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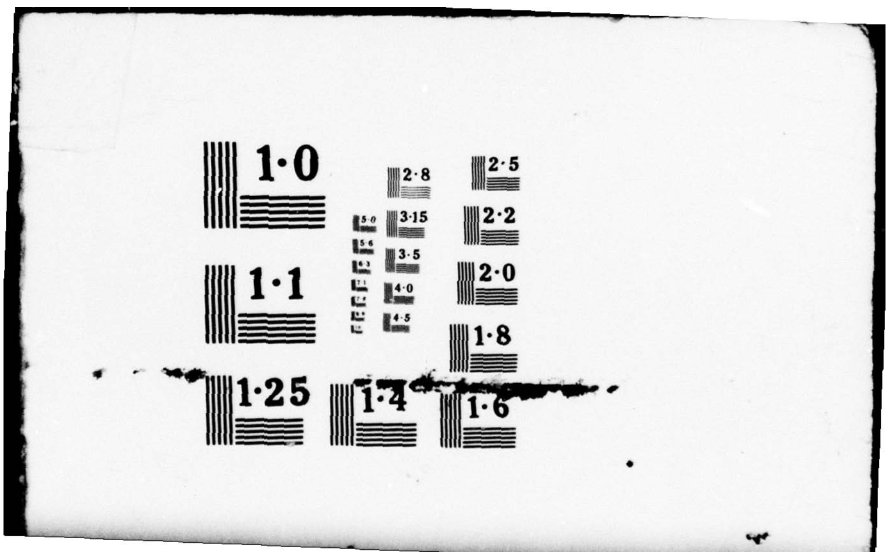
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NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Washington, D.C. 20007

EXPERIMENTAL MEASUREMENTS OF LIFT AND DRAG ON AN OSCILLATING, SMOOTH CIRCULAR CYLINDER IN CROSSFLOW

by

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HYDROMECHANICS LABORATORY

TEST AND EVALUATION REPORT

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NOTATION

A	Projected area of cylinder = $\ell \times d$
a	Half amplitude of heaving oscillation
C_D	$D/\frac{1}{2}\rho U^2 A$ = Steady drag coefficient
C_L	$L/\frac{1}{2}\rho U^2 A$ = Fluctuating lift coefficient
D	Steady drag force
d	Cylinder diameter
f	Frequency of heaving oscillation
I	Inertia force due to $M + M'$
L	Maximum half-amplitude of fluctuating lift force
L_f	Lift force on cylinder due to fluid only
ℓ	Cylinder length
M	Mass of cylinder, struts, and supports hanging from dynamometer
M'	Added mass of the fluid
n	Frequency of fluctuating lift force for non-oscillating circular cylinder
Re	Ud/ν = Reynolds number
S_f	fd/U = Forced Strouhal number for oscillating cylinder
S_n	nd/U = Strouhal number for non-oscillating cylinder
t	Time
U	Free stream velocity
y	Heave displacement
ν	Kinematic viscosity of fluid
π	3.1416 in computations
ρ	Density of fluid
ω	$2\pi f$ = Circular frequency of oscillation

INTRODUCTION

In certain Reynolds number regimes vortices are shed periodically from a smooth circular cylinder which moves normal to its axis through a viscous fluid. This alternate shedding of vortices induces a fluctuating lift force, in a direction perpendicular to its motion. A small fluctuating drag force is also induced in the direction of motion with a frequency of twice the vortex shedding frequency. For an elastic cylinder or an elastically mounted cylinder, if the frequency at which the vortices are shed is approximately equal to the free frequency of vibration of the cylinder, resonance can occur and the amplitude of vibration of the cylinder may become quite large. This phenomenon is responsible for the serious vibration problems of towed cables, submarine periscopes, and other strut appendages subject to these unsteady hydrodynamic forces. → to pg 3

Many of the experimental studies of vortex shedding have been conducted on non-oscillating cylinders in wind tunnels. The force measurements made in these studies show considerable scatter because of the non-rigidity of the mounting or because of the free stream turbulence of the flow.

Macovsky[1] was one of the first investigators to make quantitative force measurements in water by towing a cylinder in the NSRDC Miniature Model Basin. Although the cylinder was mounted on an oscillator his data were limited to the forces on a nonoscillating cylinder.

Some time later Warren[2] extended the work of Macovsky and made measurements on an oscillating cylinder. His experiments covered a Reynolds number range from 10^4 to 10^5 and amplitude-to-diameter ratios of 0.05, 0.25, and 0.50, with forced oscillation frequencies from 1.15 to 6.29 cps. This corresponded to a forced Strouhal number range from 0.032 to 2.13. The forced Strouhal number is defined as

$$S_f = fd/U$$

where f is the forced oscillation frequency

d is the cylinder diameter

U is the towing speed.

Since the natural Strouhal number for vortex shedding with frequency n from a non-oscillating cylinder S_n is approximately 0.2, this gave a range of S_f/S_n of from 0.16 to 2.13. Warren recorded his data on a heated-stylus Sanborn recorder and reduced it by measurements of the recorder traces. It was reported that the drag forces on an oscillating cylinder were greater than those on a non-oscillating cylinder and were relatively insensitive to changes in the forcing frequency and amplitude. The hydrodynamic lift force on the cylinder increased with the amplitude and frequency of oscillation. At a fixed amplitude there was a sudden drop in lift as the frequency of oscillation was decreased through the natural Strouhal shedding frequency.

Bishop and Hassan[3] conducted an experiment similar to Warren's with a 1-inch diameter circular cylinder in the Reynolds number range $3.5 \times 10^3 \le Re \le 1.1 \times 10^4$. Due to difficulties in measuring forces in this low Reynolds number range, little numerical force data were given and emphasis was placed on establishing various qualitative trends. One significant result was that when the forcing frequency of the cylinder approached the natural Strouhal frequency, only the imposed frequency of the cylinder was evident, and the natural Strouhal frequency was lost. This synchronization persisted over a range of frequencies.

Former measurements of unsteady vortex-induced forces on oscillated and stationary cylinders were obtained from oscillograph records and there was no way of determining the frequency content of the forces or the phase relations between the forced oscillations and the hydrodynamic forces. Many of the records showed variations in frequency and amplitude which indicated a multiple frequency content. Warren made a frequency analysis of a few of his records but it was too time-consuming to analyze many in this manner. Also, the force data contain inertial forces and do not represent the true fluid force acting on the cylinder. This matter is discussed in detail in the appendix of this report.

The present work was undertaken to repeat and extend some of the measurements made in the former investigations, to collect the data on magnetic tape, and to analyze the results on a high-speed computer. In addition, the data were recorded on Sanborn recorders for monitoring and for preliminary analysis. The preliminary analysis of these data are presented in this report.

The data on magnetic tape has been digitized and the IBM computer program is now ready to analyze the data. A more detailed analysis of these data will be published when funds become available to complete the work. The computer runs will provide maximum, minimum, and mean values of the forces, standard deviations, power spectra, phase relations, and total output power. With this added capability, it will be possible to deduce from the total signal, the true fluid force.

DESCRIPTION OF EXPERIMENTAL APPARATUS

The experiment was conducted in the miniature model basin of the Hydromechanics Laboratory at the Naval Ship Research and Development Center. The towing tank, with a 2 foot square cross-section and 57 foot overall length, has a precision carriage which runs along the top on specially constructed one-piece rails. The carriage, which is driven by an endless cable at constant velocities from 0.5 to 9.0 feet per second, is equipped with a mechanical oscillator that moves the attached model in a direction perpendicular to the direction of motion of the carriage. Half-amplitudes and frequencies of the oscillator range from 0 to 2 inches and 0 to 20 cycles per second respectively; however, a frequency limit is imposed by the amplitude such that the vertical acceleration does not exceed 5 g's. A more detailed description of this facility is given in reference 1.

The model-strut-dynamometer assembly, which clamps to the oscillator beams, is shown in Figure 1. The model shown is a circular cylinder 2 inches in diameter and 23.75 inches in length, which allows amplitude to diameter ratios, $\frac{a}{d}$, up to 1.0. A similar 1-inch diameter cylinder was tested after the 2-inch diameter model, thus extending the a/d range to 2.0. The cylinder was supported by two struts, each having identical symmetrical ogive fairings on its leading and trailing edges and a flat center section. Each strut was mounted to two pairs of flexures, one pair sensitive to drag forces in the direction of carriage motion and the other to lift forces perpendicular to the motion. Each flexure was instrumented with strain gages to form the four arms of a Wheatstone bridge. Since there was negligible interaction between the two pairs of flexures, one pair was sensitive only to drag force, and the other only to lift forces. The total lift and drag forces were the instantaneous sums of the forces on the respective pair of flexures.

The vertical motion of the test cylinder was measured with a variable reluctance type position transducer. The transducer was attached to the carriage and its armature attached to the oscillator beam.

The carriage velocity was measured by a 30-foot brass bar, with insulator plugs spaced at 6-inch intervals, mounted on the north rail. A point contactor on the carriage completed an electrical circuit when in contact with the brass bar and broke the circuit when in contact with the insulator plug(s). The velocity could be very accurately determined by use of an electronic counter which measured the average time elapsed between interruptions. In addition, a D.C. tach-generator, attached to one wheel of the carriage provided a continuous D.C. signal for the velocity.

The lift and drag forces, the vertical sinusoidal motion of the test cylinder, and the carriage speed were recorded on a Precision Instrument, 14-channel, F-M magnetic tape recorder. Figure 2 shows a block diagram of the instrumentation arrangement for the two lift transducers. The two drag signals were conditioned and recorded in the same manner. A Sanborn carrier pre-amplifier was used for signal conditioning (power supply and amplifier) of the position transducer. A Sanborn D-C coupling amplifier provided the signal conditioning for the velocity tach-generator.

TEST PROCEDURE

In preparation for testing, the carriage velocity was calibrated with respect to the potentiometer control of the drive motor. Hence, the desired velocity could be obtained by dialing in the corresponding potentiometer setting. The amplitude of oscillation was set with a precision dial gauge to within .0005 inches, and the period of the oscillation was determined with an electronic counter to within .001 seconds.

The first series of tests consisted of oscillating the cylinder in air at various frequencies and amplitudes and at zero forward velocity. This exact procedure was repeated in water. These static air and water data were used to determine the effect of inertia on the dynamic tests as discussed in the appendix.

The cylinder was then towed at various velocities while oscillating at the same frequencies and amplitudes tested in the static tests. The test procedure was to set a given amplitude and then run through a range of velocities and frequencies. This procedure was repeated for each amplitude keeping the same discrete values for the velocity and the frequency. Before the model was accelerated to the desired speed, the frequency of oscillation was first established. In order to prevent disturbances from propagating down the towing tank, a removeable partition isolated the model from the rest of the towing tank while the desired frequency was established. Also sufficient time was allowed between runs for the disturbance from the previous run to damp out. Wave absorbers, in the form of metal screens, were installed in each end of the towing tank.

TESTS RESULTS AND DISCUSSION

Data obtained from the Sanborn records have been reduced and are presented in Figures 3 through 5.

Figure 3 shows the variation of the drag and lift coefficients with Reynolds number for the nonoscillating cylinder. These dimensionless parameters are defined as follows:

Drag Coefficient $$C_D = \frac{D}{\frac{1}{2}\rho U^2 A}$$

Lift Coefficient $$C_L = \frac{L}{\frac{1}{2}\rho U^2 A}$$

Reynolds Number $$Re = \frac{Ud}{\nu}$$

where D is the average value of the steady drag

L is the maximum half-amplitude of the fluctuating lift force

U is the free stream velocity

d is the cylinder diameter

A is the projected area of the cylinder or length times diameter

ρ is the density of the fluid

ν is the kinematic viscosity of the fluid.

A typical drag trace consists of a steady drag component upon which is superimposed a small amplitude, oscillatory component. The drag coefficients are based on the steady drag component. On the other hand, a typical lift trace consists of a rather well defined sinusoidal trace which oscillates about the zero lift line. The lift coefficients are based on the half-amplitude value of the most frequently occurring maximum peaks in a given lift record.

The non-oscillating cylinder data in Figure 3 shows that the drag coefficient is basically independent of Reynolds number for the Reynolds number range of the test. The lift coefficient is of the same order of magnitude as the drag coefficient for Reynolds numbers near 10^4 but decreases rapidly as the Reynolds number is increased. These trends are noticed for both the one and two inch diameter cylinders and are in agreement with other investigations.[1,2]

Figures 4 and 5 show the variation of the lift and drag coefficients on the oscillating cylinder with forced Strouhal number to natural Strouhal number ratio for various Reynolds numbers and fixed amplitude to diameter ratios, a/d. The lift and drag coefficients are defined in the same manner as before. In the Reynolds number range of the experiments, between 10^4 and 10^5, the natural Strouhal number was 0.20.

It is important to point out that the forces from which the lift coefficients are defined include the inertia force. As mentioned in the appendix, in order to eliminate this contribution to the total lift signal its phase with respect to the motion must be known. However, this information is difficult and in most cases impossible to obtain from the Sanborn charts with any degree of accuracy. Even for the cases where the lift is in phase with the motion and scalar algebra is applicable, the fluid dynamic lift is the difference of two quantities of the same order of magnitude and could be within the range of error introduced by reading the Sanborn records. When the computer results are completed, phase angle information will be available.

The drag data in Figure 4 shows that for $a/d \le 1.0$, C_D is in general greater than for the non-oscillating cylinder and approximately independent of S_f/S_n for $S_f/S_n >$ about 0.4. For $S_f/S_n < 0.4$, C_D is basically the same as for the non-oscillating cylinder. The effect of Reynolds number on C_D seems to decrease as a/d is increased from 0.1 to 1.0. Whereas the spread in C_D was between 1.0 and 2.6 for a/d of 0.1, this spread decreased to between 2.0 and 2.6 for a/d of 1.0. For $a/d > 1.0$, C_D becomes more sensitive to S_f/S_n, increasing as S_f/S_n is increased to about 0.8, and falling off near S_f/S_n of 1.0. From an observation of the Sanborn records, it was found that although the steady drag coefficient decreases for $S_f/S_n > 1.0$, the unsteady drag coefficient becomes larger as S_f/S_n is increased, until this coefficient becomes several times greater than the steady drag coefficient.

The lift data in Figure 5 shows that for the smaller values of a/d and S_f/S_n ($a/d < 1$ and $S_f/S_n < 0.9$), the lift coefficient, while being somewhat larger for the oscillating case, exhibits the same trend as the nonoscillating data (C_L decreases with increasing Re). For larger values of a/d and S_f/S_n, C_L is not as sensitive to Re. As a/d and S_f/S_n become larger the inertia component of the total lift force, which depends on these two variables, becomes larger until it is the dominant component.

It will be possible to discuss some other interesting unsteady results when the computer analysis of the data is completed.

ERROR ANALYSIS

From a detailed analysis of the possible error in the various physical parameters, the maximum percent error in the significant non-dimensional numbers used to interpret the results of the test was found to be as follows:

NON-DIMENSIONAL NUMBER	MAXIMUM PERCENT ERROR
Reynolds Number (Re)	2.5%
Forced Strouhal Number (S_f)	4.7%
Lift and Drag Coefficient ($C_{L,D}$)	7.1% 0 - 5 pounds range 5.7% 5 - 10 pounds range 5.2% 10 pounds and above
Amplitude to Diameter Ratio (a/d)	0.79%

It is estimated that the error introduced from reading the lift and drag traces from the Sanborn Charts is between 3 and 6 percent. This error would have to be added to the error for the lift and drag coefficients for the results presented in this report.

ACKNOWLEDGEMENTS

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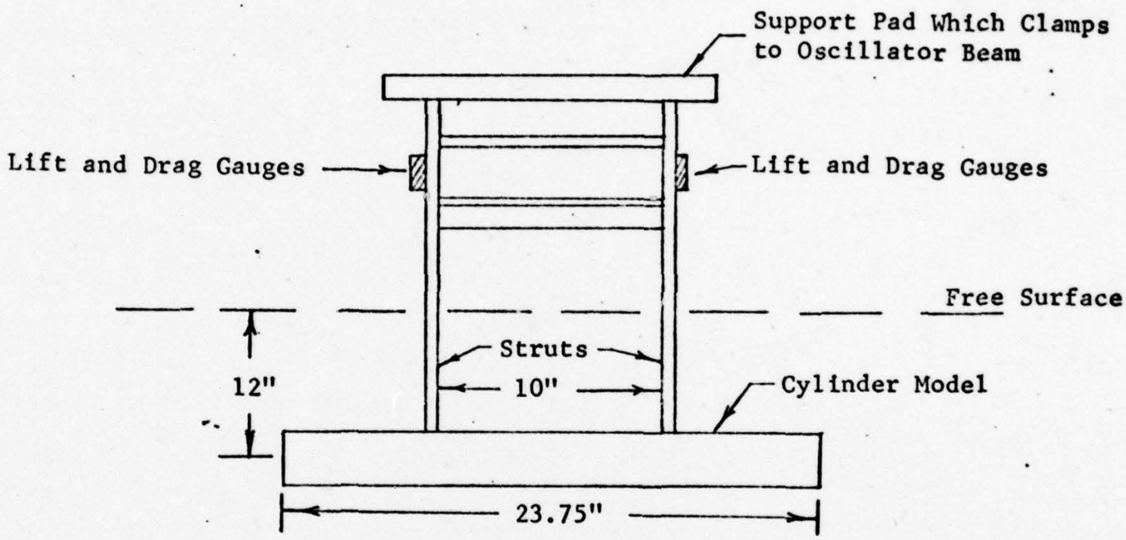

Figure 1 - Schematic of Model-Strut-Dynamometer

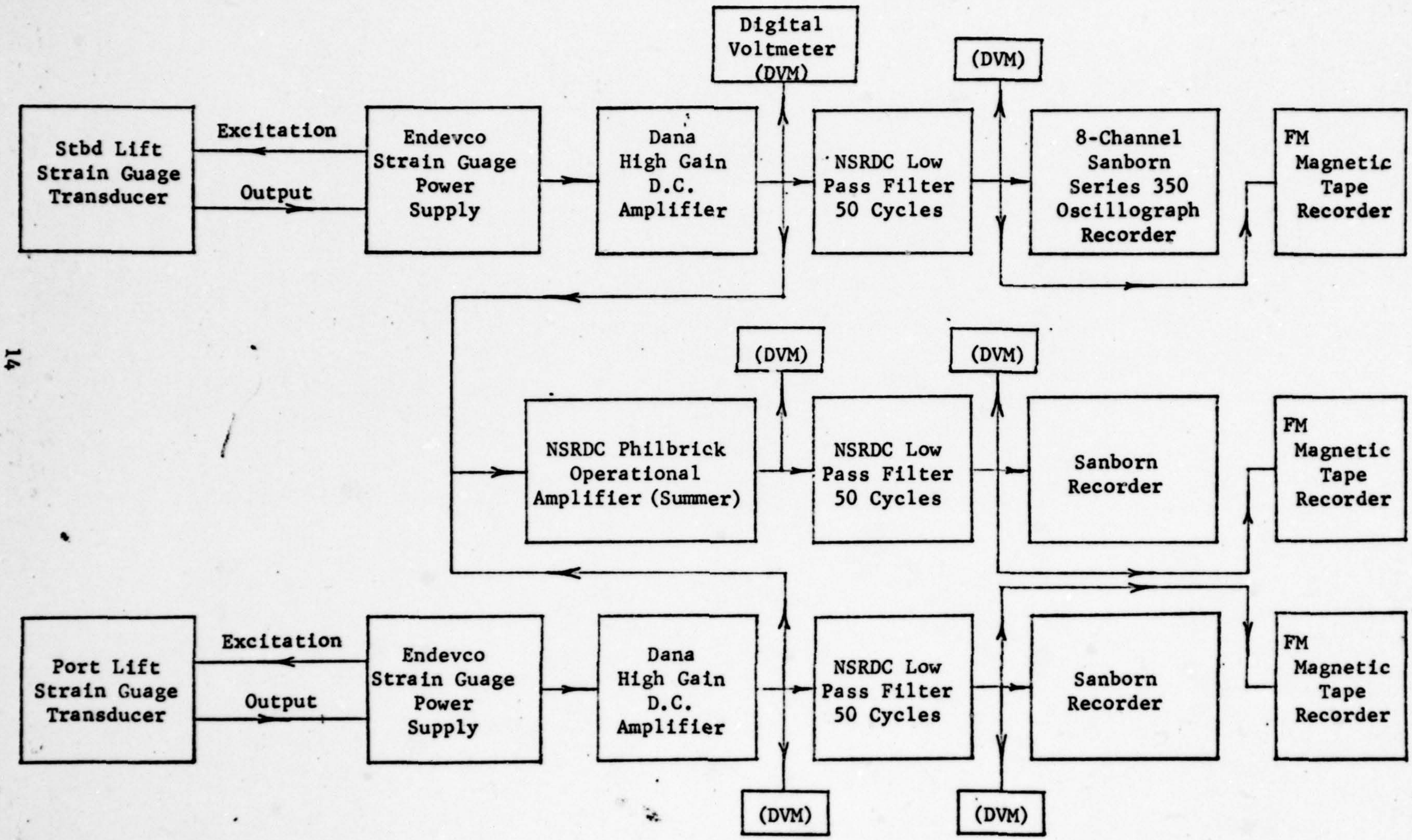

Figure 2 - Block Diagram of Electronic Instrumentation

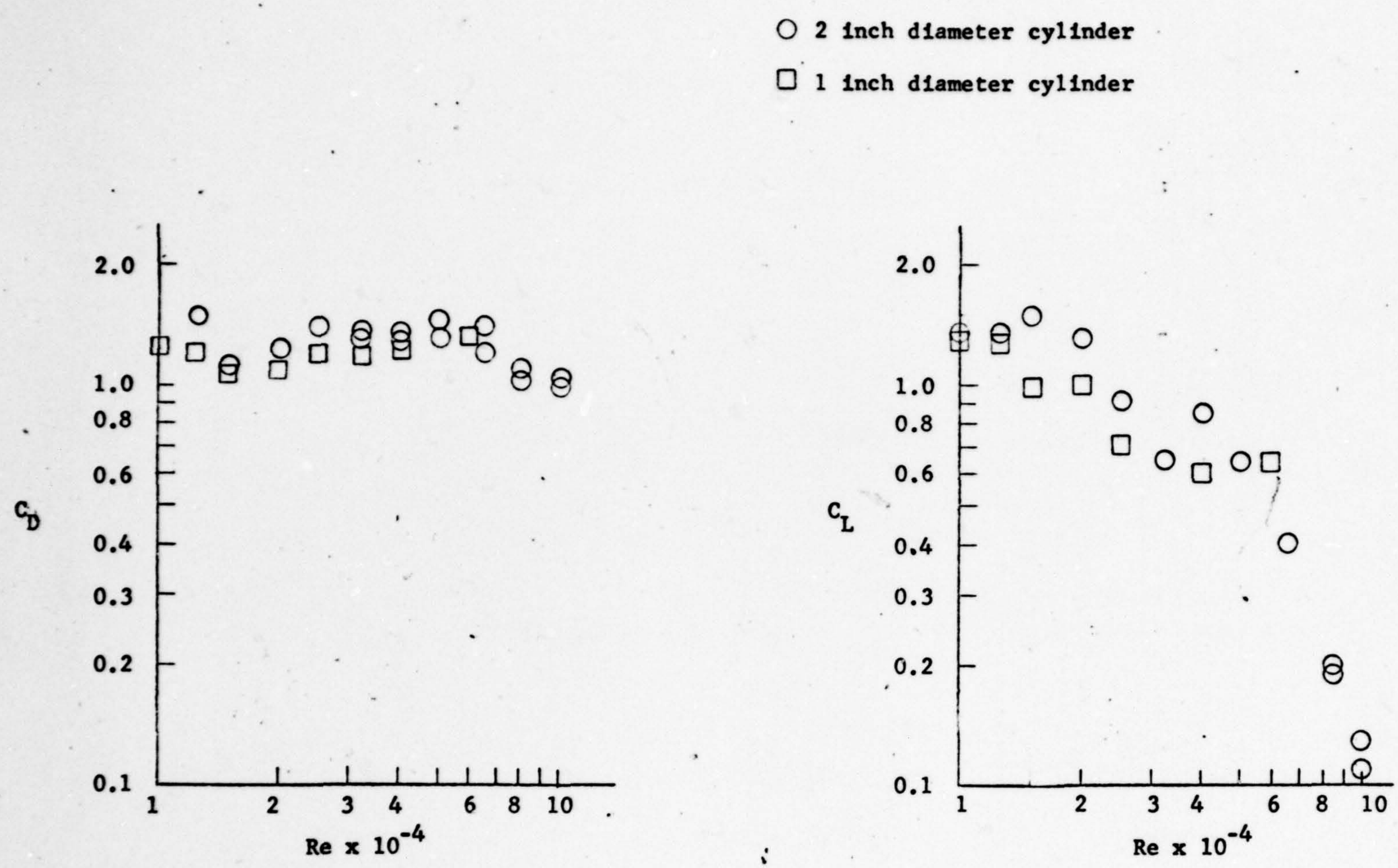

Figure 3 - Drag and Lift Coefficients of a Nonoscillating Cylinder as Functions of Reynolds Number

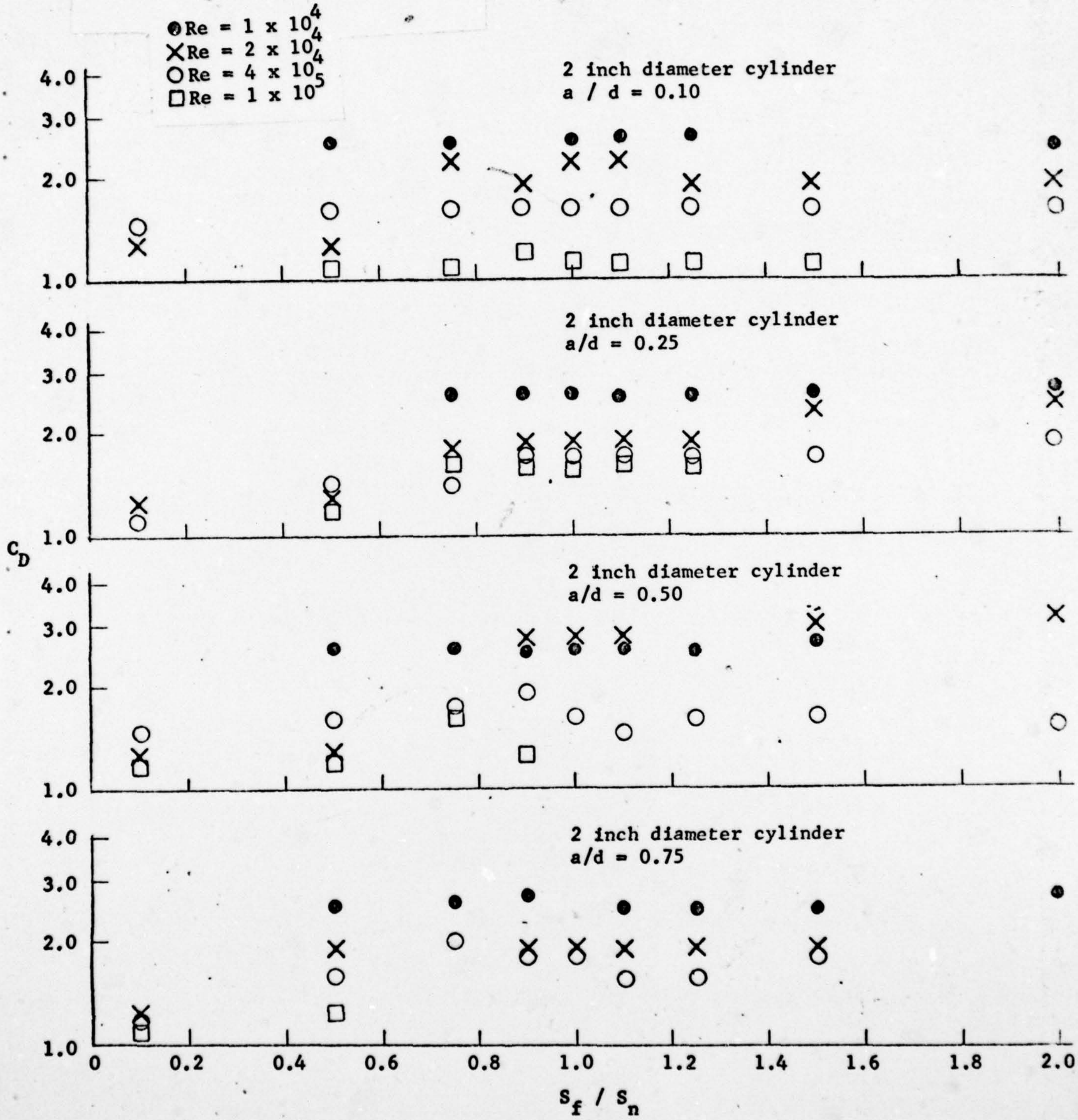

Figure 4 - Variation of Drag Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

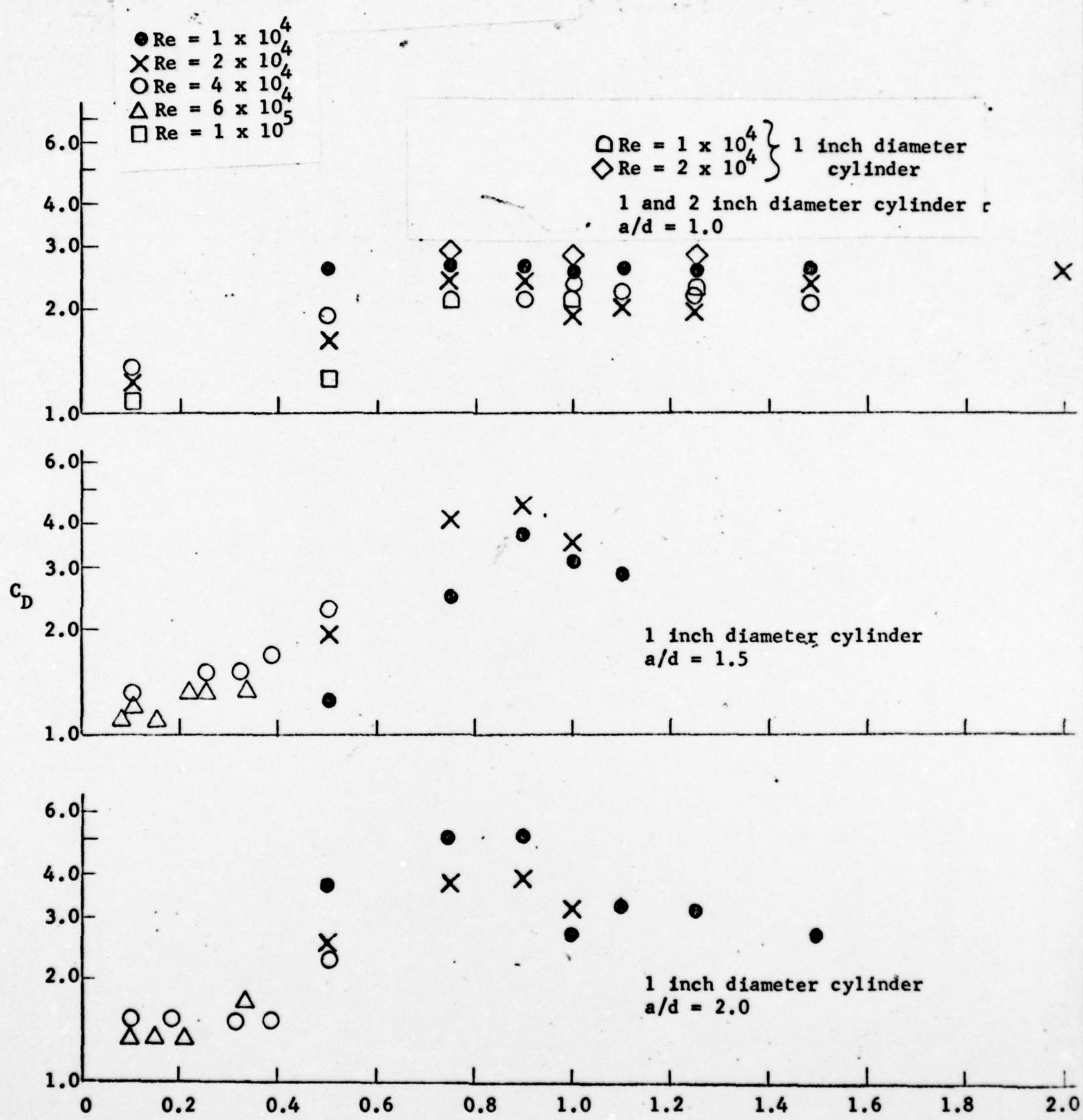

Figure 4 continued - Variation of Drag Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

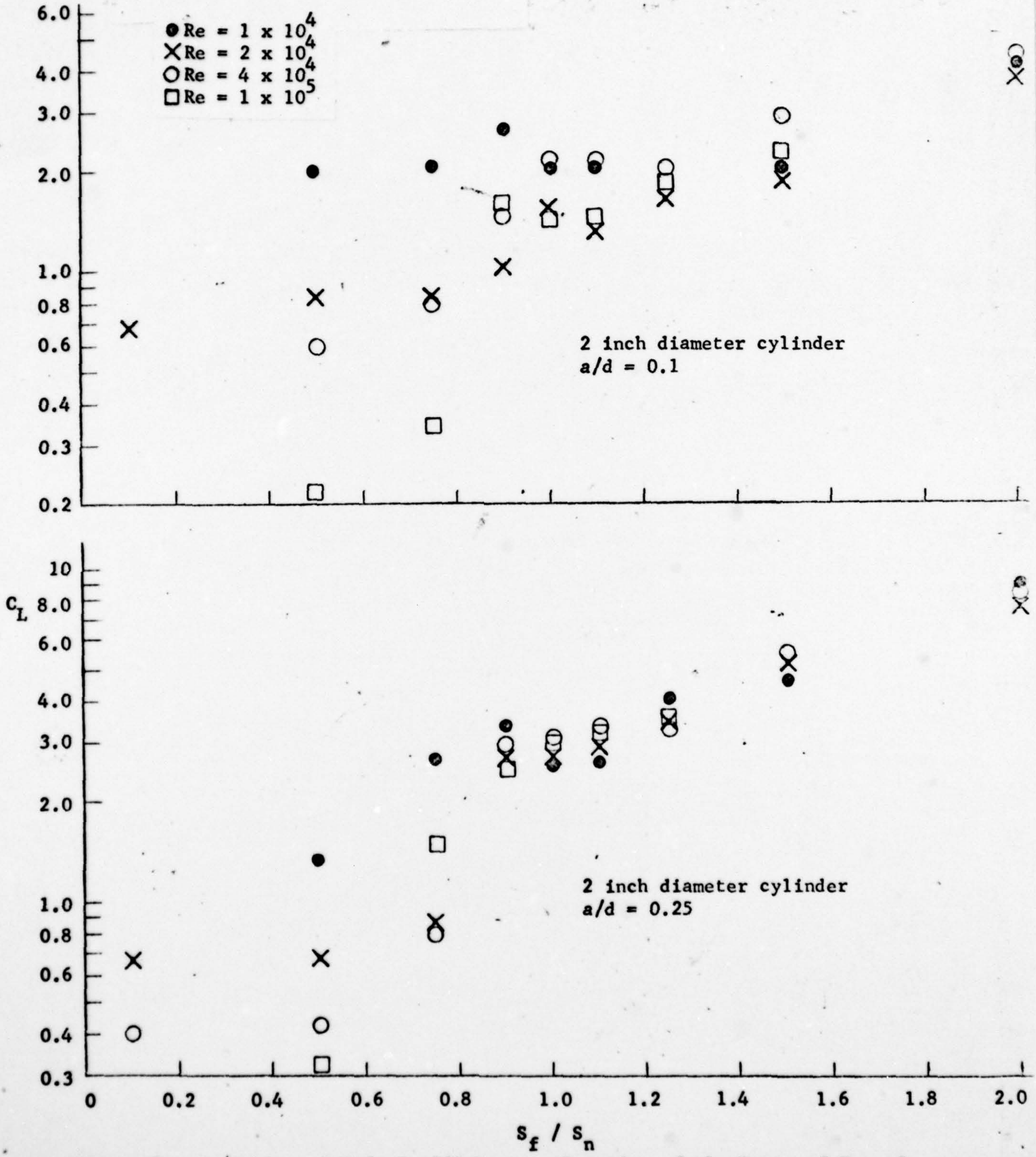

Figure 5 - Variation of Lift Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

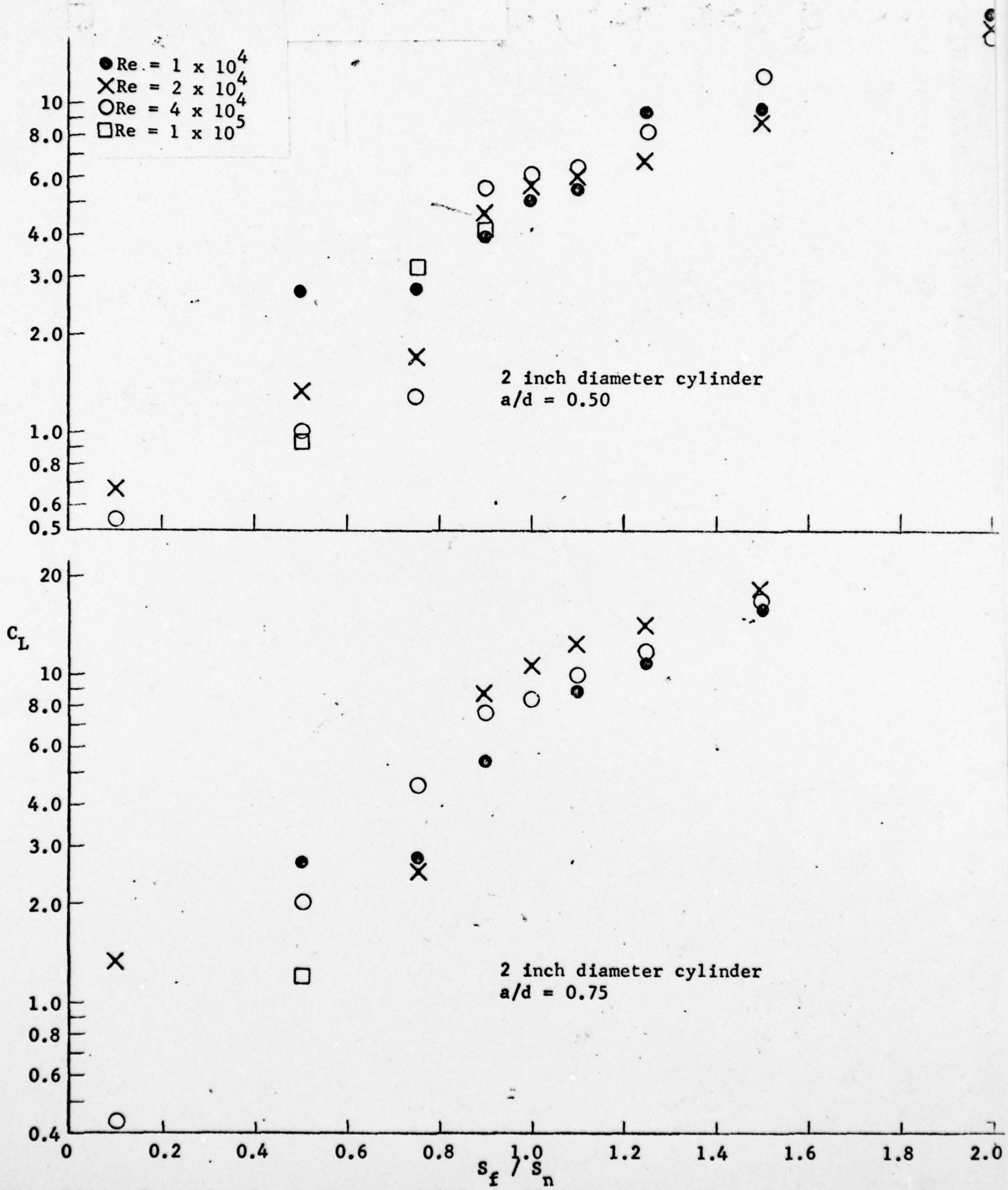

Figure 5 continued - Variation of Lift Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

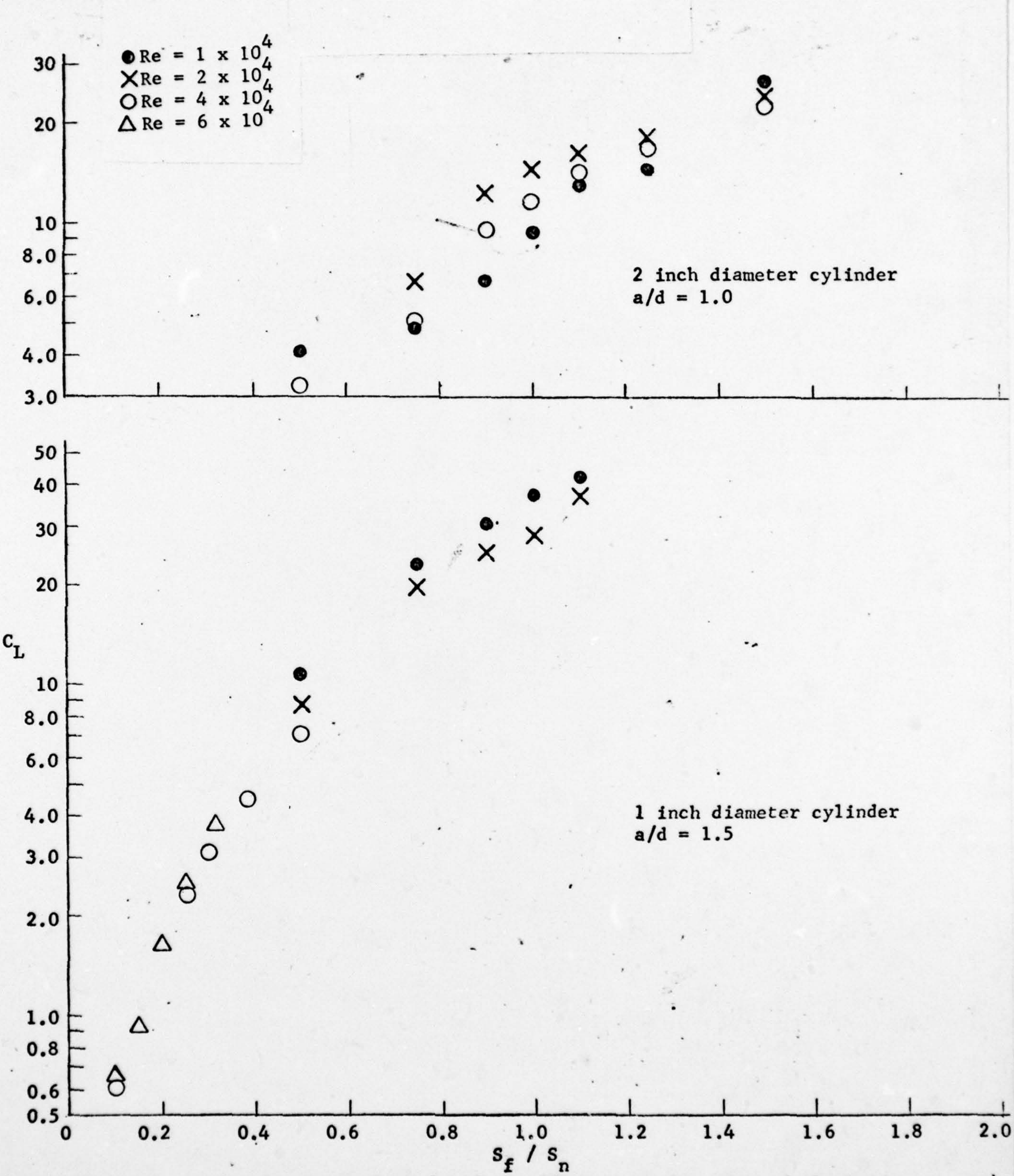

Figure 5 continued - Variation of Lift Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

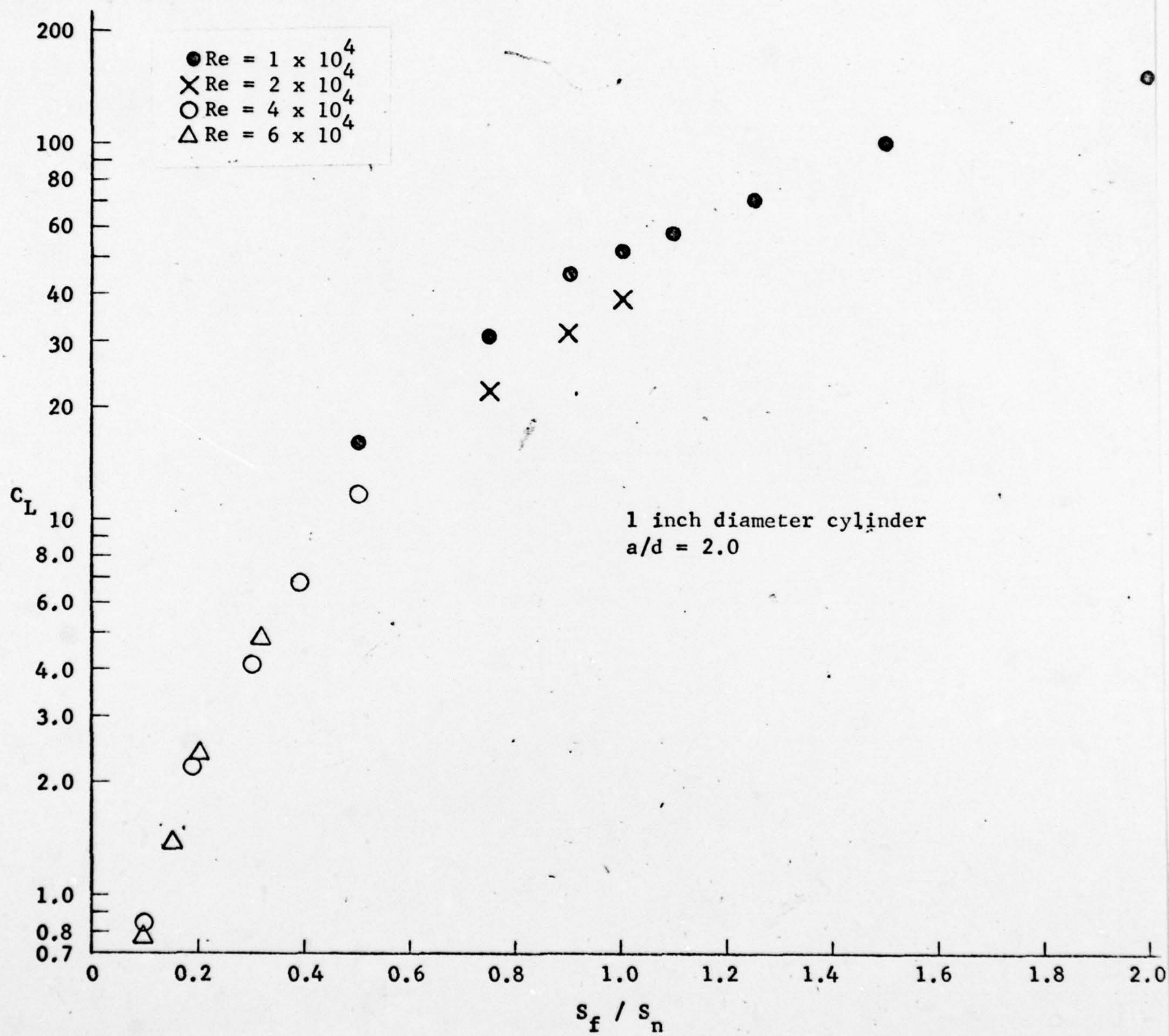

Figure 5 continued - Variation of Lift Coefficient as Function of the Ratio of Forced to Natural Strouhal Numbers for Various Reynolds Number and Ratio of Amplitude to Diameter

APPENDIX

Correction for Inertial Forces

The total lift force, L , acting on a cylinder translating, and oscillating in heave may be regarded as being composed of an inertia force, I , and a lift force, L_f , due solely to the fluid. The inertia force includes the inertia force due to the mass of the cylinder and supports hanging from the dynamometer and the added mass due to the water. If the cylinder is forced to oscillate sinusoidally in heave, the displacement and inertial force are real parts of

$$y = ae^{i\omega t}$$

then $$I = (M + M')\,\ddot{y}$$

or $$I = (M + M')\,a\omega^2 e^{i\omega t}$$

where a is the amplitude of heave displacement

M is the mass of cylinder and supports

M' is the added mass of the water, and

ω is the angular frequency, $2\pi f$

Thus $$L = I + L_f = (M + M')\,\omega^2 a e^{i\omega t} + L_f\,.$$

or $$L_f = L - (M + M')\,a\omega^2 e^{i\omega t}$$

It is seen then, that the lift force due to the fluid may be obtained by vectorially subtracting the inertia force from the total lift force.

The inertia force is obtained experimentally by oscillating the cylinder in water at a given frequency and amplitude and zero forward velocity. The inertia force is measured by the flexures and the associated mass can be calculated, since

$$(M + M') = \frac{|I|}{a\omega^2} .$$

The added mass due to the water is obtained by subtracting M from (M + M') . The mass of the cylinder and supports is obtained by oscillating the cylinder in air at the same frequency and amplitude. The inertia force due to the cylinder and support mass is measured by the flexures and M is given as

$$M = \frac{|I_{air}|}{a\omega^2} ,$$

Hence,

$$M' = \frac{|I| - |I_{air}|}{a\omega^2}$$

This experimental value can be checked, since theoretically the added mass of a cylinder is equal to the mass of the displaced fluid.

An example of how to reduce the fluid force from the total lift force is now given for a particular test run. In this example, the forces are in phase with the heave displacement, which simplifies the calculations to scalar algebra. If the forces were not in phase with the heave displacement vector algebra would be used. For a frequency of 3.34 cps and a half amplitude of 1.5 inches, the total lift force was found to be 19.9 lb oscillating in air, 21.0 lb oscillating in water, and 23.6 lb oscillating in water with a forward velocity of 6.69 ft/sec. The lift data obtained from oscillating in air represents the inertia force of the cylinder and supports from which can be calculated the corresponding mass:

$$I_{air} = M\omega^2 a = 19.9 \text{ lb}$$

$$M = 0.360 \text{ slug}$$

From oscillating in water the total inertia force is measured and the corresponding mass is calculated:

$$I = (M + M') \, \omega^2 a = 21.0 \text{ lb}$$

$$(M + M') = 0.380 \text{ slug}$$

Subtracting these two values yields the experimental added mass to be

$$(M + M') - M = M' = 0.020 \text{ slug}$$

Theoretically, from potential theory it can be shown that the added mass is equal to the density of water multiplied by the volume of the cylinder. This calculation gives $M' = 0.021$ slug. Hence the experimental and theoretical value for M' differ by 4% which is within the experimental error. The fluid force is given as:

$$L_f = L - I = 23.6 - 21.0 = 2.6 \text{ lb}$$

The fluid lift force can now be put in coefficient form and compared with the lift coefficient for translation only. By doing this one would obtain the effect oscillations have in changing the lift coefficient. The lift coefficient for 6.69 ft/sec without oscillation was found experimentally to be 0.650. With the cylinder oscillations of the above example, the lift coefficient was found to be 0.364. Thus there is a decrease in the lift coefficient due to oscillating the cylinder at the prescribed frequency and amplitude.

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