Mechatronic Suspension Design for Vehicle Dynamic Performance Enhancement

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Abstract

This paper investigates the performance advantages of a mechatronic suspension strut for a road vehicle. Mechatronic struts have attracted growing research interest since the interior electrical circuit has the potential to realise the high-order impedances with sensible size and weight to achieve desirable performance. In addition, to deal with different driving conditions (speed and road roughness, for example) it is much more straightforward to adjust electrical element values in real-time compared with changing pure mechanical elements. To obtain the optimal mechatronic suspension design, a methodology is proposed by considering all possible design parameters and constraints such as nominal voltage and current for the motor maximum force acting on the ball-screw mechanism, and size and weight of the electrical elements. Based on a qualitative analysis of a proposed mechatronic strut, it is concluded that in order to achieve improved performance, the ball-screw needs to have small rotational inertia and large bearable force, and the motor should have small inertia, internal resistance, internal inductance and large nominal current and voltage. This qualitative analysis provides brief guidance for ball-screw and motor selection. Case studies are carried out with a quarter car model using two electrical candidate layouts. They show the potential performance advantages of the mechatronic strut using a motor with sufficiently large nominal current.

1 Introduction

Mechatronic suspension struts attract wide attentions with great potential in realising complex networks and performance enhancement. Different realisations of mechatronic strut have been studied recently. In Sijing’s paper [1], two different types of Energy Harvesting Shock Absorbers (EHSAs), one with mechanical motion rectifier (MMR) and one without it, have been compared on different vehicles. It has been shown that MMR-EHSAs provide better rider performances and road handling simultaneously over the traditional shock absorber when installed on light-damped, heavy-duty vehicles. The EHSAs translate the up-and-down suspension vibration into a bidirectional rotation of the generator using a rack-pinion mechanism. A ball-screw mechanism can also be applied to translate linear motion to rotational motion in mechatronic struts, which is presented in [2]. Furthermore, a prototype of an electro-hydraulic semi-active damper was designed and manufactured, and a series of experimental tests were conducted to demonstrate its effectiveness to generate the damping characteristic for enhanced performance in Yuxin’s paper [3]. Apart from rotary motors, linear motor is also employed in the mechatronic strut. For example, in Songye’s paper [4], a novel application of electromagnetic damper using a linear motor is studied on structural vibration control, and it presents a theoretical and experimental study of linear electromagnetic dampers connected with four representative circuits. A good match between the modelling and testing results clearly demonstrates that the described model can predict the performance of the linear electromagnetic damper very well.
In mechatronic struts, rotary motors are commonly used with a mechanism to translate linear motion to translational such as ball-screw and rack-and-pinion. Rotary motors and these mechanisms bring inertia effects to the strut, which is considered and can be realised as an inerter in a mechanical network. Since the inerter [5] was proposed to substitute the mass element in the mechanical-electrical analogy, the abundant electrical theorems can be directly applied to the analysis and synthesis of mechanical networks which can be used in designing mechanical systems to achieve enhanced performance. The more high-order passive impedance, the further enhanced system performance is obtained [6]. In this analogy, the physical characteristics of inerters, dampers and springs correspond to the electrical characteristics of capacitors, resistors and inductors, respectively. The mechatronic strut, combined of mechanical and electrical networks, can be translated to an equivalent mechanical network using this analogy. The synthesis of a high-order pure mechanical system is difficult due to the limitation on the size and weight of the system. Therefore, in [2], a high-order passive impedance can be easily realised by a basic mechanical network with a high-order electrical network, where a prototype of the mechatronic strut was proposed and applied to vehicle suspension systems. Although considerable research effects have been made, some challenges still exist which hinder the large scale application of mechatronic vibration suppression devices. Firstly, a systematic optimum design approach has not been established to identify the optimum designs for both the mechanical and electrical parts, in terms of the topology and element values. Secondly, realistic considerations, such as weight, size and robustness of the whole system, have not been fully explored.

In this paper, a design methodology for mechatronic struts is presented. Based on qualitative analysis, some guidelines for ball-screw and motor selection are drawn. Case studies have shown that the mechatronic strut has a potential to achieve enhanced performance for road vehicles. This paper is arranged as follows: Section 2 introduces the principles of the mechatronic strut and represents its corresponding mechanical network and admittance function. In Section 3, the design methodology of suspension systems is presented. Furthermore, the qualitative analysis of the mechatronic strut is carried out. Section 4 presents the quarter-car model and the case studies of the mechatronic strut for different motors and suspension networks. In Section 5, some conclusions are drawn.

2 The mechatronic strut

A schematic diagram of the mechatronic strut is shown in Figure 1. The proposed mechatronic strut consists of a ball-screw mechanism, a permanent magnetic electrical motor (PMEM) and an external electrical circuit. The relative linear motion between the two terminals of the strut results in the rotational motion of

![Figure 1: The schematic diagram of the mechatronic strut system](image-url)
the ball-screw mechanism, which in turn drives the motor to generate an inductive voltage. To provide desirable performance, a suitable external electrical circuit is connected to the motor to generate corresponding torque in the rotor. The torque acts back to the ball-screw mechanism, and hence change the force-velocity behaviour across terminals 1 and 2. Furthermore, the external electrical circuit is connected to the PMEM terminals.

The working principle and electrical model of a PMEM is introduced, and then the principle of a ball-screw mechanism is presented. In Section 2.3, the admittance and equivalent mechanical network of the mechatronic strut is introduced.

2.1 The permanent magnet electric motor

In a PMEM, the armature can be modelled as a resistor $R_m$ and inductor $L_m$ in series in electrical circuits. $R_m$ is the coil resistance of the armature and the inductance $L_m$ for the armature is produced once the armature rotates. Figure 2 presents the schematic electrical circuit diagram of a PMEM with an external electrical circuit connected.

![Schematic electrical circuit diagram](image)

Figure 2: The schematic electrical circuit diagram for a PMEM model connected with an external electrical circuit.

The inductive voltage $V$ and inductive torque $T_m$ are generated by the motor simultaneously while the armature rotates. If the armature rotates with an angular velocity $\omega$, the inductive voltage $V$ can be represented as:

$$ V = k_e \cdot \omega, \quad (1) $$

where $k_e$ is the voltage constant (Vs/rad). The armature current $I$ can then be calculated from the inductive voltage $V$ and electrical admittance $Y_e(s)$, with the latter determined by the external electrical circuit and armature model. The expressions of $I$ in the Laplace domain and $Y_e(s)$ are given as follows:

$$ \tilde{I} = Y_e(s) \cdot \tilde{V}, \quad (2) $$

$$ Y_e(s) = \frac{\tilde{I}}{\tilde{V}} = \frac{1}{R_m + s \cdot L_m + Z_{ex}(s)}, \quad (3) $$

where $Z_{ex}(s)$ is the external circuit impedance. Then the inductive torque $T_m$ is calculated by the following equation:
\[ T_m = k_t \cdot I, \]  
(4)

where \( k_t \) is the motor torque constant with unit N.m/A. The inductive voltage constant \( k_e \) and inductive torque constant \( k_t \) are identical and are determined by the magnetic flux, the number of the armature coils and poles of a PMEM. Therefore, the dynamics of the motor in rotational direction can be represented as:

\[ T_E = J_m \frac{d\omega}{dt} + T_m + B_m \omega, \]  
(5)

where \( T_E \) is the total torque generated by the motor and \( J_m \) the moment of inertia for the rotor. \( B_m \) is the damping coefficient of the motor strut.

### 2.2 Ball-screw mechanism

A ball-screw mechanism is used to convert linear motions into rotational motions to drive the motor. The relation between the force \( F_M \) on the ball-screw and the torque \( T_M \) is shown as follows:

\[ F_M = (\frac{2\pi}{P}) T_M, \]  
(6)

where \( P \) is the pitch of a ball-screw mechanism. The relation between the angular velocity \( \omega \) on the ball-screw and the linear velocity \( v \) is presented as follows:

\[ \omega = (\frac{2\pi}{P}) v, \]  
(7)

If the ball-screw mechanism is ideal, for example, by ignoring friction and backlash, the torque \( T_M \) can be expressed as follows:

\[ T_M = J_T \frac{d\omega}{dt}, \]  
(8)

where \( J_T \) is the total moment of inertia of the nut and the flywheel in a ball-screw mechanism, which can produce an inerter effect for the strut. Substituting Equations 7 and 8 into 6, the force \( F_M \) is given as:

\[ F_M = (\frac{2\pi}{P})^2 J_T \frac{dv}{dt} = b \cdot a, \]  
(9)

where \( b = (\frac{2\pi}{P})^2 J_T \) is the ball-screw inerter and \( a \) the relative acceleration across its two terminals. The total ball-screw mechanism inerter \( b \) can be easily adjusted by tuning the flywheel.

### 2.3 Mechanical admittance of the mechatronic strut

To determine the admittance and equivalent mechanical network of the mechatronic strut, the dynamics of the motor in rotational direction is transferred to translational direction. As the torque \( T_E \) is acting through the ball-screw mechanism to the system, the dynamics of the motor in translation direction can be written by substituting Equation 7 and \( F = \frac{2\pi}{P} T \) into Equation 5 and simplified as follows:

\[ F_E = \frac{2\pi}{P} T_E = b_e \cdot a + F_m + (\frac{2\pi}{P})^2 B_m v, \]  
(10)

where \( F_E \) is the force produced by the motor, the inerter, \( b_e \), generated by the rotor equals \( (\frac{2\pi}{P})^2 J_m \), and the inductive force, \( F_m \), equals \( \frac{2\pi}{P} T_m \). The total force \( F \) generated by the mechatronic strut is the sum of mechanical force \( F_M \) and electrical force \( F_E \) as given by:

\[ F = F_M + F_E. \]  
(11)
Substituting Equations 9 and 10 into 11, the dynamics equation of the mechatronic strut can be rewritten as follows:

$$F = (b_e + b) \cdot \frac{dv}{dt} + F_e + \left(\frac{2\pi}{P}\right)^2 \cdot B_m \cdot v$$  \hspace{1cm} (12)$$

Substituting Equation 1 - 4 and 6 - 7 into Equation 12 to obtain the admittance $Y_{ms}$ of the mechatronic strut:

$$Y_{ms} = \frac{F(s)}{v(s)} = (b_e + b)s + \left(\frac{2\pi}{P}\right)^2 B_m + \frac{(\frac{2\pi}{P})^2 k_e k_t}{R_m + s \cdot L_m + Z_{ex}(s)}  \hspace{1cm} (13)$$

For simplification, the following parameters are defined:

$$\begin{cases}
    c_e = \left(\frac{2\pi}{P}\right)^2 B_m \\
    G_m = (\frac{2\pi}{P})^2 k_t k_e
\end{cases}$$

Therefore, Equation 13 can be rewritten as follows:

$$Y_{ms} = (b_e + b)s + c_e + \frac{G_m}{R_m + sL_m + Z_{ex}(s)}  \hspace{1cm} (14)$$

From the admittance $Y_{ms}$, the mechatronic strut can be divided into two parts: the first part $(b_e + b)s + c_e$ is considered as the mechanical inerter and damper of the motor structure, and $G_m/(R_m + sL_m + Z_{ex}(s))$ is regarded as the admittance of an external electrical circuit and motor. Using the analogy of mechanical and electrical systems, the equivalent mechanical network of the strut is presented in Figure 3, where $k_m$, $c_m$ and $G_m/Z_{ex}$ represent the corresponding internal inductor, internal resistor and external circuit, respectively. The corresponding relationships between electrical and mechanical elements are presented as follows:

$$\begin{cases}
    k_m = \frac{G_m}{L_m} \\
    c_m = \frac{G_m}{R_m}
\end{cases}$$

![Figure 3: Simplified equivalent network for the mechatronic strut](image)
3 Suspension design

3.1 Optimisation methodology

In the optimisation methodology, design parameters and physical constraints for the ball-screw mechanism, electrical motor and electrical elements are considered. These design parameters and constraints for the strut are listed in Table 1.

<table>
<thead>
<tr>
<th>Design parameters</th>
<th>Ball-screw</th>
<th>Motor</th>
<th>Electrical elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>P, J_M</td>
<td>k_e, k_t, J_m, B_m, R_m, L_m</td>
<td>R, L, C</td>
<td></td>
</tr>
<tr>
<td>Constraints</td>
<td>δx_max, F_max, b</td>
<td>V_peak, I_rms, ω</td>
<td>physical range of c, k, b</td>
</tr>
</tbody>
</table>

Table 1: Selected parameters and constraints considered for the optimisation process.

For a ball-screw mechanism, pitch \( P \) is one of the most important design parameters as analysed in Section 2.2 and 2.3. Furthermore, the nut inertia, \( J_M \) is another design parameter to represent the minimum inertia of a ball-screw mechanism. As the values of \( P \) and \( J_M \) is interact, the corresponding \( J_M \) for each value of \( P \) is different. For a motor, parameters \( k_e, k_t, J_m, B_m, R_m, L_m \) are constant properties, but values of them may vary for different motors and can be found from its specification. Based on these parameters and pitch, the value of mechanical elements \( b, k, c \) are determined for the mechanical network. In the external circuit, three basic electrical elements may be used which are resistors \( R \), inductors \( L \) and capacitors \( C \). The values of their corresponding mechanical elements can be determined from the relations shown as follows:

\[
\begin{align*}
    k &= \frac{G_m}{L} \\
    c &= \frac{G_m}{R} \\
    b &= G_m \cdot C.
\end{align*}
\]

Physical constraints for the mechatronic strut have to be determined and implemented into the optimisation process to check if the optimal values of these elements are feasible in physical realisation. For a ball-screw mechanism, \( \delta x_{max} \) and \( F_{max} \) are the maximum relative displacement between two terminals and maximum force applied to the ball-screw mechanism, respectively. In the specification, the maximum force \( F_{max} \) of a ball-screw mechanism is listed for each set value of \( P \). The minimum inertia of a ball-screw mechanism \( J_M \) produces a minimum inertance for the system, which is used as the lower boundary of \( b \) for the ball-screw mechanism. If the optimal value of \( b \) is bigger than the lower boundary, the rest inertance can be easily adjusted by tuning the attached flywheel. For a motor, to make sure the mechatronic strut system working normally in a long time, the value of \( V, I \) and \( \omega \) calculated in the optimisation process for the motor have to be smaller than its nominal voltage \( V_{nom} \), current \( I_{nom} \) and velocity \( \omega_{nom} \), respectively. For physical realisation, each electrical element should be considered within its achievable physical range. From these ranges, the constraints for their corresponding mechanical elements can be calculated. Therefore, these constraints are used in the optimisation to obtain the optimum values for these corresponding mechanical elements and meet the physical realisation requirements of electrical elements.

The optimisation methodology can be simplified as some design parameters and constraints may not able to be determined from their specifications. The simplified optimisation methodology is used to for the case studies later on. For motors used in case studies, the damping coefficient \( B_m \) is not able in their specifications. Therefore, the value of \( c_e \) can not be calculated for the suspension network, and then the element \( c_e \) is neglected for the optimisation process. However, the value of \( B_m \) can be identified through experiment and implemented into the system. Moreover, it is assumed that all values of electrical elements can be physically realised in the optimisation so the constraints of electrical elements can be neglected in optimisation. Therefore, the general network can be simplified as in Figure 3 without \( c_e \).
3.2 Qualitative analysis

In this section, it provides an analysis to qualitatively discuss what the preferred properties required of the ball-screw mechanism and motor for the mechatronic strut.

3.2.1 Ball-screw mechanism

The total ball-screw inertance $b = b_M + b_{flywheel}$ can be adjusted by adding or removing the flywheel, but there is a lower boundary, which is the inertance $b_M$ dominated by the nut with inertia $J_M$ and pitch $P$ for a ball-screw mechanism and the relation between them is given as follows:

$$b_M = \left(\frac{2\pi}{P}\right)^2 \cdot J_M$$

(15)

Therefore, the total equivalent inertance $b_{total}$ of the mechatronic strut should be larger than $b_M$ and the relation is presented as follows:

$$b_{total} \geq b_M$$

(16)

To increase the possibility of achieving more admittance functions for the strut, the value of $b_M$ should be as small as possible only considering the ball-screw mechanism. To have small $b_M$, the ball-screw mechanism requires large $P$ and small $J_M$. However, there is a trade-off between pitch $P$ and lower boundary inertia $J_M$. Sometimes, there is no detailed information available on the trade-off, a specific ball-screw for case studies in Section 4 is picked. Furthermore, the constraint of $F_{max}$ should be as large as possible to enhance the capability of force acting through the ball-screw shaft.

3.2.2 Electrical motor

Inertance $b_e$ in Figure 3, which is fixed once the motor is selected, is dominated by the motor inertia $J_m$ and pitch $P$ and its relation is given as follows:

$$b_e = \left(\frac{2\pi}{P}\right)^2 \cdot J_m$$

(17)

Therefore, combining Equations 15 and 17, $b_{total}$ should be larger than the sum of $b_M$ and $b_e$ and its relation is shown as follows:

$$b_{total} \geq b_M + b_e$$

(18)

To increase the possibility of achieving more admittance functions for the strut, the value of $b_e$ should be as small as possible. As $b_e$ is determined by Equation 17 it shows that the large pitch it has with small inertias $J_m$, the small $b_e$ it will have. Conclusively, the selected motor should have a small inertia to achieve small inertance $b_e$ for a mechatronic strut.

The internal resistance $R_m$ and internal inductance $L_m$ of the selected motor is modelled as a spring $k_m$ and damper $c_m$ in mechanical network which are in series with an external circuit shown in Figure 3. From Equation 3, to increase possibility of obtaining optimum transfer function for a system, the value of $R_m$ and $L_m$ should be as small as possible. For that reason, the values of $c_m$ and $k_m$ should be as large as possible to demonstrate this by applying translation relationships between mechanical and electrical elements. Conclusively, small values of $R_m$ and $L_m$ of a motor is more suitable for the proposed mechatronic strut.

In Section 2.3, the inductive force $F_m$ equals $\frac{2\pi}{P}T_m$. By substituting it into Equation 4, the relation of $F_m$ and $I$ is given as follow:

$$I_{nom} \geq I = \frac{F_m \cdot P}{2\pi \cdot k_t}.$$  (19)

From Equation 1 and 7, the relation of $V$ and $v$ is given:

$$V_{nom} \geq V = 2\pi \frac{k_e}{F} v.$$  (20)
For each motor, there is a set of corresponding nominal voltage $V_{\text{nom}}$ and current $I_{\text{nom}}$. To make sure the optimisation results are feasible, the optimised voltage and current in the electrical circuit must meet these constraints of nominal voltage and current. Furthermore, it is assumed that all optimised electrical elements’ values can be realised. Then Equations 19 and 20 can be combined and rearranged as follows:

$$\frac{F_m}{2\pi \cdot I_{\text{nom}}} \leq \frac{k_e}{P} \leq \frac{V_{\text{nom}}}{2\pi \cdot v} \quad (21)$$

To expand the possible solutions of $\frac{k_e}{P}$, the nominal voltage and current of a motor have to be as large as possible. In conclusion, a optimum motor for the proposed mechatronic strut should have small inertia $J_m$, internal resistance $R_m$, internal inductance $L_m$ and large nominal voltage $V_{\text{nom}}$ and current $I_{\text{nom}}$.

4 Case studies

The potential performance in ride comfort $J_1$ of the mechatronic strut is presented in a quarter-car model. A commonly used ball-screw mechanism is selected for the mechatronic strut. Using simplified optimisation procedure, case studies are carried out with specific motors and candidate external electrical circuits.

4.1 Quarter-car model and Performance index

A quarter-car model in Figure 4 (a) is applied to illustrate the performance benefits of the proposed mechatronic strut for vehicle dynamics, where $m_s$ and $m_u$ are the sprung and unsprung masses, $x_r$ the displacement of road excitation, $x_s$ and $x_u$ the corresponding displacement of the sprung and unsprung masses, $k_t$ the tyre stiffness. Figure 4 (b) presents the conventional suspension system for the quarter-car model. In this paper, the parameters of the quarter-car model are fixed as follows: $m_s = 250\, \text{kg}$, $m_u = 35\, \text{kg}$, $k_t = 150\, \text{kN/m}$ [7]. Furthermore, $k_s = 30000\, \text{N/m}$ is set to represent a normal passenger vehicle.

![Figure 4: The schematic diagram of the quarter-car model.](image)

To design a suspension system, there are a number of practical design requirements to meet such as passenger comfort, handling, tyre normal loads, limits on suspension travel etc. Although suspension design usually involve a trade-off between various performance requirements, it is useful to consider first how much improvement can be obtained in individual performance measures for various configurations. To compare the performance benefits, ride comfort $J_1$ index is selected as follows [7]:
\[ J_1 = 2\pi \sqrt{\nabla \kappa ||sT_{\tilde{x}_r \rightarrow \tilde{x}_s}||_2} \]  

(22)

where \( ||sT_{\tilde{x}_r \rightarrow \tilde{x}_s}||_2 \) is the \( H_2 \) norm of \( sT_{\tilde{x}_r \rightarrow \tilde{x}_s} \), \( V \) the driving velocity of the vehicle, while \( \kappa \) is the road roughness parameter. Values of \( V \) and \( \kappa \) are referred in [7].

The road input in time domain is also used to determine the system transient response such as \( V_{\text{peak}} \) and \( I_{\text{rms}} \). The procedure to generate the time domain signal is referred in [8].

### 4.2 Selected ball-screw

Based on qualitative analysis, the ball-screw mechanism should have small inertia and large force constraints. In Yilun’s paper [9], a common ball-screw mechanism, which the inertia and pitch are 0.02kgcm\(^2\) and 6mm, respectively, is was proposed for a novel shock absorber. The value of maximum force constraint was not presented in Yilun’s paper so it is taken from common ball-screw mechanism specification where its value is 26.5kN [10].

### 4.3 Selected motors

The qualitative analysis in Section 3.2 gives a brief guidelines for motor selection. Furthermore, the maximum terminal voltage of motors must be below 48V considering automotive 48V system [11]. As a result, two specific motors are selected with maximum available terminal voltage (48 V) and their parameter values are presented in Table 2. Comparing Motor-1 and Motor-2, Motor-1 has a small inertia \( J_m \) while Motor-2 has large nominal \( I \) and small \( R_m \) and \( L_m \). Two candidate external electrical circuits \( E_1 \) and \( E_2 \) are proposed, with the corresponding mechanical networks shown in Figure 5. The proposed two electrical circuits are used to replace the \( G_m/Z_e \) part in Figure 3.

<table>
<thead>
<tr>
<th></th>
<th>( V_{\text{nom}} ) (V)</th>
<th>( I_{\text{nom}} ) (A)</th>
<th>( R_m ) (Ω)</th>
<th>( L_m ) (mH)</th>
<th>( J_m ) (gcm(^2))</th>
<th>( k_i ) (Nm/A)</th>
<th>( k_e ) (Vs/rad)</th>
</tr>
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<tbody>
<tr>
<td>Motor-1</td>
<td>48</td>
<td>1.41</td>
<td>2.95</td>
<td>0.514</td>
<td>21.2</td>
<td>0.0429</td>
<td>0.0428</td>
</tr>
<tr>
<td>Motor-2</td>
<td>48</td>
<td>2.17</td>
<td>1.40</td>
<td>0.473</td>
<td>75.9</td>
<td>0.0585</td>
<td>0.0586</td>
</tr>
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</table>

Table 2: Parameters values for different motors.

![Figure 5: The corresponding mechanical networks of two external electrical circuits \( E_1 \) and \( E_2 \).](image)

Table 3 shows the optimisation results of four cases for two different motors in two different suspension networks using \( E_1 \) and \( E_2 \). In comparison with the optimal \( J_1 \) obtained by the conventional suspension
network $S_1$ shown in Figure 4 (b), the dynamic performance $J_1$ of four cases are much worse. It also can be seen from Table 3 that current in motors reaches the upper boundary of constraints $I_{nom}$ for all cases. To explore the impacts of the constraints on the cost function, a further investigation of $J_1$ to the change of constraint values is necessary. As the terminal voltage of motors has not reached the safe level, here only the constraint $I_{nom}$ is considered.

<table>
<thead>
<tr>
<th></th>
<th>$J_1$</th>
<th>$V_{peak}$ (V)</th>
<th>$I_{rms}$ (A)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Optimal, $S_1$</td>
<td>$E_1$</td>
<td>$E_2$</td>
<td>$E_1$</td>
</tr>
<tr>
<td>Motor-1</td>
<td>1.24</td>
<td>41.4</td>
<td>41.4</td>
</tr>
<tr>
<td>Motor-2</td>
<td>1.24</td>
<td>27.2</td>
<td>12.6</td>
</tr>
</tbody>
</table>

Table 3: Results of two different motors.

If the constraint $I_{nom}$ is released for both specific motors, the optimal $J_1$ can be further improved. Table 4 presents the optimal results of $J_1$ and their corresponding required current $I_{rms}$ and voltage $V_{peak}$ in the electrical circuit for all cases. In comparison with the optimal $J_1$ obtained by the conventional suspension system $S_1$, a maximum 14.5% enhanced performance in ride comfort can be achieved if $I_{nom}$ of Motor-2 using network $E_1$ is large enough. Furthermore, a 13.7% improvement of $J_1$ is obtained with Motor-2 using network $E_2$. Comparing these two cases, Motor-2 using $E_1$ is better for the system as it has smaller required current and larger improvement. As improvements for Motor-1 using both networks are poor, Motor-2 with large $I_{nom}$ and small $R_m$ and $L_m$ is more suitable the system.

Figure 6 presents the vertical acceleration of $m_s$ in frequency domain for all cases. Furthermore, the vertical acceleration of $m_s$ in time domain for the optimal case is shown in Figure 7 where only 25s response signal is shown. To clearly see improvements in time and frequency domain, the response of vertical acceleration of $m_s$ is presented in both domains.

In conclusion, for these two specific motors, none of them can provide a better dynamic performance $J_1$ than the optimised conventional system. However, the investigation of constraint $I_{nom}$ shows that both motors with large $I_{nom}$ can obtain better dynamic performance $J_1$. Furthermore, Motor-2 with smaller internal resistance and inductance and large nominal current can have better performance.

<table>
<thead>
<tr>
<th></th>
<th>Optimal, $S_1$</th>
<th>Motor-1, $E_1$</th>
<th>Motor-1, $E_2$</th>
<th>Motor-2, $E_1$</th>
<th>Motor-2, $E_2$</th>
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<tbody>
<tr>
<td>$J_1$</td>
<td>1.24</td>
<td>1.23</td>
<td>1.23</td>
<td>1.06</td>
<td>1.07</td>
</tr>
<tr>
<td>$I_{rms}$ (A)</td>
<td>-</td>
<td>3.3</td>
<td>3.4</td>
<td>3.4</td>
<td>4.1</td>
</tr>
<tr>
<td>$V_{peak}$ (V)</td>
<td>-</td>
<td>48</td>
<td>48</td>
<td>48</td>
<td>48</td>
</tr>
<tr>
<td>Improvement (%)</td>
<td>-</td>
<td>0.8</td>
<td>0.8</td>
<td>14.5</td>
<td>13.7</td>
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</tbody>
</table>

Table 4: Optimal results of $J_1$ for four cases and their corresponding required current, voltage and improvement.
5 Conclusion

This paper investigated the potential of mechatronic strut for road vehicles and proposed a design methodology and qualitative analysis. Cases studies of the mechatronic vehicle suspension strut were carried out and it shows that the strut can provide improvements of ride comfort for a quarter-car model. To obtain the optimal mechatronic suspension design, an optimisation methodology is proposed by considering all possible design parameters and constraints such as nominal voltage and current for the motor, maximum force acting on the ball-screw mechanism and size and weight of the electrical elements. In the qualitative analysis of the mechatronic strut, an motor for the strut should have small inertia $J_m$, internal resistance $R_m$, internal inductance $L_m$ and large nominal current $I_{nom}$ and voltage $V_{nom}$. A ball-screw mechanism should have small initial inertia $b_M$ and large bearable force $F_{max}$. Furthermore, to obtain the optimal $J_1$ for the system, the
corresponding suitable ratio of $\frac{k}{P}$ can be only a value or within a range. Therefore, there may be multiple solutions of $\frac{k}{P}$ for the system. Simultaneously, there are multiple available choices for motor and ball-screw mechanism selections for each value of $\frac{k}{P}$. In case studies, the dynamic performance of suspensions using two specific motors are determined with two candidates external electrical circuits by using a simplified optimisation procedure. It shows obtained $J_1$ for all these motors are worse than $J_1$ obtained by conventional suspension system. This is because that $J_1$ becomes poor to meet those constraints in optimisation such as $I_{nom}$ and $V_{nom}$. Then the impact of $I_{nom}$ to $J_1$ was carried out as $V_{nom}$ has not reached the safe level shown in Table 4. From the analysis for both motors, dynamic performance $J_1$ can be improved significantly if the constraint of $I_{nom}$ is released. As Motor-2 has small internal resistance and inductance, optimal $J_1$ obtained using Motor-2 is better than that obtained with Motor-1 with up to 14.5% improvement.

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