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

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

June 1972

ENGINE TEST FACILITY
ARNOLD ENGINEERING DEVELOPMENT CENTER
AIR FORCE SYSTEMS COMMAND
ARNOLD AIR FORCE STATION, TENNESSEE
NOTICES

When U.S. Government drawings, specifications, or other data are used for any purpose other than a definitely related Government procurement operation, the Government thereby incurs no responsibility nor any obligation whatsoever, and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data, is not to be regarded by implication or otherwise, or in any manner licensing the holder or any other person or corporation, or conveying any rights or permission to manufacture, use, or sell any patented invention that may in any way be related thereto.

Qualified users may obtain copies of this report from the Defense Documentation Center.

References to named commercial products in this report are not to be considered in any sense as an endorsement of the product by the United States Air Force or the Government.
A KINETIC MODEL FOR TWO-PHASE FLOW IN HIGH TEMPERATURE EXHAUST GAS COOLERS

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.
<table>
<thead>
<tr>
<th>KEY WORDS</th>
<th>LINK A</th>
<th>LINK B</th>
<th>LINK C</th>
</tr>
</thead>
<tbody>
<tr>
<td>test facilities</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>exhaust gases</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>spray evaporation</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>two-phase flow</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>coolers</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>scale model</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>humidification</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>cooling systems</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>air cooling</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
A KINETIC MODEL FOR TWO-PHASE FLOW
IN HIGH TEMPERATURE EXHAUST GAS COOLERS

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

Grateful acknowledgement is made to G. W. Lewis of the Central Computer Operations for his assistance in programming the IBM 360 computer. Acknowledgement is also made to C. E. Peters, ETF Special Projects Section, and H. K. Clark, ETF Facility Support Branch, for their valuable contributions to the technical program.

This technical report has been reviewed and is approved.

BILLY V. CLARK
Lt Colonel. USAF
Chief, Research and Development Division
Directorate of Technology

R. O. DIETZ
Acting Director
Directorate of Technology
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

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

## 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

Preceding page blank
Figures

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


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
A LISTING OF THE COMPUTER PROGRAM AND THE REQUIRED AUXILIARY EQUATIONS FOR AN EXHAUST GAS COOLER .................................................. 48

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

$A_o$ Cross-sectional area of cooler at $x = 0$, ft$^2$

$C_D$ Drag coefficient

$c_f$ Specific heat of liquid water, Btu/lbm - °R

$c_p$ Specific heat, Btu/lbm - °R

$C_v$ Mass fraction of vapor, $C_v = \frac{\rho_v}{\rho_v + \rho_{nc}}$

$D$ Particle diameter, ft

$D_{ab}$ Diffusion coefficient, ft$^2$/sec

$e_d$ Internal energy of droplet, Btu

$f_f$ Ratio of liquid flow rate to noncondensible gas flow rate, $\dot{m}_f/\dot{m}_{nc}$

$h$ 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
\( \text{Nu} \) Nusselt number for heat transfer
\( \text{Nu}_{ab} \) Nusselt number for mass transfer
\( P \) Pressure, lbf/ft\(^2\)
\( \text{Pr} \) Prandtl number
\( R \) Gas constant, ft-lbf/lbm-°R
\( \text{Re} \) Reynolds number
\( R_U \) Universal gas constant, ft-lbf/1b-mole-°R
\( \text{Sc} \) Schmidt number
\( T \) Temperature, °R
\( t \) Time, sec
\( V \) Velocity, ft/sec
\( x \) Distance along cooler, ft
\( \ddot{x} \) Mole fraction
\( \delta \) Incremental distance, ft
\( \mu \) Dynamic viscosity, lbm/ft-sec
\( \rho \) Density, lbm/ft\(^3\)
\( \sigma \) Surface tension, lbf/ft
\( \omega \) 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°F (maximum temperature of a turbojet engine exhaust gas) to approximately 550°F. 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.
SECTON 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 droplet.
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 $\delta x$ is

$$\dot{m}_V + \dot{m}_L + \dot{m}_{nc} = \dot{m}_V + \frac{\text{d}\dot{m}_V}{\text{d}x} \delta x + \dot{m}_L + \frac{\text{d}\dot{m}_L}{\text{d}x} \delta x + \dot{m}_{nc} + \frac{\text{d}\dot{m}_{nc}}{\text{d}x} \delta x \quad (1)$$

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

$$\frac{\text{d}\dot{m}_V}{\text{d}x} + \frac{\text{d}\dot{m}_L}{\text{d}x} = 0 \quad (2)$$

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 ($\delta x$) apart.
\[\dot{m}_{nc} \left[ h_{nc} + \left( V_{nc}^2 / 2 \right) \right] + \dot{m}_v \left[ h_v + \left( V_{nc}^2 / 2 \right) \right] + \dot{m}_k \left[ h_k + \left( V_k^2 / 2 \right) \right]
\]
\[= \left[ \dot{m}_{nc} + \left( d\dot{m}_{nc}/dx \right) \delta x \right] \left[ h_{nc} + \left( dh_{nc}/dx \right) \delta x \right]
\]
\[+ \left[ V_{nc} + \left( dV_{nc}/dx \right) \delta x \right]^2 / 2 \right] \left[ \dot{m}_v + \left( d\dot{m}_v/dx \right) \delta x \right] \left[ h_v + \left( dh_v/dx \right) \delta x \right]
\]
\[+ \left[ \dot{m}_k + \left( d\dot{m}_k/dx \right) \delta x \right] \left[ h_k + \left( dh_k/dx \right) \delta x \right]
\]
\[+ \left[ V_k + \left( dV_k/dx \right) \delta x \right]^2 / 2 \right] \right)
\]

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

\[\frac{dh_{nc}}{dx} + V_{nc} \frac{dV_{nc}}{dx} + \left( \frac{C_v}{1 - C_v} \right) \frac{dh_v}{dx} + \frac{dV_{nc}}{dx} \left[ \frac{h_v + \left( V_{nc}^2 / 2 \right)}{1 - C_v} \right] \frac{dC_v}{dx}
\]
\[+ f_k \left( \frac{dh_k}{dx} + V_k \frac{dV_k}{dx} \right) + \left( h_k + \frac{V_k^2}{2} \right) \frac{df_k}{dx} = 0 \]

\[ (8) \]

3.3 EQUATION FOR THE CONSERVATION OF MOMENTUM FOR ONE INJECTION STATION

The momentum equation expressing the total momentum passing two planes (\( \delta x \)) apart is:

\[\dot{m}_{nc} V_{nc} + \dot{m}_v V_{nc} + \dot{m}_k V_k + PA = \left[ \dot{m}_{nc} + \left( d\dot{m}_{nc}/dx \right) \delta x \right] \left[ V_{nc} + \left( dV_{nc}/dx \right) \delta x \right]
\]
\[+ \left[ \dot{m}_v + \left( d\dot{m}_v/dx \right) \delta x \right] \left[ V_{nc} + \left( dV_{nc}/dx \right) \delta x \right]
\]
\[+ \left[ \dot{m}_k + \left( d\dot{m}_k/dx \right) \delta x \right] \left[ V_k + \left( dV_k/dx \right) \delta x \right]
\]
\[+ \left[ P + \left( dP/dx \right) \delta x \right] \left[ A + \left( dA/dx \right) \delta x \right] \]

\[ (9) \]
Expanding and simplifying give

\[ \frac{\dot{m}_{nc}(dV_{nc}/dx)}{dx} + V_{nc}(\frac{d\dot{m}_{nc}}{dx}) + \frac{\dot{m}_{v}(dV_{v}/dx)}{dx} + V_{nc}(\frac{d\dot{m}_{v}}{dx}) \]

\[ + \frac{\dot{m}_{f}(dV_{f}/dx)}{dx} + V_{f}(\frac{d\dot{m}_{f}}{dx}) + P(\frac{dA}{dx}) + A(\frac{dP}{dx}) = 0 \]  

(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

\[ \frac{dV_{nc}}{dx} + \frac{C_{v}}{1 - C_{v}} \frac{dV_{nc}}{dx} + \frac{V_{nc}}{(1 - C_{v})^{2}} \frac{dC_{v}}{dx} + \frac{dV_{f}}{dx} + V_{f} \frac{df}{dx} + \frac{P}{\dot{m}_{nc}} \frac{dA}{dx} \]

\[ + \frac{A}{\dot{m}_{nc}} \frac{dP}{dx} = 0 \]

(11)

Multiplying by \( V_{nc}(1 - C_{v})^{2} \) and assuming that the change in area over the increment \( \delta x \) is negligible give

\[ (1 - C_{v})V_{nc} \frac{dV_{nc}}{dx} + V_{nc}^{2} \frac{dC_{v}}{dx} + (1 - C_{v})^{2} V_{nc} \left[ \frac{dV_{f}}{dx} + V_{f} \frac{df}{dx} \right] \]

\[ + \frac{(1 - C_{v})^{3}}{\rho_{nc}} \frac{dP}{dx} = 0 \]  

(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

\[ \frac{d\dot{m}_{v}}{dx} + \sum_{n} \frac{d\dot{m}_{f}}{dx} = 0 \]  

(13)
Equation (8) becomes

\[
\frac{dh_{nc}}{dx} + V_{nc} \frac{dv_{nc}}{dx} + \frac{C_v}{1 - C_v} \left( \frac{dh_v}{dx} + V_{nc} \frac{dV_{nc}}{dx} \right) + \left[ h_v + \left( \frac{V_{nc}}{1 - C_v} \right)^2 \right] \frac{dC_v}{dx} + \sum_n \left[ f_{l_i} \left( \frac{dh_{l_i}}{dx} + V_{l_i} \frac{dV_{l_i}}{dx} \right) \right] + \sum_n \left[ \left( h_{l_i} + \frac{V_{l_i}}{2} \right) \frac{df_{l_i}}{dx} \right] = 0
\]  

(14)

and Eq. (12) becomes

\[
(1 - C_v) V_{nc} \frac{dV_{nc}}{dx} + V_{nc} \frac{dC_v}{dx} + \left( 1 - C_v \right)^2 V_{nc} \sum_n \left[ f_{l_i} \frac{dV_{l_i}}{dx} + V_{l_i} \frac{df_{l_i}}{dx} \right] + \frac{(1 - C_v)^2}{\rho_{nc}} \frac{dP}{dx} = 0
\]  

(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

\[
f_{l_A} = (AA)(f_{l_1} + f_{l_2} + f_{l_3} + \cdots)
\]

(16)

\[
V_{l_A} = \frac{V_{l_1} f_{l_1} + V_{l_2} f_{l_2} + V_{l_3} f_{l_3} + \cdots}{\sum_n f_{l_i}}
\]

(17)
The properties of the liquid that strikes the piping will be

\[ V_B = 1.0 \] (20)

\[ T_{SB} = T_{SA} \] (21)

\[ D_B = \frac{13\sigma}{\rho g \left(V_{nc} - V_B\right)^2} \] (22)

The velocity \( V_B \) is arbitrarily set at a small value but not zero because it appears in the denominator of several calculations; \( T_{SB} \) is assumed equal to \( T_{SA} \) 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
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

\[
\omega_{vi}^{(m)} = k_i \frac{\bar{x}_{vi} - \bar{x}_v}{\pi D_i^2 \frac{M_{vi}}{1 - \bar{x}_{vs_i}}}
\]  

(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

\[
\omega_{vi}^{(m)} M_v = k_i \frac{\bar{x}_{vi} - \bar{x}_v}{\pi D_i^2 \frac{M_{vi}}{1 - \bar{x}_{vs_i}}}
\]  

(24)

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

\[
\frac{dM_{di}}{dt} = k_i \frac{\bar{x}_{vi} - \bar{x}_v}{\pi D_i^2 \frac{M_{vi}}{1 - \bar{x}_{vs_i}}}
\]  

(25)

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

\[
V_{li} \frac{dM_{di}}{dx} = k_i \frac{\bar{x}_v - \bar{x}_{vs_i}}{\pi D_i^2 \frac{M_{vi}}{1 - \bar{x}_{vs_i}}}
\]  

(26)

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

\[
\bar{x}_{vs_i} = \frac{P_{vs_i}}{P}
\]  

(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 $\ddot{x}_{vs}$ 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 ($\ddot{x}_v$) of the free stream is

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

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

$$\frac{df_i}{dx} = \frac{f_i}{M_{d_i}} \frac{dM_{d_i}}{dx}$$

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

$$V_{f_i} \frac{d\epsilon_{d_i}}{dx} = h \pi D_i^2 (T_g - T_{s_i}) + h_{v_g} V_{f_i} \frac{dM_{d_i}}{dx}$$

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

$$\frac{d}{dx} \left( h_{f_i} \dot{m}_{f_i} \right) = \frac{\dot{m}_{f_i}}{M_{d_i}} \frac{d\epsilon_{d_i}}{dx}$$
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,

$$e_{d_i} = c_L(T_s - T_{r_i})$$

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,

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

which may be simplified to

$$\left( V_{nc} - V_{\ell_i} \right) V_{\ell_i} \frac{dM_{d_i}}{dx} + M_{d_i} V_{\ell_i} \frac{dV_{\ell_i}}{dx}$$

$$= \rho g \frac{\pi D_i^2}{4} C_{D_i} \left| V_{nc} - V_{\ell_i} \right| \left( V_{nc} - V_{\ell_i} \right)$$

where $\rho_g$ is the density of the combined noncondensable gas and vapor in the stream.

The equation of state for the exhaust gas stream is

$$P = (\rho_{nc} + \rho_v)R_T T_{nc}$$

where

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

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 inputing the various drop diameters and their respective quantities \( f_i \) 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., \( f_0 \neq 0 \)), 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°F 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 (53°F) 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°F. 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:

\[
\begin{align*}
\dot{m}_{nc} &= 148 \text{ to } 153 \text{ lbm/sec} \\
T_{nc} &= 1066 \text{ to } 1568^\circ R \\
P_i &= 711 \text{ to } 909 \text{ psf} \\
V_{nc} &= 402 \text{ to } 1147 \text{ ft/sec} \\
f_f &= 2.5 \text{ to } 3.8 \\
T_s &= 536^\circ R
\end{align*}
\]

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.
REFERENCES


APPENDICES

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
Fig. 1 Arrangement of Spray Banks in T-1 Cooler

a. Section View Showing Internal Configuration

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

- \( \dot{m}_{g_1} = 155 \text{ lb/sec} \)
- \( T_{g_1} = 3568^\circ \text{R} \)
- \( V_{g_1} = 1077 \text{ ft/sec} \)

O Measured
- Predicted

a. Run No. 36-13
$\dot{n}_{g_1} = 154 \text{ lb/sec}$

$T_{g_1} = 3566^\circ\text{R}$

$V_{g_1} = 1103 \text{ ft/sec}$

- Measured
- Predicted

Fig. 3 Continued

b. Run No. 36-14
\[ \dot{m}_g = 154 \text{ lb/sec} \]
\[ T_{g_1} = 3548^\circ R \]
\[ V_{g_1} = 1148 \text{ ft/sec} \]

Measured

Predicted

---

c. Run No. 36-18

Fig. 3 Concluded
a. Run No. 36-13

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

\[ \dot{m}_{g1} = 155 \text{ lb/sec} \]
\[ P_1 = 909 \text{ psf} \]
\[ V_{g1} = 1077 \text{ ft/sec} \]
\[ T_{g1} = 3568^\circ \text{R} \]

- Measured
- Predicted
\[ \dot{n}_{g_1} = 154 \text{ lb/sec} \]
\[ P_1 = 883 \text{ psf} \]
\[ V_{g_1} = 1103 \text{ ft/sec} \]
\[ T_{g_1} = 3566^\circ R \]

O Measured
- Predicted

Spray Bank No. 4

Run No. 36-14
Fig. 4 Continued
\[ \dot{m}_g = 154 \text{ lb/sec} \]
\[ P_1 = 844 \text{ psf} \]
\[ V_{g_1} = 1148 \text{ ft/sec} \]
\[ T_{g_1} = 3548^\circ R \]

- **Measured**
- **Predicted**

*Spray Bank No. 4 5*

Axial Distance, ft

Fig. 4 Concluded

c. Run No. 36-18
Experimental Data

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

- $\dot{m}_{g1} = 155$ lb/sec
- $T_{g1} = 3568$°R
- $V_{g1} = 1077$ ft/sec
- Measured
- Predicted

Blockage, percent

Axial Distance, ft

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

\[ \hat{m}_{g_1} = 154 \text{ lb/sec} \]
\[ V_{g_1} = 1109 \text{ ft/sec} \]
\[ P_1 = 878 \text{ psf} \]

Average Values for Three Runs

Note: See Table 1b for Cooling-Water Conditions
Fig. 7 Predicted Relation between Exhaust Temperature and Specific Humidity for Three Cooling Water Flow Rates

- $\dot{m}_1 = 155$ lb/sec
- $\dot{m}_2/\dot{m}_1 = 4.2$ lbm/lbm
- $P_1 = 908$ psf
- $V_{e1} = 1077$ ft/sec

a. Run No. 36-13
b. Run No. 36-14
Fig. 7 Continued
3700
3300
2900
2500
2100
1700
1300
900
500

0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8
Specific Humidity, $m_v/m_{sc}$

Gas Temperature, $T_g$

$\dot{m}_{g1} = 154 \text{ lbm/sec}$
$
\dot{m}_{p1} / \dot{m}_{g1} = 2.4 \text{ lbm/lbm}$
$P_1 = 844 \text{ psf}$
$V_{g1} = 1148 \text{ ft/sec}$

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

\( m_{g_1} = 148 \text{ lbm/sec} \)
\( V_{g_1} = 402 \text{ ft/sec} \)
\( T_{g_1} = 1095^\circ R \)

- Total Pressure
- Isentropic
- Two Phase
- Static Pressure
- Isentropic
- Two Phase

Pressure, psf
Axial Distance, ft
### TABLE I
INLET ENGINE AND COOLER CONDITIONS FOR RUNS NO. 36-13, 36-14, AND 36-18

a. Measured and Calculated Exhaust Gas Parameters

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Inlet Flow Rate</th>
<th>Inlet Air Temperature, °R</th>
<th>Cooler Inlet Pressure, P, psia</th>
<th>Inlet Area, ft²</th>
<th>Calculated Cooler Inlet Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Fuel, lbm/sec</td>
<td>Air, lbm/sec</td>
<td></td>
<td></td>
<td>Moisture Mass Fraction, Cᵥ</td>
</tr>
<tr>
<td>36-13</td>
<td>7.40</td>
<td>147.57</td>
<td>778.0</td>
<td>6.31</td>
<td>30.275</td>
</tr>
<tr>
<td>36-14</td>
<td>7.36</td>
<td>146.99</td>
<td>787.0</td>
<td>6.13</td>
<td>30.275</td>
</tr>
<tr>
<td>36-18</td>
<td>7.33</td>
<td>146.97</td>
<td>782.0</td>
<td>5.87</td>
<td>30.275</td>
</tr>
</tbody>
</table>
TABLE I (Concluded)
b. Cooling Water Flow Rate Conditions

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Station or Spray Bank</th>
<th>Distance from Entrance, ( x ), ft</th>
<th>Liquid Ratio, ( \frac{\text{lbm}_2}{\text{lbm}_1} )</th>
<th>Velocity ( V_{\ell} ), ft/sec</th>
<th>Temperature, ( T_s ), °R</th>
<th>Drop Size, ( D ), ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>36-13</td>
<td>1</td>
<td>0</td>
<td>0.524</td>
<td>70.0</td>
<td>536</td>
<td>0.20 \times 10^{-2}</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>9.25</td>
<td>0.849</td>
<td>90.0</td>
<td></td>
<td>0.59 \times 10^{-3}</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>11.50</td>
<td>0.890</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>13.75</td>
<td>0.951</td>
<td>95.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>18.25</td>
<td>0.868</td>
<td>94.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>20.50</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>22.75</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>25.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>36-14</td>
<td>1</td>
<td>0</td>
<td>0.520</td>
<td>69.5</td>
<td></td>
<td>0.20 \times 10^{-2}</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>9.25</td>
<td>0.849</td>
<td>91.5</td>
<td></td>
<td>0.59 \times 10^{-3}</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>11.50</td>
<td>0.908</td>
<td>91.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>13.75</td>
<td>0.961</td>
<td>95.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>18.25</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>20.50</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>22.75</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>25.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>36-18</td>
<td>1</td>
<td>0</td>
<td>0.531</td>
<td>71.0</td>
<td></td>
<td>0.194 \times 10^{-2}</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>9.25</td>
<td>0.887</td>
<td>92.0</td>
<td></td>
<td>0.59 \times 10^{-3}</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>11.50</td>
<td>0.939</td>
<td>91.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>13.75</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>18.25</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>20.50</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>22.75</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>25.00</td>
<td>0.0</td>
<td>90.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### 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

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Inlet Flow Rate</th>
<th>Inlet Air Temperature, °R</th>
<th>Cooler Inlet Pressure, psia</th>
<th>Inlet Area, ft²</th>
<th>Calculated Cooler Inlet Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Fuel, lbm/sec</td>
<td>Air, lbm/sec</td>
<td></td>
<td></td>
<td>Moisture Mass. Frac., C_v</td>
</tr>
<tr>
<td>36-14</td>
<td>7.36</td>
<td>146.99</td>
<td>787.0</td>
<td>6.13</td>
<td>0.05950</td>
</tr>
<tr>
<td>36-15</td>
<td>4.32</td>
<td>147.23</td>
<td>779.0</td>
<td>5.58</td>
<td>0.03584</td>
</tr>
<tr>
<td>36-16</td>
<td>0.65</td>
<td>147.23</td>
<td>781.0</td>
<td>4.95</td>
<td>0.00552</td>
</tr>
<tr>
<td>35-44</td>
<td>1.319</td>
<td>152.5</td>
<td>484.2</td>
<td>4.94</td>
<td>0.00997</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Velocity, Vg, ft/sec</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Temperature, Tg, °R</td>
</tr>
<tr>
<td>36-14</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1103.4</td>
</tr>
<tr>
<td>36-15</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>864.9</td>
</tr>
<tr>
<td>36-16</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>402.0</td>
</tr>
<tr>
<td>35-44</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>407.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1066.3</td>
</tr>
</tbody>
</table>
TABLE II (Concluded)

b. Cooling Water Flow Rate Conditions

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Station or Spray Bank</th>
<th>Distance from Entrance, x, ft</th>
<th>Liquid Ratio, ( f_L ), lbm/lbm</th>
<th>Velocity, ( V_L ), ft/sec</th>
<th>Temperature, ( T_B ), °R</th>
<th>Drop Size, ( D ), ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>36-14</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.0</td>
<td>0.52</td>
<td>69.5</td>
<td>536</td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>4.8</td>
<td>0.187</td>
<td>81.0</td>
<td></td>
<td>0.0020</td>
<td>0.00059</td>
</tr>
<tr>
<td>3</td>
<td>6.9</td>
<td>0.176</td>
<td>76.6</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>9.25</td>
<td>0.849</td>
<td>91.5</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>11.5</td>
<td>0.908</td>
<td>91.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>13.75</td>
<td>0.961</td>
<td>95.5</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>18.25</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>20.5</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>36-15</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.0</td>
<td>0.534</td>
<td>71.0</td>
<td></td>
<td>0.00187</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>4.8</td>
<td>0.191</td>
<td>81.0</td>
<td></td>
<td>0.00197</td>
<td>0.00059</td>
</tr>
<tr>
<td>3</td>
<td>6.9</td>
<td>0.179</td>
<td>76.0</td>
<td></td>
<td>0.00197</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>9.25</td>
<td>0.873</td>
<td>87.0</td>
<td></td>
<td>0.00197</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>11.5</td>
<td>0.921</td>
<td>89.0</td>
<td></td>
<td>0.00059</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>13.75</td>
<td>0.979</td>
<td>90.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>18.25</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>20.5</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>36-16</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.0</td>
<td>0.547</td>
<td>72.0</td>
<td></td>
<td>0.00194</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>4.8</td>
<td>0.195</td>
<td>81.0</td>
<td></td>
<td>0.00194</td>
<td>0.00059</td>
</tr>
<tr>
<td>3</td>
<td>6.9</td>
<td>0.186</td>
<td>77.0</td>
<td></td>
<td>0.00194</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>9.25</td>
<td>0.910</td>
<td>88.0</td>
<td></td>
<td>0.00059</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>11.5</td>
<td>0.959</td>
<td>91.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>13.75</td>
<td>1.01</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>16.00</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>18.25</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>20.5</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>35-44</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.0</td>
<td>0.175</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>4.8</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td>0.00059</td>
</tr>
<tr>
<td>3</td>
<td>6.9</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>9.25</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>10.25</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>11.5</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>13.75</td>
<td>0.0</td>
<td>60.0</td>
<td></td>
<td>0.0020</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>16.00</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>18.25</td>
<td>0.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

47
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

\[ T_{fi} = \frac{T_{s1} + T_{g}}{2} \]

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:

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

\[ C_{pf_i} = \left( \frac{\tilde{x}_{vf_i}}{M_{vf_i}} \right) \left( C_{pv_i} \right) + \left( 1 - \frac{\tilde{x}_{vf_i}}{M_{nc_i}/M_{fi}} \right) \left( C_{pnc_{fi}} \right) \]

\[ D_i = \left( \frac{6M_{di}/\pi \rho_x}{1/3} \right) \]

\[ h_i = \left( \frac{Nu_{fi}}{D_i} \right) \left( \frac{C_{pf_i}}{Pr_{fi}} \right) \]

\[ k_{fi} = \left( \frac{\tilde{x}_{vf_i} k_{vf_i}}{1 - \tilde{x}_{vf_i}} \right) k_{nc_{fi}} \]
\[
\begin{align*}
  k_{xi} &= \left( \frac{\text{Nu}_{ab_{fi}}}{\mu_{fi}} \right) \left( \frac{D_i}{\text{Sc}_{fi}} \right) \left( \frac{M_{fi}}{M_c} \right) \\
  M_{fi} &= \tilde{x}_{v_{fi}} M_v + \left( 1 - \tilde{x}_{v_{fi}} \right) M_{nc} \\
  \frac{\dot{m}_{nc}}{A_o} &= \left[ \frac{V_g (1 - C_v) P}{R_u U \left( \frac{1 - C_v}{M_{nc}} + \frac{C_v}{M_v} \right)} \right]_{x = 0} \\
  \text{Nu}_{ab_{fi}} &= 2 + 0.6 \left( \text{Re}_{fi} \right)^{1/2} \left( \text{Sc}_{fi} \right)^{1/3} \quad \text{(Ref. 10)} \\
  \text{Nu}_{fi} &= 2 + 0.6 \left( \text{Re}_{fi} \right)^{1/2} \left( \text{Pr}_{fi} \right)^{1/3} \quad \text{(Ref. 10)} \\
  \text{Pr}_{fi} &= \left( \frac{\mu_{fi}}{C_{p_{fi}}} \right) \\
  R &= \frac{R_u}{\left[ \tilde{x}_v M_v + \left( 1 - \tilde{x}_v \right) M_{nc} \right]} \\
  \text{Re}_{fi} &= \left( \frac{V_g - V_{f_{i}}} {\mu_{fi}} \right) \left( \rho_{fi} \right) \left( D_i \right) \\
  \text{Sc}_{fi} &= \left( \frac{\mu_{fi}}{\left( \rho_{fi} \right)} \left( D_{ab_{fi}} \right) \right) \\
  V_g \rho_g (1 - C_v) &= \frac{m_{nc}}{A_o} / \frac{A}{A_o} \\
  \tilde{x}_{v_{fi}} &= \frac{\tilde{x}_v + \tilde{x}_v}{2}
\end{align*}
\]
\[ \tilde{x}_v = \frac{C_v/M_v}{C_v/M_v + (1 - C_v)M_{nc}} \]

\[ \tilde{x}_{v_{si}} = \frac{P_{v_{si}}}{P} \]

\[ \mu_{fi} = \left( \tilde{x}_{v_{fi}} \right)^\mu_{v_{fi}} + \left( 1 - \tilde{x}_{v_{fi}} \right)^\mu_{nc_{fi}} \]

\[ \rho_{fi} = \frac{\left( P \right)^{M_{fi}}}{R_u T_{fi}} \]

**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,ARWI
REAL*4 IRUN(2),WHAT(14)
COMMON /ENTH/,Tv,Ti,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#2569-1,I)

WT(1) = 28.0111
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(202) = 12.260
P(203) = 12.770
P(204) = 13.298
P(205) = 13.844
P(206) = 14.409
R = 1.98726

C PRESSURE DATA 32 F TO 212 F
C READ (5,100) (P(I),I=32,199)
C READ (5,100) (P(I),I=200,212,2)
C 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)
C 101 FORMAT (50F12.7)
C
C INLET CONDITIONS
C 987 FORMAT (20A4)
C
C MASS FRACTIONS
C
C 105 FORMAT (6D12,.5)
1 READ (17,EMD=999) C,P1,T1,TW1,VG1,VM1,TGUESS,IRUN,WHAT
READ (17) AREA,RAIR,RFUEL
TGUESS=TW1+.01*TI-TW1)
T2MAX=TW1+.01
T2MIN=TW1+.01
IF (P1,EQ.0. AND T1,EQ.0.) GO TO 999
C NPLT=0
C EM=0.
DO 6 1 = 1,6
EM = EM + C(I)/WT(I)
6 CONTINUE
EM = 1.0/EM
CV1 = C(6)
CNC1 = 1.0 - CV1
PV1 = (F1 * EM * CV1) / IA,
CNC1 = PI - PV1

EMNC = (PI * EM * CNC1) / PVC1
T=TI/1.8
CALL ENTHAL
SUM = 0.0
DO T = 1, 5
SUM = SUM + C(1) * H(I)
CONTINUE
MNC1 = SUM / CNC1

ITERATE FOR TEMPERATURE AS A FUNCTION OF ENTHALPY

DO1 =AI*PI
CONSTN = ((1545.*DOTM)/(28.85*144.*PI*AREA))**2
TGSI = T1/1.8

TGSO = (T1 - 0.5*T1)/1.8
T = TGSI
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT1)
T = T1/1.8
FUNC1 = ENT1 + CONSTN * T * T * HNC1

T = TGSO
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT2)
T = T1/1.8
FUNC2 = ENT2 + CONSTN * T * T * HNC1

IF (TGSO < TGS1) GO TO 410

T = TGF
CALL ENTHAL
CALL SUMIT (C, H, CNC1, ENT3)
T = T1/1.8
FUNC3 = ENT3 + CONSTN * T * T * HNC1
TSTA = FUNC1 * FUNC3
TSTR = FUNC2 * FUNC3

IF (TSTA > 404) GO TO 401

IF (TSTA < 405) GO TO 402

WRITE (6, 411)
GO TO 1

IF (TSTA) 406, 404, 402

WRITE (6, 411)
GO TO 1

FUNCTIONS = 0
CONTINUE

END OF ITERATION

VGI = (DOTM*1545.*T1)/(PI*AREA*144.,)
VG1 = VG1/28.85
WRITE (6, 413) VGI, T1, VG1
WRITE (9) C(6), VG1, T1, PI

FORMAT (* ENTHALPY = ', E12.4,' GAS TEMPERATURE = ', E12.4,
        * GAS VELOCITY = ', E12.4)
GO TO 1
CONTINUE
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), H2O(7), WT(6), COB(7), CO2B(7), H2B(7), XN2B(7),
  2 O2B(7), H2OB(7), CB(6), A(6), P(212)
  3 TTHR T=1.8

IF (T,T1.1000) GO TO 10

DO 4 J=1,6
  4 A(J)=COA(J)
    CALL HRT (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 HRT (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 HRT (T, A+H(3))
        H(3)=H(3)*TT/WT(3) +2072.3 * 1.8/WT(3)
        DO 7 J=1,6
          7 A(J)=XN2A(J)
          CALL HRT (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 HRT (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)=H2B(J)
              CALL HRT (T, A+H(6))
              H(6)=H(6)*TT/WT(6) +80164.7 * 1.8/WT(6)
              MV2=MV2

GO TO 17

DO 10 J=1,6
  10 A(J)=COB(J)
    CALL HRT (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 HRT (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 HRT (T, A+H(3))
        H(3)=H(3)*TT/WT(3) +2072.3 * 1.8/WT(3)
        DO 14 J=1,6
          14 A(J)=XN2B(J)
          CALL HRT (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 HRT (T, A+H(5))
            H(5)=H(5)*TT/WT(5) +2074.7 * 1.8/WT(5)
DO 16 J=1,6
16 A(J)=H208(J)
   CALL HTRT (T,A,H(6))
   H(6)=H(6)+11/H1(6)*60/H4.7+1.4/H1(6)
   H2=H(6)
17 CONTINUE
   MNC2 =0.0
   DO 18 I=1,5
18 MNC2=MNC2+C(I)*H(I)
   MNC2=MNC2/11.0-C(I)
   RETURN
   END
SUBROUTINE SUMIT(C,H,CN,ENTH)
IMPLICIT REAL*8 (A-M,0-Z)
DIMENSION C(1),H(1)
SIM=0.
DO 1 I=1,5
1 SUM=SUM+C(I)*H(I)
   ENTH=SIM/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,GM,GC,RC,V,CJ,WNCAO,B(4),ISTA
COMMON ISF,XXK,KOUNT,KN,KKI
COMMON DL,VL,CL,LSA
COMMON FLM,VL,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE14,GE15,GE16,GE17,GE18
COMMON PIVS,ABSM,SCF,REL,ABN,ABU
COMMON ROF
COMMON LOL/MICH

DIMENSION Y(99),Y1(99),A5(10),A6(10),A7(10),A8(10),A10(10),XVS110
1,XML(10),XMF1G(10),BAMH1(10),DM1(10),X5(10),CD1(10)
2,XH110
DIMENSION FL(10),VL(10),TS(10),D11(10),STA(11)
103 FORMAT(1H1,'BEGIN STATION',I4,12X,12HHRUIN NUMBER',BAR)
N=0
200 N=N+1
K=1
AA=.88
VLB=1.0
I=0
B=25
KN1=1
KI=1
MICH=0
READ(5,51015,END=222)ALP
$1015 FORMAT(10A8)
READ(5,102)NSTA
NS1=NSTA+1
READ(5,101)STA(I),I=1,NS1
READ(9,END=222)CJW,TC,CP
P=144.0
READ(5,101)EL(I),I=1,NSTA
READ(5,101)VL(I),I=1,NSTAA
READ(5,101)SI(I),I=1,NSTAA
READ(5,101)DI(I),I=1,NSTAA
READ(5,101)BI(I),I=1,NSTAA
100 FORMAT(4F10.2)
101 FORMAT(12)
102 FORMAT(12)
CP1=1.0
GE1=STA(2)-.2500
GE2=GE1+.100
GE3=STA(3)-.2500
GE4=GE3+.100
GE5=STA(4)-.2500
GE6=GE5+.100
GE7=STA(5)-.2500
GE8=GE7+.100
GE9=STA(6)-.2500
GE10=GE9+.100
GE11=STA(7)-.2500
GE12=GE11+.100
GE13=STA(8)-.2500
GE14=GE13+.100
THE ARRAYS USED IN DIFE ARE NOW SET UP.

Y(1)=CV
Y(2)=VG
Y(3)=TV
Y(4)=P
Y(5)=FL(1)
Y(6)=VL(1)
Y(7)=TS(1)
Y(8)=D(1)

WNC20 = Y(2) *(1.0-Y(11))*Y(4)*/(Y(3)*RV*(1.0-Y(11)))*GW+Y(11)/VW

x=0.0
WRITE(6,103)ISTA,ALP
3 CONTINUE
Dx=.0001
KOUNT =0

IF(ISTA .EQ. 1) J=8
DO 1 11=1,9999999
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
IFIX .GE. STA(2) .AND. X .LE. GE3)KKI=8
IFIX .GE. GE3 .AND. X .LE. STA(3) KKI=16
IFIX .GE. STA(3) .AND. X .LE. GE5)KKI=16
IFIX .GE. GE5 .AND. X .LE. STA(4) KKI=24
IFIX .GE. STA(4) .AND. X .LE. GE7)KKI=24
IFIX .GE. STA(5) .AND. X .LE. GE9)KKI=32
IFIX .GE. STA(6) .AND. X .LE. GE11)KKI=40
IFIX .GE. STA(7) .AND. X .LE. GE13)KKI=48
IFIX .GE. STA(8) .AND. X .LE. GE15)KKI=56
IFIX .GE. STA(9) .AND. X .LE. GE17)KKI=64
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. 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, IX, N, K, MSTA)
K = 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(L-4) + Y(L-6)) / FLN
VL = (Y(L-1) + Y(L-3) + Y(L-4) + Y(L-7) + Y(L-8)) / FLN
DA = (Y(L-2) + Y(L-3) + Y(L-4) + Y(L-5) + Y(L-8)) / FLN
TS = (Y(L-2) + Y(L-3) + Y(L-4) + Y(L-6) + Y(L-8)) / FLN
GO TO 669
909 FLN = FLN + (YJ - 3)
VL = (YJ - 3) + Y(J - 2) / FLN
DA = DA + Y(J - 3) / FLN
TS = TS + Y(J - 3) / FLN
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. GE19 .AND. X .LE. GE19 + DX) GO TO 8000
IF(X .GE. GE21) GO TO 8001
IF(X .GE. GE23 .AND. X .LE. GE23 + DX) GO TO 8001
IF(X .GE. GE25 .AND. X .LE. GE25 + DX) GO TO 8001
IF(X .GE. GE27 .AND. X .LE. GE27 + DX) GO TO 8001
IF(X .GE. GE29 .AND. X .LE. GE29 + DX) GO TO 8001
IF(X .GE. GE31 .AND. X .LE. GE31 + DX) GO TO 8001
IF(X .GE. GE33 .AND. X .LE. GE33 + DX) GO TO 8001
IF(X .GE. GE35 .AND. X .LE. GE35 + DX) GO TO 8001
IF(X .GE. GE37 .AND. X .LE. GE37 + DX) GO TO 8001
IF(X .GE. GE39 .AND. X .LE. GE39 + DX) GO TO 8001
IF(X .GE. GE41 .AND. X .LE. GE41 + DX) GO TO 8001
IF(X .GE. GE43 .AND. X .LE. GE43 + DX) GO TO 8001
IF(X .GE. GE45 .AND. X .LE. GE45 + DX) GO TO 8001
IF(X .GE. GE47 .AND. X .LE. GE47 + DX) GO TO 8001
IF(X .GE. GE49 .AND. X .LE. GE49 + DX) GO TO 8001
IF(X .GE. GE51 .AND. X .LE. GE51 + DX) GO TO 8001
IF(X .GE. GE53 .AND. X .LE. GE53 + DX) GO TO 8001
IF(X .GE. GE55 .AND. X .LE. GE55 + DX) GO TO 8001
IF(X .GE. GE57 .AND. X .LE. GE57 + DX) GO TO 8001
IF(X .GE. GE59 .AND. X .LE. GE59 + DX) GO TO 8001
IF(X .GE. GE61 .AND. X .LE. GE61 + DX) GO TO 8001
IF(X .GE. GE63 .AND. X .LE. GE63 + DX) GO TO 8001
IF(X .GE. GE65 .AND. X .LE. GE65 + DX) GO TO 8001
IF(X .GE. GE67 .AND. X .LE. GE67 + DX) GO TO 8001
IF(X .GE. GE69 .AND. X .LE. GE69 + DX) GO TO 8001
IF(X .GE. GE71) 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. STA(IISTA+1)) GO TO 2

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 = 0.004790

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 96 J=1,2
J = JJ*4*(I-1)
GO TO 20
Z(J)*1

CONTINUE

FLA = AA*FLN
FLB = (1.0-AA)*FLN
RG = Y(1)*RVAP+(1.0-Y(1))*RNC
ROG = Y(4)/(RG*Y(3))
SIG = 0.004790
DB = (418.6*SIG)/1/ROG*(Y(2)*VLR)**2

GO TO 22

CONTINUE

Y(J-2) = VLA
Y(J-3) = FLA
Y(J-1) = TSA
Y(J) = NH
KK = I+R
NEQ = NEQ+1
KI = 0

22 CONTINUE

88 CONTINUE
CONTINUE
JJJ = JJJ + 1
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 DIFF Eq
NEO + 1 = FL (ISTA)
NEO + 2 = VL (ISTA)
NEO + 3 = TS (ISTA)
NEO + 4 = DI (ISTA)
NEO = NEO + 4
Kount = 0
GO TO 3
5
GO TO 200
222
STOP
END

SUBROUTINE DIFFE (X, Y, YP, DX, IKK, NEO, KI, MST A)
IMPLICIT REAL *8 (A-H, O-Z)
COMMON B, RLL, AA, RNC
COMMON RGL, VW, GW, GC, RV, CJ, WNGAO, B(4) , ISTA
COMMON ISET, KKK, KOUNT, KN, KKI
COMMON YL, RL, CL, TSA
COMMON FLN, VLA, DA, FLA
COMMON GE1, GE2, GE3, GE4, GE5, GE6, GE7, GE8, GE9, GE10, GE11, GE12, GE13, GE14, GE15, GE16, GE17, GE18
COMMON PUTS, DBB, SCF, RE, ABN, FNU
COMMON ROF
DIMENSION Y(99), YP(99), Z(99), ZP(99), ZNI(99)
DY = 0.0
CALL YFUNC(X, Y, YP, KI, 1, MST A, IKK, DX, DY)
120 CONTINUE
DO 1 = 1, NEO
    Z(I) = Y(I) + DX * YP(I)
1 CONTINUE
X = X + DX
DX2 = 0.5D0 * DX
DO 5 J = 2, 999
    DX = DX2
5 CONTINUE
CALL YFUNC(X, Z, ZP, KI, 1, MST A, IKK, DX, DY)
K = 0
DO 40 I = 1, NEO
    Z(I) = Y(I) + DX2 * (YP(I) + ZP(I))
40 IF (DBS(Z(I) - Z(I)) - 1.0D0 < DBS(ZNI(I))) 4, 4, 3
3 K = 1
4 KK = J
  Z(I) = ZN(I)
40 CONTINUE
  IF(KK .EQ. 5) THEN
  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 .GE. 3) GO TO 2020
  DX = .5*DX
  GO TO 1212
2020 DX = 2.0*DX
  IF(DX .GE. .01) DX = .01
1212 CONTINUE
RETURN
END

SUBROUTINE YFUNC(X, Y, YP, X, Y, MSTA, IKK, DX, DY)
  IMPLICIT REAL*8(A-H, O-Z)
COMMON BA, VLB, A, A, RN
COMMON ROL, VM, GW, GC, RV, CJ, WNC, B(4), I, STA
COMMON ISET, KKK, KOUNT, KN, KK
COMMON DL, BL, CL, TSA
COMMON FLN, VLA, FA, FLA
COMMON GE1, GE2, GE3, GE4, GE5, GE6, GE7, GER, GE9, GE10, GE11, GE12, GE13, GE14
COMMON RVTS, DB, SCF, RE, ABN, FNM
COMMON R0F
DIMENSION Y(99), YP(99), A5(10), A6(10), A7(10), A8(10), A9(10), A10(10), XVS(10, 1), XHL(10), XHF6(10), BARH(10), DM(10), XX(10), CD(10)

1 X = X + DX

9899 CONTINUE
  IF(I .EQ. 1) J = 8
  IF(I .EQ. 2) J = 16
  IF(I .EQ. 3) J = 24
  IF(I .EQ. 4) J = 32
  IF(I .EQ. 5) J = 40
  IF(I .EQ. 6) J = 48
  IF(I .EQ. 7) J = 56
  IF(I .EQ. 8) J = 64
  IF(I .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
AEDC-TR-72-89

9998 CONTINUE
8001 CONTINUE
RG=Y(1)*RVAP+(1.0-Y(1))*RNC
8002 IF(K! =0) GO TO 8003
CALL ROGE(E,ROG,Y(1),Y(2),X)
GO TO 8009
8003 ROG=Y(4)/(RG*Y(3))
8009 CONTINUE
DO 1 J=1+MSTA
J=KKI+4*(1-1)
A5(I) = PIE*ROL*Y(J)**2 /2.0
CALL SUBI(Y(I),Y(3),Y(4),Y(J-1),XVS(I),XV,X(I),Y(J),DM(I),CD(I),B
1ARM(I),Y(J-2),Y(2))
IF(XVF GT. 1.0) RETURN
YP(J) = X(I)*PIE *Y(J)**2 *(XV-XVS(I))*VW/(AS(I)*Y(J-2)**2)
A6(I) = YP(J-1)
YP(J-2) = (PIE/6.0)*ROG*(Y(J)**2 )*CD(I)*DABS(Y(2)-Y(J-2)) *(Y(2)-
1Y(J-2))-(Y(2)-Y(J-2))**2*(Y(J-2)**2)
A7(I) = YP(J-2)
YP(J-3) = A5(I)**2*A6(I)**2*(Y(J-3)*DM(I)
A8(I) = YP(J-3)
SUM1 = SUM1*A5(I)**2*A6(I)**2*(Y(J-3)*DM(I)
SUM2 = SUM2 + (Y(J-3)**2 + W(J-3)**2)**2 /2.0*GC*CJ)
SUM3 = SUM3 + (Y(J-3)**2 + W(J-3)**2)**2 /2.0*GC*CJ)
SUM4 = SUM4 + (Y(J-3)**2 + W(J-3)**2)**2 /2.0*GC*CJ)
SUM5 = SUM5 + (Y(J-3)**2 + W(J-3)**2)**2 /2.0*GC*CJ)
SUM6 = SUM6 + (Y(J-3)**2 + W(J-3)**2)**2 /2.0*GC*CJ)
CONTINUE
YP(1) = -SUM1*(1-Y(1))**2
A9=YP(1)
CALL AAA3(X,V(1),Y(2),A3,AA9,DA9,ROG)
CALL DERT(RG,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,AA9,A)
YP(3) = (A11*A14-A12*A13)/(A11*(CPV*Y(1))/(1-Y(1))*CPA-A12*A0*ROG)
A15=YP(3)
YP(2) = (A13*A0*ROG*K=A15)/A11
A16=YP(2)
CALL DERT(RG,Y(1),Y(2),Y(3),R,A2,A3,A4,A9,A15,A16,DP,A)
YP(4) = DP
1 CONTINUE
AEDC-TR-72-89

IF (X .EQ. 0.0) GO TO 9999
IF (X .LT. 84) GO TO 9
R=BA+.5
WRITE(6,4)X
WRITE(6,88888)ROF

4 FORMATA(10,4F10.6)
WRITE(6,AA0,WW)

5 FORMATA(10,14A6,5X,14W/VWC=',G16.8)
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 FORMATA(10,14C4,5X,14V=*,G20.9,5X,14T=G=*,G20.9,5X,14P=*,G20.9)

IF (X .GE. GE2 .AND. X .LE. GE3-DX) GO TO 88
IF (X .GE. GE3 .AND. X .LE. GE5-DX) GO TO 88
IF (X .GE. GE5 .AND. X .LE. GE7-DX) GO TO 88
IF (X .GE. GE7 .AND. X .LE. GE9-DX) GO TO 88
IF (X .GE. GE9 .AND. X .LE. GE11-DX) GO TO 88
IF (X .GE. GE11 .AND. X .LE. GE13-DX) GO TO 88
IF (X .GE. GE13 .AND. X .LE. GE15-DX) GO TO 88
IF (X .GE. GE15 .AND. X .LE. GE17-DX) GO TO 88
IF (X .GE. GE17 .AND. X .LE. GE19-DX) GO TO 88
DO 10 J=1,9
J=J+4*(f-1)
WRITE(6,S) Y(J-3),Y(J-2),Y(J-1),Y(J)
10 CONTINUE
GO TO 1001

88 DO 988 J=1,2
J=J+4*(f-1)
WRITE(6,R)Y(J-3),Y(J-2),Y(J-1),Y(J)
WRITE(6,G)YP(J-3),YP(J-2),YP(J-1),YP(J)
988 CONTINUE

1000 CONTINUE
IF (X .GE. GE1 .AND. X .LE. GE3-DX) GO TO 1002
IF (X .GE. GE3 .AND. X .LE. GE5-DX) GO TO 1002
IF (X .GE. GE5 .AND. X .LE. GE7-DX) GO TO 1002
IF (X .GE. GE7 .AND. X .LE. GE9-DX) GO TO 1002
IF (X .GE. GE9 .AND. X .LE. GE11-DX) GO TO 1002
IF (X .GE. GE11 .AND. X .LE. GE13-DX) GO TO 1002
IF (X .GE. GE13 .AND. X .LE. GE15-DX) GO TO 1002
IF (X .GE. GE15 .AND. X .LE. GE17-DX) GO TO 1002
GO TO 1002

1002
1010 J=J+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)
199 CONTINUE

1001 CONTINUE
1002 CONTINUE
WRITE(6,7) YP(1),YP(2),YP(3),YP(4)
8 FORMAT(1HO,*FL=*,G20.9,5X,*VL=*,G20.9,5X,TS=*,G20.9,5X,D=*,G20.9)
9 FORMAT(1HO,*FLP=*,G16.6,5X,VLP=*,G16.6,5X,TS=*,G16.6,5X,D=*,G16.6)
116.6)

7 CONTINUE
3 CONTINUE
2 FORMAT(IH,7E16.6)
RETURN
END

SUBROUTINE SUBR(JY,IG,P,T,TS,X,V,T,VL,VP)
IMPLICIT REAL*A-M,H-Z)
COMMON RA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCA0,8(4),ISTA
COMMON ISFT,KKK,KN,KNK1
COMMON DL,HL,CL,TLA,TLB
COMMON FLN,VLN,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE14,GE15,GE16,GE17,GE18
COMMON PVT,DAF,SCFE,AF,AFN
COMMON RCF9899 CONTINUE

X=CV/IV)*TS*XV*VX*DA*DM*DAH*V*VL*VG)
IMPLICIT REAL*A-H,0-Z)
COMMON RA,VLB,AA,RNC
COMMON ROL,VW,GW,GC,RV,CJ,WNCA0,8(4),ISTA
COMMON ISFT,KKK,KN,KNK1
COMMON DL,HL,CL,TLA,TLB
COMMON FLN,VLN,DA,FLA
COMMON GE1,GE2,GE3,GE4,GE5,GE6,GE7,GE8,GE9,GE10,GE11,GE12,GE13,GE14,GE15,GE16,GE17,GE18
COMMON PVT,DAF,SCFE,AF,AFN
COMMON RCF
9899 CONTINUE

RE =DARS(VG-VE)*RNF#F/PF
IF(RE <LT. 0.0)GO TO 9999
AN= 2.0+0.6*DSORT(RE) * SCF** 0.3333
XM = A3*FM/(1*SCF*XM)
DM = 0**3 * 3.1416 * ROL/CX
CO = 24.0/RF/1.0+15*RE+0.0R7
IF(TF=1.4,1.700,0.0,AN=TF1=LT.4509,0)GO TO 221
IF(TFI <LT.400.0 I.N,TF1 =LT.4500)GO TO 222
CPVF = 0.4404 +167AE-4*TF1 +0.2781E-7*TF1**2.0
CPNPC = 0.2318 + 0.104E-4*TF1 + 0.7160E-8 *TF1**2
GO TO 102
221 CPVF = 0.3319 +0.143HE-3 * TF1 -0.1312E-7 * TF1**2.0
CPNPC = 0.2214 + 0.3521E-4*TF1- 0.3775E-8 *TF1**2
GO TO 102
READ(6,101)
101 FORMAT(1H,'TEMP-FI IS OUT OF RANGE')
WRITE(6,122) TG,TS
122 FORMAT(1H,'TG=',E20.6,5X,'TS=',E20.6)
9999 CONTINUE
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*(VM/VM)*CPVF+(1-XVF)*CPNCF*G/W/VM
PVF = CPF*FM/VM
FNU = 2*60/DSORT(RE)*PVF**0.3333
RETURN
END

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

SUBROUTINE HL1(TS,XHL,XHFG,XHV)
IMPLICIT REAL*8(A-H,O-Z)
XHL = TS
XHFG = -5696*TS+0.0839*TS**2+0.0927E-7*TS**3+1352.3
XHV = XHL+XHFG
RETURN
END

SUBROUTINE A3(X,CV,VG,A3,AAA0,DAAO,ROG)
IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VM,VM,GC,RV,CJ,WNCAO,B(4),ISTA
COMMON ISET,KKK,KOUNT,KNI
AAO = B(1)+B(2)*X+B(3)*X*X+B(4)*X*X*X
AAA0 = R(1)+R(2)*X*R(3)*X*R(4)*X*X
A3 = VG*(1-CV)*AAA0/((1-CV)*VG*AAA0)**2
RETURN
END

SUBROUTINE DERT(ROG,CV,VS,TS,P,A9,SUM2,SUM3,SUM4,SUM5,A0,A3,A11)
IMPLICIT REAL*8(A-H,O-Z)
COMMON BA,VLB,AA,RNC
COMMON ROL,VM,VM,GC,RV,CJ,WNCAO,B(4),ISTA
COMMON ISET,KKK
IF(VG.LT.0.0)GO TO 20
R = P/(ROG*TS)
\[ A_0 = GC \cdot (1-CV)/R_G \]
\[ A_4 = WNCAO/(VG+2 \cdot (1-CV)) \cdot AAO \]
\[ A_{11} = (1-0-CV) \cdot VG \cdot R \cdot AO \cdot A_4 \]
\[ A_{12} = (1+CV/(1-CV)) \cdot VG/(GC+CJ) \]
\[ A_1 = GW \cdot CV + VW \cdot (1-CV) \]
\[ A = WNCAO/(VG+2 \cdot (1-CV)) \cdot AAO \]
\[ A_2 = RV\cdot VW\cdot GM/(GW-VW)/(1+2 \cdot (1-CV) \cdot GW) \]
\[ A_{13} = A_0 \cdot WNCAO \cdot R \cdot A_3 - A_9 \cdot (VG+2 \cdot R \cdot A_2 \cdot AO + TG \cdot R \cdot AO) - VG \cdot (1-CV) \]
\[ CV = \frac{2 \cdot \text{SUM2}}{\text{SUM3} \cdot \text{SUM4} \cdot \text{SUM5}} \]
\[ \text{CALL} \text{ MEEV} (TG, XHV, CPV, CPA) \]
\[ A_{14} = \frac{XHV + VG \cdot 2/(OC \cdot CJ)}{(1-CV) \cdot 2} \]
\[ \text{RETURN} \]
\[ \text{WRITE}(6,31) VG \]
\[ \text{STOP} \]
\[ \text{END} \]

SUBROUTINE MEEV(TG, XHV, CPV, CPA)
IMPLICIT REAL*8(A-H, I-O-Z)
COMMON RA, VB, AA, RN
COMMON ROL, VW, GW, GC, RV, CJ, WNCAO, B4, ISTA
COMMON ISET, KKK
IF (TG .GE. 1700.0 .AND. TG .LE. 4500.0) GO TO 222
IF (TG .LT. 400.0 .OR. TG .GT. 4500.0) GO TO 221
XHV = 0.304 * TG + 0.0839E-4 * TG ** 2 + 0.0427E-7 * TG ** 3 + 0.236 * 3161 + 1042.9
CPV = 0.3304 + 0.0878E-4 * TG + 0.278E-7 * TG ** 2
CPA = 0.3319 + 0.0719E-3 * TG + 0.1312E-7 * TG ** 2
GO TO 100
222 XHV = 0.3319 * TG + 0.0719E-3 * TG ** 2 + 0.04373E-7 * TG ** 3 - 0.1312E-7 * TG ** 2
CPV = 0.3319 + 0.0438E-3 * TG - 0.1312E-7 * TG ** 2
CPA = 0.2714 + 0.0351E-4 * TG - 0.0377E-8 * TG ** 2
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 DERG(ROG, CV, VG, TG, R, A2, A3, A4, A9, A15, A16, DP, A)
IMPLICIT REAL*8(A-H, I-O-Z)
COMMON RA, VB, AA, RN
COMMON ROL, VW, GW, GC, RV, CJ, WNCAO, B4, ISTA
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 ARTN,APRN,ARM1
REAL*4 IRUN(12),WHAT(14)

COMMON /ENTH/,TT,T,T,T,R,COA(7),H(6),CO2A(7),H2A(7),XN2A(7),
1 OZA(7),M2O8(7),WT(6),CO8(7),CO2R(7),H2R(7),X12R(7),
2 O2R(7),M2OR(7),C16(1),A16(1),P(212)
 CALL ERASET(261,256,-1,1)

WT(1) = 24.011
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) = 13.244
P(209) = 13.844
P(211) = 14.404
R = 1.98726

C PRESSURE DATA 32 F TO 212 F
C
READ (5,100) (P(I),I=1,199)
READ (5,100) (P(I),I=200,212,2)
100 FORMAT (AN(10),0)

C TEMPERATURE COEFFICIENTS
C
READ (5,101) (COA(I),I=1,7), (CO8(I),I=1,7), (CO2A(I),I=1,7),
1 (CO2R(I),I=1,7), (H2A(I),I=1,7), (H2R(I),I=1,7), (XN2A(I),I=1,7),
2 (X12R(I),I=1,7), (OZA(I),I=1,7), (O2R(I),I=1,7), (M2O8(I),I=1,7),
3 (H20R(I),I=1,7)
101 FORMAT (5D16.7,2D16.7)

C INPUT CONDITIONS
C
987 FORMAT (20A4)

C MASS FRACTIONS
C
105 FORMAT (6D12.5)
1 READ (17,EN=999) C,P1,T1,TW1,VTG1,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=5
C
EM=0.
DO 6 I = 1,6
EM = EM + C(I)/WT(I)
6 CONTINUE
EM = 1.0/FM
CV1 = C(1)
AEDC-TR-72-89

CNC1 = 1.0 - CV1
PV1 = (P1 + EM * CV1) / 1A,
PN1 = P1 - PV1

EMNC = (P1 + EM * CNC1) / PN1
T = T1 / 1A
CALL ENTHAL
SUM = 0.0
DO 7 I = 1, 5
SUM = SUM + C(I) * H(I)
7 CONTINUE

HNC1 = SUM / CNC1

ITERATE FOR TEMPERATURE AS A FUNCTION OF ENTHALPY

DOTM=RAIR+RFUEL
CONSTN=((1545.*DOTM)/(128.85*144.*PI*AREA))**2
CONSTN=CONSTN/(12.778.*32.2)
TGS1=T1/1A
T=0.5*T1/T1/1A
TGS2=T1/1A
CALL ENTHAL
CALL SUMIT (C,H,CNC1,ENT1)
T=T1/1A
FUNC1=ENT1+CONSTN*T*T-HNC1
T=TGS2
CALL ENTHAL
CALL SUMIT (C,H,CNC1,ENT2)
T=T1/1A
FUNC2=ENT2+CONSTN*T*T-HNC1

T=0.5*(TGS1+TGS2)/2.

IF TOARST TTGS2= TGS1/7TGS2 LT 0.10=08 GO TO 410

T=TGS1
CALL ENTHAL
CALL SUMIT (C,H,CNC1,ENT3)
T=T1/1A
FUNC3=ENT3+CONSTN*T*T-HNC1
TSTA=FUNC1*FUNC3
TS1=FUNC2*FUNC3

IF (TSTA) 404,404,404
402 IF (TST4) 405,405,403
403 WRITE (6,411)
411 FORMAT (6,411)
GO TO 1
404 FUNC2=FUNC3
TGS2=T1IF
GO TO 401
405 FUNC1=FUNC3
TGS1=T1IF
GO TO 401
410 ENTH1=ENT3

T=T1IF+1.0

END OF ITERATION

VG1=DOTM*1545.*T1/(PI*AREA*144.)
VG1=VG1/2A.85
WRITE (6,413) ENTH1,T1,VG1
WRITE (911H1,VG1,T1,PI

413 FORMAT (6,411)
GO TO 1

999 CONTINUE
ENDFILE 9
REWIND 9
RETURN

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), HOA(7), WT(6), COB(7), CD2B(7), H2B(7), XN2B(7),
2 O2B(7), HOB(7), C16, A16, P(212)
3 TTR*='1,A
4 IF (T.LT.1000) GO TO 10
5 DO 4 J=1,6
6 A(J)=COA(J)
7 CALL HTRT (T,A,H(1))
8 H(1)=H(1)*TT/WT(1) +28498.3 *1.8/WT(1)
9 DO 5 J=1,6
10 A(J)=CO2A(J)
11 CALL HTRT (T,A,H(2))
12 H(2)=H(2)*TT/WT(2) +96290. *1.8/WT(2)
13 DO 6 J=1,6
14 A(J)=H2A(J)
15 CALL HTRT (T,A,H(3))
16 H(3)=H(3)*TT/WT(3) +2023.8 *1.8/WT(3)
17 DO 7 J=1,6
18 A(J)=XN2A(J)
19 CALL HTRT (T,A,H(4))
20 H(4)=H(4)*TT/WT(4) +2074.7 *1.8/WT(4)
21 DO 8 J=1,6
22 A(J)=O2A(J)
23 CALL HTRT (T,A,H(5))
24 H(5)=H(5)*TT/WT(5) +2074.7 *1.8/WT(5)
25 DO 9 J=1,6
26 A(J)=H2O(A(J)
27 CALL HTRT (T,A,H(6))
28 H(6)=H(6)*TT/WT(6) +60164.7 *1.8/WT(6)
29 HV2=M(6)
30 DO 10 J=1,6
31 A(J)=CD9(J)
32 CALL HTRT (T,A,H(7))
33 H(7)=H(7)*TT/WT(7) +28498.3 *1.8/WT(7)
34 DO 11 J=1,6
35 A(J)=H2R(J)
36 CALL HTRT (T,A,H(8))
37 H(8)=H(8)*TT/WT(8) +2023.8 *1.8/WT(8)
38 DO 12 J=1,6
39 A(J)=XN2R(J)
40 CALL HTRT (T,A,H(9))
41 H(9)=H(9)*TT/WT(9) +2074.7 *1.8/WT(9)
42 DO 13 J=1,6
43 A(J)=O2R(J)
44 CALL HTRT (T,A,H(10))
45 H(10)=H(10)*TT/WT(10) +2074.7 *1.8/WT(10)
46 DO 14 J=1,6
47 A(J)=H2R(J)
48 CALL HTRT (T,A,H(11))
49 H(11)=H(11)*TT/WT(11) +2074.7 *1.8/WT(11)
50 DO 15 J=1,6
51 A(J)=XN2R(J)
52 CALL HTRT (T,A,H(12))
53 H(12)=H(12)*TT/WT(12) +2074.7 *1.8/WT(12)
54 DO 16 J=1,6
55 A(J)=O2R(J)
56 CALL HTRT (T,A,H(13))
57 H(13)=H(13)*TT/WT(13) +2074.7 *1.8/WT(13)
58 DO 17 J=1,6
59 A(J)=H2R(J)
60 CALL HTRT (T,A,H(14))
61 H(14)=H(14)*TT/WT(14) +2074.7 *1.8/WT(14)
62 DO 18 J=1,6
63 A(J)=XN2R(J)
64 CALL HTRT (T,A,H(15))
65 H(15)=H(15)*TT/WT(15) +2074.7 *1.8/WT(15)
66 DO 19 J=1,6
67 A(J)=O2R(J)
68 CALL HTRT (T,A,H(16))
69 H(16)=H(16)*TT/WT(16) +2074.7 *1.8/WT(16)
DO 16 J=1,6
16 A(J)=H20B(J)
   CALL HTR1 (T,A,M(6))
   HV1=H(6)
   HV2=H(6)
   CONTINUE
   MNC2=0.0
   DO 18 I=1,5
   18 MNC2=MNC2+C(I)*M(I)
   HTRU=MNC2/110.0-C(61)
   RETURN
   END

SUBROUTINE SUMIT(C,H,CN,ENTH)
   INTEGER M, (A=1,N=2)
   DIMENSION C(1), M(I)
   SUM=0.
   DO 1 I=1,5
   1 SUM=SUM+C(I)*M(I)
   ENTh=SUM/CN
   RETURN
   END
MAIN PROGRAM FOR EXHAUST GAS SPRAY COOLER WITH ZERO BLOCKAGE

IMPLICIT REAL*8(A-H,O-Z)
COMMON 4A

COMMON ROL, VW, GW, GC, RV, CJ, WNGAO, B(4), ISTA
COMMON ISET, KKK, KOUNT
COMMON/-LO/ MICH
DIMENSION FL(10), VL(10), TS(10), D(10), Y(45), YP(45), STA(10)
103 FORMAT (14L, BEGIN STATION 1, 14, 12X, 12HRUIN NUMBER, 249)
M=0

200 M=M+1
200 BA=.25
MIC=0
READ(5,51015)ALP1, ALP2
51015 FORMAT(249)
READ(5,102) NSTA

KEN=0
NS1= NSTA+1
READ(5,101) STA(1), I=1, NSTA
READ(5,100) CV, VG, IG, P
READ(5,101) FL(1), I=1, NSTA
READ(5,101) VL(1), I=1, NSTA
READ(5,101) TS(1), I=1, NSTA
READ(5,101) D(1), I=1, NSTA
READ(5,101) B(1), I=1, 4
100 FORMAT(4E10.2)
101 FORMAT (7E10.2)
102 FORMAT (12)

CPL= 1.0
CJ= 778.0
GC= 37.2
ISET = 9
GW=24.0
VW=18.0

RV= 1.946* 77A,0
ROL=62.4
TREF= 540.0
ISTA= 1
NEQ= R

C
THF 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)

WNGAO = Y(2)*((1.0-Y(1))*Y(4))/(Y(3)*RV*((1-Y(1))/GW+Y(1)/VW))
X=0.0
WRITE (6,133) ISTA, ALP1, ALP2
3 CONTINUE
NX=.0001
KOUNT = 0

DO 1 I= 1, 499999
CALL DIFFE(X,Y,YP,DX,KKK,NEQ,1,KEN)
IF (X,GE,STA(ISTA+1)) GO TO 2
CONTINUE
1 IJJ= JJJ+1

71
C

2 CONTINUE

WE WILL NOW PLOT THE STATION JUST FINISHED

ISTA=ISTA+1

IF (ISTA .GT. NSTA) GO TO 5

WRITE(6,103)ISTA,ALP1,ALP2

C SET UP ARRAYS FOR DIFFE

ISTA=ISTA+1

IF (ISTA .GT. NSTA) GO TO 5

WRITE(6,103)ISTA,ALP1,ALP2

CONTINUE
SUBROUTINE VFUNC(X,Y,YP,K,KEN)

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

COMMON RA

COMMON RL*,VY*,GW,G*,RV,C,J,WNCAO,R(4),ISTA

COMMON KSET,KK,KOUNT

DIMENSION Y(50),VP(50),A5(10),A6(10),A7(10),A10(10),XV(10)

IECN=1
XHV(10),XHFG(10),XARH(10),DM(10),XK(10),CD(10)

PIE = 3.1416
SUM1 = 0.
SUM2 = 0.
SUM3 = 0.
SUM5 = 0.

CALL ROGEY(ROG,Y(1),Y(2),X)

DO 10 I = 1, ISTA

J = A + # J - 1

A5(I) = PIE*ROG*Y(J)**2 / 2.0

CALL SUM(Y(I),Y(3),Y(4),Y(J-1),XV(I),XK(I),Y(J),DM(I),CN(I),A

Y(I)*Y(2),RE(I),SCF(I),ABR(I),PVF(I),FNH(I))

YP(I) = XV(I)*PIE*Y(J)**2 *(XV-XV(1))*YV/A5*Y(J-2)*(1-XV(1))

A6(I) = YP(I)

YP(I-2) = (PIE/8.0)*ROG*(Y(J)**2 )/CN(I)*VAR(2)*(Y(2)-Y(J-2))*(Y(2)-

1)*YJ-1)*YJ-2)*YJ-3)*A5(I)*H(I)/(DM(I)*YJ-2)-

CXM(I)*A5(I)*H(I)/(DM(I)*A5(I)*H(I)*XHVI(I)/DM(I))

A7(I) = YP(I-1)

SUM1 = SUM1 + A5(I)*A5(I)*YJ-3)/DM(I)

CALL HLIV(YJ-1),XHML(I),XHFG(I),XHVI(I))

YP(I-1) = BAH(I)*PIE*YJ**2 *(Y(3)-YJ-1)/(DM(I)*YJ-2)-

CXML(I)*A5(I)*H(I)/(DM(I)*A5(I)*H(I)*XHVI(I)/DM(I))

A7(I) = YP(I-1)

SUM2 = SUM2 + YJ-2)*A6(I) + YJ-3)*A7(I)

SUM3 = SUM3 + A(I)*(XHML(I)*YJ-2)**2 /12.0*FRG*CJ)

SUM4 = SUM4 + YJ-3)*YJ-2)*A7(I)/(GC*CJ)

SUM5 = SUM5 + YJ-3)*A10(I)

CONTINUE

YP(I) = -SUM1*(1-Y(I)**2)

A8 = YP(I)

CALL AA33(X,Y(I),Y(2),Y(3),AA34,AA6,ROG)

CALL DERT(YR,NR,Y(I),Y(2),Y(3),Y(4),A9,SLM2,SLM3,SLM4,SLM5,AA,AA3,AL

C,AA12,AA13,AA14,AA15,AA24,AA4,AA,AA2,AX)

YP(3) = (A11*AA14-AIL*A13)/(AIL*(CPV*Y(I))**Y(I-1)) + C*AIL*A12*ROG*ROG

C

A15 = YP(3)

YP(2) = AA13-AIL*ROG*AIL/A11

A6 = YP(1)

CALL DERP(ROG,Y(I),Y(1),Y(3),R,AA2,AA4,AA9,AA13,AA16,DP,AL

YP(4) = DP

IF(X.EQ.0.0) GO TO 3

IF(X.LT.0.0) GO TO 9999

RA = BA = 25.

9999 WRITE(6,4)

WRITE(6,16794)

R7A44 FORMAT(100)

WRITE(6,16794)

STRT=1

9999 WRITE(6,4)

WRITE(6,16794)

R7A44 FORMAT(100)

WRITE(6,16794)

R7A44 FORMAT(100)

WRITE(6,16794)

WRITE(6,16794)

WRITE(6,16794)

WRITE(6,16794)

WRITE(6,16794)

WRITE(6,16794)

WRITE(6,16794)
GO TO 102
221 CPVF = 331/2 *0.1438E-3 *TF1**2 - 0.1312E-7 *TF1**2 + 0.3521E-6 *TF1 - 0.3776E-8 *TF1**2
GO TO 102

222 WRITE (6,101)
101 FORMAT (1H,**TFP** + TS OUT OF RANGE**)
WRITE (16,122) TG,TS
122 FORMAT (1H,*TG**, E20.6, 5X,*TS**, E20.6)
STOP

102 CONTINUE
FKV = 0.432*FMV
FKNC = 0.257 * (0.115 + 5.174*CPNCF)* FMNC
FK = XVFR*FKV + (1-XVFR)* FKNC
CPF = XVFR*(VM/XM)**CPVF + (1-XVFR)**CPNCF*GW/XM
PVF = CPF*FMV
FMNC = 2 + AO + ASORT(RE) * PVF**0.3333
RH = FMN*CPF/(D*PVF)
RETURN
END

SUBROUTINE RGFGE (ROG, CV, VG, X)
IMPLICIT REAL*(A-H,N-Z)
COMMON RA
COMMON ROL, VW, GW, GC, RV, CJ, WNC, AO, B(4), ISTA
AO = B(1) + B(2)*X + B(3)*X**2 + B(4)*X**3
ROG = WNC/(1-CV)*VG*AO
RETURN
END

SUBROUTINE XL1(TS, XM, XHG, XHV)
IMPLICIT REAL*(A-H,N-Z)
XM = TS - 540.0
XHG = - 566*TS + 0.239E-2 *TS**2 + 0.0927E-7 *TS**3 + 1352.3
XHV = XM + XHG
RETURN
END

SUBROUTINE AAA(A, CV, VG, AO, AAA, ROG)
IMPLICIT REAL*(A-H,N-Z)
COMMON RA
COMMON ROL, VW, GW, GC, RV, CJ, WNC, AO, B(4), ISTA
AO = B(1) + B(2)*X + B(3)*X**2 + B(4)*X**3
AO = AO + 2.00 + AO + 3.00 + AO + EE2.0
AO = VG*AO/ (VG*AO + 1)**2
RETURN
END

SUBROUTINE DPRTI (ROG, CV, VG, TG, P, A9, SUM2, SUM3, SUM4, SUM5, AO, A3, A11, A12, A13, A14, RCVP, CPA, AO, AO, A)
IMPLICIT REAL*(A-H,N-Z)
COMMON RA
COMMON ROL, VW, GW, GC, RV, CJ, WNC, AO, B(4), ISTA
COMMON T5, XEK
IF (VG**.5, 1.0, 0.0) GO TO 20
R = P/(ROG + TG)

75
\[
A_0 = \frac{G \cdot (1-CV)}{R}
\]

\[
A_4 = \frac{WNCA0}{VG \cdot (1-CV)} \cdot AA0
\]

\[
A_1 = \frac{(1-CV) \cdot VW - TG \cdot R \cdot A0}{(1-CV) \cdot VG}
\]

\[
A_2 = RV \cdot VW \cdot GW \cdot (GW - VW) / (A1 \cdot CV) \cdot (XV \cdot VW + (1 - XV) \cdot GW) / (2)
\]

\[
A_3 = AO \cdot WNCAO \cdot TG \cdot A3 - 2 \cdot A9 - AO \cdot WNCAO \cdot TG \cdot A3 - 2 \cdot A9 - AO
\]

\[
A_12 = (1 + CV) / (1-CV) \cdot VG / (CC \cdot CJ)
\]

\[
A_13 = (1 + CV) \cdot VW / (1-CV)
\]

\[
XV = GW \cdot CV / A1
\]

\[
A_2 = RV \cdot VW \cdot GW \cdot (GW - VW) / (A1 \cdot CV) \cdot (XV \cdot VW + (1 - XV) \cdot GW) / (2)
\]

\[
A_9 = A0 \cdot WNCA0 \cdot TG \cdot A3 - 2 \cdot A9 - AO \cdot WNCA0 \cdot TG \cdot A3 - 2 \cdot A9 - AO
\]

\[
CV = \frac{SIM2}{CV}
\]

\[
CALL HFFV (TG, XVH, CPV, CPA)
\]

\[
A14 = -A9 \cdot (XVH + VG) \cdot 2.0 / (12 \cdot GC \cdot CJ) / (1-CV) \cdot 2 - SIM3 - SIM4 - SIM5
\]

RETURN

31 FORMAT (F20.10)

STOP

END

SUBROUTINE HFFV(TG, XVH, CPV, CPA)

IMPLICIT REAL*(A-H, I-Z)

COMMON RA

COMMON ROLVW, GV, WCJ, WNCAO, R(4), ISTA

COMMON ISF, KKK

IF (TG .LE. 1000.0) THEN

IF (TG .LT. 400.0) THEN

GO TO 222

ENDIF

ENDIF

GO TO 100

222 FORMAT (15, "TEMP: \# IS OUT OF RANGE")

WRITE (6, 1) TG

FORMAT (15, "#TG=",F16.6)

STOP

CONTINUE

RETURN

END

SUBROUTINE DERP(RNG, CV, VG, FR, R, A2, A3, A4, A9, A15, A16, DP, A)

IMPLICIT REAL*(A-H, I-Z)

COMMON RA

COMMON ROLVW, GV, WCJ, WNCAO, R(4), ISTA

DP = ROG*R*A15 + (ROG*TG*A2 + TG*R*A) * A9 - R*G*A4 * A16 * WNCA0 * A3 * TG * R

RETURN

END