Technical Note

Damping forces on a cylinder oscillating in a viscous fluid.

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To study the damping crisis encountered in offshore structures composed of circular members, the damping forces acting on a harmonically oscillating circular cylinder were determined. An experimental setup, in which representative values of the shear-wave number together with a sufficient amplitude-to-diameter ratio can be met is very difficult to realize. This might explain the lack of experimental data so far.

1. INTRODUCTION

This note deals with a hydroelastic problem encountered in offshore structures composed of members with a circular cross section of relatively large diameter D. These structures are subject to unsteady loads caused by current, waves and wind. The resulting response depends to a large extent on the damping stemming from the interaction of the structure with the surrounding fluid. Brouwers & Meijssen[1] indicated that a considerable uncertainty exists regarding the amount of damping generated in the transition region from Stokes' solution[2], valid for vibrations of very small amplitude-to-diameter ratio X/D, to fully separated flow, valid at X/D= O(1).

2. FORMULATION OF THE PROBLEM

To apprehend the problem, consider a circular cylinder oscillating sinusoidally in a viscous fluid at rest. The relation given by Morison et al.[3] is widely used to describe the in-line forces acting on the cylinder; per unit length:

$$FM = -1/2C_d\rho Du \mid u \mid -1/4C_m\rho\pi D^2\dot{u} \qquad (1)$$

where C_d is the drag coefficient, ρ is density of the surrounding fluid, u is velocity of the relative motion and Cm is the inertia coefficient. A dot denotes time wise differentiation. The first term on the right-hand side is the damping force, non-linear in the velocity of the cylinder; the second term represents the inertia force. The quadratic damping term is expressed in an equivalent viscous damping coefficient BM, made dimensionless with Stokes' solution BSt.

$$\xi = \frac{BM}{BSt} = \frac{4\sqrt{2}Cd}{3\pi}(X/D).S \qquad (2)$$

ξ not only depends on X/D but also on the shear-wave number $S= 1/2\, D\sqrt{\omega/\nu}$, representing the ratio of the cylinder diameter to the thickness of the unsteady boundary layer. In offshore applications, $S= O(10^3)$, thus damping in the case of separation is considerably larger than damping according to Stokes' solution (Figure 1).

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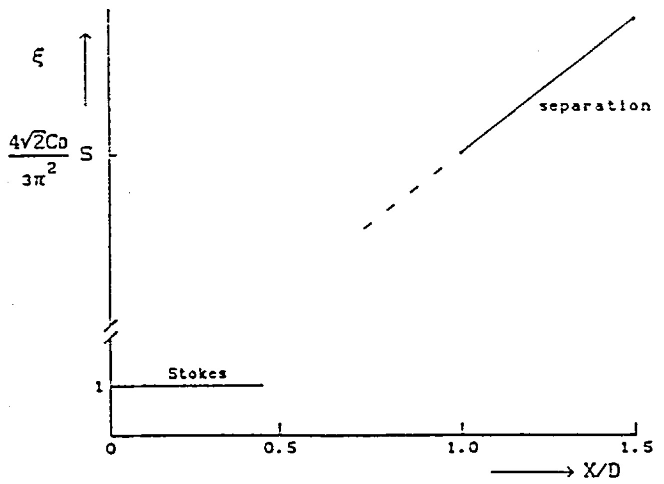

Figure 1. Dimensionless damping coefficient versus amplitude-to-diameter ratio.

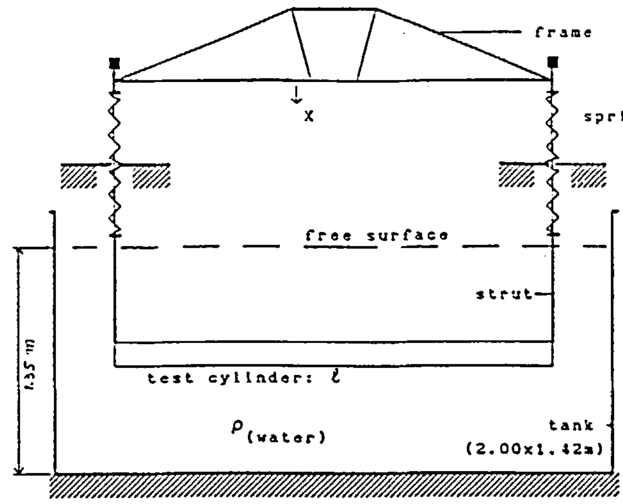

Figure 2. Schematic of experimental setup.

Values for C_d and C_m can be derived from the considerable amount of experimental data, see e.g. Sarpkaya[4] and Sarpkaya & Isaacson[5]. However, an experimental setup, in which representative values of S together with a sufficient amplitude-to-diameter ratio can be met is very difficult to realize. This might explain the lack of experimental data in literature for the parameter values considered in present work.

3. EXPERIMENTAL SETUP.

In a special designed test rig at the University of Twente, tests have been performed with an oscillating cylinder in still water, in which an interesting range of the mentioned parameters (S= 300, X/D up to 0.7) could be covered. The test cylinder (D= 0.1 m, relative surface roughness k/D= 0.14 μm) was suspended by four matched coil springs, two at each end (Figure 2). Via the frame, the cylinder could be driven by an electro-mechanical shaker. By measuring the exciting force and the resulting acceleration the concept of Frequency Response Functions was adapted to determine nonlinear effects (i.e. amplitude dependent damping) for X/D up to 0.15. Above this value free vibration tests were performed, by releasing the cylinder from an initial displacement and measuring the accelerations. Two different cylinder lengths were used (l= 0.75 m and l= 1.5 m) to enable elimination of end-effects by subtracting the results. Both tests with and without end-plates (O= 0.5 m) were carried out. For more details the reader is referred to Otter[6].

4. RESULTS AND DISCUSSION.

The results are presented as the difference of the viscous damping coefficient of the long and the short cylinder, divided by the associated Stokes' solution, thus yielding the dimensionless damping parameter ξ introduced in (2):

$$\xi = \frac{Bl - Bs}{BSt} = \frac{Bl - Bs}{(ll - ls)\pi\rho D\sqrt{2\nu\omega}} \qquad (3)$$

where Bl and Bs refer to the measured viscous damping coefficient for respectively the long and the short cylinder. First, the results of the free vibration experiments are presented in Figure 3.

This figure shows that above X/D~ 0.3 damping deviates from Stokes' solution. Although the results from the free decay experiments are rather crude, especially at larger amplitudes of vibration, they indicate that for X/D~ 0.7 damping is considerably lower than according to the Morison equation, i.e. separation damping. Next, the lower left corner of Figure 3 is magnified and the free vibration results below X/D= 0.3 are compared with the results from the constrained vibration experiments (Figure 4).

Figure 4 illustrates that the results from the free vibration tests, carried out with and without end-plates, link up well with the results from the constrained vibration experiments. No significant changes are found at the critical amplitude-to-diameter ratio following from the stability analysis by Hall[7], $(X/D)_{cr} = 6.0 \; 10^{-2}$. Considering their small magnitude, Figure 3 leads to the conclusion that, for X/D< 0.3 and S= 310, damping forces agree quite well with Stokes' solution.

It appears that the results of the experiments without end-plates are closer to Stokes' solution than the results of the tests with end-plates. For small amplitudes, when the

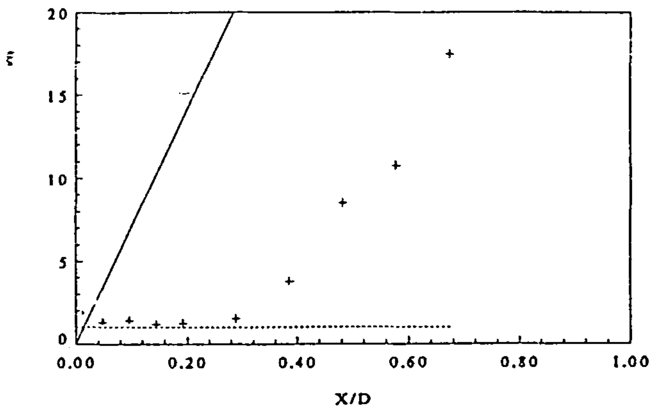

Figure 3. Dimensionless damping parameter for free vibration tests (+). -- -- -- unit (Stokes' solution), ——— drag damping (Cd = 1.2).

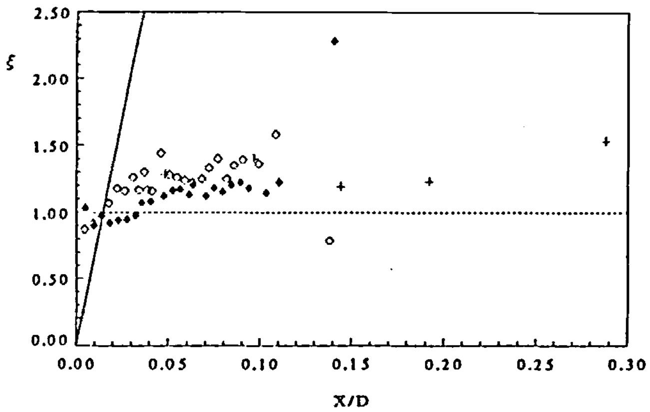

Figure 4. Dimensionless damping parameter for small amplitudes of vibration. -- -- -- unit (Stokes' solution), —— drag damping (Cd= 1.2), + free vibration, constrained vibration with end plates, constrained vibration without end-plates.

three-dimensional flow around the cylinder ends will be limited, this is likely to occur as the contribution of the end-plates to the damping forces was relatively large and susceptible to alignment errors.

During the constrained vibration experiments (X/D< 0.15), the fluid added mass coefficient for the cylinder was found to be independent of vibration amplitude: CM= 1.09 + 2%. This is somewhat larger than according to Stokes' solution at S= 310: CM= 1.01.

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