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Real-life experiences with flow-induced vibration

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Abstract

A number of occurrences of flow-induced vibration in the power-generating industry are presented, many in nuclear plant where all incidents/problems have to be reported. Specifically, cases of (i) vortex-induced vibration (VIV), (ii) fluidelastic instability in cylinder arrays, (iii) axial and (iv) annular-flow-induced vibration, (v) leakage-flow instability and (vi) shell-type ovalling are discussed. For items (ii), (v) and (vi), a few words on the mechanisms underlying the vibration are provided.

Keywords: Flow-induced vibration; Vortex-induced vibration; Nuclear reactor internals; Heat exchangers; Fluidelastic instability; Cylinder arrays; Axial-flow-induced vibration; Annular-flow-induced vibration; Leakage-flow-induced instability; Ovalling of chimneys

1. Introductory comments

Some actual experiences in which flow-induced vibrations have caused damage, sometimes extensive and expensive, in industrial equipment are reviewed. There are several aims in this presentation: (i) to motivate our collective research into the causes and mechanisms of potentially debilitating vibration; (ii) to show that it is difficult to find published information on these experiences and, when found, it is seldom complete; (iii) to sensitize BBVIV participants that vortex-induced vibration (VIV) is but one of several fluid-flow excitation mechanisms; (iv) to show that some problems that have been blamed on vortex shedding were in fact associated with other causes.

The principal causes of FIV are first enumerated: nonresonant buffeting, response to flow periodicity, fluidelastic instability, and acoustic resonance—see Fig. 1. For cross-flow, response to flow periodicity refers principally to VIV, while fluidelastic instability could be galloping for prisms, "fluidelastic instability" for cylinder arrays, or wind-induced ovalling for cylindrical shells (chimneys). For axial flow (i.e., flow along the long axis of a structure), flow periodicity refers to parametric resonances, while fluidelastic instability corresponds to static divergence or flutter, e.g., of pipes, cylinders, plates and shells.

It is difficult to obtain any information at all about flow-induced (and other) problems in industrial equipment. Reasons for this are individual, corporate or national pride, corporate image and trade-mark protection, as well as fear of litigation. Thank God for the nuclear industry and national policies of open reporting of all problems, notably by Nuclear Regulatory Commission (NRC) in USA. Even so, putting together the information to constitute a reasonably well-documented "case" necessitates a fair amount of detective work, which makes it such a challenging and enjoyable task.

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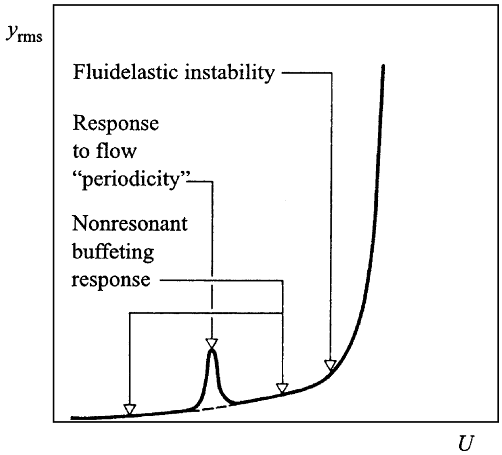


Fig. 1. Generic idealized response with increasing flow velocity of a structure in either axial or cross-flow.

The complexity of industrial equipment, e.g., a steam generator, makes the interpretation of problems therein difficult. One needs to know the details of the flow field in labyrinthine flow passages (only recently feasible via CFD), structural frequencies and nonlinear effects, response characteristics, acoustical frequencies, and so on. One also needs to have adequate criteria for VIV resonance and fluidelastic instabilities, which are still inadequate. In 1979, when most of the cases of practical experiences presented here were collected and reanalysed (Païdoussis, 1980) they were definitely less than adequate. In the original analyses of the same problems in the 1960s and early 1970s, the state of knowledge was quite unsatisfactory. For example, the existence of fluidelastic instability in cylinder arrays in cross-flow was totally unknown. Also, the Strouhal number (St) correlations or maps by Fitz-Hugh (1973) and Chen (1977) were inadequate and conflicting[1]; only later did more satisfactory St correlations become available (Weaver and Fitzpatrick, 1988) by expurgating acoustic resonance effects from the data bank and providing different correlations for each type of cylinder-array pattern (normal and rotated triangular, square and rotated square); refer to Weaver (1993).

2. Vortex-induced vibrations

Two VIV problems are reviewed first: one involving so-called in-core instrument (ICI) nozzles and guide tubes in a PWR-type nuclear reactor, and the other involving tubes in a tube-in-shell heat exchanger.

2.1. ICI nozzles and guide tubes in a PWR

ICI nozzles and guide tubes are used to guide the ICI thimbles into the core of the reactor, to monitor reactivity; see Fig. 2. In 1972, in one reactor, it was found that 21 out of 42 ICI nozzles had broken off, as well as four ICI guide tubes. This was discovered after inspection, initiated as a result of strange noises in the heat exchanger! The broken pieces mostly fell to the bottom of the reactor pressure vessel (Fig. 3), but some were carried by the flow to the heat exchanger.

The diameter of the nozzles was $D = 25.4$ mm and that of the guide tubes $D = 60.3$ mm. Their lowest natural frequencies were $f_n = 200-215$ and $80-200$ Hz, the range reflecting varying lengths. The average flow velocity was $U = 10.7$ m/s, and thus the Reynolds number, $\mathrm{Re} \approx 10^6$, is in the transitional range. Calculations were done with a Strouhal number St = 0.45 [at the high end of the possible range, see Blevins (1990), Chen (1987)], yielding vortex-shedding frequencies $f_{vs} = 190$ Hz for the nozzles and $200-215$ Hz for the guide tubes.

Based on the above, it was concluded that VIV was the culprit: partly lock-in, partly due to large values of fluctuating lift coefficients when the motion occurred at a frequency close to f_{vs}. If this appears to be tenuous, so it is! Other mechanisms, e.g., recognizing the effects of proximity to adjacent cylinders and wake interference [see Sumner et al. (2000), Zdravkovich (2003)], were not investigated. The "cure" was to redesign (beef-up) the ICI nozzles and guide tubes, such that $f_n > 4f_{vs}$. No problems were encountered thereafter.

[1]One of the Strouhal number maps is particularly intricate, and was said to look like "a map of Europe at the time of the Thirty years' War" (Païdoussis, 1980).

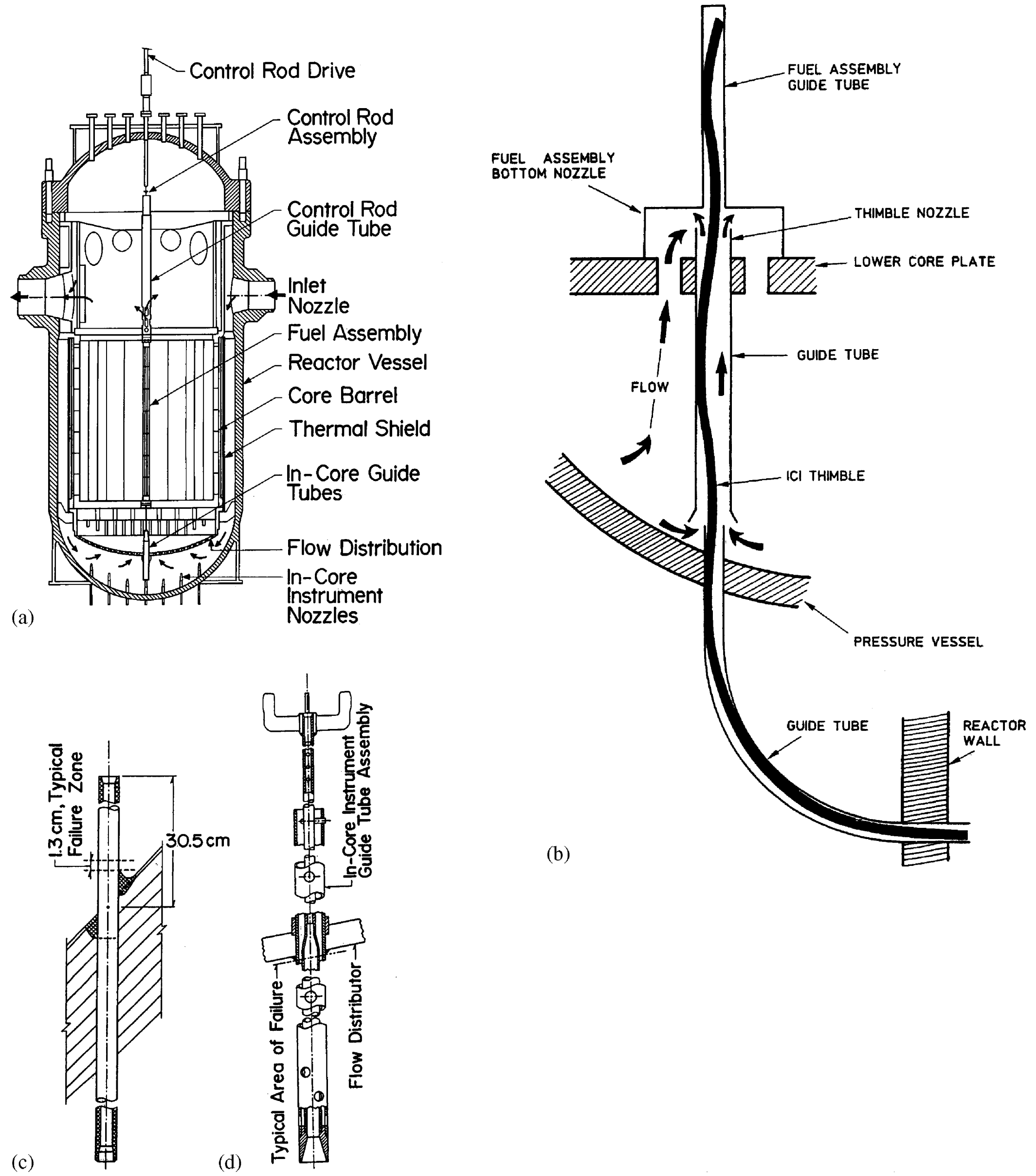


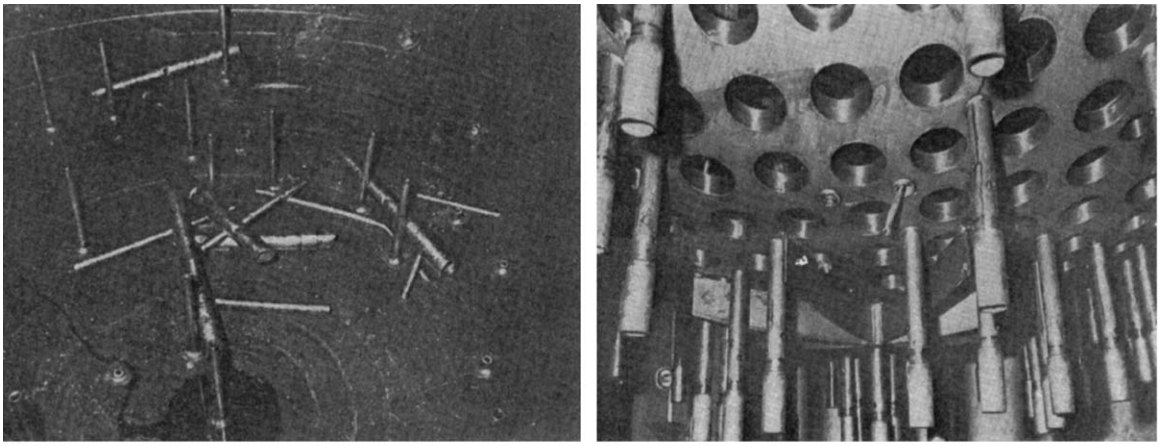
Fig. 2. (a) Schematic of a PWR nuclear reactor showing ICI nozzles and guide tubes; (b) schematic of the ICI system with the ICI thimble inserted; (c) detail of ICI nozzle; (d) detail of ICI guide tube (Païdoussis, 1980).

The replacement power costs (RPC), i.e., the costs to the utility of purchasing electricity from other suppliers were, in 1973, 0.1M\$/day for the 750 MWe plant. As the repairs took 10 months, RPC alone amounted to 30M\$. It should be noted that nowadays RPC $\approx$ 1M\$/day.

This case exemplifies that in industry (i) the real cause is often not pinned down definitively (for one thing, there is often not enough time) and (ii) the solution/cure is frequently quite pedestrian.

2.2. Heat-exchanger cylinder-array VIV

In this case the problem involved a heat exchanger with a normal triangular cylinder-array pattern (pitch-to-diameter ratio, $p/D = 1.3$). In the period 1966–1968, tubes ruptured in the same location in three different units, in the



(a) (b)

Fig. 3. Photographs of (a) the bottom of the reactor core showing missing (broken) ICI guide tubes; (b) the bottom of the pressure vessel showing broken ICI nozzles and débris (Païdoussis, 1980).

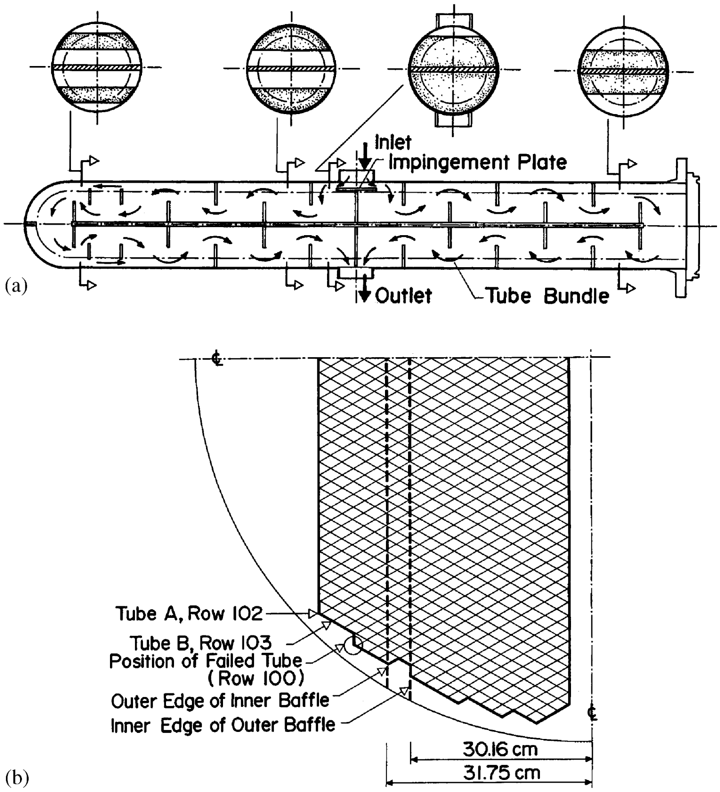


Fig. 4. (a) Cross-sectional view of the heat exchanger; (b) perpendicular sectional view, showing the outer perimeter zone where tube ruptures occurred (Païdoussis, 1980).

high-flow-velocity outer perimeter zone, Fig. 4. (A ruptured tube is serious, for it allows mixing of the inner and outer fluids.) Metallurgical analysis showed this to be due to fatigue failure; thus, fluidelastic instability was precluded.

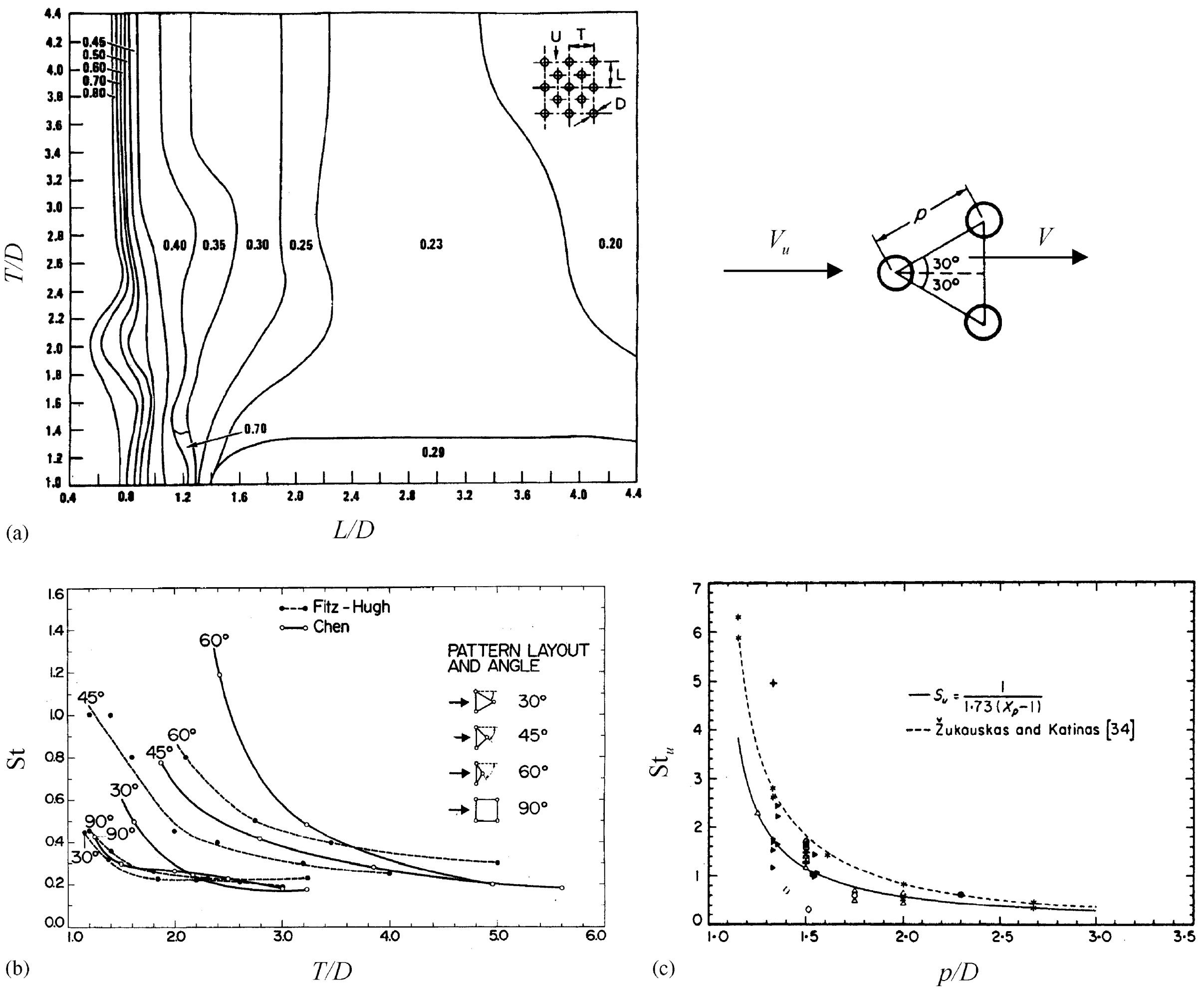


Fig. 5. (a) Strouhal numbers (St), the numbers within the chart, in staggered arrays according to Fitz-Hugh (1973); (b) Strouhal numbers (St) from the same data as reinterpreted by Païdoussis (1980); (c) St_u for normal triangular arrays (based on the upstream, free-stream flow velocity, ahead of the array) according to Weaver and Fitzpatrick (1988).

The ruptured tubes had $D = 16$ mm and $f_n = 22$ Hz. The mean rated flow velocity was $U = 0.84$ m/s, but at times could be 115% and 120% of the rated flow. The mechanism was thought to be cylinder-array-type VIV. This was reinforced by experiments, indicating an amplitude peak when U approached the rated flow.

At the time when it was reported (1970), the only available Strouhal number data for arrays of various geometries were compiled by Chen (1968). By the time the later analysis (Païdoussis, 1980) was made, there were also (i) Fitz-Hugh's (1973) map, see Fig. 5(a), (ii) Chen's (1977) improved charts and (iii) Païdoussis (1980) interpretation of Fitz-Hugh's map, Fig. 5(b). For $p/D = 1.3$ they predict (i) $St_C = 0.85$, (ii) $St_{FH} = 0.29$ and (iii) $St_{FH/MP} = 0.35$—so different as to raise questions about their reliability. Using the latter gives $f_{vs} \simeq 18.4$ Hz, which is close to f_n; at 115% and 120% of the rated flow, this gives 21 and 22 Hz. According to the newer design guide of Weaver and Fitzpatrick (1988), with the acoustical effects expurgated, see Fig. 5(c), one obtains $St_u = 1.7$, where St_u is based on the upstream flow velocity; hence $St = St_u/[p/(p - D)] = 0.39$, which gives $f_{vs} = 20.5$ Hz, and at 115% flow $f_{vs} = 23.6$ Hz, again close enough to $f_n = 22$ Hz. Based on the above, it was concluded that this was a vortex-induced failure.

However, we now know that the Strouhal number in arrays depends on how deep within the array the tubes in question are; see, e.g., Price et al. (1987) and Païdoussis et al. (1989). This was not considered. In any case, the cure was (i) to plug the tubes in the locations where failures occurred and (ii) to limit velocities below (recommended operation at 85% of) the rated flow.

3. Fluidelastic instability in cylinder arrays

Before discussing a case of fluidelastic instability due to cross-flow in cylinder arrays (typically tube arrays in heat exchangers), the mechanisms underlying this instability are reviewed, with the aid of a simplified, idealized model (Païdoussis and Price, 1988).

3.1. On the mechanisms underlying fluidelastic instability

Consider an array of cylinders, a kernel of which is shown in Fig. 6(a). The so-called negative-damping mechanism will be discussed first. Considering motions of only one cylinder (the others immobile) in the y-direction, the equation of motion is

$$ml\ddot{y} + c\dot{y} + ky = F_y, \tag{1}$$

where F_y is the fluid-dynamic force, m is the mass of the cylinder per unit length, c the damping coefficient and k the stiffness. Using quasi-static theory (Fig. 6(b)), we have

$$F_y = \tfrac{1}{2}\rho U_r^2 lD\{C_L \cos(-\alpha) - C_D \sin(-\alpha)\},$$

$$U_r = [(U-\dot{x})^2 + \dot{y}^2]^{1/2}, \quad -\alpha = \sin^{-1}(\dot{y}/U_r); \tag{2}$$

C_L and C_D are the static lift and drag coefficients, D is the cylinder diameter and ρ the fluid density. For small motions, $C_L = C_{L_0} + (\partial C_L/\partial x)x + (\partial C_L/\partial y)y$, and similarly for C_D. Then Eq. (2) may be linearized to give

$$F_y = \frac{1}{2}\rho U^2 lD\left[-2C_{L_0}\left(\frac{\dot{x}}{U}\right) + \left(\frac{\partial C_L}{\partial x}\right)x + \left(\frac{\partial C_L}{\partial y}\right)y - C_{D_0}\left(\frac{\dot{y}}{U}\right)\right]. \tag{3}$$

For symmetric geometrical patterns, $C_{L_0} = 0$ and $\partial C_L/\partial x = 0$, and Eq. (3) simplifies to

$$F_y = \frac{1}{2}\rho U^2 lD\left[\left(\frac{\partial C_L}{\partial y}\right)y - C_{D_0}\left(\frac{\dot{y}}{U}\right)\right]. \tag{4}$$

A time delay between cylinder displacements and the forces generated thereby is assumed, $\tau = \mu D/U$, where $\mu \sim \mathcal{O}(1)$. Assuming further that $y = y_0 \exp(\mathrm{i}\omega\tau)$, Eq. (4) becomes

$$F_y = \frac{1}{2}\rho U^2 lD\left[\mathrm{e}^{-\mathrm{i}\omega\tau}\left(\frac{\partial C_L}{\partial y}\right)y - C_{D_0}\left(\frac{\dot{y}}{U}\right)\right]. \tag{5}$$

Substituting in the equation of motion, we obtain

$$\ddot{y} + \left[\left(\frac{\delta}{\pi}\right)\omega_0 + \frac{1}{2}\left(\frac{\rho UD}{m}\right)C_{D_0}\right]\dot{y} + \left[\omega_0^2 - \frac{1}{2}\left(\frac{\rho U^2 D}{m}\right)\left(\frac{\partial C_L}{\partial y}\right)\mathrm{e}^{-\mathrm{i}\omega\tau}\right]y = 0, \tag{6}$$

where ω_0 is the natural frequency of the cylinder, and δ the logarithmic decrement.

For harmonic motions, the total damping is

$$\left[\left(\frac{\delta}{\pi}\right)\omega\omega_0 + \frac{1}{2}\left(\frac{\rho UD}{m}\right)\omega C_{D_0} + \frac{1}{2}\left(\frac{\rho U^2 D}{m}\right)\left(\frac{\partial C_L}{\partial y}\right)\sin\left(\frac{\mu\omega D}{U}\right)\right]\dot{y} \tag{7}$$

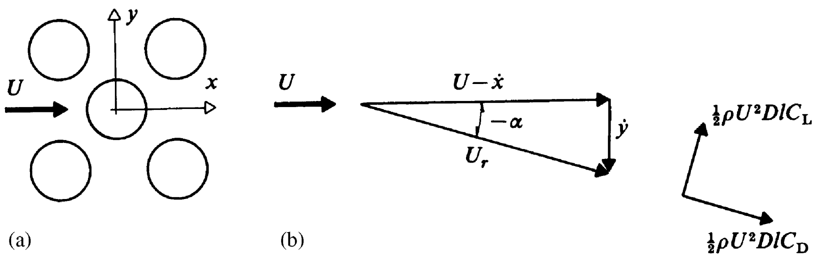


Fig. 6. (a) A kernel of a generic cylinder array in cross-flow with only the central cylinder free to move; (b) the lift and drag on that cylinder according to quasi-static fluid dynamics (Païdoussis and Price, 1988).

and instability is associated with [ ] = 0; if $\mu\omega D/U$ is small, sin( ) ≈ ( ), and

$$\frac{U_c}{f_0 D} = \left\{\frac{4}{-C_{D_0} - \mu D(\partial C_L/\partial y)}\right\}\frac{m\delta}{\rho D^2}, \tag{8}$$

where δ is the logarithmic decrement in vacuo. Hence, instability is possible only if

$$-C_{D_0} - \mu D\big(\partial C_L/\partial y\big) > 0, \tag{9}$$

i.e., if $\partial C_L/\partial y < 0$ and large. It may be shown that this can be reduced to Den Hartog's (1932) criterion for galloping. In arrays, however, the time delay is *necessary* ($\mu \neq 0$).

The negative-damping mechanism elucidated above applies for values of the mass-damping parameter $m\delta/\rho D^2 < 10^2$ approximately. For larger $m\delta/\rho D^2$, the instability is predominantly due to a displacement-dependent stiffness-controlled mechanism, involving at least two degrees of freedom—say the transverse displacements of two neighbouring cylinders; it is similar to wake-flutter of transmission lines. The critical velocity in this case is found to be

$$\frac{U_c}{f_0 D} = \left\{\frac{-64\pi^2}{\overline{\kappa}_{12}\overline{\kappa}_{21}}\right\}^{1/4}\left(\frac{m\delta}{\rho D^2}\right)^{1/2}, \tag{10}$$

where $\overline{\kappa}_{12}$ and $\overline{\kappa}_{21}$ are the off-diagonal terms of the dimensionless fluid-stiffness matrix. Hence, for this instability, the system must be nonconservative and hence $\overline{\kappa}_{12} \neq \overline{\kappa}_{21}$, but also $\overline{\kappa}_{12}\,\overline{\kappa}_{21} < 0$.

More comprehensive and elaborate models for fluidelastic instability do of course exist [see comprehensive review by Price (1995)], but the simple treatment in Païdoussis and Price (1988) does capture the essentials very nicely; see Fig. 7. Price classifies the available theoretical models into: (i) the jet-switch model of Roberts (1966); (ii) quasi-static models [e.g., Connors (1970, 1978), Blevins (1974)]; (iii) unsteady models [e.g., Tanaka and Takahara (1980, 1981), Chen (1983, 1987)]; (iv) semi-analytical models [e.g., Lever and Weaver (1986), Yetisir and Weaver (1993)]; (v) quasi-steady models [e.g., Price and Païdoussis (1984, 1986a, b), Price et al. (1990), Granger and Païdoussis (1996)]; (vi) inviscid flow models [e.g., Païdoussis et al. (1984, 1985)]; (vii) computational fluid-dynamic models [e.g., Marn and Catton (1991a, b)].

3.2. *The early history of fluidelastic instability*

Prior to 1970, the phenomenon was almost totally unknown. Roberts (1966) did some very fine work on the topic, the first ever, both theoretical and experimental; however, his theoretical model was quite complex and rather particular. A substantially simplified model was developed by Connors (1970). According to Connors (1970) and later Blevins (1974), the critical flow velocity for fluidelastic instability for *a single row of cylinders* is

$$\frac{U_c}{f_n D} = K\left(\frac{m\delta}{\rho D^2}\right)^{1/2}, \tag{11}$$

with $K = 9.9$; f_n is the (lowest) natural frequency of the cylinders. Designers from companies other than the one employing Connors mistakenly presumed that this relationship applied equally to *multi-row arrays* of cylinders. It was not till eight years later, when Connors (1978) published his work on arrays, that it became known that the same equation may still be applied, but with $K = 2.7-3.9$. This helps explain the large number of heat exchangers badly designed with $K = 9.9$ (i.e., presuming a U_c about 3 times what it should have been), and the disastrous consequences. In roughly a decade, the cumulative damages (including power replacement costs) world-wide are estimated at 1000M$. Another reason is that, long after 1970, many heat-exchanger designers ignored the existence of this instability. Indeed, in many of the cases analysed by Païdoussis (1980), the cause of damage was supposed to be vortex shedding, yet simple analysis showed that it was in fact fluidelastic instability.

A visual compendium of the disastrous effects of fluidelastic instability in heat exchangers is shown in Fig. 8. Intercylinder impacting with baffle supports (i) wears the tubes thin till they burst and (ii) cuts through the baffle supports, creating a free double-span resulting and higher amplitude vibration. In the case of a sodium–water heat exchanger, the $Na-H_2O$ chemical reaction caused additional devastation.

3.3. *A case of fluidelastic instability*

This is a case of fluidelastic instability which arose in several PWR-related steam generators of the same type, over a period of over seven years; Fig. 9(a). The damage occurred in the U-bend region, because of insufficient support by the original set of antivibration bars, resulting in the occurrence of low-frequency modes and hence fluidelastic instability at relatively low flows (Fig. 9(b)). The problem was solved by a new support arrangement in the *U*-bend region (Fig. 9(c)), such that the operating $U/f_n D$ was now smaller than $U_c/f_n D$.

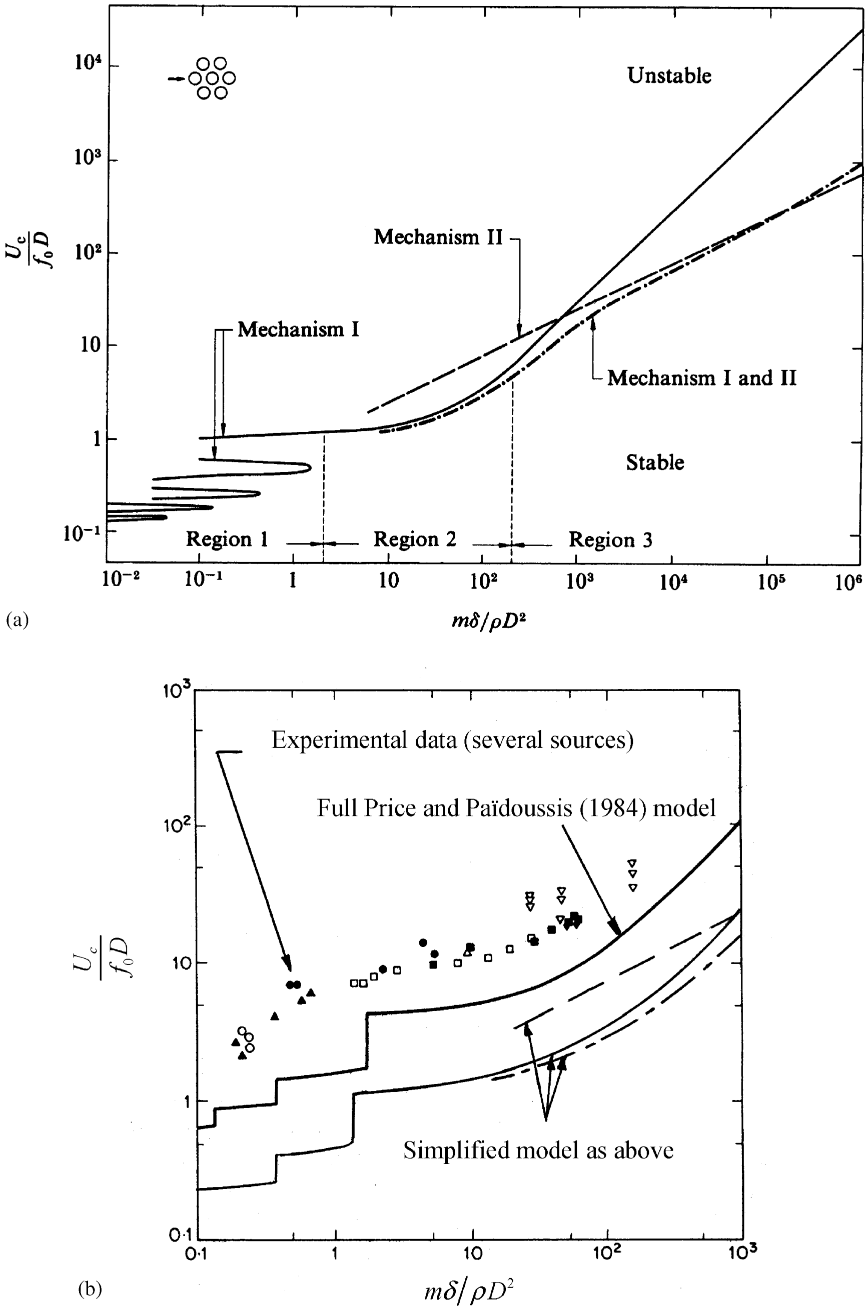


Fig. 7. (a) A chart of the critical reduced flow velocity versus the mass-damping parameter; mechanism I is the negative-damping mechanism and mechanism II is the stiffness-controlled mechanism (Païdoussis and Price, 1988); (b) the stability chart according to the simplified mechanisms of Païdoussis and Price (1988), the fuller but more elaborate Price and Païdoussis (1984) model and experimental data.

4. Axial-flow-induced vibration

A single case of axial-flow-induced vibration is presented, involving the ICI tubes in several BWR-type nuclear reactors. These are very slender cantilevered tubes supported at the bottom on the core support plate (Fig. 10).

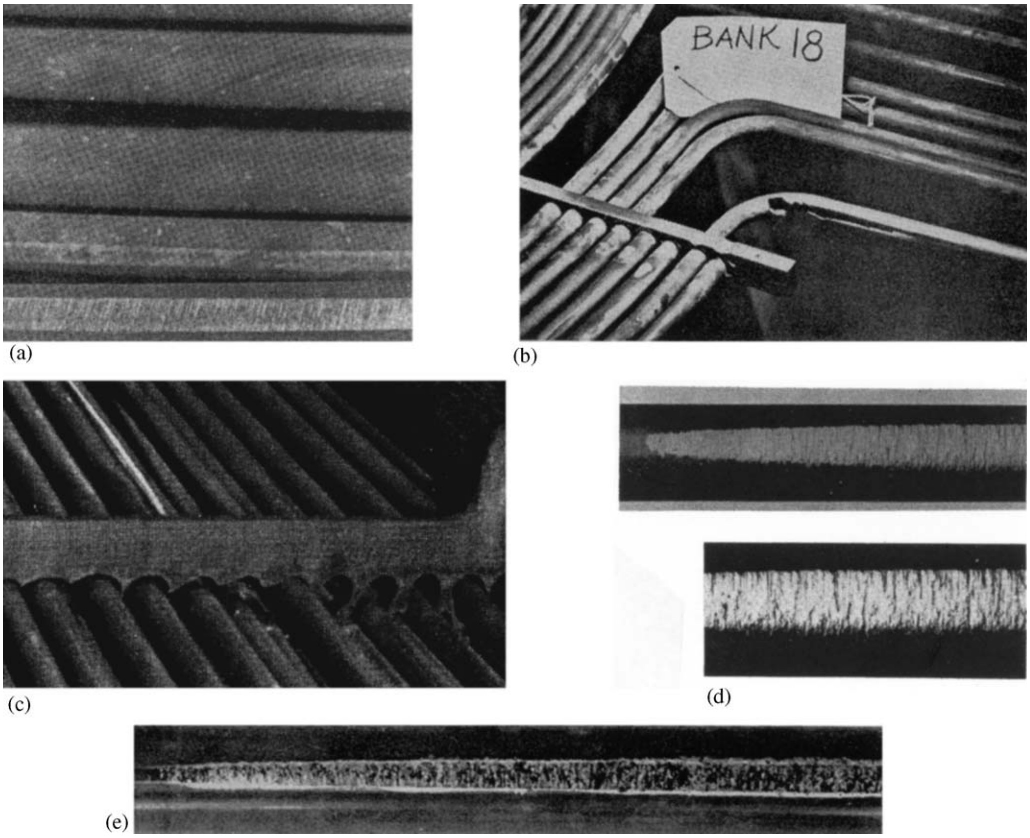


Fig. 8. A compendium of characteristic damage to heat-exchanger tube arrays due to fluidelastic instability: (a) from a CANDU steam generator; (b) from Na$-$$H_2O$ steam generator; (c) from a steam–steam heat exchanger; (d) from a steam condenser; (e) from another heat exchanger; based on cases analysed in Païdoussis (1980).

The problem arose when, to improve ICI performance, by-pass holes were drilled in the core plate to provide enhanced cooling. This allowed highly turbulent jets to issue from the core plate and excite the instrument tubes. The tubes then impacted on the corners of the channel boxes, in some cases fracturing them and producing holes as large as 8.9×12.7 cm—a serious matter, because coolant flow is thereby diverted from the fuel in the channel, also causing cross-flow in the fuel rods; missing pieces of channel box, carried away by the coolant, created added worry. In most cases, power was reduced to 40% until the problem was solved. In one reactor, of the 192 channels inspected, 65% were considered rejects; four had been perforated. In another, which also had "poison curtains" (for neutron absorption) in the channel box interstices, the curtains were found to vibrate and impact on the channels; upon removing them the problem was not solved, because then the ICI tubes were found to impact on the channels, a problem that "was hidden 'behind the curtains' for the first two years".

The vibration was diagnosed as due to high-turbulence buffeting. Means of prediction of turbulent buffeting have been developed by, among others, Païdoussis (1969), Chen and Wambsganss (1972), Mulcahy et al. (1980) for single cylinders, and by Païdoussis and Curling (1985) and Gagnon and Païdoussis (1994) for cylinder clusters; see also Païdoussis (2003). The problem was solved by plugging the by-pass holes and replacing them by a new set which directs the flow toward the core support plate.

5. Annular-flow-induced vibration

Vibrations and instabilities due to annular flow are relatively easy to excite (Païdoussis, 2003). Flow perturbations in annular geometries are easily amplified, and the loads on the annular walls can be very large.

A case of annular-flow-induced vibration of the thermal shield (see Fig. 2(a)), involving also the core barrel and the pressure vessel of another type of nuclear reactor is briefly discussed. The thermal shield is a shell used to protect the

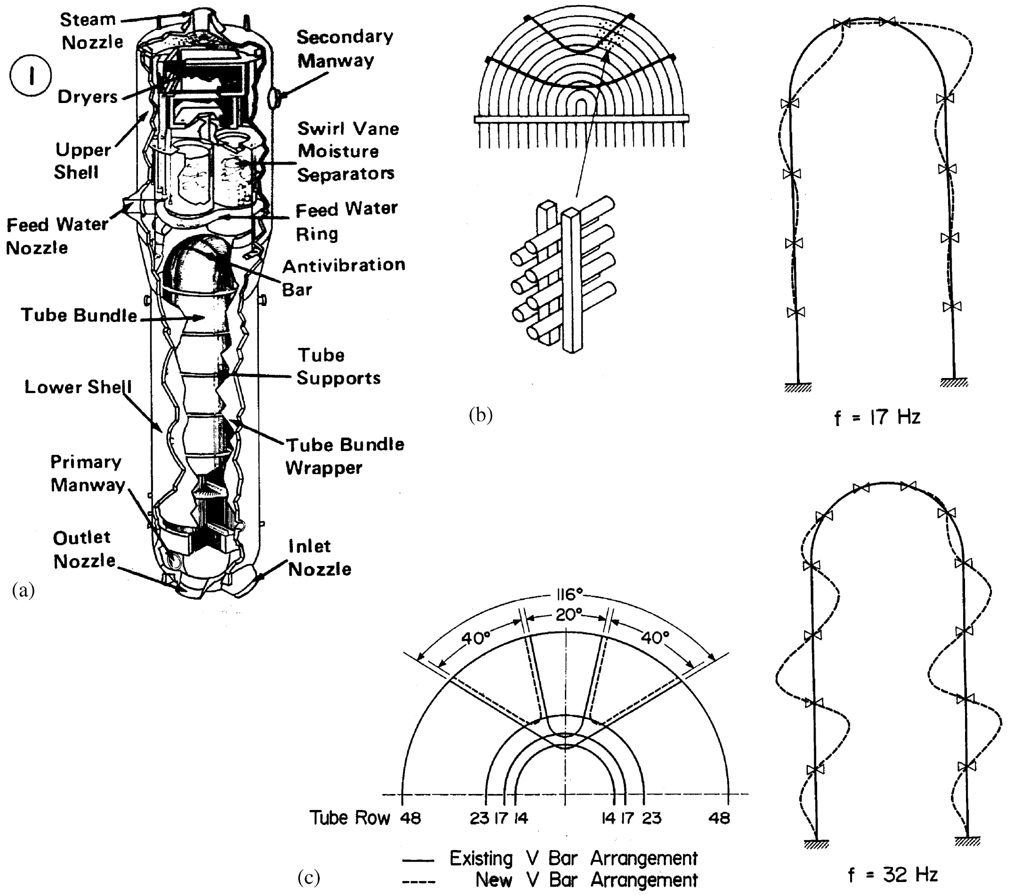


Fig. 9. (a) Schematic of a PWR steam generator; (b) antivibration-bar supports in U-bend region and one of the low-frequency modes of vibration; (c) redesigned supports in U-bend region and low-frequency mode, with a higher value of the frequency.

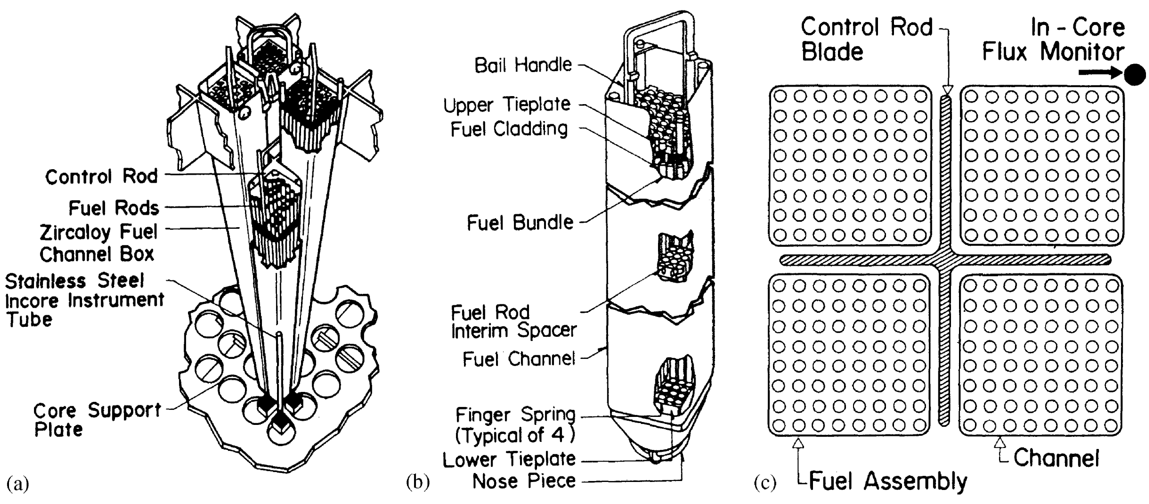


Fig. 10. (a) Schematic of part of the core of a BWR reactor, showing four fuel channels and the fuel rods, and an in-core instrument (ICI) tube; (b) a single fuel channel; (c) a cross-section of four fuel channels and an in-core flux monitor (Païdoussis, 1980).

pressure vessel from excessive neutron bombardment. It was found that: (i) several pins connecting the three segments of the thermal shield had broken off, allowing the segments to vibrate and impact on the core barrel and pressure vessel; (ii) $\frac{1}{3}$ of the pins connecting the upper and lower parts of the barrel failed, ending up in the stream generator; (iii) $\frac{2}{3}$ of the tie rods connecting the so-called lower casting to the barrel had failed; in one case the thermal shield dropped to the bottom of the pressure vessel. The shut-down, clean-up and repairs took three years.

The problem was diagnosed as due to a global instability of the flow, with large eddies forming at the top of the thermal shield, below the flow entry, capable of generating alternating loads of the order of 2 tons. The cure was to eliminate the thermal shield altogether, in most cases, but this meant removing some of the outer fuel assemblies, to protect the pressure vessel from excessive radiation. In one case, the shield was retained, but with additional, stronger supports.

For sufficiently large flow velocities, cylinders and shells in annular flow develop fluidelastic instabilities, divergence or flutter; but, in this particular case, the flow velocities (5–10 m/s) were not high enough (Païdoussis, 2003, Chapter 11).

6. Leakage-flow-induced instability

This is an enhanced form of annular-flow-induced instability, notorious for its destructiveness. Leakage flow, as the words imply, is something easy to overlook, but the forces that can be generated thereby are quite enormous (Païdoussis, 1980, 2003).

This is a negative-damping instability, the basics of which can be understood via Fig. 11 (Miller and Kennison, 1966). Consider a blade in a 2-D channel. Let us first assume the flow to be from left to right. It is supposed that the blade, which has a larger-size appendage on the left, is given an upward velocity V. As the upper sub-channel is reduced in area, the flow rate is reduced therein, and the flow must decelerate, with an attendant depression in the static pressure (if $\partial v/\partial t < 0$, then $\partial p/\partial x > 0$ in the upper channel). The opposite occurs in the lower sub-channel and there is a net pressure in the direction of the blade velocity. Hence, this results in amplified motion; if a mechanical restoring force exists, this gives rise to an oscillatory instability. If on the other hand, the flow is from right to left, the resultant pressure force acts opposite to the blade velocity, tending to damp motions. This establishes the "golden rule" for preventing leakage-flow-induced instabilities: put the constrictions in the annular flow conduit *downstream*.

A practical case involving control rods (for controlling reactivity) in guide tubes is presented in Païdoussis (1980), where this golden rule was contravened. Several holes in the guide tubes were discovered during refuelling operations at the position where the control rods reside (retracted) during normal operation. The mechanism was diagnosed as due to leakage-flow-induced instability. As a complete redesign was not feasible, the problem was "solved" by inserting reinforcing sleeves in the guide tubes.

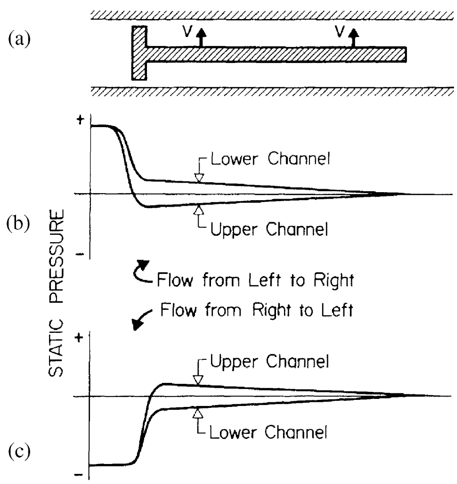


Fig. 11. Diagram to explain the mechanism associated with leakage-flow-induced instability, according to Miller and Kennison (1966). (a) A blade with a flow-constricting protuberance at the left-end in a 2-D channel. Pressure distribution (b) for the flow from left to right, and (c) for flow from right to left (Païdoussis, 2003).

7. Ovalling oscillation of shells in cross-flow

Finally, shell-type wind-induced ovalling oscillations of tall, steel chimney stacks are discussed. One such occurrence involved a 68 m tall chimney, 0.344 m in diameter and 7.9 mm wall thickness at the top, in Moss Landing Harbor, CA. The chimney developed ovalling oscillation in the second circumferential mode ($n = 2$) at a wind speed of 40 km/h, with a frequency of 1.47 Hz. Cracks developed. In another case, a chimney was totally destroyed in a typhoon, as a result of ovalling.

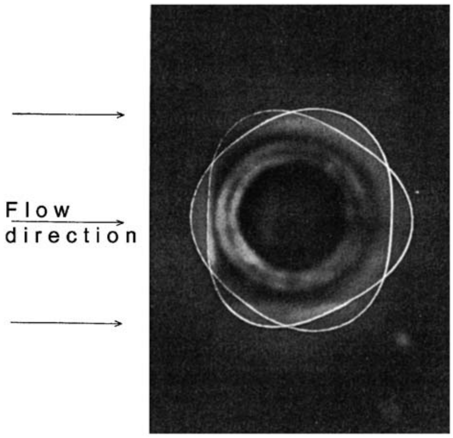


Fig. 12. A shell ovalling at $U = 21\,\mathrm{m/s}$ ($n = 3, m = 1$) with $f_{3,1} = 150\,\mathrm{Hz}$ (Paїdoussis and Helleur, 1979).

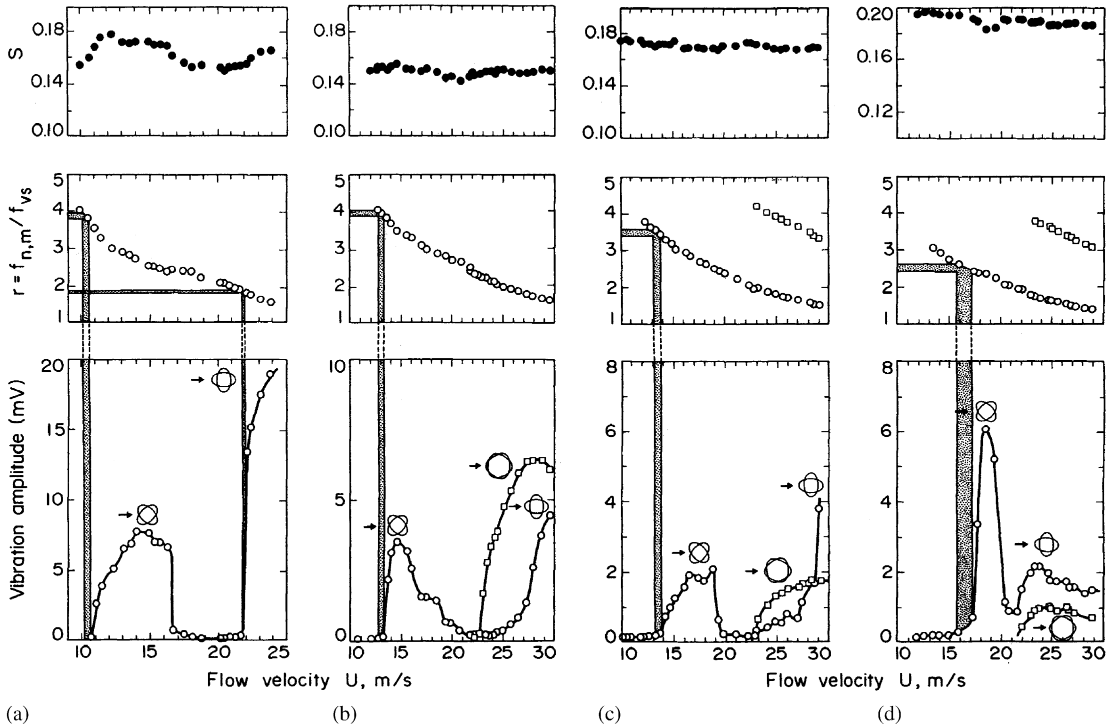


Fig. 13. Ovalling of cantilevered shells in cross-flow: (a,b) with $r =$ integer, and (c,d) with $r \neq$ integer; S is the measured Strouhal number (Paїdoussis et al., 1982b).

Prior to 1979, accepted theory supposed that ovalling oscillations were due to sub-harmonic resonance with vortex shedding, such that $f_{n,m}/f_{vs} = r =$ integer, $f_{n,m}$ being one of the shell frequencies ($n \geqslant 2$) with n the circumferential wave number and m the axial one, and f_{vs} the vortex-shedding frequency; r varied from 1 to 6. This theory was based on experiments by Johns and Sharma (1974) in the early 1970s, in which, however, f_{vs} was not measured but calculated by taking $\mathrm{St} = 0.20$ or 0.166, the latter to account for 3-D effects about the free top of the chimney.

For shells, the $f_{n,m}$ are very densely distributed. Thus, given the latitude afforded by $0.166 < \mathrm{St} < 0.20$ and $1 < r < 6$, it is not too difficult to find a value of $f_{n,m}/f_{vs}$ close to an integer.

New experiments were conducted (Fig. 12) by Païdoussis and Helleur (1979), mainly to investigate the effect on ovalling of the internal flow in the chimney. However, in the process, the basis of the vortex-shedding hypothesis was brought into question, when it was found that (i) in some cases $f_{n,m}/f_{vs} \neq$ integer at the onset of ovalling (Fig. 13), and (ii) when a splitter plate was used and f_{vs} totally disappeared, yet ovalling still occurred—indeed with larger amplitude!

A fluidelastic negative-damping model was proposed by Païdoussis and Wong (1982) to explain the phenomenon. The demise of the vortex-shedding hypothesis was valiantly resisted by the v.s.-proponents; the skirmishes in this mini-war are recounted in Païdoussis et al. (1988). However, the model was further improved and perfected (Païdoussis et al., 1982a, b, 1983, 1988, 1991; Laneville and Mazouzi, 1996), so that it is now accepted that ovalling is a self-excited fluidelastic flutter phenomenon, rather than being caused by vortex shedding. Hence, this represents yet another case of mistaken identity, when vortex shedding was thought to be the culprit, yet it was later shown that this was not a VIV but a fluidelastic instability problem.

The usual cure is to stiffen the chimney near the top by ring stiffeners. However, they must be welded on very well; spot-welded rings have been known to break loose because they did not prevent ovalling of the shell within.

8. Conclusion

Other, equally interesting phenomena are not even discussed, for brevity, e.g., in bellows, whirling shafts in narrow fluid-filled annuli, gravity/shell-weir-type instabilities. The interested reader should also refer to Axisa (1993) and Naudascher and Rockwell (1994).

It is clear that further research is needed, before truly reliable design tools are established for all these types of problems, but research funding and research effort in this area have steadily been declining over the past 15 years, partly due to the general marasmus in the power-generating industry.

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