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Effects Of Tuner Parameters On Hydraulic Noise And Vibration

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ABSTRACT

Passengers' frequent requests are for less Noise, Vibration and Harshness (NVH) in the vehicle compartment. This and the reduction of noise and vibration levels from major sources like the engine necessitate better performance of other sources of noise and vibrations in a vehicle. Some of these sources are the hydraulic circuits including the power steering system.

Fluid pulses or pressure ripples, generated typically by a pump, become excitation forces to the structure of a vehicle or the steering gear and represent a considerable source of discomfort to the vehicle passengers. Current power steering technology attenuates this ripple along the pressure line connecting the pump to the steering gear. Finding the optimum design configuration for the components (hose, tuner, tube, and others) has been a matter of experience-based trial and error. This paper is a part of a program to simulate and optimize fluid borne noise in hydraulic circuits.

INTRODUCTION

A fluid borne noise program is being developed. This program uses a database of experimentally characterized pumps, hoses and steering gears, where a test rig is built and used to develop such databases according to recent ISO standards [1]. These data are then used in simulation software to determine the optimal design configuration for a specific application. This optimization procedure determines, among other things, finding the hose, tube, and tuner lengths in a power steering configuration for a specific driving condition. The most frequent driving conditions in which the power steering noise is noticeable include:

1. Idle - no load conditions. This is usually encountered while the driver is waiting in his vehicle (for example, in front of a traffic light). The driver usually leaves the engine running at 700 to 1000 rpm. Under this condition, no load is applied at the power steering system. The pressure in the fluid line can be up to 5 bar at the pump outlet to keep the system running and overcome energy losses. Drivers may experience this condition for extended periods in a traffic jam.
2. Parking conditions. These are encountered in the parking lot. The driver here is applying some load on the power steering system that will make the pressure reach up to 80 bar, while the engine is running at 800-1100 rpm. The duration of this condition is usually for several seconds (5 - 10 seconds).

3. Driving around corners in town. The driver usually runs the engine at speeds from 1500 to 2000 rpm, and puts the steering system under pressure values near 40 bar. The duration of this condition is approximated in seconds.
4. Steady state driving. This is a condition where the drivers experience when driving on the highway, where the engine is running between 1500 to 2500 rpm. The steering pressure is approximately 5 bar at the pump outlet and may reach 10 bar when changing lanes.
5. Accelerating while parking. This is experienced when the driver drives his vehicle in the parking lot fast. The driver, under this condition, will be driving his vehicle at 1500 rpm and putting up to 80 bar of pressure in the power steering system

The last two conditions are usually investigated for loading and durability and rarely studied for noise. The reason is that engine noise levels under these conditions exceeds power steering noise levels. The first three conditions described above are considered the most critical. They produce objectionable noise levels. As can be seen, for these conditions the steering system experiences pressure values from 5 to 80 bar and engine speeds of 800 to 2500 rpm. With these values in mind, the system is usually investigated under the controlled conditions described in Reference 8. The engine usually drives power steering pumps at similar speeds. Typical pumps contain 10 pumping elements (vanes). This results in critical frequencies of the tenth order and their harmonics.

Among other things, hoses, hoses with inserts or tuners, restrictors and other devices are used to attenuate the fluid ripple. The effect of hose parameters on the hydraulic system performance was studied earlier [6,7]. In this paper, certain important tuner parameters will be studied for their effects on the whole system NVH performance. This information should be valuable for design and research engineers involved in power steering systems. It should be mentioned that extensive correlation studies are being performed to correlate the theory established here with experiment.

POWER STEERING SYSTEM MODEL

The governing relations for input pressure and flow and output pressure and flow are needed for each component at any given frequency to assemble the system. We will first study the impedance of each of these components and then concatenate the system.

PUMP

The pump model is obtained by finding the impedance matrix coefficients in the frequency domain for a particular set of environmental conditions like fluid viscosity, mean flow, pressure and temperature. This will necessitate a series of measurements of input and output flow and pressure ripple amplitude and phase. Johnston and Edge [2] demonstrated that recorded measurement data could be incorporated in the system model. The experimental procedure required to obtain the impedance coefficients was discussed by Johnston and Drew [3]. It assumes linearity in the environmental conditions that are further detailed in the references.

A series of pumps has been modeled and can be selected from the simulator library. The library contains the non-parametric data for different pressures, pump speeds and temperatures. Some are described in recent publications [5-8].

TUBE

Tubes are one of the major components of the power steering system. They are typically made of steel and relatively inexpensive compared with other power steering components.

The complex amplitudes of the pressure and flow ripple at a position x along the pipe can be determined using the following equations:

$$P_x = P_F e^{-\gamma x} + P_R e^{\gamma x}, \qquad Q_x = P_x / Z_o$$

where P_F and P_R are the pressure waves at x=0 traveling in the forward and reverse directions respectively. The damping coefficient γ and the pipe impedance Z_0 are given by

$$\gamma = \frac{j\varpi}{c_0}\xi, \qquad Z_0 = \frac{\rho c_0}{A}\xi \qquad \text{where} \quad c_0 = \sqrt{\frac{B_E}{\rho}}$$

where ρ is the fluid density, A is the cross sectional area of the pipe, B_E is the effective bulk modulus of the fluid in the pipe and ϖ is the frequency. It should be noted that the effective bulk modulus B_E should be used as this considers the bulk modulus of both the fluid and the tubing system [1]. ξ is defined by

$$\xi = \left[1 - \frac{2}{zj^{3/2}}\frac{J_1\left(zj^{3/2}\right)}{J_0\left(zj^{3/2}\right)}\right]^{-\frac{1}{2}}, \quad where \quad z = r_p\sqrt{\frac{\rho\varpi}{\mu}}$$

μ is the absolute viscosity, r_p is the radius of the pipe, and J_o and J_1 are Bessel functions. Letting l_h be the pipe length, the transfer matrix for the pipe can be formed as

$$Z = Z_0\begin{bmatrix} \coth(\gamma l_h) & \operatorname{cosech}(\gamma l_h) \\ \operatorname{cosech}(\gamma l_h) & \coth(\gamma l_h) \end{bmatrix}$$

A more detailed fundamental discussion on wave propagation along a fluid filled tubes is given in the literature [1,2,3].

HOSE

Various types of hoses are available in the market place that can be used as a part of the power steering assembly. These include low and high expansion hoses with various types of brading construction and materials. Some hoses may include tuners. These are proven to attenuate more fluid ripple than others.

For a plain hose, an impedance model must be derived relating input pressure and flow ripples to the output pressure and flow ripples. Two physical relations define the impedance of a plain hose. They represent the pressure-volume relation and the fluid motion respectively and are given by [1]

$$\frac{A_b}{A_r}\gamma \mathrm{U} + \frac{1}{B_E}\mathrm{P} = 0,$$
$$\mathrm{U} + \left(\frac{(1-j)}{r_b \varpi^2 \rho}\sqrt{\frac{2\mu}{\rho\varpi}} - \frac{1}{\varpi^2 \rho} \right)\gamma \mathrm{P} = 0$$

A_b is the bore cross sectional area of the hose (i.e., the area containing the fluid), A_r is the cross sectional area contained by the reinforcement. U is the complex fluid motion and P is the complex pressure and

$$\frac{1}{B_E} = \frac{A_b}{A_r B_{\mathrm{fluid}}} + \frac{A_l}{A_r B_{\mathrm{lining}}} + \frac{2r_r(1-\nu_x \nu_y)}{E_y}.$$

where A_l is the cross sectional area of the lining, B_{fluid} is the fluid bulk modulus, B_{lining} is the lining bulk modulus, ν_x and ν_y are Poisson's ratios in the x and y directions respectively, E_y is the modulus of elasticity in the y-direction. Assuming a harmonic solution $U = U_o e^{\gamma x}$ and $P = P_o e^{\gamma x}$, a characteristic equation can be written as:

$$\frac{B_E A_b}{A_r}\left(\frac{1-j}{r_b \varpi^2 \rho}\sqrt{\frac{2\mu}{\rho\varpi}} - \frac{1}{\varpi^2 \rho} \right)\gamma^2 + 1 = 0$$

The general solutions allows for U and P to be written at the boundaries as:

$$\mathrm{U}_{x=0} = K_1 + K_2$$
$$\mathrm{U}_{x=l_h} = K_1 e^{\gamma l_h} + K_2 e^{-\gamma l_h}$$
$$\mathrm{P}_{x=0} = K_1 - K_2$$
$$\mathrm{P}_{x=l_h} = \frac{B_E A_b}{A_r}\gamma\left(K_1 e^{\gamma l_h} - K_2 e^{-\gamma l_h}\right)$$

K_1 and *K_2* can be solved for by considering the boundary conditions at the two hose ends. With these formulations, the impedance matrix for the hose can be formulated as

$$Z = \frac{B_E A_b^2}{j\varpi A_r}\gamma \begin{bmatrix} 1 & -1 \\ e^{\gamma l_h} & -e^{-\gamma l_h} \end{bmatrix} \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \end{bmatrix} \left(\begin{bmatrix} 1 & 1 \\ e^{\gamma l_h} & e^{-\gamma l_h} \end{bmatrix} \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \end{bmatrix} \right)^{-1}$$

A more detailed fundamental discussion on wave propagation along a fluid filled hose is given by Longmore [4], with a more detailed account of the theoretical approach given by Tuc [5]. Some derivations incorporate hose wall motion [1].

TUNED HOSE

For a tuned hose, an impedance model must be derived that relates the pressure and flow into the insert at the inlet of the hose, to the pressure and flow at the outlet, where there is the gap. The gap-section for x=0…l_h, is analyzed as plain hose, and thus complies with the derivations of the previous section. The section of the hose that contains the insert is derived below. Three physical relations define the impedance of a tuned hose. They represent the pressure-volume relation and the fluid motion respectively and are given by

$$U - C_U \gamma P_i = 0$$
$$j\omega\gamma U + \left(\frac{k}{A_i} + \frac{j\omega}{B_b} \right) P_i - \frac{k}{A_i} P_a = 0$$
$$\frac{k}{A_r} P_i + \left(\frac{k}{A_r} + \frac{j\omega}{B_E} \right) P_a = 0$$

A_i is the bore cross sectional area of the insert, A_r is the cross sectional area contained by the reinforcement. U is the complex fluid motion and P is the complex pressure and

$$\frac{1}{B_E} = \frac{A_a}{A_r B_{\mathrm{fluid}}} + \frac{A_l}{A_r B_{\mathrm{lining}}} + \frac{2r_r(1-\nu_x \nu_y)}{E_y}.$$

where A_a is the area between the lining and the outer wall of the insert. Assuming a harmonic solution $U = U_o e^{\gamma x}$ and $P = P_o e^{\gamma x}$, a characteristic equation in γ can be written as:

$$\left(j\omega C_U \frac{k}{A_r} - \omega^2 C_U \frac{1}{B_E}\right)\gamma^2 +$$

$$\left(\frac{2k^2}{A_i A_r} + j\omega\left(\frac{k}{A_i B_E} + \frac{k}{A_r B_b}\right) - \omega^2 \frac{1}{B_E B_b}\right) = 0$$

where

$$C_U = \left(1 + \frac{(j-1)\sqrt{2}}{r_i}\sqrt{\frac{\mu}{\rho\omega}}\right)\frac{1}{\rho\omega^2}$$

The general solutions allows for U and P to be written at the boundaries as:

$$\mathrm{U}_{x=-l_t} = K_1 e^{-\gamma l_t} + K_2 e^{\gamma l_t}$$

$$\mathrm{P}_{x=-l_t} = C_i \gamma \left(K_1 e^{-\gamma l_t} - K_2 e^{\gamma l_t}\right)$$

$$\mathrm{U}_{x=0} = K_1 + K_2 = K_3 + K_4$$

$$\mathrm{P}_{x=0} = C_i \gamma (K_1 - K_2) = C_a \gamma (K_1 - K_2)$$

$$= \frac{B_E A_b}{A_r}\gamma (K_3 - K_4)$$

$$\mathrm{U}_{x=l_h} = K_3 e^{\gamma l_h} + K_4 e^{-\gamma l_h}$$

$$\mathrm{P}_{x=l_h} = \frac{B_E A_b}{A_r}\gamma\left(K_3 e^{\gamma l_h} - K_4 e^{-\gamma l_h}\right)$$

with K_3 and K_4 are the boundary conditions for the plain hose section, and K_1 and K_2 are for the tuned section, and

$$C_i = \frac{j\omega\left(\frac{k}{A_r} + \frac{j\omega}{B_E}\right)}{\frac{2k^2}{A_i A_r} + \frac{j\omega k}{B_b A_r} + \frac{j\omega k}{A_i B_E} - \frac{\omega^2}{B_b B_E}},$$

$$C_a = \frac{-j\omega}{\frac{2k}{A_i} + \frac{j\omega}{B_b} + \frac{j\omega A_r}{A_i B_E} - \frac{\omega^2 A_r}{k B_b B_E}}$$

The above equations can be solved by considering the boundary conditions at the two ends. With these formulations, the impedance matrix for the hose can be formulated

$$Z = \begin{bmatrix} C_i \gamma e^{-\gamma l_t} & C_i \gamma e^{-\gamma l_t} & 0 & 0 \\ 0 & 0 & \frac{B_E A_b}{A_r}\gamma e^{\gamma l_h} & -\frac{B_E A_b}{A_r}\gamma e^{-\gamma l_h} \end{bmatrix} \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \\ K_3 & 0 \\ 0 & K_4 \end{bmatrix}$$

$$\left(\begin{bmatrix} e^{-\gamma l_t} & e^{\gamma l_t} & 0 & 0 \\ 0 & 0 & e^{\gamma l_h} & e^{-\gamma l_h} \end{bmatrix} \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \\ K_3 & 0 \\ 0 & K_4 \end{bmatrix}\right)^{-1}.$$

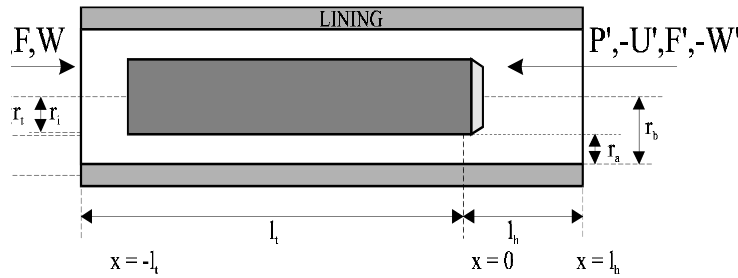

Figure 1 The geometric design of the tuned hose.

STEERING GEAR

At this point in time, the steering gear is considered to be the end of the fluid transmission line. This is true as far as high pressure is concerned. The steering gear is not modeled as a transfer or impedance matrix, but as a "point impedance", represented by a single frequency dependent equation for pressure and flow.

The relation will be read from a library of characteristic measurement files for various brands steering gear-assembly. Johnston and Edge [6] outline the experimental method of measuring component impedance.

SYSTEM CONCATENATION

A two-by-two matrix relating input and output pressures and flows will represent the components in the fluid line. This matrix can be written either as a transfer matrix or as an impedance matrix:

$$\begin{bmatrix} P_{in} \\ P_{out} \end{bmatrix} = \begin{bmatrix} z_{11} & z_{12} \\ z_{21} & z_{22} \end{bmatrix} \begin{bmatrix} Q_{in} \\ Q_{out} \end{bmatrix} \qquad \begin{bmatrix} P_{out} \\ Q_{out} \end{bmatrix} = \begin{bmatrix} t_{11} & t_{12} \\ t_{21} & t_{22} \end{bmatrix} \begin{bmatrix} P_{in} \\ Q_{in} \end{bmatrix}$$

where $t_{11} \ldots t_{22}$ are the elements of the transfer matrix and $z_{11} \ldots z_{22}$ are the elements of the impedance matrix.

SIMULATING PRESSURE AND FLOW

Once the various components of a power steering system have been modeled, it is possible to derive the pressure and flow ripple at any point along the fluid line. Generally, the high-pressure line of a power steering system will consist of a hose, connected to the steering gear and the pump with steel pipe. The model for such a system is shown in Figure 2. To compute the pressure and flow ripple P_1 and Q_1, at the outlet of the pump, first equivalent impedance must be derived for the transmission line and the steering gear. This can be found as

$$Z_{eq} = \frac{t_{22} R - t_{12}}{t_{11} - t_{21} R}$$

where R is the impedance of the steering gear and $t_{11} \ldots t_{22}$ are the elements of the equivalent transfer matrix of the pressure line. This matrix can be found by multiplying the component transfer matrices. The pressure at the outlet of the pump is

$$P_1 = \frac{Z_i Z_{eq}}{Z_i + Z_{eq}} Q_s$$

where Z_i is the internal impedance for the pump and Q_s is the pump flow ripple. The flow ripple at the pump outlet can then be found by dividing the pressure ripple by Z_{eq} as

$$Q_1 = \frac{P_1}{Z_{eq}}.$$

Pressure and flow ripples at different locations can be found by multiplying P_1 and Q_1 with the transfer matrix of the components between the pump and the particular location.

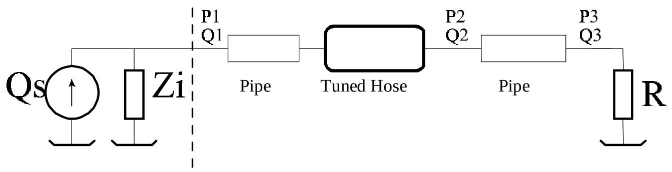

Figure 2 General model representation of a power steering system.

TUNED HOSE PARAMETERS

Simulation of the pressure ripple at the pump outlet and the steering gear inlet or any other location in the system enables us to predict the noise and vibration, excited from a particular location.

Figure 3 shows the source impedance and ripple obtained for a typical power steering pump used in a mid-size vehicle. Figure 4 shows the impedance obtained for a steering gear used in a similar vehicle. These will be used in the subsequent studies. For consistency in all the following results, a standard 0.5 m tube is attached to the outlet of the pump before the attenuation device and a similar one is linked to the steering gear.

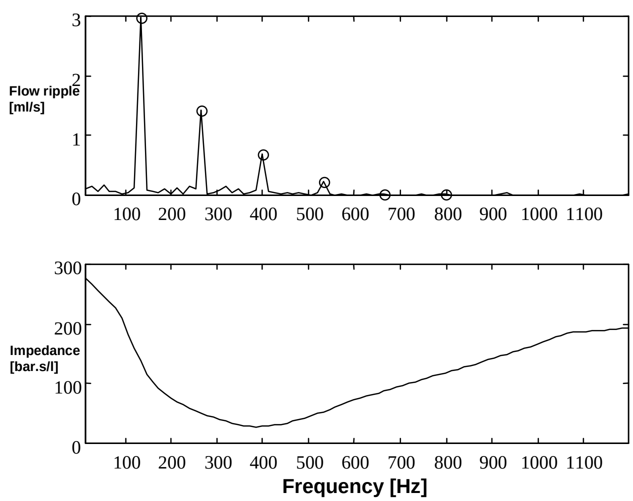

Figure 3 Flow ripple and impedance of a typical power steering pump at 10 bar, 800 rpm.

We will first use a typical 0.5 m tuned hose. The hose geometric parameters are experimentally found from a volumetric expansion test, and the material properties are found from a genetic algorithm routine.

The insert or tuner dimensions are:
Tuner inner radius = 3 mm
Tuner outer radius = 4 mm

We will study the system sensitivity to following parameters:

1. Tuner and gap lengths
2. Tuner inner radius
3. Tuner outer radius
4. Leakage coefficient

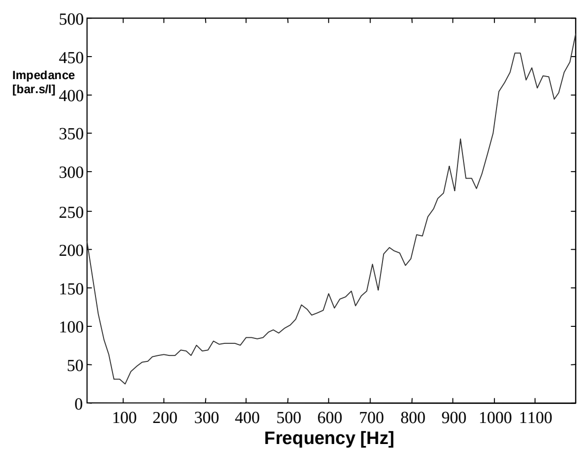

Figure 4 Impedance of a steering gear loaded in the clockwise direction at 10 bar

TUNER LENGTH

Figure 5 shows the effect of the tuner length on the tenth and twentieth orders of the pressure ripple values. As can be seen, the optimal value of the tuner length depends on the annoying frequency that needs to be attenuated. A correlation between audible noise inside the vehicle compartment and the fluid borne noise is needed to determine these annoying frequencies and their locations.

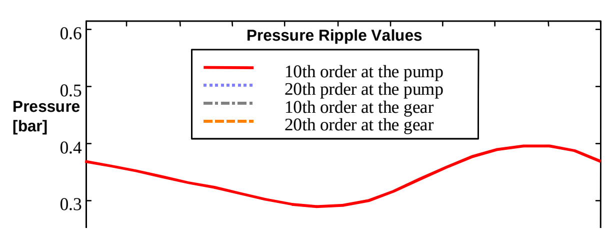

Fig. 5 Effect of tuner length on system Performance

TUNER INNER RADIUS

Figure 6 below shows the effect of the tuner inner radius on the 10th and 20th orders at the pump and steering gear. A tuner length of 0.25 m is used in these studies. The tuner inner radius is varied between 2 and 4 mm., keeping the outer radius of the tuner at 4 mm. Smaller inner radius values is showing less pressure ripple in the system. This is to be expected because the fluid in the annulus between the tuner and hose inner radius is stationary and acting as a fluid damper to the pulse.

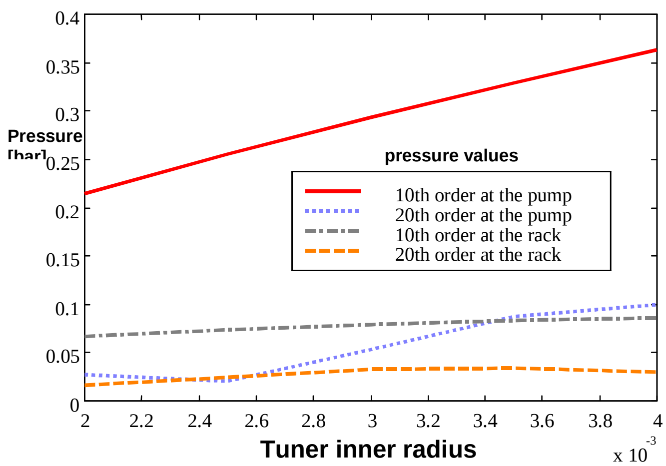

Figure 6 Tuner's inner radius effects on system performance.

TUNER OUTER RADIUS

Figure 7 below show the effect of the outer radius of the tuner on the system performance. The outer radius is varied between 3.5 and 5 mm.

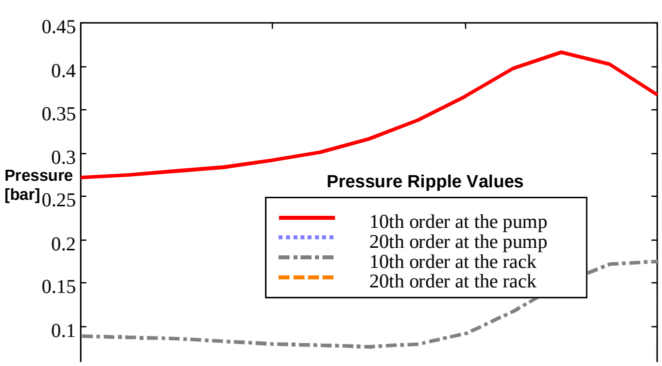

Figure 7 Effects of tuner outer radius

LEAKAGE COEFFICIENT

The leakage coefficient is varied between ($1X10^{-8}$) and ($1X10^{-11}$). These are theoretical values and correct values need to be found for the specific tuner at hand. Figure 8 below shows how the leakage coefficients can affect the system performance for the 10th and 20th orders at the pump and gear. Leakage coefficient describes the bleeding rate these tuners have from their bore area to the annulus area between the tuner and the bore radius of the hose used.

Detailed studies of the leakage coefficients can be found in reference 2.

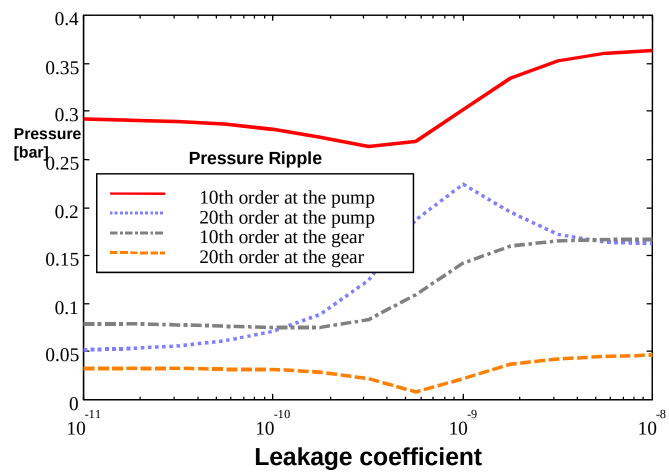

Figure 8 Effects of leakage coefficient on system performance.

CONCLUSIONS

It is found that careful studies of certain parameters in a tuned hose will yield a better understanding of the hydraulic circuit and ultimately better design from an NVH perspective. Results obtained in this study will help design engineers and manufacturers of fluid lines produce better fluid lines and inserts for attenuating ripple in hydraulic circuits.

Better optimization techniques are needed to find an optimal design for a particular condition in a vehicle. Furthermore, repeatable transfer path analysis will help relate the fluid pulses to sound levels inside the vehicle. This as well as extensive correlation studies are being included in future studies of the research team at Dana's Fluid System Products. It is hoped that such a comprehensive program to hydraulic noise attenuation will yield better performance of hydraulic circuits.

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1. Hose bore radius = 5.23 mm
2. Hose reinforcement radius = 7.51 mm
3. Hose properties:
   Ex = 557429+41886i
   Be = 291840339+26042588i

Theta= -0.18088-0.0065689I

Leakage coefficients is 1.4 e -11.