Design Development and performance evaluation of Hybrid _Exhaust gas operated air brake system for automobiles

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Abstract

In this braking system, exhaust gas from the IC engines is used to operate air brake in the automobiles. Air brake is most used braking system in vehicles. In the proposed model, instead of air brake, exhaust gas is used to operate the brake lever. Exhaust gas from engine is stored in a specially designed pneumatic tank. This exhaust gas pressure is used to operate the pneumatic cylinder and brake lever. The secondary system used will be a thermoelectric generator used to charge a 12 volt battery and the a DC compressor will be used to further pressure rise the exhaust air tank and thus the system will be dual mode operation hence it is termed as hybrid . This study can be extended for diesel engines and petrol engines. The main aim of this project is to reduce the work loads of the engine drive to operate the air compressor. In this project pressurized air from the D.C compressor and from the exhaust will be used to operate the pneumatic brake

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Introduction :

In this braking system, exhaust gas from the IC engines is used to operate air brake in the automobiles. Air brake is most used braking system in vehicles. In the proposed model, instead of air brake, exhaust gas is used to operate the brake lever. Exhaust gas from engine is stored in a specially designed pneumatic tank. This exhaust gas pressure is used to operate the pneumatic cylinder and brake lever. The secondary system used will be a thermoelectric generator used to charge a 12 volt battery and the a DC compressor will be used to further pressure rise the exhaust air tank and thus the system will be dual mode operation hence it is termed as hybrid . This study can be extended for diesel engines and petrol engines. The main aim of this project is to reduce the work loads of the engine drive to operate the air compressor. In this project pressurized air from the D.C compressor and from the exhaust will be used to operate the pneumatic brake**Present day Exhaust brake**

Exhaust brake

An **exhaust brake** is a means of slowing a diesel engine by closing off the exhaust path from the engine, causing the exhaust gases to be compressed in the exhaust manifold, and in the cylinder. Since the exhaust is being compressed, and there is no fuel being applied, the engine works backwards, slowing down the vehicle. The amount of negative torque generated is usually directly proportional to the back pressure of the engine.

An exhaust brake is a device that essentially creates a major restriction in the exhaust system, and creates substantial exhaust back pressure to retard engine speed and offer some supplemental braking. In most cases, an exhaust brake is so effective that it can slow a heavily loaded vehicle on a downgrade without ever applying the vehicle's service brakes. Exhaust brakes are manufactured by many companies. The brakes vary in design, but essentially operate as described above. More advanced exhaust brakes have exhaust pressure modulation (EPM) that controls the back pressure which in turn improves the braking performance across a range of engine speeds.

An exhaust brake is a valve which essentially creates a back-pressure in the exhaust system, which applies enough force onto the engine's pistons to slow the engine. In most cases, an exhaust brake is so effective that it can slow a heavily-loaded vehicle on a downgrade without ever applying the vehicle’s service brakes. Under these conditions, the exhaust flow from the cylinders is bottlenecked and rapidly builds pressure in the exhaust system upstream from the exhaust brake. Depending on engine speed, this pressure can easily reach up to 60 PSI maximum working pressure. Maximum working pressure is limited as part of the design of an exhaust brake. In this example, that same 60 PSI also remains in the cylinder for the entire exhaust stroke (exhaust valve open) and exerts 60 PSI on the piston top to resist its upward movement. The produces a negative torque, slowing the engine for a braking effect. Thus, simply restricting the exhaust flow can generate substantial braking.

Some innovations increase the exhaust back-pressure by various means, leading to more torque at the flywheel, and therefore more braking power. Braking effectiveness is measured in units of power and is about 60 to 80% of the engine's maximum power output. More performance can be achieved by down shifting the vehicle (increasing the leverage, or gear ratio of the engine over the wheels). See also Jake brake.

Legal implications

Compression brakes, a form of engine brakes, produce greater noise pollution than exhaust brakes. For this reason, some vehicle original equipment manufacturers prefer to use exhaust brakes despite their lower braking power. Combining compression braking with exhaust braking can increase effectiveness without being as loud as compression brakes.

Numerous cities, municipalities, states, and provinces ban the use of unmuffled compression brakes.

Most exhaust brakes in the commercial market are protected by patents by companies such as GT Emissions Systems. [1]

Pedal-operated Butterfly Valve

The butterfly valve made up of acrylic material made in the circular shape, having three peripheral circular cuts on it. These circular cuts are given because it is not expected to block whole cross section of the exhaust pipe. If there were no holes, the engine would stop instead of slowing down. In the project, actual engine is not used but a blower; still this type of the valve is used to correlate it with actual arrangement.

The butterfly valve is actuated through hydraulic linkages. This consists of a hydraulic pressure pump which is connected to the butterfly valve. This pump controls the operation of butterfly valve in following manner: When brakes are applied by the driver, the cylinder reduces its pressure so that the valve closes and restricts the path of exhaust gases. In this position the butterfly valve remains perpendicular to the flow of exhaust gases and thus creates back pressure on the engine. The butterfly valve has one to three holes in it so that there is not a complete blockage of the exhaust pipe. This assures the avoidance of damage due to high pressure.

When the brakes are released by the driver, the cylinder generates pressure so that butterfly valve is opened and allows the exhaust gases to flow into the exhaust pipe. In this position the butterfly valve remains parallel to the path of exhaust gases and thus releases the pressure on the engine and allows its speed to increase.

In this project, there is not a cylinder pressure mechanism but a simple linkage. A pedal brake is connected to the elastic rubber so that there will be pressure to apply and release butterfly valve like in an actual arrangement.

Turbocharged Exhaust Brakes

A turbocharger is commonly used to improve power and reduce fuel consumption ratio. Study on effect of turbocharger on exhaust brake for diesel engines had practical significance. When butterfly valve is closed, in the exhaust manifold there is no exhaust emissions so the turbine blades of the turbocharger cannot work, but because turbine blades has the blocking effect, and this effect can make lower inlet pressure of the diesel engine than it with no turbocharger, so air intake into the engine reduces which not only reduces the fuel consumption but also speed of the engine.

Two stage pressure control

Increase in the back pressure with the help butterfly valve can also be achieved by operating in two stages i.e. two valves can be used. This also ensures the efficient working of valve and long its long life by avoiding valve failure. Such type of arrangement is found to be very useful in case of heavy duty trucks. Because to create sufficient back pressure to slow down the engine speed, it requires large force from the valve which is possible by two valves arrangement.

ARIS Actuator Valve

Usually pneumatic or hydraulic actuators are used to operate the butterfly valve. But with the large force of exhaust such type of actuators turns to be inefficient. Hence ARIS type of valve actuator which is widely used in the industries provides effective valve-operating force.

Literature Review :

Literature review :

[1] As per Mr. Prakash T et al (1) , the research work embodied an air brake system based on exhaust gas is called “fabrication of air brake system using engine exhaust gas”. The main aim of this project is to reduce the workloads of the engine drive to operate the air compressor, because here the compressor is not operated by the engine drive. Here we are placing a turbine in the path of exhaust from the engine. The turbine is connected to a dynamo by means of coupling, which is used to generate power. Depending upon the airflow the turbine will start rotating, and then the dynamo will also starts to rotate. A dynamo is a device which is used to convert the kinetic energy into electrical energy. The generated power can be stored in the battery and then this electric power has loaded to the D.C compressor. The air compressor compresses the atmospheric air and it stored in the air tank and the air tank has pressure relief valve to control the pressure in the tank. The air tank supplies the compressed pneumatic power to the pneumatic actuator through solenoid valve to apply brake. The pneumatic actuator is a double acting cylinder which converts hydraulic energy into linear motion.

[2] As per Mr. Li He, Xiaolong Wang et al (2) , Air brake system has been widely used in heavy trucks and intercity buses for its great superiority over other brake system. The practical performance of air brake system may be greatly different from if we analyze it with static theory. Thus, it is necessary to build an integrate air brake system model to simulate the process of brake accurately. However, the dynamic mathematic model of air brake system is very complicate, which makes the model hard to be solved. In this paper, the components of air brake system are decomposed to several basic standard pneumatic components, and then build the system based on these basic standard pneumatic components. The standard pneumatic components which are built in the software MWorks based on Modelica language include cylinder, nozzle, air reservoir, volume, and air pipe. An air brake system which contains brake valve, relay valve, brake chambers and pipelines is made based on the standard pneumatic components. The simulation results show the dynamic characteristics of air brake system.

[3] As per Mr. S.Mithun, et al (3) An air brake system is used in heavy commercial vehicles for the purpose to stop or slow down the vehicle. The effective braking depends mainly on the response time of the entire system. The brake system layout configuration has to be designed in such a way that the response time should meet the vehicle safety standard regulations. This paper describes the detailed modeling of the individual brake system products, right from the actuating valves, control valves, actuators and foundation brakes. Response time prediction for a typical 4X2 Heavy commercial vehicle has been done. Also a study on comparing the transient torque generated by the existing drum brake and an equivalent disc brake model was carried out. The layout was modeled in one of the commercially available multi-domain physical modeling software employing bond graph technique and lumped system

[4] As per Mr. Fanping Bu., et al (4) Precision stopping is an important automated vehicle control function that is critical in applications such as precision bus docking, automated truck or bus fueling, as well as automatic intersection or toll booth stopping. The initial applications of this technology are most likely to be applied to heavy duty vehicles such as buses or trucks. Such applications require specific attention to brake control since the characteristics of a typical pneumatic brake system of a heavy vehicle is inherently nonlinear with large uncertainties. The feasibility of providing a smooth precision stopping brake control based on a conventional pneumatic brake system has not yet been demonstrated. This paper describes the precision stopping problem, verifies the pneumatic brake model, details the Indirect Adaptive Robust Control (IARC) design for a pneumatic brake system, and reports the successful implementation of a bus precision docking demonstration.

2.1 Literature Gap

From careful stud of the literature pertaining to the problem of air brake by application of the compressed air from the exhaust gas has been studied how ever various researchers have studied the problem no research points towards the direct application of the exhaust gas to charge the air tanks.

Thus there is scope and possibility to try and prove the application of exhaust gases to charge the air tank for application in the air brake.

Thus the paper is aimed to proposal of an alternative technology and the system components thereby used.

3. Proposal of exhaust gas operated air brake

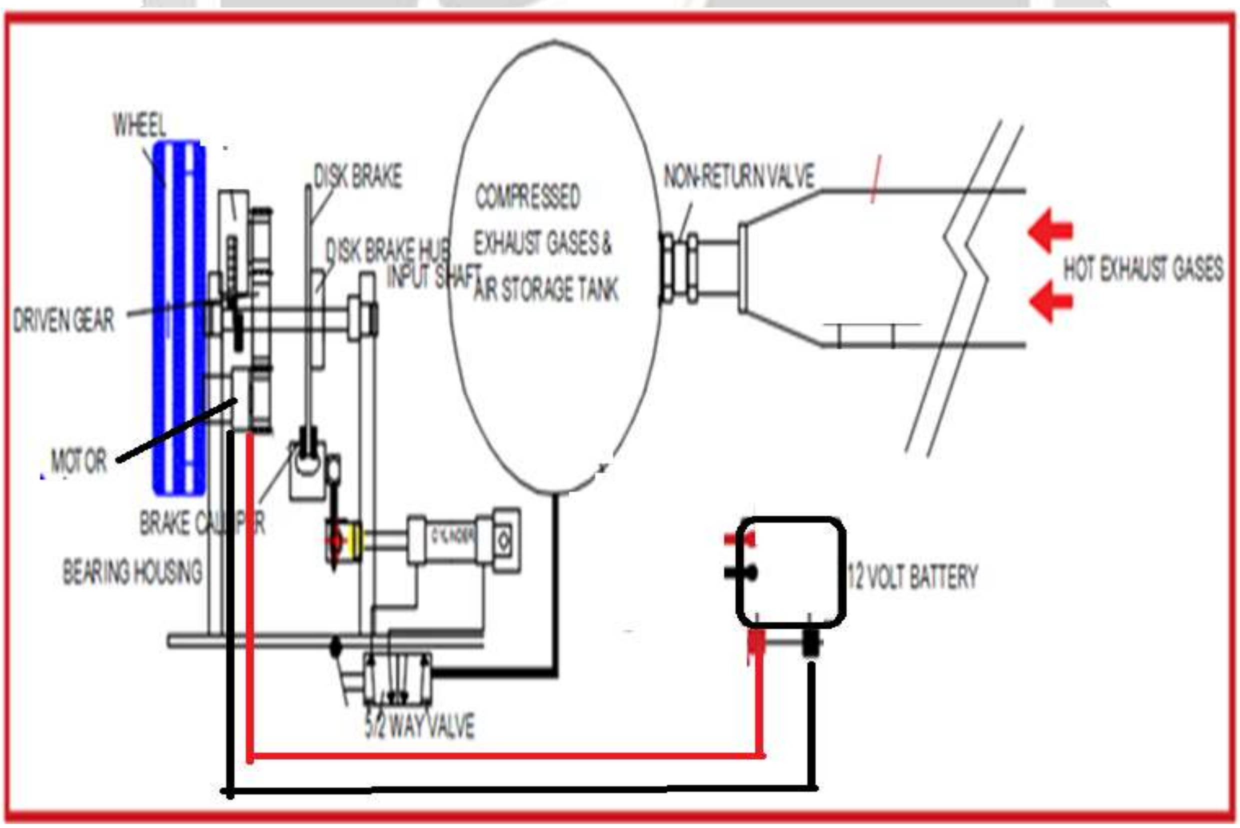
PROBLEM STATEMENT :

The conventional exhaust brake is only suitable for heavy vehicles , where as the pneumatic brakes require to compress air that in turn uses the compressor that runs on the engine power . Thus to run the pneumatic brake we are consuming engine power and more over the energy of the exhaust gases is discharged to the atmosphere which will increase pollution and so also energy is wasted.

SOLUTION :

The proposed solution is to make use of the exhaust gas pressure and compress it further to make pressure energy useful to operate the pneumatic brake.

Further more the heat carried by the exhaust gases can be used to run an Thermo-electric generator module (TEG module) that will convert the heat to useful voltage that can be used to generat electricity that can be stored in the battery which will further run a 12 volt DC compressor that will further compress air and store it in the storage tank for pneumatic braking….



Description of Components :

1. **Air tank :**
2. **Non return valve :**

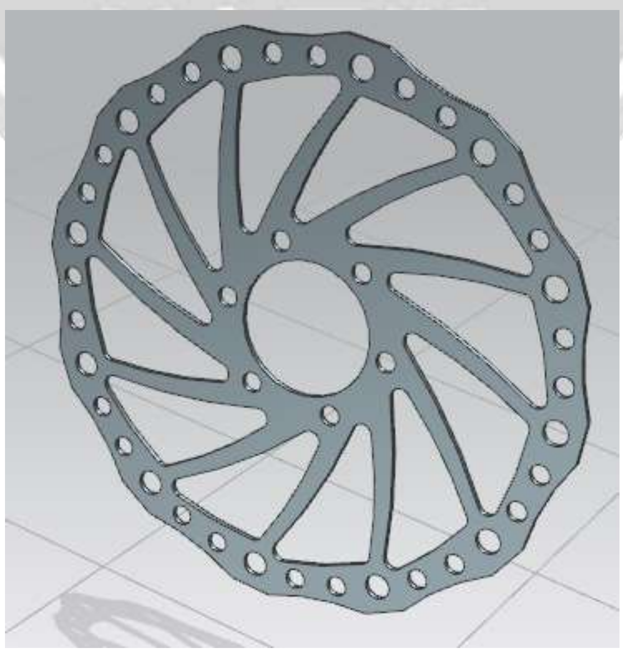
Non return valve ½ inch BSP

3. **Pneumatic Cylinder**

Compact cylinder

DNU-20-15-PPV-A.

4. **Disk Brake :**



60 RPM Side Shaft 37mm Diameter Compact DC Gear Motor

Literature Gap :

After careful review of literature it is clear that a directly exhaust gas operated braking system is not researched by any of the researchers in th hybrid form ie, using the pressure energy as well as the thermal energy of the exhaust gas .

Hence the project is conceptualized to utilize both the pressure as well as the thermal energy of the exhaust gases.

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Methodology :

Proposed Methodology:

THEORETICAL WORK

1. Literature review-

a)Study of various configuration of exhaust gas compression arrangement for storage and electricity generation using various Handbooks, United State Patent documents, Technical papers , etc.

b)Literature gap
c) Problem statement
d) Solution
2. **Design and Development:**
[a] System design as to and theoretical derivation of dimensions of the storage tank using 2-D cad software
[b] System Design and theoretical derivation of dimensions of NRV for appropriate pressure development
[c] System Design and theoretical derivations of devices of Thermoelectric generator as for electricity generation mechanism
[d] Selection of battery electricity storage
[e] Design Development and analysis for brake operation mechanism for pneumatic braking
[f] Design of pneumatic braking.
[g] Design and selection of Pneumatic Valves for system
[h] Design and selection of disk brake.
[i] Design and selection of the gear mechanism

Fabrication :
Suitable manufacturing methods will be employed to fabricate the components and then assemble the test set –up

Experimental analysis :
Testing of the set up to determine
a)Maximum Pressure attained by the exhaust gas
b Maximum Pressure attained by the compressor system
c) Maximum Pressure attained by the hybrid system
d) Brake Efficiency
f) Brake distance Vs Pressure..

PERT CHART AND PROJECT PLANNING

Scheme of implementation /project Completion schedule:

Work Task	**July 16- Aug 16**	**Sept 16- Oct 16**	**Nov 16- Dec 16**	**Jan 16 – Feb 17**
Information Gathering - Literature Survey.				
Finalization of the Aims and Objectives or Title .				
Development Experimental Set Up and Instrumentation or Case Study				
Experimentation / Data Collection				
Formulation of Model / Analysis of Model/ Data				
Critical Analysis of the Formulated Model / Optimization and Sensitivity Analysis				
Initial Report Writing and Publication				
Final Report Writing and Publication				

ORGANIZATION OF DISSERTATION :

1.ABSTRACT

2. INTRODUCTION

3.LITERATURE REVIEW

4. CONSTRUCTION

5. WORKING

6.DESIGN

7.ANALYSIS

8.MANUFACTURING PROCESSES

9.TEST AND TRIAL

10.COST ANALYSIS

11.BIBLIOGRAPHY

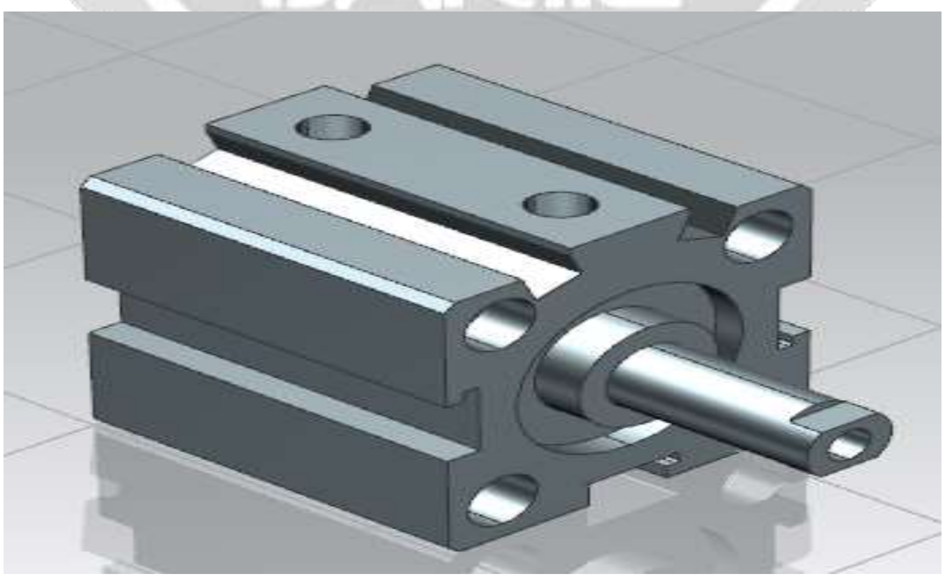
Fabrication :
Suitable manufacturing methods will be employed to fabricate the components and then assemble the test set –up.

Facilities available:-

The following facilities to carry out fabrication work are available at sponsor site

1. Centre lathe
2. Milling machine
3. DRO – Jig Boring machine
4. Electrical Arc Welding
5. Tachometer
6. Multi meter for voltage and Current measurement

DESIGN & SELECTION OF PNEUMATIC CYLINDER :



SELECTION OF PNEUMATIC CYLINDER

Standard cylinder (14167)

DNU-20-15-PPV-A.

Standard cylinders DNU/DNUL, DIN ISO 6431

FESTO

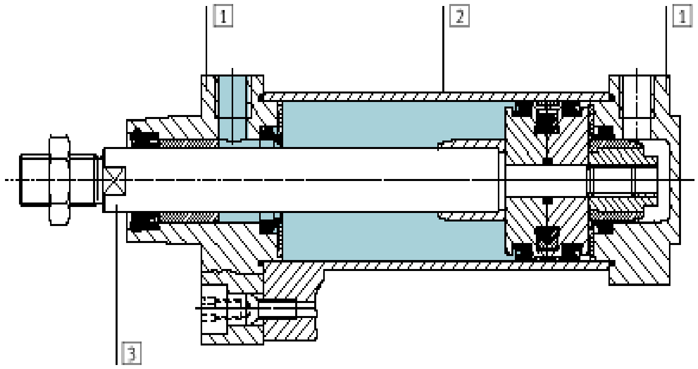
Type code

DNU – 50 – 80 – PPV – A –

Type	
Double-acting	
DNU	Standard cylinder
DNUL	Standard cylinder, non-rotating
Piston ∅ [mm]	
Stroke [mm]	
Cushioning	
PPV	Adjustable at both ends
Sensing	
A	Position sensing
Variants	
S2	Through piston rod
S3	Stainless steel piston rod
S6	Heat resistant seal for temperatures up to 150 °C
S8	High corrosion protection
S26	Through piston rod with heat resistant seals for temperatures up to 150 °C

Materials

Sectional view



Standard cylinder		
1	Bearing and end caps	Aluminium
2	Cylinder barrel	Anodised aluminium
3	Piston rod, tie rod	High-alloy steel
–	Seals	Polyurethane

General technical data							
Piston ∅		32	40	50	63	80	100
Pneumatic connection		G1/8	G1/4	G1/4	G3/8	G3/8	G1/2
Piston rod thread		M10x1.25	M12x1.25	M16x1.5	M16x1.5	M20x1.5	M20x1.5
Operating medium		Filtered compressed air, lubricated or unlubricated					
Design		Piston					
		Piston rod					
		Profile barrel					
Cushioning		Adjustable at both ends					
Cushioning length	[mm]	19	21	23	23	30	30
Position sensing		Via proximity sensor					
Type of mounting		Via accessories					
Mounting position		Any					

Operating and environmental conditions							
Piston ∅		32	40	50	63	80	100
Operating pressure	[bar]	12					
Ambient temperature[1]	[°C]	–20 ... +80					

1) Note operating range of proximity sensors

Forces [N]							
Piston ∅		32	40	50	63	80	100
Theoretical force at 6 bar,		482	753	1,178	1,870	3,015	4,712
advancing	S2	393	586	924	1,601	2,615	4,221
Theoretical force at 6 bar,		415	633	990	1,682	2,720	4,418
retracting	S2	393	586	924	1,601	2,615	4,221

Specifications of Standard cylinder (14167)

DNU-20-15-PPV-A.

Criterion	Feature
Stroke	15
Piston diameter	20
Piston rod thread	M6
Cushioning	Pneumatic cushioning, adjustable at both ends (PPV)
Assembly position	Any
Conforms to standard	ISO 6431
Piston-rod end	Male thread
Design structure	Piston Piston rod
Position detection	With proximity sensor
Variants	Single-ended piston rod
Operating pressure	0.3 - 12 bar
Mode of operation	Double-acting
Operating medium	Dried compressed air, lubricated or un-lubricated
Corrosion resistance classification CRC	2
Ambient temperature	-20 - 80 °C
Authorisation	Germanischer Lloyd
Cushioning length	30 mm
Theoretical force at 3 bar, return stroke	30 N
Theoretical force at 3bar, advance stroke	50N
Additional weight per 10 mm stroke	76 g
Basic weight for 0 mm stroke	2662 g
Mounting type	With accessories
Pneumatic connection	G3/8
Materials information for cover	Aluminium
Materials information for seals	TPE-U(PU)
Materials information for piston rod	High alloy steel
Materials information for cylinder barrel	Materials information for cylinder barrel Anodised

Theoretical force at 3 bar when advancing of piston = 50 N
Piston rod threading end = M6 x 1 pitch

Design of piston rod

Material selection:

Ref :- (PSG – 1.12)

Designation	Tensile Strength N/mm^2	Yield Strength N/mm^2
EN9	600	380

<u>Direct Tensile or Compressive stress due to an axial load</u> :-

$$fc_{act} = \frac{W}{\quad x\ dc^2}$$

$$fc_{act} = \frac{50}{(\pi/4) \times 5^2}$$

$\Rightarrow$ **<u>fc_{act} = 2.54 N/mm^2</u>**

As $fc_{act} < fc_{all}$; Piston rod is safe in compression.

<u>2. Shear stress in threaded end due to axial load</u> :-

$$fs_{act} = \frac{W}{\pi\ n\ dc\ t}$$

t = width th

t= 0.5 mm

n = No of threads in contact = 12/1 =12

$$fs_{act} = \frac{50}{\pi x12x5 \times 0.25}$$

<u>fs_{act} = 0.5</u>

As ; $fs_{act} < fs_{all}$, the screw threads are safe in shear.

<u>Stresses due to buckling of piston rod :-</u>

According to Rankine formula,

Where

$$Wcr = \frac{fc\ \ A}{l + a\,(le/k)^2}$$

Where ; Wcr =

A = Area of c/s at root (mm^2)

A= constant

Le= Equivalent unsupported length of screw (mm)

decided by end conditions.

K= Radius of gyration = dc/4 (mm)

Fc= Yield stress in compression (N/mm^2)

le = 0.707L;as one end of screw are considered to be fixed and other free(Ref . PSG Design Data Pg. No. 6.8)

Here transverse of the piston is 50 mm, total length of piston rod =172 mm

$\Rightarrow$ le = 0.707 x 72 = 50.94 mm

$$Wcr = \frac{300 \times (\pi/4 \times 10^2)}{1+(1/7500)(\ 50.94/\ (10/4))^2}$$

<u>Wcr =23.56x10^3 N</u>

As, The critical load causing buckling is high as compared to actual compressive load of 0.240 kN the piston rod is safe in buckling **.**

Design of the base vehicle that will move the VEHICLE UP AND DOWN HILL

A) Selection of motor

Assuming that the maximum weight of robot with all parts is not to exceed 6 kg the net load on all four wheels = 6 x 9.81 =58.860 approx 59 N

Thus the load on each wheel = 15 N approx.

Assuming that the maximum wheel diameter for the robot is to be 120 mm

Coefficient of friction of rubber tyre on metal = 0.5

Coefficient of friction for a range of material combinations

combination	**Static**		**Dynamic**	
	dry	**lubricated**	**dry**	**lubricated**
steel-steel	0.5...0.6	0.15	0.4...0.6	0.15
copper-steel	-	-	0.5...0.8	0.15
steel-cast iron	0.2	0.1	0.2	0.05
cast iron - cast iron	0.25	0.15	0.2	0.15
friction material - steel	-	-	0.5-0.6	-
steel-ice	0.03	-	0.015	-
steel-wood	0.5-0.6	0.1	0.2-0.5	0.05
wood-wood	0.4-0.6	0.15...0.2	0.2...0.4	0.15
leather-metal	0.6	0.2	0.2...0.25	0.12
rubber-metal	1	-	0.5	
plastic-metal	0.25...0.4	-	0.1...0.3	0.04...0.1
plastic-plastic	0.3-0.4	-	0.2...0.4	0.04...0.1

Rolling resistance

	C_r [-]
Steel wheel on rail	0.0002...0.0010
Car tire on road	0.010...0.035
Car tire energy safe	0.006...0.009
Tube 22mm, 8 bar	0.002
Race tyre 23 mm, 7 bar	0.003
Touring 32 mm, 5 bar	0.005
Tyre with leak protection 37 mm, 5 bar / 3 bar	0.007 / 0.01

The coefficient of friction between two materials in relative sliding may depend on contact pressure, surface roughness of the relative harder contact surface, temperature, sliding velocity and the type of lubricant whether the level of contamination. It's the reason that the data found in the many reference tables available may show a large variation. Motor Torque

Traction torque (Torque required to roll the tyre on pavement) is given by

T= Cr x F x r ---

Where ,

Cr- Coefficient of rolling resistance -= 0.01 for rubber on road

F= μ Rn =0.5 x 15 =7.5 N

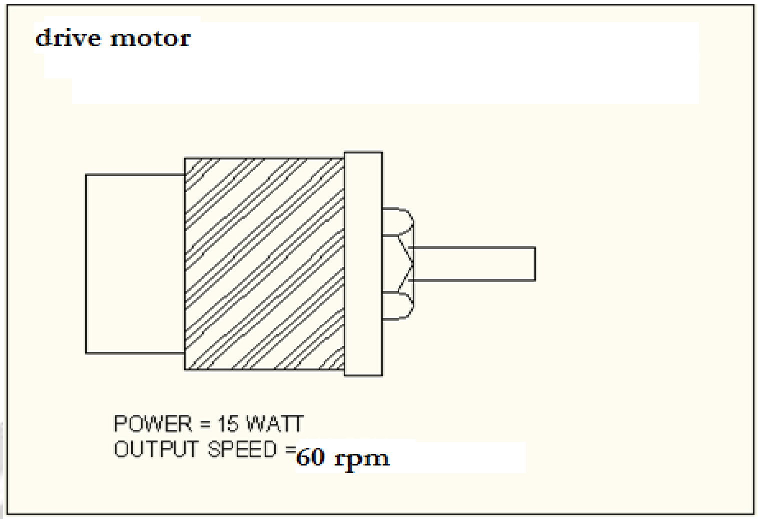
T =(0.035 x 7.5) x(60 /10) x FOS –

Where FOS = Factor of safety = 4 – assuming that only one tyre may be in contact with surface in extreme slip conditions

T= 6.3 N-cm = 0.642 kg-cm

Thus selecting motor of following specifications :

The drive motor is 12 VDC motor coupled to an planetary gear box. Specifications of motor are as follows:Power



‘15 watt ,Speed = 60 rpm

DESIGN OF WHEEL SHAFT

MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.17)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN25	800	680

ASME CODE FOR DESIGN OF SHAFT.

Since the loads on most shafts in connected machinery are not constant , it is necessary to make proper allowance for the harmful effects of load fluctuations

According to ASME code permissible values of shear stress may be calculated form various relation.

$fs_{max} = 0.18\ fult$

$= 0.18 \times 800$

$= 144 N/mm^2$

OR

$fs_{max} = 0.3\ fyt$

$=0.3 \times 620480$

$=840\ N/mm^2$

considering minimum of the above values

$\Rightarrow$ fs_{max} = 144N/mm^2

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow$ fs_{max} =108 N/mm^2

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

Assuming 100 % efficiency of transmission

$\Rightarrow$ T design = 3 KG-CM = 0.294 Nm

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

Assuming minimum section diameter on input shaft = 12 mm ---WHEEL BORE IS ATANDARD 12MM

$\Rightarrow$ d =12mm

$Td = \Pi/16 \times fs_{act} \times d^3$

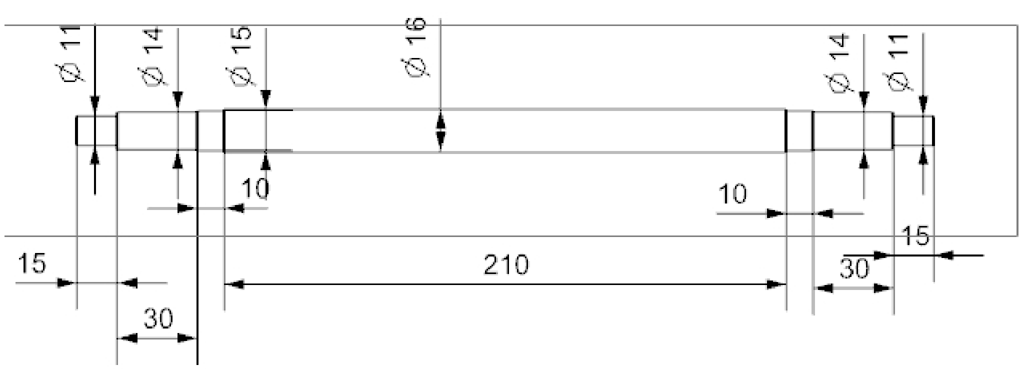
$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times d^3}$$

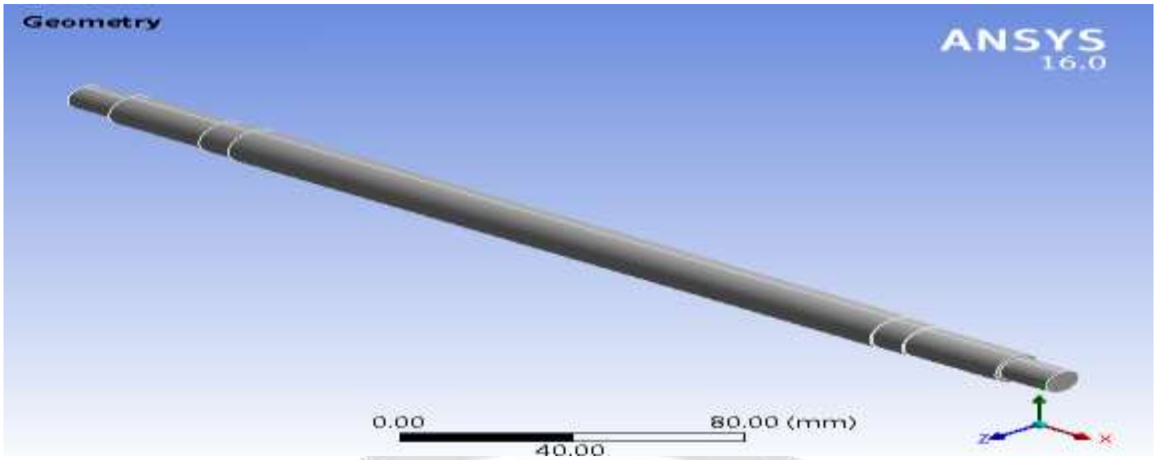
$$= \frac{16 \times 0.294 \times 10^3}{\Pi \times 12^3}$$

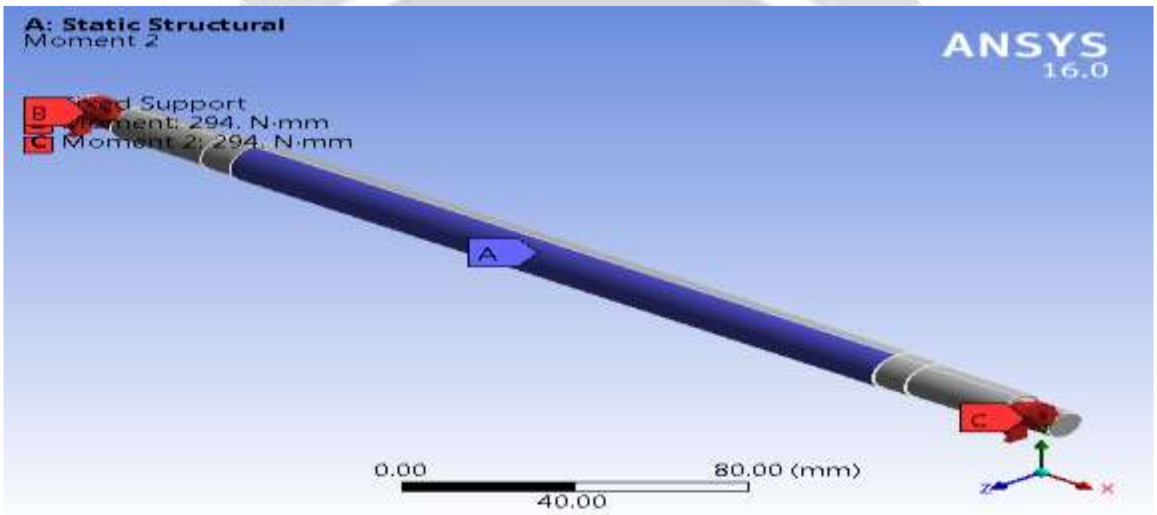
$\Rightarrow$ fs_{act} = 0.867 N/mm^2

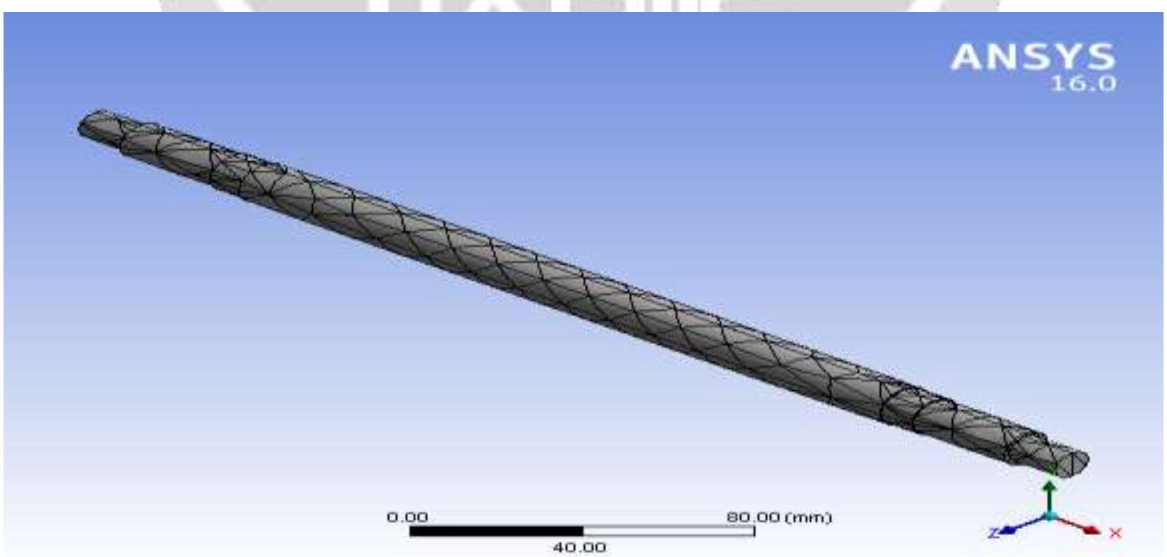
As $fs_{act} < fs_{all}$

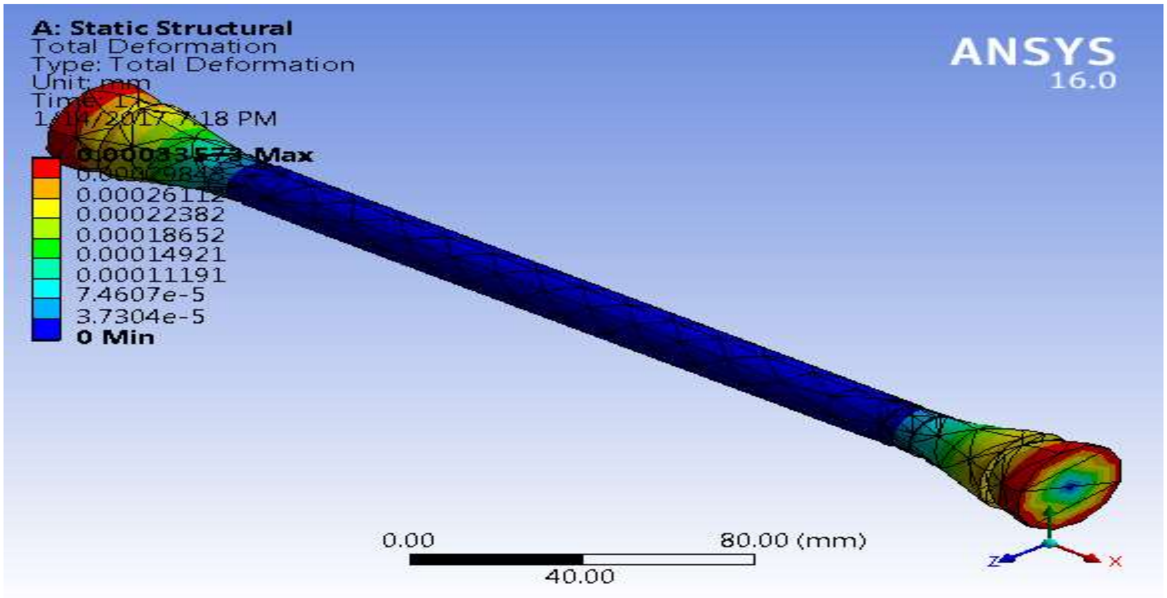
$\Rightarrow$ wHEEL shaft is safe under torsional load.

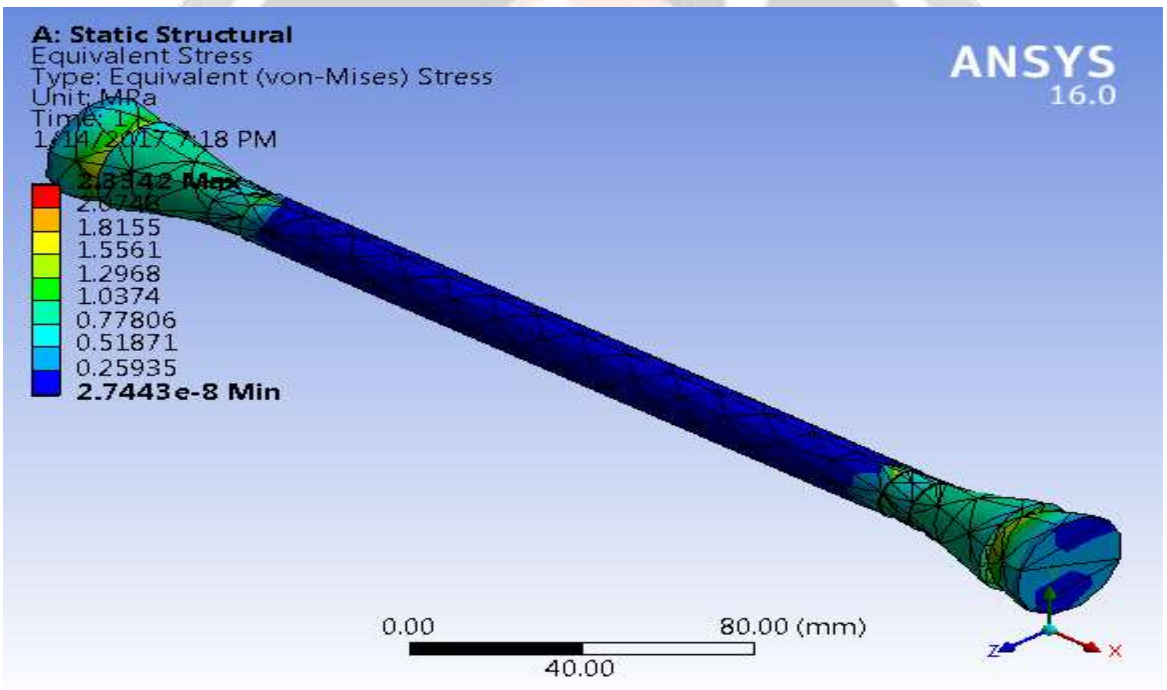



Geometry
ANSYS
16.0
0.00
40.00
80.00 (mm)


A: Static Structural
Moment 2
ANSYS
16.0
Support
294. N·mm
C Moment 2: 294. N·mm
A
B
C
0.00
40.00
80.00 (mm)

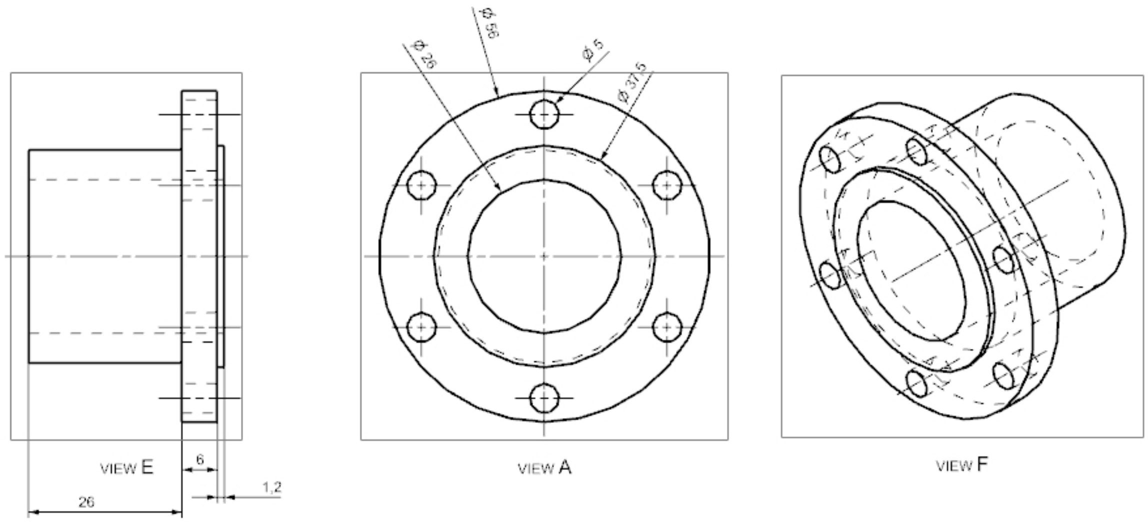

ANSYS
16.0
0.00
40.00
80.00 (mm)

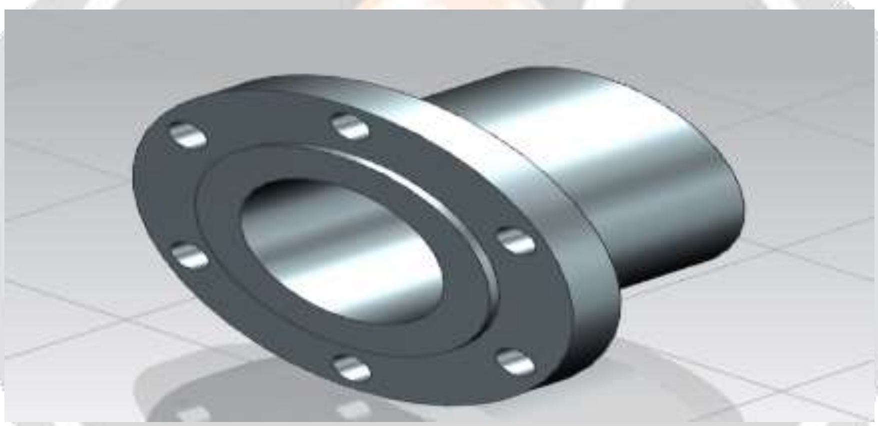




AS EQUIVALENT STRESS IS 2.8 < ALLOWABLE 108 N/MM2 THE SHAFT IS SAFE

DESIGN OF BRAKE HUB :





MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.18)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN8	520	360

⇒ fs_{max} = uts/fos = 520/2 = 260 N/mm^2

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

Assuming 100 % efficiency of transmission

⇒ T design = 0.252 Nm

$Td = \Pi/16 \times fs_{act} \times (D^4 - d^4)/D$

$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times (D^4 - d^4)/D}$$

Outside diameter of drum boss = 37.5 mm

Inside diameter of drum boss = 26mm

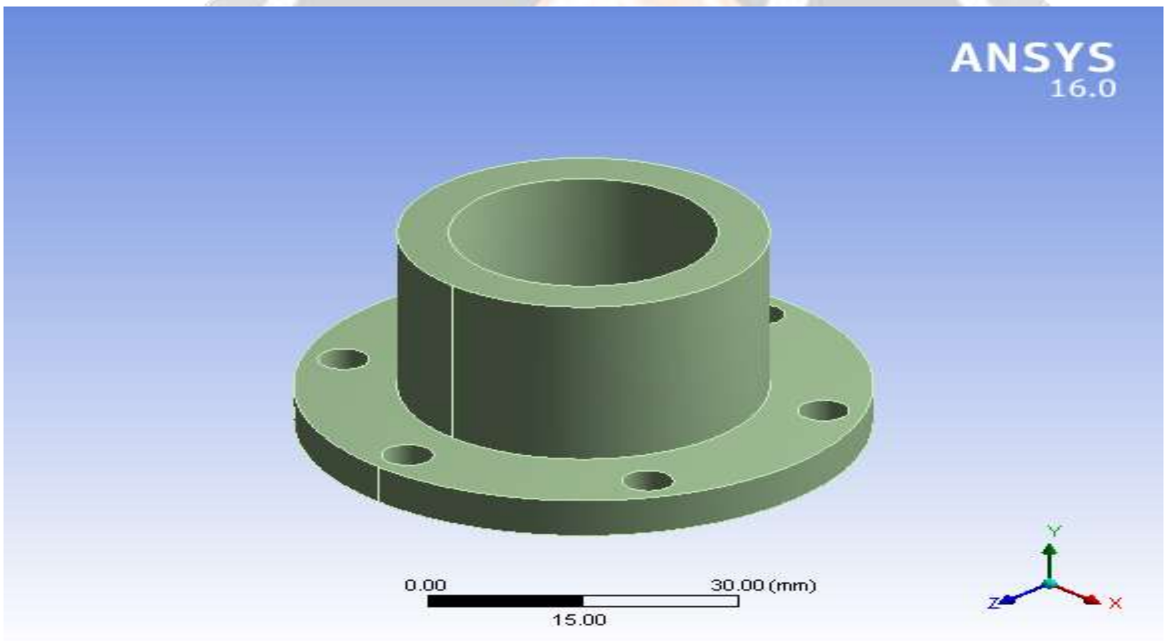
$$= \frac{16 \times 0.252 \times 10^3 \times 37.5}{\Pi \times (37.5^4 - 26^4)}$$

$\Rightarrow fs_{act} = 0.03\ N/mm^2$

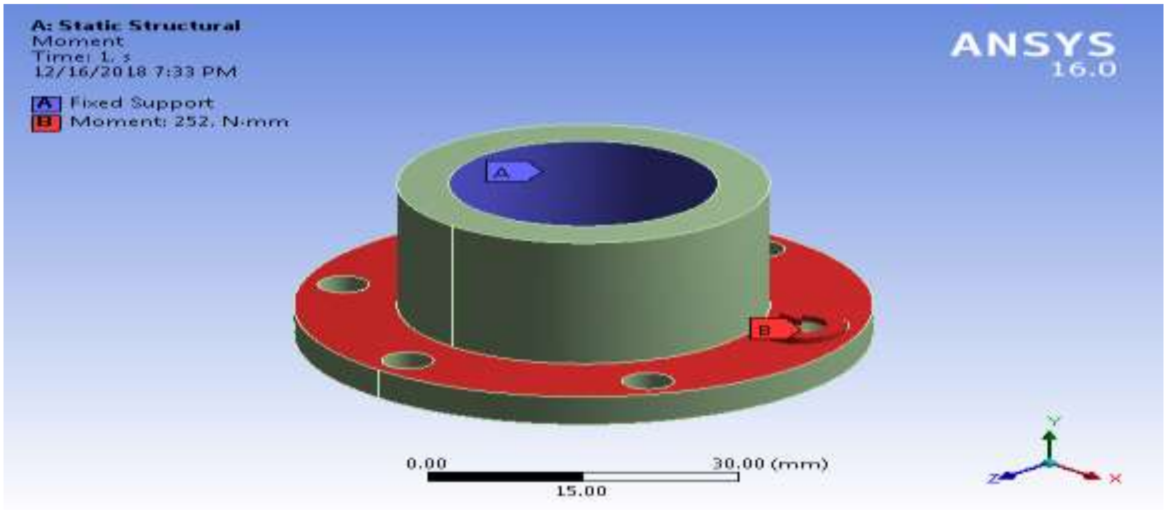
As $fs_{act} < fs_{all}$

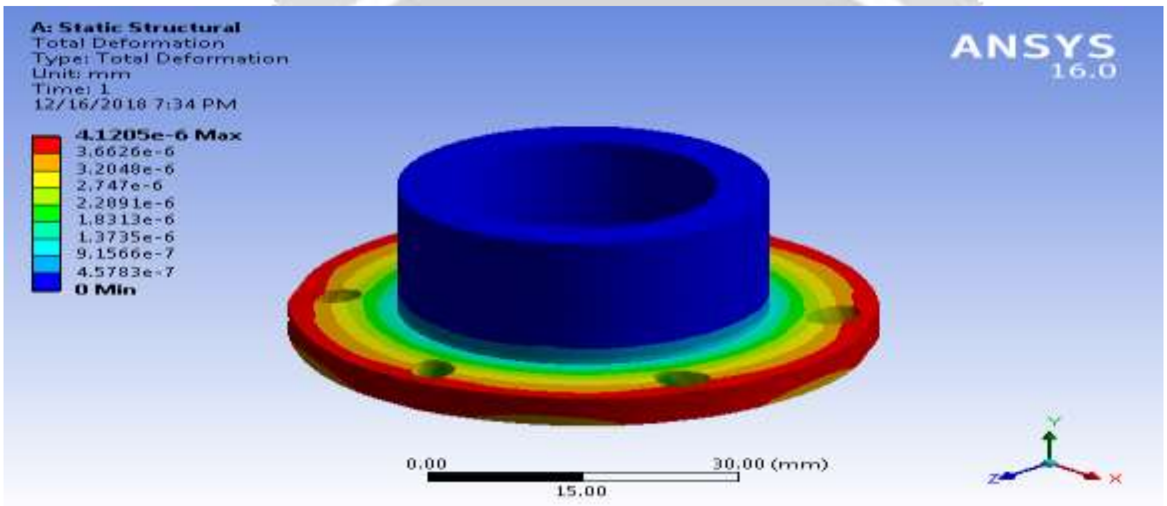
$\Rightarrow$ DISK BRAKE HUB is safe under torsional load.

ANALYSIS OF DISK BRAKE HUB

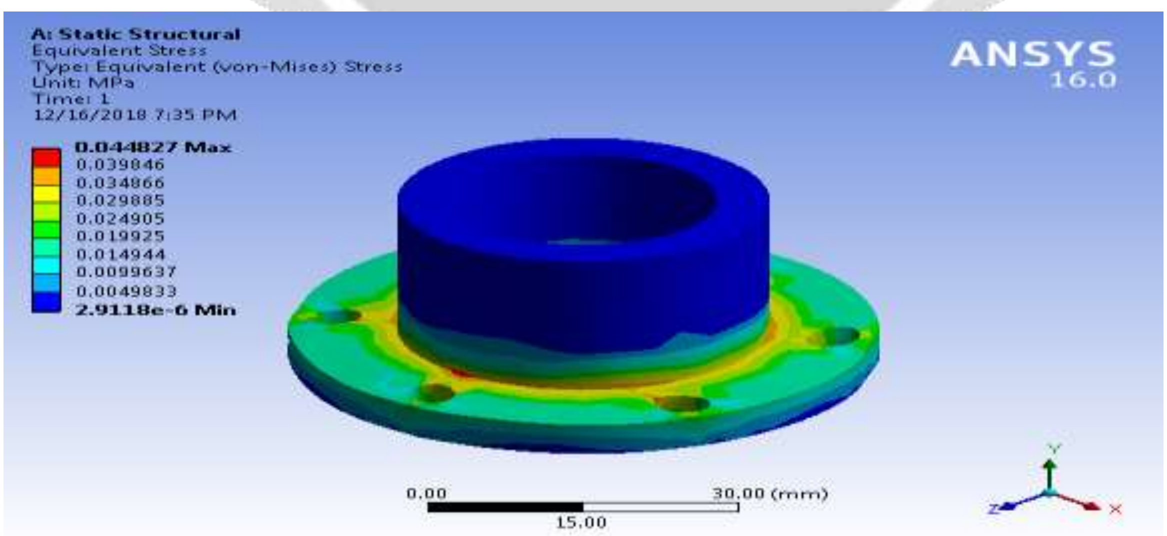


Statistics	
Nodes	3580
Elements	1814
Mesh Metric	None





As the deformation is negligible the hub is safe.

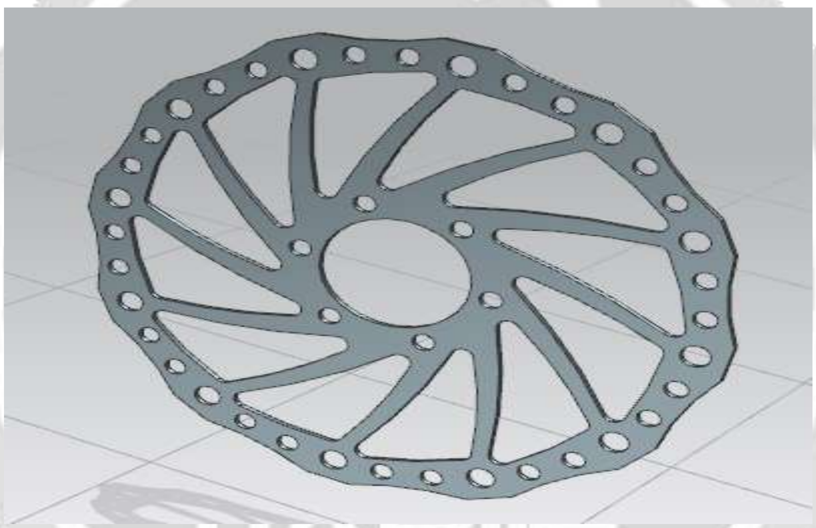


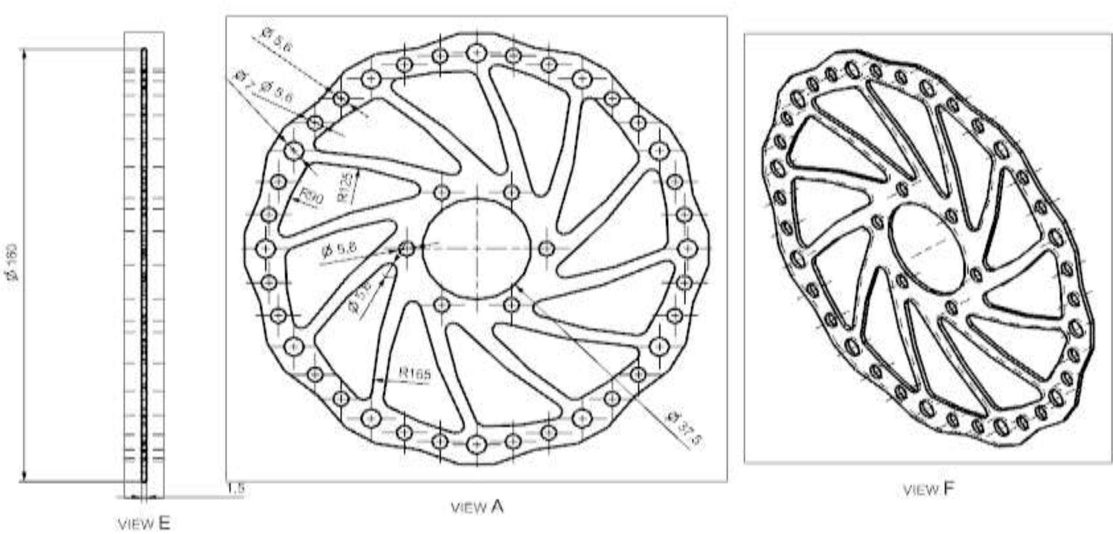
Result & discussion

Part Name	Maximum theoretical stress N/mm^2	Von-mises stress N/mm^2	Maximum deformation mm	Result
HUB	0.03	0.04	4,127E-6	safe

1. Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the HUBis safe.
2. HUB shows negligible deformation under the action of system of forces

DESIGN OF STAGGERED PROFILE DISK





MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.12) + (1.18)

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
EN8	520	360

$\Rightarrow$ fs_{max} = uts/fos = 520/2 = 260 N/mm^2

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation.

Assuming 100 % efficiency of transmission

$\Rightarrow$ T design = 0.252 Nm

$Td = \Pi/16 \times fs_{act} \times (D^4 - d^4)/D$

$$\Rightarrow fs_{act} = \frac{16 \times Td}{\Pi \times (D^4 - d^4)/D}$$

Outside diameter of drum boss = 47.2mm ---OUTSIDE DIA OF PCD CLAMP HOLE

Inside diameter of drum boss = 37.5mm

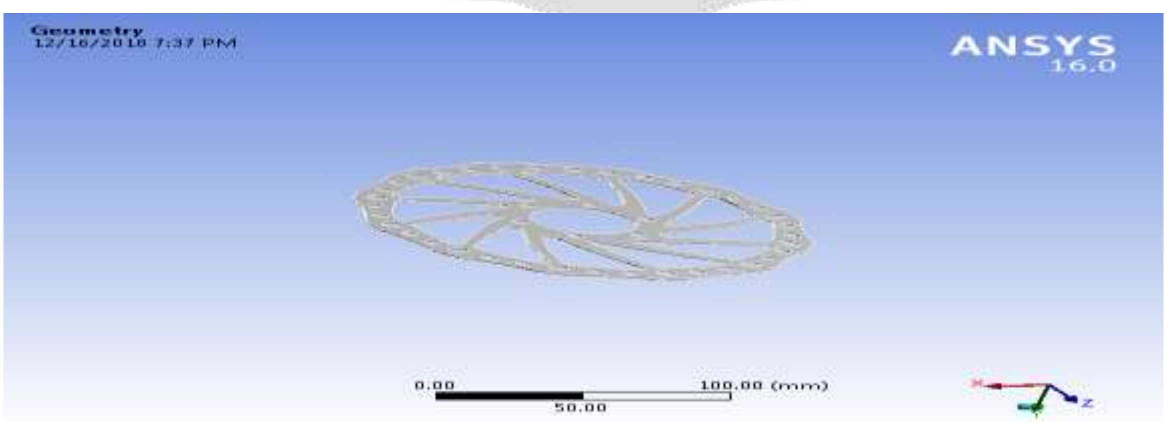
$$= \frac{16 \times 0.252 \times 10^3 \times 47.2}{\Pi \times (47.2^4 - 37.5^4)}$$

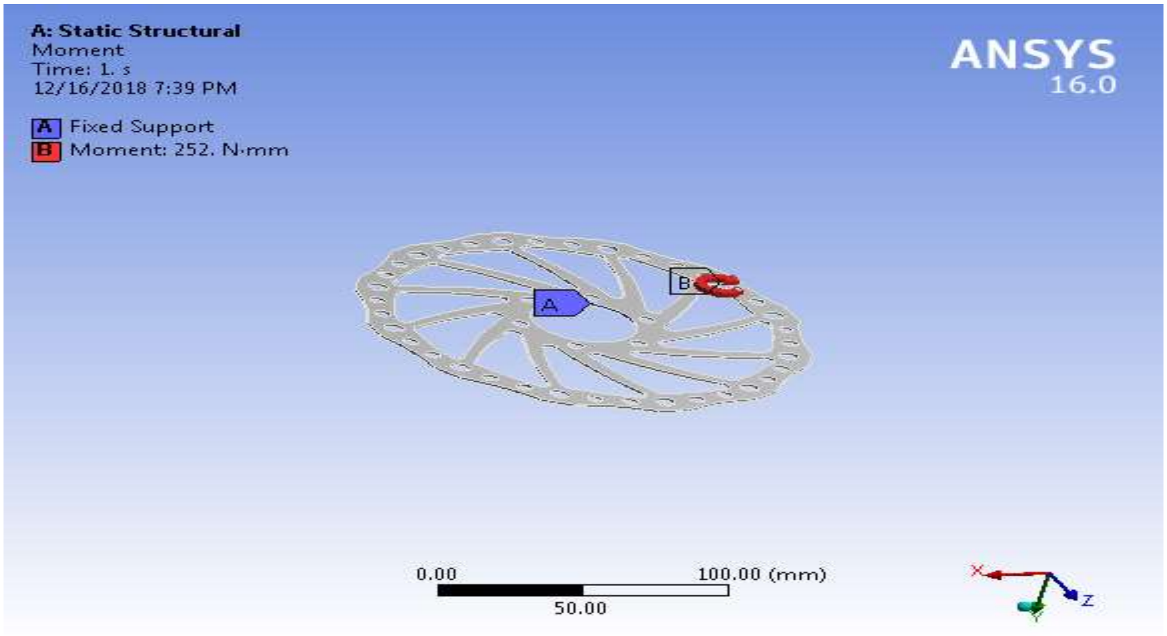
$\Rightarrow fs_{act} = 0.02\ N/mm^2$

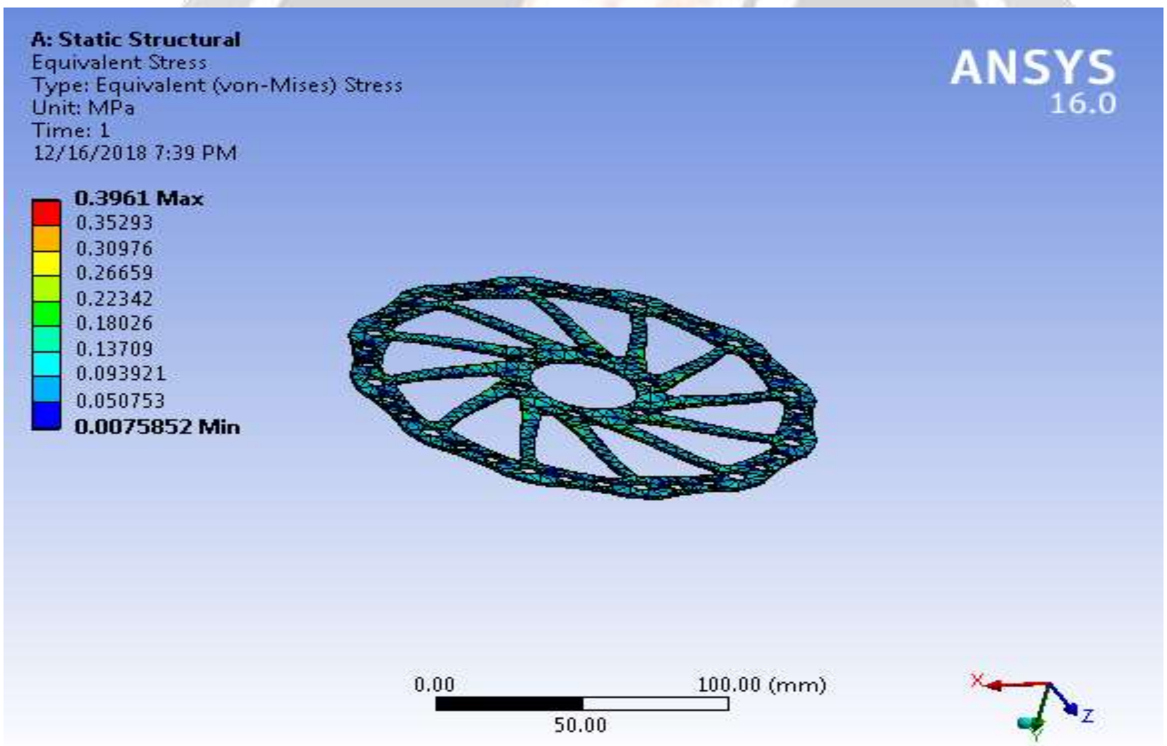
As $fs_{act} < fs_{all}$

$\Rightarrow$ CONVENTIONAL STAGGERED DISK BRAKE is safe under torsional load.

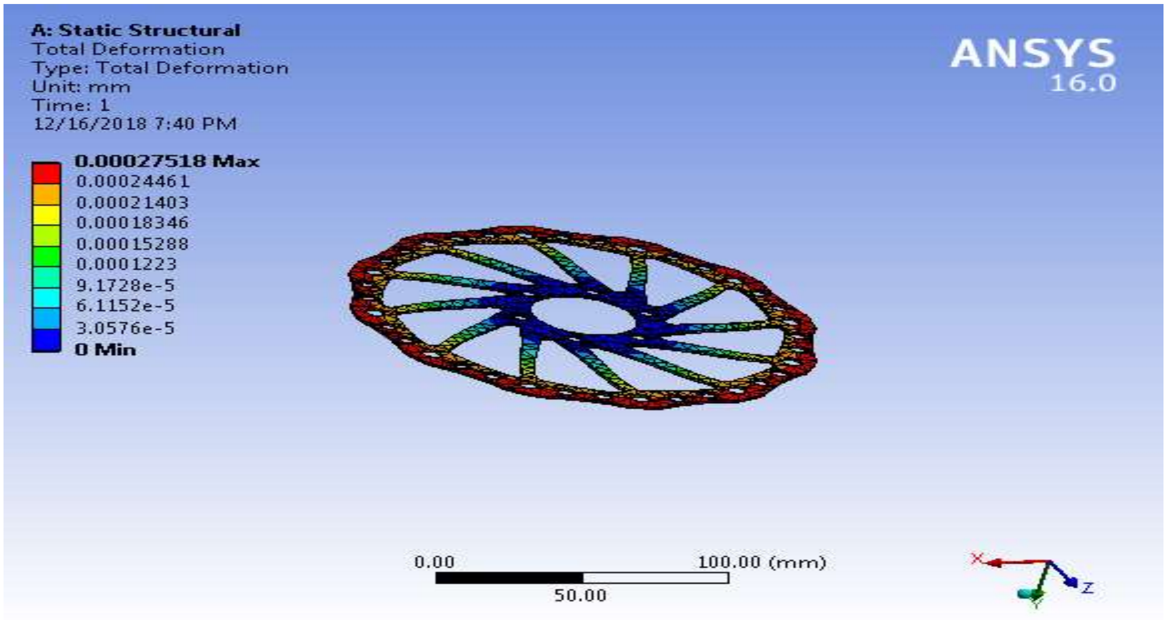
ANALYSIS OF STAGGERED BRAKE :







As the maximum stress is only 0.3961 Mpa well below the safe limit the disk brake is safe.



The deformation is well below the safe limit so the disk brake is safe

Statistics	
Nodes	6019
Elements	623
Mesh Metric	None

Result & discussion

Part Name	Maximum theoretical stress N/mm^2	Von-mises stress N/mm^2	Maximum deformation mm	Result
STAGGERED DISK	0.02	0.42	0.002	safe

1. Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the STAGGERED DISK is safe.
2. STAGGERED DISK shows negligible deformation under the action of system of forces

CONCLUSIONS

After careful review of literature it was found that no specific solution to apply air brake using exhaust gases was available. The project makes a proposal to apply one such system and the components thus needed to develop the unit have been discussed.

Thus the schematic is prepared and the future work will be to design and develop the unit and selection of standard components and design analysis of the system components to sustain the given system of forces will be discussed and done in the subsequent project work.

5. ACKNOWLEDGEMENT

In the due course of project with the valuable guidance of Guide. Prof.XXXXXXX. the project was completed as per schedule and desirable results were achieved.

6. REFERENCES

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3. Modeling and simulation of pneumatic brake system used in heavy commercial vehicle S.Mithun* , S.Mariappa* Suresh Gayakwad

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