Feasibility of Tuned Mass Damper Approach

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Abstract. His investigation is aimed at exploring the possibility of energy recovery through the suppression of vibrations. The article describes a new type of regenerative suspension using electromagnetic tuned mass damper (TMD) approach. The magnetic part of the device performs the function of the TMD, thereby providing both energy regeneration, and damping properties to the protected structure. Equations of optimal parameters of TMD, such as stiffness and damping coefficient were obtained according to the theory of TMD. Then, optimized TMD was mounted in structure with characteristics typically for car wheel. Therefore, efficiency of using TMD for such structures was revealed by amplitude and acceleration frequency response plot. Equation of velocity of TMD was obtained as well. Further estimated regenerative function of current TMD system was investigated: electromotive force for one coil equals to 0.03V, at 1.13 m/s TMD velocity. Finally, the aims of the further research were determined as well.

1. Introduction

Modern cars equipped with internal combustion engines have a number of disadvantages: low engine efficiency (up to 60%), an ever-increasing fuel price (due to limited reserves), high maintenance costs, and air pollution. In turn, the advantages of using electric motors on cars: no localized carbon-based emission, efficiency of electric drive system (up to 90%), quiet operation, and low maintenance.

All this is the reasons for the increased interest of researchers and the automotive industry to the development of electric cars. However there are two main problems which need to be solved. The first one is battery capacity of electric vehicles. The battery pack of EV is low and they still are not able to accumulate more than 200 Wh/kg of energy compared to liquid fuels (about 12,000 Wh/kg). This means that the fuel tank of conventional internal combustion engine that weights 40kg can store approximately 480kWh of energy whereas modern Li-ion battery that weight 300kg can roughly store 60kWh of electricity [1]. The second one is using electric motors as a usual mover of vehicles will require new public recharging infrastructure to be build.

To solve the problem of battery capacity, scientific research is divided into two main areas: improving the properties of the batteries and developing renewable energy technologies. In the automotive industry, solutions with recuperation systems are the most in demand lately. Systems such as regenerative braking has been implemented to increase the driving range. The latest development is the regenerative shock absorbers. Nowadays, there are few different concepts of regenerative shock absorbers, with its own characteristics and disadvantages [2]. This research provides a new concept of regenerative suspension system. Construction with two degrees of freedom consisting of an
electromagnetic damper, recovering the kinetic energy due to vibrations, and solenoid which transforms it to the electrical energy.

That concept is predicted to have many advantages. The regenerative suspension system is able to provide simultaneously: sufficient damping force and high regenerative efficiency with low energy consumption.

2. Mathematical Model and Optimization

2.1. Optimization
With the aim of obtaining sufficiently accurate and simple equations for the optimum parameters of damping, it is best to typically use a known property of a linear system with one damper. The ordinates of its presence on the frequency response are invariant points and do not depend on the value of damping. At these points the curve of frequency response of the system, corresponding to different values of viscous coefficient, is intersected. Setting and damping of the absorber are determined by minimizing the maximum ordinate of the frequency response of one or the other kinematic parameter of vibrations of the main mass at a given value of the relative weight of the damper. This in turn is prescribed on the basis of ensuring the required level of the chosen quality criterion, subject to the conditions of the strength of the elastic element and limitation on the motion of the damper [4].

Consider the system of Fig.1 in which a dashpot (damping coefficient $F_c=c$) is arranged parallel to the damper spring $k$, between the masses $M$ and $m$. The main spring $K$ remains without dashpot across itself. Newton's law applied to the mass $M$ and $m$ gives by (1).

\[
\begin{align*}
M\ddot{X} + K(X - \eta(t)) + kX - kY + c\dot{X} - c\dot{Y} &= 0 \\
m\ddot{Y} + c\dot{Y} - c\dot{X} + kY - kX &= 0
\end{align*}
\]

(1)

Then both $X$ and $Y$ are harmonic motions of the frequency $\omega$ and can be represented by vectors. The easiest manner of solving these equations is by writing the vectors as complex numbers. Where $X$ and $Y$ are unknown complex numbers, the other quantities being real. Also known as Laplace transformation. Then, in order to solve for system of equation, using Kramer's method. Finally, derive the amplitude of the motion of the main mass $M$, expressed by (2).
Thus we are in a position to calculate the amplitude in all cases. However, the number of variables can be reduced, by expressing (2) in a dimensionless form. After performing some algebra (2) is transformed into (3).

\[ X = \frac{\sqrt{K^2c^2\omega^2+(-Km\omega^2+Kk)^2}}{\sqrt{(-M\omega^3-cm\omega^3+Kc\omega)^2+(Mm\omega^4-Km\omega^2-Mk\omega^2-km\omega^2+Kk)^2}} \] (2)

\[ \frac{X}{\eta_0} = \sqrt{\frac{4c^2g^2f^2+(-f^2+g^2)^2}{c_0^2+(\mu f^2g^2-(g^2-1)(-f^2+g^2))^2}} \] (3)

This is the ratio of the main mass to exciting amplitude as a function of the four essential variables. As was said above, linear system with one damper has invariant points and coordinates of these points do not depend on the value of damping. Therefore, according to this rule it is possible to obtain coordinates of these points and set their location relatively to each other, thus make a curve of amplitude optimized. Detailed route of optimization was explained by Hartog et al [5]. Result of optimization shown in Fig. 2a.

Let us assume that current TMD system is aimed to suppress vibration from wheel of vehicle. So that, initial conditions of protected structure were obtained as follow: \( m = 45kg, K = 4500 \frac{N}{m^2} \). To derive parameters of tuned mass damper, it is essential to obtain firstly the value of the mass of TMD. Using optimal parameters of TMD (k and c) in an alphabetic way, express by mass ratio, it is possible to derive the value of amplitude of PS and TMD depending on only mass ratio coefficient. So that, plot of TMD and protected mass amplitude depend on mass ratio shown in Fig. 2b. According to Fig. 3a, to improve the damping efficiency of protected structure it needs to increase mass of TMD, but from the other hand it is able to lead to excess weight of vehicle. Because of common passenger car absorber mass has up to 20% of weight of car wheel. Moreover, as shown in Fig. 3b, increasing of TMD mass leads to reduce velocity of TMD, which in turn encourages regenerative decrease. Also, it needs to take into consideration design feature. Displacement of TMD more than 5 times bigger than excitation from road, hard to implement in current system. According to given above, more suitable parameter of mass ratio, for that particularly case, is 0.2. In that case current system has the mass almost the same with the mass of common vehicle shock absorber, sufficient damping and also is able to generate electricity energy by
given velocity of TMD. In this section, method of deriving optimal parameters, such as k, c, and m of TMD was described. The real oscillation system with car wheel and maintained TMD with optimal parameters will be considered in next section.

2.2. Efficiency of vibration absorption when applying TMD

To simplify solving problem of optimal parameters for TMD, in the previous section damping coefficient of protected structure was not taken into account, which allows deriving analytical equations for optimal parameters of TMD [6]. Consider the system with tuned mass damper, mounted on protected structure with stiffness K and damping coefficient C, as shown in Fig.1. Let us assume that current TMD system is aimed to suppress vibration from wheel of vehicle. So that, initial conditions of protected structure and optimal parameters of TMD were obtained. Then derive amplitude and acceleration frequency response of PS and analyse it with the different values of damping coefficient of PS, with the aim of obtaining the most efficiency operating mode for tuned mass damping, as damping device. Then, analyse plot of velocity of TMD and PS, with the aim of reveal regenerative efficiency of TMD. According to obtained results, shown in Fig.4, the efficiency of damping ability of TMD decrease with increasing of damping coefficient of protected structure. As seen in plot of amplitude frequency response: at low damping coefficient C=0.5 efficiency of TMD reached almost maximum value, around 99.66%, that mean that amplitude of PS reduces from 18m to 0.069m by the TMD kinetic energy. However, with the increasing damping coefficient on 99.5% from 0.5 to 100, efficiency of the TMD decreases from 99.66% to 51.3%, or from 0.069m to 0.044m. This trend continues with the further coefficient and with the plot of velocity and acceleration of PS. Decreasing of velocity of PS leads to decreasing of velocity of TMD, therefore reduce regenerative ability of suspension. In that case, the best condition of TMD operating is to protect structure with low damping coefficient. The vehicle wheel structure can be counted as structure with low damping coefficient, so that implementation TMD on car wheel would get both: sufficient damping and regeneration of electrical energy. Let us consider that kind of structure with the damping coefficient C=0.5. In that particularly case, velocity of TMD is able to reached pick value at 1.13 m/s at frequency 8.5 Hz, which is closed to resonant frequency of protected structure. In Fig.5a area I, a range of frequency from 2.5 to 11.9 Hz, the TMD works efficiently carrying out both: damping and regeneration ability. According to parameters of TMD for current system, such as peak value of velocity and mass, estimated calculate of energy regeneration will be given in the next section.

2.3. Estimated regenerative ability

Current system has a magnet, with mass m, carrying out tuned mass damper function. As was revealed in previous sections, optimal parameter for tuned damper mass should be around 20% from mass of
protected structure, so that mass of TMD equals to 9 kg. The rare-earth permanent magnet (PM), specifically NdFe30, was chosen due to their high magnetic density and availability. Size parameters of magnet were obtained according to optimal mass for TMD, as shown in Fig.5b, where $B_r$ is a magnetic density of magnet, expressed by Tesla. Electromotive force (EMF) for one coil, which is influenced by magnetic field specified by (4):

$$E_i = \frac{\Delta BS}{\Delta t}$$

(4)

Where $B$ is a magnetic density, $S$ is a square of cross-sectional area of the coil. Ring magnet response of magnetic density and distance from surface of magnet expressed by (5).

$$B = \frac{B_r}{2} \left( \frac{z+h}{\sqrt{(z+h)^2+r^2}} - \frac{z}{\sqrt{2^2+r^2}} - \left( \frac{z+h}{\sqrt{(z+h)^2+r^2}} - \frac{z}{\sqrt{2^2+r^2}} \right) \right)$$

(5)

As was obtained in previous section, movement of TMD (magnet) expressed by (6):

$$z(t) = 0.09 \sin(10t)$$

(6)
This equation due to shows peak value of EMF represents a movement of TMD in resonance condition of protected structure [7]. Substitute \( z(t) \) into equation (5) derive a plot of magnetic density and distance response, as shown in Fig.6a. Then substitute the resulting expression into equation (4), differentiate it with respect to \( t \) and derive plot of peak value of EMF for current system, shown in Fig.6b.

![Figure 6.](image)

**Figure 6.** a) Magnetic density and distance response.  b) Peak value of EMF.

The peak value of EMF for one coil reaches to 0.03V with the frequency of 10 Hz, in order to derive the particular power of regeneration for such system, further research should contain the development of design, control methods and optimization.

3. Conclusion

In this paper the idea of using the tuned mass damper for suppressing vibration from car wheel was introduced. For the system of TMD, with mass 9 kg, movement and velocity equations were obtained. With coefficient of damping \( C=0.5 \) for protected structure and initial excitation with amplitude 0.02m and frequency closed to resonance condition, that system is able to reduce resonance amplitude value for more than 260 times by the kinetic energy of TMD, thus allows to transform vibration energy into electricity energy. In that case, TMD maximum amplitude of moving is 0.136 m and peak velocity is 1.13 m/s. So that, system is able to produce EMF for one coil 0.03V. These results undoubtedly confirm of the feasibility for such method of regeneration and damping for car wheels. The aims of the further research are: to develop optimized design and methods of control, with the particularly value of the power of regeneration for such system, to consider the three degrees of freedom systems for implementation of such method of regeneration and damping for car suspensions.

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