Optimizing the Power Steering Components to Attenuate Noise and Vibrations

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Higher demands for less noise from automotive customers coupled with 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. One of these sources is the power steering system. Noise can be transmitted directly from its source, like the pump in a hydraulic system. Objectionable fluid borne noise and vibrations exhibit as a pressure ripple or flow pulsation through fluid lines. This will become an excitation force to the structure of a vehicle or the steering gear and is 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 configuration for the components (those, tuner, tube e.g.) has been a matter of experience-based trial and error.
A fluid borne noise simulation software program has been developed. It uses the MATLAB language to determine the optimal configuration for a specific application. This optimization procedure determines, among other things, finding the optimal hose length in a power steering configuration. Optimization is accomplished on the basis of ripple attenuation at any point along the fluid line. The output results have been verified and validated against measurement data and other validated simulation software.

1. INTRODUCTION

Hydraulic systems in a vehicle are sources of objectionable noise. The noise can originate from pressure and flow fluctuations generated by the pump. When the fluctuations are transmitted along the hydraulic lines, they cause forces at pipe bends and various couplings and the steering gear, which result in mechanical vibrations that in turn cause objectionable audible sounds.
Research can be done to the internal design of pumps to minimize the ripples in the output pressure and/or flow. In many practical applications this can be a costly task. Furthermore, most hose assembly manufacturers do not have a choice on the pump selected by the Original Equipment Manufacturer (OEM). It is thus the objective of this study to configure the fluid line between the pump and the steering gear such that the pressure ripple at any specified location is minimized.

In a typical application, the pump type and location, the steering gear type and location, and possible alternative routings for the fluid lines are given. In these alternative routings, the fluid line may be connected to the structure at various locations. The procedure described in this paper allows optimizing the fluid ripple with respect to a specific point along the fluid line. At the present time, many of the design rules are based on the experience and knowledge of the Noise Vibration and Harshness (NVH)-engineers.
The work that is presented in this paper will provide NVH-engineers with a simulation software program that can predict system performance in terms of pressure and flow ripple anywhere between the pump and the steering gear. It can also predict important features like transmission loss for each component in the system.
The next two sections explain how the pump, steering gear and other components are modeled and characterized. In a following section, the optimization procedure is explained. Conclusions of this study are then described at the end of the paper.

2. 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.

2.1. 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 can 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 have been modeled and can be selected from the simulator library. The library contains the non-parametric data for different pressures, speeds (i.e., flow) and temperatures. An example of the model data is shown in Figure 1.

2.2. Tube

Tube is a major component of the power steering system. It is typically made of steel and relatively inexpensive compared with other power steering components.

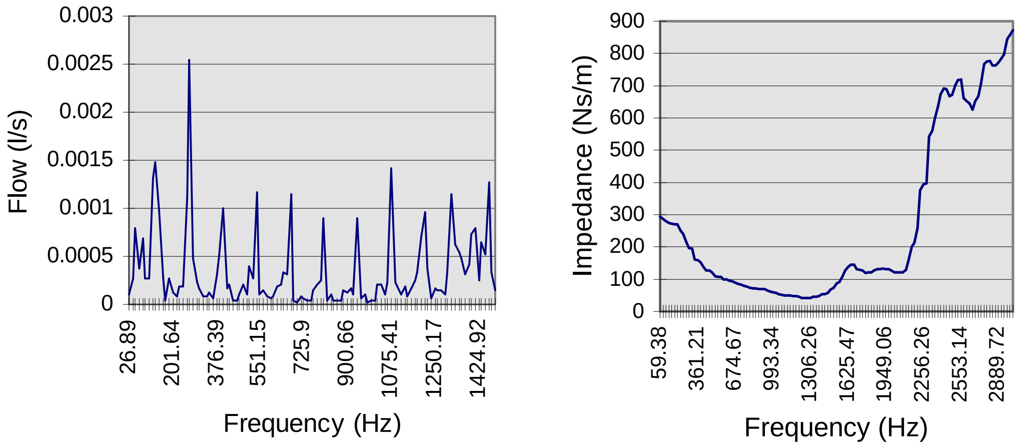

Figure 1 Typical data for the pump flow ripple and the internal impedance. This data represents the Jaguar pump at 800 rpm, 60 bar mean pressure and 70°C.

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

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

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

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

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 hose may include a tuner. These are proven to attenuate more fluid ripple than others.

The work presented in this paper can be used for various types of hoses. We will however focus solely on optimizing the length of a plain hose. 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]

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

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

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

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 in γ 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:

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

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

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

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]. The derivations that also incorporate hose wall motions can be found in [1].

2.4. 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.

2.5. 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.

3. 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.

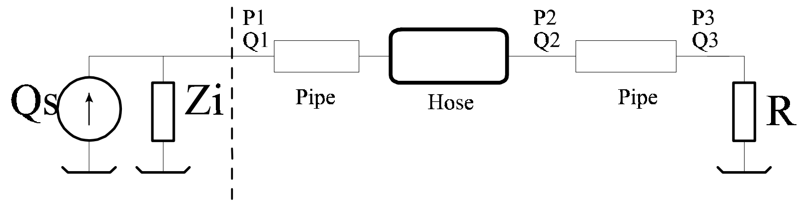

Figure 2 General model representation of a power steering system

4. SIMULATION AND OPTIMIZATION

A simple power steering configuration will be taken in the application of optimizing for noise and vibrations attenuation. The pressure line of the system consists of a pipe of 0.1 m, a hose and again a pipe of 0.1 m. A model for this system is shown in Figure 2. It can be seen from the simulation that the pressure ripple only decreases slightly due to the hose. The predicted pressures at the inlet and outlet of the hose are plotted in Figure 3.

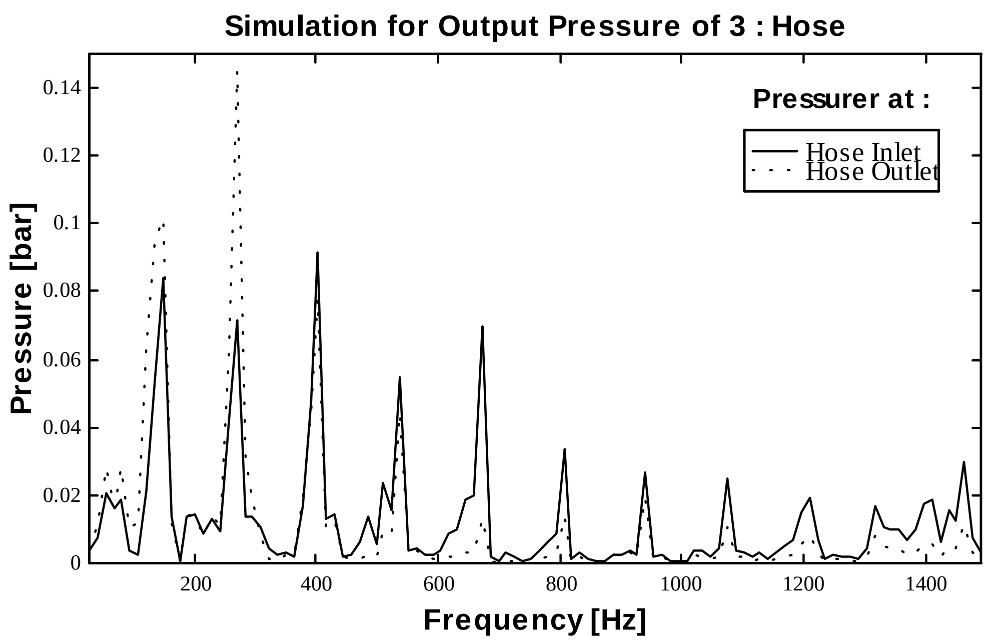

Figure 3 Pressure ripple at the inlet and the outlet of the hose.

It is the purpose of optimization to look for the hose length that minimizes the ripple (i.e. maximizes the ripple attenuation) in the hose. Pressures at the outlet for different lengths of hose show the effect of attenuation. The simulation for 4 different hose lengths is plotted in Figure 4. The graph shows that a hose length of 0.7m performs best for 133 Hz, which is the 10th order of the pump that consists of 10 vanes. However, the same length shows worst result at the 20th order.

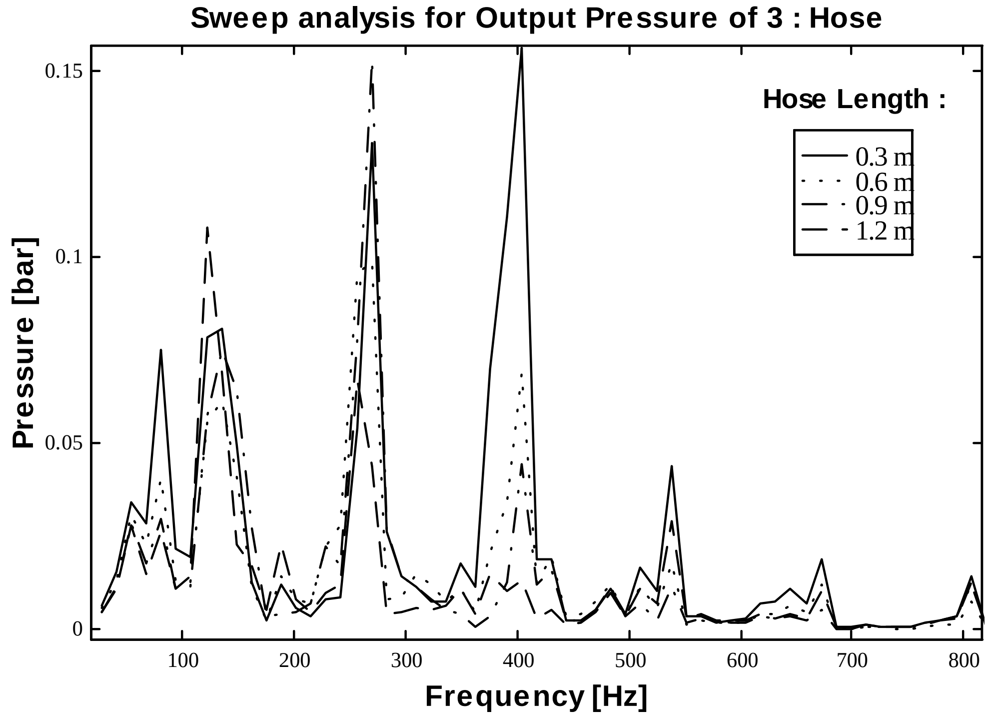

Figure 4 Pressure at the outlet of the hose for different hose lengths.

The speed at which the pump operates depends on the engine's speed. It is thus of interest to study the attenuation at the higher order frequencies of the motor speed. Results of the simulation of the pressure ripple at the outlet of the hose with respect to its length for these particular frequencies are given in Figure 5. The results show indeed that 0.7m has poor performance at the 20th order frequency. It is interesting to note that a slightly shorter hose would perform excellent for the same frequency. The graph in Figure 5 shows that best attenuation can be achieved for a hose length of approximately 0.65m. This could be expected, since it is close to the ½ wavelength of the particular frequency. At this wavelength the pressure ripple forms a standing wave inside the hose, that will absorb the energy from the pressure ripple. Since the effective bulk modulus is 2.5 10^8 N/m^2 and the density of the fluid is 840 kg/m^3[1], the speed of sound inside the hose is

$$c = \sqrt{B_E / \rho} \approx 546\ m/s$$

For the 20th order frequency at 800 rpm, which is (800/60)*20=266 Hz, this means that the ½ wavelength can be found by solving

$$l_{0.5} = \frac{\lambda}{\pi} = \frac{1}{f_n \pi}\sqrt{\frac{B_E}{\rho}}$$

which is just 0.65 m. It can also be seen, that close to, but not exactly at the ½ wavelength, the attenuation is worse than elsewhere. This is due to the fact that the pressure ripple at the hose ends are close to being in phase with the first order reflected image of the pressure ripple inside the hose. This means that a pressure ripple will enter the hose, suffers reflection at the hose end and travel almost one wavelength when it reaches the same point in the hose. Therefore the pressures will be almost in phase and thus they will add. Only very little energy will be absorbed, since there is no standing wave. The effect can be seen most clearly for the hose length analysis at the 20th order in Figure 5.

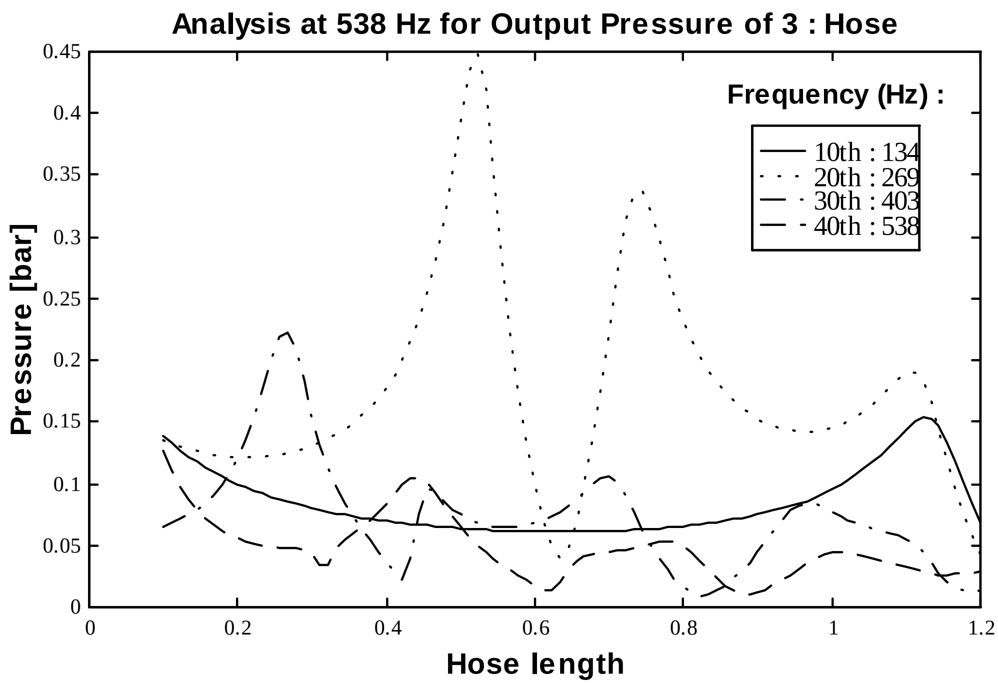

Figure 5 Pressure ripple at the outlet of the hose for varying hose lengths at excitation frequencies.

Selecting the hose length to be 0.65m, the pressure ripple is shown in Figure 6.

In order to automate this optimization procedure, weight functions W_f can be applied to each of the pressure ripple curves for the frequency modes in Figure 5, denoting the relative need to suppress the noise at the particular frequency. These weight functions are based on the noise levels, or vibration levels, produced by each of these frequencies in actual testing conditions. They can also be based on the experience of the engineer interested in minimizing noise in the power steering, or other hydraulic, systems. The pressure function is

$$P(l_h) = \sum_f W_f \, P_f \left(l_h\right).$$

A gradient search algorithm could then check for the best hose length. This will be the

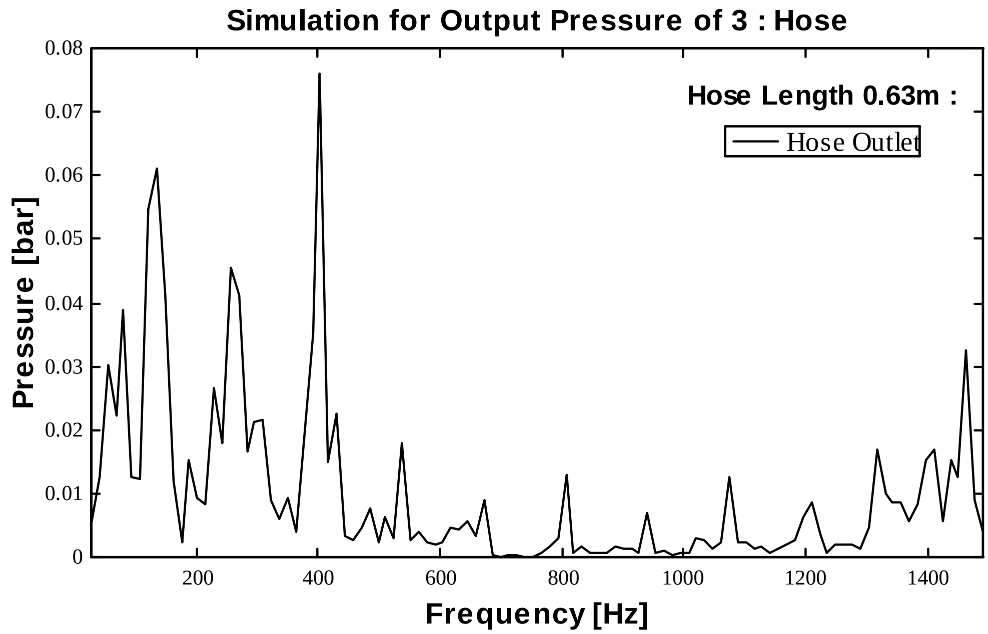

Figure 6 Simulation of the pressure ripple at the hose outlet for l_h=0.65m.

hose length where the pressure ripple is the smallest of the local minimum found for P_{lh}:

$$\hat{l}_h = \min_{P(l_h)} \left\{ l_h \left| \frac{\partial P(l_h)}{\partial l_h} = 0 \cap \frac{\partial^2 P(l_h)}{\partial {l_h}^2} \geq 0 \right. \right\}$$

Obviously, the curves in the graphs above will show different results for different speeds of the pump. One needs to know at what frequency excessive objectionable noise occurs, so that the model can be set up accurately. After finding the best attenuation from the graph in Figure 5, a frequency sweep can be performed to understand where the system generates most noise or vibration. These frequencies should not coincide with mechanical resonance in the vehicle. The results of a simulated spectrum are given in Figure 7. It can be seen there that hose length of 0.63m shows very good performance for 266 rev/min. However it always needs to be considered that performance might degrade at other frequencies. Performance degradation did not occur in this particular case.

5. CONCLUSIONS

A model of fluid axial flow in the high pressure line of a power steering system has been written in MATLAB. This allows the travel of hydraulic pressure pulses to be calculated. Simulation for different hose lengths show that plain hose attenuates best when the hose length coincides with the pressure ripple ¼ wavelength, and least efficient when it co-incides with the ½ wavelength.

The use of optimization algorithms will contribute heavily in the automation process of the simulator. Engineers could then import the vehicle characteristics, together with the pump and steering gear data, whereas the simulator would simply return the optimal

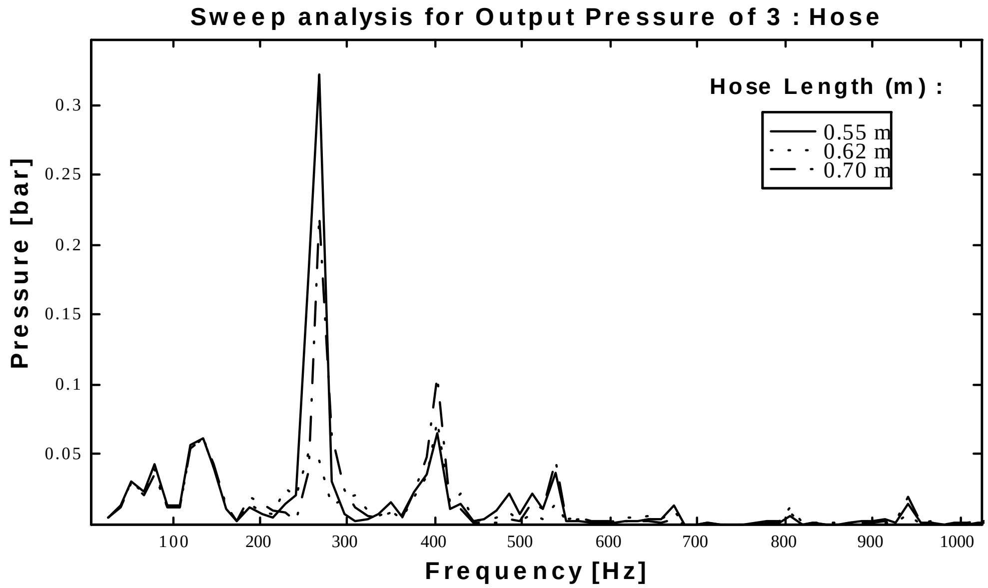

Figure 7 Frequency sweep simulation for varying hose length.

configuration for a particular vehicle. Considering various other components like a tuner, a restrictor, or a multi-chamber hose is the focus of ongoing research.

It should always be considered that the optimal hose length should attenuate all frequencies of pressure ripple that coincides with the harmonics of the mechanical structure. During the presentation, simulations will be conducted live, to show the powerfull capabilities that can be at hands in seconds for the engineers.

REFERENCES

[1] **Drew, J.E.** The use of flexible hose to reduce pressure ripple in Power Steering. *Ph.D. Thesis, University of Bath, 1997.*

[2] **Johnston, D.N., and Edge, K.A.,** Simulation of the pressure ripple characteristics of hydraulic circuits, *Proc. Inst. Mech. Eng., 1996, Vol 203, 275-282*

[3] **Johnston, D.N. and Drew, J.E.** Measurement of positive displacement pump flow ripple and impedance. *Proc. Instn Mechanical Engineers, 1996, Vol. 210, 65-74.*

[4] **Longmore, D.K.** The transmission and attenuation of fluid borne noise in hydraulic hose. *Inst. Mech.. Eng. Seminar on Quiet Oil Hydraulic Systems, 1977, C267/77, 127-138.*

[5] **Tuc, B.** The use of flexible hose for reducing pressure ripple in hydraulic systems. *Ph.D. Thesis, University of Bath, 1981.*

[6] **Johnston, D.N., and Edge, K.A.** The impedance characteristics of fluid power components: transfer matrix characteristics of two-port hydraulic components. *ASME Winter Annual Meeting, Chicago, Nov. 1994.*

[7] **Kojima, E. and Edge, K.A.** Experimental determination of hydraulic silencer matrices and assessment of the method for use as a standard test procedure. *7^{th} Bath International Fluid Power Workshop, Bath, Sept 1994.*