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TUBE CROSSFLOW FORCE MEASUREMENTS

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Introduction

Flow-induced vibrations of structures is a continuing concern in new power-generating system designs. As higher coolant-flow rates are proposed to increase thermal capacity, the fluid forces on the structures are increased. A very efficient, and most typical, configuration for purposes of heat transfer is a circular tube in cross flow. To assess the fatigue or wear, the fluid forces acting on the structure are required to make a vibration analysis. Although the circular cylinder has been the most extensively studied bluff body in cross flow, knowledge of the fluid forcing function is lacking for many flow conditions, especially for the nonuniform turbulent flows prevalent in reactor systems.

Theoretical prediction of flow-induced forces acting on bluff bodies is in the very early stages of development, and their experimental measurement 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 turbulent water cross flow. 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 occurs. 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 corresponding difficulties of the ring force transducer is that a rather complicated multi-component sensing devices must be constructed in a small space for a wide variation in the character of the forces. However, a ring-type 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 signed [4].

The flow test facility available provides a 30.5-cm (12-in.) square cross-section channel with flows up to 5.5 m/s (18 ft/sec) with an available pressure head of approximately one-half megapascal ($\sim$75 psi). A cylinder size of 2.54-cm (1-in.) diameter was chosen to minimize the need for blockage (8%) corrections to the fluid data. For this cylinder, the highest Reynolds number attainable is $\sim 2 \times 10^5$. Turbulence-producing grids under development are expected to yield intensities of 6-20% and length scales of 0.5 to 2 cylinder diameters. Forces on rigid cylinders in turbulent flow are not expected to be appreciably correlated beyond a length of three diameters, based on smooth flow results [5].

Transducer Requirements

Three forces are of interest for a cylinder in cross flow: 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. 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 which 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. Although

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turbulent flows have long been known to affect fluid forces, 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 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

$$f_v = 0.2\ U/D \quad . \tag{1}$$

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. Because of the importance of the fluctuating lift force in production of beam vibrations, the lowest structural frequency of any force-measuring transducer should be well above f_v. A factor of ten was considered adequate, based on a one-degree-of-freedom spring-mass system and limiting dynamic response at the vortex shedding frequency to one percent of the static deflection. The fluctuating drag force is usually an order of magnitude smaller than either the steady drag or fluctuating lift force. It occurs at a frequency twice that of the fluctuating lift, where dynamic response of the transducer would be an acceptable four percent of the static deflection.

Response at the resonant frequency which could mask the force signals would be eliminated by low-pass filtering, a technique made possible by the relatively high transducer resonant frequency requirement. Without filtering the force signal, spurious resonance at the transducer resonant frequency could dominate the force signal. For a relatively high structural damping of ten percent of critical damping and a force ten percent of that occurring around the vortex shedding frequency, not an unreasonable estimate especially for small scale highly turbulent flow, deformation of the transducer at resonant would be comparable to that caused by the vortex-shedding force; again based on a single degree of freedom system.

Static deflection of the active force transducer with respect to the dummy sleeve, Fig. 1, was also a concern. Surface roughness is known to change the character of the vortex shedding process; however, disagreement exists in the literature on the threshold value for which the flow is disturbed: limits on the maximum 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, would present no difficulty. However, the relative deflection of the active force-sensing ring with respect to the dummy sleeve would be difficult to maintain below the least demanding limit while still maintaining a useful transducer sensitivity. Also, construction of two adjoining, structurally separate, cylinders to within the least demanding tolerance is doubtful in practice. The singular, circumferential nature of the two very narrow discontinuities could be argued to have little effect on the wake flow; however, they form the boundaries of the active transducer surface. Thus maintenance of the longitudinal gap and radial discontinuity between the surfaces to be on the order of 0.03 mm (0.001 in.) was considered necessary, at least until more information on its effect becomes available.

The range and magnitude of the fluid forces on the transducer ring was 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 does not appear to exceed that in smooth flow. For the range of test Reynolds number, the steady drag force coefficient is 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 employed in transducer strength considerations.

Flow turbulence affects these coefficients considerably, in particular the lift coefficient becomes an exponentially decreasing function of Reynolds number in the range of interest [7,8]. Because the dynamic head increases quadratically with Reynolds number in the same range, the lift is almost constant. A lift coefficient of 0.1 at a Reynolds number of 10^5 gave an estimate of the minimum force the transducer must resolve. Because the fluctuating drag can be very small in comparison to the other forces, its resolution for all flow conditions is doubtful. Also, turbulence reduces the correlation of the forces along the length of the cylinder with the longest correlation length of approximately three diameters found in smooth flow. To assess the correlation effects, several active force rings of different lengths, from one-half to three diameterms, were considered necessary.

Contradictory information exists as to whether the amplitude of beam motion can influence the vortex shedding process and forces in turbulent flows. The amplitude influence, primarily by increased correlation of the vortex shedding and forces along the length, is quite devastating in smooth flows [5]. 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. Measurement of forces on a rigid cylinder are planned first, but a transducer design which can be incorporated on a flexible cylinder was desired for future studies.

Transducer Design

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 which 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 $\phi = 40°$ 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

$$
\begin{aligned}
M_1 &= \frac{F_D R}{3} H + \frac{F_L R}{2} (1 + 2\alpha K) \quad , \\
M_2 &= -\frac{F_D R}{6} H + \frac{F_L R}{2} (1 + \cos\phi - \alpha K) \quad , \\
M_3 &= -\frac{F_D R}{6} H + \frac{F_L R}{2} (1 - \cos\phi - \alpha K) \quad , \\
M_4 &= -M_1 + 2 \frac{F_D R}{3} H \quad , \\
M_5 &= -M_2 - \frac{F_D R}{3} H \quad , \\
M_6 &= -M_3 - \frac{F_D R}{3} H \quad ,
\end{aligned}
\tag{2}
$$

were determined by utilizing 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,

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

and αK 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 sensitivity of the drag and lift force bridges was found to be

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

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

where GF is the strain gauge factor, E the transducer elastic modules, b the width of the transducer, and t the thickness of the transducer beam. Theoretically, the effects of the moment αF_L cancels 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 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 (45 lb) respectively. Inspection of the moments (2), show that the largest moment occurs for locations 2 and 5, with a value of 0.183 Nm (1.62 lb in.), when the lift force is assumed to act through the center of the cylinder, $\alpha = 1$. In order to have a minimum active ring length of one-half diameter the width of the transducer gauge was chosen as 12.7 mm (0.5 in.), and a force 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 contemplated occurred at locations 2 and 5 and was

$$\sigma_M = \frac{86.46}{t_2} , \qquad (6)$$

in MPa for t measured in millimeters.

Static deflection of the transducer in the direction of an applied force δ_F was determined utilizing the moments (2) and Castigliano's theorem: $\delta_F = \partial U/\partial F$, where U 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

$$\delta = \sum_{n=1}^{6} \frac{12\ell M_n}{Ebt^3} \frac{\partial M_n}{\partial F_L} = 1.85 \times 10^{-2} \frac{\ell}{t^3} , \qquad (7)$$

in mm 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 ℓ 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 due to the lift force for gauges 2 and 5 of Fig. 1 are about a factor of seven larger than the others in the lift force bridge, the minimum strain $\Delta\varepsilon$ to be resolved could be determined utilizing the moment at gauge 2 due to F_L, see (2),

$$\Delta\varepsilon = \frac{3R(1 + \cos\phi)}{Ebt^2} F_1 = \frac{4.54 \times 10^{-6}}{t^2} , \qquad (8)$$

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 $\mu\varepsilon$. This was considered the minimum strain allowable to avoid effects of nonlinear time-dependent behavior of the transducer material [11]. An ℓ = 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

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

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 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) are

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

$$\frac{\Delta V_L}{VF_L} = 1.26 \times 10^{-4}\ \text{N}^{-1}\ (5.61 \times 10^{-4}\ \text{lb}^{-1}) , \qquad (11)$$

assuming a gauge factor of 2.

Results

The transducer was constructed as described and the important output sensitivities, (10) and (11), and the transducer stiffness (9) were measured by dead-weight testing. Values of $\Delta V_L/(VF_L) = 1.27 \times 10^{-4}$ lb^{-1}) and $\Delta V_D/(VF_D) = 3.37 \times 10^{-5}$ N^{-1} (1.50×10^{-4} lb^{-1}) were found, which are close to the values predicted, especially in the lift direction. The stiffness of the transducer was found to be about twice as large as predicted by (9), but the differences may be due to the difficulty in measuring such small deflections. Based on the theoretical stiffness in the lift direction, a fundamental frequency of $\sim$350 Hz was predicted and $\sim$400 Hz was measured — a more reasonable difference. Maximum fluid force frequencies, according to (1), will be $\sim$43 Hz.

Conclusions

A transducer has been designed and verified which has the potential for measuring lift and drag forces on circular rods in turbulent cross flow. The transducer is currently being used to obtain data; experience gained will be related at the time of presentation.

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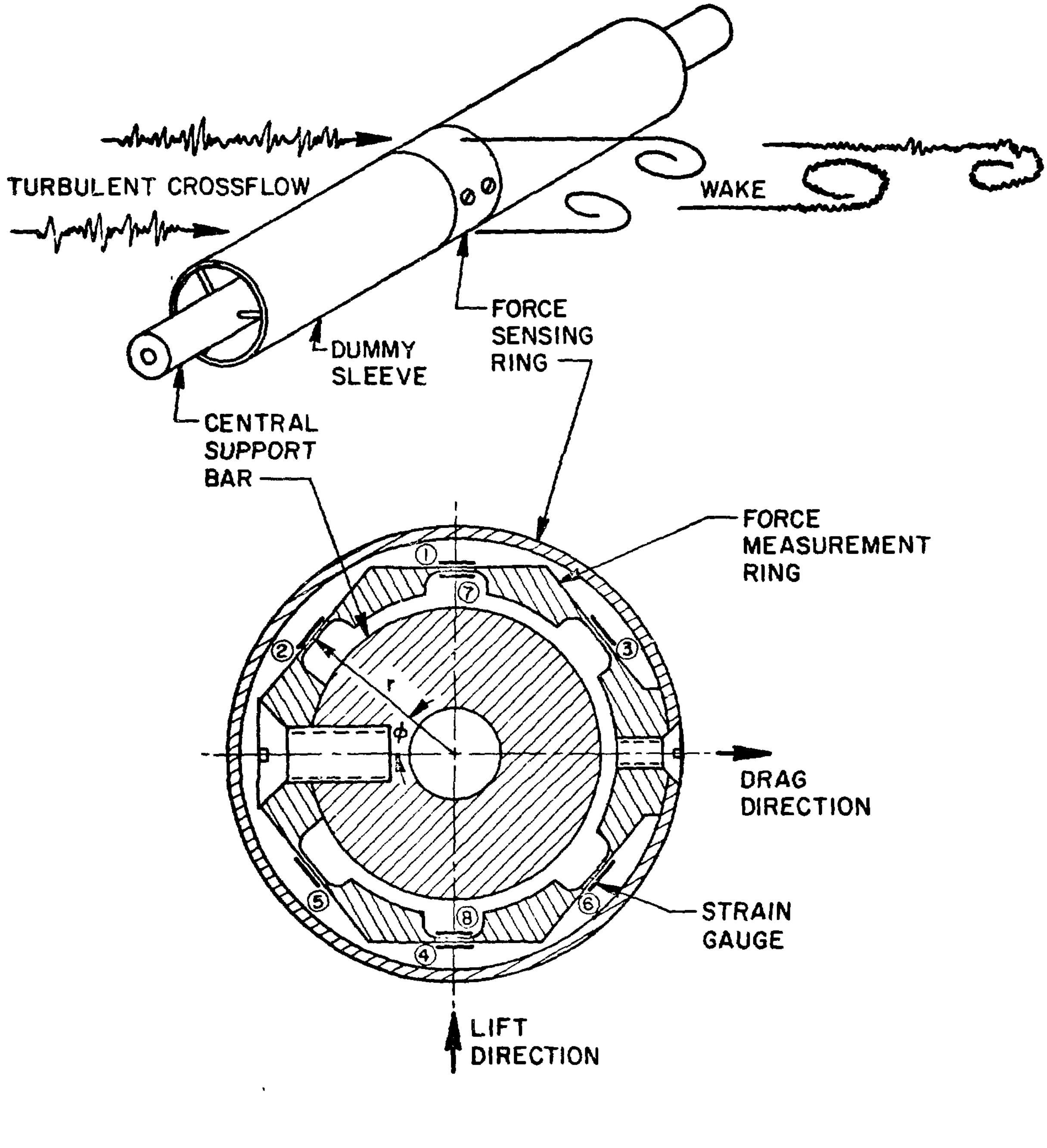


Fig. 1 Force Transducer

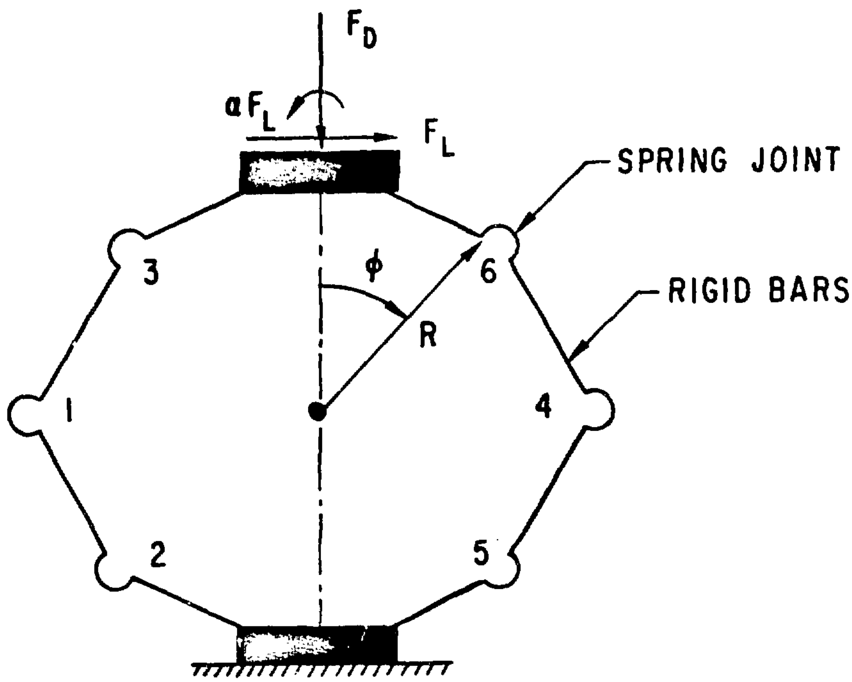
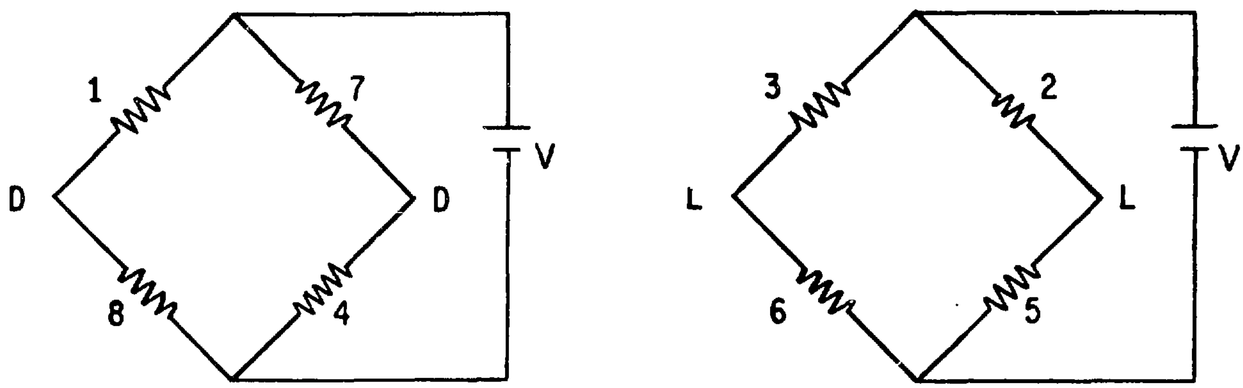


Fig. 2 Idealized Rigid Fram Connected by Springs (Beams) at Joints



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

(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