Selvan et al. 2014. *Int. J. Vehicle Structures & Systems*, 6(3), 76-81
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.6.3.06

International Journal of Vehicle Structures & Systems
Available online at www.maftree.org/eja

Investigation of Vehicle Parameters in Regenerative Coil Suspension Design

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ABSTRACT:

Regenerative type suspensions are employed to recover energy that is unproductively dissipated in dampers in automotive suspension system. In order to effectively recover energy from suspension, an attempt has been made in the design to capitalize the vehicle parameters and concoct a suitable linear generator in unison with helical spring. The performance characteristics drawing suitable inferences between the vehicle parameters and suspension specifications are well investigated.

KEYWORDS:

Shock absorber; Regenerative suspension design; Linear generator; Helical spring; Vehicle parameters

CITATION:

T.A. Selvan, S. Navin and K.R.R. Ganesh. 2014. Investigation of Vehicle Parameters in Regenerative Coil Suspension Design, *Int. J. Vehicle Structures & Systems*, 6(3), 76-81. doi:10.4273/ijvss.6.3.06.

NOMENCLATURE:

M	*Mass of vehicle with driver*
M_{wdr}	*Weight distribution ratio*
M_F, M_R	*Front and rear weight ratio*
m	*Mass of one wheel*
M_s	*Sprung mass on one wheel*
R_m	*Motion ratio*
W_R	*Wheel rate*
K	*Suspension spring stiffness*
θ	*Spring angle*
F_s	*Suspension frequency*
F_w	*Force on wheel*
g	*Acceleration due to gravity*
W_{mf}	*Wheel deflection at ride height*
S_{def}	*Deflection of shock absorber*
R_d	*Damping coefficient*
C_{adf}	*Actual Damping Force*
δ	*Deflection of spring*
P	*Load on spring*
F_V, F_L	*Vertical and lateral components of road force*
F_R	*Resultant of road forces*
h_{cg}	*Height of Centre of gravity from ground level*
b	*Wheel base*
β	*Angle of maximum repercussion on bump*
W	*Total weight of the vehicle*
β_1, β_2	*Ackerman and turn angles*
μ_r	*Coefficient of friction between road and wheel*
e	*Electromotive force (EMF)*
N	*Number of turns*
A	*Area enclosed by single turn of coil*
dB	*Change in magnetic induction*
ψ	*Angle between the axis of magnet and the plane enclosed by the coil*
B	*Magnetic induction*
B_1, B_2	*Max. and min. magnetic induction per turn*
t	*Time period*
U	*Potential energy stored in the spring*
L_g	*Permissible length of generator*
L_{magnet}	*Max. length of magnet stack*
D_{magnet}	*Diameter of magnet*
μ	*Magnetic reluctance*

1. Introduction

The irregularities and undulations of roads have ill-effects on vehicle chassis, ride-comfort and steering. These havocs, on the account of not being balanced tend to aggravate noise, vibrations and harshness levels. The functional characteristics of suspension system are to absorb the vibrations and damp the forces. Conventionally damping is done by dissipating the vibration energy as heat. Recent researches emphasize in capitulating this loss of vibration energy into useful potential or electrical energy. Thereby this conversion enhances the volumetric efficiency and damping action. Pei [1] revealed that the harvesting of vibration energy can ameliorate fuel efficiency by 10%. Mechanical regenerative technology is yester years' commonly adopted technique in which the kinetic energy is converted into potential energy, which is stored in the accumulator. The repercussions of this system would be adverse when the working fluid is heated up to its viscosity loss. The implication would worsen in the event of any leakage or rupture in the hoses.

Electromagnetic regenerative suspension system converts vibration energy into electrical energy with the aid of permanent magnet motors. This system offers great viability to vary damping force by varying shunt resistance. This system is further classified on the grounds of structural configuration into five different types. In type (1), direct-drive electromagnetic regenerative suspension converts mechanical energy to electrical energy using linear permanent magnet motors. Okada et al. [2] primarily investigated by employing linear motors in active and regenerative vibration control. In type (2), ball screw electromagnetic regenerative suspension system converts linear energy due to vibration movement into rotary movement using ball-screw arrangement. Further, the rotational energy is converted into electrical energy using motor. Arsem [3] patented an electric shock absorber that harvests vibration energy using ball-screw. Suda et al. [7-8] studied the efficiency of regenerative ball-screw type electromagnetic suspension with planetary gear to reduce or increase the rotation speed. This study culminated in productive results.

In type (3), electromagnetic suspension using rack-pinion mechanism converts linear to rotary motion and thereby rotating the motor, as a result of which electrical energy is produced. Suda et al. [4] and Beno et al. [5-6] independently investigated vehicle using electromagnetic suspension with rack-pinion mechanism. This study resulted in significant enhancement of limit speed and handling performance of the vehicle. In type (4), hydraulic transmission electromagnetic regenerative suspension system could be regarded akin to a double acting cylinder. The vertical movement causes the piston to reciprocate in the fluid filled cylinder, thus imparting kinetic energy to the fluid. The fluid in turn circulates through hoses into a chamber where gear pump is coupled to a motor. Genshock technology from Levant Power Corp [9] uses this type of regenerative suspension system. In type (5), self-powered magneto-rheological (MR) dampers strike a notch in balancing the regenerative principle and active suspension aspects. MR dampers offer high dynamic range and sturdiness. This research was under-scored by Choi et al [10] and Bogdan [11]. Shojaei et al. [12] have theoretically and experimentally investigated a quarter car suspension model with MR damper. The heuristic approach was fathomed by employing fuzzy logic, skyhook and on-off control techniques in conjunction with a Heaviside step function. This effort culminates as a threshold study in the métier of MR dampers in automobiles.

Zhang et al. [13] discussed that the regenerative suspension designs and researches to date lack fine balance of vibration control and regulation. They reiterate the fact that there is a solid vacuum in vibration control and improving riding comfort. Moreover, they aptly point out that the energy harvested in most cases exceeds the energy consumed by the system itself. Efficient regenerative system such as MR regenerative system is a costlier affair. Prabhu et al. [18] have theoretically and experimentally studied the response of hybrid magnetic vehicle system with an objective to produce vehicle body isolation. The research culminated in a result which emphasizes that amplitude of vibration of the sprung mass using active suspension is reduced from the value in the corresponding passive system. Moreover replacing conventional spring type shock absorber suspension system with electromagnetic regenerative suspension comprehensibly has apparently showed its ill impacts. The repercussions were reduced levels of comfort and steering. This can be attributed to lack of instantaneous absorption of dynamic forces, time lag in system response, discrete operation due to absence of spring. Also researches pertaining to regenerative suspension design in tandem with vehicle parameters have not been culminated to date to aid effective implementation of regenerative design to shock absorbers.

Having considered the limitations of existing models, we extend our investigation to design an effective regenerative suspension which eliminates the discrete operation and facilitates instantaneous absorption of road loads. The design execution ability is enhanced by duly encompassing the vehicle parameters.

2. Regenerative coil suspension design

The regenerative suspension design is to investigate and analyze the possibilities and implications in executing an effective design of regenerative shock absorber in vehicles. It is a consummate evaluation of vehicle parameters for effective result. In this work, a helical spring linear generator system connected in parallel unison is used. The helical spring eliminates the time lag in response to dynamic loads by absorbing road forces instantaneously and it also reverts back slowly after damping. This proves to be a cost effective model in working efficiency in comparison with MR dampers.

The regenerative coil suspension design comprises helical spring which engulfs a linear generator connected in parallel. Figs. 1 and 2 show the isometric and cut-section view of the proposed design respectively. The material selected for electromagnetic coil and spring is Copper and SAE5160 steel respectively. The yield strength of spring is 669 MPa. Considering high magnetic density (up to 1.21 Tesla) and availability, rare earth permanent Neodymium (NdFeB) magnets are selected for the proposed regenerative suspension. They are mounted onto stainless rod for its high reluctance of (μ=1.1x10^{-6} H/m). The magnets are arranged in S-N-N-S fashion to increase flux density [16]. High density steel is selected for the mounting points.

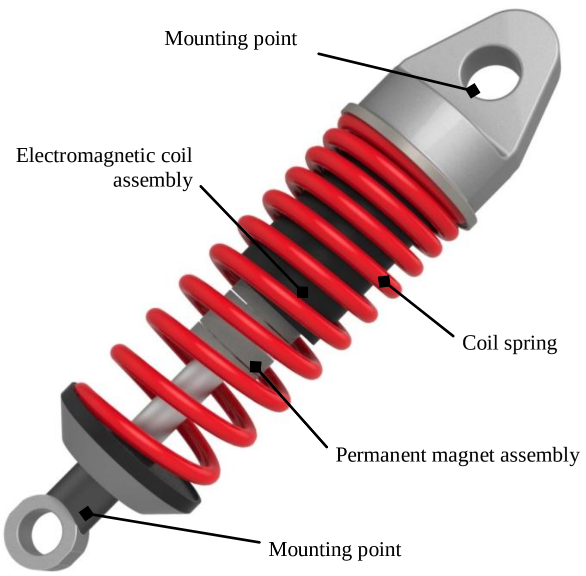

Fig. 1: Isometric view of proposed electromagnetic regenerative suspension shock absorber

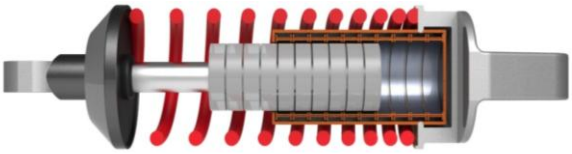

Fig. 2: Cut-section view of proposed electromagnetic regenerative suspension shock absorber

The dimensions of the helical spring and linear generator are arrived by taking into account of the characteristics of the vehicle. The frequency of suspension, $F_{s,}$ is given by [21],

$$F_s = \frac{1}{2\pi}\sqrt{\frac{W_R}{M_S}} \tag{1}$$

Where W_R is the actual rate of a spring acting at tire contact path and is given by [20],

$$W_R = R_m^2 K \cos\theta \tag{2}$$

Where $R_m = d_1/d_2$ is the motion ratio [17] as illustrated in Fig. 3. K is the spring rate. θ is the angle with respect to vertical axis of shock absorber as shown in Fig. 3. The sprung mass, M_S, on one wheel is calculated using,

$$M_s = \left(\frac{M * M_{wdr}}{2*100}\right) - m \tag{3}$$

Where M and m are the total mass of vehicle with driver and the mass of one wheel respectively. M_{wdr} is the weight distribution ration between front and rear wheels. The deflection of shock absorber, S_{def}, is related to the wheel deflection at ride height [17] as,

$$S_{def} = \frac{M_F g}{W_R} * R_m \tag{4}$$

Where M_F is the mass of front wheel and g is acceleration due to gravity. The actual damping factor of the shock absorber, C_{adf}, is calculated using [17],

$$C_{adf} = R_d 2\sqrt{W_R * M_S} \tag{5}$$

Where R_d is damping coefficient.

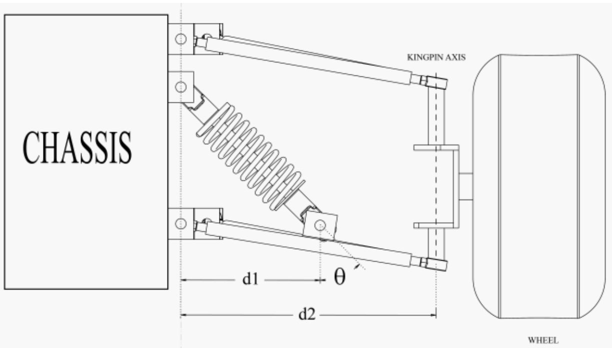

Fig. 3: Arrangement of shock absorber and motion ratio

In order to size the helical spring, the loads from selected bump road conditions during normal manoeuvre and corner turning as shown in Figs. 4 and 5 are considered. The vertical and lateral components of vehicle hitting a bump while braking are given by,

$$F_V = 1.5\left[3F_w\left(1+\left(\frac{h_{cg}}{2b}\right)\tan\beta + \frac{Mgh_{cg}}{2b}\right)\right] \tag{6}$$

$$F_L = 0 \tag{7}$$

Where F_w is the force acting on the front wheel. $h_{cg}$is height of centre of gravity from ground level. b is the wheel base. β is the angle of maximum repercussion on bump. The vertical and lateral components of vehicle cornering are given by,

$$F_V = 1.5W\left[\frac{\cos\beta_1}{\sin\beta_2} + \frac{\mu_r(h_{cg})\cos\beta_2}{2b}\right] \tag{8}$$

$$F_L = 1.5\mu_r Mg\cos\beta_1 \tag{9}$$

Where β_1 and β_2 are Ackerman and turn angle respectively. μ_r is the coefficient of friction between road and wheel.

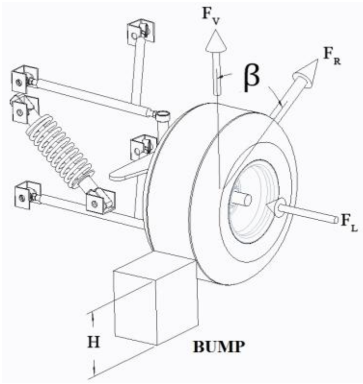

Fig. 4: Loads due to vehicle hitting a bump

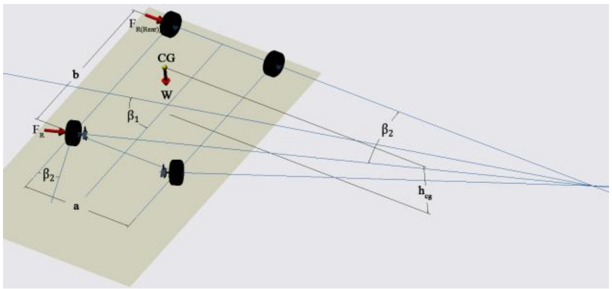

Fig. 5: Loads due to vehicle cornering

The vertical component of vehicle hitting a bump while accelerating and while cornering respectively are given by,

$$F_V = 4.5F_w\tan\beta - 0.75Mg \tag{10}$$

$$F_V = 4.5F_w\tan\beta \tag{11}$$

The load on spring is calculated as force on front wheel divided by motion ratio. The obtained loads are compared with the considered bump road cases. In the account of special case loads exceeding the load on spring, the latter is considered for spring design. Standard helical spring design equations [14-15] are used to arrive the dimensions of the spring for a given load and length of the shock absorber. A factor of safety of 2 is assumed for the shear strength of spring material that is taken as half of its yield strength.

By Faraday's law of induction, the induced electromotive force (EMF) is related to the rate of change of magnetic flux, ϕ, using,

$$e = \frac{d\phi}{dt} = \frac{d}{dt}(NBA\cos\psi) = NA\cos\psi\frac{dB}{dt} \tag{12}$$

Where N is the number of coil turns, B is the magnetic induction, A is the area enclosed by the coil and ψ is the orientation of flux. When the magnet retracts away from the vicinity of a single turn of coil, B_2 = 0 and B_1 = B. Hence, the change in magnetic induction is given by,

$$dB = B_1 - B_2 = B \tag{13}$$

The induced EMF opposes the cause that produces it. Hence, as per Lenz law, the EMF becomes,

$$e = -N\,A\,B\,f_s \tag{14}$$

Where f_s = 1/t = 1/dt is the frequency of coil. The EMF produced by the liner generator is equal to the potential energy [22] stored in the spring per unit time as,

$$e = U\,f_s = 0.5P\,\delta\,f_s \tag{15}$$

As 100% damping is not practically possible, the damping coefficient (R_d) is taken into consideration. Hence, the above Eqn. becomes,

$$e = U\,f_s = 0.5P\,\delta\,f_s\,R_d \tag{16}$$

Balancing Eqns. (14) and (16) results in,

$$N\,B = 0.5P\,\delta\,R_d\,/\,A \tag{17}$$

The main dimensions of the linear generator are shown in Fig. 8. To facilitate a smooth and effective reciprocation, the length of magnet is taken as half the length of generator and is given by,

$$L_{magnet} = L_g\,/\,2 = 0.65l\,/\,2 \tag{18}$$

Where l is the length of shock absorber. The diameter of the magnet, D_{magnet}, is taken as 80% of inner diameter of the coil to render necessary clearance.

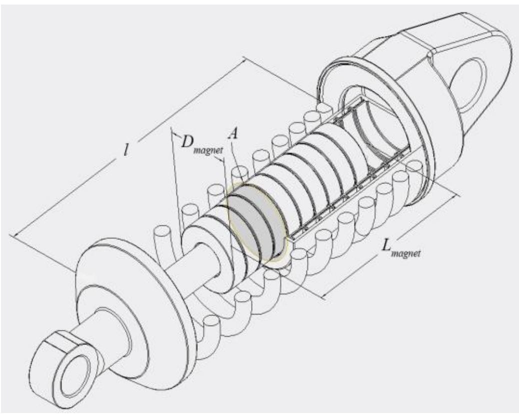

Fig. 6: Main dimensions of the linear generator

3. Results and Discussions

The mechanical variations account for the potential energy storage in the spring. This stored energy is eventually converted into electrical energy. Hence, the EMF is treated as a function of load (P), damping ratio (R_d), deflection (δ) and frequency (f_n) of spring. The variables are treated as independent. The premise of maintaining the parameters as independent variables entitles an explicit function of a single variable while treating the other variables to standard values. The functions are regarded as explicit and theoretically assumed to be devoid of non-linearity.

Figs. 7 to 10 show the variation of EMF generated with respect to the variations in the enclosed area and magnetic induction. The number of coil turns is kept as constant. It is evident from Fig. 7 that the curve has a subliminal slope up to EMF of 30 units and thereafter the slope of the graph is vividly steep. This behaviour is also coherent in the following graphs depicting symbiosis between induced EMF in the coil, enclosed area and magnetic induction. The reason could be attributed to the act of overcoming inertial forces. Fig. 8 reiterates a linear relationship between enclosed area and magnetic induction. Figs. 9 and 10 delineates that the curve is quadratic in nature up to 30 units of EMF and there after the response is linear. Hence, it is desirable to design the linear generator above 30 units of EMF.

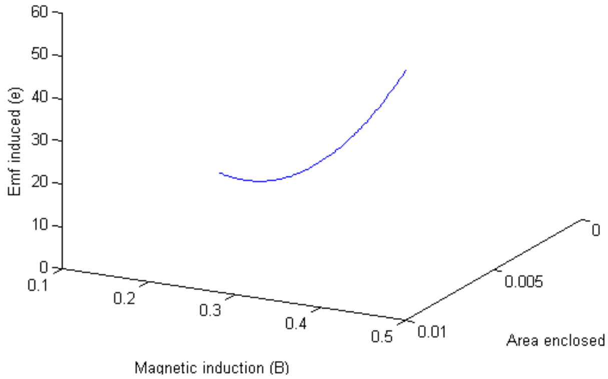

Fig. 7: Induced EMF as a function of magnetic induction and enclosed area

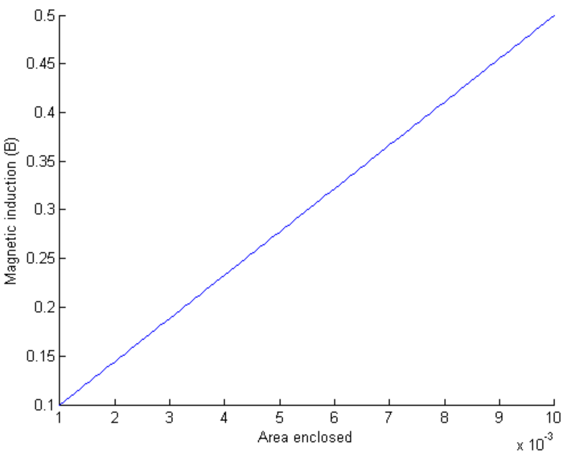

Fig. 8: Magnetic induction vs. Enclosed area

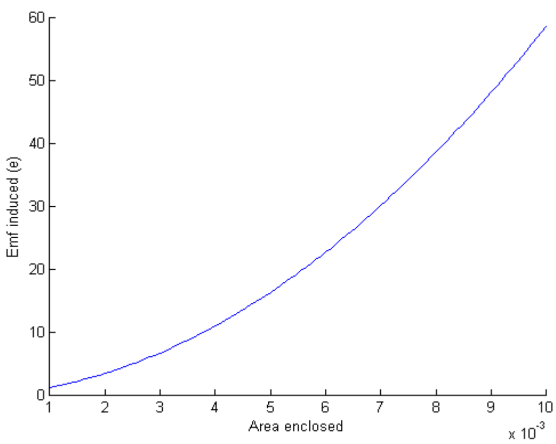

Fig. 9: Induced EMF vs. Enclosed area

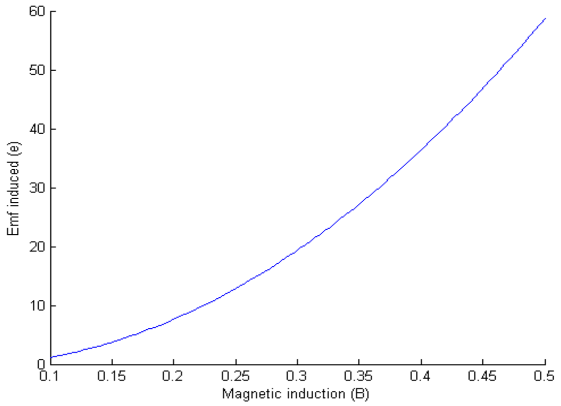

Fig. 10: Induced EMF vs. Magnetic induction

Figs. 11 to 14 indicate a steadily increasing quadratic nature of induced EMF in response to increased frequency rates and load. This behaviour demonstrates an increased production of mechanical EMF at higher loads on wheel. Hence, a possibility is to have more sprung mass. This behaviour is contradictory to conventional systems which require larger dampers at higher sprung mass. Electromagnetic regenerative suspension systems for a small race car weighing less than 200 kg and a hatchback passenger car weighing about 1 ton were designed using formulated model. The input parameters are given in Table 1. In order to counteract road undulations, the spring stiffness was calculated based on a frequency of 1.5 Hz [14-15].

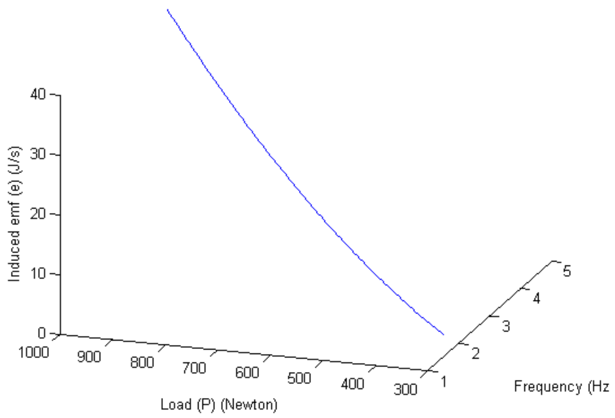

Fig. 11: Induced EMF as a function of load and suspension frequency

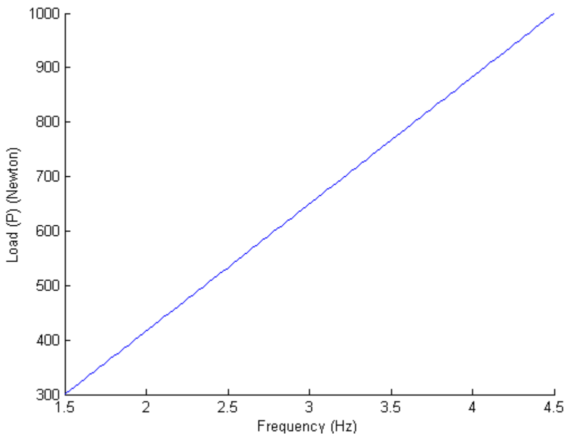

Fig. 12: Load vs. Frequency

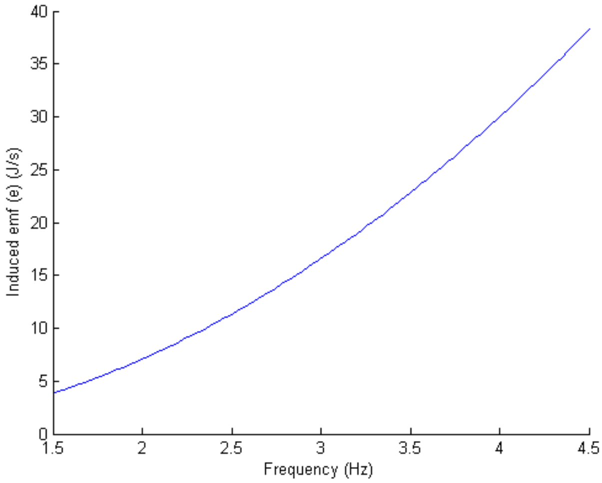

Fig. 13: Induced EMF vs. Frequency

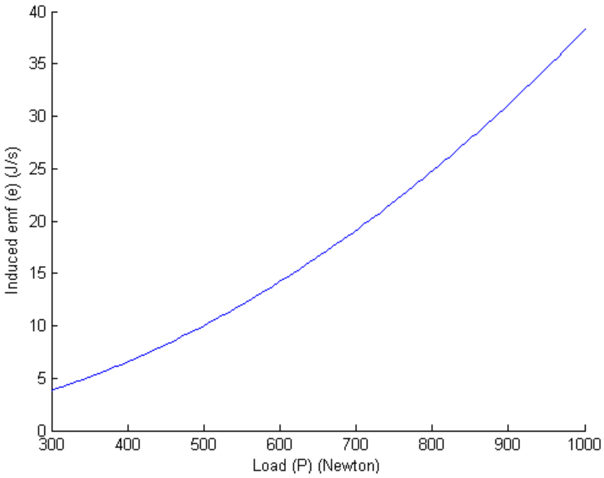

Fig. 14: Induced EMF vs. Load

Table 1: Inputs to the formulated model for suspension design

Parameters	Small race car	Hatchback car
Vehicle mass (kg)	160	1000
Weight ratio	45:55	55:45
Mass of 1 wheel (kg)	8 kg	50 kg
Motion ratio	0.83	0.98
Magnetic strength (Tesla)	0.6	1.4

All the above factors were duly incorporated in deducing the linear generator dimensions for the small race car and hatchback passenger car. Table 2 gives the deduced dimensions and other associated parameters of the linear generator designed suspension systems for both vehicles. The results are also compared with experimental results from Longxin et al. [19]. In their work, the structural parameters of electromagnetic shock absorber were calculated through experiments for 1000 kg car. The optimum internal radius of internal coil windings group was found to be 35.5 mm. Longxin et al. [19] have aptly calculated damping ratio = 0.3 for a car of 1.3 ton. This is comparable with damping ratio of 0.28 inculcated in this work of 1 ton vehicle. Comparison of enclosed area and magnet diameter has demonstrated that the proposed regenerative suspension design model correlates well with the literature [19]. Small discrepancy is attributed to the design of spring-engulfed linear generator.

Table 2: Dimensions and parameters of linear generator

Vehicle type	Small race car	Hatchback car	Longxin et al. [19]
Damping factor (Rd)	-	0.28	0.3
Coil encl. area (mm^2)	1390	3850	3959
Magnet diameter (mm)	40	70	70
Magnetic induction (T)	0.6	1.4	-
Induced EMF (J/s)	13.76	281.2	-
Magnet length (mm)	145	300	-

4. Conclusions

In this paper, an investigation is made on a formulated model of helical spring regenerative automotive suspension. The dimensions of the system were arrived in tandem with all associated vehicle parameters. The performance characteristics were duly interpreted in the form of the graphs. The design dimensions and parameters from the simulation of formulated model were compared with literature and results are found to be in good agreement.

ACKNOWLEDGEMENTS:

Authors thank Mr. Shreyas Prakash for his moral support and constant encouragement.

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