ALSEP CASK COOLING
FEASIBILITY STUDY

Prepared by B. Nordquist

Approved by L. McNaughton
1.0 INTRODUCTION

1.1 Background

1.2 Review of Cask Cooling Requirements

1.3 Study Objectives

1.4 Description of Selected Concepts

2.0 TEST RESULTS

2.1 Discussion of Data

2.2 Recommendations

2.3 Test Setup

2.4 Summary of Test Data

3.0 THEORETICAL AND EXPERIMENTAL ANALYSIS

3.1 Background

3.2 Basic Open Flow Analysis

3.3 Correlation of Open Flow to Theory

3.4 Shroud Analysis

4.0 THERMAL CORRELATION

5.0 DISCUSSION OF PRE-TEST ANALYTICAL STUDIES

5.1 Axial Flow Air Cooling Systems

5.1.1 Attached Nozzle - Open Flow

5.1.2 Detached Nozzle - Open Flow

5.1.3 Attached Nozzle - Shrouded Flow

5.2 Cross Flow Cooling Systems

5.3 Normal Flow Cooling Systems

5.4 Water Cooling Systems

5.4.1 Closed System

5.4.2 Open System

5.5 Cask Shroud Removal Investigation
### LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure No.</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>Early Cooling Concepts</td>
<td>10</td>
</tr>
<tr>
<td>1.2</td>
<td>Schematic of Test Nozzle Configurations</td>
<td>11</td>
</tr>
<tr>
<td>2.1 a &amp; b</td>
<td>Test Setup for Cask Cooling Series</td>
<td>13</td>
</tr>
<tr>
<td>2.2</td>
<td>Open Side Cask Surface Temperatures</td>
<td>16</td>
</tr>
<tr>
<td></td>
<td>Axial flow, $m = 20$ lb/min</td>
<td></td>
</tr>
<tr>
<td>2.3</td>
<td>Open Side Cask Surface Temperature Rise</td>
<td>17</td>
</tr>
<tr>
<td></td>
<td>Axial Flow, $m = 35$ lb/min</td>
<td></td>
</tr>
<tr>
<td>2.4</td>
<td>Open Side Cask Surface Temperature Rise</td>
<td>18</td>
</tr>
<tr>
<td></td>
<td>Axial flow, $m = 50$ lb/min</td>
<td></td>
</tr>
<tr>
<td>2.5</td>
<td>Open Side Cask Surface Temperature Rise</td>
<td>19</td>
</tr>
<tr>
<td></td>
<td>Conical Nozzles, $m = 20$ lb/min</td>
<td></td>
</tr>
<tr>
<td>2.6</td>
<td>Open Side Cask Surface Temperature Rise</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Conical Nozzles, $m = 35$ lb/min</td>
<td></td>
</tr>
<tr>
<td>2.7</td>
<td>Open Side Cask Surface Temperature Rise</td>
<td>21</td>
</tr>
<tr>
<td></td>
<td>Conical Nozzles, $m = 50$ lb/min</td>
<td></td>
</tr>
<tr>
<td>3.1</td>
<td>Jet Velocity for Attached Nozzle (Config. 1)</td>
<td>32</td>
</tr>
<tr>
<td>3.2</td>
<td>Jet Velocity for Attached Plugged Nozzle (Config. 2)</td>
<td>33</td>
</tr>
<tr>
<td>3.3</td>
<td>Correlation of Surface Velocity Runs 1-12 (Config. 1)</td>
<td>36</td>
</tr>
<tr>
<td>3.4</td>
<td>Correlation of Surface Velocity Runs 13-18, 30-35 (Config. 2)</td>
<td>37</td>
</tr>
<tr>
<td>3.5</td>
<td>Correlation of Surface Velocity Runs 19-21, 22-27, 28-29, 36-40 (Conf. 3, 4, 5 &amp; 6)</td>
<td>38</td>
</tr>
<tr>
<td>3.6</td>
<td>Two-Dimensional Jet Velocity $m = 23$ lb/min, $\Delta p = .10$ psi</td>
<td>41</td>
</tr>
<tr>
<td>3.7</td>
<td>Two-Dimensional Jet Velocity $m = 22$ lb/min, $\Delta p = .135$ psi</td>
<td>42</td>
</tr>
<tr>
<td>5.1</td>
<td>Normal Flow Open Shroud Purge System</td>
<td>65</td>
</tr>
</tbody>
</table>
LIST OF TABLES

<table>
<thead>
<tr>
<th>Table No.</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>Test Nozzle Configuration and Flow Conditions</td>
<td>22</td>
</tr>
<tr>
<td>II</td>
<td>Summary of Test Data</td>
<td>26</td>
</tr>
<tr>
<td>III</td>
<td>Equilibrium Temperatures</td>
<td>28</td>
</tr>
<tr>
<td>IV</td>
<td>Cask Temperature Correlations</td>
<td>51</td>
</tr>
<tr>
<td>No.</td>
<td>Assigned Drawing No.</td>
<td>Title</td>
</tr>
<tr>
<td>-----</td>
<td>----------------------</td>
<td>------------------------------------------------------------</td>
</tr>
<tr>
<td>1.</td>
<td>2334911</td>
<td>Fuel Cask Mounting Assy</td>
</tr>
<tr>
<td>2.</td>
<td>2335093</td>
<td>Axial Flow Purge System Assy</td>
</tr>
<tr>
<td>3.</td>
<td>2335343</td>
<td>Axial Flow Purge System Assy No. 2</td>
</tr>
<tr>
<td>4.</td>
<td>2335335</td>
<td>Axial Flow Purge System with Shroud</td>
</tr>
<tr>
<td>5.</td>
<td>2335355</td>
<td>Open and Closed Air/Water Shroud System</td>
</tr>
<tr>
<td>6.</td>
<td>2335346</td>
<td>Fuel Cask Mounting Assy Instrumentation</td>
</tr>
<tr>
<td>7.</td>
<td>2335347</td>
<td>Heat Shield Assy Instrumentation</td>
</tr>
<tr>
<td>8.</td>
<td>2334348</td>
<td>Mounting Bracket Instrumentation</td>
</tr>
<tr>
<td>9.</td>
<td>2335349</td>
<td>Fuel, Capsule, and Cask Assy Test Instrumentation</td>
</tr>
</tbody>
</table>
1.0 INTRODUCTION

1.1 Background

In June of 1967, NASA Headquarters held a review of the ALSEP Program as related to the overall Apollo mission safety. Specific attention was given to the safety aspects of the RTG fuel cask which maintains an on pad temperature of approximately 650°F. It was determined that such a temperature level would represent a potential ignition hazard to the vehicle propellant vapors. Therefore MSC requested that Bendix conduct a preliminary thermal study to determine the feasibility of reducing the graphite fuel cask surface temperature to a level below 400°F during the prelaunch operations.

The results of the thermal study were transmitted to MSC on June 27, 1968. The basic conclusion was that the surface temperature of the graphite cask could be maintained below 400°F if the cask were locally purged from a directed nozzle or orifice supplied with a cooling flow of 225 to 450 cfm, (17-35 lb/min) at 70°F.

As a consequence of this study, an RTG Cask/Ignition Source Committee was established with members from MSC, KSC, GAEC, Sandia, LaRC, and Bendix. The purpose of this committee was to evaluate the potential ignition source problem of the cask in the presence of the various vehicle fuel vapors and to establish the requirements for reducing the cask surface temperature in a manner consistent with existing flight hardware and KSC capability. The first meeting of the committee was held at MSC on August 10-11, 1968. As a result of that meeting, Bendix was directed to submit a proposal to MSC for the purpose of conducting an engineering feasibility study and development tests to maintain the graphite cask surface below a maximum temperature of 350°F. The study was initiated on August 15, 1967 and was completed on January 12, 1968. This report summarizes the analysis, design, and test activities conducted at Bendix during that period.
1.2 Review of Cask Cooling Requirements

The SNAP-27 graphite fuel cask (GLFC) is located to the left of the ALSEP compartment in the LM SEQ bay. At approximately T-16 hours in the Apollo launch sequence, the fuel capsule assembly (FCA) is loaded into the cask. This FCA has a thermal energy output of 1500 watts and a surface temperature of approximately 1100°F, at the time of loading. Upon loading, the GLFC surface temperature rises from ambient to a free convection level between 550° and 650°F in approximately one hour. After vehicle liftoff and the subsequent loss of the convection portion of cooling to the cask, the GLFC surface temperature further increases to the range of 700° - 800°F.

It has been estimated that the spontaneous ignition temperature (SIT) of the command module and service module RCS propellant, monomethyl hyarazine (MMH) is of the order of 380°F - while the other vehicle propellants, UDMH and N₂H₄, have spontaneous ignition temperatures of 480° and 518°F respectively. Because of the possibility of propellant leakage from the A/S or LM fuel disconnects, supply lines, or the tanks themselves, the cask assembly presents a potential ignition hazard to the vehicle.

In order to reduce this hazard, it is necessary to actively cool any exposed cask surfaces to a temperature level below that of the aforementioned spontaneous ignition temperatures.

1.3 Study Objectives

In order to guide the selection and evaluation of the various possible cask cooling systems, the following ground rules were established.

1. As a goal, the design approach should preserve current Bx/A/GE/GAEC flight interfaces in order to minimize the impact on existing design.
2. If possible, the implementation of the design should utilize existing KSC facility capabilities from both the ground and airborne systems.
3. Only on-pad cooling of the cask will be required.
4. The GE 19F graphite fuel cask design will be used for all thermal analysis and design
5. Only single fuel leakage failures will be accommodated.
In regard to implementing this cooling, several air/GN₂ vehicle purge systems are in existence in the vicinity of the cask or if deemed necessary, a water supply could be made available. Three basic cooling concepts were judged worthy of further study in regard to satisfying the above objectives. These concepts are listed below along with a brief summary of their system advantages, requirements, and constraints.

Concept I - Forced Convection, Air/GN₂ Cask Purge System

- Air/GN₂ supply is available from the S IV B instrumentation unit (IU) purge duct.
- Purge source located below nozzle yields simplest approach, with minimal impact to flight hardware.
- Implementation has the shortest lead time to hardware - and with lowest cost.
- Requirements include:
  1) Air/GN₂ flow rate of 20 to 50 lb/min
  2) Air/GN₂ supply pressure of 0.25 - 0.5 psig.

Concept II - Open and Closed Water Cooling Systems

- S IV B IU water/methanol system not recommended for supply source/per MSFC.
- Requirements include:
  1) Open system, flow rate of 4-5 lb/hr - closed system flow rate of 60 lb/hr for up to 60 hours.
  2) Overboard vent to avoid inside vapor problems.
  3) Mechanical disconnection of supply and removal of cooling system at liftoff.
  4) Monitor of water level.
  5) Additional modifications to ground supply, swing arm, unbilical plate, IU area, etc.
Concept III - Finned Graphite Fuel Cask

Passive system with free convection cooling

Requirements include:

1) 110 fins, 9 inches long, 0.1 inches thick, and 1.5 inches high

2) Removal of fins after vehicle launch

3) Total revision of the BxA/GE/GAEC/NAR interface including new cask support structure, etc.

4) Additional ALSEP weight of 5 lbs.

Based on the information presented to the RTG Cask/Ignition Source Committee in August 1967 at MSC, Bendix was directed to continue the cask cooling studies of Concepts I and II (Air/GN$_2$ purge and water jacket) and to suspend work on Concept III (Finned Cask).

1.4 Description of Selected Concepts

The scope of this study was to evaluate the feasibility of utilizing on-pad active cooling air and water systems to reduce the temperature of the external cask surface below 350°F. The BxA/GE/GAEC interface established in the ground rules of CCP #29 for the graphite fuel cask was used to define the envelope and mounting structure for the engineering models of the air/water cooling systems.

Some of the early conceptual cask cooling systems are shown in Figure 1.1. As indicated in Figure 1.1 from the standpoint of required mass flow and pressure differential, the fully enclosed systems present the most efficient means of cooling. However since the thermal dissipation from the cask is by pure radiation after the early portion of the launch sequence, it would be necessary to mechanically remove any enclosed cooling system after the GN$_2$ shutdown. Therefore the early phases of the study also included an investigation of the possible mechanisms of cooling shroud removal and/or destruction systems.

Because of the added complications and interference problems of the shrouded cooling systems, only the full axial shroud was actually fabricated for testing and the majority of testing was carried out with the various open flow configurations as pictured in Figure 1.2.
### Comparison of Air/GN₂ Purge Systems and Water Shroud Systems for On Pad Cask Cooling Configuration

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Axial Flow</th>
<th>Enclosed Flow</th>
<th>Closed</th>
<th>Open (Boiling)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air/GN₂ Purge Systems</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>nozzle attached open</td>
<td>HIGH</td>
<td>HIGH</td>
<td>HIGH</td>
<td>HIGH</td>
</tr>
<tr>
<td>nozzle attached shrouded</td>
<td>MODERATE</td>
<td>LOW</td>
<td>LOW</td>
<td>LOW</td>
</tr>
<tr>
<td>nozzle detached flow</td>
<td>HIGH</td>
<td>NO</td>
<td>NO</td>
<td></td>
</tr>
<tr>
<td><strong>Water Shroud Systems</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>enclosed flow</td>
<td>LOW</td>
<td>NO</td>
<td>NO</td>
<td></td>
</tr>
<tr>
<td><strong>Early Cooling Concepts</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Figure 1.1 Early Cooling Concepts**
Figure 1.2 Schematic of Test Nozzle Configurations

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
In Section 2 a short synopsis of the test results is given and the actual test data is summarized. Then in Section 3 the analysis and correlation of this data is developed and in Section 4 the corresponding thermal model is discussed.

2.0 TEST RESULTS

A photograph of the actual test setup is shown in Figure 2.1, and the details of this unit are given in Section 2.3. The individual test runs are tabulated in Section 2.4 along with their pertinent flow parameters and resultant range of equilibrium temperatures.

2.1 Discussion of Data

Because the actual cask cooling was more efficient than had been predicted by the pre-test analysis, the test plan was modified midway through the testing of the attached plugged nozzle, (configuration 2 of Figure 1.2). This modification was instituted for two reasons. First, it was evident that all of the proposed systems and flow conditions would provide more than adequate cooling on the cask - and thus without a test plan modification, no data would be obtained in the marginal temperature range. Secondly, on the basis of early predictions the more remote nozzle locations had been discarded as providing inadequate cooling with the available flow conditions. However upon receiving the initial test results, it became apparent that this conclusion was not entirely valid.

The fact that the actual cask cooling was more effective than had been predicted was not a result of erroneous pre-test analysis. After reduction of the test data it was found that the measured values of the film coefficients were well above both the experimental and theoretical values which exist in the open literature. The actual reasons for this apparent discrepancy probably lie in the state of the boundary layer development and are discussed in Section 3.

Various detached nozzle configurations were thereupon tested in several axial positions below the cask. In addition it was found that the thermal shield on the inboard side of the cask would function as a leeward flow deflector so that the nozzles could also be placed outboard and at an angle with respect to the cask axis. Finally a fully shrouded axial flow configuration was tested for purposes of comparison with the pre-test analysis.
Figure 2.1 Test Setup for Cask Cooling Series
The velocity of the airflow was measured at each exposed thermocouple on the cask surface so that correlations of both the flow field and the film coefficients could be made. It was found that a good correlation could be achieved between the local velocity on the open side of the cask and the local cask surface temperature differential above that of ambient. To an acceptable experimental accuracy, this result may be written as

\[
\frac{1}{T} = 9.5 \times 10^{-4} u_0^{0.71}
\]

and is general for the open side of all but the shrouded cask. Upon coupling the above to the semi-empirical velocity correlations for open flow, the temperature differential may then be written as a function of the upstream flow conditions.

Basically the correlations for open flow cooling systems may be divided into two categories - one for the essentially two-dimensional flow from a plugged nozzle configuration and the other for the essentially three-dimensional flow from conical nozzles. Because of the geometric nature of the flow from a plugged nozzle, this type of cooling configuration is only useful when the nozzle is aligned with the cask axis. On the other hand, as long as the present thermal shield remains near the back side of the cask, it is possible to have an angle of as much as 45° between the axis of the conical nozzles and the cask axis. This thermal shield then acts to direct the flow up the back side of the cask.

As noted, the correlations themselves have been developed for the front (i.e., open side) of the cask where there is effectively no airflow interference from the structure, etc. They are given below as:

**Plugged Nozzle**

\[
\angle T = 62 \left[ \frac{x + 1.9}{\Delta p^{1/4}} \frac{\sqrt{\dot{m}}}{\Delta p^{1/4}} \right]^{0.71}
\]
Conical Nozzle

\[ \Delta T = 48 \left( x + \frac{2.7 \sqrt{m}}{\Delta p^{1/4}} \cos \theta \sqrt{m \Delta p^{1/4}} \right) \]

where

\[ \Delta T = \text{temperature differential} \sim ^\circ F \]
\[ x = \text{surface distance from nozzle to cask cylinder surface point} \sim \text{feet} \]
\[ m = \text{mass flow} \sim \# \text{/sec} \]
\[ \Delta p = \text{pressure differential} \sim \text{psf} \]
\[ \theta = \text{angle between nozzle axis and cask axis}. \]

These equations are plotted for a realistic range of input parameters in Figures 2.2 through 2.7.

Within a tolerance of several degrees, the actual maximum surface temperatures measured were invariably 10% higher than the maximum open side temperatures in °F. Thus from the \( \Delta T \) of the correlations plus the ambient temperature, the maximum cask surface temperature may be predicted. Note also that the maximum \( \Delta T \) will occur in the neighborhood of the maximum downstream distance on the thin-walled section of the cask cylinder.

Because of the presence of lightweight insulation on the structure above the cask, it is also of interest to note the cooling air velocities in this region. In Table I the range of velocities three feet above the cask are tabulated for the cooling configurations tested. These are quite low and their accompanying dynamic pressures are rather insignificant in terms of any damage which might be done by the cooling air.
OPEN SIDE CASK SURFACE TEMPERATURES

vs.

LOCAL DISTANCE FROM NOZZLE

for

Axial Flow, Attached or Plugged Nozzle

(Configuration 1, 2, & 4)

$m = 24$ Lb./Min.

Figure 2.2
OPEN SIDE CASK SURFACE TEMPERATURE RISE

vs.

LOCAL DISTANCE FROM NOZZLE

for

Axial Flow, Attached or Plugged Nozzle

(Configuration 1, 2, & 4)

n = 35 Lb./Min.

DISTANCE FROM NOZZLE, X ~ FEET

Figure 2.3
OPEN SIDE CASK SURFACE TEMPERATURE RISE

vs.

LOCAL DISTANCE FROM NOZZLE

for

Axial Flow, Attached or Plugged Nozzle

(Configuration 1, 2, & 4)

\( m = 50 \text{ lb. / Min} \)

**Figure 2.1**
OPEN SIDE CASK SURFACE TEMPERATURE RISE

vs.

LOCAL DISTANCE FROM NOZZLE

for Conical Nozzles [Configuration 3, 5, & 6]

$\theta$ = Angle Between Cask Axis and Nozzle Axis

$M = 20$ Lb./Min.,

Figure 2.5
OPEN SIDE CASK SURFACE TEMPERATURE RISE

LOCAL DISTANCE FROM NOZZLE

for Conical Nozzles (Configuration 3, 5 & 6)

θ = Angle Between Cask Axis and Nozzle Axis

ṁ = 35 Lb./Min.

Figure 2.6
Open Side Cask Surface Temperature Rise

vs.

Local Distance From Nozzle

For Conical Nozzles (Configuration 1, 5, 6.)

- Angle Between Cask Axes and Nozzle Axis
  - $\phi=0.07$ psi
  - $\phi=0.2$
  - $\phi=0.4$

Figure 2.7
### TABLE I

TEST NOZZLE CONFIGURATIONS AND FLOW CONDITIONS

<table>
<thead>
<tr>
<th>Configuration Number</th>
<th>ATTACHED NOZZLE</th>
<th>ATTACHED PLUGGED NOZZLE</th>
<th>OPEN AXIAL FLOW</th>
<th>DETACHED PLUGGED NOZZLE</th>
<th>DETACHED NOZZLE AXIAL FLOW</th>
<th>DETACHED NOZZLE 45° ANGLE</th>
<th>SHROUDED AXIAL FLOW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Run Number</td>
<td>1 to 12</td>
<td>13 to 18</td>
<td>19 to 21</td>
<td>22 to 27</td>
<td>28 to 29</td>
<td>36 to 40</td>
<td>41 to 43</td>
</tr>
<tr>
<td>Mass Flow #/min</td>
<td>4-50</td>
<td>10-50</td>
<td>24-53</td>
<td>18-35</td>
<td>22-35</td>
<td>18-43</td>
<td>5-20</td>
</tr>
<tr>
<td>Pressures psi</td>
<td>.06-.83</td>
<td>.03-.42</td>
<td>.01-.05</td>
<td>.10-.33</td>
<td>.06-.13</td>
<td>.04-.56</td>
<td>.02-.19</td>
</tr>
<tr>
<td>Nozzle Area in²</td>
<td>2.80-8.40</td>
<td>3.56-9.16</td>
<td>19.6</td>
<td>3.56-7.8</td>
<td>8.95</td>
<td>3.14-19.6</td>
<td>5.17</td>
</tr>
</tbody>
</table>

Typical Flow Conditions to Maintain a Maximum Cask Wall Temperature of 200°F (using 70°F cooling air)

<table>
<thead>
<tr>
<th>Flow Rate, #/min</th>
<th>15</th>
<th>25</th>
<th>50</th>
<th>35</th>
<th>30</th>
<th>30</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Differential, psi</td>
<td>0.04</td>
<td>0.06</td>
<td>0.05</td>
<td>0.09</td>
<td>0.15</td>
<td>0.25</td>
<td>0.06</td>
</tr>
<tr>
<td>Maximum Velocity 3 feet above cask, fps</td>
<td>7</td>
<td>10</td>
<td>18</td>
<td>10</td>
<td>8</td>
<td>9</td>
<td>3</td>
</tr>
</tbody>
</table>
2.2  Recommendations

Under current planning it appears that the actual air supply for
the cask cooling would be obtained from the IU cooling duct which presently
exists in the vicinity of the cask. It has been estimated that the ductwork
could supply approximately 35 lb/min of air at a pressure on the order of
0.3 psi. However the air supply temperature may be as high as 130°F as
opposed to the 70°F of this test.

Since the maximum cask surface temperature is approximately
ten percent higher than the maximum open side cask temperature, the
maintenance of a maximum surface temperature of 350°F with an air
supply temperature of 130°F would require a maximum correlated
\[ \Delta T \] of 200°F. As can be seen in Figures 2.2 through 2.7, this \[ \Delta T \]
is well within the capability of the estimated air supply - irrespective of
the particular cooling system configuration. However it should be noted
that the current cask support hardware is now much more extensive than
the original simulated test support structure. Therefore a wider margin
(lower \[ \Delta T \]) will undoubtedly be required of the prototype configuration.
Nevertheless, there will be a considerable latitude in the placement of the
cooling nozzle in the flight vehicle. This latitude may thereby be used to
advantage in terms of minimizing interference and revision problems
in the existing hardware.

At present the IU duct consists of two deadend semicircles.
For the cask cooling it had been proposed that these be joined by a "T"
and that a second duct be routed from there to the cooling nozzle. However
this modification appears to be an unwarranted complication since the
location of this T is a considerable distance from the cask. A more simple
solution would be to tap in to the IU duct at a location adjacent to the cask
itself and merely connect the ends of the IU duct. The pressure drop
in the duct itself is essentially negligible - regardless of the placement
of the existing cooling holes and proposed nozzle. Therefore there would
be no effective flow output change from the IU duct, whether the cask
cooling air were taken from an asymmetrical point in the air supply
system or not. However there would be a weight and interference saving
if the IU tap were as close to the cask cooling nozzle as possible.
2.3 **Test Setup**

A photograph for the basic accessory hardware is shown in Figure 2. A variable speed drive was attached to the Roots blower so that the mass flow of air could be regulated from approximately twenty to fifty pounds per minute. A scrubber was attached to the output of the Roots blower in order to reduce the pulsation of the flow out of the blower. Ambient air at approximately 70°F was used in all tests and the air temperature rise across the blower for these operating conditions was essentially negligible.

The diagrams of the test plan nozzles are shown in Drawings 2, 3 and 4. However as noted in section 2.1, the test plan was modified in the course of testing and conical nozzles of 3-3/8, 2-1/2, and 2-inch diameters were fabricated on location for additional testing. Also it was part of the test plan modification to operate the plugged nozzles fully separated from the cask and to operate the ducting as a cooling source without any nozzle.

The thermocouples for these tests were placed on the cask, electric fuel capsule, and associated support hardware as shown in Drawings 6 through 9. The static pressure for the air cooling system was measured along the supply duct as well as at the nozzle orifice by means of Statham pressure transducers. Also a thermocouple was placed in the downstream end of the air supply duct. The output of these sensors was monitored by an EI Data Acquisition System connected to the thermocouples and transducers through RIC reference boxes. A Sorenson 15 Kw power supply was used to power the Electric Fuel Capsule Simulator and this output was also monitored through the DAS system.

Additional on-site, hand operated instrumentation consisted of an Alnor Velometer and Thermocon. The velometer was used to measure the air flow velocities at the nozzle orifice, adjacent to the cask surface thermocouples, above the cask, etc. for the purposes of overall test correlations. The thermocon was used to check the surface temperatures on non-instrumented areas of the cask.
2.4 Summary of Test Data

The test nozzle configurations and flow conditions are given in Table I. The individual runs are tabulated in Table II with nozzle sizes given below.

<table>
<thead>
<tr>
<th>Config. No.</th>
<th>Nozzle Area (in$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1a</td>
<td>8.40</td>
</tr>
<tr>
<td>1b</td>
<td>7.04</td>
</tr>
<tr>
<td>1c</td>
<td>5.17</td>
</tr>
<tr>
<td>1d</td>
<td>2.80</td>
</tr>
<tr>
<td>2a</td>
<td>9.16</td>
</tr>
<tr>
<td>2b</td>
<td>7.80</td>
</tr>
<tr>
<td>2c</td>
<td>5.92</td>
</tr>
<tr>
<td>2d</td>
<td>3.56</td>
</tr>
<tr>
<td>7b</td>
<td>5.17</td>
</tr>
</tbody>
</table>

The individual equilibrium temperatures are listed in Table III.
## TABLE II

**SUMMARY OF TEST DATA**

<table>
<thead>
<tr>
<th>Run Number</th>
<th>Configuration</th>
<th>Mass Flow #/min</th>
<th>Pressure Differential, psi.</th>
<th>Nozzle Exhaust Velocity ~ fps</th>
<th>Range of Surface Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1a</td>
<td>39</td>
<td>0.18</td>
<td>143</td>
<td>103-146°F</td>
</tr>
<tr>
<td>2</td>
<td>1a</td>
<td>24</td>
<td>0.06</td>
<td>81</td>
<td>117-182</td>
</tr>
<tr>
<td>3</td>
<td>1a</td>
<td>50</td>
<td>0.30</td>
<td>191</td>
<td>93-129</td>
</tr>
<tr>
<td>4</td>
<td>1b</td>
<td>22</td>
<td>0.10</td>
<td>97</td>
<td>110-154</td>
</tr>
<tr>
<td>5</td>
<td>1b</td>
<td>35</td>
<td>0.22</td>
<td>145</td>
<td>98-128</td>
</tr>
<tr>
<td>6</td>
<td>1b</td>
<td>50</td>
<td>0.43</td>
<td>228</td>
<td>92-120</td>
</tr>
<tr>
<td>7</td>
<td>1c</td>
<td>15</td>
<td>0.10</td>
<td>93</td>
<td>118-159</td>
</tr>
<tr>
<td>8</td>
<td>1c</td>
<td>20</td>
<td>0.14</td>
<td>117</td>
<td>111-145</td>
</tr>
<tr>
<td>9</td>
<td>1c</td>
<td>30</td>
<td>0.29</td>
<td>160</td>
<td>100-129</td>
</tr>
<tr>
<td>10</td>
<td>1d</td>
<td>9</td>
<td>0.14</td>
<td>92</td>
<td>127-188</td>
</tr>
<tr>
<td>10a</td>
<td>1d</td>
<td>3.5</td>
<td>0.06</td>
<td>40</td>
<td>159-233</td>
</tr>
<tr>
<td>11</td>
<td>1d</td>
<td>20</td>
<td>0.41</td>
<td>212</td>
<td>104-140</td>
</tr>
<tr>
<td>12</td>
<td>1d</td>
<td>30</td>
<td>0.83</td>
<td>317</td>
<td>96-126</td>
</tr>
<tr>
<td>13</td>
<td>2a</td>
<td>22</td>
<td>0.05</td>
<td>78</td>
<td>121-201</td>
</tr>
<tr>
<td>14</td>
<td>2a</td>
<td>35</td>
<td>0.11</td>
<td>115</td>
<td>103-166</td>
</tr>
<tr>
<td>15</td>
<td>2a</td>
<td>50</td>
<td>0.22</td>
<td>163</td>
<td>96-148</td>
</tr>
<tr>
<td>16</td>
<td>2b</td>
<td>22</td>
<td>0.07</td>
<td>83</td>
<td>119-197</td>
</tr>
<tr>
<td>17</td>
<td>2b</td>
<td>35</td>
<td>0.14</td>
<td>127</td>
<td>101-163</td>
</tr>
<tr>
<td>18</td>
<td>2b</td>
<td>10</td>
<td>0.03</td>
<td>42</td>
<td>159-259</td>
</tr>
<tr>
<td>19</td>
<td>5&quot; elbow</td>
<td>38</td>
<td>0.03</td>
<td>61</td>
<td>123-224</td>
</tr>
<tr>
<td>20</td>
<td>31&quot; elbow</td>
<td>53</td>
<td>0.05</td>
<td>85</td>
<td>109-195</td>
</tr>
<tr>
<td>21</td>
<td>cask (3)</td>
<td>24</td>
<td>0.01</td>
<td>38</td>
<td>157-277</td>
</tr>
<tr>
<td>22</td>
<td>2b on</td>
<td>23</td>
<td>0.10</td>
<td>93</td>
<td>142-254</td>
</tr>
<tr>
<td>23</td>
<td>5&quot; elbow (4)</td>
<td>35</td>
<td>0.16</td>
<td>139</td>
<td>115-206</td>
</tr>
<tr>
<td>24</td>
<td>2c on</td>
<td>35</td>
<td>0.30</td>
<td>190</td>
<td>112-196</td>
</tr>
<tr>
<td>25</td>
<td>5&quot; elbow (4)</td>
<td>22</td>
<td>0.13</td>
<td>125</td>
<td>134-237</td>
</tr>
<tr>
<td>26</td>
<td>2d on</td>
<td>22</td>
<td>0.33</td>
<td>200</td>
<td>130-226</td>
</tr>
<tr>
<td>27</td>
<td>5&quot; elbow (4)</td>
<td>18</td>
<td>0.23</td>
<td>167</td>
<td>143-246</td>
</tr>
<tr>
<td>28</td>
<td>3.4&quot; diameter</td>
<td>22</td>
<td>0.06</td>
<td>75</td>
<td>132-231</td>
</tr>
<tr>
<td>29</td>
<td>nozzle on</td>
<td>35</td>
<td>0.13</td>
<td>120</td>
<td>109-189</td>
</tr>
<tr>
<td>30</td>
<td>5&quot; elbow (5)</td>
<td>2c</td>
<td>0.10</td>
<td>102</td>
<td>132-192</td>
</tr>
<tr>
<td>31</td>
<td>2c</td>
<td>30</td>
<td>0.19</td>
<td>137</td>
<td>122-166</td>
</tr>
<tr>
<td>32</td>
<td>2c</td>
<td>11</td>
<td>0.03</td>
<td>57</td>
<td>161-246</td>
</tr>
<tr>
<td>33</td>
<td>2d</td>
<td>20</td>
<td>0.28</td>
<td>162</td>
<td>126-188</td>
</tr>
<tr>
<td>34</td>
<td>2d</td>
<td>34</td>
<td>0.42</td>
<td>208</td>
<td>116-164</td>
</tr>
<tr>
<td>35</td>
<td>2d</td>
<td>11</td>
<td>0.10</td>
<td>98</td>
<td>157-227</td>
</tr>
<tr>
<td>Run Number</td>
<td>Configuration</td>
<td>Mass Flow #/min</td>
<td>Pressure Differential, psi.</td>
<td>Nozzle Exhaust Velocity ~ fps</td>
<td>Range of Surface Temperature</td>
</tr>
<tr>
<td>------------</td>
<td>--------------------------------------</td>
<td>-----------------</td>
<td>-----------------------------</td>
<td>-----------------------------</td>
<td>------------------------------</td>
</tr>
<tr>
<td>36</td>
<td>5&quot; nozzle, 14&quot; off, cask (a) (b)</td>
<td>43</td>
<td>0.04</td>
<td>69</td>
<td>136-222°F</td>
</tr>
<tr>
<td>37</td>
<td>2.5&quot; nozzle</td>
<td>21</td>
<td>0.18</td>
<td>135</td>
<td>136-234</td>
</tr>
<tr>
<td>38</td>
<td>45&quot; off cask</td>
<td>35</td>
<td>0.44</td>
<td>224</td>
<td>114-191</td>
</tr>
<tr>
<td></td>
<td>a) 45°</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>39</td>
<td>20&quot; nozzle</td>
<td>18</td>
<td>0.24</td>
<td>178</td>
<td>129-232</td>
</tr>
<tr>
<td>40</td>
<td>45&quot; off cask</td>
<td>24</td>
<td>0.56</td>
<td>238</td>
<td>111-190</td>
</tr>
<tr>
<td></td>
<td>a) 45°</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41</td>
<td>7b</td>
<td>20</td>
<td>0.19</td>
<td>94</td>
<td>113-146</td>
</tr>
<tr>
<td>42</td>
<td>7b</td>
<td>10</td>
<td>0.09</td>
<td>63</td>
<td>131-177</td>
</tr>
<tr>
<td>43</td>
<td>7b</td>
<td>4.6</td>
<td>0.02</td>
<td>28</td>
<td>177-264</td>
</tr>
<tr>
<td>TIME</td>
<td>CASE SUPPORT BASE</td>
<td>CASE EXTERNAL SURFACE</td>
<td>CASE INTERNAL SURFACE</td>
<td>CASE SURFACE</td>
<td></td>
</tr>
<tr>
<td>----------</td>
<td>-------------------</td>
<td>------------------------</td>
<td>-----------------------</td>
<td>-------------</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1.20</td>
<td>0.70</td>
<td>0.70</td>
<td>0.70</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>1.10</td>
<td>0.60</td>
<td>0.60</td>
<td>0.60</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1.00</td>
<td>0.50</td>
<td>0.50</td>
<td>0.50</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>0.90</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>0.80</td>
<td>0.30</td>
<td>0.30</td>
<td>0.30</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>0.70</td>
<td>0.20</td>
<td>0.20</td>
<td>0.20</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>0.60</td>
<td>0.10</td>
<td>0.10</td>
<td>0.10</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>0.50</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td></td>
</tr>
</tbody>
</table>

DATM-763
Page 28
3.0 THEORETICAL AND EXPERIMENTAL ANALYSIS

3.1 Background

The original intent of this section was to present the analytical design approaches and performance predictions for the various cask cooling configurations. (Figure 1.2). However, after examining the test results, it became obvious that the functional behavior of the experimental data was not in accord with classical flow theory. Thus at this point there is little to be gained by going through rather involved analytical operations which at best do not give accurate predictions of the problem at hand. Rather these preliminary performance predictions are relegated to a discussion (Section 5.0) of this report and it is more profitable to directly examine the data in light of classical flow theory.

Note that the above comment is not intended to indicate that the classical approach to this design analysis is in error. Instead, as will be subsequently shown, the problem seldom achieves conditions whereby the classical approach may be legitimately applied.

3.2 Basic Open Flow Analysis

In this type of analysis it is conventional to first establish the particular external flow field around the cask, and from this data to determine the local boundary layer characteristics. For heat transfer purposes the boundary layer characteristics would include the effects of both the momentum and energy equations. However at the temperature ranges encountered in this problem, it is adequate and even desirable to simply use the Reynolds analogy to relate the local heat transfer to the local skin friction

For preliminary purposes the external flow may be essentially treated as that emanating from a free jet of suitable geometry. Then the boundary layer skin friction may be obtained through the application of local similarity considerations, i.e., the local external velocity is assumed everywhere constant so that typical uniform velocity solutions may be used (Blasius flat plate, power law distributions, etc.).
The application of these fundamental solutions requires that the problem is well specified. That is, the flow in the jet and/or the boundary layer must be either completely laminar or completely turbulent, the point of flow attachment must be well defined, etc. However when the problem is not well specified it is not entirely reasonable to anticipate reliable predictions from such an approach. For example, the transition Reynolds number for a jet is of the order of 100 as based on the nozzle conditions. When this value is exceeded the flow from a jet will become turbulent, but only after it has traveled a finite distance. Thus the flow must travel the order of 20 nozzle diameters downstream before a fully developed turbulent jet is established. This result is true to a large part, irrespective of the upstream flow conditions. Consequently the initial portion of the external flow on the cylinder as in configuration 1, can not properly be considered either laminar or turbulent. Further since the velocity decay rate is different for a two-dimensional laminar and turbulent jet, it is not clear just what external velocity distribution should be predicted in that region.

A similar problem exists in the boundary layer where again a finite distance is required for a turbulent boundary layer to develop. Also the problem exists that there is a minimum surface Reynolds number to generate and maintain a turbulent boundary layer. Thus even if the outer jet flow is turbulent, it is not necessarily true that the adjoining boundary layer will be turbulent. However in such a situation the corresponding heat transfer in this pseudo laminar boundary layer will be greater than that of an undisturbed laminar boundary layer. Thus from these above comments, as well as others which will be forthcoming, it is apparent that the flow patterns of the various cask cooling configurations seldom achieve a definity of character which would allow a relatively simple type of classical analysis.

Consequently in the almost total absence of any theories for transitional types of flow, it is perhaps more enlightening to look at the evidence, i.e., experimental results, and then trace this information backwards in order to ascertain the particular character of the flow.
For the velocities and temperatures encountered in this experiment, the air may be considered to be an incompressible, perfect gas. If for the moment, the external flow patterns of each cooling configuration are considered to be representative of free jets, then the theoretical velocity decay downstream of the nozzle will be proportional to:

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Laminar</th>
<th>Turbulent</th>
</tr>
</thead>
<tbody>
<tr>
<td>Two-dimensional Jet</td>
<td>$X^{-1/3}$</td>
<td>$X^{-1/2}$</td>
</tr>
<tr>
<td>Circular Jet</td>
<td>$X^{-1}$</td>
<td>$X^{-1}$</td>
</tr>
</tbody>
</table>

where $X$ is the downstream distance.

Thus for each cooling configuration it might be anticipated that the rate of change of centerline velocity with distance would at least be asymptotic to the proper values in the above table. In Figures 3.1 and 3.2, plots of maximum velocity versus downstream distance are given. Although definite slopes are in evidence in these figures, they are not in agreement with those of the above table. This lack of agreement may be traced largely to two basic sources. First the tabular values are for fully developed flows and these do not occur until the jet, whether laminar or turbulent, has proceeded the order of ten or twenty nozzle diameters downstream - thus it should not necessarily be expected that the initial portion of the flow will follow the classical decay rate. Secondly, the surface of the cask represents a constriction to the otherwise free expansion of the jet - thus the velocity decay rate will be altered by this effect. Also note that the finite transition distance between laminar and turbulent flow will introduce a gradual slope change - at least in the two-dimensional jet.

In all the configurations tested, the nozzle Reynolds numbers exceeded the critical value so that each jet eventually became turbulent. For two-dimensional jets, this transition region is quite easy to locate as in Figures 10 and 11. Upstream of this region the flow has some of the characteristics of a laminar jet, but it is basically undeveloped in terms of classical jet flow. Therefore, it is not to be expected that the laminar jet form would be followed - even if the inherent jet instability has not grown to a significant level to develop into turbulence.
Figure 3.1
Figure 3.2
The net result of the above deviations from theory is that it is easier to look for an empirical data correlation rather than to attempt to reproduce the experimental results with modifications to existing theory. Such an approach is quite justifiable in the case of turbulent jet flows since in the absence of a reliable turbulent transport theory, it is the observed rate of jet spreading which has led to the accepted velocity decay rates. On the other hand, at least until flow impingement on the cask, the circular jet will behave as theoretically predicted since the laminar and turbulent velocity decay rates are the same.

Assuming that the external flow velocities are now available, it is next necessary to establish the boundary layer characteristics and thereby the shear and heat transfer at the cask surface. For a perfect, incompressible gas, the Reynolds analogy may be written

\[ \text{St} = \frac{1}{2/3} \frac{C_f/2}{Pr} \]

Therefore the film coefficient, \( h \), will be

\[ h = \frac{Q}{A \Delta T} = \frac{St \rho g u C_p \Delta T A}{A \Delta T} \]

or

\[ h = \frac{\rho g u C_p}{Pr^{2/3}} \frac{C_f/2}{C_p} \]

By considering each segment of the cask to be essentially a flat plate, the skin friction may be related to a local Reynolds number. However the main difficulty is in establishing a reference distance upon which to base this local Reynolds number. For configuration 1 this orientation is no problem, but for the detached nozzles, the effective initial point of flow contact is not obvious. In order to temporarily bypass this problem it is possible to make direct use of the experimental data and thereby determine the sensitivity of the film coefficient to the effective point of flow attachment.
For laminar flow on a flat plate the skin friction is proportional to $Re_x^{-1/2}$ and for fully developed turbulent flow on a flat plate the skin friction is approximately proportional to $Re_x^{-1/5}$. For each location on the cask at each flow configuration, the effective surface distance is approximately constant so that

$$C_{f/2}^{Lam} \sim u^{-1/2}$$

$$C_{f/2}^{Turb} \sim u^{-1/5}$$

In addition at each location on the cask the source heat per unit area $(Q/A)$ is essentially fixed. Thus the film coefficient will be inversely proportional to the difference between the wall temperature and the free stream temperature, i.e.,

$$h \sim \frac{1}{T}$$

Consequently for each station and configuration we may write

$$\sqrt{u} \Delta T = \text{const.} \quad \text{(laminar flow)}$$

$$u^{0.8} \Delta T = \text{const.} \quad \text{(turbulent flow)}$$

Upon plotting the experimental values of $u$ and $\Delta T$ for various thermocouple locations on the cask, it is somewhat surprising to find that on log-log paper the entire set of primary test data can nearly be correlated with one straight line (Figures 3.3, 3.4, 3.5). (Note that the term primary is here applied to the non-interfered portion of the cask surface, i.e., the front side. This is the only portion of the data where some direct correlation can be anticipated a priori.) However, the slope of this experimental curve follows neither the laminar nor the turbulent value.
Figure 3.3

Correlation of Surface Velocity vs. Temperature Differential

Runs 1-12 (Configuration 1)
Figure 3.4

CORRELATION of SURFACE VELOCITY vs. TEMPERATURE DIFFERENTIAL

Runs 13-18, 30-35
(Configuration 2)

• T-68
Φ T-64
• T-60
CORRELATION
of
SURFACE VELOCITY
v.
TEMPERATURE DIFFERENTIAL

Runs 10-21, 22-27, 28-29, 38-40
(Configuration 3, 4, 5, & 6)

$\times$ T-68
@ T-64
$\ast$ T-69

TEMPERATURE DIFFERENTIAL, $\Delta T \circ ^\circ F$

Figure 3.5
Irrespective of the slope of surface velocity versus $\Delta T$, some streamwise variation in the proportionality between the velocity and temperature increment might be anticipated. However only in the case of the attached nozzle (configuration #1) is any such dependence evident. This particular case most nearly approximates the boundary layer flow along a flat plate (with an adverse pressure gradient) where a flow initiation point was well defined.

Since the streamwise boundary layer dependence decreases with increasing streamwise distance, the detached nozzle flows, which all the others are, have evidently become attached far enough upstream in relation to the placement of the three surface thermocouples on the cask cylinder (T-68, T-64, and T-60) to effectively reduce the streamwise boundary layer dependence to a negligible level. This is not to say that the boundary layer characteristics are not changing as the flow proceeds downstream, but that the cumulative effects of an external velocity decay and a lag in the adjustment to local similarity in the cask cylinder have resulted in a shear coefficient (or Stanton number) which is essentially independent of streamwise distance. Such behavior is in contrast to the classical skin friction dependence noted above where a constant external velocity is assumed throughout to develop such theories.

Note that none of the above comments include the fully shrouded case (configuration 7) which requires a separate form of analysis.

Since a correlation now exists between the velocity and the temperature differential on the open side of the cask, it is possible to combine this correlation with the previously established correlation of velocity and upstream flow conditions parameters to yield a relation between the air supply parameters and the surface temperature rise. In addition it is somewhat fortuitous that the maximum measured surface temperature, which occurs at T-61, is always within several degrees of being 10% higher than the temperature at T-60, which is one of the correlated values. Thus a correlation between the air supply conditions and the maximum surface temperature is relatively easy to establish. For the attached nozzle (configuration 1) there should be some streamwise dependence of $\Delta T$ other than what is obtained from the basic external velocity decay. However the data scatter exceeds whatever variation might be predicted so that in the interest of overall simplicity, it is accepted that all the experimental data of configurations 1 through 6 essentially obeys the relation:

$$\frac{1}{\Delta T} = 9.5 \times 10^{-4} u_e^{0.71}$$

Note that this slope (0.71) lies between the aforementioned laminar and turbulent values (0.5 and 0.8).
Although all the attached and plugged nozzles (configurations 1, 2, and 4) essentially generate two-dimensional jets at the onset of the flow, only when the nozzle is relatively close to the cask is the basic characteristic of this flow preserved. That is, as these nozzles are placed further from the cask, the expanding two-dimensional jets eventually merge because of their circular placement. Thus their velocity decay (which is directly related to the jet expansion area through the momentum equation) changes its form as the flow loses its two-dimensional characteristics.

Consequently it is not possible to conveniently apply a single velocity decay formula to the plugged nozzle. Initially an attempt was made to find a modified velocity decay versus downstream distance through logarithmic plots. However this slope is also a function of the fictitious source point in the nozzle. After a considerable amount of data manipulation, it appeared that the best compromise would be to use the basic axisymmetric form, but with a different value of the empirical constant which necessarily appears in all turbulent flow formulations (Figures 3.6, 3.7). This formulation is reasonably adequate for the attached nozzle as well, so that all the velocity distributions may be expressed in basically the same form.

Note that in the above commentary no mention is made of the velocity distribution across the jet. However in essentially all the cases, the maximum velocity occurred in close enough proximity to the cask wall so that the centerline velocity effectively approximates the edge velocity of the boundary layer along the cask wall. Also no consideration is given to the blockage of the flow by the cask itself. Because of the relative size of the cask and jet dimensions, it might be expected that this effect would be significant. However the fact that the experimental correlation is adequate is justification enough for neglecting the presence of the cask in calculating the streamwise velocity distribution.

Consequently the inviscid flow adjacent to the cask and emanating from an axially aligned two-dimensional jet may be written

\[ u_e = \frac{54 \sqrt{\dot{m}}}{x + 1.9} \frac{\Delta p^{1/4}}{\sqrt{\dot{m}/\Delta p^{1/4}}} \]
Figure 3.6

TWO-DIMENSIONAL JET VELOCITY

EFFECTIVE SURFACE DISTANCE

\[ \dot{m} = 23 \text{ lb./min.} \]
\[ \Delta p = 0.10 \text{ psi} \]

EFFECTIVE SURFACE DISTANCE, \( x + x_o \sim \text{ FEET} \)
TWO-DIMENSIONAL JET VELOCITY

vs.

EFFECTIVE SURFACE DISTANCE

\( \bar{M} = 23 \text{ Lb. /Min.} \)

\( \Delta p = 0.155 \text{ ps} \)

Figure 3.7
where from the basic form

\[ \bar{u} \xi = \frac{3}{8 \pi} \frac{K}{\xi (x + x_0)} \quad K = \int uex \, dA \]

\[ \xi = 0.043 \sqrt{K} \text{ instead of } 0.016 \sqrt{K} \text{ as in the classical circular jet. Upon combining this result with the above temperature - velocity correlation, the temperature differential becomes} \]

\[ T = 62 \left[ \frac{x + 1.9}{\sqrt{\dot{m} / \Delta p}^{1/4}} \right]^{0.71} \]

where

\[ \dot{m} = \text{mass flow } \sim \text{lb/sec} \]

\[ \Delta p = \text{pressure differential } \sim \text{psf} \]

\[ x = \text{distance from the nozzle to a point on the cask cylinder } \sim \text{ft.} \]

This equation is plotted in Figures 2.2, 2.3, and 2.4 for a realistic range of the involved parameters. Note that this temperature differential represents that of the open side of the cask. Therefore the maximum cask wall temperature will be approximately 10% higher than the absolute value of the wall temperature at maximum \( x \) of the cask cylinder. Also note that the apparent wide discrepancy between this correlation and several of the experimental runs is largely a result of the partial misalignment of the plugged nozzle relative to the cask. Thus especially at the lower mass flows and pressures, nozzle alignment is an important factor.
For the circular jets, (configurations 3, 4, and 6) a similar approach gives a velocity distribution as:

\[ u_e = \frac{77 \cos \theta \Delta p^{1/4} \sqrt{\dot{m}}}{x + \frac{2.7 \sqrt{\dot{m}}}{\Delta p^{1/4}}} \]

where in this situation, \( \theta \) is the angle between the jet axis and the cask axis \((0 \leq \theta \leq 45^\circ)\). Except for the angular correlation, this form is in agreement with the classical laminar and/or turbulent distribution. Upon combining this form with the temperature - velocity correlation, the cask temperature differential for circular jet cooling on the front side of the cask becomes (Figures 2.5, 2.6, and 2.7)

\[ \Delta T = 48 \left[ \frac{x + \frac{2.7 \sqrt{\dot{m}}}{\Delta p^{1/4}}}{\cos \theta \Delta p^{1/4} \sqrt{\dot{m}}} \right]^{0.71} \]

where

\[ \dot{m} = \text{mass flow } \sim \text{#/sec} \]

\[ \Delta p = \text{pressure differential } \sim \text{psf} \]

\[ \theta = \text{angle between nozzle axis and cask axis } (0 \leq \theta \leq 45^\circ) \]

\[ x = \text{actual distance between nozzle and front side cask surface point} \]

Again the maximum cask surface temperatures occur on the sides of the cask and are approximately 10% higher than the absolute temperatures of the open side. Note that it is precisely the presence of the radiation heat shield which directs the flow up the back side of the cask when the nozzle is off center with respect to the cask axis. Under this situation the back side temperatures are at least as low as the front temperatures. However removal of the radiation shield would completely disrupt this back side cooling.
3.3 Correlation of Open Flow to Theory

From the standpoint of future analysis as well as academic interest, it is useful to compare the effective skin friction coefficient as obtained through the above data correlation with that of the skin friction predicted through classical boundary layer theory. Since the above correlation essentially bypassed the need for film coefficients and internal thermal analysis, no effective surface area was established. However for purposes of comparison, a projected geometric area and a uniform internal heat source should provide adequate estimates of the actual values. As noted above, by the Reynolds analogy, the skin friction coefficient may be written as:

\[
C_f/2 = \frac{Pr^{2/3} h}{\rho g u C_p} = \frac{Pr^{2/3} Q}{\rho g u C_p \Delta TA}
\]

Also by the cask wall data correlation

\[
\frac{1}{\Delta T} = 9.5 \times 10^{-4} u_e^{0.71}
\]

Combining these equations yields

\[
C_f/2 = \frac{9.5 \times 10^{-4} Pr^{2/3} Q}{\rho g C_p A u^{0.29}}
\]

The net heat output is 1500 watts or 1.4 BTU/sec and the effective area of this cask is of the order of

\[
A \approx \pi \times \frac{8}{12} \times \frac{20}{12} = 3.5 \text{ sq. ft.}
\]
Using these values of Q and A plus an air Prandtl number of 0.71, the skin friction coefficient becomes

\[ C_{f/2} = \frac{0.0165}{u_e^{0.29}} \]

Because of a lack of dependence upon streamwise distance in the data correlation for \( \Delta T \), it is necessary to assume some effective surface distance if the above skin friction coefficient is to be compared to the classical values which are functions of Reynolds number, i.e.,

\[ C_{f/2} = C_{f/2} (Re_x) = C_{f/2} \left( \frac{u_e X}{\nu} \right) \]

Since the exponent of \( u_e \) is 0.29, the Reynolds number and thus \( X \) will be raised to this power. In reality \( X \) varies from approximately 0.5 to 3.0, but upon raising this to the 0.29 power, this is only a variation from 0.8 to 1.3 or a factor of +30%. This variation at least allows a rough comparison with the classical values of \( C_{f/2} \). Using a nominal value of 1.5 for \( X \), the experimental skin friction coefficient becomes

\[ C_{f/2} = \frac{0.23}{(Re_x)^{0.29}} \]

In the table below this form compared with the accepted flat plate turbulent values. (Note that the Stanton number \( \sim 1.25 C_{f/2} \)).

<table>
<thead>
<tr>
<th>( Re_x )</th>
<th>( C_{f/2} \times 10^3 ) Exp.</th>
<th>( C_{f/2} \times 10^3 ) F. P.</th>
</tr>
</thead>
<tbody>
<tr>
<td>10^5</td>
<td>8.1</td>
<td>3.8 (2.1 Lam.)</td>
</tr>
<tr>
<td>10^6</td>
<td>4.2</td>
<td>2.3</td>
</tr>
<tr>
<td>10^7</td>
<td>2.1</td>
<td>1.4</td>
</tr>
</tbody>
</table>
As can be seen, the projected experimental values of $C_f/2$ exceed the pure flat plate values by more than the uncertainty in $X$. Further if some $X$ variation with Reynolds number had been allowed in these projected values of $C_f/2$, then the disparity would be greater. Although the cask surface is quite rough, subsequent calculations indicated that this condition would at most result in a small increase in the skin friction at the lower Reynolds number. Therefore the observed skin friction increases are probably attributable to the high level of free stream turbulence and the lag in boundary level development.

With respect to the lag in boundary layer development, there are two contrasting factors which result in a seemingly anomalous behavior of the boundary layer. However upon looking more closely at the boundary layer structure, the results are seen to be plausible and even expected. For a typical laminar boundary layer on a flat plate, there is a certain similarity at the various streamwise locations. That is, the individual boundary layer profiles may all be reduced to the same profile by an appropriate choice of scale factors. However when there is turbulence in the free stream, the momentum transfer to the external portion of the laminar boundary layer from the free stream is considerably increased. Therefore the effective laminar boundary layer is much thinner, than as predicted by the similarity solutions - which results in a higher velocity gradient near the wall and thus a much higher value of skin friction. Such a situation exists in the region of flow attachment on the cask where the Reynolds number is initially low and even after reaching the transition level, there is a finite distance required to achieve transition. A similar situation, though to a lesser extent, exists with the turbulent boundary layers. That is, the free stream turbulence has somewhat of a thinning effect upon even a turbulent boundary layer where ordinarily the turbulence level decreases in the outer portions of the boundary layer. Thus the distribution and magnitude of the disagreement between classical and present experimental skin friction coefficients is relatively easily explained by the fact that all the jet flows tested were themselves turbulent - thereby resulting in a high level of free stream turbulence.

However there still exists the problem of the lack of streamwise dependence of the skin friction. At first glance this independence might appear to be the result of an excessive lag in the boundary layer development so that the streamwise position has little effect upon the wall shear. The alternative to this viewpoint is that the high level of turbulence throughout the boundary layer results in a very rapid adjustment of the boundary layer. Thus the boundary layer behaves similarly to the fully developed flow between parallel walls, where the distance between the walls is the characteristic distance. Owing to the high level of turbulence in the outer flow (and the resultant increased wall
shear) it appears that the second viewpoint is the valid one. Therefore the lack of streamwise shear dependence would indicate that there are compensating effects which result in an almost negligible change in the effective boundary layer thickness - or more properly, the momentum thickness of the boundary layer is essentially independent of the streamwise position. These compensating effects are probably located in the rate of growth of the turbulent intensity, both within and without of the boundary layer. That is, especially in the situation where the nozzle and cask are relatively close and thus a greater streamwise variation would be expected, the normal time (or streamwise) delay in the development of the turbulent intensity in both the free stream and the boundary layer results in increasing transport across the boundary layer as the flow proceeds downstream - thereby resulting in a negligible change in the boundary layer momentum thickness and thus a negligible change in wall shear as a function of position, i.e.,

$$\frac{\tau}{\rho u^2} = \frac{d\theta}{dx} + \sigma^* u \frac{du}{dx}.$$ 

Note however that there is a wall shear drop resulting from the streamwise decay of the external velocity.

3.4 Shroud Analysis

The experimental values of the heat transfer for the shrouded case also exceed the theoretical predictions by a considerable amount. The theoretical analysis, as presented in the appendix, was based on a smooth walled, fully developed channel flow. Since the geometry of the shroud rather faithfully represents a channel flow system, the differences in the heat transfer between theory and experiment must be represented by the conditions imposed upon the two-dimensional flow, i.e., the smooth wall, the fully developed flow, and possibly the form of the Reynolds analogy.

While the degree of roughness of the cask wall was not significant in increasing the shear of the relatively thick open flow boundary layers, the same roughness can be significant in a shrouded system where the boundary layer growth is constrained. Although the surface roughness of the test cask was far from uniform, the average roughness appears to be of the order of one mil. Thus the ratio of the channel height to the roughness height is of the order of 200. For the Reynolds numbers of the shroud experiments (3,000 - 10,000), it does not appear that such a roughness ratio would contribute
more than a 10% increase to the wall shear. However these estimates are based upon pipe flow results since there is very little experimental data for roughness in two-dimensional flow. Therefore because the flow conditions are in a region of high sensitivity to roughness, it is possible that the projected value of a 10% increase in wall shear could be either markedly high or low. Also note that some of the rougher local areas on the cask wall would result in a higher local shear.

In well controlled experiments, it is usually necessary for a flow to travel at least 50 inlet dimensions downstream before fully developed channel flow is achieved, (either laminar or turbulent). Up to that point the wall velocity gradients are much higher because the center portion of the flow has not yet adjusted to the wall shear effects. Consequently it is to be expected that the experimental film coefficients would exceed the predicted values over at least the majority of the cask cylinder. The Reynolds numbers for the three test runs were all sufficiently large to maintain turbulent channel flow so it appears that the shroud flow is going in that direction. In the table below, the Reynolds numbers, the predicted film coefficients, and the measured film coefficients are given for the three shrouded flow tests.

<table>
<thead>
<tr>
<th>Run</th>
<th>Re</th>
<th>$h_{\text{calc}}$ (BTU/HR°F ft$^2$)</th>
<th>$h_{T-68}$</th>
<th>$h_{T-64}$</th>
<th>$h_{T-60}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>43</td>
<td>3,200</td>
<td>6.6</td>
<td>11.</td>
<td>9.</td>
<td>7.6</td>
</tr>
</tbody>
</table>

Note that the decrease in the measured values of $h$ as the flow proceeds downstream is not entirely a result of the gradual development of the channel flow. There is also the factor of $(1 - T_e/T_w)$ which represents the decrease in heat transfer resulting from the increase in air temperature. As may be seen in the preliminary shroud design in the appendix, this factor can represent a significant reduction in heat transfer - especially in the case of low mass flows and thus low total heat capacity of the cooling air.

Since there is little valid data and no complete theory for turbulent flow, it is not possible to apportion the relative importance of the factors which effectively increased the film coefficients in the shrouded flow tests. While it might appear that a lag in the development of the channel flow was the major contributor (as opposed to the surface roughness), the accuracy of the applied form of the Reynolds analogy in such a flow is certainly subject to question.
However in light of the success of the open flow cooling systems, there does not appear to be any point in attempting to generate a valid and plausible means to predict the actual heat transfer which is attained from the more cumbersome shroud system.

4.0 THERMAL CORRELATION

Although a thermal analysis was not used (or necessary) in the correlation of the experimental data, several analyses were carried out to verify that the experimentally derived film coefficients were compatible with the measured cask temperatures. The basic model for the thermal analysis employs a finite difference approach wherein the physical configuration is subdivided into discrete finite elements - each of which is assumed to be isothermal. The mass of each element is assumed to be concentrated at its geometric center and is termed a node. These nodes have arbitrary individual heat sources and are linked together in a resistance network analogous to that of an electrical circuit. In this analogy the temperature is the analogue of the voltage and the heat flow is the analogue of the current.

For purposes of comparison, test runs 4, 5, and 6 were numerically analyzed using the graphite material properties as supplied by G. E. for the Super Temp Cask. A 26 node model was used for this analysis and in Table IV the predicted and measured temperatures are given for these nodes - along with the film coefficients as supplied from the test analysis of Section 3.4. Since no cask liner was used in these tests, the internal air essentially acted as an oven to moderate the temperatures between the capsule and the cask, as well as along the cask. Thus the various temperature gradients were not excessive, and thereby the nodal cask temperatures could be legitimately averaged in the analyses.

While the correlation between the predicted and averaged, measured temperatures is not exceptionally good, it is close enough within the bounds of the method of analysis. Further refinement of the model and the internodal properties could undoubtedly bring the predicted temperatures closer to the individual measured temperatures. However this upgrading would be a somewhat of a post factum approach. Rather the basic point to be made is that the film coefficients as developed in Section 3.4 are in the proper range for the measured temperatures.
## TABLE IV

CASK TEMPERATURE CORRELATIONS

**FILM COEFFICIENT, \( h \sim \text{BTU/HR, FT}^2 \degree F \)**

**TEMPERATURES \( \sim \degree F \), PREDICTED & MEASURED**

<table>
<thead>
<tr>
<th>Location</th>
<th>Free Conv.</th>
<th>Test Run #4</th>
<th>Test Run #5</th>
<th>Test Run #6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( h )</td>
<td>( \text{Pred/Meas} )</td>
<td>( \text{Pred/Meas} )</td>
<td>( \text{Pred/Meas} )</td>
</tr>
<tr>
<td>Barrel (SLA)</td>
<td>1.0</td>
<td>527/529</td>
<td>24.0</td>
<td>133/137</td>
</tr>
<tr>
<td>Barrel (Shield)</td>
<td>1.0</td>
<td>556/556</td>
<td>25.0</td>
<td>132/130</td>
</tr>
<tr>
<td>Band (upper)</td>
<td>1.0</td>
<td>430/463</td>
<td>5.8</td>
<td>127/140</td>
</tr>
<tr>
<td>Band (lower)</td>
<td>1.0</td>
<td>432/457</td>
<td>26.9</td>
<td>97/94</td>
</tr>
<tr>
<td>Dome (upper)</td>
<td>1.5</td>
<td>364/377</td>
<td>6.8</td>
<td>121/129</td>
</tr>
<tr>
<td>Dome (lower)</td>
<td>1.0</td>
<td>437/</td>
<td>3.9</td>
<td>169/</td>
</tr>
<tr>
<td>Shield (front)</td>
<td>1.5</td>
<td>117/105</td>
<td>25.0</td>
<td>70/70</td>
</tr>
<tr>
<td>Shield (rear)</td>
<td>1.0</td>
<td>75/</td>
<td>1.0</td>
<td>70/</td>
</tr>
<tr>
<td>Barrel In. (SLA)</td>
<td>0.3</td>
<td>560/571</td>
<td>0.3</td>
<td>183/195</td>
</tr>
<tr>
<td>Barrel In. (Sh)</td>
<td>0.3</td>
<td>586/589</td>
<td>0.3</td>
<td>182/196</td>
</tr>
<tr>
<td>Fuel</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Capsule</td>
<td>0.4</td>
<td>1017/1001</td>
<td>1.0</td>
<td>899/873</td>
</tr>
<tr>
<td>End Plate</td>
<td>0.7</td>
<td>570/581</td>
<td>1.0</td>
<td>328/312</td>
</tr>
<tr>
<td>Internal Air</td>
<td>634/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LM</td>
<td>1.0</td>
<td>77/85</td>
<td>1.0</td>
<td>70/70</td>
</tr>
<tr>
<td>SLA</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
5.0 DISCUSSION OF PRE-TEST ANALYTICAL STUDIES

5.1 Axial Flow Air Cooling Systems

5.1.1 Attached Nozzle - Open Flow

In a highly convergent nozzle such as this, (Drawing 2) the velocity at the nozzle exit is essentially uniform in distribution and equal to the Bernoulli value times some discharge coefficient

\[ U_{ex} = \sqrt{2 \Delta p / \rho} C_D. \]

It is then necessary to establish the subsequent velocity decay as the flow proceeds along the cylinder. For the present purposes of analysis, it was assumed that the velocity decay of a corresponding two-dimensional free jet would yield an adequate initial estimate. Such an approximation is justifiable because of the counteracting effects of the presence of the wall - one of which restricts the flow expansion and thereby tends to retard the rate of velocity decay, while the other, the wall friction, tends to increase the rate of velocity decay.

The basic problem lies in the determination of the character of the boundary layer flow, i.e., laminar or turbulent. The critical Reynolds number for a two-dimensional nozzle is of the order of 50 as based on the height of the nozzle. For the application under consideration here, the Reynolds number is at least an order of magnitude greater than 50. Therefore the flow issuing from the nozzle will become turbulent. However this does not necessarily mean that the boundary layer flow along the wall will be turbulent. Downstream of the nozzle, the nozzle turbulence may be regarded as free stream turbulence - and while this free stream turbulence will tend to promote the transition to turbulence in the boundary layer, unless the wall Reynolds number is sufficiently high to support turbulence, there will be no transition to turbulent flow at the wall.

Consequently the basic flow pattern on the cask wall is initially laminar with a high level of free stream turbulence. Then as the Reynolds number increases along the cask (which it does - although somewhat diminished by the jet velocity decay) the flow would enter a transition phase and eventually become fully turbulent. However in the relatively short length of the cask wall, it is not clear that the flow would ever achieve a fully developed turbulent boundary layer, and in some cases, transition itself may not be attained.
As the basic jet will be turbulent in all cases, the centerline velocity decay rate may be taken as that of the turbulent value for a two-dimensional jet. In order to calculate the heat transfer, a local similarity may be assumed with the free stream velocity equal to that of the local mean jet velocity.

It is then necessary to make some estimate of the character of the boundary layer flow at each point along the wall. Because of the presence of free stream turbulence in the main flow, the heat transfer in the laminar portion of the boundary layer will be greater than that of purely laminar flow. Then as the flow proceeds downstream and through transition, the turbulent heat transfer value will probably be approached - although it is doubtful that the actual value will be attained in such a short distance.

Since no methods exist for predicting heat transfer in transitional flow, only qualitative estimates of the heat transfer may be made where this type of flow occurs. However if the level of free stream turbulence may be estimated, then it is possible to predict the increase in heat transfer in the laminar flow regime. This value will then furnish a lower bound for the local heat transfer. Beyond that point it is purely a matter of estimation as to what level of turbulence and its corresponding heat transfer are attained as the flow proceeds downstream. In the extreme cases, the flat plate Reynolds number will furnish a criterion for the location of the transition region and fully developed turbulent flow, but in the intermediate cases, no such criteria exist. Thus a wide tolerance in the predicted heat transfer will be necessary until actual experimental data becomes available.

As a typical design example, the following case is developed.

The available pressure differential at the nozzle is taken to be 1 psi. Thus the mean exit velocity (corrected by a discharge coefficient of 0.8) would be:

\[
\overline{u_{ex}} = 0.8 \sqrt{\frac{2 \times 144 \times 1}{0.002378}} = 278 \text{ fps}
\]

For a mass flow of 50 #/min, the height of the nozzle would be

\[
H = \frac{\dot{m}}{\rho g u_{ex} b} = 0.019 \text{ ft.}
\]
Therefore the nozzle Reynolds number would be

\[ R_e = \frac{278 \times 0.019}{1.65 \times 10^4} = 3.2 \times 10^4 \]

which is well above the critical value for a two-dimensional jet, i.e., the basic jet will be turbulent.

The centerline velocity decay of a two-dimensional turbulent jet may be written as:

\[ U_L = \frac{\sqrt{3}}{2} \sqrt{\frac{K \sigma^2}{x + x_o}} \]

where

\[ K = \int U^2 dy = U_{ex}^2 \lambda \]
\[ \sigma^2 = 7.7 \text{ (experimental value)} \]
\[ x_o = \text{fictitious reference point to match } U_L \text{ with } U_{ex} \]

For the free stream velocity along the wall, the average of the jet velocity is used, which is approximately 2/3 of the centerline velocity. With the above information, the free stream velocity and the corresponding local Reynolds number may be calculated as a function of the distance along the cask. In the table below, these values are given for the initial conditions

\[ \dot{m} = 50 \text{ #/min} \]
\[ p = 1 \text{ psi} \]
\[ C_D = 0.8 \text{ (discharge coef.)} \]
## ALSEP Cask Cooling Feasibility Study

<table>
<thead>
<tr>
<th>X(in)</th>
<th>X+X₀(ft)</th>
<th>Uₑ(fps)</th>
<th>Reₓ x 10⁻⁴</th>
<th>h(BTU/Hr ft² °R)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.11</td>
<td>180</td>
<td>1.2</td>
<td>9.5 - 24</td>
</tr>
<tr>
<td>2</td>
<td>0.28</td>
<td>113</td>
<td>1.9</td>
<td>6.0 - 15</td>
</tr>
<tr>
<td>4</td>
<td>0.44</td>
<td>91</td>
<td>2.4</td>
<td>4.8 - 12</td>
</tr>
<tr>
<td>6</td>
<td>0.61</td>
<td>77</td>
<td>2.8</td>
<td>4.1 - 10</td>
</tr>
<tr>
<td>8</td>
<td>0.78</td>
<td>68</td>
<td>3.2</td>
<td>3.6 - 9</td>
</tr>
<tr>
<td>10</td>
<td>0.94</td>
<td>62</td>
<td>3.7</td>
<td>3.3 - 8.3</td>
</tr>
<tr>
<td>12</td>
<td>1.11</td>
<td>57</td>
<td>3.8</td>
<td>3.0 - 7.5</td>
</tr>
<tr>
<td>14</td>
<td>1.28</td>
<td>53</td>
<td>4.1</td>
<td>2.8 - 7</td>
</tr>
</tbody>
</table>

The film coefficient values (h) given at the right of this table represent the upper and lower bounds which might be anticipated. These values were obtained from Volume 5 of the Princeton Series, "Turbulent Flows and Heat Transfer". In order to reduce the spread in the film coefficient as given above, it will be necessary to conduct experimental testing under the properly simulated conditions.

5.1.2 Detached Nozzle - Open Flow (Drawing 3)

Although the flow pattern for this type of nozzle is somewhat different from that at the attached nozzle, the approximations inherent in any turbulent flow analysis render the two problems virtually identical in terms of the applied numerical analysis. Thus except for a relocation of the streamwise coordinate (X), the preceding analysis (extended in X) will suffice for the preliminary heat transfer estimates.
5.1.3 Attached Nozzle - Shrouded Flow (Drawing 4)

The basic purpose of the cylindrical shroud is to constrain the expansion of the axial flow so that there is no velocity decay or flow separation. The flow pattern is essentially that of two-dimensional flow between parallel plates. However although the Reynolds numbers are adequate to maintain turbulent flow, it is not evident that fully developed flow would be achieved in such a short distance. This lag is partially a result of the inlet affects and partially a result of the finite time (distance) required for flow transition.

Two approaches are available to analyze this system. One approach is to use only gross property considerations and then predict an effective mixing factor from similar existing experimental data. The other is to use the established fully developed turbulent channel flow result and then attempt to estimate the deviations which would occur in the neighborhood of the channel inlet. Note that although a laminar two-dimensional inlet solution might be useful to indicate the initial level of heating, the subsequent portions of the flow are almost immediately in a transitory state and thus not amenable to a laminar type of behavior.

On the basis of gross properties, the temperature rise times the mass flow would be related to the total heat output as

\[ K \dot{m} C_p \Delta T = Q \]

where \( K \) is essentially a mixing factor used to relate the actual heat absorbed relative to the total heat capacity of the flow. Therefore the required mass flow can be written

\[ \dot{m} = \frac{Q}{K C_p \Delta T} \]

Also by continuity

\[ \dot{m} = \rho g \bar{u} A \approx \rho g \bar{u} \pi d H \]

where \( d \) is the mean shroud diameter and \( H \) is the distance between the cask wall and shroud.
Next the pressure drop for fully developed turbulent flow in a channel may be approximated as

$$\frac{\Delta P}{L} = \frac{H}{\rho u^2} = 0.19 \left(\frac{Re_H}{K}\right)^{-1/4}$$

The combination of these three formulas then leads to the result (for the existing cask geometry).

$$\Delta T \Delta P^{4/7} = \frac{0.14}{KH^{12/7}}$$

$$H \sim \text{ft.}$$

$$p \sim \text{psi}$$

Note however that this $\Delta p$ only represents the pressure drop in the shroud. In addition there will be a $\Delta p$ required to attain the basic velocity as

$$\Delta P_u = \frac{\rho u^2}{2}$$

or

$$\Delta P_u = \frac{m^2}{2 \rho g^2 \pi^2 d^2 H^2}$$

$$= \frac{1}{2 \rho} \left[ \frac{Q}{g \pi d H K C_p \Delta T} \right]^2 = \frac{18.0}{(KH \Delta T)^2}$$

Thus the total pressure requirement as a function of the flow temperature rise and effective mixing factor becomes

$$\Delta P_T = \frac{0.032}{H^3 (K \Delta T)^{7/4}} + \frac{1.6}{(KH \Delta T)^2}$$
As a typical example, assuming a mixing factor of 0.5, an H of 0.2 inches and a maximum ΔT of 100 degrees, the total pressure requirement would be

\[ \Delta P_T = 8.8 + 2.2 \]
\[ = 11.0 \text{ psf (0.076 psi)} \]

and the corresponding mass flow

\[ m = \frac{1.4}{0.5 \times 0.24 \times 100} = 0.117 \#/\text{sec} \]

or

\[ m = 7.0 \#/\text{min} \]

In selecting a shroud gap, the interference of the supporting structure limits the size to approximately 0.25 inches. Thus for a typical shroud gap of 0.20 inches, the reference level of h would be 16 for an m of 20 \#/min and 11. for an m of 5 \#/min.

Because this shrouded system can be operated at relatively low mass flows, it is necessary to modify the above film coefficient by the net increase in the flow temperature. Since the flow will be turbulent, a total mixing may be assumed so that the flow temperature increase will be

\[ \Delta T = \frac{Q}{m C_p} \]

and will be distributed linearly down the cask. Thus the actual correction will take the form

\[ \left( 1 - \frac{T_e + \frac{Q}{m C_p} \frac{X}{L}}{T_w} \right) \]

so that the modified film coefficient will be
h = 0.36 \frac{m^{3/4}}{H} \left[ \frac{T_e + 5.0 \frac{X}{m}}{1 - \frac{T_w}{T_e}} \right]

where

h \sim \text{BTU/ft}^2 \text{hr}^0 \text{R}

\dot{m} = \text{mass flow} \sim \# \text{/sec}

T_e = \text{ambient temperature} \sim \text{^0R}

T_w = \text{wall temperature} \sim \text{^0R}

H = \text{shroud gap} \sim \text{feet}

X = \text{cylinder distance} \sim \text{feet}

The alternate form of shroud analysis is based on an attempt to predict the local wall shear in a channel inlet and then to use the Reynolds analogy to convert this result to an effective heat transfer. While plots of the experimentally established values of skin friction versus Reynolds number for turbulent parallel flow in a channel are available, the form of the resulting curve fit is rather unwieldy for use in additional derivations. Therefore the approximate form as

\frac{C_f}{2} = 0.10 (Re_h)^{-1/4}

where \(C_f/2\) here represents the skin friction on only one side of the channel wall. Thus the Stanton number would be

\text{St} \approx \frac{0.10}{Pr^{2/3}} (Re_h)^{-1/4}

= 0.12 (Re_h)^{-1/4}

and the film coefficient

h = 2.2 \times 10^{-3} \frac{\bar{u}}{(Re_h)^{1/4}}
Through the continuity equation, this equation may be rewritten (for relatively small values of the shroud gap and for the particular cask geometry)

\[ h = 0.36 \frac{m^{3/4}}{H} \]

where

\[ m \sim \#/sec. \]

\[ H \sim \text{ft.} \]

5.2 Cross Flow Cooling Systems (Drawing 5)

The basic objective behind this type of design is to attain an essentially uniform temperature distribution across the flow. If this objective can be met, then the subsequent mass flow requirement will be considerably less than that of the other cooling systems.

The flow mixing which will be necessary to achieve a uniform temperature distribution will be derived from two sources; 1. The maintenance of turbulent flow between the walls, and 2. The maintenance of the Goertler vortices which will exist in the flow between concentric cylinders at the required Reynolds numbers for turbulent flow. The existence of such mixing will not allow any significant temperature gradients in the flow. Thus this type of shroud may be analyzed on the basis of the total heat content of the flow rather than on any local boundary layer theory. However it will be necessary to maintain some temperature difference between the cask wall and the air flow so that there will be a net heat transfer, i.e., it cannot be expected to use just enough mass flow so that the exit air is at the same temperature as the cask wall.

The basic method of analysis of this type of shroud is to first estimate the streamline pattern. Then through two-dimensional channel flow analysis, the mean velocity may be calculated as a function of the mass flow, channel width, and available pressure differential. As the cask heat output is known and the temperature gradient across the flow is assumed to be zero, the net temperature rise in the flow may be related to the mass flow and subsequently to the available pressure drop. It then only remains to check that the Reynolds numbers are adequate to maintain turbulent flow between the cylinder walls. The essentials of this analysis are given below.
The net heat output of the cask is 1500 watts or 1.4 BTU/sec. Thus the required mass flux in terms of a temperature rise is:

\[
\dot{m}_{\text{req}} \frac{C_p}{\Delta T} = Q
\]

or

\[
\dot{m}_{\text{req}} = \frac{336}{\Delta T}. \quad (\dot{m} \sim \#/\text{min})
\]

By continuity the mass flux may be related to the velocity as

\[
\dot{m} = \rho g U A
\]

where A is the effective cross sectional area of the entering flow. Thus the velocity may be related to the temperature rise as:

\[
U = \frac{336}{\rho g A \Delta T}
\]

For turbulent flow between parallel plates (channel flow) the pressure drop may be written

\[
\frac{d\Delta p}{dx} = 0.11 \left( \frac{U}{H} \right)^{2/3} \left( \frac{UH}{V} \right)^{-1/4}
\]

where H is the height of the channel.

If the ratio of the effective channel width to the channel height is defined as n, then the effective area, A, will be nH^2 and the above equations can be combined to yield:

\[
\Delta p \Delta T^{7/4} = \frac{0.135}{n^{11/4}H^{23/4}}
\]

As a sample case, the presently proposed configuration has an n of approximately 6 and an H of 0.037 feet. \(\Delta T\) was arbitrarily selected as 100°R and thus:
\[ \Delta p = \frac{0.135}{6^{2.75} (0.037)^{5.75} x (100)^{1.75}} = 50 \text{ psf or } 0.35 \text{ psi} \]

also \[ \dot{m} = \frac{336}{100} = 3.4 \text{#/min} \]

and \[ \bar{u} = \frac{336}{0.076 \times 6 \times (1.37)^2 \times 100} = 90 \text{ fps} \]

so that the Reynolds number will be:

\[ R_e = \frac{UH}{\nu} = \frac{90 \times 0.037}{1.65 \times 10^{-4}} = 13,500 \]

which is well above the level required to maintain turbulent flow.

An additional pressure drop will be required to accelerate the flow to the given velocity as

\[ P_1 - P_2 = \frac{\rho \bar{u}^2}{2} \]

or

\[ \Delta p_\text{u} = \frac{\rho \bar{u}^2}{2} = \frac{0.002378(90)^2}{2} \]

\[ = 9.6 \text{ psf} = 0.07 \text{ psi} \]

Thus the total pressure drop of the system will be

\[ \Delta p_{\text{total}} = 0.35 + 0.07 = 0.42 \text{ psi} \]

Since the thermodynamic analysis requires a film coefficient, it is necessary to establish the local heat transfer—even though the system is adequately analyzed on a gross basis. To determine the local film coefficient, the wall temperature is applied to the heat transfer as:

\[ q = \dot{q}_0 \left[ 1 - \frac{T_e}{T_w} \right] \]
where \( T_e \) is the effective boundary layer temperature. Since it has been assumed that there is total mixing, the local temperature rise may be written as:

\[
\dot{m} C_p \Delta T = 2 \pi R \int_0^x \dot{q} \, dx
\]

or

\[
\Delta T = \frac{2 \pi R \dot{q}}{\dot{m} C_p} x
\]

Assuming an ambient temperature of 80°F and a cask wall temperature of 340°F, this heat transfer (and thus film coefficient) correction will be

\[
h = h_o \left[ 1 - \frac{540 + \frac{2 \pi R \dot{q}}{\dot{m} C_p} x}{800} \right]
\]

Since the correction is linear the average value of \( h \) (as calculated from the first part) and the local value of \( h \) will be equal at a point halfway along the cask - thereby furnishing a means for calculating the reference value of \( h_o \), i.e.,

\[
h_{avg} = \frac{Q}{A \Delta T} = \frac{\dot{m} C_p}{A}
\]

thus at \( x = L/2 \)

\[
h_o = \frac{h_{avg}}{540 + \frac{2 \pi R \dot{q}}{\dot{m} C_p} \frac{L}{2}} - 1 - \frac{800}{800}
\]

Consequently for the example given above,

\[
h_o = 77 \text{ BTU/ft}^2 \text{ HR}^\circ \text{R}
\]

which gives a linear distribution of \( h \) from a value of

\[
h = 25 \text{ BTU/ft}^2 \text{ HR}^\circ \text{R}
\]
at the beginning of the flow, to a value of

\[ h = 15 \text{ BTU/ft}^2 \text{ HR} \, ^\circ\text{R} \]

at the opposite end of the cooling shroud.

5.3 Normal Flow Cooling Systems (Fig. 5.1)

The internal liner of the shroud has appropriately distributed holes so that the cooling mechanism is essentially a series of stagnation region flows on the cask. For purposes of analysis the individual stagnation flow regimes must then be modified to incorporate the cross flow which exists and increases towards the ends of the shroud.

As the distance between the shroud and the cask wall is increased the cross flow velocity is reduced, thereby decreasing the asymmetric effect upon the local impingement solutions. However the geometric restrictions of the mounting hardware have reduced the allowable separation distance to a point where the cross flow in the neighborhood of the shroud ends is a significant contributor to the overall cooling. This effect in itself is not undesirable, but in the center section of the shroud where there is little or no cross flow cooling, the direct flow impingement cooling is not operating at its most efficient configuration. Also the cooling should be greatest at the center since the end cooling is already aided by internal axial conduction through the end caps of the cask. To some extent this problem can be compensated for by non-uniform placement of the shroud cooling holes, but the required iterative analysis becomes much more complicated.

For convenience on the preliminary analysis, it is assumed that the stagnation and cross flow solutions are linear, i.e., may be superimposed upon each other. The basic procedure is to develop a local stagnation region flow solution and then integrate it over the effective area covered so that an average film coefficient may be established as a function of an increment of area. With this formulation and the available pressure and mass flow of coolant, it is possible to specify the number of holes in the shroud as a function of the average value of the film coefficient. A typical example is carried out below.
NORMAL FLOW OPEN SHROUD PURGE SYSTEM

Figure 5.1
From volume 5 of the Princeton Series the Stanton number for flow normal to a cylinder is given as:

\[ St = 0.57 \left( \frac{C_p \mu}{K} \right)^{0.6} \left( \frac{BD}{U} \right)^{0.5} \left( Re_D \right)^{-0.5} \]

where for a cylinder of diameter D, BD/U = 4.0.

Thus for a Prandtl number of 0.71 for air,

\[ St_{cyl} = 1.4/\sqrt{Re_D} \]

Because of the relatively small distance between the holes and the wall (the order of several hole diameters) the effective exit velocity may be used in the stagnation point solution i.e., there will be very little velocity decay in such a short distance. Note however that the stagnation velocity is somewhat fictitious in that it does not actually exist at the stagnation point.

Using the definition of the film coefficient as:

\[ h = \frac{q}{\Delta T} = \rho g U C_p St \]

the stagnation point film coefficient becomes

\[ h = 0.25 \frac{C_D}{\sqrt{Re_D}} \sqrt{\frac{2 \Delta P}{\rho}} \]

However since it is the stagnation region rather than the stagnation point which is of interest, this relation must be modified to be a function of r (the radial surface distance from the geometrical stagnation point). Basically this local value of h may be obtained by multiplying the stagnation value of h by the local velocity ratio. This velocity ratio as a function of r may be obtained from the momentum and continuity equations, but since the functional form is not simple and since it must be integrated to yield an average value of h as a function of r (or actually the effective surface area) it will be necessary to resort to a graphical means of solution.

In the table below the local and integrated average values of the film coefficient are given as a function of the ratio of the exit diameter to the surface radius.
where $\bar{h}/h_{st} = \frac{2 \pi}{\pi r^2} \int h'/h_{st} r \, dr$

Assuming that the available pressure differential would be 1 psi and the hole diameter to be 1/8 inch, then the values of $h_{st}$ and $A_{ex}$ are specified. Then the number of holes required and the resultant mass flow as a function of $h$ may be obtained as

number of holes $= n = \frac{A_{total}}{\Delta A}$

mass flux $= m = \rho g U_{ex} A_{ex}$

and these values are given in the table below (with $A_{total} \approx 350$ sq. in.)

<table>
<thead>
<tr>
<th>$h$ (BTU/ft$^2$R HR)</th>
<th>$h$ (no. 1/8&quot; dia. holes)</th>
<th>m (#/min)</th>
<th>$V_{exhaust}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>--</td>
<td>\infty</td>
<td>--</td>
</tr>
<tr>
<td>11.6</td>
<td>1750</td>
<td>163</td>
<td>200</td>
</tr>
<tr>
<td>9.0</td>
<td>440</td>
<td>42</td>
<td>53</td>
</tr>
<tr>
<td>7.7</td>
<td>200</td>
<td>19</td>
<td>24</td>
</tr>
<tr>
<td>6.8</td>
<td>110</td>
<td>10</td>
<td>13</td>
</tr>
<tr>
<td>5.7</td>
<td>50</td>
<td>4.6</td>
<td>5.7</td>
</tr>
</tbody>
</table>
Next it is necessary to establish the heat transfer corrections for the cross flow and the increase in the boundary layer temperature as the flow proceeds from the center section to the ends of the shroud. Assuming a 1/2 inch gap between the shroud and cask wall, the exhaust velocity at the ends of the shroud are given in the above table.

If the cask heat output is uniform, then the new heat added at any point along the cask will be

$$\int_0^x \dot{q} \, dx = \dot{q} x$$

where $x$ has its origin at the middle.

The accompanying temperature rise from this heat addition would be

$$m' C_p \Delta T \gamma' = \int q dx = \dot{q} x$$

where $m'$ is the mass added from the centerline up to point $x$ and $\gamma'$ is the mixing ratio, i.e., the proportion of the total flow which is actually heated. Then as $\dot{m} = m_T (L/2) X$, the temperature rise would be:

$$T = \frac{2\dot{q}}{\gamma' C_p L \dot{m}_T}$$

and thus is independent of $x$. Further the nominal values of $\Delta T$ for a mixing factor of approximately 0.5 and the relatively low mass flow of 20 \#/min would be only 24°R. Consequently the boundary temperature correction would be at most a second order one. Also from the above table it may be noted that the end exhaust velocity for the mass flux of 20 \#/min is approximately 25 fps. Such a velocity would not yield a significant increase in the film coefficient. Therefore since these two effects (flow temperature rise and cross flow) are small and act in opposite directions, they may properly be considered small and self-canceling for the realistic levels of open shroud cask cooling.
5.4 Water Cooling Systems

Water systems have been studied for the on-pad cooling of the RTG cask. Two water cooling systems have been considered; a closed system which removes the heat by a temperature rise of the water and an open system which removes the heat by boiling off water. The individual systems and thermal analysis are described in the following paragraphs.

The RTG cask is an eight inch diameter cylinder with spherical dome ends and has an overall length of 23 inches. The dome ends are insulated and the primary heat rejecting surface is the center portion of the cask. This cylindrical surface is 14.25 inches long and has an 8 inch diameter for a rejecting area of 2.5 square feet. This cask is held by a tubular frame to the LEM vehicle and limited space is available between the cask and frame for the water jacket hardware. For purposes of analysis, a water jacket thickness of 0.25 inches and a length of 14 inches were established. Heat is released at a rate of 5120 BTU/hr from the cask and is absorbed by the water in the cooling jacket.

The volume of the water jacket is 0.054 cubic feet and has a capacity for holding 3.37 pounds of water. If cooling water is available at 60°F, the necessary flow rate of water with a 100°F temperature rise to remove the heat is

\[ \dot{w} = \frac{q}{C_p \Delta T} = \frac{5120}{(0.99)(100)(60)} = 0.86 \text{ lbm/min} \]

where \( q = \text{heat absorbed, 5120 BTU/hr} \)

\( C_p = \text{specific heat, 0.99 BTU/lbm}^\circ\text{F} \)

\( \Delta T = \text{temperature rise, 100°F} \).

The entering water requires time to pick up the heat and pass through the jacket. This time is obtained by the following calculation,

\[ \frac{\text{Capacity}}{\text{flow rate}} = \frac{3.37}{0.86} = 3.9 \text{ minutes.} \]
5.4.1 Closed System

In the thermal analysis, there are three approaches with which to look at the water jacket cooling system and establish the heat transfer film coefficient. These approaches are free convection, forced convection and conduction; however none of these methods provide individually a true model of the actual situation. The thin layer of water around the cask does not allow the natural circulation currents to develop as in free convection and also a steady supply of cool water is introduced which affects the circulation currents. Likewise the inflow velocity of cool water is low and cannot provide good conditions for forced convection. In the conduction approach, the water is assumed as a solid material and heat is conducted from the cask to the water, as the water is slowly moved through the jacket. Thus the water serves as a heat sink and is discarded when warmed up. An analysis is made for each approach and the resulting film coefficients are compared.

The free convection analysis is based on correlation data for vertical plates and cylinders heat transfer studies. This correlation data is plotted as a function of Grashof (Gr), Prandtl (Pr), and Nusselt (Nu) numbers for the laminar, transition and turbulent flow ranges. For three-dimensional shapes, a characteristic length, L, is determined by

\[
\frac{1}{L} = \frac{1}{L_{\text{hor.}}} + \frac{1}{L_{\text{vert.}}}
\]

where

- \( L_{\text{hor.}} = \) average horizontal dimension
- \( L_{\text{vert.}} = \) height dimension

Therefore, the characteristic length is

\[
\frac{1}{L} = \frac{1}{8} + \frac{1}{14} = \frac{11}{56}
\]

or

\[
L = \frac{56}{11} = 5.1 \text{ inches}
\]
The product of Gr and Pr is

\[
Gr \cdot Pr = \left[ \frac{g \beta}{N^2} \right] L^3 \Delta T \left[ \frac{C_p \mu}{k} \right]
\]

where

- \( g \) = acceleration of gravity
- \( \beta \) = temperature coefficient of volume expansion
- \( \rho \) = density
- \( \mu \) = absolute viscosity
- \( L \) = characteristic length
- \( \Delta T \) = temperature difference between surface and water, 27°F
- \( C_p \) = specific heat
- \( k \) = thermal conductivity

Substitution of values for water at 100°F gives

\[
Gr \cdot Pr = (118 \times 10^6) \left( \frac{5.1}{12} \right)^3 (27) (4.52) = 1.11 \times 10^9
\]

This value for the Gr Pr product is at the beginning of the turbulent flow range and the following relationship holds

\[
Nu = 0.021 (GrPr)^{2/5}
\]

Therefore

\[
Nu = 0.021 (1.11 \times 10^9)^{2/5} = 87.2
\]
The Nusselt number is defined by this expression

\[ \text{Nu} = \frac{hL}{k} \]

where

- \( h \) = heat transfer coefficient
- \( L \) = characteristic length
- \( k \) = thermal conductivity

Now

\[ h = \frac{k}{L} \quad \text{Nu} = \frac{0.364 \times 12}{5.1} \quad (87.2) \]

\[ h = 75 \text{ BTU/hr ft}^2 \text{ oF} \]

The heat rejected by free convection is

\[ q = A \cdot h \Delta T \]

\[ q = (2.5) (75) (27) = 5060 \text{ BTU/hr}. \]

The forced convection thermal analysis is based on the flow conditions of the water in the jacket surrounding the fuel cask. Laminar flow exists for a Reynolds (Re) number below 2100. For the water jacket, the hydraulic diameter, \( D_H \), is

\[ D_H = D_2 - D_1 \]

where the subscript 2 denotes the outer diameter of the water jacket and subscript 1 is the inner diameter. Therefore,

\[ D_H = 8.5 - 8.0 = 0.5 \text{ inches} \]

The cross-sectional flow area is

\[ A = \frac{\pi}{4} (D_2^2 - D_1^2) \]

\[ = \frac{\pi}{4} (8.25) = 6.48 \text{ sq. in.} \]
Now the velocity, $V$, is determined from the expression,

$$V = \frac{\dot{w}}{\rho A}$$

where

$\dot{w} =$ water flow rate, 0.86 lbm/min

$\rho =$ water density, 62.2 lbm/ft$^3$

$A =$ flow area, 6.48 sq. in.

With these values, the velocity is

$$V = \frac{0.86 \times 144}{62.2 \times 6.48} = 0.0051 \text{ fps}$$

The Reynolds number becomes

$$Re_D = \frac{V D_H \rho}{\mu} = \frac{(0.0051)(0.5)(62.2)}{(0.458 \times 10^{-3})(12)} = 28.9$$

and therefore the flow is definitely laminar through the water jacket.

The heat transfer coefficient for laminar flow is based on the empirical equation which correlates experimental results for liquids. This equation has the form:

$$Nu = 1.86 \left[ \frac{Re_D Pr D}{L} \right]^{0.33} \left( \frac{\mu_b}{\mu_s} \right)^{0.14}$$

where the correction factor ($\mu_b/\mu_s$) accounts for the temperature variation effects on the physical properties. Substitution of values into equation (11) gives a value of

$$Nu = 1.86 \left[ \frac{(28.9)(4.52)(0.5)}{5.1} \right]^{0.33} \left( \frac{0.458 \times 10^{-3}}{0.292 \times 10^{-3}} \right)^{0.14}$$

$$Nu = 4.59$$
Then the heat transfer coefficient is

\[ h = \frac{k}{L}, \quad \text{Nu} = \frac{0.364 \times (12) \times (4.59)}{5.1} = 3.92 \text{ BTU/hr ft}^2 \text{ °F} \]

The water flow rate of 0.86 pounds per minute requires a 100°F temperature rise of the water and an average temperature of 110°F. If the cask surface temperature is 210°F, then a 100°F temperature differential exists and the heat transferred is

\[ q = AR \triangle T \]

\[ = (2.5) (3.92) (100) = 980 \text{ BTU/hr}. \]

A third approach with which to analyze the closed water cooling system is by conduction from the cask into the water. The water jacket thickness of 0.25 inches is very small compared to the cask radius and therefore the cask surface and the water jacket are considered parallel flat surfaces. The heat transfer conduction equation has the following form

\[ q = \frac{k}{L} A \triangle T \]

where \( k = \text{thermal conductivity of water}, 0.364 \text{ BTU/hr ft °F} \)

\( L = \text{length of conduction path}, 0.125/12 \text{ feet} \)

\( A = \text{cross sectional area}, \)

Thus the term \( k/L \) is similar to the film coefficient \( h \), and has the same units. This term has the value of

\[ \frac{k}{L} = \frac{0.364 \times (12)}{0.125} = 35 \text{ BTU/hr ft}^2 \text{ °F} \]

The necessary temperature difference to remove the heat is

\[ \triangle T = \frac{q}{A \left(\frac{k}{L}\right)} = \frac{5120}{2.5 \times (25)} = 59 \text{ °F}. \]
In the three thermal analysis approaches, the heat transfer film coefficients were determined as: for free convection - 75 BTU/hr ft² °F, for forced convection - 3.92 and for conduction - 35. The free convection and conduction approaches best fit the physical conditions of this problem even though there is a slow flow of water through the jacket. A conservative film coefficient is the 35 BTU/hr ft °F from the conduction analysis and the actual film coefficient value is determined from the experimental tests of this cooling system concept.

The temperature differential between a particular point on the cask surface and the water is determined from equation (12). But the temperature is expected to vary along the axis of the cask as the water temperature increases from the inlet. For the above system a temperature drop exists across the inner wall of the water jacket which was not determined in these calculations but the temperature is the same for all the above approaches.

5.4.2 Open System

The open system has the same sized water jacket as in the closed system but water is boiled off to remove the heat from the RTG cask. The water level is maintained high on the cask by a flow rate equal to the boil-off rate. In the open system the boiling temperature is 212°F and the water enters at 60°F; therefore some of the cask heat raises the water temperature to the boiling point and the remainder of the heat vaporizes the water. The flow rate of water is determined by

\[ \dot{w} = \frac{q}{c_p \Delta T + h_{fg}} \]

where

\[ \dot{w} = \text{water flow rate, lbm/hr} \]

\[ q = \text{heat to be rejected, 5120 BTU/hr} \]

\[ c_p = \text{specific heat of water, 0.99 BTU/lbm} \ - \ ^\circ F \]

\[ \Delta T = \text{temperature use of water, (212-60) = 152}^\circ F \]

\[ h_{fg} = \text{latent heat of vaporization, 97}^\circ BTU/lbm \]

with the above values, the flow rate is 4.57 lbm/hr.
The nature of the boiling is a complex phenomena and is dependent on many variables. An excess temperature, $\Delta T_x$, is the difference between the hot surface and fluid saturation temperature. This excess temperature and the heat flux of the surface determine the boiling regime of open water cooling system. The expected temperature excess of this system is small and the heat flux is 2045 BTU/hr - ft$^2$ which indicates a pool boiling regime. An empirical relationship between the excess temperature, heat flux and physical variables is

$$\frac{c_i \cdot \Delta T_x}{h_{fg} \cdot Pr \cdot Pr^{1.7}} = C_{sf} \left[ \frac{q/A}{\mu \cdot h_{fg}} \sqrt{\frac{g_c \cdot \sigma}{g \cdot (\rho - \rho_v)}} \right]^{0.33}$$

where

$c_i = \text{specific heat of saturated liquid}, 1 \text{ BTU/lbm}^0 \text{F}$

$q/A = \text{heat flux}, 2045 \text{ BTU/hr ft}^2$

$h_{fg} = \text{latent heat of vaporization}, 970 \text{ BTU/lbm}$

$g_c = \text{conversion factor}, 4.17 \times 10^8 \text{ lbm ft/lbf hr}^2$

$g = \text{gravitational acceleration}, 4.17 \times 10^8 \text{ ft/sec}^2$

$\rho = \text{density of saturated liquid}, 59.97 \text{ lbm/ft}^3$

$\rho_v = \text{density of saturated vapor}, 0.0372 \text{ lbm/ft}^2$

$\sigma = \text{surface tension of the liquid-to-vapor interface}, 4.98 \times 10^{-3} \text{ lbf/ft}$

$Pr = \text{Prandtl number of the saturated liquid}, 1.74$

$\mu = \text{viscosity of the liquid}, 0.205 \times 10^{-3} \text{ lbf/ft-sec}$

$C_{sf} = \text{empirical constant depending on the nature of the heating surface - fluid combination}, 0.01.$

(Reference for equation: "Principles of Heat Transfer," F. Krieth, p. 406). The excess temperature is 7.5°F for the above listed values and therefore the surface temperature is approximately 220°F. The pool boiling heat-transfer coefficient, $h_b$, is defined by
and has the value of 273 BTU/hr ft² - °F.

Therefore the thermal analysis has shown that both the closed and open water systems are capable of holding the RTG cask at or below a temperature of 220°F.

5.5 Cask Shroud Removal Investigation

Although the concept of a shrouded cask cooling system represents a highly efficient approach to the problem, it also presents an interference problem to the radiation cooling of the cask on the later portions of the flight. Therefore it is necessary to include some form of removal of the shroud in the total system design. The two removal techniques which were considered in this analysis were that of post launch mechanical removal or post launch self destruct of the shroud.

The problems encountered in the mechanical removal of the shroud were the many degrees of freedom required of the shroud to clear the cask mounting hardware. For a rigid shroud it appears that at least three separate pieces plus a mechanical release system (probably thermally actuated) would be required to be able to pull such a shroud away from the cask. In addition these mechanical withdrawal links (springs, etc.) would have to be oriented in different positions. Consequently the high complexity and probably weight penalty of such a removal system led to the consideration of the second type of system, i.e., the self-destructing system.

Once the cooling GN2 system has been shut down in the post launch sequence, the heat output of the RTG cask will increase the shroud temperature (by radiation heating). Therefore a material is required which is structurally sound at the shroud temperature during cooling and which will decompose at an elevated temperature without damaging the cask and its contents.
The actual form of the decomposition could be that of melting, charring, or subliming. However for melting, such as would be attained with a thermoplastic, the subsequent deposition of material presents a problem. On the other hand most subliming materials contain highly reactive chemical constituents such as fluorine which could attack the cask surface, etc. Therefore the charring materials appear to present the most attractive approach. Such materials as certain phenolics decompose in the proper temperature range (300° - 400°) and with the proper filler result in an end product of only a fine ash. The only real question is the actual time of decomposition in the anticipated environment. If this time were too large, there could be significant overshoot in the cask temperature above that of the equilibrium space radiation temperature. Such a temperature overshoot would increase the decomposition rate, but this type of interaction would require careful study to insure capsule integrity throughout the shroud removal sequence.
SUPPLIED BY THE DESIGN CORP. EAST ROCKAWAY, L.I., N.Y. OR EQUAL.

ANY ITEM LABELED WITH BOLTS, WELD OR BEND ITEM IS AS SHOW CHASING OR ITEM 18 IN PLACES SHOWN DENS ITEM IS AS A FEATURE.

AFTER ITEM IS IN PLACE AS SHOWN, DRILL 1/8 IN HOLE IN ITEMS 475 USING HOLES IN CLIPS OF ITEM IS AS PICTURED.

DRAWING 1

ITEM #12 OMITTED FOR CLARITY

ATM-763
Page 79

FUEL CASSET MOUNTING ASSY
NOTES:

1. THIS ASSEMBLY SHALL BE AN STAINLESS STEEL INTERCHANGEABLE SUB-ASSEMBLY.

2. THRU G-9, THE SLEEVeS THEMSELVES AND THE HARDWARE REQUIRED FOR SECURING THEM.

Drawing 3
NOTES:

1. CUTTING AND INSTALLATION THREADED DUCTING PER 24
   AS SHOWN (EACH TYPE)

2. END TUBE MOUNTING DEPENDS ON SIZE AND ROOM
   REQUIREMENTS

3. ATTACH PIPING TO INSTRUMENTATION VALVES
   TO BE FIXED TO CARRIER frame AND OTHER STRUCTURES
   USING ANCHORS AND SEAT NUTS.

4. TYPICAL INSTALLATION INSTRUCTIONS:

5. IDENTIFY EACH TC TO MATCH TC SUPPORT MOUNTING
   AT POINT OF MOUNTING BY PROVIDING TYPICAL DETAIL.

6. CUTITION TO BE CONFORM TO TYPICAL DETAIL DRAWING
   SHEET BASED ON MEETING SAFETY REQUIREMENTS

7. INSTALL ALL CASES TO AS SHOWN PER THE SUPPORT (81-A)
   BRACKETS. (A, INTERIOR OR CENTER BRACKETS NOT AVAILABLE AS SHOWN)

8. DRAWING SYMBOLS PER ASA STANDARDS.

9. DETAIL A

10. 90 DEG TYPICAL