Noise and vibration suppression of electric power steering systems: a survey

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Abstract: Electric power steering (EPS) systems have become the ideal steering system for most vehicles because of the generated torque assist mechanism making steering easier for drivers. However, EPS systems generate unwanted noise and vibration that can adversely affect the comfort and health of drivers. Numerous studies have been conducted to examine the source of vibrations. Various isolation techniques including passive, active, or semi active have been proposed for achieving best noise and vibration attenuation. Discrete and distributed mathematical models have been employed to predict the dynamic response of EPS systems. This paper presents a state-of-the-art review of mathematical modelling, vibration isolation methods, and optimisation techniques used for improving the design of EPS systems.

Keywords: electric power steering; noise; vibration; isolation; optimisation.

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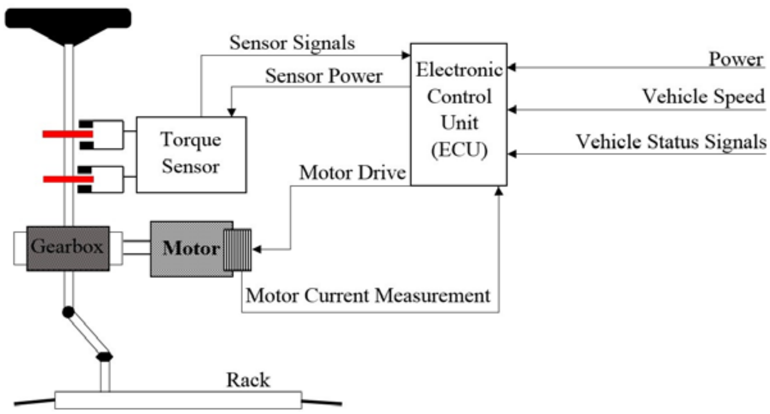
1 Introduction

Electric power steering systems assist drivers in steering by providing additional steering effort required to turn the wheel. A typical electric power steering system consists of a rack and pinion mechanism, steering column, gearbox, torque sensor, motor, and an electronic control unit (ECU) as shown in Figure 1. The ECU calculates the assisted power needed based on the applied torque by the driver. The torque sensor is implemented to sense the driver's movement on the steering wheel, and a reduction gear that increases the turning force from the motor and transfers it to the steering mechanism. Another common name used in the literature for EPS systems are electric power-assist steering (EPAS) systems, which involve the same components and control logic.

There are various types of EPS systems and they can be distinguished based on the position of the assist motor. These are column-type, pinion-type, and rack-type. The assist motor of the column-type is mounted on the steering column, which directly drives the steering shaft. Column-EPS is usually adopted for compact and medium cars. The pinion-type has the assist-motor connected to the steering gear and works well for small cars. The steering system of the rack-EPS is provided with a motor assist unit mounted on the rack shaft. This latter type works well for large size vehicle.

One of the great advantages of EPS systems over hydraulic power assisted steering (HPS) systems is the significant reduction in energy consumption. Hydraulic power steering systems constantly require oil pressure to be boosted, while electric power steering systems only operate when required using sensors resulting in much lower fuel-consumption. Though EPS systems provide a lot of benefits over HPS systems the main downfall is the increase in noise and vibration transmitted to the driver. Vehicle noise, vibration and harshness (NVH) is an important aspect to consider in all vehicle design. There are various causes of NVH in vehicles, Qatu (2012) and Qatu et al. (2009) provide a detailed review of different NVH issues in automotive vehicles.

Figure 1 EPS system components (see online version for colours)



The unwanted noise and vibration suppression in EPS systems can be achieved using passive, active, or semi-active isolators. Passive isolators offer a simple, cost-effective, reliable, and energy efficient means of vibration isolation, but they are only effective within a limited range of frequency. The incorporation of passive isolators is usually discussed in the beginning of the design stage. Rubber dampers have been increasingly used as a means of passive isolation of EPS system because of the materials good damping characteristics resulting in very low transmissibility at resonance (Sakthivel et al., 2012; Shim and Shin, 1999; Hong and Yan, 2009; Yan et al., 2009; Zhao and Wang, 2005).

Active isolation systems offer much superior performance than passive isolation, but at the expense of hefty price tag, reliability, and energy efficiency. This isolation technique involves the use of sensors and actuators in a system to produce a force where it is needed to reduce disturbance in the system (Hong and Yan, 2009; Kim and Song, 2002; Lianbing et al., 2007). Semi-active isolators are a combination of passive isolation, controllable elements, and a control mechanism (Zaremba et al., 1997; Miyazaki, 2008; Preumont, 2011; Yoo et al., 2015). This latter technique provides a trade-off between the two extremes and is potentially the best approach for suppressing noise and vibration.

To determine the best control strategy for any system, the dynamic models of the system must be first determined. Mathematical models of EPS systems abound in the literature, but most of them are discrete models, in which the column and steering wheel are modelled using a lump mass out of order actual numbers in here (31, 33) (Sakthivel et al., 2012; Shim and Shin, 1999; Kim and Song, 2002; Lianbing et al., 2007; Zaremba et al., 1997, 1998; Chatterjee et al., 2013; Li et al., 2009, 2008b, 2016; Hu et al., 2008; Ackermann et al., 1999; Kurishige et al., 1999; Xie et al., 2014; Chen et al., 2006, 2011; Yamamoto and Nishimura, 2011; Abe et al., 2016; Sugita et al., 1997; Mehrabi et al., 2011; Chabaan, 2009; Lee et al., 2005, 2017; Dong et al., 2010; Marouf et al., 2011a, 2011b, 2012; Pang, 2005; Islam and Husain, 2010; Yang, 2013; Bröcker, 2006; He and Gu, 2012; Freund et al., 2017). Recently, Freund et al. (2017) studied the free vibration of an EPS system by modelling the column as a distributed system. Their findings indicate that modelling the column as lump mass may result in inaccurate prediction of EPS system dynamic. More detail about the mathematical modelling of EPS is provided in Section 2. Section 3 addresses the different isolation techniques used for suppressing noise and vibration in EPS system. Section 4 discusses optimisation procedure and techniques used to enhance the design of EPS system. A summary of the discussed survey is presented in Section 5.

2 Mathematical modelling

There are three common types of EPS systems discussed widely in the literature, column-type, rack-type, and pinion-type. The use of these three systems into vehicle design is dependent on various characteristics of the vehicle. A detailed model of a rack-type electric power steering system is shown in Figure 2. All electric power steering systems involve the following components: a steering wheel, torque sensor, gearbox, motor, rack, and an ECU. To understand the dynamics in EPS systems a mathematical model must be developed. This section discusses the most common types of mathematical models of EPS encountered in the literature.

2.1 Lumped mass system

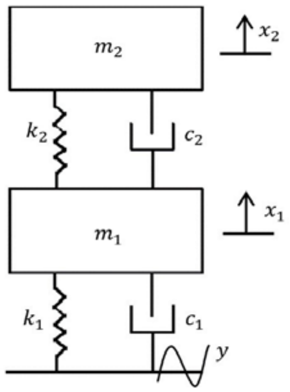
EPS systems consist of multiple components, which can make mathematical modelling very tedious. To surpass the exhaustive model containing the various parts, a simplified model of the system can be studied by grouping some of the elements (Sakthivel et al., 2012; Shim and Shin, 1999; Kim and Song, 2002; Lianbing et al., 2007; Zaremba et al., 1997; Chatterjee et al., 2013; Li et al., 2009, 2008b, 2016; Zaremba et al., 1998; Hu et al., 2008; Ackermann et al., 1999; Kurishige et al., 1999; Xie et al., 2014; Chen et al., 2006, 2011; Yamamoto and Nishimura, 2011; Abe et al., 2016; Sugita et al., 1997; Mehrabi et al., 2011; Chabaan, 2009; Lee et al., 2017, 2005; Dong et al., 2010; Marouf et al., 2011a, 2011b, 2012; Pang, 2005; Islam and Husain, 2010; Yang, 2013; Bröcker, 2006; He and Gu, 2012; Freund et al., 2017). Most simplified models of EPS systems consist of two lumped masses. This is depicted in Figure 3, in which the column and the steering wheel are modelled as one lump mass and the steering box is represented by another mass. Considering that the column is very stiff and assuming that the fundamental behaviour of the system is dominated by low frequency modes, one can make such approximation to examine the dynamic response of the system at low frequencies. Following Sakthivel et al. (2012), the equation of motion of such system can be obtained as:

$$\begin{pmatrix} m_1 & 0 \\ 0 & m_2 \end{pmatrix}\begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{pmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{pmatrix}\begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix} + \begin{pmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{pmatrix}\begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} k_1 y + c_1 \dot{y} \\ 0 \end{Bmatrix} \tag{1}$$

where m_1 denotes the mass of the steering box, m_2 is the mass of the steering column and wheel, c_1 and k_1 denote the damping and spring coefficients of the dampers, respectively. The rubber cushioning pad used between the steering box and column are represented by the spring and damping coefficients, k_2 and c_2, respectively. x_1 and x_2 denote the displacement of the steering box and steering column/wheel assembly respectively. y denotes the harmonic motion of the support base.

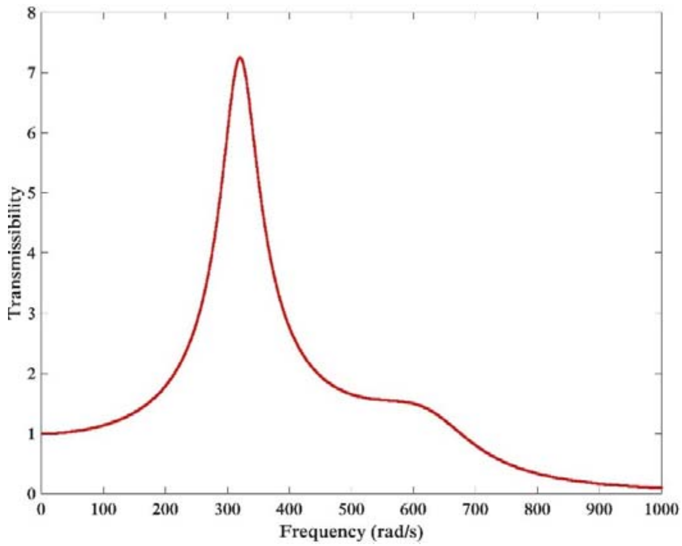
Figure 2 Common EPS system (see online version for colours)

Figure 3 EPS modelled as lumped mass system



Source: Sakthivel et al. (2012)

Figure 4 Transmissibility plot for two-dof EPS model (see online version for colours)



The transmissibility of the two degree-of-freedom can be obtained as:

$$\left|\frac{X_2}{Y}\right| = \sqrt{\frac{\left(k_1k_2 - c_1c_2\omega^2\right)^2 + \left(\left(k_1c_2 + k_2c_1\right)\omega\right)^2}{\left(k_1k_2 - \left(m_1k_2 + m_2k_1 + m_2k_2 + c_1c_2\right)\omega^2 + m_1m_2\omega^4\right)^2 + \left(\left(k_2c_1 + k_1c_2\right)\omega - \left(m_1c_2 + m_2c_1 + m_2c_2\right)\omega^3\right)^2}} \tag{2}$$

Using equation (1), a plot of the transmissibility against the excitation frequency is depicted in Figure 4. The peaks on this plot are the natural frequencies of the system and these frequencies were found to be $\omega_1 = 320.94$ rad^{1}s^{-1} and $\omega_2 = 638.32$ rad^{1}s^{-1}. It should be noted that modelling the EPS column as a discrete system can result in a less accurate understanding of the actual dynamics of the real system. As described in the next section, a continuous system model can be used to better predict the dynamic response of EPS and other key variables can be identified to optimise the design of electric power steering systems.

2.2 Continuous system

By describing the steering system using a continuous system model, more aspects of variable effects on the system can be understood, thus improving optimisation in the design process. In Freund et al. (2017), a continuous system model of the steering system is presented, in which the steering column is modelled as a rigid bar. The schematic of the EPS system studied in Freund et al. (2017) is depicted in Figure 5. In this model c_1 and k_1 represent the damping and stiffness coefficients of the rubber mount at the top of the transmission case respectively.

c_2 and k_2 denote, respectively, the coefficients of damping and stiffness of the rubber cushioning pad between the steering box and steering column base. L in Figure 5 represents the length of the steering column, while m_1 and m_2 represent the mass of the steering box and steering wheel, respectively. The engine excitation of the system is denoted by $y(t)$. $x_1(t)$ is the displacement of the steering box, and $w(x, t)$ is the longitudinal displacement of the steering column. Using Hamilton's principle, the governing equations of motion and boundary conditions can be obtained as follows (Freund et al., 2017):

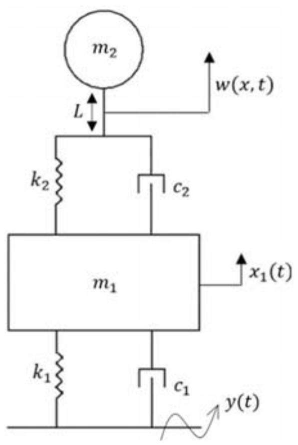
$$m_1\ddot{x}_1 + (k_1 + k_2)x_1 + (c_1 + c_2)\dot{x}_1 - k_2 w(0, t) - c_2\dot{w}(0, t) = k_1 y + c_1\dot{y} \tag{3}$$

$$\rho\ddot{w}(x, t) - EA\frac{\partial^2 w(x, t)}{\partial x^2} = 0 \tag{4}$$

$$m_2\ddot{w}(L, t) = -EA\frac{\partial w(L, t)}{\partial x} \tag{5}$$

$$k_2\left(w(0, t) - x_1\right) + c_2\left(\dot{w}(0, t) - \dot{x}_1\right) = EA\frac{\partial w(L, t)}{\partial x} \tag{6}$$

Figure 5 Schematic of system



Using the method of separation of variable, the frequency equation, and mode shapes of the EPS system can be obtained as:

$$\tan\frac{\omega}{c}l = EA\omega c\left[\frac{k_2(1-\alpha)-m_2\omega^2}{E^2A^2\omega^2+m_2\omega^2k_2c^2(1-\alpha)}\right] \quad (7)$$

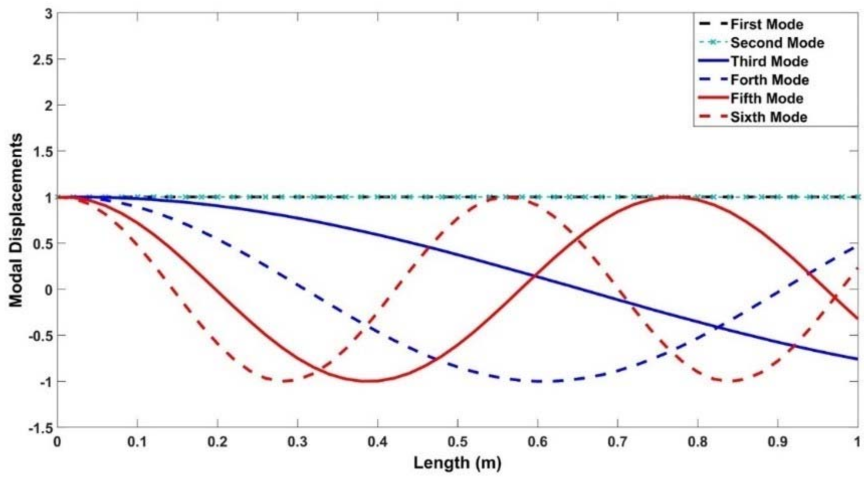
$$Y(x) = A_1\left(\cos\frac{\omega}{c}x+\frac{k_2c(1-\alpha)}{EA\omega}\sin\frac{\omega}{c}x\right) \quad (8)$$

The numerical simulation of the system discussed in Freund et al. (2017) was based on the parameters listed in Table 1 and the simulation results were obtained using MATLAB.

Table 1 Parameter for steering system 2012

Parameters	*Value*
Mass of the steering box	$m^1 = 16$ kg
Mass of steering wheel	$m_2 = 3$ kg
Stiffness of base damper	$k_1 = 2{,}740$ Nm^{-1}
Stiffness of cushioning pad	$k_2 = 1{,}490$ Nm−1
Damping coefficient of base damper	$c_1 = 1.714$ Ns/m
Damping coefficient of cushioning pad	$c_2 = 0.588$ Ns/m
Diameter of steering column	$d = 0.04$ m
Length of steering column	$L = 1$ m
Modulus of elasticity	$E = 200$ MPa
Density of steering column	$\rho = 8{,}000$ kgm^{-3}

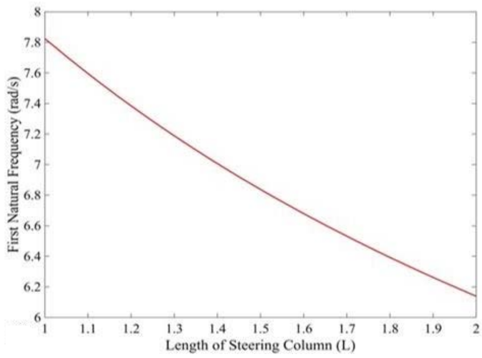
Figure 6 Six modes shapes for continuous system model (see online version for colours)



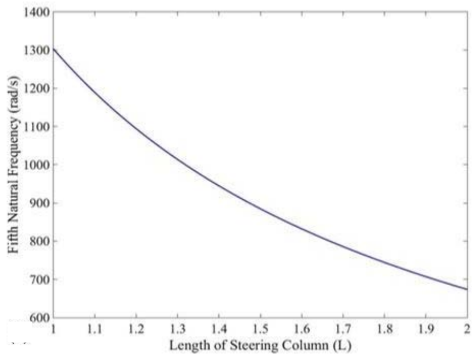
Source: Freund et al. (2017)

The first six mode shapes of the system are shown in Figure 6. This figure clearly indicates that the first two modes are associated with the motion of the discrete mass. The rest of the mode shapes are similar to those of a cantilever beam. Figure 7 shows how the length of the steering column can affect the natural frequencies of the system, which is impossible to predict with a discrete model.

Figure 7 Effect of column length on natural frequency, (a) effect of column length on first natural frequency (b) effect of column length on fifth natural frequency (see online version for colours)



(a)



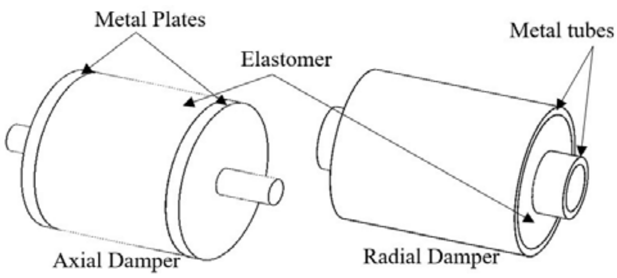
(b)

3 Vibration isolation

3.1 Passive

Passive isolation techniques have been widely used in different vehicle parts to isolate vibration. The use of passive isolators is plausible to many researchers because no sensors or actuators are required. Various types of passive isolation techniques have been reported in Sakthivel et al. (2012), Shim and Shin (1999), Hong and Yan (2009), Yan et al. (2009) and Zhao and Wang (2005). The purpose of using passive isolation is to reduce the amount of external vibration, or noise transmitted to the driver.

Figure 8 Nitrile butadiene rubber dampers with metal



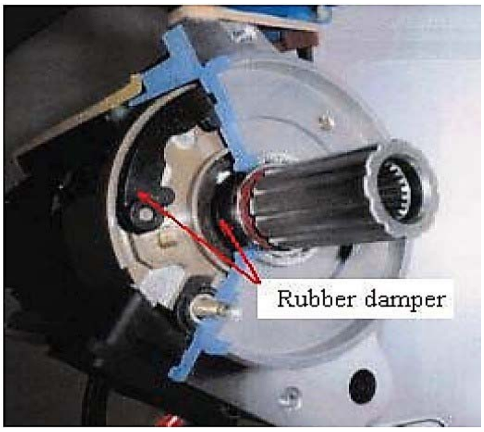
Source: Sakthivel et al. (2012)

In Sakthivel et al. (2012), a simple passive approach to reduce the resonance of a steering system implemented in an agricultural tractor is studied. The steering system mathematical model was studied as described in the previous section and the root cause of excessive vibration was the resonance of the system with engine excitation. Since dampers work best at resonance, two elastomeric dampers with metal plates, as shown in Figure 8 were proposed to help reduce much of the vibration in the operating speed range.

The damper discussed was used in the steering system model to isolate vibration from the engine to the steering column caused by engine excitation. Once the radial and axial dampers were implemented into the steering system, they were found to provide a reduction in peak acceleration of 28% and 66%, respectively (Sakthivel et al., 2012). The axial damper provided more vibration reduction than the radial damper and the accuracy of the results from the mathematical model was found to be 80%, while the accuracy of the ADAMS model was about 85% (Sakthivel et al., 2012). With a significant reduction of vibration transmitted to the driver, the use of dampers implemented into a steering system is a plausible approach and could be implemented into an automobile steering system with similar effects.

Other investigators have also studied how to suppress both noise and shock in EPS system using passive isolation. Hong and Yan (2009) developed a noise reduction structure design to reduce electromagnetic noise from the motors used in column-type EPS systems. The noise reduction structure design proposed in this paper effectively diminishes the amount of external shock in the system by adding rubber dampers. The rubbers dampers were inserted into the support sections of the worm gear, as shown in Figure 9. The placement of the dampers is particular so they can slide along the worm gear shafts and absorb the external shock force.

Figure 9 Worm reduction gear damping mechanism (see online version for colours)

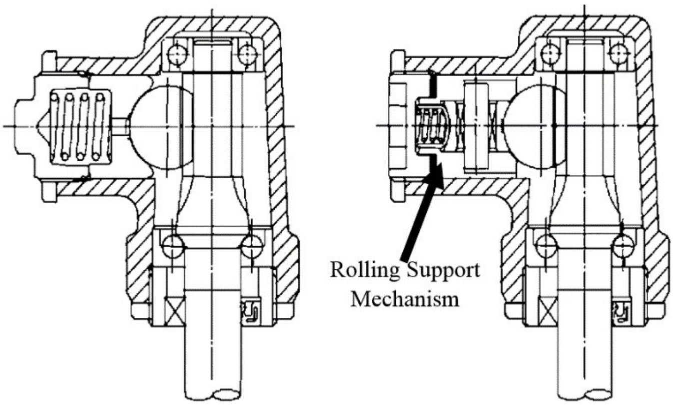


Source: Hong and Yan (2009)

A similar concept of using a rubber damper inserted into the EPS reduction gear was discussed in Yan et al. (2009), but to make the design more effective the adoption of a rolling support rack and pinion instead of the traditional rack and pinion, as shown in Figure 10, was suggested. The benefit of using a rolling support rack and pinion is that it would provide better transfer characteristics and would reduce the transmitting loss of the road-surface information Yan et al. (2009).

Another passive isolation technique was described in Hong and Yan (2009) using a passive noise filter. Passive noise filters are made up of LC circuitry consisting of capacitors to be attached to ground, common-mode coils, and capacitors to be placed across phases. The degree of attenuation of the electromagnetic interference (EMI) emission is dependent on the characteristics of the components in the filter. It is not only important to analyse the characteristics in the design phase of passive noise filter, but it is also important to consider the magnetic saturation of the coil cores, leakage current, and installation positions to develop the most optimal filter design. Although passive noise filters can help to attenuate noise, resonance problems can occur between the reactors and capacitors in the design causing the use of these filters to be less effective.

Figure 10 Conventional rack and pinion v. rolling support rack and pinion



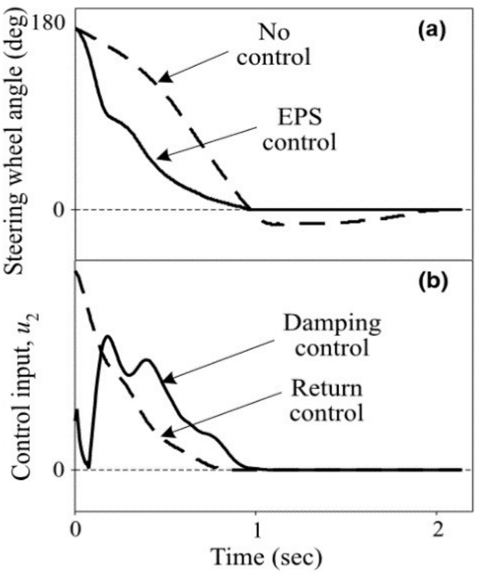
Source: Yan et al. (2009)

3.2 Active

An alternative approach to isolate unwanted noise and vibration is using sensors and actuators coupled by a controller. This type of isolation technique is considered active. Active isolation is commonly used for suppressing noise and vibration in electric power steering systems. In Lianbing et al. (2007) a proportional-integral-derivative (PID) controller is employed in the steering system using both an active damping algorithm and a return algorithm. This isolation technique allows the wheel of the steering system to return back to the centre in a smooth damped way and avoid oscillations that are present at high speeds. The model is verified through various simulations and hardware in the loop simulation (HILS)-based experiments for various driving conditions. Figure 11 shows the response of the control inputs and steering wheel angle along with both damping and return controls (Kim and Song, 2002). With the EPS control logic presented for reducing steering torque, the driver can turn the wheel with a significantly lower steering torque and the steering torque can follow the referenced steering torque independent of the load torques very closely. The EPS logic proposed for return-to-centre performance can be obtained using the proper control of the assist motor after cornering the steering wheel and without overshoot. Overall, the proposed EPS control logic discussed in this paper provided improvement to return-to-centre performance, reduction of steering torque exerted by the driver, and damping control.

In Lianbing et al. (2007), a controller design for electric power steering systems is proposed. The purpose of this design is to improve the control quality of the EPS. The two aspects of control considered are the sensor noise and the influence of road surface disturbance. The controller is designed by using a disturbance observer method. The proposed system introduces a new low-pass filter by analysing a model of the EPS system, and an analysis method of road information is proposed. Information on varying road conditions includes both disturbance information and road's back force information, which can be separated by frequency. The proposed controller is evaluated through simulation using MATLAB. The results of the simulation demonstrated that the proposed controller can satisfy EPS performance requirements and can suppress sensor noise and road surface disturbance. The study conducted in this paper provides sufficient information to conclude that the use of the proposed controller is an effective way to improve noise and vibration reduction in EPS systems.

Figure 11 Responses of the (a) steering wheel angle for both the EPS control logic and no control and (b) control inputs along with both damping and return controls



Source: Kim and Song (2002)

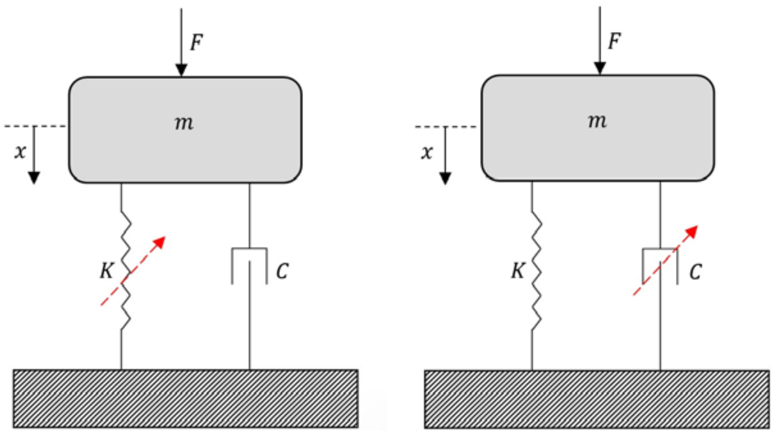
The research conducted in Hong and Yan (2009) takes into consideration both electrical and mechanical measurements of the motor. The use of different passive noise filters, as discussed in Section 3.1, invert the noise signal, but some leakage current may occur in a passive noise filter configuration when a load having a small impedance impacts the capacitors of the motor. The proposed active noise filter is used to reduce electromagnetic noise instead of a passive filter because of the overall increase of volume and cost of the inverter system. The practical use of active noise filters over passive is that it can reduce common mode and normal mode emission and eliminates the need of a transformer. By combining noise propagation and power electronics circuity into models and design noise filters discussed in this paper an overall design can be implemented in EPS applications to reduce noise, vibration, and cost.

3.3 Semi-active

In some cases, using only an active isolation system or only a passive isolation system is not optimal. This has resulted in designing systems that incorporate both. By utilising the active control techniques to tune damping and stiffness of passive isolators, as shown in Figure 12, more vibration can be suppressed than using passive isolation. These types of control mechanisms are labelled semi-active, also known as adaptive-passive. The main benefit of semi-active isolation mechanisms is that the element specified can be continually adjusted without using active force generators. In the literature involving EPS systems, there are very few control techniques studied that involve semi-active control structures. Some types of semi-active control structures utilised in other areas of research such as engine mounts are electro-rheological (ER) fluid-based variable dampers, variable damping and stiffness, and variable friction devices (Kurishige et al., 2010; Shoureshi, 1996; Jalili, 2002).

In Shoureshi (1996), an adaptive-passive vibration control system is presented. The semi-active control system involves a spring-mass vibration absorber attached to a vibrating body to absorb the energy. The presented structure of the semi-active isolation system can be adjusted using a vibration sensor attached to either the vibration absorber or the primary vibrating structure to sense the level of vibration desired. A signal is then created and sent to an electronic controller to instruct the actuator in the system to adjust the mass to compensate the vibration. In more adaptations of the described system, other variables rather than the mass can be adjusted to tune the vibration absorber.

Figure 12 Diagram of typical semi-active vibration isolation system (see online version for colours)



Another promising semi-active isolation technique discussed in the literature is the use of ER fluid (Kurishige et al., 2010; Jalili, 2002; Liu et al., 2015). The use of ER fluid is plausible for many applications such as engine mounts, vehicle suspension systems, and aerospace engineering. ER fluids can adapt their material characteristics based on the applied electric. By calculating the control force of the ER fluid mechanisms, the characteristics can be controlled by changing the voltage of the system. The use of ER fluids as a semi-active isolation system is beneficial for vibration at low frequency excitation.

Most semi-active isolation mechanisms are very sensitive to system parameters. Research could be extended by applying semi-active techniques to EPS systems in order to obtain optimal tuning parameters for the system. Semi-active isolation is a plausible cost effective and energy efficient means for noise and vibration suppression in EPS systems. By taking essential elements of an EPS, and modifying their characteristics, performance improvement with low power requirements can be obtained.

4 Optimisation

Electric power steering systems generally consist of: a motor, worm gear, worm wheel, torque sensor, mechanism, torque sensor harness, rubber damper, energy absorbing convoluted tube, keylock assembly, switch bracket, lower bracket, motor harness, and potentiometer. The design of the system is not only dependent on all of the parts separately, but on the entire system combined as well. Optimisation of EPS systems have been studied over the last few decades (Li et al., 2009, 2008a, 2008b; Zaremba et al., 1998; Hu et al., 2008; Kurishige et al., 1999; Chen et al., 2006, 2008, 2011; Yamamoto and Nishimura, 2011; Abe et al., 2016; Sugita et al., 1997; Mehrabi et al., 2011; Chabaan, 2009; Marouf et al., 2011a, 2011b, 2012; Chen and Li, 2005; Chen and Chen, 2006; Parmar and Hung, 2004; Zhao et al., 2015; Yu et al., 2001; Kozaki et al., 1999; Liao and Du, 2001; Zhang et al., 2004; Camuffo et al., 2002; Lawson and Chen, 2008). The procedure involved in the optimisation of an electric power steering system is similar to that of the engine mount optimisation procedure, which involves: modelling the system, objective selection, constraint description, and computerised parameter optimisation (Yu et al., 2001).

Before any optimisation procedure can be performed, the objective of the optimisation must be resolute. In Kurishige et al. (1999), the goal of the control algorithm is to enable reduced steering torque without any steering vibration to be transmitted to the drivers. The method used in Kurishige et al. (1999) was derived by modelling a steering mechanism and analysing the steering vibration of the mechanism using an open-loop frequency response of the linearised model. The open-loop frequency response of the linearised model showed that setting higher assist gain lead to reduced phase margin, causing EPS oscillation. To avoid this oscillation, a new control strategy based on damping for a specified frequency was introduced. The specified frequency described must be above the driver's steering frequency. The constraint in this optimisation process is that the phase margin must be increased, but the loop gain cannot be increased above the crossover frequency. If the loop gain is above the crossover frequency, unwanted electromagnetic and acoustic noise may be produced. To implement the damping algorithm, two observer methods using an estimator of the motor angular-velocity and back electromagnetic force (EMF) were discussed. The damping algorithm for specified frequencies was found to be effective in suppressing the oscillation from the short phase margin.

Conventional EPS system controllers have also been developed based on motor control. However, these controllers have complicated design processes and limitations. Chen and Li (2005), Chen and Chen (2006) and Chen et al. (2008) developed a motion controller to improve the driver's hand feeling during the steering process. The controller structure presented in Chen et al. (2008) breaks some limitations of conventional controller structures and does not require any additional sensors or actuators. The objectives for the controller structure involve two steps: motion control and motor torque control. Motor torque control can be obtained using a simple proportional integral (PI) control law, which leads to the main focus being on motion control. Two modern control approaches, H_2 and H_∞, were developed based on the modified scheme presented (Chen et al., 2008). The two-controller structure developed demonstrated superior performance over the conventional PI controller. The H_2 and H_∞ controller demonstrated more suppression of rough road surfaces on the driver's steering feel and shortened transient time.

Optimisation strategies on EPS systems also focus highly on the different types of control methods used to obtain the specific objectives. Along with the control methods discussed above, another technique used in the literature is linear quadratic (LQ) control (Yamamoto and Nishimura, 2011; Abe et al., 2016). Yamamoto and Nishimura (2011) developed a control method for EPS systems for vehicles with an active stabiliser. An active stabiliser, as shown in Figure 13, was implemented into vehicles with EPS systems to increase the roll stiffness of the vehicle. The restoring torque that acts on the steering axis during rough road running can be suppressed using a controlled suspension, such as the active stabiliser discussed in Yamamoto and Nishimura (2011). A full vehicle model with an active stabiliser coupled with a steering system model was developed. The proposed controller structure involved the use of a LQ controller for the active stabiliser and a gain scheduling (GS) controller for the EPS system with the driver's steering and rough road as the inputs. The system was tested through simulation and it was shown that the active stabiliser can significantly suppress the amplitude of the steering wheel angle by suppressing the rolling vibration of the vehicle body and suppress the rotational vibration of the steering wheel during rough road running.

In Abe et al. (2016), LQ control is also discussed, but without the incorporation of an active stabiliser. LQ control is employed to improve steering feel and reduce time-consuming manual tuning of EPS systems. A disturbance observer is incorporated into the control theory to estimate and suppress the vibrations. The incorporation of a state estimation observer with the LQ control method and disturbance observer was also discussed, but the results showed that the vibrations were even less attenuated compared to the conventional method. Mehrabi et al. (2011) discusses an optimal disturbance rejection control design that effectively attenuates external disturbances. By establishing a mathematical model of a column-type EPS system and constructing its state-space expression, three control methods were compared in regards to disturbance rejection properties, performance, and robustness. The three control methods compared were the conventional proportional-integral derivative (PID), linear quadratic Gaussian (LQG), and modified LQG. Although the proposed LQG provided good tracking properties, the performance was inferior outside nominal conditions. A modified LQG controller was proposed to make the EPS control system insensitive to disturbances and parameter variations. Through simulation the modified LQG demonstrated strong robustness in comparison to the other two controllers, by effectively strengthening the anti-interference ability of the EPS system; thereby, attenuating torque sensor noise, model parameter uncertainty, and the interference caused by random road excitation.

Parmar and Hung (2004) proposed an optimal control system that eliminates the use of the torque sensor utilised in most conventional EPS systems. The torque sensor implemented into EPS systems were developed specifically for EPS application, which makes the cost of these sensors very high. By eliminating the torque sensor, not only is the cost of the system reduced but the stiffness of the steering column increases with mechanical benefits. The dynamic model for the system discussed in Parmar and Hung was developed using Lagrangian dynamics. The state-variable feedback control system proposed used a Kalman filter to estimate the state variables and a linear quadratic regulator (LQR). A LQ regulator approach was chosen to compute optimal closed-loop pole locations. The controller developed showed multiple advances compared to conventional controller structures, such as successful attenuation of oscillations at low frequency ranges that are inherent in the open-loop system, closed-loop characteristics can be varied by fine-tuning parameters, and stable closed-loop performance is attained for high levels of assists.

Although most studies done on active control in EPS systems focus on adapting the control of the motor, in Li et al. (2009, 2008a) the focus of control shifts to the phase lag between the driver's steering torque and steering angle. The pressure ripples in EPS systems can be caused by the phase lag between the steer angle and the drivers steering torque, the disturbances from the road, sensor noise especially during high frequency manoeuvres, and nonlinear frictions. Electric power steering dynamics can be described using an eight-order nonlinear state-space model and approximated using a T-S fuzzy model with time varying delays and external disturbances. Once the T-S fuzzy model is approximated, a stabilisation approach is then used for nonlinear time delay systems through fuzzy state feedback controller in parallel distributed compensation (PDC) structure. The closed-loop stability conditions with the fuzzy controller are then parameterised in terms of linear matrix inequality (LMI) problem.

Some optimisation procedures discussed in literature encompass more than one objective (Hu et al., 2008; Chabaan, 2009; Kozaki et al., 1999). As demonstrated in Marouf et al. (2012) and Kozaki et al. (1999), a control strategy is developed to encompass multiple objectives such as:

1 improving EPAS damping characteristics
2 generate assist torque
3 stabilise the system without the need for different control algorithms and rules for switching between these algorithms.

With multiple control objectives in mind, each objective yielded different processes for reaching the desired output. For example, in order to attenuate vibration resonance produced from an EPAS systems various parameters have to be controlled without slowing down the system response. Figure 14 shows the resonance characteristics in the open loop (Kozaki et al., 1999). To ensure good steering feel in all operating conditions, it is necessary to compensate the vibrations produced by each of the system inputs.

Encompassing all objectives into one controller was carried out by developing a reference model, where the output is the reference motor angle. The inputs of the model described in Kozaki et al. (1999) are: motor speed, motor acceleration, road reaction torque, steering wheel speed, driver torque, steering wheel acceleration, road reaction torque, and desired assist torque. The control structure proposed is shown in Figure 15. It was demonstrated that unwanted vibrations can be eliminated by fine tuning the reference model parameter. To implement the developed reference model, the inputs must be estimated with the use of a sliding-mode observer and two second-order sliding mode (two-SM) differentiators to estimate the accelerations and additional outputs. The proposed control strategy reduces overall costs (i.e., no sensors are added), reduces design time, and improves robustness and overall system performance.

Figure 13 Active stabiliser

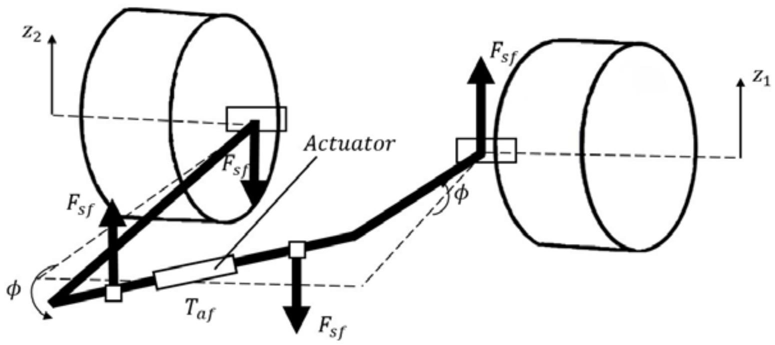
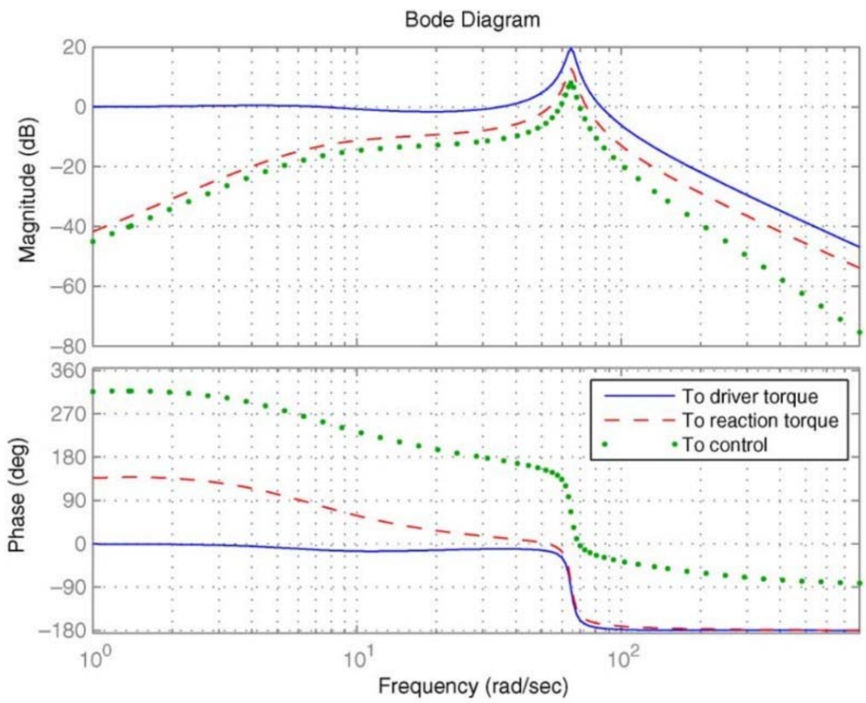


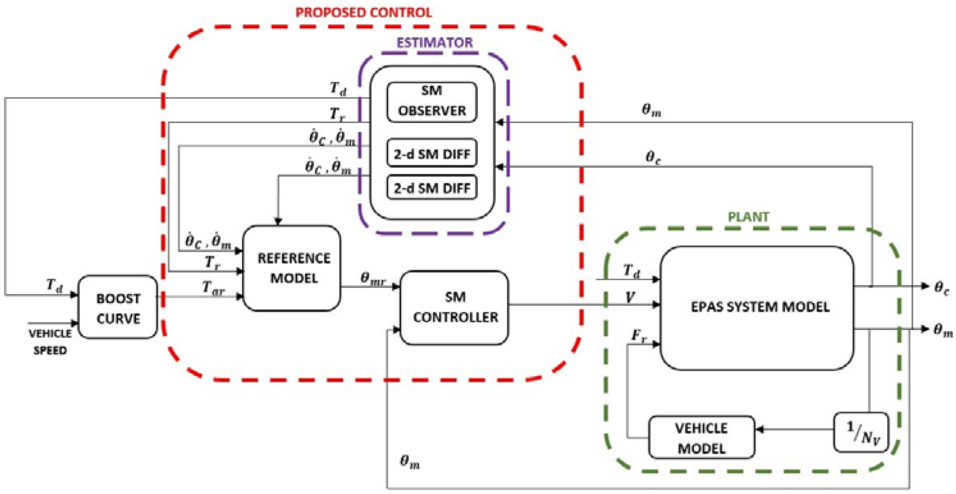
Figure 14 Open-loop frequency response of the steering torque to the driver torque, reaction force, and control input (see online version for colours)



Source: Kozaki et al. (1999)

The contemporary work on electric power steering system optimisation shows much room for improvement. With multiple control strategies being studied and implemented into EPS systems, various parameters can be controlled to improve steering feel and attenuate unwanted noise and vibration. Defining a near optimum solution or control strategy to effectively alleviate noise and vibration in EPS systems can be a complex task. Various control strategies have been used to obtain different objectives. With varying system structures and multiple control designs incorporated into different models, developing an optimal approach to encompass all would be extremely challenging, but ideal. By developing a simulation technique that can be easily tuned to fit all EPS systems, control techniques, and vehicle dynamics, more insight of the system dynamic can be realised before the system is made, leading to lower cost. Liao and Du (2001) develop a co-simulation technique involving multi-body based vehicle dynamics and EPS control system that proved to be useful for EPS performance evaluations. This technique would be useful if extended to all EPS system types and control logics.

Figure 15 Structure of proposed optimal control strategy (see online version for colours)



Source: Kozaki et al. (1999)

5 Conclusions

The dynamics of EPS systems have been presented using lumped mass modelling, but some research has also employed distributed models of the steering column so various key design variables of the system can be understood. There have been a few studies conducted using only passive isolation techniques to increase the stiffness at low frequencies. The studies on semi-active EPS isolation techniques are limited. Semi-active isolation techniques effectively used in similar areas of research were discussed showing improved performance and low implementation cost, which if implemented into EPS systems would be beneficial. Multiple active isolation techniques have been studied to further isolate the vibration of the system, but only a few isolate noise and vibration in the system without requiring additional sensors and actuators, thus increasing the cost. The described optimisation techniques have shown to be beneficial in isolating noise and vibration and improving steering feel. The performance of the EPS system is highly dependent on the overall vehicle dynamics, which makes full body simulation of vehicles an optimum technique. Further work on improving full vehicle body simulation with EPS system is needed for better understanding the nonlinearities of the system, thus improving the design before the model is developed.

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