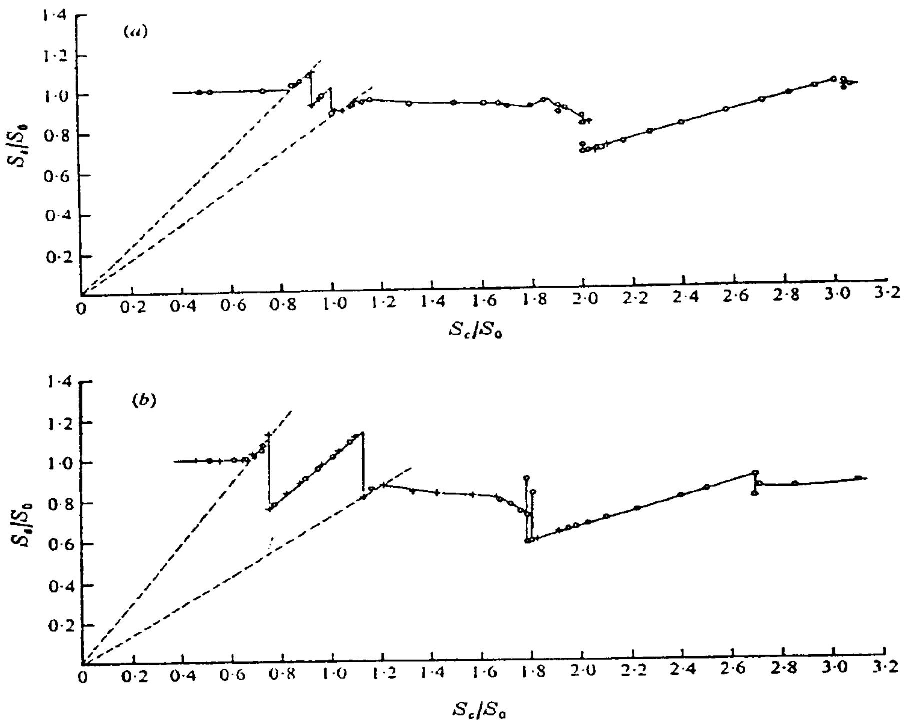


a) a/D = 0.10 and b) a/D = 0.29 for Re = 9200 (0 < Sc/So < 2.3) and 7000 (Sc/So > 2.3). O, increasing cylinder frequency; +, decreasing cylinder frequency.

FIGURE 5.6 SHEDDING FREQUENCY VS. OSCILLATING FREQUENCY (STANSBY 1976)

The data in Figure 5.6 are plotted for two amplitude ratios (0.1 diameters and 0.29 diameters). Note that increased amplitude is associated with a larger lock-in range as well as a greater reduction in shedding frequency above the primary lock-in range. Detailed observations from the experiments are described in the original reference [Stansby (1976)]. Some key observations were:

- Lock-in can occur at the frequency of cylinder motion or at sub-harmonics of the cylinder motion, in either uniform or shear flow.
- Shedding behavior at the edges of the lock-in frequency range is similar for uniform and shear flow.
- The minimum amplitude required for initiating lock-in may increase with increasing Reynolds number.
- The frequency range within which the shedding will lock-in to cylinder motion seems to narrow with increasing Reynolds number.
- At the edges of the frequency range where lock-in to the primary cylinder frequency is observed, shedding varies intermittently between the nominal frequency and the cylinder frequency.

The possible dependence on Reynolds number should be noted, as all of Stansby's data were taken at subcritical Reynolds numbers. Therefore it is difficult to judge the applicability of his conclusions to higher Reynolds numbers commonly of engineering interest.

Another interesting result is the change in vortex cell length during lock-in. Figure 5.7 illustrates how individual vortex cells in shear flow change dramatically when the cylinder frequency is within the lock-in range by presenting results of three separate tests superimposed on a single plot. Note that as the cylinder frequency changes from .354 to .280 to .168, the two cells become one, indicating locked-in shedding across the entire cylinder. Stansby also developed an approximate expression for the maximum length Δy of a locked-in vortex cell:

$$\frac{\Delta y}{D} = \left\{ \frac{4US_c}{\lambda DS_s} \right\} \frac{\alpha}{D}$$

where λ is velocity gradient, D is cylinder diameter, α is cylinder amplitude, S_c is cylinder frequency, and S_s is the nominal Strouhal frequency. The constant 4 is derived from Stansby's data at subcritical Reynolds numbers.

Peltzer et. al. (1985) studied the lock-in problem in detail and identified frequency ranges of locked-in behavior as a function of vibration amplitude. These data, along with similar data from other sources, are plotted in Figure 5.8. For example, these data indicate that for a constant vibration amplitude of 0.05 diameters, shedding will initially lock-in as the cylinder frequency is increased from 0 to about 90% of the shedding frequency, and will remain locked-in to the cylinder frequency until it exceeds approximately 105% of the nominal shedding frequency. Below the lower boundary of the lock-in range (i.e. $f_c << f_s$), the vortex-shedding frequency is essentially unchanged, while above the lock-in boundary (i.e. $f_c >> f_s$) the vortex-shedding frequency is reduced by an amount that increases with cylinder amplitude[3]. This reduction in shedding frequency is illustrated in Figure 5.9. For vibration amplitudes below 11% of diameter, the data exhibited no substantial change. This result is consistent with both Griffin's (1971), and Koopman's (1967) observations. They both noted that for forced vibration amplitudes less than 10% of diameter, no measurable increases in the correlation or coherence of the vortex shedding along the cylinder span were observed, and that above 10% there was a measurable increase in the correlation or coherence of the shedding.

For amplitudes greater than 11% of diameter, the Strouhal numbers characterizing the upper boundary of the lock-in region decreased linearly with increasing a/d. This reduction in Strouhal number with increasing a/d has been observed by Woo et. al. (1981) in their experimental studies and by Sarpkaya et. al. (1979) in experimental and numerical analyses. Peltzer also observed that the vortices were shed at a lower Strouhal number because of the increased vortex strength. The vortex strength was increased because each growing vortex was fed circulation over a longer period of time. Sarpkaya & Shoaff (1979) numerically showed that as the strength of the vortices in the near wake of a vibrating cylinder was increased, the Strouhal number decreased proportionally.

It may be important to note that all of the experimental evidence of lock-in has been developed for the case of harmonic cylinder motion. Existence of the upper and lower lock-in boundaries is not at all clear if the cylinder is undergoing random oscillations. One brief study has been attempted to investigate the effects of bandwidth on lock-in [Schargel (1980)]. In this study, harmonic, narrow-band random, and wide-band random oscillations were applied to a cylinder in a uniform flow whose Strouhal frequency was very close to the oscillation frequency, and well within the lock-in range discussed previously. Lock-in was clearly evident for harmonic motion, but it was weaker for the narrow-banded motion as measured by the coherence between cylinder motion and velocities in the wake. For the broad-band case, a shift in frequency toward the cylinder frequency was noted, but the coherence between cylinder motion and wake velocity was very low.

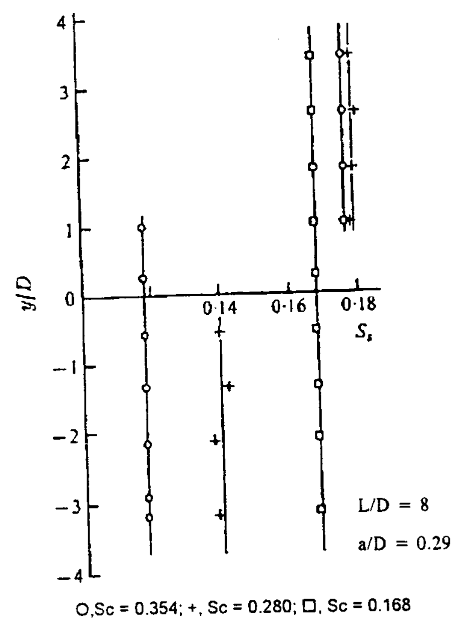
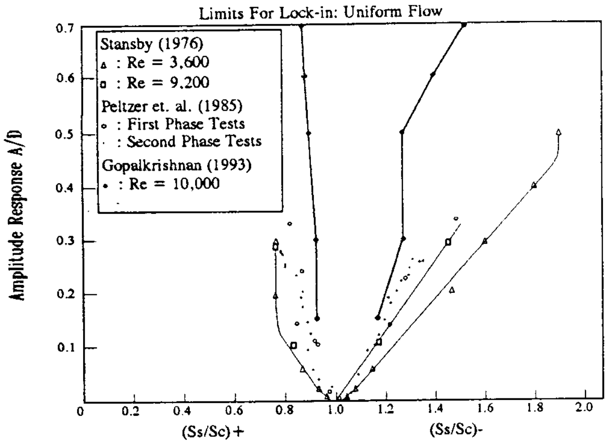


FIGURE 5.7 SPANWISE DISTRIBUTION OF SHEDDING FREQUENCIES (STANSBY 1976)

Based on the coherence data, the investigators concluded that lock-in did not occur in the case with broad-band cylinder motion. However, one potential problem with their method was that the velocity probes in the wake were fixed in position, and therefore the random swinging of the wake due to cylinder motion could reduce coherence measurements considerably. Many experiments in the literature fix velocity probes to the cylinder, such that cylinder motion does not cause motion of the probes relative to the moving wake. One could speculate that wide band, random oscillation will somehow inhibit lock-in, but at this point there is little evidence in the public domain on the effect of bandwidth.

[3] Here the terms "lower" and "upper" are used in the sense implied by Figure 5.6 and increasing cylinder frequency from "lower" to "upper". This is opposite of the usage in Peltzer's paper.



Ss = Strouhal number (vortex shedding frequency)
Sc = Strouhal number (cylinder osillation frequency)

FIGURE 5.8 LOCK-IN REGIONS

5.2.2 Phase Shift

Many of the data sets taken to investigate lift amplitude and frequency on oscillated cylinders have demonstrated a remarkably clear and rather abrupt 180° phase shift that occurs within the locked-in frequency range at subcritical Reynolds numbers. The phase shift was first investigated by Bishop & Hassan (1964) in their comparison of locked-in shedding to a mechanical oscillator model. They measured the phase of the hydrodynamic force on the cylinder relative to the cylinder motion, and found that it shifted at a "critical cylinder frequency" within the locked-in shedding range. The critical frequency was shown to be a function of cylinder amplitude and Reynolds number. The sudden phase shift was also shown to be associated with a significant jump in force amplitude, hence the comparison with a mechanical oscillator.

An example of the phase shift is given in Figure 5.10. Stansby (1976) associated the phase shift with a sudden change in wake structure. Peltzer (1985) discusses work by Zdravkovich that provides an explanation for the phase shift. Zdravkovich (1982) found that before the phase shift, vortices were shed at maximum cylinder amplitude from the "outside" cylinder edge (farthest from the wake centerline). After the phase shift, vortices are shed at maximum cylinder amplitude from the inside cylinder edge.

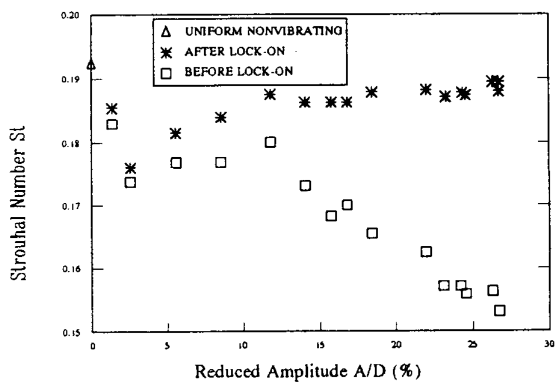
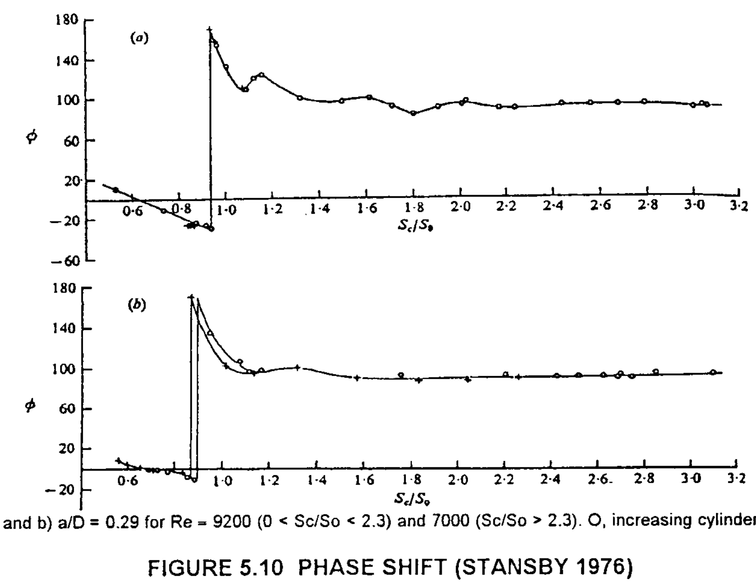
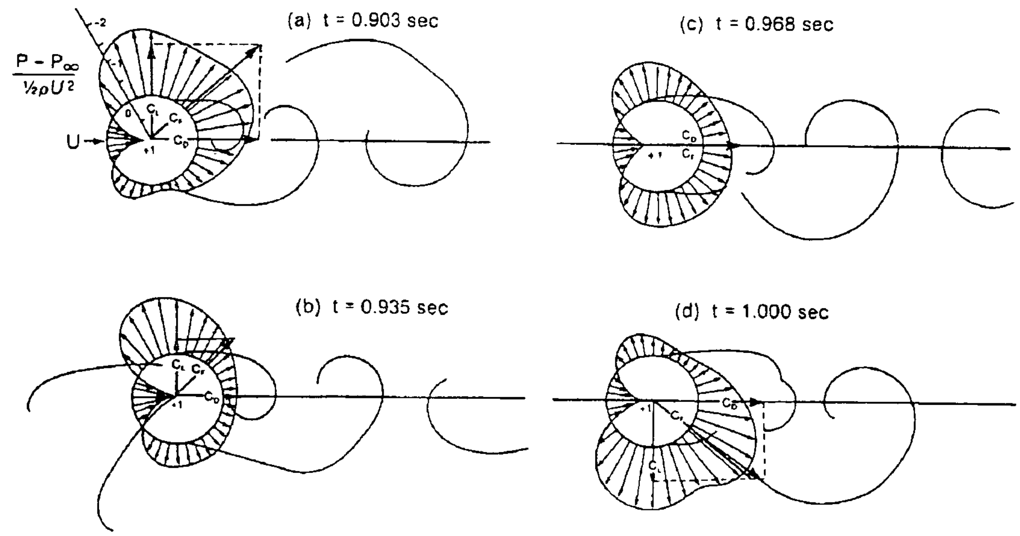


FIGURE 5.9 SHEDDING FREQUENCY VS. AMPLITUDE (PELTZER ET. AL. 1985)



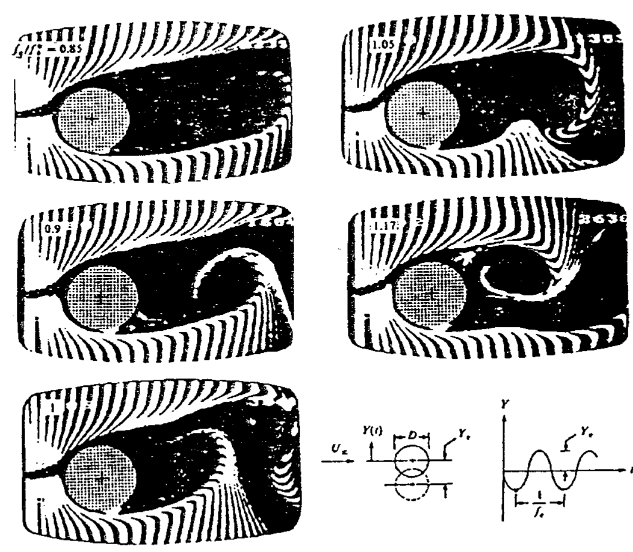
Uniform flow with a) a/D = 0.10 and b) a/D = 0.29 for Re = 9200 (0 < Sc/So < 2.3) and 7000 (Sc/So > 2.3). O, increasing cylinder frequency; +, decreasing cylinder frequency.

FIGURE 5.10 PHASE SHIFT (STANSBY 1976)



Reynolds Number Re = 112,000. One-third of one cycle of vortex shedding.

FIGURE 5.11 WAKE AND PRESSURE OF STATIONARY CYLINDER (BLEVINS 1986)



f_e = excitation frequency
f_o = natural shedding frequency
All photos taken at maximum negative displacement of cylinder.

FIGURE 5.12 WAKE OF OSCILLATING CYLINDER (ONGOREN ET. AL. 1988)

TABLE 5.3 SPANWISE COHERENCE VERSUS SPAN/DIAMETER RATIO

Curve #	Authors	Low H/D	High H/D	Medium	Comments
1	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D = 0.0
2	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D = 0.025
3	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D = 0.0375
4	Toebes, (Wootton/Scruton paper)	0.0E+00	7.0E+00	air	pressure transducer and hot wire, uniform flow α/D = 0.0
5	Toebes	0.0E+00	7.5E+00	air	oscillating cylinders α/D = 0.0
6	Toebes	0.0E+00	7.5E+00	air	oscillating cylinders α/D = 0.04
7	Toebes, (Wootton/Scruton paper)	0.0E+00	7.0E+00	air	oscillating cylinders α/D = 0.04
8	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D = 0.05
9	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D = 0.075
10	Toebes, (Wootton/Scruton paper)	0.0E+00	7.0E+00	air	oscillating cylinders α/D = 0.08
11	Toebes	0.0E+00	7.5E+00	air	oscillating cylinders α/D = 0.08
12	Howell - Novak	0.0E+00	6.0E+00	air	elastically-mounted, rigid cylinders, Re = 75,000, α/D=0.125
13	Toebes, (Wootton/Scruton paper)	0.0E+00	7.0E+00	air	oscillating cylinders α/D = 0.12
14	Toebes	0.0E+00	7.5E+00	air	oscillating cylinders α/D = 0.125

It is difficult to describe exactly how shedding occurs, but Figure 5.11 illustrates the phase between force and shedding for a stationary cylinder. Note that the peak lift force occurs in the direction of the most recently shed vortex, and at the instant the vortex detaches, there is no net lift force. If the vortex detaches at maximum cylinder amplitude from the outside cylinder edge, hydrodynamic force opposes cylinder motion. For low cylinder frequencies before the phase shift, this is consistent with the idea of hydrodynamic added mass. If after the phase shift, the vortex detaches at maximum cylinder amplitude from the inside cylinder edge, hydrodynamic force acts in the direction of cylinder motion. This is consistent with the negative added mass observed by Sarpkaya at reduced velocities above the lock-in region. Negative added mass is a result of the decelerating mass of fluid being greater than the accelerating mass [Sarpkaya (1977)].

The abrupt switch in phase of the shed vortice from the outside cylinder edge to the inside cylinder edge is illustrated in Figure 5.12. This flow visualization clearly shows that as ratio of oscillation frequency to shedding frequency exceeds 1.0, the vortex shed at maximum cylinder amplitude switches sides of the cylinder.

Similar phase shift examples are apparent in all of the available oscillated cylinder data. When interpreting these data, it is important to understand how the data were obtained and processed, especially with regard to frequency content. Within the lock-in range the data show phase between harmonic cylinder motion and net hydrodynamic force or wake velocity, including excitation (lift) force. Outside of lock-in, the data show phase only between motion and the fluid-structure interaction forces of added mass and damping. The former is defined only over a relatively narrow range of cylinder frequency, while the latter is not very important if the dominant frequencies of cylinder motion and force or wake velocity are different.

5.3 Correlation Length

It is well established that correlation of the lift force increases rapidly for oscillating cylinders, and attains maximum values during lock-in. There have been a number of studies done to quantify the effect of amplitude on correlation of the lift force. Many of these data are compiled in Figure 5.13, and related information about each model test is provided in Table 5.3.

It is characteristic in Figure 5.13 that for stationary cylinders the correlation coefficient drops quickly to low values as the separation distance (H/D) increases. This means that the cross-correlation of lift forces becomes weak. For oscillating cylinders, the correlation coefficient remains high for larger separation distances, and increases as a function of response amplitude.

It is important to note that the oscillating cylinder data in Figure 5.13 comes from tests where the cylinder oscillates at or very near the nominal shedding frequency. Therefore, the connection between greatly increased correlation length and vibration amplitude is demonstrated for locked-in shedding at the oscillation frequency only. When the cylinder oscillation frequency differs from the shedding frequency, correlation length drops off quickly toward the values for stationary cylinders.

In Figure 5.14a, where the ratio of Strouhal frequency to cylinder frequency is 1.25, note that correlation length is relatively insensitive to vibration amplitude, and shows little increase above stationary cylinder values at the maximum amplitude of 0.25 diameters. From Figure 5.8 it is apparent that a frequency ratio of 1.25 at an A/D of .25 is outside the expected lock-in range. However, in Figure 5.14b a very high sensitivity to amplitude is demonstrated, consistent with Figure 5.13. Data in Figure 5.14b are from tests at a frequency ratio of 1.0 (cylinder frequency equal to Strouhal frequency). One can conclude from these data that the increase in correlation length with amplitude is strongest when the cylinder frequency is near the Strouhal frequency, and in fact may only occur if shedding is locked-in to the cylinder motion.

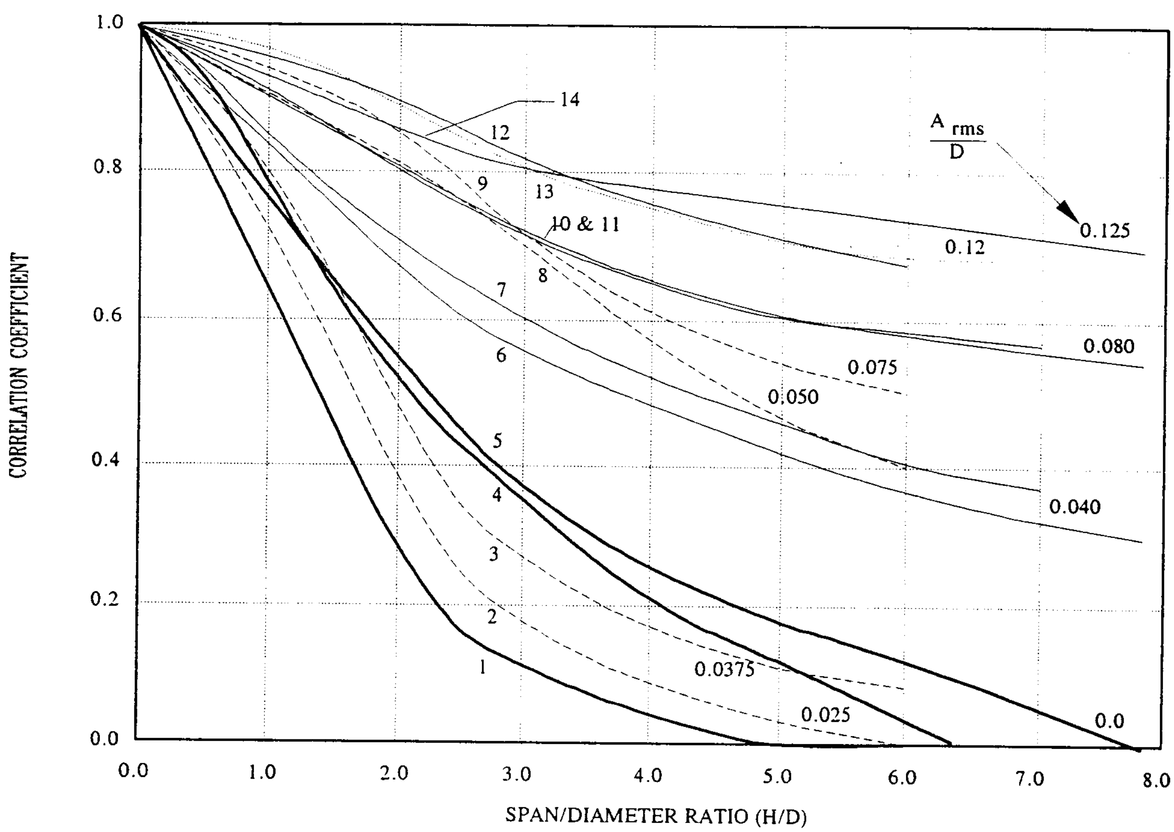
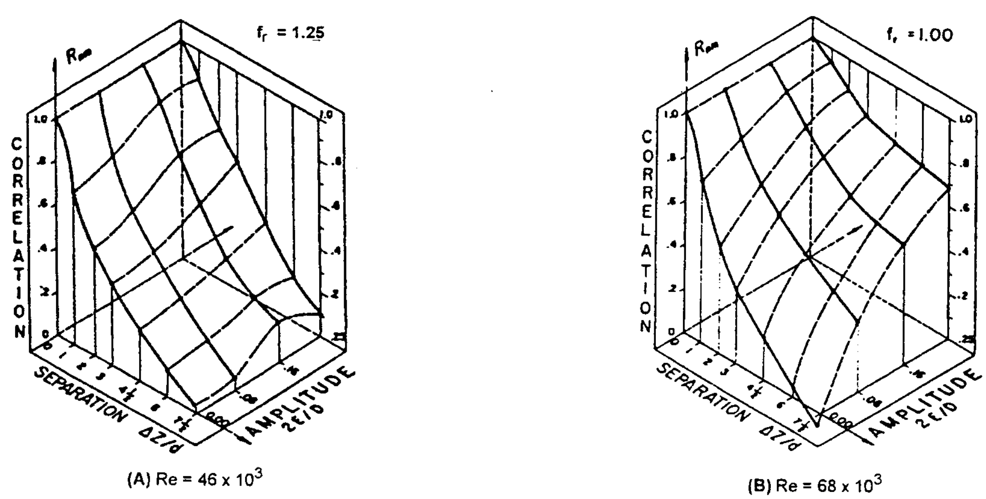


FIGURE 5.13 CORRELATION COEFFICIENT VS OSCILLATION AMPLITUDE (COMPILED DATA)



(A) Re = 46 x 10^3

(B) Re = 68 x 10^3

FIGURE 5.14 CORRELATION LENGTH VS AMPLITUDE AND FREQUENCY (TOEBES 1969)

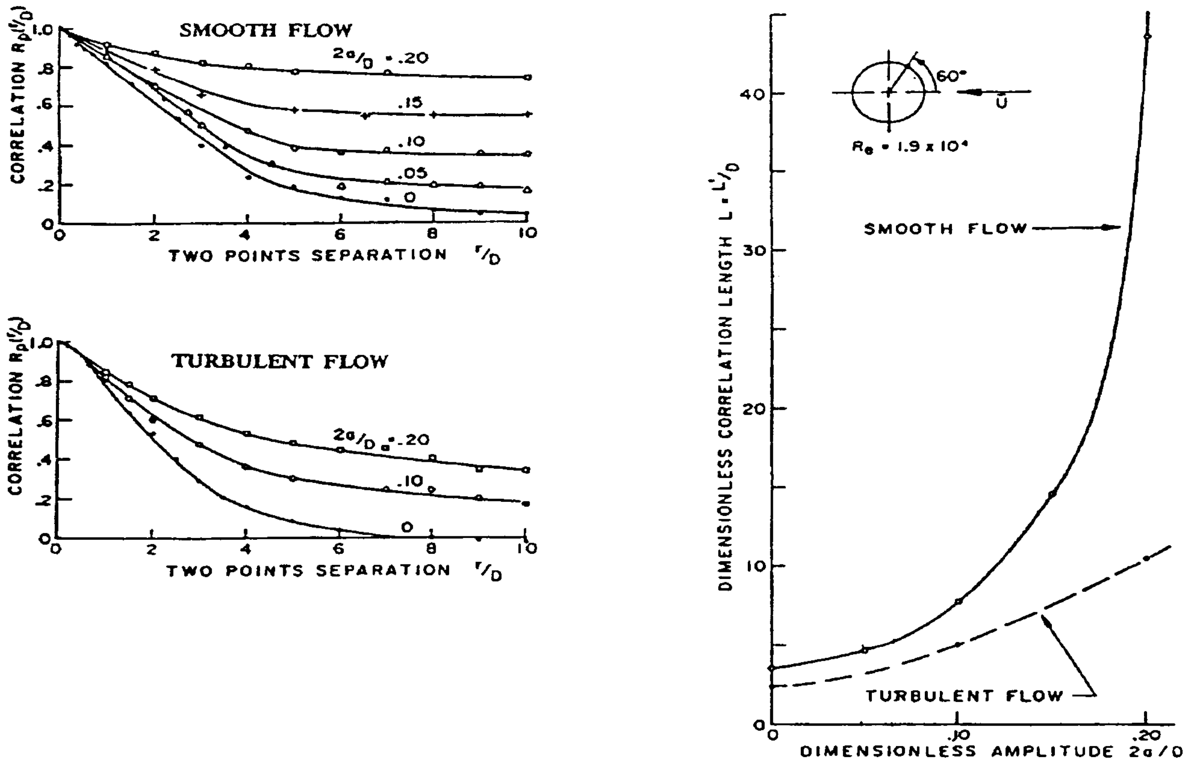


FIGURE 5.15 CORRELATION LENGTH AND TURBULENCE (NOVAK ET. AL. 1975)

5.3.1 Turbulence

Novak et. al. (1975) found that correlation length increased dramatically with the amplitude of motion. This increase was much larger in smooth flow than in turbulent flow. Their coefficients for smooth and turbulent flow are shown in Figure 5.15 (11% turbulence intensity) as a function of motion amplitude and turbulence. A response amplitude of 10% of diameter increased the correlation length from 3.5D to 43D (where D is cylinder diameter) in smooth flow and from 2.4D to 10.4D in turbulent flow. The rate of increase was steeper than linear but no abrupt change indicative of lock-in was found. These changes are accompanied by a significant reduction of the bandwidth of the power spectrum. They suggested that these changes are important and should be included in VIV prediction models.

Howell et. al (1980) performed model tests with elastically-mounted cylinders in a wind tunnel, in subcritical uniform and turbulent flow conditions, to investigate the effect of response amplitude on correlation length. The correlation of lift forces was measured by pressure taps at several locations along the cylinder, and by hot wire for the velocity fluctuations in the wake. The pressure correlation showed a similar type of decay to that of Novak et. al. (1975). The correlation of velocities in the wake compared well with those by Toebes (1969) but did not show as large an increase with amplitude as did the pressure correlations. Their results are shown in Figure 5.16 [curves 1, 2, 3, and 4].

They also found that for oscillating cylinders, increasing amplitudes increase correlation, while turbulence decreases it. For smooth flow [curve 1], an increasing of response amplitude from 5% to 10% of diameter quadrupled correlation length, from 8D to 42D, for a Reynolds number 1.9×10^4. Correlation length did not increase as significantly, up to 25D [curve 2], for a higher Reynolds number of 7.5×10^4.

For turbulent flow, correlation length did not increase as rapidly with response amplitude [curves 3, 4]. In addition, the effect of Reynolds number on the correlation length for turbulent flow was not significant. Correlation length values less than about 10D were estimated for both Reynolds numbers. These values are significantly lower than the values for the smooth flow.

EFFECT OF A/D ON Lc/D

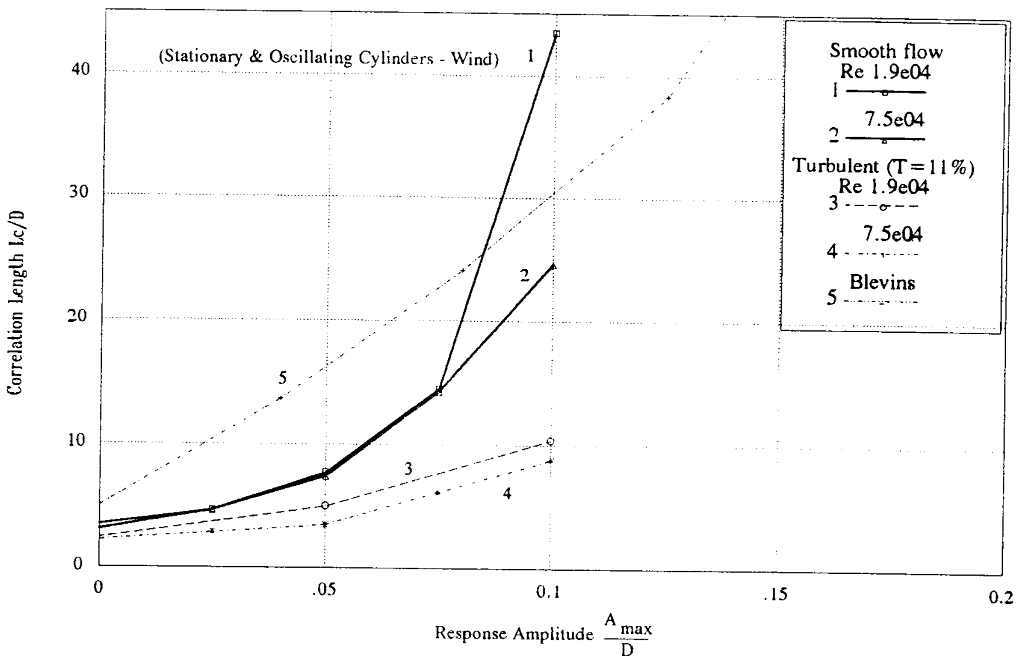


FIGURE 5.16 CORRELATION LENGTH AND TURBULENCE

5.3.2 Cables (and Flexible Cylinders)

Ramberg et. al. (1976), and Griffin et. al. (1975) performed model tests to investigate the spanwise correlation coefficient measured between two hot-wire probes in the wake (near wake) of a vibrating cable. Based on this work and later interpretations reported in conjunction with an assessment of the OTEC design [Griffin (1981)] they reported the following:

- Vortex shedding was fully correlated over most of the half-wavelength of the cable (L=12-14 D, where D is cable diameter) when the displacement amplitude was greater than a threshold amplitude of 5% - 10% of diameter.
- A change in sign of the correlation coefficient was observed when one of the hot-wire probes moved past a node of the cable, indicating a 180° phase shift corresponding to the change in displacement phase inherent to the vibration mode shape.
- Correlation value appears to be independent of frequency, amplitude, and Reynolds number within the lock-in region.
- Near wake properties along a vibrating cable are similar to those of a vibrating cylinder at the same frequency, amplitude, and Reynolds number.

These observations are important, in that they suggest that the response of a cable or flexible cylinder such as a marine riser can be modeled by a local force model calibrated to lift force and correlation lengths obtained from rigid cylinder experiments.

The observation of a phase shift around the cable node is especially noteworthy, in that it provides a basis for modeling lock-in of higher vibration modes. The observed phase shift is similar to the shift that occurs as cylinder frequency is varied around the nominal Strouhal frequency (Section 5.2.2), and it seems reasonable to assume that the hydrodynamic mechanism is similar (a change in the side of the cylinder where detachment of vortices occurs at maximum amplitude). However, the former is a difference in phase between two different locations at the same frequency, where the latter is a shift in phase at the same location as frequency is varied.

Peltzer et. al. (1985) performed forced-vibration model tests using a marine cable in water. For vibration amplitudes less than 10% of diameter, no measurable increase in correlation or coherence of the vortex shedding along the cylinder span occurred. Above 10% of diameter, there was a measurable increase in correlation or coherence of shedding, consistent with the findings of Ramberg and Griffin.

5.4 Added Mass and Damping Forces

The previous sections of this paper focused exclusively on lift, which is the source of excitation force for vortex-induced vibrations. Estimating vortex-induced vibrations of marine risers or cables also requires modeling hydrodynamic forces other than lift in order to realistically represent dynamic equilibrium. For example, consider a cable in shear flow over part of its length. The cable will respond to some extent at a variety of frequencies over its entire length. The ensuing cable motion through the fluid will generate additional hydrodynamic force that will affect its response. Hydrodynamic force on cables and slender cylinders moving through a fluid is typically viewed in terms of added mass, proportional to acceleration of the object, and damping, proportional to velocity of the object. In this section we first focus on added mass and damping forces generated by cylinder motion, then combine them with lift into a integrated representation of hydrodynamic force.

The most widely used model of hydrodynamic force on a circular cylinder is the Morison equation, which represents force on a stationary cylinder as the sum of drag force plus the force due to the acceleration of the fluid flowing around the cylinder:

$$\frac{F}{L}=\frac{1}{2}C_d\rho D|V|V+\frac{1}{4}C_m\pi\rho D^2\frac{dV}{dt}$$

where L is cylinder length, D is cylinder diameter, V is fluid velocity, ρ is the mass density of the fluid, and C_d and C_m are the damping and the inertial coefficients, respectively. This equation can be extended to cover the case of a moving cylinder by using the fluid velocity and acceleration relative to the cylinder motion. After some simplification, the force for a cylinder moving parallel to the flow is often written as:

$$\frac{F}{L}=\frac{1}{2}C_d\rho D|V-\dot{x}|(V-\dot{x})+\frac{1}{4}C_m\pi\rho D^2\frac{\partial V}{\partial t}-\frac{1}{4}(C_m-1)\pi\rho D^2\ddot{x}$$

where the velocity and acceleration of the cylinder are $\dot{x}$ and $\ddot{x}$, respectively.

The above equation is the well-known result for a cylinder in waves where V is the fluid velocity within the wave, or waves plus current where V represents the summation of wave and current velocity.

For a cylinder oscillating transversely to the flow, the geometry of the problem results in a slightly different formulation:

$$\frac{F}{L}=\frac{1}{2}C_d\rho D\left\{\sqrt{V^2+\dot{x}^2}\right\}\dot{x}-\frac{1}{4}(C_m-1)\pi\rho D^2\ddot{x}$$

The first term is the component of drag force transverse to the incident flow. This drag always opposes cylinder velocity, and therefore contributes hydrodynamic damping to the system. For relatively small, harmonic oscillations, a linearized version of this equation is [Chung (1987)]:

$$\frac{F}{L}=\frac{1}{2}C_d\rho DV\left\{1+.375\left(\frac{\dot{x}_{rms}}{V}\right)^2\right\}\dot{x}-\frac{1}{4}C_a\pi\rho D^2\ddot{x}$$

The added mass force is the second term, and C_a is the added mass coefficient. The magnitude of the added mass force depends on the type of motion of the body and motion of the fluid about the body. For a cylinder undergoing harmonic oscillations with a relative amplitude of A/D of unity in a fluid otherwise at rest, there is no vortex shedding and the added mass coefficient is equal to unity[4]. For similar values of A/D where there is vortex shedding (i.e. as in steady flow), the added mass coefficient may even be negative. Negative added mass simply means that the mass of fluid being decelerated exceeds the mass of fluid being accelerated during a cycle of motion [Sarpkaya (1977)].

Many attempts have been made to characterize the hydrodynamic force on a cylinder undergoing harmonic motion transverse to a steady flow. These experiments have provided direct measurements of added mass and damping forces as well as data on the combined effects of lift, added mass, and damping. These data are discussed in the following section.

5.4.1 Oscillated Cylinder Tests

In forced-oscillation (oscillated) model tests by Sarpkaya (1977), Rajaona et. al. (1981), Staubli (1983), Moe et. al. (1989), and Gopalkrishnan (1993), determinations of the hydrodynamic force are made by driving a short cylinder in a flow at a prescribed amplitude and frequency, and recording the necessary external force input. The cylinder boundary conditions varied from fixed-end with end-plates to spring-mounted with and without end-plates. All the tests were performed at subcritical and critical Reynolds numbers. The vectorial difference between the inertial force (force recorded on a piezoelectric force transducer while performing the prescribed cylinder oscillation in air) and the external forcing is the hydrodynamic force lift.

These researchers have typically represented their data with a force component in phase with cylinder velocity and a force component in phase with acceleration. A general equation that applies to these data is

$$\frac{F}{L}=\frac{1}{2}C_d\rho DX_d\dot{x}-\frac{1}{4}C_a\pi\rho D^2\ddot{x}$$

[4]The value typically used for C_a in dynamic analysis of many ocean structures in waves and current, including marine risers.

where X_d is a damping linearization coefficient ($=\frac{8}{3\pi}$ for harmonic motion).

Oscillated cylinder experiments generally collect force data for a variety of oscillation frequencies and amplitudes, conducted over a range of flow velocities. At each flow velocity and amplitude combination, cylinder frequencies typically range from below to well above the nominal (stationary cylinder) Strouhal frequency. These data therefore represent a variety of flow and response conditions that range from low frequency motion superimposed upon vortex shedding at a higher frequency, through lock-in, to high frequency motion superimposed on vortex shedding at a lower frequency. The raw data therefore generally contain frequencies associated with vortex shedding along with the cylinder oscillation frequency.

The typical process for recording and reducing data from oscillated cylinder experiments involves subtracting inertia forces due to structural mass directly from the recorded force time history. The remaining signal represents hydrodynamic force. While details of the procedure vary, the Fourier series for the hydrodynamic force time history is typically computed using the cylinder oscillation frequency as the fundamental frequency. Terms representing higher harmonics are then neglected, and the result is amplitude and phase (relative to cylinder displacement) of a harmonic representation for applied force.

Although the oscillated cylinder data are presented as "lift" coefficients versus either reduced amplitude or nondimensional frequency (inverse of reduced velocity), they really represent two fundamentally different force regimes. Outside of lock-in, where lift due to shedding is at the Strouhal frequency and hydrodynamic force due to cylinder motion is at the cylinder frequency, the data reduction techniques used filter out the lift force. The data contain only force terms resulting from motion of the cylinder through the fluid, typically characterized as added mass and damping. Within the lock-in range, the hydrodynamic force data includes lift, damping, and added mass forces, combined into a term in phase with cylinder velocity and a term in phase with cylinder acceleration.

Therefore, these data represent applied lift force in the sense used previously (Sections 4.0 through 5.3) only where the shedding frequency and body frequency coincide, yet the effects of added mass and damping cannot be separated from the lift. Thus lift force cannot be directly extracted from these data. However, it is likely that the magnitude and phase of the measured hydrodynamic force is representative of the complicated fluid-structure interaction. If this interaction is important to accurately estimating VIV response (and there is every indication that it is), then these data contain valuable information that is lacking in both stationary and freely-oscillating cylinder tests. It is also likely that these data contain some of the best information on the amplitude and frequency dependence of added mass and damping forces.

An example of typical oscillated cylinder data is given in Figure 5.17, which contains data extracted from Sarpkaya (1977) and replotted.

The solid line in Figure 5.17 is Sarpkaya's coefficient for the measured force in phase with cylinder velocity, and the dashed line is the coefficient corresponding to the measured force in phase with acceleration. Sarpkaya's definitions were such that the coefficients relate directly to linearized damping and added mass forces from the linearized, harmonic Morison equation.

In Figure 5.17 the approximate lock-in boundaries from Figure 5.8 are indicated. For reduced velocities from 0 to approximately 4.5, the coefficients can be interpreted as frequency-dependent added mass and damping coefficients for cylinder oscillations through steady flow at frequencies above the local Strouhal frequency (e.g. for the portion of a cable or riser that is primarily responding to shedding from higher current speeds, further up in a shear current profile). For reduced velocities above approximately 6.3, the coefficients reflect hydrodynamic forces for cylinder response at frequencies below the local Strouhal frequency. An example would be the lower mode response of a riser or cable that is primarily driven by shedding at lower current speeds elsewhere in the current profile. In either range, the reduced data reflect hydrodynamic force occurring at the cylinder frequency.

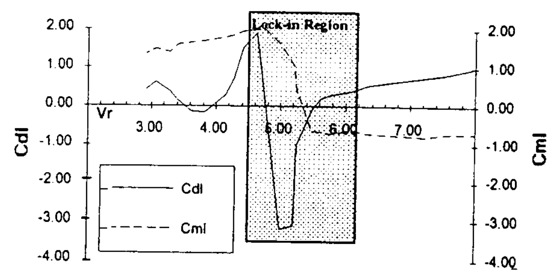


FIGURE 5.17 FORCE COEFFICIENTS FOR A/D = 0.50 (SARPKAYA 1977)

For reduced velocities between about 4.5 and 6.3, the force consists of lift, damping, and added mass. Here we have what is generally modeled as three distinct forces represented as a single amplitude and phase (or alternatively a cosine term and a sine term). Sarpkaya tested his model in this range by using the Fourier coefficients C_{ml} and C_{dl} in a dynamic equilibrium equation. Parameters of the system were selected to model published test results for a spring-mounted cylinder in uniform flow. He solved for displacement in the time domain, and his results compared very well with the measured data. The important implication of this result is that these data contain a complete and accurate description of lift force and the fluid-structure interaction effects associated with realistic cylinder motions.

It is straightforward to extract added mass and damping coefficients from these data at frequencies outside of the lock-in region. It follows that a simple model for VIV could be constructed using lift coefficient data from Section 4 through Section 5.3, augmented by added mass and damping coefficients from the data in this section. However, special consideration should be given to those portions of an oscillating cylinder predicted to become "locked-in". Sarpkaya's results indicate that the oscillated cylinder data is a good representation of the total hydrodynamic force under locked-in conditions. Therefore, it would be reasonable to create a model that utilizes stationary cylinder data for initial lift coefficients, and transitions to the use of oscillated cylinder data as cylinder motion increases.

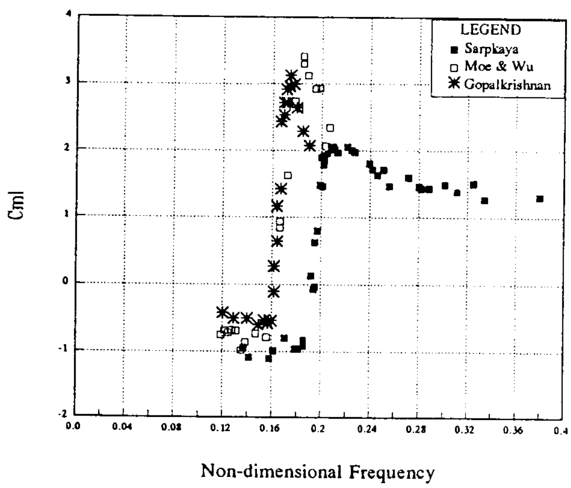


FIGURE 5.18 ADDED MASS COEFFICIENTS FOR A/D=0.25

Based on estimated response, damping and added mass effects can be calculated from coefficients derived from oscillated cylinder data. If significant locked-in response develops, then the effects of fluid-structure interaction, as measured in oscillated cylinder tests, can be incorporated by using an equivalent added mass coefficient to maintain total hydrodynamic force amplitude and phase equal to the oscillated cylinder data.

5.4.2 Added Mass Coefficient

A compilation of the added mass terms from several experiments is presented for three amplitude ratios in Figures 5.18 through 5.20. The abscissa is the cylinder oscillation frequency, f_o, multiplied by the Strouhal number, and, normalized by the Stouhal frequency of an equivalent stationary cylinder, f_s. Therefore at $f_o/f_s = 1.0$, cylinder motion occurs at the nominal Strouhal number of 0.2. The ordinate is the added mass coefficient, as defined previously. As discussed in the previous section, these data represent a single frequency approximation to applied hydrodynamic force on an oscillating cylinder. Sarpkaya (1977) gives an thorough discussion of the data reduction process, and presents a useful comparison between raw and reduced force data.

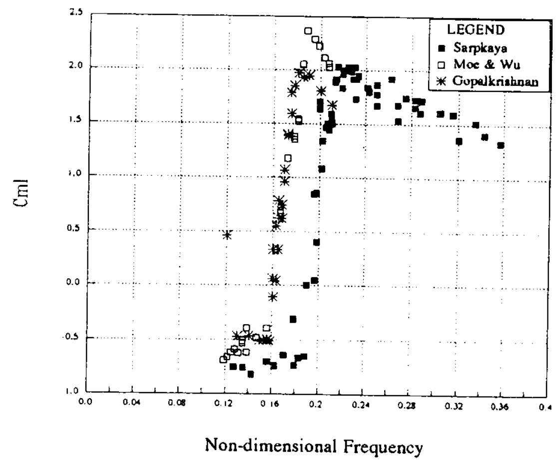


FIGURE 5.19 ADDED MASS COEFFICIENTS FOR A/D=0.50

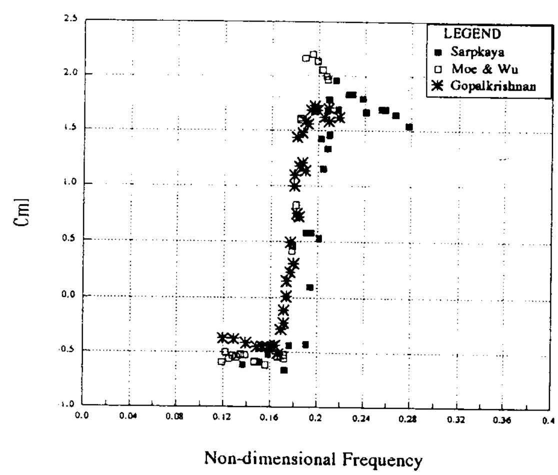


FIGURE 5.20 ADDED MASS COEFFICIENTS FOR A/D=0.75

These data support the following observations:

- For a cylinder oscillating in a steady flow, added mass is only equal to its nominal value of unity at frequencies far above lock-in.

- At frequencies above the lock-in boundary, as either cylinder frequency increases or flow velocity decreases, added mass increases to a maximum near 2.0, analogous to the case of a fluid oscillating about a stationary cylinder.

- At frequencies below the lock-in boundary, the added mass coefficient becomes negative. This is rather due to body motion than excitation force.

- The increase in added mass above lock-in, and the negative added mass below lock-in, tend to change the natural vibration frequency of the cylinder toward the lock-in region.

- Within the lock-in boundary, it is not possible to separate added mass from lift. The resulting coefficient on force proportional to cylinder acceleration is highly frequency and amplitude dependent.

- The added mass coefficient is generally frequency-dependent, but relatively insensitive to amplitude. There is some tendency for the negative added mass values to increase (i.e. become less negative) as amplitude increases.

A useful way to view these data is via a contour map, as presented in Figure 5.21. The ordinate is nondimensional amplitude (A/D), and the abscissa is nondimensional frequency (the cylinder Strouhal frequency, = f_0D/V). The author chose to plot a quantity he calls $C_{L\_A_0}$ rather than a true added mass coefficient, so the contour line values need to be converted via the following formula:

$$C_a = C_{L\_A_0} \frac{U^2T^2D}{2\pi^3AD^2}$$

Note that added mass is frequency dependent but not very sensitive to amplitude at oscillation frequencies above the nominal shedding frequency. At oscillation frequencies below the Strouhal frequency (i.e. low oscillation frequency, high flow velocity) the picture is somewhat more confusing, but modeling of the added mass force is likely to be less important in this case. Within the lock-in boundaries, added mass is both frequency and amplitude-dependent.

5.4.3 Damping Coefficient

A compilation of damping terms from several experiments is presented for three amplitude ratios in Figures 5.22 through 5.24. The abscissa is cylinder oscillation frequency, f_o, normalized by the Stouhal frequency of an equivalent stationary cylinder, f_s. The ordinate is the damping coefficient, as defined previously. Note that the variation in damping with amplitude is very pronounced. It can be concluded from these data that the damping coefficient is very dependent on amplitude, and somewhat less sensitive to frequency outside of the lock-in range. It also seems that dependence on amplitude is much stronger at frequencies above the lock-in range than at frequencies below the lock-in range. Within the lock-in range, as with the added mass data, the damping data are both frequency and amplitude dependent.

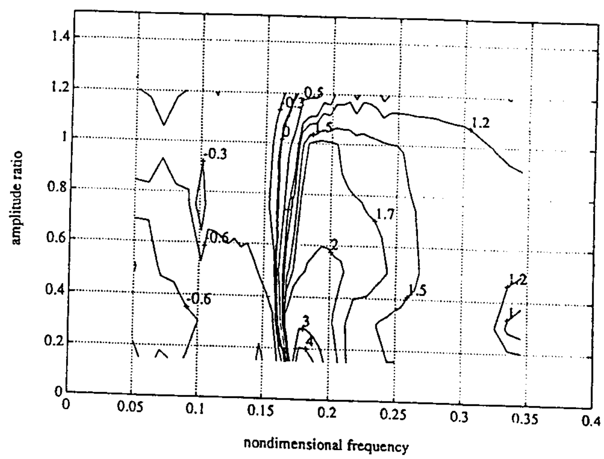


FIGURE 5.21 ADDED MASS (GOPALKRISHNAN 1993)

These data support the following observations:

- At frequencies well above lock-in, the damping coefficient seems to be consistent with typical drag coefficient data. In other words, the damping force due to transverse motion of the cylinder through a steady flow, when the motion is at a much higher frequency than the shedding frequency, is similar to drag force on a moving cylinder in still water. However, this picture is somewhat complicated by the ability of the shedding to lock-in at multiples of the fundamental frequency.

- At frequencies below lock-in, the damping coefficient also tends to be relatively well-behaved, and values are consistent with drag coefficient data.

- When the flow is locked-in to cylinder motion, it is not possible to separate damping from lift. The resulting force term proportional to cylinder velocity is very frequency and amplitude dependent.

A contour map of damping coefficient versus nondimensional amplitude and Strouhal frequency is presented in Figure 5.25. Negative values of damping coefficient represent positive damping force, given the author's definitions. The author chose to plot a quantity he calls $C_{L\_V_0}$ rather than a true damping coefficient, so the contour line values need to be converted via the following formula:

$$C_d = C_{L\_V_0} \frac{3U^2T^2}{32\pi A^2}$$

Note that at frequencies above and below the lock-in boundary, damping is very amplitude-dependent but relatively independent of frequency. These data suggest that a relatively simple damping model could suffice outside of lock-in. Within the lock-in region the data are much more complex, and a damping force model that depends on both frequency and amplitude may be necessary. A simple way to do this would be to interpolate damping coefficients directly from these (converted) data.

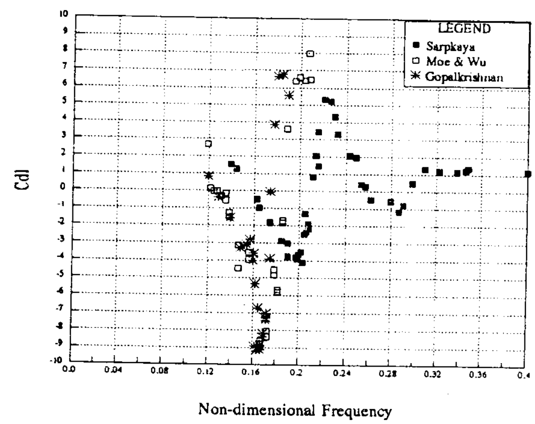


FIGURE 5.22 DAMPING COEFFICIENTS FOR A/D=0.25

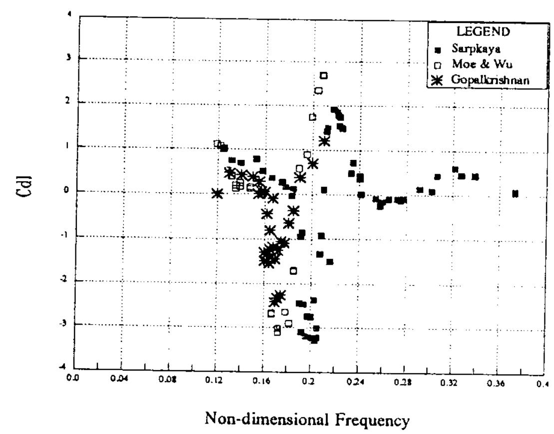


FIGURE 5.23 DAMPING COEFFICIENTS FOR A/D=0.50

5.5 Relative Velocity Model

In general, there are at least two basic empirical approaches to modeling the damping force. One approach is to simply measure the force in phase with cylinder velocity during oscillated cylinder tests under a variety of appropriate flow conditions, as discussed above in the section on oscillated cylinders. The other approach is to calculate the steady drag force using the relative velocity between structure and fluid, and take the transverse component of drag as damping [Chung (1987)]. This is generally referred to as the relative velocity model. This approach has the advantage of a wealth of data existing for steady flow drag coefficients, but requires the use of the "quasi-static" assumption.

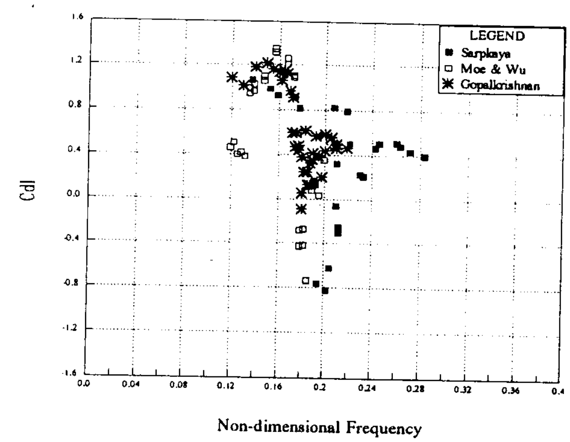


FIGURE 5.24 DAMPING COEFFICIENTS FOR A/D=0.75

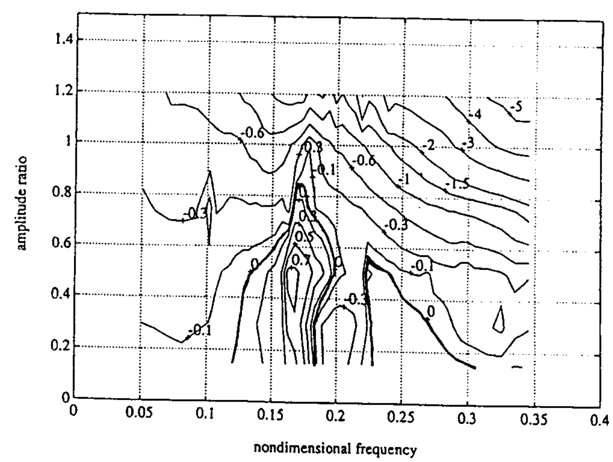


FIGURE 5.25 DAMPING (GOPALKRISHNAN 1993)

The quasi-steady assumption has been applied to a variety of fluid mechanics problems. The quasi-steady assumption states that dynamic fluid forces acting on an oscillating body can be calculated from static fluid forces measured on a stationary body. Under the quasi-steady assumption, the only effect of body motion is to modify the orientation of the incident flow vector. This concept is illustrated in Figure 5.26 for incident flow velocity V and cylinder velocity $\dot{z}$:

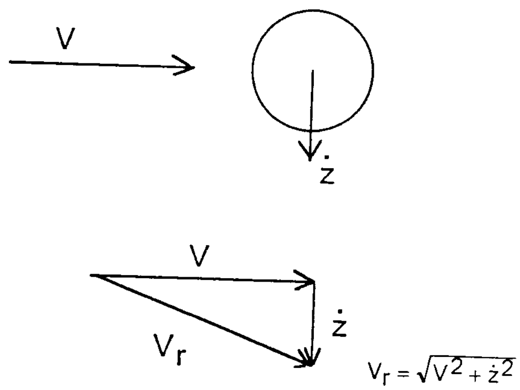


FIGURE 5.26 RELATIVE VELOCITY MODEL

Using relative velocity V_r and assuming the Morison form for drag force yields the following expression for damping in the transverse direction:

$$F_v = \frac{1}{2}\rho D C_d \left\{ \sqrt{V^2 + \dot{z}^2} \right\} \dot{z}$$

The weakness of this approach is that the quasi-steady assumption is only a reasonably good approximation if the pressure distribution around the cylinder rotates instantaneously with the resultant velocity vector (i.e. no phase lag), and that the pressure distribution is not affected by wake disturbances caused by cylinder oscillation. In general, the quasi-steady assumption has been shown to be rather poor at small values of reduced velocity. Blevins (1977) gives a qualitative explanation that indicates a minimum reduced velocity of ten is required for reasonable accuracy. However, other researchers have found the assumption to be poor for some applications out to reduced velocities of over 100 [Price, et. al. (1988)]. At best, it should be used with caution.

5.5.1 Drag Coefficient Versus Reynolds Number

Drag force varies with Reynolds number, response amplitude, surface roughness, and turbulence intensity. A few experimental studies were found in the literature which have studied these effects [Cheung et. al. (1983), Rodenbusch et. al. (1983), Sarpkaya (1977, 1979), Bearman (1984), and Schewe (1983), etc.]. These studies will provide the necessary information in forming empirical expressions for the damping coefficient.

A number of model tests have been published analyzing the effect of Reynolds number on drag force coefficient. Compilation of the data is presented in Figure 5.27. These data were collected from individual published studies and from other compiled data review studies [e.g. Sarpkaya et. al. (1981), Chen (1972), etc.]. Available information about the particular model test is provided in Table 5.4.

Chen (1972) compiled drag coefficient data in uniform flow. He compiled experimental data from a number of authors, including Schlichting (1968), Keefe (1961), and Son & Hanratty (1969). Their measured results agree reasonably well with each other. Chen observed that the skin friction drag plays a negligible role in the total drag at Reynolds numbers greater than $1\text{x}10^4$. However, its influence increases as the Reynolds number decreases. It reaches a value as 50 percent of the total drag at a Reynolds number of 5 [Chiu et. al. (1967)].

Batham (1973) performed model tests for $Re<2.6\text{x}10^5$ to measure the drag coefficient. At $Re=1.11\text{x}10^5$, he obtained a typical subcritical drag coefficient of 1.2. At $Re=2.4\text{x}10^5$, he measured drag coefficient of 0.77. Bearman (1969) in model tests at $Re=3.8\text{x}10^5$ found a drag coefficient of 0.24.

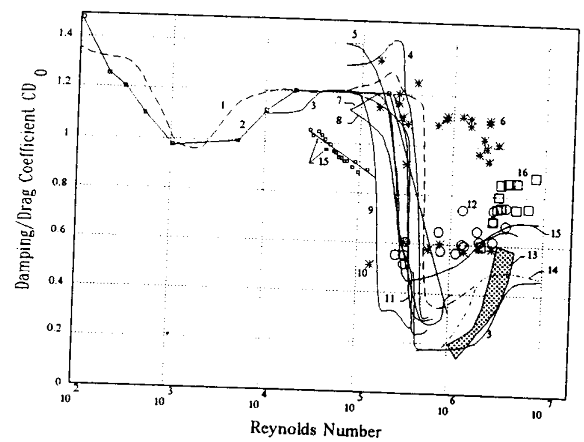


FIGURE 5.27 DRAG COEFFICIENT VERSUS REYNOLDS NUMBER

Rooney et. al. (1981) performed model tests using a low aspect ratio cylinder to study the vortex shedding patterns in a sheared flow, to measure the drag coefficient at transitional Reynolds numbers. They used a 6-in. diameter smooth cylinder. Their test cylinder exhibited a very low drag coefficient of 0.74 at $Re=1.8\text{x}10^5$, and a drag coefficient of 0.24 at $Re=3.5\text{x}10^5$. They concluded that their results did not compare favorably with the smooth cylinder drag coefficient data of Bearman (1969), and Batham (1973) because of the different geometrical characteristics of the cylinders (e.g., aspect ratio, different surface finish, etc.). Humphreys (1960) has shown that end effects dominate the circumferential flow pattern for Reynolds numbers up to $3\text{x}10^5$, at which point three dimensional turbulent effects overshadowed endplate influence.

TABLE 5.4 COMPILED DRAG COEFFICIENT DATA VS REYNOLDS NUMBER

Curve #	Authors	Low Re	High Re	Medium	Cylinder	Comments
1	Relf - Simmons	1.0E+02	2.0E+06	air	s	uniform flow, based on experiments by Wieseelberger
2	Sallet	1.0E+02	2.0E+06	air	s	uniform flow
3	Schewe	1.0E+04	7.0E+05	air	s	rigid cyl, dampened
4	Kwok	8.0E+04	8.0E+05	air	s	smooth, uniform flow, clamped - clamped
5	Achenbach	8.4E+01	8.0E+05	air	s	smooth, uniform flow
6	Rodenbusch - Gutierrez	1.8E+05	3.0E+06	water	s	Shell experiments, uniform flow, 0.02 roughness
7	Farell - Blesmann	8.0E+04	8.0E+05	air	s	smooth, uniform flow
8	Cheung - Melbourne	8.0E+04	8.0E+05	air	s	smooth, uniform flow, pressure taps
9	NPL	8.0E+04	5.0E+05			based on compiled data
10	Rodenbusch - Gutierrez	1.8E+05	3.0E+06	water	s	Shell experiments, uniform flow, 0.02 roughness
11	Guven et al.	2.0E+05	9.0E+05	water	s	smooth, uniform flow
12	Rodenbusch - Gutierrez	1.8E+05	3.0E+06	water	s	Shell experiments, uniform flow, smooth cyl
13	Schmidt	1.0E+06	6.0E+06	air	s	polished surface
14	Jones	5.0E+05	1.0E+07	air	s	smooth
15	Rajaona - Sulmont	2.0E+06	9.0E+06	water	o	forced oscillations, uniform flow, towed
16	Roshko	2.0E+06	8.0E+06	air	s	roughness = 0.0002, uniform flow

s = stationary
o = oscillating

Cheung et. al. (1983) used stationary cylinders in turbulent flow to measure the drag coefficient in wind flow at subcritical and supercritical Reynolds numbers. They used pressure taps to measure the circumferential pressure around the cylinder. The drag force was obtained by integration of the pressure. They found that mean drag coefficients decreased with turbulence intensity at subcritical Reynolds numbers and increased with turbulence at supercritical Reynolds numbers. The drop in drag coefficients occurred at a lower Reynolds number in turbulent flow than in smooth flow, indicating that there was a direct shift in the effective transitional Reynolds number due to turbulence. Nevertheless, they concluded that the total drop in drag is smaller in turbulent flow than in smooth flow.

Independent of model test conditions (fluid medium, value reported and test cylinder category) the drag coefficient follows similar qualitative characteristics, as seen in Figure 5.27. First, C_d decreases substantially with increase in Reynolds number up to 10^3. Both curves 1 and 2 explicitly captured this behavior. Then the drag coefficient remains almost constant, about 1.0, for a small range of Reynolds numbers up to approximately $7x10^3$, [curves 1, 2]. For further increase of Reynolds number the drag increases to 1.2 where it stays up to the critical Reynolds number ($1.5\text{-}2x10^5$).

As the Reynolds number enters the critical regime the drag coefficient drops to low values. The organized shedding disappears (boundary layer and wake change) in the critical flow regime, thus both the mean and the fluctuating drag drop to a lower value and then become relatively constant up to the supercritical flow regime [curves 1, 2, 3, 4, 5, 11, 13, 14, 15; points 9, 10, 12]. There is qualitative agreement within the compiled data in the critical regime. The drop in drag coefficient has been captured in most of the compiled test results. The exact Reynolds number range, however, where the drop occurs shows some discrepancy, similar to that of the occurrence of the maximum lift coefficient value. It appears that the drop of the drag coefficient occurs for Reynolds number between 10^5 - $2x10^5$. The drag coefficient value varies between 0.2 - 0.4. Slight variations in the flow turbulence and the test section roughness can justify this difference.

When the Reynolds number enters the supercritical range, values up to 10^6, C_d remains approximately constant, about 0.4 average value. The compiled data clearly show this trend. Most of the data in this regime have been produced from model tests in wind tunnel due to lack of test facilities to achieve these flow conditions in water. One exception is the tests performed by Shell Development Company [points 6, 10, 12], Rodenbusch et. al. (1983). Shell's tests used smooth cylinders under steady speed tow, and strain gages to measure forces. Points 10, and 12 represent cylinders with slightly different aspect ratios. The scatter is small about 5-10%. However, points 6 were derived from a highly roughened cylinder with surface roughness of 0.02. This is equivalent to a cylinder diameter 4% higher than the diameter of a smooth cylinder. In addition, surface roughness induces turbulence in the flow around the cylinder which increases the drag coefficient.

In the postcritical Reynolds number range C_d increases again. A limited number of model tests have been performed in this regime due to the lack of test facilities and equipment to attain high flow speeds and the very delicate flow conditions. Tests by Schewe (1983) [curve 3], Schmidt (1966) [cross-hatched area 13], Jones (1968) [curves 14], Rodenbusch [part of points 10, 12], and Roshko (1961) [points 16] are shown in Figure 5.27. All model tests were performed in a wind tunnel, except for Shell's tests [points 6, 10, 12] in water. The drag coefficient has the most scatter in this region. It varies between 0.2 - 0.7 at Reynolds number of 10^6, and 0.4 - 0.85 at the high Reynolds number of 10^7. Points 16 show the higher drag coefficient values as high as 0.85. However, the cylinder surface was not smooth but had a roughness of 0.0002. Shell's peak values are about 0.76 for the smooth cylinder.

Results from Schmidt tests were based on the drag force for an infinitely thin very smooth strip (pure 2-D flow conditions) in a wind tunnel. The drag coefficient was derived from pressure measurements and pressure integration.

The scatter is attributed to various causes. The most frequently causes are end effects and turbulence [Batham (1973), Cheung et. al. (1983), etc.]. The degree of rigidity of the mounting of the cylinders may also play a role [King (1977)] introducing cylinder oscillations at the end boundaries, and contaminating the wake and the vortex shedding. Three-dimensional effects where the vortices wrapped vertically along the cylinder length [Vickery (1962), Protos (1968), etc.] also affect the vortex shedding process and the pressure field around the test cylinder. The test cylinder aspect ratio can also affect the measured drag forces and contribute to the scatter of drag force measurements from different model tests.

Wide variation of drag forces, up to 25%, can be measured on apparently identical cylinders in different test facilities. The blockage ratio (ratio of cylinder diameter to width of test facility) may effect the flow characteristics and thus the drag coefficient [Modi et. al. (1975), Cheung et. al (1980), etc.]. Additionally, the force measuring method may influence the recorded values as it was discussed in previous paragraphs.

5.5.2 *Drag Coefficient Versus Response Amplitude*

It is well known that drag force varies with the amplitude of transverse vibrations. Therefore the natural extension of the relative velocity model is to utilize drag coefficient data from vibrating cylinders rather than stationary cylinders. Available test data were derived from model tests in the subcritical flow region ($Re<10^5$-$2x10^5$). No model tests could be located which were performed in the critical, supercritical or postcritical regions. Therefore, quantitative conclusions about the response amplitude effect on the drag coefficient are at best speculative in these Reynolds number ranges.

The drag amplification due to VIV was first discovered by Bishop and Hassan (1964). Blevins (1986), in reviewing VIV effects on hydrodynamic forces, concluded that the average (steady) drag on a cylinder vibrating at or near the vortex shedding frequency is a strong function of the vibration amplitude. Drag increased dramatically with transverse vibration amplitude, especially at resonance. The increase in drag together with VIV, can have important consequences for the design of marine pipelines, risers, and cables that are exposed to ocean currents.

Several investigators have proposed formulas for the increase in drag due to vortex-induced vibrations [Vandiver (1983), Skop et. al. (1977), Sarpkaya (1977)]. Most of these expressions are valid for the lock-in (resonance) case. At resonance, most of the known expressions give similar values (within 15%). The resonant amplitudes of vortex-induced transverse vibration of these structures is often 0.5 to 1.0 diameter with drag coefficients between 2.5 and 3.0 [Dale and McCandles (1967), Vandiver (1983), Torum and Anand (1985)].

Skop et. al. (1976) proposed an expression for the drag amplification as a function of the VIV amplitude and the response frequency. Their expression is also valid at off-resonance conditions. Their formula is:

$$(1+2Ay/D)(f/fs)>1,$$

where: f = cylinder vibration frequency
fs = stationary cylinder shedding frequency
Ay = VIV amplitude, and
D = cylinder diameter.

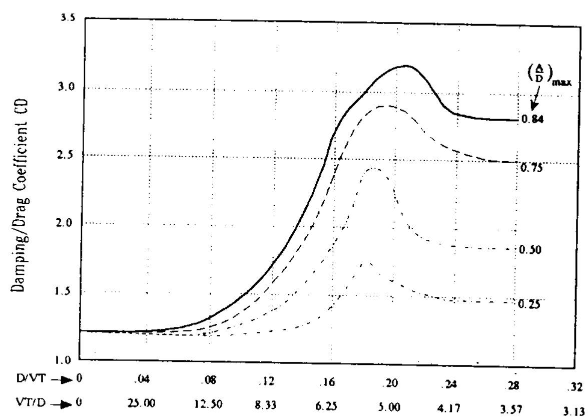


FIGURE 5.28 DRAG COEFFICIENT VERSUS AMPLITUDE (SARPKAYA 1977)

Sarpkaya (1977) has performed the most complete and consistent model tests to study the effect of VIV response amplitude and frequency on the drag amplification. In Sarpkaya's model tests, the frequency is defined as an equivalent Strouhal number D/V*To (where D=diameter, V=flow velocity, To=period of oscillation), based on the oscillatory frequency that was externally imposed on the cylinder. The model tests were performed in uniform water flow for oscillating cylinders in the subcritical Reynolds number range. The drag coefficient values were directly recorded during the model tests, and represent maximum values. Compilation of Sarpkaya's data is presented in Figure 5.28 showing the measured drag/damping versus equivalent Strouhal frequency at several maximum non-dimensional response amplitudes $\left(\frac{A}{D}\right)$.

6.0 Conclusions

The most essential element of utilizing data from the literature is consideration of all issues that can significantly affect test results including: the variety of assumptions and simplifications, the inconsistencies arisen by the researcher's implicit focus on one aspect of a very complicated problem, and the inconsistent use of terminology. The investment in the test data is so large, and the fundamental problem is so complex, that there was a great deal of incentive to derive the maximum benefit out of existing data to use in a VIV prediction model.

Reported experiments in the literature cover stationary and vibrating cylinders. Key issues identified for stationary cylinders that bear on the development of a practical predictive model are:

- The most important coefficients affecting VIV response are the lift coefficient, the correlation length, the vortex shedding frequency, and the vortex shedding frequency band width.

- The vast majority of the lift coefficient data is on stationary smooth cylinders in uniform flow. The considerable scatter in the lift coefficient data, especially in the subcritical Reynolds number region is attributable to variations in test methods, test media, instrumentation used, end-effects and boundary conditions.

- The correlation length effect on the lift coefficient C_L is crucial, rarely acknowledged in literature, and is likely the cause of the reported variation in lift coefficients. A simple model with constant lift coefficient C_L and varying correlation length Lc may be adequate to model VIV force on stationary cylinders.

- Correlation length tends to decrease with increasing turbulence.

- The mean vortex shedding frequency is generally at Strouhal number S = 0.2, independently of Reynolds number for smooth cylinders in uniform flow.

- Vortex shedding occurs in cells (a cell is a cluster of in-phase adjacent shedded vortices along the cylinder length) that may overlap. In shear flow, overlapping cells with different mean frequencies lead to a range of shedding frequencies at any given location (increased shear is associated with increased spectral bandwidth of the applied lift force). The frequency bandwidth of the VIV force is related to both turbulence and current shear profile due to overlapped random vortex cells.

Vibrating cylinder tests study fluid-structure interaction effects that become important in a VIV predictive model for long cylinders (risers) that will vibrate when placed in a flow stream. Fluid-structure interaction is a key consideration to predict accurately VIV response. The most important issues in fluid-structure interaction tests are:

- The forced displacement-excitation method (forced oscillated cylinders) permits reliable measurements of the fluid forces on the cylinder as a function of prescribed displacements to analyze fluid-structure interaction effects.

- The lift coefficient is amplitude dependent, but reported lift coefficients for oscillating cylinders represent the total hydrodynamic force, including some added mass and damping, and thus, make difficult the use of typical curves in the literature.

- Added mass, damping and lift force are often confused in literature. The added mass and damping forces on the structure are a critical part of dynamic equilibrium and because they are affected by the shedding process should be obtained from oscillated cylinder experiments to capture the fluid-structure interaction.

- Dramatic changes in lift force frequency, magnitude, and correlation length that always precede large amplitude vibrations may be viewed in terms of two broad categories of fluid-structure interaction: nonlocked-in response and locked-in response. Fluid-structure interaction effects may lead to large amplitude vibrations only if vortex-shedding is first locked-in to cylinder motion.

- In locked-in response, lift, added mass, and damping forces cannot be distinguished, and only amplitude and phase of the total hydrodynamic applied force can be determined. The proper hydrodynamic force magnitude and phase data from oscillated cylinder tests can be replicated in a traditional (added mass, damping, applied force) model by modifying the added mass term.

- The increase in correlation length Lc for locked-in shedding is dramatic, and increases with increasing amplitude. This correlation length increase is significantly moderated by the presence of turbulence and also at very high amplitudes because of the breaking up of the vortex cells at some limiting amplitude. If shedding is not locked-in, correlation length is insensitive to amplitude.

- Regions of locked-in shedding can extend over a node in a vibration shape resulting in forces that are negatively-correlated. Negative force correlation may be important to model locked-in response of modes beyond the fundamental mode.

- The best source for hydrodynamic forces on locked-in shedding appears to be the oscillated cylinder data, the most complete set of which has been published in a Ph.D. thesis [Gopalkrishnan (1993)].

The following steps seem appropriate to apply the above findings to modeling of riser VIV:

1. Identify "locked-in" regions, where flow and response interact.
2. In frequency-domain approaches use spectral characteristics of the local response like rms displacement, zero upcrossing frequency, and bandwidth to help identify locked-in regions.
3. Utilize stationary cylinder data for the force model on non locked-in segments.
4. Use data from oscillated cylinder experiments in model for hydrodynamic interaction at "locked-in" regions.
5. Account for spatial correlation in lift force model by using a constant lift coefficient and a correlation factor that depends on both Reynolds number and response amplitude and turbulence.
6. Account for both turbulence and spatial gradient of the ambient velocity profile in modeling spectral bandwidth of lift force.

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