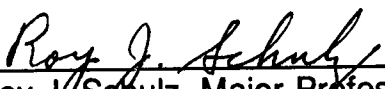
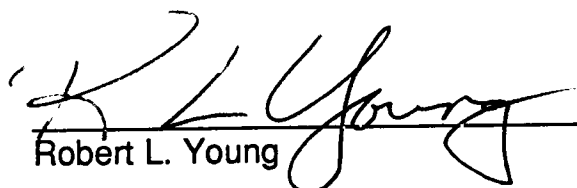


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**DESIGN AND PERFORMANCE OF A SHIELDED
HIGH VELOCITY THERMOCOUPLE PROBE
IN A HIGH-TEMPERATURE,
SEEDED-COAL-COMBUSTION
EXHAUST STREAM.**

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

Whitney J. Robinson

May 1990

DEDICATION

TO MY FATHER'S GIFT, ELAINE - WITH MY LOVE.

ACKNOWLEDGMENTS

As with any endeavor, I am indebted and grateful to the many people who have helped me both directly and indirectly during my graduate work.

I am understandably indebted to my supervisor and friend John P. Foote, PhD, for his help and unsurpassed technical knowledge in the field of Heat Transfer. I would also like to extend a special thanks to Professor Roy J. Schulz, my advisor and Chairman of my thesis committee, who obviously had a strong hand in my thesis. Special thanks also to Doctors R.L. Young and L.W. Crawford who also served on my committee and always aided me.

A sincere thanks is extended to Troy C. Shaver, William Millard and Robert Walker, Jr., for their invaluable assistance. In the typing and format problems, I am indebted to Elizabeth Huskey and Rebecka Stines. And to Jennifer Locke, Darryl Rasnake, and Jeff Holt for their patience and assistance with computer problems. Finally, the author would like to thank The University of Tennessee Space Institute for providing the opportunity to learn and the printing services for this thesis.

Any errors remaining in this work are solely my responsibility. The work and analysis for this research were performed at and for the University of Tennessee Space Institute, Tullahoma, Tennessee.

ABSTRACT

The problem of very high temperatures and particulate loadings in the exhaust streams of coal-fired magnetohydrodynamic power plants makes gas-stream temperature measurements very difficult. Therefore, effort has been expended trying to improve the design of the probe-mounted High Velocity Thermocouple (HVT), which has been used as a temperature sensor. The main problems that appear to affect the recorded temperatures are particulate blockage of the probe and radiation & conduction heat losses from the probe. As the probe must be brought to the gas temperature (the recovery temperature) by convection of the hot gases over the thermocouple, modifications were incorporated to minimize effects of particulate deposition & blockage.

The specific purpose of evaluating and redesigning the probe as needed was to obtain more accurate temperature readings in the Secondary Combustor of the U.S. Department of Energy Coal Fired Flow Facility (CFFF) at the University of Tennessee Space Institute. The objective of the research reported in this thesis was to take proper considerations of the aspirated flow velocity and all of the heat transfer mechanisms that establish the thermocouple temperature, thereby obtaining a more accurate measurement. The HVT, when properly calibrated and verified, can be utilized to correct the readings of non-aspirated thermocouples, as well as take a wider range of readings across the width of the exhaust stream.

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LIST OF SYMBOLS

A	Area	ft ²
Bi	Biot Number = hD/k	
C_p	Specific Heat at Constant Pressure	Btu/[lb °F]
C_v	Specific Heat at Constant Volume	
D	Diameter	ft
emf	Electromotive Force	
F	Shape Factor	
h	Heat Transfer Coefficient	Btu/[ft ² s °F]
J	Bessel Functions	
k	Thermal Conductivity	Btu/[ft s °F]
L	Length	ft
M	Mach Number	
m	mass	lb _m
Nu	Nusselt Number = $h_x L/k$	
P	Pressure	lbm/ft ²
Pr	Prandtl Number = ν/α	
q	Heat Flux	
R	Thermal Resistance	
Re	Reynolds Number	
r	Radius	ft; in

T	Temperature	°F; R
t	Time	s; hr
v	Velocity	ft/s
α	Thermal Diffusivity	
γ	Ratio of Specific Heats	
δ	Width	ft
ϵ	Emmissivity	
θ	$\theta \sim 0.4(T_m)^4$	°F, R
$\Re$	Recovery Factor	
λ	Thermal Conductivity	
μ	Viscosity (Dynamic)	lb/ft-s
ν	Viscosity (Kinematic)	ft ² /s
ν_o	Overall Heat Coefficient (Defined in Eq. 3-2)	
ρ	Density	lbm/ft ³
π	Pi	
σ	Stefan-Boltzman Constant	$0.1712 \times 10^{-8} \text{Btu/hr-ft}^2\text{-}^\circ\text{R}^4$

SUBSCRIPTS:

c	Convection
F	Fluid
g	Gas
i	Inner
o	Outer

p Probe
r Radiation
s Sensor
shaft Shaft of Thermocouple
tc Thermocouple
tip Tip of Thermocouple shield
w Wall
surf Surface

CHAPTER 1

INTRODUCTION.

1.A. Introduction.

Energy and its production will continue to be an important item of concern for the future. Looking upon the natural resources available, the combustion of coal delivers the majority of the electric power in the United States today. The economics of power production by fossil fuel combustion increases favorably as the combustion temperatures increase. To achieve higher efficiencies in fossil-fueled power production, the high-temperature, coal-fired, plasma-driven magnetohydrodynamic (MHD) power generation process was conceived. This process is based on burning seeded coal to temperatures exceeding 3000 K [1]. However, to maintain these more efficient, high-temperature processes requires advanced instrumentation and control systems. This thesis concerns a preliminary study of a high-temperature, adjustable measurement system for an MHD pilot test facility.

1.B. Test Facility.

In order to investigate the practical feasibility of the MHD process, the U.S. Department of Energy has funded at the University of Tennessee Space Institute (UTSI), and elsewhere, experimental and theoretical programs to

[1] R.C. Attig, et al., "MHD Bottoming Cycle Component Testing at the Coal Fired Flow Facility - 1987", Paper Presented at 26th Symposium on Engineering Aspects of Magnetohydrodynamics, June 1988.

develop MHD. At UTSI, the Department of Energy has funded the construction and operation of a facility called the Coal Fired Flow Facility (CFFF). The CFFF is the flow train for an open-cycle, coal-fired magnetohydrodynamic power system. The open-cycle MHD system has the potential of increasing conventional coal fired plants from 32 percent to 40-50 percent energy conversion efficiency. UTSI personnel operate this facility by burning seeded (K_2CO_3) pulverized coal in oxygen/nitrogen gas mixtures, creating extremely high temperatures, ash and other particulate loaded, plasma flow that creates the facility exhaust stream. A critical measure of the performance of the facility, and its components, is the temperature distribution of the exhaust gas along the flow train of the CFFF. Therefore, it is also extremely important to measure temperatures in these extreme environments as accurately as possible.

The current CFFF flow train is called the Low Mass Flow (LMF) test train number 4 which consists of an upstream coal combustor and ducting to simulate the aerodynamic and thermal characteristics of an MHD topping cycle, and the heat recovery and pollution control portion of a downstream steam plant. (See Fig. 1-1)

The CFFF utilizes fuel-rich combustion in the primary combustor to keep the exhaust products at a substoichiometric (fuel-rich) level for the control of nitrogen oxides (NO_x) in the exhaust. The exhaust gas flows into a radiant furnace, where it is cooled by heat transfer to steam generating equipment.

Downstream of the radiant furnace, a secondary combustor is employed to complete the burning process and generates additional heat.

The Secondary Combustor completes the process by burning the remaining combustibles in the flue gas. Thus it can recover additional energy for use in a steam cycle and help reduce pollution by CO. CO and H₂ are the main exhaust gas combustibles and their combustion reactions are very rapid at temperatures above 1800⁰F. Stable combustion is maintained by the injection of air to raise the stoichiometric ratio at the combustor outlet to 1.0.

The region of the flow train that concerns this thesis is the Secondary Combustor component. This component completes the combustion process started in the upstream primary combustor and as such it is necessary to accurately measure the temperatures in this component to determine the overall efficiency of the system.

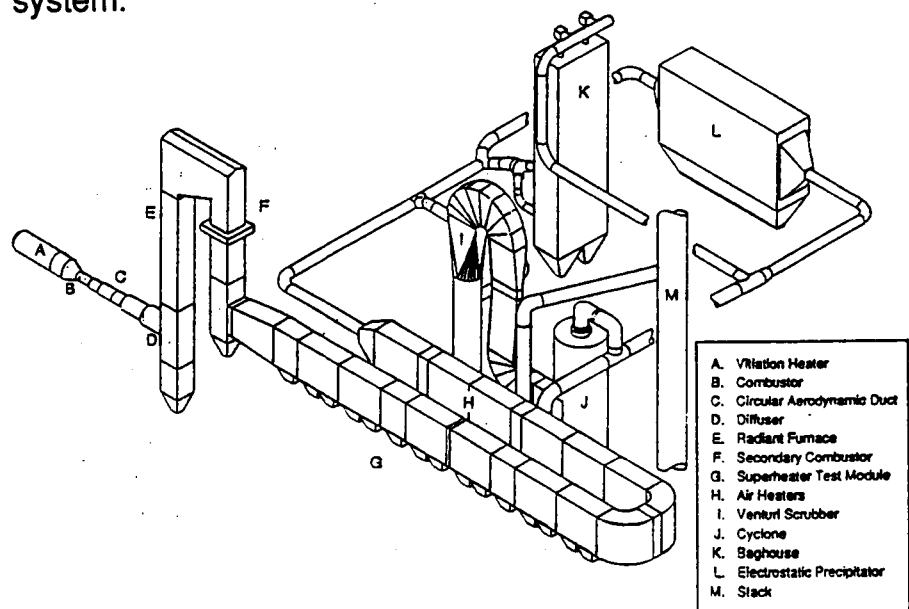


Fig. 1-1 CFFF Flow Train

1.C. HVT Probe.

The system that was selected to take temperature measurements traversing the width of the Secondary Combustor exhaust stream was developed by the Babcock & Wilcox Company (B&W) in the 1950's [2]. (See Fig. 1-2) This design was called, by B&W, a "high velocity thermocouple" or HVT probe. This design was adapted for use in the CFFF flow train, even though the temperatures in the flow train were significantly higher than those used to develop the original HVT design.

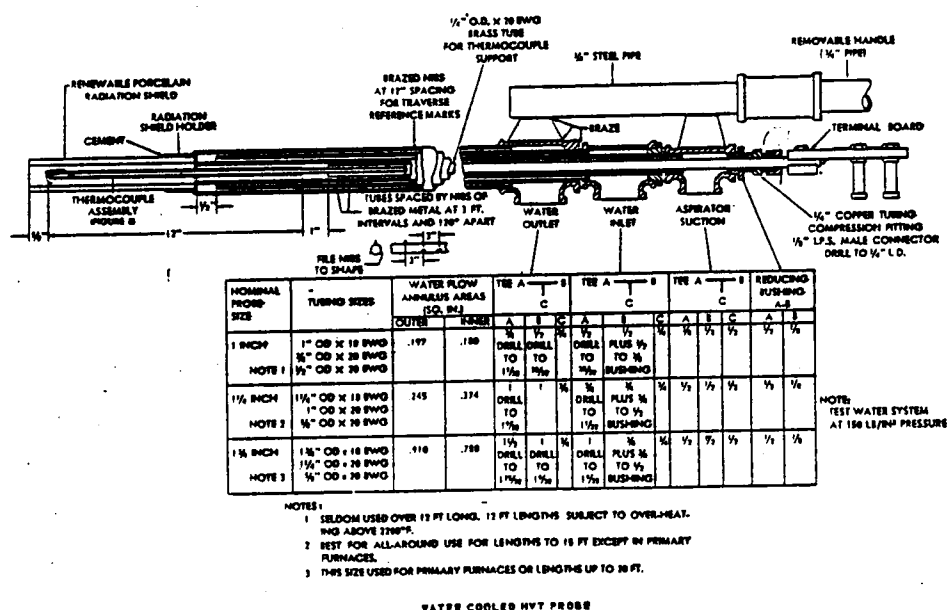


FIG. 1-2 B&W 1950 Probe Design

The basic construction of the HVT probe consisted of a long cylindrical shaft fabricated of concentric, stainless steel tubes, cooled by water. Its tip

[2] F.G. Ely, "Measurement of Gas Temperatures in Boiler Furnaces.", Bulletin 3-555 (Allaince, Ohio: the Babcock & Wilson Company), p.7.

contained a notch to hold a ceramic radiation shield and a shielded thermocouple was run down the center of the tube. The probe used a nitrogen powered inductor to draw hot exhaust gases into the probe, passing over the thermocouple, and venting the gases to the environment or to a gas collector. (See Fig. 1-3) Thus, the basic HVT design, used in lower temperature, cleaner environments is called an aspirated, shielded thermocouple probe.

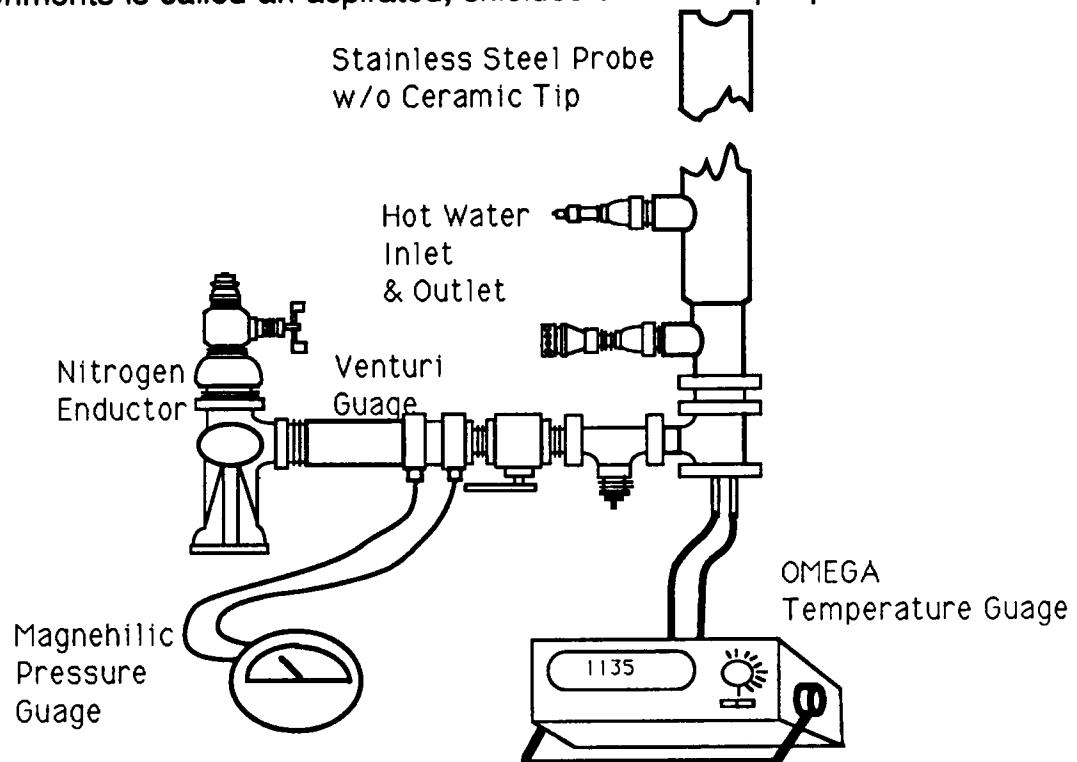


Fig. 1-3 Probe in Temperature Reading Configuration

Fig. 1-4, below, gives the recommended mass velocities of the flow over the thermocouple as determined by the B&W Company, without which, the temperature read by the measuring instrument will be too low [3]:

[3] Ely, p.11.

LEGEND:
Y Axis - Temp. indicated, °F
X Axis - Gas flow through
HVT (lb/ft²-hr)

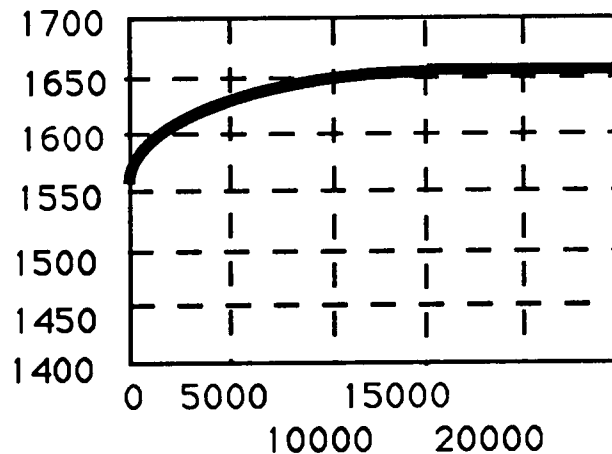


Fig. 1-4 Influence of Gas Flow vs Recorded Temperature for HVT.

The HVT-type probe first used in earlier CFFF tests copied the original B&W design, but testing showed it suffered severe blockage problems due to ash accumulation within the probe. Later modifications were made to the probe design and altered as the situation dictated and the MHD test schedule allowed.

The determination of the temperature of the flow of gas through the CFFF relates to the measure of heat transfer and the equilibrium state of the gases along the flow train. As was previously stated, the temperature at which heat is transferred is a measure of the system efficiency, versus the losses of the system, and the accuracy of such a measure is the subject and goal of this paper.

This paper will discuss the environment of concern, basic heat transfer concepts as applied to the thermocouple used in the CFFF flow train, equipment and procedures, desired and attained results, and conclusions of the thesis. A premise of this thesis is that the raw data from the wall-mounted stationary thermocouple do not correctly reflect the actual internal gas flow temperatures.

The HVT probe as discussed in this thesis was utilized in two configurations: with a single thermocouple and with a dual thermocouple scheme. (See Fig. 1-5) The single thermocouple was placed in the center of the probe with welded spikes to the 1/4 inch stainless steel sheath to hold the thermocouple in the middle of the flow. The dual thermocouple configuration had the centered thermocouple and another embedded in the radiation shield.

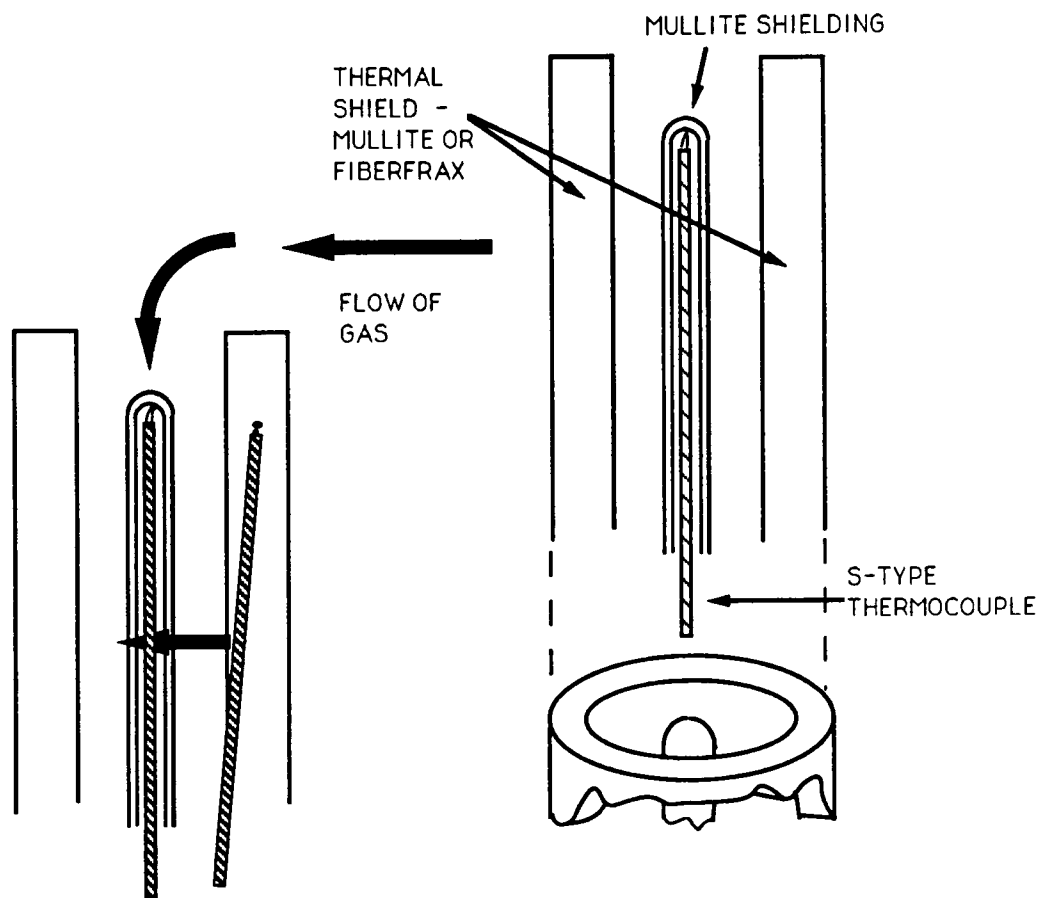


Fig. 1-5 Single and Dual Thermocouple Configurations

CHAPTER 2.

STATEMENT OF THE PROBLEM WITH DISCUSSION OF THE EXPERIMENTAL ENVIRONMENT AND EQUIPMENT.

2.A. Statement of the Problem.

The basic reason for the research was to improve the HVT probe design to ensure that it would work satisfactorily in the high temperature, particulate-loaded exhaust stream of the Secondary Combustor of the CFFF. Such an instrument will lead eventually to a better understanding of the details of the combustion processes inside the Secondary Combustor.

Temperature readings taken in the Secondary Combustor with stationary, non-aspirated thermocouples are hypothesized in this thesis to be too low, by a consistent factor of at least 200, to a maximum of 400, degrees Fahrenheit. If thermocouple probes are designed without considerations for flow velocity of the gases, shielding material, and non-steady state environment, the temperature readings from a thermocouple will be lower than that of the actual environment. Previous temperature readings are believed to be incorrect because their measurements were dominated by radiation heat losses. This effect was also observed in the temperature readings recorded by the HVT probe when it became plugged (non-aspirating).

2.A. The Secondary Combustor Design.

The CFFF-LMF Secondary Combustor is shown in Fig. 2-1. The inlet and outlet sections of the secondary combustor are 4ft x 4ft in cross section. The section

where the secondary air is injected has two converging walls which form a 4ft x 3ft restriction. The walls are water-cooled and refractory-lined. Passing through each converging wall are two rows of tubes through which air supplied by a forced draft fan is injected upstream at an angle of 120° relative to the incoming gas flow. Each wall has ten 3.07 inch diameter air tubes and nine 1.61 inch diameter tubes. (Fig. 2-2)

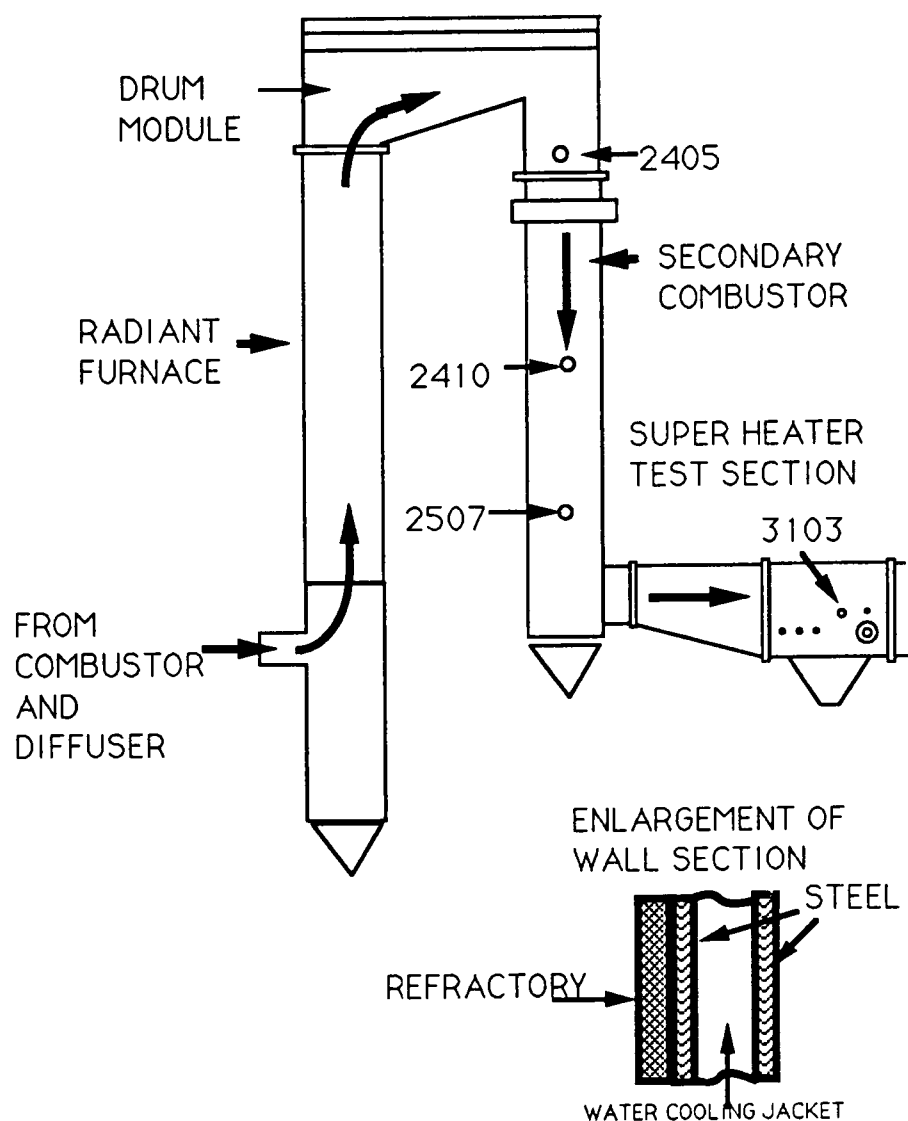


Fig. 2-1 Secondary Combustor and Port Numbers

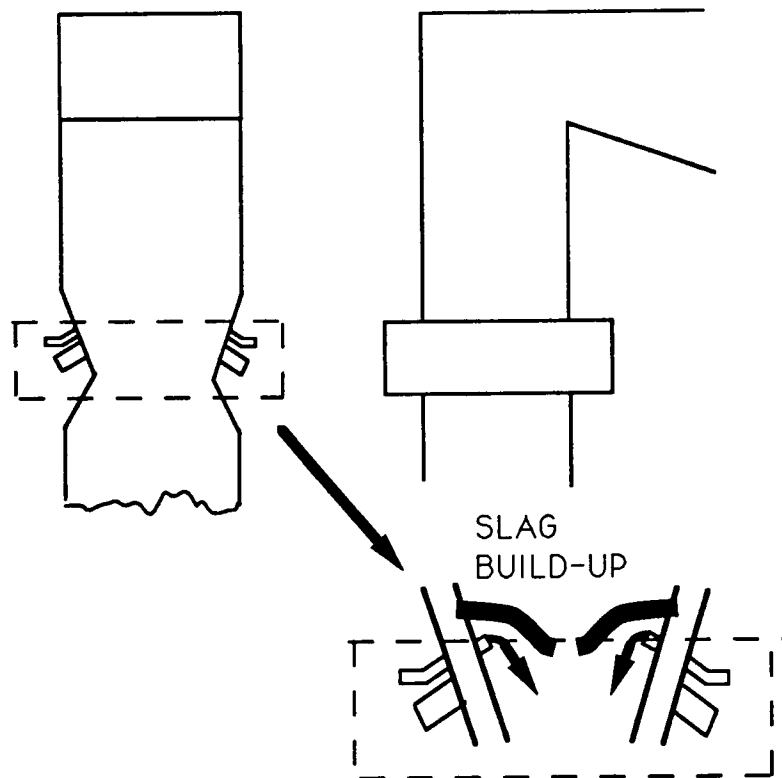


Fig. 2-2 Details of Secondary Combustor and the Air Injector

These tubes can be individually capped to provide many different injector arrangements. The combustor length after the point of injection is about 17 ft.

Temperature readings for this thesis were conducted solely in the vicinity of the Secondary Combustor and the entrance of the Superheater Test Module of the CFFF. Readings were taken above and below the air-injection ports of the Secondary Combustor, so temperatures were recorded for both reducing and oxidizing environments. Since the facility serves as a DOE test bed, there have been a variety of tests and experiments conducted on the system performance as affected by the type of coal burned, amount of water sprayed

into different parts of the channel, opening of ports during the test (which permitted additional air to be drawn into the channels), etc.

2.C. Temperature Measuring Equipment and Its Limitations.

2.C.(1) Thermocouples - Design & Construction.

A thermocouple is the temperature sensing element of the HVT measuring probe. A thermocouple consists of two dissimilar electrical conductors, insulated from each other except where joined together at two junctions.

For the HVT probe there are two primary junctions: one is the measuring junction, which will be subjected to the temperature to be measured, and the second is the reference junction, which is usually maintained at a known temperature such as the ice point. In most current thermocouple applications the ice point is reproduced electronically by resistance calibration of the measuring apparatus.

2.C.(1)a. Basic Principles.

Seebeck discovered that an electrical current will flow in a closed circuit of two dissimilar metals when the junctions of the metals are at different temperatures. The reasons for this are the Peltier electromotive force (emf) and the Thomson emf. The Peltier emf exists between two different metals placed in contact and it is dependent upon the temperature and type of metals for the magnitude of the emf. Also, for a homogeneous wire heated at one end there

is generated an emf. Again, the magnitude depends on the metal and the difference in the temperature at both ends [4].

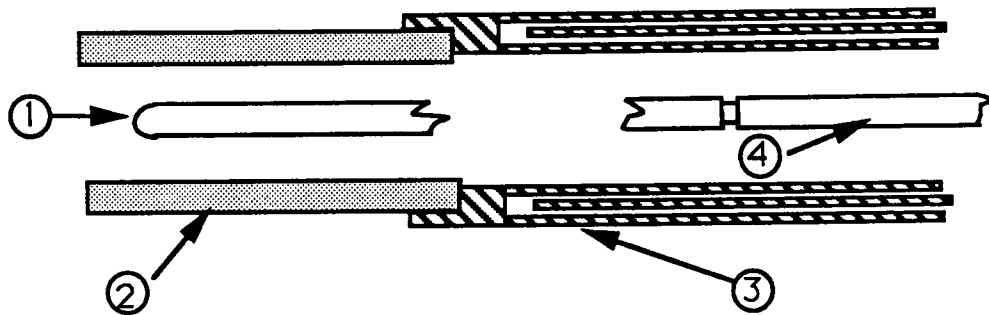
The total emf acting over a closed circuit of dissimilar metals is the sum of the Peltier and Thomson emf. Peltier emf acts over junctions and Thomson emf acts over each separate wire. If the temperature of one junction is known or fixed (i.e. 32°F) at the ice point, the temperature of the other junction can be determined by the measurement of the emf developed over the entire circuit.

The measuring junction is a union of dissimilar metals made with an electric arc welder. While welding the platinum and platinum-10% rhodium wires, care must be utilized to avoid overheating the junction and poisoning the junction's temperature response. Such excess heat will cause the wire to build a natural resistance to the electrical flow and create a smaller emf.

The thermocouple wires used in the present study were platinum and platinum-rhodium and were insulated with an alumina ceramic, as well as having a ceramic cap over the junction to prevent direct contact of the thermocouple with the exhaust gas constituents. (See Fig. 2-3) The ceramic insulation has two major drawbacks: (1) ceramics are very fragile and will take only minimal mechanical or thermal shock impact and; (2) there is a time delay associated with heat conduction through the material in this unsteady environment, so that

[4] Part 3 Temperature Measurement, Instruments and Apparatus. (New York, NY: The American Society of Mechanical Engineers, 1974), P. 18.

the thermocouple temperature changes lag behind the changes in gas temperature by some time shift.



1. Mullite cap over the S-type thermocouple
2. Mullite or FIBERFRAX radiation shield
3. Stainless steel water-cooled probe
4. Stainless steel 1/4 in. tubing to support the thermocouple

Fig. 2-3 HVT Probe Tip

Due to the high cost of platinum, extension wires are run from the primary thermocouple wires back to the measuring equipment. The extension wires must have compatible characteristics, such as reference emf and temperature response with the thermocouple or the reference junction. Incompatibility of the materials and the location of the connection within the HVT probe, which was subject to unsteady heat flux (environmental temperatures were over 1000°F), will negate part of the emf generated by the high temperature thermocouple wire junction. For this reason, the extension wire was picked for its compatibility and the reference junction occurs outside of the probe in ambient conditions for

which the temperature gauge is calibrated [5]. In the present thesis, the primary platinum and platinum-rhodium wires were attached to copper and copper-constantan wires, respectively.

2.C.(1)b. Thermocouple Materials.

Thermocouples are made for different environments from low temperature cryogenic conditions to the ultra high temperatures. In this experiment there are two basic considerations: 1. oxidizing vs reducing atmospheres and 2. temperature ranges. This thesis will not deal extensively with the topic of thermocouple materials for reducing atmospheres. Suffice it to say that reducing atmospheres have more stringent requirements for thermocouples to survive, which is one of the reasons an S-type (or noble metal type) thermocouple was chosen.

Platinum has been selected as the universal standard with which to compare materials for constructing thermocouple wires. It has been used to define the International Temperature Scale from 630.74°C to 1064.43°C. It also has a high degree of chemical inertness and stability at high temperatures in oxidizing atmospheres. The common usage range is from -300° to +3200° F.

[5] W.M.Rohsenow, J.P.Hartnett, and E.N.Garnic', Handbook of Heat Transfer Fundamentals. (New York, NY: McGraw-Hill Book Company), pp.4-37 to 4-71.

Data on S-type thermocouple emf vs temperature range was taken from the ASME Performance Test Code chart [6]:

The noble metals (Type S) include:

- (a) platinum-10% rhodium vs platinum (Will match the standard reference table to $\pm 0.5\%$ of the measured emf.);
- (b) platinum -13% rhodium vs platinum (It produces a slightly greater emf per degree of temperature than (a).);
- (c) platinum -30% rhodium vs platinum -6% rhodium (Slightly better in the higher temperature range and has greater mechanical strength than either (a) or (b).).

The base metals include:

- (a) copper-constantan (Type T; ~ 55% copper & 45% nickel; -300 to +700°F);
- (b) iron-constantan (Type J; -200 to +1400°F; can be used in reducing atmospheres with good stability);
- (c) chromel-alumel (Type K; 90% nickel & 10% chromium for the chromel and 94% nickel, 3% manganese, 2% aluminum, & 1% silicon for the alumel) is more resistant to oxidation than other base metal combinations; -200 to 2300°F;

[6] "Temperature Measurement, Instruments and Apparatus." Supplement to ASME Performance Test Code ASME PTC 19.3-1974 (New York, NY: ASME, United Engineering Center, 1974).

- (d) chromel-constantan (Type E; has the highest thermoelectric output) up to 1400°F.

2.C.(1)c. Characteristics of Thermocouple Reactions.

Tables 2.1 - 2.3 provide data on the performance of the different material combinations that are used to construct thermocouples. Table 2.1 lists the average thermoelectric power for normal operating ranges. The sensitivity of the thermocouples varies with temperature. Table 2-2 gives the recommended temperature limits of the different thermocouple wire materials, and Table 2-3 gives the limits for error in recording the temperatures of the junctions of the thermocouples (which are not the "errors" in "measured" temperatures).

TABLE 2.1. Sensitivity of Thermocouples.

THERMOCOUPLE	MICROVOLTS/DEG F
Copper-Constantan	28.0 (32-650°F)
Iron-Constantan	32.0 (32-1400°F)
Chromel-Allumel	23.0 (32-3000°F)
Platinum & 10% Rhodium - Platinum	6.3 (1000-2650°F)
Chromel-Constantan	42.0 (32-1400°F)

TABLE 2.2 Recommended Temperature Limits (F)

TC TYPE	NO. 8 GAUGE	NO. 14 GAUGE	NO. 20 GAUGE	NO. 24 GAUGE	NO. 28 GAUGE
T		700	500	400	400
J	1400	1100	900	700	700
E	1600	1200	1000	800	800
K	2300	2000	1800	1600	1600
R&S				2700	
B				3100	

NOTE: These ranges are recommended for continuous service of the thermocouple. At higher temperatures the thermocouples would have a limited life prior to becoming poisoned.

TABLE 2.3 Limits of Errors.

TC TYPE	TEMP. RANGE (F°)	LIMITS OF ERROR
T	-299 TO -74	
	-150 TO -74	2%
	-74 TO 199	1.4°F
	199 TO 700	3.4%
J	32 TO 530	4°F
	530 TO 1400	3/4%
E	32 TO 600	3°F
	600 TO 1600	1/2%
K	32 TO 530	4°F
	530 TO 2300	3/4%
R&S	32 TO 1000	2.5°F
	1000 TO 2700	1/4%
B	1600 TO 3100	1/2%

2.D. Problems Encountered with the HVT System.

2.D.(1) There were extensive problems with electrical shortages with the HVT thermocouple itself and its extension wiring. In designing the inside of the HVT probe, priority was given to keeping the flow space open for the aspirated gases and particulates. This resulted in minimum insulation space for coating the wires. Any twists in the thermocouple wire generally tore the insulation and resulted in an electrical shortage, so that in effect the thermocouple junction moved back to the shortage location.

2.D.(2) Plugging or blockage of the probe was due to ash deposition on the walls and build-up in the passage. This was not abated by the use of stainless steel in the construction of the probe to reduce the cohesive character of the walls. Plugging also seemed to depend greatly on the diameter of the passage chamber.

2.D.(3) The entire system was to some degree subject to thermal shock; especially noted by the fragility of ceramic radiation shield materials and the thermocouple itself. This increased exposure times for the HVT probe by forcing slow entries and exits when the HVT was inserted in the exhaust stream, which aggravated the ash pluggage problems.

2.D.(4) Leakage of air into the probe during temperature sampling affects the aspiration rate of the probe, thereby reducing the flow over the thermocouple and lowering the temperature readings.

2.D.(5) Uncertainty in the velocity of the aspirated hot gas flow over the thermocouple prompted the design of a venturi gauge for the probe.

2D.(6) During temperature measurements thermocouples were damaged by accidents and slag or wall insulation falling on them, and due to the high costs, they were rewelded for further use.

2.E. Problems with the Recording Gauge.

The initial instrument used to display the measured temperatures was an OMEGA ® Temperature Gauge with a visual display panel. Thus, there was no way to automatically record the maximum, minimum, or mean temperature sensed by the HVT probe. Temperature readings were recorded by visually ascertaining the mean temperature reading. At times, the temperature fluctuations were on the order of 15-20 degrees Fahrenheit. Therefore, due to the lack of an automated recording system with calibration features, there is some degree of uncertainty in the recorded temperatures.

2.F. Interpretation of Data.

In the interpretation of the recorded temperature data, the primary concern is to account for the corrections required for the effects of conduction, convection, and radiation on the thermocouple's energy balance and thereby attempt to understand the dominant thermal processes that affect the design and operation of the probe.

Interpretation of the thermocouple readings is also affected by non-steady combustion processes which might be occurring in the Secondary Combustor,

in addition to the effect of particulate in the flow (as a time related property, which affects the HVT probe by slag build-up, plugging, etc.). The radiation losses of energy of the thermocouple occur because the probe tip "sees" the walls of the Secondary Combustor, which are significantly cooler than the exhaust gases, about 800 - 1000°F. The thermocouple also "sees" the shield, which is at a somewhat lower temperature than the thermocouple. An "accurate" temperature measurement results from being able to evaluate all of the errors (systematic and biasing) that affect the recorded temperatures, and in addition, account for probe construction and sensitivity of materials and equipment used to measure the signals.

CHAPTER 3

HVT PROBE HEAT TRANSFER.

3.A. Heat Transfer Measurement Theory and Thermocouples.

This chapter will describe computational models for estimating errors in temperature measurement in fluids and solids.

The output of a sensor, such as a thermocouple, represents an approximation of the temperature at a location in the fluid. A variety of factors can cause deviations between the probe output and the actual temperature. The thermocouple configuration may alter the thermal conditions merely by its presence. The measured temperature is less than the actual temperature when heat is conducted to or away from the thermocouple junction through the lead wires or its protective mullite shielding. Another problem occurs with radiant heat exchange between the duct walls and the probe.

Computation of errors must include not only the basic understanding of the data read by the instruments, but also that the convective heat transfer to the probe cannot take place without a temperature difference between the gas and the probe. In addition, the transient nature of the process (non-steady state) will bring about a temperature difference between the sensor and its environment.

One approach to compensate for losses from the thermocouple sensor is to increase the fluid velocity over the thermocouple and thereby, increase the

convective heat transfer coefficient. Manipulating the size and shape of the thermocouple or adjusting the type of shielding to reduce radiative exchange with surrounding walls can further reduce heat losses. This portion of the thesis will be concerned with just the sources of error in temperature measurement and to discuss analytical models which may be used to estimate the errors.

The heat transfer model for the HVT probe that is discussed first will concern the lead wires and their conduction problem.

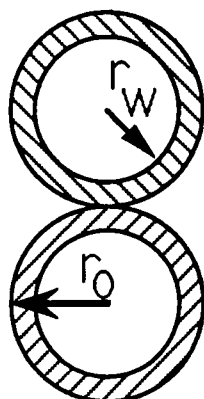
3.B. Influences of Conduction and Convection on HVT Probe.

3.B.(1) Thermocouple Wire Bundle Fin Problem.

3.B.(1)a. Analysis of the Problem.

A thermocouple has two insulation wrapped wires that act like a heat transfer fin. The heat conduction equation for this configuration is as follows.

T_{∞}



r_w = wire radius

r_o = outer radius = $r_w + \delta_i$

δ_i = insulation thickness

Fig. 3-1 Wiring Configuration.

Assuming convection heat transfer from the outer surface of the wires, the local heat loss on a small length of the wires is given as:

$$dq_{conv} = v_o A_s [T(x) - T_\infty] \quad (\text{Eq. 3-1})$$

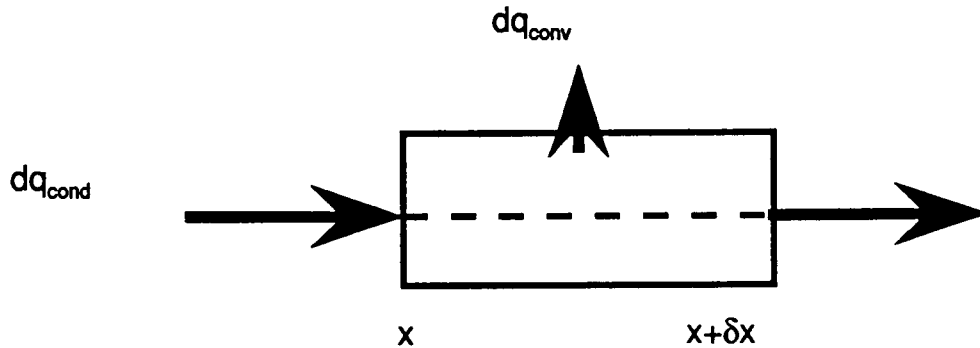


Fig. 3-2 Heat Transfer on a Single Wire.

The overall heat transfer coefficient is given as that for a single wire:

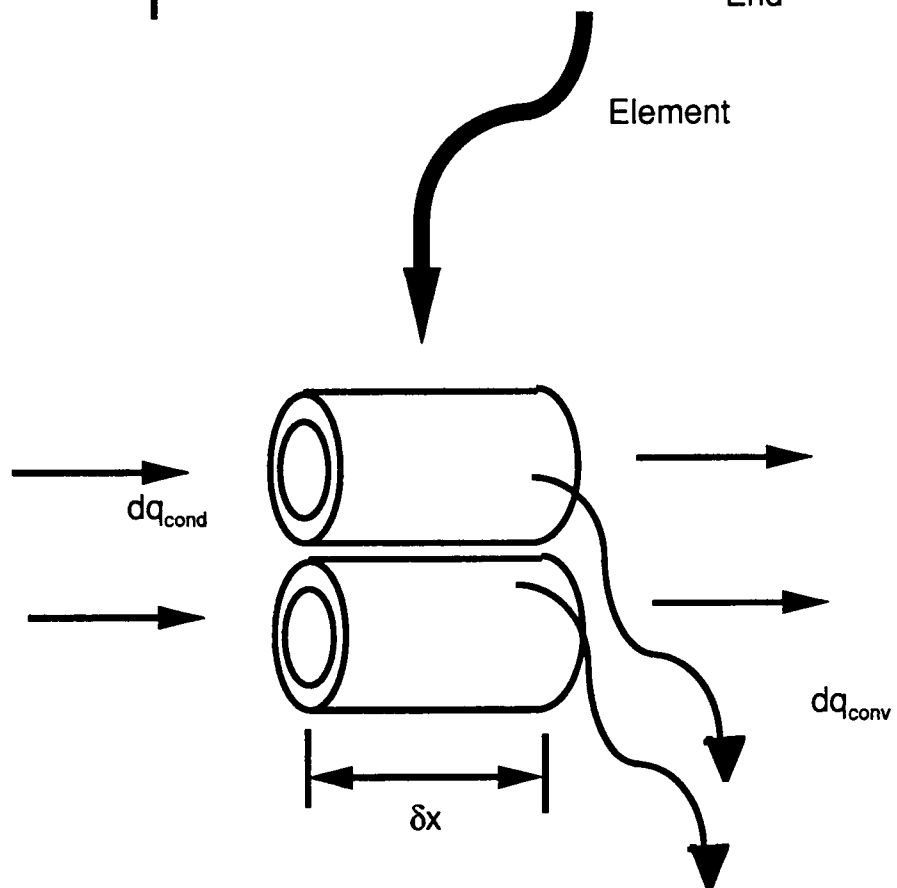
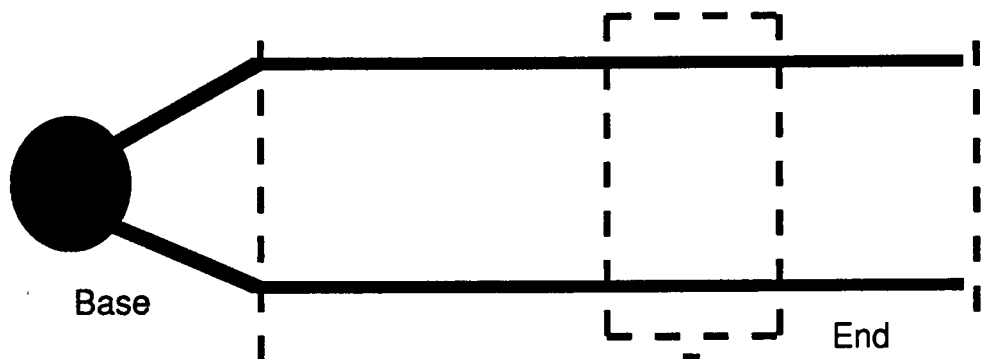
$$v_o = \frac{1}{\frac{r_w}{k_l} \ln \left(\frac{r_o}{r_l} \right) + \frac{r_w}{r_o} \frac{1}{h_s}} \quad (\text{Eq. 3-2})$$

where k_l is the thermal conductivity of the insulation on the wires, and h_s is the outer surface coefficient for heat transfer.

$$dq_{conv} = v_o A_s [T(x) - T_\infty] \quad (\text{Eq. 3-3})$$

or for two wires, (See Fig. 3-3 Fin Problem)

$$dq_{conv} = 2v_o 2\pi r_w \delta x [T(x) - T_\infty] \quad (\text{Eq. 3-4})$$



$$\theta'' - \lambda^2 \theta = 0$$

$$\theta = T(x) - T_{\infty}$$

$$\lambda^2 = \frac{2v_o}{k_w r_w}$$

$\theta(x)$ depends on the applied boundary conditions.

Fig. 3-3 Fin Problem.

The heat transfers by conduction into and out of the element of wire are:

$$dq_{cond}|_x = -k_w A_w \frac{dT(x)}{dx} \Big|_x \quad (\text{Eq. 3-5})$$

and,

$$dq_{cond}|_{x+\delta x} = -k_w A_w \frac{dT(x)}{dx} \Big|_{x+\delta x} \quad (\text{Eq. 3-6})$$

Using a Taylor series to represent $dT/dx|_{x+\delta x}$: (Eq. 3-7)

$$\frac{dT}{dx} \Big|_{x+\delta x} = \frac{dT}{dx} \Big|_x + \frac{d^2 T}{dx^2} \Big|_x \delta x + \frac{1}{2} \frac{d^3 T}{dx^3} \Big|_x \delta x^2 + \dots$$

The energy balance is:

$$dq_{cond} \Big|_x = dq_{cond} \Big|_{x+\delta x} + dq_{conv} \quad (\text{Eq. 3-8})$$

Therefore, (Eq. 3-9)

$$-k_w A_w \frac{dT}{dx} \Big|_x = -k_w A_w \frac{dT}{dx} \Big|_x - k_w A_w \frac{d^2 T}{dx^2} \Big|_x \delta x$$

$$\frac{-k_w A_w}{2} \frac{d^3 T}{dx^3} \Big|_x \delta x^2 + \dots + v_o 2\pi r_w \delta x (T_x - T_\infty)$$

Dividing through by δx and taking the limit as $\delta x \rightarrow 0$, gives the resulting differential equation for the temperature of the wire:

$$d^2 T_x - \frac{4\pi r_w v_o (T_x - T_\infty)}{k_w 2\pi r_w^2} = 0 \quad (\text{Eq. 3-10})$$

or,

$$\frac{d^2 T_x}{dx^2} - \frac{2v_o}{k_w r_w} (T_x - T_\infty) = 0 \quad (\text{Eq. 3-11})$$

Let $2v_o/k_w r_w$ be denoted as a constant parameter, λ^2 : then the thermocouple wire fin temperature is determined by the solution of the differential equation:

$$T_x - \lambda^2(T_x - T_\infty) = 0 \quad (\text{Eq. 3-12})$$

Which is subject to two applied boundary conditions. The units of λ^2 are as follows:

$$\lambda^2 = \frac{2v_o}{k_w r_w} ; \quad (\text{Eq. 3-13})$$

$$v_o \rightarrow \frac{1}{\frac{\frac{L}{E}}{L-S-T} + \frac{L}{L} \frac{1}{\frac{E}{L^2-S-T}}} \quad (\text{Eq. 3-14})$$

$$\rightarrow \frac{1}{\frac{L^2-S-T}{E} + \frac{L^2-S-T}{E}}$$

$$v_o \rightarrow \frac{E}{L^2 - S - T}$$

$$k_w r_w \rightarrow \left(\frac{E}{L - S - T} \right) L$$

$$k_w r_w \rightarrow \frac{E}{S - T}$$

$$\therefore \lambda^2 = \frac{\frac{E}{L^2} - S - T}{\frac{E}{S} - T} = \frac{1}{L^2}$$

3.B.(1)b. Solutions to the Fin Heat Transfer Problem.

The solutions to the fin heat transfer problem have been obtained long ago. They have been tabularized. For the equation $\theta'' - m^2 \theta = 0$, the following solutions have been obtained for different sets of boundary conditions:

To compare to the insulated fin problem: replace h by v_o , k by k_w :

$$m^2 = \lambda^2 = \frac{2v_o}{k_w r_w} \quad (\text{Eq. 3-15})$$

and,

$$M = (2r_w)^{\frac{3}{2}} \pi \sqrt{v_o k_w} \quad (\text{Eq. 3-16})$$

and $\theta = T_x - T_\infty$.

Table 3-1: Temperature Distribution and Heat Loss for Fins of Uniform Section.

CASE	TIP CONDITION ($x = L$)	TEMPERATURE DISTRIBUTION	FIN HEAT TRANSFER RATE
A	Convection Heat Transfer: $h\theta(L) = -k d\theta/dx$ at $x=L$	$\frac{\cosh m(L-x) + (h/mk) \sinh m(L-x)}{\cosh mL + (h/mk) \sinh mL}$	$M \frac{\sinh mL + (h/mk) \cosh mL}{\cosh mL + (h/mk) \sinh mL}$
B	Adiabatic: $d\theta/dx = 0$ at $x=L$	$\frac{\cosh m(L-x)}{\cosh mL}$	$M \tanh mL$
C	Prescribed Temperature: $\theta(L) = \theta_L$	$\frac{(\theta_L/\theta_b) \sinh mx + \sinh m(L-x)}{\sinh mL}$	$M \frac{(\cosh mL - \theta_L/\theta_b)}{\sinh mL}$
D	Infinite Fin ($L \rightarrow \infty$) $\theta(L) = 0$	e^{-mx}	M

3.B.(2) Mullite Ceramic Cap.

A simplified model of heat transfer through the protective shielding covering the thermocouple will be applied here by the theory for a temperature difference across a thin plate. This model entails some fairly hefty simplifications. Fig. 3-5 shows that the mullite cap is involved in a fairly complex heat transfer environment. The cap is the recipient of heat by direct convective heat transfer from the flow of the gases being sucked over the thermocouple's mullite cap. Also there is a heat transfer from the particulate in the gas as it adheres to the mullite. And finally there is the radiation component from the gas and to the surrounding walls and probe shield. The mullite complicates the problem by conducting heat down along its own sides and reradiating heat out to the heat shield and the gas.

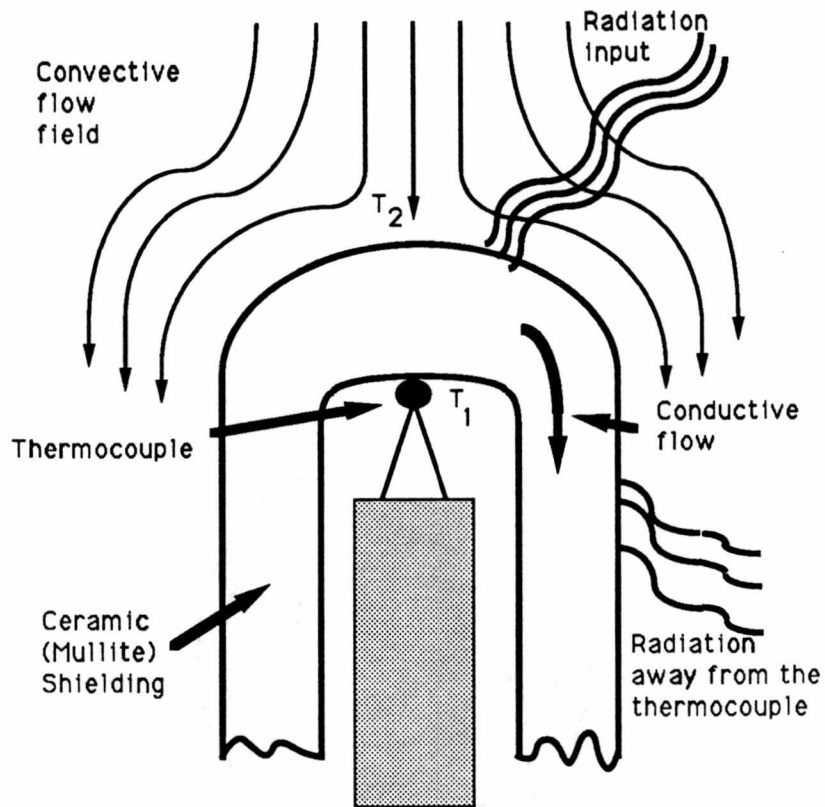


Fig. 3-4 Heat Transactions Over Mullite Cap

But the discussion initially will consider the mullite as a thin plate of a thickness, δ . It is situated between two flows, having temperatures T_1 and T_2 . (See Fig. 3-5)

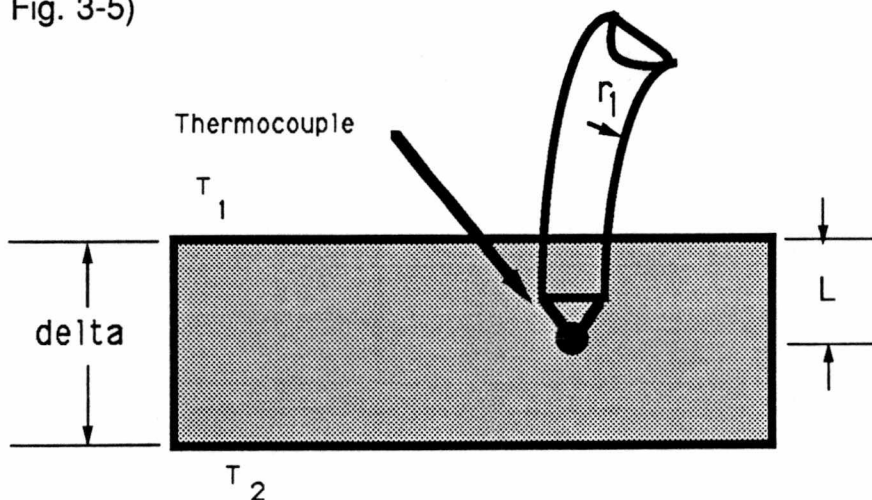


Fig. 3-5 Thin Plate Assumption

The equilibrium temperature of the plate without a thermocouple (assuming δ and the temperature variation across the plate is negligible) is:

$$T_{plate} = \frac{h_1 T_1 + h_2 T_2}{h_1 + h_2} \quad (\text{Eq. 3-17})$$

The effect of the thermocouple is to channel heat into or out of the plate, with the appropriate change of temperature of the plate or the thermocouple. The temperature error has been calculated by P.J. Schneider [7]. For T_{tc} as the temperature of the thermocouple and T_p as the temperature of the plate without the thermocouple, the error $T_{tc} - T_p$ is: (Eq. 3-18)

$$\frac{T_{tc} - T_p}{T_1 - T_p} = \left[1 + 2\pi k \delta \epsilon r_1 \left(\frac{J_1(\epsilon r_1)}{J_0(\epsilon r_1)} \right) \frac{\sqrt{kAR}}{\tanh(kAR)^{-1/2} L} \right]$$

where

$$\epsilon = \frac{\sqrt{(h_1 + h_2)}}{k \delta} \quad (\text{Eq. 3-19})$$

The symbols J_0 and J_1 denote modified Bessel functions of the second kind.

3.C. Convection Effects.

3.C.(1) Steady-State Temperature Measurements.

Historically, for heat transfer analysis, thermocouples have been treated as a fin. The thermocouple leads (and/or support) are attached to a wall whose temperature is T_w . The surrounding fluid temperature is T_f . Due to heat

[7] P.J. Schneider, Conduction Heat Transfer. (Reading, Massachusetts: Addison Wesley, 1955)

conduction in the leads, the temperature at the thermocouple junction will be between T_w and T_f .

For high-speed gas flows viscous dissipation plays an important role, such that in the absence of heat conduction in the lead wires, and radiation losses, the thermocouple would read the recovery temperature T_r , defined by:

$$T_r = T_f \left(1 + \Re \frac{\gamma - 1}{2} M^2 \right) \quad (\text{Eq. 3-20})$$

where $\Re$ is the recovery factor (on the order of unity), γ is the ratio of specific heats, C_p/C_v , and M is the Mach number of the flow. In such a flow, the temperature sensed by the thermocouple junction lies somewhere between T_w and $T_{f(\text{stagnant})}$.

For gases whose Prandtl numbers are approximately 0.7 (air with some hydrocarbon combustion products), experimental information gives: (for wires normal to the flow) [8]

$$\Re = 0.68 \pm 0.07$$

(for wires parallel to the flow)

(Eq. 3-21)

[8] R.J. Moffat, "Thermocouple Theory and Practice." 1974 ISA ASIT 74216, p. 111-124

$$\mathfrak{R} = 0.86 \pm 0.09$$

But the use of the recovery factor is only for high speed flows. It is more evident in a high speed flow as the higher kinetic energy will not all be converted to thermal energy in the boundary layer surrounding the shield of the thermocouple, but the boundary layer causes an energy conversion problem at all speeds.

The Secondary Combustor velocities have been calculated (See Chapter 5 for details and references) at a maximum of about 55 ft/sec. Since this equates to Mach = .03, $T_r \approx T_f$. Note Fig. 3-3, below, for the relationship of $\mathfrak{R}$ versus the speed of the flow normal to the wires of the thermocouple[9]:

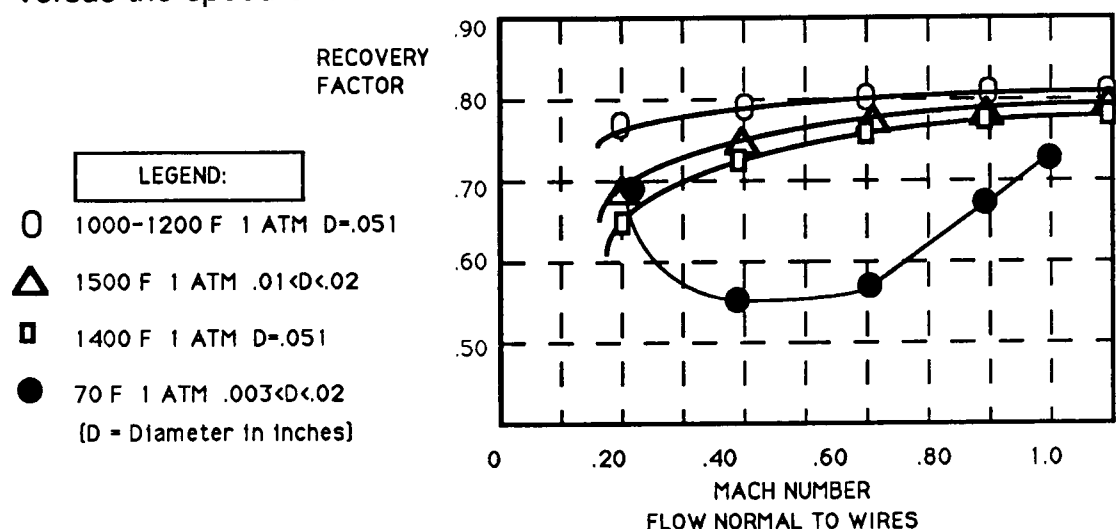


FIG. 3-6 Recovery Factor vs Mach Number

[9] R.J. Moffat, Dynamic Temperature Measurement. (New York, NY: McGraw-Hill Book Company, 1978.), p.278.

3.C.(2) Unsteady Temperature Measurements.

The traditional model for estimating errors in the measurement of unsteady fluid temperatures assumes negligible spatial temperature variations within the thermocouple sensor junction and neglects lead wire heat conduction to or from the sensor. Thus, if m , C_p , and A_s are the mass, specific heat, and surface area of the sensor, the basic energy balance is:

$$mC_p \left(\frac{dT_{tc}}{dt} \right) = hA_s(T_f - T_{tc}) \quad (\text{Eq. 3-22})$$

The temperature error $T_f - T_{tc}$ at any time is then equal to: $(MC_p/hA_s)(dT_{tc}/dt)$ and so, in principle, the error can be estimated from measured values of the temperature time derivative dT_{tc}/dt . In unsteady flow the heat transfer coefficient, h , varies in time; however for a simplified analysis, the simplification will be made that it is constant. In the present study, this type of analysis was not done because the temperature recording equipment did not permit dT_{tc}/dt to be adequately measured. For given time variations $T_f(t)$ in the fluid temperature, the error in the thermocouple output can be calculated by integrating equation 3-22.

3.D. Radiation Effects.

Radiative heat transfer is encountered in situations in which the sensor and/or the surroundings are at high temperatures or the differences in temperature between the gas, the sensor, and/or the surroundings are high.

To provide some approximated quantitative guidelines, it is useful to estimate the magnitudes of an effective radiation heat transfer coefficient. Consider an enclosure whose walls have arbitrary radiation properties. The walls are at the isothermal temperature T_w . The radiation heat loss per unit time and area of a small body situated within the enclosure is:

$$q_r = \epsilon \sigma (T^4 - T_w^4) \quad (\text{Eq. 3-23})$$

in which ϵ and T denote the emittance and temperature of the small body. Equation 3-23 is based on the assumption that the small body can be treated as a grey body.

An alternative form of equation 3-23 can be written in terms of the radiation heat transfer coefficient h_r :

$$q_r = h_r (T - T_w) \quad (\text{Eq. 3-24})$$

where:

$$h_r = \frac{\epsilon \sigma (T^4 - T_w^4)}{T - T_w} \quad (\text{Eq. 3-25})$$

If the temperature difference $T - T_w$ is substantially smaller than T_w , then h_r becomes (See derivation of following in Annex E):

$$h_r \sim 4\epsilon \sigma T_w^3 \quad (\text{Eq. 3-26})$$

Radiation arriving at the sensor and/or its leads may originate at solid surfaces or in the adjacent gas. Many common gases such as air, nitrogen, oxygen, etc. are virtually radiatively non-participating except at very high temperature levels, while others such as water vapor and carbon dioxide participate at intermediate temperature levels. Combustion gases and gas flows containing solid particles also may be radiatively active.

With respect to temperature measurement in gases, radiant energy arriving at the sensor and/or leads from solid surfaces tends to increase the error (if the temperature of such surfaces differs from the gas temperatures), while gas radiation tends to decrease the error (since the gas is hotter than the thermocouple). In cases where radiation is expected to cause a significant error in gas temperature measurements, it is common to shield the sensor so that its view of the error-inducing radiating solid surfaces is obstructed. In the CFFF, the walls are significantly cooler than the hot exhaust gases. For example, the walls are of the order 800 - 1000°F, while the gas is of the order 2100 - 2400°F.

The accounting of gas radiation introduces further uncertainties into the analysis because the radiative properties of gas bodies are not known to high accuracy. It is believed sufficient to treat the radiative interchange between the thermocouple and the gas by employing a radiation heat transfer coefficient similar to that used for radiation between the duct surface and the thermocouple.

3.E. Emitting/Absorbing Environment.

In an absorbing-emitting medium such as the CFFF, there must be a determination of the optical "thickness" of the gas. Optical thickness is said to range from "optically thin" to "optically thick", which is a function of the emission-absorption bandwidth in the gas. Most texts are vague in the exact definition of optically thick or thin. This paper is specifically concerned with heat transfer, the gases to be monitored will be the amount of CO₂, H₂O, and particulate in the gas flow.

Although, the gas is "transparent" to the eye, such that with a heat visor one, can see across the duct, at times with remarkable clarity, heat in certain bands is blocked by specific molecules of gas and the particulates in the flow. The degree of the blockage will determine whether the CFFF duct flow is optically thick. Measurements have been made up to the end of the Radiant Furnace which determined a peak gas emissivity of $\epsilon = 0.8$ declining to 0.1, but no measurements, to date, have been made in the Secondary Combustor [10].

With an optically thick gas the medium is highly absorbing. And as such the radiation can travel only a relatively short distance prior to being absorbed. The large amount of particulate also serves to redirect the heat via direct and re-reflection means. In short, the radiation penetration distance is small compared with the distance over which any significant temperature variations

[10] Conversations with B.C.Winkleman, Physicist, Advanced Measurements Group, ECP, UTSI, April 5-6, 1990.

can occur. Therefore, the local intensity of radiation will be the result of radiation coming only from the nearby, local gas which is itself within a close approximation of the temperature of the location under study.

3.F. Simplified Radiation Analysis.

3.F.(1) Combined Convective and Radiative Heat Transfer.

Because the thermocouple is shielded within a cylinder-type shield, the first step to its analysis will be to define the heat balance equation. The total heat transfer to either shield or thermocouple will be expressed in a simplified

form: $q_{net} = q_{convection} + q_{radiation} = 0$ (Eq. 3-27)

$$= h_c A (T_g - T_w) + \epsilon \sigma A (T_g^4 - T_w^4) \quad (\text{Eq. 3-28})$$

$$= A (T_g - T_w) \left[h_c + \frac{\epsilon \sigma (T_g^4 - T_w^4)}{(T_g - T_w)} \right] \quad (\text{Eq. 3-29})$$

$$q_{total} = A (T_g - T_w) (h_c + h_r) \quad (\text{Eq. 3-30})$$

where the heat transfer coefficient, h_r , is due to the radiation heat loss and is

defined as (See Annex E for derivation): $h_r = \epsilon \sigma \left(\frac{T_g^4 - T_w^4}{T_g - T_w} \right) = \epsilon \sigma \Delta \theta$

If $T_m = (T_g + T_w)/2$ and if $0.9 < T_g/T_w < 1.1$, then $\Delta \theta$ is equivalent to $0.4(T_m)^4$. With this assumption the resulting error will be less than 0.01 degree Rankine.

3.F.(2) Thermal analysis of the heat shield and thermocouple.

Heat is given by the gas to the thermocouple by convection and radiation. Heat is also given by the gas to the shield by convection and radiation. Both the thermocouple and the shield lose energy predominately by radiation. The heat lost by the thermocouple goes to the shield and possibly to colder duct walls by radiation. The shield loses energy to the duct walls by radiation in an optically thin environment. The convective heat transport to the thermocouple is

$$Q_1 = h A_{tc} (T_g - T_{tc})$$

given by:

The heat gained by the shield from the gas by convection and from the thermocouple by radiation is:

$$Q_2 = 2h_s A_s (T_g - T_s) + \frac{A_{tc} \sigma (T_{tc}^4 - T_s^4)}{\frac{1}{e_c} + \frac{1}{F_{cs}} + \frac{1-e_s}{e_s} \left(\frac{A_{tc}}{A_s} \right)}$$

Heat by convection is given to the shield on the inside and outside surface. If $F_{cs} = 1.0$, which is appropriate to a cylinder within a cylinder, then:

$$Q_2 = 2h_s A_s (T_g - T_s) + A_c \epsilon_o \sigma (T_{tc}^4 - T_s^4)$$

where

$$\epsilon_o = \frac{1}{\frac{1}{\epsilon_c} + \frac{1-\epsilon}{\epsilon_s} \frac{A_o}{A_s}}$$

Heat given by the shield to the Secondary Combustor environment is:

$$Q_s = \frac{A_s \sigma (T_s^4 - T_g^4)}{\frac{1-\epsilon}{\epsilon_s} + \frac{1}{F_{sw}} + \frac{1-\epsilon}{\epsilon_w} (A_s)}$$

Now with an estimate for $F_{sw} = 1$, Eq. 3-31 is simplified to:

$$Q_s = A_s \epsilon_s (T_s^4 - T_w^4)$$

For a steady-state environment, the heat gained by the thermocouple by convection will equal the heat loss by radiation from the tip and to the shield:

$$hA_{tc}(T_g - T_{tc}) = A_{tc} \sigma \epsilon_o (T_{tc}^4 - T_s^4) + A_{tp} \sigma \epsilon_{tp} (T_{tc}^4 - T_w^4)$$

Similarly for steady-state heat flow the total heat gained by the shield will equal the heat loss from the shield by radiation to the gas and thermocouple:

(Eq. 3-31)

$$q = h_s A_s (T_g - T_s) + h_{so} A_{so} (T_g - T_s) + A_{so} \sigma \epsilon_g F_{2-3} T_g^4 - \epsilon_s A_{so} \sigma F_{2-3} T_s^4 \\ + \epsilon_{tc} \sigma A_s F_{2-1} T_{tc}^4 - \epsilon \sigma A_s F_{2-1} T_s^4$$

Eq. 3-31 reflects an assumption for an optically thick CFFF environment. A simplified numerical analysis in Annex F demonstrates unrealistic results when assuming an optically thin environment. See Annex E for the derivation of the view-factor used in the previous equation.

One simplification made with the theories and equations just discussed was to neglect the effect of slag buildup on the mullite cap and the radiation shield. The effects of slag on heat transfer can be included as follows. Concentrating first on the mullite shielding, the determination must be made as to the type of bonding between the mullite and the slag. (This ignores another large area of question - the consistency of the slag and an associated heat transfer coefficient.) Between any two surfaces in contact, there is a resistance layer or series of pockets of resistance to heat transfer due to imperfect contact and some type of fluid filling the pockets. The overall conductance of a typical joint is a function of the materials in contact (their individual conductivity, surface finish, flatness, & hardness), contact pressure, mean temperature, and heat fluxes across the joint.

$$q'' = -k_1 \left(\frac{dT}{dx} \right)_1 = -k_2 \left(\frac{dT}{dx} \right)_2$$

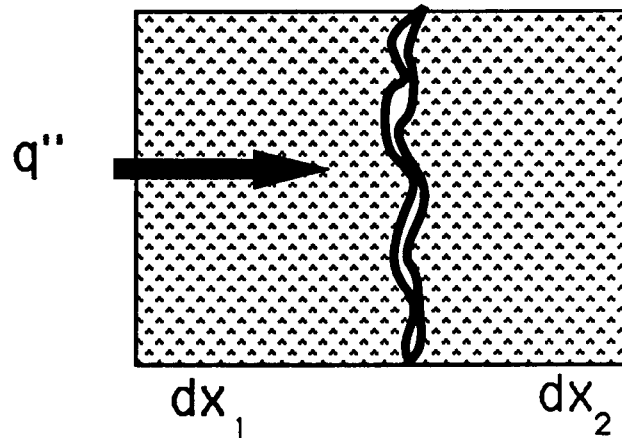


Fig. 3-7 Contact Problems.

The interface heat transfer coefficient $h_i = q''/\Delta T_i$. With perfect contact $h_i \Rightarrow \infty$ and $T_1 = T_2$. However, having given this description of the problem, its effects will be ignored in the evaluation of the theory since fundamental understanding and measurements of the particulate deposition process is lacking.

It should be mentioned that the previous analysis has simplified the actual scenario by assuming uniform heating effects on the periphery of the radiation shield. This is not the actual case because the slag buildup will be non-uniform around the periphery of the probe. The deposition is predominately on the front or upstream face of the probe. A minimum deposition occurs on the backside of the probe, where a recirculation region occurs.

CHAPTER 4

RESULTS.

In trying to analyze the combustion process in the Secondary Combustor, accurate temperature readings are required. As such, the data taken with the HVT probe's prior to the LMF4-O test (August 1988) had to be evaluated. The old data corresponded very closely to the readings of the non-aspirated stationary thermocouple and as such made it very suspect.

In the initial test program it was difficult, at best, to determine if the flue gas was being satisfactorily aspirated over the thermocouple assembly. This led to the construction and testing of a venturi gauge and its meter, which were applied to measure the flow rate of the aspirated gas, which in turn, helped demonstrate that the slag build up in the probe was the problem. After two incremental size increases in the diameter of the probe, the last probe design remained unblocked sufficiently long enough to take fully aspirated temperature readings across the width of the channel.

Additional problems occurred with electrical shortages of the thermocouple extension wire. This brought under scrutiny the axial temperature distribution within the probe. From previous testing, the temperature was believed to be circa 500°F at 20 to 24 inches inside the probe from the tip when the tip was at about 2000°F. [11] Material degradation indicated that the

[11] Discussions with P. Lynch, Environmental Sciences and Waste, ECP, UTSI, 1988.

temperature was closer to 1000°F. This led to restructuring the wiring and inside support for the thermocouple. (It was necessary to rebuild the entire probe 3-4 times during a regular test period when using the probe.)

Thermal shock of the original mullite thermal shield forced the search for a replacement material, which began with FIBERFRAX®. [12] It had excellent thermal shock characteristics but was rated only to 2300°F for continuous use. Continuous failure of the FIBERFRAX® was one of the first confirmations that the gas temperature was higher than previously indicated by the non-aspirated thermocouples.

The following is a sequential discussion of each of the results of the HVT probe channel traverses during MHD test and during testing with UTSI's Advanced Combustor. The LMF4 numbering indicates testing periods when the CFFF was run. See corresponding LMF4 material in the annexes for actual temperature readings. These readings are presented in both tabulated and plotted (graphical) formats.

4.A. LMF4-O Results.

During this test, the first HVT probe design was only one inch in outside diameter, which allowed less than .6 inches for the inside diameter to include passage way for the .25 inch tube holding the S-type thermocouple. The probe was initially cooled by water at ambient temperature. This led to severe

[12] FIBERFRAX is laminated Alumina-Silica paper. Manufactured by Metaullics® Systems, 31925 Aurora Rd., Solon, Ohio, 44139.

problems with the ash and particulate coagulating and clogging the passage.

Typical gas temperature readings were about 1900°F above the secondary Combustor air inlets and about 2100°F below the inlet, where the secondary combustion was occurring. Note that the readings in Port 2503 were around 2300°F just prior to the probe becoming blocked with ash (Annex A Table I) which forced the removal of the probe for cleaning and reconstruction.

4.B. LMF4-P Results.

The probe was redesigned and built with a larger diameter to attempt to alleviate the ash blockage problems. Temperature readings for the ports indicated in Annex A Table II were consistently higher than the LMF4-O test for the same ports. The probe tip was fabricated from FIBERFRAX® instead of mullite and it proved to be much more resistant to thermal shock. However, as the HVT probe was moved down the Secondary Combustor, the FIBERFRAX® began to fail.

A two thermocouple configuration was employed in the probe design to check the temperature difference between the thermal shielding and the thermocouple. A fairly steady difference of 160-200F was maintained between the two thermocouples, with the shield temperature being the lower one. The first three readings for Port 2405 showed a higher difference between the thermocouples than the norm but believe this to be caused by an open port upstream of the HVT, causing additional cooling of the shield.

4.C. LMF4-Q Results.

The temperatures in Table III of Annex A show that the temperatures were still relatively close to the those reported for the non-aspirated thermocouples. It is of some interest that the shield temperatures of the non-aspirated readings were higher than the main thermocouple. But the final analysis is that the probe was too plugged to attain the needed velocity of the aspirated flow over the sensor thermocouple. The pluggage actually was caused by the FIBERFRAX® thermal shield, which began to unravel and become sucked back up into the probe.

The FIBERFRAX® thermal shield began to unravel and "collapsed" about the center thermocouple, blocking the aspiration pathway. This brought the two thermocouples within a 1/4 inch of each other. The aspirator pulled a hole through the FIBERFRAX® and did not draw gases over either thermocouple. Initially this failure was believed to be due to a shipment error of the wrong type of FIBERFRAX® from the company, as there are two qualities. Since there was some doubt as to the type of material shipped from the company, no definite conclusion could be drawn about the temperatures in the channel.

4.D. LMF4-R Results.

Problems with the FIBERFRAX® continued despite insuring that the higher quality binder was purchased. FIBERFRAX® is rated to 2300°F for continuous use. It is also not subject to chemical deterioration from the products of coal combustion and seed.

In addition, the extension wire's insulating cover burned off. This occurred at a depth of over 20 inches into the probe from the tip located in the furnace; the wire's insulation was rated to 900°F.

Table 4.1 presents a comparison of the HVT readings (both aspirated and non-aspirated) with the stationary, wall-mounted, non-aspirated thermocouples.

TABLE 4.1. Non-Aspirated vs Aspirated HVT Readings.

STATIONARY	HVT (ASPIRATED)	HVT (NON- ASPIRATED)
1353	1520	
1767	1881	1764
2040	2354	1976
2244	2444	

The proximity of the HVT probe to the stationary thermocouple was roughly three feet. Note the similarity of the stationary thermocouple vs the non-aspirated HVT readings. The discrepancy in the second comparison was due to the main HVT thermocouple slipping back in the probe while the mullite cover stayed in place. This caused an internal convection problem with a gap of approximately four inches of air between the mullite cap and the thermocouple junction.

Comparing Annex A's Tables IV to V - VII is a good comparison of a "clean" vs "dirty" flow. During oil pre-burn, the gas temperature is considerable lower, and there is no particulate in the gas stream. Therefore, in this "optically thin" environment radiation is the predominant heat transfer mechanism.

4.E. LMF4-S Results.

Results of this test were similar to those of test LMF4-R. (See Annex A, Tables VIII and IX) Again the thermocouple extension wire failed after approximately 30 - 45 minutes within the CFFF, but readings were obtained prior to its failure. The non-aspirated HVT readings compared remarkable well to those of the Stationary thermocouples.

During this test the HVT probe was built with an extension wire of REFRASIL® [13], which was reported to be rated to 1600°F for continuous use. Despite initially good results with open flame testing, this wire "failed" inside the probe during CFFF testing. After completely gutting and rebuilding the probe several times, readings were still erratic, but, only when the thermocouple assembly was in the probe with its hot water circulating. The hot water circulating in the probe caused it to read erratically from 200-300°F. Apparently, the ice point moved within the probe due to the incompatibility of the S-type thermocouple wire and the REFRASIL® wire. The OMEGA® temperature measuring gauge was calibrated for an ambient temperature of 70°F, not 200 to 1000°F. Therefore, the HVT was converted back to the older style extension wire[14] for readings which were obtained until its insulation burned off.

[13] REFRASIL® is a braided thermocouple wire rated to 1600°F. It is sold by OMEGA Engineering, Inc., P.O.Box 4047, Stanford, CT 06907-0047.

[14] OMEGA Engineering, Inc.

Again during this test, non-aspirated HVT readings were almost identical to the stationary S-type thermocouple readings. At a penetration depth of fifteen inches, which approximates the depth of the tip of the stationary thermocouples in the ducting, the HVT probe read 1928°F while the stationary thermocouple read 1980°F. Upon inspection of the probe, the thermocouple had slipped back away from the mullite cap allowing a convection problem with the air surrounding the junction at the tip. This would account for the 50° temperature difference between the HVT probe and the non-aspirated, stationary thermocouples.

4.F. Advanced Combustor Results.

The Advanced Combustor is a boiler converted to burn micronized coal at temperatures of about 1900 - 2500°F. See Annex B for a listing of typical readings with the HVT probe. The Advanced Combustor's combustion environment is "clean", or optically thin, compared to the CFFF, in that the amount of particulate is relatively low. Therefore, radiation is the predominate heat transfer mechanism for the Advanced Combustor and as such served somewhat as a controlled test for the HVT probe. It provided data for comparison by contrast of the difference between the thermocouple and shield temperatures, with the relatively small difference between the thermocouple and shield temperatures indicating the more significant effect of convection in the CFFF.

4.G. Overall Comments on the Use of the HVT.

4.G.(1) Any type of air leak in the probe's rear assembly (due to gaps in the sealants) or from the gasket surrounding the probe and the port would cause a lower temperature to be read. Since the pressure in the CFFF channel is sub-atmospheric and the aspirator also creates a low pressure, any opening in the probe to the ambient atmosphere instantly becomes the source of cool air. The air would be sucked in both directions - down the aspirator which reduced tip aspiration and over the thermocouple (which would cool the same).

In addition, it was observed that if a port were opened upstream of the probe it would upset the combustion rates in the Secondary Combustor. Normally, this would bring the stoichiometric level up and begin the additional combustion early - with higher temperatures earlier in the channel.

4.G.(2) Some of the early HVT probe blockage problems were not the sole reason for the recorded temperatures being the same as those of the non-aspirated thermocouples. Earlier test data had indicated that at a distance of 24 inches into the probe, the aspirated gases should have cooled down to a temperature of around 500°F. Hence all the earlier designs (for the tests designated LMF4-O to LMF4-Q) had extension wiring and insulation rated too low to withstand the actual temperatures within the probe. Upgrading the extension wire to a 900°F capability did not alleviate the problems of burning off the insulating cover and resulting in electric shortages.

CHAPTER 5

ANALYSIS OF THE RESULTS

5.A. Discussion.

Results indicate that in the Secondary Combustor, typical differences between a non-aspirated thermocouple and the HVT probe will be in the range of 200 - 450°F. Disregarding the effect of particle adhesion to the thermocouple shield and conduction losses, the two main heat transfer mechanisms are convection and radiation. (See Fig. 5-1)

This chapter will demonstrate the numerical approach taken to analyze these effects, based on the measured data, and give a final argument about the ratio of the effects of convection vs radiation on the difference between measured temperature and gas temperature. In addition, the general structure of the probe and thermocouple assembly will be discussed.

5.B. Calculations of the Flue Gas Temperature.

5.B.(1) General.

As shown in Chapter 3, there are several energy balances that can be made on the HVT probe to check the temperature readings. This paper will utilize two equations in an attempt to assess and check the measurements; assuming an optically thick environment. In general terms, with a steady-state assumption, the heat transfer rate leaving either the shield or the thermocouple should equal that entering.

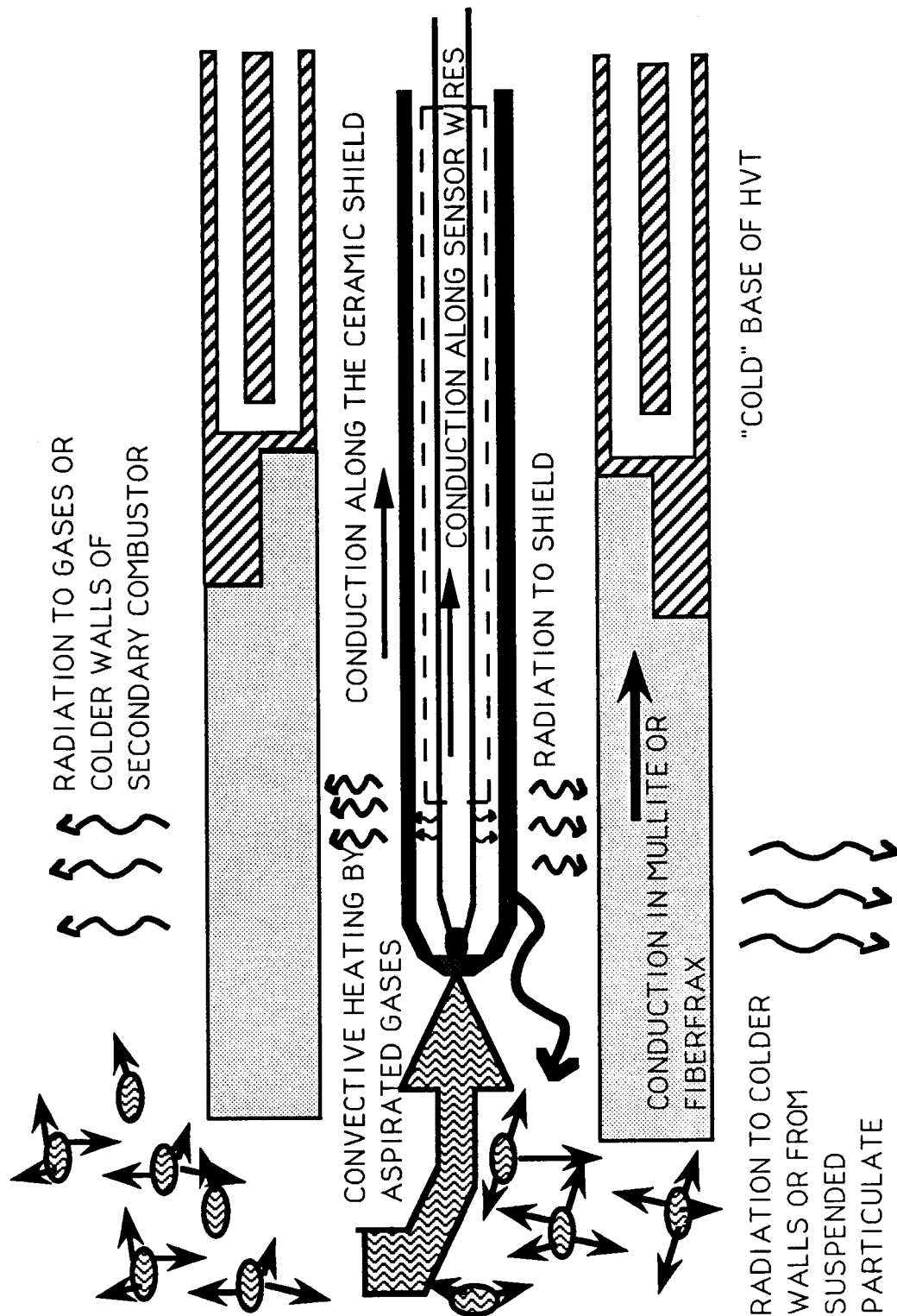


Fig. 5-1 Heat Transfer Mechanisms

5.B.(2) Optically Thin Environment.

Recalling the energy balance equation written for the shield in Chapter 3 and modifying it for an optically thin environment yields:

$$q = \text{Radiation from shield} = \begin{array}{l} \text{Convection from gas} \\ + \text{Radiation from gas} \\ + \text{Radiation from thermocouple} \end{array}$$

Or, as previously given: $q = \epsilon_s \sigma A_s (T_g^4 - T_w^4) =$ (Eq. 5-1)

$$2h_s A_s (T_g - T_s) + \epsilon_g \sigma A_s (T_g^4 - T_s^4) + \epsilon_c \sigma A_{tc} (T_{tc}^4 - T_s^4)$$

See Annex F for the solution of Eq. 5-1. The unrealistically high answer, $T_g = T_s + 906^\circ$; made without accounting for degradation of the radiation to the wall through the gas, demonstrates the need to assume an optically thick environment.

5.B.(3) Optically Thick Environment.

5.B.(3)(a) Evaluation of the optically thick environment as an energy balance on the thermal shield.

Eq. 3-31 will be used to determine the temperature of the flue gas:

q = Convection over inside of shield + over
 inside of shield + Gas radiation to outer
 shield - Shield radiation to gas
 + Thermocouple radiation to shield
 - Shield radiation to thermocouple

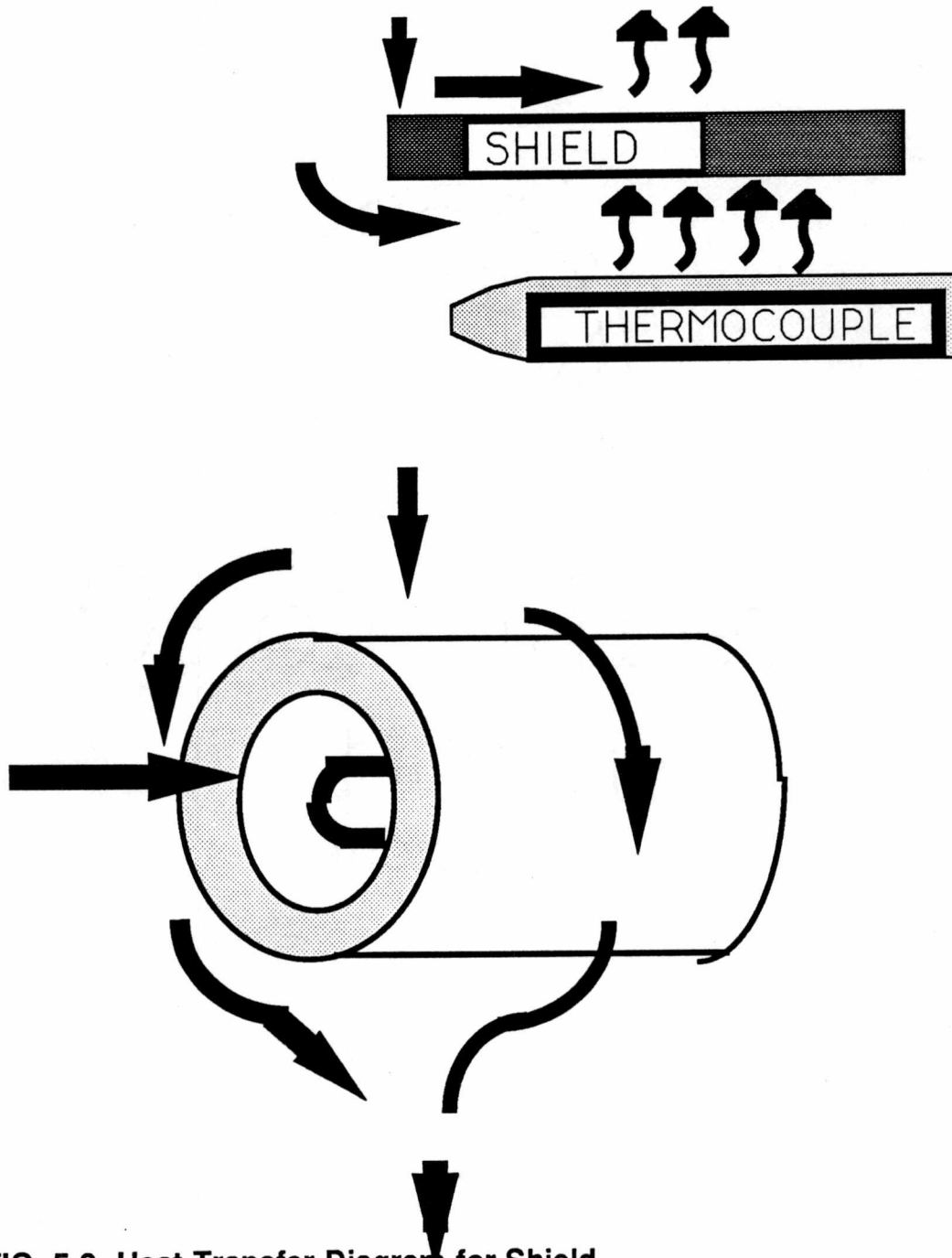


FIG. 5-2 Heat Transfer Diagram for Shield

$$\begin{aligned}
 q = & h_{sg} A_{sg} (T_g - T_s) + h_{so} A_{so} (T_g - T_s) + A_{so} \sigma \epsilon_g F_{2-3} T_g^4 - \epsilon_s A_{so} \sigma F_{2-3} T_s^4 \\
 & + \epsilon_{tc} \sigma A_{tc} F_{1-2} T_{tc}^4 - \epsilon_s \sigma A_{st} F_{2-1} T_s^4
 \end{aligned} \quad (\text{Eq. 5-2})$$

Eq. 5-2 neglects radiation heat transfer from the gas to the thermocouple in the space between the thermocouple and the radiation shield.

Now to solve Eq. 5-2 all the variables must be known or determined. In particular, the surface coefficients for heat transfer must be calculated from appropriate convective Nusselt numbers or their radiative approximations. The Nusselt numbers are, in turn, dependent upon the Reynolds numbers of the flow over the shield and thermocouple. Reynolds number, Re , is defined as:

$$Re = \frac{\rho V D_H}{\mu} \quad (\text{Eq. 5-3})$$

In this thesis, for convenience, an approximate formula for μ has been used:

$$\mu = 5.6 \times 10^{-8} \sqrt{MW} (T)^{0.6}$$

For the average molecular weight = 30.3 and temperature = 2860°R:

$$\mu = 3.653667 \times 10^{-5} \frac{\text{lbm}}{\text{ft sec}}$$

The gas density is computed from: $P = \rho RT$

$$\text{Where, } R = \frac{\bar{R}}{MW} = \frac{1545 \frac{\text{lb} \cdot \text{ft}}{\text{lb} \cdot \text{mole} \cdot \text{R}}}{30.3 \frac{\text{lbm}}{\text{lb} \cdot \text{mole}}} = 50.99 \frac{\text{ft} \cdot \text{lb}}{\text{lbm} \cdot \text{R}}$$

$$\text{Hence, } \rho = \frac{P}{mRT} = \frac{14 \frac{\text{lb}}{\text{inch}^2} (144 \frac{\text{inch}^2}{\text{ft}^2})}{(50.99 \frac{\text{ft} \cdot \text{lb}}{\text{lbm} \cdot \text{R}})(2760 \text{ R})} = 0.01433 \frac{\text{lbm}}{\text{ft}^3}$$

The gas velocity assumed for a typical case is 53.87 ft/sec [15] in the vicinity of the non-aspirated thermocouples, and the diameter of the thermocouple is $D_H = .25/12$ ft. See Annex D. Because the density and the gas constant are dependent upon the conditions in the Secondary Combustor, both ρ and MW can vary. For ρ , the value can be altered by increasing the pressure, or altering the temperature, or changing MW . However, assuming an increased pressure to 14.5 psia and dropping the temperature by 100 degrees, changes the value for ρ from 0.01433 to 0.0154 lbm/ft³; which would not be a large change for the Reynolds Number. Thus, the previously computed values

[15] Personal conversation with J.J.Lee of the ECP Section of the University of Tennessee Space Institute.

are used in the analysis of the heat transfer, and the results should be representative of general conditions.

The gas thermal conductivity, k , was determined from the B&W Steam book [16] and adjusted to apply to the CFFF exhaust gas by adding the individual molecular weights, multiplied by individual species values for k , adjusted for temperature. Another conductivity was estimated from the Prandtl number ($\mu C_p/k$). The two gas conductivities were 0.0426 and 0.047 Btu/(ft-hr-R), so they are numerically close. The Prandtl number was assumed to be 0.72, and C_p of the gas was 0.2581 Btu/(lbm R).

Substituting the appropriate values back into equation 5-3 for the Reynolds Number; Re_H was computed to be 9505, $Re_L = 38,019$ which indicates that the aspirated gas flow over the thermocouple and inside the shield was turbulent. Similar calculations for the Re of the flow over the outside of the shield showed that it too was turbulent flow. Taking Eq. 5-2 and solving for T_g :

$$(h_{si}A_s + h_{so}A_{so})(T_g - T_s) = -A_{so}\sigma F_{2-3}(\epsilon_g T_g^4 - \epsilon_s T_s^4) - \sigma F_{2-1}(A_{tc}\epsilon_{tc}T_{tc}^4 - A_{si}\epsilon_s T_s^4)$$

$$T_g = T_s - \frac{A_{so}\sigma F_{2-3}(\epsilon_g T_g^4 - \epsilon_s T_s^4) + \sigma F_{2-1}\epsilon_{tc}(A_{tc}T_{tc}^4 - A_{si}T_s^4)}{h_{si}A_{si} + h_{so}A_{so}}$$

[16] Steam, 39th Ed. G.H.Babcock and S.Wilcox. (New York, N.Y.: Babcock & Wilcox, 1978)

Using the values determined in Annex F and the specified areas:

$$h_{so} = 8.92 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ\text{R}) \quad \epsilon_g = 0.18 \text{ (Initial guess)}$$

$$h_{si} = 71.65 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ\text{R}) \quad \epsilon_{ic} = \epsilon_s = 0.45$$

$$A_{so} = 9.8 \times 10^{-2} \text{ ft}^2 \quad F_{1-2} = F_{2-3} = F_{3-2} = 1.0$$

$$A_{si} = 6.54 \times 10^{-2} \text{ ft}^2 \quad F_{2-1} = 0.16.$$

$$T_g = T_s + \frac{9.8 \times 10^{-2}(1.0)(0.1713 \times 10^{-8})(\epsilon_g T_g^4 - 0.45(2690)^4)}{71.65(6.54 \times 10^{-2}) + 8.92 \times 10^{-2}(9.8 \times 10^{-2})}$$

$$+ \frac{0.1713 \times 10^{-8}[0.45(8.2 \times 10^{-3})(2690^4) - 0.16(6.54 \times 10^{-2})2490^4]}{71.65(6.54 \times 10^{-2}) + 8.92 \times 10^{-2}(9.8 \times 10^{-2})}$$

Assuming $\epsilon_g = .18$ and simplifying: $T_g = T_s - 5.4372 \times 10^{-12} T_g^4 + 459.5$

Table 5-1 reflects iteration with $T_s = 2490$ and an initial assumption of $T_g = 2800^\circ\text{R}$. By iteration, it can be seen that the emissivity of the gas and the temperature can be adjusted to find an equilibrium. The flue gas temperature measured by the HVT probe is around 2800°R for the values used in the equation. Note in Table 5-1 that as the emissivity goes up, the flue gas temperature comes down. For an emissivity of 0.1 and assuming $T_g = 2800^\circ\text{R}$,

the gas temperature is calculated to be 2763.8, which is close to experimental data.

Table 5-1: Solving for T_g by Iteration (Shield).

ϵ	T_g Assumed	T_g Calculated
.18	2800	2615.3
.18	2700	2660.5
.18	2680	2669.0
.18	2675	2671.1
.18	2673	2671.9
.4	2800	2206.8
.4	2700	2307.4
.4	2500	2477.5
.4	2485	2488.7
.8	2800	1464
.8	2200	2383.4
.8	2300	22273.3

5.B.(3)(b) Evaluation of the optically thick environment as an energy balance on the thermocouple.

$$\begin{aligned}
 q_{net} = & h_{tip\ tc} A_{tip\ tc} (T_g - T_{tc}) + h_{shaft\ tc} A_{shaft\ tc} (T_g - T_s) \\
 & + \epsilon_g \sigma A_{tip\ tc} F_{3-1} T_g^4 - \epsilon_{tc} \sigma A_{tc} F_{1-3} T_{tc}^4 \\
 & + \epsilon_s \sigma A_{si} F_{2-1} T_s^4 - \epsilon_{tc} \sigma A_{tc} F_{1-2} T_{tc}^4
 \end{aligned}$$

$$h_{tip\ tc} = 0.023 \text{ Btu}/(\text{ft}^2 \text{ s } ^\circ\text{R}) = 81.25 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ\text{R})$$

$$A_{tip\ tc} = 6/814 \times 10^{-4} \text{ ft}^2$$

$$h_{shell\ tc} = 70.04 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ\text{R})$$

$$A_{s\ tc} = 1.621 \times 10^{-2} \text{ ft}^2$$

$$\epsilon_g = 0.18$$

$$A_{si} = 6.54 \times 10^{-2} \text{ ft}^2$$

$$\epsilon_{tc} = 0.45$$

$$F_{1-2} = 0.654$$

$$F_{2-1} = 0.16$$

$$\epsilon_s = 0.45$$

$$F_{1-3} = 0.80$$

$$F_{3-1} = 0.05$$

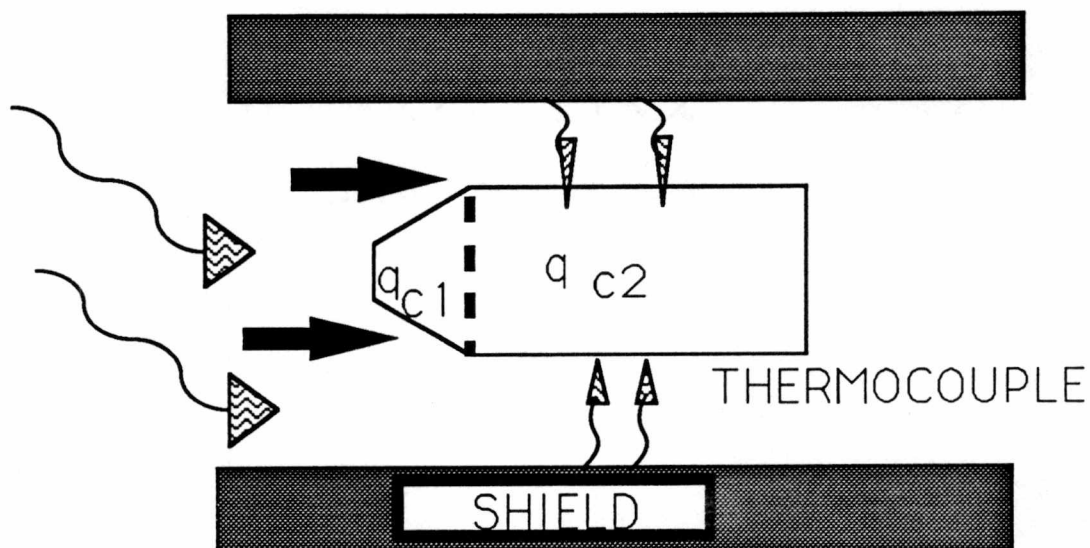


FIG. 5-3 Heat Transfer Diagram for Thermocouple

Substituting in the values and simplifying yields the following:

$$\begin{aligned}
 q_{net} = & (81.25)(6.814 \times 10^{-4})(T_g - T_{tc}) + (70.04)(1621 \times 10^{-2})(T_g - T_{tc}) \\
 & + (0.18)(.1713 \times 10^{-8})(6.814 \times 10^{-4})(0.05)T_g^4 - (0.45)(.1713 \times 10^{-8})(6.814 \times 10^{-4}) \\
 & + (0.45)(.1713 \times 10^{-4})(6.54 \times 10^{-2})(0.16)T_s^4 - (0.45)(.1713 \times 10^{-8})(1.621 \times 10^{-2})
 \end{aligned}$$

$$\begin{aligned}
 T_g - T_{tc} = & 3.5282 \times 10^{-13} T_{tc}^4 + 6.8615 \times 10^{-11} T_{tc}^4 \\
 & - 8.8203 \times 10^{-15} T_g^4 - 6.7726 \times 10^{-12} T_s^4
 \end{aligned}$$

Again using $T_{tc} = 2690^\circ\text{R}$ and $T_s = 2490^\circ\text{R}$:

$$T_g = T_{tc} + 117.4 - 8.8203 \times 10^{-15} T_g^4$$

Table 5-2: Solving for T_g by Iteration (Thermocouple).

ϵ	T_g Assumed	T_g Calculated
.18	3000	2807.4
.18	3100	2806
.18	2800	2806.8
.4	3000	2805.8
.4	2800	2806.2
.6	3000	2805
.6	2800	2805.6

Note that the emissivity of the gas seems to have little effect on the calculated temperature of the gas. This is possibly directly related to the shape factor and the lack of current information to ascertain the emissivity of the gas and, therefore, provide the proper information to determine the percentage of heat that is absorbed by the gas. Also, the values for this iterative method are higher than calculated for the thermal shield energy balance.

For the shield energy balance the predominate terms were the difference between the radiation terms for the shield and the gas. In the thermocouple energy balance the predominate terms are between the thermocouple and the shield. Again, without more research, the view factors are suspect.

In the final analysis, because of assumptions made in determining almost all of the values, a difference of little more than 100°R can be considered minimal compared to the magnitude of 2700 to 2800°R .

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.A. Conclusions.

More testing is needed to evaluate the large data sets of temperatures that have read from the stationary non-aspirated thermocouples within the ducts of the CFFF. Within the Secondary Combustor, there is a need to create calibrated temperature files that are 200° to 400° F higher than the measured values obtained with the non-aspirated thermocouples. The adjustments to the stationary non-aspirated thermocouples will need to be made dependent upon the ash and seed content of the gas in the channel. The necessary calibration can only be obtained with the improved HVT-type temperature probe, and its routine utilization during CFFF testing.

6.B. Recommendations.

In the future, additional analysis of the thermal environment of the thermocouple assembly within the HVT probe should be performed, with the thermocouple treated as a fin problem. But to obtain a better understanding of the difference in temperatures between the thermocouple's reading and that of the actual gas temperature, more thermocouples will have to be installed in the HVT probe to measure: the temperatures of the mullite shield over the thermocouple - along its entire length; the inside wall of the probe - next to the water wall; along the 1/4 inch tube holding the extension wire; and inside the 1/4 inch tube along the extension wire.

Placement will have to be judicious as additional wiring/tubing will close off the space available within the probe and tend to enhance the plugging problems. This will necessitate obtaining several readings with a limited number of additional thermocouples applied to the probe. Additional recommendations are:

- 1) Utilization of FIBERFRAX® should be limited to regions of below 2300°F.

- 2) Recording of temperatures should be done with the computer based data acquisition system at the CFFF. This would allow the recording of the multiple thermocouples in the HVT probe, and also the fluctuations in the recorded temperatures.

- 3) A heat flux probe should be constructed to measure heat flux to the walls of the Secondary Combustor, as a means to provide an alternate heat balance equation.

- 4) More HVT probes, in improved form, should be built and used in future testing. Improved form means additional thermocouples emplaced to measure the temperature degradation down the length of the probe, with a calibrated air-aspiration system for each to establish proper aspiration flow rates.

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ANNEXES

**ANNEX A:
EXPERIMENTAL RESULTS -
MHD TESTING.**

NOTE:

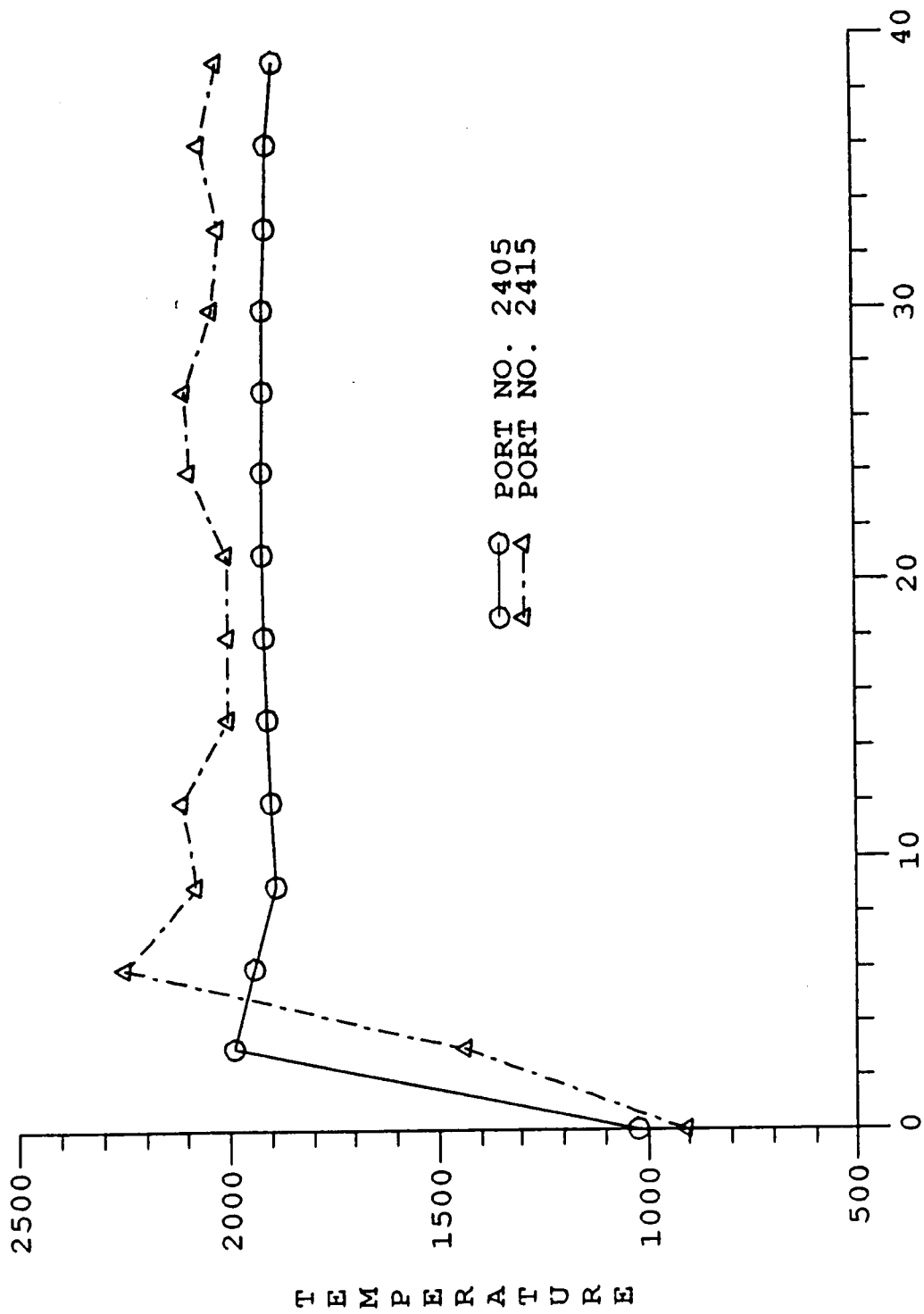
- (1) The following tables reflect raw unprocessed data taken during scheduled CFFF testing.
- (2) All readings in this annex were from the Secondary Combustor or Test Section One of the Superheater.
- (3) Port Number correspondence can be seen in Fig. 1-3.
- (4) Distance refers to the number of inches the HVT probe tip is from the inside wall.
- (5) All Temperatures are in **Fahrenheit**.
- (6) All test data was taken while burning type Illinois #6 coal.
- (7) Dates of temperature sampling:

LMF4-O	17-30	AUG '88
LMF4-P	10-12	NOV '88
LMF4-Q	26-31	MAY '89
LMF4-R	17-30	AUG '89
LMF4-S	3-4	JAN '90

TABLE I: MHD TEST AUG. 1988 RESULTS

TEST LMF4-O				
DIST. FROM WALL	TEMPERATURE (FAHRENHEIT)			
	PORT NO. 2405	PORT NO. 2410	PORT NO. 2503	PORT NO. 3103
0	1022	909		1094
3	1989	1436	2210	2203
6	1940		2251	
9	1888	2080		
12	1899	2113		
15	1906	2001		
18	1915	2001		
21	1918	2006		
24	1918	2093		
27	1914	2100		
30	1915	2034		
33	1906	2019		
36	1904	2062		
39	1886	2023		

TEST LMF4-O (CONTINUED)		
DIST. FROM WALL (INCHES)	TEMPERATURE (FAHRENHEIT)	
	PORT NO. 2405	PORT NO. 2415
0		1882
3	2376	2289
6		2354
9		2367

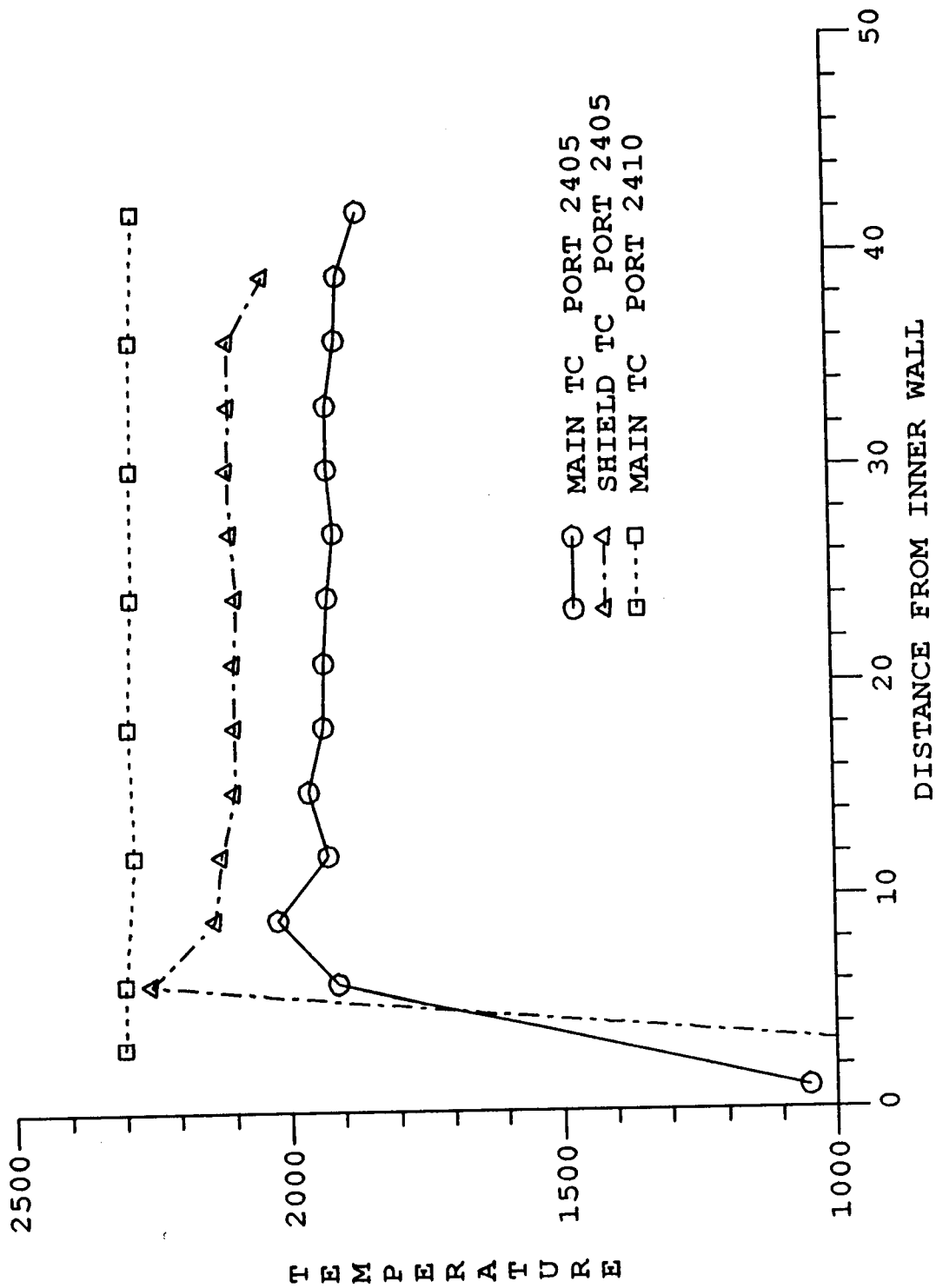


DISTANCE FROM INSIDE WALL

LMF4-O TEST AUG. 1988

TABLE II: MHD TEST NOV. 1988 RESULTS

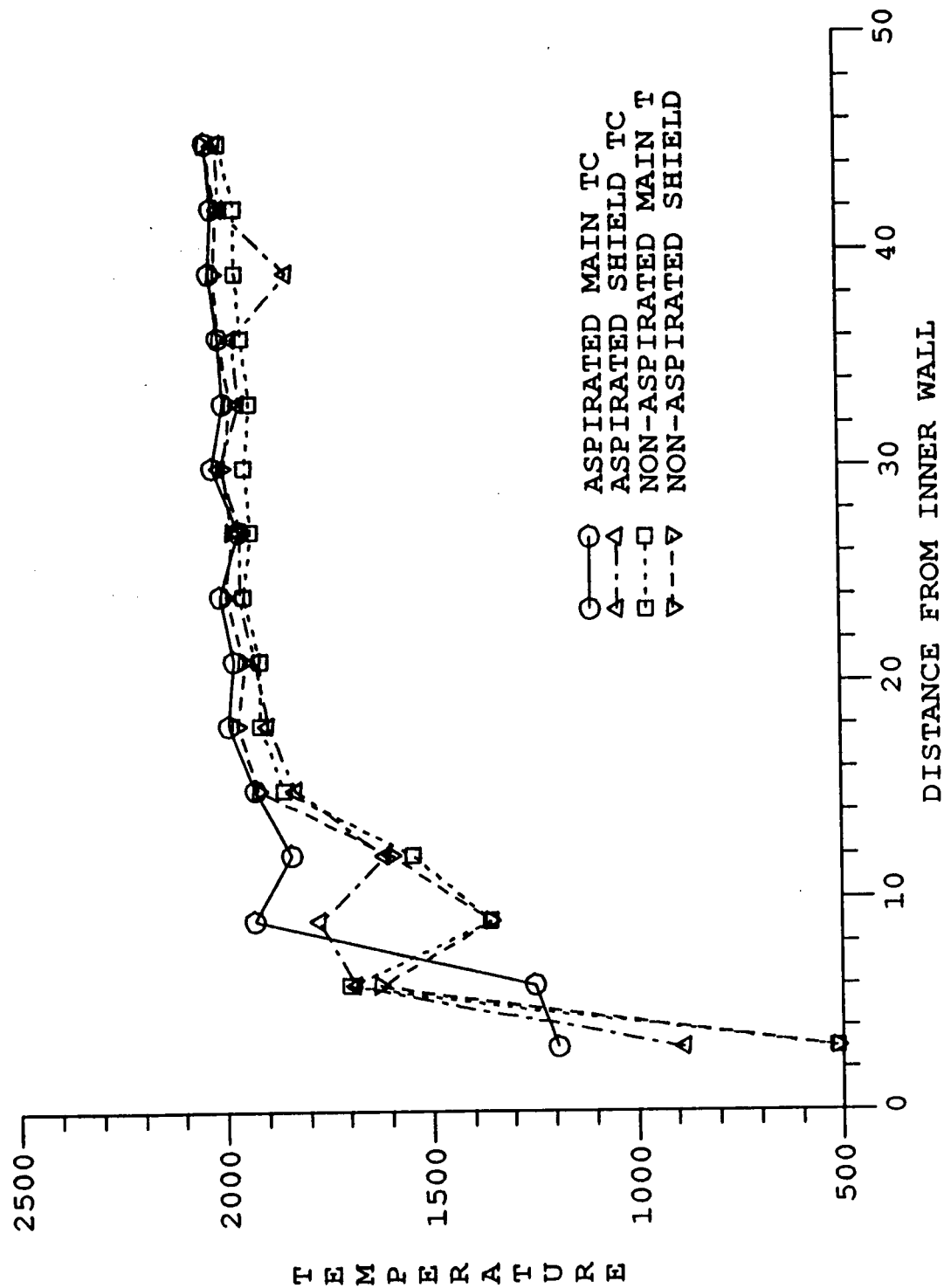
TEST LMF4-P			
DIST. FROM WALL (INCHES)	TEMPERATURE (FAHRENHEIT)		
	PORT NO. 2405 MAIN	SHIELD	PORT NO. 2410
0	1050		
3			2303
6	2165	1911	2300
9	2254	2021	
12	2135	1928	2282
15	2123	1960	
18	2097	1933	2289
21	2094	1929	
24	2095	1922	2282
27	2089	1910	
30	2097	1919	2280
33	2101	1918	
36	2097	1901	2278
39	2096	1895	
42	2033	1857	2269



LMF4-P DUAL TC PORT 2405; SINGLE TC PORT 2410

TABLE III: MHD TEST MAY 1989 RESULTS

TEST LMF4-Q PORT NO. 3110 TEMPERATURES IN FAHRENHEIT [DUAL THERMOCOUPLE CONFIGURATION - MAINSTREAM & SHIELD]				
DIST. FROM WALL	ASPIRATED		NON-ASPIRATED	
	MAIN	SHIELD	MAIN	SHIELD
3	1193	887	514	514
6	1250	1688	1702	1630
9	1933	1776	1359	1357
12	1841	1616	1544	1598
15	1927	1832	1859	1922
18	1989	1895	1913	1969
21	1976	1927	1913	1953
24	2007	1963	1952	1994
27	1963	1960	1936	1981
30	2026	2008	1947	2001
33	1998	1960	1935	1994
36	2008	1978	1953	2008
39	2030	1841	1967	2017
42	2023	2003	1965	2010
45	2034	2010	2005	2039

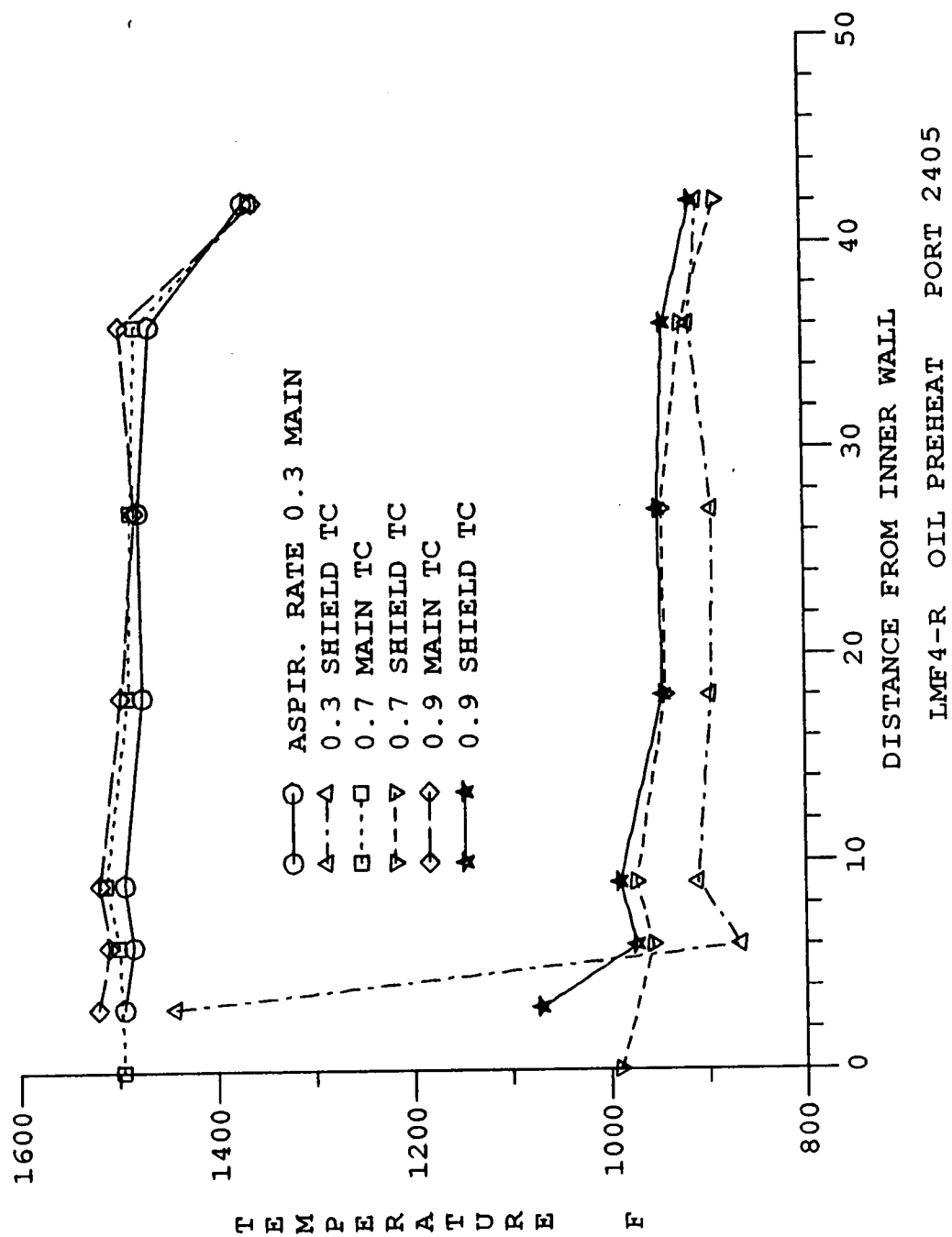


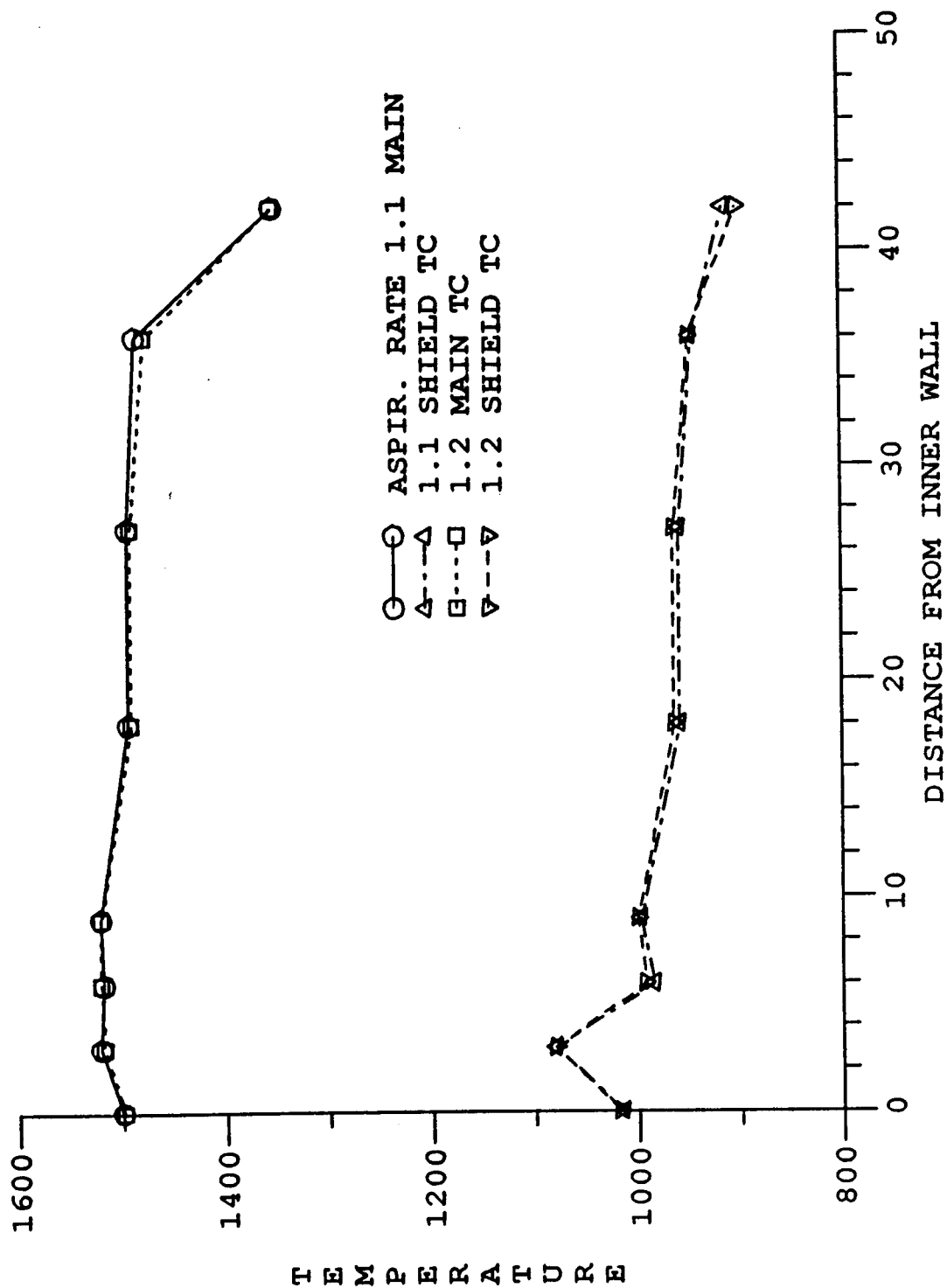
LMF4-Q PORT 3103 DUAL TC CONFIGURATION

TABLE IV: MHD TEST AUG. 1989 RESULTS

TEST LMF4-R PORT NO. 2405 TEMPERATURE IN FAHRENHEIT [DURING OIL PREHEAT W/DIFFERENT ASPIRATION RATES]						
DIST FROM WALL	ASPIRATION RATE IN INCHES H2O					
	0.3		0.7		0.9 - 1.0	
	MAIN	SHIELD	MAIN	SHIELD	MAIN	SHIELD
0			1497	990		
3	1495	1443			1522	1071
6	1485	867	1501	957	1510	973
9	1494	910	1513	973	1519	990
18	1474	896	1490	943	1497	946
27	1477	894	1485	945	1481	949
36	1465	916	1479	925	1495	943
42	1369	907	1362	889	1359	914

DIST FROM WALL	[LMF4-R PORT NO. 2405 CONTINUED]			
	ASPIRATION RATE IN INCHES H2O			
	1.1		1.2	
	MAIN	SHIELD	MAIN	SHIELD
0	1501	1015	1499	1018
3	1522	1080	1519	1080
6	1519	984	1521	993
9	1521	997	1522	999
18	1494	954	1495	963
27	1494	958	1491	964
36	1485	945	1477	948
42	1351	914	1353	901





LMF4-R OIL PREHEAT PORT 2405

TABLE V: MHD TEST AUG. 1989 RESULTS

TEST LMF4-R DURING COAL COMBUSTION PORT NO. 2405 [COOLED PROBE W/ COLD WATER & PROBE OPENING CLOGGED]					
DIST FROM WALL	ASPIRATION RATE IN INCHES H2O				
	0	0.3	0.7	0.9 - 1.0	1.1
0	1807	1909	1913	1917	1913
3	1810				
6	1805				
9	1801				
12	1789				
15	1787				
18	1787				
21	1783				
24	1776				
27	1767				

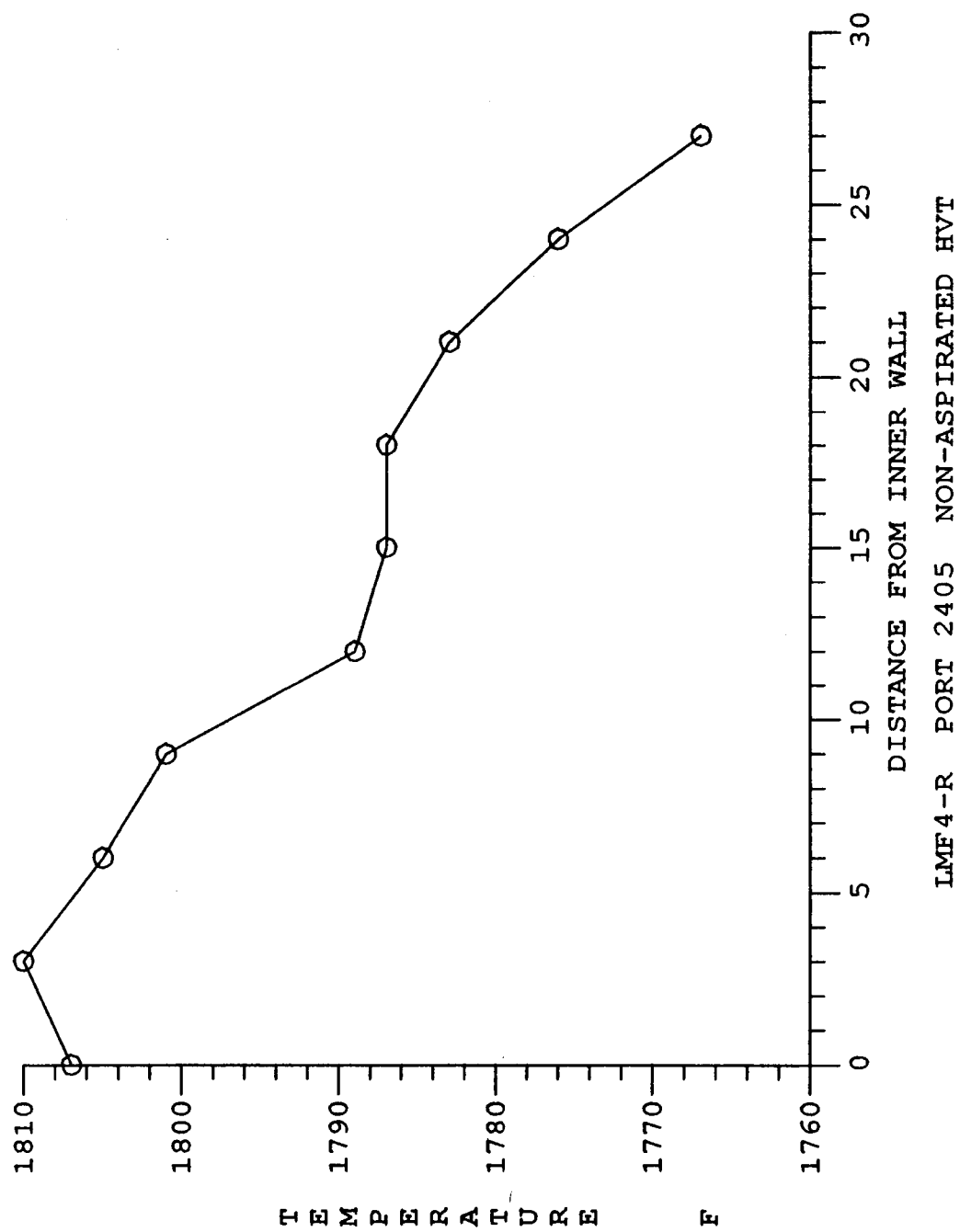
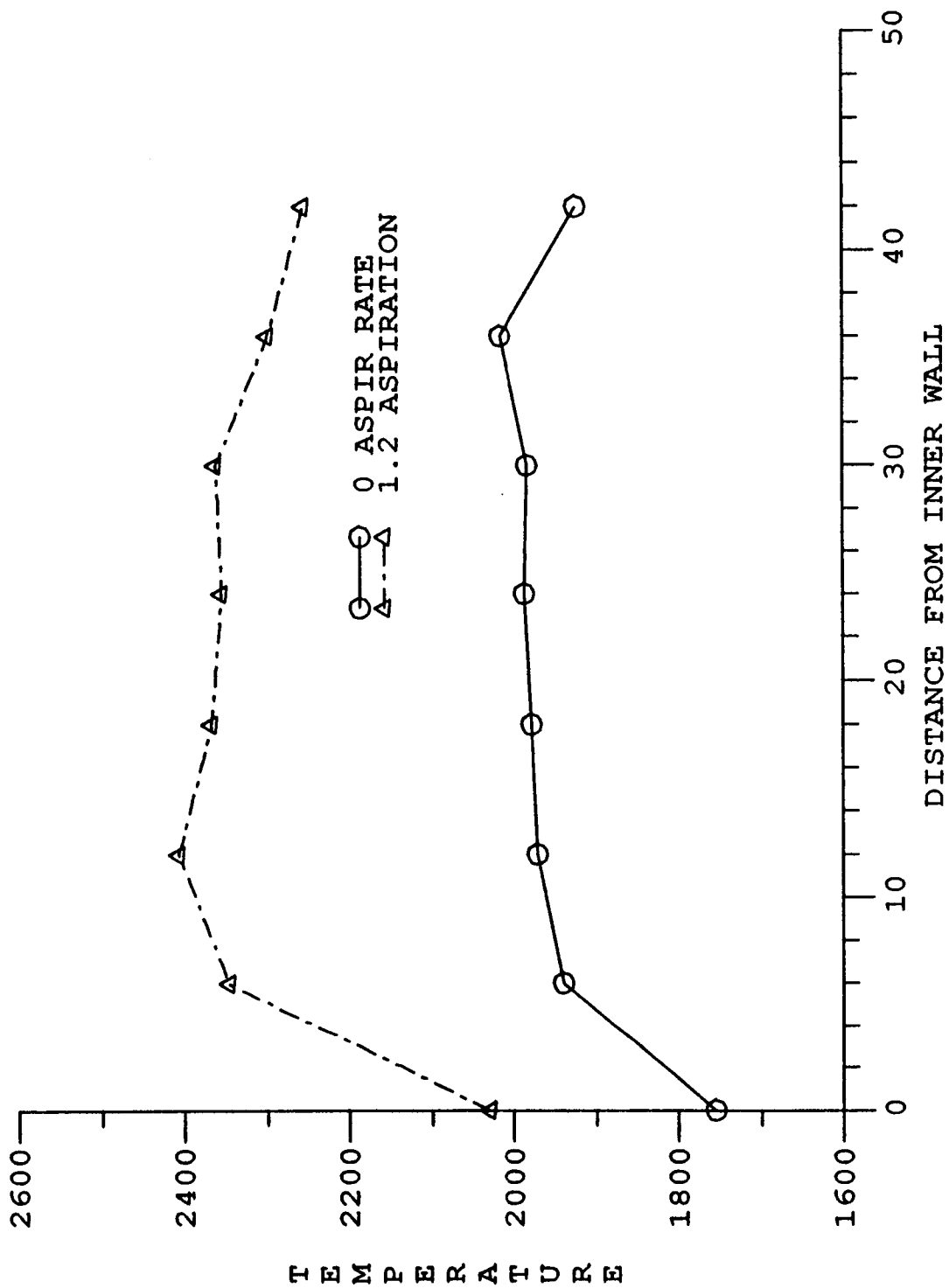


TABLE VI: MHD TEST AUG. 1989 RESULTS

TEST LMF4-R PORT NO. 2503 [READINGS WERE OBTAINED INITIALLY W/ HIGHEST ASPIRATION PRESSURE, WHICH BECAME REDUCED AS PROBE PLUGGED.] TEMPERATURE IN FAHRENHEIT					
DIST FROM WALL	ASPIRATION RATE IN INCHES H2O				
	0	0.7	0.8 - 0.9	1.0 - 1.1	1.2 - 1.3
0	1756				2030
6	1940		2321		2347
12	1972	2282	2340		2408
18	1978			2350	2368
24	1987	2323	2368		2356
30	1983			2327	2363
36	2016			2300	2300
42	1924			2237	2257

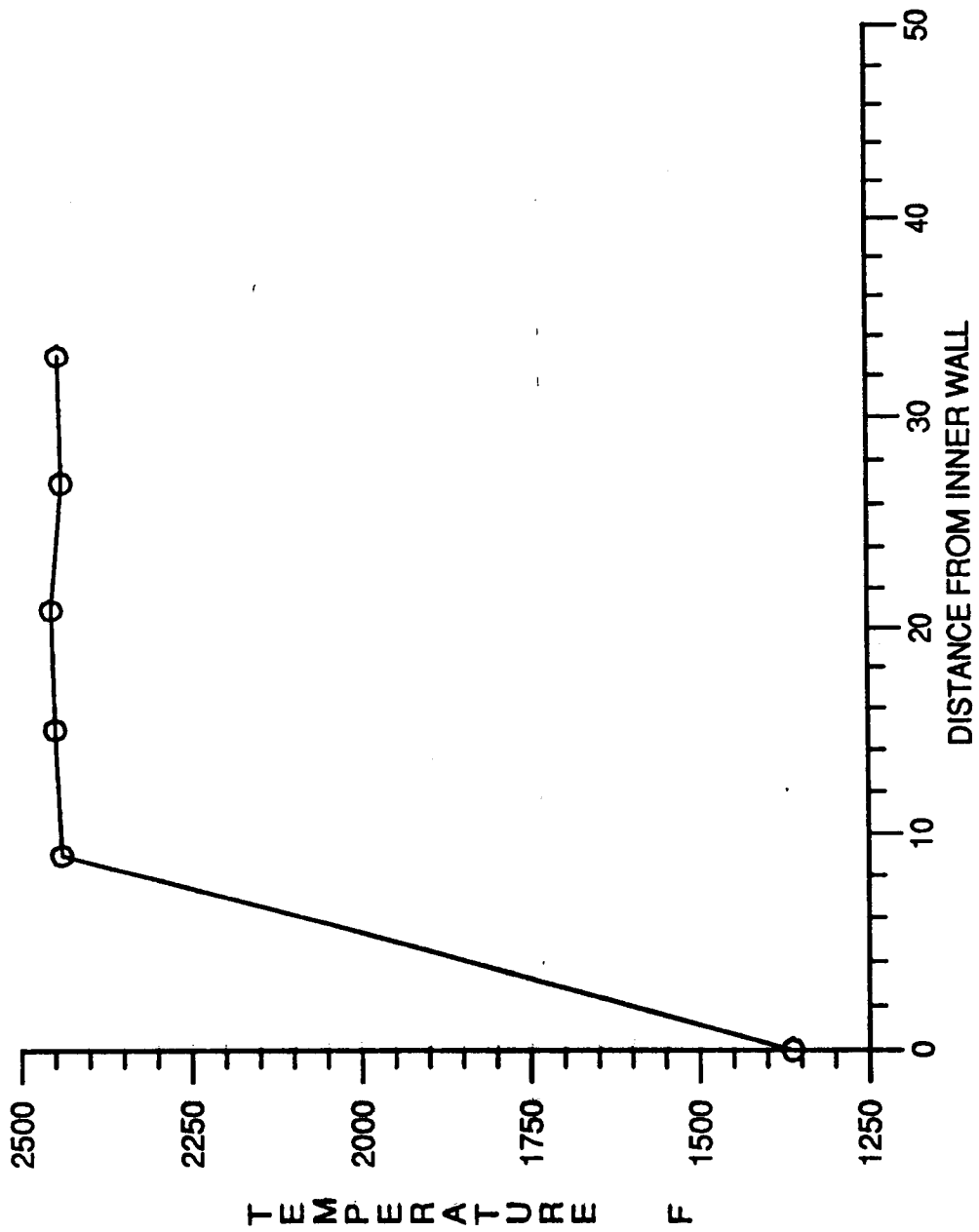


LMF4-R PORT 2503

TABLE VII: MHD TEST AUG. 1989 RESULTS

TEST LMF4-R PORT NO. 3103 TEMPERATURE IN FAHRENHEIT						
DIST. FROM WALL	ASPIRATION RATE IN INCHES H2O					
	0.7	0.75	0.8	0.9	1.0	1.2
0						1364 [I]
3	972 [W]					
6					2179 [I]	
9	2440 [W]					
12				2394 [I]		
15	2448 [W]					
18			2399 [I]			
21		2453 [W]				
24			2426 [I]			
27		2439 [W]				
30			2420 [I]			
33		2442 [W]				
36		2419 [I]				

NOTE: [I] Indicates temperature taken while probe was being inserted.
[W] Indicates temperature taken while probe was being withdrawn.



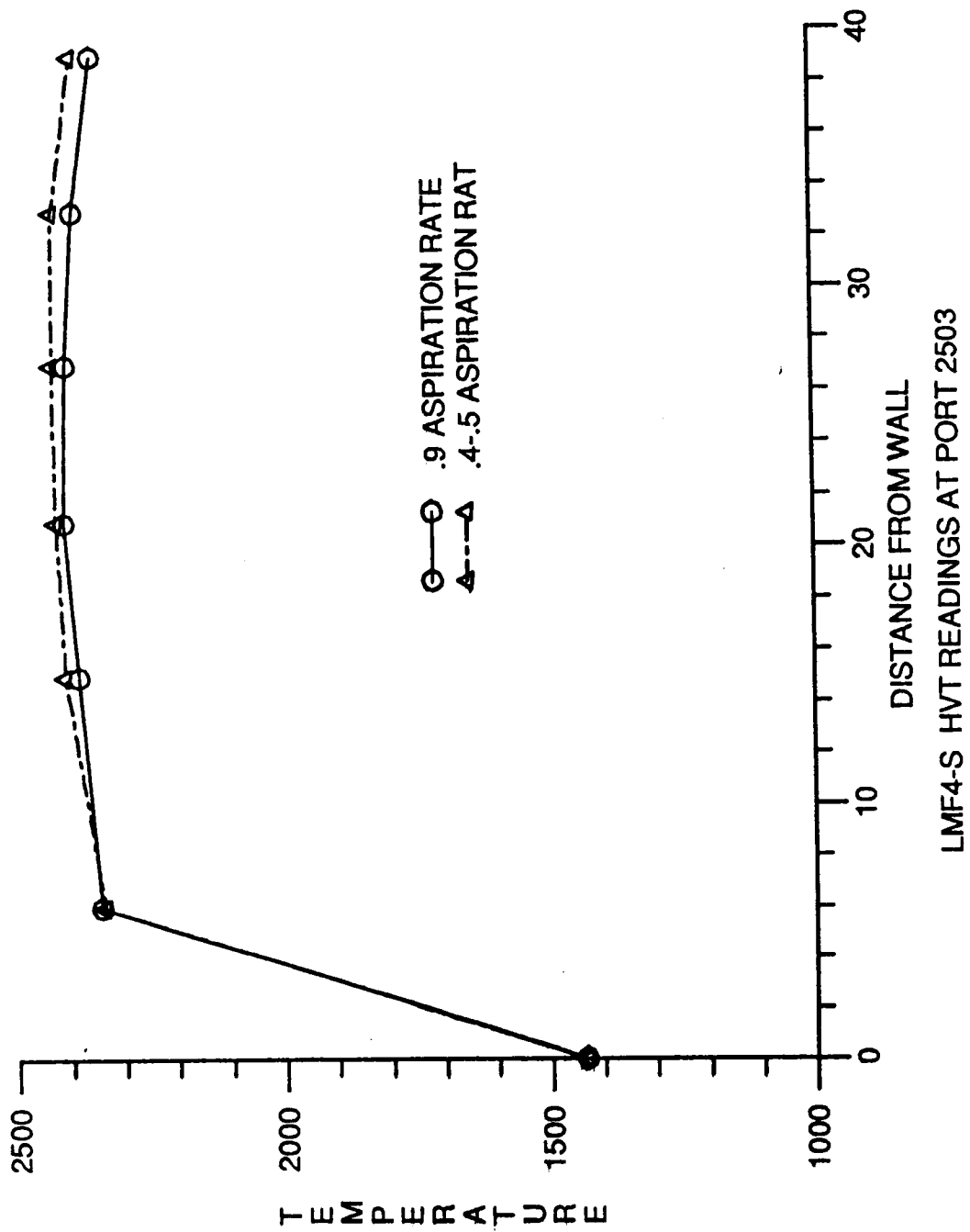
LMF4-R PORT NO. 3110

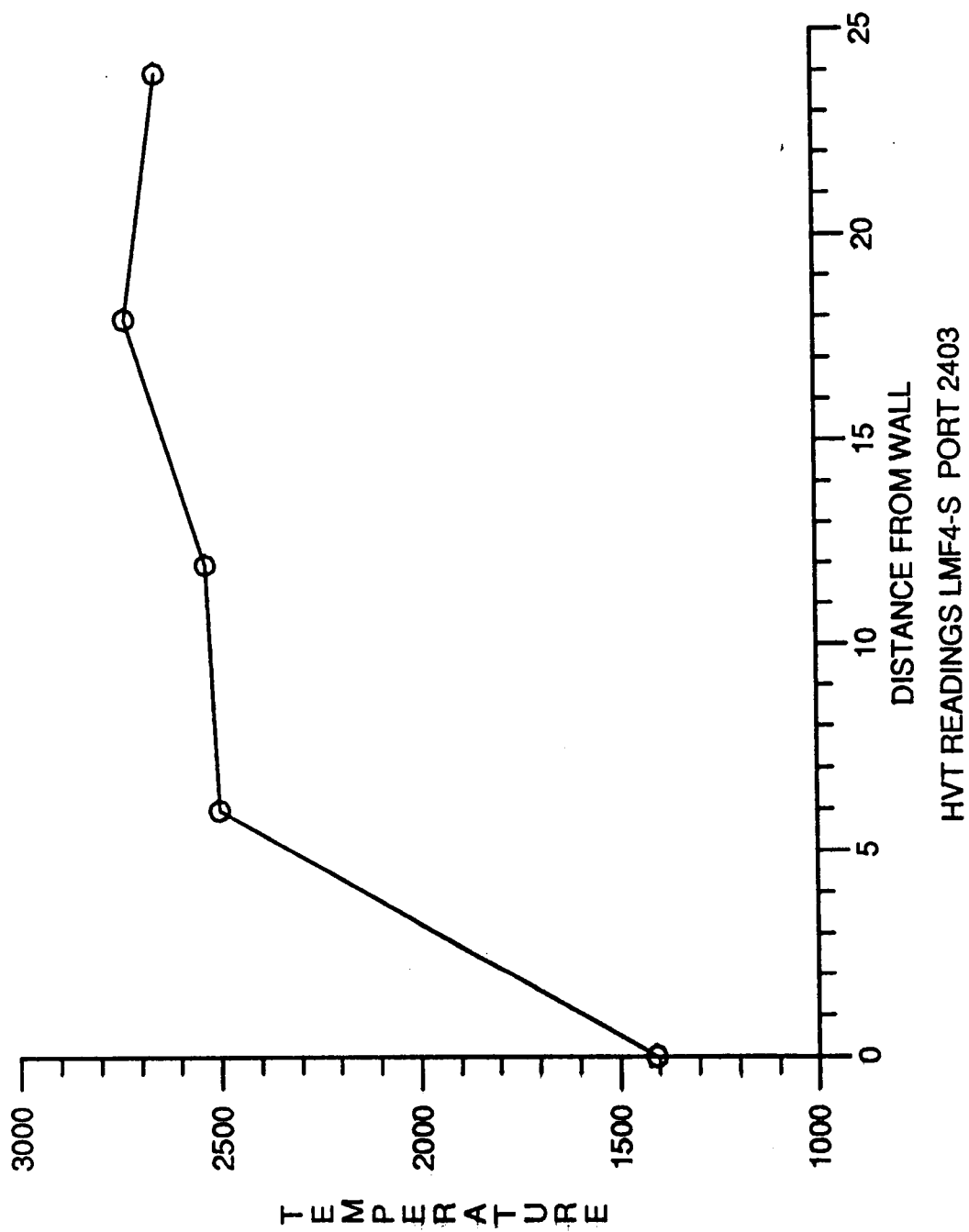
TABLE VIII: MHD TEST NOV. 1989 RESULTS

TEST LMF4-S PORT NO. 2503 TEMPERATURE IN FAHRENHEIT				
DIST FROM WALL	ASPIRATION RATE IN INCHES H2O			
	0	0.15	0.4-0.5	0.9-0.95
0	1435			2035
6	1415			2340
9			2345	
15	1928		2383	2411
21			2410	2425
27			2405	2432
33		2388		2428
39				2391
42		2352		

TABLE IX: MHD TEST NOV. 1989 RESULTS

TEST NO. LMF4-S PORT NO. 2403 TEMPERATURE IN FAHRENHEIT					
DIST FROM WALL	ASPIRATION RATE IN INCHES H2O				
	0	0.3	0.4	0.5	0.6
0	1408	2117			2385
6				2500	
12			2533		
18				2730	
24		2568	2650		





ANNEX B

TABLE I: ADVANCED COMBUSTOR DATA

Conducted burning micronized coal - May 26, 1989 (At the door w/ zero aspiration the Main read 2048°F and the wall thermocouple read 1985°F.						
Dist inches	Aspiration rate: .7 in. H2O Temp. (F)		Aspiration rate: .4 in. H2O Temp. (F)		Aspiration rate: .2 in. H2O Temp. (F)	
	Main	Shield	Main	Shield	Main	Shield
Door	2228	2122	2201	2097	2181	2046
6	2321	2140	2224	2120	2194	2071
12	2289	2161	2255	2124	2203	2066
18	2280	2136	2244	2109	2217	2077
24	2322	2160	2293	2142	2264	2113
30	2309	2147	2278	2133	2268	2115

Readings from the stationary thermocouple in the vicinity of the door location varied from 1875-1888°F; with the majority of readings running from 1885 to 1888°F.

ANNEX C

TABLE I: NON-ASPIRATED OVEN TESTS

Linberg Solar Basic Oven Oct.5, 1989 Temperature in Fahrenheit		
Stove Reading	Sheathed	Unsheathed
1310	1224	1206
1652	1571	1576
1697	1616	1625
1742	1663	1672
1787	1711	1720
1832	1755	1767
1877	1805	1814
1922	1834	1848

NOTE: Test was conducted with two S-type thermocouples. Readings for both thermocouples are lower than the stove ratings due to the opening made to insert the thermocouples. With the exception of the first reading note the quasi-constant difference caused by the "conduction error" of the mullite sheath. These readings were stagnant with no aspiration to account for.

ANNEX D

**TABLE I: VELOCITY OF GAS STREAM VS DISTANCE SOUTH TO NORTH
AT PORT 2405 ABOVE THE SECONDARY COMBUSTOR**

DISTANCE FROM NORTH WALL (UNIT 1 = 2.4 INCHES)	VELOCITY (FT/SEC)
1	0
2	35.06
3	38.14
4	39.94
5	41.65
6	43.33
7	44.97
8	46.54
9	48.02
10	49.38
11	50.60
12	51.68
13	52.60
14	53.33
15	53.87
16	54.18
17	54.21
18	53.86
19	52.81
20	48.83
21	0

Annex E

E.1. Derivation of Eq. 3-26.

Consider the ratio: $\frac{T^4 - T_w^4}{T - T_w}$

By factoring,

$$T^4 - T_w^4 = (T - T_w) (T^3 + T^2 T_w + T T_w^2 + T_w^3)$$

$$\frac{T^4 - T_w^4}{T - T_w} = T^3 + T^2 T_w + T T_w^2 + T_w^3$$

Now assume that, $T_w - T \gg T$;

hence $T_w - T = \epsilon T_w$

or, $T_w (1 - \epsilon) = T$.

Therefore, by substitution $T^3 = T_w^3 (1 - \epsilon)^3 = T_w^3 (1 - 3\epsilon + \dots)$

and $T^2 T_w^2 = T_w^2 (1 - \epsilon)^2 T_w = T_w^3 (1 - 2\epsilon - \dots)$

and $T T_w^2 = T_w (1 - \epsilon) T_w^2 = T_w^3 (1 - \epsilon) \dots$

Therefore,

$$T^3 + T^2 T_w + T T_w^2 + T_w^3 = T_w^3 (4 + (-3\epsilon - 2\epsilon - \dots$$

$$\sim 4T_w^3$$

E.2. Derivation of Shape Factor.

For shape factor - $F_{1-2} = \int_{A_1} \int_{A_2} \frac{\cos \theta_1 \cos \theta_2}{A_1 \pi s^3} dA_1 dA_2$

If $A_1 = da_1$ then: $F_{1-2} = \int_{A_2} \frac{\cos \theta_1 \cos \theta_2}{\pi s^2} dA_2$

Since, $\sum F = F_{1-1} + F_{1-2} + \dots + F_{1-n} = 1$

then, for two interacting surfaces, $F_{12}A_1 = F_{21}A_2$

For convex and flat surfaces: $F_{1-1} = 0$

For parallel plates: $A_1/A_2 = 1$; and for concentric cylinders

$A_1/A_2 = d_1/d_2 = r_1/r_2$:

$$Q = \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{\epsilon_1} + \frac{1}{F_{12}} + \frac{1-\epsilon_2}{\epsilon_2} \frac{A_1}{A_2}}$$

For heat transfer between two grey bodies with a third enclosing refractory surface:

$$Q = \frac{\sigma A_1 (T_1^4 - T_2^4)}{\left(\frac{1}{\epsilon_1} - 1\right) + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1\right) + \frac{A_1 + A_2 - A_1 F_{12}}{A_2 - A_1 (F_{12})^2}}$$

ANNEX F

NUMERICAL CALCULATIONS

F.1 Derivation of Heat Equation for an Optically Thin Environment.

As previously given,

(Eq. F-1)

$$q = \epsilon_s \sigma A_s (T_s^4 - T_w^4) =$$
$$2h_s A_s (T_g - T_s) + \epsilon_g \sigma A_s (T_g^4 - T_s^4) + \epsilon_c \sigma A_{tc} (T_{tc}^4 - T_s^4)$$

In the above equation, a simplification can be made by substituting for the effects of thermocouple radiation in the general terms of the convection terms and the same power factor. Let:

$$\epsilon_s \sigma A_s (T_s^4 - T_w^4) = h_{rs} A_s (T_s - T_w)$$

$$\epsilon_c \sigma A_{tc} (T_{tc}^4 - T_s^4) = h_{rtc} A_{tc} (T_{tc} - T_s)$$

$$\epsilon_g \sigma A_s (T_g^4 - T_s^4) = h_{rg} A_s (T_g - T_s)$$

Using these alternate representations in Eq. F-1 gives:

$$2h_s A_s (T_g - T_s) + h_{rtc} A_{tc} (T_{tc} - T_s) + h_{rg} A_s (T_g - T_s) = h_{rs} A_s (T_s - T_w)$$

$$(T_g - T_s) A_s (2h_s + h_{rg}) = h_{rs} A_s (T_s - T_w) - h_{rtc} A_{tc} (T_{tc} - T_s)$$

$$T_g = \frac{h_{rs} A_s (T_s - T_w) - h_{rtc} A_{tc} (T_{tc} - T_s)}{(2h_s + h_{rg})A_s} + T_s$$

From the above equation it can be seen that the gas temperature is going to be some increment in temperature greater than that of the thermal shield (which was measured to be about 100°F below that of the thermocouple). Since $A_{\text{shield}} \gg A_{\text{thermocouple}}$, by almost an order of magnitude, and since $T_s \gg T_w$ and $T_{tc} \approx T_s$, it is seen that the temperature difference is dependent upon the various effective heat transfer coefficients, and the difference between the inner channel wall temperature and that of the thermocouple shield. Equating $h_{rs} = h_{tc}$ and substituting in values yields:

$$T_g = T_s + \frac{29.1126 [.065 (2690 - 1460) - .016 (2940 - 2690)]}{.065 [2 (4.2) + 29.1126]}$$

$$T_g = T_s + 906$$

F.2. Derivations of Heat Transfer Coefficients.

F.2.a. Surface Coefficient for Flow over Shield.

Known parameters:	Gas Velocity	$V_g = 52 \text{ ft/s}$
	Gas Viscosity	$\mu = 3.65 \times 10^{-5} \text{ lbm/ft-s}$
	Gas Density	$\rho = 0.01433 \text{ lbm/ft}^3$
	Diameter of Shield	$D = 1.5"/12 = 0.125 \text{ ft}$
Assumed parameter:	Gas Temperature	$T_g = 3040^\circ\text{R}$

$$Re_D = (0.01433)(52(0.125))/3.65 \times 10^{-5} = 2552$$

$$A_{so} = \pi D_o L = \frac{3.14(1.5)(3.0)}{144} = 9.81 \times 10^{-2} \text{ ft}^2$$

$$N_D = C Re_D^m Pr^{\frac{1}{3}} = 0.683(2552)^{0.466} (0.72)^{\frac{1}{3}} = 23.69$$

$$h_s = \frac{N_D k}{D} = \frac{N_D (\mu Cp)}{D Pr} = \frac{23.69(3.65 \times 10^{-5})(0.258)}{0.125(0.72)}$$

$$h_s = 2.48 \times 10^{-3} \frac{\text{Btu}}{\text{ft}^2 \text{ s } ^\circ \text{R}} = 8.92 \frac{\text{Btu}}{\text{ft}^2 \text{ hr } ^\circ \text{R}}$$

F.2.b. Surface Coefficient for Flow Inside Shield.

$$\text{Flow Area} = \frac{\pi}{4} (D_o^2 - D_i^2) = \frac{3.14}{4(144)} (1.0^2 - 0.25^2) = 5.11 \times 10^{-3} \text{ ft}^2$$

$$\dot{m}'' = 20,000 \frac{\text{lbm}}{\text{ft}^2 \text{ hr}} = 5.55 \frac{\text{lbm}}{\text{ft}^2 \text{ s}}$$

$$\dot{m} = 2.836 \times 10^{-2} \frac{\text{lbm}}{\text{s}} = \rho AV$$

$$V = \frac{2.836 \times 10^{-2}}{0.01433(5.11 \times 10^{-3})} = 387.35 \frac{\text{ft}}{\text{s}}$$

The Nusselt number for inside the flow passage is obtained from the Reynold's number based on the hydraulic diameter. $D_H = D_o - D_i$ for an annulus.

$$Re_H = \frac{\rho V D_H}{\mu} = \frac{0.01433(387.35)(1.0 - 0.25)/12}{3.65 \times 10^{-5}}$$

$Re_H = 9505$; however, didn't use turbulent Nu for annulus. Assume instead a Nu for a fully developed turbulent flow in a circular pipe:

$$N_L = 0.023(Re_L)^{0.8} (Pr)^{\frac{1}{3}}$$

$$Re_L = \frac{0.01433(387.35)(3.0/12)}{3.65 \times 10^{-5}} = 38,019$$

$$N_L = 0.023(38,019)^{0.8} (0.72)^{\frac{1}{3}} = 95.11$$

$$h_{sl} = \frac{N_{Lk}}{D_H} = \frac{95.11(3.65 \times 10^{-5})(0.258)}{\frac{1.0 - 0.25}{12}(0.72)}$$

$$= 1.99 \times 10^{-2} \frac{\text{Btu}}{\text{ft}^2 \text{s } ^\circ\text{R}} = 71.65 \frac{\text{Btu}}{\text{ft}^2 \text{ hr } ^\circ\text{R}}$$

$$\text{Inner Area } A_{sl} = \pi DL = \frac{3.14(1.0)(3.0)}{144} = 6.542 \times 10^{-2} \text{ft}^2$$

Now solving for the Nusselt numbers over the thermocouple.

$$Re_D = \frac{\rho VD}{\mu} = \frac{0.01433(387.4)(\frac{.25}{12})}{3.65 \times 10^{-5}}$$

Now with $Re = 3168$, solve for the hemispherical tip:

$$Nu_D = 2 + (0.4Re_D^{1/2} + 0.06Re_D^{2/3}) Pr^{0.4} (\mu/\mu_s)^{1/4}$$

$$Nu_D = 2 + (0.4(3168)^{1/2} + 0.06(3168)^{2/3})(0.72)^{0.4} = 33.12$$

$$h_{tc \text{ tip}} = \frac{Nu_D k}{D_H} = \frac{33.12(3.65 \times 10^{-5})(0.28)}{(.25/12)(0.72)} = 0.0226$$

$$Nu_{shaft} = 0.023(Re_H)^{0.8} Pr^{1/3} = 0.023(9505)^{0.8}(0.72)^{1/3} = 31.37$$

$$h_{tc \text{ shaft}} = \frac{Nu k}{D_H} = \frac{33.07(3.65 \times 10^{-5})(0.28)}{(.25/12)(0.72)} = 0.0214 \frac{Btu}{ft^2 s ^\circ R}$$

Note that the heat transfer coefficient for both the thermocouple and the thermal shield are roughly the equivalent: $70.04 \approx 71.65 \text{ Btu}/(\text{ft}^2 \text{ hr } ^\circ R)$.

VITA

Whitney J. Robinson was born on July 28, 1953 in Manhattan, Kansas. He attended Martin High School, Martin, Tennessee and graduated in May 1971. After one year attendance at the University of Tennessee at Martin, he received an appointment to the United States Military Academy, West Point, New York. After graduating in June 1976, he served for eleven and a half years in the Infantry and Special Forces. Upon getting out of the service, he went to Tullahoma, Tennessee to attend the University of Tennessee Space Institute to work on a Master's in Mechanical Engineering.

He is married to Elaine Hall Robinson of Martin, Tennessee and they have two sons: William and Daniel.