A Design Study for a Refrigeration Plant

by

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Nomenclature

\[ H_a = \text{Heat Gain, infiltrating factor air, Btu/cu. ft} \]
\[ h_a = \text{specific enthalpy, Btu/lb} \]
\[ R = \text{Relative humidity, percent of entering dry air} \]
\[ V_h = \text{Volume of moist air, cu. ft/lb} \]
\[ W_{s1} = \text{Humidity ratio at saturation, lbs water/lb dry air} \]
\[ W_{s2} = \text{Humidity ration at saturation final, lbs water/lb dry air} \]
\[ H_{inf} = \text{Heat gain by infiltrating, Btu/hr} \]
\[ T = \text{Temperature, F} \]
\[ H_T = \text{Total Heat Transfer, Btu} \]
\[ U = \text{Overall Heat Transfer Coefficient, (h * ft2 * °F)/BTU} \]
\[ R_1 = \text{Respiration rate, entering Btu/lb./24 hours} \]
\[ R_2 = \text{Respiration rate, final Btu/lb./24 hours} \]
\[ H_{resp} = \text{Heat gain by respiration, Btu/hr} \]
\[ Twp = \text{Total product weight, lb} \]
\[ Twpc = \text{Total product container weight, lb} \]
\[ K = \text{Thermal Conductivity, (Btu * in)/(h * ft2 * F)} \]
\[ X = \text{Thickness, inches} \]
\[ q = \text{heat loss, Btu/hr} \]
\[ H_w = \text{specific enthalpy of condensed water, Btu/lb water} \]
\[ H_h = \text{Enthalpy of final moist air at saturation per pound dry air, Btu/lb} \]
\[ H_{ah} = \text{Difference of enthalpy of moist air vs. dry air, Btu/lb dry air} \]

Definitions for clarity:

Refrigeration Unit- The entire refrigeration unit design which consists of the chiller and freezer stores (boxes) , refrigeration plant, and associate areas for refrigeration plant components (evaporator coil/refrigeration plant space).

Stores- Storage area for fresh or frozen food products also known as boxes.

Refrigeration Plant – This is defined as all components that make up the plant,( i.e.) evaporator coil, compressors, thermal expansion valves, receiver tank, condenser etc.)
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Abstract

The purpose of this project was to design a reliable refrigeration unit suitable for both a military and non-military ship application. This includes sizing the refrigeration unit based on a selected number of personnel and required frozen and chill food products. This project also includes a layout examination of the various components, a discussion of materials used for box construction, and also a description of commercial components such as condensers, compressors, receiver tanks, and thermal expansion valves. Furthermore, calculations were performed to determine stores size, cooling capacity, and to examine the transfer of thermal energy between the rear chiller stores wall and refrigeration plant wall to determine if the chosen insulation is sufficient for the design. The results of this project show that for a crew size of 120, the overall size of the chiller stores and freezer stores is 486 ft$^3$ and 726 ft$^3$, respectively. The cooling capacity required for the refrigeration unit to remove the calculated heat load is .50 tons of refrigerant. Lastly, it was determined that 4 inches of Armaflex insulation applied to the rear chill wall was a satisfactory amount to maintain the rear chiller stores wall above 32°F to avoid food loss due to freezing.
1.0 Introduction

1.1 Background

The need for food preservation of perishable products is one of the most common applications where mechanical refrigeration is used. The design of the storage and required capacity of the unit is critical for at sea storage since these products may have to be preserved for long periods of time. In the earlier days of refrigeration plants, the designs were bulky, inefficient, and relatively expensive to maintain. These factors limited the use of mechanical refrigeration plants for storages on ships. In the beginning of the 20th century advanced manufacturing methods were developed, safe refrigerants were utilized, and small electrical motors that were developed; all of these helped produce smaller more efficient units. This technology was also carried to both commercial and military ships.

Marine refrigeration units that are used on commercial and military ships are primarily used to preserve and store food product in chilled and frozen stores. These refrigeration units consist of storage room(s), refrigeration plant(s), fans, duct work, control panel and a means for defrosting (dependent on a hermetically sealed unit vs. mechanical joints (serviceability)). A simple refrigeration plant consists of several components: (see the figure 1) such as a compressor, condenser, evaporator coils, thermal expansion valve, receiver tank/accumulator and heaters which are all sized based on the capacity needed to reduce and maintain temperatures within the storage rooms. The capacity of a refrigeration plant is determined by several factors which include: 1) storage size based on number of personnel; 2) associated volume of food required; 3) associated arrangement; 4) heat load; 5) refrigerant.
Figure 1.
Basic Refrigeration Schematic
Archnews, The Basic Refrigeration Cycle Reference, June 30, 2006,
<http://www.achrnews.com/Articles/Technical>

2.0 Problem Statement

2.1 Objective

The purpose of this project was to design a reliable refrigeration unit suitable for both a military and non-military ship application. This included the following:

- Determination size of stores required for food product based on number of personnel
- Creation a suitable arrangement for stores (square) and minimized foot print of entire unit
Calculation of the cooling capacity required and examination of thermal heat transfer between the rear chiller and refrigeration plant walls to determine if the chosen insulation is acceptable

- Description of commercial components such as condensers, compressors, receiver tanks, and thermal expansion valves, control panels for conceptual design
- Discussion of simple controls for interface and plant safety
- Discussion of the serviceability of the refrigeration plant components

2.2 Previous Work

In order to design a refrigeration unit, one must understand the concept of a refrigeration cycle, equipment, and various types of applications that exist. For this project, several articles, texts, and design guidance were reviewed to get an understanding of current technology. Refrigeration plants are nowadays used onboard Marine vessels for food preservation and storage in cooler temperatures and for potentially producing ice required for onboard consumption. Several calculations must be conducted in order to properly size a refrigeration plant including storage size and capacity. Most refrigeration plants on ships range from 1 to 14 ton units. Other considerations that were found included:

- Refrigerant
- Arrangement size limitations
- Electrical power requirements
- Cooling water source
- Exterior temperatures
- External Environment
- Weight
- Military requirements for testing

Over the last decade, refrigerant types like R-22 and R-114 that were widely available are now being phased out due to environmental concerns. The refrigerant of choice today for military applications is R-134a due to it being categorized as environmentally
friendly and readily available. Since space is limited on military applications, it’s necessary that the refrigeration unit be designed to meet the accommodations of the crew and take up the least amount of space available while still providing access for maintenance. Most refrigeration plants can be located on the forward, mid, or aft end of the ship depending the design layout. Another important design aspect is that the military requirement that refrigeration plants must have easy access to the refrigeration plant components for maintenance and repair (see Figure 2). The Figure 2 sketch depicts a standard refrigeration chiller stores design. This sketch shows an access door and pressure equalization valve on the exterior of the stores. The interior design includes free standing shelves, bulk head lights, modular floor, sealed joints, and portable refrigeration plant on the backside.

Figure 2.
Accessible Military Refrigeration Unit
<http://www.christalsservices.com/coldrooms.html>
In addition to the overall layout and maintenance, several conditions such as environmental conditions, at sea conditions, and potential hazards to the refrigeration plant once deployed during a mission (shock event, vibrations, and inclinations) must be considered during the design phase. However, it should be noted that this paper does not discuss qualification of these components in a military application due to the nature of the technical details required to be provided.

The main components that are commonly used in military applications include compressor/motors, thermostatic expansion valves, condensers, evaporator assemblies, piping and associated fittings, pressure equalization valve, evaporator coils or cooling tubes, and heating elements for defrost heaters. These components are thoroughly evaluated once combined in a system to ensure reliability.

### 3.0 Methodology

#### 3.1 Size of Storage

The determination of the size of the refrigeration storage areas (also referred to as stores) are based mainly on identifying the following; 1) crew size; 2) duration of voyage; 3) amount of food consumed by each crew member (volume) factor. For this design, a crew of 120 members was chosen to size both the freezer and chilled stores. This number is based on an average number of crew members from the VIRGINIA, OHIO, and LOS ANGELES class. A duration of 60 days for fresh perishables for the chiller stores and 90 days for the freezer stores was established with a volume factor of 5 percent to size the stores. Based on the aforementioned, it is first necessary to calculate the net volume of food using equation 1.

\[
Net\ Volume\ of\ Food\ (ft^3) = (\text{crew size}) \cdot (\text{voyage duration}) \cdot (\text{volume factor})
\]  \[1\]

Based on the net volume of food required and the sizing information in section 3.1, the deck area can be calculated. It is standard practice to divide the net volume of food required by a utilization factor which account for crew member consumption.

\[
Total\ Volume\ of\ Food\ (ft^3) = \frac{Net\ Volume\ of\ Food}{utilization\ factor}
\]  \[2\]
It is standard practice in refrigeration stores design to assume a standard height of 6ft to minimize arrangement area impacts and to reduce crew members from stacking food to high. It should be noted that the standard height takes into consideration the ceiling grating above, thus reducing the stack height to ensure proper air flow.

\[ \text{Deck Area for Food} = \frac{\text{Total Volume of Food}}{\text{Height}} \]  

Based on equation 3, the stores length and width was found by taking the square root of the deck area of food for each box then adding the wall thickness.

### 3.2 Refrigeration Stores and Plant Arrangement

The refrigeration unit design consists of the chiller stores and freezer stores with a common refrigeration plant. The stores were sized based on the equations defined in section 3.1 and as shown in Appendix A. An engineering sketch provided in Appendix B shows the overall layout of the refrigeration unit. The refrigeration unit consists of two independently operated plants for redundancy (in the event of one failure). Each plant contains a compressor, condenser, receiver tank, thermal expansion valve, evaporator coil, air fans, and heaters (for defrost cycle), and capacity controls for human interface/command.

Given the arrangement, the refrigeration plants can be accessed from all three exterior walls for maintenance. The inside of the refrigeration plant consists of two separate areas. In the lower half of the refrigeration plant box, the refrigeration components will be packaged with the minimum amount of mechanical joints in order to reduce the possibility of refrigerant leaks. The lower part of the refrigeration plant will be thermally isolated from the evaporator coils, heaters, and fans in order to ensure that the cooling water does not freeze in the lower half of the refrigeration plant.

The freezer stores will have an air plenum on the front side of the refrigeration box to allow the fans to push cold air into the freezer stores. The refrigeration plant will consistently produce cold air between the temperatures of 0 +/- 2°F. A return duct will return air to the evaporator coils by a means of suction created by the fans. The chiller stores will have an electromechanically actuated damper on the front side of the
refrigeration plant box wall. This will supply cool air into the chiller stores. The design range temperature for the chiller stores is approximately 32°F to 37°F. A return air duct will be located in the upper left section of the rear chiller wall to allow a path for return air. Depending on the users input, an automatic defrost cycle will initiate to melt the formed ice on the evaporator coils. This cycle will run for a minimum of 20 minutes or until a temperature of 85°F is met on the surface of the evaporator coils. A drain system will be used to collect water from the defrost cycle.

### 3.3 Heat Load Analysis

The heat load must be calculated independently for each box in order to determine the overall capacity required for the refrigeration plant (in tonnage). The assumptions for surrounding temperatures, stores space design temperatures, and number of air changes are shown in Appendix D. These assumptions are based on the guidance provided in the design data sheet referenced in section 6.0. In addition, it was assumed that the walls and ceilings were made of extruded aluminum panels and the floor was made of 2” thick aluminum with a steel base bedplate for rigidity.

The following equations were used to calculate each heat load present for both the frozen stores and chiller stores independently. Heat infiltration is defined as the uncontrolled entry of outside air that directly comes into contact with the refrigerated space. This commonly occurs due to wind and stack effects. A wind effect refers to the entry of outside air due to a pressure difference developed between the stores. The stack affect refers to the entry of outdoor air due to buoyancy effects caused by the difference in temperature between the refrigerated space and outside space. Since refrigeration plants are usually located in the ships galley area, it would be nearly impossible to eliminate these conditions. A pressure equalization valve can be installed; however, for this study it was assumed that additional heat loads due to infiltration were still present. The following equation taken from the design data sheet referenced herein, was used to calculate the infiltration factor as follows:
The heat load due to infiltration was then calculated using:

\[
H_{inf} = \text{net volume food} \cdot H_a \cdot \text{no. air changes} / 24
\]  

Following the calculation of heat infiltration, the heat transfer from the walls, floor, and ceiling was calculated by using the following equation for a 24 hour period:

\[
H_T = T \cdot \text{Area}_x \cdot U \cdot 24 \text{hr}
\]

In addition to these heat loads, heat respiration from the food was then calculated. It should be noted that the food heat respiration is for fresh perishables only which is applicable to the chiller stores and not freezer stores. Heat respiration is defined as the release of energy from food that takes place in the cells of the organism. As the food continues to break down more energy is transfer thus causing an additional heat load until the food reaches an equilibrium temperature. Food heat respiration is a considerable heat load that must be efficiently removed from the system to prevent localized increases in temperature in any one area of the box. For this design, the cooling capacity required will increase for the overall plant. Guidance will be given to the crew in order to keep the produce cool while transferring and to ensure that the damper air way is not blocked. Thus maximizing circulation of cooling air.

\[
H_{inf} = T_{wp}(R_1 + R_2)/3
\]

\[
H_{resp} = (T_{wpc} + T_{wcc}) \cdot (T_1 - T_2)/3
\]

It was assumed that the heat load from components in the lower half of the refrigerant plant space were negligible. The total heat load calculated is the summation of the heat loads discussed in the section. The heat load is then converted to tonnage.
3.4 Insulation Analysis (Rear Chiller Stores Wall)

The rear chiller stores wall is adjacent to the refrigerant plant and evaporator coil space (see Appendix C for component space layout). This is an area of concern for freezing since the chiller stores temperature is approximately 33°F and the evaporator coil space is approximately -10°F. The wall design includes a batten (see figure 4) to provide a one inch separation for air flow and reduce contact with the wall, however, if moisture is introduced it could potentially freeze the wall. Through conduction, this could effect the operation of the air damper. This area of concerned must be analyzed to ensure proper operation of the plant. Steps one through twelve were used to determine the minimum insulation required to ensure the chiller stores wall is above 32°F.

The following temperatures were assumed: 1) The evaporator coil space temperature was -10 °F; 2) chiller stores temperature was 32°F; 3) Refrigeration plant space temperature was 60°F. The overall heat coefficient is calculated by determining the resistant to the flow of heat by the materials contained within the wall. The following equation was used to calculate the individual heater transfer coefficient for the chiller stores wall, the insulation for evaporator space and the refrigeration plant space:

\[ h_{wall} = \frac{k}{x} \]  

[9]

The overall heat transfer coefficient is then calculated by:

\[ U = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2} + \frac{1}{h_3} + \frac{1}{h_4}} \]  

[10]

The heat transfer rate can then be calculated by:

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

[11]

Following the completion of eq. 12, the wall temperature is calculated by:

\[ T_{wall} = T_{air} \cdot (q \cdot R_a \cdot A) \]  

[12]

This value is used to determine if the suggested insulation is sufficient enough to ensure the wall does not freeze. This is further discussed in the results section herein.
4.0 Results and Discussion

4.1 Final Size Stores Arrangement and Construction

As previously discussed, the final chiller stores and freezer stores sizes are determined by the equations in section 3.1 and calculated in Appendix A. The chiller stores room has an interior volume of 486 ft\(^3\). The freezer stores room has an interior volume of 726 ft\(^3\). The stackable height is 6 ft. The overall size of the refrigeration unit is shown in appendix B. An engineering sketch which provides a high level component layout is shown in Appendix C.

The chiller stores and freezer stores walls are constructed with extruded aluminum so that a polyurethane insulation can be placed in between the walls (similar to foam insulation). The chillerbox and freezer box consists of welded seams to ensure no moisture can penetrate between the floor, ceiling, and walls. The ceiling and floor consists of extruded aluminum utilizing the same insulation. The floor is reinforced by adding a steel base plate to the exterior floor so that the unit can be transported and moved easily into a location. In addition, both rooms will have removable aluminum floor grates to provide ventilation and drainage. This will help facilitate maintainability (cleaning) of the store rooms. Each stores room will also contain a batten (1 inch stand off, see figure 4 in section 4.4) to allow for proper air circulation.

Two access doors, one for each stores room are located farthest away from the entrance points of cool air. The doors can be opened from both the inside or outside of the stores room. The doors are designed to have an inside safety release even when door is locked from the outside. To ensure the doors do not freeze shut, a heater strip that requires less than 120V will be placed on door contact points. Due to the low demand of power, the heater strip will remain on to ensure the door remains unaffected. In addition to the heaters, two lights will be mounted above the ceiling grates, still visible, one on each side of the stores room.
As discussed in the military refrigeration article referenced herein, it is advised to add a pressure equalization valve to the design. This would allow each room to maintain a zero pressure differential between the inside and outside of the rooms to help provide a minimal heat loss and reduce the wind and stack effects discussed in section 3.3. However, it should be noted that these effects were not taken into consideration in the design calculations to maintain a conservative approach.

4.2 Component Details

The following section describes the components used in the refrigeration plant. This design integrates two refrigeration plants inside one refrigeration plant space for redundancy. Figure 3 defines the low and high side of pressure during the refrigeration plant cycle. This will further help the reader understand the subcomponents described in this section (high and low pressure switches, etc).

Figure 3
Refrigeration Cycle Pressure Defined
Jennifer Fonder, Basic Refrigeration Cycles, Web Enhanced Course Materials
http://www.swtc.edu/Ag_Power/air_conditioning/lecture/basic_cycle.htm
One of the main components in the refrigeration plant is the refrigeration compressor. Hermetically sealed scroll type compressor are typically used in ship applications. The compressor requires a crank case heater for oil lubrication. The crank case heater ensures the oil remains viscous enough to lubricate the compressor. This also ensures liquid refrigerant does not build up in the oil which could potentially damage or degrade the compressor over time. High and low pressure switches would be equipped on the compressor to determine suction and discharge pressures. This will also monitor operation and facilitate shutdown of unit prior to damage in the event of refrigerant loss or cooling water loss. This will be further discussed in the capacity control system in section 4.5.

The condenser is also another important part of the refrigeration plant. The condenser uses cooling water to condense the refrigerant from a high pressure, high temperature gas into a high pressure liquid. The condenser is designed for continuous operation which is subject to condenser water temperatures above 90°F.

For this design, a single evaporator coil with two different paths (dual circuits) will be developed to minimize room. When liquid refrigerant reaches the evaporator coils, the refrigerant pressure is reduced dissipating its heat content, making its temperature less than the air flowing around it. This process causes the refrigerant to absorb heat from the warmer air and reach its boiling point. The refrigerant then vaporizes absorbing the maximum amount of heat. This heat flows back to the suction side of the compressor then discharges through the high pressure side to the condenser. The evaporator coil structure which holds the coil tubes in place will also contain heaters for the defrost cycle which will be further discussed in the capacity control section of this report (see section 4.5). Thermal expansion valves on the evaporator coils are located in each circuit. The valves adjust by their own thermal element. The purpose of the thermal expansion valve is to maintain a preset amount of superheat in the refrigerant leaving the evaporator coils. This prevents liquid refrigerant from returning to the compressor while still supplying the evaporator with enough refrigerant to satisfy the design load conditions.
The cooling water system consists of position valves common to both plants which provide cooling water (from an outside source) to the condenser. These valves either divert or supply water to each plant. The water regulating valve located in the cooling water system piping maintains a constant refrigerant condensing pressure by passing part of the condenser inlet water around the condenser directly to the outlet of the system piping. The valve is refrigerant pressure actuated and operates in response to the condensing pressure. The water regulating valve pressure sensing line has a shut off to isolate potential leaks.

A liquid receiver tank will be installed for each refrigeration plant. A refrigerant filter is located in line leaving the receiver tank to remove any moisture that may have built up during shutdown.

Service valves are installed on the unit to remove and/or add additional amounts of refrigerant. This also acts as an interface to perform tests that ensure the system is sealed so that no leaks occur.

4.3 Calculated Heat Loads
The following results show the rate at which the heat must be removed from the refrigerated space in order to maintain the required temperatures. Based on the material properties of aluminum, insulation (polyurethane and armaflex), stores temperatures, and ambient temperatures surrounding the exterior of the refrigeration plant the following parameters were calculated as listed in below.
Table 1. Capacity Required

Table 1 shows that the minimum required capacity of the refrigeration plant needs to be 0.52 tons to effectively cool the store sizes provided in section 4.1. The capacity required is based on analyzing the transmissions, infiltration, precooling, and respiration analysis as calculated in Appendix E. This calculation assumes that the heat generated by the components is negligible since they are separated from the evaporator coil space and that the adjacent walls are properly insulated as verified in section 4.4 herein. Most military applications require additional cooling capacity based on higher ambient temperatures, crew size, and utilization factors; however, this provides a cooling capacity that is obtainable to cool both the chiller stores and freezer stores space within a 24 hour period.

4.4 Rear Chiller Stores and Freezer Stores Wall Insulation

The refrigeration plant and evaporator coil space are adjacent to the rear chiller stores and freezer stores wall. Since the temperature in the chiller stores can be as low as 32°F and the evaporator coil space may be below -10°F, the rear wall must be analyzed. The purpose of this calculation is to ensure that the correct amount of insulation is present. This will ensure that if moisture is introduced into the chiller stores that heat transfer...
across the wall does not allow moisture to build up and freeze against the wall. This ice build up can also affect operation of the damper that allows cool air to enter the space. In addition, this will cause freezing of fresh perishable foods which would lead to food loss. Using the methodology explained in section 3.4, it is deemed necessary to utilize a 4” thick armaflex insulation. Armaflex insulation was chosen due to the fact that it is used in different platforms across the military. Armaflex insulation is a non-halogenated, chlorine-free elastometric insulation which improves efficiency and is environmentally friendly since its chemical compound off gases less in the event of a fire. This insulation has gone through several Military and ASTM specification tests and is approved by the Military. The cross section of the wall is shown in Figure 4.

![Figure 4. Cross Section of Chiller Wall](image)

Design Data Sheet 9590-1, Refrigerating Equipment for Storage Compartments-Heat Load Calculation and selection; 1967

The resultant temperature of the rear chiller stores wall was calculated to be 35°F. This is above the freezing point of water (32°F), see table 2. As discussed in the methodology section, the refrigeration plant space is separate of the evaporator coil space and is assumed to be fully insulated to ensure the cooling water in the condenser and pipe line does not freeze, therefore, reducing the overall heat transfer to the wall.
<table>
<thead>
<tr>
<th>$q_{total}$ (BTU/hr)</th>
<th>$T_{freeze}$ (F)</th>
<th>$T_{chill~wall}$ (F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.52</td>
<td>-10</td>
<td>35</td>
</tr>
</tbody>
</table>

Table 2.
Temperature of Rear Chiller Wall

In addition to this insulation, a recommendation to further evaluate the use of a partition strip curtain in the behind the door access is recommended. This will require further testing to ensure that the reduction in moisture and heat infiltration is effective compared to the burden of the crew while loading and unloading the boxes. The curtain partition is depicted in Figure 5.

Figure 5.
Partition Strip Curtains
40 x 80 Strip Curtains, 2006
http://www.etundra.com/40__x_80__Reinforced_Strip_Curtain_Set-P1005.html?token=3307|||0|24|7|1|0
4.5 Capacity Controls

The control system was developed to have a simple human interface design that allows the option of choosing which refrigeration plant to use and how to operate the overall unit. Temperature controls includes the chiller and freezer stores temperature controls. Once the set point is adjusted, these controls automatically maintain the rooms design temperature. The location of the control panel is shown in Appendix C. This is accomplished by having a chiller stores and freezer stores thermostat. The chillerbox thermostat is a microprocessor based programmable thermostat that utilizes a temperature probe in the chiller stores to determine the overall temperature. Depending on the setpoint of the thermostat, the chiller stores damper will open or close allowing the temperature to become high or lower. Similar to the chiller stores thermostat, a thermostat exists from the freezer stores. The freezer stores uses the same platform but with a different programmed setpoint. A temperature probe is also located in the freezer stores room which provides feed back to the controller. In the event the temperature is met in the freezer stores, the refrigeration plant shuts down until the temperature rises above the design temperatures. It should be noted that since the refrigeration plant is shared by both the boxes, this does not affect performance. If the temperatures was not within specification in the chiller stores but was in accordance with the freezer stores design temperatures then a failure of the damper may have occurred.

The refrigeration plant itself has its own capacity control system which includes the following components:

1. Safety switches. Operational safety switches are implemented in the design to ensure that the refrigeration plant shuts down before it becomes damaged. Each compressor has a high and low pressure switch. The high pressure switch is a pressure-operated safety device that protects the compressor and crew from damage or injury that would result in equipment failure caused by high pressure condition. There is a visual indication (LED light and label) on the control panel interface to alert the crew that the system shut down occurred due to a high pressure event. The switch must be manually reset once the
pressure falls within the set points of the system. The light would then extinguish. The low pressure switch similar to the high pressure switch mechanism will turn off the refrigeration plant. A minimum of 5 minutes is required to reset the refrigeration plant so that damage does not occur to the compressor by starting and stopping it frequently. The cooling water supply line has a flow switch which prevents the system from operating when not enough supply water is flowing through the cooling water supply lines to the condenser. Upon failure of cooling water supply, a visual indication (LED and label) on the control panel will alert the crew that the system shutdown due to low cooling water and action must be taken. A manual switch must be reset upon establishing the correct amount of water is being supplied to the refrigeration plant.

2. Defrost Cycle Timer. The system has been designed to have an automatic defrost cycle to defrost the ice build up on the evaporator coils. This sub system will automatically defrost the unit every 6 hours or upon user requests. This sub system will also turn off the refrigeration plant and fans, so that heat does not enter the chiller stores or freezer stores. This cycle is automatically terminated after 10 minutes or when the temperature in the evaporator coil space reaches above 85°F. The refrigeration plant automatically turns back on and begins the refrigeration cycle.

3. Solenoid Valves. The system has been designed to utilize solenoid valves to shut in the low pressure line which allows the compressor to pump the remaining refrigerant into the receiver tank. This will stop refrigerant from migrating around the refrigeration plant piping. This practice is used widely in the industry since this approach uses existing safety features and does not require additional system controls to shut the plant down.

4.6 Serviceability

The unit was designed so that the refrigeration plant and evaporator coil space are accessible from three sides. This provides the maximum amount of space to work on the refrigeration plant for repair. The majority of the components chosen to be a part of the design have a minimal amount of maintenance required. The unit will require that the
crew visually inspect the evaporator coil space and refrigeration plant at least once a month. A refrigerant hand held detector can be used to check for any potential leaks in the piping which could have been caused by excessive vibration during operation. Industry standard refrigerant leak detectors have a detection rate of .5 ounces; therefore, this could be the basis of determining the maximum allowable leak rate per fitting of .5 oz a year. Depending on the final design of the system piping, there could possibly be twenty to thirty mechanical joints. This would help determine the periodicity of recharging the refrigeration plants.

Since it is nearly impossible to have refrigeration plant to be fully sealed (similar to a home refrigerator), the fewest amount of mechanical joints were utilized in this design. This would also include sealing the dehydration filter (removal of moisture from refrigerant). In addition, the service valves that provide the connections to the inlet and outlet of the compressor are brazed, all system piping is brazed, and any connections to isolation valves and or solenoids are brazed.

All components have been chosen to operate at a 10 year minimum or 87,600 hours. At this point, it would be recommended to replace the compressor and motor, any non-metallic seals, and valve inserts. This would be further evaluated when all components were chosen, integrated, and endurance tested for ship use.

5.0 Conclusion

The design of a refrigeration unit touches on many aspects of mechanical engineering such as thermodynamics, fluids, and mechanics. When designing a refrigeration unit, the design engineer must first understand the basic requirements which include: 1) number of personal; 2) location; 3) environment; 4) user interface; 5) maintenance.

This project required the investigation of current military and non military refrigeration applications. Several resources from the Rensselaer Polytechnic Institute (RPI) Cole
Library publications were used to determine current refrigeration plant designs and how refrigeration plants are used in the military. Furthermore the referenced texts provided design guidance on how perform calculations to determine overall size, capacity, and verification of chosen insulation (based on its thermal properties) for an overall refrigeration plant design.

This project examined the box construction and arrangement for a design for both a chiller stores and freezer stores with a common refrigeration plant. The overall size of the boxes were calculated based on a crew size of 120 people. Based on the stores size of $486 \text{ ft}^3$ and $726 \text{ ft}^3$, the heat load for the refrigeration plant was calculated to determine the required capacity is .50 tons. The rear chiller stores wall was examined to ensure that the temperature of the wall did not fall below $32\degree\text{F}$ so that food loss did not occur. It was found that the wall temperature with the armaflex insulation was approximately $35\degree\text{F}$.

In addition to the research of previous developed refrigeration applications and design calculations this report discusses major components of the refrigeration plant, controls, and serviceability as well as how they would be implemented into a refrigeration unit design. Lastly, this project also provides the reader with a better understanding of the design process and component integration of a ship board refrigeration unit.
6.0 References

6. CP Arora, Refrigeration and Air Conditioning (Tata McGraw-Hill, New Delhi, 2001) pgs 659-663
11. Basic Refrigeration Cycle, Power Web Enhanced Course Material, 2006,
    <http://www.swtc.edu/Ag_Power/air_conditioning/lecture/basic_cycle.htm>
Appendix A

Box Sizing Calculation
**Purpose:** This calculation was developed to determine the size required to store fresh perishables and frozen goods for 120 people. The methodology and explanation of assumptions is defined in section 3.1 of this report.

**Assumptions:**
120 people
60 day trip for chiller stores
90 day trip for freezer stores
Volume Factor of .05
Utilization factor of .75
Maximum Height for Stowage = 6 ft
Length=Width for Box Size

**Calculation:**

Determine Net Volume of Food Required:

Chiller Stores:

\[
Net \ Volume \ of \ Food \ (ft^3) = (crew \ size) \cdot (voyage \ duration) \cdot (volume \ factor)
\]

\[
Net \ Volume \ of \ Food \ (ft^3) = (120) \cdot (60) \cdot (0.05 \ ft^3)
\]

\[
Net \ Volume \ of \ Food \ (ft^3) = 360 \ ft^3
\]

Freezer Stores:

\[
Net \ Volume \ of \ Food \ (ft^3) = (crew \ size) \cdot (voyage \ duration) \cdot (volume \ factor)
\]

\[
Net \ Volume \ of \ Food \ (ft^3) = (120) \cdot (90) \cdot (0.05 \ ft^3)
\]

\[
Net \ Volume \ of \ Food \ (ft^3) = 540 \ ft^3
\]

Determine Total Volume of Food Required:
Chiller Stores:

\[ \text{Total Volume of Food (ft}^3\text{)} = \frac{\text{Net Volume of Food}}{\text{utilization factor}} \]

\[ \text{Total Volume of Food (ft}^3\text{)} = \frac{360}{.75} \]

\[ \text{Total Volume of Food (ft}^3\text{)} = 480 \text{ ft}^3 \]

Freezer Stores:

\[ \text{Total Volume of Food (ft}^3\text{)} = \frac{\text{Net Volume of Food}}{\text{utilization factor}} \]

\[ \text{Total Volume of Food (ft}^3\text{)} = \frac{540}{.75} \]

\[ \text{Total Volume of Food (ft}^3\text{)} = 720 \text{ ft}^3 \]

Determine Deck Area for Food

Chiller Stores:

\[ \text{Deck Area for Food} = \frac{\text{Total Volume of Food}}{\text{height}} \]

\[ \text{Deck Area for Food} = \frac{480 \text{ ft}^3}{6 \text{ ft}} \]

\[ \text{Deck Area for Food} = 80 \text{ ft} \]

Freezer Stores:

\[ \text{Deck Area for Food} = \frac{\text{Total Volume of Food}}{\text{height}} \]

\[ \text{Deck Area for Food} = \frac{720 \text{ ft}^3}{6 \text{ ft}} \]

\[ \text{Deck Area for Food} = 120 \text{ ft} \]

Box Dimensions:

Chiller Stores:

\[ \text{Length} = \text{width} = \sqrt{80\text{ ft}} \]

\[ \text{Length} = \text{width} = 9 \text{ ft} \]

\[ \text{Height} = 6 \text{ ft} \]
Freezer Stores:

\[ \text{Length} = \text{width} = \sqrt{120} \text{ ft} \]
\[ \text{Length} = \text{width} = 11 \text{ ft} \]
\[ \text{Height} = 6 \text{ ft} \]

Conclusion: Boxes will be sized by the aforementioned dimensions. The overall size will include wall thickness and packing factor of .75. This will be shown in the arrangement drawing.
Appendix B
Refrigeration Plant Layout
Refrigeration Stores Layout

Dimensions are in inches

- Angular Mach: ± Bend ±
- Two Place Decimal: ±
- Three Place Decimal: ±

Drawing: KWH
Date: 11/1/10

Size: DWG. NO. A R-134A-1
Rev: -
Scale: 1:10

Proprietary and Confidential
The information contained in this drawing is the sole property of [Insert Company Name Here]. Any reproduction in part or in whole without the written permission of [Insert Company Name Here] is prohibited.
3-D Engineering Sketch Model
Appendix C

Refrigeration Stores Arrangement
Appendix D

Heat Load Calculation
**Purpose:** This calculation was developed to determine the heat load for both the chiller and freezer stores. This calculation will define the capacity required to pull down the temperature of the box.

**Assumptions:**
1. 120 people
2. Ambient Temperature of 77F
3. Chillerbox Design Temperature 33F (average)
4. Freezer Design Temperature 0F (average)
5. Heat Infiltration
6. Heat Respiration due to Fresh Perishables
7. Neglect Fan Heat since under 80Watts
8. Negligible heat transfer from additional components in refrigeration plant since well insulated and separate for evaporator coil space.

Variables defined in front matter of project

**Calculation:**

Chillerbox Infiltration Factor

\[ H_a = h_a + RHH_{wh} - \frac{h_h + h_w (RHW_{x_1} - W_{x_2})}{V_h} \]

\[ H_a = 20.424 \frac{btu}{lb} + 0.5 \cdot 29.02 \frac{btu}{lb \text{ dry air}} - [12.17 \frac{btu}{lb} + 1.03 \frac{btu}{lb \text{ water}}] \cdot (0.5 \cdot 0.026433 \frac{\text{water}}{\text{lb dry air}} - 0.003947 \frac{\text{water}}{\text{lb dry air}}) / 12.492 \text{ ft}^3 \]

\[ H_a = 1.821561933 \]

\[ H_a = 1.82 \]

Chillerbox Infiltration

\[ \text{Infiltration} = \text{Chillerbox Size} \cdot H_a \cdot \text{airchanges} \]

\[ \text{Infiltration} = 486 \text{ ft}^3 \cdot 1.82 \cdot 12 \]

\[ \text{Infiltration} = 10614 \text{ BTU} \]
Freezer Stores Infiltration Factor

\[ H_a = 20.424 \text{btu/lb} + 0.5 \times 29.021 \text{btu/lb dry air} - \left[ \frac{835 \text{btu/lb} - 158.89 \text{btu/lb water}}{0.5 \times 0.026433 \text{water/lb dry air} - 0.0007875 \text{water/lb dry air}} \right]/111.594 \text{ft}^3 \]

\[ H_a = 3.111466604 \]

\[ H_a = 3.11 \]

Freezerbox Infiltration

\[ Infiltration = \text{Freezerbox Size} \times H_a \times \text{airchanges} \]

\[ Infiltration = 726 \text{ft}^3 \times 3.11 \times 12 \]

\[ Infiltration = 27904 \text{BTU} \]

Chillerbox Transmissions

\[ Transmissions = T(F) \times A(\text{ft}^2) \times U(\text{BTU/sqft/F/hr}) \times 24 \text{hr} \]

\[ Transmissions \text{ ceiling} = 85 \times 9.9 \times 0.94 \times 24 = 6609.6 \]

\[ Transmissions \text{ floor} = 85 \times 9.9 \times 0.94 \times 24 = 6609.6 \]

\[ Transmissions \text{ walls} = 4 \times 85 \times 6 \times 0.94 \times 24 = 17625.6 \]

\[ \text{Chillerbox Transmissions} = 30845\text{BTU} \]

Freezerbox Transmissions

\[ Transmissions = T(F) \times A(\text{ft}^2) \times U(\text{BTU/sqft/F/hr}) \times 24 \text{hr} \]

\[ Transmissions \text{ ceiling} = 85 \times 11.1 \times 1.04 \times 24 = 9873.6 \]

\[ Transmissions \text{ floor} = 85 \times 11.1 \times 1.04 \times 24 = 9873.6 \]

\[ Transmissions \text{ walls} = 4 \times 85 \times 11.6 \times 1.04 \times 24 = 21542.4 \]

\[ \text{Freezerbox Transmissions} = 41290\text{BTU} \]

Chillerbox Precooling

\[ \text{Precooling} = (\text{Total product weight} + \text{total container weight}) - (\Delta T)/\text{days required} \]

\[ \text{Precooling} = (240 \times 26 + 240 \times 3.2) - (55 - 32)/3 \]
Pr_{ecooling} = 90094BTU

Chillerbox Precooling
Pr_{ecooling} = (Total product weight + total container weight) – (ΔT) / days required
Pr_{ecooling} = (486 \cdot 26 + 486 \cdot 3.2) – (55 – 32) / 3
Pr_{ecooling} = 90094BTU

Freezerbox Precooling
Pr_{ecooling} = (Total product weight + total container weight) – (ΔT) / days required
Pr_{ecooling} = (726 \cdot 37 + 726 \cdot 3.6) – (10 – 0) / 3
Pr_{ecooling} = 41479BTU

Total BTU for Chillerbox = 131,905.5 BTU
Total BTU/HR for Chillerbox = 5496 BTU/HR
Tonnage = 5496/12000 = .46 Tons

Total BTU for Freezerbox = 111,677 BTU
Total BTU/HR for Chillerbox = 4653 BTU/HR
Tonnage = 4653/12000 = .39 Tons

Total Capacity required .46 Tons
Margin allowable (commercial 10%) = .50 Tons
Appendix E

Rear Chillerbox Wall Temperature
**Purpose:** This calculation was developed to analyze the rear chillerbox wall to ensure that the wall temperature is above 32 F since there is a large differential between the chillerbox temperature and the evaporator coil space.

**Assumptions:**

- Evaporator coil temperature -10F
- Chillerbox temperature 33F
- Refrigeration plant space temperature 60F

**Calculation:**

Table 1.

<table>
<thead>
<tr>
<th>Component</th>
<th>Temperature</th>
<th>°F</th>
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</thead>
<tbody>
<tr>
<td>Freezer Box Temp.</td>
<td>$T_{freeze}$</td>
<td>-10</td>
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<tr>
<td>Chiller Box Temp</td>
<td>$T_{chill}$</td>
<td>33</td>
</tr>
<tr>
<td>Condenser Box Temp</td>
<td>$T_{Cond}$</td>
<td>60</td>
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<table>
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<tr>
<th>Insulation Type</th>
<th>Armaflex</th>
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<tr>
<td>Insulation thickness $X_1$</td>
<td>4 Inches</td>
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<tr>
<td>Thermal Conductivity $K_1$</td>
<td>0.035 $(\text{BTU<em>in})/(\text{h</em>ft}^2*\text{°F})$</td>
</tr>
<tr>
<td>R Value $R_1$</td>
<td>114.286 $(\text{h<em>ft}^2</em>\text{°F})/\text{BTU}$</td>
</tr>
</tbody>
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<table>
<thead>
<tr>
<th>Wall Material</th>
<th>Aluminum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall Thickness $X_2$</td>
<td>0.25 Inches</td>
</tr>
<tr>
<td>Thermal Conductivity $K_2$</td>
<td>1390 $(\text{BTU<em>in})/(\text{h</em>ft}^2*\text{°F})$</td>
</tr>
<tr>
<td>R Value $R_2$</td>
<td>0.0001799 $(\text{h<em>ft}^2</em>\text{°F})/\text{BTU}$</td>
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<table>
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<th>Room Cavity</th>
<th>Air</th>
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<td>1.00 Inches</td>
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<tr>
<td>Thermal Conductivity $K_3$</td>
<td>4 $(\text{BTU<em>in})/(\text{h</em>ft}^2*\text{°F})$</td>
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<tr>
<td>R Value $R_3$</td>
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<td>R Value $R_4$</td>
<td>0.17 $(\text{h<em>ft}^2</em>\text{°F})/\text{BTU}$</td>
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<td>Insulation Type</td>
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<tr>
<td>----------------------</td>
<td>---------------</td>
</tr>
<tr>
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<td>X₅</td>
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<tr>
<td>Thermal Conductivity</td>
<td>K₅</td>
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<td>R Value</td>
<td>R₅</td>
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</tr>
<tr>
<td>Evaporator Wall Height</td>
<td>H</td>
</tr>
<tr>
<td>Evap. Wall Surface Area</td>
<td>A</td>
</tr>
<tr>
<td>Condenser Wall Height</td>
<td>H</td>
</tr>
<tr>
<td>Cond. Wall Surface Area</td>
<td>A</td>
</tr>
</tbody>
</table>

Armaflex Panel Insulation (removable panel)

\[
R = \frac{x}{k} = \frac{0.035}{4} = 114.286 \ (\text{h*ft}^2\text{°F})/\text{BTU}
\]

\[
h₁ = \frac{k}{x} = \frac{1}{R} = \frac{1}{114.286(\text{h*ft}^2\text{°F})/\text{BTU}}
\]

Aluminum Wall

\[
R = \frac{x}{k} = \frac{0.25}{1390} = .0001799 \ (\text{h*ft}^2\text{°F})/\text{BTU}
\]

\[
h₂ = \frac{k}{x} = \frac{1}{R} = \frac{1}{.0001799 \ (\text{h*ft}^2\text{°F})/\text{BTU}}
\]

Air in Room

\[
R = \frac{x}{k} = \frac{1}{4} = .25 \ (\text{h*ft}^2\text{°F})/\text{BTU}
\]

\[
h₃ = \frac{k}{x} = \frac{1}{R} = \frac{1}{.25 \ (\text{h*ft}^2\text{°F})/\text{BTU}}
\]

Evaporator Cavity

\[
R = \frac{x}{k} = \frac{1}{6} = .17 \ (\text{h*ft}^2\text{°F})/\text{BTU}
\]
\[
\frac{1}{h_4} = \frac{k}{x} = \frac{1}{R} = \frac{1}{0.17 (h \text{ ft}^2 \text{ °F})/\text{BTU}}
\]

Wall Insulation

\[
R = \frac{x}{k} = \frac{4}{0.187} = 21.2 \ (h \text{ ft}^2 \text{ °F})/\text{BTU}
\]

\[
\frac{1}{h_5} = \frac{k}{x} = \frac{1}{R} = \frac{1}{21.2 (h \text{ ft}^2 \text{ °F})/\text{BTU}}
\]

\[
U_{top} = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2} + \frac{1}{h_3} + \frac{1}{h_4}} = 0.0087 \ BTU/(h \text{ ft}^2 \text{ °F})
\]

Using similar equation

\[
U_{bottom} = 0.0416 \ BTU/(h \text{ ft}^2 \text{ °F})
\]

Calculating heat flux

\[
q = U \cdot A_{total} \cdot \Delta T = 10.52 \ BTU / HR
\]

Final wall temperature

\[
T_{wall} = T_{air} \cdot (q \cdot R_3 \cdot A) \approx 35°F
\]