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Design and Development of Double Wishbone Electro-Hydraulic Active Suspension System

Mr. K. S. PATIL, Prof.V. R. GAMBHIRE
kspatil9393@gmail.com

Abstract : This paper is a literature survey for the electro Design and development of Double electro-hydraulic active suspension system. It is divided into three section viz. literature survey, methodology adapted and design procedure and experimentation. In literature survey. we find the theoretical information regarding the various suspension systems, performance and find the way of work. Methodology consist of different part of systems like spring, drive pinion, rack, helical cam, follower pin and push rod. Design and Experimental result to evaluate Performance of prepared model it is placed over exciter and variable load is applied over it through strut arrangement.

Keywords : Electro-hydraulic, suspension, literature survey, methodology, design and experimental result

I. Introduction

Active or adaptive suspension is an automotive technology that controls the vertical movement of the wheels with an onboard system rather than the movement being determined entirely by the road surface. The system virtually eliminates body roll and pitch variation in many driving situations including cornering, accelerating, and braking.

This technology allows car manufacturers to achieve a greater degree of ride quality and car handling by keeping the tires perpendicular to the road in corners, allowing better traction and control.

Active suspensions can be generally divided into two main classes: pure active suspensions and semi-active suspensions. Active suspensions, the first to be introduced, use separate actuators which can exert an independent force on the suspension to improve the riding characteristics. The drawbacks of this design (at least today) are high cost, added complication/mass of the apparatus, and the need for rather frequent maintenance on some implementations. Maintenance can be problematic, since only a factory-authorized dealer will have the tools and mechanics with knowledge of the system, and some problems can be difficult to diagnose. Michelin's Active Wheel incorporates an in-wheel electrical suspension motor that controls torque distribution, traction, turning makeovers, pitch, roll and suspension damping for that wheel, in addition to an in-wheel electric traction motor. Hydraulically actuated suspensions are controlled with the use of hydraulic servomechanisms. The hydraulic pressure to the servos is supplied by a high pressure radial piston hydraulic pump. Sensors continually monitor body movement and vehicle ride level, constantly supplying the computer with new data. As the computer receives and processes data, it operates the hydraulic servos, mounted beside each wheel. Almost instantly, the servo-regulated suspension generates counter forces to body lean, dive, and squat during driving manoeuvres'.

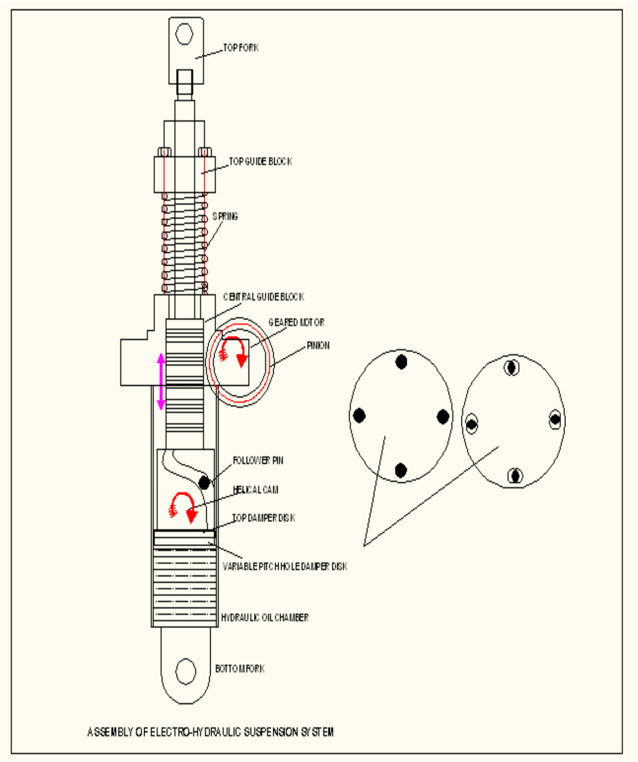
In practice, the system has always incorporated the desirable self-levelling suspension and height adjustable suspension features, with the latter now tied to vehicle speed for improved aerodynamic performance, as the vehicle lowers itself at high speed

II. Material and Methodology

The operating principle of the Hydro-electric System:
Geared motor is used to drive the pinion which will move the rack either up or down, this used to change
1) Deflection of the spring to change the spring rate as per condition of the road..ie, for large bumps spring length will be maximum and for short bumps but in series the spring length will be short. The rack moves upward thereby delfecting the spring to reduce the length of spring, similarly rack moves downward to delfect the spring to increase the length of spring.
2) To change the damping coefficient of the system by changing the damper hole size using the variable pitch disk. The downward motion of the rack makes the cam to open the holes of damper thereby allowing oil to easily pass through system disks thus enable a smooth descent of the suspension in large bump where as the upward motion of the rack makes helical cam to close holes of the damper to allow reduded oil flow to provide better damping in case of short but series of bumps.

PART LIST

Sr.No	**Part Name**
01	Spring
02	Drive Pinion
03	Rack
04	Helical Cam
05	Follower Pin
06	Push Rod
07	Bottom Fork
08	Top Fork
09	Hydraulic Chamber



Methodology

1. Design-

Design of Electro-hydraulic suspension system components like drive pinion, rack, helical cam, follower pin and push rod. Mechanical design of all components of the set up using theoretical formulae..

DESIGN OF WORM AND WORM WHEEL FOR WHEEL SHAFT DRIVE

The pair of worm and worm wheel used in the machine is designated as

1/55/10/1

The worm is made of case hardened steel 14C6 where as the worm wheel is made of Cast iron.

Z1 = 1
Z2 = 55
q= 10
M=1
I = z2/z1 =55
N = 800 rpm
N2 = 800/55 =14.5 rpm
D2 = m x z2 =1 x55 =55
tanU=z1/q = 5.71^{0}
F = 2m sq.rt (q+ 1) = 9.94
Da1 = m(q+2) =12
C= 0.2mcosU = 0.3
Lr ={ da1+2c} sin-1[F/(da1+2c)]
Lr = 632
For case hardened steel Sb =28.2
For BRASS, Sb =6.2
Xb1 = 0.25
Xb2 =0.48
$Mt_1 = 17.65\ Xb_1\ Sb_1 m\ lr\ d_2 \cos U$
$= 4.694 \times 10^6$ N-mm
$Mt_2 = 17.65\ Xb_2\ Sb_2 m\ lr\ d_2 \cos U$
$= 1.98 \times 10^6$ N-mm

The lower value of torque is on the wheel = 1.98×10^6 N-mm
Kw = 2Π n2 Mt/60 x 10^6
Kw =7.46 Kw
As the drive is capable of transmitting 7.46 Kw and we intend to transmit 0.08Kw the drive is safe.

I. DESIGN OF RACK AND PINION FOR RETRACTION

Power = 20 watt
Speed = 60rpm
b = 10 m
No.of teeth on pinion = 10
No. of teeth on GEAR = 45
Reduction ratio (i) = 4.5
Material of pinion and gear is High grade steel, EN24
Tensile strength =640 N/mm^2

$$\text{POWER} = \frac{2\,\Pi\,NT}{60}$$

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

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

$$= \frac{60 \times 20}{2 \times \Pi \times 60}$$

T= 3.18 N-m
Sult pinion = Sult gear = 240 N/mm^2
Service factor (Cs) = 1.5
dg = 90

$$\text{Now; } T = Pt \times \frac{dg}{2}$$

⇒Pt = 1413.4 N.

$$P_{eff} = \left(\frac{\text{I. } P_T \times C_S}{C_V}\right) = \left(\frac{1413 \times 1.5}{C_V}\right)$$

AS speed is very low, neglecting effect ofCv

P_{eff} = 2120 N ……. (A)

Lewis Strength equation
WT = Sbym
Where ;

$$Y = 0.484 - \frac{2.86}{Z}$$

$$\Rightarrow yp = 0.484 - \frac{2..86}{13} = 0.245$$

⇒ Syp = 156.8

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Pinion and gear both are of same material and with same number of teeth hence

$Syp = Syg = 156.8$

$W_T = (Syp) \times b \times m$

$= 156.8 \times 10m \times m$

$W_T = 1568\ m^2$----------(B)

Equation (A) & (B)

$1568\ m^2 = 2120$

$\Rightarrow m = 1.35$

selecting standard module =1.5 mm

GEAR DATA

No. of teeth on pinion =10

Module = 1.5 mm

No. of teeth on GEAR =30

Design of piston rod material selection:

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

Direct Tensile or Compressive stress due to an axial load:-

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

$$fc_{act} = \frac{240}{(\pi/4) \times 8.5^2}$$

$\Rightarrow$ **fc_{act} = 4.23 N/mm**

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

2. Shear stress in threaded end due to axial load :-

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

t = width thread at root = p/2

t= 0.75 mm

n = No of threads in contact = 21/1.5 =14

$$fs_{act} = \frac{240}{\pi \times 14 \times 8.5 \times 0.75}$$

fs_{act} = 0.85 N/mm^2

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

Stresses due to buckling of piston rod :-

According to Rankine formula,

Where,

$$Wcr = \frac{fc\ \ A}{1 + a\,(le/k)^2}$$

Where; Wcr = Crippling load on screw (N)

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

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

Wcr =23.56x10^3 N

DRIVE MOTOR

The drive motor is 12 VDC motor coupled to a planetary gear box.

Specifications of motor are as follows:

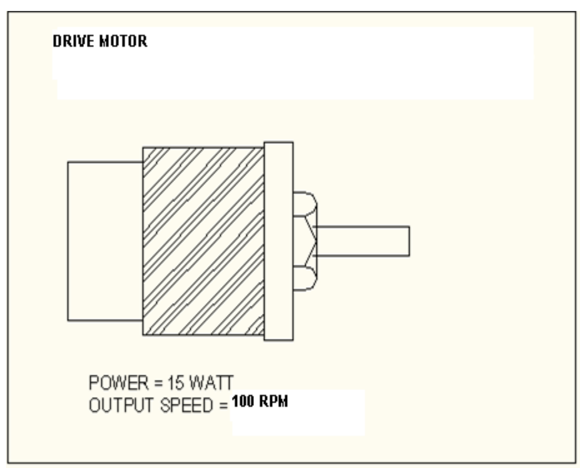
A)Power 15 watt

B) Speed = 100 rpm

C)Gear box : Planetary /epicyclic type (reduction ratio : 1:5)

D) Mounting dimensions (Face mounted M12 x 1.5) pitch

OUTPUT SPEED – 100RPM



DESIGN OF INPUT SHAFT

MATERIAL SELECTION : - 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 relations.

$= 0.18 \times 800$
$= 144\ N/mm^2$

OR

$fs_{max} = 0.3\ fyt$
$= 0.3 \times 680 = 204\ N/mm$

Considering minimum of the above values,

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

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 value of shear stress that can be induced in the shaft material for safe operation.

TO CALCULATE INTERMEDIATE SHAFT TORQUE

$$POWER = \frac{2 \Pi NT}{60}$$

Motor is 50 watt power, run at 5000 rpm, connected to intermediate shaft by belt pulley arrangement with reduction ratio 1:5

Hence input to input shaft = 1000 rpm

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

$$= \frac{60 X 15}{2 X \Pi X 100}$$

$\Rightarrow T = 1.432$ N-m

$\Rightarrow$ T design= t X REDUCTION RATIO = 1.432 X 3.5 = 5.02 N-m

CHECK FOR TORSIONAL SHEAR FAILURE OF SHAFT.

But as per manufacturing considerations we have an H6h7 fit between the pulley and shaft and to achieve this tolerance boring operation is to be done and minimum boring possible on the machine available is 16mm hence consider the minimum section on the shaft to be 16mm

Assuming minimum section diameter on input shaft = 16 mm

$\Rightarrow d = 16$ mm

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

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

$$= \frac{16 \times 5 \times 10^3}{\Pi \times (16)^3}$$

$\Rightarrow fs_{act} = 6.2\ N/mm^2$

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

$\Rightarrow$ I/P shaft is safe under torsional load

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.

2. Finite Element Analysis-

Validation of strength calculations of critical components like drive pinion, rack, helical cam, follower pin and push rod using ANSYS.

3. Fabrication of set-up-

Set-up will be fabricated using suitable method like Milling machine, DRO – Jig Boring machine, Electrical Arc Welding.

III. Results and Conclusion

Experimental testing- Testing of the Double wishbone electro-hydraulic suspension system will be done using suitable excitation methods.

The Testing of set-up will be carried out as follows-

To evaluate Performance of prepared model it is placed over exciter and variable load is applied over it through strut arrangement. Excitation conditions depicting road conditions will be changed to take several set of readings

The characteristics curves such as, excitation amplitude v/s Displacement ability, Excitation frequency v/s Damping coefficient is plotted.

This new suspension system will be useful to improve different aspects which are summarized as.

1.Increase vehicle comfort.
2.Prevent damages to chasis in deep holes and pits.
3.Reduces vibrations of chasis in frequent holes and rough road conditions.
4.Increases ride stability

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IV. Application And Future Scope

The scope research result of this new suspension system will be useful to improve different aspects which are summarized as:

1. It can be used on front and rear wheels, it is independent and most importantly, it has near perfect camber control
2.Easy to implement the system.
3. It is helpful to increase vehicle comfort and ride stability.

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