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Measurement of Crossflow Forces on Tubes

by

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ABSTRACT

A force transducer for measuring lift and drag coefficients for a circular cylinder in turbulent water flow is presented. In addition to describing the actual design, requirements for obtaining valid fluid-force test data are discussed, and pertinent flow-test experience is related.

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INTRODUCTION

Theoretical prediction of flow-induced forces acting on bluff bodies is in the very early stages of development, and therefore experimental measurement of these forces must still be relied upon. Toward this end, a force transducer was sought to provide the fluid forces for vibration analysis of circular cylindrical beams in homogeneous, turbulent crossflow. A water medium was stipulated because the applications of interest are typically in dense fluids. Although the fluid forces on relatively rigid components could be obtainable with most fluid media and proper hydraulic scaling, the eventual goal is to measure the fluid forces where beam motion and fluid flow interaction occur. The required [1] simulation of relative structural and fluid mass are more convenient using dense fluids.

For analysis of beams, characterization of the force per unit length is required and can be accomplished in more than one way. The pressures at each point around the circumference of the cylinder can be measured and integrated to obtain the resultant force [2]. Alternatively, a force transducer whose sensing element is a finite-length circumferential segment (Fig. 1) of the cylinder has the advantage of measuring an integrated force per unit length directly [3]. The difficulty of the force-ring transducer is that rather complicated multicomponent sensing devices must be constructed in a small space for a wide variation in the character of the forces. However, a force-ring transducer was sought because of the difficulties in maintaining time references and amplitude scaling in the integration of pressure to obtain resultant forces, and because measurement of pressures in pressurized water also requires considerable instrumentation effort and interpretation of the data [4].

FLOW FACILITY CONSTRAINTS

The physical size of a crossflow force transducer, incorporated inside a test cylinder, is invariably linked to the flow capacity of the available test facility. Essentially, to duplicate the high Reynolds numbers (10^4 to 10^7 based on cylinder diameter) of the turbulent flows prevalent in industry, the area of the experiment's flow channel is made as small as feasible to achieve high flow velocities. For a given ratio of test cylinder projected area to flow channel area, which is called solid blockage, a smaller flow channel area will produce a larger Reynolds number, based on tube diameter. The minimum feasible flow channel area is determined by the available pumping head of the test facility, and, more importantly, by the minimum size of the cylinder and transducer which can make the desired measurements.

A cylinder diameter of 25.4 mm (1.0 in.) was chosen to allow room for a transducer and to minimize the interaction of the cylinder's wake with the walls of the flow channel. The solid blockage of the cylinder was maintained below 8% to minimize the velocity increase caused by the flow "squeezing" between the cylinder and the flow channel walls. Of course, elimination of the walls is desirable but not practical. Based on existing data [3,7,8], corrections to force measurements would not be significant for this small blockage. A cylinder length of 0.31 m (12 in.) was chosen to achieve two-dimensional flow near the center of the cylinder where force measurements would be made. The fluid forces on long cylinder in turbulent crossflow were not expected to be correlated beyond axial distances of three diameters, based on nonturbulent flow results [5], thus a six diameter distance to the nearest wall was thought adequate. Six diameters was not adequate for simulation of two-dimensional flows, as will become apparent in the later discussion on the use of end-plates.

Based on the constraints defined above, a 0.31-m (12-in.)-square flow channel was added to the available flow facility which could provide flow velocities up to 5.5 m/s (18 ft/sec) with an available pressure head of approximately one-half megapascal (75 psi). For this flow velocity and a test cylinder size of 25.4 mm (1 in.), the maximum Reynolds number attainable, with room temperature water, was 2×10^5 . The available pressure head limited the intensity of the turbulence which could be produced by grids placed upstream of the test cylinders, but turbulence intensities of 2-15% at integral length scales of 0.5-2 cylinder diameters were produced. This range of intensity was expected to identify the major effects of turbulence on the fluid forces, even though more severe turbulence conditions can be found in some industrial applications.

DESIGN REQUIREMENTS

Three forces are of interest for a cylinder in crossflow: the steady drag force in the direction of the flow, the fluctuating drag force in the direction of flow, and the fluctuating lift force normal to the flow and the axis of the cylinder. The fluctuating forces are caused by a combination of vortices being shed in the wake and the turbulence in the flow (see Fig. 1). The interaction of the turbulence with the shed vortices is the subject of current research for which the force transducer is intended. A transducer was desired that could measure all the forces.

Information for establishing the transducer design criteria is limited because most force measurements have been made in smooth-flow wind and water tunnels where turbulence was maintained at a minimum. Turbulent flows have long been known to affect fluid forces, but only recently have efforts been made to quantify the effects for circular cylinders [2,7,8]. Although attenuated in intensity and broadened in spectral content by turbulence, vortex shedding remains a dominant source of the fluctuating forces in turbulent flows below transition Reynolds numbers of 3×10^5 based on the mean flow velocity U and cylinder diameter D .

For low turbulence intensities, the lift-force frequency spectrum is contained in a narrow band centered around the frequency

$$f_v = 0.2U/D. \quad (1)$$

For intense turbulence the lift-force spectrum begins to look like the turbulent-velocity spectrum: a relative constant spectrum up to a cutoff frequency less than f_v . Thus, the lowest structural frequency of any force-measuring transducer should be well above f_v to eliminate spurious signals caused by the dynamic (inertial) response of the transducer. Essentially a transducer which responded statically was desirable. Assuming the fluid force

is concentrated at f_v and the transducer can be modeled as a one-degree-of-freedom spring-mass system, then separation of the fundamental frequency and f_v by a factor of 10 would limit the inertial response of the transducer to 1% of its static deflection. This was acceptable. The fluctuating drag force is usually an order of magnitude smaller than either the steady drag or the fluctuating lift force. It occurs at a frequency twice that of the fluctuating lift, where dynamic response of the transducer is an acceptable 4% of the static deflection.

The fluid force to be measured is primarily due to vortex shedding in the wake flow and occurs below f_v . However, much smaller forces at higher frequencies exist due to the turbulence in the flow which are not of interest but can excite the transducer at its fundamental frequency. Signals at the resonant frequency which could mask the force signals were eliminated by low-pass filtering, a technique made possible by the relatively high transducer resonant-frequency requirement. Without filtering, the force signal could dominate the desired wake force signal. For instance, at a relatively high structural damping of 10% of critical damping, and assuming a force 10% as large as that occurring near the vortex shedding frequency, not an unreasonable estimate especially for small-scale highly turbulent flow, deformation of the transducer at resonance would be comparable to that caused by the vortex-shedding force, again based on a single-degree-of-freedom system.

Surface roughness is known to change the character of the vortex shedding process in ways similar to the expected effects of turbulence in the flow, and it therefore has to be minimized in order to study the effects of flow turbulence. Essentially, surface roughness influences the fluid forces in two ways: degradation of the correlation of the vortices along the length of the

cylinder, and thickening of the boundary layer. The latter effect only occurs for very rough surfaces and will not be considered further. The former effect is of concern.

In nonturbulent flow across smooth, stationary cylinders, wake flow measurements [12] have shown that the vortices are shed as two-dimensional sheets, three to six diameters long, provided the mean flow Reynolds number based on the cylinder diameter is less than 10^5 . As the flow Reynolds number is increased above 10^5 , the attached boundary layer undergoes transition from a laminar to a turbulent flow regime. As a result, the two-dimensionality of the vortex shedding process is substantially degraded and the fluid forces are significantly decreased. Surface roughness can influence the vortex shedding process by causing premature transition of the boundary layer, with larger surface roughnesses creating earlier transitions and larger reductions in the associated fluid forces. However, disagreement exists in the literature on the threshold value for which the boundary layer is disturbed significantly: limits on the maximum roughness size from 10^{-3} to 10^{-6} of a diameter have been proposed or employed [3,9]. Machining of an acceptable surface finish, based on the most demanding limit, presented no difficulty. However, static deflection of the active force transducer ring with respect to the dummy sleeve (Fig. 1) was a concern. The relative deflection could not be made less than the upper limit while still maintaining a useful transducer sensitivity. Also, construction of a dummy cylinder and a force sensing ring that are structurally separated by a circumferential gap less than 10^{-3} of a diameter was not possible. Maximum radial and longitudinal discontinuities of 10^{-3} of a diameter were adopted as being the best conditions for which the force-ring transducer could be constructed.

Perhaps more important than a surface roughness effect, pressure relief (venting) provided by the circumferential gap has been identified [13,14] as another mechanism that can greatly influence the axial correlation of the vortex shedding process, again by causing premature transition of the attached boundary layer. Even though the disturbance occurs at a single circumference along the cylinder axis, the effect is thought [15] to propagate over several diameters of cylinder length. The pressure relief can be solved by sealing the gap [7,14], but then transducer sensitivity and frequency response become a function of the sealant and usually the environmental conditions (especially temperature). Also, the sealed circumferential gap must be wider than an unsealed gap to provide a flexible connection relative to the transducer stiffness. As a result, the sealant introduces additional surface roughnesses which could be as effective as pressure relief in causing transition of the boundary layer. The circumferential discontinuities were rationalized as an unavoidable feature of the force-ring transducer, and the use of the transducer could only be justified by measurements which reproduce data in the literature. An unsealed circumferential gap was employed, but, as will be discussed later, end-plates had to be employed to maintain the two-dimensionality of the vortex shedding process.

Estimates of the range and magnitude of the fluid forces on the transducer ring were needed to determine the necessary strength and sensitivity of the transducer. However the forces were difficult to assess based on available data. Different turbulence intensities, length scales, blockages, and Reynolds numbers produce different force intensities and correlation lengths [7,8], and establishment of empirical relations to account for the effect of these parameters is the subject of current research. However, for a rigid cylinder, the maximum force in turbulent flow was not expected to exceed

that in smooth flow. For the Reynolds number range of the test ($< 3 \times 10^5$), fluid-force coefficients for nonturbulent flow were available. The steady drag-force coefficient is a nearly constant value of 1.1. The lift-force coefficient peaks in the Reynolds range of most interest, 10^4 to 1.5×10^5 , with an average value of about 0.7. The coefficients are defined as the ratio of the force to the product of the dynamic pressure and the projected area of the cylinder in the flow direction. These coefficients were considered good estimates of upper bounds for use in the design of the transducer's strength. However, lower bounds on the minimum force to be resolved could not be estimated as well.

In the Reynolds number range from 10^4 to 3×10^5 , flow turbulence can greatly affect the magnitude of the minimum force to be resolved. For nonturbulent flow, the lift coefficient is nearly a constant (an average value of 0.7) except for a rapid decrease to a value of ~ 0.1 over the Reynolds number range 1.5×10^5 to 3×10^5 . However, for turbulent flow the decrease apparently [7,8] may occur anywhere in the range from 10^4 to 3.0×10^5 , depending on the severity of the turbulence. Assuming that the decrease begins at 10^4 and ends at 10^5 , nearly the same range as for nonturbulent flow, then the lift force per unit length remains nearly constant because the dynamic pressures increase quadratically with Reynolds number. A lift coefficient of 0.1 at a Reynolds number of 10^5 gave an estimate of the minimum force per unit the transducer must resolve. Of course, the absolute minimum force the transducer would have to resolve depends on the axial correlation of the fluid force and the width of the force sensing ring of Fig. 1.

Turbulence also reduces the correlation of the forces along the length of the cylinder, with the longest correlation length of approximately three diameters expected in smooth flow. In turbulent flows, correlation lengths of

a diameter or less were expected. To avoid underestimation of fluid forces due to lack of axial correlation, a force sensing ring one-half diameter long was employed. However, the design (strength) was determined for a ring length of up to 3 diameters.

Because the fluctuating drag can be very small in comparison to the other forces, its resolution for all flow condition was doubtful.

Contradictory information exists as to whether the amplitude of beam motion can influence the vortex shedding process and forces in turbulent flows. In nonturbulent flows very significant increases in the forces occur with increased amplitude of motion. A large part of the increase is due to increased correlation of the vortex shedding process along the length of the cylinder. However, evidence exists [7] that the fluid-structure interaction does not occur for turbulent flows having approximately 10% intensity. Yet practical examples of large-amplitude vibrations of chimneys, light poles, ocean piles, and reactor components, undoubtedly in turbulent flow, have been observed and thought to be due to vortex shedding and structural interaction. To avoid fluid/structural interaction effects, incorporation of the ring transducer on a rigid, stationary cylinder was planned, but a transducer design which can be incorporated on a flexible cylinder was desired for future studies.

A strain-gauged force ring [10] was chosen as the basic element of the transducer because of its relative stiffness for a given sensitivity, its ability to measure steady and dynamic components of force in two orthogonal directions, and its relative compactness for incorporation into a circular cylindrical cross section without coupling to the deformation of the beam due to the flow or to the beam end supports.

A schematic of the force transducer mounted on the cylinder is shown in Fig. 1. Obviously the force-measurement ring that evolved is not a classic ring, but more a rigid frame with elastic beam joints where the strain gauges are mounted. Early analysis and testing showed a uniform-thickness ring did not yield both the desired sensitivity and rigidity. Sensitivity could be increased for the same stiffness by going to the rigid frame with beam joints concept. However, a desirable feature of a ring transducer was lost, since the strain gauges for monitoring the lift force F_L could not be located at a strain node of the drag force. A circumferential location of $\phi = 40^\circ$ was chosen to equalize maximum strains and to facilitate construction.

For purposes of design, the transducer was idealized as a frame with rigid members connected by springs (see Fig. 2). The springs correspond to the thin flexible beams, of identical dimensions, which separate each rigid segment of the force-measurement ring shown in Fig. 1. The moments acting at each beam location

$$M_1 = \frac{F_D R}{3} H + \frac{F_L R}{2} (1 + 2\alpha K),$$

$$M_2 = -\frac{F_D R}{6} H + \frac{F_L R}{2} (1 + \cos \phi - \alpha K),$$

$$M_3 = -\frac{F_D R}{6} H + \frac{F_L R}{2} (1 - \cos \phi - \alpha K),$$

(2)

$$M_4 = -M_1 + 2 \frac{F_D R}{3} H,$$

$$M_5 = -M_2 - \frac{F_D R}{3} H,$$

and

$$M_6 = -M_3 - \frac{F_D R}{3} H,$$

were determined by using the equations of equilibrium, symmetry, and compatibility of rotations around the perimeter of the frame. The moments are positive when the outside of the ring is in tension,

$$\begin{aligned} H &= 1 - \sin \phi, \\ K &= H/(2 \sin \phi + 1), \end{aligned} \quad (3)$$

and ϕR is the drag direction distance of F_L from the farthest upstream position of the transducer, the stagnation point.

Given the bending moments (2) and assuming strains due to axial loads are small in comparison to bending strains, the change in resistance for each strain gauge could be calculated. Employing nominally identical gauges in each leg, the sensitivities of the drag and lift force bridges were found to be

$$\frac{\Delta V_D}{V} = GF \frac{2RH}{Ebt^2} F_D, \quad (4)$$

and

$$\frac{\Delta V_L}{V} = GF \frac{3R \cos \phi}{Ebt^2} F_L. \quad (5)$$

The strain-gauge bridges are shown in Fig. 3, and the individual strain gauges are identified in Fig. 1. Note GF is the strain-gauge factor, E the transducer material elastic modulus, b the width of the transducer, and t the thickness of the transducer beam. Theoretically, the effects of the moment ϕF_L cancel out in both bridges, and the effects of F_D in the F_L bridge cancel and vice versa. In practice, minimization of cross sensitivity is a function of the accuracy of manufacturing identical beams with identical strain-gauge bridge legs.

DESIGN DEFINITION

Design strength loads were determined on the basis of a drag coefficient of 1.1 and a lift coefficient of 0.7. Thus, for an active transducer length of three diameters, a bound on the drag and lift forces in turbulent flow at the highest test Reynolds number was 32.0 N (7.2 lb) and 20.0 N (4.5 lb), respectively. Inspection of the moments (2) shows that the largest moment occurs for locations 2 and 5, with a value of 0.183 N-m (1.62 lb-in.), when the lift force is assumed to act through the center of the cylinder, $\alpha = 1$. To achieve a sensing-ring length of one-half tube diameter, the width of the measurement-ring was chosen as 12.7 mm (0.5 in.). A measurement-ring radius of $R = 10.2$ mm (0.4 in.) was chosen based on ease of fabrication. Thus, the maximum bending stress σ_M expected occurred at locations 2 and 5 and was

$$\sigma_M = \frac{86.46}{t^2}, \quad (6)$$

in MPa for t measured in millimeters.

Static deflection of the transducer δ_L in the direction of an applied force F_L was determined utilizing the moments (2) and Castigliano's theorem: $\delta_L = \partial SE / \partial F_L$, where SE is the total internal strain energy. The maximum deflection was expected in the lift direction and for $F_L = 20.0$ N (4.5 lb) was

$$\delta_L = \sum_{n=1}^6 \frac{12 \ell M_n}{E b t^3} \frac{\partial M_n}{\partial F_L} = 1.85 \times 10^{-2} \frac{\ell}{t^3}, \quad (7)$$

in millimeters when ℓ , the length of the beam section of the measurement ring, and t are in millimeters. In deriving (7), only the bending deformation of the transducer ring beams was assumed to contribute to the total strain

energy. Theoretically λ can be reduced to achieve any desired stiffness but practically it is limited by the strain-gauge size.

The minimum (lift) force of interest, 0.22 N (0.05 lb), occurred at a Reynolds number of 10^5 , for a lift coefficient of 0.1, on the shortest active force ring length of 12.7 mm (0.5 in.). Since the strains at gauges 2 and 5 of Fig. 1, due to the lift force, are about a factor of seven larger than the others in the lift-force bridge, the minimum strain $\Delta\epsilon$ to be resolved could be determined utilizing the moment at gauge 2 due to F_L :

$$\Delta\epsilon = \frac{3R(1 + \cos \phi)}{Ebt^2} F_L = \frac{4.54 \times 10^{-6}}{t^2}, \quad (8)$$

with t in millimeters. The elastic modulus of a steel alloy 2.06×10^{11} Pa (30×10^6 psi) was assumed.

Several iterations of the thickness to determine a design showed that the primary constraints on the design were imposed by the minimal strain requirement (8) and the maximum allowable deflection (7). The thickness chosen, $t = 0.38$ mm (0.015 in.), maintained the minimum strains of interest above $25\mu\epsilon$. This was considered the minimum strain allowable to avoid effects of nonlinear time-dependent behavior of the transducer material [11]. An $\lambda = 2.0$ mm (0.08 in.) was chosen to allow easy installation of relatively large strain gauges to minimize backing and bond-line creep. The corresponding stiffness of the transducer according to (7) was found to be

$$\frac{\delta}{F_L} = 3.43 \times 10^{-2} \frac{\text{mm}}{\text{N}} \left(6 \times 10^{-3} \frac{\text{in.}}{\text{lb}} \right). \quad (9)$$

Thus, to minimize the static transducer deflection to less than 0.03 mm (0.001 in.), the fluid loading in the lift direction would have to be maintained

below 0.74 N (0.167 lb). This is considerably less than the 20-N (45-lb) load anticipated on the longest (3 diameters) active transducer ring.

Because the maximum fluid-force estimates were thought to be considerably larger than those which would actually occur, a transducer was constructed, instrumented, calibrated, and tested. Even if the force estimates were accurate, the transducer would be usable for a force-sensing ring of one-half diameter. The transducer material chosen was 17-4 Ph stainless steel whose yield stress 7.585×10^8 Pa (110,000 psi) was more than twice that expected according to (6), again desirable to avoid nonlinear transducer material behavior [11]. The expected sensitivities of the transducer according to (4) and (5) were determined to be

$$\frac{\Delta V_D}{VF_D} = 3.91 \times 10^{-5} \text{ N}^{-1} (1.74 \times 10^{-4} \text{ lb}^{-1}) \quad (10)$$

and

$$\frac{\Delta V_L}{VF_L} = 1.25 \times 10^{-4} \text{ N}^{-1} (5.61 \times 10^{-4} \text{ lb}^{-1}), \quad (11)$$

assuming a gauge factor of 2.

TEST EXPERIENCE

A force transducer was constructed, incorporated into a cylinder, instrumented and calibrated [22]. The predicted transducer sensitivities (10-11) were realized, and the transducer was used to obtain data on fluid forces in turbulent crossflow [23]. Since the forces measured in low turbulence flows were close to those in the literature for nonturbulent flow, the transducer design is considered usable. However, one mundane problem was discovered and resolved. Although strain gauges were used which were self-compensated for temperature variations, not enough space physically existed on the transducer to make each bridge circuit self-temperature compensating [21]. As a result, upon startup of the flow for testing, cooler water would circulate into the confined space where the transducer was located and cause a significant bridge imbalance. This problem was circumvented by zeroing the bridges at 10% or less of the test flow rate. Since the forces are proportional to flow velocity squared, no corrections to the data were required.

Although no major revision to the model design was required based on test experience a modification in the flow channel design was required to obtain force data for two-dimensional flow conditions.

Leakage flow through small annular gaps between the test cylinder and the wind-tunnel walls, as well as the flow channel wall's boundary-layer flow, have been shown [12-18,19] to propagate along the cylinder, interact with the wake flow, and significantly affect the force coefficients and the base pressure in the stagnant region immediately downstream of the cylinder. To prevent such interactions, which make test results unique to the particular flow channel employed in the test, many investigators have employed two end-plates: plates mounted on the cylinder, normal to the cylinder axis, and

axially spaced equidistant from the force sensing ring (see Fig. 4). Square end-plates, 178 x 178 mm (7 x 7 in.), which were 3.18 mm (0.125 in.) thick, were successfully used at an axial spacing of 6.5 to 7.5 cylinder diameters. The center of the holes in the plates for the cylinder was placed equidistant from the top and bottom of the end-plate, but at an unsymmetrical location in the flow direction: 2.5 diameters from the rounded upstream edge of the end-plates [19].

Since the selection of the end-plate lateral separation appears to depend upon the particular flow channel employed, a series of flow tests was performed at different spacings W . The lift coefficients obtained for flows with the same, relatively small, turbulence intensity are shown in Fig. 5. Apparently the end-plates at $W/D = 6.5$ and 7.5 increase the lift coefficients ($C_L = F_L / \frac{1}{2} \rho U^2 D$) to values comparable to those obtained in nonturbulent flow, but smaller spacings of $W/D = 5.0$ and 3.5 increase the lift coefficients even more. Similar measurements have been made [14] in a much larger wind tunnel, where aspect ratios up to $30D$ were possible. The results were similar for small spacings: the lift coefficients increased with decreasing W for $W/D \leq 5$ and were the same for $5 \leq W/D \leq 7.5$. For larger W the results were different: in the wind-tunnel study the lift coefficients remained constant until $W/D = 15$ and then decreased significantly, whereas the data in Fig. 5 show a rapid decrease, with increasing W , for $W/D > 9.75$. The differences were attributed to the differences in channel size, and a spacing of 6.5 - $7.5D$ was employed in the test.

The fact that vortex shedding in nonturbulent flow has been found [12] to be significantly correlated over an axial distance (cell) of approximately six diameters is considered significant. End-plates are intended [20] to isolate, from destruction by end-effects, the highly correlated cells of vortex

shedding which naturally occur in two-dimensional flow. Therefore $W/D < 6$ were never considered for use in the test, because the vortex shedding process apparently is strengthened from that which occurs in two-dimensional flows. Of course, the need for end-plates raises another question about the use of two-dimensional data by a designer. In practical turbulent flows, do end-effects and flow nonuniformities naturally produce three-dimensional vortex shedding and turbulent attached boundary layers? This is a difficult question which can only be investigated, much less answered, on a case-by-case basis. However, the force coefficients obtained with the transducer and end-plates described above are thought to yield, at least, upper bounds on the forces which occur in practical turbulent flows. But these upper bounds are not nearly as large, in many cases, as those previously obtained in nonturbulent, two-dimensional flow. Thus the results obtained with the transducer should be valuable in optimizing designs.

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APPENDIXMAJOR NOMENCLATURE

b	Force-measurement ring width
C	Coherence
C_L	Lift-force coefficient
D	Cylinder diameter
E	Material elastic modulus
f	Frequency
f_v	Vortex shedding frequency
F_L	Lift force normal to cylinder axis and mean flow direction
F_D	Drag force in flow direction
GF	Gauge factor
H,K	Geometric parameters of force-measurement ring. See (3) and Fig. 2.
I	Turbulence intensity
λ	Length of force-measurement-ring beam sections
M	Magnitude of transfer function
M_n	$n = 1, \dots, 6$. Moments at spring locations 1, 2, ..., 6 in Fig. 2.
N_R	Reynolds number
P	Phase of transfer function
q	Dynamic pressure
R	Radius of force-measurement ring
R_i	$i = 1$ to 8, strain-gauge resistances
SE	Internal strain energy
S	Input spectrum
t	Force-measurement-ring thickness (under strain gauge)
U	Mean flow velocity
V	Bridge excitation voltage

$\Delta V_D, \Delta V_L$	Bridge voltage change due to F_D, F_L
W	End plate spacing
α	Fraction of transducer ring radius R . See (2).
δ_L	Lift-direction deflection of transducer
$\Delta \epsilon$	Minimum transducer bending strain
ϵ	Strain
ρ	Fluid mass density
σ_M	Maximum transducer bending stress
ϕ	Angle measurement. See Fig. 3.
ϕ_V	Lift-force spectral density ($\lambda = 0.5 D$)

Fig. 1. Force transducer

Fig. 2. Idealized rigid frame connected by springs (beams) at joints

$$\frac{\Delta V_D}{V} = \frac{R_7 R_8 - R_1 R_4}{(R_1 + R_8)(R_7 + R_4)}$$

(a)

$$\frac{\Delta V_L}{V} = \frac{R_6 R_2 - R_3 R_5}{(R_3 + R_6)(R_2 + R_5)}$$

(b)

Fig. 3. Strain-gauge bridge for: (a) F_D and (b) F_L

Fig. 4. Force-transducer assembly mounted in end supports attached to test-section walls

Fig. 5. Variation in C_L , for $I = 3\%$ and $L/D = 2$, with end-plate spacing W . (Symbol, W/D): Δ , 12; ∇ , 9.75; $\circ$, 6.5; $\square$, 7.5; $\diamond$, 5.0; ∇ , 3.5. Nonturbulent flow curves: 1 [14], 2 [17], 3 [18]