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Heat transfer measurements have been made from a long (L/D = 15) horizontal cylinder in water parallel to a heated vertical wall. The cylinder and wall boundary conditions were isothermal, and they were maintained at equal temperatures. The investigation covered Rayleigh numbers based on cylinder diameter from 400,000 to 10,000,000. The cylinder height above the wall leading edge was set at height to diameter ratios, H/D, of 2.07, 7.44, and 15.50. The horizontal space between the cylinder and wall spanned the range $0.05 < S/D < 0.62$. It was found that heat transfer was degraded as much as 46% for close spacings and enhanced as much as 15% at moderate spacings. The lowest position (H/D = 2.07) showed the greatest overall enhancement, while the highest position (H/D = 15.5) showed overall degradation.

Natural Convection from a Horizontal Cylinder Parallel to a Heated Vertical Wall

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TABLE OF CONTENTS

LIST OF FIGURES

Natural Convection from a Horizontal Cylinder Parallel to a Heated Vertical Wall

I. Introduction and Background

Natural convection plays an important role in many natural and man made processes. Heat transfer due to natural convection is of interest to engineers and scientists in many disciplines. From atmospheric convection to the convective currents in molten silicon, this phenomenon affects the way we live. This thesis examines a tiny part of this vast field with the hope of providing more information towards our understanding of natural convection heat transfer processes.

This thesis describes an experimental examination of the natural convection heat transfer from a horizontal cylinder as it is affected by the presence of a heated vertical wall. The heat transfer from a cylinder in an infinite medium has been thoroughly examined by many workers. The purpose of this study is to extend the data base and gain an understanding of the processes involved when a heated wall is near the cylinder. This configuration is important for the cooling of electronic components and has general applicability for the design of any natural convection heat exchange equipment containing cylindrical components.

The basic phenomenon involved is the influence of flow induced by the heated wall. The blockage effects due to the presence of a solid wall also have some significance. More generally, the heat transfer from the cylinder is highly dependent on the local fluid behaviour.

When a cylinder is placed in an infinite, undisturbed medium, the heat transfer is dependent only on the geometry, temperatures and fluid properties. Very rarely is an infinite medium encountered in practice however. In electronic equipment, the heat transfer from any given component is affected by the flow induced by all the other components. In any heat exchange process, there are unique factors that will affect the heat transfer from a given component.

By placing a cylinder adjacent to a heated wall, the effect of the flow induced by that wall can be examined. This is an approximation of an electronic component immersed in the flow induced by other components. The presence of this natural convection boundary layer flow may have contradictory effects on the cylinder heat transfer. The moving fluid resembles a forced convection flow that would tend to enhance the heat transfer from the cylinder. However, the fluid will be preheated above the ambient and as a result will be able to carry away less heat from the cylinder. The solid wall itself may have a varying effect, either blocking or possibly accelerating the flow around the cylinder. The results of this study show that there is a significant effect on cylinder heat transfer due to the presence of the heated wall. It also points toward promising areas for further research.

The configuration studied was a cylinder in water with Rayleigh numbers based on cylinder diameter ranging from 400,000 to 10,000,000. The heated plate temperature was matched with the cylinder temperature, and the cylinder and plate boundary conditions were isothermal. The parameters varied were; cylinder temperature, distance from the plate and height above the leading edge of the plate. The Nusselt number as measured from power input to the cylinder is presented as a function of the cylinder Rayleigh number and the position of the cylinder relative to the plate. Flow visualization was also employed, using the thymol blue method proposed by Baker [1].

II. Natural Convection from a Horizontal Cylinder Parallel to a Heated Vertical Wall

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ABSTRACT

Heat transfer measurements have been made from a long (L/D = 15) horizontal cylinder in water parallel to a heated vertical wall. The cylinder and wall boundary conditions were isothermal, and they were maintained at equal temperatures. The investigation covered Rayleigh numbers based on cylinder diameter from 400,000 to 10,000,000. The cylinder height above the wall leading edge was set at height to diameter ratios, H/D, of 2.07, 7.44, and 15.50. The horizontal space between the cylinder and wall spanned the range $0.05 < S/D < 0.62$. It was found that heat transfer was degraded as much as 46% for close spacings and enhanced as much as 15% at moderate spacings. The lowest position (H/D = 2.07) showed the greatest overall enhancement, while the highest position (H/D = 15.5) showed overall degradation.

INTRODUCTION

Natural convection heat transfer from a long (L/D > 20) cylinder in an infinite medium has been well researched and documented. However, this information may not be sufficient when the typical application does not resemble an infinite medium. There are often other geometrical considerations as well as a variety of thermal fluid environments that complicate the problem. This paper presents the effects on the natural convection heat transfer from a long cylinder when it is placed adjacent to a heated vertical wall. This work has application to the cooling of electronic components and many other heat exchange processes involving cylindrical elements. The heat transfer from an electronic component is highly dependent on the packaging geometry and the thermal fluid environment. The heat transfer from any one element is affected by the presence of other elements. The cylinder and plate geometry may be considered a simplified model that attempts to gain insight to general effects due to immersion of a heat exchange surface in a thermal plume.

There are two opposing mechanisms that will affect the heat exchange from the cylinder. The immersion of the cylinder in the plate boundary layer will subject it to a forced flow. This flow would be expected to enhance the heat transfer from the cylinder. However, the fluid rising near the plate will be preheated to a temperature above that of the ambient fluid. Since the temperature difference between the cylinder and fluid is decreased, the heat transfer would decrease as compared to a cylinder in an infinite reservoir. A further effect to be considered is the flow blockage due to the presence of a solid wall. This blockage is expected to degrade the heat transfer in general, but enhancement may result at certain spacings. The goal of this work is to determine how these effects vary with the parameters that define the problem.

The parameters varied for this study were; cylinder Rayleigh number, distance to the wall, and height from the leading edge of the wall. The fluid used was water, and the experiments were carried out in a plexiglass tank. The Rayleigh number based on cylinder diameter ranged from 400,000 to 10,000,000. The dimensionless cylinder to plate distance, S/D, was varied from 2.12 to 0.05, and the dimensionless cylinder height, H/D, took the values 2.07, 7.44, and 15.50. The experiment was performed in such a way as to approximate two dimensional flow at the test section and eliminate end effects. The cylinder and plate boundary conditions were isothermal to within $2^{o}C$ and the temperatures of the cylinder and plate were matched. Flow visualization was employed, using the thymol blue method of Baker [1].

LITERATURE REVIEW

Much work has been done on natural convection heat transfer from a single cylinder, but recently there has been a great deal of interest in the effects of outside influences. Influences of current interest consist chiefly of solid wall interactions and multiple cylinder arrangements. Marsters [2] studied the effect of confining adabatic walls on the heat transfer from a horizontal cylinder with Rayleigh numbers ranging from 10 to 500,000. His work identified wall spacing and wall height as the important variables. It was found that heat transfer from the cylinder was enhanced for virtually all configurations studied. Further work by Karim et al [3] and Sparrow and Pfeil [4] extend and refine this work. Tokura et al [5] worked with vertical arrays of 2, 3 and 5 cylinders in free space and between parallel confining walls, with Rayleigh numbers from 28,000 to 280,000. It was found that heat transfer from the bottom cylinder was not significantly affected, while the upper cylinders were enhanced to a maximum at a spacing of 6 to 10 diameters. Sparrow and Niethammer [6] and Razelos [7] studied vertically separated horizontal cylinders as well and came to the same conclusions as Tokura et al [5]. They also examined the effect of temperature imbalance between the two cylinders, determining that the effect is strong only at small separation distances ($S/D < 5$).

Sparrow and Boessneck [8] examined the effect of horizontal misalignment on vertically separated cylinders and found that misalignment tended to moderate both enhancement and degradation. Hunter and Chato [9] undertook to evaluate a variety of configurations, including vertically aligned, vertically misaligned and one upper with two symmetrically placed lower cylinders. Their Rayleigh numbers ranged from 100 to 600, typical of condenser and refrigerator heat exchange devices. For the first two categories the results were consistent with those of Sparrow and Niethammer [6] and others; although in their range of Rayleigh numbers, the maximum enhancement occurred at greater separation distances (S/D = 19). For the tests with two symmetrically placed lower cylinders even greater enhancement was found, with a maximum at a spacing of W/D = 3. The plumes from the two lower cylinders combined to create a chimney effect which drew cooler air past the upper cylinder resulting in enhancement.

Further work on wall effects was undertaken by Sparrow et al [10]. They examined short cylinders attached perpendicularly to a heated wall at cylinder Rayleigh numbers from 15,000 to 200,000. They determined that the lower pin fin cylinders were unaffected by those above. However, the upper cylinders showed heat transfer degradation at low Rayleigh numbers and enhancement at higher Rayleigh numbers. The enhancement was most significant at larger spacings. It was determined that short cylinders had smaller Nusselt numbers than longer cylinders, while both were smaller than the isolated cylinder case. Larson and Fries [11] did similar work with short wall attached cylinders in water. They found that for Rayleigh numbers between 500,000 and 4,000,000 and spacing to diameter ratios above 4.25 the cylinder Nusselt number was not decreased by the presence of a lower cylinder. Also, the Nusselt number increased with increasing spacing. Sparrow and Ansari [12] worked with heat transfer from a cylinder in the presence of adiabatic, partially enclosing walls at cylinder Rayleigh numbers from 20,000 to 200,000. The three configurations examined were a horizontal cylinder adjacent to 1) a vertical wall, 2) a bottom horizontal wall and 3) a corner formed by the intersection of both. The side wall interaction provided some enhancement at the largest spacing, with degradation increasing as spacing decreased. The presence of the bottom wall decreased the heat transfer, with the smallest spacing being the most degraded. The corner configuration degraded the heat transfer for each case also.

It is instructive to review the nature of the natural convection flow around an isolated isothermal cylinder and an isolated isothermal plate. Kuehn and Goldstein [13] numerically solved the full Navier-Stokes' and energy equations for laminar natural convection flow about a horizontal isothermal cylinder. Their results show that a boundary layer forms around the lower portion of the cylinder with a plume coming off the top. This boundary layer becomes thin relative to the cylinder diameter above a Rayleigh number of about 10^6. An important observation concerns the approach flow, which at higher Rayleigh numbers comes mostly from the side rather than from below. It is obvious that the presence of a solid wall would affect this radically.

Since the natural convection boundary layer of the plate is responsible for the changes in heat transfer from the cylinder, it is instructive to examine it. Profiles of velocity and temperature have been plotted in Figs. 1 and 2 from Ostrach [14] for a Prandtl number of 10. These represent temperature differences of 10 and 30 degrees C respectively at three different heights from the bottom of the plate. The vertical lines in the figures represent the diameter of the present test cylinder for comparison with the boundary layer thickness. These figures show that both the thermal and velocity boundary layers thicken appreciably at the lower temperature difference (lower Rayleigh number) as compared to the higher temperature difference. The fluid velocities increase substantially with height on the plate and with increasing Rayleigh number. However, the temperature profiles change only slightly with height on the plate. As the velocity boundary layer thickens with height, there is more fluid moving upwards. Transition to turbulence occurs at a Grashof number of approximately 1.3×10^9. This means that the plate boundary layer is probably turbulent at H/D = 15.5 when the cylinder Rayleigh number exceeds 3,500,000 ($Ra_H = Ra_D(H/D)^3$). Another interesting observation concerns the entrainment flow. Scale analysis by Bejan [15] shows that the horizontal velocity varies as the height to the negative one-fourth power. This indicates that the entrainment flow effect on the heat transfer would be strongest near the leading edge of the plate (small H/D).

EXPERIMENTAL APPARATUS AND PROCEDURES

Test Cylinder

The main focus of the present experiment was to examine the heat transfer from the cylinder, which was built and instrumented for this purpose. The cylinder was built in three sections; a test section in the center and guard heater sections at either end as depicted in Fig. 3. Each section was built of an internal copper slug of diameter 1.82 cm (0.715 in) and length 8.94 cm (3.52 in) and an outer brass sleeve of diameter 1.94 cm (0.764 in). The middle section brass sleeve was 11.51 cm (4.53 in) long while the two outer sleeves were 8.38 cm (3.30 in) long. The copper slugs were drilled on the center line to accept cartridge heaters and drilled at off-center positions (radius of 0.66 cm) to carry thermocouple and the test heater wires. The slugs were soldered into the sleeves, with the central sleeve overlapping the two outer slugs. A gap of 0.32 cm (0.125 in) was left between each adjoining slug. The test section was taken as 9.26 cm (3.65 in) in length, while the overall cylinder was 28.27 cm (11.13 in) long. It should be noted that the cylinder was constructed with as small a diameter as possible to maximize the effect of the relatively thin wall boundary layer (see Figs. 1,2). This constraint presented some machining problems and had to be relaxed more than was desired.

Ten type T 24 gauge glass insulated thermocouples were used for cylinder temperature measurements. Two thermocouples were placed at each end of the test section, 180 degrees apart. They were matched with thermocouples in the same positions at the inner ends of the guard sections. An additional thermocouple was placed at the outer end of each guard section. The thermocouples were soldered into 0.32 cm (0.125 in) diameter copper plugs that were, in turn, soldered into the holes drilled in the copper. Each thermocouple was at a distance of 0.31 cm (0.122 in) from the surface of the cylinder. The calculated temperature drop to the surface was less than 0.3 degrees C. The thermocouple wires were threaded through the length of the cylinder at the same radial position, minimizing conduction heat loss down the leads.

The cylinder was suspended in the test tank by two supports that served as conduits for the thermocouple and heater leads. Two copper elbows were soldered into the ends of the cylinder and glued over the two PVC pipes that acted as supports. The supports were then connected to the positioning apparatus. The entire assembly was checked for watertightness. The assembly was constructed so that the cylinder could be brought close to the heated wall without any part of it coming in contact with the wall.

Test Tank and Support Equipment

All tests were performed in a rectangular plexiglass tank 51.44 cm (20.25 in) tall, 54.61 cm (21.5 in) long and 35.56 cm (14.0 in) wide as shown schematically in Fig. 4. An aluminum plate 25.4 cm (10.0 in) wide by 35.56 cm (14.0 in) high was mounted in one end with its leading edge 7.62 cm (3.0 in) above the tank bottom. The plate was heated with 5 horizontal strip heaters and instrumented with 4 type T thermocouples. The thermocouples were centered between the heaters on the plate vertical centerline. A heat exchanger was placed in the tank to maintain a constant ambient water temperature. The ambient water temperature was measured with 5 type T thermocouples placed in the wall of a vertical PVC pipe and sealed with epoxy. This pipe was placed vertically 10.0 cm (3.94 in) from and centered with respect to the plate. The thermocouples were spaced 10.0 cm (3.94 in) apart with the lowest 3.0 cm (1.18 in) from the tank bottom.

Water for the tests was degassed tap water. By bringing the water up into a holding tank and applying reduced pressure, dissolved gases could be removed and bubble formation on the heated surfaces prevented for the duration of the tests. The cylinder supports were attached to a horizontal positioner that was mounted on a table capable of vertical adjustment. These devices allowed the positioning of the cylinder without restarting the experiment. Power was supplied to all heaters with auto-transformers. Each section of the cylinder was powered separately while the wall was heated with a single power source.

Instrumentation

All readings except heater current were taken with two Hewlett Packard model 3421A data acquisition units. These units measured thermocouple temperature and heater voltage. They were controlled with a Hewlett Packard 87 model computer via a BASIC program. All data were written to a floppy diskette. The heater current was measured with a Hewlett Packard model 3465B digital multimeter and typed into the HP 87 for storage on diskette. The specified accuracy of the HP 3421A is 1.0 microvolt, while the HP multimeter has a precision of 0.1 milliamperes. All of the thermocouples were calibrated in a constant temperature bath by lowering the cylinder into the bath and allowing it to reach the specified water temperature. The uncertainty of thermocouple readings was determined to be ±0.5 degrees C.

Flow Visualization

Flow visualization was done with the thymol blue method, as proposed by Baker [1]. This is an electrochemical technique where a neutral tracer fluid is produced by a pH indicator. Thymol blue is dissolved in water so that an acidic solution turns yellow while a base solution turns blue. By adding enough NaOH to the water to turn it deep blue, then titrating it with HCl until it just turns yellow, a blue tracer fluid can be produced. This is done by placing positive and negative electrodes in the solution and applying DC voltage between them. At the negative electrode an electrochemical reaction takes place that turns the local solution to a basic pH thus producing a blue tracer. In this experiment the cooling heat exchanger was used as the positive electrode, while several 26 gauge copper wires were strung in parallel for the negative electrode. These wires were arranged below and beside the test cylinder parallel to it. DC current at 20 volts was applied initially and then turned down to 5 volts after 2 or 3 seconds. The higher voltage produced more tracer but also caused undesirable hydrogen bubble formation.

Procedures

The test process was initiated by degassing the water in the tank. When a sufficient supply of water had been degassed, the tank was filled and allowed to settle. The cylinder was lowered into the water to the desired depth and all trapped air bubbles removed. To zero the horizontal positioner the cylinder was backed to the plate and the position recorded. The currents through the heaters were set to predetermined values estimated to match the test section temperature with the guard section temperature, and the wall temperature with the cylinder temperature. These values were subject to only slight modification for each run. The data acquisition program presented in appendix D was then run. The program waited ten minutes for steady state and then took five readings of all thermocouples with a minute wait between each reading. After five readings, the program would pause for repositioning and wait another ten minutes before starting the measurement process again. In this way several positions could be examined at approximately the same Rayleigh number without a restart of the process. When a sufficient number of runs were made, all data were transferred from the HP 87 to an IBM PC for data reduction as described in appendix C. The data reduction program appears in appendix D. The visualization runs were made after all other data were taken. The procedures were the same except that no data were taken.

Three types of tests were made; isolated cylinder, cylinder adjacent to a cold wall and cylinder adjacent to a heated wall. For the isolated cylinder tests, the cylinder was moved as far as possible from the wall (S/D = 3.0) and set at a middle height (H/D = 7.44). For the cold wall tests, the cylinder was brought adjacent to the wall at the same height. The heated wall tests were run at a variety of heights with the cylinder adjacent to the wall.

EXPERIMENTAL RESULTS

Data Reduction

For each run there were five sets of raw data. Out of these data, an average temperature for the cylinder and the ambient were obtained, along with average values of the test heater voltage and current. From these, Rayleigh and Nusselt numbers were calculated. The radiation loss was calculated to be less than 0.1% and was neglected. The average cylinder heat transfer coefficient was calculated from

$$h = Q/A(T_w - T_\infty) \tag{1}$$

where A is the surface area of the test section and Q was obtained from electrical power measurements. The Nusselt and Rayleigh numbers were then evaluated from

$$Nu = hD/k \tag{2}$$

and

$$Ra = g\beta(T_w - T_\infty)D^3/\alpha\nu \tag{3}$$

Fluid properties were evaluated at the film temperature

$$T_f = (T_w + T_\infty)/2 \tag{4}$$

The results are presented in terms of the Rayleigh number based on cylinder diameter and the dimensionless parameters S/D and H/D. These parameters are depicted to scale in Fig. 5. It may be seen from this figure that all tests were run with the cylinder very close to the plate.

Most of the results are presented in terms of a normalized Nusselt number. By dividing the Nusselt number by the value obtained with an isolated cylinder for the same Rayleigh number, the degree of enhancement or degradation can be easily seen. The isolated cylinder values were obtained with the present equipment for consistency.

Flow Visualization Results

Flow visualization results are presented in Fig. 6 for four cylinder positions. These photographs do not describe the flow with as much detail as is desirable, but do give some qualitative insights. Figures 6a and 6b, at an H/D of 15.5 and 2.07 respectively, show that at the largest spacing tested (S/D = 0.62) the flow from below the cylinder is pulled around through the gap between plate and cylinder. This is due to the flow requirements of the cylinder and the entrainment into the plate boundary layer. In Fig. 6c (H/D = 2.07 and S/D = 0.21) this flow through the gap is diminished and there is more flow into the cylinder plume from the side of the cylinder away from the plate. Figure 6d shows clearly that the approach flow for the cylinder and the boundary layer flow of the plate are diverted around the side of the cylinder away from the plate. In effect, most of the plate boundary layer flow is pinched off and must wash around the cylinder. As penetration of the cylinder into the plate thermal layer increases, the temperature of the fluid washing the cylinder increases and heat transfer degradation is expected. As the height on the plate increases, the thermal and velocity boundary layers thicken, and more elevated temperature fluid is moving upwards. With more fluid moving upwards, more fluid must wash around the cylinder. This description is only qualitative since the presence of the cylinder affects the overall wall flow pattern quite strongly. It was noted however that the interference of the plate boundary layer due to the cylinder was strongest above the cylinder, and the flow below was not as greatly affected. The important question is whether the moving fluid enhances heat transfer, or the elevated temperature of this fluid degrades heat transfer. In order to determine the strengths of these opposing mechanisms, we must look at the quantitative results.

Results

Three series of experiments were performed, starting with the isolated cylinder data. All comparisons with Nu_0 are based on a least squares fit of the data obtained with the present cylinder. The correlation equation used to represent these isolated cylinder data is

$$Nu_0 = 0.895Ra_D^{0.20} \tag{5}$$

These data for the isolated cylinder tests are shown in Fig. 7 and compared to the correlations of Morgan [16] and Churchill and Chu [17]. The results for the present tests fall below the accepted correlations by as much as 13 percent. There are two factors that contribute to this. The testing tank was not designed to do isolated cylinder experiments, and the tests were run at an S/D from the wall of 3.0. Blockage effects from the wall may have degraded the heat transfer by as much as 3 percent. However, it seems most likely that the biggest factor contributing to the difference was that the temperature drop between the thermocouples and the surface of the cylinder was much greater than that calculated. The outer brass sleeves and the inner copper slugs were soldered together, but the thermal resistance was probably higher than was assumed.

The next series of experiments was run to determine the blockage effect due to the presence of a cold wall. These results are shown in

Fig. 8. They indicate that the blockage effect causes a decrease in the heat transfer that is most significant at lower Rayleigh numbers and close spacing. This is expected, since at lower Rayleigh numbers the cylinder thermal boundary layer is thicker. This figure also shows that for spacings greater than S/D = 1.0, the degradation due to blockage is less than 5 percent. This indicates that the isolated cylinder results are only very slightly affected by blockage.

The majority of the tests were done with the plate heated and the cylinder in close proximity to it. The results from these tests appear in Figs. 9-16. Figure 9 shows typical raw data versus Rayleigh number for H/D = 2.07. Several values of the dimensionless cylinder to wall spacing are presented. In general, these data show that cylinder heat transfer is degraded by the presence of a heated wall except at certain spacings and Rayleigh numbers. Heat transfer degradation is greatest at lower Rayleigh numbers and close spacing. This figure also indicates an enhancement peak at moderate S/D. In appendix A, functional uncertainty of the Nusselt number is calculated based on the method of Kline and McClintock [18]. It is shown that the uncertainty due to measurement error for these data varies from 25% at the lowest Rayleigh number to less than 5% at the highest. The remainder of the raw data are plotted in Figs. 18-26 in appendix B.

To get a clearer understanding of how the heat transfer varies with the parameters, correlations of Nusselt number versus Rayleigh number were worked out for each of the 26 H/D and S/D pairs. Nusselt numbers were then calculated for four chosen Rayleigh numbers at each of the 26 positions. Figures 10-12 show the variation of Nu/Nu_0 versus S/D for the three H/D positions respectively, parameterized by the four Rayleigh numbers. The symbols represent the calculated points, not the raw data.

There are some features of these curves that are similar, while they display differences as well. The heat transfer relative to an isolated cylinder tends to increase with decreasing S/D, then drop sharply for very close spacings. Also, the heat transfer is greatest at the higher Rayleigh numbers everywhere except at the larger S/D in Fig. 10. There are some differences as well. At the lowest position (Fig. 10) the heat transfer is enhanced except at small S/D, while at the highest position (Fig. 12) the heat transfer is generally degraded. The heat transfer at the middle position (Fig. 11) falls between the other two, indicating a general trend of lower heat transfer with greater height. The other significant difference is that at the highest position (Fig. 12) there is a high dependence on Rayleigh number, while in Figs. 10 and 11 the curves for different Rayleigh numbers tend to collapse together.

These similarities and differences can be explained in terms of three identifiable mechanisms. Referring back to Figs. 1 and 2, we see that the thicknesses of the thermal and velocity boundary layers of the heated plate decrease with increasing Rayleigh number and with decreasing height from the plate leading edge. The cylinder boundary layers become thinner with decreasing Rayleigh number as well. The trends from Figs. 10-12 show an increase in cylinder heat transfer with decreasing height, and an increase with increasing Rayleigh number. Obviously, as the thermal boundary layers of the plate and cylinder thicken, the cylinder heat transfer is degraded. The peaks of Nu/Nu_0 at moderate S/D show that the highest heat transfer can be obtained by positioning the cylinder in the velocity boundary layer outside of the thermal boundary layer. These peaks are strongest at the greater heights and higher Rayleigh numbers where the velocity in the plate boundary layer is highest, and the plate thermal boundary layer is relatively thin. The high peaks in Figs. 11 and 12 correspond with the plate boundary layer becoming turbulent. Turbulent flow would be expected to increase the strength of the cylinder velocity boundary layer and plume by the transfer of momentum by eddy motion.

The sharp drop off in heat transfer exhibited by all of the figures at small S/D is a direct consequence of penetration by the cylinder of the plate thermal boundary layer. As was seen in flow visualization Fig. 6d, the flow is pinched off and must divert around the side of the cylinder away from the plate. This washes the cylinder in fluid that is at an elevated temperature with respect to the ambient, causing a decrease in heat transfer. It was also observed that the heat transfer at H/D = 2.07 (Fig. 10) was generally enhanced as compared to H/D = 7.44 and 15.5 (Figs. 11,12). Since the plate boundary layers are small at H/D = 2.07, there is much less fluid moving upwards. This means that there is not so much of the elevated temperature fluid to wash the cylinder. However, the entrainment flow into the plate boundary layer is much stronger at the lower position than at a higher position ($u = H^{-1/4}$ Bejan [15]) so that it washes the cylinder with ambient temperature fluid and enhances the cooling effect. Finally, the figures show that there is a strong Rayleigh number dependence only at the highest position (Fig. 12). This can partly be explained by the transition to turbulence of the plate boundary layer at a cylinder Rayleigh number of 3,500,000 ($Ra_H = Ra_D(H/D)^3$) which occurs at the highest position (H/D = 15.5). Also, the plate boundary layers are most developed at the highest position and would be expected to have the strongest effect.

In Figs. 13-16 plots of the same Nu/Nu_0 versus S/D data are presented with a different parameterization. Each of these figures shows very clearly that the cylinder heat transfer decreases with increasing height on the plate as discussed previously. They also show that the dependence on height is strongest at the lowest Rayleigh number where the heat transfer increases most dramatically with decreasing H/D (Fig. 13). This dependence weakens with increasing Rayleigh number as shown in Figs. 14-16 where the curves tend to collapse together.

These trends are a result of the same mechanisms discussed above. The height dependence is due to the thickening of the plate thermal boundary layer and the variation in the entrainment flow with height. The reason that this dependence weakens with increasing Rayleigh number is that both the plate and cylinder boundary layers are relatively thin and strong at higher Rayleigh numbers. This would tend to allow them to maintain their independent identities and not interact with as strong an effect. It should be noted that when S/D becomes small, the height dependence becomes much less pronounced. At these small spacings the nature of the flow is less height dependent and the plate and cylinder flows interact to affect the cylinder heat transfer.

We have seen how the various parameters influence the heat transfer from a heated cylinder parallel to a heated plate.

Basically, the effects are a competition between the enhancing mechanism of the plate velocity boundary layer and the degrading mechanism of the plate thermal boundary layer. The thickness of the plate and cylinder boundary layers tend to dictate the outcome of this competition, with thicker thermal boundary layers dominating to cause heat transfer degradation. The thickness of the boundary layers is determined by Rayleigh number and height on the plate. Horizontal spacing (S/D) is the other critical variable, with small spacings causing degradation independently of boundary layer thickness, and moderate to large spacings causing heat transfer variations that are more dependent on the other variables; height and Rayleigh number.

CONCLUSIONS

This paper has presented the results of an experimental study of heat transfer from a horizontal cylinder parallel to a heated vertical wall. The parameters examined were cylinder Rayleigh number, height from the plate leading edge, and horizontal distance from the plate. The boundary conditions were all isothermal, and the cylinder and plate were maintained at equal temperatures. All work was done in water at Rayleigh numbers from 400,000 to 10,000,000.

It was shown that each of the above mentioned parameters had a significant effect on cylinder heat transfer as compared to the equivalent heat transfer of a cylinder in an infinite reservoir at the same Rayleigh number. The most pronounced effect was due to the cylinder wall spacing S/D. At very close spacing ($S/D < 0.2$) the heat transfer drops off sharply, due to a combination of the blockage effects and immersion in the plate thermal boundary layer. This effect was strongest at the lower Rayleigh numbers where the plate boundary layer was thickest. The effects due to height from the plate leading edge were to decrease the heat transfer with increased height. This was due to the decrease in the entrainment flow and the thickening of the thermal boundary layer with height. Overall, the competition between the enhancement due to the plate velocity boundary layer and the degradation due to the plate thermal boundary layer was decided by the thickness of the thermal boundary layer. This thickness depended on Rayleigh number and height of the cylinder from the leading edge of the plate. The Rayleigh number effects at any given height were small except at the highest position ($H/D = 15.5$)

where the boundary layer was thickest and there was some transition into turbulent flow. For a given Rayleigh number, though, the differences associated with height were quite strong.

In terms of design results, it was shown that the heat transfer did change with the parameters in such a way as to indicate optimum positioning. For a given Rayleigh number and height, the Nusselt number tended to increase slowly to a maximum and then drop off sharply with decreasing S/D. Obviously, if cooling is desired, a cylinder should be placed to the side of the peak away from the plate. These results give indication of general trends and show that beneficial results can be obtained for more application oriented research.

Future work should concentrate on extending and clarifying these results. A further parameter that needs to be investigated is cylinder diameter. The ratio of cylinder diameter to plate boundary layer thickness is sure to be very important. For this study a fairly large cylinder was used, but for the same Rayleigh number range, a different diameter could show very different results. Besides extending these results, other research opportunities include; plate to cylinder temperature imbalance, end effects for small length to diameter ratio cylinders, inclined plate effects and the effect of further enclosing walls.

III. Recommendations

This thesis represents a small part of the possible research involving the heat transfer from a cylinder parallel to a heated wall. There are many other areas which can be studied with the same equipment as well as with modifications of this equipment. This work has shown how the heat transfer from a horizontal cylinder in water varies with cylinder Rayleigh number, height from the wall leading edge, H, and distance from the wall, S. To begin with, further work with these parameters is still possible. Also, more extensive flow visualization could be employed to gain a more precise understanding of the mechanisms. Extensions to the Rayleigh number range could be accomplished with different fluids. Other blockage effects could be easily explored by lowering another plate into the test tank.

With some modifications, the research possibilities expand even further. By building different cylinders, other diameters could be investigated as well as end effects for shorter cylinders. A vertical cylinder is another configuration with some application to electronic component cooling. Multiple cylinder arrangements provide a vast area for further work.

The work done for this thesis was only minimally supported so that much promising and some necessary work could not be accomplished. In order to continue to do good work with this equipment, this author believes that there are several improvements needed. The formation of air bubbles on the heat transfer surface was a problem throughout the work. Although there was a degassing tank, it had a capacity of only half that of the test tank so that complete degassing was very difficult and time consuming. Furthermore, this tank was very rusty and contaminated the water with each use. Another problem was the heated plate itself which was constructed of aluminum. The aluminum surface corroded in the water which contributed to further contamination, requiring frequent, time consuming, water changes. The use of tap water is not desirable either, particularly for the flow visualization work. A distilled water supply would be very desirable.

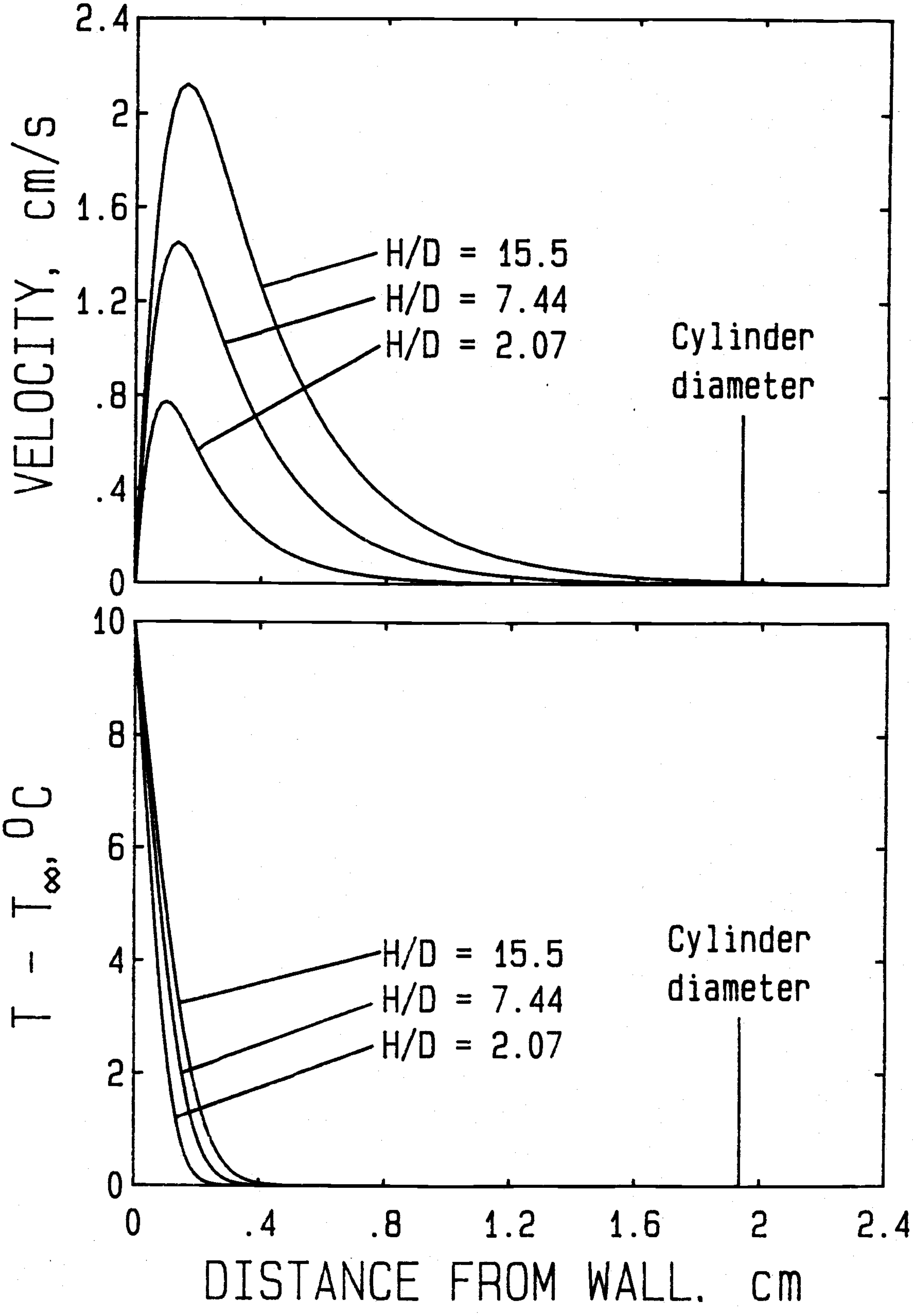


FIGURE 1 Velocity and temperature profiles for the vertical plate boundary layer for $T_w - T_\infty = 10^oC$

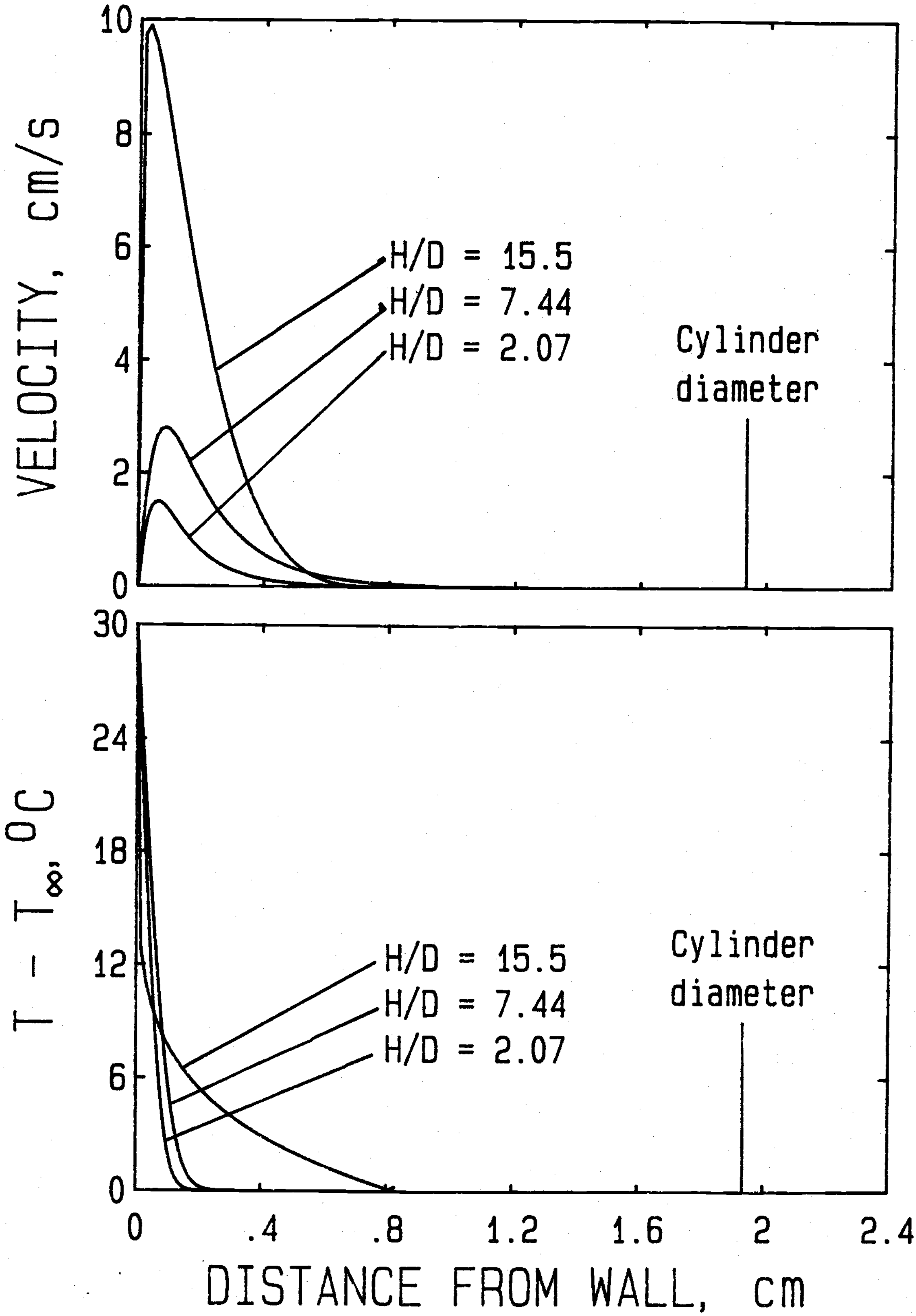


FIGURE 2 Velocity and temperature profiles for the vertical plate boundary layer for $T_w - T_\infty = 30^oC$

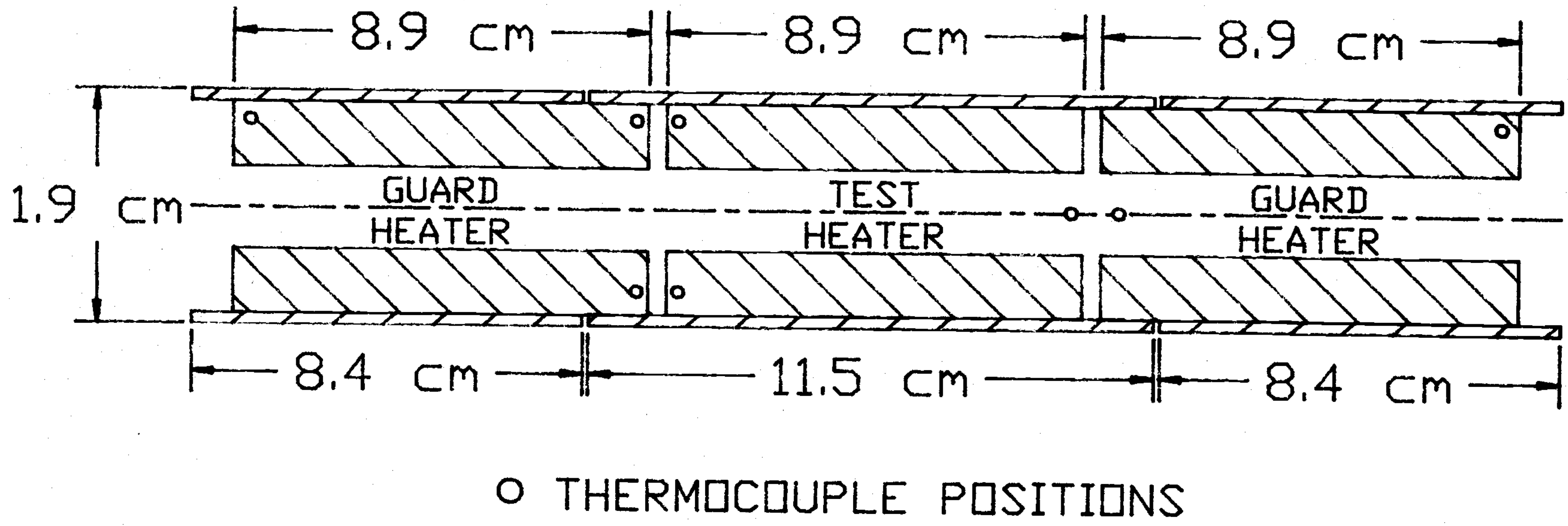


FIGURE 3 Test cylinder

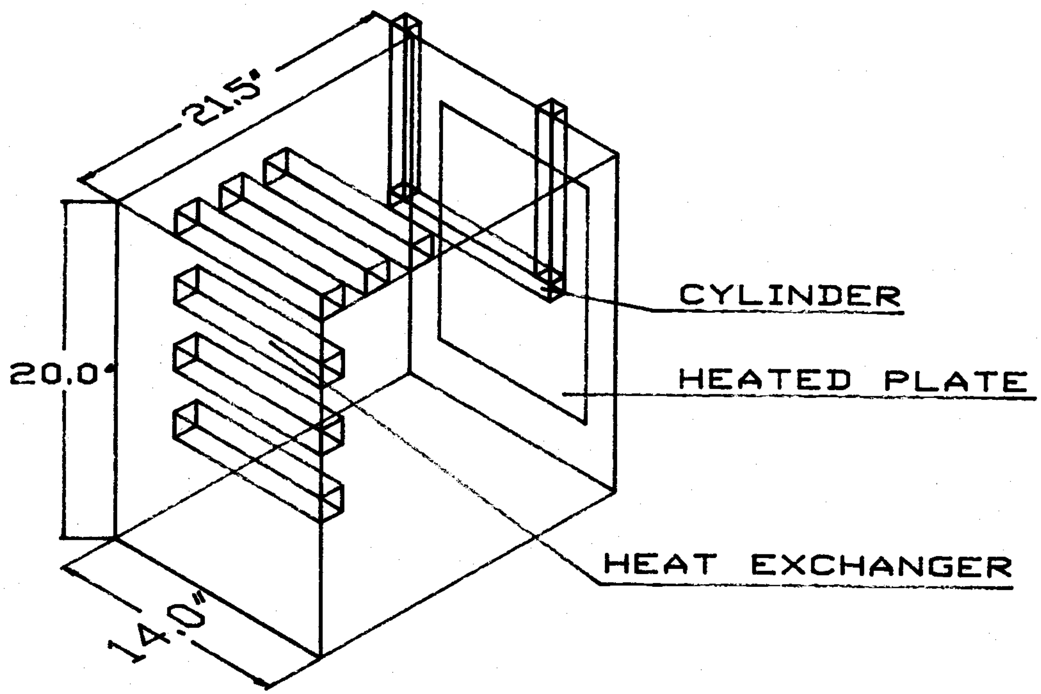


FIGURE 4 Test tank

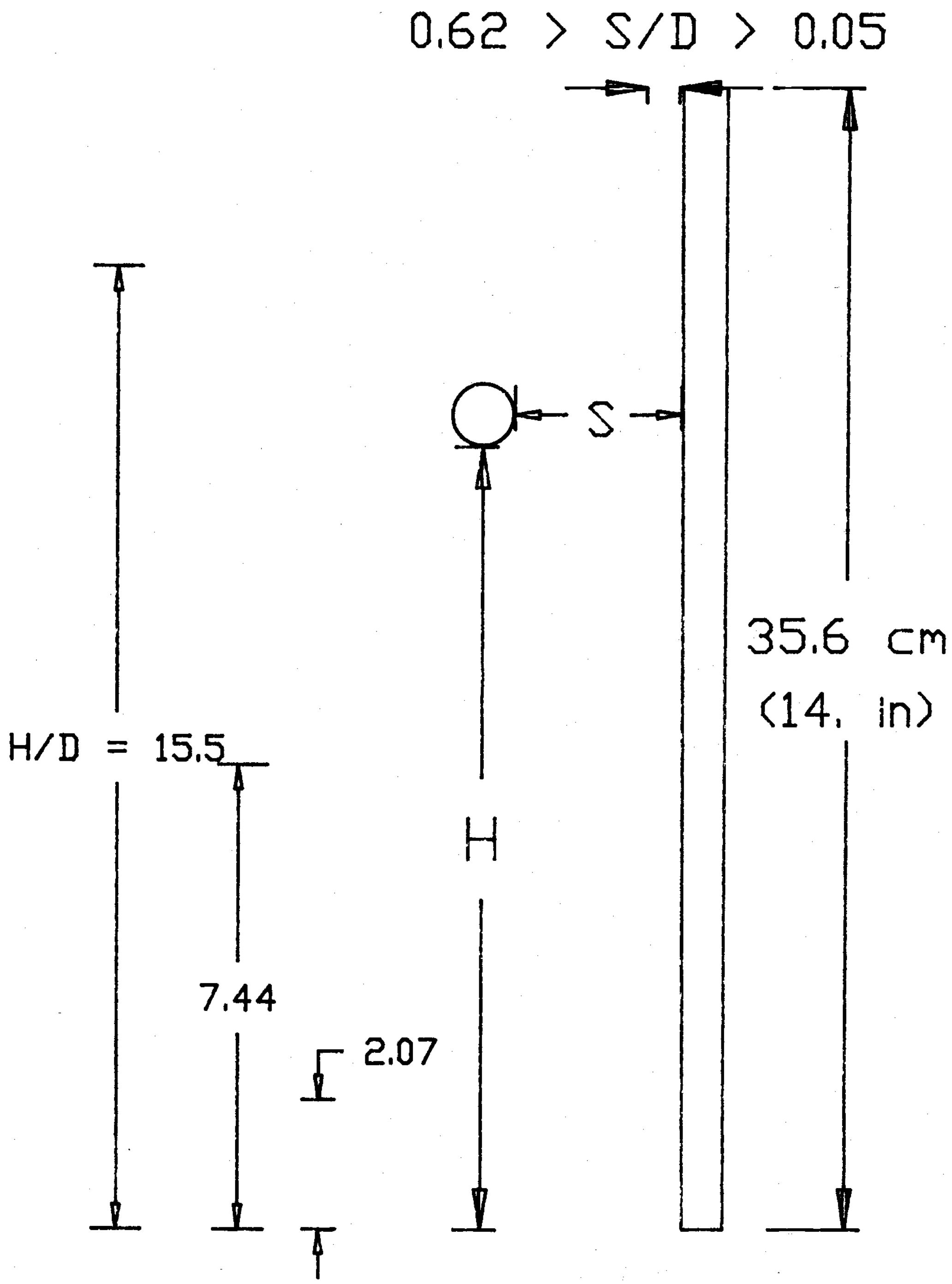
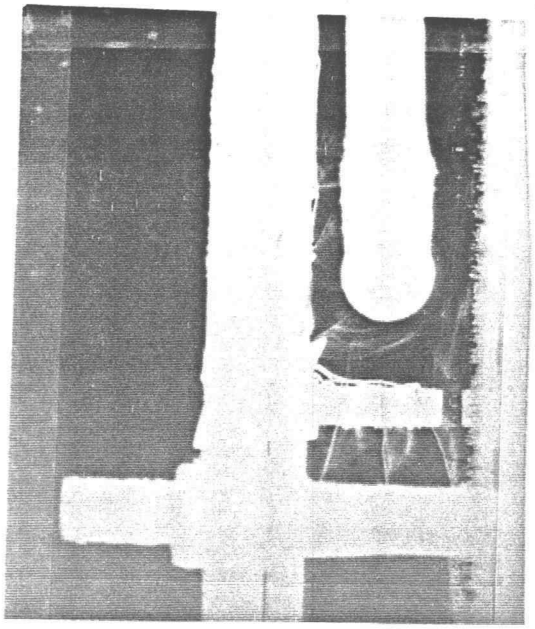
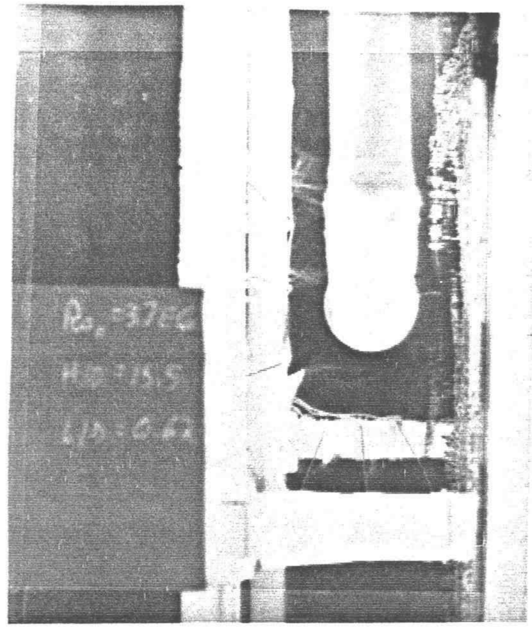


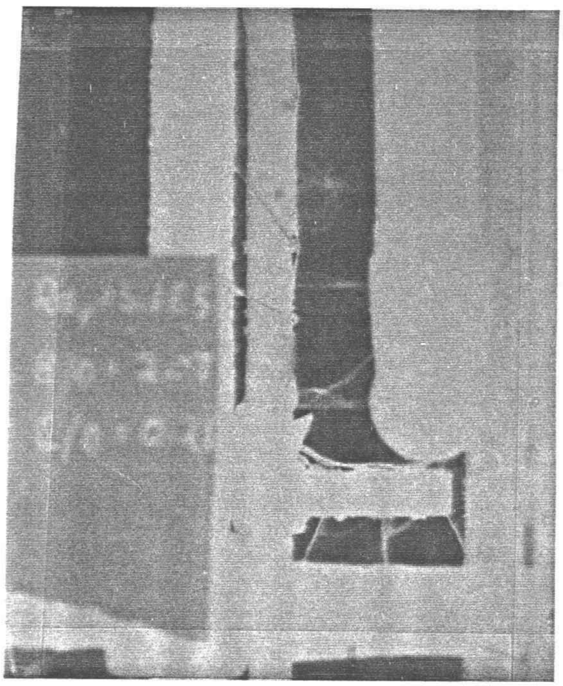
FIGURE 5 Test positions and notation



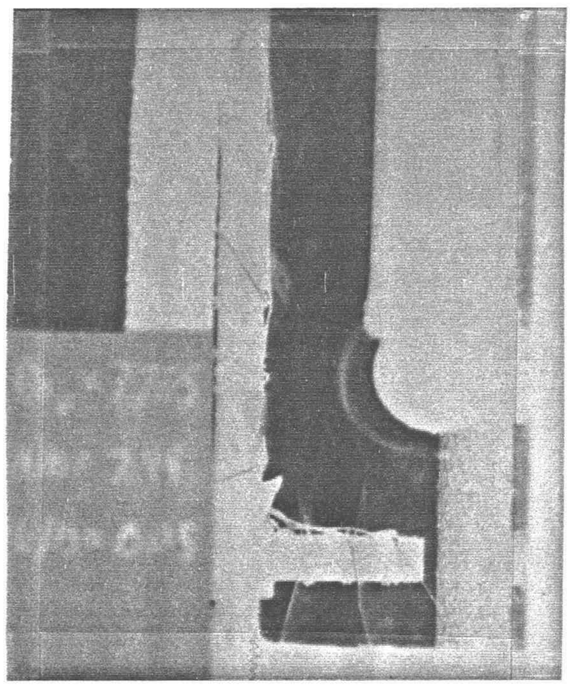
a) Ra = 3.3E6 H/D = 2.07
S/D = 0.62



b) Ra = 3.7E6 H/D = 15.5
S/D = 0.62



c) Ra = 5.1E5 H/D = 2.07
S/D = 0.21



d) Ra = 7.7E5 H/D = 7.44
S/D = 0.05

FIGURE 6 Flow visualization for four typical situations

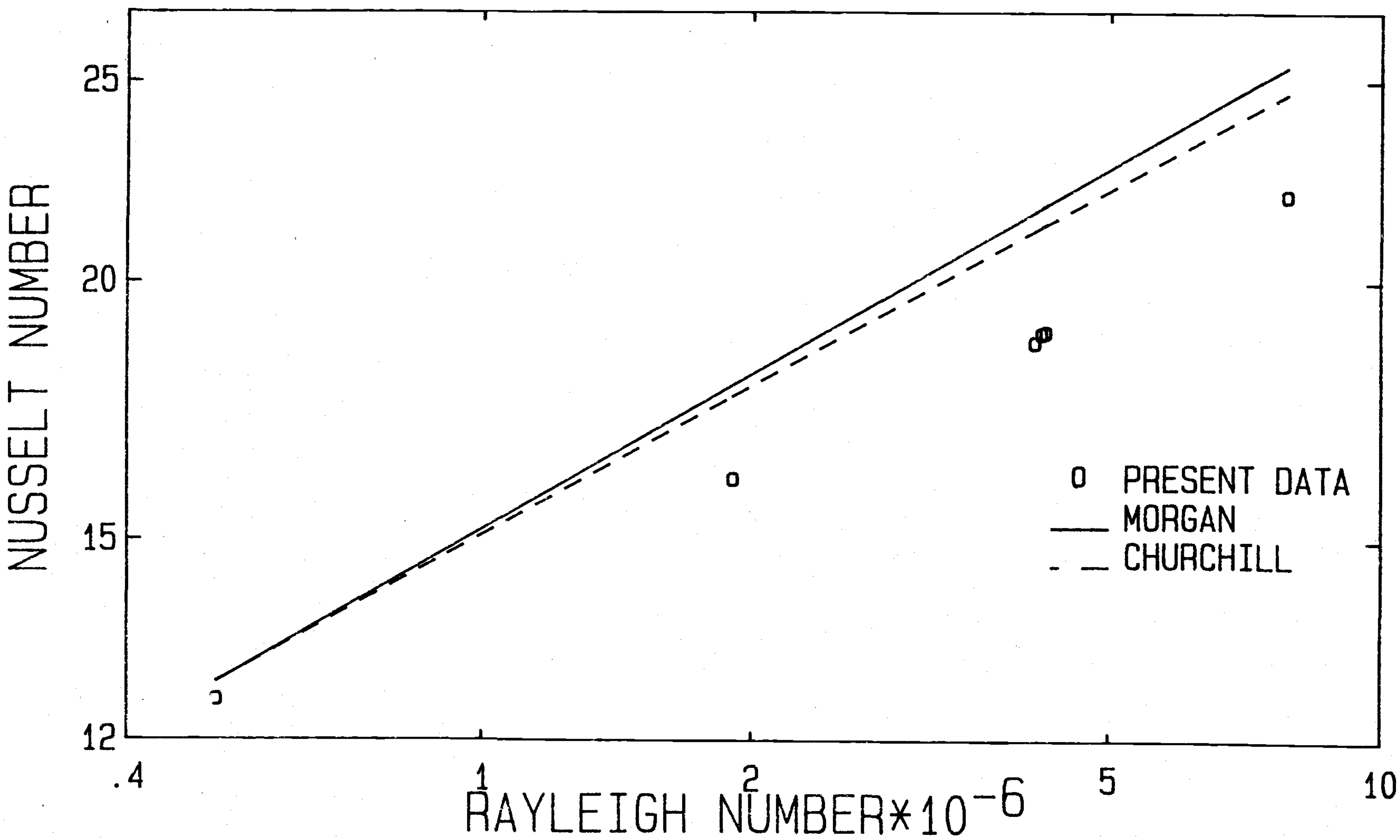


FIGURE 7 Comparison of single cylinder Nusselt number results with established correlations

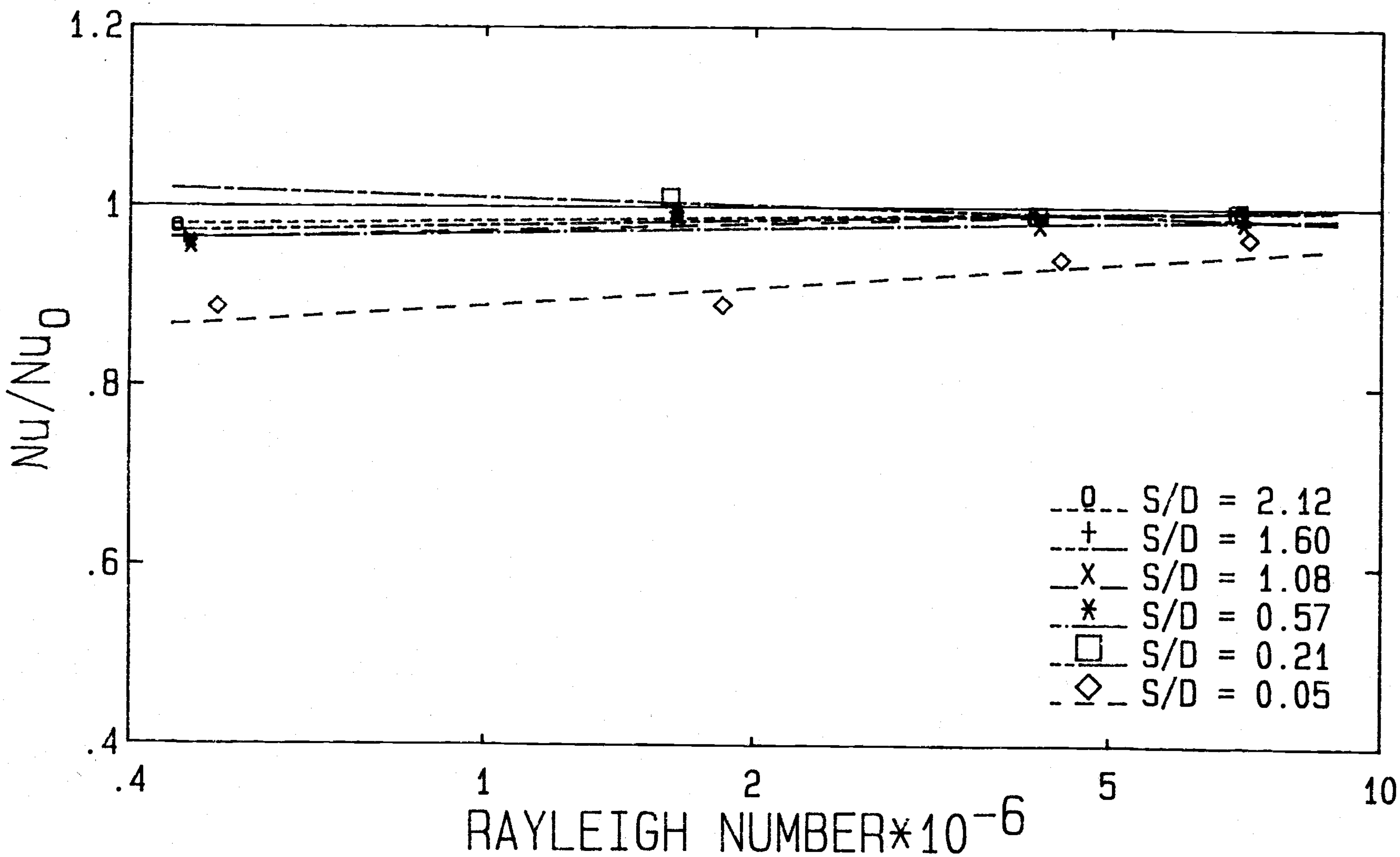


FIGURE 8 Effect of the presence of an unheated wall on cylinder Nusselt number at an H/D of 7.44

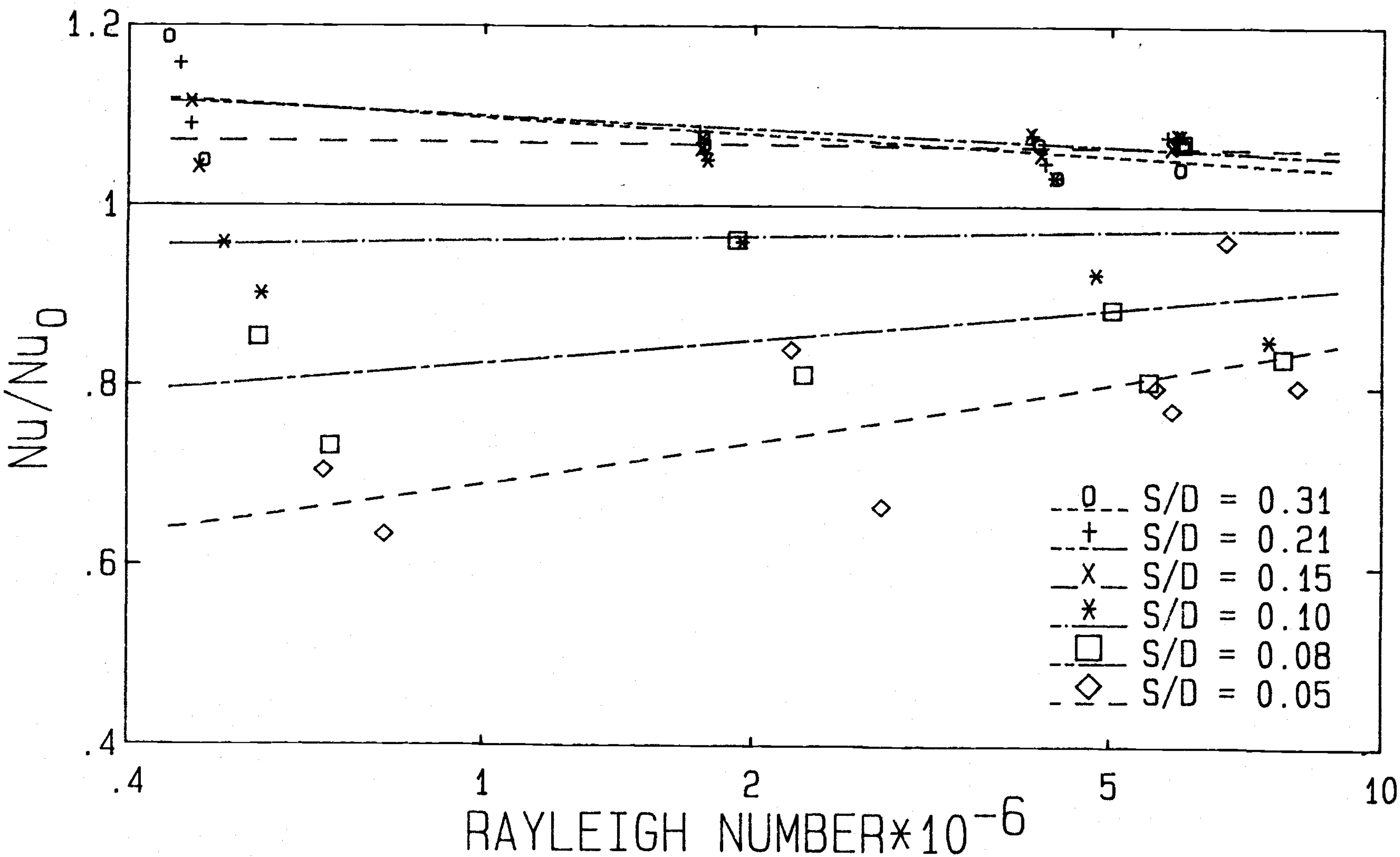


FIGURE 9 Effect of heated wall on cylinder Nusselt number at an H/D of 2.07

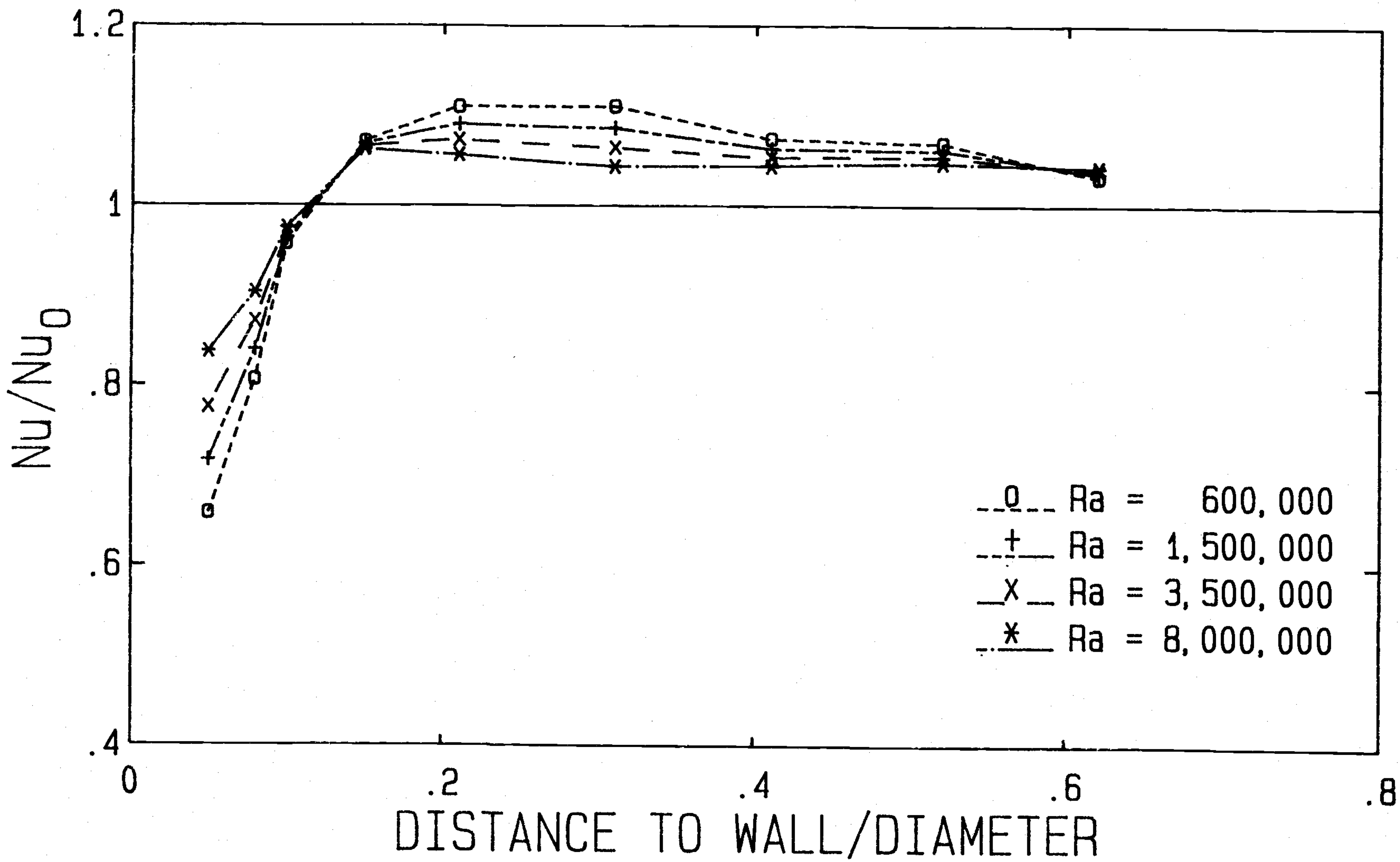


FIGURE 10 Effect of wall to cylinder spacing on cylinder Nusselt number for H/D = 2.07

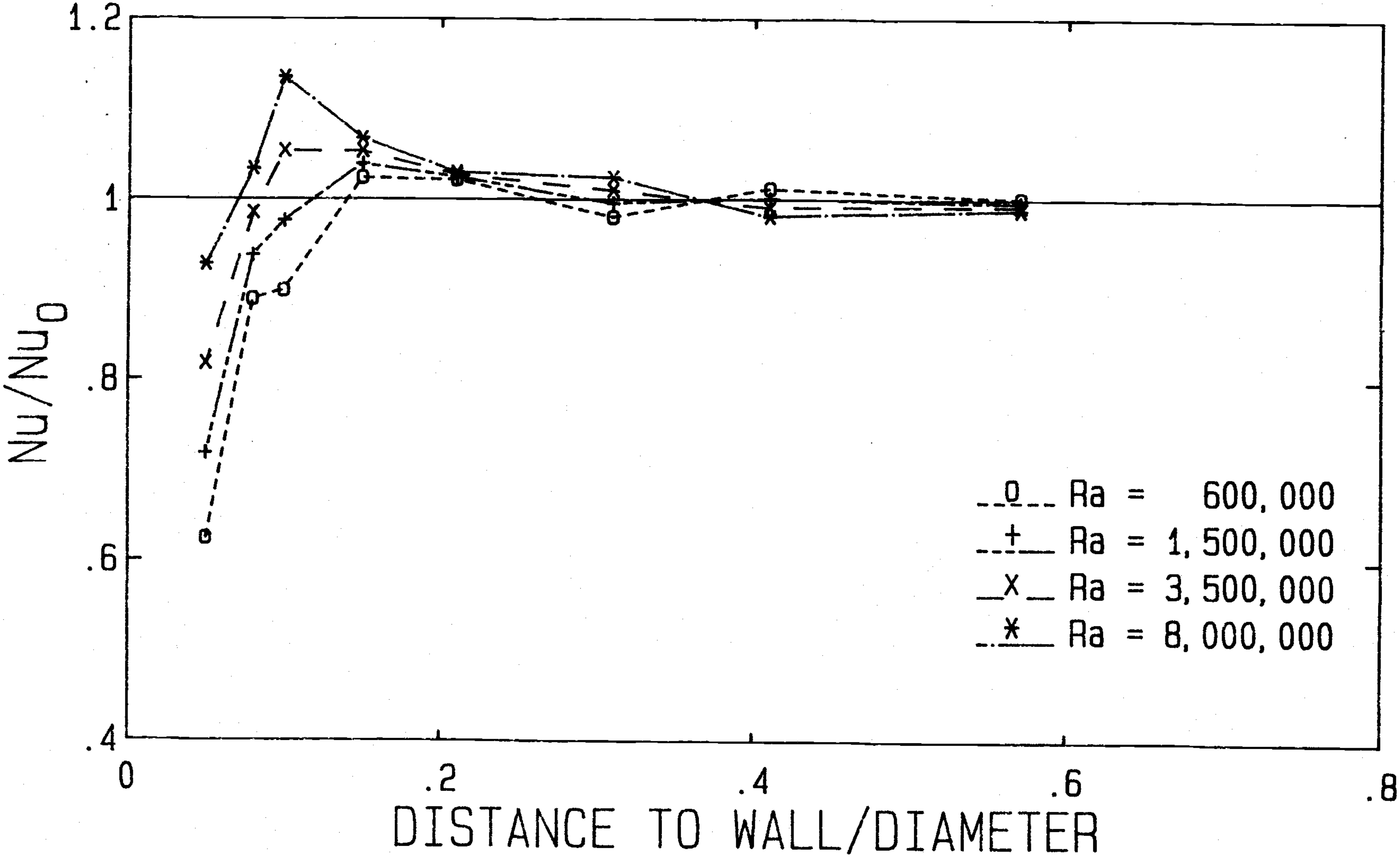


FIGURE 11 Effect of wall to cylinder spacing on cylinder Nusselt number for H/D = 7.44

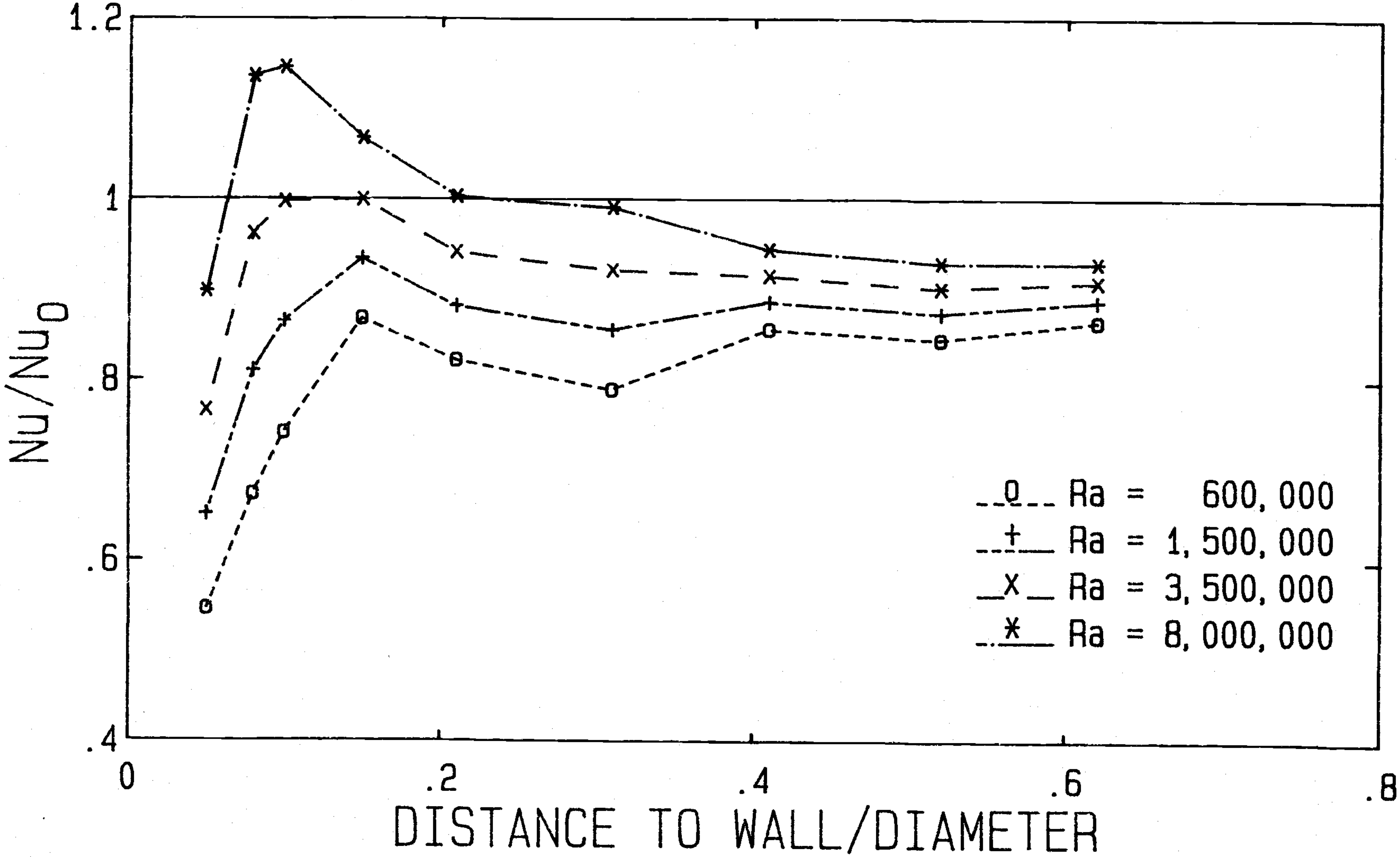


FIGURE 12 Effect of wall to cylinder spacing on cylinder Nusselt number for H/D = 15.5

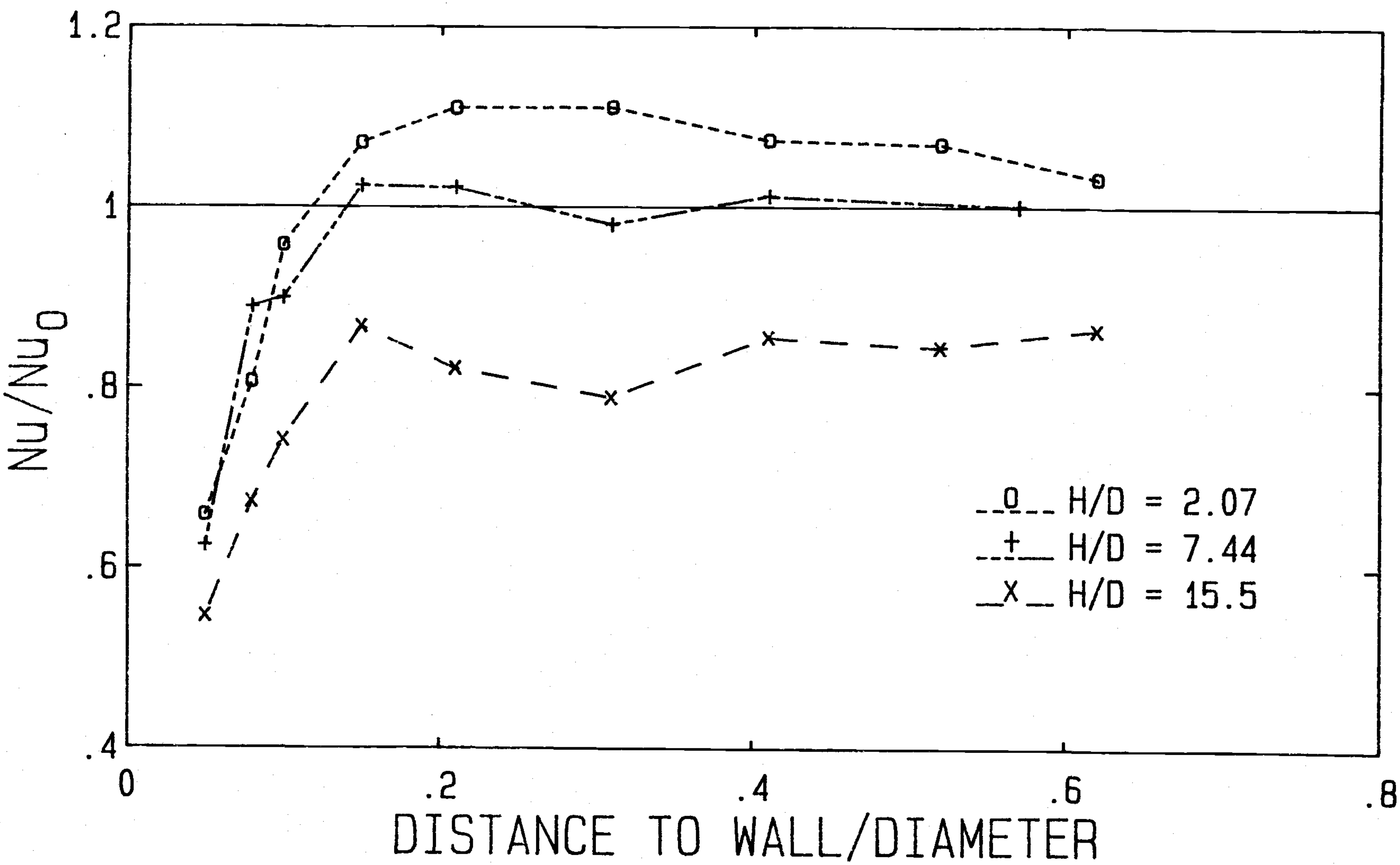


FIGURE 13 Effect of wall to cylinder spacing on cylinder Nusselt number for Ra = 600,000

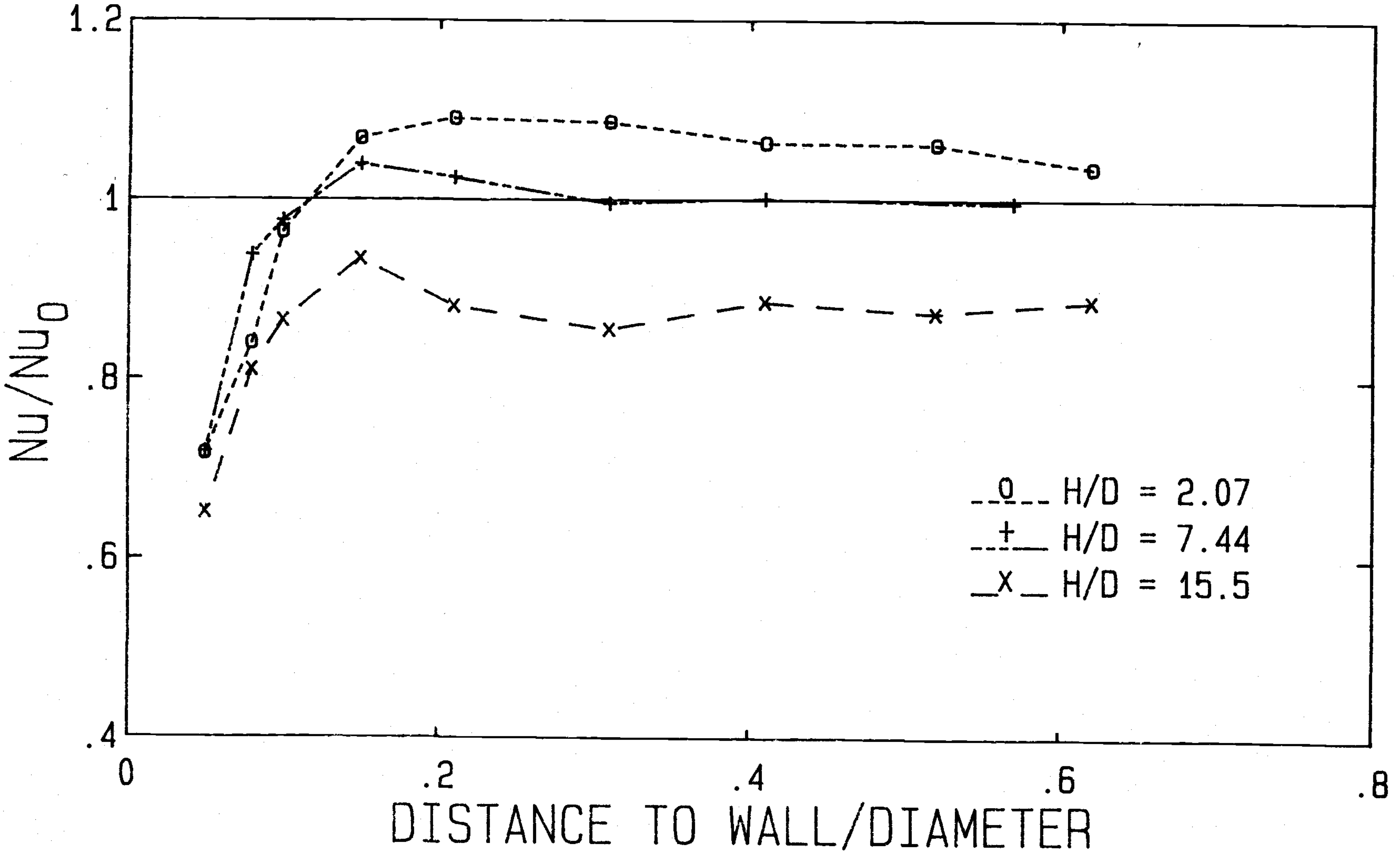


FIGURE 14 Effect of wall to cylinder spacing on cylinder Nusselt number for Ra = 1,500,000

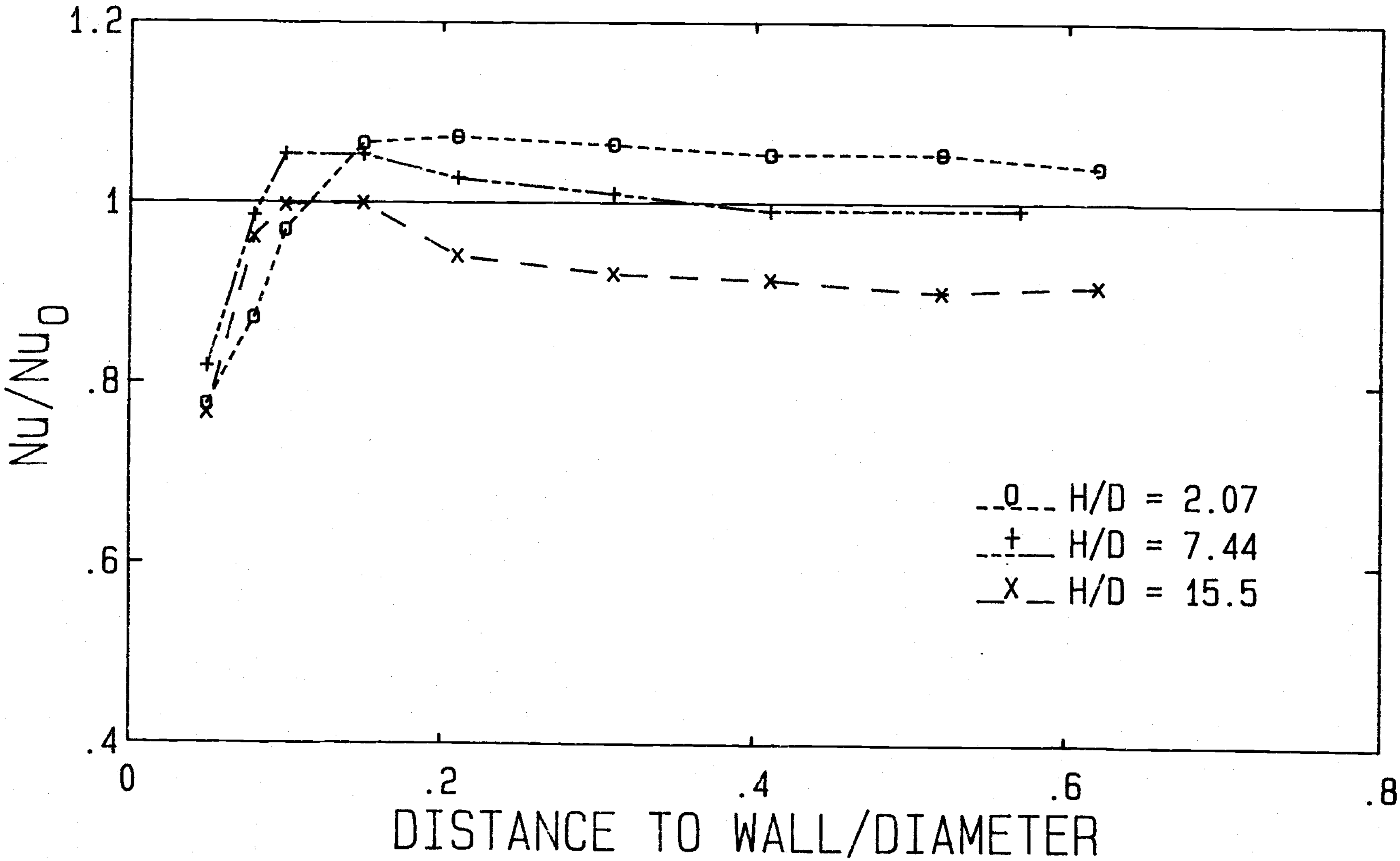


FIGURE 15 Effect of wall to cylinder spacing on cylinder Nusselt number for Ra = 3,500,000

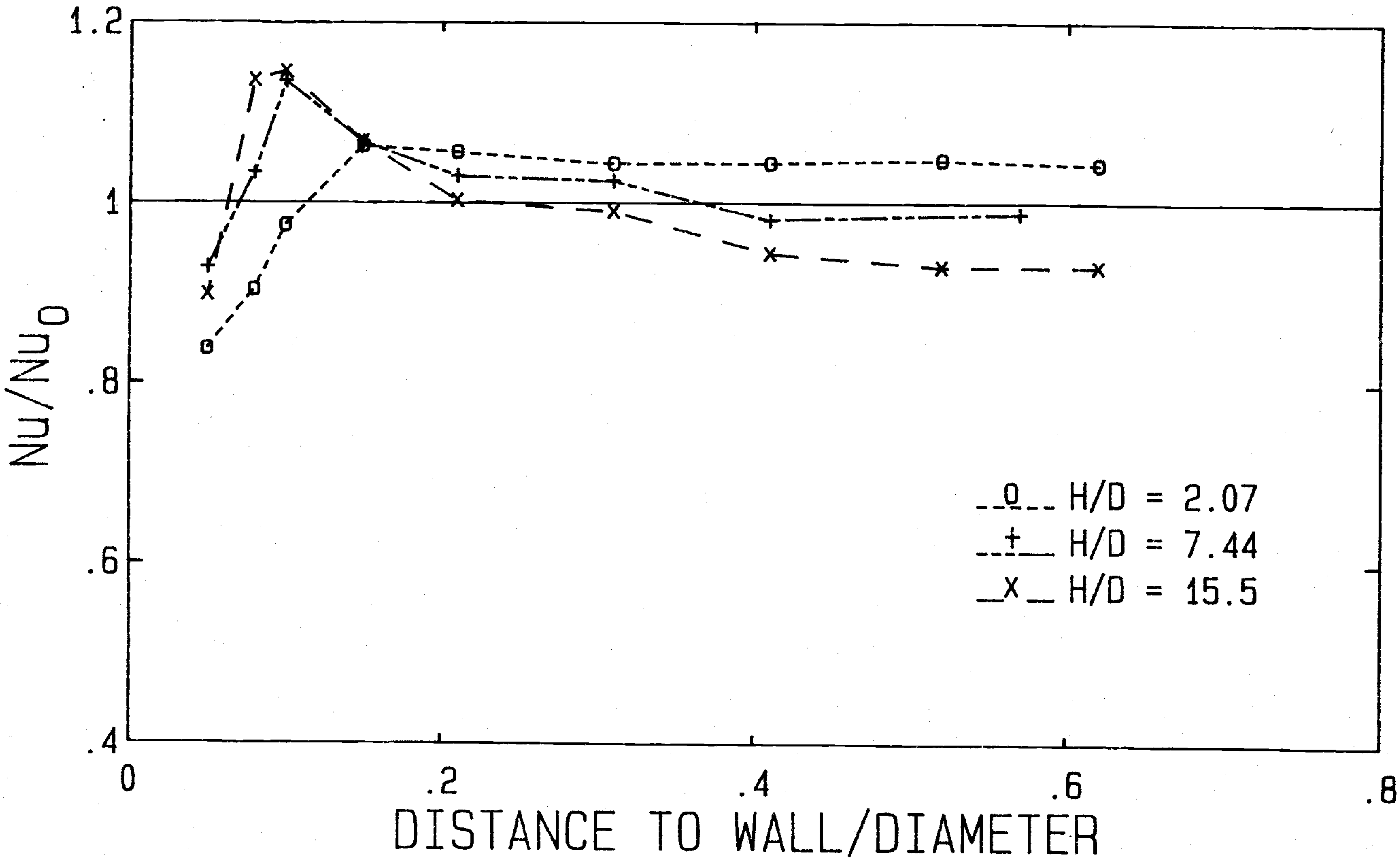


FIGURE 16 Effect of wall to cylinder spacing on cylinder Nusselt number for Ra = 8,000,000

NOMENCLATURE

A	Surface area of the cylinder test section
D	Cylinder diameter
g	Gravitational acceleration
h	Convective heat transfer coefficient
H	Height from plate leading edge to cylinder bottom
k	Thermal conductivity
L	Cylinder length
Nu	Cylinder Nusselt number
Nu_0	Isolated cylinder Nusselt number
Q	Power input to test heater
Ra_D	Rayleigh number based on cylinder diameter
Ra_H	Rayleigh number based on plate height
S	Distance between plate and cylinder
T_w	Cylinder and plate surface temperature
T_∞	Water ambient temperature
T_f	Film temperature for property evaluation
u	Horizontal velocity
α	Thermal diffusivity
β	Coefficient of thermal expansion
ν	Kinematic viscosity

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APPENDICES

APPENDIX A

Uncertainty Analysis

This appendix will evaluate the functional uncertainty of the data based on the uncertainty associated with the individual instruments as proposed by Kline and McClintock [18]. The Nusselt number was evaluated from

$$Nu = hD/k \tag{A.1}$$

where D is the cylinder diameter, k the conductivity of the water and h given by

$$h = IV/A(T_w - T_\infty) \tag{A.2}$$

I is the current through the heater and V is the voltage across it. A is the surface area of the test section, T_w is the surface temperature and T_∞ is the ambient temperature of the water. Combining equations A.1 and A.2 yields

$$Nu = IV/kL(T_w - T_\infty) \tag{A.3}$$

for the Nusselt number where L is the length of the test section.

The uncertainty is evaluated by differentiating the dependent variable with respect to each of the independent variables and putting the results into the expression

$$w_y = [(w_{x1}\delta y/\delta x1)^2 + (w_{x2}\delta y/\delta x2)^2 + \ldots\ ]^{1/2} \qquad \text{(A.3)}$$

for uncertainty of y(x1,x2,...).

This analysis yields an uncertainty in the Nusselt number that varies with the Rayleigh number as shown in figure 17. As expected, the uncertainty is highest at lower Rayleigh numbers (smaller temperature difference). The uncertainty decreases as the temperature difference increases.

This analysis does not reflect application errors. Some of these include air bubble formation on the test surface, errors in position measurement, and contamination of the water in the test tank. Care was taken to keep these factors to a minimum, but they could not always be completely eliminated.

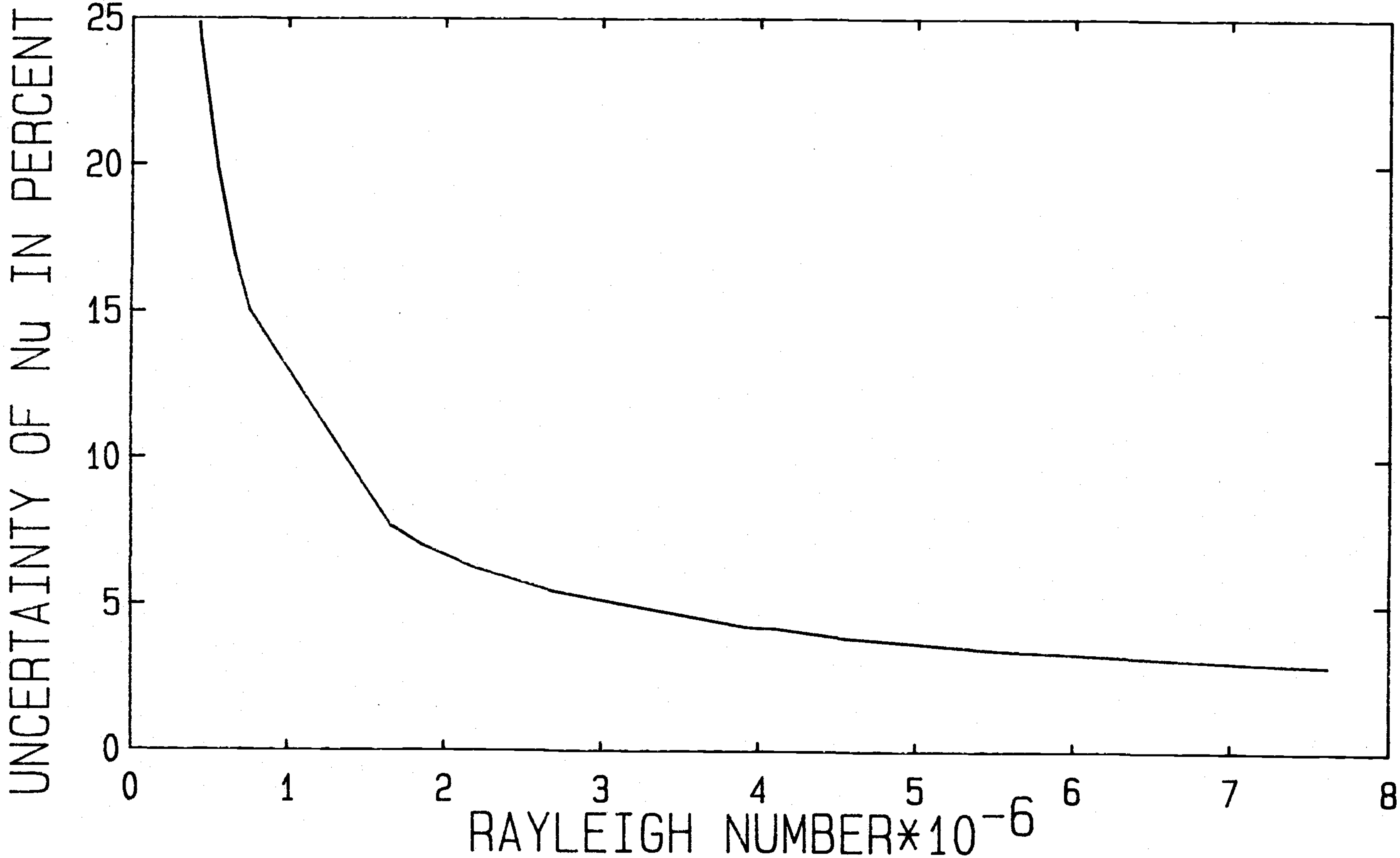

FIGURE 17 Uncertainty of Nusselt number vs Rayleigh number for typical test data

APPENDIX B

Raw Data Plots

Figures 18-26 are plots of the raw data with various parameterizations. Figures 18 and 19 show the variation of Nu/Nu_o with Rayleigh number at H/D = 7.44 and 15.5 for S/D from 0.05 to 0.31. Figures 20-26 show the variation of Nu/Nuo with Rayleigh number at S/D = 0.41, 0.31, 0.21, 0.15, 0.10, 0.08 and 0.05 with a parameter of H/D.

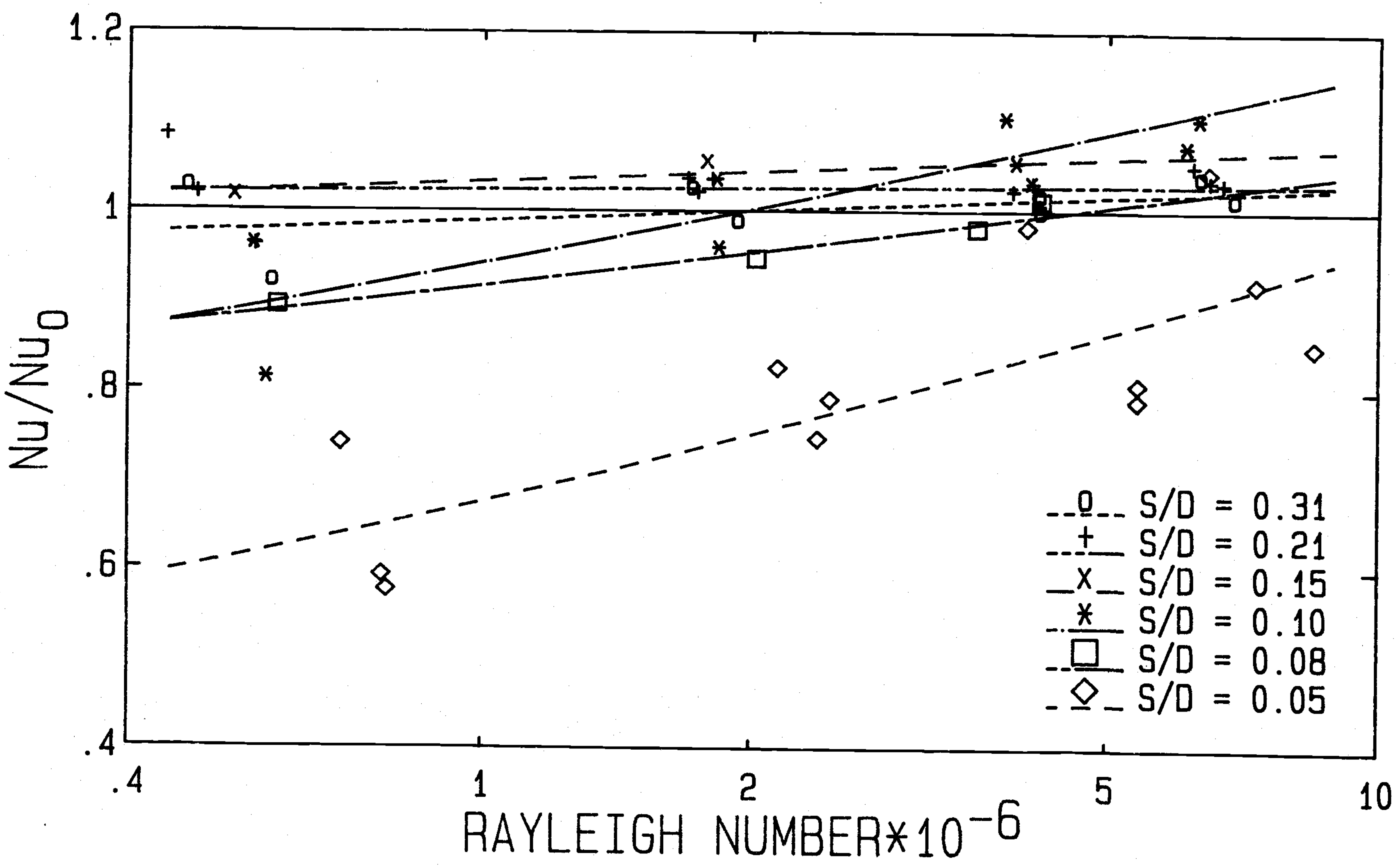

FIGURE 18 Effect of heated wall on cylinder Nusselt number at an H/D of 7.44

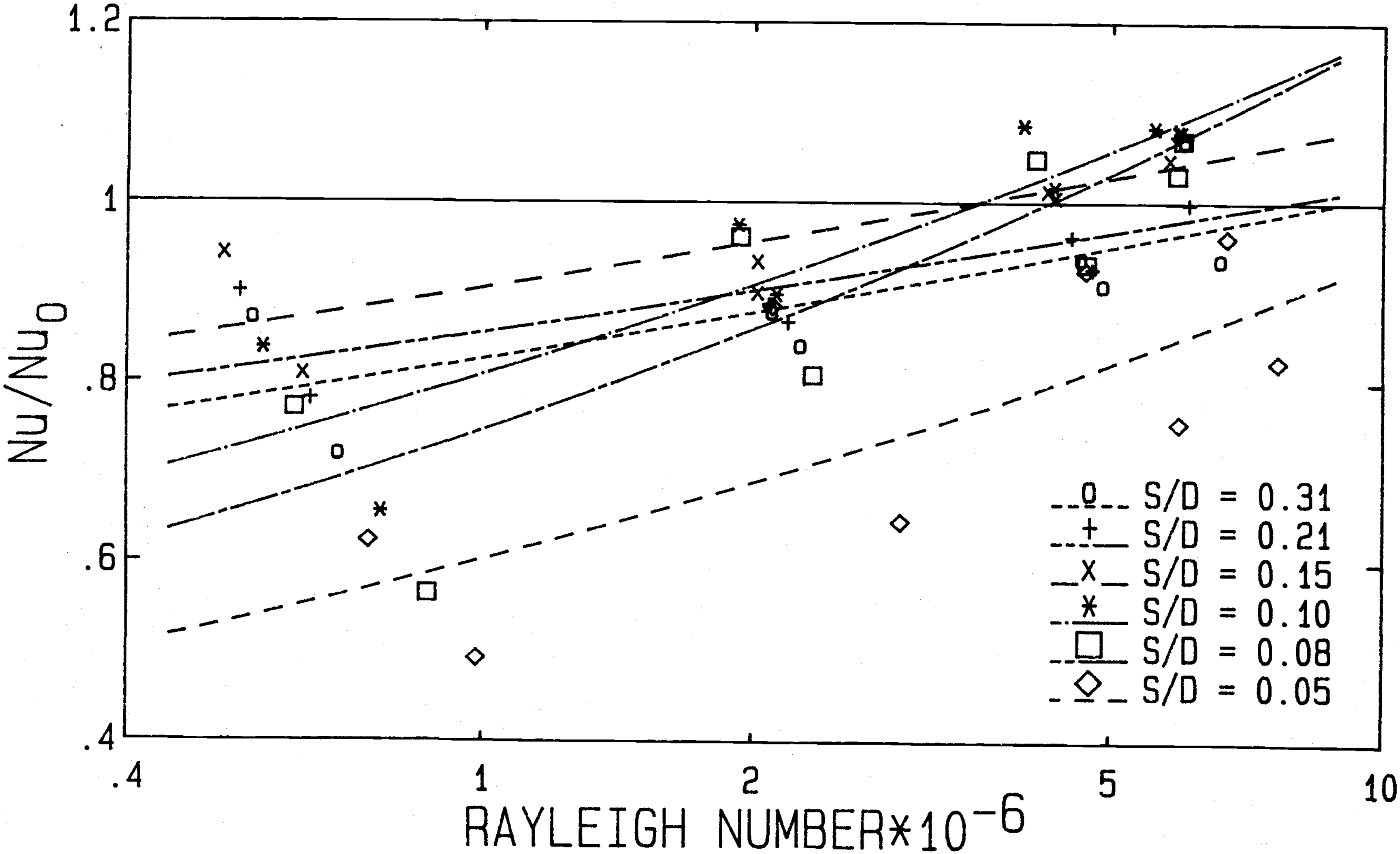

FIGURE 19 Effect of heated wall on cylinder Nusselt number at an H/D of 15.5

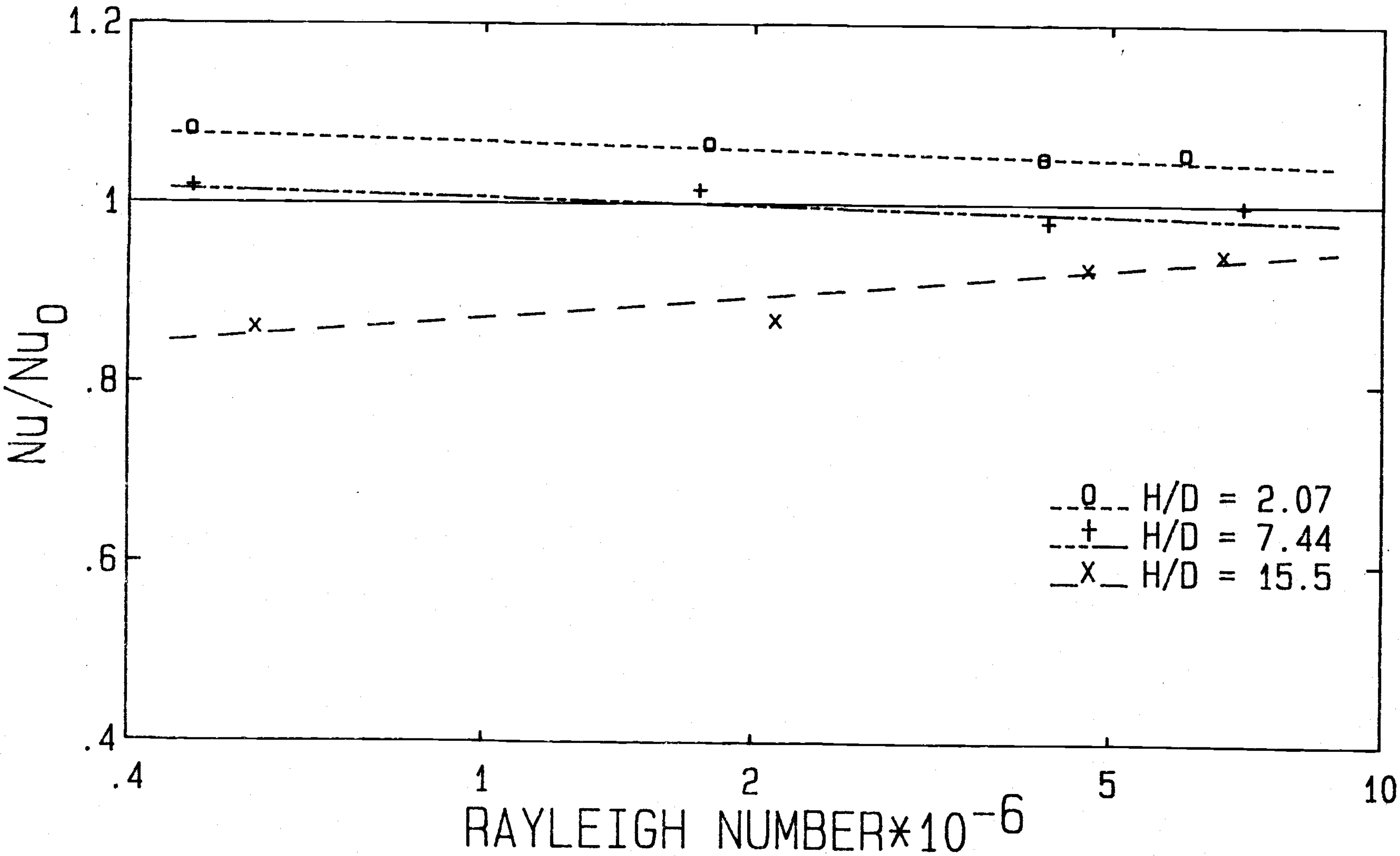

FIGURE 20 Effect of heated wall on cylinder Nusselt number at an S/D of 0.41

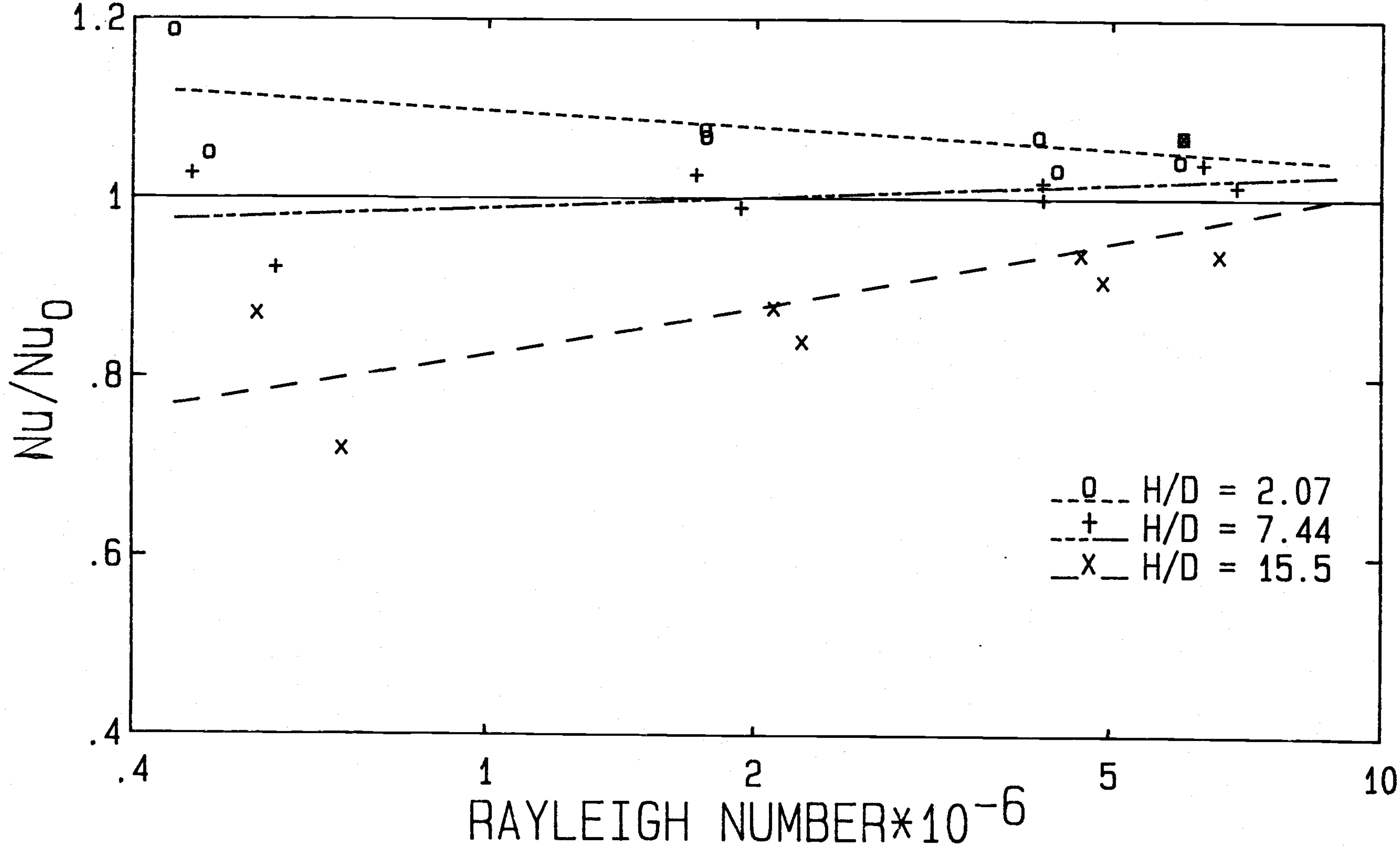

FIGURE 21 Effect of heated wall on cylinder Nusselt number at an S/D of 0.31

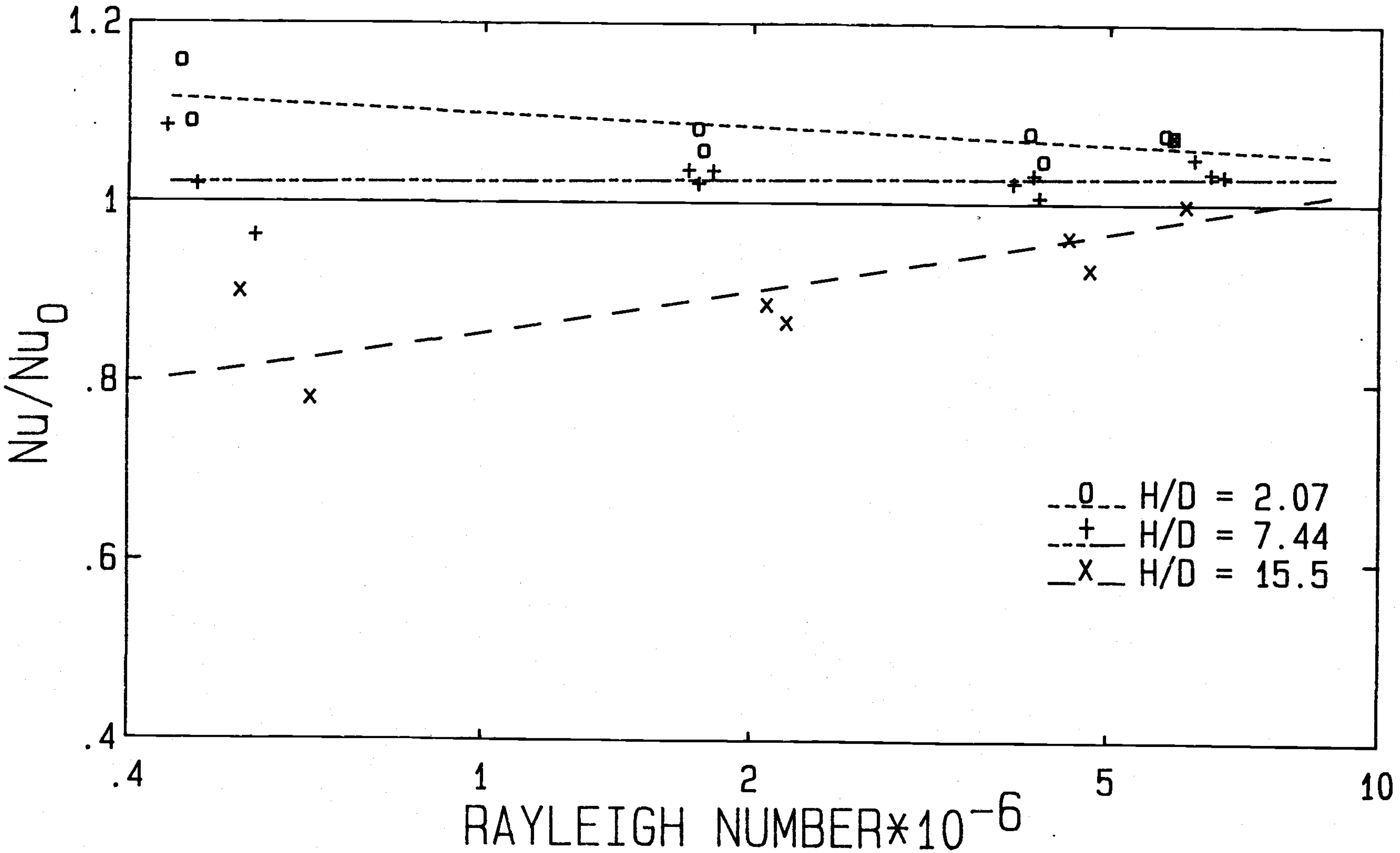

FIGURE 22 Effect of heated wall on cylinder Nusselt number at an S/D of 0.21

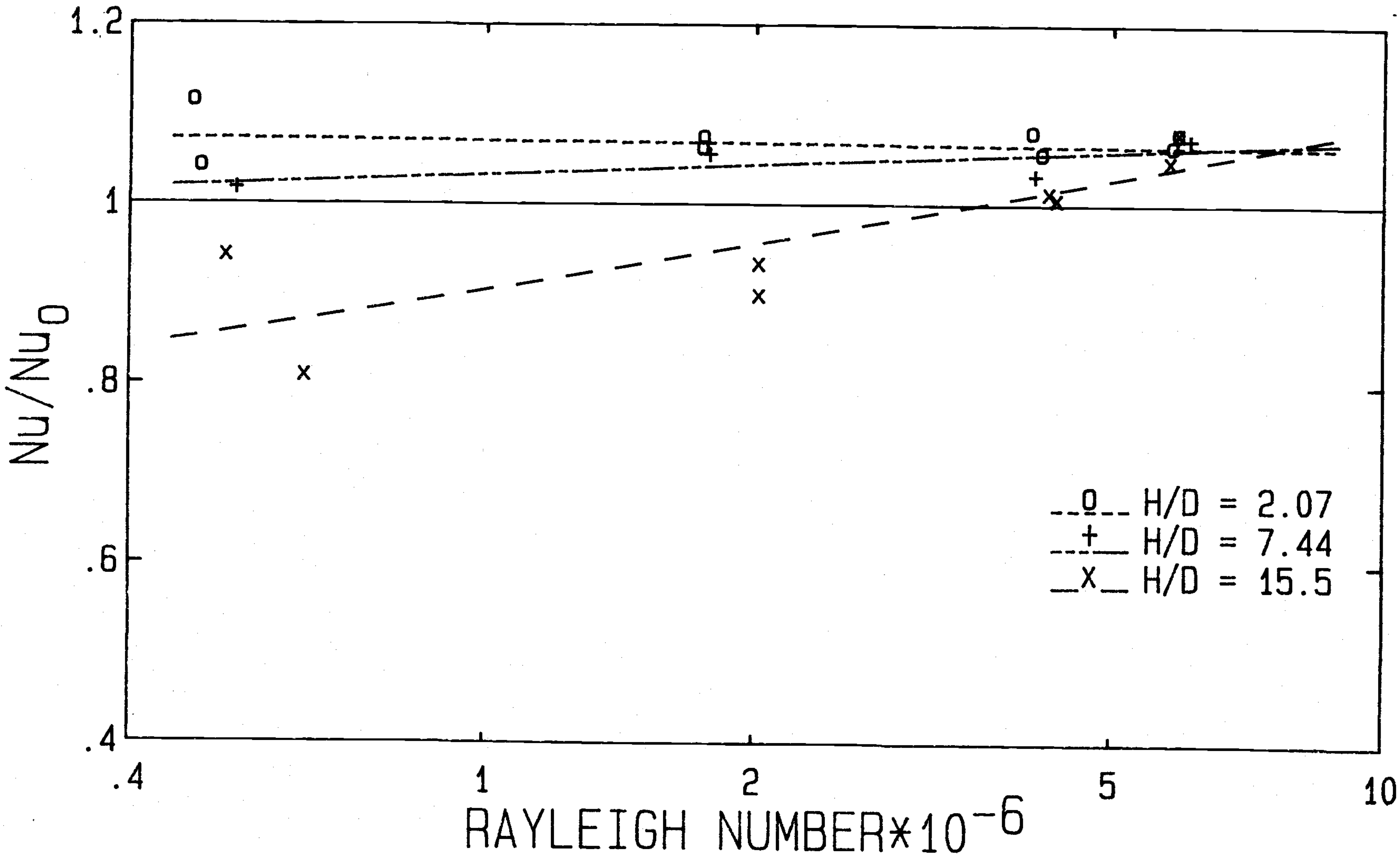

FIGURE 23 Effect of heated wall on cylinder Nusselt number at an S/D of 0.15

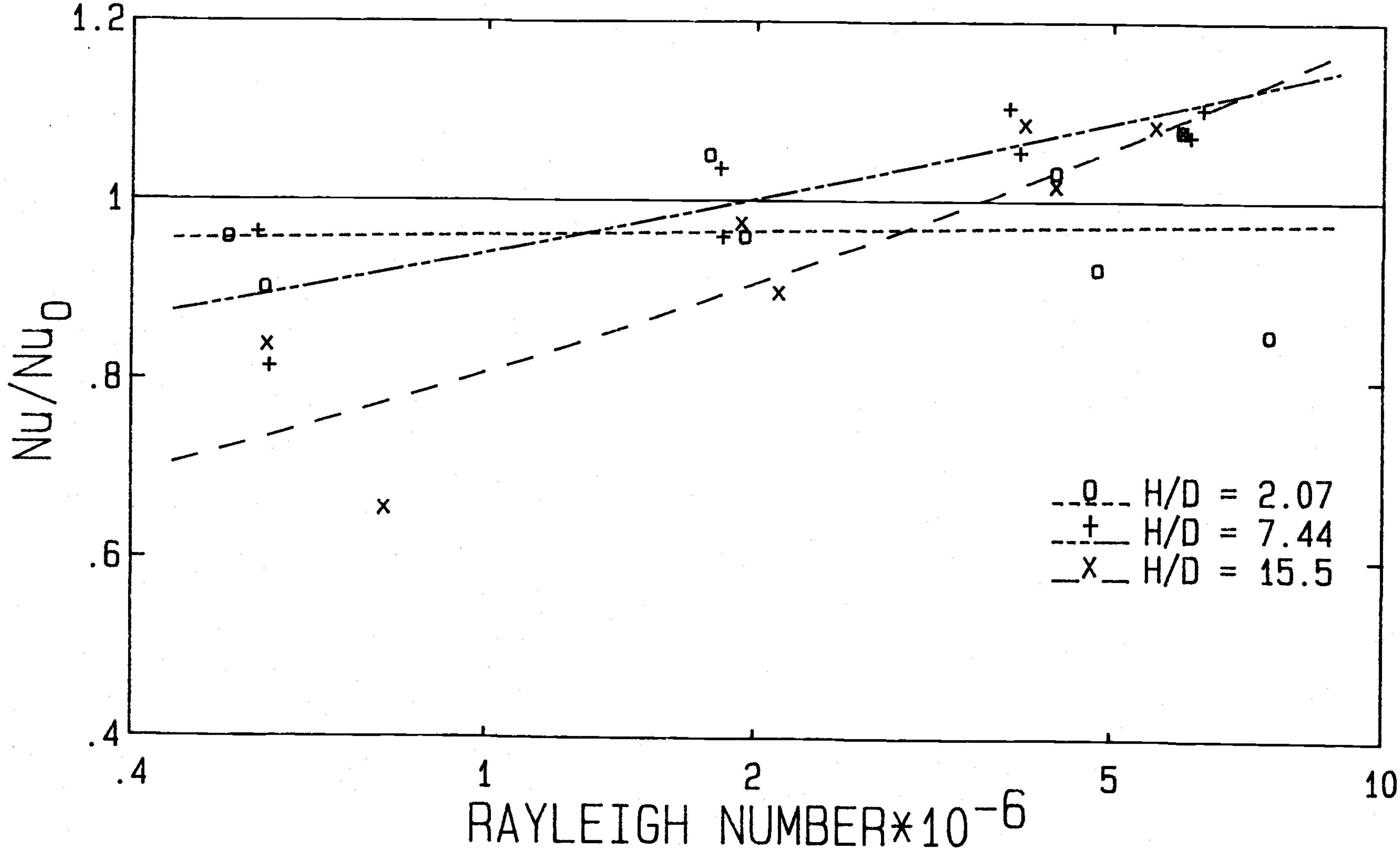

FIGURE 24 Effect of heated wall on cylinder Nusselt number at an S/D of 0.10

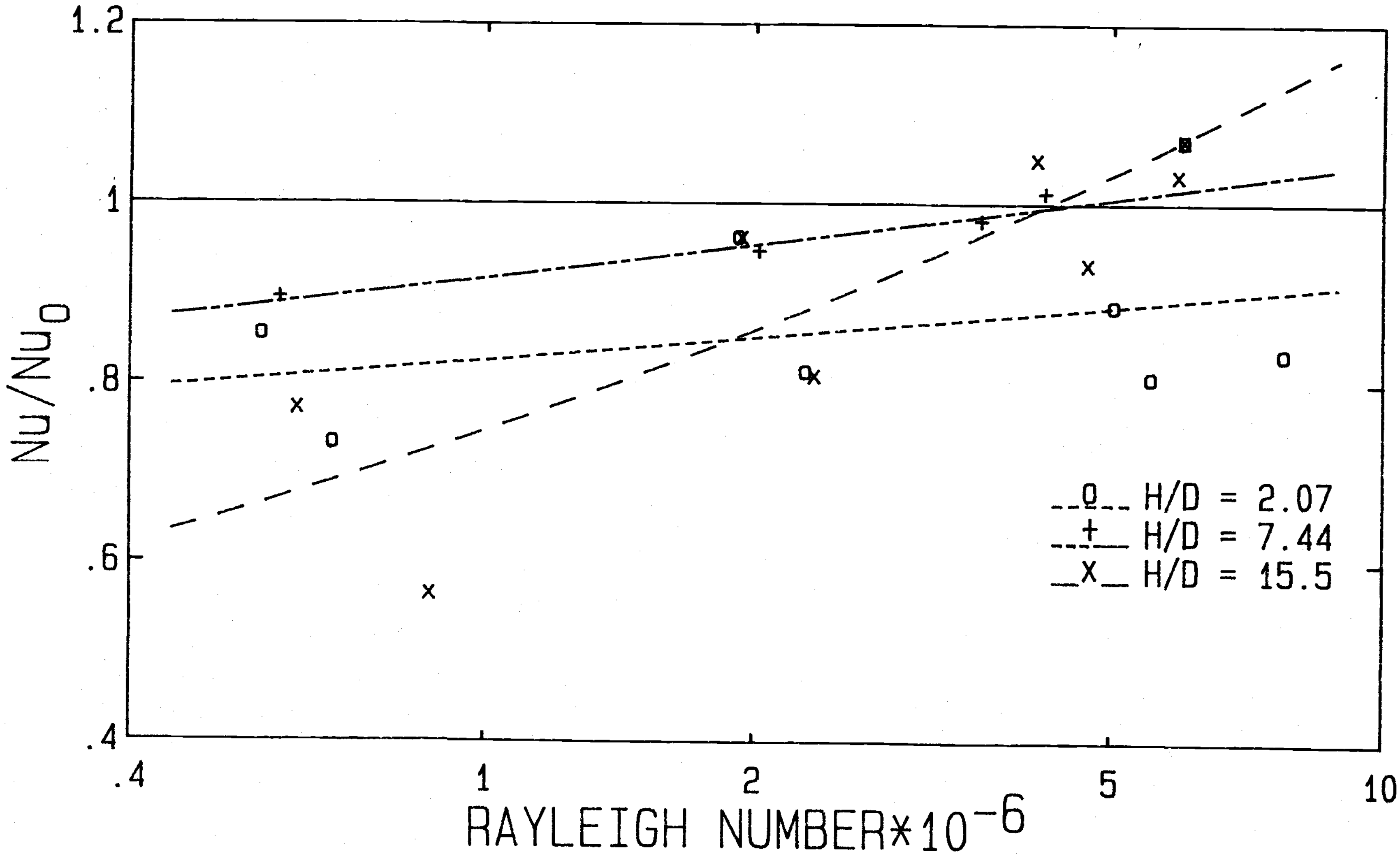

FIGURE 25 Effect of heated wall on cylinder Nusselt number at an S/D of 0.08

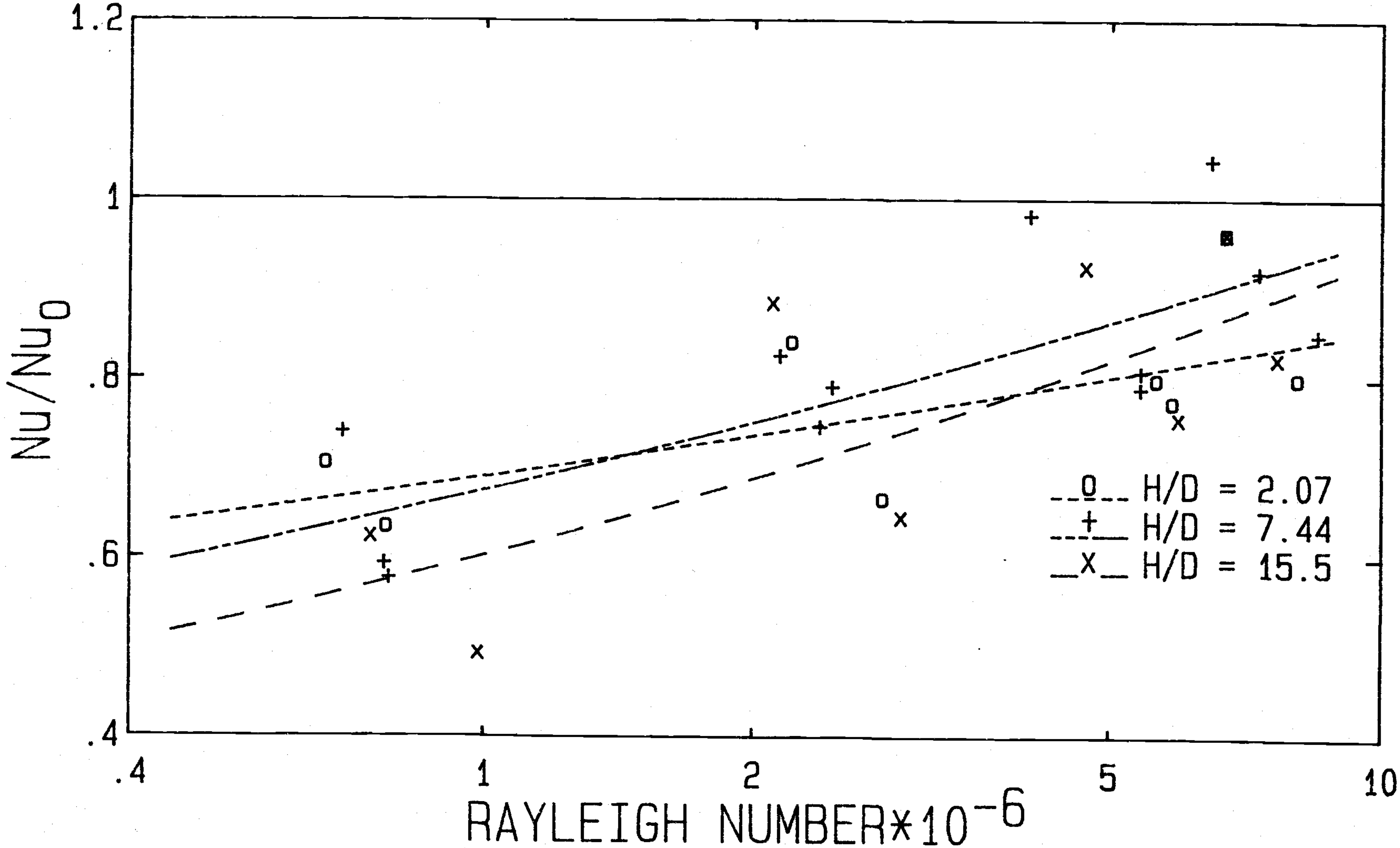

FIGURE 26 Effect of heated wall on cylinder Nusselt number at an S/D of 0.05

APPENDIX C

Procedures for Transferring a Data File from the HP-87 to an IBM-PC.

REQUIRED MATERIALS:

ASCII (AS OPPOSED TO NUMERIC) DATA FILE ON HP DISKETTE.

HP SERIES 80 DATA COMMUNICATIONS PACKAGE WITH CYBER CONFIGURATION.

CONNECT SOFTWARE WITH CONFIGURATION FOR ECYBER.

ACCOUNT ON ECYBER (PROCEDURES FOR CYBER WILL VARY).

HP-87 WITH I/O ROM, 64K MEMORY MODULE, SERIAL INTERFACE AND DISK DRIVE.

IBM-PC WITH DISK DRIVE, DOS 2.0 OR LATER, AND CONNECTION TO THE LAN LINE.

SOME FAMILIARITY WITH THE ABOVE COMPUTERS.

NOTES:

THE HP SERIAL INTERFACE MUST BE IN THE BOTTOM SLOT.

THE LAN LINES TO THE CYBER TERMINALS ARE NOT CONFIGURED FOR USE WITH THE ABOVE SYSTEMS, ANY LINE TO A PC IS OK.

TO WRITE DATA IN ASCII USE THE VAL$ FUNCTION.

UPLOAD FROM HP TO ECYBER:

LOAD "TERMEM87" (CR) - YOUR INPUT IN CAPITALS.

RUN

datacom ready for use - computer response in lowercase.

K7 (3 TIMES - CONT IN FUNCTION KEY BOXES)

K2 (GET CONFIG)

enter configuration file name?

CYBER

displays datacom status and beeps

K8 (RESUME)

datacom ready for use (BE SURE LAN LINE IS CONNECTED)

CARRIAGE RETURN (ENDLINE KEY)

#CALL 810 (LOGIN AS USUAL)

/COPF $INPUT $USER.FILENAME (CR)

? (COPY THE INPUT STREAM TO A PERMANENT FILE FILENAME)

K7 (CONT)

K2 (FNAME)

create a file (y/n)?

N (CR)

filename?

NAME OF ASCII DATA FILE.

file open.

K5 (UPLOAD)

does the host prompt (y/n)?

N (cr)

delay between lines in milliseconds:

100 (CR) (the bigger this number, the less chance of data loss)

upload in progress (wait, disk drive should click occasionally)

file transfer complete - beep

K7 (2 TIMES)

K8 (RESUME)

? CONTROL-T (CR)

command terminated

/

-- NOW CONNECT THE LAN TO THE IBM --

DOWNLOAD TO THE IBM:

BOOT UP THE IBM WITH DOS 2.0 OR HIGHER

RUN CONNECT

menu displayed

4 (RECEIVE FILE)

receive file from NOS/VE

micro file name: FILENAME

host filename: $USER.FILENAME

type fo transfer: 1 (ASCII to ASCII)

transferring file

file transfer complete - beep

(CR)

menu displayed

2 - TERMINAL MODE

(CR)

/LOGOUT

logout messages

ALT-F10

9 - LEAVE CONNECT

IF EVERYTHING WENT SMOOTHLY, THE FILE TRANSFER SHOULD BE COMPLETE.

FOR FURTHER INFORMATION CONTACT Dr. M. B. LARSON; DEPT. MECH. ENGR.

APPENDIX D

Computer Programs

Data Acquisition Program

```
10  DISP "DATA FILE"
20  INPUT FILEOUT$
30  CREATE FILEOUT$,1200,10
40  ASSIGN# 1 TO FILEOUT$
50  DISP "DATE"
60  INPUT DAY
70  SETTIME 0,DAY
80  DISP "POSTION (HEIGHT, DISTANCE) AND CURRENT"
90  INPUT H, S, CURR
110 PRINT#1 ; VAL$(DATE)
120 PRINT#1 ; VAL$(H)
130 PRINT#1 ; VAL$(S)
140 DIM M(15)
150 DIM TC(15)
160 TEMP0 = 0
170 FOR I = 1 TO 15
180     READ M(I)
190 NEXT I
200 DATA .979,.977,.979,.979,.979,.982,.985,.98,.978,.975,1.01,1.01,
210 1.01,1.01,1.01
220 FOR I = 1 TO 15
230     READ TC(I)
240 NEXT I
250 DATA .373,.403,.324,.343,.402,.313,.274,.461,.715,.813,.75,.657,
260 .759,.894,.75
270 ICOUNT = 0
280 WAIT 600000
290 OUTPUT 901 ;"TEM2-9,12,13"
300 ENTER 901 ; TEMP1
310 ENTER 901 ; TEMP2
320 ENTER 901 ; TEMP3
330 ENTER 901 ; TEMP4
340 ENTER 901 ; TEMP5
350 ENTER 901 ; TEMP6
360 ENTER 901 ; TEMP7
370 ENTER 901 ; TEMP8
380 ENTER 901 ; TEMP9
390 ENTER 901 ; TEMP10
400 TEMP1 = M(1)*TEMP1+TC(1)
410 TEMP2 = M(2)*TEMP1+TC(2)
420 TEMP3 = M(3)*TEMP1+TC(3)
```

```
430 TEMP4 = M(4)*TEMP1+TC(4)
440 TEMP5 = M(5)*TEMP1+TC(5)
450 TEMP6 = M(6)*TEMP1+TC(6)
460 TEMP7 = M(7)*TEMP1+TC(7)
470 TEMP8 = M(8)*TEMP1+TC(8)
480 TEMP9 = M(9)*TEMP1+TC(9)
490 TEMP10 = M(10)*TEMP1+TC(10)
500 DISP TEMP1,TEMP2,TEMP3
510 DISP TEMP4,TEMP5,TEMP6,TEMP7
520 DISP TEMP8,TEMP9,TEMP10
530 OUTPUT 902 ;"ACV12"
540 ENTER 902 ; VOLT
550 VOLT = 10*VOLT
560 DISP VOLT
570 OUTPUT 902 ;"TEM7,3-6,16,13-15
580 ENTER 902 ; TEMP12
590 ENTER 902 ; TEMP13
600 ENTER 902 ; TEMP14
610 ENTER 902 ; TEMP15
620 ENTER 902 ; TEMP16
630 ENTER 902 ; TEMP22
640 ENTER 902 ; TEMP23
650 ENTER 902 ; TEMP24
660 ENTER 902 ; TEMP25
670 DISP TEMP16,TEMP25
680 DISP TEMP15,TEMP24
690 DISP TEMP14,TEMP23
700 DISP TEMP13,TEMP22
710 DISP TEMP12
720 PRINT#1 ; VAL$(TIME),VAL$(VOLT),VAL$(CURR)
730 PRINT#1 ; VAL$(TEMP1),VAL$(TEMP2),VAL$(TEMP3)
740 PRINT#1 ; VAL$(TEMP4),VAL$(TEMP5),VAL$(TEMP6),VAL$(TEMP7)
750 PRINT#1 ; VAL$(TEMP8),VAL$(TEMP9),VAL$(TEMP10)
760 PRINT#1 ; VAL$(TEMP16),VAL$(TEMP15),VAL$(TEMP14)
770 PRINT#1 ; VAL$(TEMP13),VAL$(TEMP12)
780 PRINT#1 ; VAL$(TEMP25),VAL$(TEMP24),VAL$(TEMP23),VAL$(TEMP22)
790 ICOUNT = ICOUNT + 1
800 IF ICOUNT = 5 THEN GOTO 830
810 WAIT 60000
820 GOTO 290
830 BEEP 200,200
840 DISP "NEW POSITION H, S AND CURRENT"
850 INPUT H, S, CURR
860 IF H = 0 THEN GOTO 910
870 PRINT#1 ; "-1",VAL$(H),VAL$(S)
880 ICOUNT = 0
890 WAIT 600000
900 GOTO 250
910 PRINT#1 ; "-1000"
920 ASSIGN# 1 TO *
930 END
```

Data Reduction Program

```
C     THIS PROGRAM DOES DATA REDUCTION FOR CYLINDER-WALL
C     HEAT TRANSFER EXPERIMENT.  TIM McCOY JULY, 1987

      REAL*4 TCYL(10),TAMB(5),TWALL(4),TCYLAVG,TAMBAVG,TWALLAVG
      real*8 b(3),mu1,mu2,mu3,R1,R2,B1,B2
      REAL*8 rho,beta,mu,k,pr,ra,cp,g,NU1,NU2,NUEXP

      CHARACTER TITLE*20,COMMENTS*80,FILENAME*10,FILEIN*14,PRFILE*14
      CHARACTER RANUFILE*14

      b(1) = -5.81798D-03
      b(2) = 63.5704D-06
      b(3) = -79.6625D-09

      mu1 = 0.0002414
      mu2 = 247.8
      mu3 = 140.0

      R1 = -4.4D-04
      R2 = 0.9982
      B1 = 0.2136D-04
      B2 = 0.7746

      cp = 4.18
      D = 1.9355
      g = 981.0

      PI = 4.0*ATAN(1.0)
      TESTL = 9.27
      D3 = D**3
      E1 = 0.25
      E2 = 9./16.
      E3 = 4./9.

      sigma = 5.7/1000000000000.
      eps = 0.1

      WRITE(*,*) 'FILENAME: '
      READ(*,1) FILENAME
 1    FORMAT (A)
      FILEIN = FILENAME//'.DAT'
      PRFILE = FILENAME//'.PRT'
      RANUFILE = FILENAME//'.NUS'

      OPEN (UNIT=1,FILE=FILEIN)
      OPEN (UNIT=2,FILE=PRFILE)
      OPEN (UNIT=4,FILE=RANUFILE)
```

```
      WRITE(*,*) 'TITLE: '
      READ(*,1) TITLE

      WRITE(*,*) 'COMMENTS: '
      READ(*,1) COMMENTS

      READ(1,*) DATE,HEIGHT,DIST
      MONTH = INT(DATE/100)
      NDAY = INT(DATE-100*MONTH)
      HEIGHT = HEIGHT/D
      DIST = DIST/D

      WRITE(2,2) TITLE,MONTH,NDAY,COMMENTS
      WRITE(2,3) HEIGHT,DIST
  2   FORMAT (/,T30,A//,T30,'DATE: ',I2,'/',I2//,T12,A)
  3   FORMAT (/,T11,'POSITION:   HEIGHT (H/D) = ',F5.2,
     >  '    DIST FROM WALL (L/D) = ',F5.2)
  4   FORMAT (/)


100   CONTINUE

         READ(1,*) TIME
         IF (TIME .LT. 0.0) GOTO 150

         READ(1,*) VOLTS,CURRENT
         RES = 1000.*VOLTS/CURRENT
         VAVG = VAVG + VOLTS

         DO 10 I = 1,10
 10         READ(1,*) TCYL(I)
         TCYLAVG = TCYLAVG + TCYL(4) + TCYL(5) + TCYL(6) + TCYL(7)

         DO 20 I = 1,5
            READ(1,*) TAMB(I)
 20         TAMBAVG = TAMBAVG + TAMB(I)
         TAMBAVG = TAMBAVG - TAMB(1)

         DO 30 I = 1,4
 30         READ(1,*) TWALL(I)


         WRITE(2,21) TIME,VOLTS,CURRENT
 21      FORMAT (//,T15,'TIME: ',F8.1,'   VOLTS: ',F6.2,
     >            '    CURRENT (mA): ',F6.1//)
         WRITE(2,22) TCYL(2),TCYL(4),TCYL(6),TCYL(8)
         WRITE(2,23) TCYL(1),TCYL(10)
         WRITE(2,22) TCYL(3),TCYL(5),TCYL(7),TCYL(9)
         WRITE(2,24) (TAMB(I),I=1,5)
         WRITE(2,25) (TWALL(I),I=1,4)

 22      FORMAT (T30,4(F7.2))
 23      FORMAT (T11,'CYLINDER: ',T21,F7.2,T60,F7.2)
```

```
 24      FORMAT (/,T12,'AMBIENT: ',5(F7.2))
 25      FORMAT (T12,'   WALL: ',4(F7.2))
      GOTO 100
150   CONTINUE

         VOLTS = VAVG/5.0
         QEXP = VOLTS**2/TESTL/RES
         TCYLAVG = TCYLAVG/20.0
         TCYLAVG = TCYLAVG - 0.0134*QEXP
         TAMBAVG = TAMBAVG/20.0

         TBULK = 0.5*(TCYLAVG+TAMBAVG)
         DELT  = TCYLAVG - TAMBAVG

         TC = TBULK
         TK = TC + 273.15

         BETA = B1*TC**B2
         RHO = R1*(TC-20.0) + R2

         MU = MU1*10.0**(MU2/(TK-MU3))
         K = B(1) + B(2)*TK + B(3)*TK**2

         PR = MU*CP/K
         RA = G*BETA*CP*RHO**2/MU/K
         RACYL = RA*DELT*D**3

         QRAD = EPS*D*PI*SIGMA*((TCYLAVG+273)**4-(TAMBAVG+273)**4)
         QCONV = QEXP - QRAD
         WALLNU = QCONV/(PI*K*DELT)

         NU1 = 0.36+(0.518*RACYL**E1)/((1.0+(0.559/PR)**E2)**E3)
         NU2 = 0.48*RACYL**E1
         NUEXP = 0.895*RACYL**0.20
         DIFFNU = WALLNU/NUEXP

         WRITE(4,*) RACYL,WALLNU,DIFFNU

         TCYLAVG = 0.0
         TAMBAVG = 0.0
         VAVG = 0.0
         IF (TIME .LT.-10.0) GOTO 200
         READ(1,*) HEIGHT,DIST
         HEIGHT = HEIGHT/D
         DIST = DIST/D
         WRITE(2,4)
         WRITE(2,3) HEIGHT,DIST

      GOTO 100
200   CONTINUE

      STOP
      END
```