ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

School of Mechanical, Chemical and Materials Engineering

MSc project

On

Construction of a Mechanism for Harvesting Energy from Suspension System

A project submitted in partial fulfillment of the requirements for the Award of the degree of Master of Science in Automotive Technology

By

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Advisor: N. Ramesh Babu (Associate Professor)

MECHANICAL SYSTEM AND VEHICLE ENGINEERING PROGRAM

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Adama, Ethiopia

CANDIDATE'S DECLARATION

We hereby declare that the work which is being presented in the project entitled **"Construction of a mechanism for harvesting energy from suspension system"** in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Technology is an authentic record of our own work carried out from March,2016 to June,2016 under the supervision of N.Ramesh Babu, Associate Professor, Mechanical systems and Vehicle Engineering program, Adama Science and Technology University, Ethiopia.

The matter embodied in this project has not been submitted by us for the award of any other degree or diploma. All relevant resources of information used in this project have been duly acknowledged.

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This is to certify that the above statement made by the candidates is correct to the best of my Knowledge and belief. This project has been submitted for examination with my approval.

Name	Signature	Date
______________________	______________	____________

Advisor

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LIST OF ACRONYMS AND ABBREVIATIONS

σ_a	Allowable Stress
3D	Three Dimensions
AC	Alternate current
ASTU	Adama science and technology university
CAD	Computer-Aided Design
CAE	Computer-aided engineering
CAM	Computer-Aided Manufacturing
CATIA	Computer Aided Three-Dimensional Interactive Application
CAX	Computer Access Exercise
C_v	Velocity factor
DC	Direct Current
D_G	Pitch Diameter of the Gear
D_P	Pitch diameter of the pinion
ECAS	Electronically Controlled Active Suspension System
EMS	Electromagnetic Suspension
Hz	Hertz
K	Spring Rate
LMES	Linear Motion Electromagnetic System
MBSE	Model Based Systems Engineering
MMR	Mechanical Motion Rectifier
MOE	Ministry Of Education
NC	Numerical Control
N_P	Pinion Speed
N_{Sb}	Speed small bevel gear

OP	Cone distance
PGSA	Power-Generating Shock Absorber
PRV	Pressure Responsive Valves
R_m	Mean Radius
RP	Rapid Prototyping
T_b	Bevel Gear Torque
T_E	Number of Teeth
TVET	Technical and Vocational Education Training
UV	Ultra Violet
V.R	Velocity Ratio
W_N	Normal Force
W_R	Radial Force
W_T	Tangential force
X	Deflection of spring
θ_{P1}	Pitch Angle for the Pinion
θ_S	Angle between the two shaft axes
σ_o	Allowable static stress,
φ	Pressure angle

ABSTRACT

Shock absorbers are used to dampen the vibrations of the vehicle to maintain ride comfort. The energy associated with vibration of the vehicle dissipates in the form of heat from the shock absorbers. Tapping this energy which otherwise going as a waste helps in maintaining the fuel consumption and associated exhaust emissions. In this project it is aimed to design and construct a simple mechanisms consisting of rack and pinion which convert oscillatory motion of the suspension into unidirectional rotary motion, by connecting which to the generator, electricity can be produced .The energy can be used for charging the battery. All the components like rack, pinion, bevel gears, shaft etc. with proportionate dimension to fit in the shock absorber of a Sino truck HOWO6X4 336HP Dump Truck Zz3257n3847A,are designed and a 3D model was prepared using the CATIA. A prototype was prepared by assembling the components, which were manufactured. The functionality of regenerative shock absorber was successfully demonstrated.

Keywords: Regenerative shock absorber, rack and pinion, roller clutch, suspension travel

1. INTRODUCTION

Fossil fuels are being consumed with very fast rate. In addition, the cost of fuel is increasing with a very fast rate. An engineer has to work on saving of the fuel consumption. The aim of the project is to demonstrate how the kinetic energy from the suspension of a car can be utilized for producing electricity that would otherwise going waste. It is proposed to design and construct a mechanism to convert the energy of the shock absorber into electricity. The electricity generated can be used to recharge the car battery for further use for functioning of the car. There is a wide scope for regeneration of energy from shock absorber like regenerative of braking system. Shock Absorbers exhibit reciprocating motion. Although the reciprocating distance is very low the suspending mass is very high i.e. the mass of total vehicle. When vehicle is on a normal road then also shock absorbers are working due to uneven roads, sudden braking or sudden acceleration. So, this reciprocating motion of shock absorbers can be converted into rotary motion and through small gearbox attached to alternator of automobile, electricity can generated when shock absorbers are reciprocating.

1.1 Background of the study

Automotive industry is growing at a higher rate in Ethiopia. It is believed that the automobile industry has plays a crucial role in the economic development of country a like Ethiopia. Now most of the automotive industries existing in Ethiopia are assembly units. These units are assembling various types of vehicles by importing the parts. Industry has indigenous knowledge and facilities to design and manufacture efficient vehicles. There is a need to conduct a necessary research in modifying the design of vehicles for producing energy efficient vehicles. Regenerative shock absorbing system is the one, which helps in producing energy efficient vehicles.

1.2 Statement of the problem

The future of green manufacturing technology is foreseeable, especially on vehicle industry. Since the suspension is an important source of energy dissipation, it is feasible to harvest its vibration energy and convert into regenerative energy to improve the vehicle fuel efficiency. Instead of dissipating the vibration energy into waste heat, the damper in regenerative suspension will transform the kinetic energy into electricity or other potential energy and store it for later use. The stored energy can be used to tune the damping force of the damper to improve the suspension performance or to power vehicle electronics to increase vehicle fuel efficiency. In all vehicles, cars, truck, bus, train car and military vehicles a high amount of energy is being wasted during the driving condition on the road due to the vibration of the suspension system. This energy is dissipated in the form of heat to the surroundings. Therefore a good amount of energy can be used for producing electricity from this energy which otherwise be wasted, by using appropriate mechanism.

1.3 Objectives

1.3.1 General objective

The general objective of this project is to construct a mechanism for harvesting energy from suspension system of vehicles.

1.3.2 Specific objectives

- ✓ To study the vibration behavior of suspension system of vehicles.
- ✓ To design the mechanism for tapping this energy from suspension system of vehicle.
- ✓ To construct the prototype of the mechanism for producing electricity from suspension system.
- ✓ To demonstrate the functionality of the constructed mechanism.

1.4 Scope of the study

The scope of this project work has delimited to design and construction of a mechanism for harvesting energy from suspension systems of heavy vehicles.

1.5 Significance of the study

Upon successful implementation, the project will have the following significance:

- ✓ Improves fuel economy.
- ✓ Reduces the environmental pollution.
- ✓ Helps in improving ride comfort.
- ✓ Helps in reduction of oil importation expenses.
- ✓ Works as a reference for future researchers.

- ✓ Vehicle owners, Automotive industries, the society and the country as a whole get benefitted.

1.6 Limitations of the project

To show the exact regenerative shock absorber retrofit with rack and pinion mechanism in practical as designed in this project the following limitations are faced during the work.

1. Unavailability of different component parts:

Due to unavailability of component parts in the market as specification prepared like roller clutch, which are very essential, this project work is limited to show the mechanical motion rectifier components by available material.

2. Lack of equipment

Unavailability of regenerative shock absorber testing machine specially designed for mechanical motion rectifier.

3. Time constraint

The time given to complete the project was insufficient specially to perform the practical works as expected. Even though, different challenges are faced, only tried to show the full function of MMR with the prototype constructed.

2. LITERATURE REVIEW

This project started with a review of existing literature on the topic of construction of a mechanism for harvesting energy from suspension system and on regenerative shock absorber. Literature views is very much necessary a very intensive search was made step before starting project which helps to know the present status of the technology in selected area to find journal articles, thesis papers written on the subject and any literature documenting the step by step working process. It is found that very few journal papers available covering the topic of energy generation from suspension system with regard to regenerative shock absorber by using rack and pinion mechanism. Functional and design details of suspension systems and others mechanisms used for energy harvesting are covered in this chapter.

2.1 Suspension system

The function of vehicle suspension system is to support the weight of the vehicle body, to isolate the vehicle chassis form road disturbances, to enable the wheels to hold the road surface. Two main elements in suspension systems are spring and damper. The damper is designed to dissipate vibration energy into the heat to reduce the vibration, which is transmitted from road excitation. However, the dissipated heat is from fuel or electrical power. In hybrid vehicle recapture some of the energy usually lost in braking system but the dissipation of vibration energy by shock absorbers in the vehicle suspension remains unused. In a vehicle, it reduces the effect of traveling over rough ground, leading to improved ride quality, and increase in comfort due to substantially reduced amplitude of disturbances. Without shock absorbers, the vehicle would have a bouncing ride, as energy is stored in the spring and then released to the vehicle, possibly exceeding the allowed range of suspension movement. Control of excessive suspension movement without shock absorption requires stiffer (higher rate) springs, which would in turn give a harsh ride. Shock absorbers allow the use of soft (lower rate) springs while controlling the rate of suspension movement in response to bumps. They also, along with hysteresis in the tire itself, damp the motion of the unspring weight up and down on the springiness of the tire. Since the tire is not as soft as the springs, effective wheel bounce damping may require stiffer shocks than would be ideal for the vehicle motion alone. Spring-based shock absorbers commonly use coil springs or leaf springs, though torsion bars can be used in tensional shocks as well. Ideal springs alone, however, are not shock absorbers as springs only store and do not dissipate or absorb energy. Vehicles typically employ springs and torsion bars as well as hydraulic shock absorbers. In this combination, "shock absorber" is reserved specifically for the hydraulic piston that absorbs and dissipates vibration.

2.1.1 Shock absorber

A suspension system or shock absorber is a mechanical device designed to smooth out or damp shock impulse, and dissipate kinetic energy. The shock absorbers duty is to absorb or dissipate energy. In a vehicle, it reduces the effect of traveling over rough ground, leading to improved ride quality, and increase in comfort due to substantially reduced amplitude of disturbances. When a vehicle is traveling on a level road and the wheels strike a bump, the spring is compressed quickly. The compressed spring will attempt to return to its normal loaded length and, in so doing, will rebound past its normal height, causing the body to be lifted. The weight of the vehicle will then push the spring down below its normal loaded height. This, in turn, causes the spring to rebound again. This bouncing process is repeated over and over, a little less each time, until the up-and-down movement finally stops.

A shock absorber is a mechanical device designed to smooth out or damp shock impulse, and convert kinetic energy to another form of energy (usually thermal energy, which can be easily dissipated). It is a type of dashpot. A shock absorber is a device, which converts mechanical energy into thermal energy. The energy transformation occurs as the shock absorbers fluid medium is forced through orifice at high velocity.

Pneumatic and hydraulic shock absorbers are used in conjunction with cushions and springs. An automobile shock absorber contains spring-loaded check valves and orifices to control the flow of oil through an internal piston. One design consideration, when designing or choosing a shock absorber, is where that energy will go. In most shock absorbers, energy is converted to heat inside the viscous fluid. In hydraulic cylinders, the hydraulic fluid heats up, while in air cylinders, the hot air is usually exhausted to the atmosphere. In other types of shock absorbers, such an electromagnetic types, the dissipated energy can be stored and used later. In general terms, shock absorbers help cushion vehicles on uneven roads.

2.1.2 Shock Absorbers Function

A shock absorber in common phrasing is a mechanical device designed to smooth out or damp Sudden shock impulse and dissipate kinetic energy. Shock absorbers are an important part of automobile and motorcycle suspensions, aircraft landing gear, and the supports for many industrial machines. Large shock absorbers have also been used in structural engineering to reduce the susceptibility of structures to earthquake damage and resonance. [1]

In a fully active suspension, there are no passive elements, such as dampers and springs. An actuator of variable length regulates the interaction between vehicle body and wheel. The actuator is usually hydraulically controlled and applies between body and wheel a force that represents the control action generally determined with an optimization procedure. Active suspensions have better performance than passive suspensions with regard to comfort, road holding, and ride ability. However, active suspension systems are rather complex, since they require several components such as actuators, servo valves, high-pressure tanks for the control fluid, either sensors for detecting the system state or appropriate system state observers, etc.

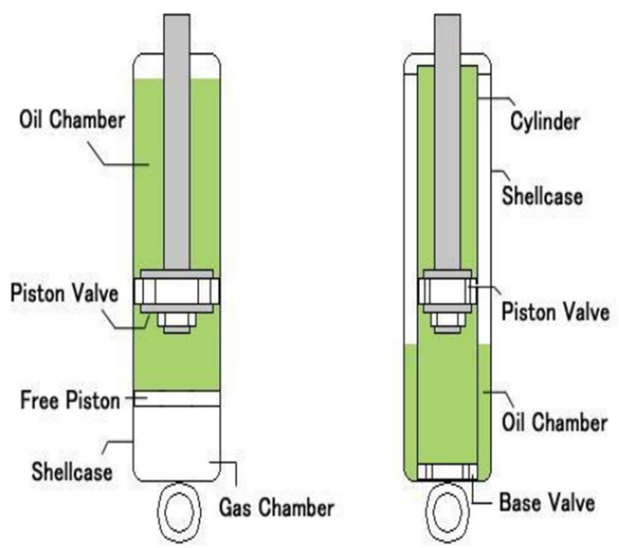


Figure 2.1: Mono Tube and Twin Tube Shock Absorber

2.2 Vibration Theory

The various masses, springs and dampers of the complete vehicle combine to make a complex vibrating system, stimulated by road roughness and control inputs. A full understanding of the role of the damper in vehicle dynamics really requires a thorough understanding of both ride and handling. However, a basic appreciation requires only an understanding of simple one and two degree of freedom vibration theory, of which a brief review is therefore included here. The heave and pitch motions of a car body constitute a 2-dof (two degree-of-freedom) system [2]

2.2.1 Free Vibration Damped

The suspension is present in order to reduce the discomfort arising from road roughness. However, as will be known later, the vehicle is very prone to vibrate at its natural frequency of around 1–2 Hz in practice; to control this, some damping must be added. Damping is dissipation of energy when there is movement. In a mechanism such as a car suspension, damping is deliberately introduced by the incorporation of the dampers. [2] There is extra damping from material hysteresis in the rubber bushes and in some cases from Coulomb friction at sliding joints. The schematics of linear damping which is also called viscous damping is shown in figure 2.2

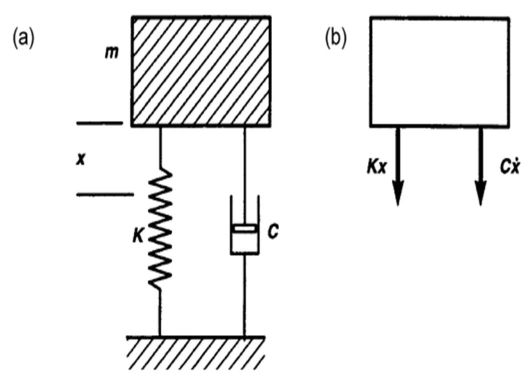


Figure 2.2: Viscous damping

2.2.3 Types of Shock Absorbers

There are two types of shock absorber are given below.

A) Air Shock Absorber

B) Damper Shock Absorber

1) Mono Tube Shock Absorber

2) Twin Tube Shock Absorber

These two types of damper shock absorbers are shown in figure 2.1.

A) Air Shock Absorber

Air shock absorber consists of an air chamber, an iron piston and a fluid.

B) Damper Shock Absorber

A damper shock absorber consists of a single chamber or two chambers; it may be fluid field or filled with air. It is commonly used to absorb the shock during the linear motion of a vehicle

1) Mono Tube Shock Absorber

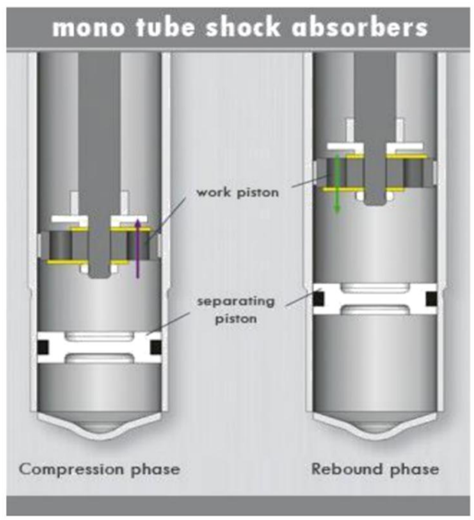


Figure 2.3: Mono Tube Shock Absorber

The mono tube damper as shown in figure 2.3.consists of single tube with two valves. It is mostly oil filled and used in larger vehicles. When the damper compresses one of the valve gets opened and when it extends the other valve gets open and the first one closes. The amount of the fluid released depends on the speed of the bumps it gets while moving. If it receives low speed small bumps the larger vents get opened and there is large amount of fluid is released. On the other hand, if gets high speed strong bumps the smaller vent gets opened and a small amount of oil is released.

2) Twin Tube Shock Absorber:

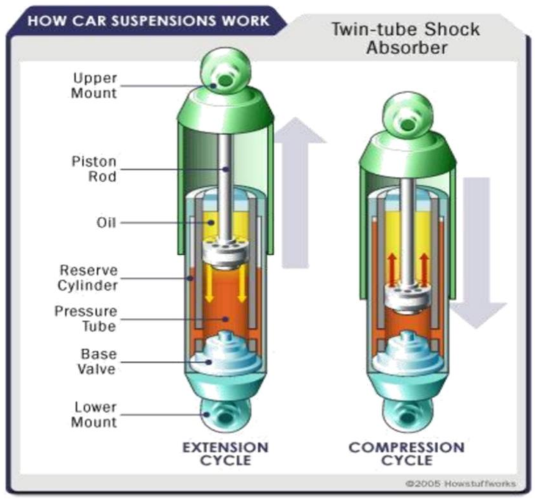


Figure 2.4: Twin Tube Shock Absorber

Also known as a "two-tube" shock absorber, this device consists of two nested cylindrical tubes, an inner tube that is called the "working tube" or the "pressure tube", and an outer tube called the "reserve tube". At the bottom of the device on the inside is a compression valve or base valve. When the piston is forced up or down by bumps in the road, hydraulic fluid moves between different chambers via small holes or "orifices" in the piston and via the valve, converting the "shock" energy into heat which must then be dissipated.

2.2.4Shock / Energy Absorbers

Shock absorbers typically aim to absorb a maximum amount of kinetic energy and sometimes potential energy, usually in the most efficient manner possible, and to bring a moving mass to a stop with minimal force or deceleration. Many shock absorbers consist of a combination of spring and damping components. These definitions are not absolute. They are presented here in an attempt to define their intended function in a specific application. There exists some overlap in the categorization of these items. For instance, isolators and shock absorbers sometimes consist of a combination of dampers and springs. Other times, a damper can be used exclusively to absorb shock. But whatever the application, these items all contribute to the mitigation of shock and vibration and result in benefits including a reduced response acceleration, reduced deflection and stress, reduced component weight, improved bio-dynamics, longer fatigue life, architectural enhancement, or reduced cost.

2.2.5 Springs

There are many types of springs. Springs react with force as a function of position or deflection. Many mechanical springs provide a force that is directly proportional to the amount of deflection. These are linear springs and obey Equation 2.1

$$\text{Force} = \text{Spring Rate} \times \text{Deflection, or } F = KX \tag{2.1}$$

Other springs are non-linear and obey a variety of different relationships with respect to spring force versus deflection. Some springs even change their characteristics as a function of frequency or velocity. This is sometimes referred to as the dynamic stiffening factor. Springs that have a rate near zero are referred to as constant force springs since their force is virtually unchanged with a changing deflection. Some types of springs are listed below:

Wound (Coil) springs – Advantages include low cost, long life, very predictable performance, readily and quickly available. Disadvantages include inherent instabilities, difficult to fixture into a stable condition, and poor availability in very large sizes and force levels.

Machined springs – Advantages include compact package, no practical size limitations, extreme accuracy, inherent stability, and the ability to incorporate attach points for any type of mechanism. The main disadvantage is their high cost.

Elastomer Springs – Advantages include low cost and moderate life under corrosive environments. Disadvantages include relatively high temperature dependency, size limitations, and sometimes-undesirable nonlinear behavior.

Pneumatic Springs – Advantages include compact designs and moderate life. Disadvantages include relatively high temperature dependency and size limitations.

Liquid Springs – Advantages include very compact designs, moderate life, and the ability to add high damping levels. Disadvantages include temperature dependency and relatively high cost. Springs can be designed into a dynamic system to provide the following types of arrangements: Un-centered – Displacement of the system changes with the addition of load. A typical example is the spring of an automobile suspension. Soft-centered – One or more springs are used to position the isolated mass to a neutral position. The removal of any external force will cause the system to move back towards this position. However, any system friction will prevent the return to an accurate position. Hard-centered the spring is preloaded to provide a means of returning the system to its initial condition and to prevent any movement under relatively low input levels.

2.2.6 Dampers

Damping is a critical tool in shock and vibration isolation. Dampers can be designed to react with force as a function of relative velocity, position, in some cases frequency, or combinations thereof. Types of dampers include hydraulic or viscous dampers, coulomb or friction dampers, elastomer dampers, structural damping materials, and tuned mass dampers, among others. The attributes of various damper types are out- lined further below.

2.2.7 Damper Construction

Main piston: In all dampers the main piston contains the primary valving components and produces the majority of the damping forces. In all three constructions, the Main Piston produces all the rebound force.

Compression Piston: The Compression Piston has several functions: Produces compression force based on the rod displacement through the Compression Piston. This results in lower compression pressures for the same damping force, typically resulting in less tire force variation and better grip. And provides a pressure balance to the main piston during the compression stroke to prevent cavitation. This enables dampers with Compression Pistons to operate with lower gas pressure. Note that the Mono tube damper does not have a Compression Piston.

Gas Separator Piston: Keeps the gas separated from the oil.

Main Piston Tube: This is the tube where the Main Piston operates. Note in the Monotube damper it is also the outer tube.

Reservoir Tube: The Reservoir Tube is the outer tube on a Twin Tube damper and creates the area for extra oil and the gas pressure in a Twin Tube shock.

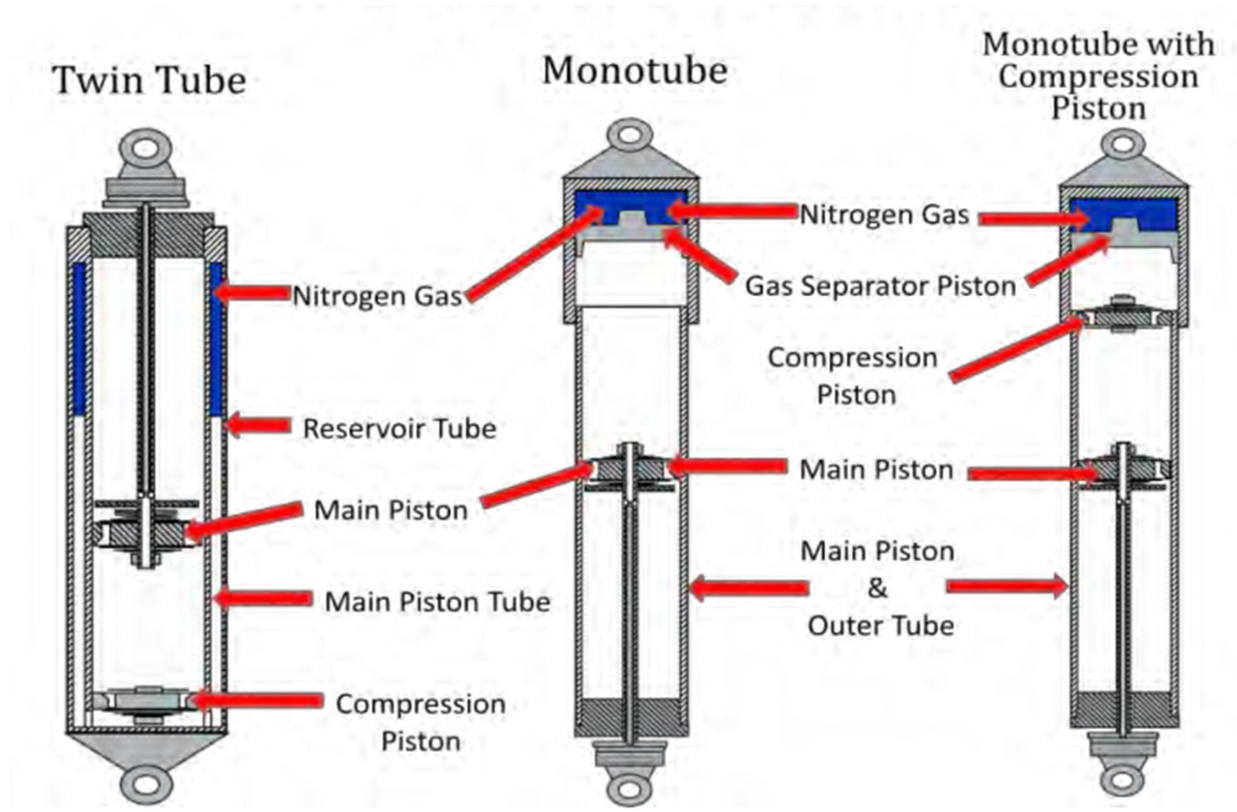


Figure 2.5: Elements of damper

Advantages and Disadvantages of Each Construction

Twin Tube Advantages

- ✓ Inexpensive

Compression Piston. This means lower compression pressures and typically lower gas pressure can be used.

Disadvantages

- ✓ Has to be mounted rod up to avoid cavitation
- ✓ No Gas Separator Piston to separate the gas from the oil. This can result in the gas and the oil mixing, reducing damping forces.
- ✓ Smaller piston diameter for a given outer tube diameter. The result is higher operating pressures and more fade.

Damper Valve Operation

As we look farther inside the damper, let us examine how the actual valving works. There are three stages to the valving operation; low speed or damper velocities zero to 50.8 mm/sec (0-2 in/sec), mid-speed or damper velocities 50.8-203.2 mm/sec (2 to 8 in/sec) and high speed, above 203.2 mm/sec (8 in/sec). The rebound and compression valves (piston and compression piston) all work the same. The piston valving in all three damper constructions work the same. The compression piston valving in a Twin Tube damper and Monotube with Compression Piston work exactly the same.

Low Speed Control

Low speed is zero to 50.8 mm/sec (0 to 2 in/sec). In the valving, this is controlled by:

- ✓ Bleed holes or bleed discs
- ✓ The by-pass jet or adjustment needle
- ✓ Leakage around the piston

Linear Damping Function:

Determining the level of damping or determining the optimal damping relationship for a given application can be a complex process. Some applications can benefit greatly by adding a level of damping consistent with proven techniques that do not require a highly complex optimization process. For oscillating sprung mass systems, the amount of damping is often characterized as a percentage of critical damping. The critical damping level is defined as that amount of damping that would cause a sprung mass to slowly come to an equilibrium position without overshooting that position (or oscillating). Mathematically, critical damping follows Equations 2.2 and 2.3:

$C_{critical}$= 2 × ω × mass, where ω is in radians per second

$$\omega = \frac{2\pi}{\tau} = 2\pi f \tag{2.2}$$

$C_{critical}=2\times\sqrt{(K \times mass)}$, where K is the system spring rate as defined above (2.3)

Inherent in the equations above is that the resulting C value is in units of force-time/displacement and this implies a linear relationship between damper force and velocity. Therefore, it follows that the damper output function obeys the following equation:

$$\text{Damper Force (N)} = \text{C (N-sec/m)} \times \text{Velocity (m/sec)} \tag{2.4}$$

As a point of reference, many automotive suspensions have dampers with 20-25% of critical. Truck suspensions might typically have 30-40% of critical. Damping used for the suppression of weapons shock in military applications is not normally categorized in terms of percent damping.

2.2.8 Types of Dampers

The following is a list of dampers that exist for use in control of dynamic systems that have proven effective for many applications: Structural Damping – Structural damping is generally considered the amount of damping that is inherent in a structure and/or its materials. Structural damping levels are relatively low. However, its magnitude varies widely depending on the types of materials used and the design of the structure. The amount of critical damping, also known as the damping ratio, that exists in structures can be 0.5% of critical (damping ratio = 0.005) or less for rigid structures and can be as high as 10% (damping ratio = 0.10) for massive structures using lightweight construction and complex joints. Coulomb or Frictional Dampers– Coulomb damping is often obtained by slippage of a joint at stress levels below that of the material yield strength. A wide range of damping ratios can be obtained, although the amount is generally limited by the frictional coefficients of the materials and the high amount of wear that can be experienced by high normal forces. Since friction is inherently high, this type of damping often leaves a permanent drift, or offset, from the original position of the system. The damping level will vary somewhat with temperature, wear, corrosion, aging, and exposure to external influences, such as water or oil.

Elastomer Dampers – Elastomers have the ability to provide both spring force (position dependent) and damping force (velocity and/or frequency dependent). Damping levels vary greatly depending on the compound type. Most elastomer dampers are highly non-linear. Care must be exercised in the design of elastomer dampers since they are temperature sensitive, they degrade with age and exposure to chemical reagents or UV light, and they are difficult to control to a high level of accuracy and consistency. However, they are generally inexpensive and standardized to some degree, making them readily available.

Active Dampers – Active dampers are those that are comprised of a power source, programmed control logic, an input measuring device, and a mechanical driver that has the ability to respond quickly to the input according to the logic and drive itself to reduce the response acceleration of the protected mass. Since the active damper has the ability to react to the input on a real time basis, active controls have the ability to provide added benefits over a completely passive damper in some instances. However, the benefits come at a price that is often substantial.

Active dampers(or active isolation systems) must possess a power source that is capable of reacting quickly the input. This requires a relatively high power source in most practical applications that consider the use of active dampers. Therefore, the weight, size and expense are difficult to overcome. Additionally, the complexity and reliability of active drivers are major factors when considering the appropriate applicability of active systems to solve a shock and vibration problem. Careful consideration must also be given to the level of improvement that can be realized through the use of active systems as opposed to passive or adaptive systems. Research has shown that only a limited amount of performance improvement, if any, can be realized with active dampers as compared to specialized passive or adaptive dampers. Hydraulic (Fluid) or Viscous Dampers – Fluid or viscous dampers fundamentally obey the following equation:

$$F \text{ (force)} = C \times V^{\alpha} \tag{2.5}$$

Where V is velocity, C is the damping constant, and αis the damping exponent. Dampers are available that can produce an exponent between approximately 0.2 and 2. Typically, fluid dampers with very low exponents near 0.2 produce this relationship through the use of pressure responsive valves (PRV's). Dampers with exponents near 2 typically obey this relationship due to the fact that the orifice type and the damping fluid is mainly governed by Bernoulli's equation of fluid flow that states that fluid pressure is a function of flow velocity squared (V^2) and the density of the fluid. As stated previously, dampers with an exponent of one are considered linear dampers.

2.3 Power Generating Shock Absorber

The Power-Generating Shock Absorber (PGSA) converts this kinetic energy into electricity instead of heat through a Linear Motion Electromagnetic System (LMES). The LMES uses a dense permanent magnet stack embedded in the main piston, a switchable series of stator coil windings, a rectifier, and an electronic control system to manage the varying electrical output and dampening load. The bottom shaft of the PGSA mounts to the moving suspension member and forces the magnet stack to reciprocate within the annular array of stator windings, producing alternating current electricity. That electricity is then converted into direct current through a full - wave rectifier and stored in the vehicle's batteries. The electricity generated by each PGSA can then be combined with electricity from other power generation systems and stored in the vehicle's batteries to increase battery life. In non-electric vehicles the electricity can be used to power accessories such as air conditioning. Several different systems have been developed recently, though they are still in stages of development and not installed on production vehicles.

Figure 2.6: Power Generating Shock Absorber

The mechatronic system we have chosen to model is Power-Generating Shock Absorber (PGSA) acting on an automotive chassis. The shock absorber will be used in conjunction with a spring to simulate one of the four-suspension systems of an automobile. When designing an automotive suspension system the key is to balance the ride of the automobile. More specifically, the suspension is meant to absorb the effects of an uneven driving surface and tilt/sway of the car. However, excess energy loss occurs due to resistance in the damper fluid and compression of the spring. The PGSA converts kinetic energy into electricity through the use of a Linear Motion Electromagnetic System (LMES). in the Figure 2.6, the absorber consists of a damper with permanent magnet stack that slide in and out of stator windings connected to two sliding blocks inside the damper casing. The Power-Generating Shock Absorber (PGSA) converts this kinetic energy into electricity instead of heat through the use of a Linear Motion Electromagnetic System (LMES`).Shock absorbers are installed between chassis and wheels to suppress the vibration, mainly induced by road roughness, to ensure ride comfort and road handling. Conventional rotational regenerative shock absorbers translate the suspension oscillatory vibration into bidirectional rotation, using a mechanism like ball screw or rack pinion gears. Figure.2.7 shows one such an implementation, where the rotary motion is changed by 90 degree with a pair of bevel gears for retrofit. And electricity generated in this mechanism. That electricity is then converted into direct current through a full wave rectifier and stored in the vehicle's batteries.

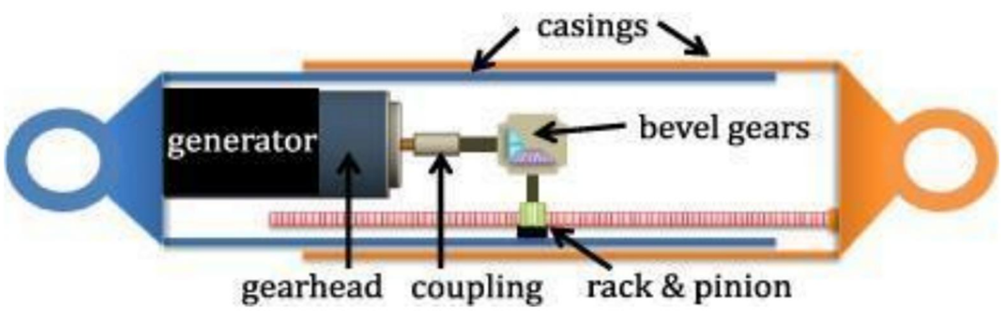


Figure 2.7: Traditional design of a rack-pinion based power generating shock absorber

The suspension system consist two types of cylinder. One have larger diameter and another have smaller diameter. When load is applied, the smaller diameter cylinder moves into the larger diameter cylinder which produces magnetic field due to repetition of movement of cylinders over coils. Then the electric generator converts the magnetic effect into electricity which is to be stored in battery.

2.3.1 Regenerative Shock Absorber

Regenerative shock absorbers were first proposed two decades ago to recover the kinetic energy dissipated by traditional oil shock absorbers in suspension vibration. Many researchers explored different principles and designs of regenerative vehicle shock absorbers, which can be classified into two main categories. The first category is based on the specialized design of a linear electromagnetic generator, which generates power from the relative linear motion between magnets and coils. Karnopp. [3] First proposed the idea of designing mechanical dampers with controllable damping coefficients using permanent magnet linear motors for application in vehicle suspensions.

2.3.2 Energy Dissipation of Vehicle Suspension

In the past, we pay little attention to energy loss of vehicle suspension. However, how much energy is dissipated by the shock absorbers of vehicle suspension? According to reference [5], only 10-20% the fuel energy is used for vehicle mobility. One of the important losses is the energy dissipation in suspension vibration. Velinskyet.al. 6] Concluded that the dissipated energy by suspension dampers is related with road roughness, vehicle speed, suspension stiff and damping coefficient. Segel et al. [7] analyzed the energy dissipation of dampers of passenger vehicle, and shown that the total power of four dampers was about 200W when running on a poor road at the speed of 13.4m/s. These data indicate that the energy dissipation of vehicle suspension can't be ignored.

2.3.3 Configuration of Regenerative Suspensions

According to the working principle, the regenerative suspension can be divided into two types: mechanical and electromagnetic regenerative suspension.

A. Mechanical Regenerative Suspensions

The mechanical regenerative suspension is reformed from the traditional hydraulic/ pneumatic suspension. It absorbs the kinetic energy of suspension and converts into potential hydraulic / pneumatic energy to be stored in accumulator. However, these hydraulic / pneumatic systems characterize some disadvantages. One, the complex pipeline system has considerable weight and need more installation room. Two, hose leaks and ruptures may disable the whole system. Three, the responding bandwidth of hydraulic / pneumatic systems is narrow, which confines the suspension performance. Four, the reuse of the regenerated hydraulic / pneumatic energy are limited, especially when the automotive industry is toward commercializing hybrid electric vehicles and full electric vehicles. Hence, the researches on hydraulic / pneumatic regenerative suspension are relative rare. Jolly et al. [8] Proposed an energy regenerative system based on hydraulic device to control the vertical vibration of vehicle seat using the regenerated energy. Nissan [9] developed a fully active suspension system with hydraulic actuators, which suppresses the suspension vibration by accumulating or releasing the energy in the accumulator under the control of valves. Noritsugu [9] investigated an active air suspension via reclaiming the exhaust to control suspension vibration for improving suspension performance and decreasing energy consumption.

B. Electromagnetic Regenerative Suspensions

On the contrary, electromagnetic regenerative suspension transforms the shock energy into electric energy that is more convenient to store and reuse, and has high performance, increased efficiency, less space requirements, and so on [10]. In recent years, electromagnetic suspension (EMS) system has drawn worldwide attention. Permanent magnets motor is favored in EMS to provide active force in actuator mode or damping force in generator mode. The damping force can be simply changed by tuning the shunt resistances. There are six types of electromagnetic regenerative suspension classified by structure configuration, and relating researches will be stated as follow.

2.3.4 Direct-Drive Electromagnetic Suspension

In direct-drive electromagnetic suspension, linear permanent magnets motor is usually used to replace the traditional shock absorber. It turns the mechanical energy of relatively motion between vehicle chassis and wheel into electric energy needing no transmission devices. Design theories of the linear motor may reference [11-12]. Okada et al. [13] studied the active and regenerative vibration control suspension using linear actuator, and its performances of vibration isolation and energy regeneration were analyzed. Suda et al. [14] investigated a self-powered active vibration control system with two linear motors for truck cabins. In this system, an electric generator that is installed in the suspension of the chassis regenerates vibration energy and stores it in the condenser. An actuator in the cab suspension achieves active vibration control using the energy stored in the condenser. Since the weight of the chassis of a typical heavy-duty truck is greater than that of the cabin, vibration energy in suspension of a chassis is expected to be greater than that in the cab suspension. So this system is self-powered. Furthermore, they proposed self-powered active vibration control using a single electric actuator [15]. Goldner [16] invented an electromagnetic linear generator to capture the suspension vibration energy. Zuo et al. [17] designed a linear energy harvester to test the amount of energy, which can be regenerated. Gysen et al. [10] utilized direct-drive electromagnetic motor to improve the suspension performance, and proved its efficiency for harvesting vibration energy. Bose

Company [18] applied the linear motor in vehicle suspension, shown as Fig.2.8. The system ends up consuming one-third of the energy used by a car's air conditioner.

Figure 2.8: Electromagnetic suspension developed by Bose

2.3.5 Ball Screw Electromagnetic Suspension

Ball screw is a common transmission device that converts linear motion into rotation. Arsem [19] invented an electric shock absorber with ball screw to harvest vibration energy. Murty[20]also invented a ball screw electric damper whose damping force can be adjusted by changing the shunt resistance. Suda et al.[21]proposed a ball screw harvester (shown as Fig.2.9), and analyzed its dynamic and regenerative characteristics. F. Yu et al. [22] designed a ball screw damper and verified its performance.

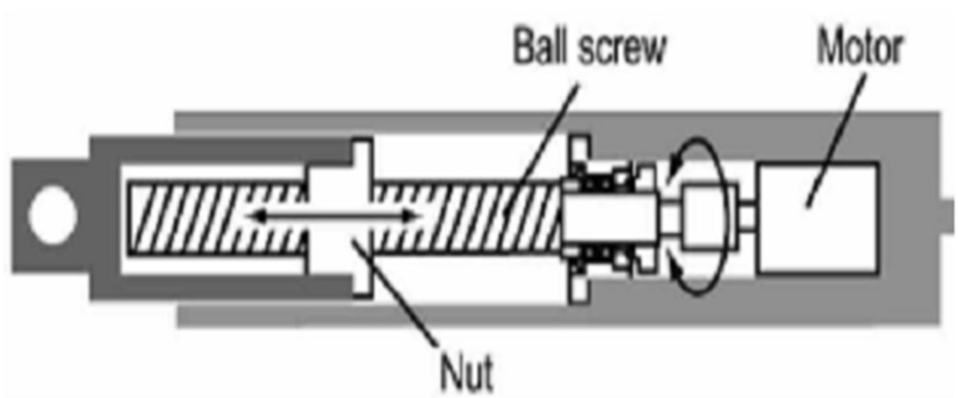


Figure 2.9: Ball screw harvester

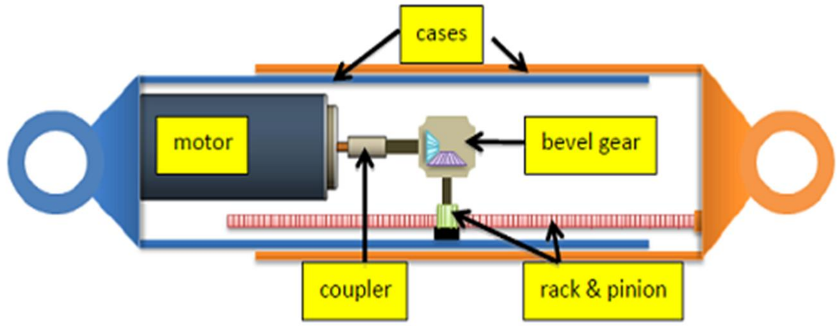


Figure 2.10: Rack-pinion electromagnetic damper

2.3.7 Rack-Pinion Electromagnetic Suspension

Rack-pinion can also convert linear motion into rotation. Suda et al. [23] studied a regenerative active suspension combining rack-pinion and rotary motor. Benoet al. [24-25] developed electronically controlled active suspension system (ECASS), which adopted the rack-pinion configuration. The experiment results indicated that the limit speed and handling performance of vehicle had been enhanced significantly. Pei [1] designed a rack-pinion damper incorporating a bevel gear for changing the motor axes to parallel with the orient of linear motion, shown as Fig.2.10

2.3.8 Planetary Gear Electromagnetic Suspension

Planetary gear is always used to reduce or increase rotate speed. Introducing the planetary gear is helpful for boosting motor efficiency and active force. For example, in order to improve the regenerative efficiency, Suda et al. [26-27] add a planetary gear to ball screw damper. Horseman incorporating with L-3 and Texas University is developing ECASS for tracked combat vehicle. All the elements including motor, planetary gear set, sensors, are integrated in road arm, shown as Fig.2.11This system has a sufficient active force, compact structure and good safety property that the combat vehicles require.

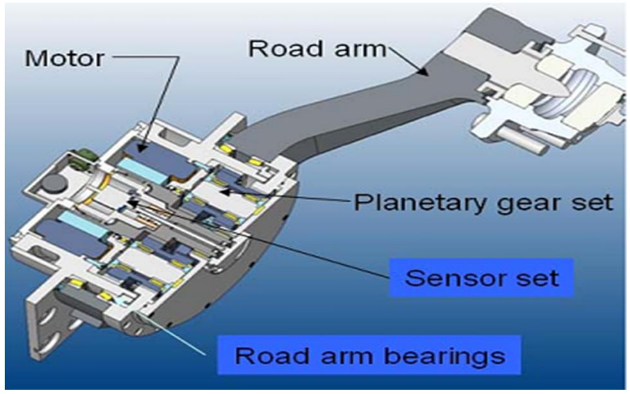


Figure 2.11: Actuator for combat vehicle suspension

2.3.9 Hydraulic Transmission Electromagnetic Suspension

Levant Power Corp. [28] is developing a regenerative damper, called GenShock, combining hydraulic transmission and electric motor. Fig.2.12 shows the working principle of this damper. A suit of rectifying pipe guarantees that the hydraulic motor driven by fluid rotates by a consistent direction whatever the piston runs up or down. Because the rotation direction of electric motor does not alternate frequently, the regenerative efficiency is enhanced obviously.

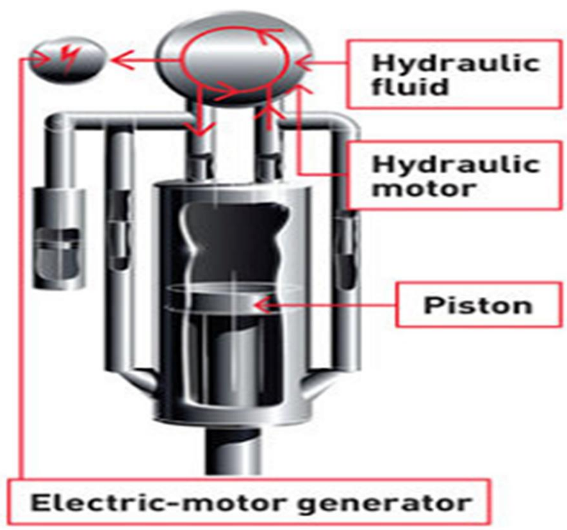


Figure 2.12: Hydraulic transmission regenerative damper

Levant Power Corp. claims that GenShock increasing fuel economy of by up to 6% for military vehicles as well as improves ride quality via an adaptable, variable-damping suspension. On the other hand, reduced heat dissipation in the damper through energy recovery helps diminish maintenance requirements. Xu et al. [29-30] proposed a hydraulic transmission electromagnetic energy-regenerative suspension, introduced its working principle, and the simulation results revealed that its comprehensive performance is superior to that of the passive suspension.

2.3.10 Self-Powered Magnetorheological Suspension

In recent decade, magneto rheological (MR) damper has been shown a cheerful prospect for structural vibration education, because of its mechanical simplicity, high dynamic range, low power requirements, large force capacity and robustness. Aims to save energy furthermore, many scholars have been studying self-powered MR damper. Jung *et al.* [34] proposed a self-powered smart damping system that consists of an MR damper and an electromagnetic induction (EMI) device to reduce vibrations of stay cable. The EMI device absorbs vibration energy to generate electrical energy and power the MR damper. Though the EMI device is separated from the MR damper, it provides a new technology scheme for self-powered vibration control system. Choi [35] and Bogdan [36] exerted some similar researches. Chen *et al.* [37] proposed a self-sensing MR damper with power generation, shown as Fig.2.13. In this structure, MR damper and power generator is integrated into an organic whole and the voltages of contiguous coils with different phase are utilized to calculate the relative velocity.

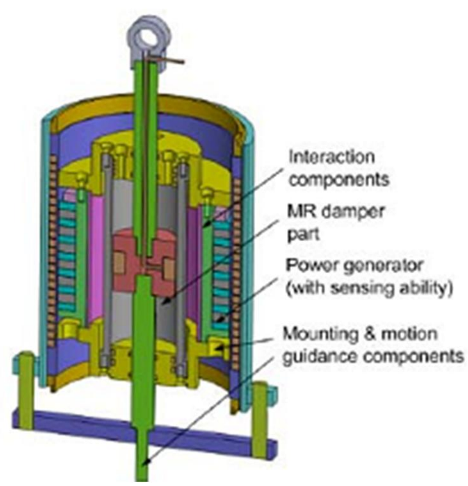


Figure 2.13: Self-sensing MR damper with power generation

3. MATERIALS AND METHODS

For accomplishing the set objectives of this project, which is designing, and construction of mechanism for harvesting energy from shock absorber, a systematic and well-designed step-by step procedure was followed. Various materials used for this purpose.

3.1 Materials

Various materials used in this project are listed in the following table

Table 3.1: List of Components of MMR

No	Items	Specification
1	Shock absorber	Standard
2	DC generator	6V ,3W bicycle motor
3	Ball bearing	8mm×20mm
4	Roller clutches	28mm×5mm
5	Thrust bearings	20mm×40×2mm
6	Large Bevel gears	14×20×66mm
7	Small bevel gear	16 mm×8 mm×21.01 mm
8	Rack	280 mm×10 mm
9	Pinion	8mm×20mm

In the above table the specification is not calculated its measured from the selected vehicle of Sino-truck.

3.2 Methodology

The methodology followed which starts with literature review include the following stages:

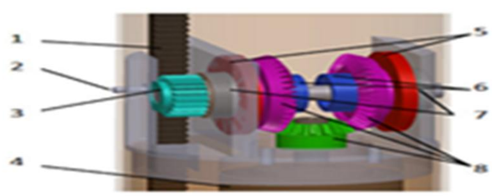
1. Literature review.
2. Designing of various components of the mechanism for energy harvesting from shock absorber.
3. Modeling of the mechanism using CATIA.
4. Collection of the materials and components required for construction of prototype.
5. Manufacturing of the components required.
6. Assembling the components.
7. Demonstrating the functionality by using the prototype developed.

These steps involves in the process are explained briefly as follows:

3.2.1 Literature review

Literature review is an important step for any project work. So, a thorough literature review was made related to the topic and regenerative shock absorber. In this respect many research articles, thesis works, books, and websites were referred. Sufficient knowledge was acquired in regards to a vibrations experienced by vehicle driving on the road, and various types of shock absorbing systems being used and their functionality. Research articles were referred to know the present status of the technology and together the information about the mechanisms being used in harvesting energy from shock absorbers. Also collected the detailed information to design and construct the rack and pinion mechanism proposed in this project.

3.2.2 Designing of various components of the mechanisms



1.rack
2.roller
3.pinion
4.planetary gears and motor
5.thrust bearings
6.roller clutches
7.ball bearings
8.bevel gears

Figure 3.1: Components mechanical motion rectifier(MMR)

Rack and pinion

A rack and pinion is a type of linear actuator that comprises a pair of gears which convert rotational motion into linear motion. A circular gear called "the pinion" engages teeth on a linear "gear" bar called "the rack"; rotational motion applied to the pinion causes the rack to move, thereby translating the rotational motion of the pinion into the linear motion of the rack. A rack is a gear whose pitch diameter is infinite, resulting in a straight-line pitch circle. Involutes of a very large base circle approach a straight line. In this project the length and width of the rack is 280mm and 10mm respectively.

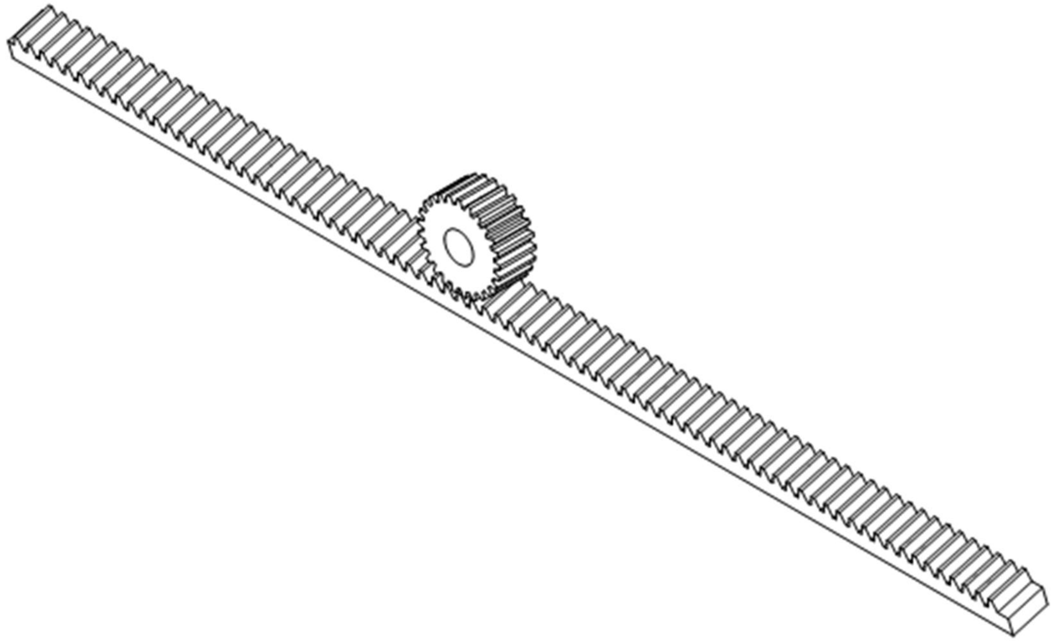


Figure 3.2: Rack and pinion 3D

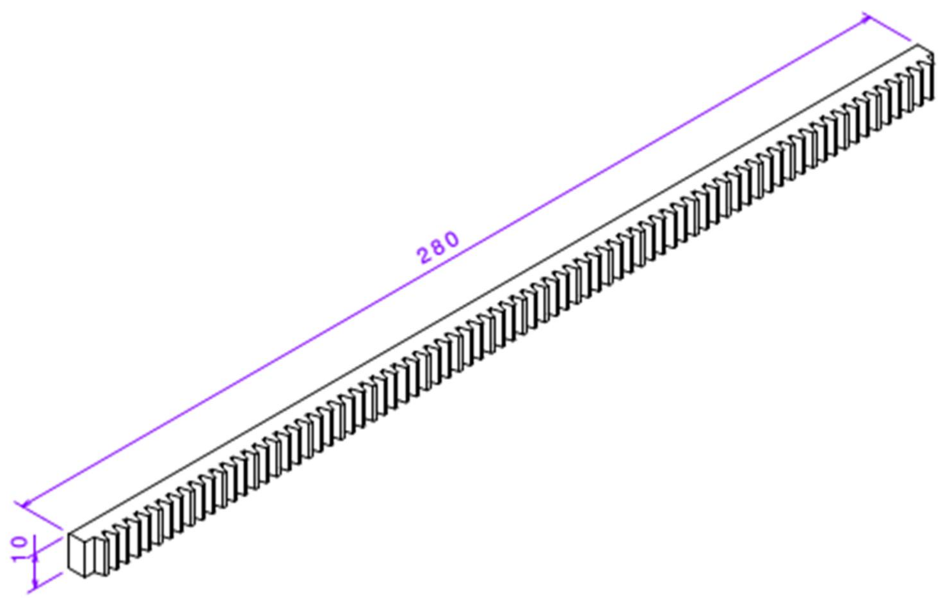


Figure 3.3: Rack 3D

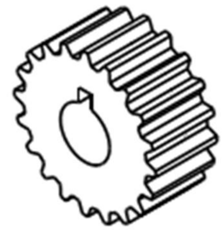


Figure 3.4: Pinion 3D

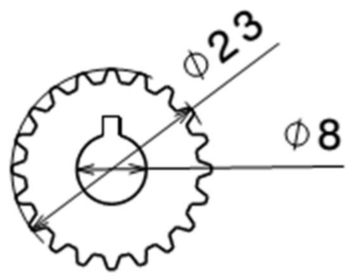


Figure 3.5: Front View of pinion

Roller Bearing

In rolling contact bearings, the contact between the bearing surfaces is rolling instead of sliding as in sliding contact bearings. The ordinary sliding bearing starts from rest with practically metal-to-metal contact and has a high coefficient of friction. It is an outstanding advantage of a rolling contact bearing over a sliding bearing that it has a low starting friction. Due to this low friction offered by rolling contact bearings these are called antifriction bearings.

Ball bearing

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 as shown in Fig. 3.6. 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 are 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.in this project the ball bearing have 8mm and15.8mm internal and external diameter.

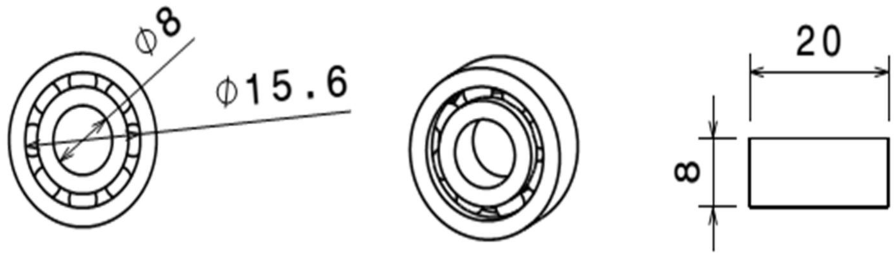


Figure 3.6: Ball bearings

Bevel Gear

The bevel gears are used for transmitting power at a constant velocity ratio between two shafts whose axes intersect at a certain angle. The pitch surfaces for the bevel gear are frustums of cones. The two pairs of cones in contact are shown in Fig. 3.7. The elements of the cones, as shown in Fig. 3.7 (a), intersect at the point of intersection of the axis of rotation. Since the radii of both the gears are proportional to their distances from the apex, therefore the cones may roll together without sliding. In Fig 3.7 (b), the elements of both cones do not intersect at the point of shaft intersection. Consequently, there may be pure rolling at only one point of contact and there must be tangential sliding at all other points of contact. Therefore, these cones cannot be used as pitch surfaces because it is impossible to have positive driving and sliding in the same direction at the same time. We, thus, conclude that the elements of bevel gear pitch cones and shaft axes must intersect at the same point.

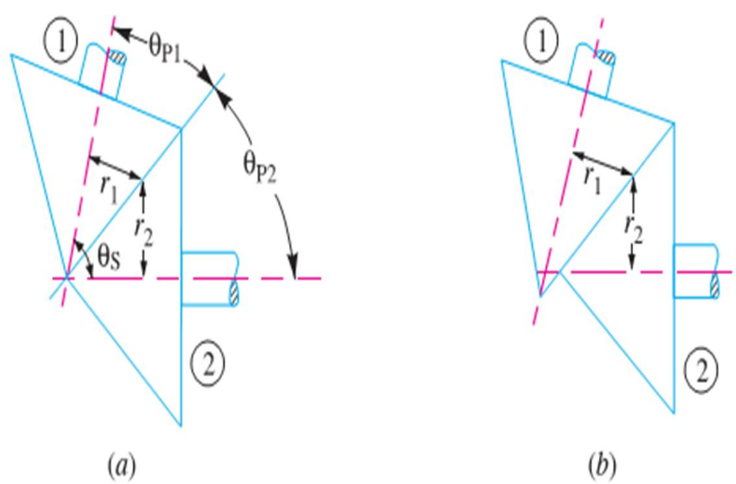


Figure 3.7: Pitch surface for bevel gears

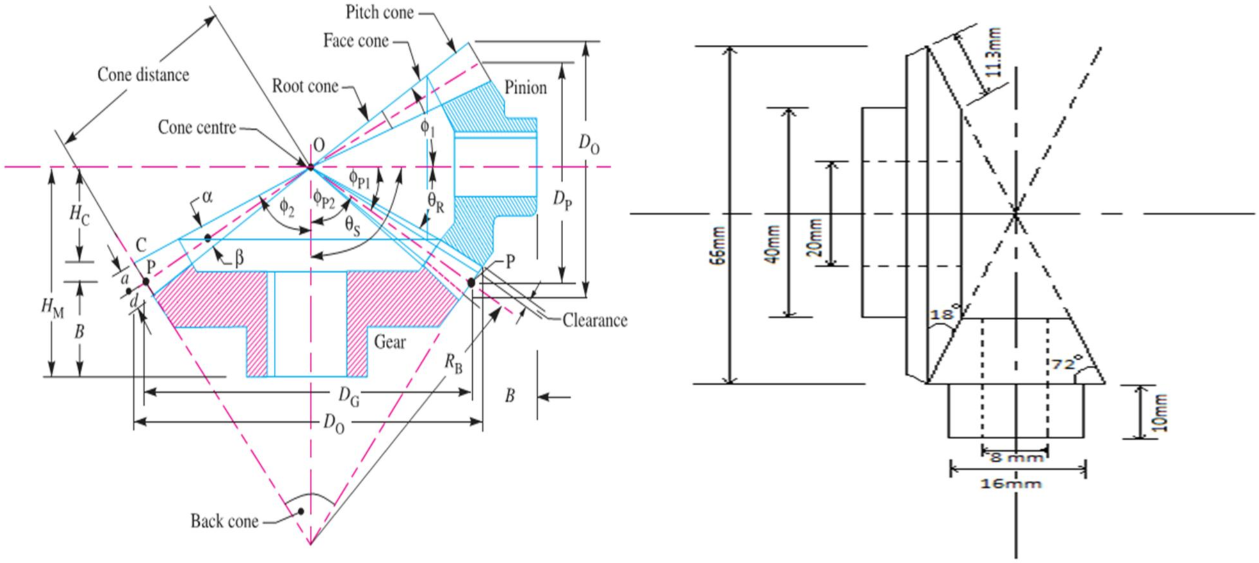


Figure 3.8: Bevel Gear

Table 3.2: The measured dimensions of the components parts of MMR

s/no	Components	Length (mm)	Width (mm)	Thickness (mm)	Internal diameter (mm)	External diameter (mm)
1	Rack	280	10	8	-	-
2	Pinion	-	-	10	8	23
3	Shaft	70	-	-	-	8
4	Ball bearing	-	-	8	8	20
5	One way clutch	-	-	5	8	28
6	Small bevel gear	-	-	16	8	21.01
7	Large bevel gear	-	-	14	20	66

3.2.3 Determination of Pitch Angle for Bevel Gears

Let θ_{P1} = Pitch angle for the pinion,

$$\theta_{P1} = \tan^{-1}(1/V.R) \tag{3.1}$$

$$= \tan^{-1}(1/3.56)$$

$$= 15.689^{o}$$

θ_{P2} = Pitch angle for the gear,

$\theta_{P2} = \tan^{-1}{}_{(V.R)}$ (3.2)

$= \tan^{-1}(3.56)$

$= 74.31^{\circ}$

θ_S = Angle between the two shaft axes,

D_P = Pitch diameter of the pinion,

= 18mm

D_G = Pitch diameter of the gear, and

= 64mm

$$OP = \frac{D_p/2}{\mathrm{Sin}\theta_{P_1}} = \frac{D_G/2}{\mathrm{Sin}\theta_{P_2}} \quad (3.3)$$

$$= \frac{18/2}{\mathrm{Sin}15.689} = \frac{64/2}{\mathrm{Sin}74.31}$$

= 33.28mm

3.2.4 Determining the number of teeth of bevel gear

✓ For large gear:

RB = R $\cos \theta_{p2}$ (3.4)

= 64/2× $\cos 74.31$

= 8.654

Where R is the diameter of large bevel gear

RB=Back cone distance or equivalent pitch circle radius of large bevel gear

✓ For small gear :

RB = R $\cos \theta_{p1}$ (3.5)

=18/2$\cos 15.689$

= 8.665

Where, R is the diameter of pinion gear.

RB=Back cone distance or equivalent pitch circle radius of small bevel gear

The equivalent (or formative) number of teeth:

$$T_E = 2\ RB / m = T\cos\theta_{p2} \tag{3.6}$$

$2\ RB / 2 = T\cos\theta_{p2}$

$RB = T\cos\theta_{p2}$

$8.665 = T\cos 74.31$

$T = 8.665/\cos 74.31$

$=32$

Where m is the module, m = 2

T is the equivalent number of teeth

The equivalent (or formative) number of teeth:

$T_E = 2\ RB / m = T\cos\theta_{p1}$

$2\ RB / 2 = T\cos\theta_{p1}$

$RB = T\cos\theta_{p1}$

$8.665 = T\cos 15.689$

$T = 8.665/\cos 15.689$

$T = 9$

Proportions for Bevel Gear

- ✓ The proportions for the bevel gears may be taken as follows:[31]
  1. Addendum = a= 1 m = 1×2 = 2mm
  2. Dedendum = d= 1.2 m = 1.2 ×2 = 2.4mm
  3. Clearance = 0.2 m = 0.2×2 = 0.4mm
  4. Working depth = 2 m= 2×2 = 4mm
  5. Thickness of tooth = 1.5708 m = 1.5708×2 = 3.14

Where **m** is the module. The bevel gears are not interchangeable; therefore, these are designed in pairs

- ✓ The proportions for the pinion gears may be taken as follows:
  1. Addendum = a= 1 m = 1×2 = 2mm
  2. Dedendum = d= 1.2 m = 1.2 ×2 = 2.4mm
  3. Clearance = 0.2 m = 0.2×2 = 0.4mm
  4. Working depth = 2 m= 2×2 = 4mm
  5. Thickness of tooth = 1.5708 m = 1.5708×2 = 3.14

Lewis factor for bevel gear

$y = 0.124 - \frac{0.684}{T}$ from Lewis factor of gear .

$y = 0.124 - \frac{0.684}{T}$ [For 14 1/2°, composite and full depth involutes system]

$y = 0.154 - \frac{0.912}{T}$ For 20°, full depth involutes system

$$y = 0.175 - \frac{0.84}{T} \text{for } 20° \text{ stub system}$$

- ✓ **For bevel gear**

$$y = 0.124 - \frac{0.684}{32}$$

$$y = 0.1026$$

- ✓ **For pinion gear (small bevel)**

$$y = 0.124 - \frac{0.684}{9}$$

$$y = 0.048$$

Strength of Bevel Gears

The strength of a bevel gear tooth can be obtained in a modified form the Lewis equation for the tangential tooth load is given as follows.[31]:

$$W_T = (\sigma_0 \times C_V)b.\pi m.\acute{y}\left(\frac{L-b}{L}\right) \tag{3.7}$$

$$W_T = (410.20\text{N/mm}^2 \times 0.194)11.3 \times \pi \times 2 \times 0.1026\left(\frac{33.28 - 11.3}{33.28}\right)$$

W_T = 19095.58N

Where, σ_o= Allowable static stress,

C_v= Velocity factor,

$= \frac{3}{3+v}$, For teeth cut by form cutters,

$= \frac{3}{3+12.5}$,

$= 0.194$

V = Peripheral speed in m / s, v = 12.5m/s

b = Face width, $8^2 + 8^2$

$= \sqrt{8^2 + 8^2}$

b = 11.3mm

m = Module, m=2

y' = Tooth form factor (or Lewis factor) for the equivalent number of teeth,

L = Slant height of pitch cone (or cone distance)

$$= \sqrt{\left(\frac{64}{2}\right)^2 + \left(\frac{18}{2}\right)^2}$$

L = 33.28mm

Where, D_G= Pitch diameter of the gear, and

D_P= Pitch diameter of the pinion.

Notes:

1) The factor $\left(\frac{L-b}{L}\right)$ may be called as bevel factor.
2) For satisfactory operation of the bevel gears, the face width should be from 6.3 m to 9.5 m, where m is the module .Also the ratio L / b should not exceed 3. For this, the number of teeth in the pinion must not less than $\frac{48}{\sqrt{1+(V.R.)^2}}$, where V.R. is the required velocity ratio.
3) The dynamic load for bevel gears may be obtained in the similar manner as discussed for spur gears.
4) The maximum or limiting load for wear for bevel gears is. [31]:

Q = Ratio factor

$= \frac{2 \times V.R}{V.R+1}$ for external gear (3.8)

$= \frac{2 \times 3.56}{3.56+1}$

$= 1.56$

$$W_W = \frac{D_p.b.Q.K}{Cos\theta_{P1}} \quad (3.9)$$

$$W_W = \frac{18 \times 11.3 \times 1.56 \times 2 \times .0.107}{Cos15.7} = 70.5N$$

3.2.5 Forces Acting on a Bevel Gear

Consider a bevel gear and pinion in mesh as shown in Fig. 3.9. The normal force (W_N) on the tooth is perpendicular to the tooth profile and thus makes an angle equal to the pressure angle (φ) to the pitch circle. Thus normal force can be resolved into two components, one is the tangential component (W_T) and the other is the radial component (W_R). The tangential component (i.e. the tangential tooth load) produces the bearing reactions while the radial component produces end thrust in the shafts.

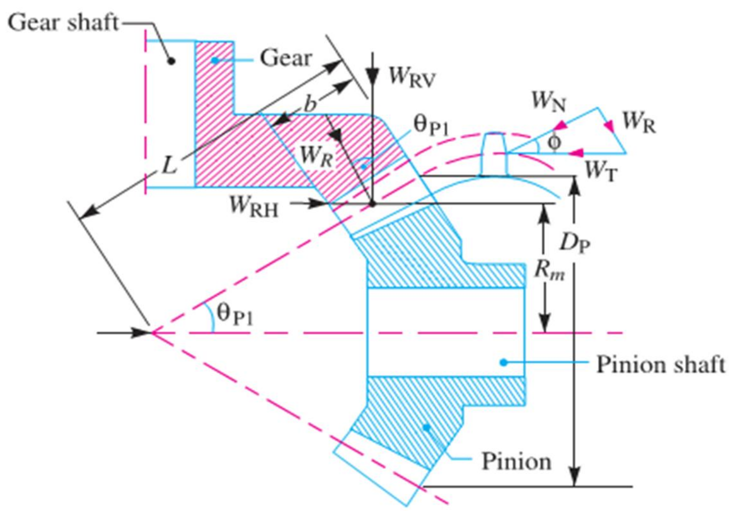


Figure 3.9: Force acting on a bevel gear

The magnitude of the tangential and radial components is as follows:

$$W_T = w_N \times \cos\varphi, \text{ and } W_R = W_N \times \sin\varphi \quad (3.10)$$

$$W_T = w_N \times \cos\varphi$$

$$w_N \times \cos\varphi = 22426.3N$$

$$w_N = 2582N$$

$$W_R = W_N \times \sin\varphi$$

$$W_R = 2582 \times \sin 20°$$

$$W_R = 883N$$

Where φ is pressure angle [31]:

These forces are considered to act at the mean radius (R_m). From the geometry of the Fig. 3.9, we find that

$$R_m = \left(L - \frac{b}{2}\right)\sin\theta_{P1} = \left(L - \frac{b}{2}\right)\frac{D_P}{2L} \tag{3.11}$$

$$R_m = \left(33.28 - \frac{11.3}{2}\right)\sin 15.7 = 7.48\text{mm}$$

Now the radial force (W_R) acting at the mean radius may be further resolved into two components, W_{RH} and W_{RV}, in the axial and radial directions as shown in Fig. 3.9. Therefore the axial force acting on the pinion shaft,

$$W_{RH} = W_R \times \sin\theta_{P1} = W_T \times \tan\varphi \times \sin\theta_{P1} \tag{3.12}$$

$$W_{RH} = 883\text{N} \times \sin 15.7$$

$$= 238.94\text{N}$$

And the radial force acting on the pinion shaft,

$$W_{RV} = W_R \times \cos\theta_{P1} = W_T \times \tan\varphi \times \cos\theta_{P1} \tag{3.13}$$

$$W_{RV} = 883\cos 15.7 = 850\text{N}$$

A little consideration will show that the axial force on the pinion shaft is equal to the radial force on the gear shaft but their directions are opposite. Similarly, the radial force on the pinion shaft is equal to the axial force on the gear shaft, but act in opposite directions.

3.2.6 Design of a Shaft for Bevel Gears

In designing a pinion shaft, the following procedure may be adopted [28]:

- ✓ Revolution of the pinion gear on the rack:

$$\text{rev} = \frac{\text{distance of pinion travel}}{circumfranceofpinion}$$

$$= \frac{25\text{mm/m}}{\pi \times 23mm}$$

$$= 0.346\text{m}^{-1}$$

$$= 0.346/\ 0.0005$$

$$N_P = 692\text{rpm}$$

- ✓ Since, the pinion, bevel gear are connected on the shaft:

$$N_P = N_b = N_{sh}$$

Where, N_P =revolution of the pinion gear

N_b = revolution of the bevel gear

- ✓ Torque produced on the pinion :

$$T = F.r_p \tag{3.14}$$

$$= 18543N \times 0.0115m$$

$$= 213.24Nm$$

- ✓ Power produced by pinion gear:

$$T = \frac{p \times 60}{2\pi N} \tag{3.15}$$

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

$$= \frac{213.24 \times 2\pi \times 692}{60} = 15.5 \text{ kW}$$

- ✓ Torque of shaft:

$$T = F.\ r_{sh}$$

$$= 18543N \times 0.004m$$

$$= 74.17Nm$$

- ✓ Power of shaft:

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

$$= \frac{74.17 \times 2\pi \times 692}{60}$$

$$= 5.4 \text{ kW}$$

- ✓ Torque of large bevel gear:

$$T_b = F.\ r_b$$

$$= 18543N \times 0.033m$$

$$= 612Nm$$

- ✓ Power of bevel gear :

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

$$P = \frac{612 \times 2\pi \times 692}{60}$$

$$= 44.4 \text{ kW}$$

✓ Speed small bevel gear:

$$V.R = \frac{T_{Lb}}{T_{Sb}}$$

$$= \frac{32}{9}$$

$$= 3.56$$

$$N_{Sb} = V.R \times N_{lb}$$

$$= 3.56 \times 692 = 2460.4 \text{rpm}$$

✓ Power of small bevel gear

$$P_{sb} = \eta lb \times p_{lb}$$

$$= .98 \times 44.\ 4 \text{ kW}$$

$$= 43.12 \text{ kW}$$

✓ Stress of small bevel gear:

$$\sigma_a = \frac{F_{damp}}{A_{pinion}}$$

$$\sigma_a = \frac{18543\text{N}}{45.2\text{mm}^2}$$

$$= 410.2 \text{ Mpa}$$

The shafts may be designed based on

Strength, and

Rigidity and stiffness.

Designing shafts based on strength

a) Shafts subjected to twisting moment or torque only,
b) Shafts subjected to bending moment only,
c) Shafts subjected to combined twisting and bending moments, and
d) Shafts subjected to axial loads in addition to combined torsional and bending loads.
e) The material used for ordinary shafts is carbon steel of grades 40C8, 45C8, 50C4 and 50 C12.

Table 3.3: Mechanical properties of steels used for shafts[31]

Indian standard Designation	Ultimate tensile strength, MPa	Yield strength, Mpa	Remark
40C8	560-670	320	Used for shafts, gears stud bolts, spdle,washer etc.
45C8	610-700	350	Used for lead screws, feed rods bigger gears, bigger shafts etc
50C4	640-760	370	
50 C12	700 Minimum	390	

From this mechanical properties of steels used for shaft, we choose 40 materials for our project by considering ultimate tensile strength of 582.5MPa

The stress included in the shaft is:

- ✓ Shear stress due to transition of torque(due to torsional load)
- ✓ Bending stress (compression n) due to force acting up on the shaft.
- ✓ Stress due to combined torsional and bending loads

1. First, find the torque acting on the pinion. It is given by

$$T = \frac{P \times 60}{2\pi N_P} \text{Nm}$$

$$T = \frac{43120 \times 60}{2\pi \times 2460.4} \text{Nm}$$

$$= 167.3\text{Nm}$$

Where, P = Power transmitted in watts, and

N_P = Speed of the pinion in r.p.m.

2. Find the tangential force (W_T) acting at the mean radius (R_m) of the pinion. We know that

$$R_m = \left(L - \frac{b}{2}\right)\sin\theta_{P1}$$

$$R_m = \left(33.28 - \frac{11.3}{2}\right)\sin 15.7$$

$$= 7.46\text{mm}$$

$W_T = T / R_m$

$W_T = \frac{167.3 \text{ Nm}}{0.00746 \text{ m}} = 22426.3N$

3. Now find the axial and radial forces (i.e. W_{RH} and W_{RV}) acting on the pinion shaft.

$$W_{RH} = W_T \times tan\varphi \times sin\theta_{p1} \quad (3.16)$$

$$W_{RH} = 22426.3 \times tan\, 20° \times sin\, 15.7$$

$$W_{RH} = 2208.8N$$

$$W_{RV} = W_T \times tan\varphi \times cos\theta_{p1} \quad (3.17)$$

$$W_{RV} = 4519.23 \times tan\, 20° \times cos\, 15.7$$

$$W_{RV} = 7857.9N$$

4. Find resultant bending moment on the pinion shaft as follows: The bending moment due to W_{RH} and W_{RV} is given by:

$$M_1 = W_{RV} \times \text{Overhang} - W_{RH} \times R_m \quad (3.18)$$

$$M_1 = 7857.9N \times 0.017 - 2208.8N \times 0.00746m$$

$$M_1 = 117.1Nm$$

And bending moment due to W_T,

$$M_2 = W_T \times \text{Overhang} \quad (3.19)$$

$$M_2 = 22426.3N \times 0.017m$$

$$= 381.25Nm$$

∴ Resultant bending moment

$$M = \sqrt{(M_1)^2 + (M_2)^2}$$

$$M = \sqrt{(117.1)^2 + (381.25)^2}$$

$$M = 398.83Nm$$

5. Since the shaft is subjected to twisting moment (T) and resultant bending moment (M), therefore equivalent twisting moment,

$$T = W_T \times D_G/2 \quad (3.20)$$

$T = 22426.3N \times 64/2$

$= 717.64Nm$

$T_e = \sqrt{M^2 + T^2}$ (3.21)

$T_e = \sqrt{(398.83)^2 + (717.64)^2}$

$T_e = 821Nm$

Now the shear stress of the pinion shaft may be obtained by using the torsion equation. We know that

$$T_e = \frac{\pi}{16} \times \tau(d_p)^3 \quad (3.21)$$

$$\frac{\pi}{16} \times \tau(d_p)^3 = 821\ N.m$$

$$\tau = \frac{821 \times 16}{\pi(d_p)^3}$$

$$\tau = \frac{821 \times 16}{\pi(0.018)^3}$$

$=582.5Mpa$

Where, d_P = Diameter of the pinion shaft, and

τ = Shear stress for the material of the pinion shaft.

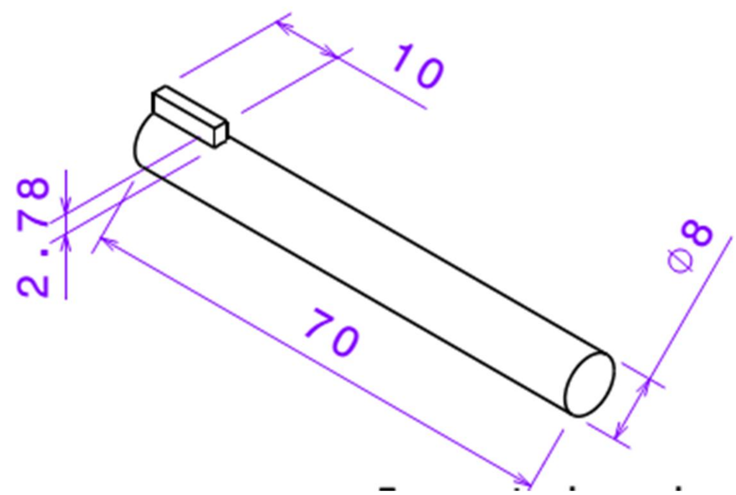


Figure 3.10: Shaft dimension

✓ **Shear force diagram (SFD) & bending moment diagram (BMD)**

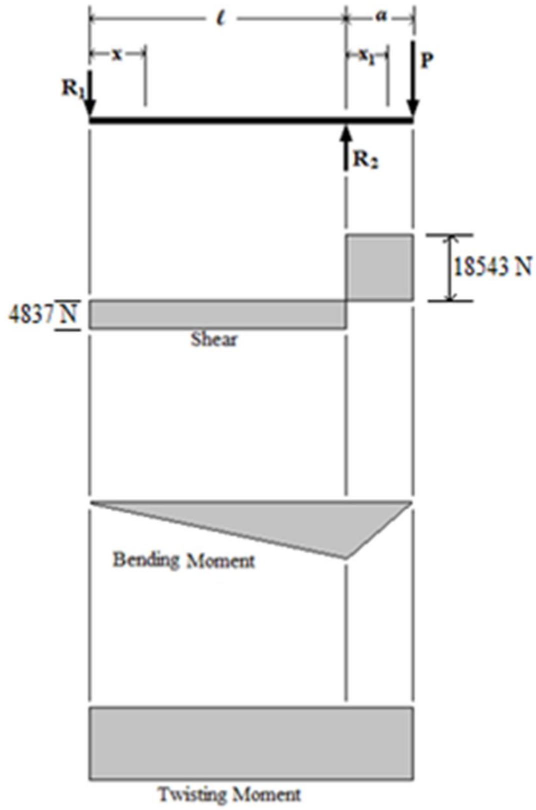


Figure 3.11: Shear Force, Bending and Twisting Moment Diagrams

a) Support Reactions

$+\uparrow \sum F_y = 0$

Where, p = 18,543 N

ℓ = 0.046 m

a = 0.012 m

$- R_1 + R_2 - 18{,}543\ \text{N} = 0$ (3.22)

$+ \circlearrowright M_1 = 0$

$R_2 \times \ell - P \times (a+\ell) = 0$

$R_2 = P \times (a+\ell) / \ell$

$R_2 = 18{,}543\ \text{N} \times (0.012 + 0.046)\ \text{m} / 0.046\ \text{m}$

$R_2 = 23{,}380\ \text{N}$

From eq. (3.22):

$R_1 = R_2 - 18543\ \text{N}$

$R_1 = 23{,}380\ \text{N} - 18{,}543\text{N}$

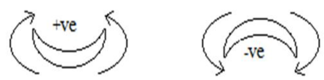
$R_1 = 4837\ \text{N} \downarrow$

M_{max} (at R_2) = P × a
= 18543 N × 0.012 m
= 222.5 Nm

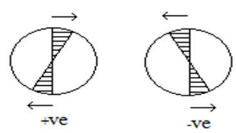
T_{max} = P × r
= 18543 N × 0.0115 m
= 213.2 Nm
Where, r is pinion gear radius = 0.0115 m

b) Shear Force

c) Bending Moment



d) Twisting Moment



- ✓ **Stress on the shaft:**

$\sigma_1 = \frac{P}{A} = \frac{18543\ N}{0.00005 m^2}$ = 368.9 Mpa

$\sigma_2 = \frac{P}{A} = \frac{18543N}{0.0001 m^2}$ = 178.5Mpa

Where, A is area of the shaft $= \pi \frac{D^2}{4}$ = 0.0001 m^2.

Based on the Maximum Shear Stress Theory (for ductile materials such as carbon steel, Mild Steel)

$$\tau_{\max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{1}{2}\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \leq \tau_{all} = \frac{\tau_{yl}}{FS} = \frac{\sigma_{yl}}{(2 \times FS)} \tag{3.23}$$

$$\frac{368.9 - 178.5}{2} = \frac{1}{2}\frac{\sqrt{368^2 - 178.5^2}}{2} + 0 = \frac{\tau_{yl}}{2 \times 1.5} = 95.2 = 285.6 \text{ Mpa}$$

Since the shear stress is within the limit, the design is safe.

Selection of Materials for Gears

The gear material should have the following properties:

- ✓ High tensile strength to prevent failure against static loads
- ✓ High endurance strength to withstand dynamic loads
- ✓ Low coefficient of friction
- ✓ Good manufacturability

Generally cast iron, steel, brass and bronze are preferred for manufacturing metallic gears with cut teeth. Where smooth action is not important, cast iron gears with cut teeth may be employed. Commercially cut gears have a pitch line velocity of about 5 m/s. For velocities larger than this, gear sets with non-metallic pinions as one member are used to eliminate vibration and noise. Non-metallic materials are made of various materials such as treated cotton pressed and molded at high-pressure, synthetic resins of the phenol type and rawhide. Moisture affects rawhide pinions. Gears made of phenolic resins are self-supporting on the other hand other two types are supported by metal side plates at both ends of the plate. Large wheels are made with fretting rings to save alloy steels. Wheel center is commonly cast from cast iron. The ring is forged or roll expanded from steel of the respective grade specified by the tooth design.

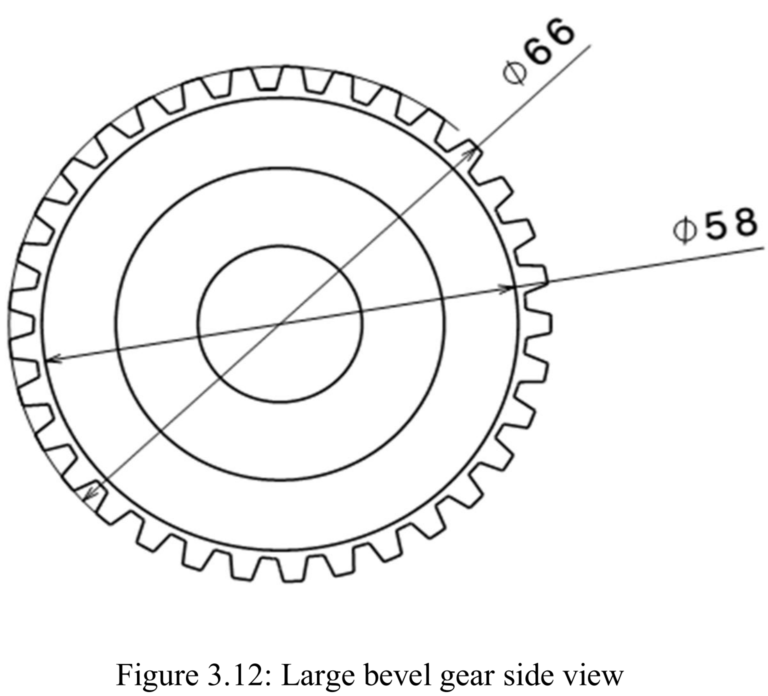


Figure 3.12: Large bevel gear side view

Figure 3.13: Front View of large bevel gear

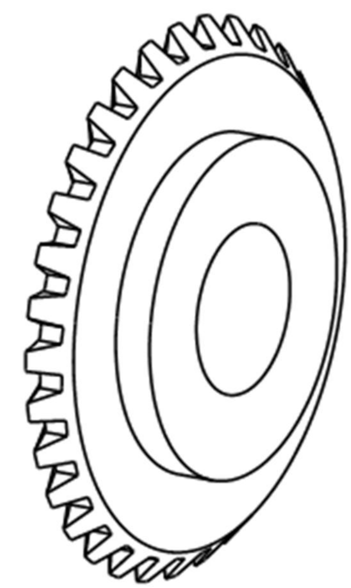


Figure 3.14: 3D large bevel gear

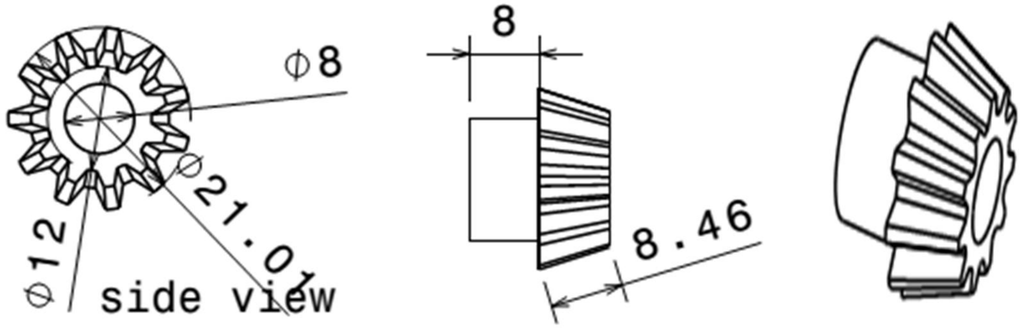
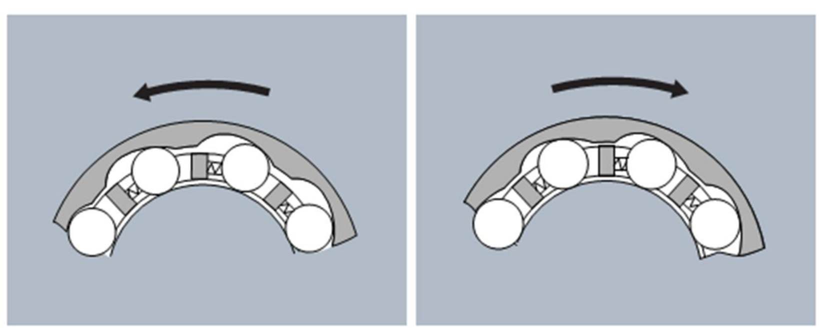


Figure 3.15: Small bevel gear

Roller Clutch (One-Way Clutch)

This is a compact and roller type one-way clutch, which formed a cam face on its outer ring. (Available shaft diameter range: 6 to 35 mm) When the outer ring begins to turn in the counterclockwise direction (direction marked on the outer ring width surface) relative to the shaft, the force of spring causes the rollers to advance to the engagement positions on the outer ring cam face, thereby the wedge action taking place between the outer ring cam face and the shaft drives the shaft. Fig.(a) when the outer ring rotates clockwise against the shaft, the shaft rotates counterclockwise relatively to the outer ring and, as the result, the rollers get away from the outer ring cam face and simultaneously the outer ring idles against the shaft Fig (b)



a b

Figure 3.16: One- way clutches

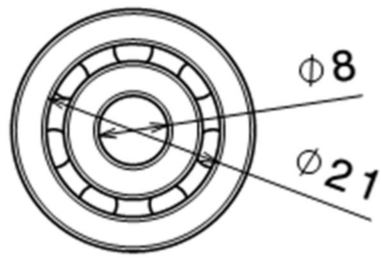


Figure 3.17: Front view of one way clutch

Figure 3.18: 3D one –way clutch

D.C. Generator

In electricity generation, an electric generator is a device that converts mechanical energy to electrical energy. A generator forces electric charge (usually carried by electrons) to flow through an external electrical circuit. It is analogous to a water pump, which causes water to flow (but does not create water). The source of mechanical energy may be a reciprocating or turbine steam engine, water falling through a turbine or waterwheel, an internal combustion engine, a wind turbine, a hand crank compressed air or any other source of mechanical energy in this project we select the 6V 3Wgenerator.

3.2.7 Basic Damping Force Equations

The compression force generated by the compression piston is proportional to the area of the rod. This is because the volume of fluid that passes through the compression piston is equal to the area of the rod times the stroke. The damping force equations for the Twin Tube with Compression Piston are show below in Figure 3.19.

Twin Tube with compression piston Damping

force calculations

$$A_{rebound} = A_{tube} - A_{rod}$$

$$= \frac{\pi(50mm)^2}{4} - \frac{\pi(16mm)^2}{4}$$

$= 1762.4mm^2$

$$F_{Rebound} = (P_1 - P_2) \times A_{rebound}$$

$$F_{compression\ Piston} = (P_2 - P_1) \times A_{compresion\ Piston}$$

The dimension on the fig 3.19: is measured from Sino truck shock absorber

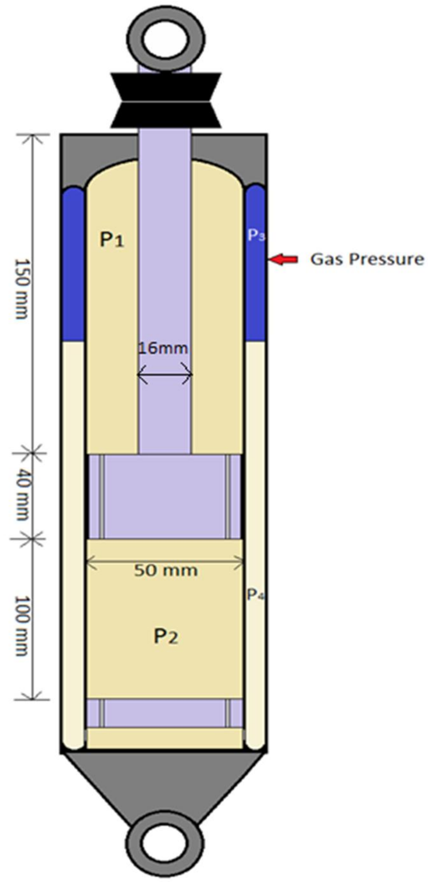


Figure 3.19:Twin tube damper

From boyle'slaw P×V is constant

$$p_1 v_1 = p_2 v_2 \tag{3.24}$$

$$P_{1p2} \times V_{1P2} = P_{2P2} \times V_{2P2} \tag{3.25}$$

Where, P_{1p2} is the pressure in the compression chamber (P_2) during fully rebound position.

V_{1P2} is the volume in the compression chamber (P_2) during fully rebound position.

$V_{1P2} = A_2 \times h_1$

$$= \frac{\pi(50mm)^2}{4} \times 100mm$$

$$= 196349.5mm^3$$

$V_{2P2} = A_2 \times h_2$

$= \frac{\pi d^2}{4} \times h_2$

$= \frac{\pi (50mm)^2}{4} \times 60\text{mm}$

$= 117809.7\ \text{mm}^3$

$P_{2\,p2} = \frac{p1p2 \times v1p2}{v2p2}$

Pressure range 0-20 Mpa [2]

We take average pressure p_{1p2} 10 Mpa

$= \frac{10\text{N/mm}^2 \times 196349.5\text{mm}^3}{117809.7\text{mm}^3}$

$P_2 = 16.67\ \text{N/mm}^2$

$$P_{1p1} \times V_{1P1} = P_{2P1} \times V_{2P1} \tag{3.26}$$

Where, P_{1p1} is the pressure in the rebound chamber (P_1) during fully rebound position.

$V_{1P1} = A_1 \times h_1$

Where, V_{1P1} is the volume in the rebound chamber (P_1) during fully rebound position.

$A_1 = A_{Piston} - A_{rod}$

$= \frac{\pi(50\text{mm})^2}{4} - \frac{\pi(16\text{mm})^2}{4}$

$= 1762.4\text{mm}^2$

$V_{1P1} = 1762.4\text{mm}^2 \times 150\text{mm}$

$= 264365\text{mm}^3$

$V_{2P2} = A_1 \times h_2$

$= 1762.4\text{mm}^2 \times 190\text{mm}$

$= 334862.4\text{mm}^3$

$p_{2p1} = \frac{p1p1 \times v1p1}{v2p1}$

$= \frac{10N/mm^2 \times 264365\ mm^3}{334862.4\ mm^3}$

$P_1 = 7.89 N/mm^2$

$$A_{compressionPiston} = A_{tube} - A_{rod}$$

$F_{compression\ Piston} = (16.67 N/mm2 - 7.89_{N/mm2}) \times 1762.4\ mm2$

$= 15473.8N \approx 15.5kN$

$A_{compressionRod} = A_{rod}$

$= \frac{\pi(16mm)^2}{4}$

$=201mm2$

$$F_{Comp-rod} = (P_2 - P_3) \times A_{rod}$$

where P_3 is gas pressure $12 - 16$ bar [33]:

$$F_{Comp-rod} = (16.67 - 1.4) \times 201$$

$$= 3069.3N$$

$$F_{compression} = F_{comp-Piston} + F_{Comp-rod}$$

$F_{compression} = 15473.8 + 3069.3 = 18543N \approx 18.5\ kN$

Where, $P_1 = ReboundPressure$

$$P_2 = \text{Compression Pressure}$$

$$P_3 = \text{Gas Pressure}$$

3.2.8 Estimation of the energy available from damper

Estimated energy Calculation from vertical displacement [28]:

Mean values

- ✓ mass of the vehicle m = 28000kg
- ✓ Weight vehicle 28,000 kg ×$9.81 m/s^2$ (Sino-truck)

- ✓ Assumed to be vertical lift of the vehicle x = 25mm per (m) meter travelled. Therefore, vertical lift per 100km travelled =2500m. so total energy due to vertical movement of vehicle per 100km travel E = (28000×9.81)×2500 =686700000 Nm

Normal city drive is assumed. Thus by assuming that the springs undergo vibrations of magnitudes 25mm at a frequency range of 0.5 Hz - 5 Hz, keeping in mind that work is done in both compressing and expansion of the spring so that energy can be harvested from both of these motions. Based on these assumptions, a two hour 100km drive generates 95.4 kWh of energy [28]

Cost analysis

Economics is one of the most important aspects of any manufacturing operation. In order for the product to be successfully marketed, its cost must be competitive, optimum cost with similar product in the market place.

Availability and cost of raw materials and manufactured components are major concerns in persistent manufacturing. If raw or processed materials or manufactured components are not available in the desired quantities, shapes and dimensions, substitutes and/or additional processing will be required, which can contribute significantly to product cost.

The various cost components are involved in manufacturing a product, manufacturing manager has to identify those factors that can help to minimize them, while maintaining quality. These components are-

- ✓ Raw materials cost.
- ✓ Operation processing cost.
- ✓ Consumable material and standard item cost.
- ✓ Overhead cost and
- ✓ Labor cost.

3.3 Collection of the materials and components required for construction of prototype

3.3.1 Raw material cost

The material and cost refers to those raw materials, which are used to produce each components of regenerative shock absorber with mechanical motion rectifier (MMR).This analysis of cost estimation of the materials carried out based on the data obtained from local markets Purchase documents of the faculty and Experience of market expertise.

Table 3.4: Material and cost

s/no	Types of materials	Specification	Qty	Unit price ETB	Total price	Remark
1	Angle iron	1000mm×2000mm×1.5mm	1	400	450	
2	Shock absorber	Standard	1	450	450	
3	Pinion	30 mm×10mm	1	100	100	
4	Ball Bearing	20 mm × 12 mm	3	50	150	
5	DC generator	6V ,3W bicycle motor	1	80	80	
6	Mild steel for bevel	80dia mm×50 mm	1	666	666	
7	Mild steel for shaft, small bevel, rack and pinion	32 mm×500 mm	1	700	700	
8	One way clutch	13 mm×48 mm×51mm	2	60	120	
9	Shaft	134mm ×8mm ×13mm	1	180	180	
10	Roller	10mm ×4mm	1	50	50	
Total					2946	

3.3.2 Operation Processes cost

Many processes are used to produce parts and shapes, there is usually more than one method of manufacturing a part from a given material all methods have different process cost. However, Selection of a particular manufacturing process depends not only on the shape to be produced but also on a large number of other factors. The type of material and its properties are basic considerations. The design and cost of tooling, the lead-time required to begin production and the effect of work piece materials on tool life are major considerations.

Therefore, the total operational time will be estimated as

1. Rack = 360 min
2. Pinion = 560 min
3. Shaft = 120 min
4. Bevel gears = 1920 min
5. Assembling = 870 min

Total time = 3830 min

If Operation processing cost is 70Birr/ hour and the operational cost will be

70 x 3830min/60 = 4468 .30Birr

3.3.3Consumable material and standard item cost

Materials that are consumed during manufacturing the products beside the raw materials for this MMR are listed in table 3.6

Table 3.5: Consumed materials

s/no	Types of materials	Specification	Qty	Unit price ETB	Total price	Remark
1	Hack-saw blade	Standard	1packet	30	30	
2	Electrode	E-1011 Dia 2.5	1 Packet	160	160	
3	Grinder disc	Dia 180mm	1 Pcs	50	50	
4	Screw	-	30 pcs	2	60	
5	Thinner	Standard size	1L	60	60	
6	Paint	standard size	1L	85	85	
7	Brush	Standard size	1pcs	25	25	
8	Sand paper	Standard size		100	100	
Total					570	

3.3.4 Overhead cost

Overhead cost includes different expenses for the production processes it may be the cost for deprecation of the machine used, house rent, electric consumption, telephone, water, taxation and other costs. It is calculated easily by multiplying the percentage of the sum total raw material and consumable material cost. It depends of the standard or income of the country and varies from 15%to20% of the material cost. So ones the raw material, consumable material and standard items cost are strictly calculated the sum total will be multiplied by the standard percentage.

The raw material cost is; = 2946

The consumable material and standard item cost = 570.00

Sum total of material cost = 3516

Therefore, the overhead cost will be for example: 15% x 3516 = 527.4

Grand total = sum total material cost +15% vat rate = 4043.4 Birr

3.3.5 Labor cost

Labor costs are generally divided into direct and indirect costs**.** The direct labor cost is for labor directly involved in manufacturing the part (productive labor).This cost includes all labor from the time; materials are first handled to the time the product is finished. The direct labor cost is calculated by multiplying the labor rate (hourly wage, including benefits) by the time that the worker spends producing the part and wage of the market.

For example - If the MMR will require 64hours

If the workers wage is 120birr per day and if the working time /day is 8 hour then

64 hour/8 = 8 days

Therefore the labor cost is = 8 days x120 Birr

= 960.00 Birr

3.3.6 Total production cost of MMR

It is clear that the total production cost of the machine is the sum of all the above mentioned cost that is:-

Raw materials cost …………………………………………………………… 2946.00 Birr

Operation processing cost.…………………………………………………………4468.300Birr

Consumable material and standard item cost. ……………………………………570.00 Birr

Overhead cost and……………………………………………………………… 527.505 Birr

Labor cost..…............960.00Birr

Total cost = 9471.70 birr

For market change (contingency) is 5%

= 5% of total production cost

= 0.05 x 9471.70birr

= 473.58 Birr

Total…………………………………………………………………9945.28 birr

3.4 Modeling of the mechanism using CATIA

CATIA an acronym of computer aided three-dimensional interactive application it is a multi-platform computer-aided design (CAD)/computer-aided manufacturing (CAM)/Computer-aided engineering (CAE) software suite developed by the French company Dassult systems.

CATIA delivers the unique ability not only to model any product, but to do so in the context of its real-life behavior: design in the age of experience. Systems architects, engineers, designers and all contributors can define, imagine and shape the connected world.

CATIA, powered by Dassault Systems' 3Dexperience platform, delivers:

A Social design environment built on a single source of truth and accessed through powerful 3D dashboards that drive business intelligence, real-time concurrent design and collaboration across all stakeholders including mobile workers. An instinctive 3D experience, for both experienced and occasional users with world-class 3D modeling and simulation capabilities that optimize the effectiveness of every user. An inclusive product development platform that is easily integrated with existing processes & tools. This enables multiple disciplines to leverage powerful and integrated specialist applications across all phases of the product development process. Commonly referred to as a 3DProduct Lifecycle Management software suite, CATIA supports multiple stages of product development (CAX), including conceptualization, design (CAD), engineering (CAE) and manufacturing (CAM). CATIA facilitates collaborative engineering across disciplines around its 3D experience platform, including surfacing & shape design, electrical fluid & electronics systems design, mechanical engineering and systems engineering.

CATIA enables the creation of 3D parts, from 3D sketches, sheet metal, composites, molded, forged or tooling parts up to the definition of mechanical assemblies. It provides tools to complete product definition, including functional tolerances as well as kinematics definition. CATIA provides a wide range of applications for tooling design, for both generic tooling and mold & die

Systems engineering

The CATIA Systems Engineering solution delivers a unique open and extensible systems engineering development platform that fully integrates the cross-discipline modeling, simulation, verification and business process support needed for developing complex 'cyber-physical' products. It enables organizations to evaluate requests for changes or develop new products or system variants utilizing a unified performance based systems engineering approach. The solution addresses the Model Based Systems Engineering (MBSE) needs of users developing today's smart products and systems and comprises the following elements: Requirements Engineering, Systems Architecture Modeling, Systems Behavior Modeling & Simulation, Configuration Management & Lifecycle Traceability, Automotive Embedded Systems Development (Autosar Builder) and Industrial Automation Systems Development (Control Build).

CATIA uses the open Modelica language in both CATIA Dynamic Behavior Modeling and Dymola, to quickly and easily model and simulate the behavior of complex systems that span multiple engineering disciplines. CATIA & Dymolaare further extended by through the availability of a number of industry and domain specific Modelica libraries that enable user to model and simulate a wide range of complex systems ranging from automotive vehicle dynamics through to aircraft flight dynamics.

CATIA's Design, Engineering and Systems Engineering applications are the heart of industry Solution Experiences from Dassault Systems to address specific industry needs. This revolutionizes the way organizations conceive, develop and realize new products, delivering competitive edge through innovative customer experiences. Computer-aided design (CAD) is defined as the application of computers and graphics software to aid or enhance the product design from conceptualization to documentation. CAD is most commonly associated with the use of an interactive computer graphics system, referred to as a CAD system. Computer-aided design systems are powerful tools and in the mechanical design and geometric modeling of products and components.

There are several good reasons for using a CAD system to support the engineering design function:

- ✓ To increase the productivity
- ✓ To improve the quality of the design
- ✓ To uniform design standards
- ✓ To create a manufacturing data base
- ✓ To eliminate inaccuracies caused by hand-copying of drawings and inconsistency between drawings

Computer-aided manufacturing (CAM) is defined as the effective use computer technology in manufacturing planning and control. CAM is most closely associated with functions in manufacturing engineering, such as process and production planning, machining, scheduling, management, quality control, and numerical control (NC) part programming. Computer-aided design and computer-aided manufacturing are often combined CAD/CAM systems.

This combination allows the transfer of information from the design into the stage of planning for the manufacturing of a product, without the need to reenter the data on part geometry manually. The database developed during CAD is stored; then it is processed further, by CAM, into the necessary data and instructions for operating and controlling production machinery, material handling equipment, and automated testing and inspection for product quality.

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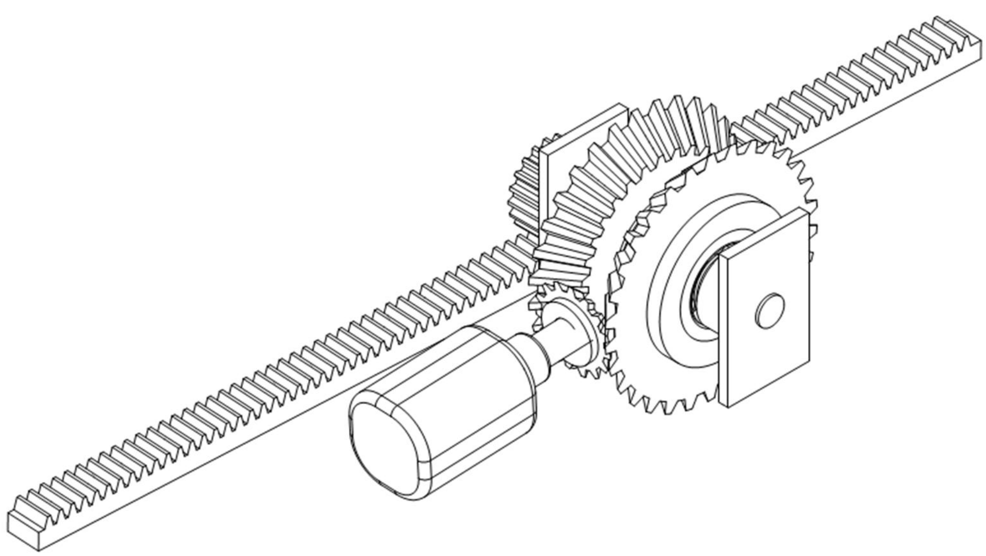

Figure 3.20: 3D Model of MMR

3.5 Manufacturing of the component required.

- ✓ Parts like rack (1), pinion (2), small bevel gear (3) and large bevel gear (4) has manufactured as per design using milling machine. These are shown figure 3.21
- ✓ Main shaft (5) and generator shafts (6) were manufactured on lathe machine using the process like cutting turning and facing.

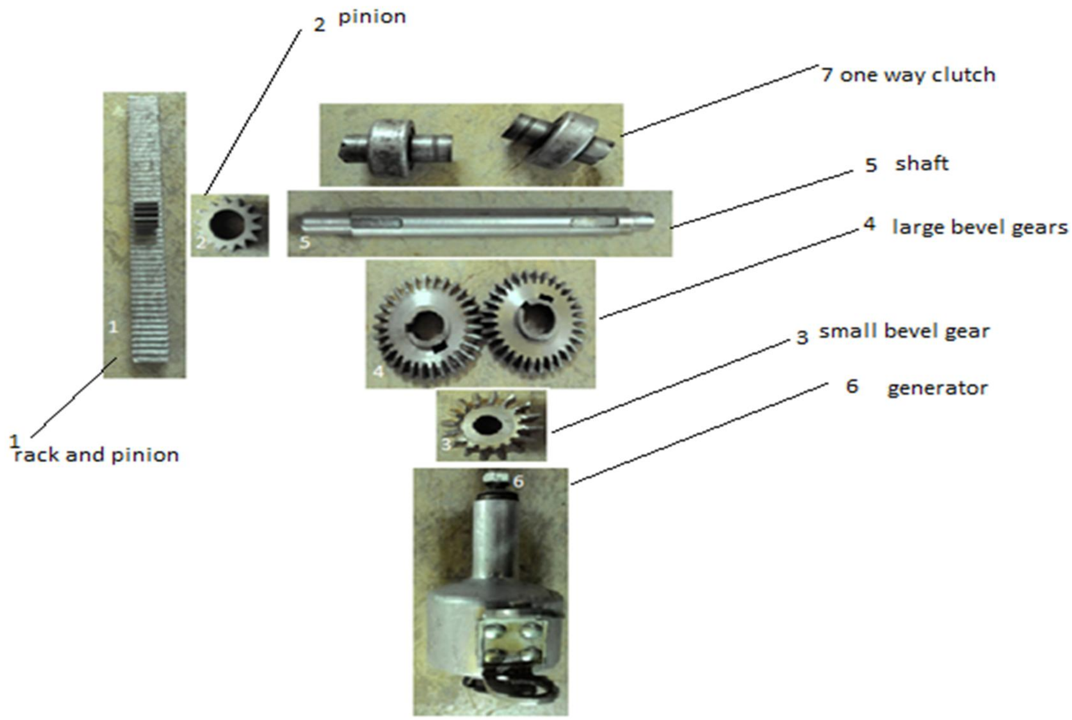

Figure 3. 21: Component Parts of MMR

3.6 Assembling the Components

After all the component parts were manufactured, they were assembled. The steps involved in the procedure of assembling the parts as follows:

- ✓ One-way clutch was joined with bevel gears by welding.
- ✓ The bevel gears and one way clutch combination was fitted on to the shafts carefully by using hydraulic press fitting machine.
- ✓ Small bevel was fitted to the generator shaft.
- ✓ Small bevel gear meshed with large bevel gears.
- ✓ All bearing were fitted to housing by hydraulic press.
- ✓ MMR was mounted on wooden frame.
- ✓ Generator output circuit was prepared.
- ✓ Pinion was inserted at the end of the main shaft and meshed with rack. Roller was inserted on wooden frame for smooth reciprocating motion of the rack.
- ✓ The MMR is ready for demonstration.
- ✓ The completed model is shown in figure 3.22.

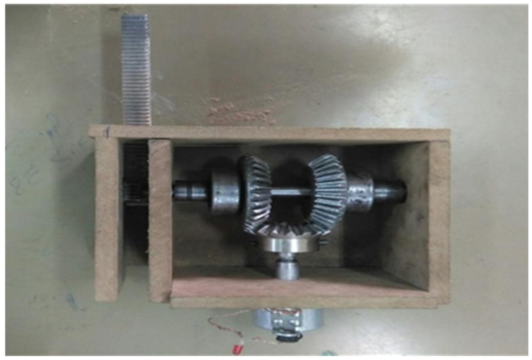

Figure 3.22: Model of MMR

3.7 Demonstrating the functionality

The functionality was demonstrated driving the rack by hand. Its clearly observed that the electricity was generated when a rack is reciprocating. All component in the model were observed to the working in coordination as per expectations.

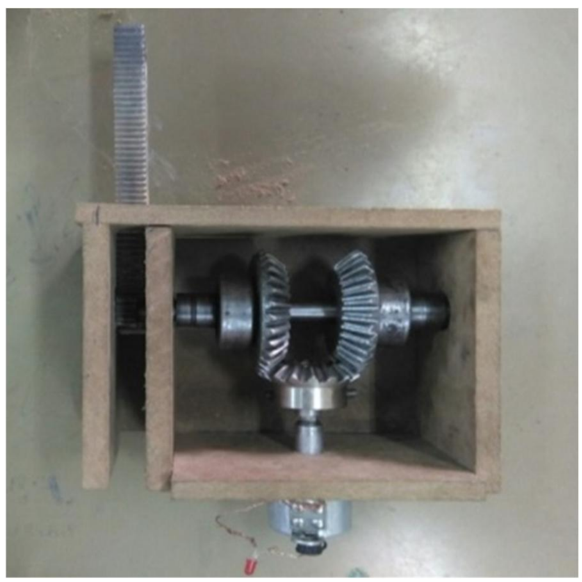

Figure 3.23: Checking functionality of MMR

4. SUMMARY, CONCLUSION AND RECOMMENDATION

4.1 Summary

Suspension systems are used in the vehicle to maintain ride comfort. Shock absorber dissipates the vibrational energy of the vehicle in the form of heat. In this project, a mechanism is designed and constructed to tap this energy which otherwise being wasted. The required components like Rack, pinion, bevel gears, shafts were manufactured. The components were assembled, mounted on wooden frame and demonstrated for its functionality.

4.2 Conclusions

- ✓ The required components for the mechanism were designed. Stress calculation shows that the design is safe with the factor of safety of 1.5
- ✓ The components were manufactured as per the design.
- ✓ The components were assembled and observed that the required motion is obtained.
- ✓ The mechanism was mounted on wooden frame and an arrangement is made to drive the mechanism by hand.
- ✓ The functionality of the mechanism was tested and it is found to be producing electricity when rack is moved testing speed of rack amount of electricity produced.
- ✓ The mechanism mounted on wooden frame observed to be a good teaching aid for teaching and learning process.
- ✓ It works as a foundation for further research in this direction of extracting energy from suspension system.

4.3 Recommendation

- ✓ Due to non-availability of precession manufacturing machine, the mechanism, the size of manufactured parts are large and the required parts is small which fits in the shock absorber, could not be made. So it is recommended to produce the mechanism, which fits in shock absorber as per the design dimensions.
- ✓ It is recommended to the department to establish the required facilities with precession manufacturing tools.

- ✓ It is recommended that the government encourage such innovative ideas for energy conservation and fuel economy.
- ✓ It is recommended that TVET institutes and other learning centers to use the model as teaching aid for imparting the knowledge and skill to the students.

4.4 Future research

There is a much scope to work on related to this project.

1. Further work is necessary in optimization of generative shock absorber. Research has to be done related to the design analysis like, stress analysis, deflection analysis and failure analysis and various operating conditions.
2. Further work is also necessary in matching the design of MMR with the nature of the vehicle shock absorber.
3. Material selection and application of precise manufacturing is also another area to be investigated.

Educational implication

The objective of the national TVET strategy is to create a competent, innovative adaptable work force in Ethiopia contributing to poverty reduction and social &economic development through facilitating demand driven, high quality technical vocational education and training, relevant to all sectors of the economy, at all levels and to all people in need of skills development and transferring new technologies.(national TVET strategy MOE,2006).

Since the industries are the stakeholders of the TVET institutions the progress and drop of industry has an impact on the out-come based training. Intensification of the industries and TVET linkage is very important to provide quality and equipment; the training program will be easy and efficient. Through industrial training, the enterprise can also develop ownership of the TVET system.

Technical and vocational skill is vital to economic development because it is needed for industrial productivity and profitability, as well as national productivity. Without the necessary skills, industry and national growth can be seriously hobbled. Technical and vocational education is of great relevance to the technological and economic status of the nation such as-

- ✓ Technological advance and progression, which naturally lead to economic release and independence
- ✓ Appropriate development and proper positioning of the technological and economic power of a nation through appropriate human resource development
- ✓ Effective mobility of labor through training and re-training
- ✓ Effective handling the issue of transfer of technology and its attendant problems
- ✓ Reduction of over dependence on imported product and conservation of the resources that could have been in engaging the services of foreign experts

This project is valuable for our country in adapting and transferring new technology from foreign country. Since energy generation from suspension system is new in our country its very important in adapting to the TVET. Generally, in this project MMR is the main body and construction of the regenerative shock absorber. This project also summarizing the major courses taken during the master's program, provide the chance to use some special machines, equipment and tools like CATIA and assuring the rightness of educational policy that the government implementing. It also in line with technology adaptation and transfer as required in TVET policy strategies.

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