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**An experimental determination of mixed and forced convection
heat transfer coefficients in a modeled nuclear waste repository**

Osborne, Rodney Layne, Ph.D.

The Ohio State University, 1992

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Ann Arbor, MI 48106**

AN EXPERIMENTAL DETERMINATION OF MIXED AND
FORCED CONVECTION HEAT TRANSFER COEFFICIENTS
IN A MODELED NUCLEAR WASTE REPOSITORY

DISSERTATION

Presented in Partial Fulfillment of the Requirements for
the Degree of Doctor of Philosophy in the Graduate School of
The Ohio State University

By

Rodney Layne Osborne, B.S.M.E., M.Sc.

* * * * *

The Ohio State University

1992

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**This dissertation is dedicated
to my wife and children:
Jeanne, Jennifer, and Amanda**

**to my parents:
Aubrey and Eleanor Osborne**

**to my "parents":
Dick and Jeanne Pfarrer**

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NOMENCLATURE

A_i	surface area of heater i [ft ²]
$A_f, A_{ls}, A_{ms}, A_{us}, A_c$	amperage of heater rows floor, lower sidewall, middle sidewall, upper sidewall, and ceiling, respectively [amp]
c_p	specific heat [BTU/(lbm · °F)]
D_h	duct hydraulic diameter, $4 \cdot (\text{duct flow area}) \div (\text{duct perimeter})$ [ft]
E_b	black body emissive power of a surface [BTU/hr · ft ²]
f	turbulent friction factor [dimensionless]
F_{i-j}	thermal radiation view factor, from surface i to surface j [dimensionless]
g	acceleration of gravity [ft/s ²]
Gr	Grashof number, $g\beta(T_{\text{wall}} - T_{\text{bulk}})\ell^3/\nu^2$ [dimensionless]
h_i	convective heat transfer coefficient for surface i [BTU/hr · ft ² · °F]
HFR	<u>H</u> eat <u>F</u> lux <u>R</u> atio for each surface, (surface heat flux)/(floor heat flux) [dimensionless]
J	thermal radiosity of a surface [BTU/hr · ft ²]
k_{air}	air thermal conductivity [BTU/hr · ft · °F]
ℓ	Grashof characteristic length [ft]
$N_f, N_{ls}, N_{ms}, N_{us}, N_c$	number of heaters in rows floor, lower sidewall, middle sidewall, upper sidewall, and ceiling, respectively [dimensionless]
Nu	Nusselt number, hD_h/k_{air} [dimensionless]
Pr	Prandtl number, ν/α
$q_{\text{net},i}$	net heat energy leaving heater pad i [BTU/hr]

$Q_{\text{electrical},i}$	electrical power input to heater pad i [BTU/hr]
$Q_{\text{loss},i}$	heat energy leaving heater pad i by conduction through the duct's walls to the room [BTU/hr]
$Q_{\text{rad},i}$	heat energy leaving heater pad i by thermally radiating to other duct surfaces [BTU/hr]
P_{atm}	atmospheric pressure [inches Hg]
Re	Reynolds number, $\rho V D_h / \mu$ [dimensionless]
$T_{\text{bulk},i}$	air bulk temperature at heater bank i [°F]
$T_{\text{wall},j}$	wall temperature of heater surface j [°F]
$V_f, V_{ls}, V_{ms}, V_{us}, V_c$	voltage of heater rows floor, lower sidewall, middle sidewall, upper sidewall, and ceiling, respectively [volt]
x	insulation thickness [inch]

Greek Symbols

α	thermal diffusivity [ft ² /hr]
β	volumetric expansion coefficient [1/(°F)]
ΔT	temperature difference across wall insulation [°F]
ΔP	flow nozzle differential pressure [mm Hg]
ϵ	surface total, diffuse emissivity for thermal radiation [dimensionless]
μ	air dynamic viscosity [lbm/(hr • ft)]
ν	air kinematic viscosity [ft ² /hr]
ρ	air density [lbm/ft ³]
σ	absolute estimated error of a measurement [units of measurement]
σ_{SB}	Stefan-Boltzmann constant [BTU/(hr • ft ² • °R ⁴)]

CHAPTER I

INTRODUCTION

The nuclear waste disposal plan for the United States, as outlined in the Nuclear Waste policy Act of 1982, calls for high level waste and/or spent fuel to be stored underground in long tunnels mined in deep geological formations. The tunnels and their methods of construction would be similar to existing mines of metals, minerals, and precious stones. Current guidelines call for the waste to be retrievable for 50 years after emplacement.

Nuclear waste, either in the form of spent fuel rods or high level waste (byproducts of fuel reprocessing), will continue to generate heat many years after emplacement. During the retrievability period, the tunnels, or *drifts*, may remain open or may be backfilled. If the drift is kept open during the retrievability period, ventilation will be required for both cooling and air supply for worker safety and comfort. If the drifts were backfilled, and retrieving the waste was required, ventilation would be required prior to actually removing the waste packages, again for worker safety and comfort. In addition, ventilation would be required to cool the walls of the drift down to workable temperatures (approximately 125°F).

To determine the ventilation and cooling requirements of the repository, accurate values of the convective heat transfer coefficients on the interior of the disposal rooms are needed. General flow and heat transfer conditions suggest that the rooms will be operating in the fully turbulent, mixed convection regime (Byrne, et al., 1979). This fact is illustrated in Figure 1.1, showing the different flow regimes for horizontal pipe flow (Metais, 1963, and Metais and Eckert, 1964). Thus, standard forced convection analyses need to be expanded to include the natural convection component of the heat transfer. Also shown in Figure 1.1 is the range of data from the present work. The Reynolds number of the repository was not difficult to match with the experimental duct. As both the repository and the duct use air as the fluid, the Prandtl numbers were equal. The Grashof number was the most difficult

Figure 1.1. Flow regimes in horizontal tubes (Metais, 1963) with estimated waste repository drift (Byrne, et al., 1979) and experimental duct conditions.

parameter to match. The Grashof number is proportional to the temperature differential between the wall and the air and the cube of a height dimension. The Grashof number of the repository is large primarily from the large height of the drift (15 to 20 feet). As the duct was a one twelfth scale model of the drift, the duct Gr number for the same wall-to-air differential would be $(1/12)^3$ or $1/1728$ of the repository Gr. With an increased temperature differential (from using high temperature heaters), this fraction was reduced to a range of $1/500$ to $1/50$.

The waste package emplacement scheme used for this study is shown in Figure 1.2. The packages are buried in the centerline of the drift floor. Other studies have considered horizontal waste package emplacement in the sidewalls (Carmel, 1987). The aim of this work was to determine the heat transfer characteristics in an experimental system which models the repository storage rooms with floor emplacement of vertical packages, both thermally and hydrodynamically. The results must be non-dimensional so that they apply in general to any similarly designed repository.

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Figure 1.2. Vertical waste package emplacement scheme.

CHAPTER II

LITERATURE REVIEW

Several studies, although mostly analytical, have addressed the ventilation in a repository. Svalstad (1983) considered "blast cooling" to cool a repository drift rapidly prior to waste retrieval. The study was analytical and used a finite element code to solve the governing equations. A constant velocity of 1 m/sec was used, which corresponds to a Reynolds number of 280,000. Flows of this type are purely forced convection. Boyd (1982) also considered the ventilation required for retrieval. In this study, also analytical, mixed convection was investigated explicitly, using convection coefficients from various authors. Boyd's results indicate mixed convection reduces the total heat transferred, but he referenced works of Yousef and Tarsuk (1982) and Abdelmeguid and Spalding (1979), which had isothermal wall conditions. For the isothermal wall boundary conditions, any increase in heat transfer for the entrance of a duct will lead to reduced heat transfer further down the duct. This is because the bulk temperature approaches the wall temperature more quickly. This is not the case for the constant axial heat flux boundary conditions used in this work, which are the boundary conditions for the repository drift. The results of several other authors (Brown and Thomas, 1965; Chang, et al., 1985; Chou and Hwang, 1984; Incropera, et al., 1987; Law, et al., 1987; Mahaney, et al., 1987; Mahaney, et al., 1989; Mahaney, et al., 1990; Mori, et al., 1966; Nandakumar, 1985; and Osborne and Incropera, 1985) indicate that the enhancement of mixed convection over purely forced convection continues axially down the duct, rather than reducing the heat transfer below that for purely forced convection. The work of Mori and Futagami (1967) indicated the same results and also included flow visualization to show that the increased heat transfer was due to secondary flows. Mori and Nakayama (1967) also discussed the secondary flow effects on forced convection, and they indicated that their study of curved pipes was analogous to mixed convection because of the secondary flows induced by both.

Dankó and Cifka (1984) determined the convection heat transfer coefficients of an underground copper mine in situ. Although their data were quite scattered, these authors determined that the effects of the natural convection component were significant when compared with the purely forced convection case.

The boundary conditions in a repository drift differ significantly from most standard mine ventilation problems. Although many mines, particularly extremely deep mines, experience high heat fluxes from the rock surfaces, the distribution of the fluxes appears to be unique to the nuclear waste repository. For floor emplacement, the waste packages create an extremely skewed heat flux profile that is present for several years after waste emplacement. Several authors reported results for skewed (circumferentially varying) heat flux profiles: Chang, et al., 1985; Incropera, et al., 1987; Law, et al., 1987; Mahaney, et al., 1987; Osborne and Incropera, 1985; and Patakar, et al., 1978. These results showed that when the heat flux was skewed toward the floor of a duct, the Nusselt numbers were substantially higher than those for purely forced convection. For the cases where the heat flux was skewed toward the ceiling of a duct, the Nusselt numbers were less than those for purely forced convection due to stratification and stagnation of the air against the ceiling. All authors also reported an increase in the friction factor, also due to the induced secondary flows, but not as much as the Nusselt number increase.

Current methods of conventional mine ventilation, which is used for air quality or temperature control for worker comfort, use standard relations for calculating heat transfer coefficients. Some methods (Carrier, 1940, and Goch and Patterson, 1940) assume that there is no surface heat transfer resistance, that is the convective coefficient is infinite. These methods use known values of the virgin rock temperature and air flow stream to calculate the amount of heat that enters the air stream. Starfield (1966a, 1966b) and Starfield and Dickson (1967) modified the results of Goch and Patterson to include a varying heat transfer coefficient, but standard correlations were used to calculate h . Starfield and Dickson recommended a correlation from Rohsenow and Choi (1961), which is the Dittus-Boelter equation, and a surface roughness multiplier, also given in Rohsenow and Choi.

The sidewall heating capabilities of the present duct appears to be unique. The present duct's sidewall is divided into three sections. No other work was found that had a sidewall profile. While many authors have reported findings in mixed convection in ducts, no one has correlated the effect of the heat flux profile on the convective heat transfer. It is this correlation that will be developed here.

CHAPTER III

EXPERIMENTAL SETUP

3.1 Duct

The experimental model, or "duct", had a cross section of 18 inches high by 12 inches wide, and a total heated length of 14.5 feet. These dimensions were chosen so that the aspect ratio of the duct was consistent with preliminary designs of repository rooms.

The air flows were in the fully turbulent regime, with Reynolds numbers ranging from 6000 to 180,000. The corresponding velocities were from 4.1 ft/sec to 25.6 ft/sec. Repository Reynolds numbers were expected to be on the order of 170,000 in the repository (Stearns-Roger, Inc., 1983, pp. 5-18). Thus, the experimental model's range covered the repository expected Reynolds number.

The four walls of the flow channel were constructed with 16 ga. (1/16-inch thick) galvanized sheet steel. Electrical resistance heaters were placed directly on the outside of the sheet steel. Another steel sheet was placed over the back of the heaters, forming a "sandwich". Blocks of high temperature mineral wool insulation, Plibrico Plisulate 201 (13 pounds per cubic foot density), 1.5 inches thick, were placed on the outside of the sandwich. One-half inch plywood and a steel angle frame provided the support for the duct. Figure 3.1 shows a cross sectional view of the duct wall construction.

Figure 3.1. Duct wall cross sectional view.

3.2 Heat Sources and Power Supplies

The wall heat flux was provide by silicon rubber electrical resistance heaters. One circumferential bank of heaters was 6 inches wide. Figure 3.2 shows the heater bank arrangement. A total of 29 of these circumferential banks were installed for the 14.5 feet of heated length. Each bank consisted of eight heaters arranged as follows: one pad, 11-7/8 inches by 6 inches, each for the floor (surface #1 in Fig. 3.2) and ceiling (surface #8); three pads, 6 inches by 6 inches, for each sidewall (surfaces #2 through #7). The heaters provided a continuous heat flux axially down the duct. This in effect smeared the discrete heat sources that would actually exist in the repository. The work of Sisson (1978) and Christensen, et al. (1981) showed that this assumption is reasonable for the waste package spacing used here. The heater rows were controlled separately, with opposing sidewall heaters wired in parallel. Therefore, here were five controllable elements per bank: 1) floor, 2) lower sidewall, 3) middle sidewall, 4) upper sidewall, and 5) ceiling.

The electrical power input to the heaters was provided by five Superior Electric Powerstat variable transformers, Model 1256D rated at 240 volts and 28 amperes. The five Powerstats made it possible to control each row of heater elements (floor, lower sidewall, etc.) separately. This flexibility allowed for a variety of circumferential boundary conditions. For example, the Powerstats could all be set to the give the same constant heat flux to all five rows. Or the Powerstats could be set to give a constant circumferential temperature for the five rows (at one bank). Two Superior Electric Stabline voltage regulators, Model EMT42078 rated at 240 volts and 36 amperes, supplied regulated power to Powerstats. These regulators held the voltage transients to $\pm 3/4$ percent of the full scale voltage. The capacity of these regulators limited the heat flux for the constant circumferential heat flux case to 0.53 watts per square inch. Higher fluxes were achieved, however, using skewed flux profiles (for example, a case with a high floor flux and decreasing fluxes up to the ceiling).

3.3 Air Flow System

Air flow through the duct was provided by a centrifugal blower, Dayton Model 2C652, rated for 2137 CFM at one inch of water column static pressure. The flow was varied through the

Figure 3.2. Heater pad arrangement.

use of two butterfly dampers on the suction side of the blower. Figure 3.3 shows the flow damper arrangement.

Wall temperature measurements showed that the flow was essentially fully developed by heater bank 26 (12.5 feet from the entrance). All results presented here are for this heater bank.

3.4 Instrumentation and Controls

The instrumentation and controls for the duct were composed of systems for temperature measurement, flow measurement, and heater power measurement and control.

Wall temperature measurements were made using 30 ga. copper-constantan (Type T) thermocouples. Thermocouple junctions were formed by arc welding in an inert argon atmosphere. One thermocouple was placed in the center of each heater pad. Each sidewall thermocouple was wired in parallel with its directly opposing thermocouple. This produced an average temperature for the heater pair, which was also wired in parallel. Each thermocouple's calibration was checked using an ice bath and boiling water and a mercury-in-glass NIST thermometer. The relatively high thermal conductivity of the sheet steel made it unnecessary to machine channels in the sheet steel for the thermocouples. The temperature drop across the steel was 0.2°F at full rated heater power (2 watts per square inch). A total of 145 thermocouple measurements were made with a Fluke Model 2200B datalogger. The datalogger was connected to a VAX 8500 minicomputer for on line storage of the temperatures.

Duct air temperature measurements were made with 1/16 inch diameter sheathed Type T thermocouples, with radiation shields per ASHRAE Standard 41.1-74 (American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1975). These temperature readings were logged through the datalogger to the minicomputer. Air temperatures could be measured at five axial locations: 0.5 feet, 5.0 feet, 10.0 feet, 12.5 feet (the beginning of Bank 26), and 14.0 feet. At each of these locations, seven holes were drilled through the ceiling, perpendicular to the duct axis. The holes were spaced 1.5 inch apart, so complete

Figure 3.3. Air flow control dampers.

air temperature profiles could be taken. The duct inlet air temperature was measured with a similar thermocouple.

Heat losses through the duct walls were measured by means of a differential thermocouple arrangement. Two thermocouples were connected in series across the insulation block, where the constantan legs of the two thermocouples were joined. The voltage was measured through the copper legs with a Keithley Model 174 digital multimeter (5-1/2 digit display). Because both of the wires that went to the voltmeter were copper, there was no need to compensate for room temperature. This voltage was the emf produced by the differential temperature across the insulation (Doebelin, 1975, pp. 530-531). The voltage was converted to temperature using thermocouple emf tables. A differential pair was located on all eight heater pads at four axial locations: 0.5 feet, 5.0 feet, 10.0 feet, and 14.0 feet.

The electrical power input to each of the five heater rows was determined by measuring the voltage and current output from each Powerstat. Fluke Model 8050A digital multimeters (4-1/2 digits) were used to measure the voltage directly, and a Fluke Model 1651 shunt resistor was used with the Fluke multimeters to measure the current.

A Brandt Flow Composer, Model B-NZP1031-12-1-4T, was used to measure the air flow rate through the duct. This flow meter was a compact (2 feet long, 1 foot in diameter) flow nozzle with a self-contained honeycomb flow straightener and an averaging pitot tube array. The flow meter was calibrated at the factory to an accuracy of 0.5% of full scale flow. The differential pressure generated by the nozzle was measured with a Daytronics Barocel differential pressure sensor, Model 570D-10T-1A1-V1. The Barocel had a 10 torr full scale range and was read with a Datametrics electronic manometer, Model 1174.

Figure 3.4 shows a schematic of the experimental system.

Figure 3.4. Experimental system schematic.

CHAPTER IV
EXPERIMENTAL PROCEDURE

For this study, the parameters of interest are the heater heat flux profile and values, and the Reynolds, Grashof, and Nusselt nondimensional numbers. The heat flux profile and values and the Reynolds number were the independent parameters, fixed at the beginning of each test. The wall and bulk fluid temperatures and the resulting Nusselt and Grashof numbers were the dependent parameters. Table 4.1 gives the values of these parameters during these reported experiments.

Table 4.1. Duct experimental parameters

Reynolds number (based on D_h)	10,000 to 180,000
Heat flux profile	Constant circumferential input flux Constant circumferential wall temperature (for Bank 26 only) Skewed profiles
Heat flux input values	0.26 to 1.0 watts/sq. in., profile dependent
Grashof number	$\sim 10^6$ to 10^9
Nusselt number	~ 50 to 400

The heat flux value parameter was limited by the maximum allowable heater temperature of 450°F and a maximum power supply current draw. The Reynolds number was limited on

the lower end to stay in the turbulent flow regime and on the upper end by the blower specifications.

4.1 Conduct of the Experiment

Data necessary to determine the local heat transfer coefficients at Heater Bank 26 were measured by the following operations:

1. The desired duct air flow rate was set by adjusting the dampers on the inlet of the blower. The flow rate was indicated by the Daytronics Barocel differential pressure sensor, in mm Hg. This pressure was converted to flow rate by an equation supplied by the flow nozzle manufacturer.
2. The Powerstats were adjusted so that the floor input power was at the desired value. The input power to the other rows (lower sidewall, middle sidewall, upper sidewall, and ceiling) was adjusted to give the desired profile. If the test was a constant circumferential temperature case for Bank 26, these four powers were continuously adjusted through the heat up transient until the five wall temperatures were within 2°F of each other. The input power to each row was calculated from the current and amperage of each Powerstat.
3. All thermocouples were monitored to determine when steady state was reached. Steady state was assumed when the wall temperatures varied less than 1°F over 15 minutes. This took about three hours after a cold startup.
4. At equilibrium, all wall temperatures were recorded through the datalogger to the VAX minicomputer. To help reduce random noise errors induced by the datalogger's high speed switching relays, three complete scans were recorded and later averaged. Heat loss differential thermocouple voltages, ambient temperature, barometric pressure, and the voltage and amperage of each Powerstat were manually recorded.

4.2 Data Reduction

The experimental data sets consisted of:

- five wall surface temperatures (floor T_f , averaged lower sidewall T_{ls} , averaged middle sidewall T_{ms} , averaged upper sidewall T_{us} , and ceiling T_c) for each of the 29 heater banks, for a total of 145 temperatures
- eight differential thermocouple voltage readings across the insulation (floor, back and front lower sidewall, back and front middle sidewall, back and front upper sidewall, and ceiling), at four axial locations (0.5 feet, 5.0 feet, 10.0 feet, and 14.0 feet), for a total of 32 differential voltages
- inlet air temperature: T_{inlet}
- voltage for each Powerstat: V_f , V_{ls} , V_{ms} , V_{us} , and V_c
- amperage for each Powerstat: A_f , A_{ls} , A_{ms} , A_{us} , and A_c
- number of heaters on each Powerstat: N_f , N_{ls} , N_{ms} , N_{us} , and N_c
- flow nozzle pressure differential, ΔP

From these data, the local heat transfer coefficient, local Nusselt number, and the local Grashof number for each heater pad and the Reynolds number were calculated.

The duct air velocity was calculated from the nozzle differential pressure from the following equation (see Appendix C: Air Flow Meter Calibration):

$$\text{Velocity (ft/sec)} = 1101.7 \sqrt{\frac{16.02 \Delta P}{P_{atm}}} \quad (4.1)$$

where ΔP is in mm Hg and P_{atm} is in inches Hg.

The Reynolds number was calculated using bulk temperature at Bank 26 for the air properties and the above velocity. Air properties as functions of temperature are given in Appendix B: Heat Transfer Properties.

For each heater pad i , the net heat convected into the airstream was calculated from

$$q_{net,i} = q_{electric,i} - q_{loss,i} - q_{radiation,i} \quad (4.2)$$

where

- $q_{net,i}$ = net heat energy leaving heater pad i
- $q_{electrical,i}$ = electrical power input to heater pad i
- $q_{loss,i}$ = heat energy leaving heater pad i by conduction through the duct's walls to the room
- $q_{rad,i}$ = heat energy leaving heater pad i by thermally radiating to other duct surfaces

The $q_{electrical,i}$ term for each heater pad i in row j was simply $V_j A_j / N_j$, where j is one of the five heater rows: floor, lower sidewall, middle sidewall, upper sidewall, and ceiling. The heater pads were inspected upon delivery for uniformity. All pads installed in the duct conformed to the manufacturer's stated uniformity of ± 0.5 percent variance in resistance.

The $q_{loss,i}$ term was calculated from the insulation temperature differential and Fourier's law of conduction. The differential thermocouples' voltage was converted to temperature using thermocouple emf tables (Omega, 1982, Type T Thermocouple EMF Tables). An approximate ΔT was calculated by using a constant value for the thermocouple emf curve slope of 40°F per millivolt. Using this slope, the average temperature of the insulation was found. This average temperature was converted back into voltage, using Eqn. 4.3:

$$emf(\text{millivolts}) = \text{function of } T \quad (4.3)$$

This voltage, which is the emf equivalent of the insulation's average temperature, was used in Eqn. 4.4 to calculate the slope of the thermocouple curve, in degrees F per millivolt.

$$\Delta T \text{ (deg. F)} = 46.44 - 2.380(mV) + 0.1212(mV)^2 \quad (4.4)$$

The slope was then multiplied by the differential thermocouple voltage to get the differential temperature. Linear interpolation between the axial differential thermocouple locations positions gave the ΔT for each heater pad. Fourier's law of conduction gave $q_{loss,i}$ as

$$q_{loss,i} = \frac{k_{insul} A_i \Delta T_i}{x} \quad (4.5)$$

where

- k = insulation thermal conductivity (see Eqn. C-7 in Appendix B: Heat Transfer Properties)
- A_i = area of the heater pad (72 sq.in. for a pair of sidewall heaters, 71.25 sq.in. for the floor and ceiling heaters)
- x = insulation thickness (1.5 inch)

The radiation heat transfer term, $q_{rad,i}$, was calculated at each heater pad using the network scheme (Oppenheim, 1956; Holman, 1976; and Appendix G: Thermal Radiation Correction). Although the radiation term was calculated for each heater pad, the view factors used were for infinitely long, parallel and perpendicular plates. Thus, the scheme only took into account the wall temperatures of a given row, and radiation exchange axially between heaters was neglected. The air was treated as a non-absorbing medium. An emissivity of 0.23 was used for the galvanized sheet steel. The determination of the emissivity is discussed in Appendix F: Sheet Steel Emissivity Measurement. Appendix G: Thermal Radiation Correction contains the development of the network scheme, calculations of the infinite length view factors, and discussions of the effects on accuracy of the assumptions of infinite length geometry and emissivity values. As stated in Appendix G, the infinite length assumption introduces an error of less than 5 percent.

The net heat input from each heater pad was summed, and combined with the known air mass flow rate, the air inlet temperature, and specific heat, the air bulk temperature were calculated for each bank.

The convective heat transfer coefficient for each heater surface i in bank n was calculated by

$$h_i = \frac{q_{net,i}}{A_i(T_{wall,i} - T_{bulk,n})} \quad (4.6)$$

where

- $T_{wall,i}$ = wall temperature of heater surface (i = floor, lower sidewall, middle sidewall, upper sidewall, or ceiling), in heater bank n
- $T_{bulk,n}$ = air bulk temperature at heater bank n
- A_i = area of the heater pad (72 sq.in. for a pair of sidewall heaters, 71.25 sq.in. for the floor and ceiling heaters)

The nondimensional convective heat transfer parameter, Nusselt number, was calculated from

$$Nu_i = \frac{h_i D_h}{k_{air,n}} \quad (4.7)$$

where

- D_h = hydraulic diameter, defined as $4 \times (\text{duct flow area}) \div (\text{duct perimeter})$. With a duct height of 1.5 feet, a duct width of 1.0 foot, $D_h = 1.2$ feet.
- $k_{air,n}$ = air thermal conductivity, evaluated at the bulk temperature at heater bank n (see Eqn. C-2 in Appendix B: Heat Transfer Properties)

The nondimensional natural convective heat transfer parameter, Grashof number, was calculated from

$$Gr_i = \frac{g\beta(T_{wall,i} - T_{bulk,n})\ell^3}{\nu^2} \quad (4.8)$$

where

- g = gravitational constant, 32.17 ft/s²
- β = volumetric expansion coefficient, evaluated at the bulk temperature at heater bank n

- ν = kinematic viscosity, evaluated at the bulk temperature at heater bank n (see Eqn. C-5 in Appendix B: Heat Transfer Properties, for the combined term of $g\beta/\nu^2$)
- ℓ = characteristic length

The Grashof number characteristic length, ℓ , of two horizontal, heated, parallel plates is the plate width for the downward facing surface, and the plate spacing for the upward facing surface. For heated, vertical walls, the characteristic length is usually the vertical distance from the bottom edge of the plate. For an enclosure, however, the vertical distance a particular surface "sees" is the distance from that surface to the top of the enclosure. For example, the upper sidewall surface, butted against the ceiling on its upper edge, had a vertical dimension of 6 inches, but its midpoint is 3 inches from the surface (the ceiling) that is impeding the natural convection flow. This surface was assigned a characteristic length of 3 inches. Likewise, the middle sidewall surface also had a vertical dimension of 6 inches, but its midpoint is 9 inches from the ceiling. Therefore, the middle sidewall surface was assigned a characteristic length of 9 inches. These alternative characteristic lengths were developed during work on the correlation presented in Chapter V. The more conventional lengths did not correlate the data well.

A set of three computer programs were used to log the wall temperature data to the VAX minicomputer and reduce the data to the above parameters. The FORTRAN program listings are given in Appendix A: Data Reduction Programs.

The first program, COLDATA, collected the wall temperature data from the Fluke datalogger and stored the raw temperature data in ASCII files. As noted earlier, three scans of data (145 thermocouple channels each) were stored using this program.

The second program, AVG2, read in the ASCII file created by COLDATA and averaged the three scans. AVG2 created a second ASCII file with the 145 averaged temperature readings.

The third program, REDDATA, read in the average data file created by AVG2 and user-supplied values of the voltage and amperage of each of the five Powerstats, the flow nozzle

differential pressure, the ambient temperature and pressure, and the heat loss thermocouple emfs. This program performed the calculations necessary to generate the net heat flux, net heat flux ratio (flux profile), Nusselt number, and Grashof number for each tunnel surface, and the duct Reynolds number.

4.3 "Real" Heat Flux Profile Determination

SAI (1976) presented some preliminary results for nuclear waste stored in salt. They considered waste packages buried along the centerlines of drifts. The waste thermal loading was the same as considered here for the HEATING5 analysis (50 kW/acre). Figure 4.1 (Figure 3.2 from their report) shows the calculated salt temperature profile, using a purely forced convection coefficient of $0.6 \text{ BTU}/(\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F})$. This corresponds to a Reynolds number of approximately 490,000. This temperature profile was exploded, and the temperature gradients were calculated from the isotherms. Using the thermal conductivity of the salt, the heat flux was calculated at the five locations similar to the experimental duct of this study (floor, lower side, middle side, upper side, and ceiling). The resulting heat flux profile is shown in Table 4.2. These numbers are relative to the floor heat flux.

Table 4.2. "Real" heat flux profile (from SAI, 1976, Figure 3.2)

Floor heat flux	1.0
Lower Sidewall heat flux	0.16
Middle Sidewall heat flux	0.086
Upper Sidewall heat flux	0.048
Ceiling heat flux	0.031

Figure 4.1 Isotherms for Ventilation Calculations (SAI, 1976, Figure 3.2)

4.4 HEATING5 Computer Program

The computer program HEATING5 (Turner, 1977) was used to model the repository drift. HEATING5 is a finite-difference heat conduction program developed at the Oak Ridge National Laboratory. The program uses a point successive overrelaxation method and a modification of the Aitken δ^2 extrapolation process to solve the finite difference equations.

The results and figures discussed in Chapter V indicate that the actual flux profile in a waste repository depends on the flow rate (Reynolds number). To try to determine the actual repository flux profile, an iterative procedure was used. The repository was modeled in HEATING5 as a two dimensional slice perpendicular to the tunnel axis, with the repository tunnel surface nodes defined similar to the experimental surfaces, with a floor, three vertical sidewall surfaces, and a ceiling. The HEATING5 node placement is shown in Figure 4.2. These surfaces were given convective boundary conditions. The tunnel was assumed to be infinite in length. The host material was salt with a temperature dependent thermal conductivity given by Eqn. C-1 in Appendix B: Heat Transfer Properties. The tunnel was 18 feet high and 12 feet wide. The experimental duct then was one-twelfth scale of the repository tunnel. Symmetry was assumed along the tunnel centerline, where an adiabatic boundary condition existed, except at the waste package location. The vertical dimensions of the model were selected somewhat arbitrarily, and were similar to those used by Sisson (1978) and Svalstad (1983). Both the top and bottom boundaries were assumed to be adiabatic. For an exact representation, these boundaries would have to extend upward to the earth's surface (with some real boundary condition applied there) and downward, possibly to the earth's geometric center (where an adiabatic boundary might truly apply). Extending the model to such bounds would certainly violate the homogeneous material constraint and would surely be impractical for numerical solution. The other vertical boundary, at the right in Figure 4.2, was set at the midway distance between two drifts. As this study is concerned with a full repository, this boundary would be adiabatic, as the drift beyond this boundary would have the same heat generation term and boundary conditions.

The waste package was modeled as a constant heat flux source 8 feet long at the tunnel centerline, 10 feet below the floor. The gross thermal loading was considered to be 50 kW/acre (Byrne, et al., 1979). The source strength in the waste package plane was then

Figure 4.2. HEATING5 repository model.

18.96 BTU/(hr • ft²). As the model was two dimensional, this strength was continuous in the axial direction; that is, the individual packages were not modeled. The work of Sisson (1978) and Christensen, et al. (1981) showed that this assumption is reasonable for the waste package spacing used here. This was also the case for the experimental tests. The coarse grid lines are shown in Figure 4.2. Fine grids were applied within the coarse grids to give a model with 1114 nodes. All calculations with HEATING5 were in the two dimensional, steady state mode. The output from HEATING5 was the nodal steady state temperatures of the solid material (salt). A sample HEATING5 input data set modeling the repository drift is given in Appendix D: HEATING5 Repository Model Input Data.

4.5 Experiment / HEATING5 Heat Flux Profile Iteration

As shown by Figures 5.9 through 5.12 in Chapter V, the host rock temperature profile, and therefore the heat flux profile, is dependent on the convective heat transfer coefficients of the surfaces. An iterative procedure using the HEATING5 conduction program and the experimental tunnel was used to predict the convective heat transfer coefficients and the host rock temperature profile. The iteration was performed for two Reynolds numbers, 10,000 and 138,000. The procedure was to choose an arbitrary flux profile for the experimental tunnel. Convection coefficients were found for the five experimental surfaces (floor, three vertical sidewall surfaces, and ceiling). These experimental convection coefficients were converted to repository convection coefficients by assuming similarity, that is:

$$Nu_{\text{experiment}} = Nu_{\text{repository}} \quad (4.9)$$

As stated previously, the experiment was a one-twelfth scale of the repository, therefore

$$D_{h(\text{repository})} = 12 D_{h(\text{experiment})} \quad (4.10)$$

Using the definition of the Nusselt number and Eqn. 4.10 and equating the thermal conductivity of the experimental air and the repository air, Eqn. 4.9 may be solved for $h_{\text{repository}}$:

$$h_{\text{repository}} = \frac{h_{\text{experiment}}}{12} \quad (4.11)$$

The five convection coefficients calculated from Eqn. 4.11 were input to the HEATING5 model, and the host rock nodal temperatures were calculated. The five heat flux values were found from the calculated rock tunnel surface temperatures and the bulk air temperature.

$$q_i'' = h_i (T_{wall,i} - T_{bulk}) \quad (4.12)$$

From the five q_i'' values, the flux profile (with respect to the floor flux) was calculated. This new flux profile was run in the experimental duct, and a new set of convection coefficients were determined. This completed one iteration. The procedure was repeated until the flux profile calculated by HEATING5 was within 3 percent of the profile that was input to the experiment. Figure 4.3 schematically shows the procedure.

4.6 ERROR ANALYSIS

Temperature. Temperature measurement errors were placed into three categories. The copper-constantan thermocouple wire had a manufacturer-stated maximum error of $\pm 1.5^\circ\text{F}$. The datalogger had a manufacturer-stated maximum error of $\pm 0.9^\circ\text{F}$. The error associated with the interconnecting extension wire and terminal strips between the thermocouples and the datalogger was found to be $\pm 1.3^\circ\text{F}$. The root mean sum of the squares of these errors is $\pm 2.2^\circ\text{F}$. The error in the term $(T_{wall} - T_{bulk})$ was therefore $\pm 3.1^\circ\text{F}$. For the temperature differentials that were measured, this amounted to ± 1 to ± 4 percent.

Electrical Power Input. The power measurement errors were attributed to the digital multimeters. The multimeters had a manufacturer-stated maximum error of ± 1 percent of the reading. For both voltage and current errors of ± 1 percent, the resulting power measurement error was ± 1.4 percent.

Air Flow Measurement. The Brandt flow meter had a manufacturer-stated maximum error of ± 0.5 percent of the full scale flow, or ± 12 CFM. For the range of the experiments, this gave flow to pressure conversion errors of ± 0.5 percent at a Reynolds number of 180,000 and ± 8 percent at a Reynolds number of 6,000. The differential pressure transducer had a manufacturer-stated maximum error of ± 0.15 percent of the reading or ± 0.005 percent of

Figure 4.3. HEATING5 / experiment iteration for heat flux determination.

the full scale, whichever was larger. For the range of the experiments, this gave pressure measurement errors of ± 0.15 percent at a Reynolds number of 180,000 and ± 9 percent at a Reynolds number of 6,000. Combining the flow meter and pressure transducer errors gave a total flow rate error of ± 0.5 percent at a Reynolds number of 180,000 and ± 12 percent at a Reynolds number of 6,000.

Net Heat Input Determination. The error of the net heat input to a surface had three components: 1) error in the measurement of the input electrical power, 2) error in assuming no sidewall vertical conduction between surfaces, and 3) the error in assuming the heater rows were infinite strips for the thermal radiation correction. The power measurement error was earlier assessed at ± 1.4 percent. The vertical conduction error was estimated in Appendix E: Sidewall Conduction Analysis to be 10 percent for the lower Reynolds number of 6,000 and 3 percent for the higher Reynolds number of 180,000. The radiation correction error was estimated in Appendix G: Thermal Radiation Correction to be 5 percent. The summation of these errors results in a net heat input error of ± 12 percent for the Reynolds number of 6000 and ± 6 percent for the Reynolds number of 180,000.

Convective Heat Transfer Coefficient. The convective heat transfer coefficient is calculated from Eqn. 4.6. From this equation, the error of the heat transfer coefficient is found from the following equation:

$$\left(\frac{\sigma_h}{h}\right)^2 = \left(\frac{\sigma_q}{q}\right)^2 + \left(\frac{\sigma_{\Delta T}}{\Delta T}\right)^2 + \left(\frac{\sigma_A}{A}\right)^2 \quad (4.13)$$

where σ is the absolute error of each the parameters in Eqn. 4.6, and $\Delta T = (T_{\text{wall}} - T_{\text{bulk}})$. The error on the surface area A was assumed to be negligible. Inserting the errors of these parameters into Eqn. 4.13 gives errors on h of ± 13 percent for the Reynolds number of 6,000 and ± 8 percent for the Reynolds number of 180,000.

CHAPTER V

RESULTS

As mentioned in Chapter IV, the heater pad arrangement allowed for a wide range of input circumferential heat flux profiles. Nusselt number results for five different profiles are presented, along with the HEATING5 iteration results. Table 5.1 shows these five boundary conditions. All cases were with each row being at a constant axial input heat flux; that is, while the lower sidewall may have been at a different input heat flux than the floor, both input heat fluxes were constant along the length of the duct. The determination of the "Real" heat flux profile was discussed in Chapter IV.

TABLE 5.1. Experimental duct boundary conditions.

BOUNDARY CONDITIONS	HEAT FLUX VALUES
CHF : Constant circumferential input heat flux	0.26, 0.53, 1.0 watts/in ²
CCT : Constant circumferential temperature, constant axial input heat flux	0.53 watts/in ²
HF1 : Hot floor profile #1.	Floor at 0.53 watts/in ² Low side @ 75% of floor Mid side @ 50% of floor Upper side @ 25% of floor Ceiling @ 10% of floor
HF2 : Hot floor profile #2.	Floor at 0.53 watts/in ² Low side @ 80% of floor Mid side @ 60% of floor Upper side @ 40% of floor Ceiling @ 20% of floor
RP : "Real" profile, taken from previously calculated repository temperature profile (SAI, 1976, Figure 3.2)	Floor at 0.8 ,1.2 watts/in ² Low side @ 16 % of floor Mid side @ 8.6% of floor Upper side @ 4.8% of floor Ceiling @ 3.1 % of floor

5.1 Nusselt Number Plots Versus Reynolds Number

Three different heat flux profiles were tested. Figures 5.1 through 5.8 show the fully developed Nusselt number for each of the five surfaces. The line labelled "Dittus-Boelter" is the classical and long-used forced convection correlation developed for turbulent flow in long tubes with constant circumferential heat flux, (Dittus and Boelter, 1930), given by Eqn. 5.1.

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (5.1)$$

Eqn. 5.1 was developed for $0.7 \leq Pr \leq 120$ and $2500 \leq Re \leq 1.24 \times 10^5$. The line labelled "Gnielinski" is a newer forced convection correlation also developed for turbulent flow in tubes (Gnielinski, 1976), given by Eqn. 5.2:

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)} \quad (5.2)$$

where the turbulent friction factor, f , for smooth ducts is given by Eqn. 5.3 (Colebrook, 1939):

$$\frac{1}{\sqrt{f}} = 1.5635 \ln\left(\frac{Re}{7}\right) \quad (5.3)$$

Eqns. 5.2 and 5.3 were developed for $0.5 \leq Pr \leq 2000$ and $2300 \leq Re \leq 5 \times 10^6$.

Bhatti and Shah (in Kadaç, et al., 1987) indicate that of these two correlations, Gnielinski's fits published data better and that the Dittus-Boelter correlation over predicts Nusselt numbers for air by 13 to 17 percent.

5.1.1 CHF : Constant circumferential input heat flux. The runs in Figures 5.1, 5.2, and 5.3 had the boundary conditions of Dittus and Boelter, and Gnielinski, i.e., all five heater rows had the same input heat flux. The three figures differ in the electrical power input. To show the effect of the magnitude of the power, two heater powers are paired in each of the three figures: Figure 5.1 shows the results for 0.26 and 0.53 watts per square inch electrical input power; Figure 5.2 shows the results for 0.53 and 1.0 watts per square inch; and Figure 5.3 shows the results for 0.26 and 1.0 watts per square inch. As would be expected, the data match the Gnielinski and Dittus-Boelter correlations reasonably well at Re greater than 50,000. For example, from Figure 5.1, the average Nu number for the five surfaces at

Figure 5.1.

Nusselt number versus Reynolds number for **CHF** : Constant Circumferential Flux, input flux of 0.26 and 0.53 watts/sq.in.

Figure 5.2.

Nusselt number versus Reynolds number for CHF : Constant Circumferential Flux, input flux of 0.53 and 1.0 watts/sq.in.

Figure 5.3.

Nusselt number versus Reynolds number for **CHF** : Constant Circumferential Flux, input flux of 0.26 and 1.0 watts/sq.in.

the highest Re number (178,300) was 310. The Gnielinski correlation gave 285 (8 percent lower than the experiment) and the Dittus-Boelter correlation gave 351 (13 percent higher than the experiment). At a Re number of 55,000, however, the data points begin to flatten out. The five-surface average Nu number is 136, while the Gnielinski correlation is 16 percent lower at a value of 114, and the D-B correlation gives 126, 8 percent lower than the experiment. At the low Re number in Figure 5.1 of 13,500, both correlations give approximately 39 for the Nu number. The average Nu number for the four surfaces with *aiding* mixed convection is 104, an increase of a factor of 2.7. The *unaiding* mixed convection of the ceiling led to the low Nu number of 23. This again is expected, as the buoyancy-driven flows would create a stagnation layer on the ceiling, thus reducing the heat transfer as predicted by forced convection analysis. The Nu number of the floor actually went up as Re was reduced, indicating a very strong natural convection component. The upper sidewall, the surface with the shortest characteristic Grashof number length (see Chapter IV, Eqn. 4.8), was affected to a lesser degree, but was still substantially about the purely forced convection correlations. Comparing the pairs (filled and unfilled) of data points of the three figures, there appears to be no difference in the Nu numbers of the three heater powers at Re numbers of above 50,000. However, at data points of Re about 50,000, the higher heat flux points are above the lower flux points, again indicating the effect of natural convection. This effect is fairly pronounced in Figure 5.3, comparing the lowest and highest heat flux tests. The floor Nu number of the higher flux (1 watt/sq. in.) is 21 percent higher than the floor Nu number of the lower heat flux (0.26 watt/sq. in.). With a fixed flow rate through the duct (that is, a fixed Re number), a higher heater power results in an increased surface-to-air temperature difference, thus increasing the buoyancy effects.

5.1.2 CCT : Constant circumferential temperature, constant axial input heat flux.

Figure 5.4 has similar results for the constant circumferential wall temperature, except the ceiling Nu did not decrease as much as in Figures 5.1, 5.2, and 5.3. The reason for this was that in order to maintain a constant circumferential temperature, the ceiling heat flux had to be reduced to counteract the buoyancy driven upward flow of hot air. This upward moving hot air increased the ceiling temperature. The reduction in the heat flux in turn reduced the stagnation layer, thus giving Nusselt numbers closer to the Dittus-Boelter correlation. As in Figure 5.1, the data curve upward as the Re is decreased, again showing the strong effect of

Figure 5.4. Nusselt number versus Reynolds number for CCT : Constant Circumferential Temperature, input flux of 0.53 watts/sq.in.

the natural convection coefficient. This shows that as the flow is decreased, the increased natural convection component more than overcomes the drop in the forced convection component.

5.1.3 HF1 and HF2 : Hot floor profiles #1 and #2. Figures 5.5 and 5.6 show the results of tests with skewed input heat flux profiles. The heat flux profiles are shown on the inset in the figures. Both figures have the same general shape as the previous plots. In both figures, the ceiling Nu number is fairly close to the forced convection correlations. This is because the ceiling heat flux was only a fraction of the floor flux, reducing the stagnation layer near the ceiling as in the constant circumferential temperature case.

5.1.4 RP : "Real" profile. Figures 5.7 and 5.8 show data from a heat flux profile that was calculated from estimated temperature profiles in a repository in salt. The heat flux profile is shown on the inset in the figure. The numbers on these profile schematics indicate that the ceiling had an input heat flux of 3.1 percent of the floor flux (0.8 watts/in² for Figure 5.7 and 1.2 watts/in² for Figure 5.8). The two figures have the same flux profile but different absolute fluxes. In the forced convection region (Re greater than 50,000), the Nusselt numbers for the five surfaces differed significantly from each other. This was due to the extremely skewed input flux profile. Sparrow, et al. (1966) did an experimental study of an asymmetrically heated duct (heated floor and ceiling, insulated sidewalls). They also showed that the lesser heated surface had a higher Nu number, the ceiling for the case of these two figures. One hypothesis for the low Nu numbers of the sidewall surfaces, especially the lower sidewall, is that with such low heat inputs for the sidewalls, the secondary flows are very small. With such low heat input from the sidewalls, the duct approaches a heated floor-only duct. Mahaney, et al., (1988) numerically considered a rectangular duct. They looked at the effect that the location of the heat input from the different walls had on each other. Floor-only heating resulted in Nu numbers 250 percent higher than purely forced convection Nu numbers. Sidewall heating was added to the floor heated case (both fluxes being equal). The floor Nu number increased by another 25 percent. This result is seen in this study when comparing Figures 5.4, 5.5, 5.6, 5.7, and 5.8 with Figures 5.1, 5.2, and 5.3. The latter have a fixed circumferential flux, while the former have circumferentially varying flux. The former are listed from less to more skewed profiles. Mahaney, et al. also started with

Figure 5.5.

Nussult number versus Reynolds number for HF1 : Hot Floor profile number 1, input floor flux of 0.53 watts/sq.in.

Figure 5.6.

Nussult number versus Reynolds number for **HF2** : Hot Floor profile number 2, input flux of 0.53 watts/sq.in.

Figure 5.7. Nusselt number versus Reynolds number for **RP** : "Real" Profile, input flux of 0.8 watts/sq.in.

Figure 5.8.

Nussult number versus Reynolds number for **RP** : "Real" Profile, input flux of 1.2 watts/sq.in.

sidewall heating only, then added floor heating. The addition of the floor heating reduced the sidewall Nu number. The more skewed the profile, the less the sidewall Nu number. This is the result of strong secondary flows induced by buoyant flow along the sidewalls.

Although the Nu numbers for the five surfaces differ significantly from each other, the slopes of the curves (the exponent of the Reynolds number) are very close to each other and to the slope of the Dittus-Boelter and Gnielinski correlations. This similarity in slope demonstrates that although the magnitudes differ, the dependence of the Nusselt numbers on the Reynolds number is consistent in the purely forced convection region. As the Reynolds number decreases, the Nusselt numbers appear to reach a horizontal asymptote around $Re = 30,000$. Below this Reynolds number, the Nusselt number curves are essentially flat, indicating that as the flow decreases, the forced convection component decreases, with, however, a corresponding increase in the natural convection component. Thus in the range of $Re = 6000$ to $30,000$, the mixed convection Nusselt number is independent of the flow rate. Although their data were scattered, the figures by Danko and Cifka (1984) show the same general trend.

This last result could be very significant if large flow rates are not required in the repository rooms, i.e. during the period that the rooms are open for emplacement activities or when the rooms are not undergoing blast cooling for retrieval. Using these results, the same heat transfer coefficient can be achieved at reduced flow rates.

5.2 Air Temperature Profiles

Figures 5.9 and 5.10 show air isotherms in the tunnel for the constant circumferential heat flux input, at Reynolds numbers of 30,000 and 138,000, respectively. Dimensions are given around the perimeter of the duct drawing: for the floor and ceiling, distance from the centerline is given; for the sidewalls, elevation from the floor is given. The two figures show drastically different profiles. The lower Re number isotherms show significant stratification vertically, although the strata are not flat across the width of the duct. The higher Re isotherms are more evenly distributed around a central core. The cool core of the lower Re number exists near the floor of the duct, similar to results found by Chou (1984).

Figure 5.9.

Air isotherms in the duct for the CHF boundary condition, $Re = 30,000$, input flux of $0.53 \text{ watts/sq.in.}$ * indicates wall temperature.

Figure 5.10.

Air isotherms in the duct for the CHF boundary condition, $Re = 138,000$, input flux of $0.53 \text{ watts/sq.in.}$ * indicates wall temperature.

Figure 5.9 confirms the presence of mixed convection. At a given elevation the isotherms indicate cooler regions in the center. This cooler region would cause air flow downward, while the warmer regions would tend to rise. This indicates a secondary flow of two counter rotating eddies, rotating clockwise in the left half of the duct and counterclockwise in the right half of the duct. It also appears that a different cellular structure exists near the floor. Chou and Hwang also found this, and their streamline plot shows a second, smaller pair of eddies near the center of floor. The small temperature gradients of the higher Re number, Figure 5.10, would preclude any substantial secondary flows.

5.3 HEATING5 Iteration

The HEATING5 iteration procedure was described in Chapter IV, Section 4.5. The iteration was performed for two Re numbers, 10,000 and 138,000. From the figures in the previous section, these two Re numbers are in the mixed convection and forced convection regimes, respectively. Table 5.2 shows the heat flux profiles at each step of the iterative process for both flow rates. Six steps were required for convergence of the heat flux profile for the Re of 10,000, and five steps were required for convergence for the Re of 138,000. The heat transfer results are shown in Table 5.3. The Nusselt numbers shown indicate a strong dependence of the flux ratio on the flow regime.

Figures 5.11 through 5.14 show temperature profiles in the repository for four different cases of heat transfer coefficients. Figures 5.11 and 5.12 show the drift isotherms for $Re = 130,000$. For Figure 5.11, the Dittus-Boelter correlation (Eqn. 5.1) was used for the heat transfer coefficients for all surfaces. Figure 5.12 shows drift isotherms using the iterative h values. The profiles are very similar, with Figure 5.12 being slightly cooler toward the ceiling. Figures 5.13 and 5.14 are for $Re = 10,000$, Figure 5.13 using the Dittus-Boelter h values, Figure 5.14 using the iterative h values. As seen from Figures 5.1 through 5.8, this Reynolds number is very much in the mixed convection regime. Using purely forced convection relations at this Reynolds number results in an over-estimation of the floor temperature by as much as 85°F. The effect of mixed convection is also apparent by comparing Figures 5.12 and 5.14. Even though the Reynolds number, and therefore the

flow rate, differs by a factor of 13, the temperature profiles are closer than would be expected by forced convection analyses.

Table 5.2 HEATING5 / experiment iteration heat flux values.

Re = 10,000						
Iteration No.	1	2	3	4	5	Final Profile
Ceiling	0.28	0.17	0.17	0.16	0.13	0.13
Upper Sidewall	0.49	0.42	0.41	0.43	0.38	0.38
Middle Sidewall	0.56	0.55	0.55	0.58	0.50	0.50
Lower Sidewall	0.67	0.78	0.79	0.78	0.78	0.79
Floor	1.0	1.0	1.0	1.0	1.0	1.0

Re = 130,000					
Iteration No.	1	2	3	4	Final Profile
Ceiling	0.56	0.86	0.76	0.62	0.61
Upper Sidewall	0.53	0.50	0.51	0.49	0.50
Middle Sidewall	0.47	0.48	0.50	0.49	0.48
Lower Sidewall	0.54	0.46	0.58	0.63	0.64
Floor	1.0	1.0	1.0	1.0	1.0

TABLE 5.3. Nusselt numbers from the HEATING5/experimental iteration.

	Re = 10,000	Re = 130,000
Ceiling	22	263
Upper Sidewall	59	243
Middle Sidewall	77	228
Lower Sidewall	111	218
Floor	127	258
Dittus-Boelter (Eqn. 5.1)	32	258
Gnielinski (Eqn. 5.2)	30	233

5.4 Nusselt Number Correlations

Four correlations were developed which predict the Nu number on a tunnel surface for a given set of flow conditions. The correlations are linear regression fits of the experimental Nu number as a function of the Re and Gr numbers for each of the five duct surfaces. The performance of the ceiling was sufficiently different from the other four surfaces that a separate correlation is given for it. Equation 5.4, for the floor and sidewalls, and Equation 5.5, for the ceiling, correlate the experimental data for low Re numbers (less than 58,000) in the mixed convection regime. Equation 5.5 uses the width of the duct as the characteristic dimension, as is common for a horizontal heated plate facing downward. Equations 5.6 and 5.7 treat the cases of higher Re numbers in the forced convection regime (Re greater than 85,000). From the Nu number versus Re graphs of Section 5.1, in the range of Re greater than 85,000, the Nu numbers of the floor and sidewalls should not have been a function of the Gr number. Indeed, the regression analysis showed this with a exponent of zero for the Gr number in this range.

Figure 5.11. HEATING5 host rock isotherms using the Dittus-Boelter Nusselt number for $Re = 130,000$.

Figure 5.12. HEATING5 host rock isotherms using the iterative Nusselt numbers for $Re = 130,000$.

Figure 5.13. HEATING5 host rock isotherms using the Dittus-Boelter Nusselt number for $Re = 10,000$.

Figure 5.14. HEATING5 host rock isotherms using the iterative Nusselt numbers for $Re = 10,000$.

For the floor, low Re range,

$$Nu = 4.20 Re^{0.172} Gr^{0.077} HFR^{0.101} \quad (5.4)$$

For the ceiling, low Re range,

$$Nu = 0.00232 Re^{1.02} Gr^{-0.0154} HFR^{-0.204} \quad (5.5)$$

where

Nu = Nusselt number for the duct surface, based on D_h

Gr = Grashof number for the duct surface, based on distance from ceiling for
Eqn. 5.4, based on the duct width for Eqn. 5.5

Re = Reynolds number, based on D_h

HFR = Heat flux ratio for the duct surface = surface heat flux / floor heat flux

Equations 5.4 and 5.5 are valid for $6400 < Re < 58,000$ and $10^5 < Gr < 10^9$. The average errors for Eqns. 5.4 and 5.5 are 8.5 and 8.3 percent, respectively.

For the floor, high Re range,

$$Nu = 0.0481 Re^{0.720} HFR^{0.050} \quad (5.6)$$

For the ceiling, high Re range,

$$Nu = 0.0391 Re^{0.748} HFR^{-0.0509} \quad (5.7)$$

Equations 5.6 and 5.7 are valid for $85,000 < Re < 186,000$ and $10^5 < Gr < 10^9$. The average errors for Eqns. 5.6 and 5.7 are 5.6 and 3.8 percent, respectively.

Figures 5.15 and 5.16 show the data used for these correlations plotted against the correlation. For a perfect fit, all data points would lie on the line $Nu_{calc} = Nu_{exp}$. Also shown on each plot are lines indicating plus and minus 10 percent of the experimental data. In Figure 5.15, 30 percent of the predictions fall outside the ± 10 percent lines, and less than 1 percent fall outside a band of ± 20 percent. In Figure 5.16, 14 percent of the predictions fall outside the ± 10 percent lines, and none fall outside a band of ± 20 percent.

Figure 5.15

Calculated Nusselt numbers, using Eqns. 5.4 and 5.6, versus experimental Nusselt numbers for the floor and sidewall surfaces.

Figure 5.16

Calculated Nusselt numbers, using Eqns. 5.5 and 5.7, versus experimental Nusselt numbers for the ceiling surface.

CHAPTER VI

CONCLUSIONS

This work presents convective heat transfer coefficients for an experimental apparatus that models a nuclear waste repository storage room. Data are shown for several circumferential heat flux ratios, including a heat flux profile expected in a room.

Below a Reynolds number of 50,000, the convection heat transfer coefficients for all surfaces except the ceiling are above the coefficients predicted by forced convection relations. The amount of increase over forced convection coefficients increases with decreasing Reynolds number, indicating an increasing mixed convection component. At a Reynolds number of 6000, the floor Nusselt number is a factor of six larger than the forced convection Nusselt number. Thus, for the floor and sidewall surfaces the mixed convection is *aiding*. The secondary flows induced by the buoyancy forces assist the bulk flow in removing the heat from these surfaces. For all but one flux profile, the ceiling Nusselt numbers in this flow range were below the forced convection Nusselt numbers, indicating *unaiding* mixed convection for this surface. With a fixed heat flux on the surface, the surface temperature increased as the flow decreased. The increased surface temperature caused a stagnant air layer to build up against the ceiling. This stagnant air layer would actually serve as an insulator, preventing the bulk flow forced convective effects from removing the heat. The thickness of this layer is therefore proportional to the both the flow rate and the ceiling heat flux. The exception to the dominance of this stagnant air layer was for the case of an extremely skewed flux profile, where the ceiling flux was only 3 percent of the floor flux. For this case only, the ceiling wall temperature was not sufficiently high to build up the stagnation layer enough to act as an insulator. The secondary flows induced from the floor and sidewalls were able to penetrate the thin stagnation layer and enhance the ceiling Nusselt number slightly above the forced convection Nusselt number. Therefore, while the thickness of the stagnation layer is proportional to the ceiling flux and the flow rate, the *disturbance* of

this layer by the secondary flows is proportional to the heat flux profile (or ratio). This profile effect would also affect the stagnation layer for the inverse of the flux skewing in this work, that is, for a profile where the ceiling has the highest flux. For the inverse profile case, however, the effect would be the opposite. The more skewed toward the ceiling the heat flux profile would be, the thicker the stagnation layer, thus reducing the ability of the bulk flow to penetrate the layer. It should be noted that any one of these effects can be overcome by the other effects. For example, the ceiling Nusselt numbers from Figs. 5.7 and 5.8 would eventually drop below the forced convection Nusselt numbers if the flow were reduced further. The flow effect would build the stagnation layer to a thickness that could not completely be penetrated by the secondary flows.

Correlations are presented for the Nusselt numbers for the five surfaces of the duct over the Reynolds number range of 6000 to 184,000. Mixed convection correlations are applicable below a Reynolds number of 58,000 where the Nusselt number is a function of the Reynolds number, the Grashof number, and the surface heat flux ratio. Forced convection correlations are applicable above a Reynolds number of 85,000 where the Nusselt number is a function of the Reynolds number and the surface heat flux ratio.

These correlations are not as straight forward to use as the typical forced convection correlations, which involve the flow as the only independent variable. The correlations shown here have the heat flux ratio as an independent variable. This ratio, however, is a function of the host rock temperature profile, which in turn is dependent on the heat transfer from the tunnel surface. The mixed convection regime correlations also include the Grashof number, so the surface temperature must also be known. Again, this temperature is a function of the heat transfer from the surface. The use of these correlations is therefore an iterative process. This is common for any process involving mixed convection. A heat flux profile would be assumed, along with surface temperatures. The convection coefficient is calculated from the correlations using these assumptions. This convection coefficient is used to calculate the heat transfer from the tunnel surfaces, and thus the surface temperatures. The procedure is continued until the assumed parameters are within some tolerance of the calculated values. It should be noted that this procedure is certainly not limited to the application to a nuclear waste repository or even a mine drift. This procedure is applicable

to any rectangular duct of similar aspect ratio operating under similar ranges of Reynolds and Grashof numbers.

The data presented here indicate that for Reynolds numbers below 30,000, the convection coefficient remains essentially constant for the range down to a Reynolds number of 6000. If repository ventilation flow rates are in this range, the same heat transfer can be achieved with a reduced flow rate. If the flow rate were reduced by a factor of five, pumping power could be reduced by a factor of 25. Thus, significant cost savings could be realized if these mixed convection results are used in place of the normally used forced convection analyses.

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APPENDIX A - DATA REDUCTION COMPUTER PROGRAMS

Appendix A.1 - Data Reduction Computer Programs - COLDATA

```

        BYTE BUFFER(20,120000),IDUMMY(20)
        CHARACTER*80 TITLE
        CHARACTER*20 FNAME
        CHARACTER*4 RUN
        CHARACTER*1 CLR(7),BELL(3),WIDE(3),SW
        DATA CLR/27,['','2','J',27,['','H'/,BELL/7,7,7/
        DATA WIDE/27,'#','6'/
        IERR=1
        RERR=0.0
        OPEN (UNIT=1,NAME='TXA6:',TYPE='UNKNOWN')
        WRITE (5,5) CLR
5      FORMAT (' '7A1//
        1      ' Make sure the data-logger is on RESET,'//
        2      ' that the printer is OFF, and that'//
        3      ' EXTERNAL ON is OFF.'////)
100    WRITE (5,10)
10      FORMAT (' Enter run number (XXX): ', $)
        READ (5,20,END=999) NR,RUN
20      FORMAT (Q,A)
        IF (NR.NE.3 .AND. NR.NE.4) GOTO 100
110    WRITE(5,25)
25      FORMAT(/' Enter switch position (A or B): ', $)
        READ (5,20,END=999) NS,SW
        IF (NS.EQ.0) GOTO 110
        FNAME='RUN'//RUN//SW//'.DAT'
        OPEN
        (UNIT=4,NAME=FNAME,STATUS='NEW',CARRIAGECONTROL='LIST')
120    WRITE (5,30)
30      FORMAT (/ ' Enter number of scans desired: ', $)
        READ (5,40,END=999) NS,M
40      FORMAT (Q,I)
        IF (NS.EQ.0) GOTO 120
        WRITE (5,45)
45      FORMAT (/ ' Enter number of channels per scan: ', $)
        READ (5,40,END=999) NS,NN
        NN1=NN+1
        IPRINT=10*NN1
        N=M*NN1
        WRITE (5,50)
50      FORMAT (/ ' Enter title of run (80 characters):' /)
        READ (5,20,END=999) NT,TITLE
        IF (NT.EQ.0) TITLE=FNAME(1:3+NR)
        WRITE (4,60) TITLE
60      FORMAT (A)
        WRITE (5,65)
65      FORMAT (////' Depress EXTERNAL ON, then set' /

```

```

1      ' the data-logger to CONTINUOUS'//)
DO 70 I=1,100
READ (1,90,ERR=70) LENGTH,(IDUMMY(J),J=1,LENGTH)
70  IF (LENGTH .LT. 4) GOTO 75
75  IPR=0
DO 80 I=1,N
IF (I/NN+1.GT.IPR) THEN
    IPR=IPR+1
    IF (IPR.GT.M) GOTO 42
    WRITE (5,41) IPR
41  FORMAT ('+', 'Scan #'I2)
ENDIF
42  READ (1,90,ERR=990) LENGTH,(BUFFER(J,I),J=1,LENGTH)
DO 80 J=1,20
IF (BUFFER(J,I) .EQ. '+') BUFFER(J,I) = ' '
IF (BUFFER(J,I) .LT. ' ') BUFFER(J,I) = ' '
80  CONTINUE
90  FORMAT (Q,20A1)
WRITE (5,*) ' '
WRITE (5,91) WIDE
91  FORMAT(' '3A1'SCAN IS FINISHED -- DECODING IN PROGRESS')
NM=0
DO 150 I=1,N
IF (BUFFER(8,I).EQ.'O' .OR. BUFFER(8,I).EQ.'B') GOTO 130
DECODE (12,115,BUFFER(1,I),ERR=146) ICAN,VALUE
115  FORMAT (1X,I3,F8.2)
IF (VALUE.EQ.0.) GOTO 145
WRITE (4,121) I-NM,ICAN,VALUE
121  FORMAT (I5,3X,I3,F8.1)
GOTO 150
130  WRITE (4,140) I-NM
140  FORMAT (I5,3X,' 0',3X,'999.9')
GOTO 150
145  NM=NM+1
GOTO 150
146  WRITE (4,121) IERR,IERR,RERR
150  CONTINUE
GOTO 999
990  WRITE (5,991) BELL
991  FORMAT (/ '3A1 ***** ERROR READING FROM DATALOGGER *****'/
1      '***** RE-RUN DATA ACQUISITION PROGRAM ****')
CALL WAIT
999  CONTINUE
CLOSE (1)
CLOSE (4)
RETURN
END

```

Appendix A.2 - Data Reduction Computer Programs - AVG2

*** This program averages data from several scans
 *** of the FLUKE 2200B data logger. It requires
 *** two input files (positions "A" and "B").

```

      DIMENSION TEMP(150,50),AVTEMP(150)
      CHARACTER*80 TITLE
      CHARACTER*20 FNAME
      CHARACTER*4 RUN
      CHARACTER*1 CLS(6),BELL(3)
      DATA CLS/27,'[','H',27,'[','J',BELL/7,7,7/
100  WRITE (5,5) CLS
5    FORMAT (' '6A1'Enter run number (XXX): '$)
      READ (5,7,END=999) NR,RUN
7    FORMAT (Q,A)
      IF (NR.NE.3 .AND. NR.NE.4) GOTO 100
      FNAME='RUN'//RUN//'A.DAT'
      OPEN
      (UNIT=1,NAME=FNAME,STATUS='OLD',CARRIAGECONTROL='LIST',ERR=990)

      FNAME(4+NR:4+NR)='B'
      OPEN
      (UNIT=2,NAME=FNAME,STATUS='OLD',CARRIAGECONTROL='LIST',ERR=990)

      FNAME(1:4+NR)='DATA'//RUN
      OPEN
      (UNIT=3,NAME=FNAME,STATUS='NEW',CARRIAGECONTROL='LIST')
      READ (1,11) TITLE
11   FORMAT (A)
      WRITE (3,11) TITLE
110  WRITE (5,15)
15   FORMAT (/ ' Input number of scans: '$)
      READ (5,18,END=999) NS,M
18   FORMAT (Q,I)
      IF (NS.EQ.0) GOTO 110

*** Define the number of channels in positions A and B

      NNA=80
      NNB=65
      NN1=NNA+1
      NN2=NNA+NNB
      DO 30 I=1,M
      DO 30 J=1,NNA
30   READ (1,40) TEMP(J,I)
40   FORMAT (11X,F8.1)
      READ (2,11) TITLE

```

```

DO 45 I=1,M
DO 45 J=NN1,NN2
45  READ (2,40) TEMP(J,I)

*** Average the temperatures for the M scans

DO 60 J=1,NN2
SUM=0.
DO 50 I=1,M
50  SUM=SUM+TEMP(J,I)
    AM=FLOAT(M)
    AVTEMP(J)=SUM/AM
60  CONTINUE

*** Average channels 124 and 134 for channel 129.

AVTEMP(130)=(AVTEMP(125)+AVTEMP(135))/2.
IBANK=1
DO 70 J=1,NN2,5
J4= J+4
WRITE(3,101)IBANK,(AVTEMP(I),I=J,J4)
WRITE(5,101)IBANK,(AVTEMP(I),I=J,J4)
101 FORMAT(' ',I9,F11.1,4F8.1)
    IBANK=IBANK+1
70  CONTINUE
    GOTO 999
990  WRITE (5,99) BELL,FNAME
99  FORMAT (' '3A1'Input data file 'A11' not found.')
999  CONTINUE
    CLOSE (1)
    CLOSE (2)
    CLOSE (3)
    WRITE (5,*) 'Press RETURN to continue'
    CALL WAIT
    RETURN
END

```

Appendix A.3 - Data Reduction Computer Programs - REDDATA

*** Data reduction program for tunnel experiments.
 *** Uses IMSL subroutine LEQT2F to solve simultaneous equations
 *** for radiation calculation. LINK with IMSLS object library.

```

REAL NU,NUAVG,NETQ
CHARACTER*1 CLS(6),BELL(3),WIDE(3)
CHARACTER TITLE*80,FNAME*30,RUN*4,BASE*3
DIMENSION V(5),A(5),TBULK(30),DIFTEM(4,5),QLOSS(29,5),
1 DT(8),X(4),TWALL(29,5),DTEM(4,8),XTEMP(29,5),QELEC(5),
2 LOCAL(4),QNET(29,5),H(29,5),Z(29,5),NU(29,5),NUAVG(29,5),
3 ZAVG(29,5),GR(29,5),GRAVG(29,5),HAVG(29,5),NETQ(30),AREA1(5),
4 TCOL(5,1),RADM(5,5),WKAREA(80),TW(5),HT(5)
DATA AREA1(1),AREA1(2),AREA1(3),AREA1(4),AREA1(5)/71.25,72.0,
1 72.0,72.0,71.25/,WIDTH/1.0/,EMISS/0.23/
*** The axial positions, in feet, of the temp. differential setups
*** are explicitly entered into the array named "X".
DATA X(1),X(2),X(3),X(4)/0.25,5.25,10.25,14.25/
DATA HT(1),HT(2),HT(3),HT(4),HT(5)/1.5,1.25,0.75,0.25,1.5/
DATA CLS/27,'I','H',27,'I','J',BELL/7,7,7/,WIDE/27,'#','6'/

OPEN (UNIT=2,FILE='IMSL.ERR',STATUS='SCRATCH')

3000 WRITE (5,1999) CLS
1999 FORMAT (' '6A1'Enter run number (XXX): '$)
      READ (5,901,END=999) NR,RUN
901  FORMAT (Q,A)
      IF (NR.NE.3 .AND. NR.NE.4) GOTO 3000
      FNAME='DATA'//RUN//'.DAT'
      OPEN (UNIT=4,NAME=FNAME,STATUS='OLD',ERR=990)

      FNAME(1:4)='RSLT'
      OPEN (UNIT=3,NAME=FNAME,STATUS='NEW',ERR=990)

      FNAME='[ME.OSBORNE1.TAKEDATA]RADMTRX'
      OPEN (UNIT=8,NAME=FNAME,STATUS='OLD',READONLY,ERR=990)

      DO 4039 I=1,5
4039  READ (8,4040) (RADM(I,J),J=1,5)
4040  FORMAT (5F10.6)

*** Read in the temperature values from the tunnel heater pads
*** "I" represents the bank number
*** "J" represents the row number

      READ (4,1) TITLE
1     FORMAT (A)

```

```

DO 100 I=1,29
  READ (4,*) K,(TWALL(I,J),J=1,5)
100 CONTINUE

  WRITE (5,111)
111 FORMAT (/ ' Input tunnel height (ft.): '$)
  READ (5,*) HITE

*** Calculate the hydraulic diameter (4A/P) in feet

  DH=4.*HITE*WIDTH/(2.*(HITE+WIDTH))

*** Input the Barocel reading and the atmospheric pressure value

  WRITE (5,2)
2  FORMAT (/ ' Input Barocel reading: '$)
  READ (5,*) BAR

  WRITE (5,3)
3  FORMAT (/ ' Input atmospheric pressure (in. Hg): '$)
  READ (5,*) PRESS

*** Input the voltage and amperage values to obtain
*** power inputs in volts and amps, respectively.

2001 DO 200 I=1,5
      WRITE (5,4) I
4      FORMAT (/ ' Input the voltage and amperage',
1          ' readings for VARIAC #'I1': '$)
      READ (5,*) V(I),A(I)
200 CONTINUE

*** Input the inlet air temperature value in degrees Fahrenheit

  WRITE (5,13)
13  FORMAT (/ ' Input inlet air temperature (Deg F): '$)
  READ (5,*) TIN

*** Input the 8 temperature differentials measured across
*** the insulation for any or all of the 4 setups

  WRITE (5,5)
5  FORMAT (/ ' Input the number of temperature differential',
1      ' setups used for this run: '$)
  READ (5,*) NUM

  WRITE (5,15) NUM

```

```

15  FORMAT (/ ' Input the 'I1' setup position numbers' /
      1      ' beginning at the point of entry (1-4): '$)
      READ (5,*) (LOCAL(I),I=1,NUM)

      DO 300 I=1,NUM
      WRITE (5,115) LOCAL(I)
115  FORMAT (/ ' Input the 8 temperature differentials measured'
      1      /' across the insulation for setup #'I1' (mV).'
      2      /' Start with the ceiling temperature, then the upper back'
      3      /' and upper front and continue as such with the floor'
      4      /' temperature last: '$)
      READ (5,*) (DTEM(LOCAL(I),J),J=1,8)

*** Convert the millivolts to degrees F

      DO 299 J=1,8
      VMV=DTEM(LOCAL(I),J)
      DTEM(LOCAL(I),J)=0.040361163*VMV**3 - 1.189841577*VMV**2 +
      1      46.42861891*VMV + 32.06221785
299  CONTINUE

      K=2

      DO 301 J=1,3
      DIFTEM(LOCAL(I),K) =
      1      (DTEM(LOCAL(I),2*J) + DTEM(LOCAL(I),(2*J+1)))/2.
      K=K+1
301  CONTINUE

      DIFTEM(LOCAL(I),1)=DTEM(LOCAL(I),1)
      DIFTEM(LOCAL(I),5)=DTEM(LOCAL(I),8)

300  CONTINUE

*** Input the bank number of interest [not implemented]

C    WRITE (5,6)
C    6      FORMAT (/ ' Input the bank number of interest: '$)
C    READ (5,*) BANKX

*** Input the number of the first bank that is turned on

      WRITE (5,7)
7    FORMAT (/ ' Input the bank number of the first bank'
      1      /' (closest to the point of entry) that is on: '$)
      READ (5,*) NBANK

*** "DATA REDUCTION IN PROGRESS" message

```

```

      WRITE (5,902) WIDE
902  FORMAT (/ ' 3A1,5X'DATA REDUCTION IN PROGRESS')

```

*** Calculate the differential temperature for each heater

```

      CALL LINES(X,NUM,LOCAL,DIFTEM,XTEMP)
      TOTAL=0.
      DO 400 I=NBANK,29

```

*** Calculate the heat lost through the insulation of
 *** each heater from the first bank on to bank 29

```

      CALL LOSS2(XTEMP,QLOSS,I,AREA,TWALL)

```

*** Calculate the "average" bulk temperature for each bank
 *** from the first heated bank to bank 29

```

      CALL THERMO(DH,TBULK,HITE,WIDTH,VEL,BAR,TOTL,
1          V,A,QLOSS,TIN,I,NETQ,NBANK,PRESS,QNET)

      TOTAL=TOTAL+TBULK(I)

```

```

400  CONTINUE

```

*** Calculate the average bulk temperature through the tunnel

```

      TBULK(30)=TOTAL/(29-NBANK+1)

```

*** Calculate the average Reynolds number, Re, in the channel

```

      AMU=FCTMU(TBULK(30))
      RHO=FCTRHO(TBULK(30),PRESS)
      RE=(RHO*VEL*DH/AMU)/60.

```

*** Calculate the heat transfer coefficient at each heater

```

      DO 500 J=NBANK,29
          REWIND 8
          DO 410 I=1,5
410      READ(8,4040)(RADM(I,K),K=1,5)

          GBNEW=FCTGB(TBULK(J))
          THERMC=FCTCON(TBULK(J))

          DO 450 I=1,5
              TWALL(J,I)=TWALL(J,I)+459.7
450      TCOL(I,1)=(TWALL(J,I)**4)*EMISS*1.714E-9

```



```

C      CALL SIMQ(RADM,TCOL,5,KKK)  ! Old DECSSP subroutine
      M=1
      N=5
      IA=5
      IDGT=0
      CALL LEQT2F (RADM,M,N,IA,TCOL,IDGT,WKAREA,IER)
      IF (IER.NE.0) WRITE (5,4501) IER
4501   FORMAT (' Error 'I3' in call to IMSL routine LEQT2F')

      DO 600 M=1,5

      EB=(TWALL(J,M)**4)*1.714E-9
      QRAD=EMISS/(1.-EMISS)*(EB-TCOL(M,1))*AREA1(M)/491.3424
      QNET(J,M)=QNET(J,M)-QRAD

      H(J,M)=(QNET(J,M)/(AREA1(M)*(TWALL(J,M)-TBULK(J))))*
1      491.3424

      GR(J,M)=GBNEW*(TWALL(J,M)-TBULK(J))*HT(M)**3

      NU(J,M)=H(J,M)*DH/THERMC

600   CONTINUE
500   CONTINUE
      CALL AVRAGE(H,HAVG,NBANK)
      CALL AVRAGE(NU,NUAVG,NBANK)
      CALL AVRAGE(GR,GRAVG,NBANK)
      WRITE (3,698) TITLE,RUN,RE
698   FORMAT ('1'20X,A//54X,'RUN ',A,4X,'Re= ',F7.0//)
      WRITE (3,6978)
6978  FORMAT (11X,'BANK',5X,'C',6X,'US',6X,'MS',6X,'LS',6X,'FL',
1      13X,'BULK'/)
      DO 6980 I=NBANK,29
      DO 6979 J=1,5
6979  TW(J)=TWALL(I,J)-459.7
      TBULK(I)=TBULK(I)-459.7
6980  WRITE (3,6981) I,(TW(J),J=1,5),TBULK(I)
6981  FORMAT (11X,I3,5F8.1,F16.1)
      IBANKON = 30 - NBANK
      WRITE (3,6982) HITE,NBANK,IBANKON,TIN,PRESS,BAR
6982  FORMAT (///9X,'HEIGHT=',F4.1,' Ft',4X,'FIRST BANK=',I3,4X,
1      '# OF BANKS=',I3,4X,'TIN=',F5.1,' F',4X,'Atm Press=',
2      F6.2,' inHg',4X,'Barocel=',F6.3,' mmHg')
      WRITE (3,6983) V(1),A(1),V(2),A(2),V(3),A(3),V(4),A(4),V(5),A(5)
6983  FORMAT (///9X,'POWER INPUTS',T32,'VOLTS',T43,'AMPS'/
1      14X,'CEILING',T30,F7.3,4X,F6.3/
2      14X,'UPPER SIDE',T30,F7.3,4X,F6.3/
3      14X,'MIDDLE SIDE',T30,F7.3,4X,F6.3/

```

```

4      14X,'LOWER SIDE',T30,F7.3,4X,F6.3/
5      14X,'FLOOR',T30,F7.3,4X,F6.3)

IBASE=5
IF (A(3)*V(3).GT.A(5)*V(5)) THEN
    IBASE=3
    BASE='QMS'
ELSE
    IBASE=5
    BASE='QFL'
ENDIF
WRITE (3,698) TITLE,RUN,RE
WRITE (3,6990) BASE
6990  FORMAT (5X,'NET HEAT FLUX RATIOS'///11X,'BANK',6X,'QC',7X,
1      'QUS',7X,'QMS',7X,'QLS',7X,'QFL',6X,A' (W/SqIn)'/)
DO 6950 I=NBANK,29
    QC=QNET(I,1)/QNET(I,IBASE)
    QUS=QNET(I,2)/QNET(I,IBASE)
    QMS=QNET(I,3)/QNET(I,IBASE)
    QLS=QNET(I,4)/QNET(I,IBASE)
    QFL=QNET(I,5)/QNET(I,IBASE)
6950  WRITE (3,6940) I,QC,QUS,QMS,QLS,QFL,QNET(I,IBASE)/AREA1(IBASE)
6940  FORMAT (11X,I3,5F10.4,5X,F8.4)

WRITE (3,698) TITLE,RUN,RE
WRITE (3,699)
699  FORMAT (' THE FOLLOWING VALUES OF THE HEAT TRANSFER',
1      ' COEFFICIENT ARE IN UNITS OF BTU/Hr-SqFt-F'/
2      ' THE FIRST FIVE COLUMNS CONTAIN THE "H" VALUES FOR',
3      ' THE TUNNEL ROWS. ',/, ' THE NEXT FIVE COLUMNS CONTAIN',
4      ' THE AVERAGED "H" VALUES FOR THE 5 ROWS.'//)

WRITE (3,700)
700  FORMAT (6X,'BANK',11X,'H1',7X,'H2',7X,'H3',7X,'H4',7X,
1      'H5',5X,'H1AVG',4X,'H2AVG',4X,'H3AVG',4X,
2      'H4AVG',4X,'H5AVG',/)

DO 702 I=NBANK,29
    WRITE (3,701) I,(H(I,J),J=1,5),(HAVG(I,K),K=1,5)
702  CONTINUE
701  FORMAT (' ',5X,I3,6X,10F9.3)

WRITE (3,698) TITLE,RUN,RE
WRITE (3,706)
706  FORMAT (' THE FOLLOWING VALUES ARE THE NUSSELT NUMBERS FOR',
1      ' THIS TUNNEL RUN.'/)

```

```

2      ' THE FIRST FIVE COLUMNS CONTAIN THE "Nu" VALUES'
3      ' FOR THE TUNNEL ROWS.'/
4      ' THE NEXT FIVE COLUMNS CONTAIN THE AVERAGED'
5      ' "Nu" VALUES FOR THE 5 ROWS.'//)

WRITE (3,707)
707  FORMAT (3X,'BANK',7X,'Nu1',7X,'Nu2',7X,'Nu3',7X,'Nu4',7X,
1      'Nu5',5X,'Nu1AVG',4X,'Nu2AVG',4X,'Nu3AVG',4X,
2      'Nu4AVG',4X,'Nu5AVG',/)

DO 708 I=NBANK,29
WRITE (3,783) I,(NU(I,J),J=1,5),(NUAVG(I,K),K=1,5)
708  CONTINUE
783  FORMAT (' ',2X,I3,2X,10F10.2)

WRITE (3,698) TITLE,RUN,RE
WRITE (3,703)
703  FORMAT (' THE FOLLOWING VALUES ARE THE GRASHOF NUMBERS FOR',

1      ' THIS TUNNEL RUN.'/
2      ' THE FIRST FIVE COLUMNS CONTAIN THE "Gr" VALUES'
3      ' FOR THE TUNNEL ROWS.'/
4      ' THE NEXT FIVE COLUMNS CONTAIN THE AVERAGED'
5      ' "Gr" VALUES FOR THE 5 ROWS.'//)

WRITE (3,704)
704  FORMAT (3X,'BANK',5X,'Gr1',9X,'Gr2',9X,'Gr3',9X,'Gr4',9X,
1      'Gr5',7X,'Gr1AVG',6X,'Gr2AVG',6X,'Gr3AVG',6X,
2      'Gr4AVG',6X,'Gr5AVG',/)

DO 705 I=NBANK,29
WRITE(3,780) I,(GR(I,J),J=1,5),(GRAVG(I,K),K=1,5)
705  CONTINUE
780  FORMAT(' ',2X,I3,10E12.4)

GOTO 999

990  WRITE (5,991) BELL,FNAME
991  FORMAT (' '3A1/' Unable to open 'A)
CALL WAIT

999  CONTINUE
CLOSE (2)
CLOSE (3)
CLOSE (4)
CLOSE (8)
RETURN

```

END

```

C***** SUBROUTINE LINES *****
C
C
C   SUBROUTINE LINES(X,NUM,LOCAL,DIFTEM,XTEMP)
C
C
C
C   DIMENSION X(4),LOCAL(4),DIFTEM(4,5),XTEMP(29,5)
C
C
C THIS SUBROUTINE CALCULATES THE EQUATION OF THE STRAIGHT LINE
C BETWEEN 2 POINTS,GIVEN THE POINT COORDINATES. IT ALSO DETERMINES
C A DIFFERENTIAL TEMPERATURE VALUE FOR EACH HEATER.
C
C THE DATA POINTS ARE ASSUMED CONNECTED BY A STRAIGHT LINE.
C
C
C   DO 1000 M=1,5
C
C DETERMINE THE EQUATION OF THE LINES FOR 5 SETS OF DATA (ONE
C SET FOR EACH "ROW" OF HEATERS).
C
C   K=NUM-1
C
C   N=1
C
C   DO 100 I=1,K
C
C
C CALCULATE THE SLOPE OF THE LINE
C
C   $   EM=(DIFTEM(LOCAL(I+1),M)-DIFTEM(LOCAL(I),M))/
C       (X(LOCAL(I+1))-X(LOCAL(I)))
C
C
C CALCULATE THE Y-INTERCEPT
C
C   B=DIFTEM(LOCAL(I),M)-EM*X(LOCAL(I))
C
C
C N AND NX REPRESENT BANK NUMBERS.
C
C   NX=(X(LOCAL(I+1))+0.25)/0.5
C

```

```

      DO 101 J=N,NX
C
C BANK IS THE X-COORDINATE (OR AXIAL DISTANCE TO BANK J).
C
      BANK=0.5*J-0.25
C
C XTEMP(J,M) IS THE Y-COORDINATE (OR DIFFERENTIAL TEMPERATURE FOR
C EACH HEATER IN EACH BANK).
C
      XTEMP(J,M)=EM*BANK+B
C
101  CONTINUE
C
      N=NX+1
C
100  CONTINUE
C
C THIS EXTRAPOLATES THE LINE UP IF BANK IS GREATER THEN OUR RANGE
C OF
C TEMPERATURE DIFFERENTIAL SETUPS.
C
      IF (N.NE.30) THEN
C
      DO 200 J=N,29
C
      BANK=0.5*J-0.25
C
      XTEMP(J,M)=EM*BANK+B
C
200  CONTINUE
C
      END IF
C
1000 CONTINUE
C
      RETURN
      END
C
C
C***** SUBROUTINE THERMO *****
C
C
C
C SUBROUTINE THERMO(DH,TBULK,HITE,WIDTH,VEL,BAR,TOTL,
$      V,A,QLOSS,TIN,I,NETQ,NBANK,PRESS,QNET)
C
C
C REAL NETQ
C DIMENSION V(5),A(5),QLOSS(29,5),TBULK(30),QNET(29,5),

```

```

$      QELEC(5),NETQ(30)
C
C
C
C
C
C
C CALCULATE THE ELECTRICAL POWER INPUT UP TO BANK X.
C
C      DO 100 K=1,5
C
C          POW=V(K)*A(K)
C          QELEC(K)=POW/(29-NBANK+1)
C
C
C 100  CONTINUE
C
C CALCULATE THE NET HEAT INPUT PER HEATER(LABELLED AS QNET(I,M)).
C NETQ(I) IS THE TOTAL HEAT INPUT FOR A HEATER BANK.
C
C      HEAT=0.
C
C      DO 101 M=1,5
C
C          QNET(I,M)=QELEC(M)-QLOSS(I,M)
C
C          HEATIN=QNET(I,M)
C
C          HEAT=HEAT+HEATIN
C
C 101  CONTINUE
C
C      NETQ(I)=HEAT
C
C CALCULATE THE VOLUMETRIC FLOW RATE IN CU.FT./MIN. FROM THE
C BAROCEL READING.
C
C      CFM=1652.6*SQRT(0.5354*BAR)
C
C CALCULATE THE VELOCITY IN FT./MIN.
C
C      VEL=CFM/(HITE*WIDTH)
C
C CALCULATE THE BULK TEMPERATURE FOR BANK X.
C SINCE DENSITY AND SPECIFIC HEAT ARE FUNCTIONS OF TBULK,
C AN ITERATIVE SOLUTION IS NECESSARY.
C
C CONVERT TIN TO DEGREE RANKINE.
C

```

```

      RTIN=TIN+459.7
C
C AS AN INITIAL GUESS FOR TBULK TAKE TBULK=TBULK(NBANK-1).
C
      TBULK(NBANK-1)=RTIN
      TBULK(I)=TBULK(I-1)
C
200  RHO=FCTRHO(TBULK(I),PRESS)
C
      CP=FCTCP(TBULK(I))
C
C THE VALUE OF TBULK CALCULATED FROM THE DEPENDENT VARIABLES,
C RHO AND CP, IS LABELED TBNEW.
C
      TBNEW=TBULK(I-1)+(NETQ(I)/(HITE*WIDTH*RHO*VEL*CP))*(3.413/60)
C
      DELTB=ABS(TBNEW-TBULK(I))
C
      TBULK(I)=TBNEW
C
C DETERMINE IF THE VALUES HAVE CONVERGED SATISFACTORILY.
C
      IF(DELTB.GT.0.1) GO TO 200
C
      RETURN
      END
C
C
C*****SUBROUTINE AVRAGE*****
C
      SUBROUTINE AVRAGE(Z,ZAVG,NBANK)
C
C      THIS SUBROUTINE CALCULATES THE AVERAGE VALUE OF Z FOR
C      N VALUES, THEN THE LAST N-1 VALUES, ETC. ALSO FOR THE
C      FIRST N-1 VALUES ETC.
C
      DIMENSION Z(29,5),ZAVG(29,5)
      DO 10 K=1,5
        SUMZ=0.
        DO 20 I=NBANK,29
          SUMZ=SUMZ + Z(I,K)
          TOTAL=I+1-NBANK
          ZAVG(I,K)=SUMZ/TOTAL
        20  CONTINUE
      10  CONTINUE
C
C
C

```

```

RETURN
END

```

```

C
C
C*****FUNCTION SUBPROGRAM TO CALCULATE "K" OF THE
AIR*****

```

```

C
C

```

```

    FUNCTION FCTCON(T)

```

```

C
C

```

```

C THIS FUNCTION SUBPROGRAM CALCULATES THE THERMAL CONDUCTIVITY
OF AIR

```

```

C IN (BTU/HR-FT-R).

```

```

C
C

```

```

    FCTCON=3.0390057E-3 + 2.0811365E-5*T + 2.4299057E-9*T**2 -
$    1.3913557E-12*T**3

```

```

C
C

```

```

    RETURN
END

```

```

C
C

```

```

C*****FUNCTION SUBPROGRAM TO CALCULATE SPECIFIC
HEAT,CP*****

```

```

C
C

```

```

    FUNCTION FCTCP(T)

```

```

C
C

```

```

C THIS FUNCTION SUBPROGRAM CALCULATES THE SPECIFIC HEAT CAPACITY,

```

```

C CP, IN (BTU/LBM-R).

```

```

C
C

```

```

    FCTCP=4.6269047E-1 - 1.1732704E-3*T +
$    2.0490854E-6*T**2 - 1.1736553E-9*T**3

```

```

C
C

```

```

    RETURN
END

```

```

C
C

```

```

C*****FUNCTION SUBPROGRAM TO CALCULATE DENSITY,RHO*****

```

```

C
C

```

```

    FUNCTION FCTRHO(T,PRESS)

```



```

C
C
C THIS FUNCTION SUBPROGRAM CALCULATES THE DENSITY, RHO, IN
C (LBM/CU.FT.).
C
C
C THE SPECIFIC GAS CONSTANT AS DEFINED IN (CU.FT.*IN.HG/LBM.R) IS:
C
    R=0.754
C
    FCTRHO=PRESS/(R*T)
C
C
C
    RETURN
    END
C
C
C*****FUNCTION SUBPROGRAM TO CALCULATE KINEMATIC
C VISCOSITY,AMU*****
C
C
    FUNCTION FCTMU(T)
C
C
C THIS FUNCTION SUBPROGRAM CALCULATES THE KINEMATIC VISCOSITY,
C AMU, IN (LBM/FT-SEC).
C
C
    FCTMU= -7.3891851E-5 + 4.1901472E-7*T -
$      6.9455950E-10*T**2 + 3.9771130E-13*T**3
C
C
    RETURN
    END
C
C
C*****FUNCTION SUBPROGRAM TO CALCULATE GBNEW *****
C
C
    FUNCTION FCTGB(TBULK)
C
C
C THIS FUNCTION SUBPROGRAM CALCULATES THE CONSTANT COEFFICIENT
C USED FOR THE GRASHOF NUMBER,GBNEW,IN (1/R-CU.FT.).
C
C
    FCTGB=9.3521348E7 - 4.2518226E5*TBULK + 6.6237664E2*TBULK**2

```

```

$      - 3.5018855E-1*TBULK**3
C
C
      RETURN
      END
C
C
C***** SUBROUTINE LOSS2 *****
C
      SUBROUTINE LOSS2(XTEMP,QLOSS,I,AREA,TWALL)
C
C
      DIMENSION QLOSS(29,5),XTEMP(29,5),TWALL(29,5)
C
C
C THIS SUBROUTINE CALCULATES THE HEAT LOST IN WATTS ACROSS THE
C INSULATION AT ANY HEATER LOCATION.
C
C
C THE THICKNESS OF THE INSULATION, IN FEET.
C
      THICK=0.125
C
C THE AREA PERPENDICULAR TO THE HEAT CONDUCTION PATH, IN SQUARE
C FEET.
C
      AREA=0.5
C
C
      DO 200 M=1,5
C
C DETERMINE THE THERMAL CONDUCTIVITY OF THE INSULATION AS A
C FUNCTION
C OF INSULATION TEMPERATURE.
C
      TINS = TWALL(I,M)-(XTEMP(I,M)/2.)
      CONDIN=FCTINS(TINS)
C
C CALCULATE THE HEAT LOSS AT A SPECIFIC BANK.
C
      QLOSS(I,M)=(CONDIN*XTEMP(I,M)*AREA/THICK)/3.413
C
C
C
200  CONTINUE
C
C
      RETURN
      END

```

C*****FUNCTION SUBPROGRAM TO CALCULATE "K" OF THE
INSULATION*****

C

FUNCTION FCTINS(T)

C

C

C THIS FUNCTION SUBPROGRAM CALCULATES THE THERMAL CONDUCTIVITY
OF

C THE INSULATION IN (BTU/HR-FT-F).

C

C

C

FCTINS=0.0272 + 7.2447E-6*T + 3.3085E-8*T**2 -
\$ 4.9768E-12*T**3

C

C

RETURN

END

C

C

SUBROUTINE WAIT

*** Waits for <CR> to be pressed before continuing

CHARACTER*1 CHAR

READ (5,10,END=999) IC,CHAR

999 RETURN

10 FORMAT (Q,A)

END

APPENDIX B - HEAT TRANSFER PROPERTIES

Appendix B - Heat transfer properties

SALT (Source 1)

B.1 - Thermal Conductivity

$$k(T) = 2.62 - (4.47\text{E}-3)T + (4.08\text{E}-6)T^2 \quad (\text{B.1})$$

where k is in units of BTU/(hr • ft • °F) and T is the salt temperature in degrees F.

AIR (Source 2)

B.2 - Thermal Conductivity

$$k(T) = (3.0390\text{E}-3) + (2.0811\text{E}-5)T + (2.4299\text{E}-9)T^2 - (1.3914\text{E}-12)T^3 \quad (\text{B.2})$$

where k is in units of BTU/(hr • ft • °F) and T is the air temperature in degrees R.

B.3 - Specific Heat

$$c_p(T) = (4.6269\text{E}-1) - (1.1733\text{E}-3)T + (2.0491\text{E}-6)T^2 - (1.1736\text{E}-9)T^3 \quad (\text{B.3})$$

where c_p is in units of BTU/(lb_m • °F) and T is the air temperature in degrees R.

B.4 - Dynamic Viscosity

$$\mu(T) = (-7.3892\text{E}-5) + (4.1901\text{E}-7)T - (6.9456\text{E}-10)T^2 + (3.9771\text{E}-13)T^3 \quad (\text{B.4})$$

where μ is in units of lb_m/(ft • sec) and T is the air temperature in degrees R.

B.5 - Grashof Parameter

$$\frac{g\beta}{\nu^2}(T) = (9.3521\text{E}+7) - (4.2518\text{E}+5)T + (6.6238\text{E}+2)T^2 - (3.5019\text{E}-1)T^3 \quad (\text{B.5})$$

where $g\beta/\nu^2$ is in units of 1/(°F • ft³) and T is the air temperature in degrees R.

B.6 - Density

$$\rho(T,p) = \frac{p}{RT} \quad (\text{B.6})$$

where ρ is in units of lb_m/ft^3 , p is the atmospheric pressure in inches of Hg, R is the ideal gas constant for air, $= 0.754 \text{ (ft}^3 \cdot \text{in Hg)/ (lb}_m \cdot ^\circ\text{F)}$ and T is the air temperature in degrees R.

PLYSULATE (insulation) (Source 3)B.7 - Thermal Conductivity

$$k(T) = (2.720\text{E}-2) + (7.245\text{E}-6)T + (3.308\text{E}-8)T^2 - (4.977\text{E}-12)T^3 \quad (\text{B.7})$$

where k is in units of $\text{BTU}/(\text{hr} \cdot \text{ft} \cdot ^\circ\text{F})$ and T is the air temperature in degrees R.

SOURCES

1. Loken, M.C.; Callahan, G.D.; Svalstad, D.K.; Wagner, R.A., 1981, "Thermomechanical Analysis of Conceptual Repository Designs for the Paradox and Permian Basins", Report RSI-0204, RE/SPEC, Inc., Rapid City, SD.
2. Kreith, F.; Black, W.Z., 1981, Basic Heat Transfer, Harper & Row, Publishers, New York.
3. Plisulate 201 Technical Specifications, Plibrico Company, Chicago.

APPENDIX C - AIR FLOW METER CALIBRATION

Appendix C - Air Flow meter calibration

BRANDT INDUSTRIES, INC.-ENG SERVICES FORM F3			
JOB NO	8210CH010U1	CONDITIONS=	ACT STD UNITS
MODEL NO	B-NZP1031-12-1-4	MW (2)	28.94 28.94 -----
TAG NO		TEMP (0)	70 68 DEG F
EFF AREA	.412 DP FOR CALC 2.250	STATIC (1)	.0 .0 IN WC
%OUTOUT	IS A SQR ROOT CURVE TO FLOW	DENS (5)	.07480 .07508 LB/CUFT

%OUTPUT	DP	ACFM	SCFM
2	0.045	350.6	349.2
4	0.090	495.8	493.9
6	0.135	607.2	604.9
8	0.180	701.1	698.5
10	0.225	783.9	780.9
12	0.270	858.7	855.5
14	0.315	927.5	924.0
16	0.360	991.5	987.8
18	0.405	1051.7	1047.7
20	0.450	1108.6	1104.4
22	0.495	1162.7	1158.3
24	0.540	1214.4	1209.8
26	0.585	1264.0	1259.2
28	0.630	1311.7	1306.7
30	0.675	1357.7	1352.6
32	0.720	1402.3	1397.0
34	0.765	1445.4	1440.0
36	0.810	1487.3	1481.7
38	0.855	1528.1	1522.3
40	0.900	1567.8	1561.9
42	0.945	1606.5	1600.4
44	0.990	1644.3	1638.1
46	1.035	1681.2	1674.9
48	1.080	1717.4	1710.9
50	1.125	1752.8	1746.2
52	1.170	1787.5	1780.8
54	1.215	1821.6	1814.7
56	1.260	1855.0	1848.0
58	1.305	1887.8	1880.7
60	1.350	1920.1	1912.9
62	1.395	1951.9	1944.5
64	1.440	1983.1	1975.6
66	1.485	2013.8	2006.2
68	1.530	2044.1	2036.4
70	1.575	2074.0	2066.1
72	1.620	2103.4	2095.5
74	1.665	2132.4	2124.4
76	1.710	2161.0	2152.9
78	1.755	2189.3	2181.0
80	1.800	2217.2	2208.8
82	1.845	2244.7	2236.2
84	1.890	2271.9	2263.4
86	1.935	2298.8	2290.1
88	1.980	2325.4	2316.6
90	2.025	2351.7	2342.8
92	2.070	2377.6	2368.7
94	2.115	2403.4	2394.3
96	2.160	2428.8	2419.6
98	2.205	2454.0	2444.7
100	2.250	2478.9	2469.5

APPENDIX D - HEATING5 REPOSITORY MODEL INPUT DATA

Appendix D - HEATING5 repository model input data

RE=10K ITERATIVE H VALUES

1000	7	24	1		1	6
4	10		1			
10			1			
1	1.E-4					
1	1	0.	6.	490.	500.	
1						5
2	1	0.	6.	482.	490.	
1		6				
3	1	0.	6.	450.	482.	
1						
4	1	0.	6.	400.	450.	
1						
5	1	6.	20.	400.	450.	
1						
6	1	20.	40.	400.	450.	
1						
7	1	6.	20.	450.	482.	
1						
8	1	20.	40.	450.	482.	
1						
9	1	6.	20.	482.	490.	
1						
10	1	20.	40.	482.	490.	
1						
11	1	6.	20.	490.	500.	
1						
12	1	20.	40.	490.	500.	
1						
13	1	6.	20.	500.	506.	
1		4				
14	1	20.	40.	500.	506.	
1						
15	1	6.	20.	506.	512.	
1		3				
16	1	20.	40.	506.	512.	
1						
17	1	6.	20.	512.	518.	
1		2				
18	1	20.	40.	512.	518.	
1						
19	1	0.	6.	518.	538.	
1				1		
20	1	6.	20.	518.	538.	
1						
21	1	20.	40.	518.	538.	
1						
22	1	0.	6.	538.	600.	
1						
23	1	6.	20.	538.	600.	
1						
24	1	20.	40.	538.	600.	
1						
1	SALT	1.0				1
1	140.0					
1	1	79.5				
2.45D-02						
2	1	79.5				
6.60D-02						

3	1	79.5					
8.67D-02							
4	1	79.5					
1.244D-01							
5	1	79.5					
1.426D-01							
6	1	350.					
18.96							
0.	6.	20.	40.				
6	7	5					
400.	450.	482.	490.	500.	506.	512.	518.
538.	600.						
5	8	8	10	6	6	6	10
4							
1	3						
1	2.623	2-4.461E-3		3	4.074E-6		
5	1	4	8	12	36		

APPENDIX E - SIDEWALL CONDUCTION ANALYSIS

Appendix E - Sidewall conduction analysis

Vertical conduction through the adjacent sidewall heaters on each bank increased or decreased the net heat flux entering the duct from the upper, middle, or lower sidewall surfaces. The temperature change between two heaters in the axial direction was less than 5 °F, so axial conduction was not considered. A computer analysis of the conduction heat transfer along the vertical wall was performed using the finite difference conduction software program HEATING5 (Turner, et al., 1977). The geometry modelled is shown in Figure E.1, and the HEATING5 input data set is shown in Table E.1. The results reveal that less than 3 percent of the electrical heat input to the heater pad was conducted to an adjacent wall surface. The exception to this was the upper sidewall, where almost 10 percent of the heat input was conducted to the middle sidewall at $Re = 10,000$. This situation arose since the flux profile along the wall was non-uniform, causing large temperature differences between adjacent wall surfaces. At higher Reynolds numbers, this conduction was on the order of 5 percent or less of the electrical power input to the wall surface.

Figure E.1. HEATING5 sidewall conduction model.

Table E.1. HEATING5 input data for sidewall conduction.

SIDE WALL CONDUCTION MODEL - RUN 058, CHF, RE=13,500

1000	7	9	1	3	18
2	10		1		
1	1000	.0001			
1	1	0.	0.0625	0.	2.0
1		1	2		
2	1	0.	0.0625	2.0	4.0
1		3	4		
3	1	0.	0.0625	4.0	6.0
1		5	6		
4	1	0.	0.0625	6.0	8.0
2		7	8		
5	1	0.	0.0625	8.0	10.0
2		9	10		
6	1	0.	0.0625	10.0	12.0
2		11	12		
7	1	0.	0.0625	12.0	14.0
3		13	14		
8	1	0.	0.0625	14.0	16.0
3		15	16		
9	1	0.	0.0625	16.0	18.0
3		17	18		
1	STEEL	1.0			1
1	291.0				
2	304.0				
3	331.0				
1	1	291.0			
1.7179					
2	1	138.0			
1.024D-2					
3	1	291.0			
1.7179					
4	1	128.0			
1.024D-2					
5	1	291.0			
1.7179					
6	1	128.0			
1.024D-2					
7	1	304.0			
1.6637					
8	1	128.0			
9.243D-3					
9	1	304.0			
1.6637					
10	1	128.0			
9.243D-3					
11	1	304.0			
1.6637					
12	1	140.0			
9.243D-3					
13	1	331.0			
1.519					
14	1	148.0			
7.30D-3					
15	1	331.0			
1.519					
16	1	163.0			
7.30D-3					
17	1	331.0			

```

1.519
18      1      203.0
7.30D-3
0.      0.0625
4
0.      2.0      4.0      6.0      8.0      10.0      12.0      14.0
16.0      18.0
4      4      4      4      4      4      4      4
4
1      3
1      2.5162      2 -1.69E-4      3 -5.88E-7

```


APPENDIX F - EMISSIVITY MEASUREMENT OF GALVANIZED STEEL

Appendix F - Emissivity measurement of galvanized sheet steel

The total emissivity of the galvanized sheet steel was found in the literature, and was confirmed with a simple experiment. Sparrow and Cess (1966, Table 2-2) and Touloukian, et al. (1970, Vol. 7, Table 251 and Vol 9, Table 781) gave the emissivity of zinc galvanized surface as 0.23.

E.1 Experimental Confirmation of Wall Emissivity.

The emissivity was determined using a thermopile radiometer. The radiometer consists of a parabolic collector, which concentrates incoming radiation on a row of blackened thermocouples, which forms the hot junction of the thermopile. The cold junctions of the thermopile are in the body of the radiometer, which is essentially at room temperature. The signal produced from the thermopile is proportional to the temperature difference of these two junctions. This temperature difference in turn is proportional to the radiant energy entering the parabolic collector (Doebelin, 1975, Measurement Systems, Rev. Ed., p. 548), and the radiant energy is proportional to the fourth power of the emitting surfaces temperature.

$$R \propto \Delta T_{thermopile} \quad (F.1)$$

$$\Delta T_{thermopile} \propto Q_r \quad (F.2)$$

$$Q_r \propto [T_{surface}^4 - T_{Thermopile}^4] \quad (F.3)$$

where

- R = radiometer output voltage
- $\Delta T_{thermopile}$ = differential temperature between the two thermopile thermocouple junctions
- Q_r = radiated heat flow
- $T_{surface}^4, T_{thermopile}^4$ = absolute temperatures of the respective surfaces

Two parameters that go into the proportionality coefficients are the emissivities of the two surfaces and the configuration or view factor between the two surfaces.

The radiant energy transfer between any two infinite parallel plates is

$$q'' = \frac{\sigma_{SB}(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad (\text{F.4})$$

where

T_1, T_2 = absolute temperatures of the two surfaces

σ_{SB} = Stefan-Boltzmann constant

ϵ_1, ϵ_2 = diffuse emissivities of the two surfaces

This equation assumes that the radiation heat transfer is gray and diffuse. The reflecting inner surface of the radiometer concentrates radiation on the thermopile. Therefore, the opening plane of the radiometer was assumed to behave as a black body at room temperature. Equation F.4 then becomes

$$Q = \epsilon_s A_s \sigma (T_s^4 - T_\infty^4) \quad (\text{F.5})$$

where

T_s, T_∞ = absolute temperatures of the surface and the room

ϵ_s = diffuse emissivities of the surface

Q in Eqn. F.3 is also equal to KR , where K is a radiometer constant, particular to an individual radiometer. If R is measured for a surface of known ϵ , the unknown ϵ of a surface can be calculated from applying Eqn. F.5 to each surface.

Dividing these two equations, we have a relationship between the known emissivity ϵ_1 of surface 1 to the unknown emissivity ϵ_2 of surface 2, both having the same area A .

$$Q_1 = \epsilon_1 A_1 \sigma [T_1^A - T_\infty^A] = KR_1 \quad (\text{F.6})$$

$$Q_2 = \epsilon_2 A_2 \sigma [T_2^A - T_\infty^A] = KR_2$$

$$\epsilon_2 = \epsilon_1 \frac{R_2}{R_1} \left[\frac{T_1^A - T_\infty^A}{T_2^A - T_\infty^A} \right] \quad (\text{F.7})$$

Figure F.1 shows schematically the test setup.

Table F.1 gives the measured emissivity of the galvanized sheet steel as a function of temperature. The experimental values correlate very well with the literature values.

Table F.1. Measured Values of Total Diffuse Emissivity of Galvanized Sheet Steel.

Temperature	ϵ
100°F	0.22
200°F	0.22
300°F	0.23
400°F	0.23

Figure F.1 Test setup for emissivity measurement.

APPENDIX G - THERMAL RADIATION CORRECTION TO NET HEAT INPUT

Appendix G - Thermal radiation correction to net heat input

The net heat transferred from a particular heater surface is composed of the electrical heat from the resistance heaters, the heat lost out the walls of the tunnel, and the heat radiated from this surface. The radiation heat transfer may be into or out of the surface.

$$q''_{net} = q''_{electric} - q''_{loss} - q''_{radiation} \quad (G.1)$$

where

- q''_{net} = net heat flux leaving the surface (energy per unit area)
- $q''_{electrical}$ = electrical power heat flux input
- q''_{loss} = heat flux leaving the surface by conduction through the tunnel's walls to the room
- q''_{rad} = heat flux leaving the surface by thermally radiating to other tunnel surfaces

This section describes the procedure for estimating the q''_{rad} term.

When the data was initially reduced, a very simple radiation correction scheme was used, based on the network method of radiation analysis (Oppenheim, 1956). In the network method, surfaces are connected to each other through "resistances". "Surface resistance" is the resistance to heat transfer because a surface is not black (i.e., $\epsilon \neq 1$), consisting of the surface's emissivity and its area. "Space resistances" are the heat transfer resistances between the considered surface and all other surfaces, consisting of the considered surface's area and the view factors between this surface and all the other surfaces. The radiosity, J , is the total radiation which leaves a surface per unit time per unit area. The *net* radiation leaving the surface may be also thought of as a current flow. This current flows because of a potential difference (the difference between the black body emissive power E_b and the radiosity, J) across a resistance (the surface resistance):

$$q'' = \frac{E_b - J}{\left(\frac{1 - \epsilon}{\epsilon A} \right)} \quad (\text{G.2})$$

where

$$E_b = \sigma T_{\text{surface}}^4 \quad (\text{G.3})$$

Similarly, the heat transfer between any two surfaces is

$$q''_{1-2} = \frac{J_1 - J_2}{\left(\frac{1}{A_1 F_{1-2}} \right)} \quad (\text{G.4})$$

For a system with N surfaces, the above equations may be combined to give (Holman, 1976):

$$J_i = (1 - \epsilon_i) \sum_{j \neq i}^N (F_{i-j} J_j + \epsilon_i E_{b,i}) \quad (\text{G.5})$$

Evaluating Eqn. G.5 for the N surfaces results in N equations with N unknowns, J_i for each surface. All the other terms in Eqn. G.5 are known: $E_{b,i}$, ϵ_i , and F_{i-j} . The system of equations is solved for the J_i terms, then q''_{rad} is calculated from Eqn. G.2.

Therefore, view factors were needed. The duct was modeled as being constructed of eight infinitely long plates all having the same emissivity. Figure G.1 shows this geometry, looking axially down the duct.

The infinite plate view factors were calculated using the geometry in Figure G.2 and the following equations.

Figure G.1. Infinite length duct surface arrangement.

Figure G.2. Geometries for infinite length view factors.

The view factor for infinitely long, directly opposed parallel plates of spacing h and the same width w (Figure G.2-a) is given by (Hottel, 1931):

$$F_{1-2} = \sqrt{1 + \left(\frac{h}{w}\right)^2} - \frac{h}{w} \quad (\text{G.6})$$

The view factor for infinitely long parallel plates of different widths b and c , centerlines of each plate are connected by perpendicular line of length a between plates (Figure G.2-b) is given by (Howell, 1982):

$$F_{1-2} = \frac{1}{2B} [\sqrt{(B+C)^2 + 4} - \sqrt{(C-B)^2 + 4}] \quad (\text{G.7})$$

$$\text{where } B = \frac{b}{a} \quad \text{and} \quad C = \frac{c}{a}$$

The view factor for infinitely long perpendicular plates of different widths h and w , having a common edge (Figure G.2-c) is given by (Siegel and Howell, 1981):

$$F_{1-2} = \frac{1}{2} \left[1 + \left(\frac{h}{w}\right) - \sqrt{1 + \left(\frac{h}{w}\right)^2} \right] \quad (\text{G.8})$$

Eqn. G.6 was used directly to calculate the view factors for 1-8 (Surface 1 to Surface 8), 2-3, 4-5, and 6-7. Eqn. G.8 was used directly to calculate the view factor for 1-2, which is equal to 1-3, 8-6 and 8-7. Reciprocity was used to calculate the reverse of these factors. For example:

$$\begin{aligned} A_1 F_{1-2} &= A_2 F_{2-1} \\ \therefore F_{2-1} &= \frac{A_1}{A_2} F_{1-2} \end{aligned} \quad (\text{G.9})$$

View factor algebra was used to calculate the remaining factors. For example, Eqn. G.7 was used to calculate the view factor for Surface 4 to the combined surface formed by Surfaces 3, 5, and 7. This factor is also formed by

$$F_{4-(3,5,7)} = F_{4-3} + F_{4-5} + F_{4-7} \quad (\text{G.10})$$

Using symmetry, $F_{4-3} = F_{4-7}$, Eqn. G.10 becomes:

$$F_{4-3} = \frac{1}{2}[F_{4-(3,5,7)} - F_{4-5}] \quad (\text{G.11})$$

Table G.1 gives the view factors for the infinitely long tunnel.

Table G.1. Infinite length duct view factors.

$F_{1-2} = F_{1-3} = F_{8-6} = F_{8-7} = 0.190983$
$F_{1-4} = F_{1-5} = F_{8-4} = F_{8-5} = 0.101910$
$F_{1-6} = F_{1-7} = F_{8-2} = F_{8-3} = 0.055719$
$F_{1-8} = F_{8-1} = 0.302776$
$F_{2-1} = F_{3-1} = F_{6-8} = F_{7-8} = 0.381966$
$F_{2-3} = F_{3-2} = F_{4-5} = F_{5-4} = F_{6-7} = F_{7-6}$ $= 0.236068$
$F_{2-5} = F_{5-2} = F_{3-4} = F_{4-3} = F_{4-7} = F_{7-4}$ $= F_{5-6} = F_{6-5} = 0.178146$
$F_{2-7} = F_{7-2} = F_{6-3} = F_{3-6} = 0.092382$
$F_{2-8} = F_{3-8} = F_{6-1} = F_{7-1} = 0.111438$
$F_{4-1} = F_{5-1} = F_{4-8} = F_{5-8} = 0.203821$

Evaluating Eqn. G.5 for the floor, Surface 1 and inserting the definition of E_b :

$$J_1 = (1 - \epsilon) [F_{1-2}J_2 + F_{1-3}J_3 + F_{1-4}J_4 + F_{1-5}J_5 + F_{1-6}J_6 + F_{1-7}J_7 + F_{1-8}J_8] + \epsilon \sigma T_1^4 \quad (G.12)$$

From observation, $F_{1-2} = F_{1-3}$. Also, since the tunnel sidewalls were heated symmetrically and were therefore at the same temperature, $J_2 = J_3$. Therefore, $F_{1-2}J_2 = F_{1-3}J_3$. The same equalities hold for the other two sidewall pairs. Eqn. G.6 then becomes for the remaining surfaces:

$$J_1 = (1 - \epsilon) [2F_{1-2}J_2 + 2F_{1-4}J_4 + 2F_{1-6}J_6 + F_{1-8}J_8] + \epsilon \sigma T_1^4 \quad (G.13)$$

$$J_2 = (1 - \epsilon) [F_{2-1}J_1 + F_{2-3}J_2 + F_{2-5}J_4 + F_{2-7}J_6 + F_{2-8}J_8] + \epsilon \sigma T_2^4 \quad (G.14)$$

$$J_4 = (1 - \epsilon) [F_{4-1}J_1 + F_{4-3}J_2 + F_{4-5}J_4 + F_{4-7}J_6 + F_{4-8}J_8] + \epsilon \sigma T_4^4 \quad (G.15)$$

$$J_6 = (1 - \epsilon) [F_{6-1}J_1 + F_{6-3}J_2 + F_{6-5}J_4 + F_{6-7}J_6 + F_{6-8}J_8] + \epsilon \sigma T_6^4 \quad (G.16)$$

$$J_8 = (1 - \epsilon) [F_{8-1}J_1 + 2F_{8-2}J_2 + 2F_{8-4}J_4 + 2F_{8-6}J_6] + \epsilon \sigma T_8^4 \quad (G.17)$$

Eqns. G.13 through G.17 were written in matrix form, with the known vector elements $E_{b,i}$, the unknown vector elements J_i , and the coefficient matrix elements being the coefficients of J_i . The resulting radiosities from the solution of this matrix equation were inserted into Eqn. G.2 and q''_{rad} were calculated.

All of the Nusselt numbers tabulated and plotted in this work were calculated using the above procedure. The diffuse surface assumption is valid because the zinc coating is not very smooth and the surface was not polished. The gray assumption is reasonable from the spectral properties given by Touloukian, et al. (Vol. 9, Table 781, 1970). Experimental and

numerical work of Chuang (1982) and Yamada, et al. (1984) showed that the assumption of a transparent gas at relatively low temperatures and relatively short mean-beam lengths is acceptable, even with real absorbing gases such as CO₂.

To justify the radiosity calculation based on 8 infinitely long plates, a more in depth radiation analysis was performed on a few runs, again using the network method, but with more surfaces. Figure G.3 shows the modeled geometry. There are a total of 22 distinct surfaces in the model. Symmetry was used to group the opposing sidewall heaters together as in the infinite plate model. The last eight heater banks (48 inches) were included. Surface 1 includes the floor heaters of Banks 22 through 25. The five surfaces of Bank 26 (Surfaces 2, 10, 14, and 18) were the surfaces of interest, as the reported data of this work were calculated for Bank 26. The surfaces were grouped this way because of similar temperatures. Reviewing several data sets showed that the floor temperatures in Banks 22 through 25 were within a few degrees F of each other. The other heaters (lower side wall, mid side, upper side, and ceiling) of Banks 22 through 25 showed this also. Similarly, the heaters of Banks 27 and 28 were grouped together. This model did not extend infinitely along its axis, so the ends also needed to be included (Surface 21 is the right end, Surface 22 is the left end). These two surfaces' temperatures and emissivities were representative of the ends. The heated surfaces all had the same emissivity. Equations G.18 through G.22 give the 22 relations, with the J terms as the unknowns.

For the floor surfaces:

$$J_i = (1 - \epsilon) \left[\sum_{j=5}^{16} (2F_{i-j}J_j) + \sum_{j=17}^{22} (F_{i-j}J_j) \right] + \epsilon \sigma T_i^4 \quad \text{for } i = 1 \text{ to } 5. \quad (\text{G.18})$$

For the sidewall surfaces:

$$J_i = (1 - \epsilon) \left[\sum_{j=1}^{22} (F_{i-j}J_j) \right] + \epsilon \sigma T_i^4 \quad \text{for } i = 1 \text{ to } 22. \quad (\text{G.19})$$

Figure G.3. Finite length duct model - 22 surfaces.

For the ceiling surfaces:

$$J_i = (1 - \epsilon) \left[\sum_{j=1}^4 (F_{i-j} J_j) + \sum_{j=5}^{16} (2F_{i-j} J_j) \right] + \epsilon \sigma T_i^A \quad i = 17 \text{ to } 20. \quad (\text{G.20})$$

For the right end:

$$J_{21} = (1 - \epsilon_{21}) \left[\sum_{j=1}^4 (F_{21-j} J_j) + \sum_{j=5}^{16} (2F_{21-j} J_j) + \sum_{j=17}^{20} (F_{21-j} J_j) + F_{21-22} J_{22} \right] + \epsilon_{21} \sigma T_{21}^A \quad (\text{G.21})$$

For the left end:

$$J_{22} = (1 - \epsilon_{22}) \left[\sum_{j=1}^4 (F_{22-j} J_j) + \sum_{j=5}^{16} (2F_{22-j} J_j) + \sum_{j=17}^{21} (F_{22-j} J_j) \right] + \epsilon_{22} \sigma T_{22}^A \quad (\text{G.22})$$

Note that in Eqn. G.19, the summation term is evaluated when $i=j$. Normally this would be for a surface that could "see" itself, as in a concave surface. In this case, however, it is used to represent the surface seeing the surface directly across from it. For example, Surface 5 cannot see itself, but it can see the surface directly across the tunnel. The same method was used to eliminate three of the unknowns in the infinite length case.

To calculate the view factors needed, the following equations were used:

Two identical, parallel, directly opposed rectangles (for floor to ceiling and sidewall to sidewall), Figure G.4, (Hottell, 1931; Hamilton and Morgan, 1952):

$$F_{1-2} = \frac{2}{\pi XY} \left[\ln \left[\frac{(1+X^2)(1+Y^2)}{1+X^2+Y^2} \right]^{1/2} + X\sqrt{1+Y^2} \tan^{-1} \left[\frac{X}{\sqrt{1+Y^2}} \right] \right. \\ \left. + Y\sqrt{1+X^2} \tan^{-1} \left[\frac{Y}{\sqrt{1+X^2}} \right] - X \tan^{-1} X - Y \tan^{-1} Y \right] \quad (G.23)$$

$$\text{where } X = \frac{a}{c} \quad \text{and} \quad Y = \frac{b}{c}$$

Two parallel, not directly opposed rectangles, all sides parallel to the axes, Figure G.5, (Gross, et.al., 1981):

$$F_{1-2} = \frac{1}{A_1} \sum_{l=1}^2 \sum_{k=1}^2 \sum_{j=1}^2 \sum_{i=1}^2 [(-1)^{i+j+k+l} G(x_i, y_j, \eta_k, \xi_l)] \quad (G.24)$$

$$G(x, y, \eta, \xi) = \frac{1}{2\pi} \left[(\xi - x) [z^2 + (\eta - y)^2]^{1/2} \tan^{-1} \left[\frac{\xi - x}{[z^2 + (\eta - y)^2]^{1/2}} \right] \right. \\ \left. - (\eta - y) z \tan^{-1} \left(\frac{\eta - y}{z} \right) + [z^2 + (\xi - x)^2]^{1/2} (\eta - y) \tan^{-1} \left[\frac{\eta - y}{[z^2 + (\xi - x)^2]^{1/2}} \right] \right. \\ \left. + \frac{1}{2} (\xi - x)^2 \ln \left[\frac{z^2 + (\eta - y)^2 + (\xi - x)^2}{(\xi - x)^2} \right] + \frac{1}{2} z^2 \ln [z^2 + (\eta - y)^2] \right. \\ \left. - \frac{1}{2} [z^2 + (\xi - x)^2] \ln [z^2 + (\eta - y)^2 + (\xi - y)^2] \right] \quad (G.25)$$

Figure G.4. View factor geometry for two identical, parallel, directly opposed rectangles, Eqn. G.23.

Figure G.5. View factor geometry for two parallel, not directly opposed rectangles, Eqn. G.24.

Two perpendicular rectangles with a common side, Figure G.6, (Hottell, 1931; Hamilton and Morgan, 1952):

$$F_{1-2} = \frac{1}{\pi W} \left[W \tan^{-1} \left(\frac{1}{W} \right) + H \tan^{-1} \left(\frac{1}{H} \right) - \sqrt{H^2 + W^2} \tan^{-1} \left(\frac{1}{\sqrt{H^2 + W^2}} \right) \right. \\ \left. + \frac{1}{4} \ln \left[\frac{(1+W^2)(1+H^2)}{1+W^2+H^2} \left[\frac{W^2(1+W^2+H^2)}{(1+W^2)(W^2+H^2)} \right]^{W^2} \left[\frac{H^2(1+H^2+W^2)}{(1+H^2)(H^2+W^2)} \right]^{H^2} \right] \right] \quad (G.26)$$

where $H = \frac{h}{l}$ and $W = \frac{w}{l}$

Two perpendicular rectangles without a common side, all sides parallel to the axes, Figure G.7, (Gross, et.al., 1981):

$$F_{1-2} = \frac{1}{A_1} \sum_{l=1}^2 \sum_{k=1}^2 \sum_{j=1}^2 \sum_{i=1}^2 [(-1)^{i+j+k+l} G(x_i, y_j, \eta_k, \xi_l)] \quad (G.27)$$

$$G(x, y, \eta, \xi) = \frac{1}{2\pi} \left[(y-\eta)(x^2+\xi^2)^{1/2} \tan^{-1} \left[\frac{y-\eta}{(x^2+\xi^2)^{1/2}} \right] \right. \\ \left. - \frac{1}{4} [x^2+\xi^2-(y-\eta)^2] \ln [x^2+\xi^2+(y-\eta)^2] \right] \quad (G.28)$$

Table G.2 gives the view factors for the 22 surfaces to all other surfaces. The factors are displayed with 6 significant digits. The calculations used 15 significant digits.

Figure G.6. View factor geometry for two perpendicular rectangles with a common side, Eqn. G.26.

Figure G.7. View factor geometry for two perpendicular rectangles without a common side, Eqn. G.27.

Table G.2. Finite length duct (22 surfaces) view factors.

A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}
1	1	0.00000	2	1	0.00000	3	1	0.00000	4	1	0.00000
1	2	0.00000	2	2	0.00000	3	2	0.00000	4	2	0.00000
1	3	0.00000	2	3	0.00000	3	3	0.00000	4	3	0.00000
1	4	0.00000	2	4	0.00000	3	4	0.00000	4	4	0.00000
1	5	1.66855e-01	2	5	3.67600e-02	3	5	4.73584e-03	4	5	9.07836e-04
1	6	9.19001e-03	2	6	1.16426e-01	3	6	1.74238e-02	4	6	1.40242e-03
1	7	2.36792e-03	2	7	3.48477e-02	3	7	1.46187e-01	4	7	3.48477e-02
1	8	2.26959e-04	2	8	1.40242e-03	3	8	1.74238e-02	4	8	1.16426e-01
1	9	7.37806e-02	2	9	3.31169e-02	3	9	8.88513e-03	4	9	2.28304e-03
1	10	8.27923e-03	2	10	3.28735e-02	3	10	1.44711e-02	4	10	2.94632e-03
1	11	4.44256e-03	2	11	2.89422e-02	3	11	5.38571e-02	4	11	2.89422e-02
1	12	5.70760e-04	2	12	2.94632e-03	3	12	1.44711e-02	4	12	3.28735e-02
1	13	3.42490e-02	2	13	1.97225e-02	3	13	7.97543e-03	4	13	2.81117e-03
1	14	4.93061e-03	2	14	1.23942e-02	3	14	7.75397e-03	4	14	2.80830e-03
1	15	3.98771e-03	2	15	1.55079e-02	3	15	2.22795e-02	4	15	1.55079e-02
1	16	7.02794e-04	2	16	2.80830e-03	3	16	7.75397e-03	4	16	1.23942e-02
1	17	1.75935e-01	2	17	1.07996e-01	3	17	4.82156e-02	4	17	1.84980e-02
1	18	2.69990e-02	2	18	6.03314e-02	3	18	4.09364e-02	4	18	1.70863e-02
1	19	2.41078e-02	2	19	8.18727e-02	3	19	1.10707e-01	4	19	8.18718e-02
1	20	4.62449e-03	2	20	1.70863e-02	3	20	4.09364e-02	4	20	6.03314e-02
1	21	1.20192e-02	2	21	3.62716e-02	3	21	1.02089e-01	4	21	3.08140e-01
1	22	1.37148e-01	1	22	2.09456e-02	3	22	1.06807e-02	4	22	5.76958e-03

Table G.2. (continued) Finite length duct (22 surfaces) view factors.

A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂
5	1	3.33711e-01	6	1	7.35201e-02	7	1	9.47169e-03	8	1	1.81567e-03
5	2	1.83800e-02	6	2	2.32853e-01	7	2	3.48477e-02	8	2	2.80483e-03
5	3	4.73584e-03	6	3	6.96954e-02	7	3	2.92373e-01	8	3	6.96954e-02
5	4	4.53918e-04	6	4	2.80483e-03	7	4	3.48477e-02	8	4	2.32853e-01
5	5	1.65269e-01	6	5	7.99563e-02	7	5	2.35671e-02	8	5	6.18697e-03
5	6	1.99890e-02	6	6	6.85896e-02	7	6	3.43367e-02	8	6	7.94841e-03
5	7	1.17835e-02	6	7	6.86733e-02	7	7	1.16654e-01	8	7	6.86735e-02
5	8	1.54674e-03	6	8	7.94838e-03	7	8	3.43366e-02	8	8	6.85896e-02
5	9	1.20606e-01	6	9	6.14372e-02	7	9	1.99586e-02	8	9	5.71269e-03
5	10	1.53593e-02	6	10	4.80641e-02	7	10	2.57806e-02	8	10	6.85466e-03
5	11	9.97927e-03	6	11	5.15611e-02	7	11	8.31712e-02	8	11	5.15612e-02
5	12	1.42817e-03	6	12	6.85464e-03	7	12	2.57806e-02	8	12	4.80641e-02
5	13	5.69530e-02	6	13	3.27411e-02	7	13	1.31616e-02	8	13	4.59001e-03
5	14	8.18527e-03	6	14	2.06093e-02	7	14	1.28989e-02	8	14	4.63760e-03
5	15	6.58081e-03	6	15	2.57978e-02	7	15	3.70634e-02	8	15	2.57979e-02
5	16	1.14750e-03	6	16	4.63758e-03	7	16	1.28989e-02	8	16	2.06093e-02
5	17	6.84979e-02	6	17	3.94449e-02	7	17	1.59509e-02	8	17	5.62235e-03
5	18	9.86123e-03	6	18	2.47884e-02	7	18	1.55079e-02	8	18	5.61660e-03
5	19	7.97543e-03	6	19	3.10159e-02	7	19	4.45591e-02	8	19	3.10159e-02
5	20	1.40559e-03	6	20	5.61660e-03	7	20	1.55079e-02	8	20	2.47884e-02
5	21	8.66976e-03	6	21	2.79422e-02	7	21	8.97178e-02	8	21	3.02548e-01
5	22	1.27481e-01	6	22	1.54486e-02	7	22	7.60796e-03	8	22	4.01457e-03

Table G.2. (continued) Finite length duct (22 surfaces) view factors.

A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}
9	1	1.47561e-01	10	1	6.62339e-02	11	1	1.77703e-02	12	1	4.56608e-03
9	2	1.65585e-02	10	2	6.57470e-02	11	2	2.89422e-02	12	2	5.89264e-03
9	3	8.88513e-03	10	3	5.78843e-02	11	3	1.07714e-01	12	3	5.78843e-02
9	4	1.14152e-03	10	4	5.89264e-03	11	4	2.89422e-02	12	4	6.57470e-02
9	5	1.20606e-01	10	5	6.14372e-02	11	5	1.99586e-02	12	5	5.71269e-03
9	6	1.53593e-02	10	6	4.80641e-02	11	6	2.57806e-02	12	6	6.85466e-03
9	7	9.97927e-03	10	7	5.15611e-02	11	7	8.31712e-02	12	7	5.15612e-02
9	8	1.42817e-03	10	8	6.85464e-03	11	8	2.57806e-02	12	8	4.80641e-02
9	9	1.65269e-01	10	9	7.99563e-02	11	9	2.35671e-02	12	9	6.18697e-03
9	10	1.99890e-02	10	10	6.85896e-02	11	10	3.43367e-02	12	10	7.94841e-03
9	11	1.17835e-02	10	11	6.86733e-02	11	11	1.16654e-01	12	11	6.86735e-02
9	12	1.54674e-03	10	12	7.94838e-03	11	12	3.43366e-02	12	12	6.85896e-02
9	13	1.20606e-01	10	13	6.14372e-02	11	13	1.99586e-02	12	13	5.71269e-03
9	14	1.53593e-02	10	14	4.80641e-02	11	14	2.57806e-02	12	14	6.85466e-03
9	15	9.97927e-03	10	15	5.15611e-02	11	15	8.31712e-02	12	15	5.15612e-02
9	16	1.42817e-03	10	16	6.85464e-03	11	16	2.57806e-02	12	16	4.80641e-02
9	17	1.47561e-01	10	17	6.62339e-02	11	17	1.77703e-02	12	17	4.56608e-03
9	18	1.65585e-02	10	18	6.57470e-02	11	18	2.89422e-02	12	18	5.89264e-03
9	19	8.88513e-03	10	19	5.78843e-02	11	19	1.07714e-01	12	19	5.78843e-02
9	20	1.14152e-03	10	20	5.89264e-03	11	20	2.89422e-02	12	20	6.57470e-02
9	21	9.17496e-03	10	21	3.08868e-02	11	21	1.07006e-01	12	21	3.51894e-01
9	22	1.49198e-01	10	22	1.65959e-02	11	22	7.98082e-03	12	22	4.14228e-03

Table G.2. (continued) Finite length duct (22 surfaces) view factors.

A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}
13	1	6.84979e-02	14	1	3.94449e-02	15	1	1.59509e-02	16	1	5.62235e-03
13	2	9.86123e-03	14	2	2.47884e-02	15	2	1.55079e-02	16	2	0.00000
13	3	7.97543e-03	14	3	3.10159e-02	15	3	4.45591e-02	16	3	3.10159e-02
13	4	1.40559e-03	14	4	5.61660e-03	15	4	1.55079e-02	16	4	2.47884e-02
13	5	5.69530e-02	14	5	3.27411e-02	15	5	1.31616e-02	16	5	4.59001e-03
13	6	8.18527e-03	14	6	2.06093e-02	15	6	1.28989e-02	16	6	4.63760e-03
13	7	6.58081e-03	14	7	2.57978e-02	15	7	3.70634e-02	16	7	2.57979e-02
13	8	1.14750e-03	14	8	4.63758e-03	15	8	1.28989e-02	16	8	2.06093e-02
13	9	1.20606e-01	14	9	6.14372e-02	15	9	1.99586e-02	16	9	5.71269e-03
13	10	1.53593e-02	14	10	4.80641e-02	15	10	2.57806e-02	16	10	6.85466e-03
13	11	9.97927e-03	14	11	5.15611e-02	15	11	8.31712e-02	16	11	5.15612e-02
13	12	1.42817e-03	14	12	6.85464e-03	15	12	2.57806e-02	16	12	4.80641e-02
13	13	1.65269e-01	14	13	7.99563e-02	15	13	2.35671e-02	16	13	6.18697e-03
13	14	1.99890e-02	14	14	6.85896e-02	15	14	3.43367e-02	16	14	7.94841e-03
13	15	1.17835e-02	14	15	6.86733e-02	15	15	1.16654e-01	16	15	6.86735e-02
13	16	1.54674e-03	14	16	7.94838e-03	15	16	3.43366e-02	16	16	6.85896e-02
13	17	3.33711e-01	14	17	7.35201e-02	15	17	9.47169e-03	16	17	1.81567e-03
13	18	1.83800e-02	14	18	2.32853e-01	15	18	3.48477e-02	16	18	2.80483e-03
13	19	4.73584e-03	14	19	6.96954e-02	15	19	2.92373e-01	16	19	6.96954e-02
13	20	4.53918e-04	14	20	2.80483e-03	15	20	3.48477e-02	16	20	2.32853e-01
13	21	8.66976e-03	14	21	2.79422e-02	15	21	8.97178e-02	16	21	3.02548e-01
13	22	1.27481e-01	14	22	1.54486e-02	15	22	7.60796e-03	16	22	4.01457e-03

Table G.2. (continued) Finite length duct (22 surfaces) view factors.

A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂	A ₁	A ₂	F ₁₋₂
17	1	1.75935e-01	18	1	1.07996e-01	19	1	4.82156e-02	20	1	1.84980e-02						
17	2	2.69990e-02	18	2	6.03314e-02	19	2	4.09364e-02	20	2	1.70863e-02						
17	3	2.41078e-02	18	3	8.18727e-02	19	3	1.10707e-01	20	3	8.18727e-02						
17	4	4.62449e-03	18	4	1.70863e-02	19	4	4.09364e-02	20	4	6.03314e-02						
17	5	3.42490e-02	18	5	1.97225e-02	19	5	7.97543e-03	20	5	2.81117e-03						
17	6	4.93061e-03	18	6	1.23942e-02	19	6	7.75397e-03	20	6	2.80830e-03						
17	7	3.98771e-03	18	7	1.55079e-02	19	7	2.22795e-02	20	7	1.55079e-02						
17	8	7.02794e-04	18	8	2.80830e-03	19	8	7.75397e-03	20	8	1.23942e-02						
17	9	7.37806e-02	18	9	3.31169e-02	19	9	8.88513e-03	20	9	2.28304e-03						
17	10	8.27923e-03	18	10	3.28735e-02	19	10	1.44711e-02	20	10	2.94632e-03						
17	11	4.44256e-03	18	11	2.89422e-02	19	11	5.38571e-02	20	11	2.89422e-02						
17	12	5.70760e-04	18	12	2.94632e-03	19	12	1.44711e-02	20	12	3.28735e-02						
17	13	1.66855e-01	18	13	3.67600e-02	19	13	4.73584e-03	20	13	9.07836e-04						
17	14	9.19001e-03	18	14	1.16426e-01	19	14	1.74238e-02	20	14	1.40242e-03						
17	15	2.36792e-03	18	15	3.48477e-02	19	15	1.46187e-01	20	15	3.48477e-02						
17	16	2.26959e-04	18	16	1.40242e-03	19	16	1.74238e-02	20	16	1.16426e-01						
17	17	0.00000	18	17	0.00000	19	17	0.00000	20	17	0.00000						
17	18	0.00000	18	18	0.00000	19	18	0.00000	20	18	0.00000						
17	19	0.00000	18	19	0.00000	19	19	0.00000	20	19	0.00000						
17	20	0.00000	18	20	0.00000	19	20	0.00000	20	20	0.00000						
17	21	1.20192e-02	18	21	3.62716e-02	19	21	1.02089e-01	20	21	3.08140e-01						
17	22	1.37148e-01	18	22	2.09456e-02	19	22	1.06807e-02	20	22	5.76958e-03						

Table G.2. (continued) Finite length duct (22 surfaces) view factors.

A_1	A_2	F_{1-2}	A_1	A_2	F_{1-2}
21	1	1.60255e-02	22	1	1.82863e-01
21	2	1.20905e-02	22	2	6.98185e-03
21	3	6.80594e-02	22	3	7.12049e-03
21	4	1.02713e-01	22	4	1.92319e-03
21	5	5.77984e-03	22	5	8.49875e-02
21	6	4.65704e-03	22	6	2.57476e-03
21	7	2.99059e-02	22	7	2.53599e-03
21	8	5.04246e-02	22	8	6.69095e-04
21	9	6.11664e-03	22	9	9.94652e-02
21	10	5.14779e-03	22	10	2.76599e-03
21	11	3.56685e-02	22	11	2.66027e-03
21	12	5.86490e-02	22	12	6.90379e-04
21	13	5.77984e-03	22	13	8.49875e-02
21	14	4.65704e-03	22	14	2.57476e-03
21	15	2.99059e-02	22	15	2.53599e-03
21	16	5.04246e-02	22	16	6.69095e-04
21	17	1.60255e-02	22	17	1.82863e-01
21	18	1.20905e-02	22	18	6.98185e-03
21	19	6.80594e-02	22	19	7.12049e-03
21	20	1.02713e-01	22	20	1.92319e-03
21	21	0.00000	22	21	2.79882e-02
21	22	2.79882e-02	22	22	0.00000

Assumptions had to be made for the end "surfaces". These two surfaces were really composites. The left end, Surface 22 in Figure G.3, was composed mostly (98 percent) of the previous 13 heated banks of heaters (Banks 9 to 21). Thus, the temperature of Surface 22 was assumed to be the average of all the surfaces from Banks 9 to 21. The emissivity of Surface 22 was assumed to be 0.25, slightly above the sheet steel emissivity to account for the fact that part of the surface was the open end of the entrance. The right end, Surface 21 in Figure G.3, was composed of the plywood duct transistion (five feet long), leading to the flow meter section. The temperature of this surface was assumed to be the average of the air bulk temperature leaving Bank 29 and the ambient, as this plywood section was unheated. The emissivity of plywood, 0.9, was used. The effects of these assumptions are discussed later.

The data were reduced again for selected runs, inserting into Eqn. G.1 the q''_{rad} terms calculated from the above procedure. This was to determine the effect on q''_{net} of using a larger number of surfaces (rather than the 5 infinitely long strips used in the data reduction program). This is also directly the effect on h and Nu . Table G.3 shows the results of these selected runs. In addition to showing the effects of the larger number of surfaces (22 surfaces, including the ends), the table shows the effects of:

- assumed surface emissivity of the galvanized sheet steel
- assumed surface emissivity of the end "surfaces",
- assumed temperatures of the end "surfaces"

The first section of Table G.3 shows the results of the calculations for three different heat flux profiles: constant circumferential heat flux input (CHF), constant circumferential temperature at Bank 26, and a hot floor (skewed) flux. This last profile had the same floor flux as the other two cases. The remaining surfaces had 20 percent, 40 percent, 60 percent, and 80 percent of the floor input flux going into the ceiling, upper sidewall, middle sidewall, and lower sidewall, respectively. Two Reynolds numbers, approximately 180,000 and approximately 10,000, were analyzed for each of the three flux profiles. The last five columns give the percent difference between the infinite strip net heat fluxes and the 22 surface net heat fluxes. For all cases, the difference is less than 8 percent, and all but two differences are less than 5 percent. For one of the other two cases (Run 058, the ceiling for

Table G.3. Finite and infinite length view factor effects on q''_{net}

Test Condition	Emissivities			Temperatures		% Difference in q''_{net} (1)				
	Sheet Steel	Left End	Right End	Left End	Right End	Floor	Lo Side	Mid Side	Up Side	Ceiling
Run 023	0.23	0.25	0.9	129	81	-2.6	-3.4	-2.2	-1.0	-1.2
CHF (3), Hi Re (2)										
Run 058	0.23	0.25	0.9	298	124	1.0	0.8	0.9	-0.4	-7.8
CHF, Lo Re (2)										
Run 041	0.23	0.25	0.9	143	74	-0.8	-1.6	-2.2	-2.9	-3.6
CCT (4), Hi Re										
Run 073	0.23	0.25	0.9	262	117	-2.8	-3.2	-3.4	-4.9	-5.7
CCT, Lo Re										
Run 092	0.23	0.25	0.9	122	74	-2.9	-3.2	-2.8	-2.9	-2.2
HF#2 (5), Hi Re										
Run 091	0.23	0.25	0.9	298	124	-4.5	-4.4	-4.3	-6.0	-3.9
HF#2, Lo Re										
Run 058	0.3	0.25	0.9	300	125	-3.2	-1.6	-1.4	-1.6	2.4
CHF, Lo Re	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.15	0.25	0.9	300	125	5.5	3.4	2.9	-0.2	-9.6
	0.23	0.5	0.9	300	125	1.3	1.1	1.2	0.0	-6.9
	0.23	0.4	0.9	300	125	1.2	1.0	1.0	-1.0	-7.2
	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.23	0.25	0.8	300	125	0.5	0.3	0.3	-0.9	-8.8
	0.23	0.25	0.7	300	125	0.0	-0.2	-0.2	-1.5	-9.9
	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.23	0.25	0.9	250	125	1.5	1.3	1.4	0.2	-6.9
	0.23	0.25	0.9	200	125	1.8	1.7	1.8	0.6	-6.1
	0.23	0.25	0.9	300	130	0.9	0.7	0.7	-0.5	-8.1
	0.23	0.25	0.9	300	125	1.0	0.8	0.9	-0.4	-7.8
	0.23	0.25	0.9	300	115	1.2	1.1	1.1	-0.1	-7.3

Table footnotes:

(1): % Difference = (infinite length strip case - 22 surface case) / 22 surface case

(2): Hi Re approx. 180,000; Lo Re approx. 10,000

(3): CHF = Constant circumferential Heat Flux input

(4): CCT = Constant Circumferential Temperature @ Bank 26

(5): HF#2 = 20%, 40%, 60%, 80% of the floor's input heat flux to the ceiling, upper side, middle side, lower side, respectively

the low Re number, CHF case), the particular surface was 100°F hotter than the other surfaces. The radiosity from this surface was easily influenced by several cooler surfaces, most notably the end surfaces, that were not considered in the infinite length correction scheme. The second of the larger error cases (Run 091, the upper sidewall for the low Re number, HF#2 case) occurred because it also took into account a cooler surface (the ceiling of the next bank, heater #19 in Fig. G.3), with a large view factor to the upper sidewall heater of interest, heater #14.

The next section of Table G.3 gives the results of a parametric study for the assumed parameters. Results are given for one of the heat flux and flow cases in the first part of the table, that is, low Reynolds number constant circumferential heat flux input. This case was chosen because it had the largest spread of errors in the first section of the table. Five sets of three rows each show the effects of varying: 1) the sheet steel emissivity, 2) left end surface emissivity, 3) right end surface emissivity, 4) left end surface temperature, and 5) right end surface temperature. In examining this section of the table, it is important to consider the change in errors of the net heat fluxes, not the actual value of an error. The effect of each of these parameters will now be discussed.

The most sensitive, but perhaps the best known of these parameters, was the sheet steel emissivity. Varying this parameter over the range shown in the table caused a swing in the signed error of up to 12 percent for the ceiling. The magnitude of this error, however, varied from 2 to 10 percent. The effects of the left and right end emissivities were not very sensitive, causing a less than a 2 percent swing in the signed differences. The effects of the left and right end temperatures were also not very sensitive, causing a less than a 2 percent swing in the signed differences. Thus, the results of the first section of Table G.3 are believed to be valid.

Therefore, the data reduction to Nusselt and Grashof numbers was not repeated with the 22 surface methodology. An error of 5 percent was assigned to the net heat flux calculation.

APPENDIX H - EXPERIMENTAL RAW DATA

Appendix H - Experimental raw data

The raw data are presented on the following pages. The data runs are in numerical order, that is, Run 024 follows Run 023. The following codes are used for the run type:

CHF1	Constant Circumferential Heat Flux, #1 (0.26 watts/in.sq.)
CHF2	Constant Circumferential Heat Flux, #2 (0.53 watts/in.sq.)
CHF3	Constant Circumferential Heat Flux, #3 (1.0 watts/in.sq.)
CCT	Constant Circumferential Temperature (0.53 watts/in.sq.)
HF1	Hot Floor profile, #1 (floor at 0.53 watts/in.sq.)
HF2	Hot Floor profile, #2 (floor at 0.53 watts/in.sq.)
RP1	"Real" Profile, #1 (floor at 0.8 watts/in.sq.)
RP2	"Real" Profile, #2 (floor at 1.2 watts/in.sq.)

The wall temperatures (in degrees F) are shown for the last 17 heater banks. Heater row current (amps) and voltage (volts) are shown next, followed by the inlet air temperature (degrees F), atmospheric pressure (inches Hg), and the air flow meter differential pressure (mm Hg). The heat loss differential thermocouple voltages (microvolts) are shown next.

The following table groups the data runs according to input boundary condition, using the run type codes shown above. The table lists the runs by increasing Re number.

Table H.1. Data runs grouped by run type

CHF1	CHF2	CHF3	CCT	HF1	HF2	RP1	RP2
28	58	32	76	78	91	93	103
27	57	31	73	77	89	94	104
26	55	30	74	61	87	95	105
25	15	29	44	45	86	116	114
24	11		43	48	85	115	107
	23		42	49	84	97	108
			40	46	92	98	109
			41			99	

Run 011 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	152.1	139.0	156.1	158.3	153.8
18	153.1	139.4	155.3	158.3	154.0
19	151.5	140.3	156.9	160.6	154.9
20	155.1	141.0	162.3	162.2	155.5
21	152.9	133.4	157.5	158.7	154.4
22	144.3	113.3	152.7	162.0	156.7
23	149.7	134.0	156.1	163.1	159.5
24	149.7	141.9	157.9	164.3	159.1
25	154.7	150.1	160.6	165.3	162.4
26	155.8	154.1	161.7	169.1	162.0
27	149.0	153.3	163.3	170.3	161.3
28	139.8	148.5	162.4	170.0	161.2
29	133.1	144.9	158.9	166.3	157.2

Amps:	6.30	6.16	6.47	6.48	6.65
Volts:	120.1	123.5	123.6	123.6	120.1

T,inlet air: 68.5 F P,atm: 29.49 in Hg
 Air flow
 meter dP: 1.53 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1301	1748	1214
UB	1822	1502	1420
UF	1492	1089	1333
MB	2025	1830	1523
MF	1689	1523	1422
LB	2060	1867	1625
LF	2155	1653	1511
F	1913	1629	1449

Run 015 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	211.8	195.0	212.1	217.8	210.4
18	214.6	196.3	212.5	219.3	211.3
19	214.4	198.9	214.6	220.3	212.8
20	217.4	199.4	217.6	221.6	212.6
21	210.4	189.3	211.2	217.9	211.7
22	201.9	166.4	209.2	220.4	214.3
23	208.3	191.1	214.9	221.8	216.7
24	211.3	204.3	220.1	225.2	216.7
25	219.9	217.0	224.1	227.0	220.3
26	223.0	221.3	225.1	230.1	217.8
27	212.2	219.5	225.8	230.1	217.2
28	179.8	212.1	223.4	227.8	216.1
29	187.0	205.4	218.1	222.4	210.4

Amps:	6.50	6.38	6.36	6.39	6.48
Volts:	120.9	124.8	124.5	125.0	120.5

T,inlet air: 79.0 F P,atm: 29.25 in Hg

Air flow

meter dP: 0.37 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2136	2697	1885
UB	2792	2340	2131
UF	2467	1826	2031
MB	2911	2633	2164
MF	2713	2377	2165
LB	2923	2707	2312
LF	3099	2567	2330
F	2770	2523	2180

Run 023 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	136.2	124.7	124.7	140.8	138.5
18	136.8	124.6	136.8	142.5	138.4
19	135.6	126.8	138.9	142.4	138.9
20	138.7	127.8	139.5	143.9	139.4
21	133.8	122.6	136.6	141.5	138.3
22	129.1	103.9	135.5	144.4	140.1
23	134.7	121.6	138.0	145.2	142.7
24	134.0	127.1	137.8	145.2	142.7
25	137.7	133.8	139.2	145.7	144.1
26	138.2	136.6	140.0	149.5	143.7
27	133.6	136.2	141.9	151.0	143.3
28	109.8	131.9	141.3	150.6	143.0
29	120.2	128.7	138.1	147.2	139.8

Amps:	6.19	6.03	6.34	6.33	6.48
Volts:	122.7	125.8	125.9	125.6	122.1

T,inlet air: 74.0 F P,atm: 29.34 in Hg

Air flow

meter dP: 3.58 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1093	1416	976
UB	1408	1149	1119
UF	1173	899	1053
MB	1519	1378	1191
MF	1357	1230	1139
LB	1546	1379	1277
LF	1770	1348	1234
F	1485	1261	1177

Run 024 - CHF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	103.4	98.7	105.6	106.5	105.3
18	104.0	98.8	105.0	107.2	105.3
19	103.6	100.0	106.0	107.1	105.4
20	105.1	100.7	106.8	107.8	105.9
21	102.7	98.1	105.2	106.3	104.9
22	100.6	89.1	104.5	107.9	105.8
23	103.3	97.5	105.6	108.2	107.1
24	103.1	99.8	105.5	108.4	106.5
25	104.3	102.6	105.9	108.3	107.4
26	104.6	104.0	106.2	110.1	107.2
27	102.3	103.7	107.1	110.6	106.9
28	91.4	101.8	106.8	110.4	106.8
29	96.3	100.3	105.5	108.7	105.1

Amps:	4.43	4.37	4.55	4.53	4.61
Volts:	84.4	87.6	86.9	86.4	83.5

T,inlet air: 74.6 F P,atm: 29.40 in Hg
 Air flow
 meter dP: 3.63 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	519	674	477
UB	690	581	573
UF	578	474	537
MB	765	705	608
MF	665	625	564
LB	783	719	659
LF	822	653	587
F	743	638	600

Run 025 - CHF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	112.9	107.6	114.8	116.7	114.6
18	113.4	107.8	114.2	117.6	114.7
19	112.9	109.2	115.4	118.0	115.3
20	114.4	109.8	115.8	118.7	115.8
21	111.7	106.2	113.7	117.4	115.1
22	109.1	95.9	113.1	119.0	116.3
23	111.8	105.9	114.8	119.5	117.6
24	111.8	109.6	115.5	120.3	117.3
25	113.7	113.5	116.4	102.4	118.6
26	114.3	115.2	116.8	122.3	118.4
27	110.9	114.8	117.8	122.9	118.1
28	98.2	112.0	117.3	122.8	116.1
29	102.9	110.0	115.5	120.7	115.9

Amps:	4.43	4.37	4.54	4.52	4.61
Volts:	84.4	87.6	86.6	86.2	83.6

T,inlet air: 73.6 F P,atm: 29.40 in Hg
 Air flow
 meter dP: 1.53 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	629	831	580
UB	885	712	689
UF	730	565	648
MB	930	827	693
MF	820	755	692
LB	961	857	747
LF	1041	816	736
F	914	793	705

Run 026 - CHF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	122.1	116.6	124.6	126.8	124.3
18	122.8	116.8	123.9	127.7	124.5
19	122.3	118.3	125.2	128.3	125.3
20	123.8	118.9	125.7	129.0	125.7
21	120.1	114.5	123.8	127.6	125.3
22	118.2	103.3	122.8	129.2	126.8
23	121.6	115.1	125.4	130.6	128.4
24	122.0	119.7	126.9	131.5	128.6
25	123.8	123.9	127.3	131.3	129.4
26	124.2	125.7	127.6	132.9	129.0
27	120.3	125.0	128.6	133.5	128.5
28	106.0	121.8	128.0	133.2	128.5
29	110.4	119.4	125.8	130.9	125.8

Amps:	4.43	4.37	4.54	4.53	4.61
Volts:	84.3	87.7	86.6	86.2	83.5

T,inlet air: 75.2 F P,atm: 29.34 in Hg

Air flow

meter dP: 0.88 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	738	979	690
UB	1057	843	809
UF	873	658	768
MB	1101	963	811
MF	979	875	816
LB	1136	1005	863
LF	1218	958	870
F	1083	947	817

Run 027 - CHF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	138.0	129.1	136.5	138.8	136.6
18	139.4	129.8	136.5	139.9	137.0
19	139.4	131.5	138.1	140.7	138.3
20	140.8	132.2	140.7	141.5	138.6
21	136.1	126.5	137.0	139.7	138.4
22	133.5	114.7	135.6	141.3	140.1
23	136.5	127.0	138.3	142.1	141.2
24	137.9	133.5	141.1	144.1	141.4
25	141.5	139.7	143.1	144.7	143.2
26	142.7	142.4	143.8	146.6	142.6
27	136.9	141.2	144.6	147.0	142.1
28	120.0	137.1	143.7	146.4	141.7
29	123.1	133.6	140.9	143.4	138.3

Amps:	4.36	4.26	4.49	4.49	4.61
Volts:	84.7	87.0	87.3	87.4	85.2

T,inlet air: 73.0 F P,atm: 29.22 in Hg

Air flow

meter dP: 0.38 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	993	1270	883
UB	1347	1222	1013
UF	1136	844	939
MB	1394	1246	1021
MF	1256	1077	982
LB	1423	1285	1070
LF	1450	1156	1050
F	1333	1190	1008

Run 028 - CHF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	180.9	167.6	170.8	171.9	166.5
18	182.3	169.0	171.2	172.5	166.8
19	183.3	170.8	172.6	173.6	168.2
20	184.6	170.9	174.9	174.3	168.1
21	177.5	163.3	171.4	171.8	167.8
22	174.8	149.1	168.3	173.3	169.6
23	178.1	164.6	171.7	174.0	170.4
24	181.5	173.8	175.8	176.4	170.8
25	187.8	180.6	178.2	177.5	172.6
26	190.6	182.6	178.9	179.3	171.6
27	183.1	180.1	178.8	179.4	170.6
28	162.2	174.3	176.6	177.4	169.5
29	161.5	170.0	172.7	173.2	164.7

Amps:	4.36	4.28	4.52	4.51	4.60
Volts:	84.5	87.3	87.7	87.4	84.5

T,inlet air: 74.0 F P,atm: 29.35 in Hg

Air flow

meter dP: 0.094 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1660	2008	1428
UB	1982	1710	1547
UF	1793	1440	1491
MB	2006	1835	1532
MF	1842	1591	1486
LB	1988	1856	1561
LF	2012	1679	1535
F	1810	1695	1441

Run 029 - CHF3

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	192.4	170.4	192.8	198.3	191.9
18	193.7	171.2	191.1	199.6	192.0
19	191.6	175.4	194.5	200.5	193.0
20	197.1	176.3	196.5	203.4	194.2
21	185.9	167.6	191.9	198.0	191.2
22	178.7	130.7	190.4	203.8	195.5
23	188.3	165.4	194.8	205.0	199.2
24	187.5	176.1	195.3	205.5	197.5
25	194.6	189.2	198.4	207.1	202.3
26	196.4	193.6	200.0	213.9	201.6
27	187.3	193.1	202.8	216.1	201.0
28	142.4	184.8	201.8	215.8	200.8
29	162.4	178.9	196.6	209.8	195.0

Amps:	8.68	8.45	8.87	8.93	9.13
Volts:	168.4	172.6	172.5	173.7	168.4

T,inlet air: 74.0 F P,atm: 29.39 in Hg

Air flow

meter dP: 3.57 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1660	2008	1428
UB	1982	1710	1547
UF	1793	1440	1491
MB	2006	1835	1532
MF	1842	1591	1486
LB	1988	1856	1561
LF	2012	1679	1535
F	1810	1695	1441

Run 030 - CHF3

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	235.4	209.6	237.5	243.0	233.0
18	237.5	210.6	237.3	244.3	233.7
19	235.7	214.5	239.6	246.5	235.9
20	241.1	214.5	242.7	249.3	236.5
21	230.0	202.2	233.5	242.7	234.1
22	219.4	162.6	233.4	248.8	239.4
23	229.4	200.9	240.4	250.6	242.9
24	230.3	216.8	244.7	253.1	242.3
25	240.6	234.0	249.7	255.2	247.6
26	243.7	240.6	251.5	261.8	246.4
27	230.9	239.2	254.3	263.9	245.3
28	179.3	228.4	252.6	262.7	245.0
29	197.1	219.6	244.9	254.8	236.3

Amps:	9.12	8.89	8.95	8.92	9.15
Volts:	168.9	173.2	174.4	174.1	169.3

T,inlet air: 74.0 F P,atm: 29.02 in Hg
 Air flow
 meter dP: 1.46 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2820	3649	2571
UB	3637	3023	2858
UF	3042	2296	2780
MB	3916	3518	3003
MF	3530	3192	2895
LB	3951	3563	3136
LF	4242	3408	3029
F	3744	3229	2846

Run 031 - CHF3

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	274.4	240.6	269.9	276.7	264.7
18	277.6	241.7	270.4	278.6	265.8
19	275.6	245.2	272.7	281.0	268.5
20	280.4	244.7	276.6	279.5	268.6
21	271.3	229.9	266.2	276.8	266.4
22	254.6	187.7	265.2	282.3	271.9
23	266.2	230.7	274.1	284.5	275.1
24	268.9	251.2	281.2	287.7	274.8
25	281.6	271.4	287.1	289.9	280.3
26	285.9	278.9	288.6	295.4	278.4
27	270.0	276.1	290.9	296.6	276.5
28	211.4	263.3	288.7	294.4	275.5
29	228.5	252.1	280.2	285.7	265.4

Amps:	9.10	8.83	8.88	8.88	9.11
Volts:	168.8	172.3	173.3	173.6	168.8

T,inlet air: 76.4 F P,atm: 29.21 in Hg

Air flow

meter dP: 0.885 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	3409	4334	3090
UB	4278	3585	3360
UF	3674	3737	3290
MB	4541	4091	3476
MF	4223	3250	3415
LB	4555	4164	3627
LF	4875	4022	3616
F	4332	3829	3331

Run 032 - CHF3

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	349.7	303.5	320.0	326.0	309.9
18	355.8	305.8	321.4	327.1	311.0
19	356.0	310.6	322.6	330.2	313.9
20	359.6	309.6	328.3	332.1	313.2
21	341.3	291.9	316.8	325.2	311.7
22	324.4	242.3	310.9	329.6	317.0
23	338.2	293.4	320.5	332.2	319.0
24	345.2	321.4	330.5	337.4	319.3
25	363.9	346.6	337.6	340.7	324.8
26	372.8	354.4	338.0	346.1	322.2
27	351.8	347.1	338.1	346.1	319.6
28	285.5	328.2	333.1	341.4	317.9
29	296.9	313.1	322.6	330.4	306.1

Amps:	9.07	8.92	8.83	8.87	9.24
Volts:	168.7	174.7	172.7	173.7	168.1

T,inlet air: 76.0 F P,atm: 29.36 in Hg
 Air flow
 meter dP: 0.37 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	4776	5792	4134
UB	5486	4631	4329
UF	4910	3875	4394
MB	5636	5114	4324
MF	5126	4558	4366
LB	5646	5258	4517
LF	5797	4894	4499
F	5185	4717	4155

Run 040 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	163.2	155.5	161.2	159.4	162.4
18	165.4	159.4	161.2	161.3	163.0
19	163.3	161.5	163.4	161.8	163.9
20	163.9	158.7	161.8	163.2	164.3
21	159.2	156.6	160.9	161.0	163.3
22	154.0	156.6	165.4	164.1	165.8
23	159.8	157.1	167.0	165.0	168.0
24	162.9	162.1	167.2	166.5	167.5
25	168.0	170.8	168.6	168.9	170.2
26	167.9	169.2	168.4	169.4	169.2
27	167.8	164.6	169.6	170.1	168.3
28	165.9	160.0	168.7	169.1	168.6
29	158.4	155.1	164.3	164.3	163.7

Amps:	6.74	6.73	6.34	5.92	6.55
Volts:	124.0	130.4	122.8	114.5	120.3

T,inlet air: 76.5 F P,atm: 29.22 in Hg
 Air flow
 meter dP: 1.53 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1828	1781	
UB	2157	1751	
UF	1469	1453	
MB	1943	1728	
MF	1671	1619	
LB	1855	1623	
LF	1920	1592	
F	1934	1633	

Run 041 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	141.5	133.0	137.8	135.1	138.6
18	143.3	136.8	137.5	136.8	138.9
19	143.3	136.8	137.5	136.8	138.9
20	143.1	136.5	137.3	137.8	139.3
21	140.5	134.9	136.5	135.5	137.4
22	135.0	135.2	141.5	138.6	140.2
23	140.2	135.1	142.6	139.4	142.6
24	141.5	137.2	142.0	140.1	141.5
25	145.1	145.6	143.1	140.4	144.0
26	143.9	143.6	142.6	143.0	143.0
27	144.6	140.5	144.3	144.0	142.9
28	142.0	137.1	143.4	143.0	142.5
29	136.3	133.4	140.2	139.1	139.2

Amps:	6.84	6.74	6.43	5.83	6.58
Volts:	125.9	130.4	124.4	112.7	120.9

T,inlet air: 75.3 F P,atm: 29.30 in Hg
 Air flow
 meter dP: 3.6 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1472	1505	
UB	1674	1355	
UF	1174	1166	
MB	1512	1354	
MF	1297	1290	
LB	1402	1208	
LF	1469	1203	
F	1474	1229	

Run 042 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	172.2	166.2	172.2	171.0	174.1
18	174.9	170.1	172.7	172.5	174.8
19	172.6	172.2	174.3	172.9	175.8
20	171.7	169.5	177.1	174.2	175.9
21	165.9	167.5	176.0	172.1	175.3
22	162.3	167.4	176.5	174.8	178.0
23	169.7	168.8	179.0	176.3	180.3
24	174.5	175.3	180.6	178.3	180.3
25	180.4	183.2	182.0	178.5	182.6
26	180.9	181.8	181.3	181.0	181.8
27	180.7	176.9	182.1	181.5	181.1
28	178.8	171.6	181.0	180.1	181.1
29	170.5	166.6	176.7	175.7	176.2

Amps:	6.52	6.56	6.24	5.91	6.54
Volts:	120.2	127.4	121.2	114.6	120.6

T,inlet air: 73.7 F P,atm: 29.30 in Hg

Air flow

meter dP: 0.88 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2089	1945	1515
UB	2472	2025	1811
UF	1696	1662	1615
MB	2298	2061	1670
MF	1930	1826	1584
LB	2184	1931	1633
LF	2174	1795	1595
F	2241	1914	1657

Run 043 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	193.7	188.4	198.6	199.2	201.6
18	197.2	192.7	199.5	200.6	202.4
19	195.3	195.6	201.3	201.7	203.9
20	193.9	191.9	206.7	202.8	203.8
21	184.7	189.6	204.6	200.4	203.4
22	182.5	189.7	202.9	203.1	206.5
23	191.8	191.8	205.5	204.3	208.4
24	199.7	201.7	208.3	207.1	208.5
25	207.7	210.2	210.3	208.0	211.6
26	210.2	209.2	209.4	210.1	210.1
27	208.8	202.7	209.9	209.6	208.7
28	206.6	195.2	206.8	207.5	208.1
29	196.2	188.9	201.3	201.5	201.8

Amps:	6.11	6.03	6.32	5.94	6.54
Volts:	112.9	117.3	122.9	116.0	120.7

T,inlet air: 73.8 F P,atm: 29.27 in Hg

Air flow

meter dP: 0.37 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2554	2245	1764
UB	2918	2405	2089
UF	2093	2003	1871
MB	2855	2599	2084
MF	2399	2251	1961
LB	2750	2503	2082
LF	2674	2294	2040
F	2764	2434	2081

Run 044 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	229.9	232.5	238.2	240.0	241.2
18	233.3	236.0	238.8	240.5	241.6
19	231.8	236.7	240.5	241.7	243.3
20	229.2	233.9	243.0	242.2	242.4
21	215.1	231.1	240.3	238.5	242.1
22	212.9	231.4	240.8	241.8	245.9
23	223.7	236.7	244.1	242.7	247.0
24	234.8	246.6	247.6	245.6	247.0
25	244.9	251.1	249.7	246.9	250.6
26	248.5	250.2	248.9	249.0	248.9
27	244.0	244.2	247.9	248.3	247.3
28	236.7	236.7	244.4	244.8	245.7
29	220.3	230.5	237.0	236.3	237.0

Amps:	5.24	5.87	6.07	5.88	6.52
Volts:	97.0	114.6	118.2	114.6	120.7

T,inlet air: 74.0 F P,atm: 29.28 in Hg

Air flow

meter dP: 0.088 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2907	2683	2197
UB	3547	3065	2820
UF	3026	2812	2612
MB	3516	3253	2779
MF	3175	2930	2678
LB	3426	3231	2788
LF	3391	2987	2793
F	3503	3160	2814

Run 045 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	118.9	157.8	183.6	208.9	234.8
18	123.2	159.8	184.5	210.1	234.9
19	124.3	161.5	186.4	210.9	236.1
20	124.4	162.2	187.0	211.6	235.0
21	119.7	159.8	186.8	208.6	234.3
22	119.1	160.3	187.1	210.7	239.1
23	122.4	161.7	188.9	211.7	240.1
24	127.6	165.8	191.1	214.8	239.7
25	131.2	167.3	192.0	215.9	243.3
26	132.0	168.6	191.9	217.3	241.9
27	130.3	165.8	191.1	215.5	240.6
28	129.5	162.3	189.1	212.9	239.2
29	126.4	158.5	184.6	207.0	230.5

Amps:	2.12	3.23	4.58	5.64	6.59
Volts:	37.4	61.1	87.4	108.2	120.4

T,inlet air: 74.3 F P,atm: 29.33 in Hg

Air flow

meter dP: 0.088 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	931	749	416
UB	1798	1587	1254
UF	1557	1416	1156
MB	2279	2082	1660
MF	2064	1875	1599
LB	2727	2573	2159
LF	2741	2354	2185
F	3399	3028	2736

Run 046 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	83.4	93.4	108.4	123.2	137.0
18	84.0	94.1	108.1	125.0	137.3
19	83.8	94.9	109.2	124.5	137.2
20	83.9	94.8	108.2	125.6	137.8
21	83.7	94.0	108.7	123.5	135.9
22	83.2	94.5	110.2	125.7	138.2
23	83.8	93.9	110.8	126.4	141.1
24	84.2	94.0	110.2	127.2	139.7
25	84.3	95.5	110.2	127.5	142.3
26	84.2	95.6	110.1	129.5	141.8
27	83.8	94.7	110.5	129.4	141.2
28	83.8	94.2	110.5	129.5	141.1
29	83.5	92.9	108.8	126.8	137.8

Amps:	2.12	3.24	4.59	5.66	6.61
Volts:	37.3	61.1	87.3	108.3	120.2

T,inlet air: 75.2 F P,atm: 29.33 in Hg

Air flow

meter dP: 3.65 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	144	130	64
UB	526	412	355
UF	345	334	302
MB	809	697	559
MF	647	635	543
LB	1122	969	863
LF	1167	911	847
F	1448	1202	1134

Run 047 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	92.3	117.8	143.3	167.5	192.9
18	94.1	119.2	144.0	169.4	193.6
19	93.9	122.2	145.8	170.2	194.8
20	93.6	121.7	148.0	171.3	194.6
21	92.0	120.2	148.4	169.5	194.0
22	91.4	120.8	147.9	171.3	198.0
23	93.1	120.5	149.4	172.5	199.6
24	95.7	123.1	150.8	175.3	199.4
25	97.2	125.5	151.6	178.0	201.9
26	97.2	126.6	151.6	178.0	201.9
27	96.4	124.6	151.7	176.7	200.9
28	96.8	122.0	150.9	176.0	200.7
29	96.1	119.1	147.8	172.0	194.4

Amps:	2.13	3.20	4.58	5.56	6.60
Volts:	38.4	61.3	88.3	107.5	121.2

T,inlet air: 73.3 F P,atm: 29.51 in Hg

Air flow

meter dP: 0.37 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	440	335	204
UB	1186	969	770
UF	847	783	675
MB	1623	1416	1097
MF	1358	1232	1028
LB	2054	1842	1513
LF	2025	1663	1488
F	2655	2287	1996

Run 048 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	86.6	104.1	125.1	145.7	168.3
18	87.6	105.0	125.3	147.7	168.9
19	87.4	107.2	126.7	148.1	169.7
20	87.4	106.7	127.1	149.3	170.0
21	86.7	105.5	127.6	147.5	168.8
22	86.1	106.2	128.7	149.4	172.1
23	87.1	105.7	129.7	150.5	174.2
24	88.5	107.1	130.3	152.6	173.7
25	89.1	108.9	130.7	153.4	176.7
26	89.1	109.3	130.6	155.4	176.2
27	88.5	107.6	130.9	154.7	175.6
28	88.7	106.1	130.6	154.5	175.7
29	88.0	104.0	128.1	151.0	170.8

Amps:	2.12	3.21	4.59	5.57	6.61
Volts:	39.1	61.5	88.4	107.6	121.4

T,inlet air: 72.2 F P,atm: 29.55 in Hg
 Air flow
 meter dP: 0.89 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	279	224	130
UB	859	693	564
UF	562	542	481
MB	1224	1058	838
MF	1011	946	787
LB	1621	1420	1201
LF	1640	1320	1189
F	2141	1801	1602

Run 049 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	87.0	101.3	120.2	138.1	156.4
18	88.1	102.0	120.1	139.7	157.0
19	87.6	103.9	121.1	139.9	157.5
20	87.6	103.3	121.4	141.1	158.2
21	87.1	102.3	122.1	139.2	156.6
22	86.7	103.0	123.2	141.3	159.8
23	87.5	102.4	123.9	142.3	162.1
24	88.7	103.5	124.3	143.8	161.4
25	88.7	105.0	124.1	144.0	164.4
26	88.9	105.3	124.3	146.4	163.7
27	88.1	103.6	124.6	145.6	163.0
28	88.4	102.6	124.4	145.8	163.2
29	87.7	100.7	122.3	142.7	158.7

Amps:	2.14	3.25	4.60	5.63	6.62
Volts:	37.8	61.2	87.5	107.6	120.7

T,inlet air: 75.0 F P,atm: 29.69 in Hg

Air flow

meter dP: 1.53 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	212	180	96
UB	712	563	476
UF	460	445	402
MB	1052	903	725
MF	852	804	674
LB	1416	1225	1062
LF	1431	1134	1027
F	1851	1525	1379

Run 055 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	302.5	265.2	264.4	264.8	251.0
18	304.3	269.7	265.1	264.8	251.5
19	302.5	267.1	266.4	265.9	253.4
20	299.9	264.2	269.6	266.6	252.8
21	286.8	260.9	266.7	262.4	252.7
22	285.6	262.8	267.0	266.4	255.9
23	302.0	271.0	270.7	267.0	256.7
24	313.5	282.5	274.8	269.8	256.9
25	323.9	287.5	277.0	270.9	260.5
26	329.3	285.5	276.0	273.4	258.6
27	322.6	276.6	274.4	273.0	256.6
28	311.2	267.3	270.1	268.3	254.7
29	285.6	260.5	261.3	259.2	245.8

Amps:	6.53	6.43	6.45	6.43	6.57
Volts:	119.9	123.8	24.3	123.8	120.1

T,inlet air: 75.6 F P,atm: 29.44 in Hg

Air flow

meter dP: 0.0825 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	4351	4212	3644
UB	4273	3723	3428
UF	3572	3333	3254
MB	4065	3785	3280
MF	3653	3385	3168
LB	3928	3714	3278
LF	3873	3437	3234
F	3650	3342	2975

Run 057 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	345.6	283.8	278.0	273.9	262.2
18	342.7	287.7	278.3	273.4	262.6
19	345.6	286.7	279.6	274.2	264.6
20	349.3	285.6	283.1	274.9	264.2
21	341.4	284.1	280.0	270.4	264.7
22	343.9	286.5	280.8	274.5	267.7
23	361.2	295.7	284.8	274.9	268.4
24	369.9	306.3	288.6	277.7	268.3
25	379.0	311.6	290.0	279.0	271.9
26	386.1	309.6	290.1	281.5	270.1
27	381.4	304.5	289.3	281.4	268.2
28	366.8	294.9	285.3	276.4	266.0
29	331.7	288.1	276.5	267.3	256.5

Amps:	6.54	6.39	6.39	6.38	6.56
Volts:	121.2	124.3	124.3	124.0	121.0

T,inlet air:	76.0 F	P,atm:	29.42 in Hg
Air flow			
meter dP:	0.04 mm Hg		

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	5333	5379	4644
UB	4693	4092	3809
UF	4191	3778	3600
MB	4319	4039	3609
MF	3966	3627	3480
LB	4073	3867	3545
LF	4030	3566	3510
F	3840	3554	3232

Run 058 - CHF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	390.9	302.9	289.0	281.7	269.7
18	391.5	307.3	289.6	281.4	270.4
19	395.8	307.2	291.2	282.5	272.5
20	401.7	307.2	294.8	283.4	272.5
21	395.6	307.9	292.8	279.3	274.0
22	399.5	311.1	294.0	283.5	276.4
23	416.7	321.1	298.1	284.1	270.0
24	420.6	331.2	302.0	286.8	276.9
25	425.5	335.7	304.4	288.0	280.4
26	431.4	333.3	303.7	209.7	278.7
27	426.9	330.8	303.8	291.2	276.9
28	412.3	232.6	300.3	286.6	274.8
29	370.5	317.0	292.1	278.0	265.6

Amps:	6.54	6.42	6.42	6.42	6.54
Volts:	121.5	125.1	125.1	124.8	120.6

T,inlet air: 79.0 F P,atm: 29.34 in Hg

Air flow

meter dP: 0.025 mm Hg

Heat Loss Differential Temperatures (mV*1000)			
	Bank 29	Bank 21	Bank 11
C	6177	6521	5213
UB	5100	4569	4030
UF	4942	4196	3855
MB	4580	4294	3790
MF	4313	3865	3670
LB	4261	4033	3706
LF	4244	3728	3668
F	3987	3711	3362

Run 061 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	136.8	173.9	198.2	220.7	246.4
18	139.8	174.4	198.2	221.1	246.2
19	140.1	174.7	199.3	221.1	247.0
20	140.3	176.1	199.5	221.4	245.5
21	135.4	174.2	199.0	217.5	245.0
22	136.3	175.2	199.4	219.7	249.9
23	141.1	176.2	201.1	220.6	250.5
24	146.2	179.5	203.1	223.4	249.9
25	148.8	180.3	203.6	224.1	253.2
26	149.2	181.9	203.3	225.4	251.7
27	146.3	179.4	202.3	223.2	250.2
28	144.6	176.2	200.2	220.0	248.6
29	141.0	172.9	195.2	214.0	238.8

Amps:	2.14	3.18	4.56	5.60	6.57
Volts:	37.7	60.1	87.0	107.4	120.0

T,inlet air: 76.6 F P,atm: 29.35 in Hg

Air flow

meter dP: 0.04 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1008	1003	725
UB	2022	1795	1541
UF	1783	1617	1439
MB	2459	2264	1962
MF	2232	2039	1885
LB	2852	2697	2434
LF	2839	2462	2445
F	3534	3161	2964

Run 073 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	242.7	248.9	256.3	261.6	262.2
18	244.7	251.3	256.6	261.2	262.2
19	246.1	252.9	258.3	261.8	263.4
20	248.3	254.6	260.9	262.5	262.3
21	243.9	255.3	260.1	258.9	262.2
22	247.6	258.2	261.7	263.1	266.0
23	257.4	262.6	265.2	264.0	266.4
24	262.0	267.8	268.6	267.0	266.3
25	264.3	270.4	270.7	268.6	270.1
26	265.4	272.2	271.0	271.7	269.1
27	261.4	272.3	271.7	272.1	268.1
28	255.2	267.9	269.4	268.5	266.9
29	236.5	261.2	260.7	259.1	256.3

Amps:	3.62	5.00	5.89	6.31	6.58
Volts:	66.5	97.1	114.4	122.5	121.3

T,inlet air: 74.2 F P,atm: 29.28 in Hg

Air flow

meter dP: 0.014 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	3159	3188	2343
UB	3810	3469	2979
UF	3797	3181	2852
MB	3931	3625	3222
MF	3719	3260	3123
LB	3922	3642	3417
LF	3944	3342	3408
F	3903	3540	3344

Run 074 - CCT

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	223.9	237.0	248.8	251.5	252.2
18	226.7	239.1	248.7	251.2	252.3
19	227.4	238.8	250.1	251.7	253.3
20	228.7	240.2	252.3	252.2	252.3
21	222.6	239.2	249.8	248.2	252.0
22	225.9	242.0	251.9	255.9	255.9
23	237.6	246.2	254.8	252.4	256.3
24	245.2	252.5	257.8	255.0	256.1
25	250.9	255.7	259.6	256.1	259.6
26	254.3	256.9	259.1	258.3	257.8
27	251.1	255.5	258.7	257.6	256.0
28	244.2	248.3	255.3	253.4	254.2
29	225.0	240.4	246.1	243.7	243.4

Amps:	4.42	5.32	6.06	6.03	6.51
Volts:	81.2	103.3	117.7	116.9	120.0

T,inlet air: 74.3 F P,atm: 29.33 in Hg

Air flow

meter dP: 0.0395 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2978	2813	2371
UB	3603	3183	2873
UF	3337	2947	2786
MB	3678	3429	3026
MF	3428	3115	2972
LB	3612	3421	3123
LF	3629	3150	3139
F	3660	3348	3120

Run 077 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	156.6	182.5	201.1	220.3	251.0
18	159.9	183.4	201.2	220.6	250.5
19	151.5	185.0	202.6	220.7	251.0
20	161.7	186.7	203.6	221.2	249.3
21	163.6	187.2	203.8	218.2	249.6
22	161.6	189.2	205.2	221.1	254.6
23	163.0	191.2	207.8	222.9	255.4
24	167.9	194.3	210.6	226.5	255.5
25	171.3	195.1	212.2	228.4	259.7
26	173.2	197.3	213.0	230.7	259.4
27	174.4	198.2	213.5	230.1	259.1
28	173.0	196.6	212.4	228.0	258.3
29	163.8	192.2	206.3	220.7	246.8

Amps:	2.08	3.18	4.53	5.54	6.59
Volts:	37.7	61.2	87.5	107.3	121.4

T,inlet air: 73.5 F P,atm: 29.34 in Hg

Air flow

meter dP: 0.01 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1667	1522	1046
UB	2329	2082	1768
UF	2323	1900	1706
MB	2709	2424	2178
MF	2561	2195	2104
LB	3048	2769	2594
LF	3075	2533	2614
F	3760	3309	3196

Run 078 - HF1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	186.2	199.5	212.2	227.5	254.7
18	187.2	200.4	212.5	227.9	254.4
19	187.4	201.7	213.9	22.1	255.3
20	188.3	203.1	215.2	228.3	254.1
21	184.6	203.4	215.6	229.0	255.3
22	185.8	205.2	216.8	226.5	260.3
23	190.6	207.3	219.6	229.5	261.4
24	193.0	210.2	223.9	231.5	265.9
25	193.9	210.9	223.9	237.2	265.9
26	194.2	212.6	224.5	239.4	265.7
27	191.9	213.0	224.9	238.9	265.5
28	189.2	210.6	223.4	236.4	264.6
29	180.1	206.0	217.3	229.4	253.7

Amps:	2.07	3.16	4.54	5.54	6.54
Volts:	37.6	60.9	87.6	107.4	120.5

T,inlet air: 75.1 F P,atm: 29.46 in Hg

Air flow

meter dP: 0.006 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1935	1936	1658
UB	2547	2352	2040
UF	2555	2159	1933
MB	2885	2637	2329
MF	2742	2303	2239
LB	3197	2922	2689
LF	3216	2669	2696
F	3853	3402	3229

Run 084 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	91.4	105.8	124.0	141.2	157.7
18	92.8	106.8	123.8	142.6	158.0
19	92.6	109.0	125.0	142.7	158.5
20	92.8	108.2	126.2	143.7	159.0
21	92.1	107.4	126.0	141.8	157.5
22	91.5	108.4	127.1	144.1	160.4
23	92.7	108.7	128.1	144.9	162.3
24	93.2	110.3	128.4	146.3	161.5
25	94.1	112.3	128.9	146.8	164.6
26	94.4	113.2	128.9	149.2	163.8
27	93.6	111.8	129.4	148.8	163.1
28	93.7	110.2	129.2	148.9	163.2
29	91.9	106.9	126.4	145.0	158.2

Amps:	2.98	4.05	4.95	5.74	6.58
Volts:	55.4	78.8	96.3	111.5	121.7

T,inlet air: 73.1 F P,atm: 29.37 in Hg
 Air flow
 meter dP: 1.53 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	389	365	227
UB	913	744	658
UF	681	612	569
MB	1235	1063	867
MF	1024	943	807
LB	1572	1380	1197
LF	1556	1227	1112
F	1908	1549	1449

Run 085 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	104.0	122.3	144.2	163.4	180.7
18	105.7	123.7	144.1	164.6	181.1
19	104.8	126.6	145.1	164.5	181.5
20	105.3	124.8	147.4	165.5	181.5
21	104.5	124.5	147.7	164.1	181.1
22	103.9	125.4	147.8	166.2	184.3
23	105.7	126.1	149.3	167.4	186.0
24	106.5	128.4	150.0	169.1	185.5
25	107.8	130.6	150.8	169.7	188.5
26	108.3	131.8	150.6	171.8	187.7
27	107.4	130.6	151.2	171.2	186.9
28	107.4	128.5	150.8	170.9	187.0
29	104.9	124.2	147.1	166.3	181.2

Amps:	3.00	4.04	4.97	5.77	6.60
Volts:	55.5	78.5	96.7	112.2	121.9

T,inlet air: 80.7 F P,atm: 29.22 in Hg

Air flow

meter dP: 0.875 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	505	461	316
UB	1169	960	819
UF	875	769	714
MB	1522	1320	1083
MF	1272	1149	986
LB	1871	1658	1440
LF	1840	1468	1357
F	2294	1938	1725

Run 086 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	109.7	135.9	159.7	180.0	198.9
18	112.4	138.3	160.5	181.7	199.4
19	112.3	142.2	162.2	182.6	200.5
20	112.2	140.4	166.0	183.7	200.5
21	109.7	139.6	165.4	182.0	199.6
22	109.5	141.1	165.3	184.2	203.4
23	112.3	142.3	166.9	185.3	204.7
24	114.6	145.9	168.7	188.0	204.6
25	117.3	148.9	169.7	189.0	208.0
26	118.7	151.2	170.0	191.0	206.8
27	117.4	149.8	170.1	189.8	205.7
28	117.2	146.9	169.3	189.0	205.4
29	113.2	140.9	164.7	183.6	198.3

Amps:	3.00	4.04	4.98	5.71	6.59
Volts:	55.6	78.5	96.8	111.2	121.8

T,inlet air: 76.2 F P,atm: 29.40 in Hg

Air flow

meter dP: 0.37 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	747	628	408
UB	1567	1268	1034
UF	1242	1063	926
MB	1925	1674	1310
MF	1647	1467	1232
LB	2264	2037	1677
LF	2241	1837	1640
F	2689	2325	2034

Run 087 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	141.3	174.8	195.5	214.8	234.0
18	145.7	176.9	196.1	216.0	234.0
19	146.4	177.8	197.6	216.8	235.1
20	146.3	178.5	199.6	217.5	234.1
21	140.4	177.4	198.4	214.8	233.6
22	140.8	178.7	199.4	217.1	237.8
23	146.3	180.3	201.3	218.0	238.3
24	151.0	184.2	203.5	220.8	237.9
25	155.7	186.1	204.9	222.2	241.7
26	157.6	187.7	204.8	223.8	240.1
27	155.7	186.2	204.3	222.3	238.6
28	153.3	181.3	201.8	219.4	237.0
29	145.8	175.4	195.1	211.5	226.8

Amps:	3.00	4.02	4.97	5.72	6.53
Volts:	55.8	78.5	97.0	111.6	121.3

T,inlet air: 73.0 F P,atm: 29.30 in Hg
 Air flow
 meter dP: 0.086 mm Hg

Heat Loss Differential Temperatures (mV*1000)			
	Bank 29	Bank 21	Bank 11
C	1409	1253	785
UB	2284	1989	1609
UF	1963	1789	1550
MB	2612	2386	1946
MF	2359	2148	1877
LB	2922	2740	2354
LF	2929	2515	2337
F	3387	3046	2769

Run 089 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	161.3	191.0	210.0	226.7	244.9
18	164.6	191.9	210.0	226.9	244.7
19	165.4	192.0	211.0	226.9	245.6
20	166.1	193.5	212.6	227.3	244.5
21	160.7	192.2	211.1	223.7	244.3
22	161.8	193.6	211.9	226.2	248.4
23	167.9	195.2	213.8	226.8	248.7
24	172.7	199.3	216.0	229.5	248.1
25	176.3	200.6	217.0	230.7	251.6
26	177.8	202.0	216.7	232.1	250.1
27	175.3	200.8	216.1	230.5	248.6
28	172.5	196.3	213.6	227.2	246.8
29	163.7	190.7	206.5	218.8	235.6

Amps:	3.06	4.05	5.00	5.73	6.54
Volts:	56.8	79.0	97.8	111.8	121.3

T,inlet air: 72.7 F P,atm: 29.50 in Hg

Air flow

meter dP: 0.04 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1690	1550	1220
UB	2500	2230	1950
UF	2280	2050	1880
MB	2800	2610	2280
MF	2580	2350	2210
LB	3060	2900	2640
LF	3070	2670	2650
F	3550	3210	3020

Run 091 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	191.8	205.7	218.8	232.4	253.4
18	194.7	207.1	219.0	232.6	253.0
19	196.4	207.9	220.4	232.6	253.8
20	198.7	210.3	222.1	233.1	252.5
21	196.1	210.7	221.8	229.8	252.8
22	198.4	212.9	222.9	232.9	256.8
23	205.2	215.7	225.7	234.1	257.1
24	208.9	219.7	228.5	237.3	256.9
25	211.1	221.4	230.1	238.9	260.8
26	212.6	223.2	230.6	241.3	260.2
27	210.5	223.3	231.0	240.9	259.5
28	207.5	220.8	229.5	238.4	258.6
29	196.4	215.9	222.7	230.5	247.4

Amps:	2.99	4.02	4.99	5.71	6.54
Volts:	55.7	78.5	97.3	111.5	121.3

T,inlet air: 74.5 F P,atm: 29.42 in Hg
 Air flow
 meter dP: 0.014 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	2300	2220	1590
UB	2840	2550	2190
UF	2820	2340	2110
MB	3070	2800	2490
MF	2900	2530	2430
LB	3260	3010	2810
LF	3290	2770	2840
F	3750	3350	3200

Run 092 - HF2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	87.8	97.9	112.1	124.3	135.4
18	88.7	98.8	111.8	125.7	135.7
19	88.6	102.0	112.6	125.3	135.8
20	88.8	99.8	112.9	126.3	136.3
21	88.4	98.9	112.4	124.2	134.6
22	87.7	99.7	114.0	126.5	136.8
23	88.9	100.1	114.7	127.0	139.1
24	88.9	101.0	114.2	127.5	137.6
25	89.4	102.7	114.4	127.9	140.5
26	89.4	103.1	114.2	130.3	139.9
27	89.0	102.0	115.0	130.4	139.2
28	89.0	100.8	114.8	130.4	139.2
29	87.1	96.6	111.2	125.7	134.2

Amps:	3.14	4.13	5.11	5.78	6.62
Volts:	57.0	79.2	98.3	111.4	121.5

T,inlet air: 72.7 F P,atm: 29.51 in Hg

Air flow

meter dP: 3.7 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	280	290	180
UB	710	570	500
UF	500	460	450
MB	920	770	670
MF	750	730	640
LB	1150	970	900
LF	1180	940	870
F	1330	1100	1010

Run 093 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	133.1	139.0	149.9	173.8	297.9
18	135.1	140.4	151.4	176.9	295.8
19	136.3	140.1	152.8	176.6	296.4
20	138.7	143.8	152.4	176.9	294.9
21	137.3	144.1	153.6	174.0	297.5
22	137.4	145.4	155.2	174.8	305.6
23	139.2	146.6	157.7	179.3	306.3
24	141.3	149.4	161.3	184.8	305.6
25	141.9	150.1	162.5	187.7	310.4
26	143.4	151.9	163.8	187.6	311.4
27	143.1	152.0	163.4	184.0	312.4
28	143.8	151.5	163.0	183.0	311.8
29	140.6	149.6	160.6	180.3	296.8

Amps:	1.44	1.73	2.33	3.18	8.04
Volts:	26.4	33.4	45.0	61.6	148.8

T,inlet air: 75.8 F P,atm: 29.23 in Hg

Air flow

meter dP: 0.006 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1130	1020	680
UB	1310	1150	890
UF	1300	1060	860
MB	1540	1320	1080
MF	1460	1210	1100
LB	1910	1700	1460
LF	1980	1550	1640
F	4970	4270	4250

Run 094 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	109.5	124.6	140.3	167.7	291.1
18	112.2	125.9	141.7	171.0	289.3
19	113.5	125.7	142.9	170.3	289.9
20	114.8	128.9	141.5	170.1	288.0
21	112.9	127.9	140.9	165.3	288.2
22	112.6	128.2	141.9	164.3	295.3
23	114.0	128.5	142.9	167.3	295.0
24	115.8	130.2	144.8	171.2	292.8
25	116.7	130.3	144.9	172.8	296.4
26	117.7	131.2	145.0	171.5	295.9
27	117.0	130.0	143.4	166.5	295.3
28	117.3	128.0	141.8	164.4	294.0
29	114.9	124.6	138.0	160.3	279.1

Amps:	1.42	1.72	2.34	3.16	8.08
Volts:	26.1	33.3	45.2	61.7	149.7

T,inlet air: 75.1 F P,atm: 29.20 in Hg

Air flow

meter dP: 0.039 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	680	580	300
UB	970	870	600
UF	850	820	600
MB	1180	1100	820
MF	1090	1020	840
LB	1560	1520	1230
LF	1610	1400	1400
F	4600	4080	3970

Run 095 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	95.1	110.1	126.7	155.6	278.9
18	97.6	111.7	128.6	160.0	277.2
19	98.8	113.4	130.5	160.3	278.3
20	100.0	115.4	130.2	160.4	276.6
21	99.2	115.6	130.8	156.6	276.8
22	99.3	116.2	131.6	155.6	284.1
23	101.1	116.9	132.9	158.9	223.8
24	103.4	119.0	134.9	163.2	282.0
25	105.3	119.9	135.4	165.4	286.0
26	106.8	121.1	135.8	164.5	285.3
27	106.5	119.9	134.6	159.3	284.6
28	107.1	118.1	133.3	157.3	283.5
29	105.4	115.1	130.2	153.8	269.5

Amps:	1.44	1.73	2.32	3.15	8.01
Volts:	26.5	33.5	44.9	61.1	148.2

T,inlet air: 72.8 F P,atm: 29.13 in Hg

Air flow

meter dP: 0.083 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	570	410	190
UB	860	720	450
UF	730	660	440
MB	1090	980	650
MF	990	900	660
LB	1480	1420	1030
LF	1520	1300	1180
F	4420	3900	3700

Run 097 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	80.6	84.3	91.8	107.7	195.2
18	81.2	84.6	92.2	110.2	194.7
19	81.2	85.7	92.6	110.3	195.7
20	81.4	85.3	92.2	110.6	195.8
21	81.4	85.1	92.6	109.0	194.9
22	81.1	85.3	93.2	108.5	199.4
23	81.2	85.3	93.6	110.5	201.4
24	81.3	85.8	94.1	112.7	200.0
25	81.2	86.0	93.9	113.9	203.8
26	81.5	86.4	94.0	114.1	203.5
27	81.1	85.5	93.8	111.5	203.2
28	81.5	85.3	93.7	111.7	203.6
29	80.6	82.7	91.3	109.3	195.6

Amps:	1.43	1.73	2.33	3.16	8.04
Volts:	26.3	33.5	45.0	61.2	148.4

T,inlet air: 73.3 F P,atm: 29.45 in Hg

Air flow

meter dP: 1.23 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	100	80	20
UB	240	170	130
UF	150	130	100
MB	390	300	220
MF	310	270	210
LB	670	560	420
LF	710	530	470
F	2770	2300	2100

Run 098 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	77.5	80.2	86.0	98.5	174.4
18	78.0	80.4	86.2	100.6	174.3
19	77.9	81.0	86.5	100.4	174.4
20	78.1	80.9	86.1	100.9	175.0
21	78.1	80.7	86.3	99.5	173.5
22	77.9	80.9	87.0	99.3	177.2
23	77.9	80.8	87.1	100.7	179.7
24	78.0	81.2	87.3	102.4	177.9
25	77.8	81.2	87.1	103.1	181.4
26	78.0	81.6	87.2	103.6	180.9
27	77.6	80.9	87.0	101.5	180.5
28	77.9	80.7	87.0	101.8	180.8
29	77.2	78.4	84.8	99.6	174.1

Amps:	1.44	1.74	2.33	3.15	8.08
Volts:	24.9	32.0	43.5	59.6	147.7

T,inlet air: 72.6 F P,atm: 29.44 in Hg

Air flow

meter dP: 2.3 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	70	60	10
UB	180	130	100
UF	120	100	80
MB	300	230	170
MF	240	210	170
LB	540	440	350
LF	560	430	380
F	2350	1940	1830

Run 099 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	79.7	81.8	86.8	97.1	163.8
18	80.2	82.1	87.1	98.9	163.7
19	80.0	82.2	87.1	98.6	163.5
20	80.3	82.5	87.1	99.2	164.3
21	80.2	82.1	87.2	97.8	162.0
22	80.2	82.5	87.7	98.0	165.5
23	80.0	82.2	87.7	98.9	168.3
24	80.3	82.6	87.9	100.4	166.3
25	79.8	82.4	87.4	100.7	169.6
26	80.1	82.8	87.7	101.3	169.1
27	79.6	82.0	87.3	99.6	168.6
28	80.1	82.1	87.5	99.9	168.7
29	79.3	79.8	85.1	97.5	162.2

Amps:	1.43	1.72	2.32	3.13	8.04
Volts:	26.2	33.3	44.9	60.7	148.1

T,inlet air: 75.8 F P,atm: 29.42 in Hg

Air flow

meter dP: 3.7 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	70	60	30
UB	160	130	110
UF	130	110	100
MB	280	230	180
MF	220	210	170
LB	490	410	340
LF	500	370	340
F	2100	1720	1600

Run 103 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	166.8	168.4	179.8	206.6	378.8
18	168.3	169.9	181.6	214.6	375.8
19	170.1	170.9	183.8	214.6	376.3
20	172.6	173.9	183.7	215.0	373.0
21	169.6	174.2	185.1	211.6	374.5
22	168.9	175.0	186.4	212.1	385.7
23	171.6	176.9	189.7	218.1	387.3
24	173.5	179.7	193.4	224.6	386.8
25	174.4	181.1	195.3	228.6	393.9
26	175.5	182.6	196.6	228.9	394.8
27	175.5	183.0	196.6	224.6	395.7
28	175.9	182.1	196.0	223.1	394.9
29	171.7	181.6	194.8	221.5	376.2

Amps:	1.79	2.15	2.86	3.91	9.98
Volts:	31.3	40.1	53.8	74.5	180.1

T,inlet air: 75.7 F P,atm: 29.25 in Hg

Air flow

meter dP: 0.0059 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1712	1617	1262
UB	1911	1709	1357
UF	1900	1592	1324
MB	2211	1948	1605
MF	2114	1785	1633
LB	2760	2471	2136
LF	2824	2272	2435
F	7019	5996	6038

Run 104 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	130.1	149.9	170.5	201.4	368.2
18	133.5	151.5	171.7	208.5	365.7
19	135.5	150.5	173.1	207.3	366.6
20	137.8	155.4	170.1	206.7	363.2
21	135.2	153.7	169.2	200.5	362.3
22	134.9	153.9	171.3	199.0	372.4
23	137.3	154.5	172.4	202.6	372.3
24	139.8	156.7	174.4	207.1	370.1
25	141.0	156.8	174.4	208.9	375.1
26	141.8	157.4	174.2	207.4	374.1
27	140.9	155.5	172.6	201.0	373.1
28	141.0	152.8	170.6	198.4	371.4
29	138.2	151.4	168.6	196.1	354.2

Amps:	1.76	2.14	2.84	3.88	9.92
Volts:	31.6	40.6	54.2	74.7	180.0

T,inlet air: 78.3 F P,atm: 28.84 in Hg

Air flow

meter dP: 0.037 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	1034	920	544
UB	1408	1266	981
UF	1248	1242	949
MB	1694	1593	1287
MF	1570	1515	1324
LB	2231	2173	1854
LF	2281	2036	2143
F	6485	5680	5633

Run 105 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	114.5	136.1	160.4	196.3	367.2
18	119.1	138.2	161.9	203.2	364.4
19	122.0	138.4	162.1	198.4	365.2
20	125.0	141.9	156.7	185.5	361.9
21	123.7	142.0	160.0	194.2	360.9
22	123.3	142.7	163.2	195.1	372.1
23	125.7	144.6	165.1	199.5	371.9
24	128.3	146.4	167.7	204.4	370.0
25	129.9	147.2	168.2	206.5	375.2
26	130.8	147.9	168.3	205.7	366.9
27	129.7	145.2	166.7	199.4	372.3
28	130.0	142.5	164.9	196.9	370.3
29	127.4	140.3	162.4	194.2	352.6

Amps:	1.78	2.14	2.89	3.84	9.89
Volts:	31.4	39.9	54.5	77.0	182.3

T,inlet air: 77.0 F P,atm: 29.05 in Hg

Air flow

meter dP: 0.083 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	860	740	340
UB	1270	1110	770
UF	1090	1070	720
MB	1620	1450	1070
MF	1490	1390	1110
LB	2230	2080	1680
LF	2270	1950	1960
F	6530	5700	5530

Run 107 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	86.7	92.1	103.8	127.4	261.3
18	87.5	92.5	104.3	133.3	260.9
19	88.0	94.0	105.3	133.1	262.0
20	88.3	93.9	104.3	133.6	262.4
21	88.1	93.5	104.7	130.7	259.2
22	87.6	93.8	105.9	130.1	267.2
23	87.9	93.8	106.6	133.1	271.1
24	88.2	94.5	107.2	136.3	268.6
25	88.0	94.9	106.9	137.8	273.8
26	88.2	95.2	107.0	138.0	273.3
27	87.8	94.0	106.9	134.2	272.8
28	88.4	93.4	106.7	134.4	272.7
29	87.8	92.0	105.6	133.3	261.3

Amps:	1.75	2.15	2.85	39.26	9.85
Volts:	31.7	41.0	54.7	75.7	182.1

T,inlet air: 75.6 F P,atm: 29.34 in Hg

Air flow

meter dP: 1.26 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	161	136	58
UB	383	313	243
UF	279	266	214
MB	623	516	390
MF	535	491	391
LB	1075	932	725
LF	1143	890	816
F	4419	3667	3361

Run 108 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	85.5	89.4	98.5	118.4	236.4
18	86.3	89.9	98.8	122.7	235.8
19	86.5	90.8	99.5	122.5	236.4
20	86.8	90.8	98.8	122.9	237.5
21	86.7	90.5	99.1	120.4	233.7
22	86.3	90.9	100.1	120.2	240.3
23	86.5	90.8	100.6	122.4	244.4
24	86.7	91.2	100.8	124.9	241.7
25	86.4	91.4	100.4	125.9	246.3
26	86.4	91.6	100.5	126.2	245.7
27	86.1	90.6	100.4	123.1	245.0
28	86.5	90.1	100.3	123.5	244.6
29	86.0	89.0	99.3	122.4	234.9

Amps:	1.75	2.14	2.85	3.89	9.86
Volts:	31.6	40.9	54.6	75.0	182.2

T,inlet air: 77.9 F P,atm: 29.37 in Hg
 Air flow
 meter dP: 2.23 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	114	98	40
UB	296	239	188
UF	212	202	168
MB	489	401	309
MF	404	377	304
LB	874	741	595
LF	900	693	648
F	3800	3121	2939

Run 109 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	85.4	88.2	95.1	110.4	210.8
18	85.9	88.6	95.3	113.9	210.0
19	86.0	89.2	95.8	113.7	210.1
20	86.2	89.2	95.1	114.1	211.8
21	86.2	89.1	95.4	111.8	207.3
22	85.9	89.4	96.4	112.1	212.7
23	86.1	89.4	96.8	114.0	217.2
24	86.2	89.6	96.8	115.7	214.0
25	85.8	89.7	96.3	116.3	218.4
26	85.9	89.9	96.4	116.9	217.7
27	85.5	89.0	96.4	114.6	217.0
28	85.8	88.7	96.4	115.1	216.4
29	85.4	87.8	95.6	114.0	208.5

Amps:	1.76	2.14	2.86	3.86	9.86
Volts:	31.8	40.9	54.8	74.3	182.2

T,inlet air: 79.6 F P,atm: 29.37 in Hg

Air flow

meter dP: 3.88 mm Hg

Heat Loss Differential Temperatures (mV*1000)			
	Bank 29	Bank 21	Bank 11
C	88	76	29
UB	245	196	156
UF	178	168	145
MB	405	331	256
MF	332	312	257
LB	741	610	501
LF	757	568	535
F	3270	2650	2530

Run 114 - RP2

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	91.9	103.2	121.9	151.6	310.8
18	93.8	104.5	123.2	161.8	309.6
19	94.1	105.1	124.4	158.3	311.2
20	94.8	106.3	123.4	161.7	310.6
21	94.4	106.4	124.5	158.7	309.6
22	93.7	106.9	125.9	157.4	318.4
23	94.1	107.0	127.1	161.2	319.3
24	95.0	108.4	128.7	165.5	317.5
25	96.5	110.4	130.0	168.8	323.9
26	97.0	111.3	130.5	168.9	322.4
27	96.8	110.4	130.6	164.4	321.0
28	97.6	109.1	130.0	163.7	319.6
29	97.1	107.2	128.9	162.3	305.6

Amps:	1.76	2.15	2.85	3.92	9.74
Volts:	32.1	41.2	54.9	75.8	180.7

T,inlet air:	74.8 F	P,atm:	29.10 in Hg
Air flow			
meter dP:	0.35 mm Hg		

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	389	330	210
UB	725	595	431
UF	587	518	398
MB	1052	894	643
MF	962	832	641
LB	1619	1475	1100
LF	1713	1365	1266
F	5464	4720	4328

Run 115 - RP1

Bank	Wall Temperatures (deg.F)				
	C	US	MS	LS	F
17	86.1	93.4	104.9	125.1	237.0
18	86.9	93.9	105.8	131.8	236.9
19	86.9	94.2	106.1	139.9	237.2
20	87.1	94.6	105.9	131.6	237.5
21	87.3	95.0	106.6	129.2	237.1
22	86.8	95.2	107.3	128.6	243.2
23	87.0	95.2	107.9	131.3	243.8
24	87.4	96.0	109.1	134.5	242.4
25	88.4	97.3	109.9	136.9	246.9
26	88.6	97.8	110.1	136.5	246.3
27	88.7	97.3	110.2	133.1	245.7
28	89.1	96.4	109.7	132.6	245.5
29	88.7	95.0	108.8	131.4	234.6

Amps:	1.46	1.75	2.33	3.14	8.01
Volts:	26.7	33.5	44.8	60.8	148.1

T,inlet air: 75.2 F P,atm: 29.20 in Hg

Air flow

meter dP: 0.368 mm Hg

Heat Loss Differential Temperatures (mV*1000)

	Bank 29	Bank 21	Bank 11
C	214	174	102
UB	418	343	243
UF	335	297	220
MB	645	544	388
MF	579	494	381
LB	1018	926	677
LF	1041	832	763
F	3651	3130	2836