A review of noise and vibration in fluid-filled pipe systems

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ABSTRACT
Pipe systems transporting liquid or gas are widely used in industrial applications and have the potential to transmit significant vibration to surrounding structures and radiate undesirable noise. This paper reviews previously published studies of the generation and transmission of noise and vibration in pipe systems. An overview of the vibroacoustic behaviour of fluid-filled pipes is given based on results from shell models of straight pipes. A simplified model of a fluid-filled beam can be adopted for pipes vibrating at low frequencies and the main elements of this approach are summarised. The fluid-filled beam is commonly implemented as a transmission matrix and one advantage of this method is realistic pipe circuits can be modelled by combining transmission matrices of sub-components such as straight and curved pipes, mechanical and fluid boundary conditions, and active sources. This review concludes with a discussion of research on identifying transmission and source characteristics of pumps.

1. INTRODUCTION
Noise and vibration in industrial situations is often associated with machinery, but pipe systems transporting liquid or gas create flanking paths and will transmit airborne, structureborne, and fluidborne noise to surrounding areas and supporting structures. Figure 1 illustrates common components in fluid-filled piping systems that affect the generation and transmission of airborne, structureborne, and fluidborne noise. Pumps induce pressure pulsations and turbulence in the fluid and excite structural components; valves and orifice plates also generate flow disturbances that excite the pipe system. The response of the system is greatly influenced by the pipe structure, including flanges, flexible pipe sections, and pipe bends, and these components increase coupling between different types of structural and acoustic waves propagating in the system. Pipe supports couple the system to various structures which may radiate acoustic noise more efficiently than the pipe system. It is therefore important to consider all elements of the pipe system when treating noise and vibration. Noise and vibration in piping systems have been studied in many applications: liquid and gas industrial processes (Norton, 1989); cooling water circuits on ships (Kinns, 1986, Purshouse, 1986, Verheij, 1982); hydraulic systems in passenger vehicles (Zhao, 2014); water-hammer and transient phenomena in liquid piping systems (Tijsseling, 1996); internal combustion engine exhausts; and heating, ventilation and air conditioning systems (Munjal, 1987).

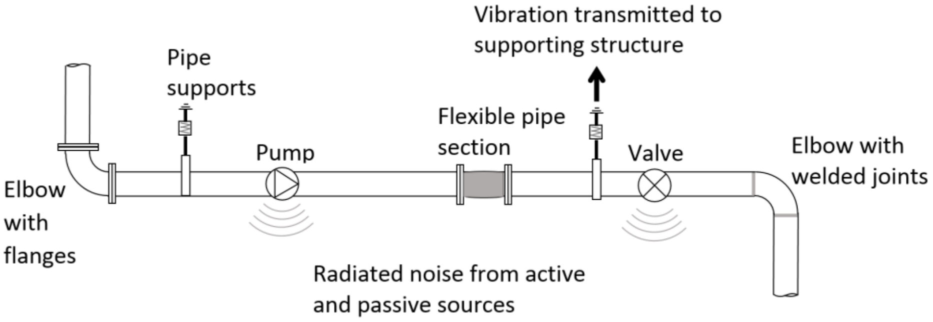

Figure 1: Common components that affect generation and transmission of structureborne and fluidborne noise in fluid-filled pipe systems.

A variety of approaches are available to model the response of pipe systems, and the physical mechanisms governing the fluid-structure interaction and vibroacoustic behaviour are common to all these applications. Differences in physical properties of the internal fluid (e.g. gas, water, or oil) and pipe or duct structure (e.g. cross-section, wall thickness) result in different mechanisms dominating the behaviour of the system. Therefore, different approximations are made when modelling the vibroacoustic behaviour of different cases. For example, at low frequencies acoustic waves propagating in gas-filled pipes can be considered as plane waves in a wave guide with rigid walls. This approximation would not be satisfactory for a duct with thin walls or for a liquid-filled pipe, except in a very restricted set of cases.

The aim of this paper is to provide a broad overview of the vibroacoustic behaviour of fluid-filled pipes and to review the main techniques for modelling pipe systems. The main focus is on liquid-filled pipe systems, for example, cooling water circuits or hydraulic systems. The following section reviews studies based on shell models of pipes, and Section 3 discusses simplified fluid-filled beam models. Section 4 describes the implementation of a fluid-filled beam in a transmission matrix and also discusses some limitations of this approach.

2. Shell models

A pipe system transporting liquid may be considered as a combination of individual components such as straight pipes, elbows, flanges, supports, pumps, etc. and each of these components can have a significant effect on the vibroacoustic behaviour of the entire system. In this context, of primary importance is the response of straight pipe sections to excitation of the structure and the internal fluid.

For pipe wall thickness h much smaller than mean pipe radius a, i.e. $h/a << 1$, a straight section of pipe can be modelled as a cylindrical shell with internal and external fluid loading. Models of cylindrical shells are described by a number of authors (Blake, 1986, Cremer et al., 1988, Fahy and Gardonio, 2007, Leissa, 1973, Norton, 1989) with Fahy and Gardonio (2007) and Leissa (1973) also discussing the effects of fluid loading. Different models vary in the extent to which they include higher-order terms proportional to a non-dimensional thickness parameter $\beta = h^2/12a^2$. Neglecting these higher-order terms yields shell equations according to Donnell-Mushtari theory. A detailed comparison of the different models is given by Leissa (1973), with further discussions by Cremer et al. (1988).

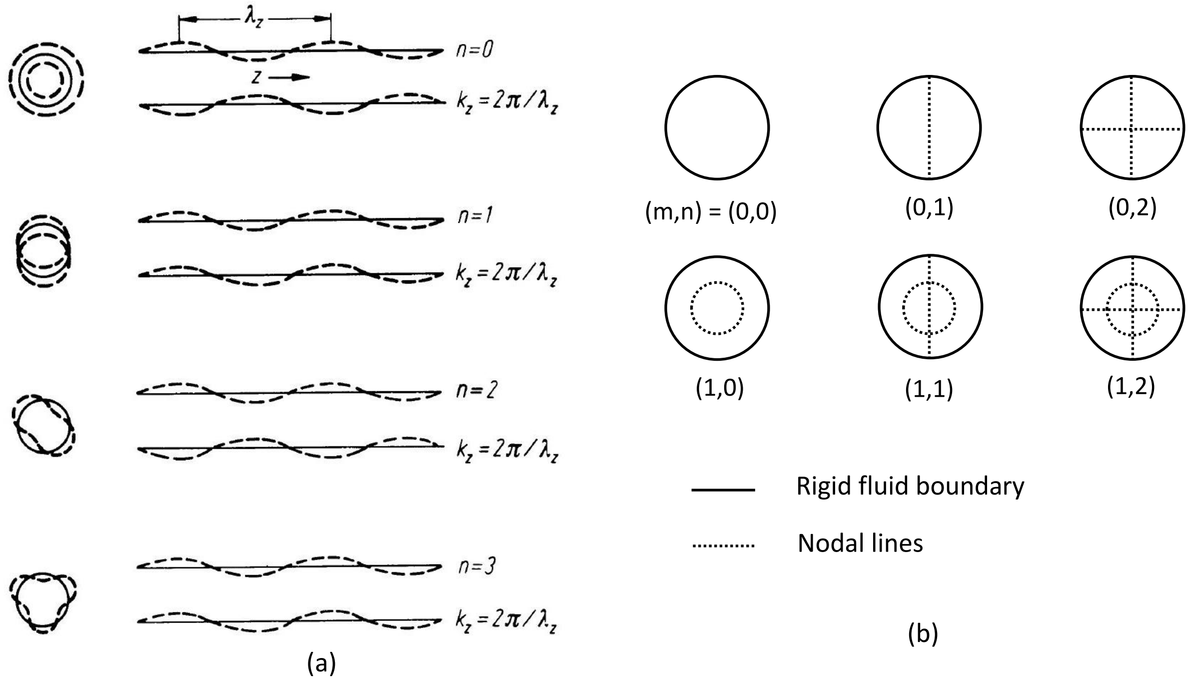

Figure 2: (a) Modes of waves in a cylindrical pipe (from Cremer et al., 1988). (b) Acoustic modes in a circular rigid duct (i.e. fluid-like waves).

Solutions to the homogeneous equations of motion for an infinite cylindrical shell describe waves that propagate in the shell and internal fluid. In certain wavenumber-frequency ranges the behaviour is dominated by shell-like waves, similar to those in a shell *in vacuo*, while in other wavenumber-frequency ranges fluid-like waves occur. The latter are dominated by fluid motion similar to that which occurs in a rigid pipe or shell. In cylindrical coordinates, the circumferential component of both shell and fluid-like waves are decomposed into distinct modes, distinguished by the number of wavelengths on a circumference in the cross section plane of the shell. Figure 2 (a) shows the first four circumferential modes of a cylindrical shell. The cross section of the shell remains circular for the n = 0 and n = 1 modes, and n > 1 are described as lobar-type modes (Leissa, 1973). The fluid-like modes, i.e. modes that exist in a rigid shell have m nodes in the radial direction in addition to the $2n$ circumferential nodes. Six low order modes are illustrated in Figure 2 (b). Note that the fluid can support an infinite number of radial modes for each circumferential mode number, although higher order modes do not propagate below their cut-on frequencies. There are also an infinite number of axial wavenumbers that exist for each circumferential mode, n. The cut-on frequencies of the $n \geq 2$ lobar modes are approximately given by (Jong, 1994, Pavic, 1992):

$$\Omega^2_{\text{cut-on}} = \beta^2 \frac{n^2(n^2-1)^2}{1+n^2+2n\mu} \tag{1}$$

where $\Omega_{\text{cut-on}} = \omega_{\text{cut-on}}/\omega_r$ is the normalized cut-on frequency; ω_r is the ring frequency (see below); $\mu = \rho_f A_f/(\rho_s A_s)$ is the ratio of mass per unit length of the fluid and shell; ρ density and A the cross-section area of the fluid (subscript f) and shell (subscript s). For example, for a steel pipe ($E_s = 2\times10^{11}$ N/m^2, ρ_s = 7800 kg/m^3, ν = 0.33; E_s, ρ_s, ν are the shell material Young's modulus, volume density, and Poisson's ratio, respectively) with external radius of 75 mm, wall thickness 4 mm, and water (ρ_f = 1000 kg/m^3) as the internal fluid, the cut-on frequencies for n = 2, 3, 4 are approximately 360 Hz, 1.1 kHz, and 2.2 kHz.

The ring frequency $\omega_r = c_l/a$ is the frequency at which the wavelength of a longitudinal wave in the shell material equals its circumference; $c_l = E_s/(\rho_s(1 - \nu^2))^{0.5}$ is the longitudinal wave speed in an infinite plate. Note that the ring frequency is not dependent on the shell thickness, only the material properties and radius. The ring frequencies for steel and rubber pipes are shown in Figure 3 for radii ranging from 10 – 150 mm. The ring frequency is 11.3 kHz for a steel pipe with 75 mm radius.

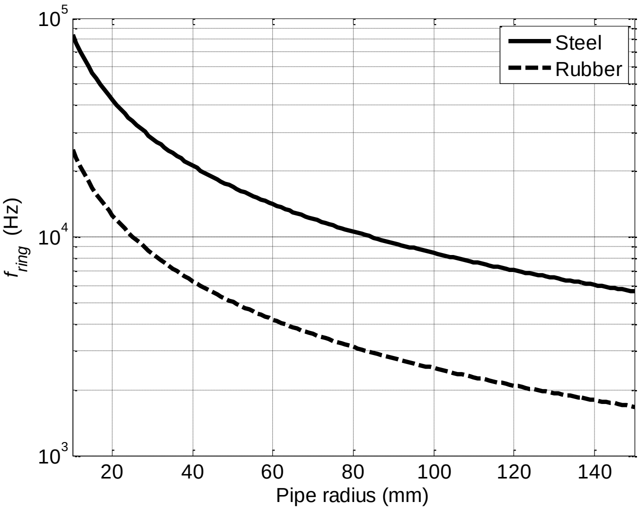

Figure 3: Ring frequency ($f_{ring} = \omega_r/(2\pi)$) vs. shell radius for steel ($E_s = 2\times10^{11}$ N/m^2, ρ_s = 7800 kg/m^3, ν = 0.33) and rubber ($E_s = 2.3\times10^9$ N/m^2, ρ_s = 1100 kg/m^3, ν = 0.4) pipes.

Below the ring frequency the deformation of the shell (pipe) is dominated by curvature; at frequencies well above the ring frequency the deformation of the shell becomes similar to that of an infinite plate; i.e. the wavelengths of circumferential modes are much shorter than the circumference. (Note that Li et al. (2015) gives an inconsistent definition of the ring frequency in terms of the wave speed in an unbounded *volume* of material).

The behaviour of free waves in an infinite fluid-filled cylindrical shell illustrates a number of fundamental aspects of the vibroacoustic response of a pipe. Fuller and Fahy (1982) derived the dispersion relations for free waves in a fluid-filled cylindrical shell for the $n = 0$ and $n = 1$ circumferential modes. They described the range of wave types that occur and also the variation in behaviour for different shell thicknesses and ratios of shell and fluid density. The dispersion relations showed that at low frequencies and $n = 0$, a fluid-filled shell exhibits a (0,0) fluid-like wave (cf. Figure 2(b)), a quasi-longitudinal extensional wave in the shell, and a torsional wave in the shell which doesn't couple to the fluid. For the $n = 1$ circumferential mode, a beam-bending wave occurs at low frequencies and as frequency increases a (0,1) fluid-like wave (cf. Figure 2(b)) is the next type of propagating wave to cut on. Higher order circumferential waves, i.e. $n \geq 2$, only propagate above their cut-on frequencies, according to equation (1). Fuller and Fahy (1982) also derive expressions for wave energy distribution in the shell and fluid. Fuller (1981) examined the effect on wave propagation of discontinuities in the shell wall, in particular, the transmission loss due to a step discontinuity in wall material, a step discontinuity in wall thickness, a finite-length discontinuity of wall thickness and also a finite-length discontinuity of wall material. These cases broadly represented the effect of different pipe components, for example, flexible dampers, flanges, and valves. A key result was that rubber pipe isolators would likely provide the highest transmission loss in a frequency range between the ring frequencies of the two pipe materials; e.g. steel and rubber; however, it is emphasised that this work studied pipes *in vacuo*.

The energy distribution in the shell and fluid has important consequences for control of pipe noise and vibration and also measurement of the vibroacoustic behaviour of pipe systems. Fuller (1983) derived the input mobility of a cylindrical shell for a line force around the circumference and also for a point force for the $n = 0, 1, 2$ modes. The distribution of energy in the shell and fluid due to these two types of excitations was also calculated. Similarly, Fuller (1984) derived the input mobility with a monopole source in the internal fluid and also calculated energy distribution in the shell and the fluid. Both these studies provided insight into how sources couple to the pipe and internal fluid, and also illustrate the mechanisms that transmit energy along the pipe. Feng (1994) derived the ratio of sound power radiated by a unit length of pipe to the vibrational energy flow through a cross section due to an external applied force. Predictions were compared with measured data in a subsequent publication (Feng, 1996). Sound radiation from a fluid-filled pipe was also investigated by Fuller (1986) who derived the radiated sound pressure due to an internal monopole source and showed that the transmission loss varied significantly with the radial location of the source in the pipe, highlighting a limitation of simplified axisymmetric models.

Pavic (1990, 1992) studied the energy flow in fluid filled pipes in terms of the different types of propagating waves and gave recommendations for a methodology to measure the energy flow in pipe systems. Earlier, Verheij (1982) had carried out similar work in the context of shipboard machinery and included results from experimental testing. Each of these studies discussed the number and direction of measurement locations on the pipe wall required to adequately measure the dominant wave types transmitting energy along the pipe. This is an important consideration when attempting to characterise the vibroacoustic response of a pipe system, especially when trying to assess the effectiveness of control measures such as flexible pipe sections. Pavic (2003) described a methodology for determining the fluid pressure spectrum and speed of sound in the internal fluid from an array of measurements along the pipe.

The effects of mean flow were discounted in the studies referenced above. Michalke (1989) investigated these effects for the case of sound propagation in a rigid pipe with pressure and mass excitation sources in the internal fluid. Leyrat (1990) carried out a comprehensive analytical and experimental study of wave propagation in infinite and finite fluid-filled pipes with mean flow. Predicted results compared favourably with experimental data. Brevart and Fuller (1993) derived the dispersion relations for an infinite pipe with mean flow and also derived the distribution of vibrational energy flow in the shell and the fluid. These investigations suggested that mean flow would affect the propagation of waves but the influence is small for low Mach numbers (Mach number $M = U/c_f$; U is the fluid velocity; c_f is the speed of sound of the internal fluid). Mean flow can reasonably be neglected at low frequencies well away from cut-on frequencies of higher-order modes and if Mach number is less than 0.01. This is likely to be the case in many applications; for example, a recommendation for maximum fluid flow rate in shipboard systems given by the Society of Naval Architects and Marine Engineers (Fischer et al., 1983) is 3.7 m/s (<< 15 m/s = 0.01*M*).

3. Simplified beam models

The studies discussed above were based on shell models of straight pipes. These models cannot be easily adapted to include the effects of other components such as elbows, pumps, or pipe supports. However, the results from shell models suggest that at frequencies well below the ring frequency of the pipe, the vibroacoustic behaviour and energy transmission is dominated by the circumferential modes where the pipe's cross section remains circular (i.e. $n = 0$ and $n = 1$). This has been exploited by researchers who have adopted simplified models to describe the vibroacoustic behaviour of pipe systems. Early work was directed at the study of transient behaviour of liquid-filled pipe systems, e.g. waterhammer, initially considering axial motion and Poisson coupling between the fluid and the structure. Models were progressively expanded to include translation and rotation in other degrees-of-freedom, and solution methods in both the time and frequency domain were developed. Comprehensive reviews of transient phenomena in pipe systems were published by Li et al. (2015), Tijsseling (1996), Wiggert and Tijsseling (2001).

The most general model, which is commonly referred to as a 'fourteen equation model', consists of equations to describe plane waves in the fluid and axial, torsional, bending and shear motion in the pipe structure (Jong, 1994, Kwong and Edge, 1996, Lesmez, 1989, Liu and Li, 2011, Tentarelli, 1990, Xu et al., 2014, Zhao, 2014). Equations (2) – (12) in Table 1 were given by Jong (1994) and coordinate definitions are shown in Figure 4.

Table 1: Equations of motion (frequency domain) for a fluid-filled beam

<u>Axial motion of pipe and fluid</u>

$$\frac{\partial p}{\partial z} = \rho_f \omega^2 u_f \qquad (2)$$

$$\frac{\partial F_z}{\partial z} = -\rho_s A_s \omega^2 u_z \qquad (3)$$

$$\frac{\partial u_z}{\partial z} = \frac{F_z}{E_s A_s} - \frac{2\nu}{E_s}\frac{a}{a_i}\frac{A_f}{A_s} p \qquad (4)$$

$$\frac{\partial u_f}{\partial z} = \left[-\frac{1}{K_f}\left(1 + \frac{2a}{h}\frac{K_f}{E_s}\right)\right] p + \frac{2\nu}{E_s}\frac{a}{a_i}\frac{F_z}{A_s} \qquad (5)$$

<u>Flexural motion of pipe (*x*-*z* plane)</u>

$$\frac{\partial F_x}{\partial z} = -\left(\rho_s A_s + \rho_f A_f\right)\omega^2 u_x \qquad (6)$$

$$\frac{\partial M_y}{\partial z} = -F_x - \rho_s I_s \omega^2 \Phi_y \qquad (7)$$

$$\frac{\partial u_x}{\partial z} = \frac{F_x}{\kappa_s G_s A_s} + \Phi_y \qquad (8)$$

$$\frac{\partial \Phi_y}{\partial z} = \frac{M_y}{E_s I_s} \qquad (9)$$

<u>Torsional motion</u>

$$\frac{\partial M_z}{\partial z} = -\rho_s J_s \omega^2 \Phi_z \qquad (10)$$

$$\frac{\partial \Phi_z}{\partial z} = \frac{M_z}{G_s J_s} \qquad (11)$$

$$\kappa_s = \frac{2(1+\nu)}{(4+3\nu)} \qquad (12)$$

a : mean pipe radius
a_i: internal pipe radius
K_f: bulk modulus of internal fluid
G_s: shear modulus of shell

$J_s \approx 2\pi^3 h$: torsional moment of inertia of pipe

I_s: area moment of inertia of pipe
κ_s: shear coefficient of thin walled pipe

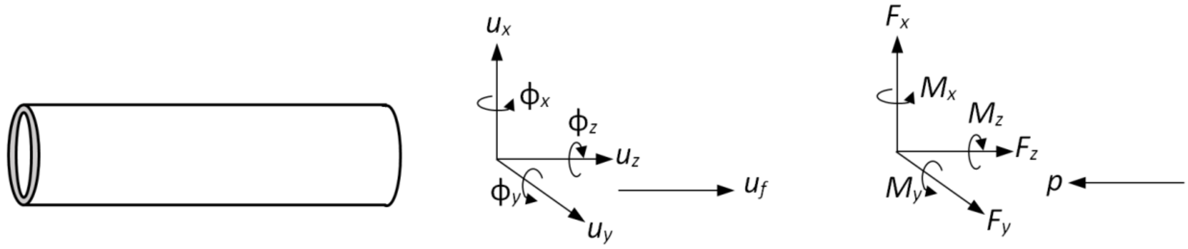

Figure 4: Coordinates and directions of displacements (*u*), rotations (ϕ), forces (*F*), moments (*M*), pressure (*p*) and fluid velocity (u_f) in a straight section of fluid-filled pipe.

Equations (2) – (5) describe axial motion of the fluid and pipe, assuming an inviscid fluid. Coupling between the fluid and the pipe due to Poisson contraction is described by the $2\nu a/(E_s a_i A_s)$ coefficient of the last term in equations (4) and (5). The coefficient of *p* in the first term of equation (5) is a modified bulk modulus of the internal fluid, accounting for the elasticity of the pipe walls. In very compliant pipes containing incompressible fluid ($K_f >> E_s$), the effective speed of sound in the fluid is decreased. Equations (2) and (3) can be modified to include frequency dependent friction and derivations are given by Kwong and Edge (1996) and Tentarelli (1990). Jong (1994) suggested that damping by viscous friction be included as a complex fluid bulk modulus, which can also account for the effects of gas bubbles in the internal fluid.

Equations (6) – (9) describe flexural motion in the *x* – *z* plane and similar equations apply to the *y* – *z* plane, after transposing coordinates. Equation (6) includes the contribution of fluid in the translational inertia term; however, a point of difference among authors is whether the fluid contributes to the total rotary inertia in equation (7). Jong (1994), Tentarelli (1990), Tijsseling (1996), Wiggert and Tijsseling (2001) do not include the rotational inertia of the fluid; Kwong and Edge (1996), Lesmez (1989), Liu and Li (2011), Xu et al., (2014), Zhao (2014), add a $-\rho_f I_f \omega^2 \Phi_y$ term to equation (7), where I_f is area moment of inertia of the fluid. An ovalisation or flexibility factor h_b can be used in equation (9) to reduce the bending stiffness; i.e. $E_s I_s / h_b$, when modelling curved pipes. Further comments on modelling curved pipes and bends are included below.

Other refinements to the fourteen equation model include accounting for the effects of gravity, internal pressure, fluid flow, centrifugal and Coriolis forces. These are more likely to be important for modelling transient phenomena or situations with very high flow speeds (Li et al., 2015, Wiggert and Tijsseling, 2001).

Jong (1994) summarised the assumptions that are the basis of the fourteen equation model of a fluid-filled pipe: fluid and pipe are each homogeneous, isotropic, and exhibit linear elastic behaviour; pipe wall thickness is small compared with pipe radius (β^2 << 1); pipe length is large compared with pipe diameter; and the effect of mean fluid flow is neglected since the fluid Mach number is small. It is emphasized that the simplified beam model only accounts for the n = 0 and n = 1 modes and for frequencies below the cut-on of the (0,1) fluid-like wave. Fuller and Fahy (1982) gave the normalised cut-on frequency of the (0,1) fluid-like wave as $\Omega_{\text{cut-on (0,1)}}$ = 0.52. Jong (1994) argued that the model in Table 1 still holds when higher order modes cut-on but becomes incomplete.

4. Transmission matrix modelling

Equations (2) – (11) (plus an additional four equations for bending in the *y* – *z* plane) can be solved in the time domain or frequency domain. Time-domain solutions, for example using the method of characteristics, have been reported for studies of transient behaviour in liquid-filled pipes and a summary can be found in Wiggert (2001). Frequency domain solutions implemented as transmission matrices have been presented for problems considering vibration in pipe systems (Jong, 1994, Kwong and Edge, 1996, Lesmez, 1989, Liu and Li, 2011, Xu et al., 2014, Zhao, 2014). Transmission matrices $\boldsymbol{T_{ij}}$ relate the structural and fluid variables $\boldsymbol{x}_i$ = {p, u_f, F_z, u_z, F_x, u_x, M_y, Φ_y, F_y, u_y, M_x, Φ_x, M_z, Φ_z}^{T} at each end (*i,j*) of a section of straight pipe and allow complicated systems to be modelled by combining the transmission matrices of individual components (Pestel and Leckie, 1963). Jong (1994), Kwong and Edge (1996) pointed out that matrix conditioning problems may arise in certain situations when long pipe lengths are modelled or when large systems are considered. Both authors suggested re-scaling the parameters in the transmission matrices to decrease the relative magnitudes of matrix elements. In addition, modelling long pipes as a series of shorter pipes was also reported to improve matrix conditioning.

An advantage of the transmission matrix methodology is that branches and boundary conditions at any junction can easily be applied. Transfer matrices of lumped mechanical impedances and fluid impedances were given by a number of authors to model the effect of flexible pipe supports; the added mass due to pipe flanges or valves; resistance to fluid flow caused by valves or orifice plates; and fluid compliance from accumulators (Jong, 1994, Kwong and Edge, 1996, Kwong and Edge, 1998, Liu and Li, 2011, Tentarelli, 1990, Zhao, 2014).

Pipe bends produce additional coupling between different degrees-of-freedom, e.g. bending and torsion; fluid pressure and bending. Bending stiffness is also affected by the presence of non-circular cross-sections of a bend. El-Raheb (1981) derived a continuous model of a pipe bend to investigate the coupling between plane acoustic waves and pipe bending deformation. The analysis showed that the plane wave assumption was valid for frequencies below *half* the first acoustic cut-on frequency. Tentarelli (1990) also derived a continuous model of a pipe bend and suggested this approach was warranted because of the extra computational cost of a discrete model of a pipe bend made from a series of small sections of straight pipe. Jong (1994) compared the results from a continuous model, a discrete model, and also a finite-element model and reported good agreement between each approach. It was noted that the flexibility factor used to reduce the bending stiffness was a key parameter in both the continuous and discrete transmission-matrix models and Wiggert and Tijsseling (2001) pointed out that this can also be affected by fluid pressure.

Experimental validation of the transmission matrix approach has typically been given for simple representative pipe circuits. For example, Tentarelli (1990) showed good agreement between predicted and experimental results over a frequency range of 0 – 1000 Hz for a hydraulic pipe system that included an elbow, T-junction, clamped pipe support and accumulator. The nominal pipe radius was 5.8mm and the internal fluid was hydraulic oil. Jong (1994) successfully modelled a water-filled pipe system including blocking masses, bends, and flexible bellows. The pipe internal radius was 75 mm and data were presented up to 1000 Hz. Kwong and Edge (1996) conducted experiments on a planar and a three-dimensional hydraulic circuit (10mm internal radius) with mean flow. Comparison of experimental and predicted results highlighted the importance of including a frequency-dependent friction factor in the model. Further work by Kwong and Edge (1998) used the transmission matrix model of the hydraulic system in an optimization routine to determine the best and worst locations for a pipe clamp. This work also included experimentally measured structural impedance as a boundary condition in the model. The best and worst pipe clamp locations were verified with experimental data; however, moderate discrepancies were identified and the accuracy of the boundary conditions applied to the model was one possible reason for the inaccuracies. This highlights a key difficulty in modelling fluid-filled pipe systems: the accuracy of the models is dependent on the mechanical properties of subcomponents and these properties may be difficult to accurately quantify, for example, the stiffness of bolted pipe flanges or valves.

Similar difficulties arise in characterising pumps and other components that generate pressure fluctuations in the fluid, for example sharp bends and orifice plates. Pumps are a source of fluid pressure excitation, structural excitation, and also airborne noise. Broadband excitation occurs due to turbulence and narrow band excitation occurs at blade pass frequency. Pumps can be modelled as a transmission matrix and a source of pressure and volume velocity (Rzentkowski and Zbroja, 2000):

$$\begin{bmatrix} p_o(\omega) \\ q_o(\omega) \end{bmatrix} = \begin{bmatrix} T_{11}(\omega) & T_{12}(\omega) \\ T_{21}(\omega) & T_{22}(\omega) \end{bmatrix} \begin{bmatrix} p_i(\omega) \\ q_i(\omega) \end{bmatrix} + \begin{bmatrix} p_s(\omega) \\ q_s(\omega) \end{bmatrix} \quad (13)$$

where *p* and *q* represent frequency dependent pressure and volume velocity; subscripts *i*, *o*, and *s* refer to inlet, outlet, and source; and T_{ij} are the elements of the frequency dependent transmission matrix of the pump.

Standards exist for the measurement of fluid pressure ripple (e.g. ISO 10767-1:2015 (2015)), and researchers have studied other methods for identifying the source and transmission characteristics of pumps. The various approaches differ in the number and location of measurement points, excitation source, and signal processing algorithms to extract the elements of equation (13). Jong (1994) tested methods for estimating the transmission matrix, pressure and volume velocity source terms for a centrifugal pump. His results suggested that the transmission matrix was approximately independent of operating condition. Carta et al. (2000, 2002) also discussed the conditions where the transfer matrix of a non-cavitating pump can be measured independently of its operating conditions. Experimental measurements on a centrifugal flow pump in air and water indicated the approach was feasible at low frequencies. Source levels were shown to be dependent on the operating point of the pump with narrowband and broadband components increasing at partial flow conditions.

Charley and Carta (2001) estimated transmission matrices of a centrifugal pump and illustrated the effect of ignoring the coupling between the fluid and the structure. Pavic and Chevilotte (2010) introduced a new method for characterising the source characteristics of a hydraulic pump, but reported difficulties in obtaining acceptable results. Qi and Gibbs (2003) considered the total acoustic power generated by heating circulation pumps due to fluid and structural excitation. They estimated the structureborne component of power from free velocity and pump mobility and noted that the power induced by moment excitation was small compared with that due to translational forces. The fluidborne component of power was measured in a pipe with semi-anechoic termination.

While the research described above illustrated the feasibility of various methods to characterise pumps, the practical difficulties in carrying out measurements has led to limited published data for use in modelling. In the absence of measured data, some empirical and theoretical models have been proposed to estimate pump transmission and source characteristics (Guelich and Bolleter, 1992, Simpson et al., 1967).

5. CONCLUSIONS

A review of literature on the generation and transmission of noise and vibration in liquid-filled pipe systems has been presented. The vibroacoustic behaviour of a pipe has been discussed based on results from shell models of straight pipes. At low-frequencies, a simplified fluid-filled beam can adequately describe the vibroacoustic response of a pipe and is often implemented with a conventional transmission matrix approach. This approach facilitates modelling of realistic pipe systems because transmission matrices of individual components such as straight pipes, bends, boundary conditions and sources can be easily combined. A key limitation is obtaining accurate data for some pipe system components such as pipe junctions and pumps.

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