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Practice for Establishing if a Structure will Undergo Vortex Induced Vibrations

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The PTS **20.062 (June 1987)** will be obsolete and removed from PTS binder / e-repository from herein onwards.

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1. INTRODUCTION

The analysis of the vulnerability of a structure (or one of its components) to Vortex Induced Vibration (VIV) in water is presented here in a simplified method. VIV is a complex phenomenon in that the loads depend on the displacements which in turn depend on the loads. As such, it does not easily lend itself to analysis as the mathematical modelling required is extremely complex. It is also a subject which has been, and is being, extensively researched, and fully acceptable methods for the evaluation of VIV have not (yet) been established. Nevertheless, an empirical technique has been developed which represents the state-of-the-art and serves to indicate whether or not VIV will occur. This technique is best used to demonstrate that VIV is **not** a problem as the in-built conservatism makes any predictions of response to VIV extremely severe.

The method, presented here, is based on the evaluation of a certain number of parameters. The values calculated for these parameters are then compared with experimental data to establish an 'upper-bound' estimate of the response. The method was compiled by D.T. Tsahalis of SDC as part of some advice provided to PETRONAS ([Ref. 1](#)) during 1984 and is an amalgamation of information available in the open literature and the in-house experience gained during SDC's research program into VIV over the period 1978 to 1984. The method is to be used in preference to the two Det Norske Veritas codes that give guidance on this subject ([Refs. 2 and 3](#)). The former DnV publication on offshore structures presents a method that is not considered to be valid as it appears to have no scientific foundation. The latter publication on submarine pipelines does not give sufficient explanation or cover sufficient ground.

In this report a definition of VIV is presented followed by a description of the vortex shedding process for a stationary cylinder. The phenomenon of VIV of a flexible cylinder is addressed in [Section 4](#) where the governing parameters are also introduced. [Section 5](#) presents the analysis method for the different fluid flow conditions for both in-line and cross-flow vibration. The analysis method is presented as a series of figures showing the interdependence of the governing parameters. Finally, [Section 6](#) discusses the simplification and assumptions used in the method and gives some recommendations for further work should the simplified method indicate VIV to be a problem. An example calculation is shown in the [Appendix](#).

2. DEFINITION OF VORTEX INDUCED VIBRATION

Vortex induced vibration of a structure, or a structural element, exposed to fluid flow, is the vibration caused by the unsteady drag and lift forces generated by the shedding of vortices.

The vortices themselves are 'rolled-up' portion of the shear layer that is formed as a fluid flows past a bluff body. Initially, at low Reynolds numbers, the shear layers are formed but still remain stable. As the Reynolds numbers increased (i.e. the fluid velocity increases) the shear layer becomes unstable. Since the innermost portion of the shear layer is moving more slowly than the outermost portion, which is in contact with the free stream, the shear layers 'roll-up' into discrete swirling vortices ([see Fig.2.1, from Ref. 4](#)). These vortices are then shed from alternate sides of the body. The unsteady pressure distributions associated with the vortex shedding process are discussed in the following section.

3. VORTEX SHEDDING FROM A STATIONARY CYLINDER

The pressure distribution created on a stationary cylinder during the generation and subsequent shedding of a vortex is shown in [Figure 3.1 \(Ref. 5\)](#). Whilst the effect of the wake pattern and the shear layer characteristics on the unsteady pressure distribution is complex, several simple observations may be made:

- 1) Large differences between the two shear layer lengths in the direction of the stream, prior to the vortex being shed, coincide with the lift force being generated (time steps 1, 2 and 4 in [Figure 3.1](#)).
- 2) At the actual moment of the vortex being shed (i.e. the 'fracture' of the shear layer) the pressure distribution on the cylinder is such that no lift force is generated (time step 3 in [Figure 3.1](#)).
- 3) The in-line force generated will have a steady component and an oscillating component. This oscillating drag component goes through one complete cycle for every vortex shed. Thus for every pair of vortices shed the oscillating lift component goes through one cycle whilst the drag component goes through two.

The frequency at which the vortex pairs are shed (f_s) is governed by the Strouhal number (S_t):

$$f_s = S_t V/D \quad (3.1)$$

where,

V = free stream velocity

D = member diameter

The Strouhal number for a circular cylinder is shown in [Figure 3.2 \(Ref. 6\)](#) as a function of Reynolds number. Comparing this figure with [Figure 2.1](#), it is apparent that when the flow regime is in the critical region (approx. $3.5 \times 10^5 < Re < 3.5 \times 10^6$), the vortex shedding frequency is not well defined and the Strouhal number varies considerably. However, for the purposes of this work, a Strouhal number of 0.2 will be adopted to represent the vortex shedding frequency for the flow regimes (subcritical and supercritical) where vortex streets are produced.

Thus equation 3.1 may be rewritten.

Frequency of shedding of vortex pairs (f_s):

$$f_s = 0.2 V/D \quad (3.2)$$

Frequency of shedding of individual vortices ($2 f_s$):

$$2 f_s = 0.4 V/D$$

Note: The Strouhal numbers for cross-sections that are not circular vary considerably from the 0.2 value (e.g. Table 5.1 in Ref. 7). It should be noted expressly that, although the procedure described in this Practice is likely to be more generally valid, all data presented here and for the empirical analysis method in [Section 5](#) are specifically for circular cylinders.

4. VORTEX SHEDDING FROM A FLEXIBLE CYLINDER

So far the discussion has been limited to stationary cylinders. For a flexible cylinder (or a spring- supported rigid cylinder) whose natural period of vibration is close to the vortex shedding frequency, the motion of the cylinder will dominate the vortex shedding process. Thus the shedding frequency (f_s) loses its dependence on the Strouhal (St) number and becomes equal to the natural frequency of vibration (f_n). This is known as the 'lock-in' effect. The motion of the cylinder also increases the spanwise correlation of the vortex shedding process, which further increases the motion. This 'interactive' nature of VIV means that an analytical prediction of response becomes extremely difficult.

However, an empirical technique based on the evaluation of a limited number of parameters has been developed and is discussed in the following sections. This technique depends on the comparison of specific parametric values with those obtained experimentally. The experimental results used relate to a test cylinder which has been excited to VIV by fluid flow acting over its entire length. The treatment of a partially submerged cylinder is described in the calculation of the effective mass parameter. This parameter and others required to perform the VIV analysis are introduced below.

Reduced velocity (V_r)

$$V_r = V/f_n D \quad (4.1)$$

where

V = free stream velocity

f_n = natural frequency of vibration in still fluid

If the Strouhal relationship given as equation 3.1 was valid for vibrating cylinders, then cross-flow resonant response ($f_s = f_n$) would occur for a reduced velocity of:

$$V_r = 1/0.2 = 5.0 \quad (4.2)$$

and in-line resonant response ($2 f_s = f_n$) for a reduced velocity of:

$$V_r = 1/0.4 = 2.5 \quad (4.3)$$

As will be seen later these values of V_r do indeed act as indicators for the type of response produced at various values of reduced velocity but do not give the full picture due to the interaction effects discussed above (see Section 5).

Stability parameter (k_s)

$$k_s = 2 m_e \delta / \rho D^2$$

where

m_e = effective mass/unit length

ρ = fluid density

δ = logarithmic decrement of structural damping = $2\pi\beta$

β = ratio of damping to critical damping, where only damping components which are not associated with hydrodynamic damping should be included (generally speaking β is of the order of 0.01)

Effective mass (m_e)

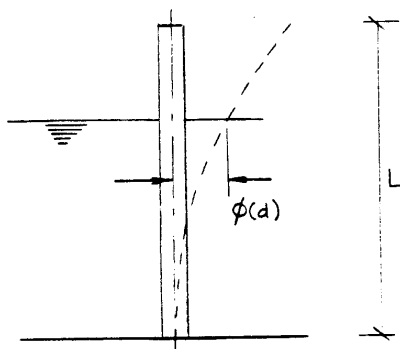
For a totally submerged member the effective mass parameter is the sum of the structural, entrained and added mass per unit length. For a partially equivalent structure that has the same cross-section as the excited structure and a length. Both have the same mode shape and natural frequency of vibration.

In this way, the total amount of kinetic energy in the 'real' vibrating structure is made equal to the amount of kinetic energy in the effective structure. The effective mass per unit length is given by :

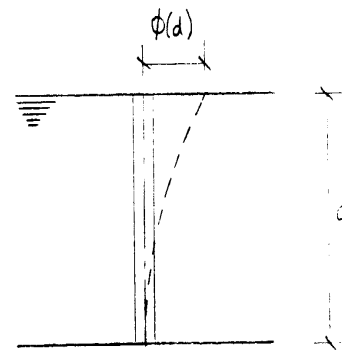
$$m_e = \int_0^L m(x) \phi^2(x) dx / \int_0^d \phi^2(x) dx \quad (4.5)$$

where

$m(x)$ = total mass per unit length
 L = total length
 d = waterdepth
 $\phi(x)$ = mode shape



Mass /unit length $m(x)$
full mode shape



Mass/Unit length m_e
mode shape truncated at free surface

Mode shape parameter (γ)

A large amount of experimental work into cross-flow VIV in current has been done with spring supported rigid cylinders. In an attempt to extrapolate these experimental results to the analysis of flexible cylinders, the mode shape parameter has been developed ([Ref.8](#)):

$$\gamma = \phi_{\max} \left| \int_0^L \phi^2(x) dx / \int_0^L \phi^4(x) dx \right|^{1/2} \quad (4.6)$$

where

$\phi_{\max}$ = the maximum value of $\phi(x)$, the mode shape

The application of these parameter is described in the following section.

5. EMPIRICAL ANALYSIS METHOD

An empirical method for assessing a structure's response to VIV is presented in this section and comprises the following two steps:

- 1) assess whether the incident flow velocity is sufficient for VIV to occur;
- 2) determine the resultant maximum amplitude of vibration.

As stated previously, all quantitative data presented for use in the analysis are specifically for circular cylinders.

It is useful here to distinguish between the different types of flow regime that may be encountered either experimentally or practically.

- Flow regime A: Steady flow with a uniform velocity profile along the length of the cylinder (uniform current).
- Flow regime B: Steady flow with a non-uniform velocity profile (non- uniform current).
- Flow regime C: Oscillatory flow with a uniform velocity profile.
- Flow regime D: Oscillatory flow with a non-uniform velocity profile.
- Flow regime E: Oscillatory flow with elliptical water particle motions and a non-uniform velocity profile (regular waves).

Flow regime A is the one that has been used by the majority of researchers in the field of VIV and is the one for which the reduced velocity parameter and stability parameter were originally developed. The extrapolation of the experimental results from flow regime A to the calculation of displacement amplitudes in flow regime B is the subject of a separate report ([Ref. 9](#)). This extrapolation relies on the use of an enhanced stability parameter.

The calculation of VIV in irregular waves is not possible using the empirical method presented here. Instead, an attempt is made to assess VIV in regular waves (flow regime E) but even this is subject to considerable uncertainty. The experimental data available for such a calculation have been obtained under flow regime C and these data are used in their original state ([Ref. 10](#)). Ideally the extrapolation performed for the steady flow data by means of an enhanced stability parameter could also be performed for oscillatory flow data. However, that would still leave the data in a form suitable for flow regime type D analysis. The extrapolation from D to E would still need to be performed.

Thus, whilst the parametric method for analysis of VIV in current may be considered to be reasonably well founded in experimental results, the method for analysis of VIV in waves is extremely approximate and should be treated as such.

Cross-flow VIV produces much larger amplitudes of vibration in 'ideal' conditions than in-line VIV. Thus the discussion below is specifically orientated towards the prediction of cross-flow VIV. Experimental data for in-line VIV in uniform currents is extremely limited and 'test model' specific, i.e. in the literature on this subject no attempt has been made to generalise the in-line VIV data with a mode shape parameter as has been done for the cross-flow data (see discussion on the definition of γ).

5.1 The maximum amplitude of displacement due to cross-flow VIV in current

The maximum vibration amplitudes reached by a cylinder as it vibrates in the cross-flow direction have been found to occur at values of reduced velocity between 6.5 and 8 ([Ref. 11](#)). To ensure that the simplified analysis technique presented here does not produce unconservative results the maximum vibration amplitudes are considered to occur at values of reduced velocity between 6.0 and 9.0. The onset of cross-flow VIV is represented as occurring at a value of reduced velocity of 4.0 (see [Fig. 5.1](#)). It is important to realise that [Figure 5.1](#) is not supposed to represent the range of reduced velocity over which maximum cross-flow VIV will occur for any one cylinder. Instead it is taken to be a conservative estimate of the envelope of vibration amplitudes that could occur for a whole range of cylinders whose maximum vibration amplitudes happen at different values of reduced velocity.