Final Report

September 1974

Space Tug Thermal Control
MCR-74-147
Contract NAS8-29670

Final Report

September 1974

SPACE TUG
THERMAL
CONTROL

Prepared for:
National Aeronautics and Space Administration
George C. Marshall Space Flight Center
Marshall Space Flight Center,
Alabama 35812

Approved

TerrL. Ward
Program Manager

MARTIN MARIETTA CORPORATION
P.O. Box 179
Denver, Colorado 80201
This document is the Final Report submitted by the Martin Marietta Corporation, Denver Division, under Contract NAS8-29670.

This study was performed for the National Aeronautics and Space Administrations George C. Marshall Space Flight Center under the technical direction of the Astronautics Laboratory, Thermal Engineering Branch, with Mr. Jack D. Loose serving as Technical Monitor. The work described herein was performed from 1 July 1973 to 30 April 1974.

The work of the following major contributors to the study is acknowledged: J. Michael Connolly and Solomon H. Eichenbaum.
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The introduction of a full capability Tug into the Shuttle mission spectrum in the 1980s will significantly broaden Shuttle's capability. To fully realize that capability it will be essential that the Tug be designed to perform its mission within a broad range of thermal environments with currently planned mission durations up to 7 days. The primary objective of this study was to develop a thermal design for the forward and intertank compartments and fuel cell heat rejection system that satisfy Tug requirements for low inclination geosynchronous deploy and retrieve missions. Key to this design was to evolve to a system that was reusable and minimized ground refurbishment requirements. Figure 1-1 presents baseline Tug configuration used in the study.

Passive concepts were demonstrated analytically for both the forward and intertank compartments. Each compartment used an external paint pattern tailored to the mission environments. The forward compartment, which contains the majority of the avionics equipment, was thermally designed with circumferential heat pipes to reduce the wide variance of skin temperatures resulting from constant attitudes. In addition, the forward shield (beta cloth) was modified to include a multilayered insulation blanket. Results indicated that the equipment used for rendezvous and docking, such as the television, laser radar, and its associated electronics, present one of the more severe thermal control problems. The most promising solution appears to be to mount the equipment on the thermal conditioning panels. The panels can be used to reduce heater power requirements. The fuel cell electrical power subsystem required an active heat rejection concept in the form of a pumped fluid radiator. Continued development of heat pipe radiators could result in their future application to thermal control of the fuel cell.

Worst-case external heating environments were determined and used in the study. All mission phases were incorporated into study with the most significant one being the heating of the Tug in the orbiter after reentry and landing. Cargo bay purging was found to be required to maintain both operating and nonoperating equipment temperature limits.

A series of three catalogues were created to provide representative equipment data for use in the thermal study. Internal distribution of the catalogues resulted in a rather wide acceptance and a desire for additional categories of information to expand their usefulness.
Key thermal control systems derived in the study were carried an additional step to preliminary sets of design and performance specifications. Three specifications were developed covering the forward compartment thermal design, battery louvers, and fuel cell heat rejection system.

A follow-on plan was developed highlighting breadboard testing of the above key areas which were advanced to the preliminary specification phase. Tests also include a honeycomb conductivity test. In addition, several areas of analytical concern were identified that were beyond the original scope of the study.

Figure 1-1 Baseline Tug Overall Configuration
New spacecraft designs generally start with studies oriented toward satisfying mission requirements. Systems-level studies of this nature generally result in identifying performance requirements, allowable system weights, power budgets, etc. New equipment (or revised existing equipment) designed to satisfy specific requirements is inherent in each new spacecraft. After some basic studies are completed the thermal designer translates the preliminary design one step further to evolve the design into thermal environments and anticipated equipment temperatures. Often the thermal designer is faced with new equipment and associated thermal data are lacking. To avoid this problem, this study began by identifying the thermal requirements, characteristics, and constraints of candidate equipment items.

The approach chosen to identify, handle, and document these data was to develop a generalized data bank containing thermal and general information for each component catalogued. The data bank was written to be dynamic in nature, allowing components to be added or deleted without affecting output of other components. A FORTRAN IV program containing four major subroutines was written to compile two catalogues using the data bank data as input data. The two catalogues contain equipment thermal requirements, and equipment physical characteristics and constraints, respectively. The data bank, catalogues, and a catalogue user's guide were published in two documents, (Ref 1 and 2).

The program and data bank provide the user a means of cataloging components for potential application to Tug or any other spacecraft in a standardized manner, while maintaining visibility to the source of the information. The data bank was organized by major system (such as the Avionics System), describing each subsystem followed by the components included within each subsystem. Table 2-1 describes the data that were catalogued and the reference used in identifying the subsystem descriptive information. Table 2-2 describes the subsystems included within the Avionics System, while Table 2-3 expands upon the Guidance Navigation and Control Subsystem describing the types of equipment, requirements, timelines, and notes. Table 2-4 presents the first component catalogued and shows the generalized and standard format used in cataloging all components.
THE SPACE TUG EQUIPMENT DATA BANK HAS BEEN PREPARED FOR NASA/MSFC UNDER CONTRACT NUMBER NAS 8-29670.

THIS DOCUMENT CONTAINS THE RAW DATA OF ALL EQUIPMENT ITEMS IDENTIFIED FOR POTENTIAL APPLICATION TO THE SPACE TUG SYSTEM.

THE FOLLOWING DATA IS INCLUDED IN THIS DOCUMENT

- EQUIPMENT THERMAL REQUIREMENTS
- EQUIPMENT PHYSICAL CHARACTERISTICS
- EQUIPMENT CONSTRAINTS

THIS DOCUMENT WAS PREPARED BY THE MARTIN MARIETTA AEROSPACE CORPORATION AND WAS SUBMITTED TO NASA/MSFC ON 1 MAY 1974.

QUESTIONS CONCERNING THE DATA CONTAINED HEREIN SHOULD BE DIRECTED TO

MR. TERRY L. WARD
PHONE 303-794-5211
EXTENSION 4702

THE SYSTEMS AND SUBSYSTEMS DESCRIBED HEREIN ARE DEFINED BY AND IN ACCORDANCE WITH

BASELINE TUG DEFINITION DOCUMENT
REVISION A
DATED JUNE 26, 1972
RELEASED BY
PRELIMINARY DESIGN OFFICE
PROGRAM DEVELOPMENT
GEORGE C. MARSHALL SPACE FLIGHT CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
TABLE 2-2

AVIONICS SYSTEM

THE DATA CONTAINED IN THE AVIONICS SYSTEM SECTION PERTAINS TO THOSE CANDIDATE EQUIPMENT ITEMS WHICH HAVE BEEN IDENTIFIED FOR APPLICATION TO THE FOLLOWING SUBSYSTEMS

GUIDANCE, NAVIGATION AND CONTROL
DATA MANAGEMENT
COMMUNICATIONS
INSTRUMENTATION
ELECTRICAL POWER
TABLE 2-3

GUIDANCE NAVIGATION AND CONTROL SUBSYSTEM

<table>
<thead>
<tr>
<th>ITEM</th>
<th>EQUIPMENT</th>
<th>QUANTITY</th>
<th>WEIGHT (POUNDS)</th>
<th>POWER (WATTS)</th>
<th>REMARKS</th>
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</thead>
<tbody>
<tr>
<td>IMU</td>
<td>2</td>
<td>80</td>
<td>40</td>
<td></td>
<td>MOUNTED AT POSITION 1 WITH STAR TRACKER.</td>
</tr>
<tr>
<td>STAR TRACKER</td>
<td>2</td>
<td>50</td>
<td>10</td>
<td></td>
<td>POSITION 1</td>
</tr>
<tr>
<td>ELECTRONICS</td>
<td>2</td>
<td>26</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HORIZON SCANNER</td>
<td>2</td>
<td>70</td>
<td>30</td>
<td></td>
<td>POSITION 3 POSSIBLY DEPLOYED.</td>
</tr>
<tr>
<td>ELECTRONICS</td>
<td>2</td>
<td>10</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LASER RADAR (A)</td>
<td>2</td>
<td>70</td>
<td>155</td>
<td></td>
<td>POSITION 2. M/3 POSITION</td>
</tr>
<tr>
<td>ELECTRONICS (A)</td>
<td>2</td>
<td>20</td>
<td></td>
<td>MIRROR YAG</td>
<td></td>
</tr>
<tr>
<td>TELEVISION (A)</td>
<td>2</td>
<td>20</td>
<td>10</td>
<td>POSITION 2° FORWARD LOOKING</td>
<td>ZOOM. ONE GIMBAL.</td>
</tr>
<tr>
<td>ACS ELECTRONICS</td>
<td>2</td>
<td>26</td>
<td>10.5</td>
<td></td>
<td>MOUNTED ON EXTERIOR AT POSITION 2 AND 4</td>
</tr>
<tr>
<td>SUN SENSOR</td>
<td>2</td>
<td>0.0</td>
<td>0.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

TOTALS

372.8 279.5

NOTES (A) INCLUDED IN RENDEZVOUS AND DOCKING CATEGORY OF MASS PROPERTIES.

TIMELINES

CONTINUOUS OPERATION

IMU

ACS

HORIZON SCANNER

STAR TRACKER

SUN SENSOR

15.3 TO 16.06, 18.45 TO 19.20, 23.40 TO 24.25
36.60 TO 37.35, 60.60 TO 61.35, 82.28 TO 83.03
87.56 TO 88.29, 98.59 TO 99.34

LASER RADAR

TELEVISION

60.35 TO 61.35

AUTO COLLIMATOR WAS EXCLUDED FROM CATALOG SINCE IT APPEARS THAT HORIZON SCANNER CAN BE ATTACHED DIRECTLY TO IMU THERE BY AVOIDING THE NEED FOR THE AUTO COLLIMATOR.

RATE GYROS WHERE INCLUDED IN CATALOG HOWEVER NO FIRM REQUIREMENT HAS BEEN ESTABLISHED.
TABLE 24

SPACE TUG EQUIPMENT DATA BANK  RAW DATA
THERMAL REQUIREMENTS, PHYSICAL CHARACTERISTICS, AND CONSTRAINTS

AVIONICS SYSTEM
GUIDANCE NAVIGATION AND CONTROL SUBSYSTEM

IMU 1 CAROUSEL 9B  DELCO ELECTRONICS  P/N 7886091-011

DESIGN OPERATING CASE TEMPERATURE  60° TO 115° DEG. F
NON-OPERATING AND STORAGE CASE TEMPERATURE  -35° TO 160° DEG. F
ACCEPTANCE TEST TEMPERATURE REQUIREMENTS  57° TO 115° DEG. F
QUALIFICATION TEST TEMPERATURE REQUIREMENTS  56° TO 118° DEG. F

PACKAGE SHAPE  RECTANGULAR
PACKAGE SIZE  LENGTH 22.7"  WIDTH 11.6  HEIGHT 12.0 (INCHES)
CASE MATERIAL  ALUMINIUM
CASE WEIGHT  20.0  POUNDS
TOTAL WEIGHT  80.0  POUNDS

SURFACE PROPERTIES  ALPHA = 0.900  EMISSIVITY = 0.900

INPUT STEADY STATE POWER  95.0  WATTS
21. AT 75° DEG. 94. AT -80° DEG. (WATTS AT DEG. FAHRENHEIT)
OUTPUT POWER  0.0  WATTS  MILLI-WATT OUTPUT

THERMAL DESIGN  ACTIVE  PASSIVE

PHYSICAL CHARACTERISTICS AND CONSTRAINTS REMARKS
NON MISSION ON-TIMES  PRELAUNCH YES  ASCENT YES  REENTRY OFF
MISSION ON-TIMES  SHUT/TUG ON  TUG/ORBIT ON  TUG/PAY ON
MAHRED WITH MAGIC 352 COMPUTER
MOUNT WITH Z-AXIS ALONG LONGITUDINAL AXIS
MAX CABLE LENGTH 1.8 METERS (6.0 FEET)
QUALIFIED FOR 9 HOUR MISSION
OPERATIONAL IN 8 HOURS

THE CAROUSEL 9B IMU IS DESIGNED AND BUILT BY
DELCO ELECTRONICS DIVISION OF GENERAL MOTORS CORPORATION
6767 HOLISTER AVE., GOLTA, CALIFORNIA  93017

THE DATA CONTAINED HEREIN WAS OBTAINED FROM
MR. BILL CATTI  PHONE 805-968-1011 EXTENSION 623
THE IMU IS CURRENTLY IN A PRODUCTION PHASE AND IS BEING
PROCURED BY SAMSO FOR USE ON THE TITAN 3C TRANSTAGE AS THE SINGLE
GUIDANCE SENSOR FOR THIS SYSTEM IT IS MARRIED TO THE MAGIC 352
COMPUTER ALSO BUILT BY DELCO AND SUPPLIED AS A TWO PACKAGE SYSTEM.
THE IMU IS A 4 GIMBAL SYSTEM AND IS QUALIFIED FOR A 9 HOUR MISSION
THE IMU IS SCHEDULED TO FLY FOR THE FIRST TIME IN 1973. A SINGLE
28 VDC SOURCE IS REQUIRED INTERCONNECTING CABLE WITH THE COMPUTER
IS LIMITED TO 1.8 M (6 FT). THE GIMBAL SET IS INHERENTLY SHOCK
MOUNTED, THE CASE IS PRESSURIZED TO 11.7 N/CM SQ (17 PSIA) AND THE
UNIT IS DESIGNED WITH AN INTERNAL ACTIVE THERMAL CONTROL SYSTEM
COMPRISED OF A FAN AND THERMOSTATICALLY CONTROLLED HEATERS. THE UNIT
IS DESIGNED TO FUNCTION WITHIN A MAXIMUM POWER BUDGET OF 205 WATTS.
APPROXIMATELY 8 HOURS ARE REQUIRED FROM POWER ON TO GO-INERTIAL

REF. BROCHURE, UNIVERSAL SPACE GUIDANCE SYSTEM, DELCO ELECTRONICS

END
Each component was catalogued in raw data form, identifying the appropriate system and subsystem. Preprinted keypunch sheets were used to reduce the amount of information to be written and correspondingly prepunched cards were used to reduce the key-punch task. This also limited the number of errors found in the review and editing of each component data sheet. One additional means of reducing errors was also applied. The data were assembled in the familiar set units and the program was used to convert the data to the International Units as shown in Table 2-5, the final data form.

Three major blocks of information were set aside for describing each component as shown separated by asterisk lines. The first block describes the component identifier (used by the program), name, manufacture, and part number. The remaining data in this block describe pertinent thermal design information of the component. Operating, nonoperating, and test box temperature limits are presented. The box shape and size, case material, and weights are specified. The exterior surface radiation properties, input power, variable power, and output power are presented. The last item describes the basic box thermal design for ground and flight operations. The word "active" to the left of the asterisk refers to a need of forced air cooling or a fluid loop on the ground, while "passive" refers to no special considerations required. The word "active" to the right of the asterisk refers to the need of special considerations in flight such as a fluid loop or other means beyond the mounting conduction and radiation capability of the box.

The second block of data contains information relative to the required on-times during the mission and pertinent characteristics and constraints remarks. The prelaunch, ascent, and reentry periods of flight were described as nonmission periods of flight because the Tug is attached to the Shuttle during these periods.

The third data block was set aside as a general narrative block to further identify the manufacturer, source of the material, expand the description of the component, development status, etc.

The first two data blocks were used by the program to build the two catalogues required by contract. The first catalogue, the Equipment Thermal Requirements Catalogue, is a summary of the data bank information in terms of allowable component temperatures as they relate to the various Tug mission phases. This summary was organized by subsystem and type of component as shown in Table 2-6. In addition, the thermal design and power dissipation are also presented. "Yes" was used to indicate that the component is on during mission phases while the Tug is attached to Shuttle, but not required to satisfy Tug mission requirements. "Int" indicates an intermittent usage during the mission.
TABLE 2-5
SPACE TUG EQUIPMENT DATA BANK FINAL DATA
THERMAL REQUIREMENTS, PHYSICAL CHARACTERISTICS AND CONSTRAINTS

AVIONICS SYSTEM
GUIDANCE NAVIGATION AND CONTROL SUBSYSTEM

---

IMU 1 CAROUSEL 5B DELCO ELECTRONICS

<table>
<thead>
<tr>
<th>DESIGN OPERATING CASE TEMPERATURE</th>
<th>289. TO 319. DEG. K</th>
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<tr>
<td>NON-OPERATING AND STORAGE CASE TEMPERATURE</td>
<td>236. TO 344. DEG. K</td>
</tr>
<tr>
<td>ACCEPTANCE TEST TEMPERATURE REQUIREMENTS</td>
<td>287. TO 319. DEG. K</td>
</tr>
<tr>
<td>QUALIFICATION TEST TEMPERATURE REQUIREMENTS</td>
<td>286. TO 321. DEG. K</td>
</tr>
</tbody>
</table>

PACKAGE SHAPE: RECTANGULAR
PACKAGE SIZE

<table>
<thead>
<tr>
<th>LENGTH (CM)</th>
<th>WIDTH (CM)</th>
<th>HEIGHT (CM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>57.7</td>
<td>27.9</td>
<td>30.5</td>
</tr>
</tbody>
</table>

PACKAGE AREA: 944.0 SQ. INCHES
PACKAGE VOLUME: 4910.2 CU. INCHES

CASE MATERIAL: ALUMINUM
CASE WEIGHT: 9.1 KILOGRAMS
TOTAL WEIGHT: 36.3 KILOGRAMS

SURFACE PROPERTIES

<table>
<thead>
<tr>
<th>ALPHA</th>
<th>EMISSIVITY</th>
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<tr>
<td>0.900</td>
<td>0.900</td>
</tr>
</tbody>
</table>

INPUT STEADY STATE POWER: 95. WATTS
OUTPUT POWER: 0. WATTS
THERMAL DESIGN: ACTIVE

PHYSICAL CHARACTERISTICS AND CONSTRAINTS

NON MISSION ON-TIMES: *PRELAUNCH* YES
ASCENT: YES
REENTRY OFF
MISSION ON-TIMES: *SHUT/TUG ON* TUG/ORBIT ON* TUG/PAY ON
MARRIED WITH MAGIC 352 COMPUTER
MOUNT WITH Z-AXIS ALONG LONGITUDINAL AXIS
MAX CABLE LENGTH: 1.8 METERS (6.0 FEET)
QUALIFIED FOR 9 HOUR MISSION
OPERATIONAL IN 8 HOURS

THE CAROUSEL 5B IMU IS DESIGNED AND BUILT BY
DELCO ELECTRONICS DIVISION OF GENERAL MOTORS CORPORATION
6767 HOLISTER AVE., GOLTA, CALIFORNIA 93017

THE DATA CONTAINED HEREIN WAS OBTAINED FROM
MR. BILL CATTOI
PHONE 805-968-1011 EXTENSION 623

THIS IMU IS CURRENTLY IN A PRODUCTION PHASE AND IS BEING
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UNIT IS DESIGNED WITH INTERNAL STEADY STATE THERMAL CONTROL SYSTEM
COMPRISED OF A FAN AND THERMOSTATICALLY CONTROLLED HEATERS. THE UNIT
IS DESIGNED TO FUNCTION WITHIN A MAXIMUM POWER BUDGET OF 205 WATTS.
APPROXIMATELY 8 HOURS ARE REQUIRED FROM POWER ON TO GO-INERTIAL.

REF. BROCHURE: UNIVERSAL SPACE GUIDANCE SYSTEM* DELCO ELECTRONICS

2-7
### TABLE 2-6

#### EQUIPMENT THERMAL REQUIREMENTS CATALOGUE

**GUIDANCE NAVIGATION AND CONTROL SUBSYSTEM**

**EQUIPMENT ITEM: STAR TRACKERS**

<table>
<thead>
<tr>
<th>NO.</th>
<th>DESCRIPTION AND MANUFACTURE</th>
<th>THERMAL DESIGN</th>
<th>POWER</th>
<th>MISSION PHASE</th>
<th>THERMAL REQUIREMENTS AND TEMPERATURE LIMITS</th>
<th>REMARKS</th>
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<td></td>
<td></td>
<td>WATS</td>
<td>DEGREES KELVIN / (FAHRENHEIT) - MIN / MAX</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>GROUND/ ORBITAL MIN/ MAX</td>
<td>CARRY</td>
<td>SHUTTLE</td>
<td>MANEUVERS</td>
<td>PAYLOAD AND TUG ORBITAL TUG</td>
</tr>
<tr>
<td>ST 3</td>
<td>B400/ PASSIVE</td>
<td>28/ OFF OFF OFF</td>
<td>253/323 253/323 253/323 253/323</td>
<td>253/323 253/323 253/323 253/323</td>
<td>253/323 253/323</td>
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</tr>
<tr>
<td>ST 4</td>
<td>5698 STAR TRACKER/ EMPE PHOTOELECTRIC PASSIVE</td>
<td>3/ OFF OFF OFF</td>
<td>218/348 218/348 218/348 218/348</td>
<td>218/348 218/348 218/348 218/348</td>
<td>218/348 218/348</td>
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<td>ST 5</td>
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<td>4/ OFF OFF INT INT</td>
<td>210/130 210/130 210/130 210/130</td>
<td>210/130 210/130 210/130 210/130</td>
<td>210/130 210/130</td>
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<td>ST 7</td>
<td>NIA ATM STAR TRKF. PASSIVE/ RENDIX CORPORATION PASSIVE</td>
<td>8/ OFF OFF INT INT</td>
<td>233/327 233/327 233/327 233/327</td>
<td>233/327 233/327 233/327 233/327</td>
<td>233/327 233/327</td>
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</tr>
</tbody>
</table>
The second catalogue, Equipment Physical Characteristics and Constraints Catalogue, presents the thermal characteristics of the components as derived from the data contained in the first data block and constraints remarks from the second block. Surface area and volume, power density, radiation time constant, adiabatic rise rate, thermal mass, and allowable sink temperature are presented. The data are presented in International units and English units. Some of the components were unable to meet their temperature limits in a 100% radiation environment, hence, the quantity of heat required to be removed via conduction was calculated and printed if the sink environment requirements were less than 0°C. Within limits, the use or need of conduction to cool a component is usually an open issue for the thermal designer. Hence, the results indicate emphasis to be placed on a given component and the potential need for special considerations such as the use of heat pipes. Table 2-7 presents an example of the catalogue.

The catalogues proved to be a valuable asset during the study. We used various groups within the Denver Division to test the applicability of the data to other disciplines and projects and found a general acceptance and desire for additional data to be included. In general, the data in the catalogues were complete within the intended scope, however, several areas for expansion are apparent. For example, each component designer in the aerospace industry compiles component information relative to the needs of his particular technical discipline, but it is rarely a complete compilation of information. The data bank approach could easily be expanded to include the functional characteristics and requirements of the components tailored to meet specific component types and a complete description of testing and test requirements. The resultant catalogues would be extremely useful to the aerospace industry and would reduce the time required by those who attempt to maintain component files while limiting the amount of misinformation that is passed along by word of mouth. Follow-on work in this area is desirable and appropriate with direct benefits to the government.
### Equipment Physical Characteristics and Constraints Catalogue

#### Guidance Navigation and Control Subsystem

#### Equipment Item: Star Trackers

<table>
<thead>
<tr>
<th>REF.</th>
<th>DESCRIPTION</th>
<th>WEIGHT</th>
<th>PACKAGE</th>
<th>SURFACE</th>
<th>VOLUME</th>
<th>RAD.</th>
<th>POWER</th>
<th>POWER</th>
<th>TIME</th>
<th>ADIABATIC</th>
<th>THERMAL</th>
<th>ALLOWABLE</th>
<th>SINK</th>
<th>OPERATION</th>
<th>NO.</th>
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<th>REMARKS</th>
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<td>862</td>
<td>.70/</td>
<td>6/</td>
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<td>1.09</td>
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<td>1</td>
<td>5.3</td>
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<td>INT</td>
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<td>19.1</td>
<td>RECT</td>
<td>11211</td>
<td>77677</td>
<td>.25/</td>
<td>10/</td>
<td>16/</td>
<td>25</td>
<td>.62</td>
<td>1</td>
<td>2</td>
<td>12.9</td>
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<td>302</td>
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<td>3</td>
<td>6</td>
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<td>247</td>
<td>280</td>
<td>226</td>
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Essential to the thermal analysis of the Space Tug and its associated equipment is an adequate definition of the expected environments to be encountered by the Tug. Many environments had to be evaluated as to their impact on the thermal design of the Tug vehicle. Both minimum and maximum heating conditions were defined. An environments timeline was generated in accordance with a major events timeline given in Table 3-1 and used for the transient mission analysis.

The thermal environments used early in the study to determine worst-case environments are summarized in Table 3-2. These environments were generated using the Tug flux model shown in Figure 3-1. The maximum on-orbit heating condition occurs in the Case 4 park orbit shown in Table 3-2. The planetary and albedo heating contributions of the park orbit and the vehicle's solar orientation make this case’s heating slightly higher than other cases considered. Also from the environments study, the minimum heating condition occurs in the Case 7 geosynchronous orbit. The minimal planetary heating in the shadow portion of the orbit led to this case being selected to evaluate cold conditions using the steady-state sink temperature model.

In addition to the hot and cold environments used in the steady-state model, additional environments were needed for the initial orbital insertion and transfer to park orbit for the mission analysis transient model. From liftoff to cargo bay door opening, the cargo bay temperature was assumed to be constant at 294°K (70°F) for the first 10 minutes and was then increased to 80°F in a linear manner to 300°K (80°F) at 0.533 hours per Reference 3. A worst-case hot environment was simulated with the Tug in the orbiter cargo bay with the radiator doors deployed with the orbiter Z-axis solar oriented as shown in Figure 3-2.

The environments timeline used in the transient mission analysis is described in Table 3-3. These environments were input to the model for the mission simulation in the form of array tables. The launch and landing environments were simulated by driving the orbiter cargo bay liner and radiator door temperatures to the values taken from Reference 3. The reentry temperatures are shown in Figure 3-3. These temperatures represent a worst-case maximum heating condition with an assumed adiabatic payload in the cargo bay.
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<th>DURATION (HOURS)</th>
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<td>.05</td>
<td>OPEN CARGO BAY DOORS AND DEPLOY SHUTTLE RADIATORS</td>
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<td>.0333</td>
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<td>.917</td>
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<td>DURATION (HOURS)</td>
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<td>-------------</td>
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INJECT INTO RETURN PHASING ORBIT
COAST 1 REV. IN PHASING ORBIT
CIRCULARIZE INTO 315 KM (170 N. MILE) ORBIT
28.5° INCLINATION
ORBIT TRIM
TERMINAL PHASE INITIATION AND TUG CAPTURE
SEARCH AND ACQUISITION OF TUG BY ORBITER
VENT TUG MAIN TANKS AND CLOSE VENTS
COELLIPTIC WINDOW
CONTROL OF TUG TRANSFERRED TO CREW
PLANE CHANGE WINDOW
ORBITER TPI BURN AND COAST
ORBITER TPF BURN
ORBITER COAST TO AND ARRIVAL AT CAPTURE POSITION
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<td>98.75</td>
<td>.0667</td>
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<td>.05</td>
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### Table 3-2 Tug Natural Environments Case Summary

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Figure 3-1 Tug Flux Model Y-Axis View

Figure 3-2 Predeployment Flux Model 3-D View
Table 3-3 Tug/Orbiter Mission Environments

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<th>DESCRIPTION</th>
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<td>160 NM Circular</td>
<td>2.805 to 4.310</td>
<td>Same as above (1 orbit)*</td>
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<td>160 NM Circular</td>
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<td>Tug deployed - orbiter continues in circular orbit until 98.917 hours - fluxes from Case 4 park orbit.*</td>
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<td>160 x 19300 NM Transfer</td>
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<tr>
<td>19300 NM Circular</td>
<td>24.350 to 84.353</td>
<td>Tug at geosynchronous fluxes from Case 7 geosynchronous (2.5 orbits).*</td>
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<td>19300 x 160 Transfer</td>
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<td>Radiator doors closed, cargo bay wall environments being boundary temperatures.</td>
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*Incident orbital fluxes calculated with vehicle x-axis perpendicular to sun vector for the hot case (see Table 3-2).
Figure 3-3 Boundary Temperatures Used for Landing Environments
4. STEADY-STATE PARAMETRIC STUDIES

Studies were performed to evaluate the influence of various parameters on the thermal design of Tug. These studies were essential in assuring adequate thermal performance of the vehicle throughout its mission and were concerned with both active and passive means of providing thermal control to the Tug and its associated equipment. The studies relied heavily on minimum and maximum heating environments. The areas investigated as part of the study are tabulated in the order they occurred in Table 4-1. A description of each thermal model that was developed and the particular studies that it was used for is discussed in the following sections. The results of each of the studies are also presented.

Table 4-1 Parametric Studies Performed

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<tr>
<td>Thermal Control Coatings</td>
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<td>Forward Compartment Heat Pipe</td>
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<td>Honeycomb Wall Structure Conductance</td>
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<td>Component Contact Conductance</td>
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<td>Component Heater Sizing</td>
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<td>Transient Mission Analysis</td>
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<td>Simplified Louver System Operations</td>
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</table>

The parametric studies began early in the program with the development of a steady-state MITAS (Ref 4) thermal math model to generate compartmental sink temperatures. This model consists of 34 nodes as shown in Figure 4-1. The Tug compartment, tank insulation, and engine are simulated by 31 arithmetic nodes (zero mass nodes) and the boundaries consisting of the LH$_2$ node, the LOX node, and space. There were 117 radiation conductors and 12 linear conductors. Radiation conductors were calculated by the model from the configuration factors and node optical properties data with the use of the SCRPFA subroutine. Also, the absorbed environmental fluxes were calculated within the model from the incident flux tables and the surface optical properties. This technique allowed for parametric variation of the surface optical properties to investigate their influence on compartment sink temperatures. The maximum and minimum incident heating conditions from Table 3-2 for Case 4 park orbit and Case 7 geosynchronous orbit, respectively, were used in the model.
Figure 4-1 Compartmental Sink Temperature Model
4.1 INITIAL COATING STUDIES

Tradeoff studies to select the external surface coatings were performed using the hot and cold environmental heating rates. The optical coating parameters $a$ and $e$ were varied along with the compartmental average power dissipation.

Figure 4-2 presents the hot-case average radiation sink temperature as a function of optical properties and selected power dissipations for the forward compartment. The specific optical property ratios used to generate the curves correspond to white paint ($a/e = 0.2/0.9$), aluminum paint ($a/e = 0.26/0.26$), and a 50% mixture of white and aluminum paint ($a/e = 0.23/0.58$). Forward compartment average sink temperature data are presented in Figure 4-3 for the shadow portion of Case 4 park orbit to show the effect of coating emissivity.

The same parametric runs were repeated using the cold-case environments and the results are presented in Figures 4-4 and 4-5 for the sun and shadow portions of the orbit, respectively.

4.2 INSULATION AND COATING SELECTION

Figure 4-5 indicates that coatings by themselves will be inadequate to maintain thermal control. This is based on maintaining the forward compartment average sink temperature above a minimum of 200$^\circ$K ($-100^\circ$F). This criterion (200$^\circ$K) was chosen based on past experience on a similar system and a survey of minimum temperatures obtained from Reference 2. Before pursuing coating selection further, an investigation of vehicle heat leaks was conducted in an effort to raise compartmental sink temperatures. It was found that a significant heat leak existed at the forward compartment beta cloth shield. By using a 24-layer Mylar insulation blanket with gold on one side of each Mylar sheet, the effective emissivity across the blanket was reduced to 0.025 per Reference 5. Using the insulation, the forward compartment heat leak was reduced to a point where selective coatings were adequate in controlling internal compartment sink temperatures.

The hot and cold cases were reanalyzed using the multilayer insulation blanket and the results are shown in Figures 4-6 thru 4-8. Figure 4-6 presents the forward compartment maximum sink temperatures versus $a/e$ ratio and shows the influence of the insulation blanket. Figure 4-7 shows similar results for the sun portion of Case 7 geosynchronous orbit. Minimum forward compartment sink temperatures are shown in Figure 4-8 for shadow portion of the Case 7 geosynchronous orbit. This curve shows an emittance of 0.475 which gives the desired minimum operating sink temperature 200$^\circ$K ($-100^\circ$F) for nominal power dissipations of 800 to 1000 watts.
Figure 4-2
Parametric Runs, Hot Case Compartment Average Sink Temperature Park Orbit Case 4 (Full Sun)
Figure 4-3
Parametric Runs, Forward Compartment Average Sink Temperature, Park Orbit Case 4 Earth Shadow Temperatures
Figure 4-4
Parametric Runs, Compartment Average Sink Temperature, Synchronous Orbit Case 7 (Full Sun)
Figure 4-5
Parametric Runs, Compartment Average Sink Temperature, Synchronous Orbit Case 7, Earth Shadow Temperatures
Figure 4-6
Parametric Runs, Forward Compartment Average Sink Temperature, Park Orbit Case 4
Figure 4-7
Parametric Runs, Forward Compartment Average Sink Temperature, Synchronous Orbit Case 7
Figure 4-8
Parametric Runs, Forward Compartment Average Sink Temperature, Synchronous Orbit Case 7
Establishing a maximum sink temperature of 297°K (75°F), from Figure 4-6 dictates an $a/e$ value of 0.50. The previous emittance value of 0.475 fixes an $a$ value of 0.2375. A similar analysis on the intertank compartment indicated an $a/e$ value of 0.60 was needed with $a = 0.246$ and $e = 0.41$.

The paint pattern needed to simulate the necessary optical properties is derived from Figure 4-9. The $a$ and $e$ for all-white paint and all-aluminum paint are plotted on the left and right abscissas, respectively, and connected by straight lines. Finding the optical property on the graph fixes the percentages of aluminum to white paint needed for a mosaic pattern. For the forward compartment 63.5% aluminum paint and 36.5% white paint is needed, and for the intertank compartment 75% aluminum paint and 25% white paint is needed.

4.3 FORWARD COMPARTMENT HEAT PIPES

Upon completion of the thermal coating studies, heat pipes were simulated in the forward compartment to isothermalize the compartment walls. This was necessary because hot-case wall temperature gradients in excess of 72°K (130°F) existed between the sun and shadowed side of the vehicle. The average compartment sink temperature was unaffected by the heat pipes as shown in Figures 4-10 and 4-11. These curves compare directly with those of the coating study, Figures 4-6 and 4-8. Heat pipe performance data for a typical high capacity heat pipe was taken from Reference 6. The pipe operates at a 2 kW load over a temperature drop of 3.89°K (7°F). Based on the performance data, six parallel circumferential heat pipes were integrated into the compartment walls for simulation in the model.

Using a fin effectiveness of 0.85 and a joint conductance of $12.1 \frac{W}{m^2 \cdot ^\circ K}$ (800 Btu/hr-°F-ft$^2$) Reference 7, a conductance value of 467 W/°K (2870 Btu/hr-°F) was calculated between each wall and each heat pipe node. The heat pipe performance data were reduced to an effective conductance between each heat pipe node of 879 W/°K (5400 Btu/hr-°F). The large heat pipe conductance caused oscillations when running the math model, resulting in excessive machine time for temperature convergence. A more efficient technique was then employed that replaced the original heat pipe nodes and network with an equivalent series network connecting adjacent compartment wall nodes with a conductance of 184.6 W/°K (1134 Btu/hr-°F). A reduction in the number of iterations was also achieved by first solving the network without the heat pipes and calculating a fourth power temperature average of the wall nodes. This temperature was applied to the wall nodes as starting wall temperatures for the heat pipe simulation.
Figure 4-9 Parametric Runs, Surface Properties
HEAT PIPES IN FORWARD COMPARTMENT

BETA CLOTH PLUS 24 LAYERS OF INSULATION

Figure 4-10
Parametric Runs, Forward Compartment Average Sink Temperatures Park Orbit Case 4
HEAT PIPES IN FORWARD COMPARTMENT

BETA CLOTH PLUS 24 LAYERS OF INSULATION

EARTH SHADOW

Figure 4-11
Parametric Runs, Forward Compartment Average Sink Temperatures, Synchronous Orbit Case 7
The effectiveness of the heat pipe in reducing circumferential gradients is shown in Figures 4-12 and 4-13 for Case 7 geosynchronous and Case 4 park orbits, respectively. The forward compartment wall temperature gradient is reduced from 50 to 2.8°K in geosynchronous orbit and from 36 to 2.2°K in park orbit.

4.4 HONEYCOMB STUDIES

A study was performed to determine the influence of the honeycomb structure on compartmental temperatures. A duplicate set of forward compartment wall nodes were added to the model simulating the fiberglass epoxy, aluminum core honeycomb structure. Figures 4-14 and 4-15 show the influence of the honeycomb conductance on the forward internal sink temperature for the hot and cold cases, respectively. The ΔTs from the above curves should be added to Figures 4-6 and 4-8, respectively, to obtain the internal sink temperatures for the honeycomb structure. The maximum conductance value of 1 watt/°K (1.94 but/°F) per 0.093 m² (1 ft²) results in a compartment sink temperature 3.3°K (6°F) warmer than no honeycomb for the hot case. The conductance value was obtained assuming an infinite value for the joint conductances. A more realistic value for the joint conductances would result in lower overall conductance values, thus increasing the effect on compartment sink temperatures. The use of a nonmetallic core, such as fiberglass would further increase the ΔT by reducing the conductivity as shown in the curves. Hence, the choice of the honeycomb structure for Tug will have an influence on the thermal design and could impact the basic passive concept chosen. A further discussion of the honeycomb structure is included in Section 7.
Figure 4-12
Forward Compartment Wall Temperatures, Geosynchronous (Vehicle in Sun)

Figure 4-13
Forward Compartment Wall Temperature, Hot Case (Vehicle in Sun)
Figure 4-14
Effect of Honeycomb Conductance on Compartment Sink Temperature, Hot Case

Power = 800 Watts
\( a/\varepsilon = 0.238/0.475 \)
\( \Delta T = T_{\text{Sink (with honeycomb)}} - T_{\text{Sink (no honeycomb)}} \)
Power = 800 Watts
\(\alpha / \epsilon = 0.238/0.475\)
\(\Delta T = T_{\text{Sink (with honeycomb)}} - T_{\text{Sink (no honeycomb)}}\)

Figure 4-15
Effect of Honeycomb Conductance on Compartment Sink Temperature, Cold Case
5. TRANSIENT ANALYSIS

5.1 MODEL DESCRIPTION AND ASSUMPTIONS

A transient mission model was constructed to simulate an actual Tug mission from liftoff through landing and subsequent cooldown. This model was used to predict individual component temperature histories along with the structural temperatures of the Tug vehicle. The model incorporated the thermal control features resulting from the previous studies using the steady-state sink temperature model. These features include the use of heat pipes in the forward compartment, multilayer insulation on the forward compartment beta cloth shield, and the external paint pattern determined from the optical properties tradeoff studies. The transient model takes both the thermal capacitance and a realistic power distribution for each component into account in arriving at temperatures.

The overall transient mission model consists of two separate submodels for the forward and intertank compartments. The forward and intertank compartment equipment is listed and described in Tables 5-1 and 5-2, respectively. Figure 5-1 is a TRASYS (Ref 9) computer plot showing the forward compartment equipment, equipment identifiers, node numbers, and their locations. An expanded rollout view of the forward compartment is shown in Figure 5-2 and top view is shown for clarity in Figure 5-3. The intertank equipment, equipment location, and node numbers are shown in Figures 5-4 and 5-5.

The radiation network for the forward compartment consisted of 214 surfaces comprised of eight forward compartment cylinder walls, the beta cloth shield, LH₂ forward dome, and 204 component surfaces. The 214 original surfaces were reduced to 44 nodes for inclusion in the thermal model. The radiation model for the intertank consists of 56 surfaces condensed into 28 nodes. These include eight interior wall nodes, LH₂ and LOX domes, and 18 equipment nodes.

The six sides of each component were used in calculating the blackbody view factors using the TRASYS program. The view factors were used to calculate the grey-body exchange factors also using TRASYS, and were then condensed to single node components using the program radiation condenser option.

Many thermal aspects of the mission analysis are common to both the forward compartment and the intertank compartment models. The time sequence of environments used is shown in Figure 5-6 and is presented in Table 3-3. The liftoff and landing environments are controlled by time varying boundary temperatures for the radiator doors and the
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<td>231, 236, 251, 256</td>
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<td>Data Management Subsystem</td>
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<td>2</td>
<td>280</td>
<td></td>
<td>Page 67</td>
<td>Grouped in pairs</td>
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<td>Electrical Power Subsystem</td>
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<td>300</td>
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<td>Battery</td>
<td>1</td>
<td>270</td>
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<td>X</td>
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<td>LH2 Sphere</td>
<td>1</td>
<td>211</td>
<td></td>
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<td>X</td>
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<td>LOX Sphere</td>
<td>1</td>
<td>716</td>
<td></td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>

*Reference 7 Baseline Tug Definition Document
Figure 5-1 Tug Forward Compartment Equipment
Figure 5-2 Tug Forward Compartment Equipment Locations
Node Numbers are Used in MITAS

Figure 5-3  Tug Forward Compartment Interior Nodes
Figure 5-4 Tug Intertank Compartment Equipment Nodes
Figure 5-5 Tug Intertank Interior Nodes

( ) Indicates Nodes On Back Side
<table>
<thead>
<tr>
<th>Event No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>185 km Circular Orbit (Tug + Orbiter)</td>
</tr>
<tr>
<td>2</td>
<td>185 km x 296 km Transfer Orbit (Tug + Orbiter)</td>
</tr>
<tr>
<td>3</td>
<td>296 km Circular Orbit (Tug + Orbiter)</td>
</tr>
<tr>
<td>4</td>
<td>296 km Circular Orbiter (Case 4 - Tug Only)</td>
</tr>
<tr>
<td>5</td>
<td>296 km x 35,800 km Transfer Orbit (Tug Only)</td>
</tr>
<tr>
<td>6</td>
<td>35,800 km Geosynchronous Orbit (Tug Only)</td>
</tr>
<tr>
<td>7</td>
<td>35,800 km x 296 km Transfer Orbit (Tug Only)</td>
</tr>
<tr>
<td>8</td>
<td>Liftoff/Landing - Cooldown (Tug + Orbiter)</td>
</tr>
</tbody>
</table>

Figure 5-6 Tug Mission Event Sequence
cargo bay liner (ref 3). All on-orbit environments consist of the natural absorbed solar, albedo, and planetary heating, and were calculated using TRASYS in conjunction with the surface optical properties that were determined from the steady-state trade-off studies.

The Tug and orbiter radiation interchange was accounted for and depends on the vehicle configuration, which follows the events timeline shown in Figure 5-7. Additionally, convection interaction between the orbiter and the Tug was accounted for at liftoff and landing. A natural convection coefficient \( h \) was calculated with the use of the following correlation from Reference 10 for a horizontal wall.

\[
N_u = 0.35 \left( \frac{G_r}{P_r} \right)^{1/4}
\]

where

\( N_u \) = Nusselt Number,

\( G_r \) = Grashof Number,

\( P_r \) = Prandtl Number.

Evaluating the properties of air at a temperature of 311°C (100°F) and assuming a constant acceleration of 2 g results in the following expression for

\[
h = K \left( \rho^2 \Delta T \right)^{1/4}
\]

where

\[
K = 0.92278 \frac{\text{watts}}{\text{meter}^2 \cdot \text{°K}} = 0.5267 \frac{\text{Btu}}{\text{hr \ ft}^2 \cdot \text{°F}}
\]

\( \rho \) = air density,

\( \Delta T \) = temperature difference between orbiter cargo bay air temperature and the Tug skin.

The air density is a function of altitude (taken from Reference 3), and input to the model as a time varying array. Also the quantity used for \( \Delta T \) assumes that the entering air will be heated to the average cargo bay temperature as it passes through the orbiter structure. The resulting \( h \) value used in the model is shown in Figure 5-8.
<table>
<thead>
<tr>
<th>Indicator No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Tug in Orbiter, Doors Closed</td>
</tr>
<tr>
<td>2</td>
<td>Tug in Orbiter, Doors Open</td>
</tr>
<tr>
<td>3</td>
<td>Tug Deployed</td>
</tr>
</tbody>
</table>

Figure 5-7 Mission Geometry Sequence
Figure 5-8 Free Convection Heat Transfer Coefficient
A circumferential heat pipe was simulated in the forward compartment similar to the heat pipe used in the steady-state model. The major difference was that the fourth power average of the eight wall node temperatures was substituted for the calculated wall node temperature at the beginning of each time step. This technique saved computer time by reducing the number of iterations needed for each transient time step.

The emergency battery used in the intertank model also included a simulated louver system as shown in Figure 5-9. The battery was modeled assuming five of the sides were insulated with an integral 5 watt thermostatically controlled heater to maintain its storage temperature at $290.3^\circ K$ ($62.5^\circ F$) $\pm 1.39^\circ K$ ($2.5^\circ F$). The base of the battery was assumed to be coupled to a louver system whose blades were fully closed at $292^\circ K$ ($65^\circ F$) and fully open at $303^\circ K$ ($85^\circ F$). The louver system radiated to the external skin of the intertank. This assumed inner honeycomb paneling was removed from the louvered area. The effective emittance of the louver system was input to the model as a function of the baseplate temperature and is shown in Figure 5-10. The battery was activated at 97.63 hours at which time 45 watts of internal energy were assumed to be generated within the battery for 0.5 hours.

The fuel cell was modeled as an insulated component that operated at a continuous boundary temperature of $356^\circ K$ ($180^\circ F$) until it is deactivated at 97.63 hours. At this time the fuel cell temperature was allowed to respond like any normal diffusion node.

A contact conductance value between the component and the mounting surface was calculated based on the number of bolted contacts assuming a $0.60 \text{ watts per } ^\circ K \left(1.13 \frac{\text{Btu}}{\text{hr}^\circ F}\right)$ conduction coupling per bolt for individual clip or rail mounts. This nominal value was taken from Reference 11 and based on aluminum bolted joints used in spacecraft application. In the final analysis, the original value had to be reduced for most of the components because the contact conductance couplings were dominating all other couplings. The component contact conductance used in the model along with other component thermal characteristics are given in Tables 5-3 and 5-5 for the forward compartment model and intertank compartment models respectively.

Transient analyses were run for two environment conditions designated "hot case" and "cold case." The hot case uses the environments time line described in Table 3-3 and shown in Figure 5-7 and the configuration time line shown in Figure 5-8. Component power dissipation cycles are indexed in Tables 5-3 thru 5-6. The hot case represents a mission consisting of a hot biased park orbit (Table 3-2, Case 4 park) and landing environment coupled with a hot geosynchronous orbit which included a cyclic shadow period (Table 3-2, case 7 geosynchronous).
Figure 5-9 Louver System/Mounting Configuration
Figure 5-10 Louver System Effective Emittance Versus Battery Baseplate Temperature
<table>
<thead>
<tr>
<th>System/Component Name</th>
<th>Node No.</th>
<th>Surface Area, m²</th>
<th>Contact Conductance, W/K</th>
<th>Thermal Mass, kgtm/hr</th>
<th>Dissipated Power, watts</th>
<th>Power Time Line (Fig.)</th>
<th>Temperature History (Fig.)</th>
<th>Heater Duty Cycle (Fig.)</th>
<th>Heater Size, watts</th>
<th>Avg Heater Power Consumption, watts</th>
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<tbody>
<tr>
<td>Guidance Navigation and Control</td>
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<td></td>
<td></td>
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<td></td>
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<tr>
<td>Inertial Meas. Unit</td>
<td>300</td>
<td>0.544</td>
<td>0.90</td>
<td>3.57</td>
<td>6.91</td>
<td>144.0*</td>
<td>(1)</td>
<td>5-46</td>
<td>N/A</td>
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<td>0.90</td>
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<td>1.16</td>
<td>5.0</td>
<td>(2)</td>
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<td>5-114</td>
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<td>0.215</td>
<td>0.90</td>
<td>1.19</td>
<td>1.16</td>
<td>5.0</td>
<td>(2)</td>
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<td>5-116</td>
<td>20</td>
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<td>Horizon Scanner</td>
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<td>0.218</td>
<td>0.90</td>
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<td>0.79</td>
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<td>5-52</td>
<td>5-118</td>
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<td>0.05</td>
<td>2.98</td>
<td>3.69</td>
<td>7.0</td>
<td>(2)</td>
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<td>5-120</td>
<td>23</td>
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<td>Laser Radar Pri</td>
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<td>7.44</td>
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<td>Computer Pri</td>
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<td>0.90</td>
<td>1.19</td>
<td>0.63</td>
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<td>(1)</td>
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<td>5-134</td>
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<td>Computer Sec</td>
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<td>0.90</td>
<td>1.19</td>
<td>0.63</td>
<td>16.0</td>
<td>(1)</td>
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<td>0.90</td>
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<td>0.90</td>
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<td>Tlmtry Frmtr Sec</td>
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<td>0.90</td>
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<td>0.63</td>
<td>7.0</td>
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<td>Data Bus Cont (Pri)</td>
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<td>0.121</td>
<td>0.90</td>
<td>1.19</td>
<td>0.63</td>
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<td>Data Bus Cont (Sec)</td>
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<td>Tape Recorder</td>
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<td>1.63</td>
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<tr>
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<td>1.79</td>
<td>0.42</td>
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<td>N/A</td>
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<td>0.122</td>
<td>0.85</td>
<td>1.79</td>
<td>0.42</td>
<td>6.2</td>
<td>(1)</td>
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<td>0.196</td>
<td>0.85</td>
<td>3.57</td>
<td>1.00</td>
<td>60.5</td>
<td>(1)</td>
<td>5-92</td>
<td>N/A</td>
<td>N/A</td>
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<tr>
<td>Transmitter, PM Sec</td>
<td>460</td>
<td>0.196</td>
<td>0.85</td>
<td>3.57</td>
<td>1.00</td>
<td>60.5</td>
<td>(1)</td>
<td>5-94</td>
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<tr>
<td>Decoder Pri</td>
<td>470</td>
<td>0.060</td>
<td>0.10</td>
<td>1.19</td>
<td>0.32</td>
<td>2.8</td>
<td>(1)</td>
<td>5-96</td>
<td>N/A</td>
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<tr>
<td>Decoder Sec</td>
<td>480</td>
<td>0.060</td>
<td>0.10</td>
<td>1.19</td>
<td>0.32</td>
<td>2.8</td>
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<td>5-98</td>
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<td>Power Amplifier Pri</td>
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<td>0.018</td>
<td>0.90</td>
<td>0.60</td>
<td>0.05</td>
<td>16.2</td>
<td>(1)</td>
<td>5-100</td>
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<tr>
<td>Power Amplifier Sec</td>
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<td>0.90</td>
<td>0.60</td>
<td>0.05</td>
<td>16.2</td>
<td>(1)</td>
<td>5-102</td>
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<td>RF Multiplexer</td>
<td>520</td>
<td>0.130</td>
<td>0.85</td>
<td>0.60</td>
<td>0.47</td>
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<td>5-104</td>
<td>N/A</td>
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<td>Hybrid Junction</td>
<td>500</td>
<td>0.045</td>
<td>0.90</td>
<td>1.19</td>
<td>0.42</td>
<td>0.0</td>
<td>(5)</td>
<td>5-106</td>
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<td>N/A</td>
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<tr>
<td>Filter</td>
<td>610</td>
<td>0.077</td>
<td>0.90</td>
<td>1.19</td>
<td>0.42</td>
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<td>5-108</td>
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<tr>
<td>Modulation Proc Pri</td>
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<td>0.153</td>
<td>0.90</td>
<td>1.79</td>
<td>1.47</td>
<td>7.5</td>
<td>(1)</td>
<td>5-110</td>
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<tr>
<td>Modulation Proc Sec</td>
<td>630</td>
<td>0.153</td>
<td>0.90</td>
<td>1.79</td>
<td>1.47</td>
<td>7.5</td>
<td>(1)</td>
<td>5-112</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

(1) Continuous power from liftoff to 98.92 hours.
(2) Off for 0.5 hours prior to each main engine burn per Table 3-1.
(3) Power on at 60.83 hours. Power off at 61.83 hours.
(4) Power on at 61.33 hours. Power off at 61.83 hours.
(5) Continuous power from liftoff through landing.

* Contains an internal heater.
<table>
<thead>
<tr>
<th>System Component Name</th>
<th>Node No.</th>
<th>Temp History (Fig.)</th>
<th>Heater Duty Cycle (Fig.)</th>
<th>Heater Size, watts</th>
<th>Average Heater Duty Cycle</th>
<th>Heater Power Consumed, watts</th>
<th>System Component Name</th>
<th>Node No.</th>
<th>Temp History (Fig.)</th>
<th>Heater Duty Cycle (Fig.)</th>
<th>Heater Size, watts</th>
<th>Average Heater Duty Cycle</th>
<th>Heater Power Consumed, watts</th>
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<tr>
<td>Guidance</td>
<td></td>
<td></td>
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<td>Communications</td>
<td></td>
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<tr>
<td>Navigation &amp; Control</td>
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<tr>
<td>Inertial Meas. Unit</td>
<td>300</td>
<td>5-47</td>
<td>5-113</td>
<td>20</td>
<td>20</td>
<td></td>
<td>Transponder, PM Pri</td>
<td>430</td>
<td>5-89</td>
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<td>Star Tracker Pri</td>
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<td>5-49</td>
<td>5-117</td>
<td>20</td>
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<td>Transponder, PM Sec</td>
<td>440</td>
<td>5-91</td>
<td></td>
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</tr>
<tr>
<td>Star Tracker Sec</td>
<td>320</td>
<td>5-51</td>
<td>5-119</td>
<td>15</td>
<td>15</td>
<td></td>
<td>Transmitter, PM Pri</td>
<td>450</td>
<td>5-93</td>
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<td>Horizon Scanner</td>
<td>330</td>
<td>5-53</td>
<td>5-121</td>
<td>25</td>
<td>25</td>
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<td>Transmitter, PM Sec</td>
<td>460</td>
<td>5-95</td>
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<tr>
<td>Horizon Scanner Elec.</td>
<td>340</td>
<td>5-55</td>
<td>5-123</td>
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<td>150</td>
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<td>Decoder, Pri</td>
<td>470</td>
<td>5-97</td>
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<tr>
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<td>350</td>
<td>5-57</td>
<td>5-125</td>
<td>150</td>
<td>150</td>
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<td>Decoder, Sec</td>
<td>480</td>
<td>5-99</td>
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<tr>
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<td>5-59</td>
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<td>150</td>
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<td>5-61</td>
<td>5-129</td>
<td>150</td>
<td>150</td>
<td></td>
<td>Power Amplifier, Sec</td>
<td>510</td>
<td>5-103</td>
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<td>5-105</td>
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<td>Television Pri</td>
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<td>5-133</td>
<td>20</td>
<td>20</td>
<td></td>
<td>Hybrid Junction</td>
<td>600</td>
<td>5-107</td>
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<td>Modulation Processor, Pri</td>
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<td>5-135</td>
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<td>Computer Sec</td>
<td>420</td>
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<td>5-73</td>
<td>5-137</td>
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<td>5-77</td>
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<td>Tlmtry Frmtr Pri</td>
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<td>5-137</td>
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<tr>
<td>Tlmtry Frmtr Sec</td>
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<td>5-81</td>
<td>5-137</td>
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<td>Data Bus Cont Pri</td>
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<td>Data Bus Cont. Sec</td>
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<tr>
<td>Tape Recorder</td>
<td>490</td>
<td>5-87</td>
<td>5-137</td>
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<tr>
<td>System/Component Name</td>
<td>Node No.</td>
<td>Surface Area, (m^2)</td>
<td>(\varepsilon)</td>
<td>Contact Conductance, Watts/°K</td>
<td>Thermal Mass, Watt-hr/°K</td>
<td>Oper. Power, watts</td>
<td>Power Time History (Fig.)</td>
<td>Heater Size, watts</td>
<td>Average Heater Power Consumption, watts</td>
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<td>-------------------------------</td>
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<tr>
<td><strong>Auxiliary Propulsion System</strong></td>
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<tr>
<td>APS Tanks</td>
<td>601*</td>
<td>1.028</td>
<td>0.10 (1)</td>
<td>Isolated</td>
<td>Heater Node</td>
<td>N/A</td>
<td>N/A</td>
<td>5-178**</td>
<td>1</td>
<td>0.04</td>
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<td>Valve Amplifier</td>
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<td>0.225</td>
<td>0.90</td>
<td>0.36</td>
<td>1.33</td>
<td>38.0</td>
<td>(2)</td>
<td>5-180</td>
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<td>Main Propulsion Sys.</td>
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<td></td>
</tr>
<tr>
<td>Helium Press Spheres</td>
<td>231*</td>
<td>1.487</td>
<td>0.10 (1)</td>
<td>Isolated</td>
<td>Arithmetic Nodes</td>
<td>N/A</td>
<td>N/A</td>
<td>5-182</td>
<td>N/A</td>
<td>N/A</td>
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<tr>
<td>Data Management Sys.</td>
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<tr>
<td>Data Acc Unit</td>
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<td>0.078</td>
<td>0.90</td>
<td>0.24</td>
<td>0.89</td>
<td>5.2</td>
<td>(3)</td>
<td>5-184</td>
<td>N/A</td>
<td>N/A</td>
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</tbody>
</table>

(1) Represents emissivity of insulation blanket.
(2) Continuous power from liftoff to 98.92 hours.
(3) Continuous power from liftoff through landing.
* 601 is representative of the eight APS tanks.
** Represents the temperature of outside insulation blanket.

** Represents the temperature of outside insulation blanket.
<table>
<thead>
<tr>
<th>System Component Name</th>
<th>Node No.</th>
<th>Surface Area $m^2$</th>
<th>$\varepsilon$</th>
<th>Contact Conductance, Watts $^\circ K$</th>
<th>Thermal Mass, Watt-hr $^\circ K$</th>
<th>Oper. Power, watts</th>
<th>Power Time, watts</th>
<th>Temperature History (Fig.)</th>
<th>Heater Duty Cycle (Fig.)</th>
<th>Heater Size, watts</th>
<th>Average Heater Power Consumption, watts</th>
</tr>
</thead>
<tbody>
<tr>
<td>LH$_2$ Sphere</td>
<td>211</td>
<td>2.088</td>
<td>0.10(1)</td>
<td>Isolated</td>
<td>Arithmetic Node</td>
<td>N/A</td>
<td>N/A</td>
<td>5-186*</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>LOX Sphere</td>
<td>216</td>
<td>1.487</td>
<td>0.10(1)</td>
<td>Isolated</td>
<td>Arithmetic Node</td>
<td>N/A</td>
<td>N/A</td>
<td>5-188*</td>
<td>N/A</td>
<td>N/A</td>
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<tr>
<td>Battery**</td>
<td>275</td>
<td>0.140</td>
<td>0.10(1)</td>
<td>Louvered</td>
<td>1.11</td>
<td>45W</td>
<td>5-190</td>
<td>5-194</td>
<td>5</td>
<td>0.02</td>
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<tr>
<td>Fuel Cell***</td>
<td>433</td>
<td>0.445</td>
<td>0.10(1)</td>
<td>0.234</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>5-192</td>
<td>N/A</td>
<td>N/A</td>
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</tr>
</tbody>
</table>

(1) Represents emissivity of insulation blanket.
* Represents temperature of outside of insulation blanket.
** Five sides of the battery are insulated 0.11 m$^2$ (1.19 ft$^2$), = 0.1. The base 0.29 m$^2$ (0.3125 ft$^2$) is covered by louvers (c shown in Figure 5-10).
*** The fuel cell temperature is held at a constant 356$^\circ K$ (180°F) until 97.63 hours when its temperature is calculated normally.
### Table 5-6 Intertank Compartment Cold-Case Summary

<table>
<thead>
<tr>
<th>System/Component Name</th>
<th>Node No.</th>
<th>Temperature History (Fig.)</th>
<th>Heater Duty Cycle (Fig.)</th>
<th>Heater Power Consumed, watts</th>
<th>Average Heater Power, watts</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aux. Propulsion System</td>
<td>APS Tanks</td>
<td>601* 5-179** N/A</td>
<td>N/A</td>
<td>.2</td>
<td>.16</td>
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<tr>
<td>Valve Amplifier</td>
<td>290</td>
<td>5-181** N/A</td>
<td>N/A</td>
<td>N/A</td>
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<tr>
<td>Main Propulsion System</td>
<td>Helium Press Sphere</td>
<td>231* 5-183 N/A</td>
<td>N/A</td>
<td>N/A</td>
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<tr>
<td>Data Mgmt System</td>
<td>Data Acc. Unit</td>
<td>280 5-185 N/A</td>
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<td>Elec. Power Subsys.</td>
<td>LH₂ Sphere</td>
<td>211 5-187 N/A</td>
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<tr>
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<td>LOX Sphere</td>
<td>216 5-189 N/A</td>
<td>N/A</td>
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<tr>
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<td>Battery</td>
<td>275 5-191 5-195</td>
<td>N/A</td>
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<tr>
<td></td>
<td>Fuel Cell</td>
<td>433 5-193 N/A</td>
<td>N/A</td>
<td>N/A</td>
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</tbody>
</table>

* 601 is representative of the eight APS tanks; 231 is representative of the four helium spheres.

** Represents temperature of outside insulation blanket.

*** Represents net heat transfer to maintain fluid at 278°C (40°F).
The cold case used environments consistent with the hot case until 24.35 hours, corresponding to the first shadow point in geosynchronous orbit. At this time the Tug was reoriented with the longitudinal axis parallel to the solar vector (Table 3-2, Case 8 geosynchronous) to minimize external orbital heating. Component power dissipation cycles continued as in the hot case. The cold-case simulation was terminated at 45 hours.

5.2 FORWARD COMPARTMENT RESULTS

The results of the hot and cold cases for the forward compartment analyzed are summarized in Tables 5-3 and 5-4. Many of the forward compartment components had simulated thermostatically controlled heaters to maintain their temperature limits.

Each component was reviewed after the initial hot case run for compatibility with its allowable temperature limits while the compartment power was at the 800-watt level. Energy balances were performed on the components that dropped below their lower temperature limits to determine major heat leaks and heater sizing requirements. As previously discussed, the mounting conduction was reduced and heaters added where required. The heaters were sized to maintain the lower temperature limit of each component in the hot case. During this exercise, it became apparent that excessive heater power was being consumed for the hot case and this was expected to be significantly worst in the later cold-case run. The cold-case run was performed to further determine heater requirements. These runs pointed to the need for an alternative thermal control concept to avoid the excessive heater power consumption.

The total heater energy required by these components was calculated by time integrating the instantaneous heater power over the total mission duration. Individual component heater powers are tabulated in Table 5-3 and the total integrated heater energy for the entire forward compartment is shown in Figures 5-11 and 5-12 for the hot and cold cases, respectively. The hot-case mission resulted in an average of 275 watts of heater power over most of the mission. The cold-case mission consumed an average of 774 watts of heater power after 25 hours in the mission. This was not sufficient to maintain the component lower temperature limits. This emphasizes the need to alter the thermal control concept originally chosen. The transient model wall nodes are shown in Figure 5-13 and the hot and cold case temperature results for these nodes are given in Figures 5-14 thru 5-45. Figures 5-46 thru 5-137 present the forward component temperatures.
The remaining areas of the forward compartment are presented in Figures 5-138 thru 5-143. Figures 5-138 and 5-139 present the outer layer of insulation on the LH₂ tank dome temperature for the hot and cold cases. Figures 5-140 and 5-141 present the forward shield inner surface temperatures and Figures 5-142 and 5-143 present the outer surface (beta cloth) temperatures for the hot and cold cases respectively. Figures 5-144 and 5-145 present the forward compartment internal sink temperatures derived from each case. Comparison of these data with the previous steady-state results accounts for the honeycomb AT and should be compared only where steady-state conditions exist.

5.3 INTERTANK COMPARTMENT RESULTS

The intertank compartment results are presented beginning with the outer and inside skin temperatures in Figures 5-146 through 5-177. Tables 5-5 and 5-6 summarize the component data and refer to the appropriate figures for the hot and cold case temperature results. This compartment contains several tanks as shown in Figures 5-4 and 5-5, hence the data presented in the report is representative of each of the various types of tanks. Figure 5-178 presents the insulation temperature for one of the eight APS tanks where each tank was controlled to 278°K (40°F). Node 231, Figure 5-182, presents representative data for the four helium pressurization spheres. The fuel cell LH₂ and LOX tank plots represent the insulation temperatures. Each tank was held at its liquid temperature during the mission simulations. Insulation properties derived from Reference 5 were used on the LH₂, LOX, and APS tanks, assuming the configuration is as applied to the forward shield.

The LH₂ tank lower dome insulation and LOX tank upper dome insulation temperatures are presented in Figures 5-196 thru 5-199. The intertank compartment sink temperature is presented in Figures 5-200 and 5-201.
FIGURE 5-11. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-12. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5.12. Transient Model Skin Nodes

Interior Nodes
TEMP NODE NO. 71 Outer Skin, Forward

MIN TEMP OF 232.722 OCCURRED AT TIME 48.800
MAX TEMP OF 338.194 OCCURRED AT TIME 101.400

FIGURE 5-14. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

TEMP NODE NO. 71 Outer Skin, Forward

MIN TEMP OF 216.374 OCCURRED AT TIME 43.300
MAX TEMP OF 312.534 OCCURRED AT TIME 1.800

FIGURE 5-15. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-16. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO. 1 Inner Skin, Forward
MIN TEMP OF 239.009 OCCURRED AT TIME 49.600
MAX TEMP OF 333.599 OCCURRED AT TIME 101.400

FIGURE 5-17. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

MISSION TIME - HOURS

TEMP NODE NO. 1 Inner Skin, Forward
MIN TEMP OF 221.890 OCCURRED AT TIME 43.300
MAX TEMP OF 308.466 OCCURRED AT TIME 1.000
Figure 5-18: Analysis of TUG 910. Comp. + Components with Heat Pipes

Temperature - Kelvin

Temperature - Kelvin

Mission Time - Hours

Mission Time - Hours

Temp Node No. 72 Outer Skin, Forward

Min Temp of 232.435 occurred at time 48.800

Max Temp of 336.113 occurred at time 101.400

Min Temp of 216.223 occurred at time 43.300

Max Temp of 304.630 occurred at time 1.800

Figure 5-19: Analysis of TUG 910. Comp. Stationed at Geo. Shadow.
FIGURE 5-20. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-21. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5-22. Analysis of TUG FWD. CORP. + COMPONENTS WITH HEAT PIPES

TEMP NODE NO. 73 Outer Skin, Forward
MIN TEMP OF 232.114 OCCURRED AT TIME 49.000
MAX TEMP OF 330.449 OCCURRED AT TIME 101.400

Figure 5-23. Analysis of TUG FWD. CORP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 73 Outer Skin, Forward
MIN TEMP OF 216.180 OCCURRED AT TIME 44.000
MAX TEMP OF 333.449 OCCURRED AT TIME 101.400
TEMP NODE NO. 3 Inner Skin, Forward
MIN TEMP OF 237.09\(^\circ\)C OCCURRED AT TIME 48.000
MAX TEMP OF 334.06\(^\circ\)C OCCURRED AT TIME 101.400

MIN TEMP OF 221.058 OCCURRED AT TIME 44.900
MAX TEMP OF 306.266 OCCURRED AT TIME 1.800

FIGURE 5-24. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-25. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
TABLE 6-26. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

<table>
<thead>
<tr>
<th>TEMP NODE NO.</th>
<th>NODE NO.</th>
</tr>
</thead>
<tbody>
<tr>
<td>NODE NO.</td>
<td>NODE NO.</td>
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<tr>
<td>MIN TEMP</td>
<td>231.021</td>
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<tr>
<td>MAX TEMP</td>
<td>330.600</td>
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</tbody>
</table>

FIGURE 6-27. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO.  Inner Skin, Forward
MIN TEMP OF 237.246 OCCURRED AT TIME 48.000
MAX TEMP OF 334.193 OCCURRED AT TIME 101.400

FIGURE 5-28
ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO.  Inner Skin, Forward
MIN TEMP OF 220.310 OCCURRED AT TIME 44.800
MAX TEMP OF 306.194 OCCURRED AT TIME 1.800

FIGURE 5-29
ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-30. ANALYSIS OF TUG FWD. COMD. + COMPONENTS WITH HEAT PIPES

FIGURE 5-31. ANALYSIS OF TUG FWD. COMD. STATIONED AT GEO. SHADOW PT.
Figure 5-32. Analysis of TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

Figure 5-33. Analysis of TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-34. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-36. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5.36. ANALYSIS OF TUG FWDO. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO. 6 Inner Skin, Forward
MIN TEMP OF 236.810 OCCURRED AT TIME 46.800
MAX TEMP OF 334.235 OCCURRED AT TIME 101.400

FIGURE 5.37. ANALYSIS OF TUG FWDO. COMP. STATIONED AT GEO. SHADOW PT.

MISSION TIME - HOURS

TEMP NODE NO. 6 Inner Skin, Forward
MIN TEMP OF 219.805 OCCURRED AT TIME 44.800
MAX TEMP OF 306.115 OCCURRED AT TIME 1.800
FIGURE 5-38. ANALYSIS OF TUG FWD. COMPONENTS WITH HEAT PIPES

FIGURE 5-39. ANALYSIS OF TUG FWD. COMPONENTS STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO. 7 Inner Skin, Forward
MIN TEMP OF 237.217 OCCURRED AT TIME 48.800
MAX TEMP OF 334.061 OCCURRED AT TIME 101.400

MIN TEMP OF 219.972 OCCURRED AT TIME 45.000
MAX TEMP OF 306.854 OCCURRED AT TIME 1.800

FIGURE 5-40. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-41. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 6.42: ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 6.43: ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS
TEMP NODE NO. B Inner Skin, Forward
MIN TEMP OF 238.153 OCCURRED AT TIME 48.800
MAX TEMP OF 333.670 OCCURRED AT TIME 101.400

MIN TEMP OF 220.524 OCCURRED AT TIME 43.300
MAX TEMP OF 308.459 OCCURRED AT TIME 1.800

FIGURE 5-44. ANALYSIS OF tug fwd. comp. + components with heat pipes
FIGURE 5-45. ANALYSIS OF tug fwd. comp. stationed at geo. shadow pt.
FIGURE 5-46. ANALYSIS OF TUG FOG. COMP. + COMPONENTS WITH HEAT PIPES

TEMP NODE NO. 300 Inertial Meas. Unit
MIN TEMP OF 287.695 OCCURRED AT TIME 100.500
MAX TEMP OF 325.533 OCCURRED AT TIME 3.300

FIGURE 5-47. ANALYSIS OF TUG FOG. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 300 Inertial Meas. Unit
MIN TEMP OF 275.939 OCCURRED AT TIME 45.000
MAX TEMP OF 325.933 OCCURRED AT TIME 3.300
Figure 5-48. Analysis of tug fwd. comp. + components with heat pipes

Figure 5-49. Analysis of tug fwd. comp. stationed at geo. shadow pt.
**FIGURE 5-50**  ANALYSIS OF TUG FWD. COOP. + COMPONENTS WITH HEAT PIPES

**FIGURE 5-51**  ANALYSIS OF TUG FWD. COOP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO. 330 Horizon Scanner
MIN TEMP OF 248.209 OCCURRED AT TIME 48.900
MAX TEMP OF 328.497 OCCURRED AT TIME 101.800

MIN TEMP OF 228.270 OCCURRED AT TIME 44.900
MAX TEMP OF 303.737 OCCURRED AT TIME 1.800

FIGURE 5-52. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-53. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-54 . ANALYSIS OF TUG FWD. CORP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-55 . ANALYSIS OF TUG FWD. CORP. STATIONED AT GEO. SHADOW PT.

- Upper Limit
- Lower Limit

TEMP NODE NO. 340 Horizon Scanner Elec.
MIN TEMP OF 253.823 OCCURRED AT TIME 48.000
MAX TEMP OF 316.203 OCCURRED AT TIME 103.300

MIN TEMP OF 228.605 OCCURRED AT TIME 45.000
MAX TEMP OF 297.399 OCCURRED AT TIME 1.800
MISSION TIME - HOURS

TEMP NODE NO. 350 Laser Radar Pri
MIN TEMP OF 291.207 OCCURRED AT TIME 48.900
MAX TEMP OF 317.372 OCCURRED AT TIME 103.000

FIGURE 5-56. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO. 350 Laser Radar Pri
MIN TEMP OF 275.599 OCCURRED AT TIME 45.000
MAX TEMP OF 299.523 OCCURRED AT TIME 4.300

FIGURE 5-57. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
**Figure 5-58.** Analysis of TUG FWD. Comp. + Components with Heat Pipes

**Figure 5-59.** Analysis of TUG FWD. Comp. Stationed at GEO. Shadow Pt.

- **Mission Time - Hours**
- **Temperature - Degrees Kelvin**
- **Temp Node No.**
  - 330 Laser Radar Sec

- **Min Temp of**
  - 291.429° occurred at time 48.500

- **Max Temp of**
  - 317.33° occurred at time 103.000

- **Min Temp of**
  - 275.631° occurred at time 45.000

- **Max Temp of**
  - 299.427° occurred at time 4.300
Figure 5-60. Analysis of Tug Fwd. Comp. + Components with Heat Pipes

Figure 5-61. Analysis of Tug Fwd. Comp. Stationed at Geo. Shadow Pt.
Figure 5.62: Analysis of XUG FAD, CORP. + COMPONENTS WITH HEAT PIPES

Figure 5.63: Analysis of XUG FAD, CORP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-64. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-65. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-66. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-67. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5-68: Analysis of TUG FWD. Comp. + Components with Heat Pipes

Figure 5-69: Analysis of TUG FWD. Comp. Stationed at GEO, Shadow Pt.
FIGURE 5.70. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

Temp Node No. 420 Computer Sec
Min Temp of 303.444 occurred at time 48.900
Max Temp of 335.482 occurred at time 102.000

FIGURE 5.71. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

Temp Node No. 420 Computer Sec
Min Temp of 247.318 occurred at time 44.500
Max Temp of 311.892 occurred at time 1.800
FIGURE 5-72. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-73. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MIN TEMP OF 251.067 OCCURRED AT TIME 48.000
MAX TEMP OF 327.022 OCCURRED AT TIME 102.000

FIGURE 5-74. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MIN TEMP OF 225.569 OCCURRED AT TIME 45.000
MAX TEMP OF 304.313 OCCURRED AT TIME 1.800

FIGURE 5-75. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MIN TEMP OF 250.615 OCCURRED AT TIME 46.900
MAX TEMP OF 326.942 OCCURRED AT TIME 102.200

MIN TEMP OF 225.242 OCCURRED AT TIME 45.000
MAX TEMP OF 303.706 OCCURRED AT TIME 1.000

FIGURE 5-76. ANALYSIS OF TUG FWD. COMP. WITH HEAT PIPES
FIGURE 5-77. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-78. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

TEMP NODE NO. 560 Telemetry Formatter, Pri
MIN TEMP OF 249.325 OCCURRED AT TIME 48.900
MAX TEMP OF 327.394 OCCURRED AT TIME 101.900

FIGURE 5-79. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 560 Telemetry Formatter, Pri
MIN TEMP OF 226.167 OCCURRED AT TIME 44.900
MAX TEMP OF 306.463 OCCURRED AT TIME 1.800
FIGURE 5-80. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

TABLE 5-80

<table>
<thead>
<tr>
<th>TEMP NODE NO.</th>
<th>MIN TEMP OF</th>
<th>MAX TEMP OF</th>
</tr>
</thead>
<tbody>
<tr>
<td>570</td>
<td>248.844</td>
<td>327.189</td>
</tr>
</tbody>
</table>

OCCURRED AT TIME

48.900

101.900

FIGURE 5-81. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

TABLE 5-81

<table>
<thead>
<tr>
<th>TEMP NODE NO.</th>
<th>MIN TEMP OF</th>
<th>MAX TEMP OF</th>
</tr>
</thead>
<tbody>
<tr>
<td>570</td>
<td>226.329</td>
<td>326.558</td>
</tr>
</tbody>
</table>

OCCURRED AT TIME

45.000

1.800

Lower Limit

Upper Limit
FIGURE 5.82. ANALYSIS OF TUG FLO. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5.83. ANALYSIS OF TUG FLO. COMP. STATIONED AT GEO. SHADOW PT.
**FIGURE 5-84.** ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

**FIGURE 5-85.** ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-86 ANALYSIS OF TUG FLG. CORP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-87 ANALYSIS OF TUG FLG. CORP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-88. ANALYSIS OF TUG FLG. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-89. ANALYSIS OF TUG FLG. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5.80. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5.81. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
TEMPERATURE - DEGREES KELVIN

MISSION TIME - HOURS

TEMP NODE NO. 450 Transmitter, FM Pri
MIN TEMP OF 226.787 OCCURRED AT TIME 40.600
MAX TEMP OF 329.264 OCCURRED AT TIME 101.700

MIN TEMP OF 230.000 OCCURRED AT TIME 45.000
MAX TEMP OF 310.028 OCCURRED AT TIME 1.800

FIGURE 5-92. ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-93. ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-94: ANALYSIS OF TUG fwd. COOP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-95: ANALYSIS OF TUG fwd. COOP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-96 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-97 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-98 ANALYSIS OF TUG FD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-99 ANALYSIS OF TUG FD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO.  500 Power Amplifier Pri
MIN TEMP OF  294.171 OCCURRED AT TIME  100.200
MAX TEMP OF  332.426 OCCURRED AT TIME  101.500

MIN TEMP OF  246.795 OCCURRED AT TIME  42.800
MAX TEMP OF  329.452 OCCURRED AT TIME  1.800

FIGURE 5-100 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-101 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO. 510 Power Amplifier, Sec

MIN TEMP OF 254.493 OCCURRED AT TIME 100.200
MAX TEMP OF 332.457 OCCURRED AT TIME 101.500

FIGURE 5-102 ANALYSIS OF TUG FWD. CORP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO. 510 Power Amplifier, Sec

MIN TEMP OF 246.765 OCCURRED AT TIME 42.800
MAX TEMP OF 329.495 OCCURRED AT TIME 1.800

FIGURE 5-103 ANALYSIS OF TUG FWD. CORP. STATIONED AT GEO. SHADOW PT.
MIN TEMP OF 240.281 OCCURRED AT TIME 48.900
MAX TEMP OF 326.480 OCCURRED AT TIME 102.000

MIN TEMP OF 240.331 OCCURRED AT TIME 45.000
MAX TEMP OF 303.814 OCCURRED AT TIME 1.800
FIGURE 5-106 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-107 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-108 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

Upper Limit

Lower Limit

MISSION TIME - HOURS

TEMP NODE NO. 610 Filter
MIN TEMP OF 242.290 OCCURRED AT TIME 48.500
MAX TEMP OF 327.729 OCCURRED AT TIME 101.300

FIGURE 5-109 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

Upper Limit

Lower Limit

MISSION TIME - HOURS

TEMP NODE NO. 610 Filter
MIN TEMP OF 220.751 OCCURRED AT TIME 45.000
MAX TEMP OF 303.135 OCCURRED AT TIME 1.800

FIGURE 5-100 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO. 620 Modulation Proc. Pri
MIN TEMP OF 231.200 OCCURRED AT TIME 48.000
MAX TEMP OF 323.418 OCCURRED AT TIME 102.300

FIGURE 5-110 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

TEMP NODE NO. 620 Modulation Proc. Pri
MIN TEMP OF 224.925 OCCURRED AT TIME 44.900
MAX TEMP OF 303.094 OCCURRED AT TIME 1.800

FIGURE 5-111 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
### Figure 5-112
**Analysis of Tug Fwd. Comp. + Components with Heat Pipes**

- **Temperature - Degrees Kelvin**
  - Upper Limit
  - Lower Limit

- **Mission Time - Hours**
  - Temp Node No.: 630
  - Modulation Proc. Sec

- **Temp**
  - MIN Temp of 251.415 occurred at time 48.900
  - MAX Temp of 323.412 occurred at time 102.300

### Figure 5-113
**Analysis of Tug Fwd. Comp. Stationed at Geo. Shadow Pt.**

- **Temperature - Degrees Kelvin**
  - Lower Limit

- **Mission Time - Hours**
  - Temp Node No.: 630
  - Modulation Proc. Sec

- **Temp**
  - MIN Temp of 225.041 occurred at time 44.900
  - MAX Temp of 303.085 occurred at time 1.800
MISSION TIME - HOURS

NODE NO. 113 Star Tracker Pri. Heater Power

MIN OF 0. OCCURRED AT TIME 110.000
MAX OF 68.260 OCCURRED AT TIME 73.500

FIGURE 5-114 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

NODE NO. 113 Star Tracker Pri. Heater Power

MIN OF 0. OCCURRED AT TIME 24.900
MAX OF 68.260 OCCURRED AT TIME 45.000

FIGURE 5-115 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-116 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-117 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

MISSION TIME - HOURS

NODE NO. 111 Horizon Scanner Heater Power

MIN OF 0. OCCURRED AT TIME 110.000
MAX OF 51.195 OCCURRED AT TIME 73.300

FIGURE 5-118 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

NODE NO. 111 Horizon Scanner Heater Power

MIN OF 0. OCCURRED AT TIME 24.900
MAX OF 51.195 OCCURRED AT TIME 45.000

FIGURE 5-119 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

NODE NO. 110 Horizon Scanner Electronics Heater Power
MIN OF 0. OCCURRED AT TIME 110.000
MAX OF 85.325 OCCURRED AT TIME 73.700

MIN OF 0. OCCURRED AT TIME 25.400
MAX OF 85.325 OCCURRED AT TIME 45.000

FIGURE 5-120 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-121 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-122 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-123 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
### Node No. 108

**Laser Radar Sec Heater Power**

- **Min**: 0.000, occurred at time: 110.000
- **Max**: 511.950, occurred at time: 108.600

---

**Figure 5.124**

Analysis of TUG FWD. Comp. + Components with Heat Pipes

**Figure 5.125**

Analysis of TUG FWD. Comp. stationed at GEO. Shadow Pt.
MISSION TIME - HOURS

NODE NO. 107 Laser Radar Electronics Pri Heater Power
MIN OF 0. OCCURRED AT TIME 110.000
MAX OF 511.950 OCCURRED AT TIME 100.300

FIGURE 5-126 ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

NODE NO. 107 Laser Radar Electronics Pri Heater Power
MIN OF 0. OCCURRED AT TIME 25.100
MAX OF 511.950 OCCURRED AT TIME 45.000

FIGURE 5-127 ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.
Figures 5-128 and 5-129 show the analysis of TUG FWD. Comp. components with heat pipes and the analysis of TUG FWD. Comp. stationed at Geo. Shadow Pt., respectively.
**FIGURE 5-130** ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

MISSION TIME - HOURS

NODE NO. 105 Television Pri Heater Power

MIN OF 0. OCCURRED AT TIME 110.000
MAX OF 69.260 OCCURRED AT TIME 73.600

**FIGURE 5-131** ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

MISSION TIME - HOURS

NODE NO. 105 Television Pri Heater Power

MIN OF 0. OCCURRED AT TIME 24.900
MAX OF 68.260 OCCURRED AT TIME 45.000
FIGURE 5-134  ANALYSIS OF TUG FWD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-135  ANALYSIS OF TUG FWD. COMP. STATIONED AT GEO. SHADOW PT.

NODE NO.  102  Computer Pri Heater Power
MIN OF 0.  OCCURRED AT TIME 110.000
MAX OF 0.  OCCURRED AT TIME 0.
Figure 5-136: Analysis of TUG FWD. Comp. + Components with Heat Pipes

Figure 5-137: Analysis of TUG FWD. Comp. Stationed at GEO. Shadow Pt.
FIGURE 5-138 ANALYSIS OF TUG FD. COMP. + COMPONENTS WITH HEAT PIPES

FIGURE 5-139 ANALYSIS OF TUG FD. COMP. STATIONED AT GEO. SHADY PT.
Figure 5.40 Analysis of Tug Fwd. Comp. + Components with Heat Pipes

Figure 5.41 Analysis of Tug Fwd. Comp. Stationed at Geo. Shadow Pt.
Figure 5-142: Analysis of TUG FWD. Comp. + Components with Heat Pipes

MIN TEMP: 55.627°C OCCURRED AT TIME 47.900
MAX TEMP: 358.712°C OCCURRED AT TIME 101.300

Figure 5-143: Analysis of TUG FWD. Comp. Stationed at Geo. Shadow Point

MIN TEMP: 49.320°C OCCURRED AT TIME 44.900
MAX TEMP: 258.130°C OCCURRED AT TIME 69.000
Mission Time - Hours

Temp Node No. 34 Forward Compartment Internal Sink Temperature

Min Temp of 249.007 occurred at time 48.000
Max Temp of 329.156 occurred at time 101.400

Figure 5-144 Analysis of Tug Fwd. Comp. + Components with Heat Pipes
<table>
<thead>
<tr>
<th>TEMP NODE NO.</th>
<th>81 Outer Skin</th>
</tr>
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<tbody>
<tr>
<td>MIN TEMP of</td>
<td>104.830</td>
</tr>
<tr>
<td>OCCURRED AT</td>
<td>48.800</td>
</tr>
<tr>
<td>MAX TEMP of</td>
<td>356.314</td>
</tr>
<tr>
<td>OCCURRED AT</td>
<td>101.400</td>
</tr>
</tbody>
</table>

**MIN TEMP OF** 100.201 OCCURRED AT TIME 45.000

**MAX TEMP OF** 338.364 OCCURRED AT TIME 1.800

**FIGURE 5-146** ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

**FIGURE 5-147** ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5-148: Analysis of TUG Int. Compartment + Components No Heat Pipe

Temperature - Degrees Kelvin

Mission Time - Hours

Temp Node No. 9 Inner Skin

Min Temp of 165.000 occurred at time 48.600

Max Temp of 355.285 occurred at time 101.400

Figure 5-149: Analysis of TUG Int. Comp. Stationed at GEO Shadow Pt.

Temperature - Degrees Kelvin

Mission Time - Hours

Temp Node No. 9 Inner Skin

Min Temp of 165.408 occurred at time 45.000

Max Temp of 336.514 occurred at time 1.800
FIGURE 5-150 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

TEMP NODE NO. 62 Outer Skin
MIN TEMP OF 163.699 OCCURRED AT TIME 48.800
MAX TEMP OF 356.399 OCCURRED AT TIME 101.400

FIGURE 5-151 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 62 Outer Skin
MIN TEMP OF 59.944 OCCURRED AT TIME 45.000
MAX TEMP OF 320.624 OCCURRED AT TIME 1.800
MIN TEMP OF 163.923 OCCURRED AT TIME 48.800
MAX TEMP OF 356.357 OCCURRED AT TIME 101.400

MIN TEMP OF 100.146 OCCURRED AT TIME 45.000
MAX TEMP OF 324.199 OCCURRED AT TIME 1.600

FIGURE 5-152 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-153 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
MIN TEMP OF 160.359 OCCURRED AT TIME 72.800
MAX TEMP OF 357.182 OCCURRED AT TIME 101.400

MIN TEMP OF 59.305 OCCURRED AT TIME 45.000
MAX TEMP OF 330.778 OCCURRED AT TIME 1.800
<table>
<thead>
<tr>
<th>Temp Node No.</th>
<th>Inner Skin</th>
</tr>
</thead>
<tbody>
<tr>
<td>Min Temp of</td>
<td>161.245</td>
</tr>
<tr>
<td>Max Temp of</td>
<td>356.939</td>
</tr>
</tbody>
</table>

**MIN TEMP**
- Min Temp of 161.245 occurred at time 72.800 min
- Max Temp of 356.939 occurred at time 101.400 min

**MAX TEMP**
- Min Temp of 98.555 occurred at time 45.000 min
- Max Temp of 331.274 occurred at time 1.800 min

**Figure 5-156** Analysis of tug int. compartment + components no heat pipe

**Figure 5-157** Analysis of tug int. comp. stationed at geo. shadow pt.
FIGURE 5-158 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-159 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5-160 Analysis of Tug Int. Compartment + Components No. Heat Pipe

Figure 5-161 Analysis of Tug Int. Comp. Stationed at Geo. Shadow Pt.

Temp Node No.: 12 Inner Skin

Min Temp of 160.949 Occurred at Time 72.800
Max Temp of 356.991 Occurred at Time 101.400

Min Temp of 99.025 Occurred at Time 45.000
Max Temp of 337.589 Occurred at Time 1.800
FIGURE 5-169 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-163 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO.  13 Inner Skin

MIN TEMP OF  161.068 OCCURRED AT TIME  72.800
MAX TEMP OF  357.006 OCCURRED AT TIME  101.400

MIN TEMP OF  98.635 OCCURRED AT TIME  45.600
MAX TEMP OF  337.692 OCCURRED AT TIME  1.000

FIGURE 5-184 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-185 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
TEMP NODE NO. 66 Outer Skin
MIN TEMP OF 160.693 OCCURRED AT TIME 72.000
MAX TEMP OF 357.272 OCCURRED AT TIME 101.400

FIGURE 5.166 ANALYSIS OF TUG INT. COUPLING + COMPONENTS NO HEAT PIPE

TEMP NODE NO. 66 Outer Skin
MIN TEMP OF 99.698 OCCURRED AT TIME 45.000
MAX TEMP OF 330.865 OCCURRED AT TIME 1.800

FIGURE 5.167 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. $46.004$ FT.
FIGURE 5-168 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

- TEMP NODE NO.: 14 Inner Skin
- MIN TEMP OF 161.450 OCCURRED AT TIME 72.000
- MAX TEMP OF 357.003 OCCURRED AT TIME 101.400

FIGURE 5-169 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.

- TEMP NODE NO.: 14 Inner Skin
- MIN TEMP OF 99.899 OCCURRED AT TIME 45.000
- MAX TEMP OF 331.304 OCCURRED AT TIME 4.000
Figure 5-170

Analysis of TUG Int. Compartment + Components No Heat Pipe

Temp Node No.  87 Outer Skin
Min Temp of 175.3 K occurred at time 48.600
Max Temp of 356.3 K occurred at time 101.400

Figure 5-171

Analysis of TUG Int. Comp. Stationed at GEO, Shadow Pt.

Temp Node No.  87 Outer Skin
Min Temp of 141.2 K occurred at time 44.600
Max Temp of 321.5 K occurred at time 1.800
TEMP NODE NO. 15 Inner Skin
MIN TEMP OF 176.341 OCCURRED AT TIME 48.000
MAX TEMP OF 356.227 OCCURRED AT TIME 101.400

MIN TEMP OF 142.275 OCCURRED AT TIME 44.600
MAX TEMP OF 324.817 OCCURRED AT TIME 1.000

FIGURE 5-172 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-173 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-174  ANALYSIS OF TUG INT. COUPLING + COMPONENTS NO HEAT PIPE

FIGURE 5-175  ANALYSIS OF TUG INT. COUP. STATIONED AT GEO. SHADOW PT.
MISSION TIME - HOURS

TEMP NODE NO. 16 Inner Skin
MIN TEMP: 169.093 OCCURRED AT TIME 48.800
MAX TEMP: 355.940 OCCURRED AT TIME 101.400

MIN TEMP: 105.704 OCCURRED AT TIME 45.000
MAX TEMP: 336.715 OCCURRED AT TIME 1.800

FIGURE 5-176 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-177 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-178 
ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

TEMP NODE NO. 201 APS Tank Insulation
MIN TEMP OF 169.283 OCCURRED AT TIME 48.000
MAX TEMP OF 358.367 OCCURRED AT TIME 101.400

FIGURE 5-179 
ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 201 APS Tank Insulation
MIN TEMP OF 121.652 OCCURRED AT TIME 45.000
MAX TEMP OF 330.857 OCCURRED AT TIME 1.800
FIGURE 5-180  ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

TEMP NODE NO.  290 Valve Amplifier
MIN TEMP OF 257.087 OCCURRED AT TIME 49.000
MAX TEMP OF 337.378 OCCURRED AT TIME 101.950

FIGURE 5-181  ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
Figure 5-182 Analysis of Tug INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

Figure 5-183 Analysis of Tug INT. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 231 Helium Press. Sphere
MIN TEMP OF 164.603 OCCURRED AT TIME 40.000
MAX TEMP OF 356.601 OCCURRED AT TIME 101.400

MIN TEMP OF 104.752 OCCURRED AT TIME 45.000
MAX TEMP OF 333.693 OCCURRED AT TIME 1.800
MISSION TIME - HOURS

TEMP NODE NO.  280 Data Acc. Unit
MIN TEMP OF  240.321 OCCURRED AT TIME  49.000
MAX TEMP OF  334.591 OCCURRED AT TIME 102.300

MIN TEMP OF 128.229 OCCURRED AT TIME  45.000
MAX TEMP OF 317.204 OCCURRED AT TIME  3.400

FIGURE 5-184 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-185 ANALYSIS OF TUG INT. COMP. STATIZED AT GEO. SHADOW PT.
TEMP NODE NO. 211 \( \text{LH}_2 \) Sphere

MIN TEMP of 104.626 OCCURRED AT TIME 45.000 MIN

MAX TEMP of 332.288 OCCURRED AT TIME 101.400 MAX

Figure 5-186 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

TEMP NODE NO. 211 \( \text{LH}_2 \) Sphere

MIN TEMP of 104.626 OCCURRED AT TIME 45.000 MIN

MAX TEMP of 332.288 OCCURRED AT TIME 101.400 MAX

Figure 5-187 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-188 ANALYSIS OF TUG INT. COMPARTMENT - COMPONENTS NO HEAT PIPE

FIGURE 5-189 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-190 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

TEMP NODE NO.  275 Battery
MIN TEMP OF  291.479 OCCURRED AT TIME  97.100
MAX TEMP OF  327.634 OCCURRED AT TIME  59.650

FIGURE 5-191 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO.  275 Battery
MIN TEMP OF  291.476 OCCURRED AT TIME  25.400
MAX TEMP OF  292.120 OCCURRED AT TIME  4.600
FIGURE 5-192 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-193 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-194  ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPES

FIGURE 5-195  ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-196 ANALYSIS OF TUG INT. COMPARTMENT - COMPONENT NO HEAT PIPE

TEMP NODE NO. 32 LH Tank Aft Dome Insulation
MIN TEMP OF 163.923 OCCURRED AT TIME 49.800
MAX TEMP OF 355.848 OCCURRED AT TIME 101.400

FIGURE 5-197 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.

TEMP NODE NO. 32 LH Tank Aft Dome Insulation
MIN TEMP OF 103.859 OCCURRED AT TIME 45.000
MAX TEMP OF 331.948 OCCURRED AT TIME 1.800
FIGURE 5-198 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-199 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
FIGURE 5-200 ANALYSIS OF TUG INT. COMPARTMENT + COMPONENTS NO HEAT PIPE

FIGURE 5-201 ANALYSIS OF TUG INT. COMP. STATIONED AT GEO. SHADOW PT.
Thermal control of the forward and intertank compartments for hot-case missions was achieved with the need of heater power. The heater power was concentrated in the low duty cycle components, namely, the primary and secondary laser radars and their electronics packages. Increasing the external coating $a/c$ ratio would reduce the amount of heater power required by increasing the internal compartmental sink temperature; however, the cold-case heater power consumption would likely remain high. The cold-case simulation resulted in all components except the fuel cell and battery dropping below the allowable lower temperature limits. Fifteen of the components were $10^\circ K$ or less below limits while 31 were 10 to $20^\circ K$ below and seven were 20 to $30^\circ K$ below limits. Out of the latter group, the IMU heater power curve was 50% of expected, which would eliminate its cold problem. Several methods are available to solve the cold-case problems, reduce the lower limit qualifications temperature, add heater power, increase conduction isolation, change component coatings, and add insulation. All of these would rely upon heater power however, and some of the components would be affected in the hot case. In any event the hot case would still require heater power which ideally should not require any heat.

An alternative forward compartment layout would be prudent to solve the cold case problems while reducing the need of heater power. The components should be grouped to allow mounting of active and inactive components on the same mount. Mounting high and low duty cycle components on thermal conditioning panels (highly conductive panels) would be a desirable configuration from a thermal point of view. The configuration considered here is shown in Figure 5-202. As shown, louvers are mounted to the skin side of the panel. The louvers provide the means of reducing panel heat losses in the cold case, while the thermal conditioning panel distributes heat between components, thus reducing the heater power required by low duty cycle components. Figures 5-203 and 5-204 present the results of a study to determine the heat flux required to maintain various panel temperatures as a function of skin temperature and internal compartmental sink temperature (TE).

Referring to the hot case curve, Figure 5-203, and assuming the skin and internal sink temperatures at 294.4°F (70°F), the panel flux range would vary from 56.7 to 179.7 watts/meter$^2$ (18 to 57 Btu/hr-ft$^2$). This corresponds to a panel temperature range of 300 to 311°F (80 to 100°F). For the 600-watt heat load a total panel area of 3.34 meters$^2$ (36 feet$^2$) would result in a panel flux of 179.4 watts/meter$^2$, yielding a panel temperature of 311°F. The advantage in using this configuration is apparent when the cold-case data for a 200°F (-100°F) skin and internal sink temperature are considered. The flux required to maintain a 272°F (30°F) panel temperature is 220.8 watts/meter$^2$ (70 Btu/hr-ft$^2$). Comparing
the hot and cold case values results in a heater requirement of 41.4 watts/meter$^2$ to maintain the selected panel temperature. Scaling this up to the assumed panel area, 138.3 watts would be required. This compares with the amount of heat required in the hot case and reduces the cold-case heater power by more than 600 watts. Increasing the total panel area by 1/3 increases the heater power significantly to 101.4 watts/meter$^2$ or a total of 508 watts. This would still yield a savings in excess of 275 watts for the cold case.

As shown, a significant reduction in heater power can be achieved using this method. Several other advantages are derived from this approach. As the Tug design evolves, the forward compartment power level will probably change. This method of thermal control provides a means of reducing the sensitivity of steady-state power on heater power requirements by maintaining preselected panel heat fluxes. Minimum cable weight can be achieved by properly grouping components on individual panels while satisfying thermal requirements. The structural design would be simplified by reducing the number of component structural interfaces to a minimum. One tradeoff would be required to determine if the reduced cable and consumable weights would offset the added weight of the louvers and thermal conditioning panels. Other tradeoffs concerning cost and design flexibility would also be in order.

The intertank compartment suffers from the lack of heat dissipated to maintain acceptable internal sink and skin temperatures. Coatings, thermal standoffs, and heaters could be used as a solution. Due to number of components expected in this compartment the louver/thermal conditioning panel concept appears to be too heavy for application.
Figure 5-202 Forward Compartment Component Mounting
Figure 5-203 Hot-Case Mounting Panel Heat Fluxes, Louvers Open

Legend:
- Panel = 300°C (80°F)
- Panel = 305°C (90°F)
- Panel = 311°C (100°F)

$\epsilon_{\text{EFF}} = 0.818$
$\epsilon_p = 0.9$

$Q/A, \text{ Btu/hr-ft}^2 (A = \text{one side of panel})$

Skin Temperature, °F

Skin Temperature, °K

$T_E = 40^\circ F$
$T_E = 60^\circ F$
$T_E = 80$

$Q/A, \text{ W/m}^2$

$280$ $285$ $290$ $295$ $300$ $305$

$31.5$ $63.0$ $94.5$ $126.0$ $157.5$ $189.0$ $220.5$ $252.0$ $283.5$
Legend:

- Panel = 283°K (50°F)
- Panel = 277.5°K (40°F)
- Panel = 272°K (30°F)

Figure 5-204 Cold-Case Mounting Panel Heat Fluxes, Louvers Closed
6. FUEL CELL HEAT REJECTION SYSTEM

Thermal control of the fuel cell electrical power subsystem represents a critical design consideration because a failure in this area could result in failure to achieve the specific mission objectives and the loss of a Tug. Two approaches were explored in this area; each used radiators. The approaches differed only in how the heat was transported from the fuel cell to the radiators. The system chosen was a redundant pumped fluid system using series-series bypassed radiators. The pumped fluid system was chosen over variable conductance (VC) heat pipes because of the current state of the art of pumped fluid systems and the current problems with VC heat pipes. The Tug is penalized in power and weight by this choice. As VC heat pipe technology expands in the future, the use of VC heat pipes in this part of the Tug design should be possible with less risk.

The fuel cell in this study was based upon design data obtained from Pratt and Whitney (Ref 13). The fuel cell heat rejection system is required to maintain the fuel cell internal fluid loop within an acceptable temperature range 349.67 to 355.2°K (170 to 180°F) independent of heat load. The baseline for the study included a single fuel cell which, when coupled with the components used in the study, resulted in an electrical load that varied from 600 to 1500 watts. The radiator design was based on rejecting resultant waste heat loads plus the fuel cell pump and radiator pump power.

Four equally sized radiator panels were assumed consistent with the baseline. The four radiators were located in each quadrant of the intertank compartment forward of and clocked 45° from the APS modules. The four panels, located as shown in Figure 6-1, reduce the effects of plume heating from the APS modules and minimize attitude influences from external heating. The apparent choice of a hydrazine APS configuration provides one of the more significant changes from earlier configurations (Ref 8), and will reduce the plume heating on the radiators to levels experienced on the Titan IIIC Transtage vehicle. These levels did not impair the radiator performance in seven flights of that vehicle.

The thermal environments were evaluated to determine the worst-case design environments for use in the radiator design. The cold-case design conditions were obvious, because at synchronous altitude the earth emitted and albedo is near zero and the Tug could be aligned with the sunline to result in no heat being applied to the radiator panels. The case 4 park orbit, $\beta = 52^\circ$, resulted in slightly higher incident fluxes than the other cases studied and was chosen for the hot case. The vehicle orientation maximized absorbed heating when two radiators were exposed to the sun and when the included angle between the center of each radiator and the sun line was 45° as shown in Figure 6-1.
Figure 6-1 Tug Exterior
The maximum heat load to be rejected was used with the hot-case thermal environment and minimum heat load to be rejected was used with the cold-case environment to obtain the thermal design conditions. These conditions are consistent with orbital altitude requirements of 296 to 35,750 km (160 to 19300 n mi) with no attitude constraints.

Operationally the fuel cell was assumed to be activated in orbit before the Tug and payload were released by the orbiter. The fuel cell was also assumed to be deactivated before the Tug was remated to the orbiter. This sequence of events was sufficient to permit the fluid system to be designed without interfacing with the orbiter for thermal control. The potential for a fluid loop failure during a 7-day mission was considered sufficient for adding a redundant fluid loop. Each loop was designed to carry the full heat load. In addition, the radiators were used to provide micrometeorite protection for the fluid lines.

The fuel cell system shown in Figure 6-2 was obtained from Reference 13. The fuel cell generates waste heat, which is removed by a fluid loop. The coolant temperature control valve, pump, and interconnecting lines are an integral part of the system. Cell performance is predicated on maintaining the coolant through the fuel cell in a narrow temperature range independent of the electrical load. The primary parameters are control of the inlet temperature to 355.4 K (180°F) ± 0 K and limiting the temperature rise through the cell to 5.6 K (10°F) under maximum load conditions. Figure 6-3 presents the waste heat rejection as a function of electrical load with the design conditions shown. The warmup heater shown in Figure 6-2 is used to heat the fluid and the cell to the operating temperature level during the activation period and is not used during the normal operational period. Pratt and Whitney suggests the use either water or FC-43 as the working fluid on the fuel cell side of the interface. FC-43 was used in the simulations; however, water could have been used because the interface temperatures chosen in the study will not result in freezing temperatures.

The reactants, H$_2$ and O$_2$, enter the cell as low-pressure gases and exit as slightly superheated steam at 355 K. The reactant consumption is presented in Figure 6-4. For this study the water vapor was assumed to be dumped continuously. However, payload contamination requirements could require a different approach. For example, the water could be stored in a tank after being condensed and dumped overboard during main engine burns, thus reducing the water vapor around the Tug during coast periods.
Flow Schematic

Regenerator Return

Coolant Temperature Control Valve

Warm-UP Heater

355.2°K

Purge Valves

H₂O Vent

Water Vent Regulator

2.758 N/m² (4 psia)

Coolant Pump

Accumulator

1.103 x 10⁵ N/m² (16 psi)

Coupled Reactant Pressure Regulator

360.9°K Max to Regenerator

Figure 6-2 Fuel Cell Flow Schematic
Figure 6-3 Waste Heat Rejection

Figure 6-4 Reactant Consumption
The fuel cell heat rejection fluid loop is presented in Figure 6-5, which shows a single fluid loop through the thermal control valve and the radiators. The schematic is presented in this manner for clarity purposes only, and should be interpreted as having a redundant loop. The regenerator was considered to be a single unit with a redundant secondary loop.

The four radiator panels are in series with tubes on each panel in series, thus the series-series description. The radiators are similar in design to the Transtage radiator using the P-tube rail concept, Figures 6-5 and 6-6, details A and B, which allows two P-tubes to be attached to a single rail. Each panel has two continuous P-tubes from inlet to outlet with the flange removed in the bend and rail crossover areas. This concept minimizes the number of fluid connections and potential leakage points. The concept also provides micrometeorite protection.

The fluid is bypassed around the radiators, Figure 6-5, as the return fluid temperature drops below a predetermined level, 333°K (140°F). The thermal control valve was envisioned as a mechanically actuated valve using an electronic controller that senses the mixed fluid temperature going to the regenerator, $T_3$, and controls 333°K (140°F). This temperature was selected to meet the heat rejection requirements while minimizing radiator area. A lower temperature would also result in lower flowrates through the radiator in the cold case coupled with lower fluid temperatures. The pump was located on the outlet side of the regenerator to maximize the fluid temperature entering the radiators in the cold case. Freon E-1, the chosen working fluid, was developed primarily to yield heat transport properties similar to Freon 21 while eliminating the compatibility problems of that fluid. The cold-case results, discussed later, indicate that a heater is not required to avoid excessively cold fluid temperatures.

The system results in a relatively constant headrise requirement on the pump because the system pressure drop should remain relatively constant. Flow trimming problems experienced on parallel flow systems are avoided with the series configuration. One concern with this design is the confirmation of the transitional flow characteristics of a single panel. Although past radiator designs have been based on a turbulent or laminar operation, the Tug radiator was designed to operate through the transition region with Reynolds numbers ranging from 27,000 in the hot case to 600 in the cold case.

Two advantages of the bypass radiator design are the limited pressure drop and reduction in heat transfer coefficient as the fluid is cooled. The maximum pressure drop through the radiators occurs at full flow when the fluid is at its high temperature and is reduced as flow is bypassed around the panels.
Figure 6-5 Fuel Cell Heat Rejection
Figure 6-6 Radiator Details
Ideally, the radiator designer desires high heat transfer coefficients at maximum heat load conditions and minimum coefficients at minimum heat load conditions. This allows the total panel area to be minimized while limiting the minimum fluid temperature. The transitional flow design permits the designer to accomplish this.

This design assumes predictable operation over the above Reynolds number range using data Colburn presented in 1936 (Ref 12). The Transtage radiator was designed to operate down to Reynolds numbers of 7000 however, the complete transition region was not explored. Successful Skylab Airlock Module radiator operation was demonstrated up to Reynolds numbers of 2500. A verification test of a single panel is needed to confirm the design philosophy considered here. A further discussion follows in the cold case results discussion.

6.1 RADIATOR MODELING TECHNIQUES

A 79-node thermal model using variable material and fluid properties to evaluate the system performance was developed. Heat transfer coefficients were evaluated for each individual radiator tube. Classical heat exchanger theory was applied in evaluating the regenerator performance.

The tube heat transfer coefficients were obtained using the Colburn J-Factor method discussed in Reference 12. Figure 6-7 was obtained from Reference 12, page 394, which relates the Colburn J-Factor to Reynolds number. The Colburn J-Factor is related to the heat transfer coefficient by the equation:

\[ J = \left( \frac{h_c}{\rho C_p V} \right) N_p^{2/3} \left( \frac{\mu_f}{\mu_w} \right)^{0.14} \]

where

\( C_p \) = fluid specific heat
\( V \) = fluid velocity in tube
\( \rho \) = bulk fluid density
\( N_p \) = Prandtl Number
\( \mu_f \) = bulk fluid viscosity
\( \mu_w \) = fluid viscosity at the tube wall
Figure 6-7 Colburn J-Factor vs Reynolds Number
\[ J = \text{Colburn J-Factor} \]
\[ k = \text{fluid conductivity} \]
\[ N_R = \text{Reynolds number} \]
\[ D = \text{tube internal diameter} \]
\[ h_c = \text{heat transfer coefficient to tube} \]

Solving for \( h_c \)

\[ h_c = Jk N_R N_p^{1/3} \left( \frac{\mu_w}{\mu_f} \right)^{0.14/D} \]

A subroutine with this equation was used in calculating the heat transfer equation and applying it to the model. Inherent in the subroutine was another technique used in evaluating radiator designs at the Denver Division for several years. This technique is directly adaptable to the finite differencing technique used by most thermal analyzer programs. Consider fluid flowing through a single tube and further consider this to be a part of a parallel flow heat exchanger.

The heat balance on the tube is governed by the following equations:

\[ \text{fluid } Q = \dot{\omega} C_p \left( T_{\text{in}} - T_{\text{out}} \right) \]
\[ \text{tube } Q = \varepsilon \dot{\omega} C_p \left( T_{\text{in}} - T_w \right) \]
\[ \dot{\omega} C_p \left( T_{\text{in}} - T_{\text{out}} \right) = \varepsilon \dot{\omega} C_p \left( T_{\text{in}} - T_w \right) \]

where

\[ Q = \text{heat rate} \]
\[ \dot{\omega} = \text{mass flow rate} \]
\[ C_p = \text{specific heat of the fluid} \]
\[ T_{\text{in}} = \text{fluid inlet temperature} \]
\[ T_{\text{out}} = \text{fluid outlet temperature} \]
\[ T_w = \text{tube wall temperature} \]
\[ \varepsilon = \text{heat exchanger effectiveness} \]
solving for \( T_{\text{out}} \)

\[ T_{\text{out}} = (1-\varepsilon) T_{\text{in}} + T_w \]

For a parallel flow heat exchange the effectiveness is

\[
\varepsilon = \frac{1 - e^{-\text{NTU} \left(1 + \frac{C_{\min}}{C_{\max}}\right)}}{1 + \frac{C_{\min}}{C_{\max}}} \]

where

\[ C = \frac{\Delta C}{p} \]

\( C_{\min} \) = the minimum enthalpy flow

\( C_{\max} \) = the maximum enthalpy flow

\( \text{NTU} \) = number of heat exchanger units = \( h_c \frac{A}{C_{\min}} \)

If the tube wall were assumed to be a constant temperature, the enthalpy flow outside the tube would approach infinity of \( C_{\max} \approx \infty \) hence \( C_{\min}/C_{\max} = 0 \). The above equation reduces to

\[ \varepsilon = 1 - e^{-h_c \frac{A}{C_{\min}}} \]

Having solved for \( T_{\text{out}} \) in terms of \( T_{\text{in}} \) and \( T_w \) and determined \( \varepsilon \), the finite difference equation was reviewed.

\[
T_{A} = \frac{\sum_{j=1}^{n} G_{j-A} T_j}{\sum_{j=1}^{n} G_{j-A}}
\]
where

\[ T_A = \text{temperature of node A} \]

\[ G_{j-A} = \text{conductance from node j to node A} \]

\[ T_j = \text{temperature of node j} \]

\[ n = \text{number of nodes conducted to node A} \]

The finite differencing equations would therefore solve for \( T_{out} \) in the following manner:

\[
(1 - \varepsilon) T_{in} + \varepsilon T_w = \frac{(1 - \varepsilon) T_{in} + \varepsilon T_w}{1 - \varepsilon + \varepsilon}
\]

which reduces to Equation [1].

The network for Equation [3] is:

The tube equation is satisfied by adding the additional conductor to the network between \( T_{in} \) and \( T_w \).

Hence the subroutine calculated the above network for each of the 16 radiator tubes, impressing the appropriate conductor values in the thermal network each iteration. In addition, the Reynolds numbers, Colburn J-Factors, and heat transfer coefficients were saved for printout purposes.
To complete the radiator evaluation, the fin effectiveness was evaluated by the following equation obtained from Reference 14.

\[ \mu_F = \frac{T \text{ANH}}{2L_F} \sqrt{\frac{\varepsilon \sigma T_R^3}{k \delta}} \]

\[ \frac{2L_F}{\sqrt{\frac{\varepsilon \sigma T_R^3}{k \delta}}} \]

where

- \( T_R \) = Fin root or rail root temperature
- \( \varepsilon \) = Surface emissivity
- \( k \) = Conductivity of the fin
- \( \delta \) = Fin thickness
- \( L_F \) = Fin width
- \( \sigma \) = Stefan-Boltzman constant

Solving for the root temperature in the model the fin heat radiated is determined by

\[ Q = \sigma A e^{\mu_F} T_R^4 \]

6.2 REGENERATOR SIZING

The regenerator was sized using the effectiveness approach described in Reference 15. For a counter flow heat exchanger, the effectiveness is defined as

\[ \varepsilon = \frac{1 - e^{-NTU}(1 - C_{\min}/C_{\max})}{1 - (C_{\min}/C_{\max})e^{-NTU}(1 - C_{\min}/C_{\max})} \]
where

\[ \text{NTU} = \text{number of heat transfer units} = \frac{\text{UA}}{C_{\text{min}}} \]

\[ C = \omega C_p \]

\[ C_{\text{min}} = \text{minimum } \omega C_p \]

\[ C_{\text{max}} = \text{maximum } \omega C_p \]

\[ A = \text{heat transfer area} \]

\[ U = \text{overall heat transfer coefficient} \]

NTU was evaluated by assuming that on an individual iteration basis the fuel cell fluid loop was at steady state. This agrees with the use of arithmetic nodes to simulate the fluid. With that assumption it follows that the heat dissipated by the fuel cell must be transferred through the regenerator. Using the previous iterations regenerator \( \Delta T \), the UA was calculated by the following equation.

\[ \text{UA} = \frac{Q}{\Delta T} \]

NTU was derived from the above equation after determining the minimum of the hot and cold side \( \omega C_p \) values.

6.3 RADIATOR PRESSURE DROP

Radiator pressure drop was evaluated directly from the following equation which was obtained from References 12 and 16.

\[ J = \frac{f}{8} \]

or

\[ f = 8J \]

where

\[ J = \text{Colburn J-factor} \]

\[ f = \text{friction factor} \]
\[ \Delta P = 8J \frac{Lp}{D} \frac{V^2}{2g_c} \]

where

- \( L \) = tube length
- \( D \) = tube internal diameter
- \( p \) = fluid density
- \( V \) = average fluid velocity
- \( g_c \) = gravity term

Substituting the velocity with the continuity equation:

\[ \Delta P = 8J \frac{L}{D} \frac{p}{2g_c} \left( \frac{\dot{m}}{\dot{A}} \right)^2 \]

where

- \( A \) = internal tube cross sectional area
- \( \dot{m} \) = fluid mass flow rate

Pressure drops for tube bends were evaluated using the above equation modified for equivalent L/D ratios obtained from Reference 16.

### 6.4 FUEL CELL MODEL

The fuel cell was modelled and integrated with the radiator model. The model schematic is shown in Figure 6-8.

Table 6-1 describes the nodes of the fuel cell model.

The conductor values used were temperature-dependent based upon FC-43 as the working fluid and were one-way conductors. The system mass flow was 5.75 kg/minute (12.67 lb/minute). The use of water in this loop would reduce the mass flow in proportion to the specific heat ratio.
Figure 6-8 Fuel Cell Model

Table 6-1 Fuel Cell Model Node Description

<table>
<thead>
<tr>
<th>Node</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>Fuel Cell</td>
</tr>
<tr>
<td>201</td>
<td>Pump</td>
</tr>
<tr>
<td>202</td>
<td>Fluid Node</td>
</tr>
<tr>
<td>203</td>
<td>Regenerator Inlet - Fluid</td>
</tr>
<tr>
<td>204</td>
<td>Regenerator Outlet - Fluid</td>
</tr>
<tr>
<td>205</td>
<td>Coolant Temperature Thermal Control Valve - Fluid</td>
</tr>
<tr>
<td>206</td>
<td>Bypass Fluid and Fast Warm-up Heater</td>
</tr>
<tr>
<td>207</td>
<td>Boundary Temperature</td>
</tr>
</tbody>
</table>

Heat

Q 200  Fuel cell heat dissipation function of electrical load
Figure 6-3

Q201  Pump heat dissipation - 30 watts constant.
The coolant temperature control valve was simulated by a linear curve assuming 100% flow through the regenerator at 356°C (181°F) and 10% flow at 353°C (176°F) regenerator outlet temperatures. The control range used was smaller than the 5.5°C (10°F) range obtained from Pratt and Whitney. The range was reduced to provide better control of the fuel cell and was based on experience with wax plug designs that tend to control in the range used. Pratt and Whitney also stated that the design of the valve is such that the minimum regenerator flow is 5 to 10% at the lower allowable fluid temperatures.

Node 207 was used as a boundary node to remove heat from the fluid using the regenerator equations and the following equations. The effectiveness is related to the heat flow by

\[ \varepsilon = \frac{q}{q_{\text{max}}} = \frac{C_h \left( T_{\text{h in}} - T_{\text{h out}} \right)}{C_{\text{min}} \left( T_{\text{h in}} - T_{\text{c in}} \right)} \]

\[ = \frac{C_c \left( T_{\text{c out}} - T_{\text{c in}} \right)}{C_{\text{min}} \left( T_{\text{h in}} - T_{\text{c in}} \right)} \]

where

- \( q \) = heat flow from hot to cold side
- \( q_{\text{max}} \) = maximum heat flow for \( \varepsilon = 1 \)
- \( C_h \) = \( \omega C_p \) for the hot side fluid (fuel cell)
- \( C_c \) = \( \omega C_p \) for the cold side fluid (fuel cell heat rejection system)
- \( C_{\text{min}} \) = minimum of \( C_h \) and \( C_c \)
- \( T_{\text{h in}} \) = fluid hot side inlet
- \( T_{\text{h out}} \) = fluid hot side outlet
- \( T_{\text{c in}} \) = fluid cold side inlet
- \( T_{\text{c out}} \) = fluid cold side outlet

Solving for \( q \)

\[ q = \varepsilon \frac{C_{\text{min}} \left( T_{\text{h in}} - T_{\text{c in}} \right)}{} \]
Using arithmetic nodes to simulate the fluid implies that the heat generated must be removed from the system because the nodes are relaxed to steady state each iteration. The heat stored in the fuel cell and pump, nodes 200 and 201, was not considered due to expected small variations from one iteration to the next. The sum of \( Q_{200} \) and \( Q_{201} \) was used along with an assumed 5.6°K (10°F) temperature drop of the fuel cell fluid through the regenerator to calculate the UA term thus enabling the effectiveness to be calculated. Node 207 was set by the maximum temperature difference.

\[
T_{207} = T_{204} - (T_{\text{h in}} - T_{\text{c in}})
\]

or

\[
= T_{204} - (T_{203} - T_{34})
\]

where \( T_{34} \) was the cold side inlet temperature.

The above equation defining \( q \) was satisfied by substituting the individual temperatures.

\[
q = \varepsilon C_{\text{min}} \left( T_{204} - T_{207} \right)
\]

\[
= \varepsilon C_{\text{min}} \left( T_{204} - T_{204} + (T_{\text{h in}} - T_{\text{c in}}) \right)
\]

hence

\[
q = \varepsilon C_{\text{min}} (T_{\text{h in}} - T_{\text{c in}})
\]

6.5 FUEL CELL HEAT REJECTION SYSTEM MODELING

The control portion of the fuel cell heat rejection system fluid loop was modelled as shown in Figure 6-9.

Table 6-2 presents a description of the nodes contained in Figure 6-9.
Figure 6-9
Fuel Cell Heat Rejection System
Flow Control Loop Model

Table 6-2 Radiator Control Loop

<table>
<thead>
<tr>
<th>Node</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Radiator fluid inlet temperature</td>
</tr>
<tr>
<td>32</td>
<td>Radiator fluid outlet temperature</td>
</tr>
<tr>
<td>33</td>
<td>Thermal control valve outlet temperature</td>
</tr>
<tr>
<td>34</td>
<td>Regenerator inlet temperature</td>
</tr>
<tr>
<td>35</td>
<td>Regenerator outlet temperature</td>
</tr>
<tr>
<td>36</td>
<td>Pump outlet temperature</td>
</tr>
<tr>
<td>37</td>
<td>Fluid temperature</td>
</tr>
<tr>
<td>208</td>
<td>Boundary temperature</td>
</tr>
</tbody>
</table>

Heat was applied to node 36 as $Q_p$ which was set at 51 watts. These data were derived from a Block II Apollo pump with Freon E-1 as the working fluid. The conductor values were obtained using temperature varying properties and represent the mass flow times specific heat. The pump flow was held constant at 1.81 kg/minute (4 lb/minute). Node 208 was used to add the heat removed from the fuel cell loop to the radiator loop and was evaluated by the following equation

$$T_{208} = T_{35} + (T_{h \text{ in}} - T_{c \text{ in}})$$

$$= T_{35} + (T_{203} - T_{34})$$

The conductor value between nodes 35 and 208 was set equal to $C_{\text{min}}$.  

6-20
The radiator model is presented for a single panel. Each panel was modelled individually and integrated into the complete model. The first panel in the loop is shown in Figure 6-10.

\[
Q_E = \text{External Heating}
\]

Figure 6-10 Radiator 1 Nodal Diagram

Nodes 1 through 9 and node 37 represent fluid nodes. Node 9 was equivalent to node 1 on panel number 2 with the entire numbering sequence contained. The series of nodes beginning with 41 were tube wall nodes while the nodes beginning with 61 were rail root nodes. Node 460 was the boundary node representing the space sink temperature of 0°C. The fluid conductors between radiator rails were \( \omega C_p \) values. The tube-to-rail root and rail root-to-rail root conductors were handled as linear conductors. \( Q_E \) represents the application absorbed external heating.

6.6 HOT-CASE PERFORMANCE

The hot-case analysis was performed to size the area of the radiators and regenerator performance for the maximum external heating and maximum heat load condition. The results of the study
resulted in the radiator being sized to 8.05 m$^2$ (22 ft$^2$) or 2.01 m$^2$ (5.5 ft$^2$) per panel. The regenerator requirements derived from the analysis indicated that an effectiveness of 0.90 or greater was achievable. Table 6-3 presents the conditions used. As previously discussed, the maximum external environment was obtained from flux case 4 in park orbit and was a transient environment. The use of higher inclination angle orbits would require resizing the radiator area for a constant solar exposure in near-earth orbit.

Table 6-3 Hot-Case Radiator Design Conditions

<table>
<thead>
<tr>
<th>Maximum External Heating</th>
<th>Flux Case 4 Park Orbit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Attitude</td>
<td>Sun Normal to Tug Longitudinal Axis</td>
</tr>
<tr>
<td></td>
<td>Two Radiators Exposed to Sun 45° from Sun Line</td>
</tr>
<tr>
<td>Maximum Electrical Load</td>
<td>1500 Watts</td>
</tr>
<tr>
<td>Maximum Heat Load</td>
<td>744 Watts Plus 81 Watts for Pumps</td>
</tr>
</tbody>
</table>

A radiator coating selection study was pursued where primary requirements for screening were a low $\alpha/c$, demonstrated stability of the properties, ease of application, ease of maintenance, and durability. White paints were eliminated by most of the above considerations. Optical solar reflectors (OSR) were deleted due to anticipated problems with handling and maintenance. Silver-coated teflon tape was selected because of its favorable optical property values, stability, ease of application, and maintenance. The properties used to represent silver Teflon in the analysis were $\alpha = 0.09$ and $c = 0.76$, obtained from Reference 17.

Figures 6-11 thru 6-13 present key temperatures of the fuel cell loop, the regenerator inlet, regenerator outlet, and coolant temperature control valve outlet temperatures, respectively. The first temperature peak is due to the initial temperature in the radiators being set at 355.4 K (180°F). Hence, the first half hour of the simulation was used to gain control of the system. Most of this time was used to allow the control valve to respond; the valve was not allowed to change more than 0.5% of full flow from one iteration to the next. This logic was to limit the valve cycling. The same logic was also applied to the radiator loop thermal control valve. The resultant regenerator inlet temperature was 359.73 K (187.75°F), as shown in Figure 6-11, while the outlet of the regenerator was 344.54 K (160.5°F), as shown in Figure 6-12. The 0.1-hour output interval accounts for the seemingly jagged minor peaks in the curves indicating some minor cycling of the coolant control valve temperature at 353.7 K (177°F), as shown in Figure 6-13. Hence, the system was controlled within the desired temperature limits under maximum heating and load conditions.
MIN TEMP OF 187.681 OCCURRED AT TIME 4.810
MAX TEMP OF 190.355 OCCURRED AT TIME 4.410

MIN TEMP OF 180.192 OCCURRED AT TIME 4.910
MAX TEMP OF 180.000 OCCURRED AT TIME 4.310

FIGURE 6-11 RADIATOR HOT CASE
FIGURE 6-12 RADIATOR HOT CASE
Figures 6-14 thru 6-16 represent the fluid inlet temperature to the radiators, fluid outlet temperature from the radiators, and the radiator loop thermal control valve outlet temperature, respectively. Figure 6-16 also represents the regenerator cold side inlet temperature and demonstrates control at the desired 333°K (140°F). Figure 6-17 presents the regenerator cold side outlet temperature. Figure 6-18 presents the heat rejected by the radiator fluid loop and Figure 6-19 presents the net heat radiated from the four radiator panels. The net heat rejected was evaluated by summing the total heat radiated from the panels and subtracting the summation of the absorbed heating rates. Figure 6-20 presents the radiator fluid mass flow, which ranged from 1.772 to 1.322 kg/minute (3.904 to 2.914 lb/minute). As shown the maximum system flow was 1.814 kg/minute (4 lb/minute. The maximum radiator flow of 1.772 kg/minute provides a 2% margin in flow in the hot case after the initial temperature transient.

Figure 6-21 presents the heat flow across the regenerator, which averaged 809 watts (2763 Btu/hour). The fuel cell loop flow through the regenerator, Figure 6-22, averaged 2.31 kg/minute (5.1 lb/minute), while the system capability was 5.75 kg/minute (12.67 lb/minute) as recommended by Pratt and Whitney. Based upon these results, the fuel cell loop flow could be reduced to 2.72 kg/minute (6 lb/minute) with adequate margin maintained.

The Reynolds numbers, Colburn J-Factors, heat transfer coefficients, and radiator pressure drop in the hot case were influenced by the tube L/D chosen for cold-case performances. With an L/D of 200 the hot-case parameters varied as shown in Table 6-4.

The radiator fin effectiveness varied between 0.908 to 0.923 for rail root temperatures of 354.2 to 330.2°K (177.8 to 134.6°F) at the maximum flow condition and 0.909 to 0.929 for rail temperatures of 352.3 to 320.8°K (174.5 to 117.7°F) at minimum flow conditions.

In reality, the hot-case electrical load on the fuel cell would occur during a main engine burn, which would result in the vehicle being oriented to the proper attitude before the burn. This required burn attitude would probably result in external heating rates less than the hot-case environment, which would yield more radiator performance margin than indicated. Further, the maximum load would be a relatively short interval, on the order of 200 to 300 seconds.
FIGURE 6-13. RADIATOR HOT CASE

FIGURE 6-14. RADIATOR HOT CASE
FIGURE 6-15 RADIATOR HOT CASE

TEMP NODE NO. 32 Radiator Fluid Outlet Temperature
MIN TEMP OF 121.200 OCCURRED AT TIME 5.310
MAX TEMP OF 180.000 OCCURRED AT TIME 4.310

FIGURE 6-16 RADIATOR HOT CASE

TEMP NODE NO. 33 Radiator Loop Thermal Control Valve Fluid Outlet Temperature
MIN TEMP OF 139.792 OCCURRED AT TIME 4.810
MAX TEMP OF 145.371 OCCURRED AT TIME 4.910
FIGURE 6-17. RADIATOR HOT CASE

MISSION TIME - HOURS

TEMP NODE NO. 35 Regenerator Fluid Outlet Temperature
MIN TEMP OF 180.000 OCCURRED AT TIME 4.310
MAX TEMP OF 198.801 OCCURRED AT TIME 4.410

FIGURE 6-18. RADIATOR HOT CASE

MISSION TIME - HOURS

NODE NO. 300 Radiator Fluid Heat Rejection
MIN OF 0.0 OCCURRED AT TIME 4.310
MAX OF 307.005 OCCURRED AT TIME 4.410
FIGURE 6.19   RADIATOR HOT CASE

- TEMPERATURE NODE NO. 301  Radiator Net Heat Radiated
  - MIN TEMP 2846.932 OCCURRED AT TIME 9.910
  - MAX TEMP 3537.411 OCCURRED AT TIME 4.310

FIGURE 6.20   RADIATOR HOT CASE

- TEMPERATURE NODE NO. 302  Radiator Loop Fluid Mass Flow
  - MIN TEMP 2.913 OCCURRED AT TIME 5.310
  - MAX TEMP 4.000 OCCURRED AT TIME 4.310
MISSION TIME - HOURS

TEMP NODE NO. 303  Regenerator Heat Flow
MIN TEMP OF 2761.49°C OCCURRED AT TIME 7.710
MAX TEMP OF 3070.79°C OCCURRED AT TIME 4.310

FIGURE 6-21  RADIATOR HOT CASE

MISSION TIME - HOURS

TEMP NODE NO. 304  Fuel Cell Loop Fluid Mass Flow
MIN TEMP OF 5.048°C OCCURRED AT TIME 4.910
MAX TEMP OF 12.421°C OCCURRED AT TIME 4.310

FIGURE 6-22  RADIATOR HOT CASE
Table 6-4 Radiator Parameters

<table>
<thead>
<tr>
<th>Flow</th>
<th>Reynolds Numbers</th>
<th>Heat Transfer Coefficient</th>
<th>Pressure Drop</th>
</tr>
</thead>
<tbody>
<tr>
<td>kg/minute</td>
<td>inlet/outlet</td>
<td>inlet/outlet</td>
<td>N/m²</td>
</tr>
<tr>
<td>(lb/minute)</td>
<td></td>
<td>watt/m² °K (Btu/hr-ft²-oF)</td>
<td>(psi)</td>
</tr>
<tr>
<td>1.772 (3.906)</td>
<td>27655/20561</td>
<td>0.00400</td>
<td>1.75058 x 10⁵</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1117/946 (680/576)</td>
<td>(25.39)</td>
</tr>
<tr>
<td>1.322 (2.914)</td>
<td>20444/14553</td>
<td>0.00467</td>
<td>9.84364 x 10⁴</td>
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<tr>
<td></td>
<td></td>
<td>842/705 (512/429)</td>
<td>(14.277)</td>
</tr>
</tbody>
</table>

COLD-CASE PERFORMANCE

The cold-case analysis was performed to verify that the radiator system performance was adequate in a minimum external heating environment with a minimum heat rejection requirement. For this case the heat load was reduced to 281 watts from the fuel cell, which results from a 600 watt electrical load. The external environment was reduced to no external heating being applied to the radiators, which would result from the vehicle longitudinal axis aligned to look at the sun.

The predicted radiator performance indicated that this environment could be flown under minimum heat load conditions without experiencing excessively cold fluid temperatures. The radiator flow was controlled at 12% of full flow.

Figures 6-23, 6-24, and 6-25 present the major fuel cell fluid temperatures. Figures 6-23 and 6-24 present the fluid inlet and outlet temperatures for the fuel cell side of the regenerator. Figure 6-25 presents the coolant temperature control valve outlet temperature. As discussed in the hot case, the high flow in this loop and the restricted response of the coolant temperature control valve resulted in the negative peak in fluid temperatures. The inlet to the fuel cell was maintained at 352.8°K (175.3°F).
TEMP NODE NO. 203 Fuel Cell Loop Regenerator Inlet Temp.
MIN TEMP OF 130.153 OCCURRED AT TIME 0.300
MAX TEMP OF 190.000 OCCURRED AT TIME 0.

TEMP NODE NO. 204 Fuel Cell Loop Regenerator Outlet Temp.
MIN TEMP OF 108.265 OCCURRED AT TIME 0.300
MAX TEMP OF 180.000 OCCURRED AT TIME 0.

FIGURE 6-23 RADIATOR COLD CASE

FIGURE 6-24 RADIATOR COLD CASE
TEMP NODE NO. 205 Fuel Cell Loop Coolant Temperature Control
MIN TEMP OF 155.315 OCCURRED AT TIME .300
MAX TEMP OF 175.316 OCCURRED AT TIME 2.000

FIGURE 6-25 RADIATOR COLD CASE

TEMP NODE NO. 1 Radiator Fluid Inlet Temperature
MIN TEMP OF 121.023 OCCURRED AT TIME .300
MAX TEMP OF 180.000 OCCURRED AT TIME 0.

FIGURE 6-26 RADIATOR COLD CASE
The radiator inlet temperature showed a similar negative peak with a resultant temperature of 344.3 K (160°F), Figure 6-26. Of major interest in this run was the radiator outlet fluid temperature, as shown in Figure 6-27, which leveled out at 227.6 K (−50°F). For the chosen fluid in the radiator loop, Freon E-1, this temperature is well above the freezing temperature of 119 K (−246°F). Figure 6-28 presents the radiator thermal control valve outlet temperature and shows that control was achieved as desired at just under 333 K (140°F). The negative peak shown in the figure also resulted from restricting the valve response. This figure also corresponds to the regenerator cold side inlet temperature. Figure 6-29 shows the regenerator outlet temperature was maintained at 343 K (158°F).

Figures 6-30 and 6-31 present the heat rejected from the radiator loop fluid and by radiation from the radiators. Figure 6-32 presents the radiator mass flow with control maintained at 0.215 kg/minute (0.474 lb/minute), which represents 12% of full flow.

Figure 6-33 presents the heat flow across the regenerator and Figure 6-34 presents the fuel cell mass flow through the regenerator.

The flow through the radiators resulted in Reynolds Numbers ranging from 2554 at the inlet to 589 at the outlet. This represents flow in the lower end of the transition region to fully developed laminar flow. The Colburn J-Factors derived from Figure 6-7 ranged from 0.0024 at the inlet to 0.0084 at the outlet, with the minimum of 0.0021 achieved in the fourth tube of the first panel. Correspondingly the heat transfer coefficients ranged from 238 \( \text{watts} \, \text{meter}^{-2} \, \text{K} \) (42 \( \text{Btu/hr-ft}^2\text{-°F} \)) at the inlet to 324 \( \text{watts} \, \text{meter}^{-2} \, \text{K} \) (57 \( \text{Btu/hr-ft}^2\text{-°F} \)) at the outlet. The minimum coefficient, 173 \( \text{watts} \, \text{meter}^{-2} \, \text{K} \) (30.5 \( \text{Btu/hr-ft}^2\text{-°F} \)) was in the fourth tube. The pressure drop through the radiators was 2096 N/m² (0.304 psi). The low pressure drop illustrates one of the desirable features of the bypassed radiator design, which allows low pressure drops in the radiator loop during cold fluid conditions while achieving essentially a constant pressure drop in the pump loop.

The transitional flow through the radiators permits the fluid to be decoupled slightly from the radiators, thus allowing warmer fluid temperatures and higher flow rates to be maintained. The Colburn J-Factor approach to radiator design has not been pursued to any great extent by the industry except on the Transtage radiators, which have experienced seven successful flights. The Transtage design did not, however, require the full transition region to satisfy the design requirements operating down to Reynolds numbers of 7000. The Airlock Module radiator on Skylab was successfully and predictably operated at Reynolds numbers up to 2500. Hence, before pursuing the radiator design further, it would be desirable to conduct some breadboard level testing on a four-tube panel to explore and verify the heat transfer and pressure drop characteristics through the transition region.
FIGURE 6-27  RADIATOR COLD CASE

TEMP NODE NO.  32  Radiator Fluid Outlet Temperature

MIN TEMP OF  -51.996  OCCURRED AT TIME  1.000
MAX TEMP OF  180.000  OCCURRED AT TIME  0.

FIGURE 6-28  RADIATOR COLD CASE

TEMP NODE NO.  33  Radiator Loop Thermal Control Valve

MIN TEMP OF  99.216  OCCURRED AT TIME  1.000
MAX TEMP OF  140.539  OCCURRED AT TIME  0.600
FIGURE 6-29 RADIATOR COLD CASE

- TEMP NODE NO. 35
- Radiation Loop Regenerator Outlet Temp.
- MIN TEMP OF 117.921 OCCURRED AT TIME .300
- MAX TEMP OF 180.000 OCCURRED AT TIME 0.

FIGURE 6-30 RADIATOR COLD CASE

- NODE NO. 300
- Radiator Fluid Heat Rejection
- MIN OF 0. OCCURRED AT TIME 0.
- MAX OF 2102.452 OCCURRED AT TIME .100
FIGURE 6-31  RADIATOR COLD CASE

NODE NO.  301  Radiator Heat Radiated
MIN OF  1392.585 OCCURRED AT TIME .600
MAX OF  4352.077 OCCURRED AT TIME 0.

FIGURE 6-32  RADIATOR COLD CASE

NODE NO.  302  Radiator Loop Fluid Mass Flow
MIN OF  .347 OCCURRED AT TIME .500
MAX OF  4.000 OCCURRED AT TIME 0.
FIGURE 6-33 RADIATOR COLD CASE

NODE NO. 303 Regenerator Heat Flow
MIN OF 573.7 W OCCURRED AT TIME .400
MAX OF 2442.0 W OCCURRED AT TIME .0.

FIGURE 6-34 RADIATOR COLD CASE

NODE NO. 304 Fuel Cell Fluid Mass Flow
MIN OF 1.267 OCCURRED AT TIME .500
MAX OF 12.421 OCCURRED AT TIME .0.
SPECIFICATIONS

Design and performance parameters of the fuel cell heat rejection system are documented in the form of a specification and are presented in Appendix I.
7. FURTHER CONSIDERATIONS

The Tug design confronts the state of the art in several areas. Inherent in the Tug mission is the goal of maximizing the payload delivery and retrieval capability. This has resulted in significant minimum weight requirements being placed on all systems. When designing the structural system, structural designers have been forced to explore the extensive use of composite structural designs aimed at minimizing weight.

7.1 HONEYCOMB STRUCTURES

A honeycomb design for the forward skirt of Tug, for example, has been proposed by most investigators. While this appears to provide a minimum weight design, further tradeoffs are necessary before arriving at the preferred baseline. The past use of the aluminum skin stringer-longeron design, while being potentially heavier than the honeycomb design, has afforded the thermal designer a significant amount of flexibility. Use of the skin as a radiation skink for compartment heat dissipation was a simple and reliable means of achieving thermal control. However, the application of honeycomb designs in this area adds an unknown to the problem, and in some cases would result in significant thermal design problems.

Heat transfer through thin aluminum skin panels results in small temperature drops (<<1°K) and is usually considered to be zero. The honeycomb material represents two surfaces separated by a core material through which heat must be transferred. Depending on the core material and the bondline characteristics, large temperature drops can result when transferring the required heat. The use of high conductivity materials such as aluminum is required because the major mode of heat transfer through the honeycomb is via conduction. The use of fiberglass or other low conductivity materials would severely impact the internal compartmental temperature in the hot case and would require large holes in the skirt to allow heat to be dissipated in local areas. To achieve the required strength characteristics such a design would probably eliminate the weight advantages gained. Continued development of lightweight skirt structural concepts should include an evaluation of the thermal design impact that each concept might yield. One of the key requirements in a supporting thermal evaluation would be to determine experimentally the thermal characteristics of each candidate concept.
7.2 APS THERMAL DESIGN

The thermal design of the auxiliary propulsion system (APS) was not specifically investigated in this study. However, experience in the design and flight of the Transtage hydrazine attitude control system provides several guidelines. The selection of a hydrazine system for Tug will simplify the thermal design problem and will make it an integral part of limit cycling requirements of the system. The thruster module thermal design is the primary concern. Depending on the individual thruster design, heat is required to maintain the catalyst temperature at some minimum level to ensure that the desired minimum impulse can be delivered upon demand. The Transtage system used engine heat to maintain the catalyst bed temperatures above 450°C (850°F). Normal limit cycling of the engines required by the guidance system to maintain the required vehicle attitudes was sufficient to supply the major portion of required heat. Computer software was added to account for the fuel consumption over 10-minute periods, comparing that against predicted cold-case fuel consumption requirements. Shortage of the required cold-case fuel consumption in any 10-minute flight interval resulted in a burn of the required thruster to make up the difference. Hence, the design used the propellant consumption instead of heaters to satisfy module thermal design requirements. Further, definition of the Tug module and engine design will be required before a thermal design can be determined. Local application of high temperature fiberous insulation will be required.

The APS propellant storage and feed system will require insulation and thermostatically controlled heaters to eliminate propellant freezing. This should not represent a significant problem. In addition, the application of low conductance tank and feedline supports will be required.
Thermal control system specifications were developed for those problem areas that required the application of specific thermal control devices. It was not considered necessary to develop a specification for the use of insulation and/or heaters. The specifications are presented in the Appendixes to this report.

The fuel cell heat rejection system specification (Appendix I) outlines the basic system's thermal design requirements. Appendix II presents the louver specification for application to the thermal control of the battery.

Appendix III presents the specification for development of the forward compartment thermal design using circumferential heat pipes, louvers, and thermal conditioning panels. The panels will provide a means to control those equipment items with low duty cycles, such as the laser radar, its associated electronics, and the TV cameras. Mounting these equipment items with other equipment which operate throughout the mission will allow components to share heat, thus reducing heat power requirements. This also provides structural panels for mounting the equipment. The heat pipes avoid excessively high or low skin temperature during constant attitudes and further enable heat to be shared between the thermal conditioning panels.
9. FOLLOW-ON PLAN

Several areas were identified for future study and test to lead to an orderly development of the Tug vehicle. In a study of this nature as many questions are identified as are answered during the course of the study.

9.1 STUDY AREAS

As the avionics system evolves in the future, the power dissipation level is expected to change. This will require altering the paint pattern and possibly revising heater power for some components. Component placement and arrangement studies on the thermal conditioning louver panels is warranted to further develop this technique. Parametric studies investigating panel Q/A, equipment Q/A, component arrangement, matching of qualification requirements, proper mix of high and low duty cycle, and environment temperature ranges should be pursued to identify the capabilities and limitations of this concept. The APS thermal control will require some future investigations as that system evolves. The use of heater power to maintain the catalyst temperature may be required; however, the limit cycle pulsing of that system will contribute significantly to maintaining the desired temperatures. Early identification of timelines will be essential to develop the engine module thermal design.

9.2 TESTING

Breadboard testing in several areas of the Tug thermal design is warranted at this time. Two areas will be explored in the follow-on to this contract. The application of louvers to the thermal control of the battery is currently being examined along with the performance of a thermal conditioning panel that will be coupled with a heat pipe radiator. Thermal conditioning panel capabilities will be further demonstrated. The design of a variable conductance heat pipe radiator will be verified. The successful demonstration of the radiator design will lend confidence in the credibility of heat pipe systems to satisfy the fuel cell heat rejection system requirements.
The pumped fluid system described here deserves further attention. The proposed design requires some breadboard-level testing to verify the radiator's operation through the transition region. This testing will verify the techniques used in the analytical models for design and mission analysis.

Testing should also be performed to determine the effective thermal conductance through honeycomb skin panels. The major unknown is the influence of the two bondlines on the overall conductance. The data generated in the study indicate that the forward compartment thermal design is sensitive to this conductance. This could have a severe impact on the compartment design concept.

The forward compartment heat pipes were envisioned as single closed circular pipes. Current technology in heat pipes has generally been limited to relatively short pipes. One 4.6-m-diameter pipe has been built and tested (Ref 18). Continued development in this area is warranted.
The analysis has shown that thermal control of Tug, exclusive of the fuel cell, can be maintained through the use of surface coatings, heat pipes, insulation, and louvers. Components can be maintained within their temperature limits by using isolation mounts, surface coatings, multilayer insulation, and in some cases thermostatically controlled heaters. A second component thermal control approach using thermal conditioning panels was also investigated, which reduced the required heater power. Both hot and cold environments for a simulated Tug mission were used to analyze the thermal control techniques. The analysis was performed for no orientation constraints during the Tug mission, thus providing flexibility in satisfying future payload requirements.

The transient analysis of the forward compartment used a paint pattern \( \alpha/\varepsilon = 0.5 \) derived from the steady-state parametric studies using 800 watts of internal power. However, initial transient analyses resulted in both hot and cold problems, with a high power (187 watts) tape recorder which had a narrow operating range of 289°K to 314°K (60°F to 105°F). A tape recorder that dissipated 8.4 watts was substituted. With the new power level for the tape recorder, the actual average power dissipation for the forward compartment was reduced to approximately 600 watts. Based upon this power level a new value of \( \alpha/\varepsilon = 0.60 \) is necessary to maintain the temperature level of the forward compartment at 297°K (75°F). This would replace the original \( \alpha/\varepsilon = 0.2375/0.475 = 0.50 \). An \( \alpha/\varepsilon \) of 0.6 is obtainable using an \( \alpha \) of 0.24 and \( \varepsilon = 0.40 \). This results in a paint pattern ratio of aluminum to white equal to 75% to 25%.

In addition to the high-power tape recorder that was subsequently replaced with a tape recorder of moderate power, other components were marginally acceptable in regards to their temperature limits. These include the laser radars and the laser radar electronics. These components have a very high lower temperature limit in both the operational and storage phases of the mission (operational minimum = 293°K (68°F), maximum = 323°K (122°F); storage minimum = 288.7°K (60°F), maximum = 323°K (122°F)). A large amount of heater power is required to maintain their temperatures, even in the hot case. Heater power for these components for the hot case included 84 watts for each of the laser radars and 65 watts for each of these four components while the rest of the components require less than 5 watts for this case. This indicates that these particular components should be requalified to temperatures more in line with the rest of the forward compartment components or additional thermal design features incorporated into individual components.
Many components exceeded their lower temperature limits in the cold-case simulation. However, this simulation used an unusually cold environment. This environment occurs only if the Tug longitudinal axis is maintained parallel to the solar vector and there is no significant planetary or albedo flux (i.e., Tug in a geosynchronous orbit). All of these component problems could be solved with additional heater power, further component isolation, and altering paint patterns. However, this reduces the flexibility of the design by making the component temperatures approach their upper limit in a hot case.

An alternative to the complex task of optimizing the isolation and heater power of each component is a new component mounting concept. In this concept, by grouping individual components with regard to electrical power output duty cycle and temperature limits on thermal conditioning panels, a reduction in heater power requirements in both hot and cold conditions can be obtained.

The thermal conditioning panels (see Appendix III) are mounting panels containing integral heat pipes and provide a means of obtaining an isothermal condition. Components are hard mounted to one side of the panel with a louver system on the other. The louvered side faces the compartment wall, which is maintained at a uniform temperature by circumferential heat pipes. The panel temperature is primarily controlled by the modulation of the temperature-sensitive louver blades. This concept offers a passive means of component control by allowing excess electrical power generation to be shared in maintaining other nonoperating components on the panel above their lower temperature limit.
11. REFERENCES


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APPENDIX I

SPACE TUG FUEL CELL HEAT REJECTION SYSTEM
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SYSTEM DESCRIPTION

The Space Tug fuel cell heat rejection system provides the means of maintaining the primary electrical power system, the fuel cell, within the desired operating temperature range during the tug mission. The fuel cell is activated in flight with power transfer occurring at T + 3.877 hours, and provides demand electrical power until T + 97.634 hours when power transfer to battery occurs.

During the mission the fuel cell rejects heat per Figure 1 and generates the byproduct water in the form of steam per Figure 2. Figure 3 is a simplified flow schematic of the fuel cell. Two major interfaces for the fuel cell heat rejection system are the internal fluid loop with the regenerator and the byproduct steam with the vent system.

The heat rejection system is comprised of the necessary plumbing and fittings, a redundant set of pumps, accumulators, thermal control valves, and controllers. The interface is accomplished with a single regenerator which has redundant secondary fluid loops. The 4 radiators are located in each quadrant around the intertank compartment with redundant fluid lines. Figures 4 through 6 schematically present the system.

The fuel cell fluid loop uses water or FC-40 Freon for a working fluid. The radiator system use E-1 Freon as the working fluid.

The fuel cell system is designed for a ΔT through the stack of 5.56°K (10°F) at electrical load of 1500 watts. This results in a heat rejection of 744.22 watts (2540 Btu/HR). The coolant pump adds an additional 30 watts to the system. The coolant temperature control valve controls the stack inlet temperature within a nominal operating temperature range 349.67 to 355.2°K (170°F to 180°F). The minimum flow to the regenerator at the lower temperature is 5 to 10% of full flow.
FIGURE 1 - WASTE HEAT REJECTION

FIGURE 2 - REACTANT CONSUMPTION
PAINT PATTERN AREAS

RADIATOR TYPICAL 4 PLACES

ACS MODULES TYPICAL 4 PLACES

FIGURE 4 - TUG EXTERIOR
DETAIL A

FLANGE THICKNESS

.254 CM
(.1 IN)

.762 CM
(.3 IN)

.635 CM
(.25 IN)

.457 CM
(.18 IN)

.762 CM
(.3 IN)

DETAIL B

FLANGE REMOVED

SCALE = 2/1

SIZE CODE IDENT NO.
A 04236

FIGURE 6 - RADIATOR DETAILS
The radiator system is a series-series-bypass flow system which has the radiators in series with flow through each radiator in series. The radiators are bypassed dependent upon the load by the thermal control valve which maintains a near constant fluid temperature to the regenerator of 333K (140°F).

Micrometeorite protection is provided by using a redundant fluid loop and a P-tube rail concept as shown in Figure 6.

The regenerator, accumulators, pumps, thermal control valves, controls, and instrumentation will be packaged within a box designated as the Thermal Control Unit (TCU) as shown in Figure 5. The TCU and the Fuel Cell will be isolated from the inter-tank compartment by thermal washers and multi-layer insulation.
SYSTEM REQUIREMENTS AND PERFORMANCE SPECIFICATIONS

Fuel Cell Requirements:
1. Maintain the fuel cell radiant to the stack within the design operating temperature range of 352.6 to 355.2 K (175 to 180°F) over the required heat load range.
2. The heat load range shall vary per Figure 1 with the 600 to 1500. The heat load is increased by 30 watts to account for the fuel cell pump heat dissipation.

Radiator System Requirements:
1. The system shall meet all fuel cell thermal requirements.
2. The system shall operate in earth orbit from 296 to 35750 kilometers (160 to 19300 nautical miles) with no attitude constraints for an inclination of 28.5°.
3. The radiators shall provide micrometeorite protection for the fluid lines.
4. The fluid system shall have a redundant loop.
5. The regenerator inlet temperature shall be maintained at 333 K (140°F) ± TBD.
6. Regenerator flow shall be maintained at 1.814 kilograms/minute (4 lb/minute).
7. The working fluid shall be Freon E-1.
8. The regenerator (counter flow heat exchanger) shall exhibit a minimum effectiveness of .900. The effectiveness (Eff) shall be defined as

\[
\text{Eff} = \frac{1 - e^{-NTU \left(1 - \frac{C_{\min}}{C_{\max}}\right)}}{1 - \left(\frac{C_{\min}}{C_{\max}}\right)^{NTU \left(1 - \frac{C_{\min}}{C_{\max}}\right)}}
\]

where:
- \(C = W C_p\)
- \(C_{\min} = \text{minimum enthalpy flow}\)
- \(C_{\max} = \text{maximum enthalpy flow}\)
- \(NTU = \text{number of heat exchanger units}\)
- \(W = \text{mass flow rate}\)
- \(C_p = \text{specific heat of fluid at constant pressure}\)
U = overall heat exchanger conductance
A = heat exchanger area

9. Either fluid loop shall be capable of carrying the heat load to be dissipated.

10. The radiators shall be sized to dissipate the maximum heat load
    minimum altitude and maximum external absorbed heating.

11. The fluid shall not be permitted to freeze 119°K (-246°F) or reach highly
    viscous state.

12. The cold case shall be defined as the minimum heat load with no external
    flux on the radiators.

13. Radiator coating shall exhibit stable thermal properties.

14. Radiator shall be sized assuming an adabatic vehicle side.
Predicted System Performance

Hot Case

Conditions:
1. Fixed attitude with respect to the sun. Normal sun to longitudinal axis with sun angle to center of 2 radiator panels of 45 degrees.
2. Attitude 296 kilometers (160 nautical miles).
3. $\beta = 52^\circ$ orbit.
5. Maximum heat dissipation is 744 watts plus 81 watts pump power.

Performance

See Figures 7 thru 10.
Cold Case

Conditions:
1. Fixed attitude with respect to the sun (parallel to longitudinal axis).
2. Altitude 35750 kilometers (19300 nautical miles).
3. Minimum electrical load 600 watts.
4. Minimum heat load is 281 watts plus 81 watts pump power.

Performance 
See Figures 11 thru 14.

Performance is based upon a transitional flow design where the fluid heat transfer is based upon the Colburn J-Factor analogy per Figure 15. The fluid heat transfer coefficient is related to the J-Factor by the equation

\[ h_c = \frac{J k N_R N_P}{3} \left( \frac{\mu_w}{\mu_f} \right)^{14}/D \]

- \( J \) = Colburn J Factor
- \( k \) = Fluid Conductivity
- \( N_R \) = Reynolds Number
- \( N_P \) = Prandtl Number
- \( \mu_w \) = Fluid Viscosity at the Tube Wall Temperature
- \( \mu_f \) = Fluid Viscosity at the Average Fluid Temperature
- \( D \) = Tube Internal Diameter

Performance is based upon an L/D per straight tube of 200.
FIGURE 11
RADIATOR FLUID TEMPERATURE

FIGURE 12
NET HEAT REJECTED

FIGURE 13
RADIATOR FLUID FLOW

FIGURE 14
RADIATOR FLUID FLOW

COLD CASE PERFORMANCE

04236
Colburn J-Factor vs Reynolds Number

![Graph showing Colburn J-Factor vs Reynolds Number with curves for different L/D ratios: L/D = 50, L/D = 100, L/D = 200, and L/D = 400.}]
## HARDWARE LIST & DESCRIPTION

<table>
<thead>
<tr>
<th>ITEM</th>
<th>QUANTITY</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>PUMP - Flow 1.59 to 2.04 KG/min. (3.5 to 4.5 lbs/min)</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>REGENERATOR - Redundant cold side loops approximately $1.379 \times 10^5$ Newtons/meter$^2$ (20 psi) pressure drop at 1.81 KG/min (4 lbm/min) flow/loop. Hot side loop pressure drop TBD.</td>
</tr>
<tr>
<td>3</td>
<td>2</td>
<td>ACCUMULATOR - Volume TBD. Pressure $\approx 3.447 \times 10^5$ Newtons/meter$^2$ (50 Psi).</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>DISCONNECTS - Primary and secondary loops 2 each. Line size TBD. Pressure Drop $\leq 6.89 \times 10^3$ Newtons/meter$^2$ (1 psi).</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>THERMAL CONTROL VALVE - Maintain regenerator inlet temperature at 333$^0$K (140$^0$F) by mixing radiator return fluid with pump outlet fluid. Flow range 0 to 2.04 KG/minute 0 to 4.5 lbs/min. Pressure drop TBD.</td>
</tr>
<tr>
<td>6</td>
<td>2</td>
<td>FILL DISCONNECT - System fill and drain, zero leakage after disconnect. Size - TBD.</td>
</tr>
<tr>
<td>7</td>
<td>10</td>
<td>PRESSURE TRANSDUCERS - Range 0 - $6.895 \times 10^5$ Newtons/meter$^2$ (0 - 100 Psia) Accuracy 1% of full scale.</td>
</tr>
<tr>
<td>8</td>
<td>6</td>
<td>TEMPERATURE SENSORS - Range 172 to 394$^0$K (-150 to +250$^0$F). Accuracy 1% of full scale.</td>
</tr>
<tr>
<td>9</td>
<td>2</td>
<td>FLOW MEASUREMENT - Range 0 - 2.04 kilograms/minute 0 - 4.5 pounds/minute Accuracy 1% of full scale.</td>
</tr>
<tr>
<td>10</td>
<td>4</td>
<td>Radiators - Each panel with 4 integral rails, minimum fin efficiency = .9. P - tubes welded to rails per Figure 6'single tube L/D = 200. Size:</td>
</tr>
<tr>
<td></td>
<td></td>
<td><strong>Length</strong></td>
</tr>
<tr>
<td></td>
<td></td>
<td>91.44 cm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(36 inches)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Area .511 meters$^2$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Panel thickness 0.0762 cm (0.030 inches). Tube ID 0.4572 cm (0.18 inches).</td>
</tr>
</tbody>
</table>
### Interconnecting Lines

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>11</strong></td>
<td><strong>10</strong></td>
<td>ALUMINUM TUBING - Length as required: OD - 0.9525 CM (3.75 inches)  ID - TBD</td>
</tr>
<tr>
<td><strong>12</strong></td>
<td><strong>16</strong></td>
<td>DISCONNECTS - Line size 0.9525 CM (.375 inches)  Pressure Drop ≤ 6.89 Newtons/meter² (1 Psi)</td>
</tr>
</tbody>
</table>
APPENDIX II

<table>
<thead>
<tr>
<th>LTR</th>
<th>SN</th>
<th>DESCRIPTION</th>
<th>DATE</th>
<th>APPROVED</th>
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MARTIN MARIETTA CORPORATION
POST OFFICE BOX 179, DENVER, COLORADO

THERMAL CONTROL LOUVER SYSTEM

<table>
<thead>
<tr>
<th>SIZE</th>
<th>CODE</th>
<th>IDENT NO.</th>
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<tr>
<td>A</td>
<td>04236</td>
<td></td>
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</table>

E-9030 (9-65)
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<td>3. Case 1 Emergency Battery Temperature History</td>
<td>9</td>
</tr>
<tr>
<td>4. Case 1 Louver Cover Temperature History</td>
<td>9</td>
</tr>
<tr>
<td>5. Case 1 Louver Blade Angle History</td>
<td>9</td>
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<tr>
<td>6. Case 1 Louver Cover Absorbed Heating</td>
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</tr>
<tr>
<td>7. Case 1 Instantaneous Heater Power</td>
<td>10</td>
</tr>
<tr>
<td>8. Case 2 Emergency Battery Temperature History</td>
<td>11</td>
</tr>
<tr>
<td>9. Case 2 Louver Cover Temperature History</td>
<td>11</td>
</tr>
<tr>
<td>10. Case 2 Louver Blade Angle History</td>
<td>11</td>
</tr>
<tr>
<td>11. Case 2 Louver Cover Absorbed Heating</td>
<td>12</td>
</tr>
<tr>
<td>12. Case 2 Instantaneous Heater Power</td>
<td>12</td>
</tr>
</tbody>
</table>
SYSTEM DESCRIPTION

The louver system provides thermal control to the fuel cell primary battery which will be used when the fuel cell is deactivated at T + 97.634 hours. The battery will also function as an emergency backup power supply unit in the event of fuel cell failure. The battery is designed to provide 450 watts of electrical power for a time period of 0.5 hours. Based on the power output and a 90% efficiency of the battery, 45 watts of thermal energy will be generated within the battery.

The louver system will dissipate the 45 watts of thermal energy and maintain the battery operational temperature below the allowable limit temperature of 305.3°K (90°F) for the required 0.5 hours of operation. The louver system will also add in controlling the non-operational temperature above 288.7°K (60°F).

The louver thermal control system consists of a component mounting baseplate attached to a set of moveable aluminum louver blades by low conductance screws. The blades are automatically actuated by temperature sensitive bimetallic spiral wound springs radiatively coupled to the baseplate. The baseplate and louver blades are housed in a conductively isolated frame which is mounted on the interior side of the tug skin using minimum conductance fasteners. The louver assembly and mounting configuration are shown in Figure 1.
FIGURE 1 - LOUVER SYSTEM/MOUNTING CONFIGURATION
A thermal model of the louver system shown in Figure 1 was constructed for the MITAS thermal analyzer (Reference 1). The model was necessary because steady state, worst condition analysis tended to overdesign the system. The model accounts for the thermal characteristics of both the louver system and the emergency battery. A GFP absorbed heating environment was simulated in the model and is shown in Figure 2. This environment was calculated assuming an $a/e$ of the external skin equal to .2/.9. The thermal capacitance of the battery, baseplate and external skin along with a time line to adequately account for the battery power generation is included. Conduction through the multi-layer insulation and through the louver system standoffs is included as well as the contact resistance between the battery and the baseplate.

The louver system parameters used in the model correspond to a commercially available bimetallic actuated louver system (Reference 2). The blade angle is determined by the baseplate temperature ($289^\circ K$ (60$^\circ F$) blades closed, $303^\circ K$ (85$^\circ F$) blades fully open). The effective emittance is then determined by the blade angle as shown in Table 1. The louvered area consists of 0.165 sq. m (1.78 sq. ft.) which was also used for the area of the baseplate and the external skin. The baseplate was assumed to be 0.32 CM (1/8 in.) thick aluminum and the external skin was assumed to be 0.25 CM (0.10 in.) thick aluminum. The emergency battery simulated was taken from the tug data bank (Reference 3) and had a thermal mass of 1.79 watt-hrs/$^\circ K$ (3.39 btu/$^\circ F$). Also a 10 watt, thermostatically controlled, heater was incorporated in the battery to maintain temperature limits in the non-operating condition.
Figure 2 External Skin Absorbed Environmental Flux
**System Requirements and Performance Specifications**

1. Maintain primary battery temperature below 305.4 Kelvin (90°F) for 0.5 hours of operation.

2. Provide means of dissipating 45 watts of thermal energy while battery is operating.

3. Control non-operational battery temperatures above 288.7 Kelvin (60°F).

4. Provide control of blade position as a function of baseplate temperature, 288.7 Kelvin (60°F) blades closed, 303.0 Kelvin (85°F) blade open.
Predicted System Performance

Two cases were simulated using the previously described math model and the absorbed environment shown in Figure 2. In both cases the initial temperatures were started at 294.4°C (700°F) and the problem was simulated for 5 orbits approximately 8 hours corresponding to the heating rate in Figure 2. In the middle of the third orbit (approximately 4 hours) the battery was activated for 0.5 hours. The two cases differ in that the second case uses only 10 percent of the absorbed heating rate shown in Figure 2. This case demonstrates the adequacy of the 10 watt heater to maintain temperature control. The results of the first case are shown in Figures 3 through 7. The results of the second case are shown in Figure 8 through 12.
Figure 3, Case 1 Emergency Battery Temperature History

Figure 4, Case 1 Louver Cover Temperature History

Figure 5, Case 1 Louver Blade Angle History
Orbit Time (hours)

Figure 6, Case 1 Louver Cover Absorbed Heating

Orbit Time (hours)

Figure 7, Case 1 Instantaneous Heater Power
Figure 8, Case 2 Emergency Battery Temperature History

Figure 9, Case 2 Louver Cover Temperature History

Figure 10, Case 2 Louver Blade Angle History
Figure 11, Case 2 Louver Cover Absorbed Heating

Figure 12, Case 2 Instantaneous Heater Power
**TABLE 1 - EFFECTIVE EMITTANCE $\varepsilon$ FOR COVERED LOUVER SYSTEM**

<table>
<thead>
<tr>
<th>BLADE ANGLE DEG</th>
<th>EFFECTIVE EMITTANCE $\varepsilon$</th>
</tr>
</thead>
<tbody>
<tr>
<td>90 (Full Open)</td>
<td>0.818</td>
</tr>
<tr>
<td>75</td>
<td>0.790</td>
</tr>
<tr>
<td>60</td>
<td>0.742</td>
</tr>
<tr>
<td>45</td>
<td>0.660</td>
</tr>
<tr>
<td>30</td>
<td>0.543</td>
</tr>
<tr>
<td>15</td>
<td>0.379</td>
</tr>
<tr>
<td>0 (Full Closed)</td>
<td>0.035</td>
</tr>
</tbody>
</table>

(for a covered louver system assuming a diffuse wall and a diffuse baseplate $\varepsilon = 0.9$)
Hardware List & Description

<table>
<thead>
<tr>
<th>Item</th>
<th>Quantity</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>Louver frame and blade assembly - minimum covered area of 0.17 $M^2$ (1.78 $ft^2$). Complete with temperature sensitive bimetallic actuators. Blades are specular and have an $\varepsilon \leq 0.5$.</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>Component mounting baseplate - 0.318 cm (0.125 in.) thick aluminum plate with a minimum surface area of 0.49 $M^2$ (5.30 $ft^2$).</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>Multilayer insulation blanket - 20 alternate layers of perforated aluminized mylar and tissue glass.</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>Interior thermal control coatings - radiating surface of component baseplate and interior of louver cover/skin, painted with a high emittance ($\varepsilon \geq 0.9$) diffuse coating.</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>Exterior thermal control coatings - a minimum area of 0.17 $M^2$ (1.78 $ft^2$) of the external cover/skin should be covered with second surface mirrors.</td>
</tr>
<tr>
<td>6</td>
<td>TBD</td>
<td>Mounting panel thermal isolators-low conductance screws, washers, standoffs, etc. for the purpose of mounting the louver assembly to the cover/skin.</td>
</tr>
</tbody>
</table>
REFERENCES

1. Conner, R. J. et al, "Martin Interactive Thermal Analysis System (MITAS)"

2. "Space Vehicle Thermal Controllers" Technical Description, Northrop Corporation

3. T. L. Ward, "Space Tug Thermal Control Equipment Thermal Requirements,
   Characteristics and Constraints Catalogue". MCR-74-145, Martin Marietta
   Corporation, April 1974.
### APPENDIX III

**Space Tug Forward Compartment Thermal Design**

**Martin Marietta Corporation**
Post Office Box 179, Denver, Colorado

**Drawn by:**

**Department:**

**Date:** 09/44

**Checking:**

**Group Eng:**

**Stress:**

**Weight:**

**Customer Appl:**

**Program Appl:**

**Size Code Ident No.:**

**A 04236**

**Scale Page Sheet:**

III-1
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System Description
System Requirements & Performance Specifications
Hardware List and Description

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1. Forward Compartment Thermal Control Concept
2. Hot Case Mounting Panel Heat Fluxes
3. Cold Case Mounting Panel Heat Fluxes
SYSTEM DESCRIPTION

The tug forward compartment is designed thermally to operate over a range of worst case environments which include a fixed attitude with respect to the sun in near earth orbit and a zero heating attitude at geosynchronous attitude. The design incorporates several thermal devices whose purposes is to provide temperature control of the avionics components.

The basic concept is to mount the components on thermal conditioning panels which are mounted to the structure with louver assemblies attached to the skin side of panels as shown in Figure 1. Heat pipes are mounted on the interior surface of the honeycomb skin to provide a relatively uniform temperature around the forward skirt.

The thermal conditioning panels are honeycomb panels with integral heat pipes. The panels are designed to permit two-dimensional heat flow, thus approaching an isothermal plate concept. Mounting of high and low duty cycle components on each panel permits distribution of heat between components thus reducing if not eliminating the need of component heaters. The skin side louvers provide the means to reduce radiation losses from the panel as the panel temperature begins to drop in cold environments by closing the blades. This permits the panel temperature to be passively controlled to a relatively narrow range thus simplifying the component thermal design problems as well as heater power requirements.

The heat pipes on the internal surface of the skin act to isothermalize the skin dependent upon the external and internal heating on the skin. Heat is transferred from the hot side of the vehicle to the cold side thus providing
Figure 1 Forward Compartment Thermal Control Concept
a more uniform environment for the panels and the components.

System Requirements and Performance Specifications

The primary purpose of this system is to maintain the tug avionics components within acceptable temperature limits during the tug mission. To achieve this objective each of the major elements shall meet the following requirements.

Thermal Conditioning Panel

Non-Operating Temperature Range 255 to 367°C 0 to 200°F
Operating Temperature Range 272 to 311°C 30°F to 100°F

Maximum Component Heat Load 300 Watts

Maximum Gradient Across Panel Surface 2.77°C 5°F
Maximum Thermal Load Density .31 Watts/cm² 2 Watts/in²

Size As Required

Bolt Pattern .1 x .1 Meters 4 x 4 inches
Panel Mass ≤ 13.8 KG/m²
Maximum Component Mass 45.4 KG 100 Pounds
LOUVERS

Size 40.64 x 20.32 x 4.9 CM 
(16 x 8 x 1.93 inches)

Weight .27 KG .6 Pounds

Blade Operating Temp. Range 288.7 to 302.60K (closed to open) 
(60 to 85°F)
End Point Adjustment ±5.60K (+10°F)
Blade Emissivity ≤ 0.1
Temperature Survivability 199.8 to 394.30K 
(-100 to 250°F)

Effective Emissivity of Baseplate

<table>
<thead>
<tr>
<th>State</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Open</td>
<td>≥ .8</td>
</tr>
<tr>
<td>Closed</td>
<td>≤ .1</td>
</tr>
</tbody>
</table>

SKIN HEAT PIPES

Number/Spacing 6 Pipes One Every 5 Inches in Longitudinal Direction
Length 14.77M, (45.03 ft.) Circumferential
Diameter 1.27 CM, (0.5 in.) (Nominal)

Non-operating Temperature Range 144 to 366°K 
-200 to 200°F
Operating Temperature Range 172 to 311°K 
-150 to 100°F
Heat Flux Capability 60 Watts/M Per Pipe
at 300°K (80°F) (19.7 Watts/Ft) Per Pipe
Evaporator to Condenser <5.60K
Maximum ΔT at 300°K (80°F) (10°F)
Heat Transport Capability TBD
Mounting panel heat fluxes are given in Figures 2 and 3 for hot and cold conditions respectively. These curves were generated from the following equation:

\[
\frac{Q}{A} = \sigma_{\text{eff}} (T_p^4 - T_s^4) + \varepsilon_p (T_p^4 - T_e^4)
\]

where:

- \( Q/A \) = Panel Net Heat Transfer
- \( \sigma \) = Stefan-Boltzmann Constant
- \( \varepsilon_{\text{eff}} \) = Louver System Effective Emissivity
- \( T_p \) = Mounting Panel Temperature
- \( T_s \) = Skin Temperature
- \( T_e \) = Interior Environmental Temperature
- \( \varepsilon_p \) = Emissivity of Mounting Panel
Figure 2 Hot-Case Mounting Panel Heat Fluxes, Louvers Open
Figure 3 Cold-Case Mounting Panel Heat Fluxes, Louvers Closed
HARDWARE LIST

6 - Circumferential Heat Pipes and Mounting Brackets
5 to 6 Thermal Conditioning Panels - Number and Size Dependent Upon Component Groupings
Thermal Control Louver Assemblies One for Each Thermal Conditioning Panel