Performance Evaluation of Locally Fabricated Public Water Cooler

M.Sc. Thesis

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Approval of Board of Examiners

We, the undersigned, members of the Board of Examiners of the final open defense by ATRSAW JEJAW have read and evaluated his thesis entitled "Performance Evaluation of Locally Fabricated Public Water Cooler" and examined the candidate. This is, therefore, to certify that the thesis has been accepted in partial fulfillment of the requirements for the award Degree of Master of Science by Thermal Engineering

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Declaration

I hereby declare that this M.Sc. Thesis entitled "Performance Evaluation of Locally Fabricated Public Water Cooler" in partial fulfillment of the requirements for the award degree of Master of Science in Thermal Engineering is an authentic record of my own work carried out from September 2017 to December 2018 under supervision of Dr. Addisu Bekele, Assistant Professor of Thermal Engineering program, Adama Science and Technology University.

The matter embedded in this thesis has not been submitted for the award of a degree or diploma in any other university. All relevant resource information used in this paper has been duly acknowledged.

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This has been submitted for examination with my approval.

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Cop	Coefficient of performance	$\mathbf{T_{out}}$	Temperature of outlet water
$\mathbf{q_e}$	Cooling effect	$\mathbf{T_2}$	Temperature at exit of compressor
$W_{net,in}$	Network input	$\mathbf{T_3}$	Temperature at exit of condenser
$\mathbf{E_{out}}$	Energy out	$\mathbf{T_4}$	Temperature at inlet of evaporator
$\mathbf{Q_r}$	Condenser heat rejecting capacity	$\mathbf{T_1}$	Temperature at exit of evaporator
$\mathbf{Q_{in}}$	Heat in	$\mathbf{S_1}$	Entropy at inlet of compressor
$\mathbf{W_{in}}$	Work in	$\mathbf{S_g}$	Entropy at saturated vapour
$\mathbf{W_{out}}$	Work out	$\mathbf{S_2}$	Entropy at exit of compressor
$\mathbf{Q_c}$	Cooling capacity	$\mathbf{T_{av}}$	Average temperature
$\mathbf{T_i}$	Inlet water temperature	$\mathbf{Q_{c,out}}$	Heat out to reservoir
$\mathbf{Q_{in}}$	Evaporator capacity	$\mathbf{h_1}$	Enthalpy at inlet of compressor
$\dot{\mathbf{w}}_\mathbf{c}$	Power of compressor	$\mathbf{h_3}$	Enthalpy at exit of condenser
$\mathbf{T_e}$	Evaporating temperature	$\mathbf{h_4}$	Enthalpy at exit of expansion valve
$\mathbf{T_c}$	Condensing temperature	$\mathbf{h_f}$	Enthalpy at saturated liquid
$\mathbf{T_l}$	Liquid refrigerant temperature	$\mathbf{h_g}$	Enthalpy at saturated vapour
$\Delta\mathbf{T}$	Over all temperature difference	$\mathbf{h_{fg}}$	Enthalpy of supper heated vapour
$\mathbf{T_{in}}$	Temperature of inlet water	$\mathbf{\Delta h_c}$	Work of compressor
$\mathbf{P_c}$	Condenser coil working pressure	$\dot{\mathbf{m}}$	Mass flow rate of refrigrant
$\mathbf{P_e}$	Evaporator coil working pressure	$\mathbf{V}$	Volume of water

$\mathbf{L_e}$	Length of evaporator	$\dot{\mathbf{v}}$	Volume flow rate
$\mathbf{d_o}$	Outer diameter	$\boldsymbol{\vartheta}$	Specific volume
$\mathbf{d_i}$	Inner diameter	$\mathbf{V_e}$	Specific volume at exit of evaporator
$\mathbf{L_c}$	Length condenser	$\dot{\mathbf{V}}_\mathbf{p}$	Piston displacement flow rate
$\mathbf{A_c}$	Area of condenser	$\mathbf{t}$	Time
$\mathbf{A_e}$	Area of evaporator	$\mathbf{g}$	Acceleration due to gravity
$\mathbf{A_s}$	Surface area	$\boldsymbol{\eta}_\mathbf{v}$	Volumetric efficiency
$\mathbf{h_i}$	Inner heat coefficient	$\boldsymbol{\eta}_\mathbf{isen}$	Isentropic efficiency
$\mathbf{h_o}$	Outer heat coefficient	$\boldsymbol{\rho}$	Density
$\mathbf{r_p}$	Pressure ratio	$\boldsymbol{\rho}_\mathbf{v}$	Density of vapour R134a
$\mathbf{x}$	Dryness factor or quality factor	$\boldsymbol{\rho}_\mathbf{l}$	Density of liquid R134a
$\mathbf{K}$	Thermal conductivity	$\mathbf{C_p}$	Specific heat capacity
$\mathbf{R}$	Thermal resistance	$\mathbf{U}$	Universal heat transfer coefficient
$\boldsymbol{\mu}$	Dynamic viscosity	$\mathbf{v}$	Kinematic viscosity
f	Friction coefficient	F	Correction factor, force
τ_W	Shear stress	G	Mass flux
$\mathbf{P_r}$	Prandalt number	LMTD	Lower mean temperature difference
$\mathbf{Nu}$	Nusselt number	Re	Reynolds number

List of Abbreviations

ASTU	Adama science and technology university	R134a	Tetrafluoroethane
VCRC	Vapour compression refrigeration cycle	R125	Pentafluoroethane
GWP	Global warming potential	R152a	Difluoroethane
ODP	Ozone depletion potential	R290	Propane
CFC	Chlorofluorocarbon	R32	Methylene Fluoride
HCFC	Hydro Chlorofluorocarbon	R1270	Propylene
HFC	Hydrofluorocarbon	R600	Butane
R11	Trichlorofluoromethane	R600a	Isobutane
R12	Dichlorodifluoromethane	R717	Ammonia
R22	Monochlorodifluoromethane	R744	Carbon Dioxide

Abstract

Water is an essential part of life for almost all organisms living on the planet. The recommended dose for human consumption of water is just under two litters per day [25]. The main challenge in rift valley and low land regions of the Ethiopia, the society directly utilize ground hot water for both drinking and shower due to this cause the society is forced to buy cooled bottled mineral water daily. The use of locally fabricated public water cooler seems to be a solution for this problem.

The objective of this thesis is performance evaluation of locally fabricated public water cooler using a vapor compression refrigeration system. The purpose of water cooler is to decrease temperature of ground hot water (at 33.6°C to 44°C) to 7°C to 13°C for quenching the thirst of the people working in hot environment. The four basic components of the water cooler, namely, the condenser, evaporator, compressor and the capillary tube are designed and selected separately. Also an insulator is designed, thermostat and filter-dryer is selected. The cycle is successfully completed nearly at the same point where it begins. The amount water to be cooled($0.083m^3$), ambient air temperature(28℃), water temperature (44℃), time required to cool water and the evaporator heat load(cooling load of water is 12847.87 KJ) are the predetermined parameters. In present study two models are developed. For the first model water cooler takes 30 minutes to cool water at cooling capacity of 7.14 KW and 300 minutes take at cooling capacity of 0.71 KW for the second model. The second model is developed from first model by scale down cooling capacity by ten for comparison of theoretical with experimental results. After design and select the component the model developed and simulated using EXCEL. R134a is selected as a refrigerant because it has less global warming potential, zero ozone depletion potential value, inexpensive and easily available. The raw materials used to fabricate public water cooler are aluminum sheet, compressor, evaporator coil, condenser coil, capillary tube, filter-dryer, refrigerant fluid, water tap, insulator and thermostat. Aluminum sheet is selected for the water tanker because of second-most-common food-grade metal next to stainless steel and a highly thermal conductive material. The storage type of water cooler is selected due to high performance of cooler because of the evaporator coil is wounded around the walls of the storage tank of the cooler. Comparison of experimental and theoretical results have been done and the COP of cooling system from theoretical analysis is 3.89 and correspondingly that of experimental result is 3.17. This shows some variation is observed because of the ideal assumption of the vapour compression refrigeration processes. Also, Comparison of present study with literature have been done.
Keywords: Water cooler, vapor compression refrigeration cycle, performance evaluation, COP, Refrigerant.

CHAPTER ONE

1 Introduction

1.1 Background of the study

Water is an essential part of life for a practically almost all organisms living on the planet today. Human being can go many days without food, but cannot go long without water. So, the recommended dose for human consumption of water is just under two litters per day [25]. But in some rift valley and low land regions of Ethiopia the society directly utilize ground hot water (about 33.6°C to 44°C) for drinking purpose due to this problem the resource is not always available of a good enough quality. To full fill this requirement public water cooler is should be locally manufactured with required capacity. A water cooler is a device which cools water by removing heat from it, either through a vapour – compression or absorption refrigeration cycle; with the key components being the compressor, condenser, evaporator and expansion device. The purpose of water cooler is to decrease temperature of ground hot water (from about 33.6 C to 44°C) to produce cold water at about 7°C to 13°C for quenching the thirst of the people working in hot environment [6].

This thesis focuses on performance evaluation of locally fabricated public water cooler based on vapor compression refrigeration cycle. The vapor-compression refrigeration cycle uses a circulating liquid refrigerant as the medium which absorbs and removes heat from hot water to be cooled and subsequently rejects that heat elsewhere. Circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapor and is compressed to a higher pressure, resulting in a higher temperature as well. The hot, compressed vapor is then in the thermodynamic state known as a superheated vapor and it is at a temperature and pressure at which it can be condensed with cooling air. Vapour is routed through a condenser where it is cooled and condensed into a liquid by flowing through a coil or tubes with cool water flowing across the coil or tubes. This is where the circulating refrigerant rejects heat from the system and the rejected heat is carried away by the air [5].

The condensed liquid refrigerant, in the thermodynamic state known as a saturated liquid, is next routed through an expansion valve where it undergoes an abrupt reduction in pressure. That pressure reduction results in the adiabatic flash evaporation of a part of the liquid refrigerant. The auto-refrigeration effect of the adiabatic flash evaporation lowers the temperature of the liquid and vapor refrigerant mixture to where it is colder than the temperature of the hot water to be refrigerated [22].

The cold mixture is then routed through the coil or tubes in the evaporator. A fan circulates the warm air in the enclosed space across the coil or tubes carrying the cold refrigerant liquid and vapor mixture. Air evaporates the liquid part of the cold refrigerant mixture. At the same time, the circulating air is cooled and thus lowers the temperature of the hot water to the desired temperature. The evaporator is where the circulating refrigerant absorbs and removes heat which is subsequently rejected in the condenser and transferred elsewhere by the water or air used in the condenser [23]. Complete the refrigeration cycle, the refrigerant vapor from the evaporator is again a saturated vapor and is routed back into the compressor.

The first refrigerated water coolers were cumbersome floor standing units that used belt driven ammonia compressors to chill the water. They were so heavy, they had to be moved by several men or with a forklift. As testimony to the quality found throughout the industry, some of these units are still in operation today. As progressed into the 1950s, the ability to offer students a cold, refreshing drink of water throughout the school day became an industry priority. This led to the design and development of wall-mounted water coolers more suitable to school environments. Over the years, Halsey Taylor developed many variations of the wall-mounted unit, including space-saving recessed models that allow for uninterrupted corridors and hallways, making them ideal for schools, hospitals and other buildings with high-traffic [23].

The next major step in the water cooler evolution process occurred in the 1960s. That is when the industry began to recognize there were physically challenged people confined to wheelchairs who had difficulty drinking from fountains and coolers. In response, the industry came out with the more easily accessible wheelchair water cooler. The 1980s also saw an enhanced focus on the design and appearance of water coolers and drinking fountains. As fountains and coolers became more accessible and more practical than ever before, they also became more attractive. Oval shapes, rounded corners and contemporary finishes that accented or blended in with a building's decor became fashionable and remain so today. In addition to looking for quality and performance when specifying or selecting a water cooler, engineers, architects, building owners and facilities managers were also seeking units that looked good. The 1990s brought about the

Americans with Disabilities Act (ADA), which more comprehensively defined the rights and needs of the handicapped for access to many types of facilities. While this resulted in some dimensional and slight design changes to water coolers and drinking fountains in order to make them even more accessible to physically challenged individuals, it also recognized the needs of disable people and those with bending difficulties. ADA declared that public facilities needed to provide both wheelchair accessible and standard height units. This resulted in the birth of the increasingly popular bi-level unit [23].

Water cooler is cooling equipment consisting of a thermally insulated compartment and a refrigeration system which works on vapour compression refrigeration cycle (VCRC). The first VCRC was developed in 1834 by Jacob Perkins using hand operations. In refrigeration process, the working fluid holding down a job as the heat absorber or cooling agent is called as refrigerant. The refrigerant used does not leave the system throughout the system alternately condensing and evaporating [23].

Heat flows in the direction of decreasing temperature, that is, from high-temperature regions to low-temperature ones. This heat-transfer process occurs in nature without requiring any devices. The reverse process, however, cannot occur by itself. The transfer of heat from a low-temperature region to a high-temperature one requires special devices called refrigerators.

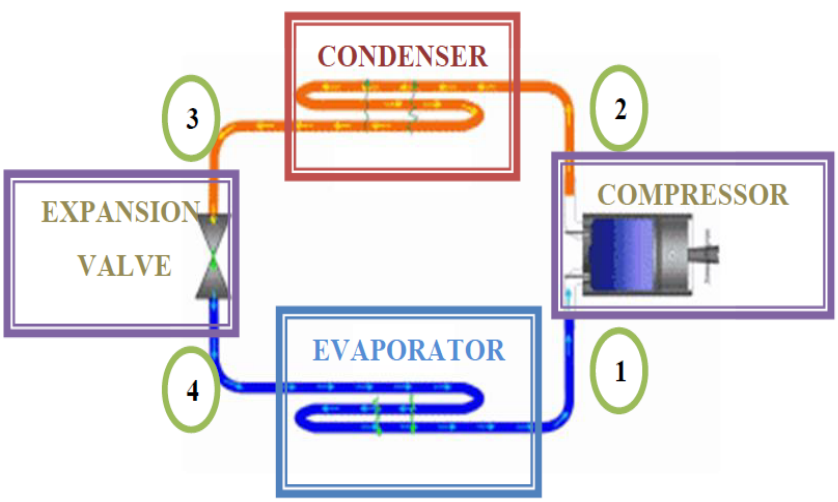


Figure 1.1: Schematic diagram of a Simple VCRC

1.2 Problem statement

In Ethiopia, social service organizations' and industries suffer to supply proper cooled water to their society and labor's in large volume. This problem broadly seen in rift valley and low land regions of the country. For instance, in different higher institutions like ASTU, Semera, Dire Dewa, Asosa, Dilla and others society directly utilize ground hot water directly for both drinking and showering. Due to this, the society is forced to buy mineral waters daily. In order to tackle this problem and unexpected expense using public water cooler is mandatory. However, obtaining the refused cooler with proper capacity in the market is not possible. Hence locally manufacturing of public cooler is found to be necessary. Water cooler is designed to decrease temperature of underground hot water (33.6 - 44°C) and to produce cold water and to maintaining the temperature of water in the range 7°C to 13°C. During this the outside temperature varied from 25°C to 31°C.

1.3 Objectives of the study

General objective of the study

The general objective of this thesis is performance evaluation of locally fabricated public water cooler for drinking purpose in ASTU student dormitory both numerically and experimentally.

Specific objectives of the study

The Specific objective of this thesis are: -

- Design public water cooler for ASTU students
- Evaluating the effect of various parameters on the performance of the water cooler numerically.
- Fabricating prototype of water cooler.
- Evaluating the performance of water cooler experimentally.
- Valid the experiment with simulation

1.4 Significances of the study

This thesis is basically to solve problem of peoples who are living in rift valley and low land regions of the country in Ethiopia, especially in different higher institutions like Adama, Semera, Dire Dewa, Asosa, Dilla and others society who directly utilize underground hot water for both drinking and showering. The materials and individual components of refrigeration system that are used to fabricate public water cooler are easily available in the market so, it is necessary to fabricate water cooler locally. Basically this thesis is used to transfer and adapt water cooler technology in our country and reduce the expenses of the society who buy mineral waters daily.

1.5 Scope and limitation of the work

This thesis focuses on performance evaluation of locally fabricated public water cooler. This include, the design and selection components of VCRC, development prototype and conducting experiment for public water cooler. The experimental results helped to conduct the validation the. The experimental results of COP of water cooler is evaluated and compare with the numerical results. In order to develop confidence about the developed model, the result was compared with the earlier published work. Obtaining electric in rural area is difficult, so design cooler using renewable energy like solar, biogas and wind energy. The cooler could be modified as both cold and hot water dispenser, but this idea are not included in this paper this are the limitation of the study. In the present study, a steady-state analysis of the refrigeration cycle has been performed. The thermos-physical properties of aluminum tanker, evaporator tube, condenser tube and capillary tube are a function of temperature, but for this study all thermos-physical properties are constant.

CHAPTER TWO

2 Literature Review

In this chapter the literatures available related to this study are reviewed within the purview of the present investigation. Review of a number of patented water cooler designs and research studies from the literature are described in the following section.

2.1 Vapour compression refrigeration system

Existing methods which are used to improve the performance of VCR system are reviewed under various titles and these have been taken for further improvement of this study.

Cabello et al. (2004) have analyzed the performance of a vapour compression refrigeration system using three different working fluids (R134a, R407c and R22). The operating variables are the evaporating pressure, condensing pressure and degree of superheating at the compressor inlet. The power consumption increase when compression ratio increases with R22 than using the other working fluids.

Qureshi and Zubair (2012) investigated the performance characteristics due to use of different refrigerant combinations in vapor compression cycles with dedicated mechanical sub-cooling. For basic designs, R134a used in both cycles produced the best results in terms of COP, COP gain and relative compressor sizing. In retrofit cases, considering the high sensitivity of COP to the relative size of heat exchangers in the sub-cooler cycle and the low gain in COP obtained due to installation of a dedicated sub-cooling cycle when R717 is the main cycle refrigerant, it seems that dedicated mechanical sub-cooling may be more suited to cycles using R-134a as the main cycle refrigerant rather than R717. With R134a as the main cycle refrigerant, no major difference was noted, by changing the sub-cooler cycle refrigerant, in the degradation of the performance parameters such as COP and cooling capacity, due to equal fouling of the heat exchangers.

Velkar et al. (2017) carried out an experimental investigation of water cooler system by using eco-friendly refrigerant (R134a). The water cooler system works on simple vapour compression refrigeration cycle and uses R134a as a refrigerant. It is environment friendly and the performance of refrigerant R134a has been found and the results were used to evaluate the behavior of the system under varying load conditions and to measure the COP of the system. R134a is considered to be the most preferred substitute for other refrigerants, since the refrigerant R-134a has less global warming potential and zero ozone depletion potential.

Agrawal and Gupta (2017) have analyzed the performance analysis of air cooled water cooler by using ecofriendly refrigerants as a possible substitute of R134a.The result were analyzed in terms of Coefficient of performance, Refrigeration capacity, compressor work, condenser capacity, specific volume, thermal conductivity and miscibility of lubricants shows that R152a, R134a/R152a-30/70 and R134a/R152a -10/90 is possible alternate for R134a.

Pande and Patil (2017) in this paper comparative study of different passages over dimpled plate evaporator for a small water cooler is conducted. Bonded coil construction have surface area coming in contact with tank after brazing is small compared with the total surface area of the coil and also there is indirect contact between water to be cooled and the evaporator coils. Due to this, type of construction most of the portion of the evaporator coil remains ineffective. Plate coil exchanger has direct surface area connect between two fluids. The whole heat exchanger surface is dipped in one of the fluid so effective area of contact is total surface area. Made from stainless steel material to maintain hygienic condition of drinking the water. Cleaning and maintenance of this heat exchanger is so easier.

Lee and Su (2002) have studied the performance of domestic vapour compression refrigeration system with isobutene (R600a) as refrigerant and compare the results with R12 and R22. They used R600a about 150g and set the refrigeration temperature about 4°C and -10°C to maintain the situation of cold storage and freezing applications. Use 0.7mm internal diameter and 4 to 4.5m length of capillary tube for cold storage applications and 0.6mm internal diameter and 4.5 to 5m length of capillary tube for freezing applications. The COP lies between 1.2 and 4.5 in cold storage applications and between 0.8 and 3.5 in freezing applications. The system with two capillary tubes in parallel performs better in the cold storage, whereas that with a single tube is suitable in the freezing applications.

Rocca and Panno (2011) have analyzed and compared the performance of a vapour compression refrigerating unit operating with R22, and with three new HFC fluids, substituting the former according to Regulation No 2037/2000. Here the plant working efficiency was first tested with R22 and then with three new HFC fluids: R417a, R422a and R422d. It is analyzed that the performance with the new tested fluids did not result as efficient as when using R22.

Mani and Selladurai (2008) have analyzed a vapour compression refrigeration system with the new R290/R600a refrigerant mixture as drop-in replacement was conducted and compared with R12 and R134a. The vapour compression refrigeration system was initially designed to operate with R12.The results showed that the refrigerant R134a showed slightly lower COP than R12.

Dalkilic and Wises (2010) have studied the performance on a vapor compression refrigeration system with refrigerant mixtures based on R134a, R152a, R32, R290, R1270, R600 and R600a was done for various ratios and their results are compared with R12, R22 and R134a as possible alternative replacements. The results showed that all of the alternative refrigerants investigated in the analysis have a slightly lower COP than R12, R22, and R134a for the condensation temperature of 50°C and evaporating temperatures ranging between −30°C and 10 °C.

Aprea et al. (2011) studied the performance of a VCR experimental plant both as water chiller and as a heat pump using R-22 and its substitute R-417A. The use of R-417A does not require lubricant change and equipment redesign. The results showed that the COP and energetic efficiency of the plant is higher for R-22 than R-417A.

Aprea and Renno (2010) experimentally investigated the energetic and exergetic performance of a VCR plant for cold storage application using both R-22 and its substitute R-417A. The results showed that COP was 15% greater than R-417A whereas exergy destruction for R-417 was greater than R-22.

Kumar et al. (1989) explained a method of carrying out exergy analysis on a vapour compression refrigeration system using R-11 and R-12 as refrigerants. The exergy-enthalpy diagrams to facilitate the analysis. The procedure used to calculate various losses in different components of the system.

McGovern and Harte (2011) presented an exergy method for compressor analysis. This is used to find and quantify defects in the use of compressor shaft power and will lead to the improvement in design of the compressor. The exergy destruction and its location are identified. They analyzed the refrigerant compressor using R-12 as refrigerant. The graphs of the instantaneous rates of exergy destruction. It is particularly suitable for applications in computer simulation of compressors and provides a sound basis for design optimization.

Chow et al. (1999) designed a VCRC using R134a as the refrigerant with 0.271 g/s flow rate. An evaporator heat load of 32 W has been obtained. The evaporation temperature was 12^{o}C and the ambient temperature was 45^{o}C. In the study, a centrifugal compressor has been used with a pressure ratio of 3.80 and a power requirement of 9.57 W, thus, a COP value of 3.34 has been reached.

Heydari (2002) in this study all components of the system were designed one by one and the refrigerator was called a miniature CPU cooling system. R134a was used as the refrigerant. The evaporation temperature was 20^{o}C and condensation temperature was 60^{o}C. The condenser was designed as a compact air-cooled heat exchanger. The junction temperature was assumed to be 86^{o}C. A positive displacement, piston-type compressor was used and the stimulation of the piston was obtained by an electrical motor instead of a crank mechanism. An isentropic compressor has been used, and the pressure lost in the valves and the manifolds have been neglected. In addition, the need for a piston-free linear vapor compression compressor has been demonstrated, and a COP value of 3.0 has been reached.

Seyfettin yildiz (2010) studied design and simulation of a vapor compression refrigeration cycle for a micro refrigerator using R134a. A heat load of 45W has been removed by the evaporator. The evaporation temperature was 0^{o}C and the condensation temperature was 50^{o}C. A Piston type compressor was used and a COP value of 3.89 was reached. For a range of evaporator heat loads, the geometrical details of the condenser and the compressor were given. For a 45W load, 0.35 g/s refrigerant mass flow rate was reported to be needed.

Phelan et al. (2004) studied a VCRC using R134a. A heat load of 100-300W has been removed by the evaporator. The evaporation temperature was 5^{o}C and the condensation temperature was 55^{o}C. A scroll type compressor was used and a COP value of 3.0 was reached. To show the effect of the refrigerant selection on the COP value of the system, NH_3 and R22 were examined too. Moreover, the efficiencies of reciprocating, screw, rotary, scroll and centrifugal type compressors have been contrasted. For a range of evaporator heat loads, the geometrical details of the condenser and the compressor were given. For a 100 W load, 0.824 g/s refrigerant mass flow rate was reported to be needed and for a 300W this value has changed to 2.47 g/s.

- From the literature survey it is observed that the following factors affected the performance of vapour compression refrigeration system.

i. Properties of working fluid.
ii. Mixture proportions of different refrigerants.

iii. Suction pressure
iv. Discharge pressure.
v. Pressure ratio.
vi. Amount of charge filled.
vii. Dimensions of capillary tubes.

2.2 Water cooler

The water coolers are two types i.e. the storage type and the instantaneous type. In the storage type water coolers, the evaporator coil is wounded around the walls of the storage tank of the cooler, generally on outside surface of the walls. The tank may be of galvanized steel or stainless steel sheets. The water level in the tank is maintained by a float valve. In this type of water cooler, the machine will have to run for long time to bring down the temperature of the mass of water in the storage tank. Once the temperature touches the set point of the thermostat, the machine cycle is stopped. When the water is drawn from the cooler and an equal amount of fresh water is allowed in the tank, the temperature will rise up slowly and the machine starts again. As such there is always a reservoir of cold water all the time [25].

In instantaneous type water coolers, the evaporator consists of two separate cylindrically wound coils made of copper or stainless tube. The evaporating refrigerant is in one of the coils and the water to be cooled is in the other coil. The water is cooled by the refrigerant in evaporator by conduction. These water coolers are further classified as bottle type, pressure type, and self-contained remote type, these are discussed, in detail, as follows:

(a). Bottle type. As the name suggests, this type of instantaneous water cooler employs a bottle or reservoir for storing water to be cooled. No city main inlet connection is required as it is normally used to cool water supplied in glass bottles, which are placed on top of the unit.

(b). Pressure type. In this type of instantaneous water cooler, water is supplied under pressure. The city main water enters the cooler through the inlet connection at the rear of the cooler. It then passes through a pre-cooler. The pre-cooler is cooled by the waste water of the cooler. As the waste water temperature is low, it is made use of cooling the supply water by passing through a pipe coil wrapped around the drainage line. This arrangement helps in reducing the cooling load for the cooler. The amount of cooling depends upon the quantity of waste water and the length of the pipe coil comprising of pre-cooler. The pre-cooled water then enters the storage chamber and loses its heat [36].

(c). Self-contained remote type cooler. This type of cooler employs a mechanical refrigeration system. The water cooled from the remote cooler is supplied to desired drinking place, away from the system. This type of arrangement does not require extra space near the place of work and is quite useful [36].

- In this study storage type water cooler is selected due to high performance of cooler because of the evaporator coil is wounded around the walls of the storage tank of the cooler.

2.3 Basic components of water cooler

2.3.1 Compressor

The compressor is considered the heart of the refrigeration system. It is a pump, like the heart in the circulatory system of the human body; however, the compressor only pumps vapour. It is used to pump heat uphill from the cold side to the hot side of the system. The heat absorbed by the refrigerant in the evaporator must be removed before the refrigerant can again absorb latent heat. The only way the vaporized refrigerant can be made to give up the latent heat of vaporization that it absorbed in the evaporator is by cooling and condensing it. Because of the relatively high temperature of the available cooling medium, the only way to make the vapour condense is to compress it. A compressor is the most important and often the costliest component (typically 30 to 40 percent of total cost) of any vapour compression refrigeration system (VCRS). The function of a compressor in a VCRS is to continuously draw the refrigerant vapour from the evaporator, so that a low pressure and low temperature can be maintained in the evaporator at which the refrigerant can boil extracting heat from the refrigerated space. The compressor then has to raise the pressure of the refrigerant to a level at which it can condense by rejecting heat to the cooling medium in the condenser [32].

Compressors used in refrigeration systems can be classified in several ways:

Based on the working principle compressors classify as: Positive displacement type and Roto-dynamic type

In positive displacement type compressors, compression is achieved by trapping a refrigerant vapour into an enclosed space and then reducing its volume. Since a fixed amount of refrigerant is trapped each time, its pressure rises as its volume is reduced.

Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

i. Reciprocating type
ii. Rotary type with sliding vanes (rolling piston type or multiple vane type)
iii. Rotary screw type (single screw or twin-screw type)
iv. Orbital compressors, and
v. Acoustic compressors

In roto-dynamic compressors, the pressure rise of refrigerant is achieved by imparting kinetic energy to a steadily flowing stream of refrigerant by a rotating mechanical element and then converting into pressure as the refrigerant flows through a diverging passage.

The selection of a compressor should be made according to the following system cooling characteristics and where it will be installed [32].

- ✓ Minimum Evaporator Temperature
- ✓ Cooling Capacity or Thermal Load
- ✓ Refrigerant Type
- ✓ Ambient Temperature
- ✓ Electrical Power Available
- ✓ Compressor and Drive Cooling

- ➢ In this study hermetic sealed compressors were selected because these types of compressor eliminate the use of crankshaft seal which is necessary in ordinary compressors in order to prevent leakage of refrigerant. The hermetic sealed compressor is widely used for small capacity refrigerating systems such as in domestic refrigerators, home freezers and window air conditioners. These compressors have the advantage of small volume, low levels of noise and energy consumption absence of leakage to the ambient especially at low back pressure and also less effect on global warming. The compressor capacity is based on the saturated suction and discharge temperatures, condensing unit capacities, on the quantity and temperature of the condensing [45].

Figure 2.1: Hermetic sealed type of compressor[45]

2.3.2 Condenser

The purpose of the condenser is to remove the amount of heat that is equal to the sum of the heat absorbed in the evaporator and the heat produced by compression. In condensers the refrigerant vapour condenses by rejecting heat to an external fluid which acts as a heat sink. The condenser receives the hot gas after it leaves the compressor through the discharge line and is forced into the top of the condenser coil by the compressor. The gas entering the condenser is so hot compared to the surrounding air that a heat exchange begins to take place immediately in the air. As the gas moves through the condenser, it begins to give up heat to the surrounding air which causes a drop in gas temperature. The gas keeps cooling off until it reaches the proper condensing temperature and the change of state takes place. The change of state begins slowly at first with small amounts of liquid and gets faster as the combination gas-liquid mixture moves toward the end of the condenser, when the refrigerant gets about 90% of the way through the condenser, it becomes almost pure liquid [32].

There are three types of condensers: air cooled, water cooled and evaporative. These have been described below.

a). Air cooled condensers

Air cooled condensers are used in small units like household refrigerators, deep freezers, water coolers, window air-conditioners, split air-conditioners, small packaged air-conditioners etc. These are used in plants where the cooling load is small and the total quantity of the refrigerant in the refrigeration cycle is small. Air cooled condensers are also called coil condensers as they are usually made of copper or aluminum coil. Air cooled condensers occupy a comparatively larger space than water cooled condensers [32].

Air cooled condensers are of two types:

- Natural convection and
- Forced convection.

In the natural convection type, the air flows over it in natural a way depending upon the temperature of the condenser coil. In the forced air type, a fan operated by a motor blows air over the condenser coil.

b). Water cooled condensers

Water cooled condensers are used for large refrigerating plants, big packaged air-conditioners, central air-conditioning plants, etc. These are used in plants where cooling loads are excessively high and a large quantity of refrigerant flows through the condenser.

There are three types of water cooled condensers:

- Tube-in-tube or double pipe type,
- Shell and coil type and
- Shell and tube type.

In all these condensers the refrigerant flows through one side of the piping while the water flows through the other piping, cooling the refrigerant and condensing it [32].

c). Evaporative condensers

Evaporative condensers are usually used in ice plants. They are a combination of water cooled and air cooled condensers. In these condensers the hot refrigerant flows through the coils. Water is sprayed over these coils. At the same time the fan draws air from the bottom side of the condenser and discharges it from the top side of the condenser. The spray water that comes in contact with the condenser coil gets evaporated in the air and it absorbs the heat from the condenser, cools the refrigerant and condenses it.

Evaporative condensers have the benefits of water cooled as well as air cooled condenser, hence it occupies less space. However, keeping the evaporative condenser clean and free of scale is very difficult and requires lots of maintenance.

- In this study, air cooled wire-and-tube type condenser was selected, in which the removal of heat is done by air. It consists of steel or copper tubing through which the refrigerant flows. The size of tube usually ranges from 6mm to 12mm outside diameter, depending upon the size of condenser. Generally copper tubes are used because of its excellent heat transfer ability. The tubes are usually provided with plate type fins to increase the surface area of heat transfer. The fins are usually made from aluminum because of its light weight. Flow of the liquefied refrigerant will be assisted by gravity, so the inlet will be at the top of the condenser and the outlet at the bottom. Rising pipes should be avoided in the design, and care is needed in installation to get the pipes level. The fins spacing is quite wide to reduce dust clogging [32].

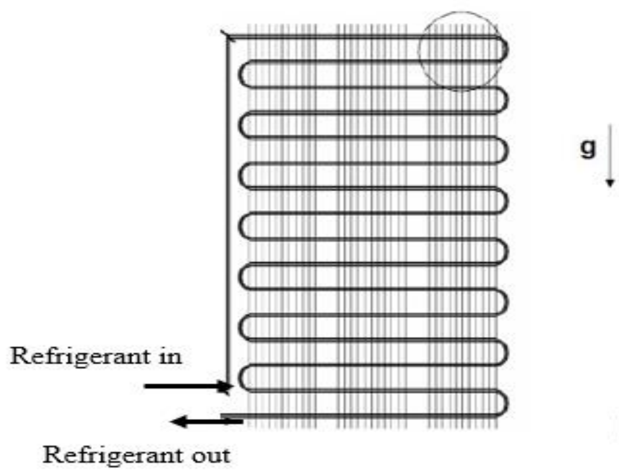


Figure 2.2: Wire-and-tube type condenser

2.3.3 Expansion device of refrigeration system

Various expansion devices are used in refrigeration systems, these includes: hand (manual) expansion valves, capillary tubes, constant pressure or automatic expansion valve (AEV), thermostatic expansion valve (TEV), float type expansion valve, and electronic expansion valve (EEV). Of the above types of expansion devices, Capillary tube which is used in this construction, is the most commonly used in small refrigeration and air conditioning systems [6]. A capillary tube is a long, narrow tube of constant diameter. It is a metering device that controls the refrigerant flow by pressure drop. The principle of operation of the capillary tube is the flow resistance caused by a long, narrow tube, throttling the refrigerant pressure. Pressure falls gradually as the liquid flows through the tube, until it starts to evaporate in the tube. The amount of refrigerant flow through the device is determined by the bore of the tube and the length of the tube. Capillary tube is a simple drawn copper tube with a very small internal diameter. It normally has a very long length and it is coiled to several turns so to occupy less space. It has an inner diameter ranging from 0.5 to 3.0 mm and the length ranging from 400 to 5000 mm. It has advantages of simplicity, inexpensiveness, and the requirement of low starting torque of compressor over other types of expansion devices and coiled type capillary tube used in the domestic refrigerator [32].

To select a thermos-expansion valve on a refrigerating system, the following design conditions must be available:

- Type of refrigerant
- Evaporator capacity, Q_e
- Evaporating temperature/pressure, T_e/ P_e
- Lowest possible condensing temperature/pressure, p_c
- Liquid refrigerant temperature, T_l
- Pressure drop in the liquid line, distributor and evaporator, ΔP

Figure 2.3: Coiled type capillary tube

- By considering the above conditions, for this study Coiled type capillary tube was selected.

2.3.4 Evaporator

The evaporator consists of a coil of copper, aluminum, or aluminum alloy tubing installed in the space to be refrigerated. Aluminum tubing with copper fins is used in ammonia systems. The liquid refrigerant enters the tubing at a reduced pressure and, therefore, with a lower boiling point. As the refrigerant passes through the evaporator, the heat flowing to the coil from the surrounding air causes the rest of the liquid refrigerant to boil and vaporize. Refrigerant temperature in an evaporator is directly related to refrigerant pressure. After the refrigerant has absorbed its latent heat of vaporization (that is, after it is entirely vaporized), the refrigerant continues to absorb heat until it becomes superheated by approximately -26.4°C. The amount of superheat is determined by the amount of liquid refrigerant admitted to the evaporator. This, in turn, is controlled by the spring adjustment of the throttling valve. A temperature ranges of -29.77° to -25.33°C of superheat is considered desirable. It increases the efficiency of the plant and evaporates all of the liquid. This prevents liquid carry-over into the compressor (flooding back). Excessive circulation of the lubricating oil will cause the evaporator temperature to increase. The main cause of slugging is improperly adjusted thermal expansion valve [32].

Defrosting of evaporator coils of a multi-box, direct expansion type refrigeration systems, and ice machines can be accomplished by passing hot vapors from the compression cycle through the coils. A re-evaporator is one way to overcome the possibility of a large slug of liquid refrigerant entering the compressor suction when hot gas defrosting a refrigeration system.

The evaporators used for the refrigeration and the air conditioning applications have different types of construction depending on the application. Based on their construction the various types of evaporators are:

a). Bare Tube Evaporators

The bare tube evaporator comprises of several turns of the tubing, though most commonly flat zigzag and oval trombone are the most common shapes. The bare tube evaporators are usually used for liquid chilling. In the blast cooling and the freezing operations the atmospheric air flows over the bare tube evaporator and the chilled air leaving it used for the cooling purposes. The bare tube evaporators are used in very few applications, however the bare tube evaporators fitted with the fins, called as finned evaporators are used very commonly [32].

b). Plate Type of Evaporators

In the plate type of evaporators, the coil usually made up of copper or aluminum is embedded in the plate so as so to form a flat looking surface. Externally the plate type of evaporator looks like a single plate, but inside it there are several turns of the metal tubing through which the refrigerant flows. The advantage of the plate type of evaporators is that they are more rigid as the external plate provides lots of safety. The external plate also helps increasing the heat transfer from the metal tubing to the substance to be chilled. Further, the plate type of evaporators is easy to clean and can be manufactured cheaply.

Plate type heat exchangers can be easily formed into various shapes as per the requirement. Thus in the household refrigerators and the deep freezers, where they are used most commonly, they can be converted into the box shape to form the closed enclosure, where various food can be kept in the frozen state. The plates can also be welded together forming the bank of the plate type of evaporators that can be used the larger evaporators of higher capacities. Plate types of evaporators provide excellent shelves in the freezers and similar applications. They can be used as the as the partitions in the freezers, frozen food display cases, ice cream cabinets, soda fountains and others. Due to various advantages and flexibility offered by the plate type of evaporators, they are used extensively [32].

c). Finned Evaporators

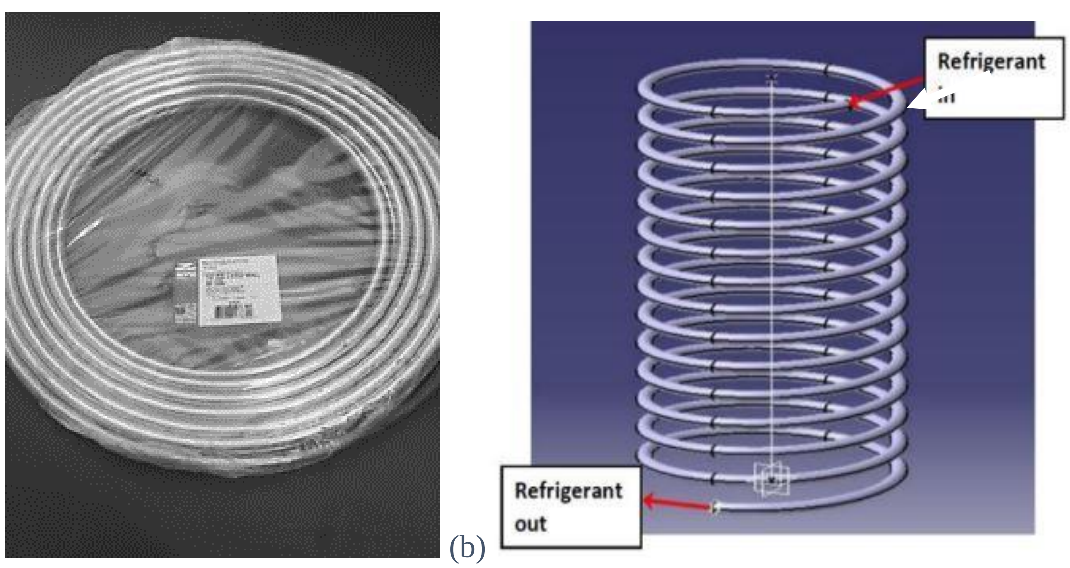
The finned evaporators are the bare tube type of evaporators covered with the fins. When the fluid (air or water) to be chilled flows over the bare tube evaporator lots of cooling effect from the refrigerant goes wasted since there is less surface for the transfer of heat from the fluid to the refrigerant. The fluid tends to move between the open spaces of the tubing and does not come in contact with the surface of the coil, thus the bare tube evaporators are less effective. The fins on the external surface of the bare tube evaporators increases the contact surface of the of the metallic tubing with the fluid and increase the heat transfer rate, thus the finned evaporators are more effective than the bare tube evaporators [32].

d). Shell and Tube types of Evaporators

The shell and tube types of evaporators are used in the large refrigeration and central air conditioning systems. The evaporators in these systems are commonly known as the chillers. The chillers comprise of large number of the tubes that are inserted inside the drum or the shell. Depending on the direction of the flow of the refrigerant in the shell and tube type of chillers, they are classified into two types: dry expansion type and flooded type of chillers. In dry expansion chillers the refrigerant flows along the tube side and the fluid to be chilled flows along the shell side. The flow of the refrigerant to these chillers is controlled by the expansion valve. In case of the flooded type of evaporators the refrigerant flows along the shell side and fluid to be chilled flows along the tube. In these chillers the level of the refrigerant is kept constant by the float valve that acts as the expansion valve also [32].

The refrigerant feed for air and liquid coolers and ice makers can be also classified mainly into the
following three types:
Dry-expansion or direct-expansion (DX). In evaporators with dry- or direct-expansion refrigerant

feed, liquid refrigerant is fed through an expansion valve and a distributer, flows inside the tubes in a finned coil, the evaporator, and is completely vaporized and superheated to a certain degree before reaching the exit of the evaporator.
Flooded refrigerant feed. In evaporators with flooded refrigerant feeds, liquid refrigerant is fed through a throttling device and vaporizes outside the tubes within a shell. The refrigerant-side surface area is always wetted by the liquid refrigerant, which results in a higher surface heat-transfer coefficient.
Liquid overfeed. In liquid overfeed evaporators, liquid refrigerant is fed by a mechanical or gas pump and is then overfed to each evaporator. The inner surface in an overfeed evaporator is also wetted by liquid refrigerant.



(a) (b)

Figure 2.4: Bare tube evaporator (a) Actual (b) 3D modeled

➢ For this study bare tube evaporator was chosen due to its construction because of simple to wounded around the wall of storage of water cooler.

2.3.5 Other complementary components

In addition to the main components of a refrigeration system, a number complementary components are required. These are filter-dryers. cold thermostat and insulators.

a). The filter-dryer

Filter-dryers are placed at the outlet of the condenser in household refrigerators and contain mesh screens to trap contaminants and chemicals to absorb moisture. This provides protection to the capillary tube which can become clogged and block the flow of liquid refrigerant to the evaporator. A completely blocked capillary tube will stop all refrigerant from reaching the evaporator and no cooling takes place. A clogged capillary tube is difficult to diagnose since it seems the unit is low on refrigerant (the evaporator doesn't receive refrigerant, but not because it has leaked out). This is one situation where piercing-valves and manifold gauges are necessary to be certain.

Filter-dryer comes in two different types: throw away and refillable type. The former is thrown away if it is not functioning appropriately and it is replaced with a new one. In case of refillable type, the desiccant granules are replaced by a fresh charge after removing the flange, thereafter the same is filled in the line. The desiccant for common use is aluminum sulfate, silica gel, zeolite etc[33].

➢ Below is the throw away type of filter-dryer used in the water cooler.

Figure 2.5: Throw away filter-dryer [33]

b). Cold thermostat

The cold thermostat is a switching device used to maintain the cold water temperature in the cooling can. The thermostat operates by using a temperature sensitive gas sealed inside the thermos bulb which is attached to bellows assembly inside the thermostat housing.

As the water temperature increases, the gas expands which forces the bellows to expand causing points inside the housing to start the compressor. Once the water temperature begins to decrease, the sensitive gas will contract until the bellows shrink and separates the two contacts, cutting off the power to the compressor, thus shutting off the refrigeration systems.

If for any reason the thermos-bulb would become broken, allowing the sensitive gas to escape, the thermostat would fail in the open position. This would result in no refrigeration and a new thermostat would have to be installed.

The cold thermostat has an external adjustment which may increase or decrease the temperature of the water approximately 2°C. Never turn thermostat to the Off position (completely-counter clockwise) as this will turn of the refrigeration system altogether [35].

Figure 2.6: Cold thermostat[35]

c). Insulation

Insulation was a necessary component of the system in order to reduce heat gain from the environment. Ideally, the subsystem consists of material with low thermal conductivity and inexpensive to reduce production costs. Low conductivity means that less heat will be transmitted through the material, decreasing the amount of heat coming from the environment into the water. Foam plastic is a material that is abundant and is already used as an insulator. Foam has a low thermal conductivity and is inexpensive. Two types of foam were tested and decided between, Styrofoam and spray foam. Styrofoam has a lower insulating value of 1.8K/W (K=0.555 ${}^{w}/_{m\ *\ °C}$) than spray foam with 2 K/W(K=0.5 ${}^{w}/_{m\ *\ °C}$). In addition, Styrofoam was more difficult to work with and required precise cutting in order to fit well onto the chamber. Spray foam was easier to form around the chamber making the system better insulated. Once a mold had been made, spray foam was distributed around the chamber and left to expand. If this insulation thermal resistance were to increase, significantly lower temperatures could be reached. This system involves different types of Styrofoam that can be used to insulate the system.

- Spray foam was chosen for this study, because of its relatively easy manufacturability and high insulating value [34].

2.3.6 Refrigerant fluid

Bolaji et al. (2011) investigated experimentally the performances of three ozone friendly Hydrofluorocarbon (HFC) refrigerants R12, R152a and R134a. R152a refrigerant found as a drop in replacement for R134a in vapour compression system.

Bolaji (2011) discussed the process of selecting environmental-friendly refrigerants that have zero ozone depletion potential and low global warming potential. R23 and R32 from methane derivatives and R152a, R143a, R134a and R125 from ethane derivatives are the emerging refrigerants that are nontoxic, have low flammability and environmental-friendly. These refrigerants need theoretical and experimental analysis to investigate their performance in the system.

Dalkilic et al. (2010) studied the performance analysis of alternative new refrigerant mixtures as substitute for R12, R134a and R 22. Refrigerant blend of R290/R 600a (40/60 by wt. %) and R 290/R1270 (20/80 by wt. %) are found to be the most suitable alternative among refrigerants tested for R12 and R22.

Wongwises et al. (2005) found that 6/4 mixture of R290 and R600 is the most appropriate refrigerant to replace HFC134a in a domestic refrigerator**.**

Mani et al. (2008) found that R290/R600a (68/32 by wt. %) can be considered as a drop in replacement for R12 and R134a.

Padilla et al. (2010) found that R413A (mixture of 88% R134a, 9%R218, 3%R600a) can replace R12 and R134a in domestic refrigerator.

Balaji et al. (2010) investigated the exergetic performance of R12 and its substitute (R134a and R 152a) in the domestic refrigerator. R152a performed better than R134a in terms of COP, exergetic efficiency and efficiency defect as R12 substitute in domestic refrigeration system

Agrawal et al. (2010) worked on eco-friendly refrigerant as a substitute for CFC (Chloroflurocarbon). The binary mixture in the ration of 64% and 36% of R290 and R600a found to be a retrofit or drop in substitute of R12 for use in the vapour compression refrigeration trainer.

EI-Awad (2011) performed the validation of model against experimental data that compared the performance of liquefied petroleum gas (LPG) to that of refrigerant R12 for domestic refrigeration.

Selection of refrigerant for a particular application is based on the following requirements:

i. Availability and performance of refrigerant
ii. Thermodynamic and thermos-physical requirements

Thermodynamic and thermo-physical properties: Generally, thermodynamic and transport properties of refrigerants are the key factors in refrigerant's selection as they determine the performance of the system. The desirable thermodynamic properties are a normal boiling point slightly less than target temperature and, thereby, an evaporating pressure higher than atmospheric pressure. The other favorable characteristics are, low liquid viscosity, high heat of vaporization, modest liquid density and slightly high gas density. It is worthwhile to mention that high heat of vaporization and gas density lead to higher capacity with a specific compressor in a refrigeration system. High liquid thermal conductivity intensifies heat transfer and results in smaller required heat exchangers. Low viscosity also causes low pressure drop in the heat exchangers. Smaller pressure ratio leads lower compression work and improve COP of the system [37].

Generally, refrigerant fluid has the following thermos-physical requirements

- Suction pressure should be above atmospheric pressure.
- Discharge pressure should be as small as possible.
- Pressure ratio should be as small as possible for high volumetric efficiency.
- Thermal conductivity should be higher for higher heat transfer coefficient.
- Viscosity should be small in both liquid and vapor phases for smaller frictional drop.

- The freezing point of the refrigerant should be lower than the lowest operating temperature of the cycle.

iii. Environmental and safety properties

Environmental effects are the main problems of common refrigerants so that non environmental friendly impacts of CFCs and later on HCFCs brought about them to be phased out despite of being stable, non-flammable and non-toxic (comparing to Sulfur Dioxide and other refrigerants used before the introduction of CFCs). Ozone depletion potential (ODP) and global warming potential (GWP) are the significant factors demonstrate the direct impact of refrigerants in case of any leakage or releasing to the surroundings. However, using low GWP refrigerants are not the only efficient way to reduce greenhouse gas emissions. In fact, it is probable to choose a low GWP refrigerant but still raise total greenhouse gas emissions. When the low GWP refrigerant causes more energy use and fuel consumption actually there are larger indirect emissions. Therefore, in developing the low GWP refrigerants always energy efficiency of the system must be studied and its indirect climate impacts should be considered besides its direct emissions [38]. Life cycle climate performance (LCCP) helps to consider overall potential of greenhouse gas emission of the system including materials, transportation, and operation, production, recycling, servicing and end-of-life. Furthermore, toxicity and flammability are the determining factors to select suitable refrigerant for any application. Low toxicity and flammability are the most desirable aspects in safety and health studies [39].

Table 2.1: Ozone depletion factors, according to AAF Ltd. (1990)

Refrigerant	*ODF*	*Proportion of Chlorine by weight*	*Atmospheric half Life in years*
CFCs			
R11	1.0	77.4	65
R12	1.0	58.6	146
R113	0.8	56.7	90
R22	0.05	41.0	20
R142b	0.06	35.2	13
R152a	0	0	1
R134a	0	0	6

Reproduced by courtesy of McQuay chillers, asnyder general brand

iv. Economic properties

The refrigerant used should preferably be inexpensive and easily available.

- Based on an intensive review and search of several refrigerants such as R12, R22, R410a, R152a, R290, R600, R143a, R134a and R125, it was found that R134a is a safe, no toxic refrigerant that is easy to obtain. Thus, it was selected to be the working fluid for this refrigeration system.

Figure 2.7: Refrigerant R134a[37]

General summary about water cooler and its components

Table 2.2: General summary about components of water cooler

Components	**Selected type for this thesis**
Refrigeration cycle	Vapour compression refrigeration system
Water cooler	Storage type
Compressor	Hermetic sealed
Condenser	Air cooled wire-and-tube type
Evaporator	Bare tube type
Expansion device	Coiled type capillary tube
Filter-dryer	Throw away type
Thermostat	Cold thermostat
Insulation	Spray foam
Refrigerant fluid	R134a

CHAPTER THREE

3 Materials and Methodology

3.1 Materials

The raw materials used in this thesis includes aluminum sheet, compressor, evaporator coil, condenser coil, capillary tube, filter-dryer, refrigerant fluid, water tap and thermostat. Aluminum sheet is the selected as material for the tanker due to its higher thermal conductivity in order to absorbed heat easily through the wall of the water tanker from the hot water. However, heat gain from the surrounding affecting the water through the aluminum. So, highly insulating material is require to reduce heat gain from the surrounding. Also, aluminum is the second-most-common food-grade metal next to stainless steel and it is less expensive than stainless steel. It is used for its light weight, low cost, and rust resistance, and is prized in cookware for the speed with which it heats. Before welding aluminum in oxyacetylene, the aluminum sheet must bend cylindrically.

The above materials are purchased from domestic market of adama city.

3.2 Methodology

Theoretical and experimental approaches used as a methodology in this study. The numerical(theoretical) approach use governing equation in each analysis of components. The software's used on this thesis were SOLID work and M.S office 2016. An excel program has been developed to determine cooling load at different initial temperature of water, mass flow rate of refrigerant, cooling capacity, the power of compressor, length of evaporator and condenser at different cooling time. Length of capillary tube segment also determined using excel. Optimization of the design has been done by the help of excel, it became easy to check the effect of change in various parameters, like cooling time and size of tanker. The experimental approach used in this thesis is to test and validation with numerically developed model results at the same working condition. Therefore, in order to achieve the required objective, this study will use both numerical and experimental method.

The overall design and fabrication of public water cooler is described in flow diagram below.

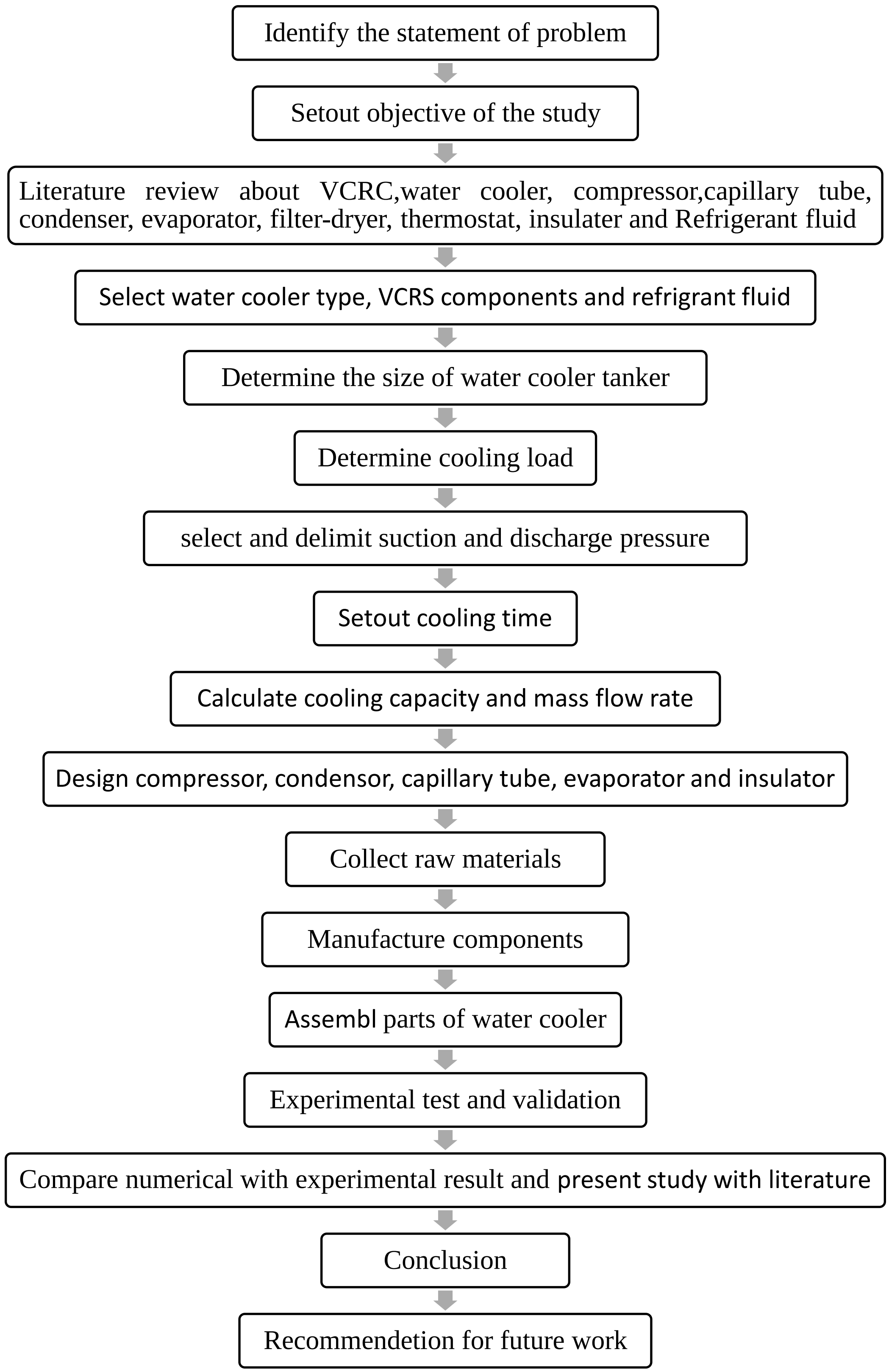


Figure 3.1: Work flow diagram of public water cooler

3.2.1 Design methodology

Assumptions

To obtain an initial design of a vapour compression refrigeration system as requested by the problem statement, the following assumptions are made:

- The four components (compressor, condenser, evaporator, thermal expansion valve) of the thermodynamics cycle are modelled as open systems.
- The four components (devices) operate under steady-state, steady-flow process.
- The changes in kinetic and potential energies across each device are negligible.
- Both the compressor and the expansion device are considered adiabatic (i.e. the heat transfer from these devices is negligible).
- Both the evaporator and the condenser are considered constant pressure devices.

3.2.1.1 Thermodynamics modelling

The basic concept of this design is to use a vapour compression cycle. Utilizing the first laws of thermodynamics [29] along with the above mentioned assumptions, a thermodynamics model for the refrigeration system can be developed as follows:

Compressor $$\dot{W}_c = \dot{m}(h_2 - h_1) \tag{3.1}$$

Condenser: $$\dot{Q}_{out} = \dot{m}(h_3 - h_2) \tag{3.2}$$

Evaporator: $$\dot{Q}_{in} = \dot{m}(h_1 - h_4) \tag{3.3}$$

Expansion Valve: $h_3 = h_4$ throttling process (3.4)

Where $\dot{W}_c$ is the power input to the compressor, $\dot{m}$ is the mass flow of the refrigerant, $\dot{Q}_{in}$ is the heat absorbed by the evaporator, $\dot{Q}_{out}$ is the heat rejected by the condenser and h is the enthalpy.

The coefficient of performance (COP) for the refrigeration system is given by [29]:

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_c} \tag{3.5}$$

3.2.1.2 Design parameters

To ensure that the system will operate over the entire temperature ranges stated in the problem statement, the refrigeration system was designed for the extreme conditions (i.e. $10^{o}C$ for the inside and $28^{o}C$ for the ambient).

3.2.2 Design procedure and calculations

There are several thermodynamics and heat transfer facts that must be satisfied while deciding on the design [30, 31].

i. From basic physics, heat always flows in the direction of decreasing temperature and in order to have heat transfer there must be a temperature difference. Therefore, the temperature of the refrigerant in the evaporator tubes must be less than the air temperature inside the cooler and the temperature of the refrigerant at the exit of the condenser is higher than the outside ambient air.

ii. The compressor operates only on gases (not liquids or two-phase). Therefore, the exit of the evaporator must be at least saturated vapour ($x_1 = 1$).

iii. The exit of the condenser must be at least saturated liquid ($x_3 = 0$).

Based on the above three facts, a thermodynamics model can be constructed and an initial design of a refrigeration system that satisfies the stated requirements can be obtained.

3.2.3 Research design

Literature studies were performed to gain an insight to the area of interest, background, influencing regulations and research. The literature was obtained through statistics databases, academic literature, articles, internet forums, websites etc. After select the component the modeled is developed and simulate the model using EXCEL.

Public water cooler is done based on hypothesis and hypothesis test of thesis. The thesis follows both qualitative and quantitative research design techniques.

3.2.4 Data sources and collection techniques

The data are collected from both primary and secondary sources of data. As primary sources of data are direct measure and secondary data sources are from internet. Reference books and international journal. For primary sources of data collection measure temperature of water (in each time duration in day) in ASTU dormitory and for secondary sources of data collection many journals, reference books and literature reviews from internet.

CHAPTER FOUR

4 Thermal Analysis of Water Cooler

4.1 Cooling load calculation

The cooling load is the amount of heat energy that would need to be removed from a space (cooling) to maintain the temperature in an acceptable range.

In order to calculate cooling load and cooling capacity, the following parameter should be known.

These are:

- Inlet temperature of water (T_{in})
- Out let temperature of water (T_{out})
- Volume of water to be cooled (V)
- Specific heat of water
- Density of water and
- Time taken to cool water

Based on these data inputs, preliminary calculations could be done to get the first estimate of the cooling load, and the size of water cooler was selected based on maximum cooling capacity and amount of cooled water required.

The water cooler is installed based on maximum number of student in one block of ASTU student dormitory. In ASTU student dormitory, four students live in one dorm and one block there are fifty to-sixty dorms. So total number of student in one block is 200 to 240.

The maximum total water consumption in one block per day = number of student in one block * consumption of water in litters per day per student.

Therefore, the cooled water require per day is 400 to 480 litters of water.

Assume One water cooler installed in one block and works six times in a day. The cooled water require per day is 400 to 480 litters of water (use maximum size) so, size of tanker (volume of water) to be cooled is 80 liters. By considering stray water volume of tanker is 83 litters ($0.083m^3$).

4.1.1 Cooling load at different initial temperature of water

Water temperature in ASTU varies with time because of time around noon temperature of water heated due to environment and earth temperature. After noon temperature of water is decrease up to 12AM

Cooling load is calculated as

$$Q_L = \mathrm{m}C_p \ \Delta T = \rho \ \mathrm{V} \ C_p \ \Delta T \tag{4.1}$$

Specific heat of water(C_p) is 4180 J/kg℃ (constant for the temperature in the range between 10℃ to 50℃ from water properties table)

volume of tanker is 83 litters (V =0.083m^3)

Density of water = ρ_i (kg/m^3)

Density of water varies with temperature so, it read from water properties table using inlet temperature of water (T_{in})

Inlet temperature of water data are collected by measure temperature (using digital thermometer) of water in ASTU at different time and taken the average value

$$Q_L = \rho_i \ (\mathrm{kg/m^3}) * \mathrm{V}(m^3) * C_p(J/kg℃) * (T_{in}\text{-}10)\ (℃)$$

Use maximum load for design case:

At water T_w= 44 ℃

- Specific heat of water(C_p) of water = 4180 $^{J}/_{Kg}$ ℃
- Density of water (ρ) = 990.16 $^{Kg}/_{m^3}$

$$Q_{load} = 990.16 \ ^{Kg}/_{m^3} * 0.083m^3 * 4180 \ ^{J}/_{Kg}\ ℃ * 34℃$$

$$= 11679.88775 \text{ KJ}$$

∴ The cooling load of water cooler is 11679.88775 KJ

Table 4.1:Cooling load at different initial temperature of water

Time duration	Inlet Temperature(T_{in})	Density(ρ_i)	cooling load(Q_L)
6- 7AM	33.6	993.90	8137.87
7- 8AM	34	993.76	8274.60
8- 9AM	35.5	993.22	8786.99
9- 10AM	37	992.68	9298.81
10- 11AM	39.5	991.78	10150.6
10- 12AM	41	991.24	10660.92
12-1PM	44.5	989.98	11849.5
1- 2PM	44	990.16	11679.89
2- 3PM	42	990.88	11000.83
3- 4PM	39	991.96	9980.37
4- 5PM	38.6	992.10	9844.14
5- 6PM	38	992.32	9639.71
6- 7PM	37.7	992.43	9537.47
7- 8PM	37.4	992.54	9435.20
8- 9PM	37.1	992.64	9332.91
9- 10PM	36.8	992.75	9230.6
10- 11PM	36.5	992.86	9128.26
11- 12AM	36.2	992.97	9025.91
12--1AM	35.9	993.08	8923.53
1- 2AM	35.6	993.18	8821.13
2- 3AM	35.3	993.29	8718.70
3- 4AM	35	993.4	8616.26
4- 5AM	34.7	993.51	8513.78
4- 6AM	34.4	993.62	8411.29

Taking the allowance factor for additional heat transfer from environment 10% [44].

$$Q_L = 1.1 * Q_{load} = 12847.88 \text{ KJ}$$

4.1.2 Time require to cool water at different temperature

If someone take n amount of water from storage tanker the same amount of hot water is add to tanker instantly. So, the basic question is how many minutes take the water cool to cool to 10℃ due to this additional hot water? So, the analysis as follow.

Temperature of hot water add into the tanker before mix T_1=44℃

Temperature of water in the tanker before mix T_2=10℃

Density of water add to tanker is 990.16kg/m^3 (ρ_1 at T_1=44 ℃)

Density of water in the tanker before mix is 998.8kg/m^3 ($\rho_2 = \rho_3$at T_2=10 °C)

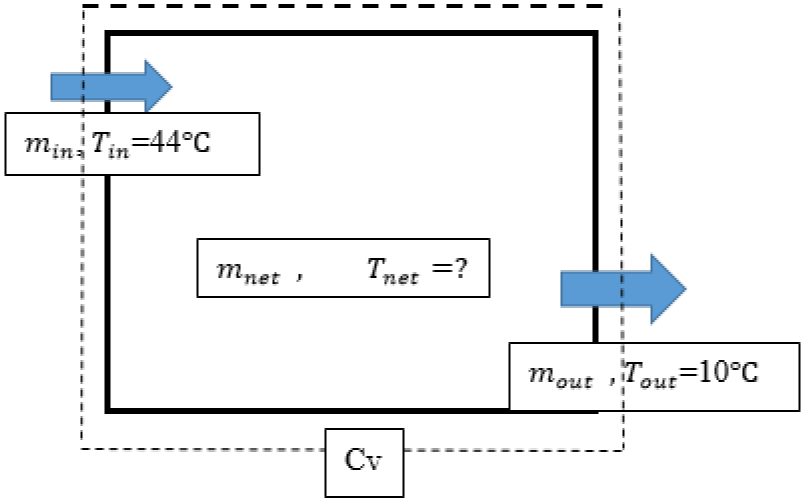


Figure 4.1: Control volume diagram of water cooler

For open system control volume

$$\frac{\partial(\mathrm{m_{net}cp})\mathrm{cv}}{\partial \mathrm{t}} = ((\mathrm{mcp})\mathrm{in} - (\mathrm{mcp})\mathrm{out} + (\mathrm{cpm})_{\mathrm{st}}\ )/\Delta\mathrm{t} \qquad (4.2)$$

$$= \mathrm{cp}(\frac{\mathrm{m_{in}} - \mathrm{m_{out}} + \mathrm{m_{st}}}{\Delta \mathrm{t}}) = 0$$

Then $m_{in} - m_{out} + m_{st} = m_{net}$

Conservation of energy

$$E_{in} - E_{out} + E_{st} = \Delta E_{system} \qquad (4.3)$$

$cpm_{in}T_{in}$-cp$m_{out}T_{out} + m_{st} * cp * T_{st} = m_{net}cpT_{net}$

$m_{net} = m_3 + m_1 - m_2$, $m_{in} = m_1$, $m_{outt} = m_2$, $m_{st} = m_3$

$T_{st} = 10°\mathrm{C}\ temperature\ of\ water\ in\ the\ storage\ before\ hot\ water\ is\ add$

$T_{net} = T_3$, T_{out}=10°C , T_{in}=44°C

T_{net}=$T_3 = Temperature\ of\ water\ in\ the\ tanker\ after\ hot\ water\ is\ add$

where n = amount of water takes out from storage in (litters)

m_1 =mass of hot water adds to the storage in(kg)

m_2 =mass of water takes out from the storage in (kg)

m_3 =mas of water remains storage in (kg)

$m_1 = \rho_1 n$, $m_2 = \rho_2 n$ and $m_3 = \rho_3(83 - n)$

Cp is constant then $m_1 T_1$-$m_2 T_2 + m_3 * T_2 = m_{net} T_3$

$$T_3 = (m_1 T_1 - m_2 T_2 + m_3 T_2)/(m_1 - m_2 + m_3) \quad (4.4)$$

$$Q_L = m_3 C_p\ (T_3 - 10) \quad (4.5)$$

Time require to cool this additional heat load due to the hot water add into tanker

$$Q_c = Q_L / time \quad (4.6)$$

Amount of time require to decrease water temperature to 10℃ due to additional hot water add to tanker is calculated as $time = m_3 C_p\ (T_3\text{-}10)/\ Q_c$ (4.7)

at different temperature time require to cool water results are presented in the next result and discussion part (in table 6.1) using MS. EXCEL at design cooling capacity.

4.2 Design of the compressor

A refrigeration compressor is the heart of a vapor compression refrigeration system. Its function is to raise the pressure of the refrigerant and provide the primary force to circulate the refrigerant. The refrigerant thus produces the refrigeration effect in the evaporator, condenses into liquid form in the condenser, and throttles to a lower pressure through the throttling device. The function of the compressor in the vapor compression refrigeration cycle is to compress the low-pressure superheated vapor leaving the evaporator to the high pressure condenser inlet state. In this study, a hermetic sealed type, isentropic compressor is designed.

In a hermetic compressor, the motor and the compressor are “sealed “or “welded”in the same housing. Hermetic compressors have two advantages: They minimize leakage of refrigerant, and the motor can be cooled by the suction vapor flowing through the motor windings, which results in a small and cheaper compressor-motor assembly. Motor windings in hermetic compressors must be compatible with the refrigerant and lubrication oil, resist the abrasive effect of the suction vapor, and have high dielectric strength. Welded compressors are usually used for small installations from $< 1\ hp\ to\ 24\ hp$ ($< 0.7\ kW\ to\ 18\ kW$) [44].

The following steps are refused in the design of compressor.

- Select discharge and suction pressure of compressor
- Take the inlet state as the exit state of the evaporator.
- Take the entropy of exit state equal to inlet entropy.
- Designate the exit state.
- Read the thermos-physical properties of the refrigerant at the inlet and exit states.
- Calculate the suction pressure.
- Calculate the compressor power requirement.
- Determine the volumetric efficiency.

a) Operating Temperature and Pressure of compressor

Referring to standard vapor compression refrigeration cycle

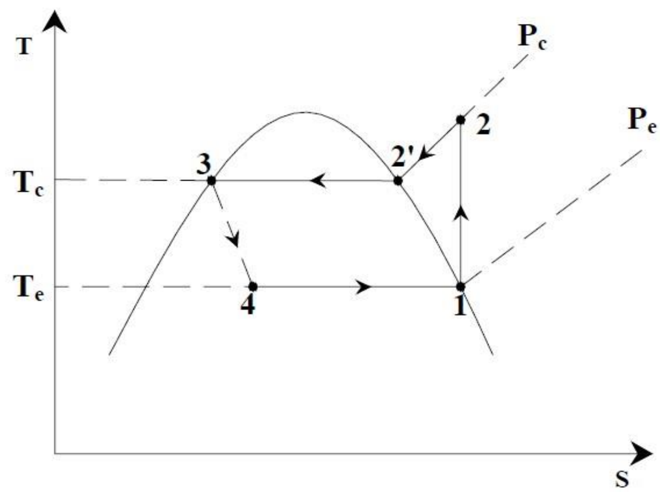


Figure 4.2: Temperatures vs. entropy diagram of an ideal VCRC [29]

where 1-2 compression process ,2-3 condensing process ,3-4 throttling process,4-1 evaporating process.

1-2 isentropic compression: - refrigerant vapour received from evaporator is compressed isentropically in a compressor by external source of energy (work in out), pressure and temperature increase

2-3 condensation: - compressor discharges vapour into the condenser where it condensed completely i.e. turns in to liquid. Heat is rejected from the refrigerant to cooling medium, usually air or water.

3-4 expansion: - from condenser liquid refrigerant passes through the expansion valve where it is throttled resulting in a drop of temperature and pressure. However, enthalpy remains constant (throttling expansion).

4-1 evaporation: - liquid refrigerant at a low temperature passes into evaporator where it extracts heat from the product to be cooled. Due to absorption of extract heat liquid refrigerant turns into vapour, and enters in to the compressor.

A simple analysis of VCRS can be carried out using assumptions

- Steady flow
- Negligible kinetic and potential energy changes across each component
- No heat transfers in connecting pipe lines.
- The steady flow energy equation is applied to each of the four components.

For open system of closed cycle neglecting the shear lose

$$E_{in} - E_{out} = \Delta E_{system} \tag{4.8}$$

$$Q_{in} - W_{in} - (Q_{out} + W_{out}) = \Delta E_{system} \tag{4.9}$$

$$Q_{e,in} + W_{c,in} - Q_{c,out} = 0 \tag{4.10}$$

$$W_{c,in} = Q_{c,out} - Q_{e,in}$$

where $Q_{c,out} = Q_H$ and $Q_{e,in} = Q_L$

Determination of discharge and suction pressure of compressor

a). Suction pressure P_e and evaporator temperature(T_e)

- ➢ From basic physics, heat always flows in the direction of decreasing temperature and in order to have heat transfer there must be a temperature difference. Therefore, the temperature of the refrigerant in the evaporator tubes must be less than the water temperature inside the cooler (10℃).
- ➢ Suction pressure must be greater than atmospheric pressure (1 $atm = 101.3\ Kpa$).

- Selected pressure must be in the range of working of refrigerant fluid.
- Consider additional heat transfer from environment to water cooler due to lack of perfect insulation to compensate this the temperature deference between refrigerant and water inside the cooler must be higher as much as possible.
- If a larger temperature difference between water inside the cooler and evaporator is chosen, the liquid cooler can be smaller, but the T_e value will be lower and compressor power input will be higher. A compromise between energy cost and initial investment should be made through life-cycle cost analysis. So it is not recommended much lower evaporator temperature.
- Most liquid cooler manufacturers have adapted a value of temperature difference between water inside the cooler and evaporator (Tw –T_e) between 6 and 10°F (3.3 and 15.6°C). For energy-saving models, Tw –T_e may be as low as 4 to 7°F (2.2 to 3.9°C) [44].

Depend on the above consideration assume evaporator temperature($T_e = -4$℃) and the Suction pressure P_e(evaporator coil working pressure)= $252.7\ Kpa$.

b). Discharge pressure P_c (condenser coil working pressure) and saturated temperature of the condenser (T_c)

- The temperature of the refrigerant at the exit of the condenser must be higher than the outside ambient air temperature (28℃) because heat always flows in the direction of decreasing temperature and in order to have heat transfer there must be a temperature difference.
- Discharge pressure must be greater than atmospheric pressure ($1\ atm = 101.3\ Kpa$)
- Saturated temperature of the condenser (T_c) must be higher than evaporator temperature
- The total heat of rejection is equal to net refrigeration at the evaporator (cooling capacity) plus the energy input into the refrigerant by the compressor (heat of compression), so the saturated temperature of the condenser (T_c) should be higher than temperature of water in the tanker (44℃).
- If the compressor discharge temperature is very high then it may result in breakdown of the lubricating oil, causing excessive wear and reduced life of the compressor valves (mainly the discharge valve). In hermetic compressors, the high discharge temperature adversely affects the motor insulation (unless the insulation is designed for high

temperatures). When the temperature is high, undesirable chemical reactions may take place inside the compressor, especially in the presence of water. This may ultimately damage the compressor [32].

- At a constant evaporator temperature as the condensing temperature increases, then the enthalpy of refrigerant at the inlet to the evaporator increases. Since the evaporator enthalpy remains constant at a constant evaporator temperature, the refrigeration effect decreases with increase in condensing temperature. The refrigeration capacity (Qe) also reduces with increase in condensing temperature as both the mass flow rate and refrigeration effect decrease. So compressor discharge temperature is not recommended much higher as much as possible.
- A high condensing temperature T_c means a higher condensing pressure P_c a higher discharge temperature. High T_c and P_c result in a higher compressor power input and, more often, unsafe operating conditions. Both are undesirable.
- The condenser temperature difference (CTD) for an air-cooled condenser is defined as the difference in saturated condensing temperature corresponding to the refrigerant pressure at the inlet and the air intake dry-bulb temperature $T_2 - T_{air}$ the total heat rejection Q_r of an air-cooled condenser is directly proportional to its CTD. The Q_r of an air-cooled condenser operated at a CTD of 30°F (16.7°C) is approximately 50 percent greater than if the same condenser is operated at a CTD of 15°F (8.3°C). On the other hand, for a specific value of Q_r an air-cooled condenser selected with a CTD of 15°F (8.3°C) is larger than a condenser elected with a CTD of 30°F (16.7°C) [44].

Depend on the above consideration assume the discharge pressure is P_c (condenser coil working pressure) $=\ 1200\ Kpa$ and the saturated temperature of the condenser) is T_c= 46.3℃.

Therefore, the saturated temperature of the condenser(at $P_c = 1200\ Kpa$) is T_c= 46.3℃ and evaporator (at $P_e = 252.7Kpa$) $is\ T_e = -4$℃ reads as from the saturated table of refrigerant R134a.

Table 4.2: Properties of R134a at evaporator temperature($T_e = -4$℃) and condenser pressure($P_c = 1200$ Kpa)

Temp	Pressure	Volume (m^3/kg)	Enthalpy (kJ/kg)			Entropy (kJ/kg.K)		
℃	kPa	v_g	h_f	h_{fg}	h_g	s_g	s_{fg}	s_g
-4	252.7	0.0799	46.50	201.61	248.11	0.1848	0.7490	0.9338
46.3	1200	0.0167	117.8	156.1	273.9	0.4245	0.4886	0.9131

In this case, the P-H diagram for standard vapor compression refrigeration system (SVCRS) preferable than T-S diagram. Because SVCRS are haled at constant pressure as well as at constant enthalpy in capillary tube.

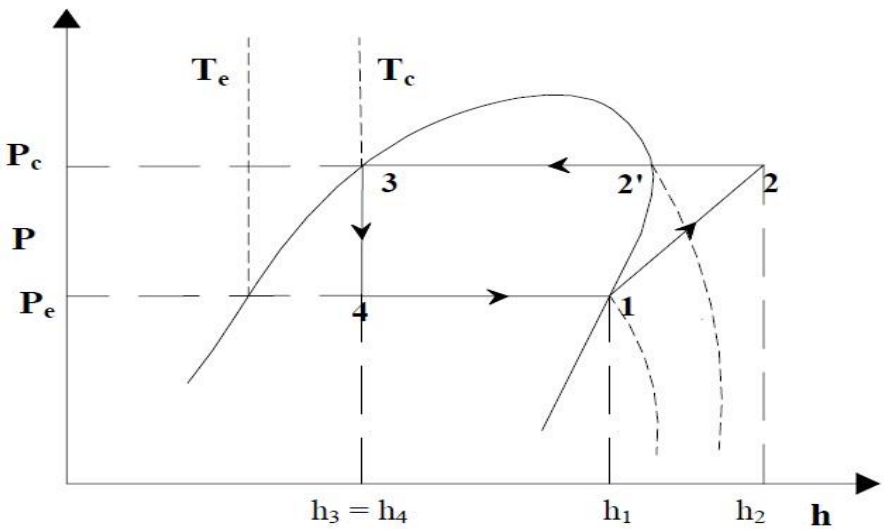


Figure 4.3: Pressures vs. enthalpy diagram of an ideal VCRC [29].

@point 1. (From Saturated vapor table of R134a @$P_e = 252.7\ Kpa$)

$$h_1 = h_g \text{ and } s_1 = s_g$$

@point 2. (From superheated table of R134a @$P_c = 1200\ kpa$)

$$s_1 = s_2$$

$$\therefore s_2 = 0.9338\ {}^{kJ}/_{kg*k}$$

From superheated section of the thermodynamic property table,

Table 4.3: Properties of superheated R134a at condenser pressure($P_c = 1200\ Kpa$)

T	v	h	s
°C	m^3/kg	kJ/kg	kJ/kg·K
P 1.20 MPa (T_{sat} =46.3°C)			
Sat.	0.016715	273.87	0.9130
50	0.017201	278.27	0.9267
T_2	v_2	h_2	0.9338
60	0.018404	289.64	0.9614

Using interpolation

h_2 and T_2 can be calculated by interpolating and get results

$$\frac{289.64 - 278.27}{289.64 - h_2} = \frac{0.9614 - 0.9267}{0.9614 - 0.9338}$$

$$h_2 = 281.542\ {}^{kJ}/_{kg}$$

and

$$\frac{60 - 50}{60 - T_2} = \frac{0.9614 - 0.9267}{0.9614 - 0.9338}$$

$T_2 = 52.5$ °C

@ Point 3(From saturated liquid table of R134a @$P_c = 1200\ \ kpa$)

i.e. $h_3 = h_f = 117.8\ {}^{kJ}/_{kg}$

and $h_4 = h_3$, because the isentropic process from state 3 to 4 due to the throttling through capillary tube (isentropic lowering of pressure and temperature)

$$\therefore h_4 = h_3 = 117.8\ {}^{kJ}/_{kg}$$

Operating pressure ratio of the compressor is given by:

$$r_p = \frac{\mathrm{P_c}}{\mathrm{P_e}} = (1200\ \ kpa)/(252.7\ kpa) = 4.7487 \quad (4.11)$$

From Saturated vapor table of R134a $@P_e = 252.7\ Kpa$

$$V_e = V_g = 0.0799\ {m^3}/{kg}$$

The cooling effect ($\mathrm{q_e}$)and the cooling capacity of the evaporator ($\mathrm{Q_c}$):

Cooling effect refers to the enthalpy difference when the refrigerant passes through the evaporator coil.

$$\text{Cooling effect}(\mathrm{q_e}) = \mathrm{h_1} - \mathrm{h_4} \quad (4.12)$$

$$=(248.11\text{-}117.8)\ {kJ}/{kg}$$

$$=130.31\,{kJ}/{kg}$$

b) **Power of compressor requires, cooling capacity and mass flow rate of refrigerant at different cooling time**

Power of compressor at different Cooling capacity is given by

Power$(kw) = \ mass\ flow\ rate * \Delta h_c$

where Cooling capacity(Q_C)= $(cooling\ load)/(time\ taken) = Q_L/(time(\mathrm{min}))$

Mass flow rate = $(Cooling\ capacity)/q_e$

Power$(hp) = power(kw) * 1000/746$

At different cooling time cooling capacity, mass flow rate, power of compressor, length and diameter of evaporator and condenser results are presented in table 4.4 below using excel.

Table 4.4: Power of compressor, length of evaporator and condenser at different cooling time.

Cooling time(min)	Cooling capacity(kw)	Mass flow rate(kg/s)	Power of compressor		Evaporator		Condenser	
			kw	hp	D(mm)	L(m)	D(mm)	L(m)

15	14.28	0.110	3.662	4.909	19.05	56.45	15.88	17.97
30	7.14	0.055	1.831	2.455	15.88	50.88	9.53	15.32
45	4.76	0.037	1.221	1.636	15.88	48.26	9.53	14.51
60	3.57	0.027	0.916	1.227	15.88	45.34	9.53	14.03
75	2.86	0.022	0.732	0.982	15.88	42.81	9.53	13.56
90	2.38	0.018	0.610	0.818	12.70	38.74	9.53	13.01
105	2.04	0.016	0.523	0.701	12.70	36.24	9.53	12.74
120	1.78	0.014	0.458	0.614	12.70	33.34	9.53	12.36
135	1.59	0.012	0.407	0.545	12.70	31.35	9.53	11.95
150	1.43	0.011	0.366	0.491	9.53	29.98	6.35	11.45
165	1.30	0.010	0.333	0.446	9.53	27.58	6.35	10.22
180	1.19	0.009	0.305	0.409	9.53	25.56	6.35	9.12
210	1.02	0.008	0.262	0.351	9.53	23.97	6.35	8.57
240	0.89	0.007	0.229	0.307	6.35	20.57	6.35	7.34
270	0.79	0.006	0.203	0.273	6.35	16.68	6.35	6.82
300	0.71	0.005	0.183	0.245	6.35	14.87	6.35	6.33
330	0.65	0.005	0.166	0.223	6.35	10.83	6.35	5.74
360	0.59	0.005	0.153	0.205	6.35	6.45	6.35	4.80

As shown in the table 4.4 the cooling time decrease the compressor power is increase. similarly increase cooling time, the corresponding cooling capacity is decrease. Since it takes longer working and cool time, the temperature of compressor is increase. This causes breakdown of the lubricating oil, causing excessive wear and reduced life of the compressor valves.

Considering the optimized data from table 4.4 it is obtained that for cooling 83 litters of water at (T_{in}= 44℃ , T_{out}= 10℃) the time required was in thirty minute (30 min).

The cooler is design to cool this load in in thirty minute (30 min) duration., so the cooling capacity is given by: -

$$\text{Cooling capacity} = cooling\ load/time\ taken$$

$$Q_c = 12847.88\ \text{KJ}/(30 * 60\ sec) = 7.14\ \text{KW}$$

Usually the cooling capacity is expressed in tons so,

$$1\ ton\ =\ 3.52\ KW$$

$$\therefore \text{Cooling capacity} =\ 2.03\ tons$$

For checking and comparing the theoretical with experimental results public water cooler is fabricate locally. It is not possible to fabricate water cooler at cooling capacity 7.14 KW for experimentation. As shown in the optimization table 4.4 at low cooling time the size of compressor is high, length of evaporator and condenser is long. So it requires large amount of refrigerant in addition to cost of compressor and coils. Due to this reason it is not possible to fabricate this types of cooler for experimentation due to financial problem. So use low capacity and fabricate this model after checking performance, for test and validation the model experimentally. Due to the higher cost of compressor and evaporator in this thesis, minimum size of compressor is selected for experimental evaluation of water cooler. So Scale down the cooling time or cooling capacity model by ten (10), then cooling capacity is 0.71 kw.

So, the time require to cool this cooling load (12847.88 KJ) in 300 minutes.

- ❖ Therefore, in present study two models are developed. For the first model water cooler takes 30 minutes to cool water at cooling capacity of 7.14 KW and 300 minutes take at cooling capacity of 0.71 KW for the second model. The second model is developed from first model by scale down cooling capacity by ten for comparison of theoretical with experimental results.
- ➢ Compressor is design for both model (Case I for Q_c=7.14 KW and case II for Q_c=0.71 kw)

Case I: Design compressor at cooling capacity 7.14 KW

Mass flow rate of refrigerant ($\dot{m}$)

$$\text{Cooling capacity } (Q_c) = \dot{m}\,(h_1 - h_4) \tag{4.13}$$

then $$\dot{m} = \text{Cooling capacity}(Q_c)/(h_1 - h_4) = (7.14\ \text{kw})/\ (130.31)\ {}^{\text{kJ}}/_{\text{kg}} = 0.05\ {}^{kg}/_{s} \tag{4.14}$$

Volume flow rate of the compressor

$$\dot{V} = \text{v}_\text{e} * \dot{m} = 0.0799\,{}^{m^3}/_{kg} * \ 0.05\ {}^{kg}/_{s} = 0.00044\ {}^{m^3}/_{s} \tag{4.15}$$

Work and $Power\ of\ compressor$

The power input to the compressor define as the product of the mass flow rate and the difference between evaporator exit and condenser inlet specific enthalpies.

$$\Delta h_c = h_2 - h_1 \qquad (4.16)$$

$$= (281.542 - 248.11)\ ^{kJ}/_{kg}$$

$$\therefore \Delta h_c = 33.432\ ^{kJ}/_{kg}$$

then

$$\text{Power of compressor} = \Delta h_c * \dot{m} \qquad (4.17)$$

$$\dot{W}_c = 33.432\ ^{kJ}/_{kg} * 0.05\ ^{kg}/_{s} = 1.83\ ^{kJ}/_{s}$$

OR $\dot{W}_c$ *=2.45 hp =5/2 hp*

The flash gas percentage after throttling: x

using the dryness factor or the quality has a relation:

$$x = \frac{h_4 - h_f}{h_g - h_f} \qquad (4.18)$$

Using the inlet enthalpy value of the compressor, $h_4 = 117.8\ ^{kJ}/_{kg}$

From thermodynamic property table

h_f and h_g values at P_e= 252.7 kpa is;

$$h_f = 46.50\ ^{kJ}/_{kg}$$

$$h_g = 248.11\ ^{kJ}/_{kg}$$

$$\therefore x = 0.3536\ \ or\ 35.36\ \%$$

Coefficient of performance of the water cooler:

The coefficient of performance is an index of performance of a thermodynamic cycle or a thermal

system. Because the COP can be greater than 1, COP is used instead of thermal efficiency.

$$\mathrm{COP} = (\text{output})/\text{input} = \frac{h_1 - h_4}{h_2 - h_1} \tag{4.19}$$

or COP=Cooling capacity of water cooler /power required to drive the compressor

$$\mathrm{COP} = \frac{7.14}{1.83} = 3.90$$

Volumetric efficiency of a typical reciprocating compressor η_v decrease from 0.92 to 0.65 when p_c/p_e increases from 1 to 6, and the isentropic or compressor efficiency (η_{isen}) decreases from 0.83 to 0.75 when $\frac{p_c}{p_e}$ increases from 4 to 6 [44].

Specification of 2.45 hp or (5/2 hp) hermetic sealed reciprocating compressor

Refrigerant used-R134a

Bore diameter of cylinder (D_{cy}) = (67.4 mm)

Stroke length of cylinder (L_{st}) = (57.2 mm)

Clearance factor (C) = 0.04

Isentropic index (n) = 1.118 (for R-134a)

Speed of the compressor (N) = 1575 rpm

Evaporator temperature (T_e) = -4^{0}C

Condenser temperature (T_c) = 46.30^{0}C

Volumetric Efficiency

It is denoted by η_v[45]

$$\eta_v = 1 + C - C\left(\frac{P_c}{P_e}\right)^{1/n} \tag{4.20}$$

$$\eta_v = 1 + 0.04 - 0.04\left(\frac{1200}{252.7}\right)^{1/1.118}$$

$$\eta_v = 0.8172 = 81.72\%$$

Refrigeration Capacity. In reciprocating compressors, the piston only performs compression work during the upward stroke in a complete rotation of the crankshaft. Piston displacement $\dot{V}_p(m^3/s)$, can be calculated as [45].

$$\dot{V}_p = \frac{\pi {D_{cy}}^2 L_{st} N}{4*60} \tag{4.21}$$

$$\dot{V}_p = \frac{\pi(0.0674)^2 * 0.0572 * 1575}{4 * 60} m^3/s = 0.0054\, m^3/s$$

Mass Flow Rate of the refrigerant can be also calculate as

$$\dot{m} = \frac{\dot{V}_p \eta_v}{v_e} \tag{4.22}$$

$$\dot{m} = \frac{0.00357 * 0.8172}{0.0799} = 0.05 \quad \mathrm{kg/s}$$

Case II: Design of compressor at cooling capacity 0.71 kw

Evaporator and condenser pressure are the same as first model, so cooling effect are the same. The only difference is mass flow rate of refrigerant and power of compressor.

Mass flow rate of refrigerant ($\dot{m}$)

$\dot{m}$ = $\frac{\text{Cooling capacity } (Q_c)}{(h_1 - h_4)}$ = (0.71 1 kw)/((130.31) kJ⁄kg) = 0.005 $kg/_s$

Power of compressor = $\Delta h_c * \dot{m}$= 33.432 $^{kJ}/_{kg}$ * 0.005 $^{kg}/_{s}$ = 0.183 $^{kJ}/_{s}$ or

Power of compressor=0.245 hp

Therefore, Specification of 0.245 hp or (1/4 hp) hermetic sealed reciprocating compressor is

Refrigerant used-R134a

Bore diameter of cylinder (D_{cy}) = (37.3 mm)

Stroke length of cylinder (L_{st}) = (27.2 mm)

Clearance factor (C) = 0.04

Isentropic index (n) = 1.118 (for R-134a)

Speed of the compressor (N) = 1080 rpm

Evaporator temperature (T_e) = -4^{0}C

Condenser temperature (T_c)) = 46.30^{0}C

4.3 Design of condenser and evaporator coils

To design condenser and evaporator coils the following parameters of must be consider

i. It is based on the time taken to extract the required heat from the hot water and refrigerant (R134a) gas capacity to vaporize (latent heat of vaporization.)

ii. It depends on the property of the materials for which the evaporator and condenser coil is made because of its thermal conductivity. If the thermal conductivity is high rapid heat exchange occurs from high temperature area to low temperature area unless longer time is required to vaporize the refrigerant. The best material with high thermal of conductivity is copper.

iii. The effect of surface area to volume ratio has its own effect

$$\text{Surface area}=2\pi rL \quad (4.23)$$

$$\text{Volume}=\pi r^2 L \quad (4.24)$$

Where: r = radius of the coil

L= length of the coil

Surface area to volume ratio becomes; $\frac{2}{r}$

The relation shows that the radius of the coil and surface area to volume ratio have inverse in relation. To optimize this relation for ensuring the effective heat transfer by considering manufacturing difficulties with cost as well as the formation of ice through the evaporator coil will cause blockage this will going to result interruption of the cyclic process.

iv. The performance of the compressor or the working pressure of both condenser and evaporator will have an effect. If the pressure is high enough it will require longer in length if not, it needs smaller coils to exchange the required heat in both a condenser and evaporator.

v. Merely it depends on the overall heat transfer coefficient of the hot and cold fluid.

Heat transfer analysis on condenser and evaporator coil

The three modes of heat transfers in evaporator are given by:

for convection $$Q_{cov}=h_i A(T_\infty - T_e) \tag{4.25}$$

for conduction $$Q_{con}= \frac{KA}{t}(T_{Al} - T_e) \tag{4.26}$$

for radiation $$Q_{rad} = A\varepsilon\sigma({T_{Al}}^4 - {T_e}^4) \tag{4.27}$$

$$Q_e = A_e U(\Delta T) \tag{4.28}$$

The heat transfers in condenser are given by:

for convection $$Q_{cov}=h_i A(T_c - T_\infty) \tag{4.29}$$

for conduction $$Q_{con}= \frac{KA_f}{t}(T_c - T_f) \tag{4.30}$$

for radiation $$Q_{rad} = A\varepsilon\sigma({T_c}^4 - {T_{cover}}^4) \tag{4.31}$$

$$Q_r = A_c U(\Delta T) \tag{4.32}$$

Where, U= overall heat transfer coefficient

4.3.1 Thermal analysis for evaporator coil

An evaporator should transfer enough heat from as small size as possible. It should be light, compact, safe and durable. The pressure loss should be as low as possible.

Design factors are:

- Evaporator temperature.
- Refrigerant properties
- Refrigeration effect.
- Tube selection.

The selected evaporator for the design is natural convection bare tube Evaporators.

Heat reaches the Evaporator by all three methods of heat transfer.

To determine the length of evaporator

$$cooling\ capacity(Q_c^{\cdot}) = U_o * A_e * F * LMTD \tag{4.33}$$

Case I: Length of evaporator at cooling capacity of Q_c = 7.14 KW

Heat absorbed by R134a vapor(Q_c) = 7.14 KW

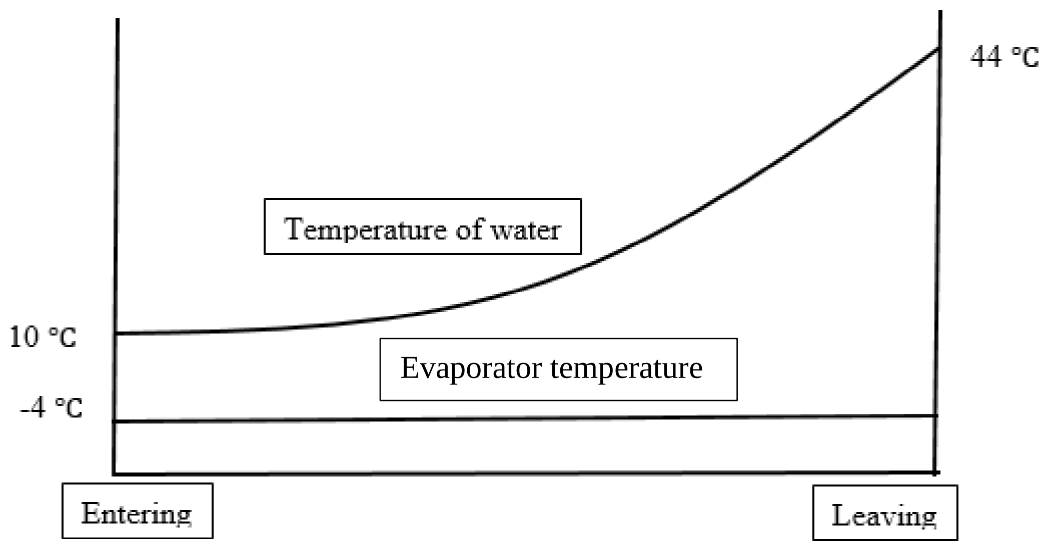


Figure 4.4: Heat-transfer process in evaporator

Inside cooler box initial Temp. of aluminum tanker at top side, T_t= 44^{0}C (because of inlet water temperature T_w= 44^{0}C) and Assume Temp. of aluminum at bottom, T_b= 10^{0}C (because of remain water temperature)

Inlet temperature of the evaporator coil (T_4) = -4°C

Outlet temperature of evaporator coil (T_1) = -4°C

The flow is cross flow use correction factor and Assuming counter flow type heat exchanger for the heat transfer.

Assuming counter flow type heat exchanger,

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \tag{4.34}$$

$$\Delta T_1 = T_t - T_1 = 44℃ - (-4℃) = 48℃ \tag{4.35}$$

$$\Delta T_2 = T_b - T_1 = 10℃ - (-4℃) = 14℃ \qquad (4.36)$$

$$LMTD = \frac{(48-14)℃}{\ln(48℃/14℃)} = 27.594\ ℃ = 300.744\ k$$

$$Qe = [U_o * A_e * F * LMTD]$$

Table 4.5: Standard pipe sizes for evaporator and condenser coil[46]

Tube Size	Outside diameter inch	Outside diameter mm	BWG	Wall thickness inch	Wall thickness mm
1/4 "	0.250	6.350	22	0.028	0.711
			24	0.022	*0.559*
3/8 "	0.375	9.525	18	0.049	1.245
			20	0.035	0.889
			22	0.028	0.711
1/2 "	0.500	12.700	18	0.049	1.245
			20	0.035	0.889
5/8 "	0.625	15.875	16	0.065	1.651
			18	0.049	1.245
			20	0.035	0.889
3/4 "	0.750	19.050	12	0.109	2.769
			14	0.083	2.108
			16	0.065	1.651
			18	0.049	1.245
			20	0.035	0.889

From standard pipe sizes for evaporator assume outer diameter = 15.875 mm and Inner diameter = 14.63 mm

Estimation of heat transfer coefficients (U_o):

Inside the tube is saturation vapour R134a.

$$Re = \frac{4\dot{m}}{\pi d_i \mu} \qquad (4.37)$$

At T_e = -4°C

- $\mu_v = 10.76 * 10^{-6}\ kg/_{m.s}$
- $\mu_l = 302.2 * 10^{-6}\ kg/_{m.s}$
- $x = 0.3536$
- $\mu = \mu_l + x(\mu_v - \mu_l)$

- μ =199.147$* 10^{-6}\, {kg}/_{m.s}$
- $\dot{m} = 0.05 \quad {kg}/_{s}$
- $k_v = 11.45 * 10^{-3}\, {w}/_{m.k}$
- $k_l = 95.3 * 10^{-3}\, {w}/_{m.k}$
- $k = k_l + x(k_v - k_l)$
- K=65.65$* 10^{-3}\, {w}/_{m.k}$
- $Cp = 1.33\ kJ/kg.k$
- a= $k/\rho cp$=kv_3/cp
- $Pr = v/a$=$\frac{\mu cp}{k}$ =4.0345

$$Re = \frac{4 * 0.05}{\pi * 0.01463 * 199.147 * 10^{-6}}$$

$Re = 23949.4$ turbulence flow inside the tube so heat transfer in fully developed turbulent

flow in smooth tubes is that recommended by Dittus and Boelter for cooling of the fluid is given by [48].

$$Nu = 0.023 * [Re]^{\frac{4}{5}} * [Pr]^{0.3} \tag{4.38}$$

$$Nu = 0.023 * [23949.4]^{\frac{4}{5}} * [4.0345]^{0.3} = 111.4$$

Therefore $h_i = Nu * k/d_i$ (4.39)

$$=111.4*65.651 * 10^{-3}/0.01463$$

$$= 499.9\ {w}/_{m^2 * K}$$

Air side heat transfer coefficients

For outside tubes

Properties of air at average temperature 23^{0}C (296 k)

- $\rho = 1.1774\, {kg}/_{m^3}$
- $v = 15.69 * 10^{-6}\, {m^2}/_{s}$
- K= 26.24$* 10^{-3}\, {w}/_{m.k}$
- α= $22.16 * 10^{-6}\, {m^2}/_{s}$

- $Pr = 0.708$
- $Prs = 0.722$ at T_{sat}=-4^{0}C =269.3k

[Above values are obtained from Refrigerating and Air-Conditioning Engineers] [22].

For the circular cylinder the characteristic length is the diameter, and the Reynolds number is defined as

$$Re = V * do/ \upsilon \tag{4.40}$$

Assume velocity of air inside cooler box is 0.25m/s

$$Re = 0.25 * 0.015875/15.69 * 10^{-6} \;\; =252.95$$

The resulting correlation for average heat-transfer coefficients in cross flow over circular cylinders by Zukauskas is of the form [47].

$$Nu = c * [Re]^m * [pr]^n * (\frac{pr}{prs})^{1/4} \tag{4.41}$$

Table 4.6: Constants for the circular cylinder in cross flow[47]

Re_D	C	m
1–40	0.75	0.4
40–1000	0.51	0.5
10^3–2×10^5	0.26	0.6
2×10^5–10^6	0.076	0.7

Hence, from Table 4.6 C=0.51and m=0.5. Also, since $Pr < 10$, n=0.37. It follows that

$$Nu = 0.51 * [252.9477374]^{0.5} * [0.708]^{0.37} * (\frac{0.708}{0.722})^{1/4}$$

$$Nu = 7.103492005$$

Therefore $$h_o = Nu * k/d_o \tag{4.42}$$

$$=7.103492005*26.24*10^{-3}/0.015875$$

$$= 11.74 \; {}^{W}/_{m^2 * K}$$

overall heat transfer coefficient:

$$1/U_o = \left[\frac{A_o}{A_i h_i} + \frac{A_o \ln({}^{d_o}/_{d_i})}{2\pi k L} + \frac{1}{h_o}\right] \tag{4.43}$$

$A_o = \pi d_o L$ and $A_i = \pi d_i L$ then

$$\frac{1}{U_o} = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o} \right]$$

$$\frac{1}{U_o} = \left[\frac{0.015875}{0.01463*499.9} + \frac{0.015875 \ln(0.015875/0.01463)}{2*401} + \frac{1}{11.74} \right]$$

Therefore, $U_o = 9.36 \ W/m^2 * k$

$$Q = U_o * A_e * F * LMTD \quad (4.44)$$

In condensation or boiling (evaporation), the fluid normally remains at essentially constant temperature and the relations are simplified. For this condition correction factor, F=1.0 for evaporation or condensation [48].

Refrigerating capacity = load taken by Evaporator = 7.14 KW= $A_e * U_o * F * LMTD$

$$A_e = 7137.7 \text{ W}/(9.36 \ W/m^2 * k * 300.744 \text{ k}) = 2.54 \ m^2$$

Diameter of the coil (d_o) = 15.875mm

$$\text{Then} \quad A_e = \pi d_o L_e \quad (4.45)$$

Length of the coil (L) = $A_e/\pi d_o$ = 2.53654/(π x 0.015875) =50.886m

Length of coil in one turn = π ∗ D_{tanker} = 128.74 cm (4.46)

Number of turns in the tank = 50.886/128.740

= 39.526 turns, say 40 turns

The evaporator coil should be arranged in the outer side of the tank that will be easy for periodic cleaning.

Case II: Length of evaporator at cooling capacity of 0.71 kw

Length of evaporator is also design at cooling capacity of 0.71 KW (or 300 min require time to cool) for check and compare the experimentally and theoretical performance of locally fabricated cooler.

Length of evaporator at this cooling capacity follows the same step with the above design. All properties are the same because of the same evaporator temperature the only difference is diameter of coil mass flow of refrigerant.

The inner and outer diameter of evaporator for this cooling capacity is assuming 5.639mm and 6.35mm respectively.

Inside convection heat transfer coefficient(h_i)

$$Re = \frac{4\dot{m}}{\pi d_i \mu}$$

Using calculated Re and properties listed in length of evaporator calculation in case I, so

$$Re = \frac{4 * 0.005}{\pi * 0.0057 * 199.147 * 10^{-6}}$$

$Re = 6213.508066$ turbulence flow inside the tube so

$$Nu = 0.023 * [6213.51\ ]^{\frac{4}{5}} * [4.0345]^{0.3} = 37.86$$

Therefore $h_i = Nu * k/d_i$

$=37.86*65.651 * 10^{-3}/0.005639$

$= 440.73\ {}^{W}/_{m^2 * K}$

outer convection heat transfer coefficient(h_o)

$$Re = V * do/\,\upsilon$$

Assume velocity of air inside cooler box is 0.5m/s

$Re = 0.5 * 0.00635/15.69 * 10^{-6} = 202.36$

$$Nu = c * [Re]^m * [pr]^n * (\frac{pr}{prs})^{1/4}$$

Hence, from Table 4.6 C=0.51and m=0.5. Also, since Pr<10, n=0.37. It follows that

$$Nu = 0.51 * [202.36]^{0.5} * [0.708]^{0.37} * \left(\frac{0.708}{0.722}\right)^{\frac{1}{4}} = 6.35$$

Therefore $h_o = Nu * k/d_o$ $=6.35*26.24*10^{-3}/0.00635$

$= 26.25\ {}^{W}/_{m^2 * K}$

Therefore, U = 24.6 ${}^{W}/_{m^2 * K}$

$Q = U_o * A_e * F * LMTD$

Refrigerating capacity = load taken by Evaporator = 713.8 = $A_e UF\ LMTD$

$$A_e = 713.8/\ (24.6\ {}^{W}/_{m^2 * K}\ x\ 300.744\ \text{k}) = 0.29646\ m^2$$

Diameter of the coil (d_o) = 6.35 mm

Then $A_e = \pi d_o L$

Length of the coil (L) = $A_e/\pi d_o$ = 0.09646/(π x 0.00635) =14.868 m

Length of coil in one turn = $\pi * D_{tanker}$ = 128.74 cm

Number of turns in the tank = 14.868/128.740

= 11.549 turns, say 12 turns

The height of tanker is 63 cm, the space occupy by coil is $6.35mm * 12 + 20\ mm = 90\ mm = 9.62\ cm$

gap between each turn=$\frac{(63-9.62)\ cm}{12} = 4.5\ cm$

Provide 4.5 cm gap between each turn of the coil. The evaporator coil should be arranged in the outer side of the tank that will be easy for periodic cleaning.

4.3.2 Thermal analysis for condenser coil

In the condenser design, the principle dimensions of the condenser are determined both the in refrigerant side and the air side. The principle dimensions of a condenser may be given as the number of turns, the diameter of tube, the length of the tube and thickness.

Since inside the condenser tubes there are two different flow regimes, namely, the superheated vapor and the two-phase mixture, the lengths of these regions are calculated separately, and sum of these lengths gives the total length of the condenser.

The superheated vapor, which leaves the compressor, enters the condenser and leaves as a saturated liquid.

$$\dot{Q}_r = U * A_C * F * LMTD = \dot{w}_c + \dot{Q}_c = \dot{Q}_{sup} + \dot{Q}_{tp} \quad (4.47)$$

The flow is cross flow use correction factor and Assuming counter flow type heat exchanger for the heat transfer.

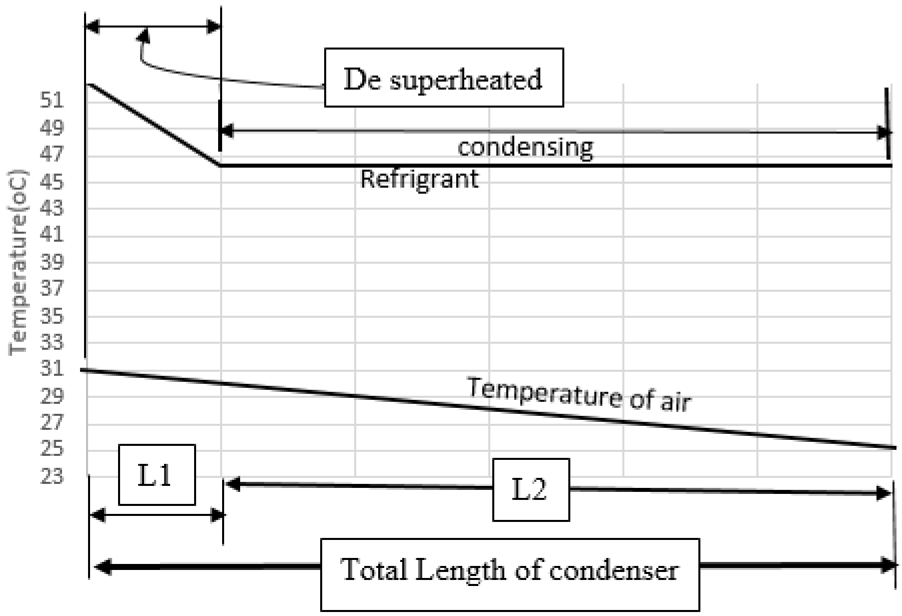


Figure 4.5 Air-cooled condenser temperature curves of refrigerant R134a and cooling air

- Temperature of refrigerant-R134a at the inlet of condenser $T_2 = 52.5\ ℃$
- Temperature of refrigerant-R134a at the out let of condenser $T_3 = 46.3℃$
- The inlet air temperature in the condenser is $T_i = 28℃$
- The outlet air temperature from the condenser is $T_o = 31℃$ (assume value)

So the change in temperature will be:

$$\Delta T_1 = T_2 - T_o = 52.5℃ - 31℃ = 21.5℃ \tag{4.48}$$

$$\Delta T_2 = T_2 - T_i = 52.5℃ - 28℃ = 14.5℃ \tag{4.49}$$

$$\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(21.5 - 14.5)℃}{\ln(21.5℃/14.5℃)} = 17.77℃ = 290.92\text{k} \tag{4.50}$$

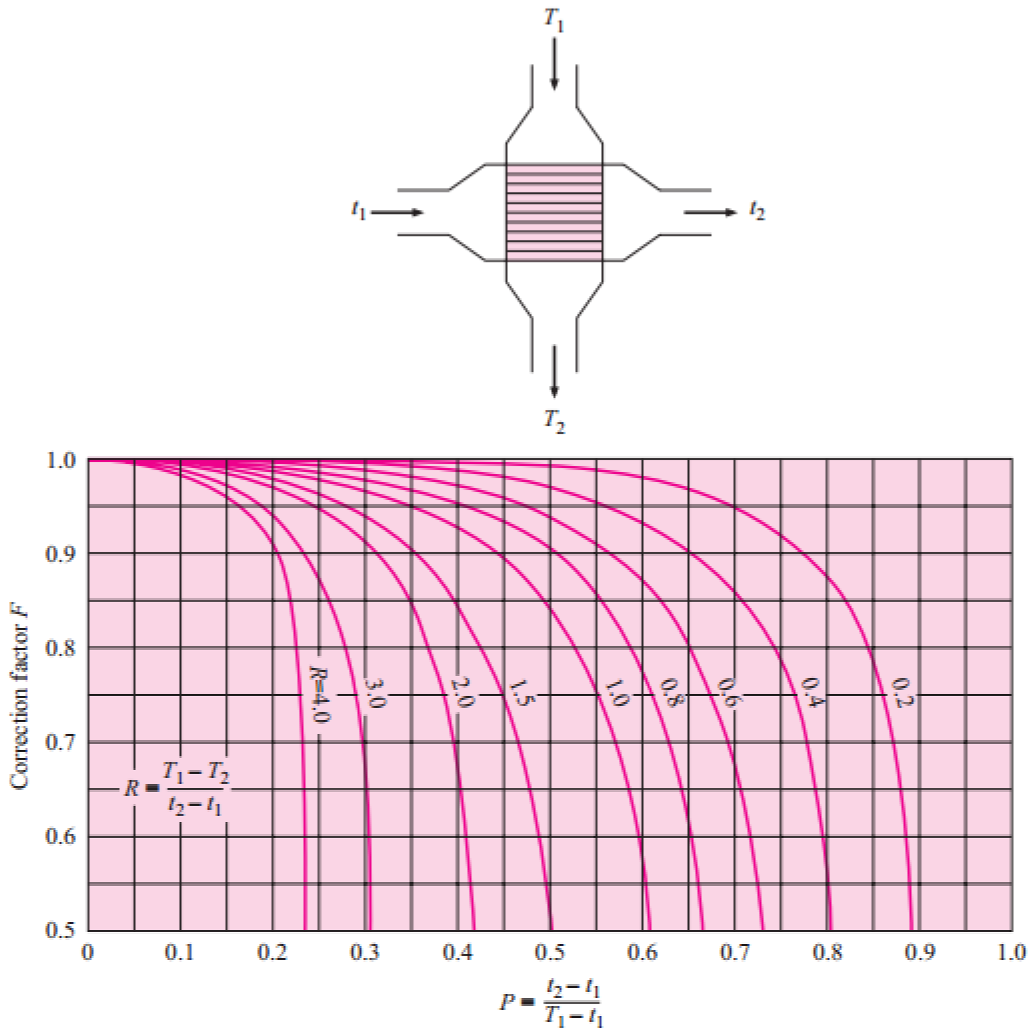


Figure 4.6: Correction-factor plot for single-pass cross-flow exchanger, one fluid mixed, the other unmixed [48]

$P = \frac{T_O - T_i}{T_2 - T_i} = \frac{31-25}{52.5-25} = 0.218$

$$R = \frac{T_2 - T_3}{T_o - T_i} = \frac{52.5 - 46.3}{31 - 25} = 1.033$$

Case I: Length of condenser at cooling capacity of 7.14 kw

a). The superheated flow regimes

$$Q_{sup} = h_2 - h_{3g}$$

From specification the inner and outer diameter of condenser are chosen

$$d_o = 0.009525m = 9.525\text{mm}$$

$$d_i = 0.00828\text{m} = 8.28\text{mm}$$

Inside the tube refrigerant is saturation vapour R134a.

$$\mathrm{Re} = \frac{4\dot{\mathrm{m}}}{\pi \mathrm{d_i} \mu} \tag{4.51}$$

At T_c= 46.31°C

- $\mu_v = 13.52 * 10^{-6}\, {kg}/{m.s}$
- $\mu_l = 16.57 * 10^{-6}\, {kg}/{m.s}$
- $\mu = \mu_l = 16.57 * 10^{-6}\, {kg}/{m.s}$
- $\dot{m} = 0.\dot{0}5\ 5\, {kg}/{s}$
- $\mathrm{k_v} = 16.34 * 10^{-3}\, \mathrm{W}/\mathrm{m.k}$
- $\mathrm{k_l} = 72.2 * 10^{-3}\, \mathrm{W}/\mathrm{m.k}$
- $\mathrm{k} = \mathrm{k_l} = 72.2 * 10^{-3}\, \mathrm{W}/\mathrm{m.k}$
  Cp=1.43 kJ/kg.k
- a= k/ρcp=kv_3/cp
- $\mathrm{Pr} = \mathrm{v/a} = \frac{\mu \mathrm{cp}}{\mathrm{k}}$ =0.3282

$$Re = \frac{4 * 0.05}{\pi * 0.00828 * 16.57 * 10^{-6}}$$

$Re = 508580.7$ turbulence flow inside the tube so,

$$Nu = 0.023 * [Re]^{\frac{4}{5}} * [pr]^{0.3} \tag{4.52}$$

$$Nu = 0.023 * [508580.7\,]^{\frac{4}{5}} * [0.3282]^{0.3}$$

$$= 604.86$$

Therefore $$h_i = Nu * k_v/d_i \tag{4.53}$$

$$=604.86*16.34 * 10^{-3}/0.00828$$

$$= 1193.66\ {W}/{m^2 * K}$$

Finding of h_o

Properties of air at 28℃ since $\mathrm{T_{AV}} = \frac{31℃+28℃}{2} = 29.5℃$

Properties of air at average temperature 29.5^{0}C (302 k)

- $\rho = 1.1774 \, {}^{kg}/_{m^3}$
- $v = 15.69 * 10^{-6} \, {}^{m^2}/_{s}$
- K= $26.24 * 10^{-3} \, {}^{W}/_{m.k}$
- a= $22.16 * 10^{-6} \, {}^{m^2}/_{s}$
- Pr = 0.708
- Prs=0.7 at T_{sat}=46.3^{0}C =319.3k

[Above values are obtained from Refrigerating and Air-Conditioning Engineers] [22]

Formulae used:

$$Re = V * do/ v \tag{4.54}$$

From methodological data free stream velocity of air in adama is the range (0.5-4.5m/s) 0.5m/s (use low velocity for design).

$$Re = 0.5 * 0.009525/15.69 * 10^{-6}$$

Re =303.5

$$Nu = c * [Re]^m * [pr]^n * (\frac{pr}{prs})^{1/4}$$

Hence, from Table 4.6 C=0.51and m=0.5. Also, since Pr<10, n=0.37. It follows that

$$Nu = 0.51 * [303.5]^{0.5} * [0.708]^{0.37} * (\frac{0.708}{0.700})^{1/4}$$

$$Nu = 7.84$$

Therefore $h_o = Nu * k/d_o$

$$=7.84*26.24*10^{-3}/0.009525$$

$$= 21.6 \, {}^{W}/_{m^2 * K}$$

Over all heat transfer coefficient:

$$ {}^{1}/_{U_o} = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln({}^{d_o}/_{d_i})}{2k} + \frac{1}{h_o} \right] \tag{4.55}$$

$$1/U_o = \left[\frac{0.0095}{0.00828*1193.66} + \frac{0.009525*\ln(0.009525/0.00828)}{2*401} + \frac{1}{21.6}\right]$$

$$U_o = 21.16\, W/m^2 * K$$

Heat removed from the condenser in superheated flow regimes ($\dot{Q}_{sup}$)

$$\dot{Q}_{sup} = \dot{m}\ (h_2 - h_{3g}) = 0.05\ (281.542 - 273.9) = 418.59\text{W}$$

also $\dot{Q}_{sup} = U_o * A_1 * F * LMTD$ (4.56)

where

F = 0.975 at P=0.218 and R = 1.033 from Figure 4.6

$A_1 = \dot{Q}_{sup}/(U_o * F * LMTD)$=$418.59/(21.16 * 0.975 * 290.92) = 0.07\ m^2$

A_1 =Area of condenser for superheated flow regimes

$$A_1 = \pi d_o L_1 \quad (4.57)$$

$then$ L_1=0.07/(3.14*0.009525) = 2.33 m

b). Two-phase mixture flow regimes

$LMTD$ = 17.77℃ = 290.92k F=1.0 because two-phase mixture flow regimes is condensation

$$U_o = 21.2$$

$\dot{Q}_{tp} = \dot{m}\ (h_{39} - h_{3f})$ =$0.05\ (273.9 - 117.8) = 8550.4$ Watt

$A_2 = \dot{Q}_{tp}/(U_o * F * LMTD)$=$8550.3775/(21.1623 * 1 * 290.92) = 0.39\ m^2$

A_2 =Area of condenser for two-phase mixture flow regimes

$$A_2 = \pi d_o L_2$$

$then$ $L_2 = \frac{0.3888}{3.14*0.009525}$ =12.999

Total length of the condenser L=$L_1 + L_2$ = 2.33m+12.99m=15.3 m

Length in one turn = (35cm + 35cm) = 70cm

Number of turns required = 15.329 / 0.70

= 21.898

= 22 turns

Case II: Length of condenser at cooling capacity of 0.71 kw

Length of condenser at cooling capacity 0.71 KW (or 300 min time to cool) follows the same step with the above design. All properties are the same because of the same condenser temperature the only difference is diameter of coil.

The inner and outer diameter of condenser for this cooling capacity is assuming 5.639 mm and 6.35 mm respectively.

Inside convection heat transfer coefficient(h_i)

Re at mass flow rate 0.005 kg/s and $d_i = 5.639\ mm$

$$Re = \frac{4\dot{m}}{\pi d_i \mu}$$

Using calculated Re and properties listed in length of condenser calculation in case is

$$Re = \frac{4 * 0.005}{\pi * 0.005639 * 16.57 * 10^{-6}}$$

$Re = 74677.217$ turbulence flow inside the tube so

$$Nu = 0.023 * [Re]^{\frac{4}{5}} * [pr]^{0.3}$$

$$Nu = 0.023 * [74677.217\ ]^{\frac{4}{5}} * [0.3282]^{0.3} = 130.35266$$

Therefore $h_i = Nu * k_v/d_i$ =130.35266*$16.34 * 10^{-3}$/0.005639= $377.7\ {}^{W}/_{m^2 * K}$

outer convection heat transfer coefficient(h_o)

$$Re = V * do/\, v$$

Assume velocity of air in adama is (0.5m/s to 4.5m/s) 0.5m/s (use low velocity for design)

$Re = 0.5 * 0.00635/15.69 * 10^{-6}$ =202.4

$$Nu = c * [Re]^m * [pr]^n * (\frac{pr}{prs})^{1/4}$$

Hence, from Table 4.6 C=0.51 and m=0.5. Also, since Pr<10, n=0.37. It follows that

$$Nu = 0.51 * [202.3581899]^{0.5} * [0.708]^{0.37} * (\frac{0.708}{0.70})^{1/4}$$

$$Nu = 6.402899$$

Therefore $h_o = Nu * k/d_o$

$$=6.402899*26.24*10^{-3}/0.00635$$

$$h_o = 26.45\ {}^{W}/_{m^2 * K}$$

Over all heat transfer coefficient:

$$1/U_o = \left[\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}\right]$$

$$1/U_o = \left[\frac{0.00635}{0.005639*377.72} + \frac{0.00635\ln(0.00635/0.005639)}{2*401} + \frac{1}{26.45}\right]$$

$$U_o = 24.45167\ {}^{W}/_{m^2 * K}$$

Heat removed from the condenser in superheated flow regimes($\dot{Q}_{sup}$)

$$\dot{Q}_{sup} = \dot{m}\ (h_2 - h_{3g}) = 0.005\ (281.542 - 273.9) = 41.86\ \text{Watt}$$

also $\dot{Q}_{sup} = U_o * A_1 * F * LMTD$

where

F=0.975 at P=0.218 and R=1.033 from Figure 4.5

$A_1 = \dot{Q}_{sup}/(U_o * F * LMTD)$=41.859/(24.45167 * 0.975 * 290.92) = 0.0060353453m^2

A_1 =Area of condenser for superheated flow regimes

$$A_1 = \pi d_o L_1$$

then $L_1 = \frac{0.0060353}{3.14*0.00635}$ =0.30269m=30.3cm

b). Two-phase mixture flow regimes

$LMTD$=17.77°C=290.92k   F=1.0 because two-phase mixture flow regimes is condensation

$$U_o = 21.1623\, {}^{W}\!/_{m^2 * K}$$

$\dot{Q}_{tp} = \dot{m}\ (h_{39} - h_{3f})$ =0.005 $(273.9 - 117.8) = 855$ W

$A_2 = \dot{Q}_{tp}/(U_o * F * LMTD)$=$\frac{855.03775}{24.45*1*290.92} = 0.12m^2$

$A_2 = Area\ of\ condenser$ for two-phase mixture flow regimes

$$A_2 = \pi d_o L_2$$

$then$ $\quad \mathrm{L}_2 = \frac{0.12020785}{3.14*0.00635}$ =6.03 m

Total length of the condenser   L=$L_1 + L_2$ = 0.30 m+6.03 m=6.33 m

Length in one turn = (35cm + 35cm) = 0.7cm

Number of turns required = 6.33147 / 0.7

= 9.047

= 10 turns

Condenser Heat Rejection Ratio (HRR)

The heat rejection ratio (HRR) is the ratio of heat rejected to the heat absorbed (refrigeration capacity), that is,

$$\mathrm{HRR} = \frac{\mathrm{Qc}}{\mathrm{Qe}} = \frac{\mathrm{Qe+Wc}}{\mathrm{Qe}} = 1 + \frac{1}{\mathrm{COP}} \quad (4.58)$$

HRR $=1 + \frac{1}{3.89} = 1.26$

4.4 Capillary tube

Capillary tube is one of the basic components of a vapor compression refrigeration cycle. The function of the tube is to reduce the pressure of the refrigerant from condenser pressure to the evaporator pressure. Therefore, a long, narrow tube is used. There are two basic reasons for the high-pressure drop in the tube: Since the diameter of the tube is very small and the length very long, the frictional pressure drop occurs in the tube. The saturated liquid exiting the condenser turns into a two-phase mixture, when entering the evaporator. Since the mass flow rate of the refrigerant is constant, the density of it reduces due to the vapor formation. It leads to a higher velocity in tube and an accelerational pressure drop occurs.

The total pressure drop inside the capillary tube may be written as

$$\Delta \mathrm{P} = \Delta \mathrm{P_f} + \Delta \mathrm{P_{acc}} \tag{4.59}$$

4.4.1 Sizing of the capillary tube

The sizing of the capillary tube is in effect selecting the length and diameter of the tube. Any different combinations of the length and diameter may give the same pressure drop in the system. There are actually two basic methods to size the tube. The first method is commonly used in mass production. After selecting the diameter and length of the hose, with greater length is installed in the system. The hose is cut until the desired pressure drop is achieved. This method is called the *cut-and-try* method. The second method is used commonly for unique system designs. The length and diameter of the hose are selected and a hose which is a little shorter than the desired length is installed in the system. The desired pressure drop is then obtained by *pinching* the hose at some points.

Either using the cut-and-try method or the pinching method, the length and the tube diameter should be decided first. Hence, in the present study, the design of the capillary tube is performed by determining these two parameters.

4.4.2 Pressure drop in the capillary tube

In this study, an isenthalpic capillary tube is considered. Actually, the flow in a capillary tube is three-dimensional, compressible and in two-phase. However, the analytical studies with incompressible, one-dimensional and single phase flow assumptions yield highly accurate results.

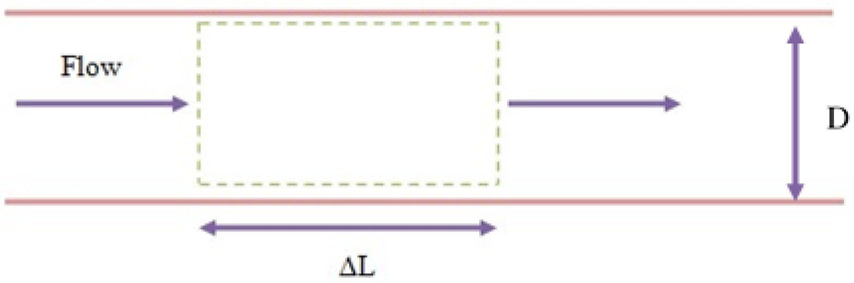


Figure 4.7: Capillary tube section

A capillary tube section is illustrated in Figure 4.7. To find the pressure drop in the section, conservation of mass and momentum equations should be solved for this control volume (CV).

Conservation of Mass:

$$\frac{\partial}{\partial t}\int_{Cv}\rho d\vartheta+\int_{Cs}\rho V.\hat{n}dA=0 \tag{4.60}$$

$$\frac{\partial(\rho V)}{\partial t}+\rho\, VA_c - \rho\, VA_c = 0 \tag{4.61}$$

where

$$A_c = \pi\frac{D^2}{4} \tag{4.62}$$

$$\frac{\partial(\rho V)}{\partial t} = 0 \tag{4.63}$$

Hence, $\rho\ V = \text{constant}$

Conservation of Momentum:

$$\sum M_{out} - \sum M_{in} = \sum F_{cv} \tag{4.64}$$

In other words, the difference between the momentums entering and leaving the CV is equal to the sum of the forces on the control volume. Including the gravitational force and shear forces in the equation,

$$A_c\left(\rho V^2 + \rho V\frac{\partial V}{\partial L}\Delta L\right) - A_c\rho V^2 = -A_c\frac{\partial P}{\partial L}\Delta L - \bar{\rho}gA_c\Delta L - \pi D\Delta L\tau_w \tag{4.65}$$

Since ΔL is an infinitesimal length, the gravitational force is neglected and higher order terms are diminished. Dividing both sides by $A_c\Delta L$ as ΔL converges to zero gives

$$\rho\, V \frac{\partial V}{\partial L} = -\frac{\partial P}{\partial L} - 4\frac{\tau_w}{D} \quad (4.66)$$

Shear stress causes frictional pressure drop in the pipe and therefore, it is written in terms of the friction factor

$$\tau_w = D\frac{\Delta p_{fric}}{4\Delta L} \quad (4.67)$$

where frictional pressure drop is calculated as

$$\Delta p_f = \rho\, f(\frac{\Delta L}{D})(\frac{V^2}{2}) \quad (4.68)$$

The friction factor is defined for laminar flow ($Re < 2300$) as

$$f = \frac{64}{Re} \quad (4.69)$$

For turbulent flow, the friction factor is defined by Blasius as

$$f = 0.33\ Re^{-0.25} \quad (4.70)$$

for $2300 < Re < 10^5$. The shear stress is written as

$$\tau_w = \rho\, f\frac{V^2}{2D} \quad (4.71)$$

Substituting the shear stress in (4.61),

$$\rho\, V\frac{\partial V}{\partial L} = -\frac{\partial P}{\partial L} - \rho\, f\frac{V^2}{2D} \quad (4.72)$$

Conservation of mass equation gives $\rho\, V = \text{constant}$, and this constant is the mass flux which can be written as

$$\rho\, V = \frac{\dot{m}}{A_c} = G \quad (4.73)$$

Substitution of G in (4.72) yields

$$G\frac{\partial V}{\partial L} = -\frac{\partial P}{\partial L} - f\frac{VG}{2D} \quad (4.74)$$

Integration of (4.74) over the infinitesimal length ΔL gives

$$G\Delta V = -\Delta P - ((f\frac{VG}{2D})_2 - (f\frac{VG}{2D})_1)\Delta L \tag{4.75}$$

Taking the friction factor and the velocity as the average values for states 1 and 2,

$$G\Delta V = -\Delta P - G\frac{\Delta L}{2*D}(fV)_{mean} \tag{4.76}$$

The total pressure drop may be found as

$$\Delta P = G\Delta V + G\frac{\Delta L}{2*D}(fV)_{mean} \tag{4.77}$$

To reduce the margin of errors in calculations, the total length of the hose is divided into segments and the total length of the hose is found as the sum of the lengths of these segments. The calculation procedure is explained next.

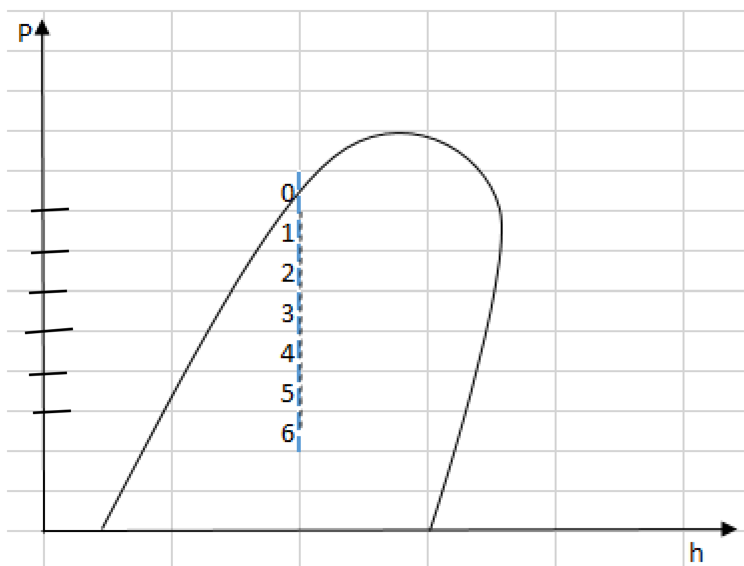


Figure 4.8: Segmentation in capillary tube

First, a differential temperature value should be assumed for all segments as seen in Figure 4.8. Thus, $T_1 = T_0 - \Delta T$. The whole temperature and pressure drops from the capillary tube inlet to the outlet is divided into suitable elementary steps of intermediate nodes, such as 46.3^0, 37.6^0, 29.2^0, 20.85^0, 12.46^0, 4.08^0 and -4.0^0c at their corresponding saturation pressures.

The properties of the refrigerant and friction factor are evaluated at state 0. Knowing T_1, the saturation pressure is found at state 1. Then, the liquid and vapor properties are found at state 1.

However, the vapor fraction must be known to evaluate the properties. Since the process is assumed to be isenthalpic, i.e. $h_0 = h_1$, the enthalpy at state 1 can be written as

$$h_1 = x_1 h_{g1} + (1 - x_1) h_{f1} \tag{4.78}$$

$$h_2 = x_2 h_{g2} + (1 - x_2) h_{f2}$$

and the vapor fraction is

$$x_1 = \frac{h_1 - h_{f1}}{h_{g1} - h_{f1}} \tag{4.79}$$

$$x_2 = \frac{h_2 - h_{f2}}{h_{g2} - h_{f2}}$$

Properties at state 1 can be calculated as

$$v_1 = x_1 v_{g1} + (1 - x_1) v_{f1} \tag{4.80}$$

$$v_2 = x_2 v_{g2} + (1 - x_2) v_{f2}$$

$$\mu_1 = x_1 \mu_{g1} + (1 - x_1) \mu_{f1} \tag{4.81}$$

$$\mathrm{Re} = \frac{4\dot{m}}{\pi D \mu_1}$$

$$V_1 = \frac{\dot{m} v_1}{A_c}$$

$$V_2 = \frac{\dot{m} v_2}{A_c}$$

At the end, the length of the hose for segment 1 is found as

$$\Delta L_1 = \frac{-\Delta P - G\Delta V}{\frac{G}{2*D}(fV)\mathrm{mean}} \tag{4.82}$$

The total length of the hose is the sum of the lengths of all segments.

One of the most important design criteria for a capillary tube is to avoid choking. If the velocity in the tube exceeds the speed of sound, choking occurs and it is not a desired condition. In case of choked flow, a capillary tube with a higher diameter should be selected.

Steps to design the capillary tube is given as follows:

i. Take the inlet state of the refrigerant as the exit state of the condenser.
ii. Read the evaporator temperature and pressure as inputs from property table R134a
iii. Choose a diameter value for the tube.
iv. Determine the temperature difference between the inlet and exit states.
v. Divide the tube into a selected number of segments with equal temperature differences.
vi. Read the inlet and exit saturation properties.
vii. Take the exit enthalpy being equal to the inlet enthalpy.
viii. Find the exit vapor fraction.
ix. Evaluate the inlet and exit state properties using the vapor fractions.
x. Evaluate the thermos-physical properties of the refrigerant as the mean values of inlet and exit states.
xi. Find the inlet velocity.
xii. Find the exit velocity.
xiii. Take the tube velocity as the mean value.
xiv. If the exit velocity is greater than the sound velocity, stop calculations and select a tube with a greater diameter.
xv. Determine the Reynolds number.
xvi. If the Reynolds number is less than 2300 calculate the friction factor from equation (4.69).
xvii. Else, use equation (4.70) to find the friction factor.
xviii. Calculate the length of the segment.
xix. Take the exit state of the first segment as the inlet state of the second segment.
xx. Apply the procedure until the last segment length is found.
xxi. Total length of the hose is the sum of the lengths of all segments.

Using the above step, length of capillary tube is calculated for cooling capacity of 7.14 KW using $d_o = 1.0mm\ and\ d_i = 0.9mm$ and for cooling capacity of 0.714 KW using $d_o = 0.8mm\ and\ d_i = 0.75mm$ (the diameter is chosen based on specification of capillary tube) for each section segments length of the hose is calculated at 46.3^0, 37.6^0, 29.2^0, 20.85^0, 12.46^0, 4.08^0 and -4.0^0c using properties at each temperature. Finally Using EXCEL, the results are presented in the result and discussion parts.

4.5 Sizing of insulation

Thermal conductivity of Spray foam K_i=0.5 $^{w}/_{m} * °C$

The outside of cooler box is rectangular with length=50 cm and width=70 cm (the same for all sides)

Properties of air at average temperature 28 ^{0}C (301 K)

- $v = 15.69 * 10^{-6}\,\mathrm{m}^2/_{\mathrm{s}}$
- K= 26.24$* 10^{-3}\,{}^{\mathrm{w}}/_{\mathrm{m.k}}$
- Pr = 0.708
- The free stream velocity ($U_\infty = 0.5\ m/s$) of air in adama

$$\mathrm{Re} = \frac{U_\infty L}{v} \qquad\qquad \mathrm{Nu} = 0.664\mathrm{Re}^{1/2}\mathrm{pr}^{1/3}4 \tag{4.83}$$

$$h_o = \frac{\mathrm{Nu} * \mathrm{K}}{\mathrm{L}} = 23.6\ \mathrm{w/m2} * {}^\circ\mathrm{C}$$

Critical radius of insulation $R_c = K_i/h_o$ (4.84)

R_C = 0.5/22.4=0.022118 m= 21.2 mm

Inner radius (R_1) = 220 mm and

Outer radius (R_2)= 250 mm are given from specification of water cooler.

Thickness of cover $t = R_2 - R_1$ =30 mm (4.84)

Critical radius of insulation

$t > R_C$

Therefore, the sufficient insulation is provided to control the heat loss to the surroundings.

4.6 Transient analysis of water temperature

The time taken to cool water was calculated using transient analysis of water temperature. To analyze a transient heat-transfer problem, we could proceed by solving the general heat-conduction equation by the separation-of-variables method.

$$\rho V c \frac{dT}{dt} = -\frac{K_{Al}A_s}{t_{Al}}(T - T_{Al}) - q_{loss} \tag{4.86}$$

$$q_{loss} = L * W * \frac{T_{Al} - T_{amb}}{(\frac{t}{k} + h_o)} \tag{4.87}$$

where K and t thermal conductivity and thickness of insulation respectively.

Rearrange and integrate both side of equation

$$-\frac{\rho V C_v\, t_{Al}}{K_{Al}A_s}\int_{44}^{T}\frac{dT}{T - T_{Al} + \frac{t_{Al}}{K_{Al}A_s} * q_{loss}} = \int_0^t dt \tag{4.88}$$

$$\frac{t_{Al}}{K_{Al}A_s} * q_{loss} = C \tag{4.89}$$

Time as function of temperature is given by

$$t = -\frac{\rho V C_p\, t_{Al}}{K_{Al}A_s}\left(\ln\frac{T - T_{Al} + C}{44 - T_{Al} + C}\right) \tag{4.90}$$

where properties of water at average temperature of water

Properties of Aluminum at average temperature of aluminum

$K_{Al} = 237 w/m.k$ $C_p = 903\, {}^{J}/_{Kg\, ℃}$ $V = 0.083 m^3$ $\rho = 2702\, {}^{Kg}/_{m^3}$

Temperature of aluminum at cooling capacity of 0.71 KW and 7.14 KW

A_s is surface area of Aluminum that contact with evaporator coil

$A_s = A_e$ t_{Al} = 1.5 mm

The surface temperature of aluminum is calculated as

$$Q_c = \frac{K_{Al}A_s}{t_{Al}}(T_{Al} - (-4)) = \frac{K_{Al}A_s}{t_{Al}}(T_{Al} + 4) \tag{4.91}$$

$$T_{Al} = \frac{Q_c t_{Al}}{K_{Al}A_s} - 4 \tag{4.92}$$

Substitute the value and get that T_{Al} =-3.8598 ℃ and C=0.741 ℃

Therefore, time as function of temperature is given by

$$t = -241.977\left(\ln\frac{47.12}{T + 3.12}\right) \text{ (min) at cooling capacity 0.71 KW} \tag{4.93}$$

$t = -24.1977(\ln\frac{47.25}{T+3.25})$ at cooling capacity 7.14 KW

4.7 Power of compressor at different size of water tanker

At cooling capacity of 7.14 kw (or time durations 30 min)

$T_{in} = 44$ ℃ (inlet temperature of water)

Specific heat of water (C_p) of water = 4180 $^{J}/_{Kg}$ ℃

Density of water (ρ_i) =990.16 $^{Kg}/_{m^3}$

V_i = volume of water in Litters

Cooling load is calculated as

$$Q_L = 990.16\ ^{Kg}/_{m^3} * V_i(litters) * 4.18\ ^{J}/_{Kg}\ ℃ * (44)℃/1000$$

At $time = 30\ min$

cooling capacity 7.14 kw

Power(kw)= mass flow rate$*$ Δh_c

$$Power(hp) = power(kw) * 1000/746$$

at different size of water tanker power of compressor results are presented in the next result and discussion part using EXCEL.

- For experimental performance evaluation 83 liters and ¼ hp compressor is selected and fabricate locally. But if we used higher horse power compressor, time to cool water is decrease but additional cost is requiring for both rating electrical power and initial cost of compressor.

CHAPTER FIVE

5 Manufacturing Process and Experimental Work

5.1 Fabrication and assembly of parts

5.1.1 Fabrication

The manufacturing process of water cooler is explained below.

Initially, all the components required for the system were gathered. Then welding of mild steel frame and water tanker was done in the welding workshop. Different types of tools are also used like tube cutter - to cut the tubes, tube bender - to bend the copper tube to the required angle, rivet set - to riveting process and welding equipment's - to joining process. To check the proper working of compressor, various tests were performed. Pressure test and vacuum test was conducted for both, compressor and condenser. After conducting all tests compressor were mounted on a wooden plate which was situated on the bottom of steel frame and electric wire connects to compressor for power supply.

Finally, the public water cooler is fabricated as per the requirement of the thesis.

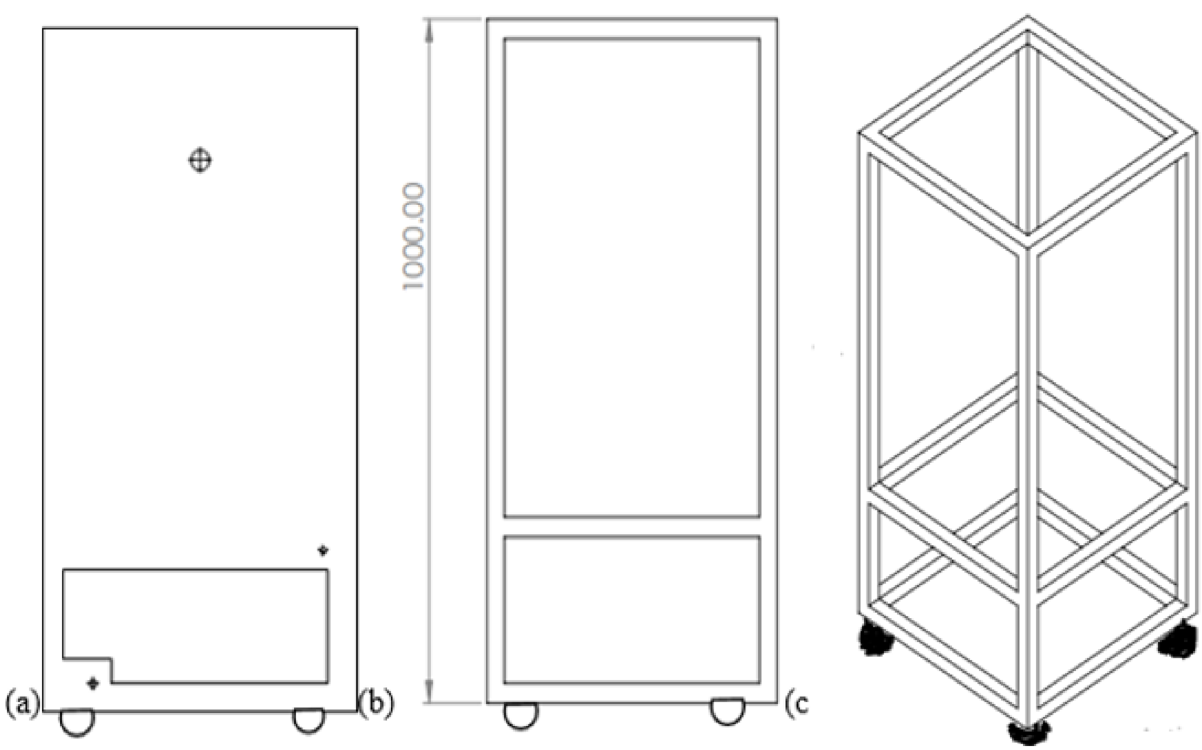


Figure 5.1: Cooler box (a) side view (b) front view (c) 3D of frame

Figure 5.2: Manufacturing process of cooler box

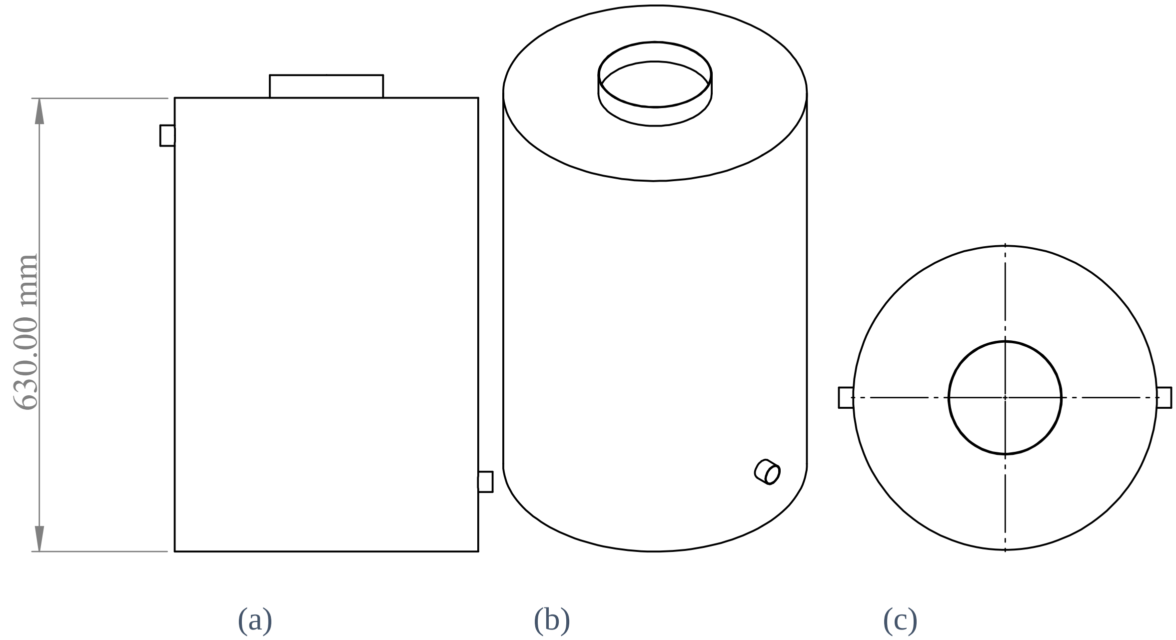


Figure 5.3: Water tanker (a) side view (b) 3D of frame (c) top view of tanker

5.1.2 Assembly of parts

The assembly of parts takes place after the construction of various parts of the cooler. The following are the assembly processes involved in this work; cutting, welding, brazing and soldering of the various parts together. A complete structure called a vapour compression refrigeration is formed when all the parts and components have been coupled and fixed in the right positions. The frame of the experimental cooler was constructed with rectangular steel pipes. The pipes are permanently joined together using arc-welding. The frame housed all other components in the system. Refrigerant pipe lines were brazed together using oxyacetylene welding. A brazing flux is used when the two metals to be joined are of different materials. e.g. iron upon copper. When the compressor is mounted on the system, a similar joining operation is carried out for the suction to compressor connection. Also, brazing operations were used for the connection of discharge line to the outlet of compressor and inlet of condenser. The liquid line is extended from condenser outlet to the evaporator inlet and it comprises of such components as the filter drier, and the capillary tube. The method employed in joining these components to the liquid line is the same with those of the suction and discharge lines connections. Finally, trolleys were welded to the four base corners of the frame. This will aid the mobility of the water cooler from one place to another. After making the connections, the system was cleaned off unwanted substances in the line and this was done by carrying out a flushing operation.

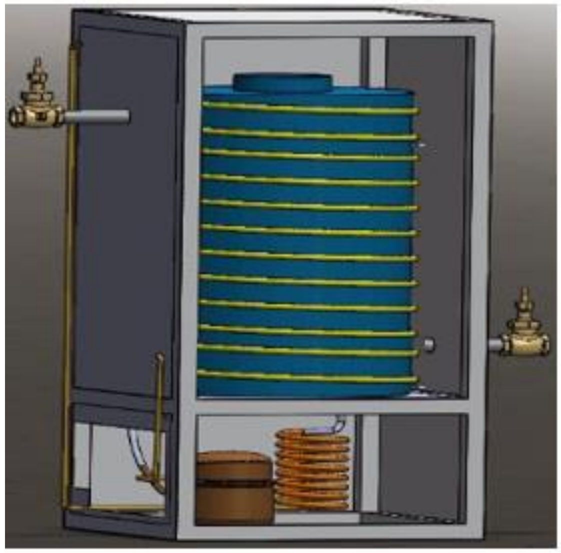


Figure 5.4: Sectional view of water cooler

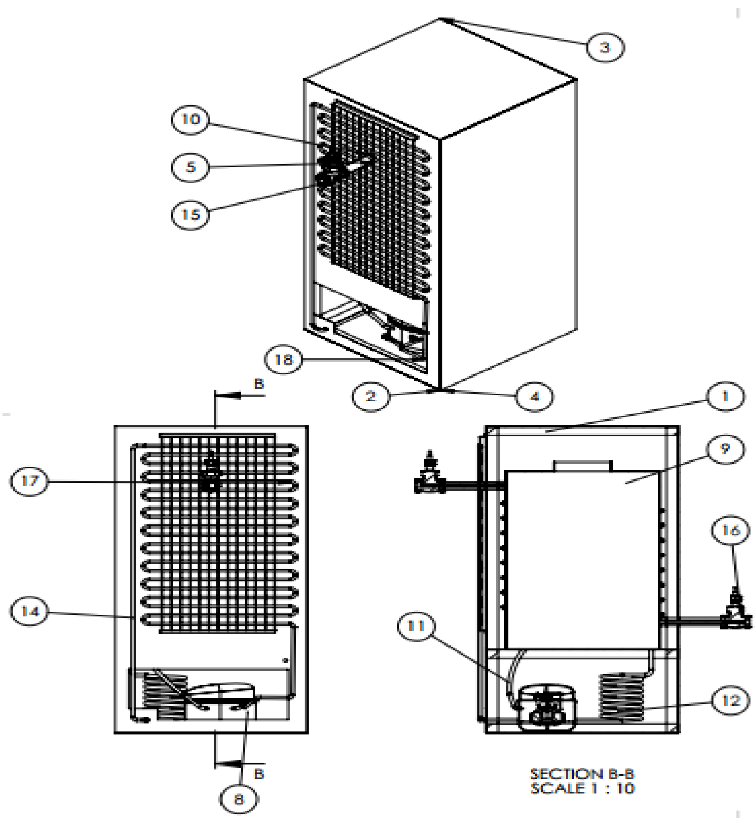


Figure 5.5: Three dimensional(3D) and sectional view with part description

Table 5.1: Water cooler part description

Item No.	Name of Parts	Quantity	Item No.	Name of Parts	Quantity
1	Frame	1	10	Evaporator	1
2	Body cover	2	11	Copper Tube1	1
3	Side cover 1	1	12	Capillary tube	1
4	Side cover 2	1	13	Copper Tube 2	1
5	Side cover 3	1	14	Copper Tube 3	1
6	Side cover 4	1	15	Pipe	1
7	Middle cover	1	16	Globe valve	2
8	Compressor	1	17	Copper Tube 4	2
9	Tanker	1	18	Copper Tube 5	1

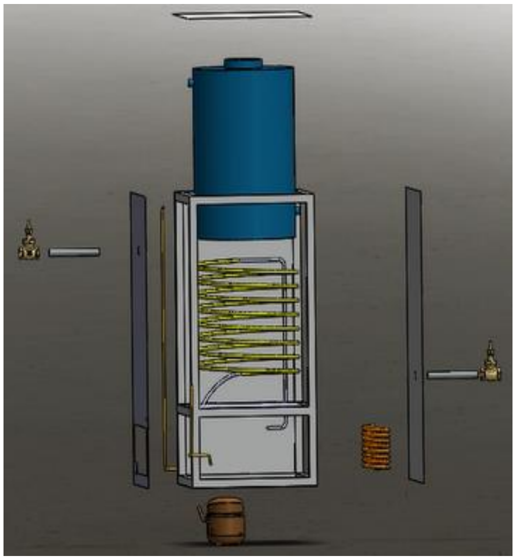
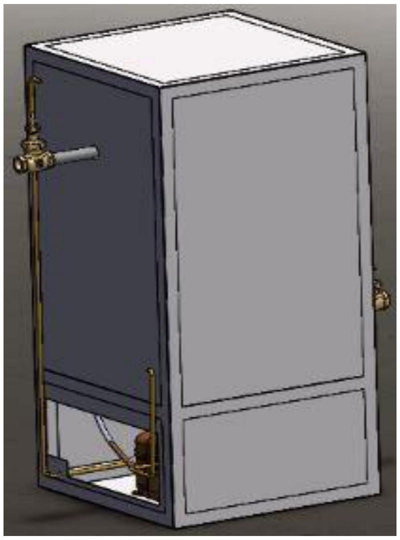
(a)  (b) 

Figure 5.6: 3D view of water cooler (a) before assembly (b) after assembly

5.2 Experimental work

The experiment result is used to conduct the validation of the model and to compare with numerical result of water cooler using refrigerant R134a. The experimental study was conducted in a thermal lab at thermal engineering laboratory, department of mechanical engineering, ASTU.

In order to know the performance characteristics of the vapour compression refrigerating system experimental, the temperature and pressure gauges are installed at each entry and exit of the component.

Experimental Set up

The following procedure is adopted for experimental setup of the vapor compression refrigeration system

i. Public water cooler is fabricated, working on vapor compression refrigeration system.
ii. Pressure and temperature gauges are installed at each entry and exit of the components.
iii. R134a refrigerant is charged in to the vapor compression refrigeration system.
iv. Leakage tests are done by using soap solution,
v. In order to further test the condenser and evaporator pressure and check purging daily for 12 hours and found that there are no leakages which required the absolutely the present investigation to carry out further experiment.
vi. Switch on the water cooler and observation is required for 1 hour and take the pressure and temperature readings at each section.
vii. The performance of the existing system is investigated, with the help of temperature and pressure gauge readings.
viii. Temperature and pressure gauge readings are taken and the performance is investigated.

The public water cooler selected for this thesis has the following specifications

Table 5.2: Specifications of public water cooler

Refrigerant used: R-134a
Capacity of water cooler: 83 liters
Thermostat used: cold thermostat
Filter-dryer used: throw away after use type
Compressor capacity: 0.25 H.P.
Condenser Length - 6.75 m Diameter- 6.35 mm Evaporator Length - 15.24 m Diameter- 6.35 mm Capillary Length - 3.6 m Diameter- 0.8 mm

The water tanker size is: Diameter = 410 mm Height = 630 mm

A 20 mm hole is made at the bottom and to side of the water tanker to give connection to the water pipe. The watercooler required many tests to evaluate the different aspects of its performance. Tests were run to evaluate the water temperature, the time it took to cool the water, the power consumption of the system, the mass/volume of the system, and the water purifying capabilities of the system, the heat dissipation, and the thermal resistance of the aluminum.

Water temperature

Since the main purpose of the system was to achieve water temperatures within the range of 7-13°C, water temperature measurements were one of the key factors measured experimentally. Evaluation of the water temperature indicated the performance of the entire cooler since many other design criteria affected the water temperature. The water temperatures were evaluated against the results from the survey that described consumer desire in terms of water temperature. Based on survey data, the target range of water temperature was 7-13°C. Temperature measurements were made every second. The uncertainty of the measurement can be taken from the uncertainty of the thermocouple, which is 0.4°C. The assumptions made in this test mostly revolved around the position of the thermocouples. The experiment assumed that the position of the thermocouples in the water represents the temperature of the cold water. It was also assumed that the thermocouple was not touching the surface of the tanker, but the water. Two thermocouples were used because the temperature throughout the water may not have been uniform, thus an average of both recorded temperatures provided a more accurate representation.

Time to Cool Water

The time taken to cool water was tested in every experiment in which the water temperature was taken. While this metric was gathered for the same tests, and at the same times, as the water temperature test previously mentioned, the time was an important unit to analyze in the experiment. Understanding how long the water took to achieve a desired temperature was an important aspect of the design.

The thermostat used for experimental set up is cold thermostat.

The function of thermostat is to stop the compressor when the temperature of water reach 10°C. if the temperature of water rises up due to additional hot water add to tanker or external heat transfer to water from surrounding the thermostat on the compressor and start removal heat until temperature of water reaches 10°C.

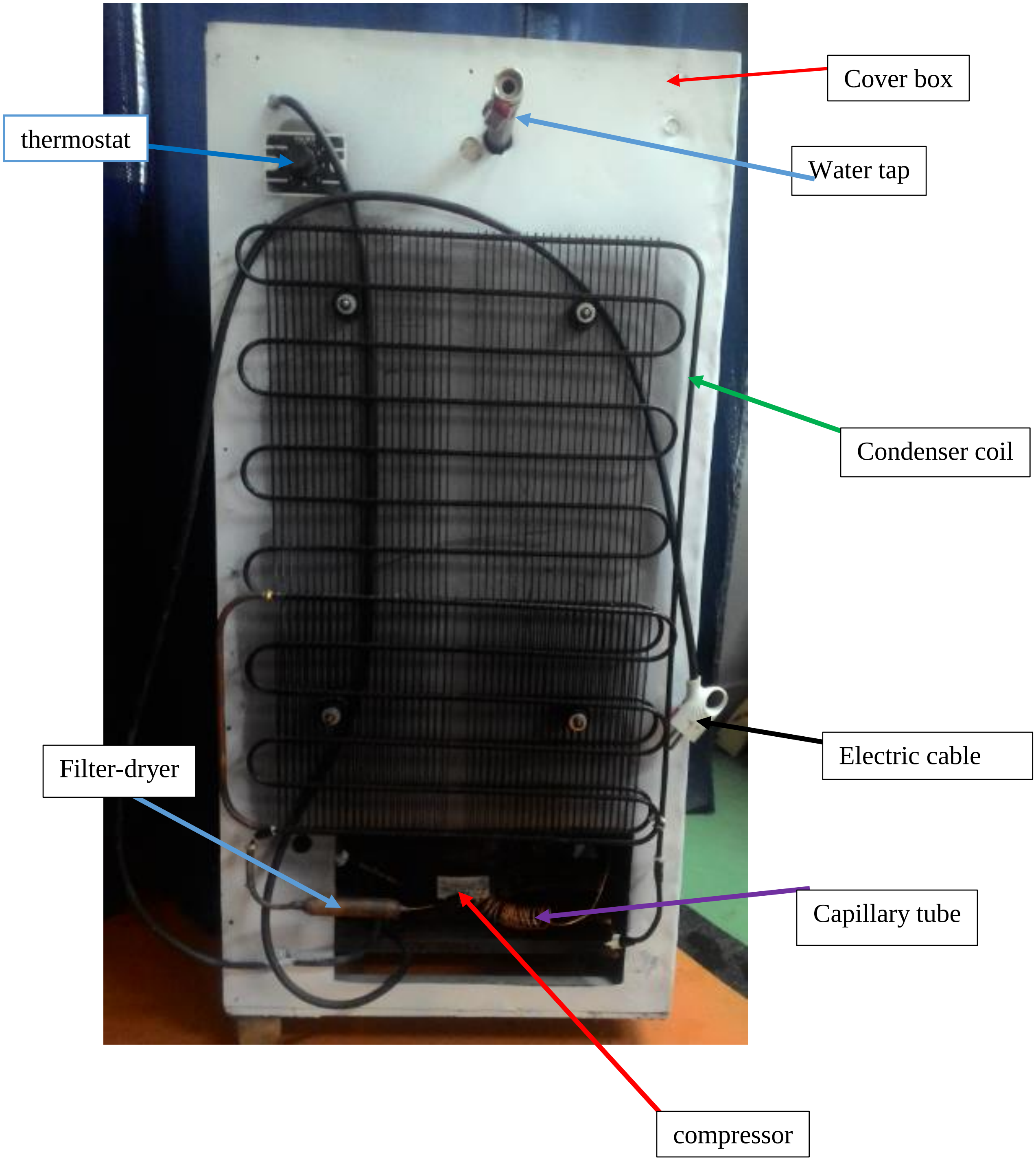


Figure 5.7: Diagram of water cooler components

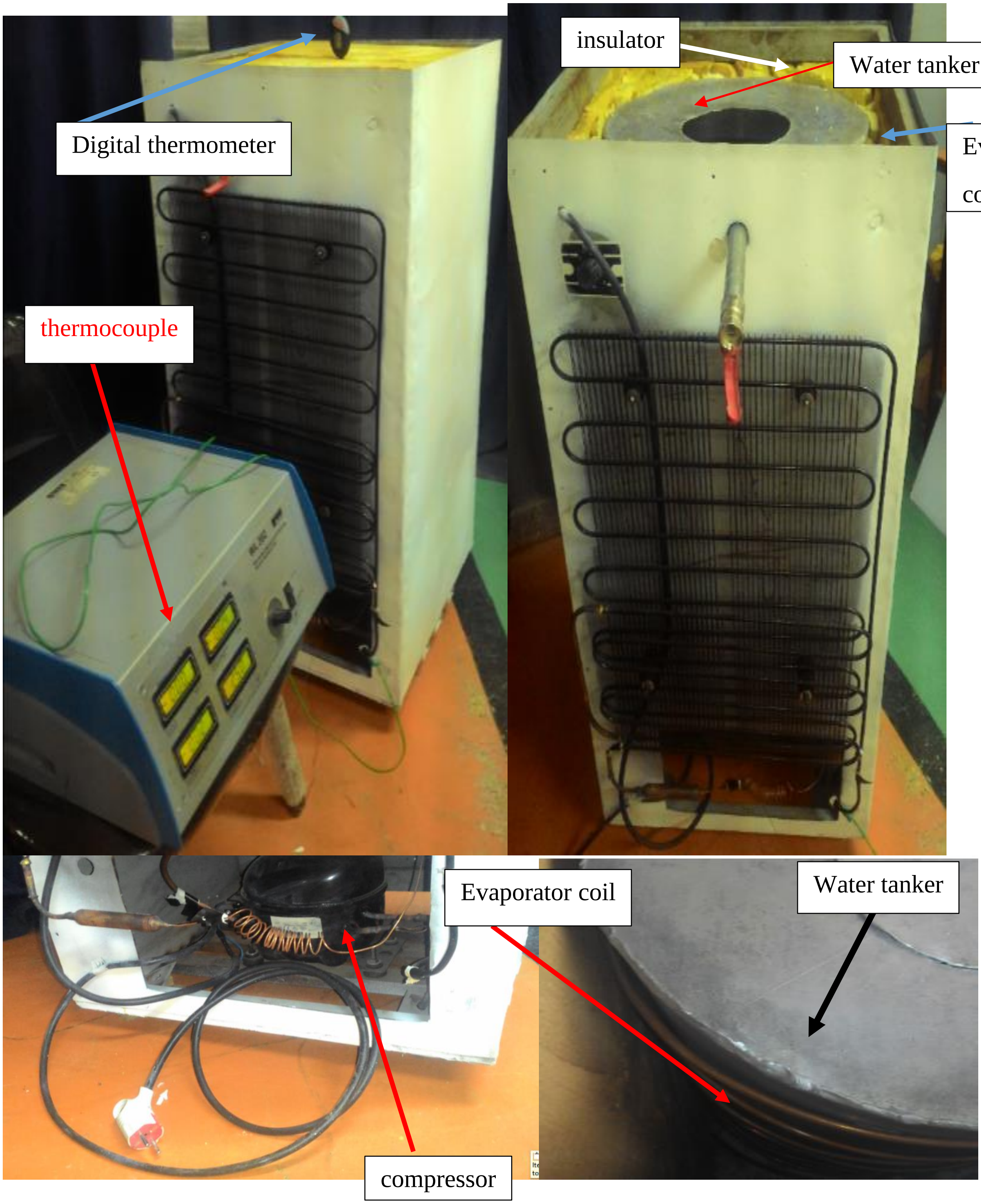


Figure 5.8: Equipment's for experimental set-up

CHAPTER SIX

6 Results and Discussions

Since the overall system performance is strongly influenced by operational parameters such as the evaporating and condensing temperatures, flow rates of refrigerant and ambient temperature, tests with the experimental cooler was planned in such a way as to allow for the evaluations of their effects on COP and various operational conditions. The experimental was designed using the fabricated public water cooler with the following inputs: evaporating temperature -4^{o}C, condensing temperature 46.3^{o}C and an average ambient temperature of 28^{o}C. Data obtained from the experiment was used to assess the quality of the model relative to the experimental tests.

Firstly, the design of public water cooler for student dormitory is started by fixing the volume of water cooled (V), V= 0.083 m^3 with inlet temperature of 44 ^{0}C and wants to decrease an outlet temperature of 10^{0}C in thirty minutes.

The results for each component and the complete cycle for theoretical results of water cooler presented in section 6.1 and those experimental results of water cooler are stated in section 6.3.

6.1 Theoretical results

a) Time require to cool water at different temperature and cooling capacity

Using equation (4.3) the results from excel

Table 6.1: Time require to cool water at different temperature and at cooling capacity of 7.14 kw and 0.71 kw.

Volume Of hot water add to tanker (Litter)	Mass of hot water add to tanker m_1(kg)	Volume of water remains in the tanker (Litter)	mass of water remains in the tanker m_3(kg)	(m_1+m_3) kg	T_3 (℃)	Q_L with10% allowance (KJ)	Time(min)	
							$at\ Q_c =$ 7.14kw	$at\ Q_c =$ 0.71kw
1	0.99	82	81.90	82.89	10.41	154.79	0.36	3.61
2	1.98	81	80.90	82.88	10.81	309.59	0.72	7.23
3	2.97	80	79.90	82.87	11.22	464.38	1.08	10.84
4	3.96	79	78.91	82.87	11.63	619.17	1.45	14.46
5	4.95	78	77.91	82.86	12.03	773.97	1.81	18.07
10	9.90	73	72.91	82.81	14.07	1547.94	3.61	36.14

15	14.85	68	67.92	82.77	16.10	2321.91	5.42	54.22
20	19.80	63	62.92	82.73	18.14	3095.87	7.23	72.29
25	24.75	58	57.93	82.68	20.18	3869.84	9.04	90.36
30	29.70	53	52.94	82.64	22.22	4643.81	10.84	108.43
35	34.66	48	47.94	82.60	24.27	5417.78	12.65	126.51
40	39.61	43	42.95	82.55	26.31	6191.75	14.46	144.58
45	44.56	38	37.95	82.51	28.36	6965.72	16.27	162.65
50	49.51	33	32.96	82.47	30.41	7739.69	18.07	180.72
55	54.46	28	27.97	82.43	32.46	8513.65	19.88	198.80
60	59.41	23	22.97	82.38	34.52	9287.62	21.69	216.87
65	64.36	18	17.98	82.34	36.58	10061.59	23.49	234.94
70	69.31	13	12.98	82.30	38.64	10835.56	25.30	253.01
75	74.26	8	7.99	82.25	40.70	11609.53	27.11	271.08
80	79.21	3	3.00	82.21	42.76	12383.50	28.92	289.16
81	80.20	2	2.00	82.20	43.17	12538.29	29.28	292.77
82	81.19	1	1.00	82.19	43.59	12693.08	29.64	296.39
83	82.18	0	0.00	82.18	44.00	12847.88	30.00	300.00

Time require to cool water is increase with temperature of water due to additional hot water is add to tanker as shown in the Figure 6.1.

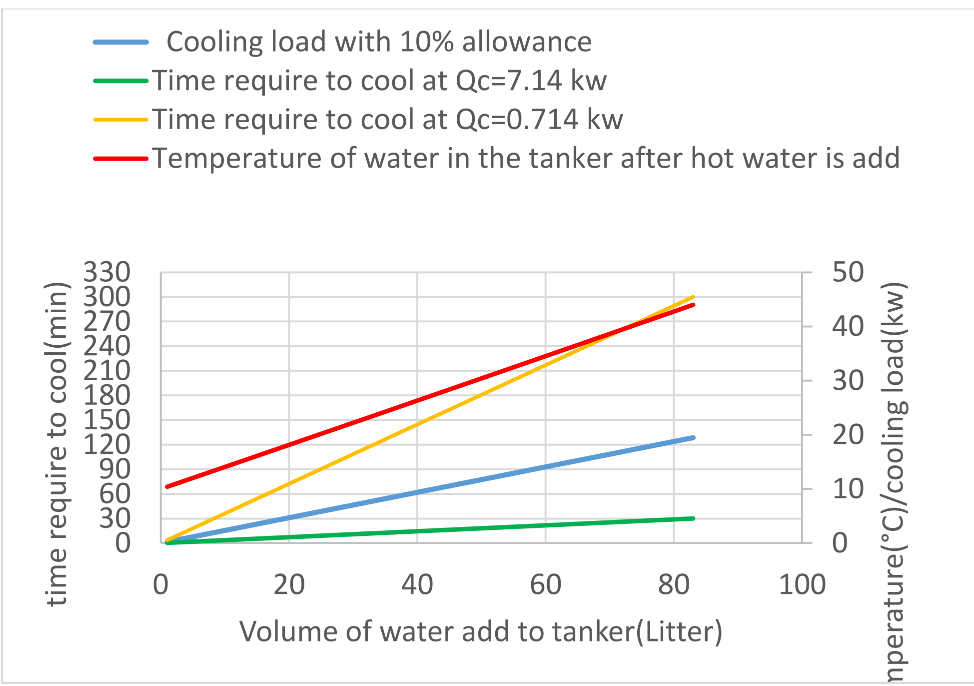


Figure 6.1: Time require to cool water at different temperature and cooling capacity

b) Power of compressor requires, cooling capacity and mass flow rate of refrigerant at different cooling time

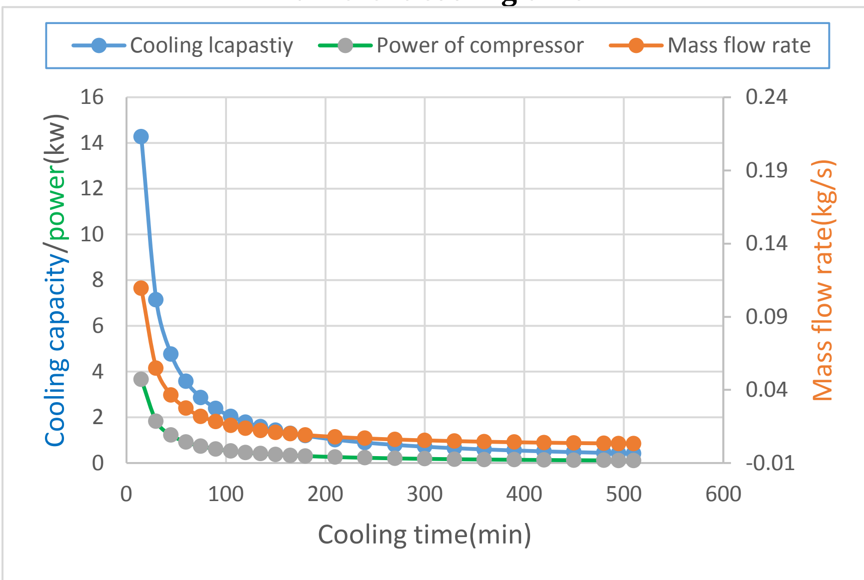


Figure 6.2: Power of compressor requires, cooling capacity and mass flow rate of refrigerant at different cooling time

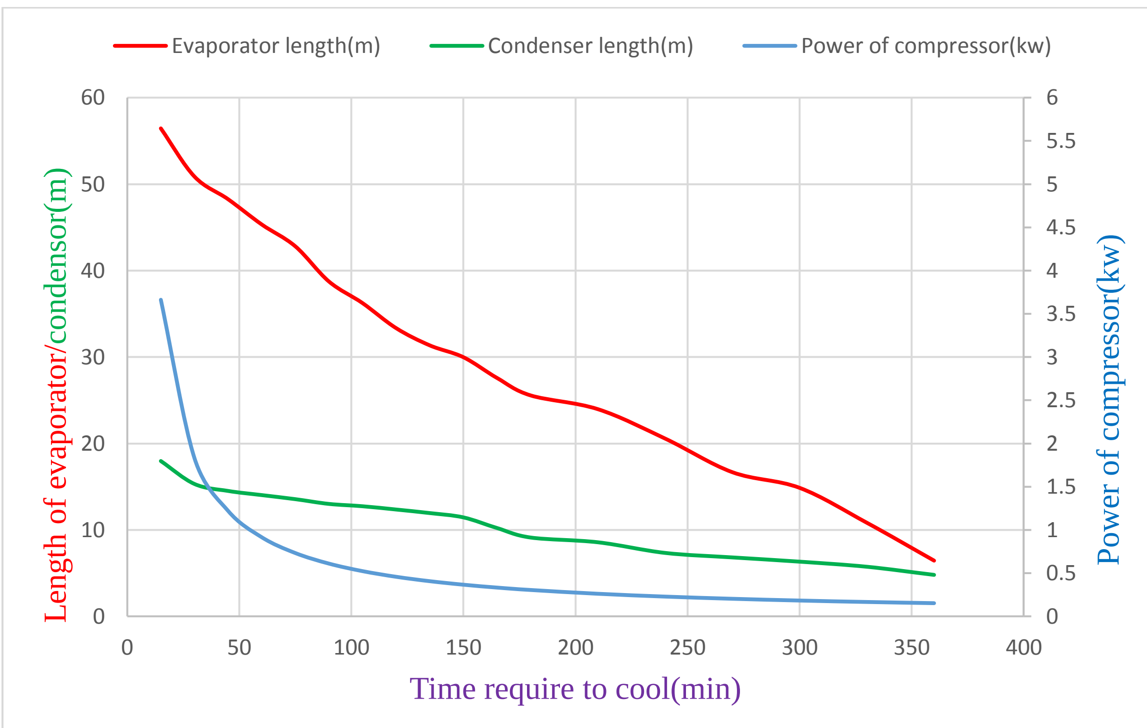


Figure 6.3: Power of compressor, length of evaporator and condenser at different cooling time.

c) Power of compressor at different size of water tanker

- At time requires to cool of 30 minutes the power of compressor is presented below.

Table 6.2:Power of compressor at different size of water tanker (at 30min time requires to cool)

Volume of water(litters)	Cooling load(KJ)	Cooling load with 10% allowance(KJ)	Cooling capacity(kw)	Mass flow rate(kg/s)	Power	
					kw	hp
10	1407.22	1547.94	0.86	0.007	0.22	0.30
20	2814.43	3095.87	1.72	0.013	0.44	0.59
30	4221.65	4643.81	2.58	0.020	0.66	0.89
40	5628.86	6191.75	3.44	0.026	0.88	1.18
50	7036.08	7739.69	4.30	0.033	1.10	1.48
60	8443.29	9287.62	5.16	0.040	1.32	1.77
70	9850.51	10835.56	6.02	0.046	1.54	2.07
75	10554.12	11609.53	6.45	0.049	1.65	2.22
80	11257.72	12383.50	6.88	0.053	1.77	2.37
83	11679.89	12847.88	7.14	0.050	1.83	2.45
90	12664.94	13931.43	7.74	0.059	1.99	2.66
100	14072.15	15479.37	8.60	0.066	2.21	2.96
110	15479.37	17027.31	9.46	0.073	2.43	3.25
120	16886.58	18575.24	10.32	0.079	2.65	3.55
130	18293.80	20123.18	11.18	0.086	2.87	3.84
140	19701.02	21671.12	12.04	0.092	3.09	4.14
150	21108.23	23219.05	12.90	0.099	3.31	4.44
160	22515.45	24766.99	13.76	0.106	3.53	4.73
170	23922.66	26314.93	14.62	0.112	3.75	5.03
180	25329.88	27862.86	15.48	0.119	3.97	5.32
190	26737.09	29410.80	16.34	0.125	4.19	5.62
200	28144.31	30958.74	17.20	0.132	4.41	5.92

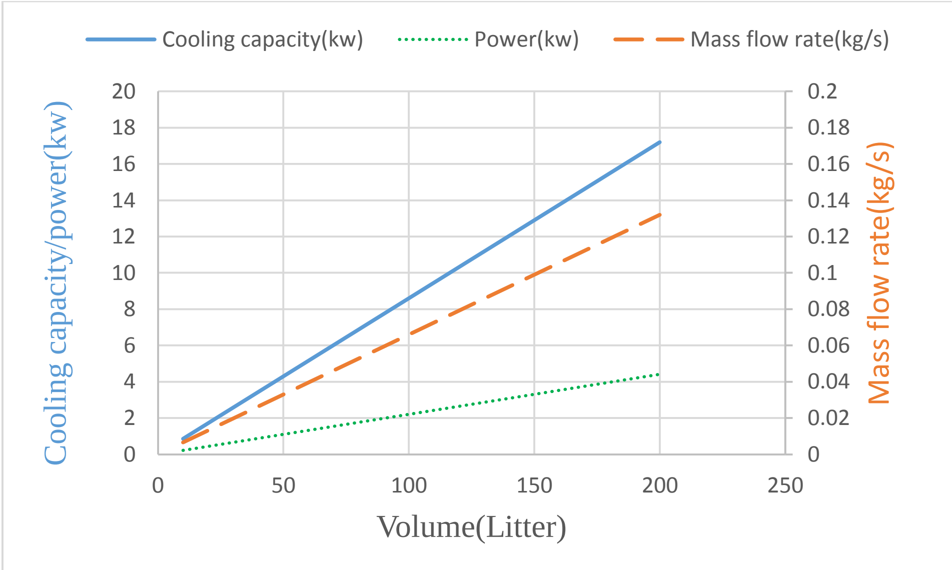


Figure 6.4: Power of compressor at different size of water tanker (at 30min time requires to cool)

- At time requires to cool of 300 minutes the power of compressor is presented below.

Table 6.3: Power of compressor at different size of water tanker (at 300min time requires to cool)

volume of water (litters)	cooling load(KJ)	cooling load with 10% allowance(KJ)	cooling capacity(kw)	mass flow rate(kg/s)	power	
					kw	hp
10	1407.22	1547.94	0.09	0.0007	0.02	0.03
20	2814.43	3095.87	0.17	0.0013	0.04	0.06
30	4221.65	4643.81	0.26	0.0020	0.07	0.09
40	5628.86	6191.75	0.34	0.0026	0.09	0.12
50	7036.08	7739.69	0.43	0.0033	0.11	0.15
60	8443.29	9287.62	0.52	0.0040	0.13	0.18
70	9850.51	10835.56	0.60	0.0046	0.15	0.21
75	10554.12	11609.53	0.64	0.0050	0.17	0.22
80	11257.72	12383.50	0.69	0.0053	0.18	0.24
83	11679.89	12847.88	0.71	0.0055	0.18	0.25
90	12664.94	13931.43	0.77	0.0059	0.20	0.27
100	14072.15	15479.37	0.86	0.0066	0.22	0.30
110	15479.37	17027.31	0.95	0.0073	0.24	0.33
120	16886.58	18575.24	1.03	0.0079	0.26	0.35
130	18293.80	20123.18	1.12	0.0086	0.29	0.38

140	19701.02	21671.12	1.20	0.0092	0.31	0.41
150	21108.23	23219.05	1.29	0.0099	0.33	0.44
160	22515.45	24766.99	1.38	0.0106	0.35	0.47
170	23922.66	26314.93	1.46	0.0112	0.38	0.50
180	25329.88	27862.86	1.55	0.0119	0.40	0.53
190	26737.09	29410.80	1.63	0.0125	0.42	0.56
200	28144.31	30958.74	1.72	0.0132	0.44	0.59

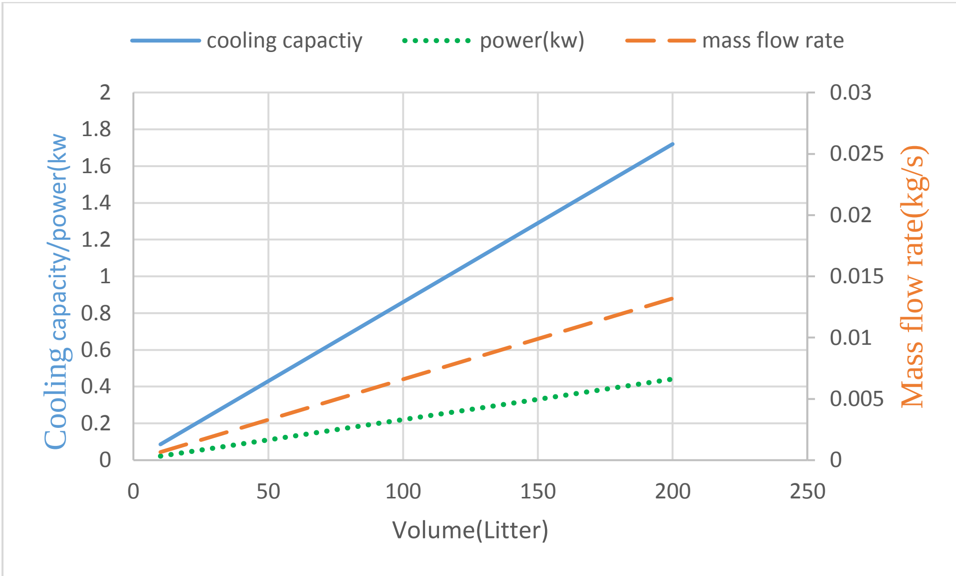


Figure 6.5: Power of compressor at different size of water tanker (at 300min time requires to cool)

d) Results of Transient analysis of water temperature

Table 6.4: Results of transient analysis of water temperature

Temperature (℃)	Time to cool		Temperature (℃)	Time to cool	
	At Qc=0.71kw	At Qc=7.14kw		At Qc=0.717kw	At Qc=7.14kw
44	0	0	25	122.4480941	12.24481
43	5.11	0.51	24	130.98	13.10
42	10.33	1.03	23	139.83	13.98
41	15.67	1.57	22	149.02	14.90
40	21.13	2.11	21	158.57	15.86
39	26.71	2.67	20	168.50	16.85
38	32.43	3.24	19	178.87	17.89
37	38.28	3.83	18	189.70	18.97
36	44.28	4.43	17	201.03	20.10
35	50.43	5.04	16	212.92	21.29

34	56.74	5.67	15	225.43	22.54
33	63.22	6.32	14	238.62	23.86
32	69.88	6.99	13	252.57	25.26
31	76.72	7.67	12	267.37	26.74
30	83.77	8.38	11	283.14	28.31
29	91.03	9.10	10	300.00	30.00
28	98.51	9.85	10	350.00	50.00
27	106.23	10.62	10	400.00	60.00
26	114.20	11.42			

Figure 6.6 and 6.7 shows the graph of temperature of water verse time require to cool water draw using equation 4.93.

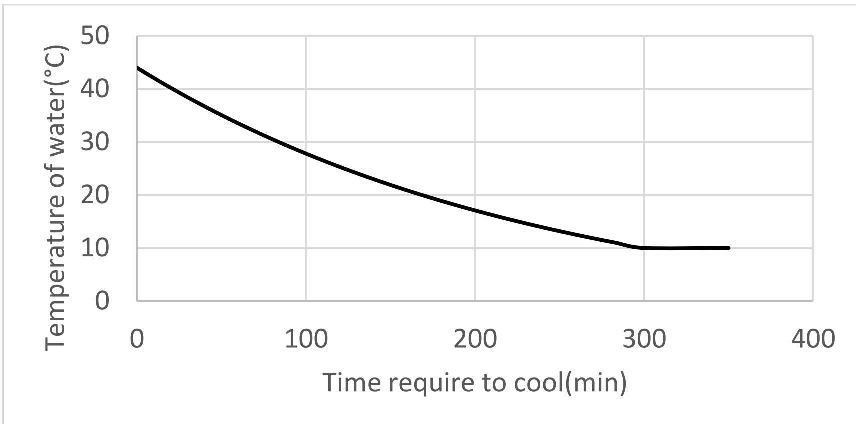


Figure 6.6: Temperature of water verse time requires to cool water at cooling capacity of 0.71 kw

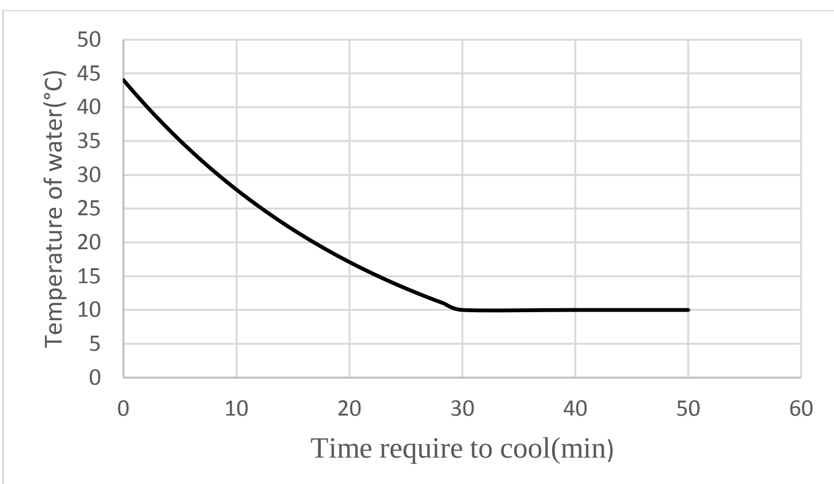


Figure 6.7: Temperature of water verse time requires to cool water at cooling capacity of 7.14 kw

Table 6.5: Compressor results

Parameter's	*Inlet state*		*Exit state*
Temperature (°C)	-4		46.3
Pressure (KPa)	252.7		1200
Enthalpy (kJ/kg)	248.11		281.54
Entropy ($kJ/kg.K$)	0.93		0.93
pressure ratio	4.75		
Coefficient of performance	3.89		
Work of the compressor($^{kJ}/_{kg}$)	33.432		
Time require to cool water	30 minutes		300 minutes
Cooling capacity(kw)	7.14		0.714
Mass flow rate of refrigerant ($^{kg}/_{s}$)	0.05		0.005
Power of compressor	*1.83 KW=5/2 hp*		0.183 kw=1/4 hp

Table 6.6: Evaporator results

Parameter's		Inlet State	Exit State
Temperature (°C)		-4	-4
Pressure (kPa)		252.7	252.7
Enthalpy (kJ/kg)		117.8	248.11
Entropy (kJ/kgK)		0.21	0.93
Time require to cool water		30 minutes	300 minutes
Diameter(mm)	outer	15.875	6.35
	inner	14.63	5.639

Length of evaporator(m)	50.88	14.87

Table 6.7: Condenser results

Parameter's		Inlet State	Exit State
Temperature (°C)		52.5	46.31
Pressure (KPa)		1200	1200
Enthalpy (kJ/kg)		248.11	117.8
Entropy (kJ/kgK)		0.93	0.42
Time require to cool water		30 minutes	300 minutes
Diameter(mm)	outer	9.525	6.35
	inner	8.28	5.639
Length of condenser(m)		15.33	6.33

Table 6.8: Capillary tube segment results at mass flow rate of 0.005 kg/s

Segment:	1	2	3	4	5	6
$T1$ (°C)	46.3	37.6	29.2	20.85	12.46	4.08
$T2$ (°C)	37.6	29.2	20.85	12.46	4.08	-4.0
$x1$	0.09	0.17	0.24	0.30	0.36	0.42
$x2$	0.17	0.24	0.30	0.36	0.42	0.47
Re	7133.70	6895.30	6659.40	6420.80	6218.10	6012.20
$V1$ (m/s)	7.40	14.78	26.03	43.46	71.02	118.31
$V2$ (m/s)	14.47	26.09	43.50	71.02	118.42	197.38
f	0.04	0.04	0.04	0.04	0.04	0.04

ΔL (m)	0.96	0.76	0.62	0.41	0.29	0.09

Table 6.9: Capillary tube results

Parameter's		Inlet state	Exit state
Temperature (°C)		46.3	-4
Pressure (kPa)		1200	252.7
Enthalpy (kJ/kg)		117.8	117.8
Entropy (kJ/kgK)		0.18	0.21
Time require to cool water		30 minutes	300 minutes
Diameter(mm)	outer	1.0	0.8
	inner	0.9	0.75
Length of evaporator(m)		3.82	3.13

Table 6.10: Results of all theoretical calculated values

No.	**Name parameters**	**Results**		**Unit**
		Time to cool		
		30 minute	**300 minutes**	
1	Total cooling load	12847.88	12847.88	KJ
2	Cooling capacity	2.028	0.203	$tons$
3	Mass flow rate	0.055	0.005	Kg/s
4	Evaporator pressure	252.7	252.7	kpa
5	Condenser pressure	1200	1200	kpa
6	pressure ratio	4.75	4.75	

7	Power of compressor	2.45	0. 245	Hp
8	Coefficient of performance	3.90	3.90	
9	Length of evaporator	50.89	14.87	M
10	Length of condenser	15.33	6.33	M
11	Length of capillary tube	3.82	3.13	M

6.2 Parameters affect performance of public water cooler

The study and analysis parameters in this topic is for cooling capacity 0.71 KW because of it design use for fabrication of water cooler.

6.2.1 Effect of evaporator temperature

The effect of evaporator temperature on performance of the system is obtained by keeping the condenser temperature (pressure) fixed.

For a given condensing temperature (or pressure), the pressure ratio r_p increases as the evaporator temperature (or evaporator pressure) decreases.

i. On refrigerant mass flow rate

The mass flow rate of refrigerant m is given by:

$$\dot{m} = \frac{\text{Cooling capacity } (Q_c)}{(h_1 - h_4)}$$

As the evaporator temperature decreases the enthalpy difference decreases, as a result of these effect, the mass flow rate of refrigerant through the compressor decreases rapidly as the evaporator temperature decrease as shown in Figure. 6.8.

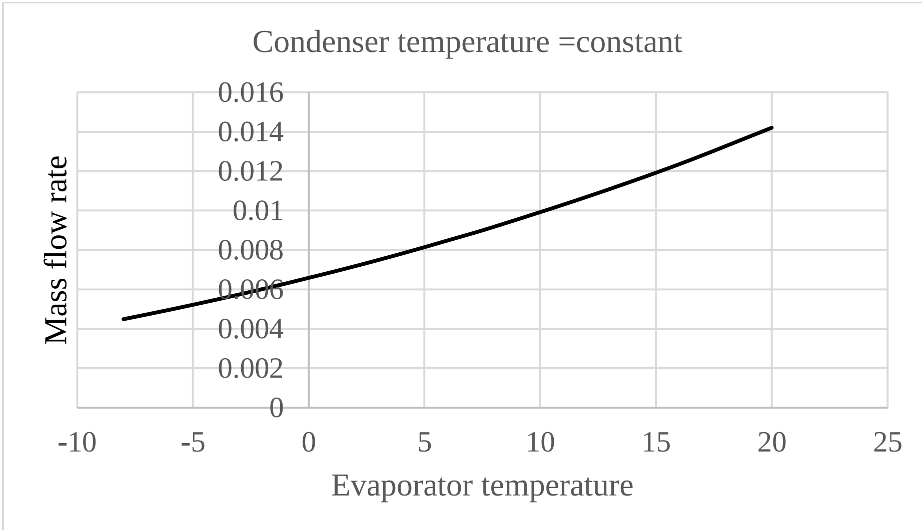


Figure 6.8: Effect of evaporator temperature on refrigerant mass flow rate

ii. **On refrigeration effect and refrigeration capacity:**

It can be seen from the figure that the refrigeration effect, (qe= h1– h4) increases marginally as the evaporator temperature is increased. This is due to the shape of the saturation vapour curve on p-h diagram. The effect of evaporator temperature on refrigerant effect is also shown in

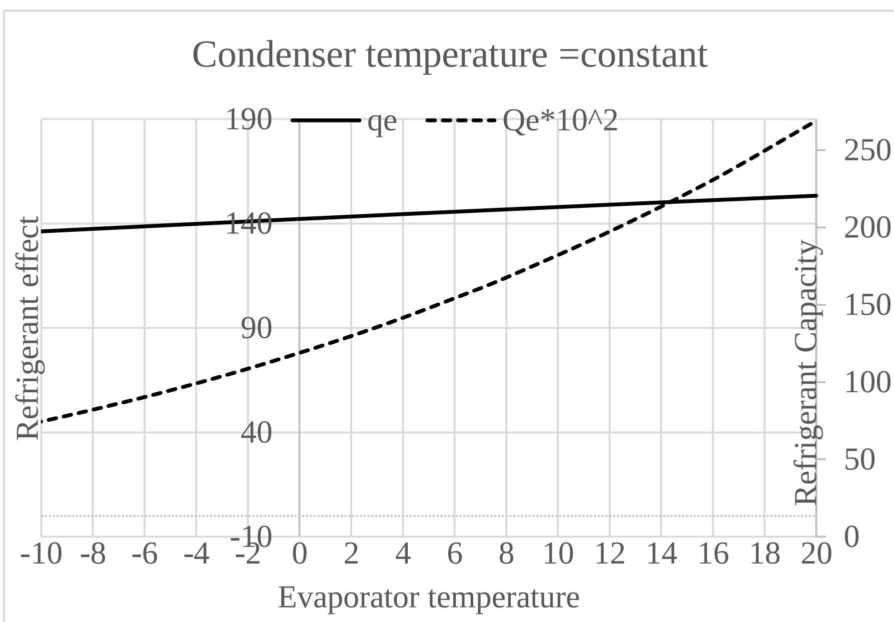


Figure 6.9: Effect of evaporator temperature on refrigeration effect and refrigeration capacity

Since mass flow rate of refrigerant increases rapidly and refrigerant effect also increases, though marginally with increase in evaporator temperature, the refrigeration capacity increases sharply with increase in evaporator temperature as shown in Figure 6.9

iii. **On work of compression and power requirement:**

At a constant condenser temperature as evaporator temperature increases the work of compression, Δh (= h2– h1) decreases as shown in Figure 6.10. This is due to the divergent nature of isentropic in the superheated region.

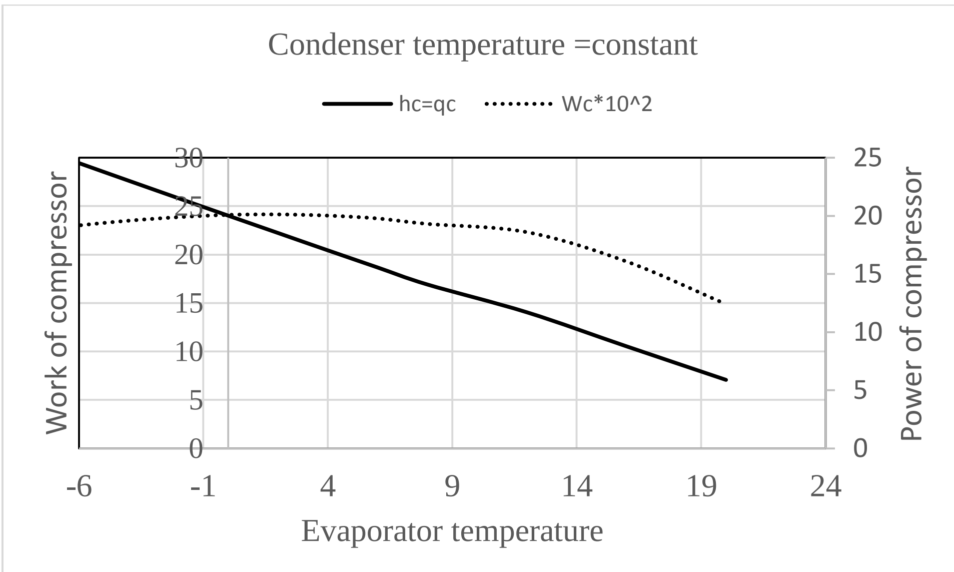


Figure 6.10: Effect of evaporator temperature on work of compression and power input to compressor

iv. On COP and volume flow rate per unit capacity:

The COP of the system is defined as:

$$COP = \frac{\text{qe}}{\Delta\text{hc}} = \frac{h_1 - h_4}{h_2 - h_1}$$

As discussed before, as the evaporator temperature increases the refrigeration effect, qe increases marginally and the work of compression, Δhc reduces sharply. As a result, the COP of the system increases rapidly as the evaporator temperature increases as shown in Figure. 6.11.

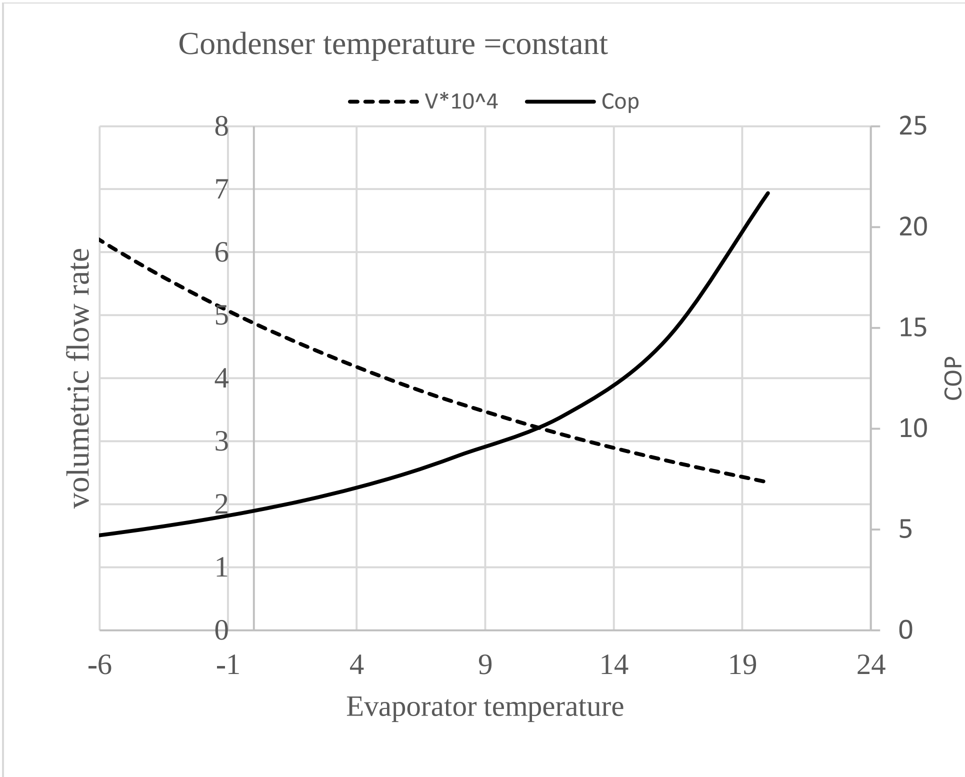


Figure 6.11: Effect of evaporator temperature on COP per unit capacity

As evaporator temperature increases the specific volume of the refrigerant at compressor inlet reduces rapidly and the refrigerant effect increases marginally.

Due to the combined effect of these two, the volume flow rate of refrigerant per unit capacity reduces sharply with evaporator temperature as shown in Figure.6.11

This implies that for a given refrigeration capacity, the required volumetric flow rate and hence the size of the compressor becomes very large at very low evaporator temperatures.

6.2.2 Effect of condenser temperature

Atmospheric air is the cooling medium for most of the refrigeration systems. Since the ambient temperature at a location can vary over a wide range, the heat rejection temperature (i.e., the condensing temperature) may also vary widely. This affects the performance of the compressor and hence the refrigeration system. The effect of condensing temperature on compressor performance can be studied by keeping evaporator temperature constant.

a. On refrigerant mass flow rate

At a constant evaporator temperature as the condensing temperature increases, the pressure ratio increases, hence, mass flow rate decrease. However, the effect of condensing temperature on mass flow rate is not as significant as the evaporator temperature as the specific volume of refrigerant at compressor inlet is independent of condensing temperature as shown in Figure 6.12.

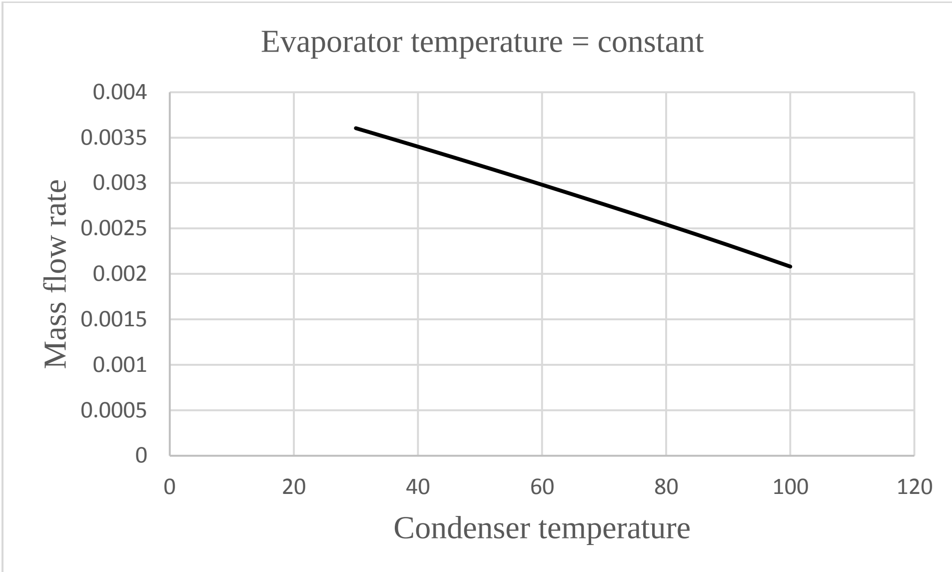


Figure 6.12: : Effect of condenser temperature on refrigerant mass flow rate

b. Refrigeration effect and refrigeration capacity:

At a constant evaporator temperature as the condensing temperature increases, then the enthalpy of refrigerant at the inlet to the evaporator increases. Since the evaporator enthalpy remains constant at a constant evaporator temperature, the refrigeration effect decreases with increase in condensing temperature as shown in Figure. 6.13.The refrigeration capacity (Q_e) also reduces with increase in condensing temperature as both the mass flow rate and refrigeration effect decrease as shown in Figure. 6.13.

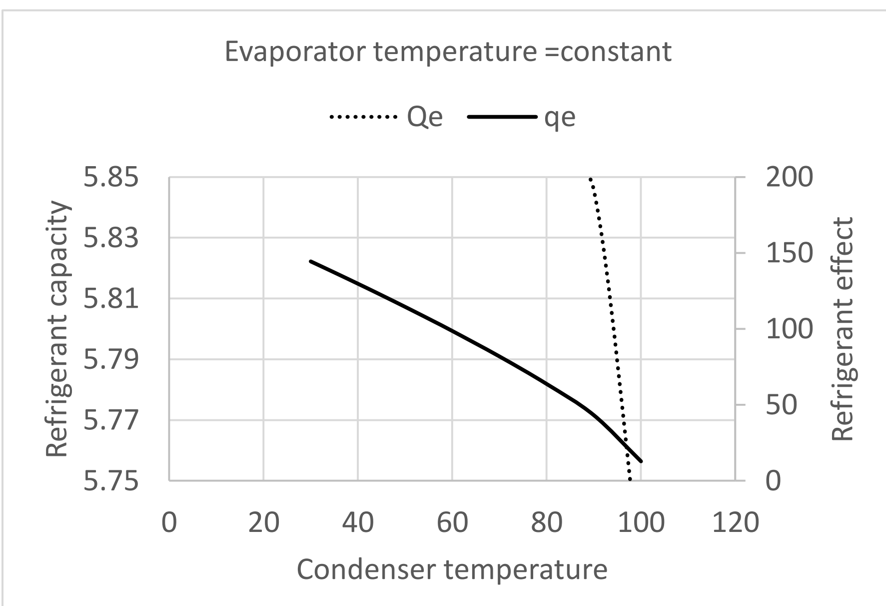


Figure 6.13: Effect of condenser temperature on refrigeration effect and refrigeration capacity

c. Work of compression and power requirement:

The work of compression is zero when the condenser temperature is equal to the evaporator temperature and at a limiting condensing temperature the mass flow rate of refrigerant becomes zero.

Hence, similar to the effect of evaporator temperature on power curve, the compressor power input increases from zero (work of compression is zero), reaches a peak and then again becomes zero at a high value of condensing temperature as shown in Figure.6.14.

However, the peak power in this case is not as critical as with evaporator temperature since the chances of condenser operating at such a high temperature are rare.

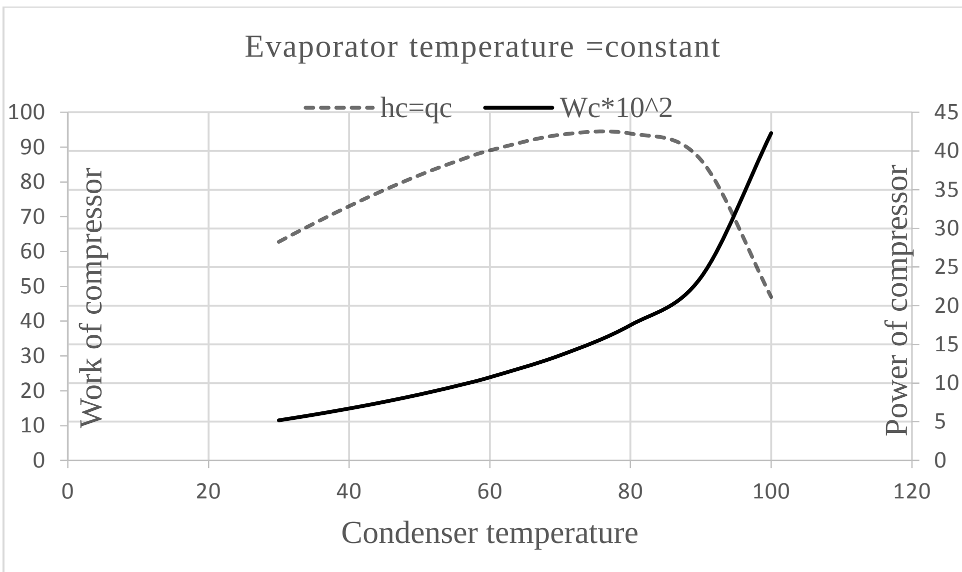


Figure 6.14: Effect of condenser temperature on work of compression and power input to compressor

d. COP and volume flow rate per unit capacity:

As condensing temperature increases the refrigeration effect reduces marginally and work of compression increases, as a result the COP reduces as shown in figure. 6.15. Even though the specific volume at compressor inlet is independent of condensing temperature, since the refrigeration effect decreases with increase in condensing temperature, the volume flow rate of refrigerant per unit capacity increases as condenser temperature increases as shown in Figure. 6.15.

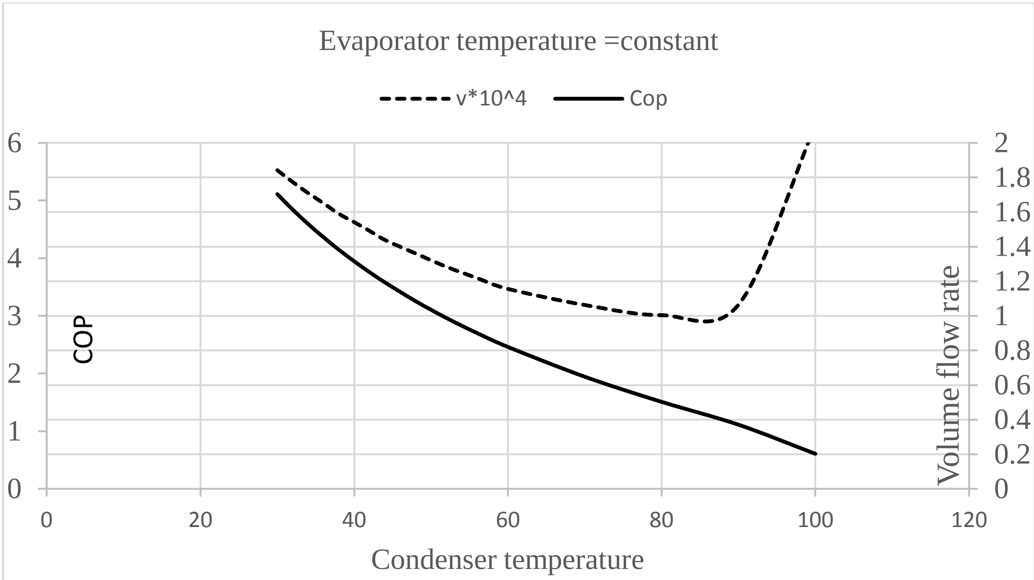


Figure 6.15: Effect of condenser temperature on COP and volume flow rate per unit capacity

The above discussion shows that the performance of the system degrades as the evaporator temperature decreases and condensing temperature increases, i.e., the temperature lift increases.

It is seen that compared to the condensing temperature, the effect of evaporator temperature is quiet significant. When the heat sink temperature does not vary too much then the effect of condensing temperature may not be significant.

6.3 Experimental results

The temperature and pressure measured from water cooler given below table

Table 6.11: Experimental results of temperature and pressure

	Compressor		Condenser		Capillary tube		Evaporator	
	Inlet state	Exit state	Inlet state	Exit state	Inlet state	Exit state	Inlet state	Exit state
Temperature (°C)	2.8	65.4	60.3	48.2	44.1	-3.6	-3.5	-2
Pressure (kPa)	326.7	1899.3	1681.8	1252.9	1130.1	256.8	257.575	272.2

Properties of refrigerant R134a read from saturated and superheated properties tables at experimentally measured temperature

Table 6.12: Experimental results of enthalpy and entropy

	Compressor		Condenser		Capillary tube		Evaporator	
	Inlet state	Exit state	Inlet state	Exit state	Inlet state	Exit state	Inlet state	Exit state
Temperature (°C)	2.8	65.4	60.3	48.2	44.1	-3.6	-3.5	-2
Pressure (kPa)	326.7	1899.3	1681.8	1252.9	1130.1	256.8	257.6	272.2
Enthalpy (kJ/kg)	251.98	293.33	284.20	120.39	114.28	118.2	118.9	249.29
Entropy (kJ/kgK)	0.93	0.90	0.91	0.43	0.41	0.45	0.45	0.93

6.4 Comparisons of theoretical and experimental results

Table 6.13 shows the comparison between the experimental and theoretical data. There is a slight variation between the theoretical and the experimental data is due to heat loss, which at present it is almost impossible to prevent since there is no perfect insulation of heat and frictional pressure drops in connecting lines.

Table 6.13: Comparison of theoretical and experimental results

No.	Name of parameters	Results				Unit
		Theoretical		Experimental	Percentage error(%)	
Time require to cool water		**30 min**	**300 min**	**300 min**		
1	Evaporator pressure	252.7	252.7	257	1.7	kPa
2	Condenser pressure	1200	1200	1252.9	4.2	kPa
3	Mass flow rate	0.055	0.0055	-------	------	Kg/s
4	Pressure ratio	4.75	4.75	4.88	2.7	
5	Power of compressor	2.45	0.25	0.25	1.2	Hp
6	Coefficient of performance	3.89	3.89	3.17	18.5	
7	Length of evaporator	50.88	14.9	15.24	2.2	M
8	Length of condenser	15.3	6.3	6.75	6.7	M
9	Length of capillary tube	3.8	3.1	3.6	13.9	M

Coefficient of performance of experimental result is less than that of theoretical result, because of condenser temperature of experimental is much larger than that of theoretical, due to heat transfer only in natural convection. Atmospheric air is the cooling medium for this refrigeration systems. Since the ambient temperature at a location can vary over a wide range, the heat rejection temperature (i.e., the condensing temperature) may also vary widely. In hermetic sealed compressors additional heat transfer from the motor winding to refrigerant takes place. The effect of this transfer is to increase the temperature of refrigerant, thereby increasing the specific volume. So, rejection in the condenser. Figure 6.16: shows the comparison between the experimental and theoretical performance of the system.

The thermostat stops the compressor when the temperature of water reach 10°C. As the temperature of water rises up due to additional hot water add to tanker or external heat transfer to water from surrounding the thermostat on the compressor and start removal heat until temperature of water reaches 10°C.

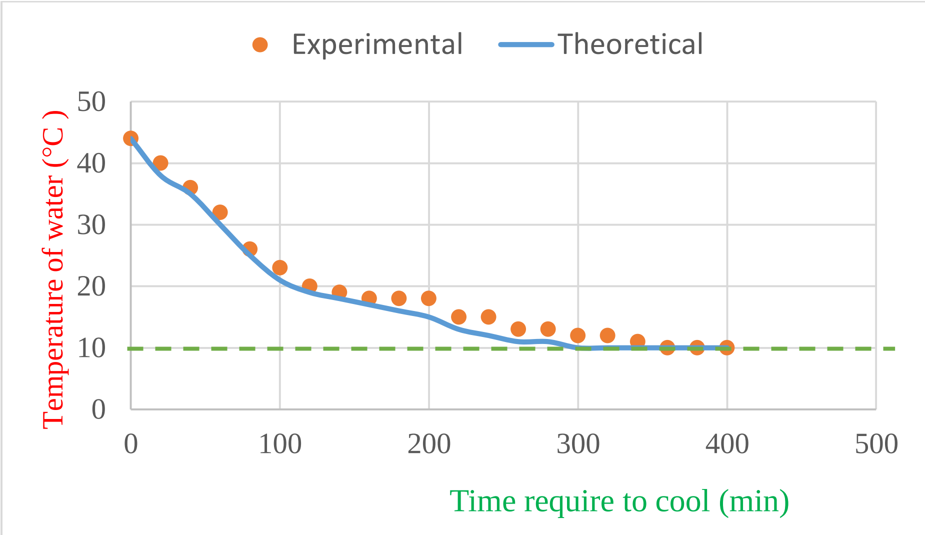


Figure 6.16: Theoretical and experimental water temperature versus time

We can be observed that in Figure. 6.16 that the theoretical and the experimental data followed the same trend. The different may be associated with heat leakage. It is interesting to note that the theoretical predicted almost a linear relationship for the COP with the experimental results. The theoretical model reaches steady state much faster than the experiment. This difference between the theoretical and experimental results might be attributed to uncertainties within the theoretical parameters. The thermal resistances between the aluminum and the water were measured based off of other experiments. However, these measurements have uncertainties, which meant that the resistances may not be exact. Additionally, the properties of the system may change as time lapses and temperatures change. Properties of the materials were assumed to be constant but may change with temperature. Water movement also was not taken into account, but water may have been moving due to vibrations of the compressor on the system and induced water movement as the temperature of the water decreased. Water movement, for example, would increase the heat transfer coefficient between the water and the cold side of the aluminum, reducing the thermal resistance. This would increase the amount of heat absorbed and decrease the temperature of the water.

Generally, actual compression processes (experimental) deviate from ideal compression processes (theoretical) due to heat transfer between the refrigerant and surroundings during compression and expansion, which these processes non-adiabatic, frictional pressure drops in connecting lines and across suction and discharge valves and losses due to leakage.

6.5 Comparison of present study with literature

To decide where to put the present study in the literature and to compare the results, Table 6.14 may help.

Table 6.14: Comparison of present study with literature

Parameters	**Chow et al. [40]**	**Heydari [41]**	**Phelan et al. [43]**	**Seyfettin [42]**	**Present Study**		
					Theoretical		**Experimental**
					Time to cool		
					30min	**300min**	**300min**
Heat load(w)	32	-	100-300	45	7137.70	713.77	713.77
T_e(°C)	12	20	5	0	-4	-4	-3.5
T_c (°C)	-	60	55	50	46.31	46.31	48.2
$P_e(kpa)$	443	571	349.85	292.8	252.7	252.7	257
$P_c(kpa)$	-	1681.8	1495.6	1317.7	1200	1200	1252.9
$T_{amb}(°C)$	45	36	33	30	28	28	28
Flow rate (g/s)	16.3	-	0.8- 2.5	0.35	54.78	5.48	------
Refrigerant	R134a	R134a	R134a	R134a	R134a	R134a	R134a
COP	3.34	3.0	3.0	3.89	3.90	3.90	3.17
Compressor type	Centrifugal	Piston	Scroll	Piston	Hermetic sealed	Hermetic sealed	Hermetic sealed

As shown in the table 6.14, in the present study, the COP for both theoretical models were 3.90 and for experimental is 3.17 and in literature COP for Chow et al. [40] is 3.34, Heydari [41] is 3.0, Phelan et al. [43] 3.0 and Seyfettin [42] is 3.89. Since the study of Phelan et al. [43] is an experimental study. So, the present study has better performance than literature for both theoretical and experimental aspect. However, it should be noted that Therefore, the present study may be thought as a milestone in the literature where a storage type public water cooler with all of the components is designed theoretically.

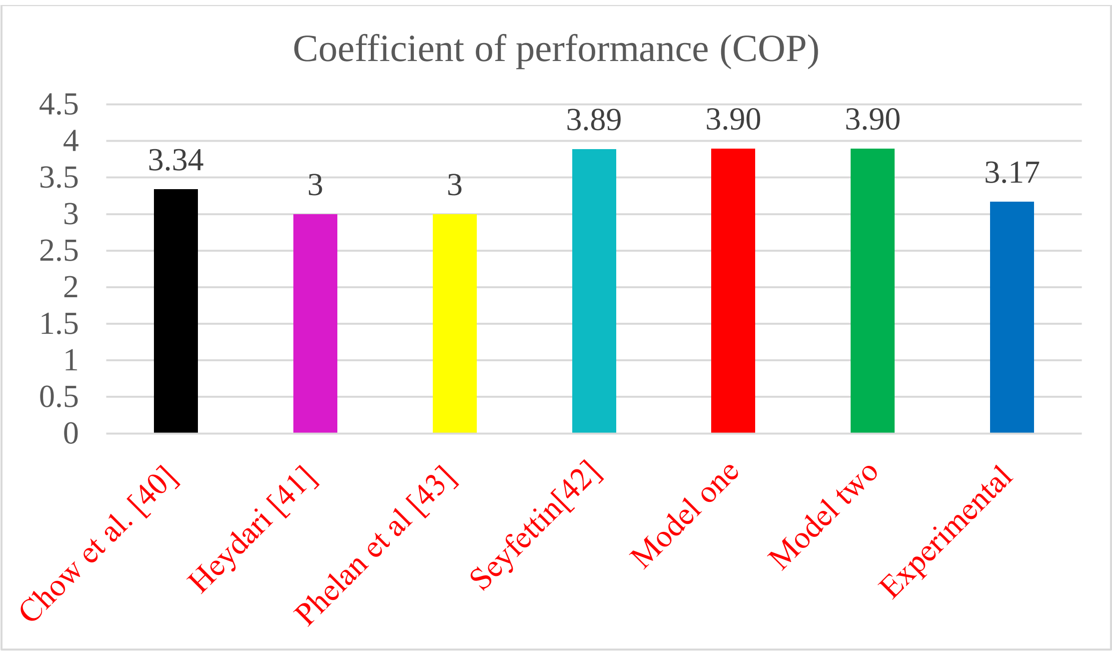


Figure 6.17: Comparisons of present study with literature in COP values

CHAPTER SEVEN

7 Conclusion and Recommendations for Future Work

7.1 Conclusion

The overall performance of water cooler is strongly influenced by the agreement between the major components, those are: condenser, compressor, evaporator and capillary tube, and the operational parameters, such as ambient temperatures, condensing temperatures, evaporating temperatures and mass flow rate. These parameters were used in this study to validate a design model developed. The maximum cooling load used for the design is 12847.87 KJ with 10% allowance. In present study two models are developed. For the first model water cooler takes 30 minutes to cool water from 44 °C to 10 °C, at cooling capacity of 7.14 KW and 300 minutes take at cooling capacity of 0.71 KW for the second model. The second model is developed from first model by scale down cooling capacity by ten for comparison of theoretical with experimental results. The power of compressor requires for first and second model is 2.46 *hp* and 0.245 *hp* respectively. In this experimental performance evaluation of vapor compression system, the performance tests are conducted for refrigerants R134a. Test results were used to assess the quality of the model by comparing the experimental results with the model predicted(theoretical) results. Tests under a wide range operating temperature interval conditions were carried out on the developed system. Experimental data was generated to observe the performance of the basic VCR system for an important parameter such as condensation (45 to 57 °C) and evaporation temperatures (-4 to 9 °C). Performance evaluation of the system was characterized in terms of cooling capacity and coefficient of performance (COP). The COP values of the theoretical results were 3.89 and the experimental results are 3.17. This shows some variation is observed because of the ideal assumption of the vapour compression. As show in the result and discussion (figure 6.16) the time require to cool water are follow the same trend and almost the same as theoretical time to cool at cooling capacity of 0.71 KW. By comparing present study with other vapor compressor refrigeration system in table 6.14 they observed that this water cooler is having excellent performance with good cooling of water and also have better efficiency. The temperature of outlet water obtain is in the range of 7 ℃ to 13 ℃. In conclusion, a water cooler has been successfully designed and analyzed within the scope of this thesis. The theoretical and experimental results of locally fabricated public water cooler actually indicate that the objective of the study is achieved.

7.2 Recommendations for future work

There are a number of areas in which further work should be carried out. The cooler could be modified as both cold and hot water dispenser and the exhaust heat from the cooler (heat is rejected by condenser) could be utilize for heating purpose, but this idea is not included in this study thus anyone who is interested to design water cooler in similar title it's better to add this idea to their thesis. The power source for water cooler in this thesis is electric energy. But in some rural area it cannot be possible to have an electric power so, it recommended that for whom interested on this topic, he/she could be design public water cooler by using solar energy. In this paper some components are used by selection standardize product but it is better to be designed and analyzed. It also recommended effect of radiation in the condenser and the evaporator may be taken into account during the design.

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APENDEX A

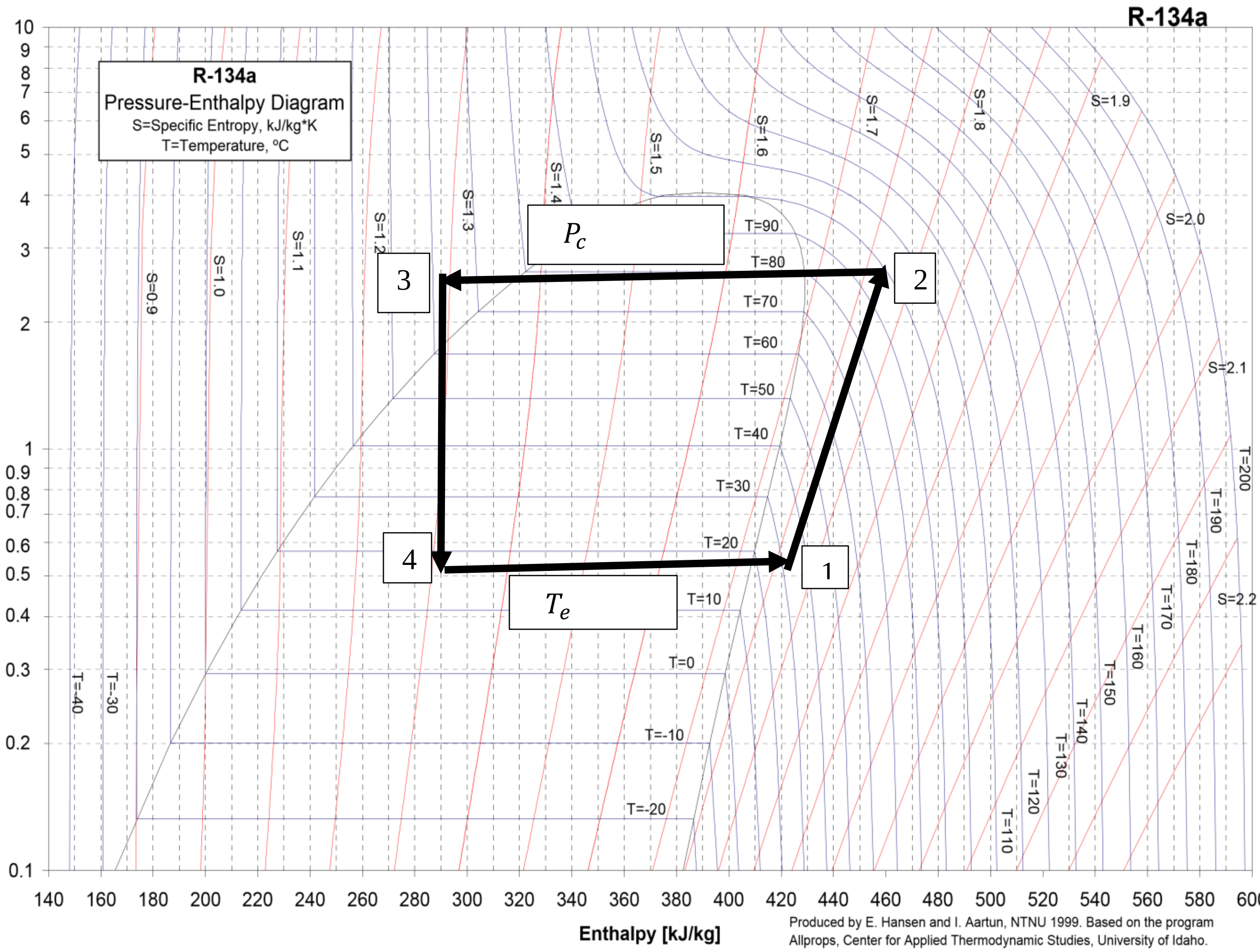

Figure A.1: Pressure vs. enthalpy curve of R-134A

APPENDIX B

Table B.1: Saturation properties of R134a temperature table[22]

Temp	Pressure	volume(m^3/kg)		enthalpy (kJ/kg)			entropy (kJ/kg.K)		
deg C	kPa	vf	vg	hf	hfg	hg	sf	sfg	sg
-40	51.2	0.0007054	0.3611	0.00	225.86	225.86	0.0000	0.9687	0.9687
-36	62.9	0.0007112	0.2977	5.04	223.35	228.39	0.0214	0.9418	0.9632
-26	101.7	0.0007265	0.1896	17.76	216.92	234.68	0.0738	0.8777	0.9515
-24	111.3	0.0007297	0.1741	20.33	215.60	235.93	0.0841	0.8653	0.9495
-22	121.7	0.0007329	0.1601	22.91	214.26	237.17	0.0944	0.8531	0.9476
-20	132.7	0.0007362	0.1474	25.49	212.92	238.41	0.1046	0.8411	0.9457
-18	144.6	0.0007396	0.1359	28.09	211.55	239.64	0.1148	0.8292	0.9440
-12	185.2	0.0007499	0.1074	35.92	207.39	243.31	0.1451	0.7941	0.9392
-10	200.6	0.0007535	0.0996	38.55	205.97	244.52	0.1550	0.7827	0.9377
-8	216.9	0.0007571	0.0924	41.19	204.53	245.72	0.1650	0.7714	0.9364
-6	234.3	0.0007608	0.0859	43.84	203.08	246.92	0.1749	0.7602	0.9351
-4	252.7	0.0007646	0.0799	46.50	201.61	248.11	0.1848	0.7490	0.9338
-2	272.2	0.0007684	0.0744	49.17	200.12	249.29	0.1946	0.7380	0.9326
0	292.8	0.0007723	0.0693	51.86	198.60	250.46	0.2044	0.7271	0.9315
2	314.6	0.0007763	0.0647	54.55	197.07	251.62	0.2142	0.7162	0.9304
4	337.7	0.0007804	0.0604	57.25	195.53	252.78	0.2239	0.7055	0.9294
6	362.0	0.0007845	0.0564	59.97	193.95	253.92	0.2336	0.6948	0.9284
8	387.6	0.0007887	0.0528	62.69	192.36	255.05	0.2432	0.6842	0.9274
12	443.0	0.0007975	0.0463	68.19	189.11	257.29	0.2625	0.6632	0.9256
16	504.3	0.0008066	0.0408	73.73	185.74	259.47	0.2816	0.6424	0.9240
20	571.7	0.0008161	0.0360	79.32	182.28	261.60	0.3006	0.6218	0.9224
24	645.8	0.0008261	0.0319	84.98	178.70	263.68	0.3196	0.6014	0.9210
28	726.9	0.0008367	0.0283	90.70	175.00	265.69	0.3385	0.5811	0.9196
30	770.2	0.0008421	0.0266	93.58	173.09	266.67	0.3479	0.5710	0.9189
36	911.9	0.0008595	0.0224	102.33	167.17	269.50	0.3761	0.5407	0.9168
38	963.2	0.0008657	0.0211	105.29	165.12	270.41	0.3855	0.5307	0.9162
40	1016.6	0.0008720	0.0200	108.27	163.01	271.28	0.3949	0.5206	0.9155
42	1072.2	0.0008786	0.0189	111.26	160.88	272.14	0.4043	0.5105	0.9147
44	1130.1	0.0008854	0.0178	114.28	158.69	272.97	0.4136	0.5004	0.9140
48	1252.9	0.0008997	0.0160	120.39	154.16	274.55	0.4324	0.4800	0.9125
52	1385.4	0.0009150	0.0143	126.60	149.41	276.01	0.4513	0.4595	0.9108
56	1528.2	0.0009317	0.0128	132.92	144.40	277.32	0.4702	0.4387	0.9089
60	1681.8	0.0009498	0.0114	139.36	139.13	278.49	0.4892	0.4176	0.9068
70	2116.8	0.0010038	0.0087	156.14	124.37	280.51	0.5376	0.3624	0.9000
80	2633.2	0.0010773	0.0064	174.25	106.42	280.67	0.5880	0.3014	0.8894
90	3244.2	0.0011936	0.0046	194.78	82.49	277.27	0.6434	0.2272	0.8706
100	3972.4	0.0015357	0.0027	225.15	34.39	259.54	0.7232	0.0921	0.8153

APPENDIX C

Table C.1: Saturation properties of R134a pressure table [22]

Pressure kPa	Temp deg C	volume (m^3/kg) vf	vg	Energy(kJ/kg) uf	ug	enthalpy (kJ/kg) hf	hfg	hg	entropy (kJ/kg.K) sf	sfg	sg
60	-36.9	0.0007098	0.3112	3.8	209.1	3.9	223.9	227.8	0.0164	0.9481	0.9645
80	-31.1	0.0007185	0.2376	11.2	212.5	11.2	220.2	231.5	0.0472	0.9100	0.9572
100	-26.4	0.0007259	0.1926	17.2	215.2	17.3	217.2	234.5	0.0720	0.8799	0.9519
120	-22.3	0.0007324	0.1621	22.4	217.5	22.5	214.5	237.0	0.0928	0.8550	0.9478
140	-18.8	0.0007383	0.1402	27.0	219.6	27.1	212.1	239.2	0.1110	0.8337	0.9446
160	-15.6	0.0007437	0.1235	31.1	221.4	31.2	209.9	241.1	0.1270	0.8150	0.9420
180	-12.7	0.0007487	0.1104	34.9	223.0	35.0	207.9	242.9	0.1415	0.7982	0.9397
200	-10.1	0.0007534	0.0999	38.3	224.5	38.5	206.0	244.5	0.1547	0.7831	0.9378
220	-7.6	0.0007578	0.0912	41.5	225.9	41.7	204.3	245.9	0.1668	0.7693	0.9361
240	-5.4	0.0007620	0.0839	44.5	227.2	44.7	202.6	247.3	0.1780	0.7566	0.9347
260	-3.2	0.0007661	0.0777	47.3	228.4	47.5	201.0	248.6	0.1885	0.7448	0.9333
280	-1.2	0.0007699	0.0724	50.0	229.5	50.2	199.5	249.7	0.1984	0.7338	0.9322
300	0.7	0.0007737	0.0677	52.5	230.5	52.8	198.1	250.9	0.2077	0.7234	0.9311
320	2.5	0.0007773	0.0636	54.9	231.5	55.2	196.7	251.9	0.2165	0.7137	0.9301
340	4.2	0.0007808	0.0600	57.3	232.5	57.5	195.4	252.9	0.2248	0.7044	0.9293
360	5.8	0.0007842	0.0567	59.5	233.4	59.8	194.1	253.8	0.2328	0.6956	0.9284
400	8.9	0.0007907	0.0512	63.7	235.1	64.0	191.6	255.6	0.2477	0.6793	0.9270
500	15.7	0.0008060	0.0411	73.0	238.8	73.4	186.0	259.3	0.2803	0.6438	0.9241
600	21.6	0.0008200	0.0343	81.0	241.9	81.5	180.9	262.4	0.3081	0.6138	0.9219
700	26.7	0.0008332	0.0294	88.3	244.5	88.8	176.2	265.1	0.3324	0.5876	0.9200
800	31.3	0.0008459	0.0256	94.8	246.8	95.5	171.8	267.3	0.3541	0.5643	0.9184
900	35.5	0.0008581	0.0227	100.9	248.9	101.6	167.7	269.3	0.3739	0.5431	0.9170
1000	39.4	0.0008701	0.0203	106.5	250.7	107.4	163.7	271.0	0.3920	0.5237	0.9157
1200	46.3	0.0008935	0.0167	116.7	253.8	117.8	156.1	273.9	0.4245	0.4886	0.9131
1400	52.4	0.0009167	0.0141	126.0	256.4	127.3	148.9	276.2	0.4533	0.4573	0.9106
1600	57.9	0.0009401	0.0121	134.5	258.5	136.0	141.9	277.9	0.4792	0.4287	0.9080
1800	62.9	0.0009640	0.0106	142.4	260.2	144.1	135.1	279.2	0.5031	0.4020	0.9051
2000	67.5	0.0009888	0.0093	149.8	261.6	151.8	128.3	280.1	0.5252	0.3767	0.9020
2500	77.6	0.0010569	0.0069	167.1	263.5	169.7	111.2	280.9	0.5755	0.3170	0.8925
3000	86.2	0.0011413	0.0053	183.1	263.4	186.6	92.6	279.2	0.6215	0.2578	0.8792

Table C.2: Properties of refrigerant R134a [22]

Temp,* °C	Absolute Pressure, MPa	Density, kg/m³	Volume, m³/kg	Enthalpy, kJ/kg		Entropy, kJ/(kg·K)		Specific Heat c_p, kJ/(kg·K)		c_p/c_v	Velocity of Sound, m/s		Viscosity, µPa·s		Thermal Cond, mW/(m·K)		Surface Tension, mN/m	Temp,* °C
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor		
−103.30a	0.00039	1591.2	35.263	71.89	335.07	0.4143	1.9638	1.147	0.585	1.163	1135.	127.	2186.6	6.63	—	—	28.15	−103.30
−100.00	0.00056	1581.9	25.039	75.71	337.00	0.4366	1.9456	1.168	0.592	1.161	1111.	128.	1958.2	6.76	—	—	27.56	−100.00
−90.00	0.00153	1553.9	9.7191	87.59	342.94	0.5032	1.8975	1.201	0.614	1.155	1051.	131.	1445.6	7.16	—	—	25.81	−90.00
−80.00	0.00369	1526.2	4.2504	99.65	349.03	0.5674	1.8585	1.211	0.637	1.151	999.	134.	1109.9	7.57	—	—	24.11	−80.00
−70.00	0.00801	1498.6	2.0528	111.78	355.23	0.6286	1.8269	1.215	0.660	1.148	951.	137.	879.6	7.97	125.8	—	22.44	−70.00
−60.00	0.01594	1471.0	1.0770	123.96	361.51	0.6871	1.8016	1.220	0.685	1.146	904.	139.	715.4	8.38	121.1	—	20.81	−60.00
−50.00	0.02948	1443.1	0.60560	136.21	367.83	0.7432	1.7812	1.229	0.712	1.146	858.	142.	594.3	8.79	116.5	7.12	19.22	−50.00
−40.00	0.05122	1414.8	0.36095	148.57	374.16	0.7973	1.7649	1.243	0.740	1.148	812	144.	502.2	9.20	111.9	8.19	17.66	−40.00
−30.00	0.08436	1385.9	0.22596	161.10	380.45	0.8498	1.7519	1.260	0.771	1.152	765.	145.	430.4	9.62	107.3	9.16	16.13	−30.00
−28.00	0.09268	1380.0	0.20682	163.62	381.70	0.8601	1.7497	1.264	0.778	1.153	756.	145.	418.0	9.71	106.3	9.35	15.83	−28.00
−26.07b	0.10132	1374.3	0.19016	166.07	382.90	0.8701	1.7476	1.268	0.784	1.154	747.	146.	406.4	9.79	105.4	9.52	15.54	−26.07
−26.00	0.10164	1374.1	0.18961	166.16	382.94	0.8704	1.7476	1.268	0.785	1.154	747.	146.	406.0	9.79	105.4	9.53	15.53	−26.00
−24.00	0.11127	1368.2	0.17410	168.70	384.19	0.8806	1.7455	1.273	0.791	1.155	738.	146.	394.6	9.88	104.5	9.71	15.23	−24.00
−22.00	0.12160	1362.2	0.16010	171.26	385.43	0.8908	1.7436	1.277	0.798	1.156	728.	146.	383.6	9.96	103.6	9.89	14.93	−22.00
−20.00	0.13268	1356.2	0.14744	173.82	386.66	0.9009	1.7417	1.282	0.805	1.157	719.	146.	373.1	10.05	102.6	10.07	14.63	−20.00
−18.00	0.14454	1350.2	0.13597	176.39	387.89	0.9110	1.7399	1.286	0.812	1.159	710.	146.	363.0	10.14	101.7	10.24	14.33	−18.00
−16.00	0.15721	1344.1	0.12556	178.97	389.11	0.9211	1.7383	1.291	0.820	1.160	700.	147.	353.3	10.22	100.8	10.42	14.04	−16.00
−14.00	0.17074	1338.0	0.11610	181.56	390.33	0.9311	1.7367	1.296	0.827	1.162	691.	147.	344.0	10.31	99.9	10.59	13.74	−14.00
−12.00	0.18516	1331.8	0.10749	184.16	391.55	0.9410	1.7351	1.301	0.835	1.164	682	147.	335.0	10.40	99.0	10.76	13.45	−12.00
−10.00	0.20052	1325.6	0.09963	186.78	392.75	0.9509	1.7337	1.306	0.842	1.166	672	147.	326.3	10.49	98.0	10.93	13.16	−10.00
−8.00	0.21684	1319.3	0.09246	189.40	393.95	0.9608	1.7323	1.312	0.850	1.168	663.	147.	318.0	10.58	97.1	11.10	12.87	−8.00
−6.00	0.23418	1313.0	0.08591	192.03	395.15	0.9707	1.7310	1.317	0.858	1.170	654.	147.	309.9	10.67	96.2	11.28	12.58	−6.00
−4.00	0.25257	1306.6	0.07991	194.68	396.33	0.9805	1.7297	1.323	0.866	1.172	644.	147.	302.2	10.76	95.3	11.45	12.29	−4.00
−2.00	0.27206	1300.2	0.07440	197.33	397.51	0.9903	1.7285	1.329	0.875	1.175	635.	147.	294.7	10.85	94.3	11.62	12.00	−2.00
0.00	0.29269	1293.7	0.06935	200.00	398.68	1.0000	1.7274	1.335	0.883	1.178	626.	147.	287.4	10.94	93.4	11.79	11.71	0.00
2.00	0.31450	1287.1	0.06470	202.68	399.84	1.0097	1.7263	1.341	0.892	1.180	616.	147.	280.4	11.03	92.5	11.96	11.43	2.00
4.00	0.33755	1280.5	0.06042	205.37	401.00	1.0194	1.7252	1.347	0.901	1.183	607.	147.	273.6	11.13	91.6	12.13	11.14	4.00
6.00	0.36186	1273.8	0.05648	208.08	402.14	1.0291	1.7242	1.353	0.910	1.187	598.	147.	267.0	11.22	90.7	12.31	10.86	6.00
8.00	0.38749	1267.0	0.05284	210.80	403.27	1.0387	1.7233	1.360	0.920	1.190	588.	147.	260.6	11.32	89.7	12.48	10.58	8.00
10.00	0.41449	1260.2	0.04948	213.53	404.40	1.0483	1.7224	1.367	0.930	1.193	579.	146.	254.3	11.42	88.8	12.66	10.30	10.00
12.00	0.44289	1253.3	0.04636	216.27	405.51	1.0579	1.7215	1.374	0.939	1.197	569.	146.	248.3	11.52	87.9	12.84	10.02	12.00
14.00	0.47276	1246.3	0.04348	219.03	406.61	1.0674	1.7207	1.381	0.950	1.201	560.	146.	242.5	11.62	87.0	13.02	9.74	14.00
16.00	0.50413	1239.3	0.04081	221.80	407.70	1.0770	1.7199	1.388	0.960	1.206	550.	146.	236.8	11.72	86.0	13.20	9.47	16.00
18.00	0.53706	1232.1	0.03833	224.59	408.78	1.0865	1.7191	1.396	0.971	1.210	541.	146.	231.2	11.82	85.1	13.39	9.19	18.00
20.00	0.57159	1224.9	0.03603	227.40	409.84	1.0960	1.7183	1.404	0.982	1.215	532	145.	225.8	11.92	84.2	13.57	8.92	20.00
22.00	0.60777	1217.5	0.03388	230.21	410.89	1.1055	1.7176	1.412	0.994	1.220	522	145.	220.5	12.03	83.3	13.76	8.65	22.00
24.00	0.64566	1210.1	0.03189	233.05	411.93	1.1149	1.7169	1.420	1.006	1.226	512	145.	215.4	12.14	82.4	13.96	8.38	24.00
26.00	0.68531	1202.6	0.03003	235.90	412.95	1.1244	1.7162	1.429	1.018	1.231	503.	144.	210.4	12.25	81.4	14.15	8.11	26.00
28.00	0.72676	1194.9	0.02829	238.77	413.95	1.1338	1.7155	1.438	1.031	1.238	493.	144.	205.5	12.36	80.5	14.35	7.84	28.00
30.00	0.77008	1187.2	0.02667	241.65	414.94	1.1432	1.7149	1.447	1.044	1.244	484.	143.	200.7	12.48	79.6	14.56	7.57	30.00
32.00	0.81530	1179.3	0.02516	244.55	415.90	1.1527	1.7142	1.457	1.058	1.251	474.	143.	196.0	12.60	78.7	14.76	7.31	32.00
34.00	0.86250	1171.3	0.02374	247.47	416.85	1.1621	1.7135	1.467	1.073	1.259	465.	142.	191.4	12.72	77.7	14.97	7.05	34.00
36.00	0.91172	1163.2	0.02241	250.41	417.78	1.1715	1.7129	1.478	1.088	1.267	455.	142.	186.9	12.84	76.8	15.19	6.78	36.00
38.00	0.96301	1154.9	0.02116	253.37	418.69	1.1809	1.7122	1.489	1.104	1.276	445.	141.	182.5	12.97	75.9	15.41	6.52	38.00
40.00	1.0165	1146.5	0.01999	256.35	419.58	1.1903	1.7115	1.500	1.120	1.285	436.	140.	178.2	13.10	75.0	15.64	6.27	40.00
42.00	1.0721	1137.9	0.01890	259.35	420.44	1.1997	1.7108	1.513	1.138	1.295	426.	140.	174.0	13.24	74.1	15.86	6.01	42.00
44.00	1.1300	1129.2	0.01786	262.38	421.28	1.2091	1.7101	1.525	1.156	1.306	416.	139.	169.8	13.38	73.1	16.10	5.76	44.00
46.00	1.1901	1120.3	0.01689	265.42	422.09	1.2185	1.7094	1.539	1.175	1.318	407.	138.	165.7	13.52	72.2	16.34	5.51	46.00
48.00	1.2527	1111.3	0.01598	268.49	422.88	1.2279	1.7086	1.553	1.196	1.331	397.	137.	161.7	13.67	71.3	16.59	5.26	48.00
50.00	1.3177	1102.0	0.01511	271.59	423.63	1.2373	1.7078	1.569	1.218	1.345	387.	137.	157.7	13.83	70.4	16.84	5.01	50.00
52.00	1.3852	1092.6	0.01430	274.71	424.35	1.2468	1.7070	1.585	1.241	1.360	377.	136.	153.8	13.99	69.5	17.10	4.76	52.00
54.00	1.4553	1082.9	0.01353	277.86	425.03	1.2562	1.7061	1.602	1.266	1.377	367.	135.	149.9	14.16	68.5	17.36	4.52	54.00
56.00	1.5280	1073.0	0.01280	281.04	425.68	1.2657	1.7051	1.621	1.293	1.395	358.	134.	146.1	14.33	67.6	17.63	4.28	56.00
58.00	1.6033	1062.8	0.01212	284.25	426.29	1.2752	1.7041	1.641	1.322	1.416	348.	133.	142.3	14.51	66.7	17.91	4.04	58.00
60.00	1.6815	1052.4	0.01146	287.49	426.86	1.2847	1.7031	1.663	1.354	1.438	338.	132.	138.6	14.71	65.8	18.19	3.81	60.00
62.00	1.7625	1041.7	0.01085	290.77	427.37	1.2943	1.7019	1.686	1.388	1.463	328.	131.	134.9	14.91	64.9	18.48	3.57	62.00
64.00	1.8464	1030.7	0.01026	294.08	427.84	1.3039	1.7007	1.712	1.426	1.490	318.	129.	131.2	15.12	63.9	18.78	3.34	64.00
66.00	1.9334	1019.4	0.00970	297.44	428.25	1.3136	1.6993	1.740	1.468	1.522	308.	128.	127.5	15.35	63.0	19.09	3.12	66.00
68.00	2.0234	1007.7	0.00917	300.84	428.61	1.3234	1.6979	1.772	1.515	1.557	298.	127.	123.9	15.59	62.1	19.40	2.89	68.00
70.00	2.1165	995.6	0.00867	304.29	428.89	1.3332	1.6963	1.806	1.567	1.597	287.	126.	120.3	15.85	61.2	19.72	2.67	70.00
72.00	2.2130	983.1	0.00818	307.79	429.10	1.3430	1.6945	1.846	1.626	1.642	277.	124.	116.7	16.12	60.3	20.05	2.46	72.00
74.00	2.3127	970.0	0.00772	311.34	429.23	1.3530	1.6926	1.890	1.693	1.695	267.	123.	113.1	16.41	59.3	20.39	2.24	74.00
76.00	2.4159	956.5	0.00728	314.96	429.27	1.3631	1.6905	1.941	1.770	1.757	256.	121.	109.4	16.73	58.4	20.74	2.03	76.00
78.00	2.5227	942.3	0.00686	318.65	429.20	1.3733	1.6881	2.000	1.861	1.830	246.	120.	105.8	17.08	57.5	21.09	1.83	78.00
80.00	2.6331	927.4	0.00646	322.41	429.02	1.3837	1.6855	2.069	1.967	1.917	235.	118.	102.1	17.46	56.6	21.46	1.63	80.00
85.00	2.9259	886.2	0.00550	332.27	427.91	1.4105	1.6775	2.313	2.348	2.231	207.	113.	92.7	18.59	54.3	22.41	1.15	85.00
90.00	3.2445	836.9	0.00461	343.01	425.48	1.4392	1.6663	2.766	3.064	2.832	178.	108.	82.6	20.15	—	—	0.72	90.00
95.00	3.5916	771.6	0.00374	355.43	420.60	1.4720	1.6490	3.961	4.942	4.424	145.	102.	70.9	22.59	—	—	0.33	95.00
100.00	3.9721	646.7	0.00265	374.02	407.08	1.5207	1.6093	—	—	—	105.	94.	53.0	28.86	—	—	0.03	100.00
101.03c	4.0560	513.3	0.00195	389.79	389.79	1.5593	1.5593	∞	∞	∞	0	0	—	—	∞	∞	0	101.03

*temperatures are on the ITS−90 scale a = triple point b = normal boiling point c = critical point

Table C.3: Superheated properties of refrigerant R134a tables [22]

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg·K
	P = 0.50 MPa (T_{sat} = 15.71°C)				P = 0.60 MPa (T_{sat} = 21.55°C)				P = 0.70 MPa (T_{sat} = 26.69°C)			
Sat.	0.041118	238.75	259.30	0.9240	0.034295	241.83	262.40	0.9218	0.029361	244.48	265.03	0.9199
20	0.042115	242.40	263.46	0.9383								
30	0.044338	250.84	273.01	0.9703	0.035984	249.22	270.81	0.9499	0.029966	247.48	268.45	0.9313
40	0.046456	259.26	282.48	1.0011	0.037865	257.86	280.58	0.9816	0.031696	256.39	278.57	0.9641
50	0.048499	267.72	291.96	1.0309	0.039659	266.48	290.28	1.0121	0.033322	265.20	288.53	0.9954
60	0.050485	276.25	301.50	1.0599	0.041389	275.15	299.98	1.0417	0.034875	274.01	298.42	1.0256
70	0.052427	284.89	311.10	1.0883	0.043069	283.89	309.73	1.0705	0.036373	282.87	308.33	1.0549
80	0.054331	293.64	320.80	1.1162	0.044710	292.73	319.55	1.0987	0.037829	291.80	318.28	1.0835
90	0.056205	302.51	330.61	1.1436	0.046318	301.67	329.46	1.1264	0.039250	300.82	328.29	1.1114
100	0.058053	311.50	340.53	1.1705	0.047900	310.73	339.47	1.1536	0.040642	309.95	338.40	1.1389
110	0.059880	320.63	350.57	1.1971	0.049458	319.91	349.59	1.1803	0.042010	319.19	348.60	1.1658
120	0.061687	329.89	360.73	1.2233	0.050997	329.23	359.82	1.2067	0.043358	328.55	358.90	1.1924
130	0.063479	339.29	371.03	1.2491	0.052519	338.67	370.18	1.2327	0.044688	338.04	369.32	1.2186
140	0.065256	348.83	381.46	1.2747	0.054027	348.25	380.66	1.2584	0.046004	347.66	379.86	1.2444
150	0.067021	358.51	392.02	1.2999	0.055522	357.96	391.27	1.2838	0.047306	357.41	390.52	1.2699
160	0.068775	368.33	402.72	1.3249	0.057006	367.81	402.01	1.3088	0.048597	367.29	401.31	1.2951
	P = 0.80 MPa (T_{sat} = 31.31°C)				P = 0.90 MPa (T_{sat} = 35.51°C)				P = 1.00 MPa (T_{sat} = 39.37°C)			
Sat.	0.025621	246.79	267.29	0.9183	0.022683	248.85	269.26	0.9169	0.020313	250.68	270.99	0.9156
40	0.027035	254.82	276.45	0.9480	0.023375	253.13	274.17	0.9327	0.020406	251.30	271.71	0.9179
50	0.028547	263.86	286.69	0.9802	0.024809	262.44	284.77	0.9660	0.021796	260.94	282.74	0.9525
60	0.029973	272.83	296.81	1.0110	0.026146	271.60	295.13	0.9976	0.023068	270.32	293.38	0.9850
70	0.031340	281.81	306.88	1.0408	0.027413	280.72	305.39	1.0280	0.024261	279.59	303.85	1.0160
80	0.032659	290.84	316.97	1.0698	0.028630	289.86	315.63	1.0574	0.025398	288.86	314.25	1.0458
90	0.033941	299.95	327.10	1.0981	0.029806	299.06	325.89	1.0860	0.026492	298.15	324.64	1.0748
100	0.035193	309.15	337.30	1.1258	0.030951	308.34	336.19	1.1140	0.027552	307.51	335.06	1.1031
110	0.036420	318.45	347.59	1.1530	0.032068	317.70	346.56	1.1414	0.028584	316.94	345.53	1.1308
120	0.037625	327.87	357.97	1.1798	0.033164	327.18	357.02	1.1684	0.029592	326.47	356.06	1.1580
130	0.038813	337.40	368.45	1.2061	0.034241	336.76	367.58	1.1949	0.030581	336.11	366.69	1.1846
140	0.039985	347.06	379.05	1.2321	0.035302	346.46	378.23	1.2210	0.031554	345.85	377.40	1.2109
150	0.041143	356.85	389.76	1.2577	0.036349	356.28	389.00	1.2467	0.032512	355.71	388.22	1.2368
160	0.042290	366.76	400.59	1.2830	0.037384	366.23	399.88	1.2721	0.033457	365.70	399.15	1.2623
170	0.043427	376.81	411.55	1.3080	0.038408	376.31	410.88	1.2972	0.034392	375.81	410.20	1.2875
180	0.044554	386.99	422.64	1.3327	0.039423	386.52	422.00	1.3221	0.035317	386.04	421.36	1.3124
	P = 1.20 MPa (T_{sat} = 46.29°C)				P = 1.40 MPa (T_{sat} = 52.40°C)				P = 1.60 MPa (T_{sat} = 57.88°C)			
Sat.	0.016715	253.81	273.87	0.9130	0.014107	256.37	276.12	0.9105	0.012123	258.47	277.86	0.9078
50	0.017201	257.63	278.27	0.9267								
60	0.018404	267.56	289.64	0.9614	0.015005	264.46	285.47	0.9389	0.012372	260.89	280.69	0.9163
70	0.019502	277.21	300.61	0.9938	0.016060	274.62	297.10	0.9733	0.013430	271.76	293.25	0.9535
80	0.020529	286.75	311.39	1.0248	0.017023	284.51	308.34	1.0056	0.014362	282.09	305.07	0.9875
90	0.021506	296.26	322.07	1.0546	0.017923	294.28	319.37	1.0364	0.015215	292.17	316.52	1.0194
100	0.022442	305.80	332.73	1.0836	0.018778	304.01	330.30	1.0661	0.016014	302.14	327.76	1.0500
110	0.023348	315.38	343.40	1.1118	0.019597	313.76	341.19	1.0949	0.016773	312.07	338.91	1.0795
120	0.024228	325.03	354.11	1.1394	0.020388	323.55	352.09	1.1230	0.017500	322.02	350.02	1.1081
130	0.025086	334.77	364.88	1.1664	0.021155	333.41	363.02	1.1504	0.018201	332.00	361.12	1.1360
140	0.025927	344.61	375.72	1.1930	0.021904	343.34	374.01	1.1773	0.018882	342.05	372.26	1.1632
150	0.026753	354.56	386.66	1.2192	0.022636	353.37	385.07	1.2038	0.019545	352.17	383.44	1.1900
160	0.027566	364.61	397.69	1.2449	0.023355	363.51	396.20	1.2298	0.020194	362.38	394.69	1.2163
170	0.028367	374.78	408.82	1.2703	0.024061	373.75	407.43	1.2554	0.020830	372.69	406.02	1.2421
180	0.029158	385.08	420.07	1.2954	0.024757	384.10	418.76	1.2807	0.021456	383.11	417.44	1.2676

APPENDIX D

Table D.1: Ideal gas properties of air [22]

T (K)	ρ (kg/m^3)	c_p (kJ/kg · K)	$\mu \cdot 10^7$ (N · s/m^2)	$\nu \cdot 10^6$ (m^2/s)	$k \cdot 10^3$ (W/m · K)	$\alpha \cdot 10^6$ (m^2/s)	Pr
Air, $\mathcal{M}$ = 28.97 kg/kmol							
100	3.5562	1.032	71.1	2.00	9.34	2.54	0.786
150	2.3364	1.012	103.4	4.426	13.8	5.84	0.758
200	1.7458	1.007	132.5	7.590	18.1	10.3	0.737
250	1.3947	1.006	159.6	11.44	22.3	15.9	0.720
300	1.1614	1.007	184.6	15.89	26.3	22.5	0.707
350	0.9950	1.009	208.2	20.92	30.0	29.9	0.700
400	0.8711	1.014	230.1	26.41	33.8	38.3	0.690
450	0.7740	1.021	250.7	32.39	37.3	47.2	0.686
500	0.6964	1.030	270.1	38.79	40.7	56.7	0.684
550	0.6329	1.040	288.4	45.57	43.9	66.7	0.683
600	0.5804	1.051	305.8	52.69	46.9	76.9	0.685
650	0.5356	1.063	322.5	60.21	49.7	87.3	0.690
700	0.4975	1.075	338.8	68.10	52.4	98.0	0.695
750	0.4643	1.087	354.6	76.37	54.9	109	0.702
800	0.4354	1.099	369.8	84.93	57.3	120	0.709
850	0.4097	1.110	384.3	93.80	59.6	131	0.716
900	0.3868	1.121	398.1	102.9	62.0	143	0.720
950	0.3666	1.131	411.3	112.2	64.3	155	0.723
1000	0.3482	1.141	424.4	121.9	66.7	168	0.726
1100	0.3166	1.159	449.0	141.8	71.5	195	0.728
1200	0.2902	1.175	473.0	162.9	76.3	224	0.728
1300	0.2679	1.189	496.0	185.1	82	257	0.719
1400	0.2488	1.207	530	213	91	303	0.703
1500	0.2322	1.230	557	240	100	350	0.685
1600	0.2177	1.248	584	268	106	390	0.688
1700	0.2049	1.267	611	298	113	435	0.685
1800	0.1935	1.286	637	329	120	482	0.683
1900	0.1833	1.307	663	362	128	534	0.677
2000	0.1741	1.337	689	396	137	589	0.672
2100	0.1658	1.372	715	431	147	646	0.667
2200	0.1582	1.417	740	468	160	714	0.655
2300	0.1513	1.478	766	506	175	783	0.647
2400	0.1448	1.558	792	547	196	869	0.630
2500	0.1389	1.665	818	589	222	960	0.613
3000	0.1135	2.726	955	841	486	1570	0.536

APPENDIX E

Table E.1: Properties of common gas, liquid and solids [48]

Properties of common liquids, solids, and foods

(a) Liquids

	Boiling data at 1 atm		Freezing data		Liquid properties		
Substance	Normal boiling point, °C	Latent heat of vaporization h_{fg}, kJ/kg	Freezing point, °C	Latent heat of fusion h_{if}, kJ/kg	Temperature, °C	Density ρ, kg/m³	Specific heat c_p, kJ/kg·K
Ammonia	−33.3	1357	−77.7	322.4	−33.3	682	4.43
					−20	665	4.52
					0	639	4.60
					25	602	4.80
Argon	−185.9	161.6	−189.3	28	−185.6	1394	1.14
Benzene	80.2	394	5.5	126	20	879	1.72
Brine (20% sodium chloride by mass)	103.9	—	−17.4	—	20	1150	3.11
n-Butane	−0.5	385.2	−138.5	80.3	−0.5	601	2.31
Carbon dioxide	−78.4*	230.5 (at 0°C)	−56.6		0	298	0.59
Ethanol	78.2	838.3	−114.2	109	25	783	2.46
Ethyl alcohol	78.6	855	−156	108	20	789	2.84
Ethylene glycol	198.1	800.1	−10.8	181.1	20	1109	2.84
Glycerine	179.9	974	18.9	200.6	20	1261	2.32
Helium	−268.9	22.8	—	—	−268.9	146.2	22.8
Hydrogen	−252.8	445.7	−259.2	59.5	−252.8	70.7	10.0
Isobutane	−11.7	367.1	−160	105.7	−11.7	593.8	2.28
Kerosene	204–293	251	−24.9	—	20	820	2.00
Mercury	356.7	294.7	−38.9	11.4	25	13,560	0.139
Methane	−161.5	510.4	−182.2	58.4	−161.5	423	3.49
					−100	301	5.79
Methanol	64.5	1100	−97.7	99.2	25	787	2.55
Nitrogen	−195.8	198.6	−210	25.3	−195.8	809	2.06
					−160	596	2.97
Octane	124.8	306.3	−57.5	180.7	20	703	2.10
Oil (light)					25	910	1.80
Oxygen	−183	212.7	−218.8	13.7	−183	1141	1.71
Petroleum	—	230–384			20	640	2.0
Propane	−42.1	427.8	−187.7	80.0	−42.1	581	2.25
					0	529	2.53
					50	449	3.13
Refrigerant-134a	−26.1	217.0	−96.6	—	−50	1443	1.23
					−26.1	1374	1.27
					0	1295	1.34
					25	1207	1.43
Water	100	2257	0.0	333.7	0	1000	4.22
					25	997	4.18
					50	988	4.18
					75	975	4.19
					100	958	4.22

APPENDIX F

Table F.1: Properties of common metals [48]

Composition	Melting Point (K)	Properties at 300 K ρ (kg/m^3)	c_p (J/kg · K)	k (W/m · K)	$\alpha \cdot 10^6$ (m^2/s)
Aluminum					
Pure	933	2702	903	237	97.1
Alloy 2024-T6 (4.5% Cu, 1.5% Mg, 0.6% Mn)	775	2770	875	177	73.0
Alloy 195, Cast (4.5% Cu)		2790	883	168	68.2
Beryllium	1550	1850	1825	200	59.2
Bismuth	545	9780	122	7.86	6.59
Boron	2573	2500	1107	27.0	9.76
Cadmium	594	8650	231	96.8	48.4
Chromium	2118	7160	449	93.7	29.1
Cobalt	1769	8862	421	99.2	26.6
Copper					
Pure	1358	8933	385	401	117
Commercial bronze (90% Cu, 10% Al)	1293	8800	420	52	14
Phosphor gear bronze (89% Cu, 11% Sn)	1104	8780	355	54	17
Cartridge brass (70% Cu, 30% Zn)	1188	8530	380	110	33.9
Constantan (55% Cu, 45% Ni)	1493	8920	384	23	6.71
Germanium	1211	5360	322	59.9	34.7

APPENDIX G

Public water cooler box

Figure G.1: Public water cooler after assembly all parts