Strength S_t for Iron and Bronze Gears at 10^7 Cycles and 0.99 Reliability

Material	Density g/cm^3	Density $lb_m/in.^3$
Stainless alloy 405	7.80	0.282
Stainless alloy 440A	7.80	0.282
Stainless alloy 17-7PH	7.65	0.276
Cast Irons		
Gray irons		
• Grade G1800	7.30	0.264
• Grade G3000	7.30	0.264
• Grade G4000	7.30	0.264
Ductile irons		
• Grade 60-40-18	7.10	0.256
• Grade 80-55-06	7.10	0.256
• Grade 120-90-02	7.10	0.256

Appendix 5: Density of several materials

Table 14–2

Values of the Lewis Form Factor *Y* (These Values Are for a Normal Pressure Angle of 20°, Full-Depth Teeth, and a Diametral Pitch of Unity in the Plane of Rotation)

Number of Teeth	Y	Number of Teeth	Y
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

Appendix 6: Values of the lewis From Factor Y

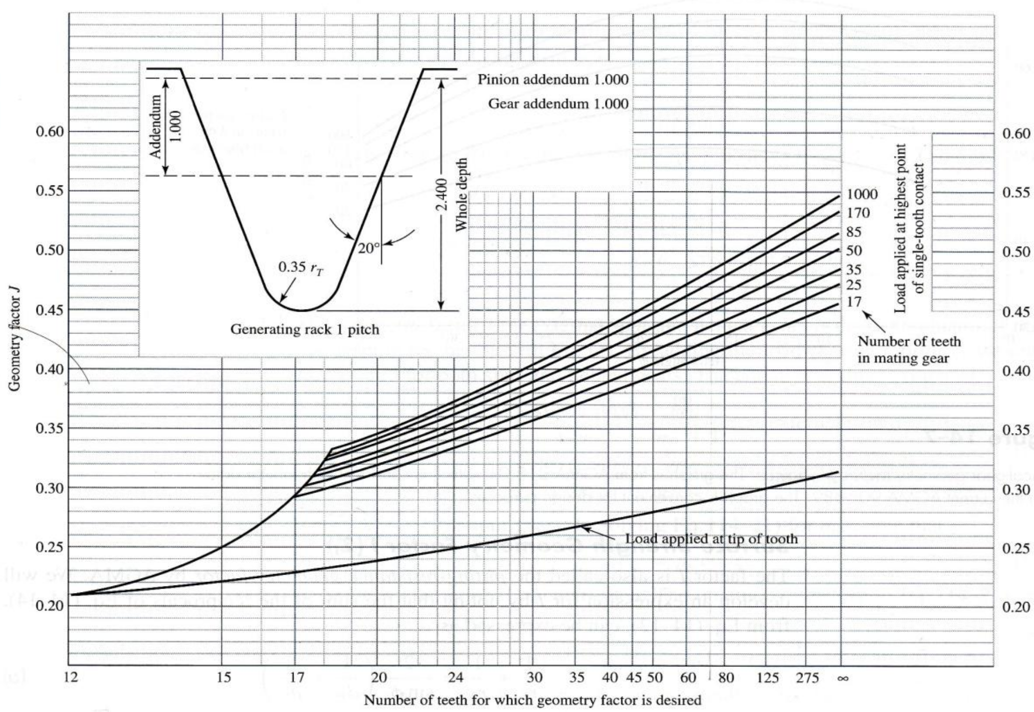


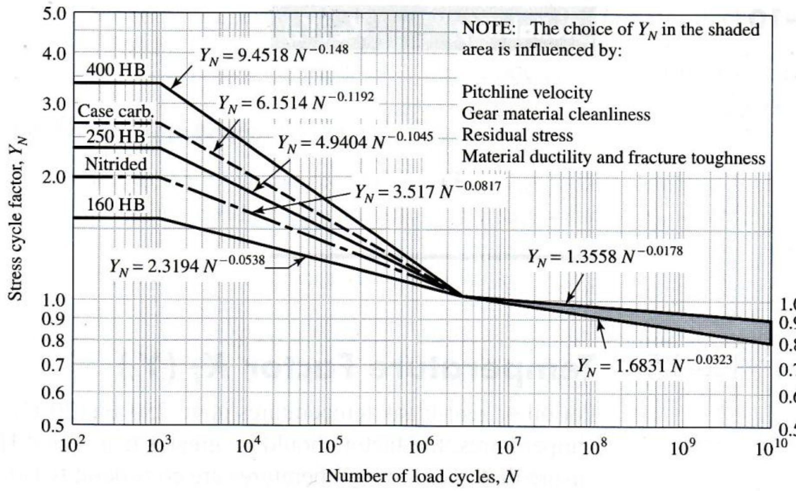
Figure 14–6

Spur-gear geometry factors *J*. *Source:* The graph is from AGMA 218.01, which is consistent with tabular data from the current AGMA 908-B89. The graph is convenient for design purposes.

Appendix 7: Spur-gear geometry factors J

Figure 14–14

Repeatedly applied bending strength stress-cycle factor Y_N. (ANSI/AGMA 2001-D04.)



Appendix 8: Repeated applied bending strength stress-cycle factor Y_N

Table 14–9

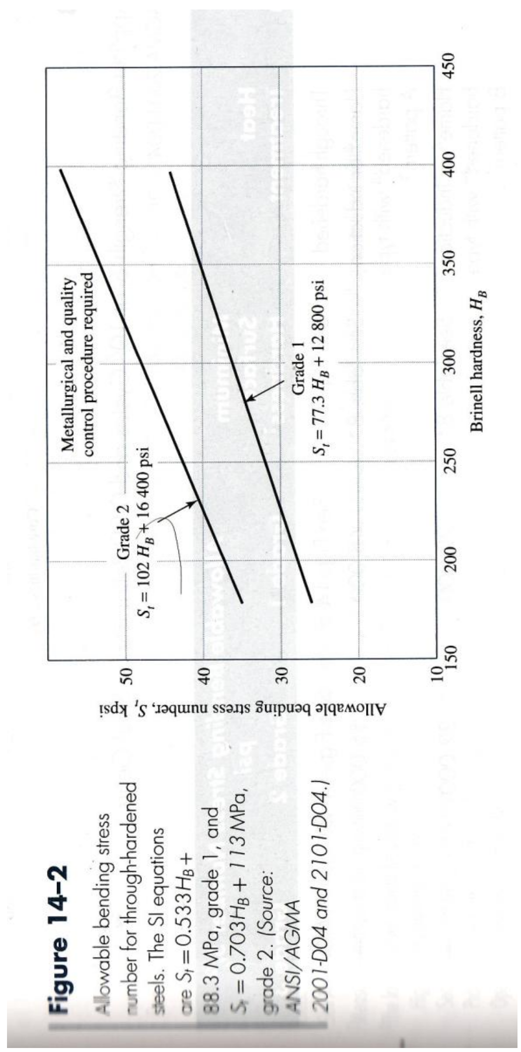
Empirical Constants *A*, *B*, and *C* for Eq. (14–34), Face Width *F* in Inches*

Source: ANSI/AGMA 2001-*D04*.

Condition	*A*	*B*	*C*
Open gearing	0.247	0.0167	$-0.765(10^{-4})$
Commercial, enclosed units	0.127	0.0158	$-0.930(10^{-4})$
Precision, enclosed units	0.0675	0.0128	$-0.926(10^{-4})$
Extraprecision enclosed gear units	0.00360	0.0102	$-0.822(10^{-4})$

*See ANSI/AGMA 2101-D04, pp. 20–22, for SI formulation.

Appendix 9: Empirical Constants A, B and C for Eq. 4.6.2



Appendix 10: Allowable bending stress number for through-hardened steels

Table 14–7

Repeatedly Applied Contact Strength S_c 10^7 Cycles and 0.99 Reliability for Iron and Bronze Gears

Source: ANSI/AGMA 2001-D04.

Material	Material Designation[1]	Heat Treatment	Typical Minimum Surface Hardness[2]	Allowable Contact Stress Number,[3] S_c, psi
ASTM A48 gray cast iron	Class 20	As cast	—	50 000–60 000
	Class 30	As cast	174 HB	65 000–75 000
	Class 40	As cast	201 HB	75 000–85 000
ASTM A536 ductile (nodular) iron	Grade 60–40–18	Annealed	140 HB	77 000–92 000
	Grade 80–55–06	Quenched and tempered	179 HB	77 000–92 000
	Grade 100–70–03	Quenched and tempered	229 HB	92 000–112 000
	Grade 120–90–02	Quenched and tempered	269 HB	103 000–126 000
Bronze	—	Sand cast	Minimum tensile strength 40 000 psi	30 000
	ASTM B-148 Alloy 954	Heat treated	Minimum tensile strength 90 000 psi	65 000

Appendix 11: Repeatedly Applied Contact Strength S_C

Table 14–8

Elastic Coefficient C_p (Z_E), $\sqrt{\text{psi}}$ ($\sqrt{\text{MPa}}$) *Source:* AGMA 218.01

Pinion Material	Pinion Modulus of Elasticity E_p psi (MPa)*	Gear Material and Modulus of Elasticity E_G, lbf/in^2 (MPa)*: Steel 30×10^6 (2×10^5)	Malleable Iron 25×10^6 (1.7×10^5)	Nodular Iron 24×10^6 (1.7×10^5)	Cast Iron 22×10^6 (1.5×10^5)	Aluminum Bronze 17.5×10^6 (1.2×10^5)	Tin Bronze 16×10^6 (1.1×10^5)
Steel	30×10^6 (2×10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162)	1900 (158)
Malleable iron	25×10^6 (1.7×10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158)	1850 (154)
Nodular iron	24×10^6 (1.7×10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156)	1830 (152)
Cast iron	22×10^6 (1.5×10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154)	1800 (149)
Aluminum bronze	17.5×10^6 (1.2×10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145)	1700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141)	1650 (137)

Poisson's ratio = 0.30.

*When more exact values for modulus of elasticity are obtained from roller contact tests, they may be used.

Appendix 12: Gear material and modulus of elasticity