Fluid-structure interaction in liquid-filled pipe systems : sources, solutions and unsolved problems

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EDF Final Report

TU/e technische universiteit eindhoven

FLUID-STRUCTURE INTERACTION IN LIQUID-FILLED PIPE SYSTEMS: SOURCES, SOLUTIONS AND UNSOLVED PROBLEMS

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Research Report

Final Report for:

ELECTRICITÉ DE FRANCE
Direction des Etudes et Recherches
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France

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Résumé

This final report is the result of an extensive literature and internet search. It is intended as an introduction to the subject and as a guide to relevant literature.

The main themes are FSI (fluid-structure interaction) and FIV (flow-induced vibration) in liquid-filled pipe systems. Roughly, one may say that FSI refers to unsteady and FIV to steady flow phenomena, exciting pipe systems. The review is specially dedicated to EDF.

The report lists 146 references.

SUMMARY

The themes of this report are fluid-structure interaction (FSI) and flow-induced vibration (FIV) in internal liquid flows (as opposed to external and gas flows) in pipe systems. Roughly one may say that FSI stands for unsteady-flow induced vibration and FIV for steady-flow induced vibration. This is explained in Section 2.

The report is a survey of and a guide to literature on FSI and FIV. For FSI, the review by Wiggert and Tijsseling (2001) is the starting point. Section 3 summarises the subject and lists new references. For FIV, the review by Weaver *et al* (2000) is the starting point. Section 4 summarises this paper and also the standard textbooks. A list of all possible excitation sources can be found in Section 5. Vibrating bends and branches, being the most important FSI mechanisms, are studied in Section 6. The Sections 7 and 8 are on codes, standards and current research activity. Future work and conclusions are given in the Sections 9 and 10. The list of references comprises 146 publications.

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1. PERSONAL BACKGROUND

1.1. The author

Arris S Tijsseling has an MSc (in Technical Mathematics and Informatics) and a PhD (in Civil Engineering) from Delft University of Technology in The Netherlands. He has spent two and half years at the renowned Dutch research institutes National Aerospace Laboratory and Delft Hydraulics Laboratory. He has been a post-doctoral research fellow with Professor Alan E Vardy, working on the suppression of waterhammer-induced vibration, in the Department of Civil Engineering of the University of Dundee, Scotland. On 1 December 1995 he was appointed Lecturer in the field of Fluid Mechanics. Per 1 September 1999 he is Assistant Professor in the Department of Mathematics and Computer Science of Eindhoven University of Technology.

The author has been working on fluid-structure interaction (FSI) in pipe systems since 1986 and has about 35 publications on this subject. In 1994 he was co-recipient of the Vreedenburgh prize of the Royal Dutch Institute of Engineers. The applicant's work is referenced in the Dutch Pipe Code NEN-3650 English (1998). He is co-author of EDF-DRD Technical Note HP-54/99/030/A (1999).

1.2. The institute

Eindhoven University of Technology (TUE) and
Eindhoven Centre for Computational Engineering (ECCE)

TUE - ECCE

TUE is a relatively young university. It was founded some 45 years ago and it is situated in the southern part of The Netherlands. ECCE is a co-operation of some 15 chairs at TUE, from the Departments of Mathematics and Computer Science, Technical Physics, Mechanical, Chemical and Electrical Engineering. Industrial orientation, and at the same time a recognised high quality of research, is evidenced by the establishment of top technological institutes (TTIs) and top scientific research schools in selected areas with a strong international profile. Indeed, one out of four technological institutes and two out of five selected top research schools in The Netherlands have their main base in Eindhoven.

Research

Theoretical and experimental science are more and more supplemented by computational science. Modelling and numerical simulation are therefore an important tool. Typical areas in which various groups work together are material technology, biomedical technology, process industry, dynamics and control, heat and mass transfer and geophysical flows. Multidisciplinary teams work here together with industrial companies on both small problems and longer-term basic issues.

Training

ECCE fosters a training programme in Computational Science and Engineering (CSE). The master's programme has a strong application orientation. Students are expected to take a minor in a technical discipline and to carry out a practical 9 months project in an industrial environment. The PhD programme is usually based on a problem originating from and often paid by industry. Also in the framework of ECMI there is a European graduate programme.

Service

There exists strong co-operation with a large variety of industries. Smaller and larger projects for companies are done by students, who often find a job there after completing their thesis. The mathematics department has a special service bureau (IWDE) for technical co-ordination of such contract work. There is expertise in doing this on an international level.

2. INTRODUCTION

2.1. Context of the study

Vibration in piping systems is a complex phenomenon not well understood by many pipework designers. Currently there are no standards and just a few guidelines to assist in determining which systems might be at risk. Through the project VICI, EDF is trying to improve its comprehension of vibrations of water pipes in the low frequency range and to propose an approachable method of analysis for the majority of the EDF piping systems (Moussou 2002).

This contract with Eindhoven University of Technology intends to fulfil a part of this objective by summarising, from an engineering point of view, the mechanisms inducing vibrations in pipe systems.

This report intends to be a guide to published work (papers and books) in the area of FSI (fluid-structure interaction) and FIV (flow-induced vibration) in liquid-filled pipe systems. This is a very broad area, and only certain aspects of importance for EDF will be highlighted. The report should leave the reader with an impression of what exists and of what still has to be developed. Furthermore, codes and standards used in practice will be compared with the present state of knowledge. An inventory of current research activities is presented.

2.2. FSI versus FIV

The terms FSI and FIV are used promiscuously in the literature. In this report, the term FSI is used for unsteady flow interacting with pipe vibration. The term FIV will be used for stationary flow inducing pipe vibration. FSI generally involves two-way (fluid $\leftrightarrow$ pipe) interaction. For FIV the interaction normally is one-way (fluid → pipe). Of course, there are regions where the above-made definitions of FSI and FIV overlap.

The stationary (or steady) flow is considered time-averaged: it includes turbulence and vorticity, and the associated small (compared to the steady state) velocity and pressure fluctuations. In this sense, the above-made definitions differ from the classification made by Blevins (1990, p. 2) which is shown in Figure 1. The top diagram refers to FIV and the bottom diagram to FSI, except that herein the random excitation by turbulence is placed under the FIV label.

The terms waterhammer and acoustics will be used in the following way, noting that both phenomena concern pressure waves. Waterhammer is transient unsteady flow of large amplitude, considered in the time domain. It is related to FSI. Acoustics concerns steady-oscillatory flow of small amplitude, which may become of large amplitude under resonance and instability conditions. It is considered in the frequency domain and related to both FSI and FIV. Waterhammer can be generated by valve and pump operation. Acoustic sources can be vortex shedding and the vibration of fluid machinery. Of course, there are many grey areas where both definitions apply.

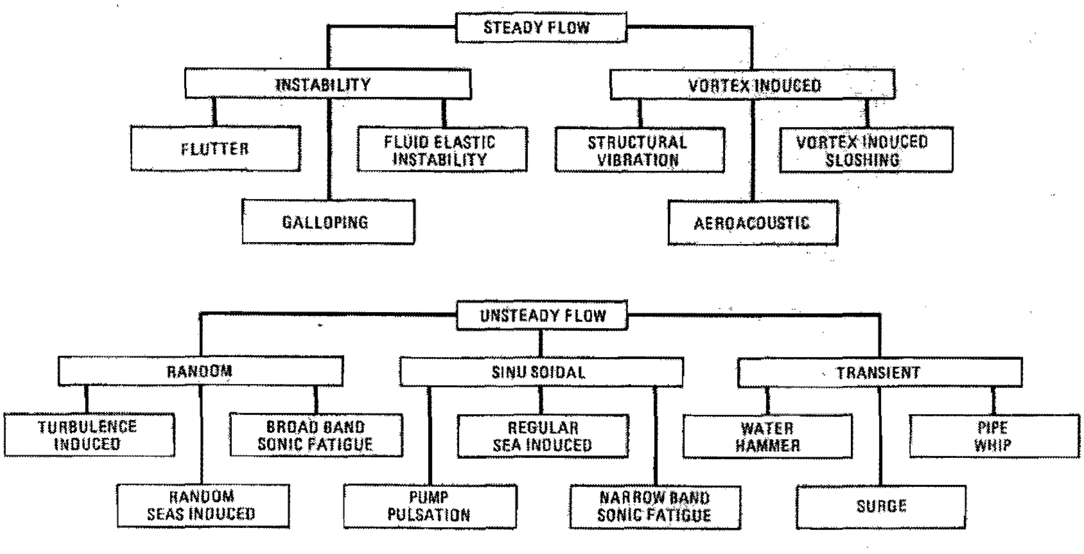


Fig. A A classification of flow-induced vibrations.

FIGURE 1 (Blevins 1990, p. 2)

2.3. Report outline

FSI and FIV are dealt with in the Sections 3 and 4, respectively. All possible sources of excitation of liquid-filled pipe systems are summarised in Section 5. The most important of these probably is the vibrating bend, which therefore deserves special attention in Section 6. What codes and standards say about FSI and FIV is reported in Section 7. Section 8 surveys current research activity and Section 9 addresses future work. Section 10 concludes the report.

3. Fluid-Structure Interaction (FSI)

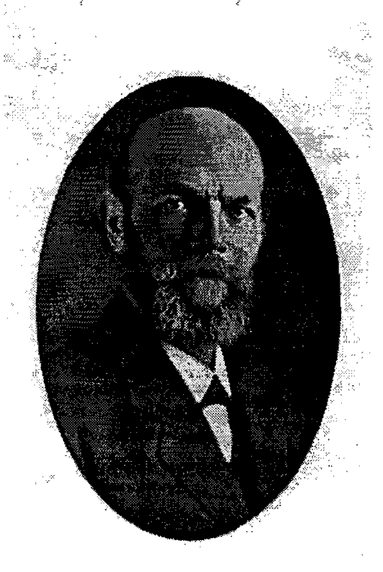
The papers by Tijsseling (1996) and Wiggert and Tijsseling (2001) give a fairly complete picture of what has been achieved in the area of fluid-structure interaction in liquid-filled pipe systems. A short summary and some additions are given here. Section 3.5 lists overlooked and new contributions to the subject that were not mentioned in the above-mentioned papers.

3.1. History

The study of the propagation of waves in liquid-filled elastic tubes has a rich history. Two excellent historical reviews, in French, are those by Boulanger (1913) and Lambossy (1950, 1951). Both authors mention the work by Von Kries (1892), who, the author recently discovered, preceded (Von Kries 1883) Joukowsky (1898) in finding - and validating experimentally - the relation between pressure and velocity in waterhammer. In those days, to actually measure the transient event of waterhammer, one needed either long pipelines or low pressure wavespeeds. Joukowsky conducted experiments in the Moscow drinking water system and had long lengths, and Von Kries used rubber hoses with low wavespeeds.

Classical papers on FSI are those by Korteweg (1878), Lamb (1898) and Skalak (1956).

JOHANNES VON KRIESS



v. Kries

3.2. Coupling mechanisms

FSI in liquid-filled pipe systems is caused by four interaction mechanisms: friction coupling, Poisson coupling, junction coupling and Bourdon coupling. *Friction coupling* is the mutual friction between the liquid and the axially vibrating pipe wall. *Poisson coupling* relates pressures in the liquid to axial stresses in the pipes through the radial displacements of the pipe walls. While friction and Poisson coupling act along the entire pipe, the more important *junction coupling* acts wherever there is a branch or a change in area or flow direction. For example, the vibration of a short unrestrained elbow generates pressure variations in the liquid, which in turn excites the elbow and other junctions in the system. *Bourdon coupling* occurs in curved fluid-filled tubes of (slightly) non-circular cross-section. Internal fluid pressure unbends the tube and externally imposed bending changes the fluid flow. The mechanism, used in Bourdon gauges to measure liquid pressures, is probably best known from unrolling paper party tooters.

Dimensional or multi-scale analysis will show that friction (coupling) is unimportant in transient events. The Poisson coupling terms generally are small, but their effect may accumulate in time. Poisson coupling is of importance for modes of vibration that are predominantly in the axial direction of the individual pipes. Junction coupling is important when the excitation is fast enough to generate large unbalanced pressure forces in the piping system.

Pipe systems vibrate in axial, radial, lateral and torsional directions. In the axial vibration pipe and fluid interact through Poisson and friction coupling. In the lateral vibration the contained liquid acts as added mass. Torsional vibration is not affected by the liquid. Unrestrained pipe junctions couple fluid and pipes, and they let the axial, lateral and torsional vibrations interact.

Chen (1987, Section 4.3) showed that two modes of <u>axial</u> wave propagation are non-evanescent at low frequencies. One mode corresponds to pressure waves with large radial pipe expansions, the other to axial stress waves with small radial pipe contractions. The two modes, corresponding to waterhammer and precursor waves, are non-dispersive at low frequencies. The constant phase velocities (wavespeeds) are defined by Chen's equation 4.15, which takes into account the influence of wall thickness; i.e. terms of the order of e/R are <u>not</u> neglected with respect to 1.

Chen (1987, Section 4.3) showed that only one mode of <u>lateral</u> wave propagation exists at low frequencies. This mode is highly dispersive. Chen followed Reissner (1955) in defining the dispersion equation 4.19, which includes both the pipe wall thickness and a frequency-dependent added mass coefficient. Figure 2, from (Chen 1987, Section 4.3, Fig. 4.5), shows that the added mass coefficient, C_m, given in his equation 4.14, is close to one at low frequencies, $\alpha = 2\pi R/\lambda$.

Figure 3, from (Au-Yang 2001, Section 2.10, Fig. 2.16), shows the influence of pipe wall thickness. In the low frequency range, due to the fluid loading, the frequency of lateral vibration is lowered in proportion to $1 / \sqrt{1 + \frac{\rho_s}{\rho_f} \frac{R}{e} C_m}$, where C_m is about 1.

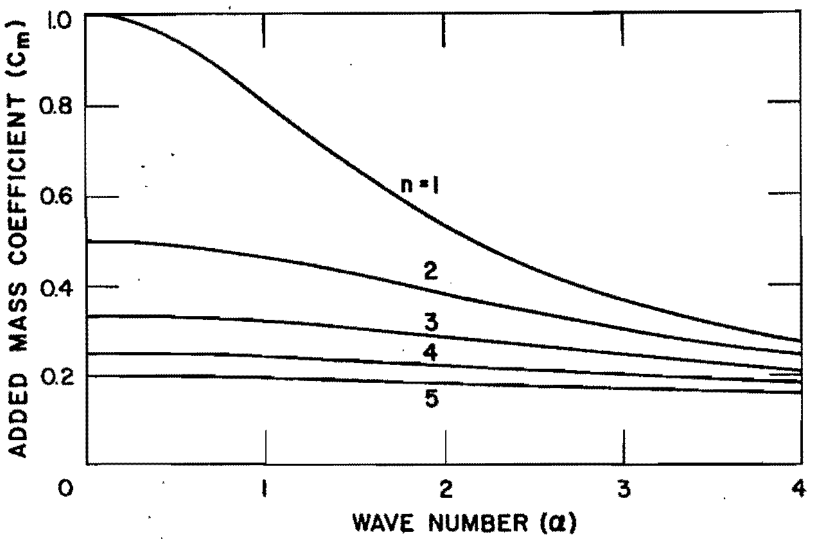


FIGURE 4.5. Values of the added mass coefficient for n = 1 to 5 (Chen and Rosenberg 1974)

FIGURE 2 (Chen 1987, Section 4.3, Fig. 4.5)

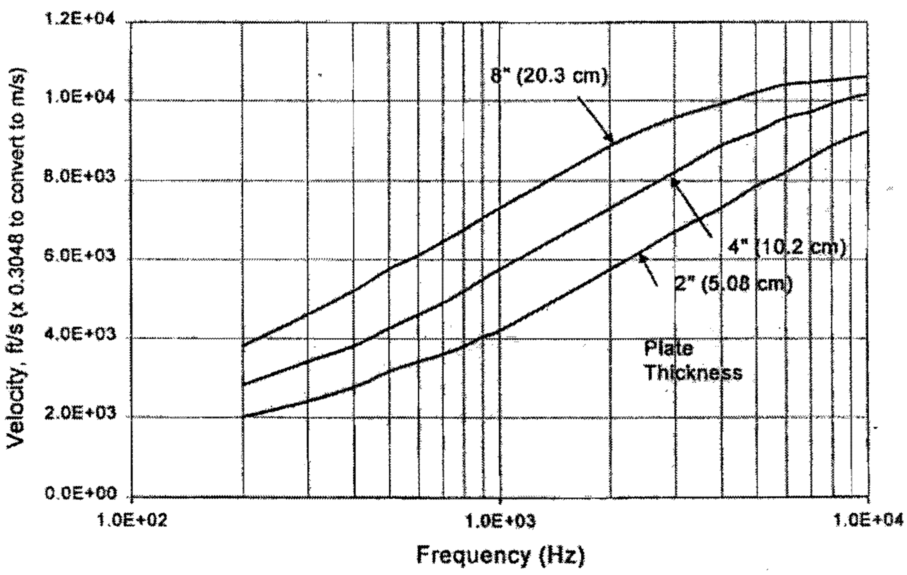


Figure 2.16 Speed of bending waves as a function of pipe wall thickness and frequencies, based on data from Mayo, et al. (Copyright © 1988, EPRI report NP-5743, Electric Power Research Institute, reproduced with permission.)

FIGURE 3 (Au-Yang 2001, Section 2.10, Fig. 2.16)

3.3. Importance of FSI

The main effects of fluid-structure interaction are problem dependent. When compared to predictions of conventional waterhammer and uncoupled analyses, predictions including fluid-structure interaction may lead to: higher or lower extreme pressures and stresses, changes in the natural frequencies of the system, and more damping and dispersion in the pressure and stress histories.

The classical theory of waterhammer predicts a square-wave pressure history (at the valve, friction neglected) in a reservoir-pipeline-valve system subjected to sudden valve closure (Chaudhry 1987, p. 14; Wylie and Streeter 1993, p. 50, p. 65). It has been shown by Tijsseling and Heinsbroek (1999) that this square wave is distorted in flexibly supported pipelines. Moving pipe (b)ends may (1) introduce higher-frequency pressure oscillations, (2) make square waves "triangular" and consequently wavefronts less steep, (3) change the system's main frequency and (4) invalidate application of Joukowsky's formula (Eq. 1 herein).

Fully fluid-structure coupled models are required to analyse flexible pipelines subjected to rapid excitation, in particular to assess accurately the anchor forces. FSI models give a more accurate prediction of pressure, stress and displacement amplitudes, of natural and resonance frequencies, of damping, and of anchor and support forces.

Amplitudes

Many investigators have shown that the pressure amplitudes during a waterhammer event in a pipeline may exceed Joukowsky's value (Eq. 1 herein) as a result of pipe flexibility. For example, figure 7c in (Tijsseling 1996) shows a Joukowsky overshoot of 100%. In general, the flexibility of pipe systems causes pressures higher than Joukowsky in the beginning of a transient event. Later on, in systems with movable bends, the pressures will be lower than Joukowsky, because energy has then been transferred from the fluid pulsation to lateral pipe vibration.

Frequencies

The <u>structural</u> natural frequencies of a fluid-filled pipe system are usually obtained from an analysis in which the fluid is dead mass. The <u>fluid</u> natural frequencies, if required, are then obtained from an analysis in which the pipe system is rigid. This approach will fail in compliant systems with structural and fluid frequencies close to each other. Fluid-structure interaction by junction coupling - e.g. movable (b)ends - will separate coinciding fluid and structural frequencies (as found without considering FSI). This is nicely shown in the Figures 4 and 5 copied from

Diesselhorst *et al* (2000) and Hara (1988), respectively. Figure 4 corresponds to a coupled fluid-structure oscillator and Figure 5 to a Z-shaped pipeline. Moussou *et al* (2000) gave a most detailed study of such a Z-system and they came to the conclusion that, due to FSI, fluid and structural frequencies can<u>not</u> coincide. Thus, FSI prevents a certain type of resonance behaviour as seen in the uncoupled calculations for a single pipe with a vibrating closed end (Müller 1989, Fig. 7; Wiggert and Tijsseling 2001, Fig. 2b) and for a pipeline with six movable bends (Kellner and Schönfelder 1982, Fig. 6), see Figure 6.

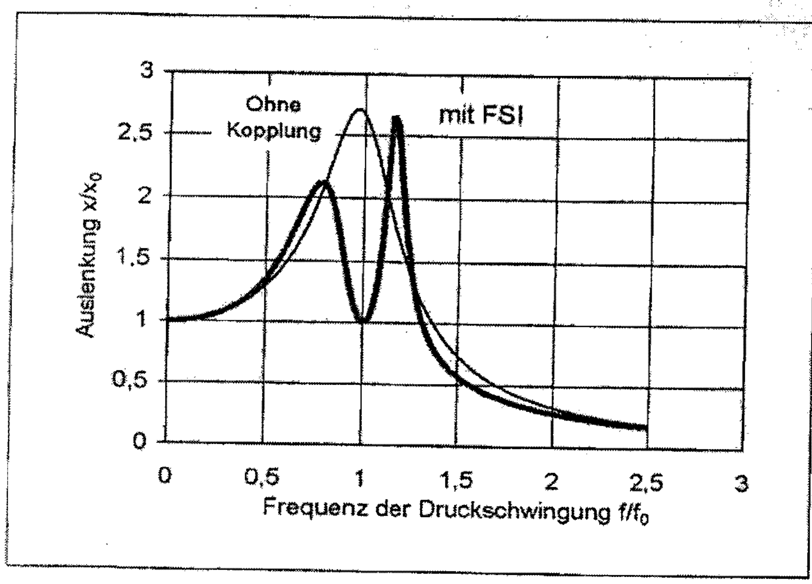
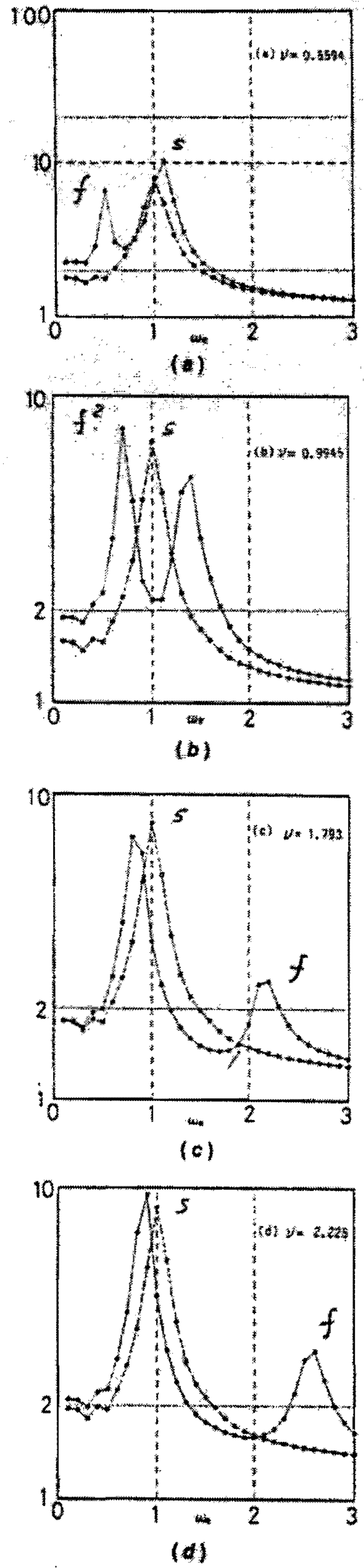


Bild 4:
Resonanz von Fluid- und Strukturschwingung; Frequenzverschiebung durch Fluid-Struktur-Wechselwirkung

Fig. 4:
Deflection of piping in case of resonance of structural and fluid oscillation; shift of frequency applying fluid-structure interaction

FIGURE 4 (Diesselhorst *et al* 2000, Fig. 4)



s ≈ f
but
separated
because
FSI

Fig. 7 Comparison of the seismic acceleration response spectrum between the dead-mass model (○) and the pressure-wave-piping-interaction model (●)—ν = (a) 0.5594, (b) 0.9945, (c) 1.793, (d) 2.225

FIGURE 5 (Hara 1988, Fig 7)

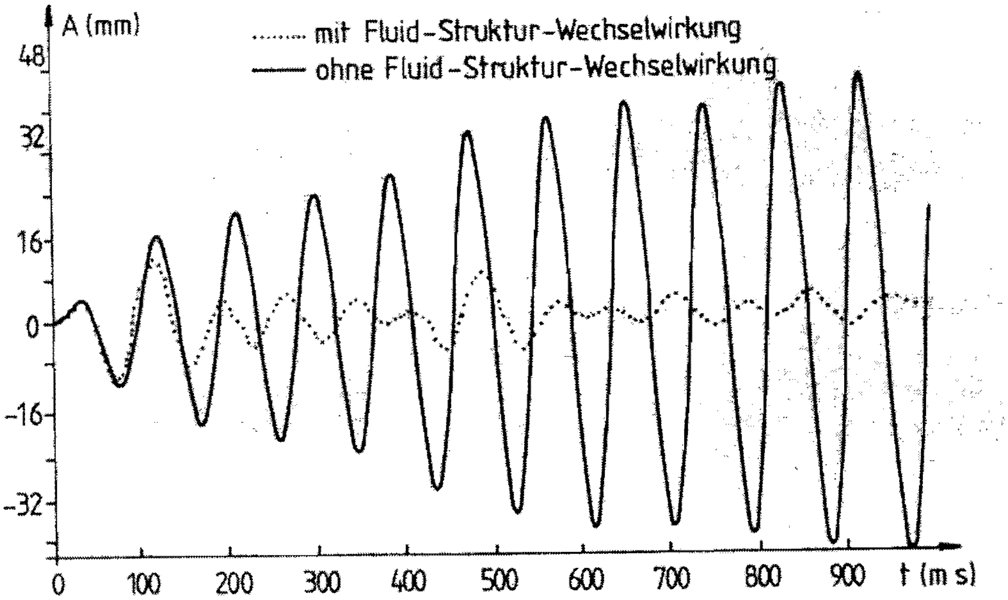


Bild 6: y-Auslenkung des ersten Krümmers
Fig. 6: y-Deflection of the first bend

FIGURE 6 (Kellner and Schönfelder 1982, Fig. 6)

FSI prevents this type of resonance behaviour, because it prohibits coinciding fluid and structural frequencies. Uncoupled calculations predict resonance, where fully-coupled calculations do not. In this way the results of uncoupled calculations are much too conservative.

Damping

This subject has been reviewed by Leslie and Tijsseling (1999). See also Blevins (1990, Chapter 8).

FSI damping (from Leslie and Tijsseling 1999)

Hydraulic engineers regard pipes as rigid containers resisting their liquid flow through wall friction. However, in waterhammer analyses the hoop elasticity of pipes is taken into account to reduce the pressure wave speed. Pipe motion and associated inertia effects are normally ignored. In doing so, they may not be aware that energy is transferred from liquid pulsation to pipe vibration, and observe more liquid damping than anticipated.

Structural engineers regard liquids as rigid columns pressurising their elastic pipes. However, in vibration analyses the mass of liquids is taken into account to reduce the structural frequencies. Liquid elasticity and associated waterhammer effects are normally ignored. In doing so, they may not be aware that energy is transferred from pipe vibration to liquid pulsation, and observe more structural damping than anticipated.

FSI engineers consider elastic liquids contained within elastic pipes and they take into account the mutual energy transfer. Liquid energy cannot only be lost to the vibrating pipe system, but it can also be gained from it.

An illustrative example is provided by a vibrating semi-infinite liquid-filled pipe with a closed end that is massless and free to move. The vibrating end will generate pressure waves in the liquid according to the formula of *Joukowsky*:

$$P = \rho_f \, c_f \, V \qquad \textit{FLUID} \tag{1}$$

The following end conditions relate the pressure pulsation to the structural vibration:

$$V = \dot{u} \tag{2}$$

FSI

$$F = A_f \, P = A_s \, \sigma \tag{3}$$

Combining the equations (1), (2) and (3) gives:

$$c\,\dot{u} = F \quad \text{with} \quad c = \rho_f \, c_f \, A_f \qquad \textit{STRUCTURE} \tag{4}$$

In this example, where the pipe vibration is imposed, the moving end is an exciter for the liquid and a dash-pot (damper) for the pipe. On the other hand, when the liquid pulsation is imposed, the moving end is an exciter for the pipe and a dash-pot (damper) for the liquid. In general, both effects occur at the same time, something fully accounted for in FSI analyses.

Coriolis damping

In systems with high flow velocities, Coriolis forces in the fluid have a dampening effect on flexural pipe vibrations. This phenomenon is fully explained by Païdoussis (1998), who also gave the governing equations including Kelvin-Voigt internal and viscous external damping.

This subject is reviewed in Section 7.2 of Wiggert and Tijsseling (2001).

Frictional damping

In systems with relative motion between fluid and structure, the friction and drag forces are proportional to $|V-\dot{u}|\,(V-\dot{u})$ instead of $|V|\,V$. For small structural velocities, $\dot{u} \ll V$, which is valid for example for small pulsations on top of a steady flow, Blevins (1990, Section 6.2.2) linearised this term to $|V|\,V - |V|\,\dot{u}$. The first term is the steady-state friction force, which is an external force for the structure. In a harmonic analysis, because of the V^2 non-linearity, this term gives rise to higher harmonics. The second term is a time-dependent damping force for the structure, which may not be neglected because it is of importance for the dynamic response near resonance conditions.

Often the damping force will not follow the ideal of the viscous damper (which is proportional to $\dot{u}$). It is possible to define *equivalent viscous damping* as the viscous damping that expends the same energy per cycle of vibration as the actual damping force. Blevins (1990, Section 8.1, Table 8.1) gives equivalent damping factors for Coulomb (friction), hysteretic (displacement dependent) and velocity power-law damping models.

3.4. Practical applications

The practical applications / importance of FSI can be captured by the following keywords:

Nuclear industry (safety)

Chemical industry (environment)

Aerospace industry (light structures)

Marine industry (noise)

Hydropower stations (stability)

Petroleum industry (long pipelines)

Earthquake engineering (Japan)

Dyke crossings (The Netherlands)

Biomedical engineering (haemodynamics)

Laboratory experiments (accuracy)

Fatigue

COSTS

3.5. Recent work

The three review papers by Wiggert (1996), Tijsseling (1996), and Wiggert and Tijsseling (2001) give a fairly complete overview of FSI research and application. There is no need to repeat what is mentioned in these papers here. What is listed below can be regarded as an addition to and extension of the above review papers through overlooked and new literature.

Reseachers not mentioned in the three FSI reviews

Altstadt *et al* (2000);
Chun and Yu (2000);
Formaggia *et al* (2001);
Giot *et al* (2001);
Hansson and Sandberg (2001);
Kollmann *et al* (1998);
Kollmann and Swidersky (2000a);
Kollmann and Swidersky (2000b);
Liu and Ohashi (1988);
Reissner (1955);
Repp (1998);
Schumann (1979);
Shu *et al* (2001);
Stoessel *et al* (1988);
Thompson *et al* (1989);
Uspuras *et al* (2001);
Vatkova *et al* (1999);
Yamaguchi *et al* (1982);
Yamaguchi and Kondo (1989)

Reseachers mentioned in the three FSI reviews, except for the papers:

Bahrar *et al* (1998b); this is the paper of Bahrar *et al* (1998a) with viscoelasticity included;
Bergant and Tijsseling (2001);
Chen and Rosenberg (1974);
Charley *et al* (1998);
Charley and Carta (2001);
Ichchou *et al* (1998);
Ichchou *et al* (1999);

Jiao *et al* (1999);
Moussou and Boyelle (1999);
Müller (1989);
Munjal and Thawani (1997);
Ohayon (1986);
Tijsseling and Vaugrante (2001);
Zhang *et al* (2000a);
Zhang *et al* (2000b)

4. FLOW-INDUCED VIBRATION (FIV)

4.1. Introduction

FIV has become a broad area of research. An excellent state-of-the-art paper has been written by six leading experts in the field: Weaver, Ziada, Au-Yang, Chen, Païdoussis and Pettigrew (Weaver *et al* 2000). This paper is the starting point for the present review. It is summarised in Section 4.2 with respect to liquid-filled pipes. What the standard textbooks, and some relevant papers, say about fluid-conveying pipes is reported in Section 4.3. Section 5 summarises all FIV and FSI excitation sources.

4.2. Review paper by Weaver *et al* (2000)

This section discusses five general excitation mechanisms that may occur in internal pipe flow.

4.2.1. Turbulence

Nearly all industrial pipe flows are turbulent. Despite DNS, LES and a whole family of k-ε models that can be used to compute turbulent flow, preference is given to a practical approach in terms of power spectral density functions (PSD). Some empirical PSD formulae are given in section 2.1 (of Weaver *et al* 2000). The PSD are input to a probabilistic structural analysis, which calculates pipe responses. Au-Yang (2001, Chapter 8) describes this method in detail. Although turbulence-induced vibration or turbulence/acoustics-induced vibrations are yet an issue in nuclear piping systems, turbulence is not further pursued here, because the next type of excitation, vorticity shedding, is considered to be much more important.

4.2.2. Vorticity shedding

Where turbulence is a random excitation, vorticity shedding is an almost periodic excitation whose frequency varies linearly with the flow velocity. The generated pressure and velocity fluctuations appear in the pressure and turbulence spectra as a discrete frequency excitation. The excitation level is generally weak, unless it is coupled with a feedback mechanism that may drastically enhance the flow oscillations such that excessive vibration and/or acute noise problems are likely to occur. Feedback can be provided by structural vibrations (for example pipe bends), by acoustic resonators (for example a closed side branch excited by grazing flow in the main) and by flow impingement on a downstream subject (this may occur, for example, in corrugated pipes). All three feedback mechanisms can coexist.

Vorticity-induced vibration. As the vortex shedding frequency, determined by the Strouhal number $St = f\,D\,/\,V$, approaches a natural frequency of the structure (piping, valve body, pump blade, etc) resonant oscillations may develop. These structural vibrations strongly enhance the vortex shedding and they lock the shedding frequency to the structural frequency. This phenomenon is called "lock-in" or "synchronisation". Lock-in can occur over a flow velocity range of about ±20% of that to produce vortex shedding resonance. Lightly damped structures in heavy fluid (liquid) flows are most susceptible to damaging vibrations caused by vortex shedding. A first step in the identification of FIV problems is the comparison of structural frequencies with Strouhal numbers. Blevins (1990, Section 3.2) gives Strouhal numbers for various bodies in external flow. He also gives a "universal" Strouhal number based on the width (D) between the flow separation points, which is valid for any bluff section, regardless of section geometry, and over a broad range of Reynolds numbers. Its value is approximately 0.2.

Vorticity-induced acoustic resonance. Howe (1975) derived the following formula for the power, P, of vorticity-induced sound:

$$P = -\rho_f \iiint \omega \cdot (\mathbf{V} \times \mathbf{v})\, \mathrm{d}\forall \tag{5}$$

where ω is the vorticity, $\mathbf{V}$ the flow velocity, $\mathbf{v}$ the superposed particle velocity of the sound field, and $\forall$ the volume containing the vorticity field. Formula (5) is valid in cases with high Reynolds and low Mach numbers.

Fluid flow in pipes containing multiple closed side-branches in close proximity can generate very powerful acoustic resonances. The side branches decouple themselves from the rest of the piping system and form an acoustic resonator with negligible radiation losses. The strongest resonance occurs when the excitation at the branch mouth is a shear layer with just one vortex. Ziada (1993) gave a Strouhal number chart to estimate the critical velocity in the main pipe at the onset of resonance, which includes the effects of diameter ratio, geometry of the side branches and location of upstream elbows. He discussed counter-measures to reduce the pulsation amplitudes and he remarked that measured source terms can be incorporated relatively easily into computer codes of pipe acoustics.

Control valves have the inherent feature that a high-speed jet issues from the vena-contracta. Another common feature is the smallness of the valve body which, in many cases, provides a downstream impingement boundary. Flow impingement in valves can cause severe noise problems and/or acoustic fatigue failures, especially when acoustic modes are excited (Ziada *et al* 1989).

Ziada *et al* (1999, 2001) and Schafbuch *et al* (1997) showed that the separation of low-speed flows in entrance ports of control valves and upstream elbows can cause acoustic resonance in the connected piping systems.

Broadband acoustic power, for example generated by high-pressure reduction valves, may cause acoustic fatigue damage to the downstream piping system. Graf *et al* (1997) proposed a systematic approach to predict and reduce the broadband noise level produced by gas throttling devices. Eisinger (1997) gave a design chart for the prevention of acoustic fatigue failure.

4.2.3. Leakage flow-induced vibration

Leakage flow is an important excitation mechanism that is not directly related to vorticity shedding and/or turbulence. It occurs, for example, in liquid control valves. When these valves are nearly closed, the local high-velocity leakage flow may cause an instability with coupled flow and valve-body oscillations. These oscillations may interact with acoustic modes in the piping and with downstream vortex streets. An upstream nearly-closed or leaking valve is generally less stable than a downstream one (Miller 1970; Mulcahy 1983, 1988).

Weaver *et al* (2000) address this subject very briefly.

Naudascher and Rockwell (1994, Section 7.4) give a nice introduction. Figure 7 is from their book. The left figure shows the system, the right one the corresponding stability diagram. The symbols in the stability diagram are explained in the sketch of the system, with γ the specific weight of the liquid.

For further information, see Blevins (1990, Section 10.4.2), Chen (1987, Section 6.10) and Au-Yang (2001, Sections 10.5 and 10.6).

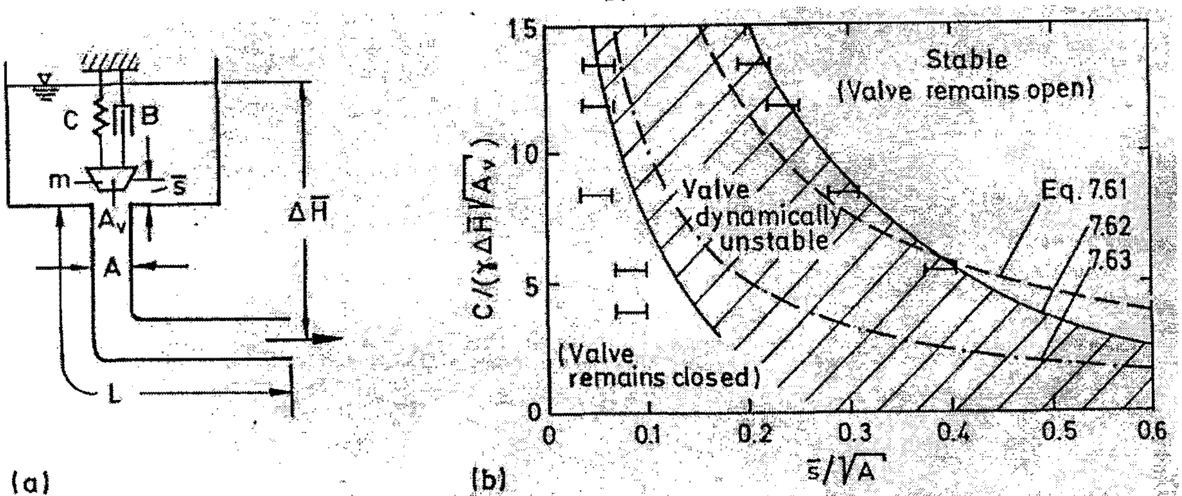


Figure 7.46. (a) Schematic of plug-valve system. (b) Stability diagram of plug valve for large fluid inertia; horizontal lines = experimental stability threshold $L/\sqrt{A_v} = 221$ (after D'Netto & Weaver, 1987).

FIGURE 7 Naudascher and Rockwell (1994, Section 7.4, Fig. 7.46)

4.2.4. Axial-flow-induced vibration

Axial-flow-induced vibration in fluid-conveying pipes refers to lateral pipe vibration caused by flow-induced centrifugal and Coriolis forces. Païdoussis (1998) gives an encyclopaedic treatise of the subject, which focuses on stability analysis. Wiggert and Tijsseling (2001, Section 7.2) summarise literature combining Païdoussis-type of FIV work with Wiggert-type of FSI work.

The basic equations describing the phenomenon as a Bernoulli-Euler beam with internal axial flow, and axial-lateral coupling mechanisms, are given by Païdoussis (1998, Section 3.3.2).

In industrial pipe systems, axial-flow-induced vibration is predicted to occur at unrealistically high flow velocities. It is of less practical importance and therefore not surveyed in this report.

4.2.5. Fluidelastic instability

Fluidelastic instability does not refer to the elasticity of the fluid, but to the mutual interaction of fluid dynamic forces and elastic structural displacements. Instability may develop if the structural motion produces a fluid force in the same direction. In the absence of the structural motion, the flow is assumed to be without periodicity, so that the behaviour is different from vorticity-induced vibration. The instability is self-excited and occurs at and above a certain critical velocity. It mostly applies to bodies, in particular tube arrays, in external flow.

4.3. The textbooks

What the standard textbooks on FIV say about internal pipe flow is shortly summarised in this section.

Chen (1987)

Some general principles for a better understanding of FIV are given in the Sections 2.1 and 2.14. Fluid viscosity, fluid compressibility and a free surface may cause damping of a vibrating structure surrounded by fluid. To be more precise: 1) in an incompressible inviscid fluid without surface waves, there is no damping and a positive added mass, 2) in a compressible inviscid fluid in a confined region, there is no damping, because energy cannot radiate away, 3) in a compressible inviscid fluid in an unconfined region, there is radiation damping, and 4) in compressible fluids, there are acoustic modes, so that the added mass may be positive or negative.

Chapter 4 describes circular cylindrical shells containing fluid. Shell theory for the pipe is combined with potential flow theory for the fluid through interface conditions at the inner pipe wall. A harmonic analysis of infinitely long pipes gives phase velocities, modes of vibration and amplitude ratios. Waterhammer and precursor waves are discussed on the pages 118-120, and bending waves on the pages 120-122.

Chapter 5 describes pipes conveying fluid. This refers to what is called Païdoussis-type of FIV work in the present Section 4.2.4. Beam theory for the pipe, combined with plug flow for the fluid, is used in stability analyses. Section 5.3.3.3 explains the role of the Coriolis force, which is a damping mechanism in non-conservative systems at low flow velocities. Section 5.6 mentions waterhammer.

Blevins (1990)

Chapter 10 describes vibrations of a pipe containing a fluid flow. The introduction on page 384 mentions waterhammer, valve chatter, pipe whip and leakage flow.

Section 10.1.1 on instability of fluid-conveying pipes is the Païdoussis-type of FIV work (see present Section 4.2.4). Equation (10-27) gives a good estimate of the influence of a steady flow on the frequency, ω_1, of the first natural mode of lateral vibration of a simply supported pipe:

$$\frac{\omega_1}{\omega_N} \approx \sqrt{1-\left(\frac{V}{V_c}\right)^2} \tag{6}$$

where V is the flow velocity, V_c is the critical velocity at which the pipe buckles, and ω_N is the natural frequency for $V = 0$. A simply-supported pipe will *buckle* because of the centrifugal force, at $\omega_1 = 0$, but a cantilever pipe will *flutter* because of the Coriolis force, at $\omega_1 > 0$. These are different kinds of instabilities. Table 10-1 gives the stability behaviour of differently supported pipes.

Section 10.3 describes *pipe whip*, which is defined as the dynamic response of a pipe to an instantaneous rupture. Pipe rupture will not only generate severe pressure waves (waterhammer) in the liquid, but also a significant lateral force acting on the pipe. If the pipe is part of a high-pressure system, fluid will blow down through the rupture into the air. The rush of fluid and the unrestrained fluid pressure place an impulsive reaction on the pipe, which can cause the pipe to whip about and threaten personnel and structures. An estimate of the initial lateral (transverse) force at the rupture is $P\,A$, where P is the pressure and A the size of the rupture.

Section 10.4.1 describes pipe acoustical forcing, which can be seen as waterhammer in the frequency domain, with turbulence excitation. Fluid flow through valves, bends, and orifices generates turbulence within the flow and radiates acoustical energy upstream and downstream (Reethof 1978; Blake 1986). See for example Figure 8. Standing acoustic waves form in the fluid and these exert forces on bends and area changes (junction coupling). If the acoustical source possesses sufficient energy, resonance may cause excessive pipe motion and fatigue.

Chadha *et al* (1980) found that turbulence generated by valves is a broad-band acoustic source with a maximum frequency of about $0.05V/D_v$, where V is the downstream velocity and D_v is the diameter of the valve throat. The induced pressure fluctuations are about 1%-2% of the steady-state pressure difference across the valve. The components of the broad-band pressure at, or near, the system's natural frequencies are dominant. The fluid natural frequencies can be estimated as $c_f/4L$ (and odd higher harmonics) for an open-closed system and as $c_f/2L$ (and higher harmonics) for an open-open or closed-closed system.

Blevins (1990, Eq. 10-58) gives a formula for the acoustic (PA) and convective ($\rho_f A\ V^2$) forces on a fixed right-angle bend. The acoustic forces are at the natural frequencies of the fluid, but the convective forces may have higher harmonic components because of V^2.

Blevins mentions that he has observed severe resonance in the case of coinciding fluid and structural frequencies, but he does not provide sufficient information on the system concerned, possibly steam (gas) piping.

Section 10.4.2 deals with leakage-flow-induced vibration (see Section 4.2.4 herein).

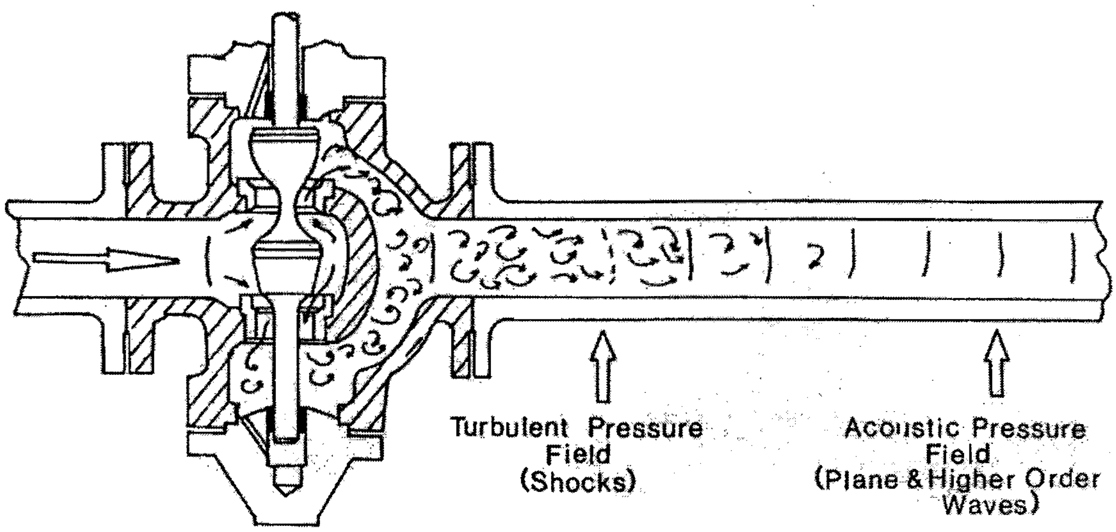


Figure 1 Schematic representation of valve noise generation and propagation.

FIGURE 8 (Reethof 1978, Fig. 1)

Naudascher and Rockwell (1994)

In their introduction, Naudascher and Rockwell give a strategy to identify possible flow-induced vibrations in a system. First, search for 1) all body oscillators, 2) all fluid oscillators, 3) all sources of extraneously induced excitation, 4) all sources of instability-induced excitation, and 5) all sources of movement-induced vibration. Second, assess all possible combinations of structural and fluid oscillations arising from (1) and (2) in conjunction with the excitations (3), (4) and (5). The coincidence of natural frequencies makes the combination of body and fluid oscillators dangerous, although in liquid flows FSI tends to separate coinciding fluid and structural frequencies. Estimating these natural frequencies, and finding the dominant excitation frequencies, is an integral part of the identification process.

A useful aid in preliminary investigations of danger spots and dangerous operating conditions of a system is a global or lumped-parameter analysis as suggested by Naudascher and Rockwell (1980) and as performed by, e.g., Wood (1968, 1969), Schumann (1979), Erath *et al* (1998), and Moussou *et al* (2000).

Section 7.4 gives a good introduction to leakage-flow-induced vibration. Press-open and press-shut devices are devices controlling the flow through small openings such that the fluid force tends to press them either open or shut. One should take care with these devices, because, if they are lightly damped, they may vibrate due to leakage-flow pulsations. More specific: valves controlling the outflow from a Helmholtz resonator of negligible fluid inertia are susceptible to movement-induced excitation if they act as press-open devices. And: devices controlling flow through small openings are susceptible to movement-induced excitation if they are press-shut and exposed to large fluid-inertia effects from the adjacent flow passage.

Chapter 8 describes fluid oscillators. The Sections 8.2 and 8.4 consider classical waterhammer theory in the frequency domain (Wylie and Streeter 1993). Section 8.5.2 describes various filters to attenuate fluid oscillations. Some practical examples are given in Section 8.6.1.

Au-Yang (2001)

Section 10.4 gives a short and clear introduction to the stability of pipes containing flowing fluid: this is the Païdoussis-type of FIV work. Example 10.3 concerns a 18 m long, 1 m diameter, 58 mm thick, water-filled, steel pipe, for which the critical velocity, V_c, is about 500 m/s.

The Sections 10.5 and 10.6 treat leakage-flow-induced vibration. Figures 9(A)(B) show common taps. If the threads are worn, the spindle may experience violent vibration that can shake the attached water pipe (waterhammer). Figure 10 shows the acceleration signal obtained from a piston check valve of the type shown in Figure 9(C). The signal exhibits a leakage-flow-induced instability.

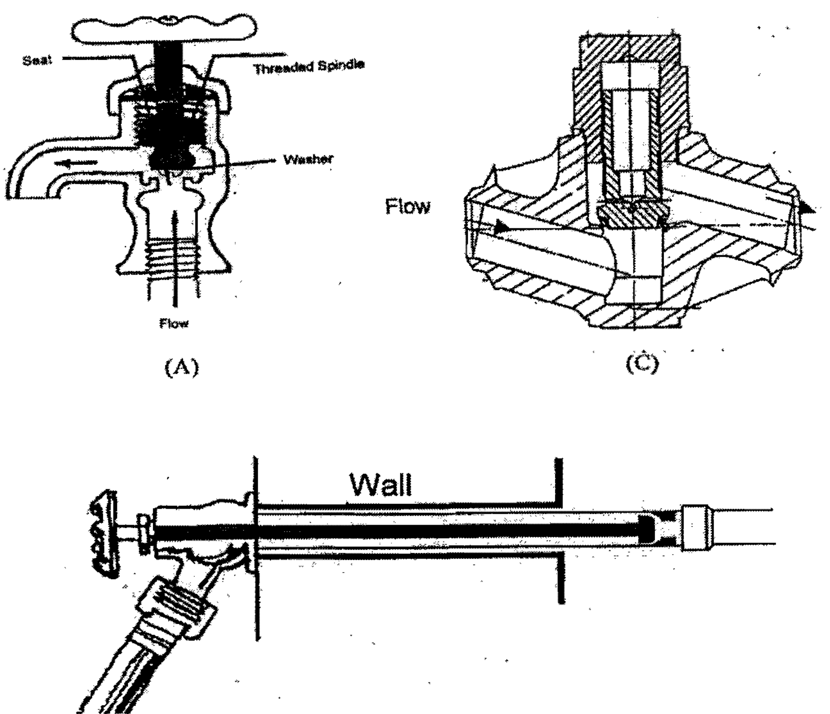


Figure 10.9 Components that have experience leakage-flow-induced instability
(A) Common faucet; (B) Frost-proof, outdoor faucet; (C) Lift check valve;
(to be continued)

FIGURE 9 (Au-Yang 2001, Section 10.6, Fig. 10.9)

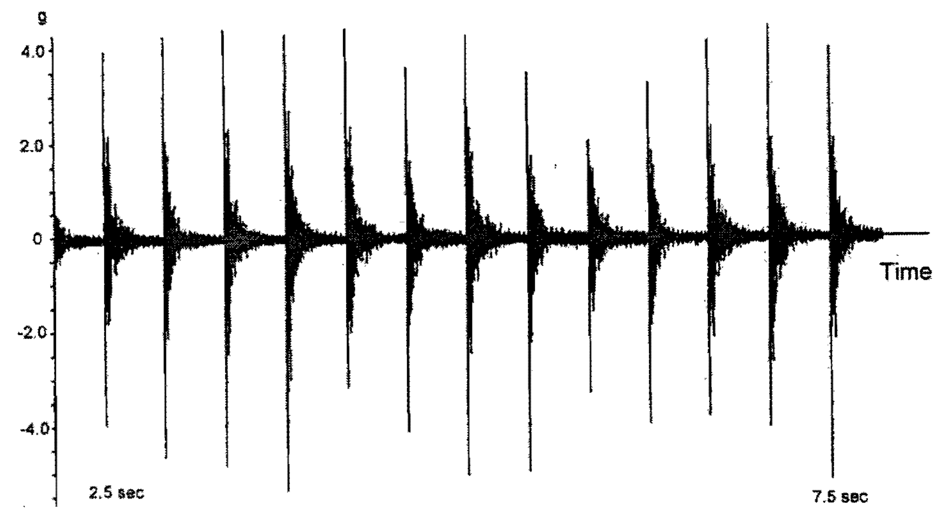


Figure 10.10 Vibration signature from an accelerometer mounted on a piston check valve showing leakage-flow-induced instability

FIGURE 10 (Au-Yang 2001, Section 10.6, Fig. 10.10)

Chapter 12 discusses acoustically induced vibration and noise. Acoustically induced vibration is probably one of the most common vibration problems in the power and process industries. Piping systems, vessels, valve cavities, heat exchanger internals, and many other components are potential resonators in which standing waves can form, while pumps, valves, elbows, orifices, etc, all have the potential to excite these standing waves. Once resonance occurs, the resulting sound intensity in most cases asks for remedial action. In some cases, acoustic excitation can cause rapid fatigue failure of piping welded points, valve internal parts and other components. The case studies in Section 12.10 are most interesting. Case Study 12.2, taken from Ziada *et al* (1999, 2001), concerns acoustic noise generated by a spherical elbow. Case Study 12.3, taken from Coffman and Bernstein (1979), concerns acoustic resonance in valve cavities. Standing waves in the cavities of relief valves, excited by the flow over them, turned out to be destructive for the valve components. The problem was solved by replacing a cylindrical nozzle by a conical one. Case Study 12.5, taken from Pastorel *et al* (2000), concerns valve-generated acoustic waves in a pipe. Acoustic fatigue caused cracks in the pipes downstream of the control valves.

Gibert (1988)

Chapter 19 gives an acoustic analysis of the low-frequency behaviour of fluid-filled pipe systems. All kinds of acoustic sources, mainly of the turbulent/acoustic nature, are described. Section 19.5

gives a practical example. This book is well known at EDF and not further summarised here.

Morand and Ohayon (1995)

Morand and Ohayon (1995) give a thorough mathematical and numerical treatment of general FSI/FIV problems without special reference to fluid-conveying pipe systems.

Païdoussis (1998)

Païdoussis's book is on axial-flow-induced vibration as discussed in Section 4.2.4 herein.

5. EXCITATION SOURCES: SUMMARY

Many excitation mechanisms exist. Here is an overview of what was found in the literature consulted. A distinction is made between exciters of the fluid and exciters of the piping. Of course, if FSI takes place in a significant manner, both fluid and piping will be excited. The excitation can be transient, periodic or random.

Types of excitation (from Wylie and Streeter 1993, pp. 309-313)

Resonance develops only when there is an exciter present at some point in the system. The form that the exciter takes can generally be placed into one of two categories. The first and most common is an explicit device that acts as a forcing function exciting the system at some frequency. The second type is one that responds to the system. The oscillations produced by this type are generally referred to as *self-excited*, as the implicit exciter is an integral part of the entire physical system.

5.1. Exciters for the fluid

Transient

valve action (closure / opening) (control valves, check valves, relief valves, etc)
pump action (shut off / start up)
turbine action (load rejection)
column separation (collapse)
steam-condensation (e.g. Chun and Yu 2000)
pipe break (generating severe waterhammer)

rupture-disk break (generating severe waterhammer)
blast / explosion
collision (with other pipe, with vehicle)
bend / junction motion
priming / start up / slug flow

Periodic

vortex-shedding (Section 4.2.2)
leakage-flow (Section 4.2.3)
check valve chatter / tapping (Figure 11)
check valve flutter (Figure 12)
surface wave in a reservoir near a pipe entrance (e.g. Naudascher and Rockwell 1994, pp. 274-276)
bend / junction motion (Figure 13(a))
pipe breathing (Figure 13(b))

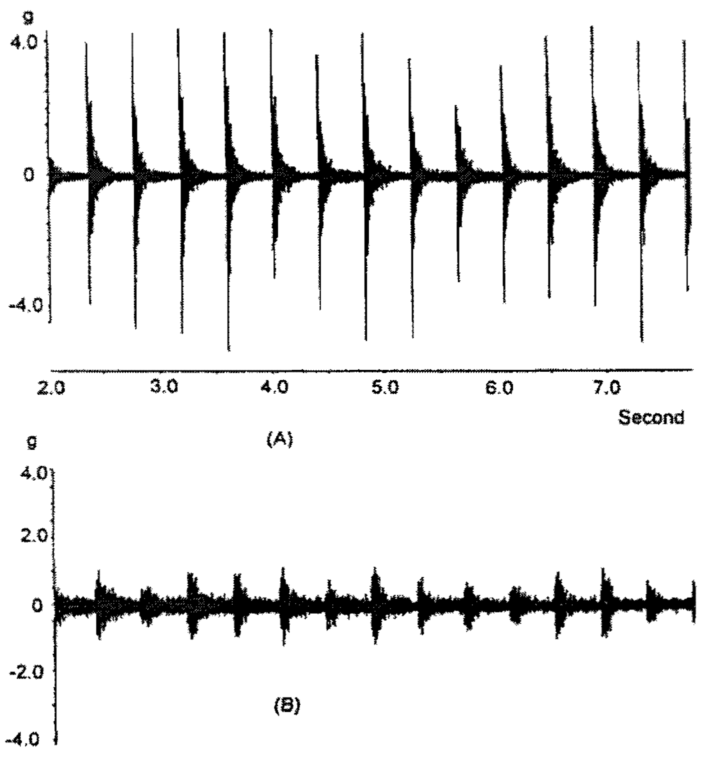


Figure 2.18 Acceleration time histories picked up by two accelerometers mounted on two piston lift check valves on two parallel pipe lines

FIGURE 11 (Au-Yang 2001, Section 2.10, Fig. 2.18)

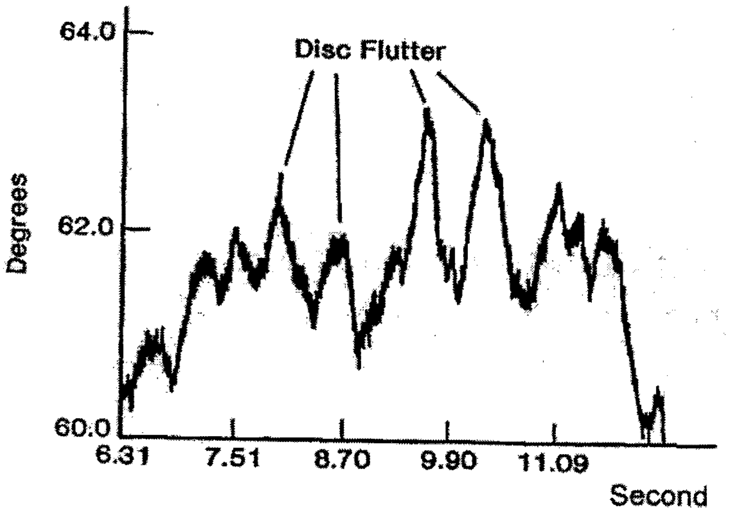


Figure 2.5 Flutter of a check valve disk measured by UT instrument (Au-Yang, 1993)

FIGURE 12 (Au-Yang 2001, Section 2.4, Fig. 2.5)

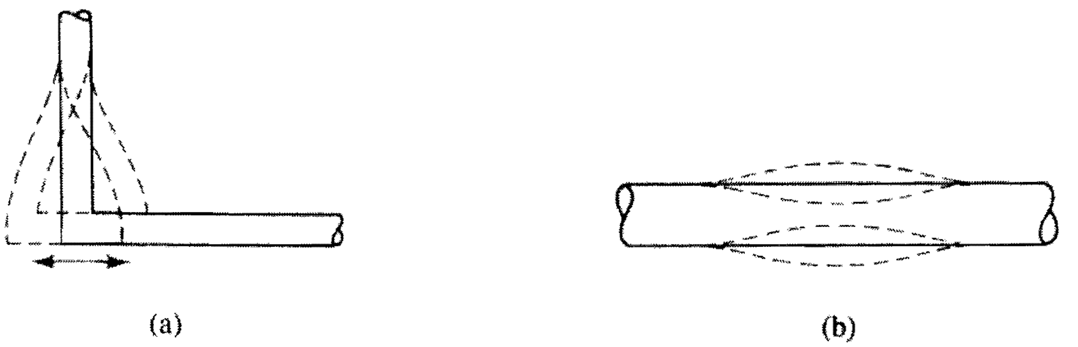


Figure 13-31 Capacitance elements.

FIGURE 13 (Wylie and Streeter 1993, Section 13-8, Fig. 13-31)

Random

turbulence (Section 4.2.1)
cavitation
two-phase flow

5.2. Exciters for the piping

Transient

waterhammer
column separation
earthquake
landslide
pipe whip (rupture of flexible pipe, where the rupture generates lateral forces that make the pipe heavily whip / flail / shake, see Blevins 1990, Section 10.3)
blast / explosion
collision (with other pipe, with vehicle)
bend / junction motion
priming / start up / slug flow

Periodic

machine vibration
axial-flow-induced vibration (Section 4.2.4)
lock-in (see Section 4.2.2)
bend / junction motion

Random

seismic motion
base motion
wind

6. EXCITATION SOURCES: VIBRATING BENDS AND BRANCHES

6.1. Introduction

Vibrating pipe junctions, in particular bends (elbows) and branches (tee-pieces), but also dead ends, are the most significant FSI mechanisms in liquid-filled pipe systems. Axially vibrating junctions generate pressure waves in the fluid, and pressure waves make unrestrained junctions vibrate. Section 6.2 attempts to list all physical experiments with respect to FSI in single-elbow pipe systems. Some aspects of mathematical and numerical modelling are discussed. Section 6.3 lists the few FSI experiments that have been performed in branched systems.

6.2. Bends

List in chronological order of experimental studies of FSI in one-elbow pipe systems:

Blade *et al* (1962);
Swaffield (1968-1969);
Davidson and Smith (1969);
Wood and Chao (1971);
Davidson and Samsury (1972);
A-Moneim and Chang (1979);
Kellner *et al* (1983);
Otwell (1984), Wiggert *et al* (1985);
Liu and Ohashi (1988);
Yamaguchi and Kondo (1989);
Tentarelli (1990), Brown and Tentarelli (2001);
Tijsseling (1993), Tijsseling *et al* (1996);
de Jong (1994, 2000);
Svingen (1996);
Repp (1998);
Jiao *et al* (1999);
Altstadt et al (2000);
Tijsseling AS and Vaugrante P (2001);
Caillaud et al (2001)

Note: The much-used experimental data of Swaffield (1968-1969) and Davidson and Smith (1969) are not valid, as will be explained below.

Many physical experiments have been performed in systems with elbows, ranging from Blade *et al* (1962) to Caillaud *et al* (2001). Nearly all of the experimental systems had at least one "fixed" end (support, anchor), for example the connection of the test pipe to a liquid supply (reservoir). "Fixed" stands for infinitely large impedance (zero mobility), something impossible in practice, especially during resonance. Some researchers, like Davidson and Samsury (1972) and de Jong (1994), measured the mobility of the pipe supports in their test systems, but it nevertheless is common practice not to measure or estimate the mobility of supports, but to neglect it on the simplifying assumption that the support is (looks) rigid. Consequently, many investigators have overlooked support mobility. The well known experiments by Swaffield (1968-1969) and Davidson and Smith (1969), the results of which have been used by many others, suffer from the ignored vibration of "fixed" points, as noted by Wilkinson (1980, p. 197) and Brown and Tentarelli (1988, p. 148), respectively. Svingen (1996, p. 76) reported "unintentional" valve motion. Care has to be taken in this respect, always. In contrast to the many other test rigs, the experimental apparatus employed by Tijsseling *et al* (1996) and Tijsseling and Vaugrante (2001) has no "fixed" points at all, it is structurally "free".

The fundamental acoustic frequency of a fluid column in a(n) (open) reservoir - pipeline - valve (closed) system is $c_f/4L_{\mathrm{eff}}$. The effective length, L_{eff}, given by Alster (1972, Eq. 40) as

$$\frac{L_{\mathrm{eff}}}{L} = \sqrt{1 + 0.48\left(\frac{D}{L}\right)} \approx 1 + 0.24\frac{D}{L} \tag{7}$$

accounts for added-mass effects at the open (reservoir) end. The semi-empirical formula (7) is valid for circular ducts of diameter D and length L. It differs from the classical value $1 + 4/(3\pi)D/L \approx 1 + 0.42\,D/L$ given in standard textbooks on acoustics (e.g. Rienstra and Hirschberg 1999, p. 105).

The correction for added mass has not been made in all of the studies listed above. Although most of the studies concern laboratory experiments in short pipes, the correction term was justifiably neglected, because the D/L ratios were small enough.

Aspects of mathematical and numerical modelling

Wiggert and Tijsseling (2001, Section 2) surveyed the mathematical models and numerical methods that are used to simulate FSI in liquid-filled pipe systems. EDF uses CIRCUS to perform computations in the frequency domain. CIRCUS is based on a spectral finite-element method. An alternative approach, based on transfer matrices, has been presented by Tijsseling and Vaugrante (1999).

One issue of concern is the influence of lumped (concentrated) masses, e.g. due to bends and branches, on pipe vibration. Lumped masses give rise to (infinitely) large accelerations when subjected to impact loads, e.g. instantaneous valve closures. In numerical calculations, if $\Delta t \to$ zero, then $m/\Delta t \to$ infinity (Tijsseling 1993, Section 4.6, Eq. 4.128), indicating the dominance of lumped inertia. Large narrow peaks appear in the solutions. A continuous representation of lumped masses (short bends, valves, etc) is not acceptable because their small L/D ratios obstruct the assumption of one-dimensionality. This particularly holds for analysis in the time domain. In the frequency domain, Tentarelli (1990, Section 7.4) has experimented with different elbow representations. He rejected the lumped-mass approach in favour of two short and stiff segments representing the elbow. The lengths of the segments were determined by the actual measured length of the elbow and the outer radii were selected such that their total mass equalled that of the elbow. In general it is difficult to find the mass and stiffness of an elbow, in particular the moments of inertia are difficult to assess.

A second issue, also addressed by Tentarelli (1990, Section 3.2.2), is numerical instability at higher frequencies (ill-conditioned matrices, round-off errors, numerical instabilities). This is a general problem in the frequency-domain solution of lateral beam (pipe) vibration. Tentarelli (1990, pp. 43-44) and De Jong (1994, p. 41, pp. 54-55) sub-divided pipes into shorter reaches to prevent numerical problems. Finnveden (1994, pp. 473-474) cured the numerical instability by choosing appropriate base and shape functions within the FEM. Charley *et al* (1998, Section 3.2) studied the matrix condition numbers in their transfer matrix model. For low-frequency flexural vibration acceptable solutions can be obtained without special measures.

6.3. Branches

List in chronological order of experimental studies of FSI in branched pipe systems:

Wood and Chao (1971);
Merkli P (1978) (air, no FSI, "analytical" solutions, definition of effective lengths);
Kellner *et al* (1983);
Tentarelli (1990), Brown and Tentarelli (2001);
Vardy *et al* (1996)

There have not been many physical experiments on FSI in branched systems, although the author has seen quite a few industrial problems in which T-pieces were involved. Some of these had to do with fatigue, others with combined mechanical-acoustical loading. As usual, these problems have not been documented for the public. However, it is well known that the connection points of small bore pipes connected to (vibrating) large bore pipes are points of potential failure (see ASME B31.3, paragraph 319.6, p. 41).

7. Codes and Standards

Leslie and Vardy (2001) inspected many codes and standards with respect to FSI, but they omitted the important ASME B31 codes. Pothof and McNulty (2001) included these codes in their review, which, however, was not exclusively dedicated to FSI. FSI was considered by Lemmens and Gresnigt (2001) in the framework of the Standards, Measurements and Testing (SMT) programme of the European Union.

There is no need to repeat the work done by the above authors. Instead, two recently revised codes will be discussed: the American ASME B31.1 (December 2001) and the Dutch NEN 3650 (August 2001). Furthermore, a few remarks on yield stress criteria.

7.1. **ASME B31.1** (December 2001)

This code prescribes minimum requirements for the design, materials, fabrication, erection, test, and inspection of power and auxiliary service piping systems for electric generation stations, industrial institutional plants, and central and district heating plants. To my best knowledge, this latest B31.1 code, does not mention FSI and/or FIV.

7.2. **NEN 3650** (Parts 1 and 2) (August 2001)

This standard (code) contains requirements for the complete life cycle of pipeline transportation systems, without regard to the material. It also gives reference to relevant European (EN) and international (ISO) standards. The standard has recently been updated and it has been open for comment until 1 December 2001. The comments are presently being used to improve the standard.

Part 1 is general, and Part 2 concerns steel pipelines. Appendix C.2.2 on pages 94-96 of Part 1 mentions three ways of calculating dynamic pressure variations: 1) with the Joukowsky formula, 2) with a classical waterhammer analysis, and 3) taking into account FSI. The description is very brief, and WL | Delft Hydraulics has commented to it.

7.3. **Stress criteria**

Standard yield stress criteria are those of Tresca, Von Mises (1913), Hill (1950) and Vegter (1991). According to Todhunter and Pearson (1960), Von Mises's material is actually a rediscovery of work by Levy (see Saint-Venant 1871). Tresca's work was largely experimental (see Saint-Venant 1885). Pijlman (2001, Section 2) gives an introduction to the subject.

8. CURRENT FSI RESEARCH ACTIVITY

If one realises that every paper in the list of 123 references in Wiggert and Tijsseling (2001) has cost time and money (man-hours), this list gives a good indication of the mainly academic research activity in this area. The category "FSI - recent work" in the list of References in this report gives latest papers to be added to Wiggert and Tijsseling's list. Also, the five European projects mentioned below, and their participating members, give an indication of the viability and importance of the subject.

Nearly all activity known to the author takes place in Europe, and in particular in Germany, UK and France. Based upon publications, web-sites and personal relations, an overview for Europe:

Universities:

Aberdeen, UK
Bathe, UK
Darmstadt, Germany
Dundee, UK
Eindhoven, The Netherlands
Imperial College, University of London, UK
King's College, University of London, UK
Lille, ENSAM, France
Louvain - Leuven, Belgium
Trondheim, Norway

Research centres:

Forschungszentrum Rossendorf, Dresden, Germany
GMD-SCAI, Sankt Augustin, Germany
UMSICHT, Oberhausen, Germany
TNO, Delft, The Netherlands
WL | Delft Hydraulics, Delft, The Netherlands

Companies:

BASF, Ludwigshafen, Germany
EDF, Clamart, France
ESDU, London, UK
FDB, Trondheim, Norway
Flowmaster, Towcester, UK
Framatome (formerly: Siemens Nuclear Power), Erlangen, Germany
KAE, Bubenreuth, Germany
KED, Rodenbach, Germany
Mannesmann-Demag, Düsseldorf, Germany

European projects:

FLODAC - BRITE/EURAM3, BRPR970394

Title: Flow in Duct Acoustics
Coordinator: Prof M Åbom, KTH, Royal Institute of Technology, Stockholm, Sweden
Period: 1997-2001
See: Dequand (2001)

LEAKING - GROWTH - G1RD - 2002 - 00677
Title: Leak detection and reliability-based life prediction of water piping
Coordinator: Dr G Becker, RISA Sicherheitsanalysen GMBH, Berlin, Germany
Period: 2002-2005

SMT4-CT97-2188
Title: Transient pressures in pressurized conduits for municipal water and sewage water transport
Coordinator: Mr G McNulty, BHR Group Ltd, Cranfield, UK
Period: 1997-2001

SURGE-NET - GROWTH - G1RT - 2002 - 05069
Title: Organisations involved in the prediction and analysis of fluid transients in pipe systems
Coordinator: Mr D Stewardson, University of Newcastle, UK
Period: 2002-2005

WAHALoads - FIKS-CT-2000-00106
Title: Two-phase flow water hammer transients and induced loads on materials and structures of nuclear power plants
Coordinator: Prof M Giot, Université catholique de Louvain, Belgium
Period: 2000-2003
See: Giot *et al* (2001)

USA:

Argonne National Laboratory, Argonne, Illinois
Michigan State University, East Lansing, Michigan

9. UNSOLVED PROBLEMS

Much work has been done and the basic mechanisms of FSI and FIV are well understood these days. Areas that need further development are listed below.

FSI

priming, start up, slug flow

two-phase flow, cavitation, steam condensation (Giot *et al* 2001)

importance of FSI, guidelines, standards and codes

dynamic behaviour of supports

curved pipes and flexible hoses

plastic pipes, visco-elasticity

unsteady friction

experimental data, empirical relations

See Wiggert and Tijsseling (2001, Section 8)

FIV

turbulence (LES, DNS, LBM instead of k-ε)

CFD

experimental data, empirical relations

See Weaver *et al* (2000, Section 7)

10. CONCLUSION

From a theoretical point of view, FSI and FIV are well understood, and powerful computational tools exist to simulate the phenomena. From a practical point of view, there is a lack of general rules and practical guidelines. Each industrial problem has to be examined on a case by case base, often with the help of specialists.

This report is a guide to the vast amount of literature on the subject.

ACKNOWLEDGEMENTS

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ACRONYMS

CFD	computational fluid dynamics
DNS	direct numerical simulation (see Moin and Mahesh 1998)
FIV	flow-induced vibration (see Section 2.2)
FSI	fluid-structure interaction (see Section 2.2)
LBM	lattice Boltzmann method (see Chen and Doolen 1998)
LES	large eddy simulation
PSD	power spectral density function(s) (see Section 4.2.1)

SYMBOLS

A - amplitude, area

c - velocity of sound, damping coefficient

C_m - added mass coefficient

D - inner pipe diameter

e - pipe wall thickness

f - frequency

F - force

L - length

m - mass

n - mode number

P - pressure, power

R - inner pipe radius

St - Strouhal number

$\dot{u}$ - structural velocity

V - flow velocity

x - deflection

y - deflection

α - R times wave number

λ - wavelength

ν - Poisson ratio, frequency ratio

ρ - mass density

σ - axial stress

ω - circular frequency, vorticity

Subscripts

c - critical

eff - effective

f - fluid

N - natural

s - structure

v - valve

0 - initial value, steady state