**Design and Analysis of a Drive Train System**
**for a Small Race Car**

by

Che Wan Mohd Hafizul Azizi Bin Wan Ismail

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

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Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

# CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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(MECHANICAL ENGINEERING)

Approved by,

Mohd Syaifuddin B Mohd
Project Supervisor

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

January 2008

# 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.

CHE WAN MOHD HAFIZUL AZIZI BIN WAN ISMAIL

# ABSTRACT

The project is a design and development project of a drive train system for a Formula-style race car. The main objective is to replace the current Universiti Teknologi PETRONAS FSAE (Formula Society of Automotive Engineers) car chain drive system to the gear drive system. The project was conducted with the target of designing the systems for entering a Formula SAE competition and the design has to abide with Formula SAE rules and regulations. This report focused on the design and analysis processes of the drive train system using relevant vehicle dynamics theories and formulas and heavily utilized computer aided engineering tools such as CATIA. No modifications were allowed to the engine as well as the transmission (gear box) due to limitations on the drive train. The design of the drive train was based on gear sizing calculations as part of the car performance analysis. The results include the determination of final drive ratio, maximum velocity, and idle changing gear speed and also the reliability of the new drive train system, which is the gear drive system. The assumptions for certain parameters such as coefficient of drag, frontal area, and overall mass of the car was based on similar existing Formula SAE race cars in order to begin with proper design calculations and performance predictions for the designed race car. The designs of the drive train component including engine drive shaft, bevel gear (ring and pinion), differential housing and bearing based on the performance requirements and analysis. Finally, the report elaborates on the positive benefits and gains from the project and some recommendations on potential improvement and possible new designs

# ACKNOWLEDGEMENT

"I would like to take this opportunity to express my highest appreciation to those who had contributed toward the success of my final year project. To Mr. Mohd Syaifuddin Bin Mohd, thank you for your guidance, supports and suggestion throughout this project. With your supervision, I managed to start my project with proper planning and proceeded until the completion of the project in the stipulated timeframe. Also, I would like to express my appreciation and special thanks to FSAE UTP Teams members for their cooperation throughout the achievement of this project. My warmest gratitude goes to the Mechanical Engineering Department of Universiti Teknologi PETRONAS (UTP) for providing the chance to undertake this final year project. My knowledge on mechanical engineering has been challenged for the purpose of completing this project. A word of heartfelt gratitude goes to the Final Year Project committee for arranging and managing the course throughout the two semesters. Last but not least, I would like to thank my entire colleagues and individuals whose names were not mentioned but had willingly lent me their hands and shared their knowledge towards the accomplishment of this project".

Che Wan Mohd Hafizul Azizi Bin Wan Ismail

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# ABBREVIATIONS AND NOMENCLATURES

| | |
|---|---|
| SAE | Society of Automotive Engineer |
| TE | Tractive Effort |
| TR | Tractive Resistance |
| FBD | Free Body Diagram |
| SF | Safety Factor |
| UTP | Universiti Teknologi Petronas |
| FDR | Final Drive Ratio |
| T | Torque |
| P | Power |
| $\tau_{allow}$ | Allowable Shear Stress |

# CHAPTER 1
# INTRODUCTION

## 1.1 Background of Study

The Formula SAE competition is for engineering students to conceive, design, fabricate and compete with small Formula style racing cars. The restrictions on the chassis and engine are limited so that the knowledge, creativity and imagination of the students are challenged. The competitions themselves give teams the chance to demonstrate and prove both their creation and their engineering skills in comparison to teams from other universities around the world According to Formula SAE® specification, a car to be develop from scratch and a group of students are formed, whereby each of them in charge of different part, such as drive train, engine intake and exhaust manifold, driver interface, suspension, braking, steering and body design.

Drive train involves the mechanism design to deliver engine power and torque from the engine to the wheels. The engine and transmission come in a package, so the gearbox setting is not changed. So the drive train design only starts from the transmission output. Current drive train components include driving sprocket, chain, driven sprocket, drive shafts that transfer the power to the wheels. The detail of drive train design includes uses of various sprockets and chain drives, differential unit, constant velocity or universal joints

## 1.2 Problem Statement

According to Formula SAE rules and regulations, the drive train should be design from scratch. This project is an improvement project of the previous FYP students and also a parallel project with UTP FSAE team. Implementation of chain drive as a torque and power transmission has some disadvantages especially the power loss during the power distribution from the engine to the wheel. The author has looked into the issue and come out with a proposal of replacing the drive chain system with a gear system which is the bevel gear system.

The author has to develop a new drive train mechanism which is using a gear system (bevel gear) to transfer power and torque from the engine to the wheels. It is believe that the use of gear system in transferring the torque and power to the wheel will minimize the amount of power loss instead of drive chain system. Hence, this new system is much safer and easy to maintain. Consideration of cost, weight reduction, performance and durability will take into account when designing the drive train mechanism.

## 1.3 Objectives and Scope of Study

The core objectives of the project are to design (redesign) and analyze the drive train system for a Formula SAE car. This includes determining the optimal drive ratio and designs the drive train mechanism from engine to joints. The main tasks in this project are:

- To design a new drive train system using gear system.
- To perform analysis on the design :
    - i. Calculate the Vehicle Performance
    - ii. Design the related new drive components such as drive shaft, differential (if required), and gears.
    - iii. Perform stress analysis on the drive train components
    - iv. Optimized the components designs and optimize designs to reduce weight

# CHAPTER 2
# LITERATURE REVIEW

## 2.1 Drive Train

Drive train system is a system is a system that transmits input from the prime mover or engine to produce useful output in power or torque. Fundamentally, the drive train serves two functions, where it transmits power from the engine to the drive wheels and varies the amount of torque. "Power" is the rate or speed at which work is performed. "Torque" is turning or twisting force. Multiple ratio gearboxes are necessary because the engine delivers its maximum power at certain speeds, or RPM (Rotations per Minute). For a racing car, the drive train shall be simple and easy to maintain. Drive train consist of engine drive shaft, power transferring mechanism (gears), differential, lubrication system, spindles and bearings. Speed reduction from the engine (input) is controlled by the final drive ratio (FDR) which will be elaborated more in Section 2.4.2.

Connection of the drive train and the engine shall be flexible in order to allow relative movement between associated parts without imposing stresses on individual components. An additional requirement would be to maintain Tripod constant velocity of rotating parts regardless of component position. In rear and four wheels drive the driveshaft transfers the engine output torque to the rear drive axle differential input. Due to relative lateral displacement between the two shafts, there needs to have a flexible connection between them in order to maintain the velocity of input and output. Since the centre distance may also vary, the driveshaft must cater for such situations which may be utilizing flexible mountings or a sliding spline arrangement.

Differential serve several purposes, contains final reduction gearing in the final drive of the wheels, splits the torque to each wheel, maintains equal torque to each wheel independent of small differences in rotational speeds of the wheels when going round corners.

## 2.2 Overview on the Existing Engine and Drive Train Specification

For this project, the engine used is a Honda engine which is Honda CBR 600 F4i. The engine has been use for the project from the previous FYP student. Following are the extracted data from the given engine specifications and to be used in the calculation. (Refer to Appendix 1)

- Primary reduction (1.822) and gear ratio are to be used and cannot be modified since the design racing car will be using original gear box.
- Final Drive Ratio (which is chain drive) can be manipulated to obtain necessary gear ratio to achieve target performance

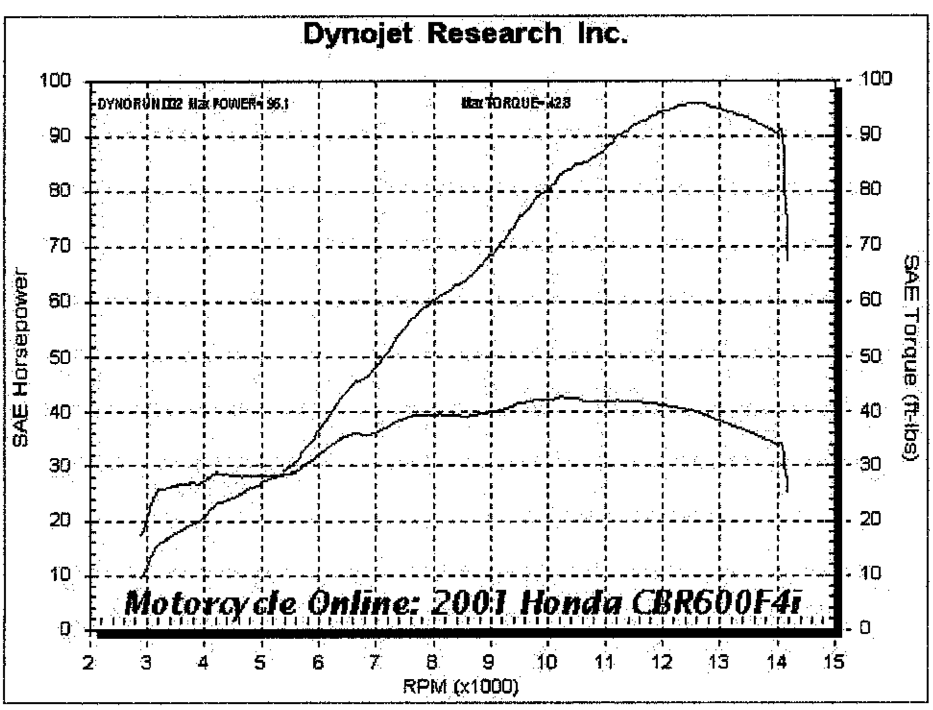



Figure 2.1: HONDA CBR 600 F4i engine performance graph [1]

From the graph, the maximum torque produce is ~42.8 ft-lb at 10100 rpm while the maximum horsepower is around 96.1 hp at 12500 rpm.

Load or tractive resistance provides resistance to the prime mover. A vehicle resistance to motion is due to three fundamental parameters, gradient resistance, rolling resistance and aerodynamic resistance. Soft start or slipping element control the load applied to the prime mover. They also allow the prime mover to achieve optimum operating conditions. Gearbox or ratio-ing element is a device that changes the speed of the prime mover to match the load applied.

$$P = 2 \times \pi \times N \times T; \quad (1)$$

With $P$ = Power,

$N$ = speed and

$T$ = Torque

For constant power input if the speed increases the torque required will reduce. Since size of the component parts are related to load and torque, if the torque reduced the size of the shaft required also will reduces. Prime movers are often run at higher speeds than the machinery in order to reduce its size. Therefore, designer aim is to maintain as high speed as possible for as long as possible to maintain low torques and small component sizes.

## 2.3 Gear-Drive System

A gear is a component within a transmission device that transmits rotational force to another gear or device. Depending on their construction and arrangement, geared devices can transmit forces at different speeds, torques, or in a different direction, from the power source with any slippage.

Current car is using chain-drive as it drive train mechanism. There are some advantages and disadvantages between the chain-drive and gear-drive. Table below show the comparison between both drive systems.

Table 2.1: Advantages of gear-drive and chain-drive

| Gear-Drive | Chain-Drive |
| --- | --- |
| • Give positive drives and constancy of speed ratio without any slippage<br>• The drive is very compact due to short centre distance used.<br>• High efficiency, reliable service and simple operation<br>• Maintenance is expensive and last longer if properly lubricated<br>Can drive much heavier loads due to it unlimited size. | • Can be employed for relatively long and short centre distances.<br>• Can have efficiency as high as 98 to 99 percent when operating under ideal condition. |

Table 2.2: Disadvantages of gear-drive and chain-drive

| Gear-Drive | Chain-Drive |
| --- | --- |
| • Manufacturing of gear is complex and requires special tools and equipment.<br>• Not suitable for large centre distance because the drive will become bulky<br>• Bad gear meshed result to power loss | • Require more service and maintenance.<br>• Due to wear of chain joints, the chain get stretched that increase the chain pitch resulting in velocity fluctuation and necessitating the use of take up drives.<br>• Noisy operation |

### 2.3.1 Bevel Gear

A set or two (2) sets of bevel gear will be used to replace the existing drive train system which use a chain-drive as it primary power/torque transfer. The selection of the number of bevel gear set is depending on the arrangement of the engine. Bevel gears are essentially conically shaped, although the actual gear does not extend all the way to the vertex (tip) of the cone that bound it. With two bevel gears in mesh, the vertices of their two cones lie on a single point, and the shaft axes also intersect at that point. The angle between the shafts can be anything except zero or 180 degrees. Often there is no room to support bevel gears at both ends because the shafts intersect. Thus, one or both gears overhang their supporting shafts. This overhung load (OHL) may deflect the shaft, misaligning gears, which causes poor tooth contact and accelerates wear. Shaft deflection may be overcome with straddle mounting in which a bearing is placed on each side of the gear where space permits. The design of the bevel gear may consider the size (diameter) of the gear, gear ratio, stress and force analysis due to torque applied and the material to be selected.

## 2.4 Design and Performance Analysis

Data for the engine performance in obtain from internet since there is no equipment to run the test for the engine performance. Formulas are used to determine the car top speed and acceleration. Following are the formulas used in measuring the performance of the race car.

| | |
|---|---|
| Engine maximum power | = 96.1 hp = 71.7 kW = 71700 W |
| Transmission efficiency | = 90% |
| Frontal area* | = 0.4 m$^2$ |
| Mass* | = Car mass + driver = (220+60) kg = 280 kg |
| Gradient | = 20$^0$ (maximum) |
| Drag coefficient $C_d$ | =0.326 |
| Coefficient of rolling resistance | = $C_R$ = 0.015 (concrete, normal road) |
| $\rho_{air}{}^{3}$ | $= 1.23 kg/m$ |

### 2.4.1 Estimated Top Speed

Using Tractive Effort (TE) and Tractive Resistance (TR) equations to built two (2) curves that will intersect each other to produce the car top speed.

$$TE \quad = \frac{P\eta}{V}$$

$$= \frac{71700 \times 0.90}{V}$$

$$= \frac{64530}{V} \qquad ; (2)$$

Equation (2) can be used to give the estimate value of the maximum speed produced by the car.

$TR_{max}$ = TR, Newton (N)

$TR_{max}$ = Drag + Rolling resistance + Gradient Resistance

$$= \frac{1}{2} C_D \rho V^2 A + mgC_R + \frac{mg}{G} \qquad (3) \qquad [1]$$

Where; $C_D$ = drag coefficient, $\rho$ = air density

V = car velocity, A = car frontal area

m = overall car mass, $C_{rr}$ = rolling coefficient

G = road gradient

The maximum speed can be obtained from of TE and TR vs Velocity graph. The intersection point between the TE and TR show the maximum speed.

### 2.4.2 Final Drive Ratio

The final drive ratio is the final speed reduction before the power transmitted to the wheels. This ratio will determine the performance of the car, balancing between acceleration and maximum velocity. The ratio has to be determined through calculation from the value of maximum power and torque of the engine. The final drive ratio is the only gear ratios that will be adjust in this project because the internal gear ratio is not changed from the manufacturer's setting.

### 2.4.3 Determination of Final Drive Ratio

Calculate the final drive ration at maximum power parameter.

Finding Overall Speed Ratio,

Overall speed ratio = primary ratio x gear ratio x final drive (4)

$\frac{rpm@\max.power}{\max.speed}$ rev/min for every km/h @ rev / meter travel

TE at wheels x 1m = 2Π x rev/meter x Torque

Where Torque is obtained from engine characteristic and necessary rpm for each torque is obtain using [1]

$$\text{Rpm} = \frac{rpm@\max.power}{\max.speed} \text{ x road speed (km/h); (for 0-169) km/h}$$

$$\text{Overall speed ratio} = \frac{rpm@\max.power}{\max.speed} \text{ x } 4\pi D_{wheel}$$

Overall speed ratio, where value for primary ratio and gear ratio is not changeable. Arrange the equation will lead to the final drive ratio value.

## 2.5 Drive Shaft

The design of the drive shaft should not exceed its Torsion Stress, $\tau_{allow}$ and Shear Stress, $\sigma_{allow}$ .For the shaft, it is preferable to use steel alloy for the material. One of this project's aims is to apply weight reduction to the car. It is recommended to use tubular shaft instead of solid shaft since the minimum diameter of the shaft between these shafts is not much in difference.

A hollow/tubular shaft can reduce weight and allow fluids to circulate to moving parts for cooling or lubrication. What important when is with hollow shaft is, user need to check for balance of the shaft if the fabrication method is drilling or boring.

### 2.5.1 Calculating drive shaft minimum diameter [5]

(a) For solid section shaft;

$$\frac{J}{c} = \frac{T}{\tau_{allow}}$$

$$\tau_{allow} = \frac{T}{J} \times c \text{ , Where } J = \frac{\pi}{2}c^4 \text{ for solid section shaft}$$

$$= \frac{T}{(\frac{\pi}{2}c^4)} \times c$$

$$= \frac{2T}{\pi c^3}, \text{ Where c is the shaft radius} \qquad (5)$$

Determination of minimum diameter for the drive shaft

$$V_1 = V_2$$

$$r_1 \omega_1 = r_4 \omega_4$$

$$\omega_4 = \frac{r_1 \omega_1}{r_4}$$

Where $V_1$ = Speed of transmission output of front gear

$V_2$ = Speed of drive shaft of driven gear

$r_1$ = Drive gear radius

$r_4$ = Driven gear radius

$$P1 = P2$$

$$P1 = T\omega_4 \text{ ; } (5)$$

$$\frac{J}{c} = \frac{T}{\tau_{allow}} \text{ ,T is T from (5)}$$

$$\frac{\frac{\pi}{2}c^4}{c} = \frac{T}{\tau_{allow}}$$

$$\frac{\pi}{2}c^3 = \frac{T}{\tau_{allow}}$$

$$c = \sqrt[3]{\frac{2T}{\pi \times \tau_{allow}}}\text{, where } \tau_{allow} = 85e6 \text{ MPA}$$

(b) For hollow / tubular shaft:

- Given the standard thickness of the hollow shaft is 3mm, inner radius, $c_i$ is an assumption, (for standard shaft with 25mm diameter or 12.5 mm outer radius, $r_o$)
- Using backward calculation to find the outer radius, $r_o$ of the hollow shaft using following formulas to find minimum allowable $r_o$
- Allowable Stress, $\tau_{allow}$ for steel is < 85 MPa

$$\frac{J}{c} = \frac{T}{\tau_{allow}}$$

$$\tau_{allow} = \frac{T}{J} \times c \qquad \text{Where } J = \frac{\pi}{2}(c_o{}^4 - c_i{}^4) \text{ for solid section shaft}$$

## 2.6 Bevel Gear Design Parameter and Equations

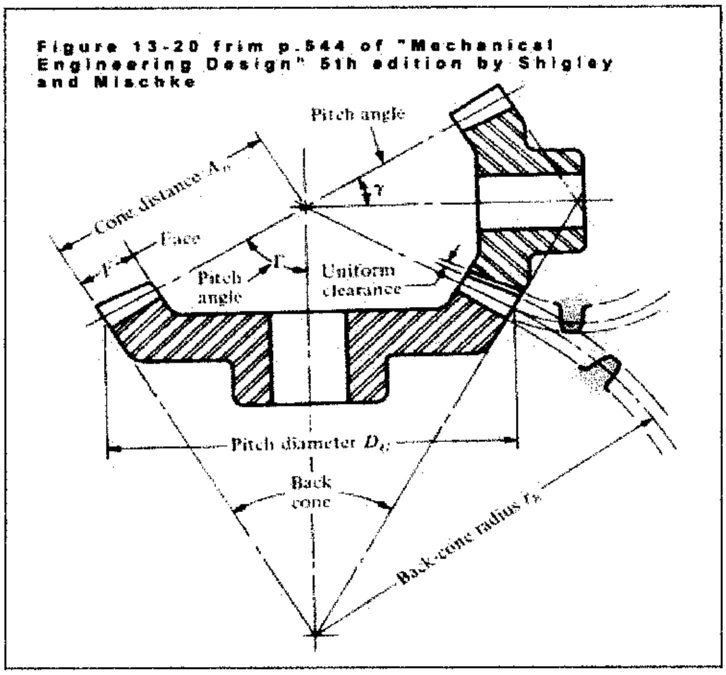



Figure 2.2: Bevel gear terminology [4]

Tan $\gamma = N_P / N_G$ ,$\gamma$=pinion pitch angle

Tan $\Gamma = N_G / N_P$ ,$\Gamma$=gear pitch angle

### 2.6.1 Calculating Bending Stress in Bevel Gear

Power and torque transfer from the engine produce stress to the bevel gear. The bending stress should not over the maximum bending stress. [4]

$$\sigma_F = \frac{W_t K_A K'_v Y_X K_H}{b m_{et} Y_\beta Y_j} \qquad (7)$$

- Application factor $K_a$, Load-distribution factor $K_m$, Size factor $K_s$, Dynamic factor, $K_V$ are the same as for spur and helical gears.
- Transmitted tangential load ,Wt
- To simplify the equation, $K_a$, $K_m$, $K_s$, $K_V$ can be set to 1

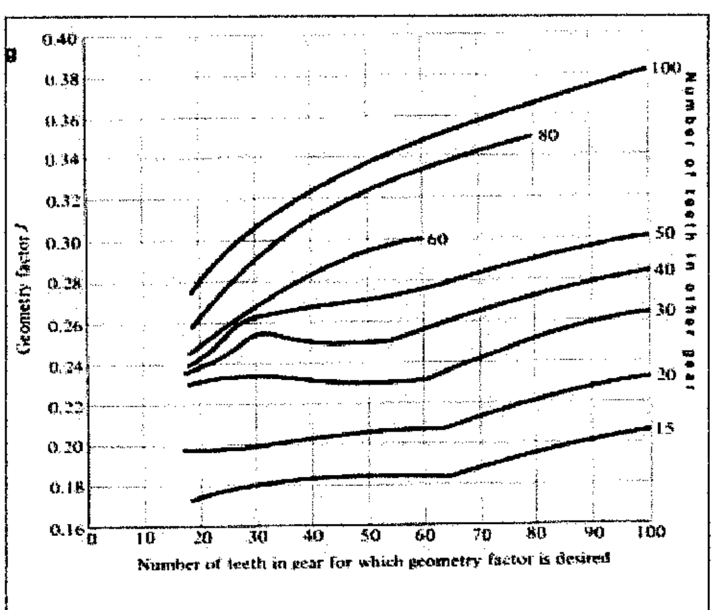



Figure 2.3: Geometry factor, J or Yj [4]

Permissible Bending Stress Equation

$$\sigma_{FP} = \frac{\sigma_{F\lim} Y_{NT}}{S_F K_\theta Y_Z} \qquad (8)$$

## 2.7 Lubrication System

Oil bath lubrication is a method where one of the gears is submersed in lubricant, which then transfers oil to the other gears, shafts and seals. Standard practice is submerging the gear for about 2/3 its teeth into the oil reservoir. In this project, the ring gear turns in the lubricant, transferring the oil between gears through direct contact. A good rule of thumb for oil level for sump-lubricated gears is that the level should completely cover the tooth of the gear sitting at the lowest position in the drive when the idle.

The oil level in gear casing must be maintained within a narrow range to assure that the component receives the correct amount of lubricant coverage. If the level of lubricant in the gear casing is too high or too low, excessive wear will be generated accelerating the degradation of the oil and shortening the life of the gears (increases the oxidation rate). Meanwhile, when the oil level is too low, contact is insufficient to lubricate (less oil film) the gear, and to act as a heat sink to carry away the normal level of heat generated. Viscosity of the lubricant plays important role in dealing with the power loss of the system. In general, the higher the viscosity of the oil the more friction the lubricant generates. As a result, it causes high frictional torque, which the gear experience greater power loss and lower gear efficiency. Therefore, viscosity of the lubricant is essential to be determined to maintain the performance of the system. Below are the relationship between the oil volume, friction loss and temperature in lubricating gears.

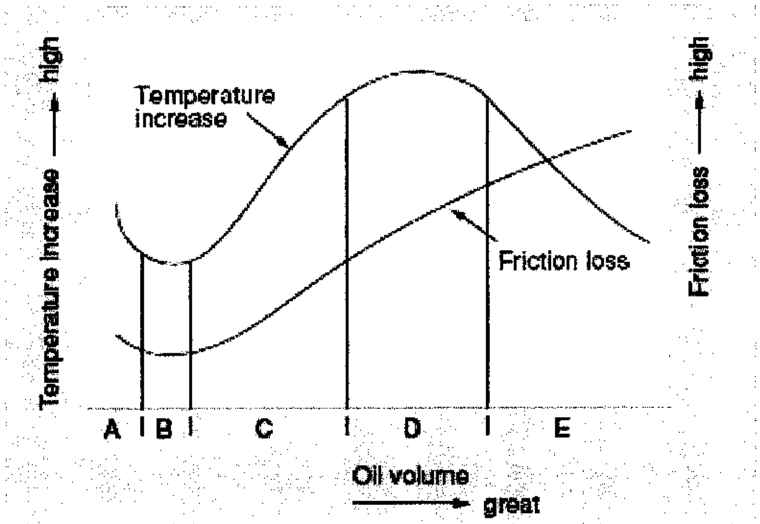



Figure 2.4: Oil volume, friction loss and gear temperature relationship

Table 2.3: Lubrication methods and its characteristics

| Range | Characteristics | Lubrication Method |
|---|---|---|
| A | Oil volume is low; direct metallic contact occurred between the sliding teeth of gears. Gear abrasion and wear occur | - |
| B | A thin oil film develops over all surface, friction is minimal and gear temperature is low | Grease lubrication, oil mist |
| C | Oil volume increase, heat buildup is balance by cooling | Circulation lubrication |
| D | Regardless of oil volume, temperature increase at a fixed rate | Circulating lubrication |
| E | Oil volume increase, cooling predominates and gear temperature decrease | Oil bath lubrication, splash lubrication |

### 2.7.1 Selection of gear lubricants

Oil selected for the lubrication of gear set should fulfil certain basic requirements which, although simple, are often overlooked. The basic requirements can be summarized as follow:

- To minimize friction and reduce wear between sliding surface by the provision of a thin film of lubricant which will prevent metal-to-metal contact between the elements of bearings and the faces of gear teeth at the point of contact.
- To protect the gear set against rust and corrosion
- To transfer hear generated in enclosed gears to the gear case walls.

For lubricating oils, viscosity is one of the most important properties and determines oil's lubricating efficiency. If viscosity is too low, formation of the oil film will be insufficient, and damage will occur to contact area of the gears. If viscosity is too high, viscous resistance will also be great and result in temperature increases and friction loss. In general, for higher speed applications lower viscosity oil suitable to used; for heavier load applications, higher viscosity oil shall be used.

# CHAPTER 3
# METHODOLOGY

## 3.1 Project Flow

The author has developed a process flow for this project which might help to plan carefully for the project as below:

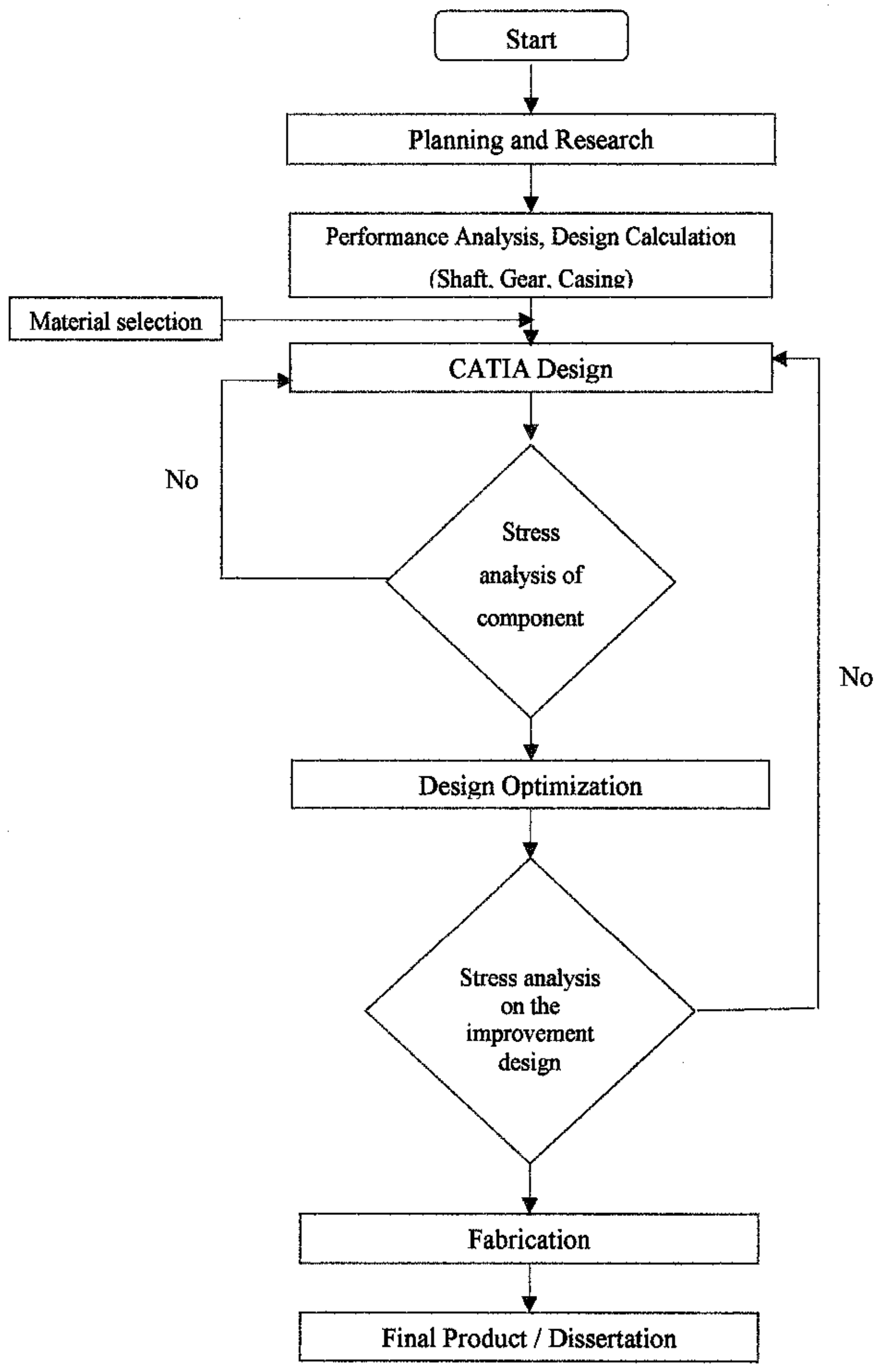



Figure 3.1: Process flow chart

Some guidelines on the process of designing the drive train system from chain drive system to bevel gear system are being set to sure this project is a success. These guidelines are explained in Section 3.1.1 and Section 3.1.2

### 3.1.1 Planning

In general, this project will be done in the following stage order; 1) Research Stage, 2) Designing Stage, 3) Material Selection Stage, 4) Analysis Stage, 5) Fabrication Stage, and 6) Testing Stage

### 3.1.2 Research and Study

Since the project objective is to design and redesign the gear drive system, it is important to start the project with research on how to determine the current performance of the car. This is important when the author come to the design stage. For the gear system, the author has studies on how to design bevel gear and the shaft connected to it. In the first stage (Research Stage); gear-drive system has more possible of power loss compare to chain-drive system but the power loss is just about 3-4% which is very small. Several things have been covered during the research development.

- Bevel gear concept , application, advantages and disadvantages
- Determination of final drive ratio
- Estimate the current performance for the car
- General material to be considered as fabrication material.
- Shaft sizing (torsion and bending stress)

For the drive shaft, two important parameters need to calculate carefully. Consideration on the Torsion and Bending stress are very important. The size of the shaft depends on the stresses applied to it.

### 3.1.3 Design and Material Selection

In design section, task is divided into 2 parts which is the manual calculation governing related equation and 3D design drawings using CATIA or any other engineering software to get a better view of the project based on the calculation done. The manual calculation is used as a basis in designing the drive train part. Stress for critical component such as gear, differential housing and shaft is determined in order to select appropriate material for the fabrication process.

### 3.1.4 Analysis

In this stage, all the design will be analyzed for its reliability and performance parameters in several types of analysis. The drive train compartment will be analyzed base on the highest torque applied which is in first gear. Design improvisations shall be done when the analysis result does not comply with the required criteria. Any adjustment done need to meet the FSAE rule. Any failed result will lead to redesign the component. Analysis is done by using CATIA software.

### 3.1.5 Fabrication

Fabrication of the prototype is based on the results from the analysis. The complexity and accuracy` of the design will help the author whether to fabricate by using UTP CNC machine, MAZAK or purchase the design product from outside market with the required specification.

## 3.2 Tool required

### 3.2.1 Equipment

CNC / EDM machining equipment will be used for the drive train prototype fabrication based on approved design.

### 3.2.2 Software

Engineering software will be used to design the drive train, bevel gear, gear casing, differential housing and drive shaft. Each part designed will undergo stress analysis by using the software as for initial evaluation on the performance and reliability. Below are the software related to this project.

- CATIA P3 V5R12
- Microsoft Excel

CATIA is software that allows the users to do design, analysis, drafting and digital mock up of the product. Compared to AutoCAD 2007, CATIA is chosen due to its ability to generate 3D design easier and faster.

# CHAPTER 4
# RESULT AND DISCUSSION

## 4.1 Performance Analysis

### 4.1.1 Estimating the Maximum Velocity

To calculate the TE, equation (2) is used. To calculate TR, an equation (3) is used and determines to be as follow:

$$TR_{max} = (0.5\times 0.326\times 1.23\times V^2\times 0.4)+(280\times 9.81\times 0.015)+\frac{280\times 9.81}{2.923}$$

$$= 0.080196V^2 + 980.9215$$

Value for TR and TE is tabulated and then use to plot graph of TR and TE vs Velocity. Intersection between the line show the maximum velocity gained.

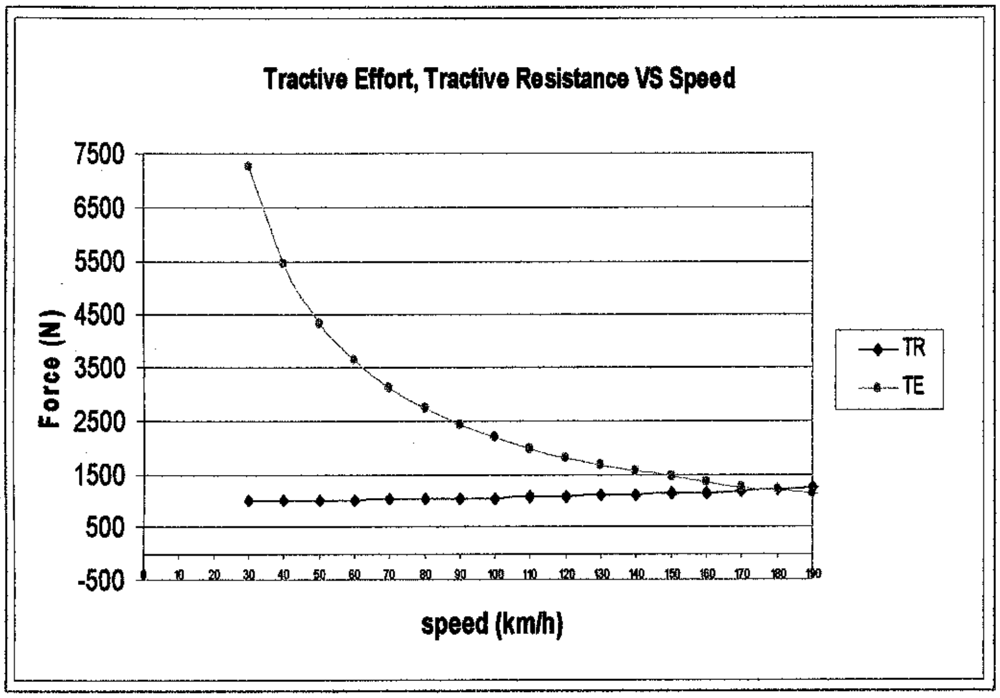



Figure 4.1: TE, TR vs Speed

Tractive effort $= \frac{T_{engine} x G_i}{r}$ (9)

Where; $T_{engine}$ = Engine Torque

$G_i$ = Gear $Ratio_i$ ( refer table below)

r = wheel radius

Table 4.1: Overall gear ratio for corresponding gear (1st gear – 6th gear)

| | Gear ratio | Primary reduction x final ratio | Overall Ratio |
|---|---|---|---|
| First gear | 2.833 | | 17.431 |
| Second gear | 2.062 | | 12.687 |
| Third gear | 1.647 | 1.822 x 3.377 | 10.134 |
| Forth gear | 1.421 | | 8.743 |
| Fifth gear | 1.272 | | 7.826 |
| Sixth gear | 1.173 | | 7.217 |

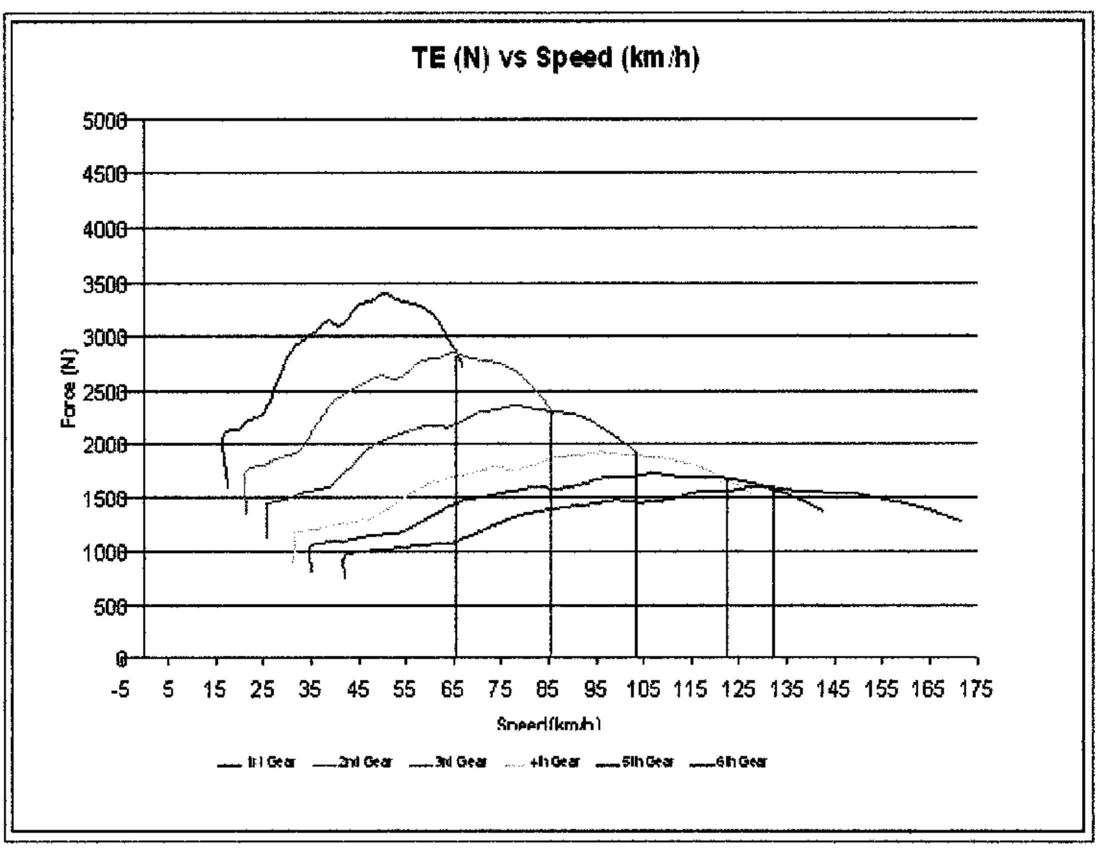



Figure 4.2: Tractive effort (1st gear-6th gear) vs speed

**Results**

Changing Gear Speed :

i. First to second gear = 65 km/h
ii. Second to third gear = 85 km/h
iii. Third to forth gear = 103 km/h
iv. Forth to fifth gear = 122 km/h
v. Fifth to sixth gear = 131 km/h

Maximum Speed at Sixth Gear is 165km/h.

## 4.2 Design and Analysis of Drive Train Components

### 4.2.1 Determination of Minimum Shaft Diameter

For solid section shaft;

$\tau_{allow} = \frac{T}{J} \times c$ Where $J = \frac{\pi}{2} c^4$ for solid section shaft

$$\tau_{allow} = \frac{2T}{\pi c^3} \quad \text{Where c is the shaft radius} \qquad (10)$$

Calculation is done at Max. Power which is 71662 W at 1309 rad/s and $\tau_{allow}$ for steel < 85MPa

Sizing the drive shaft

$$V_1 = V_2 \qquad (11)$$

$r_1 \omega_1 = r_2 \omega_2$ $r_1$ is drive gear radius = 1.325 in= 0.0336 m

$r_2$ is driven gear radius = 4.5 in = 0.114 m

$\omega_1$ =1309 rad/s

$$\omega_2 = \frac{r_1\omega_1}{r_2} = \frac{0.0336 \times 1309}{0.114}$$

$$\omega_2 = 386 rads/s$$

P1 =P2

71663 =T $\omega_2$

$$T_2 = \frac{71663}{386} = 185.6Nm$$

$$\frac{J}{c} = \frac{T}{\tau_{allow}}$$

$$\frac{\pi}{2}c^3 = \frac{T}{\tau_{allow}}$$

$$c = \sqrt[3]{\frac{2T}{\pi\tau_{allow}}} \quad c = \sqrt[3]{\frac{2\times(185.6)}{\pi\times(85\times10^6)}} = 0.0112m;$$ $\tau_{allow}$ for steel is < 85 Mpa

$$c = 11.2mm$$

So, minimum shaft diameter = (11.2x2) mm= 22.4 mm

(b) For hollow / tubular shaft:

- Given the standard thickness of the hollow shaft is 3 mm, given inner radius, $c_i$ = 9.5 mm, (for standard shaft with 25 mm diameter or 12.5 mm outer radius, $r_o$)
- From part (a) pg 30, $\omega_2 = 386 rads/s$ and $T_2$ = 185.6 Nm
- $\tau_{allow}$ for steel is < 85 Mpa

$$\tau_{allow} = \frac{T}{J}\times c \qquad \text{Where } J = \frac{\pi}{2}(c_o^{\,4} - c_i^{\,4}) \text{ for hollow shaft} \qquad (12)$$

$$85\times10^6 = \frac{185.6}{\frac{\pi}{2}(c_o^{\,4} - 0.0095^4)}\times c_o$$

$$c_o^{\,4} - (1.39\times10^{-6})c_o = 0.0095^4$$

$c_o$ = 0.0113 m (Similar to solid shaft)

So, minimum shaft diameter = (11.3 x 2) mm= 22.6 mm

We can take 1" or 1.5" diameter for shaft to fabricate.

Since, the hollow shaft produce similar result with solid shaft, it is preferable to used hollow shaft to minimize race car weight.

### 4.2.2 Project Design

For this project, the design comes from UTP FSAE SF-02 and modification is made to the drive train system. Arrangement of the engine position has been change in order to design the gear drive system. The gear drive system will eliminate the use of sprockets as mechanical device for power transfer. Current differential housing need to redesign since a ring gear will be mounted on it. Section 4.7.1 and 4.7.2 show the initial design of the car and the modification design which is the project design.

### 4.2.3 Design and Packaging Advantages

The system (gear drive) in this project will be compared to the current system which used chain drive for the drive train. The new system allows a shorter wheel base for the car and consumes less space rather than chain drive system. Overdesign the components would result to mass packaging of the drive train. The design improvement will optimize the size of the drive train. This new design provide a better safety to the power transfer component (gears) as it fully covered by a gear casing with splash lubrication inside it to allow the gear operate smoothly and lower the temperature of the system.

### 4.2.4 Gear Selection

The author has come out with several type of gears which are straight bevel gear, spiral bevel gear, hypoid gear and chain & sprocket (reference) as an option to be implemented in the new drive train system. Using parameters such as load capacity, weight, cost, noise, complexity and installation, a table is build and weightage value of 0 until 10 for worst to best is assigned to evaluate the different type of gear options. A radar chart is used to determine the best options.

Table 4.2: Evaluation on different types of gear system

| Criteria | | Straight Bevel | Spiral Bevel | Hypoid Gear | Chain & Sprockets |
|---|---|---|---|---|---|
| Weight | | Almost the same | Almost the same | Almost the same | Weight saving |
| | | 5 | 5 | 5 | 10 |
| Cost | | Medium | High | High | Low |
| | | 3 | 3 | 2 | 8 |
| Load Capacity | | Handle high load | Handle high load with more efficient | Handle high load with more efficient | Load is restricted to the chain size |
| | | 8 | 9 | 9 | 5 |
| Noise | | Low Noise | Silent | Silent | Noisy |
| Complexity | | Less complex compared to spiral and hypoid | Need high accuracy for fabrication | Need high accuracy for fabrication | Stock |
| | | 7 | 4 | 3 | 9 |
| Installation | | Standard alignment for gear installation | Require high accuracy of gear meshing | Require high accuracy for shafts offset and gear meshing | Sprockets need to be parallel to avoid chain from slipped. |
| | | 7 | 5 | 5 | 7 |

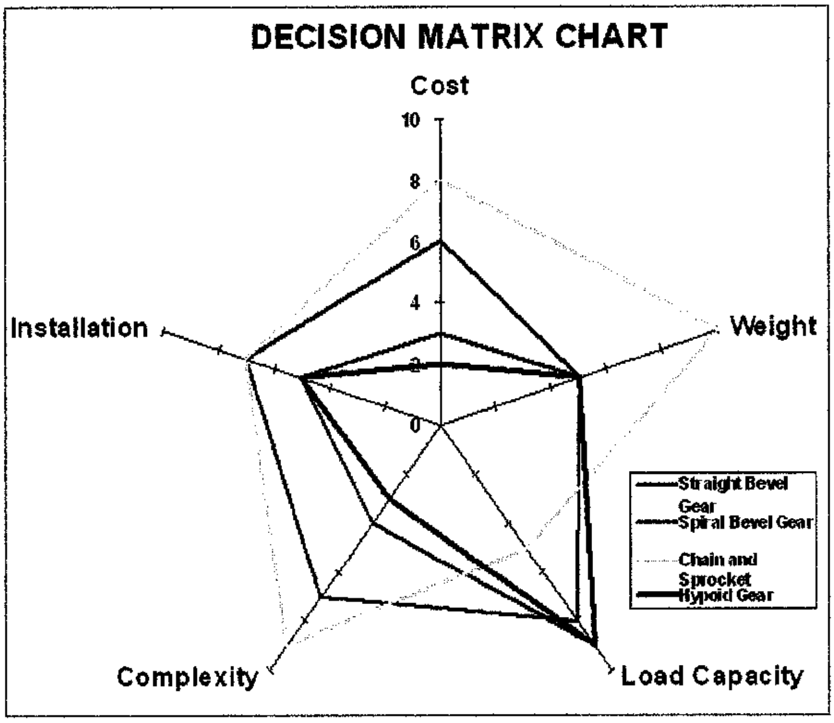



Figure 4.3: Decision Matrix Chart on gear system

**Result:**

Straight Bevel Gear (purple line) gives the best result to be included in the gear drive system.

*Note: Chain & Sprocket is just as a reference for the existing drive train system.

### 4.2.5 Design of Straight Teeth Bevel Gear

The designs of the gears are based on the equation taken from Gear Design Simplified, 3rd Edition, Franklin D. Jones, Henry H. Ryffel [3] book. Using 3.377 as the ratio of the gear, it is suitable to have a 48 teeth ring bevel gear and a 14 teeth pinion bevel gear.

Table 4.3: Straight Bevel Gear design equations [3]

| | | | |
|---|---|---|---|
| Pitch-Cone Angle, A of Gear | $\tan A = \frac{N}{n}$ | Whole Depth, W | $W = \frac{2.157}{P}$ |
| Pitch-Cone Angle, a of Pinion | $\tan a = \frac{n}{N}$ | Dedendum, K | $K = \frac{1.157}{P}$ |
| Face Width, F | $Maximum\ F = \frac{Cone\ Distance, E}{3}$ | Addendum, J | $J = \frac{1}{P} = Pc(0.3183)$ |
| Circular Pitch | $Pc = \frac{D(3.1416)}{N}$ | Cone Distance, E | $E = \frac{D}{2(\sin A)}$ |
| Addendum Angle, M | $\tan M = \frac{addendum}{cone\ distance, E}$ | Diametral Pitch | $P = \frac{N}{D}$, $P = \frac{n}{d}$ |
| Dedendum Angle, G | $\tan G = \frac{dedendum}{cone\ distance, E}$ | | |

Table 4.4: Straight Bevel Gear Specification

| | Driven Gear (Ring Gear) | Driving Gear (Pinion) |
|---|---|---|
| Teeth Number, N | 48 | 14 |
| Pitch Diameter, PD | 9 in (228.6 mm) | 2.625in (66.68mm) |
| Circular pitch, Pc | 0.59 in (14.96 mm) | 0.59 in (14.96 mm) |
| Diametral Pitch ,P | 5.33 in (135.47 mm) | 5.33 in (135.47 mm) |
| Whole Depth, W | 0.40 in (10.27 mm) | 0.40 in (10.27 mm) |
| Cone Distance, E | 4.69in (119.06mm) | 4.69 in (119.06mm) |
| Face Width, F | 1.56 in (39.69 mm) | 1.56 in (39.69mm) |
| Pitch Angle | 73.7° | 16.3 ° |
| Module | 4.763 mm | 4.763 mm |

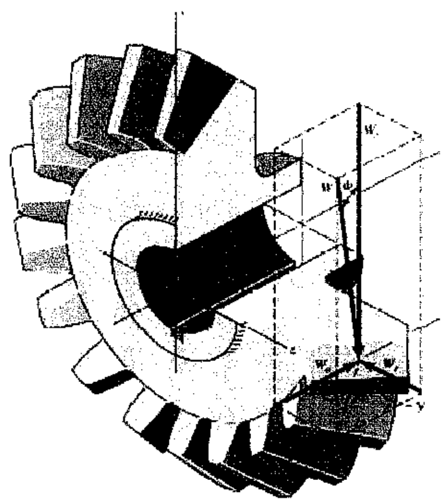
The design of the spiral bevel gear is more complex compare to straight bevel gear. For the project, the selected type of bevel gear is the straight tooth bevel gear. The author has done the calculation and sizing for the spiral bevel gear (see Appendix 2) as for reference for the further development or continuation of this project. All calculations are base on [1] 6th Edition, Mechanical Engineering Design, Joseph E. Shigley, Charles R Mischke and [3] Gear Design Simplified, 3rd Edition, Franklin D. Jones, Henry H. Ryffel.

### 4.2.6 Straight Bevel Gear Stress Calculation

The gear ratio of 3.37 is rounded up to 3.4. Rounded is made only for the gear and not for the final drive ratio. Based on the gear ratio, a combination of 48 tooth gear and 14 tooth pinion is chosen. (Similar to standard product in market)

To calculate the stress on the gear, equation (7) is used.

Known the Force exert by the system, $F=W_t = 23218$ N



$W_t$ is force acting perpendicular to the gear teeth

Figure 4.4: Bevel gear Free Body Diagram (FBD) [4]

Bending stress, $$\sigma_F = \frac{W_t K_A K'_v Y_X K_H}{b m_{et} Y_\beta Y_j}$$ [4]

Where $K_A$ =1 | B = 39.688 mm (face width)

$m_{et}$ = 4.762 mm (module) | $K_H$ = $K_{mb}$+ (5.8E-6) $b^2$ = 1.259

$K_V$ =1 | $Y_\beta$ = 1 (for straight bevel gear)

Yj = 0.19 (refer figure 3) | $Y_X$ = 0.4867+0.008339 = 0.5264

Substitute the value,

$$\sigma_F = \frac{23218\,(1)(1)(0.5264)(1.259)}{(0.039688)(0.04762)(1)(0.19)}$$

$$= 42.85 \text{ MPa} < \sigma_{allow} \text{ Steel Alloy 4340 (236 MPa)}$$

*Note: Calculation based on mechanical engineering design book by Joseph Shiegly and Charles r. Mischke for AGMA bending stress. (pg. 951-962)

### 4.2.7 CATIA Drawing of Drive Train Components

For this project, the design comes from UTP FSAE SF-02 and modification is made to the drive train system. Arrangement of the engine position has been change in order to design the gear drive system. The gear drive system will eliminate the used of sprockets as mechanical device for power transfer. Current differential housing is redesign since a ring gear will be mounting on it.

<u>1.5 " Drive Shaft</u>

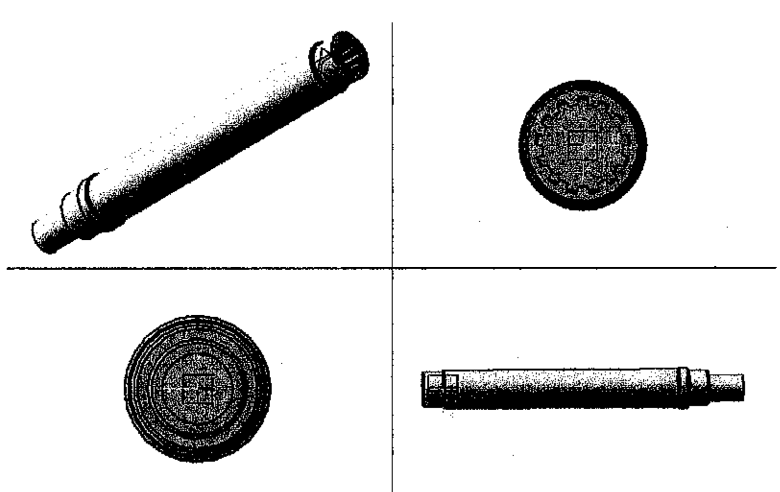


Figure 4.5: 1.5in (D) Drive Shaft

14 teeth Pinion Gear

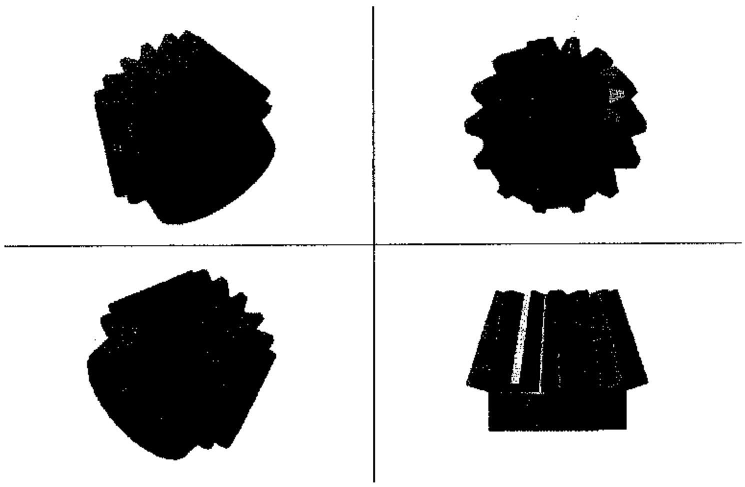


Figure 4.6: 14 teeth Pinion Gear

48 teeth Ring Gear

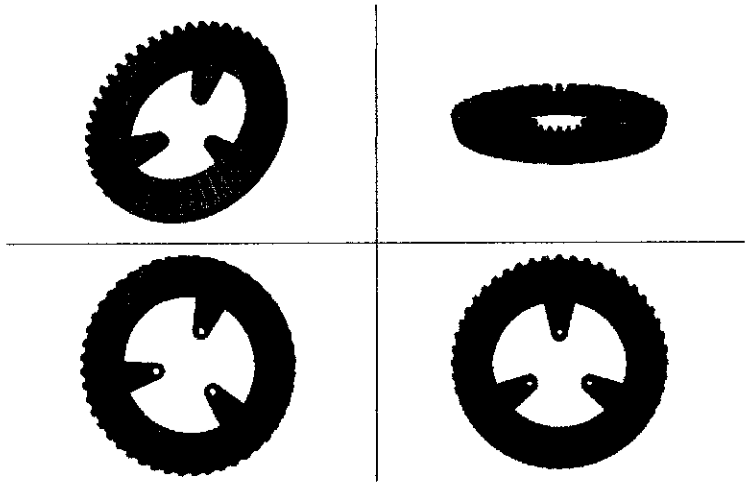


Figure 4.7: 48 teeth Ring Gear

Differential Housing

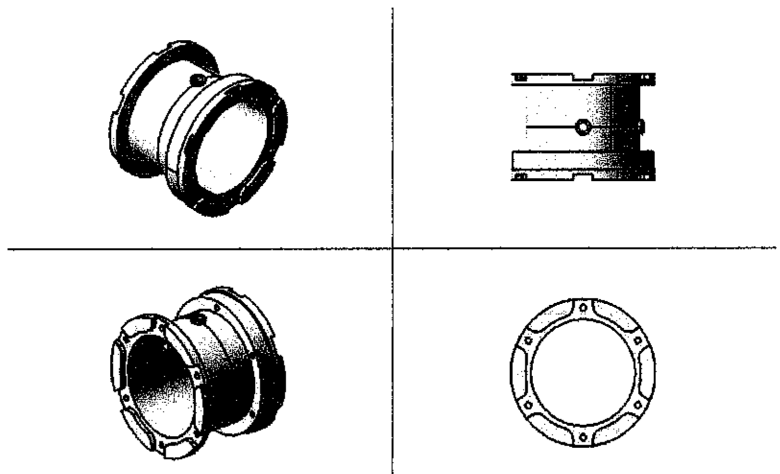


Figure 4.8: Differential housing

Gear Casing

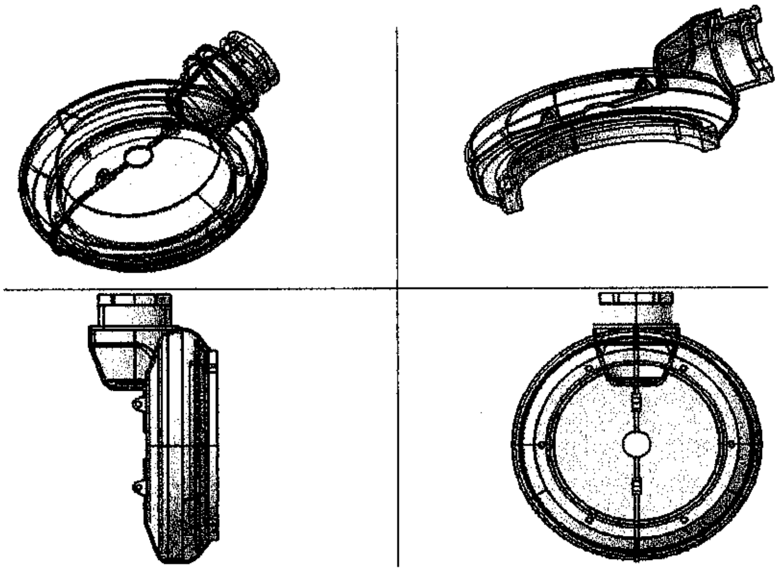


Figure 4.9: Gear Housing

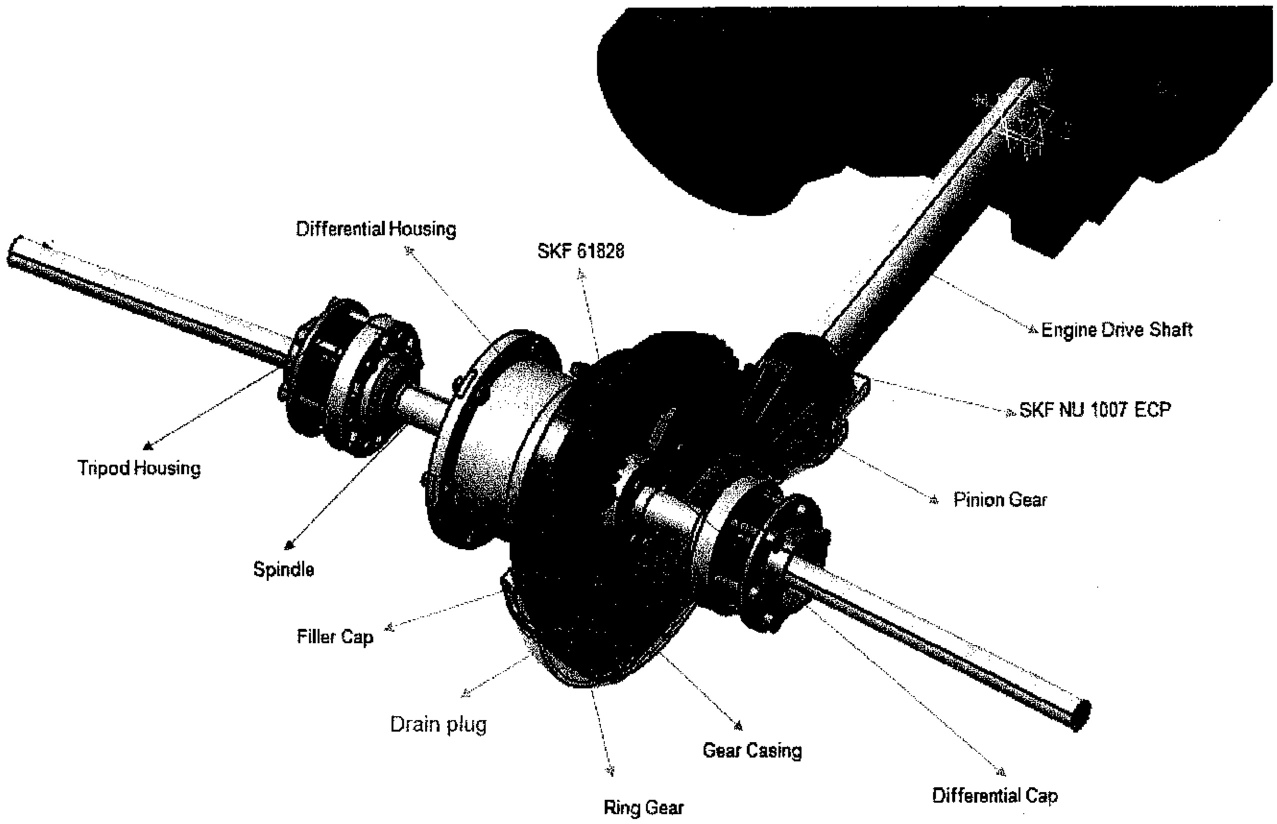



Figure 4.10: Drive train assembly with half gear casing

Table 4.5: Bill of Materials

| No. | Item | Qty | Material | Density kg/m³ | Volume m³ | Weight kg |
|---|---|---|---|---|---|---|
| 1 | Torsen Differential | 1 | Stock | | | |
| 2 | Differential Housing | 1 | AL 7075 | 2800 | 2.61E-04 | 0.732 |
| 3 | Differential Cap | 2 | AL 7075 | 2800 | 1.17E-04 | 0.328 |
| 4 | Ring Gear (9 in) | 1 | Steel 4340 | 7850 | 6.11E-04 | 4.798 |
| 5 | Pinion Gear (2.65 in) | 1 | Steel 4340 | 7850 | 1.46E-04 | 1.15 |
| 6 | Bearing Cover | 2 | AL 7075 | 2800 | 1.29E-05 | 0.036 |
| 7 | Spindle | 2 | Steel 4340 | 7850 | 2.69E-04 | 2.113 |
| 8 | Stub Axle | 2 | Steel 4340 | 7850 | 2.48E-04 | 1.945 |
| 9 | Tripod Housing | 4 | Stock | | | 1.134 |
| 10 | Tripod Joint | 4 | Stock | | | 0.692 |
| 11 | Tripod Boot | 4 | Stock | | | 0.42 |
| 12 | Engine Drive Shaft | 1 | Steel 4340 | 7850 | 1.26E-04 | 0.99 |
| 13 | Circlip | 8 | Stock | | | |
| 14 | SKF Bearing 140mm 61828 | 1 | Stock | | | |
| 15 | SKF Bearing 38mm NU 1007 ECP | 1 | Stock | | | |
| 16 | SKF Bearing 25mm 61805-2RS1 | 2 | Stock | | | |
| 17 | SKF O-Ring d1 24.99 x d2 3.53 | 2 | Stock | | | |
| 18 | Gear Casing | 1 | AL 7075 | 2800 | 3.07E-04 | 0.86 |
| 19 | Bolt, Washer | 12 | | | | |
| 20 | Bolt, Nut and Washer | 12 | | | | |
| 21 | Bolt, Nut and Washer | 8 | | | | |
| 22 | Bolt, Washer | 6 | | | | |
| 23 | Bolt, Nut and Washer | 2 | | | | |
| 24 | Bolt, Nut and Washer | 24 | | | | |
| 25 | Bolt, Nut and Washer | 2 | | | | |
| 26 | Drive Shaft | 2 | Stock | | | 1.338 |

## 4.3 Analysis Consideration for Drive Train Components

### 4.3.1 Material Selection

Most of the components will use either Steel Alloy 4340 or Aluminium Alloy 7075 for fabrication. This materials (Steel 4340 and Aluminium Alloy 7075) is a high strength and light weight. Steel Alloy is preferable to be selected for high stress operation. Both of the steel mention above is costly. Select Steel Alloy 4340 as material for gear from table below:

Table 4.6: Properties of Aluminum Alloy 7075 and Steel Alloy 4340

| Properties/material | Aluminium Alloy 7075 | Steel Alloy 4340 |
|---|---|---|
| Young modulus | 71.0 GPa | 207 GPa |
| Poisson Ratio | 0.33 | 0.3 |
| Density | 2800 kg/m$^3$ | 7850 kg/m$^3$ |
| Coefficient of Thermal expansion | 0.0000234 | 0.0000123 |
| Yield strength | 103 MPa | 472 MPa |

### 4.3.2 Analysis Load

Maximum allowable stresses of drive train components with safety factor of 2 are:

FS = 2

FS= $\sigma_{fail}$ / $\sigma_{allow}$

$\sigma_{allow}$ S.A.E 4043 = 472 MPa / 2

= 236 MPa

$\sigma_{allow}$ AL 7075 = 103 MPa / 2

= 51.5 MPa

Therefore, the drive component is safe to use when the $\sigma_{allow}$ Aluminium Alloy 7075 is below 51.5 MPa and $\sigma_{allow}$ Steel Alloy 4340 is below 236 MPa.

For fabrication process, the drive shaft, ring gear and pinion gear will be fabricated using Steel Alloy 4340 while the casing will use Aluminium Alloy 7075.

## 4.4 Result of analysis for individual drive train component on CATIA basis

To enable the simulation took place, applied load of moment, torque or force should be applied to the material. Clamping part of the material should be first defined before any load is applied. The result of the analysis is mainly dependent on the clamping part on the material and the amount of load exerted to the items. All load values entered should be based on maximum load applied to each material on real world. The clamping part on the items should be based on the fix part.

From engine characteristic curve (from Figure 1), maximum torque is ≈ 42.8 ft-lbs or 58.0291 Nm. The maximum torque from the gearbox can be obtained using the original final drive ratio.

$$T_{gearbox} = \frac{T_{wheel,bike}}{Final\ Drive\ Ratio}$$

$$T_{gearbox} = \frac{58.03}{2.875}$$

$$T_{gearbox} = 20.18Nm$$

Therefore maximum torque at wheel with given new FDR is:

$T_{wheel} = FDR \times T_{gearbox}$

$= 3.377 \times 20.18$

$= 68.15$ Nm

The maximum torque applied to the drive train components is 68.15 Nm.

### 4.4.1 Analysis of Drive Shaft

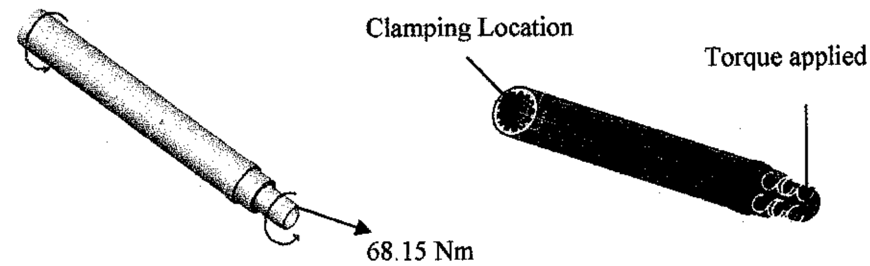



Figure 4.11: Drive Shaft (a) and components of loads (b) applied

**Specification of analysis**

| | |
|---|---|
| Material | : Steel Alloy 4340 |
| Type of load | : applied torque of 68.15 Nm |
| Load location | : Connection at pinion gear |
| Clamping zone | : Shaft spline |

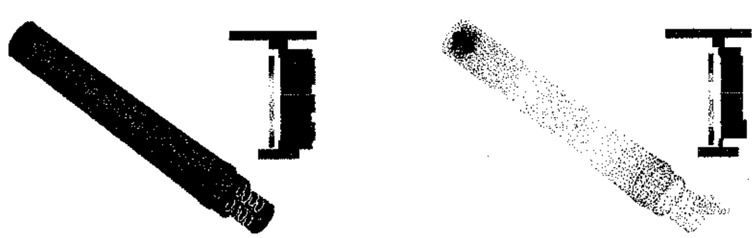


Figure 4.12: Result of (a) Von Mises Stress (nodal values)and (b) Translational displacement vector of drive shaft

**Result:**

| | |
|---|---|
| Maximum stress recorded | : 46.6 MPa |
| Stress concentration location | : Surface between bearing and pinion connection |
| Maximum allowable stress | : 236 MPa |
| Maximum translational displacement | : 0.0384 mm |
| Does the items experience plastic deformation? | : No |
| Is the items pass stress analysis | : Yes |

### 4.4.2 Analysis of Differential Housing

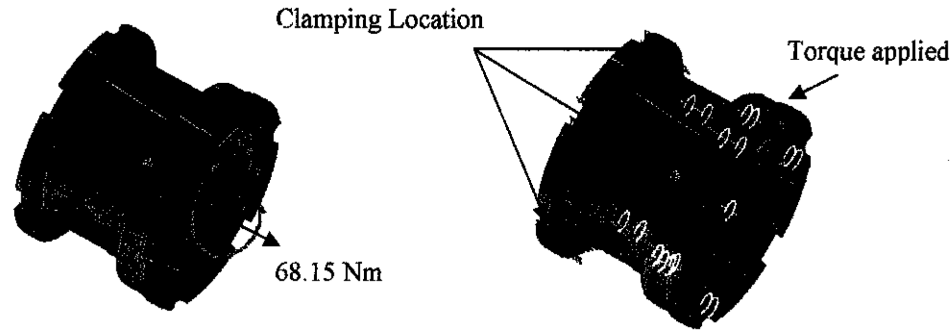



Figure 4.13: Differential housing (a) and components of loads (b) applied

**Specification of analysis**

| | |
|---|---|
| Material | : Aluminium alloy 7075 |
| Type of load | : applied torque of 68.15 Nm |
| Load location | : around the housing |
| Clamping zone | : bolt holes location |


Figure 4.14: Result of (a) Von Mises Stress (nodal values)and (b) Translational displacement vector of the differential housing

**Result:**

| | |
|---|---|
| Maximum stress recorded | : 1.22 MPa |
| Stress concentration location | : around the bolt location |
| Maximum allowable stress | : 51.5 MPa |
| Material yield strength | : 103 MPa |
| Maximum translational displacement | : 0.000332 mm |
| Does the items experience plastic deformation? | : No |
| Is the items pass stress analysis | : Yes |

### 4.4.3 Analysis of Ring Gear

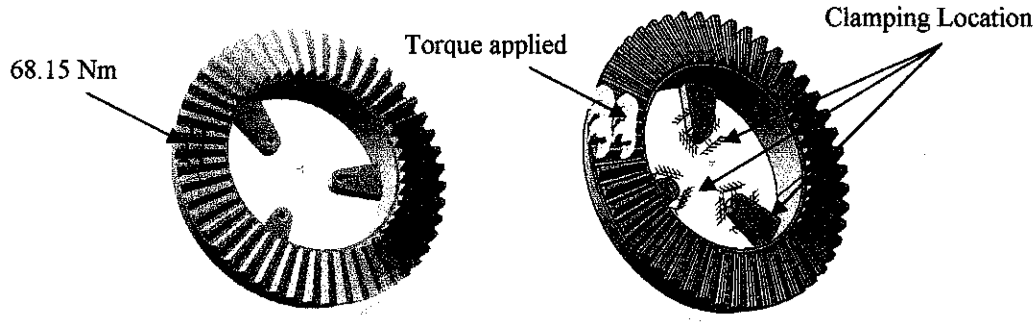



Figure 4.15: Ring Gear (a) and components of loads (b) applied

**Specification of analysis**

| | |
|---|---|
| Material | : Steel Alloy 4340 |
| Type of load | : applied torque of 68.15 Nm |
| Load location | : Surface of meshing teeth |
| Clamping zone | : Gear's bolt holes location |

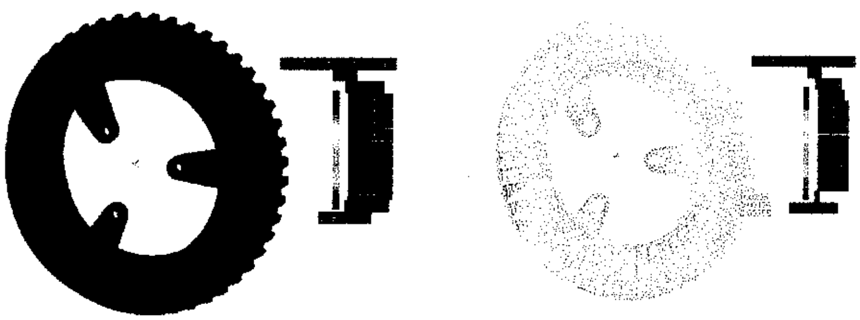


Figure 4.16: Result of (a) Von Mises Stress (nodal values)and (b) Translational displacement vector of ring gear

**Result:**

| | |
|---|---|
| Maximum stress recorded | : 49MPa |
| Stress concentration location | : At end of teeth's surface |
| Maximum allowable stress | : 236 MPa |
| Maximum translational displacement | : 0.0104 mm |
| Does the items experience plastic deformation? | : No |
| Is the items pass stress analysis | : Yes |

### 4.4.4 Analysis of Pinion Gear

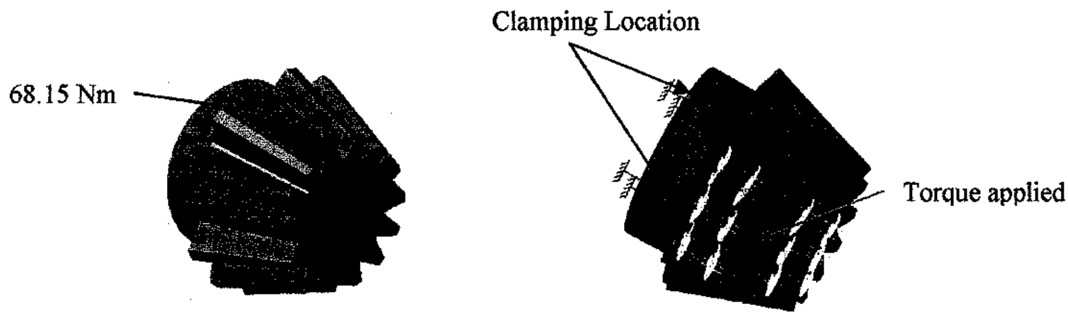



Figure 4.17: Pinion Gear (a) and components of loads (b) applied

**Specification of analysis**

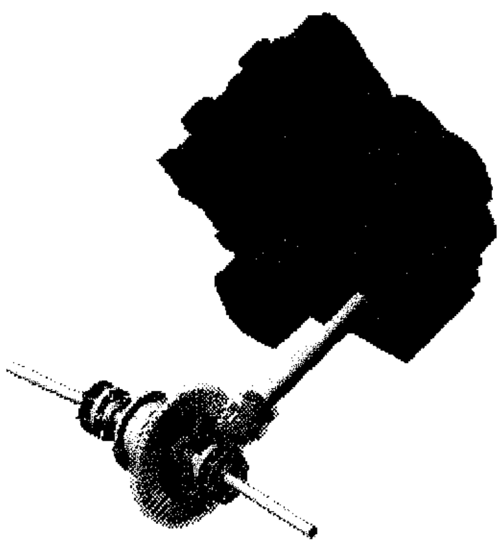
| | |
|---|---|
| Material | : Steel Alloy 4340 |
| Type of load | : applied torque of 68.15 Nm |
| Load location | : Surface of meshing teeth (Ring Gear Teeth) |
| Clamping zone | : Pin location |

Figure 4.18: Result of (a) Von Mises Stress (nodal values)and (b) Translational displacement vector of pinion gear

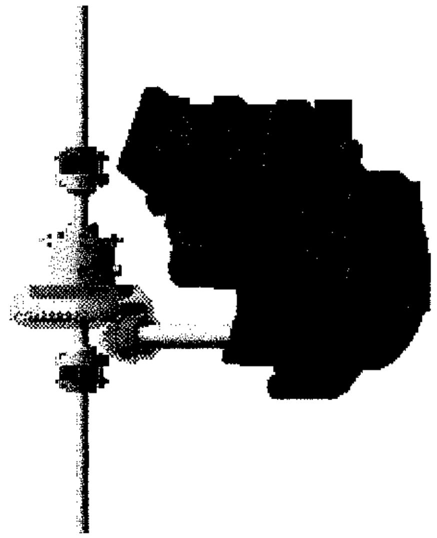
**Result:**

| | |
|---|---|
| Maximum stress recorded | : 1.5MPa |
| Stress concentration location | : At the end of teeth's surface |
| Maximum allowable stress | : 236 MPa |
| Maximum translational displacement | : 0.00329 mm |
| Does the items experience plastic deformation? | : No |
| Is the items pass stress analysis | : Yes |

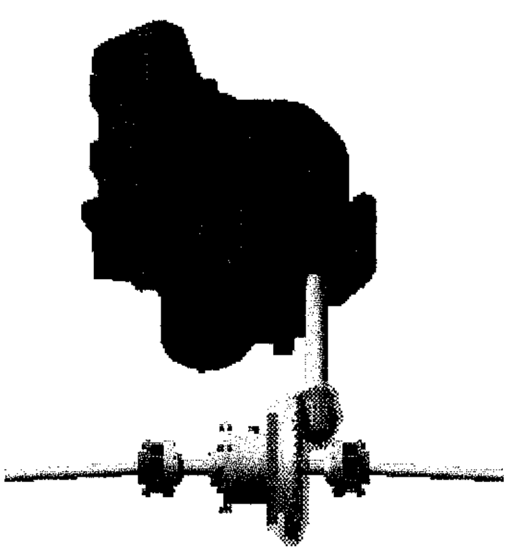
## 4.5 Drive Train Assembly



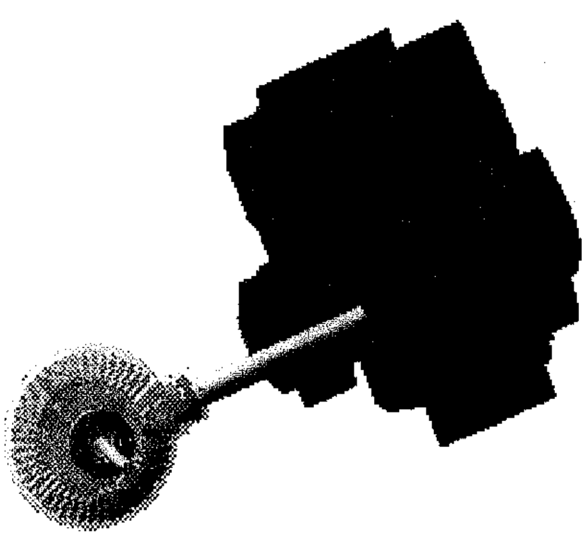
Isometric View



Top View



Rear View



Front View

Figure 4.19: Drive train assembly

## 4.6 Design Optimization

Early designs of the components (shaft, bevel gear and casing) require some design improvements since the component to be fabricated are too heavy. Applying weight reduction to the component needs to consider the stress concentration area to ensure the design is reliable after removing some part of the components. The author has followed the steps in the methodology chart (Figure 3.1) when doing the design optimization. As a result, all of the component's weight has been reduced about 40%-60% of its initial weight.

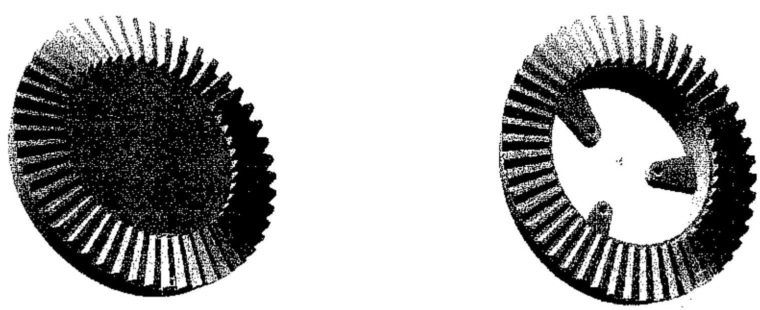


Figure 4.20: Ring Bevel Gear (a) Initial design (b) Optimizing weight design

| | |
|---|---|
| Initial Weight | : 12.724 kg |
| After Removing Weight | : 4.798 kg |
| Weight Reduction | : 7.926 kg (62.29% reduction) |

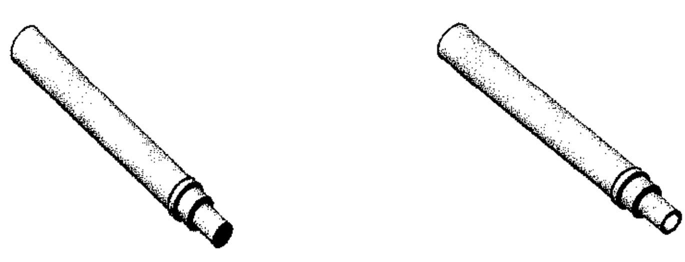


Figure 4.21: Drive Shaft (a) Initial design (b) Optimizing weight design

| | |
|---|---|
| Initial Weight | : 2.864 kg |
| After Removing Weight | : 0.99 kg |
| Weight Reduction | : 1.874 kg (65.4% reduction) |

# CHAPTER 5
# CONCLUSIONS AND RECOMMENDATIONS

## 5.1 Conclusions

Current performance (characteristic data) of the engine is not available since UTP does not have dynometer to measure the latest / current engine characteristic. Therefore, the performance analysis is done under maximum power delivers by the engine with the 90% efficiency as for correction. Below are the achieved performances of the design race car with HONDA CBR 600 F4:

i. Final drive ratio (FDR) obtained = 3.377
   a. The value is slightly lower from the previous project done by UTP student which give 3.039 for the FDR. Higher value of FDR gives better acceleration while lower value gives better top speed.

ii. Top speed of the design racing car = 165 km/h (approximation)

iii. Changing gear speed for
   a. First to second gear = 42 km/h
   b. Second to third gear = 62 km/h
   c. Third to forth gear = 78 km/h
   d. Forth to fifth gear = 102 km/h
   e. Fifth to sixth gear = 128 km/h

The weight reduction for this project cannot be compared directly with the previous project since the drive mechanism of the project is dissimilar to the previous project. The total weight of the drive train components is about 16.536 kg where the design optimization succeed to reduce the components weight about 55%-65% of its initial weight. All designed components passed the static stress analysis and safe to fabricate

## 5.2 Recommendations

Consideration for improvement of this project should be in the early stage in project design. It will beneficial the current design with better and more reliable product to fabricate. Thus, the author recommended improving the material selection of the design component since the analysis give high factor of increment before the component could fail. (Improve weight reduction). Material with lower yield strength (lower steel series) can be used and it cost much cheaper compared to the current material selected.

Further study on the lubrication system is required to increase the performance of the new drive train. Study on the lubrication method such as splash lubrication and oil bath lubrication must go deeper and come out with satisfying solution on what method shall be used. Design improvement of the components should done regularly in order to get better design with more weigh reduction can be achieved. In future, gear such as spiral bevel and hypoid gear should be used in turn to increase the performance of the system even though it is quite expensive.

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[8] Motorcycle drive shaft patent from http://www.uspto.gov

[9] Lubrication Of Gearing, Wlfried J. Bartz, English translation by A.J. Moore

# APPENDICES

## Appendix 1

Table A-1: Original Engine and Drive Train Specification for HONDA CBR 600

| | |
|---|---|
| Manufacture / Model | HONDA CBR 600 F4i |
| Bore and Stroke | 67.0 x 42.5 mm (2.64 x 1.670) inch |
| Compression ratio | 12.0:1 |
| Cylinder arrangement | 4 cylinders in line, inclined 31 degrees from vertical |
| Displacement | 599 $cm^3$ (36.5 $inch^3$) |
| Lubrication system | Forced pressure |
| Cooling system | Liquid cooled |
| Air filtration | Paper element |
| Max Power design RPM | 10 000 |
| Max Torque design RPM | 9000 |
| Carburettor type | PGM F1 (programmed fuel injection)<br>Throttle bore : 38 mm (1.5 inch) |
| Oil pump type | Trochoid |
| Fuel tank capacity | 18 litter |
| Valve train | Chain driven |
| Engine dry weight | 59 kg (130ibs) |
| Transmission | 6 speed sequential |
| Drive Type | Chain – 428M |
| Primary reduction | 1.822 |
| Final Drive Ratio | 2.875 (46/16) |
| 1st gear | (2.833) 23 mph |
| 2nd gear | (2.062) 32 mph |
| 3rd gear | (1.647) 40 mph |
| 4th gear | (1.421) 46 mph |
| 5th gear | (1.272) 52 mph |
| 6th gear | (1.173) 56 mph |

**Appendix 2**

Table A-2: Spiral bevel gear specification

| | Driven Gear (Ring Gear) | Driving Gear (Pinion) |
|---|---|---|
| Teeth Number, N | 48 | 14 |
| Pitch Diameter, PD | 9 in (228.6 mm) | 2.65in (66.67 mm) |
| Circular pitch, Pc | 0.59 in (14.96 mm) | 0.59n ( 14.96 mm |
| Diametral Pitch ,DP | 5.33 in (135.47 mm) | 5.33 in (135.47 mm) |
| Whole Depth, W | 0.354 in | 0.354 in |
| Cone Distance, E | 4.68 in (119.06mm) | 4.68 in (119.06mm) |
| Face Width, F | 1.56 in (39.68 mm) | 1.56 in (39.68 mm) |
| Pitch Angle | 73.74° | 16.26° |
| Module | 4.76 mm | 4.76 mm |
| Addendum, J | 0.09 in | 0.22 in |
| Dedendum, K | 0.26 in | 0.13 in |
| Backlash | 0.005 in → 0.007 in | |
| Addendum Angle | 1.13° | 2.76° |
| Dedendum Angle | 3.19° | 1.56° |
| Root Angle | 70.56° | 14.69° |
| Radial Dimension, X | 0.02 in | 0.21 in |
| Outside Diameter, O | 9.05 in | 3.08 in |
| Arc Thickness | 0.21 in | 0.21 in |
| Chordal Thickness, T | 0.21 in | 0.21 in |
| Chordal Addendum, J | 0.93 in | 0.23 in |
| Spiral Angle | 35° | 35° |
| Root Angle | 70.56° | 14.7° |

## Appendix 3

Table A-3: Tabulated data for Velocity and Tractive Effort for all six gears

| Torrque (ft-Ibs) | rpm engine | torque Nm | km/h | TE 1st gr | tr | km/h | TE 2nd gr | km/h | TE 3rd gr | km/h | TE 4th gr | km/h | TE 5th gr | km/h | TE 6th gr |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 20 | 3000 | 27.34 | 17.5 | 1592.0 | 50 | 21.2 | 1336.2 | 25.5 | 1109.0 | 31.3 | 902.0 | 35.0 | 807.4 | 42.2 | 744.6 |
| 26 | 3500 | 35.542 | 16.5 | 2069.6 | 50 | 21.3 | 1737.1 | 25.6 | 1441.7 | 31.5 | 1172.6 | 35.2 | 1049.6 | 42.4 | 967.9 |
| 27 | 4000 | 36.909 | 19.7 | 2149.2 | 50 | 25.3 | 1803.9 | 30.5 | 1497.1 | 37.5 | 1217.7 | 41.9 | 1090.0 | 50.5 | 1005.2 |
| 28 | 4500 | 38.276 | 21.7 | 2228.8 | 51 | 27.9 | 1870.7 | 33.6 | 1552.6 | 41.3 | 1262.8 | 46.2 | 1130.4 | 55.6 | 1042.4 |
| 28.5 | 5000 | 38.9595 | 24.0 | 2268.6 | 51 | 30.8 | 1904.1 | 37.2 | 1580.3 | 45.7 | 1285.4 | 51.0 | 1150.6 | 61.5 | 1061.0 |
| 29 | 5500 | 39.643 | 25.3 | 2308.4 | 51 | 32.5 | 1937.6 | 39.2 | 1608.0 | 48.2 | 1307.9 | 53.9 | 1170.7 | 64.9 | 1079.6 |
| 33 | 6000 | 45.111 | 28.4 | 2626.8 | 51 | 36.5 | 2204.8 | 44.0 | 1829.8 | 54.1 | 1488.3 | 60.4 | 1332.2 | 72.8 | 1228.6 |
| 36 | 6500 | 49.212 | 30.9 | 2865.6 | 52 | 39.8 | 2405.2 | 47.9 | 1996.2 | 58.9 | 1623.6 | 65.8 | 1453.3 | 79.3 | 1340.2 |
| 37 | 7000 | 50.579 | 32.8 | 2945.2 | 52 | 42.2 | 2472.0 | 50.9 | 2051.6 | 62.5 | 1668.7 | 69.9 | 1493.7 | 84.2 | 1377.5 |
| 38.5 | 7500 | 52.6295 | 36.1 | 3064.6 | 53 | 46.5 | 2572.3 | 56.0 | 2134.8 | 68.9 | 1736.4 | 76.9 | 1554.3 | 92.7 | 1433.3 |
| 39.5 | 8000 | 53.9965 | 38.4 | 3144.2 | 53 | 49.4 | 2639.1 | 59.6 | 2190.3 | 73.2 | 1781.5 | 81.8 | 1594.6 | 98.6 | 1470.5 |
| 39 | 8500 | 53.313 | 40.9 | 3104.4 | 53 | 52.6 | 2605.7 | 63.3 | 2162.5 | 77.9 | 1758.9 | 87.0 | 1574.4 | 104.8 | 1451.9 |
| 40 | 9000 | 54.68 | 43.0 | 3184.0 | 54 | 55.3 | 2672.5 | 66.7 | 2218.0 | 82.0 | 1804.0 | 91.6 | 1614.8 | 110.3 | 1489.2 |
| 41.5 | 9500 | 56.7305 | 45.1 | 3303.4 | 54 | 58.0 | 2772.7 | 69.9 | 2301.2 | 86.0 | 1871.7 | 96.0 | 1675.4 | 115.7 | 1545.0 |
| 42 | 10000 | 57.414 | 48.2 | 3343.2 | 55 | 62.0 | 2806.1 | 74.7 | 2328.9 | 91.8 | 1894.2 | 102.6 | 1695.6 | 123.6 | 1563.6 |
| 42.8 | 10500 | 58.5076 | 50.2 | 3406.9 | 56 | 64.6 | 2859.6 | 77.9 | 2373.2 | 95.8 | 1930.3 | 107.0 | 1727.8 | 128.9 | 1593.4 |
| 42 | 11000 | 57.414 | 53.0 | 3343.2 | 56 | 68.2 | 2806.1 | 82.2 | 2328.9 | 101.0 | 1894.2 | 112.9 | 1695.6 | 136.0 | 1563.6 |
| 41.5 | 11500 | 56.7305 | 56.7 | 3303.4 | 57 | 72.9 | 2772.7 | 87.9 | 2301.2 | 108.0 | 1871.7 | 120.7 | 1675.4 | 145.4 | 1545.0 |
| 41 | 12000 | 56.047 | 58.6 | 3263.6 | 57 | 75.4 | 2739.3 | 90.9 | 2273.4 | 111.7 | 1849.1 | 124.8 | 1655.2 | 150.4 | 1526.4 |
| 40 | 12500 | 54.68 | 60.8 | 3184.0 | 58 | 78.2 | 2672.5 | 94.2 | 2218.0 | 115.8 | 1804.0 | 129.4 | 1614.8 | 155.9 | 1489.2 |
| 38 | 13000 | 51.946 | 63.2 | 3024.8 | 59 | 81.4 | 2538.9 | 98.0 | 2107.1 | 120.5 | 1713.8 | 134.7 | 1534.1 | 162.2 | 1414.7 |
| 36 | 13500 | 49.212 | 65.3 | 2865.6 | 60 | 84.1 | 2405.2 | 101.3 | 1996.2 | 124.6 | 1623.6 | 139.1 | 1453.3 | 167.7 | 1340.2 |
| 34 | 14000 | 46.478 | 67.0 | 2706.4 | 60 | 86.2 | 2271.6 | 103.8 | 1885.3 | 127.6 | 1533.4 | 142.6 | 1372.6 | 171.8 | 1265.8 |

**Appendix 4**

Table A-4: Progress schedule (first semester)

| No. | Detail/ Week | 1 | 2 | 3 | 4 | 5 | 6 | 7 | | 8 | 9 | 10 | 11 | 12 | 13 | 14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Selection of Project Topic | | | | | | | | | | | | | | | |
| 2 | Preliminary Research Work | | | | | | | | | | | | | | | |
| 3 | Submission of Preliminary Report | | | | | | | | | | | | | | | |
| 4 | Seminar 1 (optional) | | | | | | | | | | | | | | | |
| 5 | Project Work | | | | | | | | | | | | | | | |
| 6 | Submission of Progress Report | | | | | | | | | | | | | | | |
| 7 | Seminar 2 (compulsory) | | | | | | | | | | | | | | | |
| 8 | Project work continues | | | | | | | | | | | | | | | |
| 9 | Submission of Interim Report Final Draft | | | | | | | | | | | | | | | |
| 10 | Oral Presentation | | | | | | | | | | | | | | | |

Suggested milestone

Process

Table A-5: Progress schedule (second semester)

| No. | Detail/ Week | 1 | 2 | 3 | 4 | 5 | 6 | 7 | | 8 | 9 | 10 | 11 | 12 | 13 | 14 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Project Work Continue | | | | | | | | | | | | | | | |
| 2 | Submission of Progress Report 1 | | | | | | | | | | | | | | | |
| 3 | Project Work Continue | | | | | | | | | | | | | | | |
| 4 | Submission of Progress Report 2 | | | | | | | | | | | | | | | |
| 5 | Seminar (compulsory) | | | | | | | | | | | | | | | |
| 6 | Project work continue | | | | | | | | | | | | | | | |
| 7 | Poster Exhibition | | | | | | | | | | | | | | | |
| 8 | Submission of Dissertation (soft bound) | | | | | | | | | | | | | | | |
| 9 | Oral Presentation | | | | | | | | | | | | | | | |
| 10 | Submission of Project Dissertation (Hard Bound) | | | | | | | | | | | | | | | |

Suggested milestone

Process

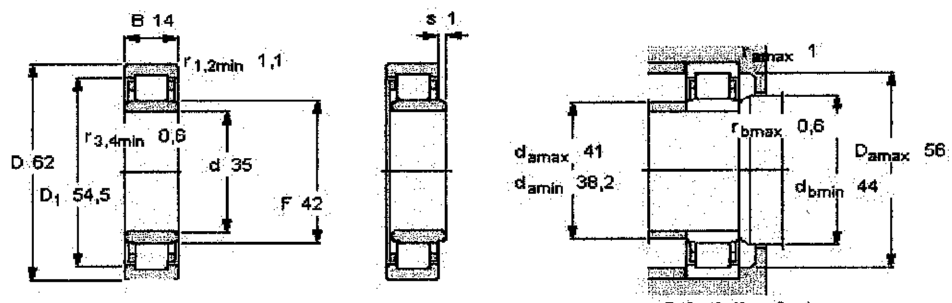
# Appendix 5

SKF

**Cylindrical roller bearings, single row**

Tolerances , see also text
Radial internal clearance, cy
Recommended fits
Shaft and housing toler

| Principal dimensions | | | Basic load ratings | | Fatigue load limit | Speed ratings | | Mass | Designation |
|---|---|---|---|---|---|---|---|---|---|
| | | | dynamic | static | | Reference speed | Limiting speed | | |
| d | D | B | C | $C_0$ | $P_u$ | | | | |
| mm | | | kN | | kN | r/min | | kg | - |
| 35 | 62 | 14 | 35,8 | 38 | 4,55 | 12000 | 13000 | 0,16 | NU 1007 ECP |




**Calculation factor**
$k_r$ 0,1

Figure A-1: SKF Bearing 38mm NU 1007 ECP

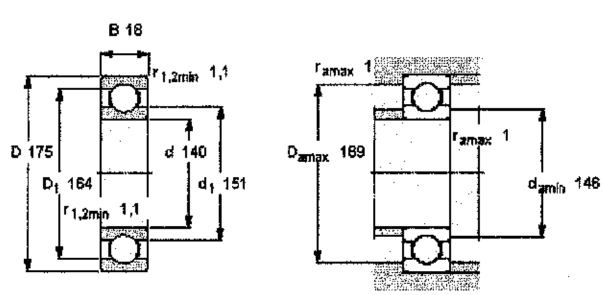
SKF

3D view | PDF | CAD

**Deep groove ball bearings, single row**

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

| Principal dimensions | | | Basic load ratings | | Fatigue load limit | Speed ratings | | Mass | Designation |
|---|---|---|---|---|---|---|---|---|---|
| | | | dynamic | static | | Reference speed | Limiting speed | | |
| d | D | B | C | $C_0$ | $P_u$ | | | | |
| mm | | | kN | | kN | r/min | | kg | - |
| 140 | 175 | 18 | 39 | 46,5 | 1,66 | 7500 | 4500 | 0,99 | 61828 |




**Calculation factors**
$k_r$ 0,015
$f_0$ 16

Figure A-2: SKF Bearing 140mm 61828