The Tangential Drag of an Axially Oscillating Cylinder

by

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Abstract

An experimental investigation of the tangential drag of an axially oscillating cylinder is presented. Deep-sea mooring cables are oscillated sinusoidally at ocean wave amplitudes and frequencies, and experimental values of the tangential (friction) drag coefficient are obtained. The experimental values of the drag coefficient are compared with values predicted by the exact solution to the Navier-Stokes equations for a smooth cylinder in laminar flow. The experimental data reports drag coefficient values that are consistently higher than those predicted by the laminar theory. An attempt is made to interpret the discrepancies in terms of the effects of roughness and/or turbulence.

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NOTATION

A or A_i	(i^{th}) oscillation amplitude
a	cylinder radius
C_1, C_2	constants
C_T	tangential (friction) drag coefficient
D	pulley diameter
d	cylinder diameter
F_h	force amplitude measured at high water level
F_l	force amplitude measured at low water level
F_m	force amplitude measured at middle water level
F_T	tangential drag force
$f(a\alpha)$	$N_1(a\alpha)/N_0(a\alpha)$
h	characteristic height of cylinder surface roughness
i	$\sqrt{-1}$, "imaginary number"
k^2	$i\alpha^2$, separation constant
l	test sample length
l_{h-l}	total test sample length between high and low water levels
l_{h-m}	total test sample length between high and middle water levels
l_{m-l}	total test sample length between middle and low water levels
$N_0^{-1}(x)$	inverse function of $N_0(x)$
n_a	average needle bearing frequency
n_i	instantaneous needle bearing frequency
n_t	turbulence frequency

q	$\frac{h}{a}$, roughness ratio
RE	"the real part of"
Re	Reynolds number
Re_A	$\frac{A^2\omega}{\nu} = A^2\alpha^2$, Reynolds number using A
Re_a	$\frac{a^2\omega}{\nu} = a^2\alpha^2$, Reynolds number using a
(r, θ, z)	circular cylindrical coordinates
S/N	peak F_T/peak F_1, signal to noise ratio
s	area of cylinder surface roughness per unit length
T	oscillation period
T_0	tension in test sample
t	time
V_0	$A\omega$, peak cylinder velocity
(V_r, V_θ, V_z)	velocities in circular cylindrical coordinates
α^2	$\frac{\omega}{\nu}$
δ	boundary layer thickness
μ	viscosity (of water)
ν	kinematic viscosity (of water)
ρ	density (of water)
τ_{rz}	sheer stress on cylinder (due to axial flow)
ω	oscillation radian frequency

I. INTRODUCTION

A. Problem Statement and Motivation for Study

An experimental study of the tangential drag of an axially oscillating cylinder is presented. The oscillating cylinders are deep-sea mooring cables forced in the axial direction at amplitudes and frequencies normally encountered in ocean moorings. No attempt is made to study the fine structure of boundary layers or to perform extensive mathematical analysis of the flow conditions. Comparisons are made, however, between the forces predicted by a simple solution to the Navier-Stokes equations for the laminar case and the measured, experimental forces.

The motivation for studying the tangential drag of an axially oscillating cylinder comes from two sources: first, from an interest in deep-sea, surface mooring dynamics and, secondly, from the lack of adequate analytic or experimental study of the specific problem. My interest in the problem comes primarily from the first motivator, the field of mooring dynamics. It has been found in recent years that (oceanographic) surface moorings have a reliability of 54% compared to a reliability of 90% for subsurface moorings (Walden and Panicker, 1973) and that current velocities recorded by current meters placed at the same depths on surface and subsurface moorings indicate current energies that are consistently greater for surface than subsurface moorings (Gould and Sambuco, 1975). Hong (1972), in a mooring dynamics study, finds that a reliable estimate of the tangential drag coefficient is essential for determining the dynamic forces in surface moorings, with the frequency response function of the mooring being heavily dependent on the choice of value of the tangential drag coefficient. Goeller and Laura (1971) also find the choice of tangential drag coefficient to be important as it has a significant effect on reducing dynamic forces in the vicinity of resonance. From these pieces of evidence, it appears that surface mooring dynamics, due to the sea surface boundary condition of the structure, are important to structural design and reliability and to oceanographic data quality.

B. Previous work

From studies, such as the four cited above, related to energy transfer from the sea surface to the mooring line, it has become apparent that several parameters in the analysis are not well-known. Tangential "external" damping coefficients are not the only ambiguous quantities, but "internal" material damping parameters of synthetic line (Hong, 1972; Goeller and Laura, 1971) are also unknown but important to the problem. A study that compares actual mooring motions with computer-predicted motions (Panicker et al, 1974) seems to show that, indeed, tangential drag on the mooring system has the tendency to reduce peaks at the natural frequency of the system in the case of the acceleration response. But, at best, it is possible to assign a tangential drag coefficient that is the value determined for cables or cylinders in uniform, axial flow. A tangential drag coefficient that is (oscillation) frequency and amplitude dependent does not exist to date (Yamamoto et al, 1974; Hong, 1972) and computer programs for mooring dynamics (Goodman et al, 1972; Kaplan et al., 1972; Sargent et al, 1972) presently use a tangential drag coefficient that is frequency and amplitude independent.

In regards to theoretical background, a related problem that has been solved for many years is that of the flow near an oscillating flat plate; Stokes' second problem (Schlichting, 1968). An exact solution of the Navier-Stokes equations can be performed for this case. For the axially oscillating cylinder, the analytical expression for the flowfield near the cylinder has recently been presented following Stokes (Stokes, 1901; Casarella and Laura, 1969; Hong, 1972). The Navier-Stokes equations, expressed in circular cylindrical coordinates, are solved for the laminar flow case. Using the laminar flow results, Yamamoto (Yamamoto et al, 1974) estimates the skin friction coefficient for axially oscillating cables. Assuming that curvature effects of cables are small, Yamamoto proceeds to derive an expression for the skin friction coefficient for the cable in turbulent flow conditions, based on the boundary layer of a flat plate. This is, to date, the extent of the theoretical work done for the flow near an axially oscillating cylinder. The laminar flowfield has been amply treated but the same is not true of the turbulent flowfield, probably the more likely condition for deep-sea moorings due to cable roughness factors and the fact that (large) normal flows and cable strumming are also typical of the flow conditions.

C. Method of Study

As stated above, the author does not intend to perform a mathematical study of the problem, but, rather, an experimental study. It was felt that a good experimental study of the problem would yield information that was much needed by and directly useful to those in the mooring dynamics field.

Further, the cylinders that are studied here are real mooring cables ranging from jacketed wire ropes to various sizes and types of synthetic lines. An experimental study of an ideally smooth cylinder would be of more value to the theoreticians but of less value to those designing mooring systems. It was felt that the jacketed wire rope came as close to being an ideally smooth cylinder as would any real-life engineering material.

The amplitudes and frequencies of oscillation chosen here also attempt to approximate the field conditions for deep-sea moorings. Sinusoidal oscillations ranging from 2.8 seconds to 18.5 seconds in period and .1 meter to .5 meters in (single) amplitude were used. Even larger oscillation amplitudes would have been used had it been practical. A Scotch yoke driving mechanism produced the sinusoidal oscillations, and data was taken using a strain gage ring dynamometer to measure force, a potentiometer to measure displacement, and strip chart and FM tape recorders for recording the data. The strip chart records were then analyzed, producing one set of results. The FM tape was played into another chart recorder, and that strip chart record (of much better quality than the first) was then analyzed. If a spectral analysis of the data or a digitized data record is ever desired, the FM tapes can be used in conjunction with the computer to obtain this information.

Following a presentation and analysis of the data, comparisons are made between the experimental results and theoretical predictions using the Casarella and Laura (1969) predictions for the comparison.

II. MATHEMATICAL ANALYSIS

As stated above, it is not the author's intention to perform an extensive mathematical study of the tangential drag of an axially oscillating cylinder. But one mathematical description of the flowfield will be presented here for later comparison with the experimental data. The assumptions inherent in the description and suggestions for future work will then be discussed. Finally, a dimensional analysis of the problem will be performed for later use with the experimental data.

A. Linear analysis - Exact solution to the Navier-Stokes equations

Casarella and Laura (1969) have presented an analytical expression for the viscous drag on a smooth rod undergoing axial and torsional oscillations. The following method of analysis comes originally from Stokes' work (Stokes, 1901) and only more recently from their work as applied to an infinite cylinder of radius (a) that is undergoing pure axial oscillations. Expressing the Navier-Stokes and continuity equations in circular cylindrical coordinates (r, θ, z) leads to the four equations given in Appendix 1. The following equations result after assuming that the flowfield is independent 1) of z because the cylinder has infinite length and 2) of θ due to symmetry, that V_r and its derivatives with respect to r are zero because of the boundary conditions at the cylinder ($V_r = 0$ at $r = a$) and the continuity equation, and that V_θ and its derivatives with respect to r are zero, also by symmetry:

$$(1) \qquad r^2 \frac{\partial^2 V_z}{\partial r^2} + r \frac{\partial V_z}{\partial r} - \frac{1}{\nu} r^2 \frac{\partial V_z}{\partial t} = 0 \ .$$

By separating variables and letting $V_z = R(r)\,T(t)$ we find that

$$(2) \quad T(t) = e^{i\omega t} \;,$$

the separation constant k^2 has a value

$$(3) \quad k^2 = i\alpha^2$$

$$(4) \quad \alpha^2 = \frac{\omega}{\nu} \;,$$

and

$$(5) \quad r^2 \frac{d^2R}{dr^2} + r\frac{dR}{dr} - k^2r^2R = 0 \;,$$

since $\frac{\partial V_z}{\partial t} = i\omega V_z$. The general solution to equation (5) is in terms of Kelvin functions (Carslaw and Jaeger, 1959) so that

$$(6) \quad R(r) = C_1I_0(kr) + C_2K_0(kr)$$

where C_1 and C_2 are constants. The total solution becomes

$$(7) \quad V_z(r,t) = R(r)\,T(t) = e^{i\omega t}\,(C_1I_0(kr) + C_2K_0(kr))$$

The two boundary conditions are

$$(8) \quad V_z(a,t) = V_o\cos\omega t$$

$$(9) \quad V_z(\infty,t) = 0$$

Applying the boundary conditions we find that

$$(10) \quad V_z(r,t) = RE\left[V_oe^{i\omega t}\,\frac{K_o(kr)}{K_o(ka)}\right]$$

where RE signifies "the real part of". The tangential drag force per unit length becomes

$$(11) \quad \frac{F(a,t)}{l} = -2\pi a\,\tau_{rz}\Big|_{r=a} = -2\pi a\mu\,\frac{\partial V_z}{\partial r}\Big|_{r=a}$$

$$(12)\quad \frac{F(a,t)}{l} = RE\left[2\pi a\mu k\, V_0 e^{i\omega t} \frac{K_1(ka)}{K_0(ka)}\right]$$

Now, using the relationships (Abramowitz and Stegun, 1968)

$$(13)\quad \frac{d}{dr}(K_0(kr)) = -k\, K_1(kr)$$

$$(14)\quad \frac{d}{dr}(K_1(kr)) = -k\, K_0(kr) - \frac{1}{r} K_1(kr)$$

and

$$(15)\quad K_0(kr) = N_0(\alpha r)e^{i\phi_0(\alpha r)}$$

$$(16)\quad K_1(kr) = iN_1(\alpha r)e^{i\phi_1(\alpha r)}$$

equation (9) becomes

$$(17)\quad \frac{F(a,t)}{l} = 2\pi a\mu\alpha\, V_0 \frac{N_1(\alpha a)}{N_0(\alpha a)} \cos\left(\omega t + \phi_1\, \alpha r - \phi_0\, \alpha r - \frac{\pi}{4}\right)$$

If desired, the above equations can be simplified if $\alpha a >> 1$. Using asymptotic expansions (Abramowitz and Stegun, 1968) for $N_p(\alpha a)$ and $\phi_p(\alpha a)$ and neglecting terms of order 1/z or smaller, equation (17) becomes

$$(18)\quad \frac{F(a,t)}{l} = 2\pi a\mu\alpha\, V_0 \cos\left(\omega t - \frac{3\pi}{4}\right)$$

The results obtained above are in exact solution of the Navier-Stokes equations for laminar flow. No assumptions about the size of the oscillation amplitude or any other parameter have been made. But the question then arises as to whether the flow is always laminar. A stability analysis of the governing equations would help to indicate if and under what conditions turbulent flow might occur. Unfortunately, stability theory for the case at hand has not yet been developed (Orszag, 1975). The inside-out case, the stability of Poiseuille flow, has been studied to some extent (Grosch and Salwen, 1972), and the stability of the incompressible boundary layer of a flat plate in a mean flow undergoing small perturbations has also been the subject of considerable research (Obremski et al, 1969). So as a stability analysis of the flow near an axially oscillating cylinder does not exist and is beyond the scope of this paper, discussion on the possibility of a turbulent flowfield near the cylinder must resort to qualitative reasoning and specific reference to the experiments performed for this work.

The most obvious experimental cause for turbulence is the roughness of the oscillating cylinder. Unlike the ideal cylinder in the equations, real cylinders and especially real mooring line materials have surface roughness of varying degrees. One might expect that, turbulent flow aside, the tangential drag of an oscillating mooring line would be greater than that of a perfectly smooth cylinder. But at what point the rough surface becomes a turbulence stimulator is a very good question. Using another line of reasoning, a deep-sea surface mooring would sometimes have a turbulent flowfield in the axial direction because many moorings not only undergo axial oscillations but also are moored in a flow normal to the mooring line (i.e., currents). If the Reynolds number describing the normal flow $Re = \frac{VD}{\nu}$ becomes sufficiently high that turbulent flow occurs in the normal direction (with cable strumming also occurring at times), it is highly improbable that the flowfield in the tangential direction would remain laminar (or uniform).

If the flowfield is turbulent, the equations describing the flow become very complex and cannot be solved easily. Yamamoto (Yamamoto et al, 1974) avoided the difficult equations, asserting that since curvature effects of mooring cables are small for laminar flow, flow over a flat plate will give good estimates of the tangential (friction) drag coefficient on a cylinder (mooring line) for turbulent flow as well. It is doubtful, though, that the assertion about small curvature effects is always true for real mooring materials. For the statement to be true, one must presumably find that the ratio of boundary layer thickness to line radius is much less than one, or

(19) $$\frac{\delta}{a} \ll 1 \quad .$$

Using Yamamoto's definition of boundary layer thickness (also see Schlichting, 1968) i.e., the distance at which the fluid velocity becomes 1% of that of the cylinder, and the frequencies, line radii, and kinematic viscosities encountered in the experiments performed here, the following calculations result:

The boundary layer thickness δ is defined as

(20) $$\delta = \left[\frac{1}{\alpha} \; N_0^{-1}(.01 \; N_0(\alpha a))\right] - a$$

where

$$.0165 \text{ ft.} \le a \le .0415 \text{ ft.}$$

$$1.07 \times 10^{-5} \text{ ft.}^2/\text{s} \le \nu \le 1.31 \times 10^{-5} \text{ ft.}^2/\text{s}$$

$$.340 \text{ rad./s} \le \omega \le 2.244 \text{ rad./s}$$

So for

$$\alpha a_{min} = (.0165 \text{ ft.}) \left[\frac{.340 \text{ rad./s}}{1.31 \times 10^{-5} \text{ ft.}^2/\text{s}} \right]^{1/2}$$

$$= 2.65$$

$$= \alpha^1{}_{min}$$

$$\alpha a_{max} = (.0415 \text{ ft.}) \left[\frac{2.244 \text{ rad./s}}{1.07 \times 10^{-5} \text{ ft.}^2/\text{s}} \right]^{1/2}$$

$$= 19.00$$

$$= \alpha^1{}_{max}$$

From tables (Abramowitz and Stegun, 1968),

$$N_o(\alpha^1{}_{min}) = .1143$$

$$N_o(\alpha^1{}_{max}) = 4.187 \times 10^{-7}$$

$$.01\ N_o(\alpha^1{}_{min}) = 1.143 \times 10^{-3}$$

$$.01\ N_o(\alpha^1{}_{max}) = 4.187 \times 10^{-9}$$

$$N_o^{-1}(1.143 \times 10^{-3}) \approx 8.5$$

$$N_o^{-1}(4.187 \times 10^{-9}) \approx 32$$

$$\delta_{min} = \left(\frac{1}{\alpha^1_{min}}(8.5) - .0165\right) \text{ ft.} = 3.6 \times 10^{-2} \text{ ft.}$$

$$\delta_{max} = \left(\frac{1}{\alpha^1_{max}}(32) - .0415\right) \text{ ft.} = 2.838 \times 10^{-2} \text{ ft.}$$

$$\left(\frac{\delta}{a}\right)_{min} \approx 2.2$$

$$\left(\frac{\delta}{a}\right)_{max} \approx .7 \quad .$$

$\therefore \; \frac{\delta}{a} \ll 1$ is not satisfied.

So it seems safe to conclude that comparing the flowfield around the cylinder to that of a flat plate is risky at best. And until a stability analysis is performed for the equations governing the axially oscillating cylinder, it will remain uncertain as to whether a turbulent flowfield would in fact be formed.

In conclusion, the main shortcoming of the mathematical analysis that exists to date for the axially oscillating cylinder is the lack of mathematical understanding of turbulent flow. A stability analysis of the problem would at least let us know whether or not turbulence might occur. And, secondly, in a more experimental vein, a better understanding of the effects of cylinder roughness on the tangential drag coefficient would be very helpful.

B. Dimensional analysis

In order to better understand the problem of the axially oscillating cylinder, a dimensional analysis of the problem will be performed here. Such an analysis is useful in choosing suitable dimensionless parameters for comparing experimental data and analytically predicted quantities. These dimensionless quantities characterize the problem. In the following analysis, we will find that the problem is defined by six dimensional parameters less three fundamental units, leaving three dimensionless parameters to be defined.

The six dimensional quantities defining the problem are an amplitude A, the cylinder radius a, density ρ, viscosity μ, and force F. The three fundamental units (of which the six dimensional quantities are made) are mass, length, and time, leaving three dimensionless parameters to describe the flowfield. One dimensionless parameter has already been defined by the analytical solution to the problem, i.e., $a\alpha = a\sqrt{\frac{\omega}{\nu}}$, and if we define a tangential drag coefficient C_T as the second parameter, we find the third parameter in the following manner: C_T is defined as

$$(21) \quad C_T = \frac{(F_T/l)}{\rho\pi 2a(A\omega^2)}$$

and

$$(22) \quad f(a\alpha) = \frac{N_1(a\alpha)}{N_0(a\alpha)}$$

so

$$C_T = \frac{\pi 2a\mu A\omega^2}{\pi\rho a(A\omega)^2} \quad f(a\alpha)$$

$$(23) \quad C_T = \frac{2a}{A} \; \frac{1}{a\alpha} \; f(a\alpha)$$

One may also develop C_T so that

$$(24) \quad C_T = \frac{2}{A\alpha} \; f(a\alpha) \quad .$$

Now, from the two equations (23) and (24) above, the third parameter can be expressed in two different ways. The third parameter can be the ratio of the line diameter 2a to the oscillation amplitude A; 2a/A; or it can be the square root of a Reynolds number $A\alpha$ where $Re = A^2\omega/\nu = A^2\alpha^2$. The parameter derived from the mathematical analysis $a\alpha$ is also the square root of a

Reynolds number $Re = a^2\alpha^2$, and depending on which of the two so-called Reynolds numbers has more meaning in the context of data comparisons and analysis, either could be chosen as the abscissa for plots of C_T. If $a\alpha$ were chosen, then C_T $A/2a$ would be plotted versus $Re = a^2\alpha^2$. If $Re = A^2\alpha^2$ were chosen, then the ordinate would be C_T and $f(a\alpha)$ would be an implicit factor as C_T would not be a function only of $Re = A^2\alpha^2$. But in either case, the problem has been defined by three and only three dimensionless parameters.

In the context of experimental data, two additions to the dimensional analysis come to mind. The analysis outlined above pertains to an ideally smooth cylinder in laminar flow. Given experimental data and a suspicion that roughness or turbulence could be having an effect on the results, it would be desirable to isolate one effect from the other or to conclude that discrepancies between experimental data and theory were due solely to experimental error. It is impossible to perform a rigorous dimensional analysis incorporating roughness and turbulence effects because, unlike the previous dimensional analysis, there is no theoretical framework into which we can fit roughness and/or turbulence parameters. Hence, a qualitative dimensional analysis for roughness and turbulence will be presented and further disucssion will take place in Chapter IV in the context of data analysis.

Roughness may be characterized by either one or two quantities in addition to the original list of six dimensional quantities discussed above. At a minimum, some sort of height parameter h characterizing the surface roughness would be included, and a dimensionless parameter that might result would be the ratio of roughness height h to cylinder radius a, the roughness ratio $q = h/a$. For a more complete description of the roughness, a second dimensional quantity dealing with the amount of surface area per unit length s occupied by the roughness could be added to the list of seven dimensional quantities. This would create a fifth dimensionless ratio, probably a ratio of roughness surface area to total surface area of the cylinder or $s/2\pi a$. The exact mathematical relationship among the five dimensional quantities C_T, $A\alpha$, $a\alpha$, h/a, and $s/2\pi a$ cannot be determined without additional mathematical theories or, perhaps, experimental data.

Turbulence is more difficult to describe in a dimensional analysis than is roughness. Again, without theory or experimental data, an exact mathematical analysis is impossible. But one might think about the quantity that best characterizes a turbulent flowfield. Usually a Reynolds number is used to indicate whether or not a flowfield should be turbulent. But such a Reynolds number is used with hindsight, usually from experimental observations. In the cases of (tangential) flow near a flat plate or in a pipe, the friction drag coefficient increases when transition and turbulence occur, producing curves of C_T versus Reynolds number that also have shallower slopes than in the laminar flow case (Schlichting, 1968; Hoerner, 1965). Roughness produces still shallower curves and larger drag coefficients. Further discussion of turbulence and roughness will occur in conjunction with the experimental data.

III. THE EXPERIMENT

A description of the experiment follows and consists of three parts: the overall configuration of the system, a description of the measuring and data-recording components of the system, and the design considerations involved in choosing the system described here.

A. Configuration

The experimental system is designed to make a force difference measurement by measuring the force while oscillating a cylinder with no water around it and then again when it is immersed in (fresh) water. Figure 1 shows the experimental set-up. The main system components are a cylindrical, vertical axis tank in which are mounted four large pulleys, two at the top and two at the bottom of the tank, a continuous test sample of mooring line that goes around the pulleys, a Scotch yoke that forces the line to move sinusoidally, a strain gaged ring dynamometer placed in-line in the forcing rod between the Scotch yoke and the test sample, and various data recording devices. The vertical tank is about 12 feet high and 5 feet in diameter with one base on the ground next to the Blake Building at the Woods Hole Oceanographic Institution (see Figures 2 and 3). The tank is approachable at its top by means of a ramp leading to the building. The Scotch yoke is mounted on the ramp next to the tank top and a forcing rod extends out over the tank to its attachment point on the rope. The attachment point occurs at the junction of the two ends of the rope, which are held together in tension by means of turnbuckles. In-line with the forcing rod is an aluminum loop upon which are mounted four strain gages. The water level in the tank is first adjusted to barely cover the two pulleys mounted on the bottom of the tank. The first set of tests is then performed for the given rope sample, using the oscillation frequencies and amplitudes that have been chosen. The water level is then increased until it is just below the top two pulleys, and the oscillations are once again performed, giving the second set of tests for that sample. The strain gage dynamometer measures the force needed to oscillate the whole system of the pulleys and the rope, and the difference between the forces necessary to move the rope through the water and through the air is the tangential drag force on the rope.

Precise dimensions and specifications of the system are as follows: the tank height is 12.0 feet and its diameter is 5.3 feet. The pulleys are 11.6 inches in diameter and are made of PVC (polyvinylchloride) that is 1 inch thick. They run on 1/2 inch stainless steel shafts and sixteen 1/8 inch stainless steel needle bearings. Two collars hold the needles in their housings in each pulley (see Figure 4). The pulleys were designed to be as frictionless as possible.

Each test sample of line was cut to a precise length. The line was kept under about 50 pounds of tension to preclude transverse vibrations in the two, long, vertical sections of line. The tensioning was performed by the turnbuckles holding the rope ends together, and cutting exact, precalculated rope lengths was very important. As the synthetic lines tended to stretch, their cut lengths were shorter than those of the same diameter wire rope to allow room for stretching. The rope ends were terminated by potting them with epoxy in 1 inch lengths of thin-walled, 1 inch diameter tubing. It was then possible to drill a hole through each termination and insert a threaded rod for attaching the turnbuckles (see Figure 5). The separations between pulleys and change in water level are given in Figure 1. From these distances, we see that the actual change in water level, or one-half the length of line over which we measure hydrodynamic drag, is 20.75 feet/2 or 10.38 feet.

The Scotch yoke drives the system through a forcing rod that is about 5.5 feet long. This "rod" is a very light weight (.43 pounds) aluminum tube that is 1 inch in diameter. One end is attached to the test sample and the other to the strain gage loop which is about 4.5 inches in diameter. On a diameter, across from the forcing pipe, the loop is attached to the Scotch yoke itself and the forcing rod is thereby continuous. The displacement of the center of the yoke is measured by a rotary potentiometer that is turned by gears running along a chain. The potentiometer (displacement) and strain gage loop (force) outputs are recorded on a two-channel strip chart recorder and on two of four channels on an FM tape recorder. The third tape channel is used for a time code from a time code generator, and the fourth channel is used for a voice record.

B. Measurements

Four strain gages mounted on a loop are used to measure the force required to oscillate the rope (see Figure 6). The loop is aluminum 7075 T6 with a yield strength of 70,000 pounds per square inch. The important force measurement in this experiment is the <u>difference</u> in force measurements between the high and low water level tests for each rope sample. In that this force