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SCALING LAWS OF HYDRODYNAMIC NOISE GENERATION FOR SIMPLE FLUID VALVE MODEL

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ABSTRACT
Noise-source mechanisms for the simple fluid valve model were observed and compared with predicted mechanisms. Hydroacoustic scaling laws were derived by dimensional analysis of the governing equations. A comparison of the predicted mechanisms with the experimental data indicated that the scaling laws could be further improved by a re-examination of the influence of the impact parameters. This was done by an extended analysis of the influence of geometrical parameters on acoustical characteristics. Additional parameters were determined statistically using non-linear multiple regression analysis of the experimental data. The estimated correlation between the observed and predicted values for the applied geometrical and hydrodynamic parameters are good.

1 - INTRODUCTION
The governing equations describing the hydroacoustical processes in fluid throttling devices were obtained by [1], continuing the previous work [2]. A dimensional investigation of the principal equations, as is shown in [3] and [4], was used to analyze the turbulent excitation of a fluid mass. The influence of the turbulence structure is defined by a proportional factor in the form of a characteristic length, as suggested in [5]. Different operating conditions and the influence of geometrical parameters on noise are described by variations in characteristic length. A comprehensive dimensional analysis with derived scaling laws is given in [6]. In this dimensional analysis only a monopole sound generating mechanism is considered.
The acoustic generation mechanism refers to the propulsion of mechanical energy from the flow. Therefore, a dimensional investigation of the principal equations which were introduced in [1] may help to explain the impact of different parameters on noise formation. Equation (1) is derived in accordance with this analysis; the turbulence excitation of the fluid mass leads to the resulting fluid-borne sound power P. P is proportional to a scale of characteristic flow velocity v and turbulence scale l at the narrowest contraction area such that:

$$P \sim \rho l^2 \frac{v^4}{c} \tag{1}$$

where ρ is fluid density and c the speed of sound. The scaling laws can be derived using dimensional analysis which takes into account the connection between geometrical parameters, the characteristic flow velocity and the turbulence scale.
Various experimental data have partly proved and partly rejected the proposed scaling laws. Experimental results [7] show that two geometrical parameters have an influence the sound producing mechanism. Following on from this idea, the appropriate functional form of the governing equations can be determined statistically using multiple non-linear regression analysis.

2 - EXPERIMENTAL PROCEDURE AND RESULTS
A clear acoustic model is needed in order to understand the relevant processes taking place in the valve. The valve model presented a flow control element with three important geometrical parameters: valve stem length a, inside tubing diameter d and valve opening s. Water was used as the fluid medium and the

valve model operated under non-whistling and non-cavitating conditions. A Bruel & Kjaer type 8103 miniature hydrophone was used to measure fluid-borne sound pressure. The output signals from the hydrophone were routed through a signal-conditioning preamplifier (Bruel & Kjaer type 2635), further through an FFT analyser (Bruel & Kjaer type 2032), and then converted for advanced data processing using appropriate PC software [8].
The experimental results were examined statistically using non-linear multiple regression to determine the joint effect of a and d, together with flow velocity v, on noise generation. For this purpose the sound pressure $\tilde{p}$ is defined in accordance with scaling laws as follows:

$$\tilde{p} \sim v^{X_1} \cdot a^{X_2} \cdot d^{X_3} \tag{2}$$

In equation (2), the exponents X_i represent the impact of the observed parameter on noise generation. Since the dimensional analysis is based on the characteristic flow velocity, the measurement results are normalised with respect to the flow velocity in the narrowest cross-section. As was pointed out in a previous parametrical study, the sound pressure can be defined using the following expanded equation in order to include the combined influence of different parameters:

$$\tilde{p} \sim v^{X_1} \cdot a^{X_2} \cdot d^{X_3} \cdot v^{(a/d) \cdot X_4} \tag{3}$$

where the last term includes the weighted influence of the combined geometrical parameter. The last term in equation (3) incorporates a new impact factor connected with valve stem geometry, namely, how the ratio between valve stem length and valve stem internal diameter influences the noise generating mechanism. This factor defines how the turbulence-generated noise source is influenced by valve geometry and causes the transition between the monopole and dipole type of noise source. As the valve stem is kept the same or lengthened and the valve's diameter is narrowed, the turbulence eddies becomes affected by a solid wall. This transition of the driving mechanism from the fluctuating masses to the fluctuating surface pressure is defined as a valve geometry-dependent parameter, with a weighted impact on the characteristic fluid velocity.
A second set of experiments were done with a nearly closed valve. Contrary to the previous measurements, where the turbulence scale varied between $l \propto a$ and $l \propto d$, here it varies between $l \propto s$ and $l \propto d$. The two parameters used were the valve opening and valve stem inside diameter. For analysis, the proposed sound pressure was defined as:

$$\tilde{p} \sim v^{X_1} \cdot d^{X_2} \cdot s^{X_3} \cdot \left(\frac{d}{s}\right)^{X_4} \tag{4}$$

where the parameter (d/s) was used for transferring the characteristic turbulence length between $l \propto d$ and $l \propto s$. Fig. 1 shows the results of the multiple non-linear regression on the experimental data. Eq. (4) gives the functional dependency of the sound pressure level:

$$SPL = 104,2 + 31,5\ \log(v) + 63,7\ \log(d) + 22,3\ \log(s) + 23,4\ \log\left(\frac{d}{s}\right) \tag{5}$$

and explains 97% of the observed variance. In Eq. (4) and (5), the influence of the valve stem length is omitted since the results showed that it has no pronounced influence either on the characteristic length or on the noise-generating mechanism. This was clearly evident when both sets of measurements were done with a wide open valve ($s>d$) and an almost closed valve ($s<d$). Under these two limiting geometrical conditions, the most clear insight into noise generation is achieved.
Equation (5) shows the governing noise mechanism, defined as the weighted factor for the fluid velocity's impact on the sound pressure level. Here, when the pulsating mass is limited to a narrow constriction at the valve seat, the monopole-type noise source prevails. The valve stem diameter had an influence on the characteristic turbulence scale similar to that seen in previous measurements. The diameter acts in a connection with the valve opening, thus defining the characteristic turbulence length. The last term in Eq. (5) proves that the characteristic turbulence length varies, as expected, between $l \propto s$ and $l \propto d$. According to the flow acoustics theory, one expects a very efficient mechanism to be responsible for sound production. There are two main causes of turbulence in this valve model: first, the narrowed entrance to the valve stem, and, second, the stem outlet, i.e. the valve seat. Where the pulsating mass in the valve stem represents the main noise source, changes in the diameter consequently change the effective pulsating mass. The main cause of the noise-generating mechanism is the mass pulsation in the narrowest cross-section, i.e. in valve seat. These pulsations are not affected by changes in valve stem length a. Valve stem length has an influence on SPL, since the eddy formation takes place outside the

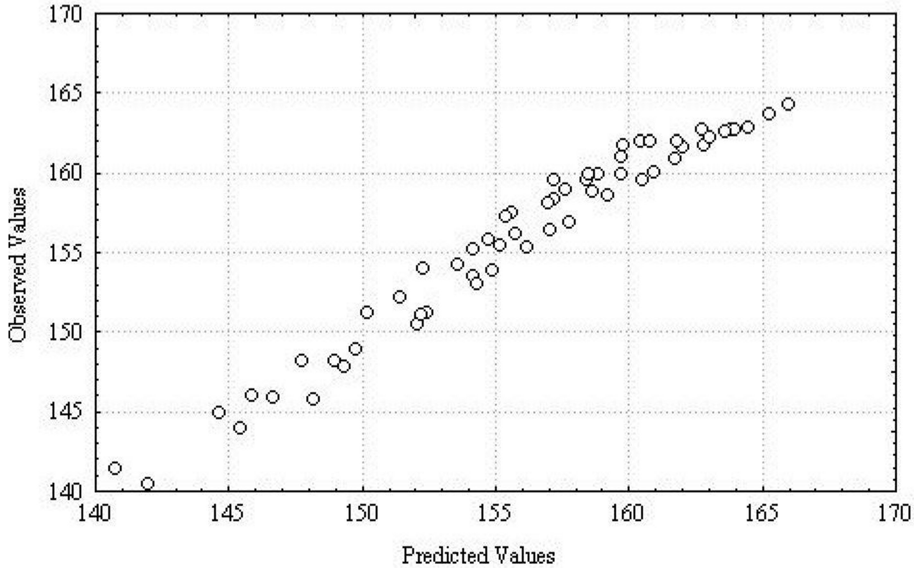

Figure 1: Observed versus predicted values of the sound pressure level (in dB re 1 μPa) for small valve opening ($s<d$).

valve stem. In contrast to the above effect is the case where eddy formation takes place in the valve stem and where length a influences the noise-generating mechanism. With the presence of the solid valve stem's wall, the dipole-type source becomes more effective, while at the same time the monopole-type source becomes less effective as a consequence of the increased mass in the stem.

3 - SUMMARY

1. Parametrical analysis enables a verification of scaling laws using the experimental data. Variation of a single geometrical parameter can be made in connection with flow characteristics or in connection with other geometrical parameters.

2. The experimental results support ideas about the combined influence of geometrical parameters on noise generation. Since, from the experiments, it was easy to detect the impact of the length and diameter on sound generation, a new measure, i.e. the ratio a/d, was implemented in order to better explain this dependency. The appropriate functional form of the equation required an additional geometrical parameter, which was determined statistically using multiple non-linear regression analysis.

3. A scaling law was experimentally derived for a nearly closed valve. As for the wide open valve, an additional geometrical parameter was introduced, in this case s/d. The expected variation of the characteristic turbulence length was explained by this parameter. No pronounced influence on the noise-generating mechanism was observed for the nearly closed valve, in contrast to measurements with a wide open valve.

4. An interesting consequence of the measurement results is the influence of the valve diameter on the noise-generating mechanism. With a wide open valve, when $d>s$, the valve diameter also influenced the noise source. This happened in a connection with valve stem length (described as the combined geometrical parameter a/d) and defines the transition between the monopole and dipole type of noise-generating mechanism.

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