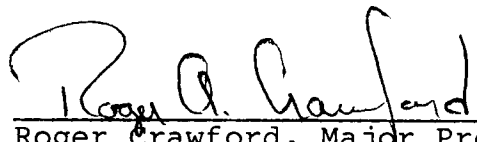
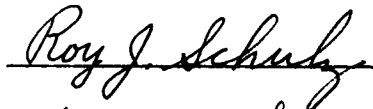
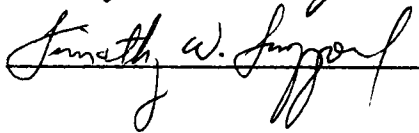


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Roger Crawford, Major Professor

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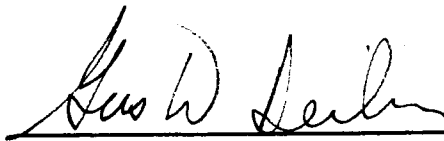

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HEAT TRANSFER IN THE AEROPROPULSION
SYSTEM TEST FACILITY MIXING HEADER

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

Gus William Deibner

March 1988

ACKNOWLEDGEMENTS

I hereby thank Doctor Roger Crawford, who served as my major professor for his assistance in the preparation of this thesis. I also thank Doctor Roy Schulz and Doctor Tim Swafford for their excellent instruction and for serving as my committee members. Lynn Moran, David Milleville, and Jim Cunningham were instrumental in helping my understanding of the problem and I give them special thanks.

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The object of this study was to determine the effectiveness of a duct liner to reduce heat transfer effects on air flow and to allow quicker temperature changes in test conditions at the Aeropropulsion System Test Facility (ASTF). The heat transfer problem was analyzed in two ways: 1) in terms of an equivalent resistance model and 2) with a transient heat conduction computer program. The resistance to heat transfer was shown to be two times greater with a liner. However, the improvement in thermal response time of the facility flow was shown not to be significant enough to warrant implementation of the liner in the actual facility. The advantages and disadvantages of this solution were investigated.

TABLE OF CONTENTS

CHAPTER	PAGE
I. INTRODUCTION	1
Background	1
The Problem	3
Physical Description of ASTF	7
Mixing Header and Cell Inlet	9
II. THE EXISTING FLUID MECHANICS MODEL OF ASTF INLET FLOW	12
Description	12
Inconsistency in Model Formulation	15
III. APPROACH TO PROBLEM	27
Suggested Solution	27
Approach	27
Data and Plant Configuration	29
Free Convection on Outside of Duct	30
Forced Convection on Inside of Duct.	30
Equivalent Resistances	31
Program for Predicting Transient Heat Transfer	33
Equations Derived.	34
Results	38
Test Cell Bypass	46
IV. SUMMARY AND CONCLUSIONS.	47
REFERENCES	50
APPENDIX	52
VITA	58

LIST OF FIGURES

FIGURE	PAGE
1. Aerial of ASTF	2
2. Aerial with Mixing Header and Cell Inlets Outlined	4
3. ASTF Capabilities Lost to Heat Transfer . .	6
4. Schematic of ASTF Inlet Air Supply Ducting System.	10
5. Expected Temperature Response of the Mixing Header	13
6. Original Model.	16
7. ASTF Mixing Header and C2 Inlet Heat Transfer Sections	17
8. Validated Equations	19
9. Locations of Thermocouples Used in Figure 10	22
10. Temperatures at Locations on Mixing Header	23
11. Plot Showing Temperature Losses in Ducting Sections	26
12. Cross Section of Ducting Liner	28
13. Equivalent Resistances of Ducting Walls . .	32
14. New Model	35
15. Test Data (Air Temperature) Vs. Predicted by New Model (Case 1)	40
16. Test Data (Metal Temperature) Vs. Pre- dicted by New Model (Case 1).	41
17. Test Data (Air Temperature) Vs. Predicted by New Model (Case 2)	42

18.	Test Data (Metal Temperature) Vs. Pre- dicted by New Model (Case 2).	43
19.	Effect of Liner (Test Data Available) . . .	44
20.	Effect of Liner (Going Cold) and Effect of Colder Inlet Temperature	45

NOMENCLATURE

A		Surface area (ft^2)
C	SPHT	Specific heat (Btu/lbm R)
D		Ducting diameter (ft)
Δx	DELX	Delta x
Δt	DTIME	Delta t
Fo		Fourier number
h	HCOE	Heat transfer coef. ($\text{Btu/hr ft}^2 \text{ R}$)
k	COND	Thermal conductivity (Btu/hr ft R)
m		Air mass flow rate (lbm/sec)
M		Metal mass (lbm)
NOINT		Number of nodes
PR		Prandtl number ($\mu C/k$)
q		Heat flux (Btu/hr)
R		Equivalent resistances (hr R/Btu)
Red		Reynolds number ($\rho VD/\mu$)
T		Temperature (R)
TAMB		Ambient temperature (R)
TAVG		Average air temperature (R)
V		Velocity (ft/sec)

SUBSCRIPTS

a	Air
amb	Ambient condition
f	Film
i	Node index

in	Inside ducting condition
j	Time index
m	Metal
out	Outside ducting condition
s	Surface
1	Inlet condition
2	Exit condition

CHAPTER I

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The Aeropropulsion System Test Facility (ASTF) (Figure 1) was designed and built for the United States Air Force, Arnold Engineering Development Center (AEDC), to provide a significantly increased capability for turbine engine altitude testing. Its design incorporates the best features of both of AEDC's older engine test facilities, called the "B" Plant and "A" Plant. The B Plant was built to provide conditions for the colder regions of turbine engine operating envelopes. Its cold capabilities are duplicated in ASTF with expansion turbines which can handle the full facility flow. The A Plant was originally built without refrigeration exclusively for providing hot test conditions. Its design included two large heaters. A Plant's hot capabilities are duplicated and expanded in ASTF with two stages of heaters. ASTF contains the best of both older test facilities and integrates them into a single, well designed system.

These design features make ASTF an ideal facility for air-breathing propulsion engine testing over a wide range of conditions not attainable in the old Engine

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These design features make ASTF an ideal facility for air-breathing propulsion engine testing over a wide range of conditions not attainable in the old Engine



Figure 1. Aerial of ASTF

Test Facility at AEDC. Unfortunately during its first operational shakedowns, a heat transfer problem became apparent that affects the facility's capability for providing desired test conditions. The purpose of this thesis was to identify the causes of the heat transfer problem and to evaluate potential solutions by a two part system heat transfer analysis. A recommendation will be made as to the best solution based on the analysis.

The Problem

Cold conditions for turbojet engines (0 to -80 degrees Fahrenheit) provide a severe test of the engine blow-out and re-light performance. Thus it is extremely important that an altitude test facility be capable of providing these conditions for turbine engine testing. In the ASTF facility, it was found that setting the cold conditions was difficult, specifically, some cold conditions at the engine inlet took hours to reach. Transient temperature changes can be achieved but they have been found to be limited to a narrow temperature band centered on mixing header wall temperature ¹ and are much slower than originally envisioned. The mixing header (Figure 2) is where the flows from the temperature conditioning sections of ASTF are "mixed" to obtain

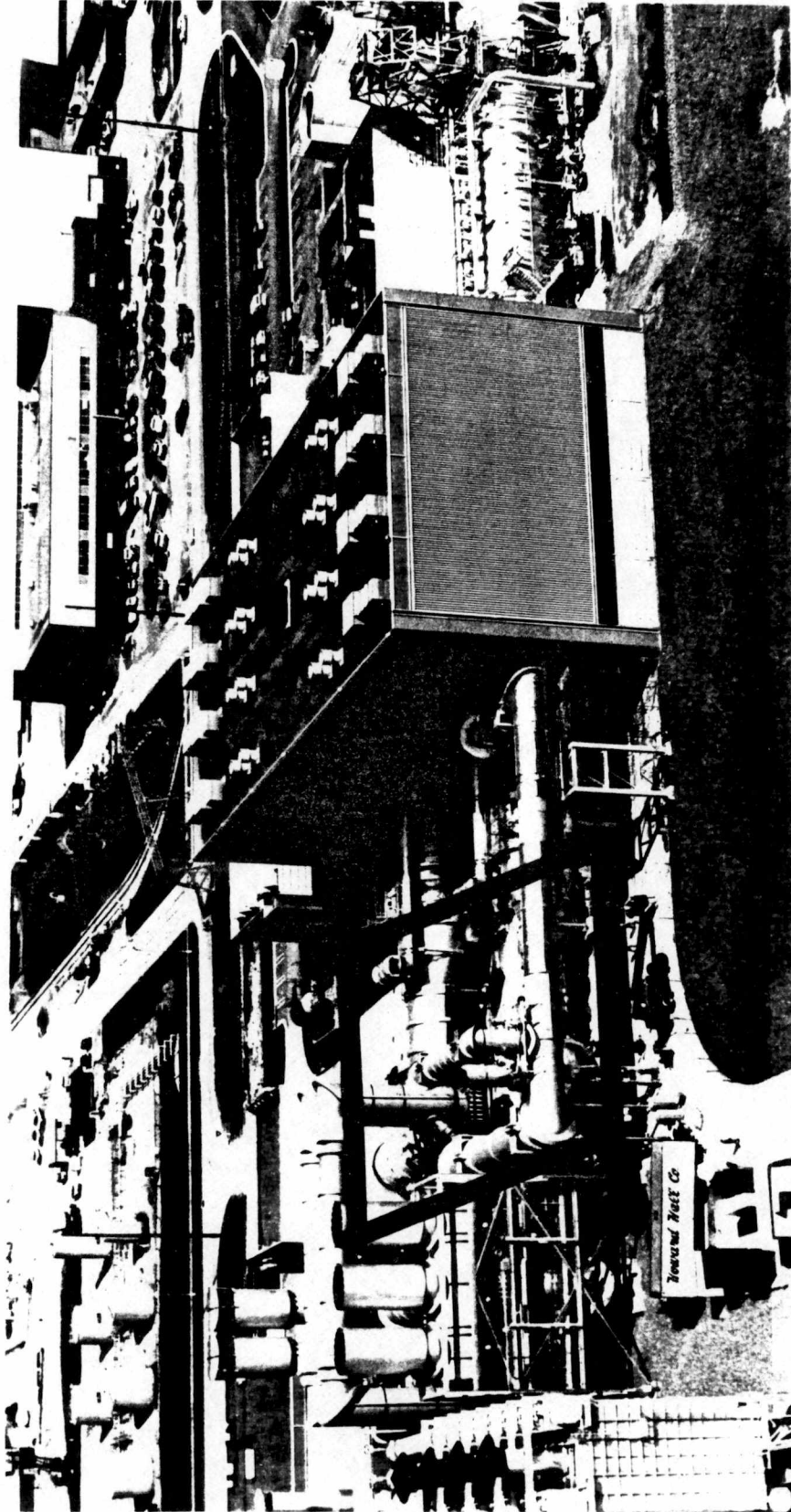


Figure 2. Aerial with Mixing Header and Cell Inlets Outlined

the proper test cell inlet temperature. Figure 3 shows the effects of facility heat transfer on the temperature performance envelope of the facility. For this figure, the ducting wall temperature is 100 degrees F. The graph shows a case where leg 1, the super cold leg, is running at -100 degrees F at 750 lb/sec and leg 3, the hot leg, is 650 degrees F at 750 lb/sec. The figure shows the outer envelope of the temperature that the facility should be supplying to the engine without heat transfer effects. The inner envelope ($\diamond$) shows what a modern gas turbine engine (F100) requires. The actual facility capability is shown by the third curve ($\circ$). Time to reach the various conditions was limited to 15 minutes which would be typical of an ideal facility. Generally as mass flow increases, the spread of the actual facility envelope increases and response times decrease. Some cell inlet temperature changes required times ranging from 30 minutes to hours. During this time, the plant is operating, consuming large amounts of electrical power along with a significant waste of manpower which stands by waiting for the test condition to be reached.

The problem of setting steady-state and transient conditions is made worse for smaller modern engines which require relatively small mass flows. Then, times to reach cold temperatures (such as 420 degree R) can

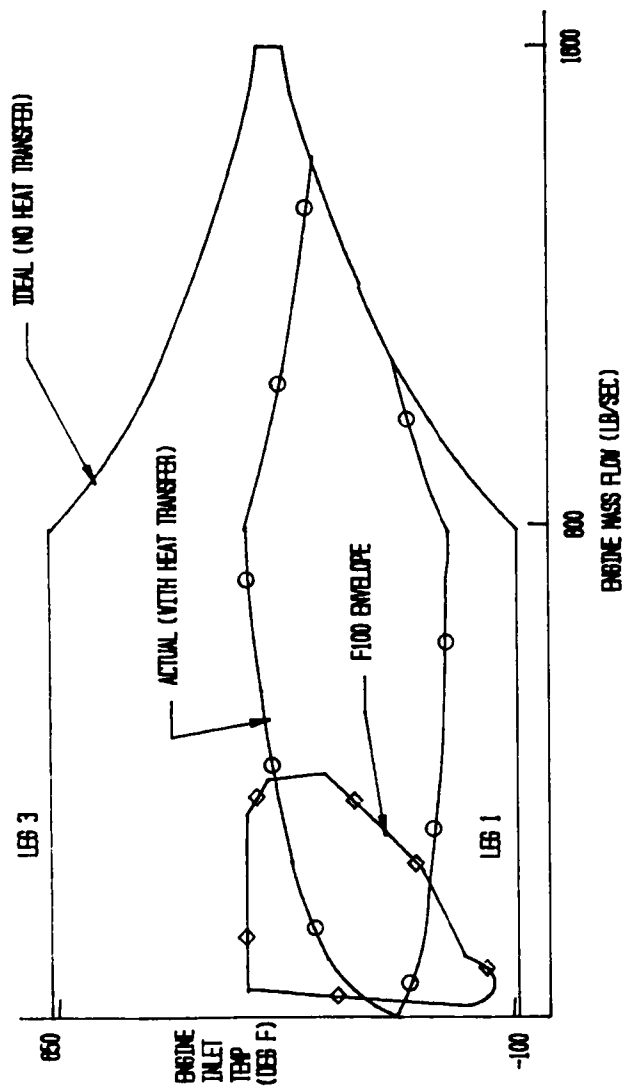


Figure 3. ASTF Capabilities Lost to Heat Transfer

take hours rather than minutes. ASTF was designed for very large engines up to 100,000 pounds thrust. Where, because of its large mass flows, a TF39 (C5 Transport Engine) could not be tested through its entire flight envelope in the old Engine Test Facility, it can be handled with ease in ASTF. The facility and its thermal system was designed to handle large mass flow engines. Initial testing showed heat transfer would not be a problem during high facility flow rates but further testing during facility activation with smaller F100 type engine air flow rates showed a significant heat transfer problem. Current Air Force needs require testing of advanced fighter engines and vectored thrust engines, which currently ASTF will test in a direct connect mode. For these engines, mass flow requirements will be in the same weight flow class as the F100, and will require cold temperatures at the test cell which probably cannot be reached with the current facility heat transfer problems and the original configuration of the plant. Obviously, this problem seriously hampers AEDC's ability to use ASTF to its full capability.

Physical Description of ASTF

ASTF is designed to provide altitude test conditions for the testing of an extremely wide range of turbine engines. This requires plant equipment to provide

conditioned air at a given pressure and temperature for the engine inlet and to pump engine exhaust gases at a given pressure to simulate the altitude conditions required. Since the heat transfer problem occurs in the mixing header and inlet ducting (as will be discussed later), a good description of this ducting geometry is required for the problem definition. Six air compressors in the air supply building can be arranged in various configurations to provide air at 40 or 150 psi. The air can then flow through a variety of coolers into the four legs of ducting located in a yard area. The legs are named for the temperature they are designed to produce, leg 1 being the super cold, leg 2 the cold leg, leg 3 the hot and leg 4 the super hot leg. The super cold leg provides air from the expansion turbines at -109 degrees F. The cold leg provides air from the RCl cooler at a minimum of -24 degrees F. The hot leg provides air reheated from the cooler (using heat of compression from the airside compressors) or air from the first stage heaters. The super hot leg provides air from the high stage heater at 1200 degrees F. The air that flows from the first three of these legs is mixed in the mixing header to provide the inlet temperatures required. The mixing process is controlled by the automatic test control system computers which set the temperature and pressure at the cell inlets. Since air

is provided to a mixing header at known temperatures, temperature transients in the flow leaving the mixing header should be easily achieved, resulting from a simple task of adjusting mixing valves just like a water faucet can be adjusted to obtain a water flow of desired temperature with a hot and cold flow.

Mixing Header and Cell Inlet

The first three legs of the plant feed into the mixing header while the super hot leg feeds directly to the inlet of C1 (Figure 4). Leg one feeds the header through a 13 foot diameter isolation valve (AV.TC.2). Pressure and temperature for leg one is controlled at the inlet guide vanes of the expansion turbines (TC1-5). Leg two and leg three feed through three valves each; CVC1a,b,c for leg two, and CVC3a,b,c for leg three. These valves dump into the mixing header and are used to set the pressure and temperature (along with the turbine inlet guide vanes mentioned above) needed to simulate flight conditions. Facility flow not used by the engine or cell auxiliary air system is vented to the atmosphere through the stacks before entering the mixing header. Thus, the actual airflow to the test cells through the mixing header is low compared to the total airflow generated by the facility compounding the heat transfer problem. The mixing header is 15 ft in diameter and

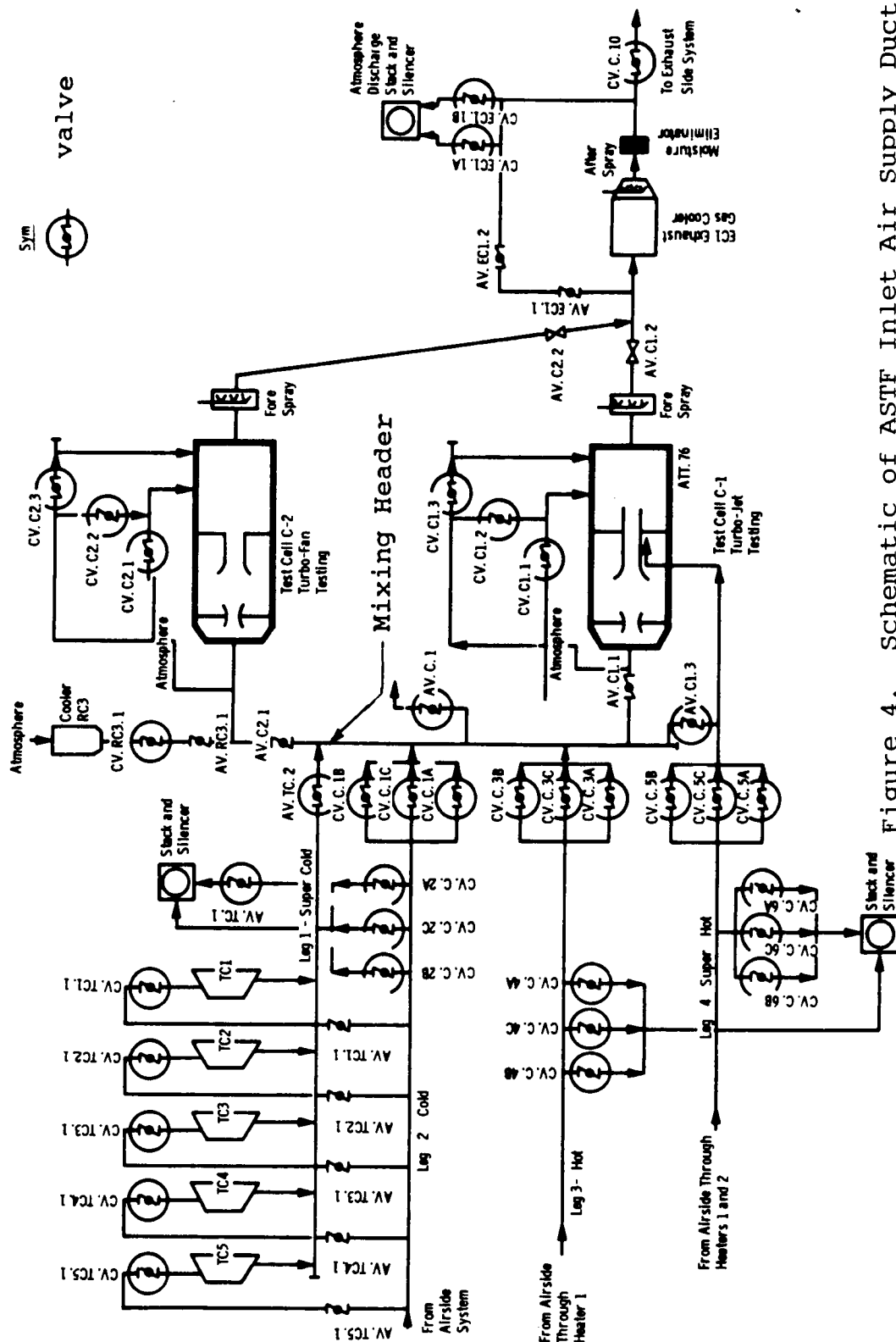


Figure 4. Schematic of ASTF Inlet Air Supply Ducting System

is made of 7/8 inch thick carbon steel with a 1/8 inch stainless steel liner bonded to its inside walls. The air then flows to the test cells from the mixing header through two flow straighteners and a set of air flow metering venturis. The test cell inlet ducting is of similar construction to the mixing header, and varies from 20 to 30 feet in diameter.

CHAPTER II

THE EXISTING FLUID MECHANICS MODEL OF ASTF INLET FLOW

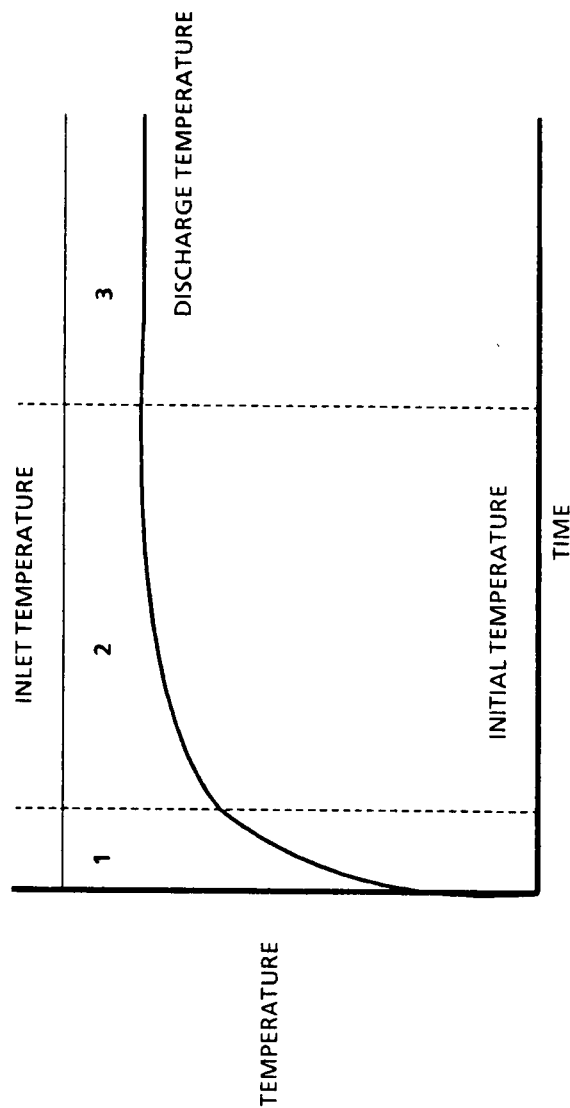
Description

An existing analytical model of ASTF ² developed by Moran was assembled from several component models that had been developed to describe the performance of various sections of the ASTF facility. Component models were first developed in 1977, and have been steadily improved since then.

The facility performance model treats the entire facility and includes several engine models. This allows complete simulation of ASTF testing for different engine classes. For the purpose of the mixing header thermal analysis, only the heat transfer component written for the mixing header was to be considered.

The original intent was to modify this model to allow analyses of changes to the ducting (such as liners). After detailed examination it was found that this could not be done but the model could be used to narrow the area of interest in the facility. A detailed explanation of why this model was not appropriate follows.

Since there are large masses of metal in the system any transient heat transfer analysis of the facility must include the heat stored in the metal. Figure 5



THE TEMPERATURE RESPONSE OF THE SYSTEM IS MADE UP OF THREE AREAS
1 IN THE FIRST FEW SECONDS AFTER A TEMPERATURE CHANGE THE LOSSES ARE;
LOSSES = MIXING + HEAT STORAGE IN THE METAL + LOSSES TO AMBIENT
2 DURING THE NEXT PERIOD THE LOSSES SLOWLY LESSEN AS THE METAL APPROACHES
THE TEMPERATURE OF THE FLOW;
LOSSES = HEAT STORAGE IN METAL + LOSSES TO AMBIENT
3 WHEN THE SYSTEM REACHES A STEADY STATE THE LOSSES ARE ONLY MADE UP OF
THE LOSSES TO THE AMBIENT SURROUNDINGS

Figure 5. Expected Temperature Response of the Mixing Header

shows the transient temperature responses of the facility that might be expected. The first time regime is a zone where the flow of the air at the new temperature is mixing with air remaining in the system at the old temperature. Heat is also being lost to the metal and surrounding environment. The second's temperature response is driven by the ducting metal temperature. As the wall approaches the temperature of the flow the slope of the line becomes smaller and smaller. In the third time regime, the wall temperature has reached a steady state and the temperature loss comes from the thermal resistance of the system to losses to the ambient conditions outside the ducting.

The original model has gone through two basic stages, hereafter referred to as the baseline and the "validated" (more appropriately, the calibrated model) forms. The baseline thermal model used equations for heat transfer coefficient on the inside of the ducting valid for circular ducts with 1D flow in them. A lumped-system, Newtonian heating (or cooling) analysis was used to predict the "heat storage" in the metal of the duct. This approach assumes that duct wall metal temperature would be the same throughout the wall. The "validated" form of the thermal model took the initial equations developed for the baseline model and adjusted them so that predicted thermal events matched test data

gathered from four ASTF tests. The adjustment required was rather extreme in some cases, indicating that the simplifying assumptions left out some important factors (Figure 6).

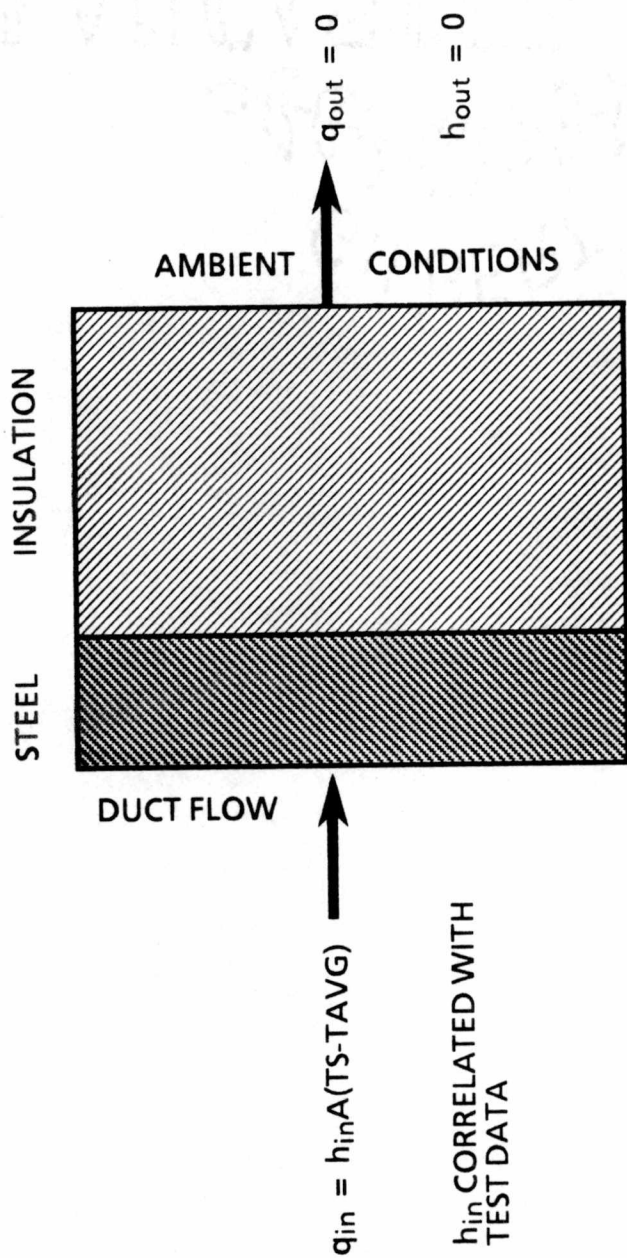
The "validated" equation's adjustment matched the model's output to within 15% of test data. Unfortunately the "validated" model does not allow changes in ducting geometry (liners, increased insulation, etc.) since the equations were corrected for the existing thermal system. The "validated" model can, however be used to show that the significant part of the heat transfer problem is in the mixing header region. The simplifying assumptions of the "validated" model were examined in detail and checked for validity so that a new model could be written which was physically more accurate allowing for the effects of geometry changes on the facility heat transfer.

To apply the model for heat transfer calculations the ducting was split into different sections from the control valves to the engine inlet. Figure 7 shows the final locations of the sections used in the model.

Inconsistency in Model Formulation

Ducting and header metal masses and heat transfer coefficient were the basic parameters adjusted in the baseline model to make predicted heat transfer match

ORIGINAL MODEL



q LOST FROM AIR = q GAINED BY METAL

Figure 6. Original Model

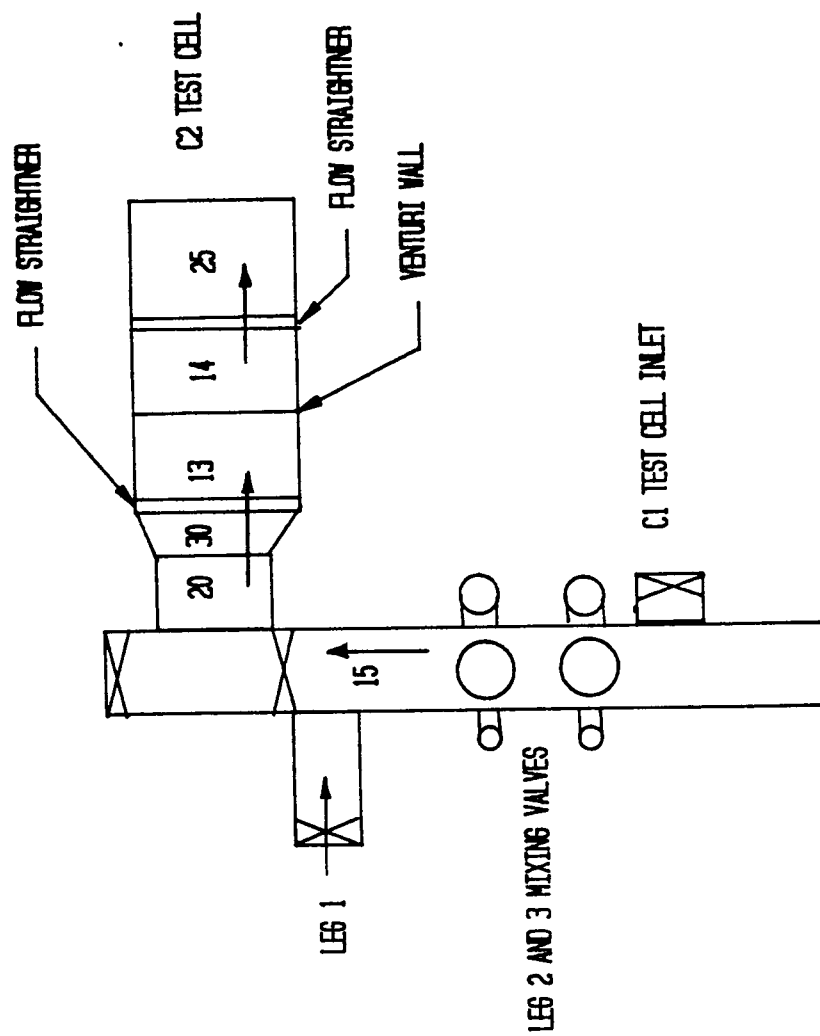


Figure 7. ASTF Mixing Header and C2 Inlet Heat Transfer Sections

actual values. Heat transfer coefficients were adjusted using a correlation method based on Reynolds Number ³. Over a moderate range of Re most convection phenomena obey a simple power relationship of the kind $y=x^n$. The baseline equation for heat transfer coefficient ⁴;

$$h = \frac{K_r (.0384) (Red)^{3/4} (PR)}{D(1 + 1.5(PR)^{-1/6} (Red)^{-1/8} (PR - 1))} \quad (1)$$

had two items adjusted using this method. The exponent of Red (i.e. 3/4) and the equation's coefficient (.0384).

Four actual values for h inside the ducting were estimated from test data and were plotted on log-log paper. Red was plotted on the x axis and the coefficient in the equation for h on the y axis. The exponent n was found from the slope of the plotted curve and the coefficient was taken at the y axis intercept. The coefficient in the equation for h varied significantly from the baseline model values in some sections (Figure 8) of the ducting (primarily the mixing header). The exponent n also exhibited differences from the baseline model value but not greater than 16 percent.

It is not surprising that the pipe flow equations for calculating h on the inside of the ducting used successfully in some parts of the ducting did not work well in the mixing header. The sections (13, 14, 25) where the equation did apply reasonably close to the

Equation for Heat Transfer Coef. on Inside of Ducting

$$h = \frac{K_r (A) (Red)^B (PR)}{D(1 + 1.5(PR)^{-1/6} (Red)^{-1/8} (PR - 1))}$$

DUCT SECTION	A	B
15	2.7367	.6333
20	.1612	.760
30	1.159	.641
13	.0384	.75
14	.0384	.75
25	.0384	.75
16	.0384	.75

FLOW STRAIGHTENER AFTER SECTION 30

Figure 8. Validated Equations

baseline formulation are smooth straight ducting, with flow straighteners. In these sections, the flow is assumed to be fully developed turbulent pipe flow which allows the use of the pipe flow equations. The mixing header, however, is supplied by several pipes emptying flow perpendicularly into the header. This results in a sudden-expansion type flow at the wall where the pipe penetrates the header and impingement flow on the opposite wall. Sudden-expansions can cause recirculation which results in a significantly higher heat transfer rate than that for a straight pipe. The coefficient h is also very high for impingement flow, compared to developed pipe flow. The flow must be very turbulent in the header because of the stirring action of the input flows. The h_{in} necessary to predict thermal response of the header was 13 times greater in magnitude than that of the initial values in the baseline equations (pipe flow) for the fifteen foot mixing header. Although this seems very high, the Reynolds number for a 50 lb/sec flow through a 15 ft diameter duct is 3.1×10^5 . The Re necessary to get a 13 times increase in h is 9.9×10^6 which could be a reasonable value for highly turbulent flow ⁵.

The mixing header was found to heat and cool in its entirety, even for cases where, when running one of the test cells, half of the mixing header should have no

direct flow. Figure 9 shows locations of thermocouples on the surface of the ducting with Figure 10 showing the temperatures of the mixing header at these locations. There is some difference in the temperatures, but at most about 9 degrees. It is clear that the whole header is heating as one piece because of the well-stirred flow within it.

Metal mass was used to adjust the baseline results to match test data. This resulted in some model masses being considerably greater than the actual weights of the ducting sections. The equation in which mass was adjusted to match actual data;

$$dT_m/dt = ((m)(C_a)/(M_m)(C_m))(T_1 - T_2) \quad (2)$$

was derived from two equations;

$$q = (m)(C_a)(T_{T1} - T_{T2}) \quad (3)$$

which shows heat transfer from the gas and,

$$q = (M_m)(C_m)dT_m/dt \quad (4)$$

showing heat transfer into the wall metal. These form a energy balance at the inner wall of the duct. Since ducting size is large the velocities are small allowing total temperatures to be taken as static temperatures for the analysis. This approach does not take into account heat transfer out of the duct due to convection on the outer surface. Taking this into account gives a better equation for the system;

$$dT_m/dt = \frac{(m)(C_a)(T_2 - T_1) - (h_{out})(A)(T_m - T_{amb})}{(M)(C_m)} \quad (5)$$

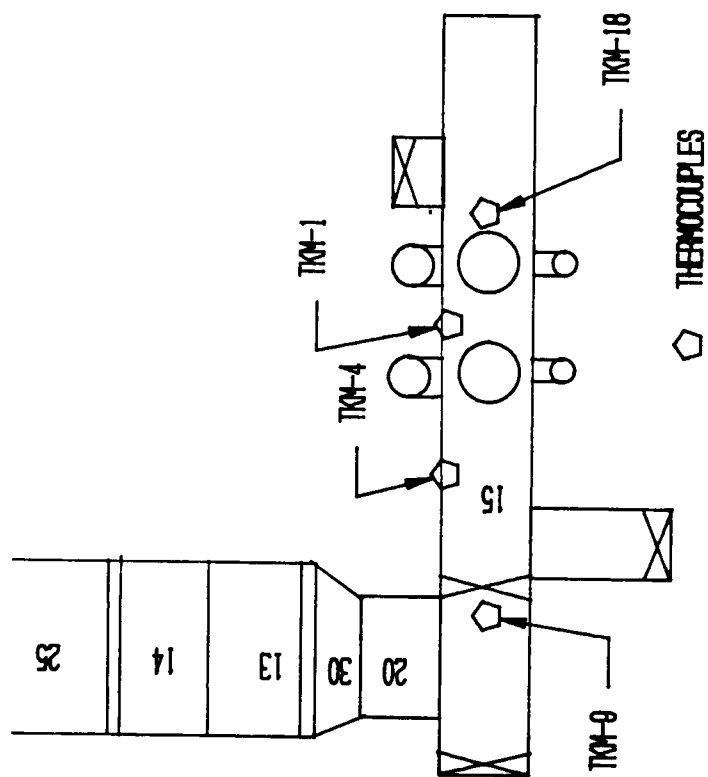


Figure 9. Locations of Thermocouples Used in Figure 10

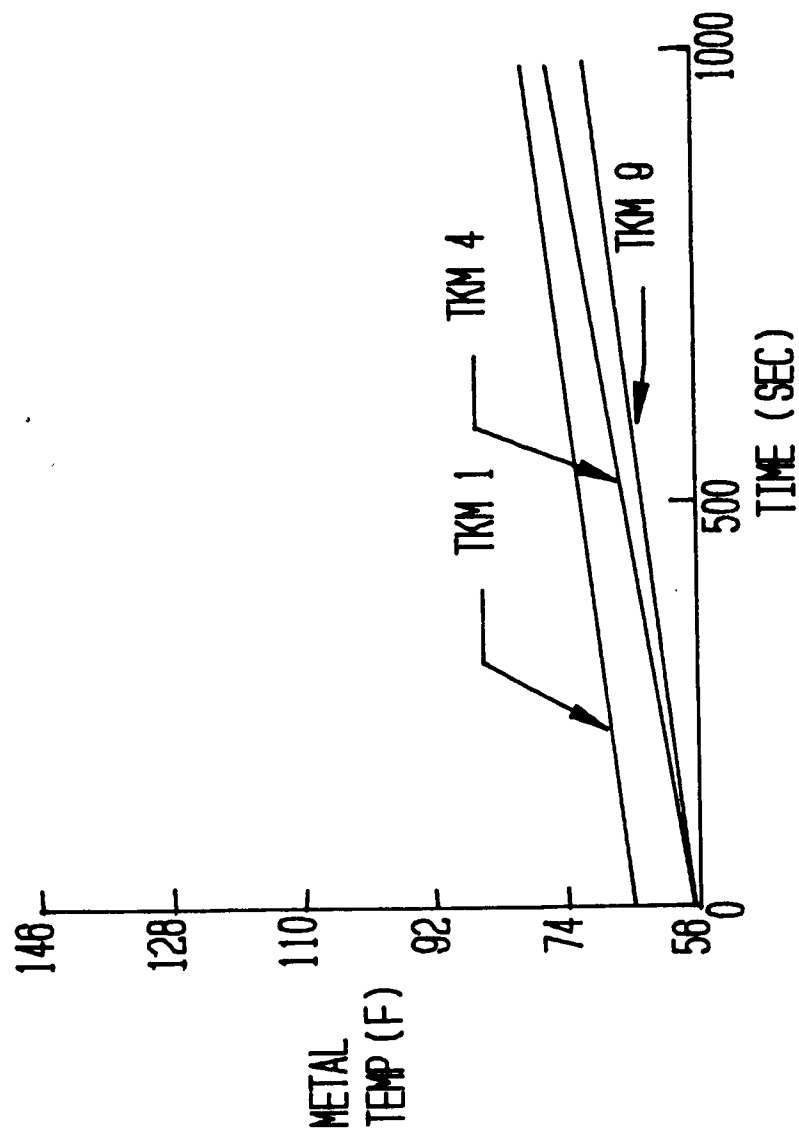


Figure 10. Temperatures at Locations on Mixing Header

Metal weights had to be increased significantly to account for the $h_{out}(A)(T_m - T_{amb})$ term. The largest discrepancy arose in the mixing header (section 15) where ducting mass was changed from the actual metal weight of 313148 lbs to a fictitious weight of 800500 lbs so predicted and actual performance would match.

In summary of the past work, the baseline equations for h_{in} worked well in ducting sections downstream of the flow straighteners in the cell inlet ducting. The baseline model performed worst in predicting transient heat loss in the mixing header where flows are expected to be the most turbulent with impingement flows. The equations for heat transfer coefficient were correlated using a recognized method and will be used later in the analyses. The change of metal mass in the "validated" model was required to account for the lack of terms for heat transfer losses (convection) outside the system. After adjustment, the "validated" model comes within 15% of the test data but does not lend itself to geometry or hardware changes such as the incorporation of interior liners since k , the wall's thermal resistance, has no part in the analysis.

The "validated" model was used to identify the section of the facility ducting that is the least well understood in flow and heat transfer, and narrow the region of interest by showing that the mixing header has

the greatest temperature loss. Figure 11 shows this by comparing temperature loss at several sections going toward the inlet of the engine. Figure 11 shows that most of the temperature loss occurs in the mixing header where the baseline equations had to be adjusted significantly. Based on these results, the rest of the new analysis treats only the heat transfer in the mixing header.

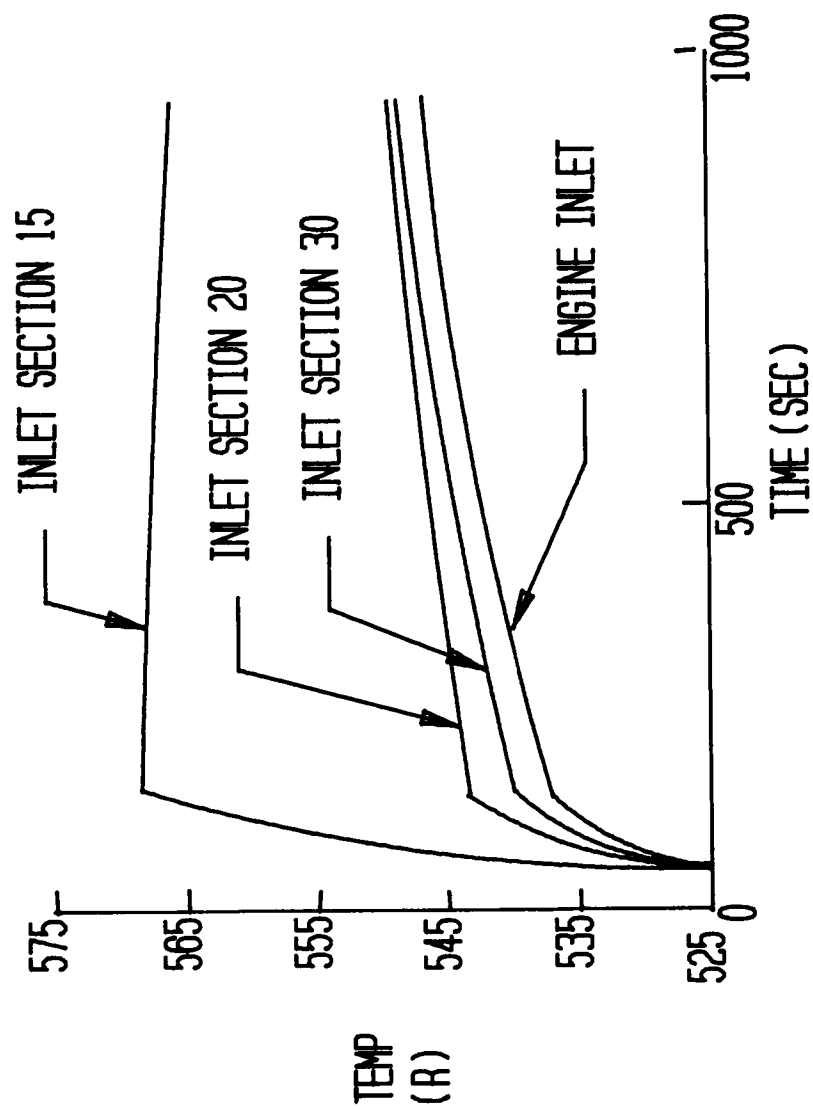


Figure 11. Plot Showing Temperature Losses in Ducting Sections

CHAPTER III

APPROACH TO PROBLEM

Suggested Solution

Several possible engineering solutions exist to the heat transfer problem in the mixing header. The concept chosen to examine in detail in this thesis involves the use of a 1/4 inch thick metal liner in the mixing header area (Figure 12) with a 4 inch air gap. The liner would reduce the mass of metal in direct contact with the process air in the ducting, which should decrease the metal temperature response time and the thermal transient time of the process air. In addition the air gap between the liner and the ducting wall is an effective insulator which should also isolate the process air from the large mass of the original ducting wall. These two advantages for the liner will be analyzed in more detail. The completed model also allowed an easy comparison of the ducting liner and a competing concept of using a test cell bypass.

Approach

This problem is approached in two steps. The first step is to analyze the thermal heat loss problem with a simple, one-dimensional equivalent resistances model.

CROSS SECTION OF DUCTING WITH LINER

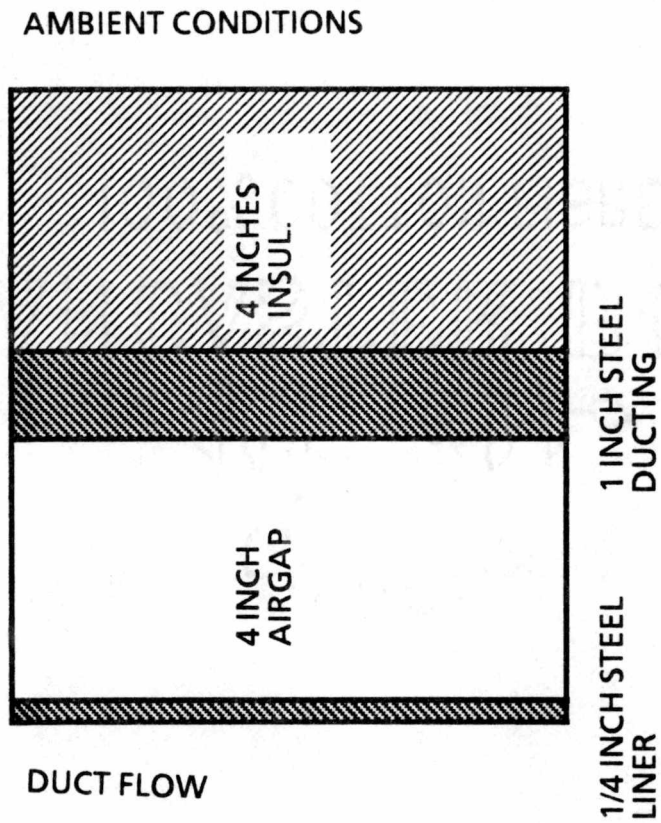


Figure 12. Cross Section of Ducting Liner

This model was used to simplify the analysis of heat loss and to test assumptions about the relative importance of the various parts of the problem. The simplifying assumptions were then included in a computer program for transient heat transfer based on a finite difference approach, to calculate the wall temperature and the heat transfer through the wall with time. From the results of calculations made with this computer program, conclusions were drawn about the effectiveness of the liner to minimize thermal transient effects and heat loss (or gain) from the facility airflow.

Data and Plant Configuration

Test data (temperature) was available from two ASTF test runs allowing the computer program to be tested. The first ASTF test run was an attempt to go from 515 degrees R to 780 degrees R as quickly as possible at the cell inlet by setting 515 degree flow in leg 2 and 780 degree flow in leg 3. At $t=0$ the hot leg was opened and the cold leg closed. Mass flow for this test through an engine simulator was 310 lb/sec. The second case was a similar transient test where cell inlet temperature were varied from 507 degrees R to 568 degrees with mass flow rate of 310 lb/sec.

Free Convection on Outside of Duct

Free convection on the outside of the duct could be a very important part of the analysis since it was not included in the baseline model of the ducting heat transfer process. A value of h_{out} was needed for the transient heat transfer computer program. This was calculated using the equation;

$$h_{out} = .537(Re)^{-.25}k_f/D \quad (6)$$

for free convection heat transfer from circular ducting ⁶. This allowed h_{out} to be calculated for a range of possible surface temperatures. It was found to be .25 to .4 Btu/hr ft² R. This value for h_{out} is about the same as a calculated value of h found using an equivalent resistance approach.

Forced Convection on Inside of Duct

Since h_{in} was experimentally determined for the existing configuration, the equation (Figure 6) for h_{in} used in the "validated" model was used for calculations involving the liner. This assumption can be made by showing that $h_{in2} = (D_2/D_1)h_{in1}$ holds true for items with similar geometries ⁷. Since the liner and ducting diameters are known the h_{in} for the liner can be calculated.

Equivalent Resistances

The use of a liner in the duct was intended to reduce the mass of the metal in contact with the air in mixing header. The liner and its air gap also provide a measure of extra insulation between the process air and ambient. The equation for the equivalent resistances takes the form;

$$q = \Delta T / \Sigma R. \quad (7)$$

The resistances (R) were calculated using $1/hA$ for convective heat transfer and x/kA for conduction heat transfer along with a modified equation ^a for the air gap that takes into account free convection in the air gap. The air gaps conduction considering free convection in the gap was five times larger than that for still air. Since the diameter of the ducting is large, and the thickness of the material small, equations for flat plates were used for convection and conduction heat transfer coefficients. The air gap becomes the primary resistance to heat flow through the ducting wall and is twice the outside convection on the outer insulation (the next highest). The original configuration had a resistance of 4.2×10^{-4} hrR/Btu where with the liner and air gap the resistance becomes 8.7×10^{-4} hrR/Btu (Figure 13). This is twice the total original thermal resistance for the ducting wall. Data from a test run was used in the thermal resistance equation to calculate

EQUIVALENT RESISTANCES

TOTAL	DUCT INTERIOR 1/hA	LINER x/kA	AIR GAP x/keqA	DUCTING S STEEL x/kA	DUCTING C STEEL x/kA	INSULATION x/kA	DUCT EXTERIOR 1/hA
A = 8437	h = 20.9	x = .25 k = 9.4	x = 5 k = .11	x = .125 k = 9.4	x = .875 k = 32	x = 4 k = .2	h = .54
8.72E-4 (LINER)	5.64E-6	2.63E-7	4.49E-4	1.31E-7	2.7E-7	1.97E-4	2.19E-4
4.21E-4 (ORIGINAL)	5.64E-6			1.31E-7	2.7E-7	1.97E-4	2.19E-4

$$A = \text{ft}^2 \quad h = \text{Btu/hr ft}^2 \text{ R} \quad k = \text{Btu/hr ft R} \quad x = \text{in}$$

Figure 13. Equivalent Resistances of Ducting Walls

what h_{out} was during a test run. Values were read from the data up to the outside of the duct where h_{out} was calculated for comparison with what had been previously calculated using the free convection equation. The convective heat transfer coefficient, h_{out} was .54 Btu/hr ft² R where a value of .4 had been predicted using the empirical equation from Wolf. The difference could easily be made up by partially forced convection on the outside of the duct from wind. This value of h_{out} (.54 Btu/hr ft² R) was used for the transient program.

Program for Predicting Transient Heat Transfer

A new computer program was written to look at the mixing header in more detail since a way to take into account important factors not included in the "validated" program was required. The ducting thermal resistance and the heat transfer through the ducting are important factors which must be included to allow analyses of the liner. The "validated" program used a lumped approach corrected to test data. Heat transfer equations considered the header as a perfectly insulated duct with no losses to the ambient environment. A finite difference method which implicitly accounts for:

1. ambient heat losses,
2. transient energy transfer,

and 3. ducting/insulation thermal resistance, was used such that more detailed analysis of the complete header system could be performed (Figure 14).

Equations Derived

A simple computer program was written to look at the transient heat transfer of the mixing header. It uses finite differences to approximate differentials and Gaussian elimination for solving the system of equations formed ⁸. The program is based on the following time-dependent temperature equation derived from the conservation of energy principle;

$$\rho C(\delta T / \delta t) = k \delta^2 T / \delta x^2. \quad (8)$$

The following finite difference approximations can be used to eliminate the differentials in equation 8;

$$(\delta T / \delta t) = (T_{i,j+1} - T_{i,j}) / \Delta t \quad (9)$$

$$(\delta^2 T / \delta x^2) = (T_{i+1,j+1} - 2T_{i,j+1} + T_{i-1,j+1}) / \Delta x^2 \quad (10)$$

$$(\delta^2 T / \delta x^2) = (T_{i+1,j} - 2T_{i,j} + T_{i-1,j}) / \Delta x^2. \quad (11)$$

The equation;

$$\delta^2 T / \delta x^2 = \delta^2 T / \delta x^2 \Big|_{j+1} + (1-\lambda) \delta^2 T / \delta x^2 \Big|_j \quad (12)$$

provides a time weighted average of $\delta^2 T / \delta x^2$ to adjust accuracy and stability. By setting $\lambda=1$ the equations take on a form that is stable for all values of Fourier number and Δt but Δt must be kept small to insure accuracy. By setting $\lambda=0$, Fourier number and Δt must be restricted to allow for stability but the equations

NEW MODEL

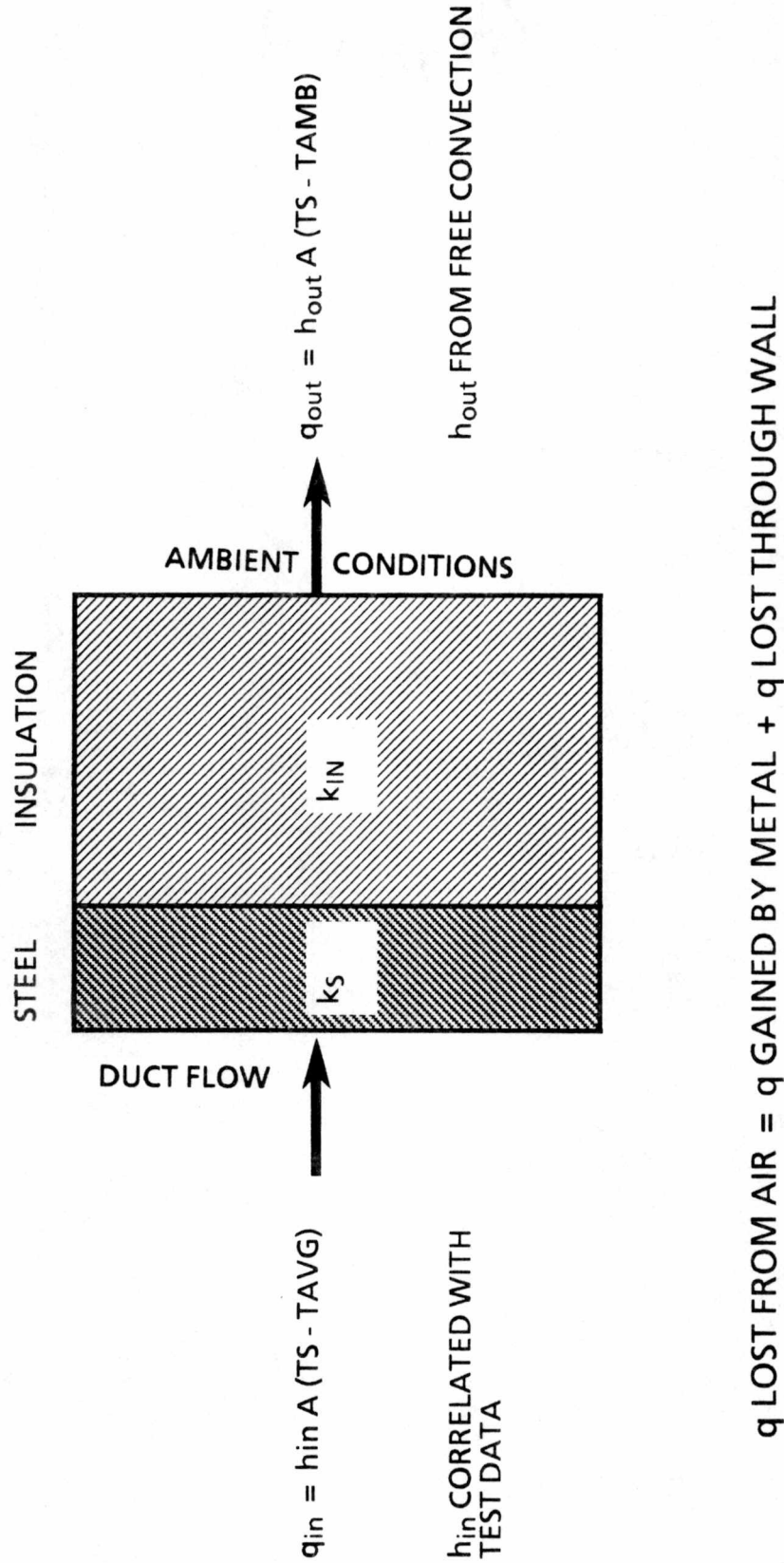


Figure 14. New Model

have a low error because of the required small Δt . By substituting 9 and 12 into 8 a finite difference equation for temperature at a node is derived;

$$\rho C(T_{i,j+1} - T_{i,j} / \Delta t) = k\lambda(T_{i+1,j+1} - 2T_{i,j+1} + T_{i-1,j+1}) / \Delta x^2 + (1-\lambda)((T_{i+1,j} - 2T_{i,j} + T_{i-1,j}) / \Delta x^2). \quad (13)$$

Fourier number can be written as $Fo = (k\Delta t) / \rho C(\Delta x)^2$ (rate of conduction of heat/ rate of storage of heat) the equation can be rewritten as;

$$-Fo\lambda T_{i-1,j+1} + (1+2Fo\lambda)T_{i,j+1} - Fo\lambda T_{i+1,j+1} = Fo(1-\lambda)T_{i-1,j} + (1-2Fo(1-\lambda))T_{i,j} + Fo(1-\lambda)T_{i+1,j}. \quad (14)$$

By setting $\lambda = .5$ the solution is always stable and has good error properties (if Δt is small) and if Fo is taken as 1, then the equation is reduced to;

$$-T_{i-1,j+1} + 4T_{i,j+1} - T_{i+1,j+1} = T_{i-1,j} + T_{i+1,j}. \quad (15)$$

This equation is used for all nodes in the body of the object in question. This approach is known as the Crank-Nicolson method.

The end points must be considered separately to allow for the convective heat transfer boundary conditions on the surfaces of the ducting. We will derive the equation for the outer surface of the duct. We know that on the surface by equating heat fluxes that;

$$k(\delta T / \delta x) = h(T_s - T_{amb}) \quad (16)$$

or using finite difference substitutions;

$$T_{i+1,j+1} - T_{i-1,j+1} / 2\Delta x = -(h/k)(T_{i,j+1} - T_{AMB}) \quad (17)$$

$$T_{i+1,j} - T_{i-1,j} / 2\Delta x = -(h/k)(T_{i,j} - T_{AMB}) \quad (18)$$

rearranging these to the form;

$$T_{i+1,j+1} = -(2\Delta x h/k)(T_{i,j+1} - T_{AMB}) + T_{i-1,j+1} \quad (19)$$

$$T_{i+1,j} = -(2\Delta x h/k)(T_{i,j} - T_{AMB}) + T_{i-1,j} \quad (20)$$

allows substitution into equation 15 above and gives the equation for the outer surface of the duct;

$$2T_{n-1,j+1} - (4 + 2(h_{out}/k)\Delta x)T_{n,j+1} = -2T_{n-1,j} \quad (21)$$

$$+ 2(h_{out}/k)\Delta x(T_{n,j} - 2T_{AMB}).$$

The equation for the other surface of the duct can be derived in a similar manor and takes the form;

$$-(4 + 2(h_{in}/k)\Delta x)T_{1,j+1} + 2T_{2,j+1} = -2T_{2,j} \quad (22)$$

$$+ 2(h_{in}/k)\Delta x(T_{1,j} - 2T_{AMB}).$$

Equations 15, 21, and 22 make up the equations necessary for the computer program to form a tridiagonal matrix. This system of equations is then solved by Gaussian elimination which gives a temperature profile across the system at a designated time. From these temperatures heat flux into or out of the system can be calculated. On the inter surface of the duct;

$$q = h_{in}A(T_M - T_{AVG}) \quad (23)$$

where T_{AVG} is;

$$T_{AVG} = (T_2 + T_1)/2. \quad (24)$$

Then for the air flowing through the duct;

$$q = mC_a(T_2 - T_1) \quad (25)$$

Equating 23 and 25 and substituting 24 into the result yields;

$$T_2 = (T_m + T_1((mc/hA) - .5)/((mc/hA) + .5) \quad (26)$$

which gives T_2 the discharge temperature of the duct at that time. This discharge temperature is averaged with the ducting inlet temperature which sets the average temperature of the air in the duct for the next iteration of the computer program. The program with sample input/output is shown in the Appendix.

Results

The computer program was used for a comparison between the liner and the current configuration of the duct. With this in mind the following simplifying assumptions were made:

1. Heat transfer coefficients h_{in} and h_{out} were held constant.
2. The mixing header was treated as a 15 ft diameter pipe with flow throughout it's length.
3. For the case with a liner the air gap has by far the highest resistance so the metal mass was taken as the mass of the liner. The original ducting was considered to be isolated from the air flow by the liner's air gap.
4. The k for the system was taken as the equivalent resistance of the system. This assumption give a correct temperature value for the surface temperatures but incorrect temperatures through the wall. Since only surface temperatures are needed for the analysis, this

is a satisfactory solution.

5. The heat capacity of the insulation was found to be only 8% of that of the steel, therefore the insulation was not considered in the analysis for heat storage.

Results generated by the program were compared to two cases where test data was available. Case 1 (Figures 15 and 16) is for a transient attempting to go to 780 degrees R. The program trends somewhat low for this case. Figure 15 shows mixing header discharge temperature where Figure 16 shows mixing header metal temperatures. Case 2 (Figures 17 and 18) shows the same except the end temperature was 568 degrees R. The computer programs output is in good agreement with this data.

To compare the lined and unlined mixing header the program was changed by changing k to the equivalent resistance with the liner and changing the metal mass to the mass of the liner. Figure 19 shows the results of the change for Case 1 above. The additional insulation makes an improvement and at $t=550$ seconds past the transient the temperature is 35 degrees R better than without the liner. The slope of the line changed only slightly and should not affect the time required to reach a desired temperature quickly. The same comments can be made about Figure 20 which is a case with a 400 degree R inlet temperature. The added insulation helps

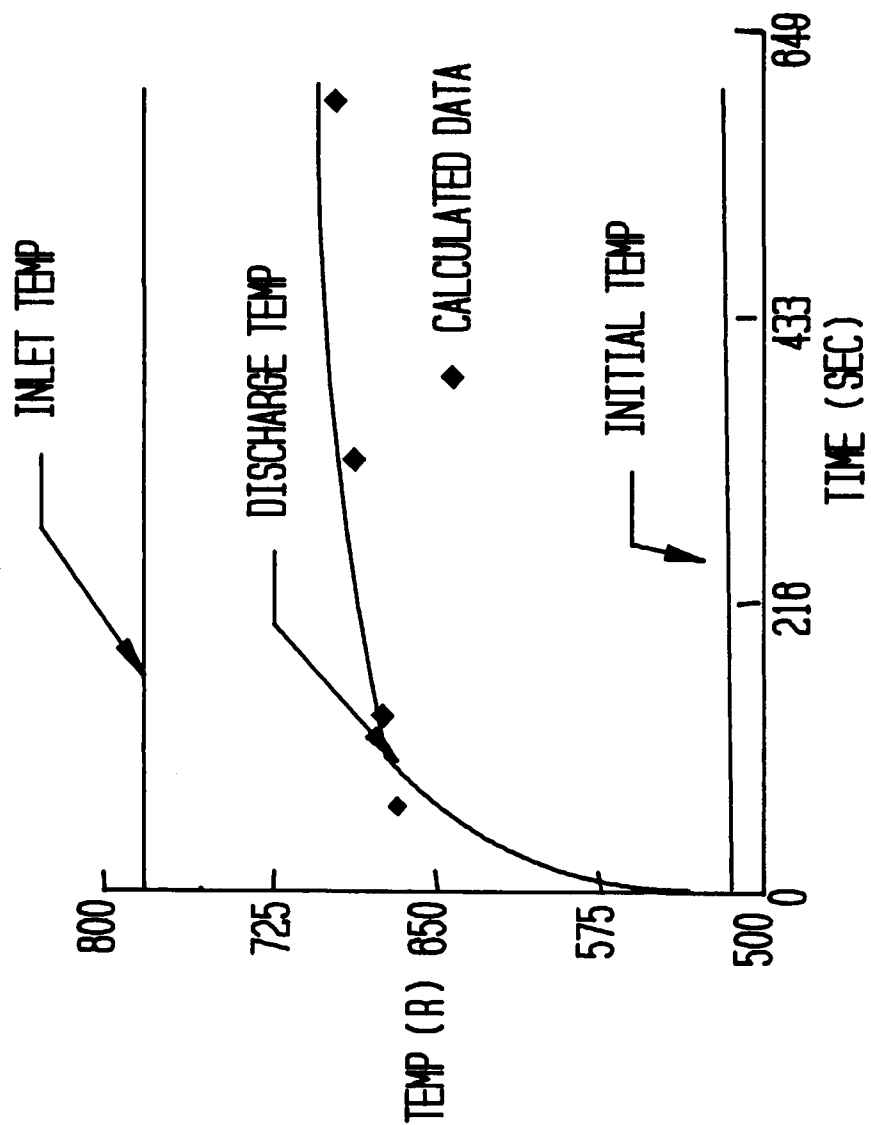


Figure 15. Test Data (Air Temperature) Vs. Predicted by New Model (Case 1)

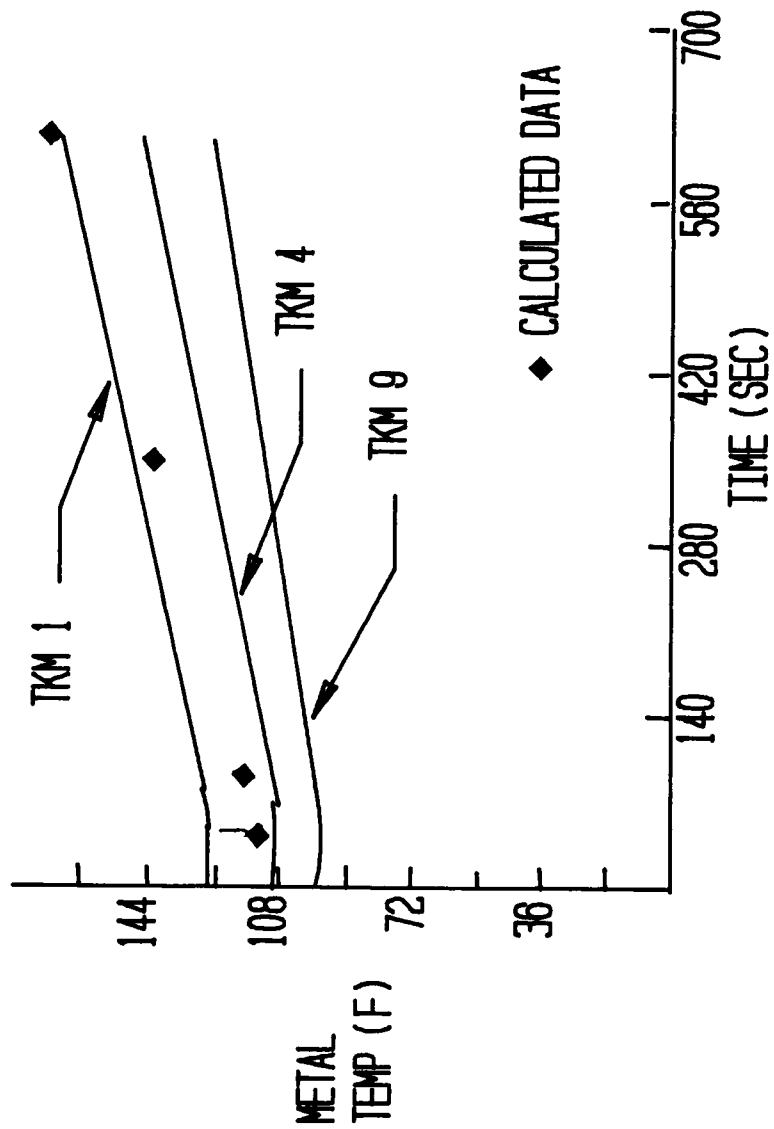


Figure 16. Test Data (Metal Temperature) Vs. Pre-dicted by New Model (Case 1)

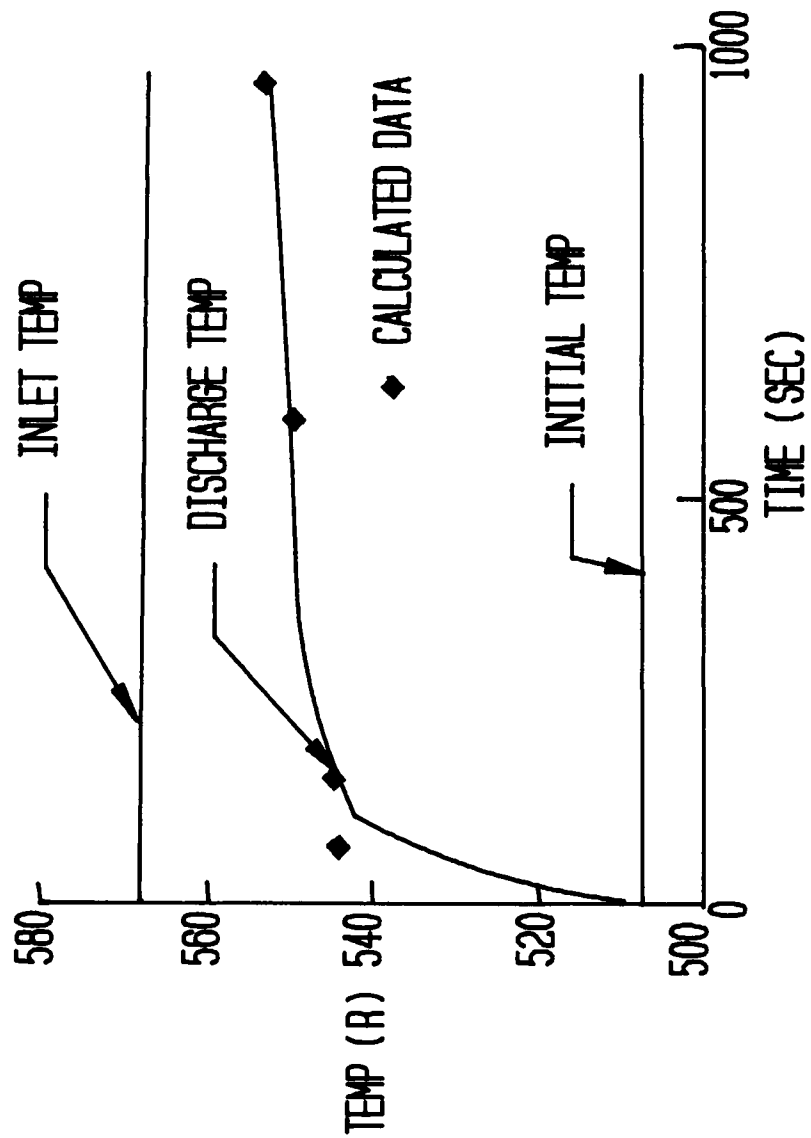


Figure 17. Test Data (Air Temperature) Vs. Predicted by New Model (Case 2)

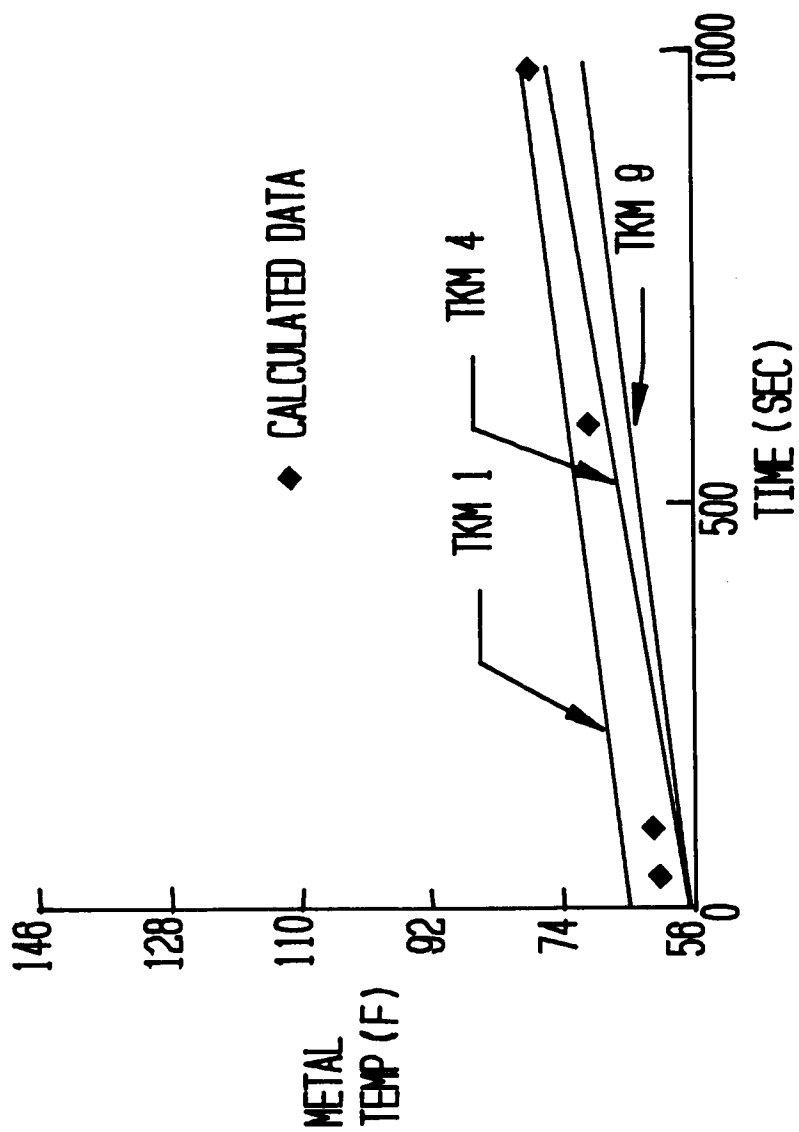


Figure 18. Test Data (Metal Temperature) Vs. Predicted by New Model (Case 2)

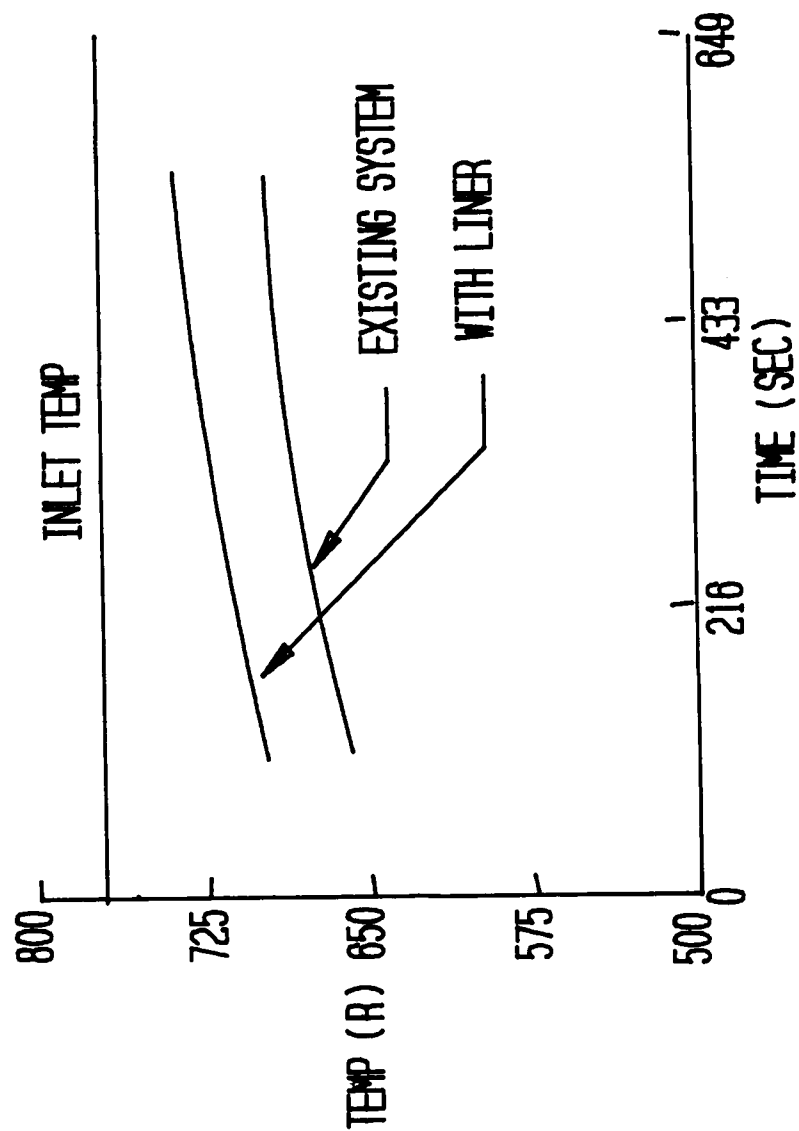


Figure 19. Effect of Liner (Test Data Available)

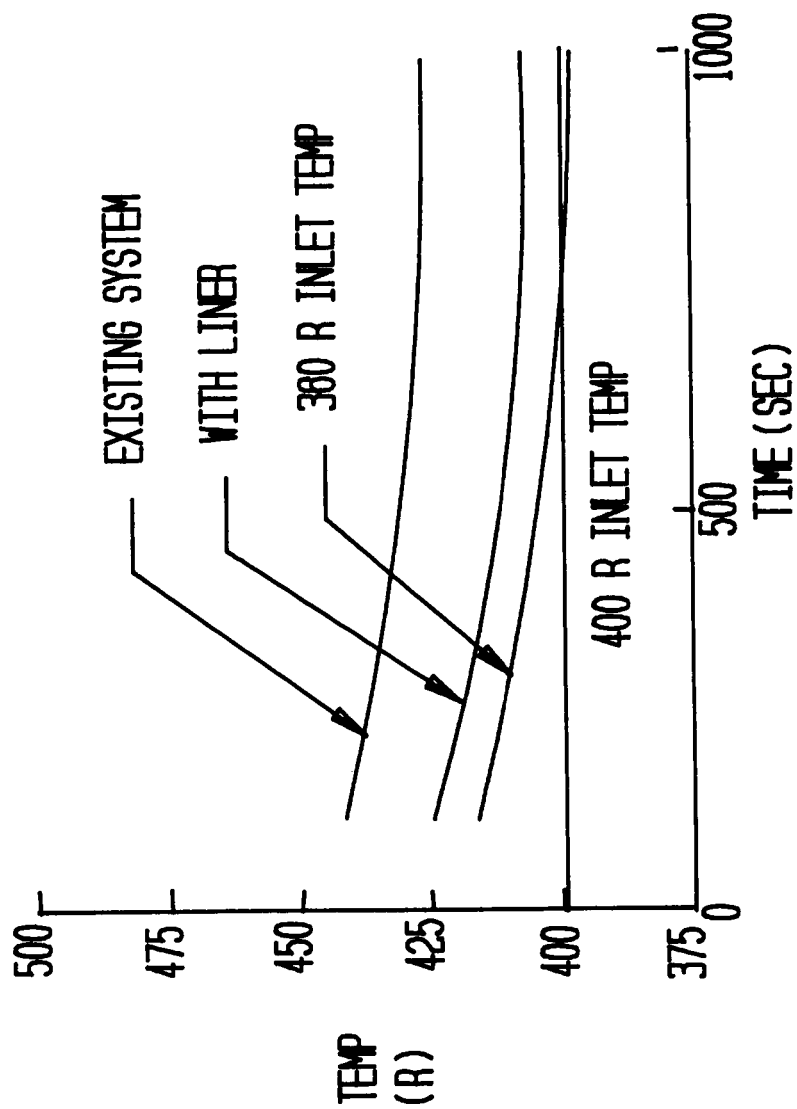


Figure 20. Effect of Liner (Going Cold) and Effect of Colder Inlet Temperature

and after 1000 seconds the insulated case is 20 degrees R colder.

Test Cell Bypass

The program could be easily modified to check the relative value of the mixing header liner vs. a test cell bypass. Since the bypass was a competing concept this was done and showed some interesting results. Figure 20 shows a case with a 360 degree R inlet temperature and the current ducting configuration (no liner). Currently the ASTF expansion turbines will not reach their rated cold temperature during a small engine test because they are rated at 300 lb/sec where actual flows are in the 50 lb/sec range. If flows could be increased through the turbines, mixing header inlet temperatures in this range (360 degree R) could be reached. Adding a test cell bypass, would increase flow through the mixing header and the expansion turbines, allowing the colder inlet temperatures, thus temperature response with a bypass is quicker than using the proposed liner. The test cell bypass proposal has been installed in ASTF and seems to work as expected although checkout time has been limited so far.

CHAPTER IV

SUMMARY AND CONCLUSIONS

In summary of the work in this thesis several interesting facts were brought out. The heat transfer coefficient in the mixing header was 13 times that found for a similar straight pipe and most of the temperature loss/gain occurred in the mixing header. A equivalent resistance model of the ducting showed that the liner increased the system resistance to heat transfer by a factor of two with the largest resistance being the liner's air gap. The air gaps resistance was twice that of the next two highest which were the insulation and the free convection on the outside of the ducting. A finite difference approach was used to calculate the temperature response of the system with and without the liner installed. It was found that the liner works as expected and speeds response times. It was also shown by a limited analysis that the test cell bypass already installed can be more effective than the liner would be.

The mixing header liner appears to be an appropriate solution to the problem of heat transfer in the mixing header but should not be further considered. The gains in temperature and response times could not justify the cost considering that the test cell bypass is more

effective and has been installed. Although the liner does work the magnitude of the improvement is not as great as that of the bypass for a roughly similar cost. There are also at least two problems that could make the liner less effective than expected. The liner would be connected to the header shell with ribs for structural support which might make the air see more metal weight and eliminate the advantage of using a thin liner. The liner also covers most areas in the mixing header but not all. This could also increase the apparent metal weight.

Liners have never been great successes in other areas of the Engine Test Facility. Problems have been experienced with aerodynamic forces causing cracking in liners which eventually result in mechanical failures. Based on previous experience in ETF and the above analysis in this thesis, the bypass was selected for implementation and has been installed.

The problem estimating heat transfer in ASTF shows the need to develop more accurate methods for determination of facility performance. Programs that require major changes to make their results match test data are not much help when designing new test facilities. Luckily the ASTF problem can be corrected because the facility's capabilities extend well beyond what is required. If it had not been for these additional capabilities the

facility might have been seriously hampered and would not have been able to live up to the expectations it was designed for. It would seem very unwise to design a facility that depends on temperature as one of its major design criteria without a model which can identify major problem areas such as the mixing header in ASTF. The work done in this thesis illustrates clearly the importance that modeling plays (or should play) in the design of new test facilities.

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REFERENCES

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APPENDIX

```

D Line# 1      7      Microsoft FORTRAN77 V3.20 02/84
1
2 C      MAIN
3 C      THIS PROGRAM SOLVES HEAT TRANSFER IN A PIPE
4      DIMENSION T(50,4), TINIT(50)
5      OPEN (2,FILE='INITIAL',STATUS='OLD')
6      READ(2,100)SPHT,DENS,COND,HCOEI,HCOEO,TAMB,T1,EL,FO,AREA,NOINT
7      COND=COND/3600
8      HCOEI=HCOEI/3600
9      HCOEO=HCOEO/3600
10 C     READ INITIAL VALUES OF TEMPERATURES IN MATERIAL
11     N = NOINT+1
12     READ(2,101)(TINIT(I),I=1,N)
13 C     COMPUTE INITIAL VALUES OF DELTA X, AND DELTA TIME
14     TIME=0.
15     TIMEP=100.
16     TAVG=T1
17     XINT=NOINT
18     DELX=EL/XINT
19     WRITE(*,*)'DELX=',DELX
20     DTIME=FO*DENS*SPHT*DELX**2/COND
21     WRITE(*,*)'DTIME=',DTIME
22     WRITE(*,199)
23     WRITE(*,200)TIME,(TINIT(I),I=1,N)
24 C     GET COEFFICIENTS OF SYSTEM
25     COEFI=HCOEI/COND
26     COEFO=HCOEO/COND
27 21     T(1,1)=0.
28     T(1,2)=-4.-2.*COEFI*DELX
29     T(1,3)=2.
30     T(N,1)=2.
31     T(N,2)=-4.-2.*COEFO*DELX
32     T(N,3)=0.
33     DO 20 I=2,NOINT
1 34     T(I,1)=1.
1 35     T(I,2)=-4.
1 36 20     T(I,3)=1.
37     T(1,4)=-2.*TINIT(2)+2.*COEFI*DELX*TINIT(1)-4.*COEFI*DELX*TAVG
38     T(N,4)=-2.*TINIT(N-1)+2.*COEFO*DELX*TINIT(N)-4.*COEFO*DELX*TAMB
39     DO 30 I=2,NOINT
1 40 30     T(I,4)=-TINIT(I-1)-TINIT(I+1)
41 C     COMPUTES VALUES. PRINT OUT ABOUT EVERY 250 SECONDS
42     TIME=TIME+DTIME
43     CALL TRDG(T,N)
44 C     PUT NEW VALUES INTO TINIT AND SEE IF TIME TO PRINT OUT
45     DO 40 I=1,N
1 46 40     TINIT(I)=T(I,4)
47 C     CALCULATE EXIT TEMP FOR HEADER
48     BB=(310*.24)/(HCOEI*AREA)
49     T2=(T(1,4)+(BB-.5)*T1)/(BB+.5)
50     TAVG=(T1+T2)/2
51     IF(TIME-TIMEP)21,22,22
52 22     WRITE(*,*)'T2=',T2
53     WRITE(*,*)'T1=',T1
54     WRITE(*,*)'TAVG=',TAVG
55     WRITE(*,200)TIME,(T(I,4),I=1,N)
56     TIMEP=TIME+250
57     IF(TIME-1500.)21,23,23
58 23     CONTINUE
59     STOP

```

```

D Line# 1      7
      60 100    FORMAT(10F7.2,I5)
      61 101    FORMAT(8F10.2)
      62 199    FORMAT(10X,'SOLUTION TO HEAT FLOW PROBLEM')
      63 200    FORMAT(5X,F6.1,'SECONDS'/(1X,10F8.2))
      64        END
      65        SUBROUTINE TRDG(X,N)
      66 C      THIS SUBROUTINE SOLVES A TRIDIAGONAL SYS OF EQ BY DIRECT
      67 C      GAUSS ELIMINATION
      68        DIMENSION X(50,4),A(50),B(50),C(50),D(50)
      69        DO 30 I=1,N
1      70        B(I)=X(I,1)
1      71        D(I)=X(I,2)
1      72        A(I)=X(I,3)
1      73 30    C(I)=X(I,4)
      74        DO 10 I=2,N
1      75        R=B(I)/D(I-1)
1      76        D(I)=D(I)-R*A(I-1)
1      77 10    C(I)=C(I)-R*C(I-1)
      78 C      BACK SUBSTITUTION.
      79        C(N)=C(N)/D(N)
      80        DO 20 I=2,N
1      81        J=N-I+1
1      82 20    C(J)=(C(J)-A(J)*C(J+1))/D(J)
      83        DO 40 I=1,N
1      84 40    X(I,4)=C(I)
      85        RETURN
      86        END

```

EXAMPLE OF INPUT DATA FILE INITIAL

.11	487.00	20.00	.50	.64	520.00	780.00	.33	1.00	90.00	20
568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00
568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00	568.00
568.00	568.00	568.00	568.00	568.00	568.00	568.00				

55

POSITION NUMBER

INPUT

1	SPECIFIC HEAT	BTU/LBM R
2	DENSITY	LBM/FT3
3	CONDUCTIVITY	BTU/HR FT R
4	CONV HEAT TRANS INSIDE	BTU/HR FT2 R
5	CONV HEAT TRANS OUTSIDE	BTU/HR FT2 R
6	TEMP AMBIENT	R
7	TEMP AVERAGE IN DUCT	R
8	DUCT WALL THICKNESS	FT
9	FORIER NUMBER	
10	DUCTING SURFACE AREA	FT2
11-31	INITIAL TEMP THROUGH WALL	R

EXAMPLE OF COMPUTER PROGRAM DATA OUTPUT

THE PROGRAMS OUTPUT PROVIDES DELX AND DTIME USED AT THE FIRST AND SECOND LINES OF THE PRINTOUT. IT THEN PROVIDES THE INITIAL TEMPERATURES THOUGH THE WALL OF THE DUCTING. THE NEXT LINE IS THE TIME OF THE NEXT SET OF DATA FOLLOWED BY T2, T1 AND TAVG. THE CURRENT TEMPERATURE THROUGH THE DUCTING WALL IS THEN PROVIDED.

DELX= 5.500000E-003

DTIME= 19.8878600

SOLUTION TO HEAT FLOW PROBLEM

.00SECONDS

516.00	515.50	515.00	514.50	514.00	513.50	513.00	512.50	512.00	511.50
511.00	510.50	509.00	507.00	506.00	505.00	504.00	503.00	502.00	501.00
500.00									

T2= 549.4130000

T1= 568.0000000

TAVG= 558.7067000

119.3SECONDS

531.05	526.08	522.05	518.94	516.62	514.93	513.67	512.69	511.83	511.02
510.18	509.29	508.35	507.37	506.38	505.42	504.53	503.76	503.15	502.71
502.48									

T2= 552.6510000

T1= 568.0000000

TAVG= 560.3256000

358.0SECONDS

537.49	533.18	529.26	525.76	522.66	519.95	517.61	515.60	513.86	512.35
511.04	509.88	508.85	507.92	507.09	506.35	505.70	505.15	504.63	504.32
504.05									

T2= 554.3702000

T1= 568.0000000

TAVG= 561.1851000

616.5SECONDS

540.90	537.03	533.38	529.98	526.85	523.98	521.38	519.04	516.94	515.07
513.42	511.96	510.68	509.55	508.58	507.73	507.01	506.40	505.88	505.47
505.14									

T2= 555.4161000

T1= 568.0000000

TAVG= 561.7081000

855.2SECONDS

542.98	539.38	535.95	532.71	529.66	526.81	524.18	521.76	519.55	517.54
515.73	514.11	512.66	511.38	510.25	509.28	508.43	507.71	507.11	506.61
506.22									

T2= 556.2703000

T1= 568.0000000

TAVG= 562.1351000

1113.7SECONDS

544.68	541.31	538.08	534.99	532.05	529.28	526.69	524.26	522.02	519.95
518.06	516.34	514.79	513.40	512.16	511.06	510.11	509.29	508.59	508.01
507.54									

T2= 556.9055000

T1= 568.0000000

TAVG= 562.4528000

1352.4SECONDS

545.94	542.75	539.67	536.71	533.86	531.19	528.65	526.26	524.02	521.95
520.03	516.27	516.66	515.20	513.89	512.72	511.69	510.79	510.01	509.36
508.82									

T2= 557.4839000

T1= 568.0000000

TAVG= 562.7419000

1610.9SECONDS

547.09	544.06	541.13	538.29	535.57	532.97	530.50	528.16	525.95	523.89
521.97	520.19	518.55	517.05	515.69	514.46	513.37	512.40	511.55	510.82
510.21									

Stop - Program terminated.

VITA

Gus William Deibner was born in Mountain Home, Arkansas on 21 June 1960. He attended elementary school and high school in the Mountain Home Public School System. In 1978 he entered the University of Arkansas at Fayetteville and received a Bachelor of Science degree in Mechanical Engineering in December of 1982.

In December of 1982 he accepted a commission in the United States Air Force and was stationed at Arnold Engineering Development Center to work in the Engine Test Facility. During this period he continued his studies on a part time basis at The University of Tennessee Space Institute. In March of 1988 he was awarded the Master of Engineering degree in Mechanical Engineering