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# Design and Analysis of Air Suspension for Three Wheel Vehicle

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A B S T R A C T

In response the defects of other types of suspensions, the air suspension system are manufactured which can be over come some problems like cost of suspension, shock and vibration. The goal of this design project is to provide confort ride, maintaining good manovering, control forces produced by tires and minimize cost by achiving better performance of the vehicle by the method of desing proper spare parts of air suspension and material selections. Finally, by following proper design procedure this design project achive/ attain the required air suspension for the three wheel vehicle.

Keywords: Design, Suspension, Three wheel

## 1. Introduction

Air suspension is the term given to the system of air springs, and with out linkages that connects a vehicle to its wheels to serve a dual purpose that contributing to the car's handling and braking, protects the vehicle itself and any cargo or luggage from damage and wear conventional suspension system. The suspension system of a vehicle refers to the group of mechanical components that connect the wheels to the frame or body.

A great deal of engineering effort has gone into the design of air suspension systems because of an unending effort to improve vehicle ride and handling along with passenger safety and comfort. In the horse and buggy days, the suspension system consisted merely of a beam (axle) that extended across the width of the vehicle, coil springs, leaf springs, in the front ends, the wheels were rotationally-mounted at the axle ends to provide steering.

Therefore, this course will study the design and application of air suspension and coil spring (air spring and springs) suspension systems. when two wheels are mounted on either side of the rigid axle then one wheel encounters the bump, both the wheel do not execute parallel up and down motion so it gives rise to gyroscopic effect and wheel wobble.

This system replaces the conventional coil spring suspension and provides automatic front and rear load leveling. The air bags, made of spring loaded cylinders support the vehicle load at the front and rear wheels it is important to understand air suspensions because there is a direct interaction between the air suspension and the leveling of a vehicle.

In fact, there are levelingsystems available that use the vehicle air suspension to do the leveling. There is different air bag with piston and cylinder air tank and valve arrangements.

Air suspension has beenintroduced, particularly their application to prime movers. The introductionof air suspension on all axle groups (with the exception of the steer axle) hasreportedly introduced a different vehicle performance than previouslyoccurred with mechanical suspension.

Air Suspension Projectnational mass limits review conducted by the NRTC during 1993-19961. While this is adesirable outcome for productivity reasons, further work to provide guidance tooperators and manufacturers in the best use and application of air suspension systems for various multi-combination vehicle configurations was considered necessary.

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As a result of the outcomes of the mass limits review, air suspension systems havebecome an important consideration for heavy vehicle operators seeking increased massconcessions because in most States and Territories, regulators now only allow vehiclesto operate at increased mass limits if their vehicles are fitted with 'road friendly'suspension systems.
An air suspension system is considered to be suitable for higher mass applications if it has been tested and certified as a 'road friendly' suspension. A road friendly suspension is considered to be less damaging to the road and therefore is allowed to carry a greater mass. Currently the only suspensions available for heavy trailers that qualify as 'road friendly' are air suspended systems.
The industry's take-up of air suspension systems for high mass operations has been rapid an in most cases, quite successful. Anecdotal evidence from industry, however, suggested that the use of air suspension systems in certain long combinations did not result in the best performing vehicles.

## 2. Statement of Problem

At the existing, the air suspension components are operated by the mixing of air and oil, which means the working principle of this system, is done by hydraulic system and air system, but due to this operation of hydraulic system, it is very costly. Due to minimize the cost, we are modifying this project by pneumatic system only and the work principle is done only by air.

## 3. Objectives of Project

**General objectives**
The main objective of the project is to design and analysis of air suspension for three wheel vehicle which is used to reduce the vibration system of the vehicle.

**Specific Objectives**

- To provide good ride and handling performance
- To ensure that steering control is maintained during maneuvering
- To provide isolation from high frequency vibration from tire excitation
- To ensure that the vehicle responds favorably to control forces produced by the tires.
- To reduce cost by changing the hydraulic system into pneumatic system.
- To achieve the better performance of the vehicle.

## 4. Methodology of The Project

For To atten the the objectives of this design follow these steps are write the introdaction, litratur reviw desgin of the compenent of the air suspension system and analisys the vibration system.
**The steps to design air suspension**

- Design of cylinder
- Design of piston
- Design of piston rod
- Design of spring
- Design of bolt
- Design of air below

**Vibration analysis**

- Free vibration
- Damped vibretion
- Forced vibration

### *4.1 Design of Cylinder*

In The Cylinder there is Stresses due to an Internal Pressur.The stresses induced in a thin cylindrical shell is made on the following assumptions:

- The effect of curvature of the cylinder wall is neglected.
- The tensile stresses are uniformly distributed over the section of the walls.

When a thin cylindrical shell is subjected to an internal pressure, it is likely to fail in the following two ways.

- Circumferentially splitting the cylinder into two troughs.
- It may fail across the transverse section splitting the cylinder into two cylindrical shells.

Thus the wall of a cylindrical shell subjected to an internal pressure has to withstand tensile stresses of the following two types:

- Circumferential or hoop stress.
- Longitudinal stress.

**Circumferential or Hoop Stress**

Consider a thin cylindrical shell subjected to an internal pressure and a tensile stress acting in a direction tangential to the circumference is called circumferential or hoop stress or on the cylindrical walls.

Let p = Intensity of internal pressure,
d = Internal diameter of the cylindrical shell,
l = Length of the cylindrical shell.
t = Thickness of the cylindrical shell, and
$\sigma_t$ = Circumferential or hoop stress for the material of the cylindrical shell.

The total force acting on a longitudinal section of the shell cylinder is equall to intensity pressure times Projected area.

F= $p \times d \times$ l ……………………………………………………….eq(1)

The total resisting force acting on the cylinder walls is equall to hoope stress times twice of the thikines of the cylinder times lengith of the cylinder.

F= $\sigma_t \times 2t \times l$ ……………………………………………………..........eq(2)

From equations eq(1) and eq(2) ,

$\sigma_t \times 2t \times l = p \times d \times l$ Then

$\sigma_t = \frac{p \times d}{2t}$ and $t = \frac{p \times d}{2\sigma_t}$ ……………………………………….eq(3)

In the design of small cylinders, a value of 6 mm to 12 mm is added in eq(3) Therefore

$t = \frac{p \times d}{2\sigma_t}$ *+6 to 12 mm*

In case of cylinders of ductile material the value of circumferential stress $\sigma_t$ istaken 0.8 times the yield point stress $\sigma_y$ and for brittle materials, $\sigma_t$ is taken as 0.125 times the ultimate tensile stress $\sigma_u$.

**Longitudinal Stress**

Consider a closed thin cylindrical shell subjected to an internal pressure and tensile stress actin in the direction of the axis is called longitudinal stress**.** In other words it is a tensile stress acting on the equation eq(1) transverse or circumferential section.

Ina longitudinal stressthe total force acting on the transverse section is equall intensity pressure times cross-sectional area of cylinder.

$F = P \times \frac{\pi}{4} d^2$ …………………. ……………………………………………eq(4)

Also the total resisting force

$F = \sigma_l \times \pi \times d \times t$ ………………………………………………………………. eq(5)

From equations eq(4) and eq(5),

$P \times \frac{\pi}{4} d^2 = \sigma_l \times \pi . d . t$ became $P \times \frac{\pi}{4} d = \sigma_l \times \pi . t$ then,

$\sigma_l = \frac{P \times d}{4t}$ or $t = \frac{P \times d}{4\sigma_l}$

From the above equation the longitudinal stress is half of the circumferential or hoop stress. Therefore, the design of a cylinder must be based on the maximum stress therefore hoop stress. Be for this it is necessary to estimate the appling force on the cylinder or on the piston to design the cylinder suspension.

Due to this,take the maximum weight of minibus to analysis the applied force to design the suspension cylinder, there for from the rule of car weight it take 3.5 tones are allowed for the 12-16 people carry(seting) minibus which means 3500kg mass M=3500kg because one tone is equal to 1000kg then multiple with gravitation then the weight is mass times gravitational acceleration.

W=M × g

W=3500kg × 9.81$m/s^2$

=34335N=34.335kN

Then another consideration also necessary to occur the single suspension System for the single cylinder force, there for by estimating the total weight divided to the total number of cylinder. But before that another consideration the rule of vehicle mass division in to sprung and un sprung mass ratio 50% by 50% is

$$\frac{W}{2} = \frac{34335N}{2} = 17167.5N.$$

This result is total mass of that car divided to two sprung and un sprung mass ratio 50% by 50%. Then total suspension cylinders are four then now divided the unsprung mass to the four cylinder.

Then $\frac{17167.5N}{4} = 4291.875 = 4292$ N

The force is applied weight by the car on one cylinder or suspension cylinder.We occur the internal pressure which is supported for this force at the inside of cylinder it supply by the compressor. pressure is force per area,

$$p_i = \frac{F}{A} \text{ ..........eq (6)}$$

Where $p_i$ =Internal pressure of cylinder in Map
F = Applied (external) force of the car support by internal pressure in N
A= area of the cylinder in $or\ mm^2$

area of internal cylinder is by considering the weight the internal diameter is $d_i$= 80mm

Area of the internal cylinder $A_i = \frac{\pi d_i^2}{4} = \frac{\pi}{4} 80^2$ mm$^2$

$= 5026.5mm^2$

There for internal pressure can occur used this formula directly then

$$P_i = \frac{F}{A}$$

$$P_i = \frac{4292N}{5026.5mm^2} = 0.85387 \text{ Mpa} = 85.387 \text{ kpa}$$

This is the supported pressure for each single piston or cylinder in the normal condition but during the working time the force is changing to shock or vibration force not a steady load then now it consider the shock factor of safety and the material is selected from mild steel therefore the value of factor of safety is 15 then the ultimate force supporting pressure is,

$P_i$ =15×0.85387Mp

=12.808Mp = 12808kp.

And the ultimate force is

$F = P_i \times A$

F=12808kp×5026.5$mm^2$

F= 64380 N is the only for one cylinder force.

Now the internal pressure is apply a force on the cylinder surface due to the force acting on the cylinder stress are produce then direct with the formula, the total force acting on a longitudinal section of the shell is equall to Intensity pressure times projected area of cylindere is,

F= p× $d$ × l ……………………………………………………eq(1)

And the total resisting force acting on the cylinder walls also,

F= $\sigma_t$×2t×l ………………………………………………………eq(2)

From equations eq (1) and eq (2)

$\sigma_t \times 2t \times l = p \times d \times l$ = $P \times \frac{\pi}{4} d = \sigma_l \times \pi.t$ *then*

$\sigma_t = \frac{p \times d}{2t} = \frac{P \times d}{2 \times t}$

$t = \frac{p \times d}{2\sigma_t}$……………………………………………………………eq(3)

In the design of small cylinders, a value of 6 mm to 12 mm (in practical is 7.6mm) is added in equation eq(3) therefore

$t = \frac{p \times d}{2\sigma_t}$*+6 to 12 mm (7.6mm)*

Where$\sigma_t$isthe tensile stress and calculated from the ultimate strength (ultimate stress) for shock load mild steel ASTM A514 material is $S_u$ = 760Mpa and the yield strength (yield stress) $S_y$ = 690Mpa therefore the tensile stress is,

$\sigma_t = 0.125\sigma_u$ = 0.125×760Mpa = 95Mpa,

Now$t = \frac{p \times d}{2\sigma_t}$+6 to 12mm (7.6mm)

$= \frac{12.808 \times 80}{2 \times 95}$ +7.6mm = $\frac{1024.64}{190}$+7.6mm

= 2.696+7.6mm= 12.96mm = 13mm

Then again the longitudinal stress can calculated

$\sigma_l = \frac{P\times d}{4t}$ or $t= \frac{P\times d}{4\sigma_l}$

Therefore longitudinal stress became

$\sigma_l = \frac{P\times d}{4t} = \frac{12.808\times 80}{4\times 13}$

$= \frac{1024.64}{52} = 19.704$Mpa

the tangential stresses

$\sigma_t = \frac{P\times d}{2\times t} = \frac{12.808\times 40}{2\times 13}$

$= \frac{1024.64}{26} = 39.41$Mpa

And mean diameter of the cylinder is

$D_m = \frac{D_0 + d_i}{2}$

Outer diameter is

$D_o = d_i + 2t = 80+(2\times 13) = 106$mm then

$D_m = \frac{D_0+d_i}{2} = \frac{106mm+80mm}{2}$

$= \frac{186mm}{2} = 93$mm

$R_o = \frac{D_o}{2} = \frac{106mm}{2} = 53mm$

$R_i = \frac{d_i}{2} = \frac{80mm}{2} = 40mm$

$R_m = \frac{D_{mm}}{2} = \frac{93mm}{2} = 46.5mm$

And the length of the cylinder is free length of spring puls strock of the piston, then

$H_c = L_f + 1.2\times d_i = 160 + 1.2\times 80 = 256$mm

### *4.2 Design of Piston*

The piston is a disc which moves up and down within a cylinder. It is either moved by the fluid or it moves the fluid which enters in the air suspension cylinder. The main function of the piston is to receive the force from the air pressure and to transmit the force to the rod through the piston the piston must also transfer the force to the farm or body of the car through the connecting rod.

The piston of internal face is flat type and full in both end and consists of the following parts,

1. **Piston seal.** The piston seal are used to sealing the cylinder in order to prevent the removing air between the piston and cylinder..
2. **Skirt.**The skirt the portion of the piston below the seal section
3. **Top land** is the distance from the top of the piston to the seal groove.

The materials that used for piston is cast aluminum because the aluminum alloy pistons is light rather than steel and the area is equal with the area of internal cylinder but it must considered the clearance of allowable between the piston and the cylinder surface.

**1. Piston seals:-**The piston seals are used to control the necessary of pressure to maintain the seal betweenthe piston and the cylinder bore. Thise is made of plastic material (natural rubber) because in this design there is no high temperature and that give good sealing and its thickness equation is equal to

$T_p = \frac{D}{10\,n}$

Where $T_p$ = thickens of piston seal

: D = diameter of cylinder

n = number of seals

Now $T_p = \frac{D}{10n} = \frac{80mm}{1\times 10} = T_p = 8$mm

**2 .Piston Skirt:-**The portion of the piston below the seal section is known as piston skirt.
. the length of the piston skirt (L) is determine. In actual practice is taken as 0.65 to 0.8 times the cylinder diameter.
Then the length of the piston skirt is taken as 0.7D,

L=0.7×80mm=56mm

**3.Top land:-** The length or the top land is the distance from the top of the piston (pressure reciving side) to the seal groove. the height of top land is $T_L$= $T_h$ to 1.2$T_h$ in this case there is no top height ($T_h$) in stmated take 10 because the seal is one, therefor,
$T_L$ = 1.2×10mm= 12mm
Now the total hight of the piston is Piston seal plus Piston Skirt plusTop land,

H= $T_{p+}$ L+ $T_L$

=8mm+56mm+12mm =76mm

***4.3 Design of Piston Rod***

The piston rod is a major link inside of the air suspension, it connects the piston to the frame (chassis) and is responsible for transmited the force from the piston to the frame (chassis).There are different types of materials and production methods used in the manufacture of piston rods. The most common types of rods are steel and aluminum.

The most common type of manufacturing processes are casting. The rod is hard, to increase its strength against compresing force. Use to balanced the internal pressure and weight of the vehicle. For this design uses cast of alloy steel containing nickel, chromium, molybdenum and vanadium, have tensile strength or maximum compressive stress is 407.7Mpa this type of rod is reliable.

In this design the pressure is applied by one face of the piston and the other side of the piston faces is fixed on the piston rod. This means that the failure of piston rod will occur due to excessive compressive stress developed in the cercular area of the piston rod and the stress (σ) can calculate.

σ = F/A

Where $\sigma$ = compressive stress of piston rod
F= force of the internal air pressure
A = area of the piston rod ($\frac{\pi d^2}{4}$)
d = diameter of the piston

Now the maximumcompressive stress of piston rod ($\sigma_c$) is given and its value is 407.7Mpa and the internal air pressure of the cylinder ($P_i$) then force is given by
F = $P_i$×$A_p$=12.808×$\frac{\pi 80^2}{4}$

= 12.808Mpa×5026.5$mm^2$ = 64379.4N

Now the working stress on the piston rod

$\sigma_w$ = F/A = 64379.4N/$\frac{\pi 30^2}{4}$

= 64379.4N/706.8$mm^2$ = 91.086Mpa

***4.4 Design Of Bolt***

**Tensile Stress In Bolt**

It is necessarily to determine the stress in the bolt due to both static and dynamic forces, from the many characteristics of bolts stress is very important. The possible failure modes of bolts are tensile and sheared failure, tensile stress tensile load is the fundamental mode of load bolts, the possible failure mechanisms under axial load are tension failure of the bolt at the shank and at the threads considering a failure in the shank of the bolt, the correctional area is calculated and the ultimate stress ($\sigma_u$) of the material is automotive ASTMA is 724MPA
$\sigma_t$ = F/$A_b$ then $A_b$ =$\frac{\pi}{4}d^2{}_b$

Where $\sigma_t$ = tensile stress of the bolt

$A_b$ = correctional area of bolt

$d_b$ = bolt diameter (M16)

Bolts subjected to tension really fracture through the shank though, and using the area of shank in capacity calculation can result in an over estimate of the bolt's actual strength. Another alternative is to use the root area of the threads in capacity calculation is multiple by 0.8 the correctional area of shank, Then
$\sigma_t$ = F/0.8 × $A_b$ and $A_b$ =0.8× $\frac{\pi}{4}d^2{}_b$

The force apply on the bolt and its value is from internal pressure 64380N

Then $\sigma_t$ = F/0.8 × $A_b$

= 64380N /0.8×$\frac{\pi}{4}$× ($16^2$)

= 64380N /160.8=400.2MPA

This design is for the piston and piston rod attachment or on lower side of the piston rod.
Similarly the force apply on the bolt at the upper side piston rod attachment thread bolt is M24 and its force value is from internal pressure 64380N and the stress is became,

$\sigma_t$ = F/0.8 × $A_b$

= 64380N/0.8×$\frac{\pi}{4}$× $(24^2)$

= 64380N/361.91 = 177.88MPA = 178 MPA

**4.4.2 Shear stress of bolt**

The shearing of bolts can take place in the threaded portion of the bolt and the area at the root of the threads, also called the tensile stress area is taken as the shear area. Since threads can occur in the shear plane, the area for resisting shear should normally be taken as the net tensile stress areaof the bolts. The shear area is specified in and is usually about 0.8 times the shank area. However, if it is ensured that the threads will not lie in the shear plane then the full area can be taken as the shear area.A bolt subjected to a shear force shall satisfy and the yield stress of the material is ASTM A is 552MPa and force 64380N from the above result is use. The shear stress in the bolt is design only for the upper side of the piston rod.

$\tau_b$ = F $/A_s$ = F/ $(\boldsymbol{\pi} d_b^2/4)$

where F =is the load acting on an individual bolt

A= is the area of the bolt and (M24)

and $\tau_b$= $F_b/A_{bs}$ = $F_b$/ $(\boldsymbol{\pi}\times t\times d_b/4)$ = 4F/$\boldsymbol{\pi}\times(24)^2$is at the shank but the thickness of the plat that use for attaching the piston rod and the chassis is unknown, and the stress at the tensile stress area is became,

$\tau_b$= F $/A_b$= $F_b$/ $(0.8\times\boldsymbol{\pi} d_{bt}^2/4)$

= F/0.8×$\frac{\pi}{4}$×$(24)^2$

= 64380N/0.8×$\frac{\pi}{4}$× $(24^2)$

= 64380N/361.91 = 177.88MPA = 178 MPA

**4.5 DESIGN OF SPRING**

**Types of springs**

There are many types of the springs, according to their shapes these,

- Helical springs.
- Conical and volute springs.
- Torsion springs

In this design it selected the helical spring to better fit with cylinder, there for the helical springs are made up of a wire coiled in the form of a helix and is primarily intended for compressive loads. The cross-section of the wire which is formed a circular. The helical springs are said to be closely coiled when the spring wire is coiled so close that the plane containing each turn is nearly at right angles to the axis of the helix and the wire is subjected to twisting.

**Material for helical springs**

The material of the spring has high fatigue strength, high ductility and it is resist creep. It largely depends upon the service for which they are used, the spring made from oil-tempered carbon steel wires containing 0.6 percent carbon and 0.6percent manganese. Materials like phosphor bronze, beryllium copper, brass etc. can also used in this as special cases to increase fatigue resistance, temperature resistance and corrosion resistance.
The helical springs are either cold formed or hot formed depending upon the size of the wire. Wires of small sizes (less than 10 mm diameter) are usually wound cold drawn and The strength of the wires varies with size, smaller size wires have greater strength and less ductility, due to the greater degree of cold working.
Therefore for this production can allow in hot formed because the diameter sizes is grater than10 mm. It is also a, oil tempered, and general purpose spring steel. It is suitable for fatigue or sudden loads, at subzero temperatures and at temperatures above 1800C. in the below shows the values of allowable shear stress, modulus of rigidity and modulus of elasticity for this materials used for helical springs.

Table 1: values of helical spring.

| **Material of spring** | **allowable shear stress(τ)MPa** | | | **Modulus of elasticity (E ) in KN/$mm^2$** | **Modulus of rigidity (G) in KN/$mm^2$** |
|---|---|---|---|---|---|
| **Carbon steel, Oil tempered wire** | **Severe service** | **Average service** | **Light service** | | |
| 8-13.5mm | 294 | 364 | 455 | 210 | 80 |

Values of allowable shear stress, modulus of elasticity and modulus of rigidity

**Terms Used In Compression Springs**

The following terms used in connection with compression springs are important from the subject point of view.

**1.Spring index.** There is an important parameter in spring design called spring index. It is denoted by letter C. The spring index is defined as the ratio of mean coil diameter to wire diameter. Or $C = \frac{D}{d_w}$ in design of helical springs, the designer should use good judgment in assuming the value of the spring index C. The spring index indicates the relative sharpness of the curvature of the coil.

A low spring index means high sharpness of curvature. When the spring index is low (C < 3), the actual stresses in the wire are excessive due to curvature effect. Such a spring is difficult to manufacture and special care in coiling is required to avoid cracking in some wires. When the spring index is high (C >15), it results in large variation in coil diameter.Spring index from 4 to12 is considered better from manufacturing considerations. and now C=4 is possible for this design ofspring.

**2. Free length.** The free length of a compression spring is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed). Mathematically,

Free length of the spring equal to Solid length pluse Maximum compression pluse clearance between adjacent coils.

$L_f = n \times d_w + \delta max + 0.15\ \delta\ max$

The following relation also be used to find the free length of the spring,

$L_f = n \times d_w + \delta max + (n - 1) \times 1$ mm

In this expression, the clearance between the two adjacent coils is taken as 1 mm.

**3. Solid length.** When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire mathematically; Solid length of the spring is,

$L_s = n \times d_w$

Where n = Total number of coils are and

$d_w$ = Diameter of the wire

**4. Deflection (compression) of Helical Springs**

Total active length of the wire is equal to length of one coil times number of active coils

$$L = \pi D_m \times n$$

where L= Length of one coiltimes number of active coils

$D_m$ = mean diametr of the helical spring

N = number of turnig coils

Andθ = Angular deflection of the wire when acted upon by the torque T.There for axial deflection of the spring is,

$$\delta = \theta \times \frac{D_m}{2}$$

then $\frac{T}{J} = \frac{\tau}{D_m/2} = \frac{G.\theta}{l}$,

$$\theta = \frac{T.L}{J.G}$$

Where G = Modulus of rigidity for the material of the spring wire from table

J= Polar moment of inertia of the spring wire

$$J = \frac{\pi}{32} \times {d_w}^4,$$

$d_w$ = being the diameter of spring wire, and

$$J = \frac{\pi}{32} \times {d_w}^4 = \frac{\pi}{32} \times 10.95^4$$

$$= \frac{564.1}{32} = 128.89 = 129mm^4$$

Now substituting the values of l and J in the above equation, then

$$\theta = \frac{T.L}{J.G} = \frac{\left(W \times {D_m}/2\right)\pi D_m.n}{\frac{\pi}{32} \times {d_w}^4 \cdot G} = \frac{16.W.n.{D_m}^2}{G{d_w}^4}$$

**5. Spring rate.** The spring rate (or stiffness or spring constant) is defined as the load required perunit deflection of the spring. Mathematically, spring rate,

$$k = \frac{W}{\delta}$$

Where W = Load, and

δ= Deflection of the spring.

**6.Pitch.**

The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state. Mathematically pitch of the coil,

$p = \frac{\text{free length}}{n-1}$

The pitch of the coil may also be obtained by using the following relation, there for Pitch of the coil,

$p = \frac{L_f - L_s}{n} + d_w$

Where $L_f$ = Free length of the spring.

$L_s$ = Solid length of the spring.

n = Total number of coils.

$d_w$= Diameter of the wire.

**End connections for compression helical springs**

The end connections for compression helical springs are suitably formed in order to apply in the center the load. Various forms of end connections are available those are by squared, ground Plain, Squared and ground ends therefore our design is ground closed end.

**Table 2 :Value of spring**.

| TYPE OF END | Total number of turn(n) | Solid length | Free length |
|---|---|---|---|
| Ground and closed end | N | n× d | n× p+(n-1)or n× p (n×d+δ+0.15 δ) |

values of total number of turns, solid length and free length for ground types of end connection spring.

**Stresses in the Helical Springs of Circular Wire**

Consider a helical compression spring made of circular wire and subjected to an axial load *W*.

Let $D_m$= Mean diameter of the spring coil,

$d_w$= Diameter of the spring wire,

n= Number of active coils from the given above,

*G* = Modulus of rigidity for the spring material,

W= Axial load on the spring,

$\tau_{max}$ = Maximum shear stress induced in the wire,

C = spring index = $D_m / d_w$ = 4 from slandered design

p = Pitch of the coils, and

δ = Deflection of the spring, as a result of an axial load W.

Now design procedure for helical compression spring of circular cross section. consider a part of the compression spring, The load W tends to rotate the wire due to the twisting moment ( T ) set up in the wire is $T = W \times D_m /2$ . Thus twisting shear stress is induced in the wire.

.Under the action of two forces (W) and the twisting moment(T) the twisting moment is

$T = W \times D/2 = \pi/16 \times \tau_t \times d_w^{\ 3}$

There for this became

$\tau_t = \frac{8W.D_m}{\pi d_w^{\ 3}}$

Where W =weight of the load

$D_m$ = mean coil diameter

$\tau_t$= twisting shear stress

$d_w$= wire spring diameter and

$C = D/d_w$

$\tau_t = \frac{8W.D}{\pi d_w^{\ 3}} = \frac{8W.C}{\pi d_w^{\ 2}} \Rightarrow d_w^{\ 2} = \frac{8W.C}{\pi \tau_t}$

$d_w = \sqrt{\frac{8W.C}{\pi \tau_t}} = \sqrt{\frac{8*4*4292}{\pi*364}} \sqrt{\frac{137344}{1143.54}} = \sqrt{120.156}$ = 10.95mm

$D_m = d_w \times C$ = 4×10.95mm = 43.8mm

$D_i = D - d_w$ = 43.8 – 10.95 = 32.85mm

$D_o = D + d_w$ = 43.8mm + 10.95mm= 54.75mm

Then $\tau_t = \frac{8W.D}{\pi d_w^{\ 3}} = \frac{8W.C}{\pi d_w^{\ 2}} = \frac{8*4282*4}{\pi*10.95^2} = \frac{137344}{377.318}$ = 364 Map

In addition to the twisting shear stress ($\tau_t$) induced in the wire, the following stresses also act on the wire:

**1** Direct shear stress due to the load W, and

**2** Stress due to curvature of wire.

The direct shear stress due to the load W is,

$$\tau_{d} = \frac{\text{load (W)}}{\text{cross sectional area of the wire}}$$

$$= \frac{w}{\frac{\pi}{4} \times d_w{}^2} = \frac{4w}{\pi d_w{}^2}$$

$$= \frac{4 \times 4292\ \text{N}}{\pi \times 10.95^2} = \frac{17168}{377.18} = 45.5\ \text{Map}$$

The stress is maximum at the inner edge of the wire, therefore maximum shear stress induced in the wire is equal to Tensional shear stress plus Direct shear stress.

$$\tau_{max} = \tau_{t+}\tau_{d},$$

$$\tau_{MAX} = \frac{8w \times D_m}{\pi d_w{}^3} + \frac{4W}{\pi d_w{}^2} =$$

$$\tau_{max} = \frac{8 \times .D_m}{\pi d_w{}^3} + \frac{4W}{\pi d_w{}^2} = \frac{8w.\times}{\pi d_w{}^3}\left(1 + \frac{d_w}{2D_m}\right)$$

$$= \frac{8w \times D_m}{\pi d_w{}^3}\left(1 + \frac{1}{2c}\right) = \frac{8w \times D_m}{\pi d_w{}^3} \times k_s = \frac{8w \times C}{\pi d_w{}^2} \times k_s$$

Where $C = D_m/d_w$ Spring index = 4

$KS = 1 + \frac{1}{2c}$ Shear stress factor

$$= 1 + \frac{1}{2 \times 4} = 1.125$$

$$\tau_{max} = \frac{8w.C}{\pi d_w{}^2} \times 1.125 = \frac{8*4292*4}{\pi * 10.95^2} \times 1.125$$

$$= \frac{386316}{377.18} = 1024.22\text{Mpa}$$

From the above equation, it can be observed that the effect of direct shear $\frac{8w.D_m}{\pi d_w{}^3}\left(1 + \frac{1}{2c}\right)$is

appreciable for springs of small index C. Also neglected the effect of wirecurvature, It can noted that when the springs are subjected to static loads, the effect of wire curvature is neglected, because yielding of the material will relieve the stresses.

In order to consider the effects of both direct shear as well as curvature of the wire, a Wahl's stress factor (K) introduced, Wahl can be used. The resultant diagram of Twisting shear stress, direct shear and curvature shear stress.

there for Maximum shear stress induced in the wire,

$$\tau_{max} = \frac{8w.D_m}{\pi d_w{}^3} k = \frac{8w.c}{\pi d_w{}^2} \times k$$

Where $k = \frac{4c-1}{4c-4} + \frac{0.615}{c} = \frac{4 \times 4 - 1}{4 \times 4 - 4} + \frac{0.615}{4}$

$$= \frac{15}{12} + 0.15375 = 1.40375$$

Then, $\tau_{max} = \frac{8w.D_m}{\pi d_w{}^3} k = \frac{8w.c}{\pi d_w{}^2} \times k$

$$= \frac{8 \times 4292 \times 4}{\pi \times 10.95^2} \times 1.40375 = \frac{137344}{377.318} \times 1.40375$$

$$= 510.965 = 511\text{MPA}$$

The Wahl's stress factor (K) composed of two sub-factors, KS and KC, such that K= KS × KC

Where KS = Stress factor due to shear, and

KC = Stress factor due to curvature

### 4.5.6 Stability

Compression coil spring may be buckle when the deflection be come to large there for the effective deflection is given by the equation

$$\delta = \frac{\alpha L_f}{Dm}$$

Where $\alpha$ is spring end condition constant the end condition constant $\alpha$ depends up on how the spring are supported the spring one end supported by flat surface perpendicular to spring axis and its value is 0.707. and the free length also given by $2.67D_m < L_f < 4D_m$ then

$$L_f < \frac{\pi * D}{\alpha}\sqrt{\frac{2(E-G)}{2G+E}} = \frac{\pi * 43.8}{0.707}\sqrt{\frac{2(210000-80000)}{2*80000+210000}}$$

$$= \frac{\pi * 43.8}{0.707}\sqrt{0.7027} = 194.62 \times 0.83827$$

= 163.144mm from this 160mm free length is satsfay for this design and free length

$L_f$ = P×n = n×d+δ+0.15 δ the deflction is becom

But $\delta_{max} = \frac{8\times W\times c^3\times n}{Gd}$

$= \frac{8\times 4292\times 4\times 4\times 4\times n}{80000\times 10.95} = \frac{2197504\,n}{876000}$ = 2.5085n

$L_f$ = p× n = n×d+δ+0.15 δ

= n×10.95 +2.5085n+ 0.376n

= n(10.95+2.5085+0.376)

= n13.835mm

From this 160mm is equalelize with n13.835mm

$L_f$ = n13.835mm = 160mm then the number of coil is

❖ Number of turn coils

n = 160/13.835 = 11.565 = 11 is number of turns, and

❖ Solid length of the spring,

$L_s = n\times d_w$ = 11×10.95 = 120.45mm

❖ Pitch of the coil,

$p = \frac{\text{free length}}{n-1} = \frac{L_f - Ls}{n} + d_w = \frac{160-120.45}{11} + 10.95$ ,

= 3.595 +10.95 =14.545mm

❖ The maximum deflection is

$\delta_{max} = \frac{8\times W\times c^3\times n}{Gd} = \frac{8\times 4292\times 4\times 4\times 4\times n}{80000\times 10.95} = \frac{2197504\,n}{876000}$

= 2.5085n = 2.5085×11

= 27.593 = 27.6mm

❖ Spring rate K = $\frac{W}{\delta}$ = 4292/27.6 = 155.5N/mm

❖ The angle is $\theta = \frac{T.L}{J.G} = \frac{\left(W\times {}^{D_m}/_2\right)\pi D_m.n}{\frac{\pi}{32}\times d_w{}^4.G} = \frac{16.W.n.D_m{}^2}{G d_w{}^4} =$

$= \frac{16\times 4292\times 9\times 43.8^2}{80000\times 10.95^4} = \frac{13174111.7n}{1150128761}$ = 0.011455(11) =0.126

θ = 0.126 × $\frac{180}{\pi}$ = 7.2199dgree

❖ Length of the wire is equale to Length of one coil times number of active coils

L = $\pi D_m$× n=π×43.8mm×11 =1513.6 mm =1.5m

**4.5.8 Energy stored in spring**

Energy is stored in helical spring of circular as the energy store in the spring equal to the work done on it by the some external load. Let W is load applied on the spring and δ is deflection produce in the spring due to the load W. assuming that the load is applied gradually then energy stored in a spring U =$\frac{1}{2}$Wδ.

Where W = 4292N

δ = 27.6 mm

from the above then the energy storing in the spring is,

U = $\frac{1}{2}$×4292N×27.6mm

= 59229.6Nmm = 59.23Nm

***4.6 Design of Air Bellow (Bag)***

An air bellow is a carefully designed rubber bellow, which contains a piece of compressed air, the rubber bellow itself does not provide force or support load, this is done by the piece of air are highly engineered elastomeric bellow designed and the end closures with metal attachment. The standard of air bellows is made up of four layers.

The air bellow is also available in high strength construction, in this case there are four plies of fabric-reinforced rubber, with an inner liner and outer cover air bellow are designed for use with compressed air. Nitrogen is also acceptable, air springs may be filled with water or water glycol (automotive antifreeze) solutions. If water is to be used rust inhibitors should be added to protect the end closures.

But in this design air bellower is only designed to air, do not use for water. Recommend that there is a minimum three times safety factor between maximum internal air pressure and burst pressure, therefore factor of safety is five for this design to more safe.

**4.6.1 Neoprene Polychloroprene(NBR)**

Neoprene is a type of rubber it has more resistant to damage from oil, in the design neoprene has been added, in addition bellower neoprene is able to withstand higher temperatures than natural rubber only and it have same material strength with the natural rubber.

Therefore this is the main case to use in the design to combined and the temperature resistance by the combined compound is (-37º to +74ºc). Advantages good inherent flame resistance, moderate resistance to fabrics and good resistance to weather, ozone, and natural aging good resistance to abrasion and flex cracking very good resistance to alkalis and acids. Generaly neoprene is an excellent for all-purpose with few practical limitations.

**4.6.2 Natural Rubber (NR)**

Natural rubber is atype of rubber with high tensile strength,superior resistance to burst pressure an in addition to excellent wear resistance, excellent rebound elasticity; good flexibility at low temperatures excellent adhesion to fabric and metals. natural rubber also good flexing qualities and service at low temperatures.

**Now the design procedure:-** the internal pressure of the maximum of air bellow is similar with the internal cylinder pressure and the ultimate tensile strength of the material neoprene (NBR) and natural rubber(NB) is 90MPAthat uses for our project by combining the two compound elements to resistance the weather conditions, ozone, heat aging , flame , cols, and gasoline.

The better understanding tensile strength in the rubber is first recall that there are intermolecular force (known as Vander walls force) helping to hold long polymer chains. These forces are at their weakens when due to structural irregularities. Then the area of the bellower is larger than the outer diameter of the cylinder by 10mm in the radius for enough air capacity, cylinder outer diameter is 106 from cylinder design therefore the area is became,

$A_b = \frac{\pi d^2{}_i}{4}$

Where $A_b$ = area of bellow

$d_i$ = internal diametr of bellow

$A_b = \frac{\pi d^2{}_i}{4} = \boldsymbol{\pi}\times 126\times 116/4 = 3364\boldsymbol{\pi} = 10568.\text{mm}^2$ and

The force applied in the internal bellow is coulculate from the internal pressure of cylinder therefore force equal internal pressure of the bellow divided by the area of internal cylindercal bellow $F = P_i/A$

Where F = force

$P_i$ = internal pressure

A = area of cylinder

but the internal pressure of bellow is equal to the internal pressure of cylinder because of the Pascal's law states that pressure applied to a internal system at any point is transmitted undiminished and equally throughout the in all directions and acts upon every part of the internal system at right angles to its interior surfaces.

$P_{be} = P_{cy} = 12.808\text{MPA}$

Now force of the bellower is

$F = A\times P_{be} = 10568.\text{mm}^2 \times 12.808\text{MPa}$

$F = 135354.944\text{N}$

Then the thickness of the bellower is from the cylinder formula equation is ocure

$t = \frac{p_b \times d_i}{2\times \sigma_t}$

$= \frac{12.808\,\text{mpa}\times 126\,\text{mm}}{2\times 90} + 6 \text{ to } 12\text{mm}(7.6\text{mm})$

$= \frac{1613.808}{180} + 7.\ 6\text{mm} = 8.96 + 7.6 = 16.56\text{mm mm}$

Now the tensiol stressof the bellow is,

$\sigma_t = \frac{p_b \times d_i}{2\times t} = \frac{12.808\times 126}{2\times 16.56} = \frac{1613.808}{33.12}$

$= 48{,}726\text{MPA}$ and

longitutdnal stress of the bellow also

$\sigma_l = \frac{p_b \times d_i}{4\times t} = \frac{12.808\times 126}{4\times 16.56} = \frac{1613.808}{66.24}$

$= 24.363\text{MPA}$

#### *4.7 Design of Self Leveling Valve*

The valve is designed to control the raising and lowering of chassis by supplying air to inflating and deflating the air bellow. Control is done automatically but a manual is also possible.This automatically adds air to air bellow, or exhausts air from air bellow to maintain a constant static height, which isresponds instantly to correct air pressure relabeling chassis to desired ride height the valve is the fastest air spring fill and exhaust system. very faster fill and exhaust rates.

The martial of the valve body is cast of aluminum alloy composed, the ultimate tensile strength is aluminum ASM6061-T4 of $\sigma_u$ is 235Mpa

Then the internal area of the valve is small and internal diameter is 4mm and outer diameter12mm from standard data book (catalog) that precise for the ultimate tensile stress less than100Mpa the outer diameter is thread, therefore the internal area is became,

$A= \frac{\pi d^2{}_i}{4} = \boldsymbol{\pi}\times4\times4/4$

$=4\boldsymbol{\pi} = 12.56\text{mm}^2$ and

The force in the internal surface is we find from the internal pressure of cylinder then force is internal pressure of the cylinder miltble by the area of internal cylinder.

$F=P_i \times A$

Where F = force

$P_i$= internal pressure

A = area of cylinder

And then force is became

$F = P_i\times A = 12.808\text{mpa}\times12.56\ mm^2$

$= 362.47N = 160.868N$ and

Then tensile stress is be came

$\sigma_t= \frac{P_i\times d_i}{2t}$

where $P_i$= internal pressure

$d_i$ = internal diameter

t = thickness of the valve

$\sigma_t$= tensile stress

Then stress $\sigma_t= \frac{P_i\times d_i}{2t}$ ,$t = \frac{d_o-d_i}{2}$

Where $d_o$= outer diametr of valve

$d_i$ = iner diametr of valve

Then $t = \frac{d_o-d_i}{2} = \frac{12-4}{2} = 4\text{mm}$

$\sigma_t= \frac{P_i\times d_i}{2t} = \frac{12.808\times4}{2\times4} = 6.404\text{MPa}$

Longitutdnal stress $\sigma_t= \frac{P_i\times d_i}{4t} =\frac{12.808\times4}{4\times4}$ =3.202 MPA

#### *4.8 Design of Air Hose (Line)*

Air suspension hose is a line which uses to transport the compressed air from the compressor to air bag (bellower) and from the bellower to the cylinder to give an air spring, the design of air hose is strongest, toughest, and the most durable hose, and the designing of air hose is an original anti-disconnect design that reduces undesired hose separations.

The internal pressure of the maximum of the air hose is similar with the internal cylinder (bellow) pressure and the ultimate tensile strength of the materialnatural rubber(NB)is 90MPA uses for this project that have high tensile strength, good flexibility and capacity to resistance the oxidation, ozone, heat aging , flame , cols, oil and gasoline.

Therefore the area of the hose (line) is small the internal diameter is 6mm and the outer diameter is 14mm from standard data(catalog) given to less than 100MPA pressure, and the internal pressure of air hoise is equal to the internal pressure of cylinder or bellow because of the Pascal's law pressure applied to a internal system at any point is transmitted undiminished and equally throughout the all directions and acts upon every part of the internal system at right angles to its interior surfaces therefore the internal pressure hose is similar with the internal pressure bellower, then area is become,

$A= \boldsymbol{\pi}\times d^2{}_i/4$

Where $d_i$ = internal hose diameter

A = internal area of the hose

$A = \boldsymbol{\pi}\times6\times6/4 = 28.3mm^2$

Stress is becam $\sigma_t= \frac{P_i\times d_i}{2t}$

Where $P_i$ = internal pressure

$d_i$ = internal hose diameter and

t = thickness of the hose ($d_o$-$d_i$)/2

$d_o$ = outer diameter of hose

$d_i$ = internal hose diameter then

$$t = \frac{14\text{mm} - 6\text{mm}}{2} = 4\text{mm}$$

Then $\sigma_t = \frac{12.808\times6}{2\times4} = \frac{76.848}{8} = 9.606$MPa

And then force is became F = $P_i$×A

= 12.808mpa×28.3$mm^2$

= 362.47N = 362.5N

### *4.9 Vibration Analysis of Air Suspension*

This chapter is concerned with the analysis of the motion excitation at the base of the four wheeled car which results a discomfort for the passengers. The motion that is created at the base of the car has a sinusoidal motion which can be studied under vibration analysis of the vehicle.

This chapter consists of the mathematical modeling the displacement, velocity and acceleration and the graphs for the transmissibility of the motion excitation. As the car moves along the road the vertical displacement at the tire acts as the motion excitant to the vehicle suspension system. The motion of this system consists of a translation motion at the center of mass and rotational motion about the center of mass.

In simple harmonic motion, we assume that no dissipative forces such as friction or viscous dray exist. Since the mechanical energy is constant the oscillation continues forever with constant amplitude. The oscillations of vibrating gradually die out as energy is dissipated. The amplitude of each cycle is a little smaller than that of the previous cycle. This kind of motion is called damped oscillation.

A vibratory system basically consists of three elements namely.

- The mass
- The spring
- The dumper

In the vibrating system there is exchange of energy from one form to anther energy is stored by **mass** in the form of **kinetic energy**& in the **spring** in the form of **potential energy.**

KE=$\frac{1}{2}$M$\ddot{x}^2$ ………………………………….eq(1)

PE=$\frac{1}{2}$K$\ddot{X}^2$ ………………………………….eq(2)

Energy enters the system which the application of external force known as exaltation. The kinetic energy is converted in to potential energy in to kinetic energy this sequences goes on repeating and the system continues to vibrate.

**Method of Vibration Analysis**

There are various methods by means of which can drive the equation of motion of vibration system. On the system is a force of spring KX, damping force C$\dot{X}$ and inertia force M$\ddot{X}$.

The equation of motion

m$\ddot{x}$ +c$\dot{x}$+kx ………………………………….eq(3)

**5.2.1 Energy method**

According to this method the same of energy associated with the system is constant

Kinetic energy + potential energy = constant

KE+PE=constant

$\frac{d}{dx}(\frac{1}{2}mx^2+\frac{1}{2}kx^2) = \frac{d}{dx}(\frac{1}{2}(m\dot{x}\dot{x}+kxx))=0$

m$\ddot{x}$ +kx=0 ……………………………….eq4

The equation of motion is simple and harmonic that written as

X=Asin $\omega t$

$\ddot{X}$ = -A$\omega^2$ sin $\omega t$

Substituting in to the equation (4)

m-A$\omega^2$ sin $\omega t$ +k Asin $\omega t$ =0

mA$\omega^2$ sin $\omega t$ =k Asin $\omega t$ =0

$\omega^2$ = k /m

$\omega = \sqrt{k/m}$ ……………………………….eq (5)

$f = \frac{1}{2\pi}\omega$ ……………………………….eq (6)

**Rayleigh methods**

This method is the extension of energy method. The method is based on the principals that the total energy of the vibrating system is equal to the maximum potential energy.

$(KE)_{max} = (\frac{1}{2}mx^2)_{max} = \frac{1}{2}m(\omega A)^2$ ……………………………eq (7)

$(PE)_{max} = (\frac{1}{2}kx^2)_{max} = \frac{1}{2}kA^2$ ………………......eq(8)

$m(\omega A)^2 = KA^2 = m\omega^2 = k$

$\omega = \sqrt{k/m}$ ………………………………………..eq(9)

Let as consider we assume that the mass of the vehicle that acts on the air suspension is it depends on the vehicle size and standard. We assume that the mass of the vehicle is 3.5 tones.

$M_{total} = 3.5\ tone \times 1000kg = 3500kg$

The total mass is divided in to two parts by the rule of sprang and un sprang weight of vehicle.

$M_{total} = 3500kg/2 = 1750kg$

This mass is that acts on the four axial wheel of air suspension

M= 1750kg/4= 437.5kg this mass is the mass of each air suspension.

To calculate the force that exerts on the air suspension is given as

$P = m \times g$ ……………………………………eq (10)

$P = 137.5kg \times 9.81 = 4291.87N$

Where p= the force that exerts on the air spring

m= the mass of the vehicle on one air spring

g =the acceleration of gravity

Let as to calculate the spring stiffens of air suspension is as fellows

$K = P/\delta$………………………………………..eq (11)

$$\delta = \frac{8 \times p \times c^3 \times N_c}{G \times d_w}$$ …………………………………eq(12)

$$\delta = \frac{8 \times 4291.87N \times 4^3 \times 11}{220 \times 10^6 KN/m \times 10.mm}$$

$\delta = 27.61$ mm

Where k = stiffens of the spring

P= the force of the vehicle to exhort the spring to dun ward.

$\delta$ = the deflection of the spring.

Nc =No of coil spring.

G= modules of elastic city

dw =diameter of wire

Substituting the deflection in to equation (11) to calculate the stiffness value of **K**

$$K = \frac{4291.87N}{27.61mm}$$

K=155.56N/mm

The value of k is substituting in to the above equation (5) or equation (9) to calculate the natural frequency of the vibration

$\omega = \sqrt{k/m} = \sqrt{\frac{155.56N/mm}{437.5kg}} = \omega = 0.639$ rad/sec

$f = \frac{1}{2\pi}\omega = \frac{1}{2\pi} \times 0.736$rad/sec = 0.118 rad/sec

**TYPES OF VIBRATION**

**Free Vibration**

Free vibration is a vibration in which energy is neither added to nor removed from the vibrating system. It will just keep vibrating forever at the same amplitude. Except from some superconducting electronic oscillators, or possibly the motion of an electron in its orbit about an atomic nucleus, there are no free vibrations in nature. They are all damped to some extent.

**Damped Vibration**

Damped vibration is one in which there is an energy loss from the vibrating system. This loss may be in the form of mechanical friction, as at the pivot of a pendulum for example, or in the form of turbulence as the vibrating system disturbs its surroundings. The amplitude of a damped vibration will eventually decay to zero.

This is the equation of motion

$m\ddot{x}+c\dot{x}+kx$ …………………………………………………..eq(12)

Let as to calculate

- Damping factor ($\xi$).
- Damping frequency ($\omega_d$).
- The critical damping ($c_c$).
- The amplitude of resonance (A).
- Maximum frequency at peak amplitude ($\omega_{max}$).

To calculate Damping factor ($\xi$)

$\boldsymbol{\omega}_n = \sqrt{k/m}$ =0.639rad/sec

$\xi = \frac{c}{2m\omega n}$ ……………………………………………………eq(13)

Where $\xi$ =damping factor

$\omega n$ =natural frequency of the system

m = mass of the system

c=the damper of the system

The damper of the air suspension is calculating as fellows.

Damper = $\frac{2}{3}$cross sectional area of the piston× height of working chamber of the system.

$C=\frac{2}{3}A_P \times H_W$ …………………………………………………eq(14)

Where $A_P = 0.001101m^2$ or $1101mm^2$ and $H_w = 0.23m$ or 23cm

The cross sectional area of the piston and the height of working chamber of the cylinder is calculated in the design parameter in the previous chapter.

$C = \frac{2}{3} \times 5026.54mm^2 \times 256mm$

C=0.857 N-sec/mm

To calculate Damping factor ($\xi$) substituting the value damper **C** in to eq(13)

$\xi = \frac{c}{2m\omega n}$

$$\xi = \frac{0.857}{2 \times 437.5kg \times 0.639rad/\sec} = 0.00583$$

$\xi = 0.00583$

There for $0 < \xi < 1$

Where $c$=the damper of the system

m =the mass of the system

$\omega n$= the natural frequency of the system

$\xi$=the damping factor of the system

Let as to calculate the damping frequency ($\omega_d$) of air suspension.

$\omega_d = \omega n\sqrt{1-\xi^2}$ …………………………………………………eq(15)

$\omega_d$ = 0.639 rad/sec$\sqrt{1-(0.00583)^2}$

$\omega_d$=0.638 rad/sec

$\omega_d \approx \omega n$

Where $\omega_d$= damping frequency

$\omega n$= the natural frequency of the system

$\xi$=the damping factor of the system

Let as to calculate the critical damping ($c_c$) of air suspension system.

$c_c = 2\sqrt{k \times m}$ …………………………………………………………eq(16)

$c_c = 2\sqrt{155.56N/mm \times 437.3kg}$

$c_c$ =521.7 unit

Where $c_c$ = the critical damping

k =the stiffness of the spring

m =the mass of the system

Let as calculate the amplitude of the system at resonance of the air suspension system.

$$A = \frac{F/K}{\sqrt{(1-\frac{\omega_d}{\omega_n})^2)^2 + (2\xi\frac{\omega_d}{\omega_n})^2}}$$ ………………………………………….eq(17)

Where $r = \frac{\omega d}{\omega n}$ =1 at resonance

$$A = \frac{F/K}{\sqrt{(1-(r)^2 + (\xi r 2)^2}}$$

$$A = \frac{F/K}{2\xi}$$

$$A = \frac{4291.88N/155.N/m}{2(0.00583)}$$

A (resonance) =237.1mm

Note

If ($\xi > 1$) = over damped system

When the value of damped ratio ($\xi$) is more than one the system is known as over damped system. The motion is periodic characteristics of over damped system.

If ($\xi < 1$) = under damped system.

When the damped factor is increases from zero to one the amplitude goes on decreasing. But when $\xi$ is $0 \leq \xi \leq 1$ the vibration is periodic displaces to both said of the equilibrium mean position.

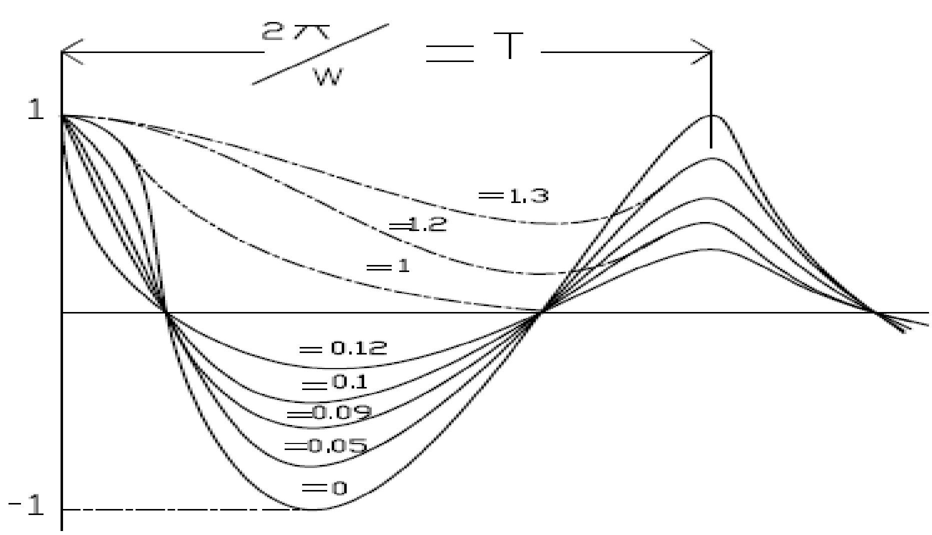


**Figure 1: (damped vibration graph).**

If ($\xi = 1$) is critical damped vibration.

When the damped factor is equal to one it is called critically damped vibration. It transits from periodic to a periodic when $\xi = 1$ so the system vibrates. X decrease at t increase and finally camas to zero as t tends to infinity this is also a periodic vibration.

**Forced vibration**

Forced vibration is one in which energy is added to the vibrating system, as for example in a clockwork mechanism where the energy stored in a spring is transferred a bit at a time to the vibrating element. The amplitude of a forced, undamped vibration would increase over time until the mechanism was destroyed. The amplitude of a forced, damped vibration will settle to some value where the energy loss per cycle is exactly balanced by the energy gained. The vibration which occurs under the influence external force is called forced vibration.

Forced exaction (vibration) may be due to external or internal force. internal exaction occurs due to unbalance in the system. Unbalance due to bent shaft loose mating parts, bearing and journal defects vibration in turning moment mass of rotating parts not distributing uniformly magnetic effect.

Equation of motion with harmonic motion.

$m\ddot{x}+c\dot{x}+kx = f\sin\theta$

The mass (m) dispels from its equilibrium position by a distance x in the down ward direction. The mass (m) is put in three forces, spring force (kx), damping force ($c\dot{x}$ ) and harmonic exaltation ($f\sin\omega t$).

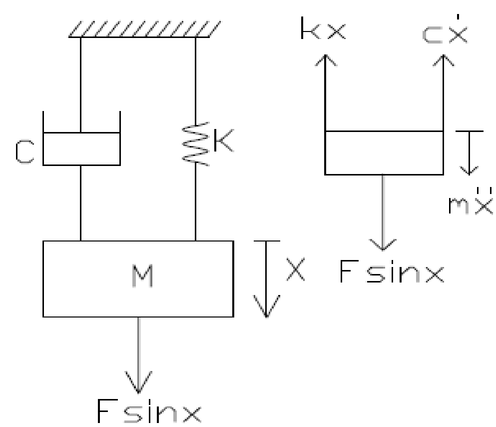


**Figure 2: (Forced vibration graph).**

The equation of motion can be written as

$c\dot{x} + kx = - m\ddot{x} + f\sin\theta$

$m\ddot{x} + c\dot{x} + kx = f\sin\theta$ ……………………………………………….eq(18)

this is the second order linear differential equation with constant coefficient. The general solution of the above equation is

$x = x_p + x_c$

where $x_c$ =complimentary solution

$x_c$ is the homogenous equation $m\ddot{x} + c\dot{x} + kx = 0$ which is salved in the previous topic on damped vibration.

$x_p$ = particular solution

let as consider the particular solution ($x_p$).

$x_p = A\sin\omega t - \phi$ ……………………………………………………….eq(19)

where A = the amplitude of the vibration.

$\phi$ = the phase angel of the displacement with respect to harmonic force.

$\dot{x}_p = \omega t\cos\omega t - \phi$

$= \omega A\sin(\omega t - \phi + \frac{\pi}{2})$

$\ddot{x}_p = \omega^2 A\cos(\omega t - \phi + \frac{\pi}{2})$

$= \omega^2 A\sin(\omega t - \phi + \pi)$

Then substituting in eq(18)

$m(\omega^2 A\sin(\omega t - \phi + \pi)) + c(\omega A\sin(\omega t - \phi + \frac{\pi}{2})) + k(A\sin\omega t - \phi)) - F\sin\theta = 0$…….eq(20)

eq(20) is the algebraic sum of forces equal to zero. so we can assume it is a condition of equilibrium.

There fore the vector diagram of fore these four forces namely inertia forces, damping forces, spring forces and harmonic forces should be a closed polygon.

fsin $\omega t$

kAsin( $\omega t - \phi$)

$c\omega A \sin(\omega t - \phi + \frac{\pi}{2})$

m($\omega^2$Asin( $\omega t - \phi + \pi$)

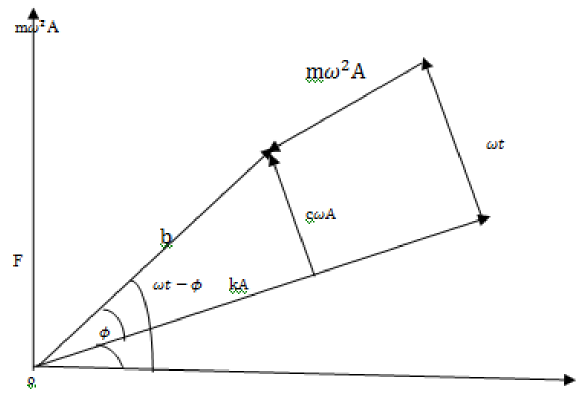


Figure 3:(forced vibration analysis diagram).

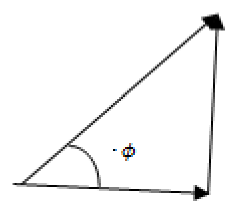


Fig (vector representetion of forced vibration referance)

<u>from⊿ oab</u>

$F^2$ =(kA-m$\omega^2$A)$^2$ +(c$\omega$A)$^2$

$F^2$= A$^2$ (k-m$\omega^2$)$^2$ +(c$\omega$)$^2$………………………………………….eq(21)

The force and ampilutid from eq (21)

$F = \sqrt{A^2(k - m\omega^2)2 + (c\omega)2}$

$F = \sqrt{(237.1mm)2((155.5N/mm - 437.5kg(0.639)^2)2 + (0.857 \times 0.639)2)}$

F=12N (the harmonic force of the system)

$$A^2 = \frac{F^2}{(K - m\omega^2)^2 + (c\omega c^2}$$

$$A = \frac{F}{\sqrt{(K - m\omega^2)^2 + (c\omega c^2}}$$

Maltypilaying with $k/_k$

$$A = \frac{F/K}{\sqrt{(\frac{k}{k} - \frac{m\omega^2}{k})^2 + (\frac{(c\omega c^2}{k})^2}}$$

$$A = \frac{F/K}{\sqrt{(1 - (\frac{\omega_d^{\ 2}}{\omega_n^{\ 2}})^2 + (\frac{c\omega}{k})^2}}$$

$$\tan\phi = \frac{C\omega A}{kA - M\omega^2 A} = \frac{C\omega}{k - M\omega^2}$$

$$\tan\phi = \frac{c\omega / k}{1 - (\frac{\omega_d}{\omega_n})^2}$$

where $\frac{c\omega}{k} = \frac{(2\times\xi\times\omega)}{\omega_n}$

$$\tan\phi = \frac{(\frac{2\xi\omega}{\omega_n})}{1-(\frac{\omega_d}{\omega_n})^2}$$

$$= (\frac{2\times\xi\times\omega_d}{\omega_n})\times(\frac{\omega_n}{1-\omega_d}) = \frac{2\times\xi\times\omega_d}{1-\omega_d}$$

$$\phi = \tan^{-1}(\frac{2\times 0.00583\times 0.6389 rad/\sec}{1-(0.6389) rad/\sec}$$

$\phi = 18^\circ$ the phease angele of the forced vibration

**Transmissiblity**

Considor a mechne which vibrats and trans miting its vaibration to the base faundetion.

Maximam ampiluted ,x =Asin $\omega t$

Maximam spring force =kA

Maximam damping force

$F_t = \sqrt{(kA)^2 + (c\omega A)^2} = A\sqrt{(k)^2 + (c\omega)^2}$

$F_t = 237.1mm\sqrt{(155.5)^2 + (0.857\times 0.639 rad/\sec)^2} = 36869.2N$

In the abave eqution $F_t$ is the force transmited to the fawndation (base) F is disturrbuting force.

The retio of $\frac{F_t}{F}$ is transmissiblty (T.R).

T.R = $\frac{F_t}{F}$

T.R = $\frac{F_t}{F} = \frac{36869.2N}{12N} = 3072.4N$

## 5. Discussion

Air suspension is one of the most important part of vehicle. The aim of this project is to design and analysis the air suspension by taking all necessary calculations concerning its basic components. In addition the most proper materials which have to be used have been determined. It has been taken into consideration that the chosen materials must resist on the maximum pressure and stresses that occur when the air suspension is working.

Another goal is to make drawings on solid work that clearly display the air suspension structure, connection and location of all parts to make the design of air suspension.The project begins with short description of the literature of air suspension and how they have benefits, the main components that designed are piston, cylinder, piston rod, bellow and helical spring as well as vibration analysis.

The project continues with an explanation of the functions of these parts and the used materials for their production. Next step is the calculation of tensile and longitutdnal stress of cylinder, tensile stress of bellow, and the different dimention, stress ,deflection rates of the spring. After making all mentioned calculation, the necessary part dimensions are achieved, and the project continues with drawings on solid work.

## 6. Conclusion

This part consists of two (2) sub-parts: the article's conclusion and suggestions or recommendations from the research. Conclude the article critically and logically based on the research findings. Please be careful in generalizing the results. The authors should also state the research limitation in these parts.

Generally, the conclusion should explain how the research has moved the body of scientific knowledge forward. In suggestion, please describe the author's recommendations for further studies regarding the author's research implication.

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