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TRANSIENT THERMAL PERFORMANCE OF AN
EXPERIMENTAL PACKED BED THERMAL
ENERGY STORAGE SYSTEM

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Arnold Glenn Graves

June 1985

ACKNOWLEDGMENTS

I would first like to thank Dr. Rao V. Arimilli, who served as my major professor, for the many helpful suggestions he made during the course of this study.

This study was conducted under Union Carbide Corporation Subcontract No. 7685 S-17 funded by the U.S. Department of Energy through the Oak Ridge National Laboratory. Special thanks are extended to the project technical monitor Mr. John J. Tomlinson.

In addition I am indebted to Mr. Thomas P. Phillips for the use of his plotting program, and to Mr. Dave Johnson as well as Mr. Dennis Higdon who are the Mechanical Engineering Department's machine shop foreman and instrument shop technician, respectively.

Finally, I would like to thank my wife, Laura and children Camille and Rachel for the support and encouragement they gave me during my graduate studies.

ABSTRACT

The transient thermal performance of an experimental packed bed thermal energy storage system, (TES), installed in the two-phase flow laboratory of the Mechanical and Aerospace Engineering Department of the University of Tennessee at Knoxville, was determined. In this study the working fluids used were refrigerants-12 and -113. Two types of packings were used; single solid phase 5/8" diameter glass spheres, and 1/2" diameter 5/16" thick encapsulated pellets of phase-change material. The phase-change thermal energy storage materials tested were of two types: PWC-73, which is a eutectic mixture of Glauber's salt that has a specified phase-change temperature in the range of 71-75 °F, and PWC-89 which is Glauber's salt and has a specified phase-change temperature of 89 °F. Both single-and two-phase working fluid results are presented.

The storage unit used consists of a randomly packed 5" diameter 4' long tube. In the case of single-phase working fluid runs with glass packing, an energy balance was carried out and was found to agree with theoretical calculations to within 10 percent. The measured heat-transfer coefficient for single-phase liquid working fluid runs showed good agreement with published literature on gas-solid packed bed systems. Single-phase working fluid data were obtained for a working fluid Prandtl number in

the range of 6 to 8, and a modified Reynolds number in the range of 30 to 1200. Two-phase working fluid data were obtained for a bed inlet quality ranging from 0.3 to 0.9 at a refrigerant mass velocity of 23.5 pounds mass per minute per square foot.

TABLE OF CONTENTS

CHAPTER	PAGE
I. INTRODUCTION	1
Storage Assisted Heat Pumps	2
Study Objective	3
II. LITERATURE REVIEW.	4
Thermal Energy Storage.	4
Heat Transfer in Packed Bed Systems	4
Heat Transfer in Packed Beds of Solid Phase Materials	6
Gas Flow through Packed Beds of Solid Phase Materials.	6
Liquid Flow through Packed Beds of Solid Phase Materials.	10
Two-Phase Flow through Packed Beds of Solid Phase Materials.	10
Heat Transfer in Packed Beds of Encapsulated Phase Change Material (PCM). .	11
Gas Flow through Packed Beds of Encapsulated PCM.	11
Liquid Flow through Packed beds of Encapsulated PCM	11
Two-Phase Flow through Packed Beds of Encapsulated PCM	11
Fluid Flow in Packed Beds	11
III. DESCRIPTION OF THE EXPERIMENTAL TEST FACILITIES	15
IV. EXPERIMENTAL PROCEDURE	25
Single-Phase Working Fluid Runs	25

CHAPTER	PAGE
Two-Phase Working Fluid Runs.	26
V. RESULTS.	28
Experiments with Glass Spheres and a Single-Phase Working Fluid	28
Experiments with Glass Spheres and a Two- Phase Working Fluid	46
VI. CONCLUSIONS.	61
LIST OF REFERENCES	63
APPENDICES	68
APPENDIX A COMPUTER PROGRAMS	69
APPENDIX B PWC-89 AND PWC-73 EXPERIMENTAL RESULTS	94
APPENDIX C INSTRUMENTATION	124
APPENDIX D UNCERTAINTY ANALYSIS.	136
VITA	150

LIST OF TABLES

TABLE	PAGE
1. Summary of Experimental Packed Bed Heat Transfer Literature	12
2. Summary of Single-Phase Working Fluid Runs (R-113, Glass Spheres).	29
3. Summary of Two-Phase Working Fluid Runs (R-113, Glass Spheres).	47
4. Summary of Single-Phase Working Fluid Runs (R-113, PWC-73)	123
5. Low Flowrate Flowmeter Calibration Data (Water)	125
6. Low Flowrate Flowmeter Calibration Data (Gasoline).	127
7. High Flowrate Flowmeter Calibration Data (Water)	128

LIST OF FIGURES

FIGURE	PAGE
1. Typical Experimental Packed Bed Heat Transfer Correlations for Gaseous Working Fluids.	7
2. Schematic Layout of Test Facility.	16
3. Vertical Packed-Bed Test Section and Details . .	18
4. Probes Used to Determine Heat Transfer Coefficients	19
5. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 77$ (Flowrate 0.25 GPM).	30
6. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 300$ (Flowrate 1.0 GPM).	31
7. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 581$ (Flowrate 2.00 GPM)	32
8. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 1200$ (Flowrate 4.00 GPM).	33
9. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Copper Heat Flux Probe, $Re_m = 552$ (Flowrate 2.00 GPM)	34
10. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 59$ (Flowrate 0.25 GPM).	35
11. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 241$ (Flowrate 1.00 GPM)	36

12. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 474$ (Flowrate 2.00 GPM)	37
13. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 955$ (Flowrate 4.00 GPM)	38
14. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Copper Heat Flux Probe, $Re_m = 835$ (Flowrate 4.00 GPM).	39
15. Comparison of Measured Nusselt Numbers with Literature Values.	42
16. Typical Bed Inlet and Exit Pressures	45
17. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 1.0	49
18. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.89	51
19. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.74	53
20. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.61	54
21. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.48	55
22. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.27	56
23. Sketch of Phase-Change Salt Capsule	95

FIGURE	PAGE
24. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 2.0 GPM	97
25. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 0.5 GPM	98
26. Transient Measurement of Temperature Distribution, "Freezing" Run, Upflow 0.5 GPM	100
27. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 1.0 GPM	101
28. Transient Measurement of Temperature Distribution, "Freezing" Run, Upflow 1.0 GPM	102
29. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 4.0 GPM	103
30. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 0.25 GPM.	104
31. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 0.5 GPM	106
32. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 1.0 GPM	107
33. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 2.0 GPM	108
34. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 4.0 GPM	109
35. Typical Bed Inlet and Exit Pressures	111

FIGURE	PAGE
36. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 701$, (Flowrate 4.0 GPM)	113
37. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 354$, (Flowrate 2.0 GPM)	114
38. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 87$, (Flowrate 0.5 GPM)	115
39. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 44$, (Flowrate 0.25 GPM).	116
40. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 583$, (Flowrate 4.0 GPM)	117
41. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 296$, (Flowrate 2.0 GPM)	118
42. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 147$, (Flowrate 1.0 GPM).	119
43. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 78$, (Flowrate 0.5 GPM)	120
44. Comparison of Single-Phase Working Fluid Runs (Glass Spheres and PWC-73).	121
45 Low Flowrate Flowmeter Calibration Data Foxboro Flowmeter Model 1/2-2-81T3C1, S/N 125504.	130

NOMENCLATURE

A = area

ATM = atmospheric pressure

C = specific heat

D_p = diameter of glass sphere

D_h = hydraulic diameter of glass sphere = $2D_p\epsilon/3(1-\epsilon)$

E = energy

f = frequency

G = mass velocity = $\dot{m}/S_{fr}$

h = enthalpy

h = heat transfer coefficient

I = current

j = Colburn-j factor = $(h/CG)Pr^{2/3}$

K = flowmeter calibration constant

k = coefficient of thermal conductivity

L = length of the packed bed

l = length

m = mass

$\dot{m}$ = mass flow rate through packed bed

Nu = Nusselt number = hD_p/k_f

P = power

P = pressure

Pr = Prandtl number = $\mu C/k_f$

Q = volumetric flowrate through bed

Q_o = heat capacity of the bed packing in a given run
 $= \dot{m} \int_i^f C(T_{\text{exit}} - T_{\text{inlet}}) d\tau$ where i is the run start time
 and f is the run final time
 Q_l = theoretical capacity of the bed packing in a
 given run $= m_s C |T_{\text{final}} - T_{\text{initial}}|$
 Q_a = heat absorbed by bed packing
 Q_b = change in energy in bed walls
 Q_r = heat rejected from bed packing
 Q_l = rate of heat transfer (losses)
 Q_{rf} = energy removed from working fluid between the bed
 inlet and exit
 Q_s = change in energy of bed packing material
 Q_{sf} = change in energy of working fluid in the storage bed
 q = electrical power input to heat transfer probes
 q_l = rate of heat transfer (losses due to axial conduction
 from heat transfer probe electrical heating wire)
 R = electrical resistance
 Re_m = modified Reynolds number $= \rho_f D_h V / \mu_f$
 r = radius
 S_{fr} = total frontal cross-sectional area of packed bed
 T = temperature
 V = superficial fluid velocity $= \dot{m} / S_{fr}$
 V = electrical potential
 Z = dimensionless axial position in bed measured from the
 bed inlet $= l/L$
 W = uncertainty in result
 x_o = thermodynamic quality
 ΔP = differential pressure

ΔT = difference between heat flux probe surface
temperature and working fluid temperature
(Table 3)

Greek Symbols

ϵ = void fraction of packed bed

μ = coefficient of dynamic viscosity

ρ = density

τ = time

Subscripts

b = bed walls

f = fluid

fg = change by evaporation

g = gas

o = outside

o = inlet

i = inside

i = outlet

s = storage material

s = surface

t = total

w = bed wall

CHAPTER I

INTRODUCTION

Financial pressures and concerns over the dwindling world energy reserves have brought about the present day drive toward more efficient energy usage. Because a large fraction of the total energy consumed in buildings goes to space heating and cooling, there is a substantial effort being made to make residential and commercial heating and cooling more efficient.

Due to their relatively low initial cost and high performance capability, electrically powered heat pumps are growing in popularity. In both heating and cooling applications, heat pumps are contributing to the increase of utility electrical power demand. Many electrical utilities experience both yearly and daily cyclical power peaks; rather than have large amounts of excess nuclear and coal-fired generating power, some utilities meet these power peaks by operating more costly gas- and oil-fired generators. Substantial savings could be achieved if it were possible to make the power demand during each day more uniform by shifting some of the demand from "peak" to "off-peak" hours. Storage assisted heat pumps are promising candidates for shifting energy demand from "peak" to "off-peak" hours as a utility load management technique [2].

Storage Assisted Heat Pumps

Storage assisted heat pumps can be broken up into several subgroups depending on the type of storage material and storage bed geometry being used, the mode of coupling, and the working fluid being used to transport thermal energy to and from the storage unit. The storage material used in a thermal storage unit could be a single-phase liquid or solid such as water or crushed rock, or a phase change material, such as wax or Glauber's salt. Because of their high thermal capacities per unit volume phase-change storage materials are ideally suited for use in storage assisted heat pumps. Due to their high surface area-to-volume ratio, packed bed thermal storage units afford a larger heat transfer rate than do bulk storage units and are therefore more desirable. Typically a thermal storage unit can be coupled to a heat pump either in series or in parallel, with the latter being more versatile. Since phase change heat transfer coefficients are usually greater than single-phase liquid or gas heat transfer coefficients, a two-phase working fluid should further enhance heat transfer and would therefore be desirable.

Study Objective

This study focused on a thermal storage unit that was installed in the two-phase flow laboratory of the Mechanical and Aerospace Engineering Department at the University of Tennessee at Knoxville. The project was sponsored by the U.S. Department of Energy through the Oak Ridge National Laboratory. Design of the thermal storage unit was carried out at the University of Tennessee by Dr. R. V. Arimilli and Mr. A. G. Graves. Testing of the thermal storage system was performed during the January, 1983 to November, 1984 period.

The objectives of this study were two-fold. The primary objective of this study was to examine experimentally the performance of a thermal storage unit operating with either a single-phase or a two-phase working fluid. The second objective was to obtain data necessary to allow verification of computer models to be developed by others.

CHAPTER II

LITERATURE REVIEW

Thermal Energy Storage

Prior to the 1920's there was little or no theoretical work published in the scientific literature on the design and performance of heat recovery devices [3]. Most of the early research done on thermal energy storage was focused on sensible heat storage devices. To date sensible heat storage systems are the most widely used systems to store thermal energy [4]. Solar ponds, water tank thermal energy storage systems, and rock-bed storage systems are among the most common sensible heat storage systems used. In recent years work has been done on latent heat storage devices. Paraffin wax and salt hydrates are some commonly used, low-temperature, latent heat-storage materials.

Heat Transfer in Packed Bed Systems

As noted by Balakrishnan and Pei [5] the phrase, "Packed Bed heat Transfer," is currently used to describe a variety of phenomena namely: (1) the convective heat transfer from the walls of the packed bed to the fluid; (2) the convective heat transfer from the particle to the fluid flowing through the bed; (3) the conductive heat

transfer from the walls of the bed to the particles constituting the bed; (4) the conduction heat transfer between the individual particles in the bed; (5) radiant heat transfer and (6) heat transfer by mixing of the fluid. The total heat transfer coefficient, h_t , is a commonly used parameter to describe the thermal performance of packed bed storage units. The total heat transfer coefficient generally incorporates the conduction mode between the bed particles, the convection mode between the bed particles and the flowing medium, and the wall to fluid convection [5]. Therefore, the total heat transfer coefficient is a function of the thermal properties of the bed and the working fluid, and since the packed bed heat transfer correlations found in the literature do not typically include parameters to take into account the bed packing material and bed wall thermal properties, the applicability of a given correlation is usually limited to the particular bed used in the development of the correlation.

The most commonly used dimensionless numbers for expressing the total heat transfer coefficient h_t are the Colburn-J factor and the Nusselt number. These are defined as

$$j = \frac{h_t}{CG} (Pr)^{2/3}$$

and

$$Nu = \frac{h_t D_p}{k_f}$$

respectively.

As pointed out previously, the total heat transfer coefficient, h_t , is a function of the different modes of heat transfer involved and the thermal properties of the bed. Therefore as can be seen by examining Figure 1 which shows the experimental correlations of several authors, different experimental techniques and bed packing materials will yield wide variations in the heat transfer coefficient.

Heat Transfer in Packed Beds of Solid Phase Materials

Gas Flow through Packed Beds of Solid Phase Materials.

Due to its wide-spread application in the chemical process industries, heat transfer in packed gas-solid systems has been studied extensively. The most commonly used experimental techniques can be classified into four groups. Each of these is discussed below.

Transient response experiments. The use of transient response techniques in determining heat transfer coefficients was initiated by Schumann [6] when he theoretically investigated the effect of introducing a step change in temperature into a fluid at the inlet of a

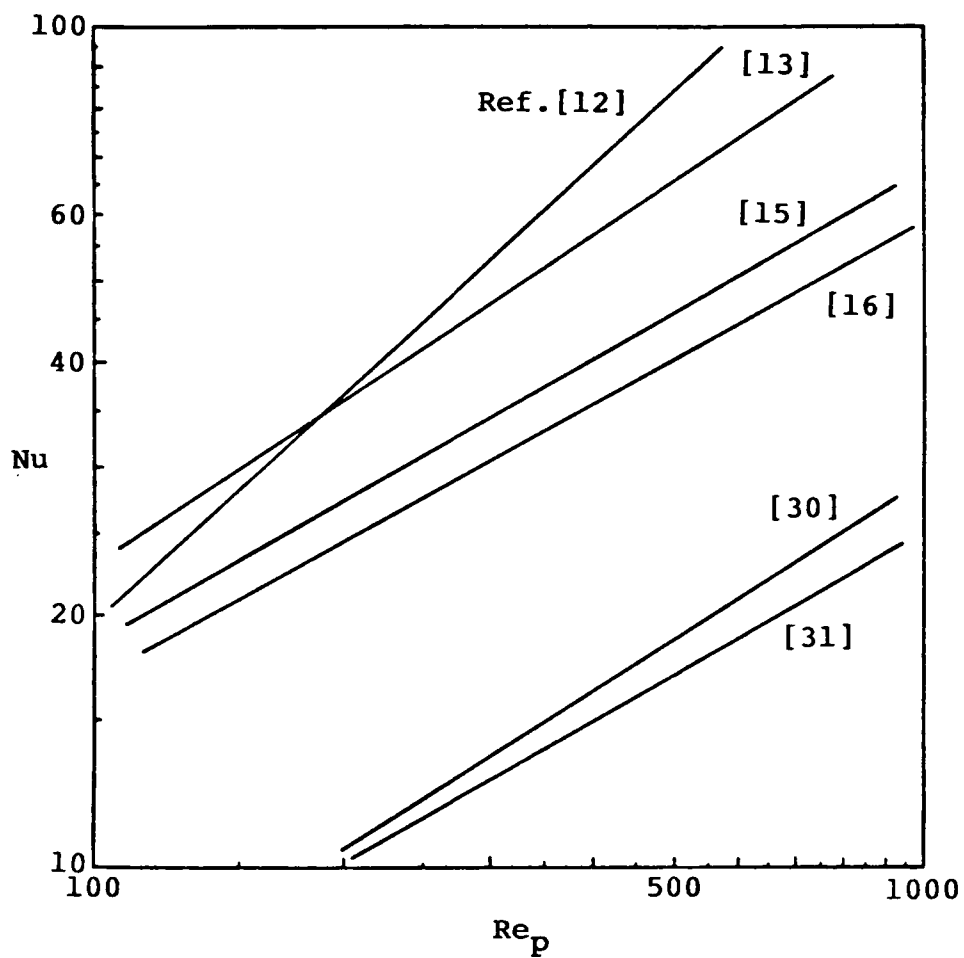


Figure 1. Typical Experimental Packed Bed Heat Transfer Correlations for Gaseous Working Fluids

packed bed. Lof and Hawley [7] used experimentally obtained packed bed exit temperature curves and Schumann's theoretically obtained bed exit temperature curves to obtain a packed bed heat transfer coefficient correlation. Many experimental investigations, using sinusoidally varying bed inlet-temperatures, have been carried out to determine packed bed heat transfer coefficients [8,9,10,11]. In these studies the change in amplitude and phase shift of the wave after passage through the bed is used to calculate the particle-to-fluid heat transfer coefficient. Typically, transient response techniques lead to bed-averaged overall heat transfer coefficients. One of the major advantages of transient response techniques is that the particle surface temperature need not be measured.

Dynamic calorimeter experiments. In dynamic calorimeter experiments, simultaneous fluid/solid temperature measurements are made during a transient, allowing the determination of the local heat transfer coefficient [11,12]. The measured change in solid temperature, assuming a lumped system, is used to calculate the rate of heat transfer. Heat transfer coefficients are based on the difference between fluid-solid temperature and the calculated heat transfer rate.

Bed heating experiments. In these experiments the steady state bed heat transfer rate and the difference

between particle surface temperature and working fluid temperature are used to calculate bed-averaged steady state heat transfer coefficients [14-18]. Specialized techniques such as particle microwave heating, particle electrical resistance heating, particle induction heating, and particle drying have been used and result in varying degrees of uniformity in particle temperature. In drying experiments the rate of heat transfer from the bed is based on the rate at which liquid is evaporated from the bed; in all other cases the working fluid mass flow rate and the difference between working fluid bed inlet and exit temperatures are used to calculate the rate of heat transfer from the bed.

Single particle heating experiments. Galloway and Sage [19] studied heat transfer from an electrically heated sphere in a packed bed of spheres. In this study an electrically heated copper sphere was provided with a small calorimeter which permitted measurements of local heat transfer as a function of polar and azimuthal angles. Another steady state experimental procedure that has been used to measure local heat transfer coefficients is the heating of a single particle in a packed bed with subsequent measurement of particle surface temperature and fluid temperature [20,21,22]. In these experiments the measured power input to the particle and the difference between particle surface temperature and the working fluid

temperature are used to calculate local steady state heat-transfer coefficients.

Liquid Flow through Packed Beds of Solid Phase Materials.

The most commonly used working fluid in packed bed heat transfer experimental research is air. Only one series of tests has been found in which the working fluid was a liquid. Baldwin [23] studied heat transfer from a single heated sphere, in a packed bed of spheres, using water as the heat transfer fluid.

Two-Phase Flow through Packed Beds of Solid Phase Materials.

No literature was found on heat transfer from packed beds to a two-phase working fluid. The most pertinent literature that was found involved studies of heat transfer from rod bundles to a two-phase working fluid [24,25], and a study on heat transfer from the outside surface of a single horizontal copper tube to a working fluid under nucleate boiling conditions [26].

Heat Transfer in Packed Beds of Encapsulated Phase Change Materials (PCM)

Gas Flow through Packed Beds of Encapsulated PCM.

No literature was found on heat transfer from packed beds of phase change material, (PCM), to a gas working fluid.

Liquid Flow through Packed Beds of Encapsulated PCM.

Wood, et al., [27] reported results of a small scale experimental packed bed of encapsulated wax spheres using water as the working fluid. The overall heat transfer coefficient for the bed was determined by matching the experimentally measured transient bed outlet temperature to that predicted by a computer model.

Two-Phase Flow through Packed Beds of Encapsulated PCM.

No literature was found on heat transfer from packed beds of encapsulated phase change material to a two-phase working fluid. A summary of the experimental packed bed heat transfer work that can be found in the literature is given in Table 1.

Fluid Flow in Packed Beds

Several experimental investigations have been undertaken to determine the mechanism of transport phenomena in gas-solid packed beds [32,33,19]. Mickley,

Table 1. Summary of Experimental Packed Bed Heat Transfer Literature

Ref.	Dimensions inch	Bed Voidage	D_t/D_p	Rep	Exp. Technique T=transient S=steady	Fluid Mat. Pr	material	Particle Shape	Dimensions inch
7	2.98 dia. x.66-1.16	0.39	36.5	25- 550	T step input	air .7	lead	sphere	.0818
8	1x1	.395,.410 .408 .354,.361	8-53	23- 18200	T frequency response sine wave	air .7	steel tungsten	sphere sphere	.03935,.069 .125 .019
9				1570- 3020	T frequency response sine wave	air .7	glass	sphere	.157,.169 .189
12	5.9 dia. x41.3		3-8	100- 4000	T dynamic calorimeter	air .7	iron	sphere	.728,1.24 1.91
13	15 dia. x48	.364	30	90- 660	T dynamic calorimeter	air .7	glass	sphere	.5
14	4 dia. x4	.354	11-26	200- 10400	S electrical induction	air .7	steel	sphere	.155,.2495 .3745
15	17/8 dia. x2	.417,.429 .477,.478 .436,.453	5-10	30- 3000	S electrical resistance heating	air .7 CO ₂ H ₂	steel steel brass	cylinder cube sphere	1/4,3/8 1/4,3/8 3/16,5/16
16	3.7 dia. x3.5	.428	6	13- 2136	S constant rate drying	air .7	celite	sphere	.626
17	2.0 dia. x1.75	.467,.392 .390	6-16	108- 833	S microwave heating	air .7	Fe ₂ O ₃ Fe ₂ O ₃	sphere cylinder	.126,.3 .2x.2
18				300- 4000	S microwave heating	air .7	comercial catalyst	sphere cylinder	
19	12 dia. x20	.2595	8	875- 3618	S sphere instrumented w/calorimeter	air .7	celluloid	sphere	1.5

Table 1. (Continued)

Ref.	Dimensions inch	Bed Voidage	D_L/D_p	Re_p	Exp. Technique T=transient S=steady	Fluid Mat. Pr	material	Particle Shape	Dimensions inch
20	6 dia. x20	.2595	3	800- 60000	S heat one sphere	air	wood	sphere	4
21	11x11 x12	.478	6	3000- 70000	S heat one sphere	H ₂ O	aluminum	sphere	1 5/16
22		.43			T step input	H ₂ O	PCM	sphere	.78

et al., [32] reported results from hot-wire anemometer measurements in a packed bed of rhombohedrally arranged spheres. In this study velocity and turbulence measurements were made in the voids between spheres and downstream from the packing. Turbulence measurements made in the central region of the packing voids showed intensity levels of 40 to 55 percent relative to the local mean velocity. No evidence of eddy shedding was found in the voids between particles. The evidence for or against eddy shedding behind the bed packing was inconclusive because the peaks in the frequency spectra were rather small. Another point of interest in this study is that although the bed voidage was kept constant by using half and quarter spheres at the wall, there was a 10 percent difference between the velocity measured in a void at the center of the bed and that measured in a void one particle diameter from the wall indicating that the wall effect in packed beds is only partly due to radial variations in voidage.

CHAPTER III

DESCRIPTION OF THE EXPERIMENTAL TEST FACILITIES

Figure 2 shows a diagram of the experimental test facility used in this study. This test facility was designed at the University of Tennessee as an add-on unit to an existing 180 degree bend burnout test facility and installed in the Two-Phase Flow Laboratory of the Mechanical Engineering Department at the University of Tennessee, Knoxville. All of the necessary machining, on the various components of this test facility, was performed by the Mechanical Engineering Machine Shop.

The nominal loop pressure can be adjusted by varying the amount of liquid in the loop; this is accomplished by contracting or expanding a stainless steel bellows within the pressurizer. The six-vane positive displacement pump used in the system can provide flow rates of up to 20 gallons per minute. Two turbine flow meters are used for flow rate measurements; one for high-flow rates and the other for low-flow rates. Calibration data for the flow meters used in this study can be found in appendix C. Three safety valves, each set at 10.2 atmospheres, have been installed on the system to relieve the system pressure should it exceed this value. The system working fluid can be heated in a vertical, stainless steel, 10-ft-long resistance heating section with a 25 KVA DC power

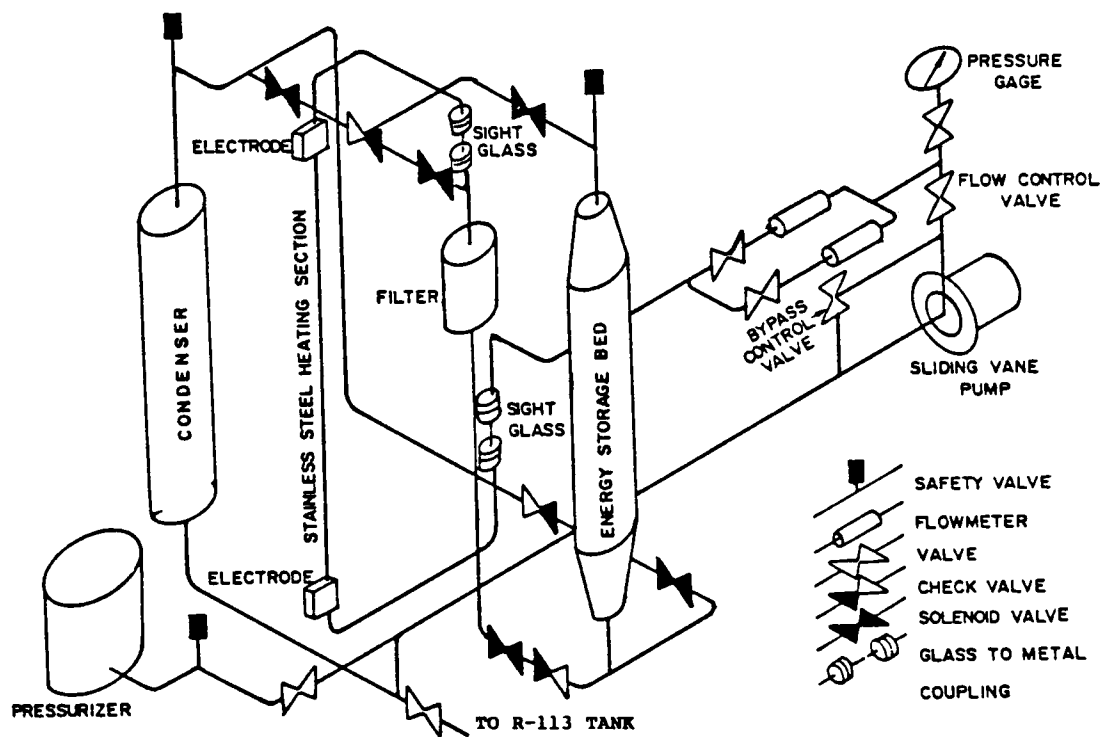


Figure 2. Schematic Layout of Test Facility

supply; both the heating length and power level are adjustable. The system working fluid can be cooled with building potable water in a single pass, vertical shell-and-tube heat exchanger. There are two sight glasses in the system; one at the exit of the flowmeters used to insure that flow through the flowmeters is always in the liquid phase, the other at the thermal storage bed inlet used to visually monitor the phase of the fluid entering the bed. The five solenoid valves in the loop allow for quick change of flow circulation from the by-pass loop to the energy storage bed and vice versa; either up-flow or down-flow through the vertical test section can be achieved.

Details of the vertical packed-bed test section used in this study are shown in Figure 3. Five ambient temperature probes were installed in the bed; these probes are labeled T1 through T5 in Figure 3. The distance between temperature probes is 1 foot with the first probe located at the bed inlet and the last probe located at the bed exit. Three electrically heated heat flux probes, labeled H-1, H-2, and H-3 in Figure 3, were installed in the bed to measure heat transfer coefficients. Figure 4 shows a diagram of the electrically heated heat flux probe along with the associated fluid temperature probe. These heat flux probes were installed at 1 foot intervals starting 6 inches down from the top of the bed. Each

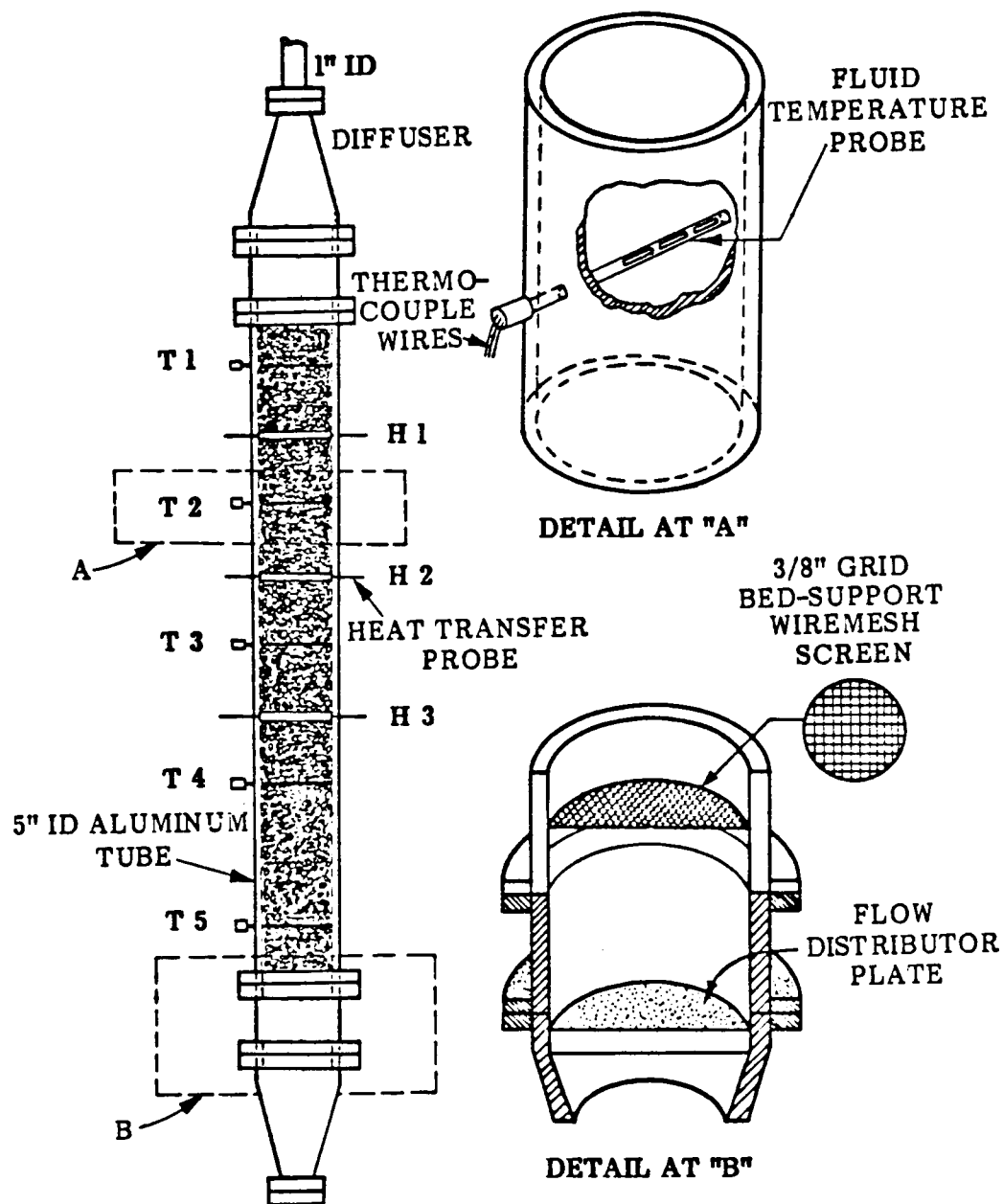


Figure 3. Vertical Packed-Bed Test Section and Details

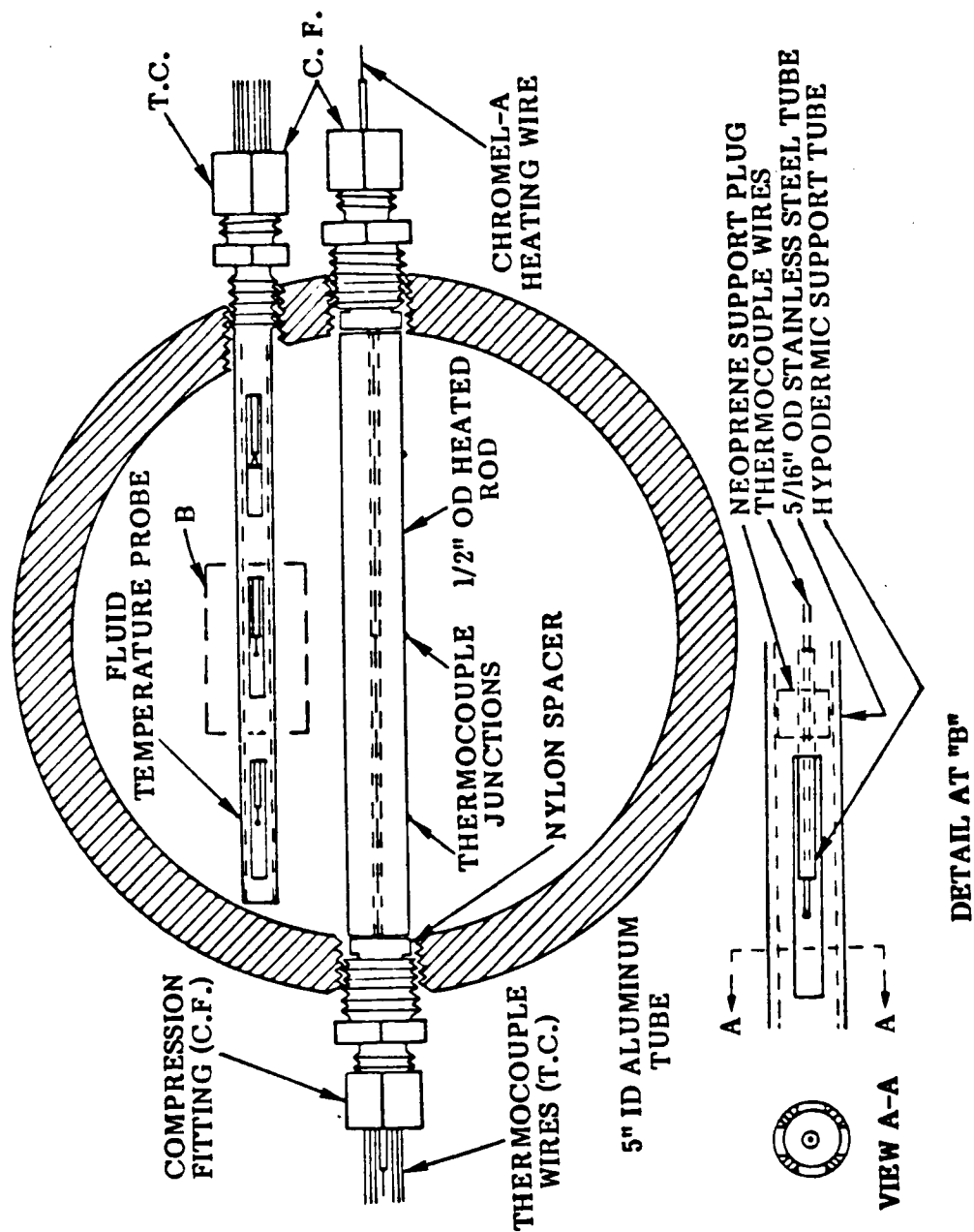


Figure 4. Probes Used to Determine Heat Transfer Coefficients

of these heat flux probes consists of a $1/2$ inch diameter, 5-inch-long, electrically heated rod instrumented on the surface with three 36 gauge copper-constantan thermocouples spaced one inch apart along the length of the rod. Both aluminum and copper heating rods were tried in order to investigate the influence of the rod thermal conductivity on the angular variations in rod surface temperature. A chromel-A wire .025 inch in diameter and five inches long, to match the length of the heating rod, was used as the electrical resistance wire to heat the probe. Copper lead wires of the same diameter were butt-welded to the two ends of the chromel-A wire. The heater wire and the lead wires were electrically insulated by a teflon sheath. A $1/16$ inch diameter hole was drilled through the center of the rod, and the wire with the teflon insulation was inserted into the hole such that the heater wire ends were aligned with the rod ends. The rod was then swaged to insure a tight fit and a good thermal contact between the heater wire, the insulation, and the rod. To measure the local fluid temperature, an ambient temperature probe was positioned parallel to each of the heater rods as shown in Figure 4. At each station in the bed, the axes of the two rods were in the horizontal plane separated by a distance of one inch between the two axes. Thus the two rods were positioned in a plane perpendicular to the direction of the flow such that one is not in the

wake of the other. At each station the average difference between the temperature of the surface of the heated cylinder and the ambient fluid was measured as a direct current emf.

To allow for pressure measurements, 1/8 inch Quick Connect pressure taps were installed at both ends of the test section. Calibration data for the differential strain gauge pressure transducers used in this study can be found in appendix C. Figure 4 shows a diagram of the working fluid temperature probes. All of the ambient fluid temperature probes, including those used in conjunction with heat transfer coefficient measurements, were of the same basic design. Each of the probes consisted of a 5/16 inch outer diameter stainless steel tube in which three thermocouples were installed at positions evenly spaced along the length of the tube. At each location where a thermocouple was placed, four 1/8 inch wide 1 inch long slots were cut at even angular increments around the periphery of the 5/16 inch tube. Each thermocouple was inserted into a 1-1/2 inch length of hypodermic size tubing with only the junction bead protruding from the end of the tube. A neoprene plug 1/4 inch thick by 3/8 inch in diameter was forced onto the hypodermic tube to form a concentric ring on the hypodermic tube. The neoprene plug assembly was then slid up into the 5/16 inch outer diameter tubing until the

thermocouple bead was centered along the length of the slots cut in the 5/16 inch tubing. This design insured that the thermocouple junction did not come in contact with either the bed packing material or the stainless steel support rod and was in direct contact with the working fluid. Further, the probe was oriented in such a way that two of the slots were parallel to the flow direction; this was done to ensure that the thermocouple senses the temperature of working fluid directly, rather than the temperature of the fluid in the wake of the probe.

To insure that the flow coming out of the conical diffuser and entering the packed bed was parallel and uniformly distributed, two one-inch thick, five-inch diameter aluminum distributor plates were installed--one each at the top and bottom of the bed. The two plates were identical, each with 476 through holes of 1/8 inch diameter drilled perpendicular to the plate surface by a numerically controlled milling machine.

To minimize heat losses, all of the tubing of the experimental test facility was insulated with 1-1/2 inch thick fiberglass insulation. The packed bed test section was insulated with one-foot thick fiberglass insulation. Thermocouples were placed at twenty locations on the inside and outside surface of the insulation to allow for loss calculations.

In order to acquire the time-dependent data from the thermocouples, a computer program was developed for digital data acquisition and processing on a system consisting of a H.P. 9826 Desk Top Computer, a H.P. 3497A Data Acquisition/Control Unit and a H.P. 2671G Printer. A listing of the programs used in this study is included in appendix A. In the two-phase flow runs, a Leeds and Northrup Trendscan model 1000 data acquisition system was used to measure insulation temperature along the electrically-heated section and other downstream tubing between the heated section and the packed bed. The measured inside and outside surface temperatures of the insulation were used to estimate losses so that accurate bed inlet quality calculations could be made.

The initial intent of this study was to evaluate the thermal performance of a thermal energy storage unit using encapsulated pellets of phase-change salt hydrates as the storage medium. Two types of salt hydrates were tested. The tests performed on the phase-change salts used in this study revealed that there were serious problems with the integrity of the encapsulation material used on the pellets. The organic resins used to encapsulate the salt pellets proved to be permeable to refrigerant resulting in a salt refrigerant mixture with unknown properties. Because of these problems, the results of these tests are included in appendix B as they can only be used for a

qualitative interpretation of the performance of PCM
packed beds.

CHAPTER IV

EXPERIMENTAL PROCEDURE

Single-Phase Working Fluid Runs

The heat transfer fluid used in these runs was R-113. The bed packing material used in these runs was 5/8 inch diameter smooth glass spheres. Both "heating" and "cooling" runs were made at flow rates ranging from 0.25 to 4 GPM. In the "cooling" runs the glass spheres were first heated by running hot refrigerant through the packed bed at a flow rate of 4 GPM for one hour to achieve a uniform initial bed temperature. Then the bed was isolated by closing the solenoid valves at the two ends of the bed, and the working fluid was then cooled to the desired bed inlet temperature in the by-pass loop prior to beginning a test-run. In all of the "cooling" runs the tube-and-shell heat exchanger coolant-water flow regulating valve was set to the full-open position. The power input to the working fluid by electrical heating was typically considerably higher than the rate of energy absorption of the packed bed. The excess energy input was removed in the tube-and-shell heat-exchanger. The purpose of this procedure was to reduce the time dependent fluctuation in bed inlet temperature. Typically during any given run, the bed energy absorption rate gradually decreased to zero

with increasing time resulting in a corresponding increase in bed inlet temperature. If the initial energy absorption rate of the bed is a large fraction of the total working fluid energy removal rate, then the bed inlet temperature rise during a run will be substantial. In most runs the working fluid flow direction was downward through the packed bed. At low flow rates, 0.5 GPM or less, natural convection effects were found to be important and it became necessary to use (1) up-flow through the packed bed to avoid working fluid flow reversals in "cooling" runs, and (2) down flow through the bed in "heating" runs. In "heating" runs the bed inlet diffuser was heated up to working fluid bed inlet temperature with a heat tape prior to the beginning of a run. This was done to achieve a steeper step in the bed inlet temperature curve; otherwise, the basic procedure followed in the "heating" runs was similar to that followed in the "cooling" runs.

Two-Phase Working Fluid Runs

The heat transfer fluid used in these runs was two-phase R-113. Six cases, each with a different bed inlet quality, were made with quality ranging from 0.24 to 1.0. In all of the two-phase working fluid runs, the working fluid flow direction was downward through the bed. Prior to beginning a two-phase run, the bed was brought up to

steady state by circulating two-phase refrigerant, of the desired inlet quality, through the bed. When the working fluid is under two-phase flow conditions, the heat transfer coefficient depends on the power level used to heat the heat flux probes. Therefore, for each two-phase run, data were recorded at three different power levels of the heat flux probes to determine the influence of the difference between rod surface temperature and working fluid temperature on the heat transfer coefficient. Typically, data were recorded for one hour at each power setting.

CHAPTER V

RESULTS

Experiments with Glass Spheres and a Single-Phase Working Fluid

Ten single phase working fluid runs were performed in this study. The working fluid mass flow rate ranged from 0.25 to 4 GPM. Both "cooling" and "heating" runs were performed. A summary of these runs is given in Table 2. In Figures 5 through 9 the measurements of the transient temperature distribution, energy absorbed, and heat transfer coefficients are presented for a bed initially at a low uniform temperature that is heated by the hot R-113 liquid. In a like manner, the measurements for a bed initially at a high uniform temperature that is cooled by cold liquid R-113 are presented in Figures 10 through 14. In all of these Figures, the axial location in the bed at which temperature measurements were made is given in dimensionless form Z , (Z = distance from bed inlet/length of bed). The energy absorbed and energy rejected curves shown in the following Figures are marked in the same manner as the temperature curves. The three locations at which heat transfer measurements were made correspond to Z = 0.125, 0.375 and 0.625 as shown in Figure 3, page 18.

Table 2. Summary of Single-phase Working Fluid Runs
(R-113, Glass Spheres)

Volumetric Flowrate (GPM)	Mass Flowrate (lbm/min)	Mode	Re _m	Nu	Q _O (Btu)	Q _L (Btu)	% difference $(1 - \frac{Q_L}{Q_O}) \times 100$
0.25	3.24	heating	77	34	547	434	26
1.0	13.01	heating	300	65	350	353	1
2.0	26.06	heating	581	98	430	434	1
4.0	51.89	heating	1200	157	334	342	2
2.0	26.34	heating	552	91	425	407	4
0.25	3.25	cooling	59	33	484	495	2
1.0	13.01	cooling	241	57	311	327	5
2.0	26.06	cooling	474	88	235	257	8
4.0	52.04	cooling	955	139	343	353	3
4.0	52.78	cooling	835	123	398	393	1

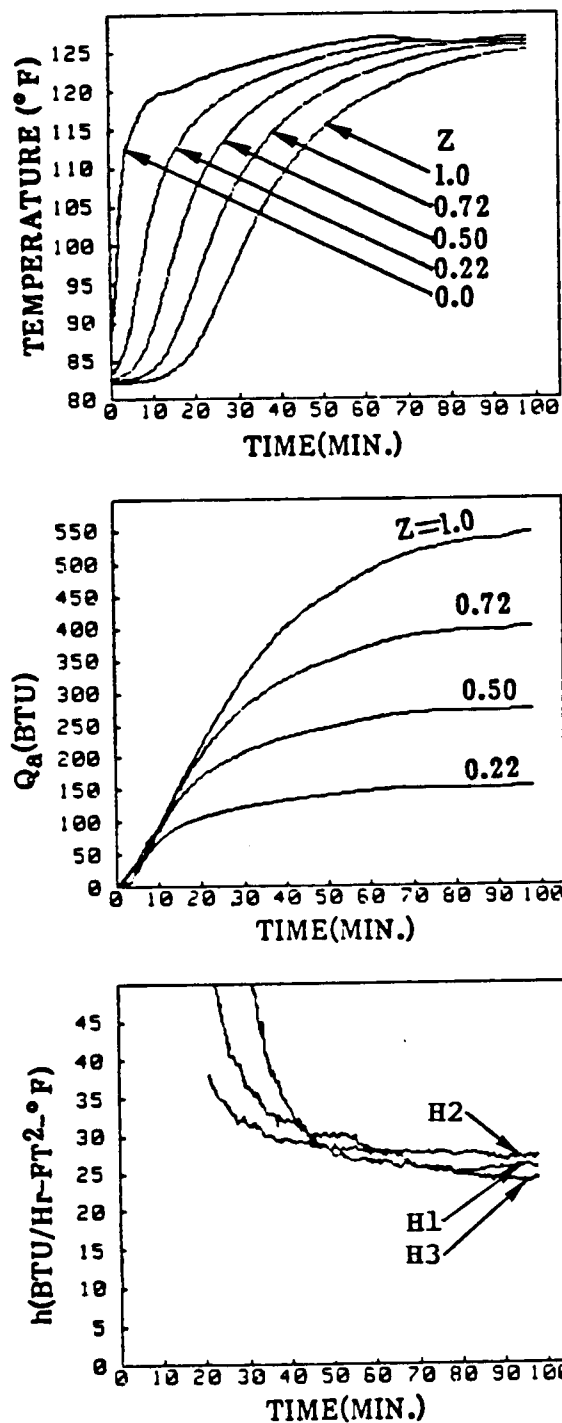


Figure 5. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 77$ (Flowrate 0.25 GPM)

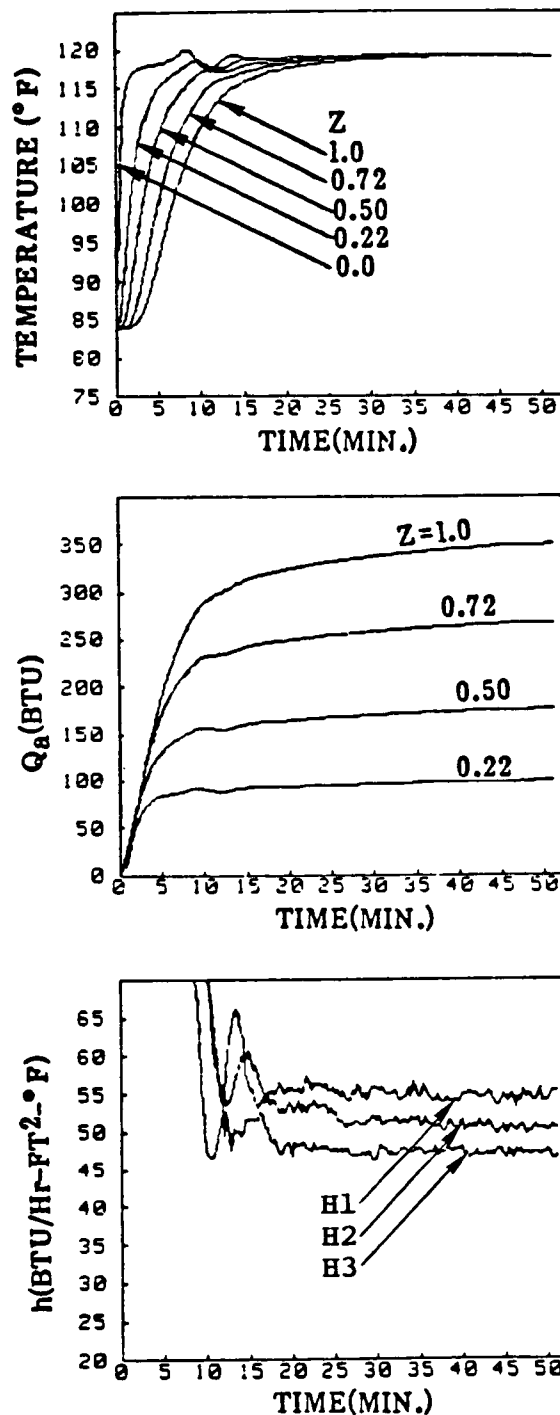


Figure 6. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 300$ (Flowrate 1.0 GPM)

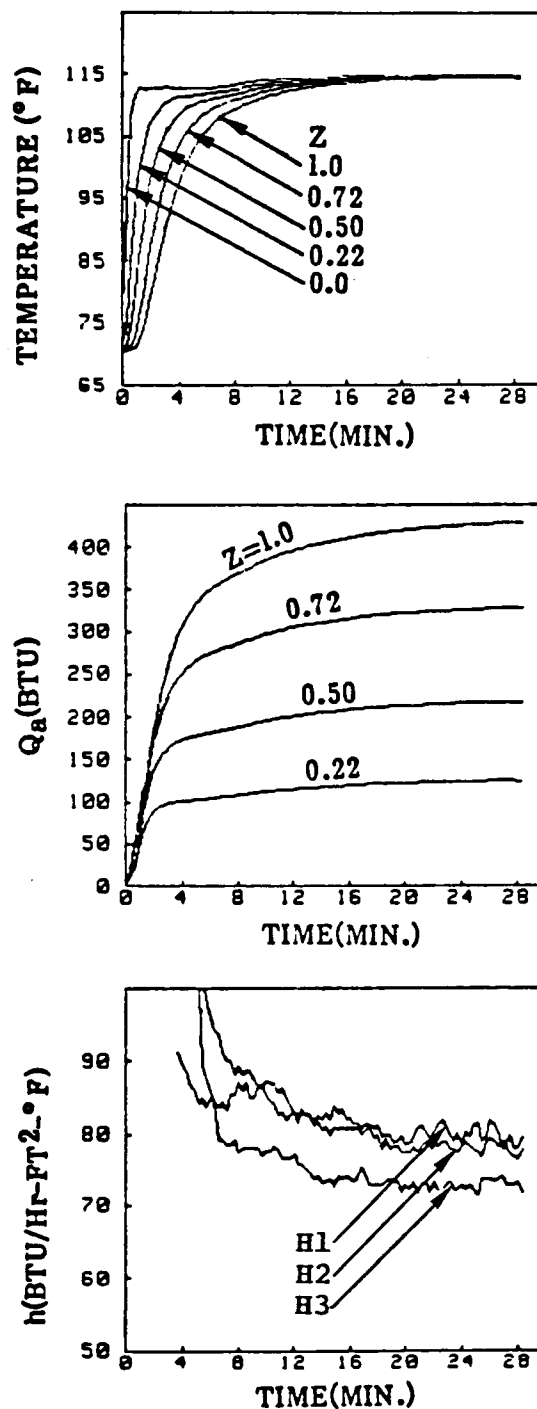


Figure 7. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 581$ (Flowrate 2.00 GPM)

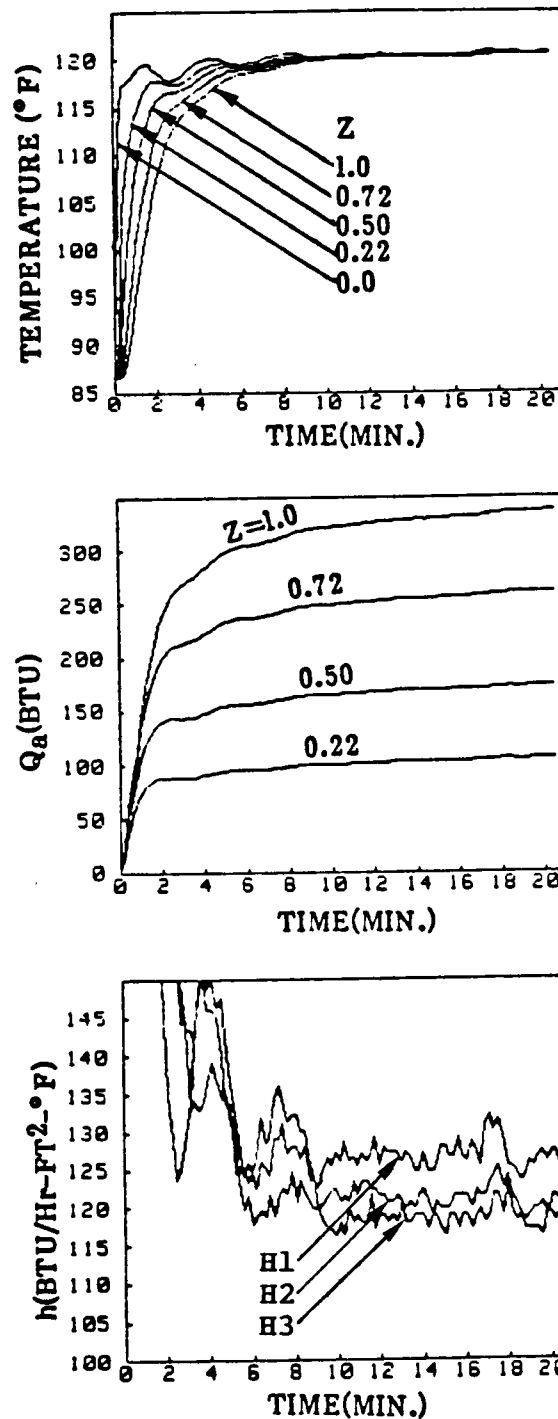


Figure 8. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Aluminum Heat Flux Probe, $Re_m = 1200$ (Flowrate 4.00 GPM)

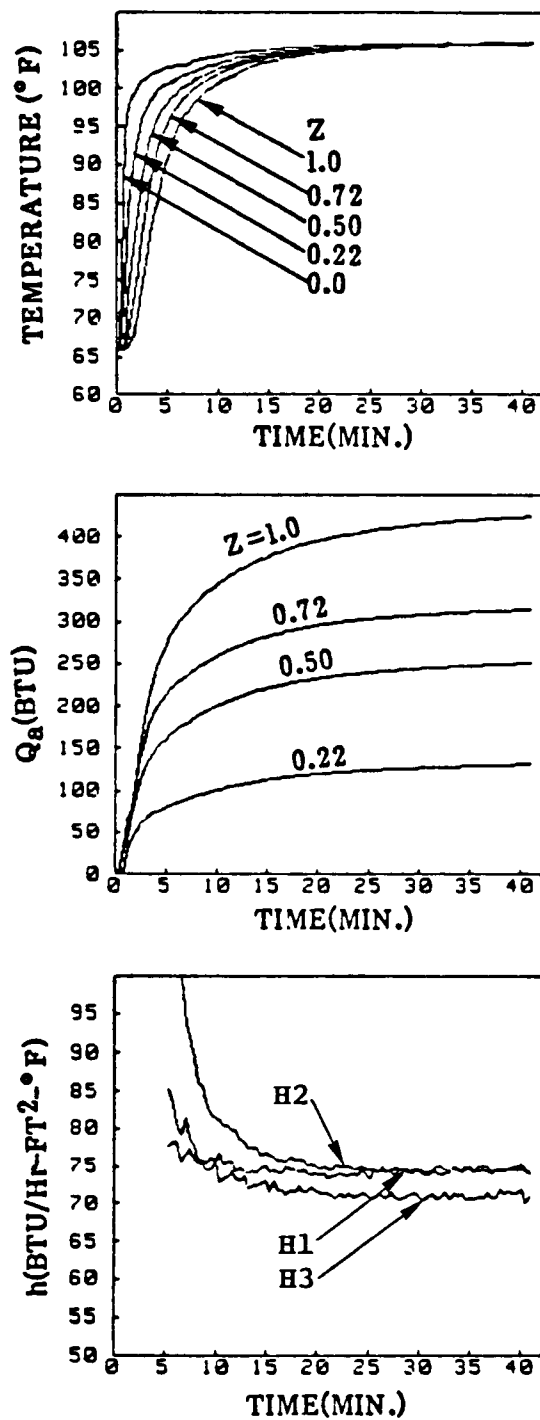


Figure 9. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, Heating Run, Copper Heat Flux Probe, $Re_m = 552$ (Flowrate 2.00 GPM)

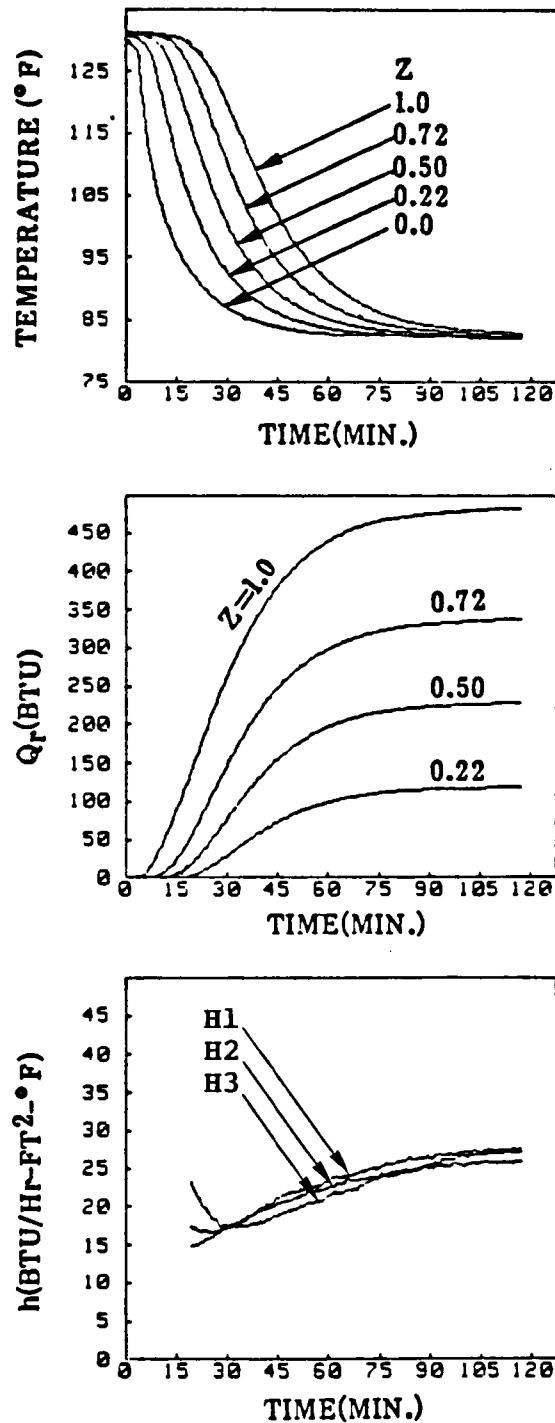


Figure 10. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 59$ (Flowrate 0.25 GPM)

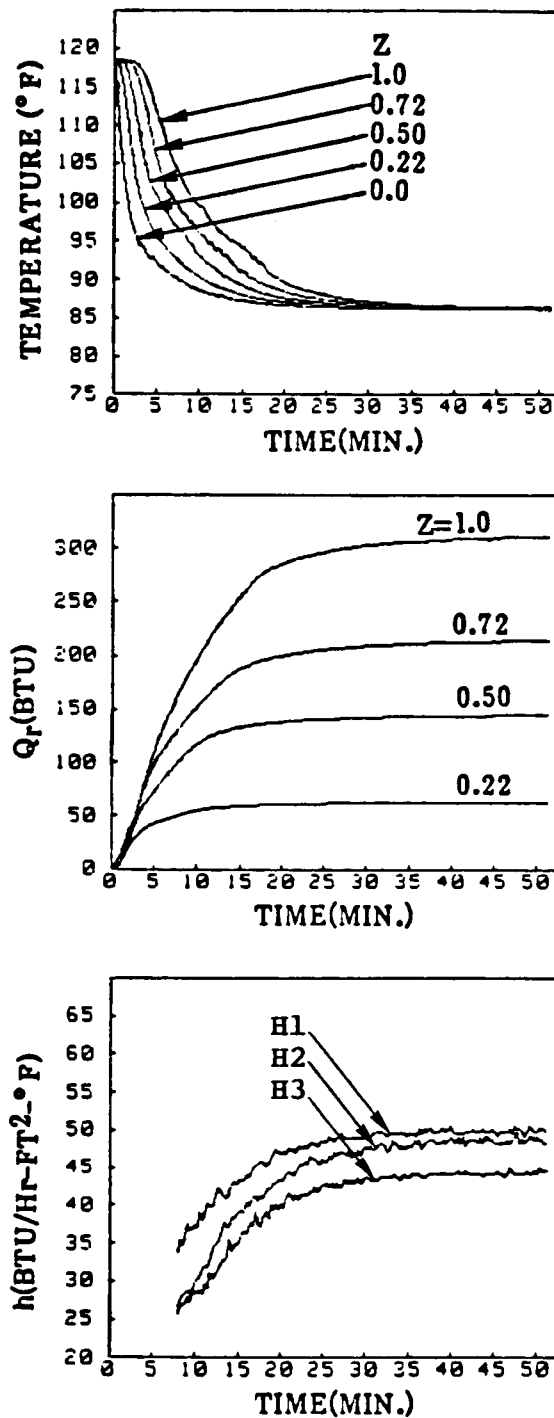


Figure 11. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 241$ (Flowrate 1.00 GPM)

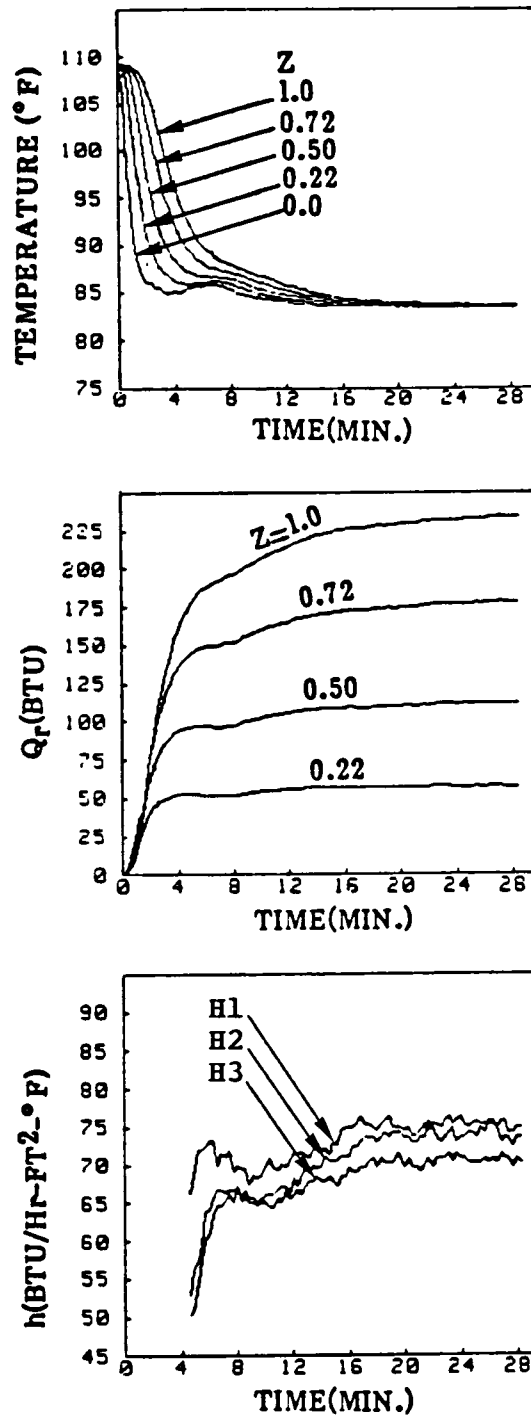


Figure 12. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 474$ (Flowrate 2.00 GPM)

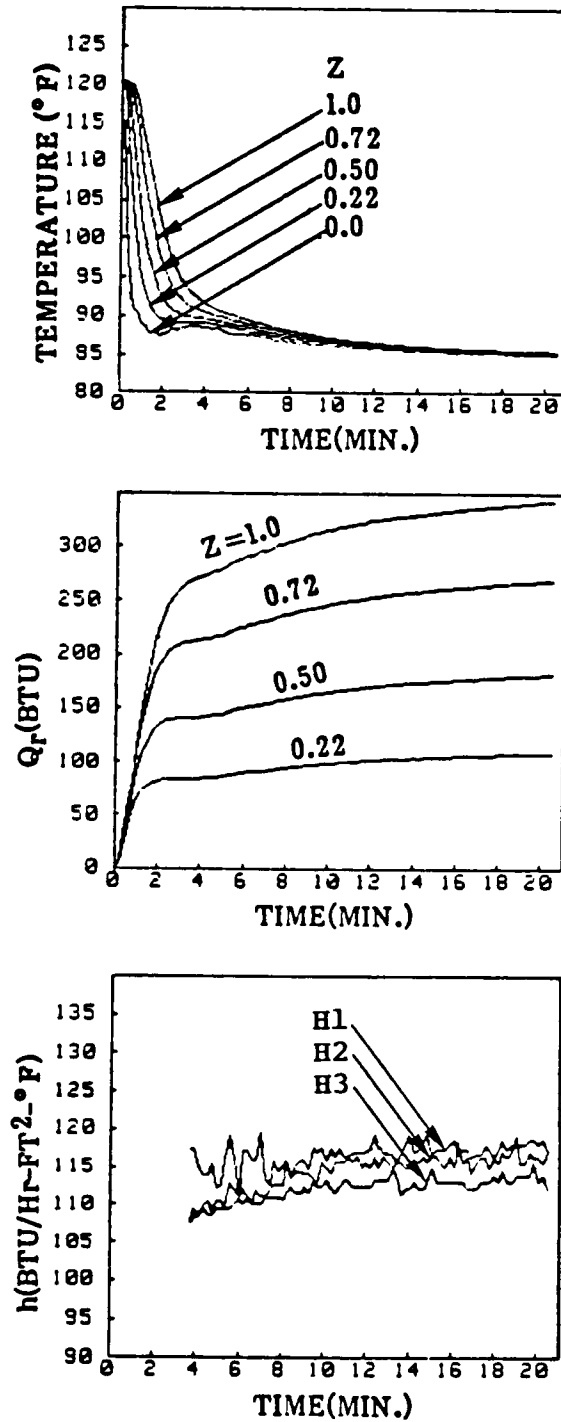


Figure 13. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Aluminum Heat Flux Probe, $Re_m = 955$ (Flowrate 4.00 GPM)

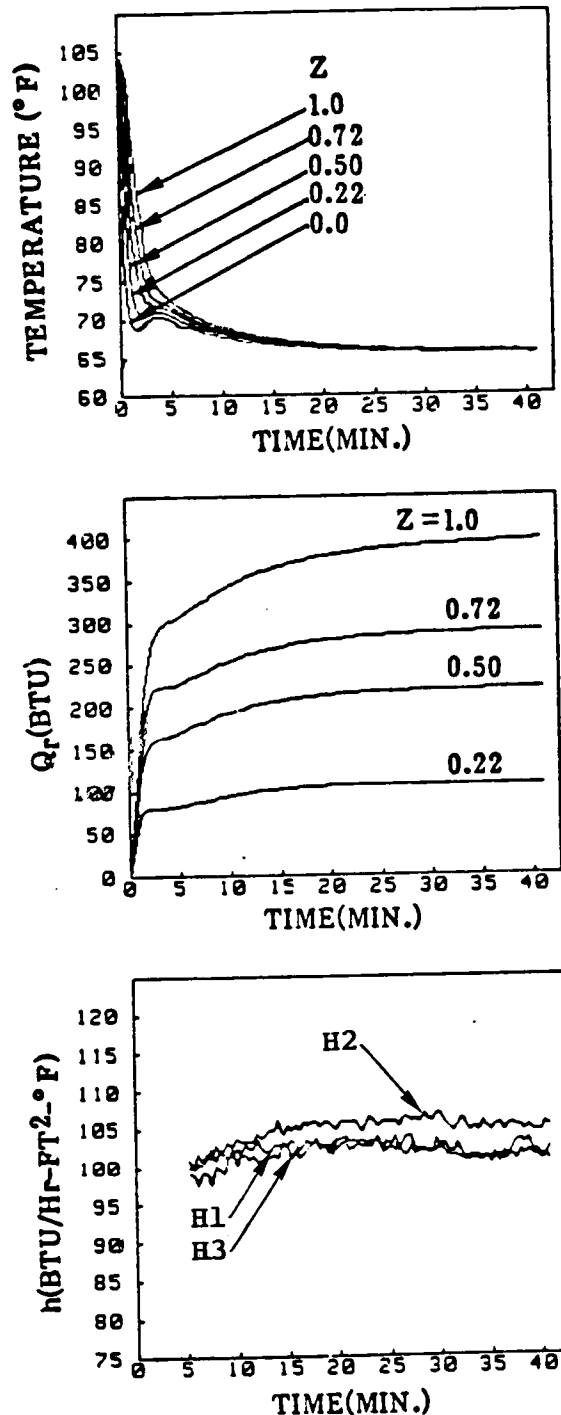


Figure 14. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, Cooling Run, Copper Heat Flux Probe, $Re_m = 835$ (Flowrate 4.00 GPM)

In all of these runs, once the initial transient associated with the heat flux probe is over, the three heat transfer coefficient values approach steady state values indicating steady conditions around the three probes. Typically in any one of these Figures the deviation in h values between the three axial locations in the bed is less than 5% of the mean of the three values; furthermore, there is no observable trend in the axial variation of h . From this it is concluded that h is independent of axial position. Therefore, for each case the average of the three values of h was used in the development of a Nusselt number correlation for this series of runs.

The heat absorbed (Q_a) and heat rejected (Q_r) curves, shown in Figures 5 through 14, pages 30-39, have been calculated from an integration of the fluid temperature history. The heat capacity of the aluminum wall and that of the working fluid were taken into account. The agreement between the measured heat capacity of the bed, Q_o , and the theoretical capacity of the bed, Q_1 , is generally good, (see Table 2). For all the runs except the 0.25 GPM "heated" run, the deviation between the measured and theoretical capacity of the bed is less than 10%. In the case of the 0.25 GPM "heated" run the deviation is 26%.

The measured heat transfer coefficients are presented in dimensionless form in Figure 15. The results of the heated and cooled runs are correlated in terms of Nu , Re_m and Pr . The correlation obtained by a least squares fit is given by

$$Nu = 2.45(Re_m)^{0.527}(Pr)^{0.174} \quad \text{for } 60 \leq Re_m \leq 1200$$

The above correlation is based on data obtained in this study for Prandtl numbers in the range between 6.8 and 8.0, and includes both "heating" and "cooling" runs. However, the data of Ref. [13] for a Prandtl number of 0.7 is also in good agreement with the above correlation (see Figure 15). Therefore, the above correlation can be considered to be valid for a Prandtl number range from 0.7 to 8.0. The standard deviation between the values of Nu measured experimentally, in this study, and those calculated using the above fit is 5.9. When comparing this correlation to other typical packed bed heat transfer correlations found in the literature it should be noted that this correlation is unique in that the exponent of the fluid Prandtl number was determined as part of the least squares fit rather than assuming a value of $1/3$ as is typically done.

The first eight runs in the above series were performed with aluminum heat flux probes. The last two

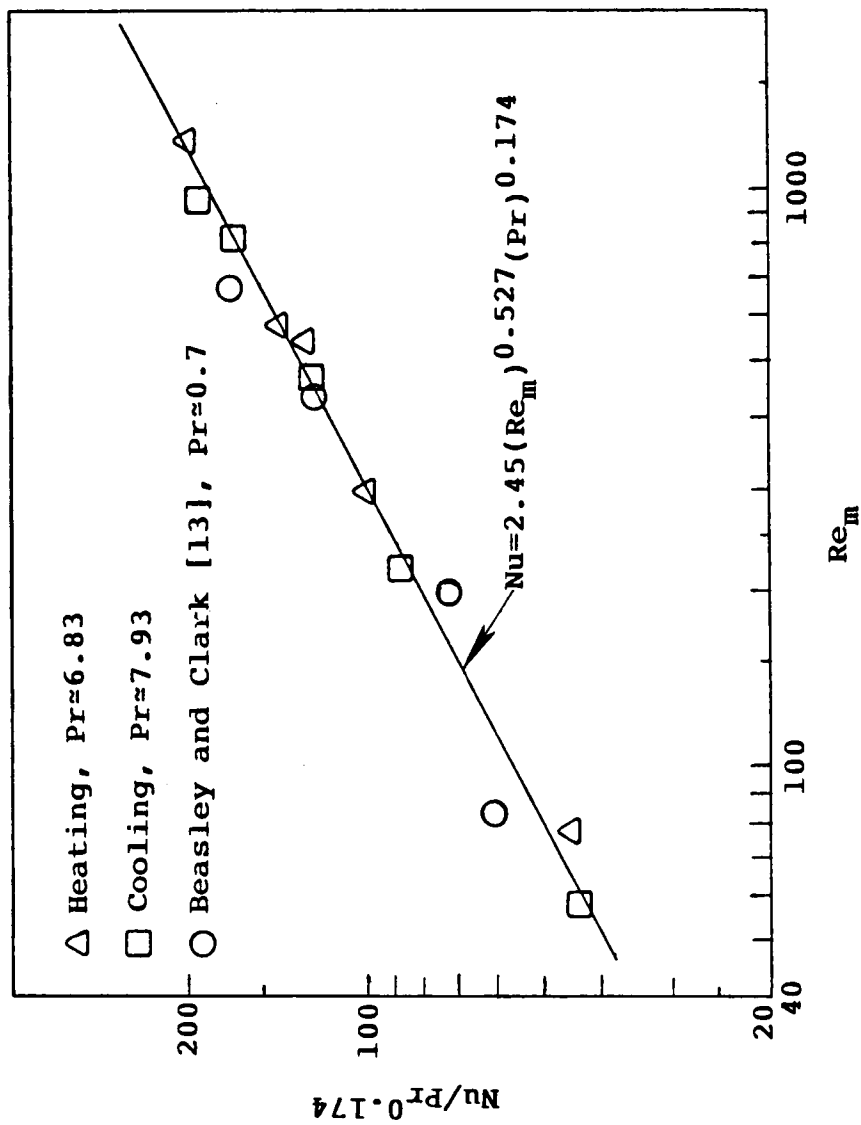


Figure 15. Comparison of Measured Nusselt Numbers With Literature Values

runs in this series were performed with copper heat flux probes; these latter runs correspond to Figures 9 and 14, pages 34 and 39. Figure 15 shows that when the heat transfer coefficients measured with the aluminum and copper heat flux probes are plotted in dimensionless form there is no measurable difference between the two types of probes. This indicates that the differences in h values that can be seen in runs using copper probes and those using aluminum probes are due to Reynolds number effects i.e., the working fluid temperature varied between runs resulting in variations in viscosity leading to variations in Reynolds number. These two types of heat flux probes, with metals of different thermal conductivity, were used to show that non-uniformities in probe surface temperature were negligible, and that the measured heat transfer coefficients represent accurate azimuthally-averaged packed bed heat transfer coefficients.

A comparison of the single-phase "heating" runs Figures 5-9, pages 30-34, to the single phase "cooling" runs Figures 10-14, pages 35-39, shows that the heat transfer coefficients for "heating" runs are time decreasing whereas the heat transfer coefficients are time increasing in "cooling" runs. These trends can be explained by examining the following equation which mathematically describes the transient behavior of the heat flux probes used in this study.

$$dE/d\tau = d(mc)/d\tau = Q_{\text{elect.}} - hA(T_s - T_f)$$

where E is the thermal energy of the probe, τ is time, m is the mass of the probe, C is the specific heat of the probe, $Q_{\text{elect.}}$ is the electrical power input to the probe, h is the probe to working fluid heat transfer coefficient, A is the surface area of the probe, T_s is the surface temperature of the probe and T_f is the working fluid temperature. For the case of a "heating" run the thermal energy of a given probe increases with time and therefore $dE/d\tau$ is positive. As time increases the probe tends toward a steady state temperature implying that $dE/d\tau$ tends toward zero as increasing time. Thus $Q_{\text{elect.}}$ is greater than $hA(T_s - T_f)$ and $hA(T_s - T_f)$ is time increasing. Assuming that h is constant this implies that $(T_s - T_f)$ is time increasing. The heat transfer coefficients presented in this study were calculated using the equation:

$$h = q/A(T_s - T_f)$$

Where q is the power input to the probe. Thus if $(T_s - T_f)$ is time increasing as argued above h will be time decreasing. Similarly in the case of "cooling" runs the heat transfer coefficients obtained with the probes used in this study will be time increasing during the transient portion of any given run.

Figure 16 shows typical single phase working fluid bed inlet and exit pressures. From this Figure it can be seen that the frictional pressure drop across the bed is small and that the difference between bed inlet and exit pressure is essentially hydrostatic.

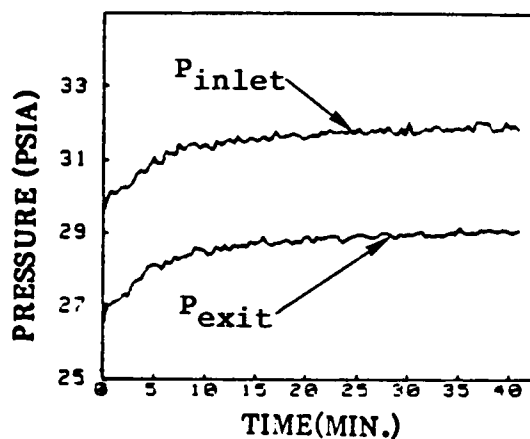


Figure 16. Typical Bed Inlet and Exit Pressures

Experiments with Glass Spheres and a Two-Phase Working Fluid

A total of five, two-phase working fluid runs and one, all vapor working fluid run were performed in this study. A summary of these runs is given in Table 3. In all of these runs the nominal working fluid mass velocity was 23.5 pounds mass per minute per square foot. The equilibrium quality of the working fluid at the bed inlet ranged from 1.0 to 0.27.

The surface area of packed beds is so large that even at low qualities most of the bed surface area is bathed in gas; that is to say there is no (or little) liquid film on the bed packing. The liquid flowing through the bed is in the form of droplets and rivulets. As the power input to the heat flux probes used in this study was increased the surface temperature of the probes and the heat flux per unit area of probe surface increased; thus liquid droplets striking the surface of a given probe evaporated at a higher rate. The net effect of this higher evaporation rate was to reduce the amount of probe surface that was operating in the two-phase regime. Since the magnitude of single and two-phase heat transfer coefficients are different the overall fluid to probe heat transfer coefficient was found to be a function probe power level when the working fluid was under two-

Table 3. Summary of Two-Phase Working Fluid Runs
(R-113, Glass Spheres)

Flowrate (lbm/min)	Quality	ΔT (°F)	h (Btu/hr-ft ² -°F)
3.25	1.0	5.1	40
3.25	1.0	13.2	24
3.25	1.0	32.9	21
3.26	0.89	2.3	192
3.26	0.89	4.4	167
3.26	0.89	11.2	130
3.26	0.74	2.1	207
3.26	0.74	3.8	191
3.26	0.74	7.6	179
3.26	0.61	1.9	241
3.26	0.61	3.4	213
3.26	0.61	6.8	193
3.25	0.48	1.8	254
3.25	0.48	3.2	230
3.25	0.48	6.0	222
3.25	0.27	1.5	300
3.25	0.27	2.8	264
3.25	0.27	5.3	246

phase conditions. Therefore, for each two-phase run, data were recorded at three different power levels of the heat flux probe to determine the influence of the difference between rod surface temperature and working fluid temperature on the heat transfer coefficient. In liquid flow runs one expects "h" to be independent of the heating level of the probes. This was confirmed in trial runs made prior to the two-phase runs.

Figure 17 shows the bed working fluid temperature profiles, the measured heat transfer coefficients, and the bed inlet and exit pressures for the case of a bed inlet quality of 1.0. The temperature measurements at the five instrumented stations in the bed indicate that local superheating of the working fluid occurred in the vicinity of the heated heat transfer coefficient probes. Bed inlet sight glass observations indicate that there was a small amount of liquid phase refrigerant entering the bed implying that the working fluid was in a non-homogeneous state. The heat transfer coefficients measured in this run are the lowest of this series of tests. The last thirty minutes of data recorded in this run were obtained at a low power setting of the heated heat transfer coefficient probe. During this portion of the run with bed inlet quality of 1.0, the difference between heat transfer probe surface temperature and ambient fluid temperature was approximately 5 °F. At this low power

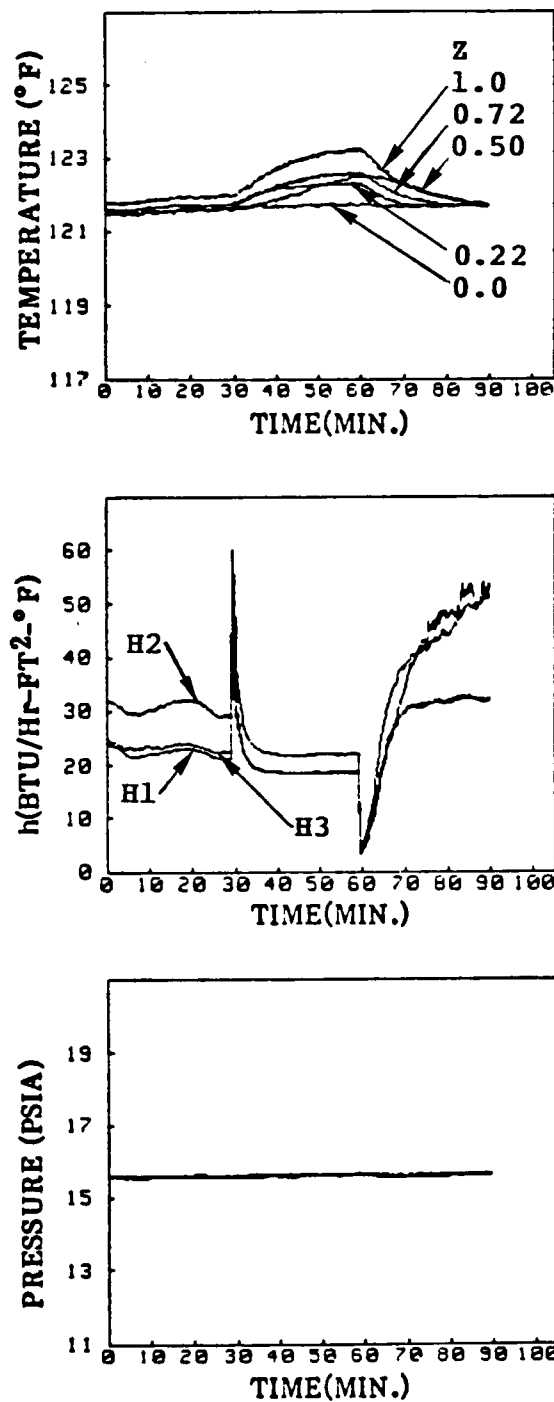


Figure 17. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 1.0

setting it can be seen that the measured heat transfer coefficients are higher and more oscillatory than those that were obtained at higher heat flux probe power settings during the first hour of this run. This indicates that the bed was operating in a two-phase regime during the last thirty minutes of this run, and that the bed was in a single-phase regime during the first hour of the run. The average difference between the working fluid temperature and the heat transfer probe surface temperature was 14 °F during the first thirty minutes of this run and 30 °F during the next thirty minutes of this run. However, the measured heat transfer coefficients are essentially the same, during these two time periods, indicating that the bed was operating in a single-phase gas regime with a heat transfer coefficient that is independent of ΔT . The pressure drop across the bed in this run was typical of that observed in this series of test. Figure 17 includes a trace of both the bed inlet and exit pressures; from this figure it can be seen that the pressure drop across the bed is very small.

Figure 18 shows the temperature, heat transfer coefficient, and pressure curves obtained with for a bed inlet quality of 0.89. From this Figure it can be seen that the difference between the bed inlet and exit temperature is very small indicating that the working fluid remained in the two-phase regime throughout the run.

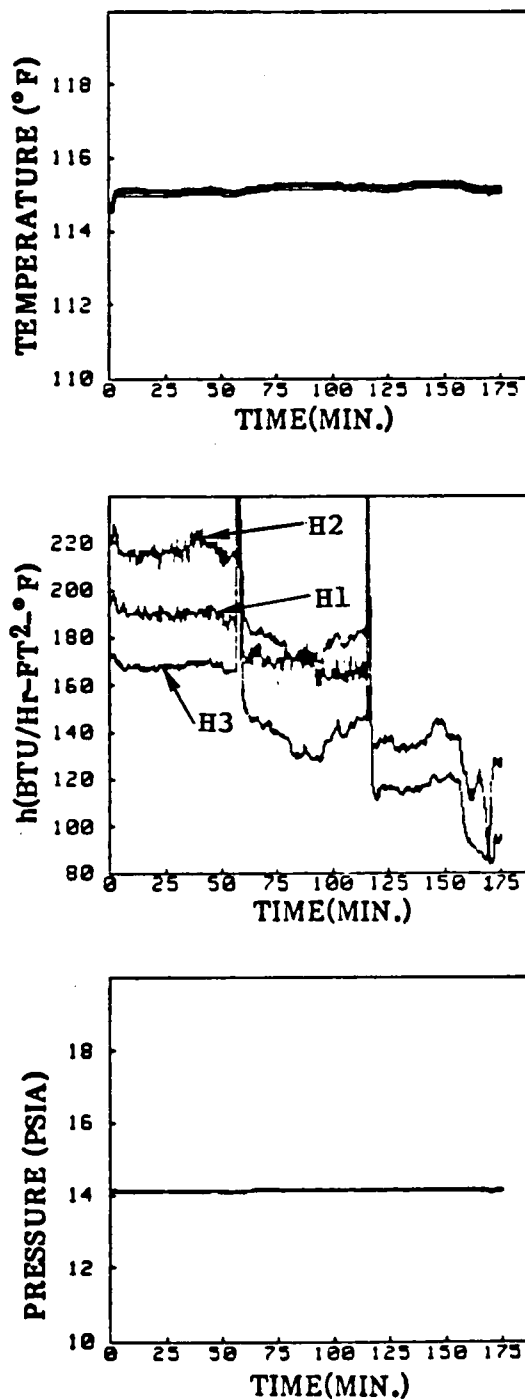


Figure 18. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.89

The difference between the heat flux probe surface temperature and the ambient fluid temperature, ΔT , was +2 °F during the first hour of this run, +4 °F during the second hour and +11 °F during the third hour. It should be noted that during the last hour of data acquisition only two of the three heat flux probes were operational. The third probe was turned off to allow an increased power input to the first two probes. When comparing this run to the 1.0 quality run of Figure 17, it can be seen that the oscillation in heat transfer coefficient and the influence of ΔT on heat transfer coefficient are much larger in the 0.89 quality run than they are in the 1.0 quality run. Further, it can be seen that the heat transfer coefficients are larger in this run than they were in the 1.0 quality run and that the heat transfer coefficient decreases with increasing ΔT .

Figure 19 shows temperature, heat transfer coefficient, and pressure data for the run with working fluid bed inlet-quality of 0.74. The temperature curves are similar to those of the 0.89 quality run. The oscillations in heat transfer coefficients have further increases as have the heat transfer coefficients.

The three other runs that were performed in this set of runs are shown in Figures 20 through 22; the working fluid bed inlet-qualities for these runs were 0.61, 0.48,

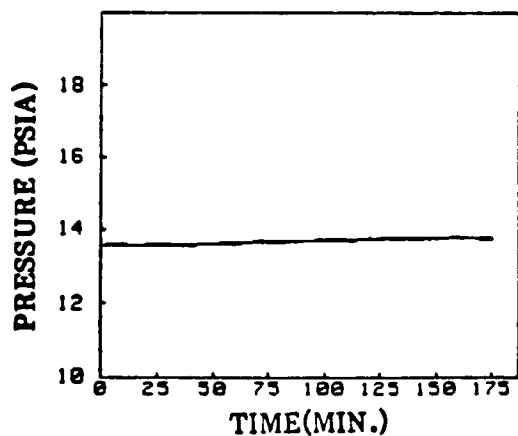
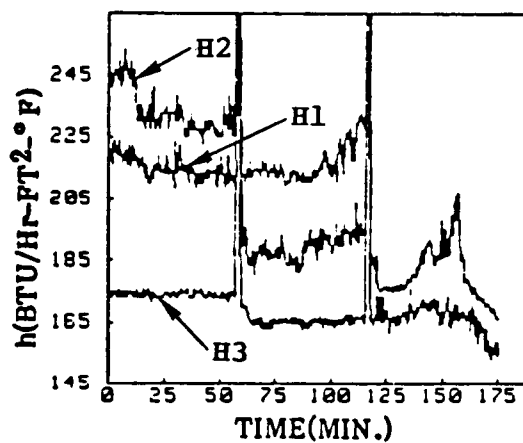
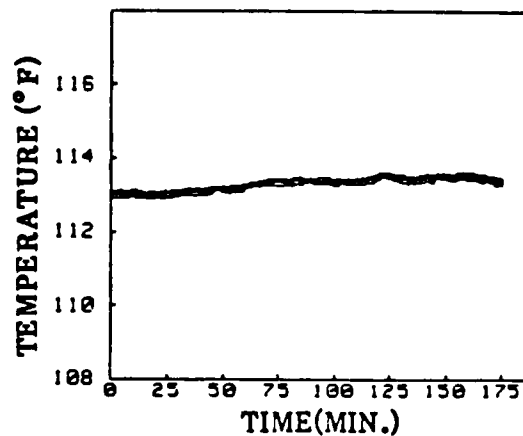


Figure 19. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.74

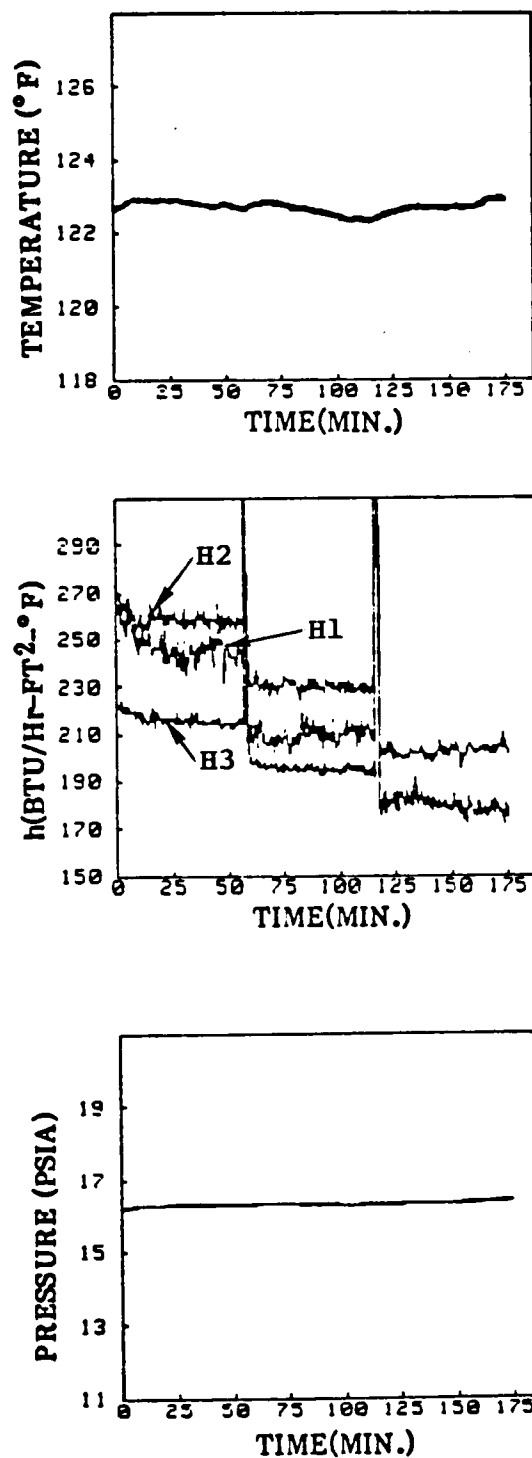


Figure 20. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.61

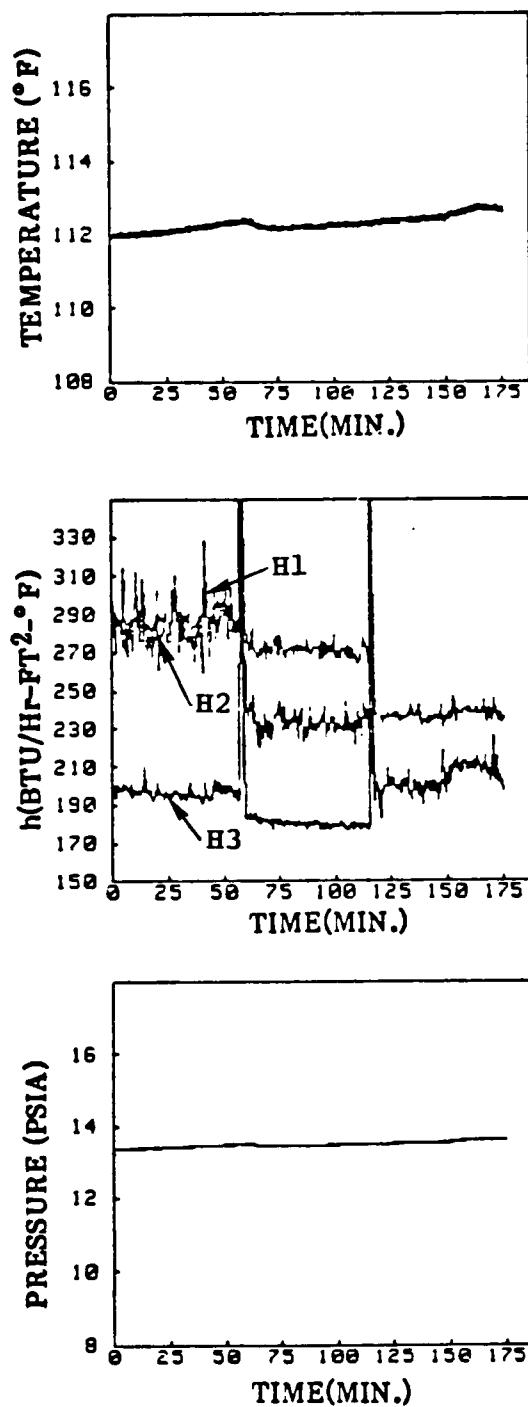


Figure 21. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.48

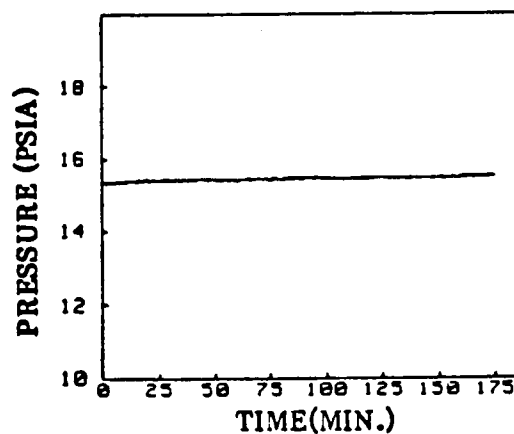
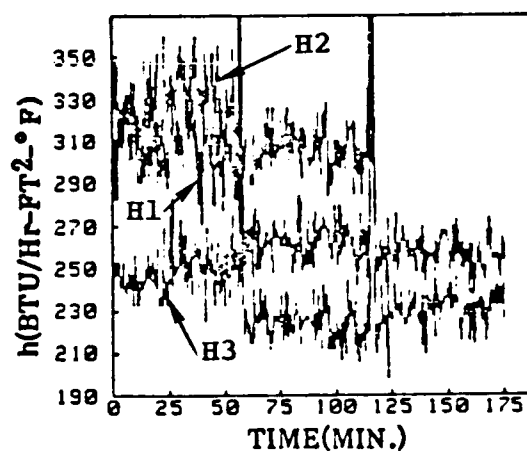
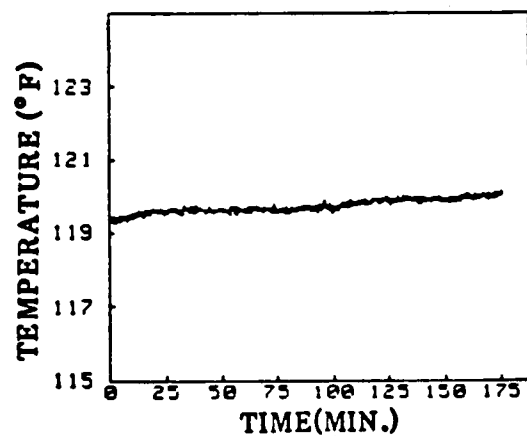


Figure 22. Transient Measurement of Temperature Distribution, Heat Transfer Coefficient, and Bed Inlet and Exit Pressures, Quality 0.27

and 0.27 respectively. The general trends in these runs are the same as those observed in the previous runs.

The two-phase working fluid heat transfer data obtained in this series of tests were correlated as a function of quality and ΔT in a least-squares fit, and the following correlation was obtained:

$$h_t = 262 \Delta T^{-0.455} x^{-0.654}$$

where h_t has units of Btu/hr-ft²-°F and ΔT is in °F. Since no two-phase working fluid packed bed heat transfer correlations were found in the literature no comparisons of the above correlation to other correlations were made.

The data obtained from the two-phase working fluid runs performed in this study indicate that the two-phase working fluid heat transfer coefficient in packed beds decreases with increasing quality and ΔT . Since increasing ΔT has the effect of increasing the quality, locally, these two observations are consistent; however, these are the opposite of what is typically observed in convective boiling. Typically in forced convection boiling in a tube the surface of the tube is covered by a film of liquid and heat transfer takes place from the tube wall to the liquid film; as the quality increases the liquid film becomes thinner and the fluid velocity and heat transfer coefficient increase until dry out and the

associated sharp drop in heat transfer coefficient occurs. In the case of packed beds the bed surface area is so large that even at very low qualities most of the bed surface area is bathed in gas; that is to say there is no (or little) liquid film on the bed packing. The liquid flowing through the bed is in the form of droplets and rivulets. As these droplets and rivulets flow over the bed packing surface some liquid evaporates causing a sharp drop in the local bed packing surface temperature. As the working fluid quality increases the number of droplets striking the bed packing surface decreases and this is the reason for the observed decrease in heat transfer coefficient oscillation with increasing quality. Furthermore, from the above description it can be seen that only a small fraction of the total bed surface is exchanging heat with a liquid undergoing phase change, most of the bed packing surface is exchanging heat with the vapor. As the working fluid quality increases the fraction of the bed that is operating in a two-phase mode decreases, and this is the reason for the observed decrease in heat transfer coefficient with increasing quality. It should be noted that no condensing flow experiments were performed in this study.

In each of the two-phase runs, the heat transfer coefficient values indicated by the three probes were within 30% of their mean value. In single-phase runs,

the three were within 5% of their mean value. This latter fact led to the conclusion that "h" in the bed is essentially independent of axial position in the bed. The power input to the heat flux probes during the two-phase flow runs was relatively small such that the change in working fluid quality between the bed inlet to exit was on the order of 0.01. Thus the relatively large variations observed in heat transfer coefficients at the three axial locations during each of these runs cannot be attributed to an axial variation in working fluid quality.

As mentioned in CHAPTER III, (see Figure 4, page 19), at each location where a heat flux probe was installed in the bed, three direct measurements of the difference between the heat flux probe surface temperature and the ambient fluid temperature were made. By observing the three differential temperature measurements of a given heat flux probe, during a low quality two-phase run, it was found that the probe surface temperature was highly non-uniform. Differences as high as 50% were observed between the three temperature measurements, (in single-phase runs the difference between any two of the three temperature measurements was typically less than 5%). Since each of the three thermocouples was located at the same angular position on the surface of the heat flux probe, (relative to the flow direction), the difference between the three measurements cannot be attributed to

angular variations in heat transfer. The heat flux probes used in this study are such that an accurate area-averaged surface temperature must be obtained in order to calculate accurate heat transfer coefficients. Thus the heat flux probes used in this study will yield more accurate results under single-phase flow conditions.

CHAPTER VI

CONCLUSIONS

Accurate transient temperature profiles for a packed bed of glass spheres have been presented. Both single and two phase working fluid heat transfer correlations have been presented. The transient single phase temperature profiles obtained can be used to validate numerical packed bed computer models. The heat transfer correlations obtained can be used to design and mathematically model packed bed thermal energy storage systems. It should be noted, however, that the heat transfer data presented in this study was obtained with the use of cylindrical heat flux probes. Since the bed packing material used in this study was essentially spherical in shape the use of cylindrical heat flux probes to make heat transfer measurements is obviously an approximation.

Further studies to clearly establish the influence of Prandtl number on single-phase working fluid packed bed heat transfer, although needed, would be far less valuable than studies involving two-phase flow through packed beds. Investigations on fluid flow regimes and the influence of the bed wall on the vapor-liquid distribution in two-phase flow through packed beds are needed to properly model the heat transfer phenomenon of two-phase working fluids through packed beds.

Due to the problems encountered with the phase change salt capsules used in this study it was not possible to obtain the necessary data to properly evaluate the potential of phase change salts for use in in-line, storage-assisted heat pump, thermal storage units.

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APPENDIXES

APPENDIX A

COMPUTER PROGRAMS

The following is a listing of the data acquisition and reduction program used in this study:

```
10      !   PACKED BED
20      !
30      !   THIS PROGRAM TAKES TEMPERATURE PRESSURE
40      !   AND HEAT TRANSFER COEFFICIENT DATA
50      !
60      !*****
70      !*
80      !*           INITIALIZE PROGRAM
90      !*
100     !*****
110     !
120     OPTION BASE 1
130     DIM Com$(8)[80]
140     INPUT "FILE NAME",Com$(1)
150     INPUT "TO TAKE DATA ENTER <TAKE> TO OUTPUT <OUTPUT>
      ",Op$
160     INPUT "FLOW DIRECTION",Fldir$
170     IF Op$="TAKE" THEN
180     INPUT "NUMBER OF SCANS",Scc
190     END IF
200     ON ERROR GOTO 310
210     Filedel1$="H"&Com$(1)
220     Filedel2$="QA"&Com$(1)
230     Filedel3$="P"&Com$(1)
240     Filedel4$="T"&Com$(1)
250     Filedel5$="TD"&Com$(1)
260     PURGE Filedel1$
270     PURGE Filedel2$
280     PURGE Filedel3$
290     PURGE Filedel4$
300     PURGE Filedel5$
310     OFF ERROR
320     IF Op$="OUTPUT" THEN
330     Numscan$="SCC"&Com$(1)
340     ASSIGN @Path TO Numscan$
350     ENTER @Path;Scc
360     Scc=Scc-1
370     END IF
380     ALLOCATE Q1(1:Scc),Q2(1:Scc),Q3(1:Scc),Q4(1:Scc),
      Q5(1:Scc)
```

```

390  ALLOCATE Dtc1(2:Sc),Dtc2(2:Sc),Dtc3(2:Sc),
      Dtc4(2:Sc)
400  ALLOCATE Dts1(2:Sc),Dts2(2:Sc),Dts3(2:Sc),
      Dts4(2:Sc)
410  ALLOCATE Ql1(1:Sc),Ql2(1:Sc),Ql3(1:Sc),Ql4(1:Sc)
420  ALLOCATE Qa1(1:Sc),Qa2(1:Sc),Qa3(1:Sc),Qa4(1:Sc)
430  ALLOCATE Ef1(1:Sc),Ef2(1:Sc),Ef3(1:Sc),Ef4(1:Sc)
440  Sc=Sc+1
450  ALLOCATE T(Sc,5,3),Ti(Sc,5),Pr1(Sc),Pr2(Sc),
      Th(Sc,3,4),Tih(Sc,3)
460  ALLOCATE Tis(Sc,5,2),Dtc(Sc,4)
470  ALLOCATE Tla(Sc),T2a(Sc),T3a(Sc),T4a(Sc),
      T5a(Sc),Tli(Sc),T2i( Sc),T3i(Sc),T4i(Sc)
480  ALLOCATE Tlha(Sc),T2ha(Sc),T3ha(Sc),Tlhi(Sc),
      T2hi(Sc),T3hi(Sc),T5i(Sc)
490  ALLOCATE Tlad(Sc),T2ad(Sc),T3ad(Sc),T4ad(Sc),
      T5ad(Sc),Dd(Sc,5 ),Tw(Sc,5)
500  ALLOCATE Hh1(2:Sc),Hh2(2:Sc),Hh3(2:Sc),
      Hh4(2:Sc),Hh5(2:Sc)
510  !
520  !*****
530  !*
540  !*          OPEN FILES
550  !*
560  !*****
570  !
580  Filesize1=Sc*57+6
590  Filesize2=Sc*10
600  Filesize3=Sc*10
610  Filesize4=Sc*4
620  Filesize5=Sc*6
630  Filesize6=Sc*8
640  A$="T"&Com$(1)
650  B$="TD"&Com$(1)
660  C$="P"&Com$(1)
670  D$="H"&Com$(1)
680  E$="QA"&Com$(1)
690  F$="C"&Com$(1)
700  IF Op$="TAKE" THEN
710  CREATE BDAT Com$(1),Filesize1,8
720  CREATE BDAT F$,8,80
730  END IF
740  CREATE BDAT A$,Filesize2,8
750  CREATE BDAT B$,Filesize3,8
760  CREATE BDAT C$,Filesize4,8
770  CREATE BDAT D$,Filesize5,8
780  CREATE BDAT E$,Filesize6,8
790  IF Op$="OUTPUT" THEN GOTO 2750
800  Numscan$="SCC"&Com$(1)
810  CREATE BDAT Numscan$,1,8
820  ASSIGN @Path TO Numscan$

```

```

830  OUTPUT @Path;Scc
840  INPUT "DATE",Com$(2)
850  INPUT "BAROMETRIC PRESSURE INHG",Atm
860  Atm=Atm*.489770685      ! CONVERT TO PSI
870  INPUT "THERMAL STORAGE MATERIAL",Com$(3)
880  INPUT "FLOWMETER S=LOW-RANGE L=HIGH-RANGE",Com$(4)
890  INPUT "FLOW DIRECTION <UP> <DOWN> ",Com$(6)
900  INPUT "COMMENTS",Com$(5)
910  INPUT "VOLTAGE",V1
920  INPUT "AMPERAGE",A1
930  INPUT "SCAN INTERVAL TIME",Dt
940  Com$(6)="C"&Com$(1)
950  !
960  !*****
970  !*
980  !*          ENTER INITIAL SYSTEM STATUS          *
990  !*
1000 !*****
1010 !
1020 CLEAR 709
1030 !
1040 !   ENTER FLOW-METER INLET FLUID TEMPERATURE
1050 !
1060 OUTPUT 709;"AC39VR1SO0"
1070 ENTER 709;I2
1080 OUTPUT 709;"AC38VR1SO0"
1090 ENTER 709;Tfs
1100 Tfs=(Tfs-I2)*1000
1110 !
1120 !   ENTER PRESSURES
1130 !
1140 OUTPUT 709;"AC40VR1SO0"
1150 ENTER 709;Pr1(1)
1160 OUTPUT 709;"AC41VR1SO0"
1170 ENTER 709;Pr2(1)
1180 !
1190 !   ENTER ICE BATHS
1200 !
1210 OUTPUT 709;"AC19VR1SO0"
1220 ENTER 709;I1
1230 OUTPUT 709;"AC39VR1SO0"
1240 ENTER 709;I2
1250 !
1260 !*****
1270 !
1280 !           SCAN INITIAL BED TEMPERATURES
1290 !
1300 OUTPUT 709;"AF0AL14VR1SO0"
1310 I=1
1320 Ib=I1
1330 FOR J=1 TO 5

```

```

1340   FOR W=1 TO 3
1350   OUTPUT 709;"AS"
1360   ENTER 709;T(I,J,W)
1370   T(I,J,W)=T(I,J,W)*1000
1380   NEXT W
1390   NEXT J
1400   !
1410   !           PAUSE
1420   !
1430   BEEP
1440   PRINT "TURN ON HEATED WIRES,  THEN SWITCH VALVES AND
        PRESS CONTINUE."
1450   PAUSE
1460   !
1470   !*****
1480   !*
1490   !*           BEGIN SCANNING
1500   !*
1510   !*****
1520   !
1530   FOR I=2 TO Scc
1540   OUTPUT 709;"AC40VR1SO0"
1550   ENTER 709;Pr1(I)           !   BED INLET PRESSURE
1560   OUTPUT 709;"AC41VR1SO0"
1570   ENTER 709;Pr2(I)           !   BED EXIT PRESSURE
1580   OUTPUT 709;"AC19VR1SO0"
1590   ENTER 709;I1               !   ICE BATH 0-18
1600   OUTPUT 709;"AC39VR1SO0"
1610   ENTER 709;I2               !   ICE BATH 20-38
1620   OUTPUT 709;"AC53VR1SO0"
1630   ENTER 709;I3               !   ICE BATH 40-52
1640   OUTPUT 709;"AF0AL25VR1SO0"
1650   Ib=I1
1660   !
1670   !*****
1680   !
1690   !           SCAN FLUID TEMPERATURES
1700   !
1710   FOR J=1 TO 5
1720   FOR W=1 TO 3
1730   OUTPUT 709;"AS"
1740   IF W=2 THEN
1750   Ti(I,J)=TIMEDATE           !   ENTER TIME
1760   END IF
1770   ENTER 709;T(I,J,W)
1780   T(I,J,W)=T(I,J,W)*1000
1790   NEXT W
1800   NEXT J
1810   !
1820   !*****
1830   !

```



```

1840 !          SCAN INSULATION TEMPERATURES
1850 !
1860 FOR J=1 TO 5
1870 FOR W=1 TO 2
1880 OUTPUT 709;"AS"
1890 IF J=3 AND W=1 THEN
1900 OUTPUT 709;"AS"
1910 Ib=I2
1920 END IF
1930 ENTER 709;Tis(I,J,W)
1940 Tis(I,J,W)=(Tis(I,J,W)-Ib)*1000
1950 NEXT W
1960 NEXT J
1970 !
1980 !*****
1990 !
2000 !          SCAN HEAT-TRANSFER PROBES
2010 !
2020 OUTPUT 709;"AF26AL37VR1SO0"
2030 FOR Q=1 TO 3
2040 FOR P=1 TO 4
2050 OUTPUT 709;"AS"
2060 ENTER 709;Th(I,Q,P)
2070 IF P=4 THEN
2080 Th(I,Q,P)=(Th(I,Q,P)-Ib)*1000
2090 GOTO 2150
2100 END IF
2110 Th(I,Q,P)=Th(I,Q,P)*1000
2120 IF P=2 THEN
2130 Tih(I,Q)=TIMEDATE      !      ENTER TIME
2140 END IF
2150 NEXT P
2160 NEXT Q
2170 !
2180 !*****
2190 !
2200 !          ENTER OUTSIDE WALL TEMPERATURES
2210 !
2220 OUTPUT 709;"AF48AL52VR1SO0"
2230 FOR G=1 TO 5
2240 OUTPUT 709;"AS"
2250 ENTER 709;Dd(I,G)
2260 Dd(I,G)=(Dd(I,G))*1000
2270 NEXT G
2280 OUTPUT 709;"AF54AL58VR1SO0"
2290 FOR Gg=1 TO 5
2300 OUTPUT 709;"AS"
2310 ENTER 709;Tw(I,Gg)
2320 Tw(I,Gg)=(Tw(I,Gg)-I3)*1000
2330 NEXT Gg
2340 IF Tih(I,3)-Ti(I,1)<Dt THEN

```

```

2350 WAIT Dt-(Tih(I,3)-Ti(I,1))
2360 END IF
2370 NEXT I
2380 !
2390 !*****
2400 !
2410 !          ENTER FLOW-METER INLET TEMPERATURE
2420 !
2430 OUTPUT 709;"AC39VR1SO0"
2440 ENTER 709;I2
2450 OUTPUT 709;"AC38VR1SO0"
2460 ENTER 709;Tfe
2470 Tfe=(Tfe-I2)*1000
2480 !
2490 !*****
2500 !
2510 !          INPUT COMMENTS
2520 !
2530 INPUT "HEATED WIRE CURRENT",A2
2540 INPUT "LOOP PRESSURE",Com$(7)
2550 INPUT "FLOW-RATE",Flr
2560 INPUT "COMMENTS",Com$(8)
2570 !
2580 !*****
2590 !*                                     *
2600 !*          OUTPUT DATA TO FILES          *
2610 !*                                     *
2620 !*****
2630 !
2640 ASSIGN @Path1 TO Com$(1)
2650 OUTPUT @Path1;T(*),Pr1(*),Pr2(*),Ti(*),Th(*),Tih(*),
Dd(*),Tfs,Tfe ,V1,A1,A2,Tw(*),Flr,Tis(*)
2660 ASSIGN @Path2 TO F$
2670 OUTPUT @Path2;Com$(*)
2680 !
2690 !*****
2700 !*                                     *
2710 !*          ENTER DATA FROM FILES          *
2720 !*                                     *
2730 !*****
2740 !
2750 IF Op$="OUTPUT" THEN
2760 ASSIGN @Path3 TO Com$(1)
2770 ENTER @Path3;T(*),Pr1(*),Pr2(*),Ti(*),Th(*),Tih(*),
Dd(*),Tfs,Tfe ,V1,A1,A2,Tw(*),Flr,Tis(*)
2780 G$="C"&Com$(1)
2790 ASSIGN @Path4 TO G$
2800 ENTER @Path4;Com$(*)
2810 END IF
2820 !*****
2830 !*                                     *

```

```

2840  !*                      DATA REDUCTION                      *
2850  !*                                                              *
2860  !*****
2870  !
2880  !          CALCULATE HEATED WIRE POWER
2890  !
2900  W1=A21/2*.428
2910  W2=A21/2*.442
2920  W3=A21/2*.423
2930  !
2940  !*****
2950  !
2960  !          CALCULATE WALL TEMPERATURE
2970  !
2980  FOR I=2 TO Scc
2990  FOR G=1 TO 5
3000  Dd(I,G)=FNDtemp(Tw(I,G),Dd(I,G))
3010  Tw(I,G)=FNTemp(Tw(I,G))
3020  NEXT G
3030  NEXT I
3040  !
3050  !*****
3060  !
3070  !          CALCULATE FLUID TEMPERATURES
3080  !
3090  FOR I=1 TO Scc
3100  FOR J=1 TO 5
3110  FOR W=1 TO 3
3120  T(I,J,W)=FNTemp(T(I,J,W))
3130  NEXT W
3140  NEXT J
3150  NEXT I
3160  !
3170  !*****
3180  !
3190  !          CALCULATE INSULATION TEMPERATURE
3200  !
3210  FOR I=2 TO Scc
3220  FOR J=1 TO 5
3230  FOR W=1 TO 2
3240  Tis(I,J,W)=FNTemp(Tis(I,J,W))
3250  NEXT W
3260  NEXT J
3270  NEXT I
3280  !
3290  !*****
3300  !
3310  !          CALCULATE FLUID TEMPERATURE (TIME)
3320  !
3330  Ti(1,1)=Ti(2,1)
3340  Ti(1,2)=Ti(2,2)

```

```

3350 Ti(1,3)=Ti(2,3)
3360 Ti(1,4)=Ti(2,4)
3370 Ti(1,5)=Ti(2,5)
3380 FOR I=2 TO Scc
3390 FOR J=1 TO 5
3400 Ti(I,J)=(Ti(I,J)-Ti(1,J))/60
3410 NEXT J
3420 NEXT I
3430 !
3440 !*****
3450 !
3460 !      CALCULATE HEAT TRANSFER COEFFICIENT
3470 !      (TEMPERATURE)
3480 !
3490 FOR I=2 TO Scc
3500 FOR Q=1 TO 3
3510 FOR P=1 TO 4
3520 IF P=4 THEN
3530 Th(I,Q,P)=FNTemp(Th(I,Q,P))
3540 GOTO 3570
3550 END IF
3560 Th(I,Q,P)=FNDtemp(Th(I,Q,4),Th(I,Q,P))
3570 NEXT P
3580 NEXT Q
3590 NEXT I
3600 !
3610 !*****
3620 !
3630 !      CALCULATE FLOW-METER FLUID TEMPERATURE
3640 !
3650 Tfs=FNTemp(Tfs)
3660 Tfe=FNTemp(Tfe)
3670 !
3680 !*****
3690 !
3700 !      CALCULATE HEAT TRANSFER COEF. (TIME)
3710 !
3720 Tih(1,1)=Tih(2,1)
3730 Tih(1,2)=Tih(2,2)
3740 Tih(1,3)=Tih(2,3)
3750 FOR I=2 TO Scc
3760 FOR Q=1 TO 3
3770 Tih(I,Q)=(Tih(I,Q)-Tih(1,Q))/60
3780 NEXT Q
3790 NEXT I
3800 !
3810 !*****
3820 !
3830 !      CALCULATE PRESSURE (INLET AND EXIT)
3840 !
3850 FOR I=1 TO Scc

```

```

3860 Pr1(I)=.8247*Pr1(I)+12.180+Atm      ! #3336
3870 Pr2(I)=.9477*Pr2(I)-.9812+Atm      ! #3324
3880 NEXT I
3890 !
3900 !*****
3910 !
3920 !   CALCULATE AVERAGE FLUID TEMPERATURES
3930 !
3940 FOR W=1 TO 3
3950 Tsi=Tsi+T(Scc,l,W)
3960 Nnn=Nnn+1
3970 NEXT W
3980 IF Tsi=0 THEN
3990 Tsi=1
4000 Nnn=1
4010 END IF
4020 Tsi=Tsi/Nnn
4030 Nnn=0
4040 FOR I=1 TO Scc
4050 FOR J=1 TO 5
4060 FOR W=1 TO 3
4070 Ts=Ts+T(I,J,W)
4080 Nnn=Nnn+1
4090 NEXT W
4100 IF Ts=0 THEN
4110 Ts=1
4120 Nnn=1
4130 END IF
4140 IF J=1 THEN
4150 T1a(I)=Ts/Nnn
4160 T1ad(I)=(T1a(I)-73)/(Tsi-73)
4170 T1i(I)=Ti(I,J)
4180 END IF
4190 IF J=2 THEN
4200 T2a(I)=Ts/Nnn
4210 T2ad(I)=(T2a(I)-73)/(Tsi-73)
4220 T2i(I)=Ti(I,J)
4230 END IF
4240 IF J=3 THEN
4250 T3a(I)=Ts/Nnn
4260 T3ad(I)=(T3a(I)-73)/(Tsi-73)
4270 T3i(I)=Ti(I,J)
4280 END IF
4290 IF J=4 THEN
4300 T4a(I)=Ts/Nnn
4310 T4ad(I)=(T4a(I)-73)/(Tsi-73)
4320 T4i(I)=Ti(I,J)
4330 END IF
4340 IF J=5 THEN
4350 T5a(I)=Ts/Nnn
4360 T5ad(I)=(T5a(I)-73)/(Tsi-73)

```

```

4370 T5i(I)=Ti(I,J)
4380 END IF
4390 Ts=0
4400 Nnn=0
4410 NEXT J
4420 NEXT I
4430 !
4440 !*****
4450 !
4460 ! CALCULATE FLUID MASS FLOW-RATE
4470 !
4480 IF Com$(4)="S" THEN
4490 Fff=560
4500 ELSE
4510 Fff=26
4520 END IF
4530 Fff=Flr/Fff
4540 Tfa=(Tfs+Tfe)/2
4550 Row1=7.65744893*Tfa-.239537209*Tfa1/2+4.0027660E-
3*Tfa3/4-3.9061137E-5*Tfa5/4
4560 Row2=2.2295816E-7*Tfa5/4-6.9040268E-
10*Tfa3/4+8.9606319E-13*Tfa1/4
4570 Row=Row1+Row2
4580 Mdot=(Fff*.133680556)*Row
4590 !
4600 !*****
4610 !
4620 ! CALCULATE ENTHALPY OF WORKING FLUID
4630 !
4640 FOR I=2 TO Scc
4650 Hh1(I)=FNHf(T1a(I))
4660 Hh2(I)=FNHf(T2a(I))
4670 Hh3(I)=FNHf(T3a(I))
4680 Hh4(I)=FNHf(T4a(I))
4690 Hh5(I)=FNHf(T5a(I))
4700 IF Fldir$="UP" THEN
4710 Hh1(I)=-Hh1(I)
4720 Hh2(I)=-Hh2(I)
4730 Hh3(I)=-Hh3(I)
4740 Hh4(I)=-Hh4(I)
4750 Hh5(I)=-Hh5(I)
4760 END IF
4770 NEXT I
4780 !
4790 !*****
4800 !
4810 !CALCULATE ENERGY REMOVED FROM WORKING FLUID
4820 !
4830 Ww=Tli(Scc)/(Scc-1)
4840 IF Fldir$="DOWN" THEN
4850 Timel=(.9688*Row*.0623)/Mdot

```

```

4860 Time2=(1.8854*Row*.0623)/Mdot
4870 Time3=(2.8021*Row*.0623)/Mdot
4880 Time4=(3.7708*Row*.0623)/Mdot
4890 ELSE
4900 Time1=(3.7708*Row*.0623)/Mdot
4910 Time2=(2.8021*Row*.0623)/Mdot
4920 Time3=(1.8854*Row*.0623)/Mdot
4930 Time4=(.9688*Row*.0623)/Mdot
4940 END IF
4950 FOR I=2 TO (Scc-1)
4960 Q1(I)=Q1(I-1)+(Ww/2)*(Hh1(I+1)+Hh1(I))*Mdot
4970 Q2(I)=Q2(I-1)+(Ww/2)*(Hh2(I+1)+Hh2(I))*Mdot
4980 Q3(I)=Q3(I-1)+(Ww/2)*(Hh3(I+1)+Hh3(I))*Mdot
4990 Q4(I)=Q4(I-1)+(Ww/2)*(Hh4(I+1)+Hh4(I))*Mdot
5000 Q5(I)=Q5(I-1)+(Ww/2)*(Hh5(I+1)+Hh5(I))*Mdot
5010 NEXT I
5020 FOR I=2 TO (Scc-1)
5030 IF T2i(I)>=Time1 THEN
5040 Q2(I)=Q1(I)-Q2(I)
5050 ELSE
5060 Q2(I)=0
5070 END IF
5080 IF T3i(I)>Time2 THEN
5090 Q3(I)=Q1(I)-Q3(I)
5100 ELSE
5110 Q3(I)=0
5120 END IF
5130 IF T4i(I)>Time3 THEN
5140 Q4(I)=Q1(I)-Q4(I)
5150 ELSE
5160 Q4(I)=0
5170 END IF
5180 IF T5i(I)>Time4 THEN
5190 Q5(I)=Q1(I)-Q5(I)
5200 ELSE
5210 Q5(I)=0
5220 END IF
5230 NEXT I
5240 !
5250 !*****
5260 !
5270 ! CALCULATE ENERGY STORE IN WORKING FLUID
5280 !
5290 FOR I=2 TO (Scc-1)
5300 Ef1(I)=Ef1(I-1)+FNFe(T1a(I+1),T2a(I+1),T1a(I),
T2a(I))
5310 Ef2(I)=Ef2(I-1)+FNFe(T2a(I+1),T3a(I+1),T2a(I),
T3a(I))
5320 Ef3(I)=Ef3(I-1)+FNFe(T3a(I+1),T4a(I+1),T3a(I),
T4a(I))
5330 Ef4(I)=Ef4(I-1)+FNFe(T4a(I+1),T5a(I+1),T4a(I),

```

```

T5a(I))
5340 NEXT I
5350 !
5360 !*****
5370 !
5380 !CALCULATE ENERGY LOSSES
5390 !
5400 FOR I=2 TO (Scc-1)
5410 FOR J=1 TO 4
5420 Dtcs1=Tis(I,J,1)-Tis(I,J,2)+Tis(I,J+1,1)-
Tis(I,J+1,2)+Tis(I+1,J,1)-Tis(I+1,J,2)
5430 Dtcs2=Tis(I+1,J+1,1)-Tis(I+1,J+1,2)
5440 Dtc(I,J)=(Dtcs1+Dtcs2)*.25
5450 NEXT J
5460 NEXT I
5470 FOR I=2 TO (Scc-1)
5480 FOR J=1 TO 4
5490 Dtc(I,J)=Dtc(I,J)*Ww*1.6254E-3
5500 NEXT J
5510 NEXT I
5520 FOR I=2 TO Scc
5530 Dd(I,3)=(Dd(I,2)+Dd(I,4))* .5
5540 NEXT I
5550 FOR I=2 TO (Scc-1)
5560 Dd(I,3)=(Dd(I,2)+Dd(I,4))* .5
5570 Dts1(I)=FNDtw(Tw(I+1,1),Dd(I+1,1),Tw(I+1,2),
Dd(I+1,2),Tw(I,1),Dd(I,1),Tw(I,2),Dd(I,2))
5580 Dts2(I)=FNDtw(Tw(I+1,2),Dd(I+1,2),Tw(I+1,3),
Dd(I+1,3),Tw(I,2),Dd(I,2),Tw(I,3),Dd(I,3))
5590 Dts3(I)=FNDtw(Tw(I+1,3),Dd(I+1,3),Tw(I+1,4),
Dd(I+1,4),Tw(I,3),Dd(I,3),Tw(I,4),Dd(I,4))
5600 Dts4(I)=FNDtw(Tw(I+1,4),Dd(I+1,4),Tw(I+1,5),
Dd(I+1,5),Tw(I,4),Dd(I,4),Tw(I,5),Dd(I,5))
5610 NEXT I
5620 FOR I=2 TO (Scc-1)
5630 Q11(I)=Q11(I-1)+1.9931*Dts1(I)+Dtc(I,1)
5640 Q12(I)=Q12(I-1)+1.9931*Dts2(I)+Dtc(I,2)
5650 Q13(I)=Q13(I-1)+1.9931*Dts3(I)+Dtc(I,3)
5660 Q14(I)=Q14(I-1)+1.9331*Dts4(I)+Dtc(I,4)
5670 NEXT I
5680 FOR I=2 TO (Scc-1)
5690 Qa1(I)=Q2(I)+(I-1)*Ww*W1*.05687-Q11(I)-Ef1(I)
5700 Qa1(I)=ABS(Qa1(I))
5710 Qa2(I)=Q3(I)+(I-1)*Ww*(W2+W1)*.05687-Q12(I)-Ef2(I)-
Q11(I)-Ef1(I)
5720 Qa2(I)=ABS(Qa2(I))
5730 Qa3(I)=Q4(I)+(I-1)*Ww*(W3+W2+W1)*.05687-Q13(I)-
Ef3(I)-Q11(I)-Q12(I)-Ef1(I)-Ef2(I)
5740 Qa3(I)=ABS(Qa3(I))
5750 Qa4(I)=Q5(I)-Q14(I)-Ef4(I)-Q11(I)-Q12(I)-Q13(I)-
Ef1(I)-Ef2(I)-Ef3(I)+(I-1)*Ww*(W1+W2+W3)*.05687

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```

5760 Qa4(I)=ABS(Qa4(I))
5770 IF Qa4(I)>Qmax THEN
5780 Qmax=Qa4(I)
5790 END IF
5800 NEXT I
5810 !
5820 !*****
5830 !
5840 !CALCULATE HEAT TRANSFER COEFF.
5850 !
5860 FOR I=2 TO Scc
5870 FOR Q=1 TO 3
5880 FOR P=1 TO 3
5890 IF Q=2 AND P=3 OR Q=3 AND P=3 THEN GOTO 5920
5900 Ts=Ts+Th(I,Q,P)
5910 Nnn=Nnn+1
5920 NEXT P
5930 IF Ts=0 THEN
5940 Nnn=1
5950 Ts=1
5960 END IF
5970 Tavg=Nnn/Ts
5980 Ts=0
5990 Nnn=0
6000 IF Q=1 THEN
6010 T1ha(I)=Tavg*64.08*W1 ! .428
6020 T1hi(I)=Tih(I,Q)
6030 END IF
6040 IF Q=2 THEN
6050 T2ha(I)=Tavg*64.50*W2 ! .442
6060 T2hi(I)=Tih(I,Q)
6070 END IF
6080 IF Q=3 THEN
6090 T3ha(I)=Tavg*62.54*W3 ! .423
6100 T3hi(I)=Tih(I,Q)
6110 END IF
6120 Ts=0
6130 Nnn=0
6140 NEXT Q
6150 NEXT I
6160 !
6170 !*****
6180 !*
6190 !* OUTPUT DATA TO PRINTER
6200 !*
6210 !*****
6220 !
6230 PRINT "FILE NAME:";Com$(1)
6240 PRINT "DATE:";Com$(2)
6250 PRINT "THERMAL STORAGE MATERIAL:";Com$(3)
6260 Fff=DROUND(Fff,4)

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```

6270 Mdot=DROUND(Mdot,4)
6280 PRINT "FLOW-RATE:";Fff;"GPM",Mdot;"LBM/MIN"
6290 Lpp=(A1*V1)/1000
6300 Lpp=DROUND(Lpp,4)
6310 PRINT "LOOP POWER:";Lpp;"KW"
6320 PRINT "LOOP PRESSURE:";Com$(7);"PSI"
6330 PRINT "NUMBER OF SCANS:";Scc-1
6340 Q2(Scc-1)=DROUND(Q2(Scc-1),4)
6350 Q3(Scc-1)=DROUND(Q3(Scc-1),4)
6360 Q4(Scc-1)=DROUND(Q4(Scc-1),4)
6370 Q5(Scc-1)=DROUND(Q5(Scc-1),4)
6380 Qa1(Scc-1)=DROUND(Qa1(Scc-1),4)
6390 Qa2(Scc-1)=DROUND(Qa2(Scc-1),4)
6400 Qa3(Scc-1)=DROUND(Qa3(Scc-1),4)
6410 Qa4(Scc-1)=DROUND(Qa4(Scc-1),4)
6420 W1=DROUND(W1,4)
6430 W2=DROUND(W2,4)
6440 W3=DROUND(W3,4)
6450 FOR I=2 TO (Scc-1)
6460 FOR J=1 TO 4
6470 Dtct(J)=Dtct(J)+Dtc(I,J)
6480 NEXT J
6490 NEXT I
6500 Dtct(1)=DROUND(Dtct(1),4)
6510 Dtct(2)=DROUND(Dtct(2),4)
6520 Dtct(3)=DROUND(Dtct(3),4)
6530 Dtct(4)=DROUND(Dtct(4),4)
6540 Q11(Scc-1)=Q11(Scc-1)-Dtct(1)
6550 Q12(Scc-1)=Q12(Scc-1)-Dtct(2)
6560 Q13(Scc-1)=Q13(Scc-1)-Dtct(3)
6570 Q14(Scc-1)=Q14(Scc-1)-Dtct(4)
6580 Q11(Scc-1)=DROUND(Q11(Scc-1),4)
6590 Q12(Scc-1)=DROUND(Q12(Scc-1),4)
6600 Q13(Scc-1)=DROUND(Q13(Scc-1),4)
6610 Q14(Scc-1)=DROUND(Q14(Scc-1),4)
6620 Ef1(Scc-1)=DROUND(Ef1(Scc-1),4)
6630 Ef2(Scc-1)=DROUND(Ef2(Scc-1),4)
6640 Ef3(Scc-1)=DROUND(Ef3(Scc-1),4)
6650 Ef4(Scc-1)=DROUND(Ef4(Scc-1),4)
6660 Wp1=DROUND(W1*Tli(Scc)*.05687,4)
6670 Wp2=DROUND(W2*Tli(Scc)*.05687,4)
6680 Wp3=DROUND(W3*Tli(Scc)*.05687,4)
6690 PRINT
6700 PRINT "ENERGY WIRE 1=";Wp1;"BTU"
6710 PRINT "ENERGY WIRE 2=";Wp2;"BTU"
6720 PRINT "ENERGY WIRE 3=";Wp3;"BTU"
6730 PRINT
6740 PRINT "DQT 1-2=";Q2(Scc-1);"BTU"
6750 PRINT "DQT 1-3=";Q3(Scc-1);"BTU"
6760 PRINT "DQT 1-4=";Q4(Scc-1);"BTU"
6770 PRINT "DQT 1-5=";Q5(Scc-1);"BTU"

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```

6780 PRINT
6790 PRINT "DQAL 1-2=";Q11(Scc-1);"BTU"
6800 PRINT "DQAL 2-3=";Q12(Scc-1);"BTU"
6810 PRINT "DQAL 3-4=";Q13(Scc-1);"BTU"
6820 PRINT "DQAL 4-5=";Q14(Scc-1);"BTU"
6830 PRINT
6840 PRINT "DQF 1-2=";Ef1(Scc-1);"BTU"
6850 PRINT "DQF 2-3=";Ef2(Scc-1);"BTU"
6860 PRINT "DQF 3-4=";Ef3(Scc-1);"BTU"
6870 PRINT "DQF 4-5=";Ef4(Scc-1);"BTU"
6880 PRINT
6890 PRINT "DQC 1-2=";Dtct(1);"BTU"
6900 PRINT "DQC 2-3=";Dtct(2);"BTU"
6910 PRINT "DQC 3-4=";Dtct(3);"BTU"
6920 PRINT "DQC 4-5=";Dtct(4);"BTU"
6930 PRINT
6940 Qmax=DROUND(Qmax,4)
6950 PRINT "QMAX=";Qmax;"BTU"
6960 PRINT
6970 PRINT "DQP 1-2=";Qa1(Scc-1);"BTU"
6980 PRINT "DQP 1-3=";Qa2(Scc-1);"BTU"
6990 PRINT "DQP 1-4=";Qa3(Scc-1);"BTU"
7000 PRINT "DQP 1-5=";Qa4(Scc-1);"BTU"
7010 PRINT
7020 PRINT "COMMENTS:";Com$(5),Com$(8)
7030 PRINT
7040 PRINT
7050 PRINT "INITIAL BED STATUS"
7060 PRINT
7070 FOR J=1 TO 5
7080 FOR W=1 TO 3
7090 T(1,J,W)=DROUND(T(1,J,W),4)
7100 NEXT W
7110 NEXT J
7120 Tstart=(T1a(1)+T2a(1)+T3a(1)+T4a(1)+T5a(1))*0.2
7130 Tend=(T1a(Scc)+T2a(Scc)+T3a(Scc)+T4a(Scc)+
T5a(Scc))*0.2
7140 Tcap=DROUND(ABS(46.4*(Tstart-Tend)*0.22),4)
7150 PRINT
7160 PRINT "THEORETICAL CAPACITY OF BED=";Tcap;"BTU"
7170 FOR I=1 TO Scc STEP 10
7180 PRINT
7190 T1a(I)=DROUND(T1a(I),4)
7200 T2a(I)=DROUND(T2a(I),4)
7210 T3a(I)=DROUND(T3a(I),4)
7220 T4a(I)=DROUND(T4a(I),4)
7230 T5a(I)=DROUND(T5a(I),4)
7240 PRINT USING "DDDDD.D";T(I,1,1),T(I,1,2),T(I,1,3),
T1a(I)
7250 PRINT USING "DDDDD.D";T(I,2,1),T(I,2,2),T(I,2,3),
T2a(I)

```

```

7260 PRINT USING "DDDDD.D";T(I,3,1),T(I,3,2),
      T(I,3,3),T3a(I)
7270 PRINT USING "DDDDD.D";T(I,4,1),T(I,4,2),T(I,4,3),
      T4a(I)
7280 PRINT USING "DDDDD.D";T(I,5,1),T(I,5,2),T(I,5,3),
      T5a(I)
7290 NEXT I
7300 PRINT
7310 PRINT
7320 PRINTER IS 1
7330 PRINT " TIME      TEMPERATURE      TEMPERATURE"
7340 PRINT "(MIN.)      (F)"
7350 FOR I=2 TO Scc
7360 T5i(I)=DROUND(T5i(I),3)
7370 T5a(I)=DROUND(T5a(I),4)
7380 T5ad(I)=DROUND(T5ad(I),3)
7390 PRINT T5i(I),T5a(I),"      ";T5ad(I)
7400 NEXT I
7410 ASSIGN @Path5 TO A$
7420 ASSIGN @Path6 TO B$
7430 ASSIGN @Path7 TO C$
7440 ASSIGN @Path8 TO D$
7450 ASSIGN @Path9 TO E$
7460 T1i(1)=0
7470 T2i(1)=0
7480 T3i(1)=0
7490 T4i(1)=0
7500 T5i(1)=0
7510 !
7520 !*****
7530 !*                                     *
7540 !*      OUTPUT DATA TO FILES      *
7550 !*                                     *
7560 !*****
7570 !
7580 OUTPUT @Path5;T1i(*),T1a(*),T2i(*),T2a(*),T3i(*),
      T3a(*),T4i(*),T4 a(*),T5i(*),T5a(*)
7590 OUTPUT @Path6;T1i(*),T1ad(*),T2i(*),T2ad(*),T3i(*),
      T3ad(*),T4i(*) ,T4ad(*),T5i(*),T5ad(*)
7600 OUTPUT @Path7;T1i(*),Pr1(*),T1i(*),Pr2(*)
7610 OUTPUT @Path8;T1hi(*),T1ha(*),T2hi(*),T2ha(*),
      T3hi(*),T3ha(*)
7620 Z=0
7630 OUTPUT @Path9;T1i(*),Z,Qa1(*),T1i(*),Z,Qa2(*),
      T1i(*),Z,Qa3(*),T1i (*),Z,Qa4(*)
7640 END
7650 !
7660 !*****
7670 !*                                     *
7680 !*      DEFINE USER DEFINED FUNCTIONS      *
7690 !*                                     *

```

```

7700  !*****
7710  !
7720  DEF FNTemp(V)
7730  P1=25.88937897
7740  P2=-.366006670
7750  P3=-.536749212
7760  P4=.388477456
7770  P5=-.135293163
7780  P6=.023396843
7790  P7=-.001588515
7800  T1=P3+V*(P4+V*(P5+V*(P6+V*P7)))
7810  Temp=V*(P1+V*(P2+V*T1))*1.8+32
7820  RETURN Temp
7830  FNEND
7840  DEF FNDtemp(Vv,Dv)
7850  P8=25.88937897
7860  P9=-7.3201334E-1
7870  P10=-1.6102476
7880  P11=1.5539098
7890  P12=-6.7646581E-1
7900  P13=1.4038105E-1
7910  P14=-1.1119605E-2
7920  T1=P11+Vv*(P12+Vv*(P13+Vv*P14))
7930  Dtemp=(P8+Vv*(P9+Vv*(P10+Vv*T1)))*Dv*1.8
7940  RETURN Dtemp
7950  FNEND
7960  DEF FNHf(Vvv)
7970  P15=7.97193634E-1
7980  P16=-1.8485485E-2
7990  P17=3.1326443E-4
8000  P18=-3.0963583E-6
8010  P19=1.7969331E-8
8020  P20=-5.669255821E-11
8030  P21=7.503588447E-14
8040  Hf1=P18+Vvv*(P19+Vvv*(P20+Vvv*P21))
8050  Hf=Vvv*(P15+Vvv*(P16+Vvv*(P17+Vvv*Hf1)))
8060  RETURN Hf
8070  FNEND
8080  DEF FNDtw(Tt1,Tt2,Tt3,Tt4,Tt5,Tt6,Tt7,Tt8)
8090  Dtw1=(Tt1+(Tt2*.5)+Tt3+(Tt4*.5))*0.5
8100  Dtw2=(Tt5+(Tt6*.5)+Tt7+(Tt8*.5))*0.5
8110  Dtw=Dtw1-Dtw2
8120  RETURN Dtw
8130  FNEND
8140  DEF FNFe(Tf1,Tf2,Tf3,Tf4)
8150  P22=7.65744893
8160  P23=-.239537209
8170  P24=4.0027660E-3
8180  P25=-3.9061137E-5
8190  P26=2.2295816E-7
8200  P27=-6.9040268E-10

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```

8210 P28=8.9606319E-13
8220 P29=7.97193634E-1
8230 P30=-3.6970970E-2
8240 P31=9.3979329E-4
8250 P32=-1.2385433E-5
8260 P33=8.9846655E-8
8270 P34=-3.4015534E-10
8280 P35=5.2525119E-13
8290 Tfavg=(Tf1+Tf2+Tf3+Tf4)*.25
8300 Row1=P25+Tfavg*(P26+Tfavg*(P27+Tfavg*P28))
8310 Row=Tfavg*(P22+Tfavg*(P23+Tfavg*(P24+Tfavg*Row1)))
8320 Dhdt1=P32+Tfavg*(P33+Tfavg*(P34+Tfavg*P35))
8330 Dhdt=P29+Tfavg*(P30+Tfavg*(P31+Tfavg*Dhdt1))
8340 Trise=(Tf1+Tf2-Tf3-Tf4)*.5
8350 Ef=Dhdt*5.6204E-2*Row*Trise
8360 RETURN Ef
8370 FNEND

```

The following is a listing of the plotting program used in this study:

```

10 !THIS PROGRAM WILL PLOT ANY 2-D CURVE
20 !(HOPEFULLY)
30 !DATA MUST BE ENTERED IN INPUTTED FILE
40 !NAME WITH ALL X COORDINATES FIRST AND
50 !ALL Y COORDINATES FOLLOWING
60 !1000 PAIRS OF DATA POINTS
70 !IS THE MAX. ALLOWABLE.
80 !
90 !
100 DIM X1(10000),Y1(10000)
110 GINIT
120 GRAPHICS ON
130 INPUT "MIN. AND MAX. X COORD.",O1,L1,"MIN. AND MAX.
Y COORD.",O2,L2
140 INPUT "INTERVAL BETWEEN TICK MARKS ON X
AXIS",Xt,"INTERVAL BETWEEN TICK MARKS ON Y AXIS",Yt
150 L1=L1+3*Xt
160 L2=L2+3*Yt
170 INPUT "DO YOU WISH TO OFFSET AXES NUMBERING",Ttt
180 IF Ttt=1 THEN
190 !
200 !
210 ! AXIS OFFSET WILL SHIFT THE AXIS UP OR DOWN
220 ! (OR LEFT OR RIGHT)BY A DESIRED DISTANCE
230 !
240 INPUT "Y AXIS OFFSET DISTANCE",Oy,"X AXIS OFFSET
DISTANCE",Ox
250 ELSE
260 Ox=0

```

```

270 Oy=0
280 END IF
290 IF O1=0 THEN
300 X=.2*(L1-O1)!---IF AXIS POSITION IS
310 Y=.2*(L2-O2) !UNACCEPTABLE ALTER CONSTANTS
320 Xw=L1+X-O1-Xt
330 Yw=L2+Y-O2-3*Yt
340 ELSE ! (1ST QUADRANT)
350 X=1.0*(L1-O1)
360 Y=1.0*(L2-O2) !----- (2ND QUADRANT)
370 Xw=L1+X-O1-3*Xt
380 Yw=L2+Y-O2-3*Yt
390 END IF
400 INPUT "NUMBER OF TICK MARKS BETWEEN AXES
NUMBERING",N
410 WINDOW O1,Xw,O2,Yw
420 AXES Xt,Yt,X,Y
430 MOVE X+L1,Y
440 PEN -1
450 IDRAW -3*Xt,0
460 PEN -1
470 IMOVE Xt/2,Yt
480 IDRAW 0,-2*Yt
490 IMOVE Xt,Yt
500 IDRAW 0,2*Yt
510 !
520 !
530 ! IF THE ORIGIN IS 0, THE NEGATIVE PORTION
540 ! OF THE X-AXIS IS ERASED
550 !
560 IF O1=0 THEN
570 PEN -1
580 MOVE X,Y
590 IDRAW -X,0
600 IMOVE X,Yt
610 Ii=1
620 FOR I=Xt TO X STEP Xt
630 IMOVE -Xt,0
640 IDRAW 0,2*Yt*(-1)1/4Ii
650 Ii=Ii+1
660 NEXT I
670 MOVE X,Y
680 ELSE
690 END IF
700 IF L1=0 THEN
710 MOVE X,Y
720 PEN -1
730 IDRAW Xw,0
740 MOVE X,Y+Yt
750 Ii=1
760 FOR I=Xt TO Xw STEP Xt

```

```

770  IMOVE Xt,0
780  IDRAW 0,2*Yt*(-1)½Ii
790  Ii=Ii+1
800  NEXT I
810  MOVE X,Y
820  END IF
830  !
840  !
850  ! IF THE ORIGIN IS 0, THE NEGATIVE
860  ! PORTION OF THE Y-AXIS IS ERASED
870  !
880  IF O2=0 THEN
890  PEN -1
900  IDRAW 0,-Y
910  Jj=1
920  MOVE X+Xt,Y
930  FOR I=Yt TO Y STEP Yt
940  IMOVE 0,-Yt
950  IDRAW 2*Xt*(-1)½Jj,0
960  Jj=Jj+1
970  NEXT I
980  END IF
990  !
1000 !
1010 ! IF THE ENDPOINT OF AXIS IS 0, NEG.
1020 ! PORTION IS ERASED
1030 !
1040 IF L2=0 THEN
1050 PEN -1
1060 MOVE X,Y
1070 IDRAW 0,Yw
1080 Jj=1
1090 MOVE X+Xt,Y
1100 FOR I=Yt TO Yw STEP Yt
1110 IMOVE 0,Yt
1120 IDRAW 2*Xt*(-1)½Jj,0
1130 Jj=Jj+1
1140 NEXT I
1150 END IF
1160 X5=X          !ERASE AXIS TICK HALVES
1170 FOR I=Xt TO 2*(L1+Xt-01) STEP Xt
1180 MOVE X5+Xt,Y
1190 X5=X5+Xt
1200 IDRAW 0,-Yt
1210 NEXT I
1220 X5=X
1230 FOR I=Xt TO 2*(L1+Xt-01) STEP Xt
1240 MOVE X5-Xt,Y
1250 X5=X5-Xt
1260 IDRAW 0,-Yt
1270 NEXT I

```



```

1280 Y5=Y
1290 FOR I=Yt TO 2*(L2+Yt-O2) STEP Yt
1300 MOVE X,Y5+Yt
1310 Y5=Y5+Yt
1320 IDRAW -Xt,0
1330 NEXT I
1340 Y5=Y
1350 FOR I=Yt TO 2*(L2+Yt-O2) STEP Yt
1360 MOVE X,Y5-Yt
1370 Y5=Y5-Yt
1380 IDRAW -Xt,0
1390 NEXT I
1400 PEN 1
1410 MOVE X+L1-3*Xt,Y
1420 IDRAW 0,L2-3*Yt-O2
1430 IDRAW -(L1-3*Xt-O1),0
1440 IDRAW 0,-(-3*Yt+L2-O2)
1450 IDRAW L1-3*Xt-O1,0
1460 IDRAW 0,O2
1470 IMOVE -(L1-3*Xt-O1),0
1480 PEN -1
1490 IDRAW 0,-ABS(L2-O2)
1500 IMOVE 0,ABS(L2-O2)
1510 PEN 1
1520 IDRAW (L1-3*Xt-O1),0
1530 IMOVE -(L1-3*Xt-O1),0
1540 IDRAW 0,-O2
1550 PEN -1
1560 IMOVE Xt,O2
1570 Jj=1
1580 PEN -1
1590 FOR I=Yt TO ABS(L2-O2) STEP Yt
1600 IMOVE 0,-Yt
1610 IDRAW 2*(-1½Jj)*Xt,0
1620 Jj=Jj+1
1630 NEXT I
1640 !
1650 !
1660 ! LABEL X-AXIS COORDINATES
1670 !
1680 INPUT "DO YOU WANT AXES NUMBERING?",Ww
1690 IF Ww=1 THEN
1700 PEN 1
1710 IF L2=0 THEN
1720 LORG 4
1730 ELSE
1740 LORG 6
1750 END IF
1760 FOR I=Xt TO ABS(O1) STEP Xt
1770 MOVE X-N*I,Y
1780 IF N*I>Xw THEN

```

```

1790 ELSE
1800 LABEL-N*I+Ox
1810 END IF
1820 NEXT I
1830 MOVE X,Y
1840 FOR I=0 TO L1-3*Xt STEP Xt
1850 MOVE X+N*I,Y
1860 LABEL N*I+Ox
1870 NEXT I
1880 !
1890 !
1900 ! LABEL Y-AXIS COORDINATES
1910 !
1920 IF L1=0 THEN
1930 LORG 2
1940 ELSE
1950 LORG 8
1960 END IF
1970 FOR I=Yt TO 2*ABS(O2) STEP Yt
1980 MOVE X,Y-N*I
1990 IF N*I>Yw THEN
2000 ELSE
2010 LABEL " " ; -1*N*I+Oy
2020 END IF
2030 NEXT I
2040 FOR I=0 TO L2-4*Yt STEP Yt
2050 MOVE X,Y+N*I
2060 LABEL N*I+Oy
2070 NEXT I
2080 END IF
2090 PEN 1
2100 !
2110 !
2120 ! INPUT AXIS TITLES
2130 !
2140 INPUT "X AXIS LABEL",A$,"Y AXIS LABEL",B$
2150 INPUT "COORD. OF X LABEL MID.",K,Q,"COORD. OF Y
    LABEL MID.",W,L
2160 LORG 6
2170 MOVE X+K-Ox,Y+Q-Oy
2180 LABEL A$
2190 MOVE X+W-Ox,Y+L-Oy
2200 DEG
2210 LDIR 90
2220 LABEL B$
2230 F=X+K-Ox
2240 G=Q+Y-Oy
2250 !
2260 !
2270 ! ALLOW FOR CORRECTION OF AXIS TITLE
2280 ! COORDINATES (ERASE OLD LABEL, REPLACE

```

```

2290 ! WITH NEW)
2300 !
2310 INPUT "ARE YOU HAPPY WITH X AXIS LABEL POSITION YES-
      1 NO-2",E
2320 IF E=2 THEN
2330 PEN -1
2340 LDIR 0
2350 MOVE F,G
2360 LABEL A$
2370 PEN 1
2380 INPUT "POSITION OF X AXIS LABEL(BOTH X AND Y
      COORD.)",F,G
2390 F=F+X-Ox
2400 G=G+Y-Oy
2410 MOVE F,G
2420 LABEL A$
2430 GOTO 2310
2440 ELSE
2450 END IF
2460 H=W+X-Ox
2470 K=Y+L-Oy
2480 INPUT "ARE YOU HAPPY WITH Y AXIS LABEL YES-1 NO-
      2",E
2490 IF E=2 THEN
2500 PEN -1
2510 LDIR 90
2520 MOVE H,K
2530 LABEL B$
2540 PEN 1
2550 INPUT "POSITION OF Y AXIS LABEL",H,K
2560 H=H+X-Ox
2570 K=K+Y-Oy
2580 MOVE H,K
2590 LABEL B$
2600 GOTO 2480
2610 ELSE
2620 END IF
2630 !
2640 !
2650 ! DATA FILE NAME IS INPUTTED FOR PLOTTING
2660 ! AS MANY CURVES AS DESIRED MAY BE PLOTTED
2670 ! ON SAME SET OF AXES. EACH CURVE MUST
2680 ! HAVE ALL X VALUES FOLLOWED BY ALL Y
2690 ! VALUES
2700 !
2710 PEN 1
2720 LDIR 0
2730 INPUT "DATA FILE NAME",R$
2740 ASSIGN @Path1 TO R$
2750 INPUT "HOW MANY CURVES/PLOT",P
2760 FOR C=1 TO P

```

```

2770 INPUT "HOW MANY SETS OF DATA POINTS ARE THERE IN
THIS PLOT?",Nnn
2780 FOR I=1 TO Nnn
2790 ENTER @Path1;X1(I),Y1(I)
2800 NEXT I
2810 MOVE X,Y
2820 FOR I=1 TO Nnn
2830 IF I<=1 THEN
2840 MOVE X1(I)+X-Ox,Y1(I)+Y-Oy
2850 GOTO 2880
2860 END IF
2870 DRAW X1(I)+X-Ox,Y1(I)+Y-Oy
2880 NEXT I
2890 NEXT C
2900 IF Tderrq=1 THEN GOTO 3180
2910 !
2920 !
2930 ! ALLOW FOR INPUT OF GRAPH TITLE, CURVE
2940 ! LABELS, ETC.
2950 !
2960 LDIR 0
2970 INPUT "DO YOU WISH TO ADD MORE LABELS? Y-1 N-2",E
2980 IF E=1 THEN
2990 INPUT "LABEL?",A$,"COORDINATES OF LABEL",A,B
3000 LONG 5
3010 MOVE A+X-Ox,B+Y-Oy
3020 LABEL A$
3030 !
3040 !
3050 ! ALLOW FOR CORRECTION OF LABEL COORD.
3060 ! (ERASE OLD LABEL, REPLACE AT NEW COORD.)
3070 !
3080 INPUT "ARE LABEL COORD. ACCEPTABLE? Y-1 N-2",E
3090 IF E=2 THEN
3100 MOVE A+X-Ox,B+Y-Oy
3110 PEN -1
3120 LABEL A$
3130 INPUT "NEW LABEL COORD. X,Y",H,K
3140 MOVE H+X-Ox,K+Y-Oy
3150 PEN 1
3160 LABEL A$
3170 GOTO 3080
3180 ELSE
3190 END IF
3200 GOTO 2970
3210 ELSE
3220 END IF
3230 INPUT "DO YOU WISH TO LABEL AXIS",Lax
3240 IF Lax=1 THEN
3250 LONG 5
3260 INPUT "LABEL",A$

```

```
3270 INPUT "LABEL COORD. ",H,G
3280 MOVE H+X-Ox,G+Y-Oy
3290 LDIR 90
3300 PEN 1
3310 LABEL A$
3320 END IF
3330 INPUT "ARE LABEL COORD. ACCEPTABLE",As
3340 IF As=2 THEN
3350 MOVE H+X-Ox,G+Y-Oy
3360 PEN -1
3370 LABEL A$
3380 ELSE
3390 GOTO 3420
3400 END IF
3410 GOTO 3270
3420 END
```

APPENDIX B

PWC-89 AND PWC-73 EXPERIMENTAL RESULTS

Experiments with PWC-89 and a Single-Phase Working Fluid

The heat transfer fluid used in these runs was R-113. The bed packing material used in these runs was 1/2" diameter 5/16" thick capsules of Glauber's salt (see Figure 23). The salt capsules used were manufactured by the Penwalt Corporation [1] and had a specified phase-change temperature of 89 °F. The Pelletization and encapsulation process used on these capsules was developed by the Penwalt Corporation under contract with Argonne National Laboratory and is described by Chen et. al. [34]. The pellets used in these runs were double coated to make them more resistant to refrigerant diffusion through the capsule wall. Both "freezing" and "melting" runs were made for flow rates ranging from 0.25 to 4 gallons per minute (GPM).

In the "freezing" runs the phase-change salt capsules were first melted by running hot refrigerant through the packed bed. Typically refrigerant at a temperature in excess of 110 °F was passed through the bed at a flow rate of 4 GPM for one hour. Once the bed was fully melted it was cooled down to approximately 90 °F by running 90 °F

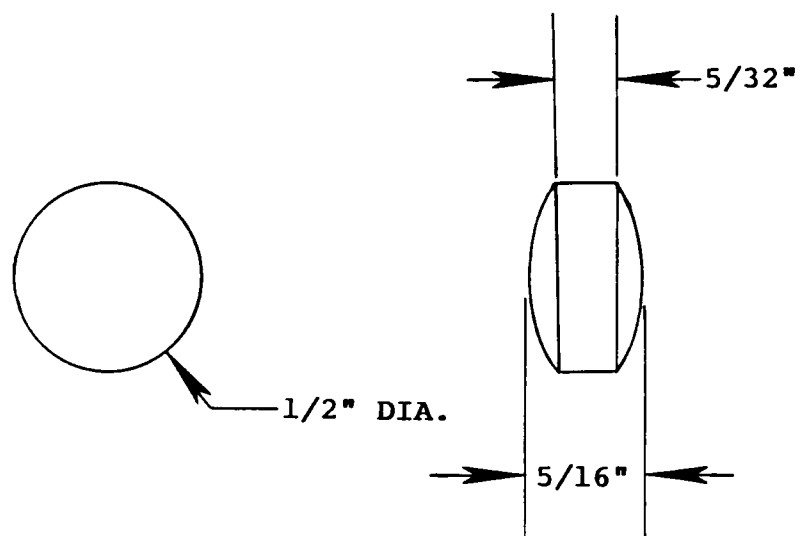


Figure 23. Sketch of Phase-Change Salt Capsule

refrigerant through the bed for one hour. The working fluid was then cooled to the desired bed inlet temperature in the by-pass loop prior to beginning a test run.

Figure 24 shows typical "freezing" temperature curves measured at the five instrumented cross-sections of the bed. This particular run is a 2 GPM "freezing" run. The bed inlet temperature at station 1 shows a sharp drop in fluid temperature starting at time = 0 seconds. The short flat section in the bed inlet temperature curve at time = 150 sec. is caused by the pellets undergoing phase change, (station one is 1" inside the packed bed). The temperature curve at station 2 clearly shows that supercooling of the liquid salt occurred, followed by a constant temperature freeze period. The other three stations in the bed clearly show a typical constant temperature freeze period followed by sensible cooling of the solid phase salt capsules. A comparison of the exit temperature curve, station 5, to the station immediately upstream of the exit, station 4, indicates that during a portion of the constant temperature phase-change period of the run there was no rise in working fluid temperature between these two stations. This indicates that the working fluid had attained pellet temperature by the time it attained station 4. Furthermore it can be seen that phase-change took place at approximately 78 °F rather than the specified 89 °F. Figure 25 shows the temperature

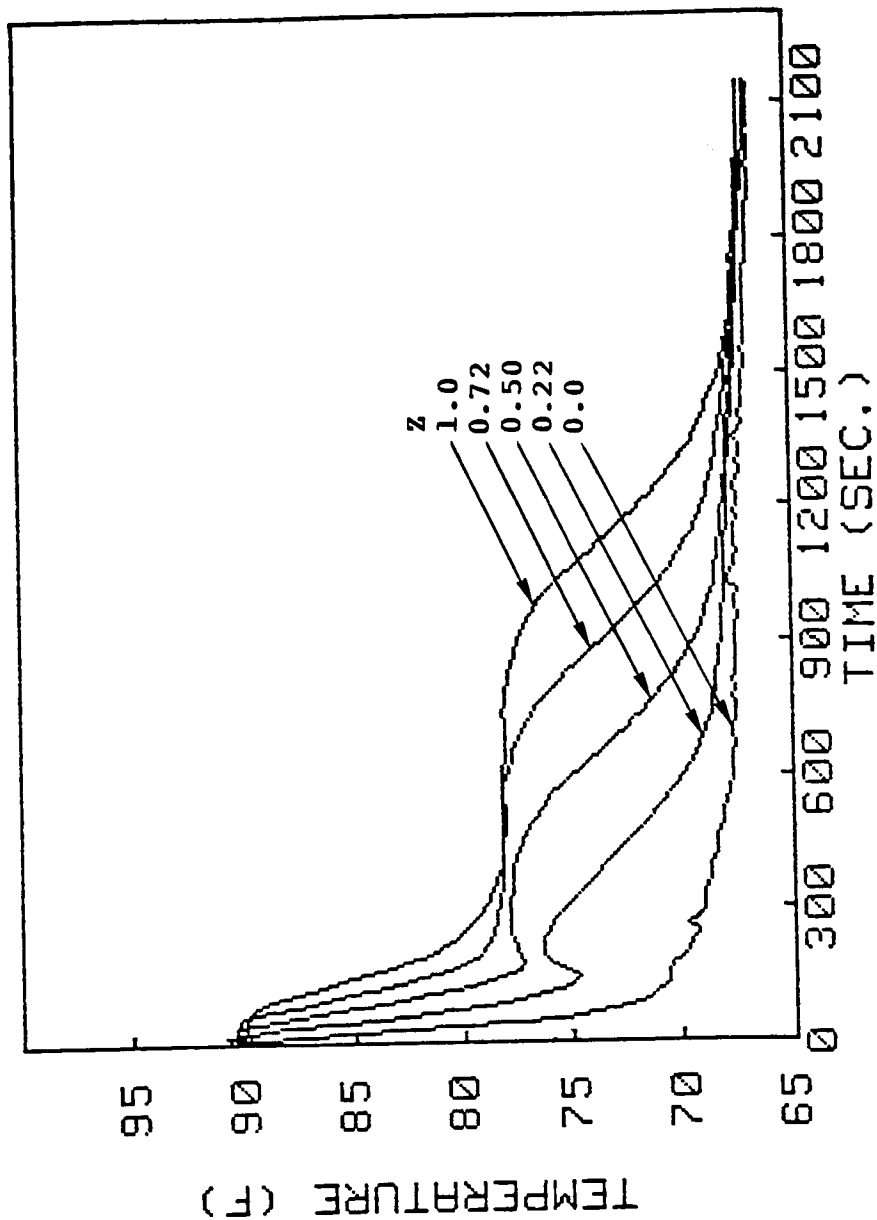


Figure 24. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 2.0 GPM

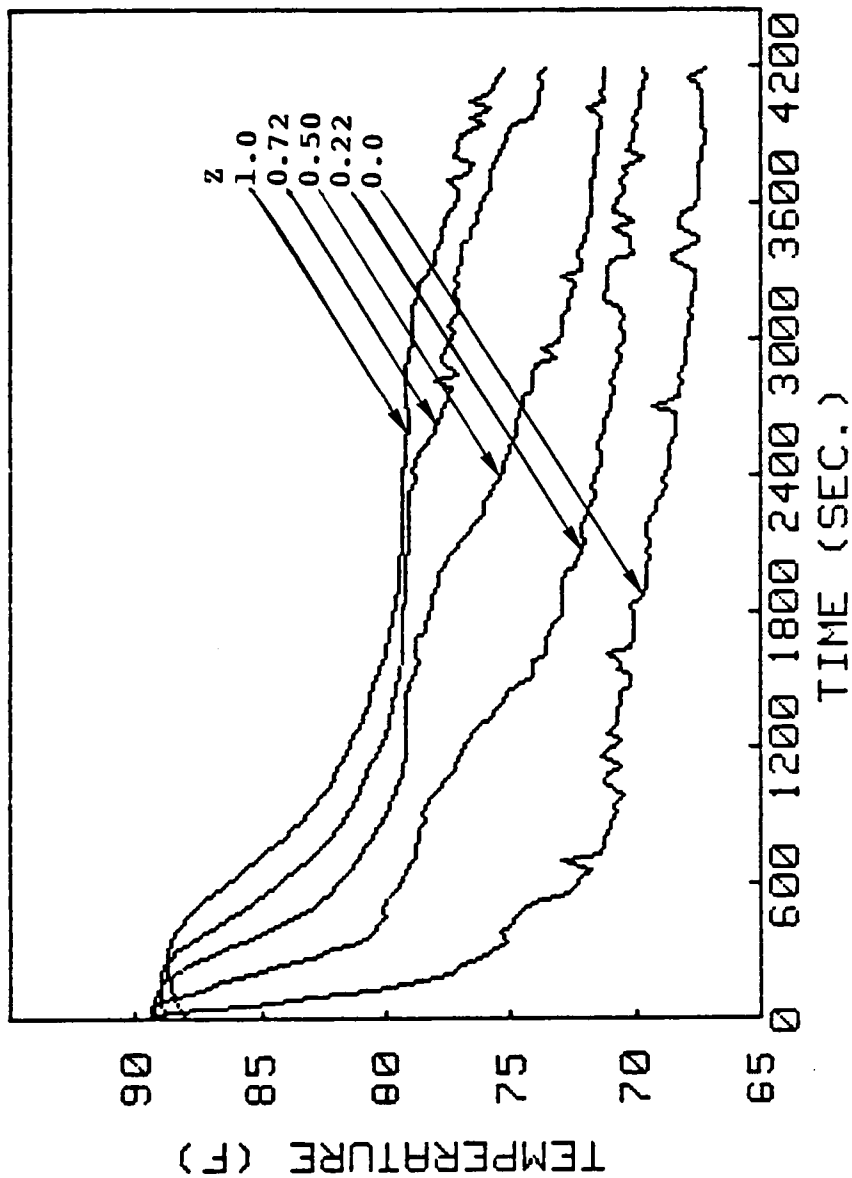


Figure 25. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 0.5 GPM

distribution for the case of a "freezing" run with downflow of cold liquid through the packed bed at a flowrate of 0.5 GPM; whereas Figure 26 shows the temperature distribution for the case of a "freezing" run with upflow of cold liquid through the bed at a flowrate of 0.5 GPM. When comparing these two runs it can be seen that the case of downflow of cold liquid through the bed, Figure 25, is unstable. This observed instability is due to buoyancy induced flow reversals. Figures 27 and 28 show the cases of downflow and upflow at a flowrate of 1.0 GPM. These Figures indicate that at flowrates of 1.0 GPM or more, the system is stable in both flow directions. The transient temperature distribution at a flowrates of 4.0 GPM is shown in Figure 29.

In the "melting" runs the phase change salt capsules were first frozen by running cold refrigerant through the packed bed. Typically refrigerant at a temperature below 80 °F was run through the bed at a flow rate of 4 GPM for one hour. Once the bed was fully frozen, it was heated to approximately 88 °F by running 88 °F refrigerant through the bed for one hour. The working fluid was then heated to the desired bed inlet-temperature in the by-pass loop prior to beginning a test run.

Figure 30 shows typical "melting" transient temperature curves measured at the five instrumented cross-sections of the bed. This particular run is a 0.25

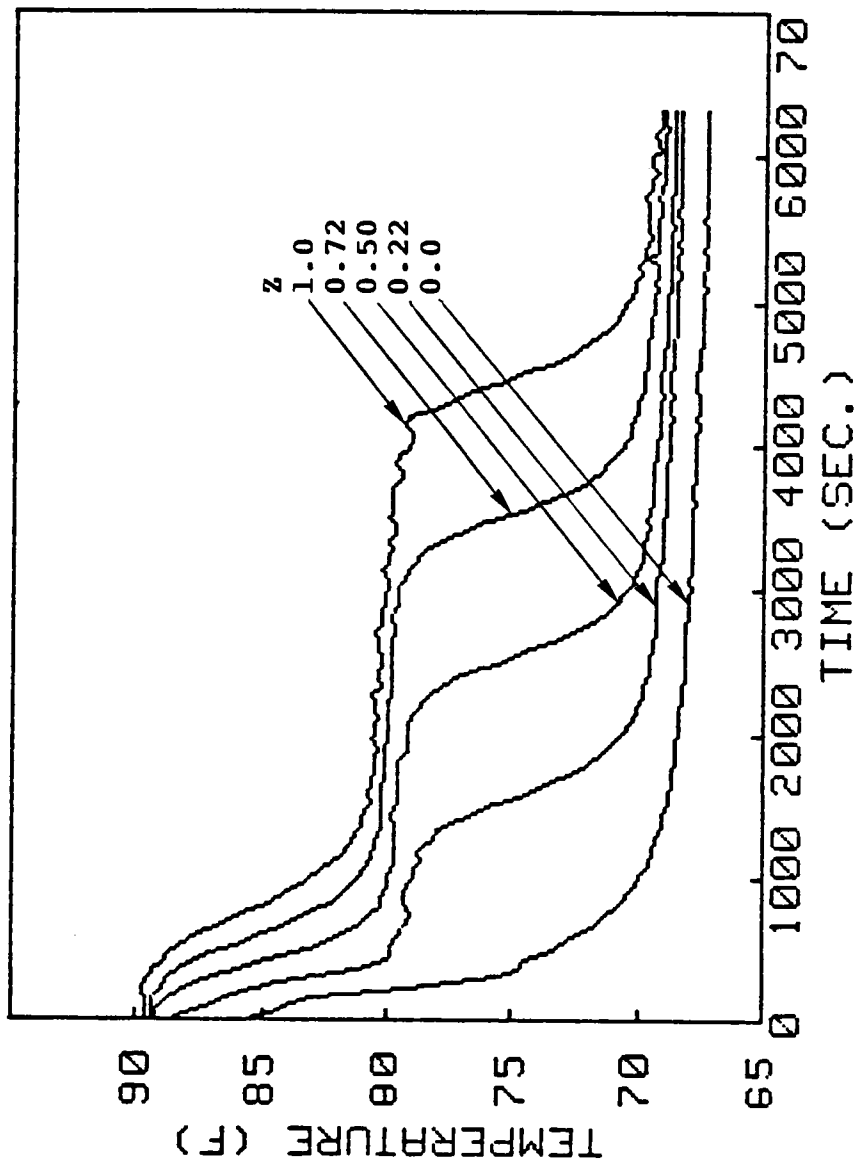


Figure 26. Transient Measurement of Temperature Distribution, "Freezing" Run, Upflow 0.5 GPM

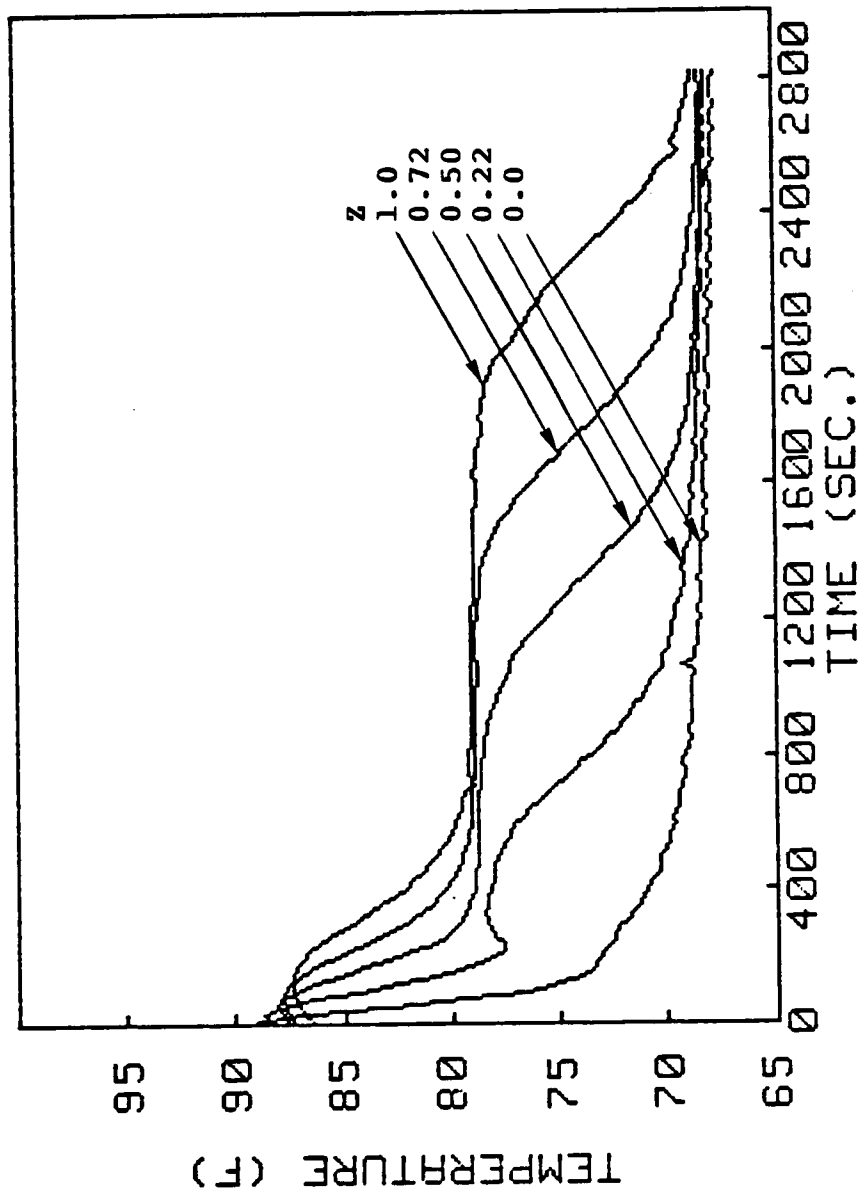


Figure 27. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 1.0 GPM

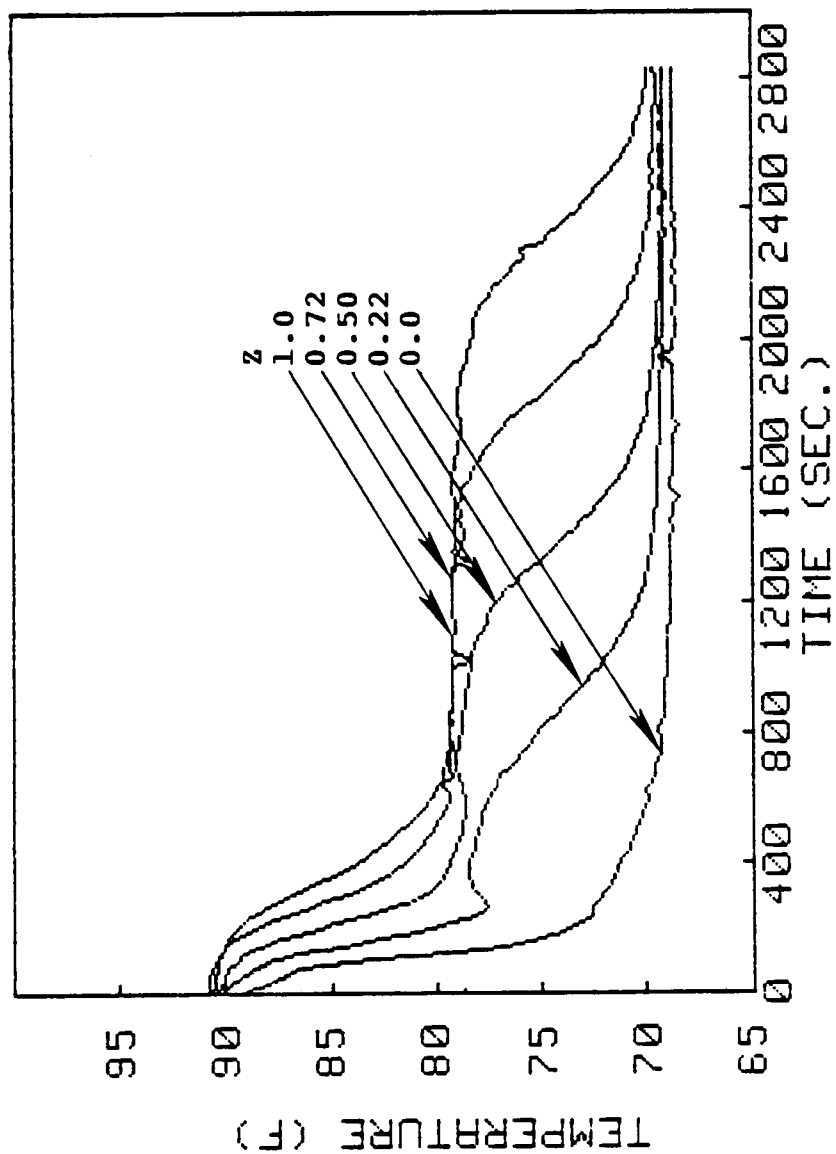


Figure 28. Transient Measurement of Temperature Distribution, "Freezing" Run, Upflow 1.0 GPM

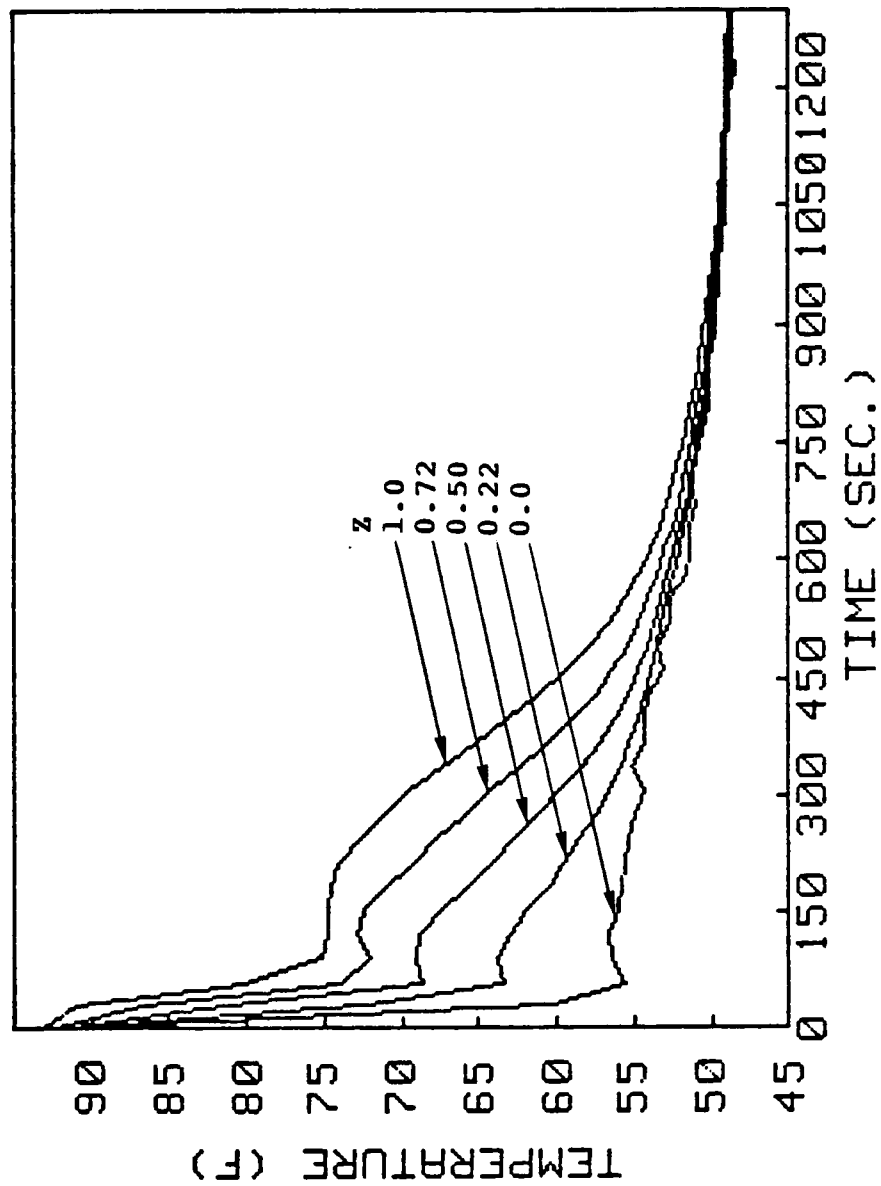


Figure 29. Transient Measurement of Temperature Distribution, "Freezing" Run, Downflow 4.0 GPM

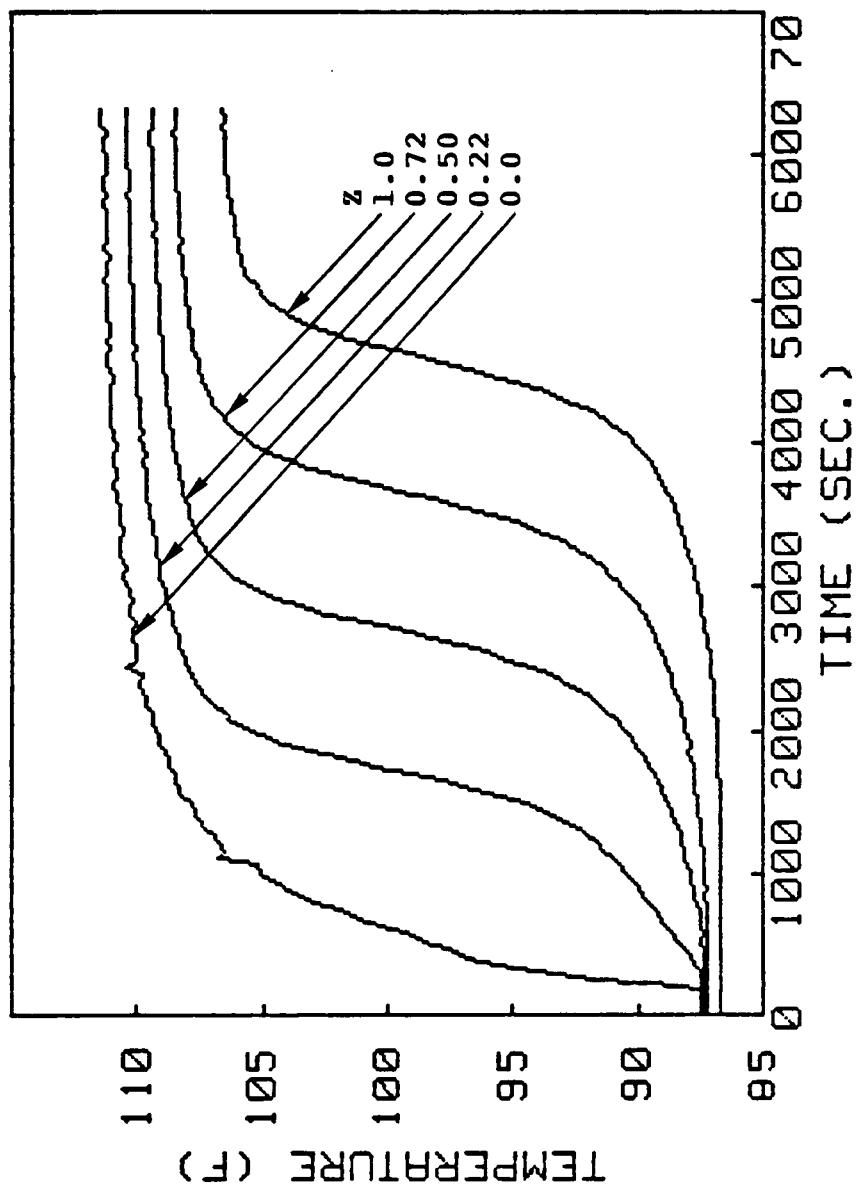


Figure 30. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 0.25 GPM

GPM "melting" run. The bed inlet temperature at station 1 shows a sharp rise in fluid temperature starting at time = 0. In contrast to the "freezing" runs the "melting" runs do not show a constant temperature duration of phase change. The transient temperature distribution for "melting" runs with flowrates of 0.5, 1.0, 2.0, and 4.0 GPM are shown in Figures 31 through 34, respectively.

It should be noted that the temperature probes used in these experiments allowed contact between the thermocouple junctions and the pellets, and therefore the temperature measurements obtained by these probes are not exact measurements of fluid temperature. Furthermore, the permeability of the salt capsule coating material allowed salt to leak out of the capsules and refrigerant to leak into the capsules making it very difficult to know either the exact amount of salt in the bed or the thermal properties of the resulting salt-refrigerant mixture. Finally it was later discovered that refrigerant had been leaking out between the thermocouple wire and the teflon thermocouple wire insulation during these pressurized runs. This refrigerant leaked out the ends of the thermocouple wires into the data acquisition/control unit junction cards. Since all measured thermocouple emfs taken during these runs were relative emfs, any non-uniformity in card temperature owing to refrigerant leakage and subsequent evaporation would lead to errors

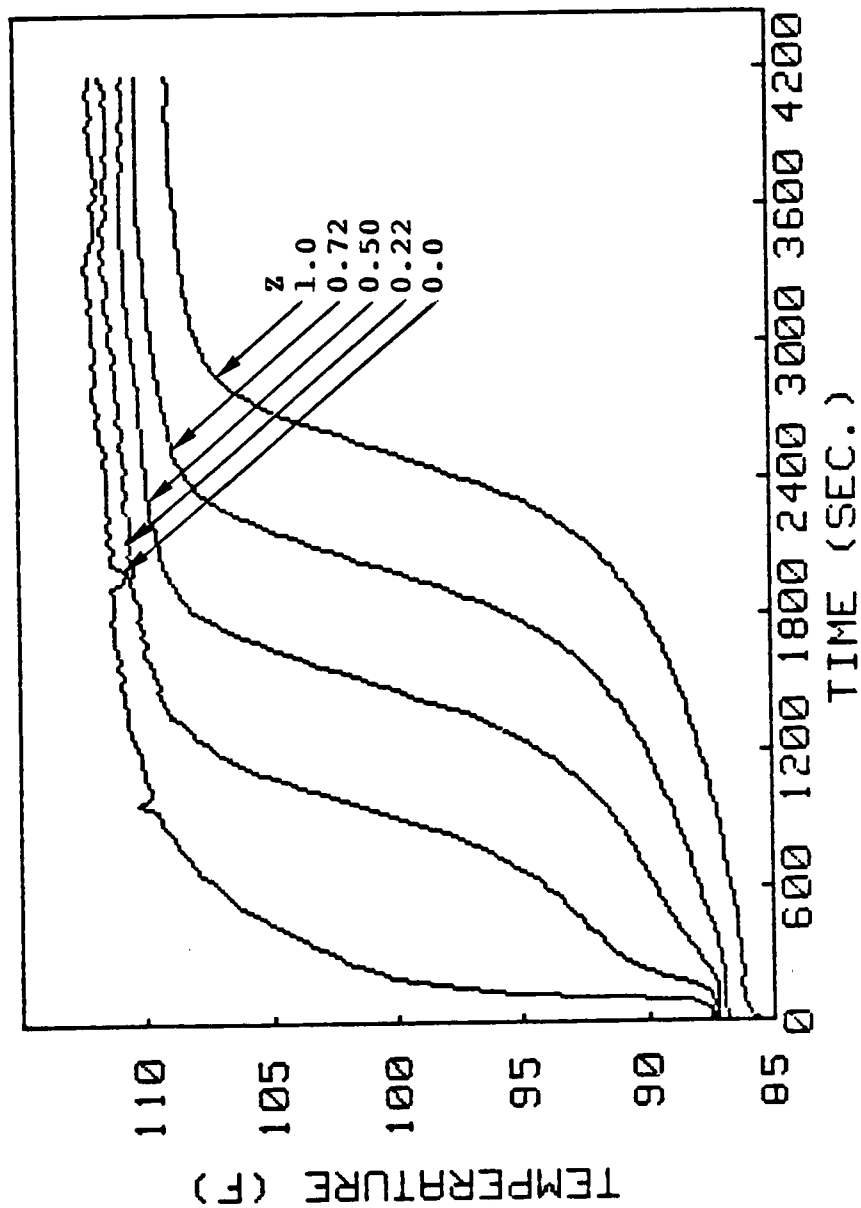


Figure 31. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 0.50 GPM

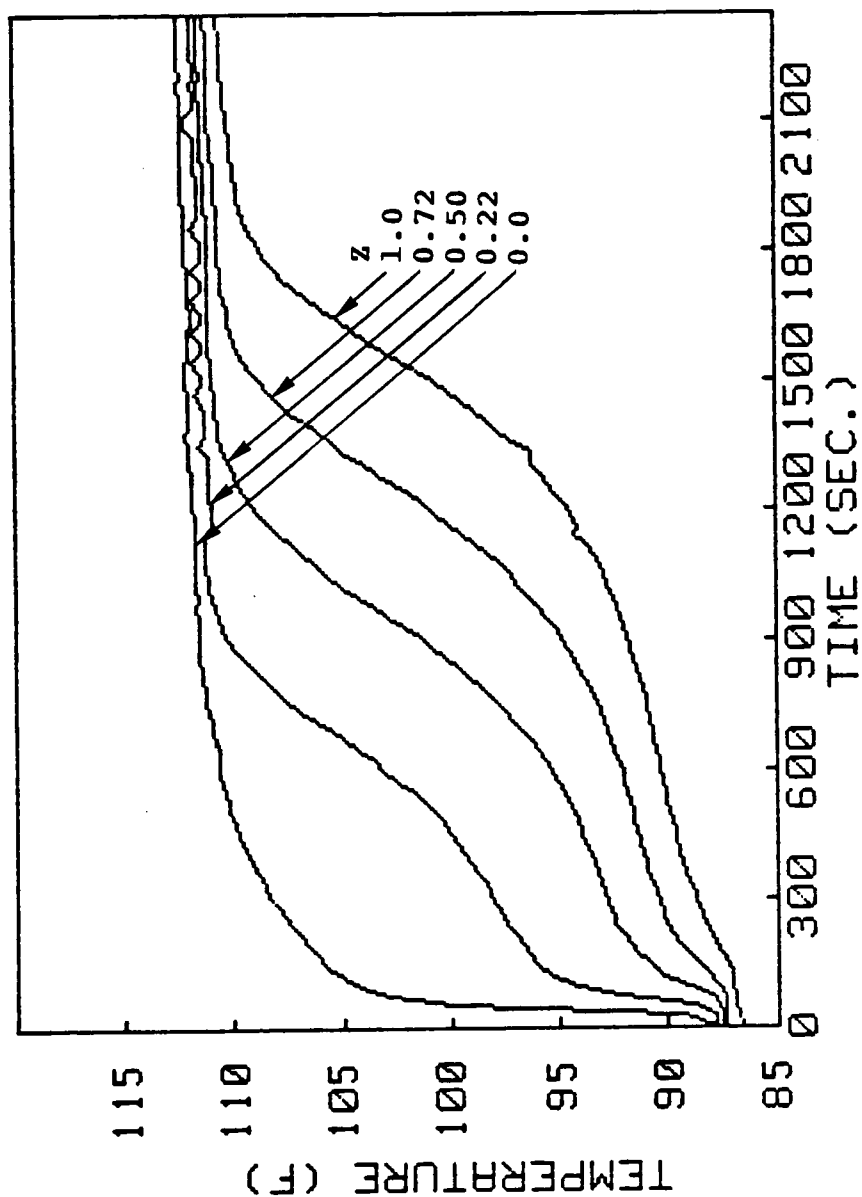


Figure 32. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 1.0 GPM

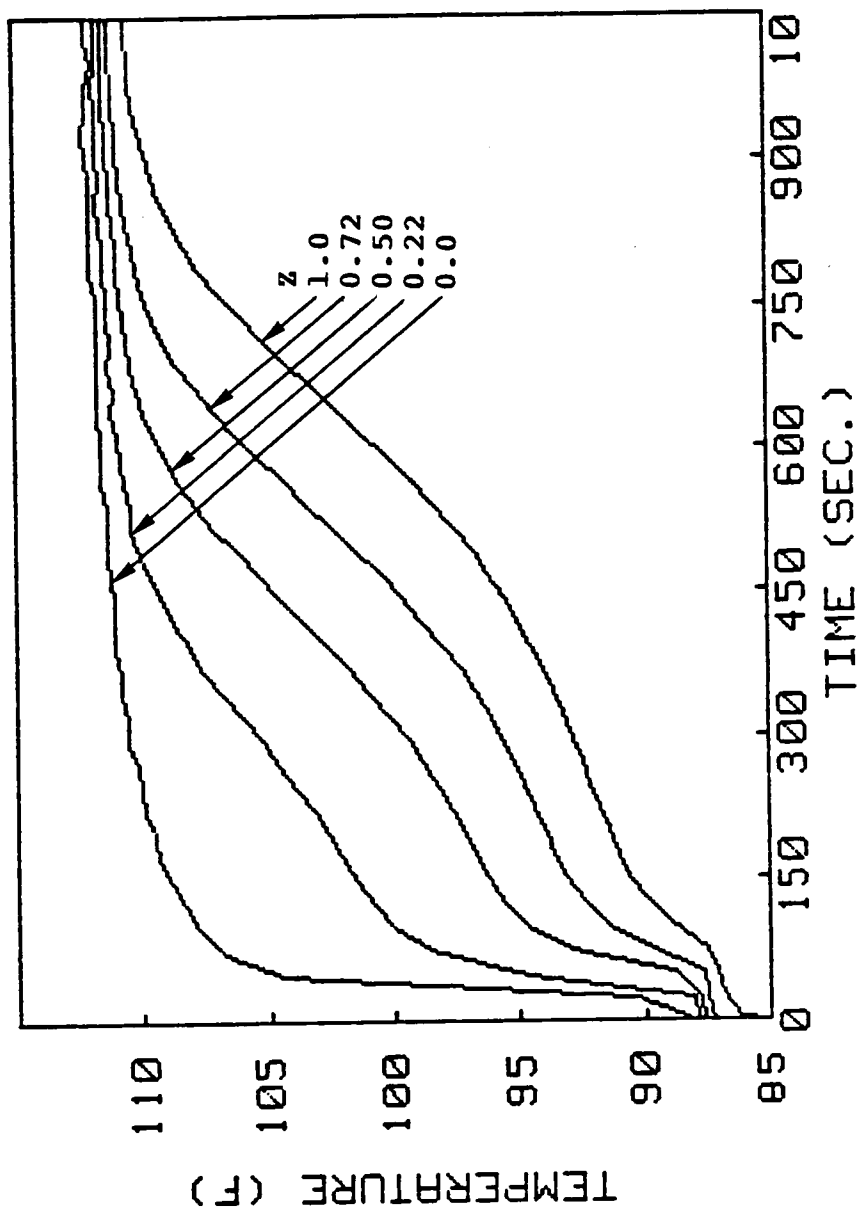


Figure 33. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 2.0 GPM

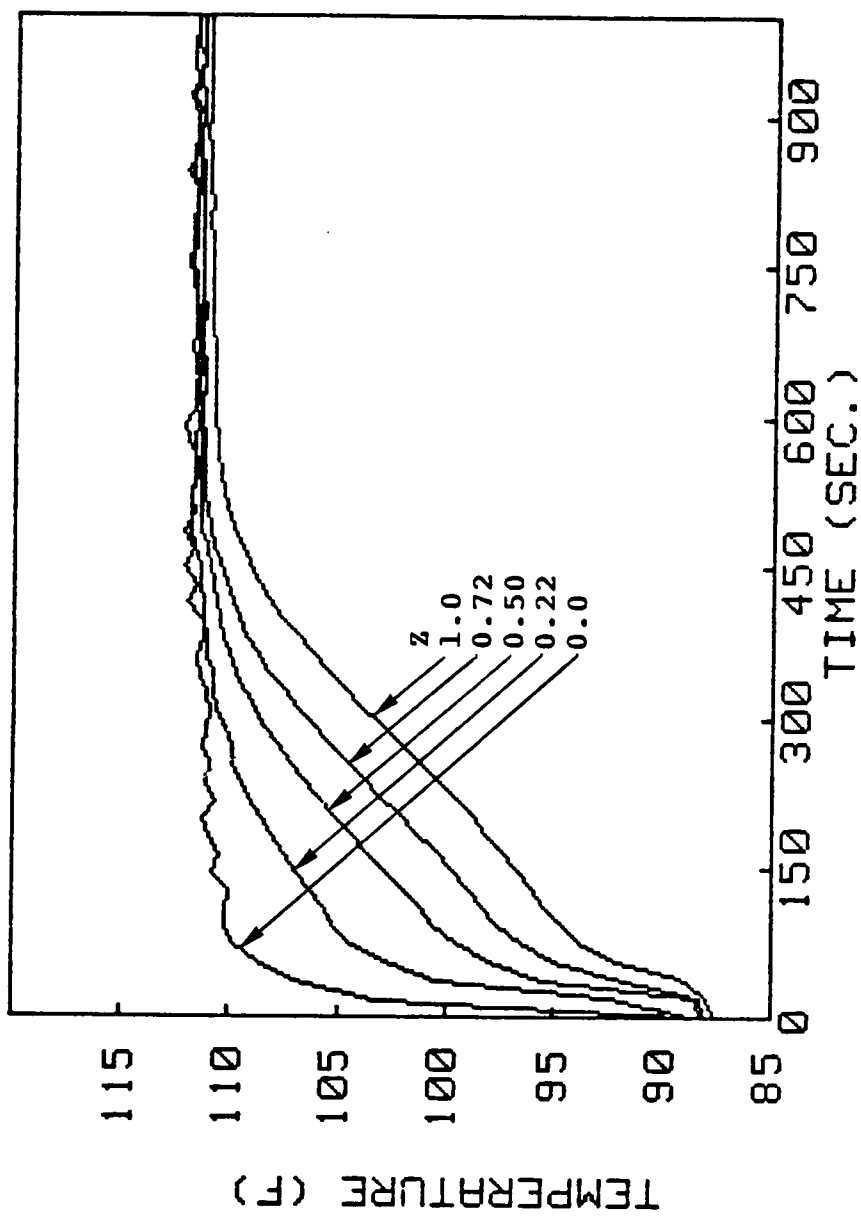


Figure 34. Transient Measurement of Temperature Distribution, "Melting" Run, Downflow 4.0 GPM

when the reference emf was subtracted out to calculate an absolute emf. The above sources of error notwithstanding, energy balances based on the time/temperature data taken during these runs indicate that the salt did undergo phase-change in both the "freezing" and the "melting" runs. If phase-change did indeed take place during the "melting" runs, the temperature at which phase-change took place is not clear but it had to have been above 88 °F since the bed initial temperature was 88 °F in the "melting" runs. This would imply an unrealistically large phase-change temperature range (77-88 °F) for the salt used during these tests further suggesting that the results derived from the temperature data taken during these test are highly uncertain.

Figure 35 shows typical bed inlet and exit absolute pressure measurements. These curves indicate that the frictional pressure drop across the bed is rather small and that the difference between bed inlet and exit pressure is essentially hydrostatic.

Due to the fact that the results obtained from the dynamic calorimeter technique used to measure heat transfer coefficients during these runs proved unreliable, no usable heat transfer data were obtained during these runs.

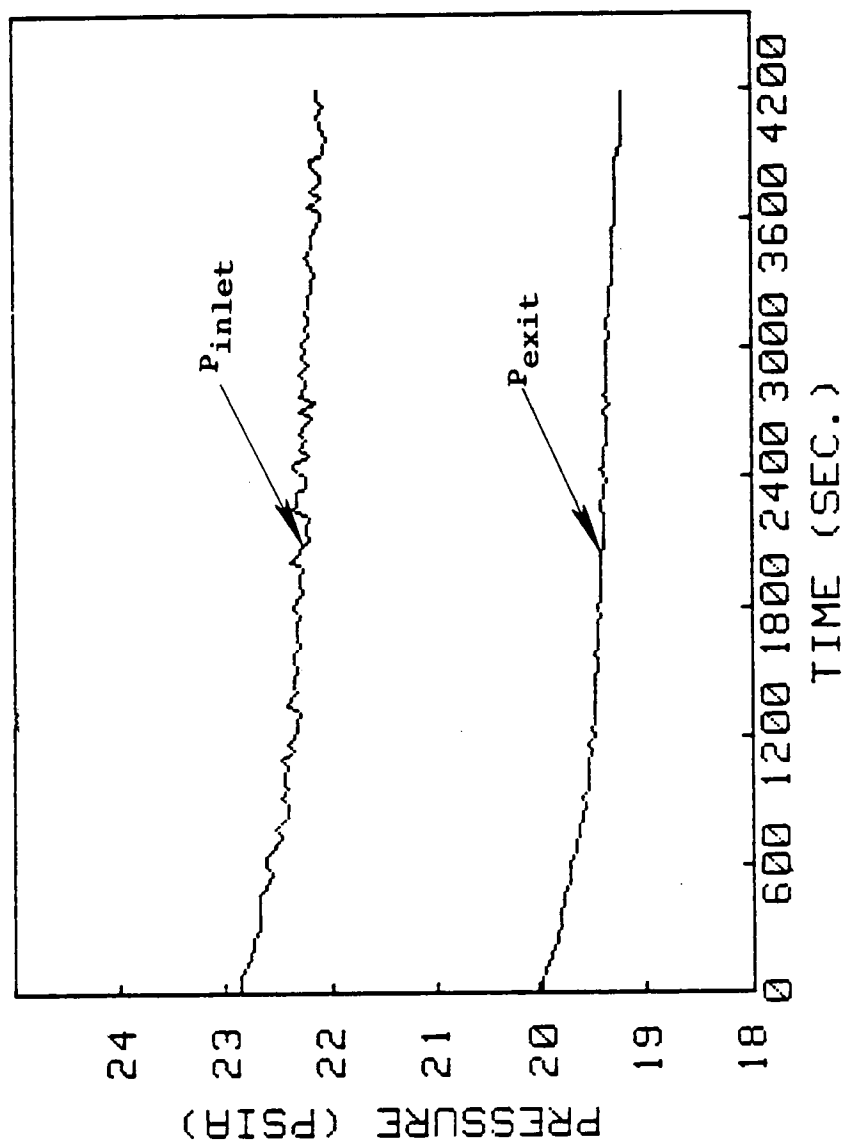


Figure 35. Typical Bed Inlet and Exit Pressures

A cast acrylic storage bed was used during these runs. To allow for flow visualization no insulation was placed on the test section during these runs. Because of the lack of insulation on the test section the losses were high during these runs, and the exit temperature failed to reach inlet temperature values at steady state.

Experiments with PWC-73 and a Single- Phase Working Fluid

Figures 36 through 39 show the measurements of the transient temperature distribution, energy absorbed, and heat transfer coefficient for a bed initially at a low uniform temperature that is subsequently heated by hot liquid R-113. In a like manner, the measurements for a bed initially at a high uniform temperature that is cooled by cold liquid R-113 are presented in Figure 40 through 43.

Figure 44 shows a comparison of the measured Nusselt numbers obtained in these runs to those obtained in the single-phase working fluid runs for glass spheres, (see pages 28-45). From this Figure it can be seen that at low Reynolds numbers the Nusselt numbers obtained for the glass spheres are larger than those obtained with the PWC-73 encapsulated salt pellets, whereas at high Reynolds numbers the Nusselt numbers obtained for the glass spheres are lower than those obtained with the PWC-73 pellets.

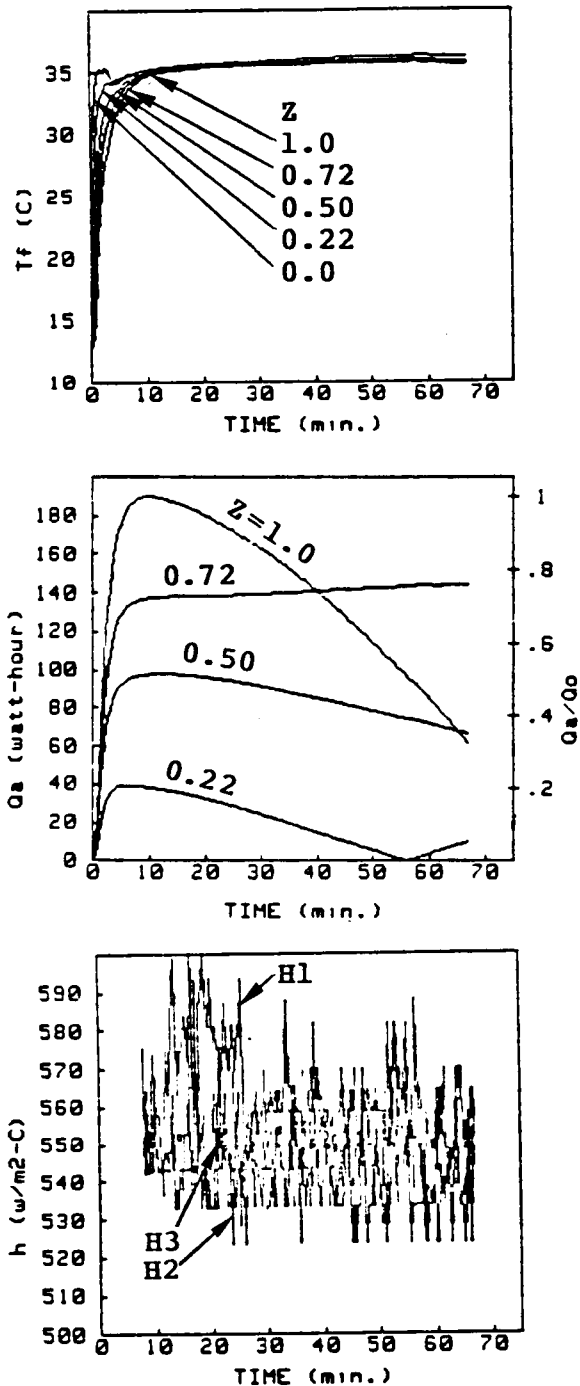


Figure 36. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 701$, (Flowrate 4.0 GPM)

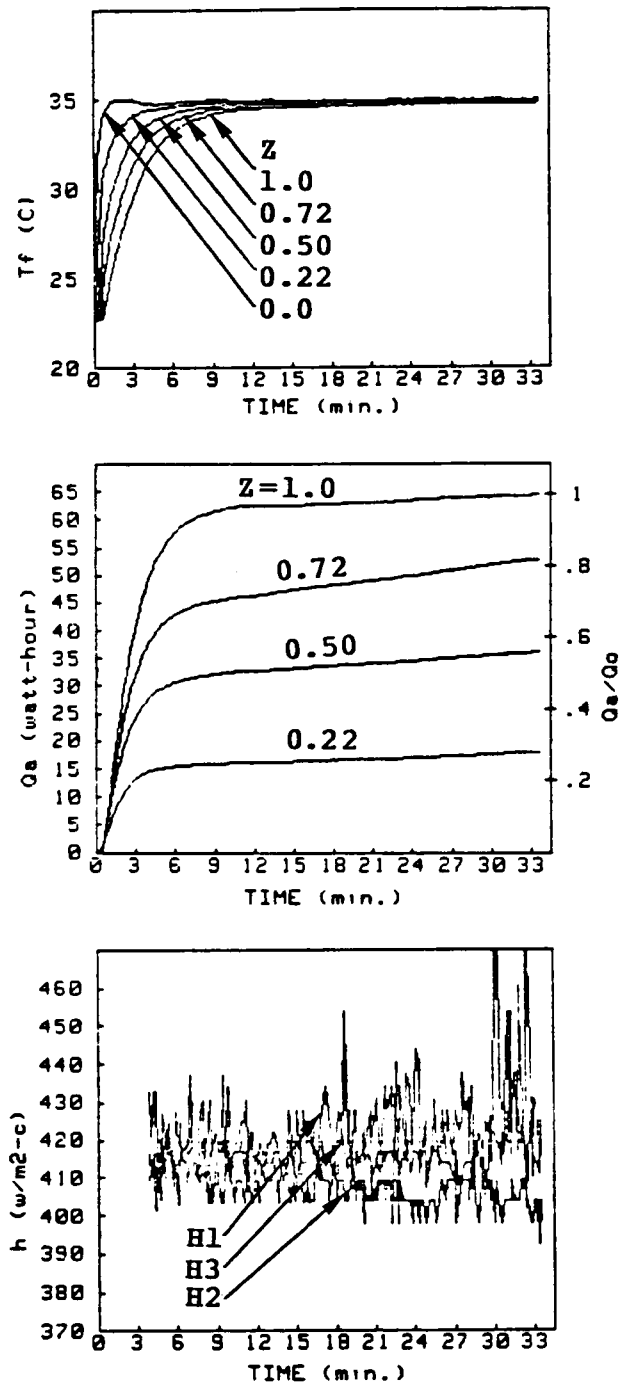


Figure 37. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 354$, (Flowrate 2.0 GPM)

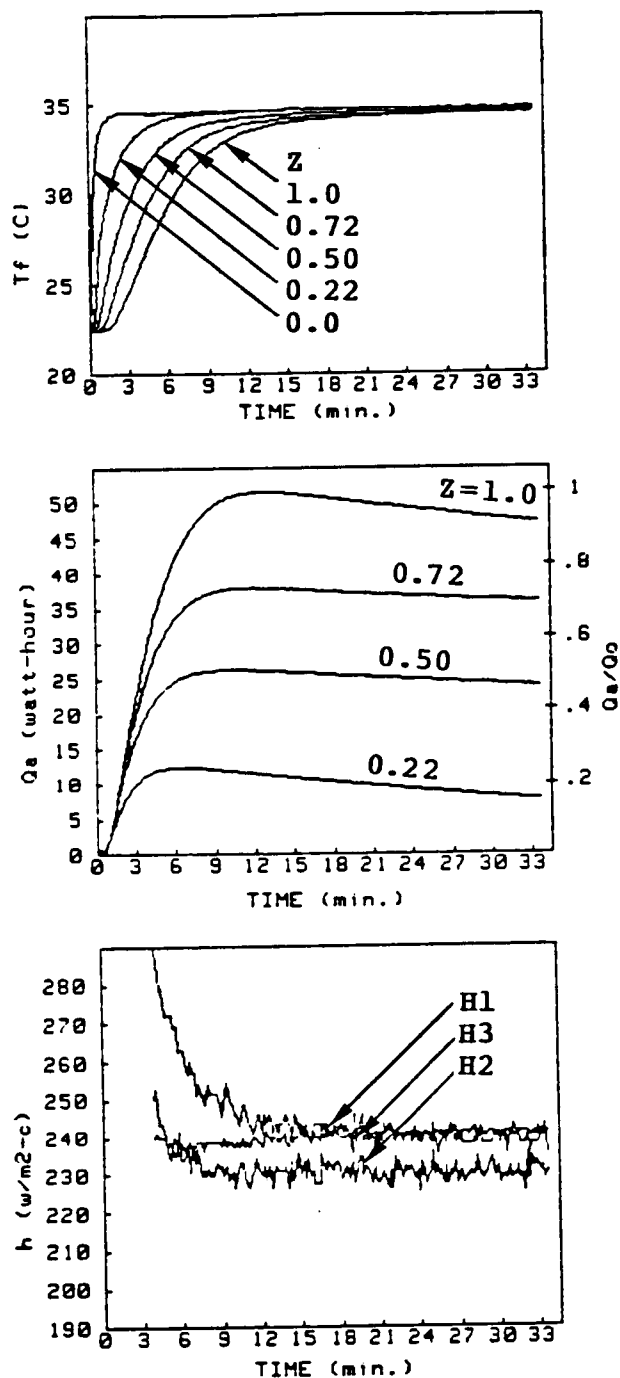


Figure 38. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 87$, (Flowrate 0.5 GPM)

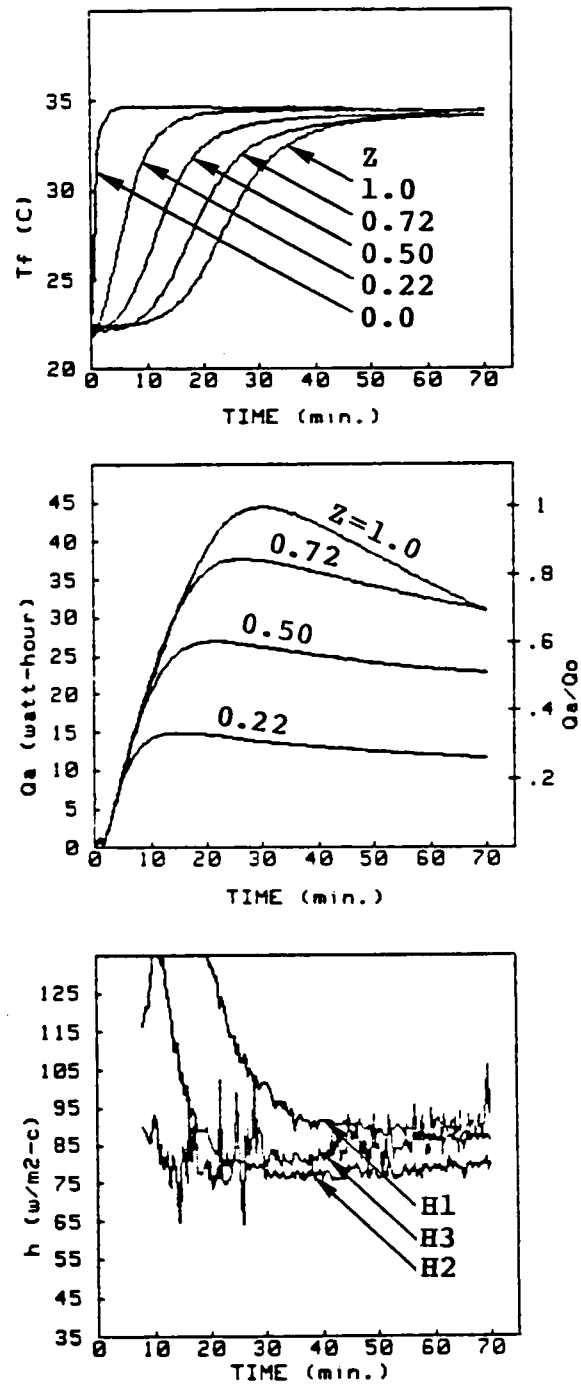


Figure 39. Transient Measurement of Temperature Distribution, Energy Absorbed, and Heat Transfer Coefficient, "Melting" Run, $Re_m = 44$, (Flowrate 0.25 GPM)

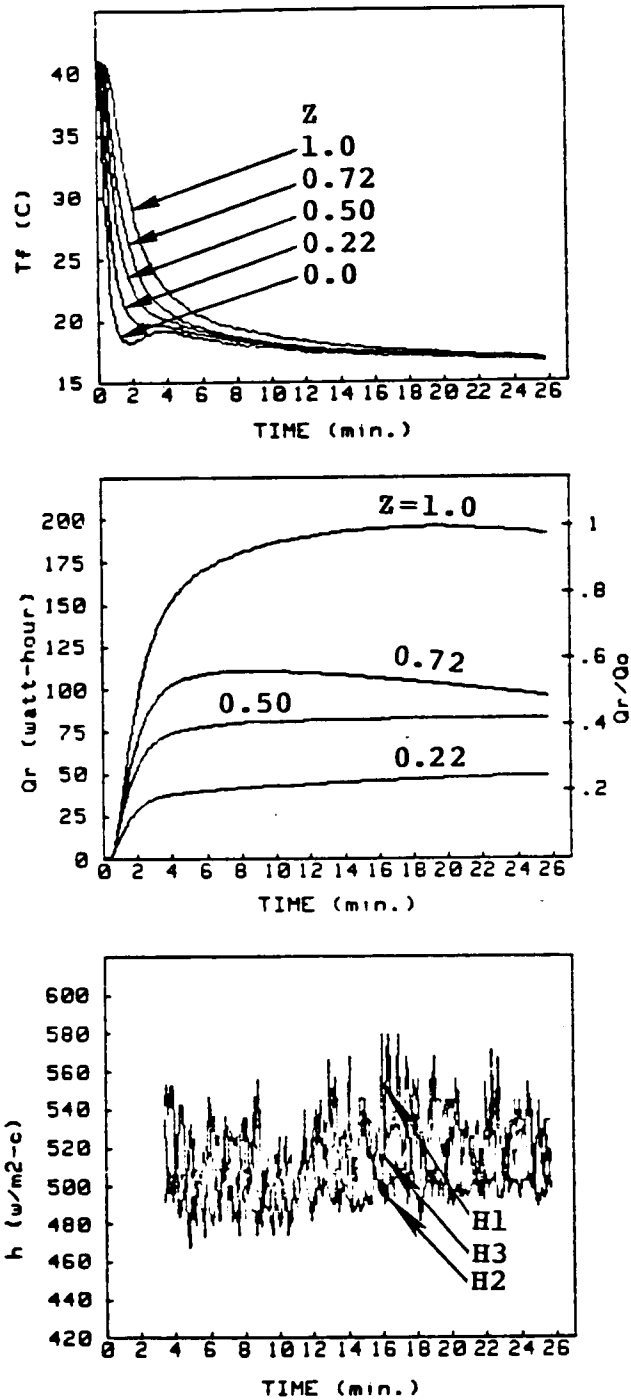


Figure 40. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 583$, (Flowrate 4.0 GPM)

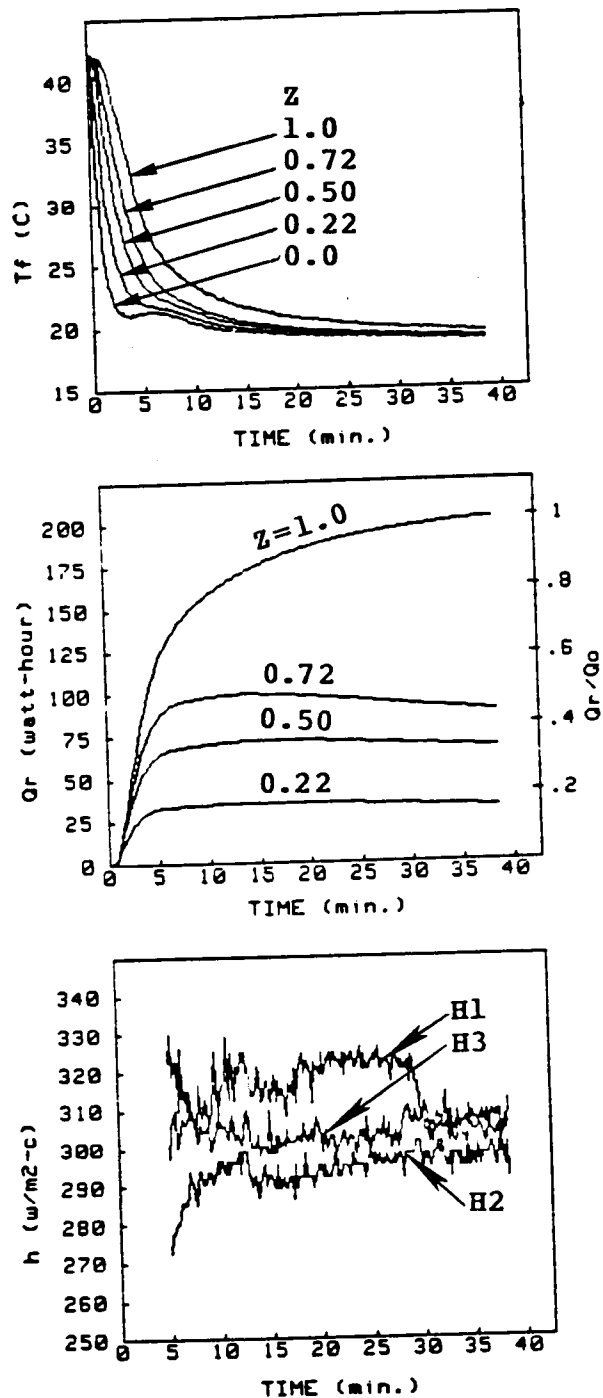


Figure 41. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 296$, (Flowrate 2.0 GPM)

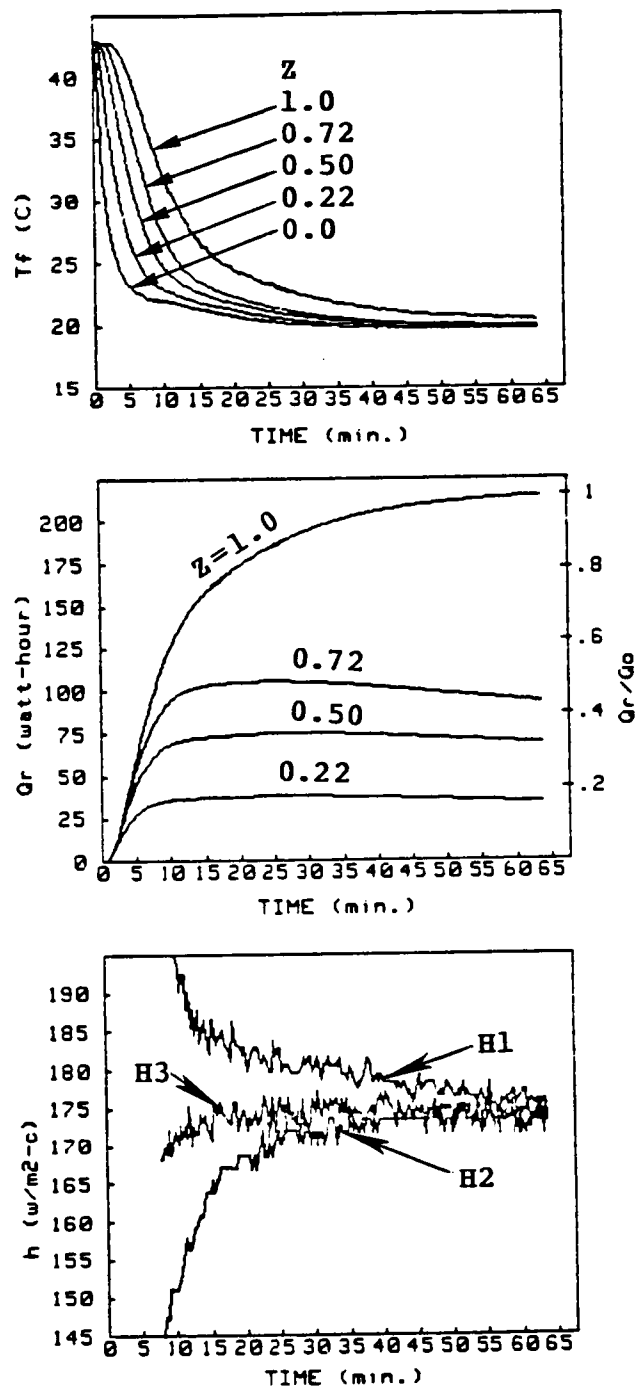


Figure 42. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 147$, (Flowrate 1.0 GPM)

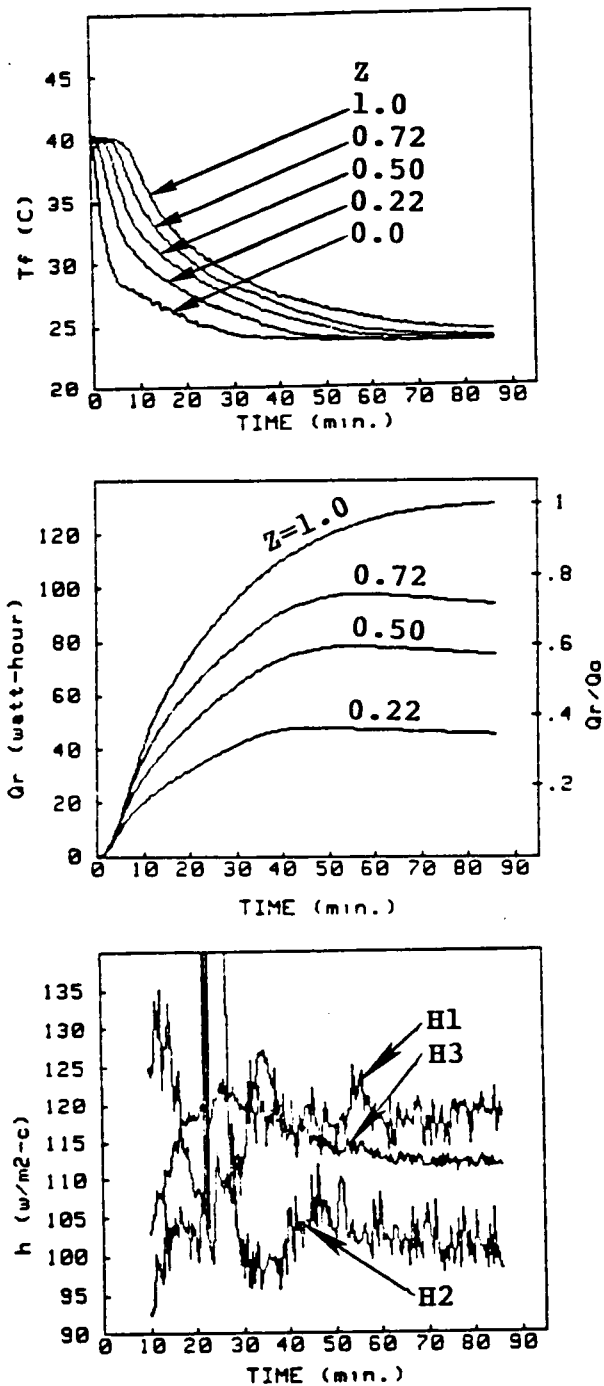


Figure 43. Transient Measurement of Temperature Distribution, Energy Rejected, and Heat Transfer Coefficient, "Freezing" Run, $Re_m = 78$, (Flowrate 0.5 GPM)

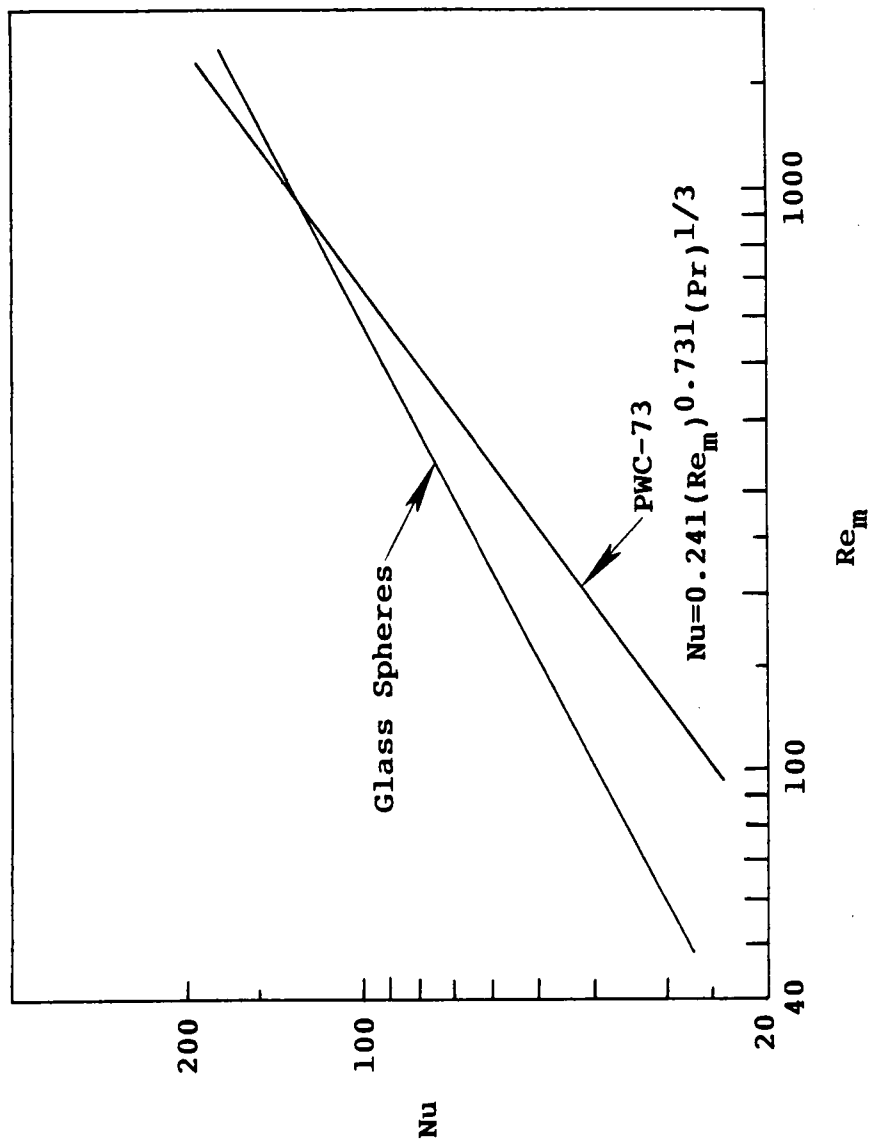


Figure 44. Comparison of Single-Phase Working Fluid Runs (Glass Spheres and PWC-73)

This observed difference in Nusselt versus Reynolds number can be attributed to the difference in the geometry of the two packing materials, and to the difference between the two beds used in these runs. The glass sphere data were measured in an aluminum bed whereas the PWC-73 data were measured in a cast-acrylic bed. It should be noted, however, that since all Nusselt numbers presented in this study were determined under steady state conditions, and that heat losses in this experiment were believed to be negligible small there was no heat transfer to the bed walls and thus the bed wall influence was negligible. Therefore, the differences shown in Figure 44 must be due to the differences in bed packing materials.

The heat absorbed (Q_a) or heat rejected (Q_r) curves, shown in Figures 36 through 43, pages 113-120, were calculated from an integration of the fluid temperature history. The heat capacity of the cast acrylic wall and that of the refrigerant were taken into account. From these Figures it can be seen that there were serious instrumentation problems during these test. Table 4 gives a summary of the PWC-73 runs performed in this study. As was true for the case of the PWC-89 data, (see pages 94-112), there were numerous possible instrumentation problems in these runs, and therefore care should be taken when interpreting these data.

Table 4. Summary of Single-Phase Working Fluid Runs
(R-113, PWC-73)

Mass Flowrate (lbm/min)	Volumetric Flowrate (GPM)	Mode	Re _m	Nu	Q _O (Btu)	Q ₁ (Btu)	% Difference $(1 - \frac{Q_1}{Q_O}) \times 100$
51.47	4.0	heating	701	97	648	380	70
26.36	2.0	heating	354	73	215	207	4
6.32	0.5	heating	87	22	171	207	17
3.31	0.25	heating	44	15	153	207	26
52.18	4.0	cooling	583	82	682	397	72
26.79	2.0	cooling	296	49	682	397	72
13.54	1.0	cooling	147	28	682	397	72
6.62	0.5	cooling	78	17	409	260	57

APPENDIX C

INSTRUMENTATION

Flow Meter Calibration Data

Two turbine flowmeters were used for flow rate measurements in this study. The high flow rate flowmeter used was a Fischer Porter flowmeter model 10C1505, serial number 5606A3583A3. This meter was originally specified to be calibrated on a fluid of 1 centistoke viscosity over a range of 1.6 to 24 GPM. The calibration constant given was 1505 pulses per gallon. The low flow rate flowmeter used in this study was a Foxboro flowmeter model 1/2-2-81T3C1, serial number 125504. This meter was originally calibrated on water at a viscosity of 1 centistoke over a range of 0.23 to 2 GPM. The calibration constant given was 31780 pulses per gallon.

An in-house calibration was performed on both of these flowmeters using a gravity feed tank. The low flow rate flowmeter was calibrated with both water and gasoline whereas the high flow rate meter was calibrated with water only. The calibration data obtained are given in Tables 5 through 7. The viscosity of the water used in these calibration experiments ranged from 1 to 0.80 centistokes where as the viscosity of the gasoline used in these calibration experiments was approximately 0.38

Table 5. Low Flowrate Flowmeter Calibration Data (Water)

f Start	f Stop	Total Mass	Temp.	Count	K	Q
(HZ)	(HZ)	(H ₂ O)	(°F)	(pulses)	(pulses/gal.)	(GPM)
1,072	1,068	32.45	90	124,570	31,876	2.014
1,068	1,065	36.00	90	138,197	31,876	2.007
1,065	1,062	32.90	88	128,858	31,902	2,007
950	945	56.65	93	218,024	31,995	2.000
945	942	56.20	82	217,012	32,108	1.763
950	945	58.25	82	224,450	32,039	1.774
802	795	54.45	83	209,563	31,996	1.497
803	796	52.30	93	201,476	32,026	1.498
800	796	52.50	84	201,491	31,900	1,501
532	530	39.85	84	153,338	31,983	0.996
533	532	40.15	84	154,084	31,899	1.002
535	533	39.70	84	152,744	31,980	1.003
267	267	26.35	85	101,322	31,957	0.501
268	267	26.15	85	100,835	32,046	0.500
267	267	26.60	85	102,349	31,978	0.501
131	131	26.45	85	101,278	31,823	0.247
131	131	26.30	85	100,601	31,790	0.247
131	131	26.25	85	100,656	31,868	0.247
103	103	13.25	85	50,684	31,791	0.194
103	103	13.25	86	50,682	31,784	0.194
103	103	13.25	86	50,630	31,751	0.194

Table 5. (Continued)

f Start	f Stop	Total Mass	Temp.	Count	K	Q
(HZ)	(HZ)	(H ₂ O)	(°F)	(pulses)	(pulses/gal.)	(GPM)
67	67	13.30	87	50,382	31,437	0.128
67	67	13.30	87	50,294	31,416	0.128
67	67	13.30	88	50,334	31,435	0.128
47	47	8.25	88	30,724	30,933	0.091
47	47	8.95	88	33,095	30,714	0.092

Table 6. Low Flowrate Flowmeter Calibration Data
(Gasoline)

f Start	f Stop	Total Mass	Temp.	Count	K	Q
(HZ)	(HZ)	(H ₂ O)	(°F)	(pulses)	(pulses/gal.)	(GPM)
815	781	9.80	82	58,643	32,889	1.413
805	750	8.30	80	49,637	32,868	1.377
731	705	8.30	80	58,324	32,877	1.272
642	630	8.25	80	49,256	32,812	1.129
520	514	6.65	80	39,742	32,845	0.917
418	412	6.45	80	38,521	32,822	0.736
323	320	5.75	80	34,334	32,816	0.570
225	224	5.65	80	33,720	32,799	0.399
103	102	5.75	80	34,077	32,563	0.183
70	70	5.65	80	34,456	32,535	0.125
41	41	2.95	80	17,201	32,021	0.074

Table 7. High Flowrate Flowmeter Calibration Data
(Water)

f Start	f Stop	Total Mass	Temp.	Count	K	Q
(HZ)	(HZ)	(H ₂ O)	(°F)	(pulses)	(pulses/gal.)	(GPM)
84	80	69.1	83	12,513	1,508	3.263
70	70	64.6	73	11,679	1,506	2.789
60	60	64.7	74	11,673	1,502	2.397
50	51	65.9	75	11,792	1,490	2.034
40	40	58.1	72	10,242	1,468	1.635

centistokes. Over the range of temperatures encountered in this study, 50-80 F, the viscosity of the R-113 working fluid ranged from 0.51 to 0.41 centistokes.

Figure 45 shows a plot of the calibration data obtained for the low flow rate flowmeter. Based on these data linear interpolation was used to obtain corrected calibrations constants at the average viscosity of the refrigerant used in this study. (It should be noted that the two curves shown in Figure 45 differ by only 5% and that an uncertainty in K of 2% was used in this study.) Thus the calibration constants for the low flow rate flowmeter was corrected to 33,600 pulses per gallon and that for the high flow rate meter was corrected to 1560 pulses per gallon. These corrections were made to compensate for the fact that the viscosity of the refrigerant used in this study is lower than the viscosity of the fluid that was used in the original calibration of the meters.

Pressure Transducer Calibration Data

Two ± 5 PSID differential strain gauge pressure transducers, serial numbers 2304 and 3324, and two ± 20 PSID differential strain gauge pressure transducers, serial numbers 3336 and 3793, were used in this study. All four of these transducers were manufactured by Statham Laboratories. These transducers were calibrated against a

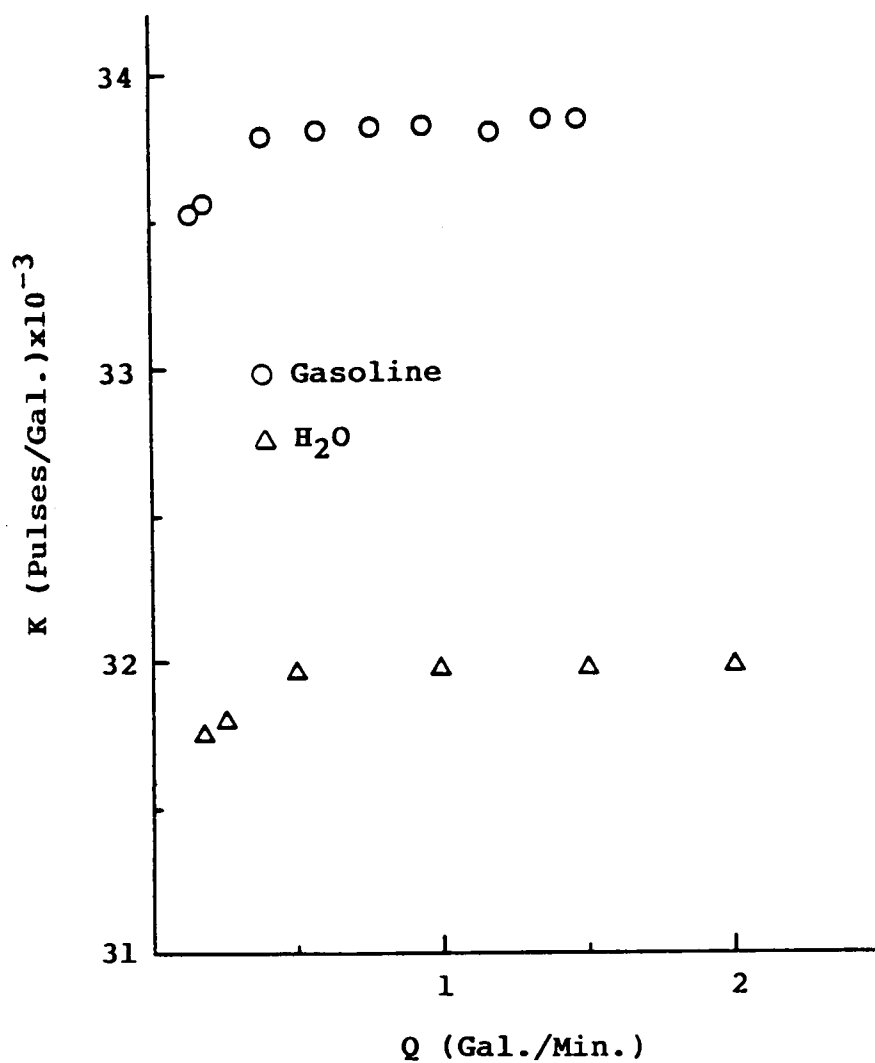


Figure 45. Low Flowrate Flowmeter Calibration Data
Foxboro Flowmeter Model 1/2-2-81T3C1,
S/N 125504

water manometer and a mercury manometer. The water manometer was used for low pressures and the mercury manometer was used for higher pressures. Calibration data for these transducers are given bellow. For each of these transducers a least squares curve fit was used to correlate the data as a linear equations for pressure as functions of voltage. The four equations thus obtained are given below. These equations were used in the data acquisition program.

<u>Transducer</u>	<u>Equation</u>
#3304	$P=198.5V+0.036+ATM$
#3324	$P=203.3V+0.011+ATM$
#3336	$P=800.4V-0.0007+ATM$
#3793	$P=975.6V-0.6532+ATM$

Where P is the absolute pressure in psi, V is the transducer output voltage and ATM is the atmospheric pressure in psi. The deviation between the measured pressures and that predicted by the above equations is less than 1%.

Heat Flux Probes

All of the heat flux probes used in this study were heated with Chromel-A resistance heating wires. Chromel-A was selected for use as resistance heating wire because of it's low temperature coefficient of resistance (the

Pressure Transducer Calibration Data

Serial # 3324 Range 5 psid

ΔP	V
(Inch H ₂ O)	(Volts x 10 ³)

0	0.0390
50	8.912
75	13.374
100	17.814
-50	-8.808
-75	-13.266
-100	-17.920

Serial # 3304 Range 5 psid

ΔP	V
(Inch H ₂ O)	(Volts x 10 ³)

0	-0.215
50	8.898
75	13.458
100	18.020
-50	-9.269
-75	-13.820
-100	-18.340

Serial # 3793 Range 20 psid

ΔP	V
(Inch H ₂ O)	(Volts x 10 ³)

0	0.775
54	2.860
136	5.820
299	11.680
429	16.340
-54	-1.327
-136	-4.360
-299	-10.543
-434	-15.420

Serial # 3336 Range 20 psid

ΔP	V
(Inch H ₂ O)	(Volts x 10 ³)

0	-0.07
50	-2.36
87	-4.04
214	-9.85
383	-17.39
456	-20.66
-40	1.74
-122	5.34

average change in resistance from 68 °F to 500 °F is +4 %). The resistance of each individual wire was obtained by measuring the voltage drop across the wire and current flow through the wire. The voltage and current measurements made on the wires used in this study are given bellow. The resistance values of the heating wires shown in bellow were used in the data acquisition program to calculate the power input to the probes. That is, the current flow through the probes was measured and used with the known resistance of the probe to calculate the power input to the probe.

Heat Flux Probe Calibration Data

<u>Wire #1</u>			<u>Wire #2</u>		
I (amps)	V (volts)	R (ohms)	I (amps)	V (volts)	R (ohms)
0.500	0.216	0.432	0.500	0.216	0.432
1.000	0.433	0.433	1.000	0.433	0.433
2.000	0.868	0.434	2.000	0.869	0.435

<u>Wire #3</u>		
I (amps)	V (volts)	R (ohms)
0.500	0.216	0.432
1.000	0.433	0.433
2.000	0.870	0.435

Equipment

The following is a partial list of the instruments used in this study.

1. Honeywell Digitest Multimeter
Model 333R
Range 0-10 volts
Least count 0.01 volts
2. Ascroft Pressure Gauge
Range 0-160 PSI
Least count 1 PSI
3. Monsanto Counter-Timer
Model 100 C
4. Lambda Regulated Power Supply
Model LC-903
Range 0-40 VDC 0-0.35 ADC
5. Mid-West Instruments Differential Pressure Gauge
Range ± 10 PSID
Least count 0.1 PSI
6. Hallmark Standards Inc. Ampmeter
Model HAC-A
Range 0-10 amps
Least count 0.2 amps
Accuracy 0.5% of full scale
7. Sola Constant Voltage Transformer
CAT. No. 23-22-112
8. The Superior Electric Co. Powerstat
Range 0-130 volts
9. Leeds and Northrup Trendscan 1000
Range -200 - +400 C
Least count 0.1 C
10. Simpson Electric Co. ampmeter
M-7980-2
Range 0 - 1500 amps
Least count 25 amps

11. Statham Laboratories differential strain gauge
pressure transducers
Serials # 3324 and 3304
Range ± 5 PSID
12. Statham Laboratories differential strain gauge
Pressure transducers
Serial # 3336 and 3793
Range ± 20 PSID
13. Hewlett-Packard 9826 Desk top computer
14. Hewlett-Packard 3497A data acquisition/control
unit
Range 0 - 0.99 volts
Accuracy ± 2 microvolts
15. Fisher Porter turbine flowmeter
Model 10C1505, Serial # 5606A3583A3
Range 1.6 - 24 GPM
Accuracy $\pm 0.5\%$
16. Foxboro turbine flowmeter
Model 1/2-2-81T3C1, Serial # 125504
Range 0.23 - 2 GPM
Accuracy $\pm 0.5\%$

APPENDIX D

UNCERTAINTY ANALYSIS

The following is based on the method presented by Kline and McClintock as outlined by Holman [36].

Assuming that a set of measurements has been made and that the uncertainty in each measurement is expressed with the same odds then the uncertainty in a calculated result based on these measurements is given by:

$$W_R = [(\frac{\partial R}{\partial x_1} w_1)^2 + (\frac{\partial R}{\partial x_2} w_2)^2 + \dots + (\frac{\partial R}{\partial x_n} w_n)^2]^{\frac{1}{2}}$$

where W_R is the uncertainty in the result and $w_1, w_2 \dots w_n$ are the uncertainties in the independent variables. W_R is also used to express values of percent uncertainty of R (i.e. $W_R = 2\%$).

1. Uncertainty in the loop volumetric flowrate

$$Q_f = f/K$$

Q_f = volumetric flow rate (gal/sec)

K = flow meter calibration constant (pulses/gal)

f = output frequency of the flowmeter pickup (pulses/sec)

$$\frac{W_{Q_f}}{Q_f} = [(\frac{W_K}{K})^2 + (\frac{W_f}{f})^2]^{\frac{1}{2}} \checkmark$$

Sample calculation:

$K = 33,600$ pulses/gal; $f = 560$ pulses/sec

$Q_f = 1.667 \times 10^{-2}$ gal/sec; $w_k = 2\%$; $w_f = 1\%$

$w_{Q_f} = 2.2\%$

2. Uncertainty in velocity

$$V = Q_f / S_{fr}$$

V = superficial velocity (ft/sec)

Q_f = volumetric flow rate (ft³/sec)

S_{fr} = bed cross-sectional area (ft²)

$$\frac{w_V}{V} = \left[\left(\frac{w_{Q_f}}{Q_f} \right)^2 + \left(\frac{w_{S_{fr}}}{S_{fr}} \right)^2 \right]^{1/2}$$

Sample calculation:

$Q_f = 1\text{GPM} = 2.2280 \times 10^{-3}$ ft³/sec; $S_{fr} = 1.3635 \times 10^{-3}$ ft²

$V = 1.6340 \times 10^{-2}$ ft/sec; $w_{Q_f} = 2.2\%$ (from item 2 above)

$w_{S_{fr}} = 1\%$; $w_V = 2.4\%$

3. Uncertainty in hydraulic diameter

$$D_h = \frac{2D_p \epsilon}{3(1-\epsilon)}$$

D_h = hydraulic diameter (ft)

D_p = pellet diameter (ft)

ϵ = bed void fraction

$$\frac{w_{Dh}}{D_h} = \left[\left(\frac{w_{Dp}}{D_p} \right)^2 + \left\{ \left(\frac{1}{1-\epsilon} \right) \frac{w_\epsilon}{\epsilon} \right\}^2 \right]^{\frac{1}{2}}$$

Sample Calculation:

$$D_p = 5.208 \times 10^{-2} \text{ ft}; \quad \epsilon = 0.4566; \quad D_h = 2.9176 \times 10^{-2} \text{ ft}$$
$$w_{Dp} = 1\%; \quad w_\epsilon = 1\%; \quad w_{Dh} = 2.1\%$$

4. Uncertainty in Reynolds number

$$Re_m = \frac{VD_h\rho}{\mu\epsilon}$$

Re_m = modified Reynolds number

V = superficial velocity (ft/sec)

D_h = hydraulic diameter (ft)

ρ = density of the working fluid (lbm/ft³)

μ = dynamic viscosity of the working fluid (ft-sec/lbm)

ϵ = bed void fraction

$$\frac{w_{Re_m}}{Re_m} = \left[\left(\frac{w_V}{V} \right)^2 + \left(\frac{w_{Dh}}{D_h} \right)^2 + \left(\frac{w_\rho}{\rho} \right)^2 + \left(\frac{w_\mu}{\mu} \right)^2 + \left(\frac{w_\epsilon}{\epsilon} \right)^2 \right]^{\frac{1}{2}}$$

Sample calculation:

$$V = 1.6340 \times 10^{-2} \text{ ft/sec}; \quad D_h = 2.9176 \times 10^{-2} \text{ ft}$$

$$\rho = 94.5 \text{ lbm/ft}^3; \quad \mu = 3.4518 \times 10^{-4} \text{ ft-sec/lbm}$$

$$\epsilon = .4566; \quad Re_m = 285.8; \quad w_V = 2.4\% \text{ (from item 2 above)}$$

$$w_{D_h} = 2.1\% \text{ (from item 3 above); } w_{\rho} = 1\%; w_{\mu} = 1\%$$

$$w_{\epsilon} = 1\%; w_{Re_m} = 3.6\%$$

5. Uncertainty in losses to room due to radial conduction through the fiberglass insulation of the tubing between the resistance heating section and the bed inlet.

$$Q_1 = \frac{2\pi k l}{\ln(r_o/r_i)} \Delta T$$

Q_1 = rate of heat loss between the inlet of the
resistance heating section and the bed inlet
(Btu/hr)

k = thermal conductivity of fiberglass insulation
(BTU/hr-ft-°F)

l = length of pipe between the bed inlet and the inlet of
the resistance heating section (ft)

ΔT = average difference between the inside and outside
surfaces temperatures of the fiberglass insulation
(°F)

r_o = outside radius of the fiberglass insulation (ft)

r_i = inside radius of the fiberglass insulation (ft)

$$\frac{w_{Q_1}}{Q_1} = \left[\left(\frac{w_k}{k} \right)^2 + \left(\frac{w_l}{l} \right)^2 + \left(\frac{w_{\Delta T}}{\Delta T} \right)^2 + \left(\frac{w_r}{r_o \ln(r_o/r_i)} \right)^2 \right. \\ \left. + \left(\frac{w_r}{r_i \ln(r_o/r_i)} \right)^2 \right]^{1/2}$$

Sample calculation:

$K = 0.02 \text{ BTU/hr-ft-}^\circ\text{F}; \quad l = 18.67 \text{ ft}; \quad \Delta T = 51^\circ\text{F}$

$r_o = 0.1667 \text{ ft}; \quad r_i = 0.0417 \text{ ft}; \quad Q_1 = 86.4 \text{ Btu/hr}$

$w_k = 5\%; \quad w_l = 1\%; \quad w_{\Delta T} = 5\%; \quad w_{r_o} = w_{r_i} = 1\%$

$w_{Q_1} = 7.3\%$

NOTE: The average difference between the inside and outside surface temperatures of the fiberglass insulation is based on 8 readings made at evenly spaced intervals along the length of the tubing between the heated section and the bed inlet. (Axial conduction was neglected.)

6. Uncertainty in power input in resistance heating section

$$P = IV$$

P = Power input to working fluid by resistance heating
(watts)

I = current (amperes)

V = voltage drop across resistance heating sections
(volts)

$$\frac{w_P}{P} = \left[\left(\frac{w_I}{I} \right)^2 + \left(\frac{w_V}{V} \right)^2 \right]^{1/2}$$

Sample calculation:

$I = 610 \text{ amps}; \quad V = 3.53 \text{ volts}; \quad P = 2,153.3 \text{ watts}$

$w_I = 2\%; \quad w_V = 1\%; \quad w_P = 2.2\%$

7. Uncertainty in bed inlet quality

$$x_o = \left[\frac{P - Q_1}{\rho_f Q_f} - (hf_o - hf_i) \right] \frac{1}{hfg_o}$$

x_o = Bed inlet quality

P = Powers input by resistance heating (Btu/hr)

Q_1 = Losses to room (Btu/hr)

ρ_f = Density of working fluid (lbm/ft³)

Q_f = Volumetric flow rate (ft³/hr)

hf_o = Enthalpy of working fluid at outlet of electrical resistance heating section (liquid) (Btu/lbm)

hf_i = Enthalpy of working fluid at inlet of electrical resistance heating section (liquid) (Btu/lbm)

hfg_o = Difference between the enthalpy of gas and liquid at the outlet of the heated section (Btu/lbm)

$$\begin{aligned} w_{x_o} = & \left[\left(\frac{1}{\rho_f Q_f hfg_o} w_P \right)^2 + \left(\frac{1}{\rho_f Q_f hfg_o} w_{Q_1} \right)^2 + \left(\frac{P - Q_1}{hfg_o Q_f \rho_f^2} w_{\rho_f} \right)^2 \right. \\ & + \left(\frac{P - Q_1}{hfg_o Q_f \rho_f^2} w_{Q_f} \right)^2 + \left(\frac{1}{hfg_o} w_{hf_o} \right)^2 + \left(\frac{1}{hfg_o} w_{hf_i} \right)^2 \\ & \left. + \left(\left(\frac{P - Q_1}{\rho_f Q_f} - hf_o + hf_i \right) \frac{w_{hfg_o}}{hfg_o^2} \right)^2 \right]^{1/2} \end{aligned}$$

Sample calculation:

$P = 7,347.4$ Btu/hr; $Q_1 = 86.3$ Btu/hr; $\rho_f = 97.4$ lbm/ft³

$Q_f = 2.005$ ft³/hr; $hf_o = 38.05$ Btu/lbm

$hf_i = 24.76$ Btu/lbm; $hfg_o = 61.31$ Btu/lbm; $x_o = 0.390$

$w_p = 2.2\%$ (from item 6 above); $w_{Q1} = 7.3\%$ (from item 5 above);
 $w_{\rho_f} = 1\%$; $w_{Q_f} = 2.2\%$ (from item 1 above)
 $w_{hf_i} = 1\%$; $w_{hf_o} = 1\%$; $w_{hfg_o} = 1\%$; $w_{x_o} = 5.5\%$

8. Uncertainty in axial conduction losses from heat flux probe lead wires.

Calculate the temperature of the heat flux probe heating wire.

$$T_i = q \frac{\ln(r_o/r_i)}{2\pi kl} + T_o$$

q = power input to probe (Btu/hr)

k = thermal conductivity of heating wire insulation
(Btu/hr-ft-°F)

l = length of the probe (ft)

r_i = outside diameter of heating wire (ft)

r_o = outside diameter of heating wire insulation (ft)

T_i = temperature of resistance heating wire (°F)

T_o = outside temperature of heating wire insulation (F°)

Note: Due to the high thermal conductivity of the metal rods used in the heat flux probes the temperature drop between the outside surface of the probe and the outside surface of the heat flux probe heating wire insulation is

negligible, therefore T_o above is taken to be equal to the outside surface temperature of the heat flux probe.

Sample calculation:

$q = 9.4 \text{ Btu/hr}; \quad k = 0.4514 \text{ Btu/hr-ft-}^\circ\text{F}; \quad l = 0.4167 \text{ ft}$

$r_i = 0.0010 \text{ ft}; \quad r_o = 0.0020 \text{ ft}; \quad T_o = 85 \text{ }^\circ\text{F}$

$T_i = 102 \text{ }^\circ\text{F}$

Calculate the uncertainty in axial conduction losses out heating wire leads.

$$q_1 = \frac{Ak}{l} (T_i - T_o)$$

$q_1 = \text{losses due to axial conduction (Btu/hr)}$

$A = \text{cross-sectional area of lead wire (ft}^2\text{)}$

$k = \text{thermal conductivity of lead wire (Btu/hr-ft-}^\circ\text{F)}$

$l = \text{length of lead wire (ft)}$

$T_i = \text{temperature of heating wire (}^\circ\text{F)}$

$T_o = \text{ambient temperature (}^\circ\text{F)}$

$$\frac{w_{q_1}}{q_1} = \left[\left(\frac{w_A}{A} \right)^2 + \left(\frac{w_k}{k} \right)^2 + \left(\frac{w_l}{l} \right)^2 + \left(\frac{w_{T_i}}{T_i - T_o} \right)^2 + \left(\frac{w_{T_o}}{T_i - T_o} \right)^2 \right]^{\frac{1}{2}}$$

Sample calculation:

$A = 3.4088 \times 10^{-6} \text{ ft}^2; \quad K = 220 \text{ Btu/hr-ft-}^\circ\text{F}$

$l = .1667 \text{ ft}; \quad T_i = 102 \text{ }^\circ\text{F}; \quad T_o = T_{\text{room}} = 80 \text{ }^\circ\text{F}$

$q_1 = 0.099 \times 2 = 0.1980 \text{ Btu/hr (two lead wires)}$

$w_A = 1\%; \quad w_K = 1\%; \quad w_l = 10\%$

$$w_{T_i} = w_{T_o} = 5\%; \quad w_{q_1} = 6.1\%$$

9. Uncertainty in power input to probe

$$q = I^2 R - q_1$$

q = power input to probe (watts)

I = current input (amps)

R = resistance of the heating wire (ohms)

q_1 = losses due to axial conduction (watts)

$$w_q = [(2IRw_I)^2 + (I^2 w_R)^2 + (w_{q_1})^2]^{1/2}$$

Sample calculation:

$I = 2.52$ amps; $R = 0.432$ ohms; $q_1 = 0.0580$ watts

$q = 9.4$ Btu/hr; $w_I = .5\%$ full scale = 0.05 amps

$w_R = 5\%$; $w_{q_1} = 6.1\%$ (from item 8 above); $w_q = 6.7\%$

10. Uncertainty in heat transfer coefficient

$$h = q/A\Delta T$$

h = heat transfer coefficient (Btu/hr)²

q = power input to probe - losses (Btu/hr)

A = Surface area of probe (ft²)

ΔT = Difference between the probe surface temperature and the working fluid temperature (°F)

$$\frac{w_h}{h} = \left[\left(\frac{w_q}{q} \right)^2 + \left(\frac{w_A}{A} \right)^2 + \left(\frac{w_{\Delta T}}{\Delta T} \right)^2 \right]^{\frac{1}{2}}$$

Sample calculation:

$$q = 9.4 - 0.1979 = 9.2 \text{ Btu/hr}; \quad A = 2\pi r l = 0.0545 \text{ ft}^2$$

$$\Delta T = 3.1 \text{ }^\circ\text{F}; \quad h = 54.4 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}; \quad w_{\Delta T} = 1\%$$

$$w_q = 6.7\% \text{ (from item 9 above);} \quad w_A = 2\%; \quad w_h = 9.6\%$$

11. Uncertainty in the energy absorbed by the bed walls

$$Q_b = mC(T_f - T_i)$$

Q_b = energy absorbed by the bed walls (Btu)

m = mass of bed wall (lbm)

C = specific heat of the bed wall (Btu/lbm- $^\circ\text{F}$)

T_f = final bed wall temperature ($^\circ\text{F}$)

T_i = initial bed wall temperature ($^\circ\text{F}$)

$$w_{Q_b} = \left[(C(T_f - T_i)w_m)^2 + (m(T_f - T_i)w_C)^2 + (mCw_{T_f})^2 + (mCw_{T_i})^2 \right]^{\frac{1}{2}}$$

Sample calculation:

$$m = 37.4 \text{ lbm}; \quad C = 0.208 \text{ Btu/lbm-}^\circ\text{F}; \quad T_i = 72.1 \text{ }^\circ\text{F}$$

$$T_f = 114.6 \text{ }^\circ\text{F}; \quad Q_b = 330.6 \text{ Btu}; \quad w_m = 1\%; \quad w_C = 1\%;$$

$$w_{T_i} = w_{T_f} = 0.2\text{ }^\circ\text{F}; \quad w_{Q_b} = 1.6\%$$

12. Uncertainty in energy stored in the working fluid

$$Q_{sf} = m_f h_f - m_i h_i$$

Q_{sf} = energy stored in the working fluid

m_f = final mass of refrigerant in bed

h_f = final enthalpy of refrigerant in bed

m_i = initial mass of refrigerant in bed

h_i = initial enthalpy of refrigerant in bed

$$W_{Q_{sf}} = [(h_f w_m)^2 + (m_f w_h)^2 + (h_i w_m)^2 + (m_i w_h)^2]^{1/2}$$

Sample calculation:

$$m_f = 26.4 \text{ lbm}; \quad h_f = 33.8 \text{ Btu/lbm}; \quad m_i = 27.5 \text{ lbm}$$

$$h_i = 26.3 \text{ Btu/lbm}; \quad Q_{sf} = 169 \text{ Btu}; \quad w_{h_f} = w_{h_i} = 0.5\%$$

$$w_{m_f} = w_{m_i} = 1\%; \quad W_{Q_{sf}} = 7.6\%$$

13. Uncertainty in losses to the surrounding room

In this experiment the test section was insulated with 1 foot of fiberglass insulation; losses to the surrounding room were believed to be negligibly small and were neglected. Losses due to axial conduction from the test section were also neglected.

14. Uncertainty in the energy removed from the working fluid as it flows through the bed.

In this experiment the energy removed from the working fluid was calculated by integrating the bed working fluid time enthalpy curves. This integrations was performed by using the mid point formula.

$$Q_{rf} = \sum_{i=0}^n \dot{m} \frac{h(i+1) + h(i)}{2} \Delta\tau |_{in-out}$$

Q_{rf} = energy removed from the working fluid as it flowed through the bed (Btu)

$\dot{m}$ = mass flow rate (lbm/hr)

$\Delta\tau$ = time step (hr)

$h(i)$ = enthalpy of working fluid at time step i

$h(i+1)$ = enthalpy of working fluid at time step i+1

in = packed bed inlet

out = packed bed exit

Assuming $w_{h(i+1)in} = w_{h(i)in} = \dots = w_h$ the uncertainty

w_q in any given term of the above summation is:

$$w_q = \left\{ \Delta\tau \left[\frac{h(i+1)+h(i)}{2} \right] |_{in-out} w_{\dot{m}} \right\}^2 + \left[\dot{m} \frac{h(i+1)+h(i)}{2} \right] |_{in-out} w_{\Delta\tau} \right\}^2 + 4 \left[\frac{\dot{m}\Delta\tau}{2} w_h \right]^2 \}^{\frac{1}{2}}$$

Sample calculation:

$$\dot{m} = 1,564 \text{ lbm/hr}; \quad \Delta\tau = 0.0047 \text{ hr}$$

$$h(i+1)_{in} = 30.2 \text{ Btu/lbm}; \quad h(i)_{in} = 30.0 \text{ Btu/lbm}$$

$$h(i+1)_{out} = 28.9 \text{ Btu/lbm}; \quad h(i)_{out} = 28.7 \text{ Btu/lbm}$$

$$w_m = 39 \text{ lbm/hr}; \quad w_{\Delta\tau} = 2.78 \times 10^{-6} \text{ hr}; \quad w_h = 0.04 \text{ Btu/lbm}$$

$$W_q = 0.3785 \text{ Btu for 99 such terms (99 terms in summation)}$$

$$W_{Q_{rf}} = 99(0.3785) = 37.50 \text{ Btu}; \quad W_Q = 4\%$$

15. Uncertainty in energy absorbed by the bed packing

$$Q_s = Q_{rf} + Q_l + Q_{sf} + Q_b$$

Q_s = energy stored in bed packing

Q_{rf} = energy removed from the working fluid between the
bed inlet and exit.

Q_{sf} = energy stored in the working fluid

Q_b = energy stored in the bed walls

Q_l = losses to surroundings

$$W_{Q_s} = \left[\left(\frac{\partial Q_s}{\partial Q_{rf}} w_{Q_{rf}} \right)^2 + \left(\frac{\partial Q_s}{\partial Q_{sf}} w_{Q_{sf}} \right)^2 + \dots \right]^{1/2}$$

Sample calculation:

$$Q_s = 430.0 \text{ Btu}$$

$$\frac{\partial Q_s}{\partial Q_{rf}} w_{Q_{rf}} = 37.5 \text{ Btu (from item 14 above)}$$

$$\frac{\partial Q_s}{\partial Q_1} w_{Q_1} \approx 0.0 \text{ Btu (from item 13 above)}$$

$$\frac{\partial Q_s}{\partial Q_{sf}} w_{Q_{sf}} = 12.8 \text{ Btu (from item 12 above)}$$

$$\frac{\partial Q_s}{\partial Q_b} w_{Q_b} = 5.2 \text{ Btu (from item 11 above)}$$

$$w_{Q_s} = 9.3\%$$

VITA

Arnold Glenn Graves was born in Grosse Pointe, Michigan on June 17, 1955. He attended elementary, junior high, and high school in various countries throughout Africa. He received his Bachelor of Science degree in Mechanical Engineering from the University of Tennessee in 1982. He received a Master of Science degree in Mechanical Engineering from the University of Tennessee in 1985.