Design of a Refrigeration System

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Abstract

This report documents the design of a refrigeration system for commercial use in a restaurant. As specified by the design requirements [1], the refrigeration system must have a load capacity of 1.5 tons (5.28 kW) and must be able to maintain the food storage area at 20˚F at an ambient temperature of 80˚F. The refrigeration system was designed based on the ideal vapor-compression refrigeration cycle, with no inefficiencies accounted for in the design procedure. Losses were considered after the system had been designed and any further work on the refrigeration system must take into account the limitations imposed by actual systems. The design was driven by an objective of a coefficient of performance (COP) near 5 for the system based upon a reading of the relevant literature [2] and [3]. The literature also provided an initial estimation for the operating temperature of the condenser and the evaporator. Based upon recommendations from industry, the condenser must operate at a temperature at least 10˚ greater than the surroundings, and the evaporator must operate at a temperature at least 10˚ colder than the refrigerated area. A number of different temperature combinations were investigated, with the final design calling for an evaporator operating temperature, T_{evap}, of 10 ˚F (260.9 K) and a condenser operating temperature, T_{cond}, of 90˚F (305.4 K). The resulting ideal COP for the refrigeration system is 4.70, which was judged to be within an acceptable range of the stated design goal. These temperatures set the rest of the system parameters based upon the assumptions of the ideal vapor-compression refrigeration cycle. The mass flow rate of the environmentally-friendly refrigerant R-134a, is 0.036 kg/s, and the compressor electrical power required is 1.12 kW assuming isentropic operation. The cost to run the compressor per hour is $0.14. The total cost of the four main components, compressor, condenser, throttling valve, and evaporator, plus copper tubing is $2127.25. The compressor, condenser, and throttling valve are combined into a single unit as is common practice in refrigeration systems [4]. An estimation of the realistic coefficient of performance of the system was carried out after the design had been completed and yielded a value of 3.18. This estimate accounted for the inefficiency that would be inherent to the compressor as well as the power required to run 2 fans. The final system will enable the safe storage of food at the restaurant and meets all the prescribed design requirements.

Table of Contents

Section 1: Introduction

A refrigeration system must be designed for use in a proposed restaurant. The refrigeration system is required to maintain perishable food storage at a temperature of 20˚ F (266.48 K) at an ambient air temperature of 80˚ F (299.82 K). The load capacity of the system must be 18000 BTU/hr (5.275 kW) in order to keep the items at the required temperature. The refrigeration system must be efficient and should operate with the environmentally-friendly R 134a as the refrigerant. The system will be designed based on the ideal vapor-compression cycle with a compressor, heat exchanger for condensing, an adiabatic throttle, and heat exchanger for evaporating the refrigerant. Losses were not accounted for in the design but were later estimated to gauge the efficiency of the system. Temperature-entropy and pressure- enthalpy diagrams were used extensively in the design of the refrigeration system.

Section 2: Nomenclature

- Subscript numbers refer to the stage of the vapor-compression refrigeration cycle
- Subscript in refers to work put into the system or heat absorbed by the evaporator
- Subscript out refers to the heat rejected by the condenser
- Subscript H refers to the temperature of the hot reservoir, the ambient surroundings
- Subscript L refers to the temperature of the cold reservoir, the refrigerated food storage area

Symbol	Description	Units
COP	Coefficient of performance	-
h	enthalpy	kJ/kg
$\dot{m}$	Mass flow rate	kg/s
P	pressure	kPa
$\dot{Q}$	Rate of heat released or absorbed	kW
s	entropy	kJ/kg K
T	temperature	K
$\dot{W}$	Power	kW
χ	Quality of refrigerant	-
Η	Efficiency of compressor	-

Section 3: Methods

3.1 Assumptions

- There is no heat loss to the environment through the pipes or tubing of the system, and there is no unintended exchange of mass or heat between the system and the surroundings
- There is no difference in gravitational potential energy or kinetic energy between any stages of the system
- The cycle works in steady state operation. The mass flow rate remains constant throughout the system and the specified operating and ambient temperatures do not change over the course of operating.
- The properties of R-134a are from [5] which is assumed to be the most accurate reference for fluid properties available to the design team.
- The refrigerant R-134a does not react detrimentally with the tubing and piping. R-134a is also environmentally friendly and safe for residential and consumer refrigeration applications [6].
- Designs with a coefficient of performance above 5 are not feasible using a simple compressor according to [1] and [4]. Designs with a COP greater than 5 were therefore discarded from the final design although a COP higher than 5 may be possible with a variable compressor.
- The condenser and evaporator are simple heat exchangers that are assumed to operate without losses and were considered out of scope for this design.
- Frictional effects were not accounted for in the design of the system.
- The design is based on the ideal vapor-compression cycle which results in the following assumptions:
- The compressor operates reversibly and adiabatically. This means that the compression process is isentropic and the entropy of the refrigerant at stage 2 is equal to the enthalpy of the refrigerant at stage 1. No losses were taken into account for the compressor in the initial design, meaning the compressor was taken to operate at 100% efficiency.
- The condenser and evaporator operate at constant pressure.
- The throttling valve operates adiabatically. The enthalpy of the refrigerant at stage 4 is therefore equal to the enthalpy of the refrigerant at stage 3.
- The refrigerant leaves the condenser as a saturated liquid and leaves the evaporator as saturated vapor. The refrigerant leaves the compressor as a superheated vapor and leaves the throttling valve as a saturated mixture of variable quality.

3.2 Design Procedure

The ideal vapor compression cycle consists of four stages, with four separate processes connecting the stages. At stage 1, the refrigerant is a saturated vapor; at stage 2, the refrigerant is a superheated vapor; at stage 3, the refrigerant is a saturated liquid; and at stage 4, the refrigerant is a saturated mixture of unknown quality. Stages 1 and 2 are connected by a compressor that is assumed to operate isentropically, meaning it is both a reversible and adiabatic process. Therefore, the entropy of the refrigerant at stage 2 is equal to the entropy of the refrigerant at stage 1. Stages 2 and 3 are connected by a heat exchanger that condenses the superheated vapor into a saturated liquid in a constant pressure process. Next, between stages 3 and 4, the refrigerant is expanded through an adiabatic throttle. However, this process is not reversible and is therefore not isentropic. The enthalpy at stage 4 is equal to the enthalpy at stage 3 because the expansion is adiabatic. Finally, the refrigerant passes through another heat exchanger from stage 4 to stage 1 that evaporates the saturated mixture into a saturated vapor in a constant temperature and constant pressure process. The stages and the processes are summarized in Table 1 and Table 2 along with the assumptions from the ideal vapor-compression refrigeration cycle. The initial design did not account for any fans or pumps that would be necessary to move the refrigerant through the system.

Stage	Refrigerant State
1	Saturated Vapor
2	Superheated Vapor
3	Saturated Liquid
4	Saturated Mixture

Table 1: Stage Characteristics

Stages	Process	Device	Ideal Assumptions
1→2	Isentropic compression	Compressor	s2 = s1
2→3	Constant pressure condensation	Condenser (Heat Exchanger)	P3 = P2
3→4	Adiabatic expansion	Throttle	h4 = h3
4→1	Constant pressure evaporation	Evaporator (Heat Exchanger)	P1 = P4 T1 = T4

Table 2: Process Definitions

Energy Conservation

In order to facilitate the design of the refrigeration system, an energy conservation equation was written for each process. Each process can be characterized in terms of the enthalpies at the beginning and end of the process and the mass flow rate of the refrigerant which was assumed to be constant through the system. The energy conservation equations are derived from [1]. The energy conservation for the compressor is written:

$$\dot{W}_{in} = \dot{m} * (h_2 - h_1) \qquad \textbf{(1)}$$

Where $\dot{W}_{in}$ is the work input by the compressor in kW, $\dot{m}$ is the mass flow rate of the refrigerant in kg/s, and h, the enthalpy, is in kJ/kg. The initial design did not account for any fans or pumps, meaning that the compressor work was the only work input to the system. The energy conservation for the condenser is written:

$$\dot{Q}_{out} = \dot{m} * (h_2 - h_3) \qquad \textbf{(2)}$$

Where $\dot{Q}_{out}$ is the heat rejected in kW by the refrigerant as it condenses from a superheated vapor to a saturated liquid, the mass flow rate is in kg/s, and the enthalpies are in kJ/kg. The condensation process occurs in a heat exchanger, with the rejected heat exhausted to the surroundings, ideally outside of the restaurant. The energy conservation for the throttle is relatively simple as it is assumed to operate adiabatically:

$$h_3 = h_4 \qquad \textbf{(3)}$$

No useful work is extracted from the throttle and it serves only to expand the saturated liquid into a saturated mixture of undetermined quality at stage 4. The energy conservation for the evaporator is:

$$\dot{Q}_{in} = \dot{m} * (h_1 - h_4) \qquad \textbf{(4)}$$

Where $\dot{Q}_{in}$ is the heat absorbed by the refrigerant in kW as it evaporates from a saturated mixture into a saturated vapor, the mass flow rate is in kg/s, and the enthalpies are in kJ/kg. The evaporation process takes place in a heat exchanger. The heat absorbed by the refrigerant is also known as the load capacity of the refrigeration system and was provided to the design team in [1]. The load as expressed by the requirements was in the standard unit of tons, which was converted into SI units in kW to fit with the conventions of this report. The work and heat inputs/outputs can be calculated for each station once the enthalpy at each station had been determined.

Coefficient of Performance

The coefficient of performance is the critical parameter that drives the design of a refrigeration system. The coefficient of performance for a mechanical vapor compression cycle is defined in [1] as the ratio of "useful refrigerating effect" to "net energy supplied from external sources." This can also be expressed as the heat removed from the system by the evaporating refrigerant to the electrical work supplied to the system to run the compressor and fans or pumps. Using the energy conservation equations, the coefficient of performance can be expressed in terms of the enthalpy of the refrigerant at the various stages as shown in [1]:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_{in}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{5}$$

Where the COP of performance is a dimensionless number, and the mass flow rates of the refrigerant cancel because they are assumed to be equal through the entire system. $\dot{Q}_{in}$, the heat absorbed by the refrigerant as it evaporates, is also the refrigerating load of the system and was provided in the system requirements. Typical coefficients of performance were found to range from 3-5 with a simple compressor [1] and[4], with higher numbers being possible using a variable speed compressor. However, the variable speed compressor was considered to be out of scope for this design, and the upper limit for the coefficient of performance was therefore selected to be around 5. Achieving a coefficient of performance near 5 was the driving objective of the design.

Condenser and Evaporator Operating Temperatures

The two variables that could be altered in the design were the operating temperatures of the condenser and the evaporator. Based on the design requirements, the perishable food storage area must be maintained at a temperature of 20˚F (266.5 K). Therefore, the evaporator must operate at a temperature lower than this value in order to absorb the required heat from the storage area. From a study of the literature, [1] and [3], a suitable temperature difference of 10˚ between the storage area and the evaporator operating temperature was selected. This temperature would allow for the correct temperature as stated in the design requirements to be achieved while limiting the size required by the heat exchanger. After a suitable evaporating operating temperature had been selected, the operating temperature of the condenser needed to be determined. Again, a study of the literature showed that a difference of 10˚ between the condenser and the ambient surroundings (at 80˚F) was acceptable for a small commercial refrigeration system. While the evaporator must operate at a lower temperature than the food storage temperature, the condenser must operate at a higher temperature than the surroundings in order to reject heat as the refrigerant condenses. The closer the condenser operating temperature to the ambient value, the larger the heat exchanger required for the heat to be rejected. Therefore, the investigation of possible designs for the refrigeration system began with an evaporator operating temperature 10˚ below that of the required food temperature, and the condenser started off with a temperature 10˚ above the ambient value. Once these two temperatures had been selected, it was possible to determine the necessary conditions at each stage to evaluate the enthalpy at each stage. Ultimately, this allowed for the coefficient of performance to be calculated and the operating temperatures could be adjusted until a suitable value was achieved.

Enthalpy at Each Stage

Determining the enthalpy at each stage was straightforward once the operating temperatures of the evaporator and condenser had been determined. The state of the refrigerant at each stage and the assumptions of the ideal vapor-compression cycle as outlined in Table 1 and Table 2 were critical to this procedure. All enthalpies were calculated using the NIST Fluid Properties Database [5] with Refrigerant-134a treated as a pure fluid. Because there is only one component present in the system, all the fluid properties for the refrigerant can be determined with a specification for two of the properties. The first enthalpy that could be determined was that at Stage 1. The refrigerant is a saturated vapor at this stage, meaning that its quality (the ratio of the mass of vapor to the total mass of the mixture) is 1.0. Moreover, the temperature at state 1 is equal to the temperature of the evaporator. Therefore, the enthalpy at state 1 can be determined using the quality and the temperature. The next enthalpy that was calculated was that at stage 3. The temperature of the refrigerant at stage 3 was the condenser operating temperature, and as the refrigerant exists as a saturated liquid, the quality is 0. The enthalpy at stage 4 is simply equal to the enthalpy at stage 3 because the throttle is assumed to operate adiabatically. The other property that characterizes stage 4 is that the pressure is equal to the pressure at stage 1 as the evaporator is assumed to operate isobarically. Finally, the enthalpy at station 2 can be determined using the assumption that the compression is isentropic meaning the entropy of stage 2 is equal to that at stage1, and that the condenser operates isobarically which sets the pressure at stage 2 equal to the pressure at stage 3. Once all of the enthalpies had been calculated, the COP for the system was found using Equation 5. The parameters used to specify enthalpy at each stage are presented in Table 3.

Stage	Parameters Used to Specify Enthalpy
1	$T_1 = T_{evaporator}$; $\chi_1 = 1$
2	$s_2 = s_1$; $P_2 = P_3$
3	$T_3 = T_{evaporator}$; $\chi_3 = 0$
4	$h_4 = h_3$

Table 3: Specification of Enthalpy

Mass Flow Rate and Compressor Work

Once the enthalpy at each stage had been computed, the mass flow rate of R-134a through the system could be calculated using the specified refrigeration load and the energy conservation equation for the evaporator. The equation for the mass flow rate is:

$$\dot{m} = \frac{\dot{Q}_{in}}{h_1 - h_4} \quad \textbf{(6)}$$

With the mass flow rate measured in kg/s. This mass flow rate was assumed to be constant throughout the system as the system was isolated from the surroundings with no inflows or outflows. The compressor work can then be solved for with the mass flow rate:

$$\dot{W}_{in} = \dot{m} * (h_2 - h_1) \quad \textbf{(7)}$$

Where the work is measured in J/s or Watts. Once the compressor work had been determined, the hourly cost to run the compressor could be calculated by multiplying the power requirement of the compressor in kW by the average price per kWh of electricity in the Ohio area as determined from [7].

$$\frac{\$}{hr} = \dot{W}_{in} * \frac{\$}{kWh} \quad \textbf{(8)}$$

Carnot COP

The maximum theoretical coefficient of performance for refrigeration is represented by the ideal Carnot Cycle. The COP for an ideal Carnot Cycle is given in terms of the temperature of the cold reservoir and the temperature of the hot reservoir, or in this case, the temperature of the food storage area and the ambient temperature. The expression for Carnot COP comes from [1]:

$$COP_{carnot} = \frac{1}{\frac{T_H}{T_L} - 1} \quad \textbf{(9)}$$

3.3 Selection of Parameters

The two parameters that could be altered in the design of the refrigeration system were the condenser operating temperature, T_{cond}, and the evaporator operating temperature, T_{evap}. The design goal of the refrigeration system was a COP near 5, which drove the temperatures chosen. From the literature, [1] and [3], it was determined that the evaporator should operate at a temperature at least 10˚ colder than the storage requirements and the condenser should operate at a temperature at least 10˚ hotter than the surroundings. Reducing the temperature difference between the condenser and the evaporator will increase the COP, but the closer the operating temperatures are to the surroundings and the refrigerated space respectively, the larger the heat exchanger that is required to transfer the heat as heat transfer is proportional to temperature difference. Therefore, the operating temperatures are limited by the physical requirements of the heat exchanger, and the temperature difference of 10˚ is reasonable for the design. Temperatures of $T_{cond} = 90^{\circ}F\ and\ T_{evap} = 10^{\circ}F$ served as the starting point for the design process. Although these temperatures would ultimately be selected as the final design specifications, a number of different temperature configurations were investigated to determine the effect of the operating temperatures on the COP. The alternatives that were explored are shown in Section 5.2. The design procedure as outlined in Section 3.2 was implemented in Excel to allow for the rapid evaluation of different temperature combinations, and the NIST fluid properties database was extensively utilized to determine the properties of R-134a at the different conditions. The final operating temperatures chosen resulted in a COP of 4.70 which was deemed to be within an acceptable range from 5, the upper limit for a refrigeration system operating with a standard compressor. The temperatures selected then determined the rest of the parameters for the refrigeration system. The resulting mass flow rate of R-134a was 0.0360 kg/s, and the power requirement for the compressor was 1.212 kW. Per hour, the compressor will cost $0.13 to run. The specifications for the refrigeration system are displayed in Table 4: Specified System Parameter

Section 4: Results

4.1 Specifications

Parameter	Value
Condenser Operating Temperature, T_{cond}	90˚F (305.4 K)
Evaporator Operating Temperature, T_{evap}	10˚F (260.9 K)
Ideal Coefficient of Performance (COP)	4.70
Mass Flow Rate of R-134a, $\dot{m}$	0.0360 kg/s
Work done by compressor, $\dot{W}_{in}$	1.12 kW
Operating Cost per Hour	$0.14
Estimated Compressor Efficiency, η	0.80
Estimated power for 2 fans	0.26 kW
Estimated Realistic COP	3.18

Table 4: Specified System Parameter

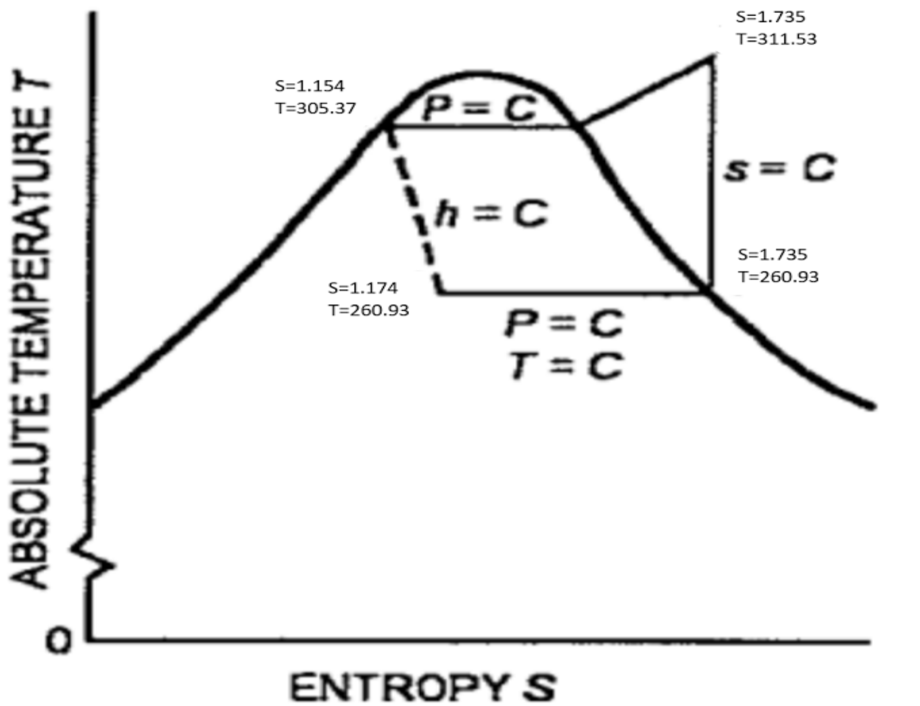

Figure 1: Temperature vs. Entropy Diagram

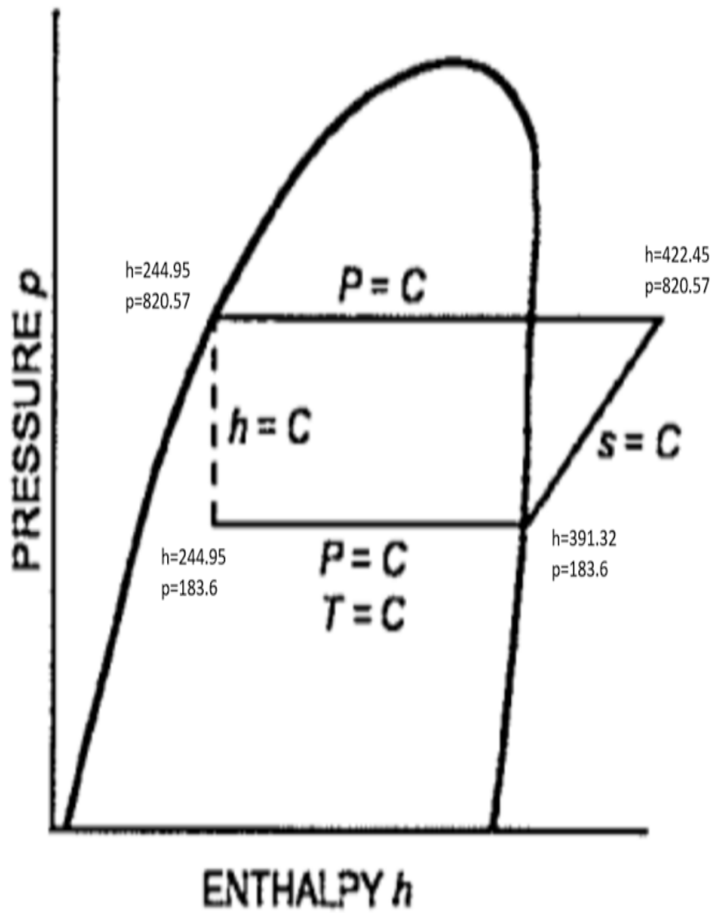

Figure 2: Pressure vs. Enthalpy Diagram

4.2 Bill of Materials and Cost Analysis

Item	Cost
64” x 5” x 2.8125” Evaporator Coil	$ 260.59
Condensing Unit with 1.1kW Compressor and throttling valve	$ 1755.71
½” Copper Refrigeration Tubing, 100’	$ 110.95
Total	**$2127.25**

Table 5: Bill of Materials from [9], [10], [11]

Cost Analysis

The first component to be selected was the compressor, to ensure that its power usage would meet the refrigeration system’s design requirements of 1.122 kW. In order to reduce the risk of mismatched or incompatible components, a condensing unit was chosen as a combined unit with the compressor, which also included the appropriate motor electronics and speed controller, in addition to condenser fans and a refrigerant dryer to ensure low refrigerant humidity and maximized lifetime. The combined compressor and condenser also includes the necessary throttling valve to expand the refrigerant from a saturated mixture to a saturated vapor. It is standard practice to combine the compressor and condenser into a single unit [1]. The condenser combined with the compressor was selected to reject the 6.4 kW required by the system. The evaporator coil was sourced from the same company to improve compatibility and reduce system variability. The evaporator coil is both physically compatible with the system and capable of sustaining the 5.3kW of heat transfer intake required.

Once the evaporator, condenser, compressor, and auxiliary components were selected, copper refrigeration tubing was also chosen to route the refrigerant from the condensing unit to the evaporating unit. One hundred total feet of compatible tubing was chosen to yield a maximum distance of fifty feet between the units, a reasonable length to promote functionality without incurring major thermal losses.

Section 5: Discussion

5.1 Inefficiencies/Losses

The system was initially designed without taking into account inefficiencies associated with the components. The reported COP of 4.70 was calculated assuming 100% efficiency of the compressor and that the compressor is the only work input required for the system. This is not indicative of an actual refrigeration system, which would require pumps and fans to drive the refrigerant through the system and a real compressor will not operate at 100% efficiency. These factors will tend to decrease the actual COP of the system. A brief analysis of the additional elements required and the efficiency of the compressor allows for a more accurate calculation of the actual coefficient of performance.

Compressor Type	Efficiency, η
Centrifugal	0.70 – 0.85
High-Speed Reciprocating	0.72 – 0.85
Low-Speed Reciprocating	0.75 – 0.90
Rotary Screw	0.65 – 0.75

Table 6: Compressor Efficiencies from [12]

Table 6 shows typical compressor efficiency values. The most common type of compressor used in a refrigeration system is a reciprocating according to [13]. Therefore, a realistic estimate of the efficiency of the compressor is 80%. According to [12], real efficiencies are difficult to determine except through testing. Therefore, the 80% efficiency is to be considered only as an estimate. The actual work required to run the compressor is increased by the inefficient realistic operation:

$$\dot{W}_{actual} = \frac{\dot{W}_{in}}{\eta} \tag{10}$$

Moreover, the actual coefficient of performance is calculated not only from the work to the compressor, but the net work into the system which includes the power to drive the fans:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_{net}};\ \dot{W}_{net} = \dot{W}_{actual} + \dot{W}_{fans} \tag{11}$$

Values for the fan power input are from the ASHRAE Fundamentals Handbook section on Thermodynamics and Refrigeration Cycle. Two fans are needed for the system, a fan on the condenser, and a fan on the evaporator. A typical condenser fan requires 0.15 kW and a typical evaporator fan requires 0.11 kW according to [1]. These are only estimates for the power requirement and were chosen to allow for a rough estimate of an actual COP for the system. The actual COP can now be calculated using the above equations:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_{net}} = \frac{5.275}{\frac{1.12}{0.8} + 0.15 + 0.11} = 3.18 \tag{12}$$

As can be seen, the estimated actual COP is much lower than the 4.70 figure calculated for ideal 100% efficiency operation of the compressor and using no fans. This is only an estimate, but it more accurately reflects the operating conditions expected in a real-world situation. Along with the decrease in COP comes an increase in the cost/hr to run the system. Further reduction of the COP would likely occur because of additional losses. The estimate of 318 includes only the inefficiency associated with the compressor and the power needed to run two fans on the evaporator and condenser Further work on the design should include a thorough examination of the actual COP and efforts to improve the efficiency of the system. This brief analysis underscores the difference between ideal and realistic system operation.

5.2 Alternative Designs

When choosing operating temperatures for the condenser and evaporator, several temperature combinations were evaluated. The designs were classified in terms of the difference between the operating temperature of the component and the ambient temperature. For example, the 15˚ option had a condenser temperature of 95˚ F (80˚ F was the ambient temperature) and the evaporator was held at 5˚ F (the refrigerated space was at 20˚ F). The primary other two options discussed below had differences of 10˚ and 5˚. The main considerations at each temperature difference were the coefficient of performance of the system, and the heat exchange that would be required with the specified operating temperatures. After performing calculations for each option, the coefficient of performance criteria eliminated the 15˚ design with a COP of 4.00. The 5˚ option had a significantly higher CoP at 5.60. However, since the difference between the ambient and operating temperatures was small, a large heat exchanger would be required to both absorb the heat from the refrigerated space and reject it to the surroundings. The heat exchangers that serve as the compressor and evaporator would be prohibitively outsized and expensive for the 5˚ option to work in a reasonable system. Therefore, this design was also eliminated, leaving the 10˚ option. This option had a COP of 4.70 which, while not as high as the 5˚ design was still judged to be close enough to the design goal of 5 while remaining a realizable system according to [1] and [4].

In analyzing the designs to see how the various temperatures affected performance characteristics, a new quantity was defined. The total operating temperature difference (TTD) is defined as the sum of the differences in temperatures between the refrigerated space and the evaporator and between the condenser and the surroundings:

$$\Delta T_{total,operating} = (T_L - T_{evaporator}) + (T_{condenser} - T_H) \quad \textbf{(13)}$$

Total temperature difference values ranged from 10 to 50˚ F. Figure 3 shows the cost/hr versus total operating temperature difference and Figure 4 shows the coefficient of performance versus the total operating temperature difference.

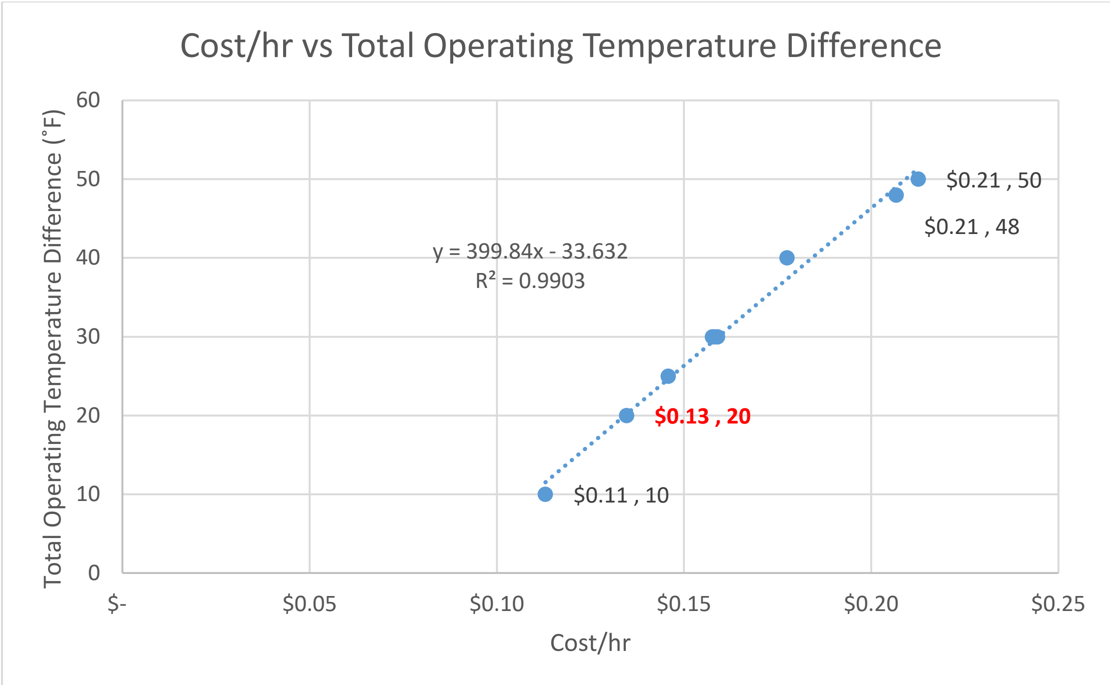

Figure 3: Cost/hr vs. Total Operating Temperature Difference

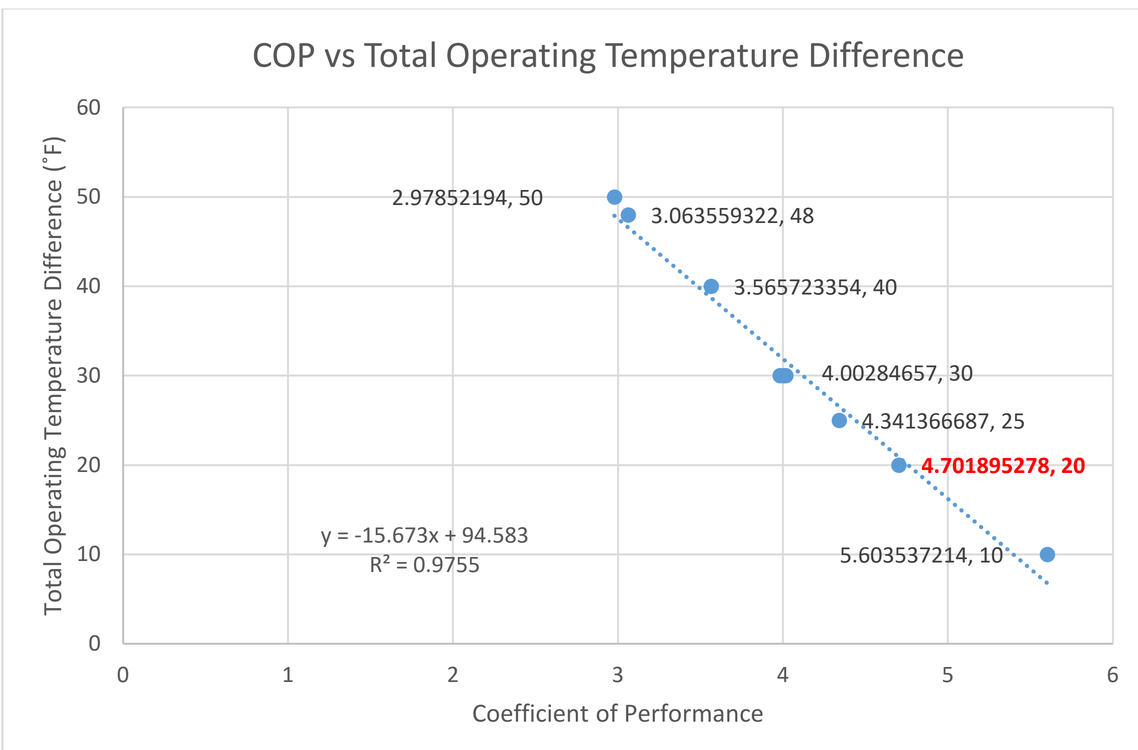

Figure 4: Coefficient of Performance vs Total Operating Temperature Difference

By studying the cost versus TTD, it can be seen that as the temperatures of cooling system elements are moved further away from ambient temperatures, the energy needed to operate those systems increases. At first, this might suggest that the design with the smallest TTD was the optimal design, however, the smaller TTD the larger the heat exchangers needed and the designs with the smallest TTDs would be unfeasible to build.

Next, the COP was graphed in relation to the TTD. As the TTD decreased, the coefficient of performance increased. This makes sense from the system analysis as the smaller temperature difference would require less power input from the compressor. This smaller requirement would result in a more efficient system when considering only the coefficient of performance and work into the compressor. Once again, though, the option with the greatest COP is not feasible because of the limitations imposed by the heat exchangers. Therefore, the 20˚ F TTD option (10˚ F difference on each element) is the final design selection based upon the acceptable COP and the feasibility of implementing the system into an actual product.

Section 6: Conclusion

A commercial refrigeration system for use in a proposed restaurant was designed in order to allow for the safe storage of perishable items. The design requirements specified the refrigeration load of the system at 1.5 tons (5.28 kW), the ambient temperature of the restaurant as 80˚F, and the required temperature for the food storage area as 20˚F [1]. The design of the refrigeration system was driven by an objective of a coefficient of performance (COP) near 5 based upon a reading of the relevant literature [1] and [3]. The refrigeration system was designed based on the ideal vapor-compression refrigeration cycle using R-134a as the refrigerant in order to comply with environmental standards. Inefficiencies were not accounted for in the initial design procedure but the effect of losses on the system COP and operating parameters was discussed and would need to be incorporated in any further iteration of the design. The literature also provided an initial estimation for the operating temperatures of the condenser and the evaporator. Based upon industry recommendations from experience, the condenser must operate at a temperature at least 10˚ greater than the surroundings, and the evaporator must operate at a temperature at least 10˚ colder than the refrigerated area in order for the heat exchangers to be a reasonable size. A number of different temperature combinations were investigated, with the final design specifying an evaporator operating temperature, T_{evap}, of 10 ˚F (260.9 K) and a condenser operating temperature, T_{cond}, of 90˚F (305.4 K). The resulting ideal COP for the refrigeration system is 4.70, which was judged to be within an acceptable range of the stated design goal. These temperatures set the rest of the system parameters based upon the assumptions of the ideal vapor-compression refrigeration cycle. The mass flow rate of the environmentally-friendly refrigerant R-134a, is 0.036 kg/s, and the compressor electrical power required is 1.12 kW assuming isentropic operation. This COP does also does not account for the power input that would be required to run the fans to drive the refrigerant through the refrigerating cycle. The cost to run the compressor per hour is $0.14. The copper tubing and the four main components consisting of the compressor, condenser, throttling valve, and evaporator, will cost a total of $2127.25. The final system design will enable the safe storage of food at the proposed restaurant and meets all the design requirements. The estimated realistic COP of the system was 3.18, indicating that actual operation is significantly less efficient than ideal operation of the system. This estimate includes the inefficiencies associated with the compressor as well as the power required to run the fans on the condenser and the evaporator. Further work should include accounting for losses in an actual system and specifying the fans and other accessories that would be required for cycle operation.

Section 7: Acknowledgements

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Section 9: Academic Integrity Statement

This report was created by the group members listed below in compliance with Case Western Reserve University's Academic Integrity policy available at:
https://students.case.edu/community/conduct/aiboard/policy.html

By signing below, each group member is giving their affirmation to the above statement.

Name:	Signature:
William Koehrsen	William
Tyler Eston	
Arturo Garcia	Arturo Garcia
Theodore Bastian	

Section 10: Appendices

I: Data Tables

Temp condenser (F)	Temp evaporator (F)	h_1	h_2	h_3	h_4
90	0	387.92	423.82	244.95	244.95
90	10	391.32	422.45	244.95	244.95
100	0	387.92	425.73	253.1	253.1
105	-5	386.21	429.51	257.24	257.24
104	-4	386.55	429.03	256.41	256.41
100	10	391.32	425.73	253.1	253.1
95	10	391.32	424.1	249.01	249.01
95	5	389.63	424.76	249.01	249.01
85	15	393	420.14	240.92	240.92

Table 7: Summary of Enthalpies

Temp condenser (F)	Temp evaporator (F)	COP	m_dot (kg/s)	W_in (kW)	Cost/hr
90	0	3.9825	0.0369	1.3246	$ 0.17
90	10	4.7019	0.0360	1.1219	$ 0.14
100	0	3.5657	0.0391	1.4794	$ 0.19
105	-5	2.9785	0.0409	1.7711	$ 0.22
104	-4	3.0636	0.0405	1.7219	$ 0.22
100	10	4.0169	0.0382	1.3133	$ 0.17
95	10	4.3414	0.0371	1.2151	$ 0.15
95	5	4.0028	0.0375	1.3179	$ 0.17
85	15	5.6035	0.0347	0.9414	$ 0.12

Table 8: Summary of Characteristics

Temp (F)	Temp (K)	Pressure (kPa)	Density (kg/m^3)	Enthalpy (kJ/ kg)	Entropy (kJ / kg K)	Quality
0	255.3722222	145.97	7.423	387.92	1.7394	1
10	260.9277778	183.6	9.2286	391.32	1.735	1
0	255.3722222	145.97	7.423	387.92	1.7394	1
-5	252.5944444	129.58	6.6319	386.21	1.7418	1
-4	253.15	132.73	6.7845	386.55	1.7413	1
10	260.9277778	183.6	9.2286	391.32	1.735	1
10	260.9277778	183.6	9.2286	391.32	1.735	1
5	258.15	163.94	8.287	389.63	1.7371	1
15	263.7055556	205.04	10.253	393	1.733	1

Table 9: Stage 1 Properties

Temp (F)	Temp (K)	Pressure (kPa)	Density (kg/m^3)	Enthalpy (kJ/ kg)	Entropy (kJ / kg K)	Quality
103.442	312.84	820.57	38.218	423.82	1.7394	Superheated
101.084	311.53	820.57	38.519	422.45	1.735	Superheated
111.884	317.53	957.35	44.976	425.73	1.7394	Superheated
120.848	322.51	1031.8	47.915	429.51	1.7418	Superheated
119.498	321.76	1016.6	47.242	429.03	1.7413	Superheated
111.884	317.53	957.35	44.976	425.73	1.735	Superheated
106.484	314.53	886.98	41.642	424.1	1.735	Superheated
107.6	315.15	886.98	41.483	424.76	1.7371	Superheated
94.604	307.93	757.97	35.72	420.14	1.733	Superheated

Table 10: Stage 2 Properties

Temp (F)	Temp (K)	Pressure (kPa)	Density (kg/m^3)	Enthalpy (kJ/ kg)	Entropy (kJ / kg K)	Quality
90	305.3722222	820.57	1178.7	244.95	1.154	0
90	305.3722222	820.57	1178.7	244.95	1.154	0
100	310.9277778	957.35	1156.1	253.1	1.18	0
105	313.7055556	1031.8	1144.4	257.24	1.1931	0
104	313.15	1016.6	1146.7	256.41	1.1905	0
100	310.9277778	957.35	1156.1	253.1	1.18	0
95	308.15	886.98	1167.5	249.01	1.167	0
95	308.15	886.98	1167.5	249.01	1.167	0
85	302.5944444	757.97	1189.6	240.92	1.1409	0

Table 11: Stage 3 Properties

Temp (F)	Temp (K)	Pressure (kPa)	Density (kg/m^3)	Enthalpy (kJ/ kg)	Entropy (kJ / kg K)	Quality
0.00	255.3722	145.97	22.672	244.95	1.1795	0.3237
10.00	260.9278	183.6	30.8	244.95	1.174	0.29475
0.00	255.3722	145.97	20.296	253.1	1.2114	0.36225
-5.00	252.5944	129.58	16.651	257.24	1.2312	0.39534
-4.00	253.15	132.73	17.315	256.41	1.2272	0.38876
10	260.9278	183.6	27.253	253.1	1.2052	0.33402
10.00	260.93	183.6	28.925	249.01	1.1896	0.31432
5.00	258.15	163.94	24.893	249.01	1.1924	0.32876
15.01	263.71	205.04	38.555	240.92	1.1563	0.26021

Table 12: Stage 4 Properties

II: Sample Calculations for Final Design

Included are the calculations for the final design. Some of the numbers may be in slight disagreement with the data tables provided. Rounding errors in the calculations presented below are to blame for any discrepancy. For the final design, numbers calculated in Excel were assumed to be correct and were the figures used in the final design specifications. Enthalpy values for R-134a were obtained from the NIST fluid properties database from the National Institute of Standards and Technology [5].

Calculations

Enthalpy at stage 1:

$$T_1 = T_{evaporator} = 260.9\ K\ ;\ \chi_1 = 1.0\ (saturated\ vapor) \rightarrow h_1 = 391.32 \frac{kJ}{kg}$$

Enthalpy at stage 2:

$$s_2 = s_1 = 1.735 \frac{kJ}{kg\ K};\ P_2 = P_3 = 820.6\ kPa\ \rightarrow\ h_2 = 422.45 \frac{kJ}{kg}$$

Enthalpy at stage 3:

$$T_3 = T_{conderser} = 305.4\ K;\ \chi_3 = 0.0\ (saturated\ liquid) \rightarrow h_3 = 244.95 \frac{kJ}{kg}$$

Enthalpy at stage 4:

$$h_4 = h_3 = 244.95 \frac{kJ}{kg}$$

Required heat absorption by condensing R-134a in condenser

$$\dot{Q}_{in} = 18000 \frac{BTU}{hr} = 5.275\ kW$$

Ideal Coefficient of Performance:

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{391.22 - 244.95}{422.45 - 391.32} = 4.702$$

Mass flow rate of R-134a:

$$\dot{m} = \frac{\dot{Q}_{in}}{h_1 - h_4} = \frac{5.275\ kW}{391.22 \frac{kJ}{kg} - 244.95 \frac{kJ}{kg}} = 0.0361 \frac{kg}{s}$$

Work performed by the compressor:

$$\dot{W}_{in} = \frac{\dot{Q}_{in}}{COP} = \dot{m}(h_2 - h_1) = 0.0361 * (422.45 - 391.22) = 1.126\ kW$$

Cost per hour to run compressor:

$$\frac{\$}{hr} = \frac{\$}{kWh} * kW = \frac{\$0.127}{kWh} * 1.126\ kW = \$0.14/hr$$

Estimated realistic coefficient of performance:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_{net}} = \frac{5.275}{\frac{1.12}{0.8} + 0.15 + 0.11} = 3.18$$

SAFETY DATA SHEET
REFRIGERANT R134A

SECTION 1: IDENTIFICATION OF THE SUBSTANCE / PREPARATION AND OF THE COMPANY / UNDERTAKING

1.1. Product Identifier

Product Name: REFRIGERANT R134a

Synonyms: 1,1,1,2 Tetrafluoroethane
HFC-134a
Norflurane

EC Number: 212-337-0

CAS Number: 811-97-2

REACH Registration Number: 01-2119459374-33-0002

If REACH registration numbers do not appear the substance is either exempt from registration, does not meet the minimum volume threshold for registration or the registration has not yet come due.

1.2. Relevant identified uses of the substance or mixture and uses advised against

Use: Refrigerant
Advised Against: No specific uses advised again have been identified, other than restrictions in the F-Gas Regulations.

1.3. Details of the supplier of the safety data sheet

Company name:

National Refrigerants Ltd.
4 Watling Close
Sketchley Meadows Business Park
Hinckley LE10 3EZ
Tel: +44(0)1455 630790
Fax: +44(0) 1455 630791
Email: sds@nationalref.com

1.4. Emergency telephone number

Emergency Tel: +44(0) 1865 407333

SECTION 2: HAZARDS IDENTIFICATION

2.1. Classification of the substance of mixture

Regulation (EC) No. 1272/2008

Warning

H280 Contains gas under pressure; may explode if heated
P410+P403 Protect from sunlight. Store in a well-ventilated place.

Directives 67/458/EEC or This substance is not classified as dangerous according to Directive 67/548/EEC.

1999/45/EC:	
Most important adverse effect:	Rapid evaporation of the liquid may cause frostbite. Vapour is heavier than air and can cause suffocation.

2.2. Label elements

Label elements under CHIP:

Risk phrases	R58: May cause long-term adverse effects in the environment
Safety phrases	None

2.3. Other hazards

Directives 67/548/EEC or 1999/45/EC:	Not a hazardous substance according to EC directives 67/548/EEC or 1999/45/EC.
Special labelling of certain mixtures:	Contains fluorinated greenhouse gases covered by the Kyoto Protocol

SECTION 3: COMPOSITION / INFORMATION ON INGREDIENTS

3.1. Substances

Hazardous Ingredients:	1,1,1,2-tetrafluoroethane	99.9%

3.2 Mixtures

SECTION 4: FIRST AID MEASURES

4.1. Description of first aid measures

Skin contact:	Rapid evaporation of liquid may cause frostbite. Take off all contaminated clothing immediately if not stuck to the skin. Flush area with lukewarm water. Do not use hot water. If frostbite has occurred call a physician.
Eye contact:	Rapid evaporation of liquid in contact with the eye will damage it. Hold eyelids apart and flush eyes with plenty of water for at least 15 minutes. Get medical attention.
Ingestion:	This is not considered a potential route of exposure.
Inhalation:	Remove from exposure, move to fresh air, and lie down. Keep patient warm and at rest. Artificial respiration and/or oxygen may be necessary. Consult a physician.

4. 2. Most important symptoms and effects, both acute and delayed

Skin contact:	Low exposure to liquid will cause redness and pain. High exposure to liquid will cause frostbite, blisters and severe pain.
Eye contact:	Exposure to liquid will cause severe pain and cornea damage.
Ingestion:	Not a route of exposure.
Inhalation:	High vapour concentrations cause severe headache, dizziness and unconsciousness.
Delayed/immediate effects:	May cause cardiac arrhythmia.

4.3. Indication of any immediate medical attention and special treatment needed

Immediate/special treatment:	Burns pack should be available on the premises.

SECTION 5: FIRE-FIGHTING MEASURES

5.1. Extinguishing media

Extinguishing media: This product is not flammable. (ASHRAE 34) All extinguishing agents are suitable. Use measures that are appropriate to local and surrounding environment. Cool cylinders/tanks with water spray.

5.2. Special hazards arising from the substance or mixture

Special hazards arising from the mixture Pressure build-up in cylinders/tanks.
Hazardous thermal decomposition products: carbon oxides, hydrogen fluoride, carbonyl fluoride.

5.3. Advice for fire-fighters

Advice for fire-fighters: In the event of fire wear self-contained breathing apparatus.
Wear neoprene gloves during cleaning work after a fire.

SECTION 6: ACCIDENTAL RELEASE MEASURES

6.1. Personal precautions, protective equipment and emergency procedures

Personal precautions: Evacuate personnel to safe areas.
Ventilate the area.

6.2. Environmental precautions

Environmental precautions: Should not be released into the atmosphere.

6.3. Methods and material for containment and cleaning up

Clean-up procedures: Material evaporates.

6.4. Reference to other sections

Reference to other sections: For handling and protection measures refer to Section 7 of SDS. Refer to Section 8 of SDS.
For disposal methods refer to Section 13.

SECTION 7: HANDLING AND STORAGE

7.1. Precautions for safe handling

Handling requirements: *Advice on handling:*
Avoid breathing vapours or mist.
Avoid liquid contact with skin and clothing.
Provide sufficient air exchange and/or exhaust in work rooms.
Advice on protection against fire and explosion:
No special measures against fire required.

7.2. Conditions for safe storage, including any incompatibilities

Storage conditions: Keep valves tightly closed.
Store in cool, dry well ventilated place.
Temperature not to exceed 45^{o}C.

Suitable packaging: Store in original cylinder only.
Protect from contamination.

7.3. Specific end use(s)

Specific end use(s) No data available.

SECTION 8: EXPOSURE CONTROLS / PERSONAL PROTECTION

8.1. Control parameters

Hazardous ingredients:
1,1,1,2-TETRAFLUOROETHANE (HFC134a)
Workplace exposure limits

State	8 hour TWA	15 min. STEL
UK	1000 ppm (4240 mg/m^3)	-

8.2. Derived No Effect Level (DNEL)

1,1,1,2-Tetrafluoroethane:

Type of Application (Use): Workers
Exposure Routes: Inhalation
Health Effects: Chronic effects, Systemic toxicity.
Value: 2476 mg/m^3

Type of Application (Use): Consumers
Exposure Routes: Inhalation
Health Effects: Chronic effects, Systemic toxicity.
Value: 2476 mg/m^3

8.3 Predicted No Effect Concentration

1,1,1,2-tetrafluoroethane:

Value: 0.1 mg/l
Compartment: Fresh water.

Value: 0.01 mg/l
Compartment: Marine water.

Value: 1 mg/l
Compartment: Water
Remarks: Intermittent use/release.

Value: 0.75 mg/l
Compartment: Fresh water sediment.

Value: 73 mg/l
Compartment: Water
Remarks: Sewage treatment plants.

8.4. Exposure Controls

Engineering measures: Ensure adequate ventilation, especially in confined areas.
Respiratory protection: For rescue and maintenance work in storage tanks use self-contained breathing apparatus. Vapours are heavier than air and can cause suffocation by reducing oxygen available for breathing.
Hand protection: Heat insulating gloves
Eye protection: Safety glasses with side shields. Wear a face shield in addition where the possibility exists for face contact due to splashing, spraying or airborne contact with this material.
Skin protection: Wear clothing that covers legs and arms.
Environmental: Gas escapes to be kept to the minimum by engineering processes and operating methods.

SECTION 9: PHYSICAL AND CHEMICAL PROPERTIES

9.1. Information on basic physical and chemical properties

State: Liquefied gas under pressure.
Colour: Clear colourless liquid and vapour.
Odour: Slight, ether like.
Molecular weight: 102.02 g/mol
Boiling Point/range: -26.2°C

Flash Point:	Non-flammable
Ignition Temperature:	n/a Non flammable
Upper explosive limit/upper flammability limit:	n/a Non flammable
Vapour pressure:	4.909 Bar (4909 hPa) at 21°C
Liquid Density:	1200 kg/m^3 at 25°C
Vapour Density:	5.368 kg/m^3 at 21°C
Water solubility:	1.5 g/l
Vapour Density (Air = 1)	3.5

SECTION 10. STABILITY AND REACTIVITY

10.1. Reactivity

Reactivity: Stable under recommended storage and transport conditions.

10.2. Chemical stability

Chemical stability: Stable under normal conditions.

10.3. Possibility of hazardous reactions

Hazardous reactions: Hazardous reactions will not occur under recommended storage and transport conditions.
May react with aluminium.

10.4. Conditions to avoid

Conditions to avoid: Heat, hot surfaces, flames.

10.5. Incompatible material

Materials to avoid: Alkali metals, alkaline earth metals, powdered metals, powdered metal salts.

10.6. Hazardous decomposition products

Hazardous decomposition products Thermal decomposition yields toxic products which can be corrosive in the presence of moisture.

SECTION 11: TOXICOLOGICAL INFORMATION

11.1 Information on Toxicological effects

Acute Oral Toxicity:
1,1,2-Tetrafluoroethane
Not Applicable.

Acute inhalation toxicity:
1,1,1,2-Tetrafluoroethane
LC_{50}/rat: 567000 ppm
/dog: Cardiac sensitization.

Acute Dermal toxicity:
1,1,1,2-Tetrafluoroethane
Not Applicable

Skin Irritation:
1,1,1,2-Tetrafluoroethane
Rabbit
Classification: Not classified as irritant.
Result: Slight irritation.

Eye Irritation:
1,1,1,2-Tetrafluoroethane
Rabbit
Classification: Not classified as an irritant.
Result: Slight irritation
Not expected to cause eye irritation based on expert review of the properties of the substance.

	Human Classification: Not classified as irritant. Result: No eye irritation.
Sensitization:	1,1,1,2-Tetrafluoroethane Guinea pig Classification: Not a skin sensitized. Result: Did not cause sensitization on laboratory animals. Not expected to cause sensitization based on expert review of the properties of the substance. Did not cause sensitization on laboratory animals. There are on reports of human respiratory sensitization.
Repeated Dose Toxicity:	1,1,1,2-Tetrafluoroethane Inhalation rat No toxicologically significant effects were found.
Mutagenicity Assessment	1,1,1,2-Tetrafluoroethane Animal testing did not show any mutagenic effects, Tests on bacterial or mammalian cell cultures did not show mutagenic effects.
Carcinogenicity Assessment:	1,1,1,2-Tetrafluoroethane Not classified as a human carcinogen.
Toxicity to reproduction Assessment:	1,1,1,2-Tetrafluoroethane No toxicity to reproduction.
Human Experience:	Excessive exposure may affect human health as follows: Inhalation Severe shortness of breath, narcosis, irregular cardiac activity.
Other information:	May cause cardiac arrhythmia. Rapid evaporation of the liquid may cause frostbite. Inhalation of decomposition products in high concentration may cause shortness of breath (lung oedema).

SECTION 12. ECOLOGICAL INFORMATION

Where sections are blank no data is available

12.1. Toxicity

Toxicity to fish:	1,1,1,2-Tetrafluoroethane LC_{50}/96 h/Oncorhynchus mykiss (rainbow trout): 450 mg/l
Toxicity to Aquatic plants:	1,1,1,2-Tetrafluoroethane EC_{50}/72 h/Algae: >118 mg/l Information given is based on data obtained from similar substances.
Acute Toxicity to aquatic Invertebrates:	1,1,1,2-Tetrafluoroethane EC_{50}/48 h/Daphnia magna (water flea): 980 mg/l
Ecotoxic values:	When discharged may contribute to the greenhouse effect.

12.2. Persistence and degradability

Persistance and Degradability:	Biodegradability /28 d Biodegradation: 3% Method: Closed Bottle test Not readily biodegradable.

12.3. Bio accumulative potential

Bio-accumulative potential: No data available.

12.4. Mobility in soil

Mobility: No data available.

12.5. Results of PBT and vPvB assessment

PBT & vPvB identification: This substance is not considered to be persistent, bio accumulating nor toxic (PBT).
This substance is not considered to be very persistent nor very bio accumulating (vPvB).

12.6. Other adverse effects

Other adverse effects:
Global Warming Potential (GWP) (CO_2 = 1) 1370

Ozone Depletion Potential (ODP) (R11 = 1) 0

SECTION 13. DISPOSAL CONSIDERATIONS

13.1. Waste treatment methods

Disposal operations: Do not allow product to be released into the environment.
Recovery Operations: Consult the manufacturer or supplier for information regarding recovery and recycling of the product. If recovery is not possible, incinerate at a licensed installation.
Disposal of packaging: De-gas and return cylinders to suppliers.
N.B. The user's attention is drawn to the possible existence of regional or national regulations regarding disposal.

SECTION 14. TRANSPORT INFORMATION

14.1. ADR

Proper Shipping Name:	Refrigerant R134a or 1,1,1,2-Tetrafluoroethane
UN Number:	3159
Class:	2
Classification Code:	2A
Labelling No.:	2.2
HI Number:	20
Tunnel Code:	(C/E)

14.2. IATA_C

Proper Shipping Name:	Refrigerant R134a or 1,1,1,2-Tetrafluoroethane
UN Number:	3159
Labelling No.:	2.2

14.3. IMDG

Proper Shipping Name:	Refrigerant R134a or 1,1,1,2-Tetrafluoroethane
UN Number:	3159
Class:	2.2
Labelling Number:	2.2

SECTION 15. REGULATORY INFORMATION

15.1. Safety, health and environment regulations/legislation specific for the substance or mixture

Special labelling of certain mixtures: Contains fluorinated greenhouse gases covered by the Kyoto Protocol.

15.2. Chemical Safety Assessment

Chemical safety assessment: A chemical safety assessment has been carried out by the supplier of this mixture.

16. OTHER INFORMATION

Other information: This safety sheet is prepared in accordance with Commission Regulation (EU) No. 453/2010.
* Indicates text in SDS which has changed since the last revision.

Legal disclaimer: *National Refrigerants Ltd. believes that the information and recommendations contained herein (including data and statements) are accurate as of the date hereof. NO WARRANTY OF FITNESS FOR ANY PARTICULAR PURPOSE, WARRANTY OF MERCHANTABILITY, OR ANY OTHER WARRANTY, EXPRESSED OR IMPLIED, IS MADE CONCERNING THE INFORMATION PROVIDED HEREIN. The information provided herein relates only to the specific product designated and may not be valid where such product is used in combination with any other methods of use of the product and of the information referred to herein are beyond the control of National Refrigerants Ltd. National Refrigerants Ltd. expressly disclaims any and all liability as to any results obtained or arising from any use of the product or reliance on such information.*

GENERAL SAFETY & HANDLING DATA

1. GENERAL

Only trained persons should handle compressed gases. Observe all regulations and local requirements regarding the storage of Cylinders.
Do not remove or deface labels provided by the supplier for the identification of the Cylinder contents.
Ascertain the identity of the gas before using it.
Know and understand the properties and hazards associated with each gas before using it.
When doubt exists as to the correct handling procedure for a particular gas contact the supplier.

HANDLING AND USE

Wear stout gloves.
Never lift a Cylinder by the cap or guard unless the supplier states it is designed for that purpose.
Use trolley or other suitable device or technique for transporting heavy Cylinders, even for a short distance.
Where necessary wear suitable eye and face protection. The choice between safety glasses, chemical goggles, or full face shield will depend on the pressure and nature of the gas being used,

Where necessary for toxic gases see that self-contained positive pressure breathing apparatus or full face airline respirator is available in the vicinity of the working area.
Employ suitable pressure regulating device on all Cylinders when gas is being emitted to systems with lower pressure rating than that of the Cylinder.
Ascertain that all electrical systems in the area are suitable for service with each gas.

Never use direct flame or electrical heating devices to raise the pressure of a Cylinder, Cylinders should not be subjected to temperatures above 45^{o}C.
Never re-compress a gas mixture without consulting the supplier. Never attempt to transfer gases from one Cylinder to another.
Do not use Cylinders as rollers or supports, or for any other purpose other than to contain the gas as supplied.
Never permit oil, grease or other readily combustible substances to come into contact with valves of Cylinders containing oxygen or other oxidants.
Keep Cylinder valves clean and free from contaminants particularly oil and water.
Do not subject Cylinders to mechanical shocks which may cause damage to their valves or safety devices.

Never attempt to repair or modify Cylinder valves or safety relief devices. Damaged valves should be reported immediately to the supplier.
Close the Cylinder valve whenever gas is not required even if the Cylinder is still connected to the equipment.

2. STORAGE

Cylinders should be stored in a well-ventilated area. Some gases will require a purpose built area.
Store Cylinders in a location free from fire risk and away from sources of heat and ignition. Designate as a no smoking area.

Gas Cylinders should be segregated in the storage according to the various categories.

The storage area should be kept clear and access should be restricted to authorized persons only, the area should be clearly marked as a storage area and appropriate hazard warning signs displayed (Flammable, Toxic etc.).
The amount of flammable or toxic gases should be kept to a minimum.
Flammable gases should be stored away from other combustible materials.

Cylinders held in storage should be periodically checked for general condition and leakage.

Cylinders in storage should be properly secured to prevent toppling or rolling.
Vertical storage is recommended where the Cylinder is designed for this.
Cylinder valves should be tightly closed and, where appropriate, valves should be capped or plugged.
Protect Cylinders stored in the open against rusting and extremes of weather.
Cylinders should not be stored in conditions likely to encourage corrosion.
Store full and empty Cylinders separately and arrange full Cylinders so that the oldest stock is used first.

FOR FURTHER INFORMATION CONTACT YOUR NEAREST DISTRIBUTION CENTRE