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OF A CIRCULAR CYLINDER IN WATER

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ARGONNE NATIONAL LABORATORY, ARGONNE, ILLINOIS

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OF A CIRCULAR CYLINDER IN WATER

H. Halle¹ and W. P. Lawrence

Engineering and Technology Division
Argonne National Laboratory
Argonne, Illinois 60439

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ABSTRACT

A circular cylinder elastically mounted to have equal flexibility in all lateral directions was exposed to cross (transverse) flow of water. The amplitudes and frequency of flow-induced vibrations were measured as a function of flow velocity. Resonance-type responses were observed when the vortex-shedding frequency coincided with one half (Condition A), and unity (Condition B) of the natural frequency of the cylinder.

As expected, Condition B resulted in large-amplitude vibrations in the lift (transverse to flow) direction. However, water excitation at the lower flow velocity of Condition A, which is generally considered unimportant for airflow exposures, resulted in drag (parallel to flow) direction vibrations of moderately large amplitudes. It appears that design applications subject to waterflow under exposure Condition A warrant individual investigation.

¹Member ASME

1. INTRODUCTION

This paper describes the experimental investigation of the vibrational behavior of single cylinders exposed to crossflow of water. This work is part of a program to develop an understanding of the response of elastic cylinders, both singly and in bundles, to the excitation of transverse liquid flow.

This task was motivated by the occurrence of damaging flow-induced vibrations in heat exchangers, presenting a design problem for these and other components of nuclear reactor installations containing elastic structural elements exposed to fluid flow. This paper will consider only circular cylinders and only exposure to crossflow conditions. While flow parallel to the cylinder axis is known to contribute to the vibration excitation, the influence of crossflow, present under many actual conditions, is regarded to be of principal significance.

The results obtained from numerous single cylinder investigations performed to date cannot be used directly, without reservations, for the prediction of the response of tubes in heat exchangers, in which the configuration of the tube bank as well as the motion of the tubes affect the flow field. Because there is a need to increase the presently rather limited understanding of flow excitation in heat exchangers (1,2)¹, the investigations of this program were aimed toward that objective. Even though only single cylinder tests are reported, the test facility can accommodate closely spaced arrays of vibrating and instrumented cylinders simulating a section of a heat exchanger tube bank.

The experiments have two features, which, in combination, do not appear to have received much attention in the past, not even for single-

¹Underlined numbers in parentheses designate References at the end of the paper.

element tests. These features are (1) use of water rather than airflow excitation, and (2) use of circular cylinders permitted to vibrate laterally with equal flexibility in all directions. Waterflow excitation was selected because, in most cases, heat exchangers and reactor components of liquid metal fast breeder reactor installations will be exposed to liquid rather than gaseous fluid flow. Compared to air, the use of the much denser water not only increases the absolute magnitude of the excitation but, in addition, can change the relative influence of the various excitation mechanisms.

The test results indicated that the water flow imposed substantial vibration amplitudes on the test cylinder in both the drag and lift directions, parallel and transverse to the flow velocity, respectively. This justifies the effort of having provided the uniform lateral flexibility feature.

2. FLOW-EXCITATION MECHANISMS

The principal mechanisms exciting flow-induced vibrations of a single cylinder appear to be turbulence and vortex shedding. Turbulence may be considered to be comprised of localized velocity and pressure fluctuations occurring over a wide range of frequencies. When the resulting random excitation is imposed on an elastically mounted, lowly damped cylinder, the latter responds by extracting vibration energy only within a narrow frequency band enclosing a natural frequency. The most significant is the cylinder's fundamental natural frequency, f_n .

Vortex shedding of the von Karman type imposes alternating lift direction forces on a single cylinder. The vortices are shed at a frequency determined by

$$f_f = \frac{SU}{D}, \quad (1)$$

where S is the Strouhal number, U is the flow velocity, and D is the cylinder diameter.

For the range of Reynolds numbers between 300 and 200,000 (including many practical applications and also the experiments reported herein) S is approximately equal to 0.2. During each cycle, two vortices shed alternately from each side of the cylinder. The resulting alternating forces render the cylinder susceptible to lift direction vibration at the vortex shedding frequency, and to drag direction vibration at twice the vortex shedding frequency.

When the flow conditions are such that the vortices are shed at or near the fundamental natural frequency (f_n) of the cylinder, the potential lift direction vibration problems are generally considered most acute. With the reduced velocity $\underline{U} = U/f_n D$ ranging in the vicinity of 5, the

resulting "resonance" effects can cause large vibration amplitudes and can synchronize the vortex shedding and cylinder vibration frequencies.

The potential for drag-direction excitation exists already at $\underline{U} = 2.5$, at only one-half the velocity discussed above. While excitation at $\underline{U} = 2.5$ has generally not been observed in air flow, water experiments (3) (including those reported herein) have shown that significant vibration amplitudes can be excited in the drag direction.

The occurrence of large vibration amplitudes may change the effective diameter in Eq. 1 and thus change the vortex shedding frequency.

In order to compare the effects of waterflow and airflow excitation, the following should be noted:

(1) Within a wide range of Reynolds numbers, the flow velocity determining the vortex shedding frequency is practically independent of the flow medium.

(2) The density (ρ) of water is about 800 times greater than that of air.

(3) The magnitude of the excitation forces generated from turbulence and from vortex shedding are generally considered to contain a factor that is proportional to the dynamic head ($\rho U^2/2$) of the flow. Since under about the same conditions of flow and thus frequency-dependent vibration susceptibility, exposure to water exerts such tremendously larger forces than air that not only the absolute but also the relative influence of the excitation mechanisms can be expected to differ.

(4) The damping effects of water immersion constitute an additional factor influencing the structural response.

3. TEST EQUIPMENT

The experiments were conducted in a rectangular flow channel (5 in. wide by 10 in. high) of a test chamber inserted into a water loop capable of a 19 ft/sec channel velocity at 3200 gpm flow capacity. After a gradual transition from the 6-in.-diameter loop piping, the rectangular channel extends for more than 20 equivalent diameters in a transition section upstream of the test chamber to obtain fully established turbulent flow conditions. A downstream transition chamber effects the return to the circular loop piping.

The rectangular flow channel is formed by four internal walls which serve to separate the internal flow from the almost stagnant water contained in the surrounding cross section of the external 12-in.-diameter pressure shell. A section of this 12-in.-diameter pipe, designated the test chamber cross (Fig. 1), is fitted with two vertical 8-in.-diameter pipe branches which accommodate a test fixture assembly (Fig. 2) oriented transverse to water flow. The test fixture, in turn, serves to support one or several test elements to be exposed to transverse flow in the region of the rectangular flow channel as shown in Figs. 2 and 3. Access ports at the end of the branches permit installation and removal of the test fixture assembly without disassembly of the test chamber cross from the loop piping.

The test element has a rigid, stainless steel body, is 10 in. long and has a 0.667-in. outside diameter. This size is representative of tubes in heat exchangers (4). Coaxial tubular springs extending from both ends suspend the element to permit vibration with equal flexibility in all directions perpendicular to its axis. The external ends of these springs, in turn, are mounted between suspension mounts fixed to support plates on both sides of the flow channel. The springs thus provide a

clamped beam-type suspension system for the test element, whose axis is oriented vertically to eliminate gravity deformation. One of the suspension mounts incorporates a slide bearing to permit a small axial motion of the spring end upon deflection. Adjustment of the mounts and relocation of the support plates permits changes in the active length of the spring and consequently, the spring constant and the natural frequency of vibration of the test element. The suspension mounts are instrumented with strain gages whose output is calibrated to measure forces and deflections in both the drag (parallel to flow) and lift (transverse to both flow and cylinder axis) directions. The task of mounting and waterproofing the strain gages and leads requires a lengthy sequence of meticulously executed steps.

The element springs are made from 0.156-in.-OD, 0.100-in.-ID tubing of Type 4130 tool steel, chosen for strength, and are externally nickel plated for corrosion resistance. Tubular springs were selected to accommodate leads from a transducer that may be mounted in the test element body at some future time. Each spring is clamped with collet-type fixtures to both the element and the mounts. Each collet consists essentially of a split collar, externally threaded (nominal 1/16-in. NPT) and tightened with an appropriate nut. Except for the tube springs, all principal components are made from stainless steel, generally Type 304, alternating with Type 17-4 PH (Trademark Armco Steel Corporation) on some threaded components to reduce the chances of seizure.

In order to create and examine critical vibrations, the fundamental natural frequencies of the elements for the different tests were selected to be in the lower range of frequencies encountered in actual heat exchanger tubes. The latter are designed to have natural frequencies that are sufficiently high - hopefully - to avoid vibration problems in the first place.

The natural frequencies of a test element depend on its mass, the spring constant of its suspension, and, to a lesser extent, the degree of damping. The effects of water immersion increase the vibrated mass and the damping and, consequently, somewhat reduce the natural frequency from that determined in air. The fundamental natural frequency is characterized by a lateral translational displacement of the test element axis.

In addition to the fundamental natural frequency, the test element at times was also subjected to a rocking-type motion corresponding to a second mode-type deformation and associated higher frequency (f_r). As discussed in Section 5, this rocking-mode frequency was excited in the drag direction at lower than expected flow velocities.

Instrumentation provided for the performance of the experiments included a hot-film anemometer probe and an optical tracker. The probe was inserted into the wake region behind the cylindrical test element to monitor the vortex shedding. The tracker was used to indicate drag (flow) direction displacements of the central portion of an element through an observation port (see Fig. 1).

4. TEST PROCEDURE AND DATA PROCESSING

Three basic tests were conducted with elements suspended to have nominal fundamental natural frequencies of 16.5, 23.5, and 11.3 Hz, respectively, in the water environment. The external size of the elements used was identical during all three tests, a moderately increased element mass was used during the last two tests. Different lengths of the tube springs determined the natural frequency.

Before insertion into the water loop, the test element, mounted in the test fixture, was calibrated in air. Upon loading, in both drag and lift directions, the deflections and strain gage signals were linear for deflections up to 0.200 in. Upon unloading, hysteresis effects were observed. The natural frequency and damping in air was determined by excitation.

Each of the three tests was conducted in two phases. In the first phase, the flow velocity was slowly increased and the signal output of the transducers was monitored on oscilloscopes to determine velocities resulting in maximum or minimum vibration amplitudes or in other phenomena of interest, e.g., ranges of frequency shifts. At certain flow velocities, including those of special interest mentioned above, there were established "test conditions." At every test condition, the flow velocity was held constant to record the transducer signals on a magnetic tape recorder for about ten minutes, the time required for a subsequent frequency analysis of the recorded signals. The tests were discontinued at the highest flow velocity that could be reached without the element contacting its limiting stops in order to avoid alteration of the test set-up caused by excessive vibration amplitudes. Subsequently, the acquired data were processed and analyzed.

Initially, the second phase of each test consisted of repeating and checking a few of the first phase test conditions and then adding one or two intermediate conditions to obtain more complete data coverage. Following this, the flow condition with incipient contacting was again established. Subsequently, the flow velocity was quickly increased in an attempt to pass beyond the "resonant" condition and to again find and record test conditions where vibration amplitudes were not large enough to avoid contacting. Even when it was possible to establish one or a few such test conditions, further velocity increases resulted again in excessive vibration amplitudes and contacting.

The final test conditions were run in the resonance region, unless the contacting phenomenon became so violent that any further testing appeared to be damaging to the equipment and futile to evaluate.

The recorded time domain signals were processed by means of a fast Fourier transform frequency analyzer, which provided power spectral density curves (as well as the integrals thereof) for each test condition. These data in turn were evaluated to plot, as a function of frequency (or frequency content), the following test outputs:

- (1) Vortex shedding frequency (anemometer signals).
- (2) Lift direction, fundamental mode vibration amplitudes and frequencies (calibrated strain gage signals).
- (3) Drag direction, fundamental mode vibration amplitudes and frequencies (calibrated strain gage and optical tracker signals).
- (4) Drag direction, rocking mode vibration amplitudes and frequency. Amplitudes, characterized by displacement of test element ends from their equilibrium position, were established from the contribution to the strain gage signals occurring at the rocking mode frequency.

In addition, the DC shift of calibrated strain gages was measured to determine drag direction steady-state displacement.

5. TEST RESULTS

Pertaining to the 16.5 Hz nominal natural frequency test, Fig. 4 presents non-dimensional frequency and single (zero to peak) amplitude data of the vortex shedding (frequency only), drag and lift vibrational displacements as a function of the reduced (non-dimensional) velocity $\underline{U} = U/f_n D$ of the water flow in the minimum gap between the test element and the channel walls. The steady-state drag displacement encountered is shown, too. Corresponding curves were prepared for the other two tests, also. Tables 1 and 2 present, respectively, the basic data and flow velocities at the occurrence of frequency and/or mode shifts for all three tests.

The non-dimensional representation indicated that the characteristic behavior of the output test parameters was similar during all three tests. This behavior will be discussed separately for each parameter in the following sections, and performance particular to any one test will be noted.

Vortex-Shedding Frequency

The anemometer probe sensing the vortex shedding frequency was located approximately where the downstream projection of the outside diameter of the test element, at midspan, will intersect a plane 0.750 in. downstream of the nominal element center. At low flow velocities up to $\underline{U} = 4$ or 5, the vortex shedding proceeded, approximately at its expected frequency, characterized by a Strouhal Number of about 0.2. Upon reaching a flow velocity $\underline{U} = 2$ to 2.5, the vortex shedding remained fairly constant until a value of $\underline{U} = 3$ to 3.5 was reached, when the expected rising trend was continued. At times, in the range $\underline{U} = 1.5$ to 2.5, the anemometer recorded vortex shedding - or perhaps here more properly called wake disturbances - at the natural frequency of the vibrating test element.

As the flow velocities were increased beyond $\underline{U} = 5$ to 6, the vortex shedding frequency remained approximately constant, synchronized with the cylinder vibration at or near the natural frequency. At the highest experimental velocities (up to $\underline{U} = 12.7$ for the 11.3-Hz test), separation of the frequencies (i.e., return of the vortex shedding frequency to the value given by Eq. 1), as observed during airflow experiments (5), had not yet been noticed.

Lift-Direction Vibration

At low flow velocities, the lift-vibration frequency followed the cylinder natural frequency. With increasing flow velocities, the power spectra indicated an increasing, and soon dominating, component at the vortex shedding frequency. Thus in effect, the cylinder frequency was forced down into synchronism with the lower vortex-shedding frequency when the flow velocity reached about $\underline{U} = 2.4$ to 2.9. Both frequencies remained synchronized throughout the upper range of flow velocities, apparently first following the vortex-shedding frequency (increasing with flow velocity) to coincidence with the natural frequency at $\underline{U} = 5$ to 6, and then remaining approximately constant at the natural frequency of the cylinder. As discussed in the following paragraph, the test element was impacting when passing through part of the velocity range; it does not appear likely that absence of such impacting would have essentially altered the synchronization, although this cannot be proven at this time.

The amplitudes of the lift direction vibrations began to rise sharply with increasing flow velocity near $\underline{U} = 5$, indicating approach of a resonance-type effect. Subsequently, the test element began to impact its limiting stops, which nominally allowed about 0.200 in. (i.e., 0.300 diameters) radial, i.e., single amplitude, displacement. However, the processed (i.e., also averaged) lift-amplitude data indicated impacting

to be occurring after only about 0.200 diameters of displacement. Reasons for this limitation include: (1) occurrence of (higher than average) peak amplitudes; (2) unsymmetric vibration with respect to clearance hole location; (3) occurrence of simultaneous vibrational as well as steady-state drag-direction displacements (clearance hole locations were shifted to compensate for 0.063 in. of steady-state drag displacement).

As described in the Section 4, the flow velocity was increased to find test conditions without contacting of the limiting stop. In other words, attempts were made to establish test conditions with a velocity high enough to have passed through resonance, but low enough not to incur combinations of lift and drag displacements that would again result in contacting. The 11.3-Hz tests permitted four such impact-free, beyond-resonance test conditions, the 16.5-Hz test only one, and the 23.5-Hz tests, performed at the highest absolute water velocity levels, none. Several test conditions were run despite impacting during the 16.5-Hz and 23.5-Hz tests. The resulting lift-amplitude data are considered questionable and noted as such on Fig. 4. During the exposure to a third, strongly impacting test condition of the 23.5-Hz element, the tube spring failed at the element mount and terminated that test.

Drag-Direction Vibration

Drag-direction vibrations occurred at the natural frequency at the lower flow velocities. As the velocity was increased, the power spectra indicated the additional contribution to the vibrations at a rocking-mode frequency; this was verified by 180° out-of-phase deformation of strain gages that are in phase during translational displacement. The rocking-mode vibrations began at flow velocities ranging from $U = 3.6$ to 4.9, indicating that this useful parameter did not provide correlation in this case. However, when the flow velocity is non-dimensionalized by incorporation

of the rocking- rather than the first-mode frequency, the corresponding variation of the $U/f_r D$ parameter narrows from 1.5 to 1.7. Thus it appears that excitation of the rocking-mode in the drag direction, unexpectedly encountered at relatively low flow velocities, may be the result of coincidence of the vortex-shedding frequency with one-third of the rocking-mode frequency. Previously, only the one-half ratio was considered of significance with regard to drag-direction excitation. However, Wooton et al (3) reported that piles driven into the ocean floor were excited by a tidal flow whose corresponding vortex-shedding frequency at $U = 1.67$ coincided with one-third of the fundamental natural frequency at which the piles vibrated in the drag direction.

As far as the part of the vibrations that occurred near the natural frequency level are concerned, upon reaching a flow velocity U of approximately 4, this vibration frequency shifted down ($\sim 10 - 20\%$) to coincidence with the vortex-shedding and lift-direction vibration frequency. With increasing velocity, the three synchronized frequencies again rose to and, subsequently, remained at or near the natural frequency level.

At the higher flow velocities, the power spectra of the drag displacements indicated that, at times, there were additional contributions occurring at twice the cylinder vibration frequency.

The amplitudes of the drag displacement incurred a peak of about 0.08 to 0.09 diameters at flow velocities U ranging between 2.25 and 2.45. Increased flow velocity resulted in a drop of amplitude, and subsequently, another smaller amplitude peak at $U = 3$ to 3.5. After rocking started, there usually appeared another peak in the vicinity of $U = 6$. Figure 4 indicates both the first-mode and rocking-mode amplitudes. These amplitudes were always smaller, and usually much smaller, than those imposed in the lift direction.

Steady Drag Direction Displacement

The determination of steady drag displacement required the reading of the DC level of oscillating strain gage signals. At low flow velocities, this could not be done with accuracy. The amplitude versus velocity curves roughly followed the parabola expected from theory. The drag displacement amplitudes appear to increase when the lift direction amplitudes are very high. This is not surprising when one considers that the transversely vibrating cylinder presents a larger frontal area to the oncoming flow than a stationary one, and subsequently higher drag forces can be expected under those conditions.

Variation of the Cylinder Vibration Frequency

The foregoing test results indicate that in an intermediate flow velocity range approaching $\underline{U} = 5$, the vortex shedding was sufficiently strong to markedly determine the frequency of the cylinder vibration. At other times, the cylinder vibrated in the vicinity of the fundamental natural frequency of the element, whose nominal value was estimated from the experimental data. In the range above $\underline{U} = 5$, the effect of vortex shedding, and possibly the associated large vibration amplitudes, appeared to modify the vibration frequency.

Damping

The pre-test calibration of the test element in air indicated variation of damping with respect to flow direction and vibration amplitude, as well as among the three test set-ups. The suspension system design subjected to the space limitations would have made control of damping very difficult, if possible at all.

Crossflow-induced vibrations experiments performed in air have shown the dynamic behavior of the cylinder to be strongly influenced by the structural damping properties (5). Evidently, immersion into water creates

an additional damping effect. With reference to the tests described in the report, it appears that, in the presence of water immersion, the influence of the structural damping may have been not insignificant, but, on the other hand, not of the major consequence attributed to air flow excitation.

Discussion

It is recognized that three experiments do not constitute a comprehensive test program. Also, experiments involving complex and sensitive resonance phenomena can be influenced by the characteristics of the particular test facility and procedures used, and thus are not necessarily directly applicable for design purposes. Present heat-exchanger design practices recommend that the maximum vortex-shedding frequency not exceed one-third to one-half the natural frequency of the tubes. The experiments did show that if the flow velocity (and, consequently, the vortex-shedding frequency) had been limited to one-half of the natural cylinder frequency, the large lift-vibration amplitudes would not have occurred. However, in that case the drag direction deflection would have just reached a resonance peak. Whether the resulting vibration amplitudes would be as significant for a design application as they were considered to be in these experiments, is a question that appears to warrant individual attention for water flow exposure conditions.

ACKNOWLEDGMENT

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Table 1

Basic Test Data

Fundamental natural frequency of test element in water (Nominal value)	(f_n)	Hz	16.5	23.5	11.3
Weight of test element		lb	0.811	0.889	0.889
Spring constant of suspension (experimental)		lb/in.	29.0	56.5	13.3
Natural frequency in air (computed)		Hz	18.7	24.9	12.1
Natural frequency in air (experimental)		Hz	17.8	25.0	12.0
Quality factor in air					
Lift direction			80-100	40-90	160-320
Drag direction	(Q)		not obtained	50-100	300-400
Rocking mode natural frequency in water	(f_r)	Hz	38.0	58.5	36.5
Frequency ratio	(f_r/f_n)		2.30	2.49	3.23
Water flow velocity					
at $\underline{U} = U/f_n D = 1$	(U)	ft/sec*	0.92	1.31	0.63
at $\underline{U} = U/f_n D = 5$	(U)	ft/sec*	4.58	6.53	3.14
Reynolds number					
at $\underline{U} = 5$			25,400	36,300	17,400

*Note: Velocity in minimum gap between test element and channel walls.

Table 2

Flow Velocity at Occurrence
of Significant Phenomena

Listed is approximate value of reduced flow velocity $\underline{U} = U/f_n D$ at occurrence indicated.

Nominal Fundamental Natural Frequency	16.5 Hz	23.5 Hz	11.3 Hz
Lift direction vibration: Shift from natural to vortex shedding frequency	2.86	2.42*	2.89
Incipient contacting	5.23	5.00*	5.20
Minimum and maximum test condition beyond resonance range without contacting	9.41 only	none without contacting 8.45 max.	9.00 min. 12.7 max.
Drag direction vibration: Amplitude peak at low flow velocity	2.38	2.28	2.43
Drag direction vibration: Initiation of rocking mode	3.69	4.23	4.86
Same as above, except in terms of reduced velocity $U/f_r D = \underline{U} f_n / f_r$	1.60	1.70	1.50

*Note: Absolute velocity levels (and possibly turbulence) were highest here.

Captions for Figs. 1 through 4 for Technical Paper
"Crossflow-Induced Vibration of a Circular Cylinder in Water"

by H. Halle and W. P. Lawrence

Fig. 1 Test chamber cross. Horizontal pipe run contains flow channel. Vertical branches contain test fixture. Transverse observation ports mount anemometer.

Fig. 2 Test fixture with mounted test element.

Fig. 3 Test element and suspension system.

Fig. 4 Vibration frequencies and amplitudes versus flow velocity during 16.5 Hz experiment.

Fig. 1

Fig. 2

Fig. 3

FIG. 4