A SIMPLE ACOUSTIC MODEL TO SIMULATE THE BLADE-PASSING FREQUENCY SOUND PRESSURE GENERATED IN THE VOLUTE OF CENTRIFUGAL PUMPS

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ABSTRACT
Rotor-stator interaction in centrifugal pumps and fans produces fluid-dynamic excitation at the blade-passing frequency that, depending on the operating point of the machine, can generate acoustic pressure waves of big amplitude. In order to quantify the effect of that fluid-dynamic interaction on the generation of noise, a simple acoustic model has been proposed for the volute, in which a number of ideal point sources are located at some position in the volute. These ideal sources are assumed to radiate plane sound waves along the volute at the blade-passing frequency towards both the positive and negative directions. Successive sound circulations along the volute are affected by several wave propagation phenomena, such as divergence (duct with variable section), sound emission through the impeller channels towards the inlet of the pump, or partial reflection at the tongue edge. The characteristics of the point sources and other model parameters are to be established for each pump and flow condition by comparison with available experimental data of pressure fluctuations, on a least square-error basis. This paper shows the acoustic model, the calculation algorithm for parameter optimisation, the theoretical predictions obtained for a centrifugal pump previously tested under a variety of flow-rates and the conclusions regarding the fluid-dynamic impeller-volute interaction as a source of sound.

INTRODUCTION
During the operation of fluid turbo-machinery, such as pumps, fans or turbines, fluid-dynamic perturbations are produced that can lead to vibration and noise emission. Consider the case of a conventional single suction centrifugal pump, such as the one depicted in Figure 1. It has a rotating impeller with passing channels delimited by a number of curved blades, that transfers energy to the fluid while forcing it to circulate through the machine from the axial to the radial direction. Around the impeller, a spiral shape casing, called the volute, collects the fluid coming out from the impeller and directs it towards the pump outlet. Typical fluid-dynamic excitations in this simple machine are associated to the rotation frequency (due to small misalignments, unbalance or manufacturing imperfections of the impeller), to the blade-passing frequency (rotation frequency multiplied by the number of blades of the impeller) and to broad-band phenomena (turbulence, cavitation). In the case of hydraulic pumps, excitation at the blade-passing frequency (f_{BP}) is usually dominant, especially when the pump operates at off-design conditions.

This f_{BP} excitation is a consequence of: *i)* the flow pulsations in the volute that follow the passage of the rotating blades (jet-wake pattern), and *ii)* the intermittent interaction of those disturbances and the volute at the tongue region [1]. The magnitude of this interaction, that is capable of generating acoustic pressure waves, is very dependent on the pump operating point

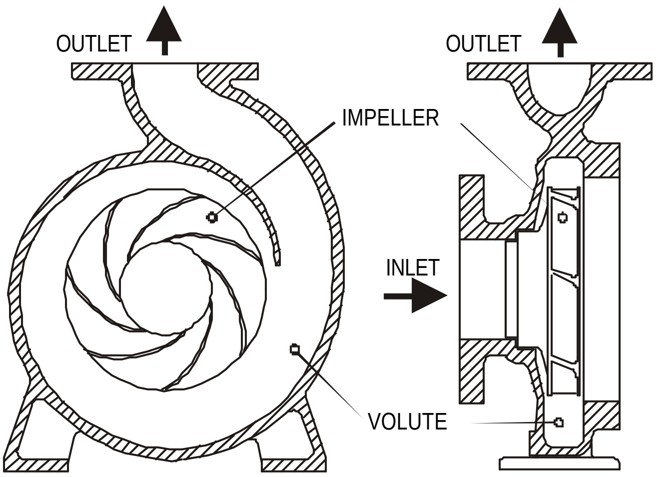

Figure 1. Centrifugal pump scheme.

[2] and on the geometry of the tongue region [3]. Since the vibration and noise levels induced by the phenomenon may become excessive under certain conditions, a low f_{BP} excitation constitutes a design objective, and there is interest in knowledge of the phenomenon and in prediction tools.

This paper presents a methodology to characterise, and even quantify, the effect of the fluid-dynamic blade-tongue interaction on the sound produced at f_{BP}. The procedure is based on a simple acoustic simulation of the pump, by imposing ideal point sources at some locations in the volute. The main parameters of the model (like position, amplitude and phase delay of each source) are to be identified by comparison between the predicted sound field and the experimental data available for each given pump and flow condition. This paper shows the basis of the acoustic model and the results obtained when applying the methodology to a conventional centrifugal pump with a non-dimensional specific speed of 0.48, previously tested in a laboratory set-up [4].

ACOUSTIC MODEL

The available experimental data suggest that the unsteady pressure field (f_{BP} component) in the volute of centrifugal pumps results from the combination of:

i) The hydraulic disturbances (jet-wake pattern of the flow behind the blades trailing edge) associated to the continuous blade rotation around the volute. In general these disturbances only affect a small region of the volute around the trailing edge of each blade, and so they may be considered to propagate along the volute at the speed of rotation of the pump (only in the positive direction).

ii) The intermittent pulsations produced from the interaction of the blade hydraulic disturbances and the volute, when a blade is passing in front of the tongue of the volute (zone of minimum gap between impeller and volute). These pulsations do generate acoustic pressure waves that propagate through the pump towards both inlet and outlet.

In order to study those two components, a simple acoustic model was considered for the volute, in which the interaction between the rotating jet-wake pattern behind the blades and the volute tongue is simulated by means of a number of ideal point sources. Each of these sources, located at some given position in the volute, is assumed to radiate plane sound waves (at f_{BP}) along the volute, both towards the direction of impeller rotation (positive) and towards the opposite direction (negative). The assumption of plane sound waves is realistic for the typical centrifugal pumps, because the wave-lengths associated to f_{BP} are usually one or two orders of magnitude greater than the dimension of the volute cross-section. Successive sound circulations along the volute are affected by: *i)* divergence, since the section of the volute increases linearly from the tongue to the exit conduit; *ii)* sound emission through the impeller channels towards the inlet of the pump; *iii)* partial reflection at the tongue edge for waves travelling in the negative direction, due to the abrupt increment in cross-section; and *iv)* sound emission through the outlet pipe for waves arriving at the exit of the volute. No incident sound was considered coming from either the inlet or the outlet pipes of the pump.

Consider one ideal source F located at the angular position φ_F (Figure 2), emitting harmonic sound at f_{BP} with a source pressure amplitude P_F and a relative time phase delay β_F. The sound transmitted from that source to another location at angular position φ can be calculated as:

$$p(\varphi,t)=P_F\left(\frac{S_\varphi}{S_F}\right)^{\alpha}\cdot e^{-j\cdot\left(\omega t-k\left|\varphi-\varphi_F\right|-\beta_F\right)} \tag{1}$$

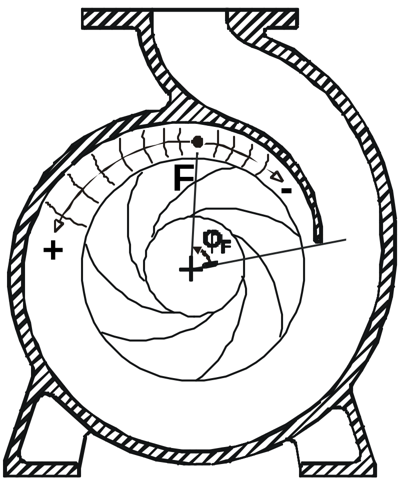

Figure 2. Test pump with one ideal acoustic source **F** at position $\varphi=\varphi_F$, radiating sound along the volute.

where $\omega=2\pi f_{BP}$, $j=\sqrt{-1}$, S_F and $S\varphi$ are the volute cross-sections at angular positions φ_F and φ respectively, and k is the angular wave number, which is equivalent to the conventional wave number but relative to angular distances rather than linear (k= number of waves per radian).

If the sound energy propagated along the volute conduit was constant then the pressure along the volute would be proportional to the square root of the volute cross-section, i.e., $\alpha=0.5$ in equation (1). However that is not the case because some sound energy is continuously 'lost' or emitted through the impeller channels towards the impeller inlet. At this position part of the incident sound waves are reflected back towards the volute (with about 180º of phase shift), due to the abrupt increment in cross-section. The net proportion of sound energy emitted through the impeller depends on the acoustic impedance of the channels. The effect on the amplitude pressure evolution along the volute is that the exponent α has to be either increased (positive propagation) or decreased (negative propagation) from the value 0.5 in a certain amount k_E: $á=-(0.5+k_E)$ for $\varphi\geq\varphi_F$ and $á=0.5-k_E$ for $\varphi<\varphi_F$, with $k_E>0$.

Eventually the sound radiated by the source F will reach the tongue edge, either from the positive or the negative directions. If the sound is propagating along the positive direction, at the tongue edge most of the sound energy will be directed towards the pump outlet, and only a small fraction, proportional to the ratio of volute cross-sections at both sides of the tongue edge, will continue for a new circulation along the positive direction. This re-circulation can be simulated by means of a new virtual source, located at the tongue edge ($\varphi=0$), with a source pressure equal to the pressure of the incident wave. This virtual source radiates sound only towards the positive direction, since no sound reflection occurs for this case.

On the contrary, the sound waves that arrive at the tongue edge from the negative direction, encounter an abrupt increment in volute cross-section, from the minimum value S_0 (for $\varphi=0$) to the maximum value $S_{2\pi}$ (for $\varphi=2\pi$). In such a situation, part of the incident sound energy is transmitted through the gap, and the rest is reflected back from the tongue edge. Under ideal conditions, the fraction of incident pressure amplitude that becomes reflected may be shown to be $k_R=(S_0-S_{2\pi})/(S_0+S_{2\pi})$, whereas the fraction of transmitted pressure amplitude is $k_T=k_R+1$ (see for instance [5]). Since $S_0<S_{2\pi}$, the sign of k_R is negative, which means a phase delay of 180º in the reflected waves with respect to the incident sound. The new circulations of the sound transmitted in the negative direction, and of the sound reflected back towards the positive direction, can be simulated by two new virtual sound sources located at $\varphi=0$, with pressure amplitudes related to the incident pressure amplitude by coefficients k_R and k_T.

In summary, the resulting pressure field $p(\varphi,t)$ at the blade-passing frequency due to N ideal sound sources, both primary and secondary (virtual sources located at the tongue edge, i.e. $\varphi_F=0$), can be modeled as:

$$p(\varphi,t)=\sum_{F=1}^{N}\left[P_F\left(\frac{S_\varphi}{S_F}\right)^{\alpha}\cdot e^{-j\cdot(ù\,t-k|\varphi-\varphi_F|-â_F)}\right]+P_B\cdot e^{-j\cdot(ù\,t-7\varphi)} \qquad (2)$$

In equation (2), the term with the P_B pressure amplitude represents the contribution of the continuously rotating jet-wake pattern to the pressure fluctuation field (hydraulic contribution to the f_{BP} pressure field). The phase delay for this term is 0, which means that the phase delay $â_F$ of each source F is relative to the fluctuations associated to the blades motion. In order to keep a reasonable degree of simplicity for the model, the amplitude of the blade pressure fluctuations, P_B, was always considered uniform along the volute, regardless the pump flow-rate.

CALCULATION PROGRAM

A special calculation algorithm was developed to compute the sound pressure field at f_{BP} in the volute of a centrifugal pump, based on the application of equation (2). The algorithm assumes the presence of some primary point sources characterised by source amplitude P_F, angular position φ_F and time phase delay â$_F$. In this algorithm the propagation of the sound waves is followed along the volute, and new virtual sound sources are generated when the waves encounter the tongue edge, either from the positive or the negative direction. This is an iterative process that finishes when the source amplitude of the new sources is sufficiently small.

In order to identify the model parameters (like the characteristics of the primary sources) corresponding to a given pump with available data of pressure fluctuations, the algorithm for pressure field calculation was incorporated in a program with a predictor-corrector algorithm. Starting from an initial set of values for the model parameters, the program evaluates sequentially if a small modification in each of the parameters can reduce the sum of square errors between the model predictions and the experimental data. Calculations end when that error is a minimum, i.e., when no additional modification in any of the parameters leads to further error reduction.

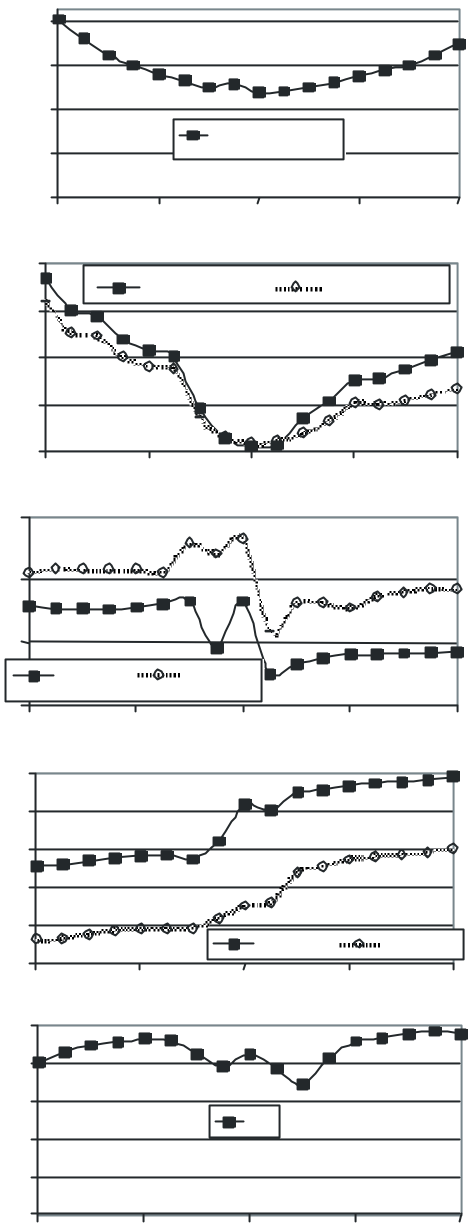

Figure 3. Parameters of sound sources (equation 2) and determination coefficient for the test pump as a funcion of flow-rate. U_2= tangential velocity at impeller outlet.

The degree of correspondence achieved between predictions and measurements was evaluated by means of the determination coefficient R^2, defined as:

$$R^2 = 1 - \left(\sum_{i=1}^{N} \left(P_i - P(\varphi_i) \right)^2 \right) \Big/ \left(\sum_{i=1}^{N} \left(P_i - \overline{P} \right)^2 \right) \qquad (3)$$

where P_i is the amplitude of the pressure measured at position φ_i, $P(\varphi_i)$ is the amplitude calculated from equation (2) and $\overline{P}$ is the arithmetic average of the N experimental data. This statistical coefficient takes values between - and 1, the closer to one the better fitting. Bearing in mind that equation (2) assumes implicitly notorious simplifications (point sound sources, uniform distribution for P_B, simplified geometry...), an R^2 coefficient of about 0.9 or greater may be considered satisfactory.

RESULTS FOR TEST PUMP

The methodology was applied to a conventional centrifugal pump, with single suction, volute casing and an impeller of 200 mm in diameter, with 7 blades of logarithmic profile. The gap between impeller and volute at the tongue edge was 10 mm (10% of radius). The pump was run in a laboratory hydraulic set-up at 1620 rpm, i.e., the blade-passing frequency was f_{BP}=189 Hz. Performance measurements indicated a best-efficiency flow-rate Q_N=0.0145 m^3 for a head of 15.04 m (non-dimensional specific speed =0.48). This pump was instrumented with miniature piezoelectric pressure transducers distributed every 10º around the front side of the volute, at 2.5 mm from the impeller outlet. Pressure signals could be digitised and FFT processed to obtain the spectra of the pressure amplitude and of the relative phase delay along the volute, for any given flow-rate. These tests were conducted for 17 flow-rates, ranging from 0 to 160% of the best efficiency

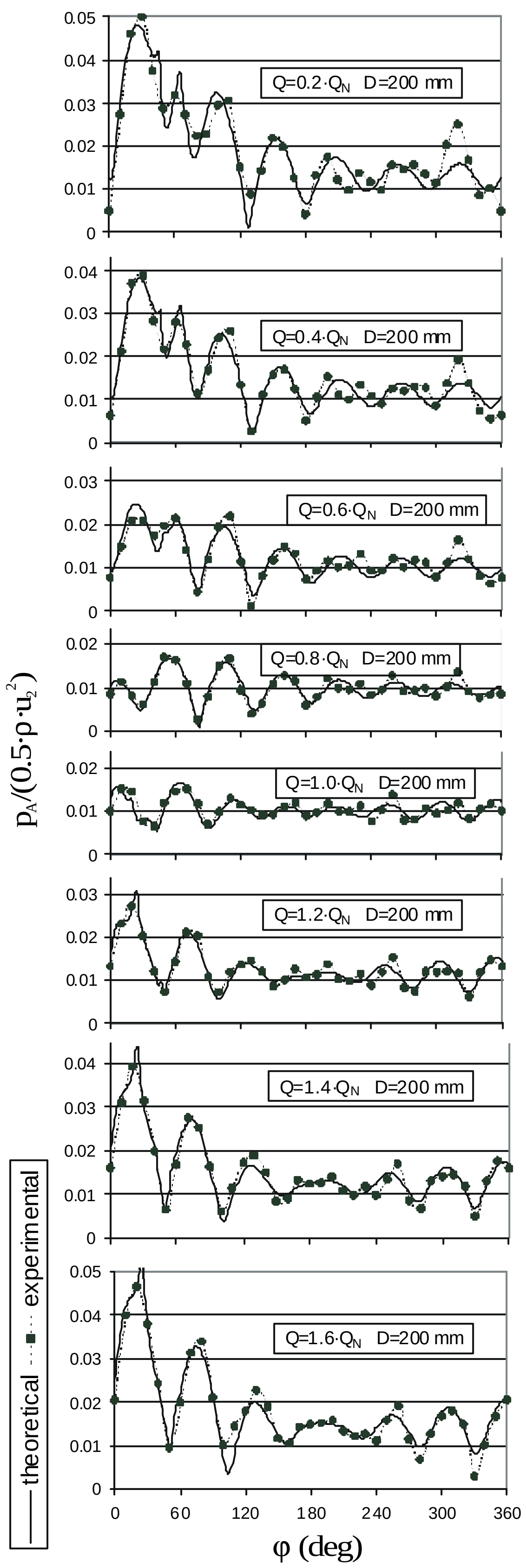

Figure 4. Comparison of the experimental and theoretical amplitude (zero-to-peak) of the pressure fluctuations coefficient at f_{BP} for several flow-rates.

flow-rate Q_N. Details of the pump geometry, hydraulic set-up, experimental procedure, instrumentation, checks and results can be found in [4].

The experimental data obtained suggested to consider not one but two different (independent) primary sources in the volute for the simple acoustic model simulations. Their pressure amplitude (P_1, P_2), peripheral position (ö$_1$, ö$_2$) and time phase delay (â$_1$, â$_2$), plus the blade pressure amplitude P_B, were estimated for each of the 17 flow-rates tested by means of the calculation program described above. Figure 3 presents the different model parameters obtained for each flow-rate, as well as the statistical determination coefficient, R^2 (the pressure amplitudes are expressed in terms of zero-to-peak values, normalised by the dynamic pressure for the impeller tangential velocity). In all cases the angular wave number was k=0.089 rad^{-1}, the reflection coefficient was k_R=-0.90 and the emission coefficient through impeller was k_E=2.0. Figure 4 shows the estimated spatial distributions of the normalised zero-to-peak pressure amplitude with the corresponding experimental data for several flow-rates.

As expected, the pressure amplitudes of both sound sources (P_1 and P_2) happen to be large at off-design conditions, thus indicating a strong blade-tongue interaction. Their order of magnitude is up to more than 10 times larger than the order of the blade amplitude fluctuation P_B. Also, they are two or three times greater than the maximum pressure amplitude measured for the respective flow-rate. For the best efficiency flow-rate region, on the contrary, P_1 and P_2 are very low, because for such range the on-coming flow-rate is expected to distribute regularly around the tongue, producing little disturbance, and thus the determination coefficients R^2 obtained with this model are rather poor.

For all flow-rates, the angular positions ö$_1$ and ö$_2$ estimated for the two primary sound sources are located in the first quadrant, close to the tongue edge, with a difference between them of about 20º and 30º for small and big flow-rates respectively. Also, the amplitude P_2 is between 70% and 100% of P_1, and the phase delays â$_1$ and â$_2$ are about 180º shifted from each other for both small and big flow-rates. All this suggests that the second sound source may actually result from the sound emission of the former along the adjacent impeller channel, followed by a negative reflection at the impeller inlet due to the abrupt increment in cross-section. Since these two ideal sound sources, though initially assumed independent, happen to be close positioned, have the same order of magnitude and radiate sound with a relative phase delay of about 180º, they behave very much like a dipole that radiates harmonic sound at the blade passing frequency f_{BP}.

In summary, the effect of the blade-tongue interaction on the fluctuating pressure field (at f_{BP}) in the volute of the pump tested can be reasonably simulated by means of two ideal sound sources that form a dipole, with amplitude that increases fast when diverging from the best-efficiency flow-rate. As expected, this blade-tongue interaction is clearly dominant in the generation of the dynamic pressure field (and the generation of noise) for off-design conditions.

CONCLUSIONS

In order to simulate and quantify the effects of the fluid-dynamic blade-tongue interaction on the blade-passing frequency excitation in centrifugal pumps with volute casing, a simple acoustic model has been considered in which a number of point sources radiate plane sound waves along the volute. The properties of these sources are to be established for each pump and flow-rate, after fitting the available experimental data of pressure fluctuations by means of a least-square error procedure. A special calculation program was developed for such purpose. The methodology has been applied to a conventional centrifugal pump with an impeller of 200 mm in diameter (gap-radius ratio of 10%), and 17 flow-rates from 0 to 160% of the best-efficiency flow-rate. The results obtained show that the effect of the blade-tongue interaction on the fluctuating pressure field (at f_{BP}) in the volute can be reasonably simulated by means of two ideal sound sources that form a dipole, except for flow-rates in the range of the best-efficiency point. The large value of the magnitude of that dipole for both low and high flow-rates confirms the dominant roll of the blade-tongue interaction in the generation of the dynamic pressure field (and the production of noise) at f_{BP}.

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