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FINAL PAPER II
PROJECT OF A COLD CHAMBER FOR THE STORAGE OF GRAPES

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Abstract. *The present work seeks the design of a cold chamber for grapes. A theoretical cold chamber is developed considering its thermal load, calculated following the steps of ASHRAE book. All the calculations are done in the IT software. The goal presents an absorption system and comparing it with a vapor compression cycle, to find which one is better for the conceived cold chamber, that will operate 18 hours per day. The thermal load was calculated as the sum of the transmission load; the product load; the internal load; the infiltration load; and the equipment load. In the results, the thermal load found was 9.774 kW, or 2.77 TR. For the evaporation and condensing temperatures, the values were - 6°C and 45°C, respectively. The vapor compression system obtained a COP of 2.43, and the equipment is found in catalogs. The absorber system obtained a COP of 0.247, and for this reason, it not be found any commercial machinery. It was expected to obtain a low-cost system, yet, more efficient. For cooling capacities below 5 TR, an absorber refrigeration system is inferior to the vapor compression refrigeration system. The results shows that this kind of cycle is more applicable to heavy industrial uses.*

Keywords: *Cold Chamber, Refrigeration, Storage, Absorption, Grapes*

TRABALHO DE CONCLUSÃO DE CURSO II
PROJETO DE UMA CÂMARA FRIA PARA O ARMAZENAMENTO DE UVAS

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Resumo: O seguinte trabalho busca o dimensionamento de uma câmara fria para uvas. Uma câmara fria teórica é desenvolvida considerando sua carga térmica, calculada seguindo os passos do livro ASHRAE. Todos os cálculos são feitos no software IT. O objetivo é apresentar um sistema de absorção e compará-lo com um ciclo de compressão de vapor, para saber qual é o melhor para a câmara fria concebida, que funcionará 18 horas por dia. A carga térmica foi calculada sendo a soma das cargas de transmissão; da carga do produto; da carga interna; da carga de infiltração; e da carga do equipamento. Nos resultados, a carga térmica encontrada foi de 9,774 kW, ou 2,77 TR. Para as temperaturas de evaporação e condensação, os valores foram - 6°C e 45°C, respectivamente. O sistema de compressão de vapor obteve um COP de 2,43, com os equipamentos encontrados em catálogos. O sistema de absorção obteve um COP de 0,247, e por este motivo não foi encontrado nenhum maquinário comercial. Esperava-se obter um sistema de baixo custo, porém, mais eficiente. Para capacidades de resfriamento abaixo de 5 TR, um sistema de refrigeração por absorção é inferior ao sistema de refrigeração por compressão de vapor. Os resultados mostram que este tipo de ciclo é mais aplicável a usos industriais pesados.

Palavras-chave: *Câmara Fria, Refrigeração, Armazenamento, Absorção, Uvas*

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1. INTRODUCTION

Food is extremely important. However, the way consumables are stored cause batch problems. In general, a major factor responsible for the loss of these items is temperature. For this, cold chambers are used to preserve them, once they are machines capable of optimizing the storage of consumables through the refrigeration process. Fruits are consumables that are extremely sensitive to changes in temperature and humidity, as these variables accelerate their rotting process, as stated by CHITARRA & CHITARRA (1990). Cold chambers are especially needed in regions where the sunlight is stronger, as consumables exposed to the sun tend to spoil more quickly, having a greater demand for air conditioning and refrigeration systems (GE et al., 2018). As described by Sonker et al. (2016), the smallest mistakes can result in losses of thousands, because of bad storage, and the prevention can be done using cold chambers. The goal is to present two refrigeration systems: the vapor compression, with R-134a refrigerant (Tetrafluoroethene, CF_3CH_2F), and the absorber cycles, with a mixture of aqua-ammonia solution (NH_3H_2O), to find out which one of them works better for the chamber conceived. This article intends to conceive a cold chamber project for the storage of grapes, aiming at a conception of a machine that prevents the fruits from spoiling due to variable temperature and humidity conditions, and will need to operate 18 hours per day. The idea is to simulate this chamber in IT software. The motivation of this article is trying to be a differential in the refrigeration area, trying to schematic the better model to stocking the grapes, once a continental country crossed by the equator line, as is the case of Brazil, with a population among the largest on earth; with a strong economic crisis occurring and the nation returning to the hunger map, the stockpiling of food must leave its mark.

2. THEORETICAL BACKGROUND

2.1 The Cold Chamber

Cold chambers are associated with the process of refrigerated storage, which refers to the concept of the thermal load, once it represents the removal of heat generated by the stored products to reduce its temperature to a desired one. The quantity of heat can be calculated knowing the variables of mass, specific heat, latent heat, the initial state and the freezing onset temperature, as said by ASHRAE (1993). In this process, the grapes are known to produce ethylene, in low quantities, but in the long term, the ethylene could be accelerating the process of senescence, as said by Kader (2002). This is why grapes better not be stocked with other fruits, according to Josué et al. (2006). To avoid this problem, the cold chambers must have a system or at least a process of ventilation. According to Neves Filho (2002), if was a form of protection at the doors of the cold chamber, the heat load can be decreased by in 80%. The sizing of the cold chamber is the first step of the development of the processing with the respective technical implications (SILVA, ALESSANDRO de, 2016). The chamber's potential charge can be determined by knowing its total volume and densities of theconsumables. The gross stocking densities come from experimental tables. They operate by taking hot air from the environment through fans, which suck air into and over the evaporator coils. In the coils, the coolant pulls heat from the air passing through the top. So, the cooled air continues to move out of the back of the chamber, now refrigerated.

2.2 The Vapor Compression Refrigeration Cycle, With R-134a (Tetrafluoroethane, CF_3CH_2F)

The vapor compression system is a refrigeration cycle used in the industry. More applied in refrigeration storage, the model of this work uses the R-134a gas, which has a great use in cold storages, in general. The vapor compression refrigeration cycle is composed by a compressor, where there is work of compressor (W_C) entering the system, and will be called in this article as state 1; by a condenser (state 2), where there will be heat loss to the environment (Q_{CO}); by an expansion valve (stage 3) and by an evaporator (stage 4), where there will be heat entering the system (Q_{EV}). Once the stages are separated, is possible to sketch the system. To get the results, the known data is the temperatures of R-134a's evaporation and condensation. Having these values, every other parameter could be calculated by the IT equations. The vapor compression refrigeration cycle can be seen in Figure I.

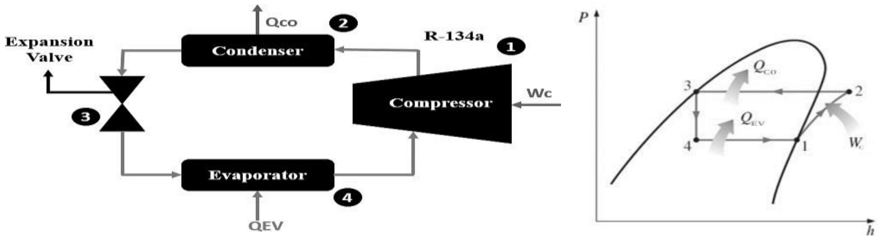

Figure I: The Compression Refrigeration Cycle, R-134a. Source: Author.

One of the ways to analyze the performance of a refrigeration cycle is the Coefficient of Performance (COP), which is the relationship between the power that is drawn out (Q_{EV}), and the energy that is supplied to the compressor (W_C).

2.3 The Absorption Refrigeration Cycle

The absorption refrigeration cycle is a system which uses a heat source to provide the needed energy on the cooling process. This is done by two coolants, where the first one performs evaporative cooling, and the second one absorbs its. In a cold chamber, this could be used to reduce the energy costs, once the process of refrigeration would take energy from the heat loss on the coolants, at the same it will be characterized by the environmental-friendly operation and a low total cost (Ghorbani et al., 2018). And, one of the best things this system has to offer is the fact that refrigerants used in absorption systems have reduced global warming potential, being less lethal to the ozone layer, as said by Gebreslassie et al. (2012). The mixture of water and ammonia that makes the refrigerant has thermal properties perfect for absorption refrigerating and air conditioning, as stated by Hmida et al. (2019). The absorption refrigeration cycle schematized is shown below in figure II.

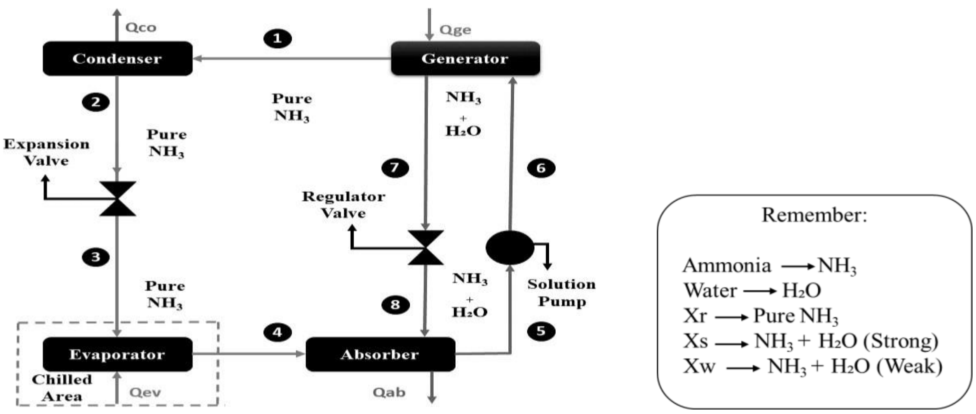

Figure II: The Absorption Refrigeration Cycle, NH_3H_2O. Source: Kramer et al. (Adapted). (2016).

The ammonia is chosen, because is a typical refrigerant gas that are used in cold chambers as an integral part of the food processing industries. In the absorption refrigeration cycle, firstly, the heat will be provided to the mixture of water and ammonia in the absorber, that will go up to the generator, passing through a pump in the way. In the generator, the temperature will rise up till the point the major part of the mixture becomes vapor. After this, the vapor will be transferred as saturated vapor to the condenser, and, at the same time the mixture will be removed from the generator (Q_{GE}), pass through an expansion valve, and go back to the absorber. Now, the ammonia vapor will flow through the condenser, where it will be liquefied, losing its heat to the environment. At the end of this process, the ammonia is now saturated liquid, and then, it is expanded through the valve to reach the evaporator low-pressure (Hmida et al., 2019). The required refrigeration capacity will be obtained at the evaporator, and it will be used to find out all the equipment by its value. From there, ammonia vapor will be transferred to the absorber, to be absorbed and become the strong solution. It is important to remember that the absorption is an exothermic process, and so, heat will be extracted from the absorber (Q_{AB}) to maintain its temperature low, in a level that the refrigerant and solution will be in synchrony.

It was necessary to insert the values of each equipment temperature, and the mass fractions, which are the percentages of Ammonia in the solution on different stages of the process. The X_r will represent the almost pure concentration of Ammonia, and for this one, can be used the state of saturated liquid and vapor, once, it is almost 100% pure. The other concentrations are the strong (X_s) and weak solutions (X_w), called as mass fraction of Ammonia in the solution and mass fraction of Ammonia in the absorbent, respectively (Bangotra, A., & Mahajan), as seen in Attachments E, F, G and H.

2.4 Composition and Properties of Grapes

The table I presents the composition and properties of grapes, and allows to calculate the heat removed from the grapes. Once the enthalpy changes in this process, it can be used to know how much energy will be needed to make the temperature change.

Table I. Grapes' Properties.

Grapes	Moisture Content, % x_{w0}	Protein, % x_p	Fat, % x_f	Carbohydrate Total, % x_c	Carbohydrate Fiber, % x_{fb}	Ash, % x_a	Initial Freezing Point, °C	Specific Heat Above Freezing, kJ/(kg·K)	Specific Heat Below Freezing, kJ/(kg·K)	Latent Heat of Fusion, kJ/kg	Specific Heat, Wh/(kg·K)
America	81.3	0.63	0.35	17.15	1.00	0.57	-1.6	3.71	2.07	272	272
Europe	80.56	0.66	0.58	17.77	1.00	0.44	-2.1	3.7	2.16	269	269

Source: ASHRAE, 2018 (Adapted).

So, according to the method of Chen (1985), the enthalpy of an unfrozen food can be calculated by:

$$H = (H_f + (T - T_f) * (4.19 - (2.30 * x_s) - 0.628 * (x_s^3))) \tag{1}$$

$$H_f = (T - T_r) * [((1.55 + 1.26 * x_s) - (((x_{w0} - 0{,}4 * x_p) * L_0 * T_f)/(T_r * T)))] \tag{2}$$

Where H is the grape enthalpy, in kJ/kg; H_f is the grape enthalpy at initial freezing temperature, in kJ/kg; T is the grape temperature, in ºC; T_f is the initial freezing temperature, in ºC, and the x_s is the grape's mass fraction of solids. As there is still no "H_f", it must be calculated by the other equation, where T_r is the reference temperature (in this case, because it is an unfrozen food, T_r is the same as the enthalpy of food at initial freezing temperature), in ºC; x_{w0} is the grape's water mass fraction; x_p is the grape's protein mass fraction and L_0 is the latent heat of fusion of water at 0°C, in kJ/kg (The latent heat of fusion of water at 0°C is 336,6 kJ/kg).

2.5 Matherials And Methods

The methods used to obtain the results was simulations on software Interactive Thermodynamics (IT). At IT, the basic calculations were done, simulating a cold chamber with capacity to maintain.

2.6 Thermal Load Equations

When a product is refrigerated, being cooled or frozen, there will be a thermal load from the removal of its heat, in order to reduce its temperature to the desired one. In the storage's process of the product, a heat load will be a function of its thermal insulation; of the door opening; of the lighting; of the people and of the engines. When there are fresh fruits and vegetables, as in this case, with the grapes, the heat of breathing must also be taken into account. However, the amount of heat that will be removed during the process of refrigeration is much higher when compared to just the storage. Thus, according to Alessandro da Silva (2016), the calculation of its capacity or thermal load basically involves four heat sources. They are: The heat transfer through walls, floor and ceiling; The heat infiltration of the indoor air through the door openings; The load of the product and others, like people or the machines, as the engines. According to ASHRAE (2006), the total refrigeration load will include the Transmission Load, or the heat transferred into the refrigerated area through its own surface; the Product Load, or the heat removed from the products, while they are in the refrigerated area; the Internal Load, or the heat produced by sources inside the refrigerated area, such as the electric motors, the lights and the people working there; the Infiltration Air Load, or the heat gained when there is air entering the refrigerated ambient, as, for example, when the door is opened; and finally, the Equipment-Related Load.

2.6.1 Transmission Load

For Steady State, the sensible heat gain through walls, floor, and ceiling is calculated by the following equation:

$$q_1 = U * A * \Delta T \tag{3}$$

The q_1 is the sensible heat gained through walls, in W; U is the global coefficient of heat transfer, in W/m^2K; A is the outside area of section, in m^2; and ΔT is difference between the external and the internal air temperature, in K.

2.6.2 Product Load

The Product Load could be divided into two steps. The first one is the heat that must be removed to bring products to storage temperature and the second one is the heat generated by the products in the storage. The heat removed can be calculated by the following equations:

$$Q_1 = mc_1 * (T_1 - T_2) \tag{4}$$

$$Q_2 = mc_1 * (T_1 - T_f) \tag{5}$$

$$Q_3 = m * h_{if} \tag{6}$$

$$Q_4 = mc_2 * (T_f - T_3) \tag{7}$$

$$q_2 = (Q_2 + Q_3 + Q_4)/(3600 * n) \tag{8}$$

These equations represent the heat removed to cool the product from initial temperature to some lower temperature above freezing (Q_1); The heat removed to cool from initial temperature to freezing point of product (Q_2); The heat removed to freeze the product (Q_3) and the heat removed to cool from freezing point to final temperature below freezing point (Q_4). At the final, q_2 is the heat that will be removed from the grapes, in W; m is the product mass, and n is the allotted time to freeze, in hours.

2.6.3 Internal Load

The internal load part considers the heat load generated by the lights, people and electric equipment, such as the motors and the forklift. It considered the heat that the equipment produces; the heat that people irradiate and the heat the lamps generate too. In the case of the lamps, with n being the number of lamps, and P the power, in Watt, the equation can be represented as:

$$q_{lights} = n * P \tag{9}$$

For the people, the equation can be written, according to ASHRAE (2006), as:

$$q_p = 272 - 6 * T_c \tag{10}$$

Where T_c is the Cold Chamber storage temperature. For the electrical part, as said in ASHRAE (2006), a 3-Phase Motor of 1750 rpm, with the equipment in, but the motor itself out, will provide 557 W, with an efficiency of 72%. And, it is said that for a forklift with a battery volt of 72 VDC, it will be necessary a power between 3 - 5 kW. So, it was chosen the average 4 kW. So, the q_3 can be written as:

$$q_3 = q_{lights} + q_{people} + q_{forklift} + q_{motor} \tag{11}$$

2.6.4 Infiltration Air Load

Because there is a temperature and density differences between the indoor and outdoor air, there will be a heat gain because of the infiltrations. According to ASHRAE (2006), that equation can be written as:

$$q_4 = \rho_r * W_{door} * H_{door}^{1.5} * Qs_{air} * \frac{1}{R_s} \tag{12}$$

Where q_4 is the heat gained by air infiltration, in W; ρ_r is the specific mass of cooled air, in kg/m^3; W_{door} is the doorway width; H_{door} is the door height; Qs_{air} is the Sensible heat load of infiltration air per square meter of doorway opening and the R_s is the Sensible heat ratio of the infiltration air heat gain.

2.6.5 Equipment Related Load

Finally, the last part of the Thermal Load calculations involves the equipment directed related load (fans). So, the q_5 has two parts, with the equations being:

$$q_{fans_{motors}} = P * n \tag{13}$$

$$q_{fans_{defrosting}} = P * z * \eta \tag{14}$$

Where P is the power, in W; n is the number of the fans; z is the number of the defrosting cycles; and η the electric efficiency (Frigoblock, 2018).

3. METHODOLOGY

3.1 Project Data And The Cold Chamber Conception

For the heat load calculations, some configurations must be pre-settled before, as the installation location; the type of assembly to be performed and the basic resources for the chamber works. The idea is the chamber be done as an indoor installation. It is important to say, in the ASHRAE meteo-info, the source of all the data that will be presented, there is no point on Alegrete City, and, because of this, the data was collected from the nearest possible point: Artigas, Uruguay.The value adopted to the outdoor dry bulb temperature in summer will be considered 37.5 ºC, but, as the installation is inside of a structure, and not exposed to the sun directly, then it will need to be consider the adjust of minus 5 ºC for indoor buildings. The “outdoor indoor” temperature of the cold chamber, will be called as external temperature and be considered 32.5 ºC. According with Oliveira et al. (2021), it’s possible to pack 550 kg of grape in a pallet. Each pallet has22 layers of boxes, 5 boxes in each layer and 10 bowls in each box. By the dimensions of 2.4 × 14.0 × 3.0 m, Oliveira etal. (2021) affirms that fits 28 grape pallets, distributed in two rows. With the Brazilian pallet pattern being 1.2 x 1.0 (Antoniolli et al. 2008), while the Brazilian grape’s box dimension is 0.63 m for the length, 0.42 m for the width, and 0.207 m for the height, the chamber itself was modelled to adapt these divergences. Considering that the standard of height is 2.4 m, this is the value chosen for the cold chamber. According to the Guialog (2001) article, the minimum width of the corridors is 1 meter, while the maximum value is 3.5 meters. So, knowing 2.4 is the standard value height, according to the Gallant, and the 5.5 x 5.5 area, as said by Hmida et al. (2019) will fit in this project. Also, the cold chamber needs an isolation system to prevent the heat loss. In the end, the dimensions of this chamber will be 5.5 m; 5.5 m; and 2.4 m. One thing that happened was that the pallet pattern dimensions, for the Brazilian type, did not synchronize with the Brazilian grape’s box pattern. So, because of that, the boxes would be left with a space missing on one side, and more space on another. By the calculations, using data provided from the references, such as Oliveira et al. (2021), the information obtained was that, with all the 6 boxes full, this cold chamber has the capacity to store 480 kg of grape. The schematic of the chamber can be seen in Figure III.

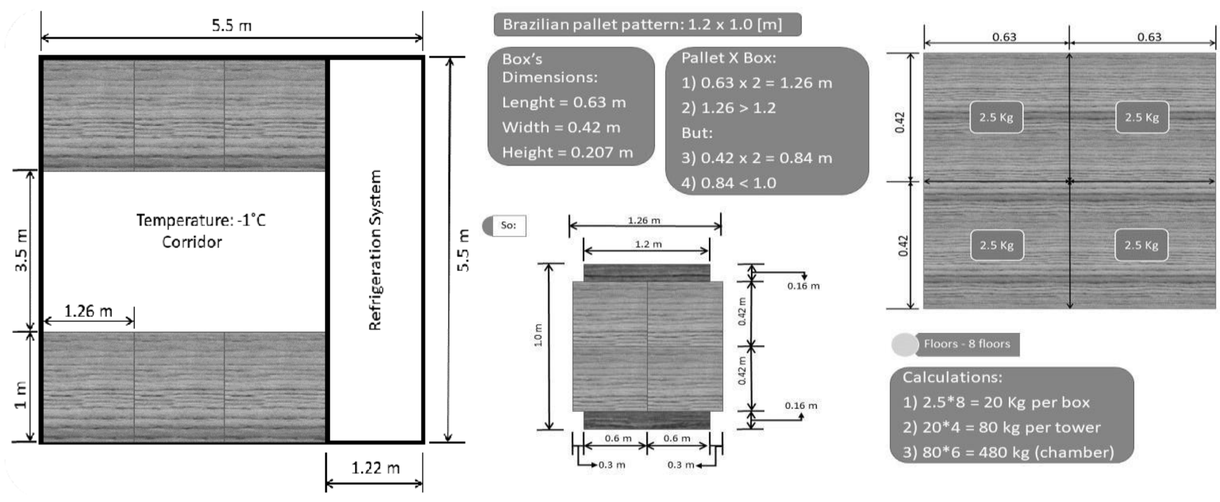

Figure III: Cold Chamber’s Design Conception. Source: Author.

3.2 Materials

So, according to the list of material characteristics of the cold room, by ASHRAE (2006), the walls of the chamber would be made of steel, for protection; polystyrene (expanded (R-142b)) for the insulation. For the ceiling, considering the gravity factor, steel, and for its structural resistance, with polystyrene (expanded (R-142b)). For the floor, concrete, once, their values on the table are lowers than the other materials, which indicates that they don't provide too much heat, and polystyrene (expanded (R-142b)) to build up an insulated place. Finally, for the door, galvanized sheet would work. It was considered a space in the sizing of the chamber to the refrigeration system works.

3.3 Input Data And Software Simulations

For each one of the cycles, it was written a code on the Software IT, attached in the end of this work (Appendixes A.1; A.2 and A.3) to calculate each component of the cycle individually by using the input data found on works such as Ferreira, 2006; Bangotra, A., & Mahajan, A. and Hmida, 2019.

3.3.1 Vapor Compression With R-134a Refrigerant Refrigeration System Simulation

For the Vapor Compression With R-134a Refrigerant Refrigeration System Simulation, the input data used is display in the Table II. One used, but not displayed yet is the thermal load value, which will be presented on the results section.

Table II: R-134a Vapor Compression Cycle Parameters

Equipment	Temperature [°C]	Temperature [K]	Temperature Adjusted With ΔTS [°C]
Condenser	50	323.15	45
Evaporator	-6	267.15	4

Source: Author.

3.3.2 Aqua-Ammonia Absorber Refrigeration System Simulation

For the Aqua-Ammonia Refrigeration System Simulation, the input data used is display in the Table III. The parameters used were the ammonia concentration, for solution, absorbent and refrigerant solution, based on Bangotra, A., & Mahajan, A. (n.d.) and the temperatures for the equipment, being them the temperatures for the generator and for the absorber, both based on Bangotra, A., & Mahajan, A. (n.d.), and the temperatures of condenser and evaporator, based on Ferreira Neto and his advisor Beyer work (Neto, 2006). Others parameters used were found on ASHRAE textbooks, in the versions: 2005, 2006, 2013, 2018 and 2021, while the project data was obtained in the ASHRAE meteo-info site.

Table III: Aqua-Ammonia Absorber Cycle Parameters.

Component	Temperature [°C]	Temperature [K]	% Ammonia	Qevaporator [kW]
Condenser	45	318.15	-	
Evaporator	-6	267.15	-	
Absorber	52	325.15	-	
Solution Pump	52	325.15	-	9.774
Generator	120	393.15	-	
Xs	-	-	0.42	
Xr	-	-	0.98	
Xw	-	-	0.38	

The Input Data For The Aqua-Ammonia Absorber Cycle. Source: Author.

Differently of vapor compression system, the Aqua-Ammonia absorber cycle needs more information. The Power used is the same as the thermal load, but in this cycle, it enters on the evaporator part. The input data of the machinery was obtained in the: Bangotra, A., & Mahajan, A. article.

4. RESULTS & DISCUSSION

The project calculations were made as follows: First, they were separated into categories, and then one by one, the equations were written in the IT, in order to obtain the necessary values.

4.1 Geral: Thermal Load

The calculation process of the thermal load was done picking each one of the equations set up early in this work, and then use the Interactive Thermodynamics software to write down a whole code that was capable of obtain the results for every equation presented on this article. The use of IT software was for its practicality, and when was necessary to change a value, it wasn't necessary to written the whole equation again. It was just done by changing the numbers, and everything was working again, with brand new values. For an adequate heat transfer, a ΔT between 5°C and 10°C must be maintained, between the exchangers, the refrigerated space and the external environment (Kipper, 2002). The final value obtained for the Thermal Load was a sum of the others mini-values, calculated as parts, which were q_1; q_2; q_3; q_4 and q_5. The sumhas given the value of 9774 W, or 9.774 kW, or 175.932 kWh (considering 18 hours as the working time of the chamber),or 2.77 TR (Tons of Refrigeration), as the results can be seen on Table IV.

Table IV: Thermal Load Calculations.

Thermal Load				
Q	W	kW	kWh (18h)	TR
q_1	2247	2.25	40.05	0.64
q_2	0.1796	0.0001	0.003	5E-05
q_3	5235	5.24	94.23	1.48
q_4	42.36	0.42	76.25	0.012
q_5	2250	2.25	40.5	0.64
qtotal	9774	9.77	175.93	2.77

Source: Author.

4.1.1 Equipment Selection For The Vapor Compression Refrigeration Cycle

The equipment selection was made by comparing the values obtained from the IT software with catalogs of companies, such as BITZER, and Danfoss. First, the process was select each one of the equipments, one by one. Entering with the thermal load of 9.97 kW as the compressor power, then the rest of equipments values could be found. The results can seen in Table V. The Attachment A presents a BITZER compressor's data, which can be used for comparison.

Table V: Equipment Values Of The Vapor Compression Refrigeration Cycle Calculated In The I.T.

Equipment	Temperature [°C]	Temperature [K]	Capacity [kW]
Condenser	45	318.15	14.55
Evaporator	-6	267.15	9.774
Expansion Valve	120	393.15	-
Compressor	78.37	351.5	3.476

Source: Author.

However, in the majority of catalogs, the equipment for the cold chamber isn't sold apart. They are all sell as one pack, called Condensing Unit, that includes the compressor and the condenser. The models select are display in the Table VI. The data used for the selections is presented in Attachments B, C and D.

Table VI: Equipment Selection For The Vapor Compressor Refrigeration System.

Equipment Selection			
Equipment	Model	Provider	Capacity
Condensing Unit	Optyma™ Slim, OP-HGZC0400U Q-114N3618 MTZ50	Danfoss	18.3 kW
Expansion Valve	Thermostatic expansion valve, TGE, R134a; R513A 067N5003	Danfoss	
Evaporator	PSM2 217-M/1.350	BITZER	10.706 kW

Source: Author.

To finish this Vapor System, it was calculated the Compressor Consumption of Electric Energy. All the values used were selected using the present values of electric energy bills on Brazil, at January, 2023. Considering that this chamber works 18 hours per day, and the cost of 1 kW/h in the Rio Grande do Sul State, at February, 2023, is R$ 0.65313, the result is, for a month, this cold chamber's electric energy consumption using a Vapor System would cost R$ 1513.63.

4.1.2 Equipment Selection For The Absorption Refrigeration Cycle

The table VII presents the values obtained for each component of the absorber cycle. For the calculations, it used the values of equipment and the mass fraction mentioned earlier, but it still needed the values of enthalpy. So, for that, these values of input data were used to find the enthalpy values in an Aqua-Ammonia Enthalpy x Mass Fraction chart (Attachments F; G and H). Analyzing the values obtained in table VII, it could be seen that, with a standard temperature for the generator, the cooling capacity is still very low, when compared with the cooling capacities of industrial generators, and its value was because of the thermal load. It could be seen that the absorber is the machine which is providing the highest value of cooling capacity, but, since this is a system entirely based on this component, it makes sense, once all the three mass fractions of the refrigerant are present at absorber, with the pure and weak solutions come in from the evaporator and the expansion valve, and the strong solution coming out for the solution pump.

Table VII: Equipment Values Of The Absorber Refrigeration Cycle. Source: Author.

Equipment	Temperature [°C]	Temperature [K]	Specific Enthalpy [kJ/kg]	Capacity [kW]	Mass Flow Rate [kg/s]	Mass Flow Rate [kg/s]
Condenser	45	318.15	1471	10.11	mr	
Evaporator	-6	267.15	152.2	9.774	mr	
Expansion Valve I	-6	267.15	152.2	-	mr	mr = 0.004906
Absorber	52	325.15	152.2	42.66	mr,ms,mx	ms = 0.1411
Solution Pump	52	325.15	1435	-	ms	mx = 0.1317
Expansion Valve II	120	393.15	50	-	mx	
Generator	120	393.15	120	39.47	mr,ms,mx	

Source: Author.

The generator has all the three mass fractions to, once, besides with the absorber, these two compose the self-called "Thermal Compressor", the equivalent of the vapor compressor in the vapor compression refrigeration cycle. It's interesting to notice that, while the absorber receives the strong solution and provides the pure and weak ones, the absorber is the reverse, receiving the pure and weak ones, and liberating the strong solution. The equipment selection for this cycle was impossible to be done, because, as said by ROBUR®, the nominal capacity of an Ammonia-Water Absorption Cycle, for commercial applications, needs, at least, 5 TR (17,5843 kW), almost the double this system can provide. The comparation between the two refrigeration cycles for the values calculated in the software could be seen in the table VIII.

Table VIII: Comparison Between The Two Refrigeration Systems For The Same Input Data.

Vapor Compression Refrigeration Cycle With R-134a			Aqua-Ammonia Absorber Refrigeration Cycle		
COP	Power (Wc_real)	Energy Consumption	COP	Power (QGE)	Energy Consumption
2.433	4.017 kW	35.23 kWh	0.2477	39.47 kW	102 kWh

Source: Author.

5. CONCLUSIONS

In conclusion, the dimensions of the chamber were selected based on Hmida et al. (2019) works, once it appeared to be a very similar process at the beginning. However, the projects have gone through very distinct paths by the start of the calculations. The 5.5 m per 5.5 m of area was chosen because it appeared to be a possible size to work with, and the height of 2.4 m was picked up, once it was found out that this was the standard value for cold chambers, not being the only one, but the most usual to work.

The compression vapor cycle proved itself as the better option to this project, once, for the low values obtained during the calculations processes, the vapor cycle shows more compatible values, having plenty of options in the catalogs for the components, like the compressors or the expansion valves. The condenser was the hard one to find, especially for the conditions set up as the main goal. But, once it was found that the condensing unit can replace it, then the rest of the process was easy to do. It's interesting to notice that the COP for the vapor compression cycle is beyond 1, while in the absorption cycle it is much lower. Another thing is that was need to be considered the local, and all the meteorologic factors. Because of that, the ASHRAE meteo-info site was a perfect guide to follow, and obtain the thermodynamics values, which were used. An absorption refrigeration cycle using an (NH3H2O) has been theoretically demonstrated as a better option that the "normal" cycle of vapor compression when cooling capacity values above 5 TR are needed. For small cooling needs, it won't be recommended, because, for such low values, the costs still don't pay themselves, and the vapor compression system seems to be a better option. By calculating the performance of both cycles, using a thermodynamic model developed in the IT software.

One thing that can be concluded by the results, analyzing the two values of COP. In reality, the value of the COP for an absorption cycle in air conditioning would be about 0.7, compared to about 3.5 for a vapour compression system (Welch, 2009). An absorption system requires about five times more energy than vapor compression cycle, however, of course, the energy for absorption is heat energy, not work (electrical) energy, which means that, knowing heat energy is cheaper than electrical energy, and in some applications this heat energy is free, or is waste heat from another use, such as waste steam, hot water, gas, solar energy etc., it makes it advantageous to use absorption, for bigger projects. Knowing that in large scale the process will consume more than just electrical energy like the vapor compressor one, then it will be a solution (Welch, 2009). And, another thing to notice is that, while the usual value for COP in a absorption cycle is 0.7, in this work, the value obtained was 0.247, just showing that for small projects this one will be much more expensive alternative to the vapor compression cycle. And, about the COP of the vapor compression cycle, it was found a lower value than the standart, however, this is an experimental work, with non-convencional values, and, the diferrence between its COP to the standart is very lower than the difference between the COP of the calculated absorber system and the standart.

This article was made as a way of encourage people who produces food, and not only grapes, to try a new refrigeration system, with a higher efficiency for a lower cost. The proposal was to instigate people to try by themselves the ammonia-water absorption cycle. This model could be extended to other products, if someone has an eye on it. An ammonia-water absorption refrigeration system could provide a great cool load, and with more efficiency for less quantity. The effects of both cycles, and their requirements were all on the table, and be considered. The effect provided by the parameters chosen, as the initial conditions, the values obtained from the cycle or the ones calculated in function of others ones, like the case of the COP. This makes this kind of machine more efficient, with lower costs, that could be made in smaller scales, and are more friendly with mother nature, while serves to feed people, is a great proposal.

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7. RESPONSIBILITY NOTICE

The author is the only responsible for the printed material included in this paper.

8. ATTACHMENTS & APPENDIXES

ATTACHMENT A – SPECIFICATIONS OF THE BITZER's COMPRESSOR MODEL: 4DES-5Y-40S.

Compressor	4DES-5Y-40S
Capacity steps	100%
Cooling capacity	10,79 kW
Cooling capacity *	11,04 kW
Evaporator capacity	10,79 kW
Power input	4,09 kW
Current (400V)	7,66 A
Voltage range	380-420V
Condenser capacity	14,88 kW
COP/EER	2,64
COP/EER *	2,70
Mass flow	289 kg/h
Operating mode	Standard
Discharge gas temp. w/o cooling	78,8 °C

Source: BITZER.

ATTACHMENT B – SPECIFICATIONS OF THE DANFOSS' CONDENSING UNIT.

All values are calculated based on the given operating conditions.
Any increase or decrease in mass flow caused by juntions is not considered.

Mass flow in evaporator: 0,07035 kg/s

Point	Description	Temperature [K]	Pressure (a) [Pa]	Density [kg/m^3]	Enthalpy [J/kg]	Entropy [J/(kg·K)]
1	Compressor suction	277	234400	11,1	403700	1762
2	Compressor discharge (estimated)	352	1318000	54,88	457900	1810
2s	Condensation dew point	323	1318000	66,26	424300	1710
3s	Condensation bubble point	323	1318000	1103	272500	1241
3a	Condenser out	318	1318000	1126	264800	1217
3	Including additional subcooling	318	1318000	1126	264800	1217
4	After expansion valve	267	234400	31,95	264800	1243
4s	Evaporation bubble point	267	234400	1315	191900	970,2
1s	Evaporation dew point	267	234400	11,65	395000	1730
1a	Evaporator out	277	234400	11,1	403700	1762

Source: Danfoss' Coolselector 2.

ATTACHMENT C DANFOSS' CONDENSING UNIT MODEL.

OP-HGZC0400U	WJ	N Q R	114N3617 114N3618 114N3619	MTZ50	90	16350	19020	21910	25000	28280	31760	35410	39220	3093	2950	7.13
					95	15700	18300	21100	24090	27260	30610	34130	37800			
					100	15010	17540	20240	23130	26200	29430	32810	36340			
					105	14290	16730	19350	22140	25090	28200	31460	34840			
					110	13530	15900	18430	21120	23960	26940	30060	33310			

Source: Danfoss' Coolselector 2.

ATTACHMENT D – EVAPORATOR SELECTION ON BITZER'S CATALOG.

MODEL: PS2M*		108	117	141	162	177	187	217	246	276	309
R404a	Watts @ 6ktd	6469	6997	8475	9693	10612	11238	13029	14749	16549	18519
	Watts @ 1ktd	1078	1166	1412	1615	1769	1873	2172	2458	2758	3087
R134a	Watts @ 6ktd	5596	6052	7331	8384	9179	9721	11270	12758	14315	16019
	Watts @ 1ktd	933	1009	1222	1397	1530	1620	1878	2126	2386	2670
R22	Watts @ 6ktd	6104	6602	7997	9146	10013	10603	12294	13916	15615	17474
	Watts @ 1ktd	1017	1100	1333	1524	1669	1767	2049	2319	2603	2912
Air flow	L/s	1458	1422	2331	2187	2133	3108	2916	2844	3645	3555
	M³/h	5249	5119	8392	7873	7679	11189	10498	10238	13122	12798
	CFM	3090	3013	4940	4634	4520	6586	6179	6027	7724	7533
Air throw-std fan (metres)		16.9	16.5	18.7	18.4	17.9	21.0	20.8	20.2	21.7	20.6
Number of 350mm fans		2	2	3	3	3	4	4	4	5	5
Fan	Total watts	330	330	495	495	495	660	660	660	825	825
Motor(s)**	Total amps	1.46	1.46	2.19	2.19	2.19	2.92	2.92	2.92	3.65	3.65
Motor heat / 24hrs watts		7920	7920	11880	11880	11880	15840	15840	15840	198000	19800

Source: BITZER.

ATTACHMENT E – DESIGNS CONDITIONS FOR THE 3 TR AQUA-AMMONIA REFRIGERATION SYSTEM.

4. Design Conditions for the 3TR Aqua Ammonia Refrigeration System.

The literature values for the design of the Aqua Ammonia vapour absorption system are:

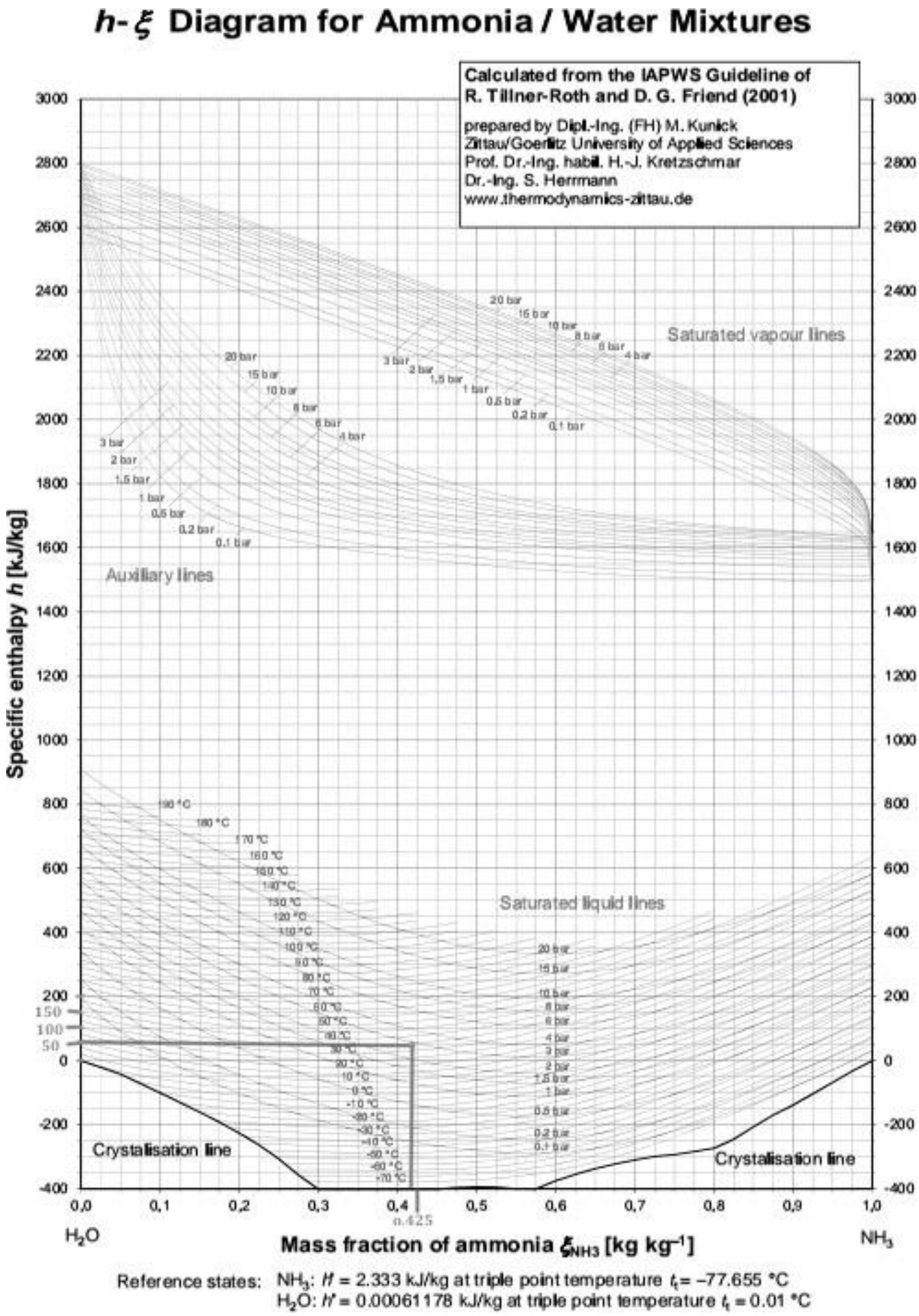
Capacity of system = 3TR(10.548KW)
Concentration of NH_3 in refrigerant, X_r = 0.98
Concentration of NH_3 in Solution, X_s = 0.42
Concentration of NH_3 in absorbent, X_w = 0.38
Temperature of the evaporator, T_E = 2℃
Generator or condenser pressure, P_H = 10.7 bar
Evaporator pressure, P_L = 4.7 bar
Temperature of the Condenser, T_C = 54℃
Temperature of the Absorber, T_A = 52℃
Temperature of the Generator, T_G = 120℃

Table 1. Values of mixture at various state points

State Points	Temperature in ℃	Pressure in bars	SpecificEnthalpy h in KJ/Kg
1	54	10.7	1135
2	54	10.7	200
3	2	4.7	200
4	2	4.7	1220
5	52	4.7	0
6	52	10.7	0
7	120	10.7	255
8	120	4.7	255

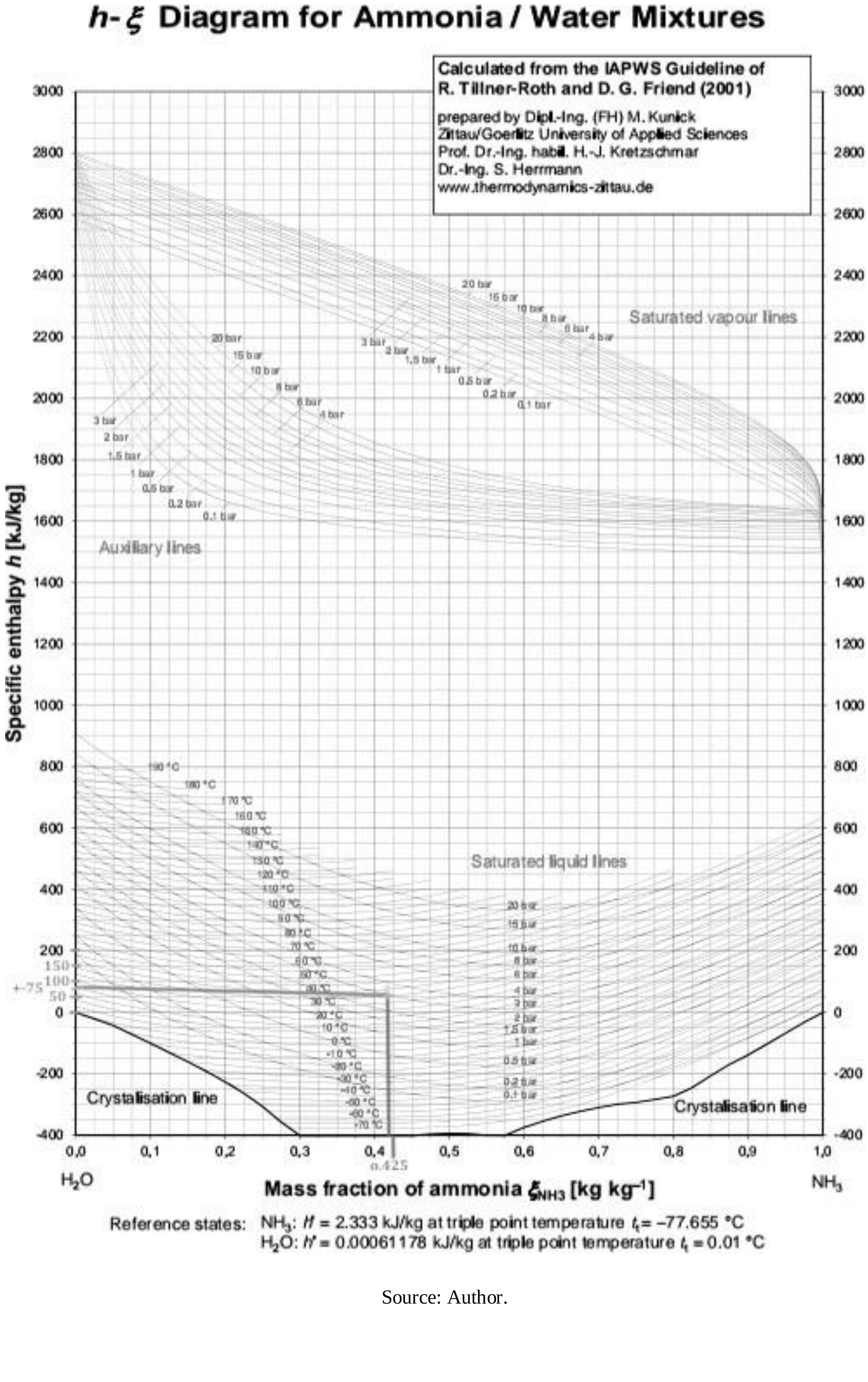
Source: Arun Bangotra, Anshul Mahajan.

ATTACHMENT F – ENTHALPY FOR THE POINT "h5" IN THE ABSORPTION CYCLE.



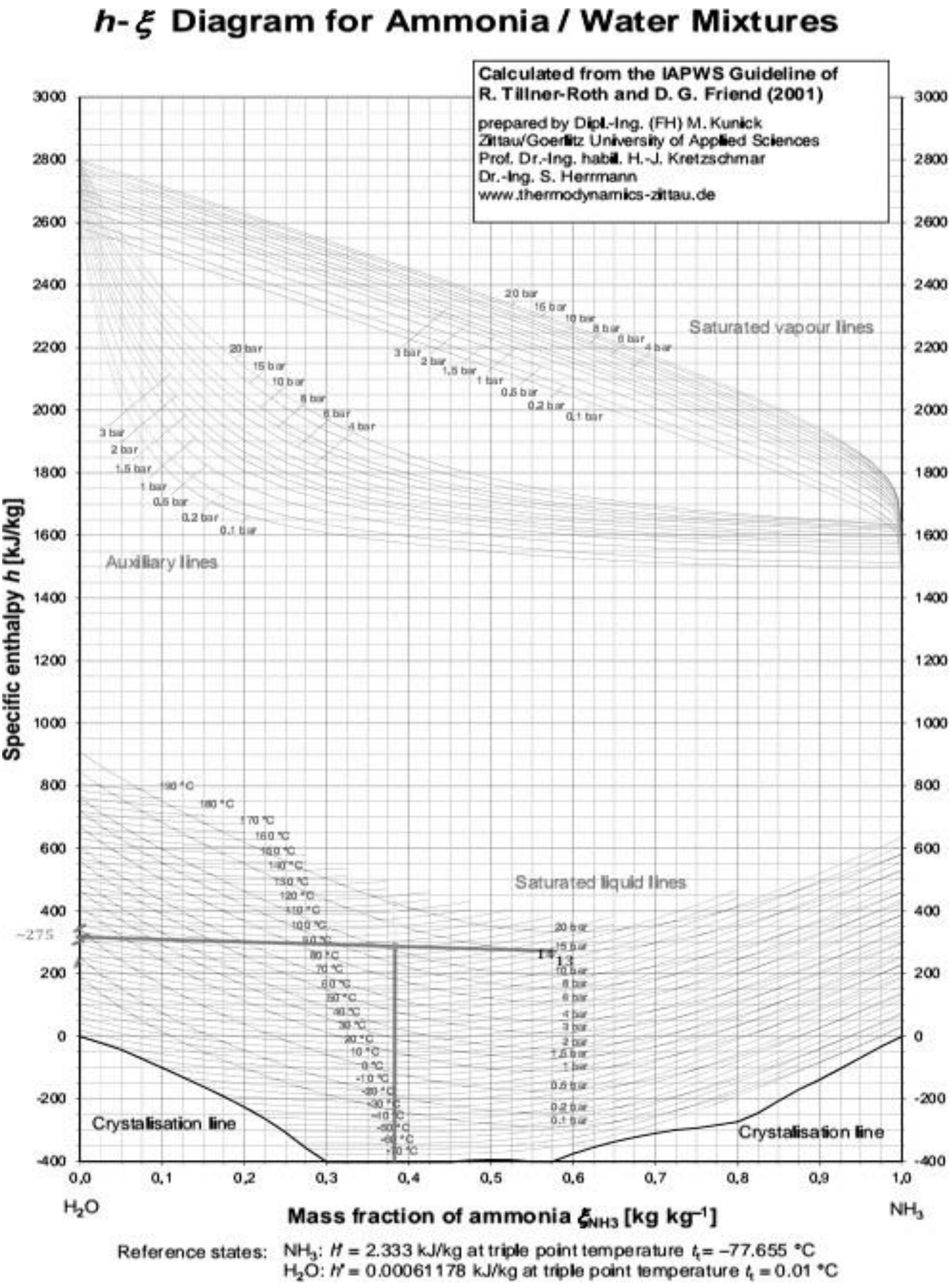
Source: Author.

ATTACHMENT G – ENTHALPY FOR THE POINT "h6" IN THE ABSORPTION CYCLE.



Source: Author.

ATTACHMENT H – ENTHALPY FOR THE POINT "h7" AND "h8" IN THE ABSORPTION CYCLE.



Source: Author.

APPENDIX A – CODES

APPENDIX A.1 – CODE I: THERMAL LOAD

8.1 Code I: Thermal Load

```
// Project Data:
// Steady State
// Reference Local: Artigas; UY
// Summer
// Covered Chamber
T_Environment = 305.85 // K, or 32.7 ºC; Standart Temperature For The Environment (3 degrees for correction)
Tm = 32.7 // ºC; From ASHARAE 2021 meteo.info
R = 8.314 // kJ/(kg mol·K); Universal gas constant (From "ASHRAE, 2006")

// Dimensions
// Chamber = 3 Full Walls + 1 Wall With A Door + Roof + Floor
Chamber_Height = 2.4 // m
Chamber_Length = 5.5 // m
Chamber_Width = 5.5 // m
Wall_Area = (Chamber_Height * Chamber_Length) // m²
Floor_Area = (Chamber_Width * Chamber_Length) // m²
Ceiling_Area = Floor_Area // m²
Chamber_Volume = (Chamber_Height * Chamber_Length * Chamber_Width) // m³
Door_Area = (1.1 * 2.1) // m³; (From "Igloodoors)

// Pallets And Boxes Dimensions:
// Mass = 2.5 // kg (Per Pallet)
// Pallet = (0.63*0.42*0.207)
// Pallets = (4*8*6)
// Weight_Of_Boxes= Mass + 0.194 // (Grapes) + Box/Crate's Weight (Brazilian Standart) = 0.194 kg
// Pallet_1Floor = (0.63*2)*(0.42*2)*(0.207)
// Floors = 8
// Mass_Per_Floors = (Mass*Floors)
// Towers = 4
// Mass_Per_Towers = (Mass_Per_Floors*Towers)
// Chamber = 6 // Towers
// Mass_Chamber = (Mass_Per_Towers*Chamber)
// Operation_Time_Per_Day = 648000 // seconds; 18 hours (Operation Time Of The Machines)
// Mass_Flow = Mass_Chamber/Operation_Time_Per_Day

// Product: Grapes
Cp = 2.07 // kJ/(kg*K); Specific Heat (From "ASHRAE, 2006")
St = 272.15 // K, or (30.2 ºF), or (-1 ºC); Storage Temperature (Could also be: -0.5 ºC) (From "ASHRAE, 2006")
T0 = 273.2 // K; freezing point of water (From "ASHRAE, 2006")
LH = 272 // kJ/kg; Grape's Latent Heat Of Fusion (From "ASHRAE, 2006")
L0 = 336.6 // kJ/kg; Latent Heat Of Fusion Of Water At 0°C (From "ASHRAE, 2006")
tf = -1.6 // ºC; Initial Freezing point of food (From "ASHRAE, 2006")
t = -1 // ºC; food temperature (From "ASHRAE, 2006")

// Xice
xw0 = 0.813 // (81.3%); Grape's Water Mass Fraction in the unfrozen food (From "ASHRAE, 2006")
Xice = ((1.105*xw0)/(1+((0.7138)/(ln(tf - t + 1))))) // (From "ASHRAE, 2006")

// Cu
Cu = (4.19 - (2.30*xs) - 0.628*(xs^3)) // kJ/(kg·K); Specific heat of the Unfrozen Food ("Unfrozen" because St < tf) (Chen (1985), From "ASHRAE, 2006")
xs = (1 - xw0) // Mass fraction of the solids in the food (From "ASHRAE, 2018")
```

// Enthalpy
H = (Hf + (t-tf)*(4.19 - (2.30*xs) - 0.628*(xs^3))) //kJ/kg; Grape Entalphy (Enthalpy of food) (From "ASHRAE, 2006")
Hf = ((t-tr) * (1.55+(1.26*xs) - (((xw0-xb)*L0*tf)/(tr*t)))) // kJ/kg; Enthalpy of food at initial freezing temperature (From "ASHRAE, 2006")
xb = (0.4*xp) // Mass fraction of bound water (From "ASHRAE, 2006")
xp = 0.0063 // (0.63%); Grape's Protein Mass Fraction (From "ASHRAE, 2006")
tr = tf // ºC; Reference temperature (For Unfrozen Food, the value is used as the initial freezing temperature of the food) (From "ASHRAE, 2006")

// Freezing Time
DeltaH = (H-Hf)
hfilm = 20 // W/m²*K
GrapeDimension = 0.006 // m; 6 mm
Dimension = (2*0.006) // m
Ste = ((Cp*(tf-Tm))/(DeltaH))
ks = 0.567 // W/m*K; Thermal Conductivity (From "ASHRAE, 2006")
Tc = t
Pfactor = 0.19665 // For Spheres; (From "ASHRAE, 2006")
Rfactor = 0.03939 // For Spheres; (From "ASHRAE, 2006")
Tetha = Tetha1*Tetha2*Tetha3
Tetha1 = ((DeltaH/(tf-Tm)))
Tetha2 = (((Pfactor*Dimension)/hfilm)+((Rfactor*(Dimension^2))/(ks)))
Tetha3 = ((1)-((1.65*Ste)/ks)*(ln((Tc-Tm)/(-10-Tm))))

// Heat Of Respiration
W = (((10.7*f)/3600)*(((9*t)/5)+32)^g) // W/kg; Heat Of Respiration (From "ASHRAE, 2006")
f = (7.056 * 10^-5) // (From "ASHRAE, 2006")
g = 3.033 // (From "ASHRAE, 2006")

// Transpiration Coefficient
tc = 123 // ng/(kg·s·Pa); Average for all varieties (From "ASHRAE, 2006")

// Surface heat transfer coefficient
qsf = (hsf*Asf*(tsf-tsft)) // Heat transfer rate (From "ASHRAE, 2006")
hsf = 30.7 // W/(m²·K) Surface heat transfer coefficient (From "ASHRAE, 2006")
tsf = t // Surface temperature of the food (From "ASHRAE, 2006")
tsft = 4 // ºC, Surrounding fluid temperature (From "ASHRAE, 2006")
Asf = ((2*(3.14)*(0.0055))*((0.0055)*(0.022))) //m²; Surface area of the food through which the heat transfer occurs (Cylinder 11) (From "ASHRAE, 2006")

// Thermal Load Calculations

qfinal = (q1+q2+q3+q4+q5) // W
qfinalkW = qfinal/1000 // kW
qfinalTR = qfinal/3517 // TR
qfinaltime = (q1perday+q2perday+q3perday+q4perday+q5perday) // kWh

// 1 - TRANSMISSION LOAD

q1 = ((qwall*4)+(qfloor)+(qceiling)) // W; Heat gained by infiltration from the surfaces
q1perday = ((q1*18)/1000) // kWh

qwall = U1*A1*DT // (From "ASHRAE, 2006")

U1 = (1/((x1/k1)+(x2/k2)+(x3/k3))) // W/m²K; is the global coefficient of heat transfer, (From "ASHRAE, 2006")
A1 = Wall_Area // m²; Outside area of section, (From "ASHRAE, 2006")
DT= (T_Environment - St) // K; Difference between the external temperature and the internal air temperature, (From "ASHRAE, 2006")

```
x1 = 0.05 // m; Wall thickness, Steel
x2 = 0.05 // m; Wall thickness, Polysterne
x3 = x1 // m

k1 = 45.3 // W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2018")
k2 = 0.037 // W/(m ·K) ; Polystyrene, expanded (R-142b); W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2006")
k3 = k1 // W/m*K; Thermal conductivity of wall material

qfloor = U2*A2*DT

U2 = (1/((x7/k7)+(x8/k8)+(x9/k9))) // W/m²K; is the global coefficient of heat transfer, (From "ASHRAE, 2006")
A2 = Floor_Area // m²; Outside area of section, (From "ASHRAE, 2006")
HTC = (qfloor/DT) // W/m²K; Heat transfer coefficient
Totalheat_exchangecoefficients = 11 // K (Combined convection and radiation); (From

x7 = 0.25 // m; Wall thickness, Concrete
x8 = 0.2 // m; Wall thickness, Polysterne
x9 = x7 // m

k7 = 1.75 // W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2018")
k8 = 0.037 // W/(m ·K) ; Polystyrene, expanded (R-142b); W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2006")
k9 = k7 // W/m*K;

qceiling = U3*A2*DT

U3 = (1/((x10/k10)+(x11/k11)+(x12/k12))) // W/m²K; is the global coefficient of heat transfer, (From "ASHRAE, 2006")

x10 = 0.05 // m; Wall thickness, Steel
x11 = 0.05 // m; Wall thickness, Polysterne
x12 = x10 // m

k10 = 45.3 // W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2018")
k11 = 0.037 // W/(m ·K) ; Polystyrene, expanded (R-142b); W/m*K; Thermal conductivity of wall material (From "ASHRAE, 2006")
k12 = k10 // W/m*K

// 2 - Product LOAD

q2 = (qrc*1000) // W (From "ASHRAE, 2006")
q2perday = (qrc/18) // kWh

// Q1, Q2, Q3, Q4 = heat removed, kJ
Q1 = (m*c1*(t1-t2)) // kJ; Heat removed to cool from initial temperature to some lower temperature above freezing (From "ASHRAE, 2006")
Q2 = (m*c1*(t1-tf)) // kJ; Heat removed to cool from initial temperature to freezing point (From "ASHRAE, 2006")
Q3 = (m*hif) // kJ; Heat removed to freeze product (From "ASHRAE, 2006")
Q4 = (m*c2*(tf-t3)) // kJ; Heat removed to cool from freezing point to final temperature below freezing point (From "ASHRAE, 2006")
m = 0.007 // kg; 1 single grape = 7 grams, mass of product (From "ASHRAE, 2006")
c1 = 3.71 // specific heat of product above freezing, kJ/(kg·K) (From "ASHRAE, 2006")
t1 = 25 // initial temperature of product above freezing, °C (From "ASHRAE, 2006")
t2 = 0 // lower temperature of product above freezing, °C (From "ASHRAE, 2006")
hif = LH // latent heat of fusion of product, kJ/kg (From "ASHRAE, 2006")
c2 = 2.07 // specific heat of product below freezing, kJ/(kg·K) (From "ASHRAE, 2006")
t3 = -1 // final temperature of product below freezing, °C (From "ASHRAE, 2006")

// Refrigeration capacity
```

```
qrc = ((Q2+Q3+Q4)/(3600*n)) // kW; Average cooling load (From "ASHRAE, 2006")
n = 4 // h, allotted time to freeze (From "ASHRAE, 2006")

// 3 - Internal Load

q3 = (qlights + qforklift +qmotor + qp) // W
q3perday = (qlightstime + qforklifttime+ qmotorday + qpday) // kWh

// Electrical Equipment

light_bulb = 100 // W; Zanotti
qlights = (4 * light_bulb) // W
qlightstime = ((qlights * time_on)/1000) // kWh
time_on = 18 // hour

// Forklifts

// 1 Forklift
// The battery is nomally sized for an 8 hour shift
Battery_Amp = 375 // Ah (Ampère Hour)
Battery_Volt = 72 // VDC; Typical reach truck batteries are 72VDC, but could be as high as 96VDC
qf = (375*72) // W/H
FLT = 3 // kW; Drive Motor
Lift_Pump = 1.3 // kW
Steering_Pump = 5000 // W
// So, somethign between 3 - 5 kW should be about right. (Choose The Average 4 kW)
qforklift = 4000 // W
// Assuming an 8 hour shift
qforklifttime = ((qforklift * 8)/1000) // kWh

// Typical Electric Motors
// 3-Phase
mr = 0.56 // kW; Motor Rated (From "ASHRAE, 2006")
rpm = 1750 // rpm; Nominal RPM (From "ASHRAE, 2006")
efficiency = 0.72 // 72%; Full Load Motor Efficiency (From "ASHRAE, 2006")
qmotor = 557 // W; Motor Out, Driven Equipment in; Type B (From "ASHRAE, 2006")
qmotorday = ((557*18)/1000) //  kWh

// People

qp = (272 - (6*t)) // W; Heat load from the people working there(From "ASHRAE, 2006")
qpday = ((qp*18)/1000) // kWh

// 4 - INFILTRATION AIR LOAD

q4 = ((qinfiltration)) // W
q4perday = ((qinfiltration*18)) // kWh

// Infiltration by Air Exchange
qinfiltration = (0.577*Wdoor*(DH^1.5)*(Qs_A)*(1/Rs)) // kW; sensible and latent refrigeration load (From "ASHRAE, 2006")
(Qs_A) = 12.5 // kW/m²; Sensible heat load of infiltration air per square meter of doorway opening (From "ASHRAE, 2006")
DH = 2.1 // m; Door Height (From "Igloodoors)
Wdoor =  1.1 // m; Doorway width (From "Igloodoors)
Rs = 0.57 // Sensible heat ratio of the infiltration air heat gain (From "ASHRAE, 2006")

// 5 - EQUIPMENT RELATED LOAD
```

```
q5 = (qfans + qfansdesfrosting) // W
q5perday = ((qfansperdays + qfansperdesfrostingdays)/1000) // kWh

// Fans Motors
qfans = (n_fans * watts) // W (From Frigosys)
n_fans = 3 // (From Frigosys)
watts = 300 // W (From Frigosys)
// Per day
qfansperdays = (qfans*operation_time) // Wh (From Frigosys)
operation_time = 18 // h; Its common to use 16 - 18 hours, because the time to the equipments recover is something between 6 - 8 hours. (From Frigosys)

// Fans Motors (Defrosting)
qfansdesfrosting = (Fans_Power * Fans_DefrostCycle * Fans_Efficiency) // W
Fans_Power = 1500 // W (From Frigosys)
Fans_DefrostCycle = 3 // Times Per Day (From Frigosys)
Fans_Efficiency = 0.3 // 30% (From Frigosys)
// Per day
qfansperdesfrostingdays = (qfansdesfrosting*defrosting_time) // Wh (From Frigosys)
defrosting_time = 0.4 // h (From Frigosys)
```

APPENDIX A.2 – CODE II: VAPOR COMPRESSION REFRIGERATION CYCLE.

8.2 Code II: R-134a Vapor Compression Refrigeration System

```
// Normal (Vapor Compression)
//Assumptions
// 1 - Steady State (From "ASHRAE, 2005")
// 2 - HFC R-134a (Tetrafluoroethane) (From "ASHRAE, 2005")
// 3 - Refrigeration Load = Qevaporatornormalcycle (From "ASHRAE, 2005")
// 4 - Direct-Expansion (From "ASHRAE, 2005")
// 5 - Single-Stage (From "ASHRAE, 2005")
// 6 - Since The R-134a Compression Vapor Cycle is considered to be normal one (Standart), them they will refered as "nc", of normal cycle.

// (8 -> 1) -> (Point 1) - Saturated Vapor
// (Point 1 -> Point 2) - (Isenthropic Compression)
// (2 -> 3) -> (Point 2) - Superheated Vapor
// (Point 2 -> Point 3) - (Condensation)
// (4 -> 5) -> (Point 3) - Subrefrigerated Liquid
// (Point 3 -> Point 4) - ((Isenthalpic Expansion))
// (6 -> 7) -> (Point 4) - Liquid + Vapor
// (Point 4 -> Point 1) - (Evaporation)

// 2-3 - Compressor -> Condenser
// 4-5 - Condenser -> Expansion Valve
// 6-7 - Expansion Valve -> Evaporator
// 8-1 - Evaporator - Compressor

// Data

// Tevapn = -6 // °C
// Tcondn = 50 // °C

T1n = -6 // °C; Tevapo
T1ninK = T1n + 273.15 // Tevapo
T11n = 4 // °C; Tevapo + DTSHn
T11ninK = T11n + 273.15
T2n    =    50    //    °C;    Tconden
T2najusted = (50 - DTSRn) // °C
T2ajustedinK = T2najusted + 273.15
T2ninK = T2n + 273.15 // °C; Tconden
T2sn = T_Ph("R134A", P2n, h2sn) // °C
T2sninK = T2sn + 273.15
T2snajusted = T_Ph("R134A", P2najusted, h2snajusted)  // °C
T2sninKajusted = T2snajusted + 273.15
DTSHn = 10 // °C
DTSRn = 5 // °C
P1n = Psat_T("R134A", T1n) // kPa; Pevap
P2n = Psat_T("R134A", T2n) // kPa; Pconden
P2najusted = Psat_T("R134A", T2najusted) // kPa; Pconden ajusted
h1n = h_PT("R134A", P1n, T1n) // kJ/kg
h11n = h_PT("R134A", P1n, T11n) // kJ/kg
s11n = s_PT("R134A", P1n, T11n) // kJ/kg.K
s2sn = s11n // kJ/kg.K
h2sn =  h_Ps("R134A", P2n, s2sn)  // kJ/kg
h2snajusted =  h_Ps("R134A", P2najusted, s2sn)  // kJ/kg

// Compressor

// Power needed with a compressor = qfinalkW
```

```
mn = (((9.774)/(h11n-h4n))) // kg/s
mnhour = mn*3600 // kg/h
//eta = (h2sn - h11n) / (h2n - h11n)
etan = 0.7
h2n = (((h2sn-h11n)/etan)+h11n)
h2najusted = (((h2snajusted-h11n)/etan)+h11n)
T2realn = T_Ph("R134A", P2n, h2n) // °C
T2realninK = T2realn + 273.15
T2realnajusted = T_Ph("R134A", P2najusted, h2najusted)  // °C
T2realninKajusted = T2realnajusted + 273.15

// Condenser

mn2 =0.08027
mn2hour = 289 // kg/h
P3n = P2n
h3n = hsat_Px("R134A", P3n, 0)
T3n = T_Ph("R134A", P3n, h3n)
T3ninK = T3n + 273.15
T31n = (T2najusted - DTSRn) // (Subref)
T31ninK = T31n + 273.15
h31n = h_PT("R134A", P3n, T31n)
Qcondensern = QLnreal + Wcrealn

// Evaporator

P4n = P1n // kPa
h4n = h3n  // kJ/kg
T4n = T1n // °C
T4ninK = T4n + 273.15
x4n = x_hP("R134A", h4n, P4n)

// Others

QLn = mn* (h11n - h4n)
QLnreal = mn2*(h11n - h4n)
QHn = mn* (h2sn - h3n)
Wcn= mn* (h2sn - h11n)
Wcrealn = mn* (h2n - h11n)
COPcoolingisen = (QLn/Wcn)
COPcoolingisreal = (QLn/Wcrealn)
```

APPENDIX A.3 – CODE III: AQUA-AMMONIA ABSORBER REFRIGERATION CYCLE.

8.3 Code III: Aqua-Ammonia Absorber Refrigeration System

```
// ABSORPTION COOLING, HEATING, AND REFRIGERATION EQUIPMENT
// AMMONIA/WATER ABSORPTION (NH3H2O)

//Assumptions
// 1 - Steady State
// 2 - Assuming thermodynamic equilibrium at all points of the cycle
// 2 - The Energy balances depends on a generator, an evaporator, a condenser, an absorber, a heat exchanger, a pump and valves (expasion valves) (From "ASHRAE, 2005")
// 3 - Pressure losses in the pipes and all heat exchangers are negligible;
// 4 - Heat exchange release to surroundings does not occur.
// 5 - No pressure changes except through flow restrictors and pump; (From "ASHRAE, 2005")
// 6 - Ammonia at the generator and evaporator outlets is assumed as saturated vapor
// 7 - Ammonia at the condenser outlet is saturated liquid
// 8 - Flow restrictors are adiabatic (From "ASHRAE, 2005")
// 9 - No liquid carryover from evaporator to absorber (From "ASHRAE, 2005")
// 10 - Single-Stage (From "ASHRAE, 2005")
// 11 - H20 -> Absorbant; NH3 -> Refrigerant/Coolant

// Data

// Qevap = Q = 9774 // W
// Tevaporator = -6 // °C
// Tcondenser = 45 // °C
// PG = Generator Pressure
// PC = Condenser Pressure
// PG = PC = PH
// PE = Evaporator Pressure
// PL = PE
// "x" are the ammonia mass fractions in the solution.
Qevaporator = 9.774 // kW
Tevaporator = -6 // °C
Tcondenser = 45 // °C
Tabsorber = 52 // °C
Tgenerator = 120 // °C
Xr = 0.98 // Concentration of NH3 in refrigerant
Xs = 0.42 // Concentration of NH3 in Solution
Xw = 0.38 // Concentration of NH3 in absorbent
TE = 2 // °C; Temperature of the evaporator
PH = Psat_T("Ammonia", Tcondenser) // kPa; High Pressure Part
PL = Psat_T("Ammonia", Tevaporator)  // kPa; Low Pressure Part

// Point 2 (Saturated Liquid)

P2 = PH
X2 = Xr
T2 = Tcondenser
h2 = hsat_Px("Ammonia", P2, 0) // kJ/kg

// Point 3 (Expansion Of refrigerant Through Expansion Valve From High Pressure To Low Pressure At Constant Enthalpy - Biphasic Region)

h3 = h2 // kJ/kg
T3 = Tevaporator // °C
P3 = PL // kPa

// Point 4 (Extraction Of Heat By Low Pressure Ammonia Vapor In The Evaporator)
```

```
P4 = PL // Saturation Pressure in Evaporator; kPa
T4 = Tevaporator
h4 = hsat_Px("Ammonia", P4, 1) // kJ/kg

// Heat Extracted By Evaporator

QE = Qevaporator // kW
QE = (mr*(h4-h3)) // kW

// Mass Balance Equation

// Mass Of Solution (ms) = Mass Of Refrigerant (mr) + Mass Of Absorbent (mw)
// ms = ?
// mr = 0.009582 kg/s
mrhour = mr*3600 // kg/h
// mw = ?

// Material Balance Equation For NH3

ms = mw + mr
// (ms * Xs) = (mw * Xw) + (mr * Xr)
((mw + mr) * Xs) = (mw * Xw) + (mr * Xr)
// mw = 0.1342 kg/s
mwhour = mw*3600 // kg/h
// ms = 0.1437 kg/s
mshour = ms*3600 // kg/h

// Point 1 (Ammonia Vapour Entering The Condenser As A Saturated Vapor)

P1 = PH
X1 = Xr
T1 = Tcondenser
h1 =  hsat_Px("Ammonia", P1, 1) // kJ/kg

// Heat rejected by condenser

QC = (mr*(h1 - h2)) // kW

// Point 5 (Strong Solution Entering The Pump As Saturated Liquid)

P5 = PL
X5 = Xs // 0.42
T5 = Tabsorber
h5 = 50 // kJ/kg; (Using enthalpy-concentration diagram for Ammonia/Water)

// Point 6 (High Pressure Saturated Strong Solution Entering The Generator)

P6 = PH
X6 = Xs
T6 = Tabsorber
h6 = 75 // kJ/kg; ((Using enthalpy-concentration diagram for Ammonia/Water)
Wp = (ms*(h5-h6)) // kW


// Point 7 (Weak Solution Leaves The Generator At Saturation Temperature Of Generator)

P7 = PH
X7 = Xw
T7 = Tgenerator
```

h7 = 275 // kJ/kg ((Using enthalpy-concentration diagram for Ammonia/Water)

// Heat Added To Generator

QG = ((mr*h1)+(mw*h7)-(ms*h6)) // kW

// Point 8 (Absorber)

P8 = PL
T8 = T7
h8 = h7

// Heat Rejected In The Absorber

QA = ((mw*h8) + (mr*h4) - (ms*h5)) // kW

// COP

COP = QE/QG // (Without Pump Work (Wp))
COPmax = (((Tevaporator + 273.15) * (Tgenerator - Tabsorber)) / ((Tgenerator + 273.15) * (Tcondenser - Tevaporator)))