

Efficiency Analysis of Hydraulic Power Steering System

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ABSTRACT

Hydraulic assisted power steering system is a high bandwidth servo with stringent performance requirements on accuracy, reliability, and cost. Design of such a system can be best achieved by using a validated and user friendly computer simulation program. Hydraulic integrated power steering (HIPS) program has been developed using basic concepts from science and engineering. HIPS provides a design and test environment for the integrated steering and suspension system subjected to disturbance forces, which may be induced by pump flow oscillations and tire loads. Two real-world automotive hydraulic steering systems are simulated with HIPS. The simulation results agree closely with the dynamometer test results. The application of HIPS for design optimization is also demonstrated.

I. INTRODUCTION

[1] Figure 1 shows a hydraulic power steering system, which is a high bandwidth nonlinear servo capable of generating a rack force of 6,000 N for small trucks and 8,000 N for heavy trucks. About 90% of the sector shaft force comes from hydraulic power assist and the remaining 10% comes from driver's effort. At one-center position of the steering wheel, a vane pump, which is driven by the engine, circulates the fluid in a closed-loop hydraulic circuit which includes a reservoir, vane pump with flow control and pressure relief, supply and return lines, and the linear motion of spool valve (SV). The flow control valve which connects the discharge and inlet manifolds of the vane pump regulates the pump flow rate into the supply line at 8.5 lpm for trucks, for engine speeds above 900 rpm. When the steering wheel is turned by the driver, SV diverts the supply line fluid flow to either side of the power piston for a right or left turn of the vehicle. At the same time, SV passes an equal amount of fluid flow from the other side of the power piston through the return line to the reservoir. This increases the differential pressure acting on the power piston and yields the desirable hydraulic power assist force.

[2]An optional speed sensitive steering controller modulates the fluid flow rate drawn by the SV from the supply line, using signals from wheel speed sensors and wheel torque sensor. The driver's steering effort is increased by lowering the hydraulic power assist during highway driving,

hence improving driver's feel of the road during city and highway driving. The steering torque is obtained when the rack force is applied to an off-center joint on the knuckle, which turns about the kingpin at the suspension strut, including a coil spring and damper assembly.

[3]The hydraulic power assists level, which is proportional to the diverted amount of fluid flow rate from the supply line by the SV, is controlled by the relative angle between the steering wheel and pinion angles. This valve relative angle, which is also called valve error angle, is equal to the twist angle of the torsion bar. The torsion bar is connected to the steering wheel intermediate shaft and the pinion at its top and bottom ends respectively. Torsion bar stiffness yields the driver's steering effort, and is designed to let the driver to turn the steering wheel with ease, and at the same time, to give a memory function for the SV in order to reduce the error angle towards zero, after the end of the driver's steering demand. Since the SV is a servo valve, during on-center operation, it generates an opposing rack force to power piston motion subjected to tire shock and vibration loads.

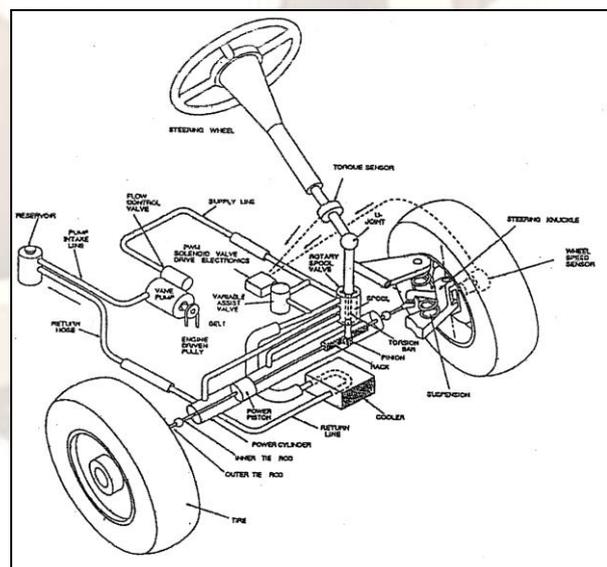


Figure 1 Hydraulic assisted power steering system

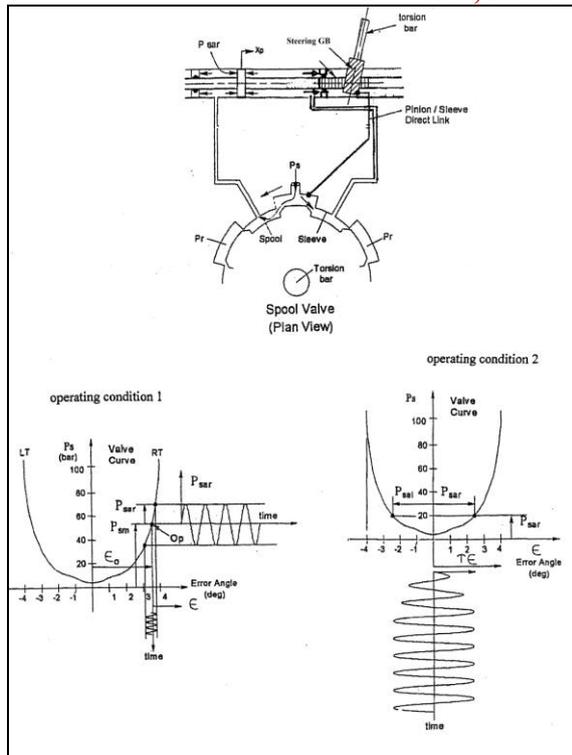


Figure 2 Spool Valve Operating Characteristics

Figure 2 shows a simplified schematic of the SV and its valve profile curves for two operating conditions. [3] In valve operating condition 1, occurring during parking, the valve error angle is large, and valve gain, which is proportional to the slope on the profile curve, is high. As shown on the related valve profile curve, any oscillatory rack motion resulting from oscillatory pump flow rates induces a high amplitude oscillatory pressure wave in the power cylinder of the steering gear. When this pressure wave excites the structural modes at the support frame, steering shudder occurs. Test results clearly show that the higher the slope of the valve profile curves at the operating point, the more the intensity of shudder. In valve operating condition 2 occurring during city and highway driving, any oscillatory rack motion caused by tire shock and vibration loads will be opposed by the hydraulic force resulting from the servo action of the rotary spool valve. This causes the formation of oscillatory pressure waves in both power cylinders. But the valve gain is low, and hence this pressure wave may not have sufficient intensity to generate shudder. The steering system noise, vibration, and harshness (NVH) related problems occur from dynamic interaction between the steering gear and suspension systems subjected to the disturbances, such as engine torque pulsation, tire shock and vibration loads. These NVH problems include:

1) Steering shudder; resulting from the excitation of the fundamental frequency at the frame support by a

fluid periodic flow force caused by the engine-driven vane pump.

2) Steering wheel nibble; resulting from the excitation of the fundamental frequency of the rack and pinion gear mechanism by a periodic rack force induced by brake disk roughness, during braking at highway speeds.

3) Mechanical front-end noise; resulting from the dynamic interaction between the rack and steering gear housing, caused by tire shock and vibration loads generated by tires riding on the bumps, stones, and pot holes on the road during low speed driving.

4) Steering wheel dither; caused by dynamic interaction between the steering gear and front suspension struts subjected to a periodic vertical tire load induced by a tire high spot at a certain highway speed.

Present solutions for steering system NVH problems mentioned above are usually achieved by using tuned hoses, shorter or longer hoses in the hydraulic lines, passive and active dampers, and reducing the SV gain. Since all of the above solutions are based on empirical rules, their adaptation to new car platforms would be time consuming and expensive. But, these solutions may cause an increase in power losses, and a reduction in the steering system bandwidth. The steering system NVH related problems are solved best, by using a validated and user-friendly computer simulation program. The hydraulic integrated power steering (HIPS) simulation program consists of two modules: Module 1 contains the steering gear model, and Module 2 contains the suspension model.

Two real-world examples are studied with HIPS. Example 1 is related to the front-end mechanical noise generated when a truck is driven at low speed on a bumpy road. Example 2 is related to the steering wheel dither generated when a truck with an unbalanced tire is driven at a certain speed on the highway.

II. MODELING

The most important requirement in modeling a system is the complete understanding of the performance specifications, physical and operational characteristics of each component in the system. The hydraulic rack and pinion power steering system of an automotive vehicle consists of hydraulic fluid lines, SV, vane pump including flow control and pressure relief valves, power actuator, inner and outer tie rods, lower and upper control arms, suspension struts, front and rear roll stabilizers, disturbances such as engine torque pulsation and tire loads. A logical modularization of the integrated system into steering gear and suspension component modules, and establishment of state variables with initial conditions in each module, is determined to avoid integration blow-up

and achieve correct interpretation of simulation results. This is a required method, since the integrated model of the real-world system under consideration consists of nonlinear components with discontinuous behavior. Figure 3 displays the modeling architecture of the HIPS simulation program.

1) Steering Gear Model.

- a) Closed-loop hydraulic circuit; including vane pump, flow control valve with pressure relief, tuned supply and return lines, cooler, reservoir, and SV.
- b) Power actuator; including the steering wheel, torsion bar, pinion gear, rack spring preload, power piston, tie rods, knuckle, tires, and housing.
- c) Driver commands for applying steering angle, steering rate, and engine rpm profiles.
- d) Disturbances; including pump flow rate oscillations and tire shock and vibration loads, which occur when driving on a rough terrain or with tire high spots on a highway.

The steering gear model is obtained by applying fundamental laws from fluid dynamics, heat transfer, and dynamics. The state variables are described by linear and nonlinear ordinary differential equations, with discontinuous behavior and temperature-dependent parameters.

2) Suspension Model.

- a) The sprung mass frame, with three degrees of freedom; heavy, roll, and pitch.
- b) Four unsprung masses, each with heavy motion.
- c) Four suspension struts. Each strut consists of a coil spring and a nonlinear damper. Each strut is connected between the movable end of the lower control arm and a corner of the frame or the upper control arm. Each control arm is pivoted about a rubber bushing connected to the frame. This gives heave motion degree-of-freedom to the tire. Each strut is tilted with respect to vertical direction, as defined by camber and caster angles, along the kingpin axis.
- d) Front-end and rear-end stabilizers; to maintain roll stability of the vehicle during cornering.

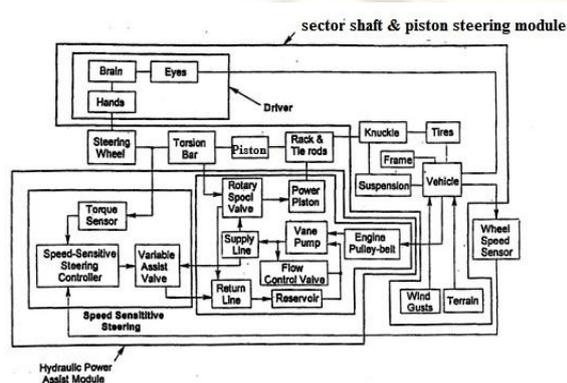


Figure 3 HIPS modeling architecture

III. LITERATURE REVIEW

HIPS is developed using the CSSL-IV language. HIPS validation is accomplished by means of two real-worlds NVH related steering problems. Example 1 is related to mechanical front-end noise, which may occur when a car is driven over a bump at low speed. Example 2 is related to steering wheel dither, which may occur when truck is driven at high speed on a highway.

1) Mechanical Front-End Noise:

The following simulation runs have been performed during the validation study:

a) Bump Test.

The Truck is driven at a speed of 8 mph and its right tire goes over a bump of 16 mm height. Figure 4 displays right-side tie rod response, denoted by FRTR, in comparison with dynamometer test data. Right side is defined with respect to driver's right side in forward direction. Figure 5 shows left-side tie rod response, denoted by FLTR, in comparison with dynamometer test data. Figure 6 and 7 shows the separation gap between rack and pinion gears, denoted by XRK3MM and ZRK3MM in forward and vertical directions respectively. The maximum amplitude and fundamental frequency values agree with those of dynamometer test data as displayed in Figures 6 through 9. Hand computations done on a 5 dof rigid-body model of the steering gear, predict an approximate value of 17.8 Hz for the fundamental frequency. The measured value of the fundamental frequency is 15.4 Hz.

b) Parking Test.

The steering wheel is rotated by the driver with a sine wave command signal having an amplitude of 360 degrees and frequency of 0.25 Hz. Figures 8 and 9 show that steering wheel command angle, denoted by ANGSW, is followed closely by pinion angle, denoted by ANGPN. In addition, the power piston differential pressure, denoted by DPPP, has correct amplitude and frequency. DPPP waveform does not show any sign of the steering shudder.

c) Optimization Test.

The system parameters representing torsion bar stiffness, rack spring preload, and bushing stiffness at the housing supports of the steering gear have been changed, one at a time, and corresponding simulation runs were carried out to obtain responses for XRK3MM and ZRK3MM.

The main goal of this study was to reduce the amplitude of above responses, without degrading the steering feel response, which is represented by the minimum error angle between ANGSW and ANGPN responses shown in Figure 8.

2) Steering Wheel Dither.

The hydraulic rack and pinion power steering system of a small truck, with 16 inch tires and independent front and solid-axle rear suspensions, has been studied using HIPS. The following results have been obtained:

a) Transient Response.

A small truck is driven at 80 mph on the highway. The right front tire, assumed to have a single high spot, is applying a periodic displacement with amplitude of 2.85 mm and frequency of 14.2 Hz to the suspension strut. The steering wheel is not held by the driver and is free to rotate. Figure 10 shows transient response for the torsion bar torque, denoted by TTB. Figure 11 shows transient response for the hydraulic power assist pressure, denoted by DPPP. It can be observed that a single tire high spot during highway drives at a speed of 80 mph, yields steering dither at a beat frequency of 1.5 Hz and carrier frequency of 14 Hz. This is shown by hand calculations as follows:

1. Simplify the steering gear model and determine the fundamental frequency, computed to be 11 Hz for the steering wheel, which is free to rotate, and 9.25 Hz for the steering wheel, which is held by the driver. An average fundamental frequency of 10 Hz is assumed.
2. Simplify the rigid body motion of the un sprung mass consisting of the tire-wheel assembly, coil spring and damper of the strut. The fundamental frequency of the simplified system is computed to be 9.4 Hz, assuming that the friction is negligible.
3. From rigid body dynamics, it is known that vibrations with beat frequencies may be generated if the forcing frequency and the average fundamental frequency of system differ slightly. For example, let the average fundamental frequency of the system be 10 Hz. Since the forcing frequency is about 14 Hz, then the carrier and beat frequencies are computed to be 12 Hz and 2 Hz respectively.

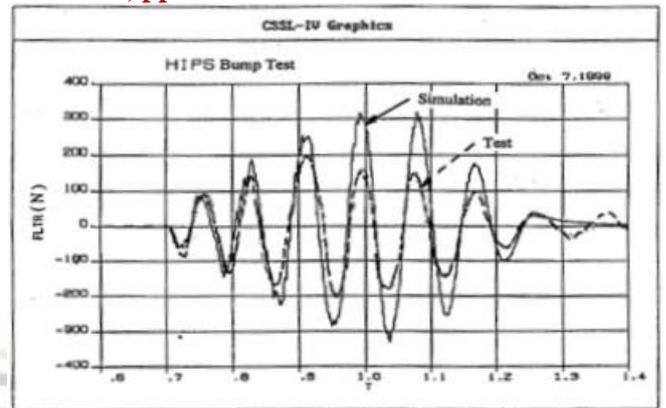


Figure 5 Left Side Tie Rod force response

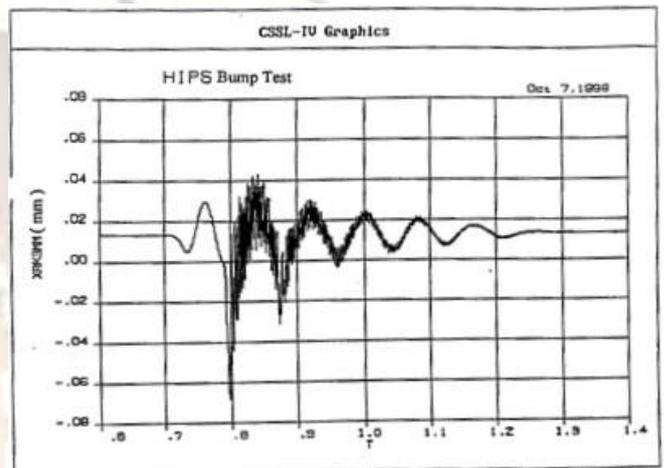


Figure 6 Transverse Gear Separation Responses

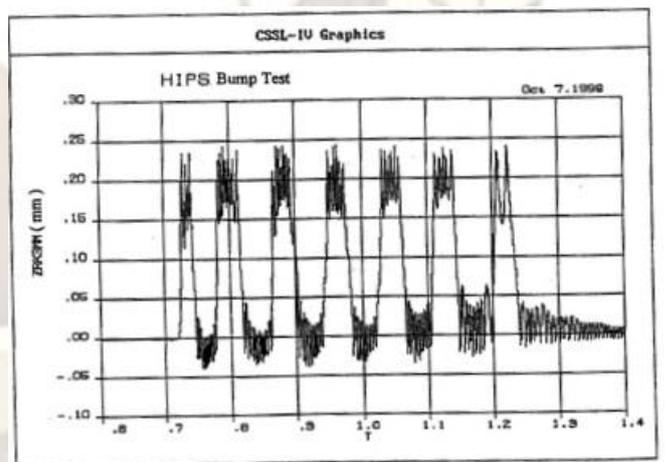


Figure 7 Vertical Gear Separation Responses

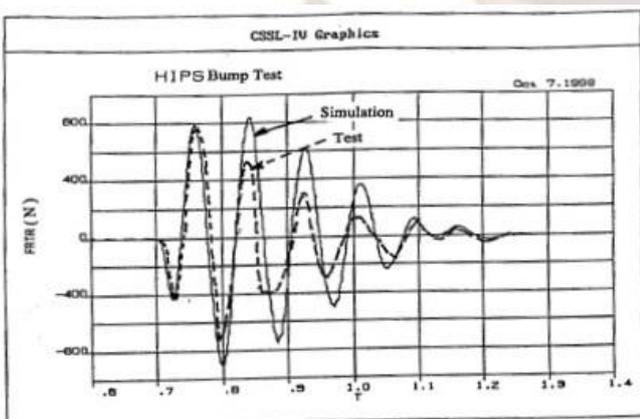


Figure 4 Right Side Tie Rod force response

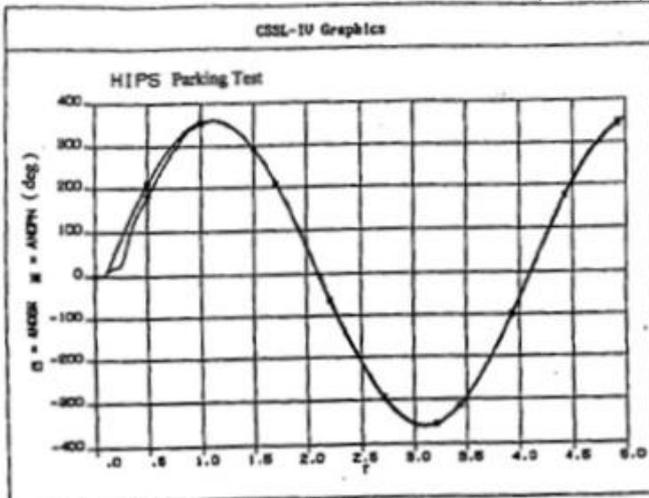


Figure 8 Steering Feel Responses

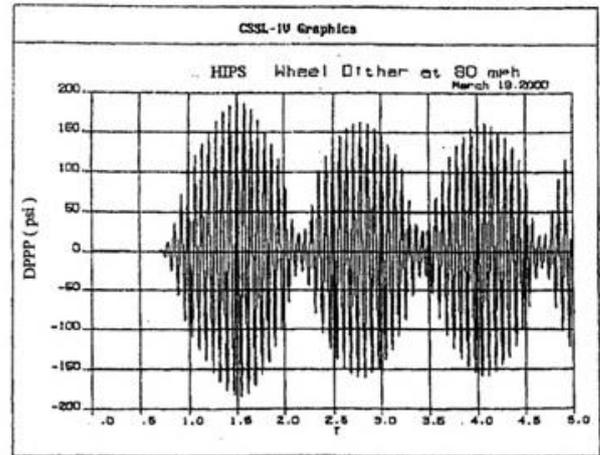


Figure 11 Hydraulic Power Assist Pressure Transient Response

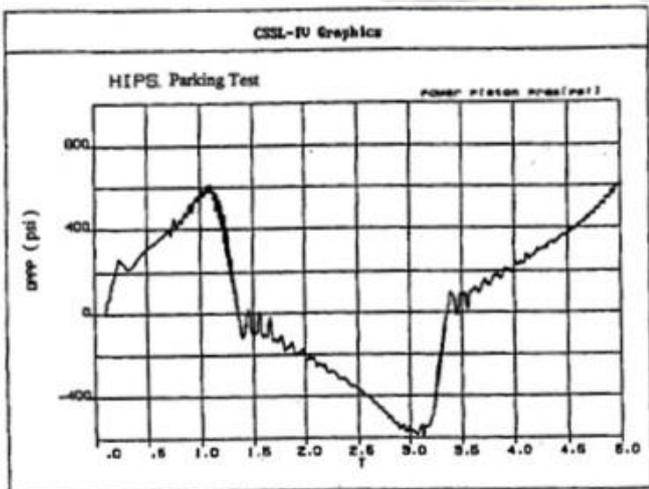


Figure 9 Steering power assist pressure Responses

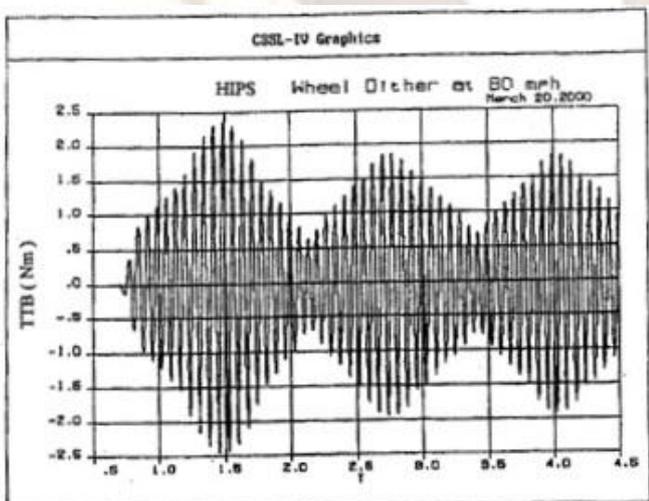


Figure 10 Torsion Bar Torque Transient Response

b) Frequency Response.

1. This task has been achieved in the following steps:
 Perform transient response for each of the following cases defined by tire high spot frequency or vehicle speed:

Vehicle Speed	Tire High Spot Frequency
50	8.8
60	10.6
70	12.5
80	14.2
90	16.0

2. For a system variable of interest, such as TTB, compute the Fourier sine and cosine coefficients over a period, after transients die away and the steady state conditions are attained, during the transient response simulation run for each of the above cases.

3. Obtain the root mean square (rms) value of TTB for each tire high spot frequency or vehicle highway speed. Figure 10 shows the frequency response with a resonance peak at 12.5 Hz.

4. This obtains the power spectrum density (psd) value of TTB by squaring its rms value and dividing it by a frequency window for each of the cases in Step 3. The frequency window is selected to be twenty times smaller than the highest forcing frequency, which is 16 Hz in this case. Figure 10 shows the TTB psd frequency spectrum, which shows a maximum power level at 12.5 Hz. This corresponds to a highway speed of 70 mph.

c) Optimization Test.

The torsion bar stiffness parameter, denoted by KTB, was changed from 1.37 to 1.13 Nm/deg, and a simulation run performed at a vehicle speed of 80 mph had shown an anticipated reduction

in dither torque at the steering wheel. Since such a change will also affect driver's steering feel, it is necessary to make additional simulation runs, as shown in Figures 10 and 11, and easily quantify the effects of such a change.

IV. CONCLUSIONS

Hydraulic power steering system design optimization can be achieved in a cost-effective manner by means of a computer modeling and simulation program including user-friendliness, modularity, flexibility, and a proven track record in the solution of NVH related problems.

The angular velocity of sector shaft applies the tangential and radial forces on teeth. The vehicle tyre applies the resistance to the linear movement of the piston.

The teeth are subjected to bending and wear stresses due to above forces. So to have proper steering gear ratio both bending and wear strength of the meshing teeth should be increased. To increase the strength the finite element analysis will be carried out to find the stresses in the teeth and suggesting the solution.

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