MEASUREMENT OF UNSTEADY FORCES ON A CIRCULAR CYLINDER IN CROSS FLOW AT SUBCRITICAL REYNOLDS NUMBERS

by

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ABSTRACT

A transducer has been developed to measure the unsteady forces associated with vortex shedding from a cylinder in cross flow. The transducer senses the force on a small isolated section of the instrumented test cylinder and operates in conjunction with an accelerometer implanted in the test cylinder to generate a simultaneous measurement of local force and cylinder motion.

An aeroacoustic measurement program was carried out in the MIT Acoustics and Vibration Laboratory's low noise, low turbulence wind tunnel. The sectional force coefficients, correlation lengths, and radiated sound intensities were determined over a Reynolds number range from 20,000 to 50,000. The sectional lift coefficient was determined to increase with increasing turbulent intensity.

The effect of large amplitude cylinder motion on the vortex shedding process was investigated by using a suitably modified transducer in the Marine Hydrodynamics Laboratory's closed circuit water tunnel. The test cylinder was subjected to forced harmonic motion in a direction transverse to the cross flow at amplitude to diameter ratios of up to 0.5 in the frequency range \(0.1 < \frac{fd}{U_\infty} < 0.3\). Lock-in boundaries were determined for a Reynolds number of 19,300. Both the locked-in and non locked-in behaviours were investigated.

The test cylinder was spring mounted in the water tunnel to determine its vibratory response to vortex shedding excitation. The system showed hysteresis in that the response was different for increasing flow velocities as compared to decreasing flow velocities. The forced vibration data were used to explain this hysteretic behaviour.

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$A_{IH}$Acceleration Sensed by Impedance Head
$A_{IN}$Acceleration Sensed by Internal Accelerometer
$A_T$Measured Transducer Acceleration
$B$Local Force per unit Length
$C$Calibration Factor of Probe (Force/Volt)
$C_6,C_7$Variable Capacitances
$C_D$Mean Drag Coefficient
$C_d'$RMS Sectional Drag Coefficient
$C_{pb}$Base Pressure Coefficient
$C_0$Speed of Sound in Air
$C_L$Total RMS Lift Coefficient
$C_k$RMS Sectional Lift Coefficient
$C_{LM}$Measured RMS Sectional Lift Coefficient
$C_{LT}$True RMS Sectional Lift Coefficient
$d$Cylinder Diameter
$E_{in}$Input Voltage
$E_0$Output Voltage
$F_{IH}$Force Sensed by Impedance Head
$F(z,t)$Force per Unit Length Acting on a Cylinder in Cross Flow
$F_m(z,t)$Measured Force per Unit Length Acting on a Circular Cylinder in Cross Flow
\[ f \quad \text{Frequency (Hz)} \]
\[ f^* \quad \text{Nondimensional Frequency, } f^* = \frac{f_s}{f_r} \]
\[ f_f \quad \text{Forcing Frequency} \]
\[ f_r \quad \text{Resonant Frequency} \]
\[ f_s \quad \text{Shedding Frequency} \]
\[ H(f) \quad \text{Frequency Response Function} \]
\[ K \quad \text{Acoustic Wavenumber} \]
\[ K(s, v) \quad \text{Transducer Response Function} \]
\[ k \quad k = \frac{2d}{\lambda_c} \]
\[ L(f) \quad \text{Fast Fourier Transform of Measured Lift Force} \]
\[ \ell \quad \text{Length of Cylinder} \]
\[ \ell_c \quad \text{Correlation Length} \]
\[ \ell_f \quad \text{Length of Force Sensing Element} \]
\[ M_{IH} \quad \text{Effective Mass of Impedance Head} \]
\[ M_m \quad \text{Mass of Magnet} \]
\[ M_e \quad \text{Effective Mass of Probe} \]
\[ P_b \quad \text{Base Pressure} \]
\[ p(a, \theta, z, t) \quad \text{Pressure on the Surface of a Cylinder in Cross Flow} \]
\[ q \quad \text{Dynamic Pressure} \]
\[ R(z_1 - z_2) \quad \text{Normalized Correlation Function} \]
\[ Re \quad \text{Reynolds Number} \]
\[ St \quad \text{Strouhal Number} \]
\[ S_P(f) \quad \text{Measured Spectrum Level Lift Force} \]
\[ S_u(f) \quad \text{Measured Spectrum Level Velocity} \]
$U_\infty$ Mean Flow Velocity

$u$ Unsteady Flow Velocity

$x$ Distance to Observer

$x_1, x_2$ Displacements

$W(f)$ Fast Fourier Transform of Transducer Output

$\delta$ Amplitude Displacement (1/2 p-p)

$\delta_n$ Amplitude Displacement per Diameter

$\gamma$ Centroid of Positive Correlation Area

$\lambda$ Wavelength of Radiated Sound

$\rho$ Fluid Density

$\theta$ Polar Angle of Observer at $x$

$\nu_a$ Kinematic Viscosity, air

$\nu_w$ Kinematic Viscosity, water
A right circular cylinder in a cross flow undergoes vortex shedding at moderate Reynolds numbers. The vorticies are alternatingly shed by one side of the cylinder and then the other. The vortex shedding induces unsteady forces transverse to the flow direction as well as steady and unsteady forces in the flow direction. These unsteady forces can cause destructive levels of vibration in many common structures such as power lines (Blevins (1)), towing cables (Blevins (1)), and trashracks (Crandall et al (2)). Associated with the unsteady forces is an acoustic radiation called the 'Aeolian Tone' (Phillips (3), Keefe (4), and Leehey and Hanson (5)). The 'Aeolian Tone' is a dipole like sound field.

The shedding of vortices by a cylinder in a cross flow is a narrowband random process. The vortices are shed in a band of frequencies around the Strouhal frequency. The shedding process is also a spatially random process. Although the geometry is two dimensional, the shedding of vortices at two points on the same cylinder is uncorrelated if the spatial separation is great. The spatial scale of the shedding process can be described in terms of the correlation length of the process (Prendergast (6), el Baroudi (7)).
The vortex shedding process is sensitive to several different disturbances such as the freestream turbulence level (Gerrard (8)), surface roughness (Szechenyi (9)), cylinder end conditions (Cowdrey (10), Standsby (11), Gerich and Eckelmann (12)), and motion of the cylinder (Bishop and Hassan (13), Sarpkaya (14), Tanida et al (15), Crandall et al (2)). Increasing the turbulence in the incoming flow tends to increase the unsteady forces on the test cylinder and decrease the correlation lengths. The end conditions are an important consideration in designing an experiment. The desired result is an experiment that is representative of the flow over a two dimensional cylinder.

The cylinder motion plays an important role in the vortex shedding process. The motion can change the character of the flow. It can cause the vortex shedding frequency to change, lock-in, to the frequency of vibration of the cylinder (Bishop and Hassan (13)). It also changes the magnitude and spatial distribution of the forces (Sarpkaya (14), Toebes (16)). The power spectra of the transverse and inline unsteady forces are also influenced by the cylinder motion. Another phenomenon associated with cylinder motion and the vortex shedding process is that there are regions of hysteresis (Crandall et al (2), Sarpkaya (17)).
It is necessary to know both the magnitude and distribution of the unsteady forces on a cylinder in a cross flow to be able to predict either the structural vibration or the acoustic radiation. A method of making a direct measurement of the local force on a moving cylinder has been developed. The force transducer and its calibration are discussed in Chapter II.

An aeroacoustic measurement program was carried out and is described in Chapter III. The force transducer was used to determine the sectional lift forces on a nearly rigid cylinder. The spatial scale of the vortex shedding process was also investigated. The force transducer was used in conjunction with a microphone in the far field to investigate the theory of vortex noise. Phillips (3), Keefe (4), and Leehey and Hanson (5) obtained a relation between the forces on a cylinder and the radiated sound. This work was based on the Lighthill-Curle (18,19) theory of aerodynamic noise. The magnitude of the forces were expressed in coefficient form and the spatial scale was described by a correlation length and a centroid of positive correlation area. This theory is reviewed in section 3.2. The measured sound pressure levels agreed well with the levels predicted by this theory.
The force measurement technique was utilized to investigate the effect of cylinder motion on the vortex shedding process and the results are presented in Chapter IV. The experiments were conducted in the recirculating water tunnel in the MIT Marine Hydrodynamics Laboratory. The effect of forced harmonic oscillation on the shedding process was investigated. The forced oscillation data were utilized to describe the non locked-in motions of a naturally oscillating cylinder. The lock-in boundaries in a nondimensional frequency, nondimensional amplitude space were used to explain a type of double amplitude response of a naturally oscillating cylinder.

The conclusions are presented in Chapter V. The techniques developed here were used to describe the vortex shedding process on a stationary cylinder and an oscillating cylinder that was constrained to move transverse to the flow. The Reynolds number range extended from 5,000 to 50,000.
CHAPTER II. THE FORCE TRANSDUCER

2.0 Force Measurement

Two methods for measuring unsteady forces on a cylinder in a cross flow are to measure the instantaneous pressure distribution around a cylinder and integrate to obtain lift and drag forces, or to directly measure the unsteady forces on the cylinder or on an isolated section of the cylinder. The pressure integration technique has been used by Schmidt (20), Szechenyi (9), Gerrard (21), and Surry (22). The direct method of force measurement has been used by Keefe (4), Kacker et al (23), Bishop and Hassan (24), Sarpkaya (14), and Leehey and Hanson (5). The force coefficient measured on a cylinder depends on the active length of the cylinder. The force coefficient decreases as the active length of the cylinder increases due to lack of correlation of the local forces at large spatial separations. The local force and total force acting on a cylinder are related through the integral correlation length and total length of the cylinder (Kacker et al (23)).

2.1 Transducer Design

The method adopted in this investigation was to develop a transducer to measure the local force at different stations on the same cylinder. The transducer's active length was small.
enough so that the flow was nearly fully correlated over its entire length. This allowed a direct measurement of the local force. The transducer sizing was partially determined by the need to make acoustic far field measurements. The wind tunnel speed range was from ten to fifty meters per second, so the smaller the transducer the better. Also, the smaller the transducer the higher the length to diameter ratio for the tests. A conflicting requirement was that the supporting cylinder had to be large enough for the transducer to fit inside and not disturb the flow. Also, the larger the cylinder the larger the forces to be resolved, thus improving the signal to noise ratio of the transducer. The cylinder diameters chosen were 1.6 cm and 1.9 cm. A small section of the test cylinder was isolated and flush mounted with the cylinder. The active lengths of the force transducers were 0.4 d and 0.5 d respectively.

The elements of the force transducer are shown in Figure 2.1. The transducer consisted of a 1.6 cm diameter aluminum ring, .05 cm thick, supported on a 1.91 cm long cantilever beam element. The beam element was steel, 0.16 cm thick. The beam element was rigidly fixed in a linen laminated phenolic base. Copper plating was epoxied to the phenolic base such that when the transducer was assembled there was a .0127 cm gap between the spring element and each copper plate. The spring element was shimmed such that the gap dimensions were maintained when the transducer was epoxied
together. The ring element was slotted to provide a good bond to the spring element and the then epoxied in place. Next, the transducer electronics were fixed in place and the probe was wired. Finally, the assembled transducer was epoxied in the outer supporting cylinder maintaining a 0.025 cm gap between the ring element and the outer cylinder. The transducer was sensitive to the relative displacement of the beam element to the copper plates fixed in the supporting cylinder. It was primarily sensitive to forces in the normal direction because it was much stiffer in the cross direction compared to the normal direction.

Initially, tests were conducted without the ring element sealed to the supporting cylinder. Abnormally low lift coefficients were measured without seals. Two other investigators had similar problems, Keefe (4), and Kacker et al (23). The gap between the ring element and supporting cylinder was sealed with natural rubber latex. Rubber has frequency dependent properties and is much stiffer at high frequencies than it is statically. Because of the frequency dependent properties of rubber, a static calibration would not necessarily yield a valid calibration for unsteady measurements. The ring element was bevelled to reduce the amount of stiffness the seals contributed to the probe.
The test cylinders were 0.54 meters long and had resonances in the frequency range of interest. The cylinders were filled with sand to provide damping to reduce their vibratory response.

2.2 Electronic Design

The transducer senses displacement of the spring element relative to the phenolic base by means of a capacitive voltage divider. The copper plates and the beam element act as two variable capacitors, with the capacitance depending on the position of the beam element.

The circuit for sensing capacitance change is shown in Figure 2.2. An electrical model of the probe is shown as an inset in Figure 2.2. The coils L2, L3, and L4 are made of ferrite, Ferroxcube 266T125,3D3. All coils were wound with No. 40 wire. The primaries were 44 turns and the secondaries were 88 turns and were center tapped. The operational amplifiers (op-amps) were National Semiconductor LM318. The op-amp A1 and coil L1 comprised a Colpitts oscillator, which drove the circuit. Coil L1 is ferrite, Ferroxcube 1041T060,3D3 and was 40 turns of No. 42 wire. The oscillator output was a stable sine wave 25 volts peak to peak at a frequency of 700 kHz. The op-amp A2 was a buffer between the probe and the demodulator. The coils L3, L4, diodes I1 to I4, and the resistors R1 to R4 comprised a diode ring
demodulator, which is described in Buck (25). The demodulator was phase sensitive, as the voltage divider went through a null, the demodulator output changed sign.

The probe is shown in Figure 2.3. The coil L2 acted as an isolation transformer with a 2:1 turns ratio. This isolation allowed the spring element of the probe to be at ground potential. All exposed parts of the probe were at ground potential. If one neglects the self inductance and coil resistance, the circuit can be modeled as in Figure 2.3. Using node voltage summation, the probe output is

\[
\frac{E_0}{E_{IN}} = 1 - \frac{2C6}{C6+C7}
\]

2.1

The capacitances C6 and C7 vary linearly with the displacement of the spring element relative to the copper plates.

The probe was driven with a 700 kHz constant amplitude sine wave. The probe output, E₀, was an amplitude modulated (AM) signal. By amplitude modulating the signal, we were able to avoid complications due to the leakage resistances of the variable capacitors at low frequencies. This kept the impedance values finite and manageable. The choice of source frequency depended on the bandwidth of the measured signals and power requirements of the coils. To avoid errors in demodulating the signal, the carrier frequency was much higher than the signal bandwidth. The probe output was an AM signal, which was buffered by op-amp A2 and demodulated by a diode
ring demodulator. The resulting output was a voltage linearly related to the relative displacement of the beam element to the copper plates fixed in the phenolic base.

2.3 Transducer Calibration

A magnet and coil calibrator was developed to calibrate accurately the force transducer. A 0.32 cm diameter permanent magnet, 1.27 cm long was rigidly attached to the ring element. The magnet was immersed in the coil a known and controlled distance. A current was put through the coil, resulting in an electromotive force on the magnet and transducer assembly proportional to the current.

The magnet and coil calibrator was calibrated by fixing the magnet to a Bruel and Kjaer 8001 impedance head. The magnet was placed in the coil and its position was varied. A voltage proportional to the current through the coil was sensed as well as the the force, $F_{IH}$ and acceleration, $A_{IH}$ outputs of the impedance head. The impedance head had mass, $M_{IH}$ moving ahead of the force guage and this effect was important. A model for the calibration is

$$F_{EM} = F_{IH} + (M_m + M_{IH}) A_{IH}$$

where $F_{EM}$ is the electromotive force and $M_m$ is the mass of the magnet.
The electromotive force depends on the position of the magnet in the coil. If the magnet was placed symmetrically in the coil or far from the coil the electromotive force was small. The optimum placement of the magnet in the coil resulted in the electromotive force being maximum. The variation of electromotive force with position is shown in Figure 2.5. At optimum position the magnet and coil calibrator was insensitive to the motion of the magnet in the coil. The displacement of the beam element to the copper plates was less than $10^{-6}$ meters.

The magnet and coil calibrator was used to do a broadband calibration of the force transducer, as shown in Figure 2.5. A white noise generator was used to drive the magnet and coil calibrator. The signals from the calibrator and force transducer were preamplified and filtered by Ithaco 452 preamps and Ithaco 4112 variable electronic filters. The data were then tape recorded on a Tandberg FM tape recorder. The taped data were analyzed on a DEC PDP 11-34 computer using the General Radio Time Series Analysis Package. The data were digitized on a ten bit analog to digital converter. A fast Fourier transform was computed on the digitized data. The spectral estimates were averaged to obtain stable estimates of the desired frequency domain quantities. The fast Fourier transforms were used to generate estimates of the cross-spectrum and the power spectra for both channels of data. The cross-spectrum was used to calculate the transfer
function and coherence. The signals dealt with here were assumed to be both stationary and ergodic.

A broad band calibration of the force transducer was done by exciting the calibrator coil with band limited white noise. The Time Series Analysis Package was used to estimate the transfer function, power spectra, and coherence. A sample calibration is shown in Figure 2.6. The transfer function is the transducer output (per unit force input) plotted against frequency. The important features of the transfer function are: that there was a resonance at 1.6 kHz indicated by a peak in magnitude and a 180 degree phase shift, and that the transfer function well below the resonance is flat and of zero phase. The coherence between the input and transducer output is shown in Figure 2.7. The coherence was good, indicating that the transfer function estimates were valid and that the system was linear. The behavior of the transducer can be modeled as a simple mechanical oscillator: mass, spring, and dashpot. The stiffness came from the beam element and transducer seal. The effective mass was from the ring element and the beam element. The transducer, being a mechanical device, had mass and responded to acceleration as well as to the force on the ring element. Because of this, it was necessary to add an accelerometer to the transducer to do acceleration equalization.
2.4 Acceleration Equalization

Because the transducer was sensitive to the motion of the supporting cylinder, it was necessary to place an accelerometer in the test cylinder, as shown in Figure 2.1. Ideally, the accelerometer would be placed at the base of the beam element, but in practice this could not be done. Due to lack of proximity of the accelerometer to the beam element support, it was not possible to do real time acceleration compensation. A method was devised to do acceleration equalization in the frequency domain and is described next.

When the experiment was performed, the acceleration, force, and microphone signals were recorded. A second experiment was performed to do the acceleration equalization. The sensing element of the transducer was covered so there were only inertial loads on the transducer. The signals from the transducer and the accelerometer were recorded. The desired result was a transfer function between the compensating accelerometer and the transducer output with no aerodynamic load on the sensing element and with the cylinder motion the same as in the test case. The wind tunnel was run at the same speeds for both the experiments and acceleration equalization tests. The loading on the cylinder was similar for both tests. The tests were conducted in the same test stand and so a transfer function calculated from the latter case may be applied to the former. The hypothesis is that
The transfer function for the case with only inertial loads can be applied to the test case. This should be correct when the cylinder was vibrating in a resonant mode where the inertial loads were the largest. Using this result, the fast Fourier transform of the lift force, \( L(f) \) can be calculated

\[
L(f) = C \left\{ W(f) - \left\{ \frac{W}{A_{IN}}(f) \right\}_{F=0} A_{IN}(f) \right\}
\]

The lift force is the transducer output minus the inertial loading.

2.5 Resolution of a Transducer of Finite Size

The finite size of the force transducer limited its spatial resolution of the force field associated with vortex shedding. The lack of resolution caused a decrease in the measured lift coefficient compared to a transducer of infinitesimal size. A one dimensional theory similar to the two dimensional theory of Corcos (26) is presented here. The results indicate that the attenuation due to finite size was not severe for the cases studied here.

Consider a right circular cylinder in a cross flow as in Figure 2.8. The force is measured on a length, \( l_F \), of the cylinder. The flow field is assumed to be stationary and
homogeneous. There is a pressure on the cylinder due to the flow field, \( p(a, \theta, z, t) \), where \( a \) is the cylinder radius. The lift per unit length, \( F(z, t) \), is the integral of the pressure around the circumference,

\[
F(z, t) = \int_0^{2\pi} p(a, \theta, z, t) \sin \theta \, d\theta. \tag{2.5}
\]

Let \( K(s, v) \) be the response of the force transducer at time \( v \) due to a unit force per unit length at location \( s \). The measured force per unit length \( F_m(z, t) \) is

\[
F_m(z, t) = \int_{-\infty}^{t} \int_{-\infty}^{\infty} F(s, \gamma) K(s-z, \gamma-t) \, ds \, d\gamma. \tag{2.6}
\]

Equation 2.6 can be simplified by assuming the transducer is an ideal transducer. The ideal transducer responds instantaneously and it responds only to forces on the sensing element. The instantaneous response is approximately true for this transducer well below its resonant frequency. That is, where the transducer response is flat in frequency and in phase with the force. The response kernel \( K(s, v) \) simplifies to

\[
K(s, v) = \begin{cases} \frac{1}{\lambda_F} & 0 < s < \lambda_F, -\infty < v < \infty \\ 0 & \text{otherwise} \end{cases} \tag{2.7}
\]

for an ideal transducer. The measured force simplifies to

\[
F_m(\xi_F, t) = \frac{1}{\lambda_F} \int_0^{\lambda_F} F(z, t) \, dz. \tag{2.8}
\]
The mean square force per unit length is

$$\frac{E[F_M(\ell_F,t)^2]}{E[F(\ell_F,t)^2]} = \frac{1}{\ell_F^2} \int_0^{\ell_F} \int_0^{\ell_F} R(z_1-z_2) dz_1 \, dz_2$$  \hspace{1cm} 2.9

where \( R(z_1-z_2) \) is the normalized correlation function. This can be integrated to yield (Frenkiel (27)),

$$\frac{C_{\ell M}}{C_{\ell T}} = \sqrt{2 \int_0^{\ell_F} R(z)(\ell_F-z) \, dz}$$  \hspace{1cm} 2.10

where \( C_{\ell M} \) is the measured sectional lift coefficient and \( C_{\ell T} \) is the true value. This is the result derived by Kacker et al (23).

The result of Equation 2.10 is that the effect of finite size can be predicted once the correlation function is known. However this function is not known apriori. The force correlations were measured and the measured results were used to determine the effect of finite size.
CHAPTER III. WIND TUNNEL EXPERIMENTS

3.0 Introduction

The force transducer was used in the M.I.T. Acoustics and Vibration Laboratory's low noise, low turbulence wind tunnel. The wind tunnel is described in section 3.1. In section 3.2 the Lighthill-Curle theory of aerodynamic noise is reviewed. The effect of end conditions on vortex shedding is discussed in section 3.3. In section 3.4 the relationship between the velocity in the potential flow near the cylinder and the unsteady force is presented. Section 3.5 deals with two point velocity correlations. The results of the force measurements are presented in 3.6. The force and sound measurements are presented in section 3.7. The effect of freestream turbulence on vortex shedding is presented in section 3.8.

3.1 Experimental Facility

The experiments were carried out using the facilities and equipment of the Acoustics and Vibration Laboratory at MIT. The low noise, low turbulence wind tunnel is shown in Figure 3.1. The basic construction characteristics of this tunnel have been described by Hanson (28). The tunnel has been modified by Shapiro (29). The wind tunnel is of open circuit construction with the test section enclosed in an
air tight blockhouse. The tunnel consists of a set of flow straighteners, screens, a settling chamber, and a 20:1 contraction leading to a square test section 38 cm by 38 cm, 4 meters long. The wind tunnel was operated in a free jet mode.

The flow in the free jet was measured using a DISA 55A22 hot wire probe with a DISA 55DO5 hot wire annemometer linearized by a DISA 55D15 linearizer. The mean velocity profile and fluctuating velocity profile are shown in Figure 3.2. The mean velocity profile is flat across the center span of the wind tunnel and the freestream turbulence is 0.3%. The spectrum level of the longitudinal component of the turbulence is shown in Figure 3.3. The spectrum shows a peak at 3 Hz which was associated with the breathing mode resonance of the wind tunnel. The broad peak at 25 Hz was associated with unsteadiness of the free jet.

The tests were conducted inside the blockhouse. The blockhouse walls, floor, and ceiling were treated with urethane foam to provide a semi-anechoic chamber to allow measurements of the Aeolian tone. The semi-anechoic treatment consisted of a base of 10 cm of foam, then a set of 10 cm foam blocks spread randomly, and finally a set of 2 cm foam blocks spread randomly. This arrangement was suggested by Crémér (30). The effectiveness of the semi-anechoic treatment was verified by using a horn driver
and measuring the sound field decay with distance. The blockhouse treatment was effective between 200 Hz and 2000 Hz. The background noise in the blockhouse is shown in Figure 3.4 for a flow velocity of 40 m/sec.

3.2 The Aeolian Tone

The Aeolian tone has been investigated by Phillips (3), Keefe (4) and Leehey and Hanson (5) to verify the Lighthill-Curle (17,29) theory of aerodynamic noise. Phillips (3), Keefe (4), and Leehey and Hanson (5) arrived at a relation which gave sound intensity and radiation pattern from a cylinder undergoing vortex shedding. The Aeolian tone is a dipole-like radiation associated with unsteady lift forces on a cylinder in a cross flow.

The theory of the Aeolian tone relates the intensity, I(x), of the radiated sound to the sectional lift coefficient, C_L, the correlation length, l_c, and a centroid, γ, of positive correlation area. The correlation length, l_c, is defined by

\[ l_c = 2 \int_0^\infty R(z)dz \]  

3.1

where R(z) is the normalized force correlation function. The centroid, γ, is defined by

\[ γ = \frac{2}{l_c} \int_0^\infty zR(z)dz \]  

3.2
The rms sectional lift coefficient, $C_{\ell}$, is defined by

$$C_{\ell} = \frac{\langle F^2 \rangle^{1/2}}{1/2 \rho U_{\infty}^2 d} \quad 3.3$$

where $F$ is the force per unit length. The result is

[Phillips (3), Keefe (4), Leehey and Hanson (5)]

$$I(x) = \frac{1}{16} \frac{c}{C^3} \frac{\cos^2 \theta}{x^2} \frac{C_{\ell}^2 U_{\infty}^6 (\lambda - \gamma) \lambda^2 \text{St}^2}{x^2} \quad 3.4$$

where $\theta$ is the polar angle of the observer at $x$ from the normal to the cylinder and to the direction of the uniform flow, and $x$ is the distance to the observer. The assumptions used to derive Equation 3.4 are; the flow is a very narrowband random process with a constant Strouhal number $f_s d / U_{\infty}$, the flow is homogeneous and stationary, the mach number, $U_{\infty} / C$ is small, that $|x| >> 1$, and $\lambda C << \lambda$, where $\lambda$ is the wavelength of the radiated sound. Leehey and Hanson (5) measured all the relevant physical quantities in the same experiment. They measured the modal force on the cylinder and used previously determined correlation lengths to determine the sectional lift coefficients from the modal force. Both Keefe (4) and Leehey and Hanson (5) used two point hot wire correlations to determine the spatial scale of the vortex shedding process. Keefe (4) measured the local force directly but did not use the same apparatus to determine the correlation lengths. Phillips (3) also used data from separate experiments. Leehey and Hanson (5)
observed good agreement between the theory and their experimental results. In this paper measurements were made of local force, radiated sound, and lateral spatial correlations in the same test apparatus.

3.3 Experimental Configuration

The experiments were conducted in an open jet inside an airtight blockhouse. Test cylinders of 1.6 cm and 1.9 cm diameter were placed in a test stand downstream of the exit plane of the ducting. The test set up is shown in Figure 3.5. The velocity profile for the free jet is shown in Figure 3.2. The velocity profile shows a region of uniform flow, 30 cm long, which was bounded on either end by a turbulent mixing region. The test cylinders penetrated the free jet. Ideally the test cylinder would be of very large aspect ratio (length divided by diameter) so that the measurements would not be influenced by the end conditions. The desired result was one that would be representative of a two dimensional cylinder in a uniform flow. The aspect ratios for the cylinders used were 19 and 16 respectively, but the cylinder end conditions were still an important consideration in the design of the experiments. The effects of end conditions were investigated by flow visualization, mean pressure measurements, force measurements, and radiated sound measurements.
An oil film flow visualization, shown in Figure 3.6, was done to qualitatively check the influence of the test cylinder penetrating the jet mixing region. The flow visualization was done using a mixture of kerosene and titanium dioxide. The mixture was painted on the test cylinder. The kerosene evaporated leaving the titanium dioxide on the cylinder surface. The flow visualization shows a uniform separation line on the front face of the cylinder over the region of potential flow of the jet. The separation line migrates to the back face of the cylinder in the mixing region. An oil drop flow visualization was also performed on the test cylinder using the titanium dioxide and kerosene mixture. Drops of the kerosene and titanium dioxide mixture were placed on the back face of the test cylinder and the motion of the drops was observed. An oil drop visualization is shown in Figure 3.7. It shows an axial inflow from the high pressure region outside the mixing layer along the low pressure region on the back face of the test cylinder.

The result of the flow visualizations is that the flow is highly three dimensional with the separation lines moving from the front face of the cylinder to the rear and an axial inflow along the rear of the cylinder. These flow visualizations point to a need to control the end conditions for a cylinder spanning a free jet.
The method selected for controlling the end conditions was to fix endplates to the test cylinder. Cowdrey (10) investigated the use of endplates on a rectangular section bluff body. He measured an increase in drag for models with endplates over models without. This was also investigated by Obasaju (31) with similar results. Standsby (11) investigated the effects of endplates on the base pressure of cylinders spanning a wind tunnel. He demonstrated that the base pressure for a cylinder with endplates was lower for than for a cylinder without endplates, which is consistent with Cowdrey's (10) results on mean drag. He further showed that the endplates made the base pressure more uniform across the span as compared to no endplates.

The endplates used in the current experiments were ten cylinder diameters in diameter. Experiments were conducted with and without end plates to have a basis for comparison.

Oil film flow visualizations were done on the cylinder and the end plates in the test stand. The separation line on the test cylinder, Figure 3.8, was a straight line up to within 0.25 diameters of the endplate. The mean flow pattern on the endplate shows a cell pattern downstream of the cylinder. It also shows a separation bubble on the leading edge of the endplate (the region on the endplate were the titanium dioxide was not moved) followed by a reattachment as evidenced by the motion of the mixture. Oil
drop flow visualizations on the back face of the cylinder, Figure 3.10 showed that there was no axial inflow. Figure 3.11 is an oil drop flow visualization on the endplate. There was a stagnation point downstream of the cylinder.

A measurement program similar to Stansby (11) was done on the axial variation of back pressure with and without endplates and is shown in Figure 3.12. The back pressure with endplates is lower than without. The back pressure is also more uniform for the case with endplates as compared to the case without.

The placement of the endplates was determined by measuring the back pressure at the center of the cylinder as a function of endplate placement. The result is shown in Figure 3.13. The back pressure was highest when the endplates were in the mixing region, decreased and leveled off as the endplates entered the potential core of the jet, and decreased again as the endplates moved to within a correlation length. The endplates were placed at the inside edge of the jet mixing region, which was where the endplates were effective and still gave the maximum length to diameter of the test cylinder.
The flow visualizations indicate that the flow between the endplates was two dimensional and there was no axial inflow at the end of the active length of the test cylinder. The mean pressure measurements showed that the back pressure was lower and more uniform compared to the case of no endplates.

Hot wire measurements were made to determine if the shedding frequency changed as the endplate was approached. At much lower Reynold's numbers this was observed by Gerich and Eckelmann (12). There was no change of shedding frequency observed from the center of the test cylinder to within 0.25 cylinder diameter of the endplate. This discrepancy may be due to the difference in Reynolds number of the tests.

The force transducer was used in conjunction with a microphone in the blockhouse to make comparisons between a cylinder without endplates to a cylinder with endplates at the same flow velocity. The endplates caused a dramatic change in the unsteady flow. The Strouhal number changed from 0.20 without endplates to 0.19 with endplates. The variation the rms lift coefficient at the center line of the test section with Reynold's number is shown in Figure 3.16. The results are summarized in Table 3.1. The corresponding spectrum levels of the unsteady lift force are shown in Figure 3.15. The plot shows the general character of the
vortex shedding process: a narrow peak at the shedding frequency. The endplates caused the peak to occur at lower frequency and the lift force was greater for the case of the cylinder with endplates. The spectral peak was also broader for the case with the endplates.

The spectrum levels for the radiated sound pressure are shown in Figure 3.16 and listed in Table 3.1. The radiated sound pressure levels were from 7 dB to 11 dB higher for the cylinder with endplates as compared to the cylinder without. The increase in radiated sound pressure levels were primarily due to the increase of the lift forces on the cylinder with endplates and not due to changes in the correlation length. The coherence between the radiated sound pressure and the local force is shown in Figure 3.17. The coherence is higher at the shedding peak for the cylinder without endplates than for the cylinder with endplates. Because of the highly three dimensional nature of the flow over the cylinder without endplates, it is not possible to use equation 3.4 to interpret these results.

The lift and drag forces are related for a cylinder undergoing vortex shedding. The base pressure was lower for a cylinder with endplates as compared to a cylinder without as can be seen in Figure 3.12. The base pressure is an indication of the mean drag force. The lower the base pressure the larger the mean drag force. The lift force was
larger for the case of the cylinder with endplates. The stronger vortex shedding resulted in a larger mean drag force. It also resulted in increased acoustic radiation.

The coherence between the lift force and the radiated sound (Figure 3.17) was higher for the case of no endplates as compared to the case with endplates. This was due to the force at the center of the cylinder being the dominant radiator for the case without endplates. The local force was measured at the center of the cylinder for both cases. In Figure 3.12 the base pressure at the center of the cylinder without endplates was a minimum. This implies that the local force was a maximum at the centerline. This result is consistent with the fact that the force at the center dominates the acoustic radiation for the cylinder without endplates.

The pressure distributions were measured around the cylinder with endplates in place. The drag coefficients for this Reynolds number range were around 1.20 and were calculated from the pressure distributions. The Strouhal number for this Reynolds number range was 0.19. The results are summarized in Table 3.1 and are consistent with data presented in the literature.
There is no consistent trend in the literature over use of endplates compared to not using endplates. Bishop and Hassan (24) did not use endplates initially in their experiments. After placing endplates on the cylinder the lift signal increased. Humphreys (32) noted a sensitivity to the end conditions of the test cylinder, as did Gerrard (8). Keefe (4) used endplates in his experiments. He noted that there was an increase in the lift as the endplates were moved to within a correlation length of each other. Graham (33) investigated the effect of endplates on correlations for D section cylinders. When moved close enough together the flow was fully correlated between the end plates. Kacker et al (23) did not use endplates in their force measurements and attributed some of their results to possible three dimensional flow. Bishop and Hassan’s (24) result that the lift increased when endplates were added was partly due to the endplates correlating the flow, their cylinder was nine diameters long which was smaller than a correlation length at the Reynolds’s number of their tests, but it may also have been partly due to the endplates changing the lift.

The use of endplates was adopted in an effort to have repeatable end conditions. Further experiments are needed to determine if the cylinders with endplates have the same local behaviour as a cylinder of large aspect ratio. For the case of a cylinder spanning a free jet, endplates
isolate the test cylinder from the end conditions and provide a region of two dimensional mean flow for experimentation. Standsby showed that end effects are also important for cylinders spanning a wind tunnel test section. Endplates were used in all of the experiments in both the wind tunnel and the water tunnel.

3.4 Force-Velocity Measurements

It is necessary to know both the magnitude and distribution of the forces to be able to predict either the cylinder vibration or the radiated sound field. Because of the comparative difficulty of making two point force correlations as compared to two point velocity correlations, the relationship between the velocity in the potential flow near the cylinder and the local force was explored. The test set up is shown in Figure 3.18. The velocity was sensed just outside the separated shear layer at ninety degrees from the flow direction by a DISA 55D22 hot wire probe followed by a DISA 55D05 hot wire anemometer. The anemometer output was linearized by a DISA 55D15 linearizer. The signals from the force transducer, compensating accelerometer, and the hot wire were preamplified by Ithaco 452 low noise preamplifiers and filtered by Ithaco 4112 variable electronic filters. The signals were recorded on a four channel Tandberg FM recorder. The data were analyzed as in section 2.3.
The test cylinder was placed in the wind tunnel and the hot wire was positioned as in Figure 3.5. Air flow was initiated and stabilized. Data for a range of velocities were recorded. The hot wire was removed and the sensing element was covered with a shield and the experiment repeated for the acceleration compensation, following the algorithm developed in section 2.4. The data were played back into the computer and plotted.

The power spectrum levels for the force signal and for the fluctuating velocity are shown in Figure 3.19. The force and velocity spectra are similar, both showing the dominant peak at the Strouhal frequency. The coherence between the force signal and the velocity signal is plotted in Figure 3.20. The coherence is good around the Strouhal frequency, where there is a significant amount of energy in the signals as can be seen from their power spectra. The transfer function was flat and in phase around the Strouhal frequency. The velocity and force signals were coherent, in phase, and had a flat transfer function. The result is that two point velocity correlations are good estimators of the two point force correlations for a nearly rigid cylinder.
3.5 **Two Point Velocity Correlations**

A measurement program of two point velocity correlations was carried out in the low noise, low turbulence wind tunnel. The two point velocity correlations were used to determine the correlation length of the vortex shedding process.

A pair of hot wires were used to do two point velocity correlations on a nearly rigid cylinder. The probes were positioned as in Figure 3.21, and the separation distance between the probes was varied. DISA 55A22 hot wire probes were used with DISA 55D05 hot wire anemometers. The hot wires were linearized using Analog Devices 433B Programmable Multifunction Modules. The linearized hot wire outputs were preamplified and filtered using Ithaco 452 low noise preamplifiers and Ithaco 4112 variable electronic filters. Two sets of data were obtained, one with 1.6 cm diameter cylinders and the other with 1.9 cm diameter cylinders. The data taken on the 1.6 cm diameter cylinders were tape recorded on a Tandberg FM tape recorder and analyzed as in section 2.3. The data taken on the 1.9 cm diameter cylinder were reduced in real time using a Hewlett-Packard 4324A Structural Dynamics Analyzer and plotted on a Hewlett-Packard 9872S digital plotter. The Structural Dynamics Analyzer had a 12 bit analog to digital converter and used Hewlett Packard generated software to compute fast Fourier
transforms, from which estimates of the power spectra, coherence, correlation functions, and transfer functions were obtained.

The data from the two point velocity correlations are shown in Figure 3.22 through Figure 3.26. Figure 3.22 shows the velocity spectra which demonstrates the characteristic peak in fluctuating velocity at the Strouhal frequency. The coherence between the two hot wire outputs is shown for two cases in Figure 3.23. In the first case, the probes were one diameter apart and the velocity signals were well correlated. In the second case the probes were ten diameters apart and the velocity signals were uncorrelated. For separations greater than this distance the vortex shedding process was uncorrelated at these Reynolds numbers.

The spatial correlation function, $R(z)$, is shown in Figure 3.24. Also shown in Figure 3.24 is an exponential decay with the same correlation length. The exponential decay fits the data well.

The correlation length is defined in Equation 3.3 and is the integral from $-\infty$ to $\infty$ of the spatial correlation function. The correlation length was computed from the spatial correlation coefficients listed in Table 3.2 and is shown as a function of Reynolds number in Figure 3.25. The correlation lengths were estimated by using the trapezoidal rule. The centroids of the correlation curves are listed in
Table 3.2 and were also computed using the trapazoidal rule. The trend in the data is for the correlation length to decrease with increasing Reynolds number.

A comparison of the current two-point velocity correlations with those of previous investigators is made in Table 3.3 and shown graphically in Figure 3.26. Two-point velocity correlations have been made by el Baroudi (7), Leehey and Hanson (5), Toebes (16), and Kacker et al (23). Two-point pressure measurements have been made by Prendergast (6), and Bruun and Davies (35). Surry (22) used integrated pressure correlations to estimate the unsteady force correlations. Humphreys (32) used a flow visualization technique. Kacker et al (23) used the force on different lengths of test cylinder to estimate correlation lengths. To date there has been no direct measurement of the unsteady force at two points on the same cylinder. Also there is a lack of correlation length information in water. The agreement between the various techniques is good.

The exponential fit to the spanwise correlation function was used with the results of Section 2.5 to check the effect of the finite resolution of the force transducer. The result is:

\[
\frac{C_{\ell M}}{C_{\ell T}} = \sqrt{2 \left( k \ell_{F} + e^{-k \ell_{F}} - 1 \right)}
\]

(3.5)
where \( k = 2d/\lambda_c \) and \( \lambda_c \) is the length of the transducer sensing element. The lift was underestimated by 2.3\% in the worst case and so no correction has been made to the data for finite resolution effects.

3.6 **Unsteady Force Results**

The force transducers were used to measure the unsteady forces on a cylinder undergoing vortex shedding. The experimental set up was the same as in Figure 3.27. The data were recorded and then reduced as in section 2.4. The background electrical noise spectrum levels for the experiments correspond to a force spectrum level of \(-80 \text{ dB re } 1\text{N}^2/\text{Hz}\). The transducers were calibrated using the magnet and coil calibrator developed in section 2.3.

The transducers were used in the Reynolds number range from 5,000 to 50,000. The spectrum level for the lift force is shown in Figure 3.28 for a Reynolds number of 42,000. The spectrum shows the peak associated with the vortex shedding at the Strouhal frequency. The Strouhal number is 0.19. The low frequency rolloff is due to bandpass filtering of the signal. The variation of lift coefficient with Reynolds number is shown in Figure 3.29. The lift coefficient was slowly varying in the Reynolds number range from 20,000 to 50,000 and the Strouhal number was constant.
The current results are compared to those of other investigators in Table 3.4 and the comparison is made graphically in Figure 3.30. The total force on a cylinder was measured by Humphreys (32), Bishop and Hassan (24), and Tanida et al (15). Leehey and Hanson (5) measured the modal force and used velocity correlations to reduce the data to a local force coefficient. The local force coefficient was measured by Keefe (4) and Kacker et al (23). McGregor (34), Gerard (21), and Surry (22) integrated pressure distributions to determine the local force coefficients. The agreement of the data is not too good. The reasons for this poor agreement are due to the sensitivity of the vortex shedding process to many different disturbances. The end conditions are of notable importance. Bishop and Hassan (24) got different results with and without endplates as did Keefe (4). Gerrard (8) reported sensitivity to end conditions as did Humphreys (32). Kacker et al (23) attributed some of their results to possible end effects. The shedding process is also sensitive to the freestream turbulence. Cylinder vibration also can influence the vortex shedding process. Because of all these effects it is not possible to collapse the reported data.
3.7 **Force-Sound Measurements**

In this section the measurements of the local force and radiated sound were done simultaneously. The measurements spanned the Reynolds number range from 20,000 to 50,000. The lower limit was determined by the shedding frequency, which had to be above 200 Hz, where the semianechoic treatment was first effective. The upper limit was determined by the top speed of the wind tunnel.

The experimental set up is shown in Figure 3.27. The force transducer was placed in a stand in the wind tunnel. A microphone was placed inside the blockhouse and out of the flow, one meter from the cylinder in a plane containing the cylinder but perpendicular to the flow direction. The signals from the microphone, force transducer, and compensating accelerometer were recorded on a Tardberg FM recorder. The wind tunnel was run and data recorded. Then the force transducer was covered and the experiment repeated to get data for the acceleration compensation per section 2.4.

The spectrum levels for the force and radiated sound are shown, Figure 3.31. The coherence between the force and sound signals is an indication of the correlation length, the lower the coherence the more independent sources in a length, \( \ell \) of cylinder, i.e. the shorter the correlation length. The coherence between the force and radiated sound
is shown in Figure 3.32. The root mean square lift coefficient and the sound pressure level were determined from their respective power spectra. The lift coefficient and correlation length, were used in Equation 3.4 to predict the sound pressure levels at the microphone position. Table 3.5 gives a comparisons of predicted and measured sound pressure levels for a single cylinder undergoing vortex shedding. There was good agreement between the predicted levels and the measured levels. Equation 3.4 tends to overpredict the sound pressure levels. This is the same trend as reported by Leehey and Hanson (5). The good agreement between the predicted and measured sound pressure levels indicate that the Lighthill-Curle (18,19) theory is valid.

The implication is that if the flow velocity and Reynolds number are known it is possible to get a good estimate of the sound field radiated from a cylinder undergoing vortex shedding. This does not mean that the intensity of the radiated sound goes as $U_\infty$ raised to the sixth power, because the lift coefficient and correlation length are functions of Reynolds number. The lift coefficient and correlation length also depend on several other parameters including surface roughness, freestream turbulence, and cylinder motion. In fact, it is possible to observe substantial departure from $U_\infty$ raised to the sixth power in a range of Reynolds numbers where the lift
coefficient and correlation length are rapidly changing, (Leehey and Hanson (5)).

Dimensionalizing Equation 3.4 allows a physical interpretation of the aeroacoustic experiments. The result is:

\[ \langle p^2 \rangle = \frac{1}{16\pi^2} k^2 \cos^2 \theta \langle B^2 \rangle \frac{x^2}{x^2} \frac{\lambda_c (\ell - \gamma)}{\lambda_F^2} \]

where \( K \) is the acoustic wavenumber and \( B \) is the force on the ring element. Equation 3.6 is a dipole source distribution of \( \ell - \gamma / \lambda_c \) sources each with strength \( \langle B^2 \rangle \lambda_c^2 / \lambda_F^2 \). That is to predict the radiated sound, it is necessary to know the unsteady forces and force distribution. Velocity does not appear explicitly in Equation 3.6. Leehey and Hanson (5) measured a modal force to predict the sound pressure level and determined the r.m.s. lift coefficient from the modal force, length of cylinder, and correlation length measurements. The total force is the equivalent of \( \langle B^2 \rangle (\lambda_c (\ell - \gamma) / \lambda_F^2) \) in Equation 3.6.

Good agreement between predicted and measured levels can only be achieved if the force and correlation length measurements are made on the same cylinder in the same facility. This is because of the sensitivity of the vortex shedding process to end conditions and freestream turbulence.
3.7 Effect of Freestream Turbulence

The force transducer was then used to investigate the effect of freestream turbulence on the vortex shedding process for a nearly rigid cylinder. The freestream turbulence was altered by using a grid and a screen across the jet exit plane. A hot wire annemometer was used to determine the properties of the turbulence.

The grid was a wire mesh 6.4 millimeter square with a wire size of 1 millimeter in diameter. The screen was a fine mesh aluminum screen. The properties of the grid generated turbulence were investigated using a DISA 55A22 hot wire probe, a DISA 55D05 constant temperature anemometer, and a DISA 55D15 linearizer. The decay of the grid generated turbulence with distance downstream from the exit plane is shown in Figure 3.34. The turbulence could be varied from 2.5% to 0.7% by varying the position of the probe behind the turbulence producing grid. The turbulent intensity in the freestream was 0.3% where there was no grid present. The spectrum levels of the grid generated turbulence are shown in Figure 3.35. The velocity spectrum levels were used to estimate the longitudinal length scales using Taylor's hypothesis. The results are summarized in Table 3.6.
The force transducers were placed in the test stand and the test stand was positioned at a fixed distance downstream of the jet exit plane. The turbulence generating grids were placed across the exit and the experiments were conducted for both grids. The lift forces on the sensing element were measured. The measured data are summarized in Table 3.7. The variation of lift coefficient with turbulent intensity is shown in Figure 3.36.

From Figure 3.36 it can be seen that turbulent intensity plays a role in the vortex shedding process. Increasing the turbulent intensity increases the unsteady forces on a cylinder undergoing vortex shedding.
CHAPTER IV. WATER TUNNEL EXPERIMENTS

4.0 Introduction

The role of large amplitude cylinder motion in the vortex shedding process was investigated using the MIT Marine Hydrodynamics Laboratory's closed circuit water tunnel. The water tunnel was used because it made it possible to achieve controlled large amplitude two dimensional cylinder motions at Reynolds numbers above 10,000. The forces required to move the test cylinder a significant fraction of the cylinder diameter were much lower in water as compared to air at the same Reynolds number and cylinder diameter. This was because the flow velocity had to be \( \frac{v_a}{v_w} \) larger in air than in water. The corresponding shedding frequency is also \( \frac{v_a}{v_w} \) larger in air than in water. The force required to move the test cylinder a fixed fraction of the cylinder diameter was the mass of the yoke and test cylinder times the acceleration. The acceleration was the desired displacement times the circular frequency squared. The force levels required to move the cylinder to significant amplitudes were realizable in water but not in air. This chapter deals with the experiments conducted in water.
The water tunnel is described in section 4.1. The force transducer was modified to work in an underwater environment and the necessary modifications are described in section 4.2. The yoke and shaker system are also described in section 4.2. The force transducer was used to determine the forces on a stationary cylinder and the results are compared to those in air in section 4.3. The test cylinder was driven with a shaker and the data for forced oscillation are reported in section 4.4. In section 4.5, the results of a spring mounted cylinder's response to vortex shedding are reported. Section 4.6 deals with an analysis of the vortex excited oscillations of a circular cylinder constrained to move transverse to the flow.

4.1 Water Tunnel Facility

The experiments were conducted in the MIT Marine Hydrodynamics Laboratory's variable pressure water tunnel. A schematic of the water tunnel is shown in Figure 4.1. It is of closed circuit construction with a test section 50.8 cm by 50.8 cm, 137 cm long. The contraction ratio of the water tunnel is 4.5 to 1. There is a honeycomb of one inch diameter acrylic tubes upstream of the contraction to reduce the freestream turbulence levels of the flow. The flow is driven by a variable speed impeller over a range from 0.1 m/sec to 10.0 m/sec. A detailed description of the facility can be found in Lewis (36). The facility has since
been modified by Kerwin (37).

The mean freestream flow conditions were measured using a laser doppler velocimeter. A hot film probe was used to determine the freestream turbulence levels. The mean velocity profile was uniform across the test section. The freestream turbulence level was measured using a cylindrical hot film probe, TSI Model 1212. The probe was connected to a TSI Model 1750 anemometer and linearized by an Analog Devices 433B Programmable Multifunction Module. The mean velocity was measured on a digital voltmeter and the root mean square was measured on a B+K true rms meter. The freestream turbulence level was 0.9% for the flow speeds used. The spectrum levels of the u component of the freestream turbulence are shown in Figure 4.2. The spectrum levels were slowly varying with frequency and did not show the impeller blade passing frequency and its harmonics. The longitudinal length scale of the turbulence was estimated from the u component spectrum levels using Taylor's hypothesis. The longitudinal length scale was determined to be about 1 cm.

4.2 Force Transducer And Yoke

The transducer had to be modified to work properly in the marine environment. Both the physical layout of the probe and the electronics were changed. When the test
cylinders were oscillated, the bending of the phenolic base of the probe became important and biased the data at the driving frequency. A pair of steel slats were added to the probes to increase their stiffness. The steel slats increased the fixed capacitance to ground of the variable capacitors, which reduced the probe sensitivity. Consequently, the beam elements of the probes were made thinner to increase the probe's sensitivity and were made of brass to facilitate construction. The thickness of the beam elements was reduced from .062 inches to 0.032 inches. This reduced the stiffness of the beam element by a factor of eight.

The transducer electronics were modified because of the long leads needed to carry the signal from the probe in the water tunnel to the demodulator on the outside. The output impedance of the probe was high, making it susceptible to noise. A buffer amplifier was added to the probe to reduce the output impedance and improve the signal to noise ratio. The buffer amplifier was built on a printed circuit board and placed inside the test cylinder near the probe.

The probe seal was changed because rubber latex is permeable to water. The probe was sealed with Dow Corning 3140 RTV. This was selected because it had a high bond strength and did not form a corrosive byproduct when curing. The water tunnel has a solution of 0.2% sodium nitrite in
the water as a soft steel preservative, which contributed to a loss of integrity for a normal RTV seal.

The demodulating circuit was modified to permit mean drag measurements. These modifications were only used in the unsteady drag measurements; the original electronics were used for the unsteady lift measurements. The new capabilities included a zero suppression stage and increased sensitivity that allowed direct use of a digital voltmeter.

The electronics for the drag measurements are shown schematically in Figure 4.3. The demodulator was changed to an active half-wave rectifier (Jung (38)). The half-wave rectifier was not a phase sensitive detector, therefore the voltage divider was biased so that a voltage null was never encountered in the experiments. The output voltage was nulled using a dc voltage and a potentiometer in the low pass filter stage following the rectifier. These changes eliminated two isolation transformers. The overall circuit was simpler and was usable to dc. The electronics required a long warm up time to settle to a stable zero and there was a slow zero drift.

The apparatus used to oscillate the cylinder in the flow was designed by Schargel (39). His apparatus was modified to include endplates on the struts. The endplates were ten cylinder diameters in diameter.
The test cylinders were supported in a yoke, shown in Figure 4.4. The yoke consisted of a pair of struts of rectangular cross section 1/4" by 1 1/2" and 25" long. The struts were supported in a steel frame. The frame consisted of two 1/4" aluminum triangular pieces (one per side). Two threaded rods held the bottom of the frame and two aluminum angle irons were used as the upper braces to provide lateral stiffness to the frame. Mounted inside the frame were four 1/2" ball bushings (two per side). The ball bushings were aligned so that the yoke could only oscillate in the vertical direction. The shafts were 1/2" diameter stainless steel rods. Two 1/2" diameter steel tubes connected the top and bottom of the steel rods. The aluminum struts were connected to the steel tubes.

The struts extended vertically downward through the plexiglass window of the test section. The window was sealed with 3/8" thick neoprene compression seals. The struts were machined to an airfoil shape to minimize their drag and were bent so that a higher aspect ratio cylinder could be used. The aspect ratio was 25.6 for these tests.

The yoke was attached to a B+K Model 4801 shaker using a B+K Model 4818 mode study shaker head. The power amplifier used to drive the shaker was a B+K Model 2707. The shaker system had a 2.54 cm peak to peak displacement limit and a force limit of 381 Newtons. The shaker was used
to oscillate the cylinder. The total mass of the yoke plus test cylinder was 2.4 kg. At low frequencies the displacement limit determined the maximum displacement. At high frequencies the available force determined the maximum displacement.

A method for calibrating the force transducer in situ was developed to check the in air calibration technique. A model for the probe is shown in Figure 4.5. The probe is sensitive to the relative displacement of the beam element to the capacitive leaves fixed in the supporting cylinder, \( x_2 - x_1 \) in Figure 4.7. A mechanical model of the probe is also shown in Figure 4.7. The equation of motion for the mass due to an imposed acceleration at \( x_1 \) is

\[
M_e \ddot{x}_2 + k(x_2 - x_1) = 0 \tag{4.1}
\]

where \( k \) is the stiffness of the beam element and probe seal, and \( M_e \) is the effective mass of the probe, due to the ring element and beam element. By adding and subtracting \( M_e \ddot{x}_1 \) to Equation 4.1 and neglecting terms of \( M_e (\ddot{x}_2 - \ddot{x}_1) \) compared to \( k(x_2 - x_1) \) the result is:

\[
(x_2 - x_1) = \frac{-M_e \ddot{x}_1}{k} \tag{4.2}
\]

The term \( M_e (\ddot{x}_2 - \ddot{x}_1) \) is small because the probe is assumed to be operating in the stiffness controlled region well below its natural frequency. The probe output can be interpreted
as responding to the inertial force $M_e \ddot{x}_1$. The probe was oscillated in air and the effective mass, $M_e$, was determined. The water tunnel was filled with water and the experiment was repeated. The effective mass plus the added mass, $M_1$, was determined from this experiment. Then a mass, $M_1$, was hung on the sensing element of the transducer and the water tunnel was filled with water. The probe was oscillated and its output, $E_0$, was determined for a fixed acceleration input, $\ddot{x}_1$. The result was:

$$E_{01} = -C(M_1 + M_1A + M_e + M_A)\ddot{x}_1$$ \hspace{1cm} 4.3

where $C$ is the probe sensitivity, output voltage per unit force input, and $M_1A$ was the added mass of the suspended weight. The experiment was repeated for a different mass, $M_2$, of a geometry similar to mass $M_1$, resulting in:

$$E_{02} = -C(M_2 + M_2A + M_e + M_A)\ddot{x}_1$$ \hspace{1cm} 4.4

Equations 4.3 and 4.4 are a set of algebraic equations with two unknowns; $C$, the probe sensitivity and $M_1A$, the added mass of the weights. The added mass of the weights was assumed to be the same because the geometry of the weights was the same. The equations were solved for the probe's sensitivity. The in situ calibration technique verified the in air calibration. The in air calibrations were used for the remainder of the tests.
4.3 Stationary Cylinder Results

The force transducer was used in the water tunnel to determine the rms sectional lift coefficient, mean drag coefficient, rms sectional drag coefficient, and the shedding frequency for a stationary cylinder undergoing vortex shedding. The test cylinder was 1.59 cm in diameter with a sensing element 0.4 diameters long. The test cylinder was supported in struts and the struts were blocked to prevent motion. There were endplates on the struts to provide repeatable boundary conditions.

The transducer was aligned to measure the unsteady lift forces. The flow was initiated and stabilized. The flow velocity was determined from the impellor RPM, which had been previously calibrated against the pressure drop across the contraction. The data were recorded on a Tandberg FM tapedeck and analyzed in real time using a Hewlett-Packard 5423A Structural Dynamics Analyzer. The shedding frequency and the rms sectional lift coefficient were determined from the spectrum levels of the lift force. The results are summarized in Table 4.1 and are plotted in Figure 4.6. Also shown in Figure 4.6 are the in air results for a similar turbulent intensity. The in water results agree with the in air results.
The spectrum levels of the lift force are shown for a Reynolds number of 19,300 in Figure 4.7. The spectrum levels are characteristic of the shedding process, showing the dominant peak at the shedding frequency. The lift coefficient and Strouhal number were slowly varying at Reynolds numbers above 12,000. This suggests that a nondimensionalization based on dynamic pressure, \( q = \frac{1}{2} \rho U_\infty^2 \), the flow velocity, \( U_\infty \), and the cylinder diameter, \( d \), may collapse the lift force spectrum levels. The spectrum levels were nondimensionalized using \( (q^2 d^2 l_F^2 d/U_\infty) \) and the frequency was nondimensionalized using \( (d/U_\infty) \). The results are shown in Figure 4.8. The water tunnel results are compared to the in air results in Figure 4.9 and the agreement is fair. The collapse is shown on a logarithmic scale and small variations are obscured by this, but the general trend for both cases studied is the same.

The transducer was used to determine the mean drag coefficient for the Reynolds number range tested. The electronics for the drag measurement were used. The probe was calibrated statically using a set of weights and a digital voltmeter and dynamically using the magnet and coil calibrator. The static calibration is shown in Figure 4.10. The probe demonstrated hysteresis because part of the static load was carried by the RTV seal, and the seal would deform with time. This resulted in the mechanical zero of the probe drifting with time. The drift was 20% per hour for a
0.10 newton load. In spite of the zero drift problem an attempt was made to use the probe statically. The observation times were kept as short as possible and the probe output was re-zeroed after each test.

The transducer was used to make estimates of the mean drag coefficients for a cylinder undergoing vortex shedding. The transducer was oriented by hanging a weight on the sensing element and rotating the probe until the output was nulled. The variation of the mean drag coefficient with Reynolds number is shown in Figure 4.11 and summarized in Table 4.2. The results tend to be high for this Reynolds number range, 7000 to 50,000.

The transducer was used to determine the rms sectional drag coefficient for this Reynolds number range. The spectrum level for the drag force is shown in Figure 4.12. It shows a broad peak at twice the shedding frequency. The leakage from the shedding frequency did not appear in the drag spectrum, so the orientation scheme seemed to be adequate. The drag spectrum also shows increasing levels with decreasing frequency at the low frequencies. The drop off at low frequency resulted from high-pass filtering of the signal. The reason for the increasing levels at low frequencies is not known. The rms sectional drag coefficients were computed from the spectrum levels of the drag force. The rms sectional drag coefficients are plotted
in Figure 4.13 and listed in Table 4.3. The drag results can be nondimensionalized similar to the lift force spectrum levels. The rms sectional drag coefficients are compared to other investigators results in Figure 4.14 and Table 4.4. There is fair agreement on the unsteady drag results. There is little information in the literature about the length scale associated with the drag forces.

4.4 Forced Oscillation Results

The cylinder spanned the water tunnel test section with the transducer on the centerline. It was supported in the previously described yoke. The yoke was oscillated harmonically by a B+K shaker. An accelerometer on the yoke sensed the motion of the yoke and transducer. The result was a simultaneous measurement of the local unsteady force and the cylinder motion.

The experiments were performed by first fixing the flow velocity and then setting the desired frequency on the oscillator. The cylinder motion was changed from at rest to the desired amplitude of motion. The flow was allowed to stabilize and then the data were taken. The data were recorded and analyzed using the Hewlett-Packard Structural Dynamics Analyzer.
The determination of the added mass of an oscillating cylinder in still water was used as a consistency test for the force transducer. The model for the probe in Figure 4.7 is useful in interpreting the results. When the probe was oscillated in air the effective mass of the probe was determined from the probe output and cylinder acceleration. Then the cylinder was oscillated in water at different amplitudes and the added mass was determined. The still water added mass was measured to be 1.34 gm compared to a theoretical value of 1.26 gm. The variation of the added mass with the ratio of amplitude to diameter is shown in Figure 4.15 for a frequency of 9.7 Hz. The error compared to the inviscid solution was ten percent. Another test on the still water behaviour of the test cylinder and yoke system was performed by forcing the cylinder with band-limited white noise. The acceleration was sensed on the yoke outside the water tunnel test section. The transducer's output was due to its inertia. The magnitude was the mass times the acceleration. In air the mass was the effective mass of the transducer. In water the mass was the effective mass plus the added mass. A transfer function was computed between the transducer output and the acceleration sensed on the yoke. The result is shown in Figure 4.16 for both the in air and in water tests. The magnitude of the transfer function was flat above 8 Hz and the two signals were 180 degrees out of phase over this
region. Below this region the coherence was poor and the transfer function is not valid. The transfer function shows that there are no structural modes in the frequency range of interest.

The transducer was used to investigate the forces on the test cylinder while it underwent forced oscillation transverse to the flow. When the cylinder was stationary, it responded to the forces on it due to vortex shedding. These forces occurred in a narrow band around the Strouhal frequency, \( f_S \). The cylinder was oscillated in the presence of a mean flow at the forcing frequency, \( f_f \) (different from the vortex shedding frequency, \( f_S \)), at a fixed amplitude. The response was observed at the forcing frequency, \( f_f \), and at the shedding frequency, \( f_S \). The excitation was a pure tone at the forcing frequency and the acceleration was nearly pure tone. The acceleration and the force were analyzed in the frequency domain. The magnitude of the force and acceleration were monitored at the forcing frequency. The transfer function between the acceleration and the force was used to determine the relative phase of the two at the forcing frequency. The magnitude of the force was also monitored at the shedding frequency. The force at the driving frequency can be broken down into two components; the force in phase with the acceleration and the force in phase with the velocity. The force in phase with the acceleration included the response of the
transducer to its effective mass. The force in phase with the velocity was the work producing force.

A phenomenon that has been observed before is that the shedding frequency would change from the Strouhal frequency to the forcing frequency under certain conditions. This phenomenon was termed lock-in (Bishop and Hassan (13)). The lock-in effect has been observed for in-line oscillations as well as transverse oscillations (Crandall et al (2)). Here we deal with only the transverse oscillations. The lock-in effect was investigated for discrete points in the nondimensional displacement, nondimensional frequency space. The amplitude displacement, $\delta$ (1/2 peak to peak) was nondimensionalized on the cylinder diameter, $\delta_n = \delta/d$. The frequency was nondimensionalized using the Strouhal frequency, $f^* = f_s/f_f$. This nondimensionalization was selected to facilitate the use of the forced data in interpreting naturally oscillating cylinder phenomena. The lock-in effect was investigated in the region $0.0 < \delta_n < 0.5$ and $0.6 < f^* < 1.4$. This corresponds to an $fd/U_\infty$ range of $0.10 < fd/U_\infty < 0.30$. This region was determined by the shaker amplitude limit and peak power. The data shows which frequencies and amplitudes were required for lock-in to occur. Both the locked-in and non locked-in behaviours were investigated.
The region of $f^* - \delta_n$ space where lock-in occurred is shown in Figure 4.17 for a Reynolds number of 19,300. Outside this region non locked-in behaviour was observed. The trend in the data was that the closer the forcing frequency was to coincidence with the Strouhal frequency the smaller was the amplitude of motion required for lock-in to occur. In Figure 4.19 the lock-in region is divided into two subregions. In Region I the work producing force is in phase with the cylinder motion. In Region II the work producing force was out of phase with the cylinder motion. Since the force is that of the fluid on the cylinder, work is done by the fluid on the cylinder in Region I whereas work is done by the cylinder on the fluid in Region II. A naturally oscillating cylinder (one that could be modelled as a mass, spring, and dashpot) can only be locked-in in Region I. In Region II the fluid force would supplement the structural damping and the cylinder could not stay at that amplitude. This type of behaviour has been observed before by Tanida et al (15) and by Sarapkaya (14).

Lock-in boundaries have been determined by other investigators. Koopman (40) and Mercier (41) used flow visualization techniques to determine when the flow was locked-in. Koopman's results were for Reynolds numbers around 100-300. Mercier's were for a Reynolds number range of 4000-8000. Figure 4.18 compares the present results to Koopman (40), Mercier (41), and Honji and Tanida (42). Both
the locked-in and non locked-in regions were investigated for a Reynolds number of 19,300.

**Non Locked-In Behaviour**

In the non locked-in region the response of the force transducer showed two peaks: a broad peak associated with the vortex shedding and a peak associated with the cylinder motion. Figure 4.19 is an example of the non locked-in response for \( f^* = 1.36 \) and \( \delta_n = 0.2 \). The peak at the forcing frequency includes the inertial force of the transducer ring element. At the forcing frequency for amplitudes up to \( \delta_n = 0.4 \) the fluid was predominantly mass-like. The motion of the cylinder at these frequencies and amplitudes did not completely disrupt the shedding of vortices near the shedding frequency. There was still the same general character to the force spectrum as when the cylinder was stationary at the same Reynolds number, except for the peak at the forcing frequency due to the cylinder motion.
Locked-In Behaviour

In the locked-in region the shedding frequency changed from the Strouhal frequency to the forcing frequency and only one peak occurred. An example of locked-in behaviour is shown in Figure 4.21 for $f^* = 1.0$ and $\delta_n = 0.20$. The spectrum levels of a stationary cylinder and the locked-in case can be compared in Figure 4.21. The cylinder motion has organized the flow, as can be seen from the narrowness of the peak for the locked-in case. The vortex shedding has become almost pure tone for the locked-in case. An example of lock-in and non lock-in at the same forcing frequency is shown in Figure 4.21, for $f^* = 0.91$ and $\delta_n = 0.05$ and $\delta = 0.32$. This shows that at a fixed frequency the flow can be locked-in or non locked-in, depending on the amplitude of motion. The non locked-in case shows two peaks and the locked-in case has only one.

4.5 Spring Mounted Cylinder Tests

Tests were undertaken to determine the response of a cylinder to vortex shedding excitation. The arrangement utilized the existing frame and yoke. Springs were mounted on the yoke outside of the water tunnel as in Figure 4.22. The springs were located such that there was a set holding the cylinder to the water tunnel and another set tied to a chain hoist that was used to pretension all springs. The
pretensioning was necessary for the springs to operate in their linear domain. The pretensioning had to be large enough so that the springs were in tension throughout the tests. The system was designed to have a resonant frequency as close to 15 Hz as possible so that the Reynolds number range tested would correspond to the Reynolds number of the forced oscillation tests.

The system characteristics were determined experimentally. The moving mass was measured to be 2.4 Kg. The natural frequency was measured by plucking the cylinder and measuring its response. The damping was determined from the same tests. The stiffness was determined from the mass and natural frequency. The tests were done in air and then the water tunnel was filled and the tests were repeated. The system parameters are summarized in Table 4.5.

An accelerometer was placed on the yoke outside of the flow and used to determine the acceleration response of the structure. The flow was initiated and stabilized. The acceleration response spectrum levels were determined using the Hewlett-Packard Analyzer for each flow condition. The path the system followed in the displacement frequency space is plotted in Figure 4.23. Also shown in Figure 4.23 are the lock-in boundaries determined from the forced motion experiments. The corresponding frequency v.s. nondimensional frequency plot is shown in Figure 4.24.
The ordinate in Figure 4.26 is the nondimensional frequency. It is defined to be the shedding frequency for a stationary cylinder at that flow velocity \( f_s = 0.2U_\infty /d \) divided by the resonant frequency of the spring mounted cylinder in water. Defined this way, the nondimensional frequency increases with increasing flow velocity. In Figure 4.23 the dots represent the path followed when the flow velocity was increasing and the x's represent the path followed when the flow velocity was decreased. Similarly, Figure 4.27 shows the variation of the cylinder response frequency with the nondimensional frequency. The circles show the path followed with increasing flow velocity and the x's show the path followed with decreasing flow velocity.

The spring mounted cylinder showed a definite hysteresis loop with the amplitude and the spectrum levels of the acceleration response depending on whether the point was approached from a higher or lower velocity. This type of double amplitude response has been observed by Feng (43). The maximum amplitude of the cylinder response did not occur at coincidence of the shedding frequency and the cylinder resonance. The cylinder did not lock-in with increasing velocity until coincidence. The maximum amplitude with increasing velocity occurred just before the cylinder detuned from a locked-in condition and returned to a non-locked-in response.
The hysteretic behaviour was investigated by both stopping the cylinder and releasing it from rest and by displacing the cylinder and releasing it. When the cylinder was released from rest it rose to the lower branch of the hysteresis loop when the flow conditions corresponded to the hysteresis region. Outside this region the cylinder returned to its initial amplitude. This is the same behaviour observed by Feng (43). When the spring mounted cylinder was displaced and released the observed behaviour depended on the nondimensional frequency. When the nondimensional frequency was outside the hysteresis region, the cylinder response would settle to the same conditions as observed without an external disturbance. When the nondimensional frequency was within the hysteresis region, the observed response depended on how close the nondimensional frequency was to unity and on the amplitude of the disturbance. At a fixed nondimensional frequency, the cylinder would not lock-in if the disturbance was small. If the disturbance was large enough the cylinder would lock-in and respond at a large amplitude. The size of the disturbance necessary to cause lock-in was smaller the closer the nondimensional frequency was to unity. At the largest nondimensional frequency that lock-in occurred with increasing velocity, it was not possible to give the test cylinder a large enough initial amplitude to achieve the locked-in condition. The result of this section is that a
naturally oscillating cylinder can be locked-in or not locked-in depending on the path the cylinder takes in a nondimensional frequency, nondimensional amplitude space.

4.6 Analysis of Data

An attempt to utilize the information gained from the forced oscillation experiments has been made. The description is only valid for cylinders constrained to move in a direction transverse to the flow. A description of the response of a naturally oscillating cylinder to vortex shedding has been formulated using the lock-in boundaries in a displacement frequency space. The results from the experiments at one Reynolds number have been utilized to predict transition from non locked-in to locked-in for a range of neighboring Reynolds numbers. The shedding frequency for a stationary cylinder at a particular flow velocity was nondimensionalized by the still water resonant frequency of the naturally oscillating cylinder, $f^* = \frac{f_s}{f_r}$. The stationary cylinder shedding frequency, $f_s'$, is the flow velocity times the Strouhal number, 0.19, divided by the cylinder diameter. The displacement was nondimensionalized on the cylinder diameter and the lock-in boundaries of Figure 4.17 were utilized. The resonant frequency replaces the forcing frequency in the nondimensionalization. In the resonant case the shedding frequency changes from the Strouhal frequency to the natural
frequency, whereas in the forced case it changes from the Strouhal frequency to the forced frequency. In both cases it is the motion of the cylinder that causes the change in character of the flow. Only the part of the forced lock-in region where the work producing force was in phase with the motion was utilized because the naturally oscillating cylinder cannot do work on the flow.

**Non Locked-in Response Prediction**

The results of the stationary cylinder experiments and the non locked-in case were utilized to predict the non locked-in response of resonant cylinders. The result utilized from the non locked-in data is that the cylinder motion does not drastically alter the force for small displacements. Therefore the stationary cylinder results can be used to predict the non locked-in response for small amplitude response.

The stationary cylinder result showed that the forces on the cylinder were similar over a range of Reynolds numbers. A two-pole fit to the nondimensional stationary cylinder lift data was used to predict the non locked-in oscillating cylinder response. The curve fit is shown in Figure 4.25 and described by
\[ \frac{S_F(f) U_\infty}{q^2 d^2} \cdot \frac{2Fd}{d} = \frac{1}{(5 - \frac{fd}{U_\infty} 138)^2 + (0.9 \frac{fd}{U_\infty})^2} \quad 4.5 \]

It was necessary to have a similar description of the structure for this model to be useful. The structure was also described by a two-pole model. The equation for the squared magnitude of the transfer function from displacement due to a force is

\[ |H(f)|^2 = \frac{1}{[k - M(2\pi f)]^2 + (c 2\pi f)^2} \quad 4.6 \]

where the parameters for the structure are its mass (including the added mass of the fluid), stiffness, and damping. The stiffness can be determined from the resonant frequency and the moving mass. The two pole model for the structure describes a transfer function, the displacement output due to the force input. The nondimensional force input is dimensionalized using a value of velocity. Then the squared magnitude of the transfer function is multiplied by the force spectrum. The nondimensional force spectrum is for a local force coefficient and so it necessary to integrate it along the cylinder length using the stationary cylinder correlation lengths. This involves the assumption that the motion does not correlate the flow for the
conditons of interest, i.e. non locked-in low amplitude flows. The result of correcting for the cylinder length is

\[ \hat{S}_\delta(f) = \frac{|H(f)|^2 q^2 \frac{d^2}{\nu_c^2} (i-\gamma) \frac{d}{U_\infty}}{[(5 - \frac{fd}{U_\infty} 138)^2 + (0.9 \frac{fd}{U_\infty})]} \]

The result is the displacement spectrum for the cylinder at that flow velocity. This estimate neglects the damping due to the fluid for low amplitude non locked-in flows. The displacement spectrum is then integrated to determine the rms displacement. The rms displacement and nondimensional frequency are checked against the lock-in boundaries to see if the response is locked-in or not. If the displacement and frequency do not correspond to a locked-in condition then the estimate is assumed to be valid.

**Lock-in Response Prediction**

When the lock-in boundary in displacement, frequency space is encountered the previous model is no longer valid. The vortex shedding frequency changes from the Strouhal frequency to the resonant frequency. The constraint for determining the response amplitude in the locked-in region is that the power flow from the fluid to the structure must be dissipated by the structural damping mechanism. The total force on the cylinder must be offset by the structural
The structural damping can be described in terms of a quality factor, $Q$. The damping force is

$$F_d = \frac{\sqrt{kM}}{Q} \frac{2\pi f_r \delta_n d}{\sqrt{2}}$$

where $f_r$ is the resonant frequency of the system. Substituting for the stiffness $k = (2\pi f_r)^2 M$ results in

$$F_d = \frac{M(2\pi f_r)^2 \delta_n d}{\sqrt{2} Q}$$

This is the damping force that must be offset by the lift force for the cylinder to remain in a steady state. The required total lift coefficient for the cylinder to respond at nondimensional amplitude, $\delta_n$ is

$$C_{LR} = \frac{F_d}{1/2 \rho U_\infty^2 d} = \frac{1}{Q} \frac{M 2\pi f_r^2 \delta_n d}{\sqrt{2} 1/2 \rho d \frac{f^*}{f_r} U_c^2}$$

where $f^* = f_s/f_r$ and $U_c$ is the flow velocity for which $f_s = f_r$ or $U_c = f_r d/0.19$. The local lift force coefficient required, $C_{lR}$ is

$$C_{lR} = \frac{1}{Q} \frac{M (2\pi f_r)^2 \delta_n d}{\sqrt{2} 1/2 \rho d \left[(\ell-\gamma) l_c\right]^{1/2} f^* U_c^2}$$
To determine the allowed amplitudes of response it is necessary to know both the variation of local force with $\delta_n$ and the variation of the correlation length with $\delta_n$. The variation of correlation length with $\delta_n$ has been measured by Toebes (16) for a $\delta_n$ up to 0.125 at a Reynolds number of 68,000. Koopman (40) used a pair of hot wires in the wake to determine the lock-in boundaries. Lock-in was defined to be when the cylinder motion organized the wake.

Example Response Prediction

Consider the spring mounted cylinder shown in Figure 4.22 and used in the spring mounted cylinder tests in section 2.4. The system properties are listed in Table 4.5. The combined mass of cylinder and yoke was 2.5 kg including the added mass of the cylinder. The quality factor of the structure is assumed to be 20 and includes the effects of the yoke in the water. The length of the cylinder was 40 cm and the diameter was 1.6 cm. The Reynolds number range corresponding to $f^*$ from 0.7 to 1.5 is $11,000 < Re < 25,000$. The correlation length for a stationary cylinder is about 7 diameters.

The non locked-in response can be estimated using Equation 4.7 and is
This result is a spectrum level with units of meters squared per Hertz. The mean square response is determined by integrating the spectrum levels. The result is plotted in a frequency, displacement space as in Figure 4.26. When the lock-in boundaries are encountered the estimate is no longer valid.

Inside the lock-in boundaries a different algorithm is used to determine the response. The force required for a cylinder to be at an amplitude, \( \delta_n \), is equal to the damping force of the cylinder. \( F = c \dot{x} \). The required lift coefficient is

\[
C_{LR} = 2.8 \frac{\delta_n}{f^*}^2
\]

This is the total lift coefficient. The prediction scheme can be simplified by assuming that at lock-in the flow is fully correlated over the cylinder. This would mean the sectional lift coefficient and total lift coefficient are the same. This is not a bad approximation for a cylinder of low aspect ratio. The variation of the lift coefficient with amplitude to diameter for an \( f^* = 1.0 \) is shown in Figure 4.27. Also shown is the variation of the required
lift coefficient with $\delta_n$. The response amplitude is where the two curves intersect. The path followed by the spring mounted cylinder in the locked-in region was estimated by assuming that within the lock-in bounds the lift coefficient is only a function of $\delta_n$ and does not depend on $f^*$. This assumption allows the use of Figure 4.28 to estimate the path followed. The result is plotted as line segment 2 in Figure 4.26.

The response predicted for the spring mounted cylinder case is shown in Figure 4.26. The response shows a region of $f^*$'s where two different amplitudes of response are possible. One response corresponds to a locked-in condition and the other corresponds to a non locked-in condition. Whether the system will be locked-in or not depends on the path followed in displacement, frequency space. If the velocity started below coincidence the cylinder will be in a locked-in state and will remain locked-in with increasing velocity until the lock-in boundary is encountered. However if the velocity started above $f^* = 1.4$ the flow will be in a non locked-in state. As the flow velocity is decreased the system will remain in the non locked-in state until the lock-in boundary is encountered. The reason for this is that even though a locked-in state exists for the velocity it requires an external stimulus to transit from non locked-in to locked-in. This behaviour was observed in the spring mounted cylinder tests.
Chapter V. RESULTS, CONCLUSIONS, AND RECOMMENDATIONS FOR FURTHER WORK

5.1 Results and Conclusions

1. A force transducer has been developed to measure the local unsteady force on an oscillating cylinder in a cross flow. An accelerometer was used to compensate for the acceleration sensitivity of the transducer. A magnet and coil calibrator was used to calibrate the transducer. The result was a simultaneous measurement of local force and cylinder motion.

2. The velocity in the potential flow near a cylinder is well correlated with the local unsteady force on the cylinder.

3. A program of two-point velocity correlations was carried out. The correlation length varied from 7d to 6d over the Reynolds number range from 20,000 to 50,000 and tended to decrease with increasing Reynolds number.

4. The root mean square sectional lift coefficients were determined for this Reynolds number range and varied from 0.4 to 0.5. The lift coefficients tended to increase with increasing Reynolds number.
5. An areoacoustic measurement program was carried out. The radiated sound pressure levels were measured for a cylinder in cross flow. The measured levels were compared to levels predicted by the theory of vortex noise developed by Phillips (3), Keefe (4), and Leehey and Hanson (5). There was good agreement between the predicted and measured levels.

6. The sectional lift coefficient was determined to increase with increasing turbulent intensities over the range from 0.3% to 2.0%.

7. The use of endplates was determined to have a large effect on the unsteady flow around a cylinder in a cross flow. The radiated sound pressure levels were from 7 dB to 11 dB higher for a cylinder with endplates. This was due to the increased lift coefficients in the case with endplates as compared to the case without and not due to changes in the correlation length. The Strouhal number was lower for the case of a cylinder with endplates.

8. The force transducer was used to investigate the effect of cylinder motion on the vortex shedding process at amplitude to diameter ratios up to 0.5 in the reduced frequency range $0.1 < \frac{fd}{U_o} < 0.3$. Lock-in boundaries were determined for this region. The lock-in boundaries were
plotted in a nondimensional amplitude, nondimensional frequency space.

9. A cylinder was spring mounted in the water tunnel. The cylinder was constrained to move transverse to the flow. Its vibratory response to vortex shedding was observed. The response showed a region where the cylinder could be locked-in or not locked-in at the same flow velocity.

10. The response of the spring mounted cylinder was estimated using the forced vibration data. The lock-in boundaries in a displacement, frequency space were used to describe the state of cylinder motion: locked-in or not locked-in. The non locked-in cylinder response was estimated using the stationary cylinder results. The locked-in response was determined from the criterion that the power in from the fluid had to be dissipated by the structural damping. The hysteretic behaviour that was observed was due to the fact that the lock-in boundaries were amplitude dependent as well as frequency dependant.

5.2 Recommendations for Further Study

1. The spring mounted cylinder provides a platform to determine the effectiveness of the various vortex shedding
suppression schemes. The methods developed here provide a means of gaining an understanding of how the suppression schemes work.

2. The case of cylinder motion in line with the flow could also be investigated using the techniques developed here. Both the forced vibration case and the spring-mounted case could be studied.

3. There is a lack of correlation information for the vortex shedding process in water. Two transducers could be built into the same probe and direct force correlation studies could be undertaken. The same probe could be used for both lift and drag measurements. The probe could also be used to investigate the effect of cylinder motion on the two point force correlations.

4. The current yoke design could be modified to include two degrees of freedom. This could be done to investigate the effect of modal overlap. The possibilities include two modes transverse to the flow, two modes in-line with the flow, or one mode transverse to the flow and one mode in-line with the flow. This behaviour would better simulate the behaviour observed for long thin cables in the ocean.
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## Table 3.2b Two Point Velocity Correlation Data

(1.9cm dia. cylinder)

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<tr>
<td>25</td>
<td>0.7%</td>
<td>1.5%</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>0.6%</td>
<td>1.1%</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>35</td>
<td>0.7%</td>
<td>1.3%</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

TABLE 3.6 SUMMARY OF GRID GENERATED TURBULENCE CHARACTERISTICS
<table>
<thead>
<tr>
<th>Re</th>
<th>$C_L$</th>
<th>Re</th>
<th>$C_L$</th>
<th>Re</th>
<th>$C_L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>5200</td>
<td>0.17</td>
<td>4200</td>
<td>0.24</td>
<td>4200</td>
<td>0.36</td>
</tr>
<tr>
<td>6400</td>
<td>0.18</td>
<td>4400</td>
<td>0.26</td>
<td>5500</td>
<td>0.40</td>
</tr>
<tr>
<td>7400</td>
<td>0.20</td>
<td>4800</td>
<td>0.22</td>
<td>5700</td>
<td>0.49</td>
</tr>
<tr>
<td>7500</td>
<td>0.18</td>
<td>5000</td>
<td>0.26</td>
<td>6600</td>
<td>0.42</td>
</tr>
<tr>
<td>8600</td>
<td>0.22</td>
<td>5400</td>
<td>0.31</td>
<td>7100</td>
<td>0.47</td>
</tr>
<tr>
<td>9700</td>
<td>0.25</td>
<td>6000</td>
<td>0.37</td>
<td>7800</td>
<td>0.38</td>
</tr>
<tr>
<td>10,200</td>
<td>0.18</td>
<td>7100</td>
<td>0.37</td>
<td>8800</td>
<td>0.47</td>
</tr>
<tr>
<td>10,600</td>
<td>0.28</td>
<td>7300</td>
<td>0.44</td>
<td>9400</td>
<td>0.45</td>
</tr>
<tr>
<td>11,200</td>
<td>0.26</td>
<td>8200</td>
<td>0.41</td>
<td>9500</td>
<td>0.49</td>
</tr>
<tr>
<td>11,900</td>
<td>0.35</td>
<td>9000</td>
<td>0.46</td>
<td>9900</td>
<td>0.45</td>
</tr>
<tr>
<td>12,800</td>
<td>0.36</td>
<td>9400</td>
<td>0.37</td>
<td>10,300</td>
<td>0.59</td>
</tr>
<tr>
<td>13,800</td>
<td>0.40</td>
<td>10,000</td>
<td>0.43</td>
<td>10,000</td>
<td>0.60</td>
</tr>
<tr>
<td>14,300</td>
<td>0.37</td>
<td>12,000</td>
<td>0.56</td>
<td>12,200</td>
<td>0.61</td>
</tr>
<tr>
<td>15,500</td>
<td>0.42</td>
<td>13,400</td>
<td>0.54</td>
<td>13,800</td>
<td>0.62</td>
</tr>
<tr>
<td>15,900</td>
<td>0.41</td>
<td>13,800</td>
<td>0.56</td>
<td>15,900</td>
<td>0.63</td>
</tr>
<tr>
<td>16,700</td>
<td>0.43</td>
<td>15,900</td>
<td>0.55</td>
<td>21,200</td>
<td>0.62</td>
</tr>
<tr>
<td>18,500</td>
<td>0.39</td>
<td>16,000</td>
<td>0.56</td>
<td>27,500</td>
<td>0.58</td>
</tr>
<tr>
<td>21,200</td>
<td>0.45</td>
<td>21,200</td>
<td>0.52</td>
<td>31,800</td>
<td>0.58</td>
</tr>
<tr>
<td>21,200</td>
<td>0.48</td>
<td>21,200</td>
<td>0.56</td>
<td>31,800</td>
<td>0.58</td>
</tr>
<tr>
<td>26,000</td>
<td>0.48</td>
<td>23,800</td>
<td>0.57</td>
<td>31,800</td>
<td>0.58</td>
</tr>
<tr>
<td>31,800</td>
<td>0.46</td>
<td>32,000</td>
<td>0.53</td>
<td>31,800</td>
<td>0.58</td>
</tr>
<tr>
<td>31,500</td>
<td>0.47</td>
<td>32,000</td>
<td>0.56</td>
<td>31,800</td>
<td>0.58</td>
</tr>
<tr>
<td>37,000</td>
<td>0.46</td>
<td>37,000</td>
<td>0.51</td>
<td>42,400</td>
<td>0.48</td>
</tr>
<tr>
<td>37,000</td>
<td>0.46</td>
<td>42,000</td>
<td>0.53</td>
<td>42,000</td>
<td>0.53</td>
</tr>
</tbody>
</table>

Note: $C_L$ represents the RMS sectional lift coefficients for $\theta = (0.3\%, 1.2\%, 2.0\%)$. The table is structured with different Re values for each lift coefficient.
<table>
<thead>
<tr>
<th>Re</th>
<th>ST</th>
<th>( C_l )</th>
</tr>
</thead>
<tbody>
<tr>
<td>2670</td>
<td>0.18</td>
<td>0.07</td>
</tr>
<tr>
<td>5080</td>
<td>0.17</td>
<td>0.12</td>
</tr>
<tr>
<td>7940</td>
<td>0.17</td>
<td>0.26</td>
</tr>
<tr>
<td>8520</td>
<td>0.17</td>
<td>0.29</td>
</tr>
<tr>
<td>11,340</td>
<td>0.19</td>
<td>0.53</td>
</tr>
<tr>
<td>15,800</td>
<td>0.19</td>
<td>0.61</td>
</tr>
<tr>
<td>19,300</td>
<td>0.19</td>
<td>0.61</td>
</tr>
<tr>
<td>20,500</td>
<td>0.19</td>
<td>0.57</td>
</tr>
<tr>
<td>28,800</td>
<td>0.19</td>
<td>0.59</td>
</tr>
<tr>
<td>38,800</td>
<td>0.19</td>
<td>0.60</td>
</tr>
</tbody>
</table>

**TABLE 4.1** MEASURED RMS SECTIONAL LIFT COEFFICIENTS AND STROUHAL NUMBERS, TI = 0.9%, WATER
<table>
<thead>
<tr>
<th>RE</th>
<th>CD</th>
</tr>
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<tbody>
<tr>
<td>6000</td>
<td>1.30</td>
</tr>
<tr>
<td>9400</td>
<td>1.30</td>
</tr>
<tr>
<td>12,200</td>
<td>1.32</td>
</tr>
<tr>
<td>15,300</td>
<td>1.36</td>
</tr>
<tr>
<td>19,300</td>
<td>1.34</td>
</tr>
<tr>
<td>23,900</td>
<td>1.39</td>
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<tr>
<td>27,100</td>
<td>1.40</td>
</tr>
<tr>
<td>31,700</td>
<td>1.38</td>
</tr>
<tr>
<td>38,600</td>
<td>1.40</td>
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</table>
TABLE 4.3  MEASURED RMS SECTIONAL DRAG COEFFICIENTS, 
TI = 0.9%, WATER

<table>
<thead>
<tr>
<th>Re</th>
<th>$C_d'$</th>
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</thead>
<tbody>
<tr>
<td>6000</td>
<td>0.013</td>
</tr>
<tr>
<td>9400</td>
<td>0.064</td>
</tr>
<tr>
<td>12,200</td>
<td>0.075</td>
</tr>
<tr>
<td>15,300</td>
<td>0.115</td>
</tr>
<tr>
<td>19.300</td>
<td>0.066</td>
</tr>
<tr>
<td>23.900</td>
<td>0.066</td>
</tr>
<tr>
<td>27.100</td>
<td>0.069</td>
</tr>
<tr>
<td>31,700</td>
<td>0.074</td>
</tr>
<tr>
<td>38.600</td>
<td>0.071</td>
</tr>
<tr>
<td>Investigator</td>
<td>Method</td>
</tr>
<tr>
<td>---------------------</td>
<td>-------------------</td>
</tr>
<tr>
<td>Current</td>
<td>Local Force</td>
</tr>
<tr>
<td>Humphreys (32)</td>
<td>Total Force</td>
</tr>
<tr>
<td>McGregor (34)</td>
<td>Integrated Pressure Distributions</td>
</tr>
<tr>
<td>Keefe (4)</td>
<td>Local Force</td>
</tr>
<tr>
<td>Gerrard (8)</td>
<td>Integrated Pressure Distributions</td>
</tr>
<tr>
<td>Bishop and Hassan (24)</td>
<td>Total Force</td>
</tr>
</tbody>
</table>
TABLE 4.5 SUMMARY OF SPRING MOUNTED CYLINDER CHARACTERISTICS

<table>
<thead>
<tr>
<th>ITEM</th>
<th>WEIGHT</th>
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</thead>
<tbody>
<tr>
<td>upper clamps (2)</td>
<td>77gm</td>
</tr>
<tr>
<td>lower clamps (2)</td>
<td>83gm</td>
</tr>
<tr>
<td>upper cross piece</td>
<td>305gm</td>
</tr>
<tr>
<td>lower cross piece</td>
<td>185gm</td>
</tr>
<tr>
<td>struts (2)</td>
<td>427gm</td>
</tr>
<tr>
<td>steel runners (2)</td>
<td>198gm</td>
</tr>
<tr>
<td>TOTAL</td>
<td>2400gm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>CONDITION</th>
<th>RESONANT FREQUENCY</th>
<th>QUALITY FACTOR</th>
</tr>
</thead>
<tbody>
<tr>
<td>dry</td>
<td>13.8 Hz</td>
<td>25</td>
</tr>
<tr>
<td>wet (still water)</td>
<td>13.4 Hz</td>
<td>14.4</td>
</tr>
</tbody>
</table>
Figure 2.1 Force Transducer Assemblage
Figure 2.2 Transducer Electronics
Figure 2.3a Transducer Model

Figure 2.3b Simple Electrical Model of Probe
Figure 2.4 Schematic of Magnet and Coil Calibrator
DISPLACEMENT FROM COIL CORE, $\delta$/mm

Figure 2.5 Variation of Electromotive Force with Position of Magnet with Respect to Driving Coil
Figure 2.6 Sample Transducer Calibration Using 1/8" Magnet and Coil Calibrator
Figure 2.7 Coherence Between Force Input and Transducer Output, Broadband Calibration of Probe
$p(a, \theta, z, t) \quad r = a$

\[ F(z, t) = \int_{0}^{2\pi} p(a, \theta, z, t) \cos \theta \, d\theta \]

Figure 2.3 Circular Cylinder in Cross Flow, Sign Conventions
Contraction
Settling Chamber Section
\[ t = \frac{1}{x} \]

Test Section in Reverberant/Anechoic Chamber (Blockhouse)
Muffler & Diffuser
Centrifugal Blower

General Specifications:
- Contraction Ratio: 20:1
- Test Section: 38cm x 38cm, shown in open duct configuration

Elevation Scale:

WIND TUNNEL FACILITY - ROOM 5-024
ACoustics & Vibrations LABORATORY
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

FIGURE 3.1
Figure 3.2 Free Jet Velocity Profile.
Figure 3.3 Spectrum Level Fluctuating Velocity, $U_\infty = 20$ m/sec.
Figure 3.4  Typical Acoustic Background Spectrum, $U_\infty = 40$ m/sec.
Figure 3.5 Wind Tunnel Test Set Up for Aeroacoustic Tests.
Figure 3.6a  Oil Film Flow Visualization, Cylinder Without Endplates

$Re = 21,000$

$V = 20 \text{ m/s}$
Figure 3.6b Oil Film Flow Visualization, Cylinder Without Endplates, Closeup of Mixing Region
Figure 3.7a  Oil Drop Flow Visualization, Cylinder Without Endplates
Figure 3.7b  Oil Drop Flow Visualization, Cylinder Without Endplates, Closeup
Figure 3.8a Oil Film Flow Visualization, Cylinder With Endplates
Figure 3.8b  Oil Film Flow Visualization, Cylinder With Endplates, Closeup of Cylinder
Re = 21,000
U = 20 m/sec
end plate, -6
3 Aug 80, #13

Figure 3.9a Oil Film Flow Visualization, Endplate
Figure 3.10a Oil Drop Flow Visualization, Cylinder With Endplates
Figure 3.10b  Oil Drop Flow Visualization, Cylinder With Endplates, Closeup
Figure 3.11a Oil Drop Flow Visualization of Endplate (a)
Figure 3.11b  Oil Drop Flow Visualization of Endplate (b)
Figure 3.12 Axial Variation of Base Pressure Coefficient, $Re = 42,000$, $\bigtriangleup$ with endplates, $\bigtriangleup$ without endplates.
Figure 3.13 Variation of Base Pressure on the Centerline with Endplate Spacing, \( 0Re = 42,000, \Delta Re = 31,500 \).
Figure 3.14 Variation of RMS Sectional Lift Coefficients with Reynolds Number,

- $O$ with endplates
- $\Delta$ without endplates
Figure 3.15  Spectrum Levels Fluctuating Force, Re = 37,000
--- with endplates
--- without endplates
Figure 3.16 Spectrum Levels Radiated Sound, Re = 37,000
--- with endplates
--- without endplates
Figure 3.17 Coherence Between Local Force and Radiated Sound, $Re = 37,000$
--- with endplates
--- without endplates
Figure 3.18 Set Up for Force-Velocity Measurements
Figure 3.19  Spectrum Levels Fluctuating Velocity and Fluctuating Force,  
$Re = 42,000$  
--- Velocity  
--- Force
Figure 3.20  Coherence Between Fluctuating Force and Fluctuating Velocity, Re = 42,000.
Figure 3.21 Two Point Velocity Correlation Set Up
Figure 3.22  Spectrum Level Fluctuating Velocity, Re = 37,000.

- Probe A
- --- Probe B
Figure 3.23 Coherence Between Hot Wire Probes, Re = 37,000
-- Separation 1d
--- Separation 10d
Figure 3.24 Spatial Correlation Function
Re = 37,000, \( l_c = 6.8d \), \( \gamma = 1.8d \)
Figure 3.25 Variation of Correlation Length with Reynolds Number
Figure 3.26 Comparison of Current Correlation Length Results with Previous Investigators
Figure 3.27  Force-Sound Measurement Set Up
Figure 3.28 Spectrum Level Fluctuating Force for a Nearly Rigid Cylinder, $Re = 42,000$
Figure 3.29  RMS Sectional Lift Coefficient for a Nearly Rigid Cylinder in Cross Flow
Figure 3.30 Comparison of Current Lift Coefficient Results with Previous Investigators
Figure 3.31 Spectrum Levels Fluctuating Force and Radiated Sound, Re = 42,000

--- Sound

--- Force
Figure 3.32  Coherence Between Local Force and Radiated Sound, Re = 42,000
Figure 3.33 Comparison of Measured and Predicted Radiated Sound Pressure Levels
Figure 3.34 Variation of Freestream Turbulent Intensity with Distance Downstream of Grid
+ Grid
0 Screen
Figure 3.35 Spectrum Level Unsteady Velocity, Grid Generated Turbulence
Figure 3.36 Variation of RMS Sectional Lift Coefficient with Turbulent Intensity
- 0.3% TI
∧ 1.2% TI
+ 2.2% TI
Figure 4.1 Schematic of Water Tunnel Facility
Figure 4.2  Spectrum Level Fluctuating Velocity, Water Tunnel, $U_\infty = 3$ m/sec
Figure 4.3 Schematic of Demodulating Circuit for Drag Measurements
Figure 4.4 Schematic of Yoke Used to Support Test Cylinder in Water Tunnel
Figure 4.5 Model of Probe for In situ Calibration

\[ m_e \ddot{x}_2 + k (x_2 - x_1) = 0 \]
Figure 4.6 RMS Sectional Lift Coefficient for a Stationary Cylinder

- △ Water $\text{TI} = 0.9\%$
- ● Air $\text{TI} = 1.2\%$
Figure 4.7 Spectrum Level Lift Force, Stationary Cylinder, Re = 19,300, Water TI = 0.9%
Figure 4.8 Nondimensional Force Spectrum, In Water Results

- - - - - - Re = 38,800
- - - - - - Re = 20,300
--- - - - Re = 15,800
Figure 4.9 Comparison of Nondimensional Force Spectra

--- Re = 42,000 In Air Results

--- Re = 39,000 In Water
Figure 4.10 Static Calibration of Probe
Figure 4.11 Mean Drag Coefficient for a Stationary Cylinder, Water, TI = 0.9%
Figure 4.12 Spectrum Levels Drag Force and Lift Force, Re = 19,300

--- Drag

--- Lift
Figure 4.13 RMS Sectional Drag Coefficient for a Stationary Cylinder, Water, $T_1 = 0.9\%$
Figure 4.14 Comparison of Current RMS Sectional Drag Coefficient Measurements with Previous Investigators
Figure 4.15  Dead Water Added Mass Measured with Force Transducer
Figure 4.16 Transfer Function, Force to Acceleration, Dead Water, --- In Air, — In Water
Figure 4.17 Lock-in Boundaries in $f^* - \delta_n$ Space, Re = 19,300
Figure 4.10 Comparison of Current Lock-in Boundaries in \( f^* - \delta \) Space with Previous Investigators, — Current, \( \Delta \) Mercier, o Honji, and Taneda, \( \Delta \) Kooman
Figure 4.19  Spectrum Level Lift Force, Forced Oscillation,  
Re = 19,300, \( f^* = 1.36, \delta_n = 0.21 \)
Figure 4.20 Spectrum Level Lift Force, Forced Oscillation, Re = 19,300
--- $f^* = 1.0$, $\delta_n = 0$
--- $f^* = 1.0$, $\delta_n = 0.21$
Figure 4.21  Spectrum Level Lift Force, Forced Oscillation, Re = 19,300

- $f^* = 0.91, \delta_n = 0.05$
- $f^* = 0.91, \delta_n = 0.32$
Figure 4.22 Schematic of Spring Mounted Cylinder
Figure 4.23  Response Diagram, Spring Mounted Cylinder
- Increasing Velocity
x Decreasing Velocity
Figure 4.24 Variation of Cylinder Response Frequency with Nondimensional Frequency for Spring Mounted Cylinder in Water
- Increasing Velocity
- Decreasing Velocity

\[ f^* = \frac{f_s}{f_r} \]
Figure 4.25 Curve Fit to Nondimensional Force Spectrum,

--- Curve Fit

--- Re = 20,300 in Water
Figure 4.26 Estimated Cylinder Response Diagram, Spring Mounted Cylinder in Water
Figure 4.27 Variation of RMS Sectional Lift Coefficient with $\delta_n$, $\text{Re} = 19,300$
REFERENCES


The author was born on February 20, 1954 and was raised in Dubuque, Iowa. He majored in Aerospace Engineering at Iowa State University, Ames, Iowa and received his B.S. Degree in June 1976. In September 1977, he received his Masters Degree in Ocean Engineering at MIT. He was employed as a research assistant for Professor Patrick Leehey working on measuring the low wavenumber levels of a turbulent boundary layer. Then the author went to The von Karman Institute for Fluid Dynamics, Brussels, Belgium and received a diploma with honors, June, 1978. In September he reentered MIT to work on his PhD in Ocean Engineering. During this period he was again employed as a research assistant for Professor Patrick Leehey.