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Acoustic source characterization of miniature pumps

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Designing a silencer system to a duct system the up- and down-stream boundary conditions needs to be known to find the final system performance. The outlet is often a straightforward open termination of the duct, while the inlet could prove more difficult. Often some sort of rotating machinery is attached to it; it can be an engine, fan or pump. Techniques for characterizing this type of machinery have been developed over the last decades and have been implemented successfully for internal combustion engines and compressors. The most common technique today is the multi-load variant where the machine under study is operated under stable conditions while a number of different, linearly independent, loads are attached to it. In-duct measurements of the system response then yields the machine´s acoustic source impedance and acoustic source pressure amplitude. In this work these techniques are applied to miniature pressure or vacuum pumps, common in medical, automotive, domestic appliances and other industries. As the name implies they are small and the ducts attached to them are also of small diameter. Special care has to be taken to losses in the narrow pipes as well as choice of sensors. Relevant data can be taken at the pump orders and are presented together with a discussion of the character of the chosen pump type.

1 Introduction

When designing a product with respect to sound it is vital to not only study the product itself but also its interaction with the surroundings. Here we are discussing duct systems with an acoustic source in them; examples are intake and exhaust systems of engines, ventilations systems, equipment including pumps and compressors and many more. The task is most often to reduce the noise of the machine delivering work to the system. Designing a silencer the idealized measure of transmission loss is often used. It however assumes anechoic terminations and does not tell us how the silencer would behave when installed, where reflections from the up and downstream ends greatly influences the result. Often the termination of the duct system is well known, e.g. an open end, and can be modelled. But the impedance at the interface to the machine is less well known and must be determined for each case. Over the last decades techniques for characterizing the acoustic source has been developed [1] and mainly applied for automotive applications [2-3] but there are examples of pumps as well [4]. The motivation for this work is to use this technique and apply it to pumps of a new type, and much smaller dimensions; known as miniature pumps. They are used in a variety of applications and comes in many different designs. Often there is strong focus on cost in the applications where they are used leaving little room for making them quiet. This work should be seen as a first test whether the technique is applicable to this type of products and giving an indication of typical acoustic source characteristics of them.

The actual pump under study is used to move air or liquid from the inlet to the outlet via two mechanically actuated membranes that moves in opposite phase. Flap valves ensures the media only moves in one direction. The operating point is simply controlled by the input voltage.

2 Theory

2.1 Linear, time-invariant 1-port source model

Assuming the source does not change (significantly) with different loads attached to it (as is the case for example for two-stroke engines) or with amplitude a linear time-invariant model can be used at steady state operating points of the machine. The acoustic source can then be described in the frequency domain by the impedance and strength of an equivalent acoustic one-port source model at a reference cross section at the inlet or outlet of the machine. A graphical representation of the equivalent source model is presented in Figure 1

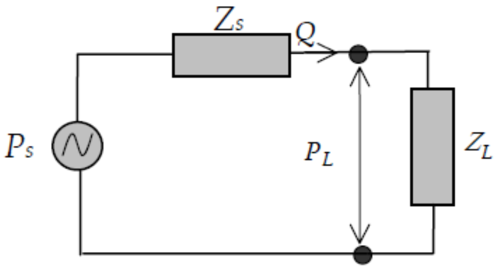

Figure 1. Electro-acoustic analogy of the 1D source model.

The source data is completely determined by the source strength (source pressure or velocity) P_S and the source impedance Z_S at the reference cross section. Often the normalized (or specific) source impedance $\varphi_S = Z_S/Z_0$ is used. Where Z_0 is the characteristic impedance of the medium. The acoustic load seen from the source is characterized by the impedance at the reference cross section: $Z_L = Z_0\varphi_L = P_L/Q$, where Q is the acoustic volume velocity. Now using the electro-acoustic analogy depicted in Figure 1 we get:

$$P_S - Z_S Q - P_L = 0, \tag{1}$$

That can be rewritten as:

$$P_S\varphi_L - P_L\varphi_S = P_L\varphi_L, \tag{2}$$

Now we need to find a method to be able to solve for the unknown source pressure and impedance.

2.2 Multiple load method

Most machines of practical use are rather strong sources making the use of direct methods difficult. Direct method refers to the case where a secondary source (a loudspeaker typically) is attached to the system. One can now achieve two linearly independent measurement to solve for the two unknowns in Eq. (2). However, it requires that the secondary source is “louder” than the source under study and also other considerable practical problems. A better choice for these types of sources are an indirect method; where instead the load on the system is changed [5].

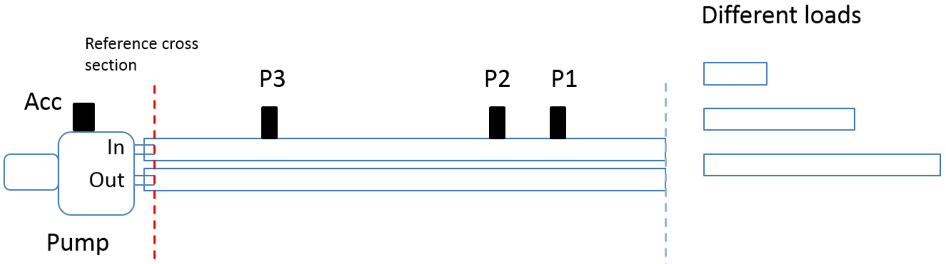

Figure 2. Schematic multiple load method applied to the pump. The procedure will be used on both the inlet and the outlet (here the inlet variant is shown).

The two unknowns P_S and φ_S are determined by solving an (overdetermined) system of equations described by the linear one-port source model presented in Section 2.1:

$$\begin{bmatrix} \varphi_{L1} & -\hat{p}_{L1} \\ \varphi_{L2} & -\hat{p}_{L2} \\ \vdots & \vdots \\ \varphi_{Ln} & -\hat{p}_{Ln} \end{bmatrix} \begin{pmatrix} \hat{p}_S \\ \varphi_S \end{pmatrix} = \begin{bmatrix} \varphi_{L1}\hat{p}_{L1} \\ \varphi_{L2}\hat{p}_{L2} \\ \vdots \\ \varphi_{Ln}\hat{p}_{Ln} \end{bmatrix} \tag{3}$$

Where subscripts $L_1, L_2 \cdots L_n$ are the different acoustics loads. Ideally one only need two cases, but with a relatively small effort, problems with measurement uncertainties are reduced significantly when using more load cases. Experience say that one need four to seven loads to get a good data set. It also yields an idea of the linearity of the system. Representing Eq. 3 as $Ax = B$ a linearity coefficient can be defined as:

$$\gamma^2 = x^{-1}x = B^{-1}AA^{-1}B, \tag{4}$$

Restricted to the range [0 1]. γ^2 is one if the source is linear, time-invariant and free of noise. In practice, we need to determine the load pressures and impedances. This is done via pressure sensors in the duct applying the "Two-microphone"-method [6-7] to decompose the pressure field in forward and backwards traveling waves. One could then determine a reflection coefficient at the reference cross section as:

$$R(f) = \frac{e^{\left(\frac{iks}{1\pm M}\right)} - H(f)}{H(f) - e^{\left(\frac{-iks}{1\pm M}\right)}}, \tag{5}$$

where *H* is the frequency response function between two microphones. The sign of the Mach number term, compensating for convective effects, will depend on the test case flow direction. Having the reflection coefficient, the load pressure and impedance can be determined as:

$$\begin{cases} \varphi_L = \frac{1}{Z_0}\frac{1+R}{1-R} \\ \hat{p}_L = \frac{\hat{p}_{ref}(1+R)}{e^{-ik_{\pm}L} + Re^{ik_{\pm}L}} \end{cases}, \tag{6}$$

$\hat{p}_{ref}$ is the complex pressure at the reference microphone for the frequency response function. L is the length from the reference microphone to the cross section where we want the source data to be defined.

The acoustic energy of most machines is concentrated in multiples of some rotational order, e.g. the firing frequency of a combustion engine. Often it is sufficient, or rather necessary, to restrict the analysis to these orders.

2.3 Sound propagation in narrow pipes

When using the multiple load method, it is essential to know the propagation wave numbers. They will be influenced by a number of factors, see a thorough investigation in [8]; the major ones are the thermo-viscous effects at the walls and convective effects of the mean flow. The thermos-viscous losses are due to diffusion in the acoustic boundary layer. Hence, the relative thickness of this boundary layer compared to the duct radius is a good indicator of the relative importance of this phenomenon and what model to apply. Tijdeman [9] proposed a "modified shear wave number" $Sh = r/\sqrt{\nu/\omega}$ as indicator; it is basically the ratio of the radius r to the acoustic wave number apart from a factor $\sqrt{2}$. If $Sh \gg 1$ we are in the "wide" duct regime. In the measurement tube we will use here $Sh \sim 10$ which is just where the narrow and wide tube solutions converge normally [10]. Hence, we will test both one narrow and one wide duct approximation.

Both solutions originate from Kirchhoff [11], where modification coefficients to the "loss-less" wave number are proposed. In the narrow duct case it is done by modifying the speed of sound as

$$c_m = c_0\sqrt{\frac{1-F(s)}{1+(\gamma-1)F(\epsilon s)}} \tag{7}$$

Where

$$F(s) = \frac{2}{s\sqrt{-i}}\frac{J_1(s\sqrt{-i})}{J_0(s\sqrt{-i})} \tag{8}$$

$$s = r\sqrt{\frac{\rho\omega}{v}} \tag{9}$$

with J being first order Bessel function and ϵ^2 the Prantl number. The modified wave number is then given by:

$$k_{m,n} = \omega/c_m \tag{10}$$

For wide ducts the thermos-viscous modification *K* to the wave number is:

$$K_0 = 1 + \frac{1-i}{\sqrt{2}s}\left(1 + \left(\frac{\gamma-1}{\epsilon}\right)\right) \tag{11}$$

Additionally the convective effects of a mean flow are included according to Documaci [12] as:

$$K^{\pm} = \frac{K_0}{1 \pm K_0 M} \tag{12}$$

Where *M* is the Mach number. Giving a modified wave number $k_{m,w} = k_0 K$.

3 Experimental setup

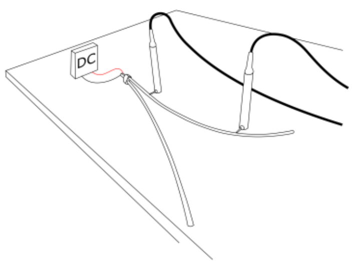

Figure 3. Schematic of the experimental setup.

The experimental setup is seen in Figure 3. The pump itself is freely suspended (not shown) and the motor of the pump is driven by a controllable DC-unit. To the air inlet and outlet of the pump a tube of 3.3 mm inner diameter is attached. To be able to measure the in-duct pressures in these narrow tubes probe microphones (Gras Type 40SA) are used; the probe is 1 mm in diameter and is simply inserted through the tube wall. The data is acquired using a NI frontend (DAQ9172) controlled via bespoke Matlab-scripts. For the results shown in this paper the microphone separation distance was 200 mm yielding a valid frequency range 68-686 Hz [13]. It is sufficient to capture the first five order for the engine operating points we ran. For each operating point four different loads are used. They are achieved by simply increasing the length of the tube (adding pieces) with: [25, 75, 105, 170] mm. The lengths are chosen to produce results that are linearly independent.

As the noise produced by this pump is very periodic with two main pressure pulses per revolution of the electric motor an order analysis will show a strong order 2, and also order 1 and harmonics, but not much in between, meaning that all the energy lies within the orders. This enabled us to perform the averaging in the crank angle domain rather than in the frequency domain, using an accelerometer, see Figure 2, as an index signal for each turn of the electric motor. The FFT on the averaged signal in the crank angle domain is used, giving us an order spectrum with a resolution of one order. The benefit of this approach is that even though the rotational speed of the electric motor varies slightly, one quickly get a stable average in the crank angle domain. In the frequency domain this would have led to frequency peaks that shifted with the rotational speed, making the average of the FFT blocks more uncertain and smeared out. An order spectrum would yield the same result as the crank angle approach.

4 Results

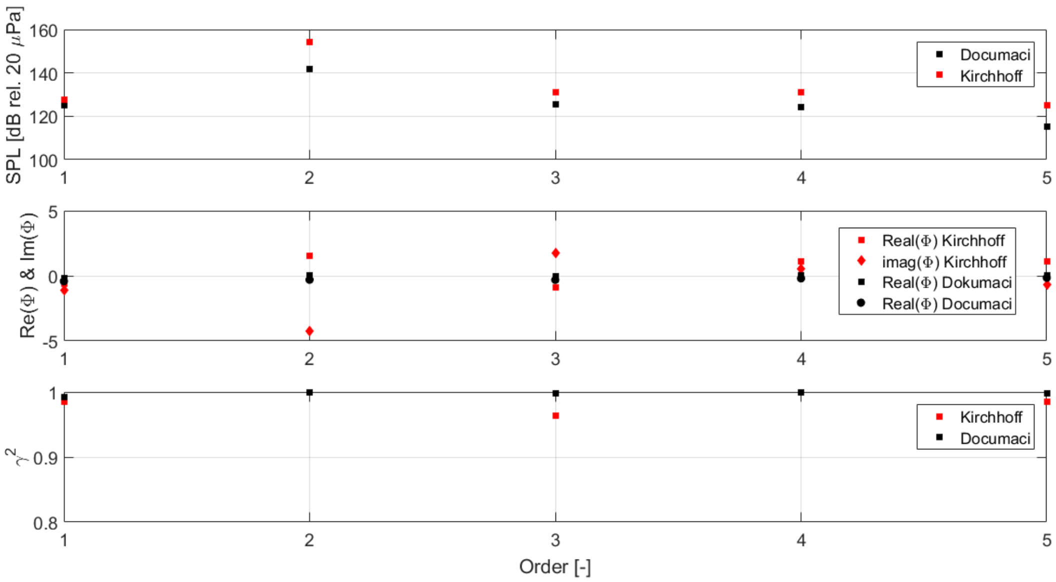

Figure 4. Source pressure, impedance and linearity coefficient for different rotational orders when running the fan at 124 rps and comparing the two models for sound propagation in the measurement tubes. (outlet side)

First we compare the two models for wave propagation to get an understanding of the problem. The actual flow velocity in the measurement section was low (<1 m/s), so the Mach number term in the Documaci model was neglected. In Figure 4 results for measurements taken at the outlet running the engine at 124 rps are postprocessed using both models. The linearity coefficient is close to unity for both version and do not yield any further information about what model to choose. The amplitude of the source pressure varies between the two cases, however not having previous experience on this type of machines both results are equally plausible. What we can learn though is that regardless of the choice of the model the second order is clearly dominating the order spectrum.

The source impedance can be represented as $Z = r + ix$, where r is a resistance and x a reactance. The resistance should be positive. If not, the termination is not passive, that is, sound is generated. This is not to be confused with the actual source we have in the machine, here we only discuss the reflection and transmission of already existing sound. This condition is fulfilled, at least at the main orders using any of the models. But finally looking at the actual amplitude of the impedance the results are more in line with previous experience in the Kirchhoff case than in the Documaci one. Remember that the results presented are normalised with the characteristic impedance, they are expected to be larger than unity, not much smaller as in the Documaci model case. We will therefore continue with the Kirchhoff model.

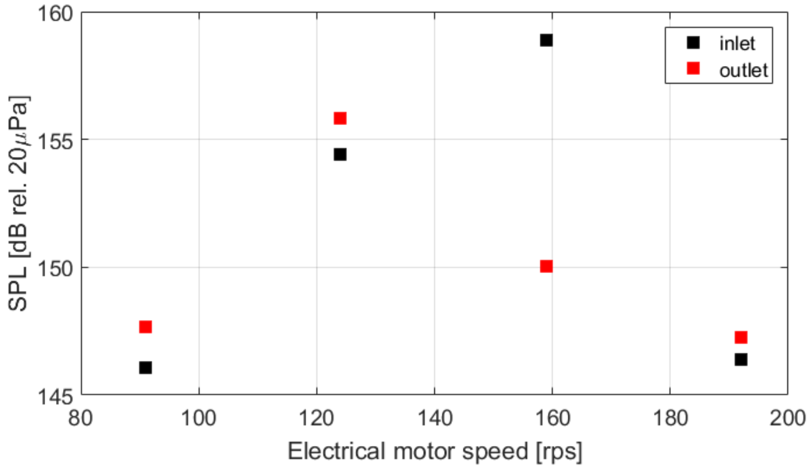

Figure 5. Source pressure for order 2 as a function of engine speed.

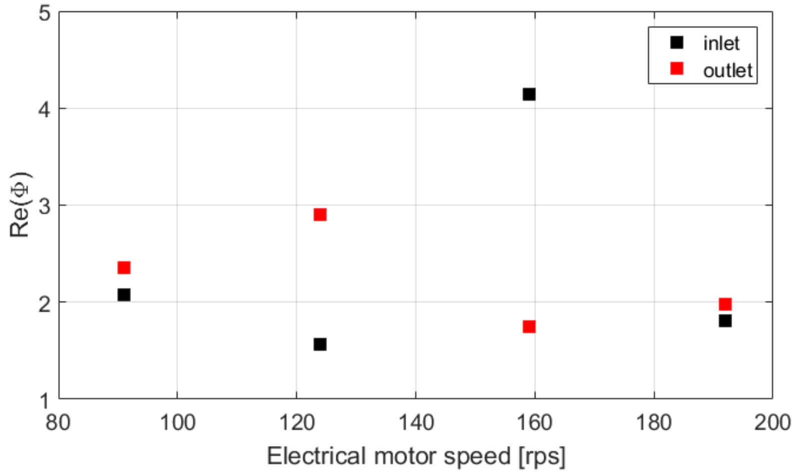

Figure 6. Real part of the source impedance for order 2 as a function of the engine speed.

The operating point is shifted by varying the input voltage to the electrical motor. Although not specifically monitored one can assume that the input power is increasing as well. In Figure 5 and Figure 6 the inlet and outlet are compared while varying the engine speed in this way. The engine speeds chosen corresponds to ~50-100% of the specified max operating point of the motor. The source strength or impedance do not increase linearly with increasing input power. They instead first increase and then drops of at higher engine speeds. It seems the pump is less noisy at the max operating point than at part load, unless reduced by half. This will obviously be important when designing a system including the pump – the optimum installation differ with the chosen operating point.

The inlet and outlet behave similarly at high and low speeds but differ in between. The source amplitude is higher (~2dB) for all speeds but one for the outlet, but then the inlet is 10 dB higher! This is also the loudest operating point seen. At the same speed the resistance at the inlet is also very high. That is, the same silencer does not necessarily work for both the inlet and outlet. The choice of operating point will make one or the other the main noise source.

5 Summary

As discussed in the beginning this study should be seen as a first investigation into the applicability of the multi load source characterization technique for small—miniature—pumps. As a bonus, it also yields an idea of the acoustic character of such devices. The study is by no means comprehensive but shows that:

- The setup using simple flexible tubes and probe microphones attached to the pump is sufficient.
- Stable results can be obtained in the order or "crank angle" domain. Requires a good reference signal, here obtained via an accelerometer on the electrical motor.
- The result indicate that the narrow pipe approximation is a better choice for the wave propagation model in the tubes. Further work is required to though to be able to make a more general judgement.
- The source strength and impedance vary significantly with operating point (and not linearly with engine speed). This is important when designing a system including the device.
- The acoustic characteristic of the inlet and outlet are not symmetric. Depending on the operating point one or the other could be the loudest.

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