DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)

BY

Daniel Tena Ebrahim

Thesis submitted to department of Mechanical Systems and Vehicle Engineering

School of Mechanical, Chemical and Material Engineering

A thesis Submitted in partial fulfillment of the requirements for the award of degree of Masters of Science in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

September, 2021

Adama, Ethiopia

DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)

Daniel Tena Ebrahim

Advisor: Dr. Ramesh Babu Nallamothu
Co-Advisor: Dr. J. Bhaskaran

Thesis submitted to department of Mechanical Systems and Vehicle Engineering
School of Mechanical, Chemical and Material Engineering

A thesis Submitted in partial fulfillment of the requirements for the award of degree of Masters of Science in Automotive Engineering

Office of graduate studies
Adama Science and Technology University

September, 2021
Adama, Ethiopia

DECLARATION

I hereby declare that this Master Thesis "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" is my original work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

Name of the student	Signature	Date

RECOMMENDATION

We, the advisors of this thesis, hereby certify that we have read the revised version of the thesis entitled "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" prepared under our guidance by Daniel Tena Ebrahim submitted in partial fulfillment of the requirements for the degree of Masters of Science in Automotive Engineering.

Therefore, we recommend the submission of revised version of the thesis to the department following the applicable procedures.

______________________	______________________	______________________
Major Advisor	Signature	Date
______________________	______________________	______________________
Co-advisor	Signature	Date

APPROVAL SHEET

We, the advisors of the thesis entitled "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" and developed by Daniel Tean Ebrahim, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

______________________	______________________	______________________
Major Advisor	Signature	Date
______________________	______________________	______________________
Co-advisor	Signature	Date

We, the undersigned, members of the Board of Examiners of the thesis by Daniel Tena Ebrahim have read and evaluated the thesis entitled "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive engineering.

______________________	______________________	______________________
Chairperson	Signature	Date
______________________	______________________	______________________
Internal Examiner	Signature	Date
______________________	______________________	______________________
External Examiner	Signature	Date

Finally, approval and acceptance of the thesis is contingent upon submission of its final copy to the Office of Postgraduate Studies (OPGS) through the Department Graduate Council (DGC) and School Graduate Committee (SGC).

______________________	______________________	______________________
Head of Department	Signature	Date
______________________	______________________	______________________
School Dean	Signature	Date
______________________	______________________	______________________
Office of postgraduate studies dean	Signature	Date

ACKNOWLEDGEMENT

First and foremost, I would like to thank GOD Almighty for giving the strength, knowledge and ability, without his blessings, this achievement would not have been possible. Next, I would like to thank my advisor Dr. Ramesh Babu and co advisor Dr. J. Bhaskaran who has been there providing his support and guidance at all times. Without his able guidance, this thesis would not have been possible. In my journey towards this paper, I have found teachers, laboratory assistants and friends in Adama Science and Technology University, Mechanical Systems and Vehicle Engineering Department who have been so helpful and cooperative in giving their support at all times helps me to achieve my goal and at last but not least I really want to give a big shout to my beloved families who were beside me throughout my journey in the university life providing their unwavering moral and financial support that kept me going till the completion of this thesis work.

TABLE OF CONTENTS

CONTENT **PAGE**

LIST OF TABLES

LIST OF FIGURES

FIGURE PAGE

ACRONYMS

MG	Motor Generator
GM	General Motors Company
DC	Direct Current
AC	Alternate Current
PGU	Power Generating Unit
PSU	Power Storage Unit
CAD	Computer Aided Design
RBS	Regenerative Braking System
HEVs	Hybrid Electric Vehicles
HRB	Hydrostatic Regenerative Braking
DC valve	Direction Control Valve
DOA	Days of Autonomy
DOD	Depth of Discharge
VRAL	Value Regulated Lead Acid
CFRPs	Carbon Fiber Reinforced Polymers
HP	Horse Power
V	Voltage
3D	Three dimension
ABS	Antilock brake system
A	Ampere
VSE	Vehicle state estimator
NEDC	New European Driving Cycle
K.E.	Kinetic Energy
DIFF	Differential
ICE	Internal combustion engine
HIL	Hardware-in-the-loop
ECE	Economic Commission of Europe
VSE	Vehicle state estimator

SYMBOLS

b	Face width	mm
T_p	Pinion gear teeth number	-
T_F	Flywheel gear teeth number	-
N_p	Pinion gear revolution speed	rpm
N_F	Flywheel gear revolution speed	rpm
W_T	Tangential tooth load	N
C_s	Service factor	-
C	Deformation factor	-
W_S	Static tooth load	N
v	Velocity	m/s
W_D	Dynamic tooth load	N
C_E	Coefficient of fluctuation of energy	-
ΔE	Maximum fluctuation energy	Nm
m	Mass	kg
t	Thickness	mm
T	Torque	Nm
Te	Equivalent twisting moment	Nm
M_e	Equivalent bending moment	Nm
P_{system}	Air brake system operating pressure	Pa
CF	Correction factor	-
t_{charge}	Battery charge time	min
KE	Kinetic energy	J

I	Moment of inertia	kgm^2
$P_{flywheel}$	Flywheel power	W
F	Force	N
P_{max}	Maximum pressure	$N/_{mm^2}$
P_{min}	Minimum pressure	$N/_{mm^2}$
ΔT	Temperature difference across the object	℃
T_{max}	Maximum temperature	℃
T_{min}	Minimum temperature	℃
q	Heat flux, measured in	W/m^2

GREEK LETTERS

σ_e	Flexure endurance	Pascal (Pa)
μ	Coefficient of friction	-
σ_b	Bending stress	Pascal (Pa)
$\eta_{battery}$	Battery efficiency	%
η_{System}	System efficiency	%
σ_{ult}	Ultimate tensile stress	Pascal (Pa)
τ_{ult}	Ultimate shear stress	Pascal (Pa)
σ_{all}	Allowable stress	Mpa
τ_{yield}	Yield shear stress	Mpa
σ_{yield}	Yield stress	Mpa
σ_h	Hoop stress or circumferential stress	Mpa
σ_l	Longitudinal stress	Mpa
σ_1, σ_2	Maximum principal stress theory	Mpa
τ_{max}	Maximum shear stress	Mpa

ABSTRACT

Now a day environmental pollution is getting worse day to day and energy crisis that would implicate to the global economy. In a car the energy is waste, environmental pollution and fuel consumption is occurred during braking. Brake is a mechanical device used to slow or stop the vehicle with the smallest distance by converting the kinetic energy of the vehicle into the heat energy which is dissipated into the atmosphere. So, just in case of car one among these useful technologies is that the regenerative braking system. Regenerative braking system is an energy recovery mechanism converting its kinetic energy into electrical energy during braking which can be either used immediately or stored until needed. This thesis aims to design and analysis of the regenerative braking system for a city bus (Anbesa bus) by direct gear engagement of regenerative braking system to the engine flywheel using pinion gear. The overall design of the regenerative braking system involves material selection for the components and the design of subsystems. The subsystems of the regenerative braking systems are Mechanical subsystem as a power transmission throughout the system while being physically connected one another, Pneumatic subsystem use compressed air from air brake reservoirs during braking to actuate the clutch by using fork lever, and Electrical subsystem responsible for converting the transmitted rotational mechanical energy into electrical energy and store the generated energy for usage at the instance of need. The generated electrical energy by the regenerative braking system is used on the bus to provide an extra electric power source used to run specific bus electrical accessories to enhances the attractiveness of *the bus. In the process of the design relevant assumptions were taken, the component specifications and dimensions are determined, component failures were checked by using mathematically and Ansys software. mechanical simulation is done by using solid work 2017 software, simulation of pneumatic and electrical system is done by using pneumatic Festo and mutism software. Finally,* the total kinetic energy recovered by the system at 2300 rpm is 26095.6 K_J and the efficiency of the system is 84%.

Keywords: *Brake, Regenerative, Energy, Flywheel, gear*

CHAPTER ONE

1 INTRODUCTION

1.1 Background of the study

Energy is defined as the ability to do work. Matters consist of atoms and molecules in constant motion, all matter has energy but energy isn't matter, and it affects the behavior of matter. Everything that happens requires energy. Energy is a very important part of most aspects of everyday life. Energy can exist in numerous forms like thermal, mechanical, kinetic, potential, electric, magnetic, chemical, and nuclear forms (Y. A. Çengel, 2006). When energy is released to do work, it's called K.E. Kinetic energy may additionally be stated as energy in motion. Stored energy is called potential energy (Jack Erjavec,2010). Energy can neither be created nor be destroyed, it can transfer from one form to another form. For instance, P.E. of a boy on a slide change to K.E. This law brings lots of advantages to the boy because it helps the boy move from the highest of the slide to the bottom. However, the K.E. of the boy is also converted to heat as there's frictional force between the boy and therefore the slide. Hence, the boy's bottom might feel hot as he goes down the slide because of the heat energy. This can be a drawback to the boy. Now, apply the above concept above to the automotive world. A moving vehicle possesses K.E. whose value depends on the load and speed of the vehicle. The engine provides this energy so as to accelerate the vehicle from rest to given speed, but this energy must be partially or totally dissipated when the vehicle is slowed down or delivered to a standstill. Therefore, it's the function of the brake to convert the K.E. possessed by the vehicle at any one time into heat by means of friction (Heisler, H et.al,2002).

1.1.1 Conventional Braking System

Brake converts the momentum of the vehicle into heat by slowing and stopping the vehicle's wheels. This can be done by causing friction at the wheels (Jack Erjavec, 2010). When a standard vehicle applies its brakes, K.E. is converted to heat as friction between the brake pads and wheels. This heat is carried away within the airstream and therefore the energy is effectively wasted. The overall amount of energy lost during this way depends how often, how hard and for how long the brakes are applied. Regenerative braking refers to a process within which a little of K.E. is stored by a short-term storage system. The energy normally dissipated within the brakes is directed by a power transmission to the energy stores during deceleration.

That energy is held until required again by the vehicle, whereby it's converted into K.E. and accustomed accelerate the vehicle (Kumar, R., 2015). The magnitude of the portion available for energy storage varies according to the kind of storage, drive train efficiency, drive cycle and inertia weight. A lorry on the motorway could travel 100 miles between stops. This represents little saving whether or not the efficiency of the system is 100%. City center driving involves many braking events representing a much higher energy loss with potential savings. With buses, taxis, delivery vans and then on there is even more potential for economy. Since regenerative braking leads to a rise in output for a given input to a vehicle efficiency is improves. The quantity of the work done by the engine of the vehicle is reduce, in turn reducing the quantity of prime energy required to the vehicle. So as for a braking system to be cost effective the prime energy saved over a specified lifetime must offset the initial cost, size and weight penalties of the system. The energy storage unit must be durable and capable of handling high power levels efficiently, and any auxiliary energy transfer equipment must be efficient, compact and of reasonable cost (S.J clegg,2018). The entire amount of energy lost during this process depends on how often, how hard, how far and for how long the brakes are applied (Patil SK, 2012).

1.1.2 Regenerative braking system

Regenerative braking not only recharges the battery pack, but the load of turning the generator's rotor helps slow and stop the vehicle. Regenerative braking systems are not a physical part of the brake system. The system's controller regulates how fast the kinetic energy is converted, and thus regulates how fast braking will occur. Regenerative braking works better when the generator's rotor spins fast. It works poorly at low speed. Regenerative braking is found on all hybrid vehicles. It is one of the emerging technologies which may prove very beneficent. Regenerative braking in a vehicle not only ends up in the recovery of the energy but it also increases the efficiency of car and saves energy, which is stored within the auxiliary battery. This technology of regenerative braking controls the speed of the vehicle by converting some of the vehicle's K.E. into another useful style of energy (Wager et al., 2017).

The energy so produced could then be stored as electrical energy within the lead-acid accumulator, or as energy in flywheels, which might be used again by the vehicle. Regenerative braking refers to a process during which some of the K.E. of the vehicle is stored by a short-term storage system. Energy normally dissipated within the brakes is directed by a power transmission to the auxiliary battery during deceleration.

The energy that's stored by the vehicle, is converted into K.E. The magnitude of the portion available for energy storage varies according to the kind of storage, drive train efficiency, drive cycle and inertia weight (Wager et al., 2017). In order for a regenerative braking system to be cost effective the prime energy saved over a specified lifetime must offset the initial cost, size and weight penalties of the system. The energy storage unit must be compact, durable and capable of handling high power levels efficiently, and any auxiliary energy transfer or energy conversion equipment must be efficient, compact and of reasonable cost (Wager et al., 2017). The most common form of regenerative brake involves using an electrical motor as an electrical generator. The working of the regenerative braking system depends upon the working rule of an electrical motor, which is that the important component of the system, motor gets activated when some current is passed through it.

The recovered energy is used for recharging the battery efficient instead of first storing it within the battery, because it doesn't contain the conversion of energies. During recharging of battery, mechanical energy is converted into electrical energy and during discharging vice-versa. So, because of these conversions' transmission loses occur and therefore the efficiency reduces. As, within the other case, there aren't any transmission loses since mechanical energy stored within the flywheel is directly transferred to the vehicle in its original form. as the energy is supplied instantly and efficiency is high, these kinds of systems are utilized in F-1 cars (Bhandari P et al.,2017).

There are several advantages of regenerative braking system that would be listed down. Firstly, it improves fuel economy - dependent on duty cycle, powertrain design, control strategy and therefore the efficiency of the individual components. After that, emissions reduction - engine emissions reduced by engine decoupling, reducing total engine revolutions and total time of engine operation (engine on - off strategy). Then, performance of car which uses regenerative braking system improves. Furthermore, reduction in engine wear - engine on/off strategy (M. Naemah and Baharuddin, 2013). Even though there are many advantages of regenerative braking system, there are still several disadvantages that would be found within the regenerative braking system. One among the disadvantages of regenerative brakes is the car is heavier, compared to the car which used conventional braking system. This is because of both of the battery and therefore the motor is already heavy.

1.1.3 How well does regenerative braking work?

There are two parameters to evaluate regenerative braking,

- ✓ Efficiency
- ✓ Effectiveness.

1.1.3.1 Efficiency

No machine is often 100% efficient (without breaking the laws of physics), as any transfer of energy will inevitably increase some loss as heat, light, noise, etc. Efficiency of the regenerative braking process varies across many vehicles, motors, batteries and controllers, but is usually somewhere within the neighborhood of 60-70% efficient. Regen usually loses around 10-20% of the energy being captured, then the car loses another 10-20% or so when converting that energy into acceleration (Mercedes-Benz, 2020). This can be fairly standard across most electric vehicles including cars, trucks, electric bicycles, electric scooters, etc. Keep in mind that this 70% doesn't mean that regenerative braking will give an 70% range increase. Lost during the act of braking are often turned back to acceleration later (Electreka, 2018).

1.1.3.2 Effectiveness

The effectiveness of regenerative braking could be a measure of how much it can increase your range. The effectiveness of RBS varies significantly based on factors including driving conditions, terrain and vehicle size. Driving conditions have a large impact. The higher effectiveness for regenerative braking in stop-and-go city traffic than in highway commuting. This could make sense, if the brake is repeatedly applied, RBS recapture lots more energy than if simply drive for hours without touching the pedal. Terrain also plays a large role here too, as uphill driving doesn’t offer you much chance for braking, but downhill driving will regenerate a much larger amount of energy because of the long braking periods. On long downhills, regenerative braking is often used nearly constantly to manage speed while continuously charging the battery (Electreka, 2018).

1.1.4 Methods of Energy Conversion and Storage

Energy is converted to another form could be a very important consideration when coupling various technologies. For instance, flywheels may be used to store mechanical energy (kinetic), but the rate at which work are often converted to K.E. and therefore the subsequent discharge rate are limited by the inertia of the flywheel. The method of energy conversion can be divided into Direct (piezoelectric: mechanical ←→ electrical) and Indirect Conversion (Diesel cycle (gas): chemical ←→ thermal ←→ mechanical ←→ electrical).

Therefore, the conversion of K.E. in to electrical energy is performed by direct energy conversion mechanism. DC generator is an electrical machine which converts mechanical energy into DC electricity. Motor is that the key portion of regenerative braking system (Harshit Pandey,2020). There are multiple methods of energy conversion in RBS including spring, flywheel, electromagnetic and hydraulic (P. Clarke et al., 2010).

1.1.5 Need for Regenerative Brakes

The regenerative braking system delivers variety of advantages over a car that only has friction brakes. In low-speed, stop- and-go traffic where little deceleration is required; the RBS can provide the majority of the overall braking force. This vastly improves fuel economy with a vehicle, and further increase the attractiveness of vehicles using RB for city driving. At higher speeds, too, regenerative braking has been shown to contribute to improved fuel economy by the maximum amount as 20% (Wager et al., 2017).

1.1.6 System overview

The regenerative braking system on the bus is used as an additional power generating unit. The generated power which is an electrical power is obtained from power generating unit (PGU) driven by the rotational output form the engine. This rotation is taken from the engine output only when the brakes are engaged and the recovered energy is then stored in the power storage unit (PSU) or batteries. Thus, the system has two main components:

1. The power generating unit (PGU): responsible for generating the electrical power to be stored during the braking action. Electricity is generated at the most electrical power plants by using mechanical energy to rotate the shaft of electromechanical generators. The mechanical energy needed to rotate the generator shaft can be produced electrical energy from the wastage kinetic energy during braking (Sector Policies and Programs Division Office of Air Quality Planning and Standards U.S. Environmental Protection Agency ,2010).
2. The power storage unit (PSU): responsible for storing the generated electrical power during braking for use at a convenient time by the operator. In the era of the rapid development of renewable energy, a major technological challenge is that the efficient storage of electrical energy (CSEE J, 2020). Among the energy storage devices, chemical batteries are increasingly used in the professional power industry (Tiba, S and van Rijnsoever, 2019). The desired characteristics of batteries are high energy density, high charging and discharging power, and a long-life cycle (Liu, et. al. 2020).

1.2 Statement of the problem

Now a day environmental pollution is getting worse day to day and energy crisis that would implicate to the global economy. In cars (vehicles) in general there are a number of moving parts that the mechanical energy generated by engine will pass through. Due to their efficiency rating they cannot transmit the overall energy throughout the system and also when the car moves forward or backward it needs to apply brake due to different conditions, in this case the developed kinetic energy is waste as a heat to the atmosphere.in this case the environmental pollution and fuel consumption of the vehicle is increased. Anbesa bus is one of the transportation buses used in Ethiopia (Addis ababa). It works for short, medium and long distances. Due to traffic jam, road conditions, and loading and unloading of the passenger it applies brake repeatedly to slow the speed or completely stop the bus. Therefore, this leads to increase in fuel consumption, energy wastage and environmental pollution. For improving the fuel economy of the vehicle, to reduce environmental pollution and to enhance the attractiveness of vehicles, regenerative braking is required.

1.3 Objectives

1.3.1 General objective

- ✓ The general objective of this thesis work is to design and simulate direct gear engagement to the flywheel regenerative braking system which helped to recover the waste energy and reuse for Anbesa bus.

1.3.2 Specific objectives

- ✓ Conceptualize the basic design of regenerative braking system.
- ✓ Design of the components of the regenerative braking system.
- ✓ Model RBS using CAD tool and numerically analyzed the system to execute appropriate result.
- ✓ Design and simulate the pneumatic and electrical Circuit to illustrate the basic working principle of the RBS

1.4 Significance

The purpose of this thesis work is to design the system for recovering waste energy by direct gear engagement with the flywheel which otherwise gets wasted during braking. The recovered energy which is to be stored in a separate battery as an electrical energy source to be used in the time of need. The stored energy is used to run electrical components of the Anbesa bus. Since, bus drivers and passenger will be spending time in the bus, waiting for loading and moving, there could be the need to run the AC fans in hot and humid condition, listen to the radio for entertainment and for other things the vehicle systems need electrical power for which the energy is provided form the battery storage. Hence, the system reduces the fear of depleting or damaging the batteries and increase life time of the bus stock lead-acid batteries and increase the performance of the engine.

1.5 Scope

The scope of this thesis s work includes the design of individual components of regenerative braking system, modeling of regenerative braking system by solid work. Simulation of mechanical system is performed by Solidwork, simulation of pneumatic system is by pneumatic Festo and simulation of electrical system is performed by Multism software for showing the working principles of regenerative braking system.

1.6 Limitation

- ✓ Being a self-sponsored candidate, the required material could not be processed due to shortage of fund.

1.7 Organization of thesis

This thesis is organized in to six chapters. Chapter one is an introduction which comprises introduction, statement of the problem, objectives including general and specific objectives, significance, limitation, scope of the work are covered. Chapter two presents a literature review; it covers reviewed literature about regenerative braking system and other content that are related with direct gear engagement to the flywheel regenerative braking system. Chapter three describes material and methods; it covers methods and material selection for design of RBS. Chapter four focused on design analysis of regenerative braking system (direct gear engagement RBS to the flywheel) which covers detail design of components of regenerative braking system, mathematical analysis of regenerative braking system and also the power flow RBS. Chapter five present result and discussion, which covers the result of RBS design and the output discussed. Chapter six is conclusion and recommendation, which covers the conclusion and recommendation and also the future works of regenerative braking system.

CHAPTER TWO

2 LITERATURE REVIEW

2.1 Introduction

Many researchers worked on the regenerative braking system to recover the waste energy during braking. This chapter reviews existing literature on regenerative braking system with the objective of revealing contributions to identifies the weakness or the gaps of the research works done. And other relevant information related to the direct gear engagement to the engine flywheel regenerative braking system is discussed.

2.2 Regenerative braking system

Regenerative braking systems for conventional vehicles are gaining attention as fossil fuels still be depleted. the main forms of regenerative braking systems include electrical and mechanical systems, with the previous being more widely adopted at this time. However mechanical systems are still feasible, including the possible hybrid systems of two energy recovery systems. A literature study was made to check the assorted energy recovery systems. These systems were compared based on their advantages and drawbacks with regards to energy storage, usage, and maintenance. based on the comparison, the foremost promising concept appeared to be one that combined the flywheel and also the pneumatic energy recovery systems. A CAD model of this hybrid system was produced to higher visualize the design. This was followed by analytical modelling of the energy recovery systems. The analysis indicated that the angular velocity had a very significant impact on the ability loss and energy efficiency. The results showed that the hybrid system can provide better efficiency but only operating within certain parameters (Ahn JK et al., 2009).

Zhang J et al. (2012) focused on maximization the efficiency of the regenerative braking system. Regeneration efficiency and brake comfort, three different control strategies, namely the maximum regeneration-efficiency strategy, the good-pedal-feel strategy, and the coordination strategy for regenerative braking of an electrified passenger car are researched in this paper. The models of the main components related to the regenerative brake and the frictional blending brake of the electric passenger car are built in MATLAB/Simulink. The control effects and regeneration efficiencies of the control strategies in a typical deceleration process are simulated and analyzed. Road tests under normal deceleration braking and an ECE driving cycle are carried out. The simulation and road test results show that the maximum-regeneration-efficiency strategy, which causes issues on brake comfort and safety, could hardly be utilized in the regenerative braking system adopted.

The good-pedal-feel strategy and coordination strategy are advantageous over the first strategy with respect to the braking comfort and regeneration efficiency. The fuel economy enhanced by the regenerative braking system developed is more than 25% under the ECE driving cycle.

Vishwakarma D et al. (2016) highlights energy crisis is one of the major challenges to the present world. The fossil-fuel resources are being depleted at an incredible rate because of their disproportionate consumption. This has put forth the widespread assumption that if resources are getting used at the current rate, the time is not any longer when all our resources will expire. Thus, there's a necessity to develop the technology that saves the energy from getting wasted. In the case of automobiles, energy recuperation can be done by introducing a Regenerative braking system. Whenever the brakes are applied, a significant amount of K.E. gets wasted within the form of heat energy as a result of friction between brake pads and rotor. Regenerative Braking System recovers K.E. as much as possible that's lost during the method of braking. It stores that energy and releases it under acceleration. This paper highlights the two different methods of recovering energy that generally gets wasted, by converting it into either electrical or mechanical energy. The kinetic energy gets converted into electrical energy with the assistance of an electric motor whereas the flywheel harnesses the kinetic energy by converting it into mechanical energy.

Nashit et al. (2016) focused on testing of regenerative braking capability of a Brushless DC The motor is designed then fabricated. This was discussing in detail the steps carried out in designing the test rig and its fabrication. The BLDC motor used in the project for the test rig is an industrial-grade BLDC Motor imported from China. The BLDC Motors are found to be more efficient at higher speeds and therefore the regenerative braking is more effectively applied only at higher speeds. It can be concluded from this intensive project that the regenerative braking cannot be used are the only way of braking in electric and hybrid vehicles and only small part of a vehicles K.E. is recovered.

Zhang et al. (2013) introduced an electrically controlled regenerative braking system integrated with an anti-lock braking system or traction control system of an electrified passenger vehicle. To achieve higher performance and efficiency of the regenerative braking system with stability low development cost and risk. It develops the layout of a regenerative braking system control strategy and algorithms for the commercial electric vehicle. The deceleration road test for a commercial electrical vehicle is done and the result shows both RBS and ABS is good in performance.

Jincui li et al. (2017) tried to get high braking energy recovery and to improve efficiency or performance of vehicle regenerative braking system. The main aim of this paper is to recover the waste energy during braking by using an electromagnetic mechanical coupling regenerative braking system. And make the simulation for longitudinal stability control of the system. Finally, the result shows the proposed ABS continuous stat control strategy based on new electromagnetic mechanical couple braking system improves the vibration of a system with good response of slip control performance, robustness, braking comfort, energy recovery, and driving range of the electric vehicle.

Varocky BJ et al. (2011) introduced two regenerative braking concepts namely serial and parallel are studied and implemented on an electrical vehicle. Also, some extent of interest is to seek out if any additional states are required from the TNO Vehicle state estimator (VSE) which might aid in regeneration. the proposed brake torque distribution strategy has been tested through the simulation on the New European Driving Cycle (NEDC) drive cycle and line braking scenario. Car has been taken to watch and adjust brake torque specified gunlock up is prevented and hence regeneration is un-interrupted. The research couldn't include any additional parameters to be added to VSE. However, it would be worthwhile to use VSE to realize a more accurate estimation of the braking force, which may aid in prolonging regeneration time and hence more energy recuperation. The results provide a decent case to speculate longer and money into developing serial regenerative braking because it clearly out-performs a parallel regenerative braking strategy. The simulation tests conducted during this research are for a longitudinal braking.

Zhang J et al. (2012) conducted this research on the regenerative braking control of an electrified bus equipped with an anti-lock braking system (ABS). The regenerative braking works simultaneously with a pneumatic ABS, thus liberating the remaining energy of the vehicle while its wheels tend to lock under an extreme brake circumstance. Based on one representative pneumatic ABS strategy and optimum control theory, the optimization for regenerative braking control is proposed, in which the frictional and regenerative brake forces are controlled integrally to obtain maximal available adhesion. The simulation results indicate that brake stability and performance on different roads profit from the optimization. Hardware-in-the-loop (HIL) tests are accomplished on the pneumatic braking system of an electrified bus. HIL tests validate the results of simulation and guarantee the advantage and reliability of the optimization. The adaptability of optimization to hardware and software of the brake controller is also ensured.

M. K. Yoong et al., (2010) worked on braking system for a car which is based on hydraulic braking technology. However, this traditional braking methodology causes a lot of energy wastage since it produces unwanted heat during braking. Thus, the invention of regenerative braking in electric car has overcome these disadvantages moreover it helps in save energy and provides higher efficiency for a car. In regenerative mode, the motor act as a generator, it transfers the kinetic to electrical energy to restore the batteries or capacitors. Meanwhile, the brake controller monitors the speed of the wheels and calculates the torque required plus the excessive energy from the rotational force that can be converted into electricity and fed back into the batteries during regenerative mode. The merits of regenerative braking over traditional braking are energy conservation, wear reduction, fuel consumption and more efficient in braking. Nowadays, the brilliant technology in automotive industry towards regenerative braking is improving. By using a flywheel and ultracapacitor with DC-DC converter it had enhance the regenerative performance. In this paper, the working principle and braking controller for the regenerative braking have been studied to promote the efficiency and realization of energy saving in the electric vehicle.

Gao Y et al. (2000) investigated on three different braking patterns for evaluating the availability of braking energy recovery. Therefore, recovering braking energy is an efficient approach for improving the driving range of EV, and therefore the energy efficiency of HEV. The results indicate that even without active braking control, a big amount of braking energy is recovered, and also the brakes don't need much change from the brake systems of conventional passenger cars.

Zhang Jingming et al. (2008) worked on the regenerative braking system existing in electric and hybrid electric vehicles. The system can restore the kinetic energy and potential energy, used during start and accelerating, into the battery through the electrical machine. The total brake force is composed of friction brake force on the front axle, friction brake force on the rear axle, and regenerative brake force when a vehicle is equipped with regenerative braking system brakes. A control strategy, parallel regenerative brake strategy, was proposed to resolve the distribution of the three forces. The parallel regenerative brake strategy was optimized on Saturn SL1 and then simulated under the urban drive cycle. As a result, through optimization, the parallel brake strategy is not only safe enough but also can restore the largest amount of the brake energy.

Gao, Yimin, and Mehrdad Ehsani, (2001) focused on the development of the electrically controlled braking system and anti-lock braking systems of front and rear wheels for an electrical and hybrid vehicle.

Both the systems are simulated and the performance is checked to know the functionality. When the electrical system is failing the system, it can function as a conventional man actuated braking system. The development of a control system controls the braking force on the rear wheel and front wheel, Mechanical braking, and regenerative braking has been developed, it's recovered or developed electrical energy is calculated. Finally, the performance of this system is perfect. Wager G et al. (2018) this research evaluates the energy gain from a regenerative braking system (RBS) in an exceedingly commercial electric vehicle. Measurements were conducted in a very controlled environment on an advertisement chassis dynamometer using international drive cycle standards. The energy recovery of the vehicle was modeled and therefore the output of the model was compared with results from the chassis dynamometer driving. The experiments were original as they coupled energy changes recovered and driving range due to the RBS settings with investigations into the time of use of the friction brake. Performance tests used two different drive cycle speed profiles and various RBS settings to compare energy recovery performance for a broad range of driving styles. The results show that because of reduced energy consumption, the RBS increased the driving range by 11–22% depending on RBS settings and therefore the drive cycle settings on the dynamometer. The results further showed that driving an EV with an RBS uses the friction brakes more efficiently, which can reduce brake pad wear. This has the potential to boost air quality because of reduced brake pad dust and reduces the upkeep costs of the vehicle. The findings were significant since they showed that friction time of use, a parameter neglected in RBS testing, plays a vital part within the efficient operation of an EV.

Grandone M et al. (2016) focused on the development of a braking control strategy that enables the most effective tradeoff between mechanical and regenerative braking on a hybridized vehicle. The research work is part of a project for the event of an automotive hybridization kit aimed at converting conventional cars into Through the Road hybrid solar vehicles. the most aspect of the project is the integration of state-of-the-art components (i.e., in-wheel motors, photovoltaic panels, batteries) with the development of an optimal controller for power management. a mild parallel hybrid structure is obtained by substituting/integrating the rear wheels with in-wheel motors and adding photovoltaic panels and a lithium-ion battery. A hybridizing equipment prototype, patented by the University of Salerno, is installed on a FIAT Grande Punto. A model useful for real-time braking control has been developed, starting from the vehicle longitudinal model and considering dynamic weight distribution in front and rear axles and related wheel slipping effects.

Different braking strategies have then been investigated, in order to maximize the advantages of regenerative braking.

P. Bhandari et al. (2017) flywheel is a component that is used to store energy so release the stored energy when needed for acceleration. A flywheel is a heavy, high-speed rotating disc that builds up kinetic energy because it spins. the quantity of energy stored depends upon how heavier it's and the way fast it rotates. Heavier weight and faster rotation end up in higher energy storage. the tactic of transmission of energy doesn't include the conversion of energies. As, during the recharging of the battery, mechanical energy is converted into electrical energy, and during discharging vice-versa. So, because of these conversions' transmission losses occur, and also the efficiency reduces. As, in the other case, there are no transmission losses since mechanical energy stored in the flywheel is directly transferred to the vehicle in its original form. As the energy is supplied instantly and efficiency is high, these systems are used in FORMULA 1 racing cars.

2.3 Research gap

Energy gets wasted in every type of motor vehicle during braking as a heat. Almost all researches are developing regenerative braking system for hybrid vehicles and electric vehicles that are small in size and use the drive electric motor as a generator which is not applicable for diesel engine vehicle. The system has more components and sophisticated structure, which is difficult to maintain and run on a diesel engine bus. In construction the flywheel has larger weight, increasing the un-sprung mass of the vehicle. There is a need to develop a simple and light weight regenerative braking system for diesel engine vehicle like Anbesa bus.

2.4 Kinetic energy recovering system (KERS)

KERS is a collection of parts which takes the K.E. of a vehicle under deceleration, stores this energy then release this stored energy back to the drive train of the vehicle, providing a power to that vehicle. For the driver, it's like having two power sources at his disposal, one of the power sources is that the engine while the other is that the stored K.E. kinetic recovery systems store energy when the vehicle is braking and return it when accelerating. During braking, energy is wasted because K.E. is mostly converted into heat energy or sometimes sound energy that's dissipated into the environment. Vehicles with KERS are able to harness some of this K.E and in doing so will assist in braking. By a correct mechanism, this stored energy is converted into K.E. giving the vehicle extra boost of power.

There are two basic kinds of KERS systems

1. Electrical and
2. Mechanical.

The main difference between them is within the way they convert the energy and how that energy is stored within the vehicle. Battery-based electric kinetic energy recovering system require variety of energy conversions each with corresponding efficiency losses. On reapplication of the energy to the driveline, the global energy conversion efficiency is 31–34%. The mechanical kinetic energy recovering system system storing energy mechanically in a rotating fly wheel eliminates the various energy conversions and provides a global energy conversion efficiency exceeding 70%, over twice the efficiency of an electrical system (G. Arun Deesh, 2017).

2.5 Brake and braking

To move a vehicle, an internal combustion engine must convert its heat energy to mechanical energy. This mechanical energy goes from the engine to the driving wheel tires by means of a system of connecting rods, shafts and gears. The final factor that moves a vehicle is the amount of traction its tires have on the road surface. Traction is the ability of a tire to grip the road surface on which it rolls. The vehicle's acceleration rate depends on the power the engine develops and the amount of traction the tires have on the road surface. Friction is the force which resists movement between two surfaces in contact with each other. To stop a vehicle, brake shoe linings are forced against the machined surfaces of the brake drums, creating friction. This friction produces heat. The engine converts the energy of heat into the energy of motion – the brakes must convert this energy of motion back into the energy of heat. Friction between brake drums and linings generates heat, while reducing the mechanical energy of the revolving brake drums and wheels. The heat produced is absorbed by the metal brake drums, which dissipate heat by passing it off into the atmosphere. The amount of heat the brake drums can absorb depends on the metal thickness of which they are made. When enough friction is created between brake linings and drums, the wheels stop turning. The final factor that stops a vehicle is not the brakes, but the traction between tires and road surface.

If a 200-horsepower engine accelerates a vehicle to 100 km/h in one minute, imagine the power needed to stop this same vehicle. Not only that, the vehicle might have to be stopped in an emergency, in as little as six seconds (just 1/10 of the time it took to reach 100 km/h). To stop a vehicle in 1/10 the time it takes to accelerate requires stopping power of 10 times the acceleration power – equivalent to 2,000 horsepower (Saskatchewan driver's licensing and vehicle registration, 2019).

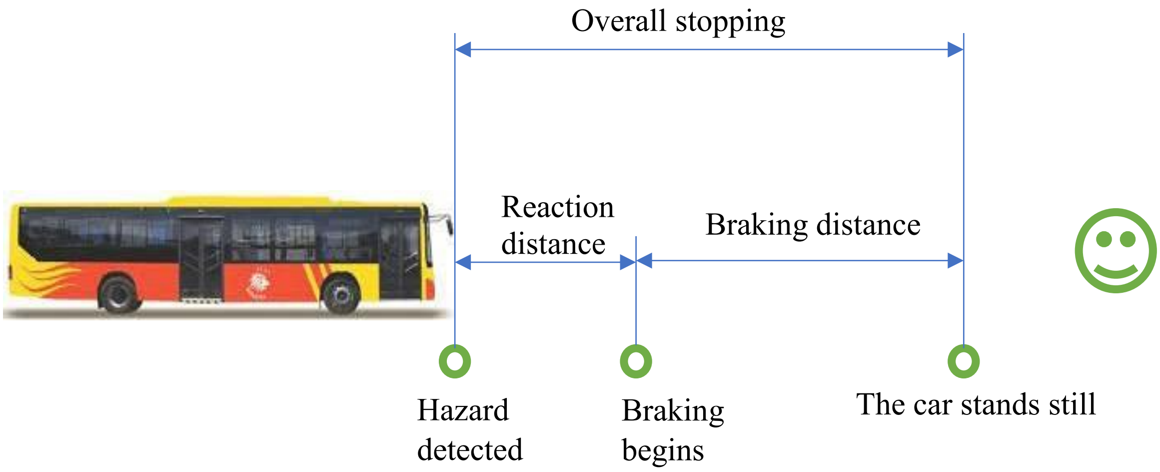


Figure 2.1. Vehicle stopping distance

2.5.1 Braking distance

Braking distance, the actual distance a vehicle travels after the brake is applied until the vehicle stops as shown in above Figure 2.1. This distance depends on the ability of the lining to produce friction. Heavy vehicles require powerful braking systems that are obtained by use of mechanical leverage and air pressure. Brakes must be used keeping in mind the heat generated by friction. If heat becomes too great, braking effectiveness will be lost. The heavier the load and the faster the speed, the greater the power needed to stop (Saskatchewan driver's licensing and vehicle registration, 2019). Therefore, the stopping distance is directly proportional to the energy wasting during braking. If the stopping distance is increased the energy wastage during barking is increased. According to the Law of Conservation of Energy, the heat energy could be converted to another energy which is certainly useful to the vehicle. Energy conversion occurs when one form of energy is changed to another. Because energy is not always in the desired form, it must be converted to a form that can be used (Jack Erjavec, 2010). Therefore, direct gear regenerative braking system is installed on the engine flywheel and it recovers the energy during the time of braking by engaging the system to the pinion gear that are freely rotate with the engine flywheel. By using RBS, the west energy during braking on the engine flywheel is recovered. The more the braking distance leads more engagement time of RBS and more energy is recovered.

2.6 Effect of Speed, weight and distance on braking

The distance required to stop a vehicle depends on its speed and weight in addition to the factors of energy, heat and friction. The brake power required to stop a vehicle varies directly with its weight and the "square" of its speed. For example, if weight is doubled, stopping power must be doubled to stop in the same distance. If speed is doubled, stopping power must be increased four times to stop in the same distance. When weight and speed are both doubled, stopping power must be increased eight times to stop in the same distance (Saskatchewan driver's licensing and vehicle registration, 2019).

2.6.1 Components of brake system

The five main components of an elementary air brake system (Saskatchewan driver's licensing and vehicle registration, 2019) and their purposes are:

1. Compressor: to build up and maintain air pressure
2. Reservoirs: to store the compressed air
3. Foot valve: to draw compressed air from reservoirs when it is needed for braking
4. Brake chambers: to transfer the force of compressed air to mechanical linkages
5. Brake shoes and drums or brake rotors and pads: to create the friction needed to stop the vehicle

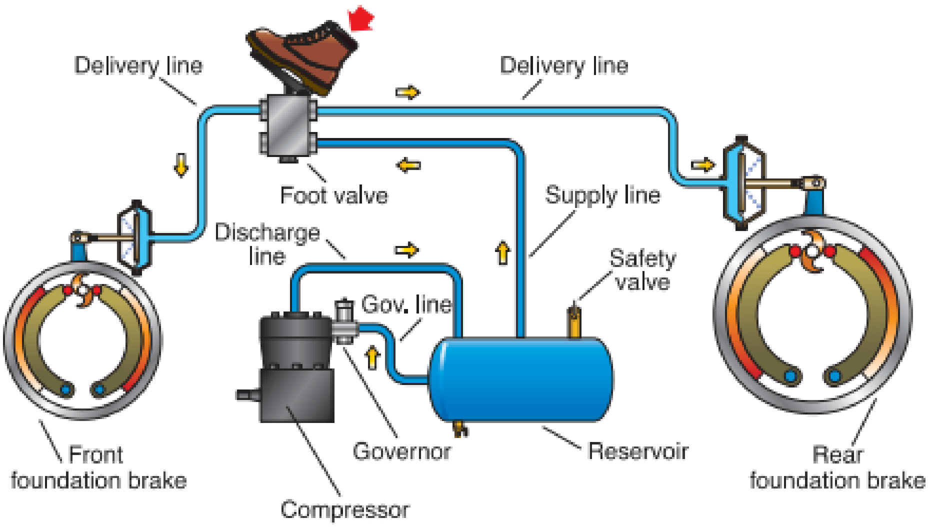


Figure 2.2. Components of air brake system (Commercial vehicle air brake)

2.7 System interpretation of direct gear engagement RBS to the flywheel

The complete RBS can be classified into three very different but strongly intertwined subsystems. These subsystems are:

i. Mechanical system
ii. Pneumatic system
iii. Electrical system

2.7.1 Mechanical system

The subsystem incorporates the components that are involved in the power transmission throughout the system while being physically connected one another. This system includes gear, clutch, shaft and flywheel.

- ❖ Gear

Gear is used to transmit torque and angular velocity in a wide variety of application (Robert L. Norton, 2006). Gears are normally mounted on a shaft, and they transmit rotating motion from one parallel shaft to another (Jack Erjavec, 2010). A gear is a toothed wheel that engages another toothed mechanism to change speed or the direction of transmitted motion. Gears are compact, positive-engagement, power transmission elements capable of changing the amount of force or torque. Gears are generally selected and manufactured using standards established by American Gear Manufacturers Association (AGMA) and American National Standards Institute (ANSI) (A.Bhatia, 2017). Most common types of gear spur gear, helical gear, bevel gear, worm gear.

Gears are generally used for one of four different reasons:

1. To increase or decrease the speed of rotation;
2. To change the amount of force or torque;
3. To move rotational motion to a different axis (i.e., parallel, right angles, rotating, linear etc.);
4. To reverse the direction of rotation.

- ✓ Spur gears

Spur gears are made with straight teeth mounted on a parallel shaft is shown in Figure 2.3 and its term is shown in Figure 2.4. Have teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another, parallel, shaft (Budynas−Nisbett ,2008). The noise level of spur gears is relatively high due to colliding teeth of the gears which make spur gear teeth prone to wear (Electro mate world press 2014). Spur gears come in a range of sizes and gear ratios to meet applications requiring a certain speed or torque output. So, for use in the RBS application spur gear construction is used for pinion this is because the vehicle flywheel ring gear that is stock is provided on the selected engine is spur gear type (Anaheim, 2016). The rule of thumb says that the smaller pinion gear should be harder than the much harder ring gear. This is because the smaller pinion turns many more times than the ring gear, and logic dictates that the pinion will probably wear out faster than the ring gear. To counter this effect, the surface hardness of the pinion is usually made harder than the ring gear (M.r. Adler, 2016).

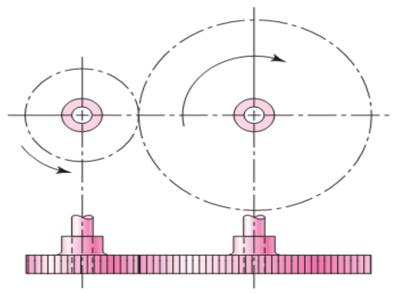


Figure 2.3. Spur gears between parallel shafts (Budynas–Nisbett, 2008 page 654).

Terms used in spur gear:

- Module (m): Is the ratio of the pitch diameter to the number of teeth.
- Pitch circle diameter: It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter.
- Pressure angle or angle of obliquity (φ): Is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point.
- Addendum: The radial distance between the top land and the pitch circle.
- Dedendum (b): Is the radial distance from the bottom land to the pitch circle.
- Diametral pitch (P): Is the ratio of the number of teeth on the gear to the pitch diameter.
- Clearance circle: Is a circle that is tangent to the addendum circle of the mating gear.

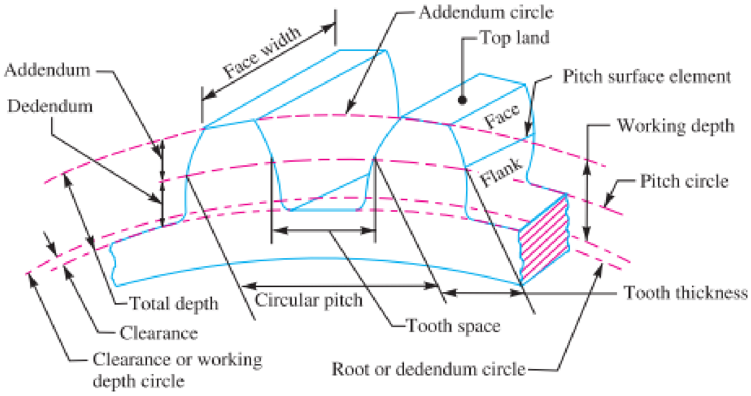


Figure 2.4. Terms used in spur gear (R.S. Khurmi and J.K. Gupta, 2005).

❖ Clutch

Clutch is a machine member used to connect a driving shaft to a driven shaft so that the driven shaft may be started or stopped at will, without stopping the driving shaft. The force of friction is used to start the drive shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces (R.S. Khurmi and J.K. Gupta, 2005).

- Shaft

Shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. In order to transfer the power from one shaft to another, the various members such as pulleys, gears etc., are mounted on it. These members along with the forces exerted upon them causes the shaft to bending. In other words, we may say that a shaft is used for the transmission of torque and bending moment. The material used for ordinary shafts is mild steel. When high strength is required, an alloy steel such as aluminum, nickel, nickel-chromium or chromium-vanadium steel is used. Shafts are generally formed by hot rolling and finished to size by cold drawing or turning and grinding (Budynas−Nisbett, 2008). Shafts are generally manufactured by hot rolling and finished to size by cold drawing or turning and grinding. The cold rolled shafts are stronger than hot rolled shafts but with higher residual stresses. The residual stresses may cause distortion of the shaft when it is machined, especially when slots or keyways are cut. Shafts of larger diameter are usually forged and turned to size in a lathe. Shaft has splines it may be internal and external. External splines broached, shaped, milled, hobbed, rolled, ground or extruded (R.S. Khurmi and J.K. Gupta, 2005).

- Flywheel

Flywheel is an inertial energy-storage device. It absorbs mechanical energy by increasing its angular velocity and delivers energy by decreasing its velocity. Flywheel is a heavy, high-speed rotating disc that builds up kinetic energy as it spins. The amount of energy stored depends upon how heavier it is and how fast it rotates. Heavier weight and faster rotation results in higher energy storage (Nagaraia.M, 2018). Flywheels are made from many different materials; the application determine the choice of material. Flywheels used in car engines are made of cast or nodular iron, steel or aluminum. By utilizing a properly designed flywheel, one or more of the following potential advantages may be realized (Jack A.Collins, 2010).

- ✓ Reduced amplitude of speed fluctuation.
- ✓ Reduced peak torque required of the driver.
- ✓ Reduced stresses in shafts, couplings, and possibly other components in the system.
- ✓ Energy automatically stored and released as required during the cycle.

- Bearing

A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load. A little consideration will show that due to the relative motion between the contact surfaces, a certain amount of power is wasted in overcoming frictional resistance and if the rubbing surfaces are in direct contact, there will be rapid wear. In order to reduce frictional resistance and wear and in some cases to carry away the heat generated; a layer of fluid (known as lubricant) may be provided. The lubricant used to separate the journal and bearing is usually a mineral oil refined from petroleum, but vegetable oils, silicon oils, greases etc., may be used (Robert L. Mott,2014).

2.7.2 Pneumatic system

The word 'Pneuma' means breath or air. Pneumatics is application of compressed air in automation. Pneumatic systems are power systems using compressed air as a working medium for the power transmission. An air compressor converts the mechanical energy of the prime mover into, mainly, pressure energy of the compressed air. After compression, the compressed air should be prepared for use. The air preparation includes filtration, cooling, water separation, drying, and adding lubricating oil mist. The compressed air is stored in compressed air reservoirs and transmitted through transmission lines: pipes and hoses. The pneumatic power is controlled by means of a set of valves such as the pressure, flow, and directional control valves. Then, the pressure energy is converted to the required mechanical energy by means of the pneumatic cylinders and motors (M.Galal Rabie,2009).

2.7.2.1 Basic Elements of Pneumatic Systems

A pneumatic system is a system that uses compressed air to transmit and control energy. Pneumatic systems are used extensively in various industries. Most pneumatic systems rely on a constant supply of compressed air to make them work. A pneumatic circuit is formed by various pneumatic components is shown in Figure 2.5, such as:

Compressor, Cylinders,

Directional control valves,

Check valves, Regulator

Gauges reservoir, actuator etc.

Pneumatic circuits have the following functions:

1. To control the injection and release of compressed air in the cylinders.
2. To use one valve to control another valve.

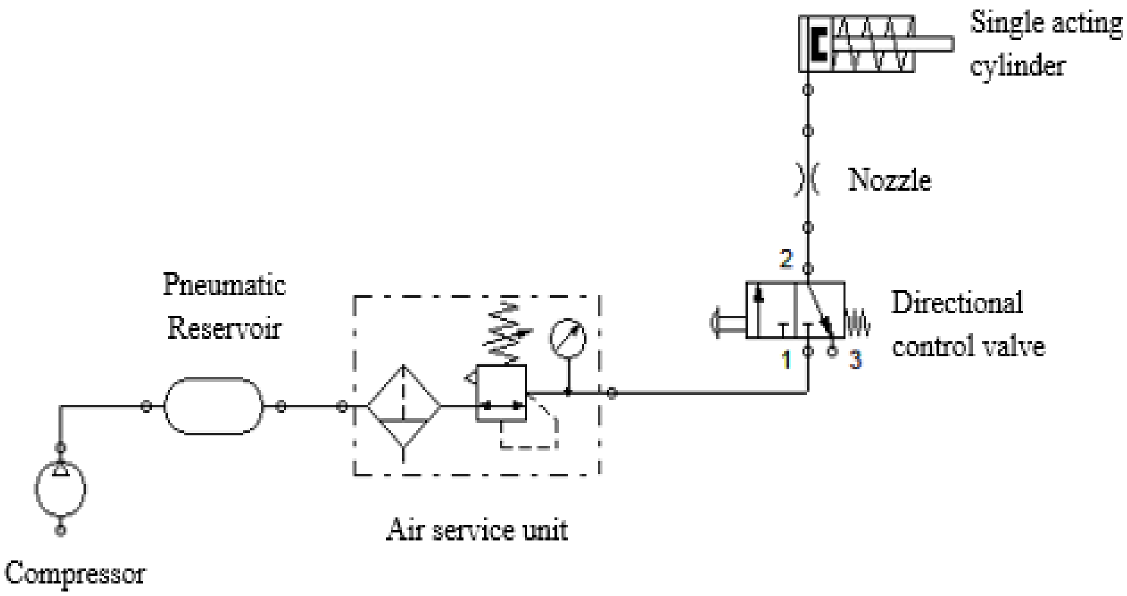


Figure 2.5. Circuit diagram of a simple pneumatic system (Festo fluid sim-p)

Air Compressors are pumps. Their purpose is to supply clean, dry air for power tools and industrial processes. They draw air in through the intake, compress it which raises its pressure and temperature, and then store it in a receiver. Air is conditioned by filtering it at the intake, by cooling it, and then by drying it to remove the water. Lubricating Oil which migrates downstream is removed to prevent problems in some applications (James A.Sullivan 2002).

Pneumatic Reservoirs: The air compressor serves to charge the compressed air reservoir. The compressor operation can be controlled by a governor to keep the air reservoir pressure within certain limits. The pneumatic system is directly fed from the reservoir (M.Galal Rabie,2009).

Air Filters: The solid particles and liquid droplets are removed from the compressed air by using air filters with either surface or depth filtering elements. The filtration process is carried out in two stages. The inlet guides force the inlet stream to rotate. The force acting on the solids and water separates them from the air stream. they're then collected within the lower part of the filter. in the second stage, a fine filter is added to separate additional impurities. Oil fog lubricators are used to lubricate the compressed air by adding a fine fog of oil using air lubricators (M.Galal Rabie,2009).

Pressure regulator: To stabilize the pressure and regulate the operation of pneumatic components is shown in Figure 2.6.

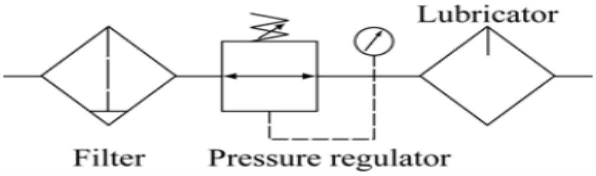
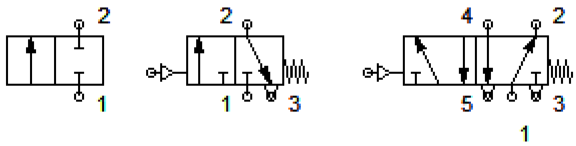


Figure 2.6. Pneumatic components within a pressure regulating component

Directional control valve: Ensure the flow of air between air ports by opening, closing and switching their internal connections is shown in Figure 2.7. Their classification is determined by the number of ports, the number of switching positions, the normal position of the valve and its method of operation. Common types of directional control valves include 2/2, 3/2, 5/2, etc. The first number represents the number of ports; the second number represents the number of positions (James A.Sullivan 2002).



(A) (B) (C)

Figure 2.7. 2/2 Way Valve (A), 3/2-way valve, pneumatically operated (B), 5/2-way valve, pneumatically operated (C) (FESTO fluid sim -P)

Pressure Reducers: Is positioned downstream of the high-pressure compressed air reservoir. It is used to control the pressure of compressed air supplied to a subsystem is shown in figure 2.8.

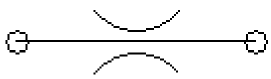
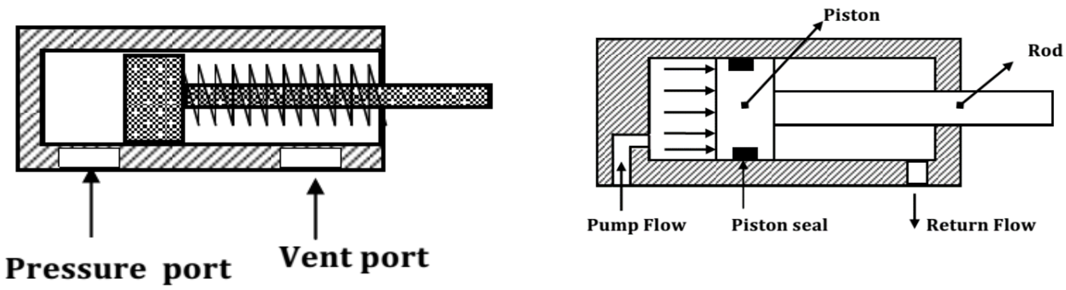


Figure 2.8. Nozzle (FESTO fluid sim -P)

Pneumatics actuators: There are the devices used for converting pressure energy of compressed air into the mechanical energy to perform useful work. In other words, Actuators are used to perform the task of exerting the required force at the end of the stroke or used to create displacement by the movement of the piston. The pressurized air from the compressor is supplied to reservoir. The pressurized air from storage is supplied to pneumatic actuator to do work. The air cylinder is a simple and efficient device for providing linear thrust or straight-line motions with a rapid speed of response. Friction losses are low, seldom exceeds

5 % with a cylinder in good condition, and cylinders are particularly suitable for single purpose applications and /or where rapid movement is required. Pneumatic cylinders can be used to get linear, rotary and oscillatory motion (M.Galal Rabie,2009).

Linear actuators: Pneumatic cylinders are devices for converting the air pressure into linear mechanical force and motion is shown in Figure 2.9. The pneumatic cylinders are basically used for single purpose application such as clamping, stamping, transferring, branching, allocating, ejecting, metering, tilting, bending, turning and many other applications (James A.Sullivan 2002). Based on cylinder action we can classify the cylinders as single acting and double acting. Single acting cylinders have single air inlet line. Double acting cylinders have two air inlet lines. Single acting cylinder compressed air is fed only on one side. Hence this cylinder can produce work only in one direction. The stroke length is limited by the compressed length of the spring and the piston moves forward in a single acting cylinder, air has to overcome the pressure of the spring and hence some power is lost before the actual stroke of the piston starts.



(A) (B)

Figure 2.9. Single acting cylinder (A) double acting cylinder (B) (M.Galal Rabie,2009).

- Advantages of pneumatic systems

Pneumatic control systems are widely used in our society, especially in the industrial sectors for the driving of automatic machines **(**Dhotare, 2019**)**. Pneumatic systems have a lot of advantages.

(i) High effectiveness

(ii) High durability and reliability

(iii) Simple design

(iv) High adaptability to harsh environment

(v) Safety

(vi) Easy selection of speed and pressure

(vii) Environmentally friendly and Economical

2.7.3 Electrical system

2.7.3.1 Role of Electricity in the Automobile

Electrical systems were basically stand-alone. For example, the ignition system was only responsible for supplying the voltage needed to fire the spark plugs. Ignition timing was controlled by vacuum and mechanical advance systems. Electrical system is used to run the AC fans in hot and humid condition, listen to the radio for entertainment and charge mobile phones for which vehicle electrical energy is play an important role (Introduction to Automotive electrical and electronic systems, 2019).

Vehicle charging system: The 'current' demands made by modern vehicles are considerable. The charging system must be able to meet these demands under all operating conditions and still 'fast charge' the battery. The main component of the charging system is the alternator and on most modern vehicles with the exception of its associated wiring this is the only component in the charging system. The alternator generates AC but must produce DC at its output terminal as only DC can be used to charge the battery and run electronic circuits. The output of the alternator must be a constant voltage regardless of engine speed and current load (Tom Denton, 2004).

Charging system consists of the following components:

Battery: The vehicle battery is used as a source of energy in the vehicle when the engine, and hence the alternator, is not running. The battery has a number of requirements, which are listed below broadly in order of importance (Tom Denton 2004).

- ✓ To provide power storage and be able to supply it quickly enough to operate the vehicle starter motor.
- ✓ To allow the use of parking lights for a reasonable time.
- ✓ To allow operation of accessories when the engine is not running.
- ✓ To act as a swamp to damp out fluctuations of system voltage.
- ✓ To allow dynamic memory and alarm systems to remain active when the vehicle is left for a period of time.

AC generator or DC generator: The DC generator provided direct current (DC) and was similar to an electric motor in construction. The biggest difference between a generator and a motor is the wiring to the armature. In a motor, the armature receives current from the battery. This creates the magnetic field that opposes the magnetic fields in the motor's coils, which causes the armature to rotate. The armature in a DC generator is driven by the engine.

It is not magnetized and the windings simply rotate through the stationary magnetic field of the field windings, inducing a voltage in the conductors inside the armature (Tom Denton 2004). A motor can become a generator by allowing current to flow from the armature instead of to it. In a DC generator, the placement of the brushes on the commutator changes the induced AC voltage to a DC voltage output. DC generators had a very limited current output, especially at low speeds. They could not keep up with demands of the modern automobile and were replaced by AC generators. AC generators are capable of providing high current output even at low engine speeds (jack Erjavec, 2010).

Voltage regulator: The output from an AC generator can reach as high as 250 volts if it is not controlled. The battery and the electrical system must be protected from this excessive voltage. Therefore, charging systems use a voltage regulator to control the generator's output. Voltage output is controlled by the voltage regulator as it varies the strength of the magnetic field in the rotor. Current output does not need to be controlled because an AC generator naturally limits the current output. To ensure that the battery stays fully charged, most regulators are set for a system voltage between 14.5 and 15.5 volts (jack Erjavec, 2010).

Charge indicator (lamp or gauge): Part of all charging circuits is some sort of charging system warning device or indicator. The sole purpose of these is to alert the driver when there is a problem. They are also one of the first things to check when a charging system problem is suspected. Vehicles have an ammeter, a voltmeter, an indicator light, and/or a message center in the instrument panel that allows the driver to monitor the charging system.

Indicator Light: This is the simplest and most common method of monitoring charging system performance. When the system fails to supply sufficient current, the light turns on (Jack Erjavec, 2010).

Cables and wiring harness: Battery Cables Battery cables must be able to carry the current required to meet all demands. Normal 12-volt cable size is 4 or 6 gauge. Various forms of clamps and terminals are used to ensure a good electrical connection at each end of the cable Connections must be clean and tight to prevent arcing and corrosion. The positive cable is normally red and the negative cable is black (jack Erjavec, 2010).

Charge controllers: The main function of a charge controller is to protect the battery by regulating the energy generation and consumption. This is done by limiting the voltage of the battery at charge and discharge. It is used to control over-charge of battery. Charge controllers must be sized to the system voltage, the maximum output current of the battery and the maximum load current. (DGS-Berlin, 2014).

CHAPTER THREE

3 MATERIALS AND METHODS

3.1 Introduction

Materials and methods are the most important part of this work of designing a regenerative braking system. Design procedure for direct gear engagement to the flywheel regenerative braking system is discussed clearly with the flow chart. The design engineer has a very large responsibility in the choice of materials to enhance the strength, lifetime, efficiency and performance and to minimize the overall cost of the system. All materials that are required for the regenerative braking system are selected by comparing the properties of the different materials that are related to the specific usage for each component of regenerative braking system. Therefore, in this chapter the design procedures (methods) and material selection based on their specific properties are discussed.

3.2 Design procedure

Design procedure of kinetic energy recovering mechanism during the time of braking is shown below in Figure 3.1.

1. Recognition: It is the complete description of the problem and analyzing the type of machine (system) to be designed. That means, know the problem and propose a solution.
2. Synthesis: In this step, proper component parts (elements or members) will be selected, designed and assembled in their right order of energy (power) flow.
3. Analysis, Material Selection, Size Determination and Optimization: Proper materials for each machine element or member has to be selected on the basis of types of loads, area of application, cost etc. Each component part (element or member) of the proposed kinetic energy recovering system subjected to various types of loads in0 the process of energy (power) transmission. The loads can be: mechanical, thermal. Hence, at this stage, the energy (power) to be transmitted and the loads acting on each component part have to be determined (calculated). Once the material is selected and load is determined then design each component properly, optimum sizes of each machine element or member can be determined. The stress analysis is to be and compared with the material mechanical properties, such as: yield stress (σ_{yled}), ultimate stress (σ_{ult}).
4. Revision and Modification: If it is necessary the size of each member and component materials has to be examined with reference to: failures, standards, availability of materials, strength, corrosivity, cost etc.

5. Design Report: In this stage, a complete and professional detail ‘2D’ and ‘3D’ part and assembly drawings together with relevant ‘models’ have to be prepared using ‘CAD software’s, and its system installation is prepared. The drawing should include all details of materials specification. And simulate the complete model using simulation software’s.

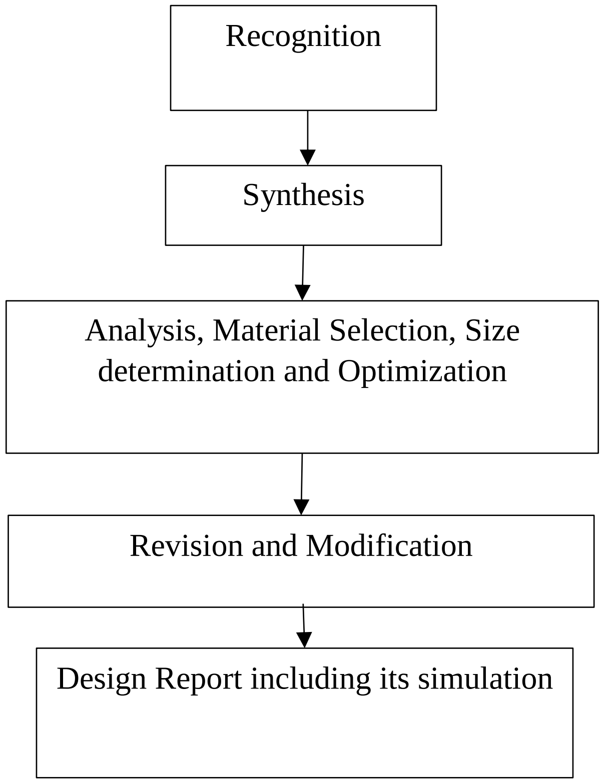


Figure 3.1. Design procedure

3.3 Anbesa bus

The Anbessa City Bus started as a share company founded in 1945 and owned by Emperor Haile Selassie and members of the royal family, before it was nationalized in 1974. It came to be a public enterprise only after it was re-established in 1994 (Anbesa bus official page). Anbessa has recently began to acquire buses assembled locally by the Metal & Engineering Corporation, a newly established military industrial complex of the Ethiopian government at bishoftu and its specification is shown on table A-13. Anbesa bus is one of the transportation buses in Ethiopia (Adis Ababa). and it works for short medium and long distance. Application of braking is very frequent due to loading and unloading of the passenger and also due to traffic jam. For such vehicles the wastage of energy by application of brake is about 60% to 65%. To improve the efficiency and attractiveness of this vehicle regenerative braking is required.

3.4 Direct gear engagement of RBS to the flywheel

A flywheel simply is a heavy rotating mass which stores kinetic or rotational energy as a function of its moment of inertia and angular velocity.

This system has a pinion consistently driven by the flywheel. The RBS engagement is performed by using a pneumatic clutch to harvest drive from the pinion shown in Figure 3.2.

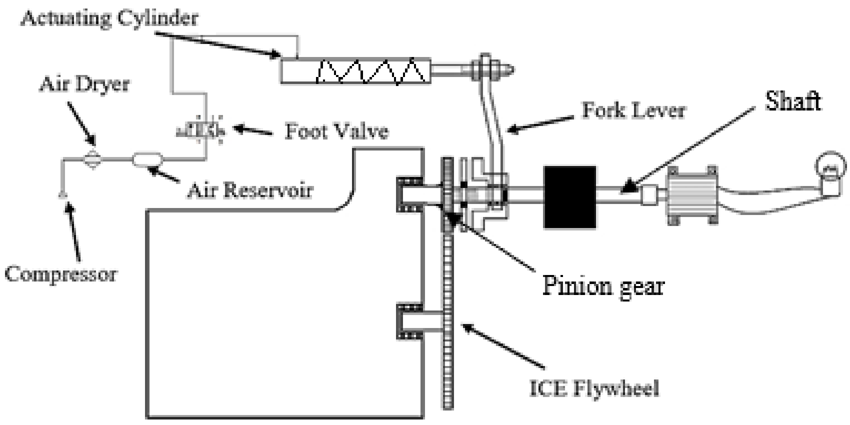


Figure 3.2. Direct Gear engagement of RBS to the flywheel

3.4.1 System and Component Concept Interpretation

The complete RBS can be classified into three very different but strongly intertwined subsystems shown in Figure 3.3. These subsystems are:

1. Mechanical system
2. Pneumatic system
3. Electrical system

The above stated subsystems further contain components that are used for power transmission, power generation, energy storage and component actuation/engagement.

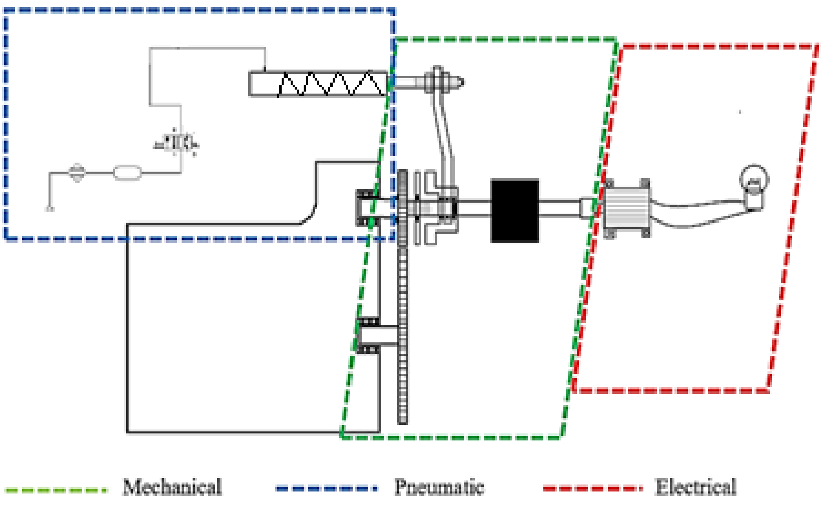


Figure 3.3. RBS Subsystem

3.4.2 Implemented operating concept

The whole system is constructed to be arranged horizontally one end emanating from the engine block shown in Figure 3.4. The shaft from the engine block carries the pinion which is consistently being driven by the flywheel of the vehicle IC engine. Then, from there on the clutch, fork lever, flywheel, splined shaft and the DC generator are arranged.

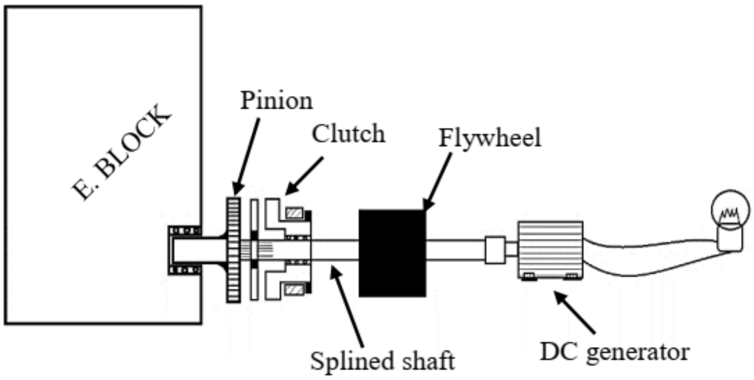


Figure 3.4. Assembled layout of the RBS

3.4.3 Operating principle

As stated in the above section 3.4.1, the system consists of three subsystems working together. The system can exist in two stages during operation:

a. System engagement
b. System disengagement

To demonstrate the two states, the following Figure 3.5 and Figure 3.6 can be used.

3.4.3.1 System engagement

- The first input to the system is the depression of the brake pedal. This action lets compresses air pass through a pneumatic DC valve to the actuating cylinder resulting in extending motion of the actuator.
- The extension of the pneumatic cylinder initiates the fork motion which pushes the engagement hub making it slide over the output shaft.
- The engagement hub continues its sliding motion until the friction plate is firmly sandwiched between it and the pinion.
- So, after the clutch is fully engaged the output shaft begins its rotation driving the DC generator.
- Since, the aim of the whole system is to drive the DC generator after the drive is provided the DC generator converts the transmitted rotational mechanical energy to electrical energy. Finally, the converted(generated) electrical energy is stored for later use to run different electrical components of the Bus.

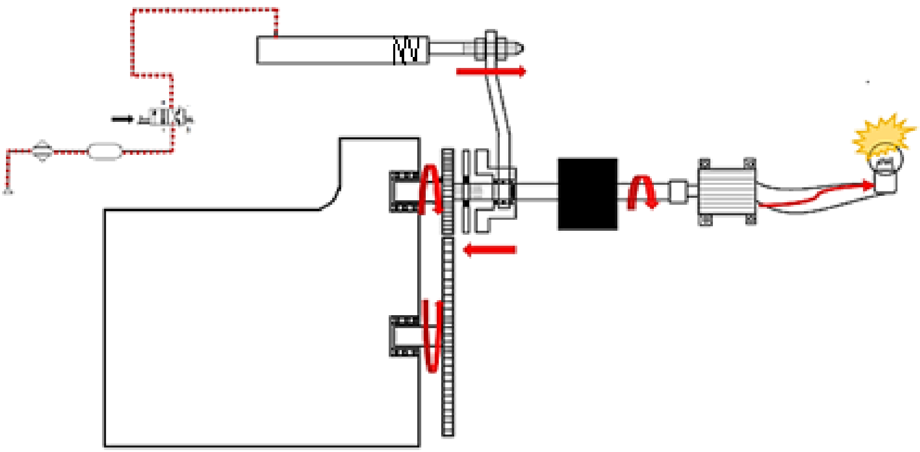


Figure 3.5. RBS engagement

3.4.3.2 System disengagement

- Same with the engagement the process begins from the brake pedal. But the brake pedal is not depressed rather released and the pneumatic DC valve returns to its initial position by spring action.
- During the position change of the DC valve the compressed air that was holding the actuating cylinder in extended position is let out to the atmosphere due to spring action to the other side of the actuating cylinder.
- The retraction motion of the cylinder initiates the fork lever movement inducing sliding movement on the engagement away from the pinion and disengaging the clutch.
- After disengaging the rotational motion to the DC generator comes to a halt gradually unless the brake is applied once more.

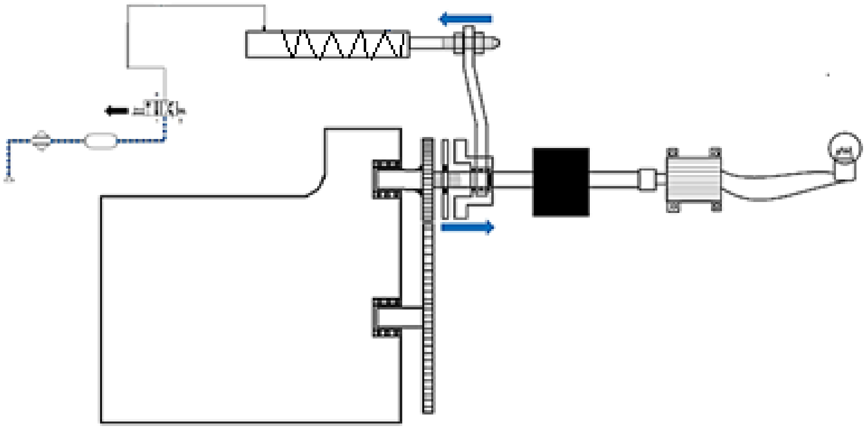


Figure 3.6. RBS Disengagement Cycle

3.5 Materials

3.5.1 Material selection

3.5.1.1 Gear (Pinion) material

The pinion gear will be subjected to fatigue due to repeated gear meshing, and in addition the hardness of the selected material determines the wear rate of the gear tooth (R.S. Khurmi and J.K. Gupta, 2005). The rule of thumb says that the smaller pinion gear should be harder than the much harder ring gear. This is because the smaller pinion turns many more times than the ring gear, and logic dictates that the pinion will probably wear out faster than the ring gear. To counter this effect, the surface hardness of the pinion is usually made harder than the ring gear. Thus, for the selection of pinion gear material the following properties must be considered:

- High tensile strength and good machinability
- Corrosion resistance
- Heat resistance
- High value of hardness number
- Cost of the material

Materials considered are:

- Cast alloy steel grade LC2-1
- Grey cast iron
- Stainless steel grade 410

Physical properties of cast alloy steel grade LC2-1, grey cast iron and stainless-steel grade 410 and its compression for the selected pinion gear material is shown in Table 3.1 and Table 3.2.

Table 3.1. Physical properties of selected pinion gear materials (Makeitfrom, 2021).

Material	Tensile stress (MPa)	Yielding strength (MPa)	Brinell hardness number (BH)	Cost ($/kg)
Cast alloy steel grade LC2-1	810	630	240	2.6
Grey cast iron	310	200	230	1.42
Stainless steel grade 410	1240	960	826	1.6

From the property considered for good gear material selection and the physical properties stated comparison is to be performed as follows by giving scores 1 to 3.

Table 3.2. Comparison of selected pinion gear materials

Parameter	Materials		
	Cast Alloy Steel Grade LC2-1	Gray Cast Iron	Stainless Steel 410
Cost	Most expensive (score = 1)	Cheapest (score = 3)	Medium (score = 2)
Tensile strength	Moderate (score = 2)	Lowest (Score = 1)	Highest (score = 3)
Galvanic Corrosion	Exposed to galvanic corrosion (score = 1)	No galvanic corrosion (Score = 3)	Galvanic corrosion may occur (score = 1)
Hardness	Moderate (score = 2)	Least hard (score = 1)	Hardest (score = 3)
Machinability	Hard (score =2)	Good (score =3)	Too hard (score = 1)
Over-all score			
Total	8	11	10

In order to avoid tooth breakage, the construction material for the pinion should have high Brinell hardness number. Thus, a material that is harder and corrosion resistant usage is a must; for which considered a Gray Cast Iron fulfills the requirements.

3.5.1.2 Clutch plate material selection

The clutch is consistently being pressed between two metal surfaces to transfer rotation between two machine components (R.S. Khurmi and J.K. Gupta, 2005). It can be used to transfer rotational motion from the pinion gear to the rest of the RBS.

To perform this task, the clutch plate material should have:

- High coefficient of friction,
- Low cost
- High operating pressure
- Minimal wear

Materials considered are:

- Sintered metal
- Rigid molded asbestos (dry)
- Rigid molded asbestos pad

Physical properties of sintered metal, rigid molded asbestos and rigid molded asbestos pad and comparison of selected clutch plate materials is shown in Table 3.3 and Table 3.4.

Table 3.3. Physical properties of the clutch materials (F. O. M. Sunny Narayan, 2018).

Material	Friction coefficient (μ)	Maximum pressure (Psi)	Cost ($/kg)
Sintered metal	0.33	400	0.2
Rigid molded asbestos (dry)	0.41	100	0.15
Rigid molded asbestos pad	0.45	750	0.25

Comparison of the selected materials is done as follows by assigning different parameters for all materials and scoring from 1 to 5. Then the material with the highest score will be chosen to be the construction material.

Table 3.4. Comparison of selected clutch plate materials

Parameter	Clutch plate materials		
	Sintered Metal	Rigid Molded Asbestos (Dry)	Rigid Molded Asbestos Pad
Cost	Intermediate (score = 4)	Cheapest of the three (score = 5)	Most expensive from the three (score = 3)
Operating pressure	Intermediate (score = 4)	Lowest (score = 3)	Has the best strength (score = 5)
Coefficient of friction	Lowest (score = 3)	Medium (score = 4)	Highest (Score = 5)
Overall score			
Total	11	12	13

Thus, from the comparison rigid molded asbestos pad is selected for clutch plate material. This is because even though it is the most expensive from the three; it makes up by providing high strength and high coefficient of friction.

3.5.1.3 Shaft material

Shaft is used to transfer rotational motion and torque from one place to another (Robert L. Norton, 2006). In RBS it transfers rotational motion from the pinion gear to the electric generator. In doing so, it is subjected to torsion from the rotation and bending from the weight of the RBS flywheel and the clutch assembly. So, the shaft material should be material of:

- High strength,
- Easy machinability
- Low cost
- Light weight

Material considered are:

- Aluminum alloy 6351 – T6
- Free cutting brass
- Carbon Steel alloy AISI 1040

Physical properties of aluminum alloy 6351 – T6, free cutting brass, carbon steel alloy AISI 1040 and comparison of selected shaft materials is shown in Table 3.5 and Table 3.6.

Table 3.5. Physical properties of the shaft materials (Makeit from, 2021)

Material	Yielding strength (MPa)	Shear strength (MPa)	Tensile strength (MPa)	Cost ($/kg)
Aluminum alloy 6351 – T6	270	207	310	1.9
Free cutting brass	310	210	469	8
Carbon steel alloy AISI 1040	415	350	620	2

Comparison of the selected materials by assigning different parameters for all materials and scoring from 1 to 5. Then the material with the highest score is chosen to be the construction material.

Table 3.6. Comparison of selected shaft materials

Parameter	Shaft materials		
	Aluminum Alloy 6351 – T6	Free Cutting Brass High strength low-Alloy Steel	Carbon Steel Alloy AISI 1040
Cost	The cheapest (score = 5)	Most expensive (score = 2)	Slightly more expensive than aluminum (score = 4)
Strength	Least (score = 2)	Moderate (score = 4)	Very high value (score = 5)
Light weight	Lightest	Moderately heavier	The heaviest

	(score = 5)	(score = 4)	(Score = 3)
Corrosion resistance	Excellent (Score = 5)	Moderate (Score = 4)	Easily corroding (score = 3)
Machinability	Easily machinable (score = 5)	Intermediate machinability (score = 4)	Very hard to machine (score = 3)
Overall score			
Total	22	18	`18

Thus, from the result shaft material should be highest score which is aluminum alloy 6351 – T6.

3.5.1.4 Flywheel material

During the material selection of the flywheel the following factors that are related to the energy storage, material cost and safety are considered (R.S. Khurmi and J.K. Gupta, 2005). The factors include:

- Material cost, material density, energy/per unit mass and operation speed limit

Material considered are:

- Cast iron
- High strength low-alloy steel
- Carbon fiber reinforced polymers (CFRP)

Table 3.7 below shows calculated values for mass, radius, operation speed, energy density, density, specific tensile strength and material cost of storing 500 J.

Table 3.7. Flywheel Material Properties (Makeit from, 2021)

Material	Mass (kg)	Density (kg/m^3)	Specific tensile strength (KJ/kg)	Operation speed (rpm)	Energy density (kWh/kg)	Cost ($/kg)
Cast iron	0.0166	7300	8-10	1467	0.0084	1.5
High strength low-alloy steel	0.004	7800	100-200	2,218	0.0316	13
CFRP	0.001	1550	200-500	3,382	0.1389	140

Note: specific tensile strength = $\frac{\sigma_t}{\rho}$, need to have high value for efficient storage.

From the above Table 3.7 the flywheel property for each material can be taken into consideration for the following comparison in Table 3.8. The comparison gives description and scores (1-5).

Table 3.8. Flywheel material comparison

Parameter	Flywheel materials		
	Cast Iron	High strength low-Alloy Steel	CFRP
Mass	Heaviest of the three (score = 3)	Second best choice for light weight construction (score = 4)	Lightest of the three (score = 5)
Operation speed	Too low for RBS (score = 3)	Best choice of speed range out of the three (score = 5)	Speed needs to be large for efficient storage (score = 3)
Radius (to generate 500 J)	Smaller radius (score = 5)	Moderate size (score = 4)	Not ideal for low-speed compact systems (Score = 2)
Cost	The cheapest (Score = 5)	Fair pricing (Score = 4)	Too expensive (score = 1)
Energy density and specific tensile strength	Lowest value (score = 1)	Fairly reasonable amount (score = 4)	The highest value (score = 5)
Overall score			
Total	17	21	16

Thus, from the comparison results, the high strength low-alloy steel has the highest score meaning that it will be used for flywheel construction material.

3.5.1.5 Alternator selection

Alternator has lower weight and efficient output is needed for restoring the braking energy. The other operation property is the speed of alternator. The alternator speed should be greater than the speed fed to it from the pinion gear to enhance the efficiency and long life of the system (Dabi Beyan, 2017).

Based on these criteria 33- SI heavy duty brushless alternator selected and the specification of 33- SI heavy duty brushless alternator is shown in A−14. The 33 SI series alternator is a brushless, heavy-duty integral charging system with built-in diode rectifier and voltage regulator, producing DC current for battery electrical systems.

And it is designed for use on large and mid-range diesel engines and gasoline engines in over the-road service, as well as for off-road, agricultural, and construction equipment. The maximum speed is up to 12000 Rpm. The 33SI alternator is a durable brushless model available in an output rating up to 110 Amps at 24 Volts (Heavy duty brushless alternator service manual, 2020).

3.5.1.6 Battery Selection

The choice for electrical energy storage the batteries taken into consideration were:

- Lead – Acid Battery
- VRLA Lead – Acid Battery
- Lithium – ion battery

Table 3.9. Battery Property Comparison for Selection (Mobbs, M., & Sussman, G., 2016).

	Flooded lead acid	VRLA lead acid	Lithium-ion
Energy Density (Wh/L)	80	100	250
Specific Energy (Wh/kg)	30	40	150
Regular Maintenance	Yes	No	No
Fast charge time	8-16 h	-	1h or less
Initial Cost ($/kWh)	65	120	600
Cycle Life	1,200 @ 50%	1,000 @ 50% DoD	1,900 @ 80% DoD
Typical state of charge window	50%	50%	80%
Temperature sensitivity	Degrades significantly above 25°C	Degrades significantly above 25°C	Degrades significantly above 45°C
Efficiency	100% @20-hr rate 80% @4-hr rate 60% @1-hr rate	100% @20-hr rate 80% @4-hr rate 60% @1-hr rate	100% @20-hr rate 99% @4-hr rate 92% @1-hr rate
Voltage increments	2 V	2 V	1.7 V

So, form Table 3.9 even though the lithium – ion battery is expensive it exceeds in performance, efficiency and weight comparison parameters. This exceeding performance makes it more suitable to be the electrical energy storage unit for the regenerative braking system being designed. Lithium – ion battery has a good power to weight ratio. The other most required property is the depth of discharge of the battery; for which the lithium – ion battery has the largest value making it more suitable.

3.5.1.7 Bearing selection

The ball and roller bearings consist of an inner race which is mounted on the shaft or journal and an outer race which is carried by the housing or casing. In between the inner and outer race, there are balls or rollers A number of balls or rollers are used and these are held at proper distances by retainers so that they do not touch each other. The retainers are thin strips and is usually in two parts which are assembled after the balls have been properly spaced. The ball bearings are used for light loads and the roller bearings are used for heavier loads (Robert L. Mott et.al.2014). Therefore, for this system 4-Bolt flange bearing and radial type ball bearings are selected. The material of the bearing is silver lead deposited because of excellent properties of fatigue strength, corrosion resistance, and conformability shown in Table A− 12. In direct gear regenerative braking system, the pinion gear is freely rotate by the engine flywheel so the pinion gear is attached to the engine with a little modification and it attached by using a 4-Bolt flange bearing shown below in Figure 3.6. At the end of the clutch housing, the system is supported by radial ball bearing. These bearing is used to support moving machine element. It permits a relative motion between the contact surfaces of the members, while carrying the load. Based on the diameter of the shaft the radial ball bearing and 4-Bolt flange bearing diameter are 30 mm and for compact design the width of the bearing is 16 mm is selected from Table A− 16.

To reduce friction and wear between the sliding parts of the bearing, to prevent rusting or corrosion of the bearing surfaces, to dissipate the heat, to protect the bearing surfaces from water, dirt etc., use a proper lubrication shown in Table A−15 (Robert L.Norton, 2006).

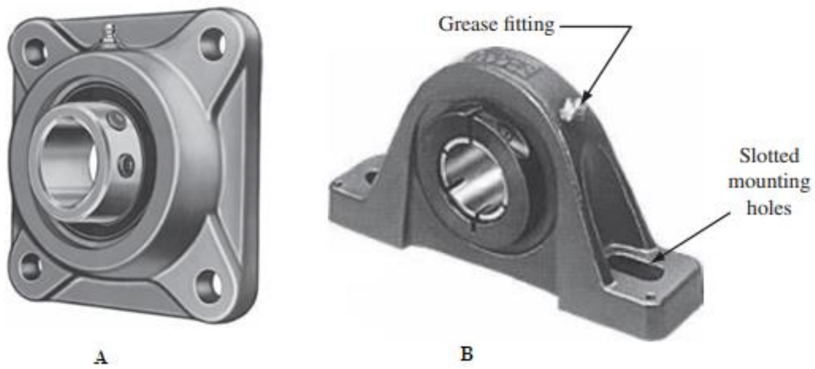


Figure 3.7. 4-Bolt flange bearing (A) and radial ball bearing (Robert L.Norton, 2006).

3.5.1.8 Fastener's selection

A fastener is any device used to connect or join two or more components. Literally hundreds of fastener types and variations are available. The most common are threaded fasteners referred to by many names, among them bolts, screws, nuts, studs, lag screws, and set screws. From different types of fasteners for this system bolts are selected to join two components. Bolt is a threaded fastener designed to pass through holes in the mating members and to be secured by tightening a nut from the end opposite the head of the bolt. Based on the size range the bolts are selected from Table A−17 for joining RBS to the vehicle body or other part.

3.6 Chapter summery

Materials and methods are the most important parts to design regenerative braking system. Design procedure for direct gear engagement to the flywheel regenerative braking system is discussed clearly using the flow chart. Power flow of the system was discussed using 2-D drawing and proper components and its materials are selected. All materials that are required for the regenerative braking system are selected by comparing the properties of the different materials that are related to the specific usage for each component of regenerative braking system.

CHAPTER FOUR

4 DESIGN ANALYSIS OF RBS

4.1 Introduction

Engineering design is a process of applying various techniques and scientific principles for the purpose of defining a device, a process or a system in sufficient detail to permit its realization. Design of a machine part is crucial to recognize that each parts function and performance. The ultimate goal of design is to size and shape the parts and choose appropriate materials and manufacturing process so that the resulting machine can be expected to perform its intended function without failure. For designing machines, it is required to analysis the force, moment, torque and stress of the components to know the functionality of the system (Robert L. Norton, 2006). In this chapter the design of direct gear engagement regenerative braking system to the flywheel is given in detail. According to the design steps the three-sub system of regenerative braking systems are clearly designed. The size, force, stress analysis etc. are determined, the failures of the component is checked to enhance the efficiency, life of the system. And the system installation based on the power flow order of the sub system of the direct gear engagement RBS to the engine flywheel is done.

4.2 Vehicle dynamics

Vehicle dynamics deals with the kinematics, kinetics, vibration, ..., of motor vehicles (automobiles).

- ✓ The maximum vehicle velocity (v_{max}) is limited by the maximum Power available at the driving wheels (P_{W-max}). The maximum engine (brake) Power (P_{b-max}) depends on the maximum Power available at the driving wheels (P_{W-max}).
- ✓ The maximum gradeability ($\emptyset_{max}$) and vehicle acceleration (a_{max}) depends mainly on the maximum vehicle tractive force (F_{tr-max}).
- ✓ The maximum vehicle tractive fornce (F_{tr-max}), in turn, depends on the maximum Torque available at the driving wheels (Tw-max) (Dabi Beyan, 2017).
- ✓ The maximum Torque available at the driving wheels (T_{w-max}), in turn, depends on the Power and angular speed available at the driving wheels. Power Flow of vehicle shown in Figure 4.1.

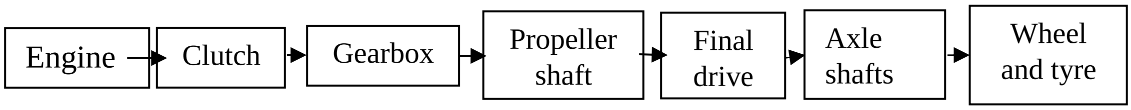


Figure 4.1. Power flow of vehicle

- Engine (Motor) is the 'power house' of a motor vehicle (automobile). The engine output is called the brake power (P_b).
- Clutch is used to connect and disconnect (engage and disengage) the engine to the rest of power train (driveline).
- Gearbox is used to change the gear ratios (GRi) or torque and angular speed of the engine depending on the driving conditions.
- Propeller shaft is used to connect the gearbox to the final drive (differential). The propeller shaft does not exist in Front-engine, Front-wheel-drive and Rear-engine, Rear-wheel-drive motor vehicles. In such vehicles, the differential is integrated with the gearbox as a unit which is called 'Transaxle'.
- Final drive is a kind of gearbox, used to adjust the speed of 'propeller shaft' with the speed of 'wheel and tire'. The differential is used to allow the drive wheels to rotate at different speeds so as to handle the motor vehicle in a curved path (road).
- Axle shafts are used to transmit power (torque and speed) from the final drive (differential) to the 'wheel and tire'.
- Drive wheels (Wheel and tire) are used to transform the engine power (torque and speed) into a tractive force (Ftr) which is used to drive or propel the vehicle.

The input and the output: power (P), torque (T) and angular velocity (ω) for each component part of a power train are illustrated below Figure 4.2.

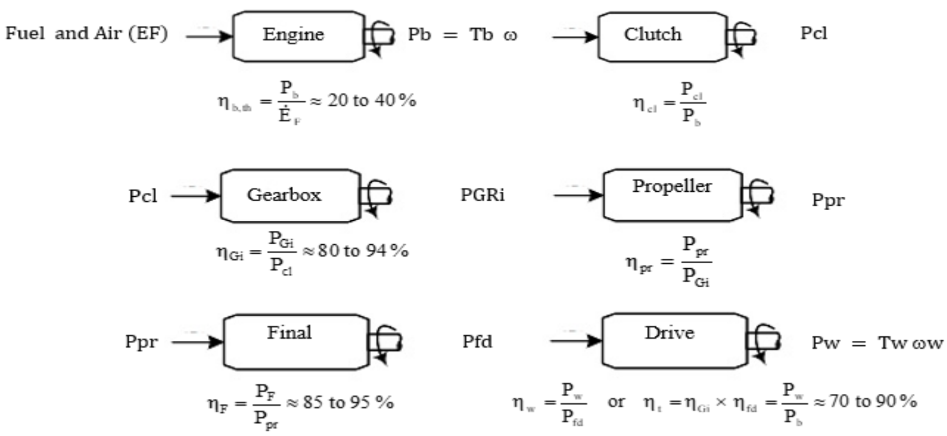


Figure 4.2. Power (torque and angular speed) transmission from the engine to the drive wheels.

4.3 RBS Component design

The design analysis consists of the design of:

- The subsystem components and the subsystems as a whole and checking for failure
- Choosing the appropriate material for the designed component and assigning.
- For the design analysis assumption of appropriate dimension and values were taken in some component design.

As stated under section 3.4.1, the whole system is comprised of three consequential sub-systems namely:

- Mechanical system
- Pneumatic system
- Electrical system

The design thus is focused on designing the components of the sub-systems and computing parameters needed for determining the amount of energy recovered by the system.

4.3.1 Mechanical system design

4.3.1.1 Pinion Gear (Spur gear) Design

The main drive for the RBS comes from the engine by receiving the drive from the engine flywheel using a pinion directly engaged with the flywheel. Figure 4.3. represents Pair of spur gears with pinion gear drives. The flywheel physical parameters are transferable to the pinion gear used for the power transmission to the RBS shown in Figure 4.4. The measurement taken from the physical flywheel of the Anbesa bus. So, the parameters transferable from the flywheel to the pinion gear are:

- The module m of the pinon will be the same as that of the engine flywheel which has the value of 3 mm.
- The construction material for the pinion is grey cast iron which is the same as that of the flywheel. This is to avoid failure during operation and reduce galvanic corrosion from having different material in contact with each other.
- The engine flywheel ring gear has $14\frac{1}{2}^{0}$ (pressure angles) full depth involute teeth which will be inherited by the pinon gear for construction.
- For proper fit the tooth face width (b = 4 cm) of the pinion and the engine flywheel is same and mass of Anbesa bus engine flywheel is 27 kg
- Pitch diameters of flywheel is 399 mm, outside diameter of flywheel is 405 mm

NB: The flywheel dimension is taken from actual measurement of Anbesa bus flywheel.

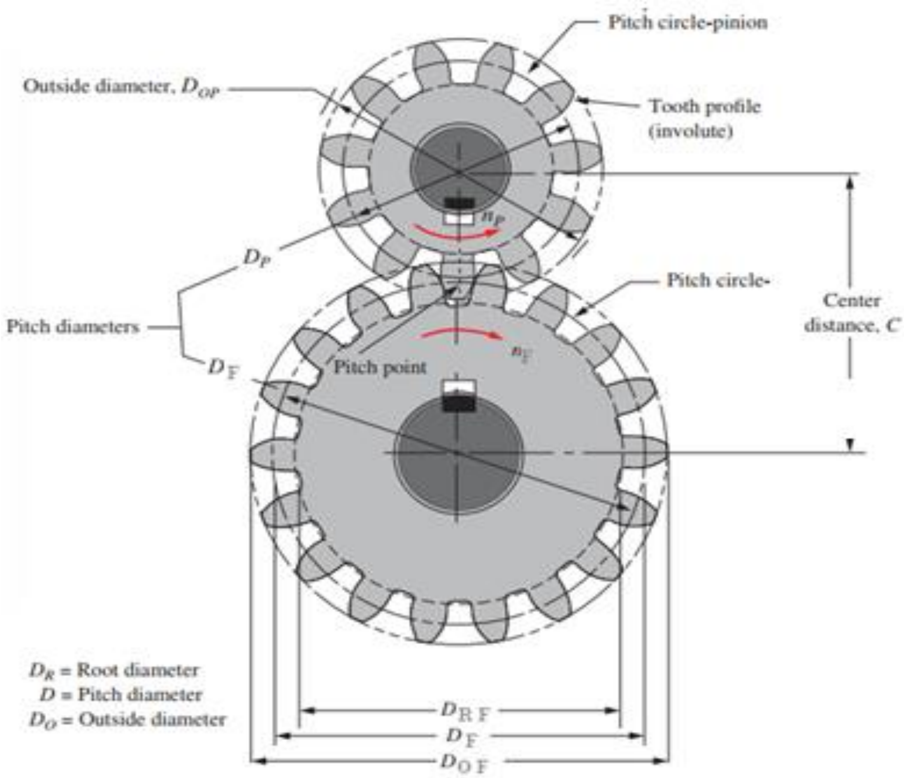


Figure 4.3. Pair of spur gears with pinion gear drives (Robert L. mott, 2016)

The rotation of the two gears are determine by considering the following two parameters (R.S. khurmi and J.K. gupta, 2005).

1. Rotation of the flywheel: It is determined from the manual of Anbesa bus which states the power output of the engine is 290 HP (213.295kW) @2300 rpm. For the analysis the flywheel rotation speed is 2300 rpm to avoid alternator over drive.
2. Rotation of the Pinion Gear: The rotation speed selection is based on the alternator output characteristics curve; which relates the rotation speed to the generator output. From Table A−14 an alternator producing 24 volts and 100 amps is selected as electric power generator. If an alternator generating 24 volts & 100 amps is selected it should produce 100% of the rated power at engine operating speed. For the analysis the 50% of the rated power is 24 volts and 50 amps which is at 2500 rpm of the Alternator.

Determining the pinion number of teeth: For a passenger car engine, idle speed is customarily between 600 rpm and 1000 rpm (Srinivasan, P et.al. 2010). The normal engine idling speed of bus is about 700 rpm (Zirker. L, et.al 2006). So, this analysis is done based on the idle speed of the engine which is 700 rpm. This is so that the alternator can be operated at low rpm of the idle engine speed to produce power.

$$\frac{N_P}{N_F} = \frac{T_F}{T_P} \qquad \text{... Eqn(3.1)}$$

Where: Flywheel rotation – $N_F = 700$ rpm

Pinion rotation – $N_p = 2500$ rpm

Pinion number of teeth – T_P

Flywheel number of teeth – $T_F = 133$ (Taken from the physical flywheel of the Anbesa bus)

$$\frac{2500}{700} = \frac{133}{T_P}$$

$$T_p = 37.24 \approx 38 \text{ teeth}$$

Thus, the pinion will have 38 teeth so that the RBS is able to recover energy at low rpm. Then using the calculated pinion teeth number, then calculate the alternator output at the normal engine operating speed. So, taking Eqn (3.1) and using the following known values the pinion rpm can be determined at normal engine operating speed of the engine to determine the alternator output.

$$\frac{N_P}{N_F} = \frac{T_F}{T_P}$$

Where: N_F= 2300 rpm, T_F = 133 teeth, T_P= 380

$$\frac{N_P}{2300} = \frac{133}{38}$$

$$N_P = 8050 \text{ rpm}$$

Thus, at engine normal operating speed the alternator will be run at 8050 rpm

The dimension of pinion gear is determined by using the relationship between module and equivalent diametral pitch P_d from Table A-11.

Then with module $m = 3$ mm the equivalent diametral pitch P_d is 8.466 teeth/in

Diametral pitch of pinion gear (D_p):

$$D_p = \frac{T_p}{P_d} \qquad \text{... Eqn(3.2)}$$

$$D_p = \frac{38}{8.466} = 4.4885 \text{ in} = 114 \text{ mm}$$

Outside diameter of pinion gear (D_o):

$$D_o = \frac{T_p+2}{P_d} \qquad \text{... Eqn(3.3)}$$

$$D_o = \frac{38+2}{8.466} = 4.7247\text{in} = 120 \text{ mm}$$

Center Distance between flywheel and pinion gear (C):

$$C = \frac{T_p - T_f}{2d_p} \quad \text{... Eqn(3.3)}$$

$$C = \frac{133 + 38}{2 \times 8.466} = 10.0992 \text{ in} = 256.52 \text{ mm}$$

Addendum (a):

$$a = \frac{1}{P_d} \quad \text{... Eqn(3.4)}$$

$$a = \frac{1}{8.466} = 0.18119\text{in} = 3 \text{ mm}$$

Dedendum (b):

$$b = \frac{1.25}{P_d} \quad \text{... Eqn(3.5)}$$

$$b = \frac{1.25}{8.466} = 0.147 \text{ in} = \ 3.75 \text{ mm}$$

Clearance (c):

$$c = \frac{0.25}{8.466} = 0.0295\text{in} = \ \ 0.75 \text{ mm}$$

Root diameter of pinion gear (D_{Rp}):

$$D_{Rp} = D_p - 2b \quad \text{... Eqn(3.6)}$$

$$D_{Rp} = 114 - 23.75 = 106.5 \text{ mm}$$

Teeth thickness (t):

$$t = \frac{\pi}{2P_d} \quad \text{... Eqn(3.7)}$$

$$t = \frac{\pi}{2 \times 8.466} = 0.1854 \text{ in } \ = 4.7 \text{ mm}$$

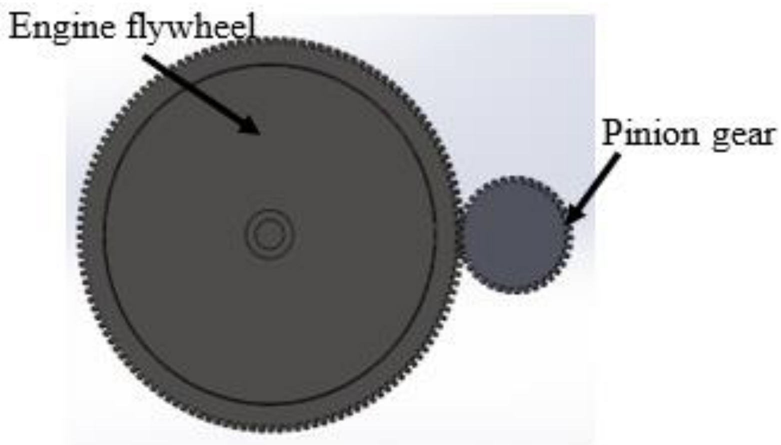


Figure 4.4. Modeling of flywheel and pinion gear assembly

The model of flywheel and pinion gear are shown in Figure 4.4. During service the pinion gear is subjected to loads that are static, wear and dynamic in nature. Hence, these forces are calculated and used to test for tooth failure. These loads include:

1. Tangential load (W_T):

$$W_T = \frac{P}{v} \times C_s \qquad \text{... Eqn(3.8)}$$

Where: W_T – Permissible Tangential tooth loads

P – Power transmitted, for which the value is taken from the engine output of the bus (290 HP or 213.295 K_W).

D− pitch circle diameter

C_S – Service factor = 0.8, it is selected from Table A–4 based on the intermittent operation and type of load (steady).

v − Pitch line velocity

$$v = \frac{\pi \times D \times N}{60} \qquad \text{... Eqn(3.9)}$$

$$D = m \times T_p \qquad \text{... Eqn(3.10)}$$

$$D = m \times T_p = 3 \times 47 = 114 \text{ mm}$$

$$v = \frac{\pi \times 0.114 \times 2300}{60} = 13.73 \ \frac{m}{s}$$

$$W_T = \frac{213.295}{13.73} \times 0.8$$

$$W_T = 12428 \text{ N} = 12..428 \ K_N$$

2. Dynamic load (W_D):

$$W_D = W_T + W_I \qquad \text{... Eqn(3.11)}$$

$$W_D = W_T + \frac{21 \times v[(b \times C) + W_T]}{21 \times v + \sqrt{[(b \times C) + W_T]}}$$

Where: W_I − Increment load due to dynamic action

b– tooth face width is the same as engine flywheel = 40 mm

C – Deformation factor

Determining the value of deformation factor (C): the value of C is determined from two tables (i.e., Table A–2 and Table A–3). it involves two steps:

I. Determining the value of maximum allowable tooth error in action (e): based on the calculated pitch velocity value (v = 13.73 m/s). Using Table, A–3 the value of e falls between 12.5 m/s and 13.75 m/s pitch line velocity (v). Interpolating the to obtain the value of e:

Pitch line velocity ($\mathbf{m/s}$)	Total tooth error
12.5	0.0300
13.73	x
13.75	0.0250

$$\frac{13.75 - 13.73}{13.75 - 12.5} = \frac{0.0250 - \mathrm{x}}{0.0250 - 0.0300}$$

$$\mathrm{e} = \mathrm{x} = 0.0251 \approx 0.02$$

Now, using the calculated value of maximum allowable tooth error in action (e) the deformation factor (C) can be determined from Table A–2. Also, for determining the deformation factor the material of the two-meshing gear is needed; for this case which is two cast iron gears are used having involute tooth form of $14\frac{1}{2}^{0}$ full depth involute.

Thus, the deformation factor (C) = 110 N-mm

$$\mathrm{W_D} = \mathrm{W_T} + \mathrm{W_I} = \mathrm{W_T} + \frac{21 \times \mathrm{v}[(\mathrm{b} \times \mathrm{C}) + \mathrm{W_T}]}{21 \times \mathrm{v} + \sqrt{[(\mathrm{b} \times \mathrm{C}) + \mathrm{W_T}]}}$$

$$\mathrm{W_D} = \mathrm{W_T} + \mathrm{W_I} = 12500 + \frac{21 \times 13.73[(40 \times 110) + 12428]}{21 \times 13.73 + \sqrt{[(40 \times 110) + 12428]}}$$

$$\mathrm{W_D} = 12428 + 11580.9$$

$$\mathrm{W_D} = 24009\ \mathrm{N} \cong 24\ \mathrm{K_N}$$

3. Static tooth load (W_S):

$$\mathrm{W_S} = \sigma_\mathrm{e} \times \mathrm{b} \times \mathrm{p_c} \times \mathrm{y} \qquad \ldots \mathrm{Eqn}(3.12)$$

Where: σ_e – Flexure endurance limit (Obtained from standard Table A–5 based on construction material). Therefore, since the construction material for the pinion gear is cast iron the Flexure endurance with Brinell hardness number 400 is 700 MPa.

Or $(\sigma_e) = 1.75 \times \mathrm{BHN}$

$$\sigma_\mathrm{e} = 1.75 \times 400 = 700\ \mathrm{Mpa}$$

$$\sigma_\mathrm{e} = 700\ \mathrm{MPa} = 700\ \mathrm{N}/_{\mathrm{mm}^2}.$$

Tooth form factor (y):

$$\mathrm{y} = 0.124 - \frac{0.684}{\mathrm{T_P}} \qquad \ldots \mathrm{Eqn}(3.13)$$

$$\mathrm{y} = 0.124 - \frac{0.684}{38} = 0.106, \qquad \text{For } 14^{1/2}{}^{\circ} \text{ stub system}$$

Circular pitch (P_c):

$$P_c = \frac{\pi \times D}{T_p} \qquad \text{... Eqn(3.14)}$$

$$P_c = \frac{\pi \times 114}{38} = 9.4 \text{ mm},$$

Tooth face width, b = 4 cm or 40 mm

$$W_S = 700 \times 40 \times 9.4 \times 0.106$$

$$W_S = 27899 \text{ N} = 27.899\ K_N$$

Wear tooth load:

The maximum load that gears teeth can carry, without premature wear, depends upon the radii of curvature of the tooth profiles and on the elasticity and surface fatigue limits of the materials.

$$W_w = D_p \times b \times Q \times K \qquad \text{... Eqn(3.15)}$$

Where: $W_w =$ Maximum or limiting load for wear in newtons

$D_p =$ Pitch circle diameter of the pinion in mm

$b =$ Face width of the pinion in mm

$Q =$ Ratio factor

$$Q = \frac{2 \times VR}{VR + 1} = \frac{2T_G}{T_G + T_P} \text{.... for external gear} \qquad \text{... Eqn(3.16)}$$

Velocity ratio (VR):

$$VR = \frac{T_G}{T_P} \qquad \text{... Eqn(3.17)}$$

$$VR = \frac{133}{38} = 3.5$$

$$Q = \frac{2 \times 3.5}{3.5 + 1} = 1.6$$

$K =$ Load-stress factor (also known as material combination factor) in N/mm^2

The load stress factor depends upon the maximum fatigue limit of compressive stress, the pressure angle and the modulus of elasticity of the materials of the gears. According to Buckingham, the load stress factor is:

$$K = \frac{{\sigma_e}^2 \times \sin\theta}{1.4}\left(\frac{1}{E_p} + \frac{1}{E_G}\right) \qquad \text{... Eqn(3.18)}$$

Where: θ = Pressure angle, ($14^{1/2°}$ full depth involute system)

E_p= Young's modulus for the material of the pinion in $N/_{mm^2}$, and

E_G= Young's modulus for the material of the gear in $N/_{mm^2}$

σ_e=Surface endurance limit in MPa or $N/_{mm^2}$

From Table A–5 values of flexural endurance limit at 400 B.H.N. is 700 MPa .

The flywheel (drive gear) and pinion gear material are gray cast iron. At high temperature Young's modulus is 48 GPa =48000 $N/_{mm^2}$

$$K = \frac{700^2 \times \sin 14^{1/2}}{1.4} \left(\frac{1}{48000} + \frac{1}{48000}\right)$$

$$K = 3.7$$

$$W_w = D_p \times b \times Q \times K$$

$$W_w = 114 \times 40 \times 1.6 \times 3.7$$

$$W_w = 26995.2 \text{ N} = 26.99 \text{ K}_N$$

For the pinion gear tooth to be safe from breakage static tooth load (W_S) and Wear toot load (W_w) should be greater than dynamic load (W_D).

So, from the obtained results W_S (27.899 K_N) and W_w(26.99 K_N) $>$ W_D (24 K_N) which implies that the pinion tooth is safe from breakage.

4.3.1.2 Flywheel Design

Used as a reservoir which stores energy during the period when the supply energy is greater than requirement during the period supply (supply energy > the requirement) and release it during the period when the requirement of energy is more. The flywheel need comes from the requirement of smooth and continuous energy recovery (R.S. Khurmi and J.K. Gupta, 2005). The flywheel brings the following attributes to the RBS:

i. Provide engine brake assistance when brakes are applied and the flywheel is run from rest. The braking effect might be small and is an additional effect that comes with the integration of flywheel in to the system.
ii. Provides smooth and continuous energy recovery during rapid intermittent braking instances by storing the kinetic energy eliminating the need to start the RBS that is coming to rest too quickly.

Design Consideration:

- Type of flywheel: Solid disc Flywheel
- Assume that the flywheel diameter does not exceed 100 mm to have compact design.

Specifications: N_1 = 8050 rpm, Maximum speed of alternator in rpm. during the cycle,

N_2 = 2500 rpm, Minimum speed of alternator in rpm. during the cycle, and

N = Mean speed in rpm, $= \frac{N_1+N_2}{2} = \frac{8050+2500}{2} = 5275 \text{ rpm}$

Determining the mean velocity of the flywheel (v):

$$v = \frac{\pi \times D \times N}{60 \times 1000} \qquad \ldots \text{Eqn}(3.19)$$

$$v = \frac{\pi \times 100 \times 5275}{60 \times 1000}$$

$$v = 27.6 \text{ m/s}$$

Determining coefficient of fluctuation of speed (C_s): The value of the coefficient to fluctuation of speed can be obtained from a standard Table A–6 on the basis of the type of power transmission. Thus, provided that the system has gear wheel transmission the value from the Table A–6 for the coefficient of fluctuation (C_s) is:

$$C_S = 0.02$$

Determining the maximum fluctuation energy (ΔE): the variations of energy above and below the mean resisting torque line.

$$\Delta E = mR^2\omega^2C_s = mv^2C_s \qquad \ldots \text{Eqn}(3.20)$$

So, to calculate the maximum fluctuation energy the following terms first need to be determined.

- Determining the Work/cycle:

$$\text{Work/cycle} = \frac{60P}{n} \qquad \ldots \text{Eqn}(3.21)$$

Where: n = Number of working strokes per minute.

n = N / 2, in case of four stroke internal combustion engines

From linear case we have $W_{done} = F \times s$, while we apply for rotational moment $W_{done} = T \times \theta$, where θ is crank angle with values of $\theta = 2\pi$ for 2 stroke engines and $\theta = 4\pi$ for 4 stroke engines.

Anbesa bus uses four stroke engines. Therefore, work done per cycle is denoted by:

$$\frac{\text{Work}}{\text{cycle}} = \frac{120P}{N} = T \times \theta = T \times 4\pi \qquad \ldots \text{Eqn}(3.22)$$

Where: $T = \text{Torque}$

$$T = \frac{60P}{2\pi N} \qquad \ldots \text{Eqn}(3.23)$$

$$T = \frac{60 \times 213.295 \times 10^3}{2\pi \times 8050} = 253 \text{ Nm} = 253 \times 10^3 \text{ Nmm}$$

$$\frac{\text{Work}}{\text{cycle}} = \frac{120P}{N} \quad \text{... Eqn(3.24)}$$

$$\frac{\text{Work}}{\text{cycle}} = \frac{120 \times 213.295 \times 10^3}{5275}$$

$$\frac{\text{Work}}{\text{cycle}} = 4852.21 \text{ Nm}$$

- Determining the coefficient of fluctuation of energy**:** it is defined as the ratio of the maximum fluctuation of energy to the work done per cycle. it is determined from standard Table A–7 based on the type of engine used to drive the flywheel. Thus, since the bus engine has a six-cylinder diesel engine the coefficient of fluctuation of energy (CE) will be:

$$CE = 0.031$$

Then,

$$\Delta E = CE \times \text{work done per cycle}$$

$$\Delta E = 0.031 \times 4852.21\text{Nm} = 150.42 \text{ Nm}$$

Determining the mass (m) of the flywheel: is calculated from the maximum energy fluctuation equation using the following rearranged equation.

$$m = \frac{\Delta E}{v^2 \times C_S} \quad \text{... Eqn(3.25)}$$

$$m = \frac{150.42}{27.6^2 \times 0.02} = 9.8 \text{ K}_\text{g}$$

Determining the width (b) of the flywheel: using the mass, density and volume relation the face width of the flywheel can be calculated.

$$b = \frac{4m}{\rho\pi D^2} \quad \text{... Eqn(3.26)}$$

Where: ρ – Density of flywheel material (High strength low-alloy steel) is 7700 ${}^{\text{K}_\text{g}}/_{\text{m}^3}$.

$$b = \frac{4 \times 9.8}{7700 \times \pi \times (\frac{100}{1000})^2} = 0.163 \text{ m} = 163 \text{ mm}$$

4.3.1.3 Shaft Design

A shaft is a rotating machine element which is used to transmit power from one place to another. The power is delivered to the shaft by some tangential force and the resultant torque (or twisting moment) set up within the shaft permits the power to be transferred to various machines linked up to the shaft. The shaft is considered to have been subjected to combined twisting and bending moments.

When the shaft is subjected to combine twisting moment and bending moments then the shaft must be designed on the basis of the two moments simultaneously (Robert L. Mott et al., 2018**).** In this system the shaft is used to transfer power from the clutch output to the alternator. The shaft will have supports on two placed to provide support for the flywheel and clutch assembly loads shown in Figure 4.5.

Design consideration:

- Flywheel mass (m_{FW} = 9.8 K_g)
- Mass of clutch assembly is assumed to be (m_c = 1.5 K_g)
- The shaft is assumed to be subjected to only vertical loading
- Material considered for design is aluminum alloy for light weight construction.
- The factor of safety for the shaft design is assumed to be F.S = 4 to have a reliable and durable shaft in operation.

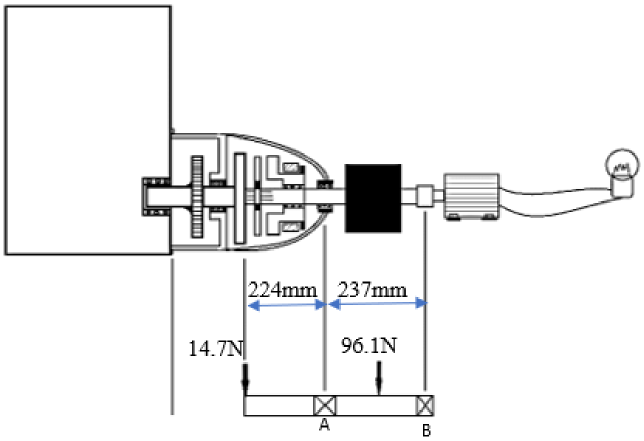


Figure 4.5. Shaft Loads and Dimensioning

Determining the twisting moment on the shaft:

$$\mathrm{T} = \frac{60\mathrm{P}}{2\pi \mathrm{N}}$$

$$\mathrm{T} = \frac{60 \times 213.295 \times 10^3}{2\pi \times 8050} = 253\ \mathrm{Nm} = 253 \times 10^3\ \mathrm{Nmm}$$

Determining the bending moment (vertical only):

For the vertical loading reaction at supports A & B will be:

$$\mathrm{R_A} + \mathrm{R_B} = 14.7\ \mathrm{N} + 96.1\ \mathrm{N} = 110.8\ \mathrm{N}$$

Taking the bending moment at B

$$\mathrm{M_B} = 0$$

$$\mathrm{M_B} = 14.7\ \mathrm{N} \times 461\ \mathrm{N} - \mathrm{R_A} \times 237\ \mathrm{mm} + 110.8\ \mathrm{N} \times 118.5\ \mathrm{mm}$$

$$\mathrm{R_A} = 83.9\ \mathrm{N}$$

To calculate reaction at B:

$$83.9\ \text{N} + \text{R}_\text{B} = 110.8\ \text{N}$$

$$\text{R}_\text{B} = 26.9\ \text{N}$$

Bending moment at location of loading (i.e., A & D):

- Bending moment at A

$$\text{M}_\text{A} = 14.7\ \text{N} \times 224\ \text{mm} = 3292.8\ \text{Nmm}$$

- Bending moment at D

Since, the flywheel is to be at the middle of the two supports the following relation will be used to calculate the bending moment at D.

$$\text{M}_\text{D} = \frac{\text{W}_\text{FW} \times \text{L}}{4} \qquad \text{... Eqn(3.27)}$$

Where: W_FW – Weight of the flywheel

L – Total length of the shaft.

$$\text{M}_\text{D} = \frac{96.1\ \text{N} \times 237\ \text{mm}}{4} = 5693.9\ \text{Nmm}$$

Maximum bending moment (M):

$$\text{M} = \text{M}_\text{D} = 5693.9\ \text{Nmm}$$

For design the maximum of the two bending moments is selected. Hence, bending moment at D (M_D) =5693.9 N-mm will be the design moment and the shear force and bending moment diagram is shown below in Figure 4.6.

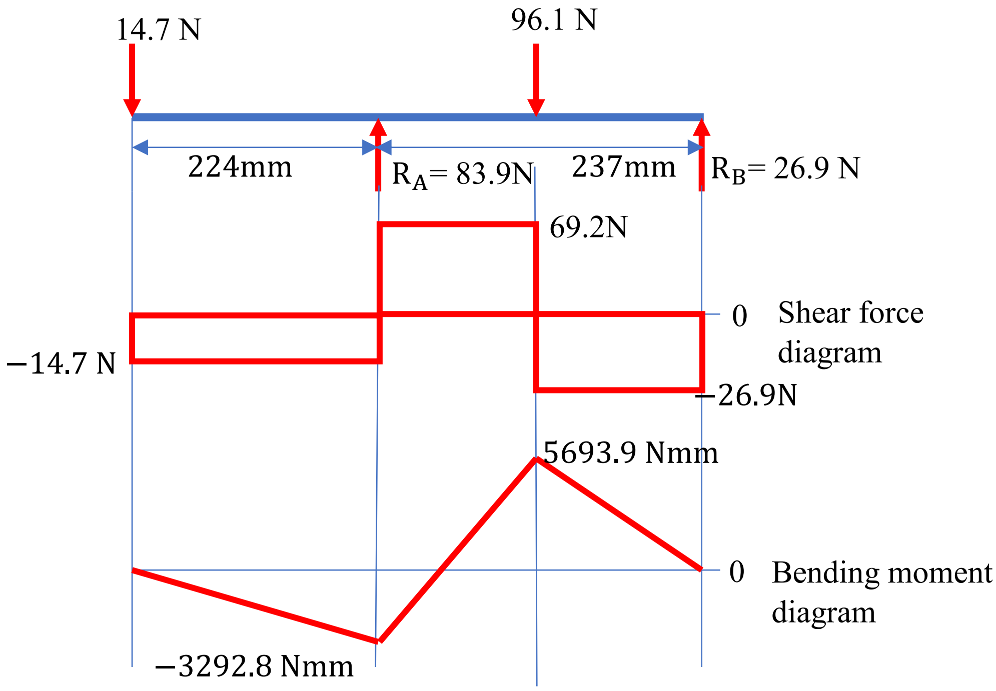


Figure 4.6. Shear force and bending moment diagram

Determining the shaft diameter:

1. Maximum shear stress theory/Guest's theory: it is used for ductile materials.

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \times d^3 \qquad \text{... Eqn(3.28)}$$

Shear stress induced due to twisting moment (τ) ($\tau_{ult} = 207$ MPa from Table: 3.5).

$$\tau = \frac{\tau_{ult}}{F.S} \qquad \text{... Eqn(3.29)}$$

$$\tau = \frac{207}{4} = 51.75 \text{ MPa} = 51.75 \text{ Nmm}$$

Then, using the above equation 3.28 the value of diameter of the shaft becomes:

$$d^3 = \frac{16 \times \sqrt{M^2 + T^2}}{\pi \times \tau} \qquad \text{... Eqn(3.30)}$$

$$d^3 = \frac{16 \times \sqrt{(5693.9)^2 + (253 \times 10^3)^2}}{\pi \times 51.75}$$

$$d = 29.74 \text{ mm}$$

$$d_{shaft} = 29.7 \approx 30 \text{ mm}$$

Spline Design (straight sided spline):

They are axial keys machined on a shaft and inside the hub which are used to transmit torque from shaft to another component (Robert L. Mott et al., 2018**)**. The spline dimensions on the shaft are determined for efficient and reliable power transmission by the shaft in the RBS is shown in Figure 4.7.

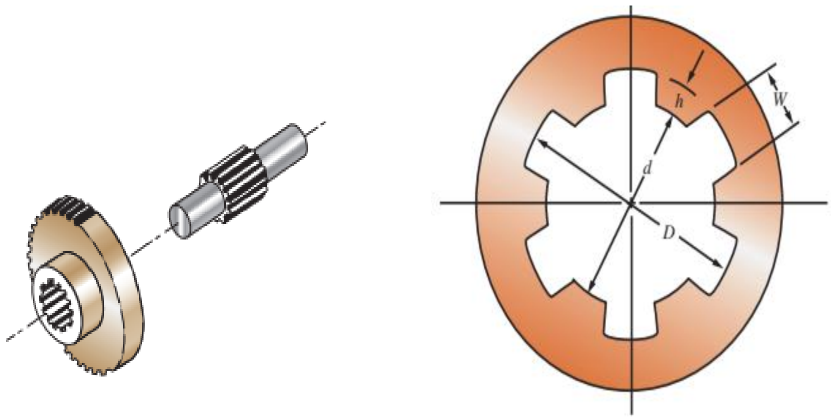


Figure 4.7. Spline (Robert L. Mott, 2018)

The torque transmitted to the spline shaft (T) is determined from the engine power output. Which is $253 \text{ Nm} = 253 \times 10^3 \text{ Nmm}$.

Selecting a permanent fit for the spline engagement having six splines from Table A–9 for firm and efficient power transmission. Then,

$$d = 0.900 \times D, \qquad \text{... Eqn(3.33)}$$

where: shaft diameter (d) is 30 mm

$$D = \frac{d}{0.900} \qquad \text{... Eqn(3.34)}$$

$$D = \frac{d}{0.900} = \frac{30\ \text{mm}}{0.900}$$

$$D = 33.33\ \text{mm} = 1.3\ \text{in}$$

Since the shaft diameter is calculated the value of inner diameter of the spline d is assumed to be the same.

Determining the width of the spline (w): the dimension can be calculated using the relation from Table A–8 using the spline numbers.

$$w = 0.25 \times D \qquad \text{... Eqn(3.35)}$$

$$w = 0.25 \times 33\ \text{mm}$$

$$w = 8.25\ \text{mm}$$

Determining the depth of the spline (h): the dimension can be calculated using the relation from Table A–8 using the spline numbers.

$$h = 0.059 \times D \qquad \text{... Eqn(3.36)}$$

$$h = 0.050 \times 33\ \text{mm}$$

$$h = 1.96\ \text{mm} \approx 2\ \text{mm to } 3\ \text{mm}$$

The value of the spline depth is approximated for machining convenience.

The assembly of spline shaft with flywheel is shown below Figure 4.8.

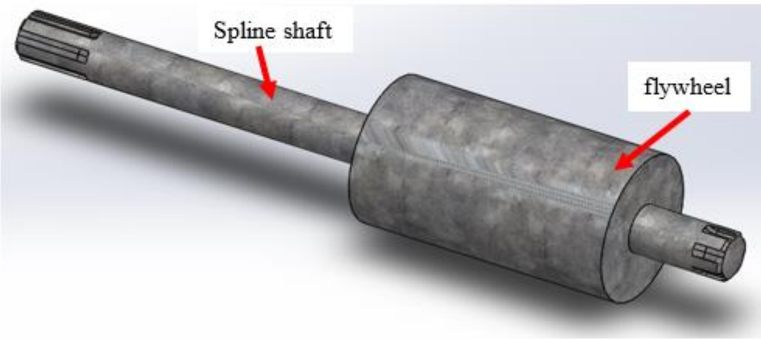


Figure 4.8. Spline shaft with flywheel

4.3.1.4 Clutch Design

A clutch is a machine member used to connect a driving shaft to a driven shaft so that the driven shaft may be started or stopped at will, without stopping the driving shaft (R.S. Khurmi and J.K. Gupta, 2005).

The system uses a single plate clutch with two effective contact faces for engagement and disengagement of the drive for the RBS is shown in Figure 4.9.

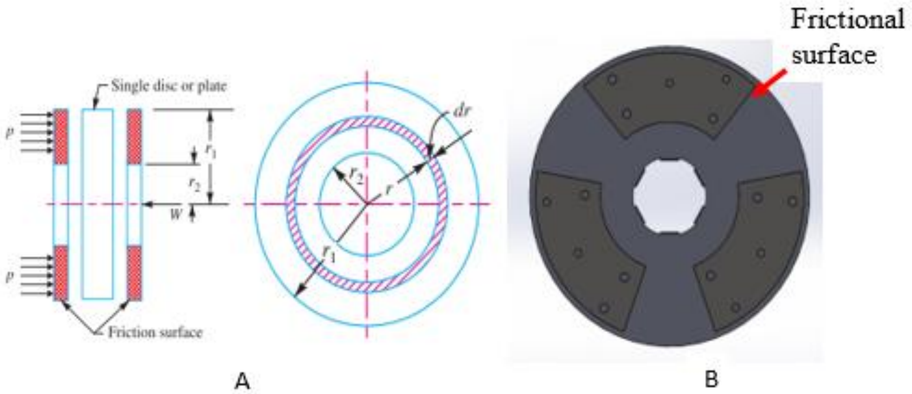


Figure 4.9.Forces on a disc clutch (A) and clutch plate modeling (B)

The RBS uses a pneumatically actuated clutch assembly. Power from air braking system is used to actuate resulting in the engagement of the clutch. This implies that the pressure of the air braking system will be acting over the clutch plate for engagement.

Design considerations

- Single driving plate with contact faces on each side
- In order to achieve compact design, the dimension upper limit of the clutch disc (r_1) is set to be 60 mm ($r_1 = 6$ cm).
- The clutch disc is assumed to be under uniform pressure
- Friction material used is rigid molded asbestos pad ($\mu = 0.45$) (Table A- 8)
- Pressure on the clutch disc (p) is assumed to be the air brake system pressure. Normal air brake system pressure range is around 690 to 828 kPa (official air brake handbook). For the analysis use average pressure 759 kPa.
- In order to mesh the system correctly Spline dimension of clutch plate is similar to the shaft spline. The only difference is on the shaft the spline is external but, in the clutch, plate uses internal spline.

Determining the inner radius of the friction surface:

$$T = \frac{P \times 60}{2 \times \pi \times N} = n\mu W \times R \qquad \text{... Eqn(3.37)}$$

Where: n −Number of contact surfaces which is 2.

μ −Friction coefficient which is 0.45.

P – Power transmitted (213.295 K_W @8050 rpm) for the pinion.

W −Axial thrust holding the friction face in place.

R − Mean radius of the friction surface

Then substituting the values of W&R in the torque equation and after some mathematical manipulation the inner radius can be calculated.

Then,

$$T = \frac{P \times 60}{2 \times \pi \times N} = n\mu W \times R$$

$$T = \frac{213.295 \times 10^3 \times 60}{2 \times \pi \times 8050}$$

$$T = 2 \times 0.45 \times 0.759 \times \pi \times (60^2 - {r_2}^2) \times \frac{2}{3}\left[\frac{60^3 - r_2^3}{60^2 - r_2^2}\right]$$

$$253 \times 10^3 = 1.43 \times (60^2 - {r_2}^2) \times \frac{2}{3}\left[\frac{60^3 - r_2^3}{60^2 - r_2^2}\right]$$

$$176.92 \times 10^3 = 60^3 - r_2^3$$

$$r_2 = 33.9 \text{ mm} \cong 34 \text{ mm}$$

Therefore, the lower limit radius (r_2) is 34 mm

$$W = \text{pressure} \times \text{area} = P_{ave} \times \pi({r_1}^2 - {r_2}^2) \qquad \text{... Eqn(3.37)}$$

$$W = 0.759 \times \pi((60)^2 - (34)^2)$$

$$W = 5827.6 \text{ N}$$

$$C = \frac{W}{2\pi(r_1 - r_2)} \qquad \text{... Eqn(3.38)}$$

$$C = \frac{5827.6}{2\pi(60 - 34)}$$

$$C = 35.7 \text{ N/mm}$$

$$P_{max} = {}^{C}\!/_{r_2} \qquad \text{... Eqn(3.39)}$$

$$P_{max} = \frac{35.7}{34} = 1.05 \approx 1 \ {}^{N}\!/_{mm^2}$$

$$P_{min} = {}^{C}\!/_{r_1} \qquad \text{... Eqn(3.40)}$$

$$P_{min} = \frac{35.7}{60} = 0.595 \ {}^{N}\!/_{mm^2}$$

Where: W= Total force acing on the friction surface

P_{max} = maximum pressure

P_{min} = minimum pressure

P_{ave} = average pressure

Mean radius of the friction surface:

$$R = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right] \qquad \text{... Eqn(3.41)}$$

$$R = \frac{2}{3}\left[\frac{60^3 - 34^3}{60^2 - 34^2}\right] = 48 \text{ mm}$$

Face width of the clutch plate is can be calculated as

$$b = \frac{R}{4} \qquad \text{... Eqn}(3.42)$$

Where: $R -$ Mean radius of the friction surface

$b -$ Face width of the clutch plate

$$b = \frac{48}{4} = 12 \text{ mm}$$

Area of friction face of clutch (A)

$$A = 2\pi Rb \qquad \text{... Eqn}(3.43)$$

$$A = 2 \times \pi \times 48 \times 12 = 3617.3 \text{ mm}^2$$

Heat conductivity of clutch material

Thermal conductivity measures the ability of the material that transfer heat. If the thermal conductivity is low means low amount of heat energy is transferred through it. Heat energy is transferred at every second per unit area is called heat flux. According Fourier's law heat flux defined as:

$$q = \frac{-\lambda \times \Delta T}{\Delta x} \qquad \text{... Eqn}(3.44)$$

Where: $\lambda -$ is thermal conductivity of materials

$\Delta T -$ temperature difference across the object

$\Delta x -$ the distance of heat transfer (the thickness of the object)

$q -$ heat flux, measured in W/m^2

N.B. The negative sign describes the direction of heat transfer. As the heat always flows from a warm body to cold body, the direction of heat is always transfer opposite to the temperature gradient.

Thermal conductivity of asbestos pad is 0.539 W/m K (Babji et.al. 2018).

$$\Delta T = T_{max} - T_{min} \qquad \text{... Eqn.}(3.45)$$

Where: T_{max} −maximum operating temperature of the material from Table $A - 12$ the maximum operating materials of asbestos pad on cast iron on dray condition is from 150^oC to 250^oC therefore for the analysis choose the maximum operating temperature 250^oC and minimum temperature (ambient temperature) 25^oC.

$$\Delta T = 250^oC - 25^oC = 225^oC = 498 \text{ K}$$

$$q = \frac{-\lambda \times \Delta T}{\Delta x}$$

$$q = \frac{-0.539 \times 498}{0.012}$$

$$q = -22368\ W/m^2$$

Convective heat transfer coefficient (h_c):

$$h_c = \frac{q}{(\Delta T)} \qquad \dots Eqn(3.46)$$

Where: q = Heat flux, measured in W/m^2

ΔT = Temperature difference across the object in ℃

$$h_c = \frac{22368}{225} = 99.4\ W/m^2.℃$$

4.3.2 Pneumatic System Design

4.3.2.1 Pneumatic Actuators

Pneumatic actuators are the devices used for converting pressure energy of compressed gas into the mechanical energy to perform useful work. In other words, Actuators are wont to perform the task of exerting the desired force at the tip of the stroke or used to create displacement by the movement of the piston (Mr. Sandeep Shrikant Karekar, 2017).

Based on application for which air cylinders are used

i. Light duty air cylinders
ii. Medium duty air cylinders
iii. Heavy duty air cylinders

Table 4.1. Material of Construction for Light, Medium and Heavy-Duty Cylinders (James A.Sullivan 2002).

Components	Types of cylinders		
	Light duty	Medium duty	Heavy duty
Cylinder tube	Hard drawn seamless aluminum or brass	Hard drawn seamless brass tubes, Aluminum	Hard drawn seamless brass tubing, brass, bronze,
End covers	Aluminum alloy casting fabricated aluminum, brass, bronze	Aluminum, Brass, bronze, fabricated Brass, Bronze	High tensile castings
Pistons	Aluminum alloy casting	Aluminum alloy castings,	Aluminum alloy castings,
Piston rods	EN 8 or smaller steel ground and polished or chrome plated	EN 8 or smaller steel, ground and polished or chrome plated.	Ground and polished stainless steel

Pneumatic system design involves:

- Determining the working pressure (operating pressure) of the system
- Determining the working condition of the components.
- Selection the appropriate components to fit the operating conditions.
- Check the stress on the cylinder and for the design factor of safety is 2

a) Determining the working pressure of the system

The working pressure of the system is the same as the maximum pressure that apply on the clutch disc or plate. Therefore, maximum pressure apply on the clutch plate is 1 $\mathrm{N/mm^2}$.

b) Determining the working condition of the components:

The pneumatic system must be selected taking the worst condition that it will operate on to have durability and long-lasting life time.

As the system is mounted on the under carriage of the bus it is exposed to:

- Environmental factors that kinder efficient operation (i.e., corrosion, mud, temperature variation, … etc.)
- Vibration and Impact from the road

Thus, due to these reasons and other factors not taken into consideration the construction of pneumatic system should be for heavy duty operation.

c) Selection of appropriate components to fit the operating conditions:

Now, for heavy duty operating condition the construction should also be heavy duty. So, from Table 4.1. being taken as a reference in selectin the actuator to be used in the system. The pneumatic actuator considered for the design is the CG1 series in the SMC pneumatics catalogue and the 3D model is shown in Figure 4.10.

General characteristics of this actuators include:

- single acting, spring Return
- Compact in construction and light weight
- High velocity

Table 4.2. Pneumatic Actuator Technical Specification (SMC, pneumatic catalog, 2015).

Fluid	Air
Proof pressure	0.15 MPa
Maximum operating pressure	0.99MPa ≈ 1MPa
Minimum operating pressure	0.05MPa
Ambient and fluid temperature	-10 to $60^{\circ}\mathrm{C}$
Piston velocity and lubrication	50 to 700 mm/sec, no required

Table 4.3. Pneumatic Actuator Construction Specification (SMC, pneumatic catalog).

Rod cover	Aluminum alloy (hard black alumite)
Body	Aluminum alloy (hard alumite)
Piston rod	Carbon steel (hard chrome plated

The actuator dimensions then can be determined from Table A–1. Therefore, for compact design and maintenance case a cylinder with bore (∅) =20 mm is selected, Stroke length up to 300 mm.

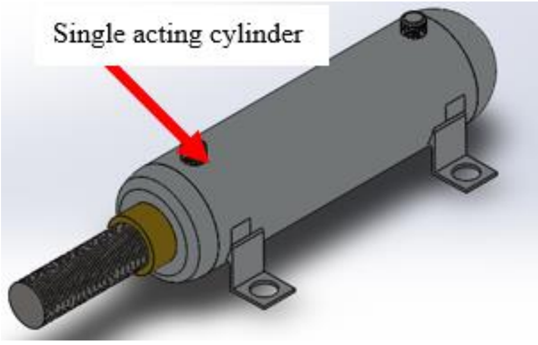


Figure 4.10. 3D model of single acting pneumatic cylinder

The pneumatic actuator force becomes

$$F = P_d \times A_P \qquad \text{... Eqn. (3.47)}$$

$$A_P = \frac{\pi {D_i}^2}{4} = \frac{\pi \times 20^2}{4} = 314.2\ \text{mm}^2$$

Maximum force applied by the pneumatic cylinder at maximum operate pressure is:

$F = 1\ \text{Mpa} \times 314.2\ \text{mm}^2 = 314.2\ \text{N} = 0.3142\ \text{kN}$ is the force that apply on the clutch by the pneumatic cylinder. Therefore, the cylinder is able to push or lift up to 32 K_g.

Minimum force applied by the pneumatic cylinder at operate pressure is:

$F = 0.05\ \text{MPa} \times 314.2\ \text{mm}^2 = 15.71\ \text{N}$ is the force that apply on the clutch by the pneumatic cylinder. The cylinder is able to push or lift up to 1.6 K_g.

Therefore, the pneumatic system is able to engage the clutch with in the given pressure range. The pneumatic cylinder is made from aluminum alloy grade 3003.

From table A- 10 ultimate stress(σ_{ult}) is 131 MPa and yield stress(σ_{yield}) is 117 MPa

Allowable stress on the cylinder (σ_{all}):

$$\sigma_{all} = \frac{\sigma_{ult}}{FS} \qquad \text{... Eqn(3.48)}$$

$$\sigma_{all} = \frac{131}{2} = 65.5\ \text{MPa}$$

$$\tau_{yield} = \frac{\sigma_{yield}}{FS} \qquad \text{... Eqn(3.49)}$$

$$\tau_{yield} = \frac{117}{2} = 58.5\ \text{MPa}$$

Checking

Hoop stress or circumferential stress (σ_h):

$$\sigma_h = \frac{P_d + R_i}{t} \qquad \text{... Eqn(3.50)}$$

$$\sigma_h = \frac{1 \times 10}{6} = 2.66 \text{ MPa}$$

Hoop stress $\sigma_h = 1.66$ MPa $\ll$ $\sigma_{all} = 65.5$ MPa Therefore, the design is safe

longitudinal stress (σ_l) is half of the hoop stress (σ_h)

$$\sigma_l = \frac{\sigma_h}{2} \qquad \text{... Eqn(3.51)}$$

$$\sigma_l = \frac{1.66}{2} = 0.83 \text{ MPa}$$

Checking, using theories of failure

A. Maximum principal stress theory

$$\sigma_1, \sigma_2 = \frac{\sigma_h + \sigma_l}{2} \pm 0.5\sqrt{(\sigma_h - \sigma_l)^2 + 4\tau^2} \qquad \text{... Eqn(3.52)}$$

$$\sigma_1, \sigma_2 = \frac{1.66 + 0.83}{2} \pm 0.5\sqrt{(1.66 - 0.83)^2 + 4 \times 0^2}$$

Where shear stress (τ) $= 0$

$$+,\ \sigma_1 = 1.66 \text{ MPa},\ -,\ \sigma_2 = 0.83 \text{ MPa}$$

According to the maximum principal stress theory $\sigma_1 < \sigma_{yield}$. i.e., σ_1= 1.66 MPa $<$ $\sigma_{yield} = 117$ MPa). Therefore, the design is safe

B. Maximum shear stress theory

$$\tau_{max} = \frac{\sigma_h - \sigma_l}{2} \qquad \text{... Eqn(3.53)}$$

$$\tau_{max} = \frac{1.66 - 0.83}{2} = 0.415 \text{ MPa}$$

$$\tau_{max} < \tau_{yield} == 0.415 \text{ Mpa} < 58.5 \text{ MPa}$$

Therefore, the design is safe

C. The maximum distortion energy theory

$$\sigma_{all}{}^2 = \sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\sigma_2 \qquad \text{... Eqn(3.54)}$$

$$\sigma_{all} = \sqrt{\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\sigma_2} \qquad \text{... Eqn(3.55)}$$

$$\sigma_{all} = \sqrt{1.66^2 + 0.83^2 - 1.66 \times 0.83}$$

$$\sigma_{all} = 1.4 \text{ MPa}$$

$$\sigma_{all(material)} = \frac{\sigma_{ult}}{FS} \qquad \text{... Eqn(3.56)}$$

$$\sigma_{\text{all(material)}} = \frac{131}{2} = 65.5 \text{ MPa}$$

$$\sigma_{\text{all}} < \sigma_{\text{all(material)}} = 1.4 \text{ MPa} < 65.6 \text{ MPa}$$

The design allowable stress is less than the material allowable stress. Therefore, the design is safe. σ_{all} obtained from the material ultimate stress is above the calculated value from the maximum distortion energy theory.

4.3.3 Electrical System Design

The electrical system design involves the determination of total loads on the system and calculating the appropriate battery size to run the loads for the specified on-hour of the loads. The RBS electrical system is integrated with the already existing vehicle electrical system and is switched on by using a toggle switch. The toggle switch is used to alternate between the stock vehicle electric power source and the RBS electric power supply.

4.3.3.1 Loading data

Determine loads per day: proper determination of the loads to be powered by the RBS is the most important part in the electrical system design. All the electrical loads of the RBS are DC loads and some power on vehicle electrical components shown in Table A-18. The average consumption of the intermittent loads is estimated using a factor of 0.1. Total wattage on some of the intermittent load that selected for the design is $510 \times 0.1 = 51$. The total wattage is 296 W is about 48% of the total wattage of the vehicle.

Commonly used correction factors to the loads due to losses:

Correction of loads: it is assumed that all equipment doesn't operate to their pick all time. Thus, it is logical to take 96% of the possible max load of the system (daily/ Wh) for design purpose shown in Table 4.5.

Table 4.4. Correction factors for the electrical systems (M. Megra, 2018).

	Loss	Efficiency
Charger controller	0.05	0.95
Mismatch	0.01	0.99
Diodes and Connection	0.005	0.995
Wiring	0.02	0.98
Soiling	0.04	0.96
System Availability	0.01	0.99
Temperature increase	0.05	0.95
Total		0.96

Decision on the voltage value (Vs) for power transfer system (i.e., 12,24,48,60,96,129 Volts)**:** this decision is made taking the normal operation source for the components of the bus. Hence, for the current system it is crucial to use a voltage source (Vs) of 24 V since the components on the bus are designed for it (DGS-Berlin, 2014).

4.3.3.2 Battery bank size

Determine days of autonomy (DOA): that is days of storage desired, DOA = 9 hrs for current design, meaning that it can carry the whole load for 9 consecutive hours without being charged or on a single charge.

DOA load storage: using the watt hour method (Kebede, M. H., 2018).

$$\mathrm{Ah} = \frac{\mathrm{load}\left(\frac{\mathrm{Wh}}{\mathrm{day}}\right) \times \mathrm{DOA}}{\mathrm{system\ voltage}}, \qquad \text{... Eqn. (3.57)}$$

- Required battery capacity: The required battery capacity is determined by dividing battery storage during the days of autonomy with depth of discharge.
- Number of batteries in series: The number of batteries needed to provide the necessary DC system voltage is determined by dividing the battery bus voltage i.e.,24 V selected battery voltage.
- Number of batteries in parallel: It is determined by dividing the required battery capacity by the amp-hour capacity of the selected battery.
- Total battery Ah capacity: The total rated capacity of selected batteries is determined by multiplying the number of batteries in parallel by the Ah capacity of the selected battery (Kebede, M. H., 2018).
- Allowable DOD limit: It is recommended that the battery should have greater depth of discharge. For this design we chose DOD of 0.80.
- Selecting Ah capacity of the battery: it is done from the manufacturer specification Sheet.

Current load design:

Step 1: Determine the total load on the system (from Table A-18).

Total DC load on the RBS: 2664 Wh/day

Step 2: Calculating the total Wh/day required from the RBS.

$$\mathrm{Required}\left(\frac{\mathrm{Wh}}{\mathrm{day}}\right) = \frac{\mathrm{Total\ Load}}{\mathrm{CF}}, \qquad \text{... Eqn. (3.58)}$$

$$\mathrm{Required}\left(\frac{\mathrm{Wh}}{\mathrm{day}}\right) = \frac{2664}{0.96} = 2775\ \frac{\mathrm{Wh}}{\mathrm{day}}$$

Where: CF = 0.96 – Correction Factor (from Table 4.5)

Step 3: Calculating the battery size

$$\text{Battery Capacity} = \frac{\text{Total}\left(\frac{\text{Wh}}{\text{day}}\right) \times \text{DOA}}{\eta_{\text{Batt}} \times \text{DOD} \times \text{V}_{\text{Sys}}}, \qquad \text{... Eqn. (3.59)}$$

Assumption:

Where: Days of Autonomy (DOA) = 9 hrs. = 0.375 days

Depth of Discharge (DOD) = 80%

Source voltage (V_s)= 24V

Lithium-ion Battery Efficiency $(\eta_{\text{Batt}}) = 97\%$

$$\text{Battery Capacity} = \frac{2775 \times 0.375}{0.97 \times 0.8 \times 24}$$

$$\text{Battery Capacity} = 55.8 \text{ Ah}$$

Thus, from different batteries available in the market 24 V & 65 Ah rating SHUNBIN lithium battery is selected (amazon). with the following specifications:

- Brand: SHUNBIN
- Nominal voltage – 24 V and Weight – 3 kg
- Dimensions (L × W × H) – 238 mm × 270 mm × 68 mm

Therefore, in order to obtain the required system voltage, the batteries connected in series are determined the by (Kebede, M. H. 2018).

$$\text{Number of batteries in series} = \frac{\text{system voltage}}{\text{Battery voltage}}$$

$$\text{Number of batteries in series} = \frac{24 \text{ V}}{12 \text{ V}}$$

$$\text{Number of batteries in series} = 2 \text{ batteries}$$

Determination of charging time and braking time of the bus

The following assumption were taken for aid in the analysis:

- The bus runs up to 10 hr. a day.
- The bus is assumed to be travelling at an average speed of 50 km/hr.
- The brakes were engaged 4 – 6% of the daily traveled distance by the bus due to traffic jam, speed brakers and loading unloading condition.
- The brake average speed of the bus was considered to be 30 km/hr.

From the above assumptions if the bus was to travel at 50 km/hr for a day's worth of work the distance covered will be:

$$\text{Travelled distance} = \text{travel speed} * \text{time taken}$$

$$\text{Travelled distance} = 50\ \frac{\text{km}}{\text{hr}} \times 10\ \text{hr}$$

$$\text{Travelled distance} = 500\text{km}$$

Now considering the brakes are engaged 4 – 6% of the distance travelled.

$$\text{Braking distance} = 0.06 \times 500\ \text{km}$$

$$\text{Braking distance} = 25\ \text{km}$$

Now taking the braking average speed of the bus to be 25 km. Then, the time taken by the total brake action will be:

$$\text{Time taken by the braking action} = \frac{\text{braking distance}}{\text{average braking speed}}$$

$$\text{Time taken by the braking action} = \frac{25\ \ \text{km}}{30\ \frac{\text{km}}{\text{hr}}} = 0.83\ \text{hr} = 50\ \text{min}$$

Determining the time taken to fully charge the batteries (t_{charge}): this value is determined form the total battery output (in Wh) and the total alternator power output (in watts).

$$\text{Battery Output} = 24\ \text{V} \times 65\ \text{Ah} = 1560\ \text{Wh}$$

$$\text{Alternator Output} = 24\ \text{V} \times 100\ \text{A} = 2400\ \text{W}$$

So, from the above calculated values the aim is to determine how much time it takes to fully charge 1560 Wh battery using a 2400 W alternator. Thus,

$$t_{chrg} = \frac{\text{Battery Output (Wh)}}{\text{Alternator (Watts)}}$$

$$t_{chrg} = \frac{1560\ \text{Wh}}{2400\ \text{W}}$$

$$t_{chrg} = 0.65\ \text{hr} = 39\ \text{min} = 2340\ \text{sec}$$

Previously stated on the battery selection charging time of lithium-ion batteries on Table 3.9 is 1 hour or less. So, to charge the batteries fully time needed for the alternator to run is 39 minutes. therefore, the selected battery is compatible for this system. Hence, the time taken by the braking action is calculated to be 50 minutes. The calculated valued (i.e., 50 minutes) is also the duration of alternator running time per day. So, to charge the batteries fully time needed for the alternator to run is 39 minutes. But for a single day the bus is operated the alternator could run for up to 50 minutes which is plenty enough to fully charge the batteries and has operating allowance for mishaps on the road.

4.4 System installation

4.4.1 Mechanical system installation

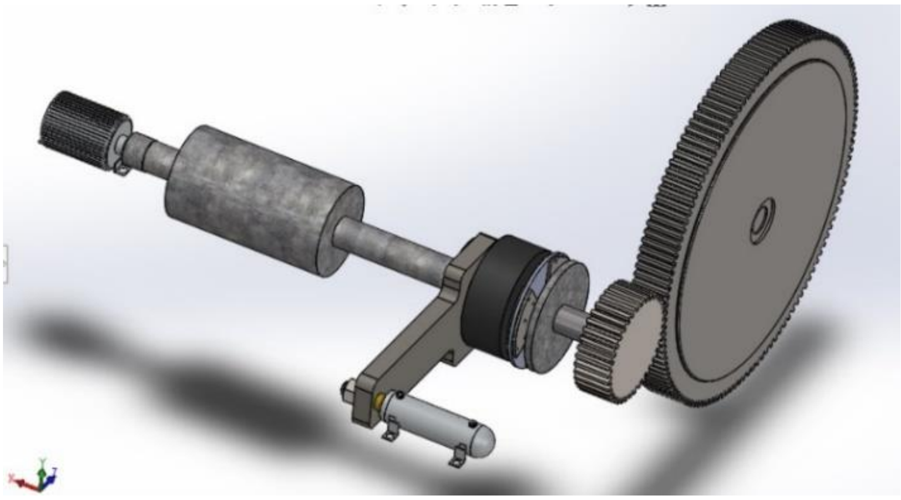


Figure 4.11. Mechanical system installation of RBS

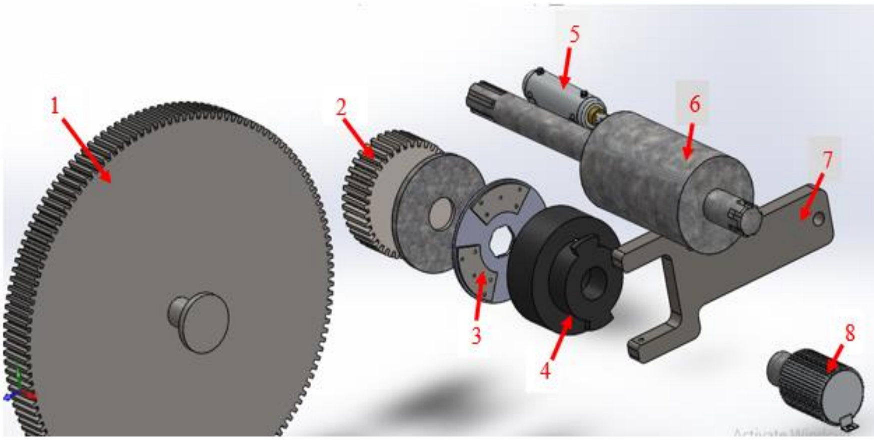


Figure 4.12. Explode view of mechanical system of RBS

Mechanical component's name:

1. Engine flywheel
2. Pinion gear with frictional plat
3. Single plate clutch disc
4. Clutch hub
5. Single acting pneumatic cylinder
6. Splined shaft with solid disc flywheel
7. Fork lever
8. Alternator

4.4.2 Pneumatic system installation

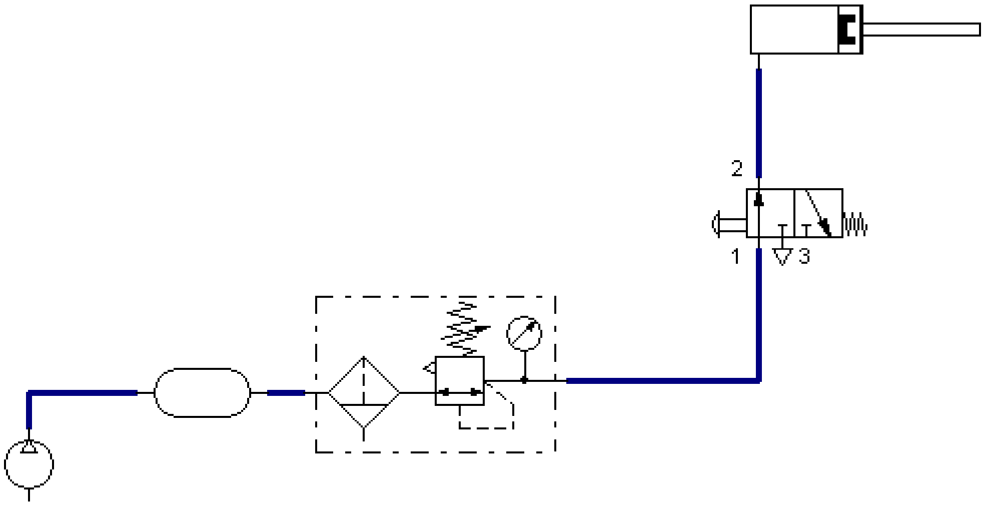


Figure 4.13. Pneumatic system installation during brake applied

The above Figure 4.13. shows the brake pedal depressed by the driver during braking. During this time the air flows from the reservoir through the spring return 3/2-way directional control valve (through port 1-2). It causes to actuate the pneumatic actuator. The linear movement helps to engage the RBS single palate clutch to the frictional surface of the pinion gear to transfer rotational motion to the rest of the RBS to develop electrical power.

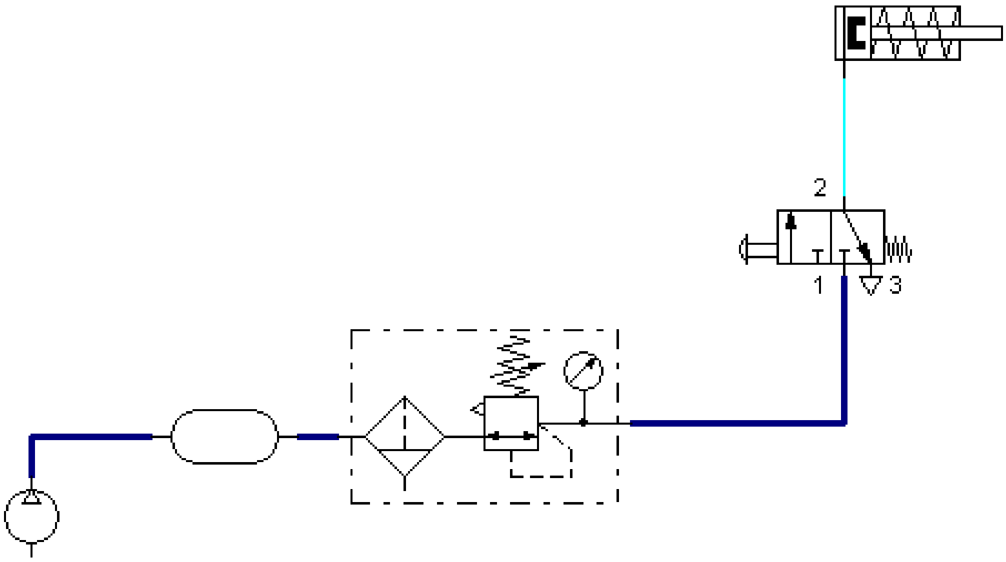


Figure 4.14. Pneumatic system installation after the brake pedal releasing

The above Figure 4.14. shows after releasing the brake pedal (after releasing brake pedal by the driver). During this time the air is flow to the atmosphere through port 3, the spring causes the pneumatic actuator retract to the original position. then RBS clutch is disengaged (the rotational motion from the pinion gear is not transferred to the rest of RBS).

4.4.3 Electrical component installation

The electrical system components of the bus to be run by the RBS will have two separate power sources. These are:

- Power from the bus batteries (24 V) and
- Power from the Li-ion battery of the RBS (24 V)

So, for the utilization of these power sources efficiently the driver/operator is provided with and electric toggle switch to change between the two power sources. As shown in Figures 4.15, 4.16 and 4.17. the electrical components can utilize power from both sources with just a flip of a switch.

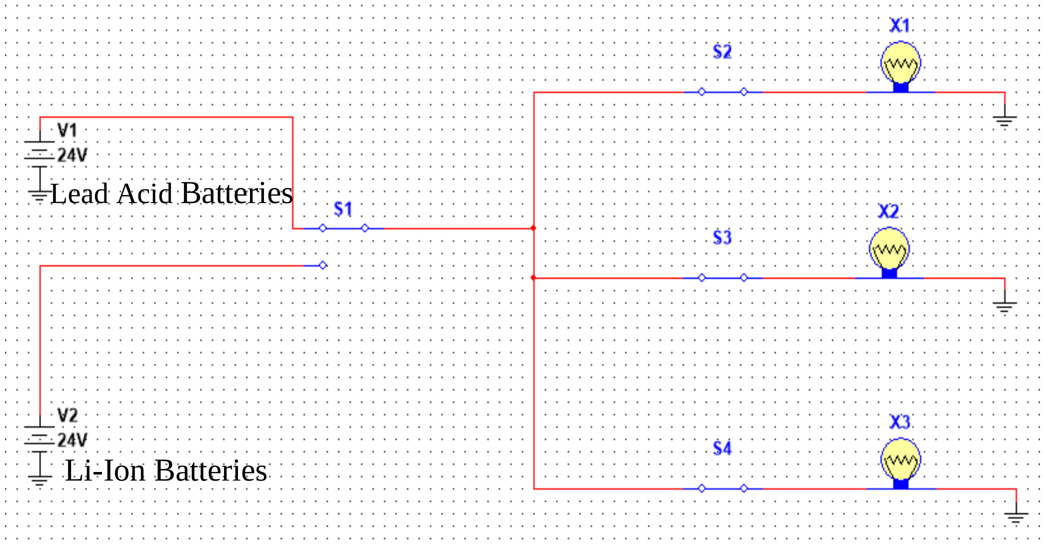


Figure 4.15. Power from The Truck Lead Acid Batteries

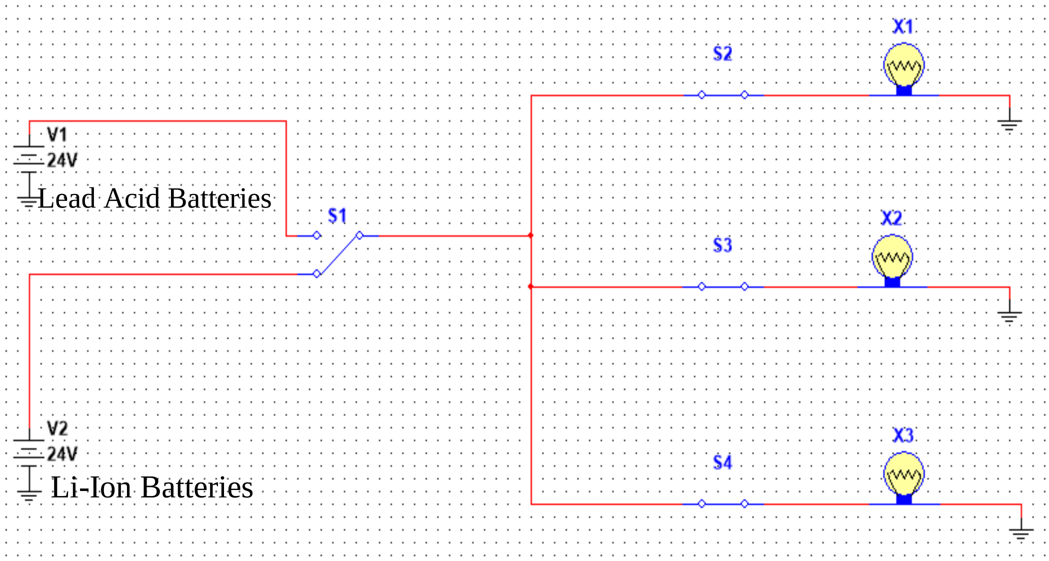


Figure 4.16. Power Source from The RBS Li-Ion Batteries

After receiving the power from either of the sources the branch circuits of the system are controlled by separate switches to turn them on and off.

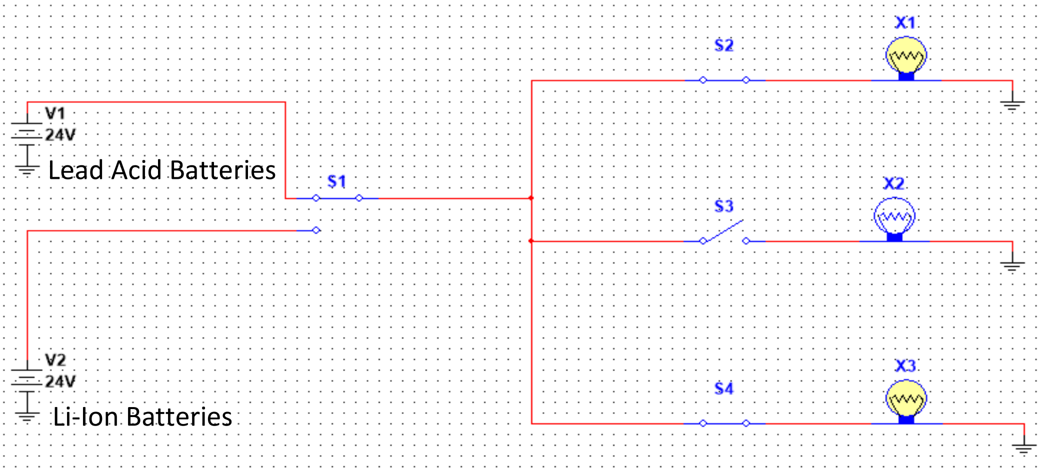


Figure 4.17. Branch Circuit Control

Where: s_n – switch

X_n – vehicle electrical components

4.5 Chapter summery

Throughout this chapter design of regenerative braking system is discussed. According to the design steps the three-sub system of regenerative braking system components are clearly designed. The size, force, stress analysis etc. are determined, the failures of the component is checked to enhance the efficiency, life of the system. Mechanical system, pneumatic system and electrical system installation is done.

CHAPTER FOUR

5 RESULT AND DISCUSSION

5.1 Introduction

Energy is neither created nor destroyed but it can transform from one form to another form. During the time of braking energy is waste as a heat, in order to recover this waste energy regenerative braking system is play important role. Therefore, this chapter covers the results and discussion of design analysis, recovered kinetic energy, recovered power, efficiency and the FEM analysis of direct gear engagement regenerative braking system to the engine flywheel in the form of Tables, Graphs and Figures.

The analysis done for the system component and sub-system are summarized as follows:

5.2 Mechanical systems

5.2.1 Pinon Gear

Design considerations:

- Module (m = 3 mm): taken from physical measurement of the flywheel.
- Tooth face width (b = 4 cm): taken from physical measurement of the flywheel.
- The construction material for the pinion is grey cast iron which is the same as that of the flywheel. This is to avoid failure during operation and reduce galvanic corrosion from having different material in contact with each other.

Using the above stated design considerations, the obtained results shown in Table 5.1.

Table 5.1. Pinion gear dimensions and loads

Teeth No	Diametral pitch (mm)	Outside diameter (mm)	Root diameter (mm)	Addendum and Dedendum (mm)	Thickness of teeth (mm)	Clearance (mm)
38	114	120	106.5	3.75& 3	4.7	0.75

Material	Tangential tooth load (kN)	Dynamic tooth load (kN)	Static tooth load (kN)	Wear teeth load (kN)
Grey cast iron	12.4	24	27.8	26.99

From the rotation obtained from the flywheel and the rotation needed to drive the DC generator the number of teeth of the pinion was determined. Then, the tooth static, tooth wear and dynamic loads were calculated for safe design of gear tooth.

Thus, for the gear tooth to be safe from breakage the static tooth load and teeth wear load must be greater than the dynamic tooth load. Gray cast iron is selected to be the pinion gear construction material and checked for tooth breakage safety then safe from breakage a material with large Brinell Hardness Number (BHN) which satisfied the require safety parameter since W_S and W_w> W_D.

5.2.2 Flywheel

Design consideration

- Type of flywheel: Solid disc Flywheel and its diameter does not exceed 100 mm to have compact design.

Table 5.2. Flywheel design

Material	Mass (kg)	Width (mm)	Diameter (mm)
High strength low-alloy steel	9.8	163	100

The flywheel material was selected by computing energy storage capacity, cost and operating speed at which the flywheel will be made to run. Then from the analysis the specification (i.e., Flywheel mass and Flywheel width) were determined shown in Table 5.2.

5.2.3 Clutch Design

Design considerations

- Single driving plate with contact faces on each side (single plate). And under uniform pressure.
- In order to achieve compact design, the dimension upper limit of the clutch disc (r_1) is set to be 60 mm (r_1 = 6 cm).
- Pressure on the clutch disc (p) is assumed to be the air brake system pressure. From, vehicle manual the air brake operating pressure is 690 to 828 kPa;). For the analysis use average pressure 759 kPa. Friction material used is rigid molded asbestos pad ($\mu = 0.45$).

Table 5.3. Clutch design

Material	Inner radius (mm)	Outer radius (mm)	Thickness of the plate (mm)	Coefficient of friction (μ)
Rigid Molded Asbestos Pad	34	60	12	0.45

After determining the working pressure, friction material friction coefficient and limiting the friction surface upper radius. The dimension of the friction surface needed for efficient torque transmission is determined shown in Table 5.3. And also due to high friction of the clutch plate heat is generated. This energy is transferred at every second per unit area is called heat flux. Therefore, heat flux of the clutch is $22368\ \mathrm{W/m^2}$.

5.2.4 Shaft and Spline

Design considerations

- Flywheel mass (m_{FW} = 9.8 kg)
- Mass of clutch assembly is assumed to be (m_c = 1.5 kg)
- The shaft is assumed to be subjected to only vertical loading
- Material considered for design is aluminum alloy for light weight construction.
- The factor of safety for the shaft design is assumed to be F.S = 4 to have a reliable and durable shaft in operation.

Table 5.4. Shaft and Spline design

Material	Shaft diameter (mm)	Maximum bending moment (N-mm)	Twisting Moment (N-mm)
Aluminum alloy 6351 – T6	30	5693.9	253×10^3
Spline			
Material	No of splines	Depth of spline (mm)	Width of spline (mm)
Aluminum alloy 6351 – T6	6	2	8.33

Shaft:

The light weight requirement for construction will require for light weight material for which aluminum alloy is selected for shaft construction. Since the shaft is subjected to both twisting and bending moment it is checked for both loads to determine the required shaft material to handle these loads with factor of safety of 4.

Spline:

Since the spline is part of the shaft it is constructed from the same material as the shaft. Then using the torque to be transmitted the spline dimensions were determined shown in Table 5.4.

5.3 Pneumatic System

Table 5.5. Pneumatic system design

Working pressure (MPa)	Operating condition	Stroke length (mm)	Cylinder Bore (mm)
0.15 – 1	Heavy duty	300	Ø20

The pneumatic system will be operating on the air brake pressure under heavy duty conditions is shown in the above Table 5.5. Then, from the SMC pneumatics catalog a CG1 series actuator is selected to be the actuating component for the clutch engagement. Maximum force applied by the pneumatic cylinder at maximum operate pressure $F = 314.2\ N = 0.3142\ kN$ is the force that apply on the clutch by the pneumatic cylinder. The cylinder is able to push or lift up to 32 kg. and with minimum operating pressure it can lift up to 1.6 kg. Therefore, the cylinder is safe, it can engage 1.5 kg clutch plate.

Table 5.6. Pneumatic cylinder stresses

Pneumatic single acting cylinder						
Material	Hoop stress σ_h (MPa)	Principal stress σ_1 (MPa)	Shear stress τ_{max} (MPa)	Yielding stress σ_{yield} (MPa)	Allowable stress σ_{all} (MPa)	Yielding shear stress (MPa)
Aluminum alloy (grade 3003)	2.66	2.66	0.665	117	65.5	58.5

For design purposes, the thickness of the pneumatic cylinder is read from Table A-1. Based on the operating pressure 1 MPa different stresses are checked. Then the cylinder is checked for failure using the theories of failure; thus, the cylinder principal pressure and the maximum shear stresses are checked with their respective yielding stress of the material. The yielding stress and allowable stress must always be greater.

Therefore, ($\sigma 1 = 2.66$ MPa $< \sigma yield = 117$ MPa and $\tau\, max = 0.665$ MPa $<$ $\tau yield = 58.5$ MPa, $\sigma_h = 2.66 < \sigma_{all} = 65.5$**)** is shown in Table 5.6.

therefore, the design is safe.

5.4 Electrical System

Design considerations

- Days of Autonomy (DOA) = 9 Hrs = 0.375days
- Depth of Discharge (DOD) = 80%
- Source voltage (V_s)= 12V
- Battery Efficiency (η_{Batt}) = 97%

Table 5.7. Electrical System Design Results

Battery type	Battery capacity (Ah)	Voltage (V)	Number of batteries
Li-ion battery	65	12	2 Batterie in series

For the required battery depth of discharge (DOD = 80%) a lithium-ion battery is the obvious efficient and low-cost choice. Then, from the analysis for the electrical system the battery size, Battery voltage and battery connection were determined shown in Table 5.7.

5.5 Mechanical System Energy Generation

The mechanical system generates energy from the rotation provided by the engine during the braking. The energy absorbed by the system is upon the type of motion of the moving body. The motion of a body may be either pure translation or pure rotation or a combination of both translation and rotation. The energy corresponding to these motions is kinetic energy. The motion of the body in direct gear engagement of RBS to the flywheel is pure rotational. Using direct gear engagement of RBS to the flywheel the bus battery is charging using two cases.

1. Direct rotational energy of the RBS using engine flywheel. This means the attached pinion gear rotates freely until the driver apply the brake, when the driver applies the brake due to road condition, traffic jam, and loading unloading condition the clutch becomes engaged and the other system start to rotate to charge the vehicle battery.
2. Stored energy RBS flywheel.: This system has a flywheel that is used to smooth the rotational motion and additionally it used to store the rotational kinetic energy. This energy is used to charge the battery after realizing the brake pedal.

Therefore, total recovered kinetic energy is the summation of absorbed rotational kinetic energy during the time of braking and the stored rotational kinetic energy on the RBS flywheel.

5.5.1 Kinetic energy recovered during braking

The RBS engagement is performed by using a pneumatic clutch to harvest drive from the pinion during the time of braking and it charges the vehicle battery until the brake pedal released. For pure rotational kinetic energy body of mass moment of inertia I(about a given axis) is rotating about that axis with an angular velocity ω rad / s. This recovered kinetic energy is directly converted to electrical power using alternator.

During breaking the recovered rotational kinetic energy is determined shown in Table 5.8.

$$K_E = \frac{1}{2} I\omega^2 \qquad (\text{ pure rotational motion})$$

Where: Rotational kinetic energy (K_E)

Flywheel moment of inertia (I) $= mr^2$

Angular velocity of the flywheel (ω) $= \frac{2\pi \times N}{60}$

Consideration

- Mass of engine flywheel 27 kg
- Radius of engine flywheel 0.911 m
- Moment of inertia (I) $= mr^2 = 27 \times 0.199^2 = 1.07\ kgm^2$
- ω_2 is final angular velocity of engine flywheel which is zero
- $\omega_1$is the initial angular velocity of the vehicle

  Therefore, absorbed rotational kinetic energy depending on the engine rotational speed and it can be determined.
- Correction factor: energy is lost due to clutch engagement and disengagement (friction) in the form of heat, gear mash, and other defects and it is assumed that all equipment doesn't operate to their pick all time. Thus, it is logical to take 70% of the possible (Fritz Faulhaber,2002 and David Pratte, 2020).

Table 5.8. kinetic energy loss and recovered on the engine flywheel during braking

Pinion gear or alternator rotational speed (rpm)	Engine rotational speed (rpm)	Angular velocity (rad/sec)	Waste kinetic energy KE_1 (K_J)	Recovered kinetic energy KE_R (K_J)
0	0	0	0	0
1700	485.7	50.8	1384	968.9
2000	571.4	59.8	1915.7	1341
2250	642.8	67.3	2424.6	1697.2

2500	714.3	74.7	2993.3	2095.3
3000	857.1	89.7	4310.4	3017.3
3500	1000	104.7	5866.9	4106.8
4000	1142.8	119.6	7662.9	5364
4500	1285.7	134.6	9698.4	6788.9
5000	1428.5	149.6	11973.3	8381.3
5500	1571.4	164.5	14487.7	10141.4
6000	1714.3	179.5	17241.6	12069.1
6500	1857.1	194.5	20234.9	14164.4
7000	2000	209.4	23467.7	16427.4
7500	2142.8	224.4	26939.9	18857.9
8050	2300	240.8	31036	21725.3

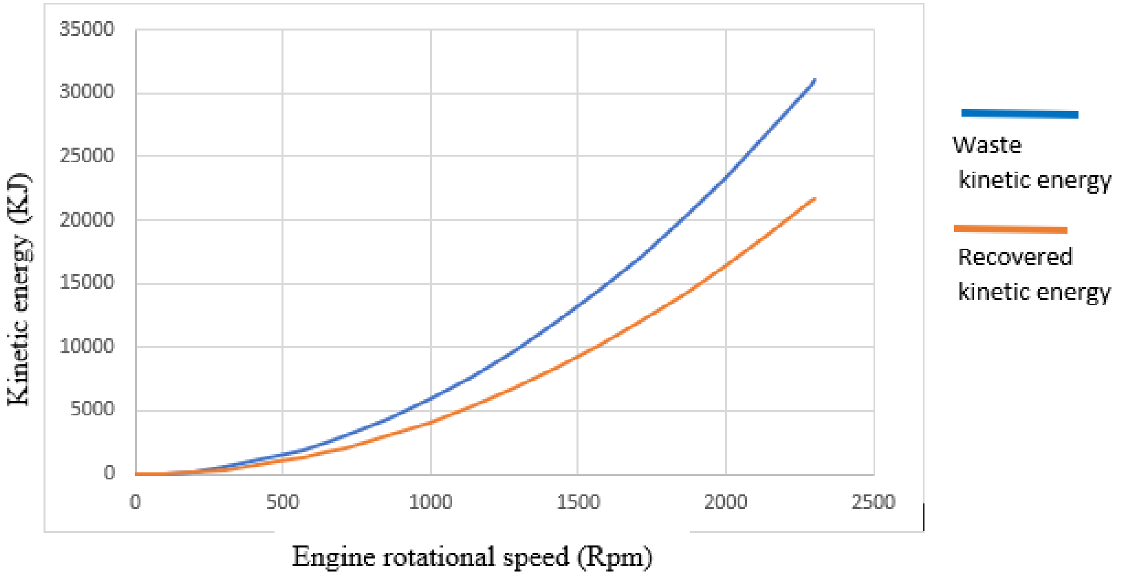


Figure 5.1. Kinetic energy loss end recovered Vs engine rotational speed

The above Table 5.8 and Figure 5.1 shows kinetic energy loss and recovered with engine rotational speed. Therefore, waste and recovered kinetic energy is directly proportional with engine rotational speed.

5.5.2 Kinetic energy stored on RBS flywheel

A flywheel is a heavy rotating mass which stores rotational kinetic energy as a function of its moment of inertia and angular velocity. A flywheel is a mechanical device which stores energy in the form of rotational momentum and also it helps to smooth the rotation. Torque can be applied to a flywheel to cause it to spin, increasing its rotational momentum. Therefore, some amount of energy is stored on the RBS flywheel.

The energy stored inside of a flywheel is related to the speed of its rotation and its moment of inertia. This stored energy also charge the battery when the system is disengaged until it stops rotation, the stored energy is shown in Table 5.9.

Considerations: ω_1 and ω_2 is initial and final angular velocity of RBS flywheel, ω_2 is zero,

Mass of engine flywheel 9.8 kg,

Radius of engine flywheel 100 mm, and

Moment of inertia I

$$I = \frac{1}{2}mr^2 = 9.8 \times 0.05^2 = 0.0123 \text{ Kgm}^2$$

Therefore, absorbed rotational kinetic energy depending on the pinion gear rotational speed.

Table 5.9. Stored kinetic energy on RBS flywheel

Pinion gear or alternator Rotational speed (rpm)	Energy stored on RBS flywheel (K_J)
0	0
500	16.8
1000	67.4
1500	151.7
2000	269.7
2500	421.5
3000	606.9
3500	826.2
4000	1079
4500	1365.7
5000	1686
5500	2040
6000	2427.9
6500	2849.4
7000	3304.7
7500	3793.6
8050	4370.4

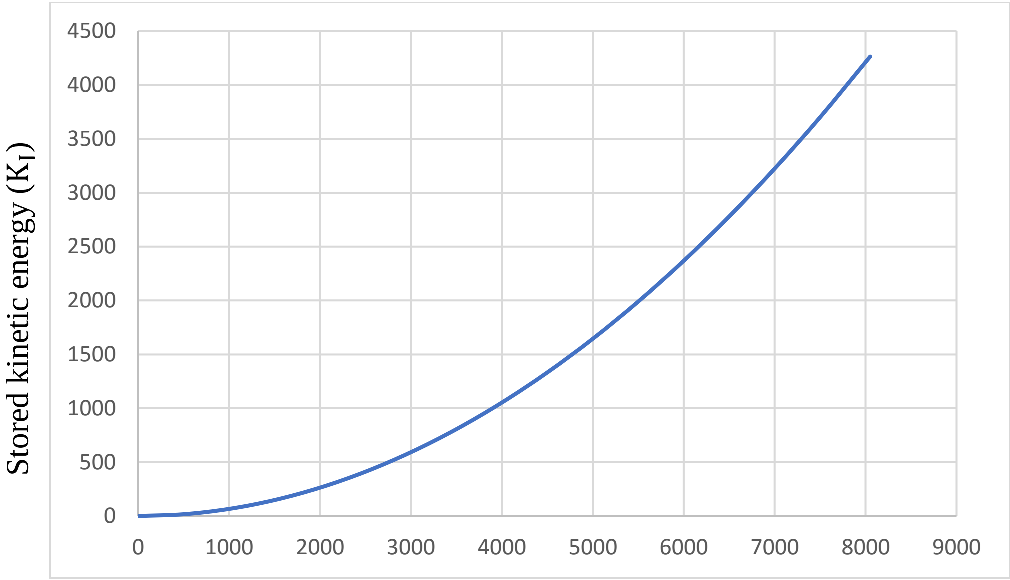


Figure 5.2. Stored kinetic energy on RBS flywheel with pinion gear rotational speed

The stored kinetic energy on the RBS relation with pinion gear rotational speed is shown in Figure 5.2. and Table 5.9. So, during braking high rotational speed has high absorbed rotational kinetic energy due to KERS.

5.5.3 Effect of mass of RBS flywheel on stored energy

The mass of RBS flywheel is greatly affecting the stored energy. If the mass of flywheel is increasing the stored energy also increased. The mass of flywheel is directly proportional with the stored energy at different mass of flywheel is shown below in Table 5.10, and Figure 5.3.

Table 5.10. Stored kinetic energy on RBS flywheel at different mass of flywheel

Pinion gear or alternator Rotational speed (rpm)	Angular velocity rad/sec	Energy stored on RBS flywheel (K_J)		
		m_1=9.8 kg	m_2=12 kg	m_3=15 kg
0	0	0	0	0
1000	104.7	67.4	82.2	102.5
2000	209.4	269.7	328.9	410.1
3000	314.2	606.9	740.2	922.8
4000	418.9	1079	1315.9	1640.5
5000	523.6	1686	2056.2	2563.3
6000	628.3	2427.9	2960.9	3691.2
7000	733	3304.7	4030	5024.2
8050	842.9	4370.4	5329.8	6644.5

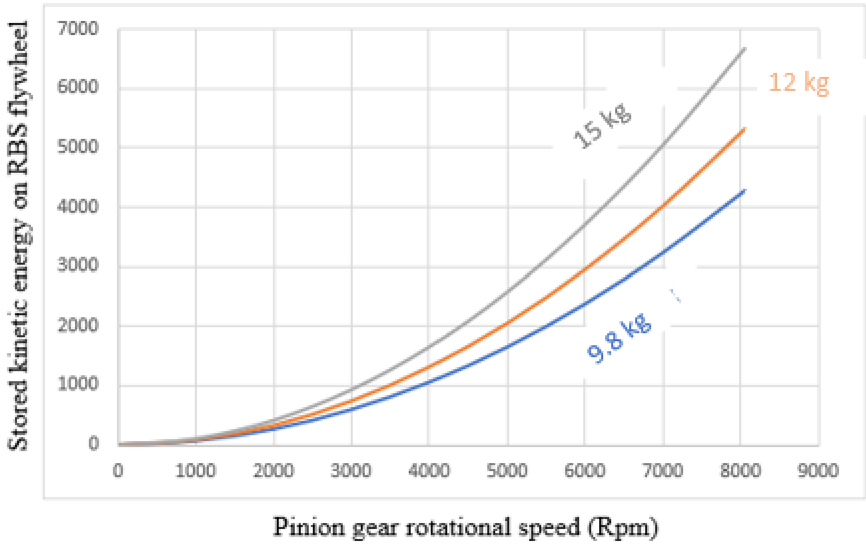


Figure 5.3. Stored kinetic energy on RBS flywheel at different mass Vs pinion gear rotational speed

5.5.4 Braking distance, braking time, waste energy and recovered energy

Direct gear engagement to the engine flywheel regenerative braking system is recover energy during braking (the time between brake pedal depressed and brake pedal released). Without kinetic energy recovering mechanism waste energy is increased with increasing braking distance and also using kinetic energy recovering mechanism recovered energy is increased with increasing braking distance. Similarly, the higher braking time leads to high amount of energy waste and recovered during braking. Therefore, both braking distance, braking time, waste energy and recovered energy are directly proportional with each other.

5.5.5 Current and power developed (recovered)

During breaking the rotational speed of pinion gear is increased from rest by the teeth ratio of the engine flywheel and pinion gear and it delivers electrical power by using the alternator. From the performance curve of 33 - SI brushless alternator the developed current is determined by using the rotational speed of pinion gear (rotational speed of alternator is the same as the rotational speed of pinion gear) from Table A-12 alternator Output Characteristics Curve (Heavy duty brushless alternator service manual) And also, Power developed can be determined by using the following formula.

$$P = V \times I$$

Where: P – Recovered power during braking

V –voltage = 24 V

I – Current

Therefore, the recovered power is shown in Table 5.11 and Figure 5.4.

Table 5.11. Recovered powers during braking

Rotational speed of pinion gear or alternator (rpm)	Developed current by alternator (A)	Recovered power during braking (Watt)
1700	0	0
2000	20	480
2250	33.3	799.3
2500	43.75	1050
3000	60	1440
3500	73.3	1759.2
4000	85.4	2050
4500	93.75	2250
5000	100	2400
5500	103.5	2484
6000	105	2520
6500	106	2544
7000	106.5	2555
7500	106.5	2556
8000	106.6	2558.4
8050	106.6	2558.4

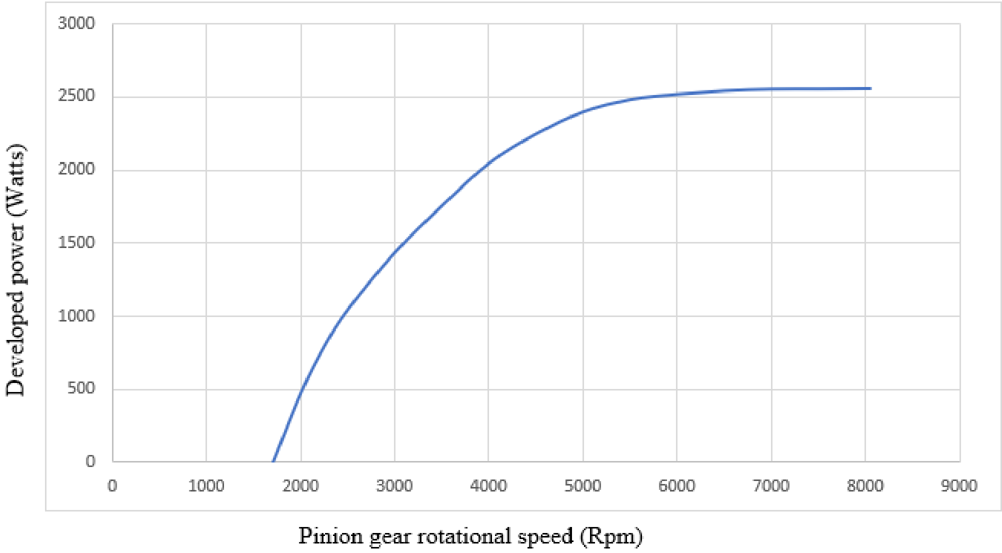


Figure 5.4. Recovered power during braking Vs pinion gear rotational speed

5.5.6 Total kinetic energy recovered absorbed during braking

The motion of the body is a pure rotational motion. The total kinetic energy is the summation of rotational kinetic energy absorbed from engine flywheel and the rotational kinetic stored on the RBS flywheel.

Table 5.12. Total kinetic energy recovered using RBS

Engine rotational speed (rpm)	Waste kinetic energy KE_w (K_J)	Recovered kinetic energy KE_R (K_J)	Pinion gear or alternator rotational speed (rpm)	kinetic energy on RBS flywheel KE_S (K_J)	Total recovered energy $KE_R + KE_S$ (K_J)	Efficiency η(%)
0	0	0	0	0	0	0
250	366.7	256.7	875	51.6	308.3	84
500	1466.7	1026.7	1750	206.5	1233.2	84
750	3300	2310.1	2625	464.7	2774.8	84
1000	5866.9	4106.8	3500	826.2	4933	84
1250	9167	6416.9	4375	1290.9	7707.8	84
1500	13200.6	9240.4	5250	1858.9	11099.3	84
1750	17967.5	12577.2	6125	2530.1	15107.3	84
1900	21179.6	14825.7	6650	2982.5	17808.2	84
2050	24655.8	17259	7175	3471.9	20730.9	84
2300	31036.1	21725.2	8050	4370.4	26095.6	84

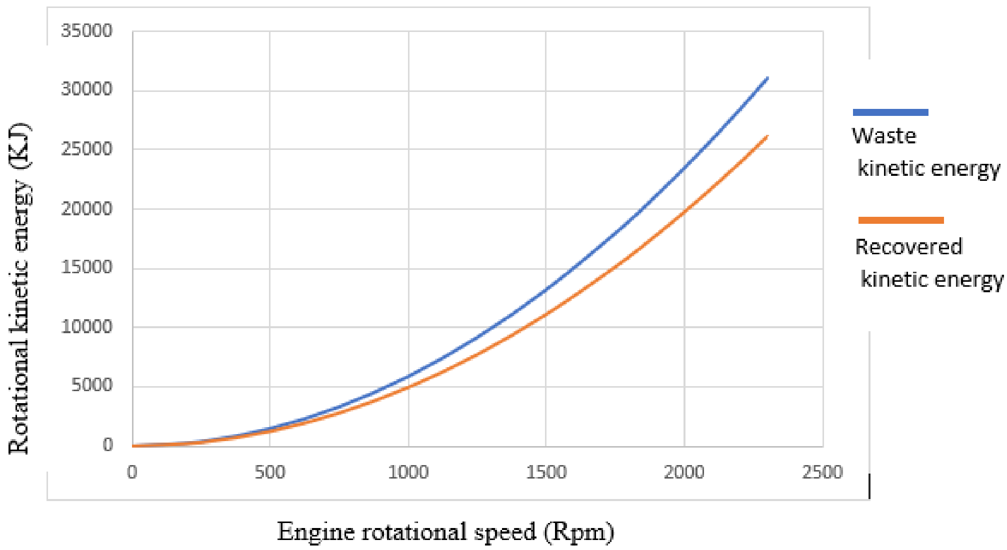


Figure 5.5. Total kinetic energy recovered Vs engine rotational speed

The stored kinetic energy on the RBS relation with pinion gear rotational speed as sown in Figure 5.5. and Table 5.12. So, during braking at high rotational speed has absorbed high amount of rotational kinetic energy using KERS. The above Figure 5.5 shows the rotational kinetic energy lost during the time of braking without RBS and the rotational kinetic energy absorbed or recovered by direct gear engagement regenerative braking system to the flywheel by taking different engine rotational speed. The kinetic energy loss and recovered of the vehicle is increase with increase its rotational speed of the engine but the efficiency of the vehicle is constant at all rotational speed. When the car is braking at 2300 rpm has 31036.1 K_J kinetic energy is loss and by using KERS this waste energy recovered. Total energy recovered is the lost energy multiplied by the correction factor plus the stored energy on the RBS flywheel. Therefore, the total kinetic energy recovered by the system at 2300 rpm is 26095.6 K_J and the efficiency of the system is 84%.

5.6 Ansys simulation result

The model is created by using solid work 2017 and it's imported into the ANSYS 18.1 software (FEA is completed by ANSYS workbench 18.1). FEM is a numerical method for solving a system of governing equations over the domain of a continuous physical system, which is discretized into simple geometric shapes called a finite element. The steps of FEM analysis are:

1. Define material properties
2. Create or import geometry
3. Create finite element mesh
4. Apply loads and boundary conditions
5. Define analysis type, options, and then solve
6. Review and evaluate the results

5.6.1 Static structural analysis using ANSYS

5.6.1.1 Flywheel with pinion gear

Input data:

Table 5.13. Input data for simulation of flywheel with pinion gear

Material	Grey cast iron
Model type	Linear elastic isotropic
Density	7200 kg m^3
poisons ratio	0.28
Ultimate tensile strength	240Mpa
Ultimate compression strength	820Mpa
Moment applies on pinion gear	253Nm

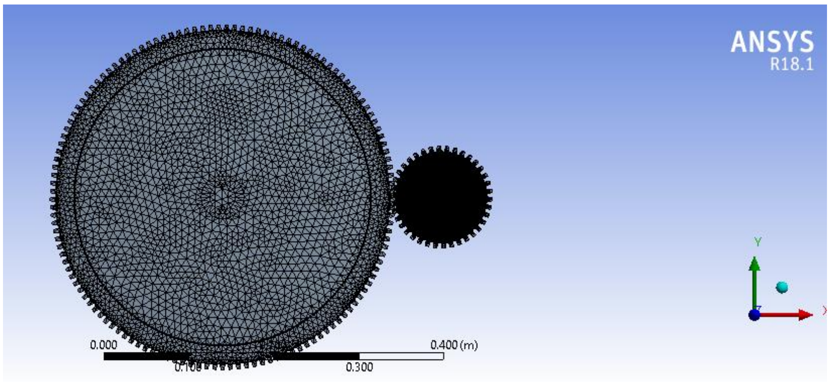


Figure 5.6. Mesh element of engine flywheel with pinion gear

Mesh element of fork lever is shown in Figure 5.6 it discretized in to   nodes and   elements with a fine mesh.

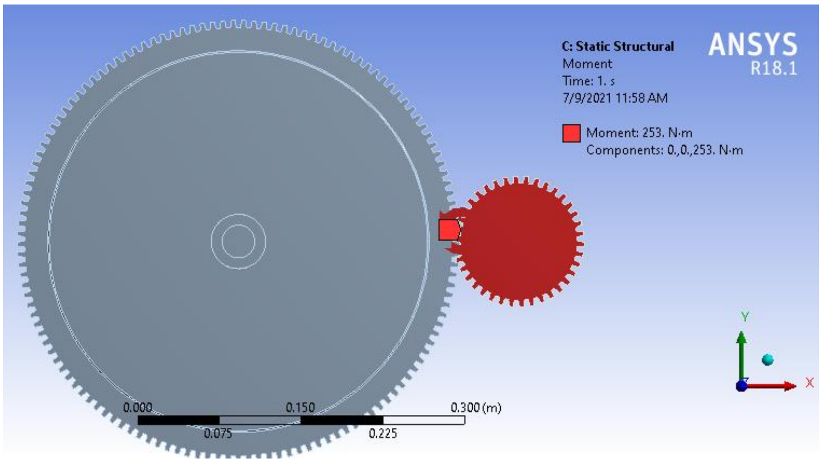


Figure 5.7. Moment applied on pinion gear

The above Figure 5.7 shows the moment (253 Nm) applied on the pinion gear from the source. And both the pinion gear and engine flywheel are rotate in opposite direction.

Total deformation:

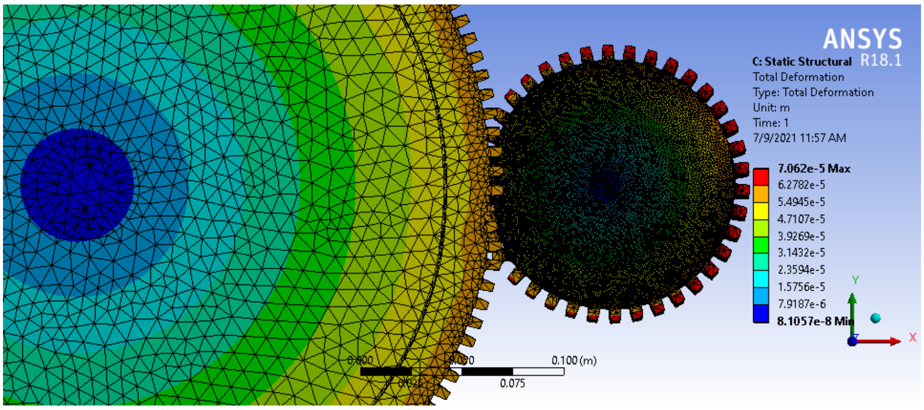


Figure 5.8. Total deformation on engine flywheel with pinion gear

The total deformation is shown in the Figure 5.8 The deformation is not the same throughout. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. The maximum deformation is 7.062e-5 m. minimum amount of total deformation is occurred at the origin of the gear.

Equivalent elastic strain:

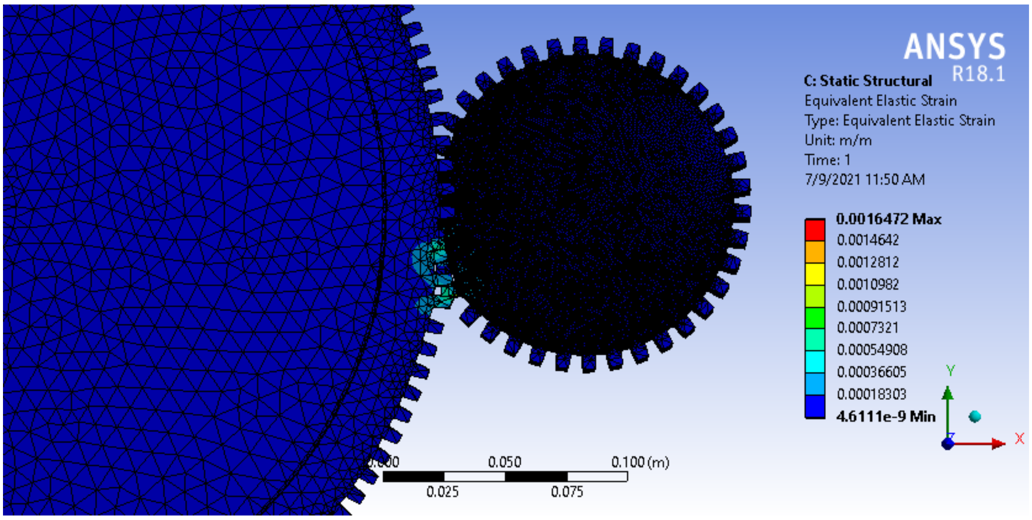


Figure 5.9. Equivalent elastic strain on engine flywheel with pinion gear

Equivalent elastic strain shown in Figure 5.9 the maximum value of strain is occurred at the contact region of the gear. The portion in red color shows that the strain in that region is maximum and the portion in blue color shows that the strain in that region minimum.

Equivalent (von- Mises) stress:

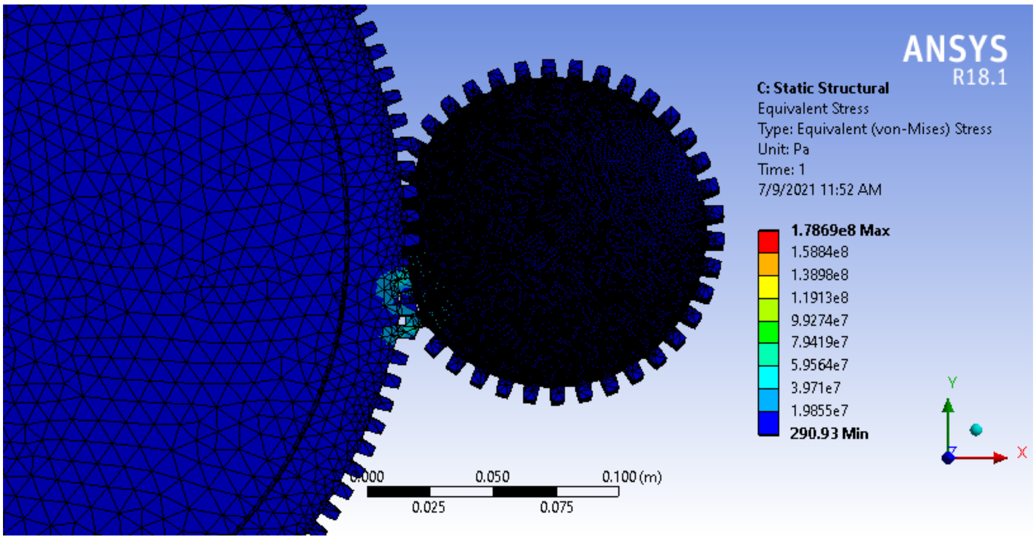


Figure 5.10. Equivalent (von- Mises) stress on engine flywheel with pinion gear

Equivalent (von- Mises) stress shown in Figure 5.9 the maximum value of stress is occurred at the contact region of the gear. The portion in red color shows that the stress in that region is maximum and the portion in blue color shows that the stress in that region minimum.

5.6.1.2 Fork lever

Table 5.14. Input data for simulation of fork lever

Material	Aluminum alloy
Model	Linear elastic isotropic
Density	2770 Kg m^3
poisons ratio	0.3
Ultimate tensile strength	460 MPa
Compressive Yield Strength	250 MPa
Load applied on the fork lever	15.71 N

Mesh on fork lever:

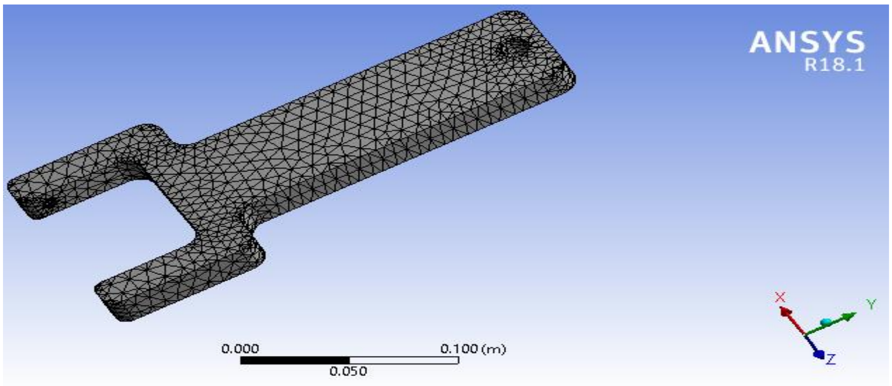


Figure 5.11. Mesh element on fork lever

Mesh element of fork lever is shown in Figure 5.19 it discretized in to 15027 nodes and 8224 elements with a fine mesh.

Load applied on fork lever:

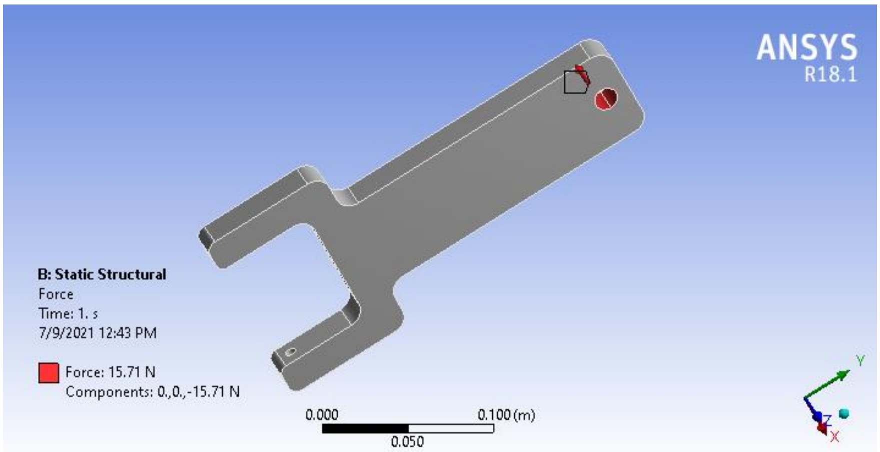


Figure 5.12. Load applied on fork lever

Load applied on the fork lever is shown in Figure 5.20. the load applied on the lever is due to the piston actuation during braking is 15.71 N.

Support on fork lever:

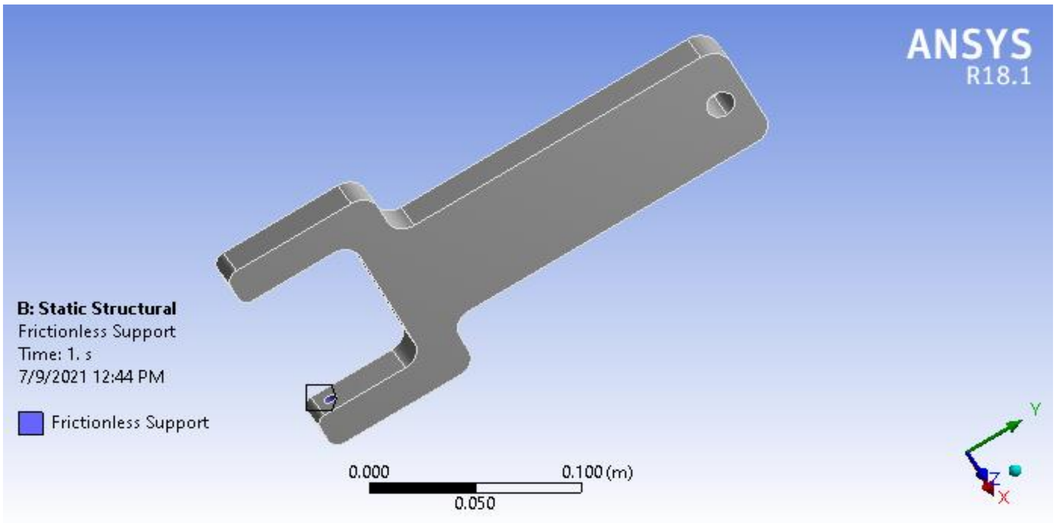


Figure 5.13. Support on fork lever

Frictional support on fork lever is shown in Figure 5.13 is frictionless support.

Total deformation on fork lever:

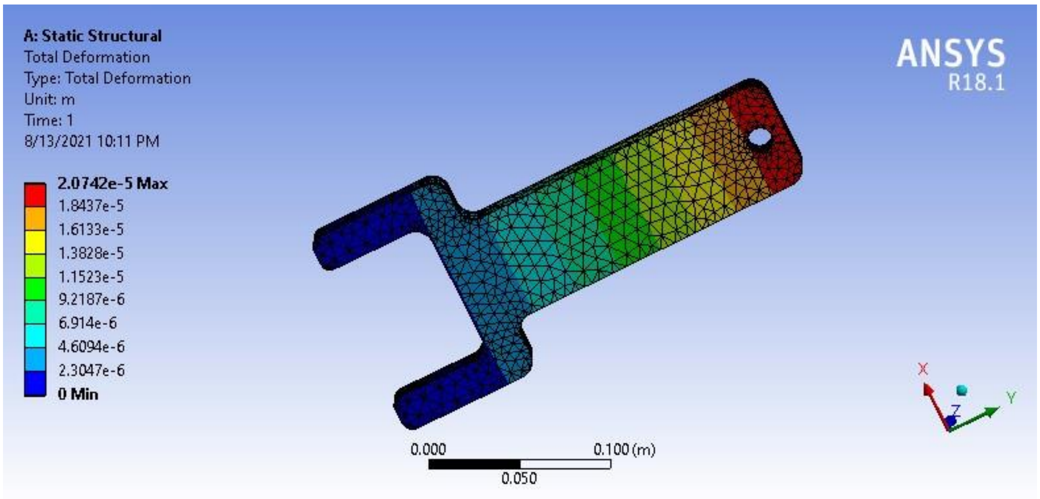


Figure 5.14. Total deformation on fork lever

The above Figure 5.14 shows the total deformation of the fork lever. The maximum deformation is occurred on the end of the pneumatic cylinder. The deformation is not the same throughout. The portion in red color shows that the deformation in that region is maximum and the portion in blue color shows that the deformation is minimum in that region.

Equivalent elastic strain on fork lever:

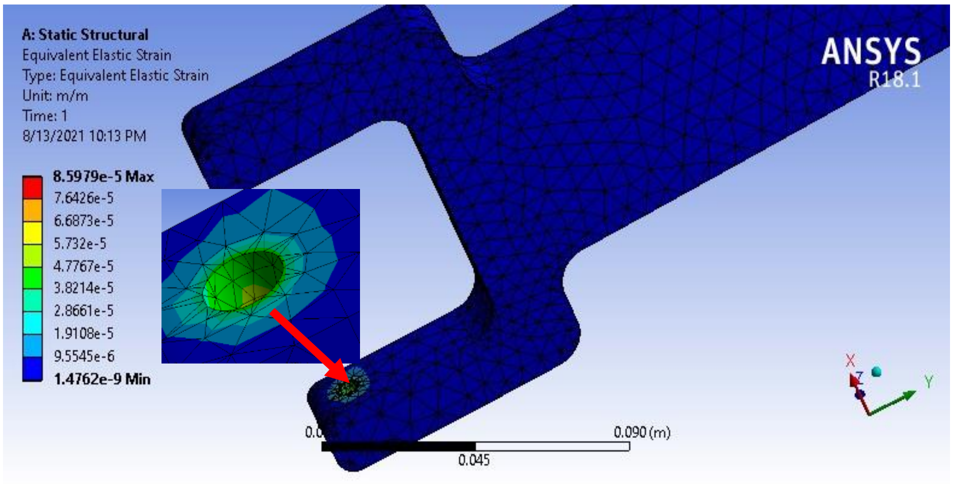


Figure 5.15. Equivalent elastic strain on fork lever

The above Figure 5.15 shows the Equivalent elastic strain on fork lever. The maximum Equivalent elastic strain on fork lever is occurred by the applied load is not the same throughout the body. The portion in red color shows that the deformation in that region is maximum and the portion in blue color shows that the deformation is minimum in that region.

Equivalent (von-Mises) stress on fork lever:

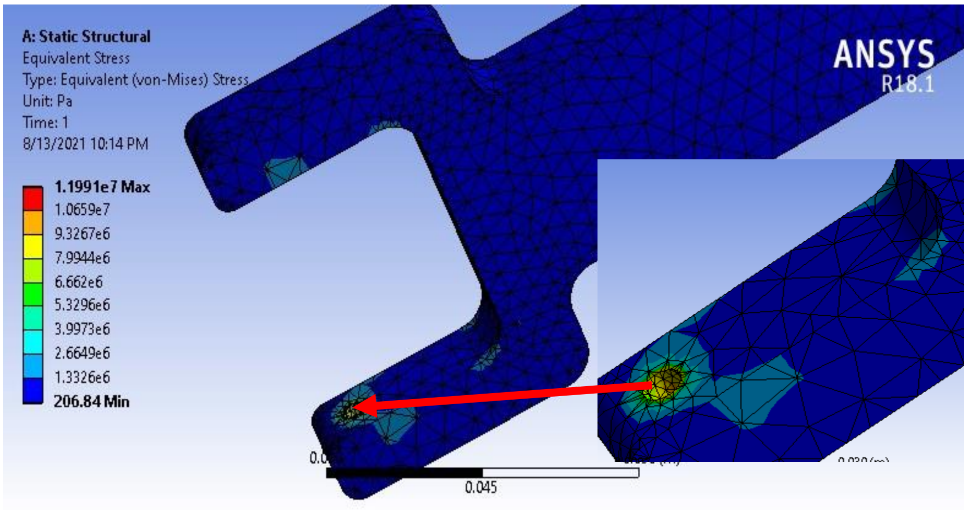


Figure 5.16. Equivalent (von-Mises) stress on fork lever

The Equivalent (von-Mises) stress on fork lever is shown in the above Figure 5.16 indicates the stress distribution is not constant throughout the body. The red color indicates the maximum stress and blue color indicates minimum value of the stress.

5.6.1.3 Splined shaft

Table 5.15. Input data for simulation of splined shaft

Material	Aluminum alloy
Model	Linear elastic isotropic
Density	2770 kgm^3
poisons ratio	0.33
Ultimate tensile strength	310 MPa
Ultimate compression strength	280 MPa
Vertical load applied on shaft	
Clutch	14.7 N
Flywheel	96.1 N

Mesh

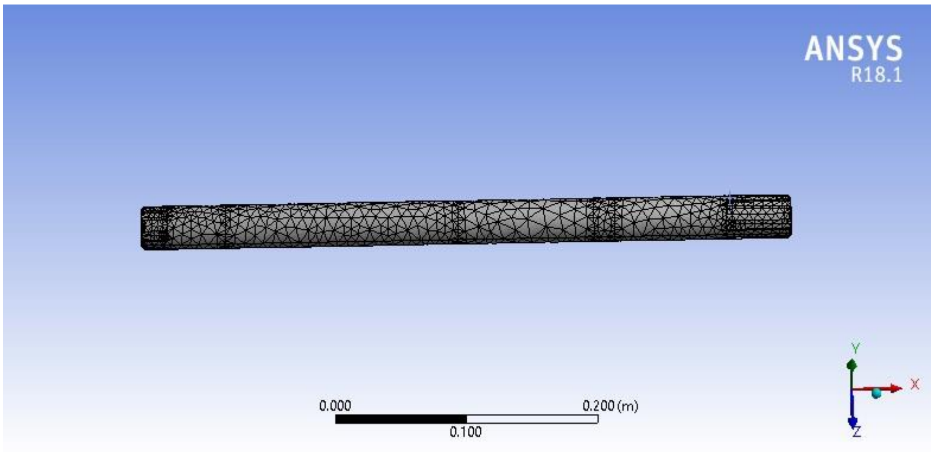


Figure 5.17. Mesh element on splined shaft

Mesh element of splined shaft is shown in Figure 5.17 it discretized in to 12481 nodes and 6798 elements with a fine mesh with mechanical preference.

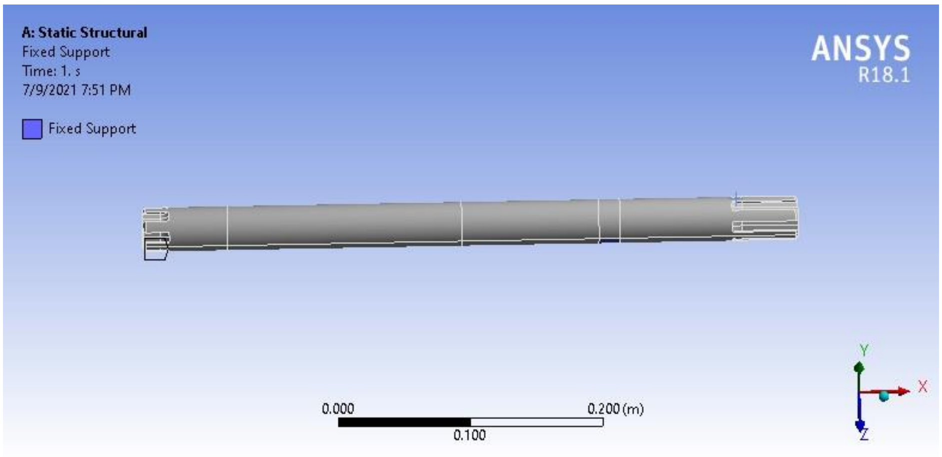


Figure 5.18. Support on splined shaft

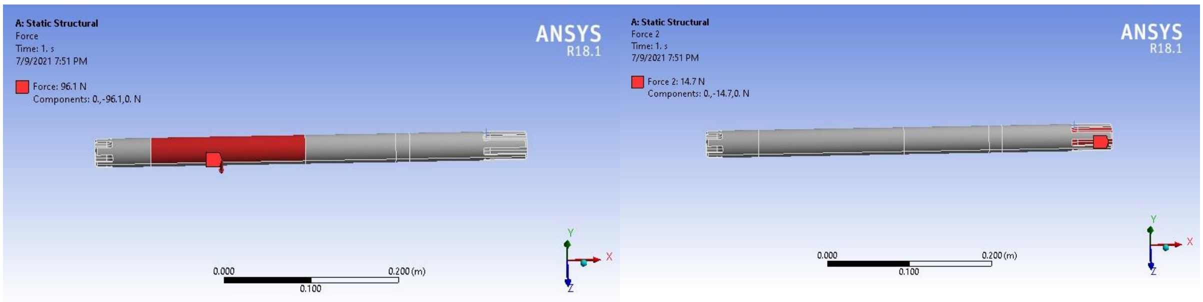


Figure 5.19. Load applied on splined shaft

The load apply on the shaft are vertical load by the RBS flywheel and the clutch palate. The above Figure 5.16 shows the applied load on the shaft 96.1 N by the RBS flywheel and 14.7N by the Sigle plate frictional clutch

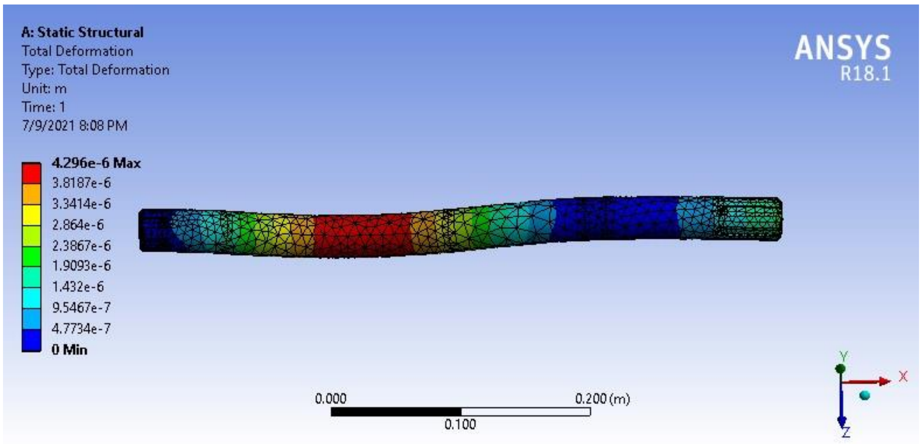


Figure 5.20. Total deformation on splined shaft

The total deformation of shaft is shown in the Figure 5.20 the deformation in the shaft is not the same throughout. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. The maximum deformation is 4.296e-6 m and the minimum total equivalent of shaft is 0 m.

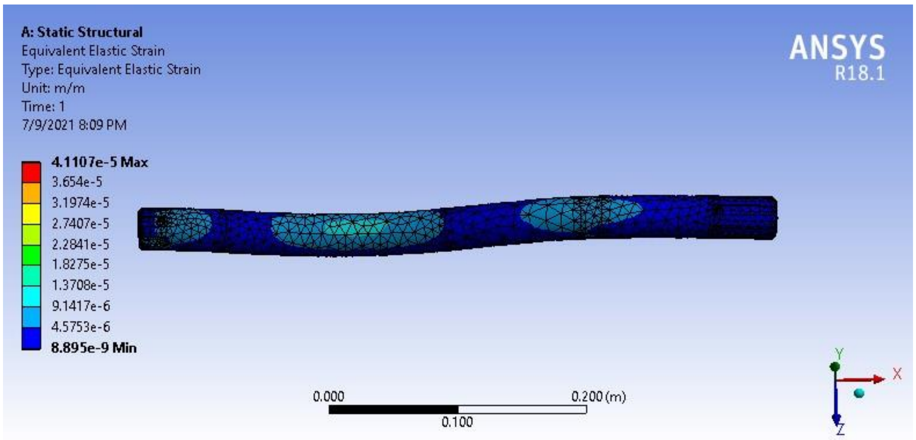


Figure 5.21. Equivalent elastic strain on splined shaft

Equivalent elastic strain on splined shaft is shown in the Figure 5.29. the maximum value of equivalent elastic strain 4.1107 e -5 and minimum value of shear stress is 8.895 e-9.

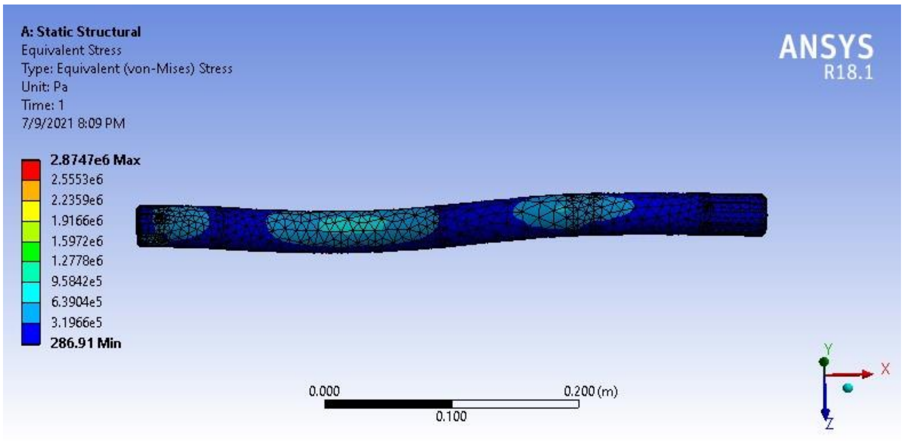


Figure 5.22. Equivalent (von- Mises) stress on splined shaft

Equivalent (von- Mises) stress on splined shaft is shown in the Figure 5.30 above the maximum value of equivalent elastic strain 2.8747e6 pa and minimum value of shear stress is 286.91 pa.

5.6.1.4 Clutch disc

Table 5.16. Input data for simulation of single plate frictional clutch disc

Material	Rigid molded Asbestos pad
Model	Linear elastic isotropic
Density	1890 kgm^3
Shear modulus	5460 Pa
Poisons ratio	0.25
Ultimate tensile strength	6800 Pa
Load applied on shaft	314.2

Mesh:

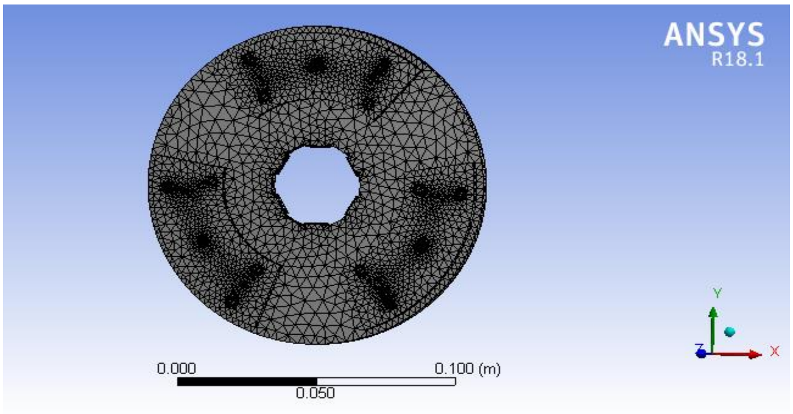


Figure 5.23. Mesh element on Sigle plate frictional clutch disc

Mesh element of clutch disc is shown in Figure 5.23. it discretized in to 78572 nodes and 46481 elements with a fine mesh.

Deformation:

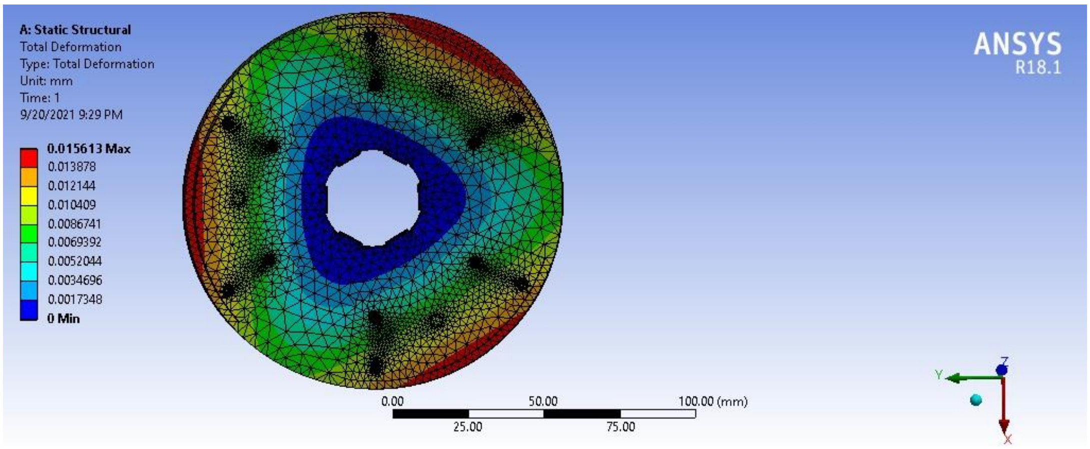


Figure 5.24. Total deformation on clutch disc

The total deformation of clutch disc is shown in the Figure 5.22 the deformation in the clutch disc is not the same throughout. The portion in red color shows that the deformation in that region is maximized and the portion in blue color shows that the deformation is minimized in that region. The maximum deformation is occurred on the upper limit of the clutch disc.

Stress:

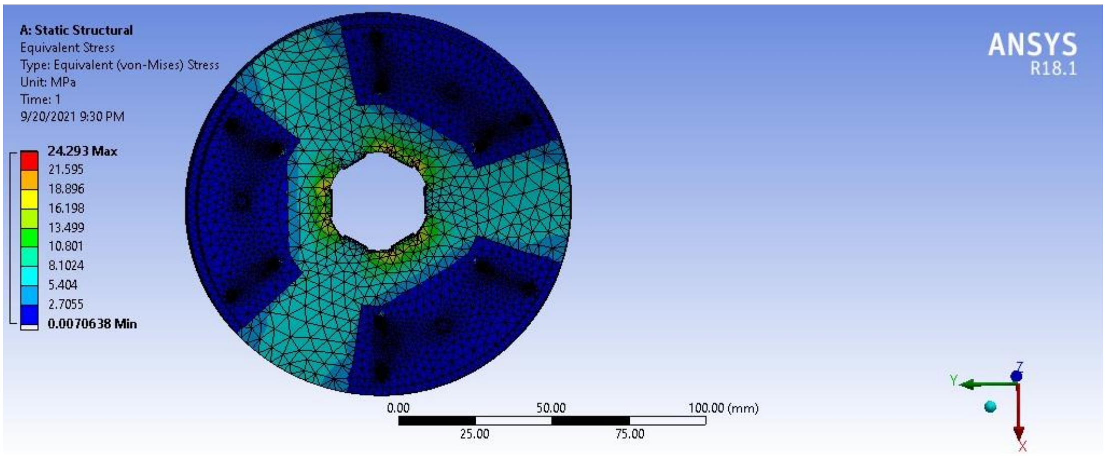


Figure 5.25. Equivalent (von- Mises) stress on clutch disc

The Equivalent (von-Mises) stress on clutch disc is shown in the above Figure 5.25 indicates the stress distribution is not constant throughout the body. The red color indicates the maximum stress and blue color indicates minimum value of the stress. Maximum stress is occurred at the center of the clutch disc around the shaft.

Strain:

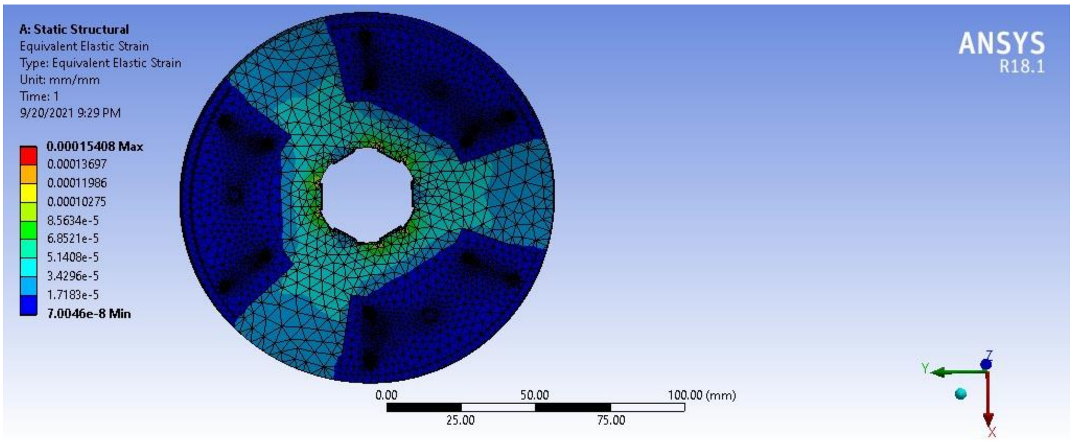


Figure 5.26. Equivalent elastic strain on clutch disc

The above Figure 5.26 shows the Equivalent elastic strain on clutch disc. The maximum Equivalent elastic strain on clutch disc is occurred by the applied load is not the same throughout the body. The portion in red color shows that the deformation in that region is maximum and the portion in blue color shows that the deformation is minimum in that region. The maximum strain is occurred at the center of the clutch disc.

CHAPTER SIX

6. CONCLUSION AND RECOMMENDATION

6.1. Conclusion

Regenerative braking is an energy recovery mechanism which slows a vehicle or object by converting its kinetic energy in to a form of electrical energy which can be either used immediately or stored until needed. The main objective was to design and development of direct gear engagement regenerative braking system to the engine flywheel. This system is installed on Anbesa bus use a light weight DC – Generator receiving drive from the engine; this is because the bus propulsion drive is fully mechanical. The work includes the design of the regenerative braking system consisting of three different sub systems which are mechanical, pneumatic, and electrical system, their components and material selection for component construction. And the system installation is done according to the power flow.

- The mechanical sub system receives drive from the flywheel through a pinion gear made from grey cast iron which is the same as that of the flywheel with 38 teeth. The drive from the pinion is then transferred through a pneumatically actuated clutch with a 60 mm and 34 mm outer and inner radius limits of friction surface respectively. Then the drive from the clutch goes to a Ø30 mm aluminum shaft with six splines on each end to transfer the drive torque. Finally, to have an efficient and continuous drive the mechanical subsystem is fitted with a high strength low alloy steel flywheel of 9.8 kg mass, 163 mm width and 100 mm diameter.
- The pneumatic sub system operates with air pressure of 785 kPa from the air brake system to actuate a Ø20 mm bore actuator (pneumatic cylinder) to be utilized for the clutch engagement and disengagement.
- The electrical subsystem converts the mechanical energy from the mechanical subsystem into an electrical energy by using a 24 V – 100 A DC generator. The generated electrical energy is then stored in two 12 V – 65 A Lithium – ion (Li – ion) batteries connected in series.

Anbesa bus apply brake at different speed due to traffic jam, road condition, loading and unloading of passenger. The kinetic energy absorbed during braking is depending on rotational speed of the engine, braking time and mass of the regenerative braking system flywheel. Kinetic energy absorbed is increases by increasing rotational speed of the engine, RBS flywheel and braking time.

In general, static structure analysis of components of RBS is done using ANSYS software and from the design analysis waste energy of Anbesa bus during braking was recovered by using direct gear engagement regenerative braking system to the engine flywheel. Kinetic energy is recovered by the system is done at different rotational speed of engine. The maximum recovered rotational kinetic energy is occur at 2300 rpm of engine rotation it recovers 26095.6 K_J and the efficiency of the system is found to be 84%.

6.2. Recommendation and Feature scope

The regenerative braking system designed in this thesis is for Anbsa bus that are entirely propelled by mechanical drive. For futuristic work, one can implement this system on all types of diesel engine vehicles (van, truck, Isuzu etc.) to recover the waste energy.

The flywheel on the current design has good performance, but from Table 3.7 it is understood that carbon fiber reinforced polymers (CFRP) have much higher efficiency rating. So, to have a more efficient flywheel CFRP should be used. But using CFRP for flywheel material has two main draw backs which are; the flywheel needs to be operated at higher speed and the cost of CFRP, which is very expensive. Therefore, if a series of highly efficient gearboxes are used so that the DC – generator is operated at higher RPMs and future improvements in CFRP production reduces costs; it is possible to use a flywheel made from carbon fiber. And this work is only covered the design and analysis of RBS, it is recommended to develop the prototype and experimental validation test.

The last but not the least; for a more complete and thorough design working with bus manufacturers or local franchise dealers is advisable. This partnership is needed to obtain different vehicle performance parameters, component dimensions and other technical assistances that might aid the design work which are difficult to come by when working solo.

REFERANCE

Ahn JK, Jung KH, Kim DH, Jin HB, Kim HS, Hwang SH. (2009). Analysis of a regenerative braking system for Hybrid Electric Vehicles using an electro-mechanical brake. International Journal of Automotive Technology. Volume 2: 229-34 pp.

A. Bhatia (2017). Basic Fundamentals of Gear Drives, Continuing Education and Development, Inc. 22 Stonewall Court Woodcliff Lake, NJ 07677, pp: (877) 322-5800.

Anaheim Automation Application assistance end customer service from U.S.A, CA 92807 established 2016, https://www.anaheimautomation.com/manuals/forms/gearbox.

Author's Guide (2016). Infogalactic, the planetary knowledge core contributors. Regenerative brake [Internet]. Infogalactic: the planetary knowledge core,2016, 20-55 p.

Babji, G., Gowrisankar, K., & Rao, D.M. (2018). Thermal and tribological behaviour of non-asbestos brake pads of passenger Cars.

B. Bolund, H. Bernhoff, and M. Leijon, (2007). Flywheel energy and power storage systems," Renew. Sustain Energy Rev., vol. 11, no. 2, 235–258 pp.

Bhandari P, Dubey S, Kandu S, Deshbhratar R. (2017) Regenerative braking systems (RBS). Int. J. Sci. Eng. Res. Volume 8, 71-4 pp.

Budynas-Nisbett (2008) Shigley's Mechanical Engineering Design, Printed in the United States of America, The McGraw-Hill Companies, Eighth Edition, ISBN: 0-390-76487-6

Corporation industry pneumatic Manufacture actuator Control energy saving (SMC), (2015)"The Pneumatics Catalog," no. North America Edition 31-32 pp.

Dabi Beyan (2017). Mechanical Regenerative braking system for Higer bus, Unpublished" MSc Thesis Submitted to Adama science and technology university, ………..pp.

Dhotare sagar, Roshan Hegde, Nikhil Kadam, Shubham Chavan, and Akash Ingale (2019). Design and Development of 360°Rotating Dumper, ISSN: 2348 7208, Volume 1, Issue 1,13-17 pp.

DGS-Berlin (2014). PV Systems for Rural Health Facilities in Developing Areas, International energy agency photovoltaic power systems program, ISBN: 978-3-906042-31-2 page 5-38.

Electro mate world press (2014). robotic and mechatronic solution, technical support information, Encyclopedia of Educational Research volume 2, 25-29 pp.

Electreka, (2018). Regenerative braking, how it works and is it worth it in small EVs, regenerative-braking-how-it-works, volume 3, 17-18 pp.

Fritz faulhaber (2002) president of engineering micro-Mo, clear water, fal, a-second-look-at-gearbox-efficiencies Retrieved 2021-05-25

Gao, Yimin, and Mehrdad Ehsani. (2001). Electronic Braking System of EV and HEV---Integration of Regenerative Braking, Automatic Braking Force Control and ABS. SAE Transactions 110 ,576-82 pp.

Gao Y, Chen L, and Ehsani M. (2000). Investigation of the effectiveness of regenerative braking for EV and HEV. SAE transactions.3184-90 pp.

Grandone M, Naddeo M, Marra D, Rizzo G. (2016). Development of a regenerative braking control strategy for hybridized solar vehicle, IFAC-Papers OnLine,497-504 pp.

G. Arun Deesh, R. Kannan, S. P. Gokul, M. Gopal, and V. Gowtham Mani Balaji (2017). Kinetic energy recovery system in bicycle by using flywheel, ISSN 2351-8014 Volume. 30,488-492 pp.

Harshit Pandey, Shivam Shukla, Prof. Pratesh Jayaswal (2020) regenerative braking system based on hydraulic, motor, and flywheel, e-ISSN: 2395-0056, p-ISSN: 2395-0072, Volume: 07 Issue: 06, 4726-4729 pp.

Heisler H, F.I.M.I, M.S.O.E, M.I.R.T.E, M.C.I.T, and M.I.L.T, (2002). Advanced vehicle technology, second edition, ISBN 0750651318, 540-542 pp.

Introduction to Automotive Electrical and Electronic Systems, (2015). Encyclopedia of Educational Research, 1-16 pp.

Jack A. Collins, Henry R. Busby & George H. Staab (2010). Mechanical design of machine elements and machines, A Failure Prevention Perspective, Second Edition, ISBN-13 978-0-470-41303-6, 798-1025 pp.

Jack Erjavec (2010). automotive technology, a system approach, 5th Edition, ISBN-13: 978-1428311497, ISBN-10: 1428311491,188-195 pp.

James A. Sullivan (2002). fluid power theory and application,3 edition, ISBN 0-13-323080-5, 391-399 pp.

Jincui li. (2017). Study on a new kind of electromagnetic mechanical coupled regenerative braking system for electric vehicles. International Journal, volume 1, 60 pp.

J. Li, E. Murphy, J. W innick, and P. Kohl (2001). Studies on the cycle life of commercial lithium-ion batteries during rapid charge–discharge cycling," J. Power Sources, vol. 102, no. 1–2, 294–301 pp.

J.M.K.C. Donev, Adhikari S, Farhan S, (2018). Energy Education - Regenerative braking [Online]. Available: https://energyeducation.ca/encyclopedia/Regenerative_braking. [Accessed: July 1, 2021].

Kebede, M. H. (2018). Design of standalone PV system for a typical modern average home in Shewa Robit Town-Ethiopia. American Journal of Electrical and Electronic Engineering, 6(2), 72-76.

Kumar, R. (2015). Regenerative brake: to harness the kinetic energy of braking. Journal of Emerging Technologies and Innovative Research, 2(1)

Liu, C. Xu, D. Weng, J. Zhou, S. Li, W. Wan, Y. Jiang, S. Zhou, D. Wang, J. Huang, Q. (2020). Phase Change Materials Application in Battery Thermal Management System: A Review. Materials volume 3, 46-242 pp.

Mach III (2021) motion control solution clutch, inc, air/spring stop and start the pilot clutch-brake working principle, USA,1-2 pp.

Makeitfrom, (2021). Material properties database, available at https://www.makeitfrom.com /Material-properties, Retrieved 2021-03-09

M. Naemah and Baharuddin (2013), regenerative braking, international Islamic university Malaysia, Review of Educational Research pp 4-11.

Mobbs, M., & Sussman, G. (2016). Lead acid vs. lithium-ion battery comparison. In Sustainable Projects. Stanford University.

M.Galal Rabie (2009). Fluid power engineering, ISBN 198-0-07-162606-4,volume 4, 367-399 pp.

Maier, M. jhonson, D. rafael. (2020). Guidelines and trends for next-generation rechargeable lithium and lithium-ion batteries. Chemical Society Reviews, 49(5), 1569-1614.

M. Megra, “Design and Simulation of Standalone PV System for Ethiopian Rural Health Center,” 2018.

Yoong, M. K., Gan, Y. H., Gan, G. D., Leong, C. K., Phuan, Z. Y., Cheah, B. K., & Chew, K. W. (2010, November). Studies of regenerative braking in electric vehicle. In 2010 IEEE Conference on Sustainable Utilization and Development in Engineering and Technology (pp. 40-45). IEEE.

Mahmoud Galal and El-Din Mohamed Rabi (2009) Fluid Power Engineering. Introduction to Pneumatic Systems https: //www.accessengineeringlibrary.com/content/book.

M.r. Adler (2016). Reliability Engineering Snapshot, Illustrated Case Studies in the Maintenance Reliability Engineering World of Failure Analysis, Predictive Maintenance, and Non-Destructive Evaluation, volume 3, Issue 4, 28-77 pp.

Sandeep Shrikant Karekar and Abhijeet Narayanrao Naik (2017). A Review on Design and Optimization of Pneumatic Actuator using Various Mechanisms and Techniques, Volume 4, Issue 3, ISSN online: 2349-6010, 139-141 pp

Nagaraja. M, Dyson Bruno.A, Anand Kumar.P, Gowtham.G, Hariharan.N, Jaya Srinath.N, (2018). Performance testing of a Regenerative braking systems, international journal of engineering research & technology (ijert) etedm, Volume- 6, Issue- 04.

Nashit S, Adhikari S, Farhan S, Avinash S, Gambhire A. (2016). Design, fabrication and testing of regenerative braking test rig for BLDC motor. Int. Res. J. Eng. Technol. IRJET. Volume 3:1881-4 pp.

Patil SK. (2012). Regenerative braking system in automobiles. International Journal of Research in Mechanical Engineering and Technology, volume 2,45-6 pp.

Pratte, David (2020). "Drivetrain Power Loss - The 15% "Rule"- Modified Magazine". Super Street Online. motor trend group, LLc Retrieved 2021-06-10

P. Bhandari, S. Dubey, S. Kandu, and R. Deshbhratar, (2017). Regenerative Braking Systems (RBS), International Journal of Scientific & Engineering Research, Volume 8, Issue 2, ISSN 2229-5518, 26-45 pp.

P. Clarke, T. Muneer, and K. Cullinane, (2010). Cutting vehicle emissions with regenerative braking, Transp. Res.Part D Transp. Environ., volume.15, no.3,160–167 pp.

Robert L. Norton (2006). Machine design, An integrated approach third edition, ISBN: 0-13-148190-8, 635 pp.

R.S. Khurmi, and J.K. Gupta, (2005). Textbook of machine design, first multicolor edition, 470-1065 pp.

Saskatchewan driver's licensing and vehicle registration (2019). air brake manual, Encyclopedia 35 pp.

Sector Policies and Programs Division Office of Air Quality Planning and Standards U.S. Environmental Protection Agency (2010). Available and Emerging Technologies for Reducing Greenhouse Gas Emissions from Coal-Fired Electric Generating Units, Research Triangle Park, North Carolina 27711,1-48 pp.

Srinivasan, P., Umashankar, K. S., Kothalikar, U. M, & Deshpande, A. (2010). Optimization of Performance and CO2 Emission of a Common Rail DI Diesel Operated Vehicle Using AVL Cruise (No. 2010-01-1526). SAE Technical Paper.

S.J clegg (2018) review of regenerative braking system, university of Leeds, Sheffield and York, 1-24 pp.

Tiba, S, van Rijnsoever, F.J, Hekkert, M.P (2019). A systematic review of responsible entrepreneurship and corporate social responsibility literature. Corp. Soc. Responsible. Environ. Manga, 265–284 pp.

Tian, S., Hong, M., & Ouyang, M. (2009). An experimental study and nonlinear modeling of discharge I–V behavior of valve-regulated lead–acid batteries. IEEE transactions on Energy Conversion, 24(2), 452-458.

Tom Denton (2004). Automobile Electrical and Electronic Systems, Third edition, ISBN 0 7506 62190,129-149 pp.

Varocky BJ, Nijmeijer H, Jansen S, Besselink IJ, Mansvelder R. (2011). Benchmarking of regenerative braking for a fully electric car. TNO Automotive, Helmond & Technische Universiteit Eindhoven (TU/e), 20-22 pp.

Vishwakarma D, and Chaurasia S. (2016). Regenerative Braking System. Imperial international journal of eco-friendly technologies (IIJET), 30-3 pp.

Wager, Guido & Whale, Jonathan & Braunl, Thomas (2017). Performance evaluation of regenerative braking systems. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 095440701772865. 10.1177/0954407017728651, 732-7 pp.

Wager G, Whale J, Braunl T. (2018). Performance evaluation of regenerative braking systems. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. Volume 1,1414-27 pp.

World heritage encyclopedia, (2002) spline (mechanical), Encyclopedia of Educational Research ID: WHEBN 0016465514, 52 pp.

Y. A. Çengel and M. A. Boles, (2006) Thermodynamics: An Engineering Approach, fifth edition, McGraw-Hill,1-50 pp.

Zhang Jingming, Song Baoyu and Niu Xiaojing, (2008). Optimization of parallel regenerative braking control strategy," IEEE Vehicle Power and Propulsion Conference, Harbin, doi: 10.1109/VPPC.2008.4677420.1-4 pp.

Zhang J, Kong D, Chen L, Chen X. (2012). Optimization of control strategy for regenerative braking of an electrified bus equipped with an anti-lock braking system. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, volume 4, 494-506 pp.

Zhang, Junzhi & Lv, Chen & Yue, Xiaowei & Qiu, Mingzhe & Gou, Jinfang & Chengkun. (2013). Development of the Electrically-Controlled Regenerative Braking System for Electrified Passenger Vehicle, SAE Technical Papers. 2. 10.4271/2013-01-1463.

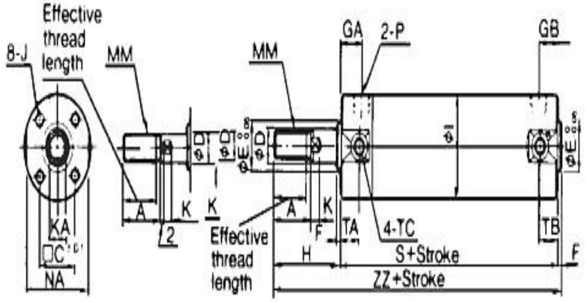
Zhang J, Lv C, Gou J, Kong D. (2012). Cooperative control of regenerative braking and hydraulic braking of an electrified passenger car. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, volume 5, 1289-302 pp.

Zirker, L., Francfort, J., & Fielding, J. (2006). Diesel Engine Idling Test (No. INL/EXT-05-00888). Idaho National Laboratory (INL).11pp.

APPENDIX

Appendix –A Standards

Table A- 1. Dimensions of Rubber Cushioned CG1 Series Pneumatic Actuator



Bore (mm)	Stroke range (mm)	Effective thread length	A	□C	øD	øE	F	GA	GB	øI	J	K	KA	MM	NA	P	S	TA	TB	*TC
20	~200	15.5	18	14	8	12	2	12	12	26	M4x0.7 Depth 7	4	6	M8x1.25	24	⅛	69	11	11	M5x0.8
25	~300	19.5	22	16.5	10	14	2	12	12	31	M5x0.8 Depth 7.5	5	8	M10x1.25	29	⅛	69	11	11	M6x0.75
32	~300	19.5	22	20	12	18	2	12	11	38	M5x0.8 Depth 8	5.5	10	M10x1.25	36	⅛	71	11	10	M8x1.0
40	~300	27	30	26	16	25	2	13	12	47	M6x1 Depth 12	6	14	M14x1.5	44	⅛	78	12	10	M10x1.25
50	~300	32	35	32	20	30	2	14	13	58	M8x1.25 Depth 16	7	18	M18x1.5	55	¼	90	13	12	M12x1.25
63	~300	32	35	38	20	32	2	14	13	72	M10x1.5 Depth 16	7	18	M18x1.5	69	¼	90	13	12	M14x1.5
80	~300	37	40	50	25	40	3	20	20	89	M10x1.5 Depth 22	11	22	M22x1.5	80	⅜	108	—	—	—
100	~300	37	40	60	30	50	3	20	20	110	M12x1.75 Depth 22	11	26	M26x1.5	100	½	108	—	—	—

*Trunnion mounting threads in flats NA are not available for Ø80, Ø100 bores

Bore (mm)	Without gaiter H	Z
20	35	106
25	40	111
32	40	113
40	50	130
50	58	150
63	58	150
80	71	182
100	71	182

Table A- 2. Values of Deformation Factor (C) (R.S. Khurmi, and J.K. Gupta, 2005).

Material		Involute	Values of deformation factor (C) in N-mm				
		Tooth	Tooth error in action (e) in mm				
Pinion	Gear	Form	0.01	0.02	0.04	0.06	0.08
Cast iron	Cast iron	$14^1/_2{}^\circ$	55	110	220	330	440
Steel	Cast iron		76	152	304	456	608
Steel	Steel		110	220	440	660	880
Cast iron	Cast iron	20° full depth	57	114	228	342	456
Steel	Cast iron		97	158	316	474	632
Steel	Steel		114	228	456	684	912
Cast iron	Cast iron	20° stub	59	118	236	354	472
Steel	Cast iron		81	162	324	486	648
Steel	Steel		119	238	476	714	952

Table A- 3. Values of Maximum Allowable Tooth Error in Action (E) Verses Pitch Line elocity, for Well-Cut Commercial Gears (R.S. Khurmi, and J.K. Gupta, 2005).

Pitch line velocity (v) m/s	Tooth error in action (e) mm	Pitch line velocity (v) m/s	Tooth error in action (e) mm	Pitch line velocity (v) m/s	Tooth error in action (e) mm
1.25	0.0925	8.75	0.0425	16.25	0.0200
2.5	0.0800	10	0.0375	17.5	0.0175
3.75	0.0700	11.25	0.0325	20	0.0150
5	0.0600	12.5	0.0300	22.5	0.0150
6.25	0.0525	13.75	0.0250	25 and over	0.0125
7.5	0.0475	15	0.0225		

Table A- 4. Values of Service Factor (R.S. Khurmi, and J.K. Gupta, 2005).

Type of load	Types of service		
	Intermittent or 3 hours per day	8-10 hours per day	Continuous 24 hours per day
Steady	0.8	1.00	1.25
Light shock	1.00	1.25	1.54
Medium shock	1.25	1.54	1.80
Heavy shock	1.54	1.80	2.00

Table A- 5.Values of Flexural Endurance Limit (R.S. Khurmi, and J.K. Gupta, 2005).

Material of pinion and gear	Brinell hardness number (B.H.N.)	Flexural endurance limit (σ_e) in MPa
Grey cast iron	160	84
Semi-steel	200	126
Phosphor bronze	100	168
Steel	150	252
	200	350
	240	420
	280	490
	300	525
	320	560
	350	595
	360	630
	400 and above	700

Table A- 6. Permissible Values of Coefficient of Fluctuation of Speed (R.S. Khurmi, and J.K. Gupta, 2005)

S. No.	Type of Machen or class of service	Coefficient of fluctuation of speed (C_S)
1	Crushing machines	0.200
2	Electrical machines	0.003
3	Electrical machines (direct drive)	0.002
4	Engines with belt transmission	0.030
5	Gear wheel transmission	0.020
6	Hammering machines	0.200
7	Pumping machines	0.03 to 0.05
8	Machine tools	0.030
9	Paper making, textile and weaving machines	0.025
10	Punching, shearing and power presses	0.10 to 0.15
11	Punching, shearing and power presses	0.10 to 0.020
12	Rolling mills and mining machines	0.025

Table A- 7. Values of Fluctuation of Energy (CE) (R.S. Khurmi, and J.K. Gupta, 2005).

No.	Type of engine	Coefficient of fluctuation of energy (C_E)
1.	Single cylinder, double acting steam engine	0.21
2.	Cross compound steam engine	0.096
3.	Single cylinder, Single acting four stroke gas engine	1.93
4.	Four-cylinder, Single acting four stroke gas engine	0.066
5.	Six-cylinder, Single acting four stroke gas engine	0.031

Table A- 8. Characteristics of Brakes and Clutches materials (Budynas-Nisbett 2008).

Material	Friction Coefficient f	Maximum Pressure p_{max}, kpa	Maximum Temperature Instantaneous, °F	Maximum Temperature Continuous, °F	Maximum Velocity V_{max}, ft/min
Cermet	0.32	150	1500	750	
Sintered metal (dry)	0.29–0.33	300–400	930–1020	570–660	3600
Sintered metal (wet)	0.06–0.08	500	930	570	3600
Rigid molded asbestos (dry)	0.35–0.41	100	660–750	350	3600
Rigid molded asbestos (wet)	0.06	300	660	350	3600
Rigid molded asbestos pads	0.31–0.49	828	930–1380	440–660	4800
Rigid molded nonasbestos	0.33–0.63	100–150		500–750	4800–7500
Semirigid molded asbestos	0.37–0.41	100	660	300	3600
Flexible molded asbestos	0.39–0.45	100	660–750	300–350	3600
Wound asbestos yarn and wire	0.38	100	660	300	3600
Woven asbestos yarn and wire	0.38	100	500	260	3600
Woven cotton	0.47	100	230	170	3600
Resilient paper (wet)	0.09–0.15	400	300		$PV < 500\,000$ psi · ft/min

Table A- 9. Formulas for SAE Spline (Robert L. Mott, et al. 2018).

		A: Permanent fit		B: To slide without load		C: To slide under load	
No. of Splines	W, for all fit	H	D	H	d	H	D
Four	0.241D	0.075D	0.850D	0.125D	0.750D		
Six	0.250D	0.050D	0.900D	0.075D	0.850D	0.100D	0.800D
Ten	0.156D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D
Sixteen	0.098D	0.045D	0.910D	0.070D	0.860D	0.095D	0.810D

Note: this formula gives the maximum dimensions for W, h, and d.

Table A- 10. Properties of aluminum alloy (Budynas-Nisbett 2008).

Aluminum Association Number	Temper	Strength Yield, S_y, MPa (kpsi)	Tensile, S_u, MPa (kpsi)	Fatigue, S_f, MPa (kpsi)	Elongation in 2 in, %	Brinell Hardness H_B
Wrought:						
2017	O	70 (10)	179 (26)	90 (13)	22	45
2024	O	76 (11)	186 (27)	90 (13)	22	47
	T3	345 (50)	482 (70)	138 (20)	16	120
3003	H12	117 (17)	131 (19)	55 (8)	20	35
	H16	165 (24)	179 (26)	65 (9.5)	14	47
3004	H34	186 (27)	234 (34)	103 (15)	12	63
	H38	234 (34)	276 (40)	110 (16)	6	77
5052	H32	186 (27)	234 (34)	117 (17)	18	62
	H36	234 (34)	269 (39)	124 (18)	10	74
Cast:						
319.0*	T6	165 (24)	248 (36)	69 (10)	2.0	80
333.0†	T5	172 (25)	234 (34)	83 (12)	1.0	100
	T6	207 (30)	289 (42)	103 (15)	1.5	105
335.0*	T6	172 (25)	241 (35)	62 (9)	3.0	80
	T7	248 (36)	262 (38)	62 (9)	0.5	85

*Sand casting.

†Permanent-mold casting.

Table A- 11. Standard module (Robert L. Mott, et al. 2018).

Module (mm)	Equivalent P_d	Closest standard P_d (teeth/in)
0.3	84.667	80
0.4	63.500	64
0.5	50.800	48
0.8	31.750	32
1	25.400	24
1.25	20.320	20
1.5	16.933	16
2	12.700	12
2.5	10.160	10
3	8.466	8
4	6.350	6
5	5.080	5
6	4.233	4
8	3.175	3
10	2.540	2.5
12	2.117	2
16	1.587	1.5
20	1.270	1.25
25	1.016	1

Table A- 12. Properties of metallic bearing materials (R.S. Khurmi, and J.K. Gupta, 2005).

Bearing material	*Fatigue strength*	*Comfor-mability*	*Embed-dability*	*Anti scoring*	*Corrosion resistance*	*Thermal conductivity*
Tin base babbit	Poor	Good	Excellent	Excellent	Excellent	Poor
Lead base babbit	Poor to fair	Good	Good	Good to excellent	Fair to good	Poor
Lead bronze	Fair	Poor	Poor	Poor	Good	Fair
Copper lead	Fair	Poor	Poor to fair	Poor to fair	Poor to fair	Fair to good
Aluminium	Good	Poor to fair	Poor	Good	Excellent	Fair
Silver	Excellent	Almost none	Poor	Poor	Excellent	Excellent
Silver lead deposited	Excellent	Excellent	Poor	Fair to good	Excellent	Excellent

Table A- 13. Anbesa bus specification (Bishoftu automotive industry official page).

Model	ZK6118HGA
L, W and H	11687mm, 2500mm, 3300mm (to the A/C)
Wheel base (mm)	5800
Min. Ground clearance(mm)	190
weight (kg)	12750
Gross vehicle weight (kg)	18000
Max. Speed (km/h)	100
Fuel Tank Capacity (L)	300
Engine	WEICHAI Engine WP 10.290
Max. power	290 HP@ 2300 RPM
Max torque	1160NM@1300-1600 RPM
Position of the Engine	Rear mounted
Type of Fuel and Tire size	Diesel and Size:295/80R22.5, 6+1 Tubeless tires
Gearbox	ZF Eco life Automatic gearbox
Steering system	BOSCH8098 integral power steering
Suspension system	Airbag suspension with ECAS
Power supplying system	Generator with 2 Batteries
Braking system	Dual-circuit air braking, WABCO ABS
Electric system	24V, single wire, minus
Lamps	Separate hand lamp, separate tail lamp, three cabin lamps on the roof
Power supplying system	Generator with 2 Batteries
Air-conditioning system	With top mounted large capacity, A/C system
Audio and Visual system	Mp3 player,8 speakers
Auxiliary equipment	With curtain; Steel handrail; 35 standing passenger rings, 2 roof hatches, 6 safety hammers
Body structure	Semi-integral body Pre-stress stretched panel skin Full metal body without fiber glass at both front and rear bumper &wing, Front and rear windscreens
Seat	44 (2+2 layout) passenger seats +1 driver seat

Table A- 14. 33 SI Alternator properties and output Characteristics Curve (Heavy duty brushless alternator service manual).

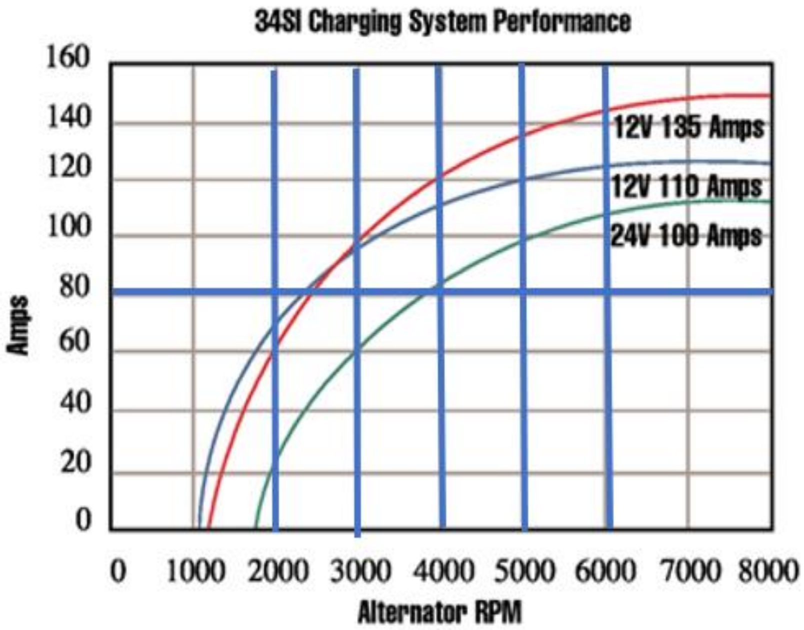
33 SI Alternator specification	
Nominal voltage	12 -28 v
Nominal current	100- 135 A
Maximum rotational speed	Continuous: 10,000 rpm Intermittent: 12,000 rpm
Corrosion coating	Special Environmental Protection Coating
Rotation	Clockwise Counterclockwise
Weight	11Kg
Application	Line-Haul Diesel Trucks Large Commercial Diesel Engines Harsh Environments Heavy Belt Loads and Vibrations
System performance curve	

Table A- 15. Absolute viscosity of commonly used lubricating oils of bearings

S. No.	Type of oil	Absolute viscosity in kg / m-s at temperature in °C											
		30	35	40	45	50	55	60	65	70	75	80	90
1.	SAE 10	0.05	0.036	0.027	0.0245	0.021	0.017	0.014	0.012	0.011	0.009	0.008	0.005
2.	SAE 20	0.069	0.055	0.042	0.034	0.027	0.023	0.020	0.017	0.014	0.011	0.010	0.0075
3.	SAE 30	0.13	0.10	0.078	0.057	0.048	0.040	0.034	0.027	0.022	0.019	0.016	0.010
4.	SAE 40	0.21	0.17	0.12	0.096	0.78	0.06	0.046	0.04	0.034	0.027	0.022	0.013
5.	SAE 50	0.30	0.25	0.20	0.17	0.12	0.09	0.076	0.06	0.05	0.038	0.034	0.020
6.	SAE 60	0.45	0.32	0.27	0.20	0.16	0.12	0.09	0.072	0.057	0.046	0.040	0.025
7.	SAE 70	1.0	0.69	0.45	0.31	0.21	0.165	0.12	0.087	0.067	0.052	0.043	0.033

Table A- 16. Principal dimensions for ball bearings (R.S. Khurmi, and J.K. Gupta, 2005).

Bearing No.	*Bore (mm)*	*Outside diameter*	*Width (mm)*
200	10	30	9
300		35	11
201	12	32	10
301		37	12
202	15	35	11
302		42	13
203	17	40	12
303		47	14
403		62	17
204	20	47	14
304		52	14
404		72	19
205	25	52	15
305		62	17
405		80	21
206	30	62	16
306		72	19
406		90	23
207	35	72	17
307		80	21
407		100	25

Table A- 17Metric Mechanical-Property Classes for Steel Bolts (Budynas-Nisbett, 2008).

Property Class	Size Range, Inclusive	Minimum Proof Strength,† MPa	Minimum Tensile Strength,† MPa	Minimum Yield Strength,† MPa	Material	Head Marking
4.6	M5–M36	225	400	240	Low or medium carbon	4.6
4.8	M1.6–M16	310	420	340	Low or medium carbon	4.8
5.8	M5–M24	380	520	420	Low or medium carbon	5.8
8.8	M16–M36	600	830	660	Medium carbon, Q&T	8.8
9.8	M1.6–M16	650	900	720	Medium carbon, Q&T	9.8
10.9	M5–M36	830	1040	940	Low-carbon martensite, Q&T	10.9
12.9	M1.6–M36	970	1220	1100	Alloy, Q&T	12.9

*The thread length for bolts and cap screws is

$$l_T = \begin{cases} 2d + 6 & l \le 125 \\ 2d + 12 & 125 < l \le 200 \\ 2d + 25 & l > 200 \end{cases}$$

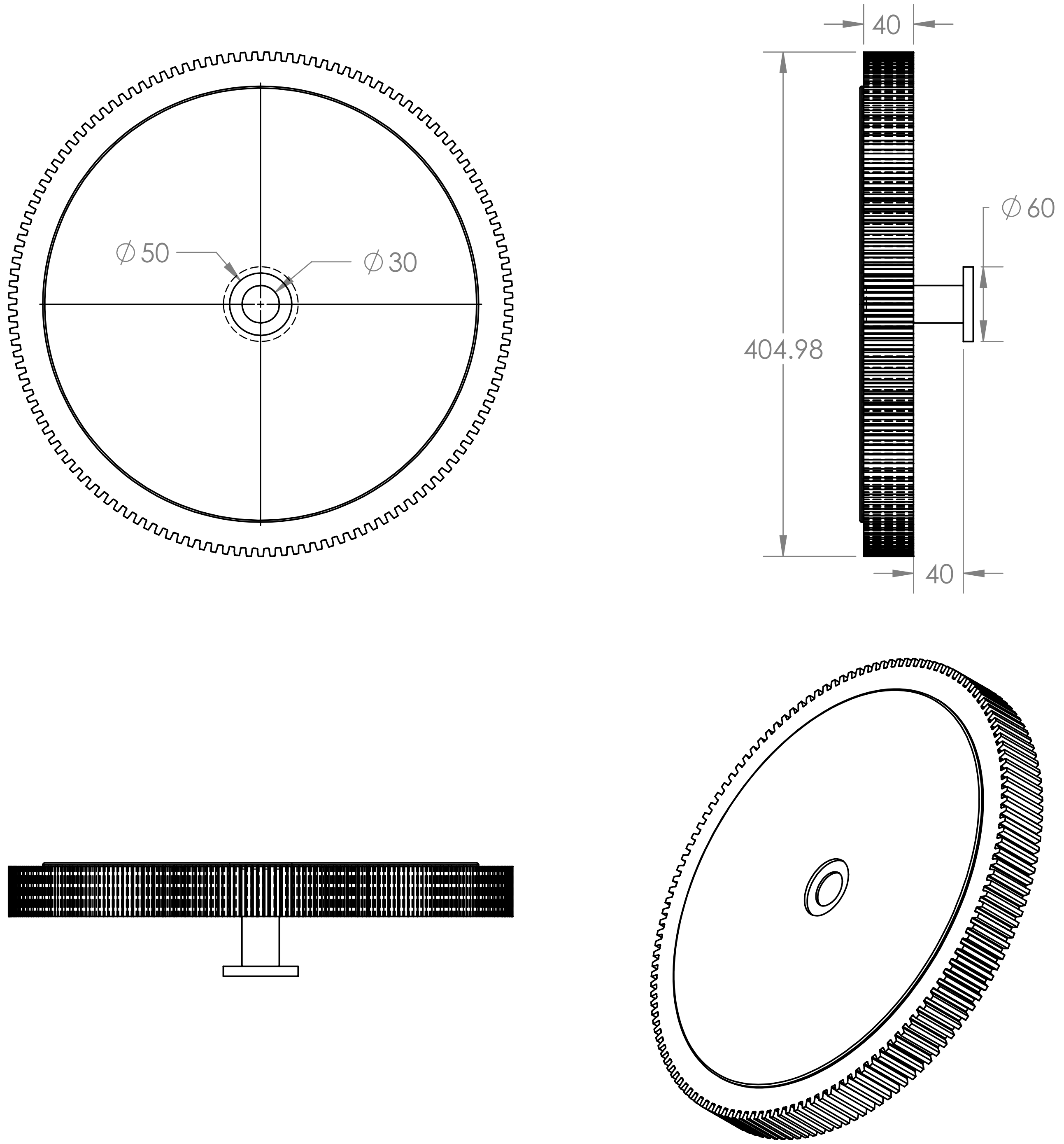
where l is the bolt length. The thread length for structural bolts is slightly shorter than given above.

†Minimum strengths are strength exceeded by 99 percent of fasteners.

Table A- 18. Some power on vehicle electrical components (some prolonging load and some intermittent loads) (Tom denton, 2004)

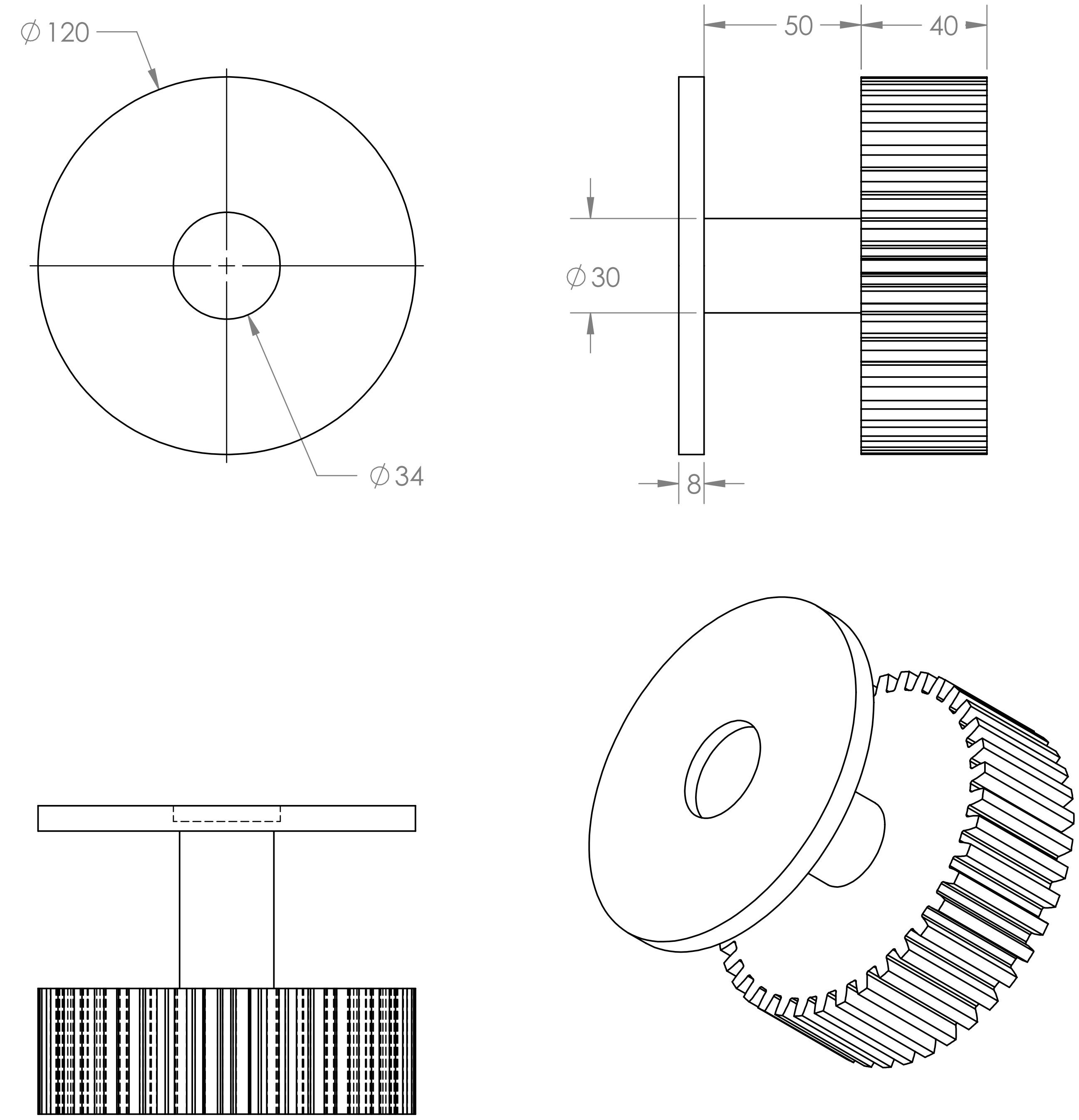
Appliance	Running wattage (W)	On (Hr/day)	Wattage (Wh/day)
Prolonging loads:			
✓ Dashboard lights	25		
✓ Side and	30	9	
Tail lights			2205
✓ Number plate lights	10		
✓ Head light	180		
Total	245		
Intermittent loads			
✓ Interior light	10		
✓ Heater	50		
✓ Rear wiper	50		
✓ Front wiper	80		
✓ Indicator	50		
✓ Auxiliary lamps	110	9	459
✓ Reverse light	40		
✓ Rear fog light	40		
✓ Horn	40		
✓ Brake light	40		
Total	$510 * 0.1 = 51$		
Total wattage	296		2664

Appendix –B Direct gear engagement to the flywheel components and assembly drawings



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

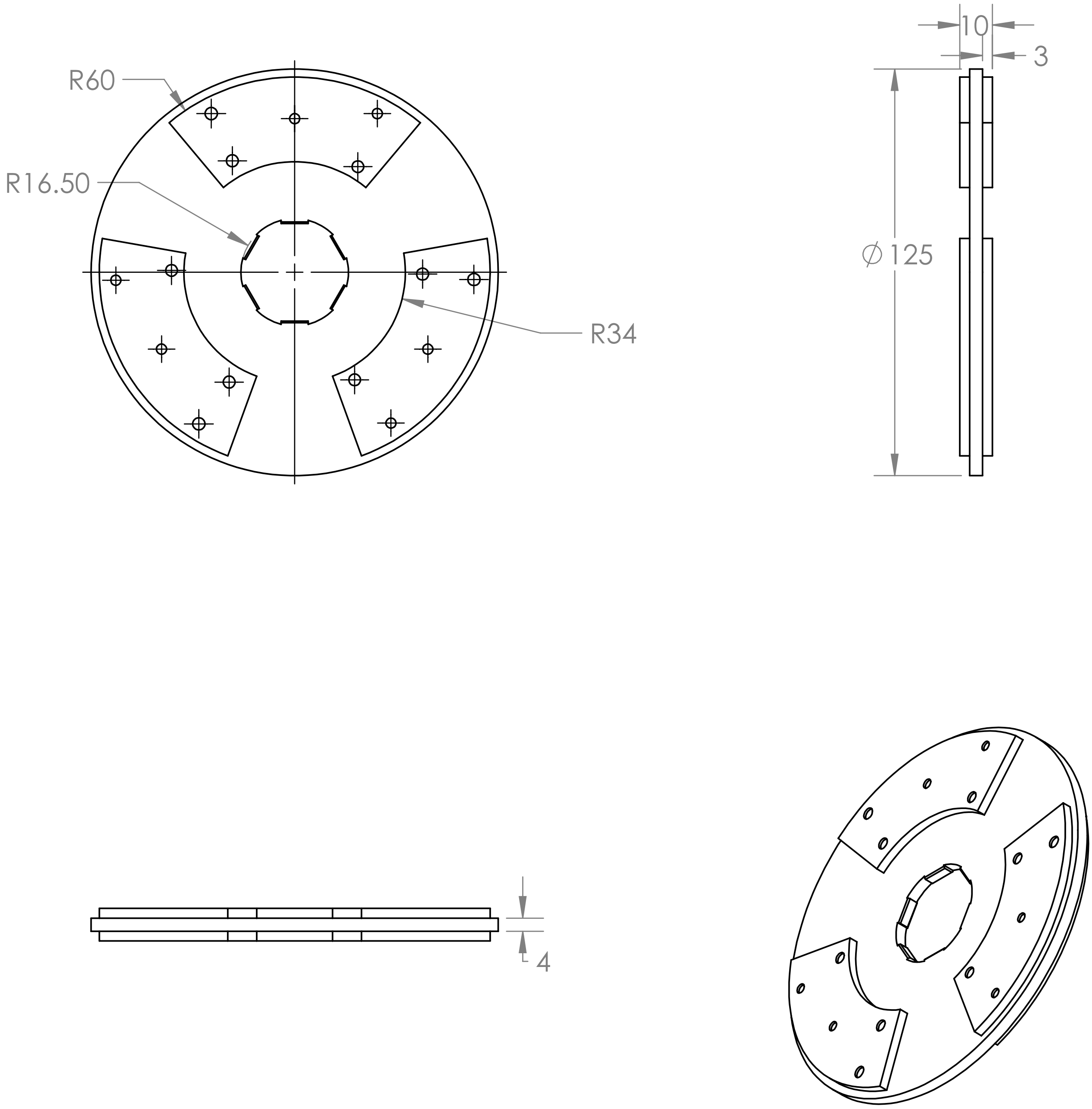
	NAME	SIGNATURE	DATE		
DRAWN	Daniel		6/28/21	TITLE:	ENGINE FLYWHEEL
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL: Grey cast iron	DWG NO. 01 — A4
				WEIGHT:	SCALE:1:5 — SHEET 1 OF 1



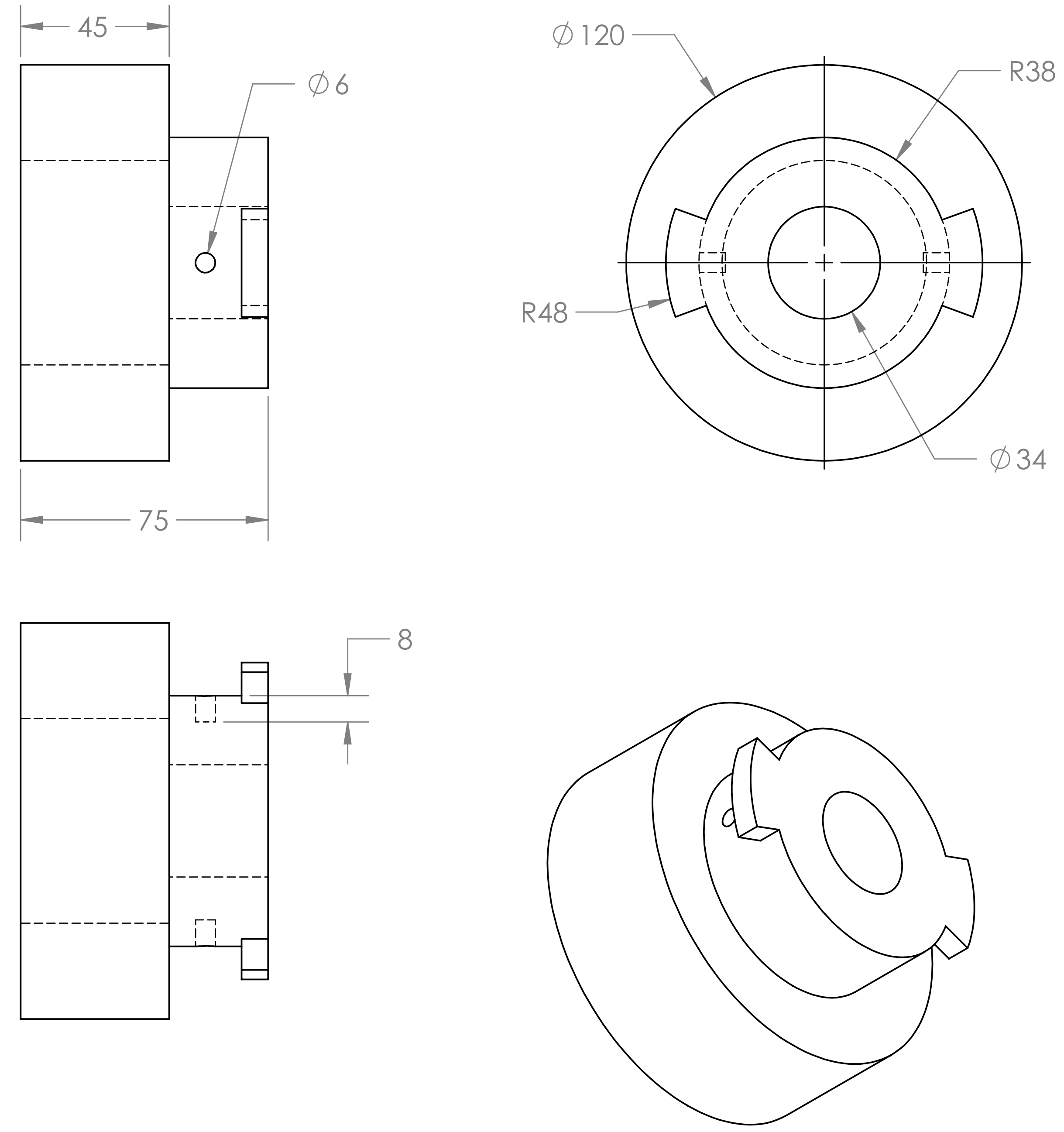
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
			REVISION

	NAME	SIGNATURE	DATE		
DRAWN	Daniel		6/28/21		
CHK'D					
APPV'D					
MFG					
Q.A	1				

TITLE:	PINION GEAR		
MATERIAL: Grey cast iron	DWG NO. 02	A4	
WEIGHT:	SCALE:1:4	SHEET 1 OF 1	



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
	NAME	SIGNATURE	DATE		TITLE:		
DRAWN	Daniel		6/28/21		CLUTCH PLATE		
CHK'D							
APPV'D							
MFG							
Q.A	1			MATERIAL: Rigid molded Asbestos pad	DWG NO. 03		A4
				WEIGHT:	SCALE:1:2	SHEET 1 OF 1	



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A	1		

TITLE:

CLUTCH HUB

MATERIAL:
Grey cast iron

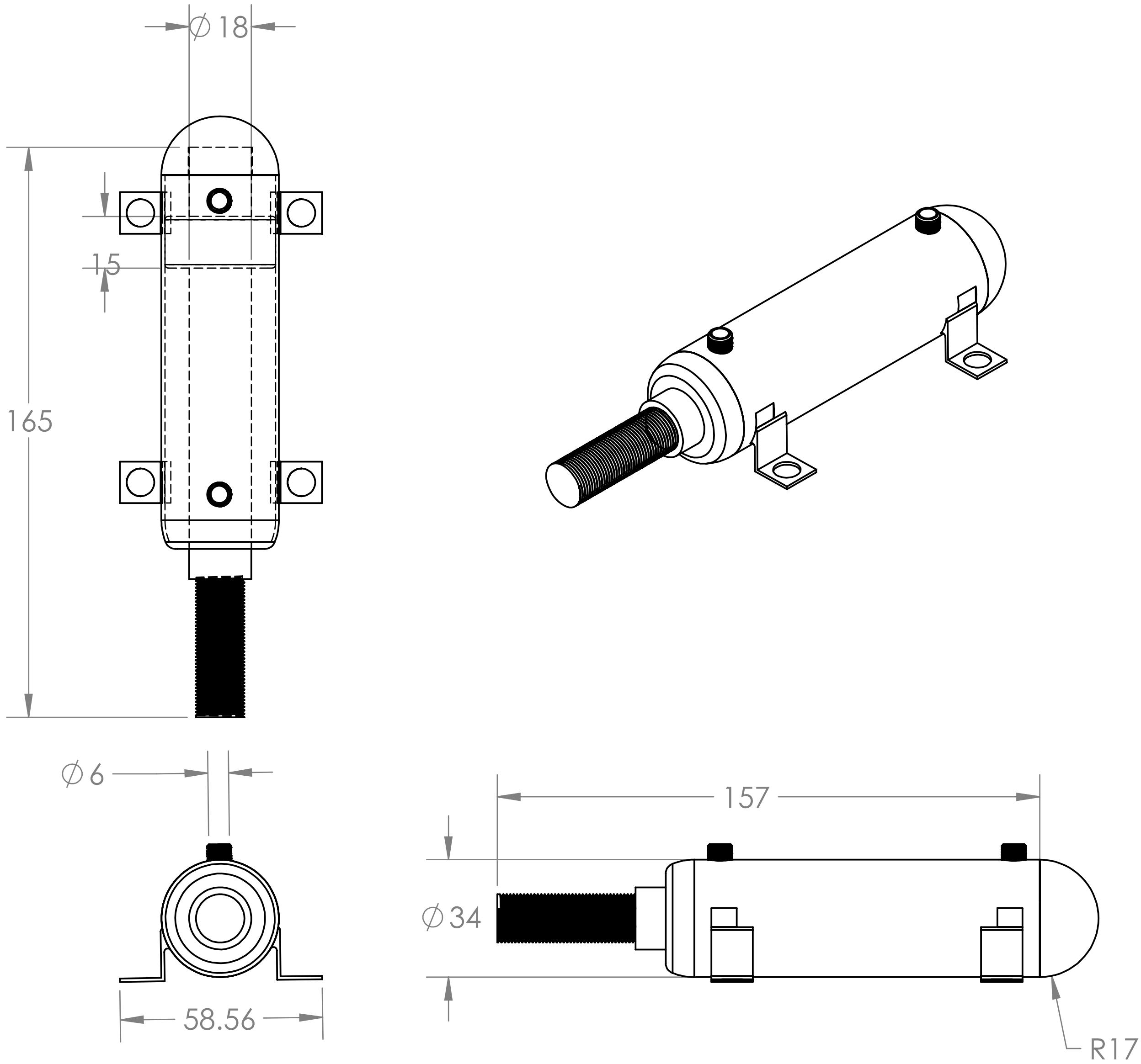
DWG NO.
04

A4

WEIGHT:

SCALE:1:2

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

SINGLE ACTING PNEUMATIC ACTUATOR

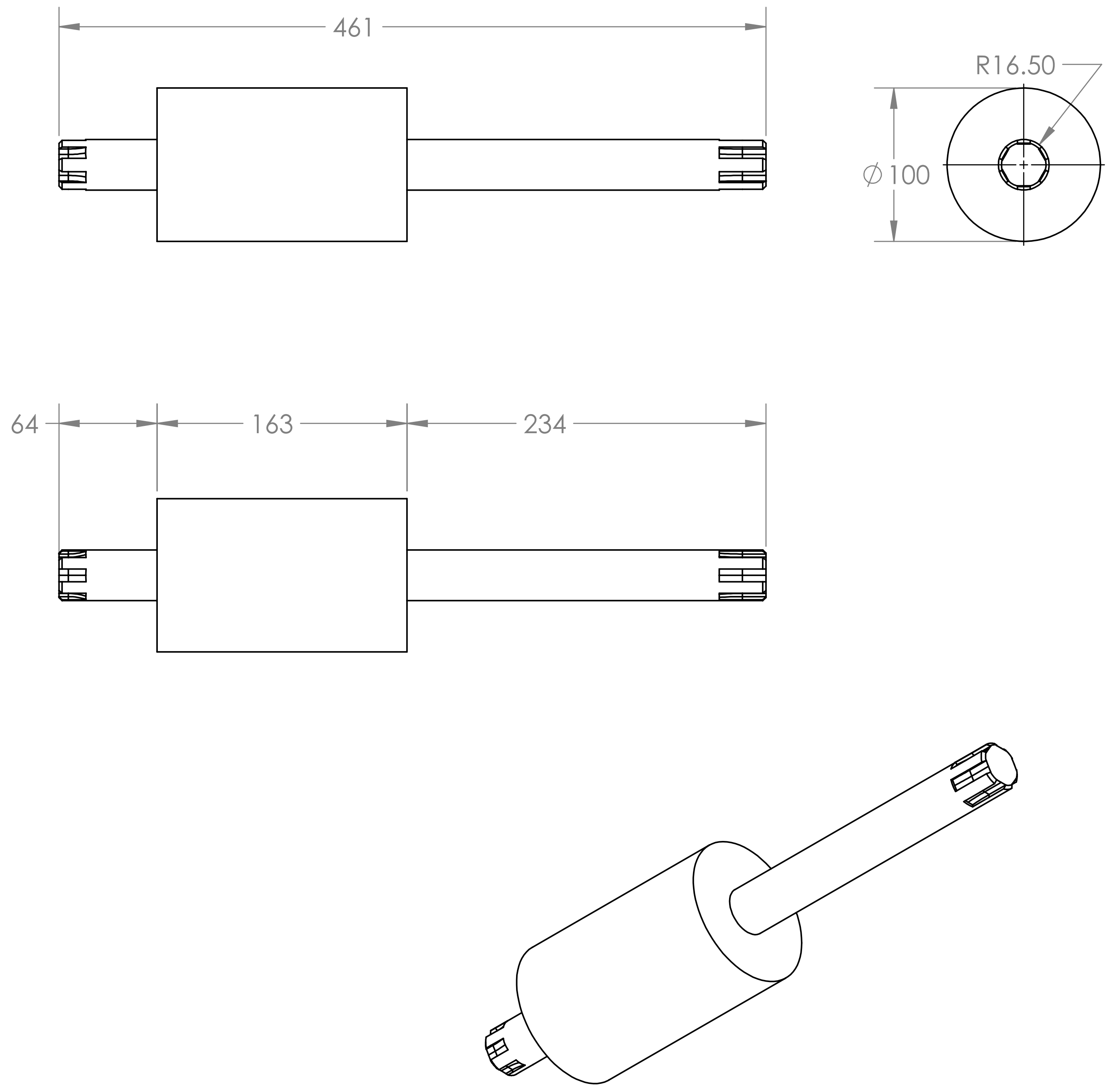
MATERIAL:
Rod cover:Aluminum alloy
Body:Aluminum alloy
Piston rod:Carbon steel

DWG NO.
05

A4

SCALE:1: 2

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A	1		

TITLE:

SPLINED SHAFT WITH FLYWHEEL

MATERIAL:
Splined shaft: Aluminum alloy
Flywheel: High strength low-alloy steel

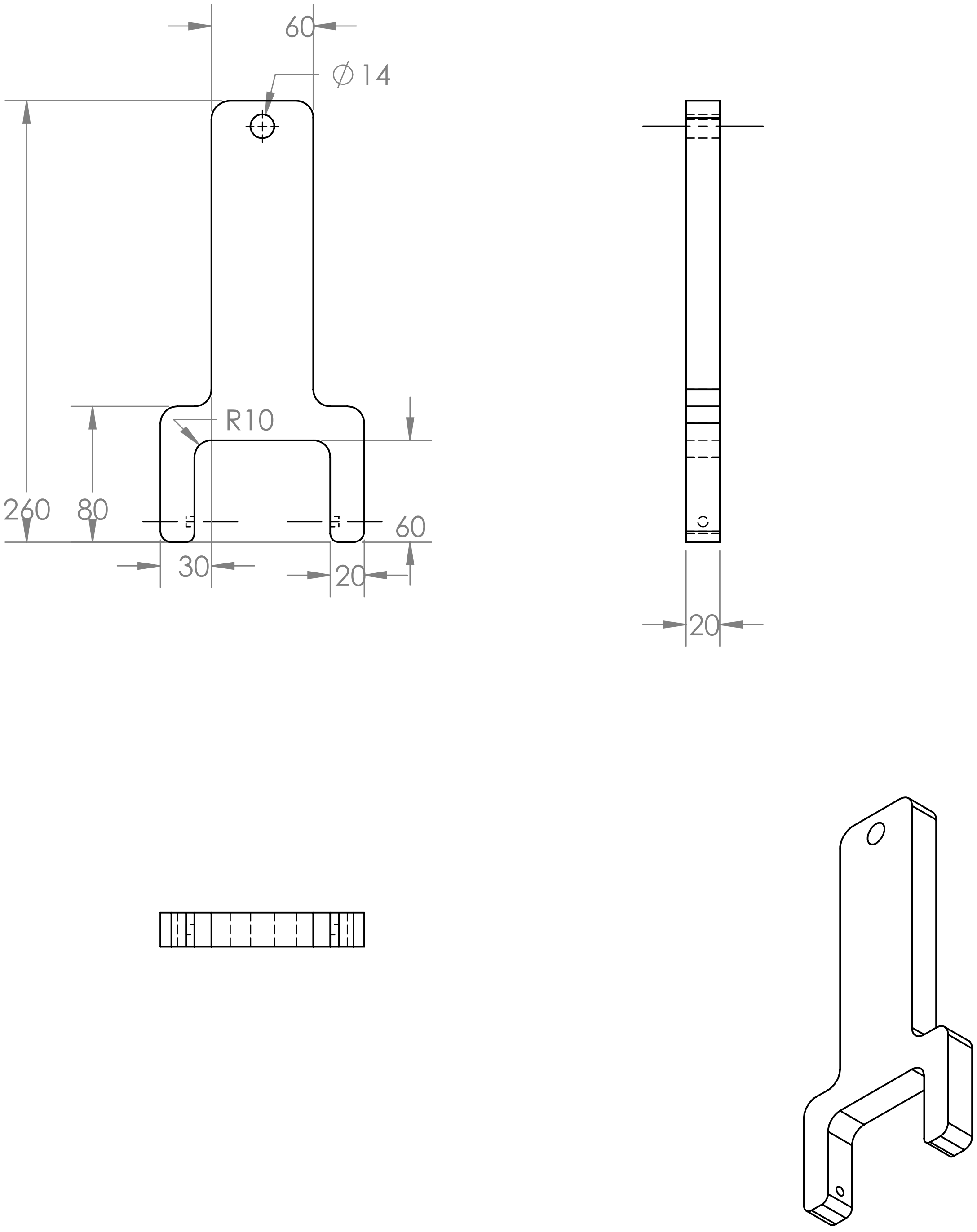
WEIGHT:

DWG NO. 06

A4

SCALE:1:2

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A	1		

MATERIAL: Aluminum alloy

WEIGHT:

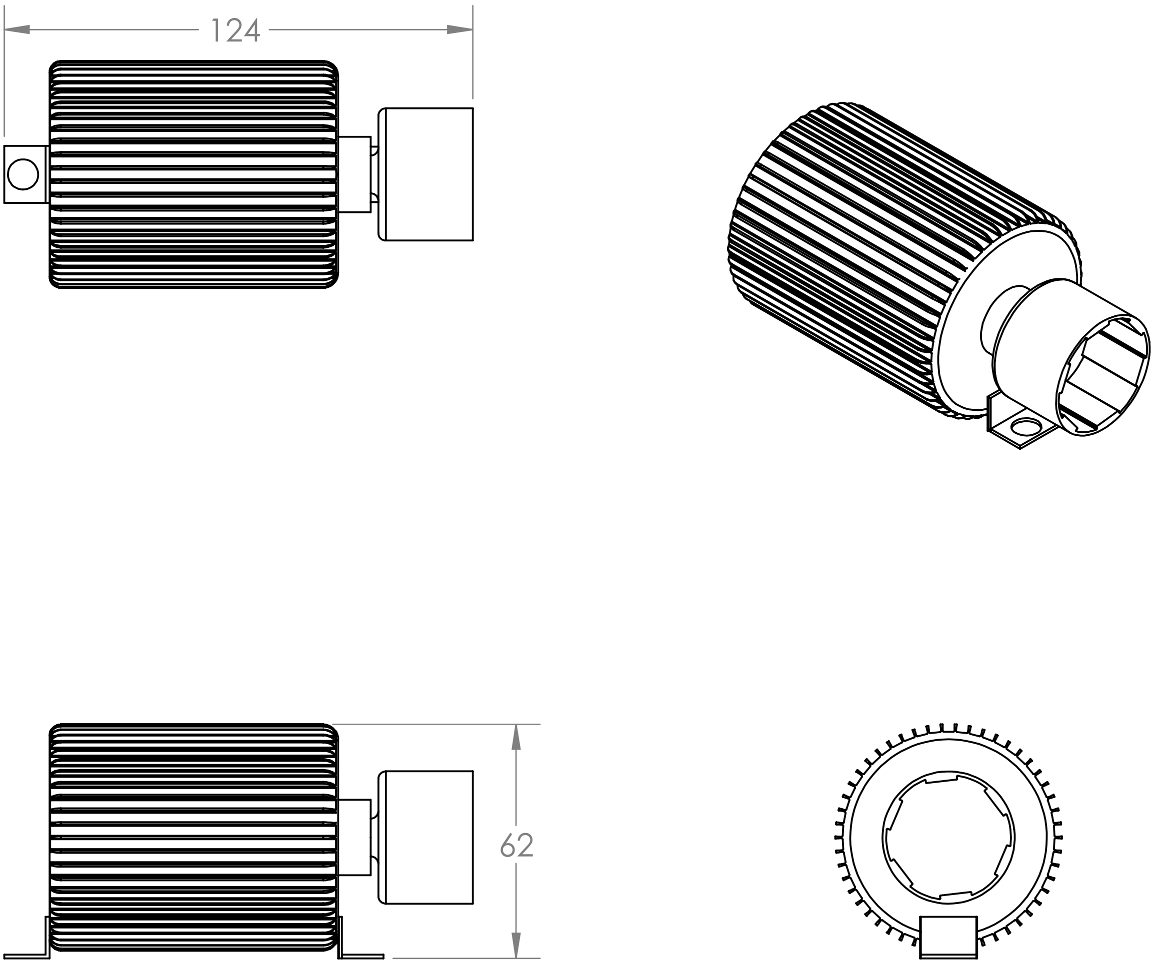
TITLE: FORK LEVER

DWG NO. 07

A4

SCALE:1:4

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A	1		

MATERIAL:

WEIGHT:

TITLE:

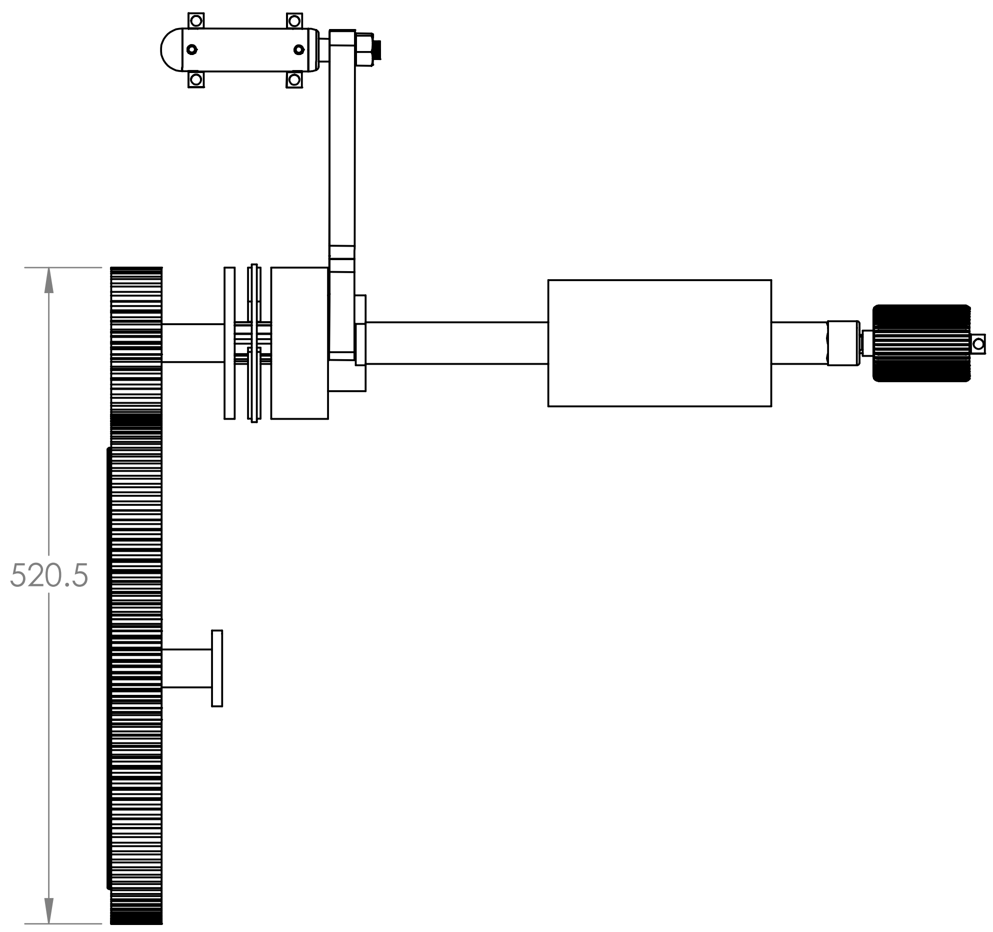
33- SI HEAVY DUTY BRUSHLESS ALTERNATOR

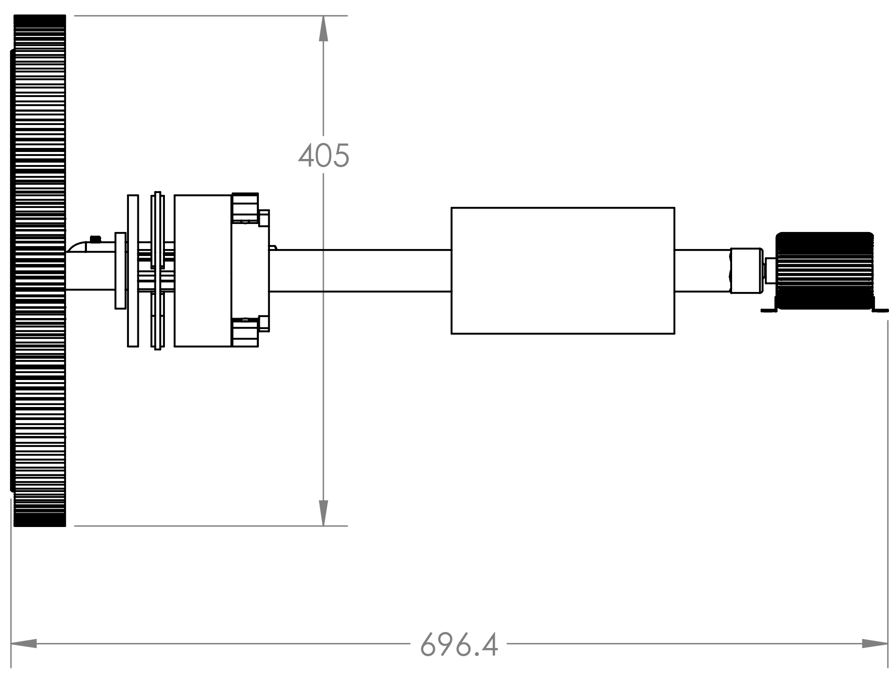
DWG NO. 08

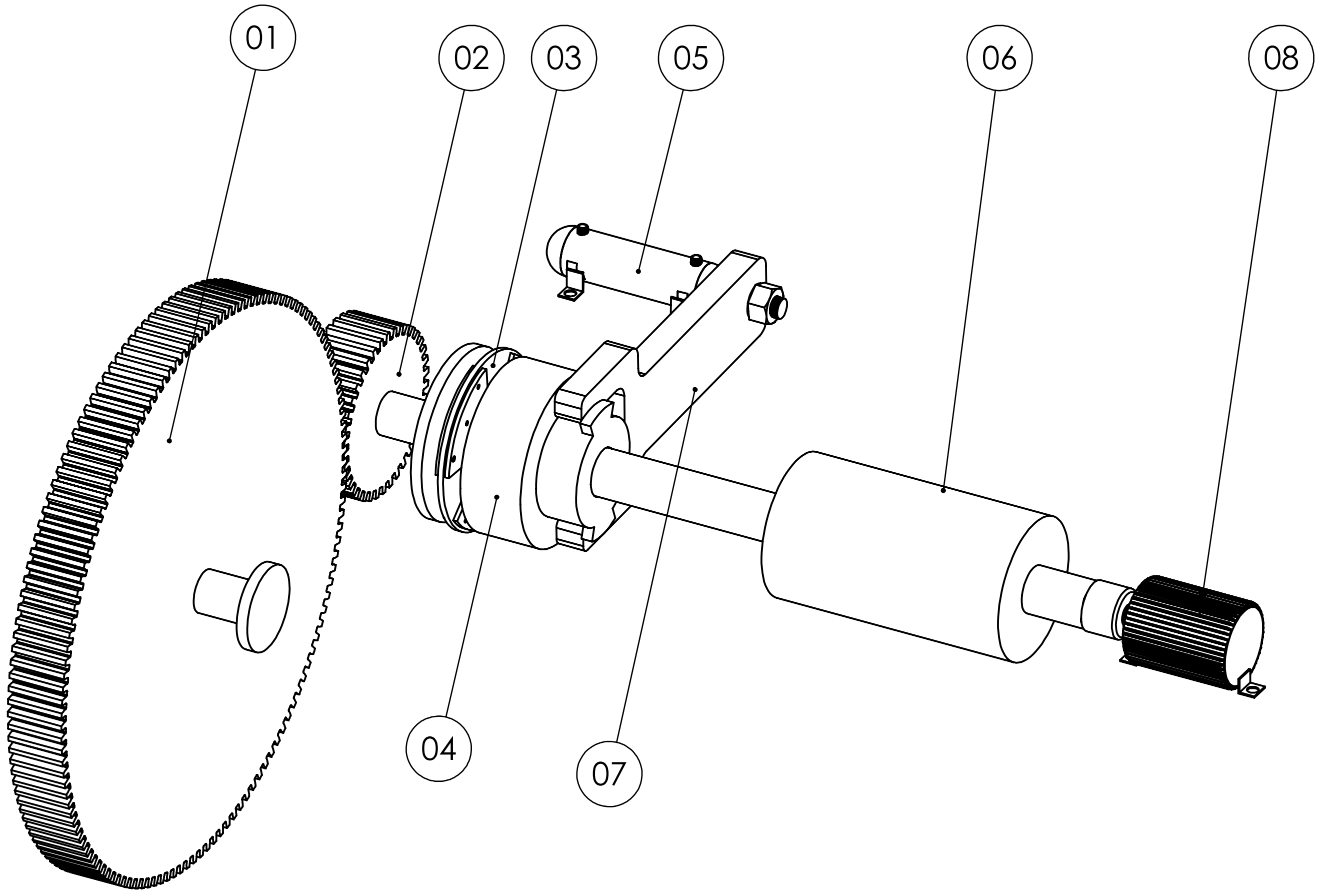
A4

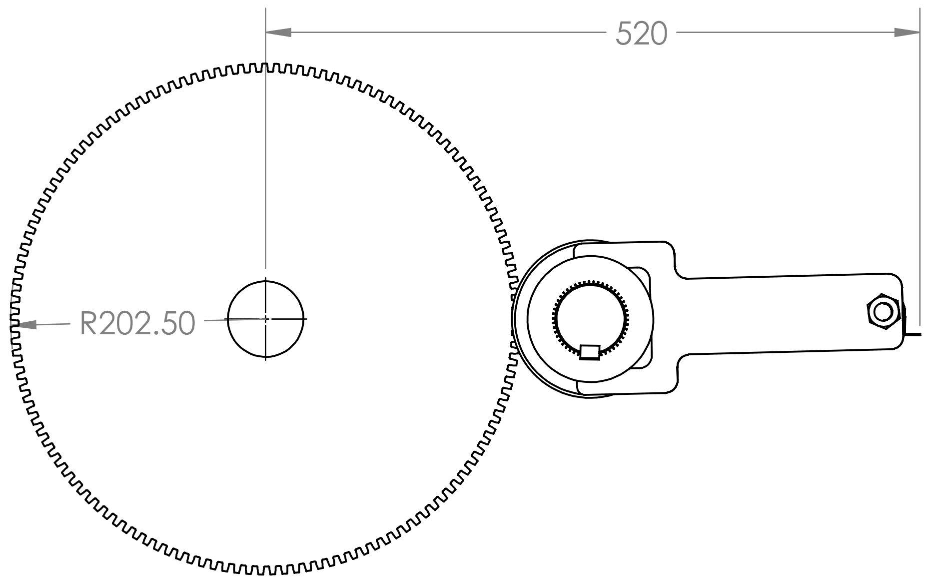
SCALE:1:2

SHEET 1 OF 1









No	Part names
01	Engine flywheel
02	Pinion gear with frictional plat
03	Single plate clutch disc
04	Clutch hub
05	Double acting pneumatic cylinder
06	Splined shaft with solid disc flywheel
07	Fork lever
08	Alternator

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE
DRAWN	Daniel		6/28/21
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

DIRECT GEAR ENGEGIMENT TO THE FLYWHEEL RBS ASSEMBLED DRAWING

MATERIAL:

DWG NO.

A3

WEIGHT:

SCALE:1:6

SHEET 1 OF 1

APPROVAL SHEET

We, the advisors of the thesis entitled "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" and developed by Daniel Tean Ebrahim, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

Major Advisor	Signature	Date
Co-advisor	Signature	Date

We, the undersigned, members of the Board of Examiners of the thesis by Daniel Tena Ebrahim have read and evaluated the thesis entitled "DESIGN AND ANALYSIS OF A REGENERATIVE BRAKING SYSTEM FOR CITY BUS (ANBESA BUS)" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive engineering.

Chairperson	Signature	Date
Internal Examiner	Signature	Date
Dr. Solomon Seid (Ph.D.)		**26 – Nov - 21**
External Examiner	Signature	Date

Finally, approval and acceptance of the thesis is contingent upon submission of its final copy to the Office of Postgraduate Studies (OPGS) through the Department Graduate Council (DGC) and School Graduate Committee (SGC).

Head of Department	Signature	Date
School Dean	Signature	Date
Office of postgraduate studies dean	Signature	Date