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Scaling the Vibrations of Water Pipes: Application to Industrial Case Studies

P. Moussou
LaMSID, UMR CNRS EDF 2832
1, avenue du Général de Gaulle
92141 Clamart Cedex, France

Th. Papaconstantinou
EDF R&D, Analysis in Mechanics and Acoustics Department,
1, avenue du Général de Gaulle
92141 Clamart Cedex, France

J. Cohen
EDF G&E, Engineering Department for Nuclear Plants in Operation
140 avenue Viton,
13401 Marseille Cedex 20, France

ABSTRACT

A scaling approach for the vibrations of water pipes was proposed in the ASME PVP 2005 Conference. Using a single hole orifice in non-cavitating conditions as a reference noise source, the vibrations of a piping system were estimated on the basis of plane wave propagation for the acoustics, and on the basis of vibrating beams for the structure. The scaling of the velocity Power Spectral Density (PSD) of a pipe is then a function of the pressure drop of the device considered, of the inner and outer diameters of the pipe, of the flow velocity, of the support spacing and of the densities of steel and water.

The present paper describes the application of the scaling approach to industrial case studies: velocity PSD were measured on a sample of piping systems with pressures varying from 3 to 100 bars, with flows varying from 30 to 1000 m^3/h and with hydraulic powers varying from 10 to 200 kW. The sources of vibrations were orifices, globe valves and butterfly valves in turbulent or cavitating regimes, and centrifugal pumps operating at nominal or partial flow regimes.

In most cases, the measured data match the scaling law based on the single hole orifice noise, as an order of magnitude. A closer look reveals that different sources of vibrations exhibit recognizable patterns; the partial flow regime of a centrifugal pump generates a velocity PSD different from the nominal regime, and the cavitation of a globe valve generates a velocity PSD very different from the one generated by the cavitation of a single hole orifice. Typical non-dimensional spectra are proposed for each type of noise generating device.

INTRODUCTION

Scientific works about water pipes vibrations in the low frequency range are rare (see the review in [1] and [2, 3, 4, 5, 6]). In order to bridge the gap between the hydraulic conditions of pressure drop devices and the vibration pattern of the pipes, a scaling law was proposed for the vibrations of water pipes in a former paper [7], with the objective of differentiating a normal pattern of vibrations from a non standard one.

Typical figures for industrial water pipes in nuclear power plants are the following: the flow velocity varies from 3 to 7 m/s, the hydraulic diameter varies from 80 to 300 mm, the hydraulic pressure varies from a few bars to several tens of bars, the pipe thickness varies from 3 to 30 mm depending on the diameter and the nominal pressure, the support spacing is equal to a few meters, and the total length of pipes is equal to a few hundreds of meters. Accordingly, the Reynolds number is higher than 10^5. As regards vibrations in standard operating conditions, water pipes differ from gas pipes because the Mach number is generally very small (typically lower than 0.01), and because only plane acoustic waves can propagate in the frequency range where vibrations usually have a significant level, i.e., below 1000 Hz. Likewise, only beam modes are relevant for the structure, except for thin pipes.

In the former paper, it was shown that the plane acoustic waves generated by turbulence downstream of a pressure drop device (such as an orifice or a butterfly valve) could be reasonably well described by the following pressure PSD:

$$G_{pp}(f) = p_d{}^2 \frac{0.1D}{V} \mathcal{G}\left(\frac{0.1fD}{V}\right), \tag{1}$$

where p_d is the hydraulic pressure drop of the device, V is the flow velocity in the current pipe, D is the inner diameter of the pipe, f is the frequency, and the factor 0.1 is a constant conversion factor such that $0.1\,fD/V$ is roughly equal to the actual Strouhal frequency of the device, i.e., a Strouhal frequency based on the maximum flow velocity and on some inner characteristic length. In the aforementioned paper, based on measurements made upon single hole orifices, a non-dimensional pressure source $\mathcal{G}$ equal to $10^{-6}\,S^{-3.2}$ for Strouhal numbers S varying from 0.1 to 10 was proposed as a general purpose source, with the idea that significant discrepancies with this power law would be due to the presence of some non-conventional excitation source.

The second part of the paper was dedicated to the elaboration of a scaling law for the velocity PSD of the pipe, with the help of Fluid-Structure Interaction theory (see details in [8]). A real water pipe was assumed to be reducible to a main pipe with a pressure drop device generating noise, and to some point on the main pipe where the velocity would be measured, the exact location of the point having only a small influence in the frequency range above the first natural frequency of the pipe. Considering the displacement of the pipe as the resonant response of its coupled modes to the excitation force $A_f^2\, G_{pp}$, the scaling law was written:

$$G_{vv}(f) = \left(\frac{0.1D}{V}\right)^2 \sum_{\text{modes}} \frac{S^2}{(S_n^2 - S^2)^2} \frac{A_f^2\, G_{pp}(f)}{16\pi^2 m_f m_s}, \quad (2)$$

where A_f is the fluid cross-section, m_f is the modal mass of the fluid, m_s is the modal mass of the structure and the terms S_n are the Strouhal numbers associated to the coupled natural modes of the pipe. Considering typical dimensions of water pipes (distance between supports, distance from an orifice to a T-piece…), it was proposed to use an average length L equal to 3 m to estimate the value of the modal masses, so that the scaling law of the velocity PSD was finally written

$$G_{vv}(f) \approx \frac{p_d^2}{4\pi^4 \rho_f \rho_s L^2} \frac{A_f}{A_s} \left(\frac{0.1D}{V}\right)^3 \Phi\left(\frac{0.1 fD}{V}\right), \quad (3)$$

where A_s is the surface of metal in a cross-section of the pipe, ρ_f is the fluid density and ρ_s is the structure density. As detailed in the former paper, the summation symbol disappears because the resonant term in (2) behaves as a frequency sampler, so that in each frequency range, only one mode is considered at a time. An underlying idea of the approach is that at the logarithmic scale, the non-dimensional excitation and the non-dimensional resonant response are roughly identical.

The present paper describes the application of the scaling laws (1) and (3) to a set of available field measurements on industrial piping systems. It is not intended to be an in-depth research paper, as each kind of noise generating device would require a thorough experimental study and theoretical and numerical investigation to accurately describe the noise generating mechanisms and their relation to the turbulent flow. It is neither an exhaustive validation of the scaling approach, because it would require too large a number of experimental conditions; for instance, butterfly valves of different sizes should be tested, with different openings and different flows and with different downstream pressures so that the cavitation intensity would be varied. Nevertheless, at the logarithmic scale, the results listed here appear sufficiently reproducible and meaningful to be useful in troubleshooting investigations; one should be able to identify a cavitating source or a pump operated in partial flow regime by its spectral signature.

In a first section, the application of the scaling approach to actual piping systems is discussed. In a second section, non-dimensional pressure measurements on test rigs and non-dimensional velocity measurements on industrial pipes are reproduced for each type of noise generating device that was available.

NOMENCLATURE

A_f	hydraulic cross section of the pipe (m^2)
A_s	surface of metal in a cross section of the pipe (m²)
d	hole diameter of an orifice (m)
D	inner diameter of the pipe (m)
Dp	pressure drop of the noise generating device (bars)
f	frequency (Hz)
$\Phi(S)$	non dimensional pressure force, expressed as a function of the Strouhal number
G_{pp}	power spectral density of the pressure (bar²/Hz)
G_{vv}	power spectral density of the structure velocity (m²/s²/Hz)
L	length of the pipe involving fluid-structure vibrations (m)
m_f	modal mass of the fluid (kg)
m_s	modal mass of the structure (kg)
$NPSH$	net pressure suction head (bars): the difference of pressure upstream and downstream a pump
Q	water flow (m^3/h)
S	Strouhal number, defined here as $0.1fD/V$
S_n	Strouhal number for the n-th natural mode,
V	fluid flow velocity (m/s)
ρ_f	fluid density (kg/m^3)
ρ_s	structure density (kg/m^3)

GENERAL RESULTS

As a first comment, it is worth pointing out that equation (3) was theoretically obtained for a main pipe with a constant section and without branch pipes, and one pressure drop device was assumed to be the source of vibrations. Actual piping systems are more complex: they exhibit branch pipes, T-pieces and section variations as the one reproduced in Fig. 1, and one would expect for instance the length L to be significantly variable from one case to another. The value of L is a compromise between the support spacing due to static design, which ranges from 3 to 7 m for usual pipe dimensions (see the ASME design code for instance), and between the average length of pipe between T-pieces, pressure drop devices and reducers, that would be about a ten of meters. Noting that the presence of concentrated masses reduces the effective length for accurately describing the modal mass of the pipe, L is taken equal to 3 m in all cases, with satisfying results, and no noticeable dispersion is observed at the logarithmic scale.

Second, the location and the direction of the measurement points did not appear crucial during the application of the method; different measurement points were always available among the field data, and they provided similar spectra in the middle frequency range. Of course, the proximity of supports does have an influence on the r.m.s. value of the pipe velocity, but essentially in the low frequency range, and not in the middle frequency range. By the way, it is worth mentioning the fact that the r.m.s. velocity values exhibit more dispersion from one pipe to another than the middle frequency velocity spectra, which is due to the fact that the r.m.s. velocity is sensitive to the location of the measurement point, and to the value of the first natural frequency of the pipe.

As regards the influence of the distance between the measurement point and the source of vibrations, all available measures were made at a distance shorter than 20 m, and no noticeable effect was observed, except when cavitation was at stake. Further work and more data would be required to quantify this influence.

Third, all excitation sources can hardly be described by a Strouhal frequency equal to $0.1fD/V$ and a unique non-dimensional source $\mathcal{G}$ (see the catalog of acoustic sources in [2] for instance); the overall level and the slope of the PSD depend on the inner details of the noise generating device considered. The fact that the acoustic signature of different types of hydraulic devices can be recognized by the velocity PSD of the pipe constitutes the major result of the present paper, and it is illustrated in the next section.

Last but not least, when performing an investigation of a given pipe, the hydraulic conditions must be known, and some reasonable guess of the major vibration source must be made. As regards broadband vibrations, experience shows that the main sources appear to be pumps in partial flow regime and pressure drop devices subjected to cavitation. In most cases, the major vibration source is easily identified with this criterion. When two vibration sources are in competition, experience shows that the overall vibration level is generally small, so that the superposition of sources has no practical consequences.

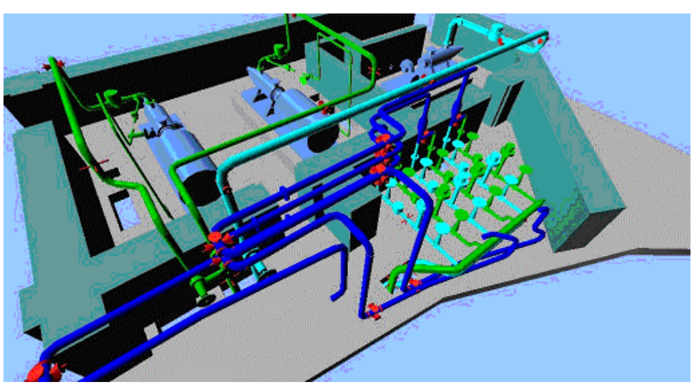

Fig. 1: partial view of a safety pipe in a nuclear power plant

ACOUSTIC SOURCES AND PIPE VELOCITIES

In the present section, non-dimensional spectra are plotted for single-hole orifices, butterfly valves, globe valves and centrifugal pumps. The choice was made to select devices for which both acoustic pressure and pipe velocities were available. In most cases, the pressure PSD are measured on test rigs and the velocity PSD are field data.

For the sake of homogeneity, all devices have their curves reported in non-dimensional form, with a Strouhal frequency equal to $0.1fD/V$ and a pressure PSD normalized by $p_d^2\,(0.1D/V)$ for pressure drop devices and $NPSH^2\,(0.1D/V)$ for pumps. The curve $\mathcal{G}(S) = 10^{-6}\,S^{-3.2}$ is plotted as a dotted line on each figure so that a visual comparison can be made with the orifice case, which is used as a reference. The dotted line ranges for Strouhal numbers from 0.1 to 10, because in the reference measurements, signal processing constraints made the pressure PSD below a Strouhal number of 0.1 meaningless.

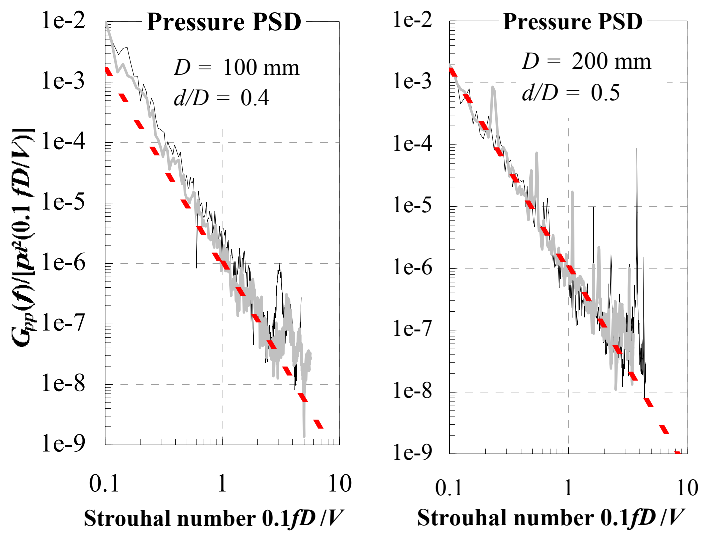

Fig. 2: non-dimensional pressure PSD generated by orifices in non-cavitating conditions

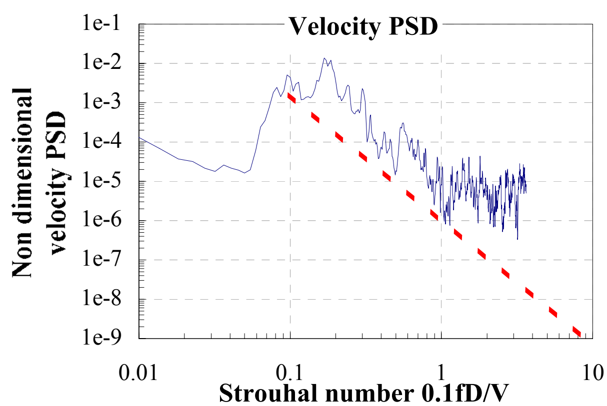

Fig. 3: velocity PSD generated by an orifice, with $p_d = 3.2$ bars, $Q = 34\ m^3/h$, $d/D = 0.25$, $t = 3.2$ mm and $D = 100$ mm

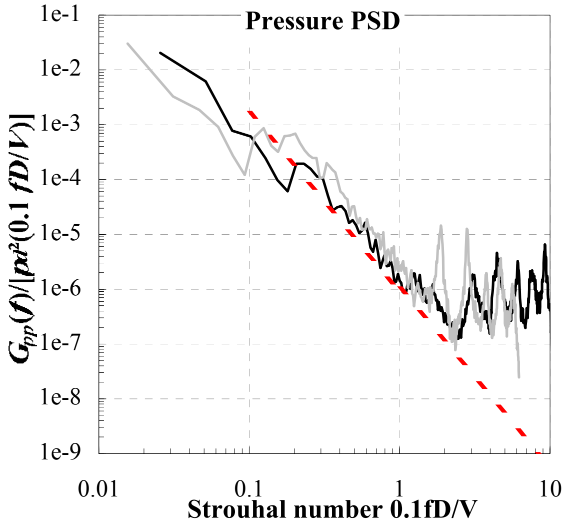

Fig. 4: non-dimensional pressure PSD generated by butterfly valves with a cavitation index equal to 1.6, an opening equal to 60% and a pressure drop of 2.8 bars and a fluid flow of 360 m^3/h (black curve), and an opening equal to 80% and a pressure drop of 1.8 bars and a fluid flow of 590 m^3/h (gray curve) (courtesy of Philippe Piteau from French CEA)

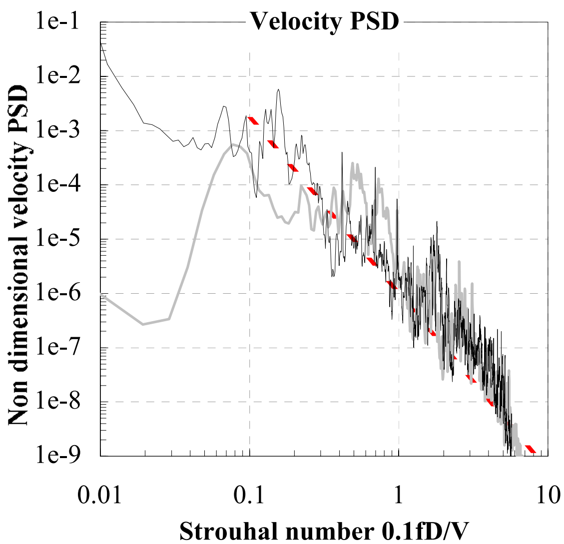

Fig. 5: velocity PSD generated by butterfly valves of two different piping systems, with p_d = 5.9 bars, Q = 845 m^3/h, t = 8.2 mm and D = 202 mm (black curve) and with p_d = 4.1 bars, Q = 180 m^3/h, t = 3.6 mm and D = 160 mm (gray curve)

Orifices and butterfly valves in standard operating conditions

The non-cavitating orifices were the base of the analysis of the former paper because of their simplicity, though they seldom constitute the dominant sources of vibrations when cavitation is not at stake. For the sake of completeness, the non-dimensional pressure PSD are reproduced in Fig. 2 for an orifice with pressure drops of 5.4 and 8.6 bars and with fluid flows of 410 and 520 m^3/h (left hand side) and another orifice with pressure drops of 5.4 and 8.6 bars and with fluid flows of 410 and 520 m^3/h (right hand side).

It can be seen that the two orifices generate similar non-dimensional pressures, and that the basis of the velocity PSD measured on the industrial site (Fig. 3) is close to the reference spectrum (dotted line). A small discrepancy is present above a Strouhal number of 1, which may be due to incipient cavitation.

The pressure PSD of butterfly valves plotted in Fig. 4 follow roughly the same trend, but for a broad hump typical of cavitation above a Strouhal number of 2, which is consistent with their cavitation index, according to the values of Tullis [9]. Two cases of pipe vibrations due to butterfly valves are reported in Fig. 5, which fairly collapse with the reference curve.

One expects then the velocity PSD of a piping system subjected to the acoustic noise of a butterfly valve or of a single-hole orifice to be close to the reference curve in the general case. As mentioned before, this type of acoustic source would seldom cause a high level of vibrations in practical cases. Nevertheless, these results are elements of validation for the scaling law.

Orifices and butterfly valves in cavitation regime

Cavitation is an issue far beyond the range of the present paper, and requires detailed investigations for each type of hydraulic component (see [10, 11, 12, 13]). A typical hydraulic indicator is the so-called cavitation index, defined as the ratio of the downstream pressure and of the pressure drop. Cavitation is present for orifices for a cavitation index approximately lower than 1, and it is present for butterfly valves for a cavitation index approximately lower than 3 (see [9] for exact values).

Cavitation is known to generate a broad hump in the middle frequency range of the acoustic pressure PSD, superposed to the decreasing turbulence-induced noise. Figures 6 and 8 provide pressure PSD for a cavitating orifice and a cavitating butterfly valve which illustrate that point. The velocity PSD of Fig. 7 was obtained with an orifice in conditions of 'super-cavitation' or choked flow [10, 11], which explains why the non-dimensional velocity PSD significantly exceeds the standard curve. The velocity PSD of Fig. 9 were generated by a butterfly valves in non-cavitating and strongly cavitating conditions [14], and the effect of cavitation is perceptible.

As pointed out by Blake [15], the scaling law of the cavitation noise cannot be similar to the turbulent noise one, because the absolute value of the pressure, the bubble size distribution and the void fraction should be involved. As a consequence, the broad humps shown in the following figures cannot be compared from one case to another. The only possible assertion is that for a given hydraulic device in given conditions of the pressure drop and flow, the broad hump amplitude increases and its average frequency decreases when the downstream pressure decreases, as shown in Fig. 6 and 8.

The non-dimensional PSD of Fig. 6 to 9 show that a significant level of cavitation generates an increase in the noise level by more than two orders of magnitude above the reference curve. The analysis of a velocity PSD on a pipe should hence easily reveal the presence of a strongly cavitating device.

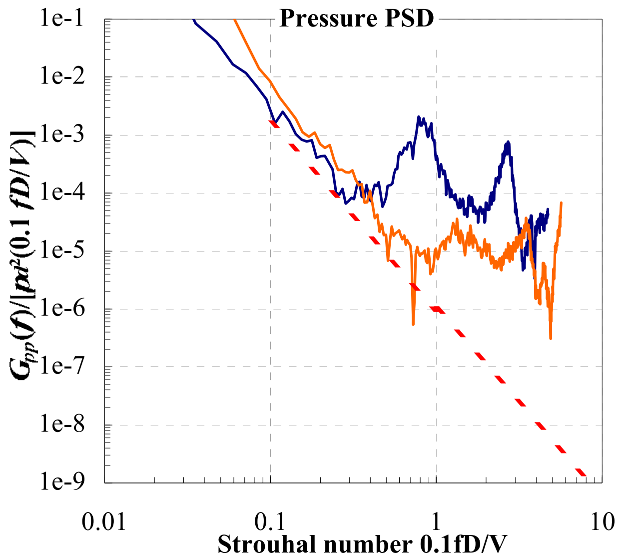

Fig. 6: pressure PSD generated by a single hole orifice with a diameter *D* equal to 100 mm, a fluid flow of 108 m^3/h and a pressure drop of 2 bars and a cavitation index of 0.3 (upper curve), a fluid flow of 90 m^3/h and a pressure drop of 1.3 bars and a cavitation index of 0.6 (lower curve)

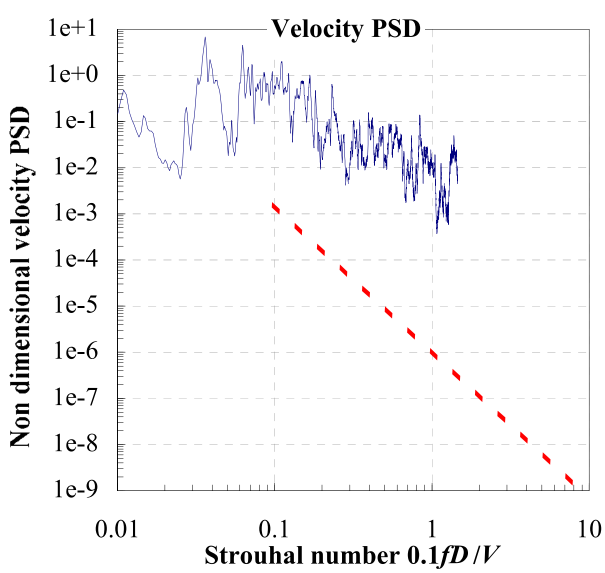

Fig. 7: velocity PSD generated by a single hole orifice with p_d = 70 bars, Q = 120 m^3/h, t = 8 mm and D = 72.9 mm, the downstream pressure being equal to about 3 bars

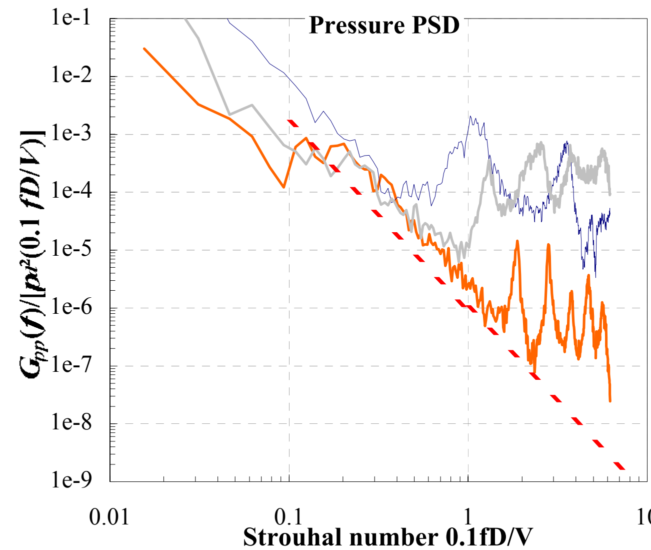

Fig. 8: pressure PSD generated by a butterfly valve with a diameter *D* equal to 200 mm, a fluid flow of 594 m^3/h, a pressure drop of 1.8 bars and an opening equal to 80 %, the cavitation index being equal to 1.6 (lower curve), 1.4 (middle curve) or 1.1 (upper curve)

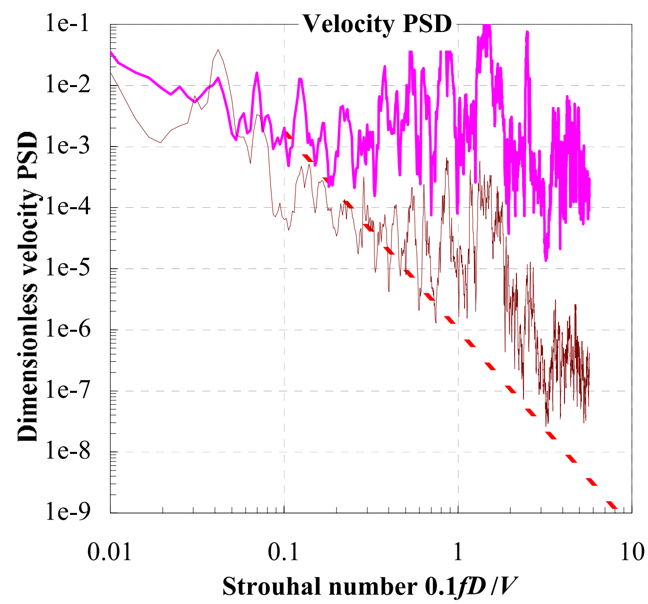

Fig. 9: velocity PSD generated by a butterfly valve with p_d = 5.9 bars, Q = 845 m^3/h, t = 8.2 mm and D = 202 mm, the downstream pressure being equal to 21 bars (lower curve) or to 5 bars (upper curve)

Globe valves

Recently, Caillaud et al. [16] have shown that globe valves in cavitation regime generate less noise than orifices and butterfly valves. Furthermore, the low frequency noise decreases below the reference curve when cavitation is present, as shown in Fig. 10 and 11. Further comments can be found in

[16], although the reason of this behavior is yet not clearly understood.

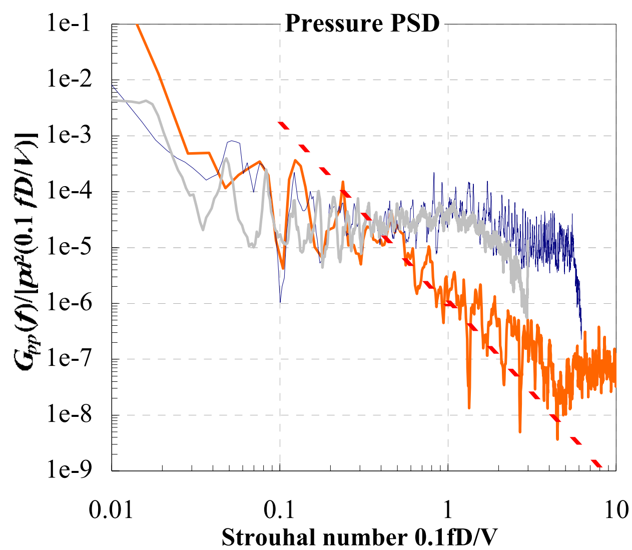

Fig. 10: pressure PSD generated by a globe valve with a diameter D equal to 74 mm: the upper curve is for a fluid flow of 38 m^3/h, a pressure drop of 7.4 bars and a cavitation index equal to 0.4, the middle curve is for a fluid flow of 79 m^3/h, a pressure drop of 32 bars and a cavitation index equal to 0.11, and the lower curve is for a fluid flow of 24 m^3/h, a pressure drop of 2.5 bars and a cavitation index equal to 1.1

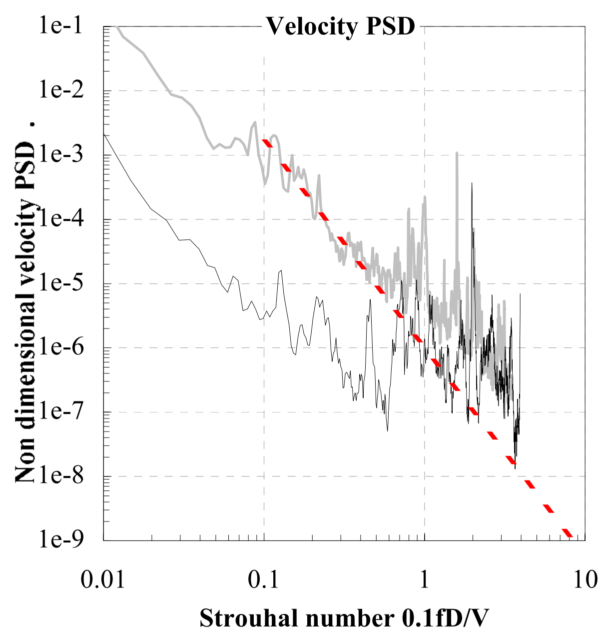

Fig. 11: velocity PSD generated by a globe valve with t = 11.12 mm and D = 67 mm, p_d = 13.3 bars, Q = 24 m^3/h and the downstream pressure being equal to 155 bars (upper curve) and p_d = 164 bars, Q = 21 m^3/h and the downstream pressure being equal to 26 bars (lower curve)

Centrifugal pumps

Though the physical mechanisms of noise generation in a pump are in all likelihood much more complex than in a pressure drop device, they deserve attention because they are the dominant source of vibrations in some practical cases. By analogy, it is proposed to scale the noise generated by a centrifugal pump by laws similar to (1) and (3), where the pressure drop is replaced by the NPSH of the pump.

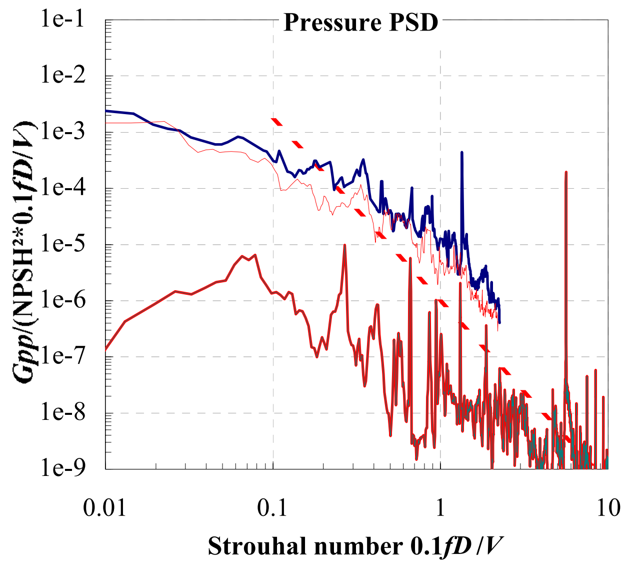

Fig. 12: pressure PSD generated by two single stage centrifugal pumps with a NPSH equal to 6.5 bars, Q = 1030 m^3/h and D = 250 mm (upper curve), with a NPSH equal to 12 bars, Q = 1050 m^3/h and D = 250 mm (middle curve), and a dual stage centrifugal pump with a NPSH equal to 67 bars, Q = 4900 m^3/h and D = 660 mm (lower curve), all pumps are operated in nominal regime

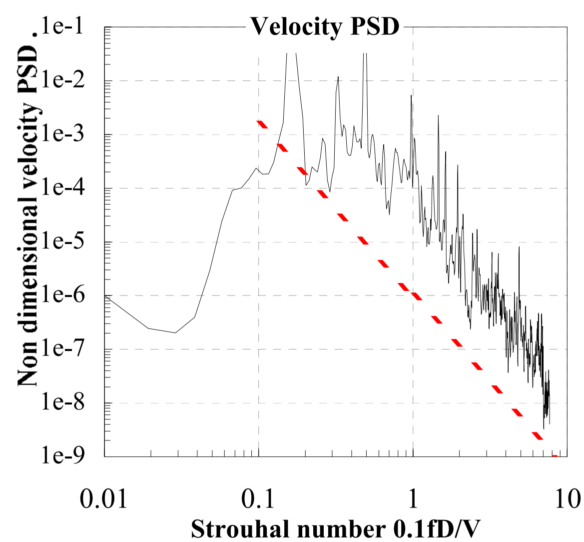

Fig. 13: velocity PSD generated by a single stage centrifugal pump in nominal flow regime, with a NPSH equal to 4.2 bars, Q = 250 m^3/h, D = 161 mm and t = 10 mm

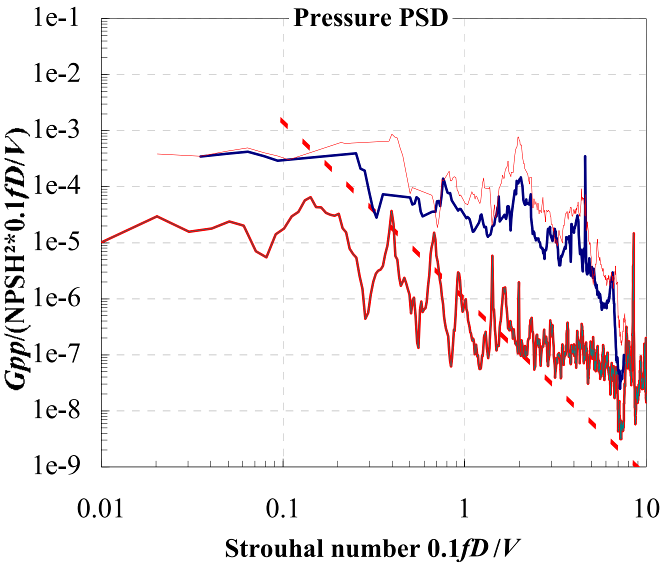

Fig. 14: pressure PSD generated by two single stage centrifugal pumps with a NPSH equal to 13.7 bars, $Q = 299$ m^3/h and $D = 250$ mm (upper curve), with a NPSH equal to 8.9 bars, $Q = 300$ m^3/h and $D = 250$ mm (middle curve), and a dual stage centrifugal pump with a NPSH equal to 69 bars, $Q = 3153$ m^3/h and $D = 660$ mm (lower curve), all pumps operated in partial flow regime

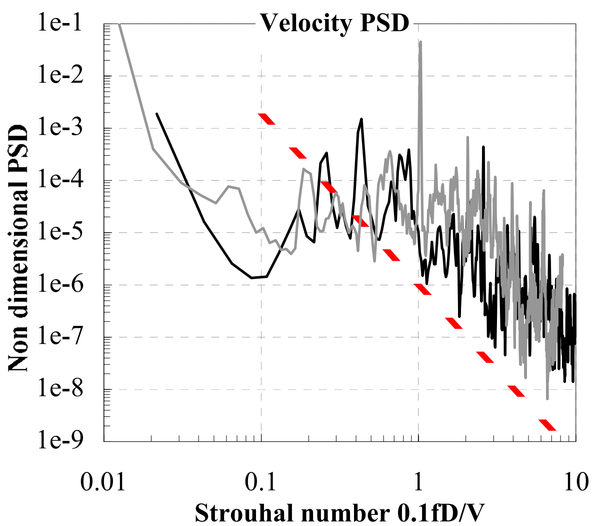

Fig. 15: velocity PSD generated by two single stage centrifugal pumps with a NPSH equal to 13 bars, $Q = 620$ m^3/h, $D = 335$ mm and $t = 10$ mm (black line), and with a NPSH equal to 8 bars, $Q = 200$ m^3/h, $D = 200$ mm and $t = 10$ mm (gray line), both pumps being operated in partial flow regime

The results of this first attempt are plotted in Fig. 12 and 13 for centrifugal pumps in nominal flow regime, and in Fig. 14 and 15 for the same pumps in partial flow regime. The curves of the single stage centrifugal pumps collapse fairly well, and they are consistent with the field data obtained with other single stage centrifugal pumps. There is hence some hope for defining a typical non-dimensional pressure PSD for pumps, but further work is needed to confirm this guess.

The PSD obtained with a two-stage centrifugal pump is reproduced for the sake of completeness; it appears much lower than the single stage pump PSD. Whether the scaling should be applied to only one stage of the pump or whether interaction of the two stages occurs cannot be determined without examining in detail the mechanism of noise generation.

DISCUSSION AND PERSPECTIVES

The vibration scaling laws proposed in [7] were applied to a broad range of hydraulic components, and validated by field test data. This method appears sound as a first level analysis tool for troubleshooting investigation, and should provide a useful help when some unidentified vibration source is present in a water pipe.

The vibrations due to orifices and to butterfly valves in turbulent and cavitating regimes seem to be fairly understood by this approach, though the amplitude of the cavitation broad hump needs further research and experimentation to be accurately predicted. If the issue is simply to determine whether there is a significant level of cavitation in a given hydraulic device, the scaling law can be successfully used.

The results obtained with centrifugal pumps are encouraging, and further validation is needed to determine if such a non-dimensional pressure PSD can be used as a reference noise source for any single stage pump, or if inner details of the device are sufficient to make the generated noise vary in large proportions. What is clearly observed is the well-known effect of partial flow upon the noise generation.

Further work is needed as well to investigate the behavior of globe valves in cavitating regimes; they might be used as low-cost anti-vibration devices in certain conditions.

One last comment can be made as regards the overall level of vibrations. As mentioned in the first section, the r.m.s. value of the vibrations appear more scattered than the average spectral line. However, when performing an integration of the non-dimensional PSD from a Strouhal number equal to 0.1 to a number equal to 10, i.e., when eliminating the low frequency components and the first natural modes of the pipes, the non-dimensional r.m.s. value obtained varies from 0.6% to 2.7% for non-cavitating orifices and butterfly valves, and for single stage centrifugal pumps. As a rule of thumb, one could use an indicative value of 1% for the r.m.s. noise generated by standard hydraulic components, which brings out a pipe velocity of the order of :

$$v_{r.m.s.} \approx 5\text{x}10^{-5} \frac{p_d}{\sqrt{\rho_f \rho_s} L} \sqrt{\frac{A_f}{A_s}} \frac{D}{V}. \tag{4}$$

This last relation does not take into account the value of the first natural frequencies of the pipe, nor the specifics of the different hydraulic components, but as an order of magnitude, it may constitute an indicator of the vibration level due to a given source. It exhibits some similarities with Eisinger's formula [17, 18] $M\,p_d$ vs. D/t, which holds for gas systems: the pressure drop is involved, and the ratio D/t as well (it is the first order term of the surface ratio A_f/A_s). However, the Mach number M is not present because in the limit of very low Mach numbers, the noise has an hydrodynamic origin and the value of the speed of sound should not be influent.

ACKNOWLEDGMENTS

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