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Numerical simulation of a freely vibrating circular cylinder with different natural frequencies

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ABSTRACT

This paper deals with the numerical simulation of low-Reynolds-number flow around a freely vibrating circular cylinder in two-degrees-of-freedom. The governing equations are written in a non-inertial system fixed to the moving cylinder and solved using finite difference method. The natural frequency of the cylinder is chosen to be constant, agreeing with the vortex-shedding frequency for a stationary cylinder at Reynolds number Re_0. Systematic computations are carried out for $\mathrm{Re}_0=80$, 100, 140 and 180 keeping the mass ratio and structural damping coefficient at $m^*=10$ and $\zeta=0$. The effect of Re_0 on the root-mean-square (rms) values of cylinder displacements and drag coefficients is analyzed. Plotting the data set belonging to different Re_0 values against $U^*\mathrm{St}_0$ makes comparison easier. Local extreme values are found in the rms of streamwise displacement and drag coefficient in the range $U^*\mathrm{St}_0=0.4$–0.65. In the vicinity of $U^*\mathrm{St}_0=0.5$ the rms of drag approaches zero and the phase angle between the *x* component of the motion and drag changes abruptly from 0° to 180°. The pressure drag coefficient seems to be responsible for the sudden change. The cylinder follows a distorted figure-eight path in most cases investigated and its orientation changes from clockwise to counterclockwise orbit at around $U^*\mathrm{St}_0=0.5$.

1. Introduction

Flow around a circular cylinder is extensively studied due to its practical importance, using both experimental and numerical approaches. The flows are usually classified using Reynolds number based on free stream velocity U_∞, cylinder diameter d and fluid viscosity ν. For stationary cylinders the flow is steady below $\mathrm{Re}\cong 47$ and twin vortices are attached to the body. At around $\mathrm{Re}=47$ Hopf bifurcation occurs, resulting in an unsteady flow of periodic vortex shedding (Thompson and Le Gal, 2004). Risers, pipes, and underwater structures are good examples of this phenomenon. Periodic vortex shedding from the body can induce high amplitude oscillations, which can cause serious damage to the structure; this phenomenon played an important role in the collapse of Tacoma Narrows Bridge in 1940. Damage to thermometer cases at the Monju fast-breeder nuclear power plant in 1995 leading to a major shutdown of the entire facility was also due to periodic vortex shedding (Nishihara et al., 2005). On the other hand, mechanical energy transferred between the fluid and the moving body can also be beneficial. Possibilities of energy harvesting have been studied e.g. by Bernitsas et al. (2008, 2009) and Mehmood et al. (2013).

Barkley and Henderson (1996), applying linear stability analysis, showed that the flow around a stationary cylinder is two-dimensional (2D) up to $\mathrm{Re}\cong 189$. Three-dimensional (3D) instability occurs at $\mathrm{Re}\cong 189$ (Mode-A) and at $\mathrm{Re}\cong 259$ (Mode B). Thus, the application of a 2D computational code above $\mathrm{Re}=189$ is not justified for a stationary cylinder. For vibrating cylinders, however, experiments by Bearman and Obasaju (1982) and Koide et al. (2002) and numerical simulations by Poncet (2002) showed that synchronization (or lock-in) between vortex shedding and cylinder motion increases the two-dimensionality of the flow compared to the case of a stationary cylinder. The upper limit of the two-dimensionality has not been determined due to the large number of influencing parameters.

For the prediction of aerodynamic forces acting on a freely vibrating cylinder researchers often use a forced or controlled oscillation model. This approach is a simplifying model and is often chosen because no equations need to be solved for the cylinder motion. A large number of papers deal with forced oscillation in one-degree-of freedom (1DoF) cylinder motion, where the cylinder is typically restricted to move only in transverse direction (e.g. Williamson and Roshko, 1988; Lu and Dalton, 1996; Meneghini and Bearman, 1997; Kaiktsis et al., 2007; Baranyi and Daróczy, 2013; Tang et al., 2017) or in streamwise direction (e.g. Okajima et al., 2004; Al-Mdallal et al., 2007; Mureithi et al., 2010). There are

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Nomenclature

b	damping [kg/s]
CCW	counterclockwise orbit on the upper loop of figure-eight
C_D	total drag coefficient, $2F_D/(\rho U_\infty^2 d)$ [−]
C_{Dp}	pressure drag coefficient [−]
C_{Dv}	viscous drag coefficient [−]
C_L	total lift coefficient, $2F_L/(\rho U_\infty^2 d)$ [−]
C_{Lp}	pressure lift coefficient [−]
C_{Lv}	viscous lift coefficient [−]
CW	clockwise orbit on the upper loop of figure-eight
d	cylinder diameter, length scale [m]
D	dilation, non-dimensionalized by U_∞/d
DoF	degrees of freedom
F_D	drag per unit length of the cylinder [N/m]
F_L	lift per unit length of the cylinder [N/m]
F_N	reduced natural frequency, $f_N d/U_\infty$ [−]
$f^*_{x,y}$	oscillation frequency in x or y directions, respectively, non-dimensionalized by d/U_∞
f_N	natural frequency of the cylinder [1/s]
f_v	vortex-shedding frequency for a stationary cylinder [1/s]
k	spring constant [kg/s^2]
K	coefficient between Reynolds number and reduced velocity for constant natural frequencies, $f_N d^2/\nu$ [−]
m	cylinder mass per unit length [kg/m]
m^*	mass ratio, $4m/(d^2\pi\rho)$ [−]
p	pressure, non-dimensionalized by ρU_∞^2
R	radius, non-dimensionalized by d
rms	root-mean-square value
Re	Reynolds number, $U_\infty d/\nu$ [−]
St	dimensionless vortex shedding frequency, Strouhal number, fd/U_∞
St$_0$	dimensionless vortex shedding frequency for a stationary cylinder at Reynolds number Re$_0$
t	time, non-dimensionalized by d/U_∞
u, v	velocities in x and y directions, non-dimensionalized by U_∞
U_∞	free stream velocity, velocity scale [m/s]
U^*	reduced velocity, $U_\infty/(f_N d)$ [−]
x, y	Cartesian coordinates, non-dimensionalized by d
x_0, y_0	cylinder displacement in x and y directions, non-dimensionalized by d
ζ	structural damping coefficient, $b/(2\sqrt{mk})$ [−]
Θ	phase angle between streamwise and transverse components of the cylinder motion [−]
ν	kinematic viscosity of the fluid [m^2/s]
ξ_{max}, η_{max}	number of grid points in peripheral and radial direction, respectively
ρ	fluid density [kg/m^3]
Φ	phase angle between x_0 and C_D [−]
Φ_p	phase angle between x_0 and C_{Dp} [−]
Φ_v	phase angle between x_0 and C_{Dv} [−]

Subscripts

L	lift
D	drag
max	maximum value
rms	root-mean-square value
n	component in the direction normal to the cylinder surface
pot	potential flow
1, 2	on the cylinder surface, at the outer boundary of the domain
0	refers to cylinder response (x_0, y_0) or to a stationary cylinder (Re$_0$, St$_0$)

relatively few papers dealing with two-degree-of-freedom (2DoF) forced motion (e.g. Jeon and Gharib, 2001; Stansby and Rainey, 2001; Baranyi, 2008; Peppa et al., 2016).

Another approach to the investigation of vortex-induced vibrations (VIV) involves an elastically supported cylinder model, where cylinder displacement is caused by lift and drag forces acting on the body. A large number of studies have dealt with this model, including Bishop and Hassan (1964), Bearman (1984, 2011), Sarpkaya (1995, 2004), Jauvtis and Williamson (2004), Williamson and Govardhan (2004), Blevins (1990), Moe and Wu (1990), and Nakamura et al. (2013). Cylinder response is highly influenced by free stream velocity U_∞, natural frequency of the body f_N, structural damping b, and the mass of the body m.

Vibrations due to vortex shedding are often modeled with 1DoF transverse-only motion. Khalak and Williamson (1999) investigated the VIV of a transversely oscillating cylinder and showed that the mass-damping parameter $m^*\zeta$ strongly influences the peak amplitude, where m^* is the mass ratio (the ratio of the mass of the vibrating body and that of the displaced fluid) and ζ is the structural damping coefficient. It was shown that at low $m^*\zeta$ three branches of cylinder response occur, namely initial, upper and lower branches, where the upper branch is associated with the highest oscillation amplitude. Feng (1968) studied high mass-damping cases where only two branches (an initial branch with low cylinder displacements and lower branch with high vibration amplitudes) are observed. Brika and Laneville (1993) and Govardhan and Williamson (2000) distinguished between the different branches based on their vortex-shedding modes. The initial branch is associated with 2S mode (two single vortices are shed in each motion cycle) while 2P mode (two vortex pairs shed in each motion cycle) belongs to the lower and upper branches. Brika and Laneville (1993) found that the transition between upper and lower branches is hysteretic and the flow is quite sensitive to incremental changes in the reduced velocity.

Klamo et al. (2006) investigated how the system transitions between two-branch and three-branch responses. In their study Reynolds number and the structural damping were varied. It was concluded that $m^*\zeta$ alone is insufficient to predict the type of response; Reynolds number is also an important influencing parameter. For small damping and high Re cases a three-branch response was observed, while a two-branch response was found for high damping and low Re cases.

Naturally, structures are not restricted to move only in one direction; in most cases two-degrees-of-freedom (2DoF) oscillations are found. Jauvtis and Williamson (2004), using an elastically supported cylinder, kept the natural frequencies identical in the two directions ($f_{Nx} = f_{Ny} = f_N$) and investigated a wide mass ratio range ($m^* < 25$) using an experimental approach. It was found that at high m^* cases ($m^* = 6$–25) in-line oscillation has only a tiny effect on transverse vibration, which was also found by Zhou et al. (1999) at low Reynolds numbers using numerical techniques. Jauvtis and Williamson (2004) found that a three-branch response occurs, as in 1DoF cylinder oscillation. Upon decreasing the mass ratio below $m^* = 6$ dramatic changes were observed. The existence of a super-upper branch was reported where the vortex-shedding mode was 2T type – two triple vortices shed in each vibration period.

However, in general, the natural frequencies in streamwise and transverse directions are not identical, $f_{Nx} \neq f_{Ny}$. Sarpkaya (1995) experimentally investigated 2DoF vortex-induced vibrations varying the natural frequency ratio f_{Nx}/f_{Ny} between 1 and 2. These results were compared with 1DoF cylinder oscillation results for $f_{Nx} = f_{Ny}$ and a 19% increase was observed in the transverse oscillation amplitude. In addition, two obvious peaks were identified at $f_{Nx} = 2f_{Ny}$. Dahl et al. (2006) showed that by increasing the ratio of streamwise and transverse frequencies in the range of $f_{Nx}/f_{Ny} = 1$–1.9, the phase angle between x and y