



Development of a transducer to measure instantaneous local heat flux to a surface immersed in a high temperature fluidized bed
by Jeffory Lawrence Smalley

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in
Mechanical Engineering
Montana State University
© Copyright by Jeffory Lawrence Smalley (1990)

Abstract:

The topic of this research was to develop an abrasion resistant, fast responding (approximately 2 ms) heat flux transducer to measure instantaneous local heat transfer coefficients to a surface immersed in a high temperature fluidized bed at combustion level temperatures. The principle of operation was to use eroding-type surface thermocouples in conjunction with analog circuitry to provide a d.c. output voltage that was related to the instantaneous local heat flux. A limited experimental study was performed in a high temperature fluidized bed to demonstrate that the heat flux transducer could survive the harsh environment imposed upon it and yield useful data. To the author's knowledge, there are no published values for the instantaneous local heat transfer coefficient to a horizontal cylinder immersed in a high temperature fluidized bed, therefore the accuracy of the measurements cannot be determined. However, the results obtained from the experimental study are in agreement with published values of the time average local heat transfer coefficient and the spatial average heat transfer coefficient. The data presented in this thesis are values of the instantaneous local heat transfer coefficient for a 5.08 cm diameter horizontal cylinder at the fluidized bed temperatures of 834.6 K, 821.3 K, and 1016.3 K.

**DEVELOPMENT OF A TRANSDUCER TO MEASURE
INSTANTANEOUS LOCAL HEAT FLUX TO A
SURFACE IMMERSED IN A HIGH
TEMPERATURE FLUIDIZED BED**

by

Jeffory Lawrence Smalley

A thesis submitted in partial fulfillment
of the requirements for the degree

of

Master of Science

in

Mechanical Engineering

MONTANA STATE UNIVERSITY
Bozeman, Montana

April 1990

1378
Sm 185

APPROVAL

of a thesis submitted by

Jeffory Lawrence Smalley

This thesis has been read by each member of the thesis committee and has been found to be satisfactory regarding content, English usage, format, citations, bibliographic style, and consistency, and is ready for submission to the College of Graduate Studies.

10 April 1990
Date

Alan H. George
Chairperson, Graduate Committee

Approved for the Major Department

April 10, 1990
Date

W. E. Larsen
Head, Major Department

Approved for the College of Graduate Studies

April 11, 1990
Date

Henry L. Parsons
Graduate Dean

STATEMENT OF PERMISSION TO USE

In presenting this thesis in partial fulfillment of the requirements for a master's degree at Montana State University, I agree that the Library shall make it available to borrowers under rules of the Library. Brief quotations from this thesis are allowable without special permission, provided that accurate acknowledgment of source is made.

Permission for extensive quotation from or reproduction of this thesis may be granted by my major professor, or in his absence, by the Dean of Libraries when, in the opinion of either, the proposed use of the material is for scholarly purposes. Any copying or use of the material in this thesis for financial gain shall not be allowed without my written permission.

Signature Jeffery L. Smalley
Date 10 April 1990

ACKNOWLEDGMENTS

The author is indebted to the following persons for their contributions to this investigation:

His advisor, Dr. Alan George, for his guidance and assistance during all phases of this investigation.

Pat Vowell, for his assistance in constructing the equipment used in this investigation.

John Rompel, for constructing the special electronic equipment used in this investigation.

The staff of the Mechanical Engineering Department at Montana State University for their assistance in all aspects of this investigation.

The National Science Foundation, for their financial assistance throughout this project (grant no. CBT-8801618).

My wife Sandra Smalley, for her support during all phases of this investigation.

Rene' Tritz, for typing and checking the final version of this thesis.

TABLE OF CONTENTS

	<u>Page</u>
LIST OF TABLES	viii
LIST OF FIGURES	x
NOMENCLATURE	xv
ABSTRACT	xix
1. INTRODUCTION	1
2. DEVELOPMENT OF HEAT FLUX TRANSDUCER	6
Design of Transducer	6
Experimental Transducer	10
Test of Experimental Transducer	11
Construction of the Heat Flux Transducer	12
Materials Required	13
Tools Required	14
Construction of Thermocouples	15
Producing the Mica Sheets	16
Assembly of Transducer	16
Initial Test of Transducer	19
3. INSTRUMENTED CYLINDER	22
Introduction	22
Design of Cylinder	23
4. SIGNAL CONDITIONING CIRCUIT	26
Principle of Operation	26
Analog Signal Conditioning	28
Test of Analog Circuit	30
Contact Resistance Effects	32

TABLE OF CONTENTS—Continued

	<u>Page</u>
Adjusted Analog Circuit	35
Validation of Analog Circuit	35
5. CALIBRATION	38
Introduction	38
Analytical Solution for β	39
Experimental Values of β	41
Investigation of Calibration Results	44
Analytical Solution for k/L	46
Experimental Values of k/L	47
6. DATA ACQUISITION SYSTEM	49
Components	49
Operation	49
Data Reduction	52
7. EXPERIMENTAL RESULTS	54
Introduction	54
Test Conditions	56
Bed Material	56
Test Procedure	56
Test Results	58
Time Average Local Heat Transfer Coefficient	58
Spatial Average Heat Transfer Coefficient	64
Instantaneous Local Heat Transfer Coefficient	65
Typical Instantaneous Local Heat Transfer Coefficients in Graphical Form	66
Description of Possible Modes of Heat Transfer	82
Relative Magnitudes of the Instantaneous Heat Transfer Coefficient	87
Local Bubble Phase Heat Transfer Coefficient	89
Defective Data for Local Heat Transfer Coefficients	92

TABLE OF CONTENTS—Continued

	<u>Page</u>
8. CONCLUSIONS	93
Recommendations	94
APPENDICES	96
Appendix A—Computer Programs	97
Data Acquisition Program	98
Data Transfer Program	103
Program To Determine Superficial Gas Velocity	107
Appendix B—Fluidized Bed Facilities	109
OSU Fluidized Bed Facility	110
Test Section and Distributor Plate	111
Air Flow Measurement	112
Fuel and Temperature Controller	112
MSU Fluidized Bed Facility	113
Test Section	114
Air Supply	114
Heating Supply and Temperature Controller	114
Appendix C—Theory of Analog Circuit	116
Appendix D—RdF Gage Calibration Literature	129
REFERENCES CITED	132

LIST OF TABLES

Table	Page
1. Analytical Values of β at the Indicated Surface Temperature	40
2. Calibrated Values of β Obtained at the Indicated Surface Temperature	44
3. Analytical Value of k/L at the Indicated Bulk Temperature	46
4. Fluidized Bed Conditions for Each of the Test Runs Considered	57
5. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s	59
6. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	59
7. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	60
8. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s	88
9. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	88
10. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	89

LIST OF TABLES—Continued

Table	<u>Page</u>
11. Apparent Local Bubble Phase Heat Transfer Coefficient Values for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s	90
12. Apparent Local Bubble Phase Heat Transfer Coefficient Values for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	91
13. Apparent Local Bubble Phase Heat Transfer Coefficient Values for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s	91
14. Values of the Magnitude and Angle of the Transfer Function as a Function of the Frequency	127

LIST OF FIGURES

Figure	<u>Page</u>
1. Eroding-Type Thermocouple Junction	8
2. Heat Flux Transducer for use in Instrumented Cylinder	9
3. Single Eroding-Type Thermocouple in Mounting Clamp	10
4. Resistance versus Time for First Thermocouple Test	12
5. Temperature versus Time for First Thermocouple Test	13
6. Punch used to Press Fit Thermocouple Clamp	14
7. Eroding-Type Transducer Mounted in Water Cooled Fixture	19
8. Typical Output of Heat Transfer versus Time for the MSU Fluidized Bed Test	21
9. Instrumented Cylinder	24
10. Boundary Conditions for Conduction Problem	27
11. Analog Signal Conditioning Circuit	30
12. Signal Conditioning Circuit Associated with the Heat Flux Transducer	31
13. Response of Original Signal Conditioning Circuit to a Step Change in Surface Heat Flux	32
14. Semi-Infinite Wall With Contact Resistance	33
15. Solution of the Indicated Heat Flux versus Time for Contact Resistance	34
16. Adjusted Analog Circuit	35

LIST OF FIGURES—Continued

Figure	Page
17. Response of Adjusted Analog Circuit to a Step Change in Surface Heat Flux	36
18. Response of Adjusted Analog Circuit to a Long Duration Step Change in Surface Heat Flux	37
19. Micro-Foil Heat Flux Transducer Used For Calibration	41
20. Schematic of Calibration Set Up	42
21. Plot of Analytical and Experimental Values of the Calibration Constant versus Ratio of Surface Temperature to Reference Temperature	45
22. Physical Model of Fluidized Bed in Operation	55
23. Angular Positions at Which Heat Flux Measurements were Made	57
24. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 0.9$ mm, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	61
25. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 2.1$ mm, $U_0 = 2.81$ m/s, $T_{bed} = 821.3$ K, and $T_{bed} = 1016.3$ K	62
26. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 2.14$ mm and $T_{bed} = 810$ K as Given by George (1984)	63
27. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 2.14$ mm and $T_{bed} = 1052$ K as Given by George (1984)	64
28. Comparison of Present Results with Correlations of the Maximum Spatial Average Nusselt Number versus Archimedes Number	66

LIST OF FIGURES—Continued

Figure	Page
29. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 0^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	67
30. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 45^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	68
31. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 90^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	69
32. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 135^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	70
33. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 180^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K	71
34. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 0^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K	72
35. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 45^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K	73
36. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 90^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K	74
37. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 135^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K	75

LIST OF FIGURES—Continued

Figure	Page
38. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 180^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K	76
39. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 0^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K	77
40. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 45^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K	78
41. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 90^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K	79
42. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 135^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K	80
43. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 180^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K	81
44. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 0^\circ$ and $T_{bed} = 834.6$ K	82
45. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 90^\circ$ and $T_{bed} = 834.6$ K	83
46. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 180^\circ$ and $T_{bed} = 1016.3$ K	84

LIST OF FIGURES—Continued

Figure	Page
47. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 135^\circ$ and $T_{bed} = 834.6$ K	85
48. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 45^\circ$ and $T_{bed} = 834.6$ K	86
49. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 180^\circ$ and $T_{bed} = 834.6$ K	87
50. Data Acquisition Program	98
51. Data Transfer Program	103
52. Superficial Gas Velocity Program	107
53. Schematic of OSU High Temperature Fluidized Bed	110
54. Schematic of MSU Low Temperature Fluidized Bed	113
55. Schematic of RC-Low Pass Filter	117
56. Block Diagram of Transfer Function for the RC-Low Pass Filter	118
57. General Heat Flux Transducer	121
58. Block Diagram of Transfer Function for the RC-Low Pass Filter with Voltage Input and Output	122
59. Bode Plot for the Filter Chosen in Analog Circuit	123
60. Block Diagram of Transfer Function with Gain Amplifiers	125
61. General Circuit Diagram for Circuit that Approximates the Transfer Function	126
62. Temperature Correction Curve from the RdF Calibration Literature	131

NOMENCLATURE

ANSI	American National Standards Institute
Ar	Archimedes Number
a	thermocouple sensitivity
C	Capacitance
c	specific heat of transducer material
cm	centimeter
d.c.	direct current
db	decibels
dp	mean particle size
e	input voltage to signal conditioning circuit
f	farad
$\bar{f}(s)$	the Laplace Transform of a function, in this case $f(t)$
$G(s)$	Laplace Transform Function
g	acceleration due to gravity
hp	Hewlett Packard
Hz	Hertz
$\langle h \rangle$	time average local heat transfer coefficient
h_b	bubble phase local heat transfer coefficient
$\bar{h}$	spatial average local heat transfer coefficient
$\bar{h}_{max}$	maximum spatial average heat transfer coefficient
$h(t)$	instantaneous local heat transfer coefficient
i	current

NOMENCLATURE—Continued

j	$\sqrt{-1}$
K	degrees Kelvin
k	thermal conductivity of transducer material
k_f	thermal conductivity of fluidizing gas at bed temperature
kg	kilogram
L	effective length of transducer
m	meter
mm	millimeter
MSU	Montana State University
$\overline{Nu}_{\max}$	maximum spatial average Nusselt Number
OSU	Oregon State University
q_{IND}	indicated heat flux with contact resistance
$q_w(t)$	instantaneous local heat flux
$\langle q_w \rangle$	time average local heat flux
$\delta q_w(t)$	instantaneous deviation of surface heat flux from time average value
$\langle q_{mf} \rangle$	average heat flux as indicated by the Micro-Foil heat flux transducer
R	resistance
R_C	contact resistance
SAE	Society of Automotive Engineers
s	Laplace transform parameter
T	temperature
T_b	bulk temperature
T_{bed}	bed temperature

NOMENCLATURE—Continued

T_L	constant temperature at $x = L$
T_w	surface temperature
$\langle T_w \rangle$	Time average surface temperature
$\delta T_w(t)$	instantaneous deviation of surface temperature from the time average value
T_1	Time average surface temperature as indicated by the first surface thermocouple
T_2	Time average surface temperature as indicated by the second surface thermocouple
T_3	Time average in-wall temperature as indicated by the in-wall thermocouple
t	time
U_0	Superficial gas velocity
V	volts
v_i	input voltage to signal conditioning circuit
v_o	output voltage from signal conditioning circuit
v_4	output voltage of signal conditioning circuit related to heat flux
W	Watts
x	position coordinate
x_1	proportionality constant between the input voltage of the signal conditioning circuit and indicated thermocouple temperature
y_1	proportionality constant between the output voltage of the signal conditioning circuit and heat flux
α	thermal diffusivity
ρ	density of transducer material
ρ_f	density of fluidizing gas at the bed temperature

NOMENCLATURE—Continued

ρ_s	density of particle
ν_f	kinematic viscosity of fluidizing gas at the bed temperature
θ	angular position on cylinder surface
μ	micro
π	constant 3.14159
ϕ	phase angle
η	dummy variable for time
ω	frequency
β	calibration constant

ABSTRACT

The topic of this research was to develop an abrasion resistant, fast responding (approximately 2 ms) heat flux transducer to measure instantaneous local heat transfer coefficients to a surface immersed in a high temperature fluidized bed at combustion level temperatures. The principle of operation was to use eroding-type surface thermocouples in conjunction with analog circuitry to provide a d.c. output voltage that was related to the instantaneous local heat flux. A limited experimental study was performed in a high temperature fluidized bed to demonstrate that the heat flux transducer could survive the harsh environment imposed upon it and yield useful data. To the author's knowledge, there are no published values for the instantaneous local heat transfer coefficient to a horizontal cylinder immersed in a high temperature fluidized bed, therefore the accuracy of the measurements cannot be determined. However, the results obtained from the experimental study are in agreement with published values of the time average local heat transfer coefficient and the spatial average heat transfer coefficient. The data presented in this thesis are values of the instantaneous local heat transfer coefficient for a 5.08 cm diameter horizontal cylinder at the fluidized bed temperatures of 834.6 K, 821.3 K, and 1016.3 K.

CHAPTER 1

INTRODUCTION

Recently a great deal of attention has been devoted to the research and development of fluidized bed combustors in the United States. There are several reasons that make fluidized beds an attractive option as opposed to using the more conventional stoker-fired and pulverized-coal-fired combustors. One of the most promising aspects of using fluidized bed combustors is improved air quality. If the bed material is composed of solid limestone or dolomite, a large portion of the sulfur dioxide released during the combustion process will be absorbed by the bed material. Furthermore, the lower temperatures characteristic of fluidized bed combustion (approximately 1123 K) tend to reduce the amount of nitrogen oxides formed during the combustion process thereby increasing air quality. Another attractive aspect of using fluidized bed combustors, according to Makansi and Schweiger (1987), is that they can be designed to burn several different types of fuels such as biomass and industrial wastes as well as coal. This means that a particular combustor may not be limited to a single type of fuel but can use whatever fuel is readily available at any particular time.

Based on information given by Makansi and Schwieger (1987) there are some drawbacks to using fluidized beds. The major drawback is that the combustion and boiler efficiency of fluidized beds are currently lower than comparably sized

pulverized-coal-fired boilers and in some cases lower than stoker-fired units. Therefore, in order to make the best use of fluidized bed combustors, it is important to understand their basic properties through research.

Many combustor designs, according to Schwieger (1985), employ arrays of horizontal tubes as heat transfer surfaces which are immersed in the fluidized bed itself. Heat transferred to the tubes is used either to generate steam or for the heating of process fluids. It is critical to have a fundamental understanding of the bed-to-surface heat transfer phenomena in high temperature fluidized beds in order to provide for accurate design and efficient operation of such systems.

Knowledge of instantaneous local heat transfer rates to surfaces immersed in low temperature fluidized beds is relatively complete. Studies involving vertical or inclined surfaces include those of Mickley et al. (1961), Tout and Clift (1973), and Gloski et al. (1984). Similar measurements for horizontal tubes have been reported by Catipovic et al. (1978, 1979, 1982). All of the above studies utilized electrically heated foils or thin metal films of low heat capacity as heat flux transducers. The operating principle of these devices, and the signal conditioning methods used, require that the transducer surface be at a higher temperature than the fluidized bed. For this reason, none of the heat flux transducers employed in the above studies are useful in high temperature fluidized beds. Despite their limitations, the above studies conducted in low temperature fluidized beds have contributed data which served to partially validate certain detailed models of the heat transfer process such as the models proposed by Adams and Welty (1979), and Decker and Glicksman (1983). However, the results obtained from these experimental studies cannot be reliably extended to the prediction of heat transfer rates to immersed surfaces in fluidized beds at combustion-level temperatures.

Time average local heat transfer coefficients for horizontal tubes immersed in high temperature fluidized beds have been reported [George and Welty (1984), Goshayeshi et al. (1985)]. While these data provide useful information concerning the local heat transfer rate around the periphery of an immersed tube, the more detailed models of the heat transfer process involve the calculation of instantaneous bed-to-surface heat transfer rates from the instantaneous flow field. These advanced analytical models cannot be validated or improved using only time average local heat transfer data.

The heat flux transducers chosen for the experiments by George and Welty (1984) and Goshayeshi et al. (1985) were thermopile type gages which use arrays of thermocouple junctions to obtain the temperature difference across thin wafers of insulating material. The problem with using this type of gage to measure the instantaneous heat transfer rate is that the gage is not abrasion resistant and must be covered with a protective film for use in fluidized beds. For use in high temperature fluidized beds, a metallic film or cover must be used. When this is done it increases the settling time of the gage. For example, Goshayeshi et al. (1985) covered a Micro-Foil heat flux gage purchased from RdF Corporation, Hudson, New Hampshire, with 0.127 mm thick stainless steel shim stock and increased the settling time from 80 ms for the gage alone to approximately 960 ms for the covered gage. The settling time referred to above is defined by Doebelin (1990), as the time, after application of a step input, for the instrument output to reach and stay within ± 5 percent of the steady state value. These values for the settling time (80 ms and 960 ms) are too long to be useful for instantaneous local heat flux measurements in fluidized beds. Therefore, in order to study instantaneous local heat transfer rates to surfaces immersed in high temperature fluidized beds, a new

heat flux transducer must be designed that can withstand the harsh environment imposed upon it.

The need for a study to measure instantaneous local heat transfer rates to surfaces immersed in high temperature fluidized beds is clear. The data are needed to validate and improve existing analytical models of the heat transfer process and for direct use in the design of immersed heat transfer surfaces. Previous attempts to accurately measure instantaneous local heat transfer rates to surfaces immersed in high temperature fluidized beds were unsuccessful due to the lack of an adequate heat flux transducer. The abrasive nature of the fluidized bed and the high temperatures involved pose significant design problems as concerns the heat flux transducer. In order to measure the instantaneous local heat flux it is necessary to develop a heat flux transducer which is abrasion resistant, capable of operation at high surface temperatures (550 K), and with sufficiently rapid response that effectively instantaneous local values of heat flux can be measured.

It was the intent of this study to develop, calibrate, and test such a heat flux transducer. There were five specific objectives in the development and testing of such a transducer:

1. To design and fabricate a heat flux transducer which was abrasion resistant, capable of operation at high temperatures, and with sufficiently rapid response that effectively instantaneous local values of the heat flux could be measured between the high temperature fluidized bed and immersed surface.
2. To design and fabricate an instrumented cylinder that contained the heat flux transducer and that could be easily mounted in the high temperature fluidized bed at Oregon State University. The work reported here was supported, in part, by the National Science Foundation under grant CBT-8801618. As part of the proposed work, it was agreed that the OSU high temperature fluidized

bed facility would be used to test the heat flux transducer and associated equipment that is the subject of this thesis. The cylinder was designed in such a fashion that the heat flux transducer was easily installed and removed from the cylinder without causing damage to either the cylinder or the transducer.

3. To design and implement a calibration scheme that determined the operational characteristics of the heat flux transducer for both steady state and transient operating conditions.
4. To program and implement a data acquisition system that interfaced with the analog signal conditioning unit which is associated with the heat flux transducer. The data acquisition system was responsible for taking time average and instantaneous temperature and heat flux readings from the analog signal conditioning unit for the heat flux transducer.
5. To prove the operation of the heat flux transducer and associated signal conditioning and data acquisition equipment by conducting a limited experimental study of instantaneous local heat transfer rates to horizontal cylinders immersed in a fluidized bed at combustion level temperatures.

CHAPTER 2

DEVELOPMENT OF THE HEAT FLUX TRANSDUCER

Design of Transducer

In the design of the heat flux transducer, several conditions had to be met. The first condition was that the transducer had to be operable at combustion level fluidized bed temperatures (approximately 1123 K). Under normal conditions the transducer will not reach a temperature this high because the instrumented cylinder, which contains the transducer, will be water cooled. However, there is a possibility that the pump controlling the water supply that cools the instrumented cylinder could fail, causing the cylinder to reach combustion level temperatures. Therefore, all materials used in the construction of the transducer can withstand temperatures up to 1123 K. Typically, however, the transducer temperature would not, in operation, exceed 550 K. Type K thermocouples, which were chosen for this application, are composed of Chromel-Alumel and have a suggested maximum operating temperature of 1533 K. This value is far greater than any expected at the OSU fluidized bed facility. Also the EMF versus temperature relationship for type K thermocouples is compatible with the circuit used in association with the heat flux transducer.

The second constraint was that the surface thermocouples had to survive in the abrasive environment caused by the solid particles in the fluidized bed. This condition was met by using thin ribbon eroding-type thermocouple junctions. Thin ribbon eroding-type thermocouple junctions of greatly different dimensions than reported here are manufactured by Nanmac Corporation, Framingham, Massachusetts. The eroding-type thermocouple junction consists of two wire ribbons pressed between three mica sheets as shown in Figure 1. The mica sheets are used as insulating material so the wire ribbons are only able to be in contact at the surface of the transducer. The actual thermocouple junction is formed by small burrs at the surface which bridge over the thin mica sheets. These burrs also contact the surface of the transducer body, creating a grounded thermocouple junction of extremely low thermal mass at the surface of the cylinder.

The next constraint that had to be met was the overall size of the heat flux transducer. The transducer was designed to fit in a 5.08 cm diameter stainless steel cylinder with a wall thickness of 0.95 cm.

The last constraint was that the heat flux transducer be easily installed and removed from the instrumented cylinder without causing damage to the cylinder or the transducer.

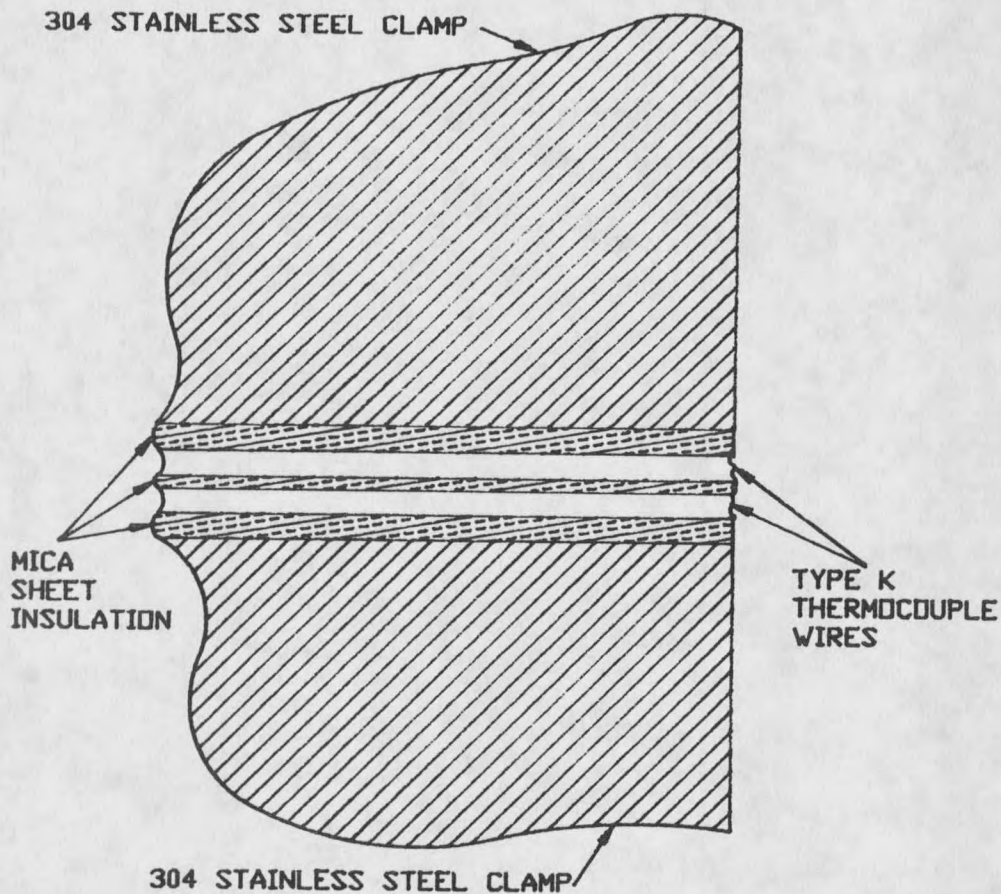


Figure 1. Eroding-Type Thermocouple Junction.

The completed design of the heat flux transducer is shown in Figure 2. The heat flux transducer, which is constructed of 304 stainless steel, contains two eroding-type thermocouples which measure the temperature at the surface of the cylinder. The transducer also contains an in-wall welded junction type K thermocouple. The welded junction thermocouple is at a depth of approximately 6 mm from the surface where the other thermocouples are positioned. The wire sleeve, which contains the thermocouple wires, has been reduced to the smallest allowable diameter so that the cooling water is not restricted at this point in the cylinder.

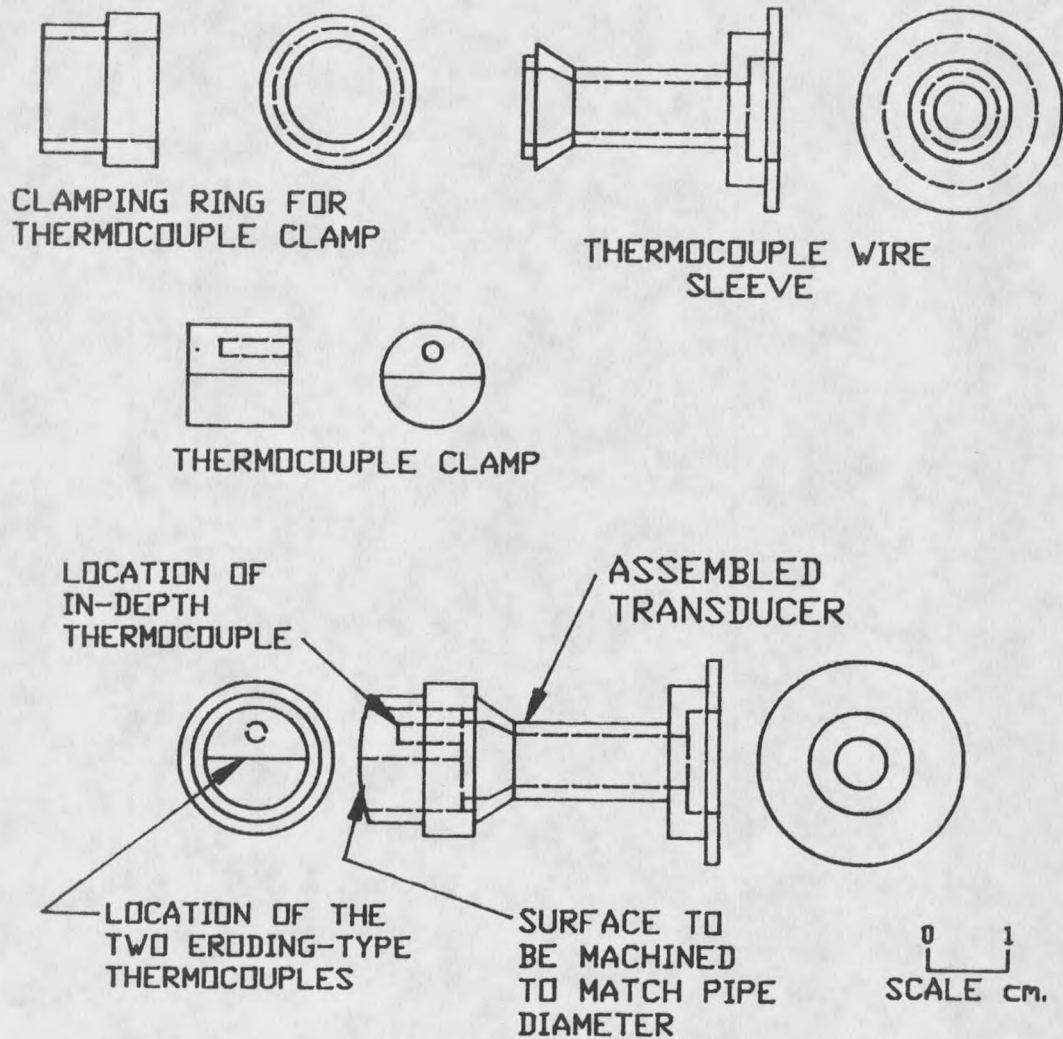


Figure 2. Heat Flux Transducer for use in Instrumented Cylinder.

It was proposed that this transducer be designed at MSU and subsequently fabricated by a vendor of specialized thermocouples. However, this was not done due to an unreasonable price relative to the project budget and long delivery time. Therefore, it was decided to build the entire heat flux transducer at MSU.

Experimental Transducer

In order to assure that an eroding-type thermocouple junction could be successfully assembled at MSU, the simple transducer with a single eroding-type thermocouple junction shown in Figure 3 was constructed and tested in the small electrically heated fluidized bed at MSU.

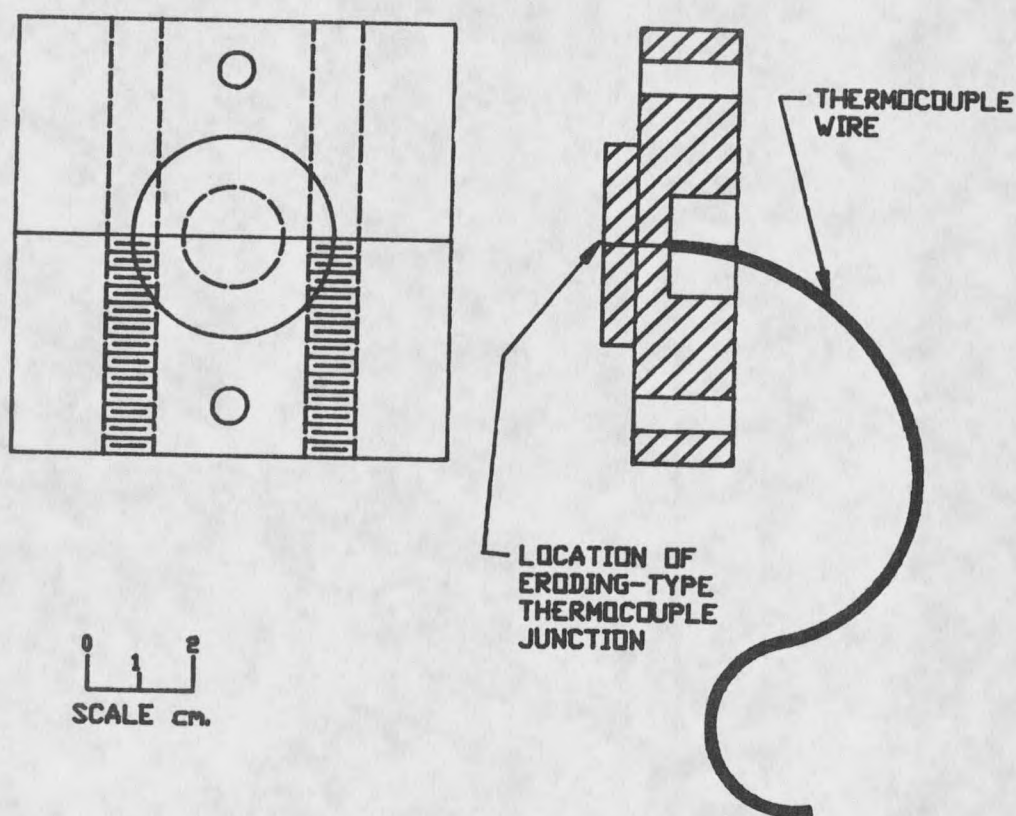


Figure 3. Single Eroding-Type Thermocouple in Mounting Clamp.

Producing the eroding-type thermocouple was more difficult than originally anticipated. Commercially available ribbon thermocouples were either too thick or too narrow to be of any use as eroding-type thermocouples for the applications considered here. Therefore, thermocouple wire was rolled at one end to a thickness

of 0.025 mm and a width of approximately 3.1 mm. This process was made more difficult because the thermocouple wires required repeated annealing.

The mica sheets used for insulating material were extracted from old capacitors because commercially available mica was difficult to locate in sheets that did not contain prepunched holes for component assembly. The two mica sheets that were used to insulate the wires from the transducer body were split to a thickness of 0.025 mm. The mica sheet used to insulate the two ribbon wires from each other was split to an approximate thickness of 0.013 mm. The eroding-type thermocouple arrangement was clamped between two pieces of 304 stainless steel and the wire tips were lightly sanded using 240 grit paper until a junction was established. This clamping arrangement was designed in such a way that it fit in the wall of the electrically heated fluidized bed at MSU so that the junction could be tested. The MSU fluidized bed is described in Appendix B.

Test of Experimental Transducer

The first test run lasted a total of two hours. During that time, the eroding-type thermocouple was exposed to a bubbling fluidized bed of approximately 500 μm diameter sand. The bed temperature was varied from 303 K to 563 K. Measurements of the combined thermocouple junction and lead wire resistance, and output voltage of the thermocouple, were taken every two seconds for the entire two hour test using an hp 85B computer and an hp 3054A automatic data acquisition/control system. The electrical resistance versus time for this test is shown in Figure 4. The electrical resistance of the thermocouple and lead wires remained within the range of 20 ± 1 ohms throughout the test. Therefore, the small burrs which form the junction do appear to renew themselves as the surface is slowly eroded by the solid particles in the fluidized bed. The output voltage

indicated by the thermocouple was converted to corresponding temperature values using an hp applications program to convert type K thermocouple voltages to corresponding temperatures. The temperature versus time for this test is shown in Figure 5. The indicated surface temperatures are reasonable for this test.

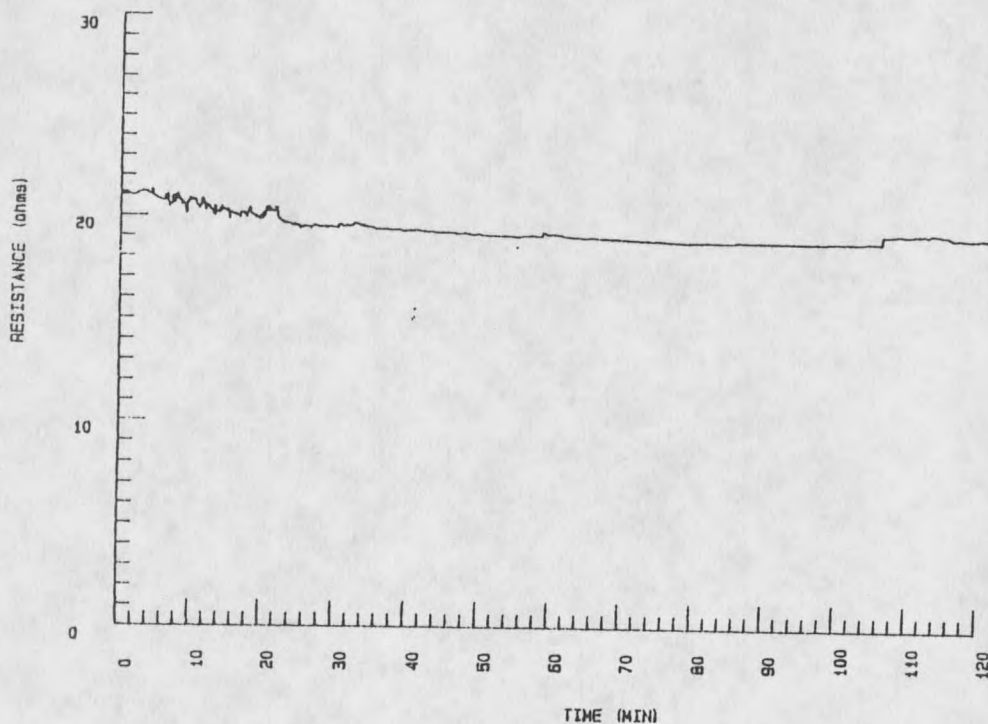


Figure 4. Resistance versus Time for First Thermocouple Test.

Construction of the Heat Flux Transducer

After initial testing of the eroding-type thermocouple was complete, the heat flux transducer shown in Figure 2 was constructed. The transducer body was machined from ANSI 304 stainless steel.

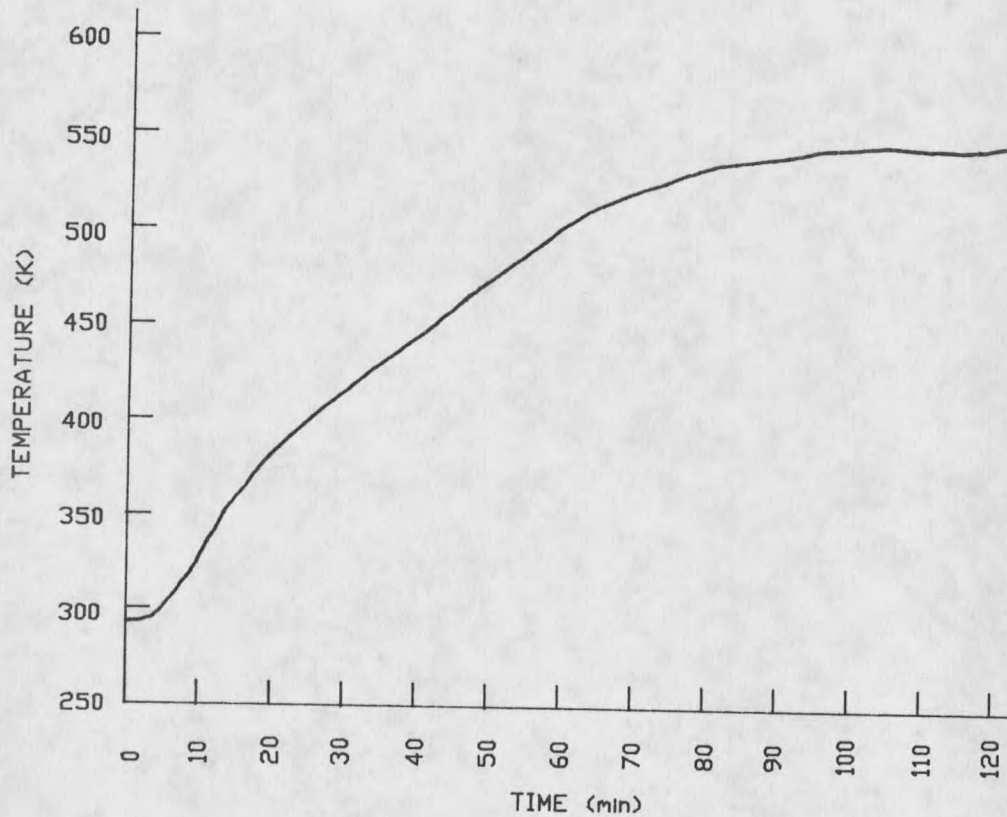


Figure 5. Temperature versus Time for First Thermocouple Test.

Materials Required

The following materials were required to construct the transducer shown in Figure 2:

1. The individual pieces of the transducer body shown in Figure 2 machined from ANSI 304 stainless steel.
2. Three mica sheets retrieved from old mica capacitors.
3. Approximately 8.25 meters of Type K thermocouple wire, insulated in asbestos and wrapped in a glass braid, with a wire diameter of 0.81 mm. Other

high temperature resistant insulations could also be used, e.g., ceramic, high temperature glass, etc.

4. Two 1.2 cm inside diameter snap rings.
5. A bottle of liquid super glue.
6. A tube of high temperature silicone gasket seal.

Tools Required

The following tools were required to construct the transducer shown in Figure 2:

1. A digital multimeter to measure the resistance of the thermocouple wires.
2. A propane torch.
3. A smooth faced hammer and anvil or other smooth surface.
4. A pair of snap ring pliers.
5. A pair of needle nose pliers.
6. A thermocouple welder.
7. A sheet of 240 and 320 grit sand paper.
8. A specially designed aluminium punch shown in Figure 6.
9. A pair of scissors.

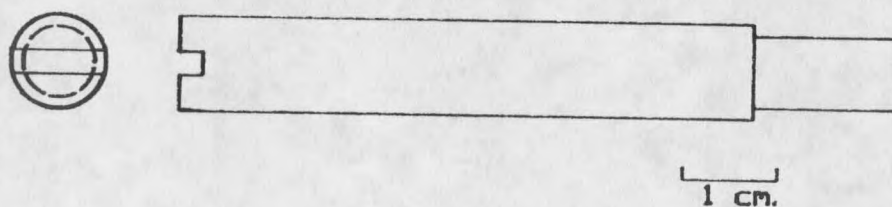


Figure 6. Punch used to Press Fit Thermocouple Clamp.

Construction of Thermocouples

There were three thermocouples that needed to be produced before the heat flux transducer could be assembled. Two of the thermocouples were eroding-type and the other was a welded junction thermocouple. To produce these thermocouples the thermocouple wire was cut into three equal pieces. With one section of wire a welded junction thermocouple was constructed. The insulation was stripped from the wire, leaving approximately 0.5 cm of exposed wire. The exposed ends were twisted together with a pair of needle nose pliers, then welded using a thermocouple welder. The other two sections of wire were used to create the two eroding-type thermocouples.

The eroding-type thermocouples were made by forging the ends of the remaining two wires into thin ribbons. The forging process was done in the following series of steps:

1. About 0.8 cm of insulation was stripped from the ends of the thermocouple wire.
2. A hammer and anvil were used to tap on the thermocouple wires lightly until they reached a thickness of approximately 0.2 mm.
3. The wires were then annealed by heating them with a propane torch and then quenching them in cool water.
4. The annealed wire ends were lightly sanded with 320 grit paper to remove any oxidation caused during the annealing process.
5. The wires were lightly tapped again with the hammer until the thickness had been reduced by approximately half. Care had to be taken to hit the wires with the flat portion of the hammer so the wires would not get creased. If this happened the process was started over again because the wire became too weak to handle during later installation.

6. Steps three through five were repeated until the wire thickness was reduced to 0.025 mm for at least 6.5 mm along the length of the wire. When this process was completed, the 6.5 mm length of the wires had a ribbon width of 6 mm.

Producing the Mica Sheets

The next step in the construction of the heat flux transducer was to make the mica sheets to be used as insulating material for the eroding-type thermocouple junctions. To do this, the mica was removed from a mica capacitor that was 7.5 cm by 5.2 cm in cross section. The mica sheets were sliced with a razor blade until two sheets were produced that were 1.3 cm by 1.3 cm and 0.025 mm thick. Also, a sheet was produced that was 1.6 cm by 1.3 cm and approximately 0.01 mm thick. The mica sheets obtained from the capacitor were tested in a heat treating oven and found to be stable at temperatures as high as 1100 K. Once the mica sheets had been produced, the transducer was ready for assembly.

Assembly of Transducer

Assembly of the transducer was done in the following steps:

1. The welded junction thermocouple was silver soldered into the hole that was drilled on one half of the thermocouple clamp shown in Figure 2. The silver soldering was done using an acetylene torch and Eutector type "T" flux made by Eutectic Castolin, Flushing, New York.
2. The two mica sheets, that were split to a thickness of 0.025 mm, were mounted on the inside face of the thermocouple clamp. The mica sheets were mounted by placing one drop of super glue on the face of each clamp and then placing one mica sheet on each face. It was important at this point to make sure that

each mica sheet covered the entire face of the thermocouple clamp. Before continuing, the super glue was allowed to dry for five minutes.

3. The ribbon thermocouples were positioned for clamping. To do this, both wires were placed in one hand and aligned so that the ribbon ends were parallel to each other.
4. The one half of the thermocouple clamp that does not contain the welded junction thermocouple was placed over the wire ribbons to check to see that the wires would not protrude from the side of the clamp. If the wires were protruding from the side of the clamp, step 3 was repeated or the width of the wire ribbons was reduced using a pair of scissors.
5. The half of the thermocouple clamp used in step 4 was removed and the remaining mica sheet was placed between the thermocouple wires.
6. One half of the thermocouple clamp was positioned on each side of the wires and a snap ring was placed over the thermocouple clamp. It was important to place the thermocouple clamp in such a fashion that the ribbon thermocouples and mica sheet extended past the end of the clamp by approximately 0.2 mm.
7. Another snap ring was placed over the thermocouple clamp to keep the thermocouple clamp and wires from slipping.
8. The thermocouple wires were inspected to ensure that they were not protruding from the sides of the thermocouple clamp. If the wires were exposed steps 3 through 8 were repeated.
9. The following resistances were checked with the multimeter:
 - a. The resistance between each pair of ribbon thermocouple wires.
 - b. The resistance between each ribbon thermocouple wire and the other three ribbon thermocouple wires.

c. The resistance between each ribbon thermocouple wire and the thermocouple clamp.

If the multimeter indicated a near infinite resistance (greater than 100 Kohms) the process continued. If not, steps 3 through 9 were repeated.

10. The thermocouple clamp was press fit into the thermocouple clamping ring. This was done using the punch shown in Figure 12 and a drill press. The punch was aligned so that the protruding ribbon thermocouple wires fit into the slit on the punch. This was done so that the wires would not get pinched during the press fit operation.
11. Step 9 was repeated.
12. The lead wires protruding from the rear of the thermocouple clamp were insulated. This was done by coating any bare spots on the lead wires with high temperature silicone gasket seal. The gasket seal was allowed to dry for two hours.
13. The lead wires were threaded through the wire sleeve and the wire sleeve was positioned in place.
14. The resistances indicated in step 9 were rechecked. If the multimeter indicated a resistance less than 100 Kohms, the wire sleeve was removed and the lead wires checked for bare spots. Any exposed wire that was found was coated with high temperature silicone gasket seal and the gasket seal was then allowed to dry.
15. Step 14 was repeated until the multimeter did not indicate a resistance less than 100 Kohms.
16. The face of the transducer where the ribbon thermocouples were exposed was sanded until a resistance was measured between each pair of wires. The

overall resistance of the junctions that were created from sanding and the lead wires was between 3 and 8 ohms.

17. The joint caused by the wire sleeve and the clamping ring was sealed with high temperature silicone gasket seal. The heat flux transducer was now ready for calibration and use.

Initial Test of Transducer

Since initial testing was desired before going to OSU, a special water cooled fixture was built as shown in Figure 7. This fixture was designed in such a fashion as to fit in the small electrically heated fluidized bed at MSU.

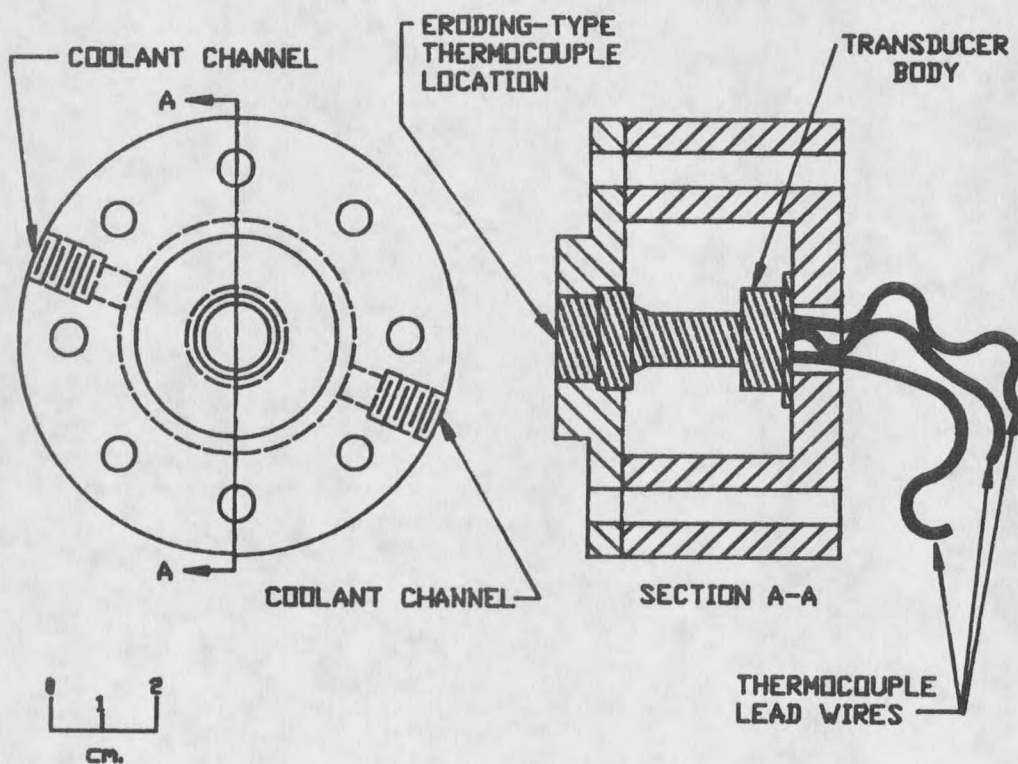


Figure 7. Erodng-Type Transducer Mounted in Water Cooled Fixture.

The preliminary testing that occurred at MSU was done in the following manner. The transducer was exposed to a low temperature bubbling fluidized bed of approximately 500 μm diameter sand. The transducer was tested at various bed temperatures ranging from 303 K to 563 K, and various cooling water flow rates. The actual test procedure went as follows. The bed was turned on and allowed to reach equilibrium before data was taken. Once the bed reached steady state, the indicated heat flux from the associated signal conditioning circuit (described in Chapter 4) was recorded. The indicated temperature from the three thermocouples was also recorded for each test run. Values for the heat flux were recorded every 5 ms during actual test runs. This data was taken using the data acquisition system and associated computer codes discussed in Chapter 6.

Testing was done for approximately 120 hours with about one hour of actual heat flux data being taken. A typical output of surface heat flux versus time is shown in Figure 8. The minimum measured heat flux values shown in the figure are negative and, therefore, not physically realistic. These inaccurate results are due in part to the use of calculated input-output relationships for the transducer and associated signal conditioning circuit rather than relationships based on accurate calibration. A malfunctioning reference junction compensator connected to one of the surface temperature thermocouples was also a significant error source during preliminary testing.

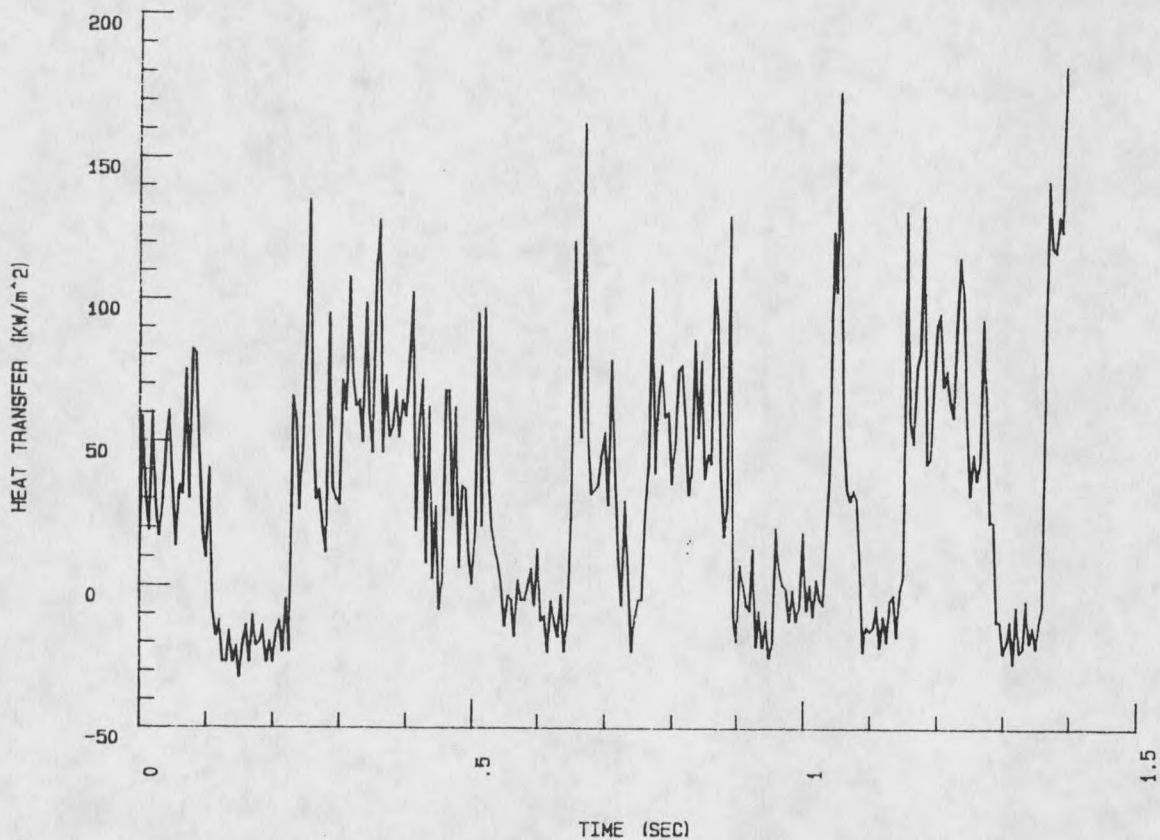


Figure 8. Typical Output of Heat Transfer versus Time for the MSU Fluidized Bed Test.

In order to correct this problem, the reference junction compensators were disconnected from the circuit and a thermocouple was installed on the signal conditioning circuit input terminals in order to establish the reference temperature. Also, for testing in the OSU high temperature fluidized bed, calibrated input-output relationships were used as described in Chapter 5.

CHAPTER 3

INSTRUMENTED CYLINDER

Introduction

There were several important criteria in the design of the instrumented cylinder. The first was that the cylinder was to be designed in such a fashion that the heat flux transducer could be installed and removed easily without causing any damage to the transducer or lead wires. The next criteria was that the transducer have a minimum number of points at which the cooling water could escape. This criteria was of utmost importance since the temperature of the environment in which the cylinder was to be placed is in excess of 1123 K. If the cooling water happened to escape the confines of the cylinder, the chance for catastrophic failure of the fluidized bed and injury of the operators was of concern. Therefore, to reduce the chance of such a failure, the cylinder had to be designed in such a fashion that there was a minimum area for water to escape. The last criteria was the dimensions of the cylinder. The overall dimensions of the instrumented cylinder were prescribed by the facility in which the actual testing of the transducer was to occur. The diameter of the cylinder was to be 5.08 cm with an overall length of 106.6 cm. Both ends of the cylinder were to be threaded to 1 $\frac{1}{2}$ inch national standard pipe thread. All other aspects of the design were left to the investigator.

Design of Cylinder

Figure 9 shows the design of the instrumented cylinder. The instrumented cylinder was composed of three pieces: the transducer cover plate, the wire channel cover plate, and the cylinder itself. The cylinder was designed so that the heat flux transducer would slip into the bored recess shown in cross section B-B, and lightly press fit in the diameter indicated at point D. A small amount of high temperature silicone gasket seal was placed at point E to produce a water tight seal. The transducer cover plate was designed to fit smoothly over the nonactive end of the transducer. This cover plate was held in place using four size 8 socket head screws. The wire channel cover plate was designed to be lightly press fit into the wire channel slot and retained by screws. This cover served to protect the wires from the fluidized bed environment. This design allowed for easy installation and removal of the transducer from the cylinder while minimizing the number of internal and external seals needed to insure that the cylinder was water tight.

The design incorporated a 0.254 mm gap between the cylinder and the clamping ring on the transducer as indicated from the detail on Figure 9. This was intended to help ensure one-dimensional heat transfer within the transducer. This technique has been used successfully with similar transducers used for measuring heat transfer rates in internal combustion engines [Alkidas and Myers (1982); Alkidas and Cole (1985)].

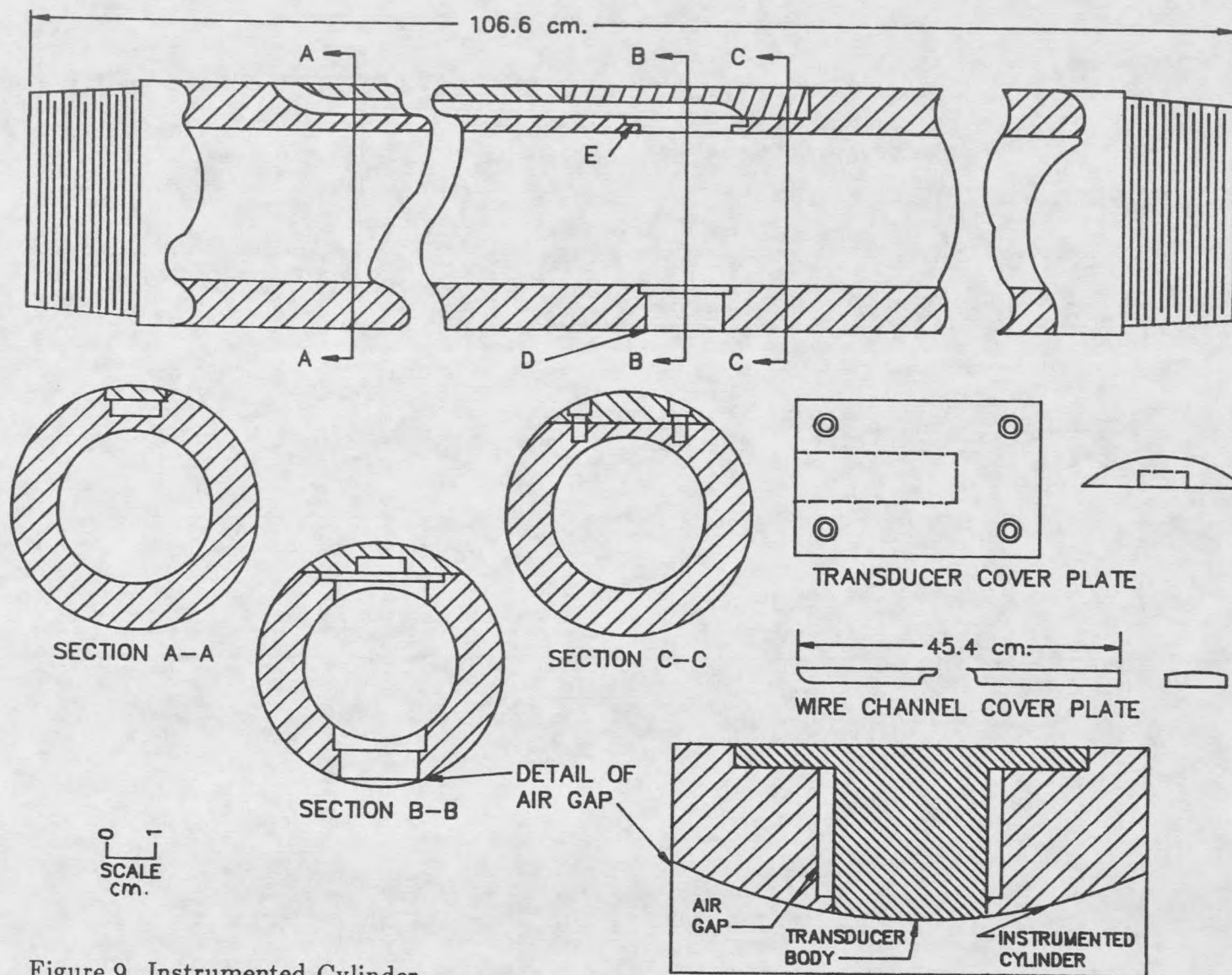


Figure 9. Instrumented Cylinder.

Coolant (water) was supplied to the instrumented cylinder through rotary joints while the cylinder was supported in packing glands. This mounting arrangement was available at the OSU high temperature fluidized bed facility and allowed obtaining instantaneous local heat flux data at several angular positions on the cylinder surface without disconnecting any piping. A very similar mounting system was used previously [George (1981); Alavizadeh et al. (1984); Goshayeshi et al. (1985)].

CHAPTER 4

SIGNAL CONDITIONING CIRCUIT

Principle of Operation

The theory of operation of the transducer and associated signal conditioning circuit was given by George (1986) and will be restated here because of its importance in the validation of the operation of the transducer.

Consider a heat flux transducer embedded in the surface of interest with the boundary conditions as shown in Figure 10. One-dimensional unsteady heat transfer is assumed to occur within the transducer between the surface and in-wall thermocouple. By design, the fluctuations in the surface temperature, $\delta T_w(t)$, are rapid enough that they are damped out before reaching the position $x = L$. Thus, the transducer is semi-infinite as far as the surface temperature fluctuations are concerned and a constant value of T_L can be assumed.

Furthermore, assume the thermal properties of the transducer are constants. This assumption is adequate for the moderate changes in temperature which occur within the transducer. The measured values of the temperatures $T_w(t)$ and T_L are used as boundary conditions to solve the unsteady conduction problem for the region. From this solution, the surface heat flux $q_w(t)$ can be computed by either digital or analog methods. An apparent problem with the formulation is that it is not possible to establish the initial temperature distribution in the transducer body from measurements of T_L and $T_w(t)$ alone. However, the solution for large values of time is independent of the initial temperature distribution within the

transducer and is given by

$$(1) \quad q_w(t) = \frac{k}{L} [\langle T_w \rangle - T_L] + \left[\frac{k\rho c}{\pi} \right]^{1/2} \int_{\eta=0}^t \frac{1}{(t-\eta)^{1/2}} \left[\frac{dT_w}{d\eta} \right] d\eta .$$

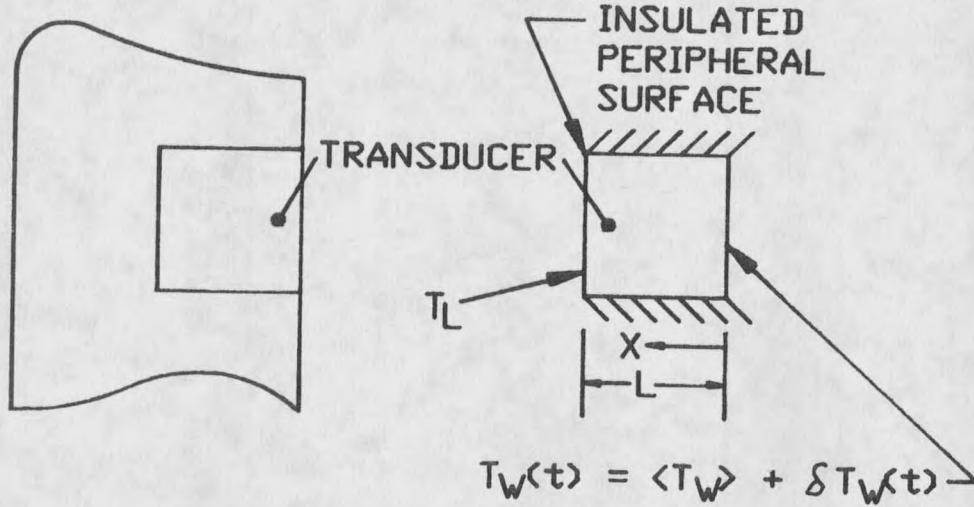


Figure 10. Boundary Conditions for Conduction Problem.

The first term in the above solution represents the constant time-average heat flux which will be denoted $\langle q_w \rangle$. To obtain the second term consider the semi-infinite medium solution for a step change in surface temperature of magnitude ΔT_w imposed at time equal zero as given by Holman (1976). This solution is given by

$$(2) \quad q_w(t) = \left[\frac{k\rho c}{\pi t} \right]^{1/2} \Delta T_w .$$

If the step change in surface temperature occurs at some later time represented by η , the corresponding surface heat flux is given by

$$(3) \quad q_w(t - \eta) = \left[\frac{k\rho c}{\pi(t - \eta)} \right]^{1/2} \Delta T_w .$$

Now apply Duhamel's superposition integral without discontinuities given by

$$(4) \quad q_w(t) = \int_{\eta=0}^t q(t-\eta) \left[\frac{dT_w}{d\eta} \right] d\eta$$

to obtain

$$(5) \quad q_w(t) = \left[\frac{k\rho c}{\pi} \right]^{1/2} \int_{\eta=0}^t \frac{1}{(t-\eta)^{1/2}} \left[\frac{dT_w}{d\eta} \right] d\eta .$$

The second term in (1) is a function of time which represents the fluctuations in the heat flux due to $\delta T_w(t)$ and will be denoted $\delta q_w(t)$.

It is very convenient to utilize analog signal conditioning to produce a d.c. voltage which is linearly related to the instantaneous local heat flux. This allows simple computer calculation of the instantaneous local heat flux without complex numerical procedures. The advantages of an analog rather than a purely numerical solution of the integral in (1) become very significant when data are to be obtained simultaneously from several heat flux transducers.

At this point it should be noted that the distance between the in-wall and surface thermocouples is less than the cylinder wall thickness, that is 6.0 mm versus 9.5 mm, respectively. Therefore, it is only necessary that one-dimensional heat transfer occur within this relatively short length of the transducer body. Furthermore, since only an approximately 1.25 mm $\times$ 6.35 mm rectangular portion of the transducer surface contains both of the surface thermocouple junctions, curvature of the surface will not invalidate the one-dimensional semi-infinite wall assumptions made in determining the input-output relationship for the transducer.

Analog Signal Conditioning

The analog solution of this problem provides a d.c. voltage which is linearly related to the heat flux fluctuations $\delta q_w(t)$. The time-average component of

the local heat flux can be computed easily by use of digital computer methods. Consequently, there is no need for analog signal conditioning to efficiently compute $\langle q_w \rangle$.

By use of the Laplace transformation and the heat conduction formulation for the semi-infinite medium, the following transfer function can be obtained:

$$(6) \quad \frac{\overline{\delta q(s)}}{\overline{\delta T_w(s)}} = \sqrt{k\rho c} \sqrt{s}$$

where s is the Laplace transform parameter. Let

$$(7) \quad \left. \begin{aligned} \bar{v}(s) &= y_1 \overline{\delta q(s)} \\ \bar{e}(s) &= x_1 \overline{\delta T_w(s)} \end{aligned} \right\}$$

where $\bar{v}(s)$ and $\bar{e}(s)$ are the output voltage and input voltage of the signal conditioning circuit, respectively, and y_1 and x_1 are constants. Therefore, the transfer function for the signal conditioning circuit is

$$(8) \quad \frac{\bar{v}(s)}{\bar{e}(s)} = \frac{y_1 \sqrt{k\rho c}}{x_1} \sqrt{s}$$

It follows that the frequency response of the signal conditioning circuit must be

$$(9) \quad \frac{v(j\omega)}{e(j\omega)} = \frac{y_1 \sqrt{k\rho c}}{x_1} \sqrt{\omega} \angle 45^\circ$$

In words, the circuit must provide an amplitude ratio proportional to the square root of frequency and a constant phase angle of 45° . An equivalent form of this relationship was first given by Skinner (1960).

The transfer function given by (6) can be approximated over a finite frequency range by several different circuits. After some experimentation, the circuit shown in Figure 11 was developed specifically for the present application. The circuit shown approximates the transfer function

$$(10) \quad G(s) = (1.194 \times 10^{-2}) \sqrt{s}$$

within 5 percent in amplitude and 2 degrees in phase angle for the frequency range 0.1 Hz to 300 Hz. The constant 1.194×10^{-2} has units of (seconds)^{0.5}. For a detailed analysis of the circuit see Appendix C.

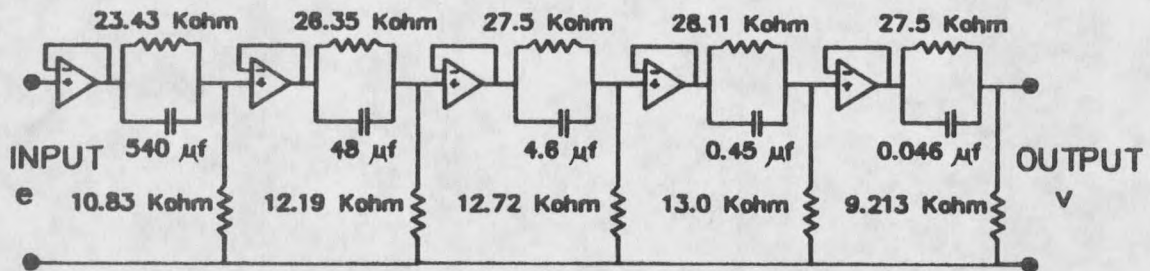


Figure 11. Analog Signal Conditioning Circuit.

A circuit which incorporates the analog signal conditioning circuit discussed above is shown in Figure 12. To improve signal-to-noise ratio, 4-pole Bessel filters were used with a cutoff frequency (-3db) of 900 Hz.

Test of Analog Circuit

The heat flux transducer was connected to the circuit shown in Figure 12 to determine the response to a step change in surface heat flux. A Bell and Howell projector was used to produce the step change in surface heat flux. Figure 13 shows the voltage response (v_4) of the circuit to a step change in surface heat flux. The maximum percent overshoot is approximately 30 percent where the maximum percent overshoot, as given by Raven (1978), is 100 times the maximum amount by which the response overshoots its final steady-state value divided by its final steady state value. The settling time in this case is approximately 2.5 seconds. The probable cause of the large overshoot is that there is a contact resistance between the thermocouple and the cylinder wall (transducer surface).

The circuit was designed with the assumption that the transducer surface temperature thermocouple would respond accurately and instantaneously to changes in the surface temperature. This is not the case if there is even a small contact resistance between the thermocouple and cylinder wall.

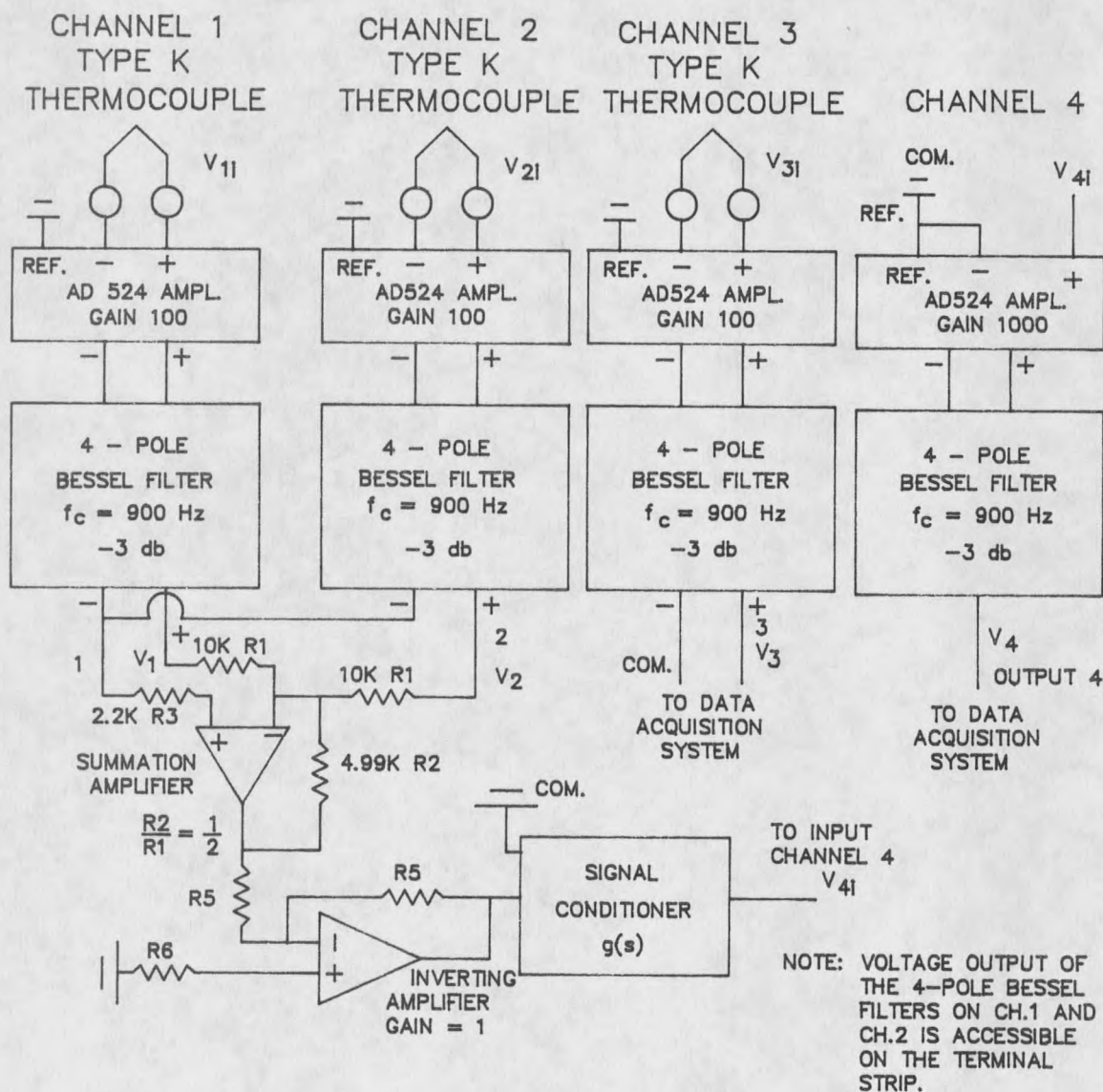


Figure 12. Signal Conditioning Circuit Associated with the Heat Flux Transducer.

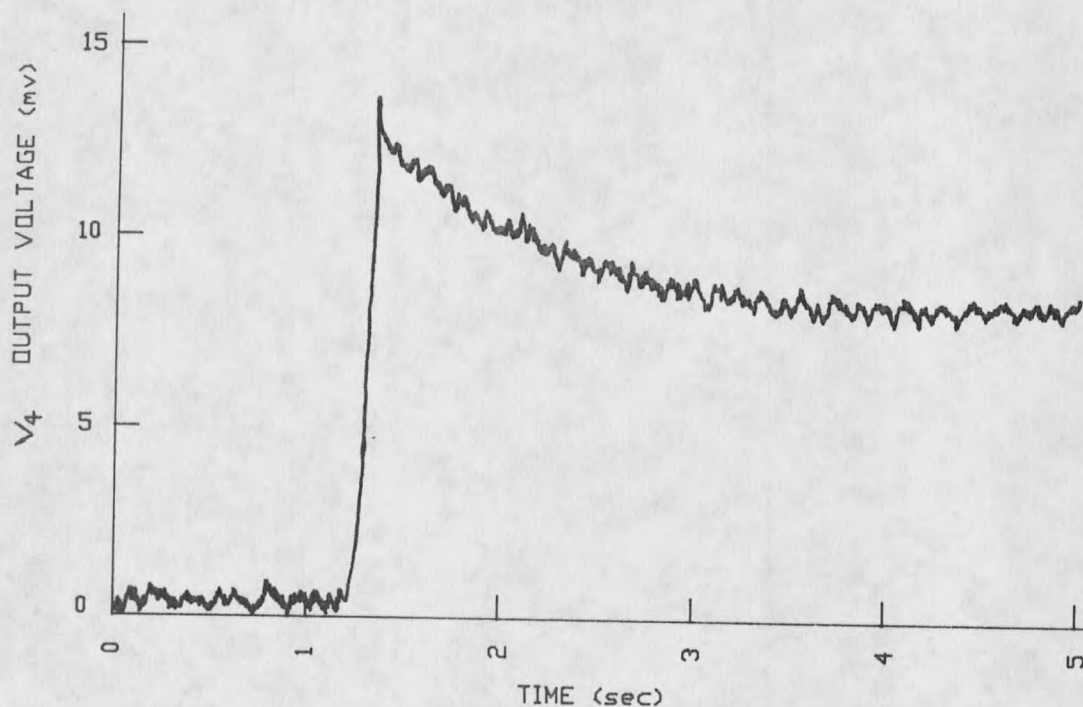


Figure 13. Response of Original Signal Conditioning Circuit to a Step Change in Surface Heat Flux.

Contact Resistance Effects

Consider the assumption that a contact resistance is present between the thermocouple and a semi-infinite medium shown in Figure 14. The corresponding heat flux is given by

$$(11) \quad q_w = \frac{1}{R_C} (T_o - T_w) .$$

Now take the Laplace Transform of (11) and rearrange the terms to get

$$(12) \quad R_C = \frac{\overline{T_o}(s)}{\overline{q_w}(s)} - \frac{\overline{T_w}(s)}{\overline{q_w}(s)}$$

or

$$(13) \quad \frac{\overline{T_O}(s)}{\overline{q_W}(s)} = R_C + \frac{\overline{T_W}(s)}{\overline{q_W}(s)} \quad .$$

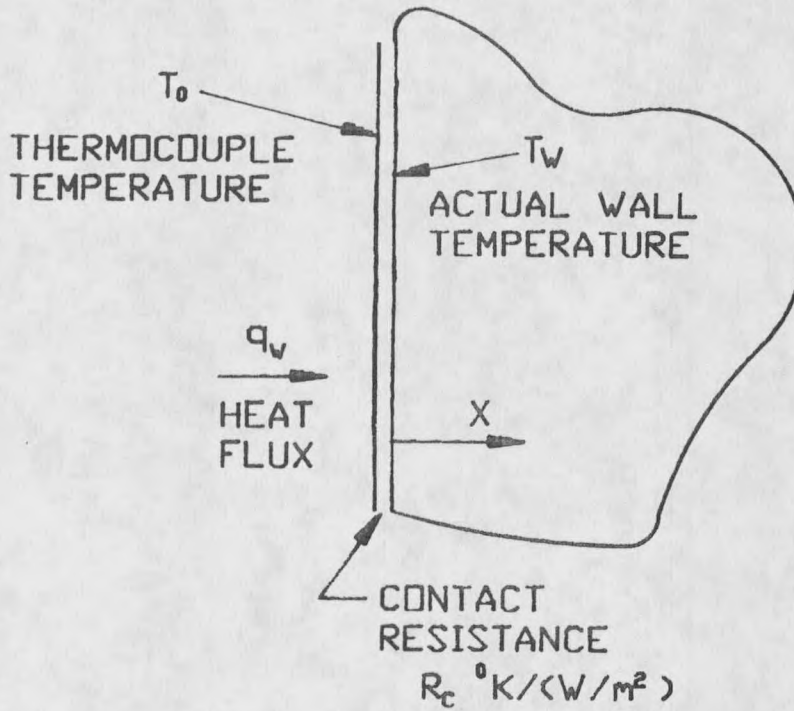


Figure 14. Semi-Infinite Wall With Contact Resistance.

Substitution of (6) into (13) yields

$$(14) \quad \frac{\overline{T_O}(s)}{\overline{q_W}(s)} = R_C + \frac{1}{\sqrt{k\rho c} \sqrt{s}} \quad .$$

Assume a step change in the surface heat flux given by (15) where Q is a constant with units (W/m^2):

$$(15) \quad \overline{q_W}(s) = \frac{Q}{s} \quad .$$

Substitute (15) into (14) to get

$$(16) \quad \overline{T_O}(s) = \frac{R_C Q}{s} + \frac{Q}{\sqrt{k\rho c} s\sqrt{s}} \quad .$$

The inverse Laplace Transform of (16) as given by Spiegel (1968) yields

$$(17) \quad T_O(t) = R_C Q + \frac{2Q}{\sqrt{k\rho c}} \sqrt{t/\pi} \quad .$$

Now assume that $T_O(t)$ is the actual wall temperature T_w and compute the indicated heat flux (q_{IND}) by replacing $\overline{T_O}(s)$ in (6) with (16). The result is

$$(18) \quad \overline{q_{IND}}(s) = \frac{\sqrt{k\rho c} R_C Q}{\sqrt{s}} + \frac{Q}{s} \quad .$$

The inverse Laplace Transform of (18) as given by Spiegel (1968) yields

$$(19) \quad q_{IND}(t) = \frac{\sqrt{k\rho c} R_C Q}{\sqrt{\pi t}} + Q \quad .$$

Figure 15 shows the solution of (19) graphically. The first term in (19) represents the overshoot that decays to zero as the time becomes large. The second term in (19) represents the actual heat flux on the surface. In order to correct for the effects of contact resistance, the circuit was experimentally adjusted until the overshoot was no longer present.

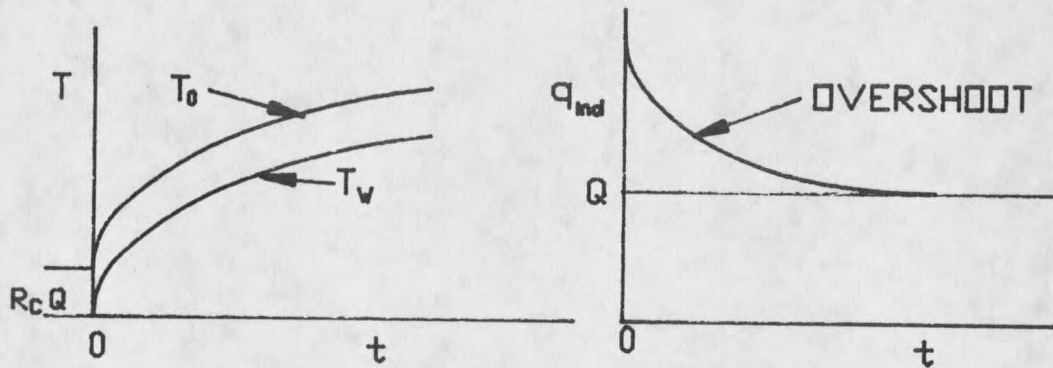


Figure 15. Solution of the Indicated Heat Flux versus Time for Contact Resistance.

Adjusted Analog Circuit

Figure 16 shows the adjusted analog circuit. The resistors were adjusted until the overshoot disappeared. The process for doing this was trial and error. One resistor was adjusted and then the signal conditioning circuit was tested for response to a step input in surface heat flux. This process continued until there was no longer any significant overshoot. The circuit shown in Figure 16 approximates the transfer function

$$(20) \quad G(s) = (1.207 \times 10^{-2})\sqrt{s}$$

within 7 percent in amplitude and 9 degrees in phase angle for the frequency range 0.4 Hz to 240 Hz. The constant 1.207×10^{-2} has units of (seconds)^{0.5}.

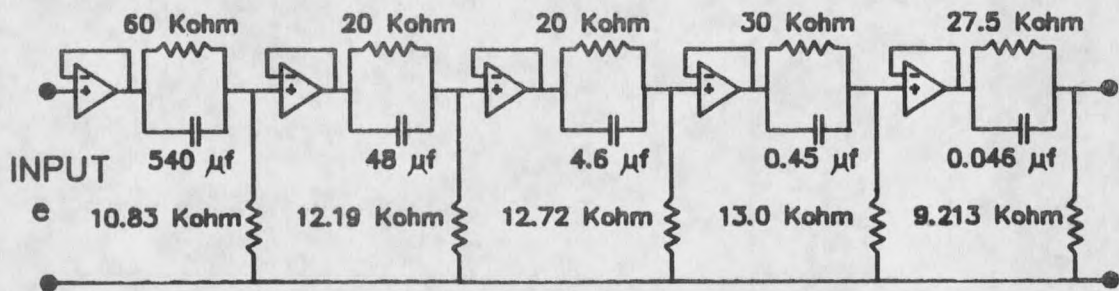


Figure 16. Adjusted Analog Circuit.

Validation of Analog Circuit

A useful check on the input-output relationship for the analog signal conditioning circuit was to subject the transducer to a step change in surface heat flux and verify that a near step change in the output voltage (v_4) is produced. A propane torch and shutter were used to produce a step change in heat flux at the transducer surface. The output voltage (v_4) was recorded on a digital storage oscilloscope. Figure 17 shows the response of the transducer to a step change in

surface heat flux level of approximately 200 KW/m^2 . The settling time as shown in Figure 17 was approximately 2 ms. The overshoot is almost nonexistent and could not be measured in this application.

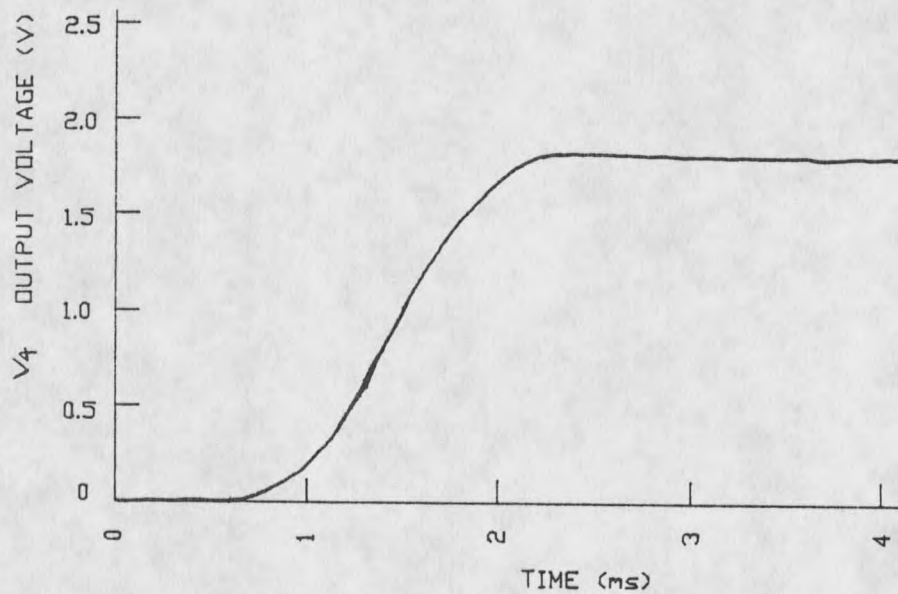


Figure 17. Response of Adjusted Analog Circuit to a Step Change in Surface Heat Flux.

The next test was to check the response of the circuit for a step change in surface heat flux of long duration (10 seconds). A Bell and Howell projector was used to produce the step change in surface heat flux. The magnitude of the heat flux applied to the surface was approximately 1600 W/m^2 . Figure 18 shows the results of this test. The output from the analog circuit, which was recorded on a XY plotter, was constant within ± 6 percent for the entire 8 seconds.

A similar test with a surface heat flux duration of 50 seconds provided an output voltage (v_4) which was constant within ± 10 percent. Therefore, the circuit and transducer combined respond accurately to events that have a longer duration than expected in the fluidized bed. The modified signal conditioning circuit shown in Figure 16 was used in all subsequent work reported in this thesis.

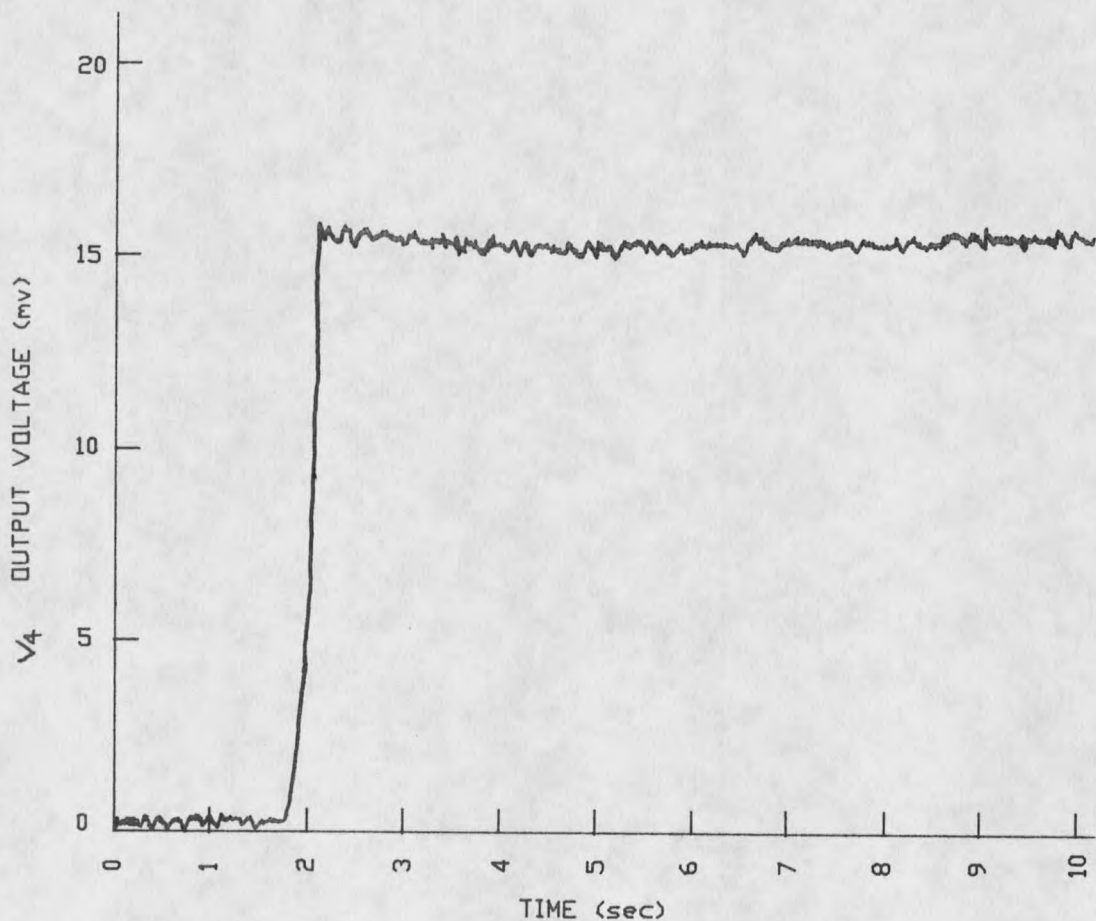


Figure 18. Response of Adjusted Analog Circuit to a Long Duration Step Change in Surface Heat Flux.

CHAPTER 5

CALIBRATION

Introduction

The instantaneous local heat flux is composed of two terms. One term represents the time average component of the heat flux while the second term represents the instantaneous changes in the heat flux. The instantaneous local heat flux is computed from

$$(21) \quad q_w(t) = \frac{k}{L} [< T_w > - T_L] + \frac{1}{\beta} [v_4(t) - < v_4 >] .$$

The reason for calibration of the device is to determine the input-output characteristics of the transducer and analog signal conditioning unit combined. The calibration constant is defined as

$$(22) \quad \beta = \frac{dv_4}{dq_w} .$$

By considering the input-output relationship of each of the thermocouples and of the circuit elements in the original analog signal conditioning unit it can be shown analytically that

$$(23) \quad \beta = \frac{10^5 a}{83.8 \sqrt{k \rho c}} .$$

The constant 83.8 has units of (seconds)^{-0.5}. For a type K thermocouple, the sensitivity at the surface temperature of interest is relatively constant over the range

of operating temperatures that will be considered in this thesis. The calibration constant can be rewritten in an alternate form since

$$(24) \quad \alpha = \frac{k}{\rho c}$$

Thus, the calibration constant becomes

$$(25) \quad \beta = \frac{10^5 a \sqrt{\alpha}}{83.8k}$$

The thermal conductivity and the thermal diffusivity of the material are both temperature dependent functions. These functions can be obtained from data found in Touloukian et al. (1970, 1973). These relationships are approximated by the following equations:

$$(26) \quad k = 0.01703T^{0.38393} \quad \text{for} \\ 273.2K < T < 700K$$

$$(27) \quad \alpha = 0.03013 + 0.00002T \quad \text{for} \\ 291K < T < 763K$$

The thermal conductivity is given in units of (W/cm K) and the units for the thermal diffusivity are (cm²/sec).

Analytical Solution for β

Given the three temperature measurements of interest, the value of β can be determined analytically. The major source of uncertainty in determining the calibration constant analytically is that the material properties must be accurately specified. The values found in Touloukian et al. (1970, 1973) are for ANSI 304 stainless steel of nominal composition. Table 1 shows values of the calibration constant for various values of the surface temperature. The surface temperature

is used to compute β since the penetration depth of the temperature fluctuation into the transducer body is small. The surface temperature is defined as

$$(28) \quad \langle T_w \rangle = \frac{1}{2} (T_1 + T_2) \quad .$$

Surface temperature (T_w) (K)	Calibration constant β (μ V/(W/m ²))
293	6.009
300	5.965
350	5.699
400	5.486
450	5.311
500	5.166
550	5.042
600	4.936
650	4.843
700	4.761

Table 1. Analytical Values of β at the Indicated Surface Temperature.

Since the circuit had been adjusted to compensate for the effects of contact resistance between the surface thermocouple and cylinder wall, it was necessary to perform an experimental calibration for β . The analytical values of β shown in Table 1 will be used only as an order of magnitude comparison for the experimental values of β .

Experimental Values of β

Calibration of the heat flux transducer and associated signal conditioning circuit to determine β was conducted using a radiant heat source to produce a step change in the surface heat flux. The radiant heat source used was a Bell and Howell movie projector with the shutter removed. The reason for removing the shutter was that initial tests showed a fluttering heat source which could be attributed to the shutter movement in front of the lens. The maximum heat flux the projector would provide at the surface of the cylinder was approximately 10000 W/m^2 .

A commercial heat flux transducer with known calibration characteristics was used to determine the step change in heat flux at the surface of the cylinder. The commercial transducer used is a Micro-Foil heat flow sensor model 20455-1 purchased from RdF Corporation, Hudson, New Hampshire. See Appendix D for calibration literature on the commercial heat flux transducer. The Micro-Foil transducer shown in Figure 19 was attached to one end of the instrumented cylinder using RdF AP cement.

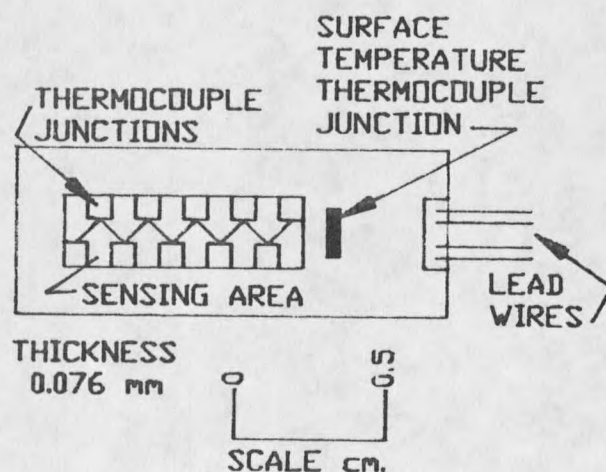


Figure 19. Micro-Foil Heat Flux Transducer Used For Calibration.

Both transducers were coated with soot from an acetylene torch with the oxygen turned off so that the surface emissivity would be the same for each transducer. A schematic of the test set up is shown in Figure 20. A metal shutter was placed in front of the projector lens and then pulled away to produce a step change in heat flux. The step change in heat flux had a duration longer than the settling time of each transducer. The calibration constant β was established by comparing the voltage output (v_4) of the circuit shown in Figure 12 with the heat flux indicated by the heat flux transducer.

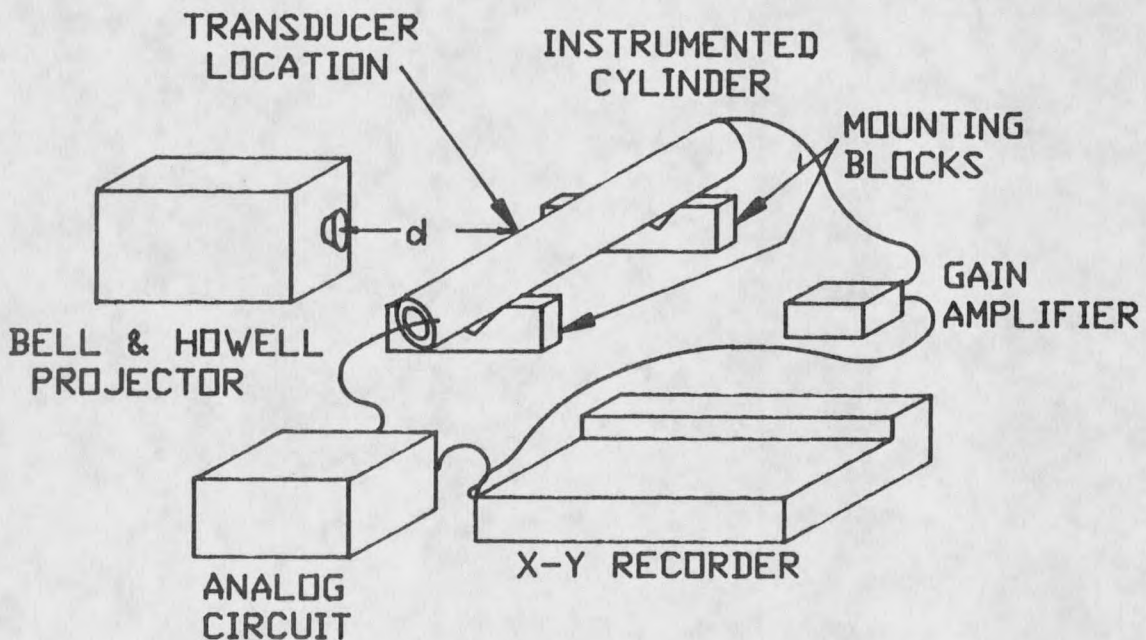


Figure 20. Schematic of Calibration Set Up.

The first step in the testing procedure was to determine the magnitude of the step change in heat flux produced by the projector. To do this the Micro-Foil transducer was placed in front of the projector at various distances (d) from the projector lens. The lead wires that indicated the output voltage for the heat

flux were connected to an instrumentation amplifier with a gain of 100 and this output was recorded on a Houston Instruments model 200 XY recorder. The surface temperature of the cylinder was also recorded for each test. This test was run 20 times at each of five different locations from the projector. Then the average heat flux and surface temperature was determined for each position.

The next step was to repeat the process using the experimental transducer at the same indicated surface temperature. The value of the calibration constant (β) was calculated from (29) at each of the different locations and then the average value of β was determined:

$$(29) \quad \beta = \frac{\langle v_4 \rangle}{\langle q_{mf} \rangle}$$

The value for β at a surface temperature of 298 K was calculated to be 9.42 $\mu\text{V}/(\text{W}/\text{m}^2)$. The calibration process was then extended to determine β for surface temperatures higher than 298 K.

The instrumented cylinder was heated in an oven to a temperature of 450 K. The cylinder was then placed on the mounting blocks at a distance of 23 cm from the projector lens and allowed to cool. When the cylinder had reached a temperature of 380 K the rate of cooling had slowed to the point that useful data could be taken. The output voltage and surface temperature measurements were recorded until the cylinder had cooled to 298 K. A 110 μf capacitor was placed at the input of the XY recorder to remove the offset voltage caused by the increased surface temperature so that the sensitivity of the XY recorder could be increased to give results that were easier to interpret. A total of 240 measurements were taken. The measurements were divided into six average surface temperature ranges and averaged.

The surface heat flux was established by positioning the Micro-Foil transducer 23 cm from the projector lens and averaging 20 heat flux measurements. The value of β was then recalculated from (29) for each of the temperature ranges. Table 2 shows the values of β obtained for each indicated surface temperature.

Surface temperature (T_w) (K)	Calibration constant β ($\mu \text{ V}/(\text{W}/\text{m}^2)$)
298	9.42
344	8.93
348	8.92
353	8.89
371	8.61
378	8.53

Table 2. Calibrated Values of β Obtained at the Indicated Surface Temperature.

Investigation of Calibration Results

Figure 21 shows both the analytical and experimental values of β versus T/T_{REF} where T_{REF} is an arbitrary constant chosen to be 298 K. The experimental values of β are of the same order of magnitude as the analytical values of β . Both values of the calibration constant display the same general characteristics; that is, they both decrease as the surface temperature increases and they also approximate a straight line on log-log paper.

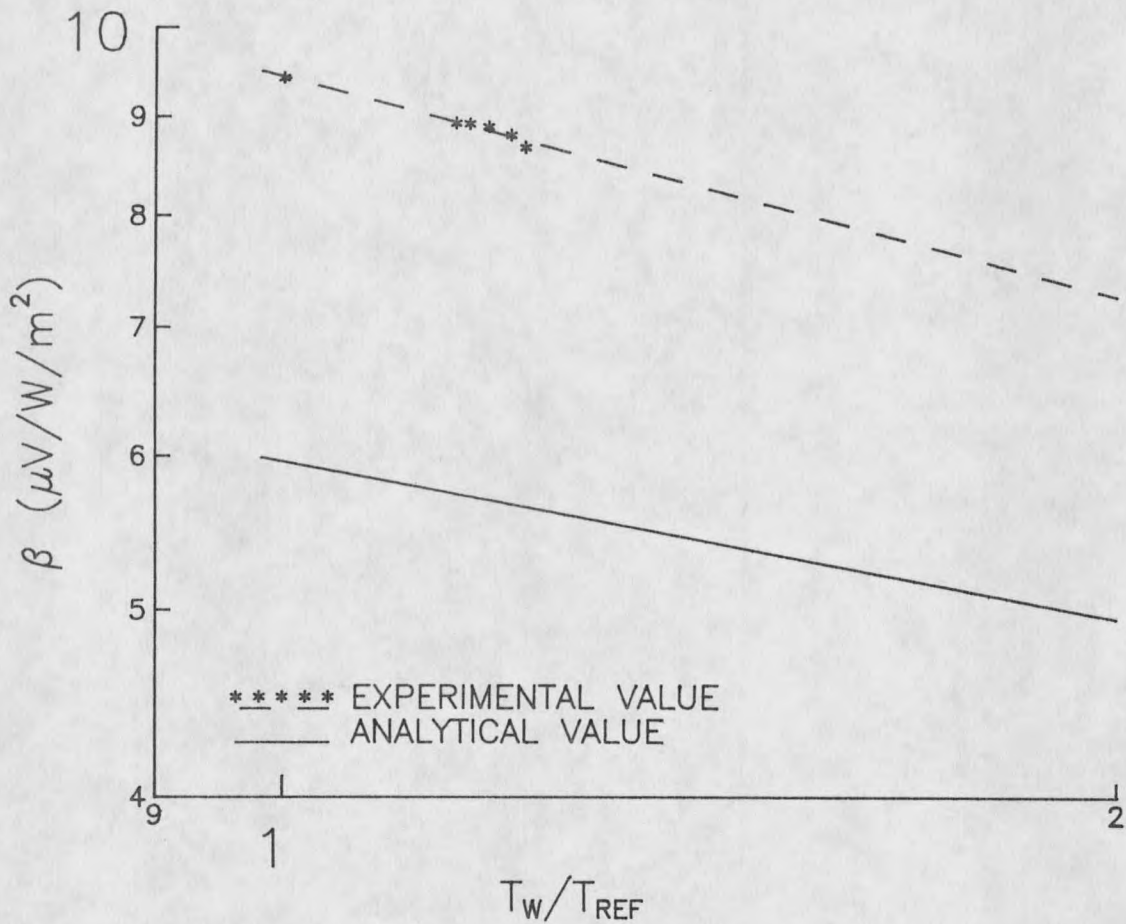


Figure 21. Plot of Analytical and Experimental Values of the Calibration Constant versus Ratio of Surface Temperature to Reference Temperature.

With the aid of a curvefitting program named CURVEFIT and a Zenith Z100 computer the equations for these functions were determined. The equation for the analytical value of β as a function of T/T_{REF} is given by

$$(30) \quad \beta = 6.027 \left(\frac{T}{T_{REF}} \right)^{-0.27} .$$

Likewise, the equation for the experimental value of β as a function of T/T_{REF} is

given by

$$(31) \quad \beta = 9.459 \left(\frac{T}{T_{REF}} \right)^{-0.409}$$

Analytical Solution for k/L

Table 3 shows values of k/L for various values of the bulk temperature. Since k/L affects only the time average component of the heat flux, the bulk temperature was used because it more closely approximates the average temperature over the length (L) of interest. The bulk temperature is given by

$$(32) \quad T_b = \frac{1}{2} \left[\frac{1}{2} (T_1 + T_2) + T_3 \right]$$

Bulk temperature T_b (K)	k/L (W/m ² K)
293	2511.7
300	2535.0
350	2690.0
400	2831.7
450	2963.3
500	3085.0
550	3200.0
600	3308.3
650	3411.6
700	3510.0

Table 3. Analytical Value of k/L at the Indicated Bulk Temperature.

Upon further inspection of Touloukian et al. (1970) some specific compositions of ANSI 304 stainless steel were listed and it turns out that the value of the

thermal conductivity varies considerably with only slight differences in the overall composition of the specimen. The value of L is also not known exactly even though the bottom of the hole that contains the welded junction thermocouple is located 6 mm from the surface of the transducer. The effective thermocouple junction may not lie in the bottom of the hole since the thermocouple was silver soldered in place. Since there exist uncertainties in the composition of the transducer material and the value for L it is necessary to carry out experimental calibration for the value of k/L .

Experimental Values of k/L

The calibration for the value of k/L was done in the OSU high temperature fluidized bed. The method used for calibration was again one of comparison. A Micro-Foil heat flux transducer purchased from RdF Corporation and used by Goshayeshi (1989) was used as the standard. This heat flux transducer was mounted on a 5.08 cm diameter cylinder made of SAE 660 Bronze. The transducer was covered with a 0.127 mm thick sheet of stainless steel to protect it from the environment of the fluidized bed. Both cylinders were placed in the fluidized bed at approximately 0.36 m above the distributor plate. The cylinders were placed approximately 0.2 m apart so as not to interfere with each other. The transducers were placed facing down so that they were in a direct line with the hot gases entering the fluidized bed. This angular position was chosen because according to Goshayeshi (1989), the overall heat transfer coefficient was least effected by the changes in superficial gas velocity at that point. In order to make a comparison between the transducers it was assumed that the time average heat transfer coefficient would be the same, since the bed condition was the same and surface temperature within 20 K for both transducers.

The data needed for calibration was the time average surface temperature of both transducers, the time average heat flux as indicated by the commercial transducer, the in-wall temperature indicated by the experimental transducer, and the steady state bed temperature.

The first step in the calculation of k/L was to determine the time average heat transfer coefficient as indicated by the commercial-transducer. The equation used was

$$(33) \quad \langle h \rangle = \frac{\langle q_{mf} \rangle}{(T_{bed} - \langle T_{mf} \rangle)} .$$

Then the time average heat transfer was determined for the experimental transducer using the time average heat transfer coefficient calculated from (33). The time average heat transfer was calculated by

$$(34) \quad \langle q_w \rangle = \langle h \rangle (T_{bed} - \langle T_w \rangle) .$$

The corresponding value of k/L was calculated from

$$(35) \quad \frac{k}{L} = \frac{\langle q_w \rangle}{(\langle T_w \rangle - T_L)} .$$

There was a problem with this calibration method related to a defect in the fluidized bed used. The distributor plate which let the hot gas into the bed was warped from repeated high temperature use. Consequently, the bubbles were not distributed equally in the bed. Therefore, instead of calculating k/L for a variety of surface temperatures, the value of k/L was determined at a single surface temperature. Several test runs were done at a constant surface temperature of ($\langle T_w \rangle = 409$ K); the value of k/L was calculated to be constant at 1571.9 (W/m²K). This assumption does not alter the accuracy of the time average heat transfer significantly because the surface temperature changed by less than 55 K for all the test runs considered in this thesis.

CHAPTER 6

DATA ACQUISITION SYSTEM

Components

The data acquisition system was composed of the following items:

1. A Hewlett Packard 85B computer.
2. A Hewlett Packard 3045A automatic data acquisition/control system.
3. A Hewlett Packard 3497A data acquisition/control unit.
4. A Hewlett Packard 3456A digital voltmeter with math capabilities.
5. A Hewlett Packard 3437A system voltmeter.
6. A Hewlett Packard 9121 dual disk drive.
7. The analog signal conditioning circuit.
8. A computer program that directed the data acquisition system to take the necessary data.

Operation

The analog signal conditioning unit has four output channels as shown in Figure 12 of Chapter 5. These channels were connected to the data acquisition system in the following way:

1. Channel one, which corresponds to one of the surface thermocouples, was connected to the hp 3497A data acquisition/control unit and was monitored by the hp 3456A digital voltmeter.

2. Channel two, which corresponds to the other surface thermocouple, was also connected to the hp 3497A data acquisition/control unit and was monitored by the hp 3456A digital voltmeter.
3. Channel three, which corresponds to the in-wall thermocouple, was connected to the hp 3497A data acquisition/control unit and was monitored by the hp 3456A digital voltmeter.
4. Channel four, which represents the instantaneous fluctuations in surface heat flux, was connected to the hp 3497A data acquisition/control unit and was monitored by the hp 3456A digital voltmeter and the hp 3437A system voltmeter.

A welded junction type K thermocouple located on the terminal strip of the analog circuit was connected to the hp 3497A data acquisition/control unit and was monitored by the hp 3456A digital voltmeter. This thermocouple was used to measure the reference junction temperature for the thermocouples located in the transducer. There is a temperature detector located in the hp 3497A data acquisition/control unit which was used as the reference junction for the thermocouple located on the terminal strip.

The data acquisition system was connected directly to the hp 85B computer via the hp-IB bus. The computer program named NSFHT in Appendix A directed the data acquisition system to perform the following operations:

1. The hp 3456A digital voltmeter took a voltage reading of the temperature detector inside the data acquisition/control unit.
2. The hp 3456A digital voltmeter took a voltage reading of the thermocouple connected to the terminal strip of the analog signal conditioning circuit.
3. The hp 3456A digital voltmeter took 4000 voltage readings from channel three, at a sampling interval of 5 ms, and averaged them.

4. The hp 3456A digital voltmeter took 4000 voltage readings from channel four, at a sampling interval of 5 ms, and averaged them.
5. The hp 3437A system voltmeter took 9996 voltage readings from channel four with a sampling interval of 5 ms.
6. The hp 3456A digital voltmeter took a reading of the temperature detector inside the data acquisition/control unit.
7. The hp 3456A digital voltmeter took a voltage reading of the thermocouple connected to the terminal strip of the analog signal conditioning circuit.
8. The hp 3456A digital voltmeter took 4000 voltage readings from channel one, at a sampling interval of 5 ms, and averaged them.
9. The hp 3456A digital voltmeter took a reading of the temperature detector inside the data acquisition/control unit.
10. The hp 3456A digital voltmeter took a voltage reading of the thermocouple connected to the terminal strip of the analog signal conditioning circuit.
11. The hp 3456A digital voltmeter took 4000 voltage readings from channel two, at a sampling interval of 5 ms, and averaged them.
12. The average voltage readings from channels one through four and corresponding reference readings were stored on disk. The 9996 voltage readings from the hp 3437A system voltmeter were then stored on disk.
13. The hp 3437A system voltmeter took another 9996 voltage readings from channel four and stored them on disk.

This whole process takes about twenty minutes with most of that time devoted to writing to the disk. For a complete listing of the program NSFHT see Appendix A.

Data Reduction

Once the raw data in the form of measured voltages has been stored on disk, another computer program named TRANS, in Appendix A, reduces the data to temperatures and instantaneous local heat transfer coefficients. This is done by the following method. The two surface temperatures and in-wall temperature were calculated using the hp subprogram Sc_ temp and the corresponding reference temperatures. The reference temperatures were calculated using the hp subprogram Sc_ temp and the corresponding reference temperature indicated by the hp 3497A data acquisition/control unit. The next step was to calculate the average surface temperature of the transducer. The average surface temperature is given by

$$(36) \quad \langle T_w \rangle = \frac{\langle T_1 \rangle + \langle T_2 \rangle}{2}$$

The time average local heat transfer coefficient was calculated next from (37). The value used for k/L was determined from experimental calibration of the heat flux transducer to be $1571.9 \text{ W/m}^2\text{K}$.

$$(37) \quad \langle h \rangle = \frac{k}{L} \frac{(\langle T_w \rangle - \langle T_3 \rangle)}{(T_{bed} - \langle T_w \rangle)}$$

Each instantaneous local heat transfer coefficient value was then calculated from (38) where β was the calibration constant that was obtained from experimental calibration of the heat flux transducer:

$$(38) \quad h(t) = \langle h \rangle + \frac{1}{\beta} \left[\frac{v_4(t) - \langle v_4 \rangle}{T_{bed} - \langle T_w \rangle} \right] 10^6$$

The value for β was determined from (39) and has the units $(\mu\text{V}/(\text{W}/\text{m}^2))$. The multiplier 10^6 is a units conversion factor. The instantaneous local heat transfer

coefficients were then stored on disk. These values were plotted using an hp 86B computer and an hp plotter. For a complete listing of the program TRANS see Appendix A.

$$(39) \quad \beta = 9.459 \left(\frac{T_w}{T_{REF}} \right)^{-0.409}$$

CHAPTER 7

EXPERIMENTAL RESULTS

Introduction

A physical model of the heat transfer process needs to be explained before the experimental results can be discussed. Figure 22 shows the physical model of what occurs in a gas fluidized bed. The gas fluidized bed is composed of an emulsion phase, which contains both particles and gas, and a bubble phase, which is almost void of particles. The modes of heat transfer that occur between the cylinder wall and the emulsion phase are conduction between the particles and cylinder wall across a very thin gas film, gas convection due to gas flowing in the interstitial channels between particles and, at high temperatures, thermal radiation. The major mode of heat transfer in the bubble phase is convection with radiation occurring across the bubble at high temperatures. At the top of the cylinder there is a stack of particles (lee stack) that is present when the fluidized bed is in operation. This lee stack is occasionally washed away by the particle motion and replaced with a fresh stack.

At the start of testing in the OSU fluidized bed, one of the surface thermocouple junctions opened and remained that way. This has happened before, during initial testing, but the junction always closed again. This time, however, the junction would not close even when the junction was filed with a metal file. The reason that this happened was most likely due to the fact that the mica sheet that is between the individual wires was too thick to allow the wire burrs to bridge

over and touch each other. Therefore, all of the testing reported in this chapter was done with only one surface thermocouple junction.

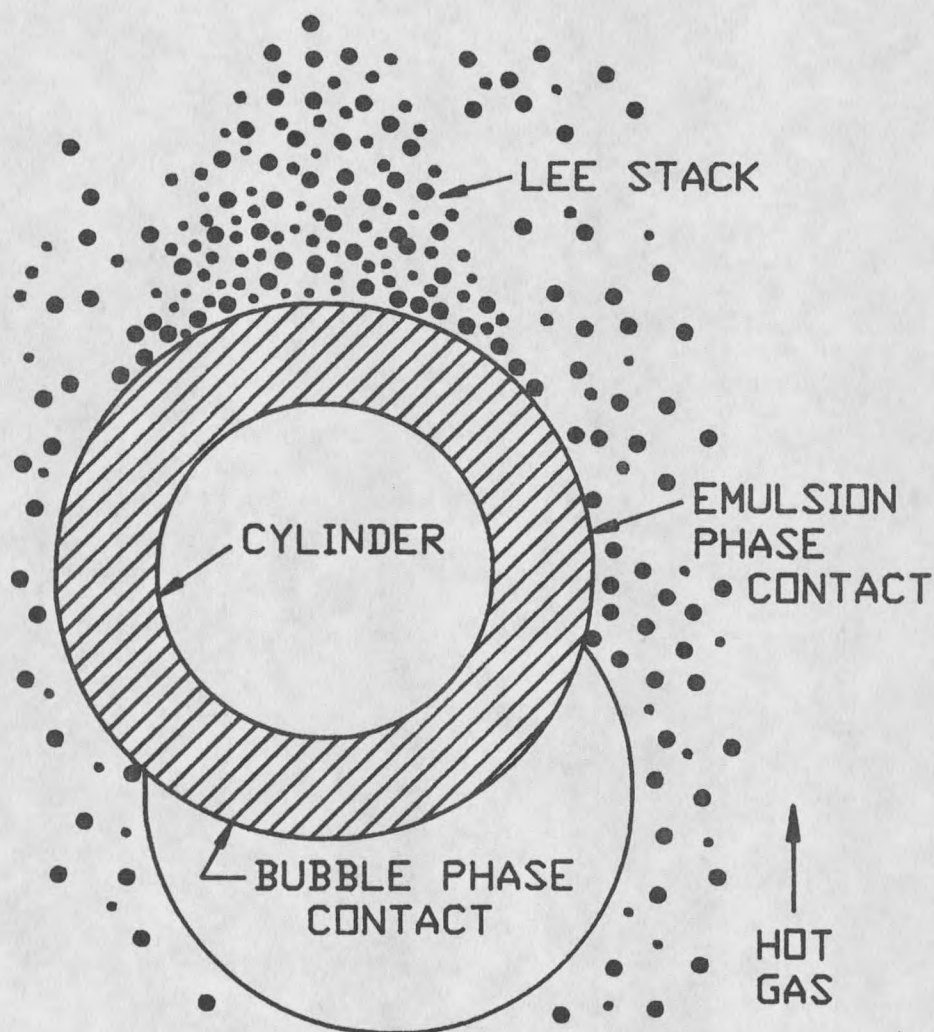


Figure 22. Physical Model of Fluidized Bed in Operation.

The value of the calibration constant (β) had to be changed since it was based on two surface thermocouple junctions. Theory predicted that it should be half of the original value. The value of the calibration constant was checked at room

temperature using the same calibration procedure as discussed in Chapter 6. The value for the calibration constant was determined to be half of the original value.

Test Conditions

The testing was done at the OSU high temperature fluidized bed facility located in Corvallis, Oregon. For a complete description of this facility see Appendix B.

Bed Material

A granular refractory material, commercially named Ione Grain, with density of 2700 kg/m^3 was used for the bed particles. Two sizes of particles were used in the testing of the transducer. They had a mean diameter of 0.9 mm and 2.1 mm. Initially, only the 0.9 mm particles were going to be used but during testing it was discovered that the maximum temperature that the fluidized bed could maintain with the small particles was approximately 820 K. The reason for this is because the flow rate of fuel gas and air needed to fluidize the bed was insufficient to heat the particles to combustion level temperatures (approximately 1123 K). Therefore, in order to reach combustion level temperatures the larger particles had to be used.

Test Procedure

The fluidized bed was in actual operation for approximately 70 hours. During that time three test runs were completed. Table 4 shows the test conditions for the three test runs that were completed. Each test run consisted of instantaneous surface heat flux measurements being recorded at five angular positions around

the cylinder. The angular positions of the transducer at which data were recorded are shown in Figure 23.

Run number	1	2	3
Bed temperature (K)	834.6	821.3	1016.3
Gas velocity (m/s)	1.0	2.81	2.81
Particle size (mm)	0.9	2.1	2.1

Table 4. Fluidized Bed Conditions for Each of the Test Runs Considered.

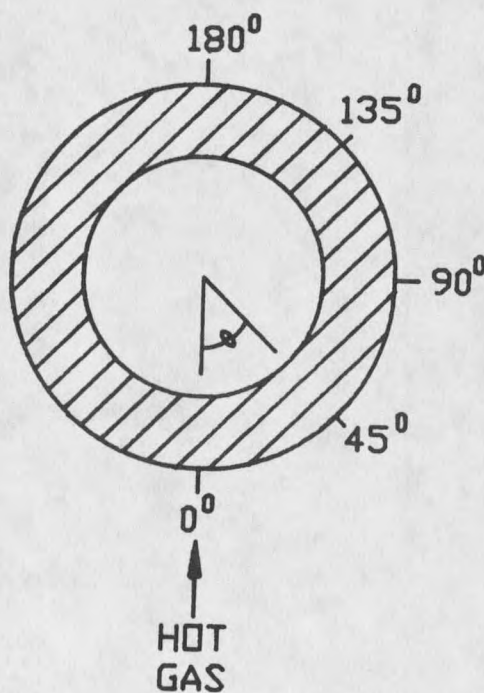


Figure 23. Angular Positions at Which Heat Flux Measurements were Made.

The first step in the testing of the transducer was to allow the fluidized bed to reach steady-state conditions. Once this happened, 100 seconds of instantaneous surface heat flux measurements were taken for each of the five angular positions using the data acquisition system discussed in Chapter 6. The first set of data was taken at $\theta = 0^\circ$ and then the cylinder was rotated in the packing glands 45° and the next data set was recorded. This procedure continued until data was taken at all five angular positions. The bed temperature was maintained to within ± 5 K of the specified value for all three test cases considered.

Test Results

The experimental results presented in this thesis are time average local heat transfer coefficient, spatial average heat transfer coefficient, and instantaneous local heat transfer coefficient. These values are presented for each of the indicated bed temperatures, particle sizes, and superficial gas velocities investigated in this thesis.

Time Average Local Heat Transfer Coefficient

Tables 5-7 show the indicated surface temperature and time average heat transfer coefficient for each of the run conditions considered.

Angular position (θ) (degrees)	Surface temperature (T_w) (K)	Time average heat coefficient < h > (W/m ² K)
0	400.2	246.6
45	403.8	261.6
90	426.3	323.7
135	448.1	384.3
180	447.0	383.7

Table 5. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s.

Angular position (θ) (degrees)	Surface temperature (T_w) (K)	Time average heat coefficient < h > (W/m ² K)
0	366.2	155.3
45	370.3	166.9
90	391.7	230.5
135	402.1	248.2
180	411.1	272.4

Table 6. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Angular position (θ) (degrees)	Surface temperature (T_w) (K)	Time average heat coefficient < h > (W/m ² K)
0	409.0	175.1
45	415.8	190.6
90	439.6	240.7
135	454.8	260.1
180	461.3	274.3

Table 7. Indicated Surface Temperature and Time Average Heat Transfer Coefficient for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Figures 24 and 25 show the time average local heat transfer coefficient for both particle sizes and the three bed temperatures considered in this thesis.

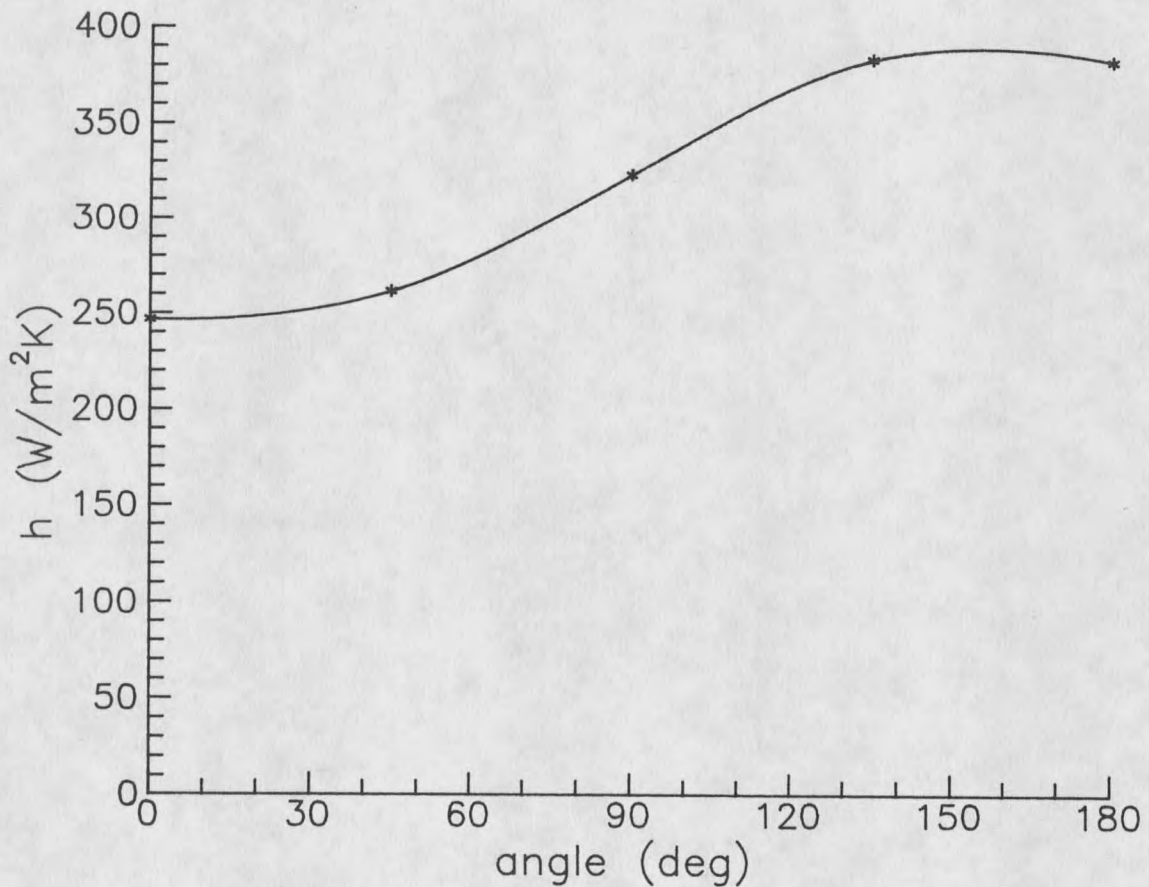


Figure 24. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 0.9$ mm, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

For comparison purposes, the results indicated by George (1984) are shown in Figures 26 and 27. Even though the bed conditions are not identical to the ones considered here, some interesting conclusions can be drawn. The superficial gas velocities that George (1984) investigated are considerably lower than the one considered here for the approximately same particle size. However, the data

considered in this thesis agrees with the trend indicated by the previous study for both of the temperatures considered.

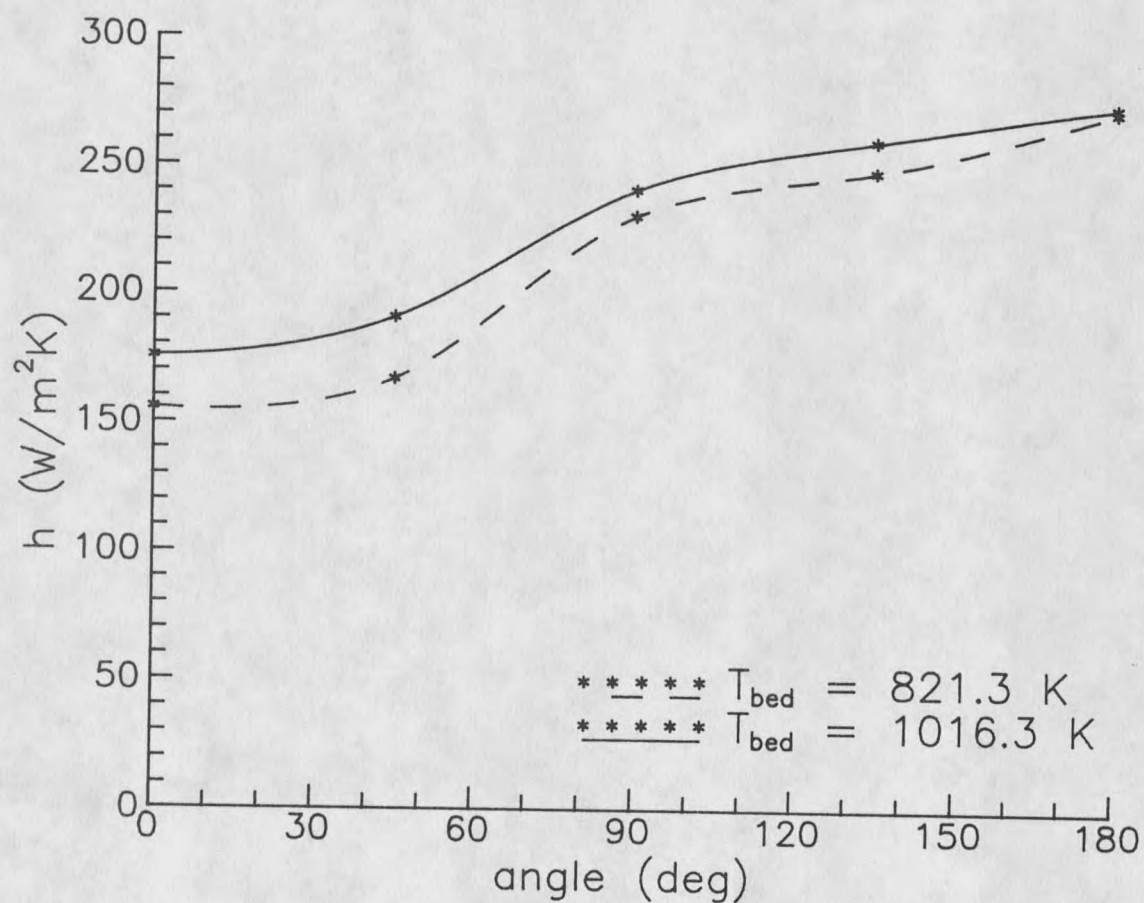


Figure 25. Time Average Local Heat Transfer Coefficient versus Angular Position for $dp = 2.1$ mm, $U_0 = 2.81$ m/s, $T_{bed} = 821.3$ K and $T_{bed} = 1016.3$ K.

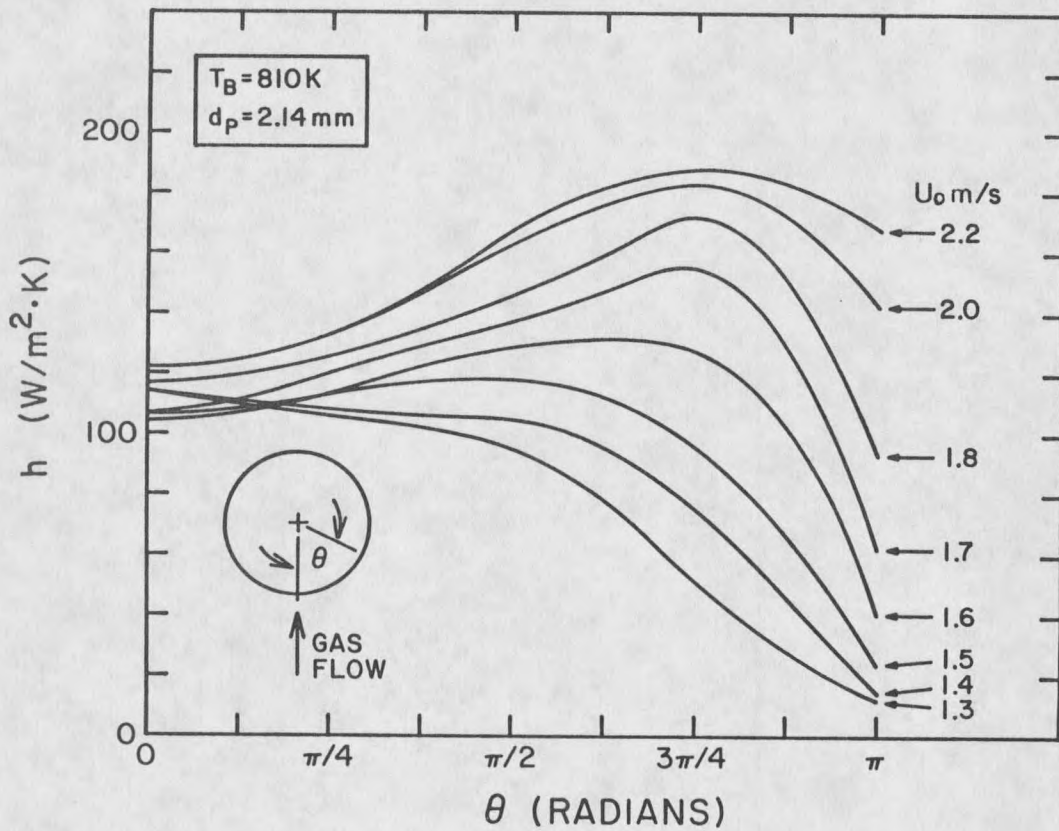


Figure 26. Time Average Local Heat Transfer Coefficient versus Angular Position for $d_p = 2.14 \text{ mm}$ and $T_{bed} = 810 \text{ K}$ as Given by George (1984).

To the author's knowledge, no time average local heat transfer coefficient data for the bed conditions considered here have been reported. Thus, in order to make accurate comparisons with previous data, the spatial average heat transfer coefficient was used.

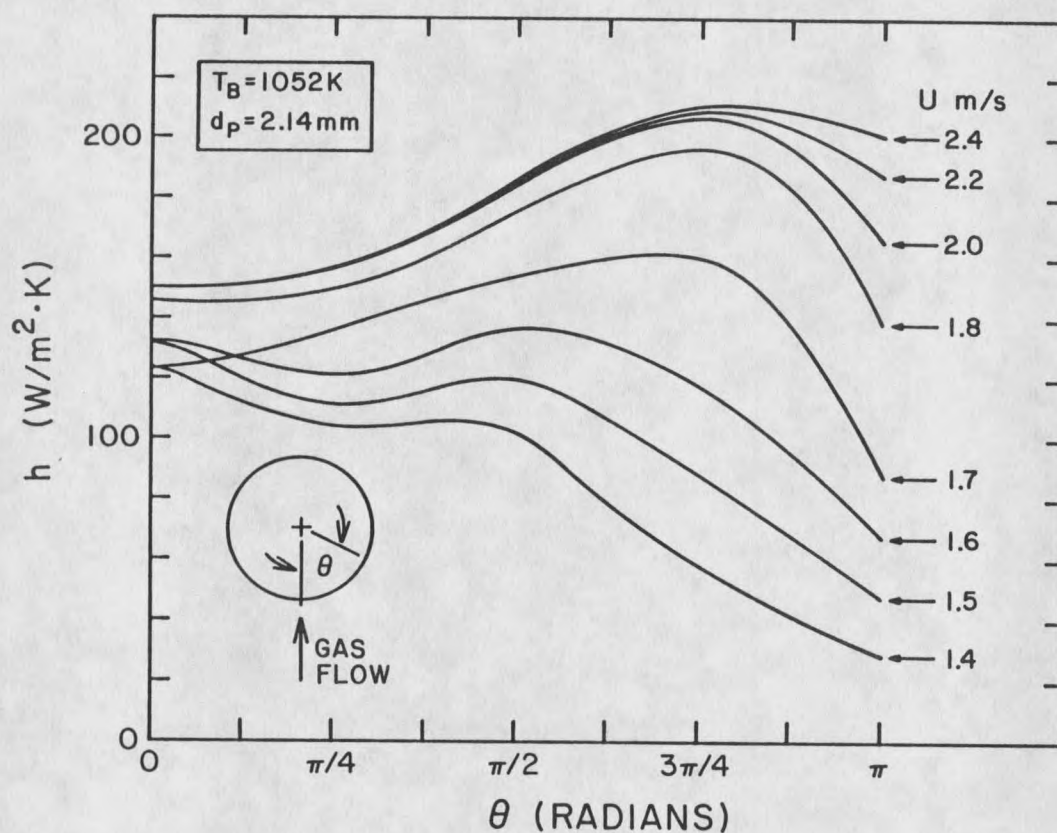


Figure 27. Time Average Local Heat Transfer Coefficient versus Angular Position for $d_p = 2.14$ mm and $T_{bed} = 1052$ K as Given by George (1984).

Spatial Average Heat Transfer Coefficient

The spatial average heat transfer coefficient was computed by applying the trapezoidal rule for numerical quadrature, given by Forsythe (1977), to the time average local values of the heat transfer coefficient.

Correlations for the maximum spatial average Nusselt Number versus the Archimedes Number have been published by several investigators. The maximum

spatial average Nusselt Number is determined from

$$(40) \quad \overline{Nu}_{\max} = \frac{dp \bar{h}_{\max}}{k_f}$$

The Archimedes Number is given by

$$(41) \quad Ar = \frac{g dp^3 (\rho_s - \rho_f)}{\rho_f \nu_f^3}$$

The correlation of the maximum spatial average Nusselt Number versus the Archimedes Number by Zabrodsky et al. (1974) is given by

$$(42) \quad \overline{Nu}_{\max} = 0.88 Ar^{0.213}$$

Other correlations valid for the Archimedes Number range considered here were developed by Baskakov et al. (1973) and Varygin et al. (1966). Almost all of the data correlated by these investigators were obtained at near room temperature with air as the fluidizing gas. These correlations are compared with results of the present study in Figure 28. The experimental data reported here are well within the range predicted by the three correlations. All gas properties have been evaluated at the corresponding bed temperature. The property data needed were obtained from Karlekar et al. (1977).

Instantaneous Local Heat Transfer Coefficient

To the author's knowledge, no instantaneous local heat transfer coefficient data for high temperature fluidized beds have been reported. Therefore, the data obtained for this thesis can only be presented and not compared with other investigators.

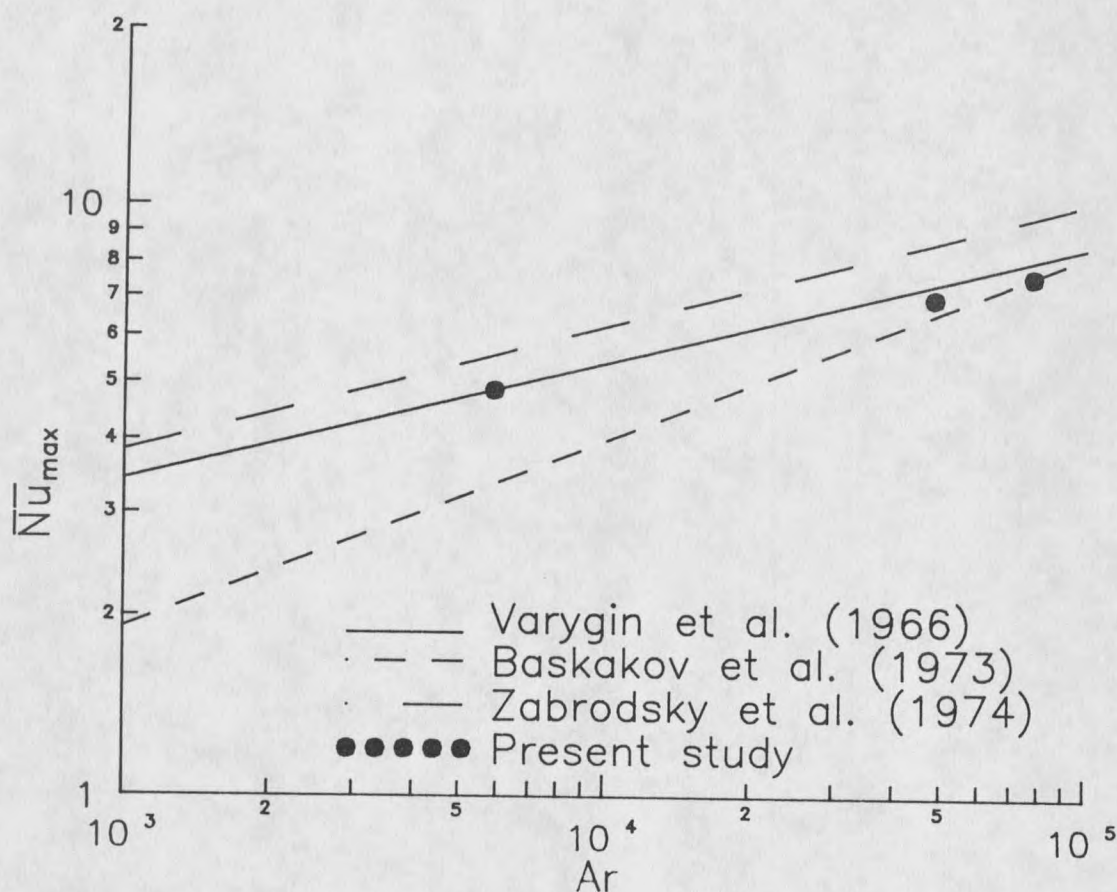


Figure 28. Comparison of Present Results with Correlations of the Maximum Spatial Average Nusselt Number versus Archimedes Number.

Typical Instantaneous Local Heat Transfer Coefficients in Graphical Form

Figures 29–43 show four seconds of typical instantaneous local heat transfer coefficient values for each of the indicated data sets considered. The electrical noise from the analog circuit was determined to be ± 37.5 mv at the output (v_4). This corresponds to approximately ± 22 W/m² K for $T_{bed} = 834.6$ K, ± 20 W/m² K for $T_{bed} = 821.3$ K, and ± 16 W/m² K for $T_{bed} = 1016.3$ K.

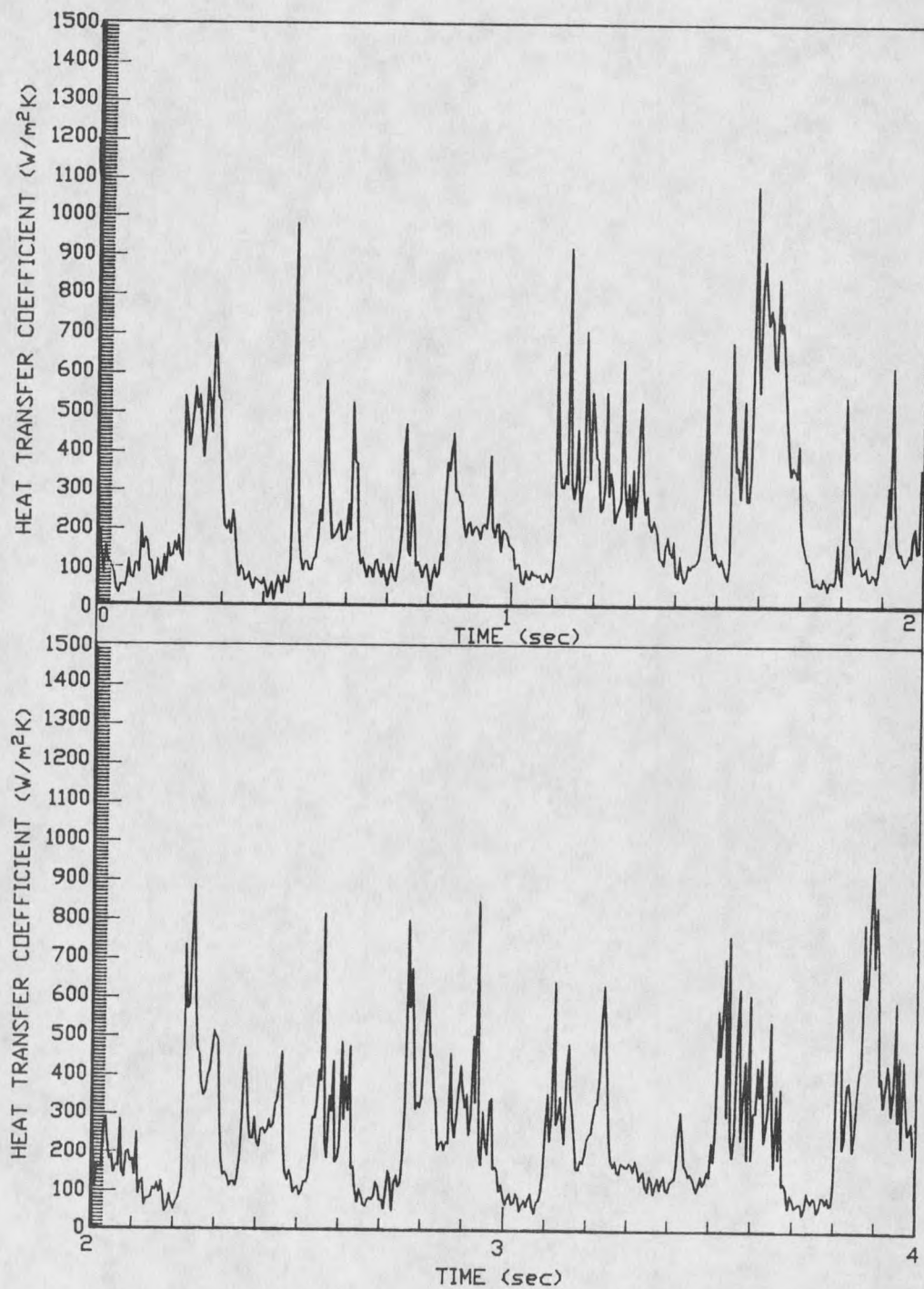


Figure 29. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 0^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

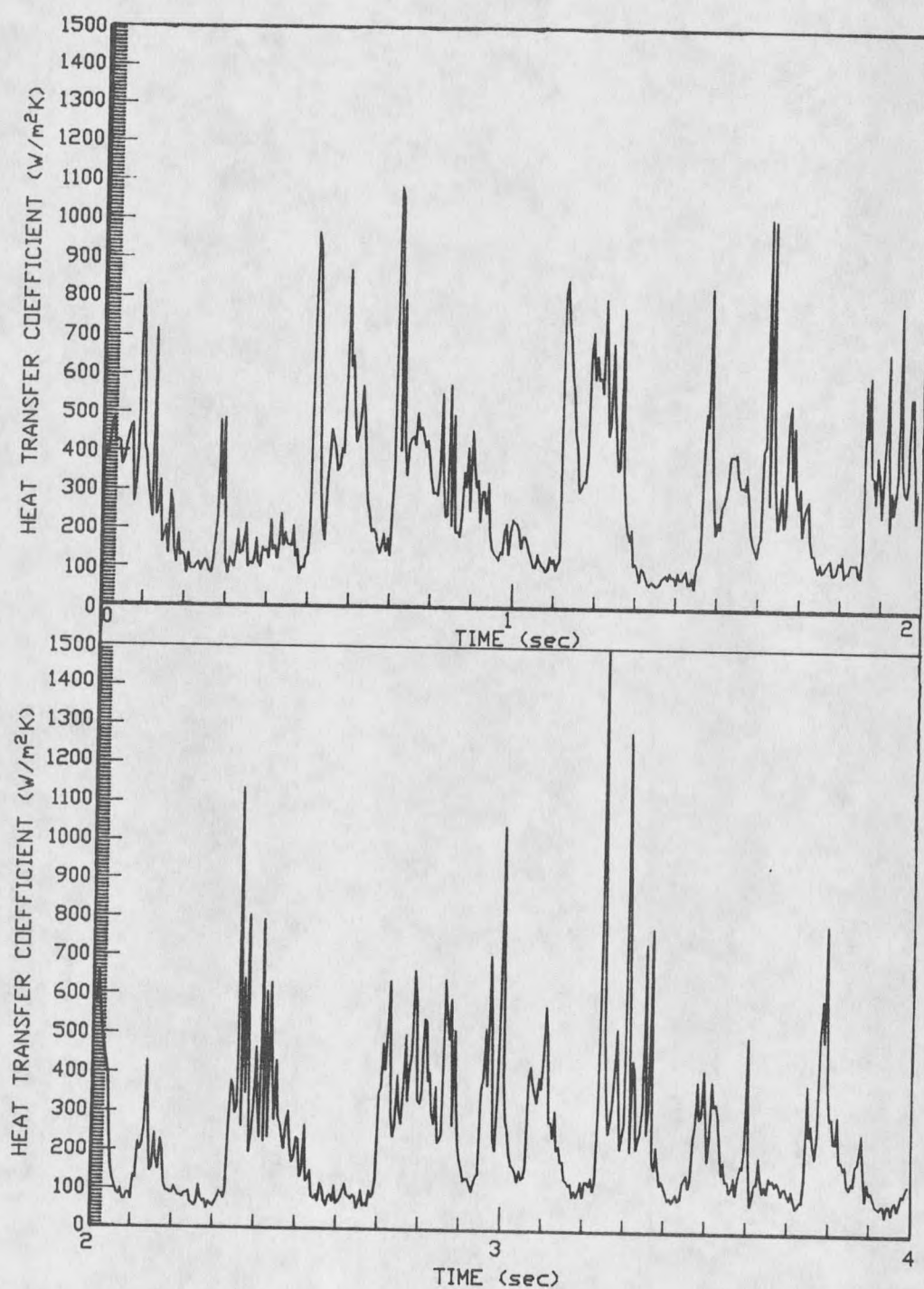


Figure 30. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 45^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

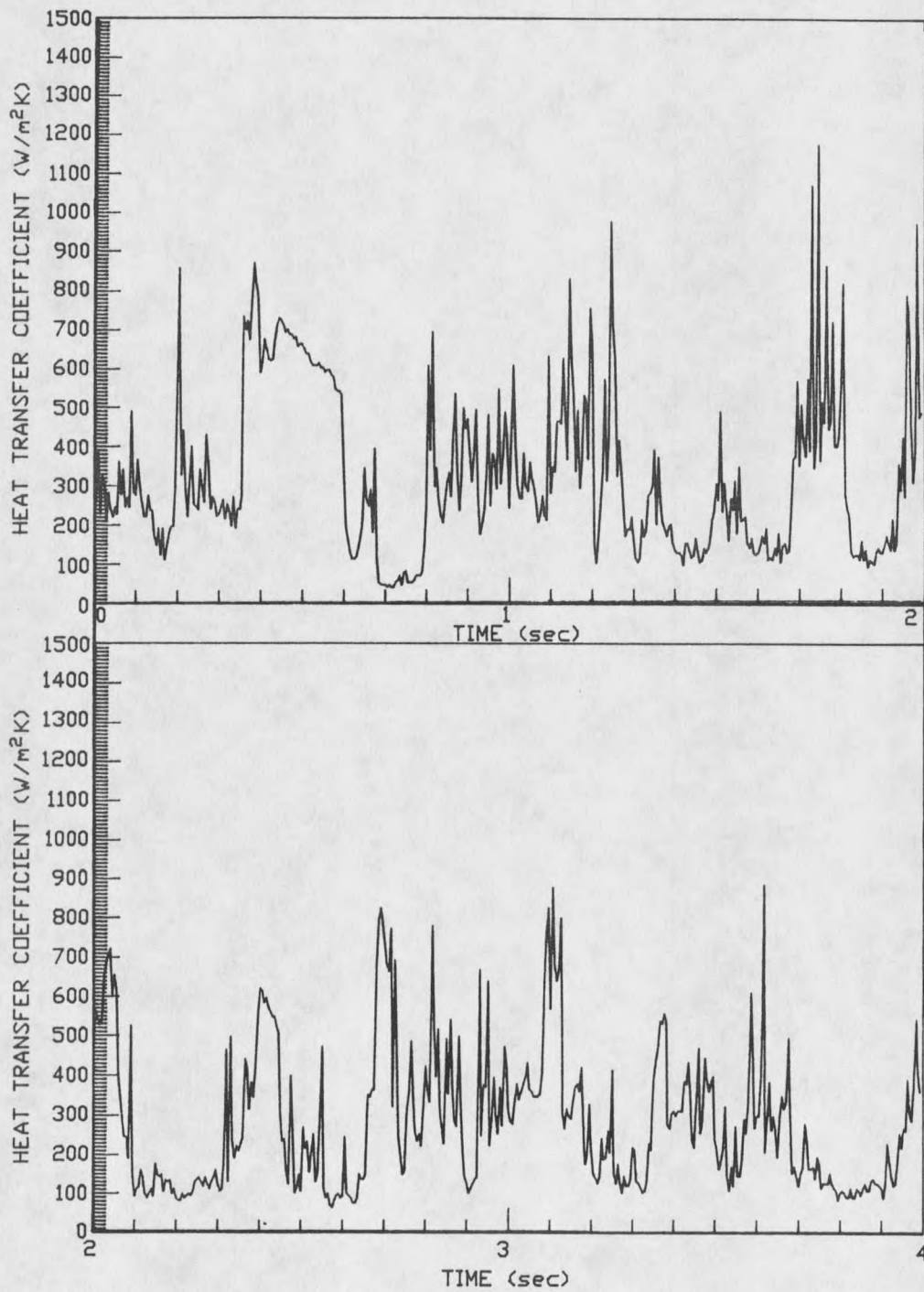


Figure 31. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 90^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

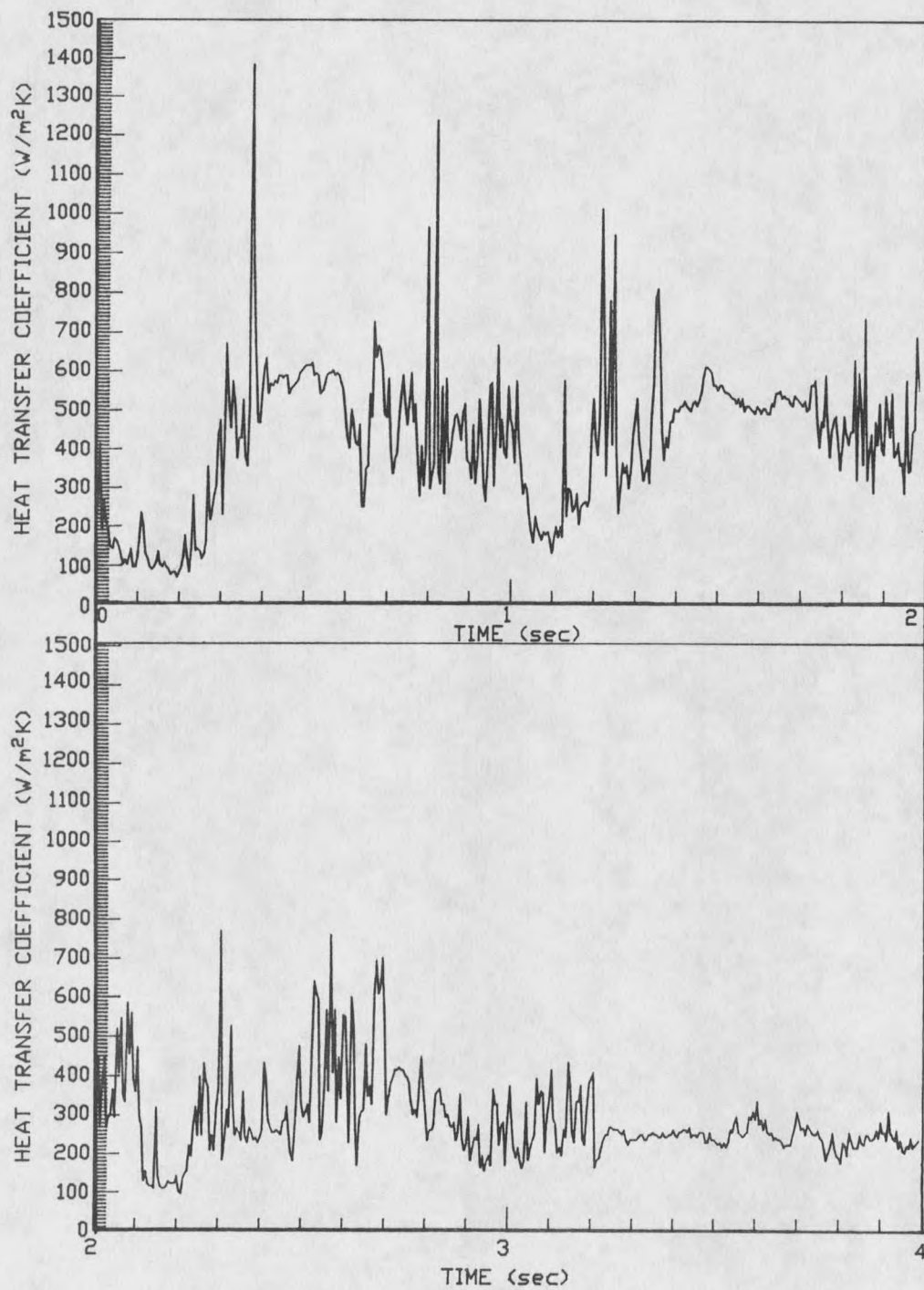


Figure 32. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 135^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

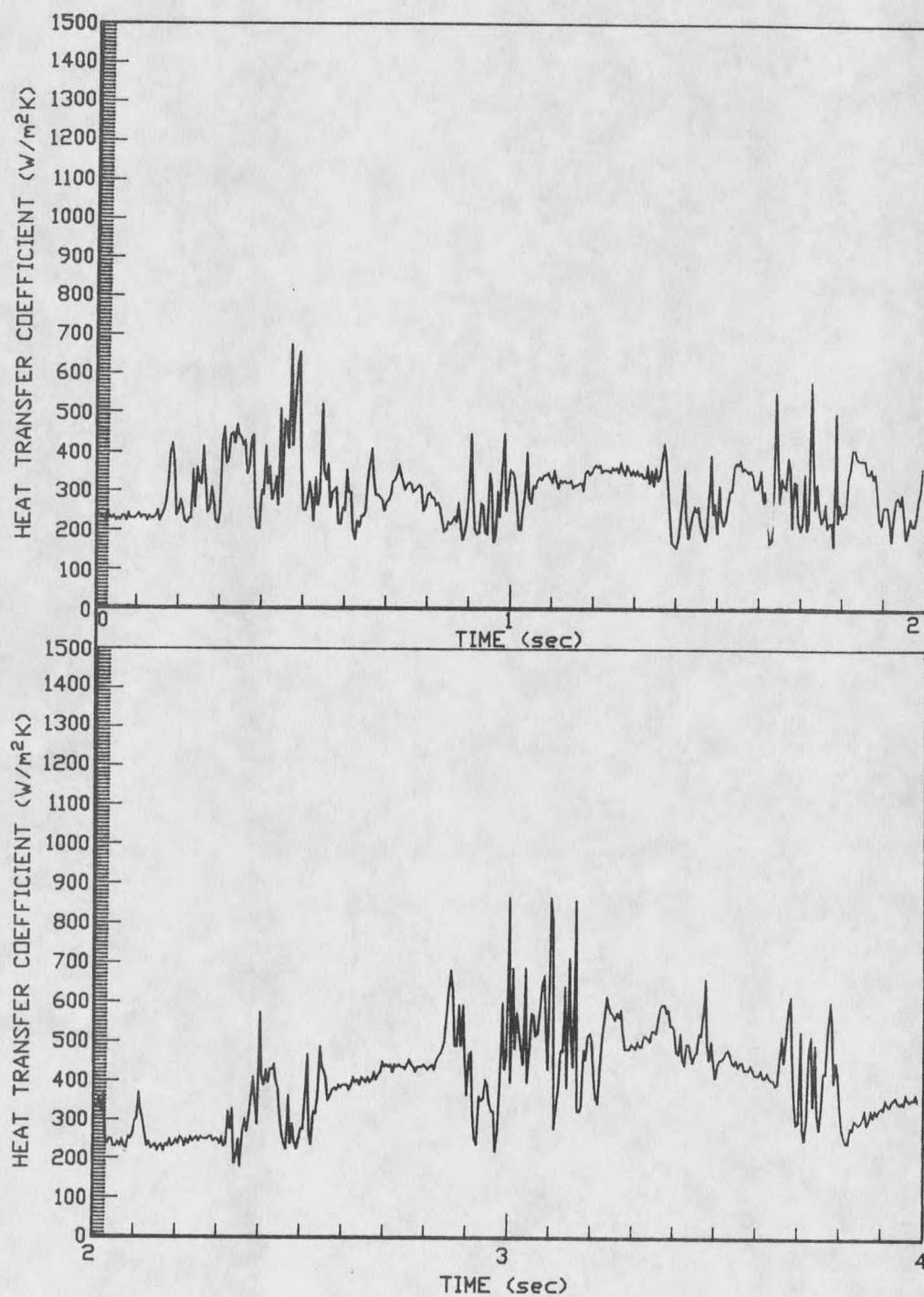


Figure 33. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 0.9$ mm, $\theta = 180^\circ$, $U_0 = 1$ m/s, and $T_{bed} = 834.6$ K.

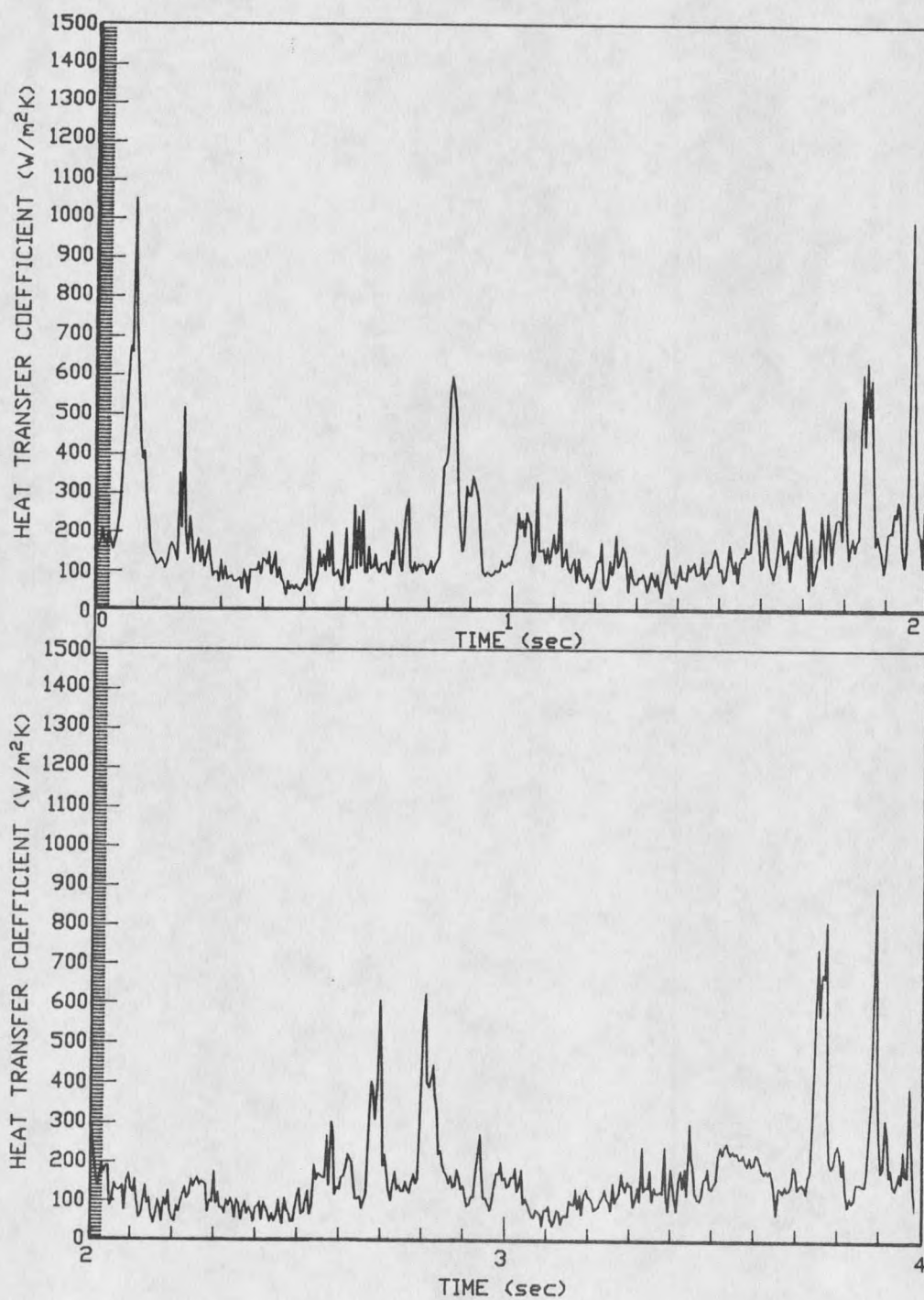


Figure 34. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 0^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K.

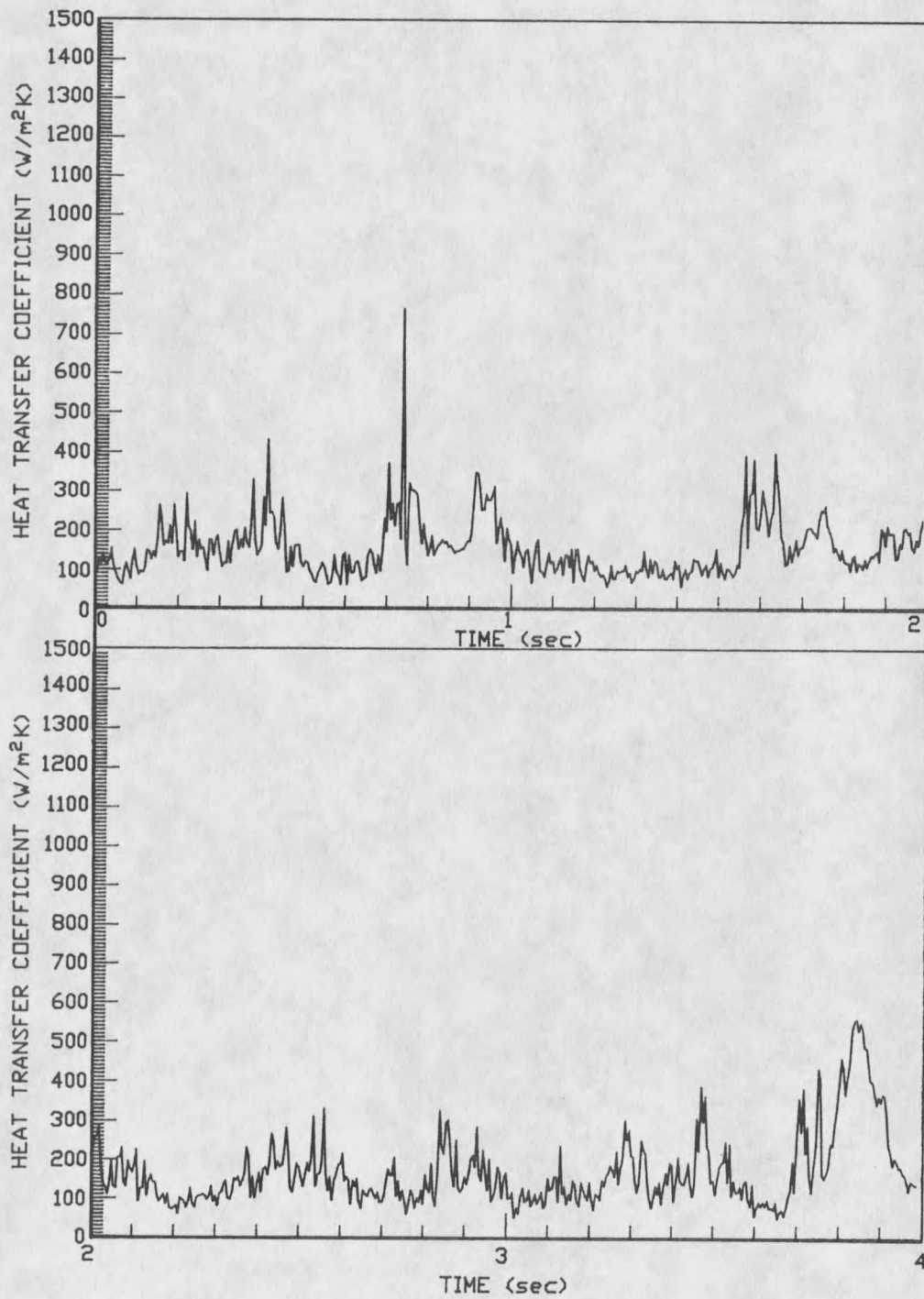


Figure 35. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 45^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K.

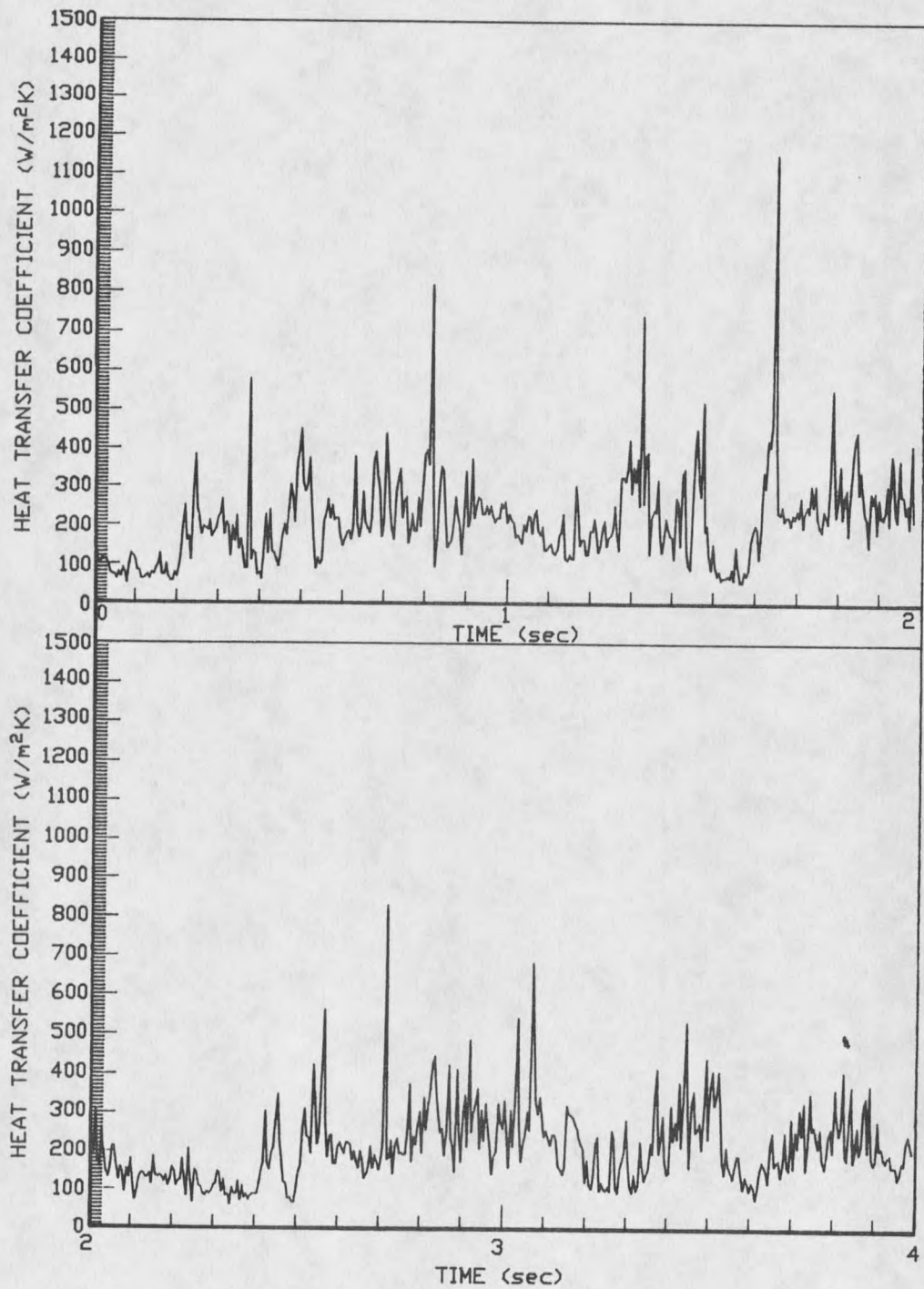


Figure 36. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 90^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K.

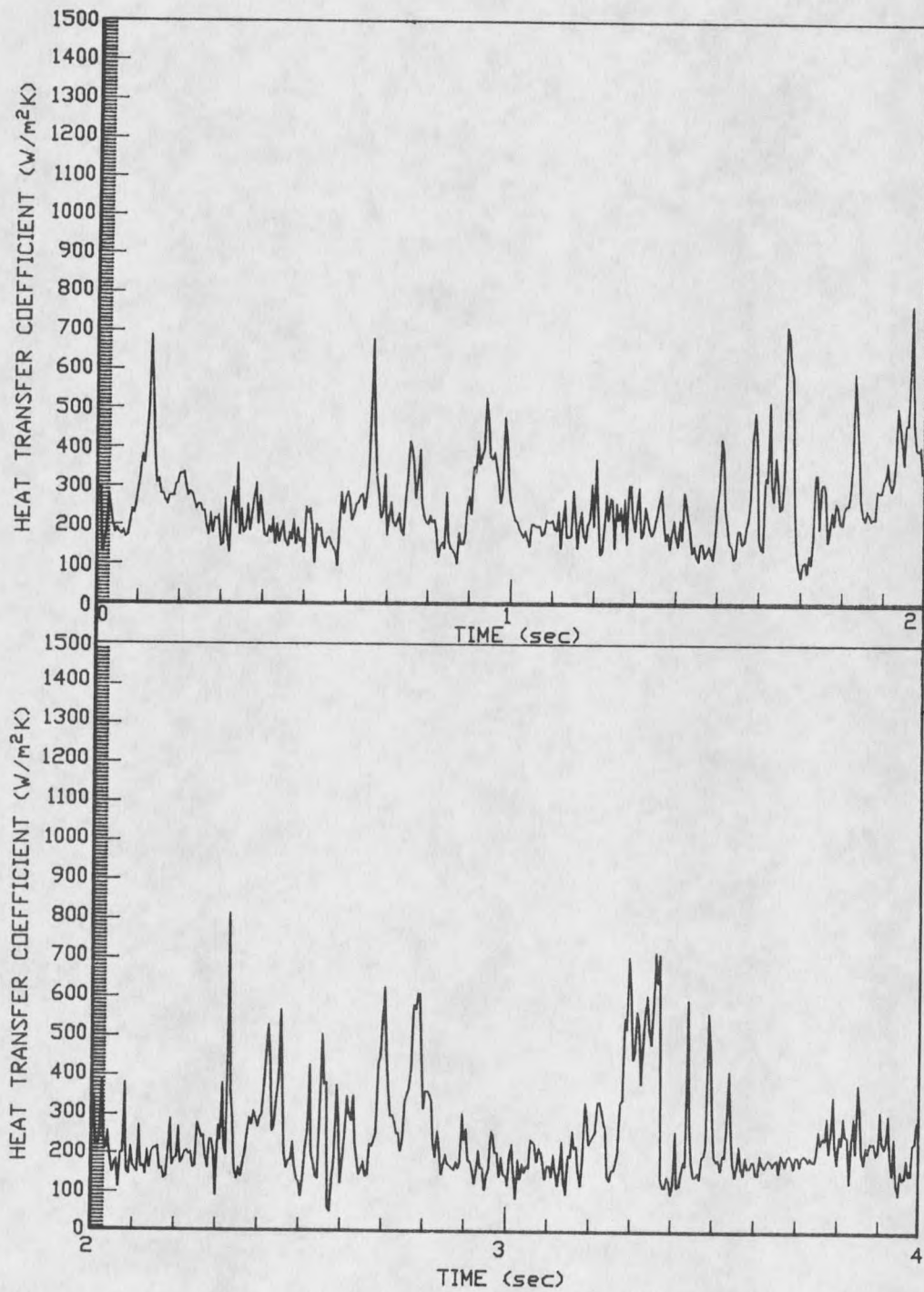


Figure 37. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 135^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K.

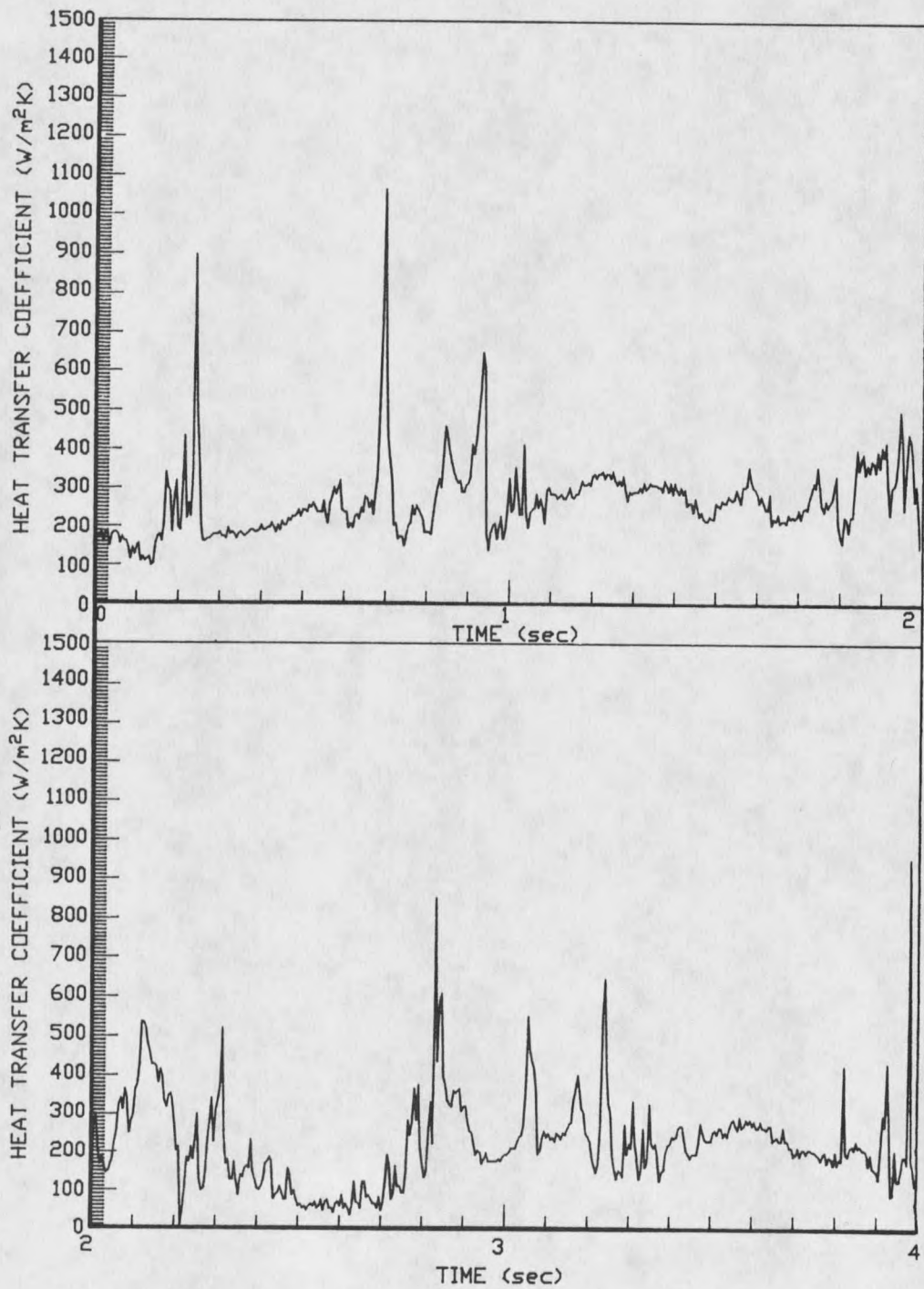


Figure 38. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 180^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 821.3$ K.

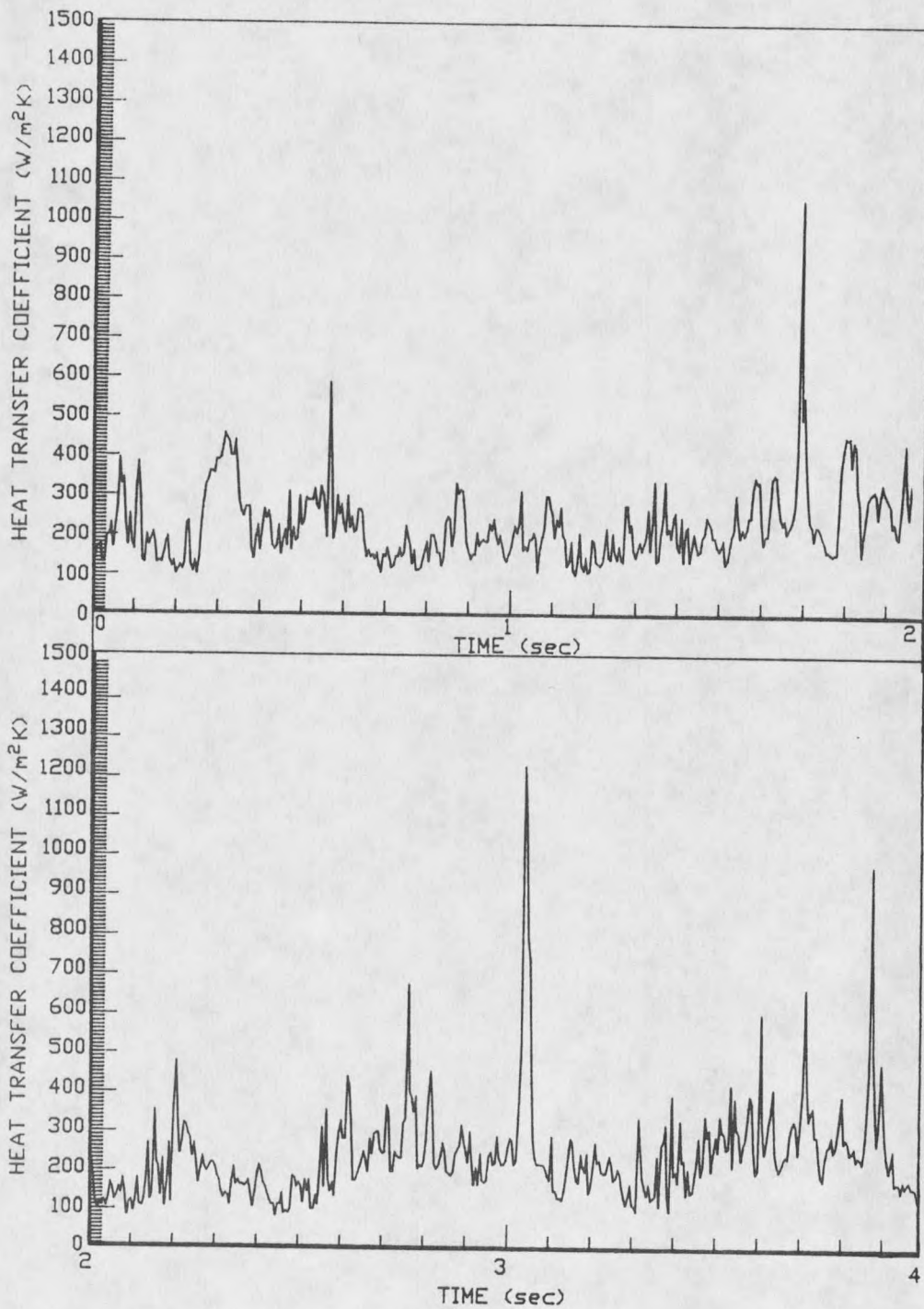


Figure 39. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 0^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K.

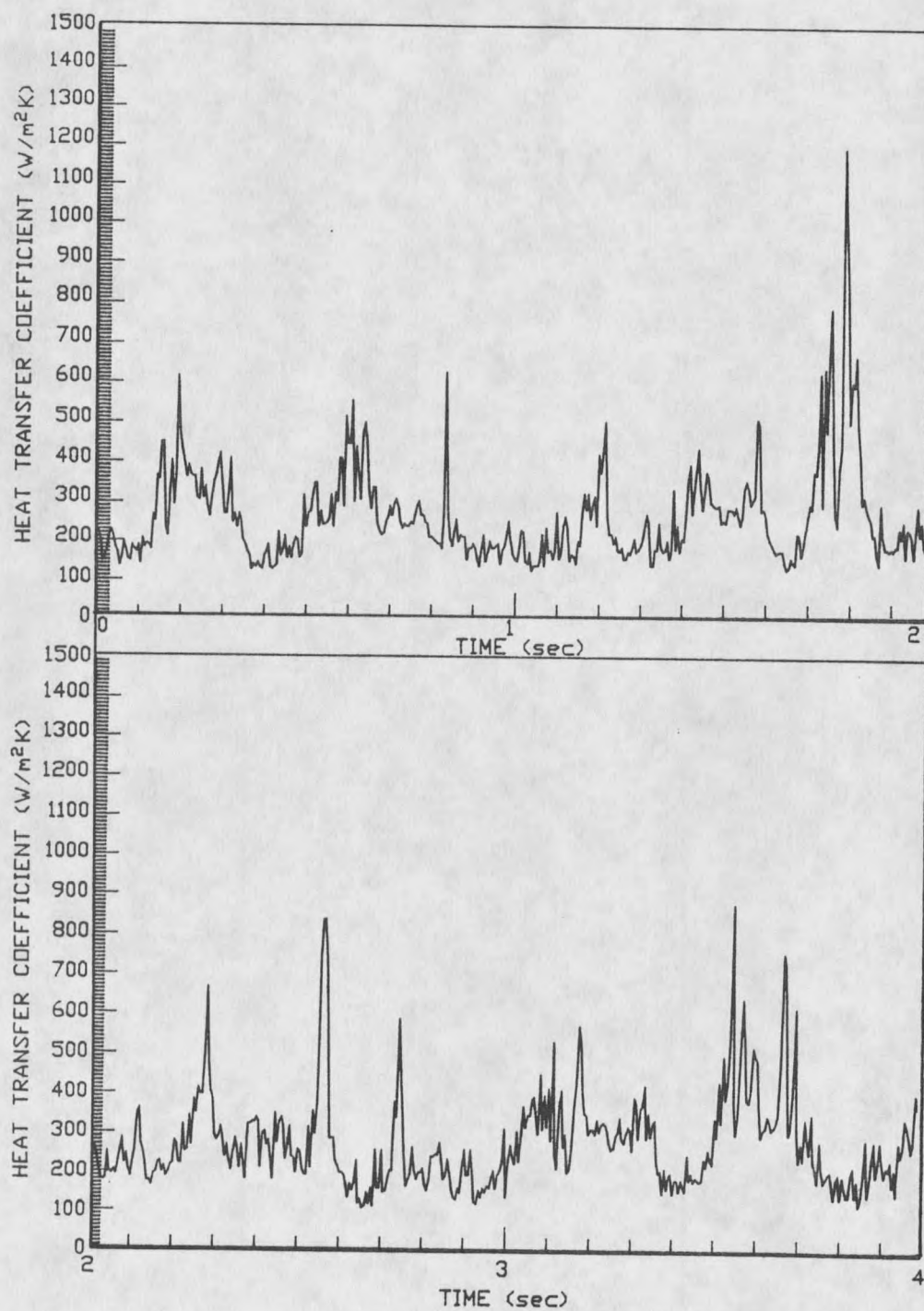


Figure 40. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 45^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K.

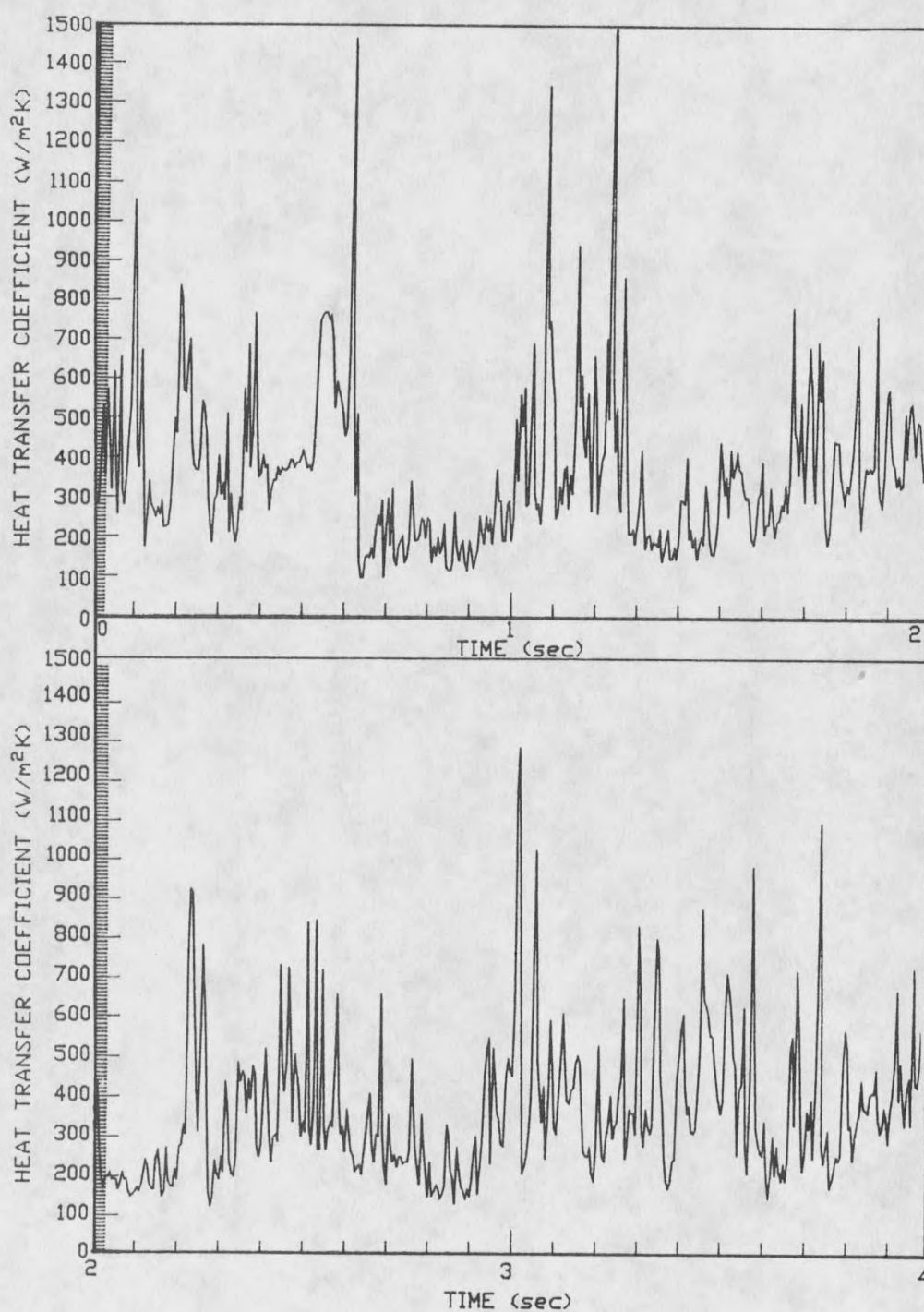


Figure 41. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 90^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K.

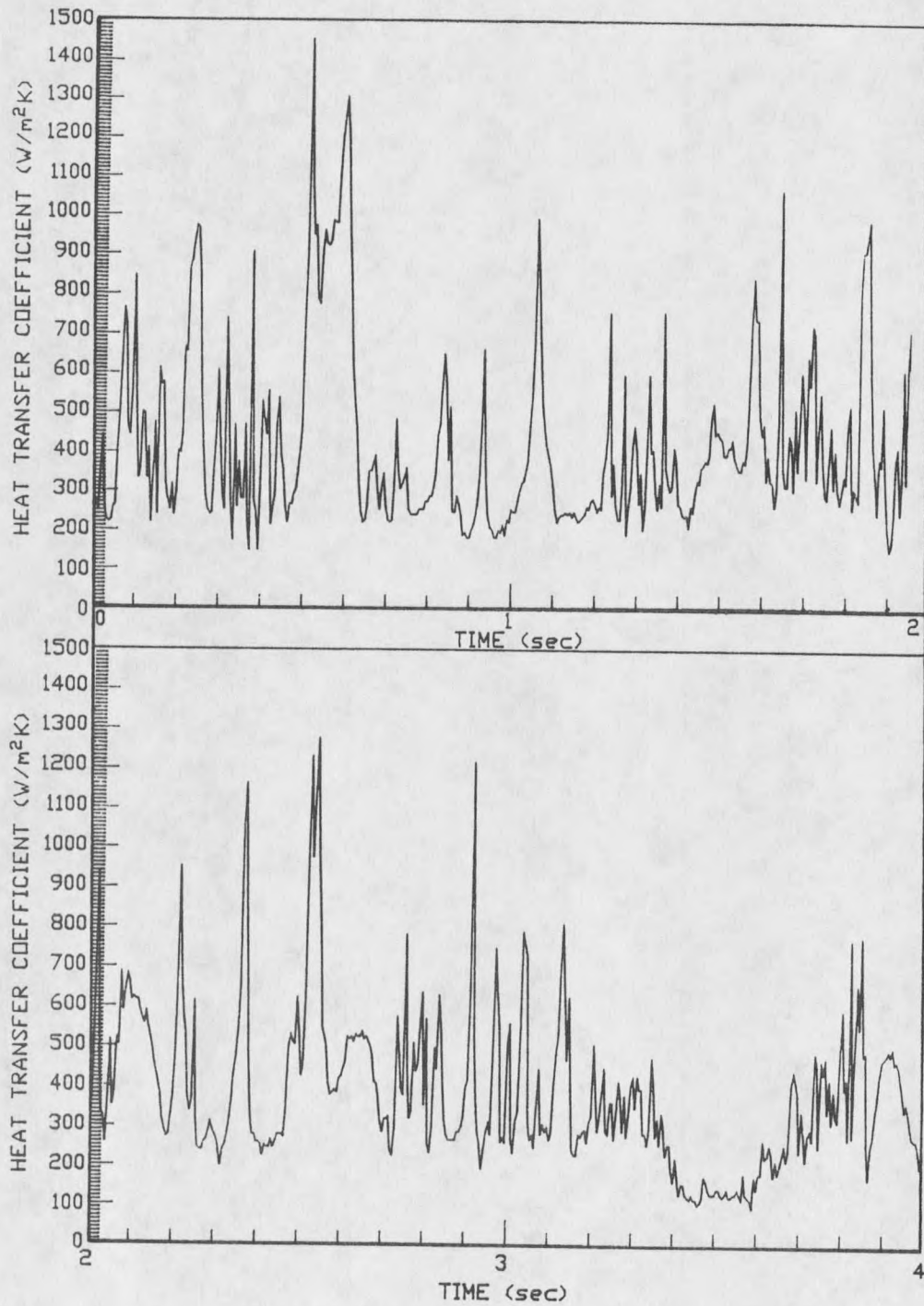


Figure 42. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 135^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K.

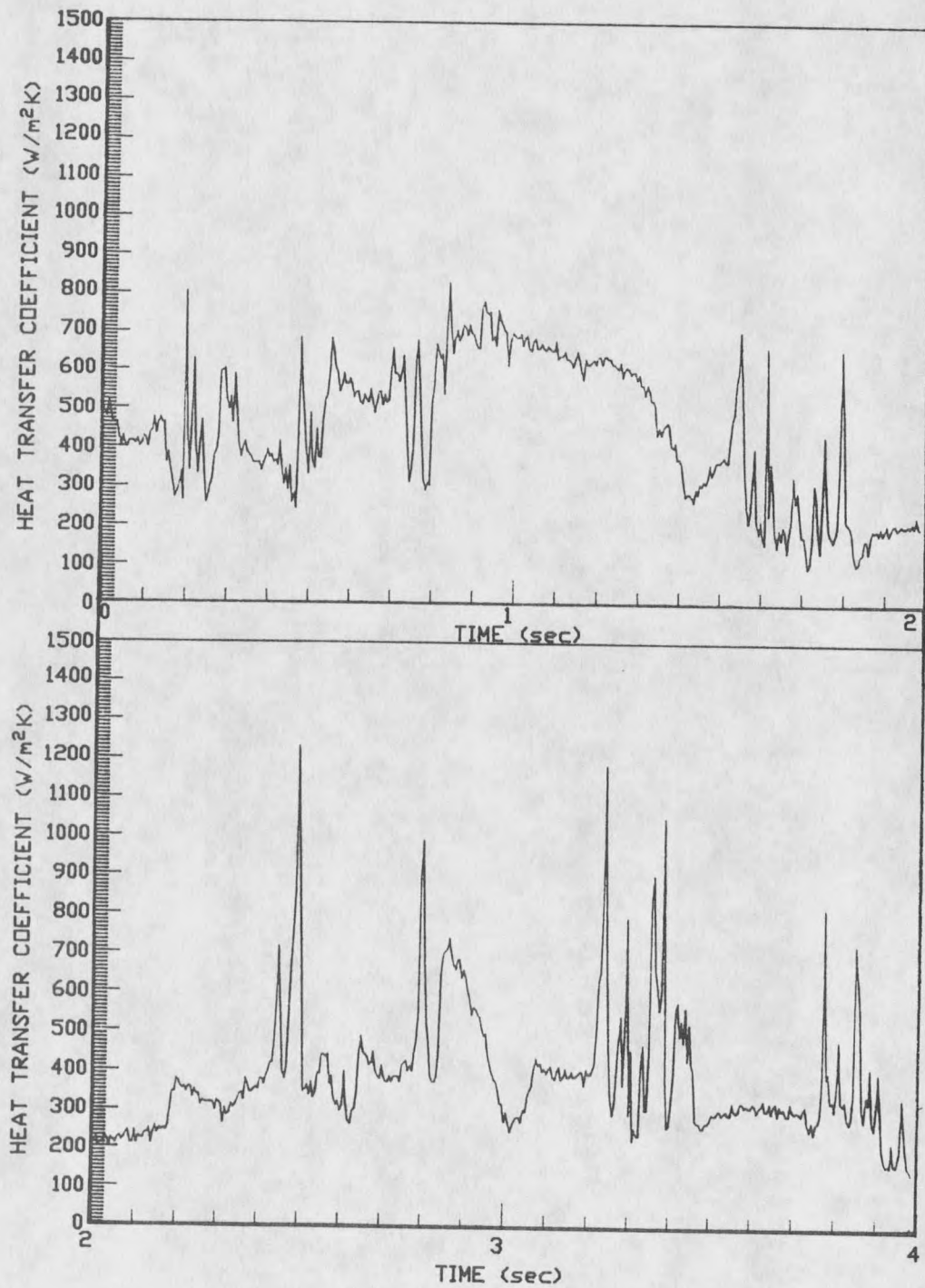


Figure 43. Instantaneous Local Heat Transfer Coefficient versus Time for $dp = 2.1$ mm, $\theta = 180^\circ$, $U_0 = 2.81$ m/s, and $T_{bed} = 1016.3$ K.

Description of Possible Modes of Heat Transfer

Visual inspection of the cylinder while in operation was not possible, however an attempt will be made to describe what may be happening in some of the figures (29–43) shown. Figure 44 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 0^\circ$.

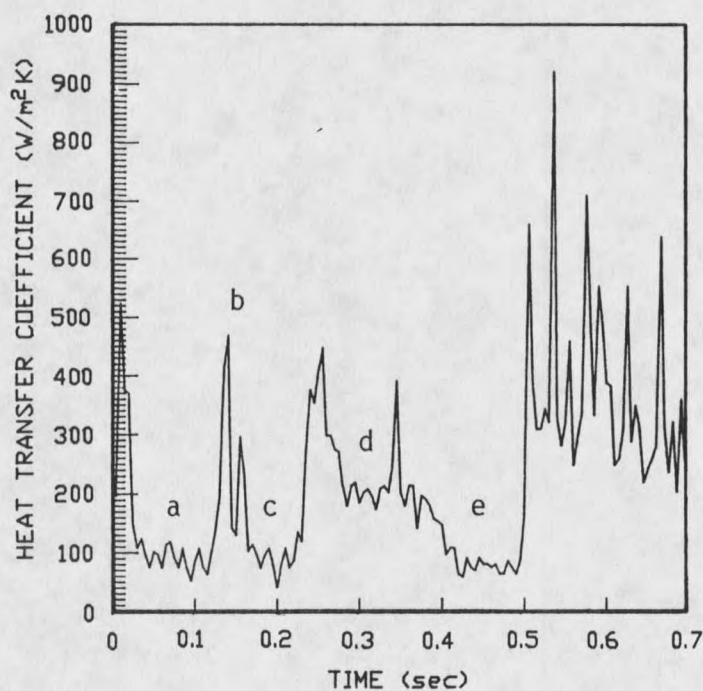


Figure 44. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 0^\circ$ and $T_{bed} = 834.6$ K.

In this figure, there is a bubble present as indicated by the low quiet region labeled (a). Then a few hot particles most likely pass through the bubble and touch the transducer face as indicated by point (b). The particles move and bubble contact is present again (c). Next the bubble moves and there is intermittent hot

gas and particle contact (d). Finally, another bubble touches the transducer, as indicated by point (e).

Figure 45 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 90^\circ$. This angular position was chosen because it has consistently the most fluctuations in the heat transfer coefficient. At points (a) and (b), there are two peaks separated by a valley. This is most likely a string of hot particles separated by hot gas in the interstitial channels between particles. The hot particle is in contact with the surface when the first reading is taken, then only gas is present, finally a hot particle is present again when the final value is taken. This figure indicates that there is intermittent particle contact on the surface of the cylinder.

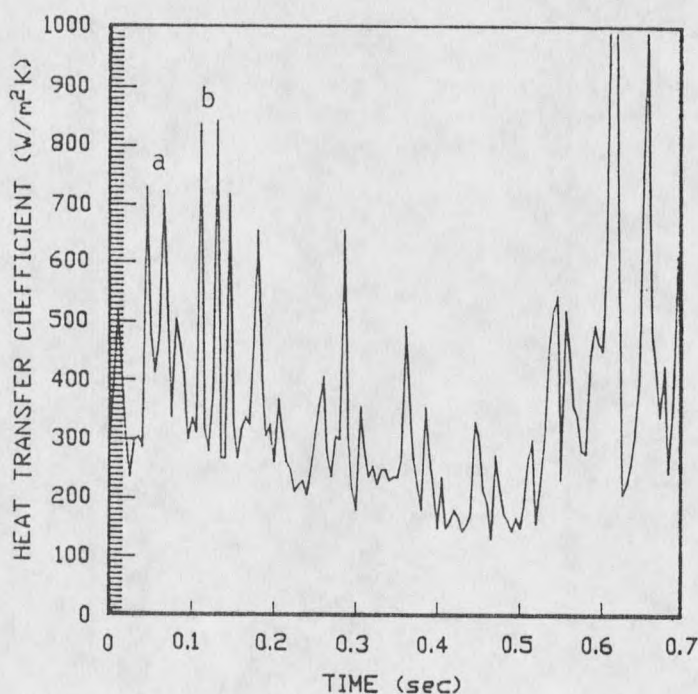


Figure 45. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 90^\circ$ and $T_{bed} = 834.6$ K.

Figure 46 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 180^\circ$. At this angular position the lee stack is present. The values shown in Figure 46 indicate that the lee stack has settled on the transducer face, and the particles that are in contact with the surface are cooling off. There is only a small amount of movement present as indicated by the small fluctuations in the heat transfer coefficient.

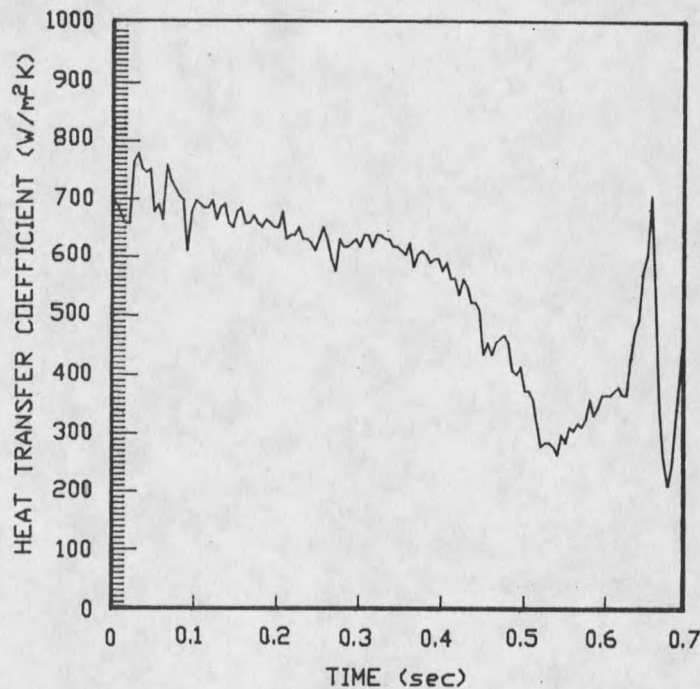


Figure 46. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 180^\circ$ and $T_{bed} = 1016.3$ K.

Figure 47 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 135^\circ$. The region indicated by (a) shows intermittent particle contact and hot gas contact. The region indicated by (b) is probably due to the lee stack covering the transducer face. This is not a bubble because the apparent

bubble phase heat transfer coefficient for this case is approximately $120 \text{ W/m}^2 \text{ K}$ as opposed to the level shown here which is approximately $250 \text{ W/m}^2 \text{ K}$.

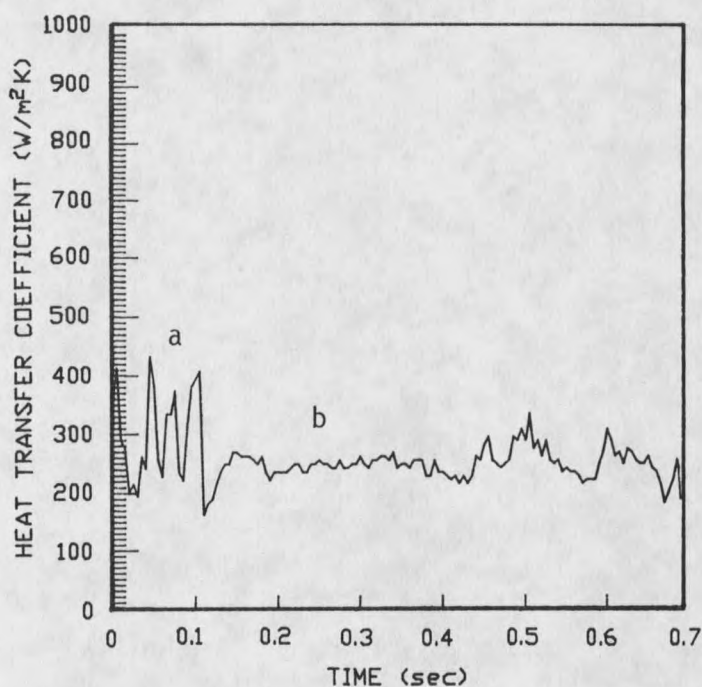


Figure 47. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 135^\circ$ and $T_{bed} = 834.6 \text{ K}$.

Figure 48 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 45^\circ$. Region (a) of this figure indicates the passage of a gas bubble over the transducer face. Note the unsteady nature of the instantaneous heat transfer coefficient during bubble contact. The fluctuations in the local heat transfer coefficient are too large to be due to electrical noise. Figure 49 shows 0.7 seconds of instantaneous local heat transfer coefficient values for $\theta = 180^\circ$. This

figure shows the approximate electrical noise level indicated at (a). The fluctuations shown in Figure 48 region (a) are much larger than these and are probably due to turbulence in the bubble.

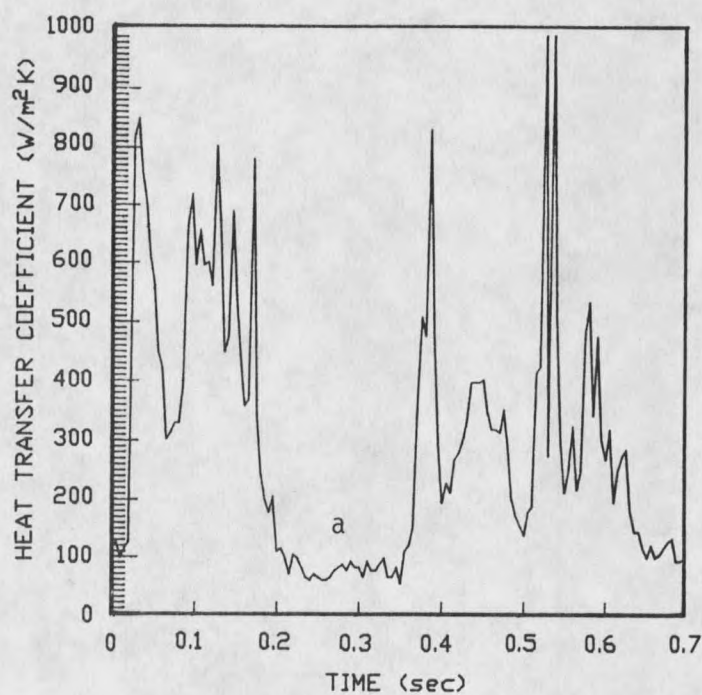


Figure 48. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 45^\circ$ and $T_{bed} = 834.6$ K.

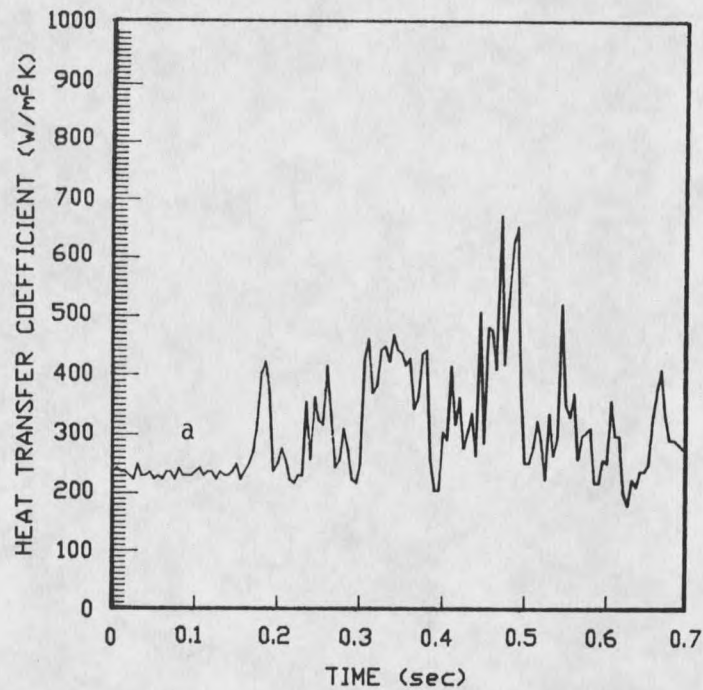


Figure 49. 0.7 Seconds of Instantaneous Local Heat Transfer Coefficient Values at $\theta = 180^\circ$ and $T_{bed} = 834.6$ K.

Relative Magnitudes of the Instantaneous Heat Transfer Coefficient

The values for the instantaneous local heat transfer coefficients are tabulated in percentage of readings above and below the time average local heat transfer coefficient. Tables 8–10 shows these values for each of the test runs considered. A total of 19978 data points were considered for each angular position.

Angle (θ°)	% below < h > (%)	% above < h > (%)	% above < h > (%)						
			1.25 h	1.5 h	2.0 h	2.5 h	3.0 h	3.5 h	4.0 h
0	61.7	38.3	26.6	18.7	9.5	4.5	2.1	1.0	0.5
45	59.8	40.2	27.7	18.7	8.3	3.7	1.7	0.9	0.5
90	54.5	45.5	28.5	16.3	5.6	1.8	0.6	0.2	0.1
135	54.7	45.3	23.9	12.1	3.0	0.7	0.2	0.06	0.01
180	58.8	41.2	20.3	8.8	2.0	0.4	0.09	0.03	0.01

Table 8. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s.

Angle (θ°)	% below < h > (%)	% above < h > (%)	% above < h > (%)						
			1.25 h	1.5 h	2.0 h	2.5 h	3.0 h	3.5 h	4.0 h
0	60.7	39.3	22.1	12.2	4.8	2.4	1.3	0.8	0.5
45	63.6	36.4	21.3	12.5	5.0	2.2	1.0	0.5	0.4
90	58.9	41.1	23.3	12.9	4.0	1.6	0.8	0.4	0.2
135	61.6	38.4	21.5	11.8	4.2	2.1	1.2	0.7	0.05
180	57.6	42.4	23.3	10.5	2.3	0.7	0.2	0.1	0.04

Table 9. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Angle (θ°)	% below < h > (%)	% above < h > (%)	% above < h > (%)						
			1.25 h	1.5 h	2.0 h	2.5 h	3.0 h	3.5 h	4.0 h
0	63.2	36.8	18.4	9.9	4.5	1.9	1.2	0.7	0.5
45	63.8	36.2	19.3	10.9	4.3	2.0	1.2	0.6	0.4
90	58.7	41.3	22.7	12.2	4.1	1.7	0.7	0.4	0.2
135	59.2	40.8	20.4	11.6	3.6	1.0	0.4	0.1	0.04
180	56.5	43.5	20.4	8.8	1.6	0.3	0.06	0.02	0.00

Table 10. Percentage of Readings Above and Below the Time Average Heat Transfer Coefficient for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Several observations can be made from these tables. The most obvious is that the amount of fluctuations above four times the average value decreases almost to zero at $\theta = 180^\circ$ for all the test runs considered. This is most likely due to the presence of the lee stack. Another observation is that a larger percentage of the values at $\theta = 0^\circ$ and $\theta = 45^\circ$ are below the average local heat transfer coefficient.

Local Bubble Phase Heat Transfer Coefficient

The apparent local bubble phase heat transfer coefficient for each of the test run conditions is shown in Tables 11–13. The value at $\theta = 180^\circ$ is not shown because the local bubble phase heat transfer coefficient is not clearly evident from any of the results. From the results shown in Tables 11–13 it is obvious that the local bubble phase heat transfer coefficient generally increases as the transducer is rotated from $\theta = 0^\circ$ to $\theta = 135^\circ$. Furthermore, the percentage of values of the instantaneous local heat transfer coefficient that are less than halfway between

the average value and the apparent bubble phase value generally decreases as the cylinder is rotated from $\theta = 0^\circ$ to $\theta = 135^\circ$. This indicates that bubbles are present for a greater percentage of the time at $\theta = 0^\circ$ and $\theta = 45^\circ$ than at other angular positions.

Angular position (degrees)	Bubble phase heat transfer coefficient (W/m ² K)	$\% \text{ below}$ $\langle h \rangle - (\langle h \rangle - h_b)/2$ (%)
0	70	41.3
45	90	42.0
90	110	30.0
135	120	21.1

Table 11. Apparent Local Bubble Phase Heat Transfer Coefficient Values
for $T_{bed} = 834.6$ K, $dp = 0.9$ mm, and $U_0 = 1$ m/s.

Angular position (degrees)	Bubble phase heat transfer coefficient (W/m ² K)	$\langle h \rangle - (\langle h \rangle - h_b)/2$ % below (%)
0	70	37.9
45	90	43.2
90	90	31.6
135	140	28.7

Table 12. Apparent Local Bubble Phase Heat Transfer Coefficient Values
for $T_{bed} = 821.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Angular position (degrees)	Bubble phase heat transfer coefficient (W/m ² K)	$\langle h \rangle - (\langle h \rangle - h_b)/2$ % below (%)
0	130	17.8
45	150	18.9
90	180	17.2
135	200	11.8

Table 13. Apparent Local Bubble Phase Heat Transfer Coefficient Values
for $T_{bed} = 1016.3$ K, $dp = 2.1$ mm, and $U_0 = 2.81$ m/s.

Defective Data for Local Heat Transfer Coefficients

There were a total of 69 negative values for the measured instantaneous local heat transfer coefficient out of a total of 299670 values. This represents approximately 0.02 percent of the values. From those 69 values, 37 can be explained by the electrical noise level which was approximately $\pm 22 \text{ W/m}^2 \text{ K}$ for $T_{\text{bed}} = 834.6 \text{ K}$, $\pm 20 \text{ W/m}^2 \text{ K}$ for $T_{\text{bed}} = 821.3 \text{ K}$, and $\pm 16 \text{ W/m}^2 \text{ K}$ for $T_{\text{bed}} = 1016.3 \text{ K}$. This leaves 32 values, or 0.01 percent of the total readings which cannot be due to the 75 mv peak-to-peak noise at the output of the analog circuit (v_4).

The analog circuit is very sensitive to fluctuations in voltage at the input. One possible explanation for the presence of these negative values is from voltage surges caused by the fluidized bed control equipment. The controller that monitors the bed temperature sends a voltage pulse to the mechanism which controls the gas flow rate when the bed temperature exceeds the indicated value. This voltage pulse could, through ground interaction, reach the analog circuit and cause the very inaccurate instantaneous readings to be taken for a short period of time (a few milliseconds).

In any case, the very few defective data points that were obtained during this preliminary study are almost certainly due to electrical equipment other than the transducer, signal conditioning circuit, and data acquisition system discussed in this thesis.

CHAPTER 8

CONCLUSIONS

The results of the experiments performed during this research demonstrate that the heat flux transducer was abrasion resistant, capable of operation at high temperatures, and had sufficiently rapid response (settling time of approximately 2 ms) that effectively instantaneous local values of the heat flux could be measured between the hot bed and immersed cylinder. Since, to the author's knowledge, there are no reported data for similar conditions of the instantaneous local heat transfer coefficients, the accuracy of the heat flux transducer cannot be determined. However, the values obtained for the spatial average heat transfer coefficient are within the range predicted by published correlations. Furthermore, the data indicates that the heat flux transducer responds to the presence of bubbles contacting the surface of the cylinder and to the notion of the lee stack at $\theta = 180^\circ$. These three facts lend evidence that the transducer is responding accurately to instantaneous changes in the heat flux to the cylinder surface.

The calibration scheme for the heat flux transducer must have provided accurate results for the calibration constant (β) and k/L . If the value of k/L was not determined accurately, the values obtained for the time average local heat transfer coefficient would not be in accordance with results obtained by previous investigators; that is, the value would either be too high or too low depending on whether the value of k/L was either high or low. If the value for the calibration constant (β) was too small, the indicated bubble phase heat transfer coefficients

would be negative, and if β was too large the value for the bubble phase heat transfer coefficient would be almost equal to the time average value of the heat transfer coefficient. Since this is not the case, β must be very close to the actual value.

Recommendations

There are a few recommendations that would improve the current system. These are:

1. Construct the heat flux transducer with only one surface thermocouple with a junction width of at least 0.6 cm. The reason for two surface thermocouples was to reduce the signal to noise ratio and to yield a more accurate average surface temperature. The signal to noise ratio is not a problem with the above transducer, and the junction width is currently almost the radius of the transducer surface. If only one surface thermocouple were used, the complexity of the transducer would be decreased and the time required for construction would be greatly reduced.
2. Attach a commercial heat flux transducer (RdF Micro-Foil) to the surface of the cylinder in the same proximity as the experimental heat flux transducer for use in calibrating k/L . This would insure that approximately the same bed conditions prevailed for each transducer. It would be necessary to cover the commercial transducer with a piece of 0.127 mm thick ANSI 304 stainless steel shim stock to protect the commercial transducer from the harsh environment of the fluidized bed.

3. Use a different data acquisition scheme to obtain the data. Currently, it takes about 20 minutes to take one data set, with most of the time devoted to storing the data on disk. If a more powerful computer were used for the data acquisition system, this time could be at least cut in half. This would allow time for a larger quantity of data to be taken. Furthermore, the current system is not compatible with IBM compatible systems, therefore data reduction is very time-consuming. For instance, it currently takes about one hour to convert the voltage readings from one data set into instantaneous local heat transfer coefficients.
4. Use a shorter sampling interval. The 5 ms sampling interval used here was approximately 2.5 times the settling time of the transducer. A 2 ms sampling interval would be more consistent with the response characteristics of the transducer and signal conditioning circuit.
5. Use two experimental heat flux transducers offset by 90 degrees. This would further reduce the time required to obtain data because two data sets could be taken before the cylinder needed to be rotated.

APPENDICES

APPENDIX A
COMPUTER PROGRAMS

DATA ACQUISITION PROGRAM

This is a listing of the computer program NSFHT. This program directs the data Hewlett Packard data acquisition system to take and store the voltages necessary in order to compute the time average local heat transfer coefficient and the instantaneous local heat transfer coefficient.

```
100 ! DATA ACQUISITION PROGRAM
110 ! FOR HEAT FLUX MEASUREMENT
120 ! IN HIGH TEMPERATURE
130 ! FLUIDIZED BEDS
140 ! BY A. H. GEORGE AND J. SMALLEY
150 ! MONTANA STATE UNIVERSITY
160 ! AUGUST 1989
170 !
180 ! NSF PROJECT CBT-8801618
190 !
200 !
210 OPTION BASE 1
220 ! DIMENSION THE VARIABLES ARRAYS
230 DIM V1$(40),V2$(30),F$(9997)
240 DIM V5$(40)
250 ! DECLARE THE DISK DRIVE FOR STORAGE
260 MASS STORAGE IS ":D701"
270 ! CREATE THE DATA FILE FOR STORAGE
280 ! OF THE VOLTAGE VALUES
290 CREATE "VDATA",24,10000
300 ! DECLARE THE PRINTER TO BE USED
310 PRINTER IS 701
320 !
330 ! INITIALIZE TIME AVERAGE VARIABLES
340 V1=0 @ V2=0
```

Figure 50. Data Acquisition Program.

```
350 V3=0 @ V4=0
360 ! CREATE THE INSTRUCTION LIST FOR
370 ! THE HP3456A VOLTMETER
380 V1$ = "H4000STN.01STIM2"
390 V1$ = V1$& "D0Z0FLOS0F1R4T3"
400 DISP " INPUT RUN NUMBER
410 INPUT N1
420 ! *****
430 ! CHANNEL 1 IS THE FIRST SURFACE
440 ! THERMOCOUPLE
450 ! CHANNEL 2 IS THE SECOND SURFACE
460 ! THERMOCOUPLE
470 ! CHANNEL 3 IS THE IN-WALL
480 ! THERMOCOUPLE
490 ! CHANNEL 4 IS THE OUTPUT VOLTAGE
500 ! THAT REPRESENTS THE HEAT FLUX
510 ! CHANNEL 6 IS THE REFERENCE
520 ! THERMOCOUPLE ON THE TERMINAL
530 ! STRIP
540 ! CHANNEL 19 IS THE INTERNAL
550 ! TEMPERATURE DETECTOR
560 ! *****
570 ! RECORD A READING OF CHANNEL 6
580 OUTPUT 709 ; "AC6"
590 WAIT 5000
600 ENTER 722 ; R3
610 OUTPUT 709 ; "AR"
620 ! RECORD THE INTERNAL TEMPERATURE
630 OUTPUT 709 ; "AC19"
640 WAIT 5000
650 ENTER 722 ; A3
660 OUTPUT 709 ; "AR"
670 ! COMPUTE AND RECORD THE TIME AVERAGE
680 ! VALUE OF CHANNEL 3
690 OUTPUT 709 ; "AC3"
700 WAIT 3000
710 DISP "CH 3 CLOSED"
720 OUTPUT 722 ; V1$
```

Figure 50 (continued). Data Acquisition Program.

```
730 WAIT 125000
740 OUTPUT 722 ; "REM"
750 ENTER 722 ; V3
760 OUTPUT 709 ; "AR"
770 REM
780 ! COMPUTE AND RECORD THE TIME AVERAGE
790 ! VALUE OF CHANNEL 4
800 OUTPUT 709 ; "AC4"
810 WAIT 3000
820 DISP "CH. 4 CLOSED"
830 OUTPUT 722 ; V1$
840 WAIT 125000
850 OUTPUT 722 ; "REM"
860 ENTER 722 ; V4
870 OUTPUT 709 ; "AR"
880 REM *****
890 ! INITIATE TAKING OF FIRST
900 ! 9996 VOLTAGE READINGS
910 ! BY HP3437A VOLTMETER
920 V2$ = "E0SR3F1D.005SN9996ST1"
930 DISP "START HP3437A"
940 OUTPUT 724 ; V2$
950 OUTPUT 722 ; "H"
960 WAIT 3000
970 REM *****
980 ! RECORD A READING OF CHANNEL 6
990 OUTPUT 709 ; "AC6"
1000 WAIT 5000
1010 ENTER 722 ; R1
1020 OUTPUT 709 ; "AR"
1030 ! RECORD THE INTERNAL TEMPERATURE
1040 OUTPUT 709 ; "AC19"
1050 WAIT 5000
1060 ENTER 722 ; A1
1070 OUTPUT 709 ; "AR"
1080 ! COMPUTE AND RECORD THE TIME AVERAGE
1090 ! VALUE OF CHANNEL 1
1100 OUTPUT 709 ; "AC1"
```

Figure 50 (continued). Data Acquisition Program.

```
1110 WAIT 3000
1120 DISP "CH 3 CLOSED"
1130 OUTPUT 722 ; V1$
1140 WAIT 125000
1150 OUTPUT 722 ; "REM"
1160 ENTER 722 ; V1
1170 OUTPUT 709 ; "AR"
1180 ! RECORD A READING OF CHANNEL 6
1190 OUTPUT 709 ; "AC6"
1200 WAIT 5000
1210 ENTER 722 ; R2
1220 OUTPUT 709 ; "AR"
1230 ! RECORD THE INTERNAL TEMPERATURE
1240 OUTPUT 709 ; "AC19"
1250 WAIT 5000
1260 ENTER 722 ; A2
1270 OUTPUT 709 ; "AR"
1280 ! COMPUTE AND RECORD THE TIME AVERAGE
1290 ! VALUE OF CHANNEL 2
1300 OUTPUT 709 ; "AC2"
1310 WAIT 3000
1320 DISP "CH 3 CLOSED"
1330 OUTPUT 722 ; V1$
1340 WAIT 125000
1350 OUTPUT 722 ; "REM"
1360 ENTER 722 ; V2
1370 OUTPUT 709 ; "AR"
1380 REM
1390 REM *****
1400 ! TRANSFER FIRST 9996
1410 ! HP3437A READINGS TO
1420 ! DISK FILE
1430 ASSIGN# 1 TO "VDATA"
1440 !
```

Figure 50 (continued). Data Acquisition Program.

```

1450 PRINT# 1 ; N1,V1$,V2$,V1,R1
1460 PRINT# 1 ; A1,V2,R2,A2,V3
1470 PRINT# 1 ; R3, A3, V4
1480 PRINT N1, V1$, V2$
1490 PRINT "TC1" ; V1,R1,A1
1500 PRINT "TC2" ; V2,R2,A2
1510 PRINT "TC3" ; V3,R3,A3
1520 PRINT "V4 = ";V4
1530 FOR I=1 TO 7
1540 IOBUFFER F$
1550 TRANSFER 724 TO F$ FHS ; EOI
1560 PRINT# 1 ; F$
1570 NEXT I
1580 REM *****
1590 ! VOLTAGE DATA FROM THE
1600 ! HP3437A ARE STORED ON DISK
1610 ! AS A STRING<<<<<<<<<<<<<<<
1620 REM *****
1630 ! INITIATE TAKING OF SECOND
1640 ! 9996 VOLTAGE READINGS
1650 ! BY HP3437A
1660 DISP " START HP3437A"
1670 OUTPUT 724 ; V2$
1680 WAIT 8000
1690 ! TRANSFER SECOND 9996
1700 ! READINGS FROM HP3437A
1710 ! TO DISK FILE
1720 FOR I=1 TO 7
1730 IOBUFFER F$
1740 TRANSFER 724 TO F$ FHS ;EOI
1750 NEXT I
1760 DISP " FINISHED WITH RUN";N1
1770 END

```

Figure 50 (continued). Data Acquisition Program.

DATA TRANSFER PROGRAM

This is a listing of the computer program TRANS. This program transfers the voltage readings taken by the program NSFHT into instantaneous local heat transfer coefficient values.

```
100 ! DATA TRANSFER PROGRAM
110 ! THAT TRANSFERS THE DATA
120 ! FROM THE HEAT FLUX
130 ! MEASUREMENT DATA DISK
140 ! BY J. SMALLEY AND A. H. GEORGE
150 ! MONTANA STATE UNIVERSITY
160 ! AUGUST 1989
170 !
180 ! NSF PROJECT CBT-8801618
190 !
200 !
210 ! DECLARE DISK DRIVE THAT VOLTAGE
220 ! DATA IS STORED IN
230 OPTION BASE 1
240 MASS STORAGE IS ":D701"
250 CLEAR
260 ! DECLARE THE PRINTER TO BE USED
270 PRINTER IS 701
280 ! INPUT THE DATA FILE THAT THE
290 ! VOLTAGE VALUES ARE STORED UNDER
300 DISP " INPUT FILE TO BE READ" ;
310 INPUT N$
320 ! DIMENSION STRING VARIABLES
330 DIM V1$(40),V2$(30),F$(9997)
340 DIM R(9,45)
```

Figure 51. Data Transfer Program.

```

350 GOSUB 1510 ! CALL Kcoef
360 ASSIGN# 1 TO N$
370 READ# 1 ; N1,V1$,V2$,V1,R1,A1,V2,R2,A2,V3,R3,A3,V4
380 ! DECLARE THE DISK DRIVE TO STORE THE
390 ! INSTANTANEOUS LOCAL HEAT TRANSFER
400 ! COEFFICIENTS ON
410 !
420 MASS STORAGE IS ":D700"
430 ! INPUT THE NAME THAT THE DATA
440 ! FILE IS TO BE CALLED
450 DISP " NEW DATA FILE NAME";
460 INPUT N$
470 ! CREATE THE NEW DATA FILE
480 CREATE N$,24,10000
490 ASSIGN#2 TO N$
500 PRINT R1,A1,R2,A2,R3,A3
510 PRINT " RUN NUMBER IS";N1
520 PRINT " OUTPUT VOLTAGE FROM CHANNEL #1 = ";V1;"V"
530 PRINT " OUTPUT VOLTAGE FROM CHANNEL #2 = ";V2;"V"
540 PRINT " OUTPUT VOLTAGE FROM CHANNEL #3 = ";V3;"V"
550 PRINT " OUTPUT VOLTAGE FROM CHANNEL #4 = ";V4;"V"
560 ! NOW DETERMINE THE THREE
570 ! TEMPERATURES
580 P1 = R1
590 P$ = "K"
600 P2 = A1*10
610 GOSUB 1750 ! CALL Sc. temp
620 P1 = V1/100
630 P2 = P
640 GOSUB 1750 ! CALL Sc. temp
650 T1 = P+273
660 P1 = R2
670 P2 = A2*10
680 GOSUB 1750 ! CALL Sc. temp
690 P1 = V2/100
700 P2 = P
710 GOSUB 1750 ! CALL Sc. temp
720 T2 = P+273

```

Figure 51 (continued). Data Transfer Program.

```
730 P1 = R3
740 P2 = A3*10
750 GOSUB 1750 ! CALL Sc. temp
760 P1 = V3/100
770 P2 = P
780 GOSUB 1750 ! CALL Sc. temp
790 T3 = P+273
800 ! INPUT THE BED TEMPERATURE
810 DISP " INPUT TBED IN KELVIN"
820 INPUT T9
830 PRINT "TEMPERATURE FROM CHANNEL #1 =" ; T1 ; "K"
840 PRINT "TEMPERATURE FROM CHANNEL #2 =" ; T2 ; "K"
850 PRINT "TEMPERATURE FROM CHANNEL #3 =" ; T3 ; "K"
860 ! DETERMINE THE AVERAGE SURFACE
870 ! TEMPERATURE
880 T5 = (T1+T2)/2
890 ! INPUT THE THERMAL CONDUCTIVITY
900 ! OF THE MATERIAL
910 DISP "INPUT k"
920 INPUT K
930 ! CALCULATE THE CALIBRATION
940 ! CONSTANT
950 T7=T5/298
960 B = 9.459*T7^ -0.409
970 ! DETERMINE THE TIME AVERAGE
980 ! HEAT TRANSFER COEFFICIENT
990 Q1 = K*(T1-T3) 1000 H1 = Q1/(T9-T5)
1010 ! SET ALL COUNTERS TO ZERO
1020 Z = 0
1030 C1 = 0
1040 C2 = 0
1050 C3 = 0
1060 C4 = 0
1070 C5 = 0
1080 C6 = 0
1090 C7 = 0
1100 C8 = 0
```

Figure 51 (continued). Data Transfer Program.

```
1110 C9 = 0
1120 ! CALCULATE THE INSTANTANEOUS
1130 ! LOCAL HEAT TRANSFER
1140 ! COEFFICIENT
1150 FOR J=1 TO 14
1160 READ# 1 ; F$
1170 FOR I = 1 TO 9989 STEP 7
1180 V5 = VAL(F$[I,I+6])
1190 Q2 = Q1+(V5-V4)*1000000/B
1200 H2 = Q2/(T9-T5)
1210 ! COMPARE INSTANTANEOUS VALUES
1220 ! AGAINST AVERAGE VALUES
1230 H=H2/H1
1240 IF H<1 THEN C1 = C1+1
1250 IF H>1 THEN C2 = C2+1
1260 IF H>1.25 THEN C3 = C3+1
1270 IF H>1.5 THEN C4 = C4+1
1280 IF H>2 THEN C5 = C5+1
1290 IF H>2.5 THEN C6 = C6+1
1300 IF H>3 THEN C7 = C7+1
1310 IF H>3.5 THEN C8 = C8+1
1320 IF H>4 THEN C9 = C9+1
1330 ! STORE THE VALUE OF H2
1340 ! ON DISKK
1350 PRINT# 2 ; H2
1360 Z = Z+1
1370 NEXT I
1380 NEXT J
1390 PRINT "# OF TIMES H<H AVG =" ; C1
1400 PRINT "# OF TIMES H>H AVG =" ; C2
1410 PRINT "# OF TIMES H>1.25H AVG =" ; C3
1420 PRINT "# OF TIMES H>1.5H AVG =" ; C4
1430 PRINT "# OF TIMES H>2H AVG =" ; C5
1440 PRINT "# OF TIMES H>2.5H AVG =" ; C6
1450 PRINT "# OF TIMES H>3H AVG =" ; C7
1460 PRINT "# OF TIMES H>3.5H AVG =" ; C8
1470 PRINT "# OF TIMES H>4H AVG =" ; C9
1480 PRINT "# OF READINGS =" ; Z
1490 DISP "TRANSFER COMPLETE"
1500 END
```

Figure 51 (continued). Data Transfer Program.

PROGRAM TO DETERMINE SUPERFICIAL GAS VELOCITY

This is a listing of the computer program UoAHG which determines the superficial gas velocity for the OSU high temperature fluidized bed facility. This program was used by Goshayeshi (1989) and the print statements were modified for use here by Dr. Alan George.

```

10 ! VENTURI DATA PROCESSING
20 G=980.6 @ D1=2.4414 @ F1=1
30 F=1.0342 @ C=.984 @ S2=3.5 @ B=.5050501
40 R1=2.7
50 REM PARTICLE SIZE IN mm
52 P=1
55 REM P IS IN mm
60 DISP "WHAT IS BED TEMP.?" @ INPUT T1
65 REM T1 IS IN DEGREES F
70 CLEAR
80 DISP "WHAT IS AIR INLET PRESSURE?" @ INPUT DP1
85 REM P1 IS IN inHg
90 DISP "WHAT IS PRESSURE DROP?" @ INPUT D
95 REM D IS IN inH2O
100 DISP "WHAT IS AIR INLET TEMP.?" @ INPUT T
105 REM T IS IN DEGREES F
110 T2=(.02*T1-1.29)/(1663.1-.085*T1)
120 R2=.6297/(459.7+T1)
130 M=.0003934 + .0000001376*(T1-1200)
140 A1=M/(P*R2)
150 A2=P*P*P*G*R2*(R1-R2)/(M*M)
160 A3=SQR(33.7*33.7+.0408*A2)
170 U1=A1*(A3-33.7)
180 U2=U1/30.48
190 P1=P1*.4912

```

Figure 52. Superficial Gas Velocity Program.

```
200 P2=14.697+P1
210 P3=P2-.03613*D
220 R=P3/P2
230 S1=R^ 1.429
240 S3=(1-R^ .2857)/(1-R)
250 S4=(1-8^ 4)/(1-8^ 4*S1)
260 Y=SQR(S1*S2*S3*S4)
270 G1=39.626/(T+459.7)
280 G2=G1*P2*.068041
290 W=5.983*C*F*D1*F1*Y*SQR(D*G2)
300 A=W/G2
310 S=A*36.0414*P2/(T+459.7)
320 P1=P1/.4912
330 B1=S*T2
340 C1=(S+2*B1)*(459.7+T1)/529.7
350 U0=C1/120
360 U0=U0*.3048
370 U3=U0/U2
375 PRINT "mm degF inHg inH2O degF m/s"
380 PRINT USING 390 ; P,T1,P1,D,T,U0"
390 IMAGE(D.DD,2X,4D,2X,D.D,2X,DD.D,2X,3D,2X,D.DD)
400 PRINT USING 410
410 IMAGE (//)
420 REM
430 END
```

Figure 52 (continued). Superficial Gas Velocity Program.

APPENDIX B
FLUIDIZED BED FACILITIES

OSU FLUIDIZED BED FACILITY

A schematic illustration of the OSU fluidized bed facility from Goshayeshi (1989) is shown in Figure 53. A brief description of the principle components follows.

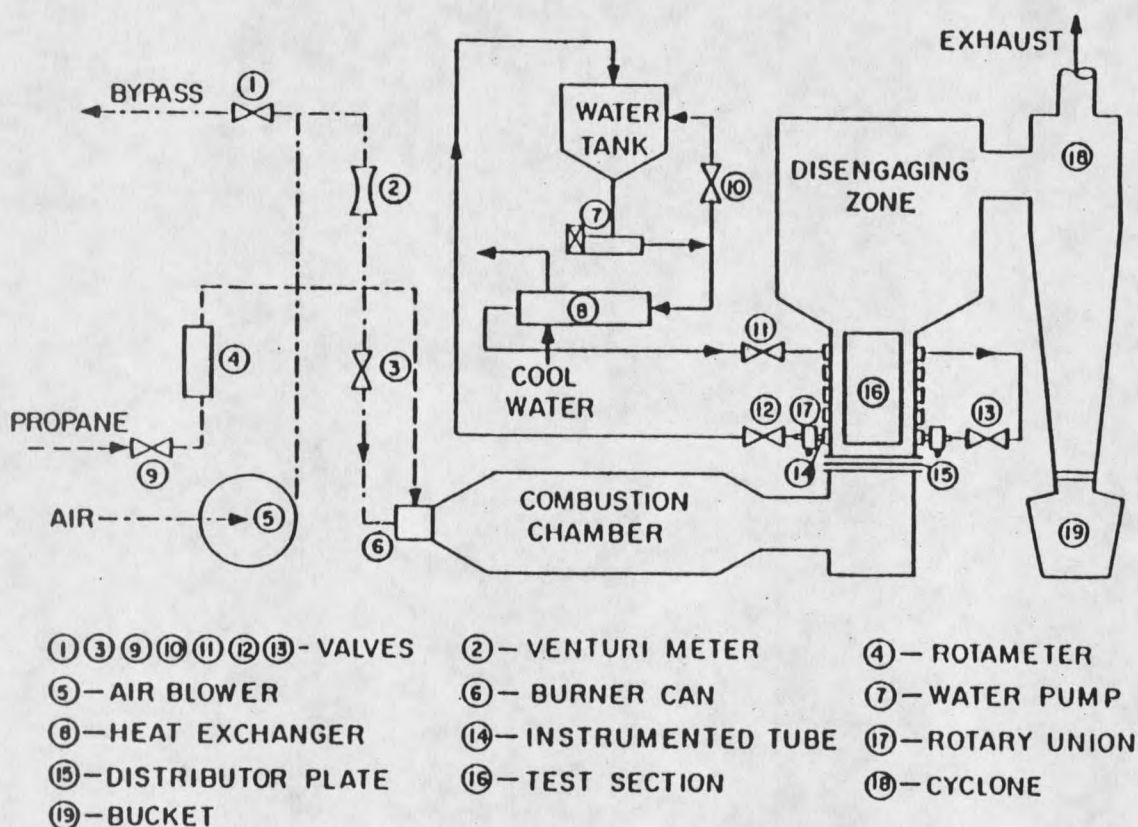


Figure 53. Schematic of OSU High Temperature Fluidized Bed.

Test Section and Distributor Plate

The test section was designed and constructed to accommodate both single tubes and tube array experiments with different tube configurations. Fifteen mounting ports were provided on each side of the test section with an equilateral triangular configuration and 15.24 cm spacing between centers. The ports were set in three columns and five rows. The centerline of the bottom row was 30.48 cm above the distributor plate. The port used for this experiment was in the first column and the third row from the bottom.

The test section was made of 15.24 cm thick refractory material with cross-sectional dimensions of 0.30 m by 0.60 m and interior height of 1.2 m. The top end of the test section flares out to a final dimension of 0.6 m by 0.6 m. A drain plug at the base of the test section, just above the distributor plate, was provided for the removal of bed media as needed. A disengaging zone with cross-sectional area of 0.6 m by 0.6 m at the top of the test section prevented particles from escaping. A glass viewport was provided at the top of the disengaging zone to allow visual observation of the top of the bed. This was also used as a filling port for adding particles into the test section.

The distributor plate was made of two inconel plates, each 3.18 mm thick. They were held together by a bolted flange connection between the gas inlet and the test section. The top plate contains 171 holes of approximately 6.35 mm diameter placed on 31.75 mm centers in a square pattern. The lower plate was identical to the top plate except for having 9.53 mm diameter holes. To prevent particles from flowing back when the bed was not fluidized, a stainless steel screen with 0.457 mm wire diameter was placed between the two plates.

Air Flow Measurement

The air flow into the system was provided by an air blower. Two silencers are in the system at the inlet and outlet of the blower, reducing the noise level. The compressed air flow is then metered through a venturi orifice and directed into the combustion chamber. The air flow rate and its inlet pressure to the propane burner are controlled by adjusting the air inlet and bypass valves. The high-temperature combustion products leaving the combustion chamber proceed to the fluidized bed test section through the distributor plate.

The air flow rate was measured by an ASME standard venturi meter. The pressure difference between the venturi meter throat and upstream section was monitored by a manometer. Downstream from the meter, air pressure and temperature were measured as well. A computer program taken from Goshayeshi (1989) and rewritten by Dr. Alan George was used to compute the superficial gas velocity. The listing of the computer program is shown in Appendix A.

Fuel and Temperature Controller

Propane was supplied to the burner by a main fuel valve. A proportional type controller was used to regulate the propane flow rate and provide the desired gas temperature at the combustion chamber exit. The fuel flow rate could also be manually controlled by a bypass valve. This was done for the high temperature run to allow the bed to get as hot as possible.

A digital thermometer and type K thermocouple were used to measure the fluidized-bed temperature. The thermocouple was located centrally at 15.2 cm above the distributor plate and protruded into the test section about 10.2 cm. The bed temperature could be set on the control panel and the fuel flow rate was automatically adjusted to meet the set temperature.

MSU FLUIDIZED BED FACILITY

A schematic illustration of the MSU fluidized bed facility is shown in Figure 54. A brief description of the principle components follows.

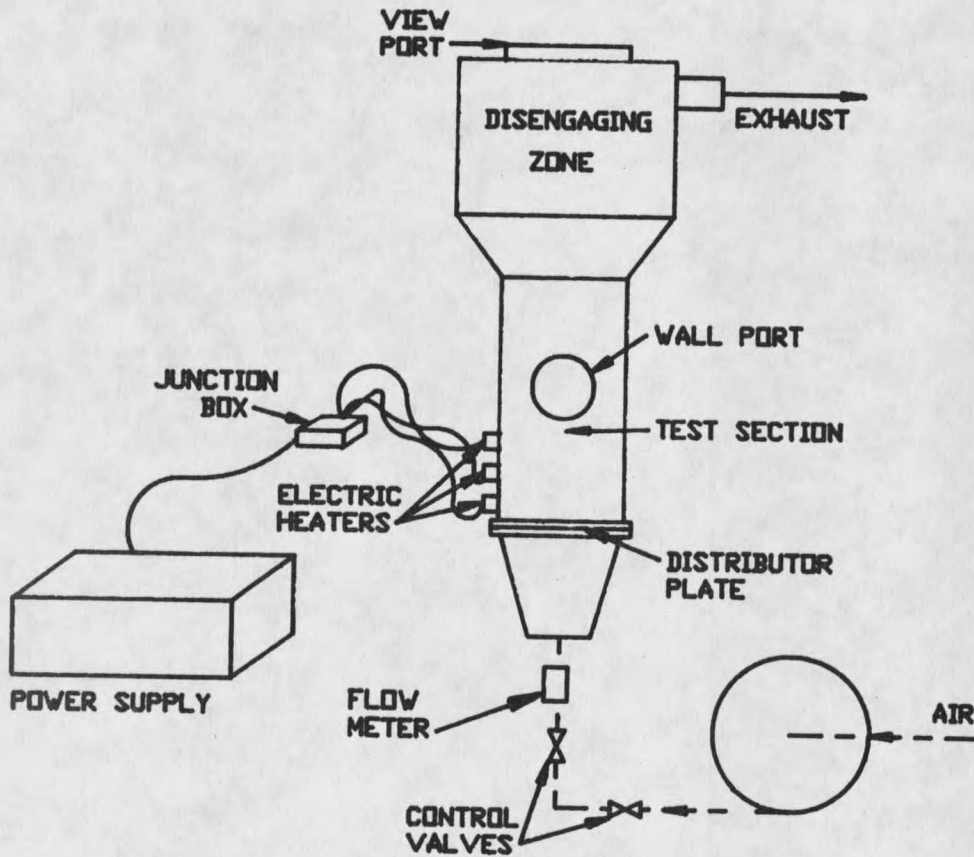


Figure 54. Schematic of MSU Low Temperature Fluidized Bed.

Test Section

The test section was designed and constructed to accommodate a single wall transducer. The mounting port has a diameter of 3.81 cm and is located approximately 15.24 cm above the distributor plate.

The test section was made of .64 cm thick plate steel. The interior cross-sectional dimension of the test section was 10.16 cm by 10.16 cm and interior height of approximately 40 cm. The top end of the test section flares out to a final dimension of 19 cm by 19 cm. A drain plug at the base of the test section, just above the distributor plate, was provided for the removal of bed material as needed. A glass viewport was provided at the top of the fluidized bed to allow visual observation of the top of the bed. This was also used as a filling port for adding particles into the test section.

Air Supply

Air flow into the system was provided by an air compressor. The air was channeled through two mufflers to remove any oil that had leaked into the system. The compressed air flow was metered through a Meriam Laminar Flow Element and directed through the distributor plate and into the test section. The air flow rate was measured by the Meriam Laminar Flow Element. The pressure difference across the laminar flow element was measured using a manometer.

Heating Supply and Temperature Controller

Heat was supplied to the fluidized bed by nine electric heaters. The heaters were spaced across the test section just above the distributor plate. A Nobatron DCR150-15A power supply was used to supply electricity to the heaters. With this arrangement a maximum safe bed temperature of approximately 650 K could

be attained. The bed temperature was monitored using a digital thermometer and a type K thermocouple. The thermocouple was located centrally at approximately 15 cm above the distributor plate. The temperature was controlled by adjusting the output voltage on the power supply until the desired bed temperature was achieved.

APPENDIX C
THEORY OF ANALOG CIRCUIT

THEORY OF ANALOG CIRCUIT

To review the concepts of transfer function and frequency response, first consider a RC-Low pass filter shown in Figure 55.

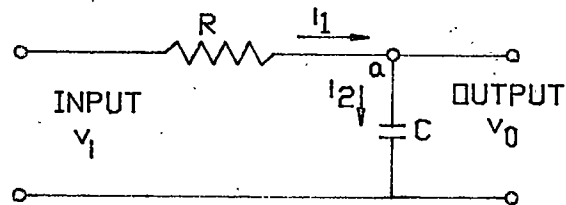


Figure 55. Schematic of RC-Low Pass Filter.

For a capacitor

$$(43) \quad i = C \frac{dv}{dt} .$$

By Kirchoff's Current Law at node a

$$(44) \quad i_1 - i_2 = 0$$

but

$$(45) \quad i_1 = \frac{v_i - v_o}{R}$$

and

$$(46) \quad i_2 = C \frac{dv_o}{dt} .$$

Therefore, (44) becomes

$$(47) \quad RC \frac{dv_o}{dt} + v_o = v_i$$

This is an ordinary differential equation which gives the relationship between the input and output.

Take the Laplace Transform with zero initial condition of (47):

$$(48) \quad RCs\bar{v}_o(s) + \bar{v}_o(s) = \bar{v}_i(s)$$

or

$$(49) \quad \bar{v}_o(s) = \frac{1}{RCs + 1} \bar{v}_i(s)$$

The transfer function is

$$(50) \quad G_1(s) = \frac{\bar{v}_o(s)}{\bar{v}_i(s)} = \frac{1}{1 + RCs}$$

thus,

$$(51) \quad \bar{v}_o(s) = G_1(s)\bar{v}_i(s)$$

This is shown schematically in Figure 56.

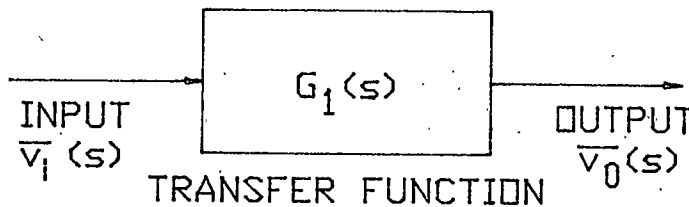


Figure 56. Block Diagram of Transfer Function for the RC-Low Pass Filter.

Now consider the problem of computing the frequency response for the same circuit, i.e., the response to a steady harmonic input. Let

$$(52) \quad v_i = V_i \cos \omega t = \text{Re} [V_i e^{j\omega t}]$$

where

$\text{Re}[\] = \text{real part}$

$V_i = \text{constant}$

For the above input, the output will be of the form

$$(53) \quad v_o = V_o \cos(\omega t + \phi) = \text{Re} [V_o e^{j(\omega t + \phi)}]$$

Substitute (53) into (47) to get

$$(54) \quad RC \frac{d}{dt} [V_o e^{j\phi} e^{j\omega t}] + V_o e^{j\phi} e^{j\omega t} = V_i e^{j\omega t}$$

Taking the derivative of (54) and reducing terms gives

$$(55) \quad [RC j\omega + 1] V_o e^{j\phi} = V_i$$

Therefore,

$$(56) \quad V_o e^{j\phi} = \frac{V_i}{[RC j\omega + 1]} = \frac{V_i}{\sqrt{(RC \omega)^2 + 1}} e^{j\gamma}$$

$$(57) \quad \gamma = -\tan^{-1}(RC \omega) = \phi$$

where ϕ is the phase angle. The amplitude is given by (58), and the amplitude ratio is given by (59):

$$(58) \quad V_o = \frac{V_i}{\sqrt{1 + (RC \omega)^2}}$$

$$(59) \quad \frac{V_o}{V_i} = \frac{1}{\sqrt{1 + (RC \omega)^2}}$$

Consider (50) and (57) with $s = j\omega$. The angle of the transfer function is the phase angle of the output for a steady harmonic input. More precisely, the corresponding phase angle is the angle of the denominator subtracted from the angle of the numerator, or, mathematically,

$$(60) \quad G_1(j\omega) = \frac{1}{1 + RC j\omega}$$

$$(61) \quad \angle G_1(j\omega) = 0 - \tan^{-1} \frac{RC \omega}{1} = -\tan^{-1}(RC \omega) \quad .$$

Therefore,

$$(62) \quad \phi = \angle G_1(j\omega) = -\tan^{-1}(RC \omega) \quad .$$

Consider (50) and (59) with $s = j\omega$. The amplitude ratio for a steady harmonic input is the magnitude of the transfer function given by

$$(63) \quad |G_1(j\omega)| = \left| \frac{1}{1 + RC j\omega} \right| = \frac{1}{\sqrt{1 + (RC \omega)^2}} = \frac{V_o}{V_i} \quad .$$

Since (63) gives the magnitude of $G_1(j\omega)$ and (62) gives the angle of $G_1(j\omega)$, it is clear that $G_1(j\omega)$ is fully determined by the frequency response characteristics of the circuit. That is, $G_1(j\omega)$ is a complex number that is fully specified by its magnitude and angle.

Now consider the heat flux transducer shown in Figure 57. Start with the one-dimensional heat conduction equation and the following boundary conditions:

$$(64) \quad \begin{aligned} \frac{\partial^2 T}{\partial x^2} &= \frac{1}{\alpha} \frac{\partial T}{\partial t} \\ T(x, 0) &= 0 \\ T(0, t) &= T_w(t) \\ T(\infty, t) &= 0 \end{aligned}$$

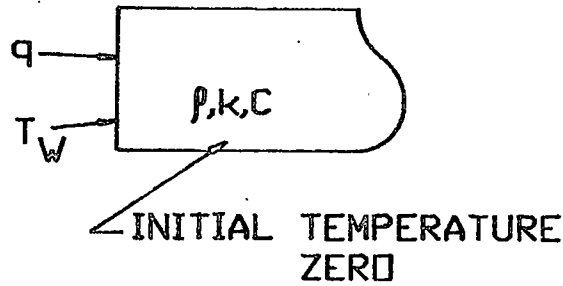


Figure 57. General Heat Flux Transducer.

Take the Laplace Transform of (64) and the boundary conditions to get

$$(65) \quad \bar{T}(x, s) = \bar{T}_w(s) e^{-\sqrt{s/\alpha} x} .$$

However, from the initial problem

$$(66) \quad \bar{q}_w(s) = -k \frac{d\bar{T}(s)}{dx} .$$

Substitute the derivative of (65) into (66) to get the transfer function

$$(67) \quad \frac{\bar{q}_w(s)}{\bar{T}_w(s)} = \sqrt{k\rho c} \sqrt{s} .$$

Both the heat flux and the surface temperature are linearly related to voltages in the circuit. That is

$$(68) \quad \bar{v}(s) = y_1 \bar{q}_w(s)$$

and

$$(69) \quad \bar{e}(s) = x_1 \bar{T}_w(s)$$

where x_1 and y_1 are constants. This assumption is valid for small changes in the surface temperature. Therefore,

$$(70) \quad \frac{\bar{v}(s)/y_1}{\bar{e}(s)/x_1} = \sqrt{k\rho c} \sqrt{s}$$

or

$$(71) \quad \frac{\bar{v}(s)}{\bar{e}(s)} = G(s) = \frac{y_1}{x_1} \sqrt{k\rho c} \sqrt{s} .$$

This is shown schematically in Figure 58.

Let $s = j\omega$, to obtain the frequency response characteristics of the circuit:

$$(72) \quad G(j\omega) = \frac{y_1}{x_1} \sqrt{k\rho c} \sqrt{j\omega} = \frac{y_1}{x_1} \sqrt{k\rho c} \left[\omega e^{j\pi/2} \right]^{1/2} .$$

Reducing (72) yields

$$(73) \quad G(j\omega) = \frac{y_1}{x_1} \sqrt{k\rho c} \sqrt{\omega} e^{j\pi/4} .$$

The amplitude ratio becomes

$$(74) \quad |G(j\omega)| = \frac{y_1}{x_1} \sqrt{k\rho c} \sqrt{\omega} .$$

The phase angle becomes

$$(75) \quad \angle G(j\omega) = \pi/4 = 45^\circ .$$

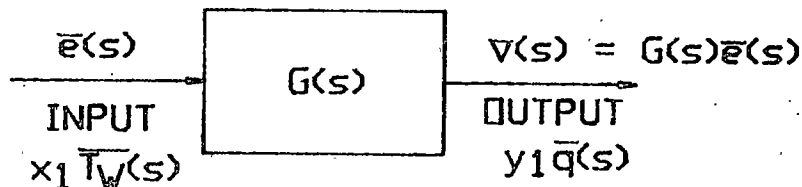


Figure 58. Block Diagram of Transfer Function for the RC-Low Pass Filter with Voltage Input and Output.

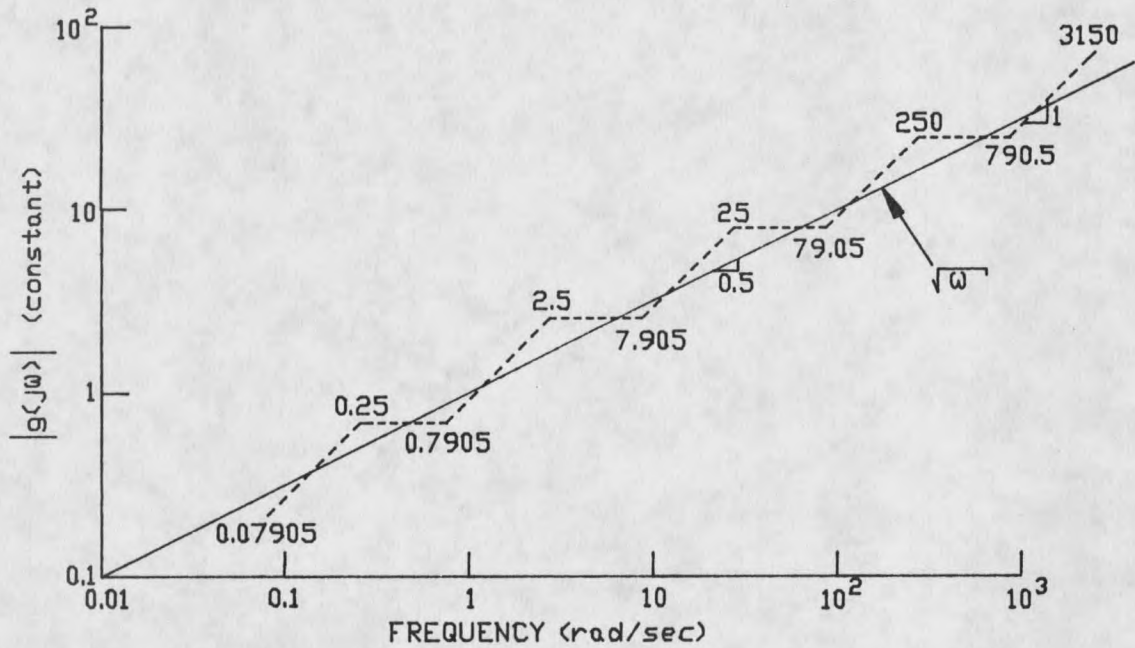


Figure 59. Bode Plot for the Filter Chosen in Analog Circuit.

Therefore, a filter with an amplitude ratio proportional to $\sqrt{\omega}$ and a phase angle of 45° is needed. Many designs are possible. Assume the form

$$(76) \quad g(s) = \frac{(s + a_1)(s + a_2)(s + a_3)(s + a_4)(s + a_5)}{(s + b_1)(s + b_2)(s + b_3)(s + b_4)(s + b_5)}.$$

To select the constants a and b use the Bode plot method as given by Cannon (1967). The Bode plot is shown in Figure 59. From Figure 59, (76) becomes

$$(77) \quad g(s) = \frac{(s + 0.07905)(s + 0.7905)(s + 7.905)(s + 79.05)(s + 790.5)}{(s + 0.25)(s + 2.5)(s + 25)(s + 250)(s + 3150)}.$$

Now set $s = j\omega$ and obtain $|g(j\omega)|$ and $\angle g(j\omega)$:

$$(78) \quad |g(j\omega)|^2 = \frac{(\omega^2 + 0.00625)(\omega^2 + 0.625)(\omega^2 + 62.5)(\omega^2 + 6249)(\omega^2 + 624900)}{(\omega^2 + 0.0625)(\omega^2 + 6.25)(\omega^2 + 625)(\omega^2 + 62500)(\omega^2 + 9.923 \times 10^6)}$$

$$\begin{aligned}
 \angle g(j\omega) = & \tan^{-1} \frac{\omega}{0.07905} + \tan^{-1} \frac{\omega}{0.7905} + \tan^{-1} \frac{\omega}{7.905} \\
 (79) \quad & + \tan^{-1} \frac{\omega}{79.05} + \tan^{-1} \frac{\omega}{790.5} - \tan^{-1} \frac{\omega}{0.25} - \tan^{-1} \frac{\omega}{2.5} \\
 & - \tan^{-1} \frac{\omega}{25} - \tan^{-1} \frac{\omega}{250} - \tan^{-1} \frac{\omega}{3150} .
 \end{aligned}$$

The functions $|g(j\omega)|$ and $\angle g(j\omega)$ are tabulated in Table 14 for the frequency range $\omega = 0.1$ Hz to 240 Hz.

A good approximation is (80), where the constant 1.194×10^{-2} is the mean value from $\omega = 0.1$ Hz to 240 Hz:

$$(80) \quad \frac{|g(j\omega)|}{\omega} = 1.194 \times 10^{-2} (\text{sec})^{1/2}$$

$$(81) \quad \sqrt{\omega} = 83.8 |g(j\omega)|$$

where the constant 83.8 has units of (seconds) $^{-0.5}$. Note, ω must be in rad/sec for the calculations in Table 14. Note also $\angle g(j\omega) = 45^\circ \pm 3^\circ$. To a good approximation (82) applies:

$$(82) \quad \sqrt{j\omega} = 83.8 |g(j\omega)| \angle g(j\omega)$$

or

$$(83) \quad \sqrt{j\omega} = 83.8 g(j\omega)$$

in the form of a Laplace Transform (let $s = j\omega$),

$$(84) \quad \sqrt{s} = 83.8 g(s)$$

From (71) and (84),

$$(85) \quad \frac{\bar{v}(s)}{\bar{e}(s)} = \frac{y_1}{x_1} \sqrt{k_{pc}} \sqrt{s} = \frac{y_1}{x_1} \sqrt{k_{pc}} 83.8 g(s)$$

and from (67) and (84),

$$(86) \quad \bar{q}_w(s) = \sqrt{k\rho c} (83.8) g(s) \bar{T}_w(s) \quad .$$

Substitute (69) into (86) to obtain

$$(87) \quad \bar{q}_w(s) = \sqrt{k\rho c} (83.8) \frac{1}{x_1} g(s) \bar{e}(s) \quad .$$

Now consider the circuit block diagram shown in Figure 60.

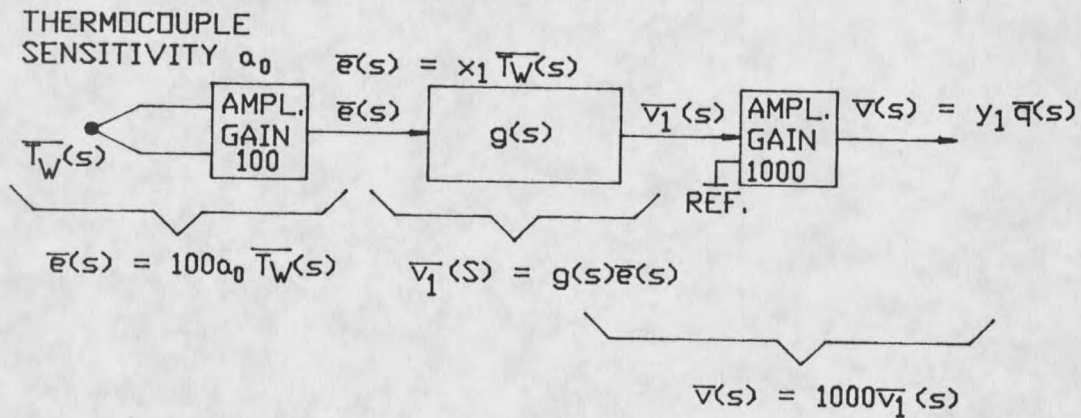


Figure 60. Block Diagram of Transfer Function with Gain Amplifiers.

From Figure 60 the following equations are obtained:

$$(88) \quad \bar{e}(s) = 100a_0 \bar{T}_w(s) = x_1 \bar{T}_w(s) \quad ,$$

$$(89) \quad x_1 = 100a_0 \quad ,$$

$$(90) \quad \bar{v}_1(s) = g(s)\bar{e}(s)$$

and

$$(91) \quad \bar{v}(s) = 1000\bar{v}_1(s) \quad .$$

Substituting (89), (90) and (91) into (87) gives

$$(92) \quad \bar{q}_w(s) = \sqrt{k\rho c} (83.8) \left(\frac{1}{1000} \right) \left(\frac{1}{100a_0} \right) \bar{v}(s)$$

where

$$(93) \quad \frac{(1000)(100a_0)}{83.8\sqrt{k\rho c}} = \beta_1 \quad .$$

Therefore

$$(94) \quad \bar{v}(s) = \beta_1 \bar{q}_w(s) \quad .$$

Now taking the inverse Laplace Transform of (92) gives

$$(95) \quad v(t) = \beta_1 q_w(t) \quad .$$

To construct the circuit that approximates the transfer function $g(s)$, consider the circuit shown in Figure 61.

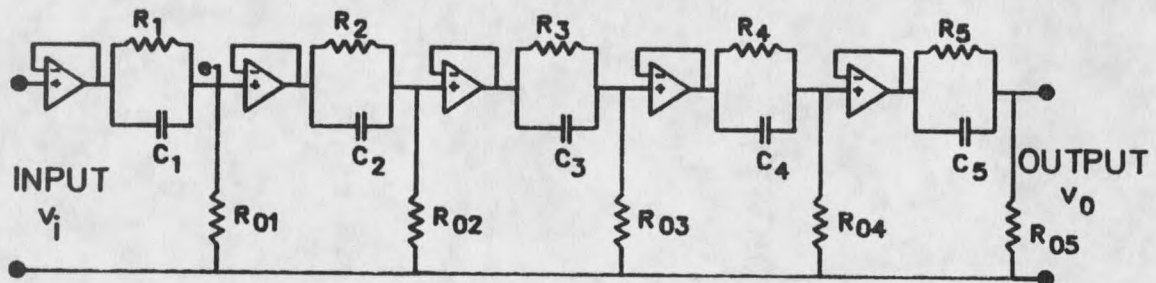


Figure 61. General Circuit Diagram for Circuit that Approximates the Transfer Function.

ω (Hz)	$ g(j\omega) $ (Dimensionless)	$\frac{ g(j\omega) }{\sqrt{\omega}}$ (Sec) ^{1/2}	$\frac{83.8 g(j\omega) }{\sqrt{\omega}}$ (Dimensionless)	$\angle g(j\omega)$ (degrees)
0	2.509×10^{-3}			0
0.1	9.232×10^{-3}	1.165×10^{-2}	0.976	42.4
0.2	1.323×10^{-2}	1.180×10^{-2}	0.989	45.6
0.4	1.939×10^{-2}	1.223×10^{-2}	1.025	44.6
0.6	2.339×10^{-2}	1.204×10^{-2}	1.009	43.3
0.8	2.644×10^{-2}	1.179×10^{-2}	0.99	43.3
1.0	2.917×10^{-2}	1.164×10^{-2}	0.98	43.9
2.0	4.184×10^{-2}	1.180×10^{-2}	0.99	46.4
4.0	6.133×10^{-2}	1.223×10^{-2}	1.02	44.9
6.0	7.395×10^{-2}	1.204×10^{-2}	1.01	43.5
8.0	8.362×10^{-2}	1.179×10^{-2}	0.99	43.5
10.0	9.227×10^{-2}	1.164×10^{-2}	0.98	44.0
20.0	0.1342	1.181×10^{-2}	0.99	46.4
40.0	0.1942	1.225×10^{-2}	1.03	44.8
60.0	0.2346	1.208×10^{-2}	1.01	43.3
80.0	0.2659	1.186×10^{-2}	0.99	43.0
100.0	0.2942	1.174×10^{-2}	0.98	43.5
120.0	0.3217	1.172×10^{-2}	0.98	44.0
140.0	0.3491	1.177×10^{-2}	0.99	44.4
160.0	0.3764	1.187×10^{-2}	0.99	44.7
180.0	0.4034	1.200×10^{-2}	1.01	44.7
200.0	0.4301	1.213×10^{-2}	1.02	44.6
220.0	0.4562	1.227×10^{-2}	1.03	44.3
240.0	0.4816	1.240×10^{-2}	1.04	43.9

Table 14. Values of the Magnitude and Angle of the Transfer Function as a Function of the Frequency.

The analysis is the same for each stage, but they have to be analyzed separately. In terms of Laplace Transforms, the analysis for the first stage gives (96), by use of the voltage divider equation and an AC circuit analysis (with $s = j\omega$):

$$(96) \quad \frac{\bar{e}_1(s)}{\bar{v}_1(s)} = \frac{R_{01}}{R_{01} + \frac{R_1}{1+sR_1C_1}} = \frac{s + \left(\frac{1}{R_1C_1}\right)}{s + \left(\frac{R_1+R_{01}}{R_{01}R_1C_1}\right)}$$

Therefore

$$(97) \quad \frac{\bar{e}_1(s)}{\bar{v}_1(s)} = \frac{s + a_1}{s + b_1}$$

where

$$a_1 = \frac{1}{R_1C_1} \quad \text{and} \quad b_1 = \frac{R_1 + R_{01}}{R_{01}R_1C_1} = \frac{R_1 + R_{01}}{R_{01}} a_1$$

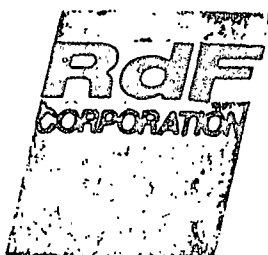
or

$$(98) \quad R_1C_1 = \frac{1}{a_1}$$

$$(99) \quad R_{01} = \frac{R_1}{\frac{b_1}{a_1} - 1}$$

The values for a_1 and b_1 are obtained from (77). This analysis continues in the same manner until all of the stages have been analyzed. The resulting circuit is shown in Figure 13 of Chapter 4.

APPENDIX D
RdF GAGE CALIBRATION LITERATURE



23 Elm Avenue
Hudson, New Hampshire 03051
Tel: (603) 882-5195 TWX: 710-228-1382

Q-116-01

MICRO-FOILTM HEAT FLOW SENSOR

CALIBRATION

RdF PART NO. 20455-1SERIAL NO. 1494

HEAT FLOW SENSOR:

Output at 70°F: 0.21 microvolts/Btu-Ft⁻² - Hr⁻¹

Polarity: (For heat flow into surface)

White - Positive (+)

Red - Negative (-)

Temperature Multiplication Factor: See Attached Graph

*Thermal Resistance: .003 °F/Btu-Ft⁻² - Hr⁻¹ (Typ)*Heat Capacity: .01 Btu-Ft⁻²/°F (Typ)Response Time: .020 sec. (62% response to step function) (Typ)

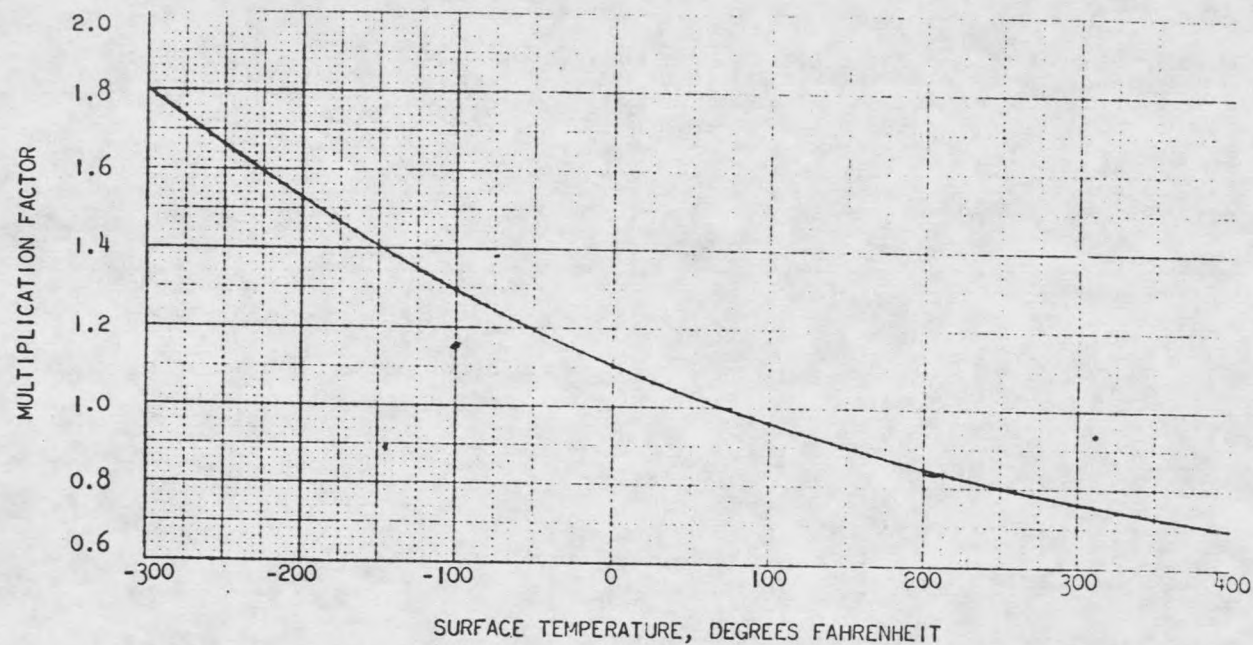
THERMOCOUPLE:

ANSI Type	Material	Polarity	Color
T	Copper Constantan	Pos. (+) Neg. (-)	Blue Red

Output per ANSI MC96.1-1975 and NBS Monograph 125

*Thermal resistance is the temperature difference between the front surface and rear mounting surface of the sensor per unit of heat flow through the sensor. Heat capacity is the amount of heat required to raise the mean temperature of the sensor 1°F. Typical values of these two properties are given primarily to indicate sensor capabilities and are required for heat flow calculations only in very rare instances.

BY: Denise FultonDATE: 12-11-87



MICRO-FOIL HEAT FLOW SENSOR

OUTPUT MULTIPLICATION FACTOR VS RECEIVING SURFACE TEMPERATURE
(70°F BASE)

RdF CORPORATION
HUDSON, NEW HAMPSHIRE

Figure 62. Temperature Correction Curve from the RdF Calibration Literature.

REFERENCES CITED

REFERENCES CITED

1. Adams, R.L. and Welty, J.R., "A Gas Convection Model of Heat Transfer in Large Particle Fluidized Beds," *AIChE J.*, 25, 395, 1979.
2. Alavizadeh, N., Adams, R.L., Welty, J.R. and Goshayeshi A., "An Instrument for Local Radiative Heat Transfer Measurement in a Gas-Fluidized Bed at Elevated Temperature," Proceedings, 22nd ASME/AIChE National Heat Transfer Conference, Niagara Falls, New York, 1984.
3. Alkidas, A.C. and Cole, R.M., "Transient Heat Flux Measurements in a Divided-Camber Diesel Engine," *ASME J. of Heat Transfer*, 107, 439, 1985.
4. Alkidas, A.C. and Myers, J.P., "Transient Heat Flux Measurements in the Combustion Chamber of a Spark-Ignition Engine," *ASME J. of Heat Transfer*, 104, 63, 1982.
5. Baskakov, A.P., Berg, B.V., Vitt, O.K., Filippovsky, N.F., Kirakosyan, V.A., Goldobin, J.M. and Maskae, V.K., "Heat Transfer to Objects Immersed in Fluidized Beds," *Powder Tech.*, 8, 273, 1973.
6. Cannon, R.H., *Dynamics of Physical Systems*, McGraw-Hill, 511-514, 1967.
7. Catipovic, N.M., Fitzgerald, T.J., George, A.H. and Welty, J.R., "Experimental Validation of the Adams-Welty Model for Heat Transfer in Large-Particle Fluidized Beds," *AIChE J.*, 28, 714, 1982.
8. Catipovic, N.M., "Heat Transfer to Horizontal Tubes in Fluidized Beds: Experiment and Theory," Ph.D. thesis, Oregon State University, Corvallis, Oregon, 1979.
9. Catipovic, N.M., Fitzgerald, T.J. and Jovanovic, G., "A Study of Heat Transfer to Immersed Tubes in Fluidized Beds," Paper 28e, AIChE 71st Annual Meeting, Miami, Florida, November 12-16, 1978.
10. Decker, N. and Glicksman, L.R., "Heat Transfer in Large Particle Fluidized Beds," *Int. J. Heat Mass Transfer*, 26, 1307, 1983.

11. Doebelin, E.O., *Measurement Systems Application and Design*, fourth edition, McGraw-Hill, 110, 1990.
12. Forsythe, G.E., Malcolm, M.A. and Moler, C.B., *Computer Methods for Mathematical Computations*, Prentice-Hall, 85, 1977.
13. George, A.H., "A Transducer for the Measurement of Instantaneous Local Heat Flux to Surfaces Immersed in High Temperature Fluidized Beds," *Int. J. Heat Mass Transfer*, 30, 763, 1987.
14. George, A.H. and Welty, J.R., "Local Heat Transfer Coefficients for a Horizontal Tube in a Large Particle Fluidized Bed at Elevated Temperature," *AIChE J.*, 30, 482, 1984.
15. George, A.H., "An Experimental Study of Heat Transfer to a Horizontal Tube in a Large Particle Fluidized Bed at Elevated Temperature," Ph.D. thesis, Oregon State University, Corvallis, Oregon, 1981.
16. Gloski, D., Glicksman, L. and Decker, N., "Thermal Resistance at a Surface in Contact with Fluidized Bed Particles," *Int. J. Heat Mass Transfer*, 27, 599, 1984.
17. Goshayeshi, A., "Experimental Studies of Heat Transfer with an Array of Horizontal Tubes Immersed in a High-Temperature Fluidized Bed," Ph.D. thesis, Oregon State University, Corvallis, Oregon, 1989.
18. Goshayeshi, A., Welty, J.R., Adams, R.L. and Alvizadeh, N., "An Experimental Study of Heat Transfer in an Array of Horizontal Tubes in Large Particle Fluidized Beds at Elevated Temperatures," 23rd ASME/AIChE National Heat Transfer Conference, Denver, Colorado, August 4-7, 1985.
19. Grewal, N.S. and Saxena, S.C., "Maximum Heat Transfer Coefficient Between Horizontal Tube and a Gas-Solid Fluidized Bed" *Ind. Eng. Chem. Process Des. Dev.*, 20, 108, 1981.
20. Holman, J.P., *Heat Transfer*, fourth edition, McGraw-Hill, 102, 1976.

21. Karlekar, B.V. and Desmond, R.M., *Heat Transfer*, second edition, West Publishing, 776, 1977.
22. Makansi, J. and Schwieger, R., "Fluidized-Bed Boilers," *Power*, 131, 5, 1987.
23. Mickley, H.S., Fairbanks, D.F. and Hawthorn, R.D., "The Relation Between the Transfer Coefficient and Thermal Fluctuations in Fluidized-Bed Heat Transfer," *Chem. Engr. Progr. Symp. Series*, 57 (32), 51, 1961.
24. Raven, F.H., *Automatic Control Engineering*, third edition, McGraw-Hill, 178, 1978.
25. Schwieger, B., "Fluidized Bed Boilers Achieve Commercial Status Worldwide," *Power*, 129, 17, 1985.
26. Skinner, G.T., "Analog Network to Convert Surface Temperature to Heat Flux," *ARS J.*, 30, 569, 1960.
27. Spiegel, M.R., *Mathematical Handbook of Formulas and Tables*, Schaums Outline Series in Mathematics, McGraw-Hill, 164, 1968.
28. Touloukian, Y.S., Powell, R.W., Ho, C.Y., and Nicolaou, M.C., *Thermophysical Properties of Matter*, Vol. 10, Thermal Diffusivity, Plenum, 346, 1973.
29. Touloukian, Y.S., Powell, R.W., Ho, C.Y., and Klemens, P.G., *Thermophysical Properties of Matter*, Vol. 1, Thermal Conductivity Metallic Elements and Alloys, Plenum, 1174, 1970.
30. Tout, J. and Clift, R., "Heat Transfer Around Single Bubbles in a Two-Dimensional Fluidized Bed," *AIChE Symp. Series*, 69 (128), 78, 1973.
31. Varygin, N.N. and Martyushin, I.G., *Khim. Mash.*, 1966, (as cited in Grewal N.S. and Saxena, S.C., 1981).
32. Zabrodsky, S.S., Antonishin, N.V., Vasiliev, G.M. and Paranas, A.L., *Visti Akad. Nauk BSSR, Ser Fiz-Energ. Nauk*, No. 4, 103, 1974 (as cited by George, A.H., 1984).

