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# HEAT TRANSFER ANALYSIS OF THE CFFF SUPERHEATER BY THE EFFECTIVENESS-NTU METHOD

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Wesley T. McMinn

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## ABSTRACT

The Superheater Test Module (SHTM) at the Coal Fired Flow Facility (CFFF) is a 124 foot long heat exchanger assembled from flanged sections. The purpose of the SHTM is to study fouling and tube metal corrosion in flue gas from a coal fired magnetohydrodynamic (MHD) power cycle, while cooling the flow to temperatures that the exhaust gas cleanup equipment can handle. The SHTM is composed of three sections with steam cooled tubes, six with water cooled tubes, and one section with air cooled tubes. The other sections are geometric transitions without cooling surface. Arrangement of the steam, water and air cooled tube banks to achieve specific temperature regimes along the flue gas flow path is critical to the test program.

The original sizing and arrangement of the SHTM was based on hand calculations checked by the Babcock and Wilcox Company with a proprietary boiler sizing computer code. After three years of testing with the SHTM, much more is known about fouling factors, flow distributions and the accuracy of measured data. Based on this operational history it appears that a major redesign of the SHTM will be necessary. The heat transfer analysis and subsequent development of a computer model were undertaken in order to accurately and conveniently evaluate options for redesign of the SHTM. The computer model is based on the "effectiveness-NTU" method of heat exchanger analysis and allows for a great deal of freedom in modeling different tube bundle arrangements.

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# NOMENCLATURE

A: area

C: heat capacity rate = $\dot{m}C_p$

$C_p$: specific heat at constant pressure

D: diameter

F: cleanliness factor or Fahrenheit

G: mass velocity

h: heat transfer coefficient

k: thermal conductivity

$\dot{m}$: mass flow rate

NTU: number of thermal units

Nu: Nusselt Number

Pr: Prandtl Number

psia: pressure (pounds per square inch, absolute)

q: heat transfer rate

Re: Reynolds Number

T: temperature

t: thickness

U: overall heat transfer coefficient

Subscripts

a: ash

c: cold fluid or convection

h: hot fluid

i: inlet or inside

m: mean

max: maximum

min: minimum

o: outlet or outside

r: radiation

s: scale

w: wall

Greek Letters

$\Delta$: difference

$\varepsilon$: efficiency

$\Sigma$: summation

$\mu$: viscosity

# SECTION 1.0

# INTRODUCTION

The Coal Fired Flow Facility (CFFF) at The University of Tennessee Space Institute (UTSI) is a pilot scale facility for research in high temperature coal combustion, heat recovery and pollution control. Primary emphasis since construction of the facility (1977-1981) has been directed towards the investigation of a coal-fired Magnetohyrodynamic (MHD) power cycle. In this process, coal and a "seed" material are mixed together and finely pulverized before being fed to a pressurized combustor. In the combustor, the coal/seed mixture is ignited by an oxygen rich flame and combusts to about 5000$^{o}$F and 80 psia. At this temperature the combustion products form a plasma (ionized gas). The "seed" material used in the CFFF/MHD process is a potassium salt which significantly improves the electrical conductivity of the plasma and thereby the efficiency of the MHD process. At the end of the combustion process the plasma is at about 85% stoichiometry, i.e. there is still unburned carbon in the gas.

The pressure in the combustor is sufficient to accelerate the plasma to supersonic (about Mach 1.1) speeds through an aerodynamically shaped nozzle and an MHD channel simulator downstream of the combustor. It is the interaction (relative motion) between the plasma in the channel and a fixed magnetic field imposed across the

channel which causes an electric current to flow in the channel electrodes. After exiting the channel, the plasma enters a series of diffusers which slow the flow back to subsonic speeds before entering a "conventional" steam-producing, water wall unpressurized boiler. The MHD channel simulator is an aerodynamically contoured duct designed to simulate the aerodynamic history (heat transfer, velocity, residence time, etc.) of plasma flow in an actual MHD channel without the added expense and complexity of the channel and magnet.

At the entrance to the boiler the MHD cycle is completed, i.e. the MHD-generated energy has been extracted, but the flue gas still contains considerable thermal energy which can be reclaimed in a "conventional" steam power cycle. The CFFF does not include a steam turbine, but does include a radiant furnace with water cooled walls, secondary combustor, the Superheater Test Module (SHTM), and a variety of pollution control devices (Fig. 1.1), all of which are intended to simulate similar components in a conventional coal fired steam power plant. The current primary mission for the CFFF is testing and evaluation of materials, processes, and systems applicable to the steam power cycle of a commercial MHD plant.

The reasons for testing these "downstream" components and materials may not be immediately apparent. The MHD process results in a flue gas with significantly different properties than those from a typical pulverized coal burner. One primary difference is the presence of significant quantities of potassium compounds (the seed material) in the flue gas. This is not common practice and their

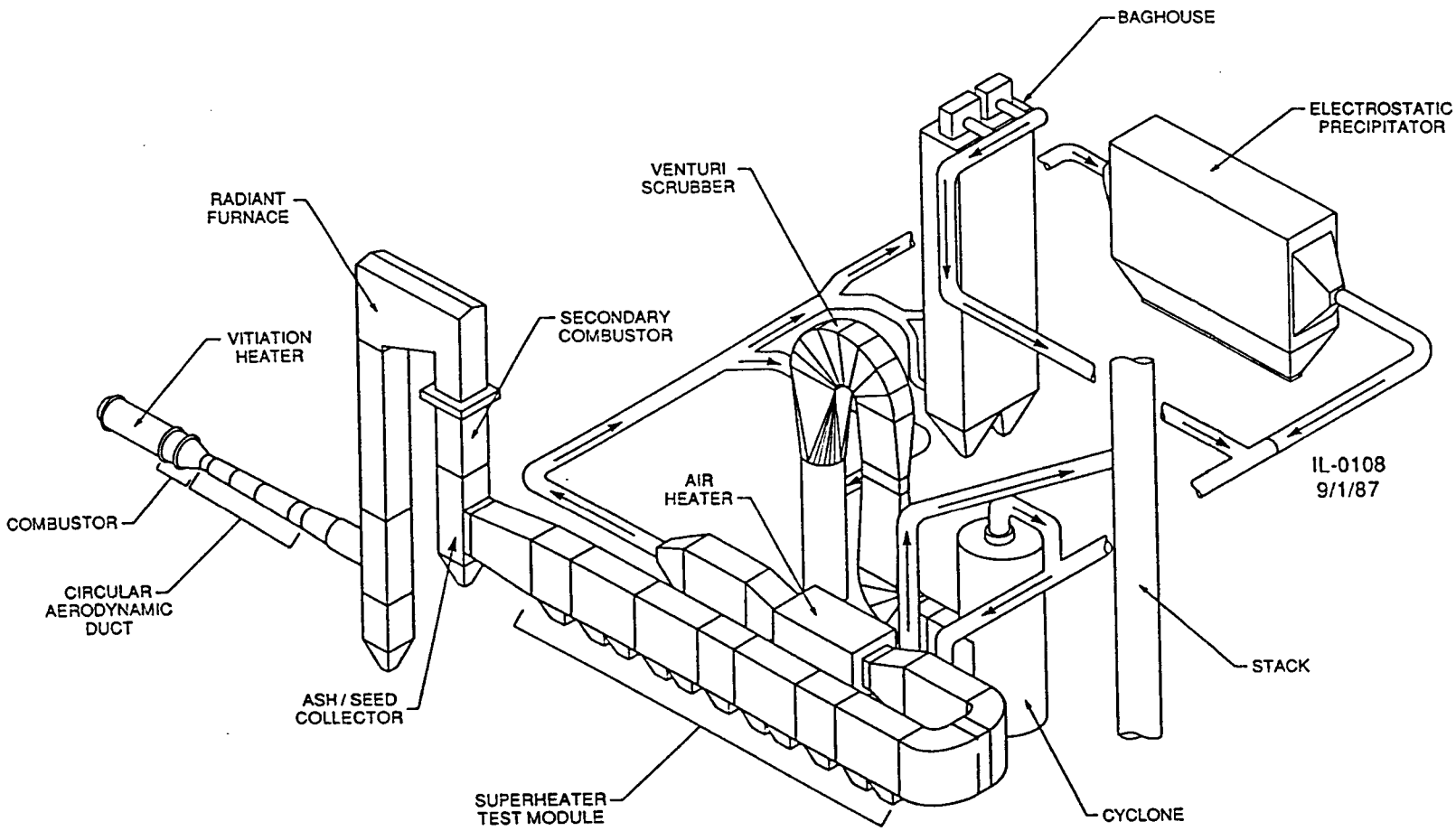



Figure 1.1. CFFF Test Train Configuration

presence in the slag and flyash have potentially serious consequences for fouling and corrosion of heat transfer materials. The extremely high temperature and substoichiometric combustion of the MHD cycle also results in particle size distributions and general flue gas chemistries which are not typical of conventional pulverized coal burners.

The Superheater Test Module (SHTM) is actually a series of individual heat exchangers positioned downstream of the secondary combustor and just upstream of the pollution control devices in the CFFF flow train. The purpose of the SHTM is to provide a test bed for the evaluation of materials and design features of heat transfer surface in steam boilers with flue gas from an MHD cycle. The SHTM is approximately 124 feet long and is assembled from fourteen individual flanged sections (Fig. 1.2). Four of these sections serve as flow interfaces and have no heat transfer surface; the other ten sections contain water, steam or air-cooled tubes. Three of the sections are known as "Test Sections" and contain tubes of various alloys which are cooled by saturated steam from a 150 psi boiler. The steam becomes superheated in these tubes which allows tube metal temperatures to reach typical commercial levels. Evaluation of fouling and corrosion on these tubes in relation to metal alloys and the flue gas temperature regime is one of the primary objectives of the current MHD program at UTSI.

Six of the other sections of the SHTM are known as "Cooling Sections" and they contain water-cooled carbon steel tubes. The

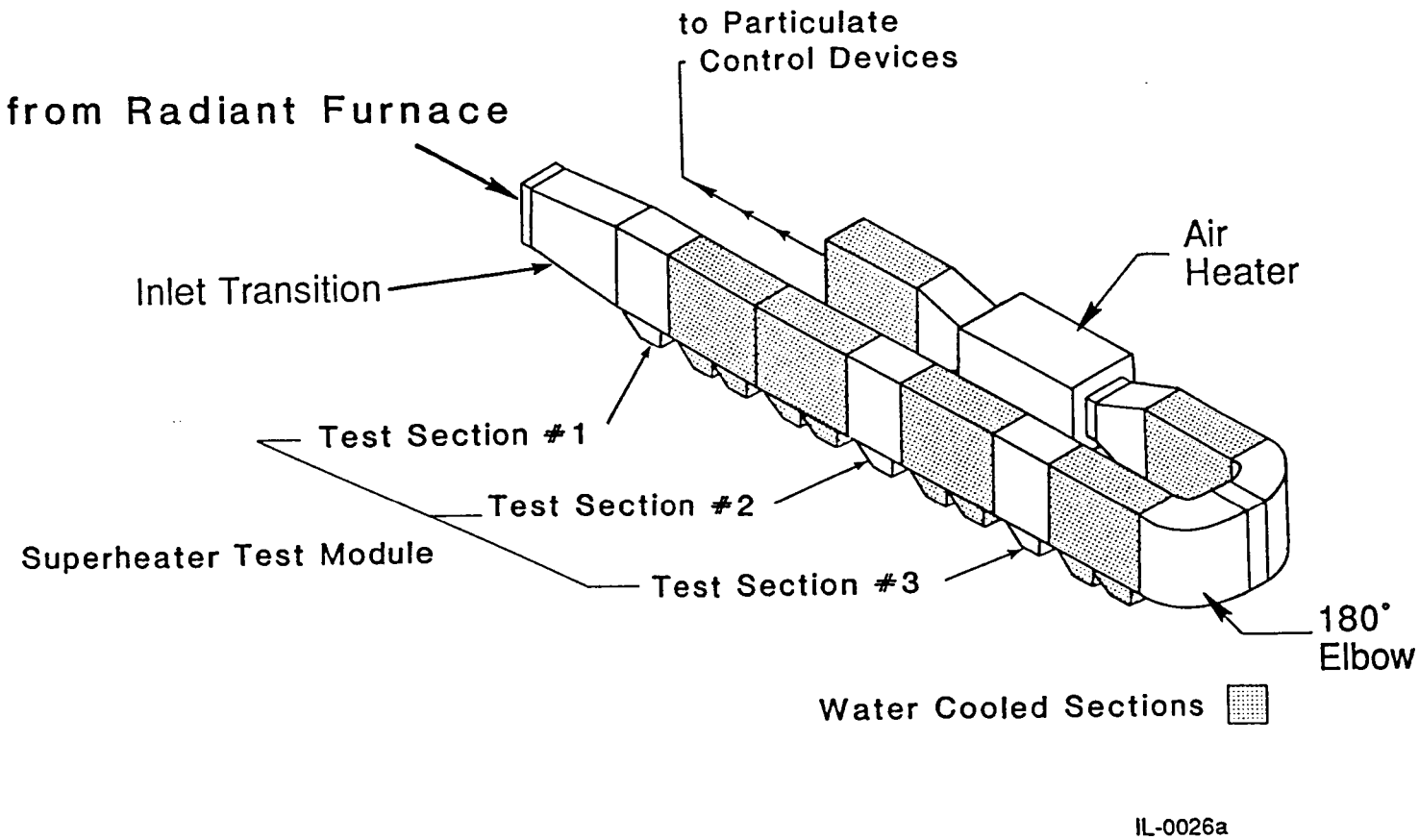



Figure 1.2. CFFF Superheater Test Module Arrangement

purpose of these sections is to reduce the flue gas temperatures to specific levels between Test Sections and to reduce the overall flue gas temperature to the operating limits of the pollution cleanup devices. Since water-cooled surfaces of this type are not typical of power plant practice, corrosion and fouling of these tubes is not of particular interest, except as it relates to the prediction of heat transfer in these sections.

The other heat transfer device is the "Air Heater". This device is used to preheat air for the secondary combustor in the furnace. The Air Heater is a recuperative heat exchanger with carbon steel tubes (air inside tubes), and operates at a relatively low temperature level of about 800$^{o}$F. Since many modern power plants use regenerative plate type air heaters, the performance of this device is only partially applicable to future designs. Tube surface fouling in this temperature regime is of interest.

Since one of the primary functions of the SHTM is the study of heat transfer surface fouling with MHD flue gas, it must follow that the original design sizing of the SHTM sections was based on incomplete knowledge. Realizing this, the design concept called for the SHTM to be as flexible as possible in terms of section arrangement and heat transfer surface distribution. It was intended that inaccuracies in the sizing could be compensated for by addition, deletion, or rearrangement of tube banks and/or whole sections. Also it was anticipated that future testing requirements might call for different arrangements of the SHTM sections. The sections of the SHTM

all have matching flanges and the tube bundles are modular.

The original SHTM sizing was performed by hand calculations and was later confirmed by the Babcock and Wilcox Company (B&W) using a company-confidential boiler sizing program. A cleanliness factor of 1.0, i.e. no fouling, was used in both the hand calculations and in the B&W analysis. The B&W analysis indicated that metal temperatures in the first test section would be too high for continuous operation. Therefore, before the SHTM was built a bank of water-cooled tubes was added upstream of the first set of steam cooled tubes. Two spare cooling sections were included in the design to allow for addition of cooling surface as needed. Subsequent operation of the SHTM revealed that fouling was more severe than anticipated and water-cooled tubes had to be added to the spare cooling sections. The sizing of this new heat transfer area was again performed by hand calculations based on extrapolations of test data. There was no overall thermal model to predict the flue gas distribution through the full SHTM.

Since the SHTM became operational in 1985, there have been several major modifications to the heat transfer geometry and/or operational characteristics. Each of these modifications has been preceded by extrapolations and hand calculations to predict their effect on the performance of individual SHTM sections, but as mentioned above, there was no overall model. During the three years of operation of the SHTM, a number of detailed analyses have been made of the heat transfer characteristics of the various sections. Particular attention has been paid to the fouling factor of each

section and this has been tracked over time.

The SHTM is not in its final configuration. Plans are being discussed for relocation of tube bundles and sections to achieve specific flue gas temperatures at the test section tubes. Also, modifications are being discussed which are intended to alleviate overheating problems in the Air Heater and downstream pollution cleanup devices. The purpose of the computer model developed for this thesis is to accurately and conveniently evaluate the effects of these proposed modifications on the thermal performance of the SHTM. Results from previous thermal analyses and evaluations of the fouling factors have been incorporated into the current analysis. The accuracy of the model will be demonstrated by comparison of predicted temperature distributions with measured temperatures for previous SHTM configurations and operating conditions. Finally, the proposed modifications will be modeled and the flue gas distributions predicted in order to evaluate the effectiveness of these changes to the SHTM.

# SECTION 2.0

# THE HEAT TRANSFER MODEL

## 2.1 Modeling Approach

In order to develop a computer model of the SHTM it was first necessary to decide on the degree of approximation which would be acceptable for the intended use. At one extreme it would be possible to develop a three dimensional model which analyzed very fine details of the SHTM geometry. Vertical, horizontal, and axial temperature differences could be investigated for small increments of length down the SHTM flow path. On the other hand, a very rough model could be developed which ignored most of the geometric details of the SHTM. For example, the heat loss to the water-cooled tubes is so dominant that the SHTM could be modeled as one single tube bank with heat transfer area equal to the sum of the water-cooled banks. A one dimensional calculation would then predict the outlet flue gas temperature with reasonable accuracy. Of course, intermediate temperatures would then be obtained by interpolation.

The primary purpose of this analysis was to examine the axial or length-wise temperature profile along the full length of the SHTM. This permits characterization of the heat transfer levels of individual sections, so that sensitivities to various operating factors, e.g. mass flows, coal composition, fouling factors, etc.,

could be studied. The model chosen, therefore, was one dimensional, but with the flow path of the flue gas divided according to the natural mechanical divisions of the SHTM.

As stated previously, the SHTM is assembled from fourteen flanged sections, ten of which contain heat transfer tubes. The tubes of primary interest are the steam-cooled tubes of the Test Sections. These tube banks are eight tubes (about 42 inches) wide by about 7 feet tall by either two passes (6 inches) or four passes (18 inches) deep. Theoretically and by actual measurement, as will be discussed in Section 4.0, it is known that there are both vertical and horizontal temperature distributions at any plane cutting the flow path. The heat transfer model, though, considers the flue gas to be at a constant temperature at the entrance and at the exit of each tube bank. The inaccuracy inherent in this approximation is considered to be acceptable for this analysis.

The water-cooled tube banks of the cooling sections are treated in a similar fashion. These tube banks are also eight tubes wide by about seven feet high, but are twelve passes (about 10 feet) deep. The model calculates a uniform outlet temperature for the whole tube bank based on a uniform inlet temperature. Even though the flue gas temperature drops across cooling sections are an order of magnitude higher than those for test sections, this one outlet temperature is considered adequate for the analysis. As mentioned in the Introduction, the cooling sections are not typical of power plant heating surface and detailed analysis of these sections is not needed.

There are two other sections with heat transfer tubes; the screen tubes of Test Section 1 and the Air Heater. The screen tubes are like cooling section tubes except they are only four passes (about 30 inches) deep. Because of the flow irregularities induced at the furnace exit, the screen tubes in actual operation have a high level of non-uniform temperature distribution at their entrance. Even so, the model only calculates a uniform outlet temperature based on a uniform inlet temperature. A three dimensional modeling of the screen tubes is beyond the scope of this analysis.

The Air Heater is a densely packed heat exchanger. It has over five times the heat transfer area of a Cooling Section in a volume only 50% bigger. The inlet and exit openings of the Air Heater have only about 38% of the flow area of a Cooling Section. This combination of reduced inlet and outlet flow area and the high flow resistance within the tube bundle tend to promote uniformity of the flue gas temperature. The Air Heater is probably the most accurately modeled of all the sections in terms of the one dimensional calculation used by the program.

Flow transitions and other "cavity" volumes in the SHTM which do not include heat transfer tubes are included in the heat transfer model for three reasons. First, these cavities are physically part of the SHTM and their inclusion helps to define the geometry of the case being modeled. Secondly, there is some minor heat loss from the walls of these sections which contributes to the overall temperature drop of

the flue gas along its flow path. Thirdly, the presence or absence of a cavity may have a significant effect on both the radiation and convection heat transfer coefficients of adjacent tube banks.

## 2.2 Heat Exchanger Analysis

As the previous section indicates, the heat transfer model for the SHTM is based on an analysis of whole tube bundles within a specified volume. Although basic heat transfer considerations are used, this type of analysis is better described as heat exchanger analysis rather than as simply heat transfer analysis. Heat exchanger analysis implies the presence of a system of heat transfer surfaces, an enclosure, and the interaction of at least two flows.

Heat exchanger analysis can follow two basic approaches. One approach is more suited to heat exchanger sizing, i.e. the calculation of the amount of heat transfer surface necessary to transfer a given amount of heat under specific operating and geometric conditions. For this case, the inlet and outlet properties of both fluids are specified to achieve thermodynamic equilibrium at a required heat rate. From this information it is possible to calculate the "log mean temperature difference" (LMTD) of the fluids for particular flow arrangements. The LMTD value is then used as a factor in determining the amount of heat transfer surface required. This process is generally known as the "LMTD method" of heat exchanger analysis (Ref. 1).

The other approach to heat exchanger analysis is better suited to determining the amount of heat transfer that can be expected from a given geometry for various flow conditions. This process is generally known as performance analysis and is applied to existing heat exchangers. In this case it is common to know the inlet temperatures of both fluids and to calculate a factor which will aid in predicting the heat transfer which can be expected from a particular heat exchanger geometry and flow arrangement. One of the better established procedures for this type of analysis is known as the "effectiveness-NTU or $\varepsilon$-NTU" method (Ref. 1).

## 2.3 The Effectiveness-NTU Method

At this point it is necessary to either give a background discussion of basic heat exchanger terminology or assume that the reader has some familiarity with these topics. The approach taken here will be to assume that the reader has a basic understanding of terms such as: parallel-flow, counterflow, crossflow, overall heat transfer coefficient, log mean temperature difference, and fouling factors. If this is not the case, the reader should review these topics before proceeding. There are many books and articles with detailed discussions of the basic terms used in heat exchanger analysis (Refs. 1,2).

The effectiveness-NTU method can be introduced by consideration of an "ideal" or perfect heat exchanger which transfers the

thermodynamic maximum amount of heat. This would be possible with a counterflow heat exchanger of infinite length (i.e. infinite area). In this theoretically perfect counterflow heat exchanger either the cold fluid (noted by subscript c) outlet temperature will reach the hot fluid (noted by subscript h) inlet temperature, or the hot fluid outlet temperature will be reduced to the cold fluid inlet temperature. When either of these conditions is met there can be no further exchange of heat and the maximum amount of heat exchange has taken place. Which case will actually occur depends on which fluid has the greater heat capacity rate, $C=\dot{m}C_p$. If the heat capacity rate of the hot fluid $C_h=(\dot{m}C_p)_h$ is greater than the heat capacity rate of the cold fluid $C_c=(\dot{m}C_p)_c$ then the cold fluid outlet temperature will eventually (at length $=\infty$ ) equal the hot fluid inlet temperature. For the case of $C_c>C_h$ the hot fluid outlet temperature will eventually drop to the cold fluid inlet temperature. These conditions are illustrated in Fig. 2.1.

For the purpose of simplicity, the case of $C_h<C_c$ will be used for the following derivation. A parallel discussion based on $C_h>C_c$ would lead to the same result. The general equation for heat transfer rate to/from a fluid of constant specific heat between two points in a heat exchanger is given by:

$$q_{1-2} = \dot{m}\, C_p\, (T_1-T_2) \qquad (2.1)$$

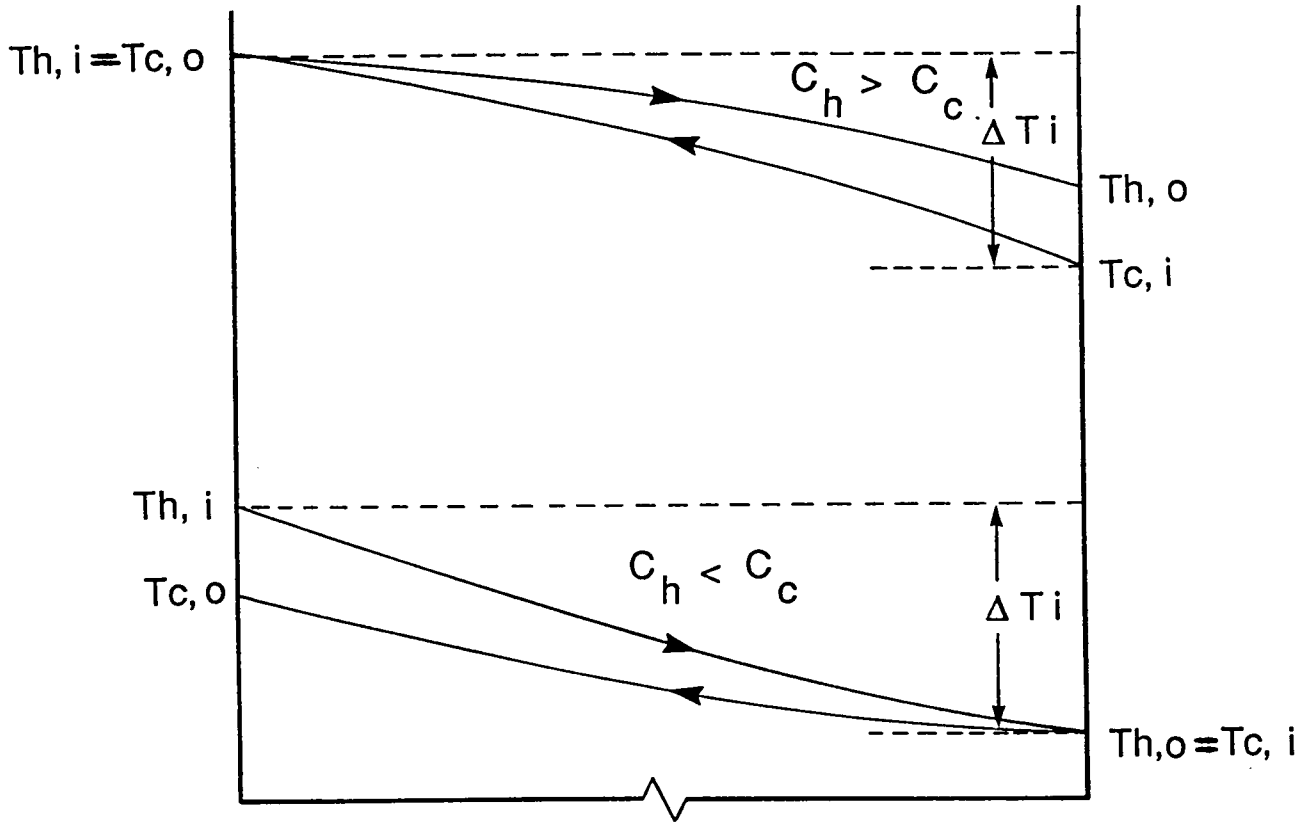



Figure 2.1. Temperature Distribution in a Counterflow Heat Exchanger of Infinite Area

Assuming a perfect heat exchanger, with no external losses, the overall rate of heat gain by the cold fluid $q_c$ is equal to the overall rate of heat loss by the hot fluid $q_h$, and this heat exchange is accomplished between the inlet and outlet of the heat exchanger:

$$C_h(T_{h,i}-T_{h,o}) = q_h = q_c = C_c(T_{c,o}-T_{c,i}) \tag{2.2}$$

Since this is an infinitely long (infinite area) counterflow heat exchanger, the overall heat exchange rate is the maximum possible, therefore:

$$q_{max} = q_c = q_h = C_h(T_{h,i}-T_{h,o}) = C_c(T_{c,o}-T_{c,i}) \tag{2.3}$$

Noting that $T_{h,o} = T_{c,i}$ then $q_{max}$ can be written in terms of known quantities:

$$q_{max} = C_h\ (T_{h,i}-T_{c,i}) = C_{min}\ \Delta T_i \tag{2.4}$$

The maximum possible heat exchange rate is equal to the lesser of the heat capacities ($C_{min}$) times the difference between the inlet temperatures of the hot and cold fluids ($\Delta T_i$).

The heat transfer rate to/from any fluid (of relatively constant specific heat) between any two points in any heat exchanger configuration is also given by Equation 2.1. Accordingly the overall

heat transfer rate for the heat exchanger (again assuming relatively constant specific heat for the fluids) is given by:

$$q_{actual} = C_c(T_{c,o}-T_{c,i}) = C_h(T_{h,i}-T_{h,o}) \tag{2.5}$$

The relationship (ratio) between this actual heat transfer rate and the theoretically maximum possible heat transfer rate for the same fluid inlet temperatures is an indication of the effectiveness ($\varepsilon$) of the heat exchanger:

$$\varepsilon = \frac{q_{actual}}{q_{max}} = \frac{C_c(T_{c,o}-T_{c,i})}{C_{min}(\Delta T_i)} = \frac{C_h(T_{h,i}-T_{h,o})}{C_{min}(\Delta T_i)} \tag{2.6}$$

or if the effectiveness were known then:

$$q_{actual} = \varepsilon(q_{max}) \tag{2.7}$$

where $q_{max}$ is taken from Equation 2.4. In other words, the heat transfer rate for any heat exchanger configuration could be calculated if the exchanger effectiveness ($\varepsilon$) were known and the inlet flow conditions were specified.

The effectiveness of a heat exchanger is highly dependent on mechanical aspects such as fluid velocities, heat transfer resistances, flow arrangement, and total heat transfer area. An expression for describing the total heat transfer rate for a heat exchanger in these "mechanical" terms is the equation:

$$q = U\ A\ (\Delta T_m) \tag{2.8}$$

where U is the overall heat transfer coefficient for the heat exchanger, A is the total heat transfer area, and $\Delta T_m$ is a representative mean temperature difference. In general, the U term takes into account many of the mechanical aspects of the heat exchanger such as heat transfer coefficients (related to velocity and direction of flow), materials (thermal resistances), and other design related features (scaling, fouling, etc.). By substituting this expression for actual heat transfer rate into Equation 2.6, the effectiveness can be given by:

$$\varepsilon = \frac{UA\ \Delta T_m}{C_{min}\ \Delta T_i} = \frac{UA}{C_{min}} \left( \frac{\Delta T_m}{\Delta T_i} \right) \tag{2.9}$$

The dimensionless ratio $UA/C_{min}$ is also known by the term NTU for "number of thermal units". It is a term related to the "thermal size" of a heat exchanger. In general a higher NTU value for a heat exchanger also indicates a higher efficiency rather than simply a larger physical size. Equation 2.9 can then be expressed as:

$$\varepsilon = (NTU)\ (\Delta T_m / \Delta T_i) \tag{2.10}$$

From these two expressions for $\varepsilon$, i.e. Equations 2.6 and 2.10, it is possible to show that heat exchanger effectiveness can be represented

by two independent dimensionless quantities and the flow arrangement of the heat exchanger. From Equation 2.6, if $C_c > C_h$ then the first equality reduces to:

$$( C_{max}/C_{min} ) (\Delta T_c/\Delta T_i) \tag{2.11}$$

while $C_h > C_c$ reduces the second equality to:

$$( C_{max}/C_{min} ) (\Delta T_h/\Delta T_i). \tag{2.12}$$

It is common to express the dimensionless ratio $C_{max}/C_{min}$ by the term $C^*$. Then from Equation 2.10 we have the dimensionless term NTU which is independent of $C^*$. The end terms of both equations are temperature difference ratios. Temperature differences from inlet to outlet of either flow are functions of the heat transfer rate (q) in the exchanger and therefore dependent on the effectiveness. In a similar fashion, $\Delta T_m$ in Equation 2.10 is dependent on $\varepsilon$ in the sense that the individual temperatures are dependent on q. The definition of $\Delta T_m$, though, is determined by the flow arrangement which is independent of q. The effectiveness of a heat exchanger may be mathematically

described by the two independent dimensionless terms $C^*$ and NTU, and the flow arrangement (parallel flow, counter-flow, cross flow, etc.) (Ref. 2).

$$\varepsilon = f(NTU, C^*, \text{flow arrangement}) \tag{2.13}$$

The ε-NTU method is particularly useful for heat exchanger performance calculations because ε-NTU relationships have already been worked out for most of the common types of heat exchangers. Recall from Section 2.2 that performance calculations assume that the heat exchanger geometry and the inlet flow conditions (mass flow rate and temperature) are known. It is also assumed (as a basic part of the previous derivation) that the fluid properties are known and are reasonably constant (particularly $C_p$) across the exchanger. With this information it is apparent that the quantities $C_{max}$, $C_{min}$, U, and A can be readily calculated. Determination of ε values are then made either graphically or analytically (Refs. 1,2,3,4) from known ε-NTU relations. The heat duty of the known exchanger is then calculated from the relation $q_{actual} = \varepsilon C_{min} \Delta T_i$ (Equation 2.7). With $q_{actual}$ known, the fluid outlet temperatures are calculated from $q = \dot{m} C_p \Delta T$ (Equation 2.2) where ΔT is the inlet to outlet temperature drop/rise of the specific fluid stream. In practice the fluid outlet temperatures are estimated so as to make estimates for such things as average fluid properties, film temperatures, and wall temperatures. It is common to iterate the calculation once or more until the

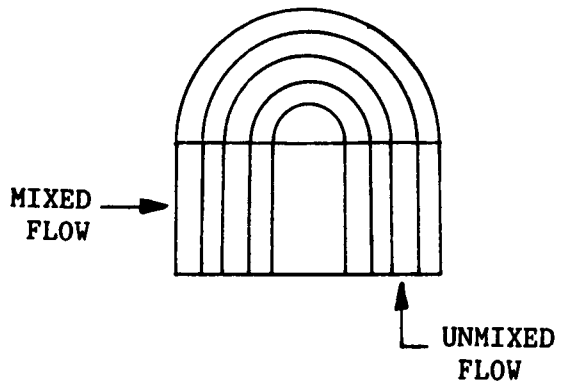
estimated and calculated outlet temperatures are in reasonable agreement.

Although there are a great many heat exchanger configurations, it happens that all of the tube bundles in the SHTM can be described as multi-pass cross-counterflow heat exchangers with one flow mixed (the flue gas) and the other flow unmixed (air, water or steam) (Fig. 2.2). This greatly simplifies the heat exchanger analysis since only two basic ε-NTU relations are needed and these relations have already been developed (Refs. 3,4).

The general ε-NTU relation for a two pass cross-counterflow heat exchanger with the single pass fluid mixed and the two pass fluid unmixed (Ref. 3) is given by:

$$1-\varepsilon C^{*} = \frac{2}{\{1-\exp(-NTU/2)+[1+\exp(-NTU/2)]/(V^{*})^{2}\}} \qquad (2.14)$$

where $V^{*} = \exp\{-C^{*}[1-\exp(-NTU/2)]\}$



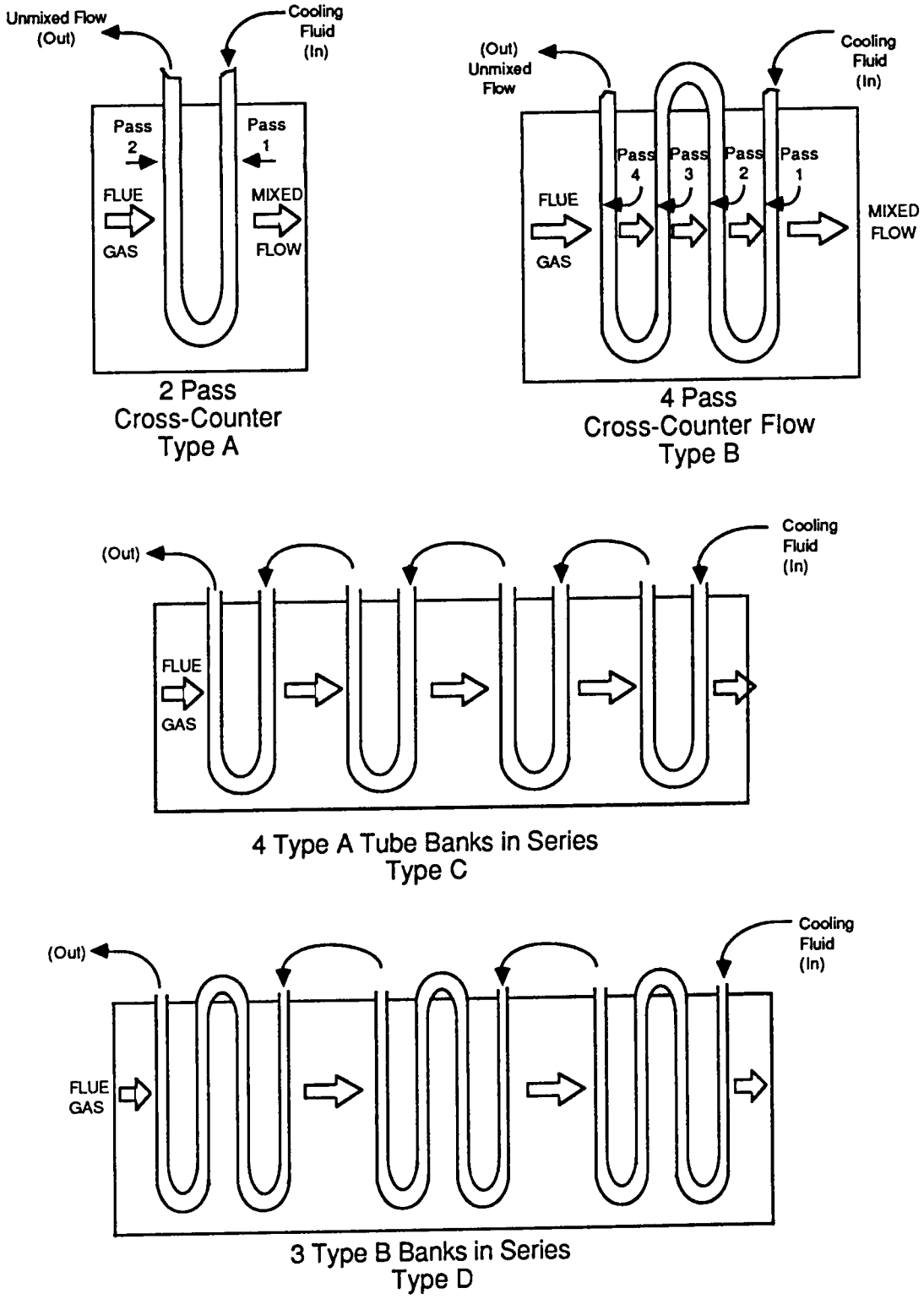



Figure 2.2. SHTM Tube Bundle Arrangements

For the SHTM analysis a recommended polynomial representation of this ε-NTU relation was used (Ref. 4). This polynomial is of the form:

$$\varepsilon = Z+B_2Z^2+B_3Z^3+B_4Z^4+B_5Z^5 \qquad (2.15)$$

where $B_2 = b_{2,0} + b_{2,1}C^* + b_{2,2}C^{*2} + b_{2,3}C^{*3} + b_{2,4}C^{*4}$

$B_3 = b_{3,0} + b_{3,1}C^* + b_{3,2}C^{*2} + b_{3,3}C^{*3} + b_{3,4}C^{*4}$

$B_4 = b_{4,0} + b_{4,1}C^* + b_{4,2}C^{*2} + b_{4,3}C^{*3} + b_{4,4}C^{*4}$

$B_5 = b_{5,0} + b_{5,1}C^* + b_{5,2}C^{*2} + b_{5,3}C^{*3} + b_{5,4}C^{*4}$

and $Z = NTU/(1 + NTU)$

This expression was used for the analysis of the Type A (Fig. 2.2) tube bundles.

No analytical expression equivalent to Equation 2.14 was found in the literature for a 4-pass cross-counterflow heat exchanger with one fluid mixed and the other unmixed. There are methods for approximating similar heat exchangers in series (Ref. 2), as will be discussed below, which could have been applied to the 2-pass equation. A polynomial expression of the type presented by Equation 2.15 was found (Ref. 4). This expression was used for the analysis of Type B tube bundles (Fig. 2.2).

The tube bundle arrangements shown in Fig. 2.2, Types C and D can be represented as 2-pass or 4-pass cross-counterflow heat exchangers in series. For example, the overall heat exchanger performance of the

Type D tube bundle (e.g. a cooling section) can be calculated by treating it like three of the Type B tube bundles in series. Likewise the Type C bundle (e.g. the Air Heater) can be modeled as four Type A tube bundles in series.

Although each tube bundle in a series could actually be calculated independently, this could take several iterations because of the need to estimate the inter-bundle temperatures and is not necessary under certain conditions. It is observed that each tube bundle has the same flow arrangement and the fluid properties can be idealized to be constant in each tube bundle. It is assumed that NTU is equal for each tube bundle and the total NTU is equal to the individual bundle NTU times the number of tube bundles. It is also assumed that the fluid temperatures between tube bundles is uniform (i.e. a homogeneous temperature). Under these conditions the generalized flow schematic of Fig. 2.3 is appropriate (Ref. 2).

The effectiveness for this arrangement is then given by the equation (Ref. 2):

$$\varepsilon = \frac{[(1-\varepsilon_p C^*)/(1-\varepsilon_p)]^n - 1}{[(1-\varepsilon_p C^*)/(1-\varepsilon_p)]^n - C^*} \qquad (2.16)$$

where $\varepsilon_p$ is the effectiveness of the individual tube bundles and n is the number of tube bundles in series. For 2-pass cross-counterflow tube bundles in series the values of $\varepsilon_p$ would be taken from Equations 2.14 or 2.15. The overall efficiency would then be calculated from

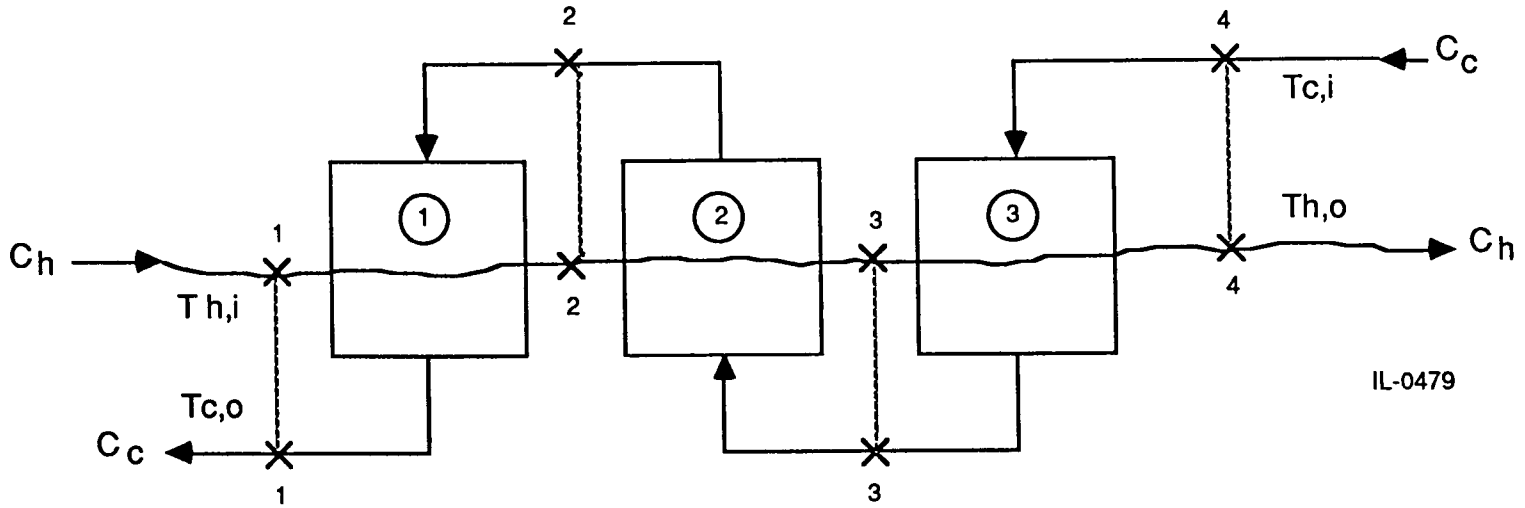



Figure 2.3. Overall Counterflow Heat Exchangers in Series

Equation 2.16 without the need to derive intermediate temperatures. A similar procedure would apply for 4-pass cross-counterflow tube bundles in series.

## 2.4 Heat Transfer Coefficients

The previous discussion of the effectiveness-NTU procedure introduced the factors required to utilize this procedure, namely $C^*$, NTU, $\Delta T_i$ and a knowledge of the physical geometry of the heat exchanger. Of these factors, $C^*$ and $\Delta T_i$ simply require specification of the hot and cold stream mass flow rates, specific heats and inlet temperatures. Two of the terms of $NTU=UA/C_{min}$, i.e. A (heat transfer area) and $C_{min}$, require either statement of the flows or a straight forward calculation of area based on the given geometry of the exchanger. It is the average overall heat transfer coefficient U which requires additional consideration.

### 2.4.1 The Overall Heat Transfer Coefficient

The heat transfer rate equation for a heat exchanger is written as $q=UA\ \Delta T_m$ (Equation 2.8) where q is the total heat energy transferred per unit time. A is a representative total heat transfer area for the exchanger and $\Delta T_m$ is a representative average overall temperature difference (hot fluid to cold fluid). U is the average overall heat transfer coefficient in heat energy transferred per unit

area per degree of average temperature difference. Total area A is chosen consistently with the definition of U. For many heat exchanger applications, and particularly for the SHTM analysis, A is based on the total outside surface of tubing exposed to the flue gas. Accordingly U must be developed on the basis of this choice for total area.

The individual factors which make up the overall heat transfer coefficient can be illustrated by looking at a local heat transfer coefficient for a representative element of the heat exchanger. Once the elements of the local coefficient are determined, it is assumed that U is reasonably constant throughout the exchanger and the value of the overall coefficient is calculated by using average fluid properties.

Consider an element of heat exchanger tubing typical of the SHTM (Fig. 2.4). The heat transfer ($q_x$) to this element is by a combination of radiation and convection from the flue gas to the tube surface. This quantity of heat is then transferred by conduction through the composite wall made up of the tube itself and whatever deposits (ash, scale, etc.) are on the outside and/or inside of the tube. Finally $q_x$ is transferred by convection from the tube inside area to the cooling fluid (water, steam, or air).

It can also be noted that at each stage of the heat transfer there is a resistance to the flow of heat. The heat flow through this resistance path is often represented by an electrical resistance analogy as illustrated in Fig. 2.5.

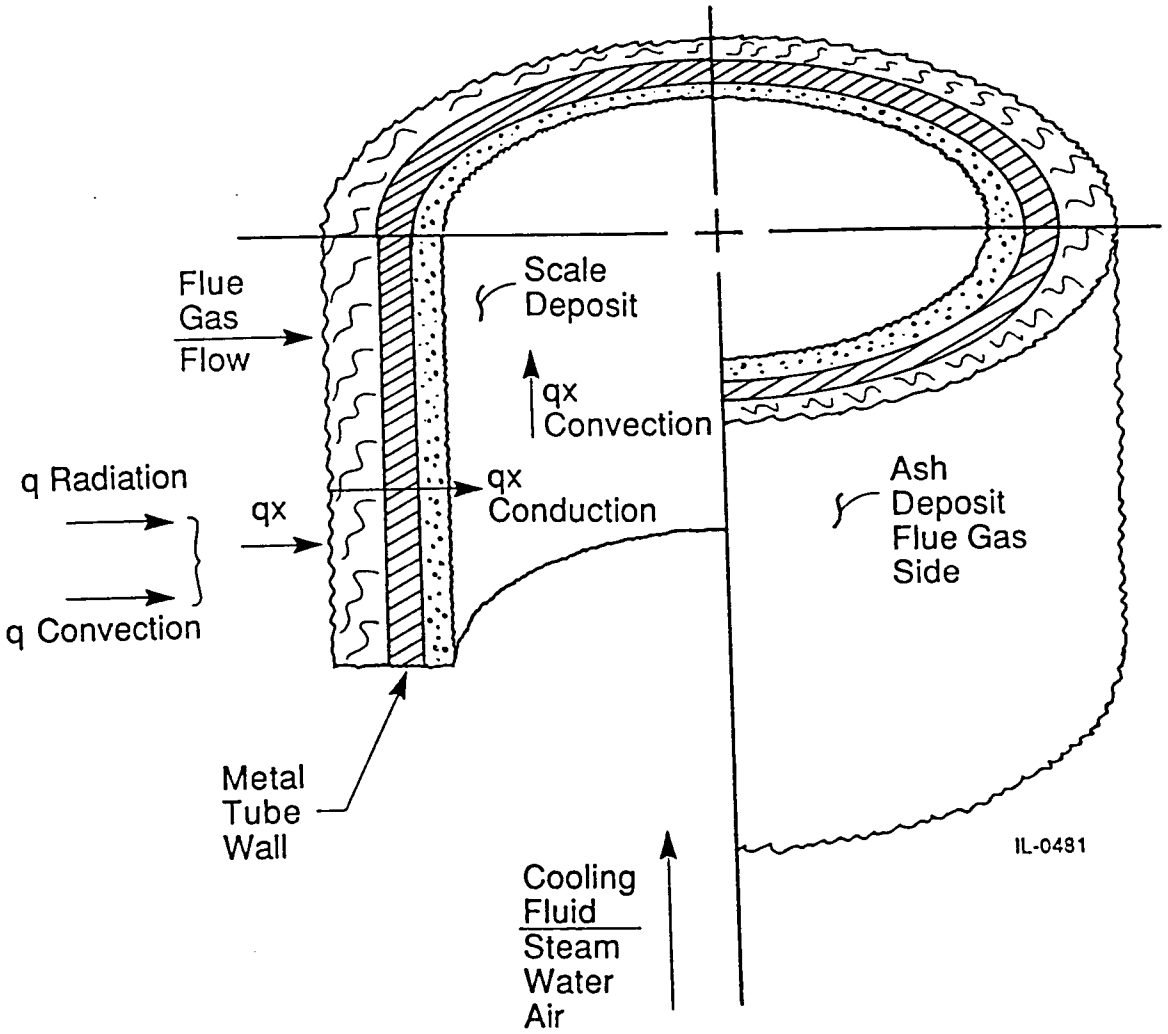
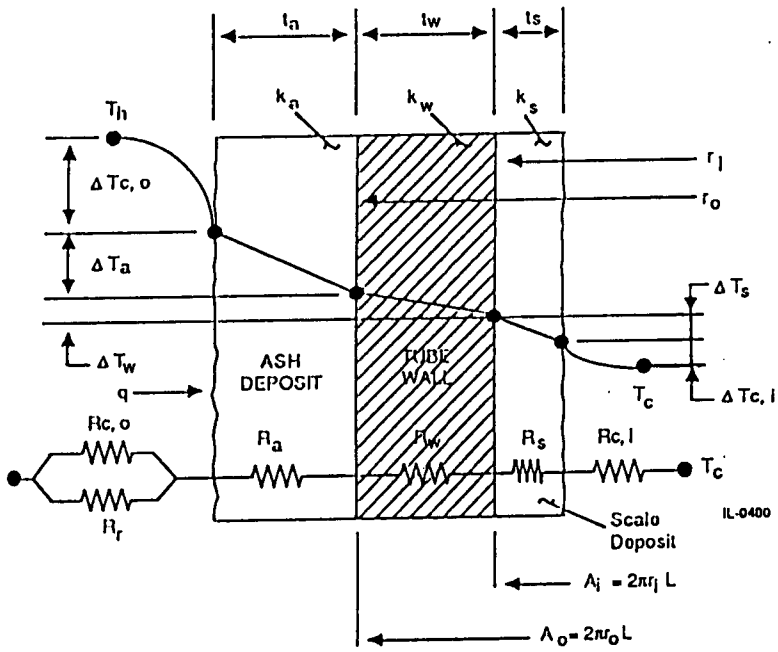



Figure 2.4. Typical Tube Wall Cross-Section




$R_{c,o}$ = gas side convection resistance = $(1/h_{c,o} A_o)$
$h_{c,o}$ is the outside convection coefficient

$R_r$ = gas side radiation resistance = $(1/h_r A_o)$
$h_r$ is the radiation coefficient

$R_a$ = gas side ash deposit conduction resistance = $(t/A\ k)_a$

$R_w$ = tube wall conduction resistance = $(t/A\ k)_w$

$R_s$ = cooling side scale deposit conduction resistance = $(t/A\ k)_s$

$R_{c,i}$ = cooling side convection resistance = $(1/h_{c,i} A_i)$
$h_{c,i}$ is the inside convection coefficient

$A_a$, $A_w$, $A_s$ are representative areas for the ash, tube wall and scale respectively

$k_a$, $k_w$, $k_s$ are the thermal conductivities for the ash, tube wall and scale respectively

$T_h$ is the flue gas local bulk temperature

$T_c$ is the cooling fluid local bulk temperature

Figure 2.5
Thermal Resistance Analogy for a Heat Exchanger Section

In this analogy the interface temperatures represent potential (or driving force), the resistances are the various radiation, convection, and conduction heat transfer coefficients, and heat is the current flow. The electrical analogy of $I = \Delta E/\Sigma R$ (Ohm's Law) has the thermal equivalent:

$$q = \Delta T/\Sigma R = (1/\Sigma R)\Delta T \tag{2.17}$$

where $\Delta T = T_h - T_c$ is the local overall temperature difference. If only convection were present on the flue gas side then $\Sigma R$ would be simply the linear sum of the resistances. The presence of parallel resistances, i.e. convection and radiation, result in the total resistance equation:

$$\Sigma R = \frac{1}{\frac{1}{R_r} + \frac{1}{R_{c,o}}} + R_a + R_w + R_s + R_{c,i} \tag{2.18}$$

or by substitution of heat transfer terms:

$$R_{total} = \frac{1}{A_o h_r + A_o h_{c,o}} + \left(\frac{t}{Ak}\right)_a + \left(\frac{t}{Ak}\right)_w + \left(\frac{t}{Ak}\right)_s + \frac{1}{A_i h_{c,i}} \tag{2.19}$$

Comparing Equations 2.8 and 2.17 shows that:

$$1/R_{total} = UA, \text{ or, } U = 1/(A\,R_{total}) \tag{2.20}$$

The final step in determining the general terms of U is to normalize the area basis. The area term A in Equation 2.8 refers to a representative area; in this case the outside surface of the heat transfer tubes. The individual area terms in the equation for $R_{total}$ (Equation 2.19) refer to the specific areas where the heat transfer is taking place. Multiplying through $R_{total}$ by the outside area then gives the general form for the terms in the overall heat transfer coefficient U.

$$U = \frac{1}{\frac{1}{h_r+h_{c,o}} + \frac{A_o}{A_a}\left(\frac{t_a}{k_a}\right) + \frac{A_o}{A_w}\left(\frac{t_w}{k_w}\right) + \frac{A_o}{A_s}\left(\frac{t_s}{k_s}\right) + \frac{A_o}{A_i}\frac{1}{h_{c,i}}} \qquad (2.21)$$

At this point the form of the overall coefficient can be simplified because of the specific nature of this analysis. It is common practice in the analysis of heat exchangers to neglect the resistance of metallic heat transfer surfaces (Ref.2). The reason for this is that the thermal resistance of a thin metallic surface such as a tube wall is typically one or two orders of magnitude lower than typical fluid flow convection coefficients. This is certainly the case for all of the tube banks of the SHTM which are steel tubes with thermal conductivities between 9 and 27 Btu Ft/Hr $Ft^2$ $^oF$ and thicknesses in the range of 0.1 to 0.2 inch.

A second simplification is made by eliminating the scaling and fouling resistances from the overall coefficient equation. In the case of the inside scaling resistance, this factor for the specific SHTM cases is negligible. Typical values for scaling resistance for steam, air, and water in heat transfer tubes (Ref. 1) are orders of magnitude lower than even the tube wall resistance. In the case of the outside resistance, this factor is quite significant, but will be handled in a different way. A study (Ref. 5) has been made by UTSI researchers on the effect of fouling on heat transfer in an MHD environment. This study has provided fouling factors in the form of multipliers which are to be applied to the "clean tube" overall heat transfer coefficients. Therefore, outside wall fouling is eliminated from the overall heat transfer coefficient equation in order to calculate a "clean wall" coefficient which is then adjusted for the fouling effect which varies with flue gas and tube surface temperature throughout the SHTM.

With these simplifying assumptions, the overall heat transfer coefficient (Equation 2.21), on a "clean tube" basis, reduces to:

$$U_{clean} = \frac{1}{\frac{1}{h_r+h_{c,o}} + \left(\frac{A_o}{A_i}\right)\frac{1}{h_{c,i}}} = \frac{(h_r+h_{c,o})\, h_{c,i}^*}{h_r + h_{c,o} + h_{c,i}^*} \qquad (2.22)$$

where $h_{c,i}^*$ is the inside convection coefficient times the area correction factor $A_o/A_i$. The operating or "dirty" coefficient is then calculated by applying a "cleanliness factor" as a direct multiplier:

$$U_{dirty} = F\ U_{clean} \tag{2.23}$$

Cleanliness factors are discussed in Section 4.2.2.

### 2.4.2 Cooling Fluid Convection Coefficient

Expressions for the individual heat transfer coefficients $h_{c,o}$, $h_{c,i}$ and $h_r$ are taken from standard industrial heat exchanger literature (Refs. 1,6). Although there are three different cooling fluids (water, steam, and air) used in the SHTM, all flow in circular ducts (tubes) at mass velocities which result in turbulent flow (Re>3000). A commonly accepted correlation for turbulent flow heat transfer in circular ducts is the Dittus-Boelter equation (Ref. 1). The dimensionless form of this equation is:

$$Nu = 0.023\ (Re)^{.8}\ (Pr)^{.4} \tag{2.24}$$

for heating of the fluid and properties evaluated at the fluid film temperature. The expanded form of this equation gives the inside convection heat transfer coefficient in terms of the applicable

geometry and fluid properties:

$$h_{c,i} = 0.023\ (GD_i/\mu)^{.8}\ (C_p\mu/k)^{.4}\ (k/D_i) \qquad (2.25)$$

where: $D_i$ is the inside diameter of the tube

$G$ is the mass velocity

$\mu$ is the fluid viscosity

$C_p$ is the fluid specific heat

$k$ is the fluid thermal conductivity

### 2.4.3 Flue Gas Convection Coefficient

All of the heat transfer tubes in the SHTM are oriented vertically with the flue gas flowing horizontally (i.e. in crossflow). Several empirical correlations for heat transfer in crossflow over tube banks were available (Refs. 1,6,7). The correlation chosen for this study was developed by the Babcock & Wilcox Company (Ref. 6) specifically for steam boiler applications. One particular feature of this correlation is the inclusion of a tube bank depth factor. This is useful for the SHTM analysis since there are tube bundles of two, four and twelve pass depth. The non-dimensional form of this equation

is similar to the Dittus-Boelter equation:

$$Nu = .287\ (Re)^{.61}\ (Pr)^{.33}\ F1\ F2 \qquad (2.26)$$

with fluid properties evaluated at the fluid film temperature. The F1 and F2 terms are tube bank geometry correction factors. The expanded form of this equation gives the outside cross flow heat transfer coefficient in terms of the applicable geometry and fluid properties:

$$h_{c,o} = .287\ (GD_o/\mu)^{.61}\ (C_p\mu/k)^{.33}\ k/D_o\ F1\ F2 \qquad (2.27)$$

where: $G, \mu, C_p$, and k are as defined for Equation 2.25

Do is the tube outside (clean) diameter

F1 is a tube bank spacing correction factor

F2 is a tube bank depth correction factor

### 2.4.4 Flue Gas Radiation Coefficient

The radiation coefficient which was chosen was developed, as above, by the Babcock & Wilcox Company (Ref. 6) for radiation from a coal flame to tubes in a steam boiler. As presented in Ref. 6, the terms of the correlation are presented in graphical form. For the computer model these graphs were represented by curve fit equations.

The form of the radiation coefficient is:

$$h_r = U'r\ K \tag{2.28}$$

where U'r is a radiation conductance based on the temperature difference between the gas and the tube surface. K is a correction factor related to the tube spacing, tube diameter, and gas composition.

The basis for applying the effectiveness-NTU method of heat exchanger analysis has now been developed. Subsequent chapters will provide more detail about the arrangement and algorithm of the analysis program (Section 3.0) and results of the analysis (Section 4.0).

# SECTION 3.0

## THE SUPERHEATER TEST MODULE COMPUTER CODE

The previous chapter discussed the theoretical and empirical aspects of the heat transfer model but did not describe the mechanics of the program itself. Some aspects of its arrangement that are appropriate to the understanding of the model are presented here. A listing of the program is provided in the Appendix.

Actually, the structure and arrangement of the program are similar to the SHTM. As discussed previously, the SHTM is assembled from individual sections. Each section has matching flanges so that, at least in concept, the sections could be arranged in any order. The computer model has a similar arrangement, in that the program is "assembled" from individual subprograms (each representing an individual section of the SHTM) which can be called in any order.

The basic sections of the SHTM are the screen tubes, the Cooling Sections, the Test Sections, the Air Heater, and the various flow transitions and cavity sections. The cooling sections are subdivided into twelve pass "original" sections (Fig. 3.1) and the "modified" cooling sections with additional heat transfer tube surface (Fig. 3.2). There are three configurations of the Test Sections; Test Section 1 (Fig. 3.3) which has a 2-pass tube bundle and provision for an upstream screen of water cooled tubes, Test Section 2 (Fig. 3.4)

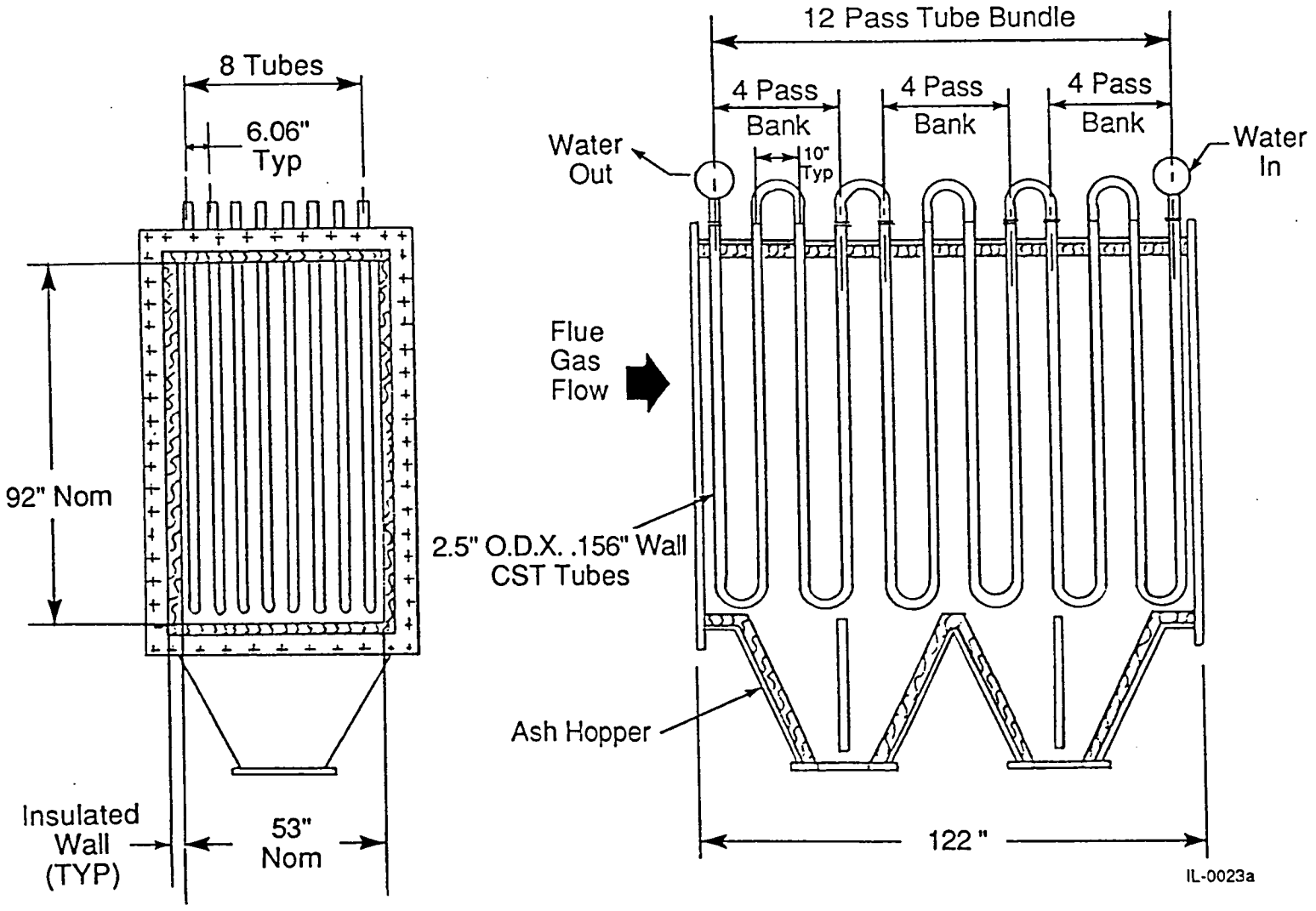



Figure 3.1. SHTM Cooling Section

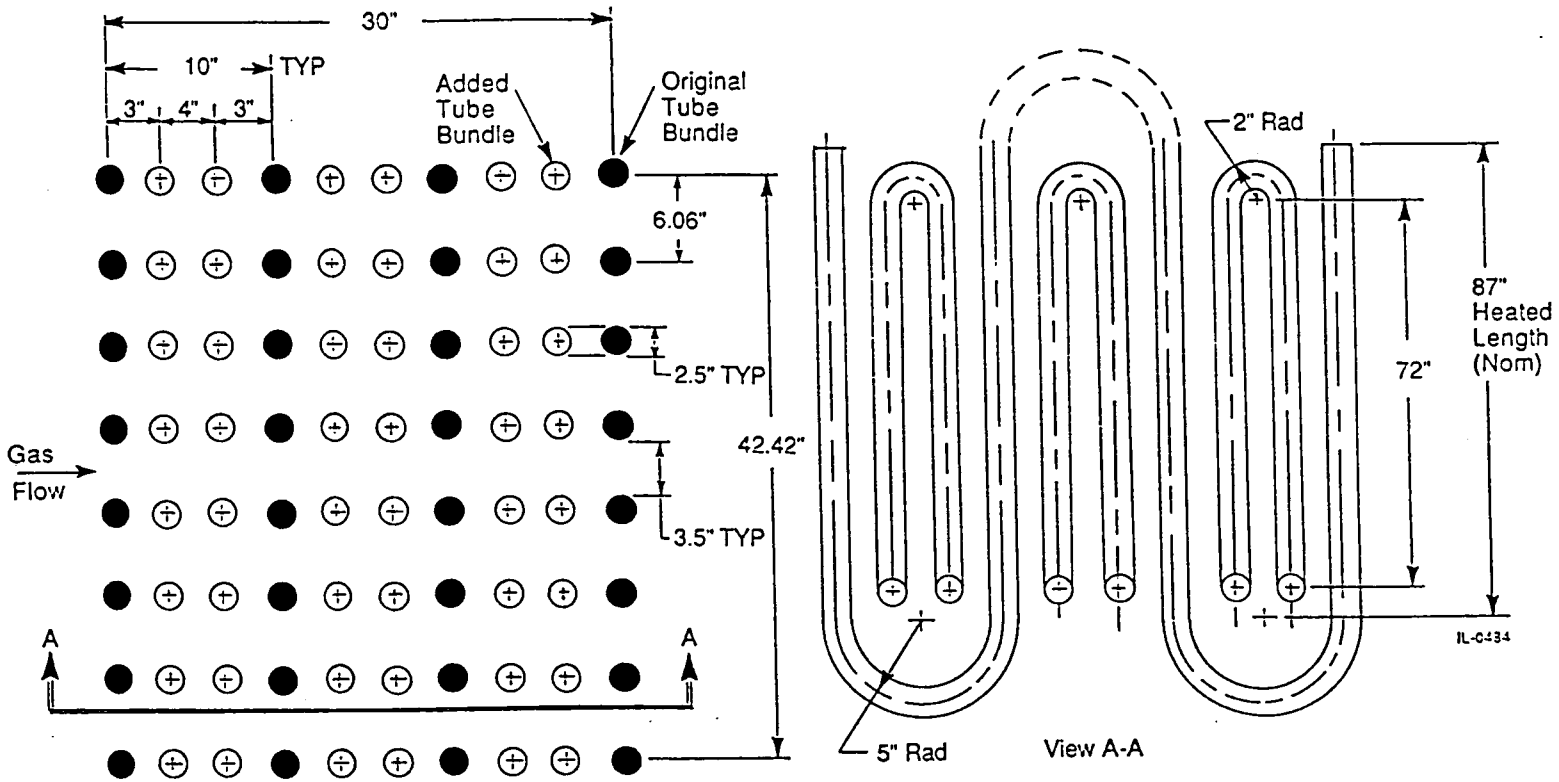



Figure 3.2. Extra Cooling Surface Added to Cooling Section

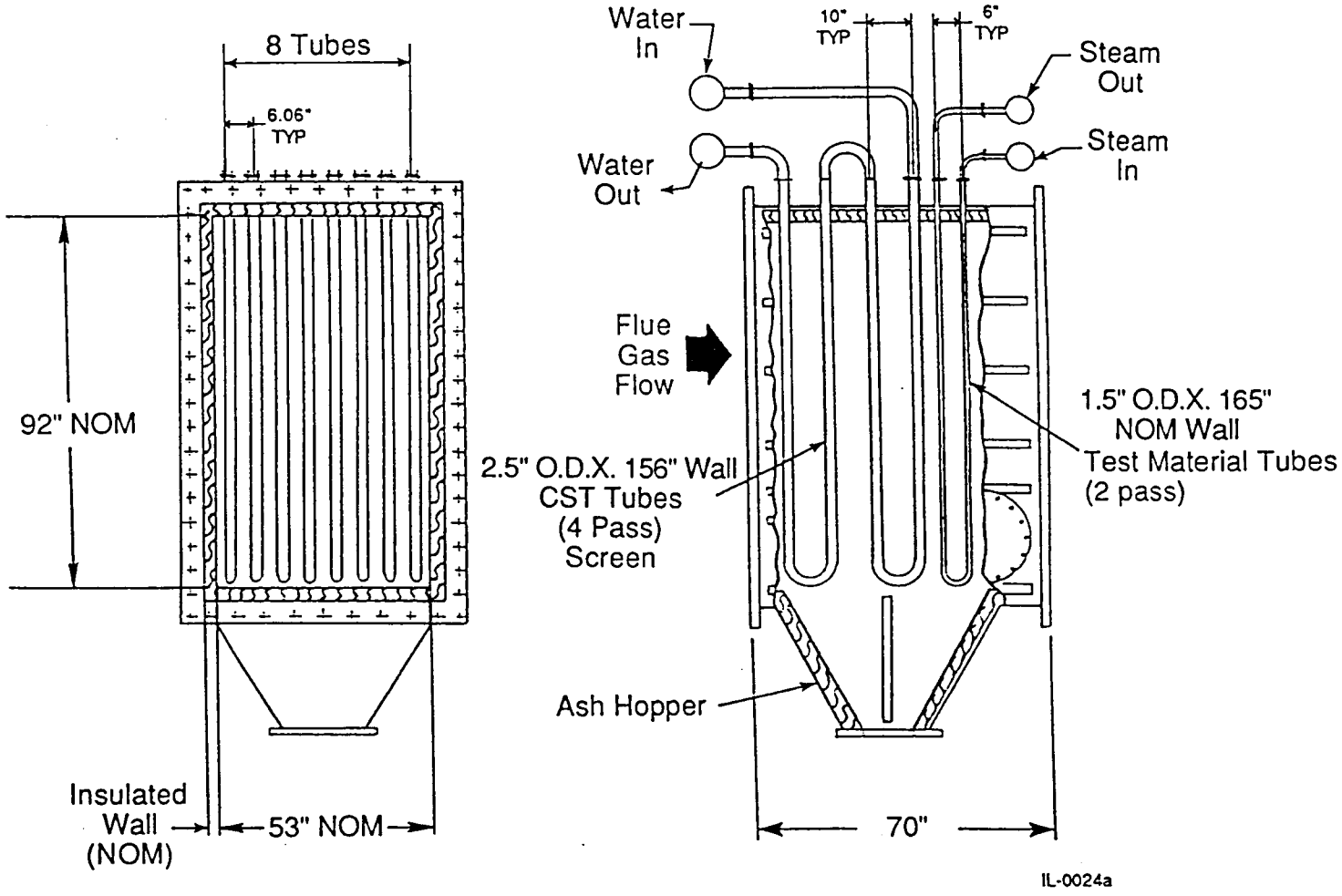



Figure 3.3. SHTM Test Section 1 with Screen Tubes

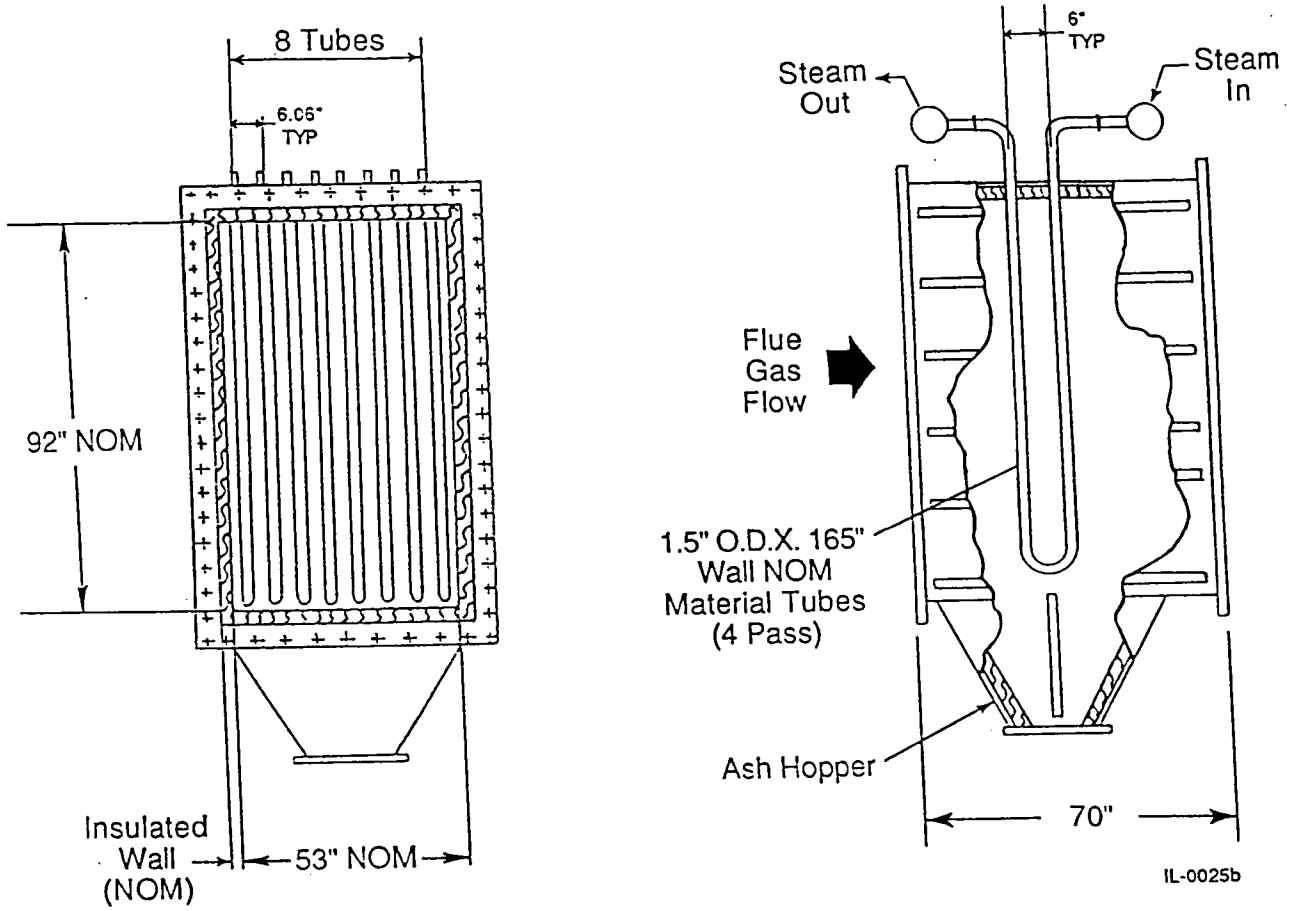



Figure 3.4. SHTM Test Section 2

which has a 2-pass tube bundle, and Test Section 3 (Fig. 3.5) which has a 4-pass tube bundle. The Air Heater is shown in Fig. 3.6.

In order to model configurations of the SHTM which have been tested in the past and to anticipate future arrangements, it is necessary to have sections which are variations of the basic sections listed above. Removal of the water cooled screen tubes or the steam cooled test section tubes to create cavities are typical examples of these special sections.

Each of the sections indicated above has a corresponding "mathematical" model in the form of a subprogram in the SHTM computer code. In these cases the subprogram is a representation of an actual piece of hardware. There are also subprograms which represent "pseudo-sections", or sections which do not exist in actual hardware but which are useful in representing possible SHTM arrangements. Each cooling section normally has three tube bundles in series. The design of the cooling sections is such that any (or all) of these water cooled tube bundles could be removed and the resulting cavity either left empty or used for a steam cooled tube bundle. Any such combination of tube bundles and cavities within the volume of a cooling section could be represented by subprograms called in the proper sequence.

The algorithm of the computer model is straightforward. An input list defines the sequence of sections (subprograms) through which the flue gas is to flow, beginning at the outlet of the radiant furnace and ending at the entrance to the pollution control devices. The

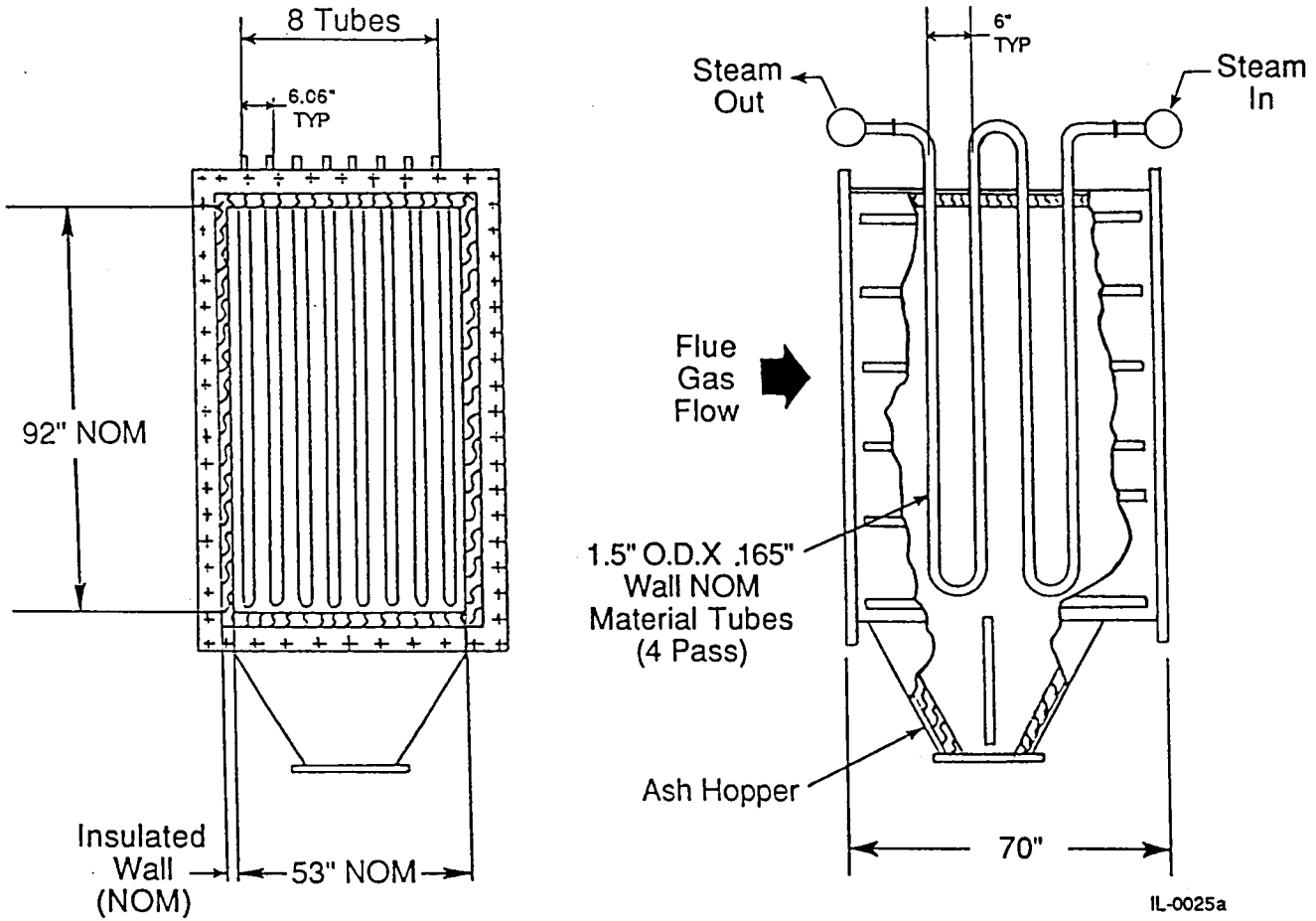



Figure 3.5. SHTM Test Section 3

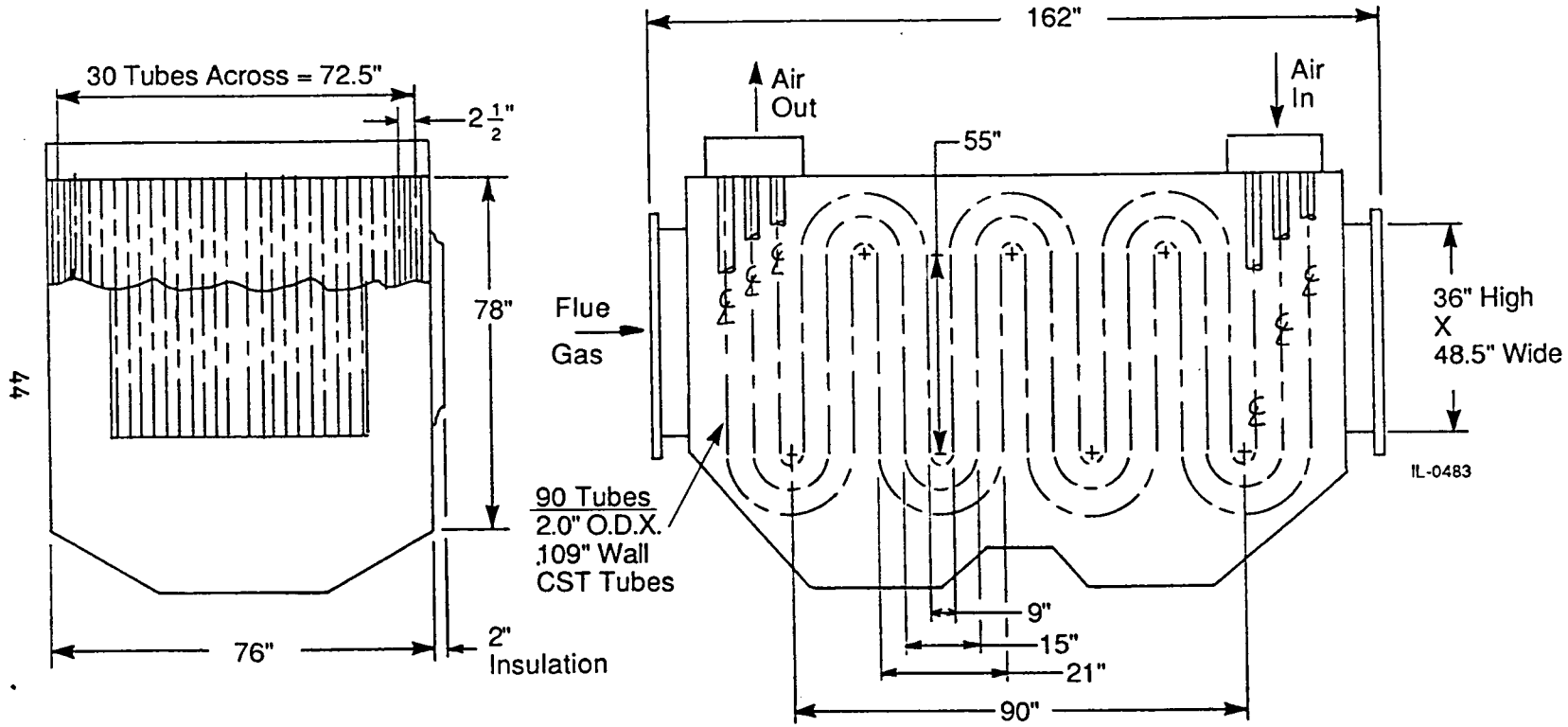



Figure 3.6. Air Heater

thermodynamic properties (temperature, specific heat, viscosity, and thermal conductivity) and initial mass flow rate of the flue gas are specified at the entrance to the first section. Equations for the temperature dependence of the flue gas properties are included. The inlet cooling fluid temperature and flow rate per tube are specified for each section with heat transfer tubes. A cleanliness factor and an estimate for inleakage of ambient air into the flue gas are provided for each section. The cleanliness factors and inleakage values are discussed in detail in Section 4.0. The other inputs required for calculation of heat transfer coefficients and total heat transfer, such as tube number and spacing, type of cooling fluid, etc., are preprogrammed into the specific subprograms.

As discussed in Section 2.0, the calculation of heat exchanger performance parameters requires the calculation of average heat transfer coefficients. Each subprogram begins its calculations with an initial estimate of the flue gas and cooling fluid outlet temperatures, then proceeds to calculate the outlet temperatures from thermodynamic considerations. The initial guess and subsequent calculation are compared for convergence. Assuming that the first iteration is not within the desired accuracy, the calculated outlet temperatures then become the "guess" for the next iteration. This process continues until the estimate and calculation converge within the tolerance specified in the computer model.

The output of the computer model includes the following calculated values for each section:

1. convection heat transfer coefficient for cooling fluid on the inside of tubes
2. convection heat transfer coefficient for flue gas on outside of tubes
3. radiation coefficients on outside of tubes
4. overall heat transfer coefficient, "clean" and "dirty"
5. total heat transfer to tubes
6. heat transfer to walls
7. NTU value and efficiency factor $\varepsilon$
8. cooling fluid outlet temperature
9. flue gas flow rate (including inleakage)
10. flue gas outlet temperature

# SECTION 4.0

# RESULTS AND DISCUSSION

Results of this analysis will be presented in three areas; heat transfer characteristics of the SHTM, validity of the model, and prediction of proposed configurations. The first area will emphasize specific aspects of the heat exchanger analysis such as heat transfer coefficients and efficiencies of the various sections of the SHTM. The second area will compare the calculations of the model against measured data. This comparison is primarily based on measured and predicted (calculated) flue gas temperatures along the length of the SHTM. In this area the inputs to the program (which are the physical operating conditions of the CFFF) are at least as important as the heat transfer equations. Particular emphasis will be given to a review of the flow and temperature measurements taken at the CFFF. An indication will be given of the effect that measurement uncertainty has on the computer model calculations. The final section will give the results of predictions of SHTM flue gas profiles for proposed configurations.

## 4.1 General Heat Transfer Characteristics

Before reviewing the heat transfer characteristics of the SHTM, it should be emphasized that the SHTM was designed for specific

purposes and to fit a specific test facility (i.e. the CFFF). The purposes of the SHTM were to: 1) achieve specific tube metal temperatures in three flue gas temperature regions, 2) provide heated air for combustion in the secondary combustor, and 3) reduce the overall flue gas temperature to an acceptable level for downstream pollution control equipment. In order to fit in the CFFF, the SHTM had specific limitations of space, orientation, and utilities (i.e. limited steam, cooling water, and air). It was also imposed on the SHTM design that it be a piece of test hardware subject to modification and rearrangement.

All of these design and operational constraints resulted in the SHTM being a relatively poor heat exchanger. Table 4.1 gives typical heat transfer values for the various components of the SHTM. The water cooled tube banks have $\varepsilon$ and NTU values of about 0.1 and 0.05 respectively, while the steam cooled banks have $\varepsilon$ and NTU values in the 0.3 to 0.5 range for both. For comparison, typical shell and tube heat exchangers have $\varepsilon$ values of about 0.63 and NTU values of about 1.0 (Ref. 2). Note from Table 4.1 that the inclusion of additional heat transfer surface in the cooling sections (520% more surface) increases their effectiveness value by about 80%. The only component of the SHTM with a relatively high effectiveness ($\varepsilon$) value is the Air Heater ($\varepsilon$=0.7). This component has a densely packed tube bundle and a $C_{min}/C_{max}$ or $C^*$ ratio of about 0.5. Note the extremely low values of $C^*$ for the other components which have low $\varepsilon$ values.

Table 4.2 gives typical heat transfer values and heat transfer

TABLE 4.1

TYPICAL NTU AND ε VALUES

| COMPONENT | FLUE GAS INLET $^{o}F$ | HEAT TRANSFER COEFFICIENTS Btu/hr $ft^2$ $^{o}F$ CLEAN | DIRTY | $C^*$ | NTU | ε |
|---|---|---|---|---|---|---|
| SCREEN TUBE BANK | 1900 | 9.33 | 5.32 | .067 | .064 | .061 |
| TS1 W/ SCREEN | 1730 | 8.26 | 3.30 | .058 | .200 | .177 |
| TS1 W/O SCREEN | 1900 | 8.81 | 3.53 | .058 | .212 | .188 |
| COOLING SECTION 1 | 1653 | 8.46 | 4.74 | .068 | .056 | .153 |
| COOLING SECTION 2 | 1370 | 7.41 | 4.15 | .067 | .050 | .137 |
| TEST SECTION 2 | 1158 | 6.70 | 5.36 | .060 | .328 | .276 |
| COOLING SECTION 3 | 1123 | 6.43 | 4.76 | .067 | .057 | .155 |
| TEST SECTION 3 | 934 | 5.12 | 3.13 | .064 | .192 | .313 |
| COOLING SECTION 4 | 904 | 5.64 | 3.72 | .067 | .044 | .124 |
| COOLING SECTION 5 | 780 | 5.19 | 2.65 | .067 | .032 | .090 |
| CS5 W/EXTRA TUBES | 808 | 4.08 | 2.08 | .049 | .026 | .165 |
| AIR HEATER | 693 | 4.12 | 3.91 | .514 | .367 | .681 |
| COOLING SECTION 6 | 467 | 4.39 | 2.28 | .066 | .028 | .079 |
| CS6 W/EXTRA TUBES | 453 | 3.48 | 1.81 | .048 | .023 | .148 |

NOMINAL OPERATING CONDITIONS

Inlet Flue Gas Flow = 38,000 lb/hr
Assumed Total Air Leakage = 20%
Outlet Flue Gas Flow = 45,600 lb/hr
Inlet Flue Gas Temperature = 1900$^{o}$F
Flue Gas Properties at Inlet:
    Specific Heat = .33 Btu/lb $^{o}$F
    Viscosity = .114 lb/hr ft
    Thermal Conductivity = .05 Btu/hr ft $^{o}$F
Cooling Water Inlet Temperature = 130$^{o}$F
Cooling Water Flow Rate (per tube) = 24,000 lb/hr
Steam Inlet Tempertature = 280$^{o}$F (saturated)
Steam Flow Rate (per tube) = 200 lb/hr
Air Inlet Temperature = 100$^{o}$F
Air Flow Rate (per tube) = 300 lb/hr

TABLE 4.2

TYPICAL HEAT TRANSFER COEFFICIENTS

| | HEAT TRANSFER COEFFICIENTS BTU/HR $FT^2$ $^oF$ | | | | | | HEAT TRANSFER BTU/HR | | |
|---|---|---|---|---|---|---|---|---|---|
| NAME | CONVECTION TUBE ID | CONVECTION TUBE OD | TUBE RAD | CAVITY RAD | CLEAN COEFF | DIRTY COEFF | TO TUBES (DIRTY) | TO WALLS | TOTAL (DIRTY) |
| INLET | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | 0.0 | $0.16 \times 10^6$ | $0.16 \times 10^6$ |
| SCREEN | 883.7 | 5.2 | 2.9 | 1.0 | 9.0 | 5.2 | $1.30 \times 10^6$ | $0.04 \times 10^6$ | $1.34 \times 10^6$ |
| TS1 | 27.1 | 6.8 | 2.9 | 2.0 | 8.2 | 3.3 | $0.19 \times 10^6$ | $0.03 \times 10^6$ | $0.22 \times 10^6$ |
| CS1 | 897.0 | 5.5 | 3.0 | 0.0 | 8.5 | 4.7 | $3.05 \times 10^6$ | $0.11 \times 10^6$ | $3.16 \times 10^6$ |
| CS2 | 885.4 | 5.3 | 2.2 | 0.0 | 7.4 | 4.2 | $2.20 \times 10^6$ | $0.11 \times 10^6$ | $2.31 \times 10^6$ |
| TS2 | 27.9 | 6.5 | 1.4 | 1.0 | 6.7 | 5.4 | $0.19 \times 10^6$ | $0.06 \times 10^6$ | $0.25 \times 10^6$ |
| CS3 | 882.5 | 5.0 | 1.4 | 0.0 | 6.4 | 4.8 | $1.99 \times 10^6$ | $0.11 \times 10^6$ | $2.10 \times 10^6$ |
| TS3 | 26.6 | 5.3 | 0.6 | 0.5 | 5.1 | 3.1 | $0.16 \times 10^6$ | $0.06 \times 10^6$ | $0.22 \times 10^6$ |
| CS4 | 873.2 | 4.9 | 0.8 | 0.0 | 5.6 | 3.7 | $1.24 \times 10^6$ | $0.11 \times 10^6$ | $1.35 \times 10^6$ |
| CS5* | 867.6 | 3.8 | 0.2 | 0.0 | 4.0 | 2.0 | $1.36 \times 10^6$ | $0.11 \times 10^6$ | $1.47 \times 10^6$ |
| A/H | 13.0 | 6.1 | 0.0 | 0.0 | 4.2 | 3.9 | $2.46 \times 10^6$ | $0.06 \times 10^6$ | $2.52 \times 10^6$ |
| CS6* | 861.2 | 3.5 | 0.0 | 0.0 | 3.5 | 1.8 | $0.57 \times 10^6$ | $0.11 \times 10^6$ | $0.68 \times 10^6$ |
| | | | | | | TOTALS | $14.71 \times 10^6$ | $1.07 \times 10^6$ | $15.78 \times 10^6$ |

* With Extra Tubes

OPERATING CONDITIONS: per Table 4.1

coefficients for the SHTM sections. The water-cooled tube banks take about 75% of the total heat transfer from the flue gas. Each test section takes about 1.0% to 1.5% and the Air Heater takes about 15%. Heat loss to the walls of the sections accounts for about 7% of the total loss. This distribution of heat loss among the sections is consistent with the study of Ref. 8 which included a more detailed analysis of the radiation and wall heat losses.

In Section 2.4.1 the overall heat transfer coefficient was developed as a sum of resistances to heat flow. The tube wall resistance was eliminated because it was found to be negligible in relation to the other factors. In a similar manner, the water side convection coefficient for the cooling section tubes could have been eliminated since it is at least two orders of magnitude greater than the other coefficients, i.e. flue gas convection and radiation. The convection coefficients for the steam and air, though, are only about two to three times greater than the flue gas convection and are significant in the overall coefficient. Note, as expected, that the radiation coefficient (Table 4.2) decreases as the flue gas temperature decreases; finally becoming insignificant at about the Air Heater entrance (i.e. about $800^{o}F$).

## 4.2 Validity of Test Data and Assumptions

The discussion above generally describes the output of the program, but nothing was said to indicate if the calculations reflect

actual operation of the SHTM. Recall that the primary purpose of performing this analysis was to develop a heat transfer model of the SHTM which would allow prediction of flue gas temperatures for a variety of configurations and flow conditions. Since actual testing of the SHTM, in a variety of configurations, has been ongoing for three years, a large body of recorded test data exists. Comparison of the model predictions to this actual data is needed to evaluate the validity of the code.

Before proceeding to compare code predictions directly with measured data, it is appropriate to review the accuracy and validity of the measured data. It is also necessary to review the validity of non-measured quantities which are part of the input data to the code. Section 2.0 detailed the parameters necessary for the calculation of heat transfer coefficients. These parameters are basically the mass flow rates of the water, steam, air, and flue gas, and their thermodynamic properties at the temperatures and pressures of interest. The mass flow rates of the water, steam, and air are direct measurements (subject to measurement inaccuracies described later) and their properties are readily obtainable from reference materials for all conditions of interest in this analysis. Obtaining the total mass flow rate and properties of the flue gas, though, presents a different problem.

### 4.2.1 Flue Gas Flow and Properties

The flue gas which enters the SHTM is a mixture of several measured flows plus a significant amount of ambient air (some intentionally introduced and some not) which is not measured. Measured components of the flue gas are: fuel oil burned in the vitiation heater, air and gaseous oxygen introduced in the vitiation heater, the pulverized coal and seed mixture burned in the combustor, fuel oil burned in the combustor, gaseous nitrogen injected into the upstream flow as instrument purges, air introduced in the secondary combustor, and air injected into the downstream flow as instrument purges. Studies of the CFFF operation have indicated that leakage of ambient air into the flow stream plus sootblowing can be on the order of 10% to 20% (Ref. 9) of the total measured flow. The effects of this uncertainty in flue gas mass flow will be reviewed later.

The flue gas thermodynamic properties are the result of combustion and high temperature chemical reactions of fuel oil, coal, seed material, oxygen, and nitrogen. It is assumed for this study that the basic chemical reactions are complete within a short distance past the secondary combustor, i.e. just prior to the SHTM. There have been some indications that secondary combustion may be proceeding into the immediate upstream sections of the SHTM, but analysis of this condition is beyond the scope of this study. Determination of the specific heat, viscosity, and thermal conductivity of the flue gas as a function of temperature must be made for each combination of the

reactant flows. This chemical reaction calculation is fairly detailed and its description is beyond the scope of this study. A chemical reaction program for general equilibrium thermochemical systems, which was developed by the National Aeronautics and Space Administration (NASA) (Ref. 10), was used for the cases presented in this thesis.

Reference was made in the paragraphs above to inleakage of ambient air and sootblowing which contributed significantly to the total SHTM gas side mass flow but were unmeasured. The SHTM typically operates at a negative pressure of 20 to 30 inches water column. Along the length of the SHTM there are 17 sootblower ports which are known to allow inleakage of air plus other instrument ports which are also prone to leakage. Also, each test section and each cooling section has gasketed interfaces where the tube bundle tube sheets bolt to the section's box structure. It is expected that there is inleakage around some of these tube sheet gaskets.

Comparison of the measured total outlet flow from the SHTM with the sum of measured inlet flows indicates that total inleakage to the SHTM is on the order of 20% of the SHTM flow (Ref. 9). Since there are no measurements of individual leakages or sootblowing, and no intermediate flow measurements between the SHTM inlet and outlet, leakage values must be assumed arbitrarily. The computer code allows for the input of an inleakage value to each section as a percentage of the total measured inlet flow to the SHTM. The general effect of leakage and the effect of different choices for leakage rates are illustrated later in this section.

### 4.2.2 Cleanliness Factors

In Section 2.0 (The Heat Transfer Model), it was indicated that the outside tube wall fouling factor would be eliminated from the formulation of the overall heat transfer coefficient and accounted for differently. Another study of the SHTM (Ref. 8) used a resistance factor for the outside fouling based on an estimate for the thickness and composition of the deposits, and incorporated this factor into the overall heat transfer coefficient. The actual thickness and composition of the deposits throughout the SHTM during operation can only be estimated from post-test inspections.

An alternate method of accounting for the effect of fouling (gas side deposits) was used for this analysis. As will be discussed below, the cooling fluid flow rates, inlet temperatures and outlet temperatures are measured for each section of the SHTM. This allows the calculation of the heat transfer, q, in each section. With a knowledge of the total heat transfer area (A) for the sections and the flue gas inlet and outlet temperatures, also measured, Equation 2.8 (page 18) is used to back calculate an overall heat transfer coefficient U. If this process is performed early in a test when the tubes are clean (new or mechanically cleaned prior to the test) and again after an ash deposit has formed, then $U_{dirty}$ and $U_{clean}$ coefficients can be determined for each SHTM section. The ratio of these coefficients ($U_{dirty}$ / $U_{clean}$) can then be used as a

"cleanliness" factor "F" applicable to tube bundles of specific geometry and for specific gas compositions and temperatures. To use this factor analytically, i.e. when $U_{dirty}$ is being calculated empirically rather than taken from measured data, it is necessary to calculate $U_{clean}$ by Equation 2.22 (page 32) and then determine $U_{dirty}$ by the relation: $U_{dirty} = F\ U_{clean}$. $U_{dirty}$ is then used in the effectiveness-NTU calculation.

The use of this type of correction factor is only appropriate if applied in specific cases. As noted earlier, this analysis is specific to the existing SHTM and overall fouling factors have already been developed from test data (Ref. 5). A listing of the fouling factors used is given in Table 4.3. These factors are specific to the SHTM tube bundle geometries, flue gas compositions and temperature regimes. Application to any other heat exchanger or flue gas composition should be carefully considered.

### 4.2.3 Measured Cooling Fluid Parameters

The input parameters discussed above are either calculated indirectly, e.g. gas composition, or arrived at empirically, e.g. leakage and cleanliness factor. The other input values needed for the analysis can be directly measured in the actual hardware. These parameters are the cooling fluid inlet temperatures, pressures, and flow rates and the flue gas inlet temperature and inlet flow rate. Of these, the cooling fluid (water, steam, air) flow rates, inlet

TABLE 4.3

SHTM CLEANLINESS FACTORS

| | LMF4-C DATA | | LMF4-A THRU E SUMMARY | |
|---|---|---|---|---|
| SECTION | FLUE GAS INLET °F | CLEANLINESS FACTOR | CLEANLINESS FACTOR AVERAGE | MINIMUM |
| SCREEN | 2251 | .57 | .58 | .57 |
| TS1 | 1866 | .40 | .39 | .25 |
| CS1 | 1858 | .56 | .60 | .49 |
| CS2 | 1569 | .56 | .61 | .52 |
| TS2 | 1414 | .80 | .60 | .44 |
| CS3 | 1392 | .74 | .71 | .59 |
| TS3 | 1187 | .61 | .46 | .37 |
| CS4 | 1161 | .66 | .59 | .48 |
| CS5* | 1007 | .51 | .50 | .45 |
| AIR HTR | 779 | .95 | .84 | .70 |
| CS6* | 506 | .52 | .58 | .47 |

* WITH EXTRA TUBES

temperatures, and pressures are considered to be accurately measured (Ref. 11). The individual flows which make up the flue gas inlet flow rate, likewise, are considered to be accurately measured. The actual flue gas flow rate in individual SHTM sections, though, is dependent on the leakage rates which are not generally known with accuracy, as described above.

### 4.2.4 Measured Flue Gas Temperatures

The other parameters which need to be reviewed before comparing predictions (calculations) from the computer analysis with measured data are the cooling fluid outlet temperatures, the flue gas intermediate (i.e. between sections of the SHTM) temperatures, and the flue gas inlet temperature. Each of these values is measured in the actual hardware and are critical in validating the model predictions. Each of the cooling fluid outlet temperatures is measured in a header or pipe by standard thermocouples. The measured values for these parameters, therefore, like the inlet temperatures are considered to be relatively accurate. The flue gas temperatures (inlet and intermediate), then are the final values to be reviewed.

It is not within the scope of this analysis to do a detailed study of the measurement accuracy of instruments used in the CFFF. It can be shown, though, that the calculation of heat input to the cooling fluids of the various SHTM sections should be accurate to within a few percent (Ref. 11). If the measurement of flue gas

parameters were equally as accurate, then the calculation of heat loss from the flue gas in the corresponding sections would approximately balance. An actual heat balance on a section-by-section basis, though, shows differences in the 20% to 50% range, and over 150% in some cases. Table 4.4 gives typical comparisons of heat gain/loss calculations for the SHTM sections based on measured data. This table takes into account two unmeasured parameters for the flue gas heat loss; first it assumes a distributed amount of inleakage of ambient air and secondly it assumes a representative value for wall losses (about 5% of total heat loss). There is no apparent pattern. In some cases the test sections have a positive difference and in others they have a negative difference. The cooling sections, likewise, have differences that change absolute magnitude and relative magnitude (i.e. positive to negative) from case to case.

In previous paragraphs it was indicated that the cooling fluid temperature and flow measurements which are used in a heat gain calculation should be relatively accurate. It would appear, then, from the comparison above that the magnitude of the difference between calculated flue gas heat loss and cooling fluid heat gain per section would be primarily due to inaccuracies in the measurement of flue gas temperatures. This can not be stated with certainty since there is no reason to assume that the instrument readings at the points of measurement are inaccurate. It could be that the temperature readings are taken at locations not representative of the true flue gas average temperature. Figure 4.1 illustrates the locations where flue gas

TABLE 4.4

COMPARISON OF HEAT GAIN/LOSS VALUES
BASED ON MEASURED DATA

| SECTION | Qgas BTU/HR ($x10^6$) | FLUID | Qcooling BTU/HR ($x10^6$) | Qwalls BTU/HR ($x10^6$) | ΔQ BTU/HR ($x10^6$) | ΔQ% % |
|---|---|---|---|---|---|---|
| SCREEN | 5.98 | WATER | 3.50 | 0.04 | 2.44 | 41 |
| TS1 | 0.15 | STEAM | 0.35 | 0.03 | -0.23 | -153 |
| CS1 | 4.52 | WATER | 5.46 | 0.11 | -1.06 | -23 |
| CS2 | 2.49 | WATER | 3.52 | 0.11 | -1.14 | -46 |
| TS2 | 0.47 | STEAM | 0.22 | 0.07 | 0.18 | 38 |
| CS3 | 3.15 | WATER | 3.59 | 0.11 | -0.56 | -18 |
| TS3 | 0.47 | STEAM | 0.30 | 0.07 | 0.11 | 23 |
| CS4 | 2.54 | WATER | 1.79 | 0.11 | 0.63 | 25 |
| CS5* | 3.40 | WATER | 2.22 | 0.11 | 1.07 | 31 |
| AIR HTR | 4.24 | AIR | 3.38 | 0.06 | 0.80 | 19 |
| CS6* | 0.84 | WATER | 1.00 | 0.11 | -0.26 | -32 |
| TOTALS | 28.25 | | 25.33 | 0.93 | 1.98 | 7 |

NOTES:
* Indicates cooling section with extra tubes
1. Qgas = heat loss from flue gas based on measured flow plus an estimate (about 2% per section) for inleakage of ambient air and measured temperatures
2. Qcooling = heat gain by cooling fluid based on measured flow and temperatures
3. Qwalls = estimated heat loss through walls of section
2. ΔQ = Qgas - (Qcooling + Qwalls)
3. ΔQ% = ΔQ / Qgas

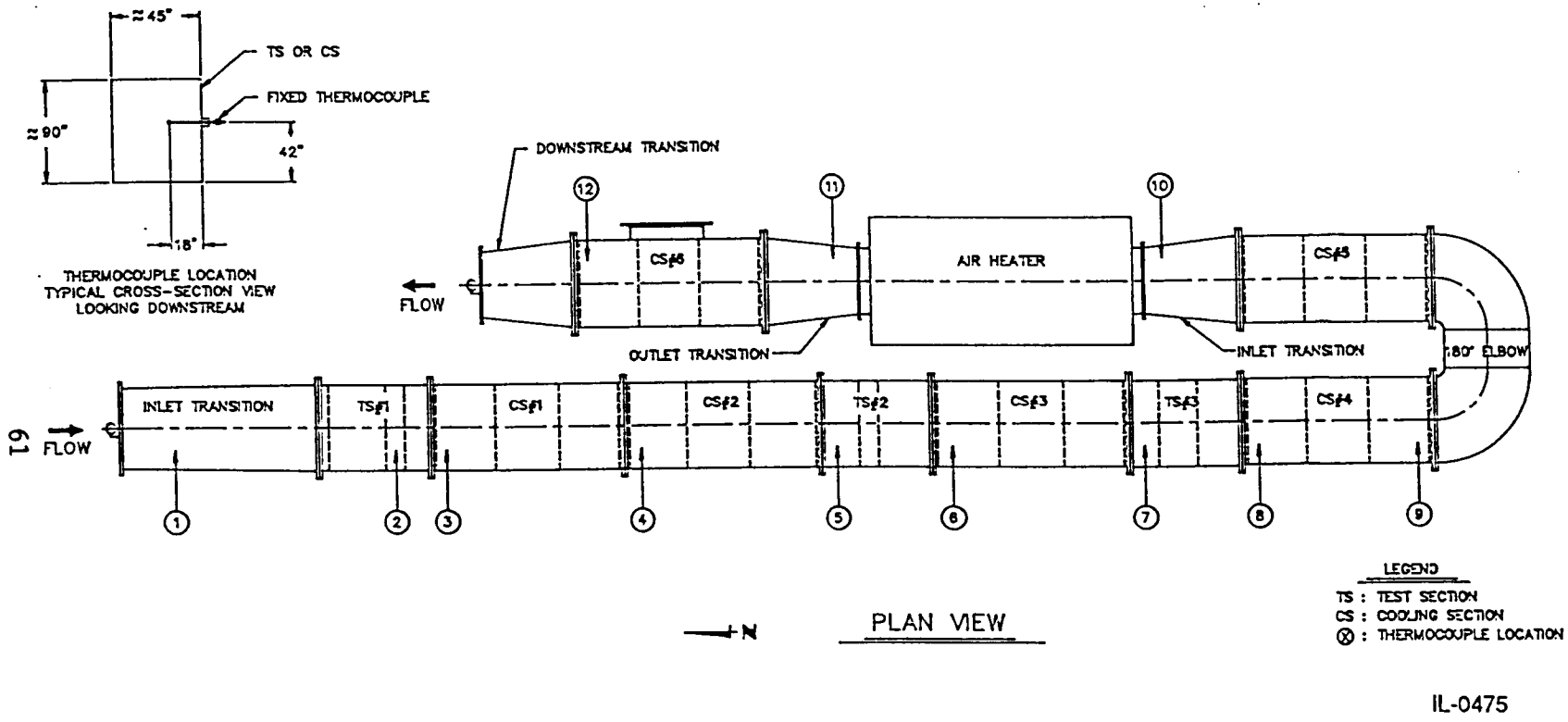



Figure 4.1. Thermocouple Locations Along SHTM

temperatures are taken in the SHTM. All of these are single point, fixed measurements at approximately the center of a 28 square foot flow area. It is particularly important to note the location of the measurement in the inlet transition (Position #1). This measurement is considered to be the SHTM inlet temperature and is used as a control parameter for operating the CFFF during testing.

Attempts have been made to measure the temperature profile (uniformity) of the flue gas at specific locations along the SHTM (Refs. 12,13). Measurements along a horizontal axis perpendicular to the flue gas flow at about the elevation of the fixed measurements have been made at several locations in the SHTM. One attempt has been made to date (Ref. 14) to take measurements at three elevations in the same plane in order to indicate a vertical profile. Typical temperature profiles for three locations are shown in Figures 4.2, 4.3 and 4.4 (Refs. 13,14). Figures 4.2 and 4.3 are horizontal profiles at the SHTM inlet transition and at a typical downstream location at about the centerline of the gas flow. Fig. 4.4 shows the three horizontal profiles taken at different elevations in Test Section 1 just upstream of the steam cooled tube bank. One problem with these measurements is that each point on the distribution takes about 3 to 5 minutes to record. There typically could be 90 minutes between the first measurement of the profile and the last measurement. Although it is intended to make these measurements during periods of steady flow, the flue gas temperature at the point of measurement could change significantly during this time.

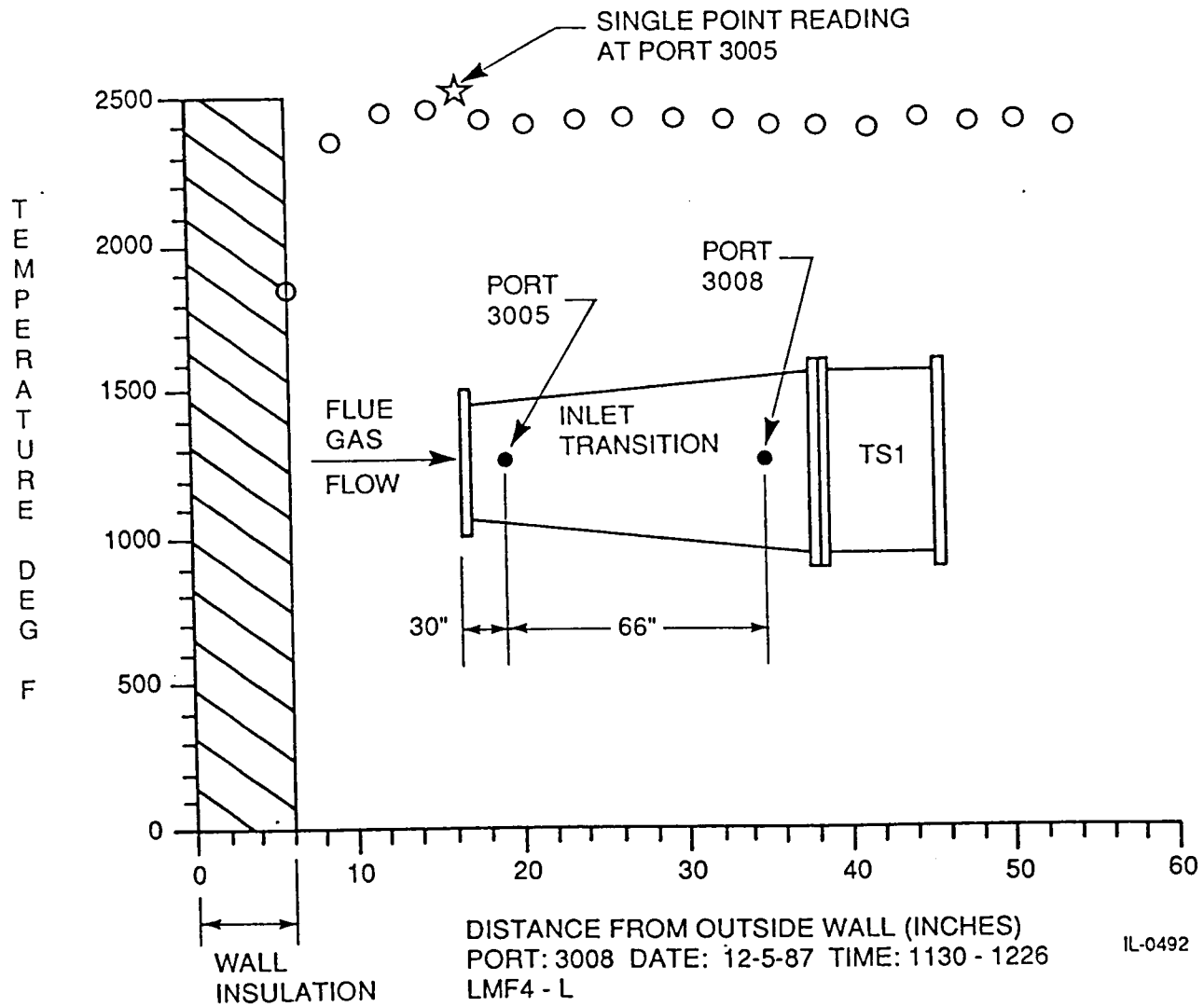



Figure 4.2. Horizontal Temperature Profile in SHTM Inlet Transition

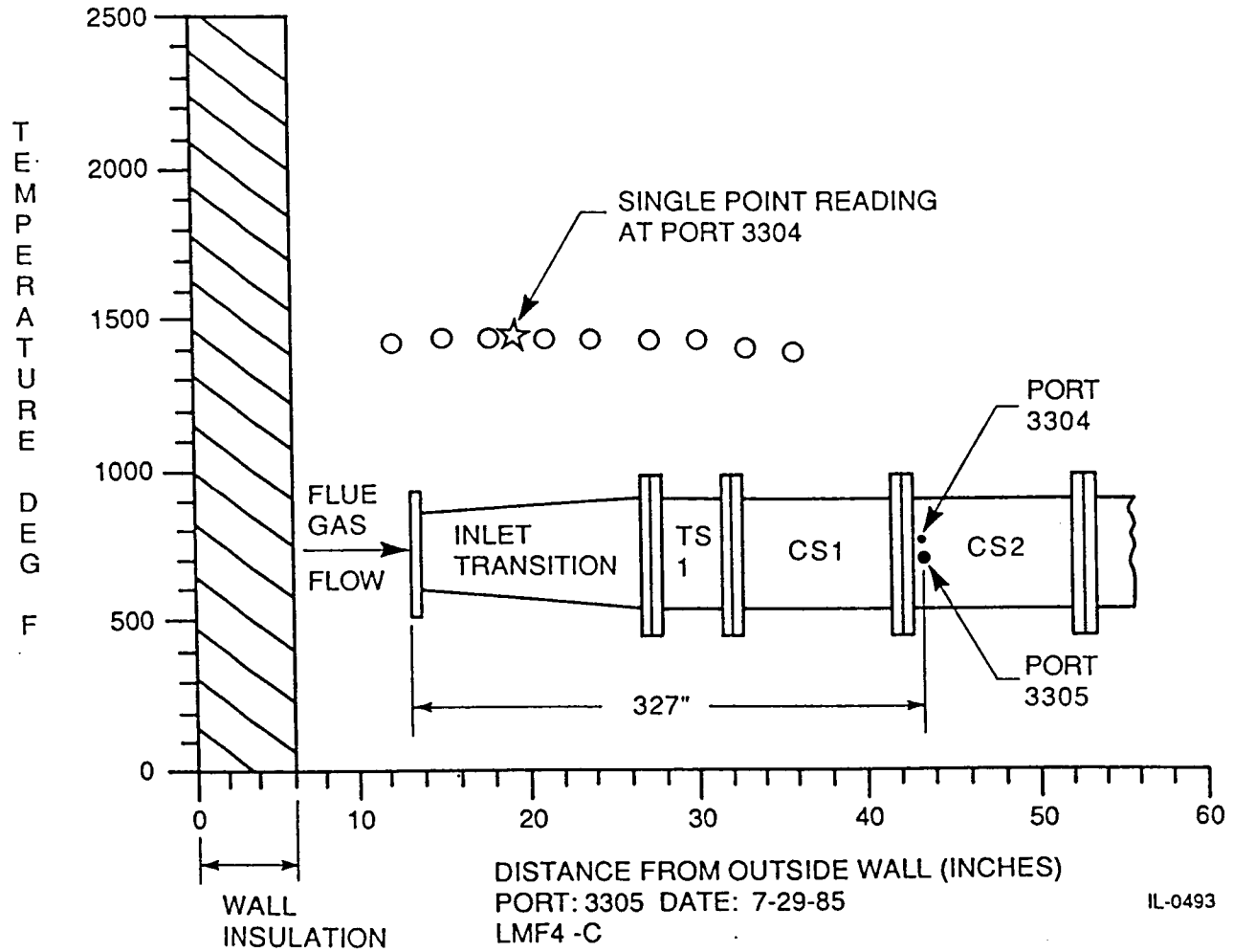



Figure 4.3. Horizontal Temperature Profile in SHTM Cooling Section 2

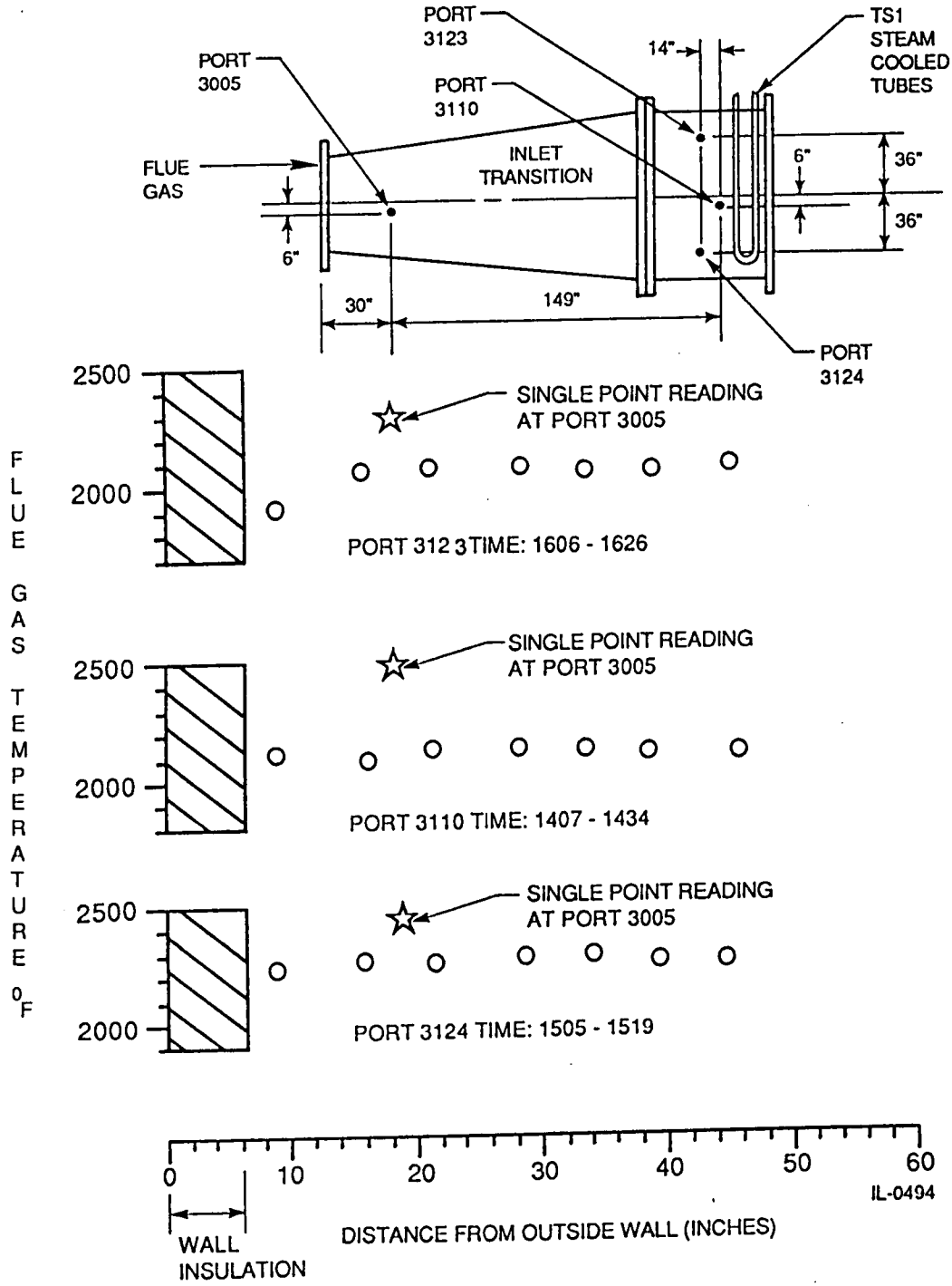



Figure 4.4. Horizontal Temperature Profile in SHTM Test Section 1

Even if horizontal and vertical temperature profiles were known at a given plane in the flue gas flow, a true average temperature could not be determined until the flow distribution was also known. A model study (Ref. 15) was performed on the SHTM geometry which indicates that the flow distribution downstream of the first tube bank is fairly uniform. The results of this flow study plus the uniformity of the horizontal temperature profiles would indicate that the single point temperature measurement and assumption of uniform flow is reasonably correct for comparison with the model for positions downstream of the first test section.

The largest discrepancy in assuming uniform conditions occurs in the inlet transition and Test Section 1. Measured temperature differences between the control thermocouple (at Position #1, Fig. 4.1,p.61) and a plane upstream of the first bank of steam cooled tubes are shown in Fig. 4.4. Note the $150^{o}$ to $200^{o}$F difference between these locations. Table 4.5 gives two cases that illustrate the heat transfer anomalies associated with the measured flue gas temperatures in this area. In Case 1 the water cooled screen tubes are between the first and second measurements. If it were assumed that the single point flue gas temperatures were representative of the total flow then the total heat loss would be calculated to be almost 6 million Btu/Hr. Similarly, the heat gain to the cooling water of the screen tubes calculated from measured data would be only about 3.5 million Btu/Hr. This magnitude of difference can not be explained by wall losses or inleakage of ambient air.

TABLE 4.5

HEAT TRANSFER ANOMALIES ASSOCIATED WITH
MEASURED TEMPERATURES IN THE
SHTM INLET TRANSISTION AND TEST SECTION 1

| | CASE 1[3] | CASE 2[4] |
|---|---|---|
| FLUE GAS FLOW (LB/HR) | 46315 | 35856 |
| TEMPERATURE #1 ($^{o}$F)[1] | 2240 | 2080 |
| TEMPERATURE #2 ($^{o}$F)[2] | 1860 | 1911 |
| FLUE GAS SPECIFIC HEAT (BTU/LB $^{o}$F) | 0.34 | 0.34 |
| FLUE GAS HEAT LOSS (BTU/HR) | $5.98 \times 10^6$ | $2.06 \times 10^6$ |
| COOLING WATER FLOW, SCREEN (LB/HR) | 246,641 | --NA-- |
| INLET TEMPERATURE $t_1$ ($^{o}$F) | 130 | --NA-- |
| OUTLET TEMPERATURE $t_2$ ($^{o}$F) | 147.2 | --NA-- |
| COOLING WATER HEAT GAIN (BUT/HR) | $3.51 \times 10^6$ | --NA-- |
| EST. HEAT LOSS TO WALLS (BTU/HR)[5] | $0.20 \times 10^6$ | $0.20 \times 10^6$ |
| UNACCOUNTED FOR HEAT LOSS (BTU/HR)[6] | $2.27 \times 10^6$ | $1.86 \times 10^6$ |

NOTES:
1. Location #1 per Figure 4.1, p.61
2. Location #2 per Figure 4.1 (approx. 10 feet downstream of #1)
3. With screen tubes in place
4. With screen tubes removed
5. For area between Temperature #1 and Temperature #2
6. Flue gas heat loss - (cooling fluid heat gain + wall loss)

An even more striking demonstration is illustrated in Case 2. Here there is a 169 $^{o}$F temperature drop of the flue gas between points 1 and 2, but in this case the screen tubes have been removed. In other words, there is an apparent 2 million Btu/Hr decrease of flue gas energy over a 10 foot flow distance in which there is no corresponding cooling surface to receive that heat.

It is apparent that the single point measurement of flue gas temperature in the SHTM inlet is not representative of the actual average temperature. It is not within the scope of this thesis to do a detailed flow/temperature distribution analysis of the SHTM inlet, but it is necessary to choose an appropriate inlet temperature for model verification. Observing from Ref. 15 that the flow becomes reasonably uniform after the first tube bank, the measured temperature at this location (near centerline of flow) will be taken as the reference. An energy balance based on the cooling fluid heat input of the preceding tube bank will be used to back calculate the effective SHTM inlet temperature. It can be noted in Table 4.4 (page xx) that if this correction to the inlet temperature were made, the overall heat balance between flue gas and cooling fluids would come to within 2%.

## 4.3 Model Verification

In review of the previous paragraphs, verification of the computer model against measured data will assume the following:

1. Cooling fluid flows and temperatures (inlet and outlet) are accurate enough to use as measured.

2. The flue gas inlet flow rate will be taken as the sum of the measured primary flows, i.e. coal, fuel oil, oxygen, primary combustion air, nitrogen and secondary combustor air.

3. The flue gas inlet properties (specific heat, viscosity, and thermal conductivity) will be calculated as the combustion products of the measured primary flows and will be only temperature dependent after entering the SHTM.

4. An allowance for inleakage of ambient air will be made so as to match the total flue gas flow measured (by Pitot tube) downstream of the SHTM.

5. Intermediate flue gas temperatures measured at about the centerline of the flow will be assumed to be representative of the average temperature in that plane.

6. The flue gas inlet temperature will be calculated from a heat balance across the first tube bank.

7. Cleanliness factors will be applied in accordance with in-house studies of the SHTM performance.

The first case of model verification (Fig. 4.5) will compare the current analysis to the original sizing of the SHTM by the Babcock & Wilcox (B&W) Company in 1983 (Ref. 16). For this comparison the calculation will assume cleanliness factors of 1.0 (i.e. no fouling) and no inleakage of air in accordance with the B&W calculation. The B&W calculation only went through the third test section since this

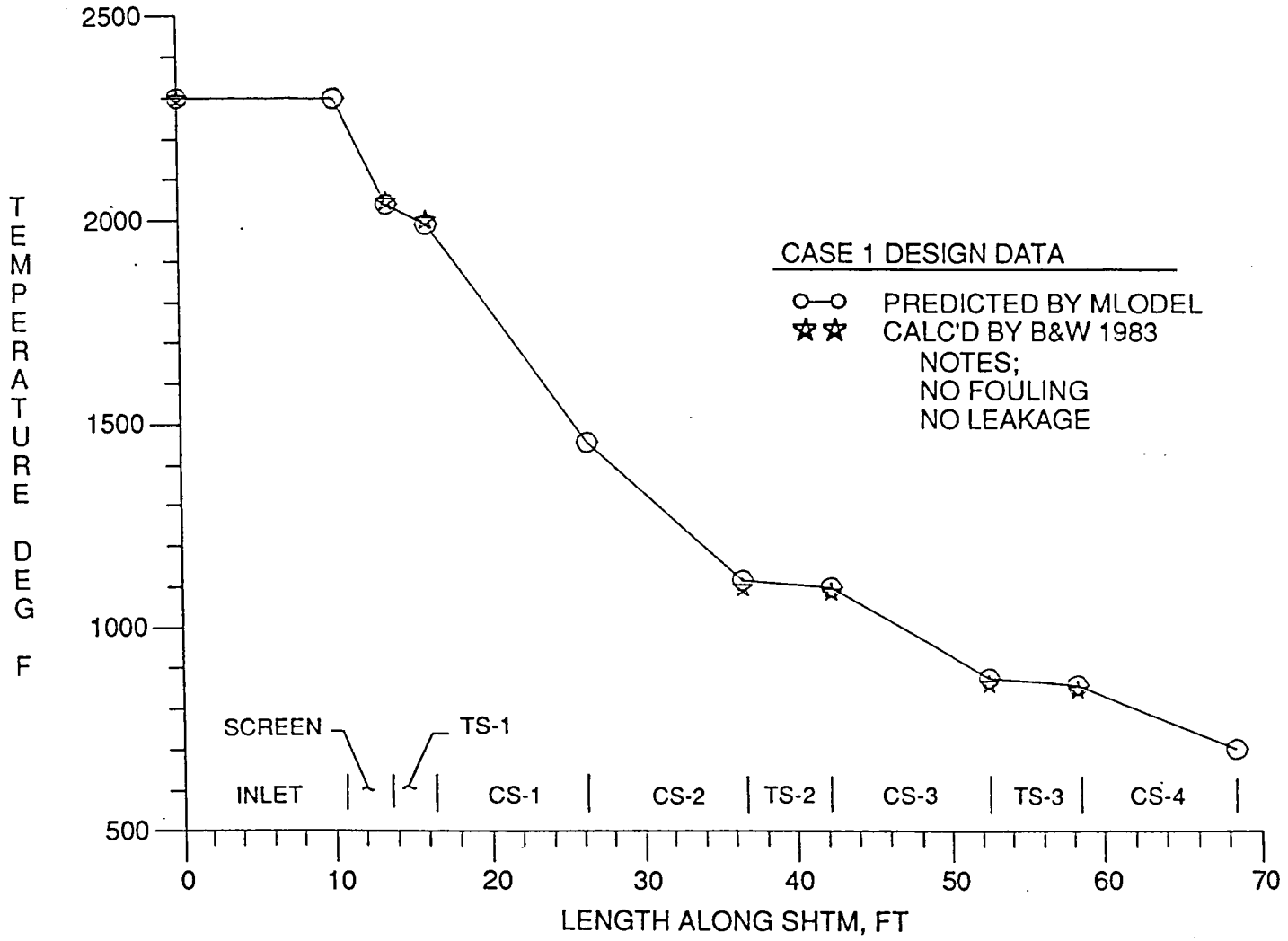



Figure 4.5. Flue Gas Temperature vs Position Along SHTM Case 1

was the area of greatest interest. This model and the B&W calculations agree within a few degrees at all points. The analytical nature of the B&W computer model is not known since it is proprietary. It is assumed that the heat transfer coefficient equations would be similar to those used in this analysis since they came from a B&W publication (Ref. 6).

Case 2 (Fig. 4.6) represents the SHTM as it was originally installed and tested in 1984. Shortly after this test, various mechanical modifications were made which changed the original configuration. Three calculations are illustrated on the graph. As a reference, the calculation was run assuming no fouling, no leakage and flue gas inlet temperature equal to the single point measurement in the inlet transition. The subsequent calculations assumed appropriate values for fouling and inleakage of ambient air. One calculation was made with the measured flue gas inlet temperature and the other with a reduced inlet temperature based on a heat balance across the first tube bank.

In operation, the original SHTM configuration was not able to achieve sufficiently low flue gas temperatures at the Air Heater entrance or at the outlet duct. This was primarily due to higher than expected fouling of the heat transfer tubes throughout the SHTM. In order to achieve better performance from the cooling sections, the number of sootblowers was increased and additional heat transfer tubing was added to the cooling sections on either side of the Air Heater. Figure 4.7 compares the measured and predicted flue gas

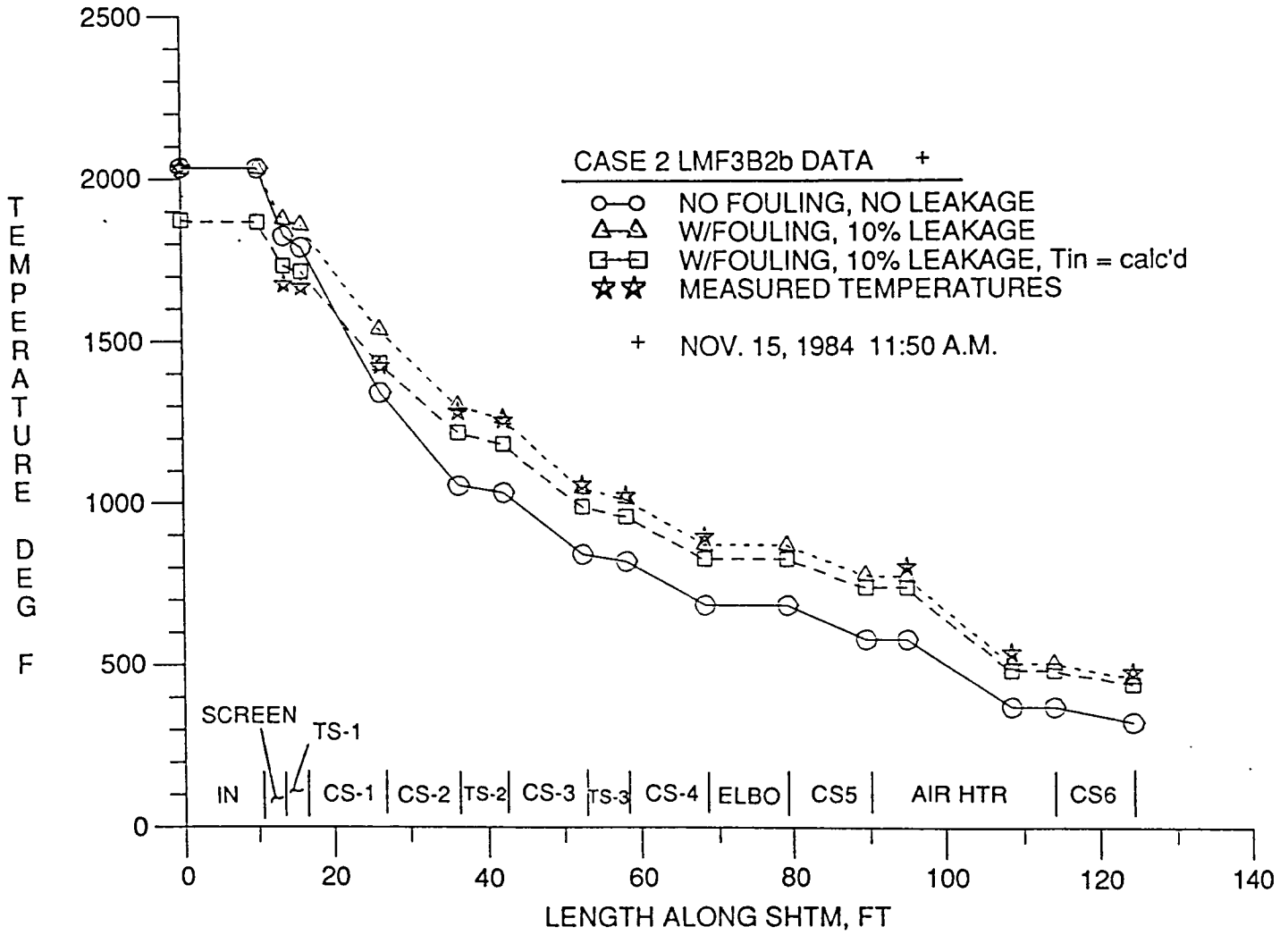



Figure 4.6. Flue Gas Temperature vs Position Along SHTM Case 2

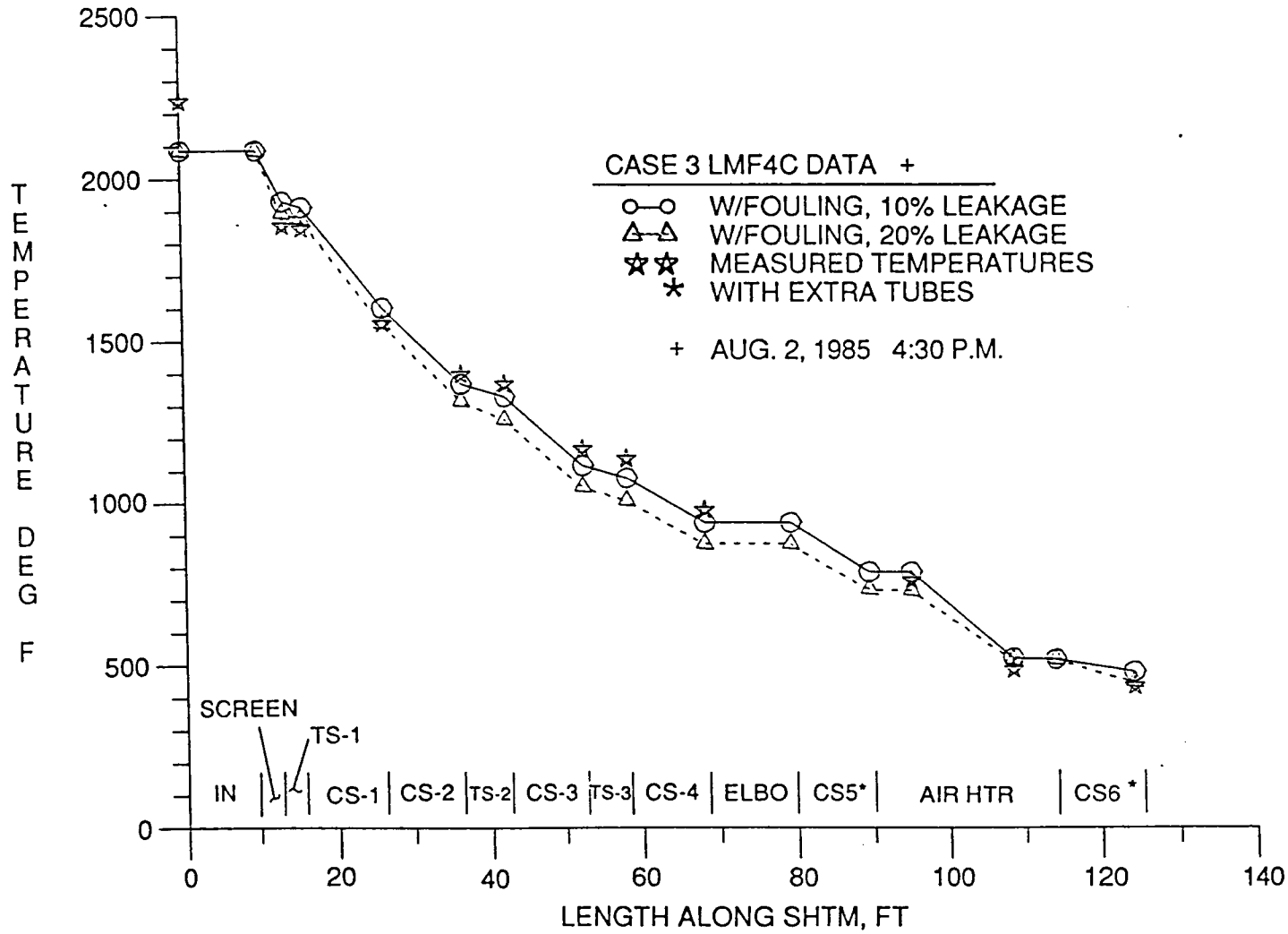



Figure 4.7. Flue Gas Temperature vs Position Along SHTM Case 3

temperatures for this configuration (Case 3). Fouling factors and leakage are included in the calculation. The flue gas inlet temperature is derived from a heat balance across the first tube bank and is 150 $^{o}$F lower than the measured temperature in the inlet transition.

The addition of heat transfer surface to two cooling sections as described above helped to reduce flue gas temperatures in the downstream leg of the SHTM, but other problems with the test section temperatures developed in later tests. In order to meet test objectives it became necessary to increase the flue gas temperature at the steam cooled tubes of Test Section 1. This was accomplished by removing the water cooled screen tubes from the first test section. Although this provided the necessary flue gas temperature, the steam cooled tubes tended to overheat because there was not sufficient steam capacity to cool all three steam cooled tube banks. Subsequently the tube banks in Test Sections 2 and 3 were removed and all available steam diverted to Test Section 1. The measured and calculated flue gas temperatures for Case 4 are shown in Fig. 4.8. As before, fouling factors and leakage are included in the calculation. The flue gas inlet temperature is derived from a heat balance across the first tube bank.

In 1987, a higher capacity steam boiler was provided for the steam cooling of the SHTM tubes which allowed reinstallation of the tubes in Test Sections 2 and 3. The screen tubes remained out and the extra tubing added to Cooling Sections 5 and 6 remained in place.

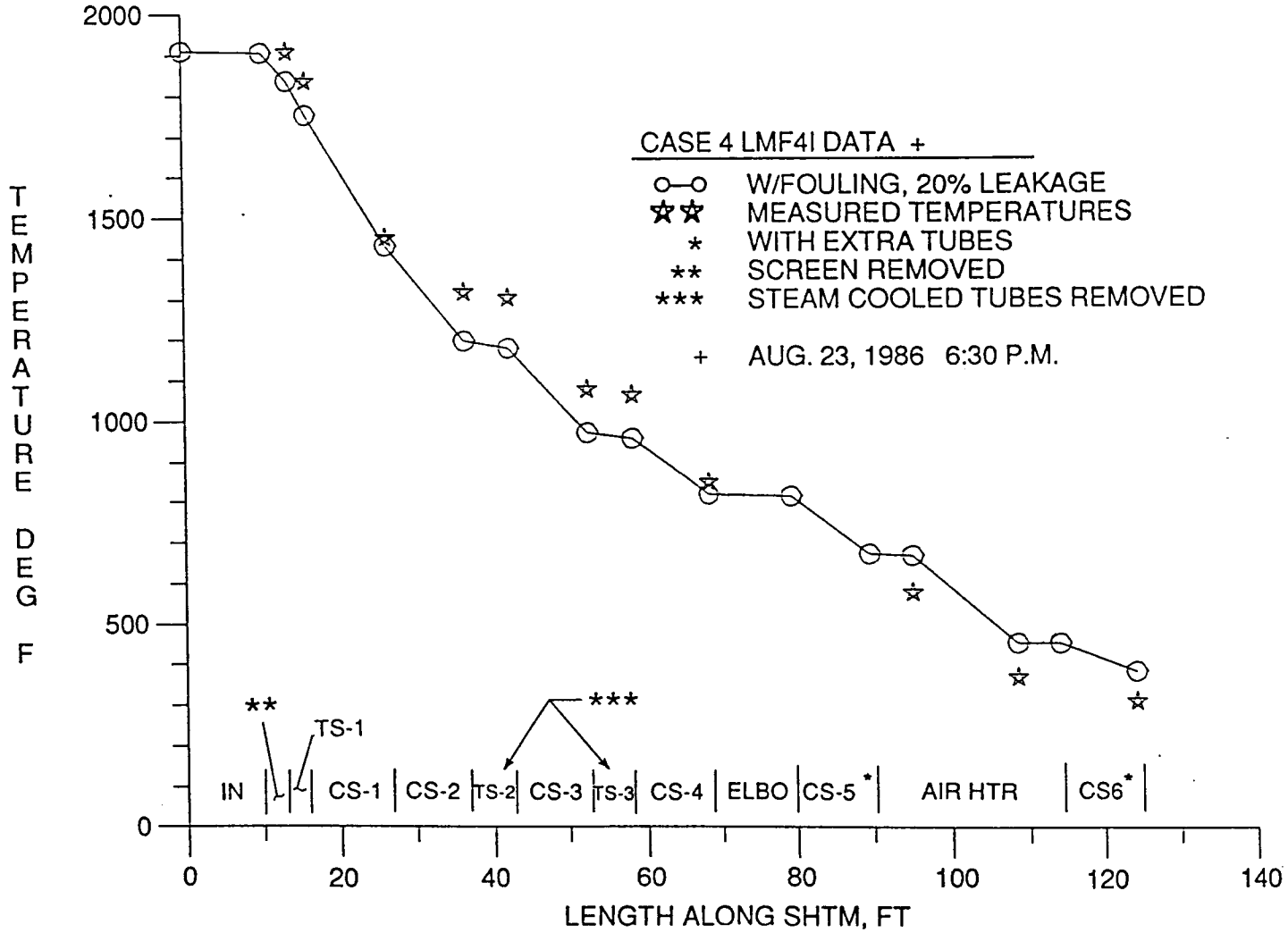



Figure 4.8. Flue Gas Temperature vs Position Along SHTM Case 4

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This is the configuration in place at the time of the writing of this thesis. Cases 5 and 6 (Figs. 4.9 and 4.10) are both based on this configuration. Case 5 has a lower flue gas inlet temperature and the coal burned during this test was from the same coal shipment (Illinois #6) as used in the previous three cases. Case 6 is included because of the higher flue gas inlet temperature and because the test was run with a new shipment of coal. The new coal is a "high sulfur Eastern" coal similar to Illinois #6 but with some slight differences in chemical analysis. This case was selected because it is this new coal which will be used in future tests for which performance predictions are presented in the following paragraphs.

For both Cases 5 and 6 the standard fouling factors were used and the flue gas inlet temperature was derived from a heat balance across the first tube bank. The measured data for both Cases 5 and 6 indicated that the total flue gas flow at the exit of the SHTM was 2.0 to 2.5 lb/sec (about 20%) greater than the sum of the measured flows into the primary combustor plus known instrument purges. As with previous cases, it was assumed that this increase in flow was due to inleakage of ambient air along the SHTM flow train. As a check on the sensitivity of the model to leakage rates, Case 5 was also run with an assumed leakage rate of only 10% (about 1.0 lb/sec).

Case 6 is noted for its relatively high flue gas inlet temperature, use of the "new coal" and the fact that the measured data was taken after about 50 hours of continuous operation (Ref. 17). These characteristics make Case 6 more representative than any of the

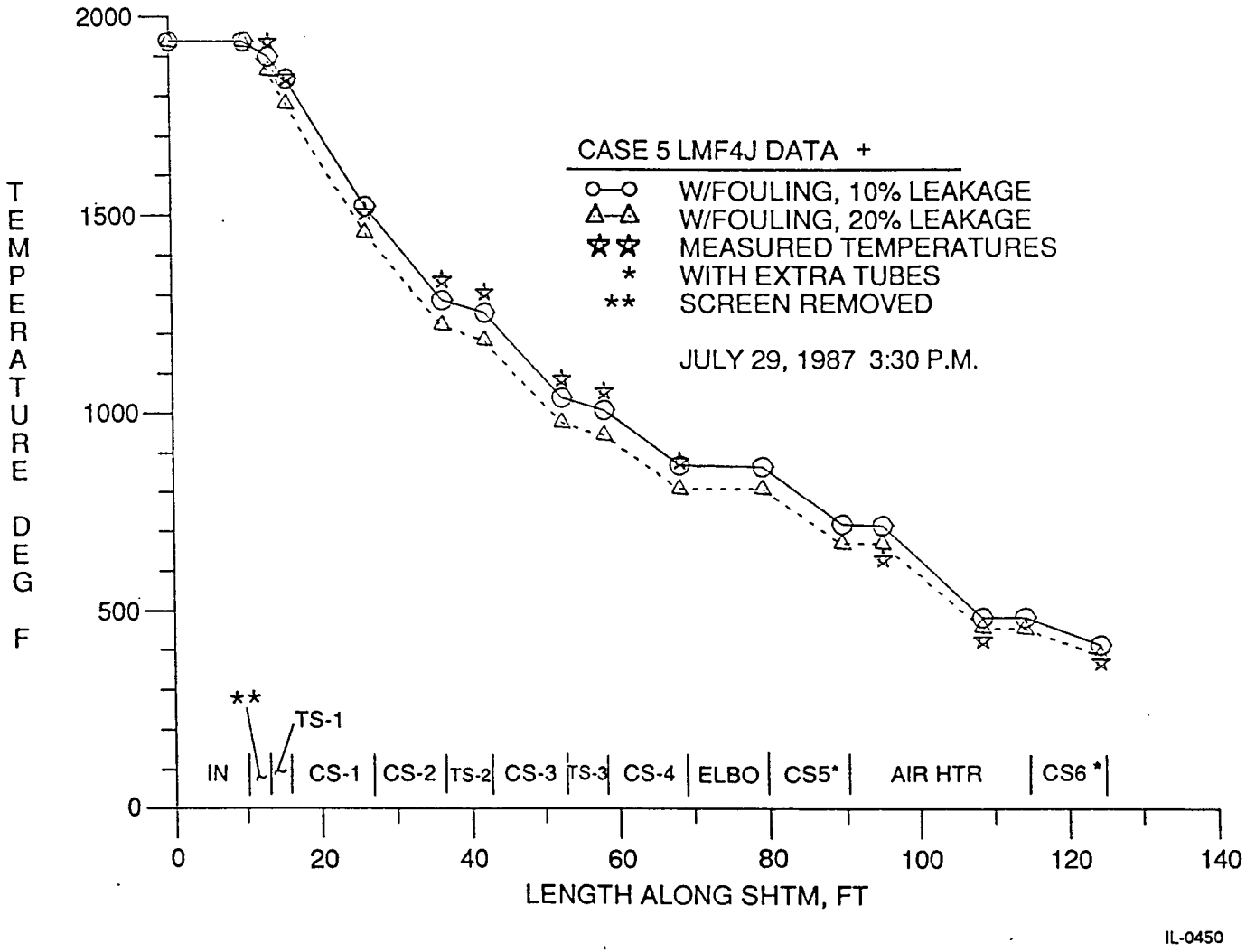



Figure 4.9. Flue Gas Temperature vs Position Along SHTM Case 5

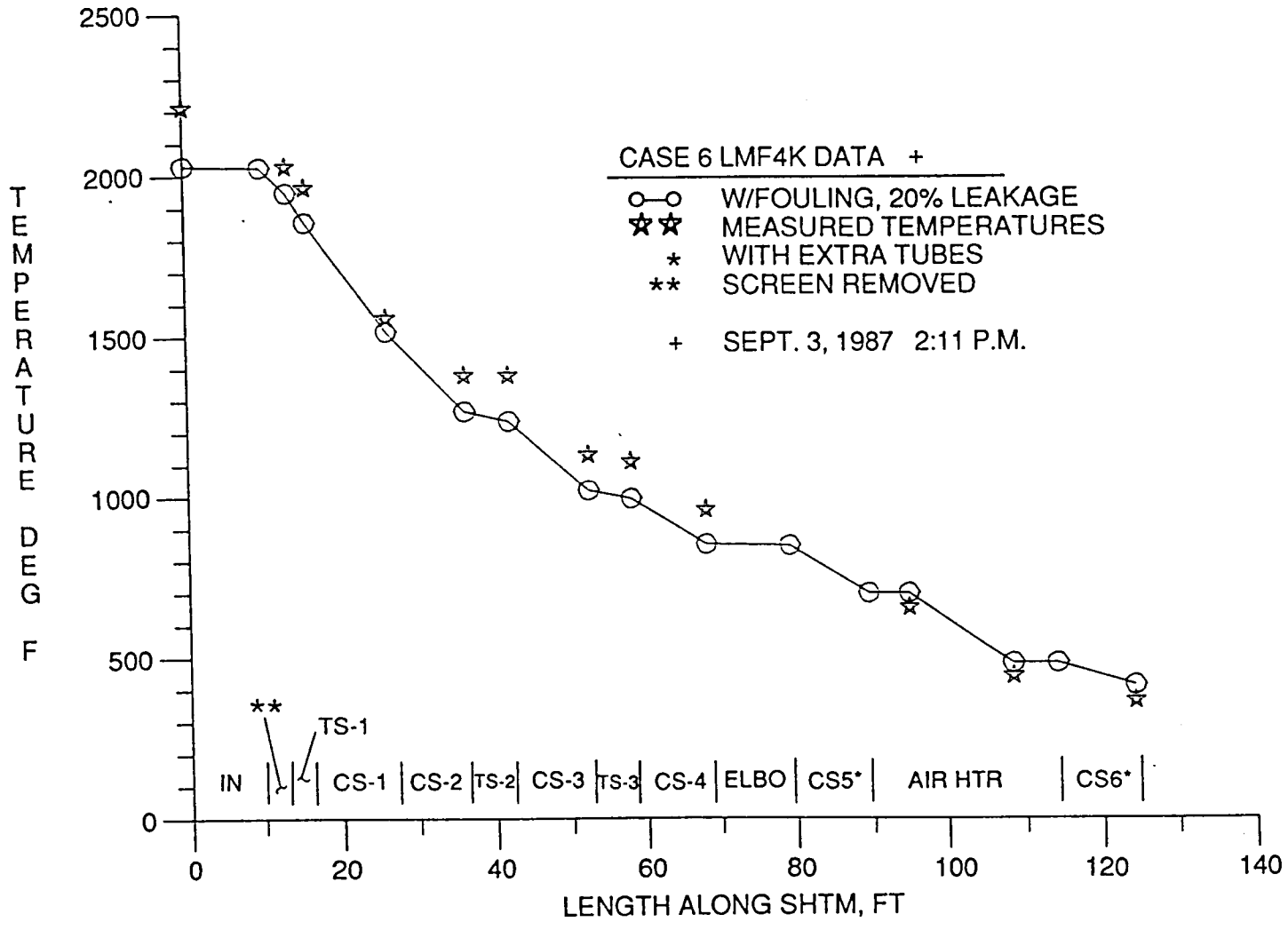



Figure 4.10. Flue Gas Temperature vs Position Along SHTM Case 6

others of expected future SHTM operation. Following paragraphs will present the results of flue gas predictions for possible future SHTM configurations. The cleanliness factors, individual section leakage rates, coal composition, and cooling fluid flow rates and inlet temperatures of Case 6 will be the basis for the following predictions.

## 4.4 Model Predictions of Proposed Arrangements

Two recurring problems during recent testing (since reinstallation of the Test Section 2 and 3 steam cooled tubes) have been too low flue gas temperatures into Test Sections 2 and 3 and too high flue gas temperatures into the Air Heater. A proposal for resolving these problems using the existing SHTM hardware with minimum modifications is illustrated in Fig. 4.11. In order to get higher temperatures into Test Sections 2 and 3, it was proposed to move these sections upstream by one cooling section. In order to lower the flue gas temperature into the Air Heater it was proposed to add water cooled heat transfer surface to either one (Case 7) or both (Case 8) of the cooling sections immediately upstream of the elbow. The resulting flue gas temperatures are given in Fig. 4.11 with Case 6 inputs as the base condition.

Although these modifications shifted the flue gas temperatures in the right direction, they tended to create other problems. Of particular concern was the creation of more closely spaced cooling

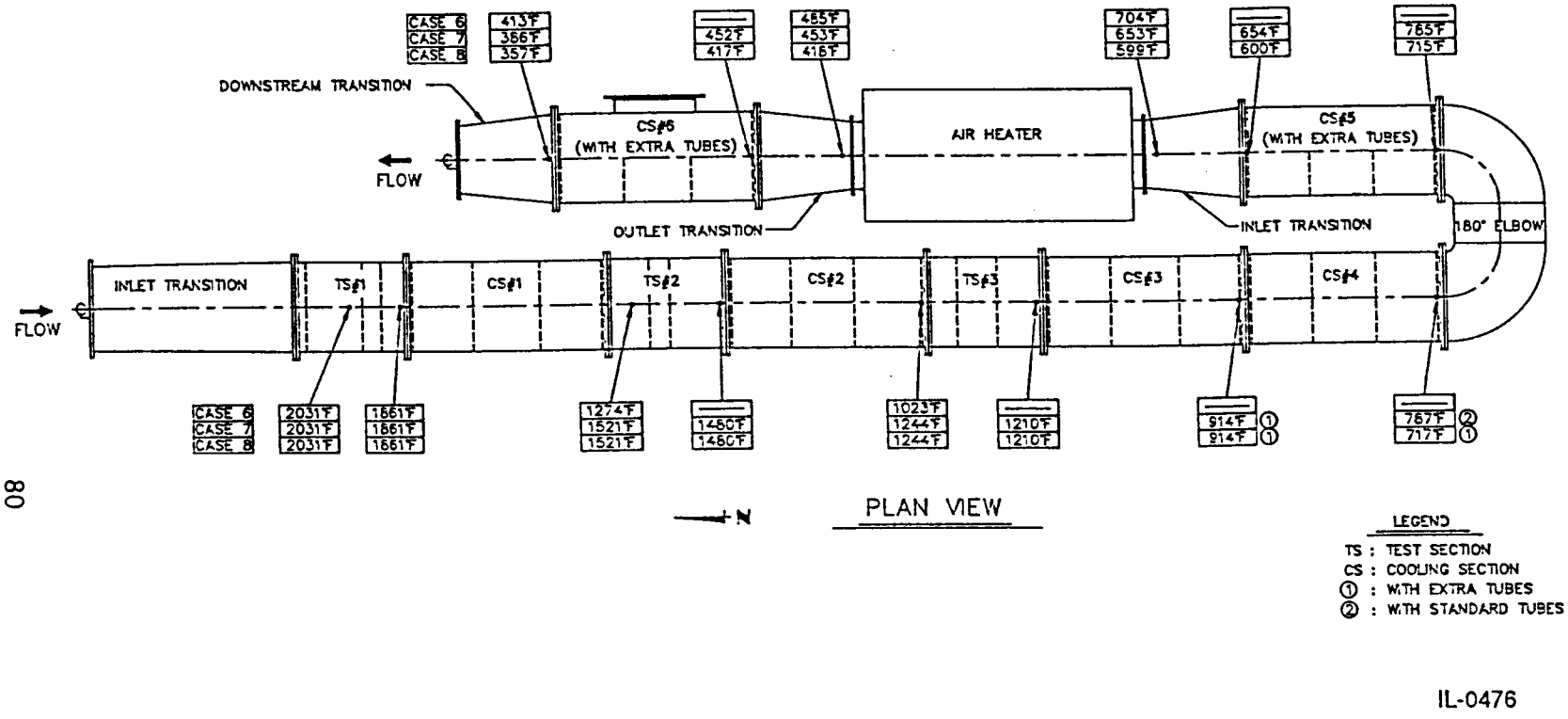



Figure 4.11. Predicted Flue Gas Temperatures for Proposed SHTM Arrangement Cases 7 and 8

tube bundles. Bridging and lane blockage by flue gas deposits is more likely to occur with closely spaced tubes. The next proposal (Case 9) was to review the effects of adding new cooling sections rather than adding area to existing sections (Fig. 4.12). The "horse shoe" arrangement of the SHTM makes it necessary to add components in equal lengths to each side of the flow train. The existing cooling sections with extra tubes have been retained and two cooling sections added. Predicted flue gas temperatures at the test sections, Air Heater and outlet are given on Fig. 4.12. Again, the inputs of Case 6 have been used as the base.

Further concern for the possibility of bridging and lane blockage in the upstream cooling sections and an envisioned need for more efficient sootblowing in all upstream sections required analysis of additional configurations. It was proposed to increase the tube-to-tube spacing (in the direction perpendicular to flue gas flow) in the first cooling section from 6.06 inches to 9.5 inches (i.e five tubes across rather than eight). It was also proposed to remove one of the three tube banks in each of the first four cooling sections so as to provide sufficient space for efficient operation of high impact sootblowers.

The net result of these requirements would be to remove so much surface from the first four cooling sections that the flue gas temperatures to Test Sections 2 and 3 would be too high if located per the previous case. Fig. 4.13 illustrates an arrangement which attempts to resolve all previous requirements. Test Section 2 has

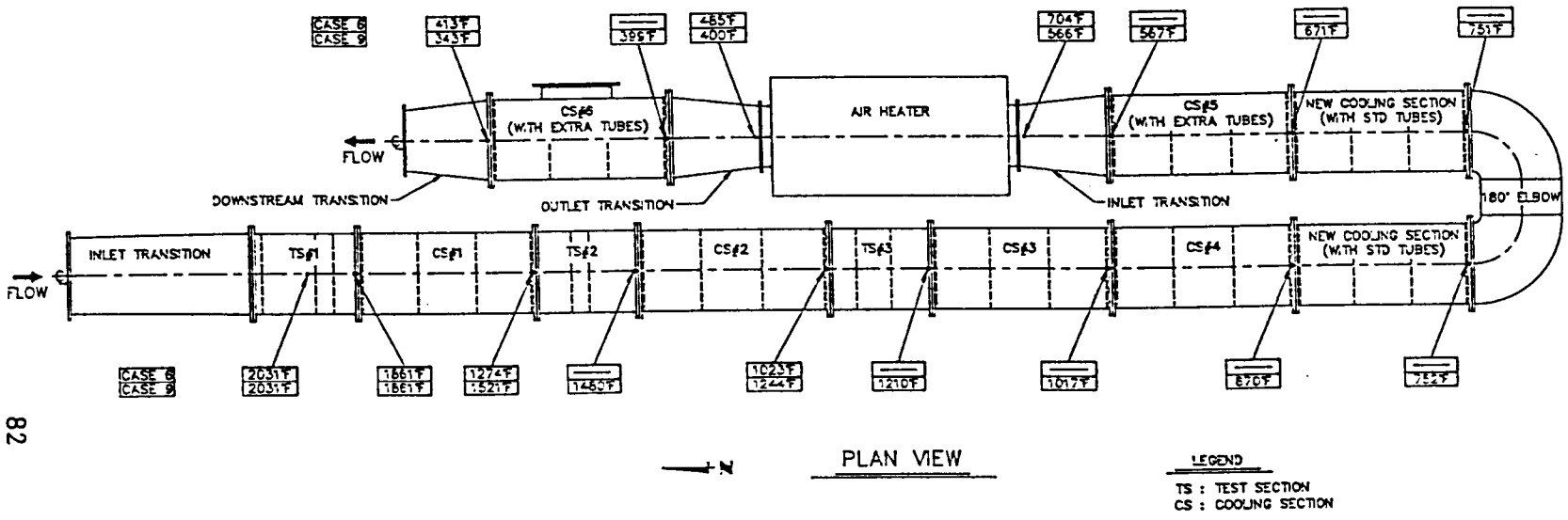
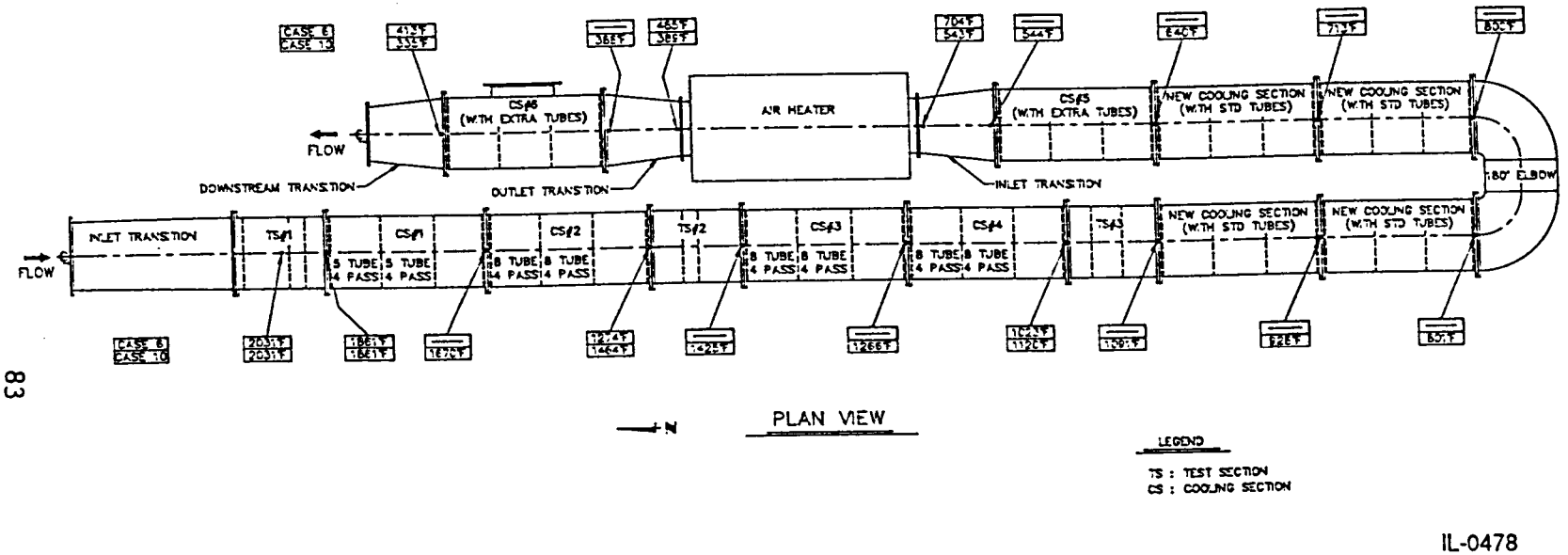



Figure 4.12. Predicted Flue Gas Temperatures for Proposed SHTM Arrangement Case 9




IL-0478

Figure 4.13. Predicted Flue Gas Temperatures for Proposed SHTM Arrangement Case 10

been returned to its original position downstream of Cooling Sections 1 and 2. Cooling Section 1, which has demonstrated the greatest tendency for tube lane plugging, has two banks of five tubes across and Cooling Section 2 has two banks of eight tubes across. Test Section 3 has been moved downstream one additional cooling section, and one tube bundle has been removed from each of Cooling Sections 3 and 4. Four new cooling sections with standard tube banks have been added downstream of Test Section 3 and the extra surface has been retained in the cooling sections on either side of the Air Heater. This arrangement (Case 10) provides the required flue gas temperatures at Test Sections 2 and 3, provides additional tube-to-tube spacing in the first cooling section, provides space for high impact sootblowers in the first four cooling sections, and provides sufficient overall flue gas cooling for the Air Heater and outlet duct.

# SECTION 5.0

# CONCLUSIONS AND RECOMMENDATIONS

## 5.1 Conclusions

The results presented in Cases 2 through 6 (Figs. 4.6, p.72, through 4.10, p.78) of Section 4.0 indicate that the computer model developed for this thesis can accurately predict the heat transfer characteristics of the SHTM with the proper assumptions for leakage, cleanliness factor and flue gas inlet temperature. Obviously, though, these are very significant factors which can change the temperature distribution calculation by hundreds of degrees if applied arbitrarily. It is significant that the agreement with measured values was achieved by using consistent assumptions. In each case the cleanliness factors were taken as the values listed in Table 4.3, p.57. Inleakage of ambient air was found in later cases to be about 20% of the measured flue gas flows and was equally distributed along the SHTM flow train. The appropriate flue gas inlet temperature for each case was found to be about $150^{o}$ to $200^{o}$F below the single point temperature measured in the inlet transition. These assumptions were used for the predictive calculations shown in Figs. 4.11, p.80 through 4.13, p.83.

It is not the purpose of this thesis to recommend a mechanical configuration for the SHTM, but rather to provide an

analytical model for the analysis of options. The configuration indicated by Fig. 4.13, p.83, is one option which the computer code predicts will achieve a set of stated objectives under a given set of operating conditions. The assumptions on cleanliness factors, leakage, and inlet temperature are consistent with the assumptions used to compare the computer analysis with measured data. As long as these assumptions remain valid the predictions should also be valid.

Several mechanical and/or operational factors could be changed which would significantly affect the validity of the critical assumptions indicated above. A change in coal/seed composition, operating temperatures, or a significant improvement in sootblower effectiveness could change the cleanliness factors through out the SHTM. A concerted effort to seal leaks, reduce (or increase) sootblower flow rates, or otherwise change the leakage rate or distribution could cause significant differences in the temperature distribution. Modification of the inlet transition to improve flow and temperature distribution and/or relocation of the inlet thermocouple could require reevaluation of the assumed inlet flue gas temperature. However, all of these changes can be easily incorporated in the analytical heat transfer model.

## 5.2 Recommendations

In the course of developing this analysis it was observed (as discussed in Section 4.2, and indicated in Table 4.4, page 60) that

calculated flue gas heat losses along the SHTM (based on measured temperatures and assumed flows with leakage) were not equal to corresponding calculated cooling fluid heat gains. This situation was noted for all test data reviewed. It was assumed that the cooling fluid flows and temperatures were more likely to be correct than the flue gas flows and temperatures, but this could be positively demonstrated only at the inlet transistion. It follows then that the primary means of verifying this computer analysis, i.e. comparison of calculated and measured flue gas temperatures for given operating data, is based on measured values of questionable accuracy. Any further adjustment of the computer model for better prediction of SHTM performance should be delayed until there is a better correlation of the flue gas and cooling fluid heat balances. Immediate attention should be given to the flow and temperature anomalies in the inlet transition, particularly since the inlet temperature measurement is used as an input in controlling the combustion process during testing.

# LIST OF REFERENCES

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# APPENDIX

# FORTRAN COMPUTER PROGRAM

```
C              ******************************************
C              ******************************************
C              ****************SHTMPZ********************
C              *****SUPERHEATER TEST MODULE PREDICTION****
C              *******WITH AMBIENT AIR INLEAKAGE*********
C              A PROGRAM TO CALCULATE HEAT TRANSFER COEFFICIENTS
C              AND TEMPERATURE DISTRIBUTIONS IN THE SHTM
C              WITH VARIABLE ARRANGEMENT OF SHTM SECTIONS
C              ******************************************
C              ******************************************
C
C ALL CALCULATIONS AND INPUTS ARE IN ENGLISH UNITS
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7)
      DIMENSION Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R
      CHARACTER*4 Q
      CHARACTER*5 R
      OPEN (UNIT=2,NAME='SHTMPZ.OUT',TYPE='NEW')
      OPEN (UNIT=1,NAME='SHTMPZ.INP',TYPE='OLD')
C READ A CASE NUMBER TO BE USED FOR REFERENCE ONLY
      READ(1,*)CASE
C READ TOTAL FLUE GAS FLOW IN POUNDS PER HOUR AND FLUE GAS
C INLET TEMPERATURE IN DEG F
      READ(1,*)FGPPHI,FGTF
C READ THE CURVE FIT COEFFICIENTS FOR SPECIFIC HEAT OF FLUE GAS IN
C BTU/LB DEG F (A1,A2,A3), VISCOSITY IN LBM/FT HR (B1,B2,
C B3), AND THERMAL CONDUCTIVITY IN BTU/HR FT DEG F (C1,C2,C3)
      READ(1,*)A1,A2,A3
      READ(1,*)B1,B2,B3
      READ(1,*)C1,C2,C3
C READ COOLING WATER INLET TEMP IN DEG F, STEAM INLET TEMP IN
C DEG F, AIR INLET TEMP IN DEG F, AND NUMBER OF COMPONENTS IN
C FLOW TRAIN
      READ(1,*)CWTF,STTF,AIRTF,NC
      FGINIT=FGTF
C FGPPHI IS THE INITIAL FLUE GAS FLOW FROM THE FURNACE INTO THE SHTM,
C FGPPH IS THE THE FLUE GAS FLOW PER SECTION WHICH CAN INCREASE
C INCREMANTALLY DUE TO INLEAKAGE OF AMBIENT AIR.
      FGPPH=FGPPHI
C CALCULATE INITIAL FLUE GAS PROPERTIES
      SPHT=A1+A2*FGTF+A3*FGTF**2
      VISC=B1+B2*FGTF+B3*FGTF**2
      THCOND=C1+C2*FGTF+C3*FGTF**2
C INITIALIZE AIR INLEAKAGE
      PCTL=0.0

C THIS DO LOOP READS THE INPUT PARAMETERS FOR THE COMPONENTS
C IN THE ORDER THEY ARE LISTED IN THE INPUT FILE
C THE IF STATEMENTS THEN CALL THE SUBROUTINES IN THE PROPER
C ORDER
      DO 400 II=1,NC
C Q(II) IS THE NAME OF THE COMPONENT, R(II) IS THE TYPE OF
C COOLING FLUID, PPHPT IS THE FLOW OF THAT FLUID IN POUNDS
C PER HOUR PER TUBE, CF IS THE CLEANLINESS FACTOR, PCTLKG IS THE %
C OF THE INITIAL TOTAL GAS FLOW AS INLEAKAGE OF AMBIENT AIR PER SECTION
      READ(1,*)Q(II),R(II),PPHPT,CF,PCTLKG

C SCR4 IS THE 4 PASS WATER COOLED SCREEN (8 TUBES ACROSS)
      IF(Q(II).EQ.'SCR4')CALL SCR4(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C CS12 IS A TYPICAL COOLING SECTION (WATER COOLED) WITH THREE
C BANKS OF 4 PASS EACH TUBES, THEREFORE "12" PASSES (8 TUBES ACROSS)
```

```
      IF(Q(II).EQ.'CS12')CALL CS12(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C MCS5 IS A COOLING SECTION MODIFIED FOR ONLY 5 TUBES ACROSS, BUT STILL
C WITH 3 BANKS OF 4 PASSES EACH.
      IF(Q(II).EQ.'MCS5')CALL MCS5(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C CSXP IS TYPICAL OF COOLING SECTIONS 5 AND 6 WITH "EXTRA PASSES"
C OF COOLING TUBES ADDED TO THE ORIGINAL BUNDLE
      IF(Q(II).EQ.'CSXP')CALL CSXP(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C CM4P REPRESENTS ONE THIRD OF A COOLING SECTION, I.E. ONE FOUR
C PASS TUBE BUNDLE (8 TUBES ACROSS)
      IF(Q(II).EQ.'CM4P')CALL CM4P(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C MCM5 REPRESENTS ONE THIRD OF A MODIFIED COOLING SECTION, I.E. ONE
C FOUR PASS TUBE BUNDLE BUT ONLY 5 TUBES ACROSS.
      IF(Q(II).EQ.'MCM5')CALL MCM5(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C CMX0 REPRESENTS ONE THIRD OF A COOLING SECTION WITH THE TUBE
C BANK REMOVED, I.E. A CAVITY WITH VOL EQ 1/3 OF A CS
      IF(Q(II).EQ.'CMX0')CALL CMX0(FGTF,II,PCTLKG,FGPPH)
C TS12 IS THE 2 PASS STEAM COOLED PORTION OF TEST SECTION #1
C (8 TUBES ACROSS)
      IF(Q(II).EQ.'TS12')CALL TS12(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C TS2P IS A 2 PASS STEAM COOLED TEST SECTION LIKE TS-2
C (8 TUBES ACROSS)
      IF(Q(II).EQ.'TS2P')CALL TS2P(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C TS4P IS A 4 PASS STEAM COOLED TEST SECTION LIKE TS-3
C (8 TUBES ACROSS)
      IF(Q(II).EQ.'TS4P')CALL TS4P(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C ARHT IS THE ORIGINAL AIRHEATER WITH INLET AND OUTLET TRANSITIONS
      IF(Q(II).EQ.'ARHT')CALL ARHT(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C CSAH IS TWO COOLING SECTION BOXES IN SERIES WITH 2/3 OF THE TUBES
C FROM THE ORIGINAL AIR HEATER (1/3 OF THE ORIGINAL AIR HTR TUBE
C BUNDLE REMOVED TO PROVIDE ADDITIONAL SPACE BETWEEN TUBES)
      IF(Q(II).EQ.'CSAH')CALL CSAH(PPHPT,FGTF,CF,II,PCTLKG,FGPPH)
C INLT IS THE INLET TRANSITION TO THE SHTM
      IF(Q(II).EQ.'INLT')CALL INLT(FGTF,II,PCTLKG,FGPPH)
C ELBO IS THE 180 DEG ELBO (W/NO COOLING SUSRFACE)
      IF(Q(II).EQ.'ELBO')CALL ELBO(FGTF,II,PCTLKG,FGPPH)
C SCR0 IS THE SCREEN SECTION OF TS-1 WITH THE WATER COOLED SCREEN
C TUBES REMOVED
      IF(Q(II).EQ.'SCR0')CALL SCR0(FGTF,II,PCTLKG,FGPPH)
C TSX0 IS A TYPICAL TEST SECTION WITH ALL TUBES REMOVED (INCLUDING
C SCREEN IF TS-1)
      IF(Q(II).EQ.'TSX0')CALL TSX0(FGTF,II,PCTLKG,FGPPH)
C CSX0 IS A TYPICAL COOLING SECTION WITH ALL TUBES REMOVED
      IF(Q(II).EQ.'CSX0')CALL CSX0(FGTF,II,PCTLKG,FGPPH)
C THE FLUE GAS INLET TEMP "FGTF" FOR THE NEXT SECTION IS SET EQUAL
C TO THE PREDICTED FLUE GAS OUTLET TEMP FROM THE SECTION JUST
C CALCULATED
      FGTF=D(II,5)
C THE FLUE GAS FLOW FOR THE NEXT SCTION IS SET EQUAL TO THE FLUE GAS
C FLOW PLUS AIR LEAKAGE FROM THE PREVIOUS SECTION.
      FGPPH=F(II,5)
C CALCULATE CUMULATIVE AIR INLEAKAGE PERCENTAGE
      PCTL=PCTL+(PCTLKG*100.)
400   CONTINUE


C THE REST OF THIS MAIN PROGRAM IS OUTPUT FORMAT
      WRITE(2,10)
      WRITE(2,15) CASE
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,20) FGPPHI
      WRITE(2,21) PCTL
      WRITE(2,22) FGPPH
```

```
      WRITE(2,30) FGINIT
      WRITE(2,40) SPHT
      WRITE(2,50) VISC
      WRITE(2,60) THCOND
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,70)
      WRITE(2,*)
      WRITE(2,80)
      WRITE(2,90)
      WRITE(2,*)

      DO 300 N=1,NC
      WRITE(2,100) Q(N),R(N),(D(N,J),J=1,5)
300   CONTINUE

      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,110)
      WRITE(2,*)
      WRITE(2,120)
      WRITE(2,130)
      WRITE(2,*)

      DO 310 N=1,NC
      WRITE(2,140) Q(N),(E(N,J),J=1,7)
310   CONTINUE

      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,150)
      WRITE(2,*)
      WRITE(2,160)
      WRITE(2,170)
      WRITE(2,*)

      DO 320 N=1,NC
      WRITE(2,180) Q(N),(F(N,J),J=1,5)
320   CONTINUE

      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,190)
      WRITE(2,*)
      WRITE(2,200)
      WRITE(2,210)
      WRITE(2,*)

      DO 330 N=1,NC
      WRITE(2,220) Q(N),(G(N,J),J=1,6)
330   CONTINUE

      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,*)
      WRITE(2,230)
      WRITE(2,*)
      WRITE(2,240)
      WRITE(2,250)
      WRITE(2,260)
```

```
      WRITE(2,*)

      DO 340 N=1,NC
      WRITE(2,270) Q(N),(H(N,J),J=1,5)
340   CONTINUE


10    FORMAT(26X,'LMF SUPERHEATER PERFORMANCE PREDICTION')
15    FORMAT(26X,'CASE NUMBER ',F5.1)
20    FORMAT(20X,'INITIAL FLUE GAS FLOW=',F6.0,' PPH')
21    FORMAT(20X,'INLEAKAGE AMBIENT AIR=',F5.2,'%')
22    FORMAT(20X,'TOTAL GAS FLOW SHTM OUTLET=',F8.1,' PPH')
30    FORMAT(20X,'FLUE GAS INITIAL TEMP.=',F6.1,' F')
40    FORMAT(20X,'FLUE GAS PROPERTIES:  INLET SP.HT.=',
     &  F5.4,' BTU/LB F')
50    FORMAT(41X,'INLET VISCOSITY=',F5.4,' LB/HR FT')
60    FORMAT(41X,'INLET TH. COND.=',F5.4,' BTU/HR FT F')
70    FORMAT(34X,'COOLING FLUID',12X,'FLUE GAS')
80    FORMAT(28X,'PPH PER    INLET    OUTLET   INLET
     &OUTLET')
90    FORMAT(15X,'NAME  TYPE    TUBE',9X,'F',7X,'F',9X,
     &  'F',9X,'F')
100   FORMAT(15X,A4,2X,A5,5F9.1)
110   FORMAT(24X,'HEAT TRANSFER COEFFICIENTS  BTU/HR
     &FT**2 F')
120   FORMAT(21X,'CONV ID  CONV OD  TUBE CAVITY
     & CLEAN   CLEAN   DIRTY')
130   FORMAT(15X,'NAME   TUBE    TUBE',5X,'RAD',3X,
     &  'RAD   COEFF   FACTOR  COEFF')
140   FORMAT(15X,A4,F10.2,6F7.2)
150   FORMAT(30X,'HEAT TRANSFER  BTU/HR',3X,'FLUE GAS FLOW PPH')
160   FORMAT(21X,'TO TUBES'7X,'TO',9X,'TOTAL',2X,'LEAKAGE    TOTAL')
170   FORMAT(15X,'NAME   (DIRTY)',7X,
     &  'WALLS',4X,'(DIRTY)',4X,'AIR',7X,'GAS')
180   FORMAT(15X,A4,3F11.1,2F10.1)
190   FORMAT(24X,'HEAT TRANSFER DIMENSIONS')
200   FORMAT(22X,'#',6X,'#',5X,'TUBE    TUBE   TUBE AREA
     &  WALL AREA')
210   FORMAT(15X,'NAME  ROWS  PASSES  OD IN  ID IN
     &  FT**2',6X,'FT**2')
220   FORMAT(15X,A4,F6.0,F8.0,2F7.2,2F11.2)
230   FORMAT(25X,'HEAT EXCHANGER ANALYSIS FACTORS')
240   FORMAT(21X,'FLUE GAS  COOLING',16X,'NTU')
250   FORMAT(21X,'"C"=W*CP  "C"=W*CP  "C"STAR')
260   FORMAT(15X,'NAME  BTU/HR F  BTU/HR F  CMIN/CMAX
     &  U*A/CMIN  EPSILON')
270   FORMAT(15X,A4,2F10.1,2F11.4,F9.3)
      STOP
      END

C SUBROUTINE ALEAK CALCULATES THE TEMPERATURE OF THE FLUE GAS AFTER MIXING
C WITH THE LEAKAGE AMBIENT AIR
      SUBROUTINE ALEAK(TFGO,AIRLKG,FGPPH,T)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R
C CALCULATE THE SPECIFIC HEAT OF THE FLUE GAS * MASS FLOW
      CGPT=(A1+A2*TFGO+A3*TFGO**2)*(FGPPH-AIRLKG)
C THE SPECIFIC HEAT OF AMBIENT AIR IS .24053 BTU/LB F
      CGAIRT=AIRLKG*.24053
C THE SPECIFIC HEAT OF THE MIXTURE "X" IS THE SUM OF THE SPECIFIC HEATS
```

```
C TIMES THEIR MASS FLOWS DIVIDED BY THE TOTAL MASS FLOW
      X=(CGPT+CGAIRT)/FGPPH
C SOLVE THE QUADRATIC EQUATION DERIVED FROM THE SPECIFIC HEAT CURVE FIT
C TO GET THE MIXTURE TEMPERATURE
      Z=SQRT(A2**2-4.*A3*(A1-X))
      T=((-1.*A2)+Z)/(2.*A3)
      RETURN
      END

C SUBROUTINE FPPG CALCULATES FLUE GAS PROPERTIES SPECIFIC HEAT (CGP),
C VISCOSITY (AMU), AND THERMAL CONDUCTIVITY (TKC).  FPP IS A FLUID
C PROPERTIES FACTOR USED IN THE EXTERNAL CONVECTION FORMULA FOR FLOW
C ACROSS A TUBE BANK.
      SUBROUTINE FPPG(TF,FPP)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      CGP=A1+A2*TF+A3*TF**2
      AMU=B1+B2*TF+B3*TF**2
      TKC=C1+C2*TF+C3*TF**2
      FPP=(CGP**.33*TKC**.76)/AMU**.28
      RETURN
      END


C SUBROUTINE EPS4 CALCULATES THE "EPSILON" FACTOR FOR A 4 PASS
C CROSS-COUNTER FLOW (ONE FLOW MIXED) HEAT EXCHANGER WITH THE MIXED
C FLOW BEING "C"MINIMUM ("C" IS MASS FLOW * SPECIFIC HEAT).  THIS IS
C USUALLY THE CASE FOR ALL WATER COOLED SURFACE IN THE SHTM. THE MIXED
C FLOW IS THE FLUE GAS
      SUBROUTINE EPS4(VK,VZ,EPSILON)
      C2=.19551+1.02662*VK-.71871*VK**2-1.74159*VK**3+1.36245*VK**4
      C3=1.76425-9.7065*VK+6.15233*VK**2+8.56018*VK**3-7.54606*VK**4
      C4=-2.27962+17.27275*VK-14.2191*VK**2-11.94207*VK**3
     &+12.63196*VK**4
      C5=.266-8.66542*VK+9.40485*VK**2+4.49907*VK**3-6.42847*VK**4
      EPSILON=VZ+C2*VZ**2+C3*VZ**3+C4*VZ**4+C5*VZ**5
      RETURN
      END

C SUBROUTINE EPS2 CALCULATES THE "EPSILON" FACTOR FOR A 2 PASS
C CROSS-COUNTER FLOW (ONE FLOW MIXED) HEAT EXCHANGER WITH THE MIXED
C FLOW BEING "C" MAX. THE MIXED FLOW IS THE FLUE GAS.  "C" OF THE
C FLUE GAS BEING "C" MAX IS THE USUAL CASE FOR ALL STEAM AND AIR
C COOLED SECTIONS OF THE SHTM.
      SUBROUTINE EPS2(VK,VZ,EPSILON)
      C2=.19910+.83387*VK-1.24416*VK**2+.19261*VK**3
     &+.11083*VK**4
      C3=1.74117-8.17810*VK+7.89864*VK**2-1.68746*VK**3
     &-.42509*VK**4
      C4=-2.23376+13.46870*VK-14.32282*VK**2+4.02622*VK**3
     &+.30024*VK**4
      C5=.23732-6.04132*VK+7.39032*VK**2-2.61667*VK**3
     &+.11384*VK**4
      EPSILON=VZ+C2*VZ**2+C3*VZ**3+C4*VZ**4+C5*VZ**5
      RETURN
      END

C             THE HEAT TRANSFER EQUATIONS USED IN THE
C             FOLLOWING SUBROUTINES COME PRIMARILY FROM
C             "STEAM/ITS GENERATION AND USE 39TH ED"
C              BY THE BABCOCK AND WILCOX COMPANY
C             REF. CHAPTERS 4 AND 14

C             THE EPSILON-NTU METHODOLOGY IS BASED PRIMARILY
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C              ON THE MATERIAL IN "HEAT EXCHANGER DESIGN VOL I"
C              BY RAMESH K. SHAH,  SECTION III, CHAPS. 3-3 THRU 3-4

C                *************************
C                *********NOTE**********
C                *************************
C SUBROUTINE CS12 (FOLLOWING) HAS MUCH MORE EXTENSIVE COMMENTS THAN
C THE OTHER SUBROUTINES.  IN GENERAL THE COMMENTS GIVEN FOR SPECIFIC
C STATEMENTS IN CS12 APPLY TO SIMILAR STATEMENTS IN THE OTHER
C SUBROUTINES.

C SUBROUTINE CS12 CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR 12 PASS COOLING SECTIONS
      SUBROUTINE CS12(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R
C FOR FIRST ITERATION ASSUME TYPICAL VALUES FOR COOLING WATER DELTA T
C "DTW", COOLILNG WATER FILM DELTA T "DTFW", FLUE GAS DELTA T "DTGAS",
C AND TUBE WALL DELTA T "DTWALL".
      DTW=15.
      DTFW=15.
      DTGAS=225.
      DTWALL=5.
      ITER=1
      E(II,6)=CF
      D(II,1)=PPHPT
      D(II,2)=CWTF
      D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH
C IT IS ASSUMED THAT THE TUBE BUNDLE IS 8 TUBES WIDE
      G(II,1)=8.
C IT IS ASSUMED THAT THERE ARE 12 PASSES
      G(II,2)=12.
C THE TUBES ARE 2.5 INCH OUTSIDE DIAMETER
      G(II,3)=2.5
C THE TUBES ARE 2.188 INCH INSIDE DIAMETER
      G(II,4)=2.188
C TOTAL HEAT TRANSFER SURFACE AREA OF TUBES IS 465 SQ FT.
      G(II,5)=465.
C OUTSIDE SURFACE AREA OF THE "BOX" IS 163 SQ FT (USED FOR
C CALCULATING HEAT LOSS TO THE ATMOSPHERE) TOP AND BOTTOM SURFACE
C AREAS ARE NEGLECTED
      G(II,6)=163.
C TBULKA IS THE AVERAGE COOLING WATER BULK TEMPERATURE
50    TBULKA=CWTF+(DTW/2.)
C TFILMW IS THE AVERAGE COOLING WATER FILM TEMP
      TFILMW=TBULKA+(DTFW/2.)
C E(II,1) IS THE CONVECTION HEAT TRANSFER COEFF ON WATER SIDE
C OF TUBES (BTU/HR FT**2 F).  THIS EQ ALREADY HAS INCLUDED IN IT
C THE I.D. OF THE TUBES, ASSUMES SUBCOOLED WATER IN THE 130 PSI
C RANGE, AND ASSUMES TURBULENT FLOW.
      E(II,1)=(.10644+1.3862E-3*TFILMW-1.0886E-6*TFILMW**2)
     &*PPHPT**.8
C TGAVG IS AVERAGE FLUE GAS TEMP
      TGAVG=FGTFF-(DTGAS/2.)
C TPS IS AVERAGE OUTSIDE TUBE WALL SURFACE TEMP
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      TPS=TBULKA+DTFW+DTWALL
C TFILMG IS AVERAGE FLUE GAS FILM TEMP
      TFILMG=(TGAVG+TPS)/2.
      CALL FPPG(TFILMG,FPP)
C E(II,2) IS THE CONVECTION HEAT TRANSFER COEFF FOR FLOW ACROSS A
C TUBE BANK.  THIS EQ ALREADY HAS INCLUDED IN IT THE O.D. AND SPACING
C OF THE TUBES AND ASSUMES TURBULENT GAS FLOW.
      E(II,2)=FPP*.0889*FGPPH**.61
C ALMTD IS THE LOG MEAN TEMP
      ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF
     &-DTGAS-CWTF))
C FACK IS A DIMENSIONLESS BEAM LENGTH FACTOR FOR INTERTUBE RADIATION
C IT ASSUMES THE O.D. AND SPACING OF THE COOLING TUBES AND ASSUMES
C THAT THE FUEL IS BITUMINOUS COAL
      FACK=.1881+.268*2.125-.0239*(2.125)**2
C URGP IS THE BASIC INTERTUBE RADIATION COEFFICIENT.
      URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C E(II,3) IS THE INTERTUBE RADIATION COEFF CORRECTED FOR THE BEAM LENGTH
      E(II,3)=URGP*FACK
C IF THIS COOLING SECTION WERE IN A REGION WITH FLUE GAS TEMP BELOW ABOUT
C 350 F, THE INTERTUBE RADIATION EQUATION COULD GIVE A NEGATIVE VALUE
C SET EQUAL TO 0.0 FOR NEGATIVE VALUES
      IF(E(II,3).LE.0.0)E(II,3)=0.0
C E(II,4) IS THE CAVITY RADIATION COEFF, WHICH IS ASSUMED TO BE ZERO
C FOR COOLING SECTIONS
      E(II,4)=0.0
C E(II,5) IS THE OVERALL HEAT TRANSFER COEFF INCLUDING CONVECTION
C AND RADIATION
      E(II,5)=((E(II,3)+E(II,2))*E(II,1))/(E(II,3)+E(II,2)
     &+E(II,1))
C E(II,7) IS THE OVERALL COEFF TIMES THE CLEANLINESS FACTOR  NOTE: THIS
C PROGRAM USES A CLEANLILNESS FACTOR WHICH IS DEFINED IN TERMS OF ITS
C EFFECT OF OVERALL COEFF.
      E(II,7)=E(II,5)*CF
C H(II,2) IS THE TOTAL COOLING WATER FLOW TIMES THE SPECIFIC HEAT, WHICH
C IN THIS CASE IS WATER AND ASSUMED TO BE EQUAL TO 1 BTU/LB F.
C FOR ALL REASONABLE CONDITIONS H(II,2) IS "C" MAX   BTU/HR F
      H(II,2)=PPHPT*G(II,1)*1.
C H(II,1) IS THE TOTAL FLUE GAS FLOW TIMES ITS AVERAGE SPECIFIC HEAT
C IN THE SECTION.  FOR ALL REASONABLE CONDITIONS H(II,1) IS "C" MIN
      H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
C H(II,3) IS THE "C STAR" FACTOR OR (C "MIN")/(C "MAX")
      H(II,3)=H(II,1)/H(II,2)
C H(II,4) IS THE "NUMBER OF THERMAL UNITS" OR NTU VALUE FOR THE MIDDLE
C TUBE BUNDLE OF THE THREE IN A COOLING SECTION.  IT IS EQUAL TO THE
C OVERALL HEAT TRANS COEFF * THE HEAT TRANS SURFACE AREA OF THAT BUNDLE
C DIVIDED BY C "MIN"
      H(II,4)=E(II,7)*155./H(II,1)
      ZZ=H(II,4)/(1.+H(II,4))
C EPS4 CALCULATES THE EPSILON FACTOR (EPSIL) FOR ONE 4 PASS TUBE BUNDLE
      CALL EPS4(H(II,3),ZZ,EPSIL)
C THE NEXT 4 LINES GIVE THE EQUATION FOR CALCULATING EPSILON FOR MULTIPLE
C TUBE BANKS IN SERIES.  IN THE CASE OF THE COOLING SECTIONS, THERE ARE
C THREE 4 PASS TUBE BUNDLES IN SERIES
      EE=1.-EPSIL*H(II,3)
      EF=1.-EPSIL
      EG=(EE/EF)**3
      EPS3=(EG-1.)/(EG-H(II,3))
C EPS3 IS THE EPSILON VALUE FOR THREE 4 PASS BUNDLES IN SERIES  I.E. THE
C EPSILON VALUE FOR A WHOLE COOLING SECTION
      H(II,5)=EPS3
C TOTAL HEAT TRANSFER = EPSILON * (TOTAL FLUE GAS FLOW * FLUE GAS SPECIFIC
C HEAT) * (DIFFERENCE IN INLET TEMPERATURES OF FLUE GAS AND WATER)
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      F(II,1)=EPS3*H(II,1)*(FGTFF-CWTF)
C HEAT LOSS FROM WALLS IS ESTIMATED AT 700 BTU/HR PER SQ FT OF BOX SURFACE
      F(II,2)=700.*G(II,6)
C TOTAL HEAT LOSS FROM THE FLUE GAS IS LOSS TO TUBES PLUS LOSS TO WALLS
      F(II,3)=F(II,1)+F(II,2)
C PREDICTED DELTA T OF COOLING WATER
      PDTW=F(II,1)/H(II,2)
C PREDICTED DELTA T OF FLUE GAS
      PDTGAS=F(II,3)/H(II,1)
C CONVERGENCE CRITERIA IS BASED ON TWO CONSECUTIVE CALCULATIONS OF THE
C FLUE GAS DELTA T
      IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
C FOR THE NEXT ITERATION SET THE "GUESS" EQUAL TO THE LATEST PREDICTION
C OF WATER AND FLUE GAS TEMPERATURES
      DTW=PDTW
      DTGAS=PDTGAS
      DTFW=F(II,1)/(E(II,1)*G(II,5))
C DTWALL IS CALCULATED ASSUMING CARBON STEEL TYPE MATL (K=27 BTU FT/
C HR FT**2 F) AND A WALL THICKNESS OF .156" (.013 FT).
      DTWALL=(F(II,1)*.013)/(27.*G(II,5))
      ITER=ITER+1
C SET A LIMIT ON NUMBER OF ITERATIONS
      IF(ITER.GE.10)GO TO 300
      GO TO 50
C PREDICTED OUTLET FLUE GAS TEMP= INLET TEMP - CONVERGED DELTA T CALC'D
300   D(II,5)=FGTFF-PDTGAS
C COOLING WATER OUTLET TEMP= INLET TEMP + CALC'D COOLING WATER DELTA T
      D(II,3)=CWTF+PDTW
C REDUCE THE OUTLET FLUE GAS TEMPERATURE BY THE AMOUNT OF MIXING WITH
C AMBIENT AIR INLEAKAGE (SOOTBLOWING).
      IF(AIRLKG.EQ.0.0)GO TO 301
      CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
      D(II,5)=T
301   CONTINUE

      RETURN
      END

C SUBROUTINE MCS5 CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR A MODIFIED (5 TUBES ACROSS) COOLING SECTION (12 PASSES DEEP)
      SUBROUTINE MCS5(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

C ASSUME TYPICAL VALUES FOR FIRST ITERATION
      DTW=10.
      DTFW=15.
      DTGAS=175.
      DTWALL=5.
      ITER=1
      E(II,6)=CF
      D(II,1)=PPHPT
      D(II,2)=CWTF
      D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH
C THE TUBE BANK HAS 5 TUBES ACROSS BY 12 PASSES, EACH TUBE IS 2.5"
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C O.D. BY 2.188" I.D., WITH 292.5 SQ FT TOTAL TUBE SURFACE AREA AND
C 163 SQ FT OF WALL SURFACE AREA
        G(II,1)=5.
        G(II,2)=12.
        G(II,3)=2.5
        G(II,4)=2.188
        G(II,5)=292.5
        G(II,6)=163.

50      TBULKA=CWTF+(DTW/2.)
        TFILMW=TBULKA+(DTFW/2.)
C CONVECTION COEFF INSIDE TUBES (FOR WATER)
        E(II,1)=(.10644+1.3862E-3*TFILMW-1.0886E-6*
     &TFILMW**2)*PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFW+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF GAS ACROSS TUBE BANK, NOTE THAT THIS EQ IS
C DIFF FROM CS12 BECAUSE OF A DIFFERENCE IN TUBE SDPACING
        E(II,2)=FPP*.0818*FGPPH**.61
        ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF-
     &DTGAS-CWTF))
        FACK=.1881+.268*2.50-.0239*(2.50**2)
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C RADIATION COEFF FOR INTERTUBE RADIATION WITH BEAM LENGTH FACTOR
        E(II,3)=URGP*FACK
        IF(E(II,3).LE.0.0)E(II,3)=0.0
C E(II,4) IS THE CAVITY RADIATION WHICH IS ASSUMED TO BE ZERO FOR
C COOLING SECTIONS.
        E(II,4)=0.0
C OVERALL HEAT TRANSFER COEFF
        E(II,5)=((E(II,2)+E(II,3))*E(II,1))/(E(II,2)+E(II,3)+
     &E(II,1))
C APPLY CLEANLINESS FACTOR TO OVERALL COEFF
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF WATER IS TAKEN AS 1 BTU/LB F
        H(II,2)=PPHPT*G(II,1)*1.
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,1)/H(II,2)
C CALC NTU FOR THE 4 PASS (5 TUBES ACROSS) TUBE BUNDLE
        H(II,4)=E(II,7)*97.5/H(II,1)
        ZZ=H(II,4)/(1.+H(II,4))
C CALL EPS4 TO CALC EPSILON (EPSIL) FOR 4 PASS CROSS-COUNTER FLOW
        CALL EPS4(H(II,3),ZZ,EPSIL)
C CALCULATE EPSILON FOR MULTIPLE (3) TUBE BANKS IN SERIES
        EE=1.-EPSIL*H(II,3)
        EF=1.-EPSIL
        EG=(EE/EF)**3
        EPS3=(EG-1.)/(EG-H(II,3))
        H(II,5)=EPS3
C CALC TOTAL HEAT TRANSFER BTU/HR TO TUBES
        F(II,1)=EPS3*H(II,1)*(FGTFF-CWTF)
C CACL HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FROM FLUE GAS
        F(II,3)=F(II,1)+F(II,2)
C CALC DELTA T'S FOR COOLING WATER AND FLUE GAS
        PDTW=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
C CHECK CONVERGENCE
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTW=PDTW
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        DTGAS=PDTGAS
        DTFW=F(II,1)/(E(II,1)*G(II,5))
C CARBON STEEL MATL ASSUMED
        DTWALL=(F(II,1)*.013)/(27.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=CWTF+PDTW
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE CM4P CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR ONE TUBE BANK (4 PASSES) OF A COOLING SECTION
        SUBROUTINE CM4P(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWTF,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R

C ASSUME TYPICAL VALUES FOR FIRST ITERATION
        DTW=10.
        DTFW=15.
        DTGAS=100.
        DTWALL=5.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=CWTF
        D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
C A SINGLE BANK HAS 8 TUBES OF 4 PASSES EACH, EACH TUBE IS 2.5"
C O.D. BY 2.188" I.D., WITH 155 SQ FT TOTAL TUBE SURFACE AREA AND
C 54.3 SQ FT OF WALL SURFACE AREA
        G(II,1)=8.
        G(II,2)=4.
        G(II,3)=2.5
        G(II,4)=2.188
        G(II,5)=155.
        G(II,6)=54.3

50      TBULKA=CWTF+(DTW/2.)
        TFILMW=TBULKA+(DTFW/2.)
C CONVECTION COEFF INSIDE TUBES (FOR WATER)
        E(II,1)=(.10644+1.3862E-3*TFILMW-1.0886E-6*
     &TFILMW**2)*PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFW+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF GAS ACROSS TUBE BANK, NOTE THAT THIS EQ IS
C DIFF FROM CS12 BECAUSE OF A DEPTH FACTOR
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      E(II,2)=FPP*.0809*FGPPH**.61
      ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF-
     &DTGAS-CWTF))
      FACK=.1881+.268*2.125-.0239*(2.125**2)
      URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C RADIATION COEFF FOR INTERTUBE RADIATION WITH BEAM LENGTH FACTOR
      E(II,3)=URGP*FACK*.76
      IF(E(II,3).LE.0.0)E(II,3)=0.0
C A SINGLE TUBE BANK IS ASSUMED TO HAVE A LARGE CAVITY EITHER UPSTREAM
C OR DOWNSTREAM
C THE NEXT 3 LINES CALCULATE THE CAVITY RADIATION HEAT LOAD ON THE
C SCREEN TUBES
      TFGR4=(FGTFF+460.)**4
      TTSR4=(TPS+460.)**4
      QCAVR=9.72E-9*(TFGR4-TTSR4)
      E(II,4)=QCAVR/(155.*ALMTD)
C THE TOTAL RADIATION HEAT FLUX IS THE SUM OF INTERTUBE PLUS CAVITY
      URGTOT=E(II,3)+E(II,4)
C OVERALL HEAT TRANSFER COEFF
      E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
C APPLY CLEANLINESS FACTOR TO OVERALL COEFF
      E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF WATER IS TAKEN AS 1 BTU/LB F
      H(II,2)=PPHPT*G(II,1)*1.
      H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
      H(II,3)=H(II,1)/H(II,2)
C CALC NTU FOR THE 4 PASS TUBE BUNDLE
      H(II,4)=E(II,7)*155./H(II,1)
      ZZ=H(II,4)/(1.+H(II,4))
C CALL EPS4 TO CALC EPSILON (EPSIL) FOR 4 PASS CROSS-COUNTER FLOW
      CALL EPS4(H(II,3),ZZ,EPSIL)
      H(II,5)=EPSIL
C CALC TOTAL HEAT TRANSFER BTU/HR TO TUBES
      F(II,1)=EPSIL*H(II,1)*(FGTFF-CWTF)
C CACL HEAT LOSS TO WALLS
      F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FROM FLUE GAS
      F(II,3)=F(II,1)+F(II,2)
C CALC DELTA T'S FOR COOLING WATER AND FLUE GAS
      PDTW=F(II,1)/H(II,2)
      PDTGAS=F(II,3)/H(II,1)
C CHECK CONVERGENCE
      IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
      DTW=PDTW
      DTGAS=PDTGAS
      DTFW=F(II,1)/(E(II,1)*G(II,5))
C CARBON STEEL MATL ASSUMED
      DTWALL=(F(II,1)*.013)/(27.*G(II,5))
      ITER=ITER+1
      IF(ITER.GE.10)GO TO 300
      GO TO 50
300   D(II,5)=FGTFF-PDTGAS
      D(II,3)=CWTF+PDTW
      IF(AIRLKG.EQ.0.0)GO TO 301
      CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
      D(II,5)=T
301   CONTINUE

      RETURN
      END

C SUBROUTINE MCH5 CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
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C T'S FOR ONE TUBE BANK (4 PASSES) OF A MODIFIED COOLING SECTION WITH
C ONLY 5 TUBES ACROSS
      SUBROUTINE MCM5(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

C ASSUME TYPICAL VALUES FOR FIRST ITERATION
      DTW=10.
      DTFW=15.
      DTGAS=75.
      DTWALL=5.
      ITER=1
      E(II,6)=CF
      D(II,1)=PPHPT
      D(II,2)=CWTF
      D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH
C A SINGLE BANK HAS 5 TUBES OF 4 PASSES EACH, EACH TUBE IS 2.5"
C O.D. BY 2.188" I.D., WITH 97.5 SQ FT TOTAL TUBE SURFACE AREA AND
C 54.3 SQ FT OF WALL SURFACE AREA
      G(II,1)=5.
      G(II,2)=4.
      G(II,3)=2.5
      G(II,4)=2.188
      G(II,5)=97.5
      G(II,6)=54.3

50    TBULKA=CWTF+(DTW/2.)
      TFILMW=TBULKA+(DTFW/2.)
C CONVECTION COEFF INSIDE TUBES (FOR WATER)
      E(II,1)=(.10644+1.3862E-3*TFILMW-1.0886E-6*
     &TFILMW**2)*PPHPT**.8
      TGAVG=FGTFF-(DTGAS/2.)
      TPS=TBULKA+DTFW+DTWALL
      TFILMG=(TGAVG+TPS)/2.
      CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF GAS ACROSS TUBE BANK, NOTE THAT THIS EQ IS
C DIFF FROM MCS5 BECAUSE OF A DEPTH FACTOR
      E(II,2)=FPP*.0736*FGPPH**.61
      ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF-
     &DTGAS-CWTF))
      FACK=.1881+.268*2.50-.0239*(2.50**2)
      URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C RADIATION COEFF FOR INTERTUBE RADIATION WITH BEAM LENGTH FACTOR
      E(II,3)=URGP*FACK*.79
      IF(E(II,3).LE.0.0)E(II,3)=0.0
C A SINGLE TUBE BANK IS ASSUMED TO HAVE A LARGE CAVITY EITHER UPSTREAM
C OR DOWNSTREAM
C THE NEXT 3 LINES CALCULATE THE CAVITY RADIATION HEAT LOAD ON THE
C TUBES
      TFGR4=(FGTFF+460.)**4
      TTSR4=(TPS+460.)**4
      QCAVR=9.72E-9*(TFGR4-TTSR4)
      E(II,4)=(.85*QCAVR)/(G(II,5)*ALMTD)
C THE TOTAL RADIATION HEAT FLUX IS THE SUM OF INTERTUBE PLUS CAVITY
      URGTOT=E(II,3)+E(II,4)
```

```
C OVERALL HEAT TRANSFER COEFF
        E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
C APPLY CLEANLINESS FACTOR TO OVERALL COEFF
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF WATER IS TAKEN AS 1 BTU/LB F
        H(II,2)=PPHPT*G(II,1)*1.
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,1)/H(II,2)
C CALC NTU FOR THE 4 PASS TUBE BUNDLE
        H(II,4)=E(II,7)*97.5/H(II,1)
        ZZ=H(II,4)/(1.+H(II,4))
C CALL EPS4 TO CALC EPSILON (EPSIL) FOR 4 PASS CROSS-COUNTER FLOW
        CALL EPS4(H(II,3),ZZ,EPSIL)
        H(II,5)=EPSIL
C CALC TOTAL HEAT TRANSFER BTU/HR TO TUBES
        F(II,1)=EPSIL*H(II,1)*(FGTFF-CWTF)
C CACL HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FROM FLUE GAS
        F(II,3)=F(II,1)+F(II,2)
C CALC DELTA T'S FOR COOLING WATER AND FLUE GAS
        PDTW=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
C CHECK CONVERGENCE
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTW=PDTW
        DTGAS=PDTGAS
        DTFW=F(II,1)/(E(II,1)*G(II,5))
C CARBON STEEL MATL ASSUMED
        DTWALL=(F(II,1)*.013)/(27.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=CWTF+PDTW
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END

C SUBROUTINE SCR4 CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR THE FOUR PASS WATER COOLED SCREEN TUBES IN TS-1
        SUBROUTINE SCR4(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWTF,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R

C ASSUME TYPICAL VALUES FOR FIRST ITERATION
        DTW=15.
        DTFW=20.
        DTGAS=300.
        DTWALL=10.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=CWTF
        D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
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C FOR THE SECTION
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
C THE SCREEN HAS 8 TUBES OF 4 PASSES EACH, EACH TUBE IS 2.5"
C O.D. BY 2.188" I.D., WITH 155 SQ FT TOTAL TUBE SURFACE AREA AND
C 50.2 SQ FT OF WALL SURFACE AREA
        G(II,1)=8.
        G(II,2)=4.
        G(II,3)=2.5
        G(II,4)=2.188
        G(II,5)=155.
        G(II,6)=50.2

50      TBULKA=CWTF+(DTW/2.)
        TFILMW=TBULKA+(DTFW/2.)
C CONVECTION COEFF INSIDE TUBES (FOR WATER)
        E(II,1)=(.10644+1.3862E-3*TFILMW-1.0886E-6*
     &TFILMW**2)*PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFW+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF GAS ACROSS TUBE BANK, NOTE THAT THIS EQ IS
C DIFF FROM CS12 BECAUSE OF A DEPTH FACTOR
        E(II,2)=FPP*.0809*FGPPH**.61
        ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF-
     &DTGAS-CWTF))
        FACK=.1881+.268*2.125-.0239*(2.125**2)
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C RADIATION COEFF FOR INTERTUBE RADIATION WITH BEAM LENGTH FACTOR
        E(II,3)=URGP*FACK*.76
        IF(E(II,3).LE.0.0)E(II,3)=0.0
C THE SCREEN HAS A LARGE CAVITY JUST UPSTREAM (THE INLET TRANSITION)
C THE NEXT 3 LINES CALCULATE THE CAVITY RADIATION HEAT LOAD ON THE
C SCREEN TUBES
        TFGR4=(FGTFF+460.)**4
        TTSR4=(TPS+460.)**4
        QCAVR=9.72E-9*(TFGR4-TTSR4)
        E(II,4)=QCAVR/(155.*ALMTD)
C THE TOTAL RADIATION HEAT FLUX IS THE SUM OF INTERTUBE PLUS CAVITY
        URGTOT=E(II,3)+E(II,4)
C OVERALL HEAT TRANSFER COEFF
        E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
C APPLY CLEANLINESS FACTOR TO OVERALL COEFF
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF WATER IS TAKEN AS 1 BTU/LB F
        H(II,2)=PPHPT*G(II,1)*1.
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,1)/H(II,2)
C CALC NTU FOR THE 4 PASS SCREEN
        H(II,4)=E(II,7)*155./H(II,1)
        ZZ=H(II,4)/(1.+H(II,4))
C CALL EPS4 TO CALC EPSILON (EPSIL) FOR 4 PASS CROSS-COUNTER FLOW
        CALL EPS4(H(II,3),ZZ,EPSIL)
        H(II,5)=EPSIL
C CALC TOTAL HEAT TRANSFER BTU/HR TO TUBES
        F(II,1)=EPSIL*H(II,1)*(FGTFF-CWTF)
C CACL HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FROM FLUE GAS
```

```
        F(II,3)=F(II,1)+F(II,2)
C CALC DELTA T'S FOR COOLING WATER AND FLUE GAS
        PDTW=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
C CHECK CONVERGENCE
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTW=PDTW
        DTGAS=PDTGAS
        DTFW=F(II,1)/(E(II,1)*G(II,5))
C CARBON STEEL MATL ASSUMED
        DTWALL=(F(II,1)*.013)/(27.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=CWTF+PDTW
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE CSXP CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR COOLING SECTIONS WITH EXTRA HEAT TRANSFER SURFACE (LIKE
C COOLING SECTIONS CS-5 AND CS/6)
        SUBROUTINE CSXP(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1;A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWTF,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R
C INITIAL GUESSES FOR TYPICAL CONDITIONS
        DTW=10.
        DTFW=10.
        DTGAS=200.
        DTWALL=10.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=CWTF
        D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
C THE "EXTRA" TUBES IN THESE SECTIONS ARE THE SAME O.D. AND I.D. AS THE
C ORIGINAL TUBES BUT ARE ARRANGED DIFFERENTLY.
        G(II,1)=11.
        G(II,2)=12.
        G(II,3)=2.5
        G(II,4)=2.188
        G(II,5)=1130.4
        G(II,6)=163.

50      TBULKA=CWTF+(DTW/2.)
        TFILMW=TBULKA+(DTFW/2.)
C CONVECTION COEFFICIENT INSIDE TUBES (WATER)
        E(II,1)=(.10644+1.3862E-3*TFILMW-1.0806E-6*
     &TFILMW**2)*PPHPT**.8
```

```
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFW+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF FOR FLOW OVER TUBE BANK, NOTE THAT THIS EQ
C IS DIFFERENT FROM CS12 BECAUSE OF THE DIFFERENT TUBE ARRANGEMENT
        E(II,2)=FPP*.07112*FGPPH**.61
        ALMTD=(DTGAS-DTW)/ALOG((FGTFF-CWTF-DTW)/(FGTFF-
     &DTGAS-CWTF))
C BEAM LENGTH FACTOR BASED ON TUBE SPACING WITH EXTRA TUBES
        FACK=.1881+.268*.875-.0239*(.875**2)
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
        E(II,3)=URGP*FACK
        IF(E(II,3).LE.0.0)E(II,3)=0.0
C THE INTERTUBE RADIATION APPROXIMATION USED HERE COULD GIVE NEGATIVE
C VALUES IN REGIONS WITH FLUE GAS TEMPERATURES BELOW ABOUT 350 F
C RADIATION WOULD BE NEGLIGABLE IN THESE AREAS ANYWAY, THEREFORE
C IT IS SET = O
        IF(E(II,3).LE.0.0)E(II,3)=0.0
C CAVITY RADIATION IS ASSUMED NEGLIGABLE
        E(II,4)=0.
        E(II,5)=((E(II,3)+E(II,2))*E(II,1))/(E(II,3)+E(II,2)
     &+E(II,1))
C APPLY CLEANLINESS FACTOR
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF WATER TAKEN AS 1 BTU/LB F
        H(II,2)=PPHPT*G(II,1)*1.
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,1)/H(II,2)
C NTU IS CALCULATED OF ONE 8 X 4 PASS TUBE BUNDLE W/O EXTRA TUBES
        H(II,4)=E(II,7)*161.5/H(II,1)
        ZZ=H(II,4)/(1.+H(II,4))
C EPSILON (EPSIL) IS CALCULATED FOR 8 X 4 PASS BUNDLE
        CALL EPS4(H(II,3),ZZ,EPSIL)
        EE=1.-EPSIL*H(II,3)
        EF=1.-EPSIL
        EG=(EE/EF)**7
C EPS7 IS THE EQUIVALENT EPSILON FOR 7 TUBE BUNDLES IN SERIES, WHICH
C IS THE APPROXIMATION OF A COOLILNG SECTION WITH EXTRA TUBES
        EPS7=(EG-1.)/(EG-H(II,3))
        H(II,5)=EPS7
C CALCULATE TOTAL HEAT TRANSFER TO TUBES
        F(II,1)=H(II,5)*H(II,1)*(FGTFF-CWTF)
C CALCULATE WALL LOSS
        F(II,2)=700.*G(II,6)
C CALCULATE TOTAL HEAT LOSS FROM FLUE GAS
        F(II,3)=F(II,1)+F(II,2)
C CALCULATE NEW TEMPS AND CHECK CONVERGENCE
        PDTW=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTW=PDTW
        DTGAS=PDTGAS
        DTFW=F(II,1)/(E(II,1)*G(II,5))
C CARBON STEEL MATL ASSUMED
        DTWALL=(F(II,1)*.013)/(27.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=CWTF+PDTW
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
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        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE TS12 CALCULATES THE HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR THE TWO PASS STEAM COOLED TUBES IN TS-1
        SUBROUTINE TS12(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWFT,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R
C INITIAL GUESSES FOR TYPICAL OPERATION
        DTS=450.
        DTFS=200.
        DTGAS=20.
        DTWALL=10.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=STTF
        D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
C THE STEAM TUBES ARE 8 TUBES WIDE BY TWO PASSES EACH.
        G(II,1)=8.
        G(II,2)=2.
C 1.5" IS TAKEN AS THE O.D. EVENTHOUGH 1 1/4" NOM PIPE (1.66"
C ACTUAL O.D.) WAS USED IN SOME PLACES
        G(II,3)=1.5
C THE "TUBES/PIPES" HAD VARIOUS WALL THICKNESSES.  1.11" I.D. WAS
C TAKEN AS THE AVERAGE
        G(II,4)=1.11
        G(II,5)=46.2
        G(II,6)=40.4

50      TBULKA=STTF+(DTS/2.)
        TFILMS=TBULKA+(DTFS/2.)
C CONVECTION COEFF FOR STEAM IN TUBE  (1.11 I.D. AND SUPERHEATED
C STEAM ARE ASSUMED)
        E(II,1)=(.37408-4.0E-5*TFILMS+1.54E-7*TFILMS**2)*
     &PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFS+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF FOR FLUE GAS OVER STEAM TUBES IF SCREEN
C TUBES ARE INSTALLED
        E(II,2)=FPP*.1*FGPPH**.61
        IJ=II-1
C CONVECTION COEFF FOR FLUE GAS OVER STEAM TUBES IF SCREEN
C TUBES ARE NOT INSTALLED
        IF(Q(IJ).EQ.'SCR0')E(II,2)=FPP*.082*FGPPH**.61
        ALMTD=(DTGAS-DTS)/ALOG((FGTFF-STTF-DTS)/(FGTFF-
     &DTGAS-STTF))
        FACK=.1881+.268*1.575-.0239*(1.575**2)
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
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C INTERTUBE RADIATION WITH SCREEN TUBES
        E(II,3)=URGP*FACK*.85
        IF(E(II,3).LE.0.0)E(II,3)=0.0
C INTERTUBE RADIATION WITHOUT SCREEN TUBES
        IF(Q(IJ).EQ.'SCR0')E(II,3)=URGP*FACK*.7
        IF(E(II,3).LE.0.0)E(II,3)=0.0
        TFGR4=(FGTFF+460.)**4
        TTSR4=(TPS+460.)**4
        QCAVR=9.72E-9*(TFGR4-TTSR4)
C CAVITY RADIATION (ONLY 59% IS ABSOSRBED BY STEAM TUBES)
        E(II,4)=(.59*QCAVR)/(G(II,5)*ALMTD)
        URGTOT=E(II,3)+E(II,4)
        E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
C APPLY CLEANLINESS FACTOR
        E(II,7)=E(II,5)*CF
C .48 BTU/LB F IS SPECIFIC HEAT FOR SUPERHEATED STEAM OVER A WIDE
C RANGE OF TEMP AND PRESS NEAR THE TYPICAL OPERATING POINT
C FOR THE TS-1 TUBES
        H(II,2)=PPHPT*G(II,1)*.48
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,2)/H(II,1)
C CALC NTU FOR STEAM TUBE BUNDLE
        H(II,4)=E(II,7)*G(II,5)/H(II,2)
        ZZ=H(II,4)/(1.+H(II,4))
C CALC EPSILON (EPSIL) FOR 2 PASS CROSS-COUNTER FLOW
        CALL EPS2(H(II,3),ZZ,EPSIL)
        H(II,5)=EPSIL
C CALC TOTAL HEAT TRANSFER TO STEAM TUBES
        F(II,1)=EPSIL*H(II,2)*(FGTFF-STTF)
C CALC HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT TRANSFER, AND PREDICTED TEMPS
        F(II,3)=F(II,1)+F(II,2)
        PDTS=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTS=PDTS
        DTGAS=PDTGAS
        DTFS=F(II,1)/(E(II,1)*G(II,5))
C ASSUMES STAINLESS STEEL TYPE MATL (K=9 BTU-FT/HR FT**2 F)
C FOR TUBE WALL AND AVG WALL THKNESS OF .195" (.016 FT)
        DTWALL=(F(II,1)*.016)/(9.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=STTF+PDTS
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END

C SUBROUTINE TS2P CALCULATES HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR 2 PASS STEAM COOLED TUBES SUCH AS IN TS-2 (THE TS-2 TUBES
C ARE POSITIONED IN THE CENTER OF THE BOX AND ARE ASSUMED TO HAVE A
C CAVITY ON BOTH SIDES WHILE TS-1 STEAM TUBES ARE POSITIONED AT THE
C DOWNSTREAM END OF THE BOX AND CAVITY RADIATION DEPENDS ON THE
C PRESENCE OF THE SCREEN TUBES
        SUBROUTINE TS2P(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
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      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWFT,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R
C INITIAL QUESSES FOR FIRST ITERATION
      DTS=400.
      DTFS=200.
      DTGAS=20.
      DTWALL=10.
      ITER=1
      E(II,6)=CF
      D(II,1)=PPHPT
      D(II,2)=STTF
      D(II,4)=FGTFF
C CALCULATE AMOUNT OF AIRLEAKAGE "AIRLKG" AND THE TOTAL FLOW "FGPPH"
C FOR THE SECTION
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH
      G(II,1)=8.
      G(II,2)=2.
C TUBE SIZES ARE AVERAGES OF TUBES AND 1 1/4 PIPE
      G(II,3)=1.5
      G(II,4)=1.11
      G(II,5)=47.0
      G(II,6)=93.3

50    TBULKA=STTF+(DTS/2.)
      TFILMS=TBULKA+(DTFS/2.)
C CONVECTION COEFF FOR SUPERHEATED STEAM IN A 1.11" ID TUBE
      E(II,1)=(.37408-4.0E-5*TFILMS+1.54E-7*TFILMS**2)*
     &PPHPT**.8
      TGAVG=FGTFF-(DTGAS/2.)
      TPS=TBULKA+DTFS+DTWALL
      TFILMG=(TGAVG+TPS)/2.
      CALL FPPG(TFILMG,FPP)
C CONVECTION FOR FLUE GAS PASSING ACROSS A TUBE BANK OF 1.5"
C O.D. TUBES TWO PASSES DEEP PRECEEDED BY ANOTHER TUBE BANK
      E(II,2)=FPP*.1*FGPPH**.61
C IF THERE IS NO TUBE BANK IMMEDIATELY UPSTREAM OF THIS SECTION
C THEN THE CONVECTION COEFF CHANGES
      IF(Q(IJ).EQ.'CSX0')E(II,2)=FPP*.082*FGPPH**.61
      ALMTD=(DTGAS-DTS)/ALOG((FGTFF-STTF-DTS)/(FGTFF-
     &DTGAS-STTF))
      FACK=.1881+.268*1.575-.0239*(1.575**2)
      URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C INTERTUBE RADIATION
      E(II,3)=URGP*FACK*.70
      IF(E(II,3).LE.0.0)E(II,3)=0.0
      TFGR4=(FGTFF+460.)**4
      TTSR4=(TPS+460.)**4
      QCAVR=9.72E-9*(TFGR4-TTSR4)
C CAVITY RADIATION COEFF
      E(II,4)=(.59*QCAVR)/(G(II,5)*ALMTD)
C TOTAL RADIATION COEFF
      URGTOT=E(II,3)+E(II,4)
C OVERALL HEAT TRANS COEFF
      E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
C APPLY CLEANLINESS FACTOR
      E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF SUPERHEATED STEAM = .48 BTU/LB F FOR ALL NORMAL
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C OPERATING CONDITIOS
        H(II,2)=PPHPT*G(II,1)*.48
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,2)/H(II,1)
C CALC NTU
        H(II,4)=E(II,7)*G(II,5)/H(II,2)
        ZZ=H(II,4)/(1.+H(II,4))
C CALC EPSILON (EPSIL) FOR 2 PASS CROSS-COUNTER FLOW GEOMETRY
        CALL EPS2(H(II,3),ZZ,EPSIL)
        H(II,5)=EPSIL
C CALC TOTAL HEAT TRANS TO COOLING TUBES
        F(II,1)=EPSIL*H(II,2)*(FGTFF-STTF)
C CALC HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FORM FLUE GAS
        F(II,3)=F(II,1)+F(II,2)
C CALC NEW TEMPS
        PDTS=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTS=PDTS
        DTGAS=PDTGAS
        DTFS=F(II,1)/(E(II,1)*G(II,5))
C ASSUMES STAINLESS STEEL MATL AND .016 FT AVG THICKNESS
        DTWALL=(F(II,1)*.016)/(9.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=STTF+PDTS
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE TS4P CALCULATES HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR 4 PASS STEAM COOLED SECTIONS, LIKE TS-3
        SUBROUTINE TS4P(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWFT,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R
C MAKE INITIAL GUESSES FOR FIRST ITERATION
        DTS=400.
        DTFS=200.
        DTGAS=20.
        DTWALL=10.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=STTF
        D(II,4)=FGTFF
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
        G(II,1)=8.
        G(II,2)=4.
        G(II,3)=1.5
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        G(II,4)=1.11
        G(II,5)=94.6
        G(II,6)=93.3

50      TBULKA=STTF+(DTS/2.)
        TFILMS=TBULKA+(DTFS/2.)
C CONVECTION COEFF FOR SUPERHEATED STEAM IN A 1.11" ID TUBE
        E(II,1)=(.37408-4.0E-5*TFILMS+1.54E-7*TFILMS**2)*
     &PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFS+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF FOR FLUE GAS ACROSS 4 PASSES OF 1.5"
C O.D. TUBES
        E(II,2)=FPP*.088*FGPPH**.61
        ALMTD=(DTGAS-DTS)/ALOG((FGTFF-STTF-DTS)/(FGTFF-
     &DTGAS-STTF))
        FACK=.1881+.268*1.575-.0239*(1.575**2)
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
C INTERTUBE RADIATION
        E(II,3)=URGP*FACK*.58
        IF(E(II,3).LE.0.0)E(II,3)=0.0
        TFGR4=(FGTFF+460.)**4
        TTSR4=(TPS+460.)**4
        QCAVR=9.72E-9*(TFGR4-TTSR4)
C CAVITY RADIATION
        E(II,4)=(.84*QCAVR)/(G(II,5)*ALMTD)
C TOTAL RADIATION
        URGTOT=E(II,3)+E(II,4)
C OVERALL HEAT TRANS COEFF
        E(II,5)=((E(II,2)+URGTOT)*E(II,1))/(E(II,2)+URGTOT+
     &E(II,1))
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF SUPERHEATED STEAM = .48 BTU/LB F FOR ALL TYP
C OPERATING CONDITIONS
        H(II,2)=PPHPT*G(II,1)*.48
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,2)/H(II,1)
C CALC NTU FOR ONE 2 PASS TUBE BUNDLE AT AVG CONDITIONS
        H(II,4)=E(II,7)*47.3/H(II,2)
        ZZ=H(II,4)/(1.+H(II,4))
C CALC EPSILON (EPSIL) FOR 2 PASS CROSS-COUNTER FLOW GEOMETRY
        CALL EPS2(H(II,3),ZZ,EPSIL)
C CALC EPSILON (EPSIL2) FOR TWO 2-PASS TUBE BUNDLES IN SERIES
        EE=1.-EPSIL*H(II,3)
        EF=1.-EPSIL
        EG=(EE/EF)**2
        EPSIL2=(EG-1.)/(EG-H(II,3))
        H(II,5)=EPSIL2
C CALC TOTAL HEAT TRANS TO TUBES
        F(II,1)=EPSIL2*H(II,2)*(FGTFF-STTF)
C CALC HEAT LOSS TO WALLS
        F(II,2)=700.*G(II,6)
C CALC TOTAL HEAT LOSS FROM FLUE GAS
        F(II,3)=F(II,1)+F(II,2)
C CALC NEW TEMPS AND CHK CONVERGENCE
        PDTS=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTS=PDTS
        DTGAS=PDTGAS
        DTFS=F(II,1)/(E(II,1)*G(II,5))
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```
C DTWALL ASSUMES STAINLESS STEEL TYPE MATL AND .016 FT
C THICKNESS FOR AVG TUBE
        DTWALL=(F(II,1)*.016)/(9.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=STTF+PDTS
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE CSAH CALCULATES HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR TWO COOLING SECTION BOXES IN SERIES WITH AIR HEATER TUBES
        SUBROUTINE CSAH(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWFT,STTF,AIRTF
        COMMON D,E,F,G,H,Q,R
C MAKE INITIAL GUESSES FOR FIRST ITERATION
        DTA=400.
        DTFA=50.
        DTGAS=200.
        DTWALL=10.
        ITER=1
        E(II,6)=CF
        D(II,1)=PPHPT
        D(II,2)=AIRTF
        D(II,4)=FGTFF
        AIRLKG=PCTLKG*FGPPHI
        F(II,4)=AIRLKG
        FGPPH=FGPPH+AIRLKG
        F(II,5)=FGPPH
C EACH COOLING SECTION BOX W/AH TUBES HAS 30 TUBES, MAKING 8 PASSES EACH,
C FOR A TOTAL OF 16 PASSES. THE TUBES ARE CARBON STEEL, 2" OD X 1.782" ID.
        G(II,1)=30.
        G(II,2)=16.
        G(II,3)=2.0
        G(II,4)=1.782
C TOTAL HEAT TRANSFER SURFACE AREA IS 1649.4 SQ FT AND THE SURFACE AREA
C OF TWO COOLING SECTION BOXES IS 326. SQ FT.
        G(II,5)=1649.4
        G(II,6)=326.

50      TBULKA=AIRTF+(DTA/2.)
        TFILMA=TBULKA+(DTFA/2.)
C HEAT TRANS COEFF FOR AIR IN A 1.782" ID TUBE
        E(II,1)=(.1183+5.068E-5*TFILMA-1.49E-8*TFILMA**2)*
     &PPHPT**.8
        TGAVG=FGTFF-(DTGAS/2.)
        TPS=TBULKA+DTFA+DTWALL
        TFILMG=(TGAVG+TPS)/2.
        CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF FOR FLUE GAS ACROSS TUBE BANK OF AIR HEATER
C TUBES IN A COOLILNG SECTION BOX.  THE ".85" FACTOR IS FOR
C THE HIGH % OF TUBE SURFACE IN PARALLEL FLOW RATHER THAN CROSS
        E(II,2)=(FPP*.0804*FGPPH**.61)*.85
        ALMTD=(DTGAS-DTA)/ALOG((FGTFF-AIRTF-DTA)/(FGTFF-
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     &DTGAS-AIRTF))
C INTERTUBE RADIATION CAN BE NEGLIGABLE IF THE FLUE GAS TEMP IS
C BELOW ABOUT 800 F, CAVITY RAD IS TAKEN AS 0.0
        FACK=.188+.268*1.0-.0239*1.0**2
        URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
        E(II,3)=URGP*FACK
        IF(E(II,3).LE.0.0)E(II,3)=0.0
        E(II,4)=0.0
C CALC OVERALL HEAT TRANS COEFF
        E(II,5)=(E(II,1)*(E(II,2)+E(II,3)))/(E(II,1)+E(II,2)+E(II,3))
C APPLY CLEANLINESS FACTOR
        E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF AIR OVER A WIDE RANGE ABOUT THE NORMAL OPERATING
C CONDITIONS FOR THE AIR HEATER IS .244 BTU/LB F
        H(II,2)=PPHPT*G(II,1)*.244
        H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
        H(II,3)=H(II,2)/H(II,1)
C CALC NTU FOR 4 PASSES OF THE AIR HEATER TUBES AT AVG CONDITIONS
        H(II,4)=E(II,7)*412.4/H(II,2)
        ZZ=H(II,4)/(1.+H(II,4))
C CALC EPSILON (EPSIL) FOR THIS "AVG" 4 PASS BANK
        CALL EPS4(H(II,3),ZZ,EPSIL)
        EE=1.-EPSIL*H(II,3)
        EF=1.-EPSIL
        EG=(EE/EF)**4
C CALC THE EQUIVALENT EPSILON FOR FOUR 4 PASS BANKS IN SERIES
        H(II,5)=(EG-1.)/(EG-H(II,3))
C CALC TOTAL HEAT TRANS TO TUBES
        F(II,1)=H(II,5)*H(II,2)*(FGTFF-AIRTF)
C CALC HEAT LOSS TO WALLS, NOTE THAT HEAT LOSS COEFF HAS BEEN REDUCED
C TO 200 BTU/HR FT**2 BECAUSE THE AIRHEATER HAS EXTERNAL INSULATION
C AND THE FLUE GAS TEMPS ARE FAIRLY LOW.
        F(II,2)=200.*G(II,6)
C CALC TOTAL HEAT LOSS AND NEW TEMPS
        F(II,3)=F(II,1)+F(II,2)
        PDTA=F(II,1)/H(II,2)
        PDTGAS=F(II,3)/H(II,1)
        IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
        DTA=PDTA
        DTGAS=PDTGAS
        DTFS=F(II,1)/(E(II,1)*G(II,5))
C ASSUMES CARBON STEEL MATL AND .009 FT THK TUBES
        DTWALL=(F(II,1)*.009)/(27.*G(II,5))
        ITER=ITER+1
        IF(ITER.GE.10)GO TO 300
        GO TO 50
300     D(II,5)=FGTFF-PDTGAS
        D(II,3)=AIRTF+PDTA
        IF(AIRLKG.EQ.0.0)GO TO 301
        CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
        D(II,5)=T
301     CONTINUE

        RETURN
        END


C SUBROUTINE ARHT CALCULATES HEAT TRANSFER COEFFICIENTS AND DELTA
C T'S FOR THE ORIGINAL AIR HEATER
        SUBROUTINE ARHT(PPHPT,FGTFF,CF,II,PCTLKG,FGPPH)
        DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
        COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
        COMMON FGPPHI,CWFT,STTF,AIRTF
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```
      COMMON D,E,F,G,H,Q,R
C MAKE INITIAL GUESSES FOR FIRST ITERATION
      DTA=500.
      DTFA=50.
      DTGAS=200.
      DTWALL=10.
      ITER=1
      E(II,6)=CF
      D(II,1)=PPHPT
      D(II,2)=AIRTF
      D(II,4)=FGTFF
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH
C THE AIR HEATER HAS 90 TUBES, MAKING 8 PASSES EACH.
C THE TUBES ARE CARBON STEEL, 2" OD X 1.782" ID.
      G(II,1)=90.
      G(II,2)=8.
      G(II,3)=2.0
      G(II,4)=1.782
C TOTAL HEAT TRANSFER SURFACE AREA IS 2474 SQ FT AND THE SURFACE AREA
C OF THE AIR HEATER BOX IS 317.6 SQ FT.
      G(II,5)=2474.
      G(II,6)=317.6

50    TBULKA=AIRTF+(DTA/2.)
      TFILMA=TBULKA+(DTFA/2.)
C HEAT TRANS COEFF FOR AIR IN A 1.782" ID TUBE
      E(II,1)=(.1183+5.068E-5*TFILMA-1.49E-8*TFILMA**2)*
     &PPHPT**.8
      TGAVG=FGTFF-(DTGAS/2.)
      TPS=TBULKA+DTFA+DTWALL
      TFILMG=(TGAVG+TPS)/2.
      CALL FPPG(TFILMG,FPP)
C CONVECTION COEFF FOR FLUE GAS ACROSS TUBE BANK OF AIR HEATER
C THE FACTOR ".75" IS A CORRECTION FACTOR FOR THE HIGH % OF
C TUBE SURFACE IN BENDS WHERE THE FLOW IS PARALLEL RATHER THAN
C ACROSS AND FOR POOR FLOW AT ENTRANCE
      E(II,2)=(FPP*.1468*FGPPH**.61)*.75
      ALMTD=(DTGAS-DTA)/ALOG((FGTFF-AIRTF-DTA)/(FGTFF-
     &DTGAS-AIRTF))
C INTERTUBE RADIATION CAN BE NEGLIGABLE IF THE FLUE GAS TEMP IS
C BELOW ABOUT 800 F, CAVITY RAD IS TAKEN AS 0.0
      FACK=.188+.268*0.7-.0239*0.7**2
      URGP=(.005*ALMTD-3.)+(2.3E-4*ALMTD+7.)*(TPS/1500.)
      E(II,3)=URGP*FACK
      IF(E(II,3).LE.0.0)E(II,3)=0.0
      E(II,4)=0.0
C CALC OVERALL HEAT TRANS COEFF
      E(II,5)=(E(II,1)*(E(II,2)+E(II,3)))/(E(II,1)+E(II,2)+E(II,3))
C APPLY CLEANLINESS FACTOR
      E(II,7)=E(II,5)*CF
C SPECIFIC HEAT OF AIR OVER A WIDE RANGE ABOUT THE NORMAL OPERATING
C CONDITIONS FOR THE AIR HEATER IS .244 BTU/LB F
      H(II,2)=PPHPT*G(II,1)*.244
      H(II,1)=FGPPH*(A1+A2*TGAVG+A3*TGAVG**2)
      H(II,3)=H(II,2)/H(II,1)
C CALC NTU FOR 2 PASSES OF THE AIR HEATER TUBES AT AVG CONDITIONS
      H(II,4)=E(II,7)*618.5/H(II,2)
      ZZ=H(II,4)/(1.+H(II,4))
C CALC EPSILON (EPSIL) FOR THIS "AVG" 2 PASS BANK
      CALL EPS2(H(II,3),ZZ,EPSIL)
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      EE=1.-EPSIL*H(II,3)
      EF=1.-EPSIL
      EG=(EE/EF)**4
C CALC THE EQUIVALENT EPSILON FOR FOUR 2 PASS BANKS IN SERIES
      EPS4=(EG-1.)/(EG-H(II,3))
      H(II,5)=EPS4
C CALC TOTAL HEAT TRANS TO TUBES
      F(II,1)=H(II,5)*H(II,2)*(FGTFF-AIRTF)
C CALC HEAT LOSS TO WALLS, NOTE THAT HEAT LOSS COEFF HAS BEEN REDUCED
C TO 200 BTU/HR FT**2 BECAUSE THE AIRHEATER HAS EXTERNAL INSULATION
C AND THE FLUE GAS TEMPS ARE FAIRLY LOW.
      F(II,2)=200.*G(II,6)
C CALC TOTAL HEAT LOSS AND NEW TEMPS
      F(II,3)=F(II,1)+F(II,2)
      PDTA=F(II,1)/H(II,2)
      PDTGAS=F(II,3)/H(II,1)
      IF(ABS(DTGAS-PDTGAS).LE.2.)GO TO 300
      DTA=PDTA
      DTGAS=PDTGAS
      DTFS=F(II,1)/(E(II,1)*G(II,5))
C ASSUMES CARBON STEEL MATL AND .009 FT THK TUBES
      DTWALL=(F(II,1)*.009)/(27.*G(II,5))
      ITER=ITER+1
      IF(ITER.GE.10)GO TO 300
      GO TO 50
300   D(II,5)=FGTFF-PDTGAS
      D(II,3)=AIRTF+PDTA
      IF(AIRLKG.EQ.0.0)GO TO 301
      CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
      D(II,5)=T
301   CONTINUE

      RETURN
      END

C SUBROUTINE CAVITY CALCULATES HEAT LOSS FROM WALLS OF SECTIONS
C WITH NO HEAT TRANSFER TUBES
      SUBROUTINE CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R
      DO 10 N=1,5
      H(II,N)=0.0
10    CONTINUE

      DO 20 N=1,5
      G(II,N)=0.0
20    CONTINUE
      G(II,6)=WA
C TAKE WALL HEAT LOSS COEFF AS 700 BTU/HR FT**2 FOR ALL CAVITIES
      F(II,1)=0.0
      F(II,2)=700.*G(II,6)
      F(II,3)=F(II,2)
      AIRLKG=PCTLKG*FGPPHI
      F(II,4)=AIRLKG
      FGPPH=FGPPH+AIRLKG
      F(II,5)=FGPPH

      DO 40 N=1,7
      E(II,N)=0.0
40    CONTINUE
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      DO 50 N=1,3
      D(II,N)=0.0
50    CONTINUE
      D(II,4)=FGTFF
C CALC FLUE GAS TEMP AT OUTLET OF CAVITY
      D(II,5)=FGTFF-(F(II,2)/FGPPH*(A1+A2*FGTFF+A3*FGTFF**2))
      IF(AIRLKG.EQ.0.0)GO TO 301
      CALL ALEAK(D(II,5),AIRLKG,FGPPH,T)
      D(II,5)=T
301   CONTINUE

      RETURN
      END

C SUBROUTINE ELBO DEFINES THE GEOMETRY FOR THE 180 DEG ELBO
      SUBROUTINE ELBO(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=286.
      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END

C SUBROUTINE SCR0 DEFINES THE GEOMETRY FOR THE SCREEN AREA OF TS-1
C WITH THE HEAT TRANSFER TUBES REMOVED
      SUBROUTINE SCR0(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=54.
      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END

C SUBROUTINE TSX0 DEFINES THE GEOMETRY FOR EITHER TS-2 OR TS-3 WITH
C THE STEAM TUBES REMOVED
      SUBROUTINE TSX0(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=94.3
      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END

C SUBROUTINE INLT DEFINES THE GEOMETRY FOR THE INLET TRANSITION
      SUBROUTINE INLT(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=232.0
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      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END

C SUBROUTINE CMX0 DEFINES THE GEOMETRY FOR ONE THIRD OF A COOLING
C SECTION WITH THE TUBE BANK REMOVED
      SUBROUTINE CMX0(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=54.3
      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END

C SUBROUTINE CSX0 DEFINES THE GEOMETRY FOR AN EMPTY COOLING SECTION
C I.E. A COOLING SECTION BOX WITH NO TUBES
      SUBROUTINE CSX0(FGTFF,II,PCTLKG,FGPPH)
      DIMENSION D(30,7),E(30,7),F(30,7),G(30,7),H(30,7),Q(30),R(30)
      COMMON A1,A2,A3,B1,B2,B3,C1,C2,C3
      COMMON FGPPHI,CWTF,STTF,AIRTF
      COMMON D,E,F,G,H,Q,R

      WA=163.
      CALL CAVITY(FGTFF,WA,II,PCTLKG,FGPPH)

      RETURN
      END
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# VITA

Wesley T. McMinn was born on January 8, 1948 in Chattanooga, Tennessee, where he attended elementary and secondary schools, graduating from Brainerd Senior High School in the spring of 1966. In the fall of that year he entered Tennessee Technological University, from which he graduated in the spring of 1970 with a Bachelor of Science degree in Engineering Science. Later that year, he was employed by the Combustion Engineering Company, Nuclear Components Division in Chattanooga, Tennessee as a Contract Engineer. During the following seven years with C. E., he held a variety of positions in the Nuclear Engineering Department, earned a Professional Engineer's License (Tennessee), and earned a Master of Science degree in Engineering Administration from the University of Tennessee. In 1978, he moved to Tullahoma, Tennessee to take a position with The University of Tennessee Space Institute as a Senior Engineer on the Magnetohydrodynamics research project. He is currently the Manager of the Design Engineering Section of the Energy Conversion Research and Development Programs (UTSI). He resides in Tullahoma with his wife, the former Elizabeth Brook McDowell, and two daughters, Amy and Katherine.