Design Development and Testing of Electrically Power Active Hybrid Suspension System

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Abstract- In these Paper experimental performance of Electrically Powered working as a Hybrid Suspension system, for means of transport was individuality of means of transport suspension systems have been widely elaborated in the past decades, be a factor to ride comfort, handling and Beneficial recovery. The suspension systems currently in use in two & four vehicle System for the absorbing load& provide comfort. However, this system cannot assure the desired functioning from a progressive suspension system. The use of electrically powered linear actuators is an Unusual for the functioning of working as a hybrid suspensions. The actuator is made with the help of power screw and electrical motor.

In this paper, it is intend an active hybrid suspension system which syndicates the simplicity of the passive. Dampers with the performance of an active suspension. Upholding the inert damper, it is possible to keep the performance of the active suspension, but using a smaller electrically powered actuator. In this Paper, the experimental results indicate that an active hybrid suspension system which will be powered electrically to Regulate. Damping and most important soft and Well-off ride of vehicle having power to Mixture according to road conditions

Keywords: Electrically Power Active Hybrid Suspension, Conventional Mc- Phreson type strut arrangement, Traction &control, Precise Linear Actuator, Damping Coefficient

I. INTRODUCTION

The primary function of vehicle suspension is to isolate the vehicle body and passengers from the oscillations created by the road irregularities and produce a continuous road-wheel contact.

At present, three types of vehicle suspensions are used: passive, semi-active and active ones. All the systems known as implemented in automobiles are based in hydraulic or pneumatic operation. However, it is verified that these solutions cannot solve satisfactory the vehicles oscillations problem or they are very expensive and contribute to the increasing of the energy vehicle consumption.

In the last decade, the evolution occurred in power electronics, permanent magnet [6]

Materials and microelectronics allowed very important improvements in the electrical drives domain. Dynamic and steady state performance, volume and weight reduction, unconstrained integration with the electronic control system, reliability, cost reduction are very important factors justifying a generalized use of electrical drives. This experimental analysis is carried out by implementing systems using electrically powered actuators in order to improve the performance of suspension system without increasing the energy consumption and the costs.

. In this paper another point of view is that an active suspension system, which keep the passenger compartment on a flat trajectory as the car wheels bounce over potholes and rough roads is a luxury concept without any practical interest in the near future.

Design Development and Testing Of Electrically Power Active Hybrid Suspension System 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.

II. SUSPENSION SYSTEMS DESCRIPTION

As already mentioned, three types of suspension systems are presently used in vehicle suspensions: passive, semi-active and active ones. All of them are constructed using hydraulic or pneumatic cylinders.

2.1 Passive Suspensions

Passive suspension systems are the most common systems that are used in commercial passenger cars. They are composed of conventional springs, and single or twin-tube oil dampers with constant damping properties. Traditional springs and dampers are referred to as passive suspensions most vehicles are suspended in this manner.

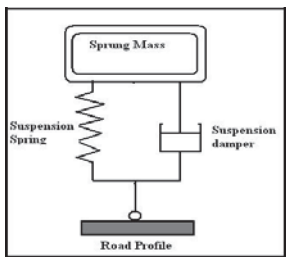


Fig.1 Passive Suspension System

2.2 Semi-active suspension systems

Semi-active suspension systems extend the possible range of damping characteristics obtainable from a passive damper. The damping characteristics of a semi-active damper can be adjusted through applying a low-power signal. Semi-active systems are a compromise between the active and passive systems. They are commercialized recently by means of either a solenoid valve as an adjustable orifice, or MR-fluid dampers.[2]

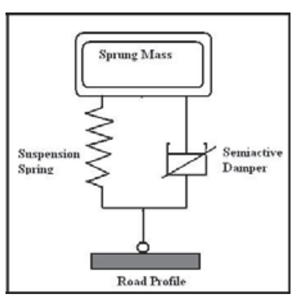


Fig.2 Semi-Active Suspension System

2.3 Active suspension systems

Active suspension system refers to a system that uses an active power source to actuate the suspension links by extending or contracting them as required.

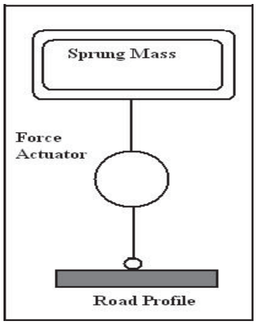


Fig.3 Active Suspension System

In an active suspension, controlled forces are introduced to the suspension by means of hydraulic or electric actuators, between the sprung and unsprang-mass of the wheel assemblies. A variable force is provided by the active suspension at each wheel to continuously modify the ride and handling characteristics.

III. SUSPENSION SYSTEM MODEL

The Fig.4 represents a model of electrically powered active hybrid suspension system for vehicles.

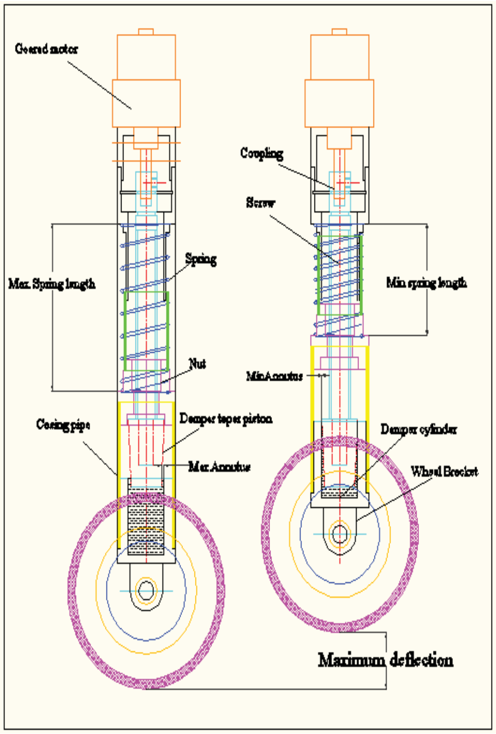


Fig.4 Electrically power active hybrid suspension system

3.1 Working principle of system

The set-up is an innovation over the conventional Mc-Pherson strut arrangement. The spring used in a helical compression spring with both end ground, the free length of the spring is adjustable. The Free length adjustment will adjust the ground clearance of the vehicle and at the same time make the suspension light thereby increasing the displacement ability of the shock absorber.

The free length adjustment is done using a precision linear actuator in the form of a12 V DC motor, coupled to power screw arrangement with precise displacement and accuracy of motion. The motor drive the power screw and thereby the nut displaces to adjust the free length of spring and also adjust the displacement of the piston of the hydraulic damper arrangement.

The second part of the hybrid system that is the hydraulic damper part, is coupled to the power screw arrangement and it adapts itself as per the motion of the power screw and nut arrangement, thereby adjusting the damping coefficient.[3]

IV. METHODOLOGY OF SYSTEM

In this Paper work to carry out Design development and testing of an electrically powered active type- hybrid form suspension for four wheel vehicles The proposed work is planned in following phases

Phase I-Literature survey.

Phase II –System design .

Phase III –Fabrication of set-up.

V. DESIGN PROCEDURE

5.1 Design of Motor

The drive motor is 12 Volt DC motor

Specifications of motor are as follows:

a)Power(P) = 15 watt

b) Output speed (N) = 100RPM

$$P = \frac{2\pi NT}{60} \quad (1)$$

$$T_d = \frac{60 \times 15}{2\pi N} \quad (2)$$

T_d = 5.02 N-m

5.2 Design of shaft

5.2.1 Material selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
EN 24	800	680

Factor Of Safety = 3 – 5

Selecting minimum diameter shaft d = 16 mm $T_d = \pi /16 \times fs_{act} \times d^3$ (3)

Fs_{act} = 6.2 N/mm^2

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

Input shaft is safe under Torsional load.

5.3 Design of Piston Rod

5.3.1 Material selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
EN 9	600	380

Factor of Safety = 3 – 5

Theoretical force Acting on piston = 240N

Pitch (p) = 1.5mm

Core diameter (dc) = 8.5mm to 10mm

No of threads (n) = 14mm

Pitch thickness (t) = 0.75mm

5.3.2 Direct Compressive stress due to an axial load

$$fc\ act = \frac{W}{\left(\frac{\pi}{4}\right) \times dc^2} \quad (4)$$

$$fc\ act = \frac{240}{\left(\frac{\pi}{4}\right) \times 8.5^2} \quad (5)$$

Fc_{act} = 4.23 N/mm^2

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

5.4 Design of Nut

5.4.1 Material selection

Designation of Material	Ultimate tensile strength N/mm^2	Yield strength N/mm^2
Phospher bronze	400	210

Factor Of Safety = 3 - 5

$$Fb = \frac{W}{\left(\frac{\pi}{4}\right) \times (do^2 - dc^2)\, n} \qquad (6)$$

$$n = \frac{240}{\left(\frac{\pi}{4}\right) \times (12^2 - 10.75^2)\, 17} \qquad (7)$$

n = 8.3= 9

Ln = n x p= 9 x 1.75 = 27mm

Considering length of nut = 30 mm

5.4.1Shear stress due to axial load.

$$fs\ nut\ (act) = \frac{W}{\pi \times dc \times \left(\frac{Ln}{p}\right)} \qquad (9)$$

$$fs\ nut\ (act) = \frac{988.75}{\pi \times 10.75 \times 9} \qquad (10)$$

$fs_{nut\ (act)} = 8.099\ N/mm^2$

As $fs\ nut_{act} < fs_{all}$, the nut is safe in shear.

5.5 Design of Spring Design of spring

Maximum load (W) = 240 N

Deflection = 25 mm

Spring index =20

Material of spring; EN48D

Maximum permissible stress = 620 N/mm^2

Modulus of rigidity = 84 KN/mm^2

Whals correction factor =K_w

$$K_w = \frac{4c - 1}{4c - 4} + \frac{0.615}{c} = 1.07 \qquad (11)$$

Maximum stress induced in spring

$$F_s = \frac{K_w \times 8W}{\pi \times d^2} \qquad (12)$$

$$620 = \frac{1.07 \times 8 \times 988.75}{\pi \times d^2} \qquad (13)$$

d= 1.45mm

Take standard wire having diameter = 1.5mm

Mean diameter of coin = 30 mm

Outer diameter of coil = D+d =30+1.5 =31.5 mm

No of turns of coil

$$\delta = \frac{8W C^3 n}{G \times d} \qquad (14)$$

$$n = \frac{84 \times 10^3 \times 20 \times 75}{8 \times 1250 \times 5^3} \qquad (15)$$

n = 12 turns

For N = n + 2

Squared and ground ends, the total number of turns

N = 14 turns [8]

5.6 Design of Cylinder

5.6.1 Hoope's stress:-
Maximum pressure induced in system due to steam (P) = 3 bar
Diameter of Piston = 95 mm
Thickness of Piston = 1 mm [6]

$$fc\ act = \frac{P \times d}{2t} \qquad (16)$$

$$fc\ act = \frac{0.3 \times 95}{2 \times 1} \qquad (17)$$

Fc_{act} = 14.25N/mm^2
As $fc_h < fc_{all}$; Cylinder is safe

5.6.2 Longitudinal stress:-

$$fcl = \frac{P \times d}{4t} \qquad (18)$$

$$fcl = \frac{0.3 \times 95}{4 \times 1} \qquad (19)$$

fcl_{act} = 7.125 N/mm^2
As $fc_l < fc_{all}$; design is safe

5.7 Design of Bearing

5.7.1 Material selection

ISI NO.	BASIC DESIGN NO.	Basic capacity	
		C	C_g
15ACo2	6002	2550	4400

P = X Fr + Yfa.

In our case;
Radial load FR =240N
a = ball bearing constant = 3

$$L_{10} = \frac{L_{10h} \times 60 \times n}{10^6} \qquad (20)$$

$$L_{10} = 240\ mrv.$$

$$L_{10} = \left(\frac{C}{P}\right)^a \qquad (21)$$

C = 873.8 N So, Bearing is saf

V. EXPERIMENTAL SET UP

For the two different conditions the piston inside the cylinder is actuated with the help electrically powered motor. The actual experimentation is divided in two different phases out of which first one is normal condition and in second phase piston is moved 5mm in upward direction

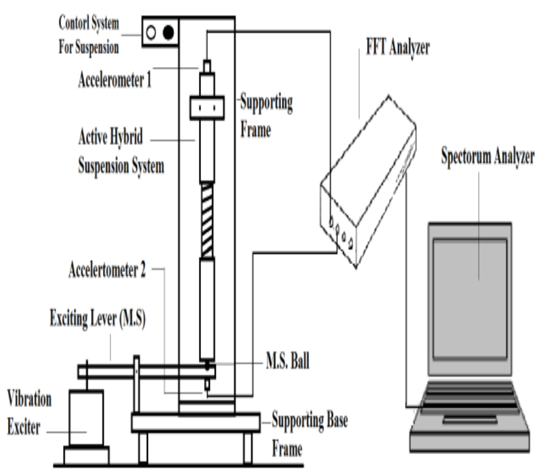


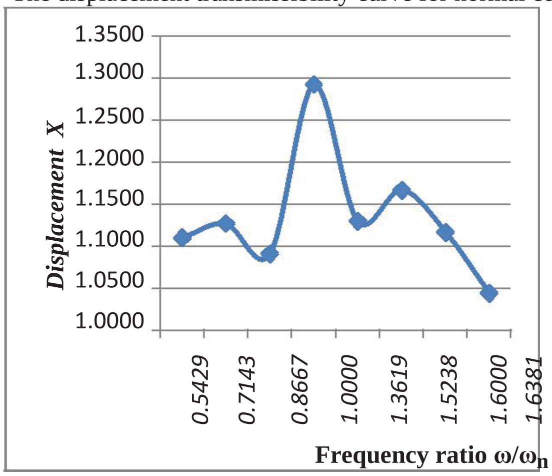
Fig 5 Experimental Set up

VI. RESULT & DISCUSSION

Case I

Piston is at its normal condition inside the cylinder

The displacement transmissibility curve for normal condition is shown in Graph 1



Graph 1 Displacement ratio vs. Frequency ratio

From above graph 1 we get the displacement transmissibility for various excitation frequencies.

Damping factor for case I

Damping factor was calculated by using quality factor and half power point method

Frequency ratio ω/ω_n

Graph 2 Displacement ratios vs. Frequency ratio

We know that damping factor is given by

$$\xi = \frac{\omega_{n2} - \omega_{n1}}{2 \times \omega_n}$$

From above equitation we get

$$\xi = 0.037894$$

From above value it's cleared that system is satisfies the under damping condition.

Case II

Piston is moved in upward direction from normal condition by 5 mm

The displacement transmissibility curve for normal condition is shown in Graph 2

1.35
1.3
1.25
1.2
1.15
1.1
1.05
1

Displacement X

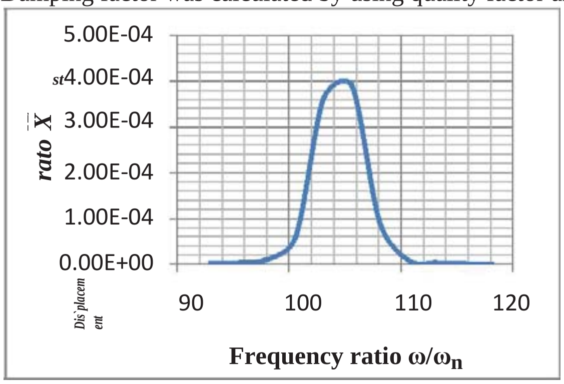
Frequency ratio ω/ω_n

Graph 3 Displacement ratio vs Frequency ratio

From above graph3 we get the displacement transmissibility for various excitation frequencies

Damping factor for case II

Damping factor was calculated by using quality factor and half power point method



Graph 4 Displacement ratios vs. Frequency ratio

$$\xi = 0.023684$$

From above value it's cleared that system is satisfies the under damping condition.Form the both case which are experimented, it's cleared that damping factor changes with respected to the height of piston inside cylinder which was adjusted by using the power screw and electrical motor.

VII. CONCLUSION

In this paper, in ordinary suspension system different problems are occurred like oil leakage, damages to chassis in deep holes and pits. To avoid these electrically power hybrid suspension system was design, the behavior of an Electrically Powered Active Hybrid Suspension System for vehicle was investigated through with design . Some of the significant factors were examined experimentally. This work supports conclusions

1. The dampening factor was less than 1, so it satisfies the condition of under damping.
2. The coefficient of damping changes with respect to height of piston inside the cylinder which is adjusted by using power screw and motor.
3. So the Electrically Powered Active Hybrid Suspension System has ability to change according to road condition.
4. Form result it is concluded that Electrically Powered Active Hybrid Suspension System is effectively used on road and off road conditions.

Abbreviations

P = power output of motor
N = speed of motor
T= Torque transmitted by motor & screw
μ = coefficient of friction of screw
d_o = Major diameter of screw
d_c = Minor diameter of screw
d = Mean diameter of screw
p = Pitch of screw
$\emptyset$ = Friction Angle of screw & nut
α = Helix angle of screw
P_b = Bearing pressure Between Screw & nut
W = Axial load acting on screw
t = Tooth thickness of thread
f_{act} = Direct compressive stress of screw
f_{ast} = Torsional shear stress of screw
n = No of thread in screw
l = Length of nut
f_{all} = Allowable compressive stress in screw

$f_{s\,all}$ = Allowable shear stress in screw
S_{yt} = Yield strength of material
S_{ut} = Ultimate strength of material
N_f = Factor of safety of material
$\sigma_{c\,act}$ = Compressive stress in piston rod
$\tau_{s\,act}$ = Shear stress in piston rod
A_s =Area of screw
a = Rankin's constant
k = Radius of Gyration
f_c = Yield stress in compression
T_d = Torque transmitted by shaft
f_s = torsional shear stress in shaft
L_{10} = Rated bearing life in million revolutions
L_{h10} = Rating life of bearing in million revolutions
n = Bearing speed in rpm
P = Equivalent dynamic load
X = Radial load constant
F_r = Radial load
Y= Yield Load constant
F_a = Axial load
C_d = Dynamic load constant
$P_{b\,nut}$ = bearing Pressure in Nut
l_n = Length of nut
$f_{s\,nut}$ = Shear stress in nut
C= Spring Index
δ = Deflection Of spring
W = Maximum load acting on spring
τ_s = maximum permissible stress in spring
G = modulus of rigidity of spring material
K_w = Whals correction factor
d = spring wire diameter
D = Mean diameter of coil
n = Number of turn in required in spring
N = Total number of turn in spring
σ_h = Hoops stress in cylinder
σ_c = Longitudinal stress in cylinder
d = diameter of piston

FUTURE SCOPE

The present set up is electrically operated, hence this limits the load carrying ability of the system, and within appropriate Sliding of screw we can reduce the fluctuation thereby improving the load carrying ability. The present system has an electric motor nit & screw thereby the need of external lubrication may arise; with appropriate modifications we can make arrangement to give lubrication oil Nut & Screw. By employing some improved processes and fine workmanship in manufacturing we can further achieve improved quality of parts and which may affect the results in positive manner.

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