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A KINETIC MODEL FOR TWO-PHASE FLOW IN HIGH TEMPERATURE EXHAUST GAS COOLERS

John M. Pelton and C. E. Willbanks
ARO, Inc.

June 1972

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**ENGINE TEST FACILITY
ARNOLD ENGINEERING DEVELOPMENT CENTER
AIR FORCE SYSTEMS COMMAND
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FOREWORD

The work reported herein was conducted at the request of the Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), Arnold Air Force Station, Tennessee. Data produced as a result of this research effort has been shared with the Federal Republic of Germany under Annex No. AF-66-G-7406 to the Mutual Weapons Development Plan Master Data Exchange Agreement between the governments of the United States and the Federal Republic of Germany.

The work involved analytical study and experimental testing conducted by ARO, Inc. (a subsidiary of Sverdrup & Parcel and Associates, Inc.), contract operator of the Arnold Engineering Development Center, AFSC, under contract F40600-72-C-0003. The test data were taken from tests conducted in the Propulsion Development Test Cell (T-1) spray cooler of the Engine Test Facility (ETF) under ARO Project Nos. RW0856, RW2116, and RW2216, and the manuscript was submitted for publication on May 16, 1972.

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This technical report has been reviewed and is approved.

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ABSTRACT

An analytical model was developed to describe the thermodynamic and fluid dynamic processes in an exhaust gas cooler employing liquid water injection. The model is based on the solution of the equations of conservation of species, momentum, and energy for the system and the equations for the exchange of these quantities between liquid and gaseous phases. These equations are programmed for solution on an IBM 360 computer. The predictions of the model are compared with measured data from a series of turbojet tests in the Propulsion Development Test Cell (T-1) spray cooler. The comparison showed that the model gave a good agreement with the measured pressure and liquid temperature at various points along the cooler. Parameters such as gas temperature and specific humidity which were not measured are discussed in terms of their relation to the overall cooler performance. From the results of the measurements and predictions, a physical description of the cooling process is presented. Based on the results of one of the tests, a possible method of reducing the pressure loss in a cooler is proposed.

CONTENTS

	<u>Page</u>
ABSTRACT	iii
NOMENCLATURE	vii
I. INTRODUCTION	1
II. EXHAUST GAS COOLING SYSTEM	
2.1 Configuration	2
2.2 Instrumentation	2
III. DEVELOPMENT OF THE ANALYTICAL MODEL	
3.1 Equation for the Conservation of Species for One Injection Station	4
3.2 Equation for the Conservation of Energy for One Injection Station	4
3.3 Equation for the Conservation of Momentum for One Injection Station	5
3.4 Equations for Multiple Injection Stations	6
3.5 Equations for the Exchange of Mass, Energy, and Momentum between Phases	8
3.6 Computer Solution of the Equations	11
IV. EVALUATION OF THE ANALYTICAL MODEL	13
V. CONCLUDING REMARKS	20
REFERENCES	22

APPENDIXES

I. ILLUSTRATIONS

Figures

1. Arrangement of Spray Banks in T-1 Cooler	
a. Section View Showing Internal Configuration	25
b. Internal View Showing Bank Configuration	26
c. Multinozzle Spray Heads, View Looking Upstream (Banks No. 4 through 11)	27
2. Schematic Showing Instrumentation Location	28
3. Predicted and Measured Exhaust Gas Cooler Pressure as a Function of Axial Location	
a. Run No. 36-13	29
b. Run No. 36-14	30
c. Run No. 36-18	31

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FiguresPage

4.	Predicted and Measured Liquid Temperature as a Function of Axial Location	
a.	Run No. 36-13	32
b.	Run No. 36-14	33
c.	Run No. 36-18	34
5.	Effect of Blockage on the Predicted Pressure in an Exhaust Gas Cooler	35
6.	Predicted Exhaust Gas Temperature for Various Amounts of Injected Cooling Water	36
7.	Predicted Relation between Exhaust Temperature and Specific Humidity for Three Cooling Water Flow Rates	
a.	Run No. 36-13	37
b.	Run No. 36-14	38
c.	Run No. 36-18	39
8.	Effect of the Difference in the Partial Pressures of the Liquid and Exhaust Gas Streams on the Specific Humidity for Run No. 36-13.	40
9.	Measured and Predicted Exhaust Gas Cooler Pressures for Approximately Constant Inlet Mass Flow Rate and Various Total Energy Levels.	41
10.	A Comparison of Two Runs to Evaluate the Influence of the Wall Spray Banks on the Predicted and Measured Cooler Pressure	42
11.	A Comparison of the Measured Cooler Pressure with Several Values of Calculated Pressure for Run 36-16 . . .	43

II. TABLES

I.	Inlet Engine and Cooler Conditions for Runs No. 36-13, 36-14, and 36-18	
a.	Measured and Calculated Exhaust Gas Parameters	44
b.	Cooling Water Flow Rate Conditions	45
II.	Inlet Engine and Cooler Conditions for Runs No. 36-14, 36-15, 36-16, and 35-44	
a.	Measured and Calculated Exhaust Gas Parameters	46
b.	Cooling Water Flow Rate Conditions.	47

	<u>Page</u>
III. A LISTING OF THE COMPUTER PROGRAM AND THE REQUIRED AUXILIARY EQUATIONS FOR AN EXHAUST GAS COOLER	48
IV. A LISTING OF A VARIATION OF THE COMPUTER PROGRAM FOR ZERO HARDWARE BLOCKAGE OF THE DUCT	66

NOMENCLATURE

A	Cross-sectional area of cooler as function of distance x along the cooler, ft ²
AA	Fraction of cooler not blocked by piping
A ₀	Cross-sectional area of cooler at x = 0, ft ²
C _D	Drag coefficient
c _ℓ	Specific heat of liquid water, Btu/lbm - °R
c _p	Specific heat, Btu/lbm - °R
C _v	Mass fraction of vapor, $C_v = \rho_v / (\rho_v + \rho_{nc})$
D	Particle diameter, ft
D _{ab}	Diffusion coefficient, ft ² /sec
e _d	Internal energy of droplet, Btu
f _ℓ	Ratio of liquid flow rate to noncondensable gas flow rate, $\dot{m}_\ell / \dot{m}_{nc}$
h	Enthalpy, Btu/lbm
$\bar{h}$	Heat transfer coefficient, Btu/ft ² -sec-°R
J	Dimensional constant, 778 ft-lbf/Btu
k	Thermal conductivity, Btu/ft-°R-sec
k _x	Mass transfer coefficient, lb-mole/ft ² ·sec
M	Molecular weight, lbm/lb-mole
M _d	Mass of droplet, lbm

$\dot{m}$	Mass flow rate, lbm/sec
Nu	Nusselt number for heat transfer
Nu _{ab}	Nusselt number for mass transfer
P	Pressure, lbf/ft ²
Pr	Prandtl number
R	Gas constant, ft-lbf/lbm-°R
Re	Reynolds number
R _U	Universal gas constant, ft-lbf/lb-mole-°R
Sc	Schmidt number
T	Temperature, °R
t	Time, sec
V	Velocity, ft/sec
x	Distance along cooler, ft
$\bar{x}$	Mole fraction
δ	Incremental distance, ft
μ	Dynamic viscosity, lbm/ft-sec
ρ	Density, lbm/ft ³
σ	Surface tension, lbf/ft
ω	Molar rate of evaporation, moles/sec

SUBSCRIPTS

A	Average
f	Film value
g	Gas phase—vapor plus noncondensable
i	Liquid originating at injection station i
l	Liquid phase
nc	Noncondensable
ref	Reference
s	Droplet surface

v Vapor
1, 2, 3, etc. Spraybank number

SUPERSCRIPT

m Molar

SECTION I INTRODUCTION

Testing of turbojet engines and rocket motors at simulated altitude in ground test facilities requires cooling of the high temperature exhaust gas to a relatively low temperature before the gas enters the exhaust gas pumping system. Cooling of the gases by water spray with direct heat and mass exchange between the water and the exhaust gas has been utilized in many test facilities. This method of cooling is often called spray cooling.

Many of the spray coolers used in the Engine Test Facility (ETF) at the Arnold Engineering Development Center (AEDC) receive exhaust gas from a rocket or turbojet engine. The cooling process reduces the temperature from approximately 4000°R (maximum temperature of a turbojet engine exhaust gas) to approximately 550°R. By means of an atomizing water spray, the exhaust gas is cooled and humidified. The cooling produces a temperature compatible with the ducting, control valves, and pumping system material limits. Water conservation is an important consideration in operation because of the large quantities of spray water required.

Previous work has developed computer models for spray coolers based on the assumption of a homogeneous two-phase flow with kinetic and thermodynamic equilibrium (Refs. 1, 2, and 3). The investigations contained in this report cover the development of a computer model of a spray cooling process that follows a typical liquid water droplet in the cooler ducting. No assumptions of kinetic or thermodynamic equilibrium between the gas and liquid are made. The model is then compared with measurements made in a spray cooler during operation. The approach is similar to that used by Shapiro (Ref. 4).

SECTION II EXHAUST GAS COOLING SYSTEM

2.1 CONFIGURATION

The configuration of the Propulsion Development Test Cell (T-1) spray cooler consists of a diverging conical inlet section followed by a constant-area duct to the end of the spray cooler (Fig. 1a, Appendix I). The cooling water is introduced through a group of nozzles

arranged in a series of banks, in which the first three banks of sprays consist of nozzles projecting a fan-type spray directed downstream along the wall to protect the ducting (Fig. 1b). The remaining banks are arranged in a wagon-wheel configuration with several spokes, each "spoke" containing several spray heads. Each spray head contains several fixed-geometry, conical spray nozzles (Figs. 1b and c) directed generally downstream. The water to each spray bank is supplied by a large header, and the flow rate to each spray bank is controlled by a valve between the header and the spray bank.

2.2 INSTRUMENTATION

Instrumentation was provided to measure flow rates and pressures of the exhaust gas stream entering the cooler, and the temperature and composition were calculated using the method of Ref. 5. Exhaust gas static pressure and liquid water temperature measurements were also made at five axial stations along the cooler. The temperature of the cooling water before injection was measured, and the flow rate was calculated from pressure measurements made across an orifice or at the control valve. The location of this instrumentation is shown in Fig. 2. Measurements taken by this instrumentation provided experimental correlation with the analytical results from the mathematical model.

The millivolt outputs from the thermocouples and strain-gage-type pressure transducers were recorded on either magnetic tape or by a photographically recording galvanometer-type oscillograph. The magnetic tape data were reduced on a digital computer, and the oscillograph data were reduced manually using electrical calibrations taken prior to testing.

SECTION III DEVELOPMENT OF THE ANALYTICAL MODEL

The analytical model is based on the equations of conservation of energy, momentum, and species for the exhaust gas cooler and the exchange of these quantities between phases. The model considers the behavior of a typical drop down the length of the cooler and calculates the changes in thermodynamic properties over a series of small incremental distances. The equations were programmed for solution on a digital computer.

Some of the key assumptions in the analysis are as follows:

1. All gases including water vapor obey the perfect gas equation of state.
2. The flow is steady and one dimensional.
3. The gas mixture at any section is homogeneous.
4. The droplets from each injection station are uniformly distributed over the cross-sectional area of the cooler.
5. The droplets are injected parallel to the gas flow and maintain this direction throughout the cooler. The influence of gravity on the droplets is considered negligible.
6. There is no aerodynamic breakup or agglomeration of the drops.
7. The drops injected at any injection station are uniform in size.
8. The maximum number of injection stations is nine, and their spacing is arbitrary.
9. The internal resistance of the drops to heat distribution is negligible, thus the temperature is uniform through the drop.
10. The drops injected at each injection station are accounted for separately.
11. Heat transfer and friction at the duct walls and piping are negligible.
12. Cross-sectional area of the cooler is a prescribed function of distance along the cooler.

3.1 EQUATION FOR THE CONSERVATION OF SPECIES FOR ONE INJECTION STATION

The conservation of species written for the exhaust gas and water (in both liquid and vapor form) over an incremental distance δx is

$$\dot{m}_v + \dot{m}_l + \dot{m}_{nc} = \dot{m}_v + \frac{d\dot{m}_v}{dx} \delta x + \dot{m}_l + \frac{d\dot{m}_l}{dx} \delta x + \dot{m}_{nc} + \frac{d\dot{m}_{nc}}{dx} \delta x \quad (1)$$

If the noncondensable flow rate is assumed constant and insoluble in water, Eq. (1) may be simplified:

$$\frac{d\dot{m}_v}{dx} + \frac{d\dot{m}_l}{dx} = 0 \quad (2)$$

The mass fraction of vapor may be written

$$C_v = \frac{\dot{m}_v}{\dot{m}_v + \dot{m}_{nc}} \quad (3)$$

and the mass fraction of noncondensable gas is

$$C_{nc} = \frac{\dot{m}_{nc}}{\dot{m}_v + \dot{m}_{nc}} = 1 - C_v \quad (4)$$

In addition, the specific humidity may be defined as

$$\frac{\dot{m}_v}{\dot{m}_{nc}} = \frac{C_v}{1 - C_v} \quad (5)$$

and the liquid water ratio (f_l) as

$$f_l = \frac{\dot{m}_l}{\dot{m}_{nc}} \quad (6)$$

3.2 EQUATION FOR THE CONSERVATION OF ENERGY FOR ONE INJECTION STATION

The energy equation is developed to equate the total energy of the exhaust gas and liquid water as they pass two planes an incremental distance (δx) apart.

$$\begin{aligned}
& \dot{m}_{nc} \left[h_{nc} + (V_{nc}^2/2) \right] + \dot{m}_v \left[h_v + (V_{nc}^2/2) \right] + \dot{m}_\ell \left[h_\ell + (V_\ell^2/2) \right] \\
&= \left[\dot{m}_{nc} + (d\dot{m}_{nc}/dx)\delta x \right] \left\{ h_{nc} + (dh_{nc}/dx)\delta x \right. \\
&\quad \left. + \left[V_{nc} + (dV_{nc}/dx)\delta x \right]^2/2 \right\} + \left[\dot{m}_v + (d\dot{m}_v/dx)\delta x \right] \left\{ h_v \right. \\
&\quad \left. + (dh_v/dx)\delta x + \left[V_{nc} + (dV_{nc}/dx)\delta x \right]^2/2 \right\} \\
&\quad + \left[\dot{m}_\ell + (d\dot{m}_\ell/dx)\delta x \right] \left\{ h_\ell + (dh_\ell/dx)\delta x \right. \\
&\quad \left. + \left[V_\ell + (dV_\ell/dx)\delta x \right]^2/2 \right\}
\end{aligned} \tag{7}$$

By expanding and simplifying the above and expressing the mass flow rates of the various components in terms of Eqs. (5) and (6), Eq. (7) becomes

$$\begin{aligned}
& \frac{dh_{nc}}{dx} + V_{nc} \frac{dV_{nc}}{dx} + \frac{C_v}{1 - C_v} \frac{dh_v}{dx} + V_{nc} \frac{dV_{nc}}{dx} + \left[\frac{h_v + (V_{nc}^2/2)}{(1 - C_v)^2} \right] \left[\frac{dC_v}{dx} \right] \\
&\quad + f_\ell \left(\frac{dh_\ell}{dx} + V_\ell \frac{dV_\ell}{dx} \right) + \left(h_\ell + \frac{V_\ell^2}{2} \right) \frac{df_\ell}{dx} = 0
\end{aligned} \tag{8}$$

3.3 EQUATION FOR THE CONSERVATION OF MOMENTUM FOR ONE INJECTION STATION

The momentum equation expressing the total momentum passing two planes (δx) apart is:

$$\begin{aligned}
& \dot{m}_{nc} V_{nc} + \dot{m}_v V_{nc} + \dot{m}_\ell V_\ell + PA = \left[\dot{m}_{nc} + (d\dot{m}_{nc}/dx)\delta x \right] \left[V_{nc} \right. \\
&\quad \left. + (dV_{nc}/dx)\delta x \right] + \left[\dot{m}_v + (d\dot{m}_v/dx)\delta x \right] \left[V_{nc} + (dV_{nc}/dx)\delta x \right] \\
&\quad + \left[\dot{m}_\ell + (d\dot{m}_\ell/dx)\delta x \right] \left[V_\ell + (dV_\ell/dx)\delta x \right] \\
&\quad + \left[P + (dP/dx)\delta x \right] \left[A + (dA/dx)\delta x \right]
\end{aligned} \tag{9}$$

Expanding and simplifying give

$$\begin{aligned} \dot{m}_{nc} (dV_{nc}/dx) + V_{nc} (d\dot{m}_{nc}/dx) + \dot{m}_v (dV_{nc}/dx) + V_{nc} (d\dot{m}_v/dx) \\ + \dot{m}_\ell (dV_\ell/dx) + V_\ell (d\dot{m}_\ell/dx) + P(dA/dx) + A(dP/dx) = 0 \end{aligned} \quad (10)$$

By dividing by $\dot{m}_{nc}$ and incorporating the specific humidity and liquid water ratio in terms of Eqs. (5) and (6), respectively, Eq. (10) may be expressed as

$$\begin{aligned} \frac{dV_{nc}}{dx} + \frac{C_v}{1 - C_v} \frac{dV_{nc}}{dx} + \frac{V_{nc}}{(1 - C_v)^2} \frac{dC_v}{dx} + f_\ell \frac{dV_\ell}{dx} + V_\ell \frac{df_\ell}{dx} + \frac{P}{\dot{m}_{nc}} \frac{dA}{dx} \\ + \frac{A}{\dot{m}_{nc}} \frac{dP}{dx} = 0 \end{aligned} \quad (11)$$

Multiplying by $V_{nc} (1 - C_v)^2$ and assuming that the change in area over the increment δx is negligible give

$$\begin{aligned} (1 - C_v) V_{nc} \frac{dV_{nc}}{dx} + V_{nc}^2 \frac{dC_v}{dx} + (1 - C_v)^2 V_{nc} \left[f_\ell \frac{dV_\ell}{dx} + V_\ell \frac{df_\ell}{dx} \right] \\ + \frac{(1 - C_v)^2}{\rho_{nc}} \frac{dP}{dx} = 0 \end{aligned} \quad (12)$$

3.4 EQUATIONS FOR MULTIPLE INJECTION STATIONS

Exhaust gas coolers similar to those shown in Fig. 1a consist of a series of spray banks or water injection stations, whereas the equations previously developed indicate that the water is injected uniformly at one station. The equations are easily expanded to include multiple injection stations by adding a term to describe the liquid injection conditions at each spray bank and the location of each bank. The previously developed equations (Eqs. (2), (8), and (12)) are modified as shown below. Equation (2) becomes

$$\frac{d\dot{m}_v}{dx} + \sum_n \frac{d\dot{m}_{\ell_i}}{dx} = 0 \quad (13)$$

Equation (8) becomes

$$\begin{aligned} \frac{dh_{nc}}{dx} + V_{nc} \frac{dv_{nc}}{dx} + \frac{C_v}{1 - C_v} \left(\frac{dh_v}{dx} + V_{nc} \frac{dV_{nc}}{dx} \right) + \left[\frac{h_v + (V_{nc})^2}{(1 - C_v)^2} \right] \left[\frac{dC_v}{dx} \right] \\ + \sum_n \left[f_{\ell_i} \left(\frac{dh_{\ell_i}}{dx} + V_{\ell_i} \frac{dV_{\ell_i}}{dx} \right) \right] + \sum_n \left[\left(h_{\ell_i} + \frac{V_{\ell_i}^2}{2} \right) \frac{df_{\ell_i}}{dx} \right] = 0 \end{aligned} \quad (14)$$

and Eq. (12) becomes

$$\begin{aligned} (1 - C_v) V_{nc} \frac{dV_{nc}}{dx} + V_{nc} \frac{dC_v}{dx} + (1 - C_v)^2 V_{nc} \sum_n \left[f_{\ell_i} \frac{dV_{\ell_i}}{dx} + V_{\ell_i} \frac{df_{\ell_i}}{dx} \right] \\ + \frac{(1 - C_v)^2}{\rho_{nc}} \frac{dP}{dx} = 0 \end{aligned} \quad (15)$$

The piping necessary for the spray banks in an exhaust gas cooler similar to the one shown in Fig. 1a can occupy a significant portion of the cross-sectional area of the duct. In the cooler of test cell T-1, the frontal area of the piping at each injection station is approximately 12 percent of the total cross-sectional area. Because of the large amount of liquid normally used for cooling and the blockage caused by water piping, the equations describing the cooling process must be further modified to incorporate terms necessary to account for the loss in momentum of the liquid that strikes this piping. This modification is made 0.25 ft upstream of each spray bank where the liquid properties from all previous stations are mass averaged to produce two new streams, one of which represents the liquid passing a spray bank without interference and a second stream which strikes the piping, loses its momentum, and is then reaccelerated. The fraction of liquid striking the piping is equal to the fraction of area occupied by the piping. Therefore, the new liquid properties passing the spray bank will be

$$f_{\ell_A} = (AA)(f_{\ell_1} + f_{\ell_2} + f_{\ell_3} + \dots) \quad (16)$$

$$V_{\ell_A} = \frac{V_{\ell_1} f_{\ell_1} + V_{\ell_2} f_{\ell_2} + V_{\ell_3} f_{\ell_3} + \dots}{\sum_n f_{\ell_i}} \quad (17)$$

$$T_{S_A} = \frac{T_{s_1} f_{\ell_1} + T_{s_2} f_{\ell_2} + T_{s_3} f_{\ell_3} + \dots}{\sum_n f_{\ell_i}} \quad (18)$$

$$D_A = \frac{D_1 f_{\ell_1} + D_2 f_{\ell_2} + D_3 f_{\ell_3} + \dots}{\sum_n f_{\ell_i}} \quad (19)$$

The properties of the liquid that strikes the piping will be

$$V_{\ell_B} = 1.0 \quad (20)$$

$$T_{S_B} = T_{S_A} \quad (21)$$

$$D_B = \frac{13\sigma}{\rho_g (V_{nc} - V_{\ell_B})^2} \quad (22)$$

The velocity (V_{ℓ_B}) is arbitrarily set at a small value but not zero because it appears in the denominator of several calculations; T_{S_B} is assumed equal to T_{S_A} since the system is adiabatic and no heat is lost to the piping. The diameter of the reaccelerated drop (D_B) is based on Eq. (12.6) of Ref. 6. The equation has been modified by neglecting the second term on the right side since it is almost negligible for the conditions encountered in this program. The final form of the equation for the drop diameter is basically a solution to the Weber number for a critical value of 13.

3.5 EQUATIONS FOR THE EXCHANGE OF MASS, ENERGY, AND MOMENTUM BETWEEN PHASES

It is now necessary to develop the equations to relate the transfer of mass, energy, and momentum between phases. Since the mass flow rate of the noncondensable portion of the exhaust stream is considered constant and negligibly soluble in water, the only exchange of mass occurs between the liquid water and vapor. The conservation of water was expressed earlier in Eq. (2) and is

$$\frac{d\dot{m}_v}{dx} + \sum_n \frac{d\dot{m}_{\ell_i}}{dx} = 0$$

Furthermore the mass transfer to or from a single drop may be developed from Eq. (21.2-26) of Ref. 7 which expresses the molar rate of evaporation as

$$\omega_{v_i}^{(m)} = k_{x_i} \pi D_i^2 \frac{\bar{x}_{vs_i} - \bar{x}_v}{1 - \bar{x}_{vs_i}} \quad (23)$$

By multiplying by the molecular weight of vapor (M_v), the results in terms of the mass rate of evaporation may be expressed as

$$\omega_{v_i}^{(m)} M_v = k_{x_i} \pi D_i^2 M_v \frac{\bar{x}_{vs_i} - \bar{x}_v}{1 - \bar{x}_{vs_i}} \quad (24)$$

Since the mass rate of evaporation is equal to the decrease in the mass of a drop per unit time, then

$$-\frac{dM_{d_i}}{dt} = k_{x_i} \pi D_i^2 M_v \frac{\bar{x}_{vs_i} - \bar{x}_v}{1 - \bar{x}_{vs_i}} \quad (25)$$

Since the equations developed will be solved for incremental distances (dx), the equation above will be more useful in terms of the distance (dx):

$$V_{\ell_i} \frac{dM_{d_i}}{dx} = k_{x_i} \pi D_i^2 M_v \frac{\bar{x}_v - \bar{x}_{vs_i}}{1 - \bar{x}_{vs_i}} \quad (26)$$

For a mixture of perfect gases, the molar concentration ($\bar{x}_{vs_i}$) can be expressed in terms of the pressure where

$$\bar{x}_{vs_i} = \frac{P_{vs_i}}{P} \quad (27)$$

is the mole fraction of vapor at the drop surface and P_{vs} is the vapor saturation pressure computed at the drop surface temperature. While this method of evaluating $\bar{x}_{vs}$ is exact only for zero mass transfer, it can be shown to give satisfactory results even at relatively high mass transfer rates. The mole fraction ($\bar{x}_v$) of the free stream is

$$\bar{x}_v = \frac{\dot{m}_v/M_v}{(\dot{m}_v/M_v) + (\dot{m}_{nc}/M_{nc})} = \frac{C_v/M_v}{(C_v/M_v) + (1 - C_v/M_{nc})} \quad (28)$$

The change in the total amount of liquid may be expressed in terms of the change for one drop and the number of drops

$$\frac{df_{\ell i}}{dx} = \frac{f_{\ell i}}{M_{d_i}} \frac{dM_{d_i}}{dx} \quad (29)$$

The transfer of energy between phases is related to the thermodynamic state of the exhaust gas stream and the liquid drops. Since the system is adiabatic and at a constant area over the distance (dx), any change in the gas stream will necessarily result in a change in the drops; therefore, the exchange of energy between phases will be expressed as a change in internal energy of a single drop, and this will then be related to the change in energy of all the liquid. The change in internal energy for a single drop over the distance (dx) is

$$V_{\ell i} \frac{de_{d_i}}{dx} = \bar{h} \pi D_i^2 (T_g - T_{s_i}) + h_{v_{s_i}} V_{\ell i} \frac{dM_{d_i}}{dx} \quad (30)$$

where the first term on the right expresses the convective heat transfer and the second term expresses the heat transfer accompanying the mass transfer and change in phase. From the known energy transfer for one drop, the total energy transfer may be expressed as

$$\frac{d}{dx} (h_{\ell i} \dot{m}_{\ell i}) = \frac{\dot{m}_{\ell i}}{M_{d_i}} \frac{de_{d_i}}{dx} \quad (31)$$

It is assumed that the resistance to heat distribution within the droplet is negligible compared with the resistance to heat transfer at the surface, that is, the temperature within the drop is uniform. Thus,

$$e_{d_i} = c_\ell (T_{s_i} - T_{ref}) \quad (32)$$

for a constant specific heat of liquid (c_ℓ).

The momentum transfer between phases will be expressed using Newton's Second Law where the force on the drop is due only to the droplet drag. Therefore, expressed in this manner,

$$V_{\ell_i} \frac{d}{dx} \left[M_{d_i} (V_{nc} - V_{\ell_i}) \right] = \rho_g \frac{\pi D_i^2}{4} \frac{C_{D_i} |V_{nc} - V_{\ell_i}| (V_{nc} - V_{\ell_i})}{2} \quad (33)$$

which may be simplified to

$$\begin{aligned} (V_{nc} - V_{\ell_i}) V_{\ell_i} \frac{dM_{d_i}}{dx} + M_{d_i} V_{\ell_i} \frac{dV_{\ell_i}}{dx} \\ = \rho_g \frac{\pi D_i^2}{4} \frac{C_{D_i} |V_{nc} - V_{\ell_i}| (V_{nc} - V_{\ell_i})}{2} \end{aligned} \quad (34)$$

where ρ_g is the density of the combined noncondensable gas and vapor in the stream.

The equation of state for the exhaust gas stream is

$$P = (\rho_{nc} + \rho_v) R_g T_{nc} \quad (35)$$

where

$$R_g = \left(\frac{1 - C_v}{M_{nc}} + \frac{C_v}{M_v} \right) R_u \quad (36)$$

3.6 COMPUTER SOLUTION OF THE EQUATIONS

The conditions in the cooler are determined by first computing the changes to the liquid and then incorporating these changes into the solution of the conservation equations for the complete system. The

computer solution is based on the modified Euler method. The changes in the liquid properties are calculated from the derivatives given in Eqs. (26), (29), (30), and (34) and a known step size (dx). These changes are calculated for each liquid stream. The sum of the changes in mass, energy, and momentum are then incorporated into Eqs. (13), (14), (15), and (35) to solve for the new gas properties. An iteration technique is used for the solution to the last three equations. The calculation procedure continues down the cooler until a point is reached 0.25 ft upstream of a spray bank. At this point, the liquid properties are averaged as discussed in Section 3.4 (Eqs. (16) through (22)). By using the gas properties last calculated and the liquid properties of stream "a" only, the calculation procedure discussed above is completed for one step (dx). At this point, stream "b" (the liquid that has impinged on the piping) is added to the calculation, and the changes to Eqs. (26), (29), (30), and (34) are calculated for the two streams separately. These changes in mass energy and momentum are included in this iterative solution to Eqs. (13), (14), (15), and (35). The calculation procedures continue until a new spray bank is reached and then these liquid properties are included in the calculation routine.

The step size (dx) is variable in this program. The initial value used is 0.0001 ft, but if convergence is achieved quickly (less than 3 iterations), the step size is increased for the next series of calculations. The step size will vary between 0.0001 and 0.01 ft depending on the number of iterations necessary for convergence in the previous set of calculations. A computer listing of the program is given in Appendix III.

The frequency of printout for the calculations may also be varied, but experience has shown that for most conditions printing the results every 0.25 ft is sufficient to see the changes in the exhaust gas cooler conditions.

Typical input for a computer run is shown in Tables Ia and b (Appendix II). Special note should be taken of the spray banks in which no water is injected ($f_l = 0$). The spray banks are included in the input because they will contribute blockage to the system and their location must be known. In each case, a fictitious velocity, temperature, and drop size is also included to prevent division by zero during the solution, but since these properties are also multiplied by f_l , they become zero and do not affect the final solution.

For conditions where two-phase flow exists, but the piping for injecting the liquid does not occupy a significant portion of the cross-sectional area, a variation of the analytical model may be used. This

variation involves only the solution of Eqs. (13), (14), (15), (26), (29), (30), (34), and (35) without the averaging of the liquid properties discussed in Section 3.4 (using Eqs. (16) through (22)). In addition, a drop size distribution may be simulated in this variation of the model by inputting the various drop diameters and their respective quantities ($f_{\ell i}$) as spray stations but with the stations at the maximum dx distance apart.

A computer listing for this model variation will be found in Appendix IV.

SECTION IV EVALUATION OF THE ANALYTICAL MODEL

An evaluation of the computer model was made by comparing the predicted exhaust gas pressure with measured data and also the predicted liquid water temperature with the value measured by an exposed junction thermocouple at several points in the exhaust gas cooler. In addition, the calculated exhaust gas temperature and the specific humidity are discussed to determine how these parameters are influenced by test conditions.

Six typical data points taken during the testing of a turbojet engine are used to evaluate the model. The input conditions for use in the computer program are shown in Tables I and II. Also shown in the tables are the inlet conditions to the spray cooler which were calculated using the method of Ref. 4 which has been included as a part of the model. These data show that, for the runs in Table I, the cooler inlet conditions are constant and the difference is in the spray banks that are in use, whereas the data in Table II show the cooling water parameters to be constant and the exhaust gas temperature and velocity to vary. Two other items of importance that should be noted are:

1. The cooling water flow rate from the individual "wagon-wheel" spray banks was kept constant, and only the number of spray banks was varied, and
2. The cooling water from the wall sprays of spray banks No. 2 and 3 were not normally included in the calculation, whereas the water from spray bank No. 1 was included.

The flow rate from the individual spray banks was kept constant to minimize the effects of variations in cooling water velocity, drop size, and distribution from the individual nozzles. The cooling water from spray

banks 2 and 3 was not included because it was believed that these wall sprays would not contribute significantly to the cooling of the exhaust gas. Calculations show that the pressure change in the ducting to the first wagon-wheel can generally be adequately described by assuming a one-dimensional isentropic flow with no cooling water present. The cooling water from spray bank No. 1 was included because liquid water must be present at the start of the computer program (i. e., $f_{\ell} \neq 0$),

and the difference between the isentropic value of pressure and that calculated using the initial spray bank was not significant.

The measured and calculated values of pressure as a function of distance along the cooler are shown in Fig. 3; the liquid water temperatures are shown in Fig. 4. The increase in pressure during the first 9 ft (the diverging portion of the ducting) followed by a drop in pressure for the constant area portion of the cooler is characteristic of nearly all runs. The initial rise in pressure is due primarily to the subsonic compression of the exhaust gas with very little acceleration of the cooling water except on the outer edges of flow. The abrupt decrease in pressure that follows occurs in the constant diameter section of the cooler where the wagon-wheel sprays are located. The drop in pressure in this section is caused by acceleration of the cooling water from the initial wagon-wheel spray bank and the loss in momentum of the previously injected liquid water as it strikes the piping and is then re-accelerated. This later loss in momentum is taken in account by the averaging of the liquid properties and the resultant use of Eqs. (34), (35), and (36). Although the area occupied by the internal water piping at each spray station is small (approximately 12 percent), it is sufficient to cause a drastic change in the pressure characteristics. The magnitude of the change caused by inclusion of this piping is best illustrated by assuming that in the constant diameter section the piping is removed but the water is still introduced uniformly over the cross section of the cooler at the various spray banks. Figure 5 shows the calculated cooler pressure for several blockages as well as the standard 12 percent used for the calculation of all data from the cooler in test cell T-1. All calculated pressures show good agreement initially, but then the pressure begins to increase for the case of zero blockage while decreasing rapidly for the remaining cases. This increase in static pressure is caused by the decrease in dynamic pressure while the total pressure of the exhaust stream remains essentially constant. The decrease in dynamic pressure is due primarily to the cooling of the exhaust gas stream. The decrease in pressure for the conditions with various amounts of blockage is due to the interference caused by the piping. The effect of blockage in a flow stream is well known and the above example illustrates its importance in a two-phase stream. The percentages in Fig. 5 cover the range of normal and extreme blockage conditions

and indicate not only the importance of including the blockage in the analytical model but also the importance of minimizing it wherever possible in cooler design.

The measured and predicted liquid temperatures for the data points in Table I are shown in Fig. 4. Three of the four measured values show good agreement with the predicted values, while the remaining value (the initial measurement) is always high. It is significant to note that the predicted liquid temperature for the first wagon-wheel spray bank at 9.2 ft rises approximately 70°R in approximately 6 in. and then levels off at an almost constant value even when additional spray banks are used. This rapid rise in temperature is due to the fact that initially the liquid water temperature is low (536°R) and its vapor pressure at the drop surface is also low. The low vapor pressure of the drop combined with the low vapor partial pressure in the gas stream results in a low mass transfer rate, while the large difference in gas and liquid temperatures gives a high heat transfer rate and a rapid rise in the temperature of the liquid with little evaporation. As the liquid temperature begins to rise, the difference between the partial pressure of the drop and gas stream increases, and the mass transfer (or evaporation from the drop) increases. This increase in mass transfer continues until a liquid temperature is approached where heat transfer to the drop is almost completely used for the evaporation of water. The final temperature approached by the liquid is its adiabatic or wet bulb saturation temperature. As additional cooling water is added through the use of additional spray banks this process is repeated, but the rate of liquid temperature rise will decrease because of the smaller temperature difference between gas and liquid and also because the hotter liquid must also be cooled. The cooling of the liquid does not become important until there is a very large amount of "hot" liquid present. The fact that the cooler liquid temperature does rise very rapidly keeps the overall cooling process from becoming extremely inefficient due to alternately heating and cooling the liquid in the stream.

The previously mentioned thermocouple located at 9.2 ft always reads high. The high reading is believed to be caused by the location of the thermocouple at the edge of the diverging section where the fan-type spray will leave a liquid deficient region near the thermocouple. Since the smaller liquid quantity is surrounded by a large amount of hot exhaust gas, the heat transfer to the liquid will be abnormally high and thus cause the liquid temperature to rise to a value higher than is predicted by the computer model which assumes a uniform liquid distribution over the cross section of the cooler.

The validity of the computer model is best determined by comparing the predicted and measured values of static pressure, gas temperature, and specific humidity. A comparison using the static pressure has already been made, but measurements of the later two quantities have not been made because of the difficulties inherent in a two-phase stream. The exhaust gas temperature is important because of the effect on pumping machinery capabilities. As the temperature of the exhaust gas entering the machines is increased, the maximum mass flow rate that can be pumped at a constant pressure decreases; or, stated another way, for a given mass flow rate of exhaust gas, the minimum upstream pressure increases as the temperature of the exhaust gas entering the machinery increases. Therefore, it is desirable to cool the exhaust gas as much as possible. The specific humidity, like the gas temperature, places a lower limit on the pressure capabilities of the exhaust machinery. As the specific humidity increases, the minimum upstream pressure at the test cell also increases because this additional vapor is additional mass that must be removed. Therefore, the optimum condition would appear to consist of the lowest exhaust gas temperature and specific humidity. The problem is that the temperature normally decreases at the expense of an increase in humidity for spray coolers like those in ETF unless very large quantities of water are injected. Since the overall process of reducing the temperature is normally by evaporation of cooling water, the specific humidity for the process increases as the temperature decreases.

The predicted gas temperatures along the length of the cooler is shown in Fig. 6 for the three data points under discussion. The decrease in temperature follows the same path for the three runs as long as the same spray banks are used. As expected the lowest gas temperature occurs for the data point using the most cooling water (Run No. 36-13), while the highest temperature is predicted with the least amount of cooling water (Run No. 36-13). The temperature curve shows a distinct change occurring at approximately 9 ft. Although the data indicated that the pressure change at this point could be treated as a subsonic compression of a gas with no mass transfer, this is not the case with the predicted cooling curve. If the gas temperature followed a subsonic compression process with no mass transfer, the predicted temperature should be at some value greater than the cooler inlet value of approximately 3570°R. If this were a subsonic compression the wall-type sprays would probably be spraying directly along the wall and the thermocouple at 9.2 ft should be reading approximately gas temperature and the first wagon-wheel spray bank should be surrounded by the high temperature gas flow. As noted earlier, the thermocouple at the end of the divergent section indicates a measured temperature higher than the predicted liquid but certainly not an exhaust gas temperature. Since

the thermocouple is measuring a liquid temperature (Refs. 8 and 10), there has obviously been heating of the water indicating that the temperature characteristics cannot be described by an isentropic compression. Therefore, some cooling and mass transfer has taken place in the divergent section of the cooler, and the predicted curve probably has the correct shape, but it is not possible to know if the temperature is absolutely correct. The remainder of the predicted exhaust gas temperature curve is typical—a rapid decrease in temperature while there is a large temperature difference between gas and liquid followed by a decreasing rate of cooling near the exit as the temperature difference decreases. The point where the cooling curves separate is the location of the next spray bank being used. The differences noted at the exit indicate the magnitude of temperature change that can be expected by using additional spray banks for the conditions of these tests.

Since one of the objects of the cooling process is to get the lowest value of exhaust gas temperature at the lowest specific humidity, the temperature as a function of the specific humidity is presented for the three data points in Fig. 7 to show how the cooling takes place. The two points immediately obvious and important to the model description are:

1. The overall process is one of humidification and not two separate processes, i. e., one of humidification followed by dehumidification, and
2. There are short periods of dehumidification downstream of each active injection station but quickly followed by a resumption of the evaporative process.

The normal method of visualizing the cooling process in an exhaust gas spray cooler is to picture first a short section of cooler in which sufficient water is present to provide saturation conditions. This water is injected into the stream and is immediately vaporized because of the large temperature difference between liquid and exhaust gas. This process involves transferring sufficient heat from the gas stream to vaporize the water. After the gas stream becomes saturated with respect to the cooling water that has been injected, any further cooling water serves to dehumidify the gas stream. This dehumidification process is generally imagined to take place very slowly when compared with the vaporization process. As shown in Fig. 7, the cooling process does not appear to follow the model described above; instead the process appears to be one of almost continuous evaporation with a few periods of slight dehumidification. Which process is correct becomes very important in the understanding and design of spray coolers. The first process

actually describes what takes place in an infinitely long cooler where only small amounts of water are injected, and this water is allowed to reach temperature and velocity equilibrium before any more water is injected. In this way, saturation may be achieved but with no excess liquid water present. The second process describes a nonequilibrium process in terms of liquid and vapor temperatures, velocity, and concentrations but is the actual process in an exhaust gas cooler.

When liquid water is injected into the gas stream, initially the liquid is at a low temperature and partial pressure, whereas the gas stream has a relatively high temperature but low vapor pressure due to the small amount of vapor in the stream (generally only the water formed during the combustion process). Since the evaporation process is controlled by the vapor concentration difference (see Eq. (16)), the mass transfer will initially be very low, but because of the large temperature differences (on the order of 3000°R), the heat transfer rate will be very high. With the high heat transfer rate, the liquid temperature rises very rapidly, and the partial pressure difference between the liquid and gas stream increases, causing a rise in the mass transfer rate. This process of rising liquid temperature and mass transfer rate continues until a liquid temperature is approached where the heat transferred into the liquid is used almost completely for vaporization. The temperature that is approached by the liquid is the adiabatic or wet-bulb saturation temperature, but although this temperature is nearly achieved, the evaporation of the liquid continues because there still exists a partial pressure difference between the droplet surface and gas stream to provide the mass transfer driving force and a temperature difference to supply heat for vaporization. This process will continue until the partial pressure and temperature difference disappears.

This process discussed above is basically for one spray bank, whereas the normal cooler operation uses several banks. What happens when fresh cooling water (from a downstream spray bank) is injected into the gas stream can be divided into two processes. These occur:

1. When the vapor pressure of the fresh liquid is greater than the vapor pressure of the exhaust stream, and
2. When the vapor pressure of the fresh liquid is less than the vapor pressure of the gas stream.

Both of these conditions are shown in Fig. 7c. The first condition occurs at the entrance to the cooler when the exhaust gas contains very little vapor (the partial pressure is almost negligible) and water is

injected with a partial pressure of approximately 64 psf. In this instance, evaporation begins immediately and continues the length of the cooler. The second case occurs when the gas temperature has reached 2500°R (at spray bank No. 4). The partial pressure of the gas stream is now 215 psf which is above that of the incoming water and should result in the gas stream being dehumidified. This is in fact what happens, but because the dehumidification takes place over such a short distance, the decrease in specific humidity does not show up. The dehumidification process is much clearer at spray bank No. 5 when the gas temperature has reached 1600°R, and the specific humidity decreases from 0.43 to 0.41 before beginning to increase again. Another way of picturing the process is shown in Fig. 8 where the partial pressure difference between the liquid and vapor is shown as a function of the specific humidity. When the pressure difference is positive, evaporation takes place, and when it is negative, dehumidification takes place. In this figure, the liquid from spray bank No. 1 is shown to be evaporating from the start, whereas the liquid from spray bank No. 4 initially causes dehumidification until the pressure difference reaches zero. Then the process for spray bank No. 4 becomes evaporative, and the specific humidity begins to increase again. The same thing will happen to the spray banks downstream as shown in Fig. 8 where dehumidification takes place immediately downstream of the injection station. The specific humidity has a greater decrease for each succeeding spray bank because the partial pressure difference is initially greater.

Three additional runs are included in this discussion to show the agreement between the model and the experimental data and also to point out some possible effects of the two wall spray banks in the diverging section which are normally omitted from the calculation procedure. The measured engine inlet parameters and the calculated cooler inlet parameters are shown in Table II, and the pressure as a function of distance data is shown in Fig. 9. These data show good agreement between theory and experiment at the cooler exit, but for Run 36-16, the agreement at the exit from the diverging section (9.2 ft) is not good. Comparing the measured and predicted pressure for the three runs shows that the agreement is good for Run 36-14, but becomes progressively poorer for Runs 36-15 and 36-16. From Table II, the test parameters, including the cooling water, are seen to be the same with only the engine fuel flow rate changing. Thus, there is poorer agreement between the model and data as the fuel flow rate decreases.

A possible explanation for this lies in the interaction between wall spray banks 2 and 3 and the relatively low velocity gas stream. At the interface between the liquid and gas streams, a portion of the hot exhaust gas is cooled by flowing radially through the wall sprays to the

area between the interface and the duct wall while the remainder undergoes essentially an isentropic compression as it flows down the duct. As the Mach numbers of the two streams decrease, their static pressure increases, and for the gas flowing through the sprays, the total pressure will also increase because of the evaporation of water. The resultant effect is to increase the static pressure above that predicted by the model because of the cooling of the radially flowing gas.

To show the effect of the wall spray banks on the agreement between measured and predicted pressure, a run (35-44, Table II) was chosen with similar cooler inlet conditions but without spray banks 2 and 3 operating. The predicted and measured pressure for the first 9.2 ft is shown in Fig. 10 for Runs 35-44 and 36-16. The agreement between predicted and measured pressure for Run 35-44 is good, indicating that the wall sprays are at least part of the problem.

A second interesting point about Run 36-16 is that the measured pressure is higher than the total pressure calculated by assuming either an isentropic expansion or a two-phase cooling process, as shown in Fig. 11. To achieve this measured pressure, mass must be added to the stream but under conditions where there is not significant loss in momentum due to the added mass being accelerated. This condition could probably be achieved by the gases flowing radially through the water stream and adding vapor to the gas stream.

The condition of abnormally high static pressure is not usually encountered because the wall sprays are not used for conditions such as encountered in Run 36-16 where the cooler inlet velocity and temperature are low. For those runs where the sprays are used, such as 36-14, the total energy level of the stream masks any influence of the wall sprays.

If this radial mass flow through the wall sprays is the reason for the extremely high static pressure at the end of the diverging section, this technique might be a useful way of supplementing the exhaust pumping machinery to achieve a lower test cell pressure.

SECTION V CONCLUDING REMARKS

An analytical model was developed to describe the process carried on by an exhaust gas spray cooler. The model consisted of the computer

solution to the equations of conservation of species, momentum, and energy and the exchange of these quantities between the gas and liquid phase.

The model was compared with data from turbojet tests conducted in Propulsion Development Test Cell (T-1). The range of spray cooler inlet conditions was as follows:

$$\dot{m}_{nc} = 148 \text{ to } 155 \text{ lbm/sec}$$

$$T_{nc} = 1066 \text{ to } 2568^\circ\text{R}$$

$$P_i = 711 \text{ to } 909 \text{ psf}$$

$$V_{nc} = 402 \text{ to } 1147 \text{ ft/sec}$$

$$f_l = 2.5 \text{ to } 3.8$$

$$T_s = 536^\circ\text{R}$$

The static pressure and liquid temperature were measured at six positions along the length of the cooler, and these pressures and temperatures were compared with similar values predicted by the model. The measured static pressure at the cooler exit agreed with the predicted value within 4 percent or less, and the measured liquid temperature agreed within 1 percent of the predicted value. With the exception of the pressure at the entrance to the cylindrical section, the agreement between model and measurement for the other data was at least this good. The predicted values of gas temperature and specific humidity are discussed, but measured values of these quantities were not available for comparison because of the lack of adequate instrumentation. Successful instrumentation was not available for making this type of measurements in a typical exhaust gas cooler stream with liquid-to-gas mass ratios on the order of 2 to 1 or greater.

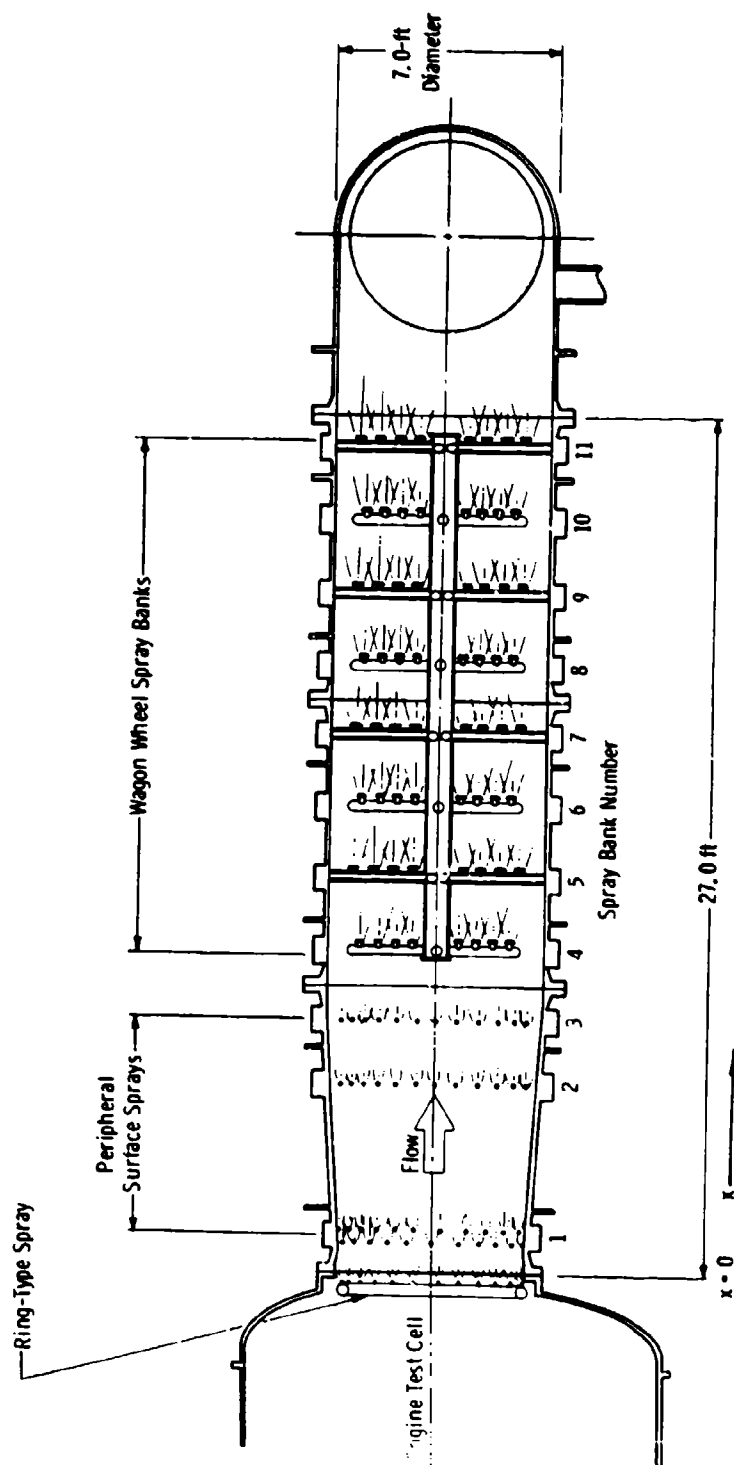
Static pressure measurements at the exit to the diverging section of the cooler gave abnormally high results for one run based on the predicted value from the model. A possible explanation for these data based on a separated recirculating flow in this area was postulated. Additional data for verifying this were not available. A possible method for improving spray cooler performance based on this postulate was also mentioned.

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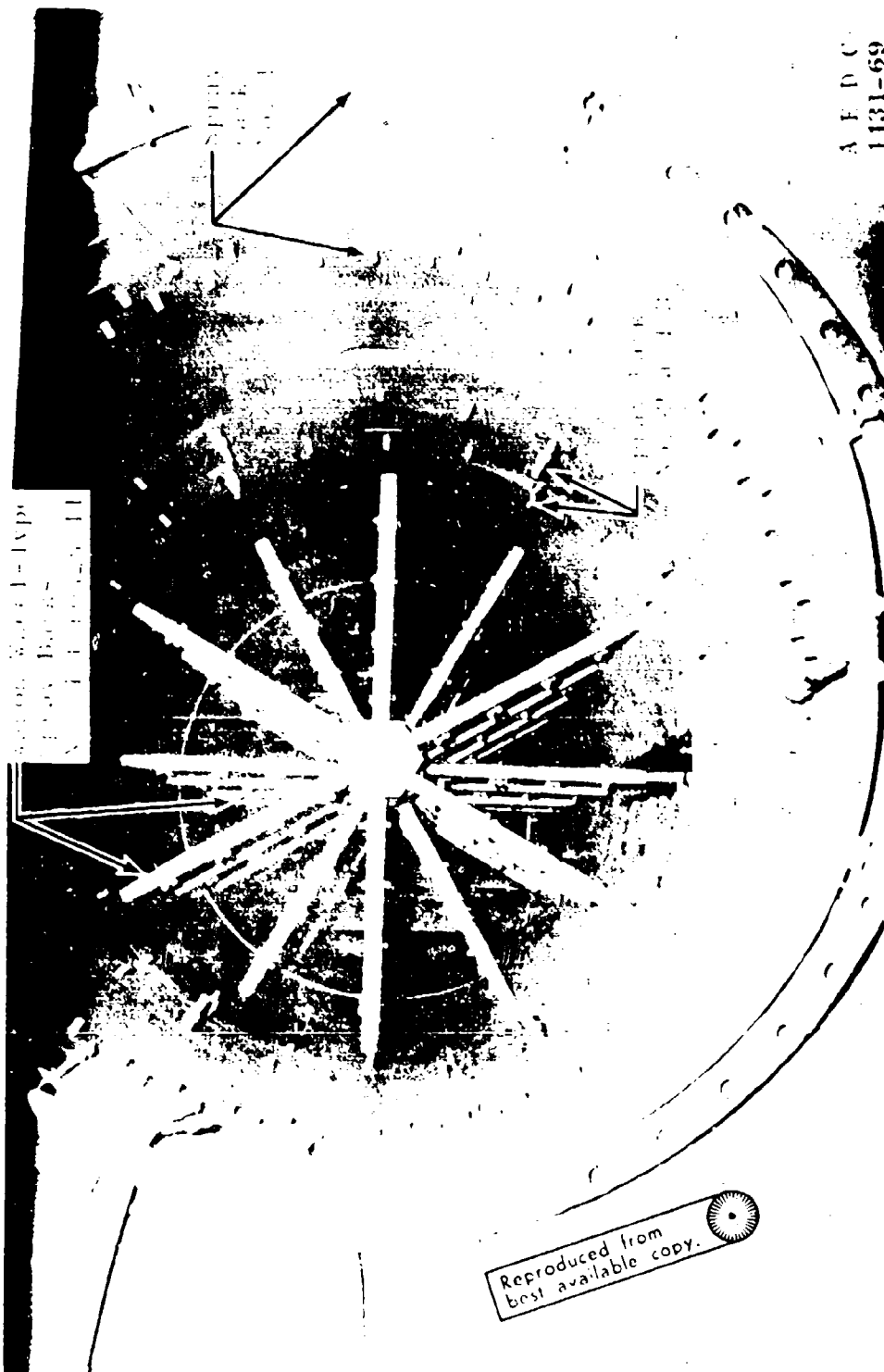
APPENDIXES

- I. ILLUSTRATIONS**
- II. TABLES**
- III. A LISTING OF THE COMPUTER PROGRAM
AND THE REQUIRED AUXILIARY EQUATIONS
FOR AN EXHAUST GAS COOLER**
- IV. A LISTING OF A VARIATION OF THE
COMPUTER PROGRAM FOR HARDWARE
BLOCKAGE OF THE DUCT**



a. Section View Showing Internal Configuration
Fig. 1 Arrangement of Spray Banks in T-1 Cooler

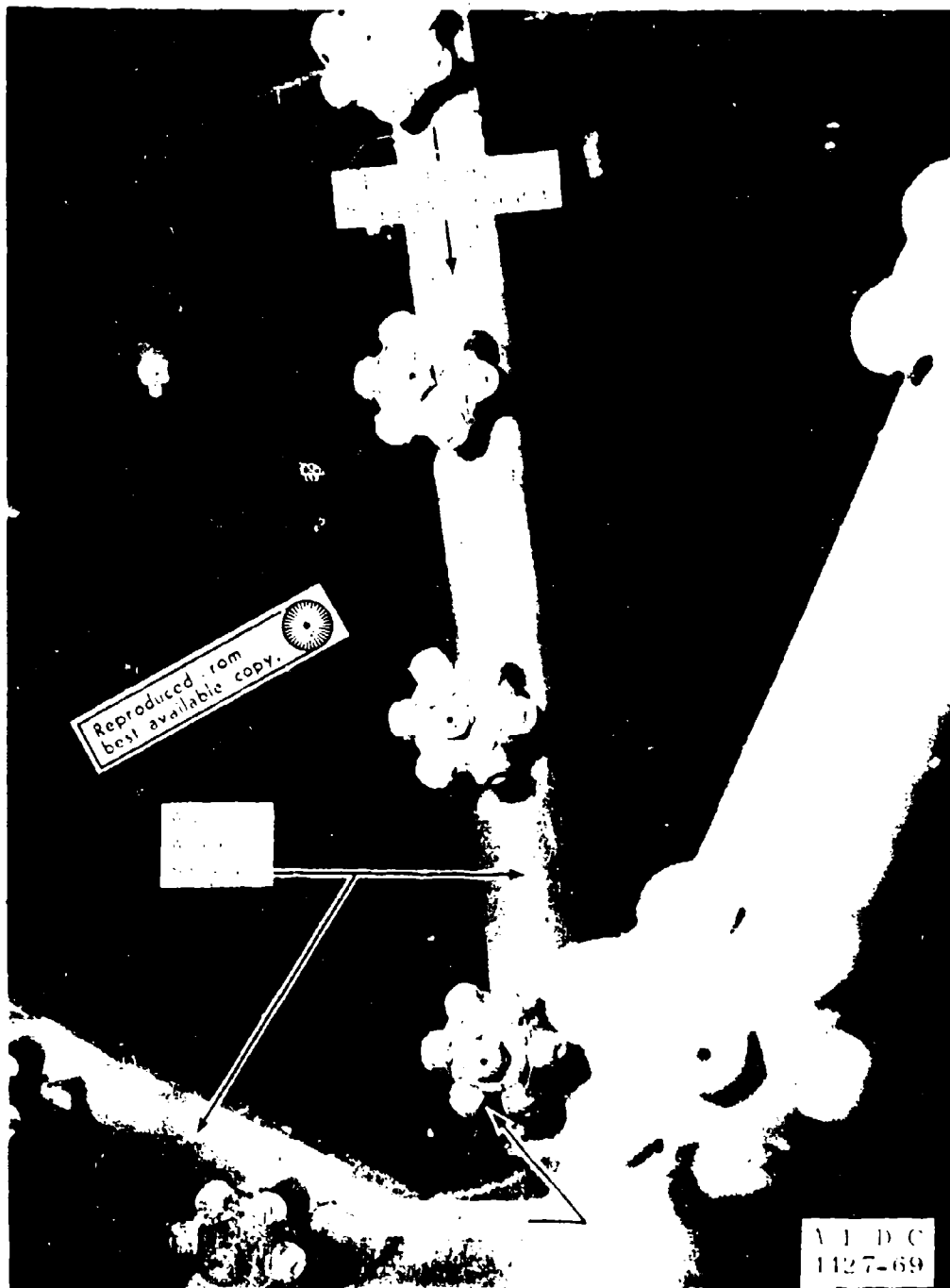
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b. Internal View Showing Bank Configuration
Fig. 1 Continued

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2. Multispectral Survey of the Area Looking North (Banks No. 4 through 11)
Fig. 1. Contour Map

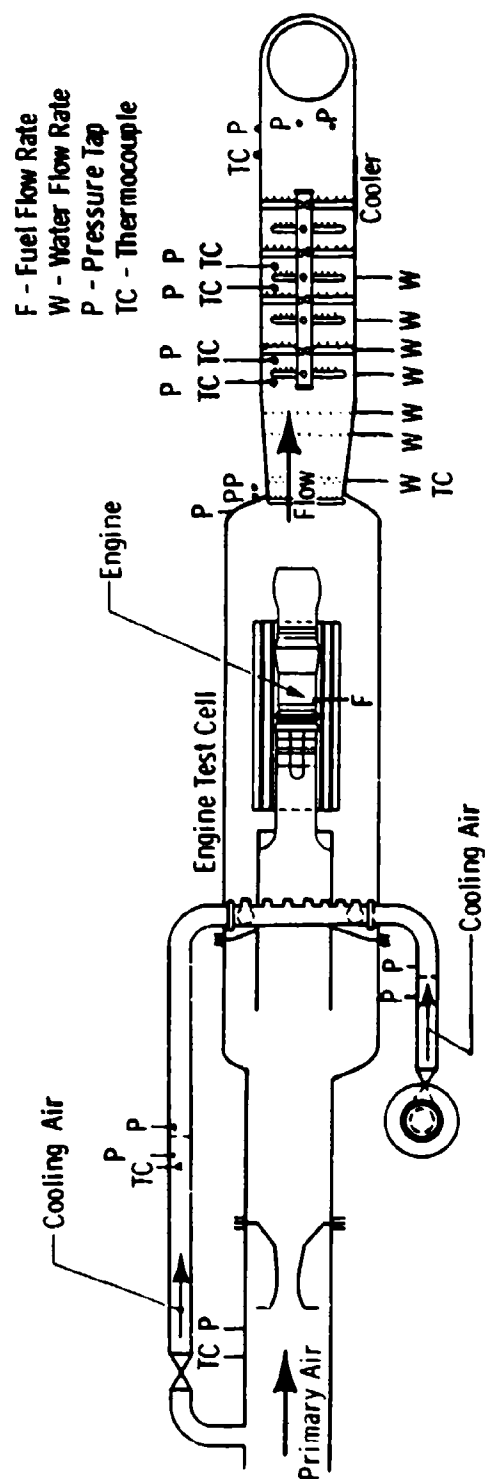
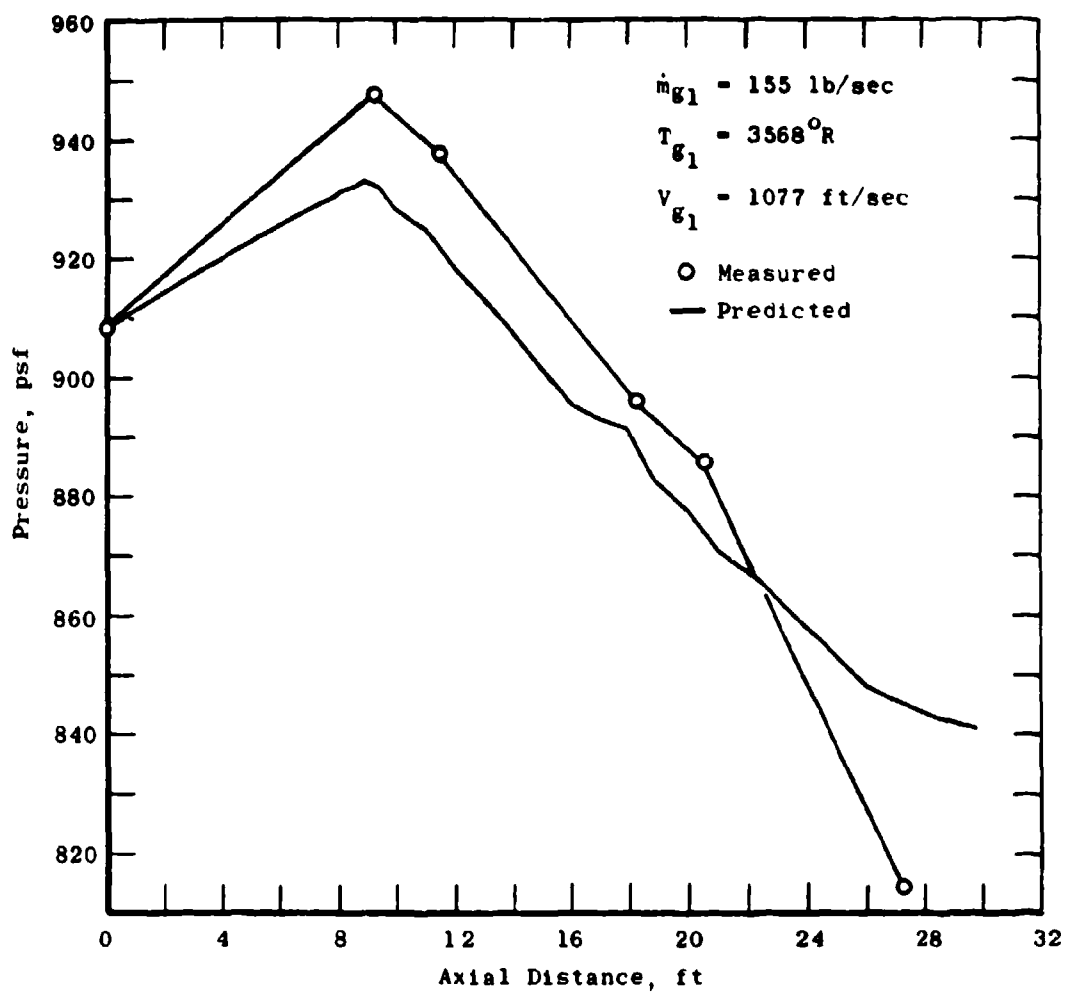
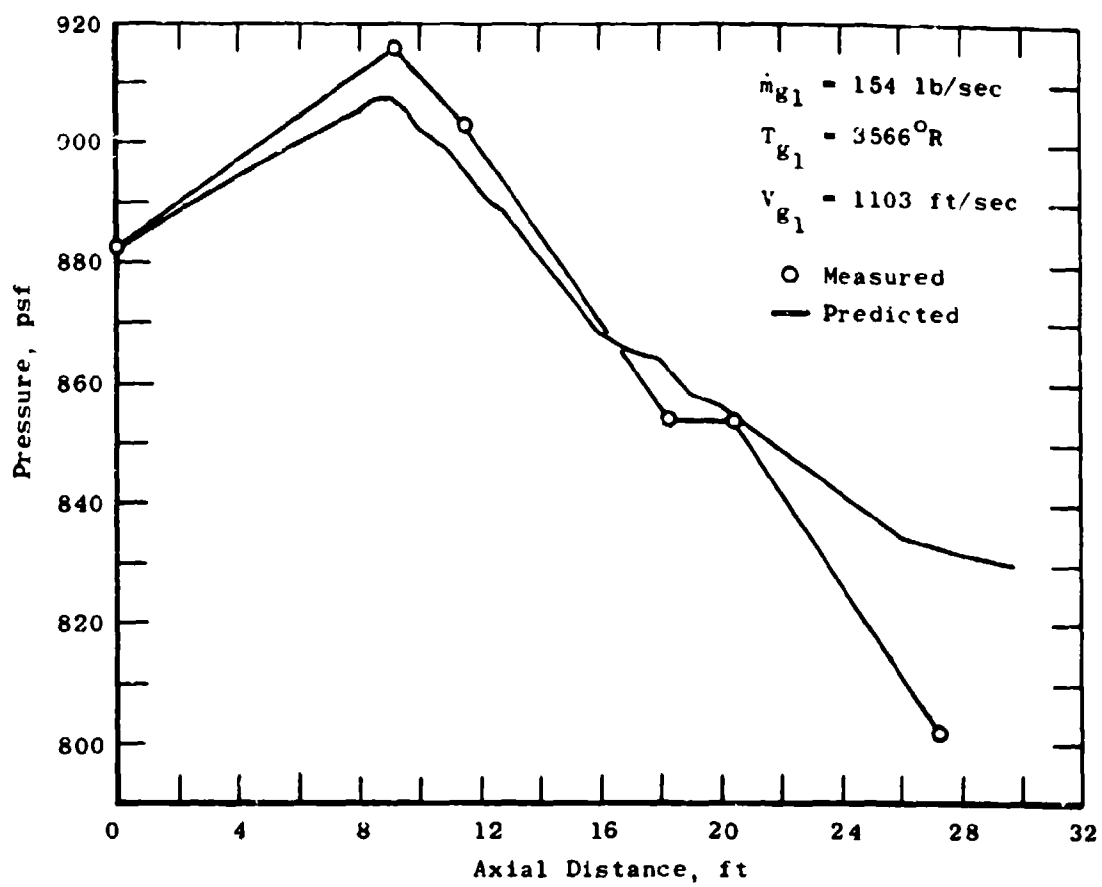


Fig. 2 Schematic Showing Instrumentation Location

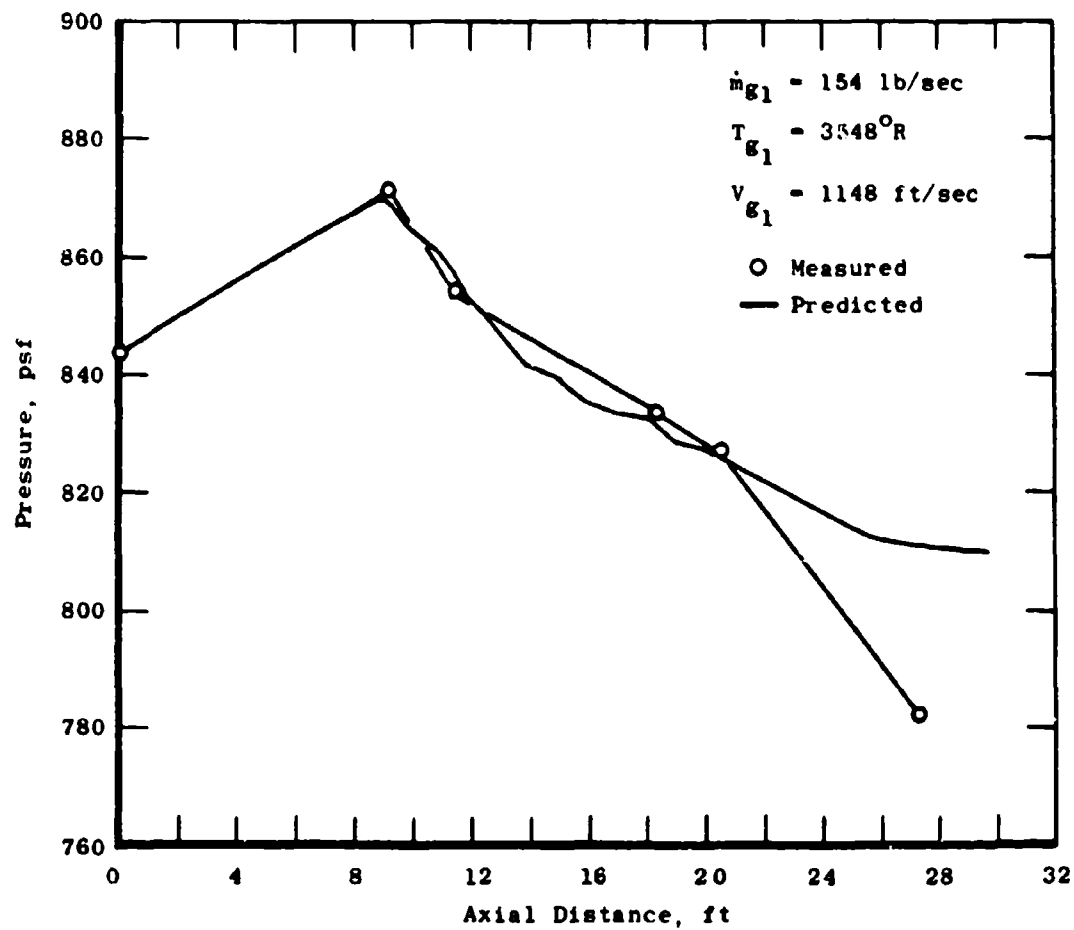


a. Run No. 36-13

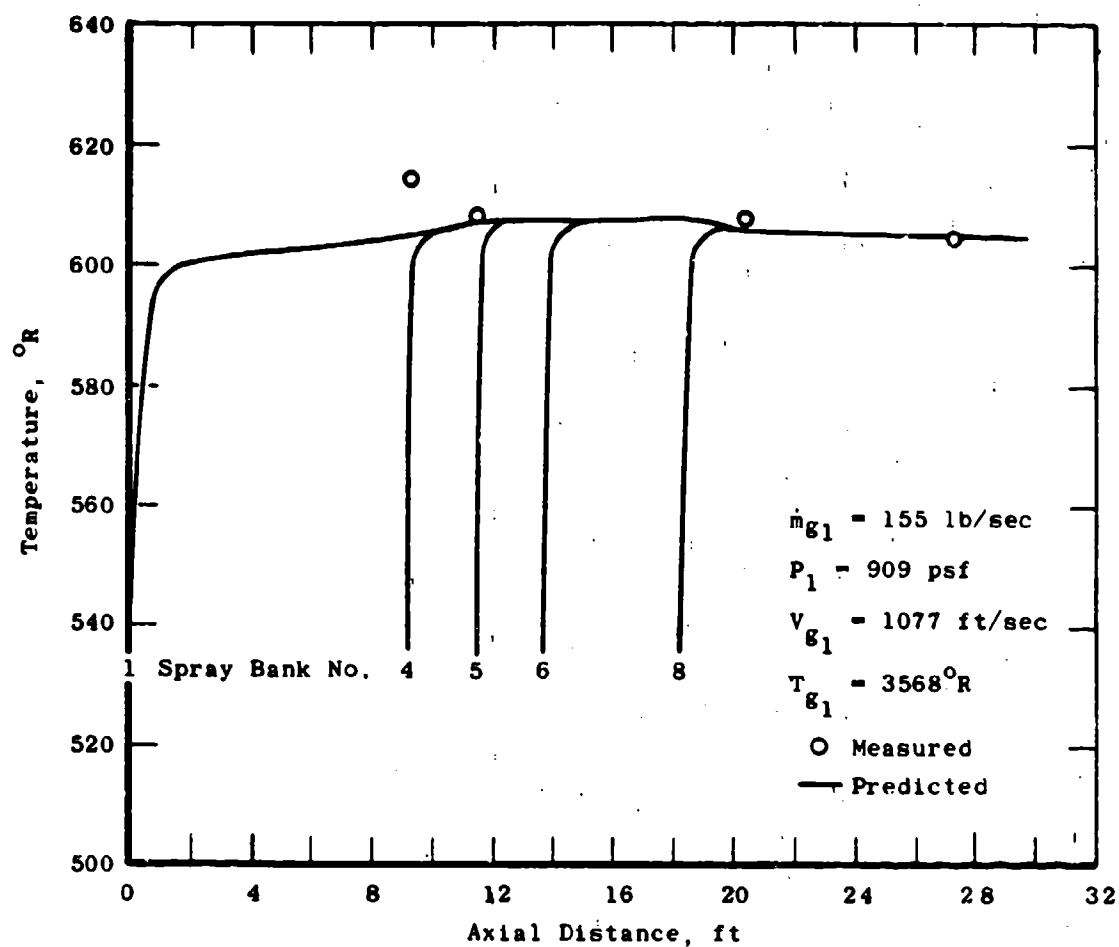
Fig. 3 Predicted and Measured Exhaust Gas Cooler Pressure as a Function of Axial Location



b. Run No. 36-14
Fig. 3 Continued

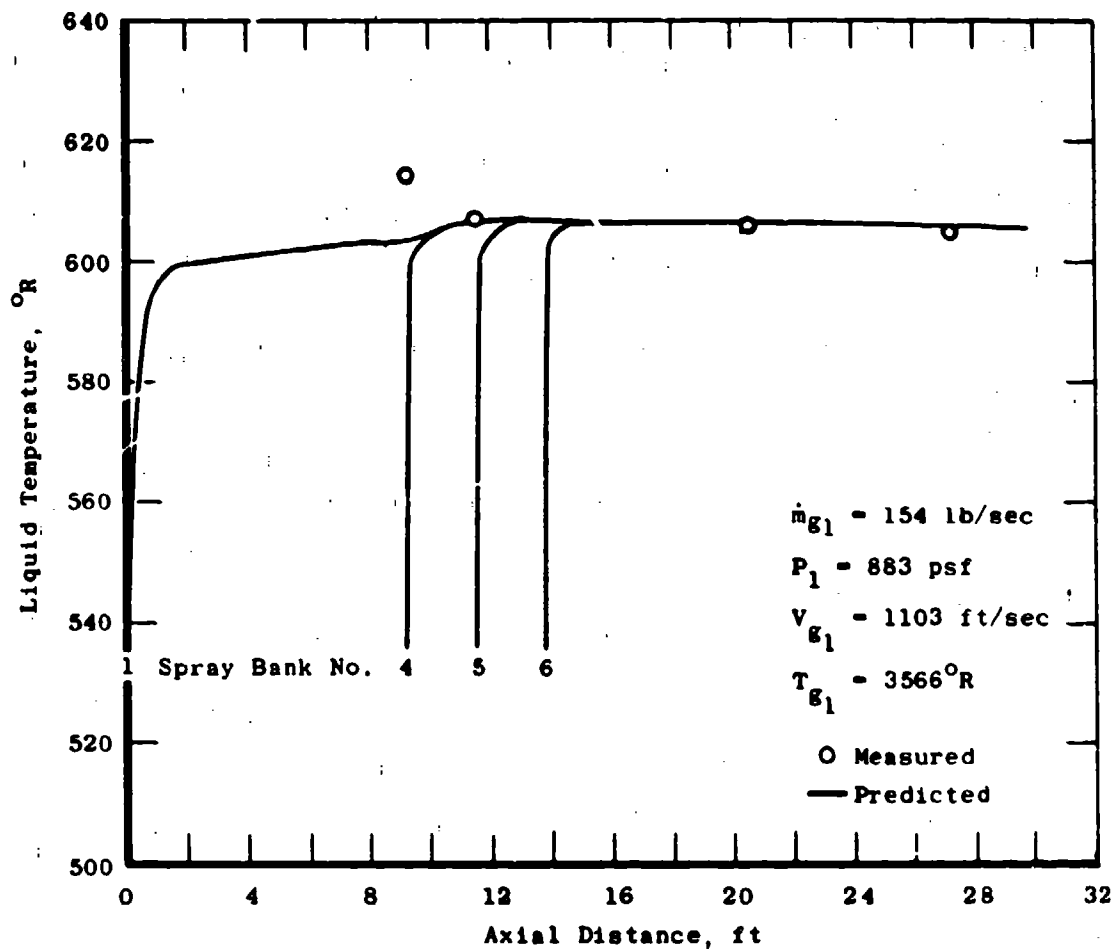


c. Run No. 36-18
Fig. 3 Concluded

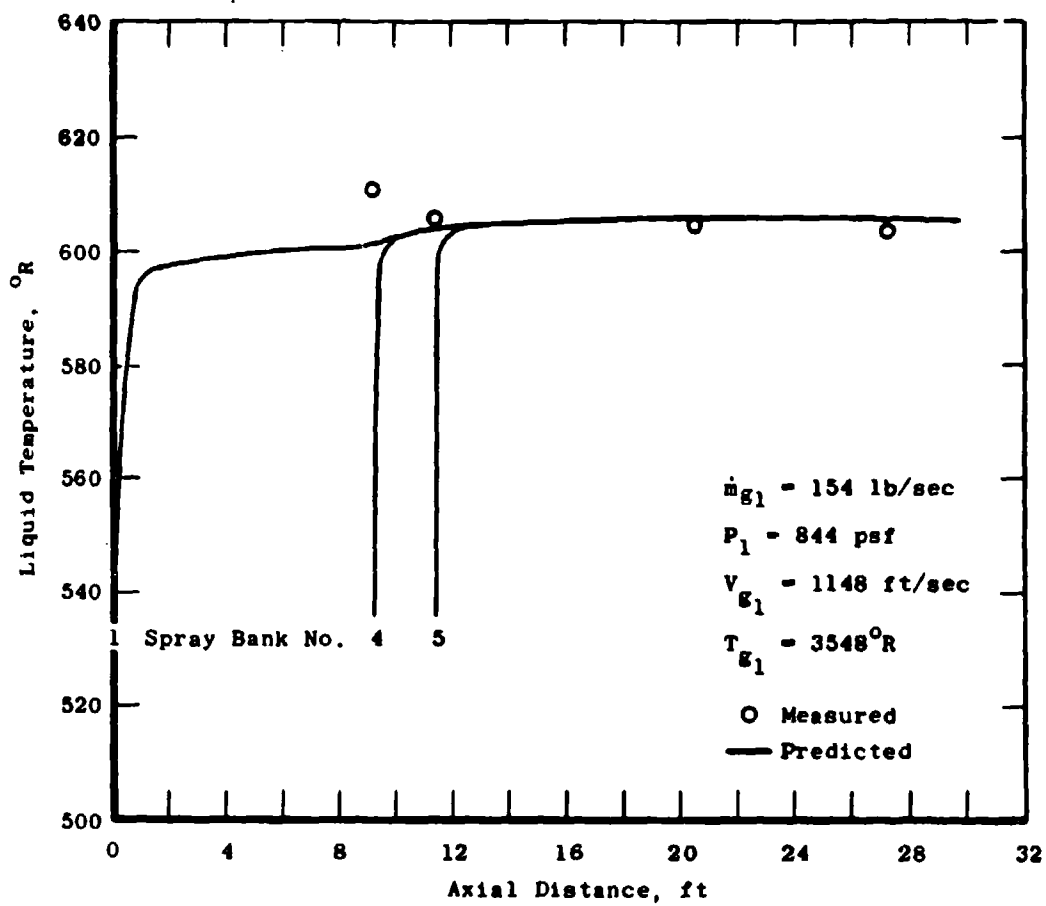


a. Run No. 36-13

Fig. 4 Predicted and Measured Liquid Temperature as a Function of Axial Location



i. Run No. 36-14
Fig. 4 Continued



c. Run No. 36-18
Fig. 4 Concluded

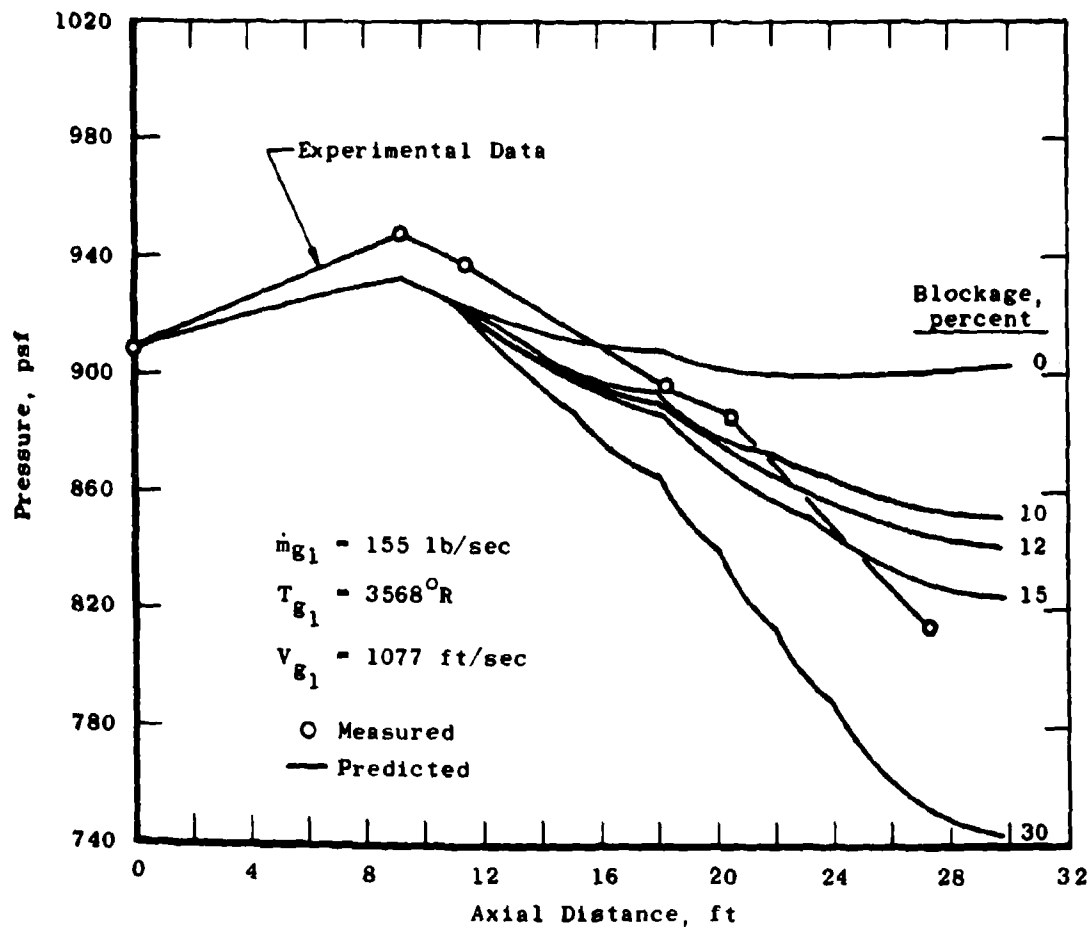


Fig. 5 Effect of Blockage on the Predicted Pressure in an Exhaust Gas Cooler

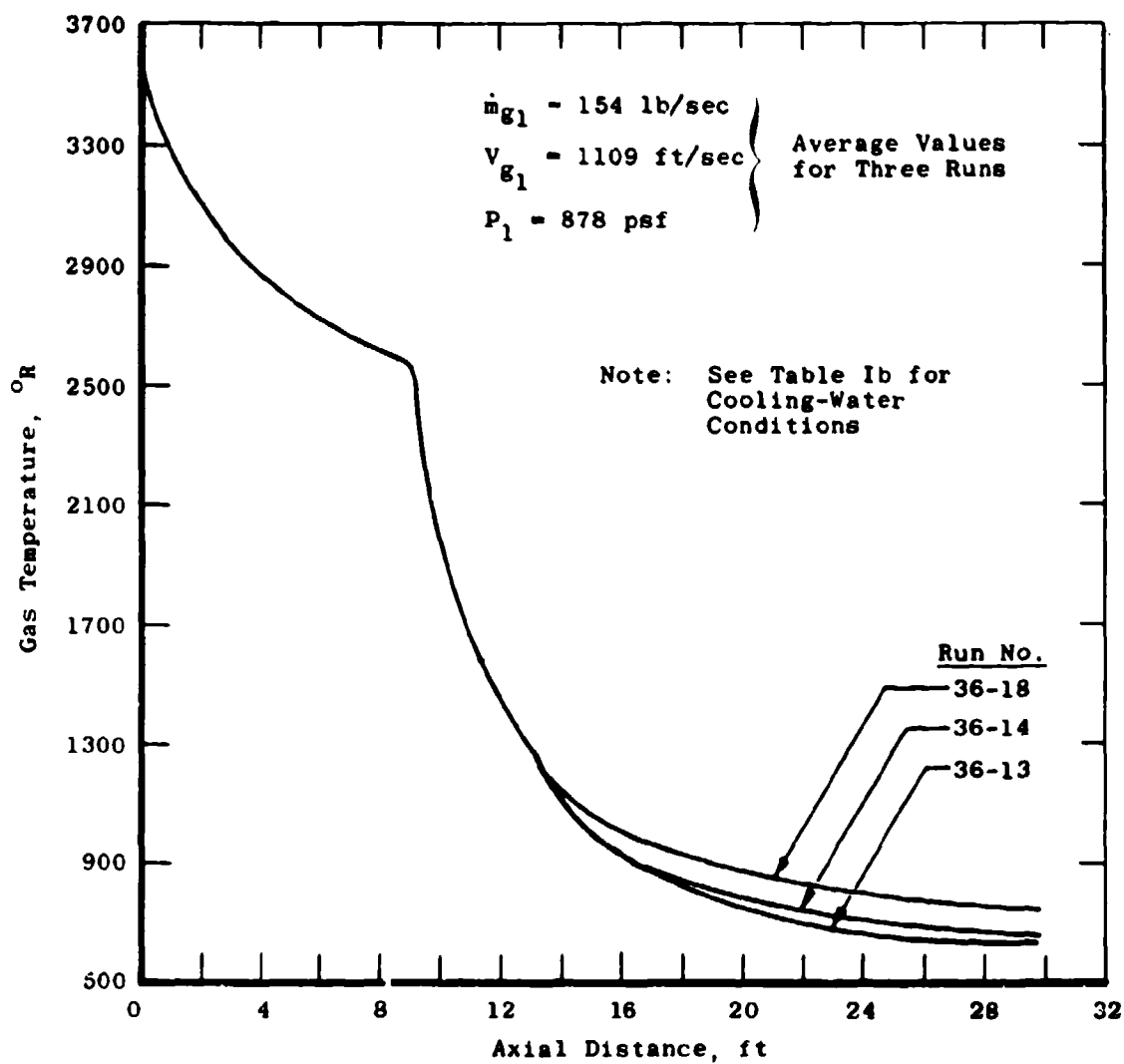
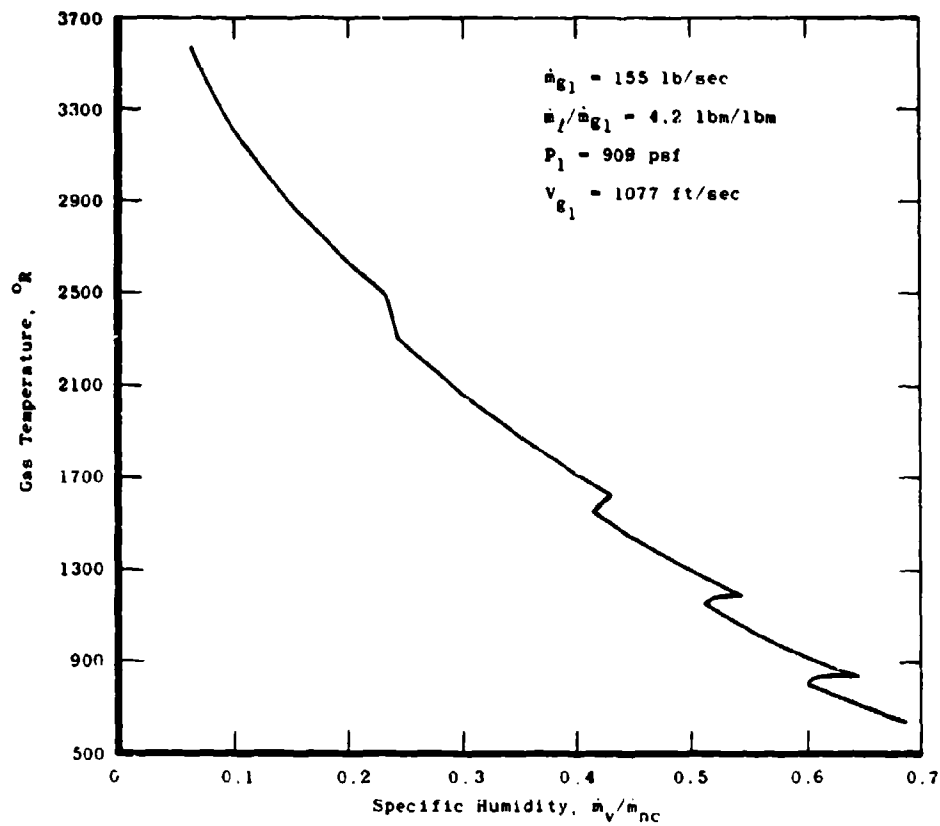
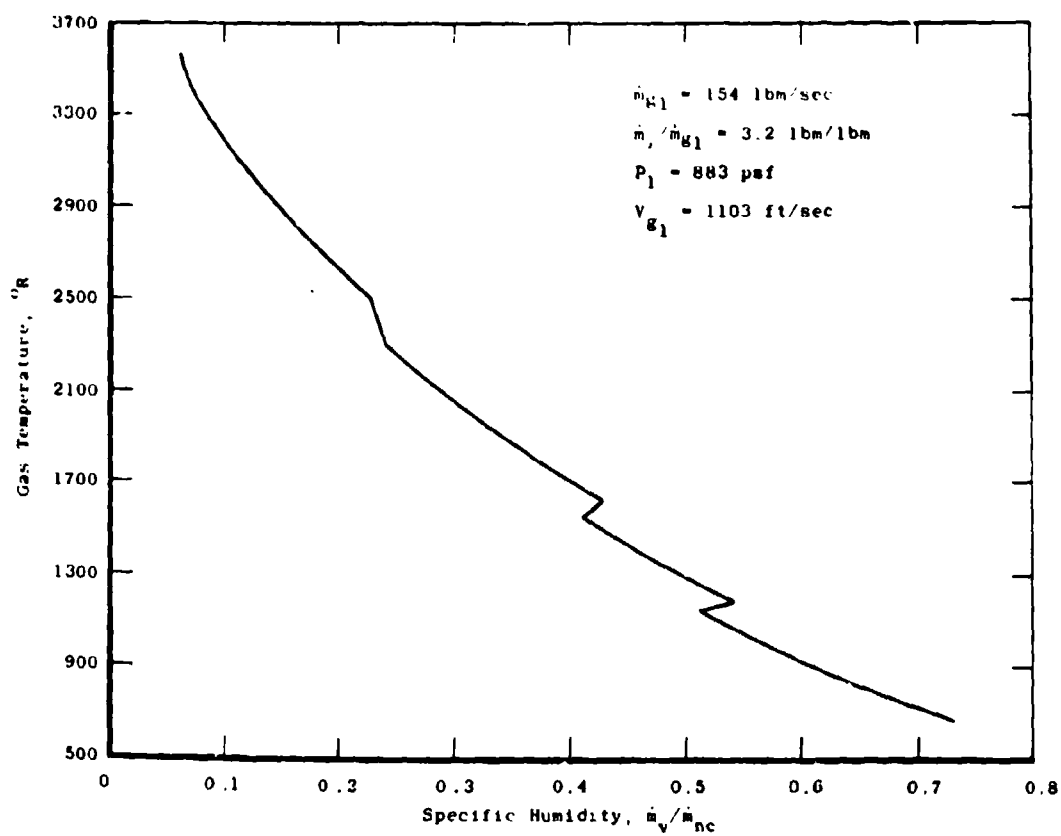


Fig. 6 Predicted Exhaust Gas Temperature for Various Amounts of Injected Cooling Water

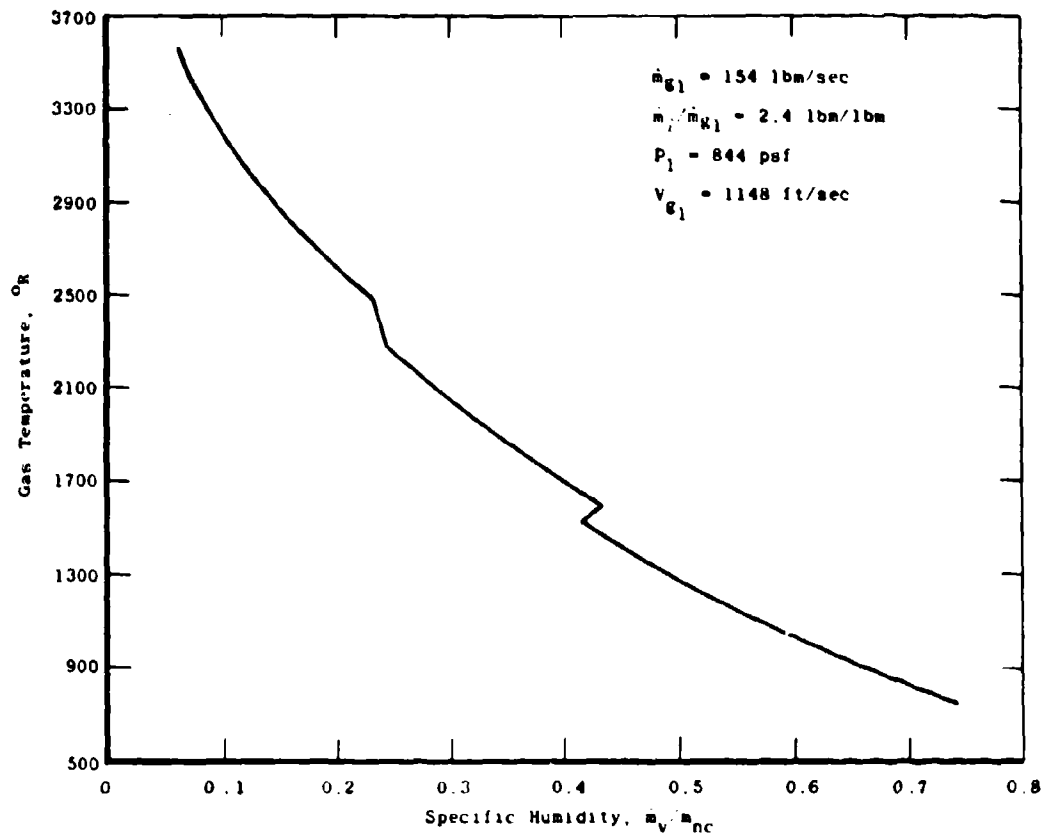


a. Run No. 36-13

Fig. 7 Predicted Relation between Exhaust Temperature and Specific Humidity for Three Cooling Water Flow Rates



b. Run No. 36-14
Fig. 7 Continued



c. Run No. 36-18

Fig. 7 Concluded

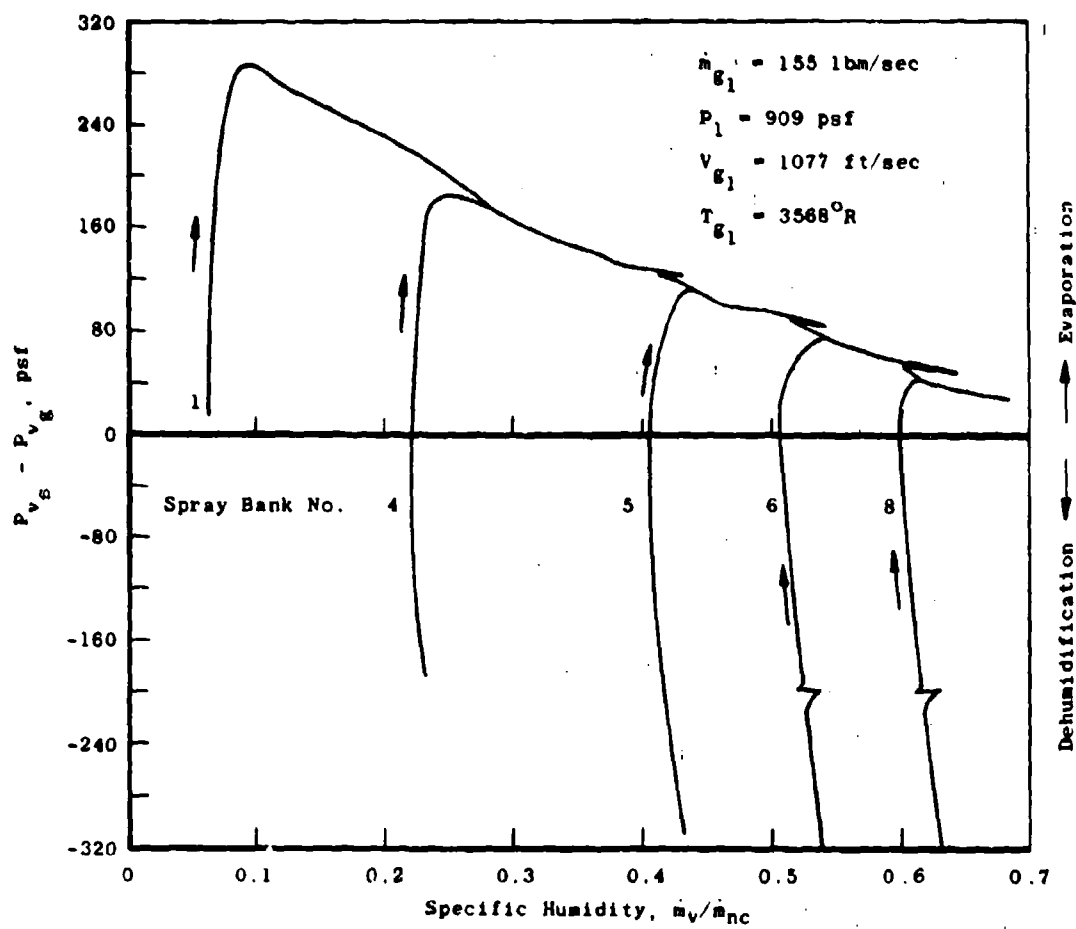


Fig. 8 Effect of the Difference in the Partial Pressures of the Liquid and Exhaust Gas Streams on the Specific Humidity for Run No. 36-13

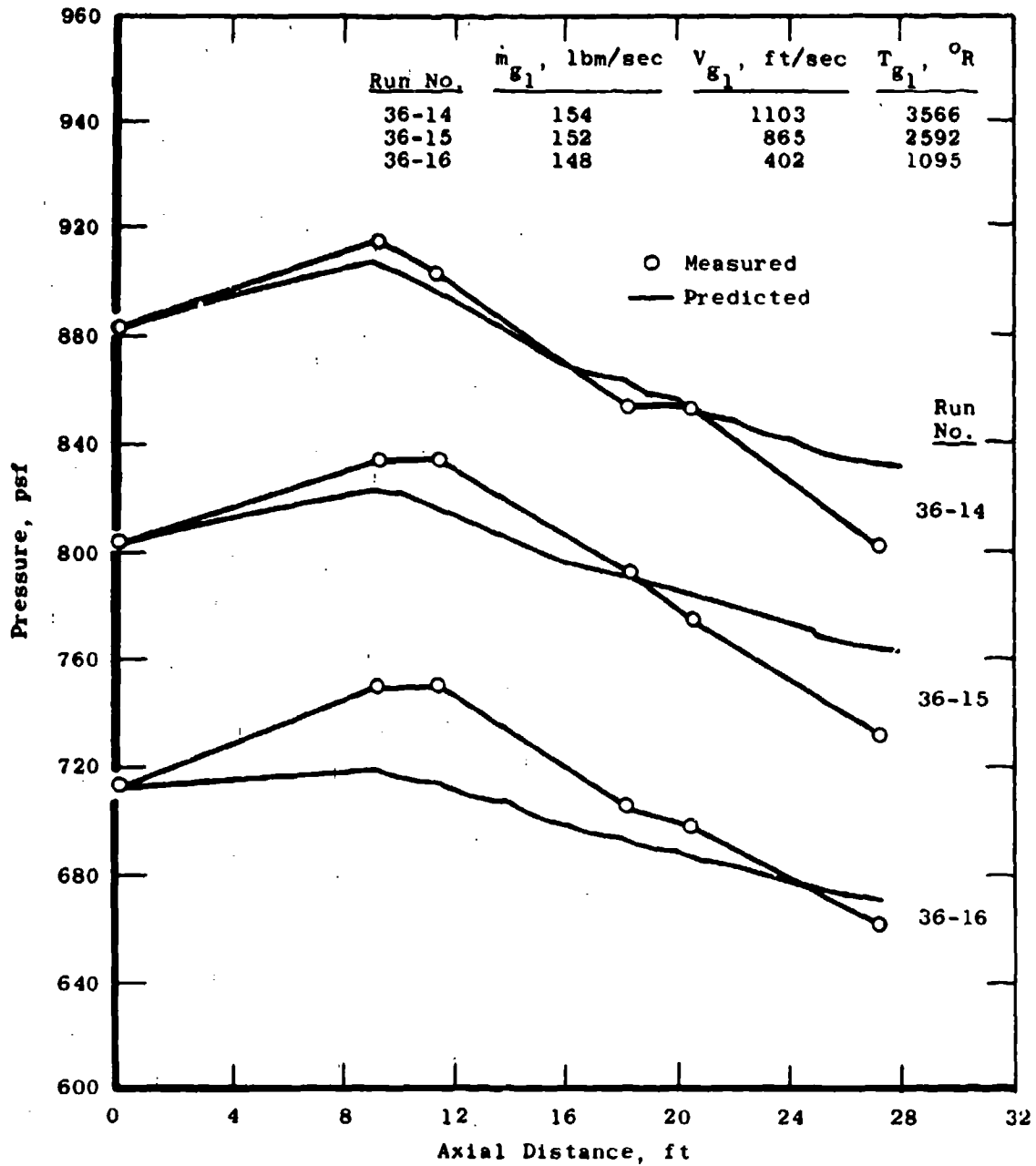


Fig. 9 Measured and Predicted Exhaust Gas Cooler Pressures for Approximately Constant Inlet Mass Flow Rate and Various Total Energy Levels

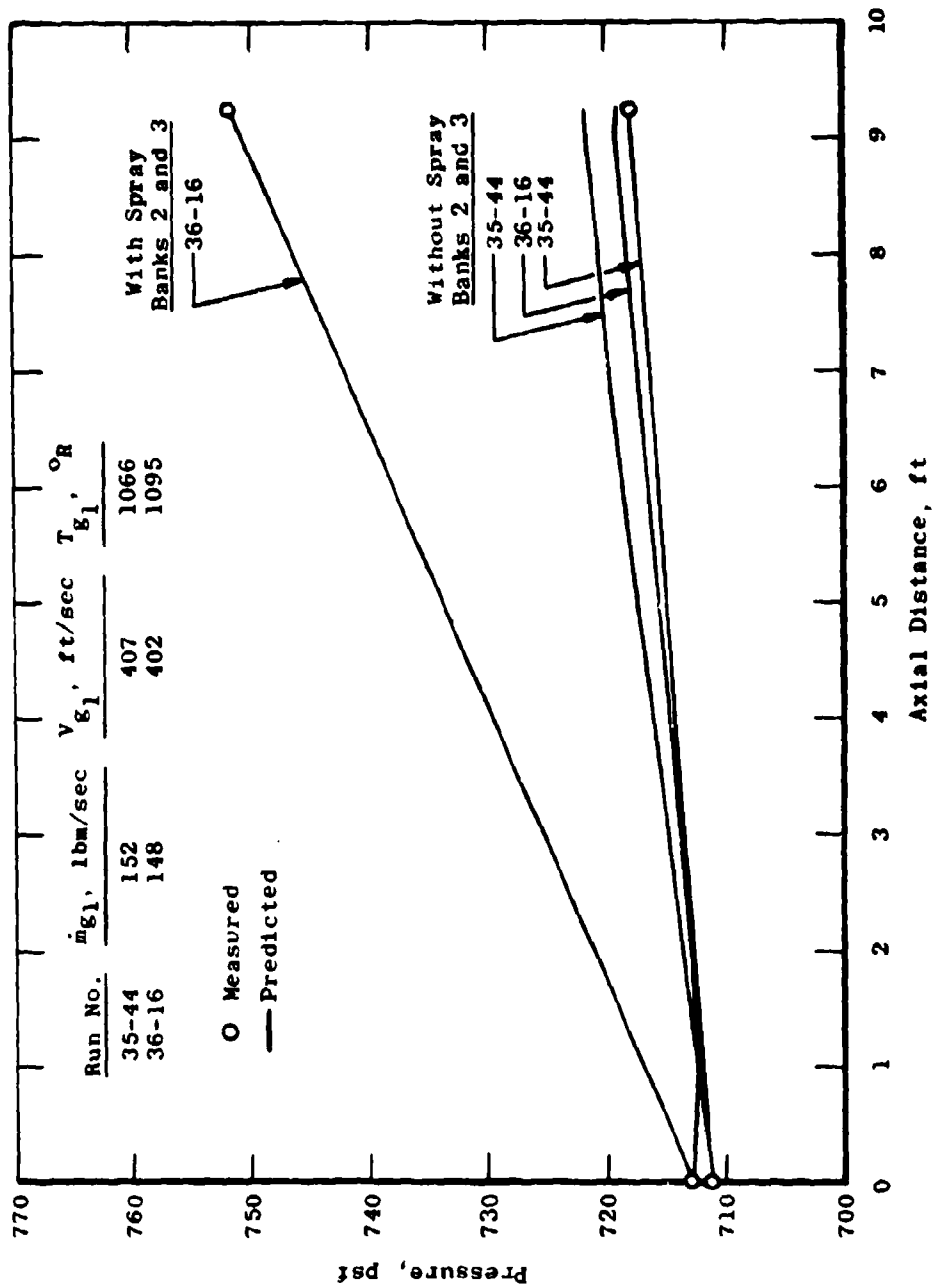


Fig. 10 A Comparison of Two Runs to Evaluate the Influence of the Wall Spray Banks on the Predicted and Measured Cooler Pressure

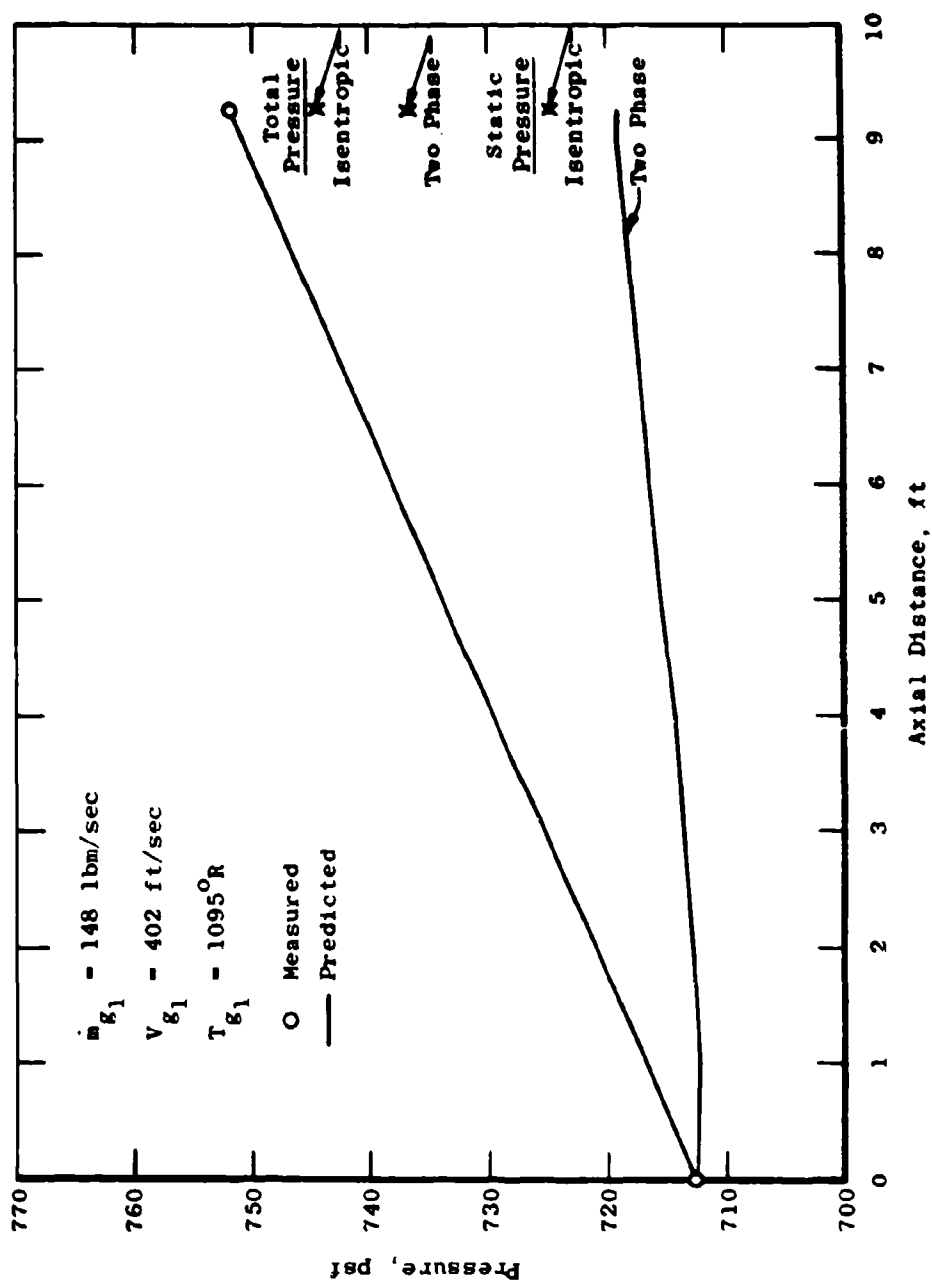


Fig. 11 A Comparison of the Measured Cooler Pressure with Several Values of Calculated Pressure for Run 36-16

TABLE I
INLET ENGINE AND COOLER CONDITIONS FOR RUNS NO. 36-13, 36-14, AND 36-18

a. Measured and Calculated Exhaust Gas Parameters

Run No.	Inlet Flow Rate		Inlet Air Temperature, °R	Cooler Inlet Pressure, P, psia	Inlet Area, ft ²	Calculated Cooler Inlet Conditions		
	Fuel, lbm/sec	Air, lbm/sec				Moisture Mass Fraction, C _v	Velocity, V _g , ft/sec	Temperature, T _g , °R
36-13	7.40	147.57	778.0	6.31	30.275	0.05959	1077.0	3567.8
36-14	7.36	146.99	787.0	6.13	30.275	0.05950	1103.1	3566.4
36-18	7.33	146.97	782.0	5.87	30.275	0.05928	1147.7	3548.2

TABLE I (Concluded)
b. Cooling Water Flow Rate Conditions

Inlet Liquid						
Run No.	Station or Spray Bank	Distance from Entrance, x, ft	Liquid Ratio, f_l , lbm/lbm	Velocity V_l , ft/sec	Temperature, T_s , °R	Drop Size, D, ft
36-13	1	0	0.524	70.0	536	0.20×10^{-2}
	4	9.25	0.849	90.0		0.59×10^{-3}
	5	11.50	0.890	90.0		↓
	6	13.75	0.951	95.0		
	7	16.00	0.0	90.0		
	8	18.25	0.968	94.0		
	9	20.50	0.0	90.0		
	10	22.75	0.0	90.0		
	11	25.00	0.0	90.0		
36-14	1	0	0.520	69.5		0.20×10^{-2}
	4	9.25	0.849	91.5		0.59×10^{-3}
	5	11.50	0.908	91.0		↓
	6	13.75	0.961	95.5		
	7	16.00	0.0	↓		
	8	18.25	↓	↓		
	9	20.50	↓	↓		
	10	22.75	↓	↓		
	11	25.00	↓	↓		
36-18	1	0	0.531	71.0		0.194×10^{-2}
	4	9.25	0.887	92.0		0.59×10^{-3}
	5	11.50	0.935	91.5		↓
	6	13.75	0	↓		
	7	16.00	↓	↓		
	8	18.25	↓	↓		
	9	20.50	↓	↓		
	10	22.75	↓	↓		
	11	25.00	↓	↓		

TABLE II
INLET ENGINE AND COOLER CONDITIONS FOR RUNS NO. 36-14, 36-15, 36-16, AND 35-44
a. Measured and Calculated Exhaust Gas Parameters

Run No.	Inlet Flow Rate		Inlet Air Temperature, °R	Cooler Inlet Pressure, P, psia	Inlet Area, ft ²	Calculated Cooler Inlet Conditions		
	Fuel, lbm/sec	Air, lbm/sec				Moisture Mass Fraction, C _v	Velocity, V _g , ft/sec	Temperature, T _g , °R
36-14	7.36	146.99	787.0	6.13	30.275	0.05950	1103.4	3567.5
36-15	4.32	147.23	779.0	5.58	30.275	0.03584	864.9	2592.3
36-16	0.65	147.23	781.0	4.95	30.275	0.00552	402.0	1095.5
35-44	1.219	152.5	484.2	4.94	30.275	0.00997	407.8	1066.3

TABLE II (Concluded)
b. Cooling Water Flow Rate Conditions

Inlet Liquid						
Run No.	Station or Spray Bank	Distance from Entrance, x, ft	Liquid Ratio, f_g , lbm/lbm	Velocity V_f , ft/sec	Temperature, T_s , °R	Drop Size, D, ft
36-14	1	0.0	0.52	69.5	536	0.0020
	2	4.8	0.187	81.0		0.0020
	3	6.9	0.176	76.6		0.0020
	4	9.25	0.849	91.5		0.00059
	5	11.5	0.908	91.0		
	6	13.75	0.961	95.5		
	7	16.00	0.0			
	8	18.25	0.0			
	9	20.5	0.0			
36-15	1	0.0	0.534	71.0		0.00197
	2	4.8	0.191	81.0		0.00197
	3	6.9	0.179	76.0		0.00197
	4	9.25	0.873	87.0		0.00059
	5	11.5	0.921	89.0		
	6	13.75	0.979	90.0		
	7	16.00	0.0			
	8	18.25	0.0			
	9	20.5	0.0			
36-16	1	0.0	0.547	72.0		0.00194
	2	4.8	0.195	81.0		0.00194
	3	6.9	0.186	77.0		0.00194
	4	9.25	0.910	88.0		0.00059
	5	11.5	0.959	91.0		
	6	13.75	1.01			
	7	16.0	0.0			
	8	18.25	0.0			
	9	20.5	0.0			
35-44	1	0.0	0.175	60.0		0.0020
	2	4.8	0.0	60.0		0.0020
	3	6.9	0.0	60.0		0.0020
	4	9.25	1.020	104.0		0.00059
	5	11.5	1.341	98.5		
	6	13.75	0.0	93.0		
	7	16.00				
	8	18.25				
	9	20.5				

APPENDIX III

A LISTING OF THE COMPUTER PROGRAM AND THE REQUIRED AUXILIARY EQUATIONS FOR AN EXHAUST GAS COOLER

A listing of the computer program for the solution of the equations developed in the text is given in this section. In addition, auxiliary equations necessary to define certain constants used for the computer solution are listed.

AUXILIARY EQUATIONS

These auxiliary equations are used to define certain dimensionless numbers which are in turn solved for certain coefficients used in the equations developed in the text. The properties of the system used in the equations are evaluated at the so-called "film temperature" which is defined as

$$T_{fi} = \frac{T_{si} + T_g}{2}$$

where the f indicates a film property and the i identifies the particular liquid stream being discussed. For this program, the noncondensable gas was assumed to be dry air, and the various constants were calculated using the properties of air. The desired constants and equations are given below:

$$C_{Di} = \frac{24}{Re_{fi}} \left[1 + 0.15 (Re_{fi})^{0.687} \right] \quad (\text{Ref. 9})$$

$$C_{pfi} = (\bar{x}_{vfi}) \left(M_v / M_{fi} \right) (C_{pvfi}) + (1 - \bar{x}_{vfi}) \left(M_{nc} / M_{fi} \right) (C_{pncfi})$$

$$D_i = (6 M_{di} / \pi \rho_l)^{1/3}$$

$$h_i = \frac{(Nu_{fi}) (\mu_{fi}) (C_{pfi})}{(D_i) (Pr_{fi})}$$

$$k_{fi} = \bar{x}_{vfi} k_{vfi} + (1 - \bar{x}_{vfi}) k_{ncfi}$$

$$k_{xi} = \frac{(Nu_{ab_{fi}})(\mu_{fi})}{(D_i)(Sc_{fi})(M_{fi})}$$

$$M_{fi} = \bar{x}_{v_{fi}} M_v + (1 - \bar{x}_{v_{fi}}) M_{nc}$$

$$\frac{\dot{m}_{nc}}{A_o} = \left[\frac{V_g (1 - C_v) P}{T_g R_u \left(\frac{1 - C_v}{M_{nc}} + \frac{C_v}{M_v} \right)} \right]_{x=0}$$

$$Nu_{ab_{fi}} = 2 + 0.6 (Re_{fi})^{1/2} (Sc_{fi})^{1/3} \quad (\text{Ref. 10})$$

$$Nu_{fi} = 2 + 0.6 (Re_{fi})^{1/2} (Pr_{fi})^{1/3} \quad (\text{Ref. 10})$$

$$Pr_{fi} = \frac{(C_{p_{fi}})(\mu_{fi})}{k_{fi}}$$

$$R = Ru / [\bar{x}_v M_v + (1 - \bar{x}_v) M_{nc}]$$

$$Re_{fi} = \frac{(V_g - V_{li})(\rho_{fi})(D_i)}{\mu_{fi}}$$

$$Sc_{fi} = \frac{\mu_{fi}}{(\rho_{fi})(D_{ab_{fi}})}$$

$$V_g \rho_g (1 - C_v) = \frac{\dot{m}_{nc}}{A_o} / \frac{A}{A_o}$$

$$\bar{x}_{v_{fi}} = \frac{\bar{x}_{v_{si}} + \bar{x}_v}{2}$$

$$\bar{x}_v = \frac{C_v/M_v}{C_v/M_v + (1 - C_v)M_{nc}}$$

$$\bar{x}_{v_{si}} = \frac{P_{v_{si}}}{P}$$

$$\mu_{fi} = \left(\bar{x}_{v_{fi}}\right)\left(\mu_{v_{fi}}\right) + \left(1 - \bar{x}_{v_{fi}}\right)\left(\mu_{nc_{fi}}\right)$$

$$\rho_{fi} = \frac{(P)(M_{fi})}{R_u T_{fi}}$$

COMPUTER LISTING

The following computer program was programmed for the IBM 360 computer and was used to obtain the calculated data of this program.

DETERMINATION OF EXHAUST GAS COOLER INLET CONDITIONS

```

IMPLICIT REAL*8(A-H,O-Z)
REAL*4  T2TW(150)
REAL*4  AREA,RAIR,RFUEL
REAL*4  ARTM,ARPR,ARW1
REAL*4  IRUN(2),WHAT(14)
COMMON /GRP/ ARTM(150),ARPR(150),ARW1(150),NPLT
COMMON / ENTH / T, TY, R, COA(7), H(6), CO2A(7), H2A(7), XN2A(7),
1  O2A(7), H2OA(7), WT(6), COB(7), CO2B(7), H2B(7), XN2B(7),
2  O2B(7), H2OB(7), C(6), A(6), P(212)
CALL ERRSET(261,256,-1,1)
WT(1) = 28.011
WT(2) = 44.011
WT(3) = 2.016
WT(4) = 28.016
WT(5) = 32.0
WT(6) = 18.016
P(201) = 11.766
P(203) = 12.260
P(205) = 12.770
P(207) = 13.298
P(209) = 13.844
P(211) = 14.408
R = 1.98726
C  PRESSURE DATA 32 F TO 212 F
C
  READ (5,100) (P(I),I=32,199)
  READ (5,100) (P(I),I=200,212,2)
100 FORMAT (8D10.0)
C
C  TEMPERATURE COEFFICIENTS
C
  READ (5,101) (COA(I),I=1,7), (COB(I),I=1,7), (CO2A(I),I=1,7),
1  (CO2B(I),I=1,7), (H2A(I),I=1,7), (H2B(I),I=1,7), (XN2A(I),I=1,7),
2  (XN2B(I),I=1,7), (O2A(I),I=1,7), (O2B(I),I=1,7), (H2OA(I),I=1,7),
3  (H2OB(I),I=1,7)
101 FORMAT (50F6.7/2D16.7)
C
C  INPUT CONDITIONS
C  987 FORMAT (20A4)
C
C  MASS FRACTIONS
C
105 FORMAT (6D12.5)
1  READ (17,END=999) C,P1,T1,TW1,VG1,VW1,TGUESS,IRUN,WHAT
  READ (17) AREA,RAIR,RFUEL
  TGUESS=TW1+.01*(T1-TW1)
  T2MAX=TW1+.01
  T2MIN=TW1+.01

  IF (P1.EQ.0.AND.T1.EQ.0.) GO TO 999
  NPLT=0
C
  EM=0.
  DO 6 I = 1,6
    EM = EM + C(I) / WT(I)
6 CONTINUE
  EM = 1.0 / EM
  CV1 = C(6)

```

```

CNC1 = 1.0 - CV1
PV1 = (P1 * EM * CV1) / 18.
PNC1 = P1 - PV1
EMNC = (P1 * EM * CNC1) / PNC1
T = T1 / 1.8
CALL ENTHAL
SUM = 0.0
DO 7 I = 1, 5
SUM = SUM + C(I) * H(I)
/ CONTINUE
HNC1 = SUM / CNC1
C ITERATE FOR TEMPERATURE AS A FUNCTION OF ENTHALPY
C
DOTM = RAIR + RFUEL
CONSTN = ((1545. * DOTM) / (28.85 * 144. * P1 * AREA)) ** 2
CONSTN = CONSTN / (2. * 7 / 8. * 32.2)
TGS1 = T1 / 1.8
TGS2 = (T1 - 0.5 * T1) / 1.8
T = TGS1
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT1)
T = T * 1.8
FUNC1 = ENT1 + CONSTN * T * T - HNC1
T = TGS2
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT2)
T = T * 1.8
FUNC2 = ENT2 + CONSTN * T * T - HNC1
401 TGIF = (TGS1 + TGS2) / 2.
IF (DABS(TGS2 - TGS1) / TGS2) .LT. 0.10 - 061 GO TO 410
T = TGIF
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT3)
T = T * 1.8
FUNC3 = ENT3 + CONSTN * T * T - HNC1
TSTA = FUNC1 * FUNC3
TSTB = FUNC2 * FUNC3
IF (TSTA) 404, 404, 402
402 IF (TSTB) 405, 405, 403
403 WRITE (6, 411)
411 FORMAT('O NO ROOT FOR INITIAL ENTHALPH')
GO TO 1
404 FUNC2 = FUNC3
TGS2 = TGIF
GO TO 401
405 FUNC1 = FUNC3
TGS1 = TGIF
GO TO 401
410 ENTNU = ENT3
T1 = TGIF * 1.8
C END OF ITERATION
VG1 = (DOTM * 1545. * T1) / (P1 * AREA * 144.)
VG1 = VG1 / 28.85
WRITE (6, 413) ENTNU, T1, VG1
WRITE (9) C(6), VG1, T1, P1
413 FORMAT('O ENTHALPY = ', E12.4, ' GAS TEMPERATURE = ', E12.4,
* ' GAS VELOCITY = ', E12.4)
GO TO 1
999 CONTINUE

```

 ENDFILE 9

REWIND 9

RETURN

 END

SUBROUTINE ENTHAL

IMPLICIT REAL*8(A-H,O-Z)

COMMON / ENTH / T, TT, R, COA(7), H(6), CO2A(7), H2A(7), XN2A(7),

1 O2A(7), H2OA(7), WT(6), COB(7), CO2B(7), H2B(7), XN2B(7),

2 O2B(7), H2OB(7), CT(6), A(6), P(212)

3 TT=R*T*1.8

IF (T.LT.1000) GO TO 10

DO 4 J=1,6

4 A(J)=COA(J)

CALL HTRT (T,A,H(1))

H(1)=H(1)*TT/WT(1) +28488.3 *1.8/WT(1)

DO 5 J=1,6

5 A(J)=CO2A(J)

CALL HTRT (T,A,H(2))

H(2)=H(2)*TT/WT(2) +96290. *1.8/WT(2)

DO 6 J=1,6

6 A(J)=H2A(J)

CALL HTRT (T,A,H(3))

H(3)=H(3)*TT/WT(3) +2023.8 *1.8/WT(3)

DO 7 J=1,6

7 A(J)=XN2A(J)

CALL HTRT (T,A,H(4))

H(4)=H(4)*TT/WT(4) +2072.3 *1.8/WT(4)

DO 8 J=1,6

8 A(J)=O2A(J)

CALL HTRT (T,A,H(5))

H(5)=H(5)*TT/WT(5) +2074.7 *1.8/WT(5)

DO 9 J=1,6

9 A(J)=H2OA(J)

CALL HTRT (T,A,H(6))

H(6)=H(6)*TT/WT(6) +60164.7 *1.8/WT(6)

HV2=H(6)

GO TO 17

10 DO 11 J=1,6

11 A(J)=COB(J)

CALL HTRT (T,A,H(1))

H(1)=H(1)*TT/WT(1) +28488.3 *1.8/WT(1)

DO 12 J=1,6

12 A(J)=CO2B(J)

CALL HTRT (T,A,H(2))

H(2)=H(2)*TT/WT(2) +96290. *1.8/WT(2)

DO 13 J=1,6

13 A(J)=H2B(J)

CALL HTRT (T,A,H(3))

H(3)=H(3)*TT/WT(3) +2023.8 *1.8/WT(3)

DO 14 J=1,6

14 A(J)=XN2B(J)

CALL HTRT (T,A,H(4))

H(4)=H(4)*TT/WT(4) +2072.3 *1.8/WT(4)

DO 15 J=1,6

15 A(J)=O2B(J)

CALL HTRT (T,A,H(5))

H(5)=H(5)*TT/WT(5) +2074.7 *1.8/WT(5)

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```
DO 16 J=1,6
16 A(J)=H2OB(J)
CALL WTRT (T,A,H(6))
H(6)=H(6)*11/W1(6)*60164.7*1.5/W1(6)
HV2=H(6)
17 CONTINUE
HNC2 =0.0
DO 18 I=1,5
18 HNC2=HNC2 + C(I)*H(I)
HNC2=HNC2/11.0-C(6)
RETURN
END
```

```
SUBROUTINE SUMIT(C,H,CN,ENTH)
IMPLICIT REAL*8 (A-H,O-Z)
DIMENSION C(1),H(1)
SUM=0.
DO 1 I=1,5
1 SUM=SUM+C(I)*H(I)
ENTH=SUM/CN
RETURN
END
```

MAIN PROGRAM FOR EXHAUST GAS SPRAY COOLER

```

IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
COMMON ISET,KKK,KOUNT,KN,KKI
COMMON DL,BL,CL,TSA
COMMON FLN,VLA,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE1
14,GE15,GE16,GE17,GE18
COMMON PVIS,DAB,SCF,RE,ABN,FNU
COMMON ROF
COMMON/LDW/ MICH
DIMENSION Y(99),YP(99),A5(10),A6(10),A7(10),A8(10),A10(10),XVS(10
1),XHL(10),XHEG(10),BARH(10),DM(10),XK(10),CO(10)
2,XHV(10)
DIMENSION FL(10),VL(10),TS(10),DI(10),STA(11)
103 FORMAT(1H1,'BEGIN STATION',I4,12X,12HRUN NUMBER ,8A8)
M=0
200 M=M+1
XVF=0.0
KI=1
AA=.88
VLB=1.0
IKK=0
BA=.25
KN=1
KI=1
MICH=0
READ(5,51015,END=222)ALP
51015 FORMAT(10A8)
READ(5,102) NSTA
NS1= NSTA+1
READ(5,101)(STA(I),I=1,NS1)
READ(9,END=222)CV,VC,TG,P
P=P*144.0
READ(5,101)(FL(I),I=1,NSTA)
READ(5,101)(VL(I),I=1,NSTA)
READ(5,101)(TS(I),I=1,NSTA)
READ(5,101)(DI(I),I=1,NSTA)
READ(5,101)(B(I),I=1,4)
100 FORMAT(4E10.2)
101 FORMAT(1E10.0)
102 FORMAT(12)
CPL=1.0
GE1=STA(2)-.25D0
GE2=GE1+.100
GE3=STA(3)-.25D0
GE4=GE3+.100
GE5=STA(4)-.25D0
GE6=GE5+.100
GE7=STA(5)-.25D0
GE8=GE7+.100
GE9=STA(6)-.25D0
GE10=GE9+.100
GE11=STA(7)-.25D0
GE12=GE11+.100
GE13=STA(8)-.25D0
GE14=GE13+.100

```

```

GE15=STA(9)-.2500
GE16=GE15+.100
GE17=STA(10)-.2*DO
GE18=GE17+.100
FLN=0.0
VLA=0.0
DA=0.0
FLA=0.0
CL=0.0
DL=0.0
TSA=0.0
BL=0.0
CJ=778.0
GC= 32.2
ISET = 9
GW=29.0
VW=18.0
RV=1.986*778.0
RVAP=RV/VW
RNC=RV/GW
ROL=62.4
TREF = 540.0
ISTA=1
MSTA=ISTA
NEQ=8
C THE ARRAYS USED IN DIFFE ARE NOW SET UP
Y(1)=CV
Y(2)=VG
Y(3)=TG
Y(4)=P
Y(5)=FL(1)
Y(6)=VL(1)
Y(7)=TS(1)
Y(8)=D(1)
WNCAL = Y(2) * (1.0-Y(1)) * Y(4) / (Y(3)*RV*((1-Y(1))/GW+Y(1)/VW))
X=0.0
WRITE(6,103)ISTA,ALP
3 CONTINUE
DX=.0001
KOUNT =0
IF(ISTA.EQ.1)J=8
DO 1 II=1,999999
IF(ISTA.EQ.1)KKI=8
IF(ISTA.EQ.2)KKI=16
IF(ISTA.EQ.3)KKI=24
IF(ISTA.EQ.4)KKI=32
IF(ISTA.EQ.5)KKI=40
IF(ISTA.EQ.6)KKI=48
IF(ISTA.EQ.7)KKI=56
IF(ISTA.EQ.8)KKI=64
IF(ISTA.EQ.9)KKI=72
IF(X.GE.STA(2).AND.X.LE.GE3)KKI=8
IF(X.GE.GE3.AND.X.LE.STA(3))KKI=16
IF(X.GE.STA(3).AND.X.LE.GE5)KKI=16
IF(X.GE.GE5.AND.X.LE.STA(4))KKI=24
IF(X.GE.STA(4).AND.X.LE.GE7)KKI=24
IF(X.GE.STA(5).AND.X.LE.GE9)KKI=32
IF(X.GE.STA(6).AND.X.LE.GE11)KKI=40
IF(X.GE.STA(7).AND.X.LE.GE13)KKI=48
IF(X.GE.STA(8).AND.X.LE.GE15)KKI=56
IF(X.GE.STA(9).AND.X.LE.GE17)KKI=64

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IF(ISTA.EQ.2.AND.X.GE.STA(2))MSTA=3
IF(X.GE.STA(2).AND.X.LE.GE3)MSTA=3
IF(ISTA.EQ.3.AND.X.GE.STA(3))MSTA=3
IF(X.GE.STA(3).AND.X.LE.GE5)MSTA=3
IF(ISTA.EQ.4.AND.X.GE.STA(4))MSTA=3
IF(ISTA.EQ.5.AND.X.GE.STA(5))MSTA=3
IF(ISTA.EQ.6.AND.X.GE.STA(6))MSTA=3
IF(ISTA.EQ.7.AND.X.GE.STA(7))MSTA=3
IF(ISTA.EQ.8.AND.X.GE.STA(8))MSTA=3
IF(ISTA.EQ.9.AND.X.GE.STA(9))MSTA=3
IF(X.GE.GE3.AND.X.LE.GE4)MSTA=1
IF(X.GE.GE5.AND.X.LE.GE6)MSTA=1
IF(X.GE.GE7.AND.X.LE.GE8)MSTA=1
IF(X.GE.GE9.AND.X.LE.GE10)MSTA=1
IF(X.GE.GE11.AND.X.LE.GE12)MSTA=1
IF(X.GE.GE13.AND.X.LE.GE14)MSTA=1
IF(X.GE.GE15.AND.X.LE.GE16)MSTA=1
IF(X.GE.GE17.AND.X.LE.GE18)MSTA=1
IF(X.GE.GE2.AND.X.LE.STA(2))MSTA=2
IF(X.GE.GE4.AND.X.LE.STA(3))MSTA=2
IF(X.GE.GE6.AND.X.LE.STA(4))MSTA=2
IF(X.GE.GE8.AND.X.LE.STA(5))MSTA=2
IF(X.GE.GE10.AND.X.LE.STA(6))MSTA=2
IF(X.GE.GE12.AND.X.LE.STA(7))MSTA=2
IF(X.GE.GE14.AND.X.LE.STA(8))MSTA=2
IF(X.GE.GE16.AND.X.LE.STA(9))MSTA=2
IF(X.GE.GE18)MSTA=2
CALL DIFFE(X,Y,YP,DX,IKK,NEQ,KI,MSTA)
KI=1
IF(X.GE.GE1.AND.X.LE.GE1+DX)GO TO 909
IF(X.GE.GE3.AND.X.LE.GE3+DX)GO TO 666
IF(X.GE.GE5.AND.X.LE.GE5+DX)GO TO 666
IF(X.GE.GE7.AND.X.LE.GE7+DX)GO TO 666
IF(X.GE.GE9.AND.X.LE.GE9+DX)GO TO 666
IF(X.GE.GE11.AND.X.LE.GE11+DX)GO TO 666
IF(X.GE.GE13.AND.X.LE.GE13+DX)GO TO 666
IF(X.GE.GE15.AND.X.LE.GE15+DX)GO TO 666
IF(X.GE.GE17.AND.X.LE.GE17+DX)GO TO 666
GO TO 669
666 FLN=(Y(LA)+Y(LA-4)+Y(LA-8))
VLA=(Y(LA+1)*Y(LA)+Y(LA-3)*Y(LA-4)+Y(LA-7)*Y(LA-8))/FLN
DA=(Y(LA+3)*Y(LA)+Y(LA-1)*Y(LA-4)+Y(LA-5)*Y(LA-8))/FLN
TSA=(Y(LA+2)*Y(LA)+Y(LA-2)*Y(LA-4)+Y(LA-6)*Y(LA-8))/FLN
GO TO 669
909 FLN=FLN+Y(J-3)
VLA=VLA+(Y(J-3)*Y(J-2))/FLN
DA=DA+(Y(J-3)*Y(J))/FLN
TSA=TSA+Y(J-1)/ISTA
669 CONTINUE
IF(X.GE.GE1.AND.X.LE.GE1+DX)GO TO 8000
IF(X.GE.GE3.AND.X.LE.GE3+DX)GO TO 8000
IF(X.GE.GE5.AND.X.LE.GE5+DX)GO TO 8000
IF(X.GE.GE7.AND.X.LE.GE7+DX)GO TO 8000
IF(X.GE.GE9.AND.X.LE.GE9+DX)GO TO 8000
IF(X.GE.GE11.AND.X.LE.GE11+DX)GO TO 8000
IF(X.GE.GE13.AND.X.LE.GE13+DX)GO TO 8000
IF(X.GE.GE15.AND.X.LE.GE15+DX)GO TO 8000
IF(X.GE.GE17.AND.X.LE.GE17+DX)GO TO 8000
IF(X.GE.GE2.AND.X.LE.GE2+DX)GO TO 8001
IF(X.GE.GE4.AND.X.LE.GE4+DX)GO TO 8001
IF(X.GE.GE6.AND.X.LE.GE6+DX)GO TO 8001

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IF(X.GE. GE8 .AND. X .LE. GE8+DX)GO TO 8001
IF(X.GE. GE10 .AND. X .LE. GE10+DX)GO TO 8001
IF(X.GE. GE12 .AND. X .LE. GE12+DX)GO TO 8001
IF(X.GE. GE14 .AND. X .LE. GE14+DX)GO TO 8001
IF(X.GE. GE16 .AND. X .LE. GE16+DX)GO TO 8001
IF(X.GE. GE18 .AND. X .LE. GE18+DX)GO TO 8001
99 IF(X.GE.STA(ISTA+1))GO TO 2
GO TO 1
8000 CONTINUE
IF(ISTA.EQ. 1)LA=5
IF(ISTA.EQ. 2)LA=13
IF(ISTA.EQ. 3)LA=21
IF(ISTA.EQ. 4)LA=29
IF(ISTA.EQ. 5)LA=37
IF(ISTA.EQ. 6)LA=45
IF(ISTA.EQ. 7)LA=53
IF(ISTA.EQ. 8)LA=61
IF(ISTA.EQ. 9)LA=69
Y(LA+1)=VLA
FLA=AA*FLN
Y(LA)=FLA
RG=Y(1)*RVAP+(1.0-Y(1))*RNC
ROG=Y(4)/(RG*Y(3))
SIG=.004790
Y(LA+2)=TSA
Y(LA+3)=DA
KI=0
GO TO 1
8001 CONTINUE
IF(ISTA.EQ. 1)JJ=8
IF(ISTA.EQ. 2)JJ=16
IF(ISTA.EQ. 3)JJ=24
IF(ISTA.EQ. 4)JJ=32
IF(ISTA.EQ. 5)JJ=40
IF(ISTA.EQ. 6)JJ=48
IF(ISTA.EQ. 7)JJ=56
IF(ISTA.EQ. 8)JJ=64
IF(ISTA.EQ. 9)JJ=72
PIE =3.1416
DO 86 J=1,2
J=JJ+4*(J-1)
GO TO(20,21),J
20 CONTINUE
FLA=AA*FLN
FLR=(1.0-AA)*FLN
RG=Y(1)*RVAP+(1.0-Y(1))*RNC
ROG=Y(4)/(RG*Y(3))
SIG=.004790
DB=(418.6*SIG)/(ROG*(Y(2)-VLR)**2)
GO TO 22
21 CONTINUE
Y(J-2)=VLR
Y(J-3)=FLR
Y(J-1)=TSA
Y(J)=DB
IKK=IKK+8
NEQ=NEQ+4
KI=0
22 CONTINUE
88 CONTINUE

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1  CONTINUE
   JJJ= JJJ+1
2  CONTINUE
   WRITE(6,12139)
12139 FORMAT(1H1)
C  WE WILL NOW PLOT THE STATION JUST FINISHED
   ISTA= ISTA+1
   IF(ISTA.EQ. 1)LA=5
   IF(ISTA.EQ. 2)LA=13
   IF(ISTA.EQ. 3)LA=21
   IF(ISTA.EQ. 4)LA=29
   IF(ISTA.EQ. 5)LA=37
   IF(ISTA.EQ. 6)LA=45
   IF(ISTA.EQ. 7)LA=53
   IF(ISTA.EQ. 8)LA=61
   IF(ISTA.EQ. 9)LA=69
   IF(ISTA.GT. NSTA)GO TO 5
   ISET = 9
   WRITE(6,103)ISTA,ALP
C  SET UP ARRAYS FOR DIFFE
   Y(NEQ+1)=FL(ISTA)
   Y(NEQ+2)=VL(ISTA)
   Y(NEQ+3)=TS(ISTA)
   Y(NEQ+4)=D(ISTA)
   NEQ =NEQ+4
   KOUNT = 0
   GO TO 3
5  GO TO 200
222 STOP
   END

7  SUBROUTINE DIFFE(X,Y,YP,DX,IKK,NEQ,KI,MSTA)
   IMPLICIT REAL*8(A-H,O-Z)
   COMMON BA,VLB,AA,RNC
   COMMON RGL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
   COMMON ISET,KKK,KOUNT,KN,KKI
   COMMON OL,BL,CL, TSA
   COMMON FLN,VLA,DA,FLA
   COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE1
14,GE15,GE16,GE17,GE18
   COMMON PVTs,DAB,SCF,RE,ABN,FNU
   COMMON ROP
   DIMENSION Y(99),YP(99),Z(99),ZP(99),ZN(99)
   DY=0.0
   CALL YFUNC(X,Y,YP,KI,1,MSTA,IKK,DX,DY)
120 CONTINUE
   DO 1 I=1,NEQ
   Z(I)=Y(I)+DX*YP(I)
1  CONTINUE
   X=X+DX
   DX2=.5D+0*DX
   DO 5 J=2,999
   DY=DX
666  CALL YFUNC(X,Z,ZP,KI,J,MSTA,IKK,DX,DY)
   K=0
   DO 40 I=1,NEQ
   ZN(I)=Y(I)+DX2*(YP(I)+ZP(I))
   IF(DABS(ZN(I)-Z(I))-1.0-05*DABS(ZN(I)))4,4,3
3  K=1

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4 KK=J
  Z(I)=ZN(I)
40 CONTINUE
  IF(K)5,6,5
5 CONTINUE
  WRITE(6,99)(ZN(I),I=1,NEQ)
  WRITE(6,90)KK
90 FORMAT(15)
99 FORMAT(4E20,10)
  STOP
6 DO 7 I=1,NEQ
7 Y(I)=Z(I)
1210 CONTINUE
  IF(J .GE. 3 .AND. J .LE. 5)GO TO 1212
  IF(J .LT. 3)GO TO 2020
  DX=.5*DX
  GO TO 1212
2020 DX=2.0*DX
  IF(DX .GT. .01)DX=.01
1212 CONTINUE
  RETURN
  END

SUBROUTINE YFUNC(X,Y,YP,KI,K,MSTA,IKK,DX,DY)
IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCA0,B(4),,ISTA
COMMON ISET,KKK,KOUNT,KN,KKI
COMMON DL,BL,CL,TSA
COMMON FLN,VLA,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE1
14,GE15,GE16,GE17,GE18
COMMON PVT5,DAB,SCF,RE,ABN,FNU
COMMON ROF
DIMENSION Y(99),YP(99),A5(10),A6(10),A7(10),A8(10),A10(10),XVS(10
1),XHL(10),XHFG(10),BARH(10),DM(10),XK(10),CD(10)
2,XHV(10)
RVAP=RV/VW
9899 CONTINUE
  IF(ISTA .EQ. 1)J=8
  IF(ISTA .EQ. 2)J=16
  IF(ISTA .EQ. 3)J=24
  IF(ISTA .EQ. 4)J=32
  IF(ISTA .EQ. 5)J=40
  IF(ISTA .EQ. 6)J=48
  IF(ISTA .EQ. 7)J=56
  IF(ISTA .EQ. 8)J=64
  IF(ISTA .EQ. 9)J=72
  SIG=.00479
  PIE =3.1416
  SUM1=0.
  SUM2=0.
  SUM3=0.
  SUM4=0.
  SUM5=0.
  IF(X .GE. GE1-DX .AND. X .LE. GE1)GO TO 8001
  IF(X .GE. GE2-DX .AND. X .LE. GE2)GO TO 8001
  IF(X .GE. GE3-DX .AND. X .LE. GE3)GO TO 8001
  IF(X .GE. GE4-DX .AND. X .LE. GE4)GO TO 8001

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IF(X .GE. GE5-DX .AND. X .LE. GE5)GO TO 8001
IF(X .GE. GE6-DX .AND. X .LE. GE6)GO TO 8001
IF(X .GE. GE7-DX .AND. X .LE. GE7)GO TO 8001
IF(X .GE. GE8-DX .AND. X .LE. GE8)GO TO 8001
IF(X .GE. GE9-DX .AND. X .LE. GE9)GO TO 8001
IF(X .GE. GE10-DX .AND. X .LE. GE10)GO TO 8001
IF(X .GE. GE11-DX .AND. X .LE. GE11)GO TO 8001
IF(X .GE. GE12-DX .AND. X .LE. GE12)GO TO 8001
IF(X .GE. GE13-DX .AND. X .LE. GE13)GO TO 8001
IF(X .GE. GE14-DX .AND. X .LE. GE14)GO TO 8001
IF(X .GE. GE15-DX .AND. X .LE. GE15)GO TO 8001
IF(X .GE. GE16-DX .AND. X .LE. GE16)GO TO 8001
IF(X .GE. GE17-DX .AND. X .LE. GE17)GO TO 8001
IF(X .GE. GE18-DX .AND. X .LE. GE18)GO TO 8001
GO TO 8002
9998 CONTINUE
8001 CONTINUE
RG=Y(1)*RVAP+(1.0-Y(1))*RNC
8002 IF(K1 .EQ. 0)GO TO 8003
CALL ROGEE(ROG,Y(1),Y(2),X)
GO TO 8009
8003 ROG=Y(4)/(RG*Y(3))
8009 CONTINUE
DO 1 I=1,MSTA
J=K1+4*(I-1)
A5(I) = PIE*ROL*Y(J)**2 /2.0
CALL SUB1(Y(1),Y(3),Y(4),Y(J-1),XVS(I),XV,XK(I),Y(J),DM(I),CD(I),B
1ARH(I),Y(J-2),Y(2))
IF(XVF .GT. 1.0)RETURN
YP(J)= XK(I)* PIE *Y(J)**2 *(XV-XVS(I))*VW/(A5(I)*Y(J-2)*(1-XVS(I)
1))
A6(I)= YP(J)
YP(J-2)=((PIE/8.0)*ROG*(Y(J)**2 )*CD(I)*DABS(Y(2)-Y(J-2))*(Y(2)-
1Y(J-2))-(Y(2)-Y(J-2))*Y(J-2)*A5(I)*A6(I))/(DM(I)*Y(J-2))
A7(I)=YP(J-2)
YP(J-3)= A5(I)*A6(I)*Y(J-3)/DM(I)
A8(I)=YP(J-3)
SUM1=SUM1+A5(I)*A6(I)*Y(J-3)/DM(I)
CALL HLI(Y(J-1),XHL(I),XHFG(I),XHV(I))
YP(J-1) = BARH(I)* PIE * Y(J)**2 *(Y(3)-Y(J-1))/(DM(I)*Y(J-2))-
CXHL(I)*A5(I)*A6(I)/DM(I)+A5(I)*A6(I)*XHV(I)/DM(I)
A10(I)= YP(J-1)
SUM2 = SUM2 + Y(J-2)*A6(I)+ Y(J-3)*A7(I)
SUM3 = SUM3 +A8(I)*(XHL(I)+Y(J-2)**2 /(2.0*GC*CJ))
SUM4 = SUM4 +Y(J-3)* Y(J-2)*A7(I)/(GC*CJ)
SUM5 = SUM5 + Y(J-3)*A10(I)
1 CONTINUE
YP(1) = -SUM1*(1-Y(1))**2
A9=YP(1)
CALL AAA3(X,Y(1),Y(2),A3,AA0,DAA0,ROG)
CALL DERT( ROG,Y(1),Y(2),Y(3),Y(4),A9,SUM2,SUM3,SUM4,SUM5,A0,A3,A1
C1,A12,A13,A14,R,CPV,CPA,A4,AA0,A)
YP(3) =(A11*A14-A12*A13)/(A11*(CPV*Y(1)/(1-Y(1))+CPA)-A12*A0*ROG*R
C)
A15= YP(3)
YP(2) = (A13-A0*ROG*R*A15)/A11
A16=YP(2)
CALL DERPI(ROG,Y(1),Y(2),Y(3),R,A2,A3,A4,A9,A15,A16,DP,A)
YP(4) = DP
IF(K.GT. 1) GO TO 3

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      IF(X .EQ. 0.0)GO TO 9999
      IF(X .LT. 84)GO TO 3
      RA=BA+.25
9999 WRITE(6,4)X
      WRITE(6,8888)ROF
      WW=Y(1)/(1.0-Y(1))
4      FORMAT(1H0,'X=',F16.10)
      WRITE(6,5) AA0,WW
5      FORMAT(1H0,'A/A0=',G16.6,5X,'WV/WNC=',G16.6)
      WRITE(6,6)Y(1),Y(2),Y(3),Y(4)
      IF(ISTA .EQ. 1)JJ=8
      IF(ISTA .EQ. 2)JJ=16
      IF(ISTA .EQ. 3)JJ=24
      IF(ISTA .EQ. 4)JJ=32
      IF(ISTA .EQ. 5)JJ=40
      IF(ISTA .EQ. 6)JJ=48
      IF(ISTA .EQ. 7)JJ=56
      IF(ISTA .EQ. 8)JJ=64
      IF(ISTA .EQ. 9)JJ=72
6      FORMAT(1H0,'CV=',G20.9,5X,'VG=',G20.9,5X,'TG=',G20.9,5X,'P=',G20.9
1)
      IF(X .GE. GE2 .AND. X .LE. GE3-DX)GO TO 88
      IF(X .GE. GE4 .AND. X .LE. GE5-DX)GO TO 88
      IF(X .GE. GE6 .AND. X .LE. GE7-DX)GO TO 88
      IF(X .GE. GE8 .AND. X .LE. GE9-DX)GO TO 88
      IF(X .GE. GE10 .AND. X .LE. GE11-DX)GO TO 88
      IF(X .GE. GE12 .AND. X .LE. GE13-DX)GO TO 88
      IF(X .GE. GE14 .AND. X .LE. GE15-DX)GO TO 88
      IF(X .GE. GE16 .AND. X .LE. GE17-DX)GO TO 88
      IF(X .GE. GE18-DX)GO TO 88
      DO 10 I=1,1
      J=JJ+4*(I-1)
      WRITE(6,8) Y(J-3),Y(J-2),Y(J-1),Y(J)
      WRITE(6,9) YP(J-3),YP(J-2),YP(J-1),YP(J)
10     CONTINUE
      GO TO 1001
88     DO 988 I=1,2
      J=J+4*(I-1)
      WRITE(6,8)Y(J-3),Y(J-2),Y(J-1),Y(J)
      WRITE(6,9)YP(J-3),YP(J-2),YP(J-1),YP(J)
988    CONTINUE
      IF(X .GE. GE1+.2500-DX .AND. X .LE. GE3-DX)GO TO 1000
      IF(X .GE. GE3+.2500-DX .AND. X .LE. GE5-DX)GO TO 1000
      IF(X .GE. GE5+.2500-DX .AND. X .LE. GE7-DX)GO TO 1000
      IF(X .GE. GE7+.2500-DX .AND. X .LE. GE9-DX)GO TO 1000
      IF(X .GE. GE9+.2500-DX .AND. X .LE. GE11-DX)GO TO 1000
      IF(X .GE. GE11+.2500-DX .AND. X .LE. GE13-DX)GO TO 1000
      IF(X .GE. GE13+.2500-DX .AND. X .LE. GE15-DX)GO TO 1000
      IF(X .GE. GE15+.2500-DX .AND. X .LE. GE17-DX)GO TO 1000
      GO TO 1002
1000   CONTINUE
      IF(X .GE. GE1-DX+.2500)JJ=8
      IF(X .GE. GE3-DX+.2500)JJ=16
      IF(X .GE. GE5-DX+.2500)JJ=24
      IF(X .GE. GE7+.2500-DX)JJ=32
      IF(X .GE. GE9+.2500-DX)JJ=40
      IF(X .GE. GE11+.2500-DX)JJ=48
      IF(X .GE. GE13+.2500-DX)JJ=56
      IF(X .GE. GE15+.2500-DX)JJ=64
      IF(X .GE. GE18-DX)GO TO 1002

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1010 J=JJ+8
      WRITE(6,8) Y(J-3),Y(J-2),Y(J-1),Y(J)
      WRITE(6,9) YP(J-3),YP(J-2),YP(J-1),YP(J)
199  CONTINUE
1001  CONTINUE
1002  CONTINUE
      WRITE(6,7) YP(1),YP(2),YP(3),YP(4)
8     FORMAT(1H0,'FL=',G20.9,5X,'VL=',G20.9,5X,'TS=',G20.9,5X,'D=',G20.9
1)
9     FORMAT(1H0,'FLP=',G16.6,5X,'VLP=',G16.6,5X,'TSP=',G16.6,5X,'DP=',G
116.6)
7     FORMAT(1H0,'CVP=',G16.6,5X,'VGP=',G16.6,5X,'TGP=',G16.6,5X,'PP=',G
116.6)
3     CONTINUE
2     FORMAT(1H,7E16.6)
      RETURN
      END

SUBROUTINE SUB1(CV,TG,P,TS,XVS,XV,XK,D,DM,CD,BH,VL,VG)
IMPLICIT REAL*8(A-H,O-Z)
COMMON RA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
COMMON ISET,KKK,KOUNT,KN,KKI
COMMON DL,BL,CL,TSA
COMMON FLN,VLA,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE1
14,GE15,GE16,GE17,GE18
COMMON PVTs,DAB,SCF,RE,ABN,FNU
COMMON RQF
9899 CONTINUE
XV=(CJ/VW)/(CV/VW+(1-CV)/GW)
X2=-9.06*((-5696*TS+.0839E-4*TS**2+.0927E-7*TS**3+1352.3)/TS-
C1.4425)
X1=672.0/TS
PVTs=2117.0*X1**5.19*DEXP(X2)
XVS=PVTs/P
XVF=(XVS+XV)/2.0
XN=XVF*VW+(1-XVF)*GW
TFI=(TS+TG)/2.0
FMV=(10.6E-7*DSQRT(TFI))/(1+1538.0/TFI)
FMNC=7.5E-7*DSQRT(TFI)/(1+216.0/TFI)
FM=XVF*FMV+(1-XVF)*FMNC
DAB=(.5375/P)*(TFI/491.0)**2.334
RQF=P*XN/(RV*TFI)
SCF=FM/(RQF*DAB)
RE=DABS(VG-VL)*RQF*D/FM
IF(RE.LT.0.0)GO TO 9999
ABN=2.0+0.6*DSQRT(RE)*SCF**0.3333
XK=ABN*FM/(D*SCF*XN)
DM=D**3*.3.1416*RQV/6.0
CD=24.0/RQF*(1.0+.15*RE**0.687)
IF(TFI.GT.1700.0.AND. TFI.LT.4500.0)GO TO 221
IF(TFI.LT.400.0.OR. TFI.GT.4500)GO TO 222
CPVF=.4304+.1678E-4*TFI+.0.2781E-7*TFI**2.0
CPNCF=.0.2318+.0.1040E-4*TFI+.0.7166E-8*TFI**2
GO TO 102
221  CPVF=.3319+.0.1438E-3*TFI-.0.1312E-7*TFI**2.0
      CPNCF=.0.2214+.0.3521E-4*TFI-.0.3776E-8*TFI**2

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      GO TO 102
222  WRITE(6,101)
101  FORMAT(1H,'TEMP-FI-1S OUT OF RANGE')
      WRITE(6,122) TG,TS
122  FORMAT(1H0,'TG=',E20.6, 5X,'TS=',E20.6)
9999 CONTINUE

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```

      WRITE(6,9990)RE,VG,VL,ROF,D,FM
9990 FORMAT(6E18.10)
      STOP
102  CONTINUE
      FKV = 0.432*FMV
      FKNC = 0.257 *(0.115+ 5.17*CPNCF)* FMNC
      FK = XVF*FKV +(1-XVF)* FKNC
      CPF = XVF*(VW/XM)*CPVF +(1-XVF)*CPNCF*GW/XM
      PVF = CPF *FM/FK
      FNU = 2+.60 *DSORT(RE) * PVF** 0.3333
      RH =FNU *FM *CPF/(D*PVF)
      RETURN
      END

```

```

SUBROUTINE ROGEET(ROG,CV,VG,X)
  IMPLICIT REAL*8(A-H,O-Z)
  COMMON BA,VLB,AA,RNC
  COMMON      ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
  AAO = B(1) +B(2)*X +B(3)*X**2.0+ B(4)*X**3.0
  ROG =WNCAO/((1-CV)*VG*AAO)
  RETURN
  END

```

```

SUBROUTINE HLI(TS,XHL,XHFG,XHV)
  IMPLICIT REAL*8(A-H,O-Z)
  XHL =TS -540.0
  XHFG = -.5696*TS + .0839E-4 *TS**2 + 0.0927E-7 *TS**3.0+1352.3
  XHV=XHL+XHFG
  RETURN
  END

```

```

SUBROUTINE AAA3(X,CV,VG,A3,AAO,DAAO,ROG)
  IMPLICIT REAL*8(A-H,O-Z)
  COMMON BA,VLB,AA,RNC
  COMMON      ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
  COMMON ISET,KKK,KOUNT,KN,KKI
  AAO = B(1) + B(2)*X +B(3)*X**2.0 +B(4)*X**3.0
  DAAO = B(2) +2.0* B(3)*X +3.0*B(4)*X**2.0
  A3 = VG *(1-CV)* DAAO/ (VG*(1-CV)*AAO)**2
  RETURN
  END

```

```

SUBROUTINE DERT( ROG,CV,VG,TG,P,A9,SUM2,SUM3,SUM4,SUM5,A0,A3,A11,
  A12,A13,A14,R,CPV,CPA,A2,A4,AAO,A)
  IMPLICIT REAL*8(A-H,O-Z)
  COMMON BA,VLB,AA,RNC
  COMMON      ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
  COMMON ISET,KKK
  IF(VG .LT. 0.0)GO TO 20
  R= P/(ROG* TG)

```

```

A0 = GC*(1-CV)/ROG
A4 = WNCAO/(VG**2*(1-CV) *AAO)
A11 = (1,0-CV)*VG-TG*R*A0*A4
A12 = (1+CV/(1-CV))*VG/(GC*CJ )
A1 = GW *CV +VW*(1-CV)
A = WNCAO/(VG*(1-CV)**2*AAO)
XV= GW *CV/A1
A2 = RV*VW*GW*(GW-VW)/(A1**2*(XV*VW+(1-XV)*GW)**2)
A13 = A0*WNCAO*TG*R*A3-A9*(VG**2+ROG*TG*A2*AO+TG*R*A *AO) -VG*(1-C
CV)**2*SUM2
CALL HEEV (TG,XHV,CPV,CPA )
A14 =-A9*(XHV+VG**2,0/(2*GC*CJ))/(1-CV)** 2 -SUM3-SUM4-SUM5
RETURN
20 WRITE(6,31)VG
31 FORMAT(E20.10)
STOP
END

```

```

SUBROUTINE HEEV(TG,XHV,CPV,CPA)
IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4) ,ISTA
COMMON ISET ,KKK
IF( TG .GE.1700.0 .AND. TG .LE.4500)GO TO 222
IF( TG .LT.400.0 .OR. TG .GT.4500) GO TO 221
XHV = .4304*TG +.0839E-4*TG**2.0+.0927E-7*TG**3.0-236.3161+1042.9
CPV = .4304 + .1678E-4*TG+.2781E-7*TG**2.0
CPA = .2318+ .1040E-4*TG+.7166E-8 *TG**2.0
GO TO 100
222 XHV = .3319 *TG + .0719E-3*TG**2.0-.04373E-7*TG**3-185.381+1042.9
CPV = .3319 + .1438E-3*TG- 0.1312E-7*TG**2.0
CPA = .2214 +0.3521E-4*TG -0.3776E-8*TG**2.0
GO TO 100
221 WRITE(6,99)
99 FORMAT(1H , 'TEMP-G IS OUT OF RANGE')
WRITE(6,1) TG
1 FORMAT(1H0,'TG=',E16.6)
STOP
100 CONTINUE
RETURN
END

```

```

SUBROUTINE DERP(ROG,CV,VG,TG,R,A2,A3,A4,A9,A15,A16,OP,A)
IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4) ,ISTA
OP= ROG*R*A15 +(ROG*TG*A2+TG*R*A )*A9-TG*R*A4*A16-WNCAO*A3*TG*R
RETURN
END

```

APPENDIX IV
A LISTING OF A VARIATION OF THE COMPUTER PROGRAM
FOR ZERO HARDWARE BLOCKAGE OF THE DUCT

The following computer program was programmed for the IBM 360 computer and has been used to calculate data for two-phase flow conditions with zero blockage of the ducting and also for cases where a drop-size distribution was to be simulated.

DETERMINATION OF EXHAUST GAS COOLER INLET CONDITIONS

```

IMPLICIT REAL*8(A-H,O-Z)
REAL*4 T2TW(150)
REAL*4 AREA,RAIR,RFUEL
REAL*4 ARTM,ARPH,ARW1
REAL*4 IRUN(2),WHAT(14)
COMMON /GRP/ ARTM(150),ARPH(150),ARW1(150),NPLT
COMMON / ENTH / T, TT, R, COA(7), H(6), CO2A(7), H2A(7), XN2A(7),
1 O2A(7), H2OA(7), WT(6), COB(7), CO2B(7), H2B(7), XH2B(7),
2 O2B(7), H2OB(7), C(6), A(6), P(212)
CALL ERRSET(261,256,-1,1)
WT(1) = 28.011
WT(2) = 44.011
WT(3) = 2.016
WT(4) = 28.016
WT(5) = 32.0
WT(6) = 18.016
P(201) = 11.766
P(203) = 12.260
P(205) = 12.770
P(207) = 13.298
P(209) = 13.844
P(211) = 14.408
R = 1.98726
C PRESSURE DATA 32 F TO 212 F
C
C READ (5,100) (P(I),I=32,199)
C READ (5,100) (P(I),I=200,212,2)
100 FORMAT (8D10.0)
C
C TEMPERATURE COEFFICIENTS
C
C READ (5,101) (COA(I),I=1,7), (COB(I),I=1,7), (CO2A(I),I=1,7),
1 (CO2B(I),I=1,7), (H2A(I),I=1,7), (H2B(I),I=1,7), (XN2A(I),I=1,7),
2 (XN2B(I),I=1,7), (O2A(I),I=1,7), (O2B(I),I=1,7), (H2OA(I),I=1,7),
3 (H2OB(I),I=1,7)
101 FORMAT (50I6.7/20I6.7)
C
C INPUT CONDITIONS
C 987 FORMAT (20A4)
C
C MASS FRACTIONS
C
105 FORMAT (6D12.5)
1 READ (17,END=999) C,P1,T1,TW1,VG1,VW1,TGUESS,IRUN,WHAT
READ (17) AREA,RAIR,RFUEL
TGUESS=TW1+.01*(T1-TW1)
T2MAX=TW1-.01
T2MIN=TW1+.01
IF (P1.EQ.0.AND.T1.EQ.0.) GO TO 999
NPLT=0
C
EM=0.
DO 6 I = 1,6
EM = EM + C(I) / WT(I)
6 CONTINUE
EM = 1.0 / EM
CV1 = C(6)

```

```

CNC1 = 1.0 - CV1
PV1 = (P1 * EM * CV1) / 18.
PNC1 = P1 - PV1
EMNC = (P1 * EM * CNC1) / PNC1
T=T1/ 1.8
CALL ENTHAL
SUM = 0.0
DO 7 I = 1, 5
SUM = SUM + C(I) * H(I)
/ CONTINUE
HNC1 = SUM / CNC1
C ITERATE FOR TEMPERATURE AS A FUNCTION OF ENTHALPY
C
DOTM=RAIR+RFUEL
CONSTN=((1545.*DOTM)/(28.85*144.*P1*AREA))**2
CONSTN=CONSTN/(2.*778.*32.2)
TGS1=T1/1.8
TGS2=(T1-0.5*T1)/1.8
T=TGS1
CALL ENTHAL
CALL SUMIT (C,H,CNC1,ENT1)
T=T*1.8
FUNC1=ENT1+CONSTN*T*T-HNC1
T=TGS2
CALL ENTHAL
CALL SUMIT (C,H,CNC1,ENT2)
T=T*1.8
FUNC2=ENT2+CONSTN*T*T-HNC1
401 TGIF=(TGS1+TGS2)/2.
IF (DABS(TGS2-TGS1)/TGS2).LT.0.1D-06/ GO TO 410
T=TGIF
CALL ENTHAL
CALL SUMIT(C,H,CNC1,ENT3)
T=T*1.8
FUNC3=ENT3+CONSTN*T*T-HNC1
TSTA=FUNC1*FUNC3
TSTR=FUNC2*FUNC3
IF (TSTA) 404,404,402
402 IF (TSTR) 405,405,403
403 WRITE (6,411)
411 FORMAT('O NO ROOT FOR INITIAL ENTHALPH')
GO TO 1
404 FUNC2=FUNC3
TGS2=TGIF
GO TO 401
405 FUNC1=FUNC3
TGS1=TGIF
GO TO 401
410 ENTHU=ENT3
T=TGIF*1.8
C END OF ITERATION
VG1=(DOTM*1545.*T1)/(P1*AREA*144.)
VG1=VG1/28.85
WRITE (6,413) ENTHU,T1,VG1
WRITE(9)C(6),VG1,T1,P1
413 FORMAT('O ENTHALPY = ',E12.4,' GAS TEMPERATURE = ',E12.4,
* ' GAS VELOCITY = ',E12.4)
GO TO 1
999 CONTINUE

```

 ENDFILE 9

REWIND 9

RETURN

END

 SUBROUTINE ENTHAL

IMPLICIT REAL*8(A-H,O-Z)

COMMON / ENTH / T, TT, R, COA(7), H(6), CO2A(7), H2A(7), XN2A(7),

1 O2A(7), H2OA(7), WT(6), COB(7), CD2B(7), H2B(7), XN2B(7),

2 O2B(7), H2OB(7), C(6), A(6), P(212)

3 TT=R*T*1.8

IF (T.LT.1000) GO TO 10

DO 4 J=1,6

4 A(J)=COA(J)

CALL HTRT (T,A,H(1))

H(1)=H(1)*TT/WT(1) +28488.3 *1.8/WT(1)

DO 5 J=1,6

5 A(J) = CO2A(J)

CALL HTRT (T,A,H(2))

H(2)=H(2)*TT/WT(2) +96290. *1.8/WT(2)

DO 6 J=1,6

6 A(J)=H2A(J)

CALL HTRT (T,A,H(3))

H(3)=H(3)*TT/WT(3) +2023.8 *1.8/WT(3)

DO 7 J=1,6

7 A(J)=XN2A(J)

CALL HTRT (T,A,H(4))

H(4)=H(4)*TT/WT(4) +2072.3 *1.8/WT(4)

DO 8 J=1,6

8 A(J)=O2A(J)

CALL HTRT (T,A,H(5))

H(5)=H(5)*TT/WT(5) +2074.7 *1.8/WT(5)

DO 9 J=1,6

9 A(J)=H2OA(J)

CALL HTRT (T,A,H(6))

H(6)=H(6)*TT/WT(6) +60164.7 *1.8/WT(6)

HV2=H(6)

GO TO 17

10 DO 11 J=1,6

11 A(J)=COB(J)

CALL HTRT (T,A,H(1))

H(1)=H(1)*TT/WT(1) +28488.3 *1.8/WT(1)

DO 12 J=1,6

12 A(J)=CO2B(J)

CALL HTRT (T,A,H(2))

H(2)=H(2)*TT/WT(2) +96290. *1.8/WT(2)

DO 13 J=1,6

13 A(J)=H2B(J)

CALL HTRT (T,A,H(3))

H(3)=H(3)*TT/WT(3) +2023.8 *1.8/WT(3)

DO 14 J=1,6

14 A(J)=XN2B(J)

CALL HTRT (T,A,H(4))

H(4)=H(4)*TT/WT(4) +2072.3 *1.8/WT(4)

DO 15 J=1,6

15 A(J)=O2B(J)

CALL HTRT (T,A,H(5))

H(5)=H(5)*TT/WT(5) +2074.7 *1.8/WT(5)

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```
DO 16 J=1,6
16 A(J)=H2OB(J)
CALL HTRT (T,A,H(6))
H(6)=H(6)*T/T(6)+60154.7*T.4/T(6)
HV2=H(6)
17 CONTINUE
HNC2 =0.0
DO 18 I=1,5
18 HNC2=HNC2 + C(I)*H(I)
HNC2=HNC2/(1.0-C(6))
RETURN
END
```

```
SUBROUTINE SUMIT(C,H,CN,ENTH)
IMPLICIT REAL*8 (A-H,O-Z)
DIMENSION C(1),H(1)
SUM=0.
DO 1 I=1,5
1 SUM=SUM+C(I)*H(I)
ENTH=SUM/CN
RETURN
END
```

MAIN PROGRAM FOR EXHAUST GAS SPRAY COOLER WITH ZERO BLOCKAGE

```

IMPLICIT REAL*8(A-H,O-Z)
COMMON RA
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
COMMON ISET,KKK,KOUNT
COMMON/LOW/ MICH
DIMENSION FL(10),VL(10),TS(10),D(10),Y(45),YP(45),STA(10)
103 FORMAT(1H1,'BEGIN STATION',I4,12X,12HRUN NUMBER',2A9)
M=0
200 M=M+1
BA=.25
MICH=0
READ(5,51015)ALP1,ALP2
51015 FORMAT(2A9)
READ(5,102) NSTA
KEN=0
NS1= NSTA+1
READ(5,101)(STA(I),I=1,NS1)
READ(5,102) CV,VG,IG,P
READ(5,101)(FL(I),I=1,NSTA)
READ(5,101)(VL(I),I=1,NSTA)
READ(5,101)(TS(I),I=1,NSTA)
READ(5,101)(D(I),I=1,NSTA)
READ(5,101)(B(I),I=1,4)
100 FORMAT(4E10.2)
101 FORMAT( 7E10.2)
102 FORMAT( 12)
CPL=1.0
CJ=778.0
GC= 32.2
ISET = 9
GW=29.0
VW=18.0
RV= 1.986* 778.0
ROL=62.4
TREF = 540.0
ISTA=1
NEQ=8
C THE ARRAYS USED IN DIFF ARE NOW SET UP
Y(1)=CV
Y(2)=VG
Y(3)=IG
Y(4)=P
Y(5)=FL(1)
Y(6)=VL(1)
Y(7)=TS(1)
Y(8)= D(1)
WNCAO = Y(2) *(1.0-Y(1))*Y(4)/(Y(3)*RV*((1-Y(1))/GW+Y(1)/VW) )
X=0.0
WRITE(6,103)ISTA,ALP1,ALP2
3 CONTINUE
DX=.0001
KOUNT =0
DO 1 I= 1,999999
CALL DIFF(X,Y,YP,DX,KKK,NFO,I,KEN)
IF(X,GE,STA(ISTA+1))GO TO 2
1 CONTINUE
JJJ= JJJ+1

```

```

2      CONTINUE
C      WE WILL NOW PLOT THE STATION JUST FINISHED
      ISTA= ISTA+1
      ISET = 9
      IF(ISTA .GT. NSTA)GO TO 5
      WRITE(6,103)ISTA,ALP1,ALP2
C      SET IIP ARRAYS FOR DIFFE
      Y(NEQ+1)=FL(ISTA)
      Y(NEQ+2)=VL(ISTA)
      Y(NEQ+3)=TS(ISTA)
      Y(NEQ+4)=D(ISTA)
      NEQ =NEQ+4
      KOUNT = 0
      GO TO 3
5      IF(M-1)200,222,222
222     STOP
      END

      SUBROUTINE DIFFE(X,Y,YP,DX,KK,NEQ,I,KEN)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION Y(50),YP(50),Z(50),ZP(50),ZN(50)
      CALL YFUNC(X,Y,YP,I,KEN)
      DO 1 I=1,NEQ
1      Z(I)=Y(I)+DX*YP(I)
      X=X+DX
      DX2=.5D+0#DX
      DO 5 J=2,9999
      CALL YFUNC(X,Z,ZP,J,KEN)
      K=0
      DO 40 I=1,NEQ
      ZN(I)=Y(I)+DX2*(YP(I)+ZP(I))
      IF(DABS(ZN(I)-Z(I))-1.D-05#DABS(ZN(I)))4,4,3
3      K=1
4      KK=J
      Z(I)=ZN(I)
40     CONTINUE
      IF(K)5,6,5
5      CONTINUE
      WRITE(6,99)(ZN(I),I=1,NEQ)
      WRITE(6,90)KK
90     FORMAT(I5)
99     FORMAT(4F20,10)
      STOP
6      DO 7 I=1,NEQ
7      Y(I)=Z(I)
      IF(J .GE. 3 .AND. J .LE. 5)GO TO 1212
      IF(J .LT. 3)GO TO 2020
      DX=.5#DX
      GO TO 1212
2020   DX=2.0#DX
      IF(DX .GT..01)DX=.01
1212   CONTINUE
      RETURN
      END

```

```

SUBROUTINE VFUNC(X,Y,YP,K,KEN)
IMPLICIT REAL*8(A-H,O-Z)
COMMON RA
COMMON ROL,VW,GW,GC,RV,CJ,WNCA0,B(4),ISTA
COMMON ISET,KKK,COUNT
DIMENSION Y(50),YP(50),A5(10),A6(10),A7(10),A8(10),A10(10),XVS(10)
1,XHL(10),XHFG(10),BARH(10),DM(10),XK(10),CD(10)
2,XHV(10),RE(10),SCF(10),ABN(10),PVF(10),FNU(10)
PIE = 3.1416
SUM1=0.
SUM2=0.
SUM3=0.
SUM4=0.
SUM5=0.
CALL ROGEE(ROG,Y(1),Y(2),X)
DO 1 I=1,ISTA
J= 8+4*(I-1)
A5(I) = PIE*ROL*Y(J)**2 /2.0
CALL SUB1(Y(1),Y(3),Y(4),Y(J-1),XVS(I),XV,XK(I),Y(J),DM(I),CD(I),A
1ARH(I),Y(J-2),Y(2),RE(I),SCF(I),ABN(I),PVF(I),FNU(I))
YP(J)= XK(I)* PIE *Y(J)**2 *(XV-XVS(I))*VW/(A5(I)*Y(J-2)*(1-XVS(I)
1))
A6(I)= YP(J)
YP(J-2)=((PIE/8.0)*ROG*(Y(J)**2 )*CD(I)*DARS(Y(2)-Y(J-2))*(Y(2)-
1Y(J-2))- ( Y(J-2)-Y(2))*Y(J-2)*A5(I)*A6(I))/(DM(I)*Y(J-2))
A7(I)=YP(J-2)
YP(J-3)= A5(I)*A6(I)*Y(J-3)/DM(I)
A8(I)=YP(J-3)
SUM1=SUM1+A5(I)*A6(I)*Y(J-3)/DM(I)
CALL HL1(Y(J-1),XHL(I),XHFG(I),XHV(I))
YP(J-1) = BARH(I)* PIE * Y(J)**2 *(Y(3)-Y(J-1))/( DM(I)*Y(J-2))-
CXHL(I)*A5(I)*A6(I)/DM(I)+A5(I)*A6(I)*XHV(I)/DM(I)
A10(I)= YP(J-1)
SUM2 = SUM2 + Y(J-2)*A8(I)+ Y(J-3)*A7(I)
SUM3 = SUM3 +A8(I)*(XHL(I)+Y(J-2)**2 /12.0*GC*CJ)
SUM4 = SUM4 +Y(J-3)* Y(J-2)*A7(I)/(GC*CJ)
SUM5 = SUM5 + Y(J-3)*A10(I)
1 CONTINUE
YP(1) = -SUM1*(1-Y(1))*2
A9=YP(1)
CALL AAA3(X,Y(1),Y(2),A3,AA0,DAA0,ROG)
CALL DERT( ROG,Y(1),Y(2),Y(3),Y(4),A9,SUM2,SUM3,SUM4,SUM5,A0,A3,A1
C1,A12,A13,A14,R,CPV,CPA,A2,A4,AA0,A)
YP(3) = (A11*A14-A12*A13)/(A11*(CPV*Y(1)/(1-Y(1))+CPA)-A12*A0*ROG*R
C)
A15= YP(3)
YP(2) = (A13-A0*ROG*R*A15)/A11
A16=YP(2)
CALL DERT(ROG,Y(1),Y(2),Y(3),R,A2,A3,A4,A9,A15,A16,DP,A)
YP(4) = DP
IF(K.GT. 1) GO TO 3
IF(X.EQ. 0.0)GO TO 9999
IF(X.LT. RA)GO TO 3
BA=BA+.25
9999 WRITE(6,4)X
WRITE(6,87654)KEN
87654 FORMAT(130)
WW= Y(1)/(1.0-Y(1))
4 FORMAT(1H0,'X=',F16.10)
WRITE(6,5) AA0,WW

```

```

5      FORMAT(1H0,'A/A0=',G16.6,5X,'WV/WNC=',G16.6)
      WRITE(6,6) Y(1),Y(2),Y(3),Y(4)
6      FORMAT(1H0,'CV=',G20.9,5X,'VG=',G20.9,5X,'TG=',G20.9,5X,'P=',G20.9
1)
      DO 10 I= 1,ISTA
      J=R+4*(I-1)
      WRITE(6,8) Y(J-3),Y(J-2),Y(J-1),Y(J)
8      FORMAT(1H0,'FL=',G20.9,5X,'VL=',G20.9,5X,'TS=',G20.9,5X,'D=',G20.9
1)
      WRITE(6,9) YP(J-3),YP(J-2),YP(J-1),YP(J)
      WRITE(6,989)CD(I),BARH(I),RE(I),SCF(I)
989    FORMAT(1H0,'CD=',G16.6,5X,'BARH=',G16.6,5X,'RE=',G16.6,5X,'SCF=',
1G16.6)
      WRITE(6,899)ARN(I),PVF(I),FNU(I)
899    FORMAT(1H0,'ARN=',G16.6,5X,'PVF=',G16.6,5X,'FNU=',G16.6)
10     CONTINUE
9      FORMAT(1H0,'FLP=',G16.6,5X,'VLP=',G16.6,5X,'TSP=',G16.6,5X,'DP=',G
116.6)
      WRITE(6,7) YP(1),YP(2),YP(3),YP(4)
7      FORMAT(1H0,'CVP=',G16.6,5X,'VGP=',G16.6,5X,'TGP=',G16.6,5X,'PP=',G
116.6)
1001   CONTINUE
1000   CONTINUE
3      CONTINUE
2      FORMAT(1H,'7E16.6)
      KEN=KEN+1
      RETURN
      END

```

```

SUBROUTINE SUB1(CV,TG,P,TS,XVS,XV,XK,D,DM,CD,BH,VL,VG,RE,SCF,ARN,
1PVE,FNII)

```

~~IMPLICIT REAL*8(A-H,N-Z)~~

COMMON RA

COMMON RDL,VW,GW,GC,RV,CJ,WNCAO,B(4) ,ISTA

COMMON ISFT, KKK

$$XV = (CV/VW) / (CV/VW + (1-CV)/GW)$$
$$X2 = -9.06 * (1 - .5696 * TS + .0839E-4 * TS^2 + .0927E-7 * TS^3 + 1352.31 / TS - 1.4525744)$$
$$x_1 = 672,0/15$$
$$PVTS = 2117.0 * X1 ** 5.19 * \exp(X2)$$

XVS = PVT S/P

$$XVF = (XVS + XV) / 2.0$$
$$XM = XVF + VW + (1 - XVF) * GW$$
$$TFI = (TS + TG) / 2.0$$
$$FMV = (10.6E-7 * DSORT(TFI)) / (1 + 1538.0 / TFI)$$
$$FMNC = 7.5E-7 * DSQRT(TFI) / (1 + 216.0 / TFI)$$
$$FM = XVF * FMV + (1 - XVF) * FMNC$$
$$DAR = (1.5375/P) * (TFI/491.0) ** 2.334$$
$$RGF = P * XM / (RV * TF)$$
$$SCF = FM / (RIF \#1) \Delta B)$$
$$R_F = 0.895 (V_G - V_L) * R_{DF} * D / F_M$$
$$\Delta H_N = 2.0 + 0.6 * \ln \text{SQRT}(\text{RE}) * \text{SCF} ** 0.3333$$
$$XK = ARN * FM / (1 * SCF * XM)$$
$$DM = D \times 3 \quad 3.1416 \times RDL / 6.0$$
$$C.D = 24.0 / RF * (1.0 + .15 * RF * .687)$$

```
IF( TFI .GT.1700.0 .AND. TFI .LT.4500.0) GO TO 221
```

```
IF (TF1 .LT. 400.0 .OR. TF1 .GT. 4500) GO TO 222
```

$$CPVF = .4304 + .1678E-4 * TFI + 0.2781E-7 * TFI ** 2.0$$
$$C_{PNCr} = 0.2318 + 0.1040E-4 * TFI + 0.7166E-8 * TFI**2$$

```

      GO TO 102
221  CPVF = .3319 + 0.1438E-3 *TFI - 0.1312E-7 *TFI**2.0
      CPNCF = 0.2214 + 0.3521E-4*TFI - 0.3776E-8 *TFI**2
      GO TO 102
222  WRITE(6,101)
101  FORMAT(1H,'TEMP-FI-IS OUT OF RANGE')
      WRITE(6,122) TG,TS
122  FORMAT(1H0,'TG=',E20.6, 5X,'TS=',E20.6)
      STOP
102  CONTINUE
      FKV = 0.432*FMV
      FKNC = 0.257*(0.115+ 5.17*CPNCF)* FMNC
      FK = XVF*FKV + (1-XVF)* FKNC
      CPF = XVF*(VW/XM)*CPVF + (1-XVF)*CPNCF*GW/XM
      PVF = CPF *FM/FK
      FNU = 2+.60 *DSQRT(RE) * PVF** 0.3333
      RH = FNU *FM *CPF/(D*PVF)
      RETURN
      END

      SUBROUTINE RRGFE(ROG,CV,VG,X)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON RA
      COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,B(4),ISTA
      AAO = R(1) + R(2)*X + R(3)*X**2.0 + B(4)*X**3.0
      ROG = WNCAO/((1-CV)*VG*AAO)
      RETURN
      END

      SUBROUTINE HLI(TS,XHL,XHFG,XHV)
      IMPLICIT REAL*8(A-H,O-Z)
      XHL = TS - 540.0
      XHFG = -.5696*TS + .0839E-4 *TS**2 + 0.0927E-7 *TS**3.0 + 1352.3
      XHV = XHL + XHFG
      RETURN
      END

      SUBROUTINE AAA3(X,CV,VG,A3,AAO,DAAO,ROG)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON RA
      COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,R(4),ISTA
      AAO = R(1) + R(2)*X + R(3)*X**2.0 + R(4)*X**3.0
      DAAO = R(2) + 2.0* R(3)*X + 3.0*R(4)*X**2.0
      A3 = VG*(1-CV)* DAAO/ (VG*(1-CV)*AAO)**2
      RETURN
      END

      SUBROUTINE DFRT(ROG,CV,VG,TG,P,A9,SUM2,SUM3,SUM4,SUM5,A0,A3,A11,
1A12,A13,A14,R,CPV,CPA,A2,A4,AAO,A)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON RA
      COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,R(4),ISTA
      COMMON ISST,KKK
      IF(VG .LT. 0.0)GO TO 20
      R = P/(ROG* TG)

```

```

A0 = GC*(1-CV)/ROG
A4 = WNCAO/(VG**2*(1-CV)*AAO)
A11 = (1.0-CV)*VG-TG*R*A0*A4
A12 = (1+CV/(1-CV))*VG/(GC*CJ)
A1 = GW*CV+VW*(1-CV)
A = WNCAO/(VG*(1-CV)**2*AAO)
XV = GW*CV/A1
A2 = RV*VW*GW*(GW-VW)/(A1**2*(XV*VW+(1-XV)*GW)**2)
A13 = A0*WNCAO*TG*R*A3-A9*(VG**2+ROG*TG*A2*AO+TG*R*A*AO)-VG*(1-C
CV)**2*SJM2
CALL HEEV(TG,XHV,CPV,CPA)
A14 = -A9*(XHV+VG**2.0/(2*GC*CJ))/(1-CV)**2 -SUM3-SUM4-SUM5
RETURN
20 WRITE(6,31)VG
31 FORMAT(E20.10)
STOP
END

```

```

SUBROUTINE HEEV(TG,XHV,CPV,CPA)
IMPLICIT REAL*8(A-H,O-Z)
COMMON RA
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,R(4),ISTA
COMMON ISET,KKK
IF(TG.GE.1700.0.AND.TG.LE.4500)GO TO 222
IF(TG.LE.400.0.OR.TG.GT.4500)GO TO 221
XHV = .4304*TG+.0839E-4*TG**2.0+0.0927E-7*TG**3.0-236.3161+1042.9
CPV = .4304+.1678E-4*TG+.2781E-7*TG**2.0
CPA = .2318+.1040E-4*TG+.7166E-8*TG**2.0
GO TO 100
222 XHV = .3319*TG+.0719E-3*TG**2.0-.04373E-7*TG**3-185.381+1042.9
CPV = .3319+.1438E-3*TG-0.1312E-7*TG**2.0
CPA = .2214+.0.3521E-4*TG-0.3776E-8*TG**2.0
GO TO 100
221 WRITE(6,99)
99 FORMAT(1H,'TEMP-G IS OUT OF RANGE')
WRITE(6,1)TG
1 FORMAT(1H0,'TG=',E16.6)
STOP
100 CONTINUE
RETURN
END

```

```

SUBROUTINE DERP(ROG,CV,VG,TG,R,A2,A3,A4,A9,A15,A16,DP,A)
IMPLICIT REAL*8(A-H,O-Z)
COMMON RA
COMMON ROL,VW,GW,GC,RV,CJ,WNCAO,R(4),ISTA
DP = ROG*R*A15+(ROG*TG*A2+TG*R*A* )*A9-TG*R*A4*A16-WNCAO*A3*TG*R
RETURN
END

```