

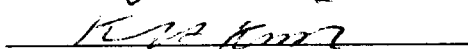
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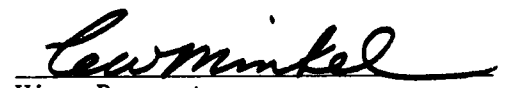


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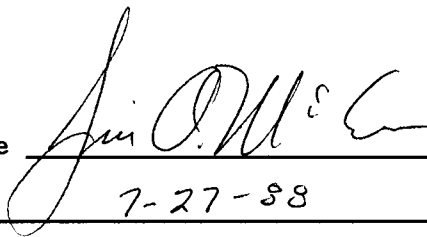

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DESIGN OF A CONTROLLED TEMPERATURE FLUE GAS CORROSION PROBE

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Jim A. McEvers

August 1988

DEDICATION

I wish to dedicate this thesis to my family; my wife Dona, my daughter Angela, and with special love to my son Robert Allan McEvers.

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ABSTRACT

The need to determine suitable materials for use in industrial waste heat recovery systems requires assessment of materials degradation in various flue gas environments. Such an assessment generally involves exposing candidate materials to the environments, then analyzing the effects of the exposure conditions. Degradation of materials is related to flue gas chemical composition, temperature, and thermal cycling behavior. Temperature variations, especially atypical thermal cycling, complicate the task of characterizing such phenomena as corrosion rates and corrosion mechanisms. A relatively constant temperature would allow a more accurate determination of the effects of flue gas chemical composition and temperature. In response to this need, a flue gas corrosion probe has been designed for exposure tests of materials. The probe allows materials to be tested at nominally constant temperatures within a wide span of either constant or changing flue gas temperatures. This thesis describes the design, fabrication, and verification of the controlled temperature flue gas corrosion probe system.

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LIST OF SYMBOLS

A	Area
C	Circumference
C_p	Specific heat
D	Diameter
Δt	Incremental change in time
ΔT	Incremental change in temperature
e	Base of natural logarithm
F	Geometric view factors
h	Heat transfer coefficient
k	Thermal conductivity
L	Length
m	Mass flow rate
Q	Volume flow rate
q_c	Rate of convective heat transfer
q_r	Rate of radiative heat transfer
R	Radiosity
Re	Reynolds number
r	Reflectance, radius
t	Thickness, time
T	Temperature
v	Velocity
V	Volume

Greek Letters

- ϵ (epsilon) Emissivity
- μ (mu) Viscosity
- ρ (rho) Density
- σ (sigma) Stephan-Boltzman Constant

Subscripts

- a Average, annulus (relating to the annulus formed by the specimen and the heating element).
- A Relating to the cooling air
- c Convective
- f Relating to the flue
- g Relating to gases (flue)
- h Relating to the heat source, high
- p Relating to the probe (specimen)
- r Radiative
- s Sum (total)
- w Flue wall
- 1 Low
- 1 With reference to the heating element (geometric view factor)
- 2 With reference to the probe specimen (geometric view factor)
- 3 With reference to the flue wall (geometric view factor)

1. INTRODUCTION

Exposure tests in industrial furnaces have been used in recent years to compare the high-temperature corrosion of various candidate materials for heat exchangers. Specimens have been placed in the flues of aluminum remelt furnaces,¹⁻³ steel soaking pits,² steel reheat furnaces,² and glass furnaces.⁴ In most cases, specimens were exposed to ambient furnace conditions for a time determined by the furnace production schedule. Specimens were installed during a furnace shutdown and removed during a subsequent shutdown. During the exposure period, temperatures sometimes could be monitored and a temperature-time history determined, but specimen temperature could not be controlled. Exposure conditions--maximum and minimum temperatures, heating and cooling rates, temperature-time history, and composition of combustion products--all have been dependent on production requirements. The disadvantages of this type of materials exposure test are:

- Specimen temperature is not controlled, and sometimes is not measured accurately.
- Flue gas temperature cycles frequently in some furnaces, and therefore specimen temperature also cycles.
- Corrosion data are based on a complicated temperature-time history.
- Corrosion data for a specific temperature are not determined.
- The effect of temperature on corrosion rate is not determined.
- Extrapolation of corrosion data to other furnaces with different temperature-time histories is difficult.

- Stresses induced by thermal cycling cause additional degradation.

Exposure of materials to ambient conditions within a particular furnace might provide useful information for an intended application, especially if the furnace operates normally. Alternatively, results might serve only as a guide to materials behavior because of atypical temperature-time history. In the latter case, results would represent conditions unmatched in other furnaces of the same type and would not be applicable under a different set of conditions. Furthermore, data interpretation is difficult if temperatures vary over a wide range because corrosion rates cannot be correlated to specific temperatures. As a result, additional exposure tests would be required or results would have to be estimated. Estimation of corrosion results is unwise if unanticipated failure of a heat exchanger occurring as a result of these estimates would cause production losses.

A possible solution to this problem of temperature variation is instrumented, temperature-controlled corrosion probes. Ideally, several such probes are operated at different temperatures of interest in a typical combustion environment. Preselected constant temperatures or temperature-time histories are then maintained by heating or cooling as required. In this manner results are obtained for test conditions representing a particular heat exchanger or for more generic conditions. The main features of a controlled-temperature corrosion probe are:

- thermocouples for monitoring temperatures,
- internal heating to increase temperature,

- internal cooling to decrease temperature,
- automatic temperature control, and
- continuous recording of temperature.

Advantages and disadvantages of conventional and controlled temperature exposure tests are presented in Table 1. A disadvantage of the corrosion probe is that relatively few specimens can be tested with the current design, which increases the cost per specimen per test. The conventional exposure test, therefore, can be used to screen a large number of materials, and the corrosion probe can be used to obtain pertinent data for the better materials. The probe can be installed or removed from the furnace flue at any time, even when the furnace is operating, and companion specimens having no

Table 1. Comparison of corrosion test methods.

Comparison Criteria	Conventional Exposure Test	Controlled Temperature Corrosion Probe
Number of materials per test	Many	Few
Expense per specimen	Low	Moderate
Temperature known	Yes	Yes
Temperature controlled	No	Yes
Temperature-time history	Complicated	Simple
Corrosion rate at temperature of interest	No	Yes
Effect of temperature on corrosion rate	No	Yes
Extrapolatable data	No	Yes
Furnace shutdown required for installation or removal	Yes	No

temperature control can be installed along with the probe(s). Corrosion data representative of the temperature fluctuations of the furnace obtained with these specimens might be useful for a heat exchanger on that particular furnace. The probe data, on the other hand, could be applied to heat exchangers on other furnaces having different temperature-time histories.

The objective of this work has been to develop a conceptual design for an instrumented, temperature controlled corrosion probe suitable for testing candidate heat exchanger materials (both metallic and ceramic) in high-temperature combustion environments. The conceptual design involves size, assembly features, heating and cooling methods, and temperature control instrumentation. The probe has a tubular configuration with temperature control provided by a regulated internal heater and by the flow of cooling air through the bore. The cooling method is similar to that employed in metallic corrosion and fouling probes,⁵⁻⁷ which were designed to operate at temperatures considerably less than flue gas temperatures by regulating the flow of air through a tubular configuration in contact with the flue gases. Practically no control was obtained in the corrosion probes because of equipment malfunction,⁵ whereas good control was achieved in the fouling probe. Neither the corrosion probes nor the fouling probes contained a heat source for increasing the operating temperature.

Computer models were used to evaluate the characteristics of the corrosion probe discussed here. The effects of size, spacings, cooling air flow rate, heater capacity, and flue gas temperature and

flow rate were evaluated to determine the possible range of temperature control. The models permitted a theoretical determination of the effect of exposure conditions on probe performance before actual fabrication of the probe.

The analysis was performed based on silicon carbide as the probe specimen material. This material is considered to have good thermal shock characteristics and good thermal conductivity, and it can be used at high temperatures. These properties make it a good candidate for waste heat recovery application.

2. DESIGN BASIS

Although the probe is intended for use in various furnaces in which the flow rate and temperature of flue gases are different, the heater and air flow are sized for a typical industrial aluminum remelt furnace. The reference furnace burns natural gas with 10% more air than the stoichiometric amount required for combustion. At the highest firing rate (Table 2), the burners produce 4.7 MW (16E + 06 Btu/h) of heat and flue gas temperatures of about 1300°C (2400°F). At the lowest firing rate, the quantities are 0.9 MW (3E + 06 Btu/h) and about 750°C (1400°F). The flue gases consist mostly of N₂, CO₂, H₂O, and O₂, as shown in Table 2. Figure 1 illustrates a possible flue gas temperature variation in an industrial furnace. The temperature increases rapidly from 750 to 1300°C, holds for about 2 h, then decreases to 750°C. After 1 h at 750°C, the cycle is repeated. The purpose of the probe control system is to maintain the probe at a constant temperature during the thermal cycles. Calculations were conducted to determine the amount of internal heating and cooling required to maintain a constant temperature by adding heat when the flue gas temperature is below the probe setpoint and by cooling when the flue gas temperature is too high.

Table 2. Reference furnace combustion conditions.

Combustion Products ^a	High Fire	Low Fire
Heat content		
MW	4.7	0.9
Btu/h	16 E+06	3 E+06
Temperature		
°C	1300	750
°F	2400	1400
Mass flow rate		
kg/s	1.69	0.29
lb/h	13,385	2295

^a~71% N₂, 12% CO₂, 12% H₂O, 2% O₂.

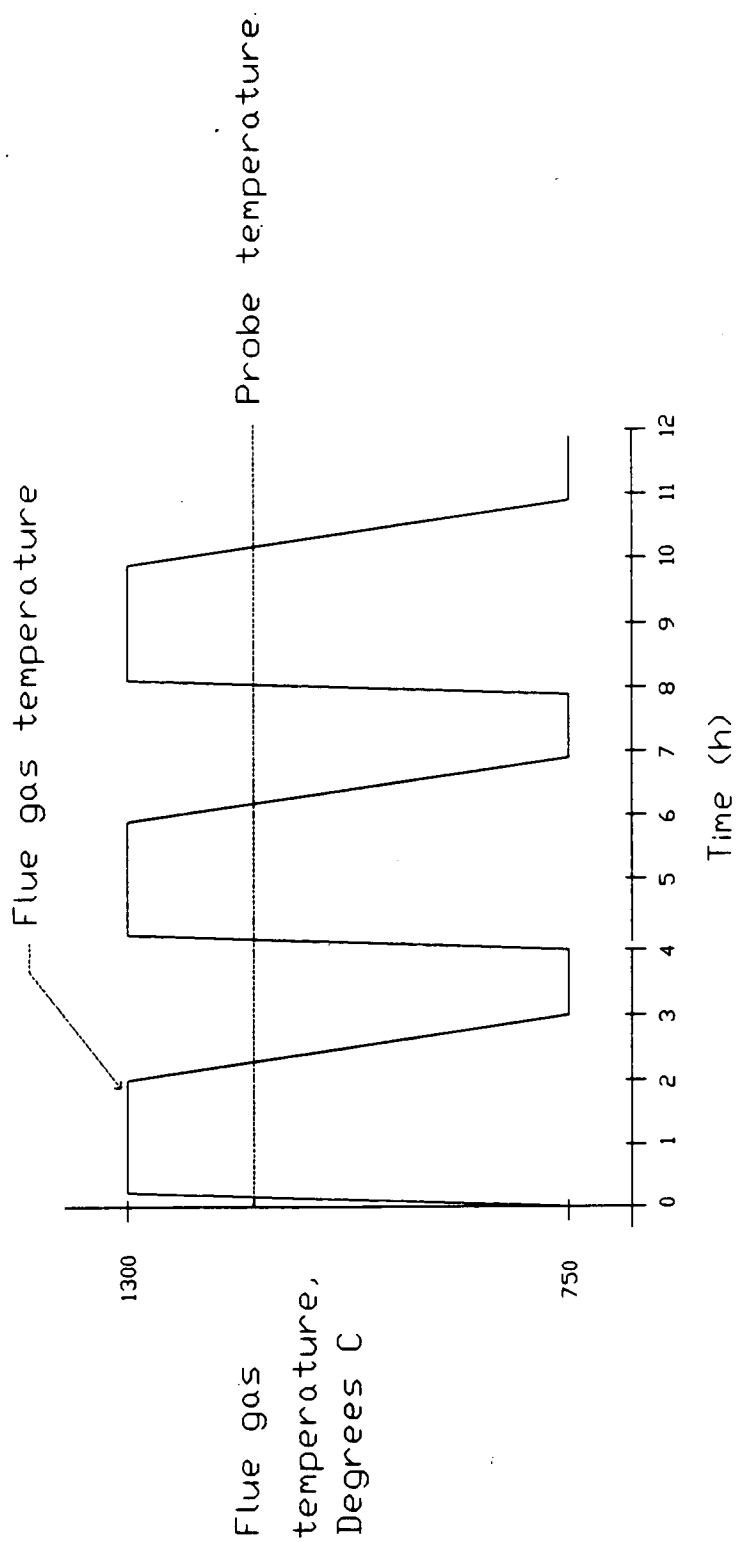


Figure 1. Example of flue gas vs corrosion probe temperature.

3. CHARACTERISTICS OF A CORROSION PROBE

The corrosion probe has been designed to provide features that are not available in conventional exposure tests.

- It can be installed in and removed from an industrial furnace flue without affecting furnace operation.

- It can automatically maintain the desired temperature within a range of flue gas temperatures and flow rates.

- It can provide a continuous, permanent temperature-time history. The conceptual design is a tubular configuration (Figure 2) composed of the specimen (test section) and other commercially available components. The probe projects a predetermined distance through a port in the flue wall and is immersed in flue gases in a crossflow mode. The test section in the reference design is ~2.5 cm OD $\times$ 1.9 cm ID $\times$ 38 cm long (1 in. OD $\times$ 0.75 in. ID $\times$ 15 in. long). These dimensions limit the size and capacity of the internal heat source. Exposure of the probe may be terminated at any time by withdrawing the probe from the flue wall and closing the port in the wall.

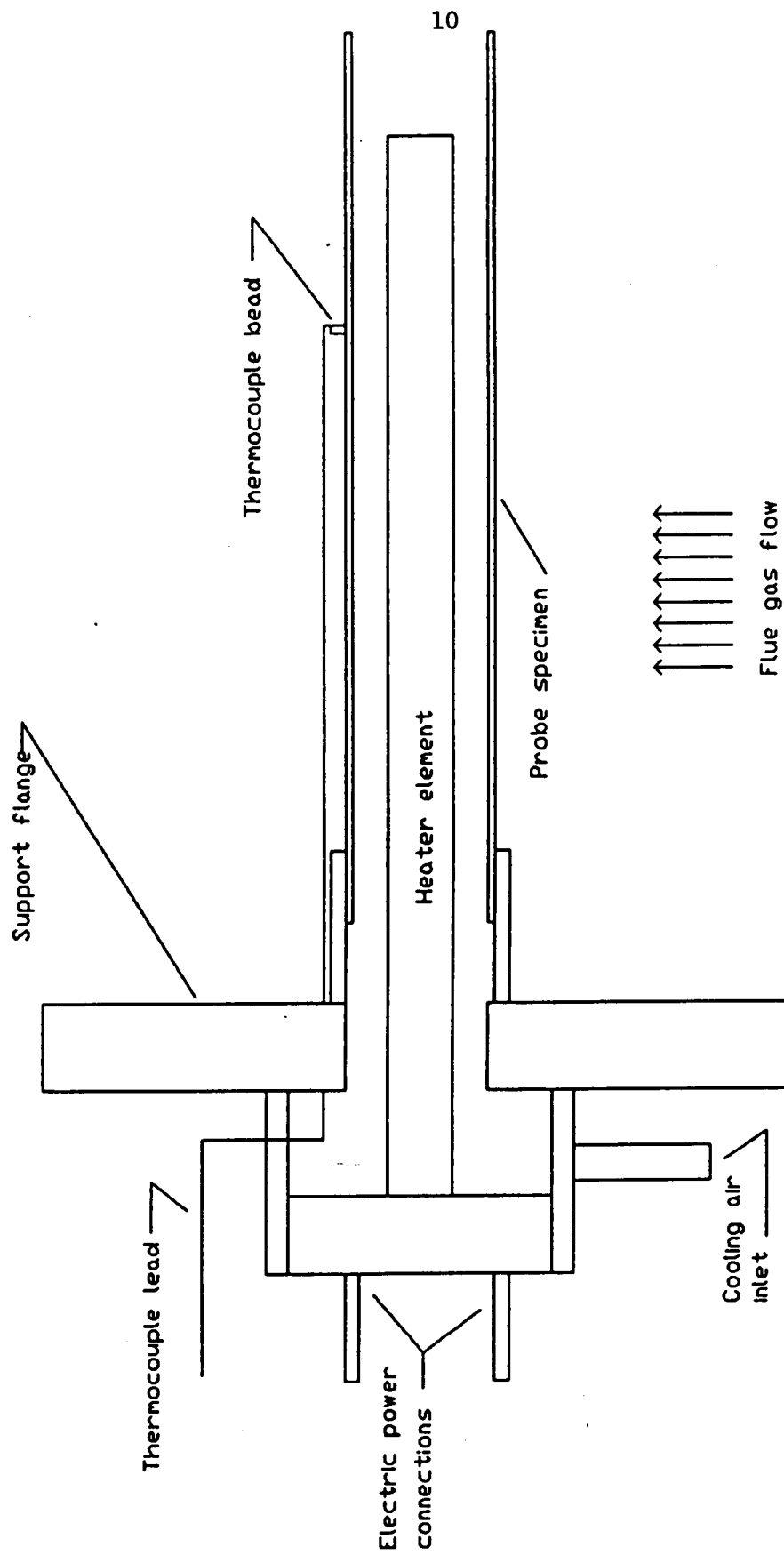


Figure 2. Corrosion probe detail.

4. PROBE TEMPERATURE CONTROL

The probe temperature control system used to regulate the cooling gas flow and heat source power consists of a Barber-Coleman model 570 digital proportional-integral-derivative (PID) process controller, Badger Meter, Inc., Research Control flow control valves (for cooling gas and fuel gas flow control), an Electronic Control Systems, Inc., Model No. 7810 single-phase, silicon controlled rectifier (SCR) power controller (for electric heating power control), and Bellofram Corp. Type 1000 current-to-pressure converters (I/P). The temperature is measured using an unsheathed type S (platinum/platinum-10% rhodium) thermocouple.

The thermocouple wires are fed through alumina insulating tubes and fixed to the probe using alumina cement.

A diagram of the control system is provided in Figure 3. The thermocouple is mounted along the top (downstream) surface of the probe. Although the bead is exposed to fouling, by cementing the probe into a recess at this location, a more representative value of average probe temperature is obtained. Also, the thermocouple is not exposed to the direct force of the flue gas stream. The thermocouple, which is attached to the outer surface of the test section, provides a signal to the controller. The controller regulates an internal heat source and cooling air flow valve to maintain temperature. Actual temperature control is accomplished by the conventional PID controller. This unit measures probe temperature by means of the thermocouple and computes an appropriate output signal to drive the control elements, an electric power control or fuel gas flow control

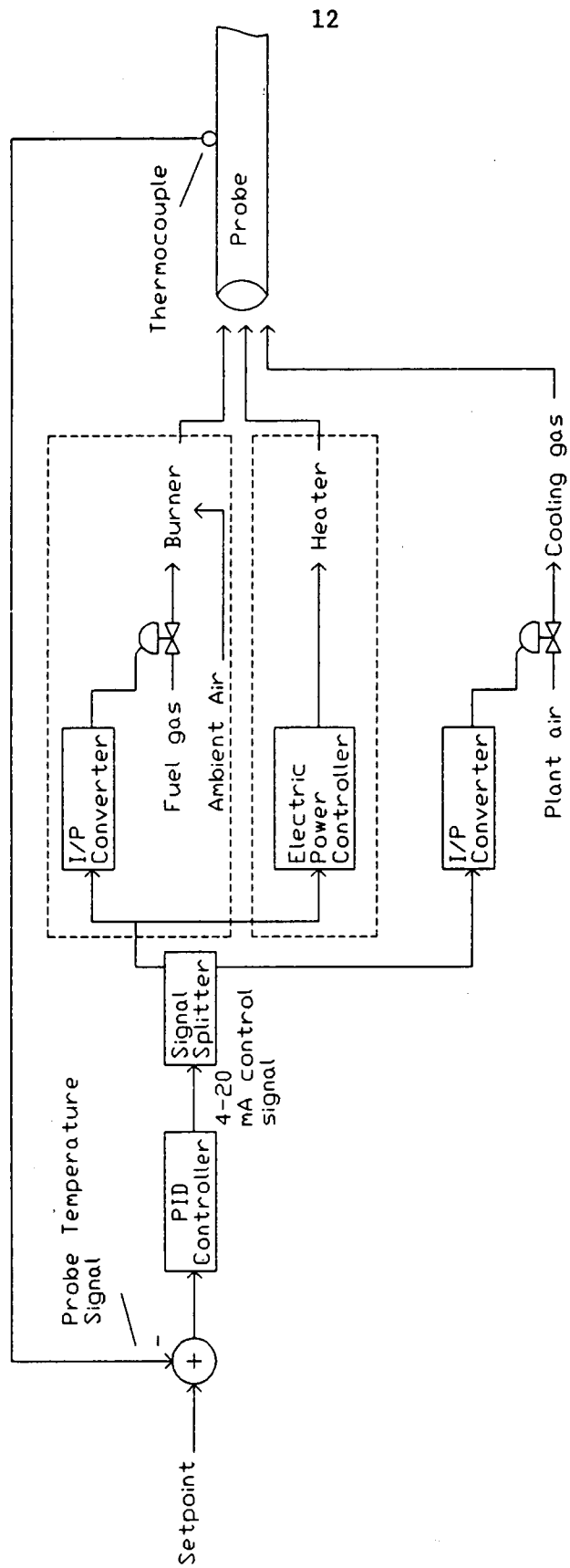


Figure 3. Probe assembly/control system block diagram.

valve, depending on the heat source being used and a cooling air flow control valve.

The PID controller is an electronic device that utilizes the following mathematical operation to maintain a process at a desired operating point:⁸

$$P = K_p \left[\text{error} + \frac{1}{T_I} \int \text{error} \, dr + T_D \frac{d\text{error}}{dt} \right]$$

where

P = change in controller output signal,

K_p = controller proportional gain value,

error = difference between desired temperature (controller setpoint) and actual temperature

T_I = integral time,

r = dummy variable,

T_D = derivative time,

$d\text{error}/dt$ = time rate of change of error.

The consideration of two possible alternative heating sources was based on convenience and availability versus required temperature span. Electric heating is by far the more convenient method but does not provide sufficient power to compensate for large temperature differences. Gas heating is capable of providing higher power levels but at the cost of having to supply fuel gas and supplemental combustion air. The basic probe design is such that either method may be utilized without modifying the basic probe heat source mounting arrangement.

The power controller is a single-phase, 115-V, phase-fired silicon controlled rectifier (SCR) unit. The fuel gas flow control and cooling air flow control valves are conventional proportional pneumatic valves, which are driven by separate I/P (current-to-pressure) converters. The output from the process controller is a standard 4- to 20-milliamp (mA) signal. The 12-mA point is used as a control "null" point. A signal greater than 12 mA results in the heat source being energized, and a signal below 12 mA results in the cooling air control valve being actuated.

A special signal converter was designed and built to permit heating from one source and cooling from another. The converter accepts the standard 4-20 mA signal from the Barber-Coleman PID controller, provides a 4-20 mA signal to the electric power controller or fuel gas control valve when heating is needed, and provides a 4-20 mA signal to the cooling gas flow control valve when cooling is required. A schematic diagram of the circuit is provided in Appendix A. This configuration provides appropriate control with a minimum of components.

Figure 4 is a photograph of the portable cabinet containing the PID controller, electric power controller, and signal splitter.

Figure 5 is a photograph of the portable cabinet containing the flow control valves, pressure gages, and cooling air flow meter.

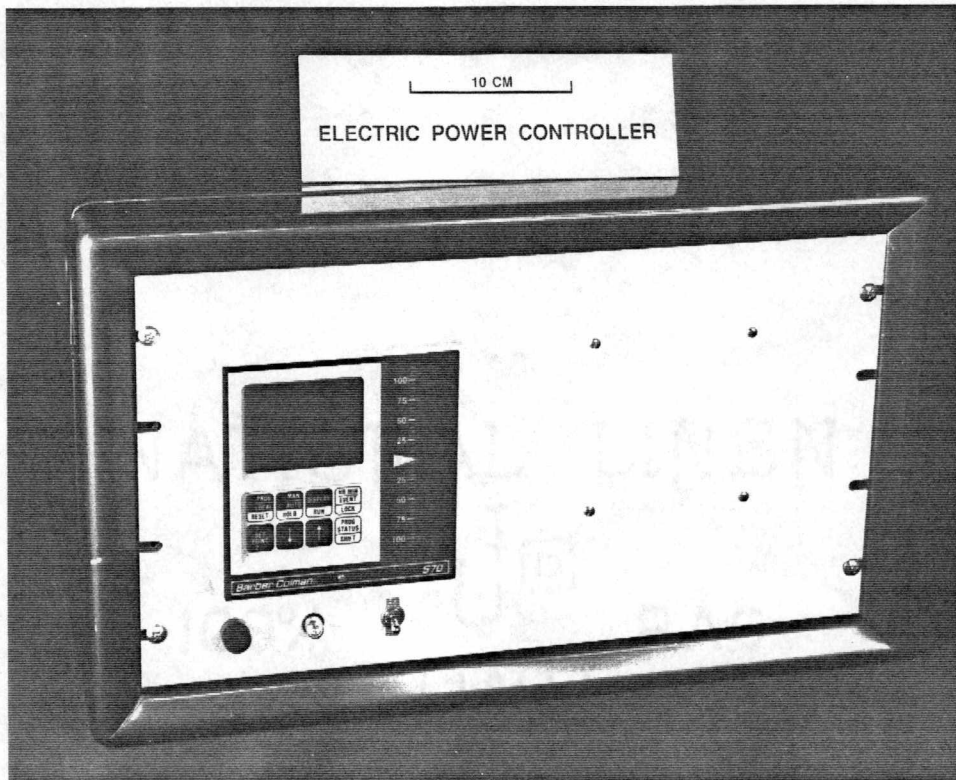


Figure 4. Electric power controller cabinet.

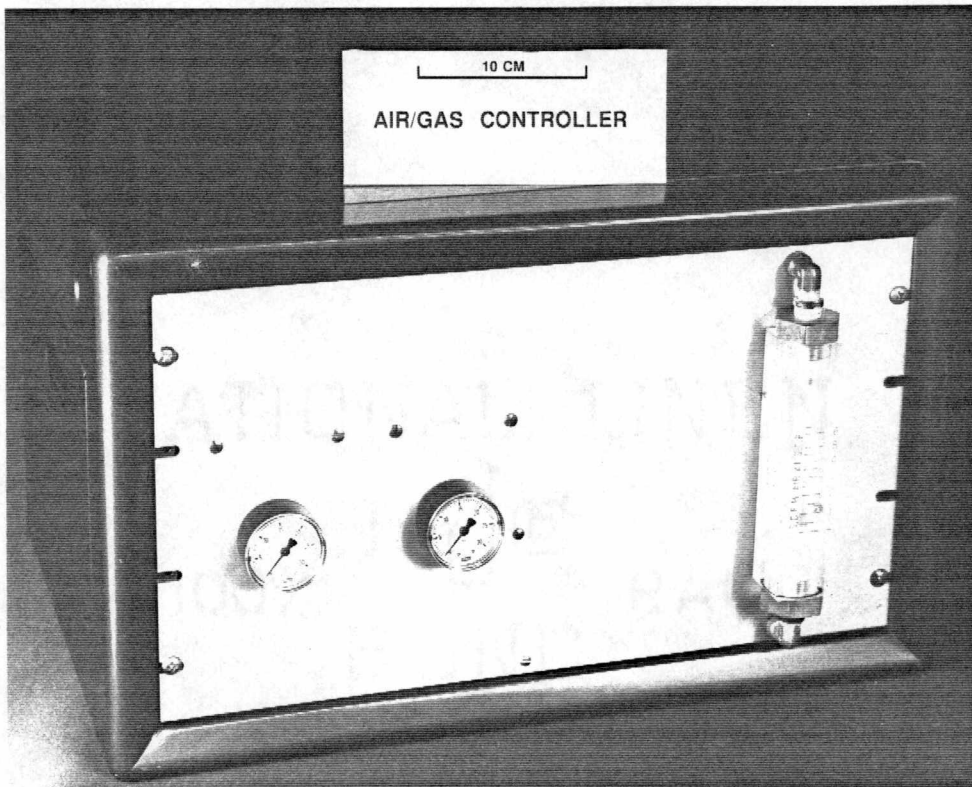


Figure 5. Cooling air/fuel gas control cabinet.

5. HEAT TRANSFER CALCULATIONS

A proper controlled temperature flue gas corrosion probe design requires an analysis of the heat transfer between the probe and its environment. This is necessary to determine the rate at which energy must be added to the probe when the flue temperature is below the desired probe temperature or the rate at which energy must be removed from the probe when the flue temperature is above the desired probe temperature. Heat transfer calculations have been conducted for a silicon carbide (SiC) corrosion probe having the characteristics shown in Table 3, for flue gases having the characteristics shown in Tables 2 and 4, for a refractory furnace flue having the characteristics shown in Table 5, and for cooling air having the characteristics shown in Table 6. Accurate temperature control requires balancing the thermal energy received by and transferred from the probe. The balance of thermal energy within the probe (q_s) can be described in terms of the various components:

$$q_s = q_{cg} + q_{rf} + q_{ch} + q_{rh} + q_{ca} \quad , \quad (1)$$

where

q_{cg} = convective heat transferred to/from the probe by the flue gases,

q_{rf} = radiative heat transferred to/from the probe by the flue wall,

q_{ch} = convective heat transferred to/from the probe by the heater,

q_{rh} = radiative heat transferred to/from the probe by the heater,

Table 3. Characteristics of SiC probe and heater used in heat transfer calculations.

Parameter	Symbol	Value
Probe		
length	L_p	38.10 cm
diameter	D_p	2.54 cm
surface area	A_p	303.8 cm ²
emissivity ^a	ϵ_p	0.9
reflectance ^b	r_p	0.1
wall thickness	t_p	0.318 cm
radius (inner)	r_i	0.948 cm
temperature	T_p	°C (varies with operating conditions)
specific heat	$C_{p,p}$	1121 J/kg-K
density	ρ_p	3100 kg/m ³
volume	V_p	9.29×10^{-5} m ³
Heater		
radius (outer)	r_h	0.635 cm
temperature	T_h	°C (varies with operating conditions)
Annulus		
cross-sectional area	A_a	1.5598 cm ²
average radius	r_a	0.793 cm

^aStandard Oil Engineered Materials Co.,
Niagara Falls, New York (formerly Carborundum Resistant
Materials Company).

$$^b r_p = 1 - \epsilon_p.$$

Table 4. Characteristics of flue gases used in heat transfer calculations.

Parameter	Symbol	Value
Viscosity	μ_g	0.05134 mPa·s
Density		
high fire	ρ_{gh}	0.2579 kg/m ³
low fire	ρ_{gl}	0.0513 kg/m ³
Thermal conductivity	k_g	0.081497 mW/m·K
Mass flow rate		
high fire	$\dot{m}_{gh}$	1.588 kg/s
low fire	$\dot{m}_{gl}$	0.318 kg/s
Velocity		
high fire	v_{gh}	414.132 cm/s
low fire	v_{gl}	82.814 cm/s
Reynolds number	Re_g	528.382

Table 5. Characteristics of flue used in heat transfer calculations.

Parameter	Symbol	Value
Cross-sectional area	A_f	14,864 cm ²
Emissivity ^a	ϵ_{fw}	0.8
Reflectance ^b	r_{fw}	0.2
Geometric view factors ^c	F_{23}	1
	F_{33}	1
	F_{32}	0

^aEstimated.^b $r_{fw} = 1 - \epsilon_{fw}$ (for solid body).^c F_{23} = view factor from probe to flue wall. F_{33} = view factor from flue wall to flue wall. F_{32} = view factor from flue wall to probe.

Table 6. Characteristics of cooling air used in heat transfer calculations.

Parameter	Symbol	Value
Density ^a	ρ_A	0.3204 kg/m ³
Thermal conductivity ^a	k_A	0.081497 mW/m·K
Viscosity ^a	μ_A	0.05134 mPa·s
Specific heat ^a	C_{pA}	1222.55 J/kg·K
Volume flow rate	Q_A	102412.6 cm ³ /s
Cooling air inlet temperature	T_{IN}	27°C

^aO. W. Eshbach, ed., *Handbook of Engineering Fundamentals*, 3d ed., John Wiley and Sons, New York, p. 650, 1975.

$q_{c,a}$ = convective heat transferred to/from the probe by cooling air.

The convective heat flow between the probe and the flue gas is given by Newton's Law of Cooling,

$$q_{c,g} = (T_g - T_p) h_c A_p, \quad (2)$$

where

T_g = temperature of the flue gases,

T_p = temperature of the probe,

h_c = convective heat transfer coefficient,

A_p = surface area of the probe.

The convective heat transfer coefficient is given by:⁹

$$h_c = \frac{k_g C Re_g^n}{D_p}, \quad (3)$$

where all terms except C and n are identified in Tables 3 and 4. The values of C and n depend on the value of the Reynolds number (Re_g), and were obtained from the following table.⁹

Re_g	C	n
0.4 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4000	0.615	0.466
4000 - 40,000	0.174	0.618
40,000 - 400,000	0.0239	0.805

The total radiative heat flow between the probe and the flue wall can be determined as follows. Let:

A_i = area of surface i ,

ϵ_i = emissivity of surface i ,

r_i = reflectance of surface i ,

R_i = radiosity of surface i ,

A_j - area of surface j ,

ϵ_j - emissivity of surface j ,

r_j - reflectance of surface j ,

R_j - radiosity at surface j .

(E.g., i could correspond to the probe; j could correspond to the flue wall.)

The total power leaving a surface i is:

$$A_i R_i = \text{total emitted power} + \text{total reflected power} , \quad (4)$$

where

$$\text{total emitted power} = A_i \epsilon_i \sigma T_i^4 , \quad (4a)$$

$$\text{reflected power} = r_i [\text{total incident power}] , \quad (4b)$$

$$\text{total incident power} = \sum_{j=1}^n A_j F_{j,i} R_j ,$$

$$n = \text{total number of other surfaces or objects}. \quad (4c)$$

Substituting Equations (4a) through (4c) into Equation (4):

$$A_i R_i = A_i \epsilon_i \sigma T_i^4 + r_i \sum_{j=1}^n A_j F_{j,i} R_j . \quad (5)$$

Note that the geometric definition of $F_{j,i}$ requires that $A_i F_{i,j} = A_j F_{j,i}$.

Therefore, Equation (5) may be written as:

$$A_i R_i = A_i \epsilon_i \sigma T_i^4 + r_i \sum_{j=1}^n A_i F_{i,j} R_j . \quad (6)$$

Eliminating the surface area we get:

$$R_i = \epsilon_i \sigma T_i^4 + r_i \sum_{j=1}^n R_j F_{i,j} . \quad (7)$$

The net power or rate of energy transfer from surface i can be written as:

$$A_i q_i = \text{incident power} - \text{leaving power}, \quad (8)$$

or, from Equations (4c) and (4):

$$A_i q_i = A_i \sum_{j=1}^n R_j F_{ij} - A_i R_i \quad (8a)$$

From Equation (6), we can obtain an expression for $A_i \sum R_j F_{ij}$:

$$A_i \sum R_j F_{ij} = \frac{A_i R_i}{r_i} - \frac{A_i \epsilon_i \sigma T_i^4}{r_i} \quad (9)$$

Substituting Equation (9) into Equation (8a), we get:

$$q_i = \frac{R_i}{r_i} - \frac{\epsilon_i \sigma T_i^4}{r_i} - R_i \quad (10)$$

Collecting terms:

$$q_i = R_i \left(\frac{1}{r_i} - 1 \right) - \frac{\epsilon_i \sigma T_i^4}{r_i} \quad (11)$$

Using the relationships,

$$\frac{1}{r_i} - 1 = \frac{1 - r_i}{r_i}, \text{ and } 1 - r_i = \epsilon_i,$$

and making the appropriate substitutions into Equation (11):

$$q_i = \frac{\epsilon_i}{r_i} \left(R_i - \sigma T_i^4 \right) \quad (12)$$

For the specific case of the probe relative to the flue wall,

Equation (12) becomes

$$q_{rf} = \frac{\epsilon_p}{r_p} (R_p - \sigma T_p^4) A_p \quad (13)$$

where σ is the Stefan-Boltzman constant ($5.67 \times 10^{-8} \text{ W/m}^2 \text{K}^4$) and new terms except R_p are identified in Table 3.

The term R_p , the probe radiosity relative to the flue, is determined from the following. From Equation (7), where subscript i is in reference to the probe and subscript j is in reference to the flue, we can write:

$$R_p = r_p F_{23} R_f = \epsilon_p \sigma T_p^4 \quad (14)$$

Also:

$$R_f = r_f F_{33} R_f = \rho_p F_{32} R_p = \epsilon_f \sigma T_f^4 \quad (15)$$

Solving Equations (14) and (15) for R_p , we obtain:

$$R_p = \frac{\epsilon_p \sigma T_p^4 (1 - r_{fw} F_{33}) + r_p F_{23} \epsilon_{fw} \sigma T_g^4}{1 - r_{fw} F_{33} - r_p^2 F_{23} F_{32}} \quad (16)$$

where new terms are identified in Tables 3 and 5. The total heat transfer between the probe and its surroundings is obtained by combining the convective and radiative contributions

$$q_t = q_{cg} + q_{rf} \quad (17)$$

Table 2 shows that the flue gas temperature varies between 750 and 1300°C. Either heating or cooling is required to maintain the probe temperature within this range. When the flue gas temperature is below the desired probe temperature, heat is added to the specimen from the internal heater. Initially, electric heating is employed, although heat may be provided by either electricity or by gas combustion. The electric heater is essentially completely enclosed by the specimen; therefore, in the steady state condition, the heat input to the probe by convection and radiation is equal to heater input power. Since this heater can provide about 1.6 kW,

$$q_h = q_{ch} + q_{rh} = 1.6 \text{ kW} \quad (18)$$

When the flue gas temperature is above the desired probe temperature, excess heat must be removed from the probe. This is accomplished by flowing a stream of cooling air through the bore of the probe. The component q_{ca} can be derived as follows. Consider a differential volume, $A_a dx$, as shown in Figure 6. Let:

q_h = convective heat transfer to the differential volume
from the heater,

q_p = convective heat transfer to the differential volume
from the probe specimen,

q_{in} - heat energy transported into the differential volume
by incoming cooling air,

q_{out} - heat energy transported out of the differential volume
by outgoing cooling air,

q_{stored} - rate of heat energy storage in differential volume,

dx - length of differential element,

A_a - cross-sectional area of annulus formed by probe and
heater,

C_i - inside circumference of probe,

C_h - outside circumference of heating element,

h_1 - convective heat transfer coefficient between the
heater and the cooling air as a function of cooling
air flow rate,

h_2 - convective heat transfer coefficient between the inner
surface of the probe and the cooling air as a function
of cooling air flow rate.

T - temperature of cooling air in $A_a dx$.

Performing energy balance on the differential volume $A_a dx$:

$$q_{stored} = q_{in} - q_{out} + q_h + q_p \quad (19)$$

The rate of energy storage into the differential volume $A_a dx$ can be
expressed as:

$$q_{stored} = \rho_A C_{pA} A_a \frac{\partial T}{\partial t} dx \quad (20)$$

The rate of energy transfer into $A_a dx$ by the cooling air is:

$$q_{in} = \rho_A C_{pA} v_A A_a T_x \quad (21)$$

where

T_x - temperature at the left side of element $A_a dx$.

The rate of energy transfer out of $A_a dx$ by the cooling air is:

$$q_{out} = \rho_A C_{pA} v_A A_a T_{x+dx} - \rho_A C_{pA} v_A A_a \left(T_x + \frac{\partial T}{\partial x} dx \right) \quad (22)$$

The rate of energy transfer into $A_a dx$ from the heating element across the heat transfer film coefficient h_1 is:

$$q_h = C_h h_1 (T_h - T) dx \quad (23)$$

The rate of energy transfer into $A_a dx$ from the probe specimen across the heat transfer film coefficient h_2 is:

$$q_p = C_p h_2 (T_p - T) dx \quad (24)$$

Substituting into Equation (19) and putting in standard form:

$$\frac{\partial T}{\partial x} + \left(\frac{C_h h_1 + C_p h_2}{\rho_A C_{pA} v_A A_a} \right) T = \frac{C_h h_1 T_h + C_p h_2 T_p}{\rho_A C_{pA} v_A A_a} - \frac{1}{v_A} \frac{\partial T}{\partial t} \quad (25)$$

Since we are concerned only with the steady state temperature change of the cooling air as it passes through the probe, let:

$$\frac{\partial T}{\partial t} = 0 \quad (26)$$

Equation (25) then becomes:

$$\frac{dT}{dx} + \left(\frac{C_h h_1 + C_p h_2}{\rho_A C_{pA} v_A A_a} \right) T = \frac{C_h h_1 T_h + C_p h_2 T_p}{\rho_A C_{pA} v_A A_a} \quad (27)$$

Let

$$\beta = \frac{C_h h_1 + C_p h_2}{\rho_A C_{pA} v_A A_a} \quad (28)$$

and

$$\phi = \frac{C_h h_1 T_h + C_p h_2 T_p}{\rho_A C_{pA} v_A A_a} \quad (29)$$

The differential equation for the change in cooling air temperature as it passes through the annulus is then:

$$\frac{dT}{dx} + \beta T = \phi \quad (30)$$

We can let $T(x) = 0$ at $x = 0$ since we are interested only in the change in temperature.

The solution of Equation (30) is therefore:

$$T(x) = \frac{\phi}{\beta} [1 - e^{-\beta x}] \quad (31)$$

The temperature of the cooling air at the exit of the probe is then

$$T_{out} = \frac{\phi}{\beta} [1 - e^{-\beta L_p}] \quad (32)$$

The rate of energy transfer by the cooling air can be expressed as:

$$q_{cA} = \rho_A Q_A C_{pA} [T_{out} - T_{in}] \quad (33)$$

Since $T_{out} - T_{in}$ represents the rise in cooling air temperature, substituting Equation (32) into Equation (33) gives:

$$q_{cA} = \rho_A Q_A C_{pA} \left[\frac{\phi}{\beta} [1 - e^{-\beta L_p}] \right] \quad (34)$$

Expanding Equation (34), we get:

$$q_{cA} = \rho_A Q_A C_{pA} \left\{ \frac{C_h h_1 T_h + C_p h_2 T_p}{C_h h_1 + C_p h_2} \left[1 - e^{-\left[\frac{C_h h_1 + C_p h_2}{\rho_A C_p Q_A} \right] L_p} \right] \right\} \quad (35)$$

Since $C = 2\pi r$, Equation (35) may be written as

$$q_{cA} = \rho_A Q_A C_{pA} \left[\frac{h_1 r_h T_h + h_2 r_i T_p}{h_1 r_h + h_2 r_i} \right] \left[1 - e^{-\frac{(h_1 r_h + h_2 r_i) 2\pi L_p}{\rho_A Q_A C_{pA}}} \right] \quad (36)$$

The terms in brackets in Equation (36) were developed for design analysis of probe performance as a function of cooling air flow rate, Q_A , heat transfer film coefficients, h_1 and h_2 , and the location within the probe, L_p . The terms are equivalent to the actual rise in temperature of the cooling air through the probe, a quantity that can be measured during probe operation with a suitably placed thermocouple.

Using Equation (17), the quantity q_t was calculated for the temperature range 750 to 1300°C. Figure 7 shows that q_t , the amount of heat to be added or removed from the probe, increases with increasing difference between T_g and T_p . This figure defines the operating regime of the probe if sufficient heating and cooling capacity are available to maintain the desired probe temperature. Because the available electrical heater can provide only 1.6 kW of heat, the probe can be maintained at a temperature ~125°C above T_g . Gas heating is expected to extend operation to >250°C. Better control can be obtained if cooling is required. The cooling capacity obtained from Equation (36) allows the probe to operate at a temperature ~375°C below T_g .

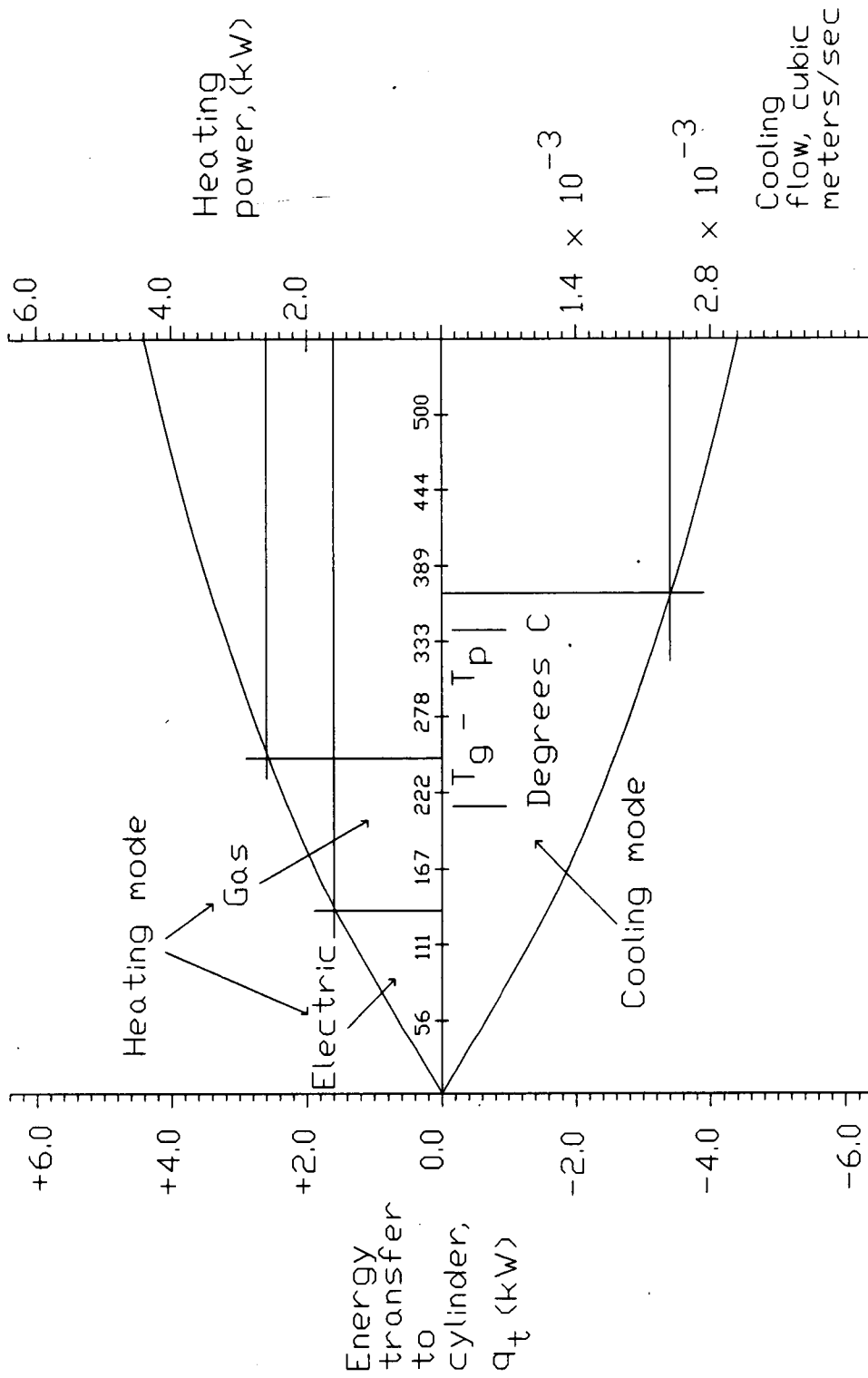


Figure 7. Controlled temperature corrosion probe operating regime.

6. SYSTEM MODELING

Mathematical analysis of anticipated probe performance was accomplished by the development of a steady state model and a separate dynamic system model. The steady state effects of probe size, heater output, cooling air flow rate, and flue gas temperature and flow rate on the temperature of the probe have been investigated by a steady state computer model developed using the MathCAD engineering calculation software package. The model calculates thermal balance in the probe as affected by heat flow due to convection and radiation associated with the flue gas and walls, the heater output, and cooling air flow rate. This simplified model was used initially to assess the feasibility of attaining the necessary heat energy transfer capacities. A listing of this model is provided in Appendix B.

A more comprehensive dynamic mathematical model of the flue gas corrosion probe, including the expected flue gas conditions and the control loop elements, was developed to evaluate the dynamic response of the probe. The model permits variation in probe configuration and conditions prior to actually making physical alterations.

The dynamic model is implemented using the ACSL (Advanced Continuous Simulation Language) package by Mitchell and Gauthier Associates, Inc., 290 Baker Avenue, Concord, Massachusetts 01742. This package is based on the FORTRAN programming language. A listing of the dynamic model program is provided in Appendix C.

The model simulates the probe operating from initial startup and through a step change in setpoint. Plots of the probe temperature and energy transfer rates versus time are shown in Figures 8 and 9.

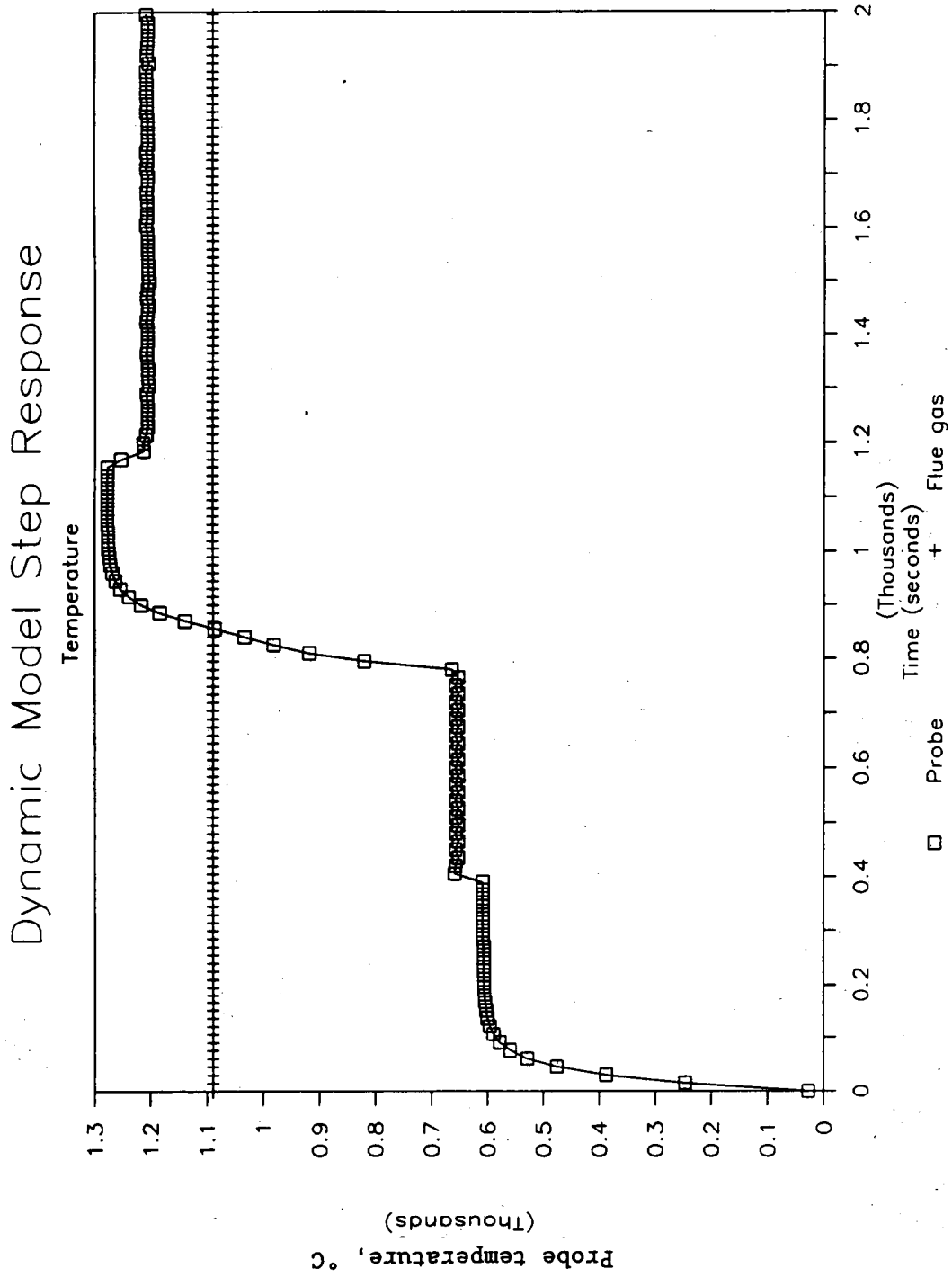


Figure 8. Dynamic model step response, temperature.

Dynamic Model Step Response

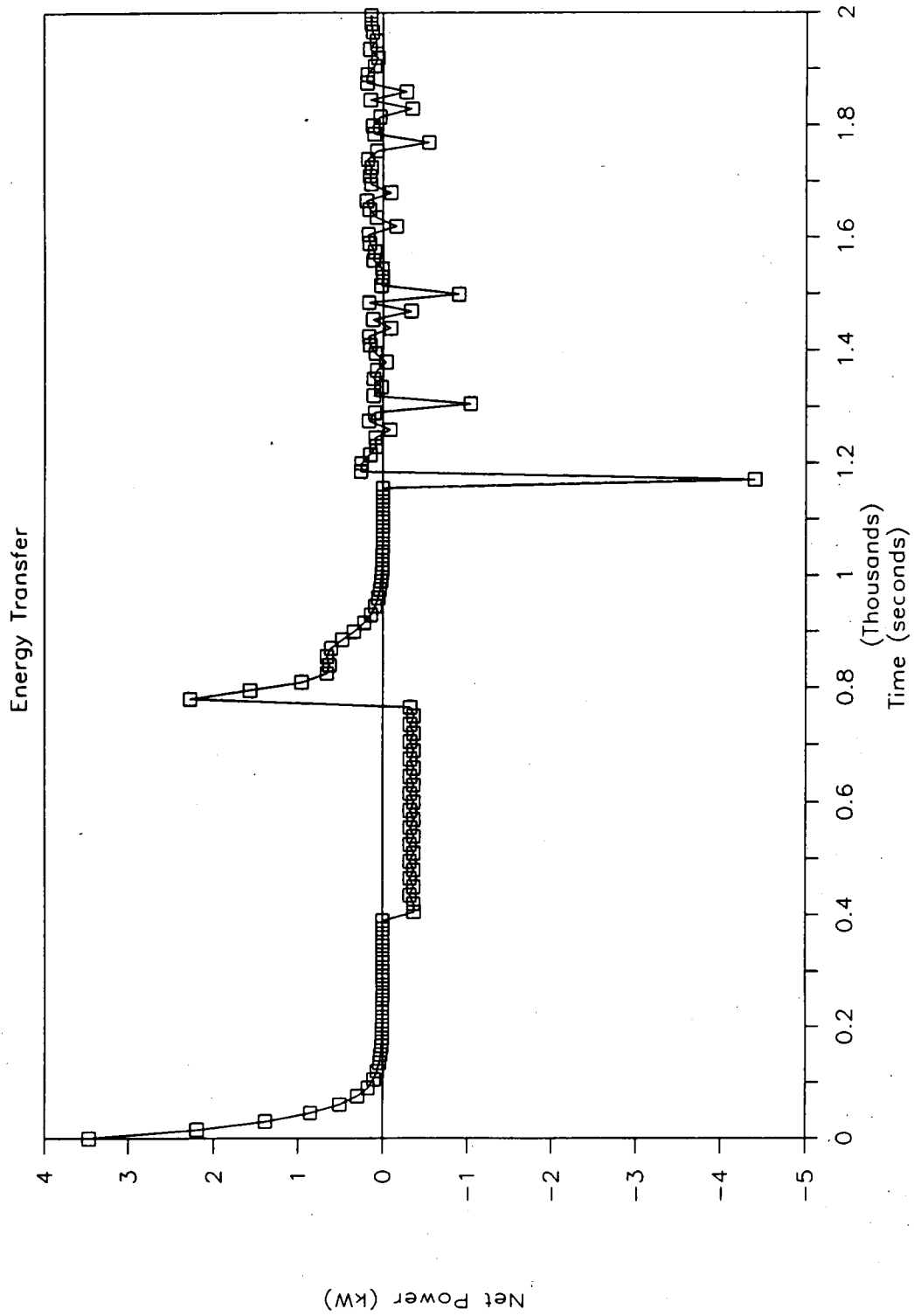


Figure 9. Dynamic model step response, energy transfer.

Figure 9 is a plot of the net power being transferred to/from the probe by the hot flue gases, the electric heating element, and the cooling air. The initial setpoint for the controller is 27°C. The probe is initially exposed to the hot flue gases which are at a temperature of 1090°C. As a result, there is a high rate of energy transfer to the probe. In an effort to maintain the probe at setpoint, the controller opens the cooling air flow control valve. The net rate of energy transfer to the probe decreases toward zero as the probe reaches an equilibrium temperature of 600°C. This temperature does not correspond to the controller setpoint, since there is not sufficient cooling capacity to compensate for such a large difference between setpoint and flue gas temperatures. After 400 s has elapsed, the controller setpoint is stepped to 1200°C. The controller does not respond immediately because the integral term is "wound up" due to the continued error between the setpoint of 27°C and the actual probe temperature. At approximately 750 s, the controller begins to respond by closing the cooling air flow control valve and turning on the electric heating element. The net energy transfer to the probe steps from zero to a new level, which is the sum of the energy transfer from the flue gases and the electric heating element. The large negative excursion at approximately 1150 s is due to the controller going fully into cooling mode to correct for the overshoot. The net energy transfer again decreases toward zero and fluctuates as the probe attains a new controlled temperature at 1200°C.

7. ACTUAL LABORATORY PROBE PERFORMANCE

The fabricated probe assembly and associated control components were assembled in the laboratory to allow preliminary tests of heating and cooling capacities during actual operation. Figure 10 is a photograph of the assembled corrosion probe. The probe was mounted on a steel plate for later installation in an industrial boiler firebox for field tests.

The probe was initially heated by an internal electric heating element, and probe surface temperature data were obtained.

After the probe had returned to room ambient air temperature, the electric heating element was removed and the gas burner assembly was installed. The gas burner was operated in three modes: (1) fuel gas with normally entrained ambient air, (2) fuel gas with normally entrained ambient air with additional air for combustion being supplied through the cooling air inlet, and (3) fuel gas with normally entrained ambient air with 100% O_2 being supplied through the cooling air inlet to support complete combustion. Probe surface temperature data were obtained during each mode to assess the efficiency of gas heating and effectiveness of providing additional combustion air or oxygen.

7.1 Cooling Mode Performance

When the probe was expected to operate at a point below the flue gas temperature, it was necessary to remove excess heat energy. This was accomplished by injecting a stream of air at a nominal temperature

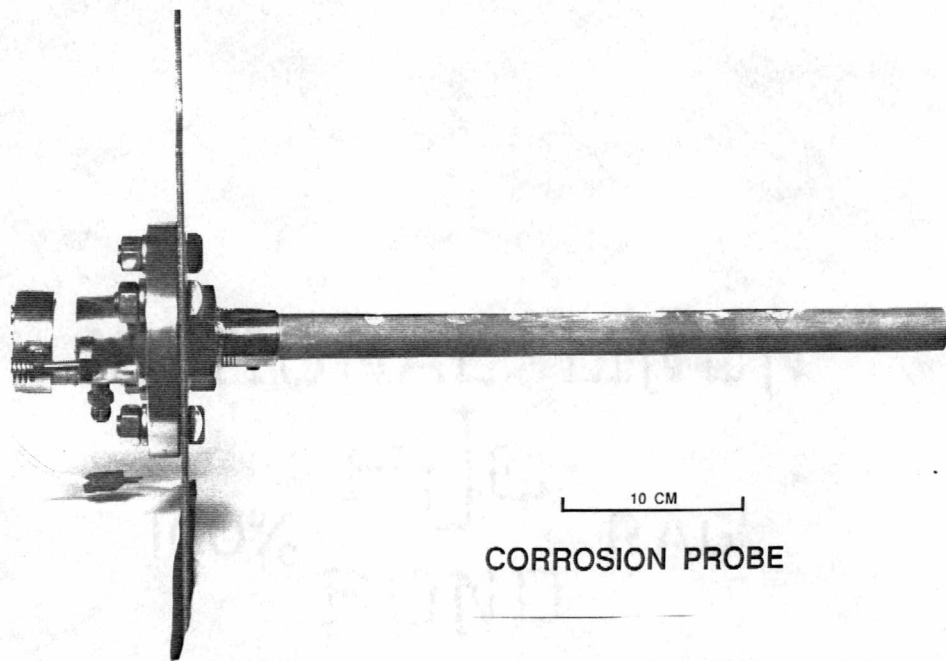


Figure 10. Assembled corrosion probe.

of 27°C into the air inlet connection. The air passed through the center of the probe and exited at the open end, discharging into the flue gas stream. The rate of change of probe temperature versus time was measured and recorded. Equation (37) can be used to obtain energy transfer estimates for incremental changes in temperature versus time.

$$q_p = C_{pp} \rho_p V_p (\Delta T / \Delta t) \quad , \quad (37)$$

Substituting for specific probe properties and parameters, we get:

$$q_p = (0.293 \text{ kW-s/}^\circ\text{C}) (\Delta T / \Delta t) \quad . \quad (37a)$$

Plant air at a pressure of approximately 690 kPa and flow rate of $5 \times 10^{-3} \text{ m}^3/\text{s}$ was used for the coolant. Plant air was chosen for the coolant gas because it is readily available at most facilities at essentially no cost. The temperature values, relative times of measurement, and corresponding rates of energy transfer are shown in Table 7. Plots of these values are provided in Figures 11 and 12. The erratic values plotted for the probe cooling in Figure 11 are considered to be the result of intermittent contact between the thermocouple bead and the probe surface. The bead was cemented to the surface, but it was observed that the cement, and therefore the bead, was not making consistent contact with the probe surface. As stated earlier in this section, the cooling rate data were obtained following the heating of the probe by the internal electric heater. Therefore, the heating element was also being cooled. This reduced the effective cooling of the probe specimen. The cooling capacity is expected to be greater in the case of heating with a fuel gas since the effective thermal mass is significantly reduced.

Table 7. Cooling mode probe temperature response/rate of energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	517	
15	499	0.38
30	469	0.63
45	427	0.88
60	388	0.81
75	347	0.85
90	302	0.94
105	271	0.66
120	232	0.80
135	201	0.65
150	175	0.55
165	152	0.49
180	129	0.47
195	110	0.41
210	93	0.35
225	83	0.21
240	71	0.26
255	60	0.23
270	53	0.15
285	47	0.12

Flue Gas Corrosion Probe

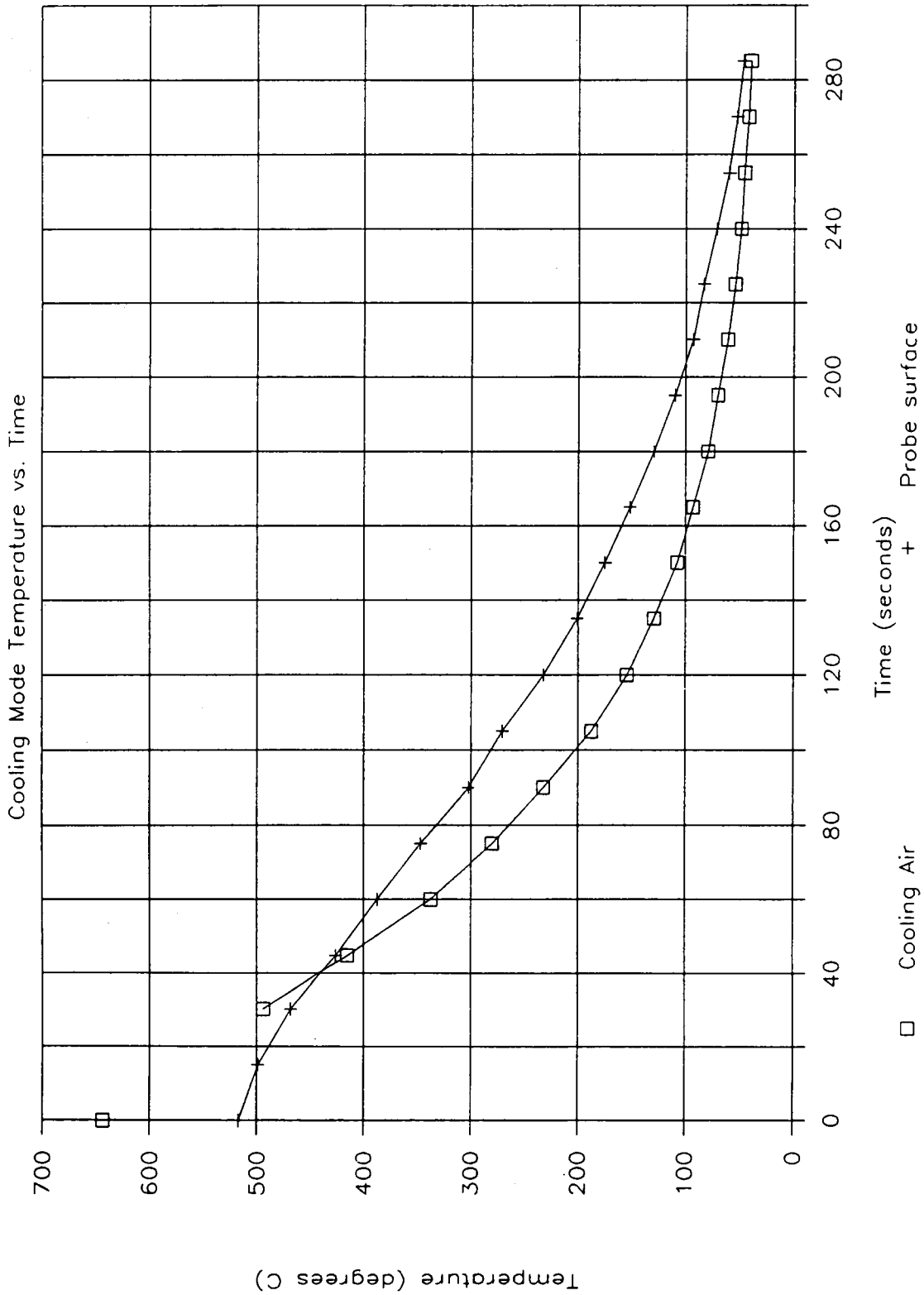


Figure 11. Cooling mode temperature vs time.

Flue Gas Corrosion Probe

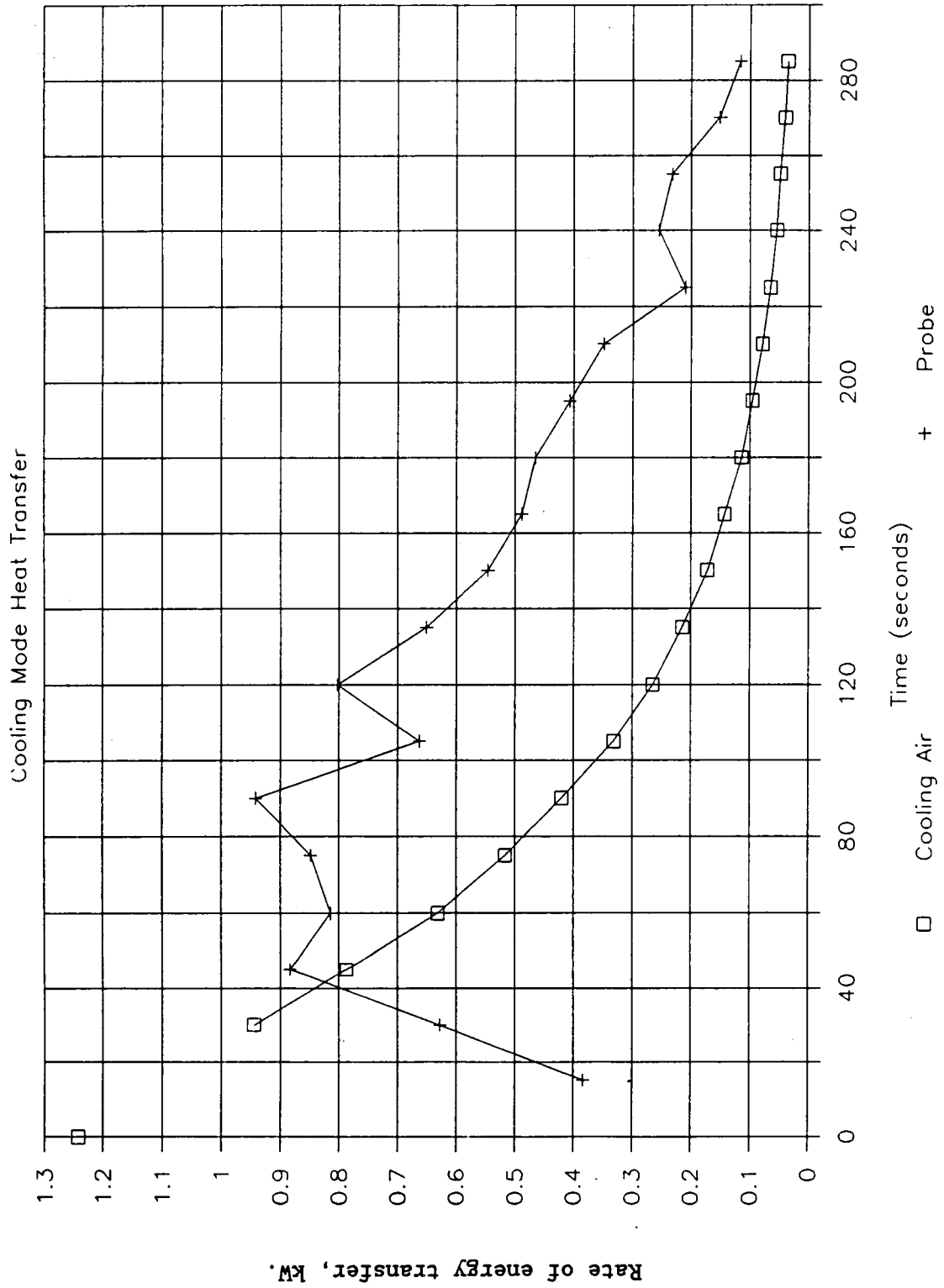


Figure 12. Cooling mode energy transfer.

7.2 Electric Heating Mode Performance

The probe was initially fitted with a modified silicon carbide heating element to evaluate heating with electricity. The modification consisted of coating the element with alumina cement to serve as an electrical insulator. This was necessary since it is possible, although unlikely, that the element can contact the inside surface of the probe cylinder. For this series of tests, both the probe and heating element were made of silicon carbide and could therefore form an electrical short circuit if they made direct contact. Efforts made to locate suitable ceramic tubing which could be slipped over the element, thereby providing a more convenient means of insulating, were not successful. A photograph of the coated electric heating element (with test specimen removed) is provided in Figure 13. Although initial calculations indicated that sufficient heat energy input could probably not be achieved with this particular element, the possibility of operating the element at higher than rated voltage to increase heat output and the fact that the element did not have to operate all of the time made it an option worth considering. The tests were performed to compare actual heat transfer values with values predicted by the models.

Measurements of the probe surface temperature were made as the probe was heated from room ambient air temperature, approximately 27°C, to the maximum attainable temperature, 494°C. The measurements were made at 15-s intervals. These values were then used with Equation (37a) to obtain estimates of the rate of energy transfer between the heating element and the probe. The relative time of

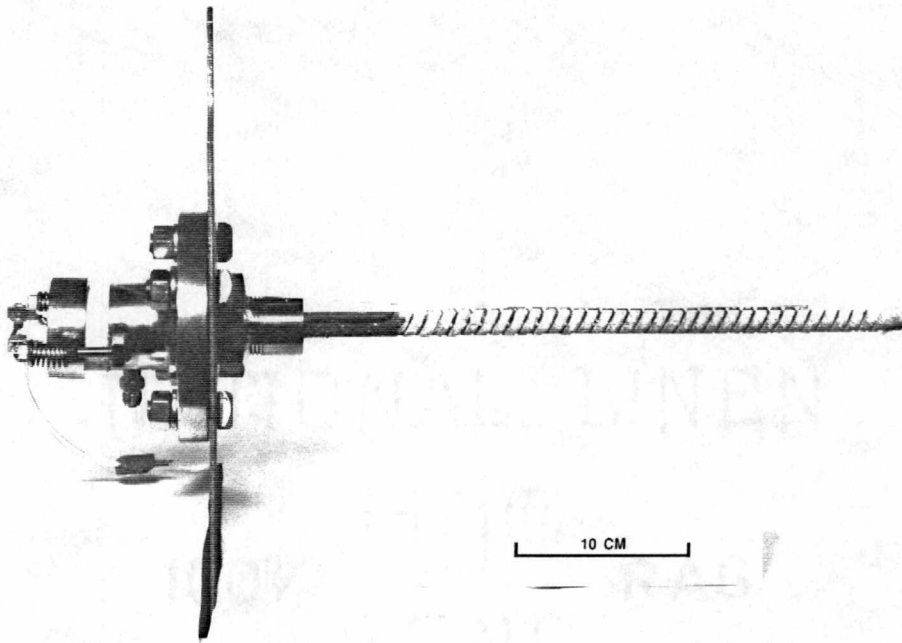


Figure 13. Electric heating element.

measurement, temperature values, and corresponding approximate energy transfer rates are shown in Table 8 and plotted in Figures 14 and 15.

The limitation on operating at a higher than rated voltage is watt density, or how fast energy can be transferred from the element surface to prevent it from overheating locally and burning out. Other types of heating elements can be considered, such as carbon. Of course, the problem with carbon is its oxidation at high temperatures. If the cooling gas is a suitable inert blanket gas such as argon, it can be supplied at a very low flow rate during the heating mode to shield the carbon from oxygen.

The data obtained during the electric heating mode tests indicate a slightly lower than expected rate of energy flow.

The test results show that, while electric heating is convenient, the present configuration does not provide sufficient heat energy input to compensate for a significant temperature difference between the probe and the flue gas.

7.3 Gas-Fired Heating Mode Performance

The gas heating assembly was fitted to the probe assembly to permit heating with a gas flame. Gas heating was employed to evaluate an extension of the operating regime of the probe in the heating mode.

The gas heating assembly is made from a modified Model 403 Prest-O-Lite air-acetylene torch unit manufactured by L-TEC Welding and Cutting Systems. The modification consists of replacing the curved piping portion of a No. 4, medium-type torch tip with a length

Table 8. Electric heating mode probe temperature/rate of energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	16	
15	17	0.03
30	21	0.08
45	27	0.12
60	37	0.21
75	56	0.40
90	78	0.47
105	107	0.60
120	138	0.66
135	174	0.76
150	214	0.84
165	249	0.73
180	285	0.74
195	313	0.58
210	342	0.62
225	366	0.49
240	384	0.40
255	404	0.41
270	418	0.30
285	431	0.26
300	442	0.23
315	451	0.20
330	459	0.17
345	465	0.12
360	469	0.08
375	473	0.09
390	477	0.07
405	480	0.07
420	482	0.05
435	485	0.06
450	487	0.03
465	489	0.05
480	492	0.06
495	493	0.02
510	493	0.01
525	494	0.02
540	494	0.00

Flue Gas Corrosion Probe

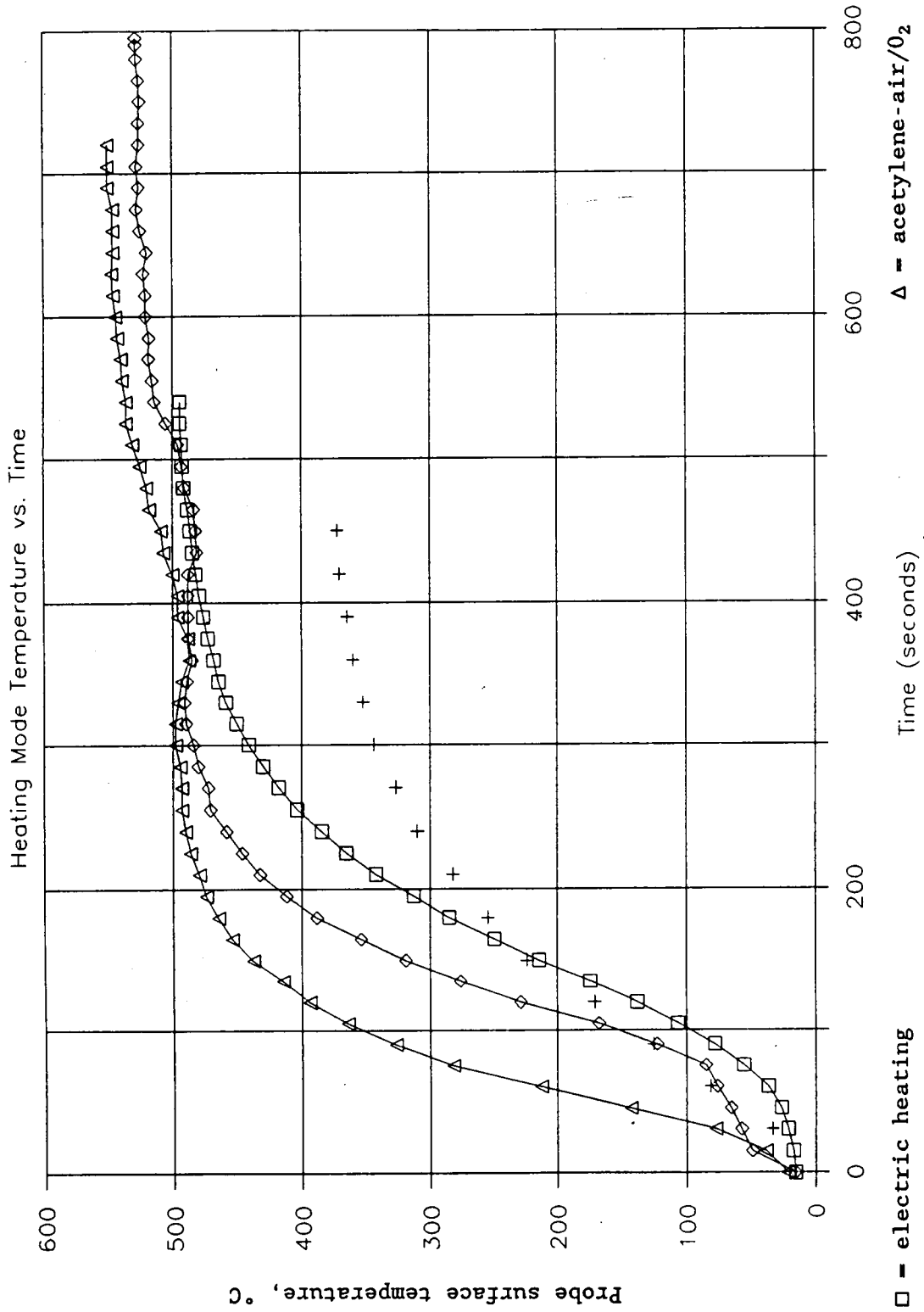
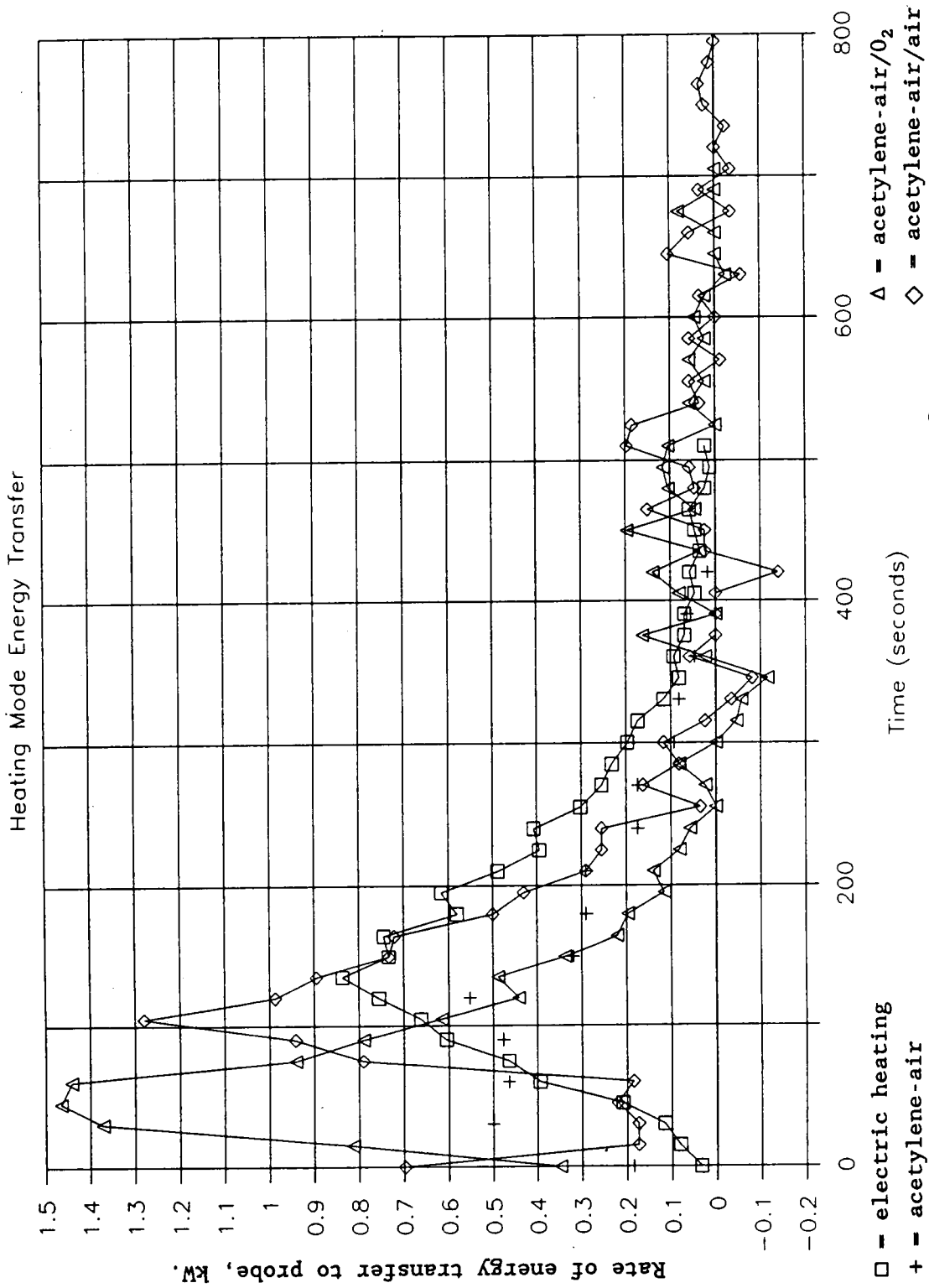


Figure 14. Heating mode temperature vs time.

Flue Gas Corrosion Probe



of straight pipe with an inside diameter of 6.4 mm. A photograph of the gas heating assembly is provided in Figure 16.

The burner assembly is designed such that it can be readily inserted in place of the electric element. The fuel gas supply connection is made on the left end of the tube (threaded portion) and the flame is produced on the right end. The right end protrudes into the probe assembly a distance of 8.6 cm. This results in the flame being positioned at the entrance to the probe specimen.

Initial gas-fired tests were performed using acetylene with normally entrained (air drawn into the burner assembly by the action of the fuel gas stream or jet) ambient air for combustion. An integral 3000-V spark gap assembly was used to ignite the gas mixture within the probe.

Subsequent gas-fired tests were performed with additional combustion air supplied to the interior of the probe through the cooling air inlet. This was necessary since, in normal operation, a torch unit such as that used is operated in open air where there is sufficient oxygen to allow complete combustion. When operated in a confined space, such as inside the probe, there is insufficient oxygen to allow complete combustion.

The probe was operated using both ordinary plant air and 100% oxygen as the supplementary gas. Plant air is compressed air at 689 kPa, which is available at most industrial facilities. This was most desirable as the supplementary gas because of its ready availability. Pure oxygen was considered also, since the additional

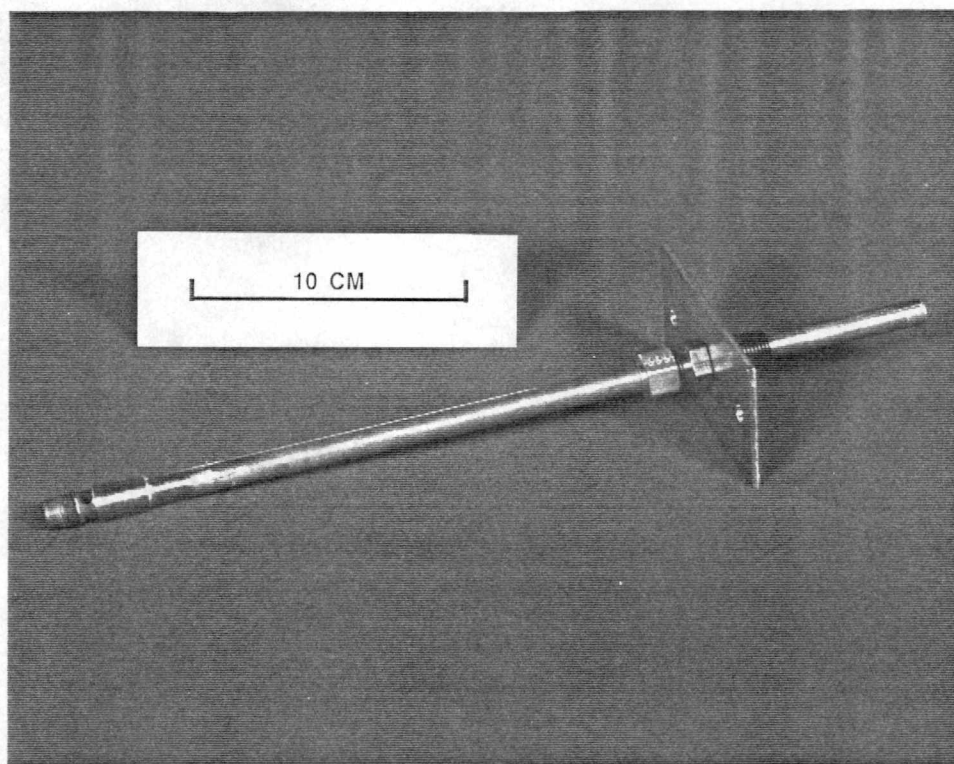


Figure 16. Gas heating burner assembly.

heat provided would offset the inconvenience of having to provide bottled gas.

The flow rate of acetylene fuel gas during each of these tests was nominally $4.21 \times 10^{-4} \text{ m}^3/\text{s}$. The flow rate of supplementary air was approximately $2.53 \times 10^{-3} \text{ m}^3/\text{s}$ during initial heating and $1.71 \times 10^{-3} \text{ m}^3/\text{s}$ at optimum combustion conditions. The flow rate of supplementary oxygen during similar operation was essentially the same.

The rate of energy input to the probe was estimated in the same manner as for cooling and heating with electricity, that is, by Equation (37a).

The relative time of measurement, temperature values, and corresponding approximate energy transfer rates are shown in Tables 9, 10, and 11. Negative values of energy transfer rates indicate that the probe is essentially at equilibrium and that small quantities of energy are being carried off due to air currents.

Results obtained from the steady state model indicate that an energy input of approximately 1.6 kW is required to compensate for a flue gas temperature that is 125°C below the probe setpoint. Data in Table 11 indicate that the acetylene-air flame with pure oxygen as the supplementary gas is capable of providing such a level of energy input to the probe. A possible alternative to utilizing oxygen to increase the available heat energy might be the substitution of an alternative fuel gas.

Table 9. Gas heating mode, acetylene-air, probe temperature/
rate of energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	16	
30	33	0.19
60	81	0.50
90	126	0.47
120	171	0.48
150	224	0.55
180	254	0.32
210	282	0.29
240	310	0.29
270	327	0.17
300	343	0.17
330	352	0.09
360	360	0.08
390	364	0.05
420	371	0.06
450	372	0.02

Table 10. Gas heating mode, acetylene-air/air, probe temperature/
rate of energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	16	
15	49	0.70
30	57	0.17
45	66	0.17
60	76	0.22
75	85	0.19
90	120	0.79
105	168	0.94
120	229	1.28
135	276	0.99
150	319	0.90
165	354	0.73
180	388	0.72
195	412	0.50
210	433	0.43
225	447	0.29
240	459	0.26
255	471	0.26
270	473	0.03
285	481	0.16
300	484	0.08
315	490	0.12
330	491	0.02
345	489	-0.03
360	486	-0.08
375	488	0
390	488	0
405	488	0
420	488	0
435	482	-0.14
450	483	0.02
465	484	0.02
480	491	0.15
495	493	0.05
510	496	0.06
525	506	0.20
540	514	0.19
555	516	0.03
570	519	0.06
585	518	-0.01
600	521	0.06
615	521	0
630	523	0.03
645	520	-0.06

Table 10. (Continued)

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
660	525	0.10
675	528	0.06
690	526	-0.03
705	528	0.03
720	526	-0.03
735	526	0
750	525	-0.02
765	526	0.02
780	528	0.03
795	528	0
810	528	0

Table 11. Gas heating mode, acetylene-air/O₂, probe temperature/rate of energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	22	
15	38	0.35
30	77	0.81
45	143	1.37
60	213	1.46
75	282	1.44
90	327	0.94
105	364	0.79
120	394	0.62
135	415	0.44
150	438	0.49
165	454	0.34
180	465	0.22
195	474	0.20
210	480	0.12
225	487	0.14
240	491	0.08
255	493	0.06
270	493	0
285	494	0.02
300	498	0.08
315	498	0
330	496	-0.05
345	493	-0.06
360	488	-0.12
375	489	0.02
390	497	0.16
405	497	0
420	501	0.08
435	507	0.14
450	509	0.03
465	518	0.20
480	521	0.05
495	526	0.1
510	531	0.12
525	536	0.1
540	536	0
555	539	0.06
570	540	0.02
585	543	0.06
600	544	0.02
615	546	0.05
630	547	0.02
645	546	-0.02

Table 11. (Continued)

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
660	546	0
675	546	0
690	550	0.08
705	550	0
720	550	-0.03

The substitution of propane gas for acetylene as the fuel gas would be expected to increase the available energy. This is based on the heat value of acetylene being 45,223 joule/m³ and that of propane being 72,882 joule/m³, an increase of ~61%. These heat values are low (net) values based on 0°C, 760 mm Hg and are used for comparison only. Propane gas was not used during the initial testing because it was unavailable. It is presumed, however, that, owing to the ready commercial availability of propane, it would certainly be available at most test sites.

The flame pattern within the probe was considered to be fairly laminar, owing to the orientation of the burner and the flow rates. An enhancement that would certainly improve the rate of heat transfer to the probe cylinder would be to incorporate a diffuser in the flame path. This could be fabricated from a material such as silicon carbide and, in fact, could be a portion of a modified electric heating element such as that used for electric heating. Although this addition might slow the response time (adding more thermal mass), it would certainly introduce turbulence into the flow stream and decrease the boundary layer on the inside surface of the probe.

Graphs of the temperature versus time and energy transfer rate data contained in Tables 8, 9, 10, and 11 for each of the heating methods are shown in Figures 14 and 15. (see pages 45 and 46). As can be seen from these figures, the acetylene-air/O₂ mixture, that is, acetylene gas as the fuel mixed with normally induced ambient air for primary combustion with 100% O₂ added to the hot gas stream for complete combustion, yields the fastest rate of temperature rise (and

therefore the highest rate of energy transfer). The upward trend at approximately 400 s is the result of decreasing the O_2 flow rate from $2.6 \times 10^{-3} \text{ m}^3/\text{s}$ to $1.8 \times 10^{-3} \text{ m}^3/\text{s}$. It is evident that a flow rate higher than that required for complete combustion results in a cooling effect similar to that when the probe is operating in the cooling mode. The supplementary oxygen (or air) flow rate could be regulated in proportion to the fuel gas by the addition of another control valve operated in a manner similar to the fuel gas flow control valve.

As mentioned earlier, propane is expected to yield a higher rate of energy input, owing to its higher heat value. However, since the rate of heat energy transfer depends directly on the relative temperature difference between the objects--in this case, the hot gas stream and the inside surface of the probe--the higher theoretical flame temperature of the acetylene flame (2632°C) versus propane (2115°C) may be of more benefit.

All measurements represented in Figures 14 and 15 were made with the probe in still ambient air at a nominal 27°C . The measurements were made under room ambient conditions to determine the relative performance of various methods prior to installation in an actual field environment.

8. BOILER FIREBOX TEST RESULTS

The heating and cooling capacity and rate measurements made on the assembled probe in a laboratory environment support the calculations used in the design of the probe. In an actual flue environment, however, there is significant radiant heat energy emanating from the flue gas and flue walls. Also, the flue gas circulating over the surface of the probe results in convection heat transfer which will either heat or cool the specimen.

The increase in temperature experienced during the firing of the alumina cement used to retain the thermocouple confirmed that the probe heat source was capable of elevating the probe temperature above the ambient environment. The lack of a high temperature, flowing gas stream over the surface of the probe made it desirable to evaluate the probe in a more realistic environment. This was accomplished by inserting the probe into the viewing port of an industrial steam plant boiler firebox. This exposed the probe to both hot flowing gases and high levels of radiant heat energy. The approximate conditions present in the firebox are shown in Table 12.

Table 12. Boiler firebox conditions.

Parameter	Symbol	Value
Mass flow rate of gases	m_B	6.3 kg/s
Average density	ρ_B	0.32 kg/m ³
Estimated cross-sectional area of firebox	A_B	13.9 m ²
Firebox temperature	T_B	920°C

The probe was inserted through the viewing port with the boiler at full operation. Measurements of the probe surface temperature were started as soon as the probe was inserted and continued at 15-s intervals until the probe reached steady state temperature. These values were recorded and used with Equation (37a) to estimate energy input to the probe. The relative time of measurement, temperature, and corresponding rates of energy transfer are provided in Table 13.

The electric heating element was then energized and temperature measurements made each 15 s until the probe reached a new steady state temperature. These values were also used in Equation (37a). The relative time of measurement, temperature, and corresponding rates of energy transfer values for this test are provided in Table 14.

The electric element was then deenergized and cooling air supplied to the probe. The cooling air was obtained from the building plant air supply through a regulator. The cooling air flow rate was approximately $2.77 \times 10^{-3} \text{ m}^3/\text{s}$. Measurements of the probe surface temperature were started as soon as the cooling air was turned on and continued at 15-s intervals until the probe again reached constant temperature. The values obtained during this test were recorded and used in Equation (37a). The relative time of measurement, temperature, and corresponding energy transfer values for this test are provided in Table 15.

Figure 17 shows the temperature response experienced through each of the three measurement periods as contained in Tables 13, 14, and 15. As can be seen, the probe reached a maximum temperature of approximately 910°C under normal firebox conditions. The electric

Table 13. Boiler installed temperature response/energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	30	
15	99	1.35
30	260	3.14
45	501	4.71
60	627	2.46
75	699	1.41
90	750	1.00
105	787	0.72
120	815	0.55
135	835	0.39
150	851	0.31
165	864	0.25
180	875	0.21
195	879	0.08
210	892	0.25
225	900	0.16
240	900	0
255	902	0.04
270	907	0.10
285	912	0.10
300	915	0.06
315	913	-0.04

Table 14. Boiler installed heating mode temperature response/
energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	909	
15	918	0.16
30	938	0.39
45	954	0.31
60	963	0.18
75	973	0.20
90	977	0.08
105	982	0.10
120	986	0.08
135	987	0.02
150	983	-0.08
165	980	-0.06
180	980	0
195	980	0

Table 15. Boiler installed cooling mode temperature response/
energy transfer.

Time (s)	Probe External Surface Temperature (°C)	Rate of Energy Transfer (kW)
0	930	
15	930	0
30	917	-0.25
45	911	-0.12
60	890	-0.41
75	870	-0.39
90	850	-0.39
105	841	-0.18
120	828	-0.25
135	818	-0.20
150	807	-0.21
165	803	-0.08
180	798	-0.10
195	795	-0.06
210	798	0.06
225	800	0.04
240	799	-0.02
255	793	-0.12

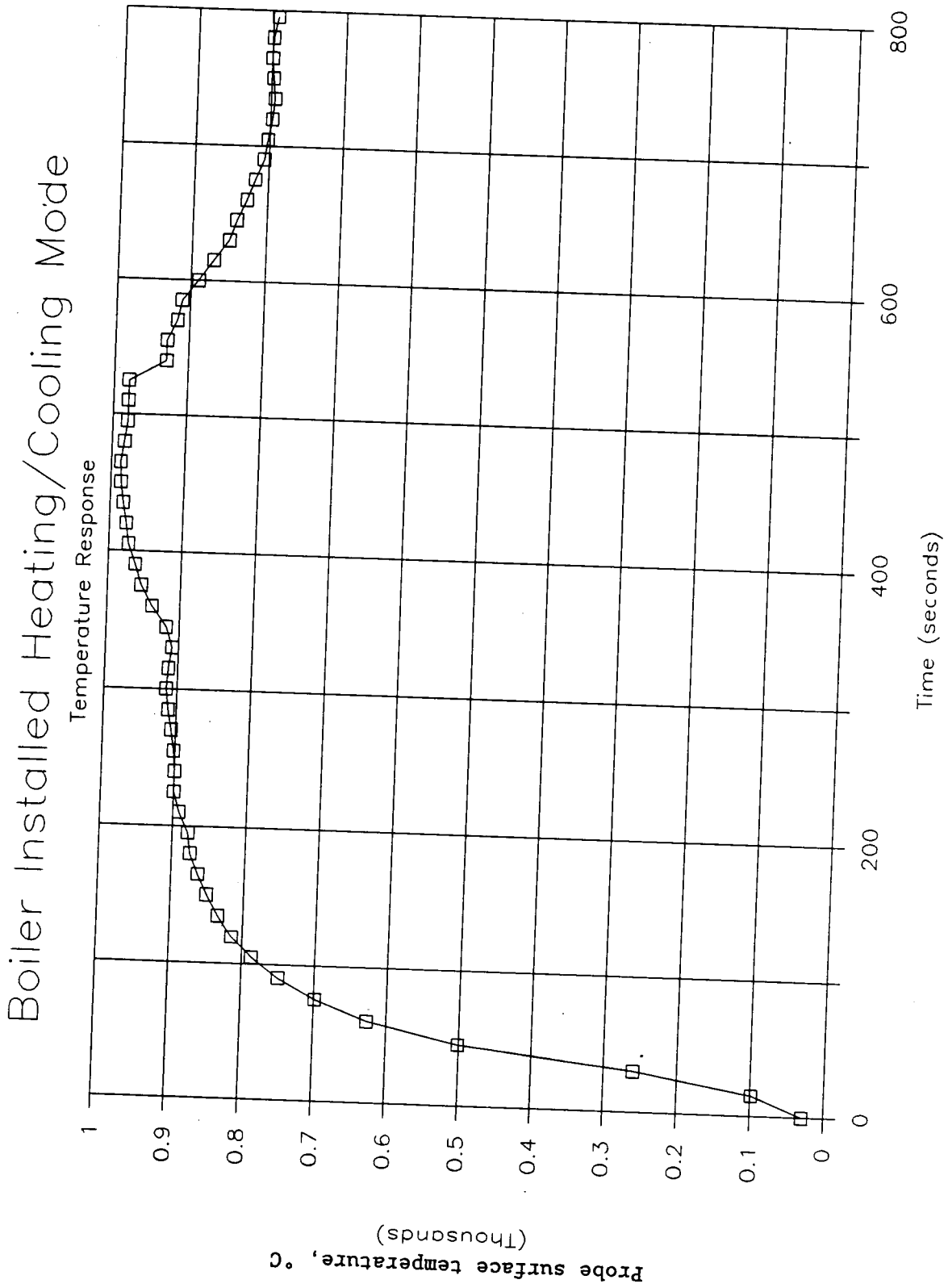


Figure 17. Boiler installed temperature response.

heating element was capable of elevating the temperature an additional 90°C, and the cooling air was able to depress the temperature by a little over 100°C.

A plot of energy transfer data. (Tables 13, 14, and 15) during the three test phases is provided in Figure 18. This is particularly useful for comparing with the maximum available energy transfer rates from the laboratory test.

During the initial heating phase, the probe received a maximum of 5 kW of heat energy. The electric element supplied a maximum of approximately 0.5 kW, and the cooling air reached a maximum (negative) value of approximately 0.5 kW.

Gas heating was not used during these tests because of time and equipment constraints.

The results obtained indicate that it is reasonable to expect to adequately elevate and depress corrosion probe temperature using combustible fuel gases and higher cooling air flow rates.

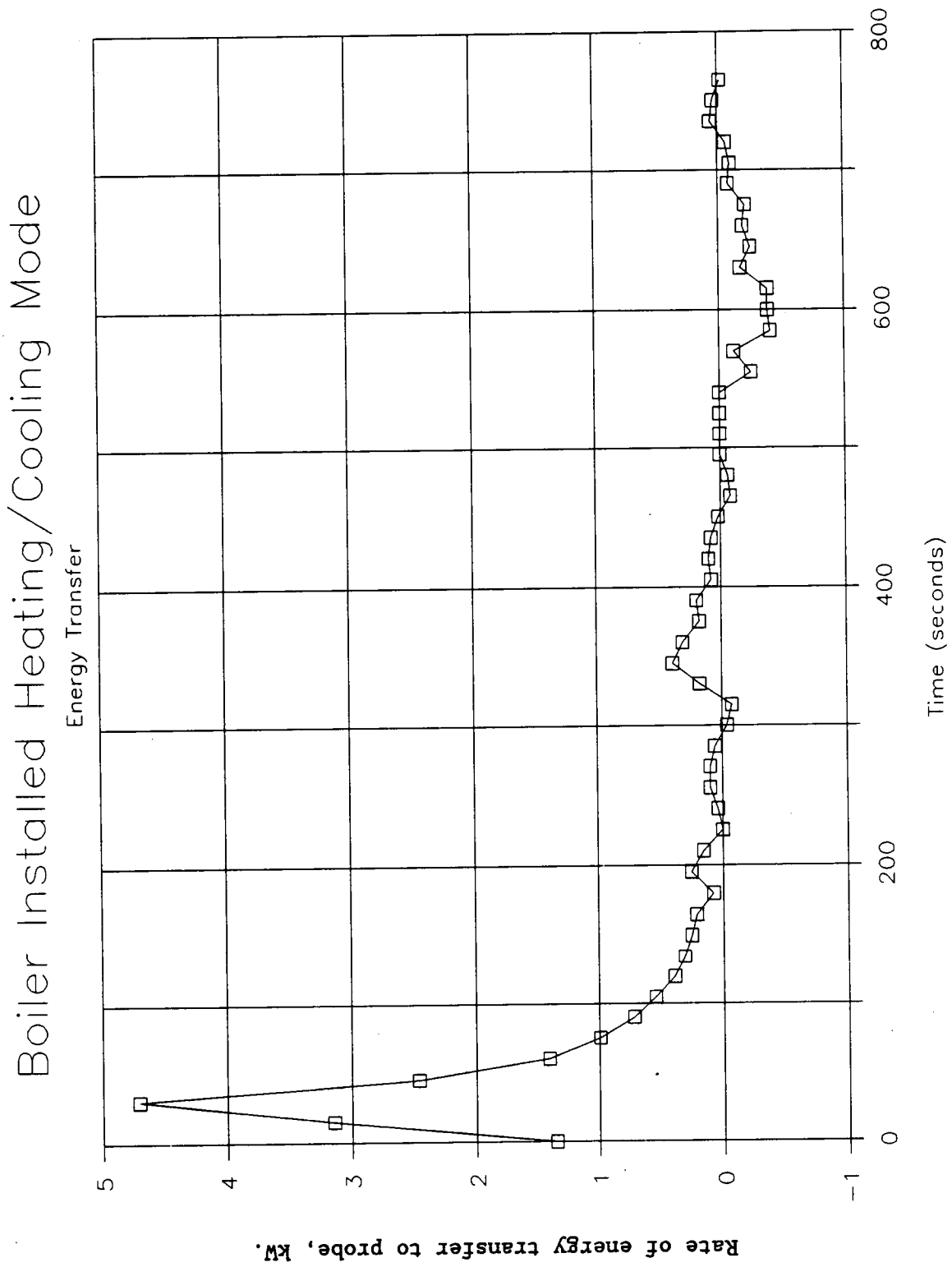


Figure 18. Boiler installed energy transfer.

9. RESULTS, CONCLUSIONS, AND RECOMMENDATIONS

The initial design basis reference conditions and anticipated flue gas temperature excursions about the probe setpoint indicate that the probe could be exposed to a maximum 550°C temperature excursion during uncontrolled operation.

Results of the steady state analysis of probe performance requirements show that a heat energy transfer rate (power level) of approximately 3.2 kW would be required to compensate for an absolute maximum probe to flue gas temperature differential of 540°C under anticipated flue conditions, gas/flue temperature and flue gas density and velocity.

The dynamic model analysis revealed that a cooling air flow rate of 0.028 m³/s would permit the probe to operate at a temperature 500°C below the flue gas temperature under reference conditions. A step change in the probe setpoint from 29°C to 1200°C required a maximum rate of energy input from the electric heating element of 1.1 kW to achieve a probe temperature 200°C above the ambient flue temperature.

Results of laboratory tests indicate that it is possible to achieve energy transfer rates in excess of the 0.95 kW experienced in the cooling mode. This conclusion is based on the fact that the cooling air flow used during the tests was 0.0037 m³/s, which is significantly below the maximum achievable flow. These tests also showed that electric heat could elevate the probe temperature approximately 540°C above the surrounding environment when in still ambient room air. Results obtained during the firing of the alumina cement used to secure the thermocouple, that is, while the probe was

being internally heated by the electric element and externally by a bench top furnace capable of reaching 1010°C , showed that the probe could be elevated almost 200°C above ambient. Of course, the fact that these were still air conditions has a significant effect on the rate of energy transfer. The maximum effective energy transfer rate to the probe during electric heating in the laboratory was 0.82 kW.

Laboratory tests conducted using gas heating resulted in an increase in the effective rate of energy transfer to the probe from 0.82 kW to 1.44 kW. This confirmed that gas heating was the more effective method for accommodating significant flue gas temperature excursions. While the gas heating method does provide more power, it is the more inconvenient and costly method of heating the probe. The ready availability of propane gas for fuel and plant air as a source for both primary and supplementary combustion air should make this a reasonable and practical method.

Actual field tests of the probe installed through a viewport in the firebox of an industrial steam plant boiler show that the present controlled temperature flue gas corrosion probe design is both practical and functional. The conditions present in the firebox are considered to be reasonably representative of conditions considered in the design basis. The probe temperature was elevated approximately 90°C above the firebox temperature using electric heating and depressed approximately 100°C below the firebox temperature. Gas heating was not used during the firebox tests because a suitable fuel supply was not readily available. Although the temperature depression

was only 100°C, higher volumetric flow rates of cooling air are certainly possible with the current probe design.

It can be concluded from the controlled temperature flue gas corrosion probe design, fabrication, and testing activities described herein that a simple, low cost, portable probe and control system design has been achieved. Although the probe has not been tested over the full temperature range defined in the design basis, it is possible both to heat and cool the probe over a reasonable range of temperatures, as may be encountered in an industrial flue environment using generally available fuel gas and cooling/combustion air supplies.

Recommendations for further improvements in the design include: utilizing the largest possible gas burner that can be accommodated by the inside diameter of the probe, incorporating a method for introducing additional turbulence in the hot gas and cooling air streams as they pass through the probe interior to improve energy exchange, and examining higher power capacity electric heating elements. Although the specimen is currently attached to the mount by means of a single retaining screw, a threaded adapter can be fitted to permit screwing of samples into the probe mount.

10. SUMMARY

A need to evaluate candidate waste heat recovery materials under essentially constant temperature conditions has been identified. Appropriate modeling and design calculations demonstrate the feasibility of constructing a controlled-temperature corrosion probe. These calculations also indicate that sufficient energy may not be available from existing electric heating elements. While the initial configuration utilized such an element, provisions have been made to incorporate a gas burner as an alternative heat source to permit a broader operating regime in the heating mode.

A corrosion probe was fabricated according to the conceptual design described herein and tested in an industrial boiler firebox. Fabrication of the probe revealed that there were few, if any, assembly problems and provided a prototype for laboratory testing and simulated furnace flue testing using natural gas combustion products as the flue gases. The responsiveness of the heat source and cooling air to various flue gas temperatures is considered to be adequate, and control system configuration and performance are suitable for installation and testing in an actual industrial flue gas environment.

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APPENDICES

APPENDIX A
SIGNAL SPLITTER SCHEMATIC

* DENOTES DAVEN 1350 SERIES 0.5%
20 ppm/deg C WIREWOUND RESISTORS.
OTHER RESISTORS 1% MF, CAPACITORS 5

APPENDIX B
STEADY STATE MODEL LISTING

Temperature control:

The temperature control for the probe is maintained through regulating the flow of probe cooling gas, usually air, combined with the level of input power to the heating element located within the specimen (refer to figure 1). The cooling gas serves to carry off excess heat energy when the flue gas is at a higher temperature than the probe setpoint while the heating element serves to add additional heat energy when the flue gas temperature is below the setpoint. A "split-range" controller arrangement is employed to permit control in this configuration. The temperature on the inside surface of the test specimen is measured by a type K thermocouple. A conventional single loop PID controller is used to sense the specimen temperature, compare it with the desired setpoint and output an appropriate control signal. The control signal is sent to two converters in parallel. The converter which drives the cooling gas flow rate is configured as a reverse acting unit such that an input signal of 4 milliamps from the controller produces an output signal from the converter of 20 milliamps and an input signal greater than or equal to 12 milliamps produces an output signal of 4 milliamps to the control valve. The converter which controls the heater input power is configured such that a controller signal of less than or equal to 12 milliamps produces a signal of 4 milliamps and a controller signal of 20 milliamps produces a signal of 20 milliamps to the power control.

Probe energy balance:

The temperature of the test specimen is the algebraic result of the total energy received by the specimen and the total energy leaving the specimen. The sources of energy into the specimen are the flue gas and the heater. The paths by which energy leaves the specimen are the cooling gas and the flue gas.

Energy flow from flue gas:

With regard to the energy received from the flue and flue gas, the governing equations are shown in the following example:

we first define the following quantities:

for the flue gas:

corresponding viscosity of flue gas; $\mu_g := 3.45 \cdot 10^{-6} \frac{\text{lbm}}{\text{ft sec}}$

density of flue gas; $\rho_g := 0.0161 \frac{\text{lbm}}{\text{ft}^3}$

thermal conductivity; $k_g := \left[\frac{0.0471}{3600} \right] \frac{\text{BTU}}{\text{sec ft degF}}$

mass flow rate (at burn level); $\dot{m}_g := 3.50 \frac{\text{lbm}}{\text{sec}}$

volume flow rate; $Q_g := \frac{\dot{m}_g}{\rho_g}$

for the flue:

width of the flue; $W_f := 4 \text{ ft}$

length of the flue (not height); $L_f := 4 \text{ ft}$

corresponding cross sectional area; $A_f := W_f L_f$

outside diameter of probe: $D_p := \left[\frac{1}{12} \right] \text{ ft}$

length of probe: $L_p := \left[\frac{15}{12} \right] \text{ ft}$

outside surface area: $A_{po} := \pi D_p L_p$

the velocity of the flue gas is then given by: $v_g := \left[\frac{Q_g}{A_f} \right]$

$v_g = 13.587 \frac{\text{ft}}{\text{sec}}$

The Reynolds number for the flue gas stream, relative to the probe, under these conditions is:

$$R_d := \frac{\rho_g V D}{\mu_g} \quad \text{or,} \quad R_d = 528.382$$

The convective heat transfer film coefficient at the surface of the probe is then given by:

$$\text{for } R_d = 528.382 \quad ; \quad C := 0.615 \quad \text{and} \quad n := 0.466$$

with C and n being obtained from the following table:

R_d	C	n
0.4-4	0.891	0.330
4-40	0.821	0.385
40-4,000	0.615	0.466
4,000-40,000	0.174	0.618
40,000-400,000	0.0239	0.805

$$h_{cav} := \frac{k_g C R_d^n}{D_p} \quad \text{or,} \quad h_{cav} = 0.002 \frac{\text{BTU}}{\text{ft}^2 \text{ degF sec}}$$

and, the total convective energy flow across the film boundary, in BTU/sec, is given by:

$$q_{gc} [T_g, T_p] := [T_g - T_p] h_{cav} A_{po}$$

The other mechanism for heat energy transfer to the probe is by means of radiation from the flue walls as shown by the following:

assuming that the flue walls are at the temperature of the flue gas (i.e.; steady state) and defining appropriate quantities:

Stefan-Boltzmann constant; $\sigma := \frac{1.73 \cdot 10^{-9}}{3600} \frac{\text{BTU}}{\text{ft}^2 \text{ sec degR}^4}$

emissivity of the probe surface; $\epsilon_p := 0.8$

reflectance of the probe surface; $r_p := 0.2$

emissivity of the flue wall; $\epsilon_{fw} := 0.8$

reflectance of the flue wall; $r_{fw} := 0.2$

geometric view factors;

$F_{23} := 1$ $F_{33} := 1$ $F_{32} := 0$

the radiosity or total power per unit area leaving the probe is given by:

$$R_2 \left[\begin{matrix} T_g \\ T_p \end{matrix} \right] := \frac{\epsilon_p \sigma T_p^4 \left[1 - r_{fw} F_{33} \right] + r_p F_{23} \epsilon_{fw} \sigma T_g^4}{1 - r_{fw} F_{33} - r_p^2 F_{23} F_{32}}$$

$$q_{gr} \left[\begin{matrix} T_g \\ T_p \end{matrix} \right] := \frac{\epsilon_p}{r_p} \left[R_2 \left[\begin{matrix} T_g \\ T_p \end{matrix} \right] - \sigma T_p^4 \right] A_{po}$$

The total energy supplied to the probe by both convection and radiation from the gas and flue walls is then:

$$q_{gt} \left[\begin{matrix} T_g \\ T_p \end{matrix} \right] := q_{gc} \left[\begin{matrix} T_g \\ T_p \end{matrix} \right] + q_{gr} \left[\begin{matrix} T_g \\ T_p \end{matrix} \right]$$

for the maximum and minimum cases:

$$q_{gtmin} := q_{gt} (1380 \text{ degF}, 2400 \text{ degF})$$

$$q_{gtmax} := -1 \cdot q_{gtmin}$$

$$q_{gtmin} = -4.316 \frac{\text{BTU}}{\text{sec}}, \quad q_{gtmax} = 4.316 \frac{\text{BTU}}{\text{sec}}$$

Energy supplied from heater:

When the flue gas is operating below the setpoint for the probe, additional heat must be added in the form of electrical energy to the heating element. The maximum amount of electrical power required as a function of flue gas temperature and probe setpoint can be determined as follows:

Since the heater is completely enclosed by the cylinder, they are treated as a single unit. From the preceding calculations, under steady state conditions, in order to hold the probe at a desired temperature when the gas temperature is below the setpoint, it is necessary to replace the energy being carried off by the gas. The worst case is when the probe temperature is at or near the highest value and the gas is at the lowest value. From above, the most energy is removed at:

$$q_{gtmin} = -4.316 \frac{\text{BTU}}{\text{sec}} \quad \text{or,} \quad q_{gtmin} = -4.55 \cdot 10^3 \text{ Watts}$$

Therefore, the maximum energy required from the heater is then:

$$q_{Hmax} = -1 q_{gtmin} \quad \text{or,}$$

$$q_{Hmax} = 4.55 \cdot 10^3 \text{ Watts}$$

over the range of flue gas operating temperatures:

$$T_g = 2400 \text{ degF}$$

$$T_p = 1380 \text{ degF, } 1460 \text{ degF, } 2400 \text{ degF}$$

$q_{st} \left[\frac{T}{s}, \frac{T}{p} \right]^{-1}$	$T - T_{ps}$
Watts	degF
$4.55 \cdot 10^3$	$1.02 \cdot 10^3$
$4.379 \cdot 10^3$	940
$4.186 \cdot 10^3$	860
$3.969 \cdot 10^3$	780
$3.726 \cdot 10^3$	700
$3.452 \cdot 10^3$	620
$3.147 \cdot 10^3$	540
$2.806 \cdot 10^3$	460
$2.427 \cdot 10^3$	380
$2.007 \cdot 10^3$	300
$1.541 \cdot 10^3$	220
$1.027 \cdot 10^3$	140
460.921	60

Since the heaters currently being considered have a maximum output on the order of 1800 Watts, the maximum heating mode temperature difference is approximately 250 degrees F. It is expected that this range can be expanded as new elements are examined.

Energy removed by cooling gas stream:

When the flue gas is operating above the setpoint for the probe, it is necessary to remove excess energy by means of the cooling gas stream. The worst case cooling gas requirement occurs when the flue gas is at the highest reasonable temperature and the probe setpoint is at the lowest reasonable value. This condition is examined in the following example:

defining appropriate quantities:

nominal air inlet temperature; $T_{in} := 80 \text{ degF}$
 nominal density of air; $\rho_{air} := 0.02 \frac{\text{lbm}}{\text{ft}^3}$

nominal thermal conductivity of air; $k_{\text{air}} := \left[\frac{.0471}{3600} \right] \frac{\text{BTU}}{\text{sec ft degF}}$

nominal viscosity of air; $\mu_{\text{air}} := 3.45 \cdot 10^{-5} \frac{\text{lbm}}{\text{ft sec}}$

thickness of cylinder; $t_{\text{cyl}} := \frac{1}{8 \cdot 12} \text{ ft}$

radius of inside of cylinder; $R_{\text{ci}} := \frac{D_p - 2 t_{\text{cyl}}}{2}$

radius of the heating element; $R_{\text{heater}} := \frac{0.25}{12} \text{ ft}$

nominal cooling gas specific heat; $C_{p_{\text{air}}} := 0.292 \frac{\text{BTU}}{\text{lbm degF}}$

cross sectional area of heater/cylinder annulus;

$$A_d := \pi \left[R_{\text{ci}}^2 - R_{\text{heater}}^2 \right]$$

average radius of duct; $R_{\text{avg}} := \frac{R_{\text{ci}} + R_{\text{heater}}}{2}$

the amount of energy which needs to be removed for the possible range of probe operating temperatures is:

$T_p := 1380 \text{ degF}, 1460 \text{ degF} \dots 2400 \text{ degF}$

$T_s := 2400 \text{ degF}$

q	T	T	sec
ft	g	p	BTU
	4.316		
	4.154		
	3.971		
	3.765		
	3.534		
	3.275		
	2.985		
	2.662		
	2.302		
	1.903		
	1.462		
	0.974		
	0.437		

T	- T
g	p
	degF
	3
1.02	10
940	
860	
780	
700	
620	
540	
460	
380	
300	
220	
140	
60	

T	P
	degF
	3
1.38	10
1.46	10
	3
1.54	10
	3
1.62	10
	3
1.7	10
	3
1.78	10
	3
1.86	10
	3
1.94	10
	3
2.02	10
	3
2.1	10
	3
2.18	10
	3
2.26	10
	3
2.34	10

determining the film coefficient between the heater and cooling air:

$$D_{\text{equiv}} := 2 \frac{A_d}{\pi \left[R_{ci} + R_{\text{heater}} \right]}$$

$$R_{Dair} [Q_{air}] := \frac{\rho_{air} \frac{Q_{air}}{A_d} D_{deqiv}}{\mu_{air}}$$

values of the Reynolds number over the expected air flow range are:

$$Q_{air} := 1 \frac{ft^3}{Min}, 20 \frac{ft^3}{Min} \dots 200 \frac{ft^3}{Min}$$

R _{Dair}	[Q _{air}]
118.098	
3	
2.362	10
3	
4.606	10
3	
6.85	10
3	
9.094	10
4	
1.134	10
4	
1.358	10
4	
1.583	10
4	
1.807	10
4	
2.031	10
4	
2.256	10

$$C := 0.174 \quad n := 0.618$$

$$h_1 [Q_{air}] := \frac{k_{air} C \left[\rho_{air} \frac{Q_{air}}{A_d} \frac{D_{deqiv}}{\mu_{air}} \right]^n}{D_{deqiv}}$$

and between the cylinder and air:

$$h_2 [Q_{air}] := h_1 [Q_{air}]$$

$$H_1 [Q_{air}] := [h_1 [Q_{air}] R_{heater}]$$

$$H_2 [Q_{air}] := [h_2 [Q_{air}] R_{ci}]$$

$$\tau [Q_{air}] := \rho_{air} Q_{air} C_{p_{air}}$$

defining the input forcing function;

$$\phi [Q_{air}, T_{heater}, T_{cyl}] := \frac{2\pi}{\tau [Q_{air}]} [H_1 [Q_{air}] T_{heater} + H_2 [Q_{air}] T_{cyl}]$$

defining the coefficient for the steady state output term;

$$\beta [Q_{air}] := [H_1 [Q_{air}] + H_2 [Q_{air}]]^2 \frac{\pi}{\tau [Q_{air}]}$$

the temperature of the cooling gas as a function of distance along the cylinder is then:

$$f[x, Q_{air}] := \left[1 - e^{-1 \beta [Q_{air}] x \text{ ft}} \right]$$

the average or bulk temperature of the cooling air at the exit from the probe as a function of the flow rate is:

$$\alpha [Q_{air}] := \left[\frac{1}{L/P} \right] \int_0^{\frac{L}{P}} f[x, Q_{air}] dx \text{ ft}$$

$$T_{\text{avg}} [Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}] := \frac{\phi [Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}]}{\beta [Q_{\text{air}}]} \alpha [Q_{\text{air}}]$$

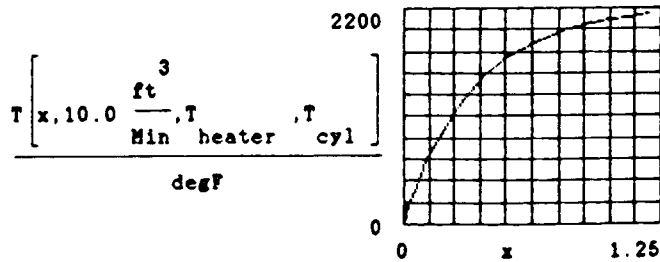
the rate of rise in temperature of the cooling gas along the length of the probe is shown for an arbitrary flow rate:

$$T_p := 2200 \text{ degF}$$

$$x := 0, .1 \dots 1.25$$

$$T[x, Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}] := \frac{\phi [Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}]}{\beta [Q_{\text{air}}]} f[x, Q_{\text{air}}]$$

$$T_{\text{cyl}} := T_p \quad T_{\text{heater}} := T_p$$



defining the rate of energy transfer by the cooling gas as a function of the flow rate:

$$q_{\text{cg}} [Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}] := r [Q_{\text{air}}] [T_{\text{in}} + T_{\text{avg}} [Q_{\text{air}}, T_{\text{heater}}, T_{\text{cyl}}]]$$

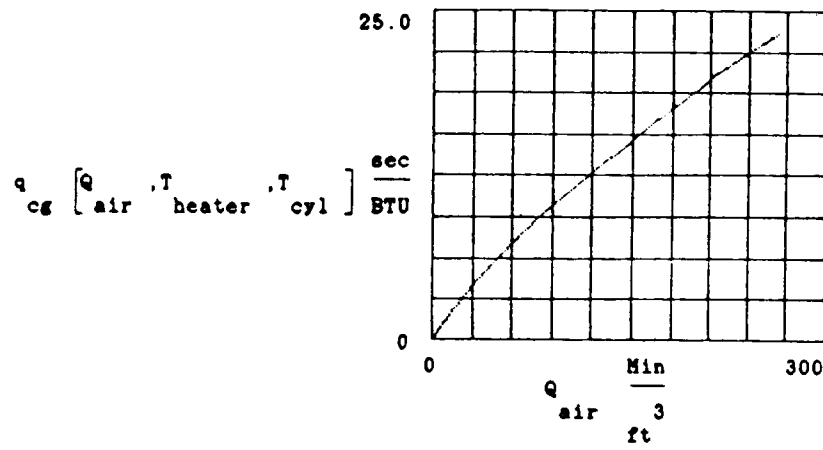
now, we can determine the rate of energy removal over the following range of cooling air flow rate:

$$Q_{\text{air}} := 1 \frac{\text{ft}^3}{\text{Min}}, 30 \frac{\text{ft}^3}{\text{Min}} \dots 300 \frac{\text{ft}^3}{\text{Min}}$$

Min		T				T	
Q	air	q	Q	T	heater	T	cyl
ft	3	cg	air	heater	cyl	degF	
1			0.187			1.949	10
30			4.216				3
59			7.33			1.364	10
88			10.075				3
117			12.592			1.196	10
146			14.946				3
175			17.176			1.096	10
204			19.306				3
233			21.353			1.026	10
262			23.329			971.744	
291			25.245			928.36	
						892.281	
						861.531	
						834.832	
						811.309	

By comparing the maximum value for heat energy removal by the cooling gas from the above table to the required rate of energy removal relative to the difference between the flue gas temperature and the desired probe temperature as defined in the preceeding table, we can determine the maximum, reasonable operating span during cooling.

Therefore, a reasonable span would seem to be 620 degrees F differential at a cooling air flow rate of 290 actual cubic feet/minute.



APPENDIX C
DYNAMIC MODEL LISTING

```

PROGRAM FLUE GAS PROBE
"WRITTEN BY: J. A. MCEVERS, 7/2/87"

"CALL DEBUG"

"MACRO -- AIR CONTROLL LIMITER"
MACRO ACLIM(ACOUT,CNTLI)
PROCEDURAL (ACOUT=CNTRI)
CNTLA=CNTRI
IF (CNTLA .GT. 12.0) CNTLA=12.0
ACOUT=-2.0*CNTRI+28.0
END $ "OF PROCEDURAL"
MACRO END

"MACRO -- POWER CONTROL LIMITER"
MACRO PCLIM(PCOUT,CNTLI)
PROCEDURAL (PCOUT=CNTRI)
CNTLP=CNTRI
IF (CNTLP .LT. 12.0) CNTLP=12.0
PCOUT=2.0*CNTRI-20.0
END $ "END OF PROCEDURAL"
MACRO END

"MACRO -- CONTROLLER OUTPUT LIMITER"
MACRO CLIM(CTLOUT,COUT)
PROCEDURAL (CTLOUT=COUT)
CTLOUT=COUT
IF (CTLOUT .GT. 20) CTLOUT=20.0
IF (CTLOUT .LT. 4) CTLOUT=4.0
END $ "END OF PROCEDURAL"
MACRO END

"MACRO -- CONTROLLER ANTI-RESET WIND-UP"
MACRO ANTIR(LINTRM,INTRM)
PROCEDURAL (LINTRM=INTRM)
LINTRM=INTRM
IF (ABS(LINTRM) .GT. 1000) LINTRM=LINTRM/ABS(LINTRM)*1000
END $ "END OF PROCEDURAL"
MACRO END

```

INITIAL

CINTERVAL CINT=0.1

TFIN=2000.0

```
'SET CALL DEBUG WINDOW FOR DETAIL DUMPS'
BUGON=390
BUGOFF=BUGON+0
CONSTANT PI=3.14159
CONSTANT EXP=2.718281
'DEFINE STEFAN-BOLTZMAN RADIATION CONSTANT, BTU/FT^2-SEC--DEG R^4'
SIGMA=4.81E-13
'DEFINE PROBE COOLING GAS (AIR) PARAMETERS'
'DENSITY, LBM/FT^3'
RHOAIR=0.03
'VISCOSITY, LBM/FT-SEC'
MUAIR=2.4E-5
'THERMAL CONDUCTIVITY, BTU/SEC-FT-DEG F'
TCAIR=8.33E-06
'HEAT CAPACITY, BTU/LBM-DEG F'
CPAIR=0.26
'COOLING GAS CONTROL VALVE COEFFICIENT, CV'
CV=0.5
'TYPICAL PRESSURE DROP ACROSS CONTROL VALVE, PSI'
DELTA P=50.0
'COOLING GAS SUPPLY PRESSURE, PSIA'
USP=114.7
'
END OF COOLING GAS PARAMETERS'
'
DEFINE FLUE GAS PARAMETERS'
'THERMAL CONDUCTIVITY OF FLUE GAS, BTU/SEC-FT-DEG F'
TCGAS=8.33E-06
```

'VISCOSITY OF THE FLUE GAS, LBM/FT-SEC'
MUGAS=MUAIR

'DENSITY, LBM/FT^3'
RHOGAS=0.02

"SPECIFIC HEAT, BTU/LBM-DEG F"
CPFG=0.2835

"DEFINE FUEL CONSUMPTION PARAMETERS"
"FUEL BURNING RATE, BTU/SEC, HI-16E6, LO-3E6"
BRNRAT=4444.44

"NATURAL GAS, CH4, HEAT VALUE, BTU/LBM"
CH4HV=23000.0

"MASS FLOW RATE OF NATURAL GAS(CH4), LBM/SEC"
MFRCH4=BRNRAT/CH4HV

"FLUE CROSS SECTIONAL DIMENSIONS, FT"
FLENTH=4.0
FWIDTH=4.0

'FLUE GAS AIR PRESSURE, PSI'
APDUCT=14.7

'STANDARD ATMOSPHERIC PRESSURE, PSI'
SAPRES=14.7

'EMISSIVITY OF FLUE WALLS'
FLUEEM=0.8

'REFLECTANCE OF THE FLUE WALLS'
FLUEFP=1-FLUEEM

'TEMPERATURE OF THE FLUE GAS, DEG F'
TCAS=2000.0

'TEMPERATURE OF THE FLUE WALLS, DEG F'
TFLUE=TCAS

'VOLUME FLOW RATE OF THE GAS, FT^3/SEC'
VFRGAS=350


```

"      END OF FLUE AND FLUE GAS PARAMETERS'
"
"      DEFINE SAMPLE CYLINDER PARAMETERS'
"NOTE: ALL PARAMETERS ARE FOR SINTERED ALPHA SILICON CARBIDE.'
"LENGTH, FT"
CYLEN=15.0/12.0
"OUTSIDE DIAMETER, FT"
CYLODI=1.0/12.0
"INSIDE DIAMETER, FT"
CYLIDI=(1.0-1.0/8.0)/12.0
"THERMAL CONDUCTIVITY, BTU/SEC-FT-DEG F"
CYLTC=.012
"SPECIFIC HEAT, BTU/LBM-DEG F"
CYLCP=3.35E-1
"EMISSIVITY"
CYLEMI=0.9
"REFLECTANCE"
CYLRPF=1-CYLEMI
"DENSITY, LBM/FT^3"
CYLDEN=193
"CYLINDER MASS"
CYLMAS=CYLDEN*CYLEN*PI*((CYLODI/2)**2.0-(CYLIDI/2)**2.0)
'
'      END OF SAMPLE CYLINDER PARAMETERS'
"
"      DEFINE HEATER PARAMETERS'
"PARAMETERS BASED ON SILICON CARBIDE HEATING ELEMENT."
"LENGTH, FT"
HEATLN=18.0/12.0
"DIAMETER, FT"

```



```

"EQUIVALENT CROSS SECTIONAL AREA, FT^2"
SPCCSA=(1.0/8.0/12.0)**2.0*3.0

"INSIDE DIAMETER, FT"
SPCIDI=HEATDI

"OUTSIDE DIAMETER, FT"
SPCUDI=CYLIDI

"
      END OF SPACER PARAMETERS'
'
      GENERAL COMPUTED VALUES'

"VISCOSITY OF AIR IN THE DUCT"
MUAIRD=MUAIR

"VISCOSITY OF GAS IN THE FLUE"
MUAIRF=MUAIR

"VISCOSITY OF AIR AT THE WALL"
MUAIRW=MUAIR

"CROSS SECTIONAL AREA OF FLUE, FT^2"
FAREA=PLENTH*FWIDTH

"VELOCITY OF FLUE GAS."
GASVEL=VFRGAS/FAREA

"CROSS SECTIONAL AREA OF SPACE BETWEEN HEATER AND CYLINDER."
DAREA=PI*(CYLIDI**2.0-HEATDI**2.0)/4.0

"VOLUME OF SPACE BETWEEN HEATER AND CYLINDER."
DUCTV=DAREA*CYLLEN

"EQUIVALENT DIAMETER OF DUCT FORMED BY HEATER AND CYLINDER:"
EQDDIA=4.0*DAREA/(PI*(HEATDI+CYLIDI))

"EQUIVALENT DIAMETER OF FLUE"
EQFDIA=(4.0*FAREA)/(2.0*PLENTH+2.0*FWIDTH)

"AXIAL AREA OF HEATER, FT^2 (IGNORE END OF HEATER)."
HAREA=PI*HEATDI*HEATLN

"COMPUTE AXIAL AREA OF INSIDE OF CYLINDER, FT^2."
CYLIAR=PI*CYLIDI*CYLLEN

```

```

"COMPUTE AXIAL AREA OF OUTSIDE OF CYLINDER."
CYLOAR=PI*CYLODI*CYLLEN

"DEFINE GEOMETRIC VIEW FACTORS."
F12=1.0
F21=AREA/CYLAR
F22=1.0-F21
F23=1.0
F33=1.0
F32=1.0

"DEFINE INITIAL CONDITIONS PRIOR TO ENTERING DYNAMIC SECTION"
"INITIAL CYLINDER TEMPERATURE, DEG F"
TCYL=80.0
TCYL=TCYLIN

"INITIAL HEATER TEMPERATURE, DEG F"
THETIN=80.0
THEAT=THETIN

'INITIAL AIR TEMPERATURE, DEG F'
TAINIT=80.0

"INITIAL ENERGY OF HEATER"
HQINIT=(THETIN+460.0)*HEATCP*HETMAS

'INITIAL ENERGY OF CYLINDER'
CQINIT=(TCYLIN+460.0)*CYLCP*CYLMAS

'INITIAL CONTROLLER ERROR'
ERRINT=0.0
COUT0=12.0

'TIME IN SECONDS TO STEP CHANGE OF SETPOINT.'
TO=400.0
'MAGNITUDE OF SETPOINT CHANGE, DEGREES F'
TCHNG=2120.

'INITIAL CONTROLLER ERROR TERM.'
ITRMO=0.0
ERTRM=0.0

END $"OF INITIAL SECTION"
      BEGIN DYNAMIC SECTION'

```

```

DYNAMIC
  SETPNT-SET0+TCHNG*STEP(T0)
DERIVATIVE
  *CALL DEBUG"
  *IMPLEMENT THE PID CONTROLLER."
  ERRTRM-SETPNT-TCVL
  INTRM-KI*INTEG(ERRTRM,ITRMO)
  LINTRM-ANTIR(INTRM)
  COUT-COUT0+KP*(ERRTRM+LINTRM)
  CTLOUT-CLIM(COUT)
  *SPLIT THE CONTROLLER OUTPUT FOR HEATER AND AIR FLOW CONTROL."
  ACNTLI-ACLIM(CTLOUT)
  HCNTLI-PCLIN(CTLOUT)
  PROCEDURAL(THQDOT,TCQDOT)
  *IN PREPARATION FOR THE FOLLOWING, IN EITHER CASE NEED TO KNOW.."
  *ENERGY BEING SUPPLIED FROM THE FLUE GAS AND SURROUNDING MASS."
  *REYNOLDS NUMBER FOR FLUE GAS."
  RDNUMF-(GASVEL*RHO*GAS*CYLODI)/MUGAS
  *NUSSELT NUMBER FOR FLUE BASED ON REYNOLDS NUMBER."
  *FIRST DETERMINE THE COEFFICIENTS BASED ON REYNOLDS NUMBER."
  IF (RDNUMF .GT. 4.0)GOTO 1020
  C=0.891
  M=0.33
  GOTO 1060
1020..CONTINUE
  IF (RDNUMF .GT. 40.0)GOTO 1030
  C=0.821
  M=0.835
  GOTO 1060
1030..CONTINUE
  IF (RDNUMF .GT. 4000.0)GOTO 1040
  C=0.615
  M=0.466
  GOTO 1060
1040..CONTINUE
  IF (RDNUMF .GT. 40000.0)GOTO 1050
  C=0.174

```

```

M=0.618
GOTO 1060
1050..CONTINUE
C=0.024
M=0.805
"NOW ACTUALLY CALCULATE THE NUSSELT NUMBER."
1060..CONTINUE
NUNUMF=C*RDNUMF**M

"RADIOSITY OF CYLINDER RELATIVE TO FLUE."
R2CF=(CYLEMI*SIGMA*TCYL**4.0*(1-FLUERF*F33)+CYLRF*F23*FLUEEM*SIGMA*...
TFLUE**4.0)/(1-FLUERF*F33-CYLRF**2.0*F32)

"RADIATIVE ENERGY TRANSFER FROM FLUE TO CYLINDER, BTU/SEC"
CRQDOT=(CYLOAR*(CYLEMI/CYLRF*(R2CF-SIGMA*TCYL**4.0)))

"HEAT TRANSFER FILM COEFFICIENT BETWEEN THE CYLINDER AND THE FLUE GAS."
HCG=(TCGAS*NUNUMF)/CYLODI

"CONVECTIVE ENERGY TRANSFER FROM FLUE GAS TO CYLINDER, BTU/SEC"
CCQDOT=(TGAS-TCYL)*HCG*CYLOAR

"CONDUCTIVE HEAT TRANSFER FROM CYL. TO FLUE BY SUPPORT, BTU/SEC"
"TOTAL HEAT ENERGY INPUT FROM FLUE GAS TO CYLINDER, BTU/SEC"
FCQDOT=CRQDOT+CCQDOT

"NOW DETERMINE IF THE COOLING GAS IS ON OR THE HEATING ELEMENT"
"DETERMINE WHICH MODEL TO USE, HEATING OR COOLING, BASED ON..."
"CONTROLLER OUTPUT, < 12 MA OUT IS COOLING, -> 12 MA OUT IS HEATING."
IF( CTLOUT .GE. 12.0)GO TO 2000

"COOLING MODEL IS IN EFFECT. (FORCED CONVECTION)"
"DETERMINE THE VALVE OPENING AS A FUNCTION OF CONTROL SIGNAL."
"REVERSE ACTING"
VLVOPN=(ACNTLI-4.0)/16.0

"VOLUME FLOW RATE OF AIR AS FUNCTION OF VALVE OPENING, FT^3/SEC"
VFAIR=((CV*0.37*(USP*DELTA P)**0.5)/(460+TAINIT)**0.5)*VLVOPN

"MASS FLOW RATE OF AIR AS FUNCTION OF VALVE OPENING, LBM/SEC"
MFAIR=VFAIR*RHOAIR

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```

'VELOCITY OF COOLING GAS, FT/SEC'
AIRVEL=(VPAIR/DAREA)*(TAAVG/TAINIT)

'COMPUTE PRANDTL NUMBER FOR DUCT BASED ON CURRENT CONDITIONS'
PRNUD=MUAIRD*CPAIR/TCAIR

"COMPUTE REYNOLDS NUMBER FOR DUCT BASED ON CURRENT AIR FLOW"
RDNUD=(AIRVEL*RHOAIR*EQDDIA)/MUAIRD*(TAINIT/TAAVG)

"COMPUTE NUSSELT NUMBER FOR DUCT BASED ON REYNOLDS NUMBER"
IF (RDNUD .GT. 5000.0) GOTO 1000

NUNUD=1.86*(RDNUD*PRNUD*EQDDIA/CYLLEN)**(1.0/3.0)*...
(MUAIRD/MUAIRW)**0.14

'CHECK FOR AIR TURNED OFF.'
IF (RDNUD .EQ. 0)NUNUD=0.0
GOTO 1010

1000..CONTINUE
NUNUD=0.021*(CYLIDI/HEATDI)**0.45*RDNUD**0.9*PRNUD**((1.0/3.0)*...
(MUAIRD/MUAIRW)**0.14

1010..CONTINUE

"NUSSELT NUMBER HAS BEEN DETERMINED, COMPUTE FILM COEFF."
"HHA IS THE FILM COEFF. BETWEEN HEATER AND THE AIR."
"HCA IS THE FILM COEFF. BETWEEN CYLINDER AND THE AIR."
"UNITS OF HHA ARE: BTU/SEC-FT^2-DEGF"
HHA=NUMUD*TCAIR/EQDDIA
HCA=HHA

"COMPUTE ENERGY TRANSFER COEFFICIENT BETWEEN HEATER AND MID-POINT..."
"OF DUCT., HTA=HEATER-TO-AIR"
HTA=HHA

"COMPUTE ENERGY TRANSFER COEFFICIENT BETWEEN MID-POINT OF DUCT..."
"AND CYLINDER., CTA=CYLINDER-TO-AIR"
CTA=HCA

IF (AIRVEL .NE. 0)GOTO 1011
TAAVG=ABS(HEAT-TCYL)/2
GOTO 1012

1011.. CONTINUE

```

```

      BETA=(1/(RHOAIR*(TAINIT/TA AVG)*CPAIR*AIRVEL))*(HTA*(PI*(HEATDI/DAREA))...
+CTA*(PI*(CYLIDI/DAREA)))

      K1=(HEATDI*HTA*THEAT*CYLIDI*TCYL*CTA)/(HEATDI*HTA*CYLIDI*CTA)

      "CALCULATE AVERAGE TEMPERATURE."
      TAAVG=K1*(1-(1/(BETA*CYLLEN))*(1-EXP*(-BETA*CYLLEN))))

1012.. CONTINUE

      "RATE OF ENERGY TRANSFER FROM HEATER TO AIR, BTU/SEC"
      HTAQDT=HTA*PI*HEATDI*(THEAT-TAAVG)*HEATLN

      "RATE OF ENERGY TRANSFER FROM CYLINDER TO AIR, BTU/SEC"
      CTAQDT=CTA*PI*CYLIDI*(TCYL-TAAVG)*CYLLEN

      "PROVIDE FOR RADIATIVE HEAT TRANSFER FROM CYLINDER TO HEATER"
      "RADIOSITY OF HEATER RELATIVE TO CYLINDER."
      RHIC=(HEATEM*SIGMA*THEAT**4.0*(1.0-CYLRF*F22)+(HEATRF*F12*CYLENI...
*SIGMA*TCYL**4.0))/(1.0-CYLRF*F22-HEATRF*F12*CYLRF*F21)

      "RADIATIVE ENERGY TRANSFER FROM CYLINDER TO HEATER, BTU/SEC"
      HRQDOT=(HAREA*(HEATEM/HEATRF*(RHIC-SIGMA*THEAT**4.0)))

      "CONDUCTIVE ENERGY TRANSFER THROUGH THE SPACERS, BTU/SEC"
      HSQDOT=(THEAT-TCYL)*TCSPAC*NUMSPC*(SPCODI-SPCIDI)*SPCCSA

      'ASSUME NO HEATER TO CYLINDER CONVECTIVE HEAT TRANSFER.'
      HCQDOT=0.0
      HIQDOT=0.0

      "JUMP AROUND HEATING MODEL (IN COOLING MODE)"
      GO TO 3000

      "HEATING MODEL IS IN EFFECT (FREE CONVECTION)"
      "HEATER POWER AS FUNCTION OF CONTROLLER CURRENT, BTU/SEC/MA"
      HIQDOT=HMAX*((HCONTLI-4.0)/16.0)**2.0

2000..
      "HEAT TRANSFER FILM COEFF. FOR HEATER, BTU/FT^2-DEG F."
      "FOR THE TIME BEING, THE SAME METHOD IS USED FOR AIR TO..."
      "CYLINDER FILM COEFFICIENT."

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```

HCT=0.98
THEAT0=530.
TCYLO=530.
TAAVG=(THEAT+TCYL)/2
HHA0=0.281*(ABS (THEAT-TAAVG)/HEATDI)**0.25
HHA=(-.0000001+HHA0*HCT*(THEAT0/TAAVG)**0.25*...
(APDUCT/SAPRES)**0.5)/3600.0
HCA0=0.281*(ABS (TAAVG-TCYL)/CYLIDI)**0.25
HCA=(-.0000001+HCA0*HCT*(TCYLO/TAAVG)**0.25*...
(APDUCT/SAPRES)**0.5)/3600.0
"CONVECTIVE HEAT TRANSFER FROM HEATER TO CYLINDER, BTU/SEC"
HCQDOT=(THEAT-TCYL)/(1.0/(HAREA*HHA))+...
(ALOG(CYLIDI/HEATDI)/(2.0*PI*HEATIN*TCAIR))+(1.0/(CYLIAR*HCA))
"RADIOSITY OF HEATER RELATIVE TO CYLINDER."
R1HC=(HEATEM*SIGMA*THEAT**4.0*(1.0-CYLRP*F22)+(HEATRF*F12*CYLEMI...
*SIGMA*TCYL**4.0))/(1.0-CYLRP*F22-HEATRF*F12*CYLRP*F21)
"RADIATIVE ENERGY TRANSFER FROM HEATER TO CYLINDER, BTU/SEC"
HRQDOT=(HAREA*(HEATEM/HEATRF*(R1HC-SIGMA*THEAT**4.0)))
"CONDUCTIVE ENERGY TRANSFER THROUGH THE SPACERS, BTU/SEC"
HSQDOT=(THEAT-TCYL)*TCSPAC*NUMSPC*(SPCODI-SPCIDI)*SPCCSA
'ASSUME NO CYLINDER OR HEATER TO COOLING AIR HEAT TRANSFER.'
CTAQDT=0.0
HTAQDT=0.0
VLVOPN=0.0
VFRAIR=0.0
MFRAIR=0.0
AIRVEL=0.0

3000..CONTINUE
'TOTAL CYLINDER HEAT ENERGY'

```

```

TCQDOT=HCQDOT-HRQDOT+CRQDOT+CCQDOT+HSQDOT-CTAQDT
'TOTAL HEATER ENERGY'
THQDOT=-HCQDOT+HRQDOT+HIQDOT-HSQDOT-HTAQDT
CONTINUE

END $ 'END OF PROCEDURAL'

'TOTAL NET ENERGY EMITTED FROM/RECEIVED BY HEATER'
HQ-INTEG(THQDOT,HQINIT)

'TOTAL NET ENERGY EMITTED FROM/RECEIVED BY CYLINDER'
CQ-INTEG(TCQDOT,CQINIT)

END $ "END OF DERIVATIVE SECTION"

'DETERMINE THE CURRENT TEMPERATURE OF THE CYLINDER, DEG F.'
TCYL=CQ/(CYLCP*CYLMAS)-460.0
'CONVERT TO DEGREES C'
TCYLC=(TCYL-32)*.556

'DETERMINE THE CURRENT TEMPERATURE OF THE HEATER, DEG F.'
THEAT=HQ/(HEATCP*HETMAS)-460.0
'CONVERT TO DEGREES C'
THEATC=(THEAT-32)*.556

IF (T .GT. BUGON .AND. T .LT. BUGOFF) CALL DEBUG
IF((T - INT(T/15)*15).EQ. 0)....
write(5,5050) t,TCYLC,THEATC,CTLOUT,TCQDOT,THQDOT,SETPT,TAAVG,VFPAIR,CTAQDT
5050.. format(' ',F5.0,1X,3(F5.0,1X),2(E10.3,1X),F5.0,2(1X,F5.0),E10.3)

TERMT(T .GT. TFIN)
END $ "END OF DYNAMIC SECTION"
END $ "END OF PROGRAM"

```

VITA

Jim A. McEvers was born on December 22, 1945, in Herrin, Illinois and presently resides in Knoxville, Tennessee. He is married to the former Miss Dona M. Friend. They have two children, Angela Dawn McEvers and Robert Allan McEvers.

He received his Associate of Arts degree in June 1968 from Orlando Junior College in Orlando, Florida, his Bachelor of Science Degree in Engineering from the University of Central Florida in Orlando, Florida, in March 1974, and will receive his Master's Degree in Engineering Science and Mechanics from The University of Tennessee in Knoxville, Tennessee, in August 1988.

He was a student member of IEEE and has been a professional member of IEEE.

He is employed as a staff engineer/Research Support group leader in the Instrumentation and Controls Division at Oak Ridge National Laboratory.