

|                                                                                                                      |                                             |                        |         |
|----------------------------------------------------------------------------------------------------------------------|---------------------------------------------|------------------------|---------|
| <br><i>An SAE International Group</i> | <b>AEROSPACE<br/>INFORMATION<br/>REPORT</b> | <b>SAE</b> AIR1168/13  | REV. A  |
|                                                                                                                      |                                             | Issued                 | 1999-11 |
|                                                                                                                      |                                             | Reaffirmed             | 2006-03 |
|                                                                                                                      |                                             | Stabilized             | 2011-07 |
|                                                                                                                      |                                             | Superseding AIR1168/13 |         |
| Spacecraft Equipment Environmental Control                                                                           |                                             |                        |         |

## RATIONALE

This document has been determined to contain basic and stable technology which is not dynamic in nature.

## STABILIZED NOTICE

This document has been declared "Stabilized" by the AC-9 Technical Committee and will no longer be subjected to periodic reviews for currency. Users are responsible for verifying references and continued suitability of technical requirements. Newer technology may exist.

## PREFACE

This document is one of 14 Aerospace Information Reports (AIR) of the Third Edition of the SAE Aerospace Applied Thermodynamics Manual. The manual provides a reference source for thermodynamics, aerodynamics, fluid dynamics, heat transfer, and properties of materials for the aerospace industry. Procedures and equations commonly used for aerospace applications of these technologies are included.

In the Third Edition, no attempt was made to update material from the Second Edition nor were SI units added. However, all identified errata were corrected and incorporated and original figure numbering was retained, insofar as possible.

The SAE AC-9B Subcommittee originally created the SAE Aerospace Applied Thermodynamics Manual, and, for the Third Edition, used a new format consisting of AIR1168/1 through AIR1168/10. AIR1168/11 through AIR1168/14 were created by the SAE SC-9 Committee.

The AIRs comprising the Third Edition are shown below. Applicable sections of the Second Edition are shown parenthetically in the third column.

|           |                                                              |               |
|-----------|--------------------------------------------------------------|---------------|
| AIR1168/1 | Thermodynamics of Incompressible and Compressible Fluid Flow | (1A,1B)       |
| AIR1168/2 | Heat and Mass Transfer and Air-Water Mixtures                | (1C,1D,1E)    |
| AIR1168/3 | Aerothermodynamic Systems Engineering and Design             | (3A,3B,3C,3D) |
| AIR1168/4 | Ice, Rain, Fog, and Frost Protection                         | (3F)          |

SAE Technical Standards Board Rules provide that: "This report is published by SAE to advance the state of technical and engineering sciences. The use of this report is entirely voluntary, and its applicability and suitability for any particular use, including any patent infringement arising therefrom, is the sole responsibility of the user."

SAE reviews each technical report at least every five years at which time it may be reaffirmed, revised, or cancelled. SAE invites your written comments and suggestions.

Copyright © 2011 SAE International

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or otherwise, without the prior written permission of SAE.

TO PLACE A DOCUMENT ORDER: Tel: 877-606-7323 (inside USA and Canada)  
Tel: +1 724-776-4970 (outside USA)  
Fax: 724-776-0790  
Email: CustomerService@sae.org  
http://www.sae.org

SAE WEB ADDRESS:

Full document details: SAE AIR1168\_13A

SAE values your input. To provide feedback on this Technical Report, please visit <http://www.sae.org/technical/standards/AIR1168/13A>

Documents

Document Sharing Hub

## PREFACE (Continued)

|            |                                                                                                                 |               |
|------------|-----------------------------------------------------------------------------------------------------------------|---------------|
| AIR1168/5  | Aerothermodynamic Test Instrumentation and Measurement                                                          | (3G)          |
| AIR1168/6  | Characteristics of Equipment Components, Equipment Cooling System Design, and Temperature Control System Design | (3H,3J,3K)    |
| AIR1168/7  | Aerospace Pressurization System Design                                                                          | (3E)          |
| AIR1168/8  | Aircraft Fuel Weight Penalty Due to Air Conditioning                                                            | (3I)          |
| AIR1168/9  | Thermophysical Properties of the Natural Environment, Gases, Liquids, and Solids                                | (2A,2B,2C,2D) |
| AIR1168/10 | Thermophysical Characteristics of Working Fluids and Heat Transfer Fluids                                       | (2E,2F)       |
| AIR1168/11 | Spacecraft Boost and Entry Heat Transfer                                                                        | (4A,4B)       |
| AIR1168/12 | Spacecraft Thermal Balance                                                                                      | (4C)          |
| AIR1168/13 | Spacecraft Equipment Environmental Control                                                                      | (4D)          |
| AIR1168/14 | Spacecraft Life Support Systems                                                                                 | (4E)          |

F.R. Weiner, formerly of Rockwell International and past chairman of the SAE AC-9B Subcommittee, is commended for his dedication and effort in preparing the errata lists that were used in creating the Third Edition.

## TABLE OF CONTENTS

|     |                                                                |     |
|-----|----------------------------------------------------------------|-----|
| 1.  | INTRODUCTION.....                                              | 9   |
| 1.1 | Scope .....                                                    | 9   |
| 1.2 | Environmental Control Systems.....                             | 9   |
| 1.3 | Nomenclature .....                                             | 10  |
| 1.4 | Common Abbreviations .....                                     | 14  |
| 2.  | ESTABLISHMENT OF DESIGN REQUIREMENTS .....                     | 16  |
| 2.1 | Parameters to be Studied .....                                 | 16  |
| 3.  | REGULATION, SENSING, AND CONTROL OF ENVIRONMENTAL SYSTEMS..... | 18  |
| 3.1 | Temperature .....                                              | 18  |
| 3.2 | Compartment Pressure Regulation.....                           | 23  |
| 4.  | TEMPERATURE CONTROL .....                                      | 25  |
| 4.1 | Passive Temperature Control Methods.....                       | 26  |
| 4.2 | Technical Approach, Passive Methods.....                       | 30  |
| 4.3 | Semipassive Temperature Control Methods.....                   | 51  |
| 4.4 | Technical Approach, Semi-Passive Methods.....                  | 59  |
| 4.5 | Active Temperature Control .....                               | 94  |
| 4.6 | Technical Approach, Active Temperature Control Methods.....    | 101 |
| 5.  | PRESSURE CONTROL.....                                          | 159 |
| 5.1 | External Leakage Control .....                                 | 159 |
| 5.2 | Examples .....                                                 | 161 |
| 6.  | REFERENCES .....                                               | 170 |

## LIST OF FIGURES

|           |                                                                                                                                          |    |
|-----------|------------------------------------------------------------------------------------------------------------------------------------------|----|
| FIGURE 1  | Temperature Control by Fluid Flow Modulation .....                                                                                       | 19 |
| FIGURE 2  | Temperature Control by Mixing and Bypass .....                                                                                           | 20 |
| FIGURE 3  | Temperature Control by Heat Addition .....                                                                                               | 20 |
| FIGURE 4  | Temperature Control by Varying Surface Emissivity and Solar Absorptivity .....                                                           | 21 |
| FIGURE 5  | Compartment Pressure Schedule Chart .....                                                                                                | 24 |
| FIGURE 6  | Compartment Relief Valve .....                                                                                                           | 25 |
| FIGURE 7  | Temperature Control by Fixed Resistance .....                                                                                            | 26 |
| FIGURE 8  | Heat Disposal by Radiation .....                                                                                                         | 29 |
| FIGURE 9  | Simple Passive Temperature Control System .....                                                                                          | 31 |
| FIGURE 10 | Thermal Conductance Across Interface of Conductor Joint .....                                                                            | 34 |
| FIGURE 11 | Thermal Conductance Versus Contact Pressure for Joints With Shim Material .....                                                          | 35 |
| FIGURE 12 | Section A-A of an Electrical Separation Washer Using Glass Balls as<br>Dielectric Material .....                                         | 36 |
| FIGURE 13 | Typical Component Mounting Joint .....                                                                                                   | 38 |
| FIGURE 14 | Thermal Conductivity Versus Density<br>Contribution by Various Modes of Heat Transfer (for Fibrous Type Insulation) .....                | 39 |
| FIGURE 15 | Variation of Thermal Conductivity $k$ With Pressure for Various Powdered<br>Insulations of Temperatures Between 138 and 540 °R .....     | 40 |
| FIGURE 16 | Thermal Balance Solution for Flat Plate Normal to Sun's Rays at<br>Top of Atmosphere .....                                               | 44 |
| FIGURE 17 | Solution of Radiation Equation (Equation 13) .....                                                                                       | 45 |
| FIGURE 18 | Multilayer Shielding .....                                                                                                               | 46 |
| FIGURE 19 | Multishield Heat Transfer Coefficient (Box in Free Space and No Solar Heat) .....                                                        | 49 |
| FIGURE 20 | Heat Transfer in Space Using Three Shields .....                                                                                         | 50 |
| FIGURE 21 | Heat Transfer in Space Using Four Shields .....                                                                                          | 51 |
| FIGURE 22 | Thermal Switch for Control of Conductivity .....                                                                                         | 53 |
| FIGURE 23 | Shutter for Controlling Emissivity .....                                                                                                 | 54 |
| FIGURE 24 | Forced Convection Loop .....                                                                                                             | 55 |
| FIGURE 25 | Expendable Heat Sink System .....                                                                                                        | 56 |
| FIGURE 26 | Vacuum Cooling System .....                                                                                                              | 57 |
| FIGURE 27 | Silica Gel Cooling System .....                                                                                                          | 58 |
| FIGURE 28 | Differential Metallic Expansion Device for Controlling a Thermal Path .....                                                              | 60 |
| FIGURE 29 | Thermal Switch for On-Off Conductivity Control .....                                                                                     | 60 |
| FIGURE 30 | Thermal Switch for Variable Conductivity Control .....                                                                                   | 61 |
| FIGURE 31 | Separate Expendable Coolant Storage .....                                                                                                | 63 |
| FIGURE 32 | Integral Expendable Coolant Storage .....                                                                                                | 63 |
| FIGURE 33 | High Latent Heat Liquids .....                                                                                                           | 66 |
| FIGURE 34 | Moderately High Latent Heat Liquids .....                                                                                                | 67 |
| FIGURE 35 | Vapor Pressure of Liquids With High Latent Heats; for Dashed Lines,<br>Data Computed Using Clausius-Clapeyron Equation .....             | 68 |
| FIGURE 36 | Vapor Pressures of Liquids With Moderately High Latent Heats;<br>for Dashed Lines, Data Computed Using Clausius-Clapeyron Equation ..... | 69 |
| FIGURE 37 | Expendable Water Method (see Table 12) .....                                                                                             | 69 |

## LIST OF FIGURES (Continued)

|           |                                                                                                                                                                          |     |
|-----------|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----|
| FIGURE 38 | Liquid Heat Sink Storage Weight; Includes Tank, Insulation Where Applicable, Supports and Fill, Vent, and Safety Valving .....                                           | 72  |
| FIGURE 39 | Liquid Heat Sink Storage Volume (Reference 26) .....                                                                                                                     | 73  |
| FIGURE 40 | Capacity of Expendable Heat Sinks (200 °F Equipment Surface Temperature) (Reference 26) .....                                                                            | 73  |
| FIGURE 41 | Forced Convection Loop Temperature Control Method .....                                                                                                                  | 76  |
| FIGURE 42 | Turbulent Flow Liquid Coolant Parameters .....                                                                                                                           | 79  |
| FIGURE 43 | Typical Closed Cycle Gaseous Coolant Loop .....                                                                                                                          | 80  |
| FIGURE 44 | Vacuum Cooling Method; (a) Basic Method; (b) Two Loop Method .....                                                                                                       | 84  |
| FIGURE 45 | Silica Gel Cooling Method .....                                                                                                                                          | 85  |
| FIGURE 46 | Venetian Blind Shutter, Hinged Type .....                                                                                                                                | 85  |
| FIGURE 47 | Thermal Analog Network for Venetian Blind Shutter .....                                                                                                                  | 87  |
| FIGURE 48 | Cross Section Through Shutter Arrangement .....                                                                                                                          | 87  |
| FIGURE 49 | Radiosity Analog Network for Venetian Blind Shutter .....                                                                                                                | 88  |
| FIGURE 50 | Incident Radiation Absorbed .....                                                                                                                                        | 89  |
| FIGURE 51 | Translating and Rotating Mask Temperature Control Units .....                                                                                                            | 92  |
| FIGURE 52 | Mercury Film Temperature Control Method .....                                                                                                                            | 93  |
| FIGURE 53 | Temperature-Entropy Diagram for Reverse Carnot Refrigeration Cycle .....                                                                                                 | 94  |
| FIGURE 54 | Temperature-Entropy Diagram for a Vapor Refrigeration Cycle .....                                                                                                        | 95  |
| FIGURE 55 | Vapor Refrigeration Cycle .....                                                                                                                                          | 95  |
| FIGURE 56 | Temperature-Entropy Diagram for a Gas Refrigeration Cycle .....                                                                                                          | 96  |
| FIGURE 57 | Absorption Refrigeration Cycle .....                                                                                                                                     | 97  |
| FIGURE 58 | The Vortex Tube .....                                                                                                                                                    | 99  |
| FIGURE 59 | Peltier Heat Pump .....                                                                                                                                                  | 100 |
| FIGURE 60 | Comparison of Volumes of Freon 11 Plate-Fin, Vortex Tube, and Plain Fin-Tube Evaporators, All Evaporators Finned on Air Side .....                                       | 104 |
| FIGURE 61 | Comparison of Volumes of Freon 11 Plate-Fin, Vortex Tube, and Plain Fin-Tube Evaporators; Same Conditions as for Figure 60 Except That $\Delta P_{air} = 0.1$ psia ..... | 105 |
| FIGURE 62 | Comparison of Weights of Freon 11 Plate-Fin, Vortex Tube, and Plain Fin-Tube Evaporators .....                                                                           | 105 |
| FIGURE 63 | Environmental Control System Evaporator for Zero Gravity Operation .....                                                                                                 | 106 |
| FIGURE 64 | Proposed Design of an Evaporator Using Wick Material .....                                                                                                               | 107 |
| FIGURE 65 | Radiator Area Versus Radiator Temperature for Cooling System; Evaporator Temperature = 40 °F .....                                                                       | 110 |
| FIGURE 66 | Domains for Two Phase Flow Stability; $\alpha$ = Wave Number = $2\pi\delta/\lambda$ .....                                                                                | 112 |
| FIGURE 67 | Influence of Surface Tension on Tapered Tubes .....                                                                                                                      | 113 |
| FIGURE 68 | Geometry of Tapered Tube Condenser-Radiator .....                                                                                                                        | 114 |
| FIGURE 69 | Condenser-Radiator Characteristics with Freon 11 .....                                                                                                                   | 116 |
| FIGURE 70 | Flow in Curved Passages .....                                                                                                                                            | 117 |
| FIGURE 71 | Spiral Condenser (Fluids in Counterflow) (Reference 31) .....                                                                                                            | 118 |

## LIST OF FIGURES (Continued)

|           |                                                                                                                                                                                                                                        |     |
|-----------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----|
| FIGURE 72 | Comparison of Volumes of Freon 11 Spiral Condenser (Zero g) with Large Tube-Fin Condensers (1 g); Staggered Circular Finned Tubes .....                                                                                                | 118 |
| FIGURE 73 | Volume and Pressure Drop for Freon 11 Spiral Condenser .....                                                                                                                                                                           | 119 |
| FIGURE 74 | Variation of Freon 11 Spiral Condenser Volume and Pressure With NaK Side Hydraulic Diameter .....                                                                                                                                      | 120 |
| FIGURE 75 | Rotating Condenser (Reference 31) .....                                                                                                                                                                                                | 120 |
| FIGURE 76 | Condenser System Dead Weight Versus Number of Discs .....                                                                                                                                                                              | 121 |
| FIGURE 77 | Total Power Requirement Versus Number of Discs; Condenser Angular Velocity and System Capacity are the Parameters .....                                                                                                                | 122 |
| FIGURE 78 | Performance of Vapor Cycle (Reference 36) .....                                                                                                                                                                                        | 123 |
| FIGURE 79 | Coefficient of Performance of Freon Refrigerants .....                                                                                                                                                                                 | 124 |
| FIGURE 80 | Refrigeration Effect Per Unit Volume for Freon Refrigerants .....                                                                                                                                                                      | 125 |
| FIGURE 81 | Ratio of Condenser Pressure to Evaporator Pressure for Freon Refrigerants .....                                                                                                                                                        | 126 |
| FIGURE 82 | Temperature Entropy Diagram of the Brayton Cycle .....                                                                                                                                                                                 | 130 |
| FIGURE 83 | A Gas Cycle Cooling System .....                                                                                                                                                                                                       | 131 |
| FIGURE 84 | Relationship of Gas Cycle Coefficient of Performance to Pressure Ratio .....                                                                                                                                                           | 133 |
| FIGURE 85 | Relationship of Gas Cycle Coefficient of Performance to Pressure Ratio; $\Delta T_E/T_4 = 0.0492$ , $\eta_c = \eta_t = 0.85$ , $\Delta P = 0$ .....                                                                                    | 133 |
| FIGURE 86 | Radiator Requirements of Gas Cycle Systems; $\Delta T_E/T_4 = 0.0588$ , $\eta_t = \eta_c = 0.85$ .....                                                                                                                                 | 136 |
| FIGURE 87 | Radiator Requirements of Gas Cycle Systems; $\Delta T_E/T_4 = 0.0492$ , $\eta_c = \eta_t = 0.85$ .....                                                                                                                                 | 136 |
| FIGURE 88 | Relationship of Radiator Area Ratio to Cycle Pressure Ratio; $\Delta T_E/T_4 = 0.0588$ , $\eta_c = \eta_t = 0.85$ .....                                                                                                                | 137 |
| FIGURE 89 | Relationship of Radiator Area Ratio to Cycle Pressure Ratio; $\Delta T_E/T_4 = 0.0492$ , $\eta_c = \eta_t = 0.85$ .....                                                                                                                | 137 |
| FIGURE 90 | Compressor for Gas Cycle .....                                                                                                                                                                                                         | 140 |
| FIGURE 91 | Temperature Profile Across a Peltier Refrigerator .....                                                                                                                                                                                | 148 |
| FIGURE 92 | Peltier Coefficient of Performance as a Function of Temperature Differential; $Z_0 = 0.002 \text{ K}^{-1}$ .....                                                                                                                       | 150 |
| FIGURE 93 | Equivalent Series and Series-Parallel Thermoelectric Circuit Configurations; (a) $N_r = 1$ row; (b) $N_r = 3$ rows; Total Thermoelement Area Same in Both Cases and Total Number of Elements Per Row, $N_e$ , Same in Both Cases ..... | 153 |
| FIGURE 94 | Volume of Active Thermoelectric Material as a Function of Cold Junction Temperature; $Z_0 = 0.002 \text{ K}^{-1}$ .....                                                                                                                | 155 |
| FIGURE 95 | Minimum Volume of Active Thermoelectric Material Per Watt of Refrigeration Capacity as a Function of Cold Junction Temperature .....                                                                                                   | 156 |
| FIGURE 96 | Volume of Active Thermoelectric Material as a Function of Temperature Differential; $Z_0 = 0.002 \text{ K}^{-1}$ .....                                                                                                                 | 156 |
| FIGURE 97 | Optimum Temperature Differential as a Function of Cold Junction Temperature (Based on Minimum Weight) .....                                                                                                                            | 157 |
| FIGURE 98 | Minimum Volume of Active Thermoelectric Material as a Function of Figure of Merit .....                                                                                                                                                | 157 |

## LIST OF TABLES

|          |                                                                                                                                   |     |
|----------|-----------------------------------------------------------------------------------------------------------------------------------|-----|
| TABLE 1  | Types of Automatic Controllers for Temperature Control Systems .....                                                              | 21  |
| TABLE 2  | Temperature Sensors and Thermostats .....                                                                                         | 22  |
| TABLE 3  | Thermal Properties of Some Heat Storage Materials .....                                                                           | 28  |
| TABLE 4  | Heats of Solution of Representative Substances .....                                                                              | 29  |
| TABLE 5  | Comparative Thermal and Electrical Resistances of Various Electrical<br>Insulating Materials (Washers) .....                      | 37  |
| TABLE 6  | Effect of Electrical Insulators on Thermal Resistance of Typical Component<br>Mounting Joint .....                                | 37  |
| TABLE 7  | Effect of Air Removal Upon Thermal Conductivity of Various Materials at<br>100 °F Mean Temperature.....                           | 38  |
| TABLE 8  | Absorptance and Emittance of Materials Found in Explorer VII Satellite Vehicle .....                                              | 42  |
| TABLE 9  | Variation of Thermal Conductivity at 100 °F of Various Insulating Materials<br>with Interstitial Pressure.....                    | 62  |
| TABLE 10 | Properties of Some Substances Suitable as Expendable Coolants .....                                                               | 64  |
| TABLE 11 | Liquid Properties and Trouton's Rule .....                                                                                        | 65  |
| TABLE 12 | Comparison of Water Method to Ammonia Method (see Figure 37).....                                                                 | 70  |
| TABLE 13 | Weight Comparison of Expendable Coolants<br>( $Q = 1000 \text{ W}$ , $t_E = 130 \text{ °F}$ , $\Delta\theta = 1 \text{ h}$ )..... | 75  |
| TABLE 14 | Comparison of Liquids for Semipassive Closed Loop Cooling .....                                                                   | 78  |
| TABLE 15 | Comparison of Gases for Semipassive Closed Loop Cooling .....                                                                     | 82  |
| TABLE 16 | Comparison of Artificial Atmospheres.....                                                                                         | 83  |
| TABLE 17 | Physical Properties and Absorption Characteristics of Wick Materials .....                                                        | 108 |
| TABLE 18 | Typical Vapor Velocities $N_{Re}$ and $N_{We}$ .....                                                                              | 112 |
| TABLE 19 | Minimum Ratios for Condensing Side to Radiating Side<br>Heat Transfer Coefficients for Integral Condenser-Radiators.....          | 114 |
| TABLE 20 | Summary of Properties of Selected Refrigerants .....                                                                              | 126 |
| TABLE 21 | Vapor Cycle System Weight and Performance Characteristics .....                                                                   | 127 |
| TABLE 22 | Vapor Cycle System Weight and Performance Characteristics .....                                                                   | 128 |
| TABLE 23 | Range of Suitable Specific Speeds for Various Compressor Types .....                                                              | 138 |
| TABLE 24 | Range of Suitable Specific Speeds for Turbines .....                                                                              | 141 |
| TABLE 25 | Weight and Performance Characteristics of a Gas Cycle Refrigeration System .....                                                  | 142 |
| TABLE 26 | Leakage Rates of Vehicles .....                                                                                                   | 160 |



## 1. INTRODUCTION:

### 1.1 Scope:

This part of the manual presents methods for arriving at a solution to the problem of spacecraft inflight equipment environmental control. The temperature aspect of this problem may be defined as the maintenance of a proper balance and integration of the following thermal loads: equipment-generated, personnel-generated, and transmission through external boundary.

Achievement of such a thermal energy balance involves the investigation of three specific areas:

1. Establishment of design requirements.
2. Evaluation of properties of materials.
3. Development of analytical approach.

The solution to the problem of vehicle and/or equipment pressurization, which is the second half of major environmental control functions, is also treated in this section. Pressurization in this case may be defined as the task associated with the storage and control of a pressurizing fluid, leakage control, and repressurization. Although secondary to the thermal energy balance, the function of pressure control is nevertheless an important aspect of the overall environmental problem.

### 1.2 Environmental Control Systems:

Environmental control systems (ECS) must operate in a continuously changing radiant thermal energy exchange condition with the Sun, Earth, and space, and with variable internal equipment loads. Frequently, during the operational phase, concentrated amounts of energy must be dissipated from small components with small external surfaces. The internally generated heat loads may be cyclic but also random with respect to the orbit.

Some equipment such as actuators, attitude control devices, or storage tanks may be partially exposed and require special treatment. Other components will function properly only if pressurized, while in some instances pressurization is used as a means of eliminating the need for qualification in a hard vacuum. These installation techniques will introduce their own thermal control problems and contribute to the selection of a specific ECS configuration.

### 1.3 Nomenclature:

Wherever the same symbol is used in the text for different properties, the paragraph number is included in the definition as a convenience and to avoid ambiguity. Variations of symbol usage in an equation may be defined in the explanatory list following the equation. Thermoelectric nomenclature is listed separately in 4.6.4.1.

|                                  |   |                                                                                     |
|----------------------------------|---|-------------------------------------------------------------------------------------|
| A                                | = | Heat transfer surface, ft <sup>2</sup> or in <sup>2</sup>                           |
| a                                | = | $1 - \frac{D_e}{D_0}$ = Taper parameter (4.6.1.6, Equation 101)                     |
| C                                | = | Specific heat, W-s/°F, Btu lb-ft <sup>3</sup>                                       |
| C                                | = | Parameter, dimensionless (Equations 24 to 27)                                       |
| COP                              | = | Coefficient of performance, dimensionless                                           |
| c <sub>p</sub>                   | = | Specific heat capacity at constant pressure, Btu/lb-°F                              |
| c <sub>v</sub>                   | = | Specific heat capacity at constant volume, Btu/lb-°F                                |
| D                                | = | Diameter, ft, in                                                                    |
| E                                | = | Blackbody emissive power, Btu/h-ft <sup>2</sup> (4.4)                               |
| E                                | = | Planetary emission, Btu/h                                                           |
| E                                | = | Potential, Btu/h-ft <sup>2</sup> (4.4.6.1)                                          |
| F                                | = | Configuration factor, dimensionless (4.4.6.1, Equation 63)                          |
| F                                | = | Fraction of vapor condensed in tube, dimensionless (4.6.1.6)                        |
| F                                | = | Wind force, lb (4.4.3.3)                                                            |
| °F/W                             | = | °F per watt (4.2.1.2) (Tables 5 and 6)                                              |
| f                                | = | Fractional volume of insulation occupied by fibers, dimensionless (4.2, Equation 2) |
| f                                | = | Friction factor, dimensionless                                                      |
| f                                | = | Interchange factor, dimensionless                                                   |
| f                                | = | Mean friction factor, dimensionless (4.6.1)                                         |
| G                                | = | Sum of all incident radiation, Btu/h-ft <sup>2</sup>                                |
| G                                | = | Mass velocity, lb/s-ft <sup>2</sup> (5.2.3)                                         |
| g                                | = | Gravitational force, in/s <sup>2</sup>                                              |
| g'                               | = | Artificial gravitational force, in/s <sup>2</sup>                                   |
| g <sub>0</sub>                   | = | Gravitational acceleration on Earth, cm/s <sup>2</sup>                              |
| g <sub>p</sub> or p <sub>g</sub> | = | Specific weight (density), lb/ft <sup>3</sup>                                       |
| g <sub>p<sub>f</sub></sub>       | = | Fin specific weight, lb/ft <sup>3</sup>                                             |
| g <sub>p<sub>t</sub></sub>       | = | Tube specific weight, lb/ft <sup>3</sup>                                            |
| H                                | = | Film heat transfer parameter, Btu/h-ft <sup>2</sup> -°F (4.4.3.2, Equation 52)      |
| H                                | = | Adiabatic head, ft                                                                  |
| h                                | = | Altitude, miles (4.2)                                                               |
| h                                | = | Surface heat transfer coefficient, Btu/h-ft <sup>2</sup> -°F (4.4.2.3)              |
| h <sub>c</sub>                   | = | Film condensation coefficient, Btu/h-ft <sup>2</sup> -°F (4.6.1.6)                  |
| h <sub>r</sub>                   | = | Radiation heat transfer coefficient, Btu/h-ft <sup>2</sup> -°F                      |
| ID                               | = | Inside diameter, in                                                                 |
| J                                | = | Radiosity, Btu/h-ft <sup>2</sup>                                                    |
| K                                | = | Constant, dimensionless (4.4.2.3)                                                   |
| K                                | = | Fin thermal conductivity, Btu/s-ft-°F                                               |

## 1.3 (Continued):

|                  |   |                                                                                |
|------------------|---|--------------------------------------------------------------------------------|
| K                | = | Heat transfer coefficient, dimensionless (Equation 31)                         |
| K                | = | Parameter for physical properties of fluid (4.6.1.2, Equation 89)              |
| k                | = | Thermal conductivity, Btu-ft/h-ft <sup>2</sup> -°F                             |
| k'               | = | Apparent thermal conductivity, Btu-ft/h-ft <sup>2</sup> -°F                    |
| L                | = | Fin length, ft (4.6.1)                                                         |
| L                | = | Latent heat of evaporation, Btu/lb                                             |
| M                | = | Mach number, dimensionless                                                     |
| m                | = | Molecular weight (5.2)                                                         |
| m                | = | Fluid mass, lb                                                                 |
| N                | = | Number of parallel tubes (4.6.1.6)                                             |
| N <sub>s</sub>   | = | Specific speed, dimensionless (4.6.3.4, Equation 122)                          |
| N <sub>Bo</sub>  | = | Bonde number = $\frac{D_h^2 \rho g k g_0}{\sigma}$ (4.6.1.1)                   |
| N <sub>We</sub>  | = | Weber number = $\frac{D_h \rho g V^2}{\sigma}$ (4.6.1.1)                       |
| N <sub>Re</sub>  | = | Reynolds number = $\frac{\rho V \delta}{\mu}$ (4.6.1.5)                        |
| n                | = | Number of shields (4.2.2.1)                                                    |
| P                | = | Pump power parameter, dimensionless (4.4.3.2)                                  |
| P                | = | Pressure, psi (4.4)                                                            |
| Q                | = | Internal heat load rate, Btu/h (4.2)                                           |
| Q                | = | Heat input, Btu (4.2)                                                          |
| Q                | = | Heat load, Btu/min (4.4.3.3), Btu/s (4.6.2.3)                                  |
| Q <sub>abs</sub> | = | Incident solar radiation absorbed, Btu/h-ft <sup>2</sup> (4.4.6.1)             |
| Q <sub>c</sub>   | = | Cooling load, Btu/min (4.6.1)                                                  |
| Q <sub>f</sub>   | = | Volumetric flow rate, ft <sup>3</sup> /min                                     |
| Q <sub>R</sub>   | = | Condenser-radiator heat rejection rate, Btu/s (4.6.1.4)                        |
| q                | = | Heat transfer rate, Btu/h (4.2.1.1 and 4.4.2.3)                                |
| q'               | = | Heat transfer rate/unit area, Btu/h-ft <sup>2</sup>                            |
| Q/V <sub>1</sub> | = | Refrigeration effect/unit volume of refrigerant, Btu/ft <sup>3</sup> (4.6.2.2) |
| R                | = | Universal gas constant, ft-lb/lb-°R                                            |
| R                | = | Earth radius, miles                                                            |
| R                | = | Reflected solar energy from planet, Btu/h-ft <sup>2</sup>                      |
| R <sub>R</sub>   | = | Radiation resistance, h-°F/Btu (4.4.6.1, Equation 80)                          |
| R <sub>T</sub>   | = | Combined conduction and radiation resistance, h-°F/Btu (4.4.6.1, Equation 81)  |
| r                | = | Pressure ratio, dimensionless (4.6.1 and 4.6.3.2)                              |
| r                | = | Albedo, dimensionless                                                          |
| r                | = | Parameter, dimensionless (Equation 6)                                          |
| S                | = | Solar radiation, W/ft <sup>2</sup> (4.2.2.1)                                   |
| S                | = | Solar constant = 430 Btu/h-ft <sup>2</sup> (Figure 20)                         |
| S                | = | Entropy, Btu/°R                                                                |
| T                | = | Temperature, absolute, °R                                                      |

## 1.3 (Continued):

|                |   |                                                                                             |
|----------------|---|---------------------------------------------------------------------------------------------|
| t              | = | Temperature, °F, °C                                                                         |
| u              | = | Velocity, ft/h                                                                              |
| u              | = | Volume flow, ft <sup>3</sup> /h (4.4.3.3)                                                   |
| V              | = | Axial velocity, ft/s                                                                        |
| W              | = | Weight, lb                                                                                  |
| W              | = | Combined tube and fin weight, lb                                                            |
| w              | = | Weight flow, lb/h (4.4.2.3)                                                                 |
| w              | = | Net heat flow per unit area, W/ft <sup>2</sup> (4.2.2.1, Equation 9, Figure 16)             |
| X              | = | Tube length, ft (4.6.1.5)                                                                   |
| Y              | = | Expansion factor, dimensionless                                                             |
| Y              | = | Blade twist, dimensionless (4.6.1.2, Equation 88)                                           |
| Z <sub>0</sub> | = | Combined figure of merit, 1/K (4.5.5, Equation 83)                                          |
| $\alpha$       | = | $\frac{2\pi\delta}{\lambda}$ = Wave number (4.6.1.5, Figure 66)                             |
| $\gamma$       | = | Ratio of specific heats, ( $c_p/c_v$ ), dimensionless                                       |
| $\delta$       | = | Fin thickness, in                                                                           |
| $\delta$       | = | Film thickness, ft (4.6.1.5, Equation 98)                                                   |
| $\epsilon$     | = | Porosity factor (4.6.1.3)                                                                   |
| $\epsilon$     | = | Emissivity, dimensionless                                                                   |
| $\epsilon$     | = | Dielectric constant, dimensionless (Table 14)                                               |
| $\eta_f$       | = | Efficiency, dimensionless                                                                   |
| $\eta_c$       | = | Compressor efficiency, dimensionless                                                        |
| $\eta_{ct}$    | = | Efficiency based on Carnot cycle, dimensionless                                             |
| $\eta_t$       | = | Turbine efficiency, dimensionless                                                           |
| $\theta$       | = | Angle between incident solar rays and the normal to the surface, deg                        |
| $\theta_s$     | = | Angle between Earth-Sun line and vertical from planet to satellite, deg                     |
| $\theta_1$     | = | Slant angle measured from skin surface, deg                                                 |
| $\theta_2$     | = | Irradiation angle with respect to the normal to surface, deg                                |
| $\lambda$      | = | Wavelength, microns                                                                         |
| $\mu$          | = | Micron (10 <sup>-6</sup> m)                                                                 |
| $\mu$ in       | = | Microinch (10 <sup>-6</sup> in)                                                             |
| $\mu$          | = | Viscosity, poise (4.4.2.4)                                                                  |
| $\mu$          | = | Viscosity, lb/ft-s                                                                          |
| $\rho_g$       | = | Specific weight (density), lb/ft <sup>3</sup>                                               |
| $\sigma$       | = | Surface tension, lb/ft (4.6.1.3), dynes/cm (4.6.1.1)                                        |
| $\sigma$       | = | Stefan-Boltzmann constant (0.173 × 10 <sup>-8</sup> ) Btu/h-ft <sup>2</sup> -T <sup>4</sup> |
| $\tau$         | = | Time, s, min, h                                                                             |
| $\Delta\tau$   | = | Time interval                                                                               |
| $\phi$         | = | Slant angle, deg (4.6, Equation 64)                                                         |
| $\omega$       | = | Rotational speed, rpm (4.6.1.2)                                                             |

## 1.3 (Continued):

## Subscripts:

|     |   |                                          |
|-----|---|------------------------------------------|
| a   | = | Air                                      |
| a   | = | Axial                                    |
| abs | = | Absorbed                                 |
| ad  | = | Adiabatic                                |
| b   | = | Boiling                                  |
| C   | = | Condenser                                |
| c   | = | Cold                                     |
| c   | = | Compressor                               |
| c   | = | Conductance                              |
| c   | = | Critical flow (5.1.1)                    |
| c   | = | Critical point (5.2)                     |
| cd  | = | Conductivity of gas (4.2.1.3)            |
| ct  | = | Carnot                                   |
| cv  | = | Conductivity due to convection (4.2.1.3) |
| E   | = | Earth or planet (4.2)                    |
| E   | = | Equipment or compartment (4.4.2.3)       |
| E   | = | Source heat exchanger (4.6.1 and 4.6.3)  |
| e   | = | Earth                                    |
| e   | = | Evaporator                               |
| e   | = | Smaller (4.6.1.6)                        |
| eff | = | Effective                                |
| ex  | = | Exit                                     |
| f   | = | Fin                                      |
| f   | = | Fluid                                    |
| fg  | = | Evaporation (4.6.1.3)                    |
| g   | = | Gas                                      |
| gen | = | Generated                                |
| H   | = | Plumbing hardware                        |
| Hx  | = | Heat exchanger                           |
| h   | = | Hot                                      |
| h   | = | Hydraulic                                |
| in  | = | Inlet                                    |
| L   | = | Length                                   |
| m   | = | Mean                                     |
| n   | = | Number                                   |
| o   | = | Internal body surface                    |
| P   | = | Pump                                     |
| R   | = | Reflected solar                          |
| R   | = | Rejection                                |
| r   | = | Radiation                                |
| ra  | = | Radiation (4.2.1.3)                      |
| s   | = | Solar                                    |

## 1.3 (Continued):

|      |   |                                                      |
|------|---|------------------------------------------------------|
| s    | = | Specific speed                                       |
| sc   | = | Conductivity of solid conductor due to fiber content |
| sink | = | Heat sink                                            |
| T    | = | Tank (4.4.2.3)                                       |
| T    | = | Overall or total (4.2 and 4.4.6.1)                   |
| t    | = | Tube                                                 |
| turb | = | Turbine                                              |
| *    | = | Reference area                                       |
| 0    | = | Reference properties                                 |

## Superscripts:

|           |   |           |
|-----------|---|-----------|
| $\bar{x}$ | = | Average x |
|-----------|---|-----------|

## 1.4 Common Abbreviations:

|         |   |                                          |
|---------|---|------------------------------------------|
| %       | = | Percent                                  |
| abs     | = | Absolute                                 |
| AC      | = | Alternating Current                      |
| A       | = | Ampere                                   |
| Al, AL  | = | Aluminum                                 |
| ARS J.  | = | American Rocket Society Journal          |
| ASD     | = | Aeronautical Systems Division (USAF)     |
| ASME    | = | American Society of Mechanical Engineers |
| atm     | = | Atmosphere                               |
| bp (Bp) | = | Boiling Point                            |
| Btu     | = | British Thermal Unit                     |
| °C      | = | Degrees Centigrade                       |
| cal     | = | Calorie                                  |
| cm      | = | Centimeter                               |
| comp    | = | Compressor                               |
| COP     | = | Coefficient of Performance               |
| cps     | = | Cycles per second                        |
| ct      | = | Carnot                                   |
| DC      | = | Direct current                           |
| deg     | = | Degree                                   |
| dia.    | = | Diameter                                 |
| ECS     | = | Environmental control system             |
| eff     | = | Efficiency                               |
| Emf     | = | Electromotive force                      |
| °F      | = | Degrees Fahrenheit                       |
| ft      | = | Feet                                     |
| g       | = | Gravity                                  |
| gal     | = | Gallon                                   |

## 1.4 (Continued):

|                        |                                               |
|------------------------|-----------------------------------------------|
| g                      | = Gram                                        |
| gpm                    | = Gallons per minute                          |
| hp                     | = Horsepower                                  |
| h                      | = Hour                                        |
| Hx                     | = Heat exchanger                              |
| Hz                     | = Hertz, cps                                  |
| in                     | = Inch                                        |
| Inst.                  | = Institute                                   |
| J.                     | = Journal                                     |
| K                      | = Kelvins (formerly called degrees Kelvin)    |
| kg                     | = Kilogram                                    |
| kW                     | = Kilowatt                                    |
| lb                     | = Pound                                       |
| m                      | = Meter                                       |
| max                    | = Maximum                                     |
| mil                    | = 0.001 in                                    |
| Mil.                   | = Military                                    |
| min                    | = Minimum                                     |
| min                    | = Minute                                      |
| MIT                    | = Massachusetts Institute of Technology       |
| mm                     | = Millimeter                                  |
| mol                    | = Mole                                        |
| mol.                   | = Molecular                                   |
| NACA                   | = National Advisory Committee for Aeronautics |
| p. (pp.)               | = Page(s)                                     |
| psfa                   | = Pounds per square foot absolute             |
| psi                    | = Pounds per square inch                      |
| psia                   | = Pounds per square inch absolute             |
| psig                   | = Pounds per square inch gage                 |
| °R                     | = Degrees Rankine                             |
| rms                    | = Root-mean-square                            |
| rpm                    | = Revolutions per minute                      |
| SAE                    | = Society of Automotive Engineers             |
| sat                    | = Saturated                                   |
| s                      | = Second                                      |
| std (STD)              | = Standard                                    |
| TN                     | = Technical Note                              |
| ton (of refrigeration) | = 200 Btu/min                                 |
| TR                     | = Technical report                            |
| Trans                  | = Transactions                                |
| USAF                   | = United States Air Force                     |
| vol                    | = Volume                                      |
| W                      | = Watt                                        |
| WADC                   | = Wright Air Development Center (USAF)        |

#### 1.4 (Continued):

|          |   |                                                  |
|----------|---|--------------------------------------------------|
| WADD     | = | Wright Aeronautical Development Division (USAF)  |
| wt       | = | Weight                                           |
| $\Delta$ | = | Increment                                        |
| $\mu$    | = | Micron, $10^{-6}$ m ( $3.937 \times 10^{-5}$ in) |
| $\Sigma$ | = | Summation                                        |

## 2. ESTABLISHMENT OF DESIGN REQUIREMENTS:

Satisfactory thermal design of equipment environmental control systems (ECS) is dependent in a large measure on accurate as well as adequate knowledge of the environmental limits of the equipment. This area is of special concern to the environmental control system designer because it is obviously one of the controlling factors in the weight, size, and power requirements, and particularly in the configuration of the environmental control system. It is not only essential that complete design criteria be established but also that repeated efforts be made to reduce and simplify requirements during the course of development.

### 2.1 Parameters to be Studied:

In order to evolve sufficient criteria for the task of providing a satisfactory environment for equipment, the following parameters must be studied:

1. Heat input rate and required heat dissipation rate in watts or Btu/h. Where parts of equipment operate intermittently or at a varying percentage of their full load rating, this should be noted together with heat input versus time, absolute pressure, or other pertinent variables.

It should be also noted that the heat input to the accessory package and the required heat dissipation rate are equal only in the prolonged steady-state condition. The difference between the two is ordinarily the heat that is absorbed into the equipment itself as a function of the specific heats of the various materials used in its construction.

Where the mass of the accessory is large and the time of the accessory temperature stabilization long, the required heat dissipation rate can be much lower than the heat input rate for short-time operation. Where this factor is important, the heat dissipation rate should be stated as a function of time or other pertinent variables.

2. Maximum permissible ambient temperature conditions versus minimum average coolant supply to the equipment and versus absolute pressure and heat input rate. Ambient temperature conditions are here defined as the temperature of the ambient gas and/or the temperature of surrounding solid surfaces (wall temperatures of test chamber or enclosing compartment).
3. Temperature rise of critical equipment elements versus time and heat input in maximum ambient conditions noted in item 2.

## 2.1 (Continued):

4. Life of critical elements versus temperature of critical elements.
5. Approximate envelope dimensions, position and size of louvers or other openings, location and performance (versus absolute pressure) of any internal fans, location and dimensions of any special heat transfer fins, and color and type of surface finish.
6. Provisions for attachment of equipment to vehicle. For example, accessories mounted by shock mounts are effectively isolated against conduction from the vehicle structure, but accessories mounted metal to metal on structure may achieve considerable cooling or heating by direct conduction.
7. Maximum permissible temperature of outside surface of equipment versus heat input rate.
8. Temperature rise of surface of package versus time and heat input, with stated ambient temperature and coolant supply conditions.
9. Quantity of gas required for pressurization of equipment, in lb/min, versus compartment absolute pressure.
10. Minimum required flow rate and maximum permissible temperature of coolant into the package versus various stated ambient temperature conditions; items 1 to 9 versus compartment absolute pressure and heat input rate.
11. Anticipated maximum coolant exit temperatures corresponding to inflow conditions noted.
12. Pressure loss of coolant flowing through package or heat transfer surface versus coolant flow rate and density.
13. Temperature rise of critical accessory elements versus time and heat input, with no cooling and with specified quantities of coolant.
14. Location and size of coolant interfaces.
15. Layout of heat transfer surfaces in package and mean operating temperature of these surfaces.
16. Specification of type or class of package seal. MIL-S-8484 (USAF) "Seals and Seal Testing Procedure for Electronic Enclosures" may be used as a guide.
17. Positive and negative pressure differential capability of the package, along with surface deflection.
18. Humidity and contamination control requirements.

## 2.1 (Continued):

19. Ground support requirements; interfaces, temperature, pressure, and flow of coolant.
20. Storage environmental requirements under sea level and vacuum conditions.

## 3. REGULATION, SENSING, AND CONTROL OF ENVIRONMENTAL SYSTEMS:

### 3.1 Temperature:

- 3.1.1 Performance: Often, temperature control systems designed for space vehicle applications have not performed as well as expected. As a result, several major redesign efforts are required before an acceptable system is developed. This poor system performance can usually be traced to the improper choice of locations and types of sensors, controller and regulator devices, and the neglect of dynamic response analysis for prediction of system performance.

From a consideration of the analysis approach used for most designs, the heat transfer and thermodynamic analyses appear satisfactory, but the control and flow rate choices are usually based on maximum and minimum temperature and heat load considerations. Obviously, this procedure does not lead to an optimum and/or a well-controlled system, since neither consideration for the heat storage of the equipment, coolant, and components nor prediction of control system response is included.

The real importance of dynamic response analysis for space vehicle temperature control systems is not apparent unless a detailed breakdown of the system heat loads is considered. The basic heat loads can be listed as:

1. Solar, reflected, and planetary radiant heat (variable with vehicle spin, orientation, distance from Sun and orbit conditions).
2. Equipment and personnel heat load to heat exchangers and cold plates (variable with time and/or function).

As can be seen, the heat load for most vehicles must vary with time. Thus, there should be no doubt that such factors as speed of response, damping factor, phase lag and dead band, stability, control band, and prediction of temperature versus time for individual components are necessary outputs to any analysis. (Note that, in most cases, the use of dynamic response analysis methods is necessary for optimization, since the system designed without dynamic response analysis will be overdesigned to supply control for the maximum and minimum heat load conditions.)

Apparently, the main cause of neglect of dynamic response analysis is the fact that most texts on dynamic response systems are directed toward hydraulic, pneumatic, and electrical systems, with little or no mention of temperature control applications.

### 3.1.1 (Continued):

Another cause is the educational background requirement of three or four courses in linear and nonlinear system analysis needed for basic understanding of the block diagram Laplace transform analysis method. Also note that a block diagram Laplace transform solution becomes extremely complex or impossible when a temperature control system with fluctuating heat loads and temperature control by fluid flow modulation is used.

Another method of analysis, the differential equation method, uses the basic differential equations for temperature, heat transfer, and fluid flow in a form that can be solved by computer programs or analog networks.

- 3.1.2 Components: Components associated with the temperature control circuits are sensors, anticipators, amplifiers and controllers, servomechanisms, and feedback position indicators. Figures 1 through 4 illustrate the relationship of the controller circuit to some simple semipassive temperature control methods. Even with the controller circuit setup as shown in these figures, numerous controller circuit component problems exist in choice of sensors, controllers, and servomechanisms. To show some of the choices available, tables of methods of controller operation and of sensor and thermoswitch operation and accuracy are shown in Tables 1 and 2.

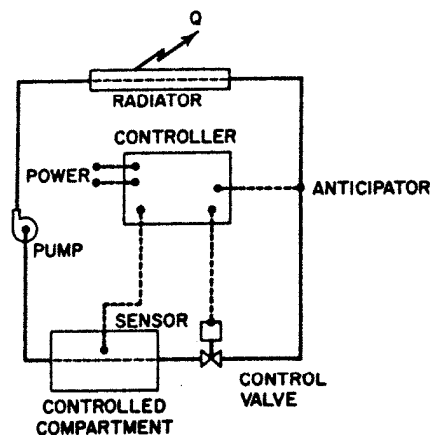


FIGURE 1 - Temperature Control by Fluid Flow Modulation

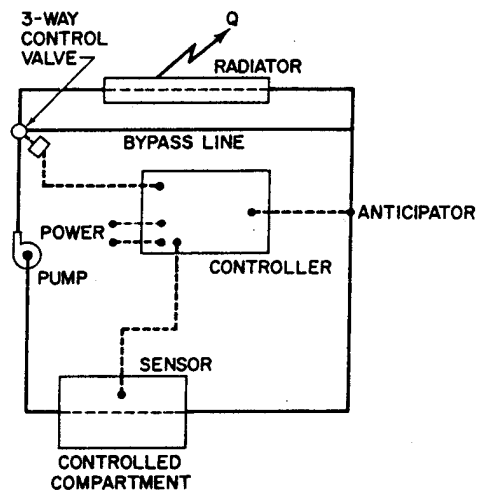


FIGURE 2 - Temperature Control by Mixing and Bypass

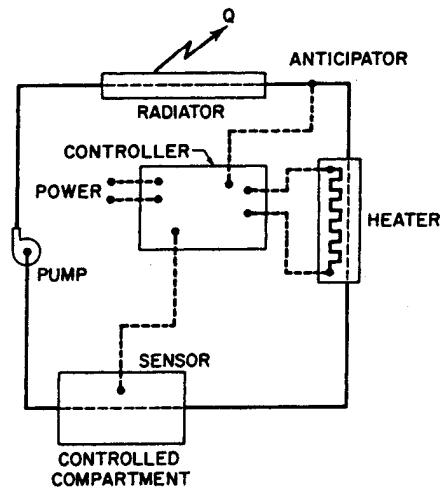


FIGURE 3 - Temperature Control by Heat Addition

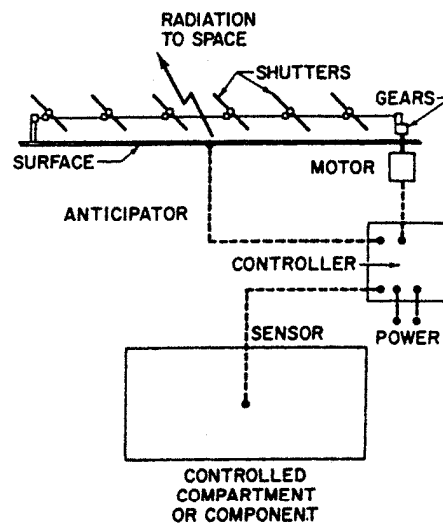


FIGURE 4 - Temperature Control by Varying Surface Emissivity and Solar Absorptivity

TABLE 1 - Types of Automatic Controllers for Temperature Control Systems

| Type of Controller                  | Description                                                                                                                                                                                                                              |
|-------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| On-Off                              | Receives signals, compares, amplifies, and opens or closes a servo such as a solenoid valve                                                                                                                                              |
| Proportional                        | Receives signals, compares, amplifies, and modulates servo proportional to signal                                                                                                                                                        |
| Constant Frequency, Pulse Modulated | Receives signals, compares, amplifies, and modulates pulse width of constant frequency on-off servo action                                                                                                                               |
| Feedback                            | Position indicator sends signal to controller for comparison to rate of response of servo. This allows speedup of response and increased accuracy in positioning of servo. Can be used with proportional or pulse modulated controllers. |

TABLE 2 - Temperature Sensors and Thermostats

| Type                                | Range,<br>Low | Range,<br>High | Sensi-<br>tivity | Controller<br>Required? | Description                                                                                        |
|-------------------------------------|---------------|----------------|------------------|-------------------------|----------------------------------------------------------------------------------------------------|
| Wire Resistor                       | -460          | +5000          | $\pm 0.05$ °F    | Yes                     | Resistance of wire increases with temperature                                                      |
| Thermistor                          | -200          | +900           | $\pm 0.05$ °F    | Yes                     | Resistance decreases with temperature                                                              |
| Thermocouple:                       |               |                |                  |                         |                                                                                                    |
| Copper-Constantan                   | -460          | +600           | $\pm 0.05$ °F    | Yes                     | Emf at junction increases with temperature                                                         |
| Iron-Constantan                     | 0             | +1600          | $\pm 0.1$ °F     |                         |                                                                                                    |
| "Chromel-Alumel"<br>(Hoskins Corp.) | +600          | +2100          | $\pm 0.2$ °F     |                         |                                                                                                    |
| Platinum-Platinum<br>Rhodium        | +1300         | +2800          | $\pm 0.1\%$      |                         |                                                                                                    |
| Mercury Thermostat                  | -80           | +750           | $\pm 0.1$ °F     | No                      | Length of mercury column increases with temperature, contact with points can be used for switching |
| Bimetallic Disk<br>Thermostat       | -100          | +1000          | $\pm 3\%$        | No                      | Bimetallic disc snaps into reverse position at desired temperature and can be used for switching   |
| Bimetallic Expansion<br>Thermostat  | -100          | +1000          | $\pm 1\%$        | No                      | Differences in thermal expansion of bimetallic strip cause switching                               |
| Thermal Expansion<br>Thermostat     |               | +1800          | $\pm 1\%$        | No                      | Thermal expansion of material causes switching                                                     |

## 3.1.2 (Continued):

In consideration of the controller circuit, the controller is usually the key component. These controllers can be extremely complex or nonexistent, depending on the temperature control system. For example, in a complex system with precise temperature control requirements, the controller may consist of electronic circuits to:

1. Interpret sensor and anticipator measurement signals.
2. Amplify the interpreted signal.
3. Direct a signal to the servomechanism for corrective action.
4. Alter the output signal to linearize the servo action.
5. Interpret and compare feedback signals from position indicators with the desired servo control position to speed up the rate of response.

### 3.1.2 (Continued):

The opposite in complexity is the thermostitch and solenoid valve type of combination wherein the thermostitch directly controls the valve and no controller is needed.

At any rate, in most cases a controller of some form is required. This controller is usually electronic for space applications because electronic controllers are usually lighter, more reliable, and smaller than hydraulic or pneumatic controllers. (Also, electricity is more readily available.) Much useful information on controllers for particular applications can be obtained from the electronic controller companies. Some of these companies will not only design a controller for the application, but will also analyze the dynamic response of the system.

### 3.2 Compartment Pressure Regulation:

Compartment pressure regulation for space vehicles is dependent on the allowable positive and negative pressure differentials established by structural requirements for ascent and reentry flight phases, respectively, and the total pressure level that must be controlled during orbit of space flight under hard vacuum conditions.

Generally, positive pressure relief is required during ascent, to reduce internal total pressure to the level that will be maintained throughout the space mission. Once the prescribed compartment total pressure level has been reached during ascent, further loss of compartment gas occurs only from leakage. Internal pressure regulation is accomplished by controlled addition of makeup gas. The method and rate of makeup gas addition depends on the allowable control tolerances and leakage rate.

During reentry and subsequent terminal flight to touchdown, negative pressure relief is required to prevent excessive crushing pressures on the structure. If inflow of hot boundary layer air cannot be tolerated, internal pressure increase to match ambient pressure increase must be accomplished by the onboard pressure regulation system. A typical compartment pressure schedule is shown in Figure 5.

A compartment pressure regulation system may consist of a supply of makeup gas stored at high pressure, a pressure regulator, a flow control valve and/or orifice, pressure sensors, and a positive and negative pressure relief valve.

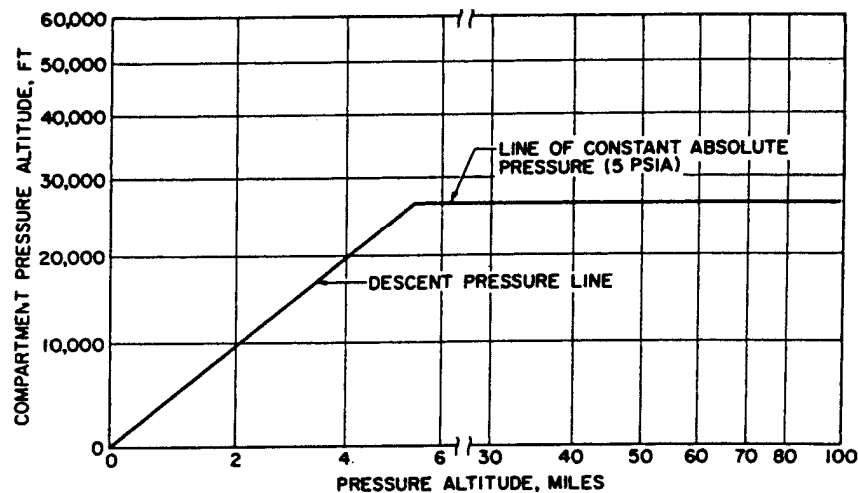


FIGURE 5 - Compartment Pressure Schedule Chart

- 3.2.1 High-Pressure Regulation: A high-pressure regulator is required to reduce the supply makeup gas pressure from its stored level to a constant reduced level on which to base flow control valve and/or orifice sizes and the supply flow rate. A typical regulator is a balanced poppet type valve operated pneumatically and automatically by a metal diaphragm. Single stage pressure reductions as high as 75:1 are possible.
- 3.2.2 Low-Pressure Regulation: Low-pressure regulation is accomplished by the controlled addition of a makeup gas quantity to maintain the required compartment total pressure level. Excess quantities of gas are relieved overboard through the compartment pressure relief valve.

Makeup gas control can be accomplished by either an on-off shutoff valve controlled by compartment total pressure sensing or by an orifice sized to provide a gas flow equal to the total leakage flow from the compartment. In the latter case, a shutoff valve will be required in series with the orifice, to prevent loss of makeup gas until pressure control is required.

The compartment pressure relief valve functions to maintain the internal-to-ambient pressure differential within allowable limits for all phases of the flight. Such a valve (Figure 6) contains an aneroid that senses compartment pressure and is adjusted to close a small pilot valve when the compartment pressure reaches the desired absolute pressure. The small pilot valve, in turn, blocks the escape of gas from the outflow valve head chamber when the pilot valve is in this position.

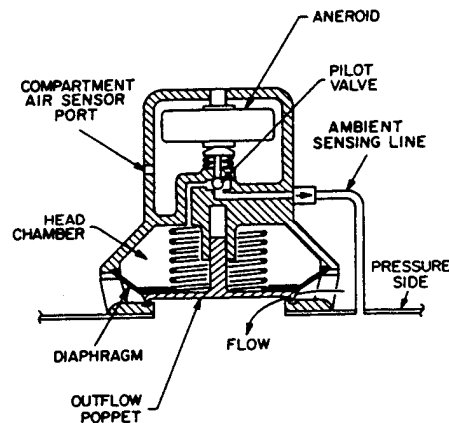


FIGURE 6 - Compartment Relief Valve

### 3.2.2 (Continued):

As the pressure outside the compartment drops, the cabin-to-ambient differential pressure provides a closing force on the outflow poppet, which helps prevent leakage. Regulators may be adjustable and limit the compartment pressure range within less than  $\pm 5\%$  of the nominal setting. Leakage of a typical regulator with 0.5 psia outlet pressure and 5 psia inlet pressure may be expected to be of the order of 0.0005 lb/h.

Regulators are generally mounted inside and require a sensing line to the atmosphere. The pneumatic sensing line is provided with a check valve which causes the regulator to open automatically when atmospheric pressure exceeds compartment pressure.

For maximum reliability, it may be necessary to provide a duplicate regulator that will act in the capacity of a safety valve. This regulator would be identical to the compartment pressure regulator described above except for calibration. Obviously, the setting of the standby regulator would be at pressure somewhat higher than the primary regulator. Pressure setting of the backup unit is established by the structural criteria.

## 4. TEMPERATURE CONTROL:

The basic problem involved in aerospace equipment temperature control is the provision of a transfer loop from the internal heat source to various types of heat sink systems. If this loop consists of no fluids and no moving parts, it is considered "passive." The system becomes "semipassive" when moving parts or fluids are introduced in order to energize the loop and reject heat at a value less than the source temperature. If a heat pump is employed, heat is rejected at a higher temperature than the source and the system is classified as "active."

#### 4.1 Passive Temperature Control Methods:

##### 4.1.1 Fixed Resistance Heat Path (Insulation):

Figure 7 shows how a fixed resistance is used to control temperature.

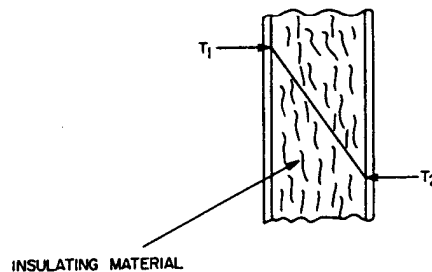


FIGURE 7 - Temperature Control by Fixed Resistance

The function of insulation in the control of temperature is to impede the transfer of heat across a space. This is often done by breaking up this space into very small volumes, thereby preventing the formation of the convection currents by which most of the heat transfer would occur. This effect is produced by the use of materials such as foams, powders, and matted hairlike or fibrous substances.

Heat transfer across a space by convection or conduction can be eliminated almost completely by producing a vacuum between the walls. However, this does not prevent radiation transfer, and the so-called superinsulations become necessary. These evacuated insulation types are laminated combinations of aluminum foil and Fiberglas paper or layers of thin metallized plastic film which provide multiple radiation barriers to reduce transfer of radiant energy.

Characteristics of the types of insulation described above include:

1. Power – No external power is required.
2. Weight – Densities vary from 3 to 15 lb/ft<sup>3</sup> for thermal conductivities in the range of 0.015 Btu/h-ft-°F at a mean temperature of 560 °R for unevacuated materials, to 0.005 Btu/h-ft-°F for evacuated materials at 560 °R. Laminated foil-Fiberglas insulation at an average temperature of 300 °R has a conductivity less than  $5 \times 10^{-5}$  Btu/h-ft-°F.
3. Method – Vacuum of outer space may be used to evacuate a desired region of vehicle. Insulation can be combined with self-sealing of walls.
4. Limitations – Extra weight. May prevent easy access to some structural areas.
5. General – Simple, no power requirements, nontoxic, noncorrosive; easy to apply.

- 4.1.2 Fixed Conductance Heat Path: This method of temperature control differs from the one previously described in that it has paths of high conductivity interposed between two areas rather than paths that are thermally isolated from each other. The purpose here is to remove heat, and this is done by joining the heat source and the heat sink by paths of high conductivity. The best conductive materials are metals, so that rods or plates of solid metal provide the necessary service.

Characteristics include:

1. Power – No power is required.
  2. Weight – Depends primarily on length and cross-sectional area of path and upon density and thermal conductivity of metals used. The density of common materials varies from 114 lb/ft<sup>3</sup> for beryllium to 556 lb/ft<sup>3</sup> for copper.
  3. Integration – Sections of structure, enclosures, fluid conduits, or protective shielding can all be used.
  4. Limitations – The best conducting materials have the highest densities. Radiation from conductive paths may necessitate insulation.
  5. General – A simple, reliable process involving easily predicted behavior of familiar materials and numerous possibilities for integration.
- 4.1.3 Heat Storage: Heat storage devices may be used to minimize the temperature excursion of space vehicles subjected to extremes of cooling and heating. A storage device may also be utilized as a heat sink for active systems during periods when normal rejection to space is impossible or inadequate.

These devices may become solid or liquid materials, but not, ordinarily, gases. The important properties to be considered in selecting heat storage materials are high thermal capacity and high latent heat of fusion. While gases may possess these required properties, the specific volumes of gases at practical pressures are excessive.

Thermal properties of some promising materials for heat storage are tabulated in Table 3.

TABLE 3 - Thermal Properties of Some Heat Storage Materials

| Material             | Melting Point, °F | Latent Heat of Fusion, Btu/lb | Heat Absorbed from 70 °F Through Fusion, Btu/lb | Chemical Symbol             |
|----------------------|-------------------|-------------------------------|-------------------------------------------------|-----------------------------|
| Tristearin           | 133               | 82.1                          |                                                 | $(C_{17}H_{35}COO)_3C_3H_5$ |
| Beeswax              | 143               | 76.1                          |                                                 |                             |
| Methyl Bromobenzoate | 178               | 54.4                          |                                                 | $C_8H_2O_2Br$               |
| Catechol             | 220               | 88.9                          |                                                 | $C_6H_4(OH)_2$              |
| Phenacetin           | 279               | 58.8                          |                                                 | $CH_3CONHC_6H_4OC_2H_5$     |
| Hydroquinone         | 342               | 111.1                         |                                                 | $C_6H_6O_2$                 |
| P-Aminobenzoic Acid  | 371               | 65.7                          |                                                 | $C_6H_4(NH_2)COOH$          |
| Lithium              | 367               | 59.5                          | 270 <sup>1</sup>                                | Li                          |
| Beryllium            | 2330              | 468                           | 2032 <sup>1</sup>                               | Be                          |
| Silicon              | 2570              | 776                           | 1330 <sup>1</sup>                               | Si                          |

<sup>1</sup> Computed from available data.

#### 4.1.3 (Continued):

Characteristics of these materials are as follows:

1. Power – In themselves, storage systems require no power, but circulation may be required to transport heat to storage.
2. Weight – Based on Table 3 and allowing 100% extra for containers and circulating system, the range is 4.2 to 24.3 lb/kW-h.
3. Method – Storage material may be any body, solid, or liquid that is on board and which will not be materially altered by the addition of heat.
4. Limitations – The rate of transfer may be slow and recovery more difficult than accretion. Containment of material may be difficult.
5. General – Simplicity and wide possibilities of integration make this method attractive.

- 4.1.4 Heat of Solution Storage System: This is a unique method for storing heat, or more accurately, the effects of heat. It depends upon the fact that the addition of water to concentrated solutions of some alkaline materials is attended by the production of considerable quantities of heat. After reaching a certain degree of dilution, no more heat is given off, so that the solution must then be reconcentrated.

Excessive heat can be used to drive off water, which is condensed and then re-used to dilute the reconcentrated solution. Some representative values are given in Table 4.

TABLE 4 - Heats of Solution of Representative Substances

| Solution                                        | Molecular Weight | Heat of Solution<br>(18 °C, 1 atm) | Heat of Solution<br>(18 °C, 1 atm) |
|-------------------------------------------------|------------------|------------------------------------|------------------------------------|
|                                                 |                  | kg-cal/g-mol                       | Btu/lb                             |
| CaCl <sub>2</sub>                               | 111.0            | 18.0                               | 292                                |
| NaOH                                            | 40.0             | 10.3                               | 463                                |
| MgBr <sub>2</sub>                               | 184.2            | 43.3                               | 423                                |
| BeBr <sub>2</sub>                               | 168.5            | 62.6                               | 668                                |
| AlCl <sub>3</sub>                               | 133.3            | 77.9                               | 1054                               |
| Al <sub>2</sub> (SO <sub>4</sub> ) <sub>3</sub> | 342.1            | 126.0                              | 662                                |

## 4.1.4 (Continued):

Characteristics of these substances include:

1. Power – No external power is required beyond that needed for circulation of the heat exchange medium
2. Weight – A representative weight is indicated by the heat of solution value of 249 W-h/lb of solution for 1 mol of aluminum chloride, AlCl<sub>3</sub>, dissolved in 2 mols of water.
3. Method – This system can act to store heat and release it on demand, or it can be used to generate energy, and so can be integrated with either a heat sink or a heat source.
4. Limitations – Materials are usually highly corrosive. Temperatures of system operation are relatively low.
5. General – Materials are cheap and readily obtainable in pure form. The system is reliable and easily controlled. The high-temperature requirement of meltable heat storage materials is avoided.

4.1.5 Radiators: A radiator is any surface whose purpose it is to emit heat by electromagnetic radiation. Heat may be transported to the radiator by means of conduction, free or forced convection, or radiation (Figure 8). In the interior of a vehicle, heat may be removed from the face of a radiator by radiation or by the passage of convective air currents, whereas in space the only means of transmission is through radiation.

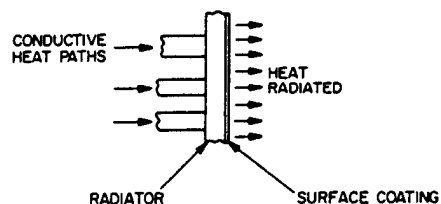


FIGURE 8 - Heat Disposal by Radiation

#### 4.1.5 (Continued):

The interchange of radiant energy between the face of a radiator and its surroundings is in accordance with the Stephan-Boltzmann law, which states that this interchange will be a function of the emissivity and the fourth power of the absolute temperatures of the surfaces.

Emissivity is a property of the material and condition of the surface, being low for gold, silver, and aluminum, and high for black oxides of metals. Ideally, a radiator should have high thermal conductivity and low density in order to minimize weight.

Characteristics of radiators are:

1. Power – No power is required to operate the radiator, although a fluid transporting heat to the radiator would have to be moved by some mechanical means.
2. Weight – The specific weight (weight per unit of heat transfer) would be a function of the overall effectiveness of the radiator, the density of the material, and other factors, such as whether the radiator forms an integral part of the vehicle structure or surface.
3. Method – Can be integrated with vehicle outer surface or auxiliary surfaces, such as radar antennas.
4. Limitations – Ordinarily not subjected to control, although movable shutters may alter emissivity of surface. Surface properties are subject to change through meteoritic damage, high vacuum, and other space environmental factors.
5. General – Insensitive to gravity field; long lived; high reliability; no power requirements; and (ordinarily) no moving parts.

#### 4.2 Technical Approach, Passive Methods:

Passive methods previously discussed in this section depend upon conduction, radiation, or a combination of conduction and radiation as a means for heat transfer. Since natural convection does not occur in a zero gravity environment, heat transfer by this process will not be considered here.

Figure 9 illustrates a simple passive temperature control system that depends on conduction and radiation for heat transfer.

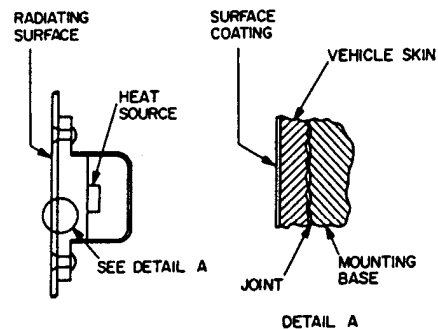


FIGURE 9 - Simple Passive Temperature Control System

#### 4.2 (Continued):

Passive methods, because of their relative simplicity, have been the sole means for temperature control of first generation space vehicles. Although more complex control methods will be required in future vehicles, passive methods will be an integral part of the overall temperature control system.

High conductance heat paths can be achieved by utilizing high-heat conducting material and by placing the heat source close to the heat sink, such as the vehicle skin. The conduction path should be designed with a minimum number of joints.

An important consideration for the efficient design of conduction paths is the effect of joints on heat transfer. Currently (circa 1965), there is a need for more data, particularly at very low pressures, and a development of methods for analysis. Limited information currently available indicate thermal resistance of joints can be minimized by:

1. Increasing contact pressure.
2. Reducing surface roughness.
3. Maintaining surface flatness.
4. Utilizing shim materials that have lower hardness than joint materials.

Through continued development efforts, high efficient insulations have been developed. Most efficient insulations require very low interstitial pressure to minimize or eliminate gas conduction, which is the major mode of heat transfer, accounting for 50 to 90% of the total conductance. In addition, radiation heat transfer is minimized by use of powder and fibrous materials and by utilizing multilayer foils. Foil type insulations are available for vehicle thermal conductivity of 0.000025 Btu/h-ft-°F. Being nonload bearing, this type of insulation usually requires additional structure for rigidity.

#### 4.2 (Continued):

Proper selection of surface coating for space vehicles is a very important consideration for the efficient design of a temperature control system. Vehicle skin temperature has direct influence on the interior temperature and thus affects temperature control requirements. Many surface coatings have been developed and provide a wide range of values for  $\alpha_s/\epsilon$ .

An analysis for multilayer radiation shields and the plots of equations developed indicate the marked effect of shields to reduce radiation heat transfer. As shown on Figure 19, three or four shields should be used for maximum reduction in the value of heat transfer coefficient. Increasing the number beyond four has only a small effect in reducing the heat transfer coefficient. Shielding material should have low values of emissivity, such as aluminum foil, for which  $\epsilon = 0.05$ .

The principal mode of heat transfer from heat source to the outer skin surface is by conduction. From the outer skin, surface heat is dissipated to space, the ultimate heat sink, by radiation. The prime requirement of this simple system is to provide an efficient heat path from the source to the sink. One very important consideration, which will be discussed, is the heat transfer across the interface of mating surfaces, illustrated in detail A of Figure 9.

In addition to discussing means for providing efficient heat transfer by conduction and by radiation, methods for providing effective heat resistance or insulation will also be considered. This is very important in providing thermal protection for space vehicles while in orbit and particularly during reentry.

- 4.2.1 Conduction: The subject of heat conduction has been treated extensively in many books on heat transfer, such as References 1 and 2. Therefore the discussion here is limited to selected material on the subject.

Study of heat flow through many materials has established the fact that materials differ in their ability to conduct heat. Those that are good conductors of heat are also good conductors of electricity. This includes well-known metals such as silver, copper, and aluminum. Many nonmetallic materials such as wood, liquids, and gases are poor heat conductors, and are classified as insulators. Thus, depending upon the material utilized, either high thermal conductance or high thermal resistance can be achieved.

- 4.2.1.1 Thermal Conductance: To obtain efficient heat transfer by conduction requires more than just the consideration of the conducting material. As indicated by the fundamental law of conduction (Fourier's equation)

$$q = \left( \frac{kA}{L} \right) \Delta T \quad (\text{Eq. 1})$$

high heat flow rate can be achieved by obtaining large values for the term  $(kA/L)$  or the temperature difference  $\Delta T$ .

#### 4.2.1.1 (Continued):

To obtain a large temperature difference between the heat source and sink requires, for a given heat source temperature, the determination of a heat sink at the desired lower temperature. For the temperature control system illustrated in Figure 9, the intermediate heat sink is the vehicle skin, so that  $\Delta T$  is dependent upon the skin temperature, which in turn is governed by the net energy exchange by radiation.

The term  $(kA/L)$  is a combination of the conducting property of the conductor and its physical dimensions. Good heat conductors, such as copper and aluminum, have specific gravity values greater than 1; therefore weight is an important consideration in addition to conductivity. Increase in area  $A$  would increase heat flow rate, but it would also result in a larger conductor, and hence an increase in weight. The most significant quantity is the length  $L$ , since a smaller value would result not only in higher heat flow rate, but also a lighter-weight conductor.

For the simple system illustrated in Figure 9, the heat source should be placed as close to the thermal radiating surface as possible. The conduction path shown is composed of several solid conductors, which introduce joints; thus, joint effect on heat flow becomes an important consideration.

- 4.2.1.2 Joint Resistance: For efficient design of conduction paths, knowledge of joint resistance is necessary. A survey of available literature (circa 1965) indicates a lack of analysis and experimental data directly applicable to spacecraft, particularly for extremely low pressure environment. Many variables (which include contact pressure, temperature, heat flow rate, interstitial pressure, interstitial fluid, joint materials, surface finish, material hardness, and surface flatness) must be taken into consideration to provide useful data on this subject.

Some of the more important earlier work on joint resistance was published in 1949. Weills and Ryder (Reference 3) conducted tests using copper, steel, and aluminum for varying temperature and pressure. Brunot and Buckland (Reference 4) used laminated and machined joints, using steel with cement, steel, and aluminum foils sandwiched between the surfaces; only pressure was varied.

In 1953, WADC (Reference 5) reported an experimental method of measuring thermal resistance by using a Mach-Zehnder interferometer. The results of these measurements were tabulated. Fenech (Reference 6) developed an idealized model for thermal contact and compared the theoretical analysis with experimental data for Armco-iron contact surfaces of 600 rms roughness for varying pressure and temperature. Groff (Reference 7) introduced a dimensionless conductance  $(h_c p/kp)$ , which was used to correlate test data presented by other investigations.

Investigators for NACA (References 8 to 11) have presented a considerable amount of experimental data on aircraft joints at atmospheric pressure for surface finish ranging from 10 to 120  $\mu\text{in}$ , rms surface roughness. Contact pressure, temperature, and heat load were the principal variables. Figure 10 presents a portion of the data given in Reference 9. The influence of surface finish and contact pressure on joint conductance is clearly indicated.

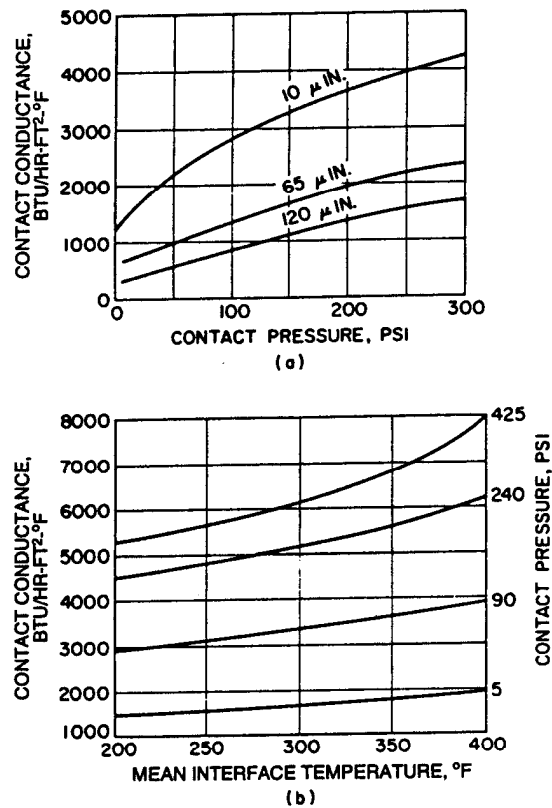


FIGURE 10 - Thermal Conductance Across Interface of Conductor Joint  
 (a) 75S-T6 Aluminum-to-Aluminum Joint;  
 (b) 75S-T6 Aluminum-to-Aluminum Joint, 10  $\mu$ in rms Surface Roughness

#### 4.2.1.2 (Continued):

Recently, Fried and Costello (Reference 12) have presented the results of experimental work for tests conducted at ambient pressures of  $10^{-4}$  to  $10^{-6}$  mm Hg abs and relatively low range of contact pressures (2 to 35 psi). The experimental values indicate that joint conductance is lower at vacuum conditions. Tests were also conducted utilizing various interface shim materials. The results of these tests are presented in Figure 11, indicating the marked influence of soft materials, such as lead foil, in increasing the joint conductance.

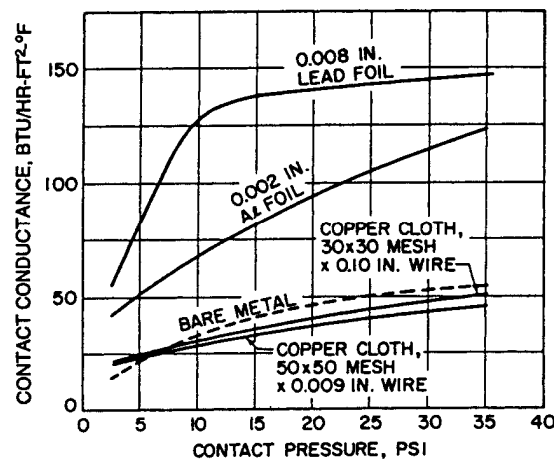


FIGURE 11 - Thermal Conductance Versus Contact Pressure for Joints With Shim Material

4.2.1.2 (Continued):

Based on the review of available reports, joint resistance can be decreased by:

1. Increasing contact pressure.
2. Reducing surface roughness.
3. Maintaining surface flatness.
4. Utilizing shim materials with lower hardness than that of the joint material.

The problem of analyzing thermal resistance of joints consists of two steps: determining the true contact area through which heat flows by solid conduction, and identifying the modes of heat transfer, other than solid conduction, and their relative importance. True contact area is difficult to establish because it is made up of many small points of contact, which depend upon the surface roughness, contact pressure, contact material hardness, and surface flatness. The amount of deformation of the contact points establishes the true or effective contact area.

Reference 12 suggests gaseous, or molecular or other conduction through the interstitial fluid or filler and radiation as possible modes of heat transfer in addition to solid conduction through the contact area. The contribution of each mode to the total heat transfer has not been established (circa 1965).

In the transfer of heat by conduction, the thermal resistance across the mounting surfaces can decrease the overall heat transfer. This decrease in heat transfer becomes particularly important when considering cold plate mounted components where the small change in heat transfer can cause the junction temperature to operate 10 °F higher or more.

## 4.2.1.2 (Continued):

For cold plate mounted components, a spacer with high electrical resistance and low thermal resistance is required. Two new methods of decreasing joint resistance have been used:

1. RTV rubber between component, washer, and cold plate.
2. Special washer with high dielectric and low thermal resistance.

RTV rubber has been used successfully in at least one company in the aerospace industry. For example, a newly developed washer (spacer) material, which may be classified as a breakthrough in material development, has resulted. It consists of 30% (by weight) of 0.002 in dia. glass balls bonded into a single layer solid by Epoxy 907, which constitutes the remaining 70% finished material. This is sandwiched between two layers of 0.020 in thick dead-soft aluminum (110 SO) and then bonded to it.

As an alternate, a single layer of aluminum may be used; the washer would then be bonded to the electronic element during the packaging phase. The dielectric layer thickness of 2 mils is fixed by the glass ball diameter (state-of-the art of 1965); however, the aluminum thickness is limited only by physical considerations (see Figure 12).

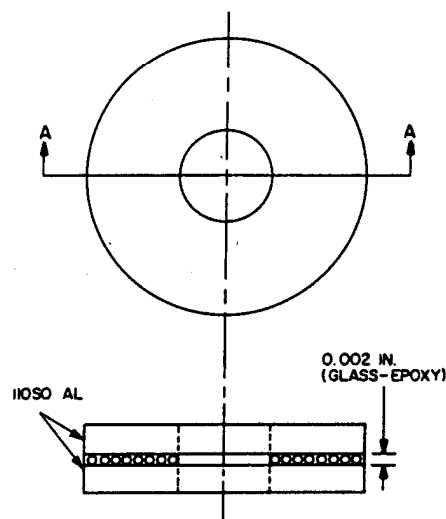


FIGURE 12 - Section A-A of an Electrical Separation Washer Using Glass Balls as Dielectric Material

Preliminary tests have been conducted on glass-epoxy and comparisons made with other materials (see Table 5) including beryllium oxide (ceramic). The latter is an excellent electrical dielectric and is also favorable for thermal conductivity. The major disadvantage of beryllium oxide, other than brittleness, is its extremely high cost.

TABLE 5 - Comparative Thermal and Electrical Resistances of Various Electrical Insulating Materials (Washers)

| Resistance             | Type of Washer<br>Mica (0.002 in) | Type of Washer<br>BeO | Type of Washer<br>Glass-Epoxy | Type of Washer<br>Anodized<br>Aluminum |
|------------------------|-----------------------------------|-----------------------|-------------------------------|----------------------------------------|
| Thermal, °F/W          | 2.61                              | 1.7                   | 1.5                           | 2.6                                    |
| Electrical,<br>megohms | $150 \times 10^3$                 | $0.250 \times 10^4$   | $150 \times 10^3$             | Range<br>(100-750) $\times 10^3$       |

## 4.2.1.2 (Continued):

Along with the above-mentioned glass-epoxy and BeO, the often discussed mica and anodized aluminum are also compared for electrical and thermal resistance. It should be emphasized that these are comparative values, due only to the nature of the preliminary tests, and are not to be used as absolute values for design purposes.

From the summary of the preliminary test data shown in Table 5, it can be seen that beryllium oxide is by far the best material, although, in addition to extremely high cost, it is quite brittle and requires a form of grease or smear gasket for maximum usefulness. Based on these test results, the glass-epoxy has great promise. As a complete sandwich (between two layers of aluminum), the dead-soft aluminum provides a smear gasket, with a good thermal bond between the aluminum and glass-epoxy joint. Used with a single sheet of aluminum, this epoxy joint is made direct to the heat source or heat sink.

More extensive tests of this nature are planned (circa 1965). This is expected to provide more data on joint resistance in general. Table 6 shows the results of a series of laboratory tests designed to indicate the magnitude of thermal resistance of the materials investigated as compared to mechanical joints with no electrical insulation.

TABLE 6 - Effect of Electrical Insulators on Thermal Resistance of Typical Component Mounting Joint (refer to Figure 13)

| Run No. | Washer<br>A  | Washer<br>B         | Temperature Gradient,<br>°F/W |
|---------|--------------|---------------------|-------------------------------|
| 1       | Bare Al      | (2) Mica (0.006 in) | 1.05                          |
| 2       | Bare Al      | None                | 0.94                          |
| 3       | Mica (0.006) | (2) Mica (0.006)    | 2.61                          |
| 4       | Anodized Al  | (2) Mica (0.006)    | 2.55                          |
| 5       | None         | (2) Mica (0.006)    | 1.17                          |

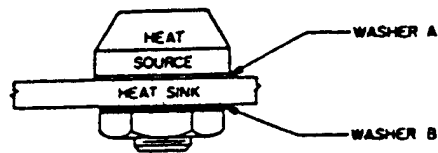


FIGURE 13 - Typical Component Mounting Joint

- 4.2.1.3 Thermal Resistance: There are many materials available which can be used to achieve good thermal resistance. Solids, solid particles, liquids, and gases are used in various combinations to provide the necessary insulation. The selection of the material is dependent upon, in addition to low thermal conductivity, whether structural rigidity is important. For example, insulation for cryogenic storage vessels must have not only high thermal resistance but also structural rigidity.

Solid insulating material, such as wood, stone, and plastics, are not considered here because there are other insulations that have not only much lower thermal conductivity but also lower densities. Table 7 presents a list of representative insulations that are commonly used and are readily available.

TABLE 7 - Effect of Air Removal Upon Thermal Conductivity of Various Materials at 100 °F Mean Temperature

| Material                  | Density,<br>lb/ft <sup>3</sup> | Thickness<br>in | Thermal<br>Conductivity, k<br>At 1<br>atm. | Reduced<br>Pressure<br>μ Hg | Thermal<br>Conductivity<br>At Reduced<br>Pressure<br>k | Percent<br>Reduction |
|---------------------------|--------------------------------|-----------------|--------------------------------------------|-----------------------------|--------------------------------------------------------|----------------------|
| Air Space-Copper Surfaces |                                | 1               | 0.572                                      | 0.07                        | 0.187                                                  | 67.3                 |
| Corkboard                 | 6.67                           | 1               | 0.283                                      | 0.06                        | 0.106                                                  | 62.5                 |
| Corkboard                 | 12.4                           | 1               | 0.314                                      | 0.13                        | 0.120                                                  | 61.8                 |
| Cotton                    | 0.78                           | 1               | 0.288                                      | 0.07                        | 0.072                                                  | 75.0                 |
| Cotton                    | 1.53                           | 1               | 0.257                                      | 0.12                        | 0.052                                                  | 79.8                 |
| Cotton                    | 2.62                           | 1               | 0.248                                      | 0.14                        | 0.033                                                  | 86.7                 |
| Coarse Glass Wool         | 2.8                            | 1               | 0.267                                      | 0.04                        | 0.055                                                  | 79.4                 |
| Fine Glass Wool A         | 0.58                           | 1               | 0.2505                                     | 0.08                        | 0.050                                                  | 80.0                 |
| Fine Glass Wool A         | 1.94                           | 1               | 0.2145                                     | 0.05                        | 0.0195                                                 | 90.0                 |
| Fine Glass Wool A         | 3.8                            | 1               | 0.211                                      | 0.11                        | 0.012                                                  | 94.3                 |
| Fine Glass Wool B         | 0.78                           | 1               | 0.302                                      | 0.04                        | 0.0665                                                 | 78.0                 |
| Hairfelt                  | 11.1                           | 3/4             | 0.260                                      | 0.10                        | 0.031                                                  | 88.1                 |
| Kapok                     | 0.26                           | 1               | 0.366                                      | 0.10                        | 0.091                                                  | 75.1                 |
| Kapok                     | 0.99                           | 1               | 0.2445                                     | 0.10                        | 0.0375                                                 | 84.7                 |
| Kapok                     | 3.96                           | 1               | 0.244                                      | 0.12                        | 0.0175                                                 | 92.8                 |
| Mineral Wool              | 7.7                            | 1               | 0.258                                      | 0.05                        | 0.037                                                  | 85.7                 |
| Silica Aerogel            | 8.6                            | 1               | 0.1715                                     | 0.07                        | 0.0255                                                 | 85.1                 |
| Special Specimen          | 3.1                            | 1               | 0.231                                      | 0.00                        | 0.0145                                                 | 93.7                 |
| Wood Fiberboard           | 15.5                           | 3/4             | 0.353                                      | 0.05                        | 0.075                                                  | 78.8                 |
| Wood Fiber                | 3.5                            | 1               | 0.2715                                     | 0.08                        | 0.052                                                  | 80.8                 |
| Wood Fiber                | 6.9                            | 1               | 0.277                                      | 0.09                        | 0.033                                                  | 88.1                 |

Source: References 15 and 42.

## 4.2.1.3 (Continued):

These insulating materials are nonhomogeneous and porous so that thermal conductivity is a function of composition, density, temperature, and (particularly) pressure. For materials of this type, heat transfer is effected not by conduction alone but also by convection and radiation. Reference 13 suggests that the term "apparent thermal conductivity" should be used for fibrous insulation rather than the term "thermal conductivity" and gives the following expression for the overall thermal conductivity:

$$k'_T = \frac{(k_{cd} + k'_{cv} + k'_{ra})}{1-f} + k_{sc} \quad (\text{Eq.2})$$

Table 2 of Reference 13 shows measured thermal conductivity values for various densities of fiber insulation with four gases (helium, air, carbon dioxide, and Freon 12) at atmospheric pressure, except for one test at 76 mm Hg pressure. The results indicate that at atmospheric pressure, gas conduction is by far the most important mode of heat transfer in fibrous insulation, contributing from 50 to 90% to the overall thermal conductivity. Also, at atmospheric pressure, helium gas gives the highest thermal conductivity and Freon 12 gives the lowest value.

Figure 14, which is reproduced from Figure 6 of Reference 13, shows the contribution by each of the different modes of heat transfer, at atmospheric pressure, for changes in insulation density. It is readily apparent that air conduction is the major mode of heat transfer and does not vary with change in insulation density. Radiation and convection decrease rather rapidly with increase in insulation density (up to about 3 lb/ft<sup>3</sup>) and further increase in density has less influence on these two modes of heat transfer. Solid conduction due to fiber contacts contributes very little to the heat transfer in the fibrous insulations.

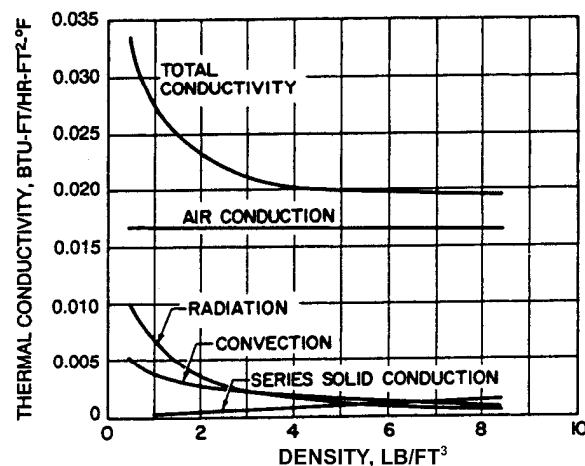


FIGURE 14 - Thermal Conductivity Versus Density.  
Contribution by Various Modes of Heat Transfer (for Fibrous Type Insulation)

## 4.2.1.3 (Continued):

The predominant mode of heat transfer (gas conduction) can be minimized or eliminated by evacuating the gas from the insulation. The marked effect of low pressure on decreasing the thermal conductivity is clearly indicated in Table 7. The variation of thermal conductivity with pressure for powdered insulation is given in Figure 15.

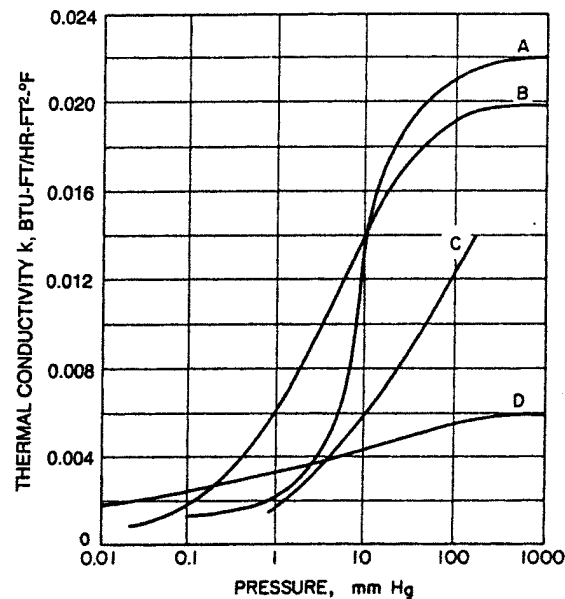


FIGURE 15 - Variation of Thermal Conductivity  $k$  With Pressure for Various Powdered Insulations of Temperatures Between 138 and 540 °R;  $T_{av} = 339$  °R; A = Diatomaceous Earth; B = Perlite; C = Lampblack; D = Santocell (Reference 15)

When gas conduction is eliminated by lowering the insulation pressure, radiation becomes the predominant mode of heat transfer. This mode of heat transfer can be minimized by the use of powdered or solid particle insulation, by fibrous materials, or by multiple layers of highly reflective surfaces. This latter method is discussed more fully in the part on radiation.

Highly efficient insulating materials with lower thermal conductivities than those listed in Table 7 have been developed through improvements in cryogenic insulation. Table 6 of Reference 14 lists a number of such materials. As indicated in Reference 14, the best forms of insulation are the laminated types of insulation, which have mean thermal conductivities on the order of 0.000025 Btu/h-ft-°F. However, they are not load bearing, so that an application such as insulating a cryogenic storage vessel requires additional structural members, which introduce the additional problem of heat leaks. Evacuated powders and fibers have been used extensively, particularly in small vessels. The thermal conductivities of these materials are approximately ten times as high as those of laminated materials.

#### 4.2.1.3 (Continued):

Christensen, Reference 15, draws the following conclusions with regard to some of the commonly used insulation materials.

1. Cellular foams are not the best insulation for space environments because the totally enclosed interstices cannot be evacuated by the vacuum of space.
2. Fiberglas mats may be the best conventional insulation for space application because they are vibration- and pressure-resistant and have conductivities as low as 0.001 Btu-ft/h-ft<sup>2</sup>-°F at low temperatures.
3. Fiberglas mats and powders must be used at the relatively high densities of 5 lb/ft<sup>3</sup> and more in order to reduce radiation transfer through the insulation.
4. Powders, while having the best thermal properties as insulators, have the disadvantages of being nonload bearing or vibration resistant, and are susceptible to effects of moisture.

These insulating materials are applicable primarily for extremely low temperatures and are not suitable for specific application such as for reentry bodies. The insulation requirements during reentry are low thermal conductivity at high temperatures for short duration. This insulation problem is discussed in Reference 19.

- 4.2.2 Radiation: Radiation is a very important means of heat transfer for temperature control of equipment. Surface temperatures, disposal of waste or excess heat, highly efficient insulation, all are dependent upon radiation or the control of radiation. Knowledge of thermal radiation and means for its control are vital for the design of temperature control systems. Reference 16 gives a comprehensive treatment on radiation heat transfer analysis. Reference 17 gives a detailed analysis for the design of efficient radiating surfaces. A more general discussion of thermal radiation is presented in many books on heat transfer, such as References 1, 2, and 18.

The selection of surface coatings for space vehicles is one of the important tasks to provide proper temperature control. Surface coatings must have appropriate values of solar absorptivity and emissivity, to keep the vehicle and equipment surface temperatures within required ranges. Table 8 illustrates the variety of materials and finishes found on the internal or external surfaces of Explorer VII (Reference 20).

TABLE 8 - Absorptance and Emittance of Materials Found in Explorer VII Satellite Vehicle

| Material                     | Short Wave Absorptance,<br>Solar Radiation | Long Wave<br>Emittance |
|------------------------------|--------------------------------------------|------------------------|
| Sandblasted Al               | 0.42                                       | 0.21                   |
| Polished Al                  | 0.31                                       | 0.07                   |
| Sandblasted Mg               | 0.63                                       | 0.54                   |
| Rokide A                     | 0.15                                       | 0.77                   |
| Gray TiO <sub>2</sub> Paint  | 0.87                                       | 0.87                   |
| White TiO <sub>2</sub> Paint | 0.19                                       | 0.94                   |
| Solar Cells                  | 0.80                                       | 0.40                   |
| Sauereisen                   | 0.34                                       | 0.88                   |
| Fused Silica                 | 0.06                                       | 0.84                   |
| Gold Foil                    | 0.21                                       | 0.07                   |

Source: Reference 20.

- 4.2.2.1 Equipment Surface Radiation: Consider a piece of heat generating equipment. It may be electronic or electromechanical. A fluid moving (hydraulic or pneumatic) equipment item is ruled out because it would fall in the semipassive class. If the equipment is considered to be uniform in temperature at any time, it can be treated thermally as a unit.

The parameters of interest in steady-state cooling of the equipment are:

1. Heat generated in the equipment,  $Q_{\text{gen}}$ . (In the steady-state condition, this is also the heat to be dissipated.)
2. Effective radiating surface area  $A$ .
3. Allowable equipment surface temperature  $T$ .

If transients are being looked at, the thermal capacity  $C$  is also of interest.

For an object cooled only by radiation, the heat balance is

$$C \left( \frac{dT}{d\tau} \right) = Q_{\text{gen}} - Q_{\text{net}} \quad (\text{Eq.3})$$

where:

$Q_{\text{gen}}$  = Rate of heat generation in equipment, watts

$Q_{\text{net}}$  = Rate of net heat loss by radiation from equipment, watts

## 4.2.2.1 (Continued):

If the equipment is reasonably small compared with an enclosing spherical satellite, the following equation can be used to a good approximation, provided the satellite inner surface is at a uniform temperature:

$$Q_{\text{net}} = \sigma A \epsilon T^4 \quad (\text{Eq.4})$$

If the satellite inner surface is nonuniform in temperature, the same equation may apply, with  $T^4$  replaced by  $\bar{T}^4$ , an effective radiation temperature, provided the equipment is near the center of the satellite or is (essentially) flat and wall mounted. In either case, the simple term  $T^4$  can describe the heat sink temperature; further, this temperature can be obtained from a gross vehicle heat balance of the type:

$$\frac{\pi D^2}{4} \epsilon_T \sigma \bar{T}^4 = \alpha_s S \frac{\pi D^2}{4} + \sum Q_{\text{gen}} \quad (\text{Eq.5})$$

Equation 5 can be rearranged as

$$\left( \frac{\epsilon_T}{\alpha_s + r} \right) \sigma \bar{T}^4 = S \quad (\text{Eq.6})$$

where:

$$r = \sum Q_{\text{gen}} / \left[ \left( \pi D^2 / 4 \right) S \right] \quad (\text{Eq.7})$$

Referring to Figure 16,  $\omega$  is defined as

$$\omega = \frac{\epsilon_T}{\alpha_s + r} \quad (\text{Eq.8})$$

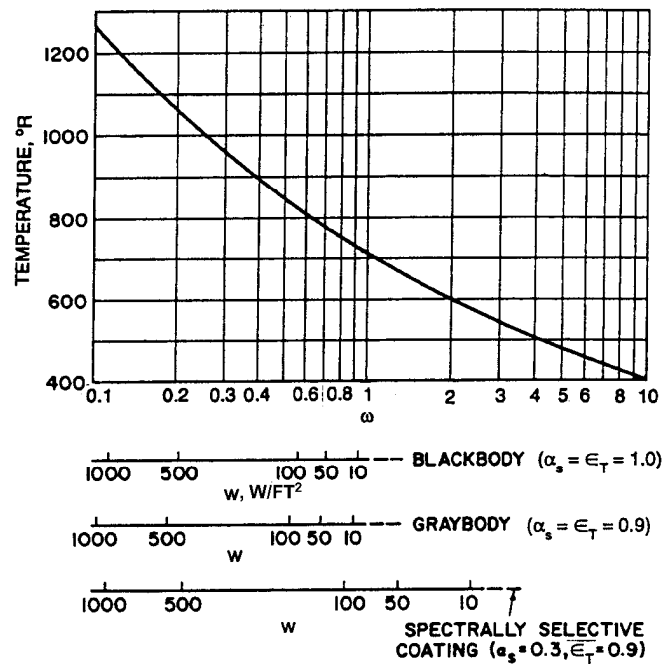


FIGURE 16 - Thermal Balance Solution for Flat Plate Normal to Sun's Rays at Top of Atmosphere

#### 4.2.2.1 (Continued):

If  $w$  in Figure 16 is defined as

$$w = \frac{\sum Q_{\text{gen}}}{\pi D^2 / 4} \text{ W/ft}^2 \quad (\text{Eq.9})$$

then Figure 16 can be used to give the sink temperature:

$$T_{\text{sink}} = (\bar{T}^4)^{1/4} \quad (\text{Eq.10})$$

$T_{\text{sink}}$  will vary from 400 to 600 °R for most Sun-illuminated satellites.  $T_{\text{sink}} = 400$  °R corresponds to  $\epsilon_T/\alpha_s = 10$  and  $w = 0$  W/ft<sup>2</sup>.  $T_{\text{sink}} = 600$  °R corresponds to  $\epsilon_T = 0.9$ ,  $\alpha_s = 0.3$ , and  $w = 20$  W/ft<sup>2</sup>.

Also in Figure 16,

$$r = \frac{w}{S} \quad (\text{Eq.11})$$

## 4.2.2.1 (Continued):

where  $w$  is the thermal input to the plate in  $\text{W/ft}^2$  in addition to solar radiation  $S$ , the value of which is

$$S = 130 \text{ W/ft}^2 \quad (\text{Eq.12})$$

Higher vehicle surface thermal loadings than these are possible, of course, for equipment compatible with the resulting temperature level. At  $w = 50 \text{ W/ft}^2$ ,  $\epsilon_T = 0.9$ , and  $\alpha_s = 0.3$ ,  $T_{\text{sink}} = 650^\circ\text{R}$ . These thermal loadings are per unit disc area of the spherical satellite; they should be divided by 4 to get the equivalent loading per unit of satellite surface area.

Figure 17 is a generalized plot of the basic radiation equation:

$$\frac{Q}{A} = \sigma(T^4 - T_{\text{sink}}^4) \quad (\text{Eq.13})$$

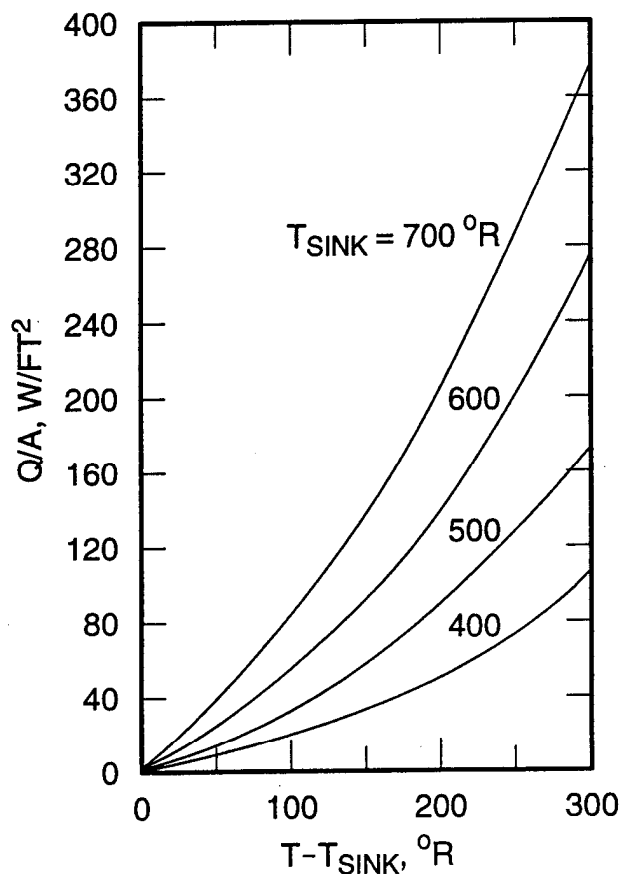


FIGURE 17 - Solution of Radiation Equation (Equation 13)

## 4.2.2.1 (Continued):

It may be used to show temperature levels associated with cooling by radiation alone, even under favorable conditions.

At reasonable sink temperatures, in the range of 600 °R, it takes a temperature rise of 110 °R to dissipate 60 W/ft<sup>2</sup> from equipment. The loading of 60 W/ft<sup>2</sup> is a rather light thermal loading, typical of electronic elements in a power limited installation. The conclusion is that radiation cooling is satisfactory for items of low to moderate thermal loading. For high thermal loadings, additional cooling means are desirable, to avoid excessive temperature rise. The exception to this would be single components of high thermal output, designed to be radiation cooled at fairly high temperature rises.

4.2.2.2 Thermal Radiation Shield: The heat transfer equations derived here are for a vehicle whose surface is shielded by  $n$  layers of protecting materials. It is assumed that a vacuum exists in the space between the body and the shield, as well as between the individual layers, and that no heat flows by conductivity between the elements under consideration. One further assumption is that the layers are flat, parallel, and infinitely large, so that the configuration factor is equal to 1.

Each of the several layers is assumed to have an emissivity  $\epsilon$  except for the outside surface of the last layer (the most external one), which has an emissivity  $\epsilon_m$ . The rate at which radiant energy leaves the body, whose temperature is  $T_0$  and emissivity is  $\epsilon_0$ , can be analyzed by use of Figure 18.

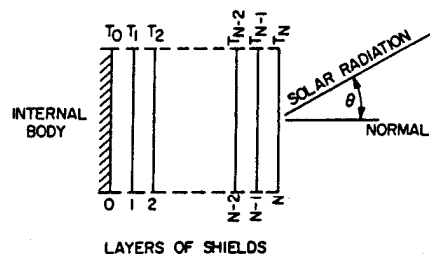


FIGURE 18 - Multilayer Shielding

Heat transfer between surface 0 and layer 1 is

$$\begin{aligned}
 q_{0,1} &= \frac{1}{(1/\epsilon_0) + (1/\epsilon) - 1} \sigma (T_0^4 - T_1^4) \\
 &= \frac{\epsilon_0 \epsilon}{\epsilon + \epsilon_0 - \epsilon_0 \epsilon} \sigma (T_0^4 - T_1^4)
 \end{aligned}
 \tag{Eq.14}$$

## 4.2.2.2 (Continued):

and the heat transfer between layers 1 and 2 is

$$\begin{aligned} q_{1,2} &= \frac{1}{(1/\epsilon) + (1/\epsilon) - 1} \sigma (T_1^4 - T_2^4) \\ &= \frac{\epsilon}{2 - \epsilon} \sigma (T_1^4 - T_2^4) \end{aligned} \quad (\text{Eq.15})$$

Similarly, it follows that heat transfer between layers  $n - 1$  and  $n$  is

$$q_{n-1,n} = \frac{\epsilon}{2 - \epsilon} \sigma (T_{n-1}^4 - T_n^4) \quad (\text{Eq.16})$$

The heat transfer equation between the  $n$ th layer and the environment is

$$q_n = \epsilon_n \sigma T_n^4 - S \alpha_s \cos \theta \quad (\text{Eq.17})$$

For steady-state condition, heat flow rate is the same through all the layers; thus, by equating Equations 15 and 16, the following relation is obtained:

$$T_1^4 - T_2^4 = T_{n-1}^4 - T_n^4 \quad (\text{Eq.18})$$

Equation 18 is typical for any of the layers and shows the change in the value of the fourth power of the temperatures between succeeding layers. For  $n$  layers, there will be  $n - 1$  changes in value between the first and the  $n$ th layer, or

$$T_1^4 - T_n^4 = (T_{n-1}^4 - T_n^4)(n-1) \quad (\text{Eq.19})$$

Now, equating Equations 14 and 16,

$$\frac{\epsilon_0 \epsilon}{\epsilon + \epsilon_0 - \epsilon_0 \epsilon} \sigma (T_0^4 - T_1^4) = \frac{\epsilon}{2 - \epsilon} \sigma (T_{n-1}^4 - T_n^4) \quad (\text{Eq.20})$$

It follows that

$$T_0^4 = T_1^4 + \frac{\epsilon + \epsilon_0 - \epsilon_0 \epsilon}{\epsilon_0 (2 - \epsilon)} (T_{n-1}^4 - T_n^4) \quad (\text{Eq.21})$$

Also, equating Equations 16 and 17 and simplifying,

$$T_{n-1}^4 - T_n^4 = \frac{\epsilon_n (2 - \epsilon)}{\epsilon} T_n^4 - \frac{2 - \epsilon}{\epsilon \sigma} S \alpha_s \cos \theta \quad (\text{Eq.22})$$

## 4.2.2.2 (Continued):

Combining Equations 19, 20, and 22,

$$T_0^4 = T_n^4 + \left[ \frac{\epsilon_n (2 - \epsilon)}{\epsilon} T_n^4 - \frac{2 - \epsilon}{\epsilon \sigma} S \alpha_s \cos \theta \right] \left[ \frac{\epsilon + \epsilon_0 - \epsilon_0 \epsilon}{\epsilon_0 (2 - \epsilon)} + (n - 1) \right] \quad (\text{Eq.23})$$

which may be expressed simply as

$$T_0^4 = T_n^4 + (C_a T_n^4 - C_c) C_b \quad (\text{Eq.24})$$

where:

$$C_a = \frac{\epsilon_n (2 - \epsilon)}{\epsilon} \quad (\text{Eq.25})$$

$$C_b = \frac{\epsilon + \epsilon_0 - \epsilon_0 \epsilon}{\epsilon_0 (2 - \epsilon)} + (n - 1) \quad (\text{Eq.26})$$

$$C_c = \frac{2 - \epsilon}{\epsilon \sigma} S \alpha_s \cos \theta \quad (\text{Eq.27})$$

Finally,

$$T_n^4 = \frac{T_0^4 + C_b C_c}{C_a C_b + 1} \quad (\text{Eq.28})$$

Substituting Equations 28 and 17, the general equation for the amount of heat radiated through a cover consisting of n layers as a function to  $T_0$  and angle  $\theta$  is obtained as

$$q_n = \epsilon_n \sigma \frac{T_0^4 + C_b C_c}{C_a C_b + 1} - S \alpha_s \cos \theta \quad (\text{Eq.29})$$

When in free space, if a side of the vehicle is not exposed to the Sun,  $\theta = 90^\circ$ ,  $\cos \theta = 0$  and  $C_c = 0$  (see Equation 27), therefore

$$q_n = \frac{\epsilon_n \sigma T_0^4}{C_a C_b + 1} \quad (\text{Eq.30})$$

## 4.2.2.2 (Continued):

Further, in the particular case where  $\epsilon_n = \epsilon$ , Equation 30 reduces to

$$q_n = K_{\epsilon, \epsilon_0} \sigma T_0^4 \quad (\text{Eq.31})$$

where:

$$K_{\epsilon, \epsilon_0} = \frac{\epsilon \epsilon_0}{\epsilon + \epsilon_0 (1 - \epsilon)} + \epsilon_0 [2(n-1) - (n-1)\epsilon] \quad (\text{Eq.32})$$

If  $\epsilon_0 = \epsilon$ , Equation 32 reduces to

$$K_{\epsilon, \epsilon_0} = \frac{\epsilon}{1 + n(2 - \epsilon)} \quad (\text{Eq.33})$$

The heat transfer coefficient  $K_{\epsilon, \epsilon_0}$  versus  $n$ , the number of shields, is plotted in Figure 19 for the case in which an enclosed body is in free space receiving no solar energy. The figure indicates a rapid decrease in the value of heat transfer coefficient as the number of shields increases from zero to 2. As  $n$  increases beyond 3, the reduction in the value for  $K_{\epsilon, \epsilon_0}$  becomes negligible.

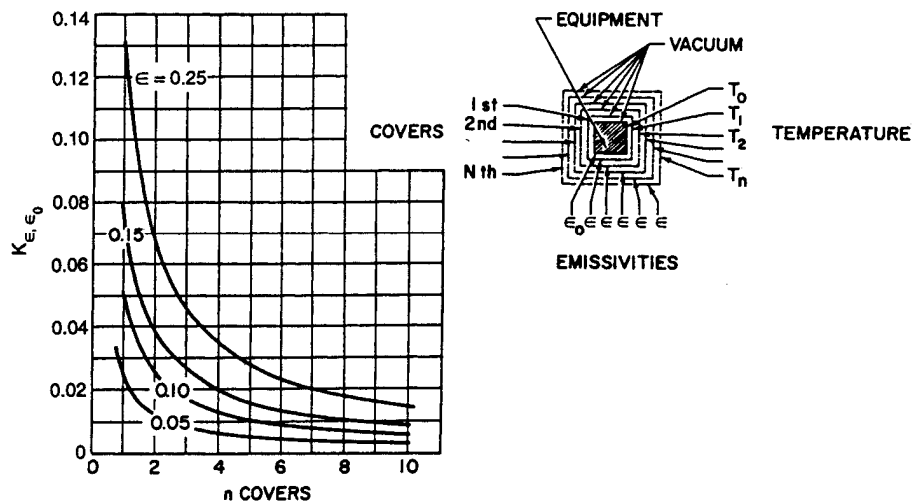


FIGURE 19 - Multishield Heat Transfer Coefficient (Box in Free Space and No Solar Heat);

$$Q = K_{\epsilon, \epsilon_0} \sigma T_0^4 \text{ Btu/h-ft}^2$$

A value of  $\epsilon_0 = 1.0$  Was Used, Since Only Small Deviations  
Result From Using Values as Low as 0.5

## 4.2.2.2 (Continued):

Also indicated by Figure 19 is that the coefficient  $K_{\epsilon, \epsilon_0}$  is reduced by decreasing the value for emissivity. By using three or four shields of low emissivity material, such as aluminum foil ( $\epsilon = 0.05$ ), very low values for the heat transfer coefficient can be obtained.

Figures 20 and 21 show plots of Equation 29 for  $n = 3$  and  $n = 4$ , for the case where the body is exposed to solar radiation for which the incident angle varies from 0 to 90 deg. Curves are plotted for several values of body temperature from 400 to 800 °R. For a given Sun angle  $\theta$ , the heat transfer rate increases almost exponentially with increase in body temperature. Comparison of Figures 20 and 21 indicates the reduction of heat transfer rate by increasing the number of shields from three to four, which is as expected.

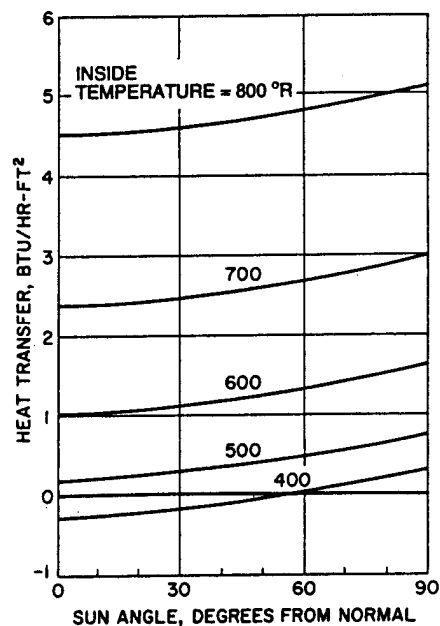


FIGURE 20 - Heat Transfer in Space Using Three Shields  
Assumed Conditions:  $S = 430 \text{ Btu/h-ft}^2$ ;  $\epsilon_0 = 0.80$ ;  $\epsilon = 0.05$ ;  $\epsilon_n = 0.95$ ;  $\alpha_s = 0.18$