

Design of Experiments on Steering Shimmy in Hydraulic versus Electric Power Steering Systems

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ABSTRACT

Research was conducted on automotive power steering systems at a component level to examine system sensitivity to an oscillatory loading condition. This series of experiments was aimed at quantifying system response of hydraulic power steering (HPS) and electric power steering (EPS) to objectively compare vibration attenuation capability of the two systems.

Steering shimmy is caused by an imbalance in a wheel/tire rotational assembly. The energy of this phenomenon is usually too great for a HPS system to completely attenuate while an EPS system is commonly capable of absorbing most of the vibration. With a HPS system, vibration can be partially attenuated through steering system parameter variation such as increasing friction or inertia. In an EPS system, the motor inertia, programming control, and gear reduction make it comparatively simple to attenuate the vibration before it reaches the steering wheel. This ability of an EPS system to partially or completely attenuate vibration caused by other chassis components has yet to be quantified. Furthermore, it has yet to be objectively compared with HPS and is thus the primary focus of this research.

The test results confirm that increased inertia and friction have a dramatic effect on steering vibration regardless of application. There is a friction threshold that can be used in partially mitigating some small vibrations such as tire imbalance. Furthermore, it shows that in an EPS system, the motor inertia and electric motor gear reduction are beneficial in stopping vibration from reaching the steering wheel. Through this research, we have shown that future developments are better equipped to attenuate steering shimmy per the application of an EPS system.

1. Introduction

In the manufacturing of tires and wheels, it is typical for an imbalance to exist about the rotational axis and/or rolling center line of the component. While the modern process of wheel/tire balancing is largely effective, it is neither permanent nor impervious to the inevitable effects of year-round vehicle operation. Dynamic wheel imbalance can result from uneven tire wear, dislodged wheel balance weights, and build up of mud and snow within the wheel. This imbalance has no effect on the continued safe operation of the wheel/tire assembly, but does transmit an oscillatory load into the chassis that in turn can create objectionable levels of vibration throughout the passenger compartment. When this wheel/tire imbalance causes vibration at the steering wheel, it is referred to as shimmy.

Vehicle manufacturers go to great lengths to attenuate such vibrations to improve the marketability of the vehicle. With the recent widespread application of EPS, this capability has grown because of the unique capabilities of that system. Past research on steering rotational vibration⁽¹⁾⁻⁽³⁾ explored the response

of an entire chassis system, including an HPS system, to an oscillatory input load. These studies looked into system correlation and prediction as well as component level factor analysis. The research described in this paper expands on that topic by exploring HPS versus EPS component level parameter variation and the benefits of EPS with regard to attenuation of steering shimmy.

2. Design of Experiments on Steering Shimmy

Steering shimmy is typical in the event of a wheel/tire imbalance because these assemblies are directly connected as shown in Fig. 1. However, because steering systems inherently possess characteristics that can attenuate vibration, it is possible to tune-out this reaction at least partially.

System characteristics such as friction do not exist solely for vibration attenuation, so a balance must be reached. In order to reach that balance, every system characteristic affected by friction must be considered. To that end, it is prudent to first look at the system at a fundamental level.

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The free body diagram of the system (Fig. 2) shows why this phenomenon occurs. An oscillatory load parallel to the ground plane causes a rotational vibration of the wheel/tire assembly about the kingpin axis. This vibration is transmitted to the steering system by the now axial vibration into the gearbox (G/box). This forcing function then acts on a seemingly simple mass, spring, and damper system, although the “damper” is actually a by-product of the power steering fluid in the gearbox cylinder. In an EPS system, the damper is replaced by a gear reduction and motor inertia, although there is damping designed into the software. For the purposes of this fundamental analysis, this section focuses on HPS only.

To more clearly show the areas of focus for attenuation of this vibration, this system can now be characterized by its equations of motion, Eq. 1 and Eq. 2:

$$F_{input} \left(\frac{r_{input}}{r_{gbox}} \right) + c_{gbox} v_{rack} + (-F_f) + \frac{k_{col} \theta_{col}}{r_{col}} + \frac{I_{str} \alpha_{str}}{r_{str}} = F_{response} \quad (1)$$

Where F_{input} is described by the equation for centripetal force:

$$F_{input} = m_{imbalance} r_{imbalance} \left(\frac{v_{vehicle}}{r_{tire,dynamic}} \right)^2 \quad (2)$$

The term $r_{tire,dynamic}$ is the dynamic loaded radius of the tire at the vehicle speed, $v_{vehicle}$, of interest. Shimmy happens when there is a net imbalance described by the mass of that imbalance, $m_{imbalance}$, and it's radius from the wheel center, $r_{imbalance}$. When there is shimmy it causes some oscillatory steering rack velocity, v_{rack} . The vibration is then felt in the

steering wheel as an oscillatory angular acceleration, α_{str} at some radius, r_{str} . The fourth term in Eq. 1 is for the torsional spring rate of the steering column where r_{col} is the outer radius. Since the displacement terms are determined by the input loading and the gearbox damping (c_{gbox}) is fixed, the tuning variables are friction (F_f), torsional spring rate (k_{col}), and rotational inertia (I_{str}). Within the realm of tuning friction we can change normal force and rack friction material to change the coefficient, μ . These are the areas of focus in this design of experiments.

This research treated rack, housing, and mounting stiffness as rigid. This is a safe assumption because the relative stiffness to that of the components described in the above equations of motion are several orders of magnitude higher.

Up until now, shimmy has been attenuated to an acceptable level by trial and error, but the full range of this solution has not yet been quantified. Commonly this attenuation is achieved by increased friction. So in addition to quantifying friction changes, we will see the empirical effects of changes to the other components as well.

In this research, the steering system was isolated on bench and studied at a component level to quantify the steering system's response to varying inputs and component parameter variation. Through this study we will demonstrate the similarities and differences of HPS and EPS steering response, and finally highlight the advantages for future developments with the application of EPS systems.

3. Test Method

3.1. Test Setup

Figure 3 shows a cross section of a steering G/box rack-and-pinion interface.

In order to isolate the steering system on-bench, a test fixture had to be created to exactly mimic the setup on a vehicle. Once built, the test setup would be able to be mounted alongside a power steering pump tester so a HPS system can function just as it would in vehicle at speeds that shimmy is most likely to occur. The required equipment to mimic vehicle speed with EPS is mobile and therefore not a concern in the

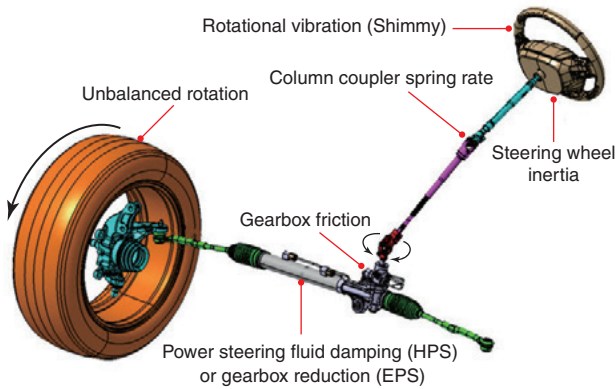


Fig. 1 Steering system model of shimmy condition

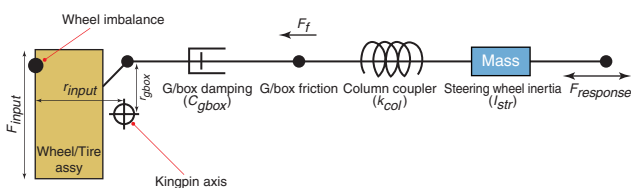


Fig. 2 Steering shimmy free body diagram

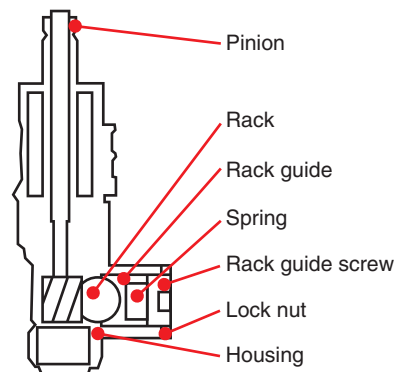


Fig. 3 Cross-section of a rack-and-pinion interface

fixture's design. Figure 4 shows the test fixture with a steering system mounted to it.

To capture the input measurements of force and acceleration, a force transducer and accelerometer are mounted at the interface of the shaker and G/box. The response at the steering wheel is captured via the accelerometer mounted at the nine o'clock position. The rotational vibration at the pinion was measured to look at potential damping provided by the steering column. As shown in Fig. 4, several other points were measured for vibration response to make sure that results were not being skewed by vibration energy being lost in shaking the test fixture itself.

3.2. Test Setup Confirmation

3.2.1. HPS correlation of on-vehicle to on-bench data

Before this testing began, measurements of tie rod load and steering wheel vibration were taken during another similar steering vibration phenomenon, brake judder. Though this mode has greater amplitude than shimmy, it provides a legitimate reference point for validating the test setup since resonant frequency is independent of amplitude. Figure 5 shows the comparison between this data and the test setup for an HPS system. A steering wheel vibration response of 1.9 m/s^2 was measured with a 750 N input force at a frequency of about 15 Hz on-bench and in-vehicle. This test setup proved to be a good model of the in-vehicle situation.

3.2.2. EPS correlation of on-vehicle to on-bench data

A similar trend was found with an EPS system. As shown in Fig. 6, a natural frequency peak exists at 10 Hz on-bench as well as on-vehicle. Though the HPS and EPS correlations were done at different load

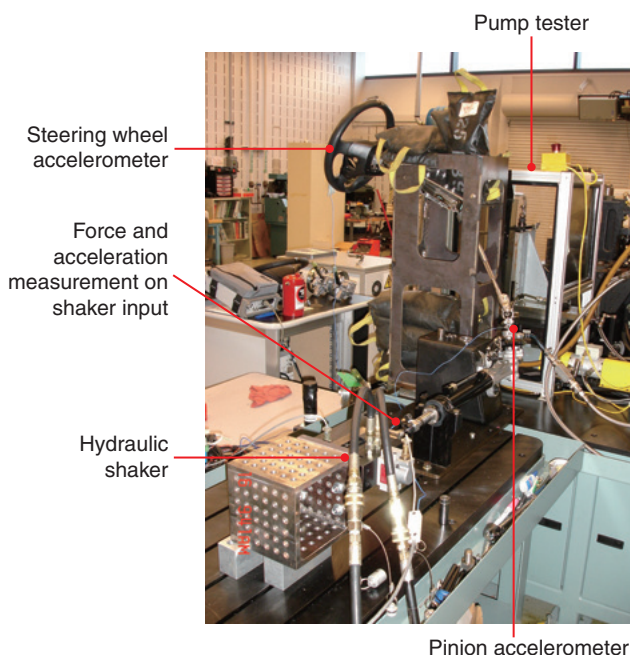


Fig. 4 Steering vibration sensitivity test bench setup

levels (750 N and 250 N respectively), once again the amplitude of a sinusoidal forcing function is not related to the response frequency, only the response amplitude. Therefore this has no effect on system resonant response and offers a sound method for test setup validation.

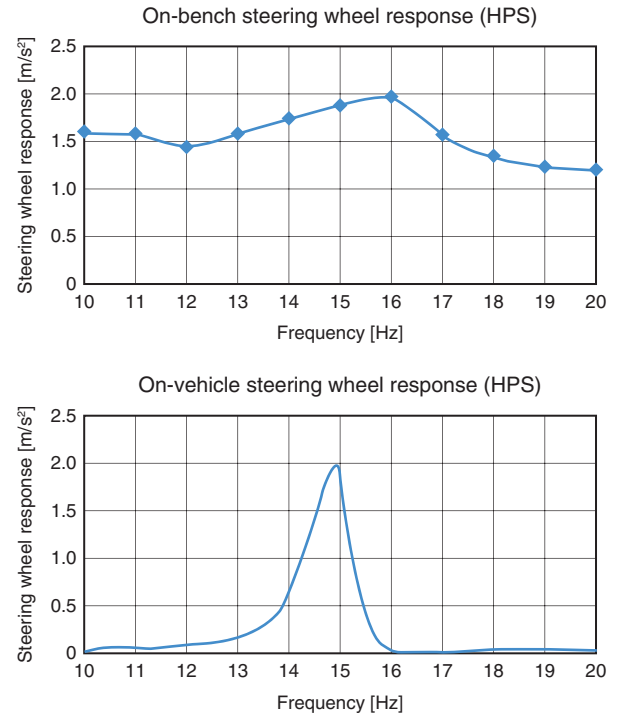


Fig. 5 HPS vehicle versus bench data

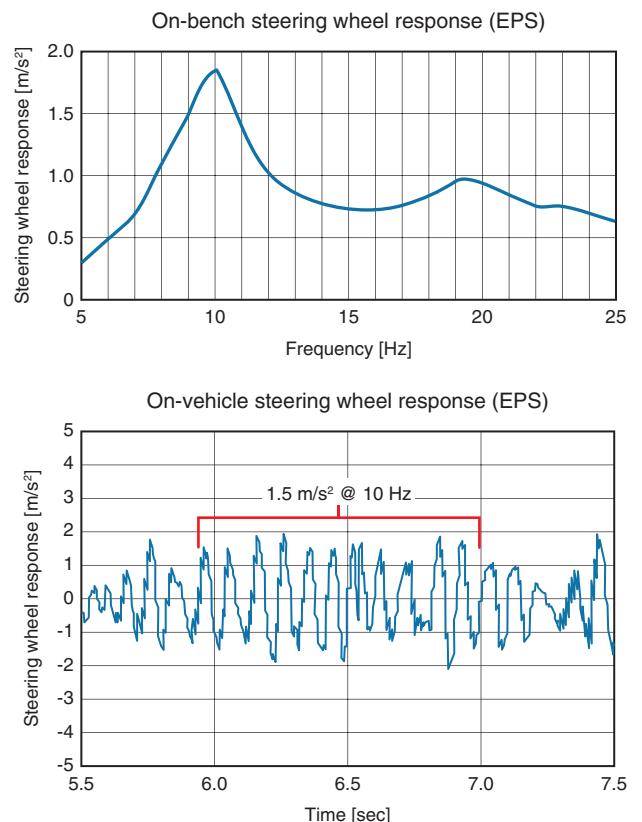


Fig. 6 EPS vehicle versus bench data

3.3. Test Procedure

In the HPS and EPS systems, it was initially important to find where the natural frequencies of the systems were because they would be the most likely to occur on-vehicle. After resonate frequencies are determined, testing can proceed to other frequencies if necessary. However, since the frequency peaks found on-bench were the same as those found on-vehicle, it was immediately apparent that this needed to be the area of focus. As shown in Figs. 5-6, the HPS and EPS systems being examined in this research have peak responses around 15 Hz and 10 Hz, respectively. After discovering the systems' behavior in the frequency domain, testing from then on was in the form of a force sweep. This was done for two reasons. First, it allowed for observing a particular setting's small and large input behavior all at once. Secondly, if there is a resonance frequency shift, it is important to see that effect at the natural frequency of the vehicle chassis, that is, 10 Hz for EPS or 15 Hz for HPS. Therefore, a force sweep was run while changing each variable component of the system in the following design of experiments:

- (1) G/box Rack Guide Screw Torque
- (2) G/box Rack Guide Material Friction
- (3) Steering Column Coupler Rate
- (4) Steering Wheel Rotational Inertia
- (5) P/S assist On and Off
- (6) Simulated Vehicle Speed

With regard to variable (6), in order to simulate different vehicle speeds, the P/S pump speed was varied to change flow rate and pressure. Later on during testing it was found that there was no change in response from this variation, so these results are

omitted. This response is expected because the system typically does not provide power assist on-center, where this event occurs.

For EPS, software was used to send the ECU an ignition signal and CANalyzer software was used to send the ECU a speed signal. It is important to simulate vehicle speed because at any given speed, there are unique power assist characteristics that have been intentionally tuned into the system; therefore, system behavior will vary from a static to dynamic condition. This is discussed in more detail in section 4.

After the validity of the test setup was confirmed, another sampling of on-vehicle steering shimmy data was referenced to find the proper load range for testing that phenomenon. As shown in Fig. 7, shimmy occurs at around 250 N at the tie rod. Therefore this would be our primary area of focus.

4. HPS Parameter Variation

4.1. Friction

When the setup was verified and ready, testing began by varying the aforementioned HPS variables (1-5) to find the more sensitive areas in the system. Each setup was run through a force sweep to study response at all input levels. First, the rack guide friction variables (1) and (2) were investigated. Figure 8 shows that varying rack guide screw torque changes the steering response. This was tested by tightening or backing off the rack guide screw by incremental amounts, thereby changing the rack guide's normal force and therefore the friction. Changing the rack guide material (2) has the same effect on system response.

This result is because of the threshold of static and kinetic friction between the steering rack and the rack guide. As that friction is increased via increased normal force, the threshold at which the rack transitions into dynamic oscillation increases. If this point is increased high enough, shimmy can be completely attenuated. However, even if that threshold cannot be moved above the shimmy load region for a particular vehicle, it still has an effect of reducing the vibration response due to the increased kinetic friction.

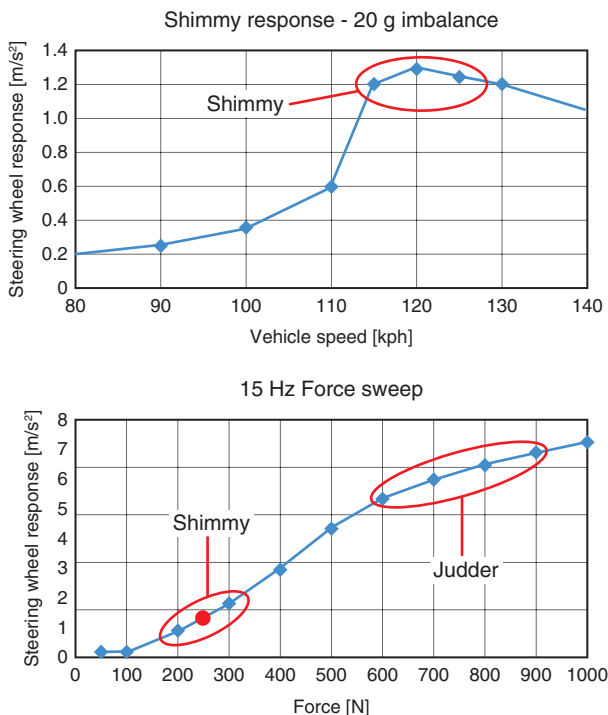


Fig. 7 Vehicle versus bench shimmy data

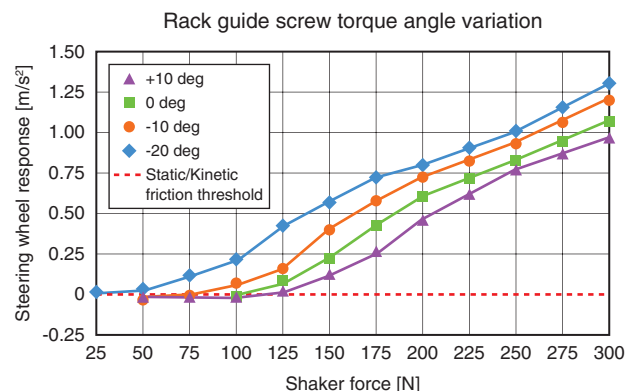


Fig. 8 G/box rack guide screw torque with HPS

4.2. Spring Rate

Next, the angular spring rate of the steering column coupler was varied to determine its effect on vibration attenuation. This potential solution had a notably different effect than a change in rack guide friction. As shown in Fig. 9, it had a noticeable effect at higher vibration levels, but had no discernable trend in the oscillatory loading range where shimmy happens.

The reason for this response is due to the nature of changing spring rates in a mass-spring-damper system. Given the frequency sweep run early on in this testing, it is clear that there exists a system natural frequency for this HPS system around 15 Hz. It is a physical property of harmonic motion that if the spring rate of a singular component of a system changes, the natural frequency of the entire system also changes. However, given the relatively small spring rate variation possible with this component, it is certain that any shift in resonant frequency will be too small to mitigate the rotational vibration at the steering wheel. This is shown at the much higher loads, where any frequency shift at all only resulted in a small change in response as opposed to meaningful attenuation.

4.3. Inertia

Moving further up the steering system, variation in steering wheel inertia was tested. Figure 10 shows the effects of this variation under a 250 N oscillatory load

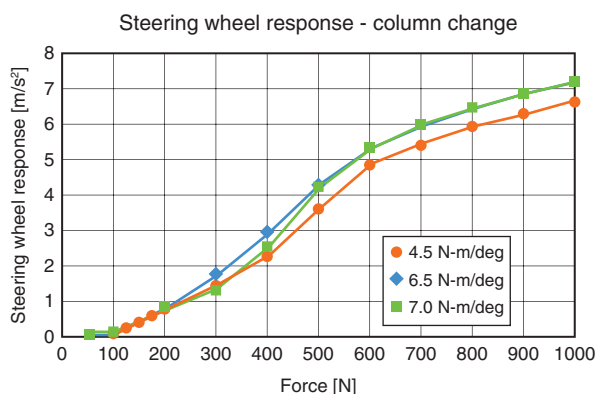


Fig. 9 Steering column coupler rate change

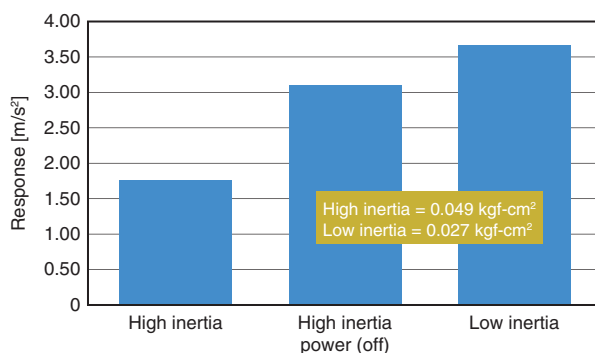


Fig. 10 Steering wheel inertia and power off

at 15 Hz. By cutting the steering wheel inertia in half, the vibration response is doubled.

This result is expected considering the basic equation $T = I\alpha$, which relates torque, T , to an object's moment of inertia, I , and angular acceleration, α . Since rotational vibration is a form of angular acceleration, at a given pinion torque, the response is directly dependent upon the inertia of the system. If the system has half the inertia, then it should have twice the angular acceleration for a given torque.

Another test involved checking the response with the P/S pump completely turned off. Although this is a scenario that is unlikely, it was tested for further understanding. As shown in Fig. 10, the damping of the system is enhanced with P/S fluid continuously circulating through the system.

5. EPS Parameter Variation

In a rack EPS system, a radial motor drives a ball screw mechanism that in turn provides power steering assist by pushing on the rack in the desired direction. Spatial limitations in an engine room require the motor to be relatively small and consequently, a sizeable gear ratio is needed to provide adequate assist. This gear ratio and rotational inertia of the motor have effects on system response to external inputs.

5.1. EPS versus HPS

5.1.1. Similarities

When varying the aforementioned mechanical variables, the EPS steering system response was predictably similar to HPS. For example, as shown in Fig. 11, changes in friction can cause the static/kinetic friction threshold to shift just as it did in the HPS setup. However, in comparison to Fig. 8, the EPS friction threshold and dynamic response is better attenuated than HPS. As mentioned before, the sizeable gear reduction provides a great deal of disturbance rejection to inputs such as vibration or kickback.

With changing variables (3) and (5), the EPS system response is virtually identical to HPS. This is an expected result. The principals that governed the

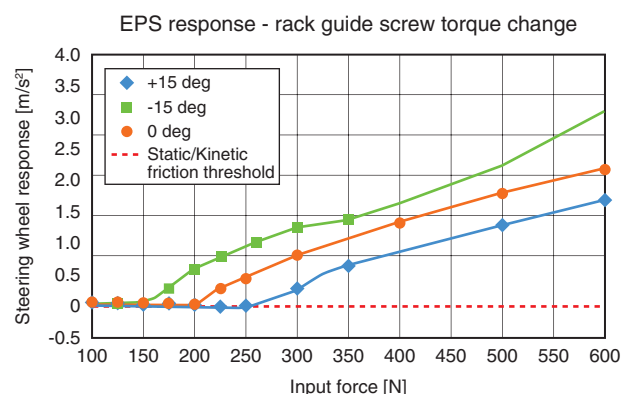


Fig. 11 G/Box rack guide screw torque with EPS

response to a column coupler rate change on HPS still apply in an EPS system. Also, as mentioned, EPS inherently possesses some level of damping as a result of the motor programming; therefore, it would respond just as an HPS system with P/S fluid flowing through it.

5.1.2. Differences

The effects of changing steering wheel inertia did have a unique result. As shown in Fig. 12, varying steering wheel inertia had no effect on system response in the shimmy load range. This is due to the relatively large gear reduction and motor inertia in a rack assist EPS system. The changes in inertia offered by different steering wheels are small by comparison although they did begin to have an effect at higher loads. The reasoning behind this result is somewhat similar to that of changing column coupler rates. Removing the steering wheel airbag had no discernable effect because the component's mass is essentially along the rotational axis.

5.2. Software Programming Variation

EPS system response was also tested by software tuning. In an EPS system, the steering control unit is constantly receiving a vehicle speed signal while the vehicle is in operation. This signal is useful because some steering assist software maps are speed-based, which allows an engineer to, for example, decrease

power assist when the vehicle is being driven straight (on-center) as speed increases. Such adjustments can enhance on-center stability.

Using CANalyzer software, the vehicle speed signal was varied and system response to the resonance frequency during a force sweep was recorded. Figure 13 shows the results.

There is no change in system response in the load range of the shimmy phenomenon. There is a pronounced effect at the higher load range, and the dynamic threshold is shifted when the vehicle is in motion versus statically. This shows the need to simulate vehicle speed, but in-motion response is virtually the same at all speeds.

6. Conclusion

In this study, it is shown that the mechanical tools for attenuating steering shimmy include increases to:

- Rack guide friction by increasing
 - Normal force (rack guide screw torque)
 - Material coefficient of friction
- Steering wheel inertia

However, contrary to what one might expect given the properties of system response to forced harmonic motion, this research showed that varying the column coupler rate has little effect on mitigating vibration for a small input. Another interesting point is that a power assist "on" condition is beneficial toward attenuation of steering shimmy. As shown in this research, EPS system response holds similarities with that of HPS. For example, steering shimmy can be attenuated by increasing system friction. However, this increase may not be necessary because of the already higher static/kinetic threshold that exists. There are differences also such as the system response to steering wheel inertia variation. Largely in part to the significant inertia of the EPS motor, changes to steering wheel inertia had no effect on system response in the load range of steering shimmy.

Additionally with EPS, software programming had to be considered. This research showed that the vehicle speed simulation caused system response to change everywhere except in the shimmy region. However, a speed signal did cause the static/kinetic threshold to shift for the resonant frequency of the system.

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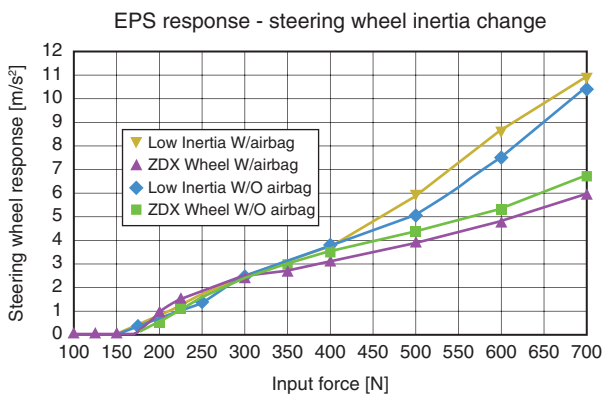


Fig. 12 Steering wheel inertia variation

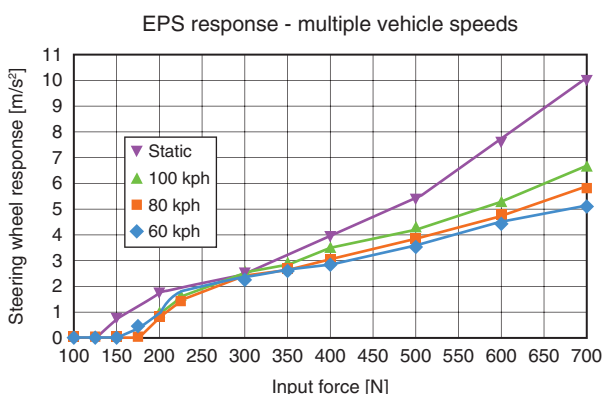


Fig. 13 EPS response under varied vehicle speeds

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