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FINAL REPORT
SPACE SHUTTLE/ FOOD SYSTEM STUDY
VOLUME II

APPENDIX C
FOOD COOLING TECHNIQUES ANALYSIS

APPENDIX D
PACKAGE and STOWAGE – ALTERNATE CONCEPTS ANALYSIS

prepared for

NATIONAL AERONAUTICS and SPACE ADMINISTRATION
Johnson Spacecraft Center
Houston, Texas 77058

Contract NAS9-13138

Prepared by

Pillsbury

THE PILLSBURY CO.

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CONTENTS

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ILLUSTRATIONS

ILLUSTRATIONS CONT'D

PRELIMINARY RESULTS COOLING ANALYSIS
SPACE SHUTTLE FOOD SYSTEM STUDY

1.0 INTRODUCTION

This study considered the relative penalties associated with various techniques for providing an on-board cold environment for storage of perishable food items. The techniques were evaluated in terms of vehicle penalties of weight, volume and power, and were assessed for their capability to maintain both a 40-45°F refrigerated temperature and a 0°F and 20°F frozen environment temperature. Data are presented for the following freezer and refrigerator concepts:

a) Phase Change (Heat Sink) Concept

b) Thermoelectric Concept

c) Vapor Cycle Concept

d) Expendable Ammonia Concept

A sublimator concept was dropped from consideration and the expendable ammonia concept discarded after inputs were received from RI/SD that overboard venting and/or dumping would not be permitted.

In the studies presented, the following assumptions are implicit in the analyses.

a) The mission is a 6-man-7day mission.

b) Two freezer/refrigerator sizes have been generated by TPC based on the smallest and largest number of frozen and/or refrigerated items likely to appear on the menu. The small freezer/refrigerator internal dimensions are 14" x 9" x 10"(1260 in.^{3}) and the large freezer/refrigerator is 15" x 13" x 13" (2535 in.^{3}).

1.0 Cont'd

c) Freezer temperature to be 0°-5°F.

d) Refrigerator temperature to be 40°-45°F.

e) A liquid heat sink loop is available in the Shuttle for absorbing heat. Liquid temperature range 70-100°F; flow available 550 lbs/hr (pure H_2O); penalty 0.1 lb/$\frac{Btu}{hr}$.

f) Maximum cabin dew point temperature 61° with dry bulb from 65° to 80°F.

g) A negative penalty equivalent to the heat dissipated penalty can be applied for heat absorbed by the freezer and refrigerator.

h) System penalties include considerations of weight, heat loss to cabin (calculated as 0.133 lbs per average Btu/hr over a 24-hour period), and electrical energy consumed (1.5.4 lbs per Kw hr.)

i) Supplementary information pertaining to food data are shown in Table I.

A summary matrix of the study results is presented in Table 2, and ROM type cost estimates are shown in Table 3.

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TABLE 1. FROZEN FOOD DATA AND ASSUMPTIONS
(As Supplied From The Pillsbury Co.)

Food Item	Weight Each	Package Dimensions (Inches)	Small Number	Large Number
Sandwich	4 oz	$5 \times 5 \times 1\frac{1}{4}$	6	12
Entre	9 oz	$4 \times 4 \times 1\frac{1}{4}$	6	12
Ice Cream	4 oz	$2\frac{1}{2} \times 2\frac{1}{2} \times 1\frac{1}{4}$	6	6
Bakery	2 oz	$3 \times 3 \times 1\frac{1}{2}$	6	12
Bread (6 slices)	6 oz	$4\frac{1}{2} \times 5 \times 3\frac{1}{4}$	6	12
Butter (42 pats)	20 oz	$5 \times 5 \times 4\frac{1}{4}$ or $5 \times 2\frac{1}{2} \times 9$	1	2

TABLE 2. SUMMARY MATRIX - REFRIGERATION ANALYSIS

Technique	Temperature °F	Weight lbs		Volume ft^3		Power Watts
		Super-Insulation	Conventional Insulation	Super-Insulation	Conventional Insulation	
Phase Change Heat Sink	0	34.1(1)	109.4	4.7(1)	7.47	
	20	29.2(1)	76.7	4.14(1)	6.53	
	45	26.9(1)	72.0	3.75(1)	5.61	
(Cavity 15 x 13 x 13)	0	56.1(2)		3.27(2)		
	20	48.6(2)		2.98(2)		
	45	50.8(2)		2.53(2)		
Expendable Ammonia(9)						
(Cavity 15 x 13 x 13)	0	40.9		3.6		
(Cavity 14 x 10 x 9)	0	29.7		1.9		
Thermo-electric	0	34.2-47.6(3)	58.5-73.2(5)	2.31	3.91-5.87	12.5-26.5(4)
	20	31.5-44.9(3)		2.31		8.5-22.5(4)
(Cavity 15 x 13 x 13)	45	31.7-36(3)	43.5(6)	2.31	5.87(6)	4.2-9.2(4)
(Cavity 14 x 10 x 9) (7)	45	24.3-27.2(3)	33.3(6)	1.32	4(6)	2.9-6.3(7)
Vapor Cycle (8)	0	23.5(10)	29.7(10)	2.22(10)	6.01(10)	16
(Cavity 15 x 13 x 13)	20	20.0(10)	25.5(10)	2.05(10)	6.01(10)	12
	45	15.6(10)	20.2(10)	1.84(10)	6.01(10)	6.5

Notes: (1) Optimized weights and resultant volumes.

(2) Optimized volumes and resultant weights.

(3) Weight range based on heat rejection to cabin (high value) or to a liquid loop heat sink (low value).

(4) Power range based on heat rejection to cabin requiring a fan (high value) or to a liquid loop where fan is deleted (low value).

(5) Weight range based on calculating penalties only for 2.6" to 4" of insulation. Hardware and insulation weights were not calculated due to high penalties which make conventional insulation impractical.

(6) Weight and volume based on 4" foam insulation.

(7) Weight, Volume and Power are estimated based partially on ratio of surface areas of small to large size refrigerator.

(8) This is a high risk system due to O-g phase separation requirements.

(9) System not acceptable due to constraints on overboard venting.

(10) Weights and volumes exclude presently undeveloped 0-g phase separation hardware.

TABLE 3. COST ESTIMATES - REFRIGERATION SYSTEMS

Technique	R & D Cost	Production Cost (5 systems)	Total Program Cost
Phase Change (Heat Sink)	400K	250K	650K
Thermoelectric	525K	375K	900K
Vapor Cycle	875K	625K	1.5M

2.0 DISCUSSION

2.1 Introduction

The approach used to satisfactorily complete the NASA contract requirements for analyzing in-flight food refrigeration methods, was primarily a 3-step process as follows:

a) Research and review previous studies, data, reports, techniques and equipments that may be applicable.

b) Perform a preliminary screening of these data and select candidates that are Shuttle feasible, by means of preliminary analysis and in-review with North American.

c) Perform basic analysis of these candidates to determine performance characteristics of interest to the program (i.e., power, weight, volume, cost and temperature effects).

The following points must be recognized when reviewing the above documents.

- The intent of the analysis was to provide a comparative basis for assessing the various systems.
- A substantial number of techniques and conditions were analyzed with the most obvious and logical variations considered within the allocated effort. Undoubtedly, additional variables could be conceived which could indefinitely extend the scope of the study.

2.1 Cont'd

- A fixed time and effort expenditure was allocated for this task which is only one element of the total program. The total refrigeration analysis effort was, therefore, scoped in magnitude and depth to be consistent with the balance of program tasks. For this commitment of effort the level to which each analysis was carried produced comparable data and results.
- The results of the analysis are valid and correct, and have been based on certain thermal, food, and system assumptions. While the actual values presented for power, weight, volume, temperature effects, and cost may be subject to discussion due to the assumptions made, the relative ratings will not be substantially affected. By altering the assumptions, the final penalties can be recalculated.
- The technical competency of the analysis and the confidence level of the results provides a reasonable basis for selecting a particular technique and recommending such a technique for shuttle use.

2.2 Phase Change (Heat Sink) System

The phase change concept is based on the utilization of a material that changes phase and absorbs heat at a constant temperature. By using this material in the walls of a freezer or refrigerator, a desired compartment temperature can be maintained over a selected time period. Since the phase material is of high density, insulation is employed to optimize the amount of phase change material required over the mission length. In an actual situation, the phase change unit would be pre-conditioned to the desired temperature prior to vehicle installation. The analysis then considered an additional 24 hours elapsed time prior to launch, and a subsequent mission of 7 days. Weights and volumes of the freezer/refrigerators have been optimized for this time period, and the phase change material will have undergone a complete phase change from solid to liquid at a constant temperature. The design permits reuse of the system by re-freezing prior to the next mission.

The assumptions made in the analyses are as follows:

1) As an initial condition, the insulation temperature distribution is an equilibrium temperature distribution between the cabin environment and the phase change material.

2) A liquid zone exists adjacent to the freezer/refrigerator compartment wall due to heat leaks attendant on door openings.

3) The conductances at the insulation-liquid interface and inner compartment surface-liquid interface are small compared with the conductances between these surfaces and the solid portion of the phase change material. Consequently, the temperatures throughout the liquid zones remain constant at the phase change material temperature and the thermal capacitances of the liquid zones can be neglected.

2.2 Cont'd

4) At each opening of the freezer/refrigerator compartment door, a complete air change occurs with the air mass temperature assumed as an average of cabin and compartment temperature.

The results generated and summarized on Table 2, indicate that at all temperature ranges considered (0° to 45°F) for the large sized unit (15" x 13" x 13"), penalties for conventional insulation are too severe to be considered. Utilizing vacuum insulation produced more competitive results. It should be noted that optimized weights produce high volumes and when volume is optimized the weights increase.

A decision is required of RI/SD as to which criteria drives the design, weight or volume.

Since the unit is essentially a passive type system, no power is required to operate the refrigeration system.

2.3 <u>Expendable Ammonia Freezer</u>

The analysis for an expendable system was conducted for the freezer temperature of 0°F. However, in discussions with Rockwell International/Space Division, it was stated that overboard venting or dumping will not be permitted on the Shuttle, thereby negating the possibility of using any expendable system. No additional efforts were therefore expended on either the 20°F freezer or 40-45°F refrigerator utilizing this technique.

2.3 Cont'd

The analysis was based on a 42 man/day mission and sized in accordance with data received from the Pillsbury Company based on potential Shuttle menus. The freezer temperature range was taken as 0°F to 5°F, and a maximum of 12 air changes per day was assumed in the initial calculations. A single door opening was also evaluated to assess the impact of door openings versus penalties, with final results indicating that weight and volume are reduced by a small factor, approximately under 5% savings.

2.4 Thermoelectric Freezer

The thermoelectric freezer is based on the use of a commercial thermoelectric (T/E) module installed in a double walled honeycomb box, so that the T/E cold end is in the freezer cavity and the hot end terminates in an external heat sink. The analysis shows that a single T/E module (with redundancy provided in the event of module failure) will carry the entire load.

The freezer design was sized for a 15" x 13" x 13" cavity with a 1" honeycomb evacuated insulation around the cavity. The analysis was performed for both a 0°F and 20°F freezer and for an extreme of 2 or 12 door openings per day.

Approximately 5% savings in weight are obtained with the lower restriction in door openings due to reductions in the electrical and heat rejection penalties. The values presented in Table 2, therefore, are based on 2 door openings/day. The volume, which is independent of door openings, remains the same.

An analysis was performed to assess the impact of conventional insulation, rather than super-insulation. Utilizing the minimum thickness required to prevent condensation on the freezer walls, it was found that the electrical and heat rejection penalties alone were almost double the total system penalty for super-insulation. An attempt was made to lower these conduction loads by increasing the insulation thickness to a practical limit of 4". The penalties still exceeded the system weights of super-insulation by a substantial amount. The severe penalties of weight and volume in the use of conventional insulation makes it mandatory that only evacuated insulation be considered.

The relationship between the 0°F and 20°F freezer requirements are approximately 10% savings which is considered substantial for the weight critical Shuttle. Food data indicates that the 20°F freezer will be satisfactory to support the menu, therefore the weight and power savings should be taken advantage of with use of this design.

2.5 Thermoelectric Refrigerator

The thermoelectric refrigerator is similar in design to the freezer discussed in Section 2.4 using the same T/E module but rated at lower power. The analysis considered maintaining food temperatures between 40°-45°F and was based on two sizes of refrigerator cavity, 15" x 13" x 13" and 14" x 10" x 9". The basic analysis was performed for the larger refrigerator and heat rejection load penalties were scaled for the smaller size. Equipment weight and volume were calculated for both sizes.

A significant weight savings occurs, approximately 20%, when the refrigerator volume is reduced from 2.31 to 1.32 cubic feet. The smaller refrigerator may be attractive in that it offers advantages of on-board chilling and refrigeration, at minimum penalties. Again, evacuated insulation must be considered if the technique is to be competitive. An analysis of 4" foam insulation resulted in weights substantially higher than super-insulation and volumes 3 times larger.

2.6 Vapor Cycle System

A vapor cycle employing Freon-12 refrigerant was analyzed at three temperature ranges of 0°F, 20°F and 34°F. Penalty curves were generated at each temperature as a function of insulation thickness (super-insulation) and based on no air changes and one complete air change. The door opening penalty can therefore be determined by interpolating between the two curves, and it can be seen that the penalties are not critical or significant to the final results.

2.6 Cont'd

Although the values plotted in Table 2 do not show severe weight or volume penalties, the technique is not recommended due to the high development risk associated with the zero-gravity phase separation requirement at the condenser. Equipment does not presently exist to accomplish this in zero-gravity, consequently these weights and volumes cannot be estimated and are excluded in the values presented in Table 2.

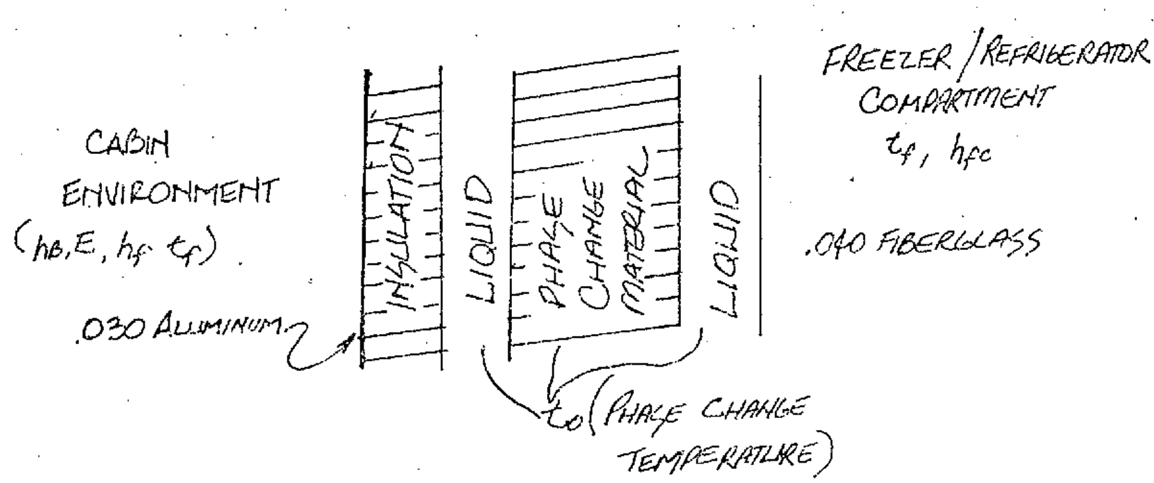
The penalties for 4" of conventional insulation are also shown in Table 2, and it can be seen that volume is unacceptable.

It is believed that this system would be competitive if a development program produced a simple, reliable and minimum cost (weight, volume, power) phase-separator.

3.0 DETAILED ANALYSIS

3.1 Phase Change Material

The concept employed here is the utilization in the walls of the freezer or refrigerator of a material that changes phase and absorbs heat at a constant temperature.



3.1 Cont'd

Insulation is employed to minimize the amount of phase change material, which changes phase at the desired freezer/refrigerator compartment, t_o. The liquid zone adjacent to the freezer/refrigerator compartment wall is the result of heat leaks attendant on door openings.

3.1.1 Assumptions

1) Assume as an initial condition that the insulation temperature distribution as an equilibrium temperature distribution between the cabin environment and the phase change material.

2) Assume that the temperatures throughout both liquid zones remain constant at t_o. This will be the case if the conductances at the insulation-liquid interface and at the freezer/refrigerator compartment surface-liquid interfaces are small compared with the conductances between these surfaces and the solid phase change material. As a consequence of this assumption, the thermal capacitances of the liquid zones can be neglected. In addition, since the temperature distribution through the insulation does not change, its thermal capacitance also can be neglected, as well as the thermal capacitances of the cabin side and freezer/refrigerator compartment side surface materials.

3.1.1 Cont'd

3) Assume that whenever the compartment door is opened, a complete air change of compartment air occurs. The air mass introduced is calculated on the basis of compartment free volume, cabin pressure, and a temperature that is the average of cabin and compartment temperature. The energy transferred to the compartment walls is taken to be that removed from the air change mass in cooling from cabin temperature to compartment temperature, t_o. The time to dissipate the heat leak is assumed to be proportional to the freezer/refrigerator compartment free volume fraction, N.

3.1.2 Thermal Analysis

$$U(t_f - t_c) + h_{fc}(t_f - t_c) = -\rho F_f,$$

WHERE $U = 1 \Big/ \left\{ \frac{1}{h_b + \epsilon h_f} + \frac{\bar{t}_i}{k_i} \right\}$

$h_{fc} = \frac{W C_p}{\Delta \tilde{T}_o A}$ ~ FREEZER/REFRIGERATOR SIDE CONNECTIVE COEFFICIENT

$W = \frac{P_{CABIN}}{R(HCO + .5(t_f + t_o)}$ V_f ~ FREEZER/REFRIGERATOR ARE CHANGE MASS

$\bar{T}$ ~ FREEZER/REFRIGERATOR TEMPERATURE ($\bar{\tau} = \tau_o$)

R ~ GAS CONSTANT

C_p ~ air specific heat at constant pressure

$\Delta\tau_D$ time to dissipate heat leak due to complete air change in an empty freezer/refrigerator compartment. (.25 Hr)

A ~ freezer inner surface area (A=7.76 Ft.^{2})

h_B ~ Cabin side convective coefficient (h_B=1.45 BTU/Hr.Ft.^{2}°F)

Eh_f ~ cabin side radiative heat transfer coefficient

(Eh_f=.20(1.05) .21 BTU(Hr.Ft.^{2}°F)

δ_i ~ insulation thickness

k_i ~ insulation thermal conductivity

t_f ~ cabin temperature (t_f=75°F)

t_e ~ phase change temperature

ρ ~ phase change material density

F ~ phase change material heat of fusion

δ ~ phase change material initial thickness

V_f ~ freezer volume (V_F = 1.47 Ft.^{3})

Define N_D ~ number of days per mission (N_D=7)

N_A ~ number of door openings per day

N ~ fraction of freezer volume not occupied by food

n_A ~ number of door openings to date

Food volume removed per day V_F/N_D

Food volume removed per door opening $V_F/N_D N_A$

$$N = 1 - \left(V_F - \frac{V_F}{N_D N_A} n_A\right) \Big/ V_F$$

$$N = \frac{n_A}{N_D N_A}$$

3.1.2 Cont'd

Time to dissipate air heat leaks is proportional to freezer/refrigerator free volume fraction,

Thus $\Delta \tau_{n_A} = \Delta \tau_D \dfrac{n_A}{N_A N_D}$

At the end of a one day hold plus elapsed mission time, τ'

$$\delta = 24 \frac{U}{\rho f}(t_f - t_o) + \frac{U}{\rho f}(t_f - t_o)\tau + (t_f - t_o)\int_0^{\tau'} h_{fc}\, d\tau$$

Since the integral equals a discreet number of terms

$$\delta = 24 \frac{U}{\rho f}(t_f - t_o) + \frac{U}{\rho f}(t_f - t_o)\tau + (t_f - t_o)\frac{h_{fc}}{\rho f} \sum \Delta \tau_{n_A}$$

The heat leak through the insulation must be accounted for up to the last door opening. Air change heat leaks need be considered up to the next to last door opening.

Thus:

$$\delta = 24 \frac{U}{\rho f}(t_f - t_o) + \left(24 N_D - \frac{24}{N_A + 1}\right)\frac{U}{\rho f}(t_f - t_o) + \frac{h_{fc}}{\rho f}(t_f - t_o)\sum^{N_D N_A - 1} \Delta \tau_D \frac{n_A}{N_D N_A}$$

THE SUM OF THE FIRST $N_D N_A - 1$ INTEGERS IS GIVEN BY $N_D N_A \left(\frac{N_D N_A - 1}{2}\right)$

$$\delta = \left\{ 24\left(N_D + 1 - \frac{1}{N_A + 1}\right)\frac{U}{\rho f} + \frac{N_D N_A - 1}{2}\,\frac{h_{fc}}{\rho f}\,\Delta \tau_D \right\}(t_f - t_o)$$

3.1.3 <u>Weight & Volume Analysis</u>

The system weight is given by

$$W_c = \rho\left\{(S_1 + 2\delta)(S_2 + 2\delta)^2 - S_1 S_2^2\right\} + \rho_i\left\{(S_1 + 2\delta + 2\delta_i)(S_2 + 2\delta + 2\delta_i)^2 - (S_1 + 2\delta)(S_2 + S_{\delta}\ldots\right.$$

$$+ W_{fg} + \rho_A \delta_A \left\{4(S_1 + 2\delta + \delta_i)(S_2 + 2\delta + 2\delta_i) + 2(S_2 + 2\delta + 2\delta_i)^2\right\},$$

where ρ ~ density of phase change material

δ ~ thickness of phase change material

ρ_i ~ insulation density

δ_i ~ insulation thickness

W_{fg} weight of fiberglass compartment side surface

$$W_{fg} = \rho_{fg}\,\delta_{fg}\left\{H L_1 L_2 + 2L_2^2\right\}$$

δ_g ~ fiberglass density $\rho_{fg} = 110$ PCF

3.1.3 Cont'd

δ_{fg} ~ fiberglass thickness $\delta_{fg} = .040$ IN

ρ_A ~ aluminum density $\rho_A = 173$ PCF

δ_A ~ aluminum surface thickness $\delta_A = .030$ IN

$$S_1 = L_1 + 2\delta_{fg}$$
$$S_2 = L_2 + 2\delta_{fg}$$

For the large freezer/refrigerator configurations,

L_1 =15.0 in., and L_2=13.0 in.

The system volume is given by

$$V = (S_1 + 2\delta + 2\delta_i + 2\delta_A)(S_2 + 2\delta + 2\delta_i + 2\delta_A)^2$$

3.1.4 Phase Change Material Properties

The feasibility of maintaining freezer/refrigerator temperatures by means of phase change materials was investigated utilizing Trans Temp. phase change materials, which are commercially available preparations developed to maintain temperatures within shipping containers for long periods of time.

The pertinent properties of the materials are:

t_o	ρ	F
0°F	66.2 PCF	117 Btu/lb
20	63.3	114
45	94.8	73.0

3.1.5 Insulation

Two insulation systems were investigated: a super-insulation and a conventional fiberglass insulation. The properties of each are as follows:

hinde SI-12 evacuated to 10 microns mercury abs.

$$\rho_i = 3.0 \text{ PCF}$$

$$k_i = .37 \times 10^{-3} \text{ Btu-ft/hr.ft.}^{2\circ}\text{F}$$

Johns-Manville Microlite AA

$$\rho_i = .6 \text{ PCF}$$

$$k_i = .02083 \text{ Btu-ft/hr.ft.}^{2\circ}\text{F}$$

The thermal conductivity of the hinde insulation was increased by an order of magnitude ($k = .37 \times 10^{-2}$) as an allowance for heat leaks through structural attachments between the freezer/refrigerator inner and outer surfaces. The thermal conductivity of Microlite was not increased since it was assumed that attachments could be designed having approximately the same conductance as the insulation.

3.1.6 Compartment Sizing

The freezer/refrigerator compartment size utilized was that given by the Pillsbury Co. based on the greatest number of frozen/refrigerated items likely to appear on the menu for a 6 man/7 day mission: 15" x 13" x 13". The compartment inner surface was assumed to be fabricated from 0.040 gage fiberglass (ρ = 110PCF) and the freezer/refrigerator outer surface from 0.030 gage aluminum.

3.1.7 Material Pre-Conditioning

According to the manufacturer of Trans Temp materials, the preparations must be solidified by conditioning at the appropriate temperature for 16 hours. It is assumed that this is done outside the vehicle and that the freezer/refrigerator is installed 24 hours before launch.

3.1.8 Results

The results of the analyses optimizing hinde SI-12 super-insulation are given in Figures 1, 2, and 3 for a 0°F and 20°F freezer, and a 45°F refrigerator, respectively.

3.1.8.1 Weight Optimized

If the freezer/refrigerators are optimized on a weight basis:

TABLE 4 -- Weight Optimized Phase Change System

T_o	N_A	δ_i	δ	W_T	V	ECS PEN.
0°F	3	2.80 in.	.358 in.	35.5 lb	4.70 ft^3	-1.37 lb
	6	2.85	.368	36.0	4.78	-1.39
	12	2.90	.378	36.7	4.87	-1.42
20	3	2.45	.290	30.2	4.14	-1.02
	6	2.50	.295	30.6	4.21	-1.03
	12	2.60	.300	31.0	4.35	-1.03
45	3	2.25	.180	27.5	3.75	- .60
	6	2.45	.183	28.0	4.00	- .60
	12	2.65	.188	28.3	4.30	- .61

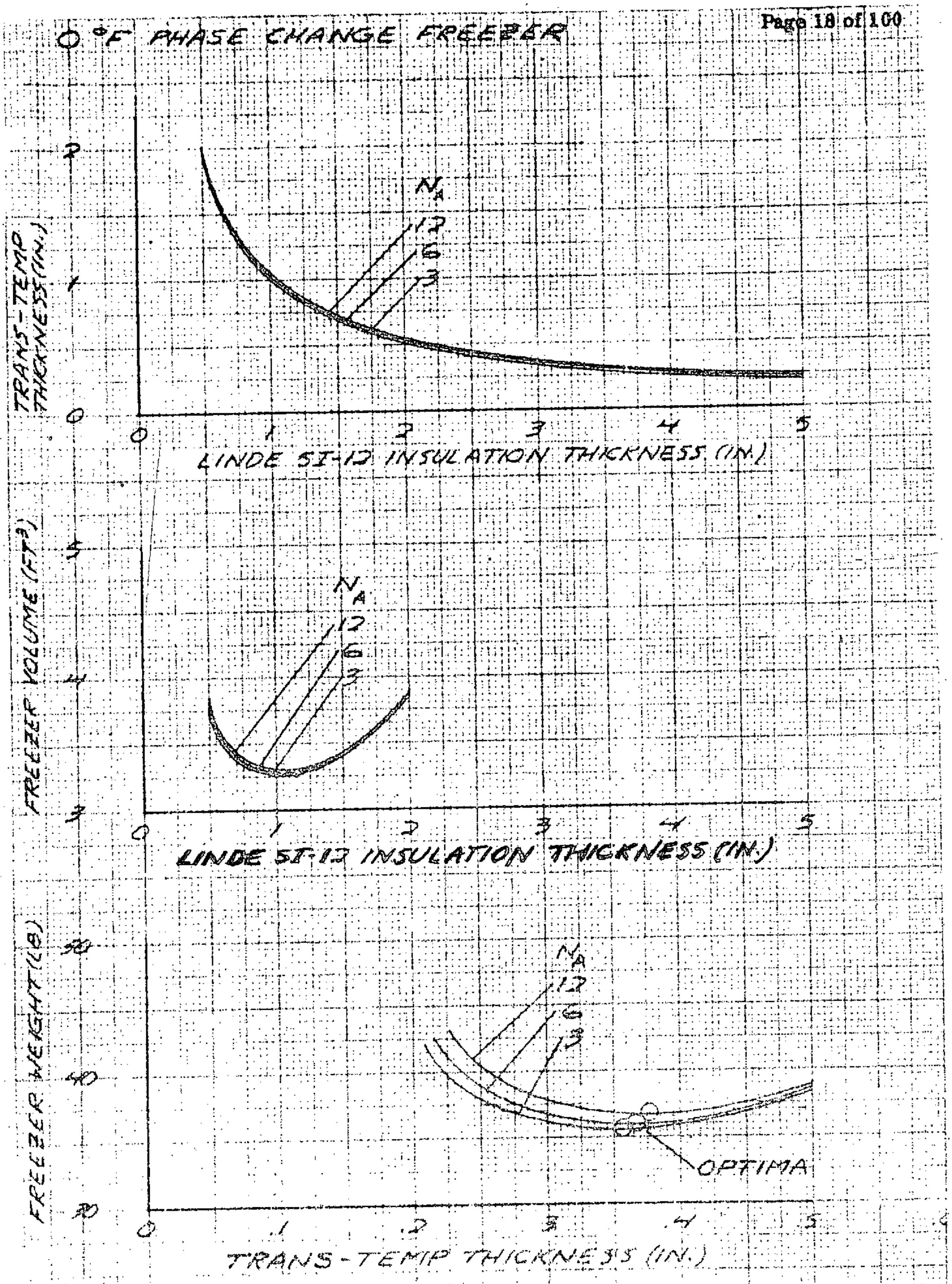


FIG. 1 - 0° PHASE CHANGE FREEZER - WT & VOL.

20 °F PHASE CHANGE FREEZER

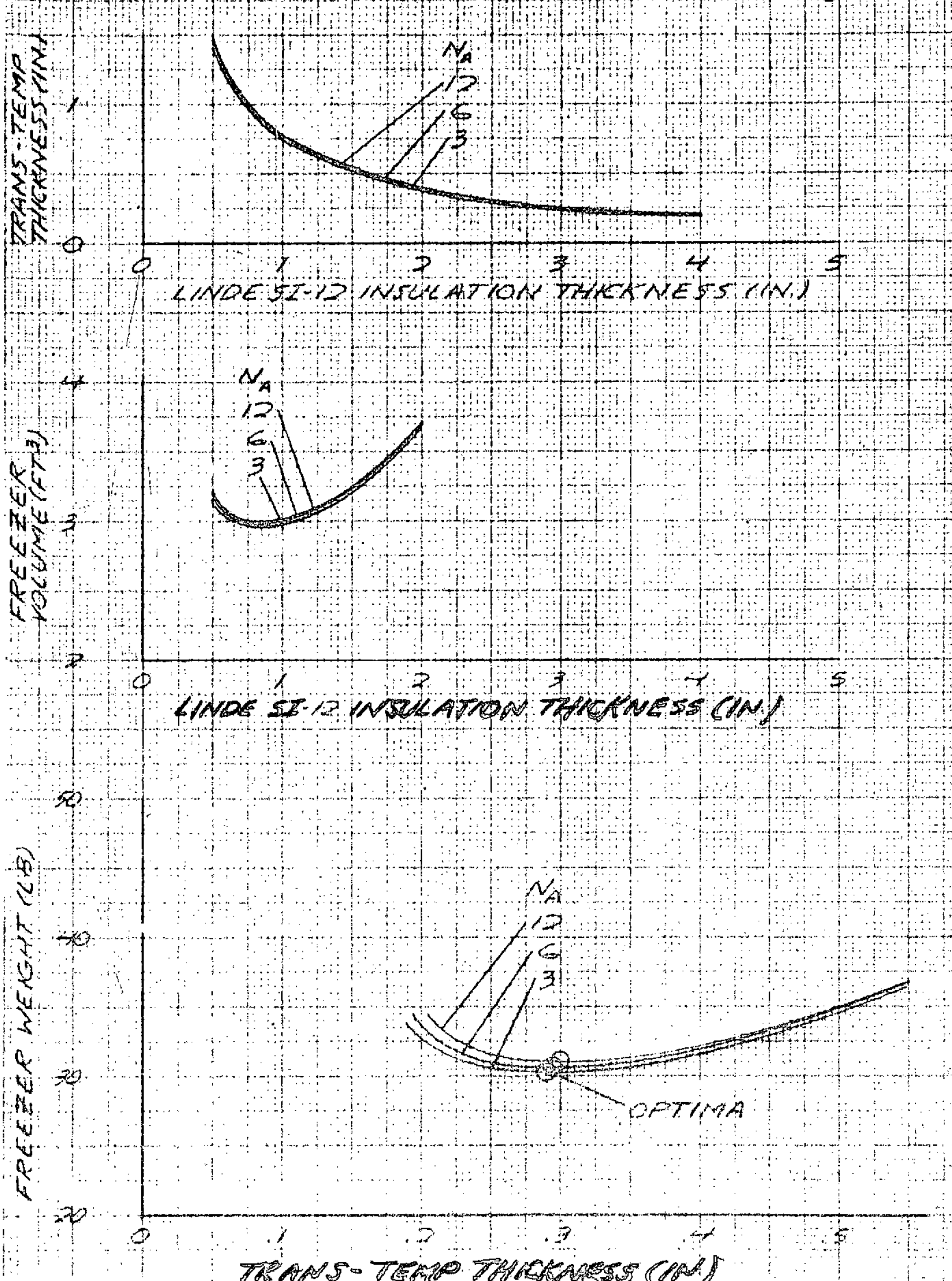


FIG 2 - 20° PHASE CHANGE FREEZER - WT & VOL.

45 °F PHASE CHANGE REFRIGERATOR

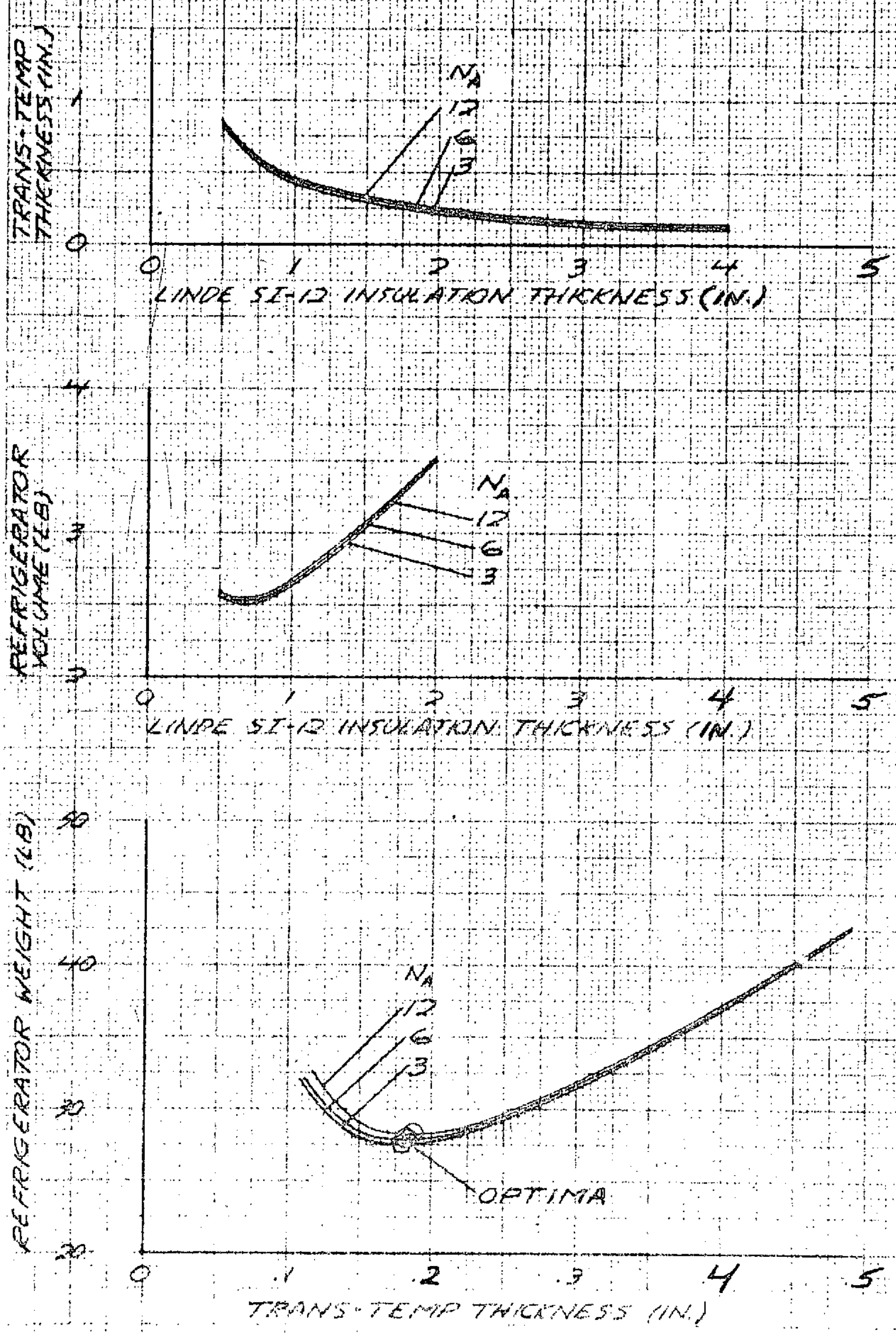


FIG 3 - 45°F PHASE CHANGE REFRIGERATOR - WT & VOL

3.1.8.2 ECS Penalty

The last column, denoted "ECS Penalty", is the penalty imposed on the shuttle environmental control system due to heat absorption by the freezer/refrigerator. This penalty is given by:

ECS Penalty = $\left[\rho\{(S_1+2\delta)(S_2+2\delta)^2 - S_1 S_2^2\}F/\{24(ND+1-\frac{1}{N_A+1})\}\right](-.133)$

The quantity within the square brackets represents an average heat transfer rate to the phase change material. The quantity, 0.133 Btu/lb is the ECS penalty by North American.

The total equivalent weight penalty to the shuttle is the sum of W_t and the ECS penalty.

3.1.8.3 Volume Optimized

If the freezer/refrigerators are optimized on a volume basis:

TABLE 5 - Volume Optimized Phase Change System

t_o	t_i	δ	W_t	V	ECS PENALTY
0°F	1.08 in.	.970 In.	60.0 Lb.	3.27 ft.^{3}	-3.95
20	.85	.880	51.9	2.98	-3.30
45	.67	.640	53.0	2.53	-2.22

Figures 1, 2, and 3 show that optimization as a volume basis occurs at virtually the same insulation thickness for all door openings studied, and that optimum volumes are almost identical for all door openings. The ECS penalty, however, is based on N_A = 12.

3.1.8.4 Conventional Insulation Penalty

For comparison purposes, results were generated utilizing Johns Manville Microlite AA insulation, which is a conventional, unevacuated, glass fiber material. Only one insulation thickness (3.0 in.) was studied since it was felt that the thickness represents an approximate practical maximum. Weight optima would occur at approximately an 8.0 inch thickness. The results for δ_i = 3.0 are:

TABLE 6 - Weight & Volume-Conventional Insulation

t_o	N_A	δ_i	δ	W_T	V	ECS PENALTY
0°F	3	3.0 in.	1.84	118.0lb	7.47 ft^3	-8.59
20	3	3.0	1.35	82.2	6.53	-5.53
45	3	3.0	.769	74.8	5.61	-2.78

Results were not generated for the smaller 9" x 10" x 14" freezer/refrigerator configuration since it is felt that the analyses of the larger units provide sufficient data for a relative assessment of the food storage concepts studied.

3.1.9 Summary

The results of the cooling analysis are summarized in Table 7.

A weight summary is given in Figure 4 for the various ranges of freezer/refrigerator temperatures considered. The range of weight differential at optimized weight vs. optimized volume indicates the necessity for a trade decision on the governing vehicle parameter. This input must be supplied by North American. A similar situation exists with the volume as shown in Figure 5.

TABLE 7 - SUMMARY - PHASE CHANGE MATERIAL COOLING ANALYSIS *

UNIT TEMP °F	DOOR OPENINGS PER DAY	OPTIMIZED WT. (#)	VOL. (FT^3)	WT. (#)	OPTIMIZED VOL. (FT^3)	TOTAL SYSTEM PENALTY (#) FOR OPT. WT.	OPT. VOL.
0	3	35.5	4.70	60.0	3.27	34.13	56.05
	12	36.7	4.87	60.0	3.27	35.28	56.05
20	3	30.2	4.14	51.9	2.98	29.18	48.6
	12	31.0	4.35	51.9	2.98	29.97	48.6
45	3	27.5	3.75	53.0	2.53	26.9	50.78
	12	28.3	4.30	53.0	2.53	27.69	50.78

* DATA PRESENTED IS BASED ON SUPER-INSULATION

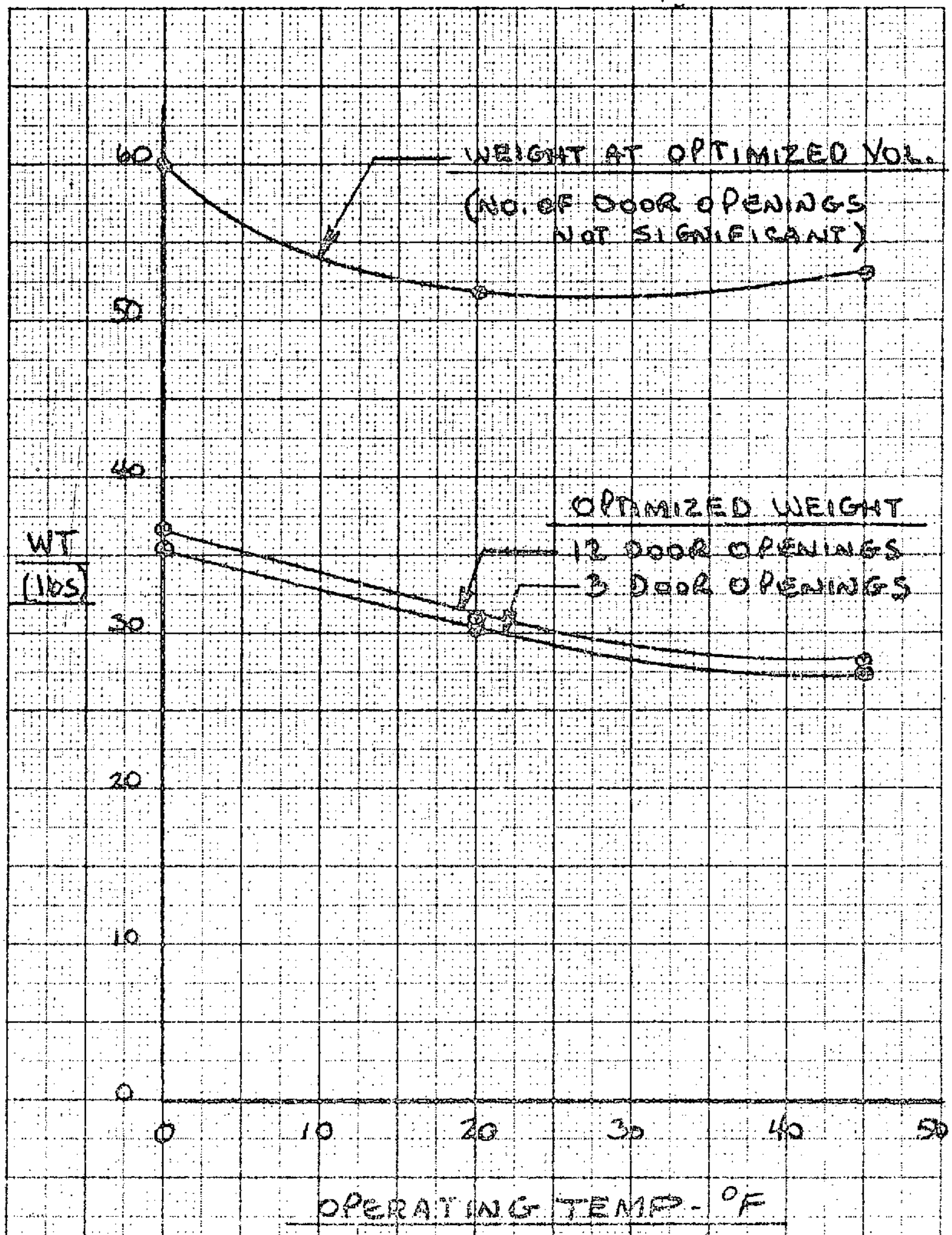


FIG 4 - WEIGHT SUMMARY

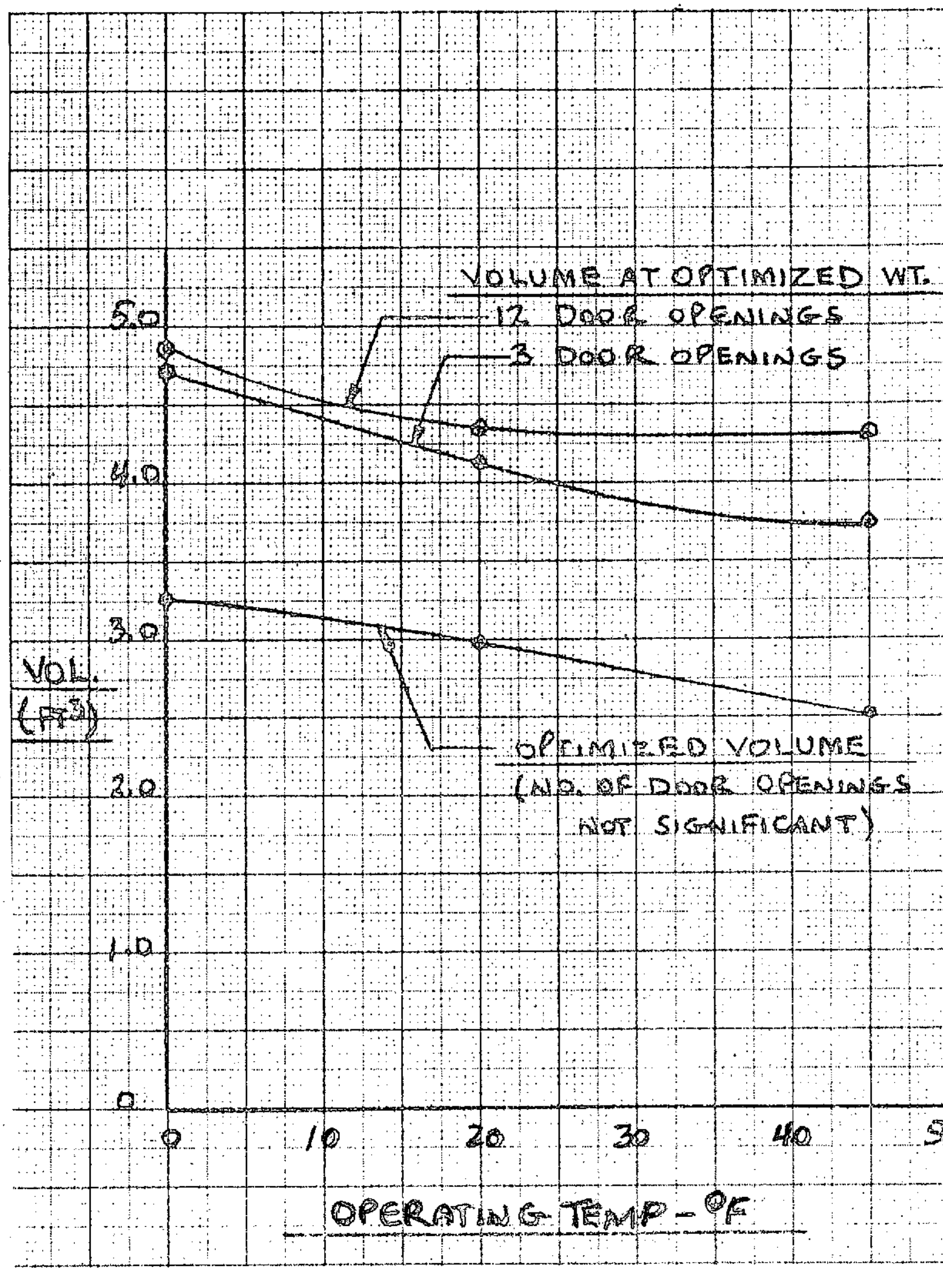
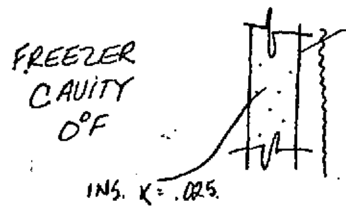


FIG 5 - VOLUME SUMMARY

3.2 Expendable Ammonia Freezer

3.2.1 Insulation Required to Keep Freezer Cabinet Surface Above Dewpoint:



T_{SKIN} MUST BE >61°F = MAXIMUM CABIN DEW. PT.

$h = 1.45 \ BTU/_{HR\text{-}ft^2}$

THERMAL RESISTANCE OF INSULATION & FILM $= \frac{1}{h} + \frac{t}{K}$

$$= \frac{1}{1.45} + \frac{t}{.25} = .69 + 4t$$

$$\frac{.69}{.69 + 4t} = \frac{T_{CABIN} - t_{SKIN}}{T_{CABIN} - 0°F}$$

SETTING T_{SKIN} = 61°F, CAN RELATE REQUIRED insulation thickness to cabin temperature. The results are plotted in Figure 6.

It is seen that to avoid surface condensation with a minimum cabin temperature of 65°F requires a minimum insulation thickness ($K = 0.25 \frac{\text{Btu-in}}{\text{hr-ft}^2\text{-°F}}$) of 2.63 inches.

Figure 7 presents the freezer (15" x 13" x 13" cavity size) conduction heat load as a function of insulation. For the required minimum thickness of 2.63", the conduction load is about 72 Btu/hr with the load value rapidly flattening out as thickness is increased.

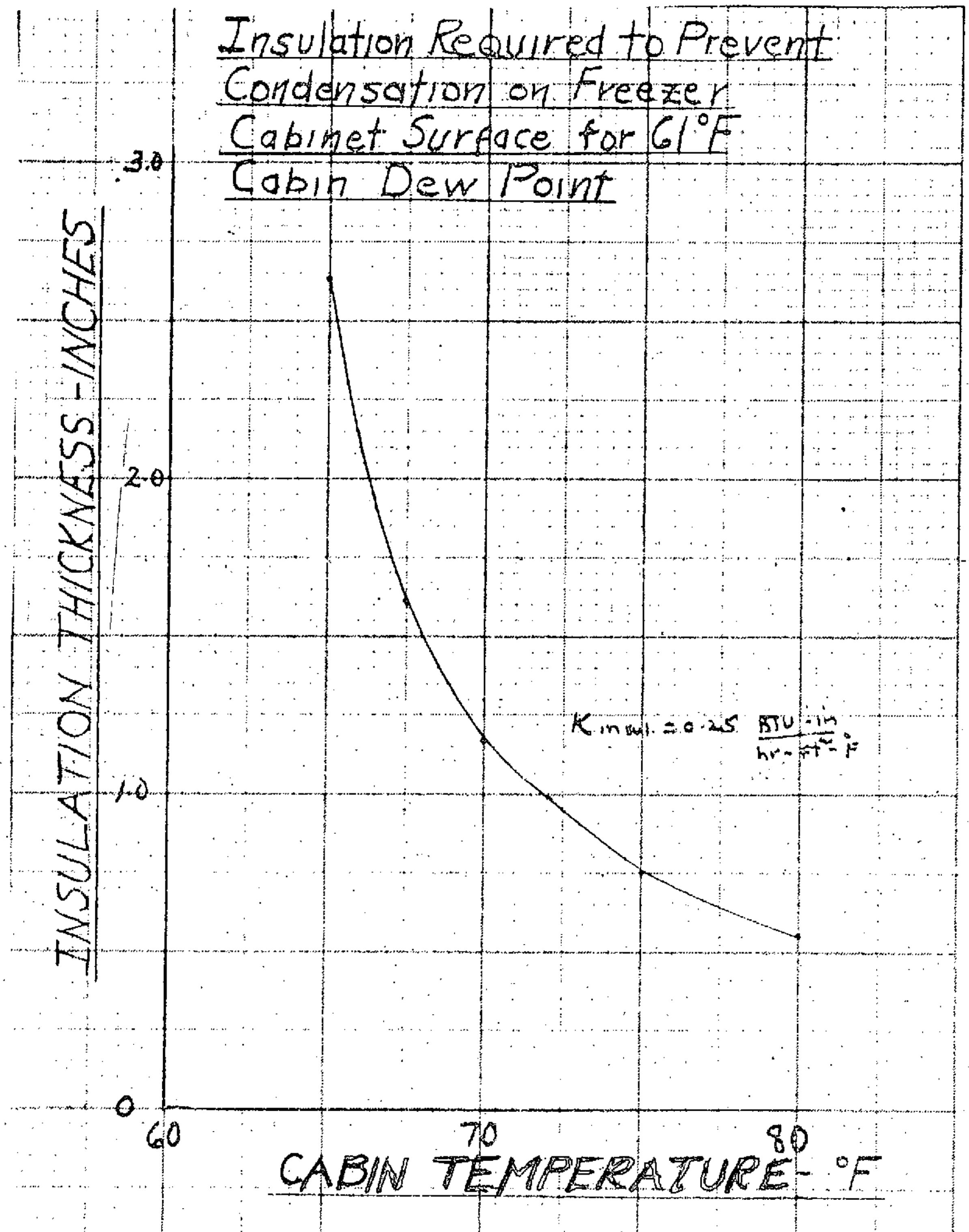


FIG. 6 - INSULATION REQUIREMENTS

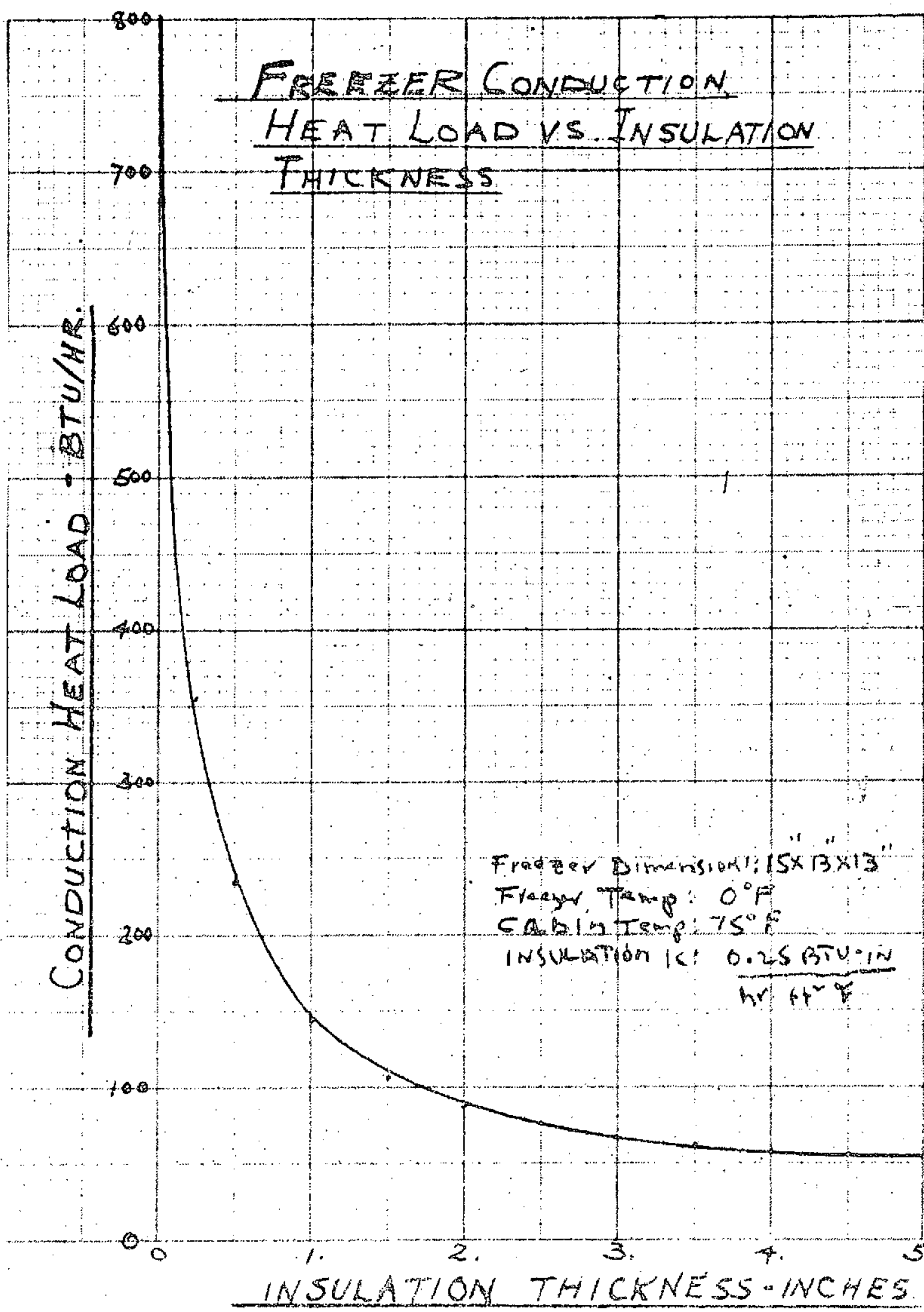


FIG. 7 - LOAD VS. THICKNESS

ORIGINAL PAGE IS OF POOR QUALITY

3.2.1 Cont'd

Considering an expendable evaporant such as ammonia which is a subcooled liquid at moderate pressures can be throttled to 0°F at 30 psia, a latent heat of approximately 500 Btu/lb is available. For the conduction load alone, the theoretical quantity of 72 Btu/hr x 24 hrs/day x 7 days x 1 lb NH_3/500Btu = 24 $\frac{\text{lbs}}{NH_3}$

Allowing for evaporation inefficiency and reserve liquid, it is seen that the NH_3 weight becomes excessive. Accordingly, multiple-layer vacuum insulation is extremely attractive and will be used for the freezer designs.

3.2.2 Design Guidelines

- 6 Man/7 day mission
- Freezer cavity dimensions - 15" x 13" x 13" (in accordance with data from the Pillsbury Co. based on Shuttle menu plans)
- Temperature range - 0°F - 5°F
- 12 air changes in freezer per day
- Cabin Temperature - 75°F

A design concept of an expendable ammonia freezer is shown in Figure 8. The subsequent analysis is based on this design.

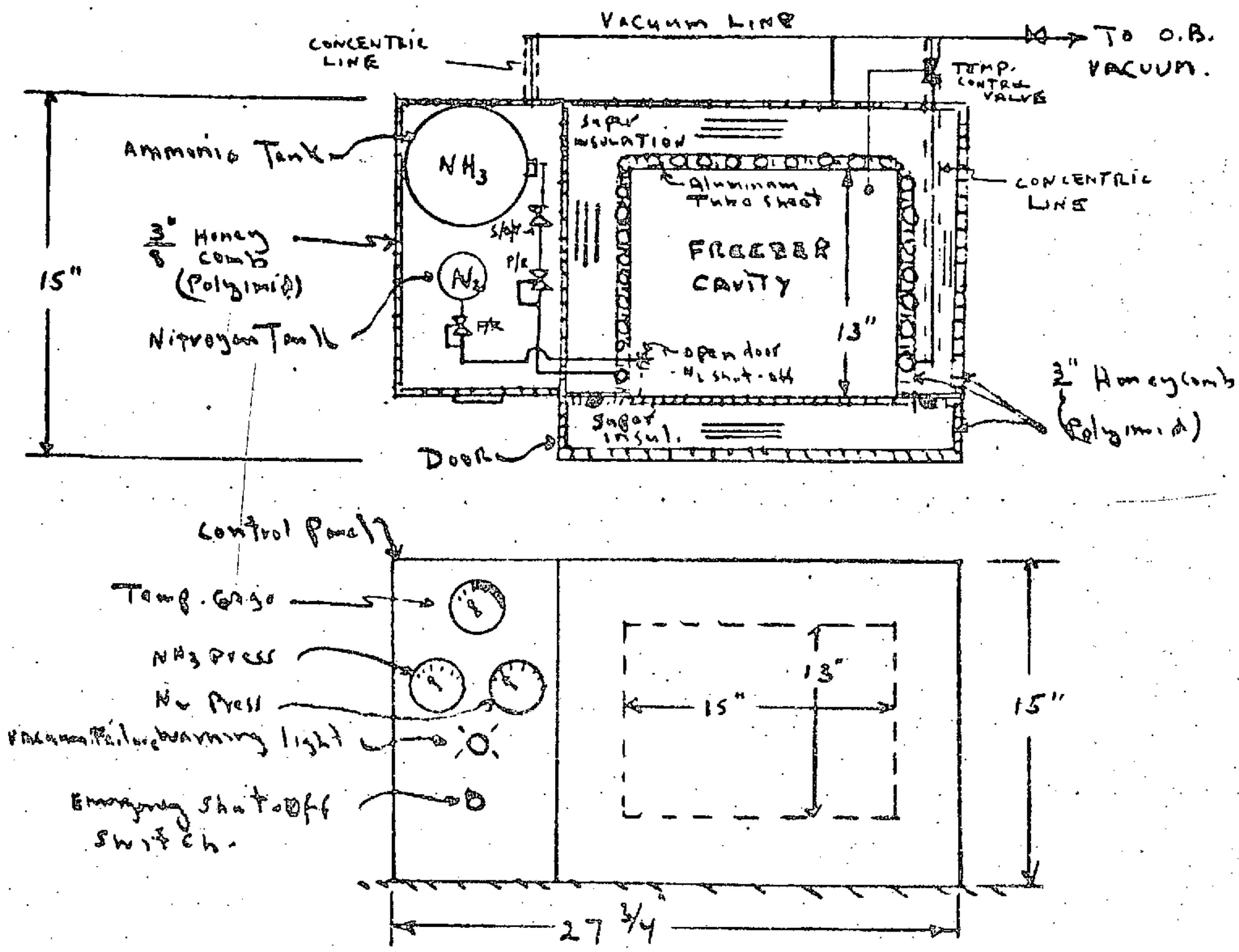
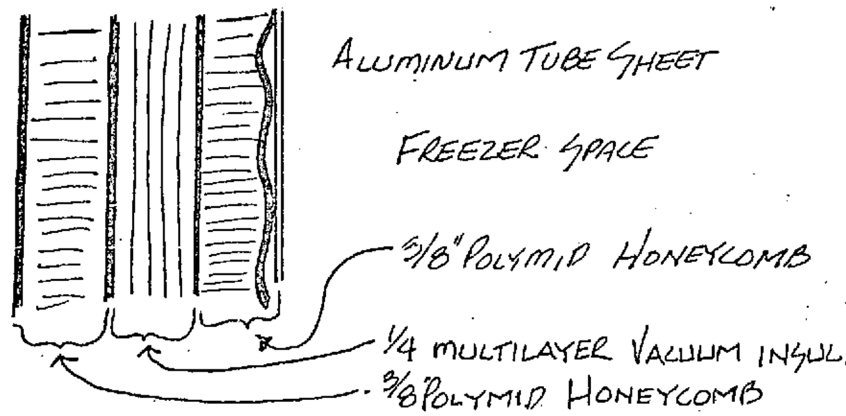


FIG. 8 - DESIGN SCHEMATIC CONCEPT

3.2.3 Wall Construction

The freezer wall construction is:



- Freezer Door Construction same as above except for absence of aluminum tube sheet.
- Evacuated enclosure containing ammonia and nitrogen bottles surrounded by simple wall of single sheet of 3/8" polyimid honey comb.

3.2.4 System Description

Vacuum insulation is connected to space vacuum line as is the NH_3 storage space. All NH_3 lines and fittings, outside the evacuated storage space are enclosed within concentric tubing to intercept any leaks.

NH_3 stored at 500-600 psia. Pressure regulator throttles the fluid to 20-30 psia (0°F to -15°F). The cold gas flows through the tube sheets to absorb the heat load. The ammonia temperature is monitored at the discharge end of the tube run to regulate a flow control valve which maintains the design temperature.

When the freezer door is closed, a high pressure N_2 bottle lined to the freezer cavity maintains a slightly positive pressure therein to counteract the effect of trapped ambient temperature in the cavity decreasing in pressure below ambient as its temperature is lowered making door reopening difficult. This pressure was arbitrarily assumed at approximately 3 inches of water differential. When the freezer door is opened, the N_2 is cut off.

A control panel is available with gages indicating freezer temperature, remaining NH_3 and N_2. A warning light is illuminated if the vacuum in the insulation is lost. This signal will permit the crew to select a preponderance of freezer foods for the meals immediately following a failure and thus consume them before spoilage occurs. An ammonia storage bottle shut-off valve is available to cut off flow if, despite all systems precautions, the presence of ammonia is sensed in the cabin.

3.2.5 Cooling Load Calculation: $\rho = 7\#/FT^3$

Use SI-4, type vacuum insulations

$K - 0.025 \times 10^{-3}$ Btu/hr-ft-F — Honeycomb panels structurally self sustaining under vacuum load eliminating supportive structure heat leaks.

Evacuated gap between honeycomb structure = 0.25"

Assume ammonia temperature @ -10°F

$\therefore$ DT = 75° - (-10) = 85°F

BOX SURFACE AREA.

$$\dot{Q}_{LENS} - \frac{0.025 \times 10^{-3}}{0.25\ IN} \frac{BTU}{H^{\prime}\text{-}FT\text{-}^{\circ}F} \times 12 \frac{IN}{FT} \times \frac{1118}{144} FT^2 \times 85 = .78 \frac{BTU}{HR}$$

Allow 200% increment for heat leaks through the insulation:

$2 \times 0.78 = 1.56$

Total Conduction = 0.78 + 1.56 = 2.34 Btu/hr

3.2.6 Air Change Load

Assume 12 door openings in 24 hours

$$\text{Cavity volume} = \frac{15 \times 13 \times 13}{1728} = 1.47 \text{ ft}^3$$

Take air at 0°F, = 460°R, then

$$\text{air} = \frac{14.7 \times 144 \times 1.47}{53.3 \times 46°} = 0.127\text{\# air/door opening}$$

12 openings x 0.127 = 1.52# air

Assuming air enters at 80°F

$$\dot{Q}_{AIR} = 1.52 \times 0.24 \times (80-0) = 29.2\text{Btu}/24\text{ hrs.} = 1.22\text{ Btu/hr.}$$

with contingency, assume $\dot{Q}_{AIR} = 2$Btu/hr.

$$\dot{Q}_{TOTAL} = \dot{Q}_{COND} + \dot{Q}_{AIR}$$

$= 2.34 + 2 = 4.34$ Btu/hr.

Total load in 7 day mission:

$$4.34\ \frac{\text{Btu}}{\text{hr}} \times 24\ \frac{\text{hrs}}{\text{day}} \times 7 \text{ days} = 735 \text{ BTU}$$

h_1, Enthalpy of subcooled NH_3 liquid at 75°F storage temperature - 125Btu/# (See Fig. 9)

h_2, enthalpy of saturated NH_3 vapor at 0°F = 614 Btu/

∴ Heat absorbed per pound NH_3 vaporized =

$h_2 - h_1 = 614-125 = 489$ Btu/#

Theoretical Total weight g NH_3 required -

$$\frac{735\text{Btu}}{489\text{ Btu/\#}} = 1.5\text{\#}$$

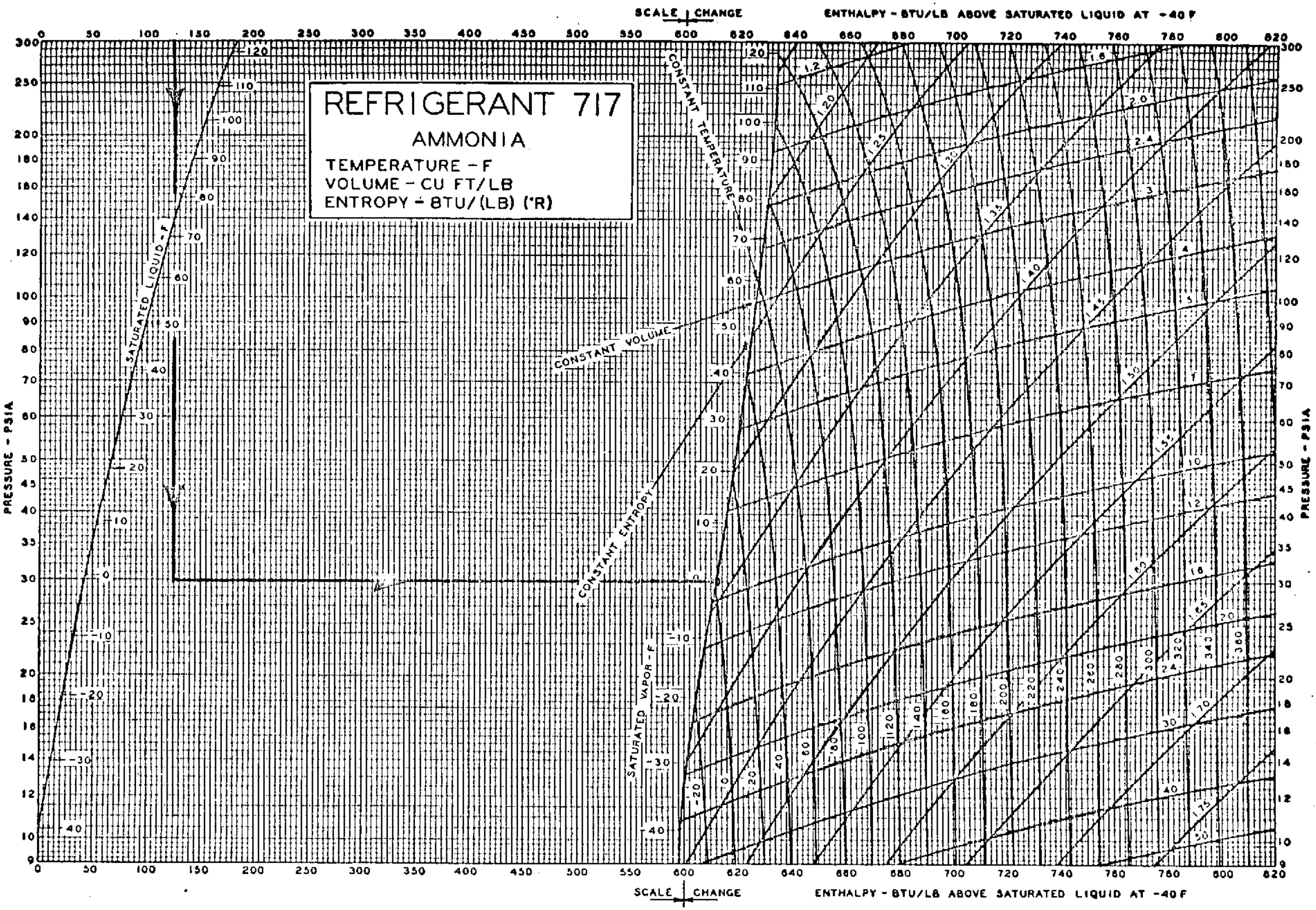


Fig. 9 Pressure-Enthalpy Diagram for Refrigerant 717

3.2.6 Cont'd

Assume 50% evaporation efficiency

$\frac{1.5}{0.5} = 3\#$ required

Allow 100% extra NH_3 for contingencies

$2 \times 3 = 6\#$ NH_3 design requirement

NH_3 liquid density - $40.6\#/ft^3$

$\therefore$ Volume NH_3 liquid required $= \frac{6}{40.6}\ ft^3 \times \frac{1728\ in^3}{ft^3} = 255\ in^3$

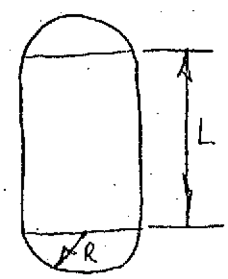
Assume 33% of NH_3 occupied by pressurizing gas

(e.g. N_2 acting on a separating diaphragm)

NH_3 tank volume required $255/\frac{2}{3} = \underline{382\ in^3}$

3.2.7 Storage Tank Design

Assume tank shape as cylinder capped with hemispherical ends:

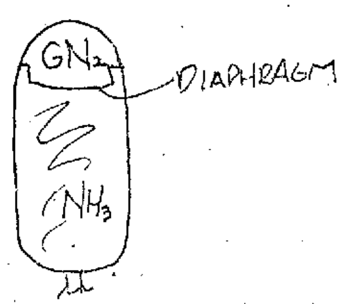


$R = 4''$

$VOL = \frac{4}{3}\pi(4)^3 + \pi r^2 L = 382$

$L = 2.77''$

Tank pressure requirements should assure that at the maximum cabin temperature, the stored NH_3 should remain liquid until completely empty.



$\therefore$ When the gas expands to the full tank volume as the liquid is drained the final pressure should be greater than the saturation pressure of NH_3 at the maximum cabin temperature, say 90°F.

NH_3 saturation pressure corresponding to 90°F=180 psia $\therefore$

Initial gas pressure must be $> 3 \times 180 = 540$ psia

(assuming isothermal expansion, which is good assumption since expansion occurs very slowly)

This pressure level is so low that tank wall thickness is determined by loadings other than stress induced by pressure.

Assume wall thickness (aluminum tank) = 0.03"

3.2.7.1 Tank Weight

Take tank wall as 0.03" thick

Volume of tank wall material

$[4\pi(4)^2 + 2\pi \times 4 \times 2.77] \times 0.03$

$V = \underline{8.1 \, IN^3}$ ALUMINUM WEIGHS $0.1 \, \#/IN^3$

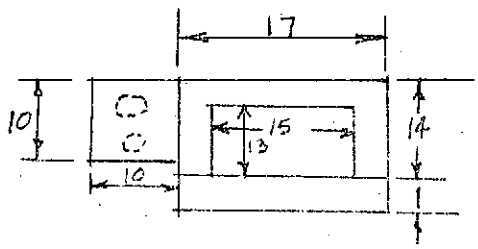
SHELL WT. $= 0.1 \times 8.1 = .81$

Allow 100% increment for fittings, connectors, lines, mounts.

$\therefore$ Tank weight penalty = $2 \times 0.81 =$ $\underline{1.6\#}$

3.2.8 Freezer Weight

3.2.8.1 Honeycomb Shell



Cavity outer honeycomb area = 17 x 14 (3) + 15 x 14 (2) = 1134

Cavity inner honeycomb area = 15 x 13 (3) + 13 x 13 (2) = 923

Box - Door contact area = 1 x 15 (2) = 30

Σ_1 = 2087

Door Faces 17 x 15 (2) 510

Door Edges 17 x 1 (2) + 15 x 1 (2) 64

Σ_2 574

Area-Gas Tank Storage enclosure =
10 x 15 (2) + 14 x 15 (1) + 10 x 14(2)
= Σ_3 = 790

$\therefore$ Total honeycomb area = $\Sigma_1 + \Sigma_2 + \Sigma_3$ = $3451\ in^2$

$$\text{Area} = \frac{3451}{144} = 24.0\ ft^2$$

Use polyimid honeycomb panels capable of taking 14.7psi differential @ wt = $0.75\#/ft^2$

$\therefore$ Honeycomb Shell wt. = 24 x 0.75 = 18#.

3.2.8.2 Vacuum Insulation

t = 0.25" thick

Area = 17 x 14(2) + 17 x 13(1) + 13 x 13(2) = $1035\ in^2$

$$\text{Vol.} = \frac{1035 \times 0.25}{1728} = 0.15\ ft^3$$

Weight = $0.15\ ft^3 \times 7\#/ft^3$ = 1.05#

Note: More detailed design could result in optimizing insulation, NH_3 and tank wts. to minimum penalties.

3.2.8.3 Nitrogen Cavity Pressurization System

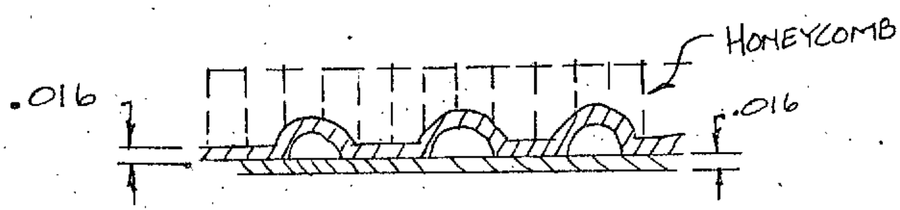
Assume system maintains 0.1 psi positive pressurization differential when door is closed. Gas is turned off when door opens.

$$\text{Cavity vol.} = \frac{15 \times 13 \times 13}{144} = \underline{1.46 \text{ ft}^3}$$

With partial pressure of N_2 lost in 12 air changes per day for 7 days = $W = \dfrac{0.1 \times 144 \times 1.46}{55.5 \times 460} \times 12 \times 7 = 0.($

$\therefore$ Weight of bottle + lines + misc = 1.5#

3.2.8.4 Aluminum Cavity Liner with Tubes

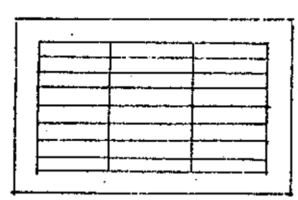


Aluminum Vol. = [15 x 13 x 3 + 13 x 13 x 2].03

= 27.69 in^3

Aluminum Weight = 27.69 x 0.1 = $\underline{2.8\#}$

3.2.8.5 Aluminum Shelves & Separators in Cavity



Gross Area = [13 x 13 x 2[+ [15 x 13 x 6] = 1508 in^2

Aluminum Vol. = 1508 x .03 thick = 45.2 in^3

If solid - wt. = 45.2 x 0.1 = 4.5#

but assume grilling, holes, perforations, etc.

$\therefore$ Wt = $\underline{2.5\#}$

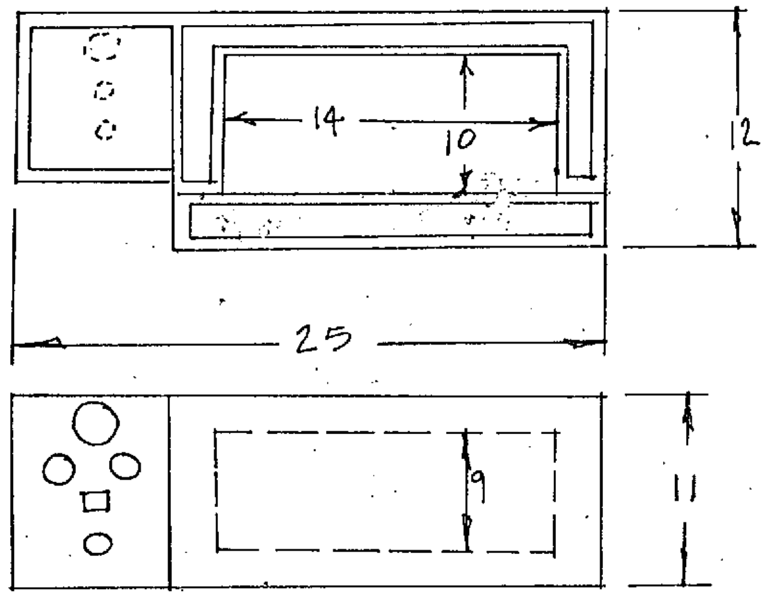
3.2.9 Ammonia Freezer (15" x 13" x 13" Cavity) Weight Summary

Honeycomb	18. lbs.
Vacuum Insulation	1.
Aluminum Cavity Liner + Tube	2.8
Aluminum Shelves & Separators	2.5
NH_3	6.0
NH_3 Tank	1.6
Valves	4.0
Gages	1.5
NH_3 Line Jackets	0.5
N_2 System, tank, gas, fittings	1.5
Switch, light, links, mounts, misc.	1.5
Total System Wgt. (incl. NH_3)	= 40.9 lbs.

3.2.10 Volume Summary

$$\text{Volume: } \frac{27.75 \times 15 \times 15}{1728} = 3.6 \text{ ft}^3$$

3.2.11 Small Ammonia Freezer (14" x 10" x 9" Cavity)



Honeycomb Panel Wt = 2330 in^2 (large freezer 3451 in^2)

Heat Transfer Area = 712 in^2 (large freezer - 118 in^2)

$$\text{Conduction Load Ratio} = \frac{\text{Small freezer}}{\text{Large freezer}} = \frac{712}{1118}$$

$$\text{Air Change Load} = \frac{\text{Vol. Small Freezer}}{\text{Vol. Large Freezer}} = \frac{1}{2}$$

$$\therefore \text{Small Load} = \frac{712}{1118} \times 2.34 \frac{\text{BTU}}{\text{hr}} + \frac{1}{2} \times \frac{2\text{Btu}}{\text{hr}} = 2.5 \frac{\text{Btu}}{\text{hr}}$$

$\therefore$ Load Dependent quantities ratio is

$$\frac{2.5}{4.34} \quad \frac{\text{Small load}}{\text{Large load}}$$

Rationing above gives following for small freezer:

3.2.11 Cont'd

Ammonia Freezer (14" x 10" x 9") Weight Summary

	12. Lbs
Honeycomb	0.7
Vacuum Insulation	1.9
Aluminum Cavity Liner and Tube	1.7
Aluminum Shelves & Separators	3.5
NH_3	0.9
NH_3 Tank	4.0
Valves	1.5
Gages	0.5
NH_3 Line Jackets	1.5
N_2 System	1.5
Switches, light, links, mounts, misc.	29.7 Lbs.

$$\text{Volume} = \frac{25 \times 12 \times 11}{1728} = \underline{1.9 \text{ ft}^3}$$

3.3 Thermoelectric Freezer

3.3.1 Description

The preliminary design of the thermoelectric freezer will be based on a double wall polyimid honeycomb with multiple layer vacuum insulation. A commercial thermoelectric module will be installed so that the cold end will be in intimate contact with the lauminum wall liner of the freezer cavity and the hot end will terminate in an air cooled heat exchanger heat sink mounted external to the freezer.

Since the structural configuration is similar to that of the expendable heat sink type previously analyzed in Section 3.2, the thermal load will be the same, 4.34 Btu/hr.

The initial design approach considered the use of an active thermoelectric (T/E) module in each of the freezer walls, except the door. The initial calculations indicated, however, that this approach split the load so that each module was carrying a very small Btu/hr. loading, resulting in very low efficiency. The proposed design is therefore based on the use of a single module carrying the entire load (with redundancy provided in the event of module failure) and acting as the condenser end of a series of heat pipes integrated into the aluminum wall liner which pick up the load evenly over each inner wall of the freezer.

3.3.1 Cont'd

Alternative means would be to:

a) Incorporate fins on the outer surface of the aluminum liner radiating from the cold element of the T/E module; or

b) Employ a sufficiently thick liner to minimize temperature gradients throughout, or

c) Use an additional active module on a second wall of the freezer in conjunction with a) or b) above. A simplified analysis shows that for alternative b), the gradient along one of the sides of the liner with a centrally located module mounted on a 2" diameter boss, is very small (0.22°F for a 0.03" thick liner wall), and that the overall differential from a remote point on the liner of the box to a centralized module is only slightly above 5°F for an ordinary aluminum 0.03" thick liner.

3.3.2 Temperature Variation with Centrally Located T/E Module

3.3.2.1 Assumptions

a) 13 x 13" Wall

b) Module mounted on 2" dia. boss

c) Module is 1.17 x 1.17

d) Intimate contact and no radial gradient in boss.

3.3.2.2 Calculations

The T/E module is mounted as shown in Figure 10.

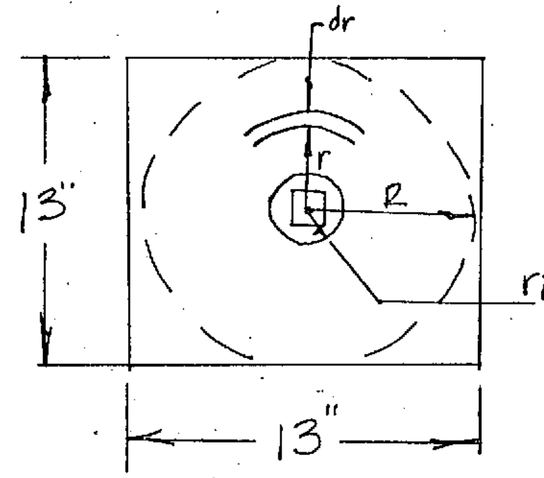


Figure 10 - T/E Module Instl.

$$\frac{\dot{Q}}{A} = \frac{BTU}{Hr\text{-}ft^2} \quad = \text{average heat flux to plate}$$

$$K_{alum} = 1200 \ \frac{Btu - in}{hr - ft^2 - °F}$$

t = temperature

$$\frac{\dot{Q}}{A}\,\pi(R^2-r^2) = -K 2\pi r m \frac{dt}{dr}$$

$$-\frac{\frac{\dot{Q}}{A}(R^2-r^2)}{2Km}\,\frac{dr}{r} = dt$$

$$-\frac{\dot{Q}/A}{2Km}\int_R^{r_i}\left[R^2\frac{dr}{r} - r\,dr\right] = \int_{t_R}^{t_i} dt$$

$$-\frac{\dot{Q}/A}{2Km}\left[R^2 \ln\frac{r_i}{R} - \frac{1}{2}(r_i^2 - R^2)\right] \ \Delta t$$

$$\Delta t = \frac{\dot{Q}/A}{2Km}\left[R^2 \ln\frac{R}{r_i} - \frac{1}{2}(R^2 - r_i^2)\right]$$

Where: R = 6.5

r = 2

$$\dot{Q}/A = 0.565 \ \text{Btu/hr-ft}^2$$

$$\Delta t = \frac{0.565}{2 \times 1200 \times .03}\left[(6.5)^2 \ln\frac{6.5}{2} - \frac{1}{2}(6.5^2 - 2^2)\right]$$

$$\Delta t = \frac{0.565}{2 \times 1200 \times .03}\left[\frac{28.2}{47.5 - 19.3}\right]$$

Δt = 0.22°F differential between module boss & 13" radius circle

Since this differential is very small along the wall - NEGLECT

3.3.3 Temperature Variation to Remote Point

Consider the freezer as a developed box as in Fig. 11.

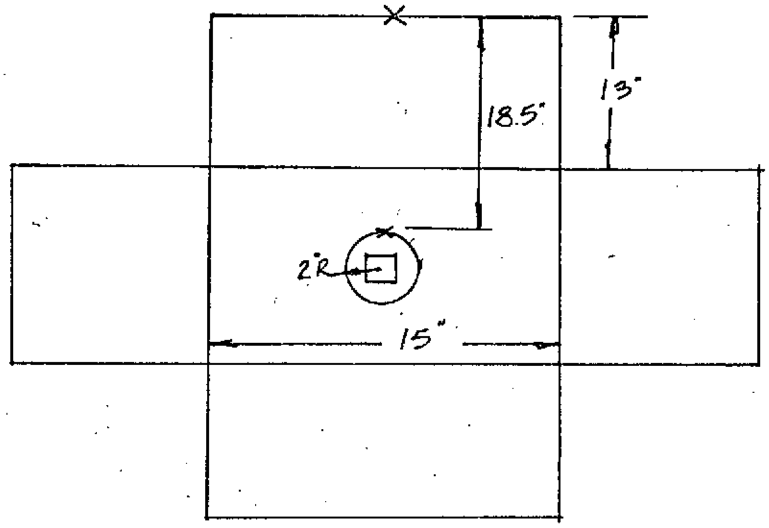


Figure 11 - Freezer Developed Box Config.

$$t \doteq \frac{0.565}{2 \times 1200 \times .03} \left[19.5^2 \ln \frac{19.5}{2} - \frac{1}{2} \left(19.5^2 - 2^2\right)\right]$$

$$t = 5.2°F$$

With single module and simple .03 aluminum liner, t from module to distant points along the freezer wall are relatively small.

A small amount of finning or heat piping will produce uniform temperature. An opened view of the freezer showing such a design is shown in Figure 12.

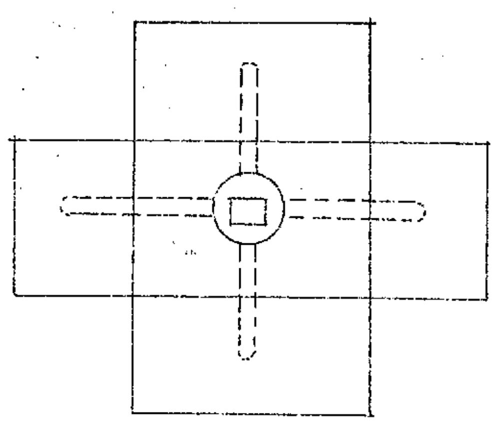


Figure 12 - Heat Pipes

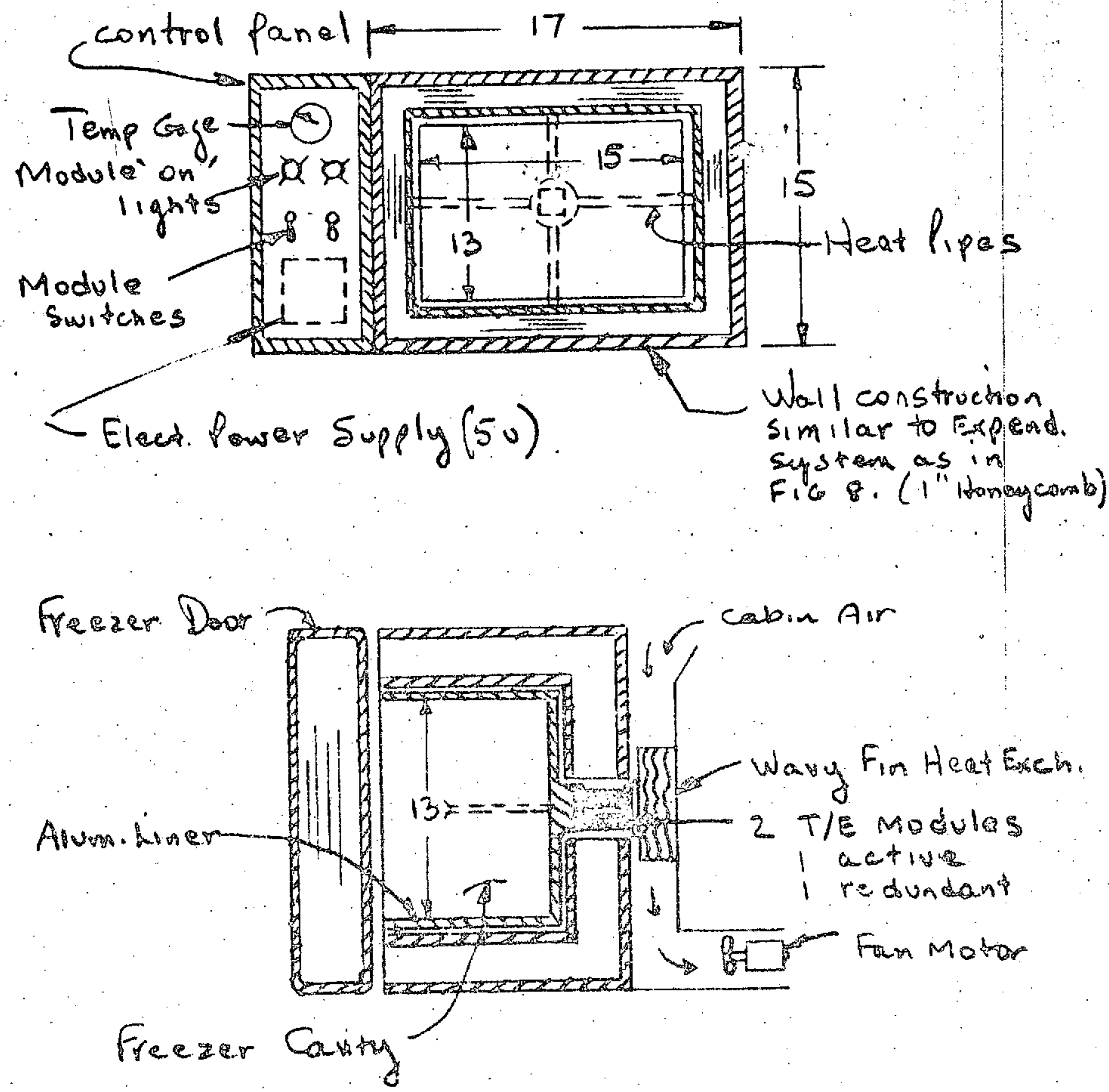
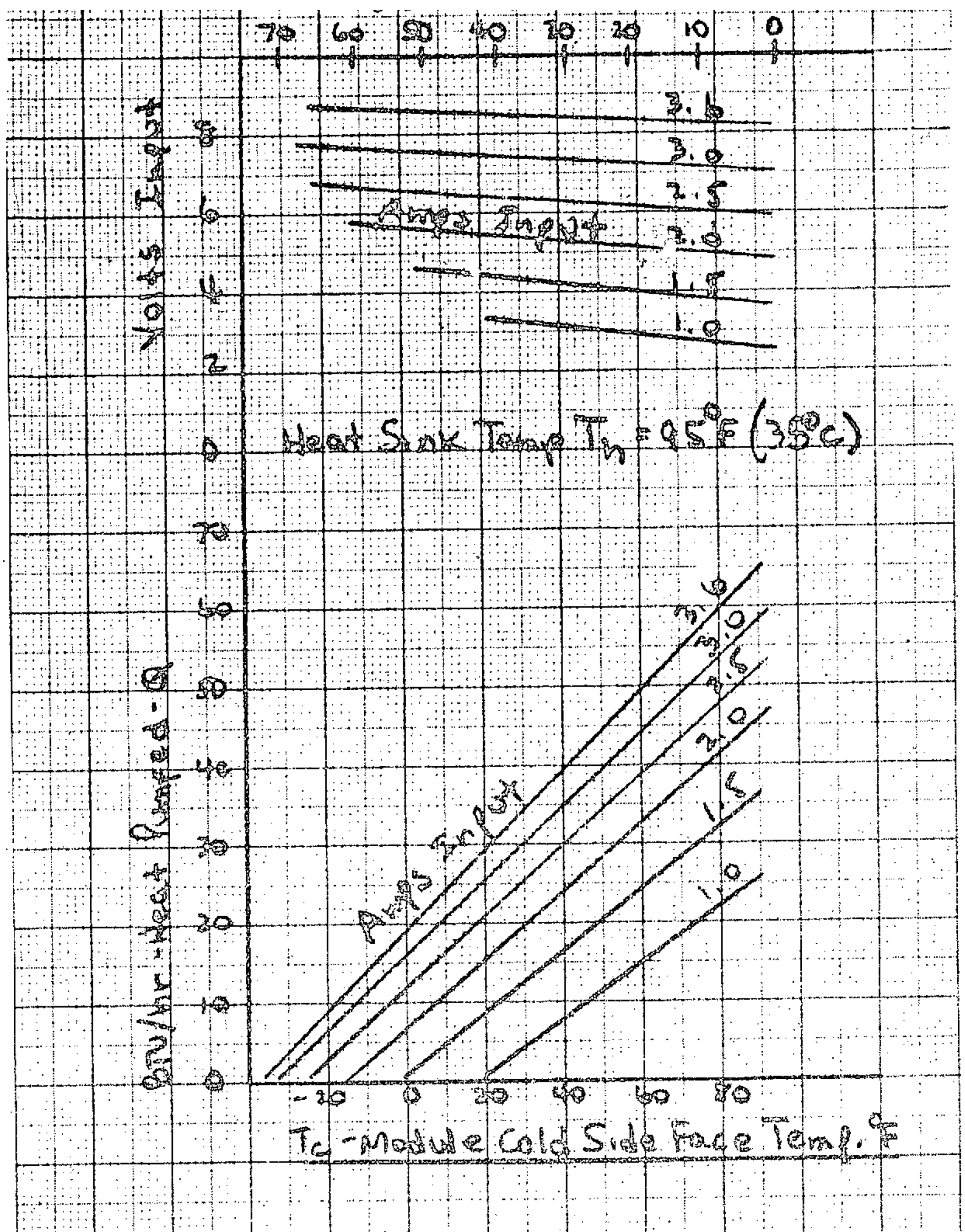


Figure 13 - Freezer Design Configuration



MATERIALS ELECTRONIC PRODUCTS CORP.
T/E MODULE CP1.4-71-10

FIGURE 14 - MODULE CHARACTERISTICS

3.3.4 Design Configuration

The design configuration proposed for the thermoelectric freezer is shown in Figure 13.

3.3.5 Preliminary Results

The T/E Module selected is a Melcor T/E Module CP 1.4-71-10 to provide the required cooling requirements.

Module characteristics are shown in Figure 14.

3.3.5.1 Weight

Honeycomb Shell	18.0#
Aluminum Shelves & Separators	2.5
Aluminum Liner & Heat Pipes	2.8
Super Insulation	1.0
(2) T/E Modules	0.2
Heat Sink Core	0.1
Elect Power Supply & Control	1.5
Ducts	0.1
Fan Motor	0.4
Control Panel	0.4
Mounts, Supports, Switches etc.	0.5
Miscellaneous & Contingency	1.5
	29.0 #

3.3.5.1 Cont'd

∴ Freezer Box = 29# to which penalties must be added for total system weight.

3.3.5.2 Heat Rejection Penalty Weight

Use TRW Globe Model 3A1246 Fan Motor Power = 14 Watts

Electrical load for T/E Module = 15.5 Watts

Electrical Penalty - 14 + 15.5 = 29.5 Watts

$$\text{Total Weight Penalty} = \frac{29.5\text{W} \times 7 \times 24}{1000 \text{ W/Kw}} \text{ Kw hr} \times 1.514 \frac{\#}{\text{Kwhr}}$$

$$= 7.5\#$$

3.3.5.3 Heat Rejection Penalty Weight

Electrical Penalty = 29.5 Watts

$$29.5 \text{ W} \times 3.41 \frac{\text{BTU}}{\text{Watt hr}} \times 0.133 \frac{\#}{\text{Btu/hr}} = 13.4\#$$

3.3.5.4 Total System Wt.

Total Wt = $\sum$ of Hardware wt. + Penalties =

$$29 + 7.5 + 13.4 = 49.9\#$$

Above weight is based on rejecting heat to the cabin. If a liquid loop heat sink is employed the following savings occur.

$$\text{Elect. Penalty} = \frac{15.5 \times 7 \times 24}{1000} = 2.6\#$$

(Fan Motor deleted)

Heat Rejection Penalty = 15.5 x 3.41 x 0.1 = 5.3#
(Fan Motor deleted)

Fan Wt. = 0.4# deleted

∴ Total Penalty for a liquid loop heat sink

$$29 - 0.4 + 2.6 + 5.3 = 36.5\#$$

3.3.5.5 Volume

Basic Freezer Cavity = 15 x 13 x 13

Allow 1" Honeycomb around cavity

$V = 17 \times 15 \times 15 = 3825 \text{ in}^3$

$V = 2.21 \text{ ft}^3$

Control Panel = 15 x 4 x 3 = .10 ft^3

Total Volume = 2.21 + .10 = 2.31 ft^3

3.3.6 Conventional Insulation Analysis

Based on previous analysis in Section 3.2.1, it has been determined that for a 61°F cabin dewpoint, the insulation thickness to prevent condensation will be 2.63" and the conduction load is 72 Btu/hr.

To determine penalties, the load on the T/E Module can be estimated for the best coefficient of performance (COP). For T_c = 0°F and T_h = 95°F, assume a range of Q valves and read amps on Figure 14 (Ref). For each valve, read a corresponding voltage. Results are shown in Table 8.

Q_i Heat Absorbed	I Amps	V Volts	Qi/IV Est. Cop
0.5	1.5	4.6	0.072
7.0	2.0	5.6	0.625
11.5	2.5	6.6	0.696
16.0	3.0	7.7	0.692
18.5	3.6	8.8	0.58

Table 8 - Module Optimization

3.3.6 Cont'd

Based on Table 7 (Ref), peak COP is at .696

$\therefore$ No. of Modules required = $\frac{72 \text{ Btu/hr}}{11.5}$ = 6.25 or 7

Power Required = (6.25)(2.5 amps)(6.6 volts) = 103 watts

Equiv. BTU = 351.2 BTU/hr

Heat Rejected = Q_R = Power + Heat load

= 351.2 + 72 = 351.2 $\frac{\text{BTU}}{\text{hr}}$

However since 72 BTU/hr is also the heat absorbed neglect in this calculation.

Penalties = Elect. + Heat Rej + insulation wt.

Elect (excluding fan) = $\frac{103}{1000}$ x 7 x 24 x 1.514 = 26.2#

Heat Rejection = 351.2 x 0.133 = 47#

Total 73.2#

Plus wt. of insulation.

The Elect. + Heat penalties alone for 2.63" of insulation are almost double the total penalty for super insulation and therefore unacceptable.

In order to lower the conduction load, thereby reducing the Elect and Heat Rejection penalties, a practical limit of 4" thickness insulation was considered. From Figure 7 (Ref), the heat load is 58 Btu/hr at 4" thickness insulation.

No. of Modules = $\frac{58}{11.5}$ = 5

Heat Rejection Penalty = $\frac{5}{6.25}$ x 47# = 37.5#

Elect. Penalty = $\frac{5}{6.25}$ x 26.2 = 21 #

58.5#

Plus wt. of insulation.

3.3.6 Cont'd

Conclusion

Conventional Insulation results in severe penalties of weight and is not considered feasible for this application. Additional penalties not calculated would be an increase in heat rejection to the heat exchanger and increased blower requirements.

3.3.7 Freezer Door Opening Impact

Assume 2 air changes/day rather than 12.

Conduction Load remains unchanged $= 2.34 \frac{\text{BTU}}{\text{hr.}}$

(as per Section 3.2.5)

Ratio 12 changes to 2 air changes

2/12 x 2 Btu/hr = 0.33 Btu/hr

2.67 Btu/hr.

For T/E Module Cp1.4-71-10 @ 10°F

I = 1.6 amps

V = 5 Volts

Power = 1.6 x 5 = 8 Watts

Power Supply = 0.64 Efficiency

$$\text{Power} = \frac{8}{0.64} = 12.5 \text{ Watts}$$

Use same blower @ 14 Watts

Elect Power = 12.5 + 14 = 26.5 Watts

$$\text{Elect. Penalty} = \frac{26.5}{1000} \times 7 \times 24 \times 1.515 = 6.7\#$$

Heat Reject Penalty = 26.5 x 3.41 x 0.133 = 12.1#

3.3.7 Cont'd

Weight Savings

Elect Penalty = 7.5 (Sect. 3.3.5.2)

-6.7

0.8#

Heat Reject Penalty = 13.4 (Sect. 3.3.5.3)

-12.1

1.3#

Total Wt. Savings = $\sum$ = 0.8 + 1.3 = 2.1#

plus slightly smaller heat reject. heat exchanger;

assume saving of 0.2#

Total System Wt. Savings = 2.1 +.2 = 2.3#

For cabin heat rejection

Wt = 49.9 - 2.3 = 47.6#

For liquid loop heat sink

Wt = 36.5 - 2.3 = 34.2#

3.3.8 Thermoelectric 20°F Freezer

Cabin temp = 75°F

Inner Liner temp = 10°F

$\Delta T = 75 - 10 = 65°F$

$\therefore Q = \frac{65}{85} \times 0.78 = 0.6$ Btu/hr

Ref. Sect. 3.2.5

Load with 200% increment for leaks

= 3 x 0.6 = 1.8 BTU/hr.

Use 2 air changes/day

Freezer air at 20°F

Wt of air in cavity (empty) = $\frac{460}{480} \times 0.127 = 0.122$#/door opening

3.3.8 Cont'd

At 2 openings/day = 2 x 0.122 = 0.244#/day

Allowing air to enter at 80°F

$$Q'_{air} = \frac{0.244 \times 0.24\ (80-20)}{24\ \text{hrs}} = 0.147\ \text{Btu/hr}$$

$$Q_{Total} = 1.8 + 0.147 = 1.95 \quad - \quad \text{approx } 2\frac{\text{BTU}}{\text{hr.}}$$

Use Melcor CP 1.4-71-10 T/E Module

T Module @ + 10°F T_h = 95°F

Load - 2 Btu/hr

I = 1.35 amps

V = 4 Volts

Power = I V = 1.35 x 4 = 5.4 Watts

Power Supply Eff. = 0.64

$$\therefore \text{Power Required for module} = \frac{5.4}{0.64} = 8.45\ \text{Watts}$$

3.3.8.1 Electrical Penalty Wt.

Fan load - Heat rejection = 14 watts

Total Electrical load = 8.45 + 14 = 22.45 Watts

$$\text{Elect. Penalty} = \frac{22.45}{1000} \times 7 \times 24 \times 1.515 = 5.7\#$$

3.3.8.2 Heat Rejection Penalty Wt.

$$Wt = 22.45\ \text{Watts} \times 3.41\ \text{Btu/Watthr.} \times 0.133\ \frac{\#}{\text{Btu/hr.}}$$

$$= 10.2\#$$

3.3.8.3 Total System Weight

Elect. Penalty = 7.5# (Sect. 3.3.5.2)

-5.7#
1.8#

Heat Rejection Penalty = 13.4# (Sect. 3.3.5.3)

-10.2#
3.2#

Savings = 1.8 + 3.2 = 5.0#

3.3.8.3 Cont'd

Total System Weight

For cabin heat rejection - 49.9 - 5.0 = <u>44.9#</u>

For liquid loop heat sink - 36.5 - 5.0 = <u>31.5#</u>

3.3.8.4 <u>Volume</u>

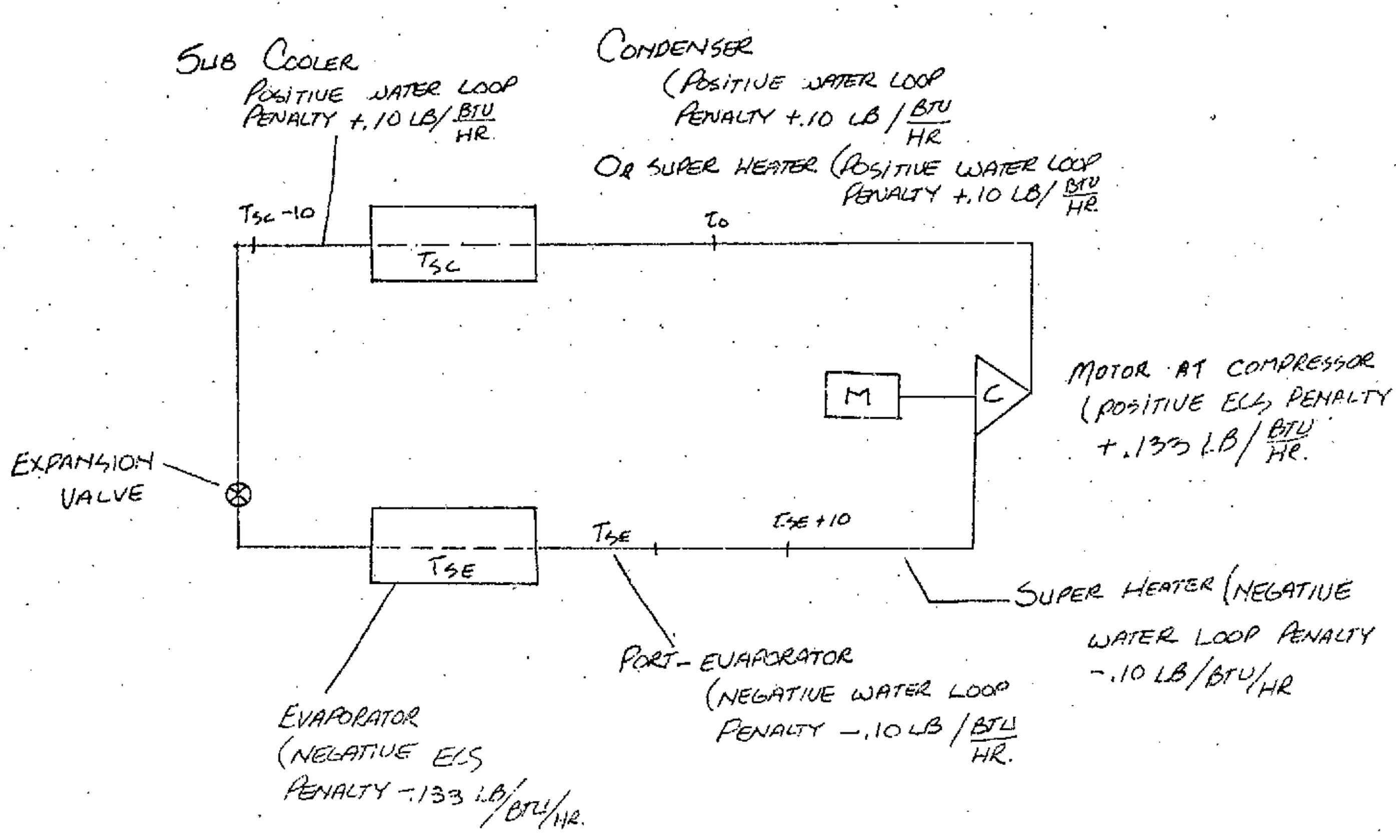
Volume - <u>2.31 ft^3</u>

See Sect. 3.3.5.5

3.4 Vapor Cycle Freezer/Refrigerator

3.4.1 Description

A vapor cycle employing Freon-12 refrigerant was studied as one of the food storage concepts for the space shuttle. The cycle schematic and the identification of the various heat rejections and shuttle interface penalties are as follows:



3.4.1.1 Thermal Interfaces

Heat rejection from compressor: $(1-\eta_{ad,c})\,HP_c$

Heat rejection from electric motor: $\frac{1-\eta_m}{\eta_m}\,HP_c$

Evaporator heat gain: $W_R L(t_{SE})(N_{OUT}-N_{IN})$

Port evaporator heat gain: $W_R L(t_{SE})(1.0-N_{OUT})$

Super-heater heat gain: $W_R C_{PR}(t_{SE}+10-t_{SE})$

De super-heater heat rejection: $W_R C_{PR}(t_{SC}-t_o)$

Condenser heat rejection: $-W_R L(t_{SC})$

Sub-cooler heat rejection: $W_R C_{PL}\{(t_{SC}-10)-t_{SC}\}$

Motor power input: $\frac{HP_c}{\eta_m}$

Electrical interface penalty 1.514 Lb/Kw.Hr

3.4.1.2 Definitions:

t_{SC} ~ Condenser saturation temperature (= 90°F)

t_{SE} ~ evaporator saturation temperature

t_o ~ refrigerant vapor temperature after compression

$\eta_{ad,c}$ ~ compressor efficiency (=.70)

HP_c ~ compressor power input

η_m ~ motor efficiency

W_R ~ refrigerant flow rate

$L(t_{SE})$ ~ refrigerant latent heat at

N_{OUT} ~ refrigerant quality at evaporator exit

N_{IN} ~ refrigerant quality at evaporator inlet

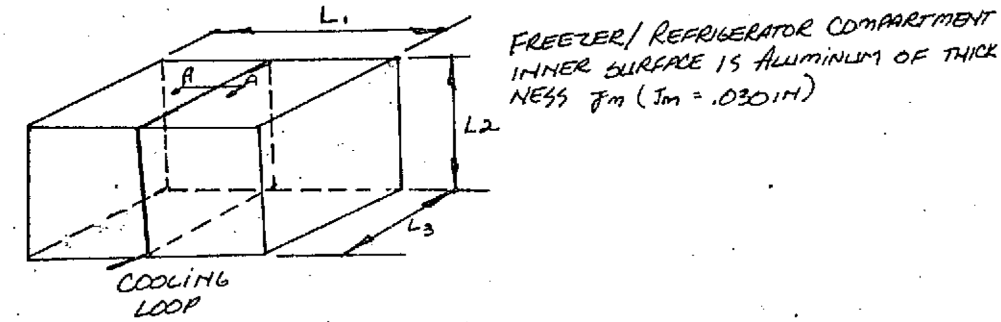
C_{PR} ~ refrigerant vapor specific heat

$L(t_{SC})$ ~ refrigerant latent heat at

C_{PL} ~ refrigerant liquid specific heat

3.4.2 Thermal Analysis

The evaporator saturation temperature required to maintain a given freezer or refrigerator temperature was calculated as follows:



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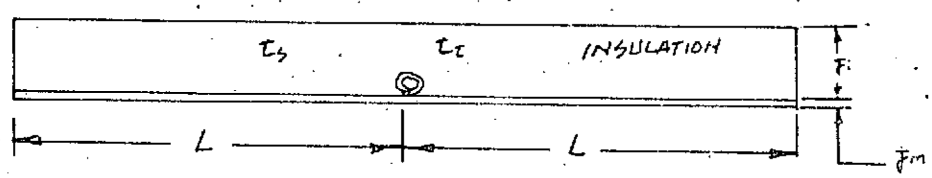

(1.) $\pi D_i h_E (t_t - t_s) = 2\sqrt{\frac{H}{k_m \delta_m}}\, k_m \delta_m \left\{\tanh\sqrt{\frac{H}{k_m \delta_m}}\, L\right\}(t_f - t_t)$, HEAT CONDUCTORS TO HAVE OF FIN.

Where D_i tube inner diameter

h_e tube internal heat transfer coefficient

t_t tube temperature

t_s saturation temperature

k_m fin thermal conductivity (k_m =100BTU-ft./hr.ft^2°F)

L fin length

t_f cabin temperature (t_f = 75°F)

H sum of conductances on both sides of fin (h= + h_{fc})

h_B cabin side convective coefficient(h_B=1.45Btu/hr.ft^2°F)

h cabin side ratiative heat transfer coefficient (h =.20(1.05) =.21 Btu/hr.ft.^{2}°F)

3.4.2 Cont'd

k_i insulation thermal conductivity

h_{fc} compartment side heat transfer coefficient due to air heat leak attendant on door opening.

3.4.2.1 Air Changes

Assume that whenever the compartment door is opened a complete change of compartment air occurs. The air mass introduced is calculated on the basis of compartment free volume, cabin pressure, and a temperature that is the average of cabin and compartment temperature. The energy transferred to the compartment walls is taken to be that removed from the air change mass in cooling from cabin temperature to compartment temperature, $\bar{t}$.

The time to dissipate the heat leak is assumed to be proportional to freezer/refrigerator compartment free volume fraction, H.

$$h_{fc} = NWC_P / \Delta \tau A$$

$$W = P_{CABIN} / R(460 + .5(t_F - \bar{t}))$$

R ~ air gas constant

C_p ~ air specific heat at constant pressure

$\Delta\tau$ ~ .25 Hr. for = 1

A ~ freezer/refrigerator surface area

V_F ~ freezer/refrigerator volume

3.4.2.2 Compartment Temperature

The freezer/refrigerator compartment temperature, $\bar{t}$, is defined as the average temperature of the compartment aluminum inner surface. The temperatures on this surface vary with distance from the cooling loop. The first is due to the thermal resistance of the surface material, while the second is due to the variation of heat transfer coefficient with refrigerant quality. The average temperature, $\bar{t}$, is defined by first integrating the fin equation to find the average temperature over the distance from the cooling loop, then utilizing an average coolant loop heat transfer coefficient to find an average coolant loop tube temperature (base of fin).

The average coolant loop tube temperature is then utilized to calculate the evaporator saturation temperature required to maintain a given freezer or refrigerator temperature.

3.4.2.3 Coolant Loop

(2) $$\bar{t} = t_f - (\bar{t}_t - t_f)\,\frac{\tanh\sqrt{H/k_m F_m}\,L}{\sqrt{H/k_m F_m}\,L}$$ ← Integrating the fin equation.

where $\bar{t}$ average surface temperature over distance from coolant loop, freezer/refrigerator temperature,

$\bar{t}_t$ ~ average coolant loop tube temperature.

Rearranging (2)

(3) $$\bar{t}_t = t_f + (\bar{t} - t_f)\,\left(\sqrt{H/k_m F_m}\,L\right) / \tanh\sqrt{H/k_m F_m}\,L$$

From (1)

$$(4) \quad t = \bar{t}_t + \frac{2\left(\sqrt{H/k_m \delta_m}\right) L \tanh\left(\sqrt{H/k_m \delta_m}\, L\right)\left(\frac{k\,\delta_m}{L}\right)(\bar{t}_t - t_s)}{\pi D_i \bar{h}_e}$$

Thus, specifying $\bar{t}$ determines $\bar{t}_t$. The value $\bar{t}_t$ and the average heat transfer coefficient, $\bar{h}_e$, determines t_s. Since preliminary calculations showed that coolant loop tube temperature would be fairly constant over a range of refrigerant quality 0.2 = = 0.8, the value of $\bar{h}_e$ used was that averaged up to a quality of 0.8. The variation of h_e with Freon-12 quality is given in Figure 15 taken from Reference 1. Since these values were measured at high refrigerant vapor velocities in vertical tubes, the effect of body force due to gravity would be minimized and the values would be applicable to the care of zero g. The variation of $\bar{h}_e$ with t_s is given in Figure 16. This relationship is arrived at by expanding the refrigerant at constant enthalpy from a condenser temperature of 90°F with 10°F sub cooling to t_s. This expansion determines refrigerant quality, , into the evaporator. The coefficient, $\bar{h}_e$, is the integrated average value over the quality range to 0.8.

Subsequent to the evaporator (which exchanges heat with the cabin), the refrigerant completes evaporation in a port-evaporator, and then is superheated 10°F to insure a single phase at the compressor inlet. Both port evaporator and superheater exchange heat with the shuttle water loop as do the de-superheater, the condenser and the subcooler.

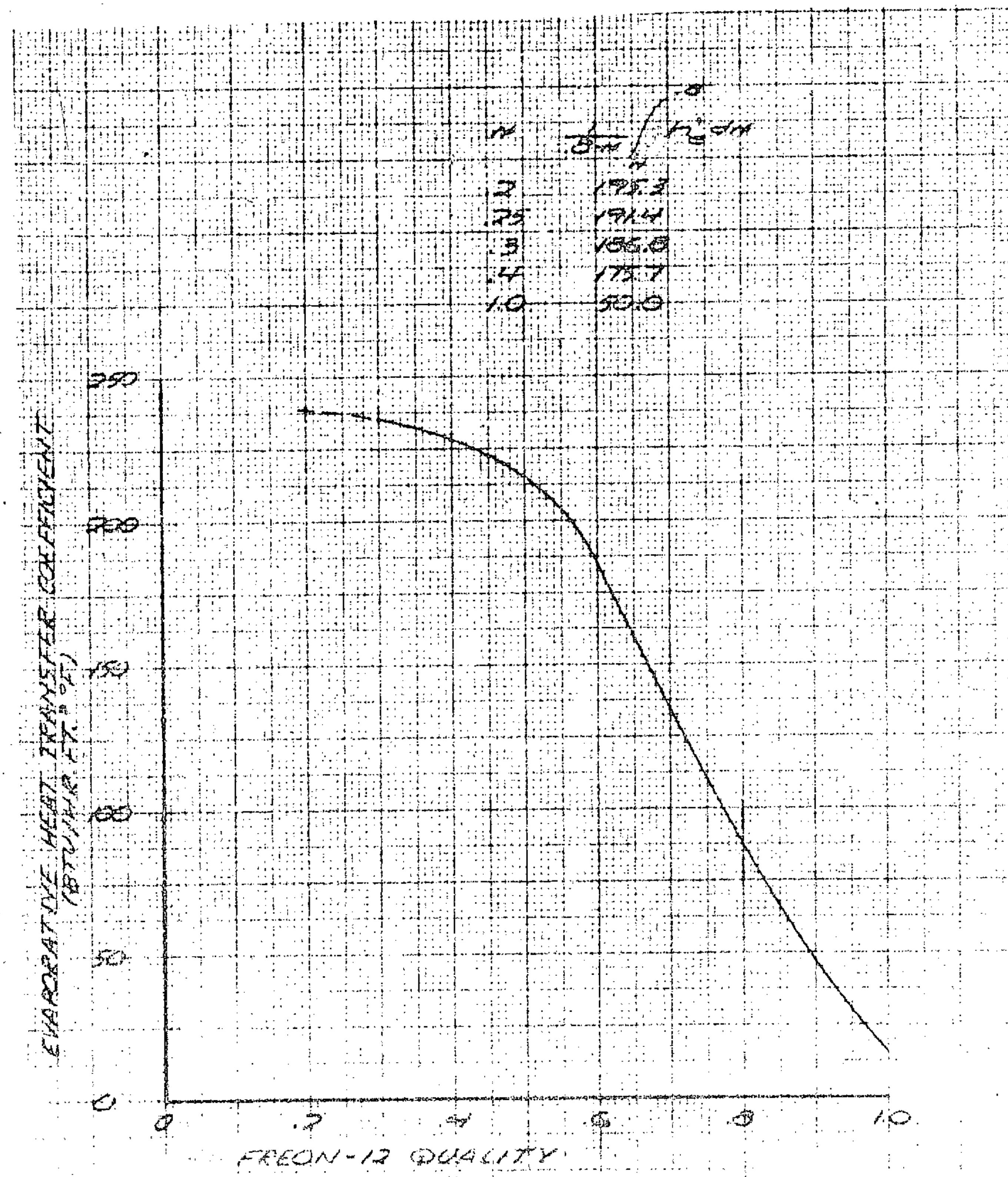

FIGURE 15 - VARIATION OF h_e WITH FREON-12

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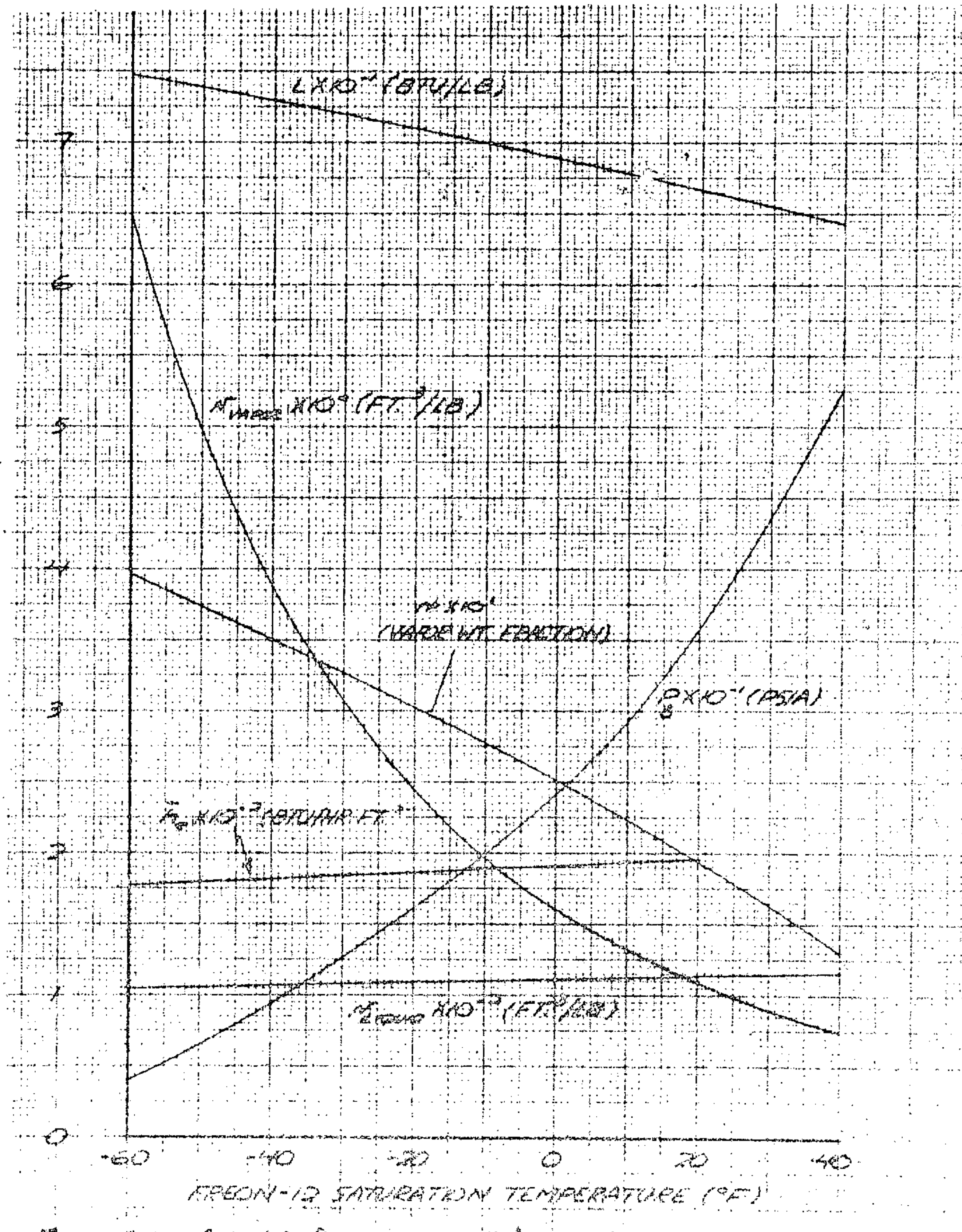

FIGURE 16 – VARIATION OF h_c WITH TEMPERATURE

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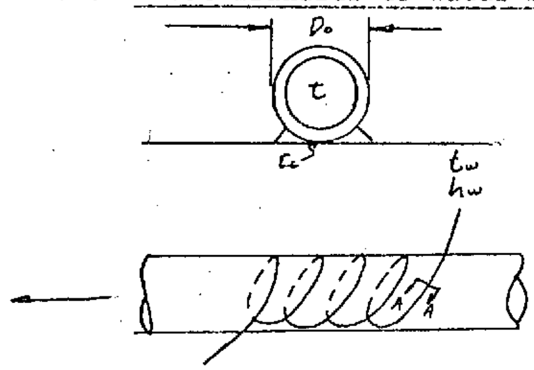
5.4.2.4 Heat Transfer to Water Loop



ASSUME FILLET AREA EFFECTING FOR HEAT TRANSFER IS $\frac{2}{3} D_o$ AND THAT REFRIGERANT TUBE PERIMETER IS AT t_o.

$$h(\pi D_i)(t - t_o) = h_w(\tfrac{2}{3} D_o)(t_o - t_w)$$

$$UP = 1 \Big/ \left\{ \frac{1}{h' \pi D_i} + \frac{1}{h_w(\tfrac{2}{3} D_o)} \right.$$

Vapor-super heater, Sub cooler, De-superheater

$$UP\,ds\,(t - t_w) = W_R\,C_{PR}\,dt$$

$$\frac{-UPS}{W_R C_{PR}} = \ln\left(\frac{t - t_w}{t_{IN} - t_w}\right)$$

$$\frac{t - t_w}{t_{IN} - t_w} = EXP\left(-\frac{UPS}{W_R C_{PR}}\right) \quad 5.$$

where t ~ refrigerant temperature out

t_W ~ shuttle water loop temperature

t_{in} ~ refrigerant temperature in

S ~ coolant tube length

W_R ~ refrigerant flow rate

C_{PR} ~ refrigerant specific heat

The refrigerant flow rate is given by

$$W_R = S\left\{\pi D_i \bar{h}_e (\bar{t}_c - t_s)\right\} \Big/ \left\{L(N_{OUT} - N_{IN})\right.$$

Where S ~ evaporator coolant tube length

3.4.2.4 Cont'd

Condenser

$$UPdS(t_S - t_N) = -L(t_S)\bar{w}_R\, dN$$

$$UPS(t_S - t_w) = -L(t_S)W_R\{N(S) - N(0)\}$$

$$N(0) = 1.0 \text{ AND } N(S) = 0$$

(6). $UPS(t_S - t_w) = L(t_S)w_R$

Port Evaporator (h - 50 Btu/Hr.Ft.^{2}°F)

$N(0) = .8$ and $N(5) = 1.0$

7. $UPS(t_S - t_w = -.2\, \ell(t_S)W_R$

3.4.2.5 Heat Transfer Coefficient (h_w) within Water Loop

Water flow 550

Water temperature t_w = 75°F

Tube O.D. 1.0 In.
I.D. .93 In.

Turbulent flow

$$\frac{h_w D_i}{k_b} = .023\left(\frac{DG}{\mu_b}\right)^{.8}\left(\frac{C\mu}{k}\right)_b^{.4}$$

where subscript "b" denotes properties evaluated at bulk temperature.

3.4.2.6 Heat Transfer Coefficient Within Superheater, De-superheater

Tube O.D. 0.235 In.

I.D. 0.055In.

Turbulent flow :

$$\frac{h D_i}{k_b} = .023\left(\frac{D_i G}{\mu_b}\right)^{.8}\left(\frac{C_p \mu}{k}\right)_b^{.4}$$

3.4.2.7 Heat Transfer Coefficient Within Sub Cooler

Tube O.D. 0.125In.

I.D. 0.055 In.

Turbulent flow (R_e 2000)

$$\frac{h\,D_i}{k_B} = .023\left(\frac{D_i G}{M_B}\right)^{.8}\left(\frac{C_p M}{k}\right)_b^{.4}$$

FLOW ($R_E = 2000$)

$$\frac{h\,D_i}{k} = 1.75\left(\frac{W c_p}{k S}\right)^{1/3}$$

3.4.2.8 Heat Transfer Coefficient Within Condenser

From Reference 2,

$$\frac{h_c M_\ell}{k_\ell \rho_\ell^{1/2}} = .065\left(\frac{C_p m}{k}\right)_\ell^{1/2} F_{vc}^{1/2}$$

where C_p, , k and are liquid properties

$$F_{vc} = \frac{f}{2}\frac{G_m^2}{\rho_v}$$

is a function of Reynolds number based on an average vapor mass flow rate, G_m, per unit area, vapor properties and a surface roughness parameter $\frac{k}{D_i}$. Assume K = .000005 (drawn tubes). If it is assumed that the condensing rate is uniform and that, as a result, the vapor velocity decreases with length, then the proper average value (one which will give the same total fraction) is derived to be

$$G_m = \left(\frac{G_1^2 + G_1 G_2 + G_2^2}{3}\right)^{1/2},$$

where G_1 and G_2 are the inlet and outlet valve, respectively.

For $G_2 = 0_2$ $G_m = .58\ G_1$

Thus, f is a function of $.58G_1\ D_i/$ and K/D_i.

3.4.3 System Weights and Penalties

3.4.3.1 Approach

Equations (5), (6), and (7) are utilized to calculate refrigerant line lengths once evaporator saturation temperature (t_{sE}), compressor discharge temperature (t_D), condensor saturation temperature (t_{sc}), and water loop temperature (t_w) are known. The latter temperatures were fixed at 90°F and 75°F respectively. Compressor discharge temperature was calculated by the following relationships:

$$H_{AD,C} = \frac{r}{r-1} R (t_{SE}+10)\left\{\left(\frac{P}{P_S}\right)^{\frac{r-1}{r}} - 1\right\}, \quad r = 1.13, \quad R = 12.78$$

where P ~ saturation pressure at 90°F

P_S ~ saturation pressure at t_{se}

$$HP_C = \frac{\rho_{IN}}{550\,\eta_{AD,C}} Q H_{AD,C}, \quad Q \sim \text{VOLUMETRIC FLOW RATE AT INLET}, \ \rho \sim \text{DENSITY}$$

$$t_D = (t_{SE}+10) + H_{AD,C}/\{\eta_{AD,C}\, C_{PR}\}, \quad C_{PR} = .148$$

3.4.3.2 Assumptions

Refrigerant lines and the Shuttle water loop line were assumed to be of aluminum. The length of water loop line required for the various heat transfers was that required to accomodate the refrigerant lines helically wrapped with no spacing between loops. The freezer refrigerator inner surface was assumed to be 0.040 fiberglass with a density of 110 Lb/Ft.^{3}. The outer, cabin side surface was assumed to be 0.030 aluminum with a density of 173 Lb/Ft.^{3}. Two insulation systems were studied: a conventional fiberglass with a density of 0.6 Lb/Ft.^{3} and hinde SI-12 superinsulation with a density of 3.0 Lb/Ft.^{3}

3.4.3.3 Equipment Weights.

Motor, compressor, and expansion value weights and volume were estimated by means of relationships given in Reference 3:

D.C. motor weight - - $W_m = 1.5 + 3.83(HP_c)^{5/6} \cdot (N \times 10^{-4})^{-1.25}$ $N \sim RPM$ $N = 12{,}000$ $W_M \sim LB$

D.C. motor volume - - $V_m = W_m/0.1 \sim IN^3$

Compressor weight - - $W_c = 1.5\,W_m$

Compressor volume - - $V_c = 1.5\,V_M$

Expansion valve weight - - W_E 0.5 LB

Expansion valve volume - - $V_E = W_E/0.1$

The motor efficiency also was estimated by means of a relationship given in Reference 3.

$$\eta_m = 1.02\{1 - .281(HPC)^{-.169}\}$$

3.4.3.4 Penalty Factors

System penalties in terms of equivalent weight were determined by means of the relationships given on the page following the schematic and by means of the penalty factors given on the schematic.

It was found that refrigerant line pressure drops were negligible due to the low refrigerant flow rates required.

3.4.3.5 Optimization

The thermal conductivity of the hinde insulation was increased by an order of magnitude (k=.37 x 10^{-2} BTU=Ft/Hr.Ft^2°F) as an allowance for heat leaks through structural attachments between the freezer/refrigerator inner and outer surfaces. The thermal conductivity of fiberglass was not increased since it was assumed that attachments could be fabricated having approximately the same conductance as insulation.

3.4.3.5 Cont'd

The freezer/refrigerator compartment size utilized was that given by the Pillsbury Co. based on the greatest number of frozen/refrigerated items likely to appear on the menu for a six man/7 day mission: 15" x 13" x 13" (L_1 = 15", L_2 = 13")

The results of the analyses optimizing hinde SI-12 super-insulation are given in Figures 17 through 20 with freezer/refrigerator temperature and freezer/refrigerator free volume fraction, $\mathcal{N}$; a parameters. Results are provided only for optimization on a weight basis since optimization and a volume basis has been shown to result in much higher weights (see analysis on phase change materials).

Weight optimized results for super insulation are plotted in Table 9 and for conventional insulation in Table 10.

Results were not generated for the smaller 9" x 10" x 14" freezer/refrigerator configuration since it is felt that the analyses of the larger units provide sufficient data for a relative assessment of the food storage concepts studied.

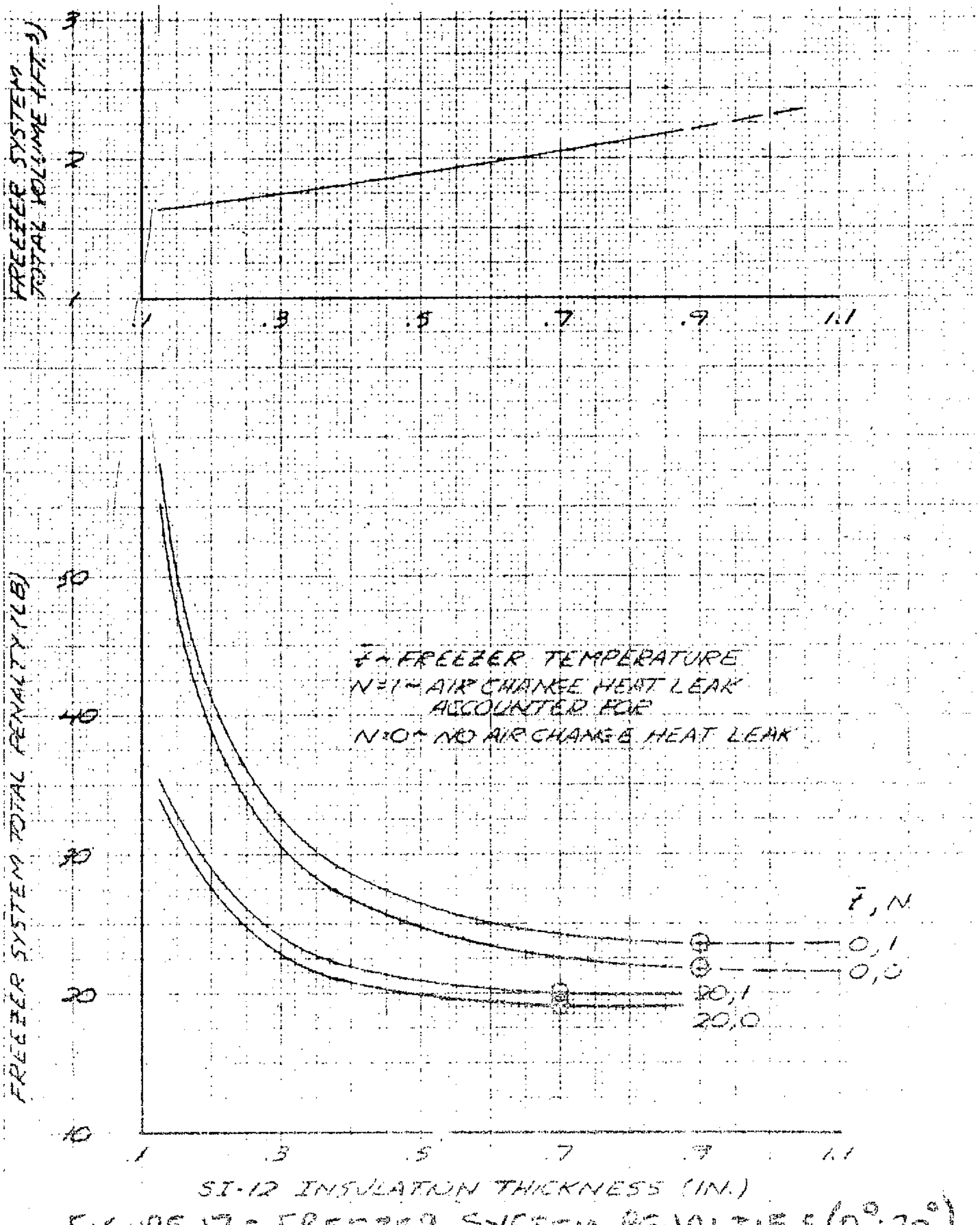


FIGURE 17 - FREEZER SYSTEM PENALTIES (0°; 20°)

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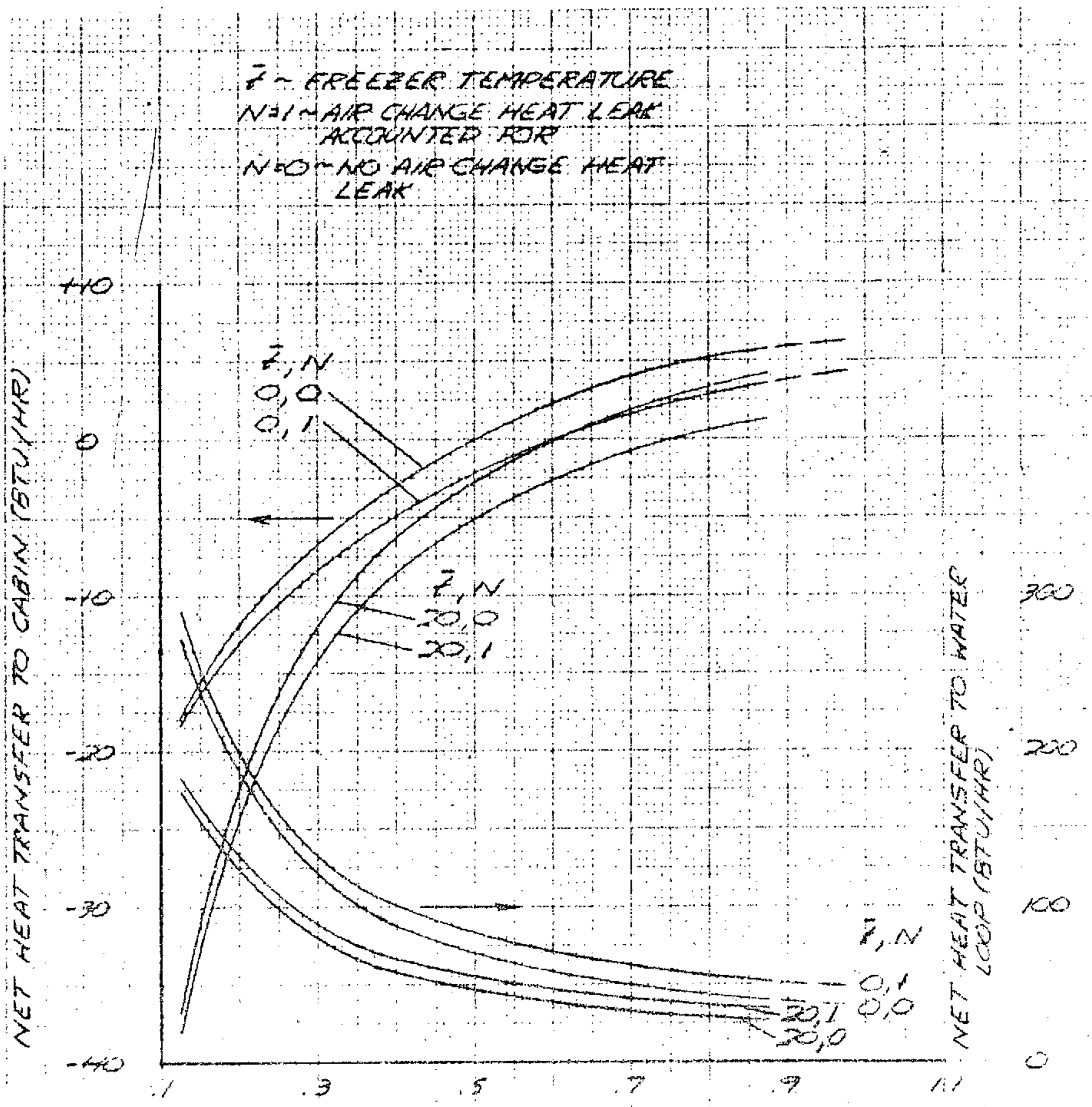

FIGURE 18 - HEAT TRANSFER TO CABIN

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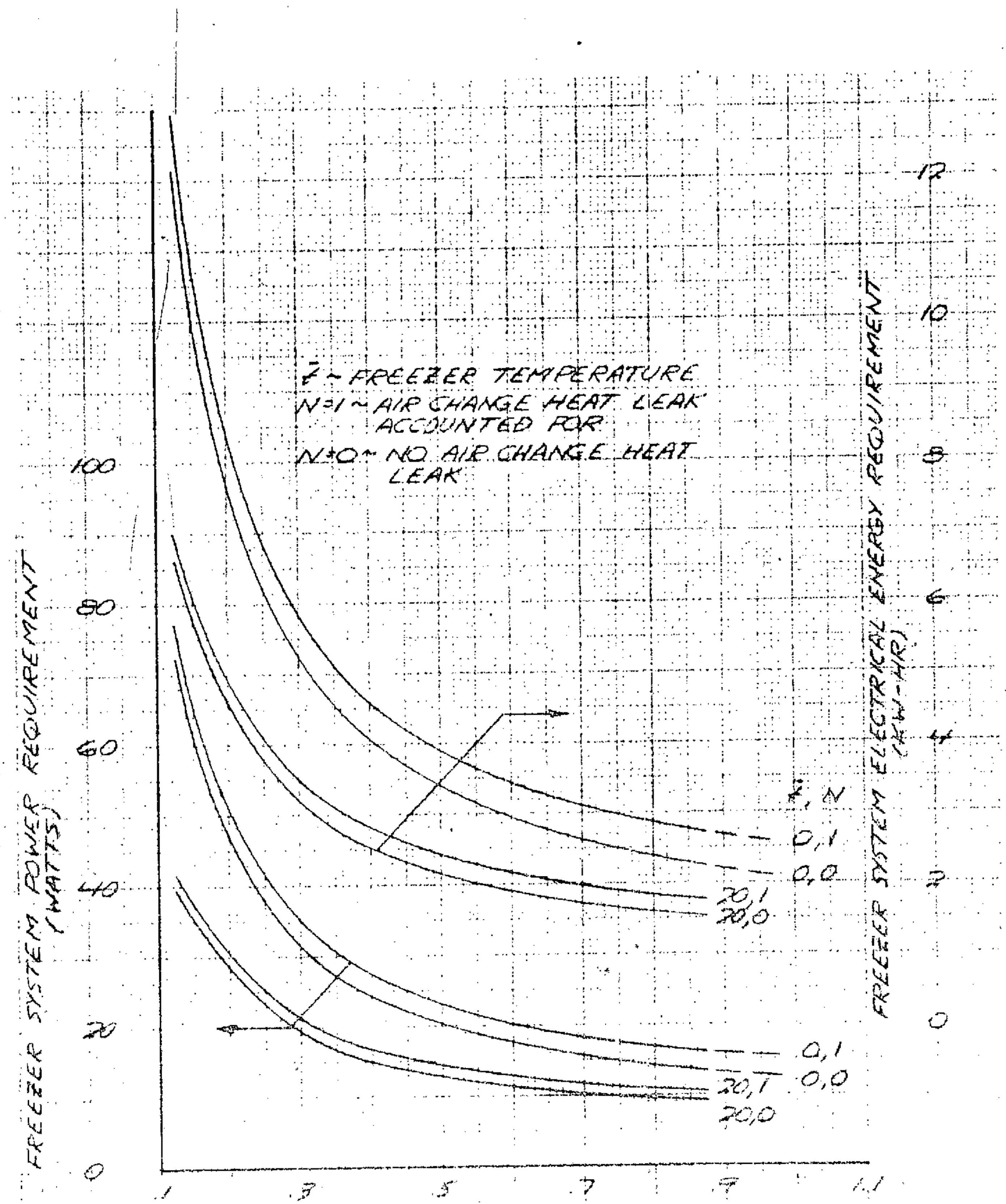


FIGURE 19 - FREEZER SYSTEM POWER (0°, 20°)

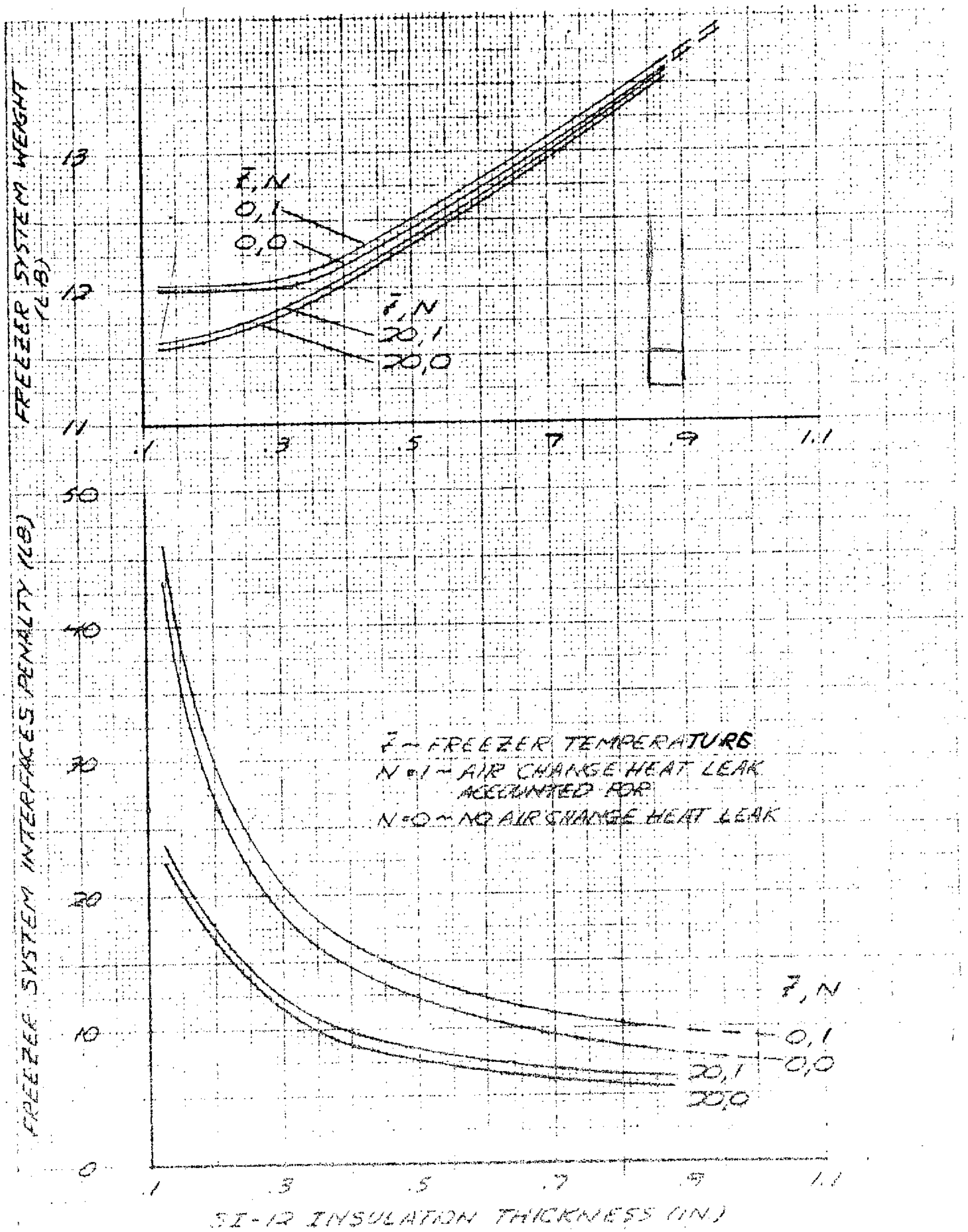


FIGURE 20 - HARDWARE & INTERFACE PENALTIES UIT.

TABLE 9 - Optimum Design - hinde SI-R Insulation

t_i	$\bar{t}$	N	H_A Weight	Equivalent Penalty Wt.	Total Penalty	Total Volume	Net Cabin Heat Transfer	Net Water Loop H.T.	Electrical Energy
.90 IN	0	1	13.8 LB	9.70 LB.	23.5 LB	2.22 Ft^3	3.80BTU/Hr.	51.0 BTU/HR	2.69 KW-HR
.70	20	1	13.0	7.00	20.0	2.05	-.80	33.0	1.98
.45	45	1	12.0	3.62	15.6	1.84	-6.55	10.5	1.09
.90	0	0	13.8	8.00	21.8	2.22	6.00	40.0	2.20
.70	20	0	13.0	6.20	19.2	2.05	.18	33.0	1.71
.45	45	0	12.0	3.95	16.0	1.84	-7.10		1.10

For comparison purposes, results were generated utilizing a conventional fiberglass insulation. Only one insulation thickness (4.0 In) was studied since it was felt that this thickness represents an appropriate practical maximum. For the same column headings as in Table 8;

TABLE 9 - Optimum Design - Fiberglass Insulation

4.0	0	1	18.7 Lb.	11.0 Lb	29.7 LB	6.01 Ft^3	1.50BTU/Hr	60.0BTU/Hr	3.06 KW-HR
	20	1	18.5	7.0	25.5	6.01	-.80	40.0	1.96
	45	1	18.2	2.0	20.2	6.01	-3.68	15.0	.58
	0	0	18.5	9.5	28.0	6.01	3.60	50.0	2.60
	20	0	18.5	6.2	24.7	6.01	1.70	33.0	1.70
	45	0	18.5	2.1	20.6	6.01	-.68	11.8	.58

3.4.4 References

1) Yader, Richard J. and Dodge, Barnett, F., "Heat Transfer Coefficients of Boiling Freon-12", General Discussion on Heat Transfer, Sept. 11 to 13, 1951, Section I, the American Society of Mechanical Engineers, New York, N. Y. , 1951

2) Carpenter, F.G. and Colburn, A.P., "The Effect of Vapor Velocity on Condensation Inside Tubes", General Discussion on Heat Transfer, Sept. 11 to 13, 1951, Section I, The American Society of Mechanical Engineers, New York, N. Y., 1951

3) Dieckmann, R.R., Watson, A.C., and Glover, S.F., Development of Integrated Environmental Control System Designs for Aircraft, Vol. I - ECS Design, AFFDL-TR-72-9, Vol.I, May 1972

3.5 Thermoelectric Refrigerator

3.5.1 Description

The thermoelectric refrigerator is similar in design to the thermoelectric freezer described in section 3.3.1. Sizing is the same and the configuration is based on the design shown in Figure 13 (Ref.)

The large refrigerator (15 x 13 x 13) is used in the analysis.

T_{food} = 40 - 45°F T_{module} @ 30°F

T_{cabin} = 75°F

3.5.2 Cooling Load Calculation

Use SI-4 vacuum insulation = 7#/ft^3

Use honeycomb structure

Box surface area = 1118 (Sect. 3.2.5

$\dot{Q} = 0.78$ Btu/hr (Sect. 3.2.5)

$$\dot{Q} = 0.78 \times \frac{75-30}{85} = 0.415 \text{ BTU/Hr.}$$

Allow 200% increment for heat leaks thru insulation

∴ Add 2 x 0.415 - 0.830

∴ Total Conduction Q_c = .415 + .830 = 1.245 BTU/Hr.

3.5.3 Air Change Load

Assume only 2 door openings in 24 hours.

Cavity Volume = 1.47 ft^3 (Sect. 3.2.6)

$$W_{air} = \frac{14.7 \times 144 \times 1.47}{53.3 \times 500} = .116 \ \frac{\# air}{\text{Door opening}}$$

460° + 40°F = 500°R

Air Change = 2 x .116 = .232# air

3.5.3 Cont'd

$$Q_{air} = \frac{.232 \times 0.24 \times (75-30)}{24 \text{ hrs.}} = .104 \text{ BTU/Hr.}$$

$$Q_{total} = Q_{cond} + Q_{air} = 1.245 + .104 = 1.349 \text{ BTU/Hr.}$$

Use 1.5 BTU/Hr.

3.5.4 Module Power Extrapolation

Try Module - Melcor CP 1.4-71-10 (Ref Fig. 14).

Use $T_c = 30°F$

For 30°F

BTU/Hr.	Amp
3.5	1.0
12.0	1.5
19.5	2.0
25.0	2.5
30.0	3.0
34.0	3.6

Extrapolate to 1.5 BTU/Hr. @ 0.89 amps

For 30°F

I	V
1.0	3.1
1.5	4.3
2.0	5.4
2.5	6.4
3.0	7.6
3.6	8.6

Extrapolate to 0.89 amps @ 3 Volts for 1.5 $\frac{BTU}{hr.}$

3.5.4 Cont'd

Net Module Power = I x V = 0.89 x 3 = 2.67 Watts

Assume power supply eff (n) = 0.64

Power required for Module $= \frac{2.67}{0.64} = \underline{4.2 \text{ Watts}}$

3.5.5 Electrical Penalty Weight

Elect power penalty $= \frac{4.2w \times 7 \times 24}{1000} \times 1.514 = \underline{1.07\#}$

3.5.6 Heat Rejection

Q = 4.2w x 3.41 - 14.3 BTU/Hr.

Q_{total} = 14.3 + 1.5 = 15.8 BTU/Hr.

Liquid Loop penalty = 15.8 BTU/Hr. x 0.1 = 1.58#

3.5.6.1 Heat Rejection to Cabin Air

Cabin h_c = 1.45 BTU/hr-ft^2°F

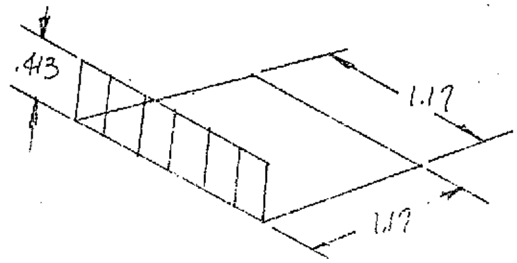
UAΔT = 15.8 BTU/Hr.

$A = \frac{1.58}{1.45(95-80)} = 0.73 \text{ ft}^2$ Reqd.

Try wavy fin 17.8 - 3/8 W @ 514 ft^2/ft^3

Volume reqd. $= \frac{0.73}{514} \times 1728 = 2.45 \text{ in}^3$

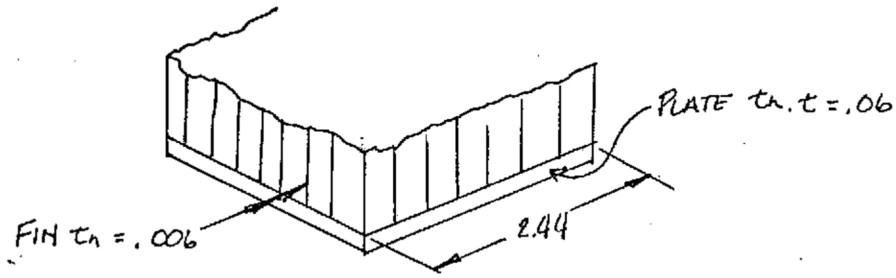
Fin area/total area = 0.892



$.413 x^2 = 2.45 \text{ in}^3$

$x = 3.44''$

3.5.6.1 Cont'd



Vol = Fin metal vol + plate vol.

$= 2.44 \times 17 \times .413 \times .006 \times 2.44 + (2.44)^2 \times .06$

$= 0.61 \text{ in}^3$

Surface wt (alum) = 0.61 x 0.1 = 0.061#

If normal cabin airflow available in area of heat exchanger, then free convection air rejection can be used to dissipate the heat load.

Heat (to air) Weight Penalty =

$15.8 \text{ Btu/Hr.} \times 0.133 \frac{\#}{\text{BTU/Hr}} = \underline{2.1\#}$

3.5.6.2 Potential Fan Penalty

If cabin airflow is not available to dissipate the heat load, assume a small fan will be required in the system.

Assume a 5 watt fan.

Elect Penalty = $\frac{5}{1000} \times 7 \times 24 \times 1.514 = \underline{1.29\#}$

Heat Rejection Penalty = 5 x 3.41 x 0.133 = 2.26#

Fan Weight = 0.25# (Assumed)

Total fan penalty = 1.29 + 2.26 + 0.25 = 3.80#

3.5.7 Total System Penalty

Summing previously calculated penalties

Elect Power Penalty = 1.07# (Sect. 3.5.5)

Heat Rejection (Liquid loop) Penalty = 1.58# (Sect. 3.5.6)

Heat Rejection (cabin air) Penalty = 2.1# (Sect.3.5.6.1)

Hardware Weight = 27.5# (Sect. 3.3.5.1) (Delete int. press. source)

a) Air Heat Rejection

$W = 29 + 1.07 + 2.1 = \underline{32.17\#}$

b) Liquid Loop Heat Rejection

$W = 29 + 1.07 + 1.58 = \underline{31.65\#}$

c) Air Heat Rejection + Fan Penalty (if required)

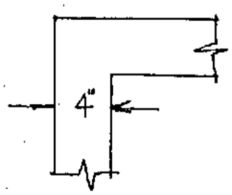
$W = 32.17 + 3.80 = \underline{35.97}$

Volume = 2.31 ft^3 (Sect. 3.3.5.5)

3.5.8 Conventional Insulation Analysis

Thermal Resistance = R Insulation Th = 4"

$$R = \frac{1}{h_c} + \frac{t}{K} + \frac{1}{1.45} + \frac{4}{0.25} \qquad R = 16.69$$



Inside Area = 1118 (Sect. 3.2.5)

$$\text{Outside Area} = 1118 + 164t + 6\,t^2$$
$$= 1118 + 164(4) + 6\,(4)^2$$
$$= 1870 \text{ in}^2$$

$$\text{Average Area} = \frac{1118 + 1870}{2} = 1494 \text{ in}^2 = 0.86 \text{ ft}^2$$

$$Q = \frac{A\,\Delta T}{R} = \frac{0.86(75-30)}{16.69} \qquad Q = 1.8 \text{ BTU/Hr.}$$

Allow 200% for leakage

Q design = 3 x 1.8 = $\underline{5.4 \text{ BTU/Hr.}}$

3.5.8 Cont'd

Air Load - Use 2 changes/24 hours

Cavity Volume - $15 \times 13 \times 13 = 2535 \text{ in}^3$

$$V = \frac{2535}{1728} \times 2 \text{ changes} = 1.3 \text{ ft}^3 \text{ air}$$

$$\rho \text{ air at } 40° = \frac{14.7 \times 144}{53.3 \times 500} = 0.079\#/\text{ft}^3$$

$$W_{air} = 0.079 \times 1.3 = 0.103\# \text{ air}$$

$$Q_{air} = .103 \times \frac{0.24 \times}{24 \text{ hrs.}} (75-40) = .04 \text{ BTU/Hr.}$$

Use Melcor CP 1.4-71-10

$$Q_{total} = 5.4 + .04 = 5.44 \text{ BTU/Hr.}$$

$I = 1.1$ amps $V = 3.4$ Volts

Power = $1.1 \times 3.4 = 3.74$ Watts

Power Supply Eff (η) = 0.64

$$\text{Power} = \frac{3.74}{0.64} = \underline{5.84 \text{Watts}}$$

3.5.8.1 Penalties

Electrical Penalty (Module Only) =

$$\frac{5.84}{1000} \times 7 \times 24 \times 1.514 = 1.49\#$$

Heat Rejection = $5.84 \times 3.41 + 5.44 = 25.3$ BTU/Hr.

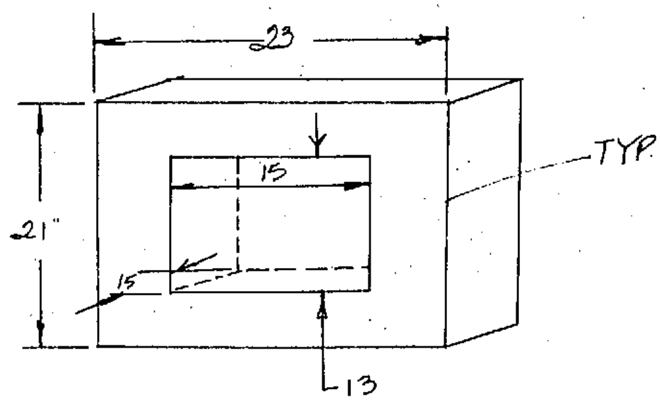
Air Heat Rejection Penalty =

$$5.84 \times 3.41 \times \frac{0.133\#}{\text{BTU/Hr.}} = \underline{2.64\#}$$

Liquid Loop Rejection Penalty =

$$25.3 \times 0.1 \frac{\#}{\text{BTU/Hr.}} = \underline{2.53\#}$$

3.5.8.2 Weight



$Vol._{out} = 23 \times 21 \times 21 - 10143 \ in^3$

$Vol_{ins} = 15 \times 13 \times 13 = \underline{2535} \ in^3$

$Vol._{insul} = 7608 \ in^3 = \underline{4.4 \ ft^3}$

$Wt. = 4.4 \times 3\#/ft^3 = \underline{13.2\#}$

Aluminum Shell Wt.

Aluminum Volume

Box - (3) 15 x 13 +(2) 13 x 13 + (3) 23 x 17 + (2) 21 x 17

Cover -(2) 23 x 21 +(2) 23 x 4 +(2) 21 x 4 =

$4128 \ in^2$ - Area of Alum. Sheet

Volume = 4128 x 0.03 thick = $124 \ in^3$

$\therefore$ Shell Wt. = 124 x 0.1 = $\underline{12.4\#}$

Wt. calculation based on T/E freezer design shown in Figure 13 and analysis in Sect. 3.3.5.1. The weight of the honeycomb structure and superinsulation are replaced by the weight of the aluminum shell and 4" of foam insulation (assume foam for rigidity).

Total Wt.

$\therefore \Delta Wt = \Sigma$ Al. Shell + 4" Insul. - Honeycomb +Super Insul.

= 12.4 + 13.2 - 18 + 1

= +6.6#

3.5.8.2 Cont'd

Hardware Wt. - 29 + 6.6 = 35.6#

Elect. Penalty = 1.5# (Sect. 3.5.8.1)

Heat Rej. Penalty = 2.64# (Sect. 3.5.8.1)

Fan Penalty = 3.80# (Sect. 3.5.6.2)

Total System Wt. = 35.6 + 1.5 + 2.64 + 3.80 =

43.54#

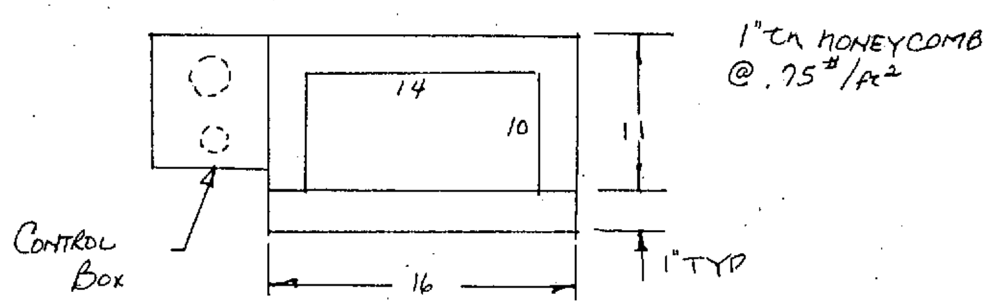
3.5.8.3 Volume

Refrigerator Volume with 4" conventional foam insulation

Vol. = 23 x 21 x 21 = 5.87 ft^3

3.5.9 Small Cavity Refrigerator

Assume a 14 x 10 x 9 Cavity size:



Cavity outer honeycomb area	=(3) 16 x 11 + 2(12 x 10)	=	768
Cavity inner " "	=(3) 14 x 10 +2(10 x 9)	=	600
Box-door contact area	=(2) 1 x 14	=	28
Door faces	=(2) 16 x 11	=	352
Door edges	=(2) 16 x 1 +(2) 11 x 1	=	54
Control Panel + tank enclosure (Sect. 3.2.8.1)		=	790
			2600 in^2

Area = $\frac{2600}{144}$ = 18.06 ft^2

Honeycomb Shell Wt. = 18.06 x 0.75 = 13.55#

3.5.9.1 Weight

Honeycomb Shell	-13.55
Super Insulation	- 0.7
Aluminum Shelves + Separators	- 1.7
Aluminum Liner + heat pipes	- 1.9
(2) T/E Modules	- 0.2
Heat Sink Core	- 0.1
Elect. Power Supply & Control	- 1.5
Ducts	- 0.1
Fan Motor	- 0.4
Control Panel	- 0.3
Mounts, supports, switches, etc.	- 0.5
Miscellaneous + Contingency	- 1.5
	22.45

Ratio Penalties in Sect. 3.5.7 by relationship of surface area of small to large refrigerator size.

$$\text{Ratio} = \frac{\text{Exposed surface A small refrig.}}{\text{Exposed surface A large refrig.}}$$

$$= 0.68$$

Elect Power Penalty = 1.07 x 0.68 = .73#

Heat Rej. Liq. Loop = 1.53 x 0.68 = 1.07#

Heat Rej. (cabin air) = 2.1 x 0.68 = [illegible]#

Fan Penalty = 3.80 x 0.68 = 2.58#

3.5.9.1 Cont'd

System Weight =

a) Air Heat Rejection

$W = 22.45 + 0.73 + 1.43 = \underline{24.61\#}$

b) Liquid Loop Heat Rejection

$W = 22.45 + 0.73 + 1.07 = \underline{24.25\#}$

c) Air Heat Rej. + Fan (if required)

$W = 24.61 + 2.58 = \underline{27.19\#}$

3.5.9.2 Volume

$V = 16 \times 12 \times 11 = 2112 = 1.22\ ft^3$

+ Control Panel .1

$= \underline{\underline{1.32\ ft^3}}$

FINAL REPORT
SPACE SHUTTLE/ FOOD SYSTEM STUDY
VOLUME II

APPENDIX D
PACKAGE and STOWAGE – ALTERNATE CONCEPTS ANALYSIS

prepared for

NATIONAL AERONAUTICS and SPACE ADMINISTRATION
Johnson Spacecraft Center
Houston, Texas 77058

Contract NAS9 - 13138

Prepared by

THE PILLSBURY CO.

PACKAGE AND STOWAGE

CONTENTS

C-2

CONTENTS (continued)

ILLUSTRATIONS

ILLUSTRATIONS CONT'D

1.0 INTRODUCTION

This study was concerned with developing packaging and vehicle stowage data, in terms of vehicle imposed weight and volume penalties. Certain assumptions were made for food packaging sizes based on a preliminary Shuttle menu generated by The Pillsbury Company. Utilizing the assumed packaging sizes, a series of stowage options were assessed to determine the impact on the Shuttle. The options were based on providing:

a) A fixed menu plan with no-choice in-flight for the crew or passengers

b) A single meal choice (dinner) of entrees and secondary meal components per day with the balance of the days meal fixed (no choice)

c) A two-meal choice (dinner and lunch) of entrees and secondary meal components per day with breakfast fixed (no choice)

d) A full choice of all the food on board throughout the mission

The above options were analyzed for a design concept consistent throughout each option in order to maintain a viable range of data. If the design concept is changed, it is possible that the absolute values may vary as to weight and volume; however, the additional penalties for each increasing complexity of choice should be valid as to percent increase of penalty to the vehicle.

In addition to the vehicle stowage penalties, a simple liner concept was analyzed for weight and volume without consideration of the vehicle requirements. In this case, food would be packaged in a no-choice configuration within the liner and stowed in a food locker in the galley. By only defining the food liner or package, the design of the vehicle interface is left open.

The assumptions made in the design analyses are as follows:

a) Mission time is 42 man-days (6 men - 7 days)
b) All food packages are of a 3" x 3.5" formed base and varying heights
c) The meals are packaged in a primary meal package (PMP) containing an entree and 2 side dishes (heatables) and a secondary meal pack (SMP) containing the meal R.T.E. (ready-to-eat) food
d) Separate beverage and snack packs are available for all meals at all times during the mission
e) Weights of all food stowage cabinets include a 10% contingency factor applicable for design variations and to provide a growth potential indication
f) Total system weights also include a 15% vehicle interface structure allowance.

A summary matrix of the analysis is presented in Table 1 for the range of meal choices considered, and compared for a 42 and 28 man-day mission. A graphic presentation of the results is shown in Figure 1 - Stowage Summary.

TABLE 1. FOOD STOWAGE SUMMARY

(Weights and Volume of Structure and Installation - But Less Food + Packaging)

42 VS 28 Man-Days

Meal Choice	Weight (lbs)		Volume			
			42 M-D		28 M-D	
	42 M-D	28 M-D	Ft^3	$In.^3/M\text{-}D$	Ft^3	$In.^3/M\text{-}D$
None	42.73	$29.5^{(1)}$	8.64	355.5	5.97	363.4
1-Meal	64.35	-	10.29	423.4	-	-
2-Meals	$\approx 77^{(2)}$	-	$\approx 10.7^{(2)}$	$\approx 440^{(2)}$	-	-
All Meals	99.28	$69.6^{(1)}$	10.81	444.8	7.57	467.2

(1) Wts. are extrapolated from wt/vol ratios obtained in 42 man-day evaluation

(2) Scaled from Figure 1

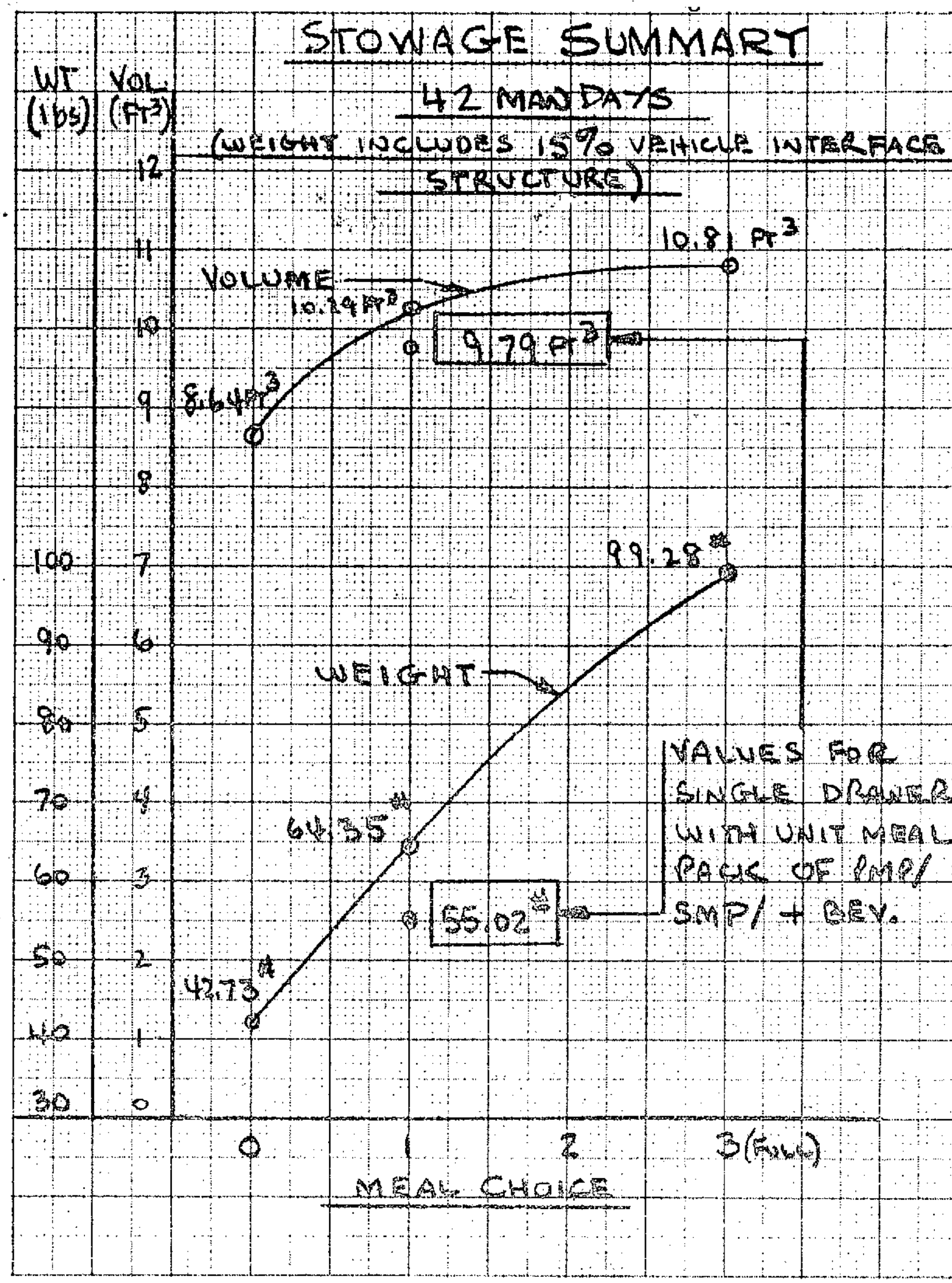


Figure 1 - Stowage Summary

2.0 DISCUSSION

2.1 Introduction

The approach used to satisfactorily complete the NASA contract requirements for analyzing food packaging and stowage methods was based on a determination of appropriate Shuttle menus, the development of package sizes to accommodate the food items, and a range of options for stowing the food aboard the Shuttle.

The vehicle penalties for weight and volume were calculated for each of the stowage options considered. The food stowage penalties represent installed weight and volume of the particular cabinet design for each of the stowage options, but exclude the food weight. Consideration was given to zero gravity operations, launch loads, vehicle installation arrangement for 1-g flight, ground servicing, maintainability and meal preparation, shipping and handling.

The following points must be recognized when reviewing this report:

* The intent of the analysis was to provide a comparative basis for assessing the various systems.

* A number of techniques and conditions were analyzed with the most obvious and logical variations considered within the allocated effort. Undoubtedly, additional variables could be conceived which could indefinitely extend the scope of the study.
* A fixed time and effort expenditure was allocated for this task which is only one element of the total program. The total stowage analysis effort was, therefore, scoped in magnitude and depth to be consistent with the balance of program tasks. For this commitment of effort the level to which each analysis was carried produced comparable data and results.
* The results of the analysis are valid and correct, and have been based on certain design, food, and system assumptions. While the actual values presented for weight and volume may be subject to discussion due to the assumptions made, the relative ratings will not be substantially affected. By altering the assumptions, the final penalties can be recalculated.

* The technical competency of the analysis and the confidence level of the results provides a reasonable basis for selecting a particular technique and recommending such a technique for shuttle use.

2.2 Options for On-Board Meal Selection

2.2.1 No Meal Choice

All menus are pre-selected, bulk packaged as crew meals, and stowed sequentially in mission day order. This technique permits a single module storage concept for yet to be eaten food and waste food and packaging.

The single unit food and waste storage module is 22.7 in. W x 22.7 in. D x 29.1 in. H in overall dimensions. Access at the upper face of the module permits withdrawal of one tray-like divider structure containing one six-man day supply of food divided into dinner, lunch, breakfast and bev/snack overwraps. See Figures 2 and 3 following.

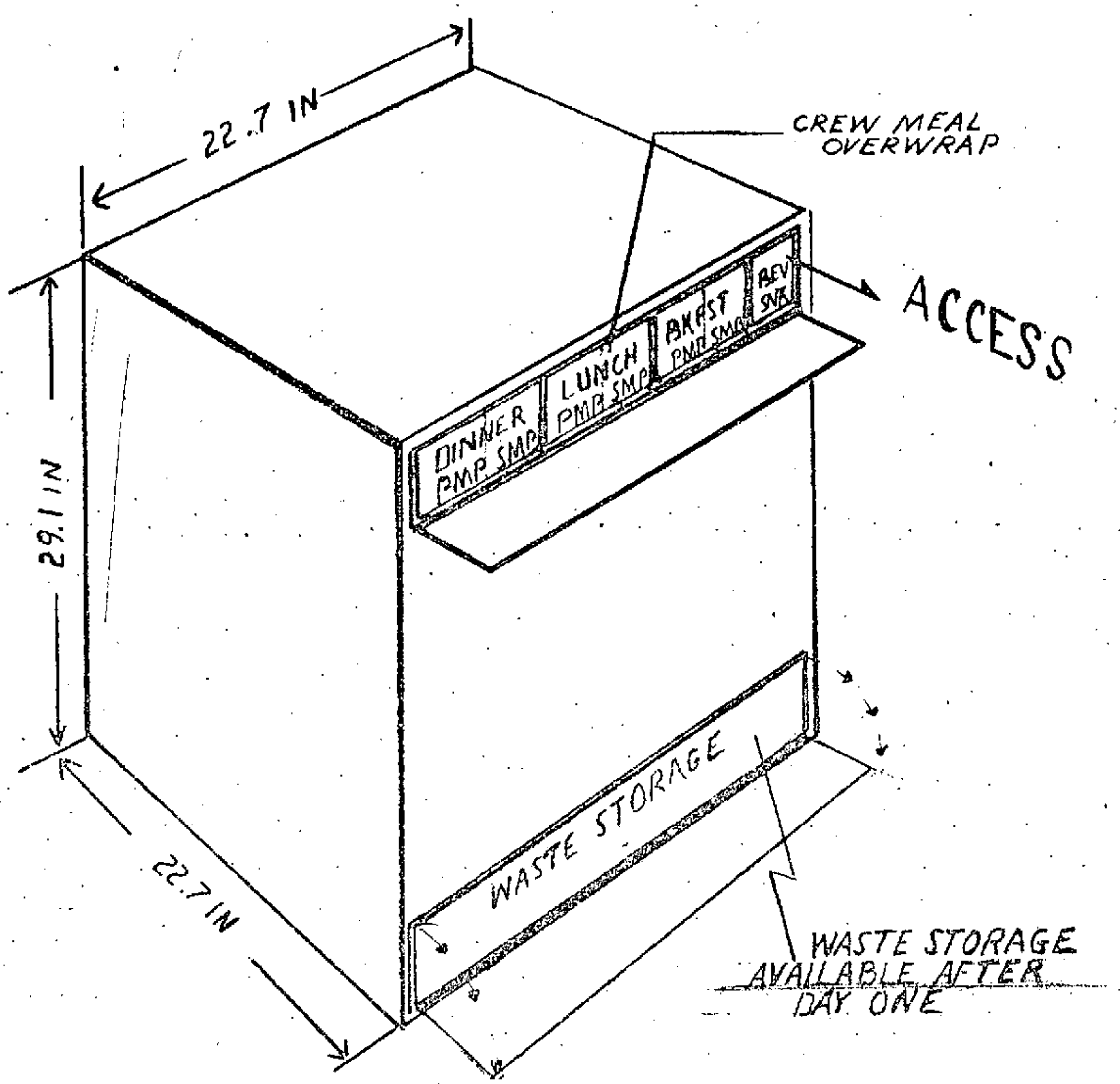


Figure 2. No Meal Choice Module

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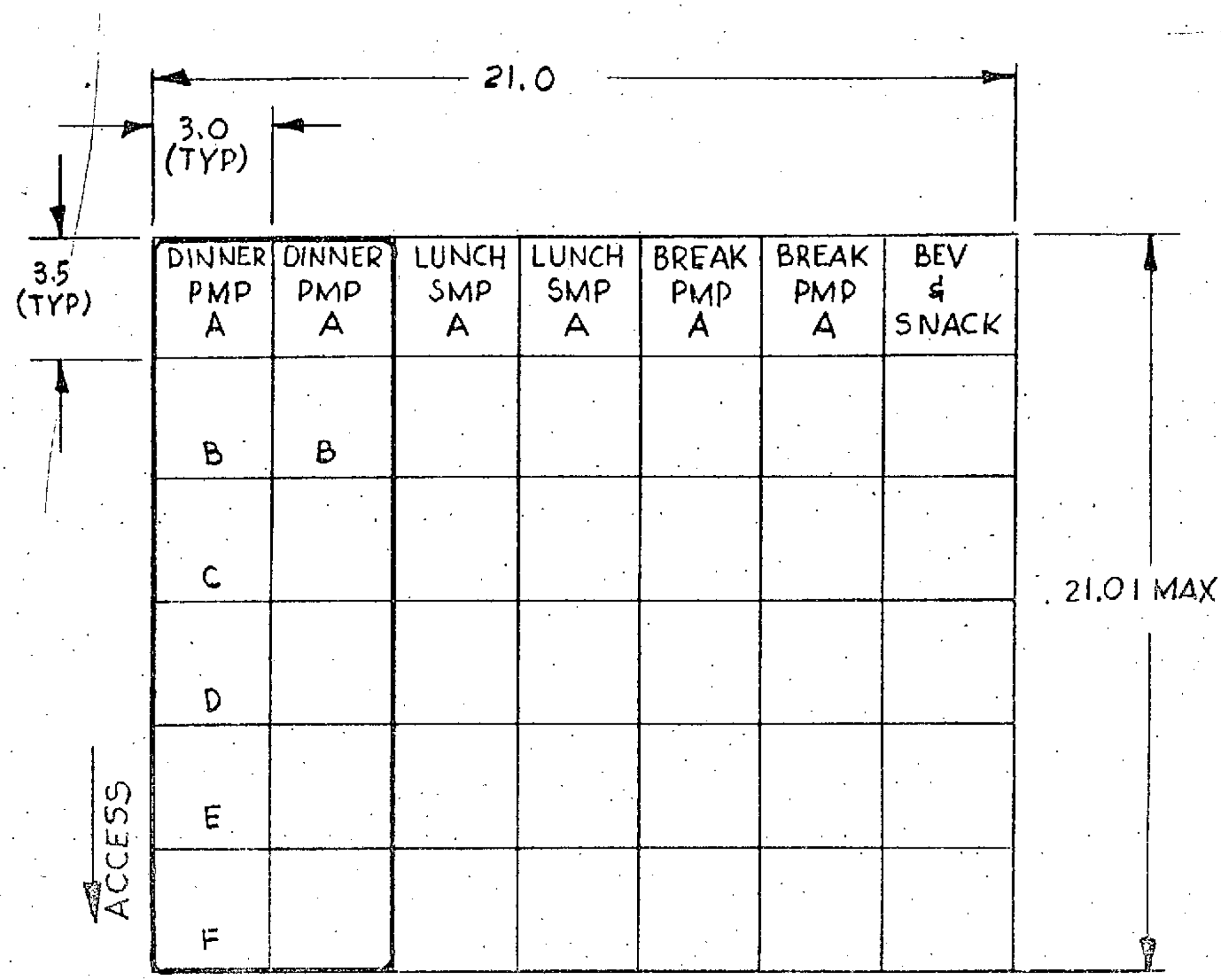


Figure 3. Overwrap Scheme

This figure shows a typical overwrap surrounding primary, PMP, and secondary, SMP, dinner meal packs for each of six crewmen A through F. The same concept applies to lunch, breakfast and snack meal packs. Dimensions are in inches on this top view.

Each tray-like divider containing one mission day menu stacks onto the next mission day menu down to and including the seventh day menu which is on the food support plate. The food support plate is attached to negator springs such that when the day-one menu is removed from the top of the module, all succeeding menu trays move up. This design permits storage of waste packaging and food in the vacated module space.

Volume requirements for this no meal choice module are 8.6 $ft.^3$ where W = 22.7 in., D = 22.7 in., and H = 29.1 in.

Weight penalty including 10% contingency and 15% (5.07 lb) vehicle interface structure if required is 42.23 lb.

In the case of the four man seven day mission the discussion of overwrap and mission day menus holds. Only the width dimension and module weight change. The resulting volume with the new 15.636 in. W is 5.97 $ft.^3$. The weight approximation is 29.5 lbs.

2.2.1.1 Liner Concept

A simple rigid liner concept was analyzed for crew interface weight and volume.

The food is packaged without overwrap in a no-choice configuration. It is nested in a "crew-day" layer arrangement, i.e., each layer is made up of a crew breakfast, lunch, dinner, and snack-beverage overwrap. There are no provisions for waste storage.

Volume requirements for this no meal choice food liner are 21.16 in. W x 21.13 in. D x 27.55 H with a resulting volume of $7.13 ft^3$.

Weight penalty without regard for contingency or vehicle interface structure is 12.5 lb.

2.2.2 One Meal Choice

The dinner entree and two side dishes are packaged as a unit (Primary Meal Pack) and stowed such that any remaining PMP unit is available for selection during the mission. The lunch and breakfast menus are overwrapped on a mission day basis as in the no meal choice system. This scheme permits:

* Flexibility in dinner menu selection
* Single module storage

* Limited food and package waste storage in same module

2.2.2.1 One Meal Choice - Single Drawer Concept

The one meal choice single drawer module is 23.3 in. W x 22.1 in. D x 31.0 in. H in overall dimension. Access at the top front of the module permits withdrawal of one tray-like divider structure containing preselected crew breakfast and lunch overwraps including beverage and at least one daily snack/beverage overwrap. The drawer access permits selection, by each crew member, his choice of one complete dinner menu. See Figure 4 following.

This figure shows the cutaway module as viewed from the front. The A, B, C, D, E, F letters designate crew member. Access for crew breakfast and lunch overwrap packs is at the upper level which will be emptied at breakfast of day 2. At that time the divider tray is removed and the internal negator springs will lift the complete food supply up one level such that luncheon 2 pack (L2) will be in view and ready for withdrawal.

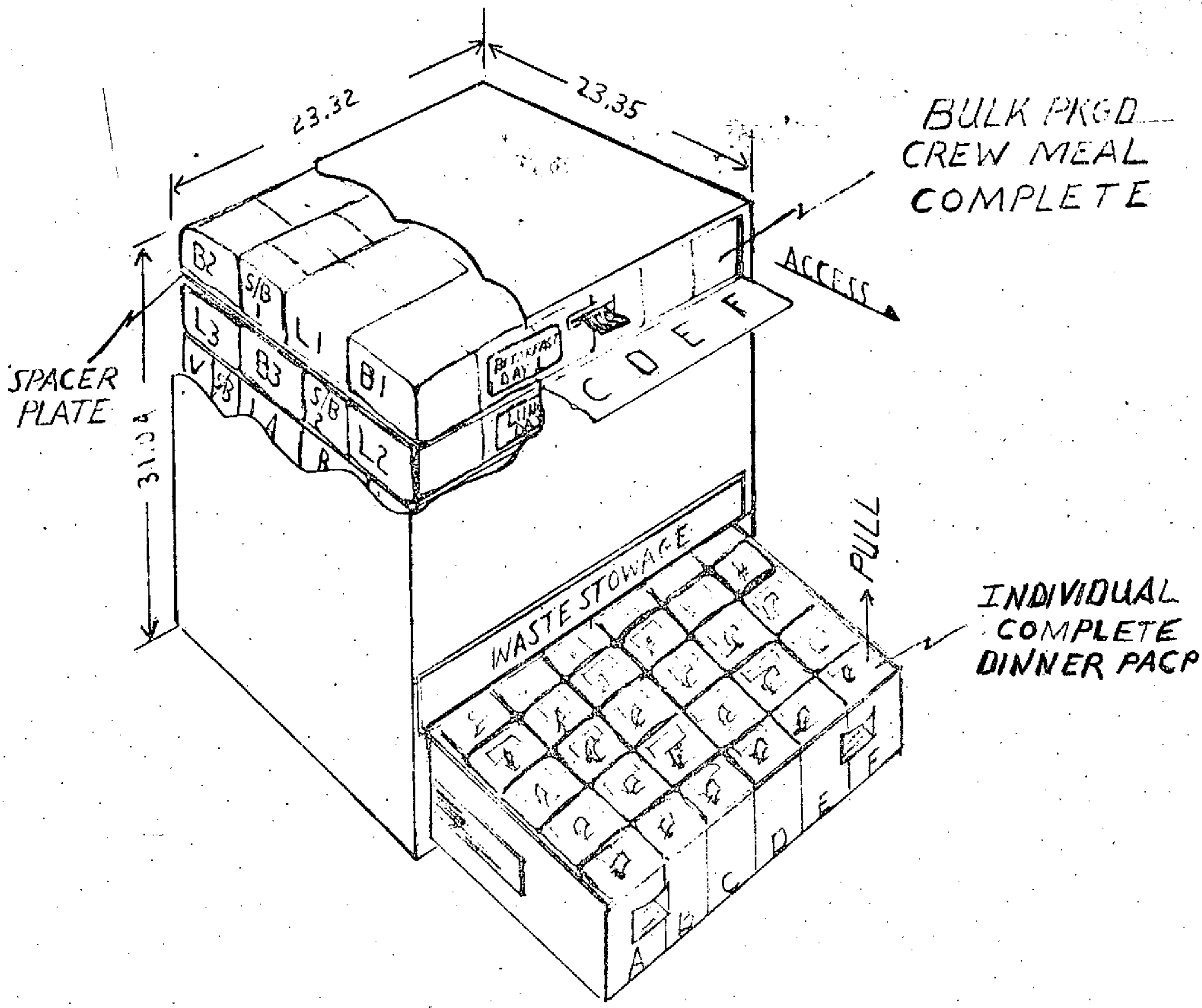


Figure 4

Single Drawer Module
One Meal Choice

Access for individual dinner is provided by the dinner drawer. This drawer pulls out to permit the crew member to select from his own file (A, B, C, etc.) one of the dinner meals remaining from the original seven. The supplied meal for him is complete with PMP, SMP, and BEV. Waste stowage space is available after breakfast of day two when the top level supply is exhausted and succeeding levels move up to allow vacant space below.

Volume requirements for this one drawer dinner meal choice system are 9.79 ft^3 where W = 23.4 in., D = 23.3 in., and H = 31.0 in.

Weight penalty including 10% contingency and 15% (6.52 lb) vehicle interface structure if required is 55.02 lb. Extrapolating the case of the four man seven day mission the discussion of overwrap and mission day meals menus will hold. The resulting volume with the new 16.35 in. W is 6.85 ft^3. The weight approximation is 39.14 lb.

2.2.2.2 One Meal Choice - Double Drawer Concept

As in the one meal choice, single drawer concept discussed in section 2.2.2, this system provides for the stacked storage of breakfast, lunch, and snack/beverage bulk mission day packages.

The dinner entree and secondary meal pack are packaged each in a separate drawer. This system permits added flexibility of menu selection in that the crew member can choose any remaining dinner entree of meat and two vegetables and accompany that with the appetizer, bread, dessert and beverage pack of his choice.

The impact of this double drawer system on the weight and volume penalties is shown in the following table comparing the two one meal choice modules.

TABLE 2. WEIGHT AND VOLUME COMPARISON SINGLE AND DOUBLE DRAWER MODULES

	Single Drawer	Double Drawer
Width	23.326 in.	23.326 in.
Depth	22.023 in.	22.023 in.
Height	31.039 in.	32.649 in.
Volume	9.79 ft^3	10.29 ft^3
Weight *	55.02 lb	64.35 lb

* 10% contingency plus 15% vehicle interface structure

These data reflect the requirements of the six man-seven day, forty two man-day mission. In the case of the four man-seven day, twenty eight man-day mission only the module width will change to reflect the decrease in crew members (see fig 3).

2.2.3 Two Meal Choice Module

No detailed analysis has been done for this configuration. Based upon previous discussion of the one meal choice single and double drawer system, the two meal choice system could provide for the following alternatives:

* Choice of lunch and supper
* Choice of breakfast and lunch
* Choice of breakfast and supper

Each of these concept systems can be accomplished utilizing a module of two or four drawers to allow individual meal selection. The bulk packaged meals and snack/beverage packages would be stowed as in the single drawer module shown in Figure 4. Volume and weight penalties for the two meal choice system are 10.7 ft^3 and 77 lb respectively. These data are not as a result of detailed analysis but rather from scaling Figure 1 presented in section 1 above.

2.2.4 Full Choice Module

All three meals are free choice to each crew member so that he can select any remaining combination of breakfast, lunch, dinner and snack/beverage up to the last day of the mission. The system is accomplished utilizing a module of 23.3 in. W x 22.0 in. D x 36.4 in. H equipped with eight pull out drawers.

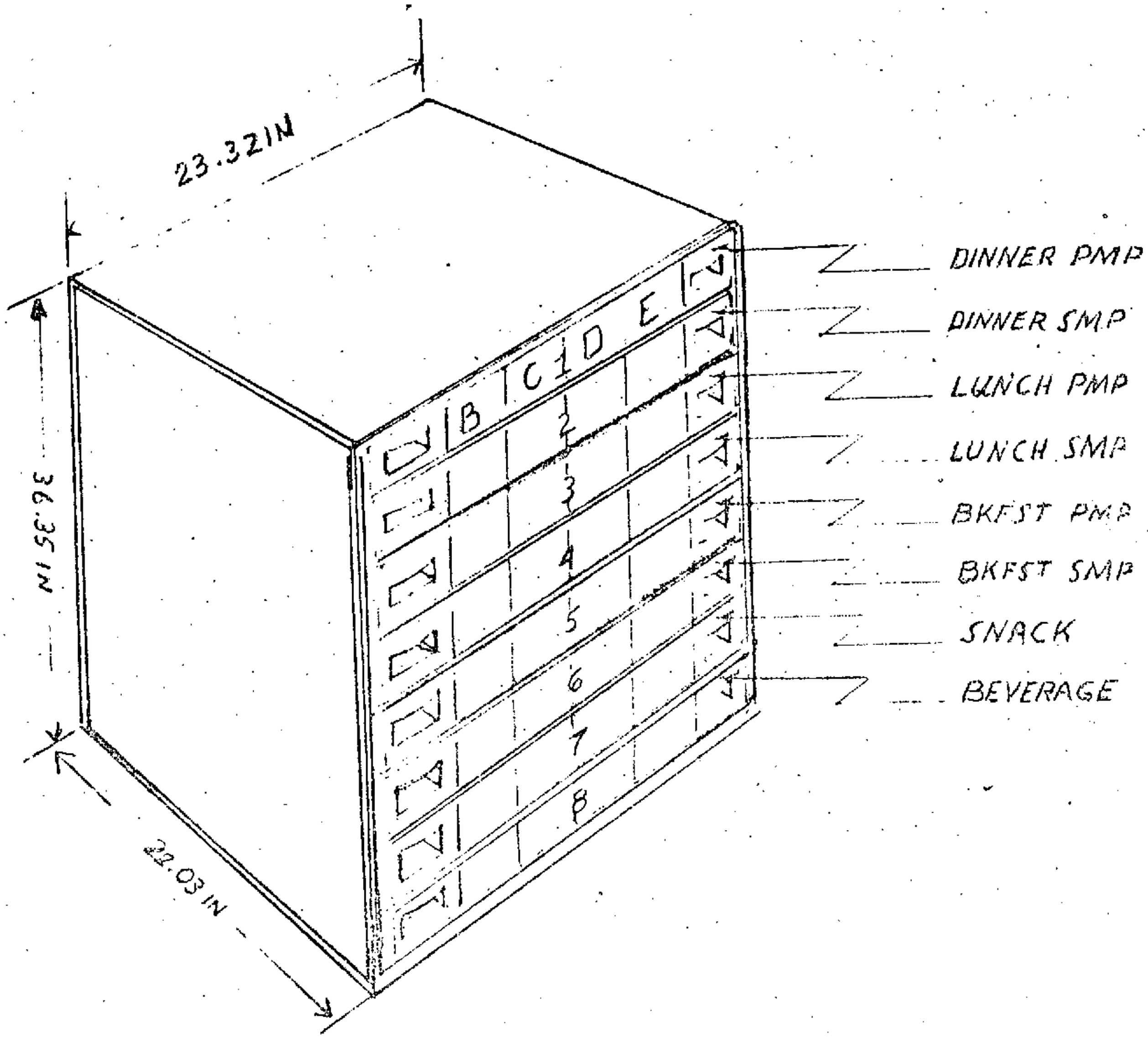


Figure 5. Full Meal Choice Module

At the breakfast meal, day one, the crewman selects one packet each from his file (A, B, C, etc.) in drawers five and six. This selection comprises a complete breakfast including beverage. At lunch and dinner the crewman merely repeats the selection process at drawers designated for those meals. Snacks and extra beverages are available ad-lib from drawers seven and eight, respectively.

Volume requirements for this eight drawer module — full meal choice meal system are 10.8 ft^3.

Weight penalties associated with this system including 10% contingency factor and 15% vehicle structural interface if required are 99.28 lb.

Extrapolating the case for the four man seven day mission the discussion of the eight level module and individual menu selection will hold, only the drawer width will change. The resulting volume with the new 16.33 in. W is 7.57 ft^3. The weight approximation is 69.6 lb.

3.0 Detailed Analysis

3.1 Packaging Assumptions

For purposes of the following preliminary study all packages are 3 in. x 3.5 in. in cross section.

Entrees - use 12 in.^{3} loose fill of food/pkg. For vacuum pack use 60% reduction.

$$12 \text{ in}^3/\text{pkg.} \times .60 = 7.2 \text{ in}^3/\text{pkg.}$$

For package stowage and fill efficiency assume 65%

$$7.2 \text{ in}^3/\text{pkg} \times 1.65 = 11.88 \text{ in}^3/\text{pkg.}$$

$$\left.\begin{array}{l}\text{Vegetables}\\ \text{Side Dishes}\\ \text{Soups}\\ \text{Snacks}\end{array}\right\} \text{- use same size as entree}$$

Beverage use 5 in^3/pkg (assumes packaging and stowage inefficiencies for beverage pack.

Entree H dimension

Using the 3.0 x 3.5 in. cross section constraint and 11.88 in^3/pkg. volume requirement minimum H dimension is

$$3.0 \text{ in} \times 3.5 \text{ in} \times (H)\text{in.} = 11.88 \text{ in}^3$$

$$(H)\text{in.} = \frac{11.88 \text{ in}^3}{10.50 \text{ in}^2}$$

$$(H) \text{ in.} = 1.13 \text{ in.}$$

Allowing 15% contingency in H

$$H = 1.13 \text{ in.} \times 1.15 = 1.2995 \text{ in.}$$
$$H = 1.3 \text{ in.}$$

Beverage use 5 in^3/container (assumes packaging and stowage inefficiencies for beverage pack)

Using the 3.0 in. and 3.5 in. cross section constraint and the 5 in^3/pkg. volume requirement minimum H dimension is

$$3.0 \text{ in.} \times 3.5 \text{ in.} \times (H) \text{ in.} = 5.0 \text{ in}^3$$

$$(H) \text{ in.} = \frac{5.0 \text{ in}^3}{10.50 \text{ in}^3}$$

$$(H) \text{ in.} = .476$$

$$\text{to the nearest tenth } H = .5 \text{ in.}$$

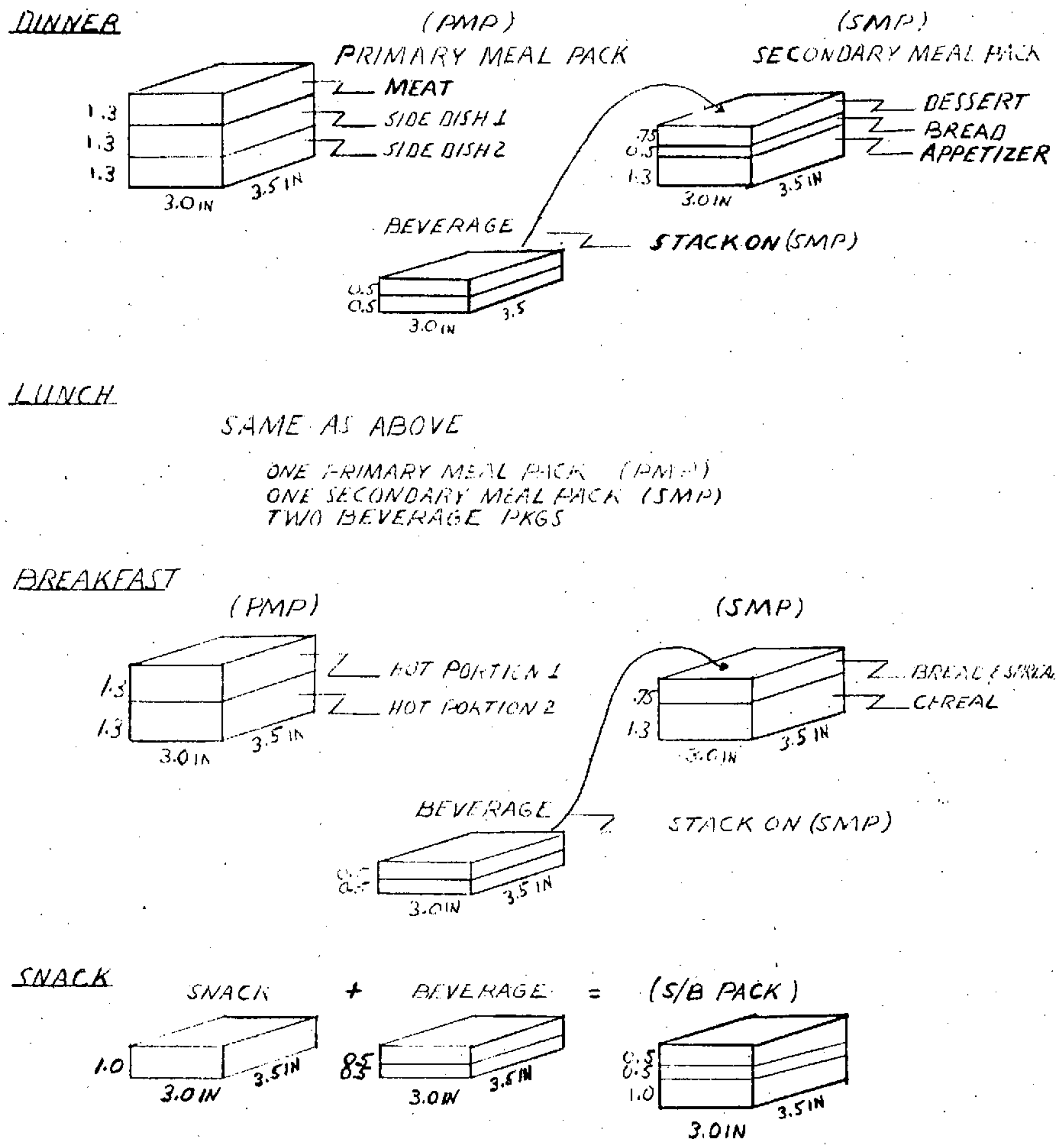


FIGURE 6 - PACKAGE SIZING AND MEAL GROUPING

3.1 Cont'd

3.1.2
Packaging Volumetric Summary

Allow for presently undefined valve on rehydratable packages. Assume old value.

		W	D	H	IN^3
Dinner	Main Course (PMP)	3.0 x	3.5 x	3.9	= 40.95
	appetizer, bread, dessert & beverage (SMP)	3.0 x	3.5 x	3.55	= 37.28
Lunch	Same as above				= 40.95
					= 36.00
Breakfast	Hot portions 1 & 2 (PMP)	3.0 x	3.5 x	2.55	= 26.78
	Bread & spread, cereal and beverage	3.0 x	3.5 x	2.85	= 29.92
Snack	Snack, Beverage – S/B pack	3.0 x	3.5 x	2.0	= 21.00
					232.88

$$\frac{232.88\ in^3}{\text{Man Day}} \times 42 \text{ man days} \times \frac{1\ ft^3}{1728\ in^3} = 5.66\ ft^3$$

3.2 Options for Onboard Meat Selection

No choice - Ref 3.2.1 and 3.3.1 - all food pre-selected, packaged, and stowed in shuttle in sequence of predetermined consumption.

One meal choice - Ref 3.2.2 & 3.2.3 - Main meal (dinner) entree and two side dishes are packaged as a unit such that any remaining unit is available for selection. All other meals are stowed in sequence of predetermined consumption.

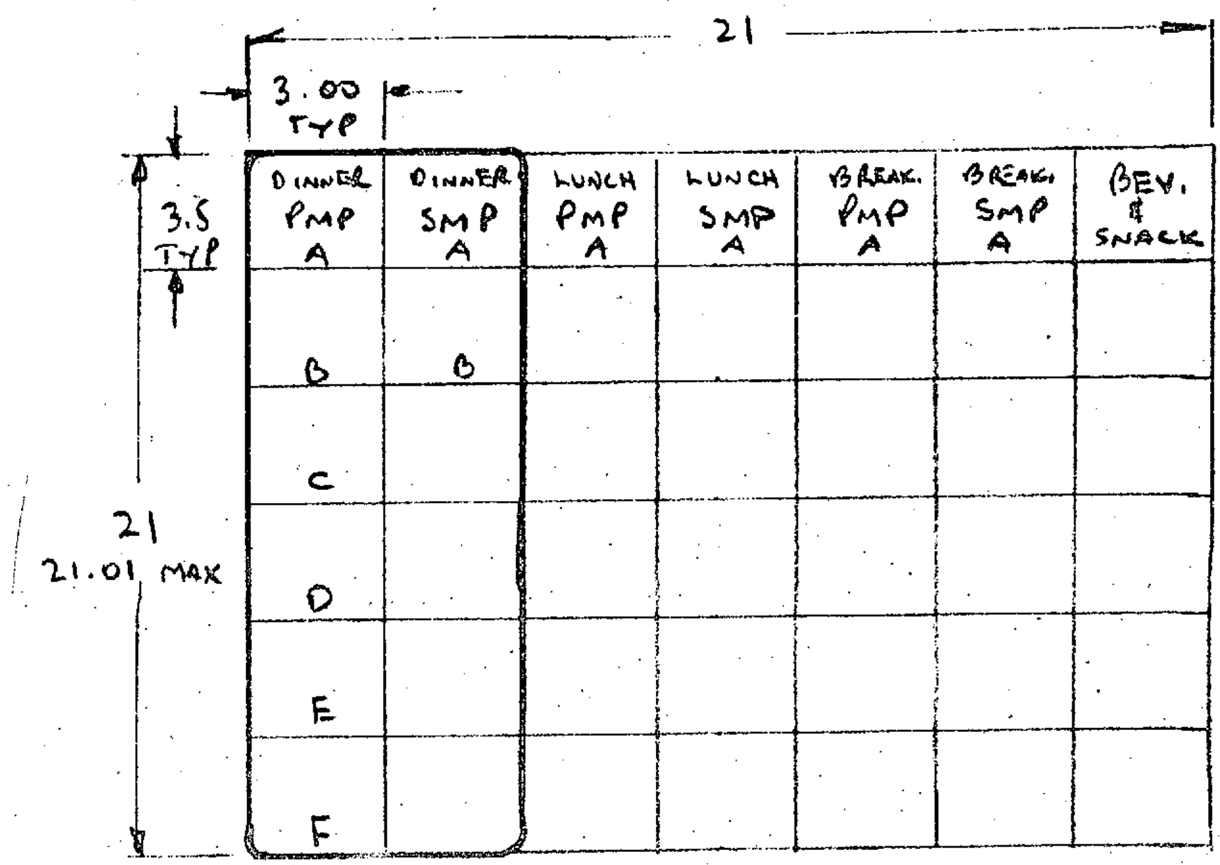
Two meal choice - Ref. 3.2.4 - Main meal pack and lunch pack units are available for selection during the mission.

Three meal choice - Ref. 3.2.5 - All three meals are free choice to each crewman so that he can select any remaining combination of three meals during each successive mission day.

3.2 Cont'd

3.2.1 No Meal Choice System

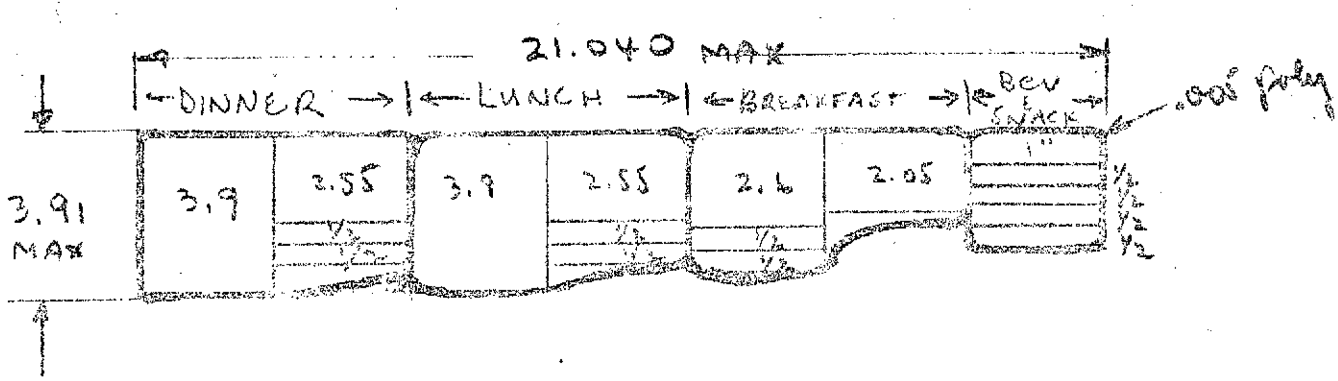
3.2.1.1 Packaging Plan



PLAN VIEW

Overwrap each set of PMP and SMP (incl. beverage for that meal) for all crewman.

One layer = 1 day supply for 6 men (6 man days)



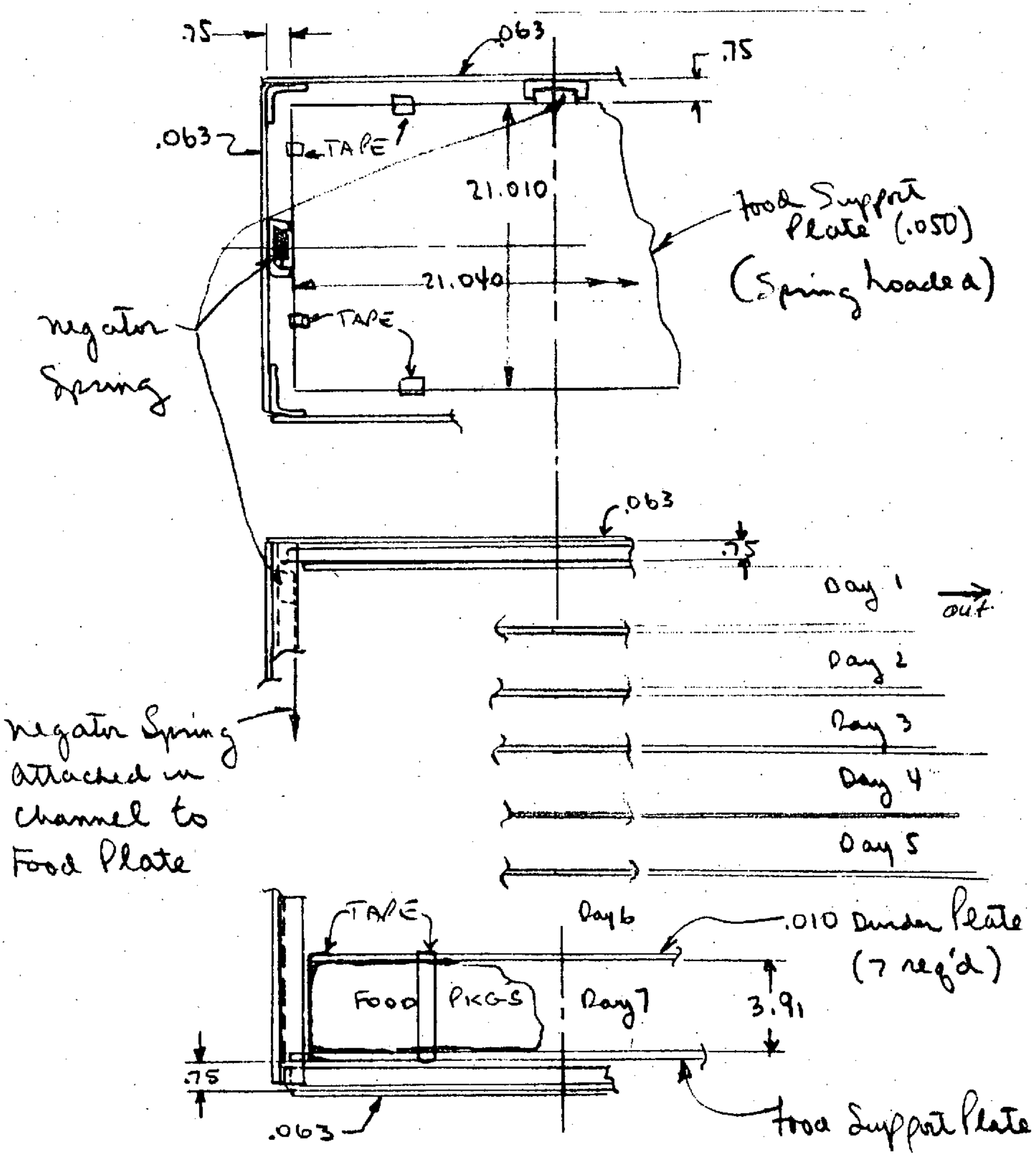


FIGURE 7 — NO CHOICE SYSTEM MODULE DETAIL

3.2 Cont'd

3.2.1.2 Cont'd

Volume = 21.16 x 21.13 x 27.55 = 12317.902 = 7.13 ft^3

Liner Wt. - Bottom + Top - 2 [21.16 x 21.13] = 894.22

Sides - 2[21.13 x 27.43] =1159.192

Front + back - 2[21.04 x 27.43] =1154.252

3207.668

Wt = 3207.668 (.06) (.065) = 12.5#

Module Volume

W = 21.010 + 2 (.75) + 2 (.063) = 22.636

D = 21.040 + 2 (.75) + 2 (.063) = 22.666

H = (3.91)7 + 7(.010) + .050 + 2(.75) + 2(.063) = 29.116

V = 14938.47 in^3

V = 8.64 ft^3

3.2.1.3 Preliminary Weight Analysis

No Choice System

Cabinet Sheets (t = .040)

2 side panels (22.666 x 29.116) 2 = 1319.88

1 back panel (22.636 x 29.116) = 659.07

1 front panel (22.636 x 29.116) = 659.07
(incl. access. door)

2 end panels $(22.666 \times 22.636)^2$ = 1026.14

3664.16

Wt = (3664)(.040)(0.10) = 14.66#

3.2 Cont'd

3.2.1.3 Cont'd

Cabinet Food Storage

Support Plate (21.010 x 21.040 x .050) = 22.103

Divider Plate (21.010 x 21.040 x .010)7 = 30.944

53.047

Velcro Tape Strips (5 x .5 x .06)

8/layer x 7 layers = 56 req

56 (.150)(.04) = .336#

Wt = (53.047) .10 = 5.30

+ .34

= 5.64#

Support Structure

4 Corner Supports .090

Section Area = 1.72 in^2 x 29.116

4 (.172) (29.116) = 20.03

8 Corner Closure Angles .050

Section Area = .098 in^2

Length = 22.6 - 2(.063) - 2(.09) = 22.3

8 (.098) (22.3) = 17.48

4 Guide Channels .75 .090

Section Area = .209 in^2

Length = 29.116 - 2(.063) = 28.99

4 (.209) (28.99) = 24.24

8 Intercortals .75 .75 .050

Section Area = .073 in^2

Av. Length = 22.6 - 2(.063) - 2(.09) = 22.3

8 (.073) (22.3) = 13.02

Negative Springs (4 Reqd) & Mounts

Ust total weight = 6#

Wt = (20.03 + 17.48 + 24.24 + 13.02) 0.10 = 7.48 + 6

= 13.48#

3.2.1.3 Cont'd

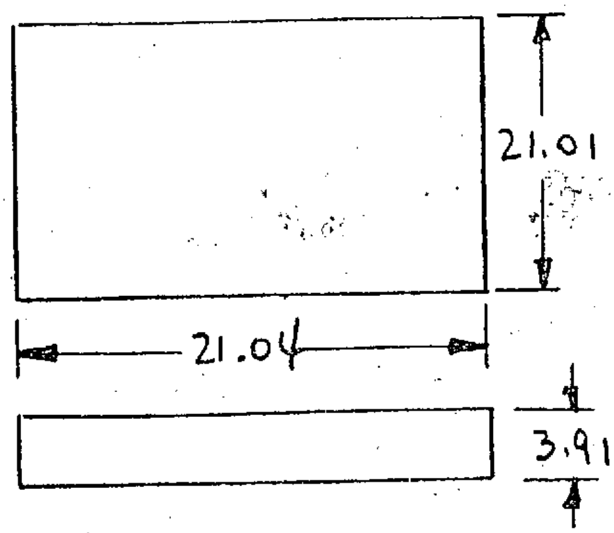
<u>Total Weight</u>

Storage Module for no choice inflight menu

Module Outer Shell	- 14.66
Storage Shelf & Dividers	- 5.64
Support Structure	- 13.48
Storage Module	- 33.78
10% Contingency	- 3.38
25% Vehicle Interface Structure	- 8.45
	45.61#

Storage Module	33.78
10% Contingency	3.38
15% Vehicle Interface Structure	5.07
	42.23#

3.2.1.4 Food Liner Concept Analysis



Basic arrangement of

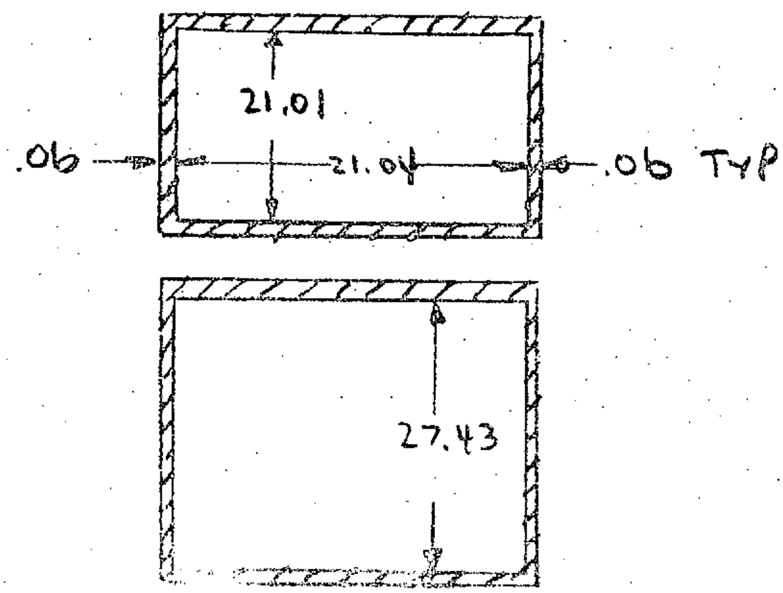
1 layer = 1 day supply

for 6 men per page 12

Package Height = 7(3.91) + 6 (.01) = 27.43"

spacing between layers

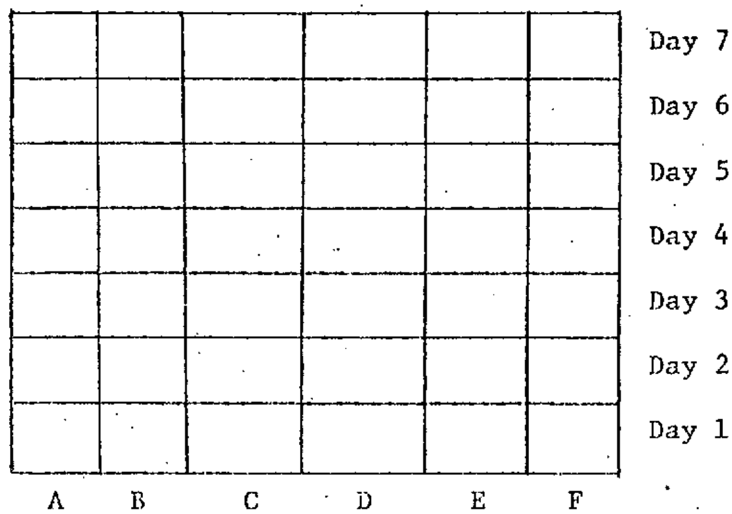
Liner Volume - Assume .06 th fiberglass @ .065#/in^3



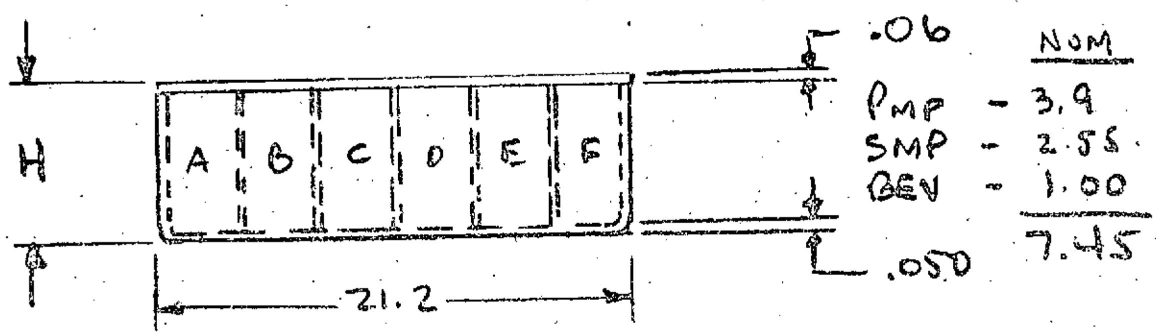
3.2.2 One Meal Choice Single Drawer System

3.2.2.1 Packaging Plan.

Use Single Drawer for Dinner PMP & SMP & Bev.



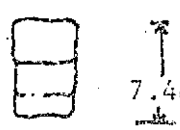
Plan View



3.9 + .01

2.55 + .01

1.00 + .01

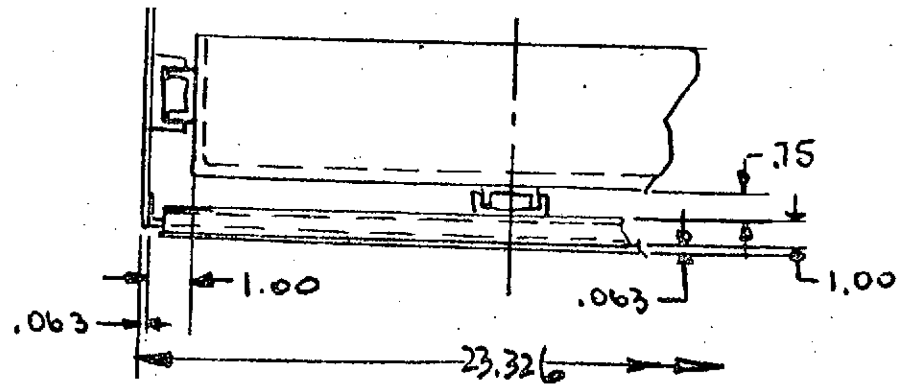


Overwrap each PMP, SMP & Bev.- use .005 poly

H = 7.48 + .06 + .050 = 7.59

Figure 8 Single Drawer Packaging Scheme

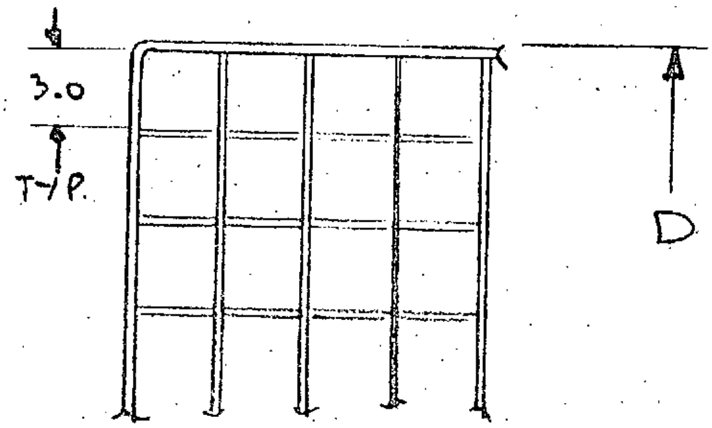
3.2.2.1 Cont'd



Cabinet Width = 21.2 + 2(1.00 + .063) = 23.326

Figure 9 Typical Drawer Installation

Cabinet Depth

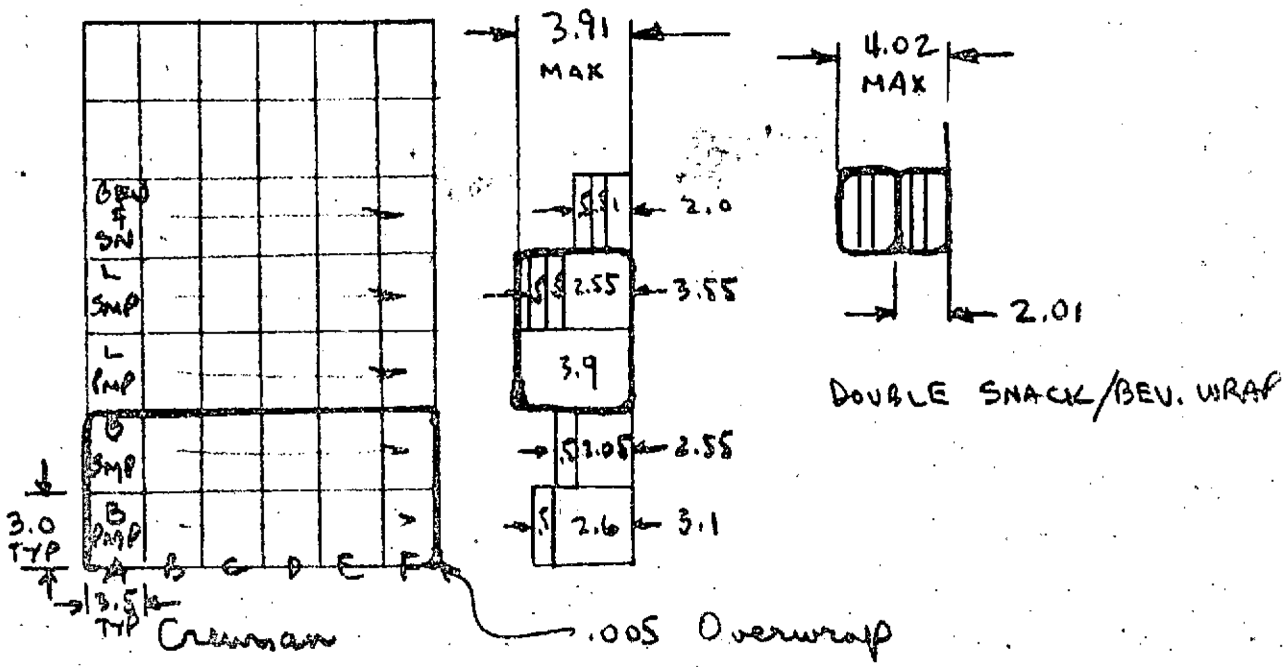


D = 7(3.0) + 6(.020) + 2(.050) = 21.22

End Closure = 0.813

Cabinet Depth = 21.22 + 0.813 = 22.033 min.

Use Layer approach for fixed part of meal

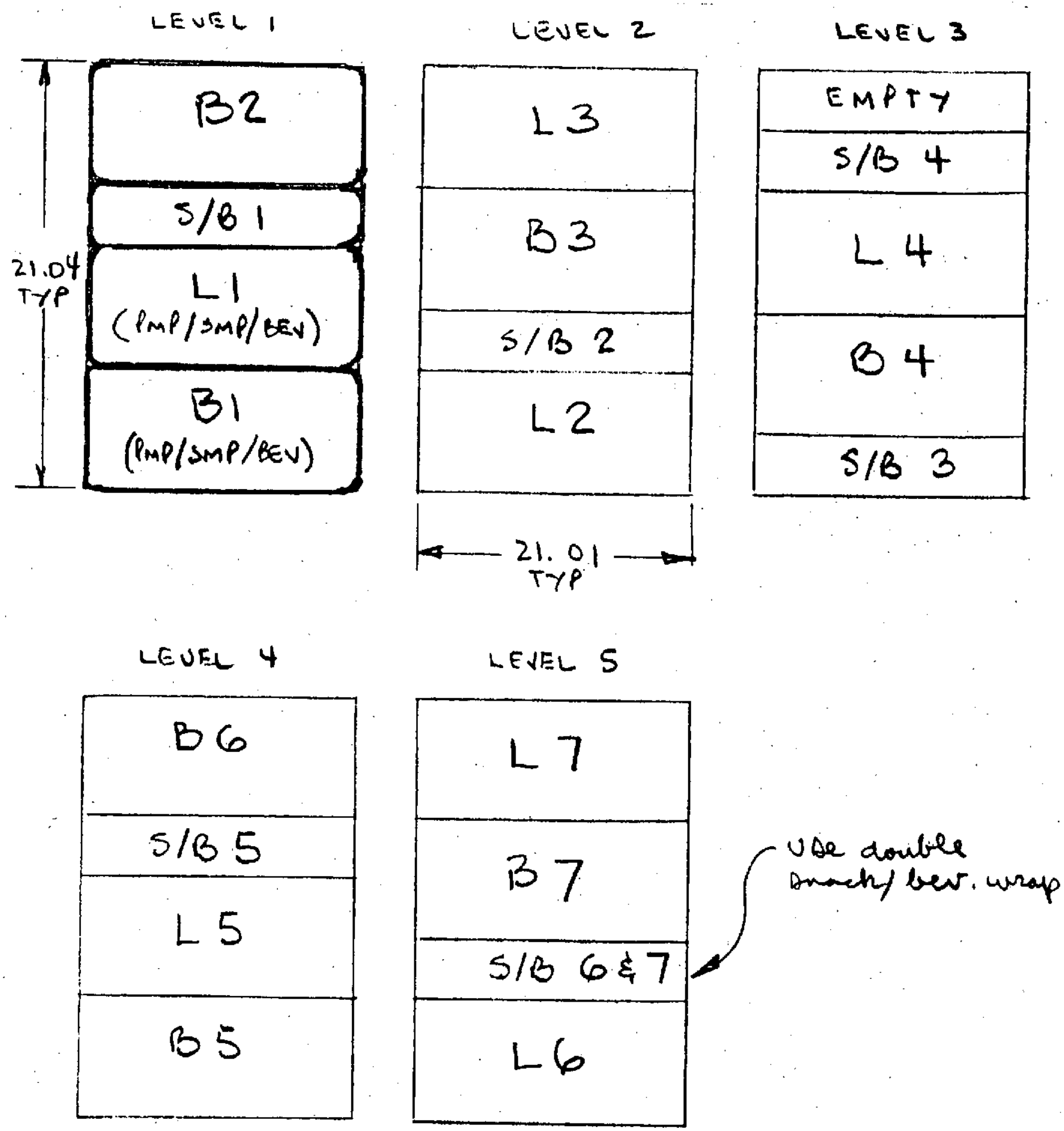


Note: 1. Overwrap B (Breakfast) SMP and PMP to form complete breakfast menu for 6 men for 1 day (incl. bev. for that meal)

2. Same for L (Lunch) SMP and PMP

3. Overwrap 1 days supply for 6 men of beverage balance (2bev.) + snack.

Stowage Arrangement - Single Drawer System



S/B -Snack & Beverage	Bev - Beverage	Days
B - Breakfast	PMP Primary Meal Pack	1, 2, 3 etc.
L - Lunch	SMP - Secondary Meal Plan	

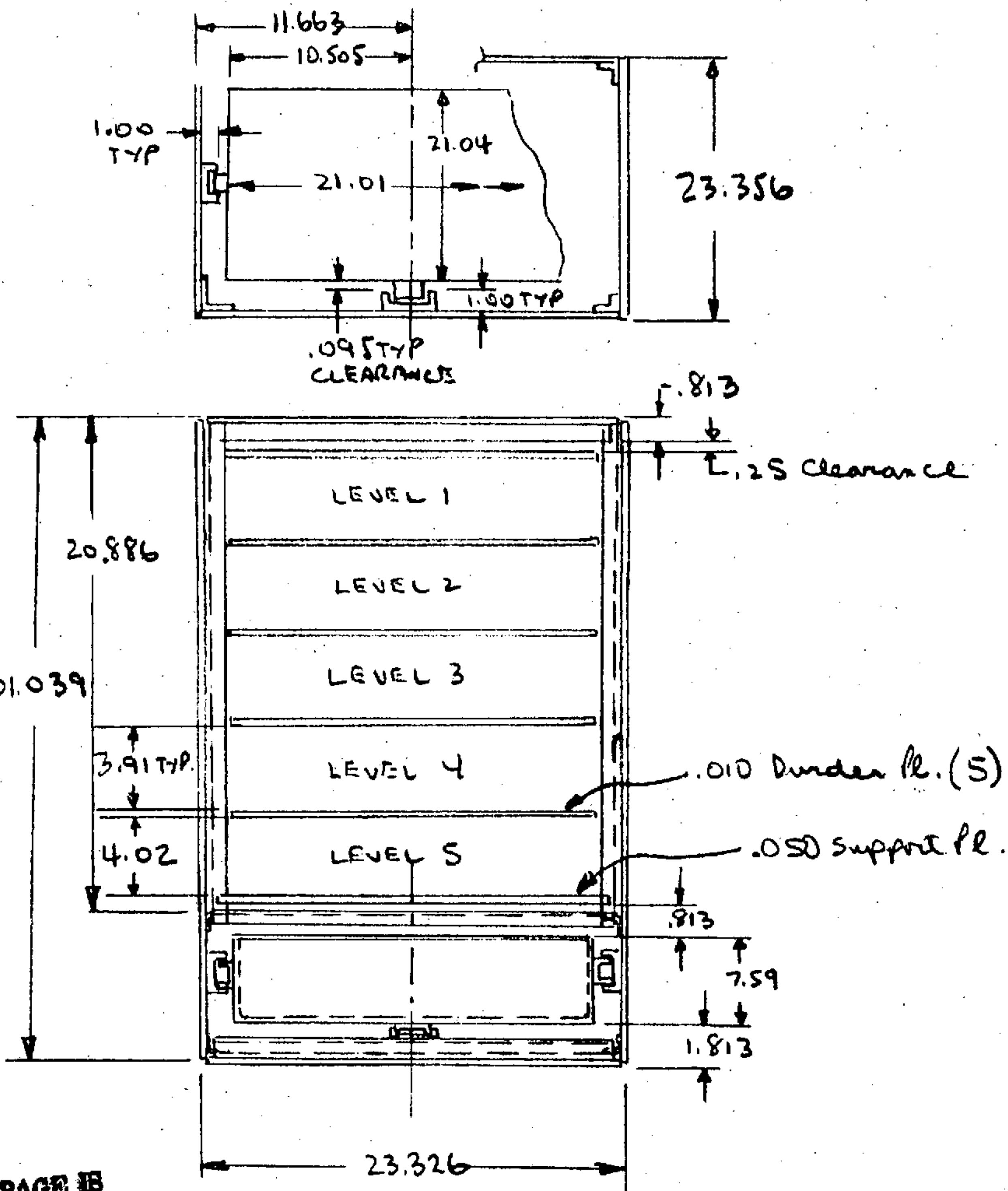


FIGURE 10 - DESIGN DETAIL SINGLE DRAWER MODULE

3.2.2.2 Cont'd

Cabinet Height

Drawer Installation	- 1.813
	7.590
	.813
Support Plate	- .050
Level 5	- 4.020
Levels 4, 3, 2, 1 (4 x 3, 91)	-15.640
Divider Plates (.010 x 5)	- .050
Top Clearance & Instl.	- .813
	.250
	31.039

Cabinet Volume

$$23.356 \times 23.326 \times 31.039 = 16910.111$$

$$= 9.79 \text{ ft}^3$$

3.2.2.3 Preliminary Weight Analysis- 1 Meal Choice - 1 Drawer

Cabinet Sheets (t = .040)

2 Side Panels	(23.356 x 31.034)2	= 1449.894
1 Back panel	(23.326 x 31.039)	= 724.016
2 End Panels	$(23.356 \times 23.326)^2$	= 1089.604
1 Front panel (assumes cutout for drawer & access door for balance of food)	(23.326 x 20.823)	= 485.717
		3749.231

Wt = (.040) 3749.231 (.10) = 14.997#

3.2.2.3 Cont'd

Cabinet Drawer

Drawer Base	= (21.22 x 21.2 x .050)	= 22.493
2 Sides	=2(21.22 x 7.48 x .050)	= 15.873
2 Sides	=2(21.1 x 7.48 x .050)	= 15.783
5 Dividers	=5(21.12 x 7.48 x .010)	= 7.899
Removable Dividers	=6(21.05 x 7.48 x .010)	= 9.447
0-g cover/restraint	=(21.2 x 21.22 x.020)	= 8.997
		80.492
Drawer Wt.	= (80.492)0.10	= 8.05#

Fixed Stowage Area

Support Plate	(21.01 x 21.04 x .050)	= 22.103
Divider Plates	(21.01 x 21.04 x .010)5	= 22.103
		44.206

Valero Strip Tapes (5 x .5 x 06)
8/layer x 5 layers = 40

Wt = 40(.150) (.04) = .24#

Wt = (44.206)(.10) = 4.42#

+ .24

4.66#

Support Structure

2 Drawer Guide Channels .75 x .090

Section Area = .209 in^2 x 23.356 lg.
2(.209)(23.356) = 9.763 in^3

1 Drawer Center Guide 3/4 x .090

Section Area = .209 in^2 x 23.356 lg.
1(.209) 23.356 = 4.881 in^3

3 Drawer Mounted Guides .75 78 x.090

Section Area = .13 in^2 x 23.356 lg.
3(.13) 23.356 = 9.109 in^3

4 Corner Closure Angles x .050

Section Area = .098 in^2 x 23.17 lg
4(.098) 23.17 = 9.083 in^3

3.2.2.3 Cont'd

4 Corner Closure Angles x .050

Section Area = .098 in^2 x 23.14 lg
4(.098) 23.14 = 9.071 in^3

6 Intercostal Supports x .050

Section Area = .073 in^2 x 23.17 lg
6(.073) 23.17 = 10.148 in^3

6 Intercostal Supports x .050

Section Area = .073 x 23.14 lg
6(.073) (23.14) = 10.135 in^3

4 Corner Supports x .090

Section Area = .172 in^2 x 20.886 lg
4(.172) 20.886 = 14.369 in^3

4 Guide Channels x .090

Section Area = .254 in^2 x 20.886 lg
4(.254) 20.886 = 21.220 in^3

Negator Springs & Mounts (4 req'd)

Use Total weight = 6#

Support Structure Wt. = (9.763 + 4.881 + 9.109

+9.083 + 9.071 + 10.148 + 10.135 + 14.369

+ 21.220) 0.10 + 6 = <u>15.78#</u>

3.2.2.3 Cont'd

Total Weight

Storage Module for 1-Meal Choice

Module Outer Shell	-	15.00
Storage Drawer + Shelves	-	12.71
Support Structure	-	15.78
Storage Module		43.49
10% Contingency		4.35
25% Vehicle Interface Structure	-	10.87
		58.71#

Food Stowage Structure + 10% Contingency	-	47.84
15% Vehicle Interface Structure	-	7.18
		55.02#

3.2.3 One Meal Choice Double Drawer System

3.2.3.1 Packaging Plan

Use 2 Drawers for PMP & SMP + Bev.

Drawer 1 - PMP only - 3.91 in/package

Drawer 2 - SMP + Bev -3.57 in/package

$H_{Drawer\ 1} = 3.91 + .06 + .050 = 4.02$ in.

$H_{Drawer\ 2} = 3.57 + .06 + .050 = 3.68$ in.

3.2.3.2 Module Dimensions and Volume

Cabinet Width (Ref 3.2.2.1) = 23.326 in.

Cabinet Depth (Ref 3.2.2.2) = 23.356 in.

Cabinet Height (inches)

Drawer Installation	-	1.813
H_{D1}	-	4.020
Drawer Spacing	-	1.500
H_{D2}	-	3.680
Clearance to support	-	0.813
Support Plate	-	0.050
Level 5 ht.	-	4.020
Levels 4,3,2,1 (4 x 3.91)	-	15.640
Divider Plates (5 x .010)	-	0.050
Top clearance & installation	-	0.813
		0.250
	H =	32.649

Cabinet Volume

23.326W x 23.356D x 32.649H = 17787.242 in

= 10.29 ft^3

3.2.3.3 Preliminary Weight Analysis

1 Meal Choice - 2 Drawers

Cabinet Sheets (t = .040)

2 Side panels	(23.356 x 32.649)2	=1525.100
1 Back panel	(23.326 x 32.649)	= 761.571
2 End panels	(23.326 x 23.356)2	=1089.604
1 Front Panel	(23.326 x 20.823)	= 485.717
(assumes cutouts for drawers & access door for balance of food)		
		3861.992

Wt = 3861.992 (.040) (.10) = 15.45#

Cabinet Drawers

Drawer 1 - PMP only

Drawer Base	= (21.22 x 21.2 x .050)	=22.493
2 Sides	=2(21.22 x 3.91 x .050)	= 8.297
2 Sides	=2(21.1 x 3.91 x .050)	= 8.250
5 Dividers	=5(21.12 x 3.91 x .010)	= 4.129
Removable Div.	=6(21.05 x 3.01 x .010)	= 4.938
0 g Cover/	= (21.2 x 21.22 x .020)	= 8.997
		57.104

Wt = 57.104 (.10) = 5.71#

3.2.3.3 Cont'd

Drawer 2 - SMP + Bev.

Drawer Base -	(21.22 x 21.2 x .050)	= 22.493
2 Sides	2(21.22 x 3.57 x .050)	= 7.576
2 Sides	2(21.1 x 3.57 x .050)	= 3.770
5 Dividers -	5(21.12 x 3.57 x .010)	= 4.509
0-g Cover/ Restraint	(21.2 x 21.22 x .020)	= 8.997
		54.878

Wt = 54.878 (.10) = 5.49#

Fixed Stowage Area

(See P. 30) - 4.66#

Support Structure

4 Drawer Guide Channels

Ref. P. 35 2(9.763) = 19.526 in^3

2 Drawer Center Guides

Ref. P. 35 2(4.881) = 9.762 in^3

6 Drawer Mounted Guides

Ref P. 35 2(9.109) = 18.218 $in.^3$

4 Corner Closure Angles (Ref. P. 35) = 9.083 in^3

4 Corner Closure Angles (Ref. P. 35) = 9.071 in

8 Intercostal Supports (Ref. P.35) $\frac{8}{6}$ (10.148) = 13.531 in^3

8 Intercostal Supports (Ref. P. 35) $\frac{8}{6}$ (10.135) = 13.513 in^3

2 Tee Stiffeners x .090

Section Area = .158 in^2 x 23.230 lg
2(.158) (23.230) = 7.341 in^3

3.2.3.3 Cont'd

4 Corner Supports (Ref. P. 35) = $14.369\ in^3$

4 Guide Channels (Ref. P. 35) = $21.220\ in^3$

Negator Springs & Mounts (4 reqd.)

Use total weight = 6#

Support Structure Wt. = (19.526 + 9.762 + 18.218 + 9.083 + 9.071 + 13.531 + 13.513 + 7.341 + 14.369 + 21.220).10 + 6 = 19.56#

Total Weight

Storage Module for 1-Meal Choice/2 Drawers

Module Outer Shell	–	15.45
Storage Drawers & Shelves	–	15.86
Support Structure	–	19.56
Storage Module	–	50.87
10% Contingency	–	5.09
25% Vehicle Interface Structure	–	12.72
		68.68#
Food Stowage Structure + 10% Contingency		55.96
15% Vehicle Interface Structure		8.39
		64.35#

3.2.4 Two Meal Choice System

No detailed analysis has been done for this system. See Section 2.2.3

3.2.5 Three Meal Choice System

3.2.5.1 Packaging Plan

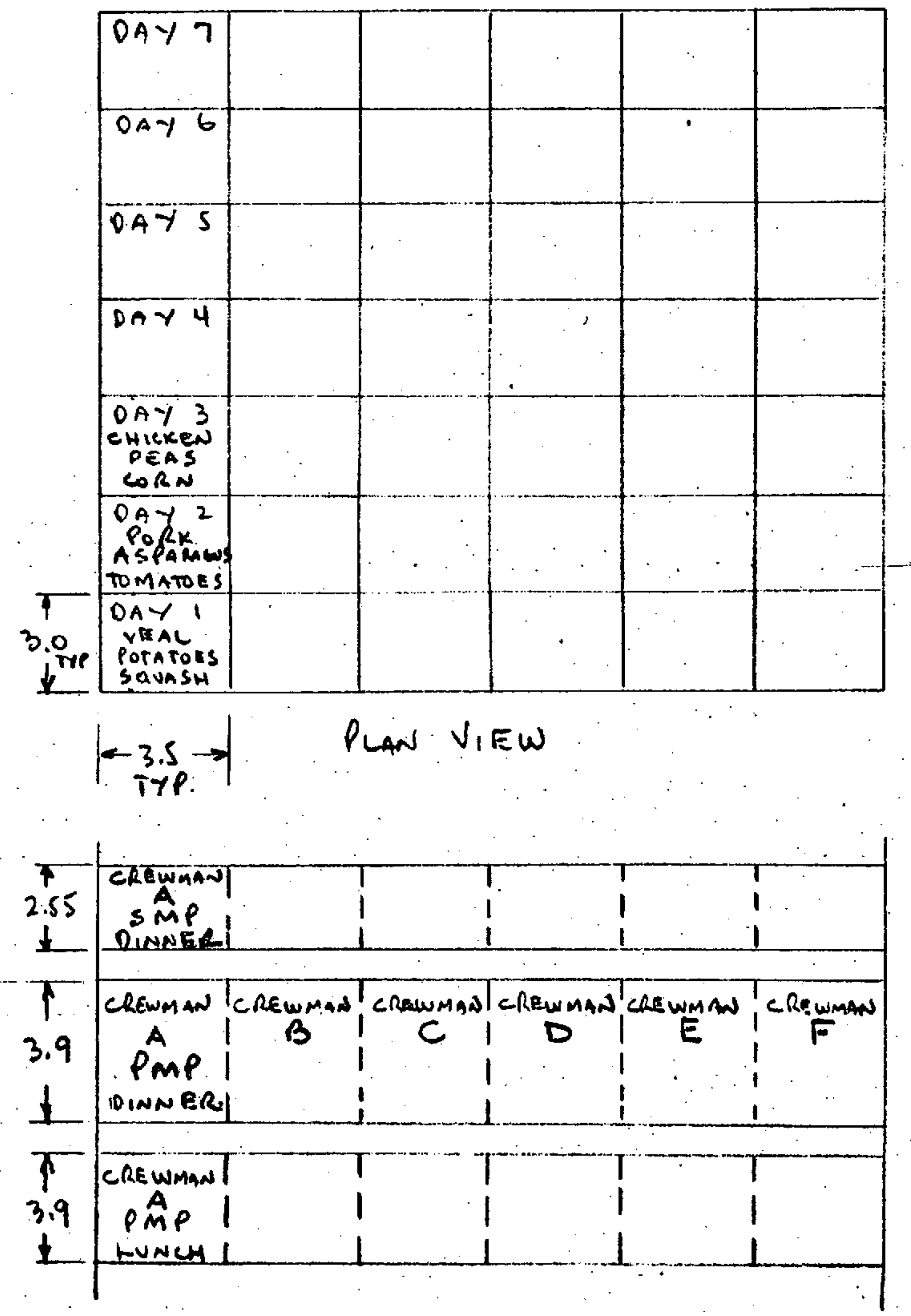
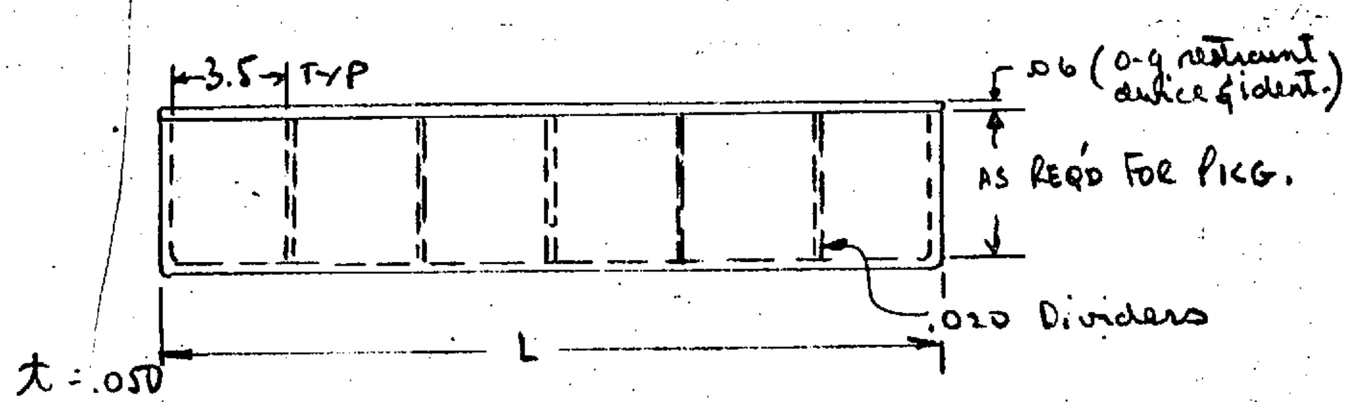


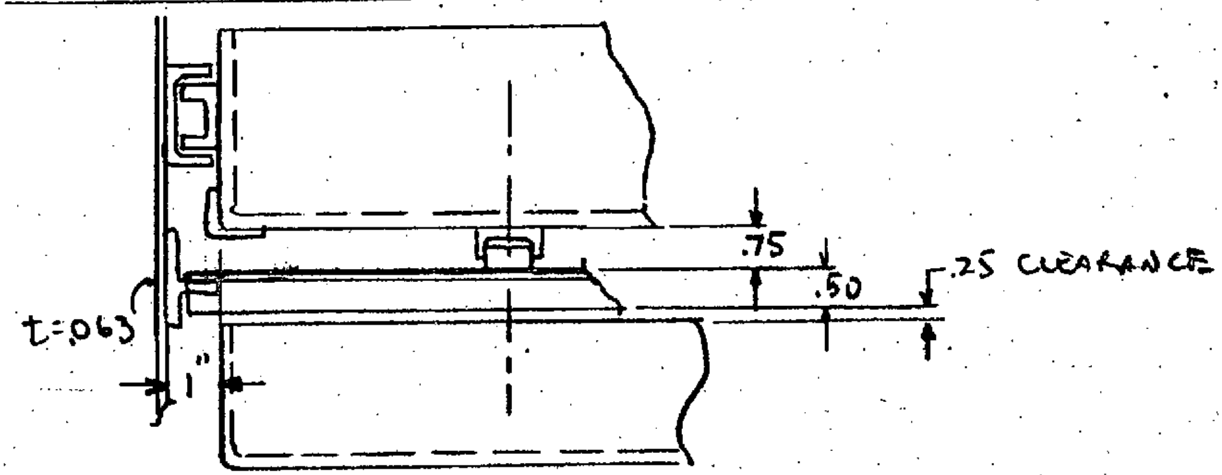
FIGURE 11 - THREE MEAL CHOICE SYSTEM

PACKAGING PLAN



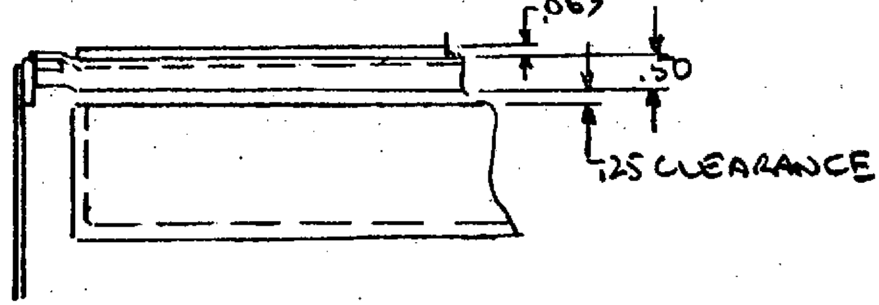
$$L = 6(3.5) + 5(.020) + 2(.050) = 21.2''$$

DRAWER INSTL



CABINET WIDTH $= L + 2(1.00 + .063) = 21.2 + 2.126 = \underline{23.326}$

TOP CONSTRUCTION



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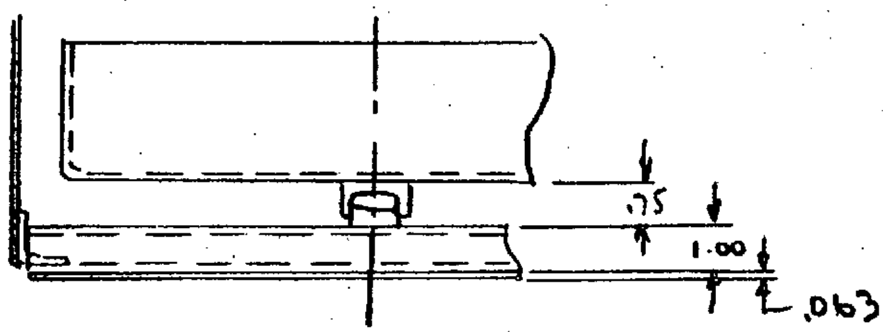
3.2.5.2 Cont'd

Bottom Construction



Assume end construction similar to top construction.

Cabinet Height = Sum of	Dinner PMP	=	3.9
	Dinner SMP	=	2.55
	Lunch PMP	=	3.9
	Lunch SMP	=	2.55
	Breakfast PMP	=	2.6
	Breakfast SMP	=	2.05
	Snack	=	1.0
	*Beverage	=	3.8
			22.35"
+ No. of Spaces between drawers x cap (7 x 1.5")		=	10.5
+ Top clearance		=	.813
+ Bottom Clearance		=	1.813
+ 8 Drawers (.06 + .05)		=	.88
			36.356

* Bev based on 5 in^3/package x 8/Man day = 40 x 42 = 1680 in^3

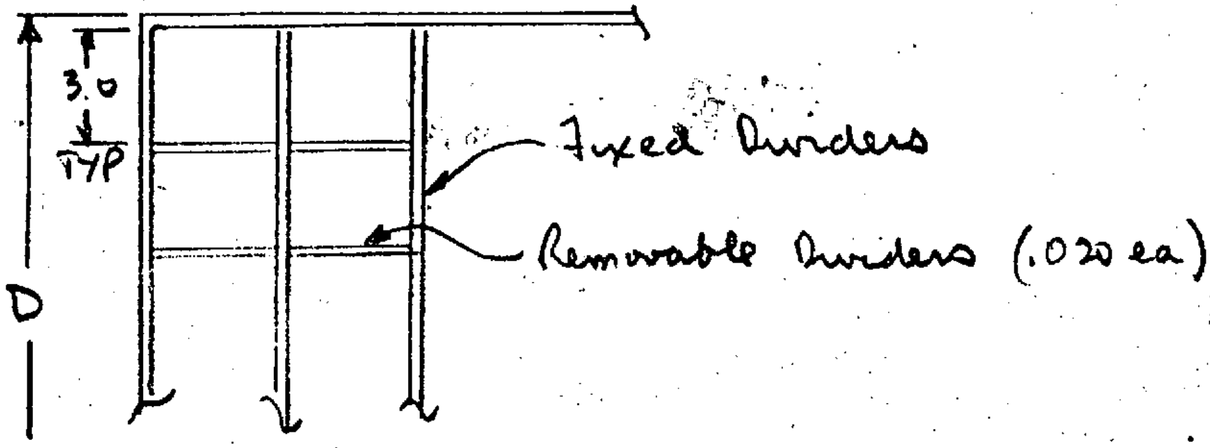
Using 3 x 3.5 package x 42 packages (6 x 7)/drawer= 441 in^2

Bev. package ht. = 1680/441 = 3.8"

3.2.5.3 Module Dimensions and Volume

Cabinet Width (Ref. 3.2.5.2) = 23.326

Cabinet Depth



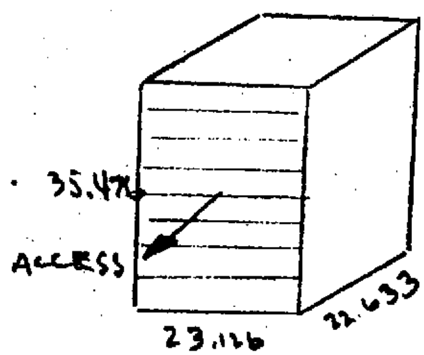
$$D = 7\ (3.0) + 6\ (.020) + 2\ (.050) = 21.22$$

Cabinet Depth	= Sum of D	=	21.22
	+ End cl.	=	.813
			22.033

Cabinet Volume = 23.326 x 22.033 x 36.356

= 18684.87 in^3

= 10.81 ft^3



3.2.5.4 Preliminary Weight Analysis

<u>Full Choice System</u>

Cabinet Sheets (t - .040)

2 Side Panels	(22.633 x 35.476)2	1605.86
1 Back Panel	23.126 x 35.476	820.42
2 End Panels	(23.126 x 22.633)2	1046.83
1 Front Panel	(Neglect due to cutouts)	
		3473.11
Wt = .040 (3473.11).10	=	13.89#

<u>Cabinet Food Storage Drawers</u>

2 Dinner & Lunch PMP @ 3.9 pkg. ht.

Drawer Base	= (21.22 x 21.2 x .050)	=	22.493
2 Sides	=2(21.22 x 4.01 x .050)	=	8.509
2 Sides	=2(21.1 x 4.01 x .050)	=	8.461
5 Dividers	=5(21.12 x 3.9 x .010)	=	4.118
Remov. Dividers	7(21.0 x 3.9 x .010)	=	5.733
0-g Cover/ Restraint	=(21.22 x 21.2 x .020)	=	8.99
			58.311
2 Drawer Wt. = 2(58.311) 0.10		=	11.66#

2 Dinner & Lunch SMP @ 2.55 pkg. ht.

Drawer Base	= (21.22 x 21.2 x 050)	=	22.493
2 Sides	=2(21.22 x 2.66 x .050)	=	5.645
2 Sides	=2(21.1 x 2.66 x .050)	=	5.639
5 Dividers	=5(21.12 x 2.55 x .010)	=	2.693
Removable Dividers	=7(21.0 x 2.55 x .010)	=	3.749
0-g Cover/ Restraint	= (21.22 x 21.2 x .020)	=	8.997
			49.216
2 Drawer Wt. = 2(49.216) 0.10		=	9.84#

3.2.5.4 Cont'd

1 Breakfast PMP @ 2.6 pkg. ht.

Assume same weight as Dinner & Lunch SMP @

2.55 pkg. ht. = 4.92#

1 Beverage @ 3.8 Pkg. ht.

Assume same weight as Dinner & Lunch PMP @

3.9 pkg. ht. = 5.83#

1 Breakfast SMP @ 2.05 pkg. ht.

Drawer Base	= (21.22 x 21.2 x .050)	=	22.493
2 Sides	=2(21.22 x 2.16 x .050)	=	4.583
2 Sides	=2(21.1 x 2.16 x .050)	=	4.558
5 Dividers	=5(21.12 x 2.05 x .010)	=	2.165
Removable Dividers	=7(21.0 x 2.05 x .010)	=	3.014
0-g Cover/ Restraint	= (21.22 x 21.2 x .020)	=	8.997
			45.81

Drawer Wt. = (45.81) 0.10 = 4.58#

1 Snack @ 1.0 pkg. ht.

Drawer Base	= (21.22 x 21.2 x .050)	=	22.493
2 Sides	=2(21.22 x 1.11 x .050)	=	2.355
2 Sides	=2(21.1 x 1.11 x .050)	=	2.342
5 Dividers	=5(21.12 x 1.0 x .010)	=	1.056
Removable Dividers	=7(21.0 x 1.0 x .010)	=	1.470
0-g Cover/ Restraint	= (21.22 x 21.2 x.020)	=	8.997
			38.713

Drawer Wt. = (38.713) 0.10 = 3.87#

Total Drawer Wt. = 11.66 + 9.84 + 4.92 + 5.83 +

4.58 + 3.87 = 40.7#

3.2.5.4 Cont'd

Support Structure

16 Drawer Guide Channels .090

Section Area = .209 in^2 x 21.22 lg ea
16 (.209) (21.22) = 70.96 in^3

7 Tee Stiffeners .090

Section Area = .158 in^2 x 22.633 lg ea
7(.158) (22.633) = 25.03 in^3

14 Intercostal Supports .050

Section Area = .073 in^2
Length = 23.126-(.063)2-(.090)2 = 22.82

14(.073)(22.82) = 23.32 in^3

8 Corner Closure Angles .050

Section Area = .098 in^2 x 22.82 lg ea
8 (22.82)(.098) = 17.89 in^3

8 Drawer Center Guides .090

Section Area = .209 in^2 x 21.22 lg ea
8(21.22)(.209) = 35.48 in^3

24 Drawer Mounted Guides .090

Section Area = .13 in^2 x 21.22 lg ea
24 (21.22)(.13) = 66.21

Total Area = 70.96 + 25.03 + 23.32 + 17.89 +

35.48 + 66.21 = 238.89

Wt = (238.89) 0.10 = 23.89#

Total Weight

Storage Module for Full Choice In-flight Menu

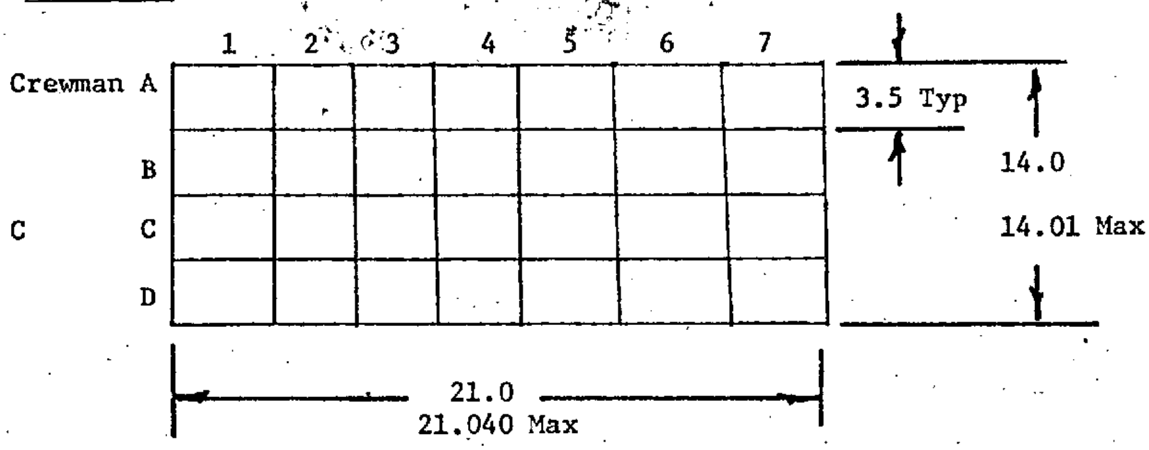
Module Outer Shell	-	13.89
8 Storage Drawers	-	40.7
Support Structure	-	23.89
Storage Module	-	78.48#
10% Contingency	-	7.85#
25% Vehicle Interface Structure	-	19.62#
		105.95#

3.3 Options for On-board Meal Selection - 4 Men - 7 Days

3.3.1 No Meal Choice System

28 Man-Days (4 Men x 7 Days)

• Packaging



• Cabinet Volume

W = 14.01 + 2(.75) + 2(.063) = 15.636

D = 21.040 + 2(.75) + 2(.063) = 22.666 Ref. P. 14

H = Ref. P. 14 29.116

Volume = 10318.873 in^3

= $5.97\ ft^3$

• Weight Approximation

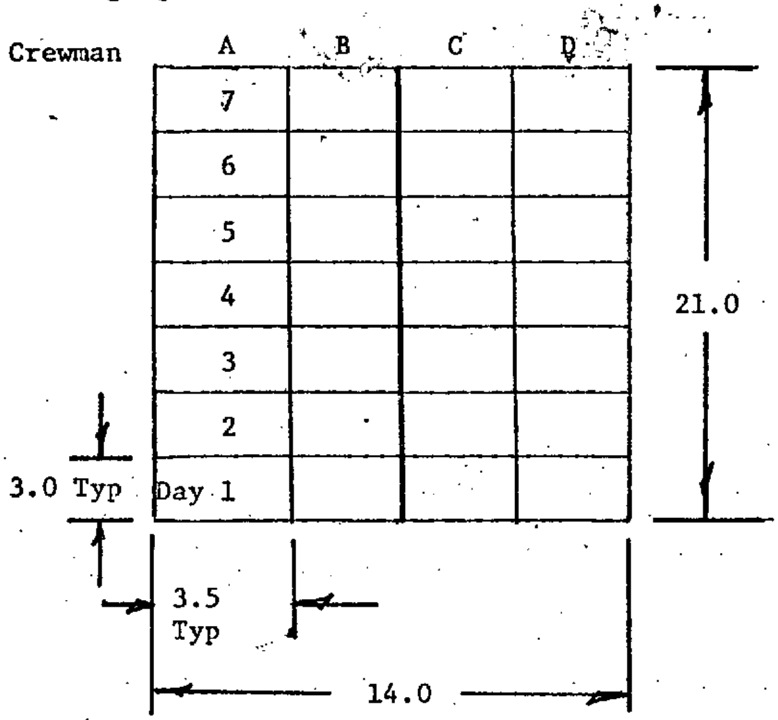
Use ratio for 42 man days - $\frac{42.73\#}{8.64\ ft^3} \cong 4.95\#/ft^3$

Wt = 5.97 x 4.95 $\cong$ 29.5#

3.3.2 Full Meal Choice System

<u>28 Man-Days (4 Men x 7 Days)</u>

•Packaging



(Ref. 3.2.5.2 Cabinet Width) = 4(3.5) +5(.020) + 2(.050) +

2(1.00 +.063) = 16.326

(Ref. 3.2.5.2 Cabinet Height) = 36.356

(Ref. 3.2.5.2 Cabinet Depth) = 22.033

- Cabinet Volume = 16.326 x 36.356 x 22.033

  = 13077.644

  = 7.57 ft^3

- Wt = Using 42 man day ratio $\frac{99.28}{10.81}$ = 9.2#/ft^3

  Wt = 7.57 x 9.2 = 69.6#

3.4 Stowed Weight of Food 6 Men - 7 Days

	Wet Weight		Solids		Dry Wt.
Entree	6 oz.	x	25%	=	1.5
Side Dishes	15 oz.	x	20%	=	3.0
Beverages	8 oz.	x	10%	=	0.8
Desserts	3 oz.	x	*20%	=	0.6
Soup	4 oz.	x	*10%	=	0.4
					6.3 oz.

* Assumed Values

Per Man Day - Dinner 6.3

Lunch 6.3

Breakfast 5.1 (assume 80% of main meals)

Snack 3.6 (assume 16 oz. bev. = 1.6 & 2 oz. snack)

21.4 oz./man day

21.4 x 42 man days = 898.8 oz. = 56.18# Food

3.5 Package Weight

Dinner	-	8 packages	Ref. Fig. 6
Lunch	-	8 packages	
Breakfast	-	6 packages	
Snack	-	3 packages	
		25 packages/man day	

25 x 42 man days = 1050 pkgs. x*15 $\frac{\text{grams}}{\text{pkg}}$ = 34.69# pkg.

90.87 Total