MODELING NOISE SOURCES AND PROPAGATION IN DISPLACEMENT MACHINES AND HYDRAULIC LINES

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ABSTRACT

The study of displacement machines, and in particular external gear pumps has improved the understanding of the important features of operation. Applying these underlying phenomena to new design methodologies has brought new advances in quieter machines by designing for the reduction of fluid-borne noise, cavitation, and pressure peaks. The present work seeks to expand on the previous modeling and experimental efforts by directly considering the effect that design changes to the pump and to the system apply on the total sound power and the sound quality emitted from displacement machines and the attached lines. In particular, the current document focuses on efforts related to the noise propagation into hydraulic lines and out of the lines to the environment where a pump is the primary excitation of this noise.

KEY WORDS

Fluid power, Noise, External gear machines

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>c</td>
<td>wave speed</td>
<td>[m/s]</td>
</tr>
<tr>
<td>p</td>
<td>pressure</td>
<td>[bar]</td>
</tr>
<tr>
<td>f</td>
<td>frequency</td>
<td>[Hz]</td>
</tr>
<tr>
<td>K</td>
<td>bulk modulus</td>
<td>[Pa]</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate</td>
<td>[kg/s]</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity</td>
<td>[cSt]</td>
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<tr>
<td>ρ</td>
<td>density</td>
<td>[kg/m³]</td>
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INTRODUCTION

Fluid power applications generally include heavy loads where noise is not a primary requirement and often include large machinery in construction and agriculture. With the increase of environment health and safety standards, the amount of noise generated by hydraulic components has increased in importance. Fluid power is also expanding into new industries with lighter applications such as passenger vehicles. In lighter and quieter machines, the importance of the level of noise transmitting from the hydraulic components to the...
passengers and environment becomes a primary design concern. In order to increase the range of applications where fluid power is advantageous, the noise generation must be better understood and ultimately reduced. Besides environmental concerns, decreasing the noise generation of hydraulic components has potential additional benefits of smoothing control and increasing machine life and reliability. The present work focuses on noise generation from displacement machines and the attached lines through simulation and experimental techniques.

The study of displacement machines, and in particular the case of external gear pumps, has improved the understanding of the important features of their operation [1]. Applying these underlying phenomena to new design methodologies has brought new advances in quieter machines by designing for the reduction of fluid-borne noise, cavitation, and pressure peaks [2]. Similar work has also been done by other authors.

The study of noise in displacement machines has many contributions from different authors over the past several decades [3][4]. The present work seeks to expand on the previous modeling efforts on the pump body alone [5] by directly considering the effect that design changes to the pump and to the system apply on the sound power and the sound quality emitted from displacement machines and the attached lines. Several research efforts have resulted in new designs that have reached the commercial sector [6][7].

The current work has the advantage of a strong base pump model which has been validated for a wide range of different units [1], and also multiple options for experimental validation and comparison available on test rigs at the Maha Fluid Power Research Center of Purdue University. The previous modeling efforts [5] predicted radiation only directly from the pump body. This work was completed for a typical external gear pump and is similar in scope to the work of Yamazaki and Kojima which focused on axial piston pumps [8]. The results from the acoustic model did not have a favorable magnitude comparison to experimental sound power measurements due mainly to the differences between the idealized model boundary conditions compared to the real experimental setup. The main differences include the presence of hydraulic lines, the attached electric motor, and the frame of the test rig. These structures contribute to both noise generation and noise propagation out of the sound power closed volume. An experimental study on the noise generated by several different models of gear pump was previously presented [9]. Importantly, the difference in sound power predicted between the pump-only model and the full experimental setup is expected due to exactly these issues. The pump is the main source of noise due to pressure and force oscillations, but the transmission of this noise into the environment is a much more complicated and coupled problem.

The present work seeks to solve this problem by exploring the impact of additional sources inside the sound power volume with a special focus placed on the line connecting the pump to a load. This line in particular has a relatively large amount of surface area under high pressure fluctuations (typical gear unit applications result in 1 to 10 bar pressure fluctuations in the outlet lines). Efforts by previous research have shown the importance of the lines and the noise transmission through similar structures [10][11]. Additionally, the lines provide less impedance to noise transmission out to the environment than the body of the pump due to the difference in material thickness and stiffness.

**MODELING**

Modeling efforts include a lumped parameter model HYGESim for simulating the features of pump operation and fluid phenomena in both the pump and the attached lines. A separate case with simplified models for the lines are also considered in order to better understand the system operation. Finally, numerical geometrical techniques for analyzing the lines and solid structures are also considered.

**Lumped Parameter Model**

The main focus for noise modeling is on the force and pressure oscillations caused by the main unit in pumping operation. The unit under consideration is a typical 38 cc/rev external gear pump with 14 teeth on each gear. Other noise sources such as cavitation and pressure peaks are included in the lumped model, but are not considered specifically in their individual contributions to the noise radiating from the pump. The main characteristics of the pump flow operation are determined the lumped parameter model HYGESim as described in previous publications and shown in Figure 1 [1].

![Figure 1 Layout of pump model](image_url)
Pressures and flows inside the pump casing and the interior surfaces of the lines are found through a combination of an advanced geometric model and customized submodels based in the AMESim environment. The outputs of this model are used as the main inputs for the acoustic models. The geometric model calculates the variable control volumes of the inlet, outlet, and tooth-space volumes throughout a pump revolution. The pressure in each control volume is found through the pressure build-up shown in Eq. 1 is given by Vacca and Guidetti [1].

\[
\frac{dp_j}{dt} = \frac{1}{V_j \rho_{\gamma_j}} \left[ \sum \dot{m}_{w_j} - \sum \dot{m}_{s_j} - \rho_{\gamma_j} \left( \frac{dv_j}{dt} - \frac{dV_{w_j}}{dt} \right) \right]
\]

It also calculates the equivalent orifice connections between tooth space volumes including the turbulent flow equation for the connections between the tooth space volumes and the ports in Eq. 2. For Equations 2 and 3, h, L, and b are the height, length, and width of the relevant gap.

\[
m_{s,j} = \frac{(p_j - p_{i,j})}{(p_j - p_{i,j}) + \rho_{\gamma_j} \left( R_{\gamma_j} \right)} \cdot \left( \frac{2(p_i - p)}{\rho(p_i)} \right)
\]

Also, the laminar flow equation for the flow around the tooth tips of adjacent tooth space volumes in Eq. 3.

\[
m_{s,j} = \rho \left( \frac{b}{12 \mu} \frac{h^3}{L} + \frac{u}{2} \right) b
\]

The combined model also allows for calculation of the radial motion of the gears, casing wear, and various other physical phenomena such as air release and absorption. An example output of the model is shown in Figure 2 for one revolution of the pump.

![Figure 2 Modeled pressure for the pump and outlet](image)

The meshing region for a reference tooth space volume occurs from approximately 330° through 30° where 0° is the point of minimum volume. The pressure rise at 110° in the tooth space volume corresponds to the connection to the high pressure at the outlet port through a narrow back-flow groove. The peak at 350° comes from the displacing action of the gears in the meshing zone while the tooth space volume is isolated from both ports.

**Additional Pump Model Features**

A coupled CFD model shown in green in Figure 1 allows for calculation of the lateral balance and leakage flows of the pump between the lateral faces of the gear teeth and the pressure-compensated bushing plates. This model runs in co-simulation with the lumped parameter model using the lumped parameter model pressure boundary conditions and returning the leakage flows through the gap [10].

The combined finite element and boundary element vibro-acoustic model for the pump is also included in Figure 1 in orange. This part of the model was used previously to determine the acoustic radiation from the pump body due to the internal excitation forces and pressures [5]. This model is currently being extended for greater accuracy in comparison to experimental data by including additional components such as the lines, which is one of the motivations for the present document.

**Line Model Study**

Modeling line contributions to noise is important since the lines provide a key path for noise to propagate from the sources out into the environment. The geometry and dynamics of the lines and the load can play an important role in how the forces and pressures are applied to the interior of the lines, and also the subsequent propagation through the walls of the lines and out to the environment. In order to understand the behavior of the system, it can be broken down into two main categories for discussion here, the harmonics of the fluid, and of the steel pipe.

First, the harmonic frequencies of the oil in the pipe. Similar to many wind instruments, the fluid inside a cylindrical shell of appropriate length to diameter ratios acts in a harmonic way according to the geometry of the cavity and the properties of the fluid. This dominates when the system does not have a strong forced excitation at certain frequencies. To understand the harmonic behavior of the media, the speed of sound must be found.

\[
c = \frac{K}{\sqrt{\rho}}
\]

K is the bulk modulus and \(\rho\) is the fluid density. For ISO 46 oil at 50°C and 250 bar, \(c=1474\text{m/s}\). The relevant free wave propagation speeds for relevant materials and fluids in question are shown in Table 1.
Table 1 Speed of wave propagation in relevant mediums

<table>
<thead>
<tr>
<th>Medium</th>
<th>Free wave propagation speed [m/s]</th>
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<tbody>
<tr>
<td>Oil</td>
<td>1474</td>
</tr>
<tr>
<td>Steel</td>
<td>6000</td>
</tr>
<tr>
<td>Braided rubber hose[10]</td>
<td>900-1150</td>
</tr>
<tr>
<td>Air</td>
<td>340</td>
</tr>
</tbody>
</table>

Basic equations for fluid motion inside a cylindrical shell can be derived from the geometry by making simplifying assumptions. An example simplified diagram of the fluid behavior in the steel pipe is shown in Figure 3 where the pump has been replaced by a single oscillating piston, and the load orifice is represented by a simple mechanical impedance.

The fluid harmonic behavior can be modelled as forward and backward travelling plane waves from left to right in the diagram and in Equation 5.

\[
p = Ae^{j(\omega t + kL - x)} + Be^{j(\omega t - kL - x)}
\]  \hspace{1cm} (5)

By assuming different end conditions, one can approximate harmonics of the fluid analytically. The pipe is 1m long with 25.4mm internal diameter. Rigid end cap at right:

\[
f = \frac{nc}{4(L + 0.4d)} \hspace{1cm} n=1,3,5…
\]  \hspace{1cm} (6)

Open pipe at right:

\[
f = \frac{nc}{2(L + 0.3d)} \hspace{1cm} n=1,2,3…
\]  \hspace{1cm} (7)

Harmonic frequencies of the fluid based on the geometry of the pipe and the fluid properties are shown in Table 2. Harmonics take the form of standing waves in the pipe at different frequencies if the excitation is not being forced to specific frequencies. The fluid harmonic behavior in the test setup pipe is between the two extreme values based on the mechanical impedance of the orifice plate. The harmonics actually couple to the excitation frequencies in a forced excitation similar to the oscillation of flow introduced by the pump, which results in the observed fluid behavior in simulation and practice.

The hydraulic line model used in the AMESim environment is based on a specialized distributed line with lumped elements. This model takes into account friction and flexure of the line walls, which allows for determining source terms for acoustic excitation and radiation, and also the distribution of pressure profiles down the length of a specific line. The line excitations can then be applied to a simplified acoustic model to estimate the acoustic response. Considering the fidelity of the line model is an important detail that greatly affects the simulation results. If too few nodes along the pipe length are considered, a poor approximation of the loading conditions is provided. If too many elements are added, the effective stiffness of the fluid exceeds the true behavior and results in amplification of higher order frequencies due to an increased number of potential fluid harmonics.

With only a single node in the line, the resulting predicted pressure is very smooth because higher frequency features cannot be captured. With more nodes, a tradeoff is reached between simulating higher frequencies, and the stiffness of the distributed hydraulic fluid reaching non-physical values. Higher frequencies are amplified due to the change in fluid harmonics because of added stiffness when using more nodes. The recommended number of nodes for this particular pipe model is 5 nodes across a 1 meter with 25mm internal diameter. For example, the pressure down the length of the pipe for a 20 node simulation is shown in Figure 4.
Choosing the correct number of nodes results in a more physically accurate model, but limits the fidelity of the pressure loading conditions for noise considerations. Ideally, more nodes could be used since this gives better loading functions.

Trends typical of standing waves can be seen in the time domain, but the frequencies are forced by the pump. Several periods for primary pump excitation frequency are shown in the figure. The pressure is mainly pulsing back and forth from the end at the pump to the end with the orifice, which can be seen in Figure 5 if the previous graph is expanded into the spatial domain and made into a surface.

![Figure 5 Simulated standing wave behavior in pipe](image)

Pressure is pulsing back and forth in the 1m pipe with a node at center point of pipe (0.5m). This behavior dominates performance, out of phase by 180 degrees at opposite ends as shown in Figure 5. The second bar.

![Figure 6 Frequency spectra magnitude of Figure 5](image)

If excitation frequencies align with harmonic frequencies of the fluid, large outputs can occur in the line pressures. Such as around 600 to 700 Hz, which corresponds well with the values calculated in Table 2. This result is also shown in Figure 7 in the highlighted bands around 700 and 1400 Hz in the frequency spectra, the pressure ripple outputs are interacting with the harmonics of the line, which results in a larger output. In Figure 7, the outlet pressure for a range of pump shaft speeds is shown, and the harmonic behavior of the fluid is visible in the heavier bands.

![Figure 7 Simulated pressure spectra under varied speeds](image)

There is coupling between the forcing frequencies of the pump based on speed and number of pumping chamber, and the natural harmonics of the fluid in the pipe. The interaction between the fluid harmonics and the shape of the pipe has a large impact on the fluid behavior inside the pipe volume. If the forcing frequencies of the pump change (due to change in speed or number of pumping chambers), or if the harmonics of the fluid volume change (due to adjusting the pipe length or diameter or load impedance), then the coupled results will also be different. This makes for a very complicated problem, but understanding the interactions is a good step towards improving the system noise performance.

**Geometric Considerations**

With a better understanding of the fluid harmonics, focus can now be placed on the interaction between the fluid and the structure of the steel pipe. The main structure exhibits sinusoidal geometric behavior at its own modal frequencies with two examples shown in Figure 8. The shape is affected by both the geometry and material
The methodology for a combined finite element and boundary element acoustic model for predicting noise radiation from an external gear pump was previously published [5]. That work focused on the coupling of the loading forces and pressures to the pump body, and then the transmission out to a simulated environment. The present work considers the effects of the lines on the total noise radiation. The study of just the pump body showed it to act as nearly a pure impedance to the loading oscillating pressures. Due to the stiffness and material of the structure, there is very little interaction between the two parts of the pump body (modal frequencies above 7 kHz), and the excitation frequencies (between 0 and 2500 Hz). As shown in Figure 9, the steel pipe is predicted to have more interacting harmonic behavior between the excitation frequencies and the structural modes due to the presence of structural harmonics in the lower frequency range.

**Measurement Setup**

The modeling and experimental efforts are focused on matching the interaction between the pump and the lines. A 1m steel pipe with 25.4mm internal diameter and 3mm thick steel walls as shown in Figure 10. The pipe is terminated by an orifice plate with a hole of 4.7mm diameter. This is the same setup as used in the model.

The steel pipe has the added benefits of three embedded piezo-electric pressure sensors for measuring the fluid-borne noise, and represents an easier component to model with respect to rubber hoses. Typical rubber hoses reinforced with braided steel wire are also common in application, but are not the focus of the current study. The elastic walls and consistent cylindrical geometry of the steel pipe are both easier to model and also more straightforward to match simulation to experiment.

The experimental layout was implemented at the Maha Fluid Power Research Center of Purdue University. The key details of the setup are shown in Figure 11. The steel pipe is shown connected to the pump.

**EXPERIMENTAL**

Where there are structure/fluid harmonic interactions, it should be apparent in experimental measurements of the fluid pressure fluctuations, and also in the air-borne noise. More details on similar previous experiments were presented [9]. The current work more closely examines the experimental and simulated data to find trends and deeper understanding of the key feature to be used when designing new quieter displacement machines and hydraulic systems.
The angled lines in Figure 12 and Figure 13 represent the frequencies present in the fluid-borne noise. That is, they are excited by the interaction of the flow ripples generated by the pump passing through the orifice at the end of the pipe, as well as the wave propagation occurring throughout the pipe. The vertical bands seen in the figure are present even with changing excitation frequencies and are most visible around 600 Hz, 1100 Hz, and 1400 Hz. These frequencies correspond well with the predicted harmonic frequencies of the fluid in the pipe as demonstrated in Table 2 and Figure 7.

The recorded air-borne noise is shown in Figure 13 for the same speed and frequency range. Both figure unit scales are logarithmic. The reference values for the two plots are 1 bar for Figure 12, and 1 Pa for Figure 13. The two results of the fluid-borne noise and the air-borne noise show that the main frequencies excited by the pump are transmitted almost directly from the oil and out into the environment. The vertical bands seen in both figures are present even with changing excitation frequencies, which means they are inherent to the system and structure, which includes the pump body, the lines, and other objects coupled to the same test bed.

A clear interaction between the fluid-borne noise and the harmonics of the fluid and the structure can be seen. Where the pump excitation frequencies align with the vertical bands in both figures, they tend to be amplified in the air-borne noise, with the darkest red sections occurring almost exclusively at these intersections. In particular, the air-borne noise shown in Figure 13 demonstrates that while the frequencies measured in the air correlate well with the measured fluid-borne noise, there is not perfect matching.

The presence of other frequency content in the air-borne noise is largely due to additional sources measured using a microphone which are not present in the hydraulic fluid; such as noise from downstream valves and from the electric motor operation.

CONCLUSIONS

This work was primarily motivated by the need for quieter fluid power systems and the needs identified by prior research and publication efforts. The preliminary results identify the importance of considering the entire system when modeling or measuring the noise generated by a displacement machine. The coupled harmonic behavior between the source excitation frequencies and the geometry of the fluid and system structures was presented. The predicted interactions are also shown to be present in real experimental data using a sweep through the operating speed range of a typical external gear machine.

The future work includes advancing the line modeling to fully predict the transmission of noise from source to receiver, and combining the line model with previous work focused only on the pump body. Better and quieter displacement machines and systems can be designed through better understanding the phenomena related to noise generation and transmission.
REFERENCES