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GRADUATE SCHOOL OF

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EXPERIMANTAL STUDY ON EVAPORATIVE COOLING FOR PRE-COOLING OF DISCHARGE LINE OF SPLIT AIR CONDETIONERS COMPRESSOR

M.Sc. THESIS

In

MECHANICAL ENGINEERING

BY

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Experimental Study on Evaporative Cooling for Pre-cooling of Discharge Line of Split Air Conditioner Compressor's

M.Sc. Thesis

in

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Supervisor

Prof. Dr. Sait SÖYLEMEZ

By

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This is to certify that we have read this thesis and that in our consensus/majority opinion it is fully adequate, in scope and quality, as a thesis for the Master of Science.

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Ayaz Ayden HURMUZI

ABSTRACT

EXPERIMENTAL STUDY ON EVAPORATIVE COOLING FOR PRE-COOLING OF DISCHARGE LINE OF SPLIT AIR CONDITIONER'S COMPRESSOR

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Air conditioners consume high energy especially in very hot climate regions, and then performance of the system decreases. Therefore, reducing the energy consumption of the conditioners is one of the most important problems for an air conditioner. In this thesis, an experimental study on split air conditioner having 24000 Btu/h cooling capacity is performed by applying evaporative cooling of the condenser. In order to cool the air before passing on the condenser, it is cooled by spraying cold water. Thermal performance of the cooling unit is determined by measurement of performance parameters for the cooling unit and its conditions. The parameters are inlet and outlet temperatures, mass flow rate of the fluids, voltage and current used by the compressor and fans of the air conditioner. Data for the air conditioner before and after modification are collected from June to August 2014 to obtain the performance parameters of the unit. The testes are performed between 32 and 45 C°, and for two relative humilities of 45 and 80 % to compare the system performance in medium and high relative humilities. Results show that the energy consumed by the air conditioner can be saved between 23 and 27.5 % at the testing conditions while the COP of the unit increases between 36 and 49 %. The results indicate that this type of air conditioner can be used reliably and efficiently for Gaziantep climatic conditions.

Keywords: Air conditioner, split type air conditioner, evaporative cooling, energy saving

ÖZET

SPLİT KLİMA TAHLİYE HATTI ÖN SOĞUTMASI İÇİN EVAPORETIF SOĞUTMA ÜZERİNE DENEYSEL ÇALIŞMA

HURMUZI, Ayaz Ayden

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Tez Yöneticisi: Prof. Dr. Sait SÖYLEMEZ

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90 sayfa

Klimalar özellikle sıcak iklim bölgelerinde yüksek enerji tüketmektedirler ve o zaman da performansları düşmektedir. Bu yüzden, klimaların enerji tüketimini azaltmak, klimaların en önemli problemlerinden birisidir. Bu çalışmada, buhar sıkıştırmalı çevrime göre çalışan ve 24000 Btu/h soğutma kapasiteli siplit tip bir klimanın yoğuşturucusuna buharlaştırmalı soğutma uygulayarak deneysel bir çalışma yapılmıştır. Soğutma havası yoğuşturucuya girmezden önce üzerine soğuk su püskürterek buharlaştırmalı soğutma yapılmıştır. Klimanın ısıl performansı test şartlarında ölçülen parametreler kullanılarak belirlenmiştir. Bu parametreler ise, akışkanların giriş ve çıkış sıcaklıkları, debileri ve çevre sıcaklıkları ile klimanın kompresör ve fanlarının kullandığı voltaj ve akımlardır. Klimanın performans parametrelerini elde etmek için Haziran ile Ağustos 2014 tarihleri arasında testler yapılarak veriler elde edilmiştir. Düşük ve yüksek bağıl nemlerde klimanın performansını karşılaştırmak için, testler dış hava sıcaklığı 32 ile 45 C° arasında ve bağıl nem ise % 45 ve 80 olduğu şartlarda yapılmıştır. Sonuçlar, test şartlarında % 23 ile 27.5 arasında enerji tasarrufu yapıldığını gösterirken, % 36 ile 49 arasında performans katsayısının yükseldiğini göstermiştir. Bu sonuçlar, bir klima kondenserinin buharlaştırmalı soğutma yapılarak, Gaziantep hava şartlarında güvenli ve verimli bir şekilde kullanılabileceğini göstermiştir.

Anahtar kelimeler: Klima, split tip klima, buharlaştırmalı soğutma, enerji tasarrufu

This thesis dedicated to

Aydan NAKİP, Fidan ÜMİT, and to my kids ARZU, MUHAMMED, and NİLÜFER

for their endless support, and encouragement

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TABLE OF CONTENT

LIST OF FIGURES

page

LIST OF TABLES

LIST OF SYMBOLS

C_p	Constant pressure specific heat	KJ/Kg °C
K	Thermal conductivity	W/m °C
T_{db}	Dry bulb temperature	°C
T_{wb}	Wet bulb temperature	°C
h	Specific enthalpy of refrigerant	KJ/Kg
s	Specific entropy	KJ/Kg °C
$\dot{m}$	Mass flow rate	Kg/s
Q	Heat transfer rate	KW
W	Power	KW
P	Pressure	bar
ω	Moisture contain	
ϕ	Relative humidity	%
f	Friction factor	----
L	Length of pipe	m
D	Diameter of pipe	m
ρ	Density	Kg/m^3
V	Velocity	m/s
μ	Viscosity	Kg/m s
R_e	Reynolds number	---
Nu	Nusselt number	---
ε_e	Evaporative effectiveness	%
A	Pipe crosectional area	m^2
V	Voltage	Volt
I	Electric curent	A

ABBREVIATIONS

HVAC	Heating ventelating air conditioning systems
COP	Cofficinet of performance
DBT	Dry bulb temprature
WBT	Wet bulb temprature
RH	Relative humidirty
EER	Evaporation effective ratio
ECI	Evaporative cooling index

CHAPTER 1
INTRODUCTION

1.1 General

Small air conditioners are used widely in homes and small workplaces, and the big needs for these types of air conditioners occurred because of the ease its installation and low cost, but these types of air conditioners consume high energy compared to units of central air-conditioning, as the central air conditioning units conditioned all parts of the building which leads to lack of heat exchange between the interior walls of the building.

The small air conditioner units such as split or window types used forced air condensing and this type of condenser have good performance in low and medium ambient temperatures.

Due to the global warming, many countries start to get hotter summer especially in Middle East countries, and temperature rise up to 49 C, since the temperature and the pressure of the condenser increase and the compressor is forced to work under the greater pressure ratio, which results in more power consumption and low cooling performance.

Another problem which was reported with application of air condenser in hot weather area is related to the high stories buildings. In these buildings the hot air from air conditioners of lower stories rise up and provide a hot flow field around the air conditioners of higher stories as shown in Fig. 1. The increase in the air temperature is so high that the air conditioner trip down. In order to prevent this problem the hot air is required to be cool down before passing over the condenser.

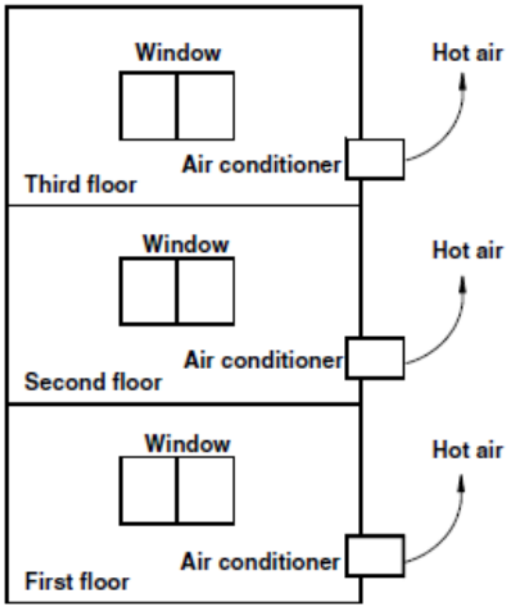


Figure 1.1 Hot air around the building

1.2 Evaporative cooling

Evaporative cooling is the process by which the temperature of a substance is reduced due to the cooling effect from the evaporation of water. The conversion of sensible heat to latent heat causes a decrease in the ambient temperature as water is evaporated providing useful cooling (figure 1.2). This cooling effect has been used on various scales from small space cooling to large industrial applications.

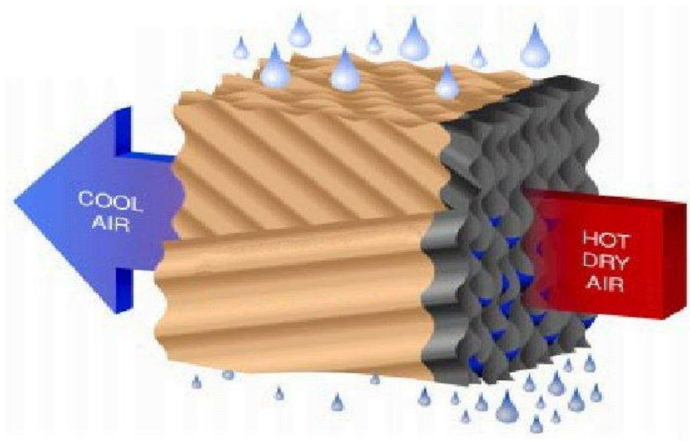


Fig. 1.2 Evaporative cooling principle

1.3 Evaporative condensers

Evaporative condenser used to increase the condenser efficiency without need to increase the physical size of the condenser, and this system work base on evaporation water. When water vaporize and absorb the latent heat of the media, cooling effect will appear, there are two main types of evaporative condensers

1.3.1 Direct evaporative condensers

In this type the condenser will put inside the evaporation space and the water will inject directly on the condenser, in other side an air blower will push the air to the evaporation area, this type of condenser calls also direct evaporation condenser figure 1.3

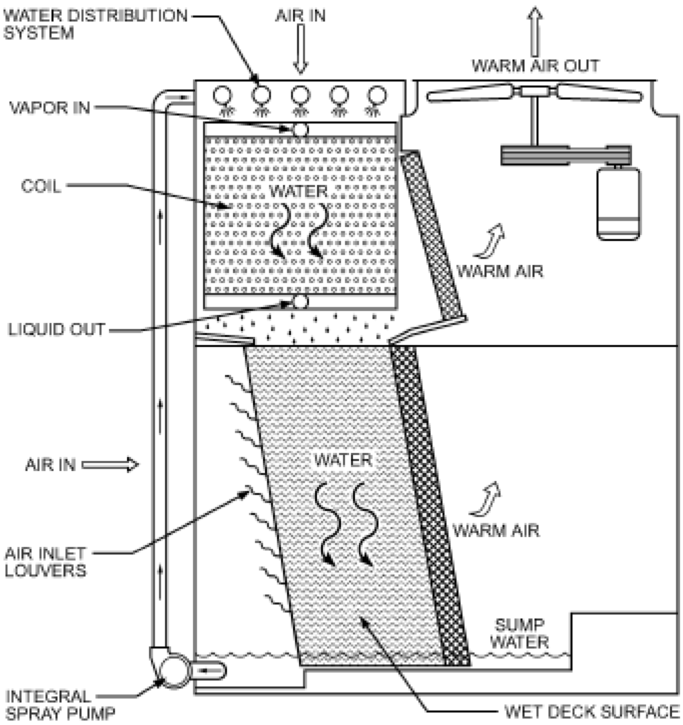


Figure 1.3 Combined Coil/Fill Evaporative Condenser

1.3.2 Indirect evaporative condenser

In this type of evaporative condenser , there were no contact between water and condenser, the cooling effect on the air which is appear due to water evaporation will use to cooling the condenser, and this type of evaporative condenser will used in this thesis.

1.4 Application of evaporative condenser in small air-conditioners

Generally the evaporation cooling applied on small air-conditioners by using water injection on a filling material, and the condenser fan used as air blower to suck the air on the wet pad. Figure1.4 shows the general applying of evaporative cooling on air-conditioner condenser.

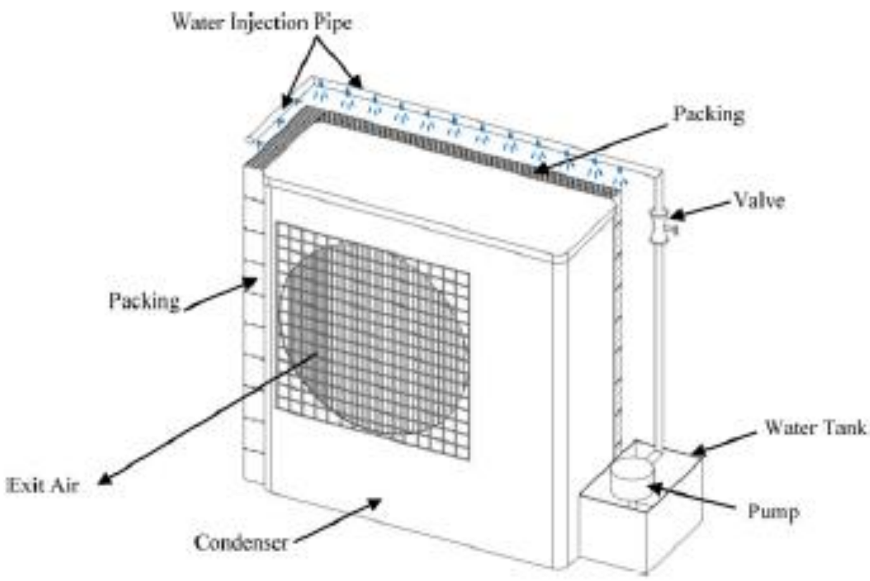


Figure 1.4 Applying evaporative cooling on condenser

1.5 Discharge line pre-cooling

The main aim of this study is to evaluate how the pre-cooling will affect one the air-conditioner if the pre-cooling applied on discharge line of the compressor. The cooling will done by extend discharge line and pass it through cold water, evaporative cooling will apply to the outside of the air conditioner as figure 1.4, and due to evaporation of water, like air washer systems, it's will get cold air which it pass through the air-conditioner condenser, and as a result of this process a cold water will produce which will used to cool the discharge line.

1.6 Effect of discharge pre-cooling on condenser efficiency

In any simple refrigeration cycle which it's working based on compression method, the compressor, take the refrigerant from low pressure side of the system (evaporator) by suction line and compress it to condenser in high temperature and pressure (proses 1-2) as shown in ph diagram in figure 1.5

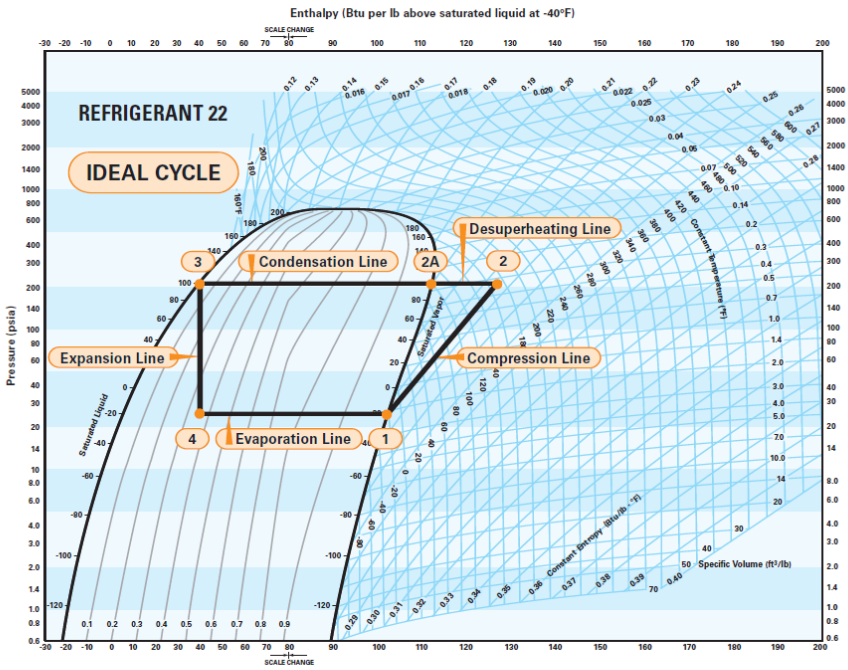


Figure 1.5 Simple compression refrigerant cycle

The superheated refrigerant at point 2 enter to the condenser, which it convert to saturated liquid at point 3, in this process the superheated gas should be change to saturated gas first at point 2A, this conversion done in the first third of condenser, the discharge line pre-cooling will make the superheated gas change to saturated gas before entering to condenser, thus the condenser effectiveness will increase.

The pre-cooling idea in all small air-conditioner are applied simply by extending the discharge line, so the superheated gas can take more area to transfer the heating from the discharge line before entering the condenser as shown in figure 1.6,but in this work, the pre-cooling will done by cold water instant of air.

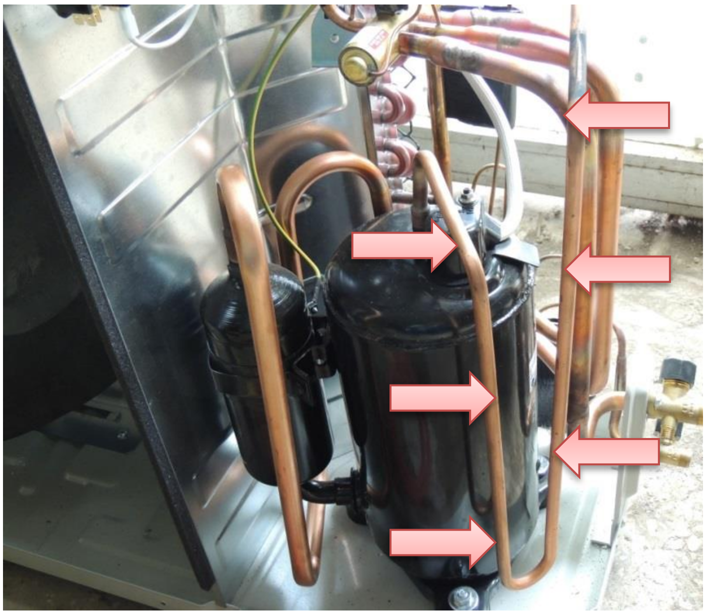


Figure 1.6 Discharge line extensions in small air-conditioner

1.7 Objective of Study

Small air conditioners are often used to conditioned rooms and offices in our life, and it's used generally in summer season operating at cooling mode, and this kind of air-conditioners are consume large electric power every season spatially in the hot summer areas like south eastern of Turkey. In this thesis, we will focus on how we decrease the electric power consume of the small air-conditioners operating in cooling mode.

The experimental study will collect operating data of 24000 BTU/h split air-conditioner which it modified by applying indirect evaporation cooling system to the outside unit of the air-conditioner and applying pre-cooling to the discharge line using the cold water which it's produce from evaporative cooling system.

Three types of data are collected from under different conditions:

- The original operating without any modifying
- Operating with evaporating cooling
- Operating with modified pipe line applying pre-cooling on discharge line

The investigation will do in several outside temperature based on the wide range temperature difference; the experiments were done in medium and high relative humidity to compute the system performance in several weather conditions.

1.8 Layout of Thesis

This thesis consists of seven chapters which are organized in such a way that; in the following chapter two, comprehensive literature survey of research on the object is summarized, the historical background on the, evaporating cooling, small air-conditioners and how the evaporating cooling applied on the system.

In the third chapter, present basic information about vapour compression sysems.
The forth chapter will mention a background and information about evaporative cooling and evaporative condensers.
Chapter five will cover the experimental setup and theoretical analyses of applying the evaporative cooling on the air conditioner condenser, and applying the pre-cooing on the compressor discharge line.

Chapter six presents the experimentally measured result data that have been recorded during the tree types of operations and the air-conditioner performance in sevral out room conditions. The results calculated are turned into graghics and tabels in order to compare three operating types. Chapter seven will contain the tables and counculusion.

CHAPTER 2
LITERATURE SURVEY

2.1 Introduction

Evaporative cooling can be applied on air conditioner which its work on vapor compression system principle, by two methods, the first method is to used direct evaporative condenser, where the condenser installed inside the evaporative cooler, and the second method is to use the evaporative cooling system to pre-cool the air which it used to cold the condenser in the air conditioner.

In this chapter, previous works on evaporative cooling improvement and combining of the evaporative cooling and vapor compression system will explain in the next sections.

2.2 Evaporation rate improvement studies

(Giabaklou, 1996) added a evaporative cooling system on the front face of a building, the water in this study was moved from the top of the building to the bottom by gravity, and they found the air which it's inters to the building is reduced by 9.9 C°, while Kant and (Mullick, 2003) attempt to add the evaporative cooling in room with exposed roof.

(Taufiq, et al., 2007) have made an energy analysis of evaporative cooling for reducing energy use in a building. A correlation has been developed between relative humidity and energy efficiency, and between ambient temperature and energy efficiency. The result of the study was show that when relative humidity is increased, energy use also increases.

(Hoyano, 2010) have studied the cooling effects of passive evaporative cooling wall constructed of porous ceramic materials. Experimental results showed that the cooling efficiency reached a maximum of 0.7 during sunny daytime periods. A higher cooling efficiency is obtained under windy conditions where wind at a speed of 1-3 m/s is continuously blowing. (Heidarinejad, et al., 2010) have investigated a hybrid system of nocturnal irradiative cooling and direct evaporative cooling and up to 13.5 ºC reduction in indoor temperature is reported

2.3 Studies on cooling pad effectiveness

The cooling pad is one of the important factors which effect on any system works with evaporative cooling principle, it's can effect of the efficiency of the system by 50 %, many students are made to evaluate the pad performance and how it's effect on the system. Table 2.1 shows the previous working on cooling pad effectiveness.

Table 2.1 Studies related to the effect of cooling pad material

No	Author(s)	Pad materials tested	Temperature drop or Evaporative cooling efficiency reported
1	Taha et al., (1994)	Charcoal granules	10- 13 °C
2	Liao, (2002)	Coarse fabric PVC Sponge	82 to 85 %
		Fine fabric PVC Sponge	77 to 92 %
3	Al-Sulaiman, (2002)	Date palm fiber	39 %
		Jute	62 %
		Luffa	55 %
		Commercial pad	50 %
4	Anyanwu, (2004)	Porous	0.1 – 12 °C
5	Gunhan, et al. (2007)	Fine pumice stones	93 %
		CELdek	82 %
		Volcanic tuff	81 %
		Coarse pumice stones	76 %
		Greenhouse shading Net	51 %
6	Darwesh et al. (2009)	Rice straw	5.67 – 8.66 °C, 77 %
		Palm leaf fibers	5.01 – 7.50 °C, 88 %
7	Ahmed et al. (2010)	CELdek	10.63 °C, 82 %
		Straw	11.77 °C, 79 %
		Sliced wood	9.53 °C, 86 %

2.4 Studies on combining evaporative cooling with vapor compression system

There have been several studies on improving the performance of an air-cooled condenser taking advantage of evaporative cooling. The studies reviewed in this section concluded that the methods are effective in increasing the performance of an air conditioner.

(Goswami, et al., 1993) examined experimentally a tonnage air-conditioner performance by adding evaporative cooling on the system condenser, by using a wetted media pad surrounded the condenser to pre-cool the ambient air, data collected for three week operating and found 20% EER improvement with the evaporative cooling system installed because of the lower compressor power consumption and the gain in cooling capacity.

(Grant, et al., 2001) also experimented with indirect evaporative cooling for window type air conditioner; a wetted media pad was used to pre-cool the ambient air, in this test one more setup are used in investigation, by using a desiccant to lower the relative humidity of the ambient air before entering the evaporative cooling zone, this extra setup was done to decrease the interring wet bulb temperature to obtain more evaporative cooling performance, the study found 18% power reduction in the system after applying the evaporative cooling.

(Kutscher, 2002) made a simulation was done on a geothermal power plant using indirect evaporative cooling for the air-cooled condenser, The study aim was to increase the output of the plant by pre-cooling the ambient air. Four different methods were considered and then economically analyzed, and all the four methods shows increasing in the capacity of the plane, but the economic study shows the methods were very expensive.

(Aglawe, 2013) made an experimental study to adding evaporative cooling to improve the cooling efficiency, they used a 18000 BTU/h split type air conditioner, they found there is increasing in the cooling capacity by 17%.

(M. Youbi-Idrissi, 2004) made a Numerical model of sprayed air cooled condenser coupled to refrigerating system, the model was not including cooling pad, and instant of it, he used direct water spraying on the condenser, and he get 55% improvement in COP.

2.5 Selecting evaporative cooling weather

Evaporative cooling performance depend on two main factors in weather, dry bulb temperature (DBT), and wet bulb temperature (WBT), and studying these two factors in weather are so important to insure the evaporative cooler will working in good efficiency and more economically.

According to (Joudi, et al., 2000) the wet bulb temperature can be calculated under normal pressure range by equation 2.1

$$T_{wb} = 2.265\sqrt{1.97 + 4.3\,T_{db} + 10^4\,\omega} - 14.85 \tag{2.1}$$

In equation 2.1 the moisture contain ω are express by the saturated pressure P_{sat} and relative humidity ϕ as in equation 2.2

$$\omega = \frac{0.622\,P_{sat}\,\phi}{1.013\ X\ 10^5 - \phi\,P_{sat}} \tag{2.2}$$

(Musa, 2012) used the equations 2.1 and 2.2 to build a simulation to compare the inlet and outlet temperature in a cooling evaporative system, and he found that the outlet air temperature are decreasing when the relative humidity are increase in the inlet air, figure 2.1 shows the change of outlet dry bulb temperature by time for constant inlet temperature (43°C) for various relative humidity while figure 2.2 shows the same simulation result for (35°C)

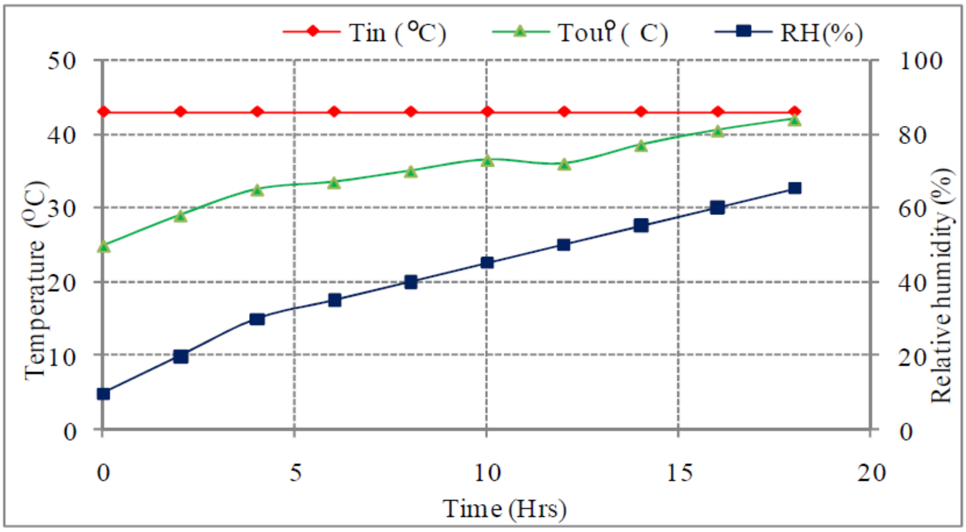


Figure 2.1 Timely variation of outlet air temperature with relative humidity for 43°C DBT

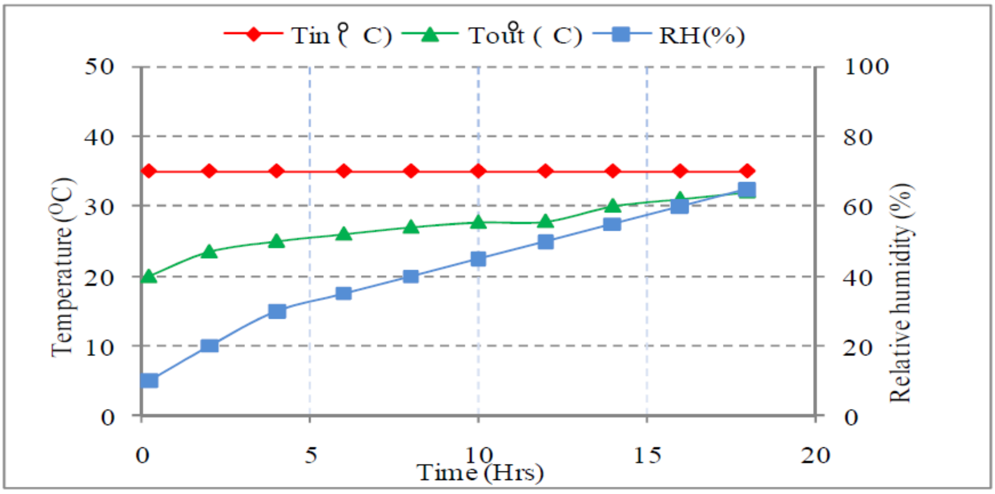


Figure 2.2 Timely variation of outlet air temperature with relative humidity for 35°C DBT

The results shows that the decreasing amount in outlet air temperature are depend on the relative humidity of the outside air, and the efficiency of the evaporative cooling decrease in high relative humidity.

For same reason, (M.Ateş, 2005) make an study to investigate how much the evaporative cooling can used in Turkey, in this study the evaporative cooling index (ECI) were found for seven major regions in Turkey depending on the weather dataset from Turkey Meteorological authority, as shown in figure 2.3, the study found that the evaporative cooling can be used efficiently in middle, south, and south east of Turkey

Region	Population		CDD		ECI weighted by CDD							PCDD		ECI weighted by PCDD						
			(^{0}C·Day)		Avg	(R)	% of Annual CDD					(People·^{0}C·Day)		Avg	(R)	% of Annual PCDD				
	x 10^3	(R)		(R)			0	1	2	3	4	x 10^6	(R)			0	1	2	3	4
Aegean	3,787	(4)	1,147	(4)	3.12	(6)	1	2	21	38	39	1,232	(4)	2.69	(6)	2	5	36	39	19
Black Sea	429	(7)	402	(7)	3.77	(2)	0	0	4	14	81	54	(7)	3.87	(2)	0	0	2	8	89
Central Anatolia	6,985	(2)	1,097	(6)	3.94	(1)	0	0	0	5	95	873	(5)	3.90	(1)	0	0	0	10	90
East Anatolia	1,286	(6)	1,224	(3)	3.77	(3)	0	0	2	20	78	343	(6)	3.82	(3)	0	0	1	16	83
Marmara	12,778	(1)	1,144	(5)	3.21	(5)	1	0	11	54	34	1,251	(3)	3.08	(5)	1	1	13	59	26
Mediterranean	4,356	(3)	2,041	(2)	2.27	(7)	26	7	14	17	35	1,780	(2)	2.62	(7)	30	9	17	16	28
SE Anatolia	3,184	(5)	2,669	(1)	3.54	(4)	1	2	8	20	69	1,794	(1)	3.65	(4)	0	1	6	19	74
TURKEY	**32,805**		**9,723**		**3.27**		**6**	**2**	**9**	**24**	**59**	**7,327**		**3.18**		**8**	**3**	**14**	**27**	**48**

Figure 2.3 ECI for seven regions of Turkey

CHAPTER 3
REVIEW ON VAPOR COMPRESSION SYSTEMS

3.1 Introduction

Vapor compression refrigeration systems are the most commonly used among all refrigeration systems. As the name implies, these systems belong to the general class of vapor cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapor compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems. Vapor compression refrigeration

3.2 Vapor Compression System Components

The main components for any vapor compression system include compressor, two heat exchanger as condenser and evaporator, and expansion valve.

These main components connect in closed system and the working fluid is circulating in the system known as refrigerant.

3.2.1 Compressor

It's one of the important parts of any compression cycle, it's sucked the refrigerant from low side pressure of the system (evaporator) and compressed the refrigerant to the high side pressure (condenser), and the compressor performed the circulation of the refrigerant in the system. Many types of compressors are available, which are:

1. Reciprocating type
2. Rotary type
3. Centrifugal type
4. Scroll type

Reciprocating type is one of the most compressor types which used in vapor compression cycles, as shown in Figure 3.1 a piston cylinder principle are used, when the piston down the suction valve will be opened while the discharge valve will be closed, and when the piston go up, the section valve will closed and the discharge valve will opened, so the refrigerant will compressed to the discharge line.

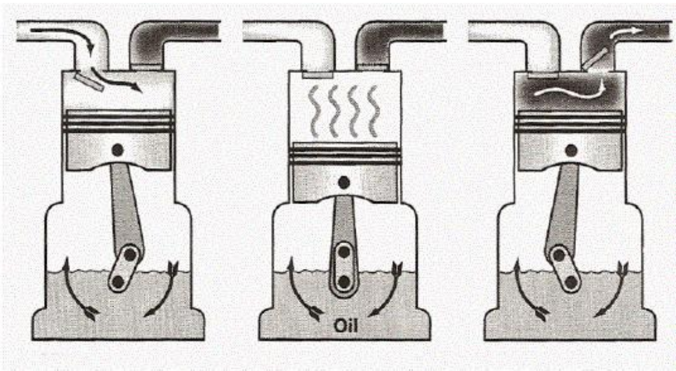


Fig. 3.1 Reciprocating compressor principle

Because reciprocating compressors do not rely on refrigerant flow to deliver oil and cool the surfaces with boundary lubrication, they are uniquely suitable for low-suction-pressure applications, such as medium- and low temperature refrigeration. Efficiency of these compressors is lower than more advanced designs, but reliability is extremely high.

Rotary compressor type is one of the common compressors which used in small air conditioners, it's very reliable and work quit. This type of compressor uses a roller mounted on the eccentric of a shaft with a single vane or blade suitably positioned in the nonrotating cylindrical housing, generally called the cylinder block. The blade reciprocates in a slot machined in the cylinder block. This reciprocating motion is caused by the eccentrically moving roller Fig. 3.2 shown the rotary compressor working

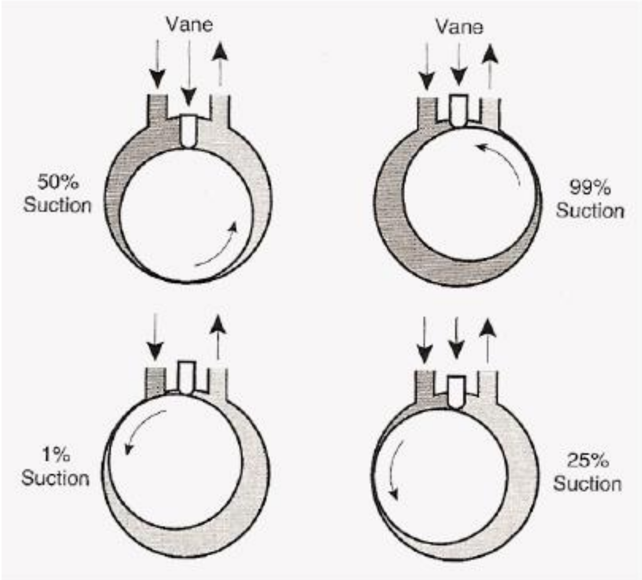


Fig. 3.2 Rotary compressor working principle

The centrifugal compressor type as shown in Fig. 3.3 are used centrifugal force with a rotating impeller to compress refrigerant vapor, it’s used in large water cooled chillers.

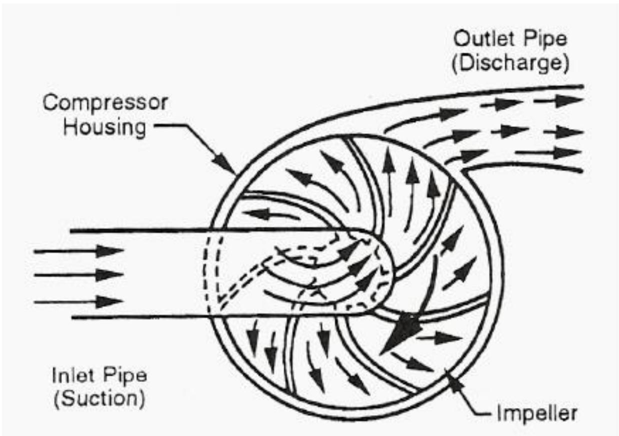


Fig. 3.3 Centrifugal compressor

Scrolled type Fig. 3.4 are one of the common compressors which used in HVAC marketplace, it’s reliable and have less moving parts, efficient work on principle of trapping and compressing refrigerant vapor.

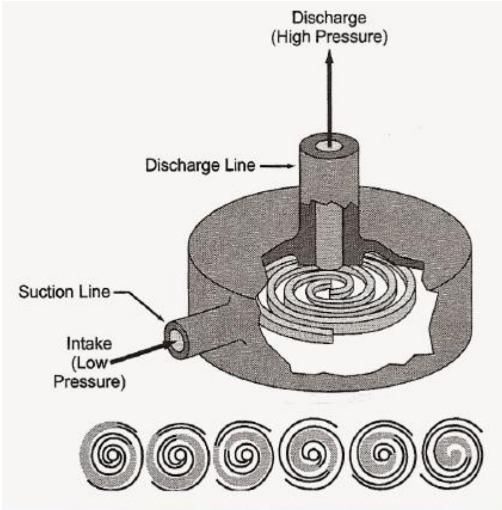


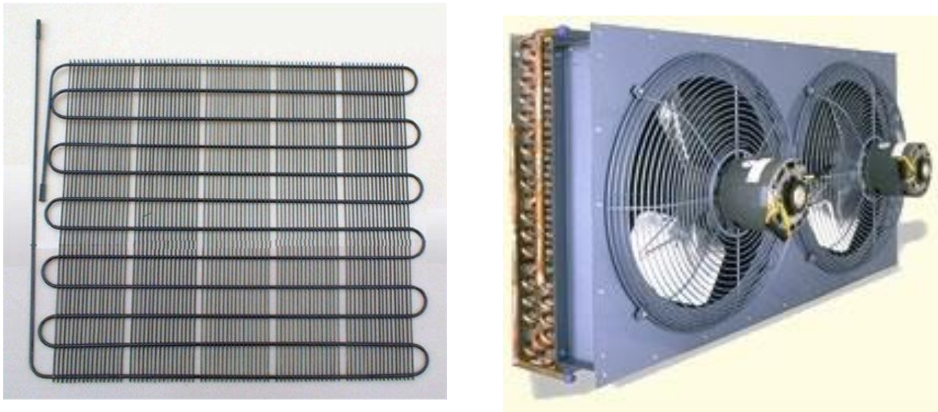
Fig. 3.4 Scrolled compressor

3.2.2 Condenser

Condensers used in vapor compression cycles as heat exchanger, the high pressure and temperature refrigerant which it generate by compressor will inlet the condenser by discharge line, the condenser will reject the heat to the ambient and the refrigerant will condensed to the liquid stat without decreasing in refrigerant pressure.
There are three main types of condensers and they can mentions as:

i. Air cooled condenser, it's used in small unit's like domestic refrigerators and water cooler, the heat exchange between the condenser and the ambient air by natural load of convection, air cold condensers usually made from copper or aluminum, with grill fin to increase the area of heat exchange as shown in Fig. 3.5a

ii. Forced air cooled condenser, it's have the same principle work of natural air condenser, but in smaller size, a fan push the air through the condenser, this kind of condenser can be found in small air conditioners and big size refrigerators as shown in Fig. 3.5b



a b

Fig. 3.5 Types of air cold condenser

iii. Water cooled condenser, these types of condenser used in large load units like central air conditioning systems, a circulation water cooled the condenser, it's also can be classified as three types, double pipe type it's also called tube-in-tube type, sell-coil type and shell-tube type. The cooling water inlet to the heat exchanger and cooled down the refrigerant, and the water in this case usually re-cooled by cooling tower.

iv. Evaporative condensers, in this type the discharge line will inlet to the evaporative condenser, which it's contain the condenser, water spry system and fan, the water sprayed over the condenser's coils and the fan will push the air inside the condenser, due to the water vaporizing ,the condenser will cooled down, Fig. 3.6 shows the principle of evaporating condenser work.

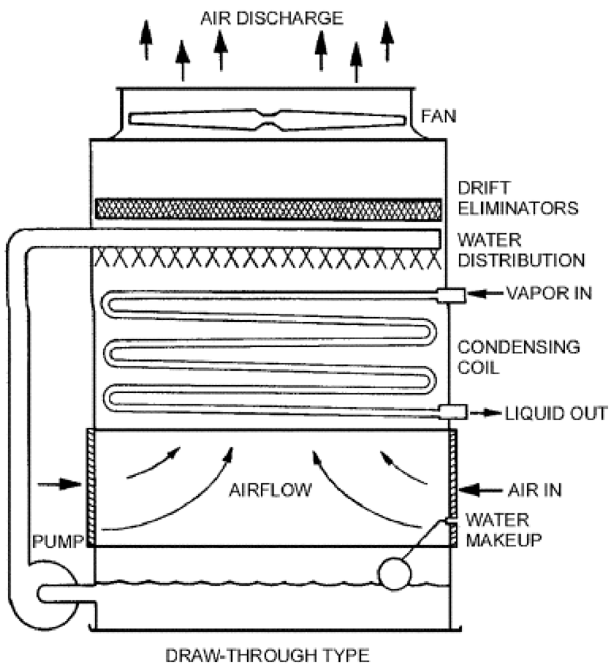


Fig. 3.6 Evaporative condenser

3.2.2 Expansion Valve

It's the device that reduce the refrigerant pressure from high pressure (condenser pressure) to low pressure (evaporator pressure) , it's can be as a capillary tube in small size units like small refrigerators or air conditioners, or can be found as expansion device.

The most common expansion valve are thermostatic expansion valve which it can control the flow of refrigerant which it inlet to the evaporator.

3.2.3 Evaporator

Evaporator is the type of heat exchanger that the refrigerant absorbs heat from the surrounding and changed from liquid to vapor stat, it can found as natural convection type like domestic refrigerators or as fan coil unit as in air conditioner, or shell tube as in chillers.

3.3 Miscellaneous Parts of Vapor Compression System

There are some secondary parts for any system works on vapor compression system, these parts are necessary to operate the unit such as pipes, control units ,refrigerant dryer .. Etc. in this section a little summery will mention of these parts

3.3.1 Control units

Control units are used to control the system operation more economical such as thermostat, which controlled the compressor operation when the system reaches to the aimed temperature, some controls are used to protect the system such as high and low pressure switches and overload switch.

3.3.2 Pipe Connection

Pipes are made from copper or aluminum, it's connect the main system parts, the size of pipes, length, pipe metal and welding should be choice carefully to insure the best operation for the system.

3.3.3 Refrigerants

It's the working fluid in any HVAC or refrigeration system, it's absorb the heat from surrounding in evaporator and reject it at condenser, refrigerant can be a substance or mixture, the chemical and physical properties can be change from refrigerant to other, but there are some criteria should be in any refrigerant like, nontoxic, nonflammable, and environmentally.

The most common types of refrigerants can be mention as:

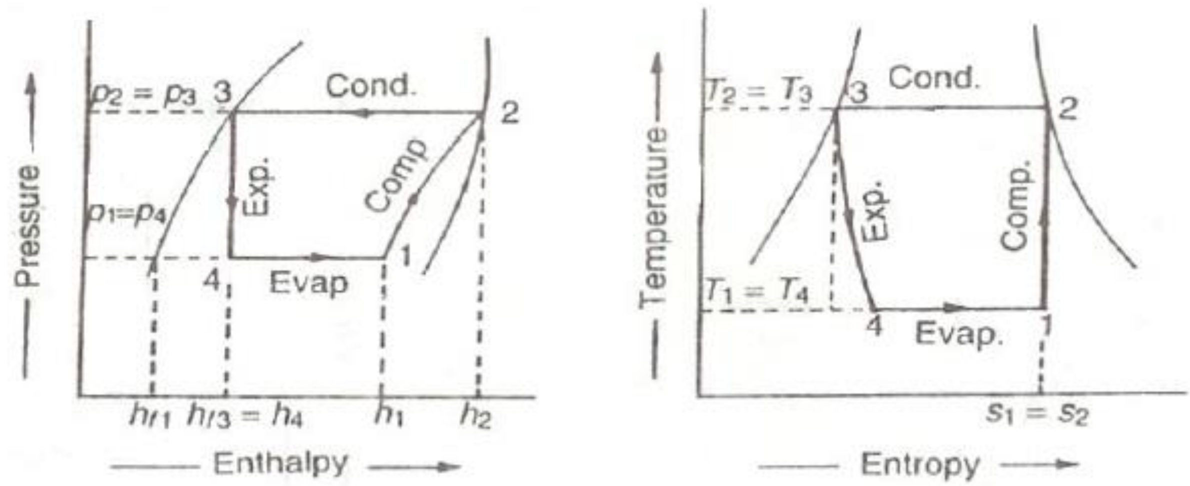
- CFC-12
- CFC-114
- R-22
- R-502

However, some of refrigerant found that it's effect negatively to ozone layer, spatially the chlorine content; some of these refrigerants are excluding to use like R-11/12/12, R-113/14/15, and R-500/502.

A new generation of non-chlorine refrigerant found and stared to use, like R-134a, R-410a, and R-404a.

3.4 Simple Vapor Compression Cycle

A vapor compression cycle with dry saturated vapor after compression is shown on T-s diagrams in Figures 3.7(a) and (b) respectively at point 1,T_1, P_1, s_1, the temperature, pressure and entropy of the vapor refrigerant respectively. The four processes of the cycle are as follows:



a b

Fig. 3.7 Ts & ph diagram for simple compression cycle

- **Compression Process**

  The vapor refrigerant at low pressure P_1 and temperature T_1 is compressed isentropically to dry saturated vapor as shown in line 1-2 on the T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature increase from P_1 to P_2 and T_1 to T_2 respectively.

The work done during isentropic compression per kg of refrigerant is given by

$$w = h_2 - h_1$$

Where:

h_1 Enthalpy of vapor refrigerant at the suction of compressor

h_2 Enthalpy of vapor refrigerant at the discharge of compressor

- **Condenser Process**

  The high pressure and temperature vapor refrigerant after leaves compressor, is passed through the condenser where it is completely condensed at constant pressure and temperature as shown by the horizontal line 2-3 on T-s and p-h diagrams. The vapor refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

- **Expansion Process**

  The liquid refrigerant at pressure $P_3 = P_2$ and temperature $T_3 = T_2$, is expanded by throttling process through the expansion valve to a low pressure $P_4 = P_1$ and Temperature $T_4 = T_1$ as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram In some smaller or domestic a capillary tube used instant of expansion device, some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator. In this process, no heat is absorbed or rejected by the liquid refrigerant.

- **Vaporizing process**

  The liquid-vapor mixture of the refrigerant at pressure $P_4 = P_1$ and temperature $T_4 = T_1$ is evaporated and changed into vapor refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapor refrigerant absorbs its latent heat of vaporization from the medium which, is to be cooled, this heat which is absorbed by the refrigerant is called refrigerating effect.

The process of vaporization continues up to point 1 which is the starting point and thus the cycle is completed.

3.5 Actual Vapor Compression Cycle

The actual vapor compression cycle have two main differences from the ideal cycle, the fluid frictions in pipes which cause pressure drop and heat loss to or from surrounding. In Fig. 3.8 an actual cycle process are shown,

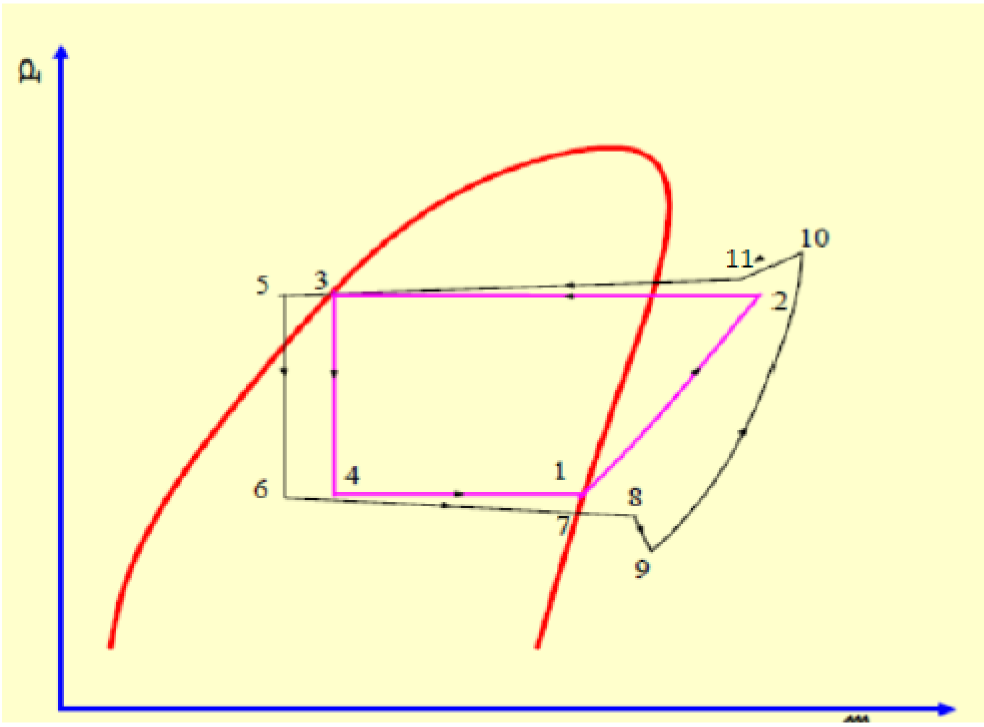


Fig.3.8 ph diagram for actual compression cycle

In (1-2-3-4) cycle, an ideal cycle is appeared, while (7-8-9-10-11-5-6) are represent to the actual vapor compression cycle. The main difference can be mentioned as:

- Pressure drop in the system components as process 11-5 in condenser, 6-7 in evaporator, 8-9 in compressor suction valve, and 10-11 in compressor discharge line.

- The compression process 9-10 is polytropic while it's assumed as constant entropy process in the ideal cycle.
- The liquid refrigerant are leave the condenser in low temperature than the condenser temperature (subcooling) as in process 5-6
- The vapor refrigerant are leave the evaporator in temperature higher than the saturate temperature (super heat)

3.5.1 Pressure Drop

Pressure drop occurs during fluid flow as a result of frictional forces within the fluid and frictional forces between the moving fluid stream and the stationary pipe walls. The amount of pressure drop depends on a number of variables, including:

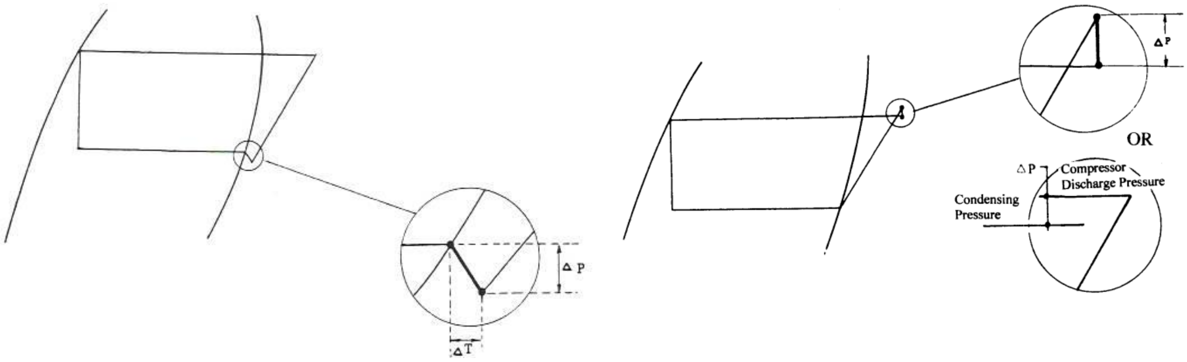
- Type of flow, e.g., laminar, turbulent, etc.
- Physical properties of fluid, e.g., viscosity, density, etc.
- Pipe characteristics, e.g., diameter, roughness, etc.
- Velocity of flow in pipe

Pressure drop increases in proportion to the length of pipe. Pressure drop is also increased by anything which disturbs the flow, such as valves, tees, elbows and other fittings.

In refrigerant piping, some pressure drop occurs in both vapor and liquid lines. These pressure drops can have a significant impact on system performance. The effect of these pressure drops must be anticipated and compensation made in the total design.

Suction penalty is the combination of suction superheat and suction pressure drop as shown in Fig. 3.9a Therefore, the suction penalty is not a horizontal, nor a vertical line. The correct expression shall be a slope as shown in the P-H Diagram.
Compressor discharge pressure is to be higher than the condensing pressure to overcome the pressure resistances in the discharge line. This pressure difference ($\triangle$P) is shown in the P-H Diagram of Fig. 3.9b.

Too much discharge pressure drop allowed would increase the power consumption of the compressor. But, too small pressure drop allowance might increase the size of discharge piping and fittings. In any case, the discharge penalty should be allowed for the compressor selection for the proper function of the refrigeration system. For screw compressor application, the ΔP is usually larger than the reciprocating and centrifugal compressors. The ΔP for air cooled or evaporative condenser is larger than the water cooled condenser.



(a) (b)

Fig. 3.9 Pressure drop in suction and discharge line

Also the pressure drop can occur in condenser and evaporator, and generally the pressure drop can be calculated as shown:

$$\Delta p = f \frac{L}{D} \frac{V^2}{2} \rho$$

Where:

Δp Pressure drop

f Friction factor

L Length of pipe

D Diameter of pipe

V Velocity

ρ Density

The friction factor is a function of Reynolds number and its can find from Moody chart after calculating Reynolds number, which can find by:

$$R_e = \frac{d\ \rho\ V}{\mu}$$

Where:

R_e Reynolds number

d Inside diameter of the pipe

ρ Density of refrigerant

μ Viscosity of refrigerant

V Average refrigerant velocity

3.5.2 Polytropic Compression

In ideal vapor compression cycle, it's assumed that the compressor is compressed the refrigerant isentropic, that mean the entropy in the compression process remain constant

$$\Delta S = 0$$

This assumption is ideal process, but in actual compression process, the compression done with polytropic compression

$$pv^n = C$$

3.5.3 Subcooling

Condensed liquid refrigerant is usually subcooled to a temperature lower than the saturated temperature corresponding to the condensing pressure of the refrigerant, shown in Fig. 3.10 as point$3'$. This is done to increase the refrigerating effect. The degree of subcooling depends mainly on the temperature of the coolant during condensation, and the construction and capacity of the condenser.

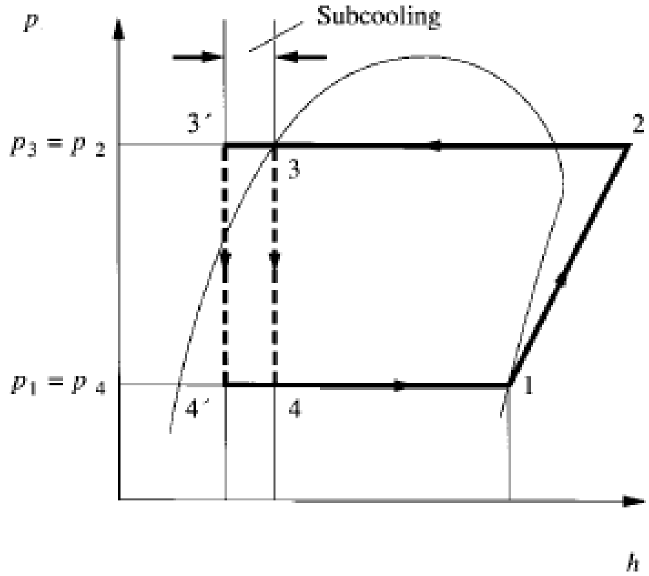


Fig. 3.10 Subcooling on ph diagram

The enthalpy of subcooled liquid refrigerant can be calculated as:

$$h_{sc} = h_{s,con} - c_{pr}\left(T_{s,con} - T_{sc}\right)$$

Where:

h_{sc} Enthalpy of subcooled liquid

$h_{s,con}$ Enthalpy of saturated liquid refrigerant at condensing temperature

c_{pr} Specific heat of liquid refrigerant at constant pressure

$T_{s,con}$ Saturated temperature of liquid refrigerant at condensing pressure

T_{sc} Temperature of subcooled liquid refrigerant,°

3.5.4 Superheating

Superheating done in ant vapor compression cycle when the refrigerant which leave the evaporator temperature are higher than the saturation vapor temperature as shown in Fig. 3.11

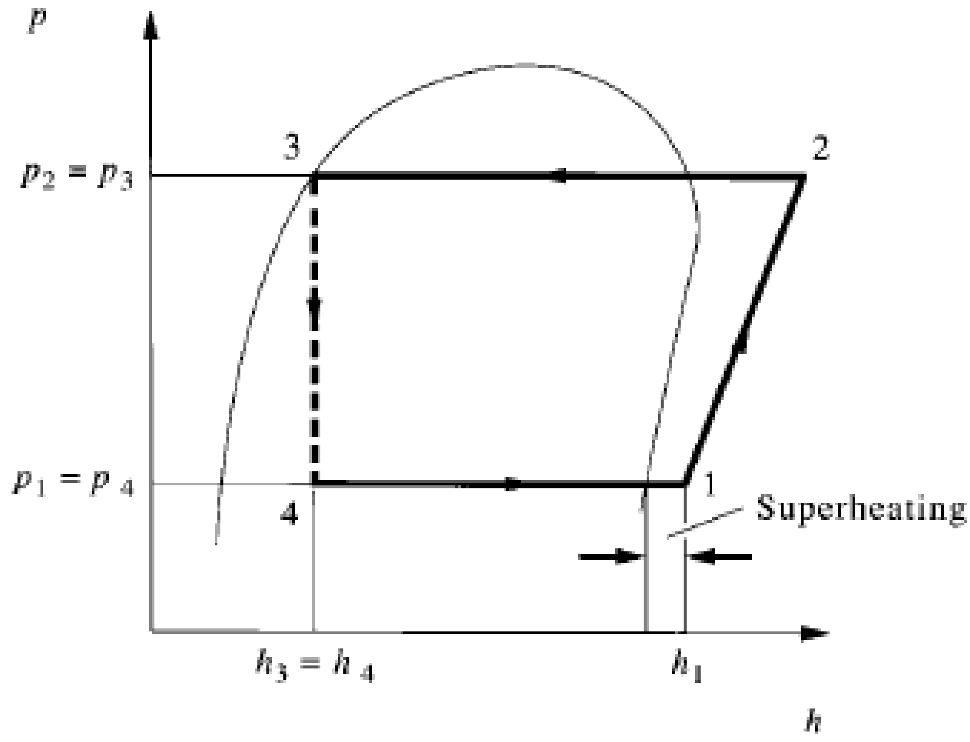


Fig. 3.11 Superheating in suction line

CHAPTER 4
REVIEW ON EVAPORATIVE COOLING

4.1 Introduction

Evaporative coolers, often called "swamp coolers", are cooling systems that use only water and a blower to circulate air. When warm, dry (unsaturated) air is pulled through a water soaked pad, water is evaporated and is absorbed as water vapor into the air. The air is cooled in the process and the humidity is increased.
It’s one of the oldest methods of cooling that known by human, Frescoes or plaster paintings from about 2500 B.C. show slaves fanning jars of water to cool rooms for royalty. The earliest archeological trail of buildings incorporating mechanisms for evaporative space cooling starts in Ancient Egypt with paths spreading quickly to other regions having hot and dry climates.

4.2 Evaporative Cooling Applications

Evaporative cooling reduces the dry-bulb temperature and increases the relative humidity of the air It is most commonly applied to dry climates or to applications requiring high air exchange rates like kitchens, laundries, agricultural, and industrial applications.

4.3 Evaporative cooling components

4.3.1 Pads

Cooling pads are used in evaporative cooling systems to Impedes the water and give the water more time and area to contact with air, there is many types of cooling pads using in evaporative cooling, and every type have individual working efficiency and selecting the pad type are effect to the system efficiency.

The aspen wood fiber pads Fig. 4.1 encased in chemically treated cheesecloth is widely used option. Literally wood fibers are packed together loosely, which offer the least amount of resistance to air flow through the cooler.

Fig. 4.1 Aspen wood fiber pad

Water was introduced across the top of this pad and air was pulled through it by a blower. These types of pads are economically.

A cellulose fiber media or a slab as shown in Fig. 4.2. This media is said to be uniform throughout, to provide consistent cooling performance and to last for several seasons. This type of pad is more expensive than the traditional aspen wood fiber pad but is designed to last for several years if the cooler is operated in compliance with the owner's manual.

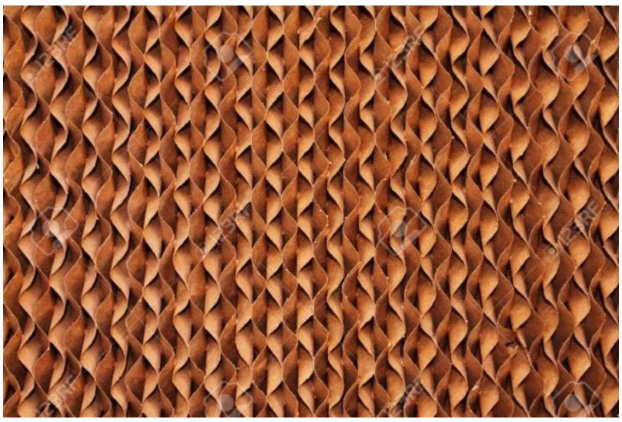


Fig. 4.2 Cellulose fiber media

The amount of pad area needed depends upon several factors including the type of pad material used. The pads should be continuous along the entire length of the wall. If aspen pads are used, it is recommended that one square foot of pad be provided for each 140 cubic feet per minute (CFM) of air moved by the fans.

Cellulose pads can be used with airflows of up to 230 cubic feet per minute per square foot of pad. The higher airflow rates of cellulose pads, means that fewer area of pad area is needed than if aspen pads are used.

4.3.2 Fan

Fans are used in evaporative cooling to forced air inside the cooler to contact with circulated water. Two general types of fans are used in evaporative cooling equipment:

- **Axial fan**

In an axial fan Fig. 4.3 the air flows in parallel to the shaft. It is common to classify axial fans upon their wheel like:

1. C-wheel - Blades can be adjusted when running. High efficiency, small dimensions, variable air volume.
2. A-wheel - Blades can be adjusted only when the fan is standing still. High efficiency, small dimensions, adaptive to recommended air volume.
3. K-wheel - Blades cannot be adjusted. Simple and small dimensions.

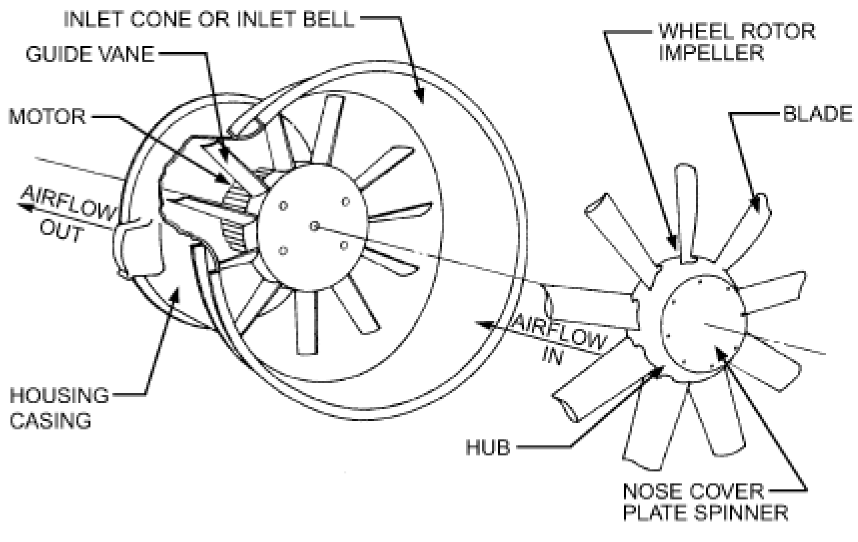


Fig. 4.3 Axial fan

- **Centrifugal fans**

In centrifugal fan Fig.4.4 the air flows is in a radial direction relative to the shaft. Centrifugal fans can be classified by their wheel like:

1. F-wheel - Curved forward blades. High efficiency, small dimensions, changing in pressure has little influence on pressure head.
2. B-wheel - Curved backward blades. High efficiency, low energy consumption, changing in pressure has little influence on air volume. Low noise emission, stable in parallel running.
3. P-wheel - Straight backward blades. High efficiency, self-cleaning, changing in pressure have little influence on air volume
4. T-wheel - Straight radial blades. Self-cleaning.Suitable for material transport.

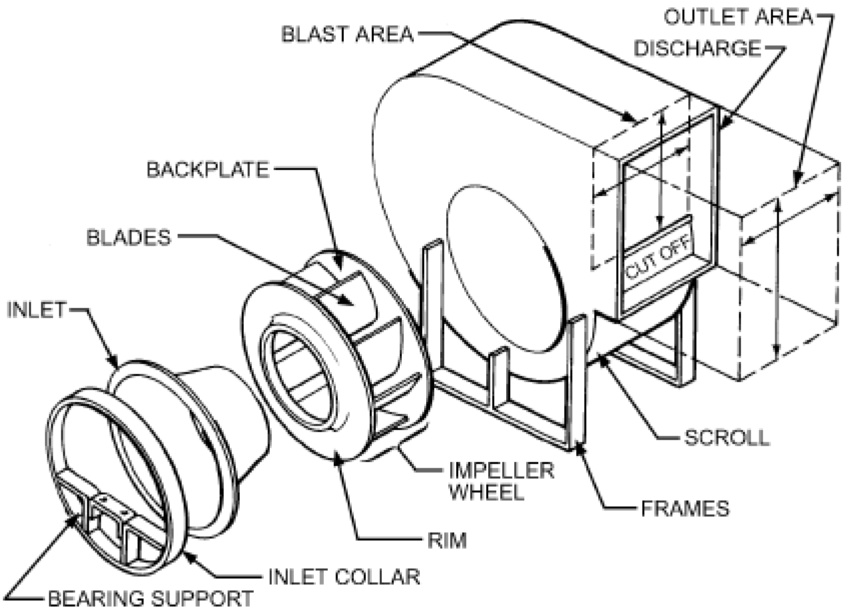


Fig. 4.4 Centrifugal Fan

4.3.3 Nozzle

Some of small evaporative cooling systems didn't use the nozzles, in this case water will supply on the top of cooling pads by multi hole piping system.

In large units and evaporative condenser units, nozzles are used to spray the water on the cooling pad, it's very efficient to use the nozzle in this type of cooling systems, water when got spray, more amount of water will contact with air, in Fig. 4.5 a common usage nozzle are shown.

Fig. 4.5 Common evaporative cooling nozzle

4.3.4 Water pump

Water pumps are used in evaporative cooling to circulate the water in the system, it's should select according to needed water flow rate, if the water flow rate are less than needed, a reducing in cooling efficiency will appear, and if high flow rate pump are select, the system will consume more energy, rather than an high water circulation will fold the cooling pad, and it may cause losing sprayed water with exhausted air. In Fig. 4.6 (a) and (b) a sample of water pumps in large and small evaporative system respectively.

(a) (b)

Fig 4.6 Water pumps in evaporative cooling

4.4 types of evaporating cooling

There are two main types of evaporative , direct and indirect evaporative cooling, sometimes it's can be found a combined (direct and indirect) and for both types, the effectiveness depends on the low wet bulb temperature (WBT) in the supplied air stream.

- **Direct cooling**

In this type, water sprayed directly to the air stream, a blower pulls air through a permeable, water-soaked pad. As the air passes through the pad, it is filtered, cooled, and humidified. A recirculation pump keeps the media wet.

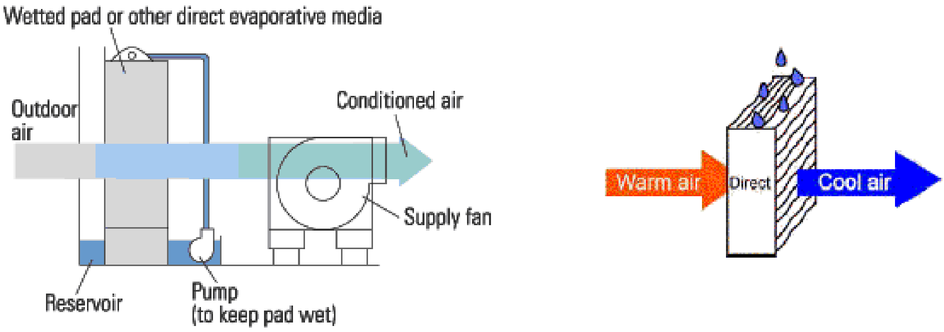


Fig 4.7 Direct evaporative cooling system

As shown in Fig. 4.8 the air blower will pull the air inside the evaporative cooler, a water pump will pumped the water to spray directly on pad. The efficiency of direct cooling depends on the pad media. A good quality rigid cellulose pad can provide up to 90% efficiency while the loose aspen wood fiber pad shall result in 50 to 60% contact efficiencies. The evaporative cooling efficiency can be calculated by equation:

$$\varepsilon_e = \frac{t_{db1} - t_{db2}}{t_{db1} - t_{wb1}}$$

Where:

ε_e Direct evaporation (saturation) effectiveness, %

t_{db1} Dry-bulb temperature of entering air, C°

t_{db2} Dry-bulb temperature of leaving air, C°

t_{wb1} Wet-bulb temperature of entering air, C°

- **Indirect evaporative cooling**

Indirect evaporative cooling as in Fig. 4.8 lowers the temperature of air via some type of heat exchanger arrangement, in which a secondary airstream is cooled by water and which in turn cools the primary airstream. The cooled air never comes in direct contact with water or environment.

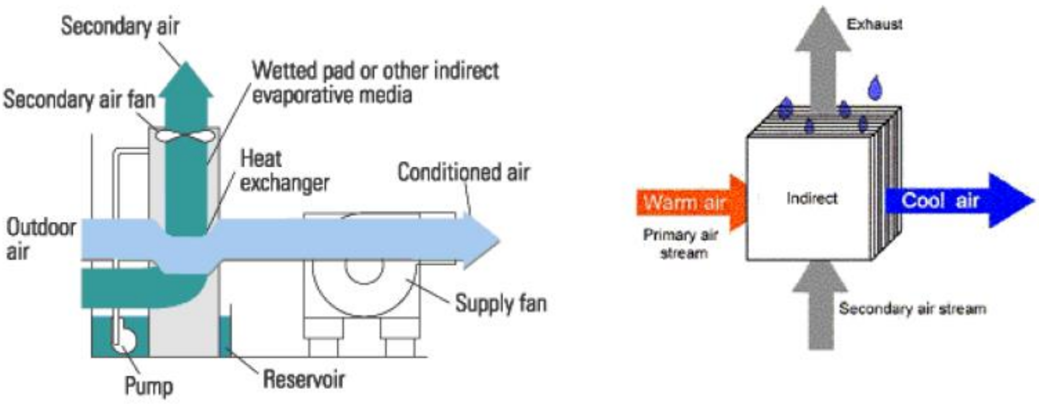


Fig. 4.8 Indirect evaporative cooling system

Indirect evaporative coolers do not add humidity to the air, but cost more than direct coolers and operate at a lower efficiency. The efficiency of indirect cooling is in the range of 60-70%. The evaporative cooling efficiency can be calculated by equation:

$$\varepsilon_{ie} = \frac{t_{db1} - t_{db2}}{t_{db1} - t_{wbs}}$$

Where:

ε_{ie} Indirect evaporative cooling effectiveness, %

t_{db1} Dry-bulb temperature of entering primary air, C°

t_{db2} Dry-bulb temperature of leaving primary air, C°

t_{wbs} Wet-bulb temperature of entering secondary air, C°

- **Two stage (combined) evaporative cooling**

Two stage evaporative coolers as shown in Fig. 4.9 combine indirect with direct evaporative cooling. This is done by passing air inside a heat exchanger that is cooled by evaporation on the outside. In the second stage, the pre-cooled air passes through a water-soaked pad and picks up humidity as it cools.

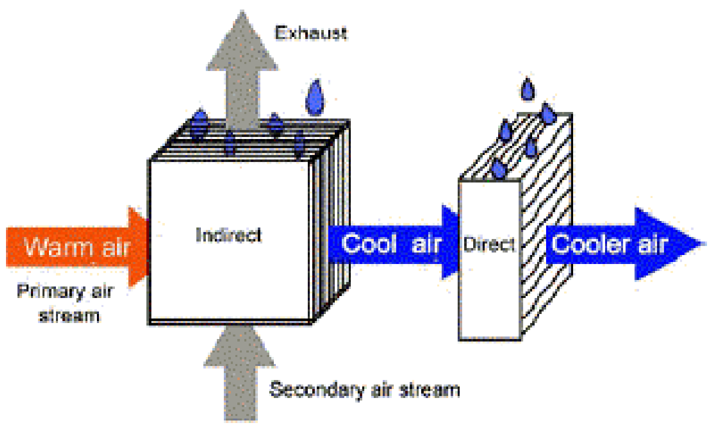


Fig. 4.9 Two stage evaporative cooling

4.5 Evaporative cooling process

The evaporative cooling is an adiabatic process, in another term its constant enthalpy process, in Fig 4.10 the line AB in psychometric chart represent to the evaporative cooling, both temperature and humidity are change.

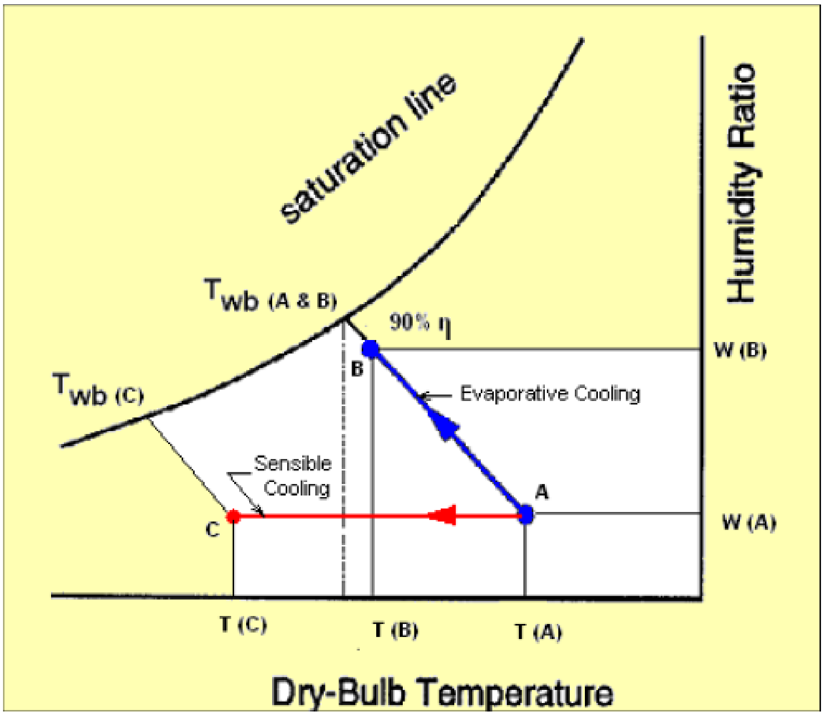


Fig. 4.10 Evaporative process on psychometric chart

In evaporative cooling process, changes occur in dry bulb temperature, specific volume, relative humidity, humidity ratio, dew point temperature, and vapor pressure of the moist air. No change occurs in wet bulb temperature.

4.6 Water Evaporation rate

Water evaporation is the main factor of any evaporative cooling system, spatially when city main water line are used, water cost changed from country to others, but still water consume is main operating cost in evaporative cooling.

The water consumption rate due to evaporation varies depending on the air flow rate, the temperature and humidity of the outside air and the pad characteristics.
Water evaporation rate can be calculated by

$$\dot{m}_{e} = \rho \dot{V} (w_2 - w_1)/1000$$

Where:

$\dot{m}_e$ Water consumption rate, kg/hr

ρ Air volumetric flow rate, m3/hr

$\dot{V}$ Air density, 1.2041 kg/m3

w_1, w_2 Humidity ratios of entering and leaving air, g moisture/kg dry air

CHAPTER 5

EXPREMANTAL SETUP AND THEROTICAL ANALYSIS

5.1 Introduction

The objective of this chapter is clearly mention of experimental setup to adding evaporative cooling system with pre- cooled discharge line on vapor compression system (split air conditioner). The system installed on the engine laboratory of mechanical engineering department in Gaziantep University.

5.2 Experimental system and setup for evaporative cooling

The experimental setup about adding evaporative cooling system to decrease the inlet air temperature of the split air conditioner condenser, in addition a modifying of air conditioner pipe line made to cold the discharge line before interring the condenser.

5.3 Testing zone

Testing rooms have dimensions of (450x720) cm the room has four different walls; there was three walls and roof supply on unconditioned spaces, while on wall supplied on outside condition. Fig. 5.1 shows the testing room details.

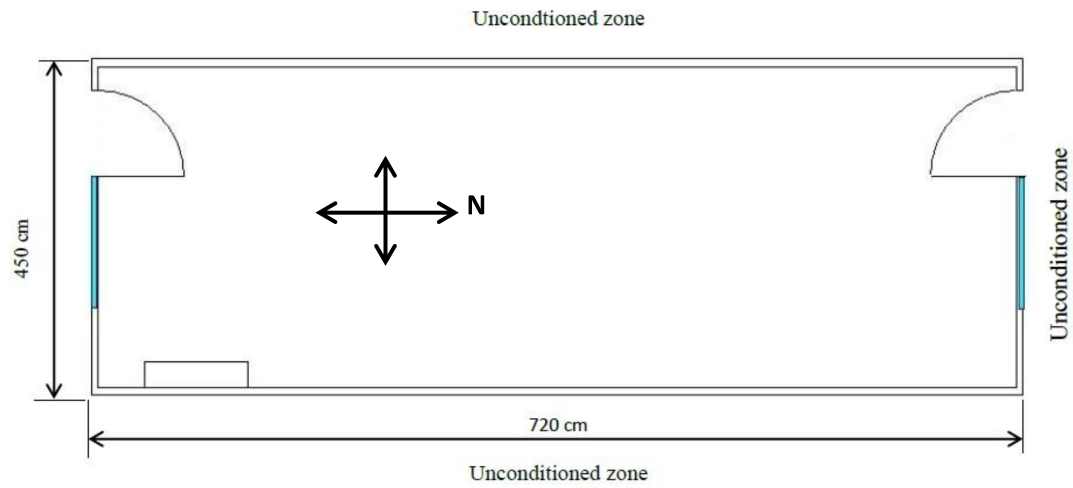


Figure 5.1 Test room detail

The all walls construction was built from brick without any plasters from both inside and outside of the walls.

5.4 The combination of vapor compression and evaporative cooling

An existing split-air-conditioner (24000 BTU/h) was used in the experiments. Consistent with the shape of the condenser, a frame was built and filled by media pad with 5 cm thickness and installed in front of the condenser as shown in Fig 5.2, for the water circulation system, a small water pump (52 W) used to circulate the water in fixed flow rate by using house connection, the water are injected by the pump on the top of the media pad, after water injected on bad, the water will accumulate in the primary tank which in width 15 cm and height 10 cm. The primary tank was made in same shape and profile of the condenser, and then the water accumulates in the secondary tank which it connects to the circulated pump.

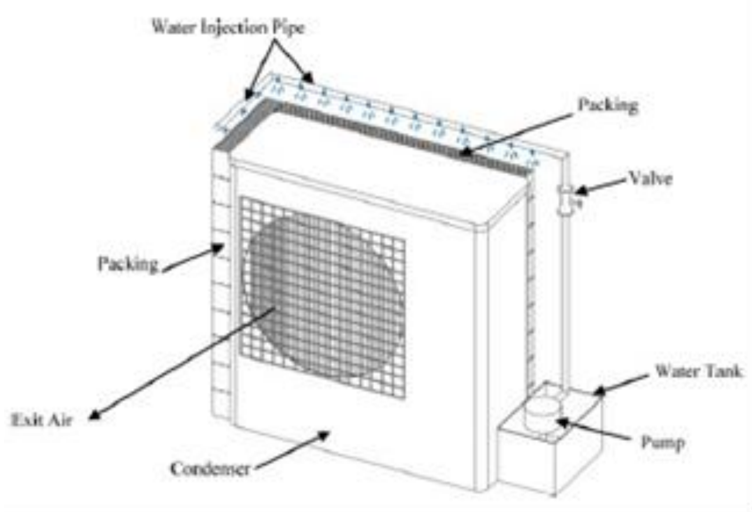


Fig. 5.2 Schematic of modified condenser unit

5.4.1 Adding evaporative cooling on system

The evaporative cooling system was made in the general workshop of mechanical engineering department. It's contained from three parts and connected together by arc welding, Fig. 5.3 shows the maid of the frame for evaporating part.

Fig. 5.3 The general frame of evaporative cooling

The first part is the lower part which used as primary water tank; it's in width 10 cm and height 6 cm, with a 1 cm diameter hole in the end of the tank as water drain, the hole made in 4 cm height from the bottom of the tank, so the primary tank contain water in all operating time.

The second part was 6 bars in 70 cm height welded on the outer face of the primary tank to hold the water distributer. Third part is the water distributor; it's made from 1.5 mm plate, and gives (V) shape by pressing machine , the water distributor have 20 holes in 2 mm diameter to separate water on the cooling pad.

A cellulose media cooling pad in 5 cm thickness was used in evaporating cooling system, the pad reinforced by wire mesh then connect to the frame by wires.

5.4.2 Adding pre-cooling discharge line

One of the main parts of this study is making pre-cooling for the discharge line, The discharge line are modified by extended copper pipe with same dimensions of the discharge line, the extended pipe modified to pass through the primary tank and flood with the cold water that generate from the evaporating cooling as shown in Fig. 5.4, a copper tube was used in same original discharge line of the system.

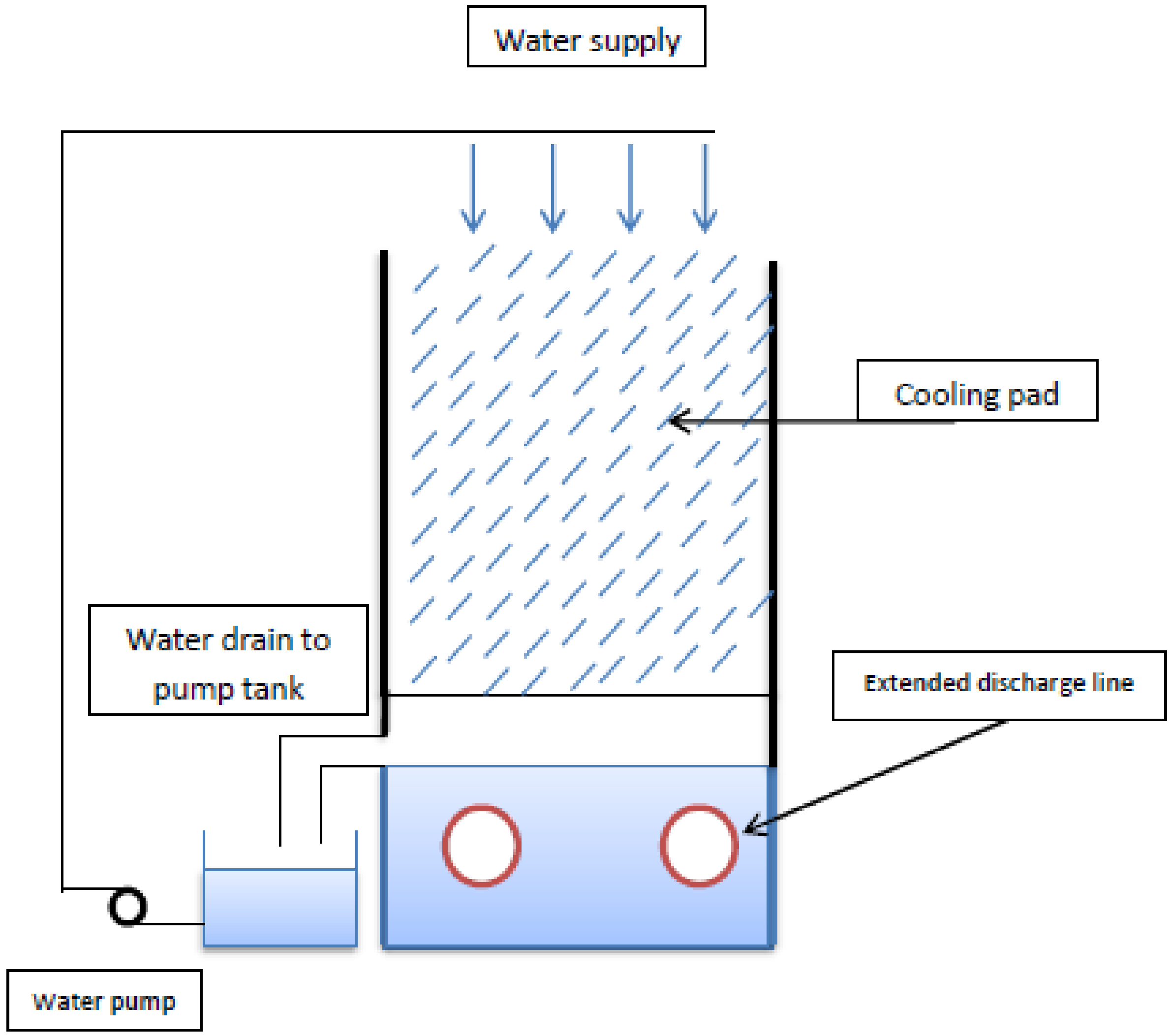


Figure 5.4 Side views for modified evaporative system

Two solenoid valves are connected on the modified piping system to control refrigerant flow as shown in Fig. 5.5, so three types of operations can get from the system. When the both solenoid valves 1 and two are closed the original operating can obtain, and when the solenoid valve 1 are open while the solenoid valve 2 are closed the novel design going work and we can get the pre-cooling for the discharge line. In this work, we make three kinds of operations which are the original operation, original operation with evaporating cooling, and modified pipe line with evaporating cooling operation.

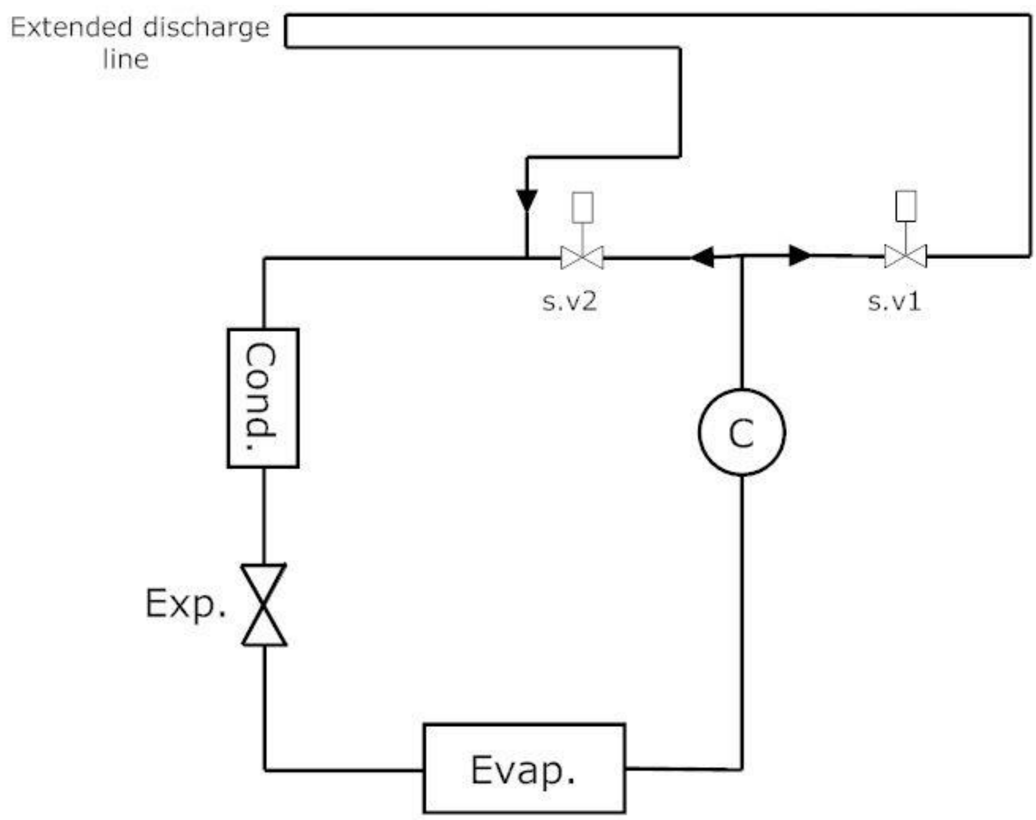


Fig. 5.5 Modified piping system

5.4.3 System charging

Due to modified discharge line, the system refrigerant are sucked, and after done from adding the solenoid valves, and all temperature, pressure measurement kits, we start to recharge the system.

Before recharging the system, an air pressure applied on the system (400 psi) to insure there is no leakage in welded connections, then pipe cleaner are applied on all pipes in the system to insure there is no dust or welding slag inside the pipes.

After that a system vacuumed by vacuum pump, and using digital manifold gauge, digital weight scalar, the system had been charged by R-22 refrigerant, figure 5.6 shows the refrigerant charging operation.

Figure 5.6 System charging

5.5 Measurement equipment

Measurements should to obtain thermal performance for each part of the system, temperature and pressure for the refrigeration cycle, and power consumed by water circulation pump and air conditioner compressor. Figure 5.7 shows the temperature and pressure measurement equipment on system.

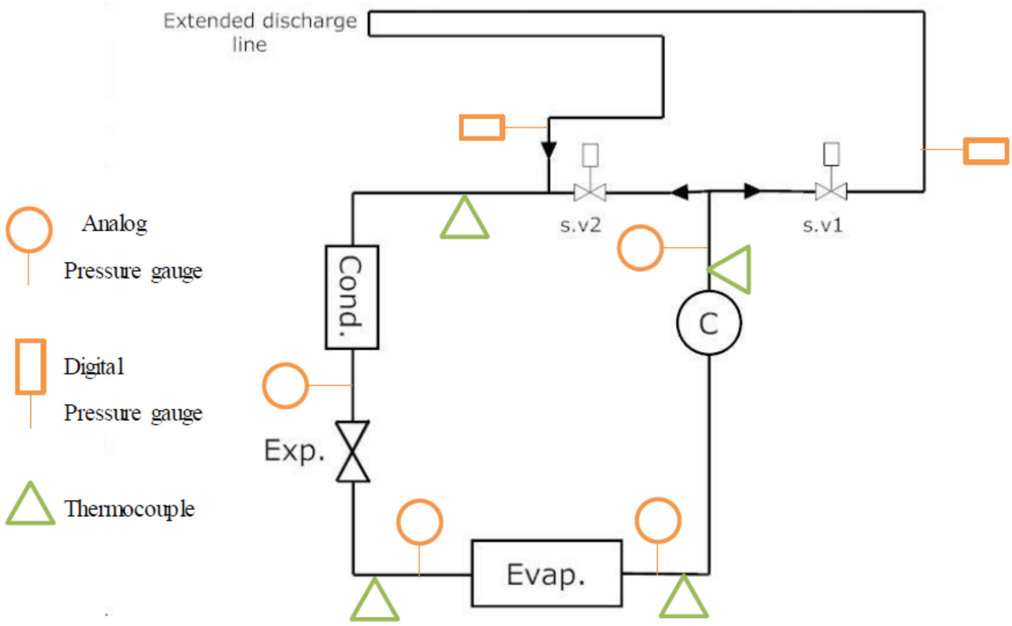


Figure 5.7 Measurement equipment on system

5.5.1 Pressure measurement

Pressure measurement in the system, at condenser and evaporator inlet and exit, two pressure gauges were installed on the inlet and outlet of the extended discharge line to measure the pressure drop. The pressure transmitter was CAREL SPKT0031C0 coded piezo-resistive, it's have 0-30 pressure measurement range and 4-20 mA output. Figure 5.10 shows a schematic of the pressure probe develop by CAREL for the application in the refrigeration and air conditioning.

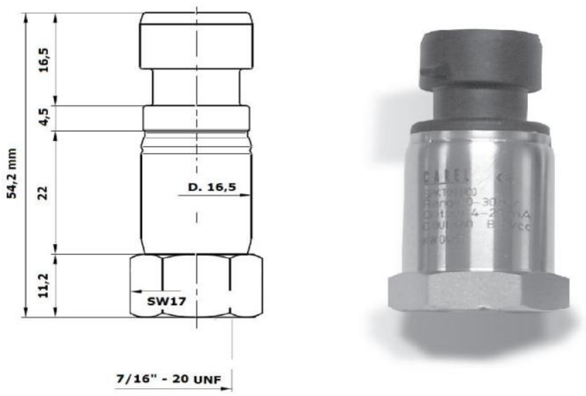


Figure 5.8 Pressure transmitter

The pressure transmitter has specifications as:

- Power supply: 8-28 V DC
- Output:4-20 mA
- Linearity: $\pm$ 0.5% FS typical $\pm$ 1% FS max

A high accuracy analog oiled gauges are connect with the system which it made especially for R-22 to calculate the degree of subcooling and superheating in the system, the accuracy of the gauge is 0.5%.

5.5.2 Temperature measurement

All nodal temperatures on the system, ambient air, inlet room air, and point after the cooling pad temperatures were measured using K-type thermocouples, each thermocouple were connected with display high accuracy screen having six digits.
For the nodal temperature on the system as shown in figure 5.9, before temperature measurement, the surface of the tube was polished for removing any dust or rust and then the thermocouple probe was laid down on the surface. In order to reduce the thermal contact resistance between thermocouple probe and tube surface, thermal grease was used in the point of contact. Insulating tape was wrapped around the probe to make good contact and also prevent any convection effect of ambient air on the temperature readings as shown in figure 5.9.

Figure 5.9 Insulating of contact point between pipe and thermocouple

5.5.3 Voltage and current measurement

Measurement of the compressor current is one of the most important parameter in this study, to calculate the power consume by the compressor, and since the power is function of current and voltage, a voltmeter are used also in measurement, due to the fixed power consumption by air conditioner fan and circulation water pump, the current measurement equipment are applied on the compressor only.
The current and voltmeter was manufacture by KYORITSU model KEW2200R and all specification and accuracy data are mentioned in appendix A.

5.6 Measurement equipment calibration

It's known that calibration is very important for correct measurements of the performance parameters, thermocouple and pressure meter calibration will explain in this section.

5.6.1 Pressure calibration

Carel SPKT0031C0 pressure transmitter, a data logger allowed to make scan for 30 second for each measurement, and using calibrated voltmeter to take direct readings, for each scan, fabric calibration result was done,0-10 vdc output of transmitter was changing linearly with change voltmeter reading, it's found the error rate was about 2% .

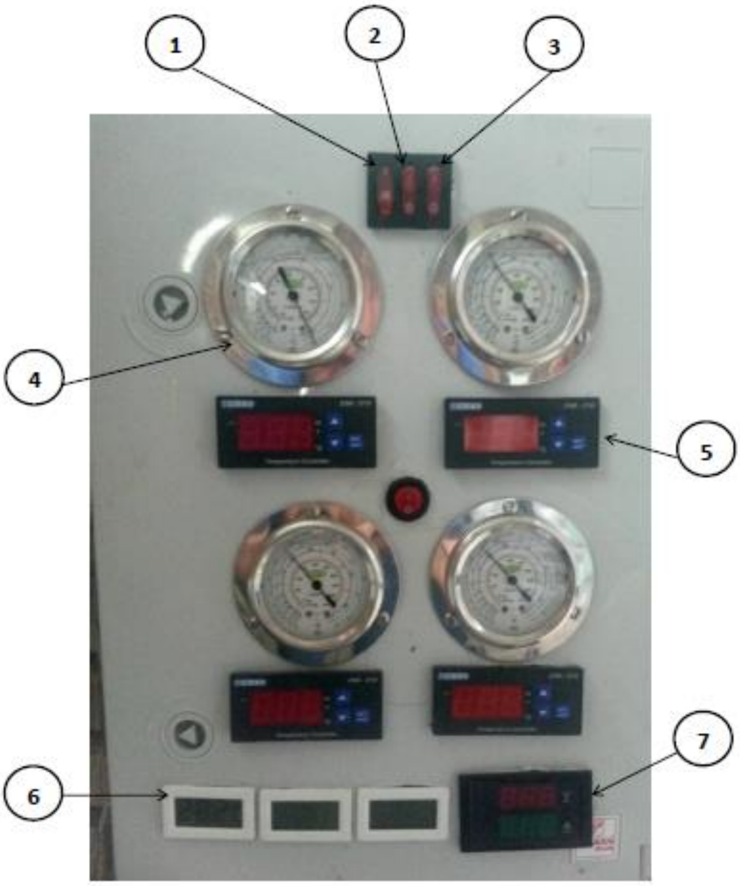
For the analog pressure gauge, the two types of analog gauge (high and low pressure types) are connected to a small steel cylinder, the calibrated pressure transmitter also connected with the cylinder, using a vacuumed pump, the cylinder vacuumed then then, and making a comparing with the pressure transmitter and analog pressure gauge, we made the calibration with adjusting escrow which it on the analog pressure gauge, for the high pressure analog gauge, same procedure had been done, but this time by rising the pressure in the cylinder by small compressor.

5.6.2 Temperature calibration

A water tank with small mixer fan and heater element were used to calibrate the thermocouples, an thermostat connected to the heater element to control the water temperature, using mercury glass thermometer having sensitivity of 0.1 °C, water start to heat by heater elements, to an individual temperature, when the thermostat turn-off ,the element and mixer fan was kept working, a compering between the thermocouple and glass thermometer reading were taken, the test were repeat three times for each reading, same think were made for lower temperature reading using ice mixture with salt, the calibration test shows that there were $\pm$ 0.5 °C different from mercury glass thermometer.

5.7 Control and data panel

The high accuracy digital display were connected with the thermocouples, analog pressure gauges, digital voltage and current meter, the solenoid valves and circulation water pump switches are combined in one control panel as shown in figure 5.10.



1 first solenoid switch	5 thermocouple digital screen
2 second solenoid switch	6 digital thermometers
3 water pump switch	7 volt & current digital screen
4 analog pressure gauge	

Figure 5.10 Control and data board

5.8 Extra setups

According to the Turkey Meteorological authority, and our weather parameter measurement for Gaziantep city, the average maximum temperature was 40°C, while the relative humidity was in (40%-50%), and due to the temperature average are larger than 40°C in some other areas in Turkey, and Middle East, we make our maximum temperature in the testes 45°C, and this done by adding a small room electric heater behind the outside air conditioner unit, the distance between heater and the air conditioner was not fixed, we adjust the distance so we get the right inlet air temperature.

For the same reason, we want to make the testes in a relative humidity larger than the normal averages, to insure the modified system performance in high relative humidity, and this done by spraying water on the air behind the air conditioner, the amount of water sprayed and the distance between the spray and air conditioner was adjusted to get 80% inlet relative humidity as shown in figure 5.11a and b.

a b

Figure 5.11 Spray water by air humidifier

5.9 Theoretical analysis

5.9.1 Effect of adding evaporative cooling on system

For any small air-conditioner which it's work on compression cycle, as shown in the figure 5.12, Circulating refrigerant vapor enters the compressor and is compressed to a higher pressure, resulting in a higher temperature as well. The hot, compressed refrigerant vapor is now at a temperature and pressure at which it can be condensed and is routed through a condenser. Here it is cold by air flowing across the condenser coils and condensed into a liquid. Thus, the circulating refrigerant moves heat from the system and the heat are carried away by the air.

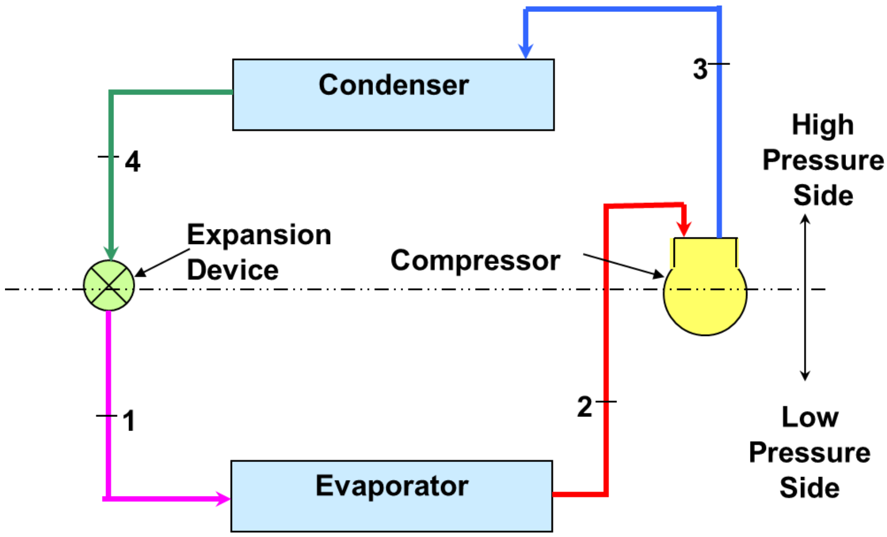


Figure 5.12 Working principle of compression cycle

The condensed and pressurized liquid refrigerant is next routed through an expansion valve where it undergoes an abrupt reduction in pressure. That pressure reduction results in flash evaporation of a part of the liquid refrigerant, lowering its temperature. The cold refrigerant is then routed through the evaporator.

By applying these data on ph. diagram (figure 5.13) we can see the four proses on the chart we find

- **Process 4-1**: two-phase liquid-vapor mixture of refrigerant is evaporated through heat transfer from the refrigerated space.
- **Process 1-2**: vapor refrigerant is compressed to a relatively high temperature and pressure requiring.
- **Process 2-3**: vapor refrigerant condenses to liquid through heat transfer to the cooler surroundings.
- **Process 3-4**: liquid refrigerant expands to the evaporator pressure.

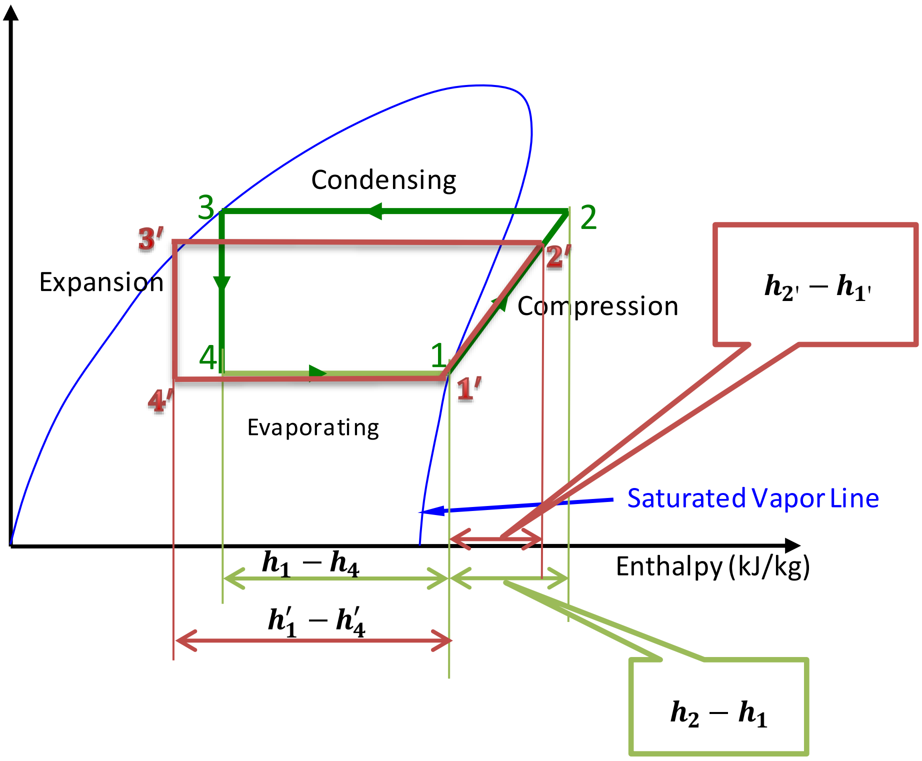


Figure 5.13 Simple compression systems on ph diagram

Applying evaporative cooling on condenser of the air-conditioner will decrease the temperature of the condenser from the presses 2-3 to 2′-3′ as shown in the figure 5.12 and the new cycle will appear as 1′-2′-3′-4′.

Since the compressor work is

$$W_c = \dot{m}\, \Delta h \tag{5.1}$$

Then:

$$W_{original} = \dot{m}\,(h_2 - h_1) \tag{5.2}$$

And

$$W_{\text{modified}} = \dot{m}\,(h_{2'} - h_{1'}) \tag{5.3}$$

Due to the decreasing in condenser pressure and temperature of superheated refrigerant when a evaporative cooling adding on the air conditioner condenser, at the point (2), the enthalpy change in conventional system is less than the enthalpy changed in original system referring to equations (5.1), (5.2), and (5.3), an decreasing in compressor power consumption will done after applying the evaporative cooling.

Referring to figure 5.13, we can see that the enthalpy change in the evaporator is rise when evaporative cooling is applied on condenser comparing with the enthalpy change in original operation.
Since cooling capacity

$$Q_e = \dot{m}\,\Delta h \tag{5.4}$$

For the original operating:

$$Q_{e.original} = \dot{m}\,(h_1 - h_4) \tag{5.5}$$

And for modified operation:

$$Q_{e.modified} = \dot{m}\,(h_{1'} - h_{4'}) \tag{5.6}$$

The increasing in enthalpy at the point (4) make the enthalpy change value increase, which will lead to increasing in cooling capacity of the system.

5.9.2 Effect of adding pre-cooling discharge line

The main aim of this study is to evaluate how the pre-cooling will affect one the air-conditioner if the pre-cooling applied on discharge line of the compressor. The cooling will done by extend discharge line and pass it through cold water, evaporative cooling will apply to the outside of the air conditioner, and due to evaporation of water, like air washer systems, it's will get cold air which it pass through the air-conditioner condenser, and as a result of this proses a cold water will produce which will used to cool the discharge line.

In any simple refrigeration cycle which it's working based on compression method, the compressor, take the refrigerant from low pressure side of the system (evaporator) by suction line and compress it to condenser in high temperature and pressure (proses 1-2) as shown in ph diagram in figure 5.14

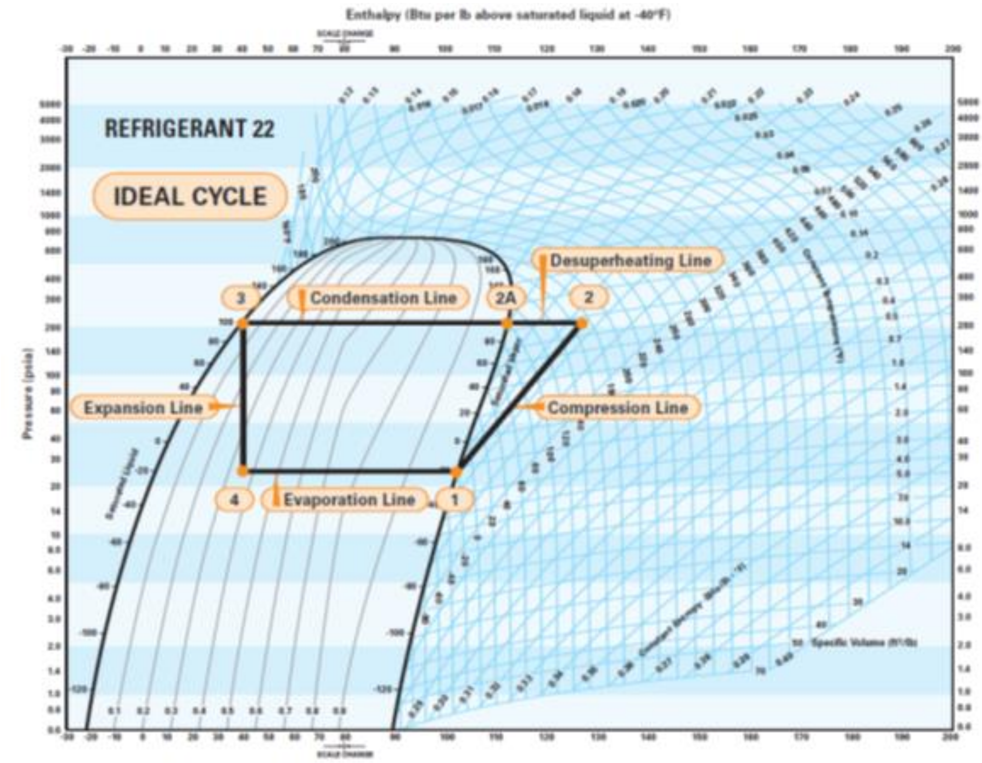


Figure 5.14 Simple compression refrigerant cycle on Ph diagram

The superheated refrigerant at point 2 enter to the condenser, which it convert to saturated liquid at point 3, in this process the superheated gas should be change to saturated gas first at point 2A, this conversion done in the first third of condenser, the discharge line pre-cooling will make the superheated gas change to saturated gas before entering to condenser, then the condenser effectiveness will increase.

The pre-cooling idea in all small air-conditioner are applied simply by extending the discharge line, so the superheated gas can take more area to transfer the heating from the discharge line before entering the condenser as shown in figure 5.15

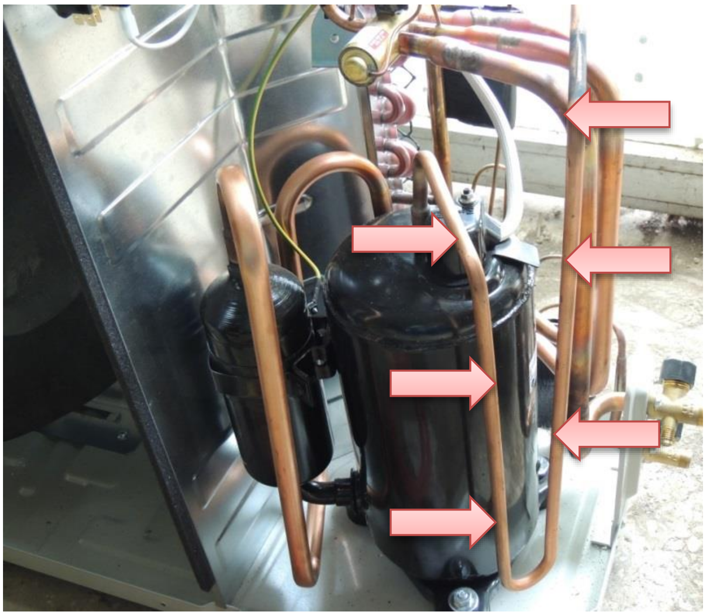


Figure 5.15 Discharge line extensions in small air-conditioner

The advantage and disadvantage of extend the discharge line are study using the following formulas:

(A) Advantage of the pre-cooling compressor discharge line is concentrated in heat transfer from the hot discharge line to the cold water.

Applying heat balance on the discharge extended by which it flooded by water to find the existed refrigerant temperate (T_2):

$$\int_{T_1}^{T_2} \dot{m}_g \, cp_g \, dT = \int_0^A U \, dA \, (T - T_w) \tag{5.7}$$

$$\ln \frac{T_2 - T_w}{T_1 - T_w} = \frac{-\,U\,A}{\dot{m}_g \, cp_g}$$

$$\frac{T_2 - T_w}{T_1 - T_w} = e^{\frac{-\,U\,A}{m_g \, cp_g}} \tag{5.8}$$

For heat transfer from pipe to cooling water the weather of flow type examined by computing Reynolds number using equation (5.9):

$$Re = \frac{V\,D\,\rho}{\mu} \tag{5.9}$$

The flow were found turbulent flow type, using equation (5.10) to find Nusselt number

$$Nu = 0.023 Re^{0.8}\ Pr^{0.3} \tag{5.10}$$

The convection heat transfer coefficient for refrigerant found from equation (5.11)

$$h = \frac{Nu\,K}{D} \tag{5.11}$$

The amount of overall heat resistance found from equation (5.12)

$$\sum R = \frac{1}{h_g\,A_i} + \frac{1}{h_w\,A_o} \tag{5.12}$$

Finally the heat transfer from the extended discharge pipe to the water is compute by equation (5.13)

$$Q = \frac{T_s - T_w}{\sum R} \tag{5.13}$$

While thickens of the pipe is too small, and made from copper which have high coefficient conduction, the temperature of the pipe surface (T_s) are assumed equal to the refrigerant temperature

(B) Disadvantage of extending the compressor discharge line is the pressure loses in pipe, which effect on the performance and power consumption by the compressor, the pressure drop in the extended discharge pipe are computed using equation (5.14)

Pressure drop per 1 meter = $f\ \frac{1}{D}\ \frac{V^2}{2}\rho$ (5.14)

Where the friction factor (f) found from Moody diagram after testing the flow type. The pressure drop computed at all testes and it's evaluated in the result computing which will explain in Chapter 6

For the pressure drop by the solenoid valves, the manufacturer information is mentioned in appendix B.

CHAPTER 6
RESULTS AND DISCUSSIONS

6.1 Introduction

In this chapter, the major performance parameters related with experimental system will presented in graphical and table forms. The performance are computed by using measurement values for experimental testes which done on a modified spilt air conditioner by applying evaporative cooling condenser and discharge line pre-cooling. The measurement values of ambient temperature, room temperature, pressure and temperature for refrigerant R-22 used in the air conditioning cycle at each inlet and outlet states of the system components, current and voltage during operating are collected to compute system performance.

The performance parameters of the added cooling pad are compute by measurement the dry bulb and wet bulb temperature at the inlet and outlet of the cooling pad.

All the tests are done in University of Gaziantep between July and August months, the temperature of ambient air was ranged between 32°C and 45 °C.

The results which it's mentioned in this chapter will covered the changing in the power consumed by compressor, cooling load, COP, and refrigerant mass flow rate of the system in three type operations:

1. The original system without any modifying.
2. The adding of evaporative cooling.
3. The pre-cooling discharge line with evaporative cooling.

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1. The original system without any modifying.
2. The adding of evaporative cooling.
3. The pre-cooling discharge line with evaporative cooling.

All these operations done in same temperature range with two relative humidity 45% and 80%, the testes repeated for each temperature and relative humidity three times, and the average measurements are used in computing the system performance.

The measuring results of pressure and temperature at all nodes are analyzed by digital enthalpy-pressure diagram as shown in figure 6.1

All measurements were taken after the system stated operation by 15 minutes so the system can reach the stable operation.

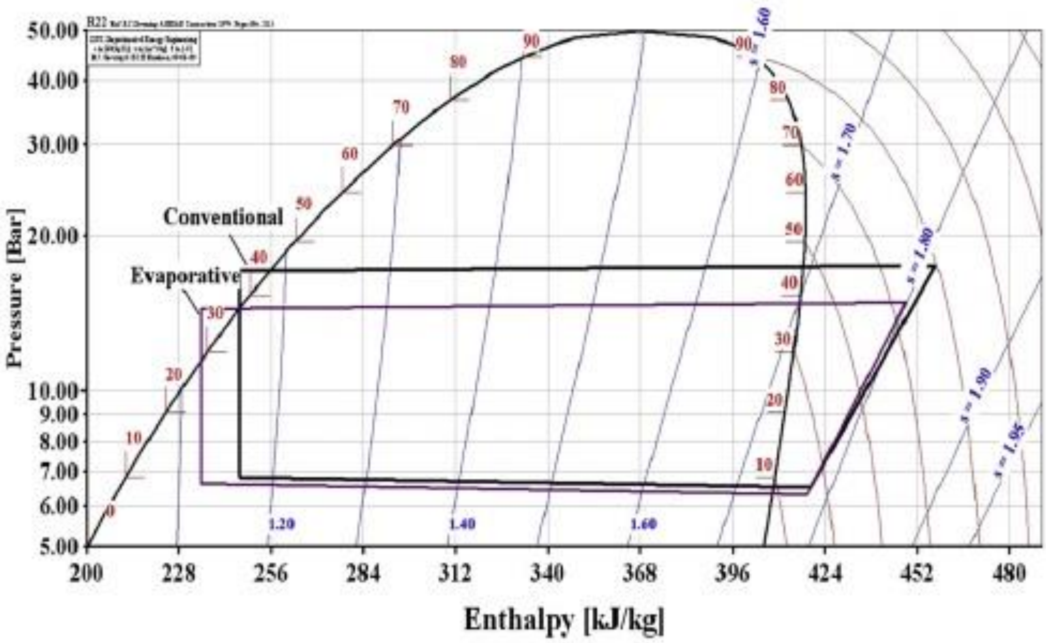

Figure 6.1 ph diagram analyzing

6.2 Experimental results for cooling pad performance

The cooling pad efficiency can be defined as in equation 6.1:

$$\varepsilon_{ie} = \frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{wb}} \tag{6.1}$$

Table 6.1 listed the measured cooling efficiency with the cooling pads fitted around the condenser at 45% RH. The average cooling efficiency was approximately 65%, while the efficiency was about 43% at 80% RH as mentioned in table 6.2. The figure 6.2 shows the efficiency of the cooling pad change by inlet air temperature and RH

Table 6.1 Effectiveness of the cooling pad at 45% RH

DBT in °C	WBT in °C	DBT out °C	Efficiency %
32	22.4	25.44	68.3
34	24	27.28	67.2
36	25.6	29.13	66
38	27.2	30.95	65.2
40	28.8	32.80	64.2
42	30.5	34.74	63.1
43	31.3	35.68	62.5
44	32.1	36.70	61.3
45	32.9	37.71	60.2

Table 6.2 Effectiveness of the cooling pad at 80% RH

DBT in °C	WBT in °C	DBT out °C	Efficiency %
32	28.9	30.6019	45.1
34	30.8	32.5696	44.7
36	32.7	34.5645	43.5
38	34.6	36.555	42.5
40	36.5	38.5545	41.3
42	38.4	40.5528	40.2
43	39.3	41.5385	39.5
44	40.3	42.5829	38.3
45	41.2	30.6019	37.1

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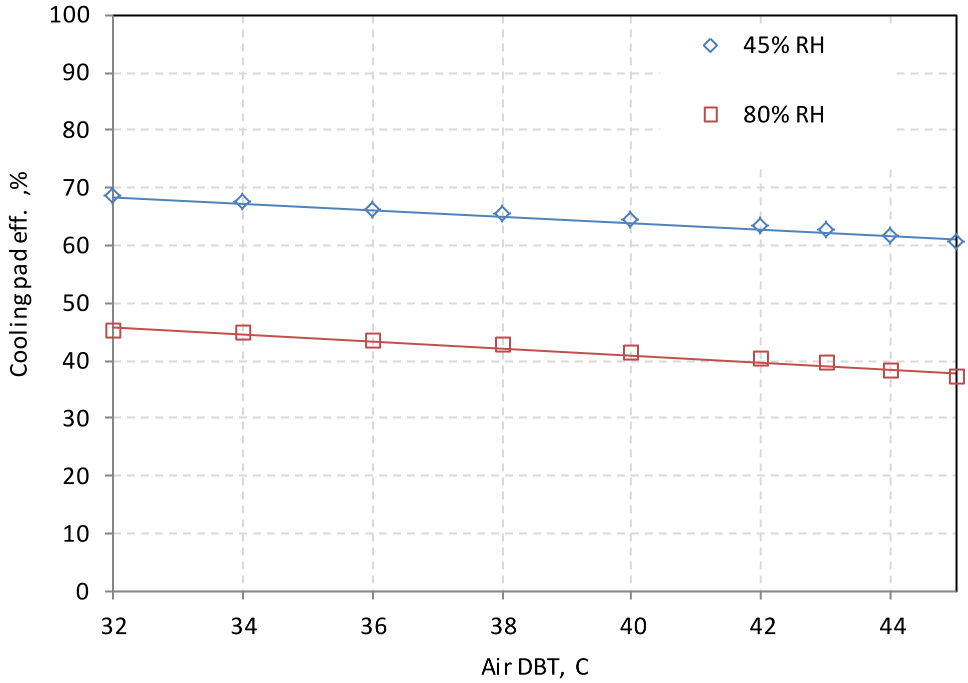


Figure 6.2 Cooling pad effectiveness

Water consumed by the evaporative cooler was compute; to determine the economic feasibility of the adding evaporative cooling on the system, the water consumed can be calculate by equation 6.2:

$$\dot{m}_e = \rho \dot{V}(w_2 - w_1)/1000 \quad (6.2)$$

Figure 6.3 shows the water consumed by the evaporative cooling at various temperatures and 45%.

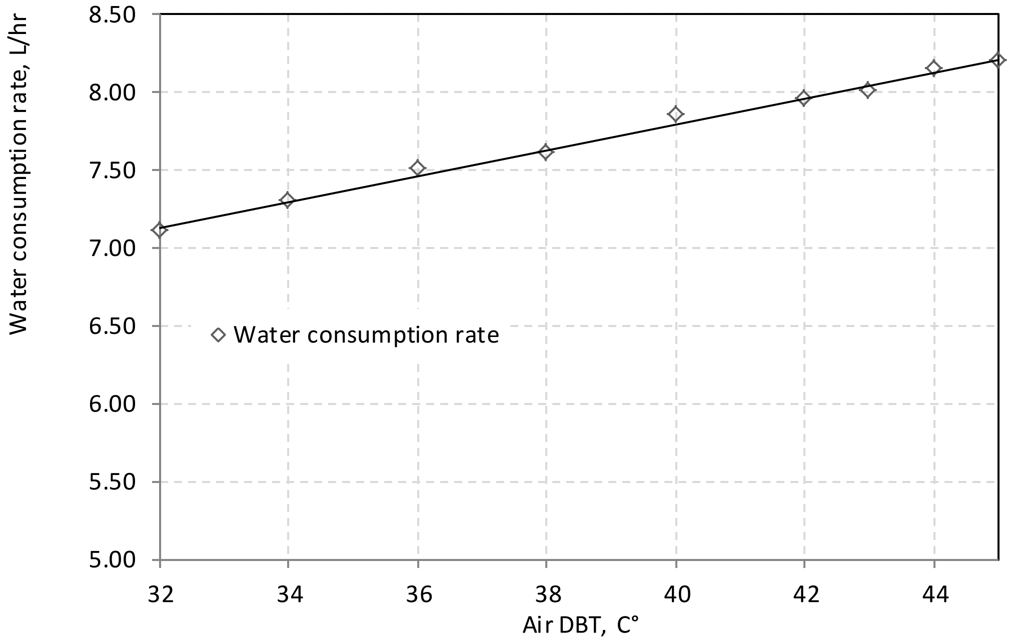


Figure 6.3 Water consumption rates

The maximum water consumption rate was 8.2 L/hr occurred at the maximum inlet temperatures 45°C, while the minimum water consumption rate was 7 L/hr at the 32°C.

6.3 Experimental results for cooling cycle

6.3.1 Power consumed by compressor

The power consumed by the compressor compute by measuring the electric current and voltage of the compressor, the power of the compressor can be found from the equation 6.3

$$W = I\ V\ cos\theta \qquad (6.3)$$

Figure 6.4 shows the power consumed by the compressor at the relative humidity 45%; it's found that the maximum power consumed by the compressor was at 45°C in original operation without adding cooling pad or pre-cooling for discharge line.

After adding the evaporative cooling on the air conditioner condenser, and at the same conditions (45°C DBT, 45%RH), it's found there are 16.5 % reduction in power consumption.

The adding pre-cooling discharge line at the same conditions (45°C DBT, 45%RH) made a power consume reduction by 6.6% of the evaporative cooling operation, that's mean the total power reduction after pre-cooling discharge line is 23.1%.

The power consumption by the compressor for other weather conditions is mentioned in the figure 6.5.

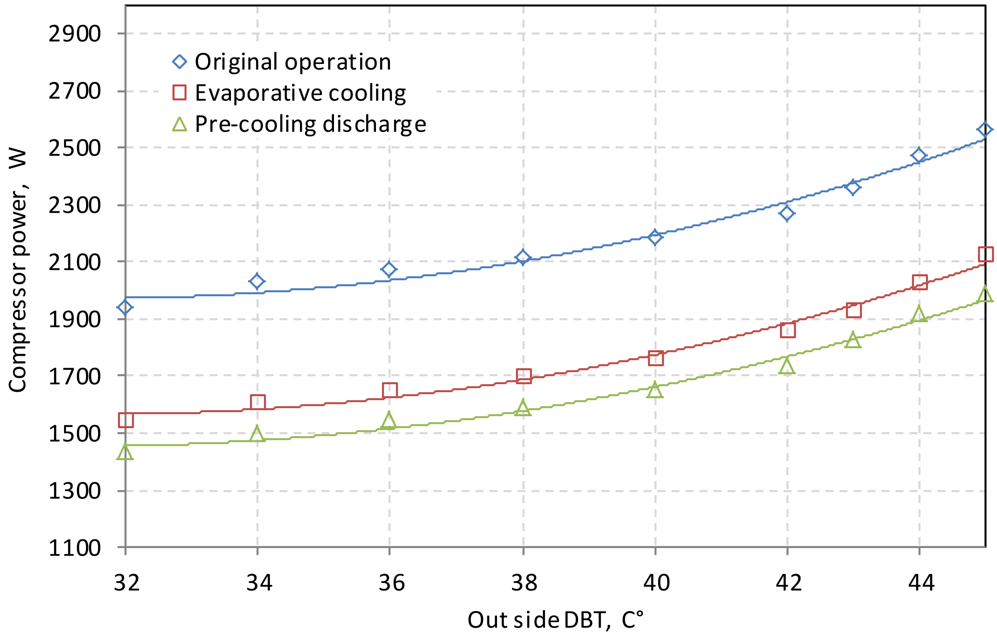


Figure 6.4 Compressor power consumption at 45%

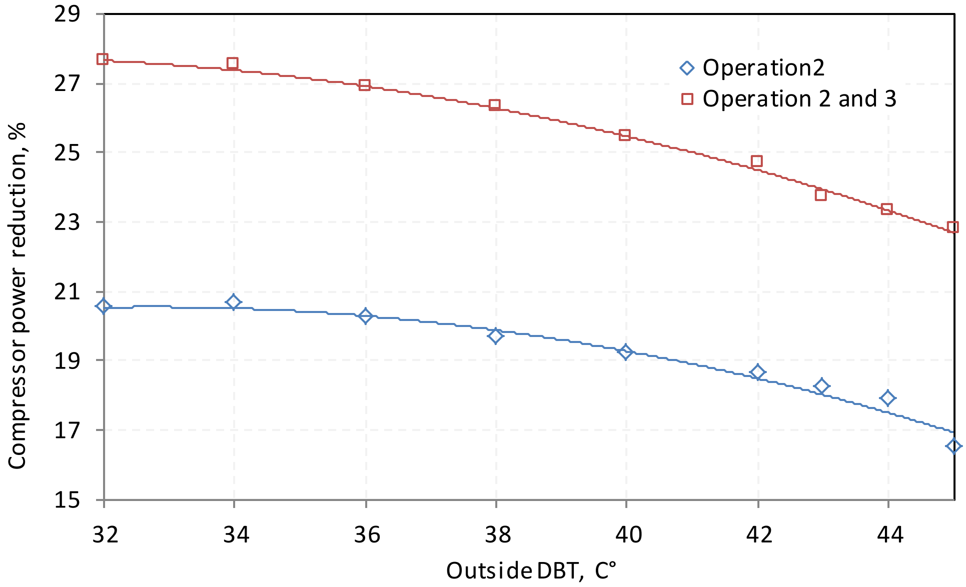


Figure 6.5 Compressor power reduction rates by outside DBT at 45 RH%

Figure 6.6 shows the power consumed by the compressor at high relative humidity 80% RH, the maximum power consumed by the compressor was at 45°C for the three operation types, in the original operating, the power consumed remain same comparing with the same DBT and 45% RH, because no evaporating cooling used in the original operating, and the condenser of the air conditioner performance don't changed with relative humidity change.

For second type operation (evaporative cooling operation) the maximum power consumed by the compressor was at 45°C, the power consumed reduction in the same temperature but in 80% RH was about 3.95%, while it was16.5 % in the same DBT but in 45% RH, this 12.55% reduction due to the high relative humidity which it caused to decrease the evaporative cooling efficiency then increasing in the inlet condenser air DBT.

For the discharge pre-cooling operation the maximum power consumed was at same condition (45 °C DBT and 80% RH), the power reduction was about 4.28%, while it was 6.6 % at (45 °C DBT and 45% RH), the reduction in this operation was about 2.32%, and it's less than the reduction ratio in the evaporative cooling operation, and this due to the water temperatures which it was used to pre-cooling the discharge line didn't decrease in big range. The power consumed by the compressor for other DBT was mentioned in the figure 6.7.

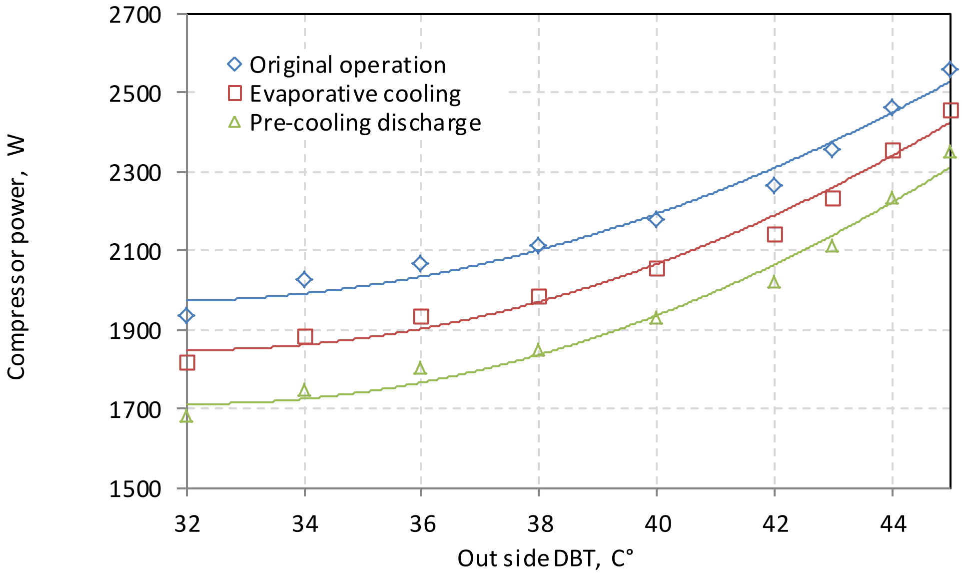


Figure 6.6 Compressor power consumption at 80% RH

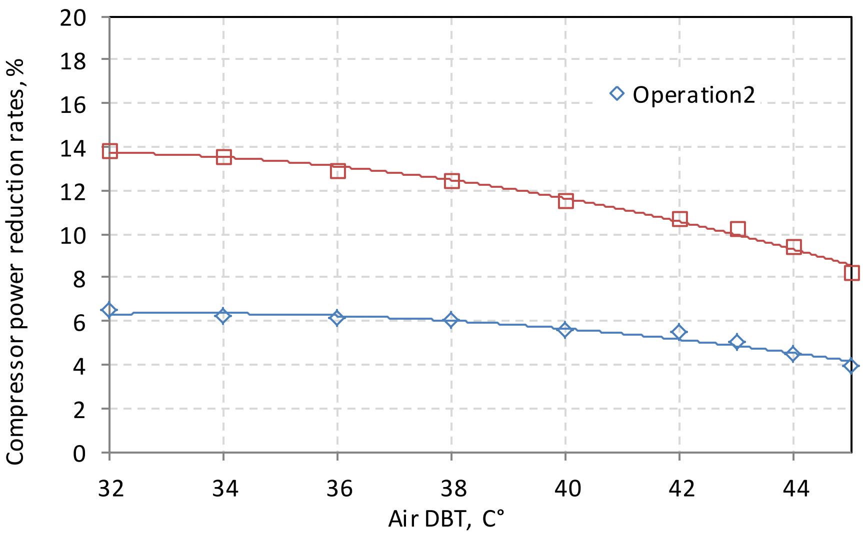


Figure 6.7 Compressor power reduction rates by outside DBT at 80 RH%

6.3.2 Mass flow rate

The studying of refrigerant mass flow rate changing in the system are very important to calculate the cooling load of the system, then computing the COP of the system.

The actual power consumed by compressor is computed in the last section, so the refrigerant mass flow rate can be calculate by equation 6.4:

$$\dot{m} = \frac{\dot{W}_{comp.}}{h_2 - h_1} \tag{6.4}$$

The enthalpies of states in entering and exiting of the compressor are taken from the R-22 refrigerant properties at the temperature and pressure measured during the tests.

As shown in figure 6.8, and at the condition 45°C, 40% RH, the refrigerant mass flow rate increased by 5.3% when applying the evaporative cooling on the air conditioner condenser, while it's increase by 7.2% when applying pre-cooling on the compressor discharge line, to make a total increasing by 12.5%, the maximum increasing in refrigerant mass flow rate noted on the minimum DBT of air which it's entrance to the condenser, figure 6.9 shows the changing in the refrigerant mass flow rate for other air DBT at 45% RH.

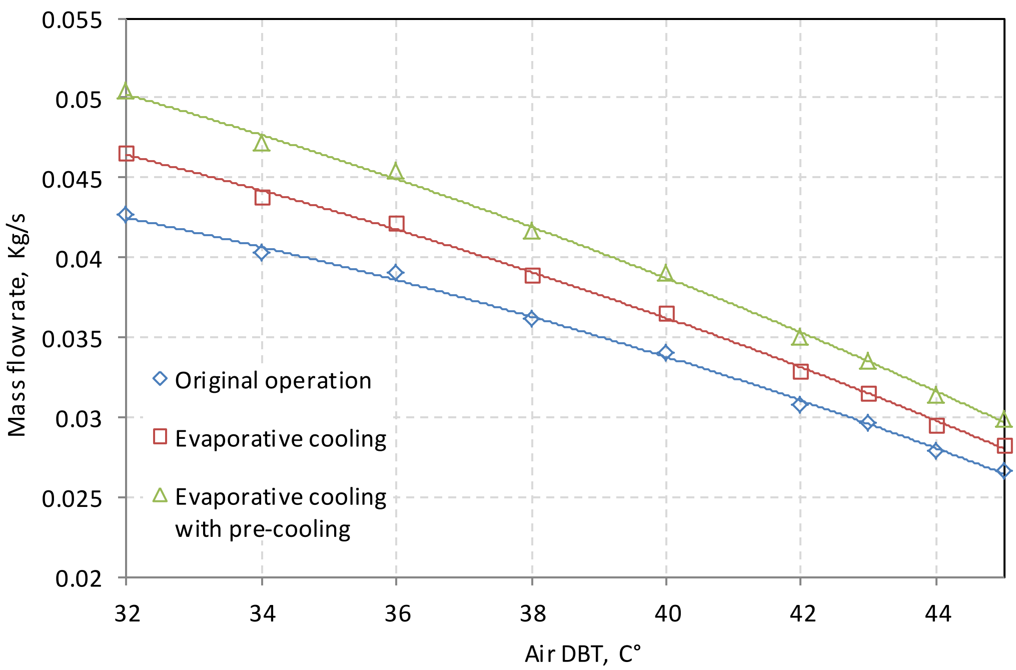


Figure 6.8 Refrigerant mass flow rates at 45%RH

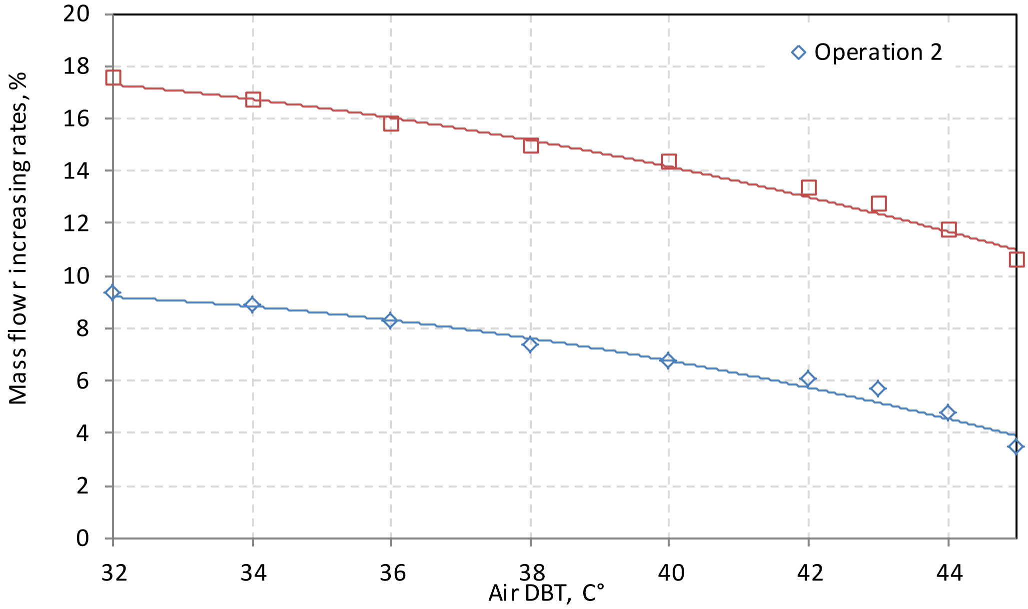


Figure 6.9 Mass flow rate increasing by air DBT at 45 RH%

For the system operation at high relative humidity, and at the condition 45°C DBT,80% RH, and after applying evaporative cooling on air conditioner condenser, the mass flow rate increasing was about 3.50%, this shows 1.8% reduction compering with 45% RH test, while the increasing in mass flow rate was 3.31% when pre-cooling of discharge line applied, the reduction was about 0.3% compering with the test at same DBT but in 45%RH, figure 6.10 shows the changing of refrigerant mass flow rate at 80% RH, and the figure 6.11 shows the changing for other air DBT.

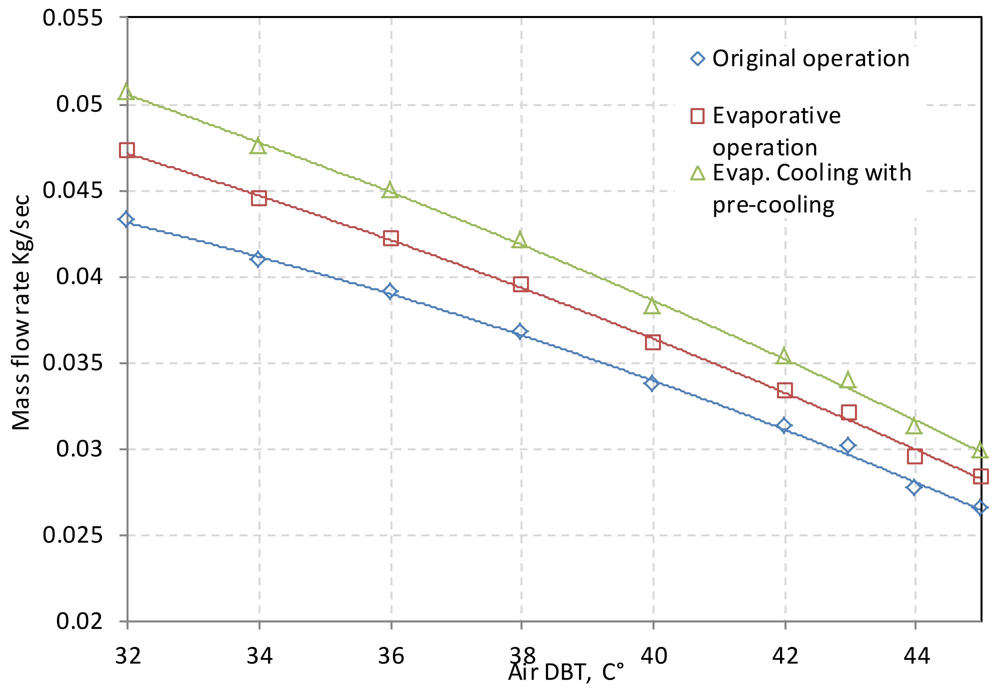


Figure 6.10 Refrigerant mass flow rates at 80% RH

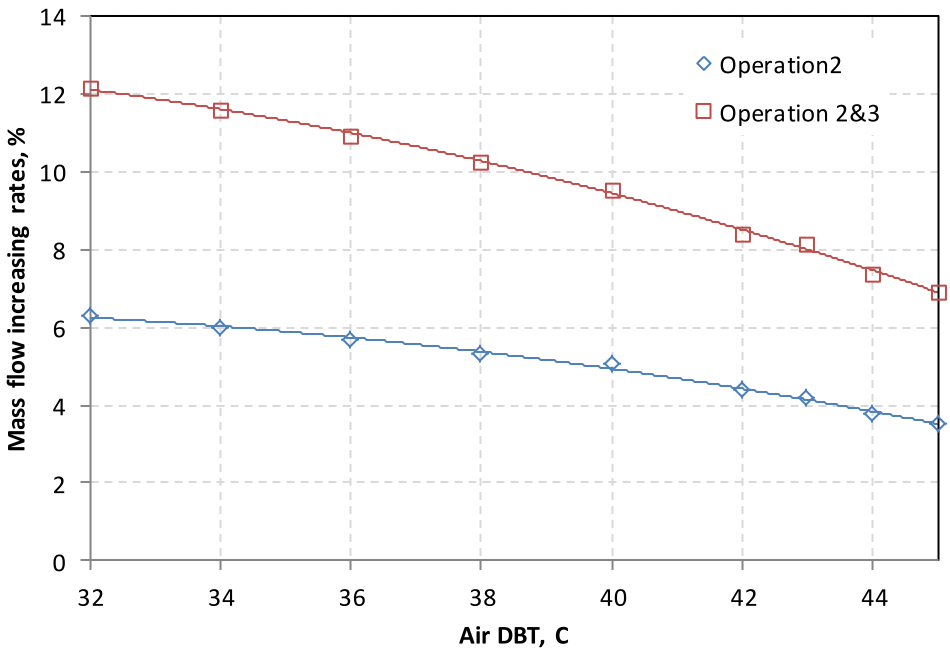


Figure 6.11 Mass flow rate increasing by air DBT at 80 RH%

There are two main factors that cause increasing in the refrigerant mass flow rate, the first factor is reducing the power consumption by the compressor due to applying evaporative cooling and pre-cooling discharge line which they reduce the total operating pressure of the system, the second factor is reducing in the enthalpy at compressor exit, and this is due to reduction of the temperature of the super heat refrigerant.

6.3.3 Cooling capacity

Cooling capacity is one of the other important factors which effect to operate the system economically, because of the increasing and decreasing in the cooling capacity effect on the compressor running time period, increasing in cooling capacity will make the system reach to demand temperature in shorter time so the compressor will shut down faster, from other hand, the increasing of outside temperature will increase the heat transfer from outside to inside the conditioned room, therefore a high cooling capacity in high outside temperature will be make the system operation more comfortable and economical.

The cooling capacities are compute by the equation 6.5

$$Q_{evap.} = \dot{m}\,(h_1 - h_4) \qquad (6.5)$$

We used the computed refrigerant mass flow rate from last section, and the enthalpies are compute from R-22 tables at the measuring temperature and pressure.

As shown in figure 6.12, the increasing of outside temperature is make the cooling capacity decrease, at the condition 45°C 45 % RH, and after applying the evaporative cooling on the air conditioner condenser, an 12.55% increasing in cooling capacity are noted, while when applying the pre-cooling on discharge line, the increasing was 6.11%, to make 18.66% in total, other cooling capacity increasing rate are mentioned in figure 6.13

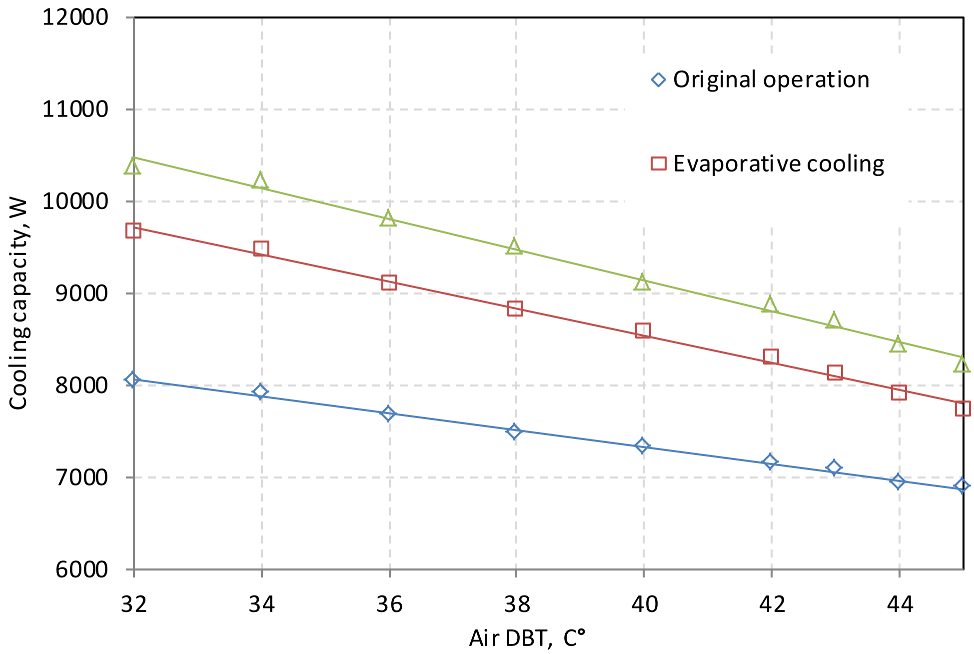


Figure 6.12 cooling capacity increasing at 45%RH

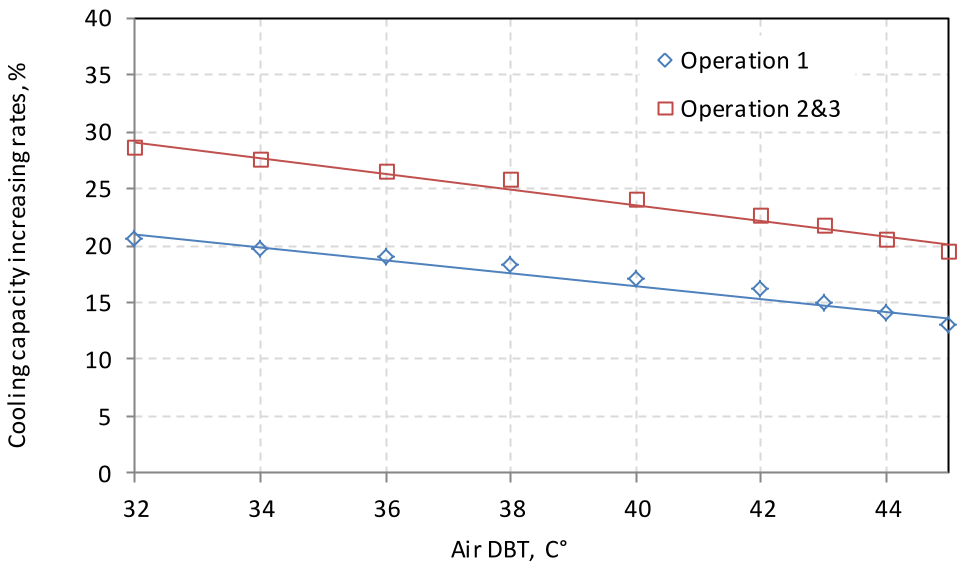


Figure 6.13 Cooling capacity increase rates by air DBT at RH 45%

For the system operating in high relative humidity, and due to decreasing in evaporative cooling efficiency as mentioned before, the increasing in the cooling capacity at 45°C, 80% RH are noted as 3.91%, and it's about 8.86% reduction comparing with the same DBT but in 45%, as shown in figure 6.14, the increasing in cooling capacity was 3.95% when pre-cooling on discharge line are applied, that's shows reduction by 2.16% comparing with the same DBT but in 45% RH, the figure 6.15 shows increasing rates in cooling capacity for other tested temperature.

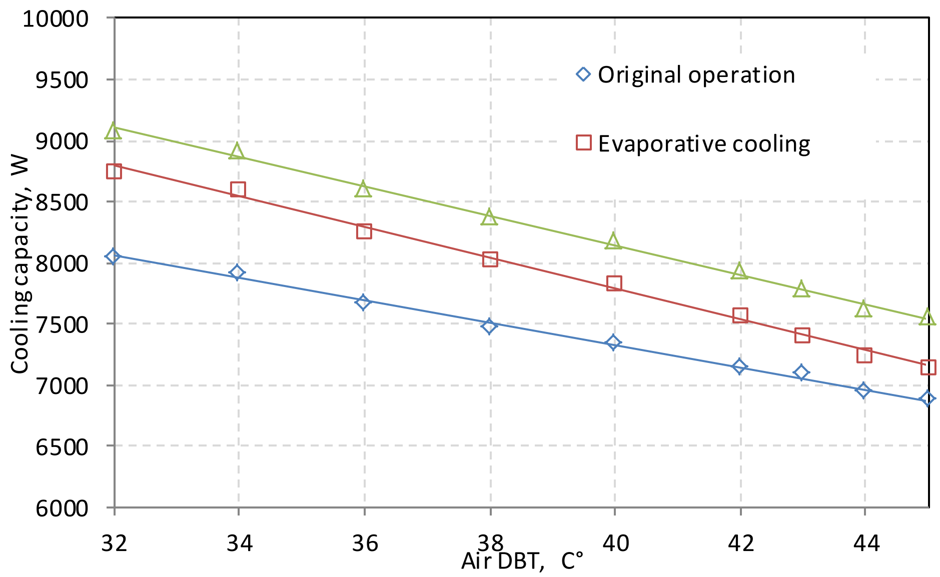


Figure 6.14 Cooling capacity at 80% RH

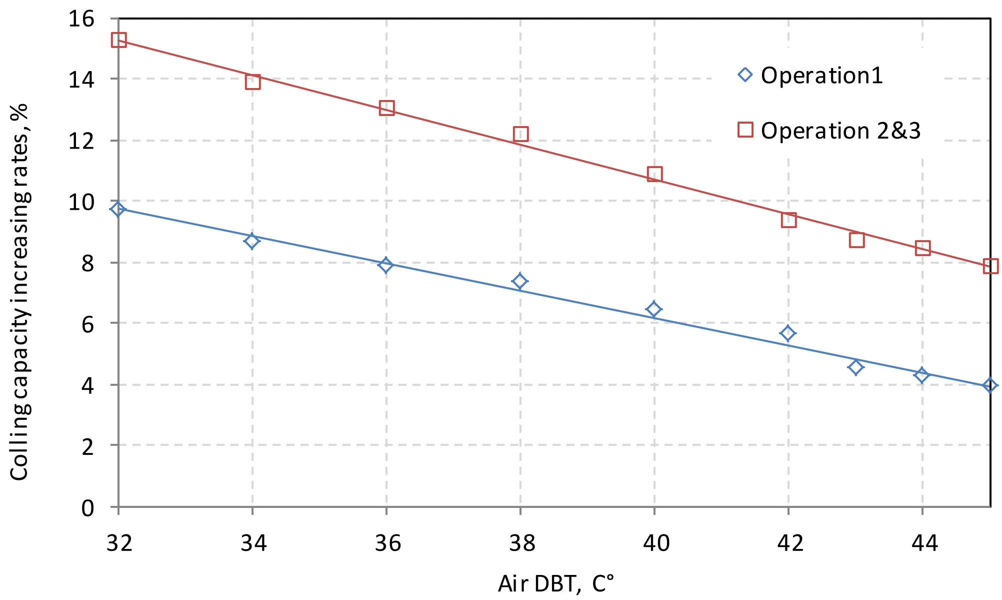


Figure 6.15 Cooling capacity increase rates by air DBT at RH 45%

the cooling capacity increasing done by two factors, increasing in refrigerant mass flow rate as mentioned in last section of this thesis, and decreasing in enthalpy change in the system evaporator, which it caused by the changing in the system pressure ratio and temperature.

6.3.4 System coefficient of performance (COP)

The coefficient of performance (COP) for the system compute after computing the power consumption by the compressor and cooling load for each test by using the equation 6.6

$$COP = \frac{Q_{evap.}}{W_{comp.}} \qquad (6.6)$$

As shown in the figure 6.16, and after applying evaporative cooling on the air conditioner condenser, an increasing in COP by 25.47% indicated when the system operating at 45°C DBT and 45% RH, while the increasing in the COP for the same temperature and relative humidity was 11.26% after applying the pre-cooling on the compressor discharge line, figure 6.17 shows the increasing rates of COP for other tested temperature.

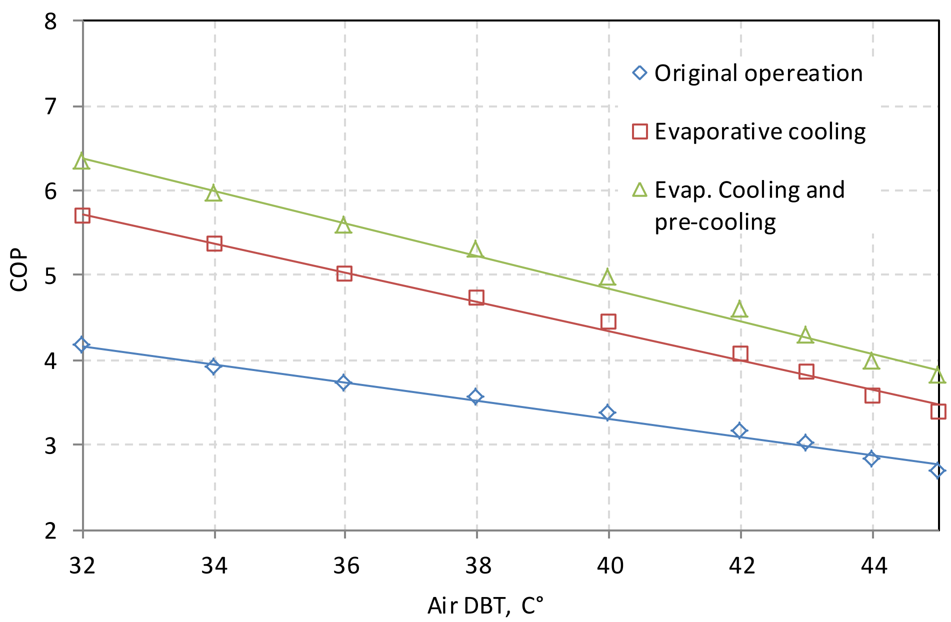
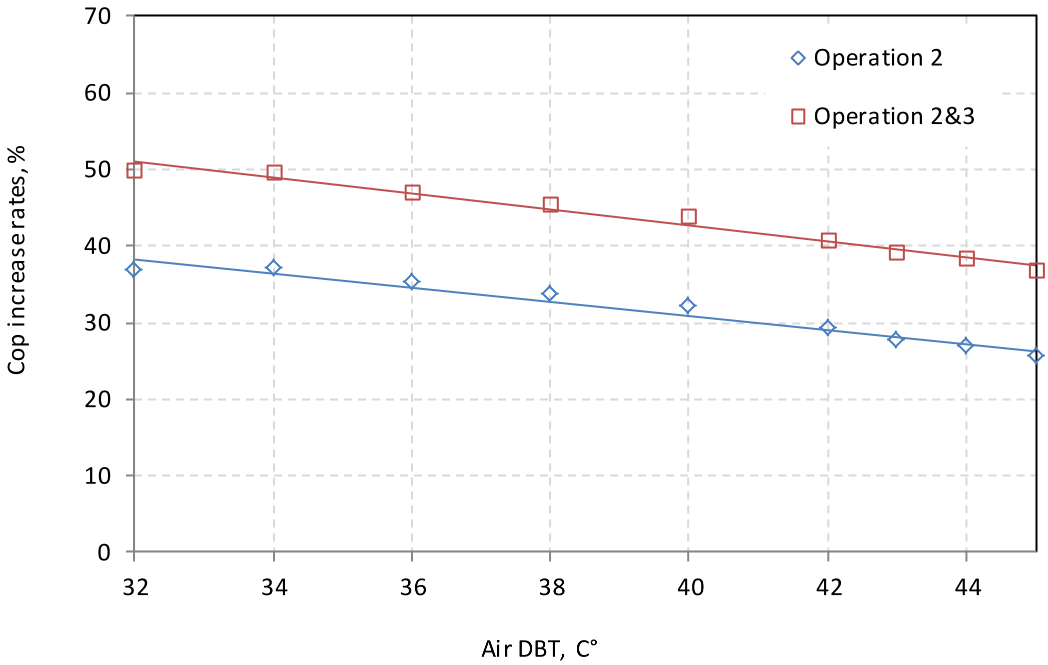


Figure 6.16 COP of the system at 45%RH



6.17 COP increasing rates at 45 % RH

When test the system at high relative humidity as shown in figure 6.18, the increasing in COP noted about 8.18% at 45°C and this is about 17.29% reduction comparing with the same DBT but in 45% RH, while the increasing was about 8.8% after applying the discharge pre-cooling, which it's 2.46% reduction comparing with the same DBT and 45%RH, figure 6.19 shows the increasing rates of COP in other tested DBT.

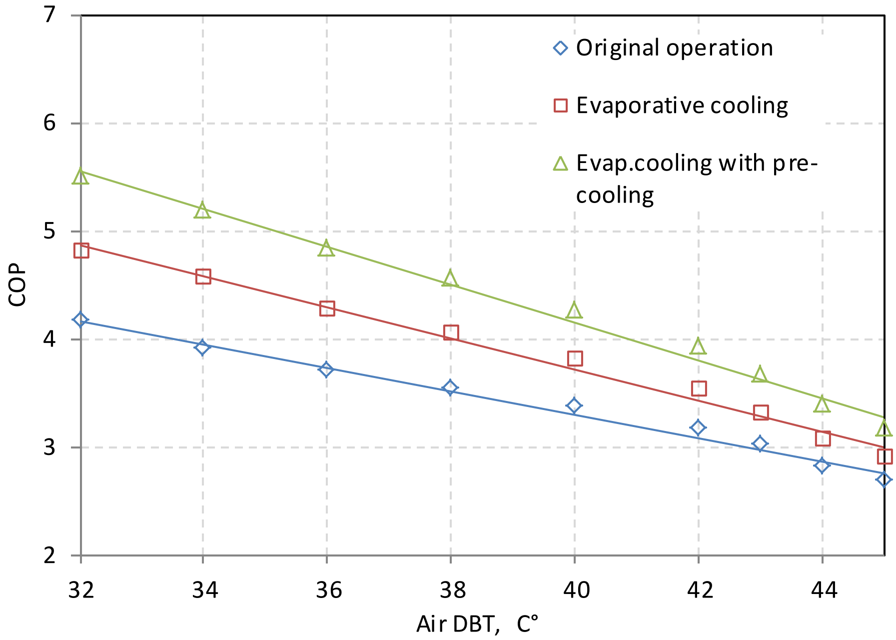


Figure 6.18 COP of the system at 80%RH

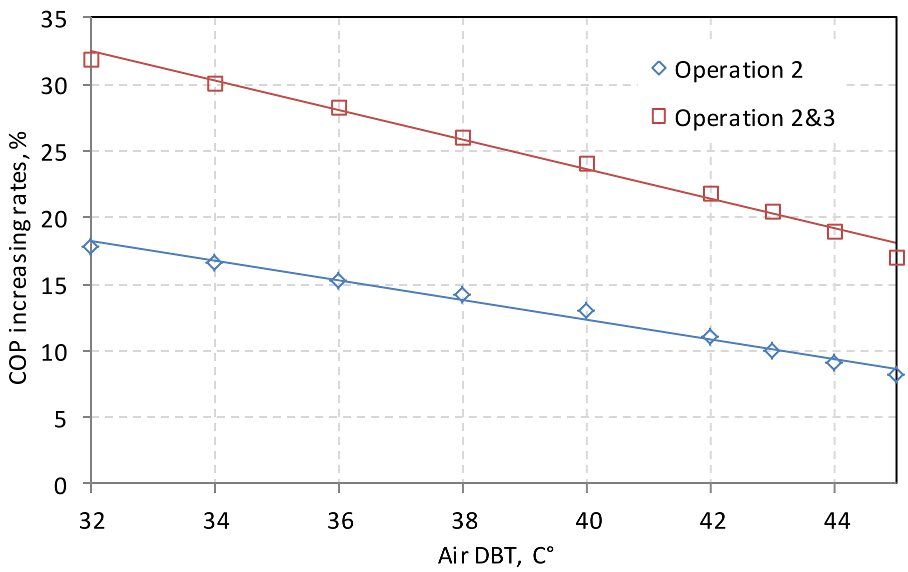


Figure 6.19 COP increasing rates at 80% RH

The increasing in coefficient of performance for the system after applying evaporative cooling on the condenser, and pre-cooling on compressor discharge line was caused by the reduction of compressor power consumption and increasing in cooling capacity as mentioned in last sections.

6.4 The economic feasibility

As mentioned in last sections, the adding of evaporative cooling on air conditioner condenser, and pre-cooling on compressor discharge line, will effect to increase the power consumption by the compressor.

For Gaziantep city, the average high temperature in summer season is 38 °C, and referring to table 6.3, the compressor power consume reduction or the operating cost reducing is 26.3%

At the same weather condition, there are some adding elements on the system which make increasing in operating cost, like the circulation water pump power consumed, and evaporated water cost by the evaporative condenser, since the circulation water pump are consumed power by 52W, although it is so small value comparing with the power which saving by evaporative cooling, the water pump power calculated in the economic feasibility of this work.

At the same outside condition (38 °C) and referring to figure 6.2 the water evaporation rate is 7.55 L/hr, and according to the cost lists for electric and water for Turkey republic in 2015 year, the cost of water per one hour of the tested system operation is 0.538% of the cost of electric for one hour operating, so the total cost saving will be 24.76% to operate the system for one hour when the outside temperature is 38°C.

This coast saving rate will decrease more when the system working in morning or evening time which the outside temperature are lower as shown in figure 6.20

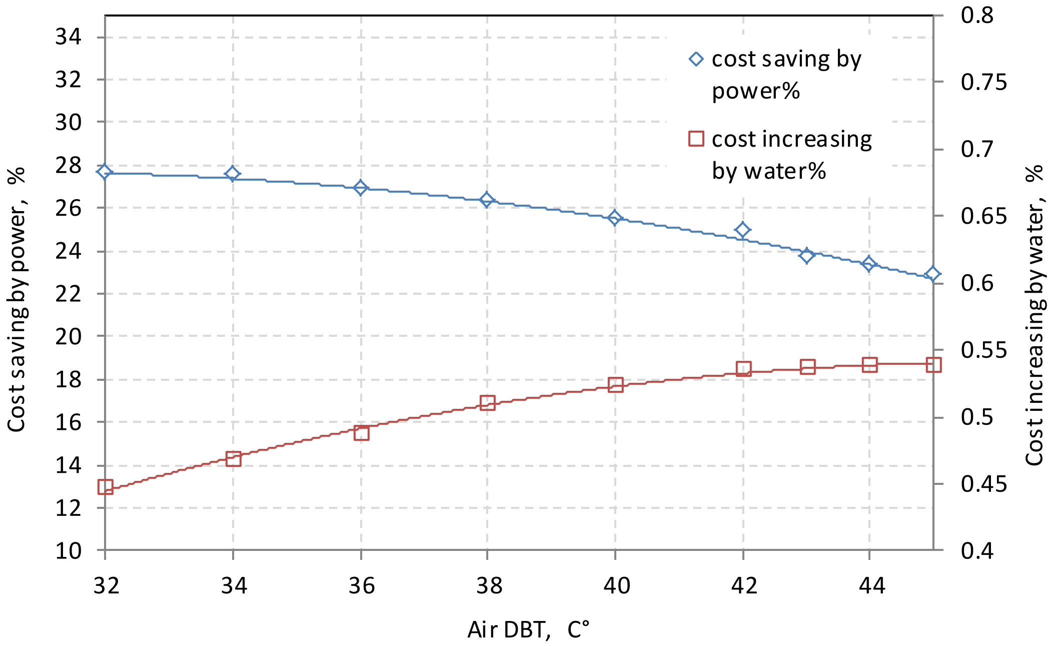


Figure 6.20 Cost analyzing by outside temperatures at 45% RH

Using the same procedure, the cost saving by decreasing in air conditioner compressor power consumption, and the additional cost by evaporating the water in evaporative cooling are compute when the system operates at high relative humidity weather, figure 6.21 shows the cost analyzing at relative humidity 80%.

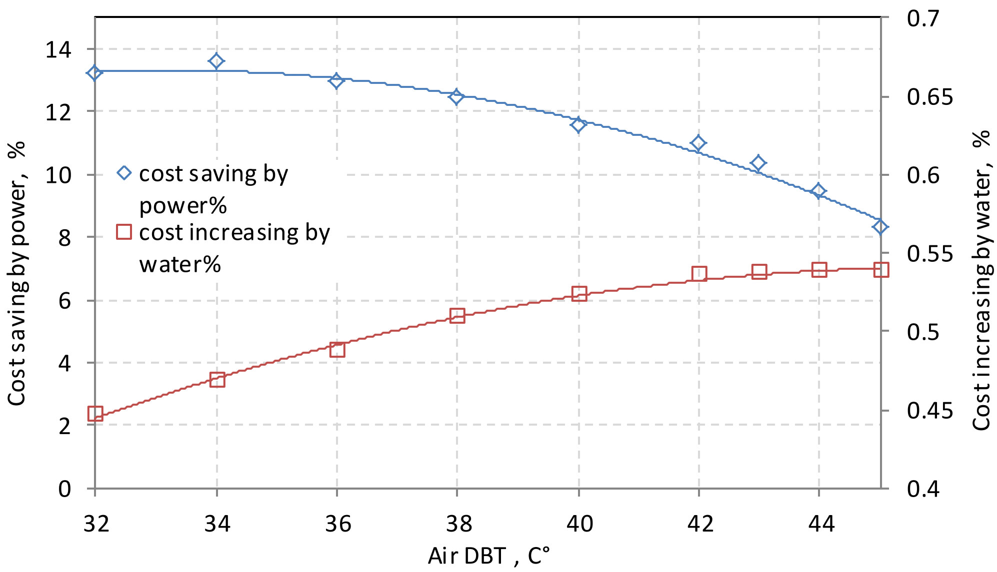


Figure 6.21 Cost analyzing by outside temperatures at 80%

CHAPTER 7
CONCLUSION AND RECOMMENDATIONS

Evaporative cooling is one of the oldest methods which used by human in the world, it's have low initial cost and low operating cost too, evaporative cooling characterized as environment friendly because it's cold air without and pollution in environment, while the vapor compression cycle used in the last hundred years, but still have operation cost problem due to the high power consumption by system compressor, from other hand, the high power consumption led to use the electric which generate by fossil fuel, in other meaning the improvement in power consumption by vapor compression system will decrease the pollution in the world because it's one of the most used system to condition in the world.

To improve the power consumed by vapor compression cycle a novel system designed and tested experimentally, the new system include combination between a 24000 BTU spilt air conditioner which it's work by vapor compression cycle principle and evaporative air cooler which it's design spatially for the tested air conditioner, and to appraisement the cold water which generate in evaporative cooling a new piping design made to pre-cooling the compressor discharge line which increase the air cold condenser performance.

In order to examine advantage of the new designed system experimentally, in this thesis, an experimental system determined performance installed in scope of this thesis investigation on performance of new designed system. All nodal temperature, pressure, voltage and current were measured and collect during the cooling operation mode in summer season (June 1-august 31, 2014). Performance parameters of the system were compute using the measurement values, the result obtained from experimental study and mentioned as figures and tables in Chapter 6. For measurement days in cooling season, the most important results and some recommendations are offered in this chapter.

1. The experimental results show that the power consumed by compressor can be decrease by (16.2 %-20.54 %) depending on outside temperature when an evaporative cooling added on system to pre-cool the air before inlet to the air conditioner condenser, while the power improvements compute (6.6 %-7.14 %) when adding pre-cooling on the discharge line to make (22.8%- 27.59) power saving in total when the system works in low and medium relative humidity weather.

2. Due to the decreasing in the system pressure ratio, an increasing in cooling capacity indicated by (7.82 %-14.39 %) and (7.2 %-6.11 %) when adding evaporative cooling and pre-cooling on discharge line respectively.

3. The increasing in cooling capacity and decreasing in compressor consumed power depend on the relative humidity of the weather; the experimental testes show the system performance decreasing when the it's work in high relative humidity weather when the evaporative cooling adding on the system.

4. Due to cooling capacity increasing, its can reduce the size of the compressor or condenser when the new design adding on new manufactured air conditioners to improve the manufacturing cost.

5. When adding pre-cooling on discharge line on the system works in high relative humidity, the system performance decreasing was less than the decreasing when applying just the evaporative cooling, and this decreasing depend on the supply water temperature.

6. The performance of cooling pad are compute in the tested temperature rate for two relative humidity (45%, 80%) and it's found the pad efficiency was about 70% and 50% respectively.

7. The water evaporation rates found (7-8.2) L/hr depending on the temperature and relative humidity of the weather.

8. Since the power of the circulation pump is very low (52 W) comparing with power of the compressor, its can operated by solar panel to decrease the power consumed by all system.

9. The solenoid valves which added on the system used to change the refrigerant flow direction for comparing reason, so the pressure drops by adding the solenoid valves and the cost of adding it can be neglected when applying the new design on any air conditioner.

10. It's found experimentally, the decreasing of the air temperature by 1°C lead to decrease the power consumed by the compressor (2%) without any modifying, the choosing of air conditioner installing place to be on shadow area of any building have effects on power consumption by the air conditioner.

11. The moisture on inside room air is condensed to water when the air pass through the air conditioner evaporator, and it's wasted to outside by drain hose, although the amount of the drain water depend on the humidity of the room air, it's found the drain water amount reach to 1 L/hr when applied the evaporative cooling and discharge pre-cooling due to the decreasing in evaporator temperature, this waste water can used as refilled water in evaporative cooling to reduce the evaporated water cost.

12. For the medium temperature range weathers, it can be used aspen wood cooling pad instant of the Cellulose fiber media, to improve the initial cost, because the aspen wood media have good efficiency in low and medium temperature range and low cost.

13. It does can make the evaporative cooling frame by plastic instant of iron frame to improve initial cost and to get rid of the corrosion.

14. For economic analysis based on local prices shows the energy saving can pay for the cost associate with retrofitting the condenser and discharge line in less than 1one year.

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APPENDEX A

KEW 2200R Digital clamp meter specification

	KEW 2200R	KEW 2200
Detection method	RMS	Averaging value
AC A	40.00/400.0/1000A(Auto-ranging) ±1.5%rdg±5dgt(45~65Hz) ±2.0%rdg±5dgt(40Hz~1kHz)	40.00/400.0/1000A(Auto-ranging) ±1.4%rdg±6dgt(50/60Hz) ±1.6%rdg±6dgt(45~65Hz)
AC V	4.000/40.00/400.0/600V(Auto-ranging) ±1.8%rdg±7dgt(45~65Hz) ±2.3%rdg±8dgt(65~500Hz)	
DC V	400.0mV/4.000/40.00/400.0/600V(Auto-ranging) ±1.0%rdg±3dgt* *400mV range is excluded	
Ω	400.0Ω/4.000/40.00/400.0kΩ/4.000/40.00MΩ(Auto-ranging) ±2.0%rdg±4dgt(0~400kΩ) ±4.0%rdg±4dgt(4MΩ) ±8.0%rdg±4dgt(40MΩ)	
Continuity buzzer	buzzer sounds below 50±30Ω	
Conductor size	φ33mm max.	
Applicable standards	IEC61010-1 CATⅣ300V*, CATⅢ600V Pollution degree2(AC A) *2200R only CATⅢ300V, CATⅡ600V Pollution degree2(AC/DC V) IEC61010-031, IEC61010-2-032, IEC61326(EMC), EN50581(RoHS)	
Power source	R03/LR03(AAA)(1.5V)×2	
Continuous measuring time	Approx.120 hours	Approx.350 hours
	Auto power off : approx.10 minutes	
Dimensions/Weight	190(L)x68(W)x20(D)mm / Approx.120g(including batteries)	
Accessories	7107A (Test leads), 9160 (Carrying case), R03(AAA)×2, Instruction manual	
Optional	8008 (Multi-tran)	

APPENDEX B

Pressure transmitter readings and corresponding calibration equations for Gigalog data loggers

	Gauge pressure (bar)			
Transmitter	0	1	5	10
	Readings (mA)			
1	4,112	4,566	6,653	9,357
2	4,031	4,577	6,210	9,433
3	4,077	4,456	6,160	9.654
4	4,109	4,554	6,479	9,562

APPENDEX C

Pressure drop curves for solenoid valve