A model based approach for the evaluation of noise emissions in external gear pumps

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This paper contributes to the topic of modelling noise generation and propagation in hydraulic pumps, particularly focusing on the external gear pumps. By using proper methodologies for the fluid, structure, and air domains, the model proposed in this study can predict the resultant noise emissions coming from the interactions between these three domains. Two cases of numerical simulations were performed, considering a different complexity for reproducing the pump mounting conditions. For validation purposes, noise measurements were taken in a semi-anechoic chamber on a commercial unit. The effects of the mounting situation on the overall emitted noise as well as the level of the agreements between simulation and experiments are discussed.

Keywords: External gear pumps, Vibro-acoustical modelling, Fluid-borne noise, Structure-borne noise, Air-borne noise.

1 Introduction

Noise emissions are one of the most known detrimental aspects of fluid power technology, and a limiting factor for many applications. To overcome this problem, fundamental research has been carried out during the last decades on the basic mechanisms of noise generation in fluid power systems. As outcome of this effort, nowadays several criteria for identifying the noise sources and for limiting the overall noise emissions in fluid power systems are available. It has now understood that the pump is a crucial element as concerns the generation of noise and vibration in a hydraulic system; moreover, it is now a common practice to consider port flow oscillations as main source of fluid borne noise. However, methods for simulating the transfer path from the fluid borne noise sources to the actual airborne noise, depending on the pump architecture, are still incomplete, if not entirely unavailable. Moreover, current literature often does not consider some important sources of noise, at both fluid-borne and structure-borne noise level.

This paper contributes to the field of modelling noise generation and propagation in hydraulic pumps. Taking as reference the case of external gear pumps, a simulation method to predict the actual airborne noise starting from the main noise sources is presented. A fluid dynamic model previously developed at the Maha Fluid Power Research Center for the simulation of the main flow through external gear pumps is used to evaluate the instantaneous pressures within the internal fluid domain of the unit. These pressures are used as internal load functions applied to the internal parts of the pumps. A finite element model (FEM) and a boundary element model (BEM) are properly combined, to perform the vibro-acoustic analysis of the pump. In this way, all fluid-borne and structure-borne noise sources are properly considered for the estimation of the actual air-borne noise.

The proposed model is validated within this research through comparisons between the simulated and the measured sound pressure levels (SPLs) and sound power levels (SWLs), considering a reference commercial pump produced by Casappa. For this purpose, a test campaign was performed by utilizing a semi-anechoic chamber available at the authors. For validation purposes, the simulations were performed in such a way that the acoustic environment, including the main features of the pump-flange assembly as well as the presence of reflecting planes, was properly considered.

The results presented in the paper illustrate the potential of the proposed approach to predict the actual noise emissions of an external gear pump. Relevance is given to the important effect of the pump mounting on the overall emitted noise, as well as the features of the anechoic room for the understanding of the features of noise directionality.

The modelling approach of this paper presents a significant step forward with respect to what already published by the authors in /1/. In particular, the results of this work highlight the new findings related to the pump mounting conditions. While in the cited work the pump was considered as stand-alone element, in the present work it is shown how the pump mounting has an important effect on the emitted noise, particularly as pertains to the noise directivity.

2 Modelling Overview

This section provides an overview of the approach used to model each fluid, structure, and air domain as well as their mutual interactions. Being this paper mostly focused on the new findings related to the pump mounting effects, this section will only provide a general overview for most of the modules of the model, for the sake of completeness. The readers interested in the more details about the different parts of the model are recommended to refer to /1/.

Noise generation in the pumps and motors is commonly classified into three categories in many studies of noise in hydraulic applications: fluid-borne noise (FBN), structure-borne noise (SBN), and air-borne noise (ABN). This classification can be useful in understanding the mechanisms involved in the generation and transmission of noise. Thus, the acoustic model in this study uses the different methodology for each domain and predicts the resultant noise coming from the interaction between the three domains.

Figure 1 shows the overview of the vibro-acoustic model. For the evaluation of the fluid-borne noise sources, the simulation tool developed by the authors’ research team over the last decade is used: HYdraulic GEar machine Simulator (HYGESim) /2/. This simulation tool can accurately predict the dynamic pressure in the inner pump tooth space volumes (TSVs) as well as the forces acting on the gears and the internal journal bearings. The predicted pressures within the internal fluid domain of the unit are properly classified into 4 types of the internal load functions applied to the correct areas of the pump casing in a realistic way. This will be further discussed in the next section. For the vibro-acoustic behaviors of the pump, a combined FEM/BEM approach is used. FEM mesh is generated directly from the CAD drawings of the pump. Then, a numerical modal analysis is performed...
with the proper consideration of the pump mounting condition to obtain the modal frequencies and mode shapes of the structure. The resulting modal information is used to implement the modal superposition technique, which enables an efficient calculation of the structural response. After that, the boundary element surface mesh is generated around the exterior surfaces of the structural finite element mesh, and the field point mesh is imported considering the acoustical features of the environment. Both the structural vibration and the emitted noise are finally predicted by solving the coupled FEM/BEM equations. More details are described in the following sections.

The reference pump considered for this study is a commercial gear pump produced by Casappa (model PLP20Q). The unit has a displacement of $22 \, \text{cm}^3/\text{rev}$, and it can operate up to 300 bar. It has 12 teeth working under dual flank conditions (zero backlash) $\frac{3}{5}$. The pump body is in aluminum and it is composed of three pieces (flange, main pump casing, and end cover). Pressure compensated lateral bushings are present to enhance the lateral sealing of the tooth space volumes.

3 Fluid-borne noise

The evaluation of the fluid-borne noise sources is performed through the use of the authors’ HYGESim tool [2]. HYGESim basically uses a lumped parameter approach, which divides the fluid domain inside the unit into the four main control volume groups as shown in Figure 2a: the inlet port volume ($V_{i}$), the outlet port volume ($V_{o}$), the set of control volumes for the TSVs in the drive gear ($V_{LP}$), and the set of volumes for the TSVs in the driven gear ($V_{HP}$). The fluid properties inside each control volume are assumed uniform and only time-dependent. Since the control volumes for the TSVs have the deformed shapes near the meshing zone, each control volume is defined according to the minimum distance based geometrical criteria, as described in [4]. The pressure inside each control volume is evaluated from the mass conservation law. For this purpose, the model evaluates the instantaneous internal flows connecting each control volume. These flows are given by the flow from the tooth space volumes and the inlet/outlet ports, and the flows between the tooth space volumes. These internal flows are evaluated with turbulent orifice flow equations and, for the case of the leakages, with laminar flow equations [2, 5].

From the evaluation of the pressure in each control volume, HYGESim can also estimate the radial forces acting on each gear and transmitted to the journal bearings, as detailed in [6]. In the journal bearing model, based on a 2D CFD approach, the Reynolds equation is solved in order to evaluate the pressure distribution in the gap. The model takes into account the deformation of the components as well as the dynamic effect caused by the time-varying load. The resulting pressure inside the TSV is normalized with respect to the delivery pressure (which is in the order of magnitude of 100 bar). From this result, important features of the FBN noise sources can be observed. As the gear rotates, the TSV first goes through a steep pressure gradient due to the pressurization at around $50^\circ$. From approximately $50^\circ$ to $230^\circ$, the TSV pressure almost coincides with the outlet pressure ripple. This is mainly due to an internal connection machined in the lateral bushings, which connects the tooth space volumes to the outlet environment along the edge of the lateral bushing. As the gear rotates, the tooth space volume faces a pressure peak at around $300^\circ$, which is caused by the reduction of the trapped volume during the meshing process. From $305^\circ$ through $320^\circ$, the pressure in the TSV can fall below atmospheric due to the increasing volume of the trapped TSV, causing a slight onset of localized aeration [5]. The contributions of all these noise sources are considered in a direct or indirect way by mapping the loads to the corresponding areas.

![Figure 2](image1.png)

Figure 2: (a) control volume definition and angular convention $\theta$ defines the TSV position with respect to the start of the internal casing) and (b) instantaneous volume of a tooth space chamber (TSV) (dashed line) and drive gear TSV dimensionless pressure (solid line), obtained from HYGESim. Figure 2b shows the parts of the HYGESim results in terms of the working volume and pressure inside a particular tooth space volume (TSV) calculated by HYGESim, according to the angular convention shown in Figure 2a. Note that the resulting pressure inside the TSV is normalized with respect to the delivery pressure (which is in the order of magnitude of 100 bar). From this result, important features of the FBN noise sources can be observed. As the gear rotates, the TSV first goes through a steep pressure gradient due to the pressurization at around $50^\circ$. From approximately $50^\circ$ to $230^\circ$, the TSV pressure almost coincides with the outlet pressure ripple. This is mainly due to an internal connection machined in the lateral bushings, which connects the tooth space volumes to the outlet environment along the edge of the lateral bushing. As the gear rotates, the tooth space volume faces a pressure peak at around $300^\circ$, which is caused by the reduction of the trapped volume during the meshing process. From $305^\circ$ through $320^\circ$, the pressure in the TSV can fall below atmospheric due to the increasing volume of the trapped TSV, causing a slight onset of localized aeration [5]. The contributions of all these noise sources are considered in a direct or indirect way by mapping the loads to the corresponding areas.

![Figure 3](image2.png)

Figure 3: Areas of application of noise sources (a) and dynamic pressure ripples applied to the subdivided regions (b) inlet, (c) outlet, (d) TSV pressure regions at $\theta = 51^\circ$, and (e) journal bearing regions)
that it is sufficient to split the case in a number of regions so that each TSV has an angular length of 4 segments. Figure 3b–3e show the simulated loads that are applied to the subdivided areas. Note that all the pressure functions in these figures are normalized, for confidentiality, with respect to the reference pressure \(p_{ref}\), which is the maximum pressure of interest in this study. The operating condition considered for these plots is 1500 rpm shaft speed and 0.4 \(p_{out}\) outlet pressure (Figure 3b: inlet, Figure 3c: outlet, Figure 3d: TSV pressure regions, and Figure 3e: journal bearing regions). The left side of these figures shows the normalized pressure fluctuation depending on time, and the right side of the figures shows Fourier transform of the normalized pressure fluctuation. These dynamic pressure loads serve as the inputs to predict the structural response and the radiated noise.

4 Structure-borne noise

To investigate the effects of the pump mounting on the both structural responses and emitted noise, two cases were compared as shown in Figure 4. Figure 4a pertains to the simplest case, that considers the pump only, as standalone element; further complexity is added in Figure 4b, by including the mounting plate to which the pump is connected during the actual experiments of Section 5. In the remaining part of the paper, the first case is referred as “standalone pump” case while the latter case is referred as “pump with structure” case.

For the assembly of the parts, the bonded contact is applied to the entire joint surfaces for FEM analysis in ANSYS. This is the fastest and simplest way to model the bolted connections and the whole assembled structure can be considered as a single part. Thus, the acoustic model can be further developed in future by considering the more complicated but accurate way to model the bolted connections such as using the beam elements or solid elements.

After creating the FEM mesh for both cases starting from the 3D CAD of the structures, the numerical modal analyses are performed. Figure 5 shows the normalized modal frequencies for the standalone pump (dark blue) and the pump with structure (yellow). Note that – for confidentiality – all the modal frequencies for both cases are normalized with respect to the first numerical modal frequency of the standalone pump. It can be seen that the inclusion of the attached structure significantly changes the modal frequencies; it lowers the first modal frequency almost by half and the more number of modes are observed in the same frequency range. Therefore, for the standalone pump, the total 16 modes are considered in the audible frequency range whereas the total 20 modes are concerned up to 10 kHz for the pump with structure case. Taking into account the fact that the first few modes can dominate the overall structural response and the frequency range of interest in this work is up to 5 kHz (which is the capability of the experimental setup, described in section 5), the number of modes and the frequency ranges considered in the modal analyses for both cases seem to be enough to provide the appropriate results.

After evaluating the modal frequencies, some observations can also be done as concern the mode shapes. The first four mode shapes for the standalone pump case are shown in Figure 6. The first and second modes are the bending modes in vertical and horizontal directions, the third mode is the torsional mode, and the fourth mode is the longitudinal mode. The corresponding bending, torsional, and longitudinal modes of the pump with structure are shown on the right side of each mode of the standalone pump in Figure 6. It can be observed that the inclusion of the attached structure brings to mode shapes with a motion of the mounting plate. Another observation one can make is that the attached structure changes not only the modal frequencies, as discussed previously, but also the order of the modes: the order of torsional mode (3rd mode) is ahead of the longitudinal mode (4th mode) for the standalone pump case while the order of longitudinal mode (7th mode) is ahead of the order of the torsional mode (7th mode).
The modal frequencies and mode shapes obtained from the modal analysis are used for implementing the modal superposition technique to get the forced response of the structure. Using the modal superposition technique, the structural response in terms of displacement at a given excitation frequency can be described using a linear combination of modal vectors (mode shapes) of the vibrating structure:

\[
[w_i] = \sum_{k=1}^{m} q_k \Phi_k = [\Phi] \cdot [q]
\]

where \([w_i]\) is the structural displacement at node \(i\), \(q_k\) is the modal participation factor of the \(k\)th mode, \(\Phi_k\) is the \(k\)th modal vector (mode shapes), \([q]\) is the vector of modal participation factors, and \([\Phi]\) is the matrix of modal vectors. In particular, the modal participation factors determine how strongly the corresponding modes contribute to the total structural response at the given frequency. The advantage of this method is in the orthogonality of the modal vectors with respect to the mass matrix, which makes it possible to reduce and uncouple the equations of motion. Thus, the computational efforts can be substantially reduced.

Figure 7 shows the structural responses (displacement) for both the standalone pump (left) and the pump with structures (right) at the frequency of (a) 2700 Hz and (b) 4500 Hz.

Figure 8 shows the structural responses (displacement) of the standalone pump (left) and pump with structures (right) at the frequency of (a) 2700 Hz and (b) 4500 Hz.

5 Airborne noise

5.1 Numerical acoustic analysis and pump noise measurement procedures

The coupled FEM/BEM approach is used to predict the airborne noise radiated by the pump in the numerical model. The acoustic environment implemented in the numerical model is shown in Figure 8a. Firstly, the boundary element surface mesh was generated around the exterior surfaces of the structural finite element mesh. This mesh is coarser and has more uniform distribution of the elements than the structural mesh. This feature of BEM surface mesh allows us to obtain the fairly accurate acoustic results in a reduced calculation time. The maximum size of the boundary element mesh was chosen to have 6 elements per the shortest wavelength of interest (i.e., at maximum frequency). Then, the field point mesh and two reflecting planes were generated in such a way as to mimic the semi-anechoic chamber, where the experimental noise measurements were conducted (Figure 8b) so that a fair comparison can be made. The field point mesh is a visualization mesh in the air domain, which can be regarded as the microphone positions: For the field point mesh geometry, a 1-meter radius spherical mesh from the center of the pump was first created and the mesh elements located behind the wall and below the floor were removed. In addition, two reflecting planes were imported at the same position of the wall and floor inside the sound chamber. These symmetry planes remove the boundaries and generate the mirror image of the sound sources, which is acoustically equivalent to the presence of the rigid boundary surface and enables the efficient calculations. Once all the acoustic environmental setup is done, the coupled FEM/BEM equation is solved to compute the normal displacement and the pressure on the boundary surface. (Note that when solving this equation, the modal superposition technique is involved so that the modal participation factors are calculated to determine the normal displacement of the structural surface as discussed in the section 4). The normal displacement and pressure results on the boundary surface are used in the post-processing step to determine the acoustic pressure, particle velocity, and the sound intensity at the field point mesh. By integrating the sound intensity over the area of the surface, overall sound power levels are finally obtained. The reference /8/ provides the details of the coupled FEM/BEM equations and post-processing step.

To validate the acoustic model, the experimental noise measurements were taken at the semi-anechoic chamber available at the Mahi Fluid Power Research Center of Purdue University as shown in Figure 8b. The pump was tested in the open hydraulic circuit including the needle valve, pressure relief valve, heat exchanger, filter, and...
reservoir. The experimental set up is the same to the one used in /1/, the reader can refer to this reference for further details. The pump under testing and the short hydraulic connecting lines were inside of the sound chamber, while all other elements including the electric motor (prime mover), the reservoir, the loading valve and the water cooler were placed behind the reflecting wall to minimize the noise radiated by other components to the inside of the chamber. During the operation of the pump, the inlet temperature was maintained at 50 °C.

The sound power levels (SWLs) of the tested pump were measured based on ISO 9614-1 /9/. The sound power represents the total sound energy radiated by the noise source per unit time and can be regarded as the intrinsic characteristics of the noise source because the sound power is independent of the surrounding environment and distances between noise sources and measurement points (i.e. microphone positions) while the sound pressure level (SPL) is highly dependent on them. Therefore, it is more preferable to use sound power level rather than sound pressure level to make comparisons or assessments concerning the model accuracy although the sound power measurements require a more demanding task than measuring the sound pressure at one point. In this work, for the determination of the sound power levels, the sound intensities were measured at the evenly distributed 107 discrete grid points at the distance of 1 m from the pump, and the shape of measurement surface is the same with the shape of field point mesh shown in Figure 8a. A robot was used to move the intensity probe (GRAS, three microphones Type 40A0—Sensitivity 0.2 dB ref 20 µPa, ½" diameter) to the selected measurement points.

5.2 Numerical and experimental acoustic results

Both numerical and experimental acoustic analyses were performed in the frequency of up to 5 kHz, which was the maximum limit of the frequency range of the sound intensity probe used in the measurement. The sound power levels and sound pressure level distributions at four operating conditions were compared between the measurement and numerical results. The operating conditions were properly chosen to see the effect of the change in both the delivery pressure and the shaft speed.

The overall sound power levels between numerical and experimental results are compared as shown in Figure 9 and Table 1. Note that the SWLs were normalized with the reference of the sound power of the experimentally measured mean noise floor inside the semi-anechoic chamber. Therefore, the normalized SWLs indicate how much levels were increased from the ambient noise level with the absence of the noise sources. Similar trends can be observed in both experimental and numerical results; in particular, the noise level tends to be increased as the shaft speed and delivery pressure are increased. However, this trend is not always true in the measurement when comparing the noise levels at 1500 rpm, 200 bar and 2000 rpm, 200 bar. Furthermore, the numerical results of both standalone pump case and pump with structure case are close to the numerical results as the differences from the measurements can be seen in the parenthesis in Table 1. Although the agreements with the measurements become better or worse depending on the operating conditions, the discrepancies are smaller, and reduce from [-1.8 ~ 3.4 dB] to [-2.1 ~ 2.0 dB] after including the attached structure in the model. According to this result, it can be concluded that the slight improvement was achieved by the inclusion of the attached structures in the numerical model in terms of the sound power levels.

More significant improvements after including the attached structure can be derived by observing the noise radiation patterns. Figure 10 shows the normalized SPL distribution at the distance of 1 m from the pump. The reference of normalization is the global minimum SPL of the entire results. There are two features that both numerical and experimental results have in common: 1) noisy areas are present near the inlet and outlet sides. 2) an increase of the delivery pressure does not greatly change the overall shapes of the SPL distributions, but it results in an increment of the overall noise level. However, the numerical results for the standalone pump present a great contrast to the experimental results in a regard that the noise radiations in the axial direction are hardly observed. On the other hand, after including the attached structure, the noisy areas in the axial direction appear in the numerical results, and the pattern becomes similar to the experimental results. This clear difference of noise radiation in the axial direction can be explained by the structural response as described in the section 4. For both cases for numerical analyses, no longitudinal motions were observed in the structural responses at the frequency range of interest as far as the motions of the pump itself were concerned. However, contrary to the standalone pump case, the axial motions of the attached structures were always observed, resulting in the noise emissions in the axial direction. Furthermore, after including the attached structure, the numerical results capture the change in the noise radiation patterns with respect to the shaft speed in the experimental results. For example, the red noisy areas exist in the upper side of the sphere at 1500 rpm, while they disappear at 2000 rpm in both numerical and experimental results. In conclusion, including the attached structures to the pump has a great impact on the noise radiation patterns in the numerical models and results in the better agreements with the experimental results. These results also imply that noise radiation patterns for a given pump can significantly change depending on the actual mounting conditions of the pump.

<table>
<thead>
<tr>
<th>Operating conditions</th>
<th>Measurement</th>
<th>Standalone pump</th>
<th>Pump with structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500 rpm, 100 bar</td>
<td>40.5 dB</td>
<td>40.0 dB (+0.5 dB)</td>
<td>40.2 dB (+0.3 dB)</td>
</tr>
<tr>
<td>1500 rpm, 200 bar</td>
<td>46.9 dB</td>
<td>45.1 dB (+1.8 dB)</td>
<td>44.8 dB (+2.1 dB)</td>
</tr>
<tr>
<td>2000 rpm, 100 bar</td>
<td>42.5 dB</td>
<td>42.6 dB (+0.1 dB)</td>
<td>41.9 dB (+0.6 dB)</td>
</tr>
<tr>
<td>2000 rpm, 200 bar</td>
<td>44.7 dB</td>
<td>48.1 dB (+3.4 dB)</td>
<td>46.7 dB (+2.0 dB)</td>
</tr>
</tbody>
</table>

Table 1: Experimental and numerical overall (normalized) sound power levels. The differences from the measurements are shown in the parentheses.
Conclusions

This paper presented a numerical methodology to evaluate the noise emitted by external gear pumps in conjunction with an experimental validation. The acoustic model in this study is based on different methodologies for modeling the fluid, structure, and air domains and it predicts the resultant noise coming from the interaction between these three domains. The evaluation of fluid-borne noise sources benefitted from the tool HYGESim developed by the authors’ research team. The internal noise sources were classified into the four types of effective load functions and applied to the correct areas of the pump structures in a realistic way. Vibro-acoustic behaviors of the unit were predicted using the combined FEM/BEM approach. Meanwhile, the modal analyses were performed to implement the modal superposition technique for the efficient calculation of the coupled FEM/BEM equations.

Taking a 22 cm³/rev commercial pump as a reference unit, two modelling assumptions were compared to evaluate the effect of the pump mounting on the overall noise prediction: 1) the standalone pump and 2) the pump with its mounting flange. It was found that the inclusions of the attached structures have significant impacts on the both modal frequencies and mode shapes.

Experimental noise measurements were taken in a semi-anechoic chamber at the authors’ research center for the purpose of the model validation. The numerical results for both cases were compared with the experimentally measured SWL and SPL distributions. While the both numerical results showed a good agreement with the experimental results, a slight improvement was achieved by the inclusion of the attached structures in the numerical model in terms of the SWLs. The maximum error between the predicted and measured SWL is of 2.1 dB.

The inclusion of the attached structure brings to clear improvements with respect to the modelling of the sound directivity patterns. In fact, the predicted SPL distributions are quite similar to the measured ones, but only for the case in which the attached structure is considered. These results showed the importance of including the mounting situation in the numerical model to achieve a more realistic noise prediction.

In the future, the acoustic model can be further developed by including further complexities such as the inclusions of the hydraulic lines, the presence of the oil, and more detailed consideration for the bolts that implement the connections between the structures. Furthermore, it is expected that the proposed acoustic model will be able to serve future numerical studies aimed at reducing the noise emissions of external gear pumps.

References