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EFFECT OF VIBRATION ON HEAT TRANSFER FROM
CYLINDERS VIBRATED SINUSOIDALLY WITHIN
A VERTICAL PLANE IN FREE CONVECTION

THESIS

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EFFECT OF VIBRATION ON HEAT TRANSFER FROM CYLINDERS VIBRATED SINUSOIDALLY WITHIN A VERTICAL PLANE IN FREE CONVECTION

THESIS

Presented to the Faculty of the School of Engineering of

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Master of Science

By

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Preface

This report is concerned with heat transfer rates from a vibrating cylinder in free convection in air. It has not only been an interesting experience in heat transfer, but also a long, laboring experience in vibration techniques for obtaining higher and higher vibration intensities from a cylinder when excited at an end. The higher vibration intensities were possible but they were always in three degrees of motion, which for the purposes of this investigation was not applicable.

I wish to express my appreciation to those members of the Mechanical Engineering Department who graciously gave their time and energy. I am especially grateful to Dr. Andrew J. Shine for suggesting this thesis topic. As my thesis advisor, Dr. Shine gave timely suggestions that were invaluable. Mr. Flahive and Mr. Brown assisted me with many practical problems associated with the test equipment.

I wish to thank the Electrical Engineering Department for the loan of equipment and calibration of the electrical instruments used in the experiment.

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List of Symbols

A	Lateral Surface Area of Test Cylinder	ft^2
a	Amplitude of Vibration	in
d	Diameter of Test Cylinder	ft
E	Voltage Drop Across the Test Cylinder	volts
f	Frequency of Vibration	cycles/sec
Gr	Grashof Number	dimensionless
h	Heat Transfer Coefficient	BTU/hr ft^2 F
I	Current Through Test Cylinder	amps
k	Thermal Conductivity of Air at T_f	BTU/hr ft F
L	Length of Test Cylinder	ft
Nu	Nusselt Number	dimensionless
Pr	Prandtl Number	dimensionless
Q	Total Heat Input to Test Cylinder	watts
Re	Reynolds Number $\frac{2afd\rho}{\mu}$	dimensionless
T_a	Temperature of Ambient Environment	F
T_f	Temperature of Boundary Layer Fluid $\frac{T_a + T_w}{2}$	F
T_w	Temperature of Test Cylinder Surface	F
ΔT	Temperature Difference $(T_w - T_a)$	F
ΔT_o	Static Temperature Difference	F
V	Vibration Velocity (2af)	in/sec
v	Vibration Intensity, af	in/sec
μ	Absolute Viscosity	lbm/hr ft
ρ	Density of Air at T_f	lbm/ft^3

Abstract

Methods to increase the heat transfer rate from a body to its surroundings include the effects of vibration. This study is an investigation of the effect of sinusoidal vibration in a vertical plane on the heat transfer rate from a cylinder in free convection in air.

Three different cylinders with diameters 0.072 in, 0.120 in, and 0.25 in were vibrated at amplitudes from 0 to 1.5 in, at frequencies from 16 to 80 cps, and at surface temperatures of 100 F and 200 F above room temperature.

The heat transfer rate changes from that obtained in a free convection process to that obtained in a forced convection process at the conventional forced convection curve of McAdams and generally parallels this curve to Reynolds numbers as high as 1100 which was the highest obtained. For a given Reynolds number, the heat transfer rate is independent of the cylinder temperature and test position along the length of the cylinder. The results of this investigation agree closely with results previously presented for cylinders vibrated in free convection in air, regardless of the mode or method of vibration.

The scope of the investigation was limited by vibrating characteristics of the cylinders, failure of the cylinders at higher temperatures, and the limited mass of the mounting assembly for the cylinders.

EFFECT OF VIBRATION ON HEAT TRANSFER FROM CYLINDERS VIBRATED SINUSOIDALLY WITHIN A VERTICAL PLANE IN FREE CONVECTION

I. Introduction

Background

Free convection is conduction of heat from a body to its boundary layer with subsequent fluid movement caused by changes in fluid density in the boundary layer. The boundary layer is an effective barrier to the transfer of heat from a body to its surroundings; and to increase the rate of heat transfer, the boundary layer must be altered. Common techniques employed to alter the boundary layer are to reduce the thickness of the boundary layer, increase transverse fluid motion in the boundary layer, or both. Vibration is one method of altering the boundary layer by mechanically inducing turbulence in the boundary layer and by making the boundary layer thinner. This is illustrated in the Schlieren photographs in Figure 13.

Within the past twenty years, a number of investigations have been performed to study the effect of various methods of vibration on the heat transfer rate from flat plates and cylinders in free convection in air. The following conclusions were made from these investigations:

1. The heat transfer rate increases with vibration beyond a critical vibration intensity.

2. The effect of vibration is independent of temperature in the range of temperature tested.
3. The heat transfer rate is independent of the mode of vibration, and is only dependent on the intensity of vibration (Ref 6:3).

Knowledge of heat transfer rates as a function of vibration intensity has possible applications in the design of equipments which contain components operating at elevated temperatures and vibrating at known intensities.

In 1962, Shine and Jarvis investigated the effects of vibration on the heat transfer rate from cylinders using a different method of vibrating the cylinders than had been previously used. They mounted the test cylinders horizontally at the ends and excited the cylinders at a point near one end. An investigation of the effect of vertical vibration on the heat transfer rate from cylinders vibrated in a sinusoidal wave form in air was made. The cylinder diameters were 0.032 in and 0.072 in. The frequency range was from 15 to 75 cycles per second and the amplitude range was from 0.002 to 0.99 in. They reported that the variation of the heat transfer coefficient with vibration intensity generally paralleled the forced convection curve recommended by McAdams in the range of vibration intensities tested (Ref 8:2).

This investigation was an extension of the work of Shine and Jarvis to study the effects at higher Reynolds numbers, at two cylinder surface temperatures, and at different test positions along the cylinder. Reynolds number as used throughout this report to express vibration intensity is computed from $2afd\rho/\mu$. The vibration intensity for a given cylinder diameter is defined as 2af which is in fact the average

velocity of the cylinder motion. Therefore, the vibration intensity (2af) and Reynolds number for a vibrating cylinder are related because the vibration intensity is used in computing the Reynolds number.

Purpose

The purpose of this investigation was to determine the effect of vibration on the heat transfer rate from a heated, horizontally-mounted cylinder vibrated in a sinusoidal wave form in a vertical plane in free convection. The specific objectives of this investigation were to

1. Investigate the effect at Reynolds numbers higher than previously obtained and correlate the results with the conventional forced convection curve of McAdams (Ref 5:275).
2. Determine the effect of the location of the test position along the cylinder during vibration in a sinusoidal wave form.
3. Determine the effect of cylinder temperature.
4. Determine the effect of cylinder diameter.

Scope

This investigation was concerned with heat transfer from cylinder diameters of 0.072 in, 0.120 in, and 0.25 in.

The cylinders were vibrated over a frequency range from 16 to 80 cycles per second and an amplitude range from 0 to 1.5 in. The 0.072 in cylinder was tested at a surface temperature 100 F above room temperature and the 0.120 in and 0.25 in cylinders were tested at 100 F and 200 F above room temperature. The heat transfer rates were determined at different positions along the cylinder while it was vibrated in a sinusoidal wave form. Schlieren photographs of the boundary layer were

taken to correlate the behavior of the boundary layer with heat transfer data in the vicinity of the critical Reynolds number.

Approach

The cylinder was vibrated at a frequency and amplitude while the surface temperature of the cylinder, which was electrically heated, was maintained at a predetermined constant temperature above room temperature. The electrical power input to the cylinder provided a measure of the heat flux from the surface of the cylinder. Nusselt numbers were computed from the power input data and Reynolds numbers were computed from the vibration intensity data. These dimensionless parameters were plotted to present the results in graphical form for correlation with previous investigations.

II. Experimental Equipment

Test Cylinders

The test cylinders were stainless steel tubing of three different diameters. The lengths of the cylinders were also different because the vibration responses differed, and the lengths were chosen to obtain maximum amplitude response. They are listed below:

Body	Surface	Diameter	Length	Wall Thickness
Stainless Steel	Polished	0.072 in O.d	25.5 in	0.020 in
Stainless Steel	Polished	0.120 in O.d	37.47 in	0.020 in
Stainless Steel	Gray-oxidized	0.250 in O.d	39.17 in	0.020 in

A movable, iron-constantan thermocouple was threaded through the opening of the cylinder with one lead extending out of each end. Copper, electrical connectors were fitted tightly to the extreme ends of the cylinders to provide connections for electrical power input and voltage pick offs (see Figure 3). The cylinders were mounted at points immediately to the inside of the connectors, and the vibration input shaft from the vibrator to the cylinder was fastened to the cylinders at a point 6 in from the left-end mounting (see Figure 3).

Mounting Assembly

The mounting assembly for the test cylinders consisted of two vertical posts fastened to a horizontal beam which was supported at each end (see Figure 3). The space above the mounting assembly including the test cylinder was enclosed with a rectangular hood which

measured 48 × 12 × 30 in. The bottom and top of the hood were vented to allow free convection around the cylinder while protecting the cylinder from random convective currents in the room (see Figure 4). An adjustable weight was attached to the right end of the test cylinder through a pulley arrangement to compensate for thermal expansion and thus maintain a constant tension on the cylinder.

Vibrator

The source of energy for vibrating the cylinder was an electromagnetic exciter having a variable output of up to 25 pound force. The exciter was powered by a power supply which contained an audio frequency oscillator and a variable power output unit. Frequency of vibration was controlled by the oscillator, and the amplitude of vibration was controlled by the variable power output rheostat. The exciter was positioned directly below the test cylinder, and was connected to the cylinder by a 3/8 in diameter, 12 in long aluminum drive rod (see Figure 3). The rod constrained the cylinder to vibrate within a vertical plane in a sinusoidal wave form.

Heat Power Source

Direct current electrical power for heating the test cylinder was taken from a 28 v, 540 amp rectifier which serves as the central source of D.C. power for the M. E. Laboratories at AFIT. The output voltage was adjustable from 0 to 50 volts, and to provide finer adjustments in power through the test cylinder, four 5.8 ohm, 10 amp, variable resistors were connected in parallel and placed as a unit in series with the power circuit to the test cylinder.

Measurement Equipment

Electrical power to heat the cylinders was measured on a Weston Electric model 931 ammeter and voltmeter.

Frequency of vibration of the cylinder was measured with a strobotac model number 631-B manufactured by General Radio Company.

Amplitude of vibration was measured from photographs taken of the vibrating cylinder motion using a light source, mirrors, and a camera. The arrangement of these equipments is shown in the schematic drawing of Figure 1.

The source of light was a mercury lamp unit manufactured by the George W. Gates Company. The lamp was hooded and its light passed through a condensing lens onto a 1/32 in pin hole in the forward end of the hood. This provided a point source of light which was passed onto a series of mirrors, across the cylinder, and onto a photographic film.

Two 5.5 in diameter, 35.5 in focal length, parabolic mirrors and a 2 in diameter flat mirror mounted on the rotating shaft of a small D.C. electric motor were used to direct light across the cylinder and onto the photographic film. The first parabolic mirror was positioned so that it directed the light across the test cylinder in a parallel light beam. The second parabolic mirror was positioned on the opposite side of the cylinder from the first mirror. It received the light from the first mirror and passed it onto the 2 in rotating flat mirror which was located at the focal point of the second parabolic mirror. The rotating mirror reflected the light in a sweeping motion across the aperture to the camera (see Figure 5).

A Polaroid-back camera containing type 47 Polaroid film was used to photograph the motion of the cylinder. The camera was mounted on the aft end of a special plywood housing with an extendable bellows section which contained the aperture to the camera, and which prevented stray light from entering the camera.

Temperatures were measured with iron-constantan thermocouples and a Bristol's Dynamaster Recorder potentiometer. Iron and constantan wire with varnished fiberglass sheathing was used in constructing the thermocouples. The junction was formed by spot welding the ends together. The thermocouple used inside the test cylinder was further sheathed in teflon spaghetti except for the junction which protruded through a hole in the spaghetti. The thermocouples and recorder as a combination were calibrated against an accurate mercury-in-glass thermometer using an ice bath, ambient air, and boiling water as references.

Boundary Layer

The boundary layer was photographed using Schlieren photography, and the same equipment used to photograph the cylinder motion was used with the following exceptions. The light source was a locally constructed condenser discharge spark unit and an ionizable-gas-filled lamp. The lamp was housed in the same hood used with the mercury lamp and was adjusted within the hood in the same way as the mercury light. The hole in front of the light hood served as the first knife edge and an adjustable knife edge was used at the focal point of the second parabolic mirror.

III. Measurements

The raw data required in the computation of the dimensionless parameters, which are used to present graphically the results of this investigation, were the variable quantities of ambient air temperature, cylinder surface temperature, frequency of vibration, amplitude of vibration, electrical current and voltage, and barometric pressure. The measurement of these variables is discussed in the following paragraphs. An analysis of errors in variables is presented in Appendix D.

Temperature

The two temperatures--ambient air and cylinder surface--were measured to the nearest 1/2 F with an iron-constantan thermocouple connected to an automatic recording potentiometer. The ambient temperature was measured at a point level with the test cylinder and 6 in away in a direction perpendicular to the plane of vibration. At this distance the thermocouple was outside the effect of the temperature of the cylinder. Ambient temperature measurements were recorded once each minute during a test run.

The surface temperature at any test point along the cylinder was measured from the inside surface of the cylinder with a movable iron-constantan thermocouple. Since the entire thermocouple was within the cylinder, it did not affect the boundary layer. The thermocouple was connected to 15 consecutive channels of the recording potentiometer which recorded each 4 seconds giving an essentially continuous indication of the temperature to assist in adjusting the power input. During test runs, the ends of the cylinder were closed with a pliable material to

prevent heat losses due to air movement through the inside of the cylinder. The temperature of the cylinder was measured at the same point along the cylinder as the amplitude.

Vibration

The frequency of vibration was set on the dial of the audio frequency oscillator, but a strobotac was used to measure the frequency of vibration on the cylinder because the scale of the strobotac was marked in smaller graduations than the dial of the oscillator.

The amplitude of vibration was measured from the photograph taken of the cylinder motion at the time the temperature was recorded. Preliminary to data runs for each cylinder installation, a photograph was made of the stationary test cylinder. This photograph was used to determine the scale factor for that specific cylinder and optics adjustment by comparing the width of the cylinder on the photograph to the actual diameter of the cylinder (see Figure 6). This factor was used for subsequent test runs to compute the true amplitudes of vibration. Amplitudes were measured from the photographs with an optical measuring instrument which was graduated to 0.0001 in.

Electrical Heating Power

The electrical power for heating the test cylinder was adjusted by varying the resistors while monitoring the cylinder temperature readings. When the power was finally adjusted to give the desired cylinder temperature, the power was obtained from the current and voltage readings using the formula $P = EI$.

IV. Experimental Procedure

Testing with Cylinder Stationary

With the cylinder stationary and the optical equipment adjusted, the cylinder was heated to a predetermined surface temperature, and a photograph was taken of the stationary cylinder for determining the scale factor. The temperatures and electrical power measurements were recorded for determining the free convection heat transfer rate.

Testing with Cylinder Vibrating

The cylinder was vibrated at a selected frequency and amplitude while the surface temperature of the cylinder was maintained at the predetermined value by adjusting the electrical power input. Since the surface temperature of the cylinder responded quickly to changes in power, the temperature usually stabilized quickly as was observed on the recorder. When the temperature was stabilized, the measurements of temperature, electrical power, frequency and amplitude of vibration, and barometric pressure were recorded.

Visualization of Boundary Layer

Special test runs using the 0.25 in diameter cylinder heated to 200 F above room temperature were performed at vibration intensities in the vicinity of the critical Reynolds number to photograph the behavior of the boundary layer. The same procedure as in stationary and vibratory testing was followed with one exception--the Schlieren optical system was used. When the desired image was obtained, the test was conducted as usual, and a photograph of the boundary layer was taken at the time the variables were recorded.

V. Results

The results of this investigation are presented in both tabular and graphical form. Measured variables and calculated parameters for all test runs are contained in Tables I through III in Appendix E. The calculated parameters are presented graphically in Figures 7 through 9.

There exists for each cylinder diameter a critical Reynolds number. Vibration intensities below this Reynolds number give heat transfer rates expressed as Nusselt numbers that are obtained in a free convection process. Vibration intensities above this Reynolds number give Nusselt numbers that are obtained in a forced convection process. The Nusselt numbers corresponding to Reynolds numbers above the critical Reynolds number were generally along the forced convection curve of McAdams to a Reynolds number of 1100 which was the maximum obtained in this investigation. The critical Reynolds number was either on or near the conventional forced convection curve.

The heat transfer rates expressed in Nusselt numbers at a given Reynolds number were not different for a cylinder surface temperature of 100 F and 200 F above room temperature.

The heat transfer rates expressed as Nusselt numbers showed no consistent trend for the effect of cylinder diameter. The rates were not independent of cylinder diameter, nor did they consistently increase or decrease with cylinder diameter. Rather, the results indicate that the rate increases with cylinder diameter up to a certain diameter and then decrease with larger cylinder diameters.

The heat transfer rate is independent of the test position along

the cylinder, and is only dependent upon the vibration intensity at the test point.

The boundary layer gradually became thinner and showed an increase in turbulence above the cylinder as the vibration intensity was increased.

VI. Discussion of Results

The variation of the heat transfer rate with vibration intensity is presented for each cylinder in the non-dimensional forms $\frac{Nu}{Pr^{.3}}$ and Re in Figures 7, 8, and 9. Vibration intensity is expressed in Reynolds numbers, and the highest Reynolds number shown for each cylinder was the maximum possible with the test apparatus used.

Since the primary objective of this investigation was to test to the highest possible Reynolds numbers, most of the data were obtained with the cylinders vibrated in the fundamental mode, for this mode produced the highest Reynolds numbers. Some data were obtained with the two smaller cylinders vibrating in their second and third harmonics. The 0.25 in. diameter cylinder could be vibrated only in the fundamental mode. The heat transfer rate from each individual cylinder was independent of all modes, positions along the cylinder, and cylinder temperatures. It depended only upon the vibration intensity but showed some variation between the different diameter cylinders. The broken lines on the graphs show the limits of scatter in the data from which the conventional forced convection curve of McAdams is derived. Most of the data of this investigation fall within the limits of scatter shown by these lines.

The critical Reynolds number for each cylinder falls either on the conventional forced convection curve or near it. Beyond the critical Reynolds number, the heat transfer rate increases with Reynolds number and generally parallels the forced convection curve to a Reynolds number of at least 1100 which was the maximum obtained.

An analysis of the data for the three cylinders shows no consistent

trend of the effect of cylinder diameter on the heat transfer rate expressed as a Nusselt number. The Nusselt number at a given Reynolds number for the 0.120 in diameter cylinder was significantly larger than for the 0.072 in diameter cylinder, but the Nusselt number for the 0.25 in diameter cylinder was slightly less than for the 0.120 in cylinder.

Comparison with Other Authors

A graphical comparison with the results of other authors who investigated heat transfer from vibrating cylinders in free convection is presented in the non-dimensional forms $\frac{Nu}{Pr^{.3}}$ and Re in Figure 12. An average curve is shown for the data of this investigation for purposes of comparison, and a discussion is presented in the following paragraphs comparing this curve to the data curve of other authors.

McAdams. The data for the 0.072 in diameter cylinder show a heat transfer rate which is an average of 8.6% below the conventional forced convection curve but generally parallel to the curve. Transition from the free convection heat transfer rate to the forced rate occurs at a Reynolds number of 25; whereas, the transition on the conventional forced convection curve occurs at a Reynolds number of 7.

The data for the 0.120 in diameter cylinder are within a few percent of the conventional forced convection curve and generally parallels it. The Reynolds number at which the heat transfer rate changes from a free convection rate to a forced convection rate appears to coincide with the Reynolds number at which the free convection rate crosses the conventional forced convection curve.

The data for the 0.25 in diameter cylinder fall along the conventional forced convection curve except at Reynolds numbers near 1000

where the heat transfer rates are approximately 10% higher than the conventional curve. The heat transfer rate changes from that obtained in a free convection process to that obtained in a forced convection process at a Reynolds number that nearly coincides with the Reynolds number at which the free convection rate crosses the conventional forced convection curve.

The data as a whole fall within a band which is approximately 10% either side of the conventional forced convection curve of McAdams, and which is generally parallel to this forced convection curve.

Shine and Jarvis. A comparison of the data for the 0.072 in diameter cylinders shows the heat transfer rates for the cylinder of this investigation to be 26% higher than the rates obtained by Shine and Jarvis, though the slopes of the two curves are identical. A similar difference occurred between the results of Neely and the results of Shine and Jarvis. As did Neely, this author examined the data, equations, and computations of Shine and Jarvis but was unable to find an explanation for the difference.

Russ and Neely. Since Russ and Neely both used similar test equipments and Neely's investigation was in extension of the investigation performed by Russ, a comparison is made jointly with the results of these authors. They vibrated the entire length of the cylinder uniformly and transverse to the length. Russ obtained a maximum Reynolds number of 152 using cylinders of diameters 0.085 in, 0.25 in, and 0.75 in which were heated from 125 to 167 F. Neely obtained a maximum Reynolds number of 896 using cylinders of diameters 0.085 in and 0.25 in which were heated from 60 to 120 F above room temperature (Ref 6:6).

The only difference between the data of Russ and Neely and the

data of this investigation is in the range of Reynolds numbers from 0 to 100. The data of this investigation show that the heat transfer rate changes from that obtained in a free convection process to that obtained in a forced convection process at the conventional forced convection curve and generally parallels this curve; whereas, the heat transfer rate from the investigations of Russ and Neely changes at a 60% higher Reynolds number, and the transition is abrupt. Also, in this range of Reynolds numbers, the heat transfer rates of Russ and Neely are as much as 50% lower than the rates of this investigation. Russ and Neely computed the free convection heat transfer rates indirectly; whereas, the rates were directly measured in this investigation. Further, Schlieren photographs taken of the boundary layer in the vicinity of the critical Reynolds number show a gradual transition in the boundary layer which correlates with the data of this investigation (see Figure 13).

Above a Reynolds number of 100 the heat transfer rates are identical for the 0.085 in and 0.25 in diameter cylinders of Neely compared to the 0.120 in and 0.25 in diameter cylinders respectively of this investigation (see Figures 10 and 11).

Fand and Kaye. Fand and Kaye used a 0.875 in diameter cylinder for which they obtained Reynolds numbers from 955 to 2190 with the surface temperatures ranging from 25 to 185 F above room temperature (Ref 6:5). The range of Reynolds numbers obtained during the investigation by Fand and Kaye began approximately at the maximum value of Reynolds number obtained during this investigation. Though the heat transfer rates at this Reynolds number differ between the two curves by 16%, the slopes of the curves are parallel to the conventional forced convection curve.

The 16% difference between the curves possibly indicates an effect of cylinder diameter on the heat transfer rate. The data from this investigation show the rate increasing from the 0.072 in diameter cylinder to the 0.120 in diameter cylinder, and then decreasing slightly between the 0.120 in diameter cylinder and the 0.25 in diameter cylinder. The results for the 0.875 in diameter cylinder presented by Fand and Kaye show a further decrease in the heat transfer rate between the 0.25 in diameter cylinder and the 0.875 in diameter cylinder. Therefore, the results of this investigation in conjunction with the results of Fand and Kaye point to a possible trend of the effect of cylinder diameter on the heat transfer rate. This trend would indicate that there is a certain diameter cylinder between 0.072 in and 0.25 in diameter that gives a maximum heat transfer rate for all Reynolds numbers.

<u>Overall Correlation with Other Authors</u>. This investigation 1) provides what may be a more accurate behavior of the heat transfer rate in the vicinity of the critical Reynolds number, 2) substantiates the heat transfer rates in the range of Reynolds numbers from 100 to 1000, and 3) further provides substantiation of previous results above a Reynolds number of 1000 because of close correlation with these previous results. Therefore, the results of this investigation in conjunction with the results of Russ, Neely, Shine and Jarvis, and Fand and Kaye show that the heat transfer rate from a cylinder of diameter from 0.072 to 0.875 in vibrated in air, in any way transverse to the length, at any temperature up to approximately 200 F above ambient temperature will change from being a free convection process to a forced convection process and follow the well-established forced convection curve within ±10% up to a Reynolds number of at least 2190.

Accuracy of Results

Because the magnitude of the heat transfer rates and vibration intensities were generally much larger than the errors associated with the variables, the errors were relatively insignificant. Certainly, the errors incurred in the experiment were no more significant in calculating the dimensionless parameters than the errors in reading the slide rule.

Errors which might have been caused by room convective air currents and conductive heat losses from the cylinder were avoided by enclosing the test assembly in a suitable hood and by insulating the cylinder from all objects except the copper electrical power connections. The cylinder was uniformly heated and the longitudinal temperature gradient was nearly zero along the length of the cylinder within 4 in either side of the test point.

The maximum error which occurred in the Nusselt number based on all combined errors was 3.7% for the 0.072 in cylinder at the lowest vibration intensity. Likewise the maximum error in the corresponding Reynolds number was 2.0%. The errors decrease with increasing vibration intensities, cylinder diameter, and temperature difference. A detailed discussion of errors is presented in Appendix D.

Test Limitations

The scope of this investigation was limited by cylinder vibration characteristics, cylinder failure at higher temperatures, and the capability of the mounting assembly to support a more massive cylinder.

Higher vibration intensities within a vertical plane were not obtained because the cylinder motion changed from two-dimensional to

three-dimensional beyond a critical intensity. Since this investigation was restricted to motion in a vertical plane, the vibration intensities achieved were limited by the intensities at which motion of the cylinder left the vertical plane.

Tests to attain high Reynolds numbers with cylinder surface temperatures above 290 F were not possible because of failures of the 0.25 in cylinder. The failures occurred too frequently at higher temperatures to continue testing.

The mass of the mounting assembly was not adequate to support a larger diameter test cylinder than the 0.25 in cylinder. High intensity vibration of the 0.25 in cylinder excited the mounting assembly which in turn caused the motion of the cylinder to be too unsteady to allow accurate data to be recorded.

VII. Conclusions

Conclusions drawn from the results of this investigation are:

1. The heat transfer rate (expressed in a Nusselt number) from a vibrating cylinder in air changes from that obtained in a free convection process to that obtained in a forced convection process at a critical Reynolds number. This Reynolds number increases with cylinder diameter and occurs at or near the conventional forced convection curve of McAdams, and beyond this Reynolds number, the heat transfer rate will follow the conventional forced convection curve within approximately ±10% up to a Reynolds number of at least 1100. Further, based on close correlation with the results of Fand and Kaye, the heat transfer rate can be expected to follow the conventional forced convection curve up to a Reynolds number of at least 2190.
2. The heat transfer rates expressed in Nusselt numbers at a given Reynolds number were not different for a cylinder temperature of 100 F and 200 F above room temperature.
3. The heat transfer rate is independent of the position of the test point along the cylinder. It is only dependent upon the vibration intensity at the point.
4. No general effect of cylinder diameter on the heat transfer rate was determined by this investigation; however, in conjunction with the results of Fand and Kaye, there appears to be a trend for the rate to increase up to a certain cylinder diameter and then decrease with larger diameter cylinders.

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Appendix A

Figures

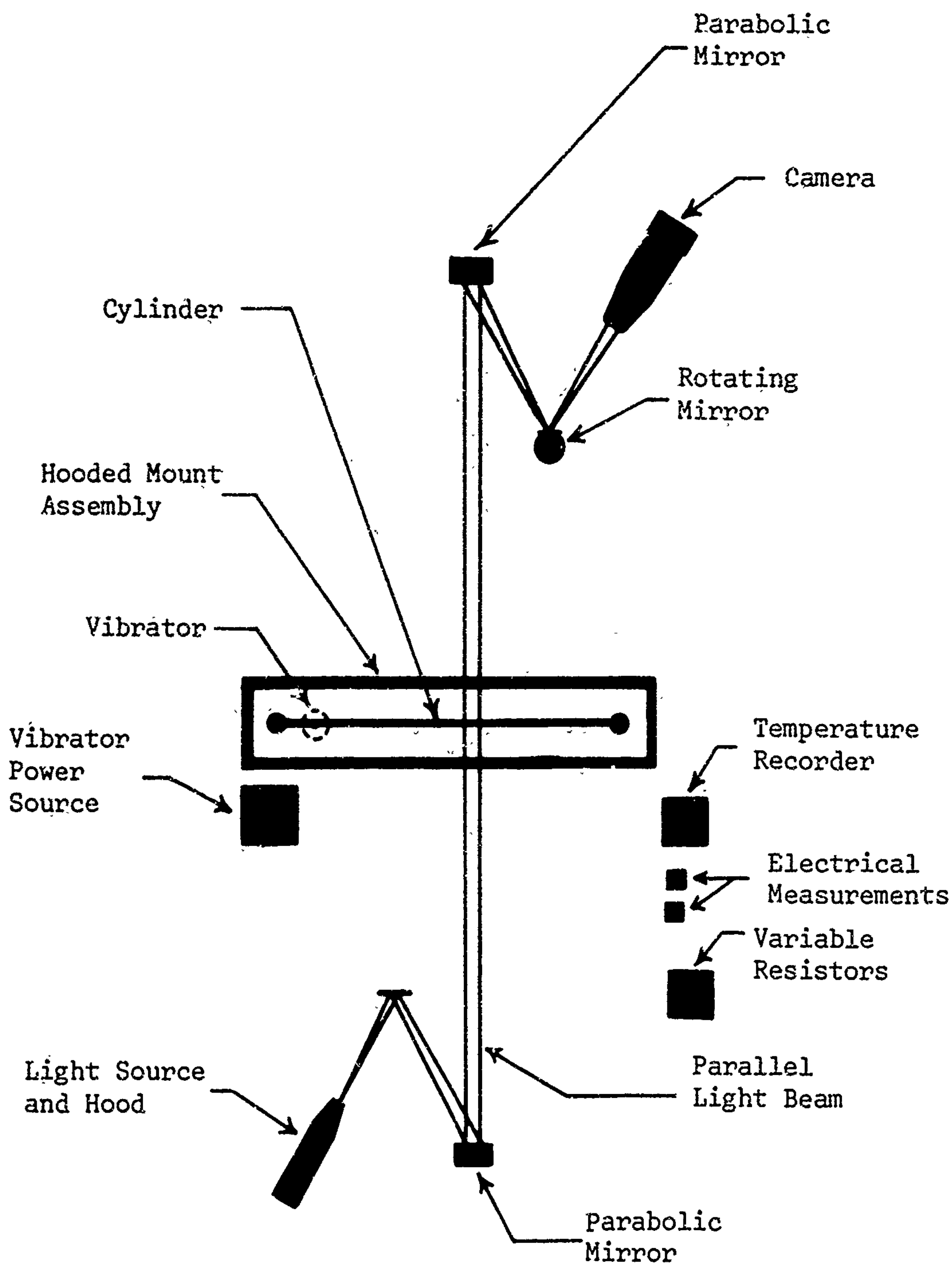

Figure 1

Schematic Layout of Experimental Equipment

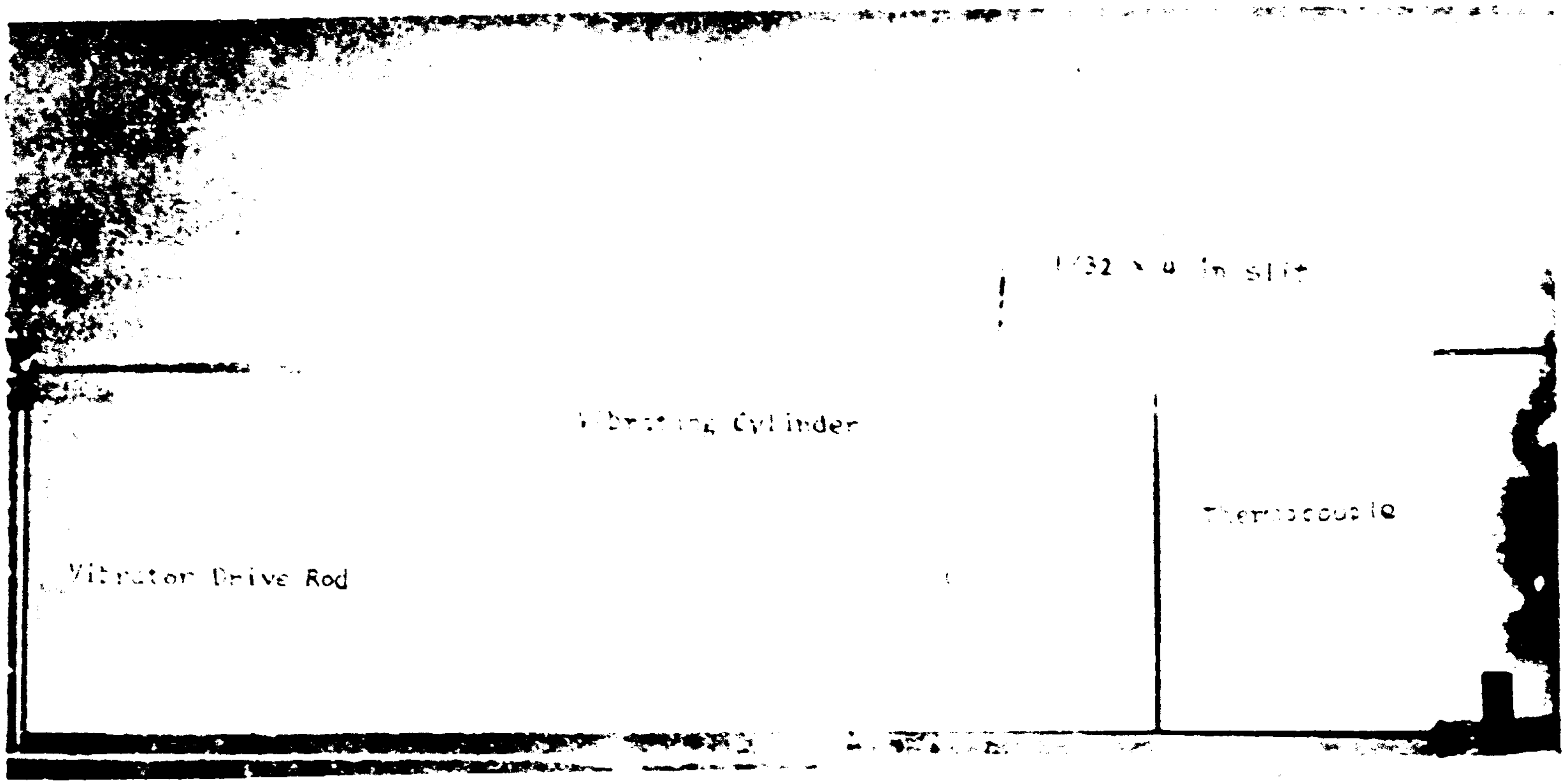

Figure 2

Photograph of Test Cylinder Mounted and Vibrating in Sinusoidal Wave Form

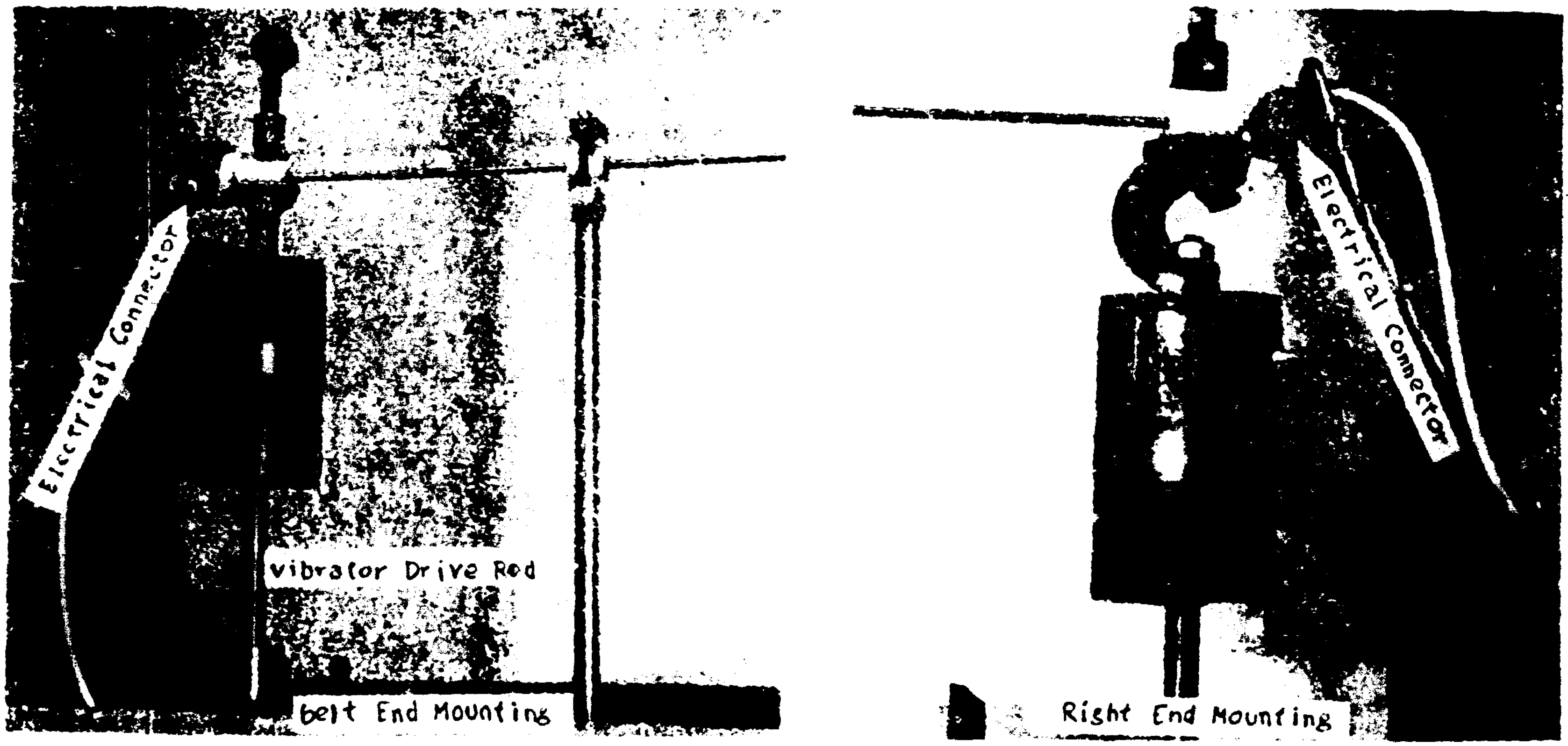

Figure 3

Photographs of Left End and Right End Mountings of the Cylinder

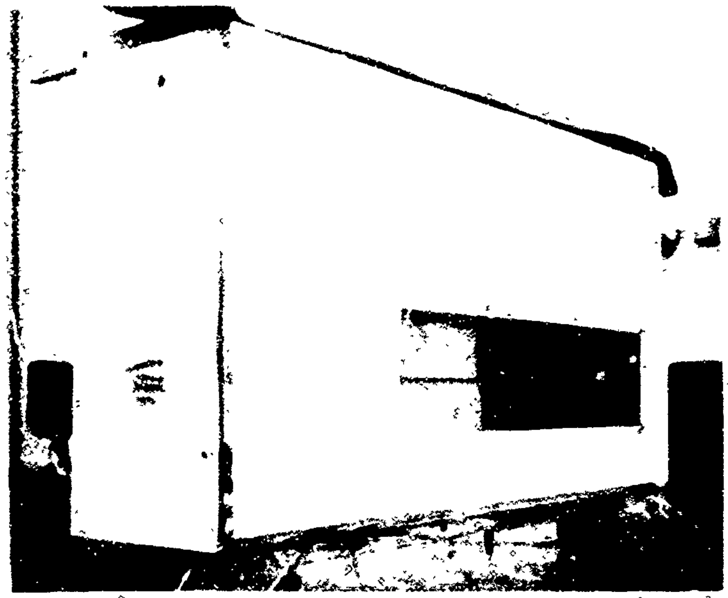

Figure 4

Photograph of Hood Which Encloses the
Cylinder and Surrounding Space

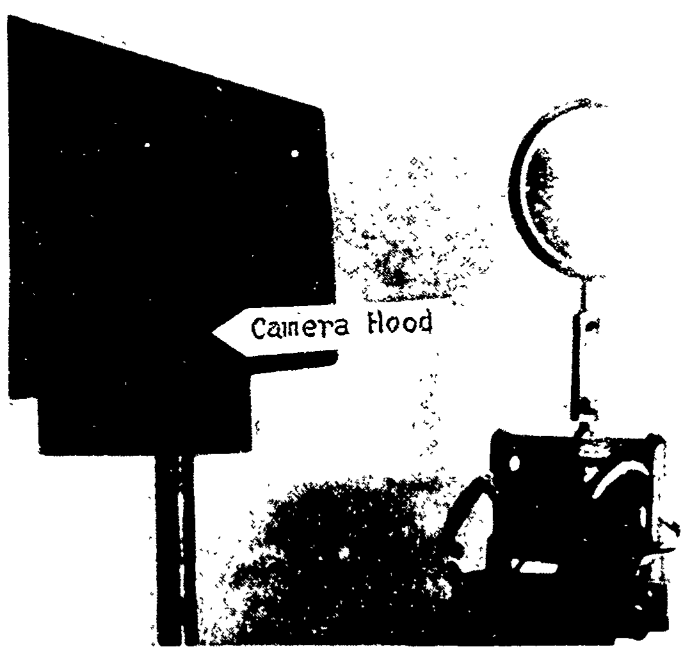

Figure 5

Photograph of Rotating Mirror on Small D.C.
Electric Motor in Front of Camera Aperture

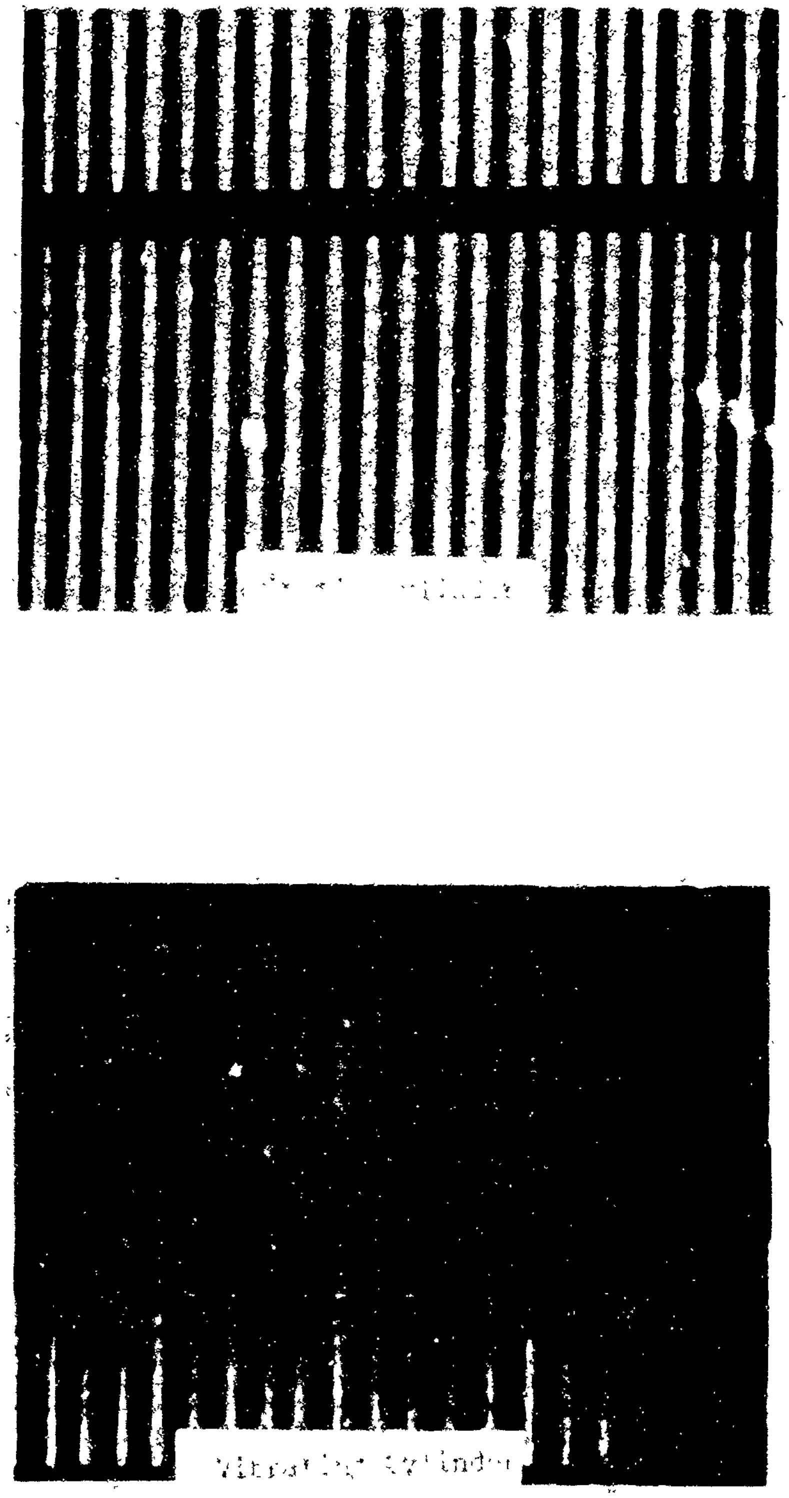

Figure 6

Photographs of Cylinder While Static and Vibrating

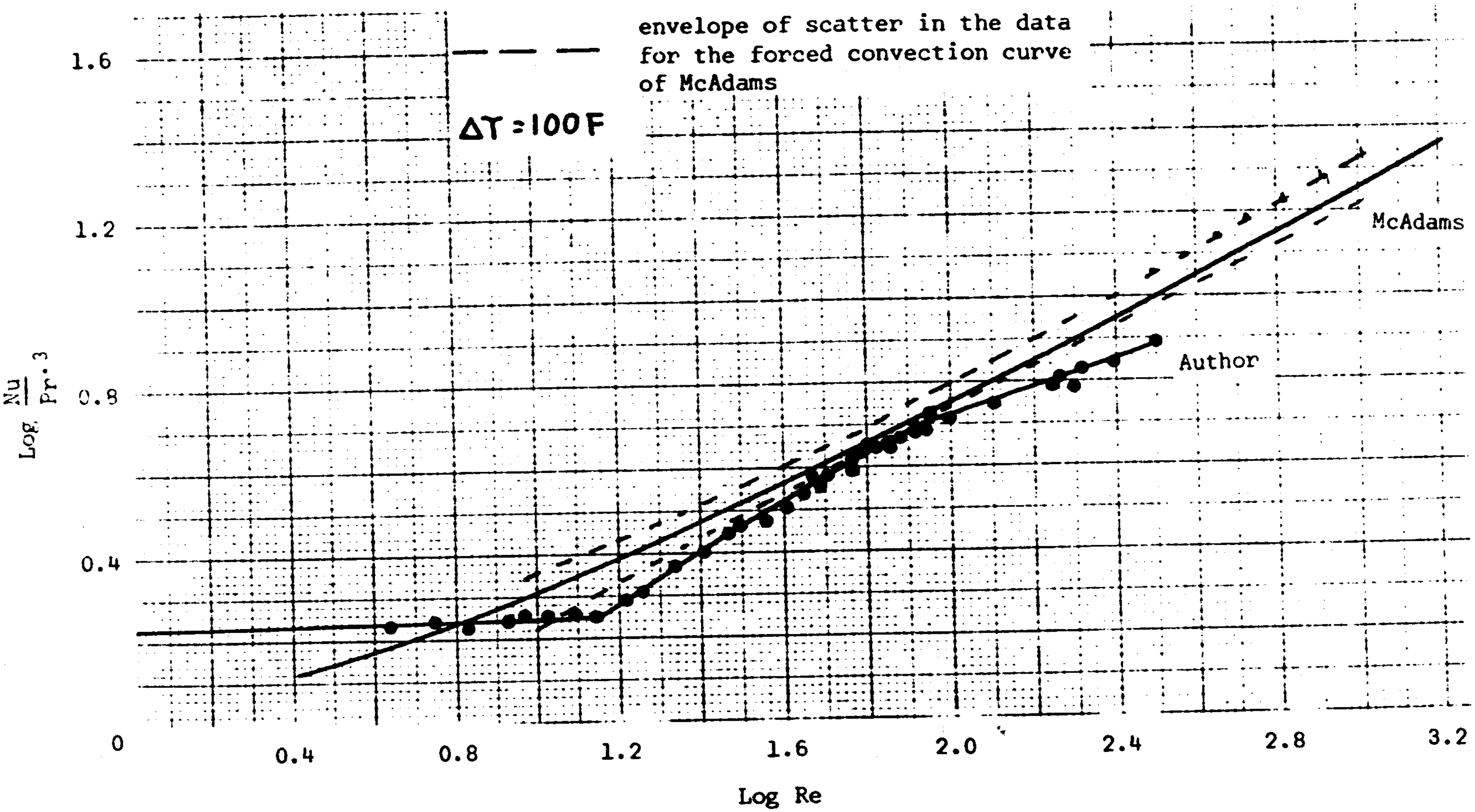

Figure 7

Variation of the Heat Transfer Rate with Vibration Intensity
For the 0.072 in Diameter Test Cylinder

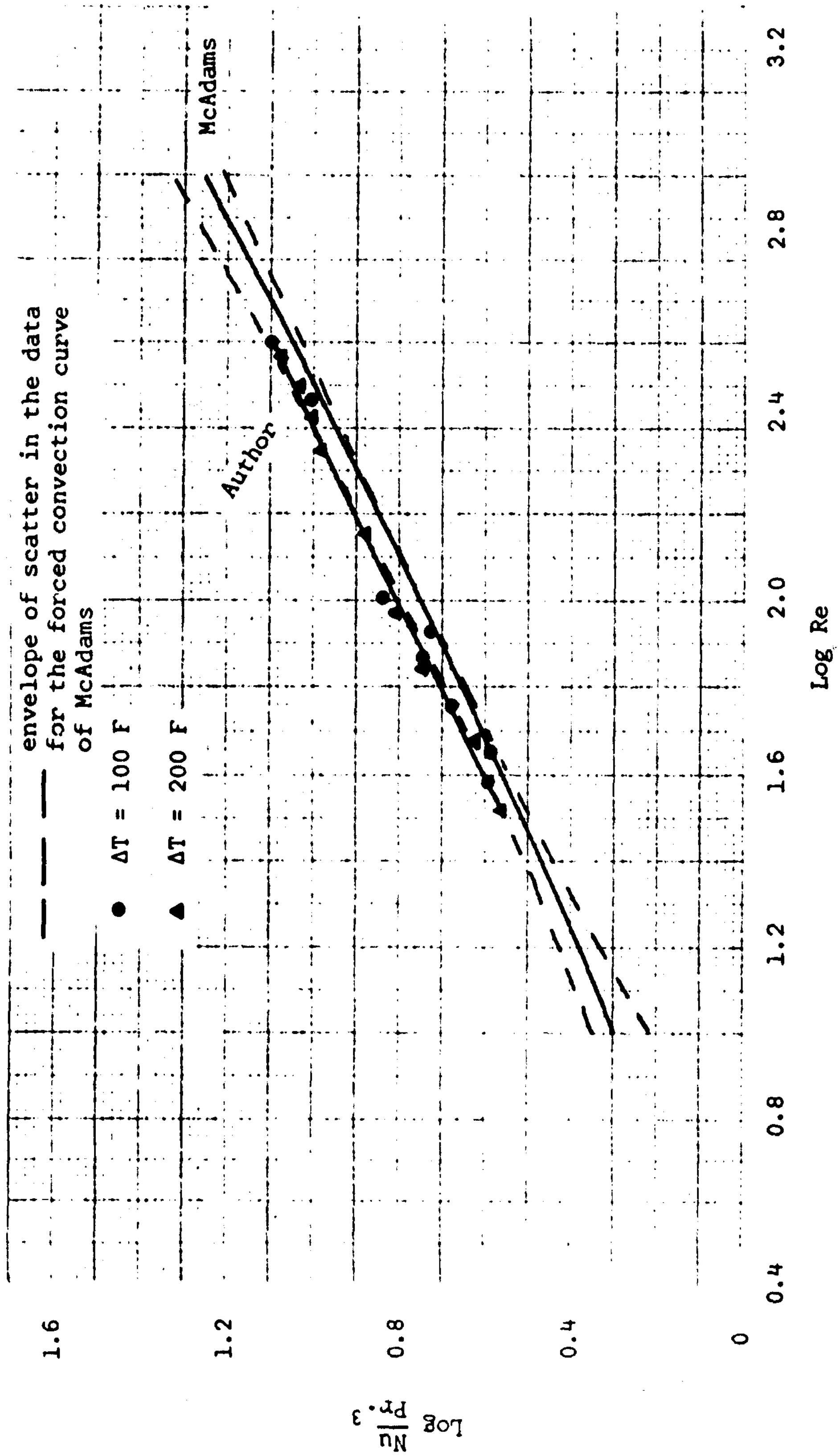

Figure 6

Variation of the Heat Transfer Rate with Vibration Intensity for the 0.120 in Diameter Test Cylinder

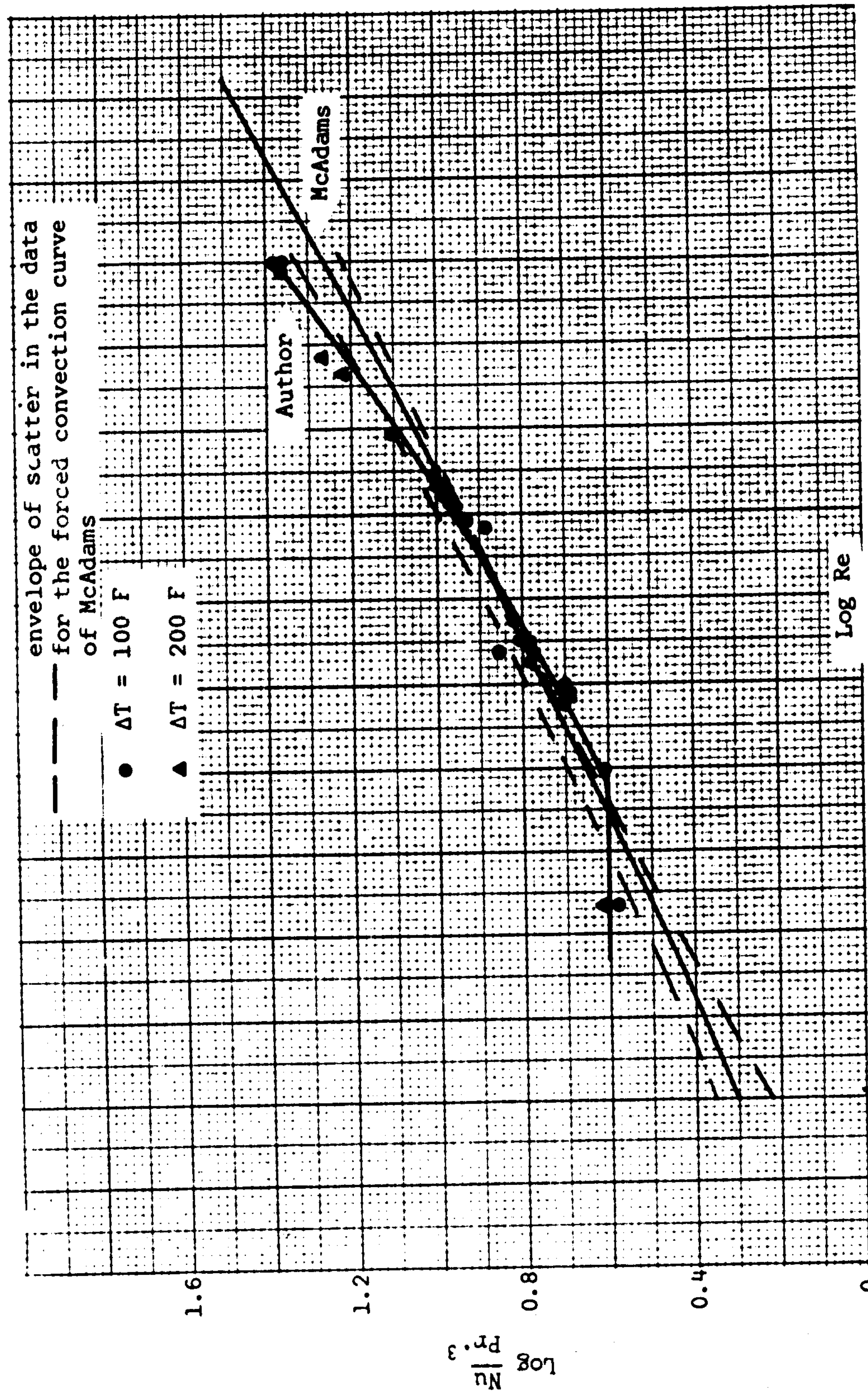

Figure 9

Variation of the Heat Transfer Rate with Vibration Intensity for the 0.25 in Diameter Test Cylinder

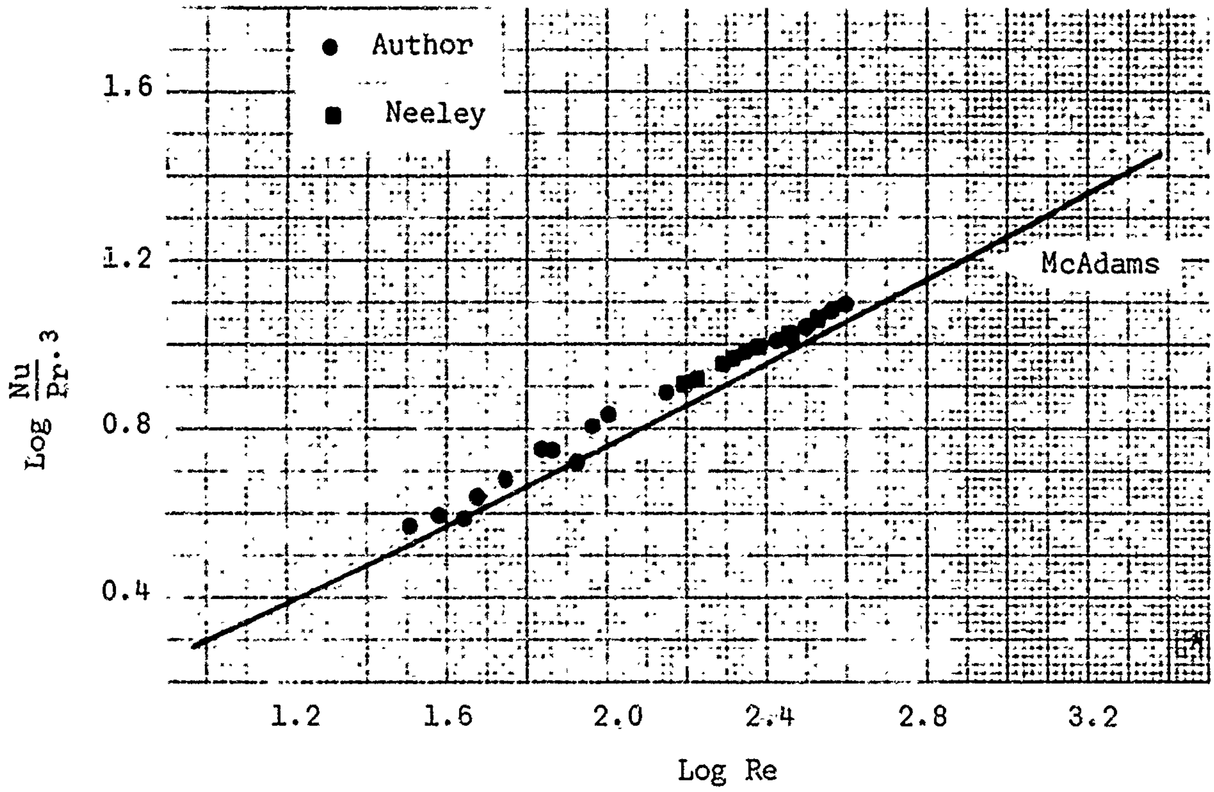

Figure 10

Comparison of the Heat Transfer Rates with Vibration Intensity Between the 0.120 in Diameter Cylinder and the 0.085 in Diameter Cylinder of Neely's Investigation

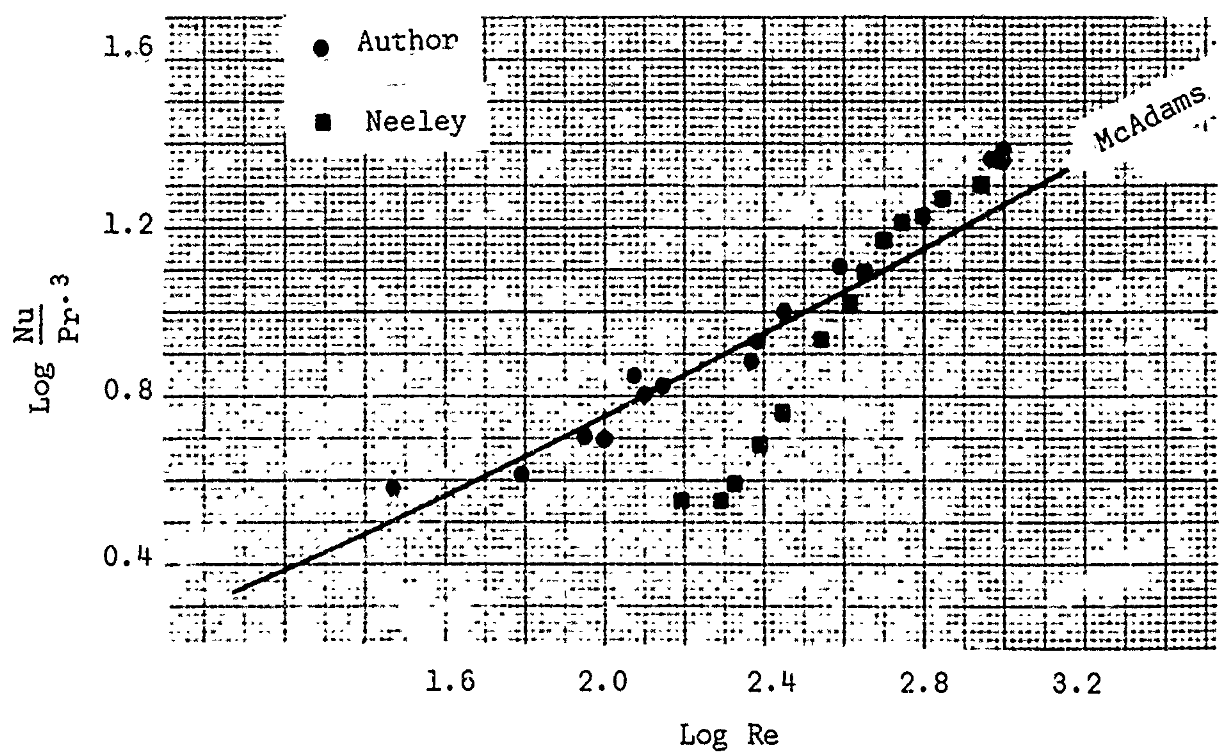

Figure 11

Comparison of the Heat Transfer Rates with Vibration Intensity Between the 0.25 in Diameter Cylinder and the 0.25 in Diameter Cylinder of Neely's Investigation

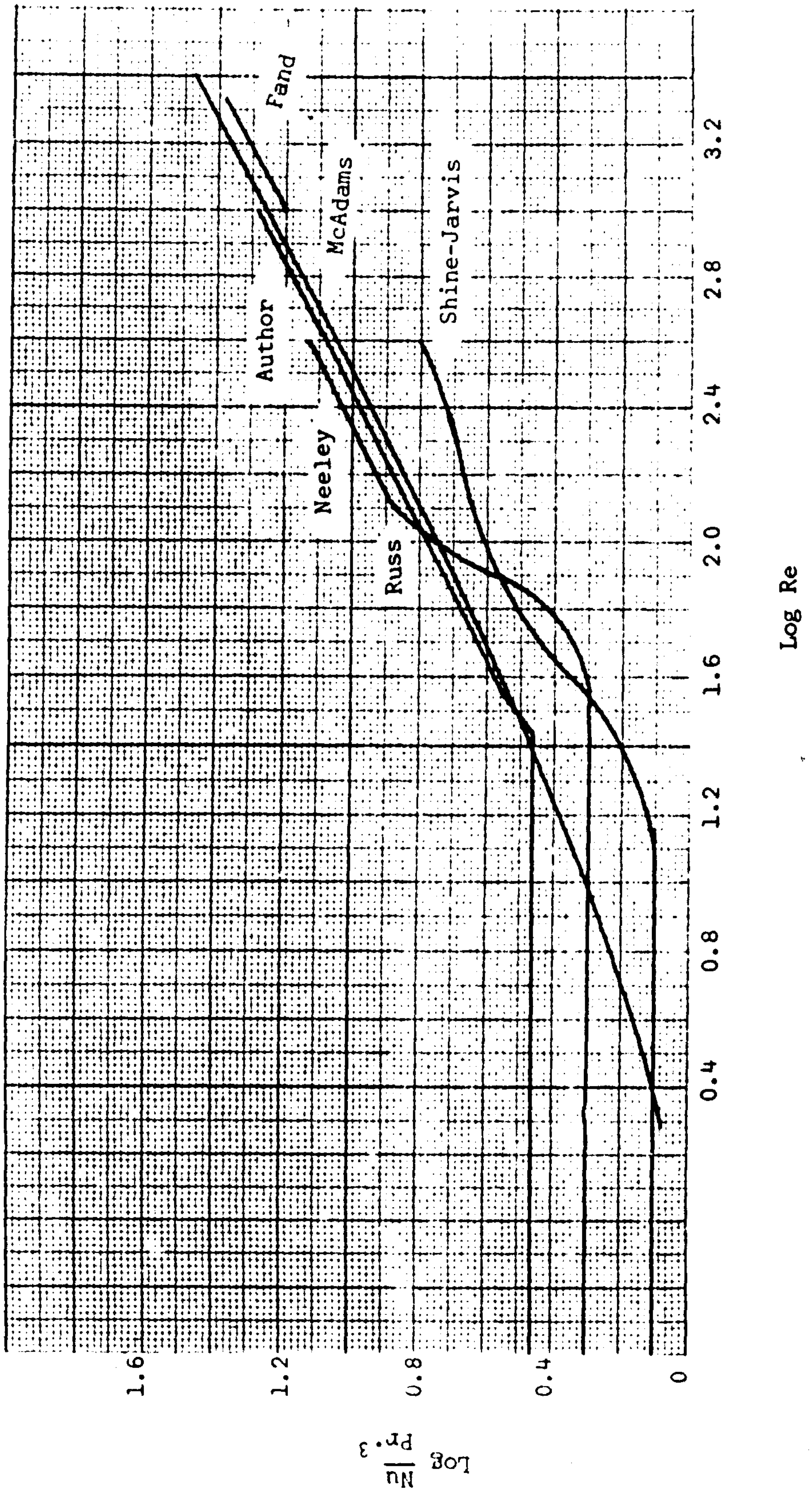

Figure 12

Comparison with Other Authors

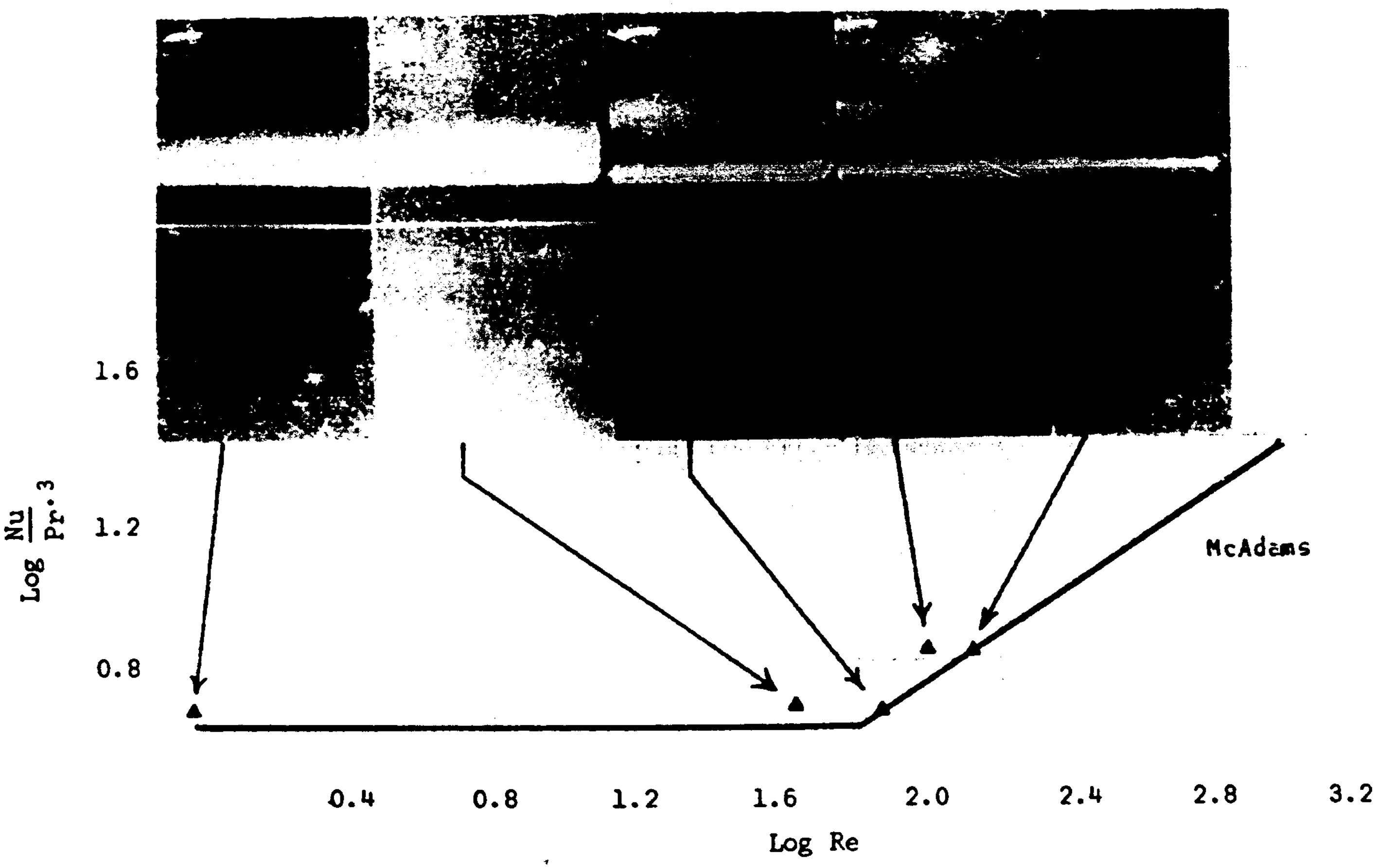

Figure 13

Photographs of the Boundary Layer on Cylinder While Vibrating in the Vicinity of the Critical Reynolds Number

Appendix B

Equations

Nusselt Number

$$Nu = \frac{hd}{k} = 3.413 \frac{IE\ d}{A\ k\ \Delta T} \quad \frac{B}{watt\ hr} \quad \frac{watt}{ft^2} \quad \frac{ft\ hr\ ft\ F}{B\ F}$$

Reynolds Number

$$Re = \frac{vd\rho}{\mu} = \frac{2afd\rho}{\mu} \quad \frac{ft}{sec} \quad \frac{ft}{lbm} \quad \frac{hr\ ft}{12} \quad \frac{lbm}{ft^3} \quad \frac{3600\ sec}{hr}$$

$$\doteq 600 \frac{afd\rho}{\mu}$$

where a is measured in inches.

Appendix C

Sample Calculation

The following calculations are based on run #20 for the 0.25 in diameter cylinder. The corrected values of the measured variables are:

$T_a = 79$ F	$I = 20.0$ amps	$a = 0.3740$ in
$T_w = 179$ F	$E = 1.75$ volts	$f = 42.70$ cps

B.P. = 29.385 in Hg

From tables in Eckert and Drake (Ref 1), the fluid properties were determined as

$$k_f = 0.16395 \text{ BTU/hr ft F}$$

$$\mu = 0.04777 \text{ lbm/hr ft}$$

$$P_r = 0.7042 \quad \text{dimensionless}$$

The following computations were performed:

$$T_f = \frac{T_w + T_a}{2} = \frac{179 + 79}{2} = 129 \text{ F}$$

$$\Delta T = T_w - T_a = 179 - 79 = 100 \text{ F}$$

$$Q_t = IE = (20.0)(1.78) = 35.0 \text{ watts}$$

The radiation heat loss computed from the following formula assuming the hood was very large compared to the diameter of the cylinder (Ref 5:390).

$$Q_r = 0.173\ \varepsilon\ A_w \left[\left(\frac{T_w}{100}\right)^4 - \left(\frac{T_a}{100}\right)^4\right]$$

$$= 0.173 \times 0.65 \times \left(\pi \times \frac{.25}{12} \times 3.14\right) \left[\left(\frac{179}{100}\right)^4 - \left(\frac{79}{100}\right)^4\right]$$

$$= 1.123 \times (.2055)(10.25 - 0.039)$$

$$= .01465\ \frac{BTU}{hr}$$

$$= 0.05 \text{ watt}$$

The conduction losses were assumed to be zero because the cross-section area of the cylinder was only 0.0375 in^2 and the temperature gradient along the cylinder was too small to be significant. Bednorz showed experimentally that conduction losses are insignificant (Ref 1:12). The convection heat transfer was

$$Q_c = Q_t - Q_r = (35.00 - 0.05) = 34.95$$

Nusselt Number

$$Nu = \frac{0.333\ Q_t}{k\ \Delta T}$$

$$= \frac{0.333 \times 35.0}{0.016395 \times 100} = 7.11$$

$$\frac{Nu}{Pr^{.3}} = \frac{7.11}{.902} = 7.88$$

Reynolds Number

$$Re = 12.5\ af\ \frac{\rho}{\mu}$$

$$= 12.5 \times 0.374 \times 42.7 \times \frac{0.0661}{0.04777} = 276$$

The overall error in the Nusselt number (see Appendix D)

$$\Delta Nu = \frac{3.413\ d}{A\ k}\left[\Delta I\ \frac{E}{\Delta T} + \frac{I}{\Delta T}\ \Delta E + IE\ \frac{\Delta(\Delta T)}{(\Delta T)^2}\right]$$

$\Delta I = .1125 \qquad \Delta E = 0.0085 \qquad \Delta(\Delta T) = 0.5$

$$\Delta Nu = \frac{3.413}{\pi \times 3.14 \times 0.016395}\left[\frac{.1125 \times 1.75}{100} + \frac{20 \times .0085}{100} + \frac{35 \times 0.5}{100 \times 100}\right]$$

$$= 20.9 \times .00542$$

$$= 0.113$$

The overall error in the Reynolds number

$$Re = 600\ \frac{d}{\mu}\ (f\rho\Delta a + a\rho\Delta f + af\Delta\rho)$$

$\Delta a = 0.002 \qquad \Delta f = 0.617 \qquad \Delta\rho \simeq 0 \qquad \rho = .0661$

$$\Delta Re = 600\ \frac{0.25}{12 \times 0.04777}\ (0.002 \times 42.7 \times .0661 + 0.374 \times 0.617 \times 0.0661)$$

$$= 261 \times 0.0209$$

$$= 5.45$$

The maximum overall error will be different for each run and will decrease as the size of the cylinder increases, and as the magnitude of the Nusselt number and Reynolds number increases the percent error decreases.

Appendix D

Analysis of Errors

Both human and instrument errors were possibly introduced into the raw data during the measurement of each of the seven variables. An analysis of the individual errors is presented in the following paragraphs.

Power Measurement

Electrical power input to the cylinder was computed using the formula $P = EI$.

Though the maximum error of the ammeter and voltmeter was 1% of the full scale reading, the effect of this error was reduced by calibrating the instruments against a meter which had a maximum error of 1/4 of 1% of the full scale reading. The only other error was the human error in reading the scales of the instruments. The accuracy of data read on the instruments are presented in Table D-1.

Vibration Intensity

Vibration intensity is the product of amplitude of vibration and frequency of vibration, and errors were possible in measuring each of these factors.

Frequency. The frequency of vibration was measured with a Strobotac which was calibrated prior to each series of test runs. When calibrated, the accuracy of the instrument was ±1% of the high scale which amounts to a possible error of ±0.617 cylces per second. The total error in frequency data is shown in Table D-1.

Table D-1

<u>Data Errors</u>

Instrument	Scale	Smallest Scale Graduation	Closest Approximation	Maximum Instrument Error	Maximum Total Error in Data
Ammeter	10 amp	0.10 amps	±0.01 amps	±0.025 amps	±0.035 amps
Ammeter	25 amp	0.25 amps	±0.05 amps	±0.0625 amps	±0.1125 amps
Ammeter	50 amp	0.50 amps	±0.10 amps	±0.125 amps	±0.225 amps
Voltmeter	3 volt	0.02 volts	±0.001 volts	±0.0075 volts	±0.0085 volts
Voltmeter	15 volt	0.10 volts	±0.050 volts	±0.0375 volts	±0.0875 volts
Strobotac	600-3700 cpm	10 cpm	±3 cpm	±37 cpm	±0.667 cps

Amplitude. The amplitude of vibration was measured on photographic film. The possible contributions to this error are errors in measuring the amplitude of the cylinder motion from the photographs. The amplitudes were scribed with a knife and read on a micro compensator which can be read to the fourth decimal place. Thus the primary error in measuring from the photograph was in scribing the amplitude on the photograph which contributed an error of ±0.001 in. The optical system was calibrated to within an accuracy of ±0.001 in error. Therefore, the maximum total error in obtaining the amplitude was ±0.002 in. Hence, the total error introduced in obtaining the vibration intensity was ±0.89 in/sec as computed from $\Delta(af) = \Delta af + a\Delta f$.

Temperature

The temperature measuring system, iron-constantan thermocouples and a Bristol's Dynamaster Recorder potentiometer, was calibrated against a mercury-in-glass thermometer in an ice bath, in ambient air (75 F), and in boiling water. The 65 to 212 F scale of the recorder used during the tests with a ΔT of 100 F agreed with the thermometer as close as the two scales could be read. The 40 to 320 F scale used during the test with a ΔT of 200 F agreed with the thermometer at 32 F and 75 F but differed at 212 F by 2 F. This error was applied to subsequent readings above 212 F. The temperature response to power changes and vibration intensity was immediate. This can be attributed to a positive contact which was maintained between the cylinder surface, a small size thermocouple junction, and the small heat capacity of the cylinder and thermocouple. Because of the quick response in temperature, the temperatures read were well stabilized, thus minimizing any error

in readings. The stability of cylinder and room temperatures was further increased by the hood which enclosed the space surrounding the test cylinder. Therefore, a close estimate of the accuracy with which temperatures were obtained is 1/2 F--the closest the scale could be read.

Overall Error

The error in Nusselt number due to errors in variables is

$$\Delta Nu = \frac{3.41\ d}{A\ k}\left[\Delta I\ \frac{E}{\Delta T} + I\ \frac{\Delta E}{\Delta T} + IE\ \frac{\Delta(\Delta T)}{(\Delta T)^2}\right]$$

The error in Reynolds number due to errors in variables is

$$\Delta Re = 600\ \frac{d}{\mu}\ (\Delta a f \rho + a \Delta f \rho + a f \Delta \rho)$$

The maximum overall error computed for the test run which should have the largest error during this investigation was a 3.72% error in Nu and a 2.0% error in Re.

Appendix E

Experimental Data

Table I

Experimental Data for 0.072 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
1	17	73	169	96	3.35	1.63	5.46	16.5	.202	1.973	.295	17	1.23
2	17	73	171	93	4.02	1.98	7.96	16.5	.359	2.8	.447	30	1.48
3	17	73	174	100	3.33	1.62	5.40	25.0	.139	1.866	.2705	17.5	1.24
4	15	73	173	100	3.29	1.6	4.65	34.6	.100	1.823	.261	13	1.1
5	15	74	174	100	4.77	2.365	11.30	34.6	.342	3.925	.594	59	1.77
6	13	75	174	99	5.18	2.58	13.38	34.6	.452	4.68	.67	78	1.89
7	11	74	172	98	5.39	2.68	14.42	34.8	.577	5.14	.71	101	2.01
8	17	75	176	101	5.48	2.73	14.97	44.4	.452	5.13	.71	100	2.0
9	19	75	174	99	5.07	2.52	12.80	44.45	.327	4.47	.651	73	1.86
10	14	77	182	105	3.3	1.62	5.34	0	0	1.778	.25	0	$-\infty$
11	14	78	180	102	3.25	1.59	5.16	16.55	.069	1.751	.244	5.6	0.75
12	14	78	180	102	3.3	1.615	5.33	16.55	.131	1.81	.258	10.7	1.03
13	19	78	180	102	3.4	1.67	5.68	16.55	.209	1.925	.284	17	1.23
14	19	78	177	98	4.19	2.07	8.67	16.55	.475	3.01	.478	39	1.59
15	17	79	180	101	4.49	2.22	9.95	16.55	.506	3.44	.536	41	1.62
16	15	79	181	101	4.93	2.46	12.13	24.7	.537	4.18	.621	60	1.78
17	15	79	179	100	4.75	2.37	11.25	24.7	.475	3.9	.591	58	1.76
18	15	79	181	101	4.6	2.3	10.58	24.7	.412	3.6	.556	50	1.70
19	15	79	178	98	4.2	2.09	8.78	24.7	.351	3.05	.484	43	1.63
20	15	79	179	99	3.88	1.92	7.45	24.7	.256	2.58	.412	31	1.49
21	15	79	179	100	3.4	1.66	5.58	24.7	.131	1.95	.29	16	1.20
22	17	80	182	102	4.58	2.29	10.50	34.3	.287	3.55	.55	48	1.68
23	17	80	178	98	4.93	2.46	12.12	34.3	.381	4.28	.632	65	1.81

Table I (cont.)

Experimental Data for 0.072 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
24	17	80	183	102	5.25	2.63	13.80	34.4	.491	4.81	.682	84	1.92
25	17	81	181	100	5.3	2.66	14.10	34.4	.530	4.85	.686	90	1.96
26	19	81	182	101	5.2	2.6	13.50	48.0	.334	4.63	.666	80	1.9
27	17	80	180	99	5.2	2.58	13.40	43.5	.287	4.68	.67	62	1.79
28	11	81	169	87	5.2	2.58	13.40	43.5	.413	5.35	.728	89	1.95
29	17	70	171	101	3.17	1.530	4.85	0	0	1.670	.22272	0	--
30	17	70	170	100	3.21	1.560	5.01	16.51	.054	1.725	.23679	4.4	0.64
31	17	70	171	101	3.3	1.600	5.28	16.51	.116	1.800	.25527	9.4	0.97
32	17	71	170	98	3.35	1.615	5.41	16.52	.147	1.890	.25744	12	1.08
33	17	72	172	100	3.59	1.678	6.02	16.52	.226	2.072	.31639	18	1.26
34	17	72	169	96	3.7	1.800	6.66	16.52	.272	2.385	.37749	22	1.34
35	17	72	175	102	3.98	1.878	7.47	16.52	.320	2.510	.39967	26	1.41
36	17	72	171	99	4.15	2.040	8.46	16.52	.382	2.960	.47129	31	1.49
37	17	72	171	99	4.25	2.100	8.93	16.52	.461	3.121	.49429	37	1.57
38	17	72	176	103	4.96	2.630	13.04	25.0	.507	4.340	.63749	62	1.79
39	17	72	172	100	4.67	2.310	10.79	25.2	.414	3.74	.57287	51	1.71
40	17	72	171	98	4.47	2.21	9.87	25.2	.367	3.45	.53782	45	1.65
41	17	72	172	100	3.93	1.93	8.93	25.2	.304	3.09	.48996	38	1.57
42	17	72	175	103	3.29	1.60	5.26	25.2	.1161	1.760	.24551	14	1.16
43	17	72	174	101	3.24	1.58	5.13	25.2	.070	1.740	.24055	9	0.93
44	17	72	174	101	3.2	1.542	4.94	25.2	.054	1.675	.22401	7	0.82
45	17	72	170	97	3.16	1.522	4.81	0	0	1.700	.23045	0	$-\infty$
46	17	69	168	99	3.16	1.521	4.81	0	0	1.672	.223	0	$-\infty$
47	17	70	168	98	4.7	2.320	10.90	34.7	.273	3.830	.583	46	1.67
48	17	71	169	98	5.1	2.520	12.85	34.8	.382	4.520	.655	65	1.81

Table I (cont.)

Experimental Data for 0.072 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
49	17	71	174	103	5.25	2.620	13.76	34.8	.460	4.600	.663	78	1.89
50	17	71	168	97	6.0	3.000	18.00	43.7	.882	6.400	.806	189	2.28
51	17	69	174	105	3.28	1.59	5.22	34.4	.070	1.710	0.233	12	1.06
52	17	70	170	100	4.18	2.65	11.10	34.4	.212	3.820	.582	36	1.55
53	19	70	170	100	5.08	2.52	12.70	34.5	.398	4.380	0.642	67	1.83
54	19	71	168	97	5.6	2.78	15.58	44.7	.632	5.53	0.743	139	2.13
55	19	71	170	99	5.99	2.98	17.85	44.4	.820	6.20	0.792	178	2.25
56	19	71	171	99	5.98	3.00	17.96	48.9	.835	6.20	0.792	200	2.30
57	19	72	171	99	6.6	3.10	20.42	53.8	.943	7.11	0.852	248	2.40
58	19	71	170	98	6.25	3.15	19.67	63.5	.678	6.86	0.836	211	2.32
59	19	72	170	98	6.6	3.3	21.76	69.8	.928	7.65	0.884	317	2.50

Table II

Experimental Data for .1202 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
1	29	77	178	101	6.75	1.75	11.8	0	0	2.96	.472	0	$-\infty$
2	29	79	180	101	12.08	3.35	40.50	43.80	1.052	9.8	.991	376	2.58
3	41	77	174	97	8.05	1.72	15.4	44.05	0.107	3.9	.60	39	1.59
4	27	75	173	98	12.70	3.50	44.50	39.35	.886	11.2	1.005	293	2.47
5	27	75	177	101	14.50	3.80	55.50	39.35	1.243	12.6	1.1	408	2.60
6	26	76	286	209	9.75	3.0	29.22	0	0	3.01	.479	0	$-\infty$
7	26	76	268	192	10.25	3.15	32.30	16.75	.276	3.67	.565	33	1.53
8	26	77	274	197	11.25	3.5	39.40	16.70	.403	4.36	.639	48	1.68
9	26	77	276	199	12.75	4.0	51.00	16.78	.601	5.7	.756	72	1.86
10	26	78	280	202	13.75	4.32	59.50	16.78	.787	6.49	.812	94	1.97
11	26	78	274	196	14.75	4.7	69.30	21.5	.918	7.89	.897	142	2.15
12	26	78	282	204	15.75	5.1	80.20	25.1	.961	8.68	.939	171	2.23
13	26	78	285	197	16.75	5.48	91.70	24.9	1.26	9.76	.990	223	2.35
14	26	79	286	207	17.25	5.63	97.00	25.04	1.483	10.34	1.014	264	2.42
15	26	80	284	204	17.75	5.82	103.20	33.7	1.29	11.15	1.047	309	2.49
16	26	80	282	202	18.5	6.05	112.00	33.7	1.56	12.22	1.087	375	2.57
17	23	78	181	103	9.7	2.92	28.3	36	.349	6.92	.84	103	2.01
18	23	78	176	98	8.9	2.55	22.7	36	.255	5.65	.752	75	1.88
19	23	79	177	98	8.4	2.51	21.1	36	.286	5.3	.724	84	1.93
20	23	80	176	96	8.0	2.34	18.72	36	.192	4.8	.681	57	1.75
21	23	80	180	100	7.4	2.18	16.12	36	1.56	3.96	.597	46	1.66

Table III

Experimental Data for 0.25 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
1	29	74	173	99	14.25	1.25	17.82	0	0	4.07	0.61	0	$-\infty$
2	29	76	172	96	20.15	1.8	36.2	42.2	0.335	8.54	0.931	246	2.39
3	29	75	177	98	25.0	2.3	57.5	42.4	.536	13.18	1.12	389	2.59
4	29	77	172	95	32.0	3.0	96.0	42.4	1.410	22.8	1.358	1037	3.02
5	29	82	177	94	32.6	3.1	101.0	42.2	1.448	24.08	1.382	1039	3.02
6	29	79	177	98	19.5	1.75	34.1	42.3	.325	7.83	.894	237	2.38
7	29	78	170	92	17.5	1.55	27.1	20.9	.391	6.66	.824	143	2.16
8	29	81	179	98	32.5	3.01	97.8	43.2	1.313	22.65	1.355	975	2.99
9	29	77	176	99	15.6	1.4	21.82	20.88	.297	4.99	.698	108	2.03
10	29	78	178	99	16.1	1.6	28.95	20.88	.357	6.55	.816	130	2.12
11	29	74	178	104	14.4	1.28	18.43	16.85	.1015	3.82	.582	30	1.47
12	29	74	177	103	14.6	1.30	19.00	16.85	.212	4.14	.617	63	1.80
13	29	75	176	101	16.0	1.40	22.40	16.85	.305	5.00	.699	90	1.96
14	29	76	178	102	19.5	1.70	33.15	16.75	.423	7.30	.863	124	2.08
15	35	76	177	101	14.5	1.30	18.85	0	0	4.215	.625	0	$-\infty$
16	35	75	182	107	15.0	1.32	19.80	16.77	.1019	4.17	.620	29.5	1.47
17	35	78	180	102	16.0	1.40	22.40	22.23	.2490	4.94	.694	95.8	1.98
18	35	78	177	99	17.5	1.51	26.40	22.23	.2895	6.03	.780	111.4	2.05
19	35	79	180	101	18.0	1.57	28.25	24.05	.3050	6.29	.799	126.5	2.10
20	35	79	179	100	20.0	1.75	35.0	42.70	.3740	7.88	.897	276	2.44
21	35	80	173	93	23.0	2.05	47.15	42.50	.4180	11.44	1.058	306.5	2.49
22	35	83	181	98	28.0	2.58	72.25	42.50	.5800	16.63	1.221	538	2.73

Table III (cont.)

Experimental Data for 0.25 In Diameter Cylinder

Run	Test Pt from Left End in	T_a F	T_w F	ΔT F	I amps	E volts	IE watts	f cps	a inches	$\frac{Nu}{Pr^{.3}}$	$\log \frac{Nu}{Pr^{.3}}$	Re	log Re
23	35	84	184	100	30.0	2.80	84.00	42.50	.7400	18.85	1.275	532	2.73
24	35	83	183	100	30.0	2.80	84.00	42.50	.8150	18.85	1.275	592	2.77
25	28	73	288	215	22.5	2.12	47.7	0	0	4.69	.671	0	$-\infty$
26	28	74	284	210	22.5	2.12	47.7	16.85	0.172	4.8	.681	43	1.63
27	28	76	283	207	25.0	2.35	58.7	16.85	0.298	4.52	.655	74.5	1.87
28	28	76	262	186	21.1	2.10	44.3	16.85	0.406	6.79	.831	104.4	2.01
29	28	75	284	209	26.8	2.40	64.4	16.85	0.531	6.56	.817	132.7	2.13

Vita

Major William J. Watson was born [redacted] He was graduated from the University of Alabama in 1952 and was commissioned as a second lieutenant in the United States Air Force through the ROTC program. At that time he entered pilot training, and upon completion of training was assigned duty with the 44th Fighter Bomber Squadron at Clark AFB, P. I. Upon completing an eighteen month tour of duty with the 44th FBS, he was assigned to duty with the Air Force Systems Command at Elgin AFB, Florida and Wright-Patterson AFB, Ohio. Subsequently, he entered resident training for Graduate Aeronautical Engineering (Air Weapons Option) at Wright-Patterson AFB, Ohio.

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This thesis was typed by Mrs. Imogene J. Hoffer.