DESIGN DATA SHEET
DEPARTMENT OF THE NAVY
NAVAL SHIP ENGINEERING CENTER

1 February 1967
DDS 9590-1

REFRIGERATING EQUIPMENT FOR STORAGE COMPARTMENTS - HEAT LOAD CALCULATION AND SELECTION.

Supersedes DDS 9590-1, dated 1 Jan. 1962, w/Rev. dated 1 March 1964.

9590-1-a. References

(1) General Specifications for Ships of the United States Navy.

(2) Design Data Sheet DDS 9390-1 (ex 3901-1).

(3) ASHRAE Guide and Data Book (1965 Edition).

(4) Standard Drawing - NAVSHIPS No. S3801-921784, Vaneaxial Fan.

9590-1-b. General design

(1) The refrigeration heat load for each refrigerated compartment shall be calculated for a pull-down condition, and also for a normal operating condition, and shall include the sum of the estimated transmission, infiltration, ventilation, and product heat loads of each compartment. Chilled storerooms shall be designed for 0 degrees F. as well as 33 degrees F. Pull-down condition shall be considered as the period during which the temperature of the products within the compartment is being reduced to the design holding temperature. Normal condition shall be the period during which design temperatures are being maintained after the pull-down condition has been accomplished.

(2) The basis for selection of condensing unit loads shall be as follows:

(a) Continuous operation of all condensing units on a system for pull-down condition.

(b) Condensing unit operation shall not exceed 18 hours per day for normal operation with standby compressor secured.

(3) Cooling coils shall be selected on the basis of compressor selection for pull-down operation, or design coil refrigerant temperature for normal operation, whichever establishes the maximum coil surface requirements.

9590-1-c. Symbols

A Area, surface, square feet.

c Specific heat at indicated condition, Btu/lb.

H_a Heat gain, infiltrating air, Btu/cu. ft.

h_a = Specific enthalpy of entering dry air, Btu/lb.

h_{a_h} = Difference between enthalpy of entering moist and dry air, Btu/lb. dry air.

h_h = Enthalpy of final moist air at saturation per pound dry air, Btu/lb.

h_w = Specific enthalpy of condensed water, Btu/lb. water.

RH = Relative humidity, percent of entering dry air.

R_1 = Respiration rate at entering temperature, Btu/lb./24 hours.

R_2 Respiration rate at final temperature, Btu/lb/24 hours.

t_r Temperature, design refrigerant, degree F.

t_{rc} - Temperature, refrigerant based on compressor capacity, degree F.

t_1 = Temperature, entering, degree F.

t_2 = Temperature, final, degree F.

U = Overall transmission coefficient, Btu/hr./sq. ft./degree F.

V = Cubical contents of storage compartment, cu. ft.

V_h = Volume of final moist air at saturation per pound, cu. ft./lb.

W_c Weight, container, pounds.

W_p Weight, product, pounds.

W_{s1} Humidity ratio at saturation, entering air, lbs. water/lb. dry air.

W_{s2} Humidity ratio at saturation, final air, lbs. water/lb. dry air.

9590-1-d. Detail design

(1) Cooling load estimate. - The equations for determining total refrigeration cooling load and the basis for selection of equipment, are summarized in Table 1.

(2) Transmission heat load for each refrigerated space shall consist of the heat gained by temperature differential between the inner and outer surfaces of the compartment boundaries; equation (1).

(a) The following ambient temperatures (t_1) for the outer surfaces shall be used in the calculation of heat gains to refrigerated storage spaces:

Boundary surface	Temperature (degrees F.)
Adjacent machinery spaces	120
Other nonrefrigerated spaces	100
Surface exposed to sun	140
Surface exposed to water	90

The maximum design storage temperature of an adjacent refrigerated storage space shall be used as the outer ambient temperature of the area being estimated. Where the design temperature is lower, so as to constitute a heat loss condition, the heat loss shall be neglected in the calculation.

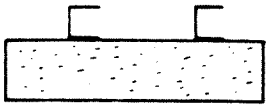
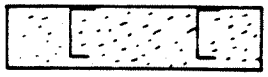
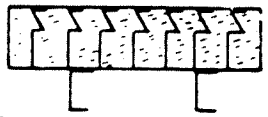
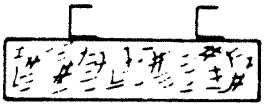
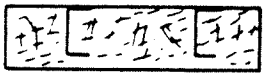
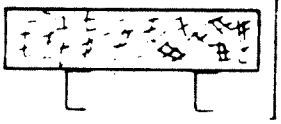
(b) The overall transmission coefficients U for various types of construction for shipboard application are shown in Table 2. The U values for construction using fibrous glass insulation are developed and established by Reference 2. For standard Navy refrigerator construction, using either six inches of three-pound/cubic foot density fibrous glass insulation or polyurethane foam insulation with a two-pound/cubic foot density in walls and overheads and a four-pound/cubic foot density in decks, the values in Table 2 may be used.

TABLE I

EQUATIONS FOR LOAD ESTIMATION AND EQUIPMENT SELECTION

EQUATIONS		PULL-DOWN OPERATION	NORMAL OPERATION	EQUATION NO.
REFRIGERATION COOLING LOAD Transmission: $AU(t_1 - t_2)$ X 24 hrs.	=	Btu/24 hrs...	Btu/24 hrs...	(1)
Infiltration: VH_a X air changes/24 hrs.	=	Btu/24 hrs...	Btu/24 hrs...	(2)
Ventilation: $\frac{VH_a \text{ X } 24 \text{ hrs.}}{3}$	=	.....	Btu/24 hrs...	(3)
Product: Precooling, internal heat: $(W_p c + W_c c)(t_1 - t_2)$	=	Btu/24 hrs...	.....	(4)
Precooling respiration: $\frac{W_p(R_1 + R_2)}{2}$	=	Btu/24 hrs...	.....	(5)
Storage respiration: $W_p R_2$	=	.....	Btu/24 hrs...	(6)
Total cooling load.....		Btu/24 hrs...	Btu/24 hrs...	
REQUIRED CONDENSING UNIT CAPACITY				
$\frac{\text{Total pull-down cooling load}}{\text{24 hours}}$	=	Btu/Hr.....	.....	(7)
$\frac{\text{Total normal cooling load}}{\text{18 hours}}$	=	.....	Btu/hr.....	(8)
REQUIRED COOLING COIL SIZE				
$\frac{\text{Compartment pull-down cooling load}}{U(t_2 - t_{r_c}) \text{ X 24 hours}}$	=	Sq.ft. Surface	.....	(9)
$\frac{\text{Compartment normal cooling load}}{U(t_2 - t_r) \text{ X 18 hours}}$	=	.....	Sq.ft. Surface	(10)

TABLE 2 - VALUES OF U

REFRIGERATOR SURFACE AND TYPE	CONSTRUCTION	INSULATION	U Btu/sq. ft./°F./hr. 4"	5"	6"
Wall, external stiffeners		Fibrous glass	- -	- -	0.045
Wall, internal stiffeners		Fibrous glass	- -	- -	0.100
Overhead, internal stiffeners		Fibrous glass	- -	- -	0.100
Deck, nonmetallic internal stiffeners		Fibrous glass	- -	- -	0.110
Wall, external stiffeners		Polyurethane foam	0.041	0.034	0.029
Wall, internal stiffeners		Polyurethane foam	0.087	0.071	0.060
Overhead, internal stiffeners		Polyurethane foam	0.090	0.073	0.061
Deck, no internal reinforcement		Polyurethane foam	0.041	0.034	0.029

(3) Infiltration heat load for each refrigerated space shall consist of the heat gained through admittance of air by door openings and cracks; equation (2).

(a) The condition of the entering air t_1 shall be assumed to be 100 degrees F. with 60 percent relative humidity, where air is admitted direct from an adjoining space or vestibule which is not refrigerated, or 50 degrees F. with 80 percent relative humidity, where the air enters from a refrigerated vestibule or thaw room.

(b) The quantity of admitted air is assumed proportional to the compartment size and is designated as average air changes per 24 hours, as shown in Figure 1.

(c) The entering air is considered to have the temperature and moisture content of the space adjacent to the door opening, as indicated in paragraph 9590-1-d-(3)-(a), and the heat gained per cubic foot is determined by the following equation.

$$H_a = \frac{h_a + RHh_{ah} - \left[h_h + h_w \ (RHW_{s_1} - W_{s_2})\right]}{V_h} \qquad \text{Eq. (11)}$$

For conditions established by paragraph 9590-1-d-(3)-(a), the heat gain of infiltrating air is graphically shown in Figure 2.

(4) Ventilation heat load, equation (3), shall be considered for fruit and vegetable refrigerated cargo spaces and shall consist of the heat gained by admittance of outside air required for displacement of vitiated compartment air. The ventilation heat load shall be used in heat load calculations only where it exceeds the infiltration load and then shall be used in lieu of the infiltration load for the operating condition being calculated.

(a) The outside air condition shall be assumed to be 100 degrees F., with an 80 percent relative humidity and a unit heat gain H_a determined by equation (11), or as shown in graph, Figure 2.

(b) The quantity of outside air shall be based on the use of a supply fan with an hourly capacity equal to the gross cubical contents of the compartment. The fan shall be assumed to operate under the following limitations:

1 A maximum of 20 minutes in any one hour.

2 Only one compartment on a refrigeration plant shall be ventilated during a 20-minute period.

3 During normal operation period only.

(5) Product heat load for each refrigerated space shall consist of the heat gain due to the internal heat of the product and containers entering the compartment for storage; equation (4). In addition, for chilled storage, the product heat load shall also include respiration heat from fruits and vegetables; equations (5) and (6).

(a) The assumed maximum temperature t_1 of the entering product and the time (days) allowed for temperature reduction is listed in Table 2. Where a compartment is designated as requiring two temperatures, such as 33 degrees F. and 0 degrees F., the computation for each condition shall assume that the product will be that associated with the temperature shown in Table 3.

(b) The specific heat c and respiration rates R_1 and R_2, of various products, are available in reference (3).

(c) For general storage of food products and where the kind and proportion of sundry products is indefinite, the product heat load can be approximated by assuming the maximum permissible loading of composite products which will allow satisfactory operation of the refrigerating equipment and the average heat load of the assumed products.

TABLE 3

AVERAGE PRODUCT LOAD CONDITIONS

	Storage space	
	Chill	Freeze
Final temperature, degrees F., (t_1)	33	0
Entering temp. max., degrees F., (t_2)	55	15
Time permitted for reducing temperature, days:		
Ships refrigerated stores	2	2
Refrigerated cargo spaces	3	5
Average, product wt. lbs/cu. ft. (W_p)	29.7	35.9
Average, product specific heat Btu/lb (c)	0.85	0.40
Average, container wt. lbs/cu. ft. (W_c)	3.2	3.64
Average, container specific heat Btu/lb (c)	0.65	0.65
Respiration rate, Btu/24 hours:		
Entering condition (R_1)	3.20	
Final condition (R_2)	1.08	

1 The available volume of a compartment for maximum product storage is the volume remaining after allowances for package spacing, air circulation, deck gratings, battens, coils, and other obstructions. The available volume based on typical installation is shown in Figure 3.

2 The average weight W_p per volume, specific heat c and respiration R_1 and R_2 of assumed composite storage is established in Table 3 from data of typical ship loading for recommended rations. The total weight of W_p, and of container W_c, for equations (4), (5), and (6) is then found by the equation

$$W_p \text{ total} = V \times \text{ratio } \frac{\text{usable volume}}{\text{total volume}} \times W_p \text{ per volume}$$

or

$$W_c \text{ total} = W_c \text{ per volume}$$

Eq. (12)

(d) For small refrigerators, or for specific storage for other than food products, and where the product load is not a high heat source, a usage heat gain load plus infiltration load shall be assumed as shown in Figure 6.

(6) The electric motor heat load for each refrigerated space shall consist of the heat gained from the motors used with circulating fans and shall be included as follows:

HEAT LOAD FROM ELECTRIC MOTORS

Horsepower range:	Btu per 24 hrs. per horsepower
1/20 to 1/8	132,000
Greater than 1/8 to 1/2	102,000
Greater than 1/2 to 3	88,800
Greater than 3 to 20	70,000

(7) Circulating fans shall be included in all freeze and chill spaces.

(a) Overhead fans shall be installed on the following basis:

Overhead area (sq. ft.)	No. fans
Up to 120	1
121 to 200	2

NOTE: When volume is above 1200 cu. ft., refer to paragraph (b) below. The rated horsepower of overhead fans shall be considered as 1/20 horsepower.

(b) Vaneaxial fans shall be installed in all spaces having a gross volume of 1,200 cubic feet or more. The total fan capacity shall be approximately 1/3 cfm per cubic foot of gross space volume. The horsepower rating of these fans can be found in reference (4).

(8) The total cooling load shall be proportioned between multiple condensing units to allow flexibility of operation under pulldown and normal operation and to permit the use of identical equipment in each system. Units shall be selected, where feasible, to equalize loadings on multiple condensing units during both pulldown and normal operation. Final selection of units shall provide an optimum number of units, minimum unit size and reasonable cooling coil size.

(9) The required cooling coil size for each refrigerated space shall be based on the compartment cooling load and the temperature differential between the room and the refrigerant at the cooling coils; equations (9) and (10). The coil selected shall be the maximum coil required for either normal or pulldown condition.

(a) The condensing unit suction temperature can be determined from manufacturer's capacity tables or curves for a selected compressor.

(b) The refrigerant temperature in the cooling coil calculations is found by considering the following operating limitations:

1 The pressure line loss between the compressor and the cooling coils is equivalent to a reduction of the saturated vapor temperature of 3 degrees F. for compressor operation at an equivalent temperature below 0 degrees F. and a 2 degree F. assumed loss for compressor operation at or above an equivalent temperature of 0 degrees F.

2 The temperature differential between room temperature t_l and refrigerant temperature t_r at the cooling coils.

a Refrigerated Ships Stores Space - for the normal condition, a maximum temperature differential of 20 degrees F., and for pulldown condition a variable temperature differential that will be dependent upon the compressor selected to meet the normal operating condition.

b Refrigerated Cargo Space - The average refrigerant temperature not more than 12.5 degrees F. below the design room temperature for normal condition and a 20 degree F. difference for pull-down.

(c) The transmission coefficient U for cooling coils in gravity air, is shown in Figures 4 and 5. Provision for any additional surface for dryer effect is not included and must be separately considered. For usual installations, 10 percent additional coil surface may be used.

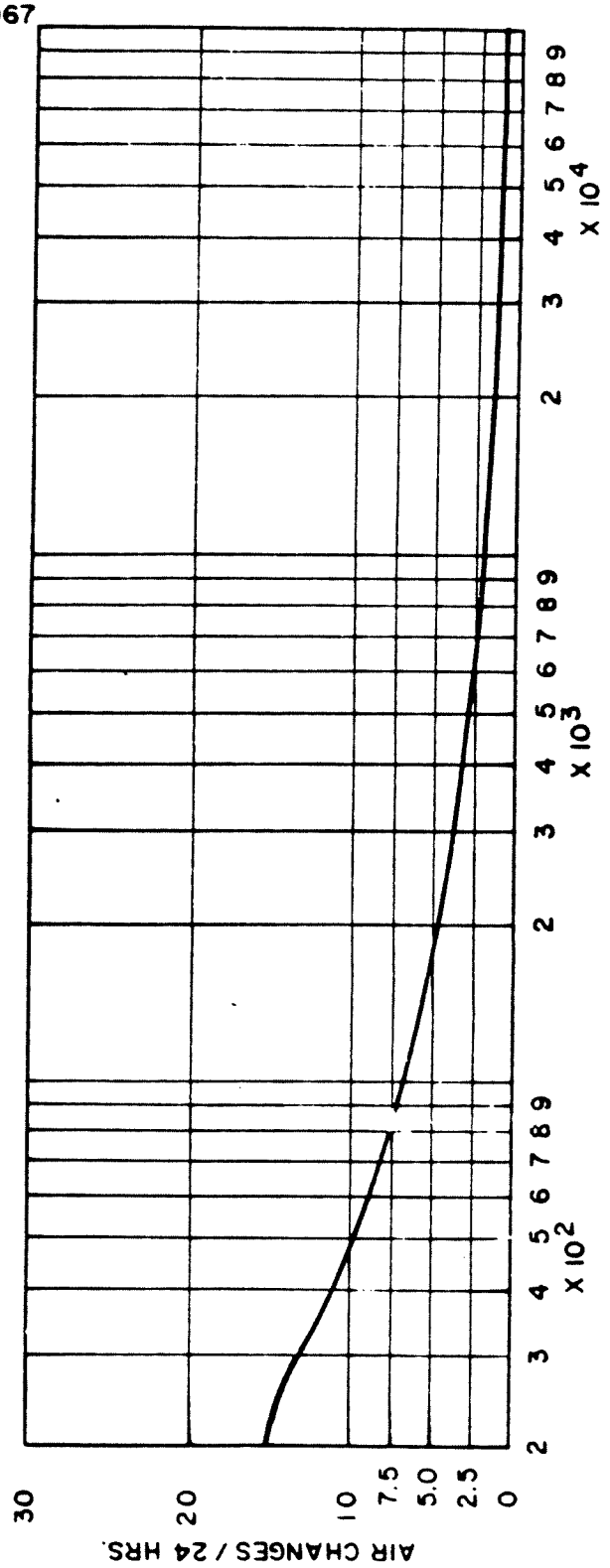

AIR CHANGES / 24 HRS.
30
20
10
7.5
5.0
2.5
0
2 3 4 5 6 7 8 9
X 10²
2 3 4 5 6 7 8 9
X 10³
2 3 4 5 6 7 8 9
X 10⁴
REFRIGERATOR VOLUME (V) - CUBIC FEET
REFRIGERATOR AVERAGE AIR CHANGES PER 24 HOURS
DUE TO INFILTRATION AND DOOR OPENINGS

Figure 1

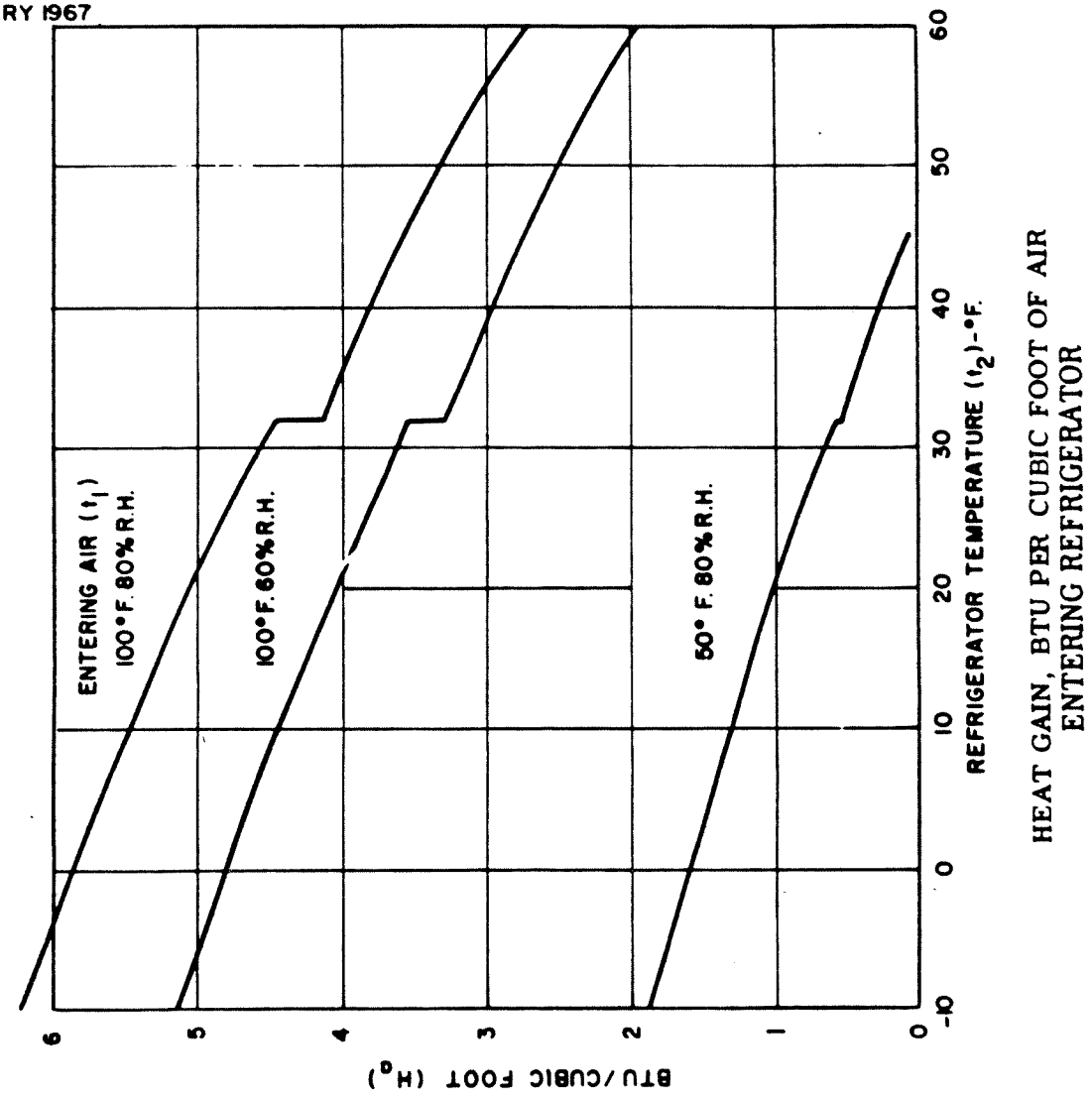

ENTERING AIR (t_1)
100°F. 80% R.H.
100°F. 60% R.H.
50° F. 80% R.H.
REFRIGERATOR TEMPERATURE (t_2)-°F.
60
50
40
30
20
10
0
-10
0
1
2
3
4
5
6
BTU/CUBIC FOOT (H_D)
HEAT GAIN, BTU PER CUBIC FOOT OF AIR ENTERING REFRIGERATOR

Figure 2

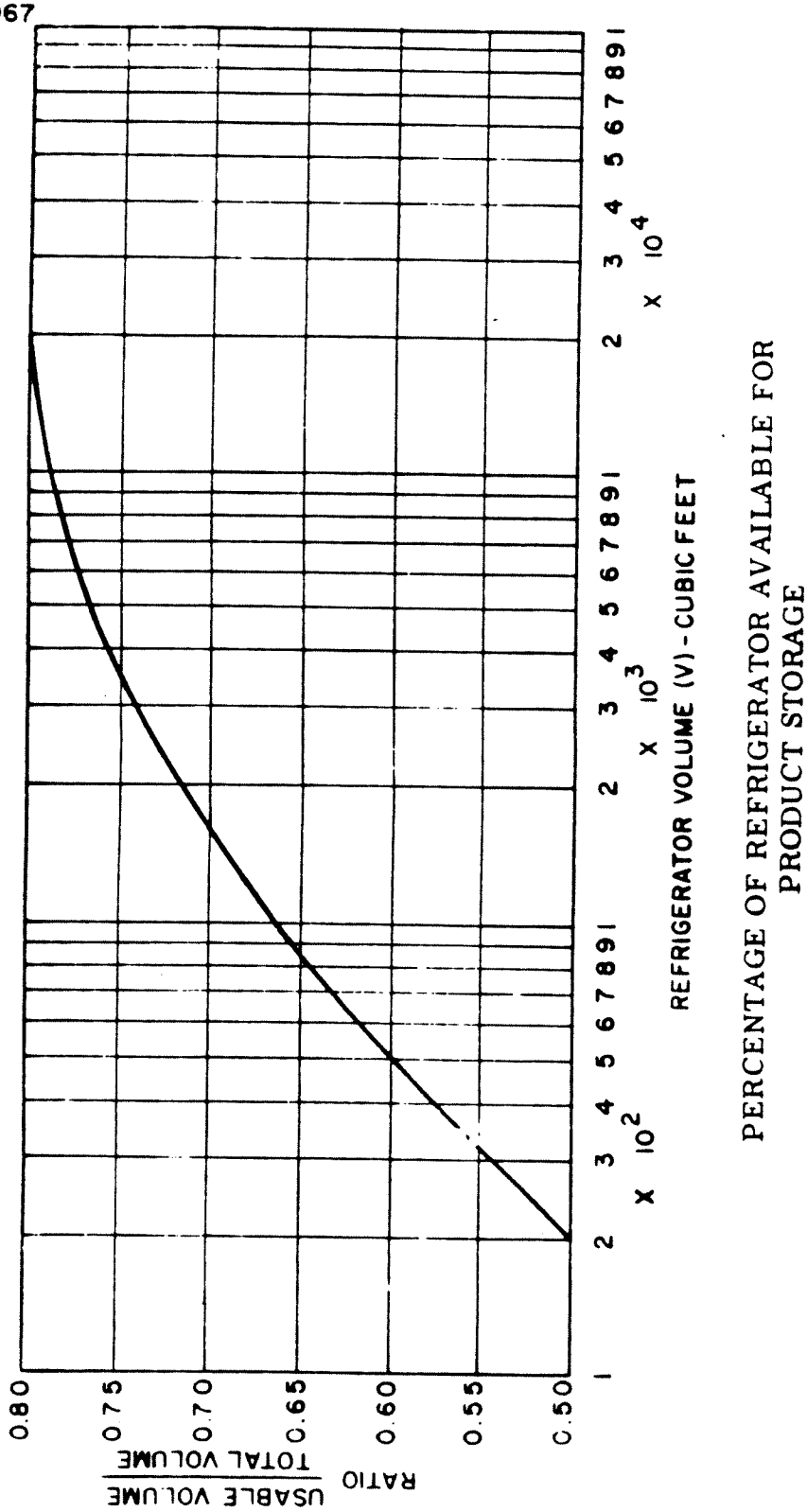
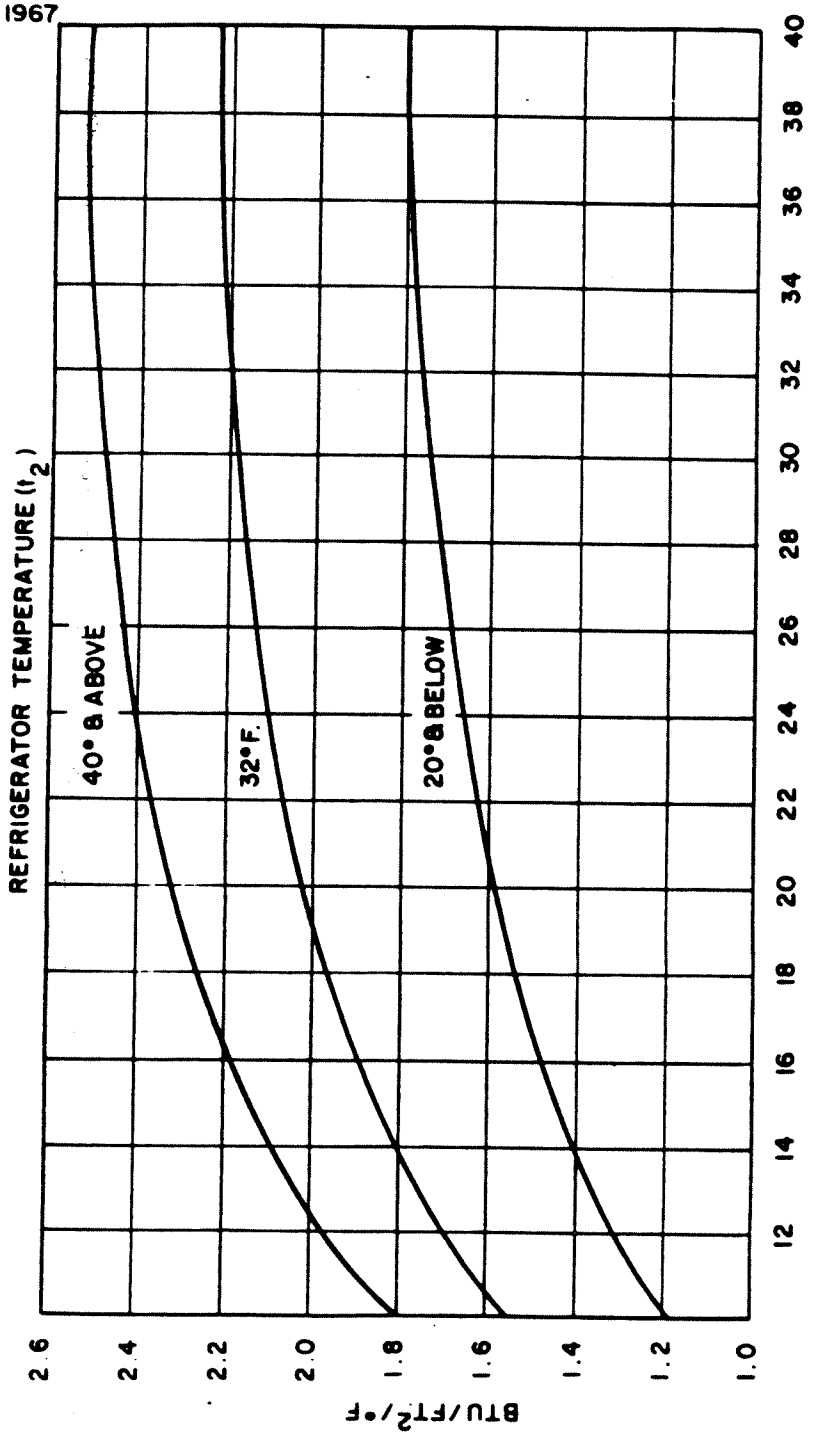
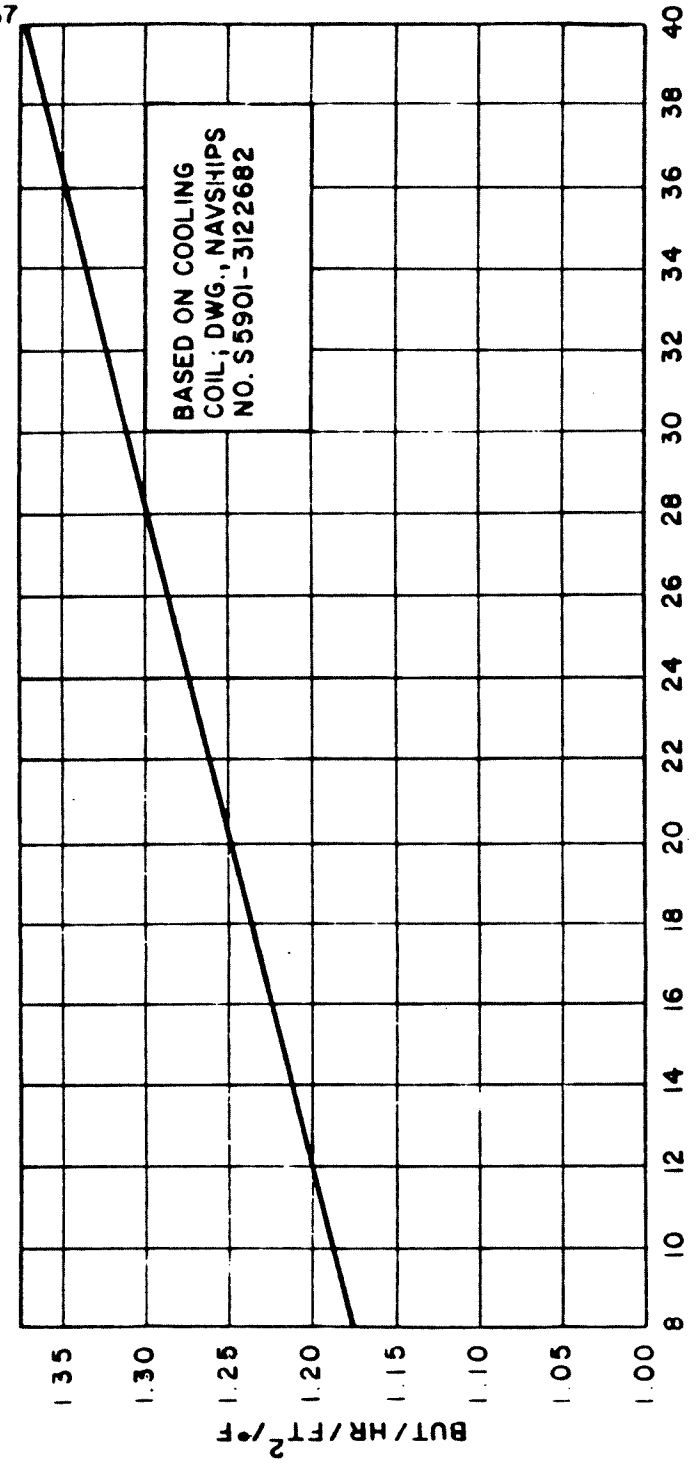

PERCENTAGE OF REFRIGERATOR AVAILABLE FOR PRODUCT STORAGE
RATIO USABLE VOLUME / TOTAL VOLUME
0.80
0.75
0.70
0.65
0.60
0.55
0.50
REFRIGERATOR VOLUME (V) - CUBIC FEET
1
2 3 4 5 6 7 8 9 1
X 10^2
2 3 4 5 6 7 8 9 1
X 10^3
2 3 4 5 6 7 8 9 1
X 10^4

Figure 3



TRANSMISSION COEFFICIENT (U) FOR BARE SURFACE
COPPER TUBE - HORIZONTAL - FOR GRAVITY AIR

Figure 4



TRANSMISSION COEFFICIENT (U) FOR FINNED SURFACE
COPPER FIN AND TUBE - GRAVITY TYPE

Figure 5

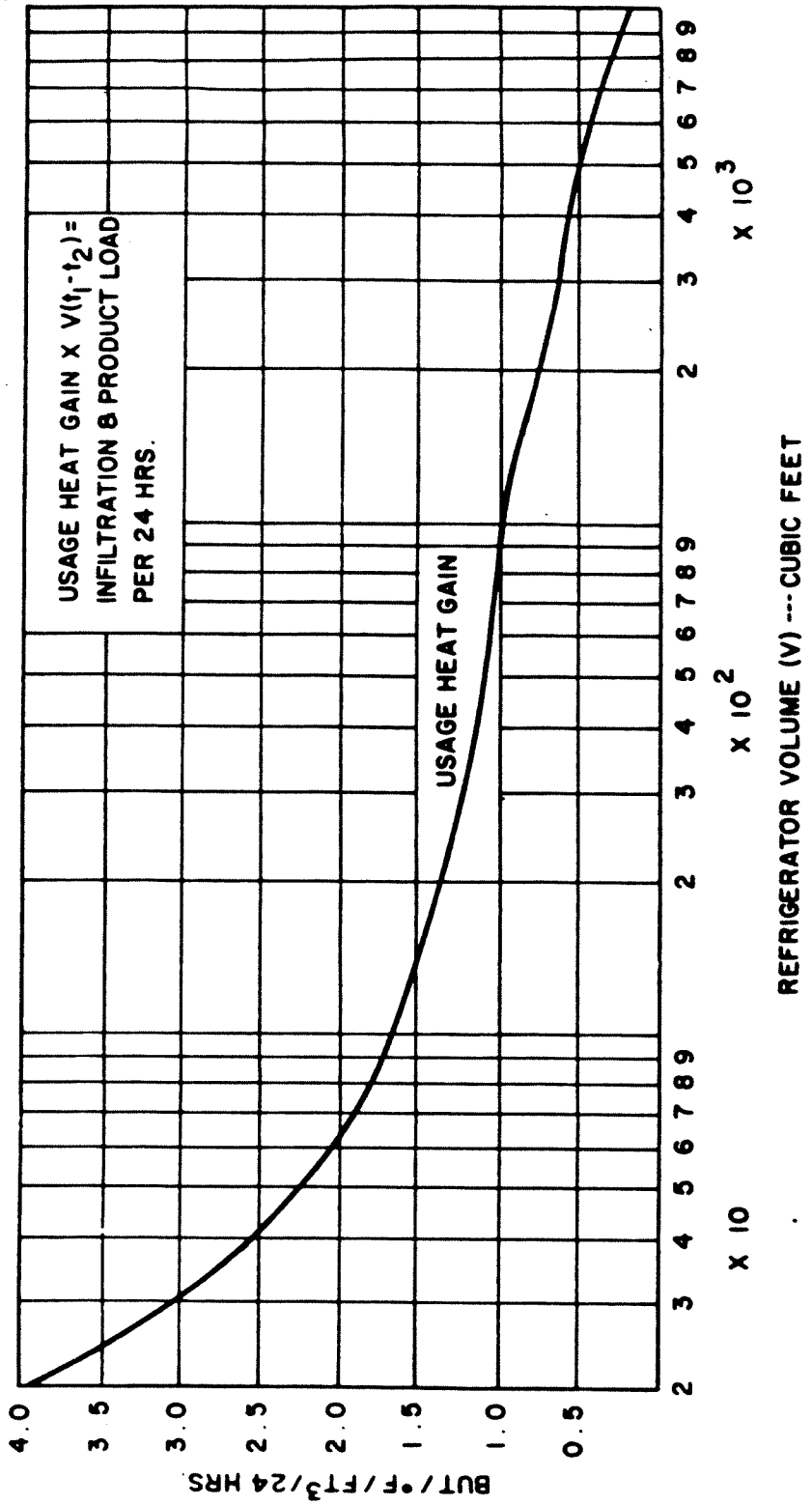

USAGE HEAT GAIN X V(t1-t2)= INFILTRATION & PRODUCT LOAD PER 24 HRS.
USAGE HEAT GAIN
BUT/°F/FT3/24 HRS
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
2 3 4 5 6 7 8 9
X 10
X 10^2
X 10^3
REFRIGERATOR VOLUME (V) --- CUBIC FEET

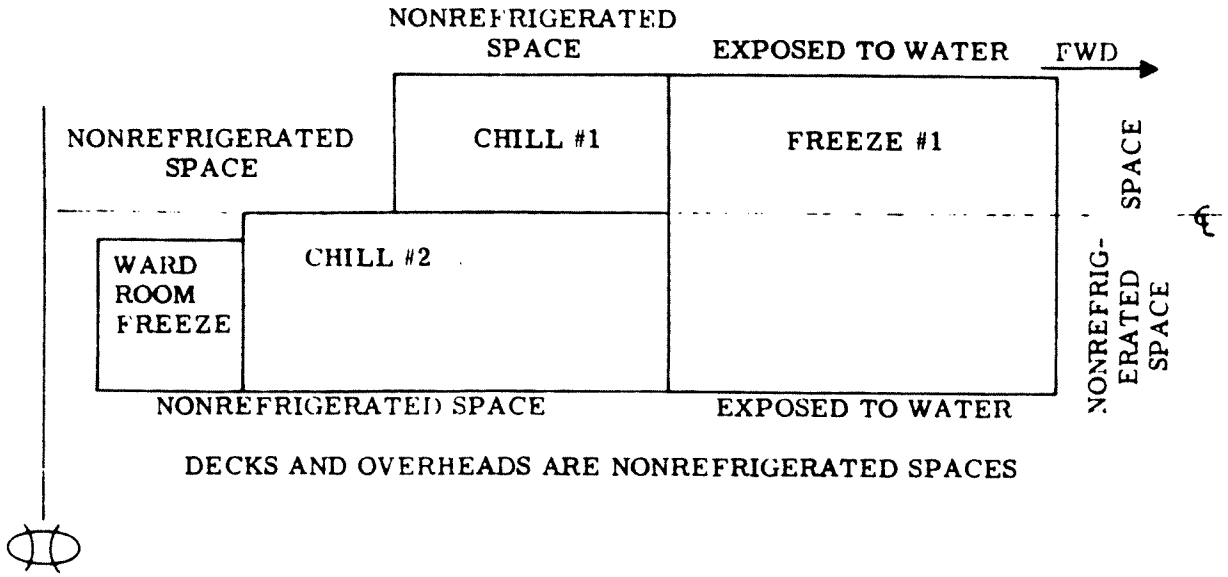
Figure 6

9590-1-e. Example I

(1) Calculate the cooling load, and determine the required compressor and cooling coil capacities for the refrigerated stores compartments shown in the diagram below with all boundary areas constructed in accordance with Navy standard practice using six inches of polyurethane foam insulation. The walls are assumed to have stiffeners arranged to face inboard on longitudinal bulkheads and to face midship on transverse bulkheads.

(2) Solutions

(a) See the following refrigeration load calculations.



1 FEBRUARY 1967

REFRIGERATION LOAD CALCULATIONS

REFERENCE: DDS 9590-1

KEY:
1. Overhead
2. Deck
3. Port Blk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft. Blk.

SPACE: Freeze No. 1 Des. Temp. 0 °F.

DIMENSIONS: 35 x 49 x 7

VOLUME (INSUL TO INSUL) = 12005 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio x W_p) 12005 x .79 x 35.9 = 340,474 lbs.

TOTAL CONTAINER WEIGHT
(Volume x Ratio x W_c) 12005 x .79 x 3.64 = 34,522 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu/24 hrs.	NORMAL OPERATION Btu/24 hrs.
	Transmissions Δt X AREA X U X 24		
1.	100 X 35 X 49 X 0.061 X 24	251,076	251,076
2.	100 X 35 X 49 X 0.029 X 24	119,364	119,364
3.	90 X 35 X 7 X 0.060 X 24	31,752	31,752
4.	90 X 35 X 7 X 0.060 X 24	31,752	31,752
5.	100 X 49 X 7 X 0.060 X 24	49,392	49,392
6.	33 X 49 X 7 X 0.029 X 24	7,878	7,878
	FAN (No. X hp X Btu/24 hrs.) 1 fan x 3 hp x 88,800	266,400	266,400
	INFILTRATION (Vol. X H_a X Air changes/24 hrs) 12005 x 2 x 4.8	115,248	115,248
	VENTILATION (Vol. X H_a X 24 hrs/3) X X8	-	-
	PRECOOLING INTERNAL HEAT $(W_pC + W_cC)(t_1-t_2)$/Days [(340,474 x 0.40) + (34,522 x 0.65)] X (15 - 0)/2	1,189,717	-
	PRECOOLING RESPIRATION $W_p(R_1 + R_2)/2$ X (+) /2	-	
	STORAGE RESPIRATION W_p R_2 X	-	
	SUBTOTAL	2,062,579	872,862
	SUBTOTAL Btu/hr. (24)	85,941	-
	(18)	-	48,492
	TONNAGE (SUB/Btu + 12,000)	7.16	4.04

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

REFERENCE: DDS 9590-1

KEY:
1. Overhead
2. Deck
3. Port Blk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft. Blk.

SPACE: Ward Room Freeze Des. Temp. 0 °F.

DIMENSIONS: 12 X 15 X 7

VOLUME (INSUL TO INSUL) = 1260 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio X W_p) 1260 x .67 x 35.9 30,307 lbs.

TOTAL CONTAINER WEIGHT
(Volume x Ratio X W_c 1260 x .67 x 3.64 = 3,073 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu 24 hrs.	NORMAL OPERATION Btu 24 hrs.
	Transmissions Δt X AREA X U X 24		
1.	100 X 12 X 15 X 0.061 X 24	26,352	26,352
2.	100 X 12 X 15 X 0.029 X 24	12,545	12,545
3.	100 X 12 X 7 X 0.029 X 24	5,842	5,842
4.	100 X 12 X 7 X 0.060 X 24	12,096	12,096
5.	33 X 15 X 7 X 0.060 X 24	4,990	4,990
6.	100 X 15 X 7 X 0.029 X 24	7,303	7,303
	FAN (No. X hp X Btu/24 hrs.) 1 Fan x 1/5 hp x 88,800	17,760	17,760
	INFILTRATION (Vol. X Ha X Air changes/24 hrs.) 1260 X 6 X 4.8	36,288	36,288
	VENTILATION (Vol. X H_a X 24 hrs./3) X X8	--	
	PRECOOLING INTERNAL HEAT ($W_pC + W_cC$) (t_1-t_2)/Days [(30,307 x 0.40) + (3,073 x 0.65)] x (15 - 0)/2	105,902	--
	PRECOOLING RESPIRATION $W_p(R_1 + R_2)/2$ X ( )/2	--	
	STORAGE RESPIRATION W_pR_2 X	--	
	SUB TOTAL	229,078	123,176
	SUBTOTAL Btu/hr. (24)	9,544	
	(18)	--	5,132
	TONNAGE (SUB/Btu ÷ 12,000	0.80	0.43

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

REFERENCE: DDS 9590-1

KEY:
1. Overhead
2. Deck
3. Port B lk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft. Blk.

SPACE: Chill No. 1 (Alternate A) Des. Temp. 0 °F

DIMENSIONS: 23 X 24 X 7

VOLUME (INSUL TO INSUL) = 3864 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio X W_p) 3864 x .75 x 35.9 = 104,038 lbs.

TOTAL CONTAINER WEIGHT
(Volume x Ratio x W_c) 3864 x .75 x 3.64 = 10,549 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu/24 hrs.	NORMAL OPERATION Btu/24 hrs.
	Transmissions Δt X AREA X U X 24		
1.	100 X 23 x 24 X 0.061 X 24	80,813	80,813
2.	100 X 23 x 24 X 0.029 X 24	38,472	38,472
3.	100 X 23 x 7 X 0.060 X 24	23,184	23,184
4.	33 X 23 x 7 X 0.060 X 24	7,651	7,651
5.	0 X 24 x 7 X 0.060 X 24	--	--
6.	100 X 24 x 7 X 0.029 X 24	11,685	11,685
	FAN (No. X hp X Btu/24 hrs) 1 x 1 x 88,800	88,800	88,800
	INFILTRATION (Vol. X H_a X Air changes/24 hrs) 3864 X 3 X 4.8	55,642	55,642
	VENTILATION (Vol. X H_a X 24 hrs/3) X X8	--	
	PRECOOLING INTERNAL HEAT (W_pC + W_cC) (t_1 - t_2)/Days [(104,038 X 0.40) + (10,549 X 0.65)]x(15 - 0)/2	363,540	--
	PRECOOLING RESPIRATION W_p (R_1 + R_2)/2 X (+) / 2		--
	STORAGE RESPIRATION W_p R_2 X	--	
	SUBTOTAL	669,987	306,447
	SUBTOTAL Btu hr. (24)	27,916	--
	TONNAGE (SUB/Btu ÷ 12,000)	2.33	1.42

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

REFERENCE: DDS 9590-1

KEY:
1. Overhead
2. Deck
3. Port Blk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft Blk.

SPACE: Chill No. 1 (Alternate B) Des. Temp. 33 °F

DIMENSIONS: 23 X 24 X 7

VOLUME (INSUL TO INSUL) = 3864 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio x W_p) 3864 X .75 X 29.7 86,071 lbs.

TOTAL CONTAINER WEIGHT
(Volume x Ratio x W_c) 3864 X .75 X 3.2 9,274 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu/24 hrs.	NORMAL OPERATION Btu/24 hrs.
	Transmission Δt X AREA X U X 24		
1.	67 X 23 X 24 X 0.061 X 24	54,145	54,145
2.	67 X 23 X 24 X 0.029 X 24	25,776	25,776
3.	67 X 23 X 7 X 0.060 X 24	15,533	15,533
4	X X X 24		
5	X X X 24		
6	67 X 24 X 7 X 0.029 X 24	7,828	7,828
	FAN (No. X hp X Btu/24 hrs) 1 X 1 X 88,800	88,800	88,800
	INFILTRATION (Vol. X H_a X Air changes/24 hrs.) 3864 X 3.2	37,094	37,094
	VENTILATION (Vol. X H_a X 24 hrs/3) X X8	--	
	PRECOOLING INTERNAL HEAT ($W_pC + W_cC$) ($t_1 - t_2$)/Days [(86,071 X 0.85) + (9,274 X 0.65)] X (55 - 33)/2	981,073	
	Precooling RESPIRATION $W_p (R_1 + R_2)/2$ 86,071 X (3.20 + 1.08)/2	184,192	
	STORAGE RESPIRATION $W_p R_2$ 86,071 X 1.08		92,947
	SUBTOTAL	1,394,441	322,133
	SUBTOTAL Btu/hr. (24)	58,102	--
	(18)	--	17,896
	TONNAGE (SUB/Btu ÷ 12,000)	4.84	1.49

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

REFERENCE: DDS 9590-1
KEY:
1. Overhead
2. Deck
3. Port Blk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft. Blk.

SPACE: Chill No. 2 (Alternate A) Des. Temp. 0 °F.

DIMENSIONS: 35 X 25 X 7

VOLUME (INSUL TO INSUL) = 6125 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio X W_p) 6125 x .77 x 35.9 = 169,313 lbs.

TOTAL CONTAINER WEIGHT
(Volume x Ratio x W_c) 6125 x .77 x 3.64 = 17,167 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu/24 hrs.	NORMAL OPERATION Btu/24 hrs.
	Transmissions Δt X AREA X U X 24		
1.	100 X 35 X 25 X 0.061 X 24	128,100	128,100
2.	100 X 35 X 25 X 0.029 X 24	60,984	60,984
3.	100 X 12 X 7 X 0.060 X 24	12,036	12,036
3.	33 X 23 X 7 X 0.060 X 24	7,651	7,651
4.	100 X 35 X 7 X 0.060 X 24	35,280	35,280
5.	0 X -- X -- X 24	--	--
6.	100 X 10 X 7 X 0.029 X 24	4,869	4,869
	FAN (No. X hp X Btu/24 hrs) 1 X 11/4 X 88,800	111,000	111,000
	INFILTRATION (Vol. X H_a X Air changes/24 hrs) 6125 X 2 X 4.8	58,800	58,800
	VENTILATION (Vol. x H_a x 24 hrs/3) X X8 --	--	
	PRECOOLING INTERNAL HEAT (W_pC + W_cC) (t_1 - t_2)/Days [(169,313 X 0.40) + (17,167 X 0.65)] X (15 - 0)/2	591,628	--
	PRECOOLING RESPIRATION $W_p(R_1 = R_2)/2$ X (+)/2		
	STORAGE RESPIRATION W_p R_2 X		
	SUBTOTAL	1,110,648	519,020
	SUBTOTAL Btu/hr. (24)	47,110	--
	(18)		28,834
	TONNAGE (SUB/Btu ÷ 12,000)	3.93	2.40

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

REFERENCE: DDS 9590-1

KEY
1. Overhead
2. Deck
3. Port Blk.
4. Stbd. Blk.
5. Fwd. Blk.
6. Aft. Blk.

SPACE: Chill No. 2 (Alternate B) Des. Temp. 33 °F.

DIMENSIONS: 35 X 25 X 7

VOLUME (INSUL TO INSUL) = 6125 cu. ft.

TOTAL PRODUCT WEIGHT
(Volume x Ratio x W_p) 6125 X .77 X 29.7 = 140,073 lbs.

TOTAL CONTAINER WEIGHT
(Vol. x Ratio x W_c) 6125 x .77 x 3.2 = 15,092 lbs.

KEY REF.	CALCULATIONS	PULL DOWN OPERATION Btu/24 hrs.	NORMAL OPERATION Btu/24 hrs.
	Transmissions: Δt X AREA X U X 24		
1.	67 X 35 X 25 X 0.061 X 24	85,826	85,827
2.	67 X 35 X 25 X 0.029 X 24	40,659	40,659
3.	67 X 12 X 7 X 0.060 X 24	8,104	8,104
4.	67 X 35 X 7 X 0.060 X 24	23,638	23,638
5.	-- X -- X -- X 24	--	--
6.	67 X 10 X 7 X 0.029 X 24	3,262	3,262
	FAN (No. X hp X Btu/24 hrs) 1 X 1-1/4 X 88,800	111,000	111,000
	INFILTRATION (Vol. X H_a X Air changes/24 hrs) 6125 X 2 X 3.2	39,200	39,200
	VENTILATION (Vol. X H_a X 24 hrs/3) X X8		
	PRECOOLING INTERNAL HEAT ($W_pC + W_cC$) ($t_1 - t_2$/Days [(140,073 X 0.85) + (15,092 x 0.65)] (55-33)/2	1,417,590	--
	PRECOOLING RESPIRATION $W_p(R_1 + R_2)/2$ 140,073 X (3.20 + 1.08)/2	299,756	
	STORAGE RESPIRATION W_pR_2 140,073 X 1.08		151,270
	SUBTOTAL	2,029,036	462,960
	SUBTOTAL Btu/hr. (24) / (18)	84,543	25,720
	TONNAGE (SUB/Btu ÷ 12,000)	7.05	2.14

REFRIGERATION LOAD CALCULATIONS (Cont'd.)

SHIP: (Sheet of)

SPACE	t_1, °F.	Tons/Space	Tons/Circuit	COMPRESSOR DESIGNATION	CONDENSING UNIT SUCTION Temp. °F.
Alternate "A" Pull down operation					
Freeze No. 1	0	7.16	7.16	1	-24
Ward Rm. freeze	0	0.80			
Chill No. 1	0	2.33	7.06	2	-25
Chill No. 2	0	3.93			
Normal Operation					
Freeze No. 1	0	4.04			
Ward Rm. freeze	0	0.43	8.29	1	-20
Chill No. 1	0	1.42			
Chill No. 2	0	2.40			
				2 (Standby)	
Alternate "B" Pull down operation					
Freeze No. 1	0	7.16	7.96	1	-21
Ward Rm. Freeze	0	0.80			
Chill No. 1	33	4.84	11.89	2	-11
Chill No. 2	33	7.05			
Normal Operation					
Freeze No. 1	0	4.04			
Ward Rm. freeze	0	0.43	8.10	1	-21
Chill No. 1	33	1.49			
Chill No. 2	33	2.14			
				2 (Standby)	

SHIP: REFRIGERATION LOAD SUMMARY (Sheet of)					
SPACE	STORAGE SPACE DES. TEMP.	PULL DOWN OPERATION		NORMAL OPERATION	
		Btu hr.	TONS	Btu hr.	TONS
Alternate "A"					
Freeze No. 1	0	85,941	7.16	48,492	4.04
Ward room freeze	0	9,544	0.80	5,132	0.43
Chill No. 1	0	27,916	2.33	17,025	1.42
Chill No. 2	0	47,110	3.93	28,834	2.40
TOTAL		170,511	14.22	99,483	8.29
Alternate "B"					
Freeze No. 1	0	85,941	7.16	48,492	4.04
Ward room freeze	0	9,544	0.80	5,132	0.43
Chill No. 1	33	58,102	4.84	17,896	1.49
Chill No. 2	33	84,543	7.05	25,720	2.14
TOTAL		238,130	19.85	97,240	8.10

(b) Required condensing unit capacity and selection.

1 The cooling load calculations indicate that for normal operation the maximum load is "Alternate A". This condition requires 8.29 tons of refrigeration with one unit secured.

2 The cooling load calculations indicate that for pull-down operation "Alternate B" is the maximum load. This condition requires 19.85 tons of refrigeration with all units operating. However, during pull-down of the chilled boxes with the circuits as indicated on the summary sheet, the additional refrigeration capacity obtained with higher suction temperatures may be utilized.

3 Combining steps 1 and 2 the selection will be based on the requirements of the normal operation for Alternate "A". It is more practical to select two units of the 8.29 tons each than either three units of 7.16 tons or four units of 4.25 tons.

4 The selection will be two units of 8.29 ton capacity, and the following refrigeration load summary demonstrates this selection.

(c) Compressor capacity based on condensing unit selected.

1 Table of typical rating for Model 'X' condensing unit 105 degrees F. condensing temperature.

SATURATED SUCTION TEMPERATURE, °F.	TONS OF REFRIGERATION
-26	6.9
-25	7.1
-22	7.6
-21	8.1
-20	8.3
-18	9.1
-16	9.7
-14	10.5
-10	12.1
0	16.3

(d) Required cooling coil size.

1 Table of refrigerant temperatures for each compartment.

SPACE	t_a °F.	PULL-DOWN CONDITION			NORMAL CONDITION		
		Comp. Suction Temp. °F.	Line Loss °F.	t_{r_c}, °F.	Comp. Suction Temp. °F.	Line Loss °F.	t_r, °F.
Alternate "A"							
Freeze No. 1	0	-24	-3	-21	-20	-3	-17
Ward room freeze	0	-25	-3	-22	-20	-3	-17
Chill No. 1	0	-25	-3	-22	-20	-3	-17
Chill No. 2	0	-25	-3	-22	-20	-3	-17
Alternate "B"							
Freeze No. 1	0	-21	-3	-18	-21	-3	-18
Ward room freeze	0	-21	-3	-18	-21	-3	18
Chill No. 1	33	-11	-3	-8	-21	*	13
Chill No. 2	33	-11	-3	-8	-21	*	13

* Back pressure regulator required.

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COIL SURFACE CALCULATIONS

Alternate "A" - Pull Down

Freeze No. 1 - $\frac{85,941}{1.26\ [0-(-21)]}$ = 3248 ft.^{2}

Ward room freeze - $\frac{9,544}{1.26\ [0-(-22)]}$ = 344 ft.^{2}

Chill No. 1 - $\frac{27,916}{1.26\ [0-(-22)]}$ = 1007 ft.^{2}

Chill No. 2 - $\frac{47,110}{1.26\ [0-(-22)]}$ = 1700 ft.^{2}

Alternate "A" - Normal

Freeze No. 1 - $\frac{48,492}{1.23\ [0-(-17)]}$ = 2319 ft.^{2}

Ward room freeze - $\frac{5,132}{1.23\ [0-(-17)]}$ = 246 ft.^{2}

Chill No. 1 - $\frac{17,025}{1.23\ [0-(-17)]}$ = 814 ft.^{2}

Chill No. 2 - $\frac{28,834}{1.23\ [0-(-17)]}$ = 1380 ft.^{2}

COIL SURFACE CALCULATIONS (Cont'd.)

Alternate "B" - Pull down

Freeze No. 1 - $\frac{85,941}{1.24\ [0-(-18)]}$ = 3,850 ft.^{2}

Ward room freeze - $\frac{9,544}{1.24\ [0-(-18)]}$ = 428 ft.^{2}

Chill No. 1 - $\frac{58,102}{1.38\ [30-(-8)]}$ = 1,027 ft.^{2}

Chill No. 2 - $\frac{84,543}{1.38\ [33-(-8)]}$ = 1,494 ft.^{2}

Alternate "B" - Normal

Freeze No. 1 - $\frac{48,492}{1.24\ [0-(-18)]}$ = 2,173 ft.^{2}

Ward room freeze - $\frac{5,132}{1.24\ [0-(-18)]}$ = 230 ft.^{2}

Chill No. 1 - $\frac{17,896}{1.25\ [33-13]}$ =

Chill No. 2 - $\frac{25,720}{1.25\ [33-13]}$ = 1,029 ft.^{2}

MAXIMUM CALCULATED COIL SIZE		
SPACE	CONDITION	COIL-ft.^{2}
Freeze No. 1	Pulldown, Alternate "B"	3850
Ward room freeze	Pulldown, Alternate "B"	428
Chill No. 1	Pulldown, Alternate "B"	1027
Chill No. 2	Pulldown, Alternate "A"	1700