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UNSTEADY HEAT TRANSFER RESULTING FROM
 THE RAPID CHARGING OF AN EVACUATED
 TANK WITH CONDUCTING WALLS

THESIS

Dennis E. Lilekis
 Second Lieutenant, USAF

AFIT/GA/AA/86D-9

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THESIS

Presented to the Faculty of the School of Engineering
of the Air Force Institute of Technology

Air University

In Partial Fulfillment of the
Requirements for the Degree of
Master of Science in Astronautical Engineering



Dennis E. Lillekis, B.S.
Second Lieutenant, USAF

December 1986

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Preface

The time will come when diligent research over long periods will bring to light things which now lie hidden. A single lifetime ... would not be enough for the investigation of so vast a subject ... Many discoveries are reserved for ages yet to come ...

Seneca, Natural Questions
Book 7, first century

As is so often the case in scientific research, the experimenter, in his desire to uncover the truth, unearths more questions than he answers. I can now empathize with that scientist after having worked for so long on this thesis. As data was slowly acquired, so too were theories to explain the results. However, further data reduction would often prove the hypothesis in error, and a re-thinking was in order. Yet when data and theory finally agree, you enjoy a wonderful sense of accomplishment. As Democritus of Abdera once said, "I would rather understand one cause than be King of Persia."

None of this, however, would have been possible without the help of several individuals. First and foremost is Dr. James Hitchcock, my thesis advisor. I first became interested in heat transfer during an introductory course which he had taught. Throughout my entire heat transfer curriculum, I have always been eager to learn from and study under Dr. Hitchcock. Secondly, I would like to thank 2Lt.

James Shoemaker, a member of the AFWAL Propulsion Lab, for the use of his digitization equipment. Thanks also go out to Mr. John Brohas and the AFIT lab technicians for their help in constructing my apparatus.

Computer used for word processing Apple IIc
Program used Appleworks
Printer used Imagewriter II

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List of Symbols

	<u>Quantity</u>	<u>Units</u>
A	Area	ft ²
α	Thermal Diffusivity	ft ² /s
c_p	Specific Heat at Constant Pressure	lbf-ft/lbm-R
c_v	Specific Heat at Constant Volume	lbf-ft/lbm-R
e	Stored Energy per Unit Mass	BTU/lbm
E	Voltage or Stored Energy	Volts or BTU
G	Amplifier Gain	---
γ	Ratio of Specific Heats c_p/c_v	---
g_c	Conversion Constant	32.2 lbm-ft/lbf-s ²
h	Convection Coefficient	BTU/ft ² -s-F
i	Total Enthalpy per Unit Mass	BTU/lbm
I	Current	Amps
m	Mass	lbm
M	Mach Number	---
P	Pressure	lbf/ft ²
q	Heat Flux	BTU/ft ² -s
Q	Heat	BTU
R	Resistance or Gas Constant	Ohms or lbf-ft/lbm-R
ρ	Density	lbm/ft ³
s	Entropy per Unit Mass	BTU/lbm
S	Entropy	BTU
t	Time	s
T	Temperature	F or R

u	Velocity	ft/s
V	Volume	ft ³

Subscripts

c.v. Control Volume

f Final condition

g Gage property

l Initial condition or index of summation

N A given time increment in a summation

o Ambient or atmospheric condition

p Pressure in tank equal to atmospheric pressure

s Surface condition

t Derivative with respect to time

th Nozzle throat condition

uc Unchoked condition

x Derivative with respect to x

Overstrike

• Derivative with respect to time

Abstract

↙ The two classical problems of charging a tank with an ideal gas concern themselves only with the initial and final states of the gas. The first problem involves an insulated tank. From initial conditions, the final temperature of the gas in the tank can be determined. The second problem involves a tank with conducting walls. In this case, it is the total heat transferred from the gas to the tank walls which can be found. This study went beyond these equilibrium analyses to acquire a fundamental understanding of the transient phenomenon involved in charging a tank.

By way of experimentation, transient pressure and heat flux measurements were made on an evacuated cylindrical tank as it was charged by sudden exposure to the atmosphere. Then, an average convection coefficient and driving temperature potential for the tank were found. A method was then developed for estimating heating rates for other tanks and different ambient conditions. Also, information about the transient flow dynamics occurring within the tank was gleaned from the heat transfer data. This information includes rates of kinetic, internal, and thermal energy generation and dissipation as well as rates of entropy creation in the gas.

↑

UNSTEADY HEAT TRANSFER RESULTING FROM THE RAPID CHARGING OF AN EVACUATED TANK WITH CONDUCTING WALLS

I. Introduction

The classical problem of charging a tank with an ideal gas is well documented for two conditions [1:80-87] [2:203-210]. If an insulated tank is considered, the final equilibrium temperature can be determined. If, on the other hand, the tank has a conducting wall, the total heat transferred from the gas to the wall can be determined. In either case, the solution is derived from an elementary application of the first law of thermodynamics. The change in stored energy in the tank must equal the difference between the energy which came into the tank, be it internal, kinetic, or flow energy, and the heat lost from the fluid to the tank walls.

In contrast, little if any work has been done to understand the actual transient process involved in charging a vessel. Thorough literature searches, both manual and computer, revealed no useful information regarding this process. Perhaps one reason for the lack of information is that no analytical solution exists for the unsteady process of charging a tank with a fluid. Because of the complexity of the flow, an analytical solution would require that the

Navier-Stokes equations be solved in their entirety. Significant simplifications cannot be made to these equations since both compressibility and diffusive transport of momentum and energy dominate the entire process. The diffusive processes are what dictate the rate of kinetic energy dissipation in the tank as well as the rate of heat transfer to the walls. As a consequence, a simple energy balance will not work. The only alternative, then, is an experimental investigation. With the rapid-response, thin-film resistance gages currently available, the rate of heat transfer encountered during the charging of a tank may be observed directly, thereby yielding some insight into the dynamics of the flow.

There were three main objectives of the study. The first objective was to detail the distribution of the measured thermal energy along the walls of the tank by direct observation. The second objective was to use the measured heat flux data to derive an average convection coefficient and its associated temperature difference. These could then be used to predict heat flux rates encountered during the charging of other tanks. Finally, the third objective was to gain some knowledge of the flow dynamics that occur within a tank during the charging process.

II. Charging Theory

Charging is the process by which a vessel containing a fluid at low pressure is filled from a reservoir of fluid at a high pressure. The resulting pressure gradient forces fluid from the reservoir into the vessel until the pressures inside and out are equal. The work done in moving this mass of fluid is then manifested in the internal and kinetic energies of the fluid in the vessel as well as any heat lost to the surroundings.

For example, assume that a perfectly evacuated, insulated tank is suddenly opened to atmospheric pressure. This is one of the two classical problems that has been solved. The mass which enters the tank carries with it internal energy as well as flow work done by the atmosphere in pushing the mass into the tank. This total enthalpy per unit mass is just $c_p T_0$. After the air comes to rest in the tank, it has energy per unit mass of $c_v T_f$. See Figure 2-1. Since the tank is perfectly insulated, no energy is lost to the walls by way of heat transfer. Thus, a simple energy balance can be written to find the final temperature of the air in the tank. Enthalpy in, $mc_p T_0$, must equal energy stored in the tank, $mc_v T_f$. The final temperature of the gas, therefore, equals $T_0 c_p / c_v$. For air, if $T_0 = 530$ R, the final temperature would be 742 R, an increase of 212 R.

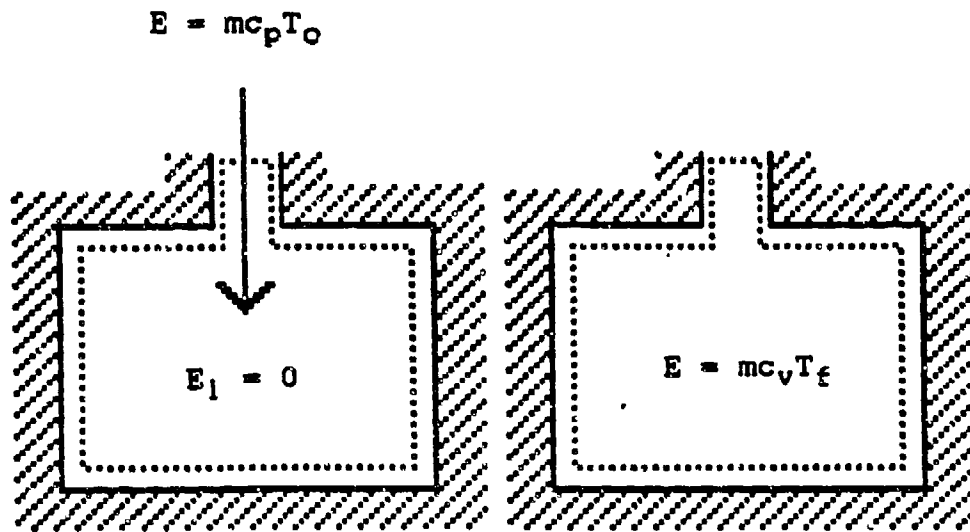


Figure 2-1 The Charging of an Evacuated, Insulated Tank

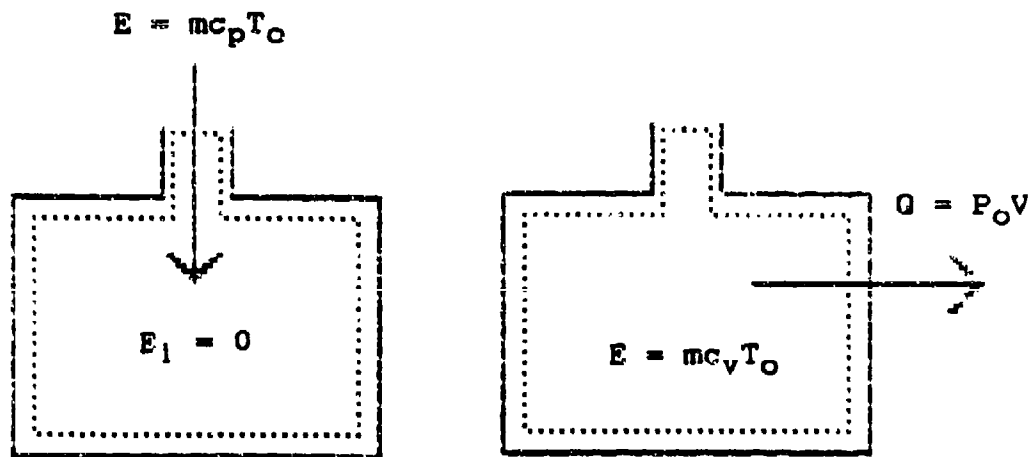


Figure 2-2 The Charging of an Evacuated Tank with Conducting Walls

In the second classical problem, the tank walls are conductors of heat, and the total heat transferred to these walls is sought. Once again, assume an evacuated tank that is opened suddenly to the atmosphere. After the air in the tank comes to equilibrium, it will have the same pressure and temperature as it had before entering the tank. See Figure 2-2. All the flow work done by the atmosphere will be manifested in the heat transfer to the tank walls. This flow work per unit mass is P_0/ρ_0 . Therefore $Q = mP_0/\rho_0 = P_0V$ where P_0 is the atmospheric pressure and V is the volume of the tank.

During the transient charging period, the actual rise in fluid temperature within the uninsulated tank is a complex function of the rate of kinetic energy dissipation in the tank as well as the rate of heat dissipation to the tank walls. To determine the heat that is lost as a function of time, the first law of thermodynamics in its open, unsteady form is needed. This law states that the increase in stored energy within the tank must equal the enthalpy added by the mass which enters the tank minus the heat that is removed from the gas at each instant in time. Consider the control volume pictured in Figure 2-3. Mathematically,

$$\int dE = \int \dot{m} i dt - \iint q dA dt \quad [2-1]$$

Also,

$$dE = d(mc_v T + mu^2/2g_c) \quad [2-2]$$

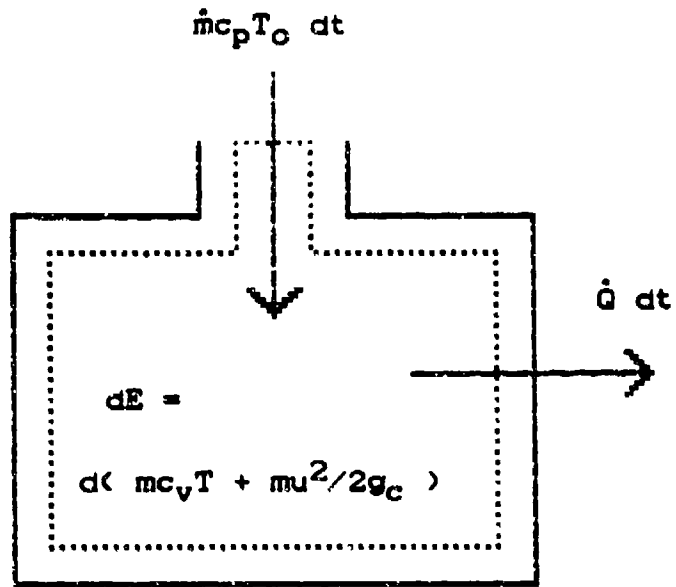


Figure 2-3 Control Volume Used in the Transient Energy Balance of Equation [2-3]

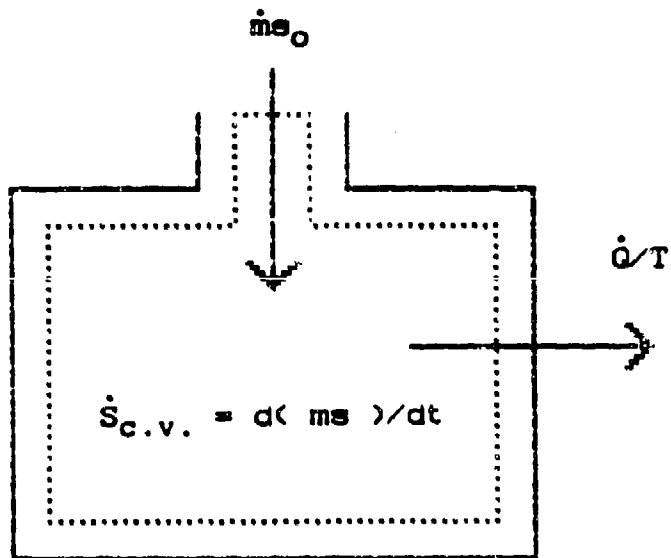


Figure 2-4 Control Volume Used in the Transient Entropy Balance of Equation [2-4]

T and u represent some average temperature and velocity of the mass within the tank.

Equation [2-2] can be substituted into Eq. [2-1] and, assuming constant enthalpy entering the tank, the resulting integrals evaluated, yielding

$$m[c_v T + u^2/2g_c] - m_1 c_v T_0 = (m - m_1)c_p T_0 - Q \quad [2-3]$$

The term $m_1 c_v T_0$ accounts for the initial mass in the tank if it was not fully evacuated. Q is the total heat that has been transferred from the gas to the tank walls.

A second law analysis of this transient problem can also be accomplished. The second law of thermodynamics states that the rate at which entropy is created within the control volume must be greater than or equal to the rate at which entropy enters with the entering mass minus the rate at which entropy leaves by way of heat transfer. See Figure 2-4.

This can be expressed as

$$\dot{S}_{c.v.} \geq \dot{m}s_0 - \dot{Q}/T \quad [2-4]$$

$\dot{S}_{c.v.}$ is the rate of entropy increase within the tank, s_0 is the specific entropy of the fluid entering the tank, and \dot{Q}/T is the rate at which entropy leaves the gas due to heat transfer. T is the temperature of the inner tank wall which, for walls having large heat capacity and conductivity, equals T_0 . The entropy per unit mass of the air within the tank can be found by

$$s(t) = s_0 + c_p \ln[T(t)/T_0] - R \ln[P(t)/P_0] \quad [2-5]$$

Thus, data derived from Eq. [2-5] can be numerically differentiated to find $\dot{S}_{c.v.}(t)$ providing, of course, that the mass within the tank is known since

$$\dot{S}_{c.v.}(t) = d(m(t) s(t)) / dt \quad [2-6]$$

III. Thin-film Gage Theory

Surface Temperatures

Thin-film resistance gages were used to measure the temperature at the surface of a solid substrate. On the surface of the substrate is a thin strip of platinum through which a small, constant current is passed. The resistance change of the platinum film is directly proportional to its change in temperature above some reference temperature. The resistance of the film at a given temperature is

$$R_g(T) = R_g(T_{ref}) + (T - T_{ref}) (\Delta R / \Delta T)_g \quad [3-1]$$

$(\Delta R / \Delta T)_g$ is a property of an individual gage determined from calibration. [3:5]

The resistance change of the film can be measured by measuring the voltage change across it. A gage can be connected to a bridge and amplifier circuit as shown in Figure 3. The bridge element resistor that is shown in series with the gage should be large enough to ensure that the change in gage resistance does not induce an appreciable change in current. For this type of constant-current bridge circuit, the resistance change is given by the measured voltage change.

$$\Delta R_g = \Delta E (R + R_g) / R I_g G \quad [3-2]$$

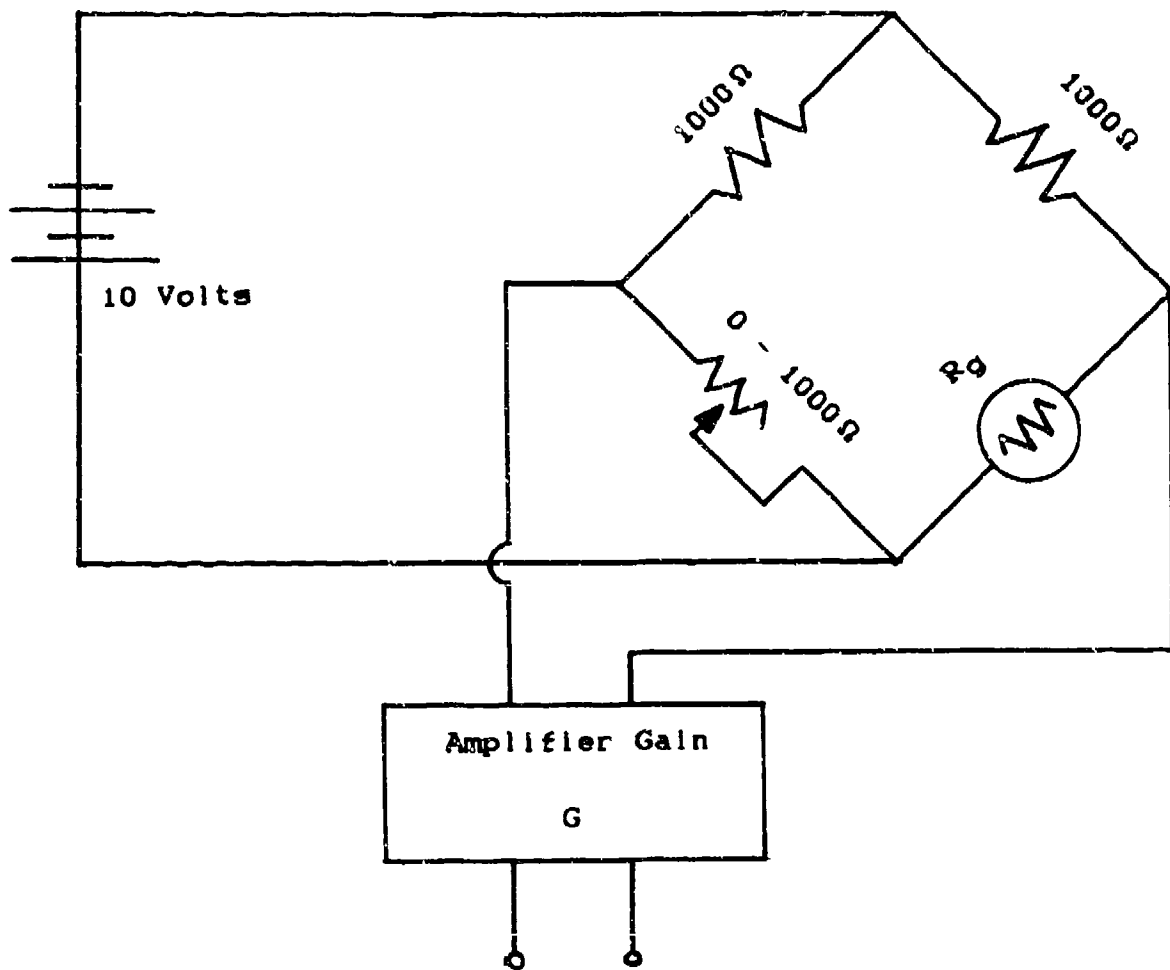


Figure 3 Bridge Circuit Used with the Thin-film Gages

E is the measured voltage across the bridge, I_g is the constant current through the gage, and G is the gain of the amplifier. [3]

By using Eq. [3-1] and [3-2], the change in voltage can be directly converted to change in temperature with the following expression.

$$\Delta T = \Delta E (R + R_g(T_{ref}) + R' [T - T_{ref}])^2 / RVGR' \quad [3-3]$$

where $R' = (\Delta R / \Delta T)_g$. Note that current has been eliminated in favor of the applied voltage, V.

Surface Heat Fluxes

Although thin-film gages measure temperature, heat flux can be determined by using a solution to the basic heat equation with two assumptions imposed. First, the gage is treated as a semi-infinite solid during the test time. Second, the heat flux through the gage is assumed to propagate in a one-dimensional fashion. These elements allow heat flux rates to be determined from the temperature data.

To solve for the surface heat flux in terms of surface temperature, the one-dimensional heat equation can be used

$$T^+_t = \alpha T^+_{xx} \quad [3-4]$$

where $T^+ = T^+(x,t) = T(x,t) - T(x,0)$.

The boundary conditions are

$$\begin{aligned} T^+(0,t) &= T_s^+ \\ T^+(x \rightarrow \infty, t) &\rightarrow 0 \\ T^+(x,0) &= 0 \end{aligned}$$

The solution to this equation can be found by using a similarity variable or by the the Laplace transform technique. [4:384-390]

$$T^+ = T_s^+ \operatorname{erfc}(x / [2(\alpha t)^{1/2}]) \quad (3-5)$$

An energy balance at the surface of the gage requires that the heat flux incident on the gage must equal the heat flux into the gage. Mathematically,

$$q(t) = -k T_x^+|_{x=0} \quad (3-6)$$

Cook and Felderman [5] used Eq. [3-5] and [3-6] along with Duhammel's superposition method [4:403-411] to obtain a numerical solution for surface heat flux given a discrete number of measured surface temperatures. Their equation, slightly rearranged, is

$$q(t_N) = 2 \sqrt{\frac{\rho c k}{\pi}} \left[\frac{T_s^+(t_0)}{2t_N^{1/2}} + \sum_{l=1}^N \frac{T_s^+(t_l) - T_s^+(t_{l-1})}{(t_N - t_l)^{1/2} + (t_N - t_{l-1})^{1/2}} \right] \quad (3-7)$$

Thus, given the measured temperatures from a thin-film resistance gage, surface heat fluxes may be calculated as a function of time.

IV. The Experiment

Apparatus

The Tank. The tank was fabricated from a section of stainless steel pipe 0.375 inch thick. The tank had an inner diameter of 10 inches and a length of 35 13/16 inches. Holes were bored and tapped at various locations on the tank to accommodate instrumented brass plugs. See Figure 4-1. The top and bottom plates were made from steel and provided with sites for instrumented plugs as well. The bottom plate was welded in place, while the top one was bolted to the tank. This provided a means of entering the tank should the need arise. All touching metal parts that were not welded together had O-ring seals or rubber gaskets between them to prevent leakage.

The entrance to the tank consisted of a simple converging nozzle. The nozzle had an entrance diameter of 4 inches and a throat diameter of 1 inch, yielding an area ratio of 16. A rubber stopper commonly found in chemistry labs was used to plug the nozzle during evacuation. The stopper was attached to one arm of a lever which was bolted to the top of the tank. This lever was used to initiate the flow quickly by pulling the stopper out and away from the nozzle inlet.

Thin-film Gages. Five thin-film gages were used to gather data on the rates of heat transfer at 19 different

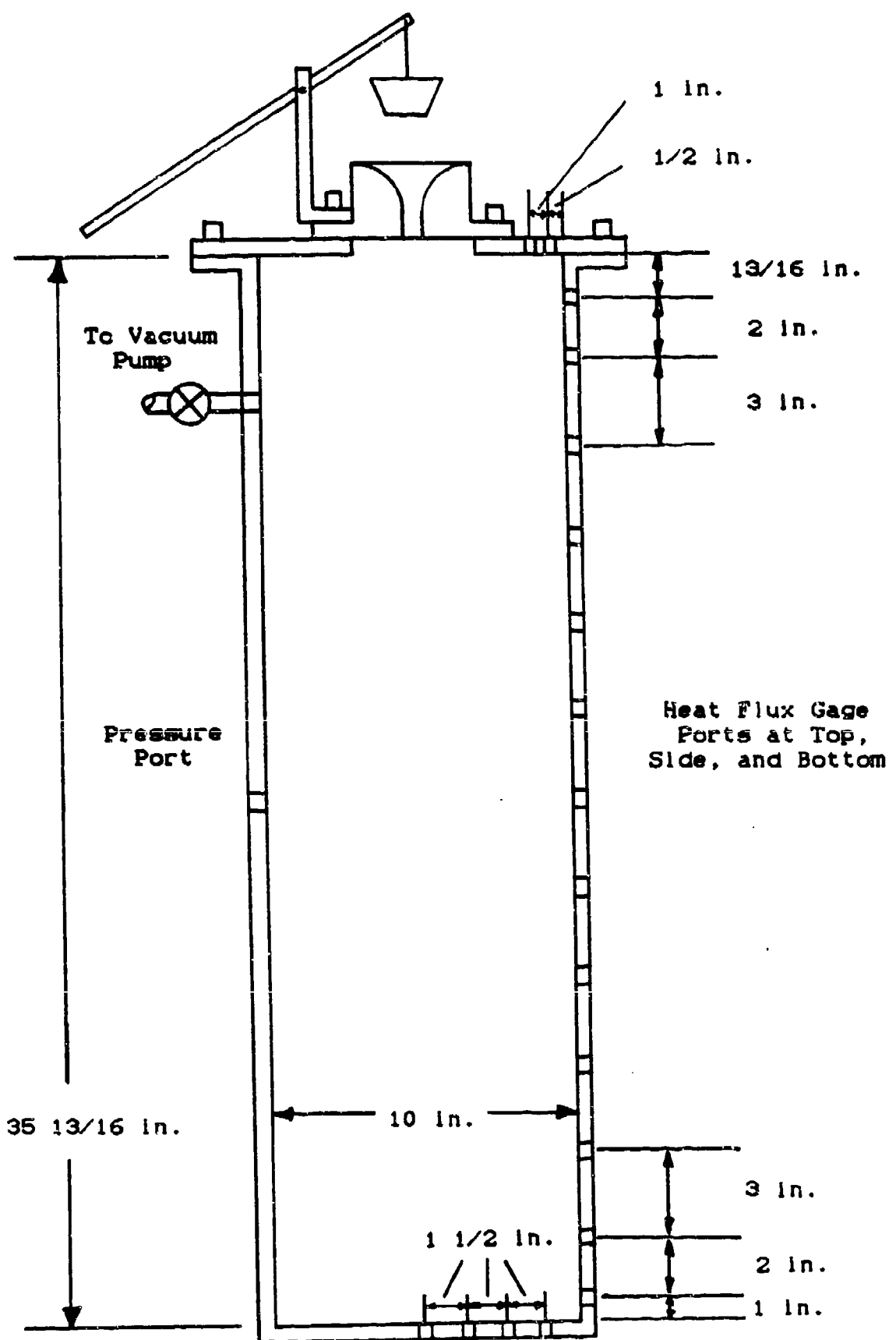


Figure 4-1 Tank Apparatus Used In Charging Experiments

stations over the tank. The gages were commercially produced and consisted of a thin strip of platinum which was deposited onto one end of a cylindrical piece of quartz. The quartz substrate was 0.25 inch long and had a diameter of 0.125 inch. Each gage was inserted into a hollow plexiglas rod which had an outer diameter of 0.3875 inch. This rod was then affixed inside a hollow brass plug. See Figure 4-2. Epoxy was used to hold the plexiglas and quartz in place as well as to ensure an airtight seal. When inserted into the tank, the center of each gage was flush with the inner wall of the tank; the edges were slightly recessed due to the curvature of the tank wall.

The gages were connected to bridge/amplifier circuits. The output from each bridge was stored using a digital waveform recorder. After a run, each of the five recorded channels was displayed on an oscilloscope, and a polaroid photograph was taken of the resulting trace. The photo was later digitized and the data stored in a computer for data reduction.

Pressure Transducer. An Endevco 0-100 psia pressure transducer was used to measure the transient pressure within the tank. The transducer and its signal conditioner fed data to the same digital waveform recorder. A polaroid photograph was taken and digitized and the data was stored in a computer for later reduction. A mercury manometer was used to

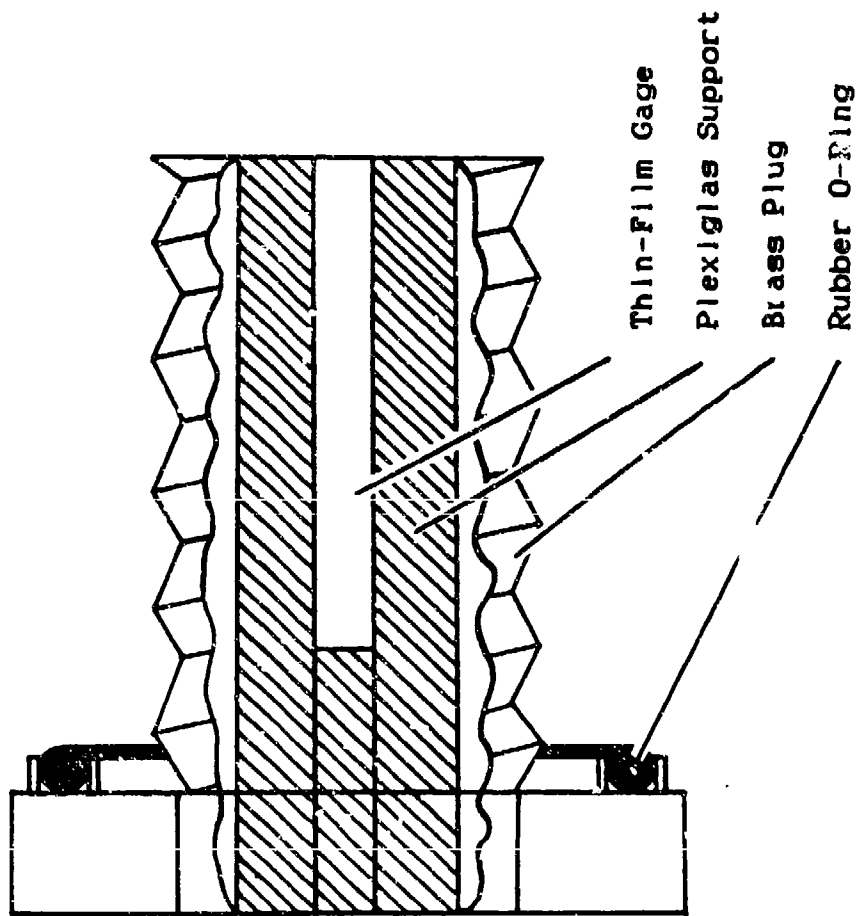


Figure 4-2 Thin-Film Gage in Brass Mounting Plug

calibrate the transducer while a mercury barometer was used to determine atmospheric pressure.

Vacuum Pump. A mechanical vacuum pump was used to evacuate the tank. The pump was connected to a valve which was attached to the side of the tank. It was consistently able to reduced the tank pressure to between 2 and 3 psia within 5 to 7 minutes.

Digital Waveform Recorder. A Datalab 8 channel digital waveform recorder was used to capture the data from the thin-film and pressure gages. It was desired to use the digitized data directly, but a suitable computer was unavailable. Thus, the information stored in the recorder was displayed on an oscilloscope, photographed, and then digitized using other equipment.

Digitizer/Computer. A Hewlet-Packard plotter/ digitizer was used to digitize the thin-film and pressure traces. This data was then input by hand into an Apple IIc computer for reduction and manipulation.

Pressure Gage Calibration

To calibrate the pressure transducer, the tank was evacuated eight different times to various pressure levels. Each time, the pressure from a mercury manometer was recorded along with the deflection of the transducer trace on the oscilloscope. These measurements gave conversion constants (from millivolts to psi) which varied by not more than 4%.

These eight constants were then averaged together to give a conversion constant of 2.88 mV/psi.

Next, to determine which location on the tank would give the "best" average pressure, the transient pressure was recorded at the top, the middle, and the bottom of the tank. There was no perceptible difference in the three traces. As a result, the pressure was assumed to be essentially constant throughout the entire tank at each instant in time. Since the location of the transducer was unimportant, the middle location was chosen for all subsequent runs as a matter of convenience.

Heat Flux Gage Calibration

Due to the difficulty in calibrating these types of gages [3], the calibration from the manufacturer was assumed to be correct. Each of the gages had different resistances and different rates of resistance change with temperature change. Care was taken during all test runs to keep track of each gage and the position on the tank which it occupied at that time.

Because thin-film gages work on a one-dimensional heat flow model (see Section III), the long times of this experiment (i.e. > 1 second) caused some concern as to whether two-dimensional effects might alter the results. In order to find out, one of the gages was exposed to a constant heat flux and the resulting signal recorded. Since an

analytic solution exists for this boundary condition, the response of the gage could be compared to a true one-dimensional model. The source of the constant heat flux was the stagnation region of a hot stream of air. This stream was generated with a hot air gun which is normally used to shrink electrical insulation. In actuality, it is the Nusselt number or convection coefficient which remains constant in stagnation region heat transfer. However, if the surface temperature of the gage is assumed to remain essentially constant, then heat flux will also be constant.

The test was crude and the hot stream only somewhat constant, but the results indicated that the gage was giving approximately one-dimensional results all the way out to 7 seconds. According to the analytic solution, a constant heat flux would result in an increase in surface temperature proportional to the square root of time. This would generate a horizontal line as plotted in Figure 4-3. Between 0 and 1 second for the low heat flux levels, there were small deviations from a horizontal line. This was due to the transient effects associated with establishing the stream of air onto the gage. The difference was much greater for the two highest heat flux levels where it was necessary to have the fan on the gun set on high. Apparently, this increased the transient flow establishment time. After the transient time period, however, the curves do approximate a straight, horizontal line. However, the curves generated with the two

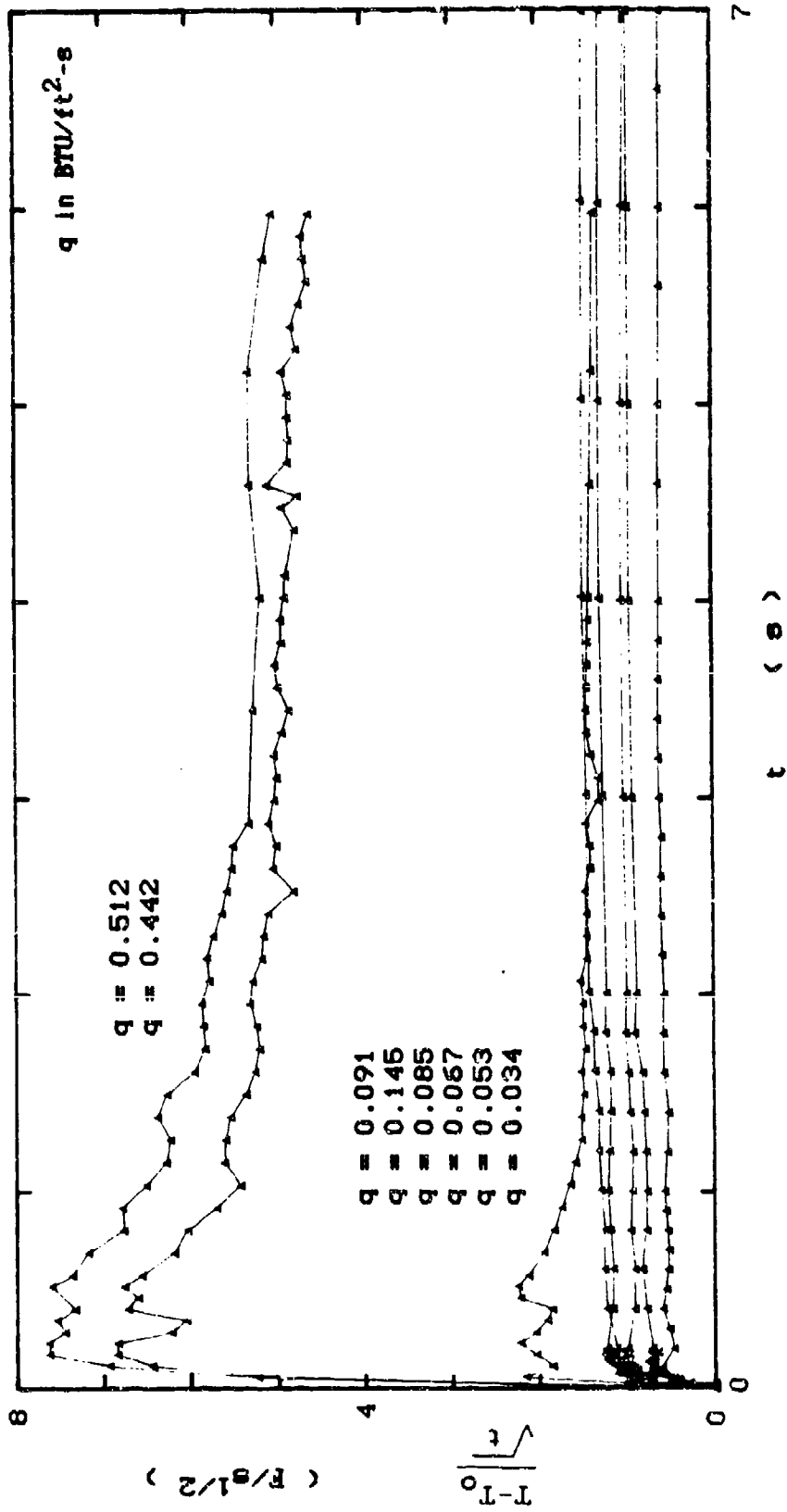


Figure 4-3 Calibration Curves for 2D Effects in Thin-film Gages

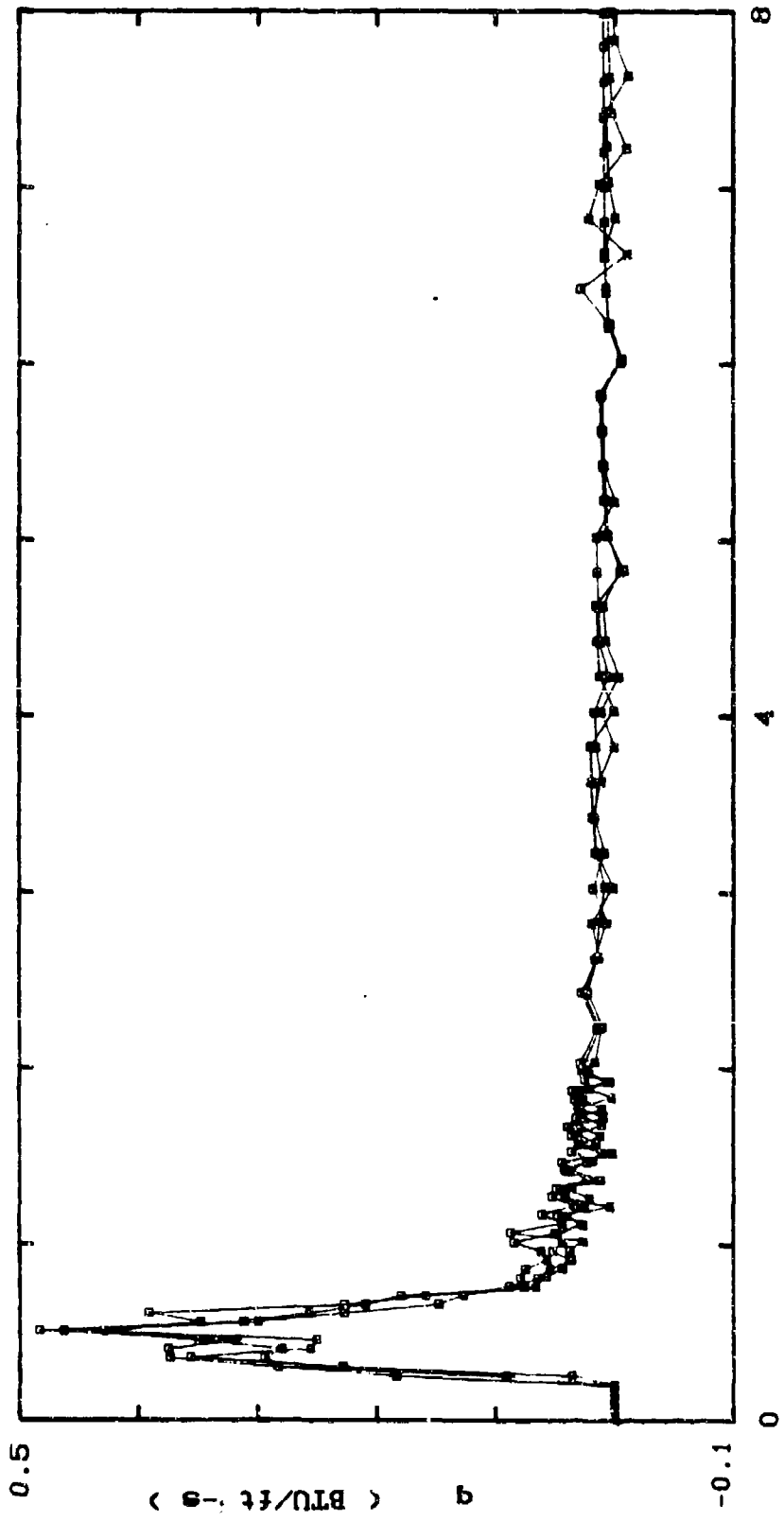


Figure 4-4 Heat Flux Curves Used in Determining Experiment Repeatability

highest heat flux levels seem to continue decreasing after the initial transients. Because these levels produced an appreciable surface temperature change (near 10 F), the heat flux was no longer a constant and was expected to decrease. As a result, no correction was employed to account for two-dimensional effects.

To ensure that the main experiment was repeatable, three preliminary runs were made with the gages mounted in the tank. The results of one of the gages is shown in Figure 4-4. The three traces were nearly identical; the slight variations due largely to the slightly differing initial pressures in the tank. Although convinced of the repeatability of the experiment, one gage was maintained in the same location throughout all subsequent runs to ensure that this assumption remained valid.

Test Procedure

Before a test was made, the electronics were given at least 20 minutes to warm up, allowing all transients to die out. The tank was then evacuated to the capacity of the vacuum pump; this was between 2 and 3 μ ia. During the evacuation, atmospheric pressure and temperature were recorded. After that, the bridge circuits were balanced, and the digital recorder was checked to ensure that the settings employed were the correct ones. The first five channels of the recorder were used for the five heat flux gages, the

sixth for the pressure transducer, and the remaining two went unused. The recorder was set to trigger from any change in the pressure transducer since, from earlier tests, it seemed to have the fastest response. The digital recorder was also set to record data for 8 seconds.

Once the tank had been evacuated, the valve between it and the pump was closed; then, the pump was turned off. The recorder was armed and the tank stopper lever was pulled down sharply to ensure a rapid opening. After the 8 seconds were over, each of the recorded channels was displayed on an oscilloscope. The image was then enlarged and expanded using the controls on the recorder. This was done to facilitate the future digitization of the photograph. Because previous tests indicated that the significant heat transfer occurred within the first 4 seconds, only this portion of the signal was photographed. To take the pictures, the grid on the oscilloscope was illuminated and the intensity of the trace made slightly brighter than the grid. The camera was set with a shutter speed of 1/30 second and an f-stop of 4. Each photograph was marked with an identifying code immediately after processing. This code was used to identify the gage and tank position for that particular trace. Figures 4-5 and 4-6 are sample photographs of a heat flux trace and pressure trace, respectively.

At a later date, the photographs were digitized using a Hewlett-Packard plotter/digitizer. Because the computer to be used in the data reduction could not be made to communicate with the digitizer, data from the digitizer had to be entered manually into the computer. This was the most laborious task of the experiment and was also a driving factor in limiting the recorded signal time to 4 seconds.

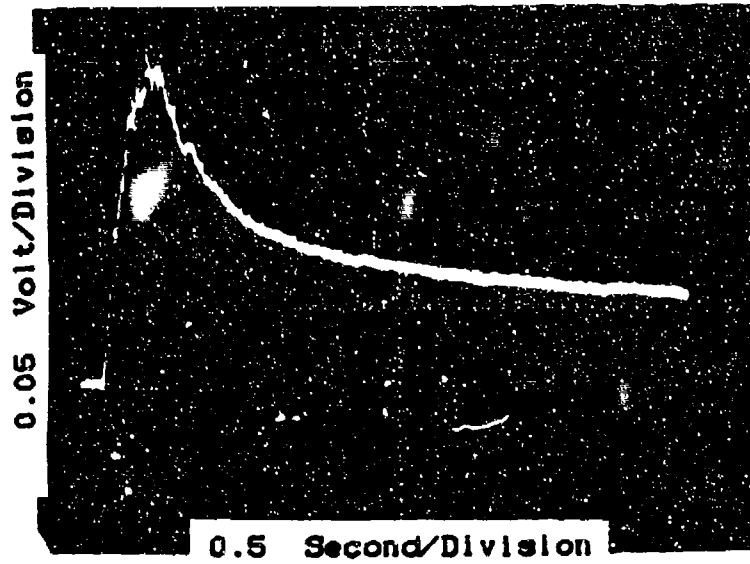


Figure 4-5 Sample Oscilloscope Output from Thin-film Gage

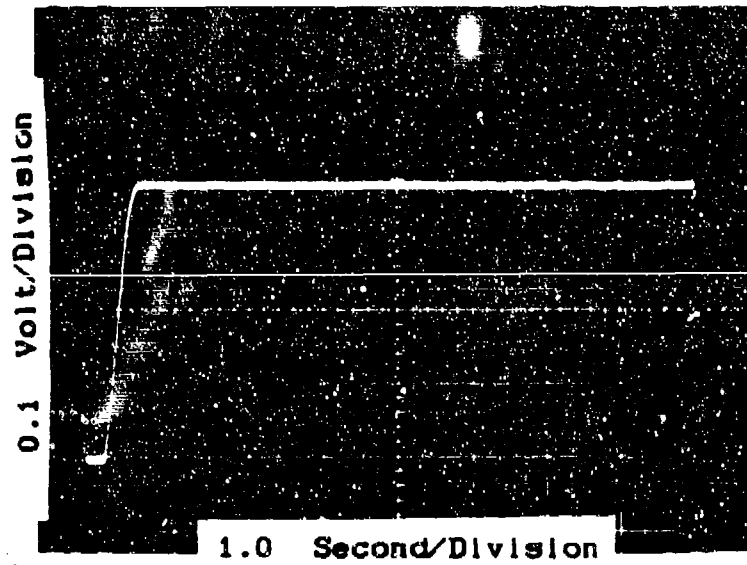


Figure 4-6 Sample Oscilloscope Output from Pressure Transducer

V. Data Reduction

General Comments

From the five different tests needed to accumulate data from all 19 locations on the tank, average initial conditions were calculated and used in all data reduction.

The initial conditions used were

$P_0 = 2066.65$	lbf/ft ²	0.6 %
$P_1 = 200.365$	lbf/ft ²	1.3 %
$T_0 = 540$	R	0.9 %

The numbers in the far right column indicated the largest percentage difference from the average.

In all subsequent calculations, numerical derivatives were calculated using a centered difference formula for interior points or a three-point formula for end points. For any integrations, cubic splines were fit to the data points and the resulting cubic polynomials integrated exactly.

Measured Data

One of the first quantities calculated from the measured data was the mass in the tank as a function of time. The only data needed for this calculation was the transient pressure data and the initial conditions. Because the entrance to the tank was a simple converging nozzle, isentropic relations could be used to calculate the mass flow as a function of the pressure in the tank. In terms of Mach

number at the throat, the mass flow through the nozzle is given by [6:84-85]

$$\dot{m} = P_0 A M (\gamma/RT_0)^{1/2} [1 + M^2(\gamma-1)/2]^{(\gamma+1)/2(1-\gamma)} \quad [5-1]$$

For a perfect gas under isentropic conditions, the Mach number can be replaced with the following expressions.

For unchoked flow,

$$M = [2/(\gamma-1)]^{1/2} [(P/P_0)^{(1-\gamma)/\gamma-1}]^{1/2} \quad [5-2]$$

For choked flow,

$$M = 1$$

P is the pressure within the tank and P_0 is the atmospheric pressure. Substituting [5-2] into [5-1] and letting $P^+ = (P/P_0)^{(1-\gamma)/\gamma}$ yields the following.

For unchoked flow,

$$\dot{m} = P_0 A [2/[(\gamma-1)RT_0]]^{1/2} [P^+-1]^{1/2} P^{+(\gamma+1)/2(1-\gamma)} \quad [5-3]$$

For choked flow,

$$\dot{m} = P_0 A (\gamma/RT_0)^{1/2} [(\gamma+1)/2]^{(\gamma+1)/2(1-\gamma)}$$

According to the measured pressure data, see Figure 5-1, the nozzle unchoked after 0.18 second, and the pressure in the tank reached atmospheric pressure after 0.46 second.

Equation [5-3] was integrated numerically from $t=0$ to $t=0.46$ second to obtain the mass in the tank as a function of time.

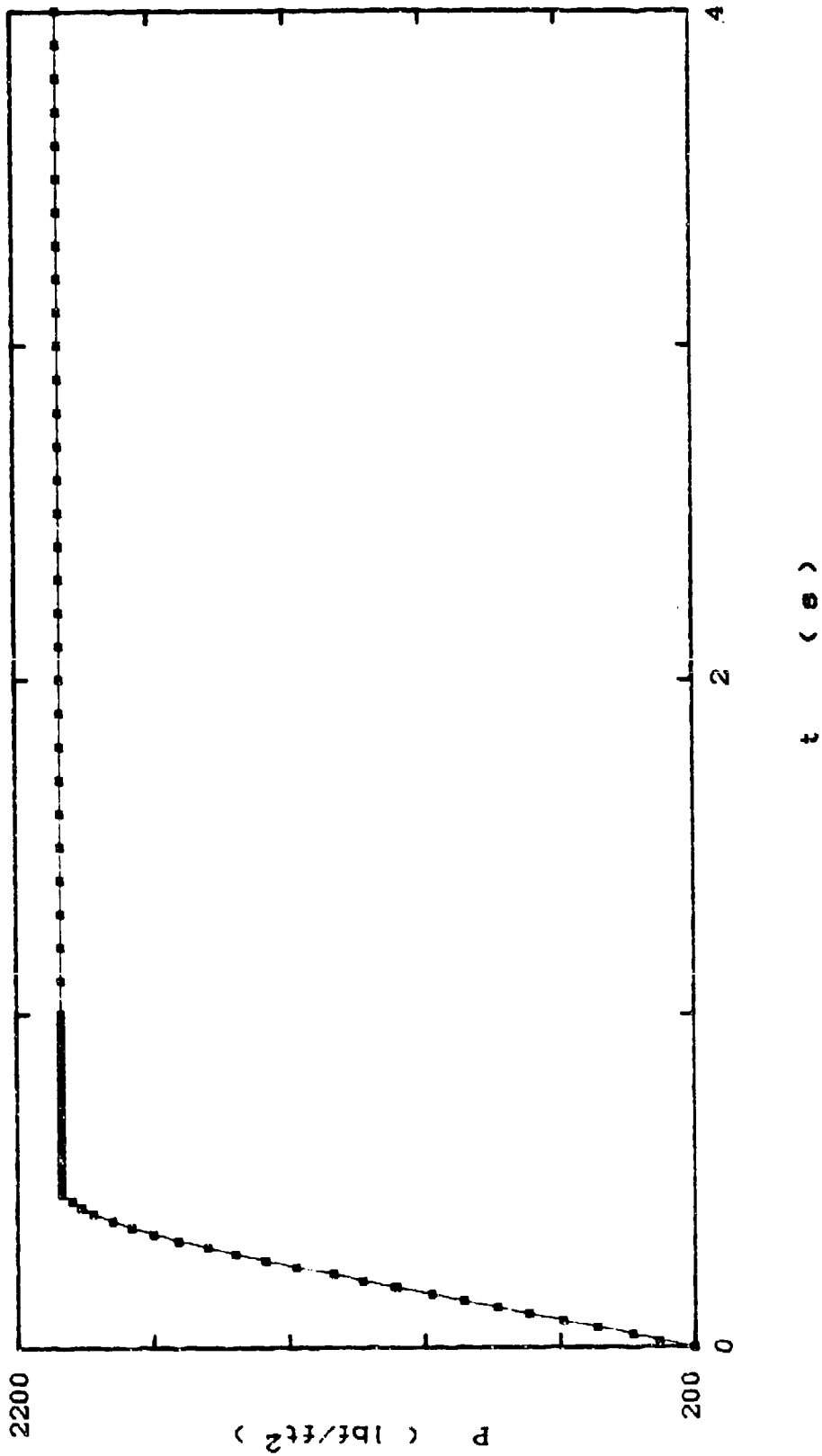


Figure 5-1 Pressure Within Tank: 0-4 Seconds

The result is shown in Figure 5-2. It should be noted that the mass in the tank at 0.46 second, 0.11017 lbm, is somewhat less than the mass the tank could hold at equilibrium, 0.11749 lbm. This indicated that the temperature of the gas was still above T_0 .

From the pressure data and the newly calculated mass data, the average temperature of the air within the tank was found as a function of time. This temperature was found by using the ideal gas law

$$T(t) = P(t) V / m(t) R \quad [5-4]$$

The results are plotted in Figure 5-3. Because the partial derivative of temperature with respect to mass is proportional to the reciprocal of the mass squared, small errors in calculating the mass were magnified when the temperature was calculated. This explains the somewhat erratic nature of Fig. 5-3. However, the general trend of increasing temperature is readily apparent.

Next, from the voltage data of the thin-film gages, surface temperatures were calculated using Eq. [3-3]. These temperatures were then used in the Cook and Felderman equation [3-7] in order to determine the surface heat fluxes. The value for $\rho c k / \pi$ used in this equation was the well documented value of $0.001819 \text{ BTU}^2/\text{ft}^4\text{-R-s}$. [3] This data spanned the time period between 0 and 4 seconds. The heat flux of each gage was calculated separately, and then it was

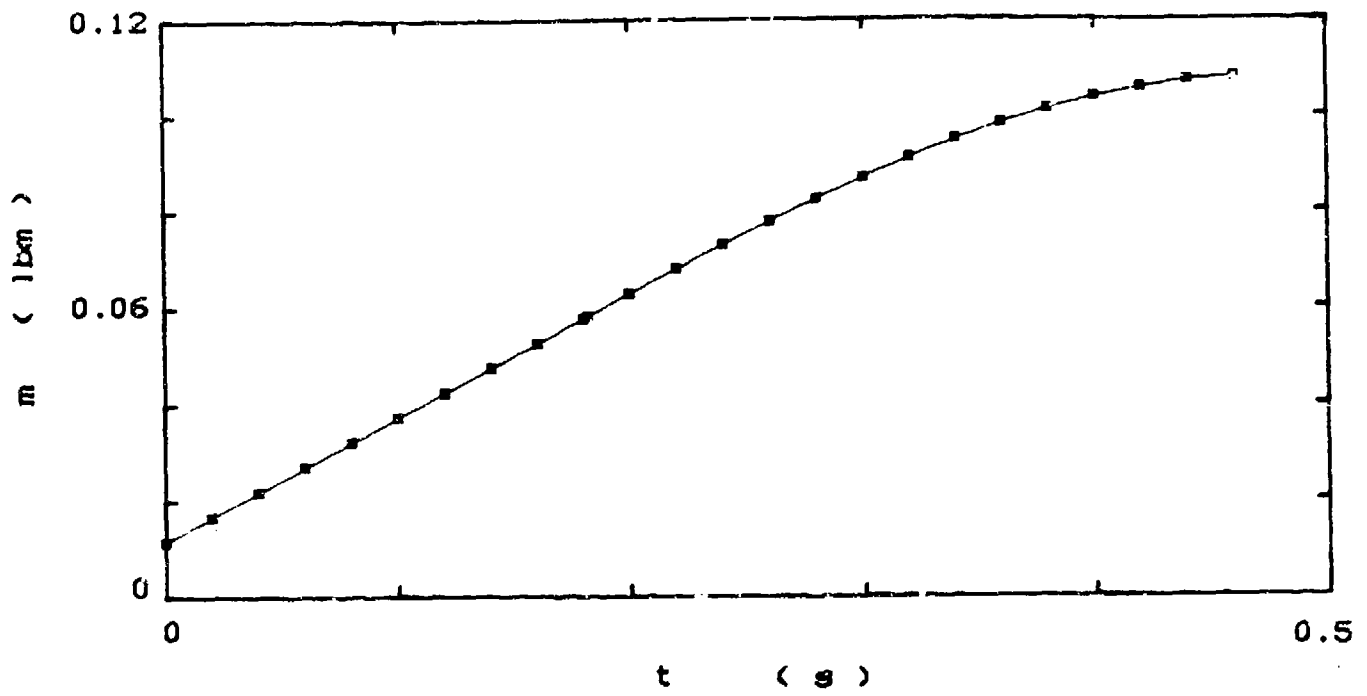


Figure 5-2 Mass of Air Within Tank: 0-0.46 Second

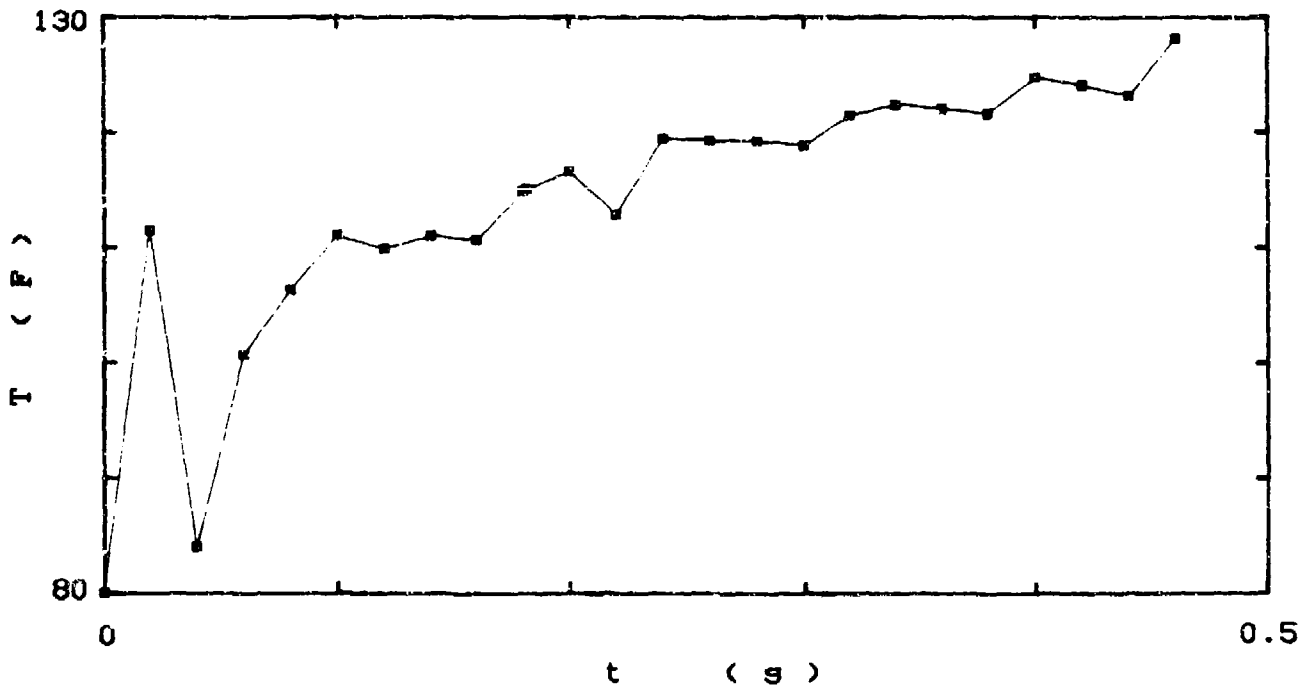


Figure 5-3 Average Air Temperature in Tank: 0-0.46 Second

multiplied by the area of that section of tank it was representing. The results from all the gages were then added together to get the total heating rate for the entire tank. See Appendix B for this information in tabular form. The data was then integrated numerically to determine the heat transferred to the tank walls as a function of time. This data is displayed in Figure 5-4.

At the end of the 4 seconds, the total heat transferred was approximately 80% of the total amount which should have been transferred had equilibrium been reached. That maximum is 3.92 BTU and depends solely on the initial and final tank pressures as well as the volume of the tank. (See Section II: the second classical problem.) As a result, the curve was extrapolated out to 10 seconds to account for the missing energy, but that will be discussed shortly. The heat flux data from the individual gages versus their position in the tank are plotted in Figure 5-5 at specific moments in time. These plots give an indication of the heating distribution along the tank wall.

From the mass, temperature, and heat data, the average internal and kinetic energies of the gas within the tank were calculated. Recall Eq. [2-3],

$$m[c_v T + u^2/2g_c] - m_1 c_v T_0 = (m - m_1) c_p T_0 - Q$$

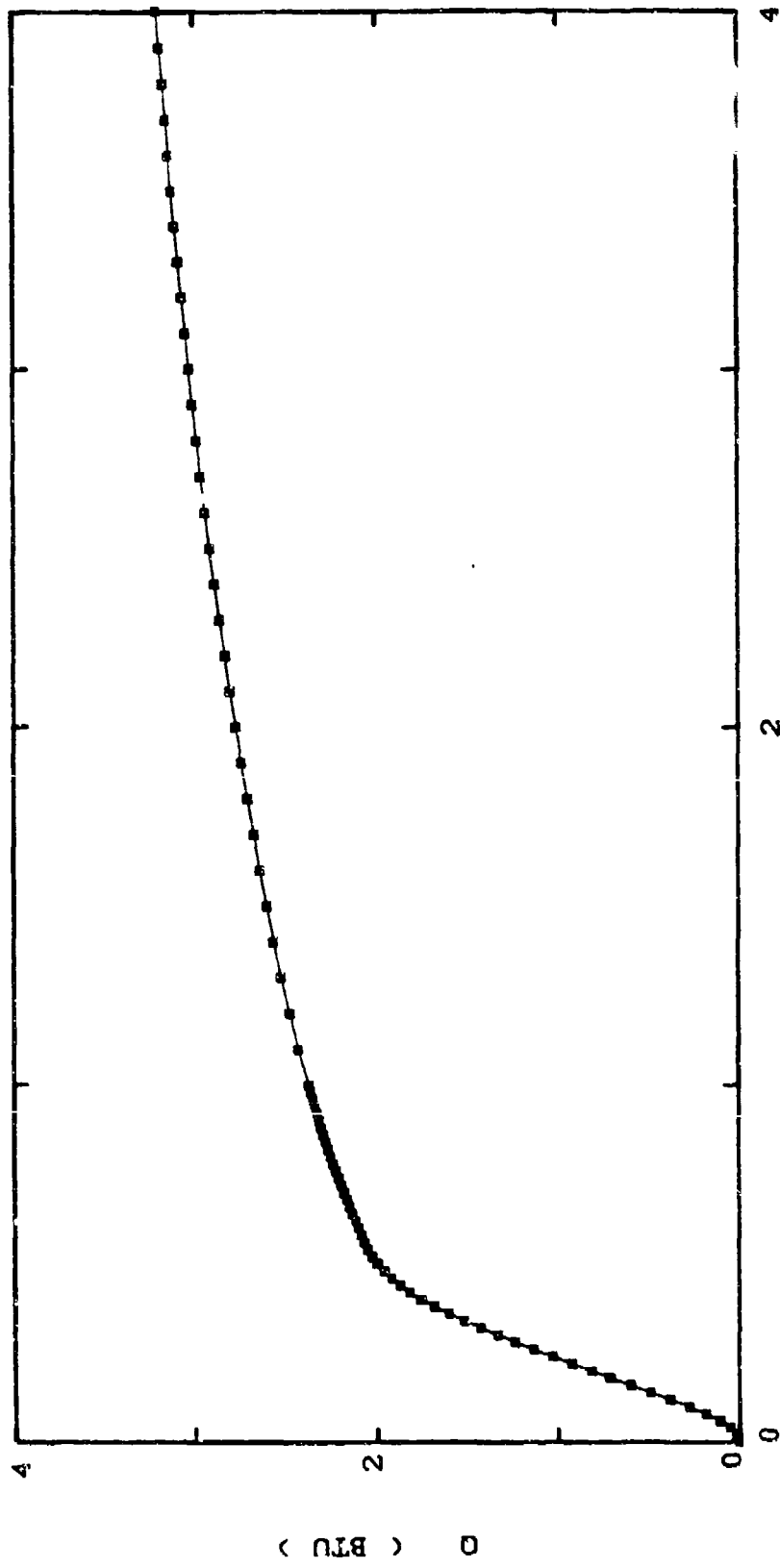


Figure 5-4 Heat from Air to Tank Walls: 0-4 Seconds

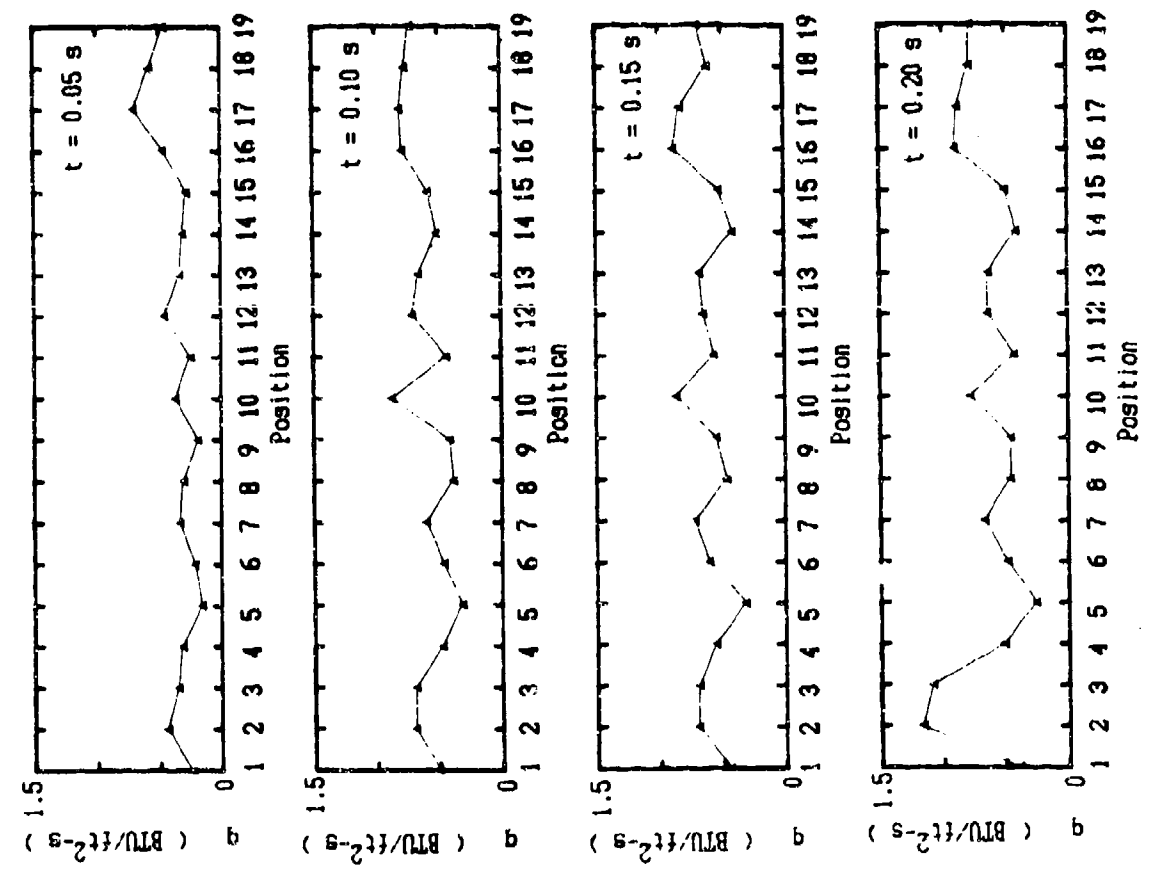
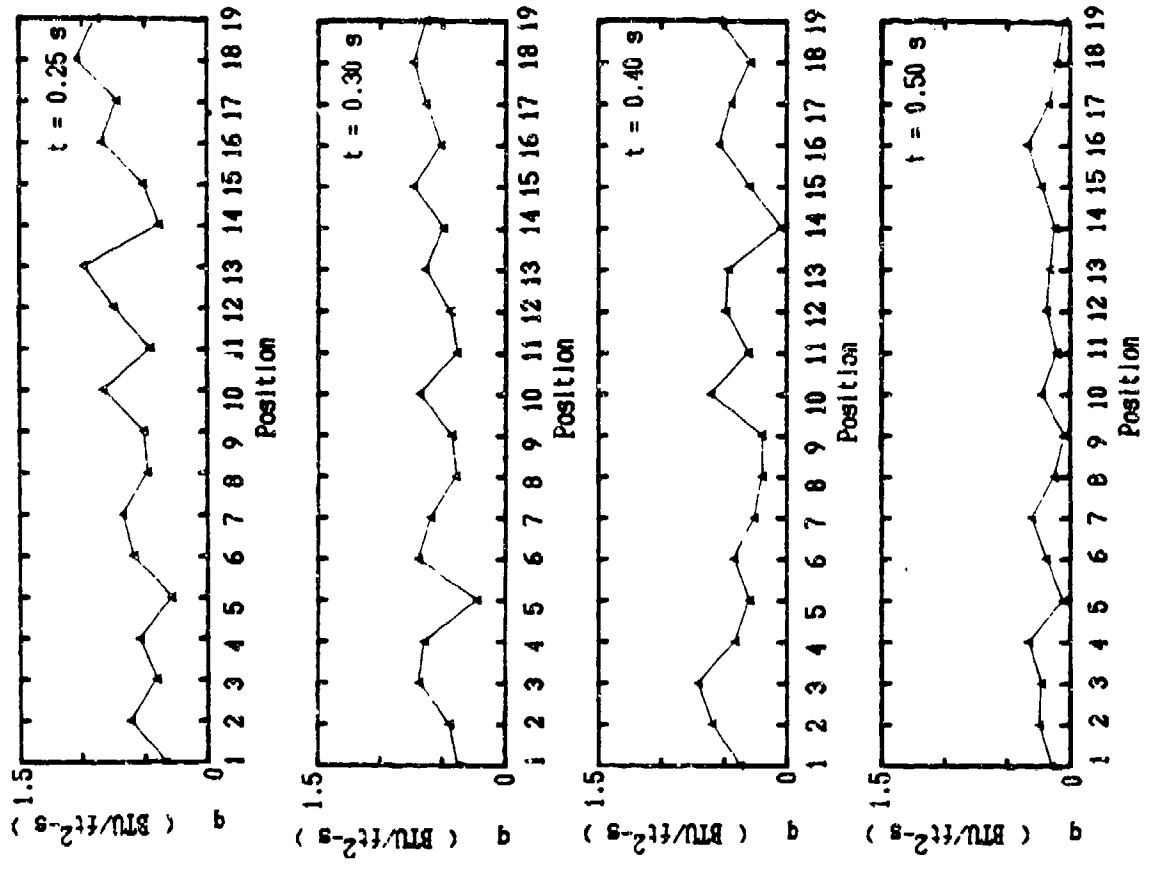


Figure 5-5 Heat Flux Versus Wall Location

Since the only unknown in Eq. [2-3] is the kinetic energy term, $\mu^2/2g_c$, it could be determined. Each of the energy terms appearing in Eq. [2-3] is plotted in Figure 5-6.

Because rapid charging involves high-speed flow, the adiabatic wall minus wall temperature difference was chosen as the driving temperature potential for heat transfer. Due to the unknown dynamics of the flow, a recovery factor of unity was chosen for lack of a better estimate. This reduced the driving potential to be the total temperature minus the wall temperature. The total temperature, T^* , is shown in Figure 5-7.

$$T^*(t) = T(t) + [u(t)]^2/2g_c c_p \quad [5-5]$$

To determine an average convection coefficient for the tank, $h(t)$, the following equation was used.

$$h(t) = q(t) / [T^*(t) - T_o] \quad [5-6]$$

To get $q(t)$, the data for $Q(t)$ was divided by the total surface area of the tank and numerically differentiated. The plot of $h(t)$ is shown in Figure 5-8. Anticipating a relationship between $h(t)$ and Reynolds number, $\log(h)$ was plotted versus the $\log(\rho u)$. The velocity, u , was determined from the kinetic energy data. This logarithmic plot is shown in Figure 5-9.

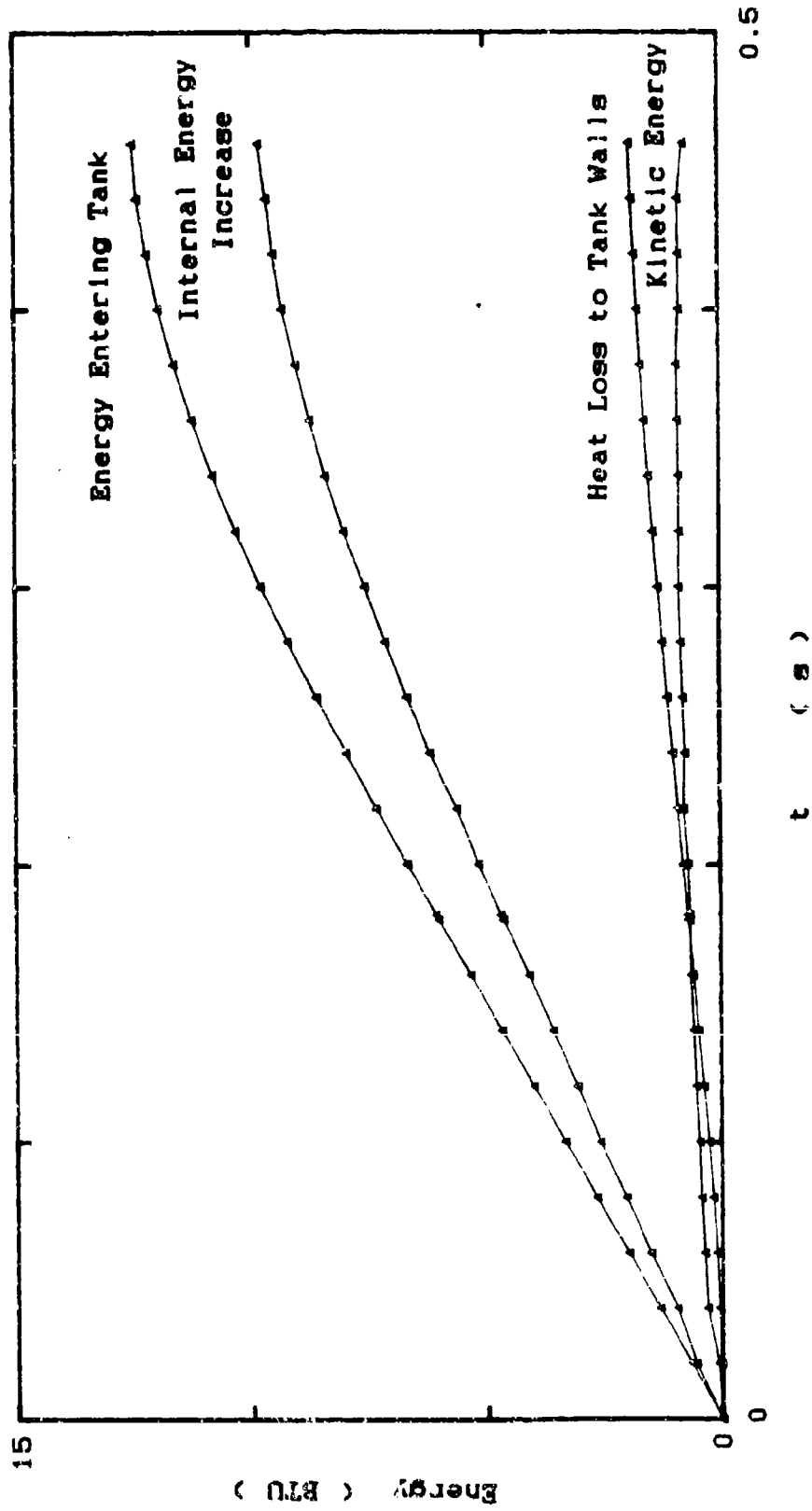


Figure 5-6 Terms in the Energy Balance of Eq. [2-3]:
0-0.46 Second

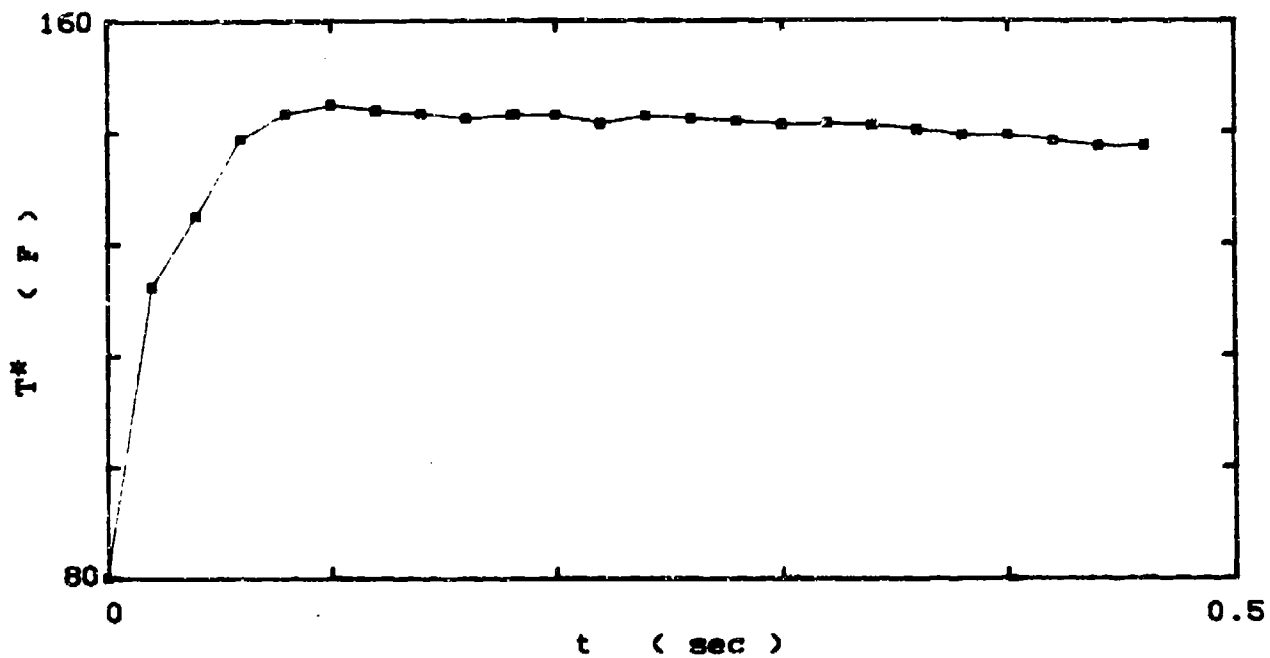


Figure 5-7 Average Total Air Temperature: 0-0.46 Second

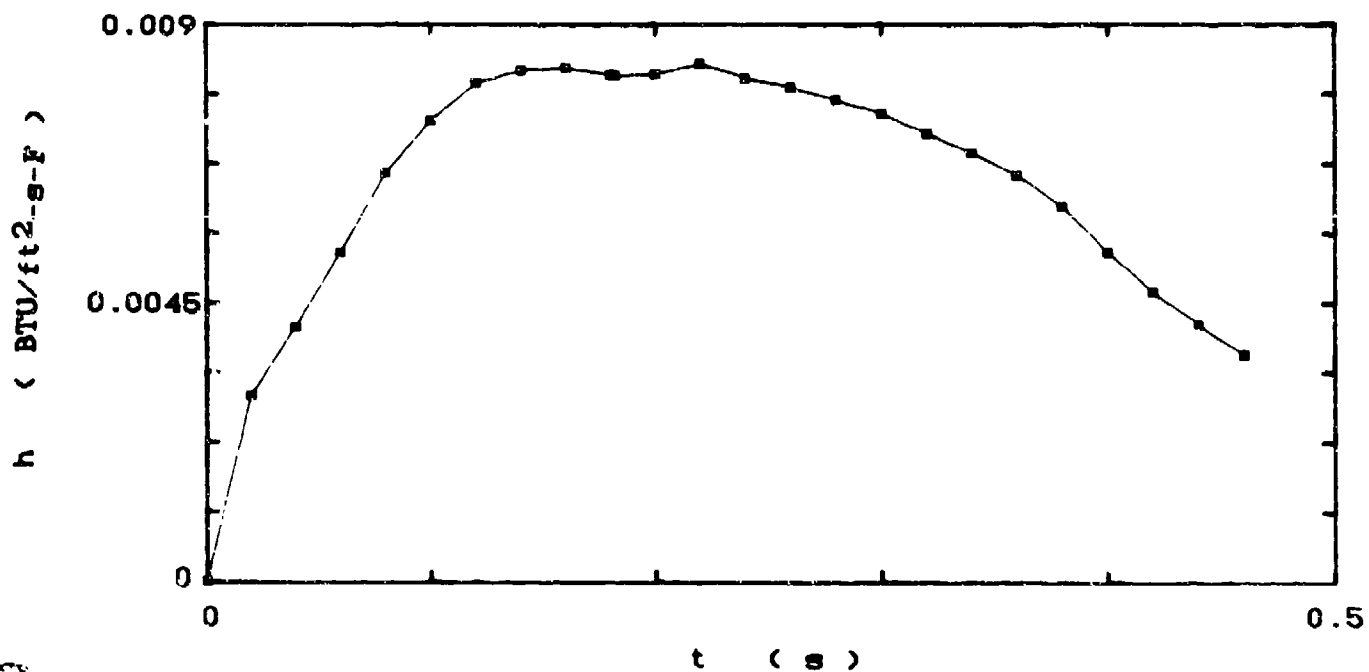


Figure 5-8 Average Convection Coefficient: 0-0.46 Second

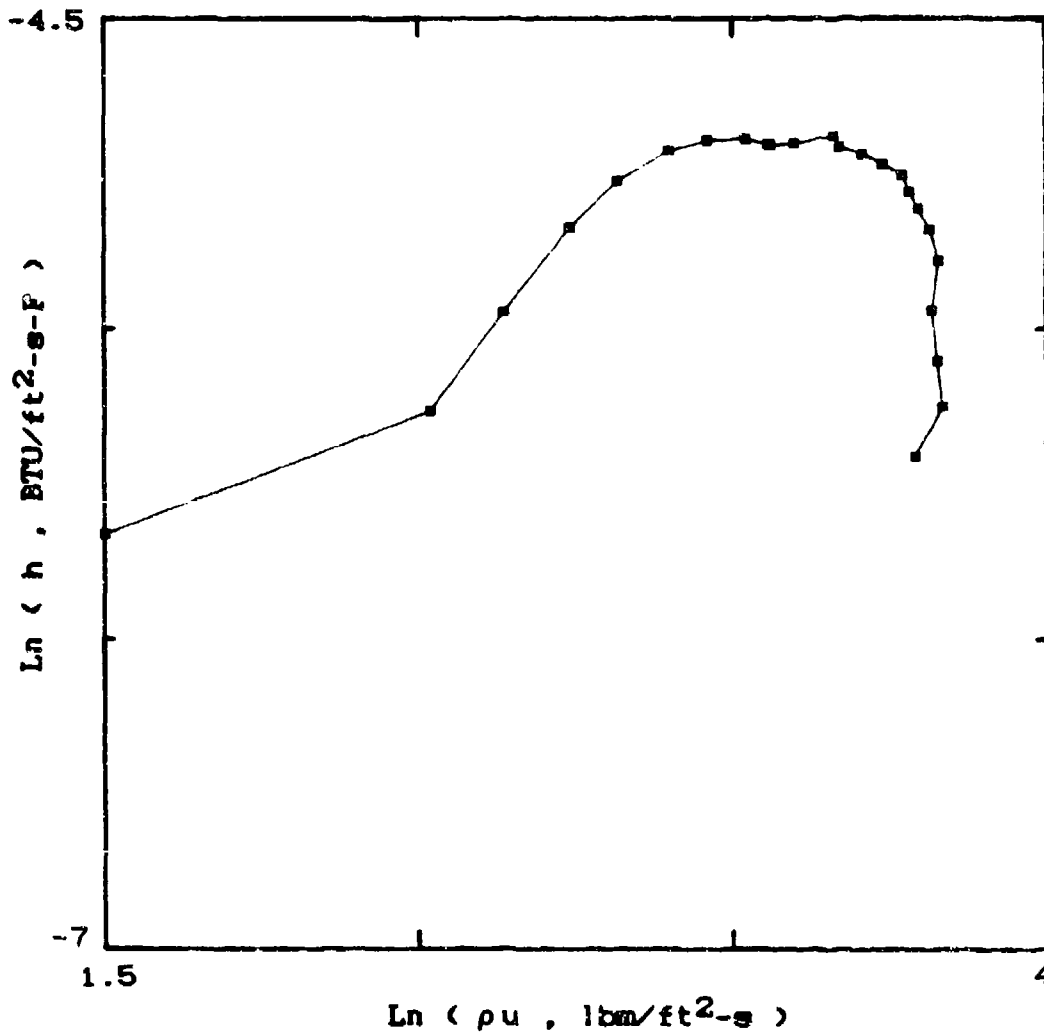


Figure 5-9 $\ln(h)$ Versus $\ln(pu)$: $t=0.46$ Second

Extrapolated Data

Because the heat transferred after 4 seconds was short of the theoretical equilibrium value, an attempt was made to account for this energy by extending the time for equilibrium to be established. After several attempts, 10 seconds was found to be the time at which the heat transfer was completed and at the maximum value. In order to extrapolate all the data out to 10 seconds, three quantities had to be chosen as independent variables. Since the pressure within the tank was known, it was selected as one of the independent variables. The two other quantities finally chosen were the convection coefficient, h , and the velocity, u .

The convection coefficient was chosen as an independent variable because prior trials at extrapolation indicated that $h(t)$ remained fairly constant after 1 second. Therefore, it was assumed that from 1 to 10 seconds $h(t) = 0.0014385$ BTU/ft²-F-s . This modified curve is displayed in Figure 5-10.

The velocity was also chosen as a known quantity because there was some theory with which to predict its behavior. The rate of momentum dissipation of a fluid within a tank depends upon the force on that fluid due to the shear stress at the wall. Since the mass changed very slowly after 0.46 second, it may be approximated as a constant. Doing so, the momentum

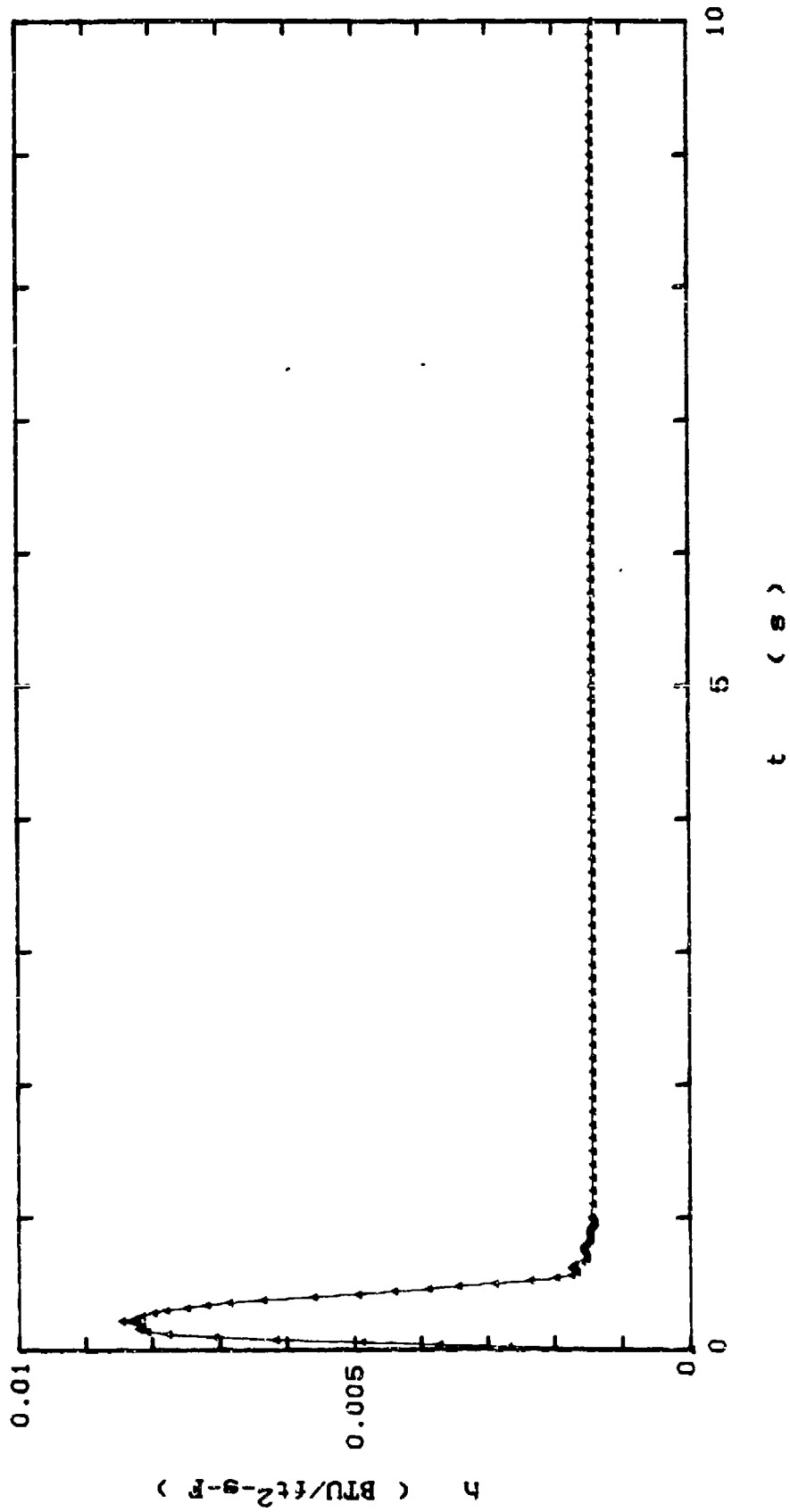


Figure 5-10 Modified Average Convection Coefficient:
0-10 Seconds

dissipation may be expressed as

$$d(\mu u(t))/dt = -\tau_w A_{\text{surface}} = -\mu u(t) A_{\text{surface}}/L \quad [5-7]$$

After integration,

$$u(t) = u(0.46) \exp(-(\mu A_{\text{surface}}/mL)(t-0.46)) \quad [5-8]$$

Where τ_w , wall shear stress, has been approximated by $\mu u(t)/L$, and L is some characteristic length. From Eq. [5-8], the velocity should decrease exponentially. Because the velocity prior to 0.46 second was calculated from measured quantities, a simple exponential decay was fit to the velocity curve from 0.46 to 10 seconds. The time constant, $1/(\mu A_{\text{surface}}/mL)$, was taken to be 0.5 sec in order to have $u(10)$ approximately equal to zero. This curve is shown in Figure 5-11.

To find the mass in the tank as a function of time, the energy equation, Eq. [2-3], can be rewritten in terms of P , h , m , and u .

$$c_v PV/R + mu^2/2g_c - m_1 c_v T_0 = mc_p T_0 - hA(PV/mR + u^2/2g_c c_p - T_0) \quad [5-9]$$

Differentiating with respect to time and solving for m yields

$$\dot{m} = \frac{(m\dot{u}/g_c) + d[hA(PV/mR + u^2/2g_c c_p - T_0)]/dt}{c_p T_0 - u^2/2g_c} \quad [5-10]$$

Equation [5-10] was numerically integrated and the resulting mass function is plotted in Figure 5-12. With the mass in

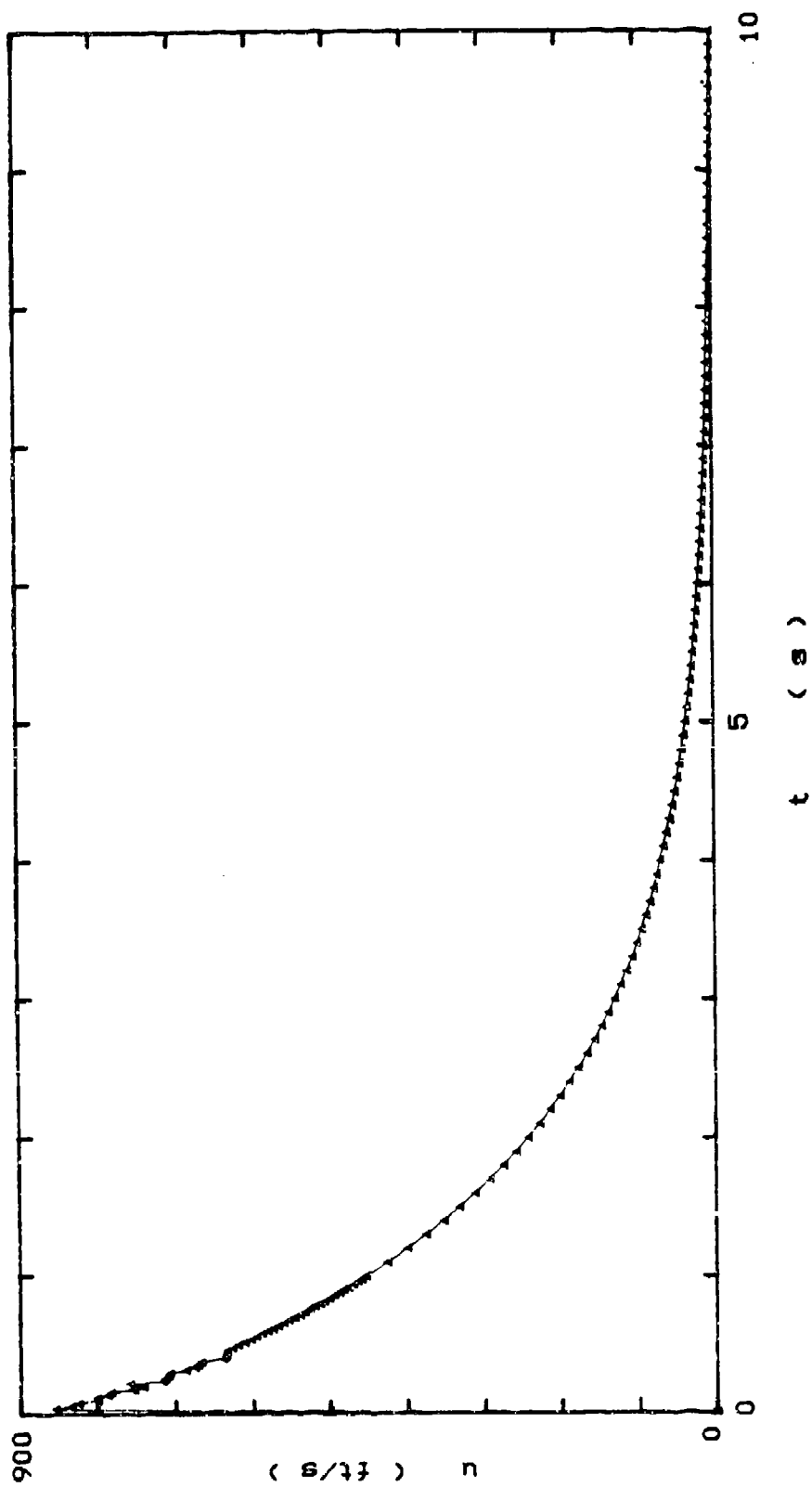


Figure 5-11 Modified Average Velocity: 0-10 Seconds

the tank now estimated, the average and total temperatures were found using Eqs. [5-4] and [5-5], respectively. They are shown in Figure 5-13. From this resulting data, the energy terms in Eq. [2-3] were calculated. They are displayed in Figure 5-14. Also, the plot of $\log(h)$ versus $\log(\rho u)$ was found and is shown in Figure 5-15.

To verify that the second law of thermodynamics was not violated as a result of the modified data, each of the three terms of Eq. [2-4] was calculated from the estimated data. As a convenience, entropy per unit mass was taken to be zero at atmospheric conditions. This eliminated the need to calculate the ms_0 term.

The tank wall temperature was taken to be T_0 since the wall had such a large heat capacity and the heat flux was so small. In fact, for a constant heat flux applied to the boundary of a semi-infinite solid, the ratio of temperature change for a stainless steel solid (tank wall) to that of a quartz solid (thin-film gage) is given by

$$\Delta T_{\text{tank}} / \Delta T_{\text{gage}} = (\rho ck)_{\text{gage}} / (\rho ck)_{\text{tank}} \quad [5-11]$$

For the material properties involved, this ratio is approximately 0.2. Since the maximum gage temperature differences were between 3 and 5 F, the maximum temperature change of the tank wall would be about 1 F. Thus, the wall temperature was assumed to remain constant at T_0 .

To calculate $\dot{S}_{c.v.}$, Eq. [2-5] was used.

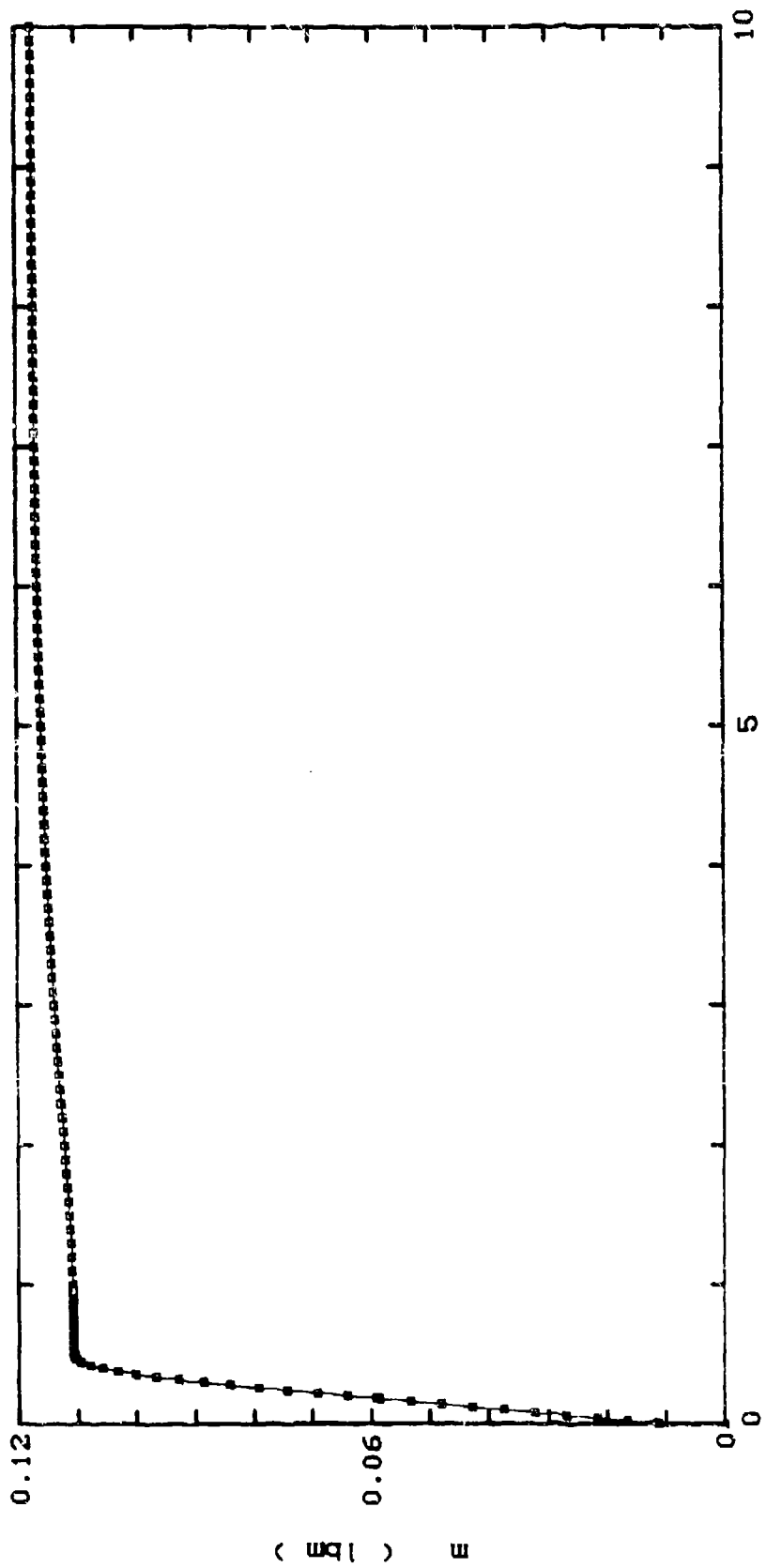


Figure 5-12 Extrapolated Mass in Tank: 0-10 Seconds

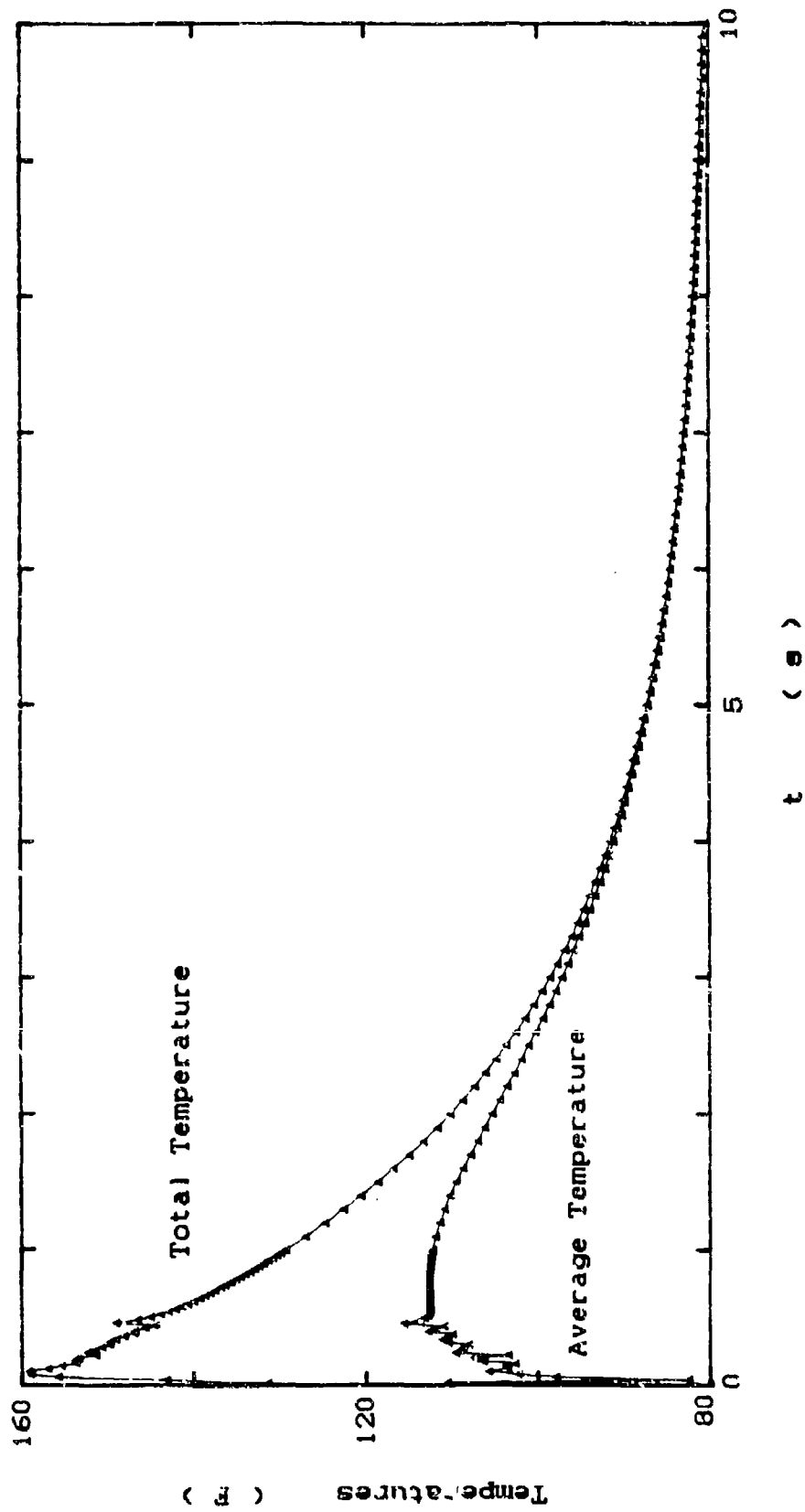
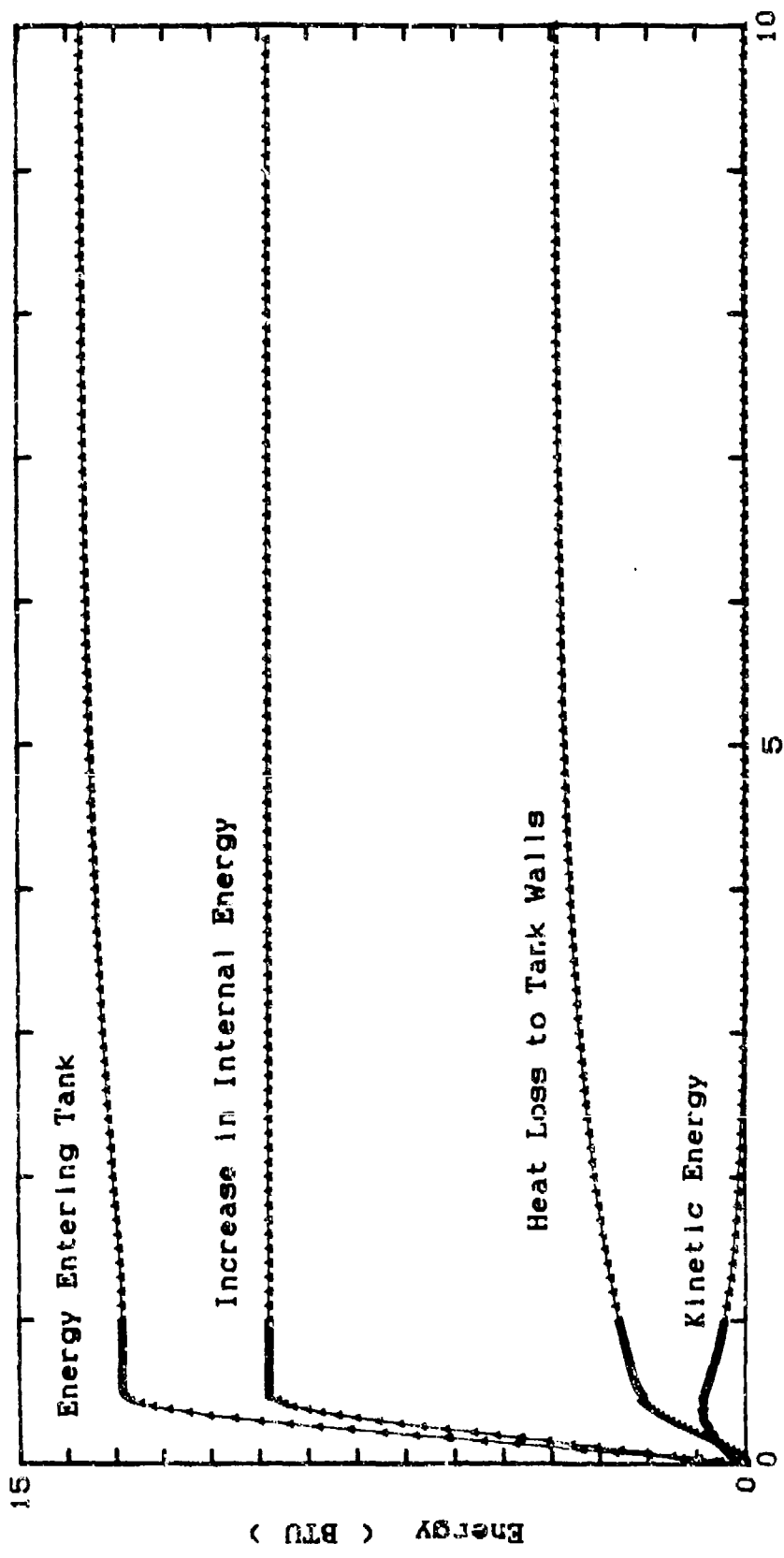
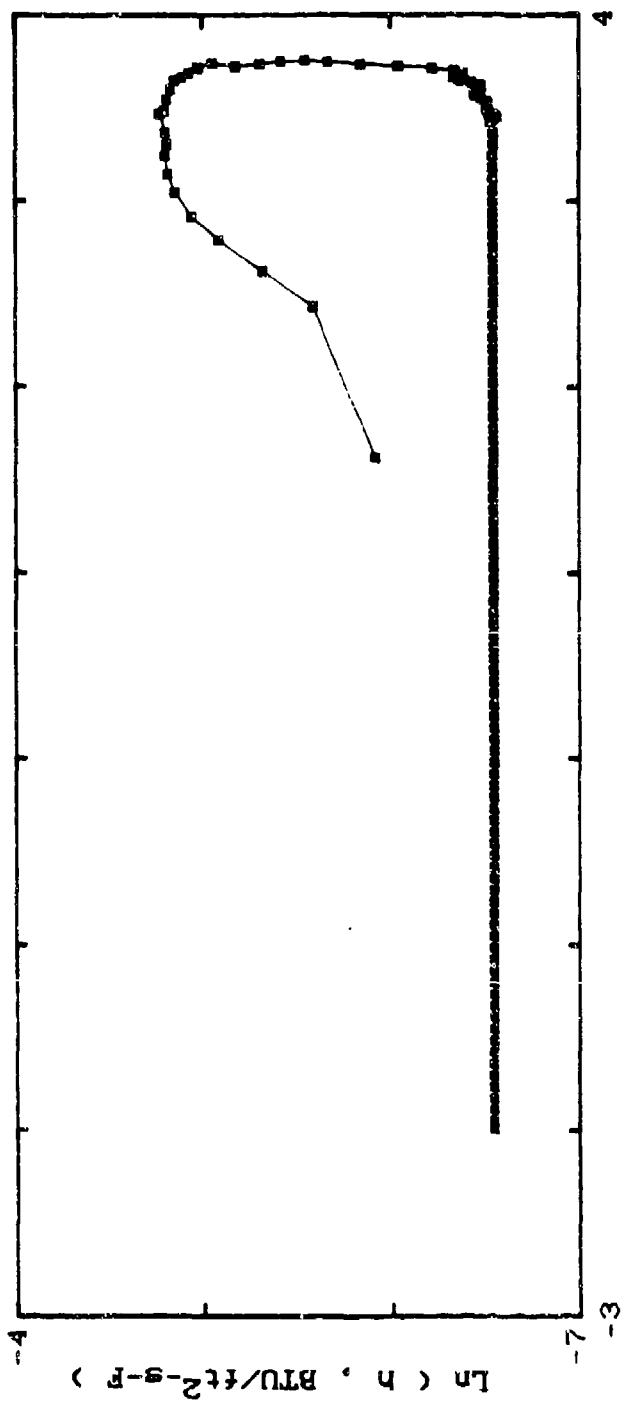


Figure 5-13 Extrapolated Average and Total Temperatures:
0-10 Seconds



t (s)

Figure 5-14 Extrapolated arms in the Energy Balance of Eq. 12-31: 0-10 Seconds



$\ln(pu, \text{lbm}/\text{ft}^2\text{-s})$

Figure 5-15 $\ln(h)$ Versus $\ln(pu)$: 0-10 Seconds

$$s(t) = c_p \ln[T/T_0] - R \ln[P/P_0] \quad [5-12]$$

Thus,

$$S_{c.v.}(t) = m(t)s(t)$$

or,

[5-13]

$$\dot{S}_{c.v.}(t) = d(m(t)s(t))/dt$$

The results of these calculations are shown in Figure 5-16.

Recall that $s_0 = 0$. Equation [2-4] then requires that

$$\dot{S}_{c.v.}(t) + \dot{Q}/T_0 \geq 0 \quad [5-14]$$

Figure 5-17 shows the sum of these entropy terms, and indeed, the sum is initially greater than and quickly approaches zero.

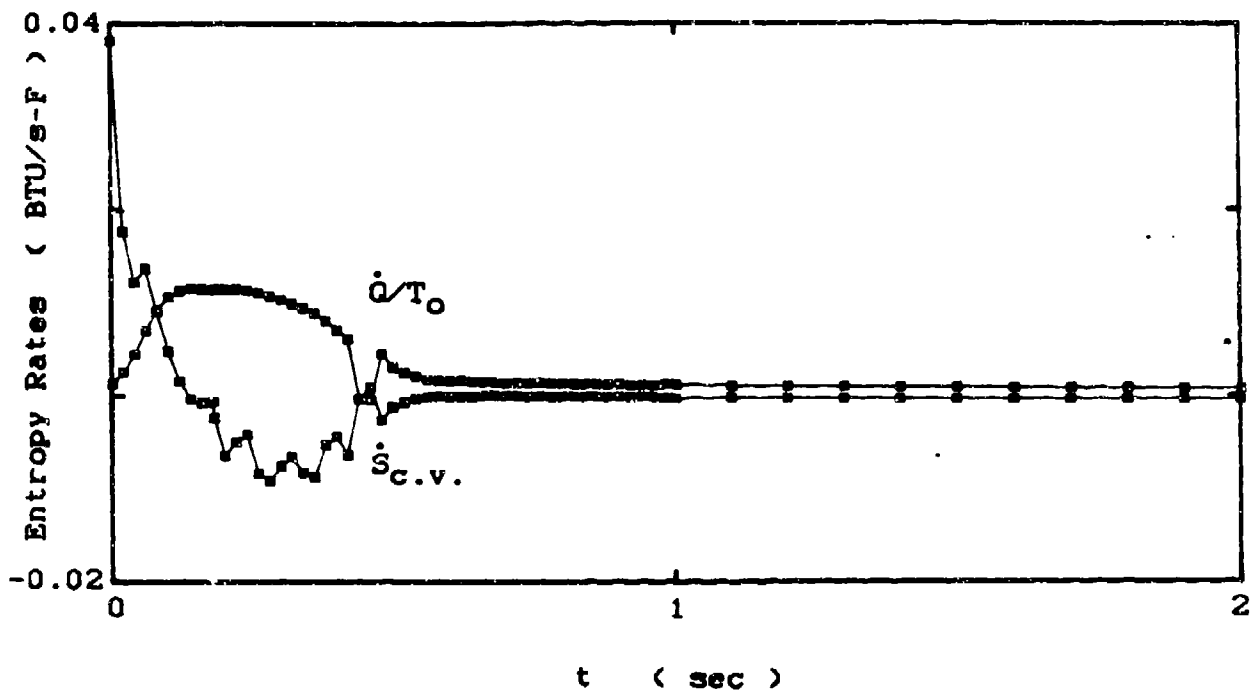


Figure 5-16 Entropy Entering, Leaving, and Stored in Tank: 0-2 Seconds

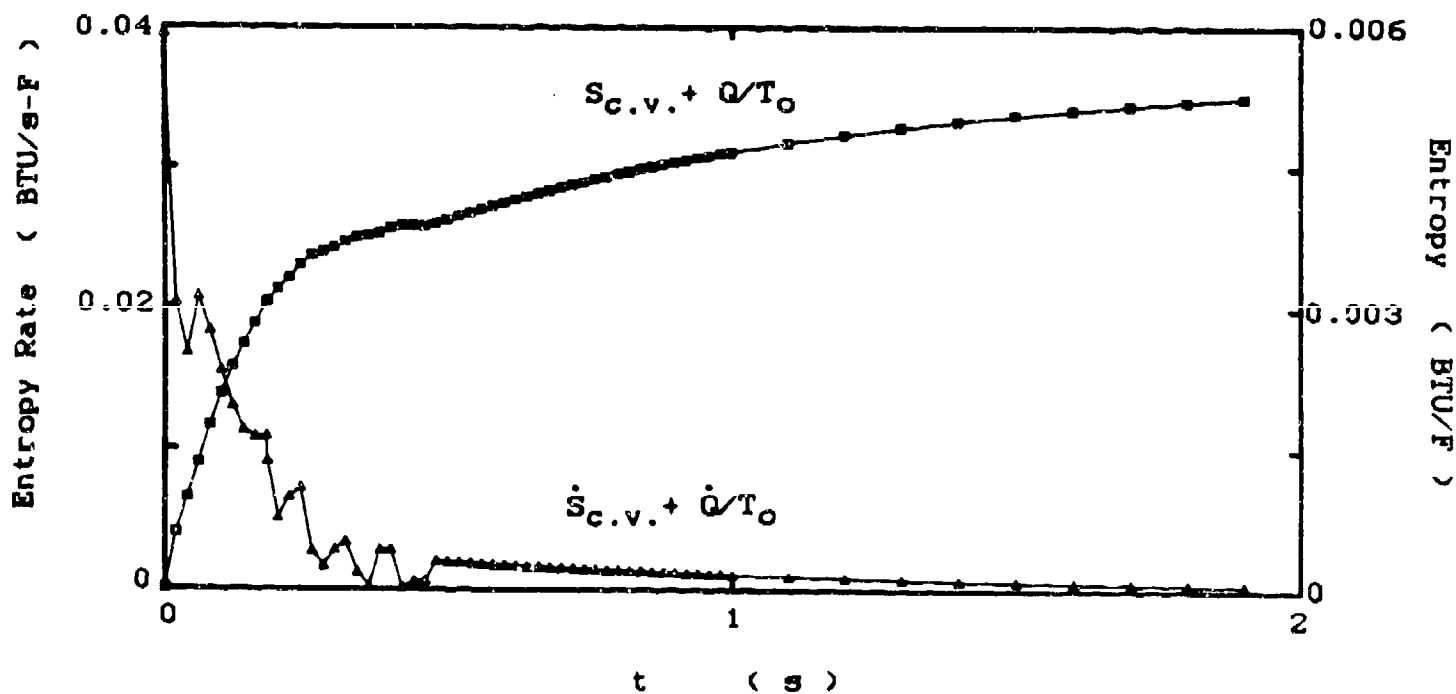


Figure 5-17 Sum of Entropy Terms from Fig. 5-16: 0-2 Seconds

VI. Results

As shown in Figure 5-5, the heat flux level along the tank wall at any given time is fairly uniform. Deviations do exist, however, and may imply recirculation regions within the tank. With the exception of the positions on the bottom of the tank, the crests in heat flux remain crests and the troughs remain troughs. What may be happening in the tank is this: the instant the tank is opened, air flowing out of the choked nozzle expands immediately into the upper corners of the tank. The flow then continues down the tank and finally stagnates at the base. As the tank fills, the flow out of the nozzle becomes more jet-like and swirling begins. This swirling seems to reach a peak around the time at which the nozzle unchokes. After that time, the velocity of the entering air decreases as does the kinetic energy of the air within the tank. The flow pattern during the time of maximum swirling may appear as that shown in Figure 6-1.

From all the figures in Section V, it's clear that the bulk of the charging process is over in a very short time. The pressure within the tank rose to atmospheric pressure in 0.46 second. The mass nearly rose to its maximum value in 0.46 second. From that time on, mass slowly entered the tank as the average gas temperature within gradually decreased. After 0.46 second, slightly over 50% of the heat energy had already been transferred from the gas to the tank walls. By

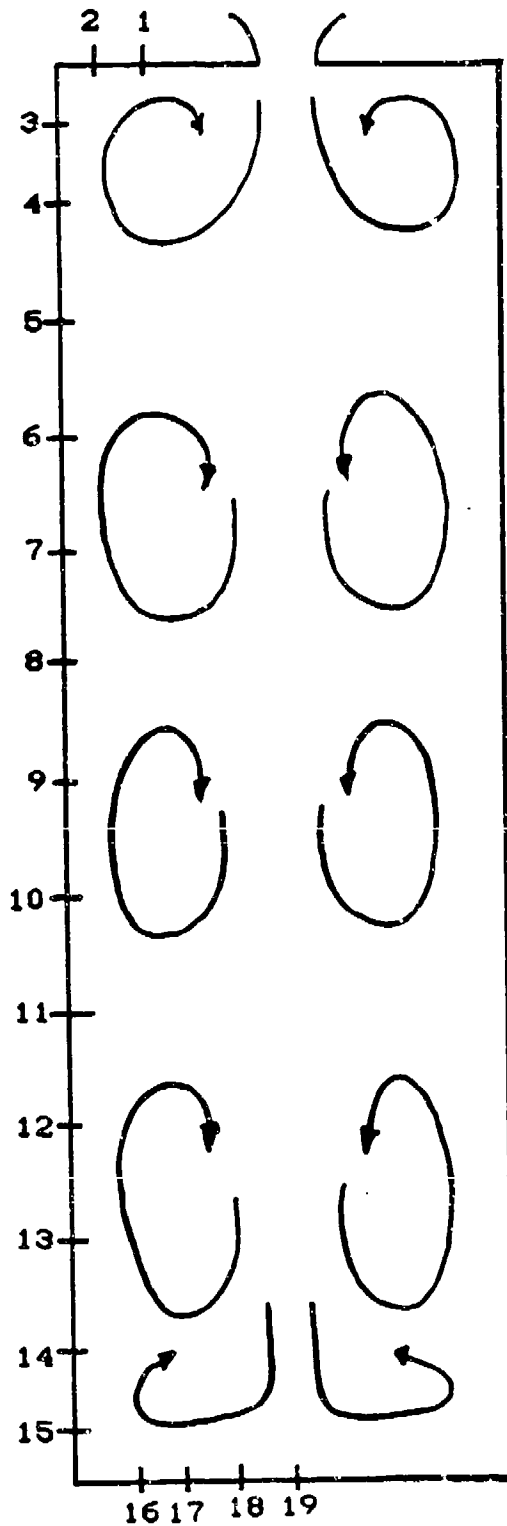


Figure 6-1 Possible Flow Pattern During Charging

four seconds, 80% had been transferred; by 8 seconds, 95%; and by 10 seconds, almost 100%.

A convection coefficient was derived and plotted in Figure 5-10. This $h(t)$ was also plotted on a log scale and is shown in Figure 5-15. The plot contains the same strange behavior as the plot in Fig. 5-9. This data indicates a complex flow pattern within the tank with two, possibly three, different regimes. During the early times of the process, h is proportional to $(\rho u)^{1.114}$. A transition then occurs and h seems to be independent of (ρu) . The third flow pattern is the straight line in the plot and is a result of choosing $h(t)$ to be constant after 1 second. Little can be said concerning the meaning of that portion of the curve. Because of this strange behavior, it was not possible to establish a relationship between $h(t)$ and Reynolds number.

This erratic behavior of the convection coefficient also precluded its use as a tool for predicting heating rates in other tanks. Another approach was needed. To this end, the heating rate for the entire tank, $\dot{Q}(t)$, was found from differentiating $Q(t)$ and is displayed in Figure 6-2.

$\dot{Q}(t)$ depends upon the initial pressure conditions, the volume of the tank, and the size of the inlet. Also, it will depend upon the time during which the nozzle is choked and the time for pressure within the tank to equalize with the ambient pressure. These factors were considered in extending the measured results to other tanks with conducting walls

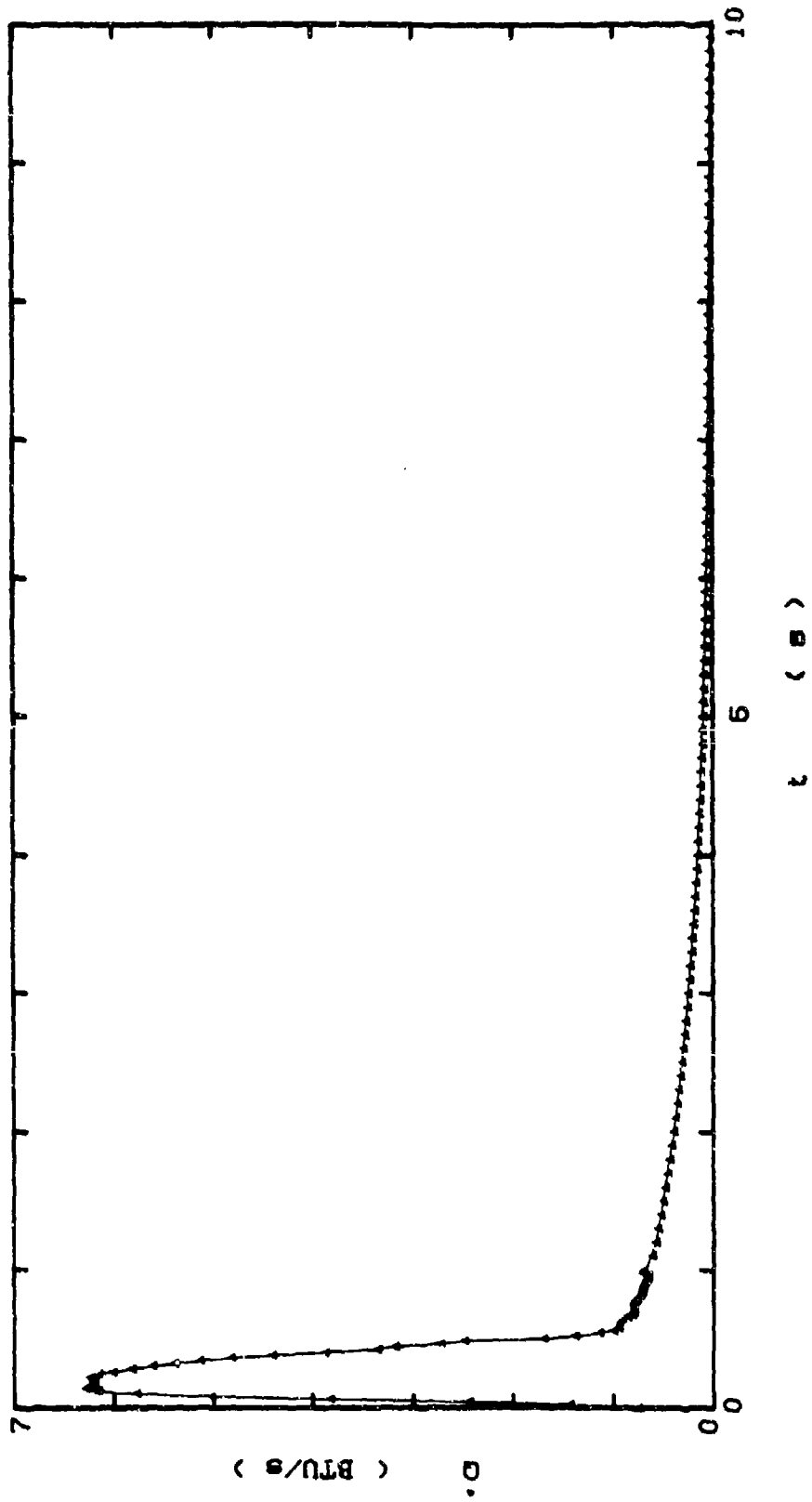


Figure 6-2: Heating Rate of Tank Wall: 0-10 Seconds

which have roughly the same thermal properties as stainless steel. Those properties are $\alpha = 0.147 \text{ ft}^2/\text{hr}$ and $\rho c k = 435 \text{ BTU}^2/\text{ft}^4\text{-hr-F}^2$. It should be noted here that the following calculations (with the exception of Q_{tot}) are only estimates.

As was discussed previously, the total heat transferred from the gas to the tank walls is just the total flow work done on the gas. This energy, Q_{tot} , equals the difference in initial pressures multiplied by the volume of the tank. This relationship is plotted in Figure 6-3 for various heat levels.

To find the time at which the nozzle unchokes as well as the time at which the pressures equalize, the measured pressure curve was represented by two straight lines. The first line describes the pressure rise while the nozzle is choked. Since this relationship is a linear one, no approximation needs to be made. The second line represents the pressure rise between the time from the unchoking of the nozzle to the time the tank pressure reaches P_0 . This is an approximation to the actual curve. See Figure 6-4. The slopes of these lines were calculated from the measured data, and they yielded the following results.

$$t_{\text{uc}} = (0.52828P_0 - P_1) / 4952.25$$

[6-1]

$$t_p = t_{\text{uc}} + (1 - 0.52828) P_0 / 3481.71$$

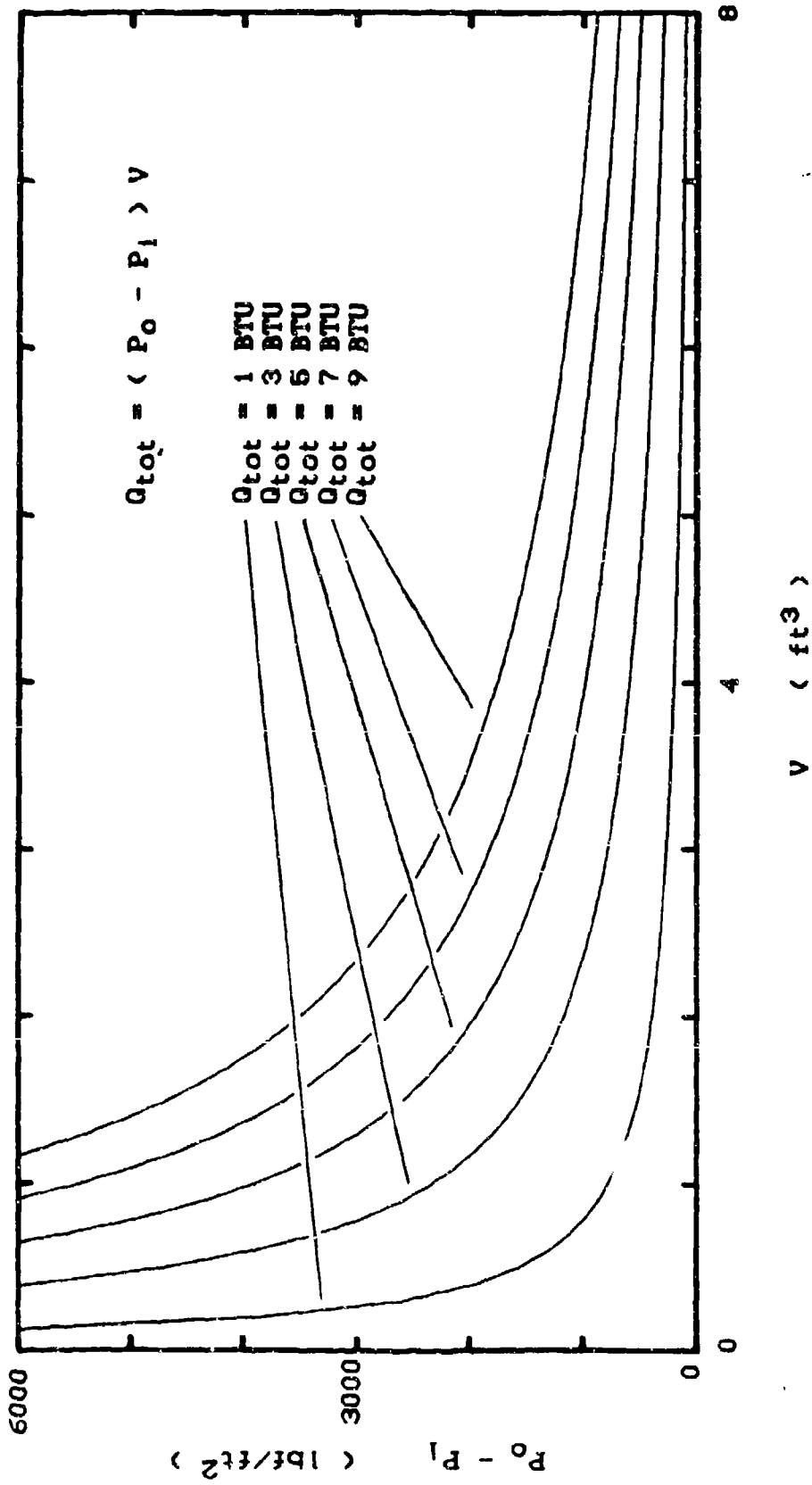


Figure 6-3 Relationship Between Pressure Difference, Tank Volume, and Total Heat Transferred

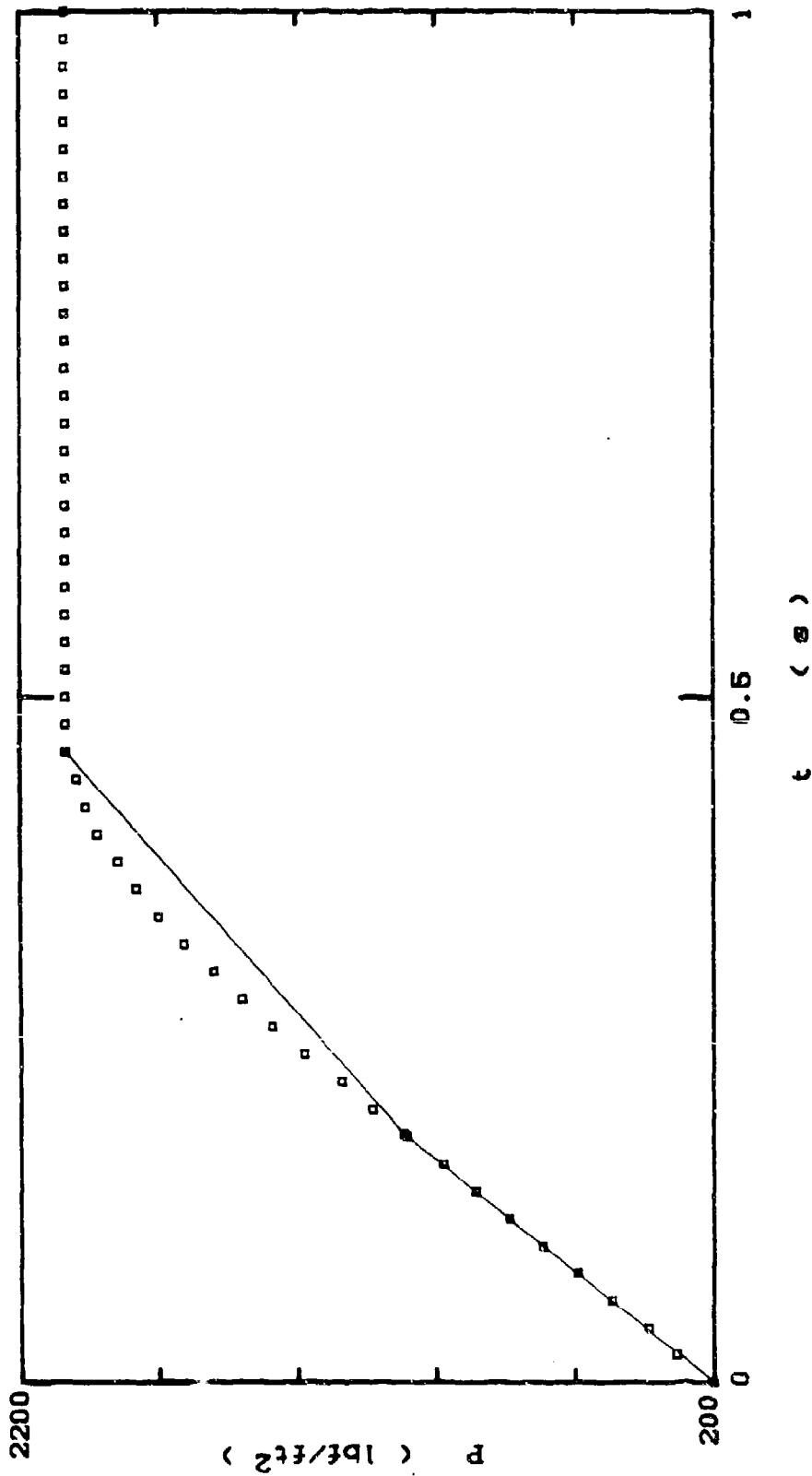


Figure 6-4 Straight Line Approximation to Pressure Curve
Used in Finding t_{uc} and t_p

The pressures are measured in lbf/ft², and the times are in seconds. These two equations are only valid for the nozzle and tank that was used in this study. Since the time to fill a tank ought to be proportional to the volume of the tank, Eq. [6-1] was multiplied by the ratio of new tank volume, V, to the tank volume used in this study. Also, a nozzle with a large throat area will fill a tank faster than one with a small throat area. Thus, Eq. [6-1] was divided by the ratio of new throat area, A_{th}, to the area of the throat that was used. These equations are

$$t_{uc} = 0.003362 (V/A_{th})(0.52828P_0 - P_1) / 4952.25 \quad [6-2]$$

$$t_p = t_{uc} + 0.001585 (VP_0/A_{th}) / 3481.71$$

The tank volume is measured in ft³ and the throat area in ft². Equation [6-2] can now be used to estimate the time it takes for a given nozzle to unchoke as well as the time required for the gas in the tank to reach equilibrium pressure.

From the data in Figure 5-14, the the heat transferred by t_p is approximately 50% of Q_{tot} . Also, from Figure 6-2, $\dot{Q}(t)$ can be accurately approximated by a second order polynomial between $t=0$ and $t=t_p$.

$$\dot{Q}(t) = \dot{Q}_{max} [at^2 + bt + c] \quad [6-3]$$

The following equation can then be used to find \dot{Q}_{\max} .

$$0.5 \dot{Q}_{\text{tot}} = \dot{Q}_{\max} \int [at^2 + bt + c] dt \quad [6-4]$$

The three conditions needed to find the coefficients are

$$\dot{Q}_{\max} = \dot{Q}(t_{uc})$$

$$\dot{Q}(0) = 0$$

$$\dot{Q}(t_p) = 0.18 \dot{Q}_{\max}$$

Therefore,

$$\dot{Q}(t) = \dot{Q}_{\max} t [8.875 - 18.444 t] \quad [6-5]$$

After substituting into Eq. [6-4] and integrating,

$$\dot{Q}_{\max} = 0.5 \dot{Q}_{\text{tot}} / [t_p^2(4.4375 - 6.148t_p)] \quad [6-6]$$

Thus, by knowing only the volume of a tank, the area of the nozzle throat, the ambient pressure, and the initial pressure within the tank, the total heat transferred to the tank walls can be calculated. Also, the times to unchoking and pressure equalization can be estimated as can the peak heat flux and the shape of the $\dot{Q}(t)$ curve. The remainder of the $\dot{Q}(t)$ curve can also be estimated after t_p by assuming another second order polynomial. This part of the curve represents long times with small heating rates. Because of this, the first part of the curve is most important and was all that was found. However, the total time for charging to

be completed can be estimated by

$$t_f = 22 t_p$$

[6-6]

This equation was found by taking the measured ratio of the total time for charging, 10 seconds, to the time at which the tank pressure equaled P_0 , 0.46 second.

VII. Summary

This study was a new look into the old problem of charging a vessel with an ideal gas. The actual transient process was observed and calculated. A major surprise was the uniformity of heat flux along the walls of the tank. There were no significantly hot or cold spots at any point on the tank wall. Another surprise was the strange behavior of the convection coefficient with respect to (ρu) . As a result, it was difficult to make any conclusive statements as to the actual flow dynamics. It was also difficult to formulate a method of using $h(t)$ for the prediction of heat transfer rates in other tanks. However, a method of estimating peak heating rates as well as the general shape of the $\dot{Q}(t)$ data was found.

VIII. Recommendations

Further study of the charging process is definitely warranted. This study, being the first of it's kind, has only scratched the surface of the complexities involved in the flow dynamics and heat transfer during the charging of an evacuated tank. Further experimentation should involve the recording of heat transfer data throughout the entire process. Also, testing with different nozzle sizes and various pressure conditions should be carried out to verify or disprove the estimation techniques developed here. It would also be of interest to vary the length to diameter ratio of the tank in order to determine effects resulting from geometry. All these would aid in developing a means of accurately predicting heat flux during the charging process.

Appendix A

Model Numbers of Equipment Used

Endevco 0-100 psia Pressure Transducer Model # 8530A-100

TSI Miniature Flush Mount

Sensor Elements 1268 Platinum Hot Film

Probes used: 1, 2A, 3, 4, 8

Datalab dl1200 Waveform Recorder Serial # 119

Appendix B

Table of Heats Measured by Individual
Thin-Film Gages

* LOCATION ON THE TANK WALL	TRIAL NUMBER FOR GAGE	HEAT LOSS PER AREA BTU/FT ²	WALL AREA REPRESENTED IN ²	TOTAL HEAT LOST BTU
1	**	0.2655275	28.274	0.0521356
2	E2	0.4253386	47.124	0.1391921
3	E3	0.4508400	56.941	0.1782728
4	D3	0.3764253	78.540	0.2053086
5	D4	0.2168532	94.248	0.1419304
6	D5	0.3276899	94.248	0.2144730
7	C3	0.4223368	94.248	0.2764194
8	C4	0.2916098	94.248	0.1908586
9	E4	0.3129244	94.248	0.2048090
10	B3	0.4977912	94.248	0.3258043
11	B4	0.3131344	94.248	0.2049465
12	B5	0.4033936	94.248	0.2640211
13	F3	0.4254639	94.248	0.2784661
14	F4	0.2767622	78.540	0.1509507
15	F5	0.3774505	62.832	0.1646942
16	F2	0.3742415	34.361	0.0893008
17	B2	0.3950619	28.274	0.0775693
18	C2	0.3618079	14.137	0.0355200
19	D2	0.3845312	1.767	0.0047185
		TOTALS	1279.022	3.1993912

* For locations, see Fig. 6-1, page 50.

**	B1	0.2407778	
	C1	0.2777149	
	D1	0.2814635	Gage left in same spot on
	E1	0.2444963	tank for 5 runs to verify
	F1	0.2831850	experiment repeatability.
AVERAGE		0.2655275	

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