Optimized Drag Reduction and Wake Dynamics
Associated with Rotational Oscillations of
a Circular Cylinder

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Abstract

The drag reduction on a circular cylinder through rotational oscillations is globally maximized by coupling a CFD solver with a novel parallel global deterministic optimization algorithm. The simulations are performed at a Reynolds number of 150 and the amplitude and frequency of the rotational oscillations are respectively constrained to the ranges $0.1 \leq \Omega \leq 1.0$ and $0.1 \leq f_\Omega \leq 3.0$. The novelty of this work lies in the use of a massively parallel derivative free optimization algorithm to find the globally (not locally, as is typically done) optimal values of
the rotational amplitude and frequency for minimum drag. Rotational oscillations significantly affect the pattern of shedding vortices and the mean drag is reduced at high forcing frequencies in relation to the shedding frequency. The results also show that there is a threshold of the rotational oscillation amplitude below which the mean drag is not reduced for any excitation frequency. The wake dynamics associated with the globally optimized and nonoptimized configurations are presented. The contributions of the viscous and pressure components to the total drag in these configurations are also determined and discussed.

**Keywords:** Rotational oscillations, VTDIRECT95, Global optimization, Synchronization, Critical amplitude/frequency, Drag reduction

1 Introduction

The flow behind a circular cylinder exhibits an organized and periodic motion of a regular array of concentrated vorticity that sheds from the cylinder. For a stationary cylinder, these vortices are shed with a specific frequency represented by a nondimensional Strouhal number $St$. This vortex shedding results in oscillatory forces on the body, which can be decomposed into lift and drag components along the cross-flow and in-line directions, respectively ([24, 33, 6]). Modifying the shedding mechanism or controlling the motion of the cylinder in an appropriate manner is thus a way of controlling the fluid forces.

Many active and passive control mechanisms for the vortex shedding have been explored for the purpose of reducing the forcing components. For instance, splitter plates have been used to modify the shedding patterns ([16, 31]). Active controls such as suboptimal blowing and suction ([19, 3]), acoustic feedback ([7, 14]), inline and transverse oscillations ([28, 18, 27]), and rotational oscillations ([30, 8, 26, 23, 29]) have also been proposed. [3] observed a suppression of the fluctuating forces and obtained more than 40% reduction in the mean drag coefficient using a couple of suction actuators on a stationary cylinder. [7] experimentally observed that the shedding frequency and forces can be altered by introducing sound waves with frequencies close to the natural vortex shedding frequency. [14] introduced local sound waves into the flow to influence the shear layer on one side of the cylinder. Suppression of the vortex shedding was observed by the destructive interference of the two shear layers. [27] observed a reduction in the drag force for a moderately spaced staggered arrangement of circular cylinders.

[30] performed experiments on a rotationally oscillating circular cylinder in a steady uniform flow at Reynolds number $Re = 15,000$. They observed a reduction in the oscillatory component of the drag up to 80% for optimal amplitude and frequency of oscillation. [29] experimentally investigated the effect
of rotatory oscillations at low Reynolds number Re = 150. They observed that the forcing parameters (oscillation amplitude and frequency) affect the pattern of vortex shedding. Numerical studies on the flow past an oscillating rotating cylinder have been performed by [26] over a range of 150 \( \leq \text{Re} \leq 15000 \). Their results are in qualitative agreement with those obtained experimentally by [30]. They observed that drag reduction is mainly caused by the generation of multipole vorticity structures. These structures are generated as a result of bursting of the boundary layer, triggered by appropriate rotati-

on oscillations. They observed that this phenomenon leads to an overall separation delay and thus drag reduction. They also found that the impact of rotational oscilla-

tions depends strongly on the Reynolds number and would be effective only if Re \( \geq 3000 \) because the viscosity suppresses the generation of multipole vortic-

ty in the case of low Reynolds number. The Reynolds number dependence for mean drag reduction was also observed by [8]. They studied the effects of rotational oscillations in an unsteady laminar flow past a circular cylinder at Re = 100. They varied the rotational speed and frequency in the ranges 0.2 \( \leq \Omega \leq 2.5 \) and 0.02 \( \leq f_\Omega \leq 0.8 \), respectively. They found a reduction in the mean drag of 12% for Reynolds number 100 and also performed numerical simulation at Reynolds number 1000 and found a reduction in the mean drag of 60%. They observed that the lock-on frequency range becomes wider as rotational speed increases. They also determined that local minimum points for the mean drag are near the boundaries of the lock-on regions. [23] numerically investigated the flow over a cylinder undergoing rotational oscillations at Re = 150. They observed a reduction in the mean drag coefficient of 25% at the higher forcing frequencies. They also suggested that the mean drag coefficient does not monotonically decrease as the forcing frequency is increased, but there is an optimal forcing frequency beyond which the mean drag again increases. The value of this optimal frequency depends on Reynolds number and rotational amplitude. More recently, [25] numerically and experimentally studied the suppression of aerodynamic forces through rotational oscillations. They observed a complete suppression of vortex shedding phenomena at a relatively high rotation rate. [5] and [21] showed the existence of multiple lock-in regions in the flow past a circular cylinder subjected to a circular motion. [20] showed that the flow topology can be significantly affected by sinusoidal rotations of a cylinder placed in a flow.

The present study aims to globally optimize the drag reduction on a circular cylinder through rotational oscillations, and understand the wake dynamics that are associated with maximum drag reduction. Beyond merely sampling in the parameter space, or following a descent direction, a parallel global optimization code VTDIRECT95 is coupled with a CFD solver to find the rotational amplitude and frequency that globally minimize the mean value of the drag. Section 2 discusses and validates the numerical methodology used to sim-
ulate the flow past a rotating cylinder in an unsteady uniform flow. Section 3 presents the optimization algorithm VTDIRECT95 and its interface with the CFD solver. A discussion of numerical results and the frequency/amplitude response data is in Section 4, and Section 5 concludes.

2 Numerical Simulations

2.1 Numerical Scheme

In the CFD solver, the time dependent incompressible continuity and Navier-Stokes equations are solved in the generalized coordinates (ξ, η). These equations are written in a strong conservative form as

\[
\frac{\partial U_m}{\partial \xi_m} = 0, \tag{1}
\]

\[
\frac{\partial (J^{-1}u_i)}{\partial t} + \frac{\partial F_{im}}{\partial \xi_m} = 0, \tag{2}
\]

where the flux

\[
F_{im} = U_m u_i + J^{-1} \frac{\partial \xi_m}{\partial x_i} p - \frac{1}{Re_D} G^{mn} \frac{\partial u_i}{\partial \xi_n}, \tag{3}
\]

\[J^{-1} = \det \left( \frac{\partial x_i}{\partial \xi_j} \right)\]

is the Jacobian or the volume of the cell, \(U_m = J^{-1} \frac{\partial \xi_m}{\partial x_j} u_j\) is the volume flux normal to the surface of constant \(\xi_m\), and \(G^{mn} = J^{-1} \frac{\partial \xi_m}{\partial x_j} \frac{\partial \xi_n}{\partial x_j}\) is the “mesh skewness tensor”. In the flow solver, the governing equations are nondimensionalized using the cylinder diameter \(D\) as a reference length and the incoming free-stream velocity \(U_\infty\) as a reference velocity. The Reynolds number is defined as \(Re = U_\infty D/\nu\).

The equations are solved on a nonstaggered grid topology ([34]). The Cartesian velocity components \(u, v, w\) and pressure \(p\) are defined at the center of the control volume in the computational space. A second order central difference scheme is used for all spatial derivatives except for convective terms, which are discretized using QUICK ([17]). The temporal advancement is done using a fractional step method where a predictor step calculates an intermediate velocity field, and a corrector step updates the velocity by satisfying the pressure Poisson equation at the new time step. A semi-implicit scheme with the Adams-Bashforth method is used for the convection terms and the Crank-Nicolson scheme for the diffusion terms. In this study, an “O”-type grid is employed to simulate the flow over a circular cylinder as shown in Figure 1. Further details of the numerical discretization and the parallel implementation can be found in Refs. ([4, 2, 22]).
In this study, we perform two-dimensional numerical simulations of the flow passing over a circular cylinder undergoing rotary oscillations. For the inflow and outflow boundaries, Dirichlet and Neumann boundary conditions are used, respectively. To simulate the flow field past a rotationally oscillating cylinder in a uniform stream, we enforce the boundary conditions

\[(u, v) = (1, 0) \quad \text{(inlet)},\]
\[\frac{\partial u}{\partial n} = 0 \quad \text{(outlet)},\]
\[(u, v) = (-\Omega y, \Omega x) \quad \text{(surface)},\]

where \(\Omega\) is the angular velocity of the cylinder and is nondimensionalized as \((\Omega D)/(2U_{\infty})\).

The fluid force on the cylinder is the manifestation of the pressure and shear stresses acting on the surface of the cylinder. The net force is decomposed into two components, namely the lift and drag forces. These forces are nondimensionalized with respect to the dynamic pressure. The coefficients of lift and drag can thus be written in terms of the pressure and shear stresses as

\[C_L = -\int_0^{2\pi} (p \sin \theta - \frac{1}{Re_D} \omega_z \cos \theta) d\theta,\]  

\[C_D = -\int_0^{2\pi} (p \cos \theta + \frac{1}{Re_D} \omega_z \sin \theta) d\theta,\]

where \(\omega_z\) is the spanwise vorticity component on the cylinder surface and \(\theta\) is the angle that the outer normal of the area component makes with the flow direction. We employ curvilinear coordinates \((\xi, \eta)\) in an Eulerian reference frame. For the specific case of a circular cylinder, the generalized coordinates \((\xi, \eta)\) can be represented by polar coordinates \((r, \theta)\).

### 2.2 Validation of CFD solver

We validate the rotational oscillation numerical results by comparing them with other numerical simulations. Figure 2 shows a comparison of the drag coefficients as obtained from the current simulations and those of [8] at \(Re = 100\) and \(Re = 1000\) for three different oscillation frequencies. The comparison shows good agreement. Note that the values are normalized by the respective values of the stationary cylinder. We also compare our numerical results for the drag reduction with those obtained by [23] as shown in Figure 3. In both simulations, it is shown that forcing the rotational oscillation at frequencies near or below the shedding frequency increases the drag coefficient. On the other hand, forcing it at a much higher frequency results in a reduction of this coefficient. The plot also shows good agreement between the results of the two simulations.
Figure 1: A 2-D layout of an “O”-type grid in the \((r, \theta)\)-plane showing the inflow and outflow directions.

Figure 2: Comparison of the normalized mean drag coefficient on a rotationally oscillating cylinder between current simulations and those of Choi et al. ([8]). The comparison encompasses simulations at \(Re = 100\) and \(Re = 1000\) for different nondimensional forcing frequency \(f_\Omega\) and amplitudes \(\Omega\).

3 Global Optimization

For the global optimization, the parallel CFD code ([1]) that computes the flow past a circular cylinder undergoing rotational oscillation is coupled with the parallel deterministic global optimization algorithm VTDIRECT95 ([11, 10, 12]) through an interface function that computes the drag coefficient.
Figure 3: Comparison of the mean drag coefficient as a function of the forcing frequency of the rotational oscillations between current simulations and those of Protas et al. ([23]). The vertical dashed line is set at the Strouhal number of the stationary cylinder. The horizontal dashed line is set at the drag coefficient for the stationary cylinder. The Reynolds number in both simulations is $Re = 150$ and the amplitude of the oscillations is set at $\Omega = 2.0$.

VTDIRECT95 is a Fortran 95 package for deterministic global optimization that contains the subroutine VTdirect (serial) and pVTdirect(parallel) implementing a version of the algorithm known as DIRECT ([15]). This algorithm is widely used in multidisciplinary engineering problems and physical science applications. It is an efficient global optimization method that avoids being trapped at local optimum points and performs the search for global optimum points through three main operations:

1. selection of potentially optimal boxes that are the regions most likely to contain the global optimum point;
2. sampling at points within those selected boxes;
3. division of those selected boxes into smaller boxes.

A detailed description and both serial and parallel implementations of the code are provided in [11, 10, 12]. A distinctive characteristic of deterministic algorithms like VTDIRECT95 is their frugal use of function evaluations, compared to population based evolutionary algorithms.

We consider a cylinder undergoing rotational oscillations and investigate the reduction in the mean value of the drag coefficient as the amplitude $\Omega$ of the rotational oscillations and forcing frequency $f_\Omega$ are varied. As such, the optimization problem is formulated as

$$\min_{v \theta \in D} C_D(v_\theta),$$
where \( v_\theta = (\Omega, f_\Omega) \), \( D = \{ v_\theta \in \mathbb{R}^2 \mid l_\theta \leq v_\theta \leq u_\theta \} \) is a 2-dimensional box, and \( C_D \) is the mean value of the drag coefficient.

In order to show the advantage of combining the optimization code VT-DIRECT95 with the CFD solver, we note that most of the experimental or numerical works dealing with drag reduction through oscillatory motion of circular cylinders have considered only variations of one control parameter ([8, 23, 29]). To identify the optimal point, a refined sweep must be performed in a specified range of the control parameter. In the example of reducing the mean drag, the frequency is usually taken as the control parameter while keeping the amplitude constant. Adding more control parameters would require a large number of experimental runs or numerical simulations to determine the optimal point where the mean drag is minimized. To show the significance of coupling the CFD code with an optimizer, we keep the rotational speed constant \( (\Omega = 1.5) \) and vary the frequency in the range \( 0.1 \leq f_\Omega \leq 1.0 \) with a constant increment of 0.1. As a brute force approach, we perform ten simulations to obtain the frequency response curve as shown in Figure 4). Note that the results are presented in terms of percentage reduction in the mean drag coefficient relative to that of the stationary cylinder. From this figure, we observe that the maximum drag reduction of approximately 20% occurs at \( f_\Omega = 0.7 \). In order to further refine our results, we would require another set of simulations in the proximity of this locally optimal point, thus the total number of simulations may be of the order of 100. In comparison we use a one-dimensional version of the original algorithm DIRECT for one control parameter, i.e., frequency, and consider the same range of variations. DIRECT specifies the value of the input frequency and the simulations are performed to compute the mean drag. The interface function feeds back the computed mean drag to DIRECT, which specifies the rotational frequency for the next simulation. In this way, DIRECT searches for the globally optimum point where minimum mean drag is achieved. The frequency search for the minimum mean drag is also plotted in Figure 4 for the sake of comparison. The results show that DIRECT was able to converge to a drag reduction factor of 20% corresponding to \( f_\Omega = 0.65 \) within five simulations. The brute force method would have required many more simulations to identify the optimal point. Thus, DIRECT provides an efficient algorithm to locate the optimal configuration for reducing the mean drag. Considering other parameters would add significantly to the computational cost of parameter sweeps. The same argument can be extended to the application of DIRECT for more than one control parameter where each parameter has its own sweeping range. If we require \( N_s \) simulations for one control parameter, then using a brute force method, \( P \) control parameters would require \( N_s^P \) simulations, many orders of magnitude more than VT-DIRECT95 would need.
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4 Results and Discussion

To optimize the drag reduction using oscillatory rotation of the cylinder as a control mechanism and understand the wake dynamics associated with optimal reduction, we vary the amplitude \( \Omega \) and frequency \( f_\Omega \) of rotation as control parameters. The upper and lower bounds of the amplitude of the rotational oscillations \( \Omega \) and forcing frequency \( f_\Omega \) are presented in Table 1.

Table 1: Lower and upper bounds of the frequency and amplitude of the rotational oscillations.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Lower bound</th>
<th>Upper bound</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Omega )</td>
<td>0.1</td>
<td>3.0</td>
</tr>
<tr>
<td>( f_\Omega )</td>
<td>0.1</td>
<td>1.0</td>
</tr>
</tbody>
</table>

To perform the optimization, we specify the maximum number of iterations and function evaluations, the minimal relative change in the objective function and minimum box diameter. These limits serve as the stopping conditions for VTDIRECT95. In the current study, we specify the limit on the number of function evaluations used by VTDIRECT95 as 41. By specifying the above stopping conditions, the optimization algorithm yielded the four best results reported in Table 2.

Table 3 shows a summary of the percentage reduction of the mean values of the drag coefficient with the forcing amplitude \( \Omega \) and frequency \( f_\Omega \). The

Figure 4: Percentage variation of the reduction in the mean drag coefficient with the forcing frequency \( f_\Omega \) for a rotationally oscillating cylinder at \( \text{Re} = 150 \). The rotation amplitude is kept constant at \( \Omega = 1.5 \).
percentage reduction is calculated relative to its value on a stationary cylinder. The first observation to be made here is that below a certain oscillation frequency \( f_\Omega < 0.41 \), the drag coefficient is not reduced for any rotational oscillation amplitude, as shown in Table 3 by the vertical double dashed line. As a matter of fact, increasing the amplitude of the oscillations results in a significant increase in the value of the drag coefficient in this region. As the forcing frequency is increased to values above 0.41, the drag coefficient becomes smaller than that of the stationary cylinder. The results also show that there is a threshold of the rotational oscillation amplitude below which the mean drag coefficient does not decrease for any rotational oscillation frequency, as shown in Table 3 by the single horizontal dashed line. These observations indicate that there is a minimum threshold for the rotational oscillation frequency/amplitude to obtain drag reduction. The highest reduction of about 21% is obtained over the forcing frequency range between 0.75 and 0.85, i.e., 4–5 times that of the Strouhal frequency and for oscillation amplitudes in the range 2.1–2.3.

Experimental ([30, 29]) and numerical studies ([26, 8]) show that changes in the value of drag coefficient as a result of rotational oscillations are associated with changes in the structure of the wake of the cylinder. Depending on the oscillation amplitude \( \Omega \) and forcing frequency \( f_\Omega \), the wake of the cylinder exhibits different flow patterns. For some specific values of these control parameters, the vortices are shed with the external excitation frequency and the phenomenon is then referred to as “lock-in” or “synchronization” whereby the frequency of vortex shedding becomes identical to the cylinder’s forced oscillation frequency.

To determine the wake effects of forcing the cylinder with rotary oscillations, we use flow over a stationary cylinder at \( \text{Re} = 150 \) as the reference case. Three snapshots of contours of the instantaneous vorticity in the wake of the stationary cylinder are shown in Figure 5. The plots show an alternating pattern for the vortex shedding from the upper and lower halves of the cylinder. Snapshots of vorticity contours for the optimal case of drag reduction where \( \Omega = 2.08 \) and \( f_\Omega = 0.78 \) obtained from VTDIRECT95 are shown in Figure 6.
Table 3: Summary of percentage reductions in the mean drag coefficient as obtained from VTDIRECT95, the rotational amplitude $\Omega$ versus the forcing frequency $f_0$ for a rotationally oscillating cylinder at Re = 150, threshold points in *italics*, best points in **boldface**.

| $\Omega$ | -41.0 | 20.3 | 21.0 | 17.2 | 17.8 | 18.9 | 19.7 | 20.1 | 19.6 | 14.4 | 20.0 | **20.5** | **20.8** | **20.3** | 2.50 |
|----------|-------|------|------|------|------|------|------|------|------|------|------|-------|-------|-------| 2.45 |
| 2.37     | 5.34  | 2.20 | 13.2 | 2.30 | 20.3 | 21.0 |
| 2.20     | 14.4  | 20.0 | **20.5** | **20.8** | **20.3** | 2.08 |
| 1.97     | 7.65  | 1.87 | 3.28 | 11.5 | 15.8 | 19.6 | 19.7 |
| 1.76     | 9.06  | 1.55 | 10.8 | 17.2 | 20.1 | 19.7 | 17.8 | 14.5 |
| 1.20     | 13.9  | 18.3 | 13.2 |
| 0.90     | 16.0  | 0.85 |
| 0.58     | -1.38 | 0.26 |
| 0.26     | -0.98 | 0.26 |

The contours show that the oscillatory cylinder releases single vortices of opposite signs during each half cycle. The time history of the lift coefficient, presented in Figure 7, shows periodic variations with a frequency that is equal to the forcing frequency. This is confirmed by the power spectrum, presented in the same figure, which shows a high peak at the rotational oscillation forcing frequency and a smaller peak at the shedding frequency of the stationary cylinder. The time history of the drag coefficient shows variation at a frequency twice that of the forcing frequency.

For the sake of comparison, snapshots of the vorticity contours for the case where the mean drag coefficient is significantly increased (by 42%) are presented in Figure 8. In this case, the wake structure is also synchronized with the forcing frequency, as evident from the time series of the lift coefficient and its power spectrum, which are presented in Figure 9. The lift coefficient also exhibits a third harmonic with a significant amplitude. The time series of the drag coefficient shows variations at twice the frequency of the lift coefficient. What is different from the optimal case is the wake dimension. In this case, the wake exhibits large-scale vortices behind the cylinder that are shed in an alternating manner. Furthermore, the shed vortices are further away from the center line, resulting in a larger wake in comparison to the wake in the optimal case, which explains the significant increase in the drag coefficient. The above results show that rotational oscillations at a large enough amplitude force the
Figure 5: Three snapshots of the vorticity contours for the flow over a stationary cylinder.

Figure 6: Three snapshots of the vorticity contours for the case where the highest percentage reduction in the mean drag coefficient ($\approx 21\%$) at $\Omega = 2.08$ and $f_\Omega = 0.78$ is attained.

shedding to take place at the imposed oscillation frequency no matter what it is. However, relatively slow rotational frequencies, near or less than the shedding frequency of the stationary cylinder, result in larger wakes than the wake of the stationary cylinder. On the other hand, high frequency forcing results in a smaller wake than that of the stationary cylinder and, thus, in a smaller drag coefficient.

The nature of the flow passing over a cylinder strongly affects the induced fluid forces. When a fluid separates from the rear surface of the cylinder, it forms a separated region between the cylinder and the fluid stream. The difference between the high pressure in the vicinity of the stagnation point and low pressure on the opposite side in the wake produces a net force on the cylinder in the direction of flow. At low Reynolds numbers, the major contribution to the drag force is from the viscous component. At high Reynolds numbers (i.e., $Re > 5000$), pressure drag is the major contributor to the total drag. In the intermediate range Reynolds number, both effects are significant. Figure 10(a) shows time histories of the viscous drag, pressure drag, and total
Figure 7: Time histories of the (a) lift coefficient, (b) drag coefficient, and (c) power spectrum of the lift coefficient for the optimal case where $\Omega = 2.08$ and $f_\Omega = 0.78$.

Figure 8: Three snapshots of the vorticity contours for the case where the mean drag coefficient is increased by $\approx 42\%$ at $\Omega = 1.55$ and $f_\Omega = 0.25$.

Drag for the case of the flow over a stationary cylinder at $Re = 150$. At this Reynolds number, the viscous drag and pressure drag contribute 22% and 78%, respectively, to the total drag. The time histories of the drag and its components for the optimal case are shown in Figure 10(b). The plots show that the viscous drag contributes 22% and the pressure drag contributes 78% to the total drag. However, for the nonoptimal cases, the viscous drag contributes only 12% to the total drag and the pressure drag contribution increases to 88%. This is a result of the fact that the wake is significantly larger (Figure 8) than the wakes of the stationary cylinder (Figure 5) and optimal configuration.
(Figure 6). In the nonoptimal case, the pressure drop from shoulder to tail is so sharp that the boundary layer falls near the shoulder (maximum thickness of the cylinder), yielding a broad eddying wake. Consequently, a large pressure gradient exists between the front and rear parts of the cylinder, resulting in a larger pressure drag. Figure 11(a) shows the effect of the rotational oscillations on the pressure drag. In the nonoptimal case, the pressure drag is increased by 52% when compared to the pressure drag of the stationary case. On the other hand, this drag is reduced by 20% in the optimal configuration. This stands in contrast with the viscous drag that is decreased by 24% and 28% in the optimal and nonoptimal cases, respectively, of rotational oscillations as shown in Figure 11(b). As such, the rotational oscillations of the cylinder reduce the viscous drag irrespective of the frequency of oscillation. On the other hand, the large difference is in the pressure drag that is significantly affected by the rate of rotation.

5 Conclusions

For harmonic rotational oscillation of a circular cylinder global minimization of the drag coefficient in a flow with Reynolds number 150 has been done. Optimal drag reduction happens over a frequency range that is five times larger than the stationary Strouhal frequency and with oscillation velocity ampli-
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Figure 10: Time histories of drag coefficient for (a) stationary case, (b) optimal case, and (c) nonoptimal case at Re = 150.

Figure 11: Time histories of (a) pressure drag, (b) viscous drag, and (c) total drag at Re = 150.

tudes that are near $4.5(2U_\infty)/(D)$. The maximum drag reduction is about 21%. There are thresholds of the rotational oscillation amplitude and frequency below which the mean drag does not decrease for any forcing frequency or amplitude. Arguably the wake characteristics in terms of relatively small and large vortices are responsible for low and high pressure gradients, thus modifying the drag force. These modifications in the wake are associated with the injection of external vortices generated by the rotational oscillations of the cylinder. Irrespective of the oscillation frequency, the viscous drag is reduced when performing rotational oscillation.

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