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Title EFFECT OF VIBRATION ON FORCED CONVECTION TO WATER FROM A CYLINDER AT REYNOLDS NUMBERS IN THE RANGE OF STABLE VORTEX SHEDDING

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An experimental investigation has been made of the effect of vibration on heat transfer from a cylinder in crossflow. The cylinder used was 3/16-inch in diameter and was sinusoidally oscillated in the vertical direction by a mechanical drive mechanism while immersed in an open water channel. Nominal amplitude ratios, a/d, varied from 0.125 to 1.43 while frequency varied from 0 to 6.5 cycles per second. Data was taken at three flow Reynolds numbers: 64; 103 and 144. Temperature difference between the cylinder and the water was held at approximately 10° F. A hydrogen bubble technique was used in conjunction with dye studies to observe the flow near the cylinder.

It was found that the heat transfer could be increased or

decreased by vibration, depending on the amplitude ratio and the vibrational Reynolds number, $(N_{Re})_v = a\omega\rho d/\sqrt{2}\,\mu$. Below a critical vibrational Reynolds number, which increased with the flow Reynolds number, oscillation had no effect on the heat transfer. After this critical point was passed, the data for the amplitude ratios 0.125, 0.250, and 0.500 showed a decrease in the heat transfer while the data for higher amplitude ratios (a/d = 1.00; 1.43) gave a steady increase. After reaching a minimum, the data for a/d = 0.500 increased to converge with the data of higher amplitudes; this trend was also suggested by the lower amplitude ratios but was not completely substantiated because of the frequency limitation of the drive mechanism. The maximum decrease in the heat transfer below stationary conditions was about 20 percent; the maximum increase in the heat transfer was 180, 125, and 90 percent for flow Reynolds numbers of 64; 103 and 144 respectively.

The vortex shedding frequencies were in the range of frequency used for mechanical oscillation; noting this, an attempt was made to oscillate the cylinder through the immediate frequency range of vortex shedding to see if any special heat transfer effects would result. No change occurred other than the general increase due to increasing the vibrational intensity.

The increases in heat transfer are attributed to vibrationally induced turbulence. An adequate explanation of the decreases in heat transfer was not provided by this investigation; however, it is thought that the phenomenon occurring may be similar to the observations of Kubanskii (11) who imposed acoustical vibrations on a cylinder in crossflow. By appropriately locating the cylinder with respect to the nodes of the sound field he found that the point of flow separation shifted toward the upstream side of the cylinder resulting in lower heat transfer rates.

A critical $(N_{Re})_v/(N_{Re})_f$ required to alter the heat transfer was established for this investigation and although it had a weak dependence on the amplitude ratio, a value of 0.35 serves as a good approximation for the data of this experiment.

EFFECT OF VIBRATION ON FORCED CONVECTION TO WATER FROM A CYLINDER AT REYNOLDS NUMBERS IN THE RANGE OF STABLE VORTEX SHEDDING

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TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

a	Sinusoidal amplitude of oscillation, ft.
af	Intensity of vibration, ft/sec.
c_p	Specific heat at a constant pressure, Btu/lbm °F.
d	Outside cylinder diameter, ft.
f	Frequency of vibration, cycles/sec.
h	Convection coefficient of heat transfer, Btu/sec ft^2 °F.
k	Thermal conductivity, Btu/sec ft°F.
n_c	Angular velocity of cylinder oscillation, rpm.
n_v	Frequency of vortex shedding, cycles/min.
V	Flow velocity, ft/sec.

Greek Symbols

ρ	Density, lbm/ft^3.
μ	Absolute viscosity, lbm/sec ft.
ω	Circular frequency, rad/sec.
ν	Kinematic viscosity, ft^2/sec.

Dimensionless Groups

a/d	Amplitude ratio
$(N_{Nu})_s$	Stationary Nusselt number, hd/k.
$(N_{Nu})_v$	Vibrational Nusselt number, hd/k.
N_{Pr}	Prandtl number, $c_p\mu/k$.
$(N_{Re})_f$	Flow Reynolds number, $\rho vd/\mu$.
$(N_{Re})_v$	Vibrational Reynolds number, $a\omega\rho d/\sqrt{2}\,\mu$.

EFFECT OF VIBRATION ON FORCED CONVECTION TO WATER FROM A CYLINDER AT REYNOLDS NUMBERS IN THE RANGE OF STABLE VORTEX SHEDDING

INTRODUCTION

In the last few years interest in vibration as a means of increasing heat transfer rates has grown considerably. There have been numerous experimental investigations undertaken and some attempt made to define the basic phenomena involved in the mechanism of heat transfer coupled with vibration. At this time it seems that there yet remains a considerable amount of work to be completed before the basic concepts involved are adequately defined. A review of the pertinent literature manifests some apparent discrepancies between the conclusions of some investigators. However, a few qualitative conclusions have emerged which are widely accepted.

Fand and Kaye (8, p. 133) conclude that the convective heat transfer is unaltered unless the vibrational intensity is great enough to change the basic structure of the boundary layer flow around the heat transfer surface; otherwise any vibration of the surface or fluid medium is only a weak perturbation superimposed upon the basic convective field. No experimental results have contradicted this; however, the critical intensity of vibration required to affect the heat transfer varies a great deal from experiment to experiment.

The purpose of this investigation is not to fully evaluate the effects of oscillation upon heat transfer but to extend the range of test conditions investigated to this date. The major portion of work previously completed has been concerned with a heating surface in a free convection field vibrated at relatively small amplitudes (those much smaller than the characteristic dimension of the heating element) and high frequencies (above ten cycles per second). It is the author's intent to expand the work in forced convection using relatively large amplitudes and low frequencies.

An apparatus has been constructed which will vertically oscillate a 3/16-inch diameter cylinder in a water environment at amplitudes from 0.02 to 0.280-inch and frequencies of 0 to 6.5-cycles per second. A special aspect of this experiment is that the forced flow regime is in the range of stable vortex shedding for a circular cylinder which includes a Reynolds number of flow from 40 to 150. Since the apparatus is capable of being oscillated at the same frequency as the vortex shedding rate, it was initially planned to pass the frequency of vibration through this range to determine if any heat transfer effects would result from so doing.

BACKGROUND INFORMATION

There have been two basic methods used in experimental research on the effects of vibration coupled with heat transfer. One method employs a mechanical oscillation of the heated surface in an otherwise undisturbed fluid medium while the other imparts a disturbance to the fluid medium by use of acoustical vibrations. These two methods accomplish the same objective, agitation of the boundary layer around the heating surface, and in general yield the same qualitative results.

Fand and Peebles (7, p. 13) conclude that the physical mechanism involved in the interaction of vibration and free convection of a horizontal cylinder is basically the same regardless of the nature of vibration (mechanical or acoustical). The critical intensity of vibration required to increase the heat transfer from the cylinder was approximately the same for both mechanical and acoustical vibrations which seems to strengthen their conclusions. The vibration vector was horizontal with respect to the gravity field and air was used as the fluid medium.

Of considerable interest is Fand's conclusions (6, p. 18) on the comparison of vertical mechanical oscillations and horizontal transverse mechanical oscillations of a horizontal cylinder in free convection. He points out that the critical intensity of vibration is the same for both cases; yet, the boundary layer flow is radically different for each of the two methods. The increase in heat transfer for transverse horizontal vibrations is attributed to thermoacoustic streaming which is marked by the shedding of vortices from the upper half of the cylinder. In the case of vertical vibrations the prime mechanism appears to be vibrationally induced turbulence. Conclusions on the interaction of vibrations and heat transfer by other investigators have been summarized in Fand's report (6) and the reader is referred to his work for more detailed explanations.

Vibrational intensity or a vibrational Reynolds number has been used as the fundamental variable by most experimenters in presenting their data. Although useful for correlating data of closely related experiments it has been illustrated that any gross extrapolation of data may result in large errors. Fand and Peebles (7, p. 2) point out that for intense vibrations of a wire with natural convection experiments have shown the heat transfer coefficient, h, to be a function of $(af)^2$; while in another experiment using a 3/4-inch diameter cylinder with horizontal acoustical vibrations, h was proportional to $(af)^{2/3}$ for sufficiently large vibrational intensities. Another study using vertical mechanical oscillations of a 7/8-inch cylinder showed h to be proportional to (af) above a certain vibration level. The ratio a/d is an important parameter (6, p. 19) and large differences in a/d ratios may account for some of the large differences in the results of related experiments.

In any event it should be recognized that the coupling of vibrations and convection is a non-linear phenomenon and any superposition or extrapolation may not be valid.

SURVEY OF LITERATURE

Vibration With Free Convection

One of the earlier investigations to determine the effects of vibration on heat transfer was conducted by Martinelli and Boelter (13) who mechanically vibrated a horizontal cylinder in the vertical direction while immersed in a tank of water. The cylinder was 3/4-inch in diameter and 12 5/8-inches long; the amplitude range was 0 to 0.10-inch while frequency varied from 0 to 40-cycles per second. They defined a Reynolds number of oscillation equal to $a\omega\rho d/\sqrt{2}\,\mu$ to correlate their data. Below a Reynolds number of 1,000 no significant increase in the heat transfer was observed. As the Reynolds number was increased to 7,500 a more than four fold increase in the heat transfer occurred. Unfortunately there is some question concerning the accuracy of their data as later work reported by Boelter was unable to reproduce their original data (8, p. 134).

Swanson (20) used an experimental arrangement closely resembling the design of Martinelli and Boelter. His Nusselt numbers were consistently higher (20, p. 32) than those of Martinelli and Boelter and the maximum heat transfer increase was 260 percent.

Lemlich as referenced by Fand (6, p. 3) observed up to 400 percent increases in the heat transfer by vibrating electrically heated wires of three different diameters (0.0253, 0.0396, and 0.0810-inch).

Sinusoidal amplitudes up to 0.115-inch were used with a frequency range from 39 to 122 cycles per second. The range of temperature difference between the wires and the air medium varied from 7 to 365 degrees F. Similar results were reported for both horizontal and vertical vibrations.

An investigation by Deaver, Penny, and Jefferson (5) concerned itself with relatively large amplitudes of 0.09 to 2.76-inches (total) and low frequencies from 0 to 4.25 cycles per second. The apparatus was a 0.007-inch platinum wire immersed in a water tank. Temperature differentials varied from 10 to 100 degrees F. A Reynolds number of oscillation was defined using the mean velocity of the wire. This mean velocity was equal to $4af$. They concluded that their results could be divided into three regions: free convection, mixed-flow convection, and forced convection. In the free convection region the Nusselt number was independent of the Reynolds number. In the mixed-flow region the Nusselt number was a function of the Reynolds number as well as the Grashof-Prandtl (Rayleigh) number. The forced convection region showed the Nusselt number to be dependent only on the Reynolds number while being independent of the Rayleigh number. In fact a correlation was obtained in this latter region almost identical to McAdams' (14, p. 267) recommended curve for flow normal to single cylinders. An accurate correlation was not possible in the mixed region but the data illustrated that the critical Reynolds number required to affect the heat transfer increased with an increasing Rayleigh number.

Russ (17) oscillated three horizontal cylinders in air using diameters of 3/4, 1/4, and 0.085-inches. Amplitude ranged from 0 to 0.165-inch and frequency varied from 53 to 130 cycles per second. He observed four distinct regions: a free convection region, two regions in the mixed-flow region each identified by a definitely different rate of increase with vibrational intensity, and a final region which approached the forced convection correlation of McAdams (14, p. 267).

Vibration With Forced Convection

Scanlan (18) vibrated a heating surface in contact with water in a direction normal to the surface. The article is not clear as to what the experimental arrangement was; however, it is mentioned that amplitudes from 0 to 0.002-inch and frequencies from 20 to 600 cycles per second were used. Data was taken at four different Reynolds numbers of flow, 360; 720; 1460 and 2170. Maximum heat transfer resulting from vibration was about 280 percent over that of stationary values and peaks for all amplitudes occurred in a frequency range of 50 to 80 cycles per second. In general the lower Reynolds number of flow used, the greater increase in heat transfer for a particular amplitude and frequency. A unique peculiarity of

Scanlan's data was that after the peak in heat transfer was passed the heat transfer coefficient decreased sharply and tended to stabilize at a relatively constant value regardless of frequency except for the highest amplitude setting (0.002-inch). This higher amplitude setting had a second peak which decreased with increasing Reynolds number of flow. Scanlan predicted a decrease in the heat transfer with the following reasoning: if the acceleration of the surface away from the liquid becomes great enough, cavitation will occur and a resulting "blanketing effect" will counteract the increase in heat transfer due to vibration. Explanation of the second peak observed at higher amplitudes was somewhat nebulous.

In contrast were the conclusions of Larson (12, p. 42) who imposed ultrasonic vibrations on a liquid medium and observed the heat transfer from a one-inch diameter sphere. The liquid was subjected to a sound field ranging in frequency from 20 to 1000 kc per second with intensities up to 6 watts/cm^2. For forced convection of water his results showed an increase in the Nusselt number for the entire range of flow Reynolds numbers tested (200-6000) at lower frequencies. At 1000 kc per second no measurable increase in heat transfer was detected for Reynolds numbers above 1000. He concluded that the general behavior seemed to support the conclusion that the mechanism of heat transfer at lower frequencies is due to cavitation.

Kubanskii (11) studied the combined effect of forced air convection and sound waves using a 2.4-cm. diameter cylinder subjected to sound intensities of 0.03 to 0.16 watts/cm^2 at frequencies from 8 to 30 kc per second. Direction of propagation of the sound waves, flow direction, and axis of the cylinder were all mutually perpendicular. Heat transfer was increased by up to 50 percent. Kubanskii also observed that the point of flow separation could be moved by changing the position of the cylinder axis with respect to the nodes of the sound field. If the point of flow separation was moved toward the upstream side of the cylinder, the heat transfer was lowered; a shift in flow separation towards the downstream side resulted in greater heat transfer rates.

A study carried on at the Southwest Research Institute (21) imposed acoustical vibrations on a pipe with water flowing over the perimeter through an annular passage. Increases of up to 450 percent in the heat transfer were observed using a range of flow Reynolds number from 540 to 20,000, the lower the Reynolds number the larger the increase in heating rate.

Koerkenmeir (9) investigated the effect of vibration coupled with parallel forced air flow over small cylinders. The cylinder sizes were 0.072 and 0.082-inches in diameter and were heated by an electrical current. Amplitudes of vibration reached a maximum of 0.10-inch while frequency was varied between 25 and 90 cycles per second. Heat transfer was effectively increased by 103 percent for the smaller cylinder. He concluded that a critical Reynolds number of vibration was established for a given flow velocity and increased with the velocity. No increase in heat transfer was observed below this critical value. Also of interest was his establishment of a critical $(N_{Re})_v/(N_{Re})_f$ which was a constant equal to 0.035 for his experiment.

Anantanarayanan and Ramachandran (2) also did a study in forced air convection from a wire in parallel flow. They used a 0.018-inch nichrome wire oscillated at frequencies of 75 to 120 cycles per second and amplitudes from three to twenty millimeters. Qualitative results were in general the same as those of Koerkenmeir and a maximum increase of 130 percent was observed in the heat transfer rate.

APPARATUS

The apparatus components included a test cylinder, a drive mechanism, a water table, an electrochemical system for flow visualization, a hot film anemometer, and the necessary instrumentation for amplitude, frequency, temperature, and power measurements. The assembled apparatus, excluding supporting instrumentation, is shown in Figure 1, page 16.

The test cylinder was a six inch length of 3/16-inch brass tubing internally heated by a core wrapped with constantan resistance wire. For temperature measurement one thermocouple was soldered on the surface of the cylinder at its mid-point with leads passing through the inside and out the cylinder end. A filler of low conductivity was used to seal the ends of the tubing. Figures 3 and 13 and the section on construction of the test cylinder in the appendix will furnish the reader with more detail.

Mounting of the cylinder was done with the use of 1/4-inch plexiglass struts which allowed visual observation of the flow in the vicinity of the cylinder. Some concern was felt for the boundary layer growth that would occur on the sides of the plexiglass struts; hence, the cylinder was mounted near the front side of the struts. Calculations using flat plate boundary layer theory showed the boundary layer to be very thin at the cylinder location for the velocities used and any error from assuming a constant velocity across the cylinder length to be negligibly small. The cylinder was positioned with the thermocouple on the upstream side. Mounting and plugging of the cylinder ends reduced the effective heated length to 5 5/8-inches.

The drive mechanism, Figure 2, page 16, consisted of an adjustable eccentric driven by a direct current electric motor coupled through a worm gear speed reducer. Rubber mounts and flexible shaft couplings were used in mounting the gear reducer and motor to minimize the vibrational disturbance resulting from the electric motor. Motor speed variation from 0 to 1900 rpm was accomplished by a motor speed control powered from an A. C. line. The eccentric permitted a maximum amplitude of approximately 0.3-inch. A yoke constructed of aluminum and plexiglass transmitted the motion of the eccentric to the test cylinder. The entire drive mechanism was mounted on a cantilever frame to position the system above the water table.

The water table consisted of an 11-inch wide channel placed inside of a two feet wide outer tank; the entire construction utilized plexiglass sheets. Water flow into the channel was controlled by two valves upstream from a water manifold submerged in the table. Stabilization and filtering of the flow was accomplished by a series of fine mesh screens and linen cloth. The outflow was regulated by an adjustable gate before passing into a dump tank. For more detail the reader is referred to Figure 4 and the section on construction of the water table in the appendix.

The electrochemical system used for flow visualization was comprised of a 0.005-inch diameter steel wire serving as a cathode with a sheet of brass in the bottom of the channel serving as the anode. By placing a voltage between the anode and cathode hydrogen bubbles were shed from the wire and forced downstream with the water flow, thus serving to illustrate the flow pattern. The wire was kinked by passing it through a small gear reducer which caused the hydrogen bubbles to be shed in parallel rows. This in turn amplified the clarity of flow visualization. Position of the wire with respect to the cylinder is shown in Figure 3, page 17.

A Lintronic hot film anemometer using a platinum probe was used for determination of the channel velocity. The output signal of the anemometer was monitored by a Sanborn chart recorder.

Cylinder and water temperatures were measured by a potentiometer using an ice bath for the reference junction.

Motor rpm was determined by a strobotac and the resulting frequency of oscillation calculated by utilizing the gear reduction ratio. Below motor speeds of 350 rpm the frequency of oscillation was determined by a stopwatch and visual counting of the eccentric revolutions. Amplitude of the eccentric was measured by a dial indicator.

Power consumption of the test cylinder was measured by a voltmeter-ammeter combination.

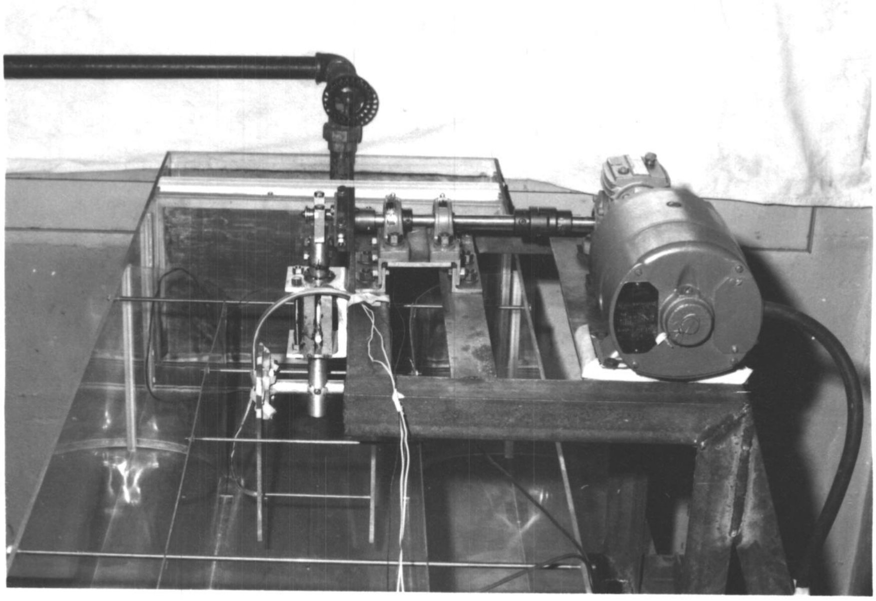

Figure 1. The assembled apparatus.

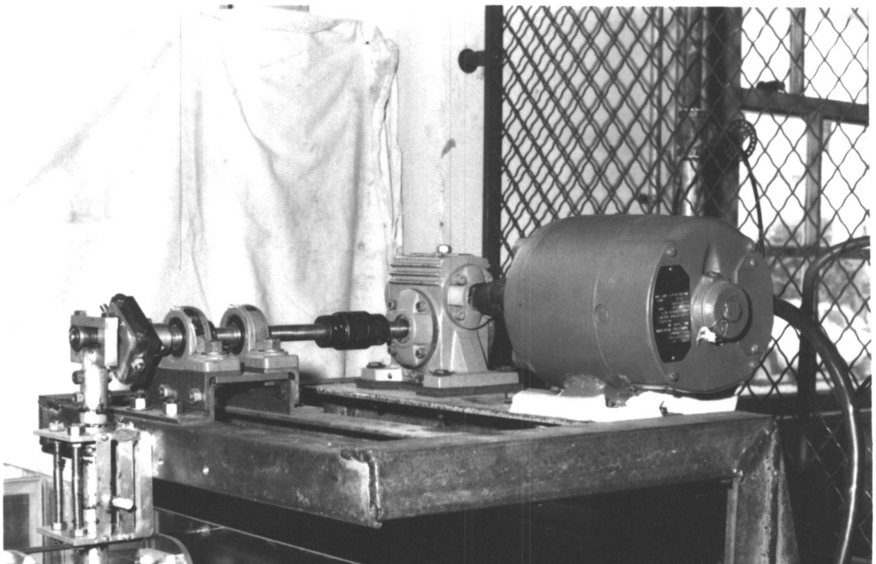

Figure 2. The drive mechanism for the test cylinder.

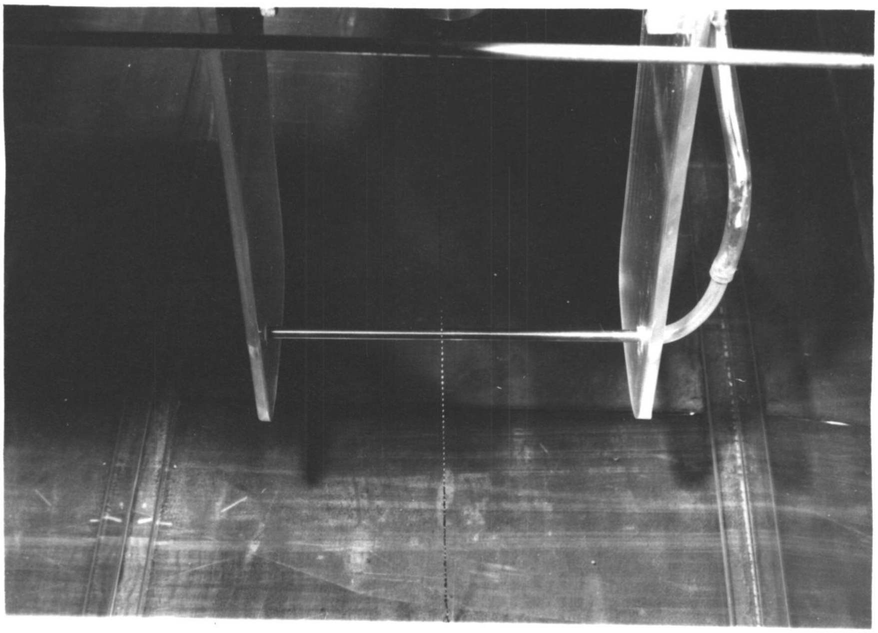

Figure 3. The test cylinder positioned in the water channel.

Figure 4. The water table.

METHOD OF TESTING

Data was taken at three distinct flow conditions which in terms of flow Reynolds numbers were 64; 103 and 144. Velocity measurement in each case was accomplished by use of the hot film anemometer mentioned in the previous section; however, the velocity was not directly measured. Previous work by Roshko (16) has demonstrated a correlation between the frequency of vortex shedding from a circular cylinder and the free stream velocity. His results (16, p. 11) were used to predict the flow velocity by recording the signal output of the anemometer on the Sanborn recorder. A vortex frequency was determined by counting the signal peaks and simultaneously observing the elapsed time. The resultant velocity was assumed constant throughout the run series (a series being at a particular flow condition). The position of the hot film probe with respect to the test cylinder is shown in Figure 5, page 21 and a typical readout of the vortex signal is illustrated in Figure 6, page 21.

It is noted that Roshko's work was done in air; however, his results are presented in terms of dimensionless parameters and should thus be reasonably accurate for any fluid. This method has also been successfully used by Abbot and Kline (1, p. 22) to calibrate a hot film anemometer for measuring low speed water velocities.

In the case of the author's experiment the method was particularly attractive because there was no need to calibrate the anemometer and a conventional flow measurement using (e.g.) a venturi meter would have required velocity corrections due to boundary layer effects on the channel walls. A venturi meter was located upstream from the water table and was used at one flow condition to check the velocity as determined from the vortex frequency. The two methods gave essentially the same result.

Cylinder data were obtained by setting the amplitude on the eccentric drive and then taking power and temperature measurements at several randomly spaced frequencies. The cylinder was occasionally wiped off to insure a bubble-free surface. Five nominal amplitude ratios, a/d, of 0.125, 0.250, 0.500, 1.00, and 1.43 were used for each flow condition. Because it was such a time consuming task to set the amplitude exactly, a small variance from these nominal amplitude ratios was allowed. A stationary Nusselt number was defined by averaging stationary data taken throughout the run series.

All data were taken at a nominal temperature difference of 10° F. within plus or minus 1° F. Since the tap water temperature did not vary a great deal, all data were taken at a nearly constant mean temperature.

The Reynolds number proposed by Martinelli and Boelter (13) using the root mean square velocity of vibration was used as the parameter of vibrational intensity ($(N_{Re})_v = a\omega\rho d/\sqrt{2}\,\mu$). All physical properties were evaluated at the mean temperature of the cylinder and water. Kreith (10, p. 537) was the source used for obtaining the physical properties of the water.

The first two run series were taken at flow Reynolds numbers of 63 and 104 using a 15 to 1 gear reducer to couple the electric motor and the eccentric. After these two runs were completed, the cylinder began giving erratic data which was apparently caused by water leakage into the cylinder ends. A second cylinder was installed and a 5 to 1 gear reducer replaced the 15 to 1; this combination was used to obtain all subsequent data. The flow conditions used for the first cylinder were duplicated within one or two percent error and two new run series were obtained with the second cylinder. The flow condition using the highest channel velocity ($(N_{Re})_f = 144$) used only the latter cylinder with the 5 to 1 reducer. The bulk of the data presented was taken with the second test cylinder, particularly the data where significant changes in heat transfer occurred because little change was noticed in the lower frequency range while using the 15 to 1 gear reduction.

For purposes of reference, hereafter the test cylinders will be designated cylinder number one and two respectively.

Figure 5. The anemometer probe and the test cylinder.

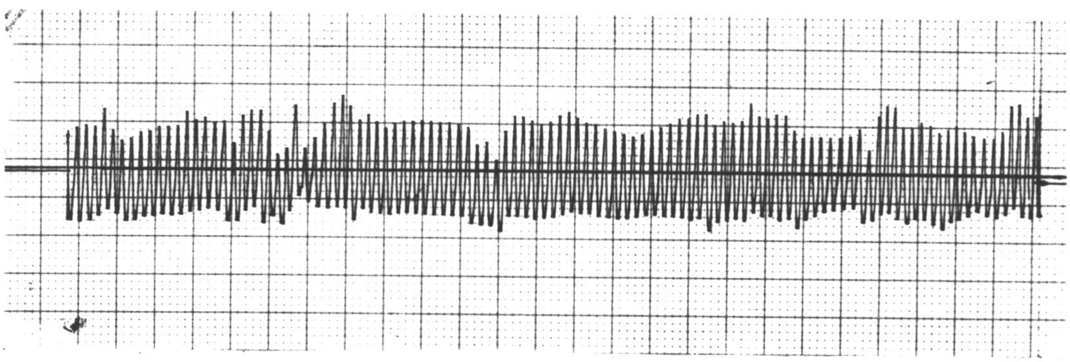

Figure 6. Anemometer signal at $(N_{Re})_f = 103$.

RESULTS

The experimental results are plotted graphically in Figures 7, 8, 9, 10, 11, and 12. Stationary data for both cylinder number one and two are compared with McAdams correlation (14, p. 267) in Figure 7. Figure 8 shows the result of passing the oscillation frequency through the frequency of vortex shedding while Figures 9, 10, and 11 give ratios of $(N_{Nu})_v/(N_{Nu})_s$ plotted against $(N_{Re})_v$ for each flow condition tested. Figure 12 presents the data independently of the flow Reynolds number.

All heat transfer data has been plotted as the ratio $(N_{Nu})_v/(N_{Nu})_s$ or $(N_{Nu})_v/(N_{Nu})_s$ minus one allowing the data of cylinder number one and two to be plotted together. Otherwise it would have been necessary to plot the data for each cylinder separately because the magnitude of the Nusselt numbers recorded individually by each cylinder was quite different. In calculating the Nusselt numbers, no correction was made for heat losses through the cylinder ends. In any case this loss was small because the cylinder ends contained a filler of low conductivity and the area of the ends was small compared to the total area of the cylindrical surface.

The maximum vibrational intensity resulted in a vibrational Reynolds number of about 800 which gave the maximum heat transfer increase for each run series. The maximum increase in heat transfer above the stationary value was about 180 percent and occurred at $(N_{Re})_f$ equal 64.

The maximum experimental uncertainity of the Nusselt number is about seven or eight percent, the average error being close to four or five percent. This estimate does not contain the uncertainities of the physical properties nor the effect of heat loss through the ends of the cylinder. Experimental error in the vibrational Reynolds number is almost entirely due to the uncertainity of frequency measurements. This uncertainity was one percent at the highest frequency measured and approached three or four percent at the lowest frequency measurement. There was a very small variation in the flow Reynolds number throughout a particular run series due to fluctuating property values because of slight changes in the mean temperature. The maximum variation was about one percent and was ignored in the calculations.

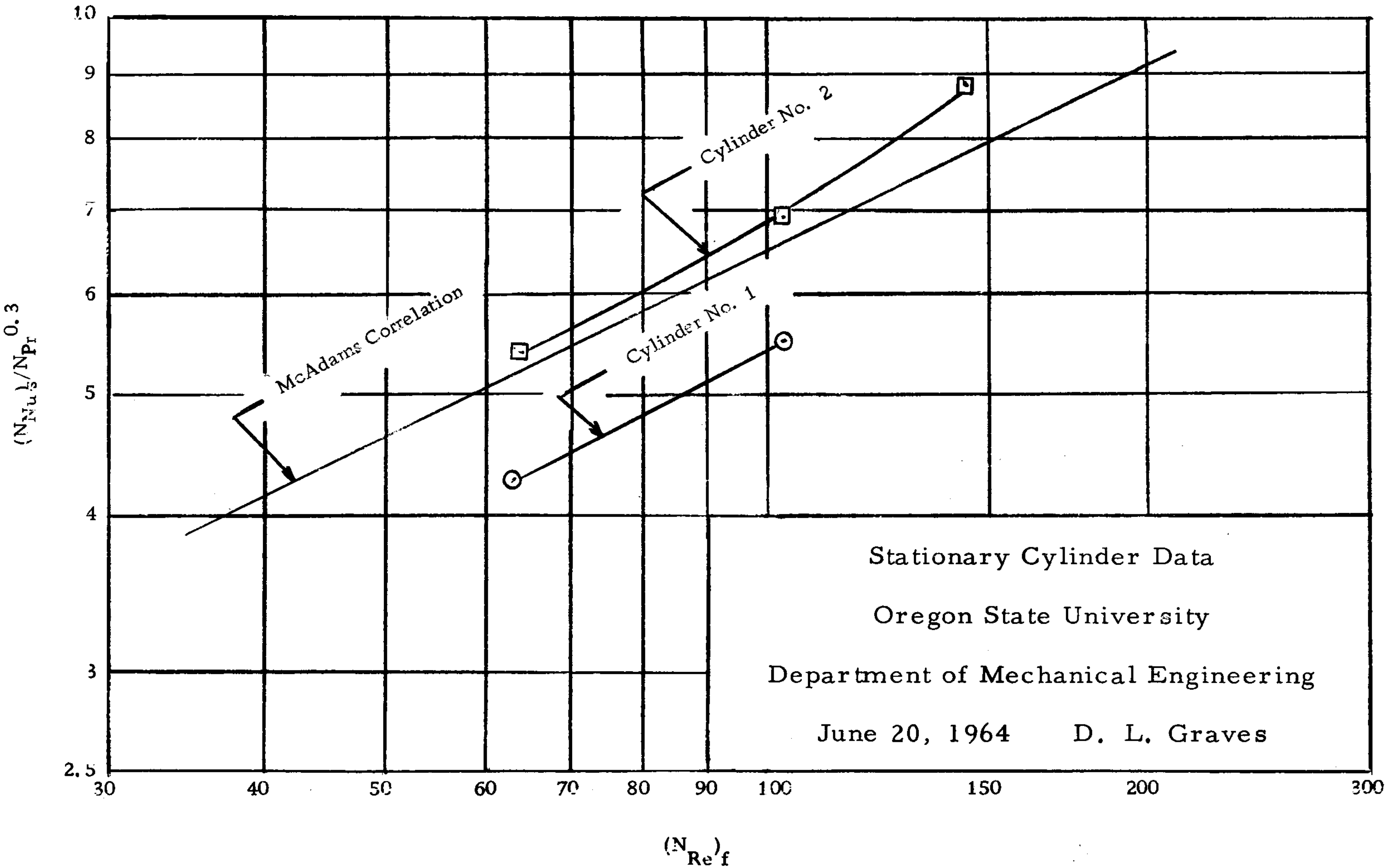

Figure 7. Stationary cylinder data.

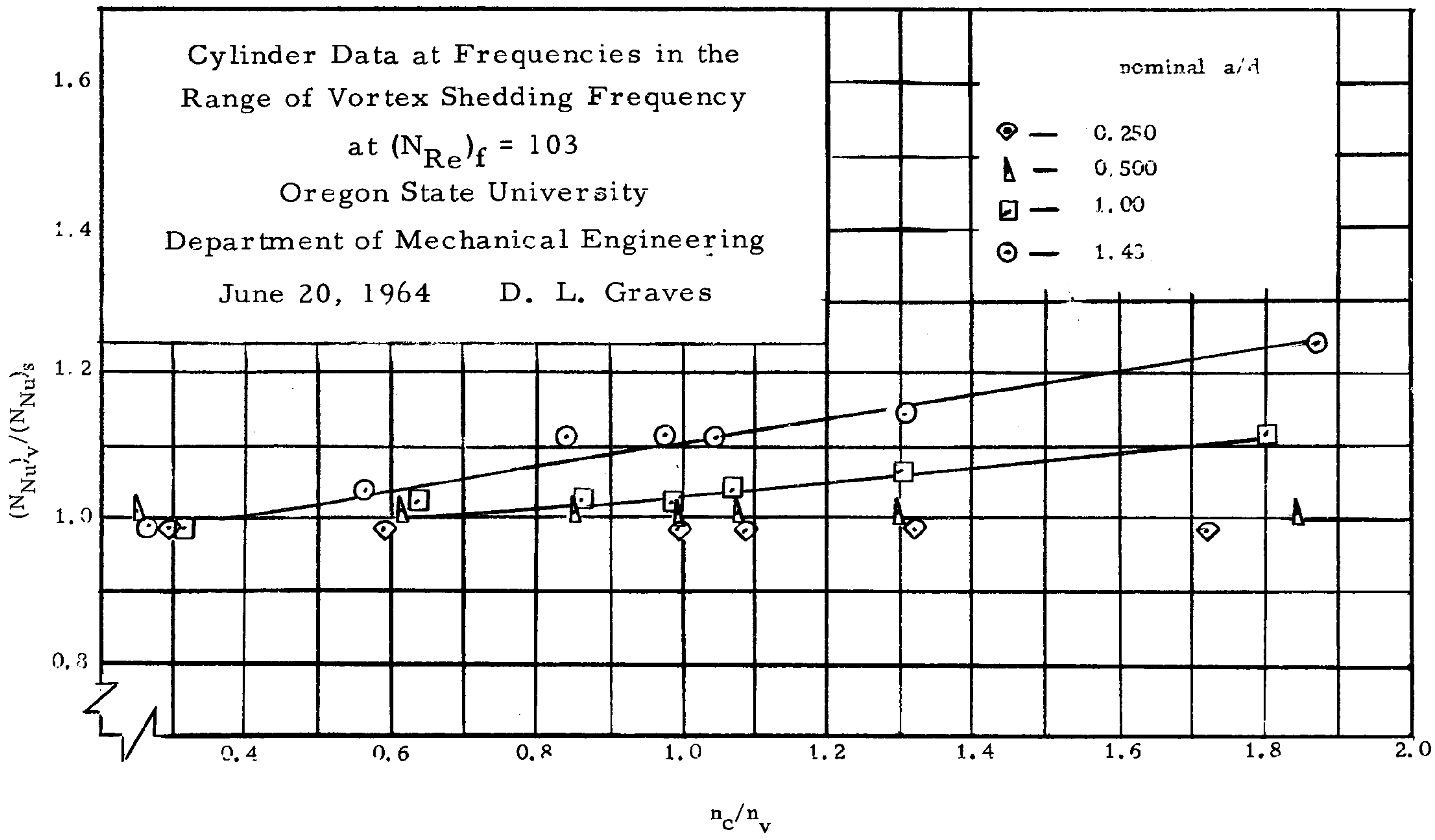

Figure 8. Cylinder data at frequencies in the range of vortex shedding frequency.

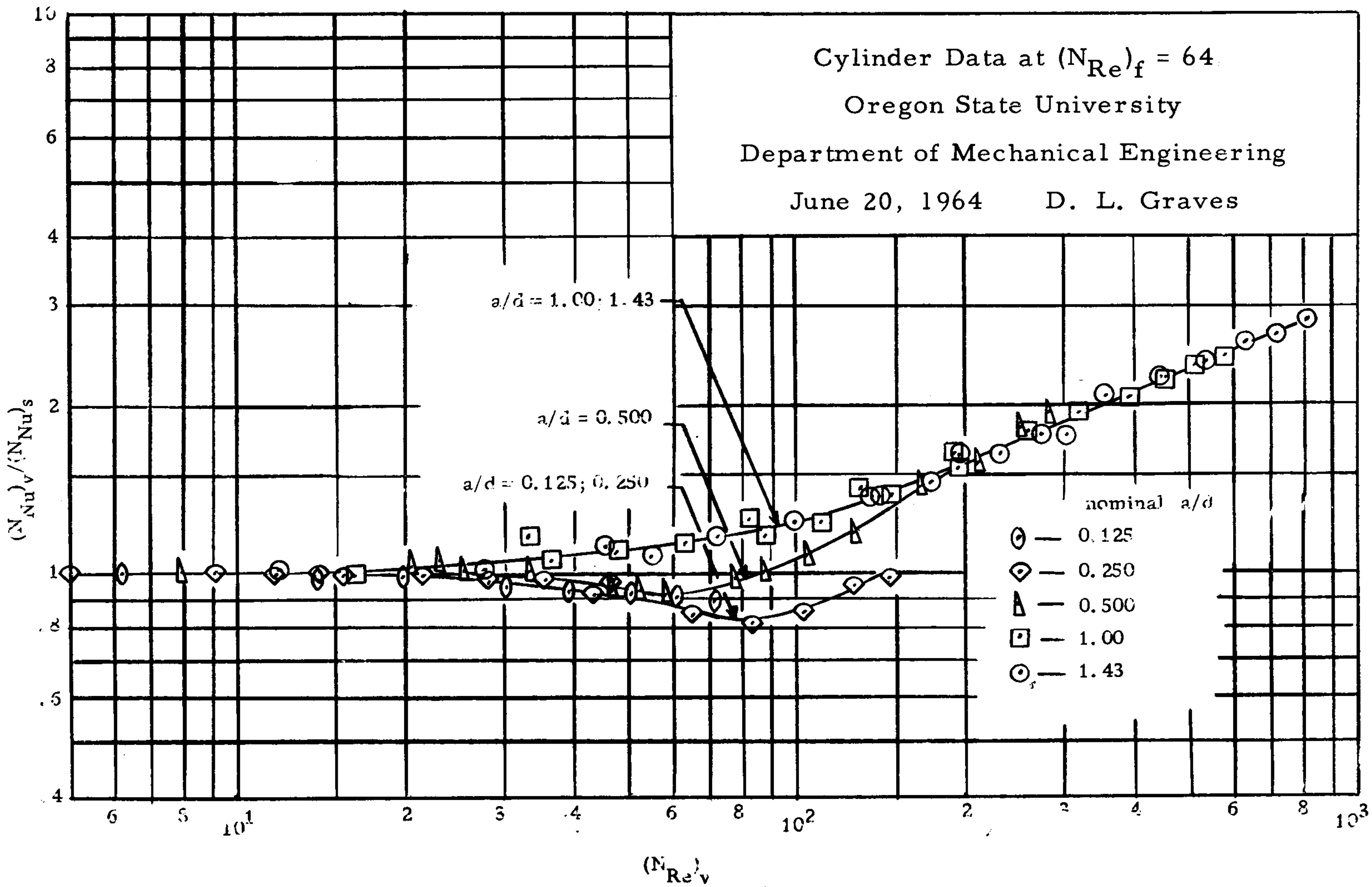

Figure 9. Test cylinder data at $(N_{Re})_f$ = 64.

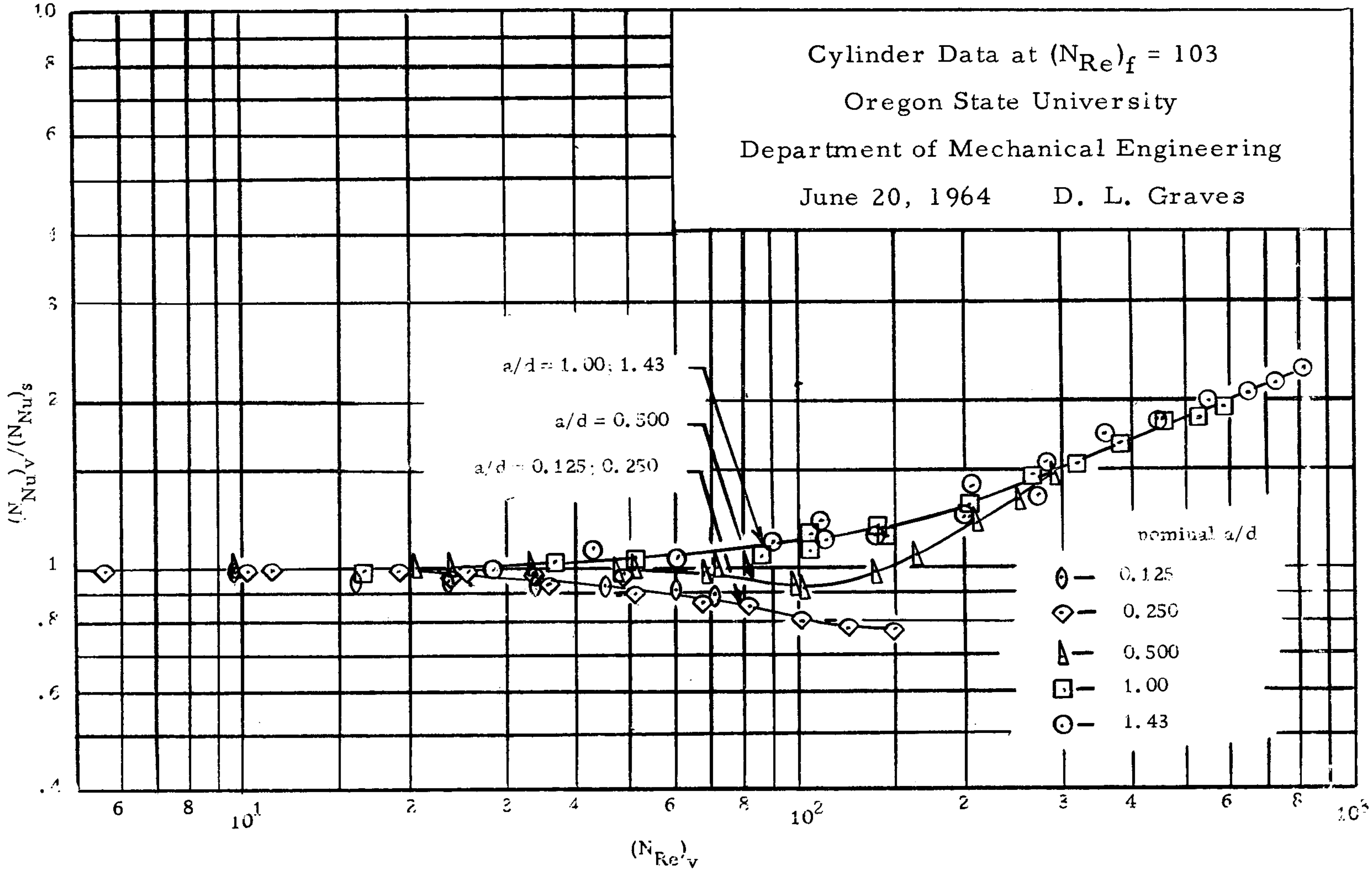

Figure 10. Test cylinder data at $(N_{Re})_f$ = 103.

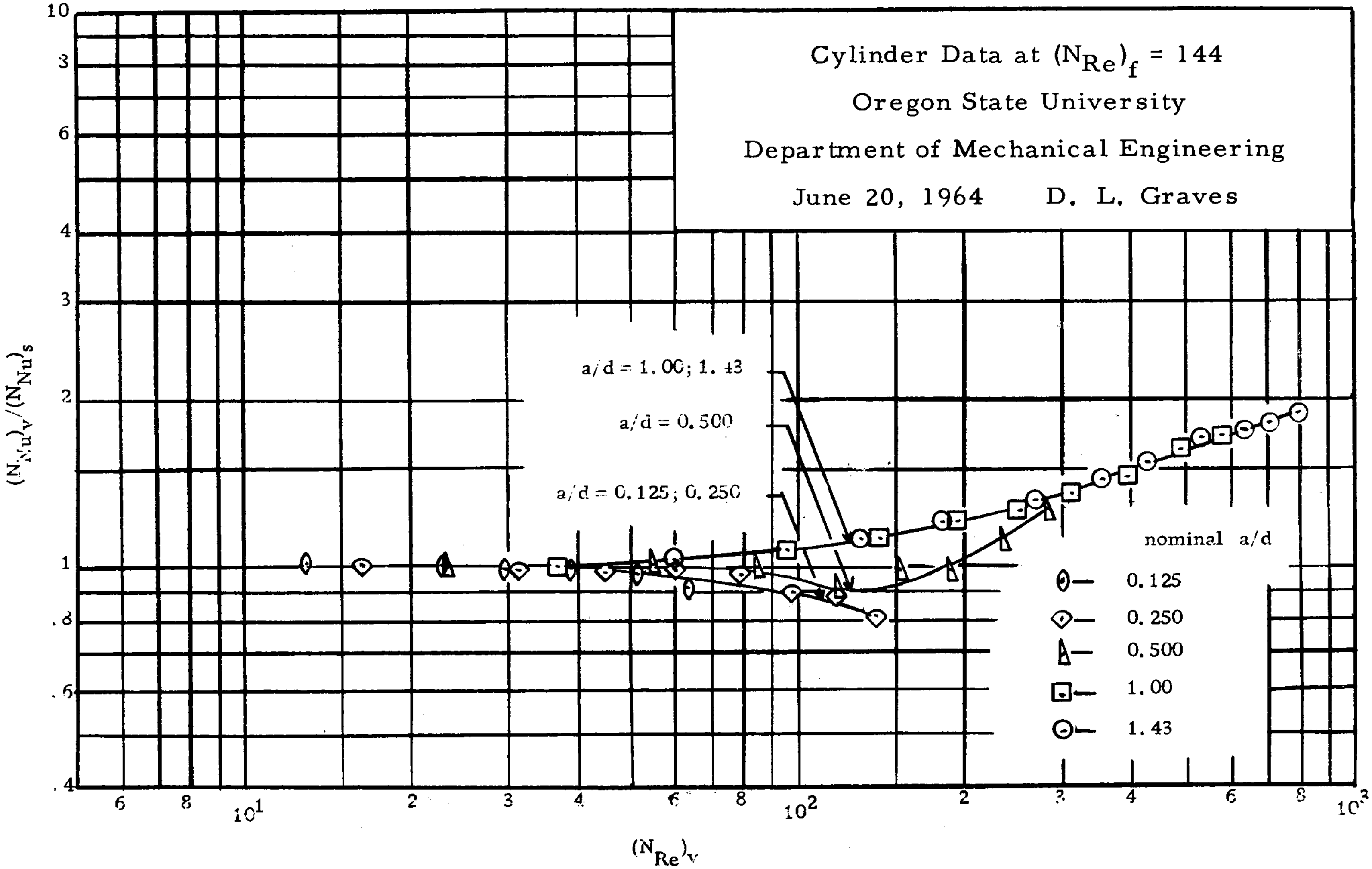

Figure 11. Test cylinder data at $(N_{Re})_f$ = 144.

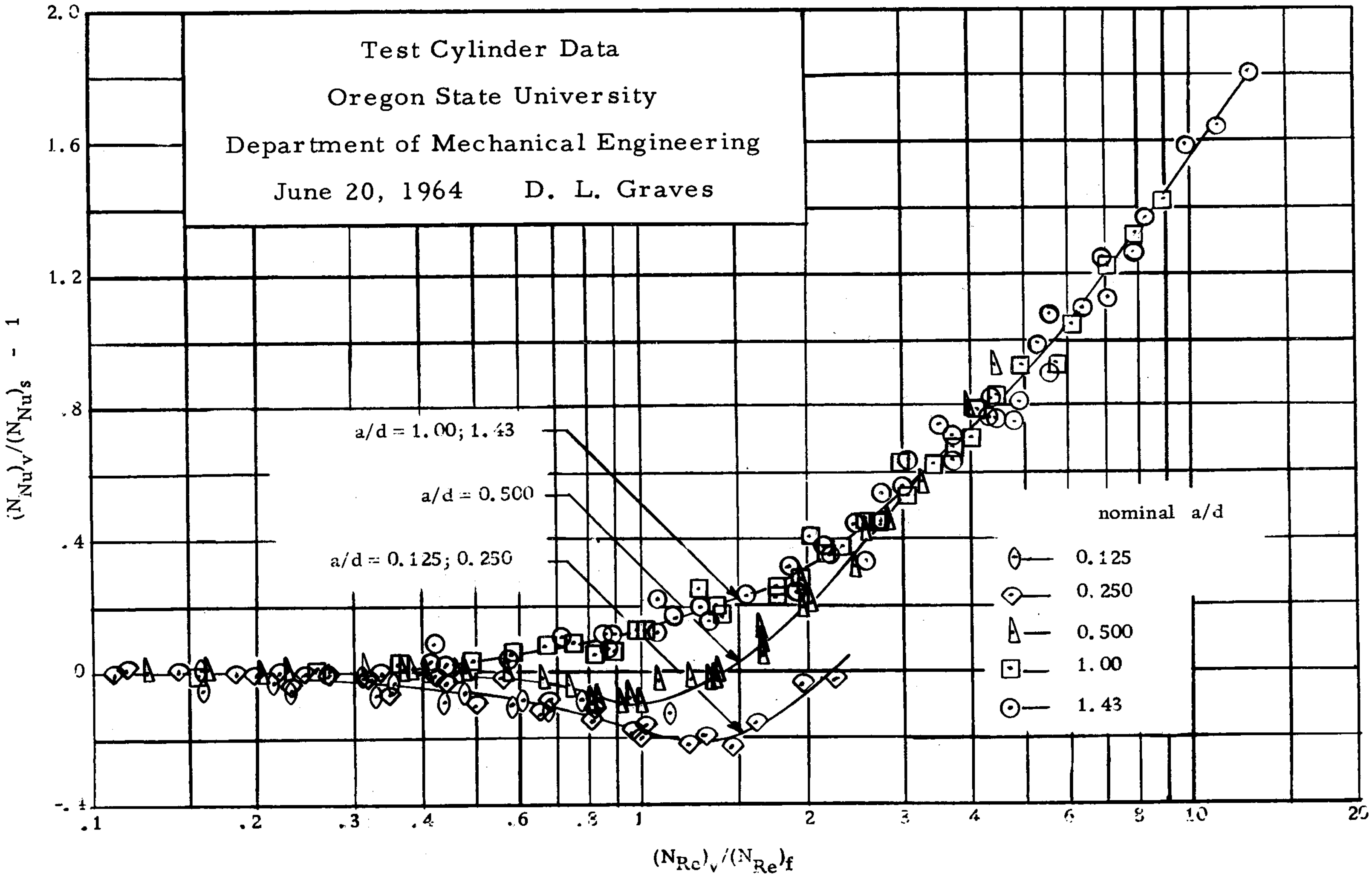

Figure 12. Test cylinder data.

DISCUSSION OF RESULTS

Figure 7, page 24, shows the magnitude of stationary Nusselt numbers recorded by the two cylinders to be quite different. The Prandtl number used in the McAdams correlation varied but a small degree in this investigation. Data from cylinder number one lie about 17 percent below the McAdams correlation while data from cylinder two lie from four to ten percent above McAdams correlation. This variance in the stationary data of different cylinder constructions is primarily attributed to the error in assuming the cylinder surface to be at a uniform temperature equal to the temperature measured by a single thermocouple and the variability that occurred in the soldering of the thermocouple junction (see the section on construction of the test cylinder in the appendix). The deviation of cylinder number two from the McAdams correlation increases with increasing Reynolds numbers. Again this is attributed to the variability in soldering the thermocouple junction to the cylinder because after the run series at $(N_{Re})_f$ equal 144 was completed the cylinder began to leak after which it was disassembled, dried out, and reassembled with the thermocouple being resoldered to the cylinder surface. The cylinder gave satisfactory data after this reconstruction although the magnitude of the Nusselt numbers were probably slightly less for the two remaining run series at the lower

Reynolds numbers than they would have been otherwise. Although the data from both cylinders varied in absolute magnitude, the ratio $(N_{Nu})_v/(N_{Nu})_s$ was essentially independent of the test cylinder used.

Figure 8 shows that there is no apparent effect on the heat transfer from oscillating the cylinder through the same frequency as vortex shedding except the general increase which results from increasing the intensity of vibration. Of more interest and significance are the results shown in Figures 9, 10, 11, and 12.

Each flow condition shown in Figures 9, 10, and 11 exhibits the same general trend. Vibration of the cylinder has no effect on the heat transfer until a critical intensity is reached after which the heat transfer increases or decreases depending on the amplitude ratio. The curves of different amplitude ratios show a tendency to converge if the intensity of oscillation is sufficiently increased. Due to frequency limitation of the drive mechanism this observation is not completely substantiated at the lower amplitudes.

The occurrence of a decrease in the heat transfer was not expected; however, it is seen that this decrease exhibits the same trend regardless of the flow condition. The greatest decrease was about 20 percent for each run series occurring at a nominal a/d of 0.250. Three different curves were drawn through the data points in the range of vibrational Reynolds numbers where heat transfer decreased to aid the reader in viewing the data trends. In the case of the two lower amplitudes one curve was used but it should be borne in mind that if higher frequencies had been attainable the lowest a/d (0.125) may have diverged from this latter curve at higher vibrational intensities. The data for the amplitude ratio 0.500 decreased to about nine percent below the stationary data for each run series and then increased to converge with the data for the two highest amplitudes. This shows that there is a maximum decrease which is intuitively expected; this minimum is again illustrated for a/d equal 0.250 in Figure 9. The minimum for this latter a/d is not clearly shown in Figures 10 and 11 because the appropriate frequency range was unattainable at the higher Reynolds numbers of flow.

A significant decrease in the heat transfer due to mechanical oscillation is apparently a unique result of this investigation. The available literature sources neither discovered this phenomenon nor gave reference to its occurrence elsewhere; however, a cylinder in crossflow has a flow characteristic peculiar to other geometries. This unique characteristic is, of course, the occurrence of flow separation from the upper and lower surfaces of the cylinder at a stable frequency. Kubanskii (11) was the only literature source that investigated a cylinder in crossflow and although he used a sound field to agitate the fluid medium, his results are comparable to this work. As mentioned in the survey of literature section, Kubanskii could move the point of flow separation by changing the position of the cylinder with respect to the nodes of the sound field; by moving the point of flow separation upstream the heat emission was decreased. Unfortunately the magnitude of this decrease was not mentioned. He concluded that the point of flow separation was moved upstream when the acoustically induced currents coincided with the returning flow at the downstream portion of the cylinder. Under such conditions the returning flows are amplified resulting in a lower heat transfer rate.

Dye studies were attempted at the points of minimum heat emission but the results were inconclusive. There did appear to be more of a tendency for the dye to cling to the cylinder than in the unagitated case which suggests a thickening of the boundary layer which in turn would explain the decrease in heat transfer. The general flow pattern around the cylinder appeared similar to the stationary flow condition and the vortex shedding pattern also appeared unchanged. Of related interest is the fact that significant increases in heat transfer were accompanied by a disappearance of the stationary vortex pattern. In any event this resultant decrease in the heat transfer is a complicated phenomenon depending on the parameter a/d as well as the vibrational Reynolds number and the scope of this investigation does not suffice to adequately explain the mechanisms involved. It is likely that the phenomena involved here are equivalent to Kubanskii's observations except that the induced currents are produced by mechanical agitation instead of an acoustical field. More work is needed to validate the observations made in this investigation. Frequency and amplitude of vibration should be extended and it would be of interest to determine the local Nusselt number around the cylinder perimeter.

The higher amplitude ratios (a/d = 1.00; 1.43) did not show a decrease in heat transfer at any time. After a critical intensity of vibration is passed, the data for these amplitudes curve slowly upward and seem to converge to a straight line whose slope is independent of the flow Reynolds number. The point of this convergence in terms of the vibrational Reynolds number increases with an increase in the flow Reynolds number. The heat transfer increase due to a particular vibrational Reynolds number decreases as the flow Reynolds number increases. Maximum increases in the heat transfer over stationary values were approximately 180 percent, 125 percent, and 90 percent for flow Reynolds numbers of 64; 103, and 144 respectively. A critical vibrational Reynolds number is required to alter the heat transfer and it increases with larger flow Reynolds numbers. With the exception of the observed decreases in heat transfer these qualitative results have also been observed by Anantanarayanan and Ramachandran (2) and Koerkenmeir (9) who vibrated cylinders in parallel flow.

Figure 12, page 29, shows all data points plotted independently of the stationary flow condition. The ordinate is the fractional increase in the Nusselt number above that of the stationary value and the abscissa is the ratio of the vibrational Reynolds number to the flow Reynolds number. Although more scatter is apparent in this method of plotting, it is not dependent on the flow Reynolds number. The main advantage of this plot is that it indicates a critical $(N_{Re})_v/(N_{Re})_f$ below which no change occurs in the heat transfer. The convergence of the data to this critical value is relatively independent of the amplitude ratio; however, there is a small variation. The curve for the two largest amplitudes converges to zero at a Reynolds number ratio of about 0.35; the 0.500 a/d curve converges to zero at an abscissa value of 0.45 while the two lowest amplitudes converge to zero at a value of approximately 0.25. The minimum of the lower amplitudes is quite distinct showing that a greater vibrational intensity is needed to achieve this minimum as the amplitude decreases. The data of the two lowest amplitudes would likely converge to the data of higher amplitudes if the appropriate frequencies had been attainable. This is clearly shown by the 0.500 a/d data. The data at the higher intensities of oscillation again plots as a relatively straight line as was the case for the separate plots in Figures 9, 10, and 11. In this region the vibrational Reynolds number is much larger than the flow Reynolds number; hence, it is expected that vibration is the dominant mechanism of heat transfer. The curved region of the data is apparently influenced by both the vibrational velocity and the flow velocity since they are of the same order of magnitude.

No attempt was made to correlate the data. In view of the complicated interactions occurring in the region where heat transfer decreased, it is doubtful if a correlation of practical value could be obtained. This investigation distinctly illustrates that the parameter a/d is an important variable as well as the vibrational Reynolds number for cylinders oscillated in crossflow. At sufficiently high vibrational intensities it appears that the heat transfer may be independent of a/d; however, an extension of the range of variables involved is needed to confirm this.

The mechanism causing a decrease in heat transfer with vibration is still uncertain; fortunately, the concepts involved in an increased heat transfer rate are more tangible. Dye studies and operation of the hydrogen bubble technique at the higher amplitudes showed that as the oscillation was increased past the critical intensity the stationary vortex pattern eventually disappeared and the cylinder soon began to shed vortices at the same frequency as that of oscillation. With further increases in the oscillation intensity the flow around the cylinder became highly turbulent and these latter vortices also disappeared. Pictures were taken while using the hydrogen bubble technique but the results were not successful. It is concluded that the mechanism causing the heat transfer to increase with vibration is the vibrationally induced turbulence mentioned by Fand (6, p. 18).

CONCLUSIONS

1. Vertical oscillation of a cylinder in crossflow can either decrease or increase the heat transfer depending on the amplitude ratio, a/d, and the vibrational Reynolds number, $(N_{Re})_v$.

2. Above amplitude ratios of one the heat transfer is always increased if the intensity of vibration is greater than the critical value. An increase of the order of 180 percent was the maximum observed in this work. The maximum increase for each flow Reynolds number decreased with an increasing flow Reynolds number. The maximum decrease in heat transfer observed was approximately 20 percent for each Reynolds number of flow and occurred at an a/d of 0.250.

3. A critical $(N_{Re})_v/(N_{Re})_f$ is established for this investigation; vibrational intensities below this critical ratio have no effect on the heat transfer. This critical ratio seems to have a weak dependence on the amplitude ratio; however, a numerical value of 0.35 can be used as a fair approximation regardless of the amplitude ratio.

4. The mechanism causing an increase in the heat transfer due to vibration is attributed to vibrationally induced turbulence.

More work is needed to adequately explain the mechanisms causing a decrease in the heat transfer.

5. Oscillating a cylinder at the frequency of vortex shedding has no special effect on the heat transfer.

RECOMMENDATIONS FOR FUTURE WORK

1. The range of variables investigated in this work should be extended. Smaller amplitude ratios and greater frequencies should be used and higher Reynolds numbers of flow should also be investigated to determine if the observations made in this investigation are valid for all flow conditions. The region where heat transfer decreases might be given special attention. It would be interesting to determine the upper and lower bounds of the amplitude ratios which cause a decrease in the heat transfer and to also determine the condition at which the greatest decrease will occur.

2. A better system of flow visualization should be designed to aid in explaining the mechanism causing a decrease in heat transfer. This might be more easily accomplished by using air as the fluid medium.

3. It should prove profitable to examine the local Nusselt number around the cylinder perimeter to determine what effect oscillation has on local film coefficients. This could be done by building a cylinder with a guarded test section.

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APPENDIX

CONSTRUCTION OF THE WATER TABLE

Since the flow system used tap water from a pipe line, the water table required a design that would change the initial turbulent flow into a stable uniform velocity profile. Also the table had to be of sufficient dimensions to provide a channel Reynolds number above the laminar region. Otherwise the velocity profile across the test section would have been parabolic and the assumption of a constant Reynolds number along the test cylinder's length would have been in considerable error.

The water table was first made as a rectangular tank two feet wide, six feet long, and thirteen inches deep constructed of 1/4-inch plexiglass sheets. An adjustable gate was placed at the rear of the table to control the outflow. A bed of smooth gravel was then placed downstream of the water feed to even out the flow. It was found that the gravel bed did not completely stabilize the flow in the water table. More troublesome was the creation of tiny air bubbles from the tap water which were primarily produced by cavitation at the flow control valve and became more numerous with higher flow rates. Many of the bubbles would rapidly attach themselves to the test cylinder as they progressed downstream. Needless to say this condition could not be tolerated.

A combination of several alterations properly alleviated the above problems. Two valves were placed in series to control the flow rate into the tank. This design change reduced the pressure drop across each valve which virtually eliminated the cavitation created by the use of a single valve. The gravel bed was replaced by a series of fine mesh copper screens (0.007-inch diameter; 25 per inch) and linen cloth which helped to filter the water, provide uniform flow, and break up the troublesome air bubbles. Finally an 11-inch wide convergent channel constructed of 1/16-inch plexiglass was placed within the original tank. This served to reduce the flow rates required at the test section and better stabilize the flow pattern. The inner channel of the table was constructed to converge downstream from the test section to protect the test section from eddy currents caused by the square corners of the outer channel. Metal rods were positioned at various points across the top of the table to strengthen the walls against the static pressure of the water.

After completion of the water table, dye studies were made in the test channel to verify the existence of a satisfactory flow pattern.

CONSTRUCTION OF THE TEST CYLINDER

The manufacture of the test cylinder, though quite simple in design, proved to be a tedious task because of the cylinder's small size. The diameter dimension arrived at was controlled by two considerations. First, if the cylinder was too large, the flow velocities required for the desired range of Reynolds number would be extremely low and thus increase the difficulty of accurately regulating the flow. Secondly, the diameter had to be large enough to accommodate a heating element and a thermocouple. Essentially the final choice consisted of two parts: a six-inch length of 3/16-inch brass tubing as the outer cylinder surface and a small cylindrical core as the inner heating element. It was realized that copper tubing would have been more desirable because of its higher thermal conductivity; however, it was not available below a diameter of 1/4-inch.

The heating core was made by wrapping #31 gauge enamel insulated constantan wire around a six-inch length of 1/8-inch brass tubing. Before wrapping the wire a small hole was filed in the center of the core to provide a route to the outer cylinder for the thermocouple. It was found that the wire would short out quite easily after wrapping, apparently because of chipping of the enamel insulation on the wire. To combat this the 1/8-inch core was coated with a thin layer of insulating varnish and baked in an oven; the wire was then wrapped around the core with varnish being applied at the same time; after the core was wrapped, a thin layer of varnish was applied over the wire and the entire core was then baked in the oven again. This last step allowed the fit between the heating core and the outer cylinder to be a slight force fit thereby promoting better conductivity between the core and the cylinder and providing a more uniformly heated test cylinder.

A small hole was filed at the center of the 3/16-inch outer tubing for the thermocouple after which the core was placed inside. At this point thermocouple leads were threaded through the hole in the outer cylinder and into the center of the core until the free end of the leads emerged from the end of the cylinder allowing connections to be made. The thermocouple junction was soldered to the cylinder, turned in a lathe, and carefully sanded until the solder joint conformed to the cylinder dimensions.

Before further work was continued the assembly was checked for short circuits. After this precautionary step a plastic filler of low thermal conductivity was placed in both ends of the cylinder to furnish a seal against the liquid environment and minimize heating loss through the ends.

The finished cylinder was then fitted in the plexiglass yoke and assembled to the drive mechanism.

It was discovered after initial tests that the thermocouple indicated a temperature difference between the cylinder and the water of almost twice the "true" value. This discrepancy was checked by temporarily soldering another thermocouple to the outside surface of the cylinder. Apparently the junction had been partially soldered to the hotter core and thus read a higher temperature than that of the cylinder surface.

A new cylinder was reconstructed with more care given to the thermocouple junction. This latter construction gave satisfactory data and was used until it developed leaks, after which an identical construction was built to obtain the remaining data.

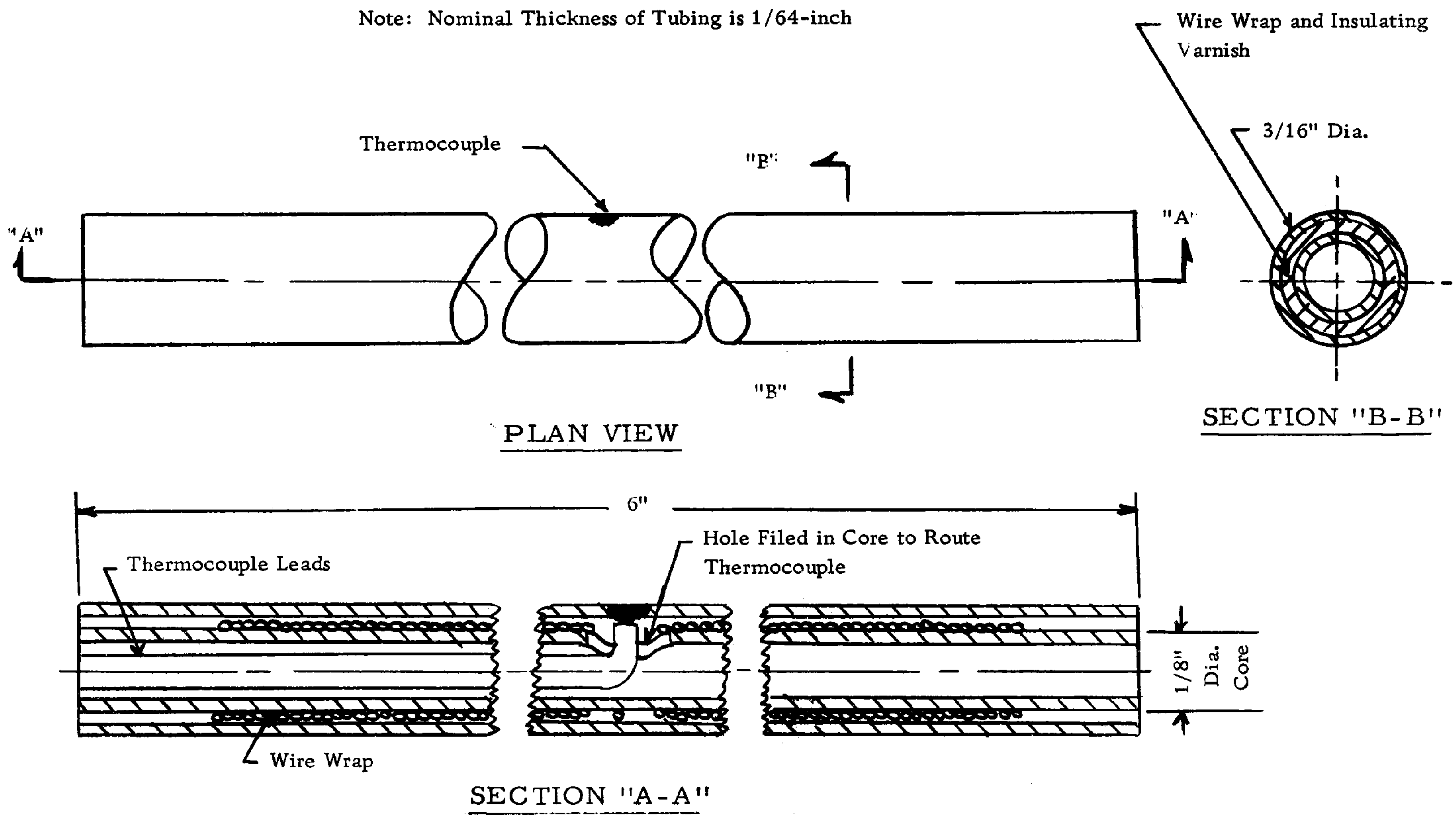

Figure 13. Detail of the Test Cylinder

TABULATION OF RESULTS

Diameter of Cylinder 3/16-inch

Data of Cylinder No. 1

For runs 1_s - 33: nominal $(N_{Re})_f$ = 103

n_v = 52 cycles/min

V = 0.0818 ft/sec

Run No.	a/d	n_c(rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
1_s	-	0	0	10.24	-	59.2
2	.248	15.4	5.68	10.2	.984	59.5
3	.248	30.8	11.36	10.2	.984	59.5
4	.248	51.8	19.1	10.2	.984	59.5
5	.248	56.0	20.6	10.2	.984	59.5
6	.248	69.0	25.5	10.2	.984	59.5
7	.248	89.7	33.1	10.15	.981	59.5
8	.248	132.0	48.8	10.1	.976	59.5
9_s	-	0	0	10.43	-	59.8
10	.501	13.1	9.72	10.43	1.01	60.3
11	.501	32.0	23.9	10.43	1.01	60.3
12	.501	44.4	33.1	10.43	1.01	60.3
13	.501	51.8	38.6	10.43	1.01	60.3
14	.501	56.1	41.8	10.43	1.01	60.3
15	.501	67.6	50.4	10.43	1.01	60.3
16	.501	96.2	71.6	10.4	1.01	60.3
17	.501	132.8	99.0	9.63	.930	60.3

Run No.	a/d	n_c (rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
18	1.03	10.7	16.5	10.16	.982	60.5
19	1.03	33.3	51.2	10.62	1.03	60.5
20	1.03	45.0	69.3	10.62	1.03	60.5
21	1.03	51.2	78.8	10.62	1.03	60.5
22	1.03	55.7	85.9	10.78	1.04	60.5
23	1.03	67.9	104.2	11.01	1.065	60.5
24	1.03	95.0	146.2	11.52	1.115	60.5
25	1.03	132.4	204	12.78	1.235	60.5
26	1.38	13.9	28.5	10.20	.987	60.5
27	1.38	29.5	60.3	10.76	1.04	60.5
28	1.38	44.0	90.0	11.52	1.115	60.5
29	1.38	50.9	103.8	11.52	1.115	60.5
30	1.38	54.7	111.8	11.52	1.115	60.5
31	1.38	68.0	139	11.83	1.143	60.5
32	1.38	97.8	200	12.85	1.242	60.5
33	1.38	132.4	271	13.75	1.33	60.5

For runs 34 - 66: nominal $(N_{Re})_f = 64$

$n_v = 24.3$ cycles/min

$V = .0464$ ft/sec

34_s	-	0	0	7.81	-	64.4
35	.230	14.0	5.05	7.81	1.00	64.3
36	.230	25.1	9.06	7.81	1.00	64.3
37	.230	32.2	11.6	7.74	.991	64.3

Run No.	a/d	n_c(rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp.°F.
38	.230	42.3	15.3	7.74	.991	64.3
39	.230	58.6	21.2	7.74	.991	64.3
40	.230	73.6	26.6	7.74	.991	64.3
41	.230	97.7	35.3	7.60	.973	64.3
42	.230	130.4	47.1	7.60	.973	64.3
43	.512	9.80	7.91	7.86	1.007	64.3
44	.512	25.0	20.2	8.08	1.032	64.3
45	.512	31.3	25.3	7.90	1.011	64.3
46	.512	41.3	33.4	7.90	1.011	64.3
47	.512	57.5	46.5	7.48	.958	64.3
48	.512	71.8	58.0	7.10	.909	64.3
49	.512	95.7	77.3	7.65	.979	64.3
50	.512	128.9	104.0	8.56	1.097	64.3
51	.966	10.6	16.2	7.74	.991	64.3
52	.966	24.0	36.5	8.26	1.058	64.3
53	.966	31.2	47.5	8.51	1.09	64.3
54	.966	41.5	63.1	8.86	1.133	64.3
55	.966	58.0	88.3	9.11	1.166	64.3
56	.966	72.6	110.8	9.55	1.222	64.3
57	.966	97.0	147.9	10.70	1.370	64.3
58	.966	129.8	197.2	11.73	1.502	64.3
59	1.496	11.8	27.8	7.90	1.011	64.5
60	1.496	23.3	54.8	8.37	1.071	64.5
61	1.496	30.7	72.3	9.06	1.159	64.5
62	1.496	42.0	99.0	9.60	1.228	64.5
63	1.496	58.4	137.6	10.75	1.374	64.5
64	1.496	73.6	173.6	11.35	1.452	64.5
65	1.496	99.0	233	12.63	1.618	64.5
66	1.496	129.8	306	13.62	1.743	64.5

Data of Cylinder No. 2

For runs 1_s - 51_s: nominal $(N_{Re})_f$ = 144

n_v = 77.5 cycles/min

V = 0.113 ft/sec

Run No.	a/d	n_c (rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
1_s	-	0	0	16.65	-	59.2
2	.113	78.0	13.0	16.8	1.018	59.2
3	.113	136.5	22.7	16.68	1.010	59.2
4	.113	178	29.7	16.50	.998	59.3
5	.113	233	38.9	16.50	.998	59.3
6	.113	306	51.1	16.0	.968	59.4
7	.113	378	63.1	15.10	.913	59.6
8_s	-	0	0	16.40	-	59.2
9_s	-	0	0	16.60	-	59.1
10	.244	45.6	16.5	16.80	1.018	59.1
11	.244	87.1	31.4	16.42	.991	59.2
12	.244	125	45.1	16.10	.973	59.3
13	.244	166	59.9	16.43	.991	59.3
14	.244	219	79.0	16.28	.984	59.3
15	.244	271	97.7	14.82	.895	59.8
16	.244	325	117	14.6	.882	59.9
17	.244	382	138	13.33	.806	60.4
18_s	-	0	0	16.90	-	59.2
19_s	-	0	0	16.80	-	59.2
20	.496	31.6	23.1	16.72	1.010	59.2
21	.496	74.9	54.8	16.84	1.020	59.2

Run No.	a/d	n_c (rpm)	$(N_{Re})_f$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
22	.496	115	84.1	16.63	1.006	59.2
23	.496	159	116.5	15.2	.919	59.6
24	.496	209	153	16.3	.985	59.2
25	.496	259	190	16.25	.982	59.3
26	.496	324	237	18.68	1.130	59.1
27	.496	383	281	21.2	1.282	59.1
28_s	-	0	0	16.4	-	59.2
29_s	-	0	0	16.5	-	59.3
30	1.028	24.5	37.0	16.6	1.004	59.3
31	1.028	63.2	95.5	17.94	1.084	59.2
32	1.028	93.0	141	18.6	1.126	59.4
33	1.028	130	196.5	19.8	1.198	59.2
34	1.028	166	251	20.8	1.258	59.3
35	1.028	207	313	22.3	1.350	59.3
36	1.028	261	395	24.0	1.452	59.2
37	1.028	324	490	26.8	1.621	59.2
38	1.028	384	580	28.1	1.701	59.4
39_s	-	0	0	16.5	-	59.3
40_s	-	0	0	16.4	-	59.6
41	1.42	28.5	59.8	16.9	1.022	59.4
42	1.42	61.5	129	18.36	1.10	59.6
43	1.42	87.6	184	19.7	1.191	59.6
44	1.42	128	269	21.6	1.307	59.5
45	1.42	169	354	23.8	1.440	59.6
46	1.42	204	428	25.6	1.549	59.7
47	1.42	256	536	28.2	1.706	59.4

Run No.	a/d	n_c(rpm)	$(N_{Re})_f$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
48	1.42	304	638	29.1	1.760	59.7
49	1.42	337	706	29.8	1.804	59.8
50	1.42	382	801	31.2	1.890	59.8
51_s	-	0	0	16.3	-	59.7

For runs 52_s - 106_s: nominal $(N_{Re})_f$ = 103

n_v = 50.5 cylces/min

V = 0.0794 ft/sec

Run No.	a/d	n_c(rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
52_s	-	0	0	12.6	-	60.8
53	.123	52.5	9.65	12.60	.979	60.8
54	.123	88.2	16.22	12.10	.940	61.0
55	.123	127.7	23.5	12.10	.940	61.0
56	.123	184	33.8	11.9	.925	60.8
57	.123	245	45.1	11.9	.925	60.8
58	.123	328	60.4	11.62	.902	60.7
59	.123	382	70.4	11.4	.885	60.7
60_s	-	0	0	12.75	-	60.7
61_s	-	0	0	12.6	-	60.7
62	.256	26.5	10.2	12.6	.979	60.7
63	.256	61.2	23.6	12.2	.947	60.7

Run No.	a/d	n_c (rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
64	.256	92.8	35.8	12.0	.931	60.8
65	.256	132.1	51.0	11.5	.893	60.8
66	.256	175	67.5	11.1	.861	60.7
67	.256	214	82.5	10.9	.846	60.6
68	.256	263	101.5	10.25	.796	60.7
69	.256	323	124.5	10.0	.776	60.7
70	.256	387	149	9.90	.770	60.8
71_s	-	0	0	12.85	-	61.0
72_s	-	0	0	12.85	-	60.9
73	.501	27.3	20.6	12.85	.998	60.9
74	.501	63.1	47.6	12.85	.998	60.9
75	.501	91.0	68.5	12.75	.990	60.9
76	.501	135.2	102	11.7	.909	61.3
77	.501	184	138.5	12.6	.979	60.9
78	.501	224	168.5	13.5	1.049	60.9
79	.501	279	210	15.5	1.203	60.6
80	.501	332	250	17.05	1.322	60.6
81	.501	382	288	18.9	1.469	60.5
82_s	-	0	0	12.63	-	60.9
83_s	-	0	0	13.04	-	60.9
84	1.012	24.1	36.7	13.1	1.018	60.9
85	1.012	68.8	104.8	14.55	1.130	60.7
86	1.012	91.1	139	15.2	1.18	60.9
87	1.012	133.2	203	16.5	1.281	60.5
88	1.012	173	264	18.7	1.452	60.5
89	1.012	211	321	19.6	1.522	60.7
90	1.012	251	382	21.5	1.671	60.7

Run No.	a/d	n_c (rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
91	1.012	299	455	23.6	1.830	60.7
92	1.012	348	530	23.7	1.841	60.8
93	1.012	385	586	24.8	1.925	60.8
94_s	-	0	0	12.95	-	60.9
95_s	-	0	0	13.48	-	60.6
96	1.430	20.0	43.0	14.1	1.094	60.5
97	1.43	51.2	110	15.75	1.222	60.4
98	1.43	95.7	206	18.08	1.402	60.4
99	1.43	130	280	19.85	1.540	60.6
100	1.43	167	358	22.4	1.740	60.6
101	1.43	209	449	23.5	1.825	60.6
102	1.43	255	549	25.6	1.989	60.5
103	1.43	305	655	27.0	2.10	60.6
104	1.43	338	726	27.4	2.126	60.8
105	1.43	377	810	29.2	2.26	60.7
106_s	-	0	0	13.10	-	60.7

For runs 107_s - 160_s: nominal $(N_{Re})_f$ = 64

n_v = 26.2 cycles/min

V = 0.0495 ft/sec

Run No.	a/d	n_c (rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp. °F.
107_s	-	0	0	9.96	-	60.9
108	.128	32.2	6.19	9.96	.993	60.9
109	.128	72.2	13.9	9.79	.975	61.0
110	.128	103.8	19.9	9.90	.986	60.9
111	.128	158	30.3	9.44	.940	61.1
112	.128	203	39.0	9.25	.922	61.2
113	.128	261	50.1	9.25	.922	61.2
114	.128	318	61.1	9.09	.906	60.9
115	.128	372	71.5	8.92	.889	61.0
116_s	-	0	0	9.90	-	60.9
117_s	-	0	0	10.02	-	60.8
118	.256	36.3	14.0	10.02	1.00	60.8
119	.256	73.0	28.1	9.66	.962	60.6
120	.256	112.8	43.4	9.16	.913	60.7
121	.256	169	65.0	8.42	.839	60.7
122	.256	216	83.1	8.10	.807	60.7
123	.256	267	103	8.49	.846	60.7
124	.256	328	126.2	9.60	.956	60.7
125	.256	378	145.5	9.79	.975	60.6
126_s	-	0	0	10.1	-	60.5
127_s	-	0	0	10.1	-	60.5
128	.496	30.9	22.9	10.52	1.050	60.5
129	.496	71.0	52.5	9.35	.932	60.3

Run No.	a/d	n_c(rpm)	$(N_{Re})_v$	N_{Nu}	$(N_{Nu})_v/(N_{Nu})_s$	Mean Temp.°F.
130	.496	118.1	87.4	10.02	1.00	60.7
131	.496	170	126	11.97	1.191	60.4
132	.496	224	165.6	14.38	1.432	60.2
133	.496	284	210	15.88	1.580	60.0
134	.496	338	250	18.28	1.821	59.7
135	.496	380	281	19.42	1.938	59.9
136_s	-	0	0	10.00	-	60.0
137_s	-	0	0	10.1	-	59.8
138	1.016	22.0	33.0	10.85	1.18	59.4
139	1.016	54.5	81.8	12.55	1.250	59.3
140	1.016	86.5	129	14.1	1.405	59.3
141	1.016	127.2	191	16.35	1.630	59.5
142	1.016	172	258	18.0	1.795	59.7
143	1.016	211	316	19.3	1.923	59.7
144	1.016	260	390	20.6	2.06	59.6
145	1.016	304	456	22.1	2.20	59.4
146	1.016	340	510	23.3	2.32	59.6
147	1.016	383	575	24.3	2.43	59.6
148_s	-	0	0	10.02	-	59.6
149_s	-	0	0	10.1	-	59.5
150	1.45	21.2	45.5	11.20	1.117	59.4
151	1.45	65.3	140	13.62	1.358	59.5
152	1.45	127.5	274	17.7	1.765	59.2
153	1.45	91.3	196	16.50	1.642	59.5
154	1.45	166	356	20.9	2.08	59.0
155	1.45	207	443	22.6	2.25	59.2
156	1.45	248	531	23.8	2.37	59.6
157	1.45	293	628	26.0	2.59	59.4
158	1.45	336	720	26.5	2.64	59.6
159	1.45	380	815	28.2	2.81	59.6
160_s	-	0	0	10.00	-	59.6