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PERFORMANCE STUDY OF WINDOW TYPE AIR CONDITIONING UNIT WITH SUPERHEATING AND SUBCOOLING THAT USES R407c AS AN ALTERNATIVE REFRIGERANT TO R22

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This Thesis was Submitted in Partial Fulfillment of the Requirement for The Master's Degree of Science in Mechanical Engineering

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The University of Jordan

November, 2003

This Thesis (Performance Study of Window Type Air Conditioning Unit with Superheating and Subcooling That Uses R407c as an Alternative Refrigerant to R22) was successfully defended and approved on 6 th November, 2003.

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DEDICATION

To my mother who gave me hope and love, to my wife who helped and encouraged me to complete this work, to my children and to every one who gave me support and encouragement.

ACKNOWLEDGEMENT

I would like to express my deep gratitude and to thank my supervisor professor Mohammad Alsa'ad for his valuable encouragement and help to complete this work.
I would like to thank col. eng, Hassan Omer Mustafa, Director of Royal Maintenance Corps for his great help and support. I wish to extend my thanks and appreciation to all those who helped in advancing this work, which enabled me to accomplish this project.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

T	Temperature, °C.
T_e	Evaporating temperature, °C.
T_c	Condensing temperature, °C.
P	Pressure, MPa
S	Entropy, kJ/kg. K
h_1	Enthalpy at compressor in let, kJ/kg
h_2	Enthalpy at compressor out let, kJ/kg
h_4	Enthalpy at condenser out let, kJ/kg
h_{2s}	Isentropic enthalpy at compressor out, kJ/kg
v_s	Specific volume at suction of compressor, m3/kg.
v_d	Specific volume at discharge of compressor, m3/kg
m	Mass flow rate, kg/s
M	Percent clearance of the compressor.
q_{ref}	Refrigeration effect, kJ/kg
Q_{ref}	Refrigeration Capacity, kW
q_c	Heat rejection, kJ/kg
Q_c	Heat rejection rate, kW
η_o	Overall efficiency, %
η_v	Volumetric efficiency, %
$\Delta T(sub)$	Degrees of superheating, °C
$\Delta T(sup)$	Degrees of subcooling, °C.
T_s	Compressor discharge temperature, °C
η_{isent}	Isentropic efficiency, %

ABBREVIATIONS

ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
ASERCOM	Association of European Refrigeration Compressor Manufacturers
CFC	Chlorofluorocarbon
COP	Coefficient of Performance
GWP	Global Warming Potential
HCFC	Hydrochlorofluorocarbon
HFC	Hydroflourocarbon
ODP	Ozone Depletion Potential
ppm	Parts per million

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ABSTRACT

This research investigates the effect of superheating and subcooling when replacing the refrigerant R22 by the refrigerant R407c in a 5 kW window-type air conditioning unit. The need of such replacement comes from the fact that R22 is an ozone depleting gas.

Results of this work indicate that the refrigerant R407c can be used as an alternative refrigerant for R22 in this type of air conditioning units. The obtained results of this work indicate that the COP of R407c and R22 increases slightly as ΔT (sub) and ΔT (sup) increase separately or in a combined way and it reached a value of 4.35 and 5.18, respectively at $T_e = 13°C$, ΔT (sub) = 17°C, ΔT (sup) = 6°C and $T_c = 40°C$. This indicates that the COP of R407c is less than that of R22 by 16%. Also, as a result of this work, the refrigeration capacity of R407c when it replaces R22 is decreased from a value of 5.6 kW to 4.66 kW, at $T_e = 13°C$, $T_c = 40°C$, ΔT (sub) = 17°C, ΔT (sup) = 6°C. This means a reduction of refrigeration capacity of about 16.8%.

1-INTRODUCTION

Refrigeration and air conditioning is the art of controlling an atmospheric environment that is used for the comfort of human beings or for the proper performance of some industrial or scientific processes.

Refrigerants are the working fluids in refrigerators, air conditioning and heat pumping systems.

The refrigerant of a refrigeration unit should whenever possible, be non-flammable, non explosive, and of very low toxicity for the humans and the environment.

Several chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) are being extensively used as refrigerants for air conditioning and refrigeration purposes. They posses most of the desirable characteristics such as thermal and chemical stability, thermodynamic efficiency, non-flammability, availability, safety, ease of handling and low cost.

The environmental consequences of refrigerant that leaks from system must be considered when selecting suitable refrigerants. Some refrigerants such as CFCs and HCFCs and hallons molecules break down and release chlorine and bromine atoms, which destroy the ozone layer around the earth (ozone depletion). Therefore, they allow harmful ultra violet radiation to pass through the atmosphere unfiltered. Each refrigerant has a different ozone depletion potential (ODP) value. The impact of refrigerants containing chlorine atoms on the environment has been assessed. As a basis, an index of 1 is used which relates to the environmentally destructive potential of CFC11 (R11) within a particular decay period. Therefore, R11 has an ODP of 1.

By comparison with this scale, R22 has an ODP of 0.055; R407c contains no chlorine and therefore, has an ODP of zero.

Due to the emission of CFCs to the atmosphere, a serious environmental problem affecting the whole climate of the earth arises to be a global problem. Montreal protocol of 1987 limited the production of CFCs and HCFCs and hallons to 0.5% by January 1, 2020 for R22 and the other refrigerants.

For the above reason, scientists looked forward to have alternatives with zero ozone depletion potential and low global warming potential. Such alternative refrigerants should posses good thermodynamical and physical properties, high chemical and thermal stability, low toxicity, good miscibility with lubricants, compatibility with materials, less expensive and low flammability with no environmental side effect.

Thermodynamic properties are of most importance since they decide whether a substance is applicable as a refrigerant in a certain temperature range or not. If the thermodynamic properties meet the requirements, the other characters must be taken into consideration and at least to be acceptable.

As a replacement to R22, R407c with the above properties encouraged the scientist to use it in air conditioning systems.

Refrigerant R407c which is a blend of R32, R125 and R134a (all of them are free of chlorine atoms) is a new refrigerant being used in U.S.A and Europe for the last ten years. It has proven to have friendly effect on the ozone layer and low effect on the global warming compared with the other traditional refrigerants. It also gives good performance when compared with R22 with minimum extra cost and minimum design changes.

In this work the effect of superheating and subcooling on the performance of a window type air conditioning unit that uses the proposed refrigerant R407c as a replacement for R22 is experimentally investigated. The work aims to find advantages and disadvantages of using R407c under the above conditions. Many parameters such as the operating pressures, compressor exit temperature, superheating and subcooling effects, refrigeration effect, refrigeration capacity, heat rejection, heat rejection rate, COP, compressor power, work of compression, isentropic efficiency, and volumetric efficiency will be investigated and compared with those of R22.

2-LITERATURE SURVEY

The Montreal Protocol on substances that deplete the ozone layer was drawn up in September 1987 under the United Nations Environment Programme. This protocol required all developed countries to phase out CFC'S by the year 1996. It further calls for the complete phase out of HCFC'S by 2030 in developed countries.

The future of refrigeration and air conditioning has never been discussed so controversially as at present. The world trend towards environmental issues made it eager for the researchers and scientists to conduct experiments to find appropriate substitutes of CFCs and HCFCs.

Many studies and researches were carried out on different refrigerant alternatives to R22 concerning their properties, system performance (theoretically and experimentally).

In this chapter, some of the previous works and efforts concerning the proposed area of study are presented.

The German Compressors Manufacturing Company (BITZER) studied the performance of R407c on their new generation of compressors to cope with the world trend regarding the environmental cases. BITZER found that R407c refrigerant has nearly similar thermodynamic properties and performance as that of R22 in air conditioning systems for medium temperature cooling range.

Dilok, (1998) examined the R407c refrigerant performance in retrofitting the R22 refrigerant. He concluded that the refrigeration capacity of R407c and R22 refrigerants are very close to each other and the power consumption of R407c is less than that of R22.The COP is simultaneously slightly higher.

Ferrari et al. (1994) examined R407c (which is a mixture of 52% R134a, 25% R125, 23% R32) as a replacement for R22. They showed that evaporator heat rate at 40 °C condenser temperature with no superheating or subcooling varied from 82% to 98% as the evaporator temperature varied from – 40 to 10 °C while compressor work rate varied from 85% to 98%.

Mongey, (1996) constructed a refrigeration tests facility to examine the performance of potential alternative refrigerants to R22. It was found that the performance of R407c approached that of R22 at higher evaporator temperatures. Reductions in evaporator capacity and COP were found with decreasing evaporator temperature.

Dawood, (2001) made a performance study of a 20-ton chiller using R407c as a replacement to R22. He concluded that the refrigerant R407c is a suitable alternative for R22 since it is ozone friendly and has similar physical and thermodynamic properties as that of R22. Performance tests were conducted under different condensing and evaporating temperatures to study the performance parameters such as the operating pressure, compressor exit temperature and COP.

Makahleh, (2001) studied the performance of a 2-ton air conditioning split unit using R407c instead of R22. He concluded that at medium and high evaporating temperature, COP, refrigeration capacity, isentropic efficiency and volumetric efficiency reached a value of 100% of that of R22.

Hatzikazakis. , (1994), presented a comparison of the dual compressor performance when R22 is replaced by R407c. He showed that the COP of the system using R407c reached 95% of that of R22. This result was concluded at 32°C ambient temperature, 11.1 °C superheating temperature and 8.3 °C subcooling temperature.

The Association of European Refrigeration Compressor Manufacturers (ASERCOM), (1997), conducted several experiments on the use of R407c to replace R22. This study was carried out to evaluate the coefficient of performance using R407c at different evaporation and condensing temperatures. They concluded that advantages of R407c are, it is non-flammable, nontoxic, has lower compressor discharge temperature than that of R22 and large subcooling effects.

Sormani, (1994), tested the performance of an air-cooled water chiller charged with R407c. He concluded that it is possible to improve the performance by using the tenary mixture (R407c), provided that the chiller is equipped with a symmetric shell and tube evaporators. This allowed taking advantages of the characteristics of the individual components in the R407c mixture, which means increasing in temperature during the evaporation process.

Muir, (1994), studied the performance of R134a, R407c and R410a as alternative refrigerants to R22. He found that R134a has the lowest refrigeration capacity and pressure ratio of the three alternatives. The use of R134a would require redesigning all R22 equipment in order to utilize R134a efficiently. He concluded that R134a is the least likely of all alternative refrigerants to be utilized in residential and commercial air conditioning system. On the other hand, he found that R407c is the simplest alternative refrigerant because it requires minimal changes to the current R22 operating equipment. The only major change required is the use of polyester lubricant, which is a hydroscopic material that have greater tendency to absorb moisture. Muir concluded that the coefficient of performance (COP) of R407c is slightly less than that of R22, (95% of R22) and the cost of the system using R407c is slightly greater than that of R22. In addition, he concluded that the COP of the system that uses R410a has about (5%) higher as compared with R134a and R407c, for the same cost.

Luzzatto et al. (1994) presented the efforts carried out for applying R134a and R407c as substitution for R22 on portable room air conditioning system. They showed that for R134a, the compressor displacement should be 50% higher than that for R22 for the same refrigeration capacity. They showed that the lubricant oil, the expansion device and the filter should be changed. Also they showed that when using R407c the COP reached 91% of that of R22 without any modification on the system. They made a cost comparison between systems when R134a an R407c were used. They showed that the system cost would increased by 2% and 7% by using R407c and R134a, respectively, higher than the same system working with R22.

Herz, (2003) studied experimentally the performance of window type air conditioning unit using R407c as an alternative refrigerant. He concluded that the refrigerant R407c could be used as an alternative refrigerant for R22 in this type of air conditioning units with optimum charge quantity of 900 gram. He concluded also that the COP of R407c and R22 reached a value of 3.6 and 4.2, respectively at T_e=10 °C and T_c=40 °C. This indicates that the COP of R407c is less than that of R22 by 14.3 %. On the other hand, he found that the refrigeration capacity of R407c when it replaces R22 was decreased from a value of 4.65 to 3.9 kW, at T_e=10 °C and T_c=40 °C.

In this work, a study is carried out experimentally to determine the performance parameters of R407c, when it is used instead of R22 in 5 kW window type air-conditioning unit under the effect of superheating and subcooling parameters. The obtained performance curves of R407c are compared with those for R22 at different evaporating and condensing temperatures and different values of degrees of superheating and subcooling.

3-THEORITICL ANALYSIS

3.1 General

Hydrochlorouflourocarbons (HCFCs) are being extensively used as refrigerants for air conditioning and refrigeration purposes.

R22 was one of the original refrigerants developed in 1930s. It has been the workhorse of the air conditioning and refrigeration industry and is the predominant refrigerant in use today. The low ozone depletion potential of R22 compared to R12, along with its excellent refrigerant properties has helped facilitate the transition from CFCs. However, HCFCs including R22 are scheduled for eventual phase out under the Montreal protocol. R407c has been developed as the similar pressure replacement for R22 in positive displacement, direct expansion air conditioners and heat pumps.

R407c is a non- azeotropic blended refrigerant. Its components by mass proportion are:

- R32	23%	(2H2) Difluoromethane
- R125	25%	(F3CHF2) Pentafluoroethane
- R134a	52%	(F3CH2F) Tetrafluoroethane

It evaporates over a temperature range of 5-6 °C, when at the same pressure. A non-azoetropic mixture contains components, which boil at different temperatures. This difference in boiling points results in a variance in temperature through the length of an evaporator or condenser as the mixture ratio changes and is referred to as the temperature glide. An azoetrope mixture has no glide. Mixtures with a very small amount of glide (fraction of 1) are some times referred to as near azeotropes.

for a specific application. Selected thermodynamic, and physical properties of R407c and R22 are compared and presented in table (1).

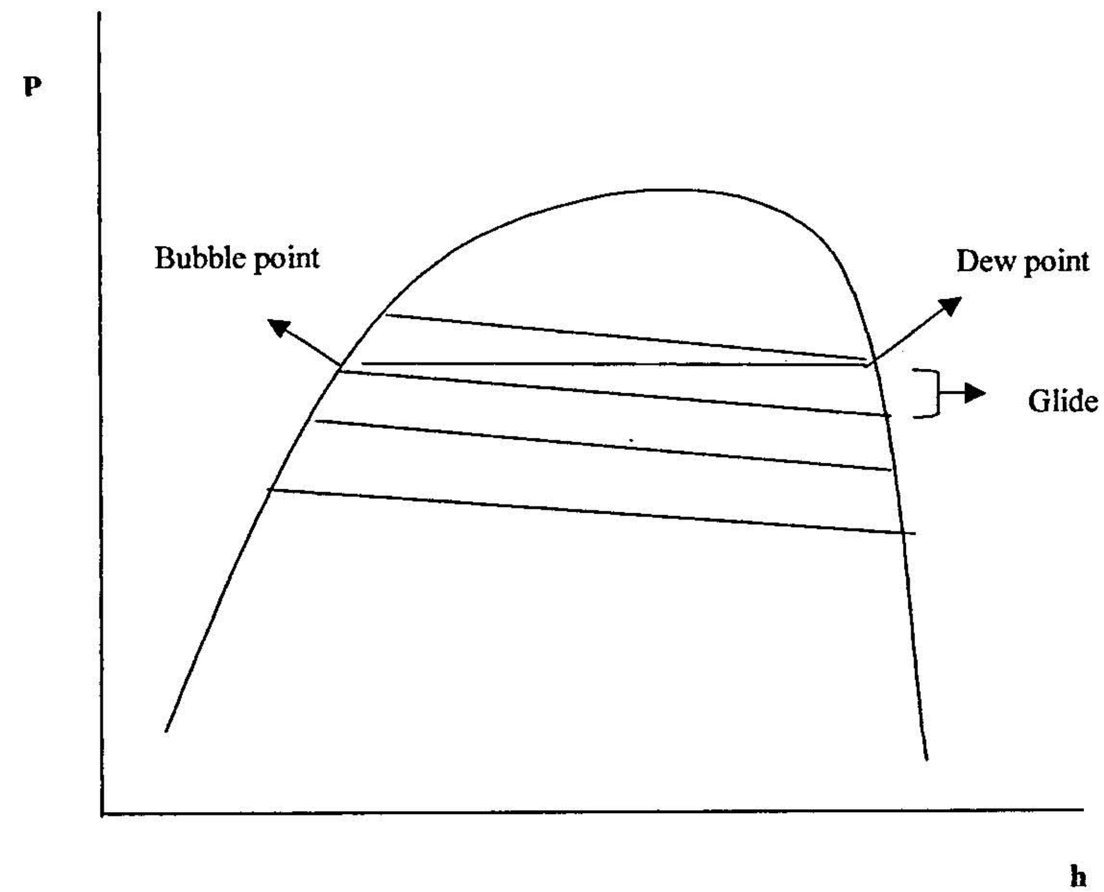

Figure (3.1): Temperature glide of non-azeotropic refrigerant mixture

Table (1): Thermodynamic and physical properties of R407c and R22

(Adopted from Dupont products catalog 1999)

PROPERTY	R407c	R22
Boiling point at 1atm(°C)	-43.56	-40.8
Critical temperature (°C)	86.74	96.24
Critical pressure (kPa)	4619.1	4980.7
Critical density (kg/m^3)	527.3	524.21
Liquid density at 25 °C (Kg/m^3)	1134	1195
Sat. vapor density at 25 °C (Kg/m^3)	41.96	44.21
Liquid specific heat at 25 °C (kJ/kg. K)	1.54	1.24
Vapor specific heat at 25 °C (kJ/kg. K)	0.83	0.685
Sat. liquid vapor pressure (kPa)	1173.4	1043.1
Heat of vaporization (kJ/kg)	245.1	233.5
Liquid thermal conductivity (W/m. K)	0.0819	0.0849
Vapor thermal conductivity (W/m. K)	0.01314	0.01074
Liquid viscosity (Pa.s)	0.00016	0.000159
Toxicity (ppm)	1000	1000
Temperature glide at 25 °C	7	0
Vapor viscosity (Pa.s)	0.0000123	0.000013
Ozone depletion potential	0	0.055
Global warming potential	1600	1700

3.3.2 Thermodynamic Properties

Some thermodynamic properties such as boiling point, critical temperature and pressure and latent heat of vaporization are discussed below.

- **Boiling Point**

Boiling point is defined as the temperature at which the vapor pressure of liquid is equal to the pressure on its surface when the substance changes from liquid state to vapor state. It can be seen from table (1) that the boiling temperature of R407c is slightly lower than that of R22 which means more compressor work is needed to reach the temperature that the refrigerant starts boiling to accomplish the vapor phase. Fluid with higher boiling points tends to have greater efficiency, higher pressure drop in some cases and lower heat transfer coefficient.

- **Critical temperature and critical pressure**

The critical temperature is defined as the temperature above which the gas cannot be liquefied however great a pressure it may be applied. The critical pressure is the lowest pressure at which a gas can be liquefied at its critical temperature, i.e., it is the saturation pressure at the critical temperature. The critical temperature and pressure of a refrigerant should be higher than the temperature and pressure in the condenser in order to prevent any decomposition of the refrigerant material. Table (1) indicates that the critical values for R22 are higher than that for R407c.The refrigerant should have high critical temperature because this means low work of compression. The critical temperatures for R407c and for R22 as shown in table (1) are high enough for ordinary air conditioning applications.

- **Latent heat of vaporization**

The latent heat of vaporization of a substance is the quantity of heat required to change unit mass of it at its boiling point from liquid to vapor without change of temperature at standard atmospheric pressure.

As shown in table (1), R407c has higher latent heat of vaporization than that of R22. Refrigerant that have higher value of latent heat of vaporization produces higher refrigeration effect per unit mass of the refrigerant. High values of the refrigeration effect gives low rate of refrigerant flow rate and consequently low losses.

3.3.3 Physical and chemical properties

The following physical and chemical properties for R22 and R407c are discussed:

- **Thermal conductivity**

The vapor thermal conductivity of R407c is higher than that of R22, which implies a higher value of a heat transfer coefficient and a higher heat transfer rate in the condenser. The liquid thermal conductivity of R407c is slightly lower than that of R22, which means a slightly lower heat transfer in the evaporator.

- **Viscosity**

Viscosity is a measure of flowing quality and it is desirable to use refrigerants with low viscosity in both liquid and vapor phases for higher heat transfer rates in the evaporator and the condenser, low pumping power and small pressure drops during flow. As shown in table (1), both refrigerants have nearly the same values of viscosity.

- **Density**

If the density of a refrigerant at compressor suction is high, the refrigerant mass circulated through the refrigeration system is low and so smaller compressor can be used.

R407c has slightly lower value of vapor density and hence slightly larger compressor is used for same value of refrigeration capacity.

- **Specific heat**

Low specific heat of liquid will increase the subcooling state of liquid and decrease the subcooling temperature, which will increase the refrigeration effect. As shown in table (1) the specific heat of R407c is higher than that of R22 in both liquid and vapor states.

- **Temperature glide**

As discussed in section (3.2) the high temperature glide can have negative influence upon the size of the heat exchanger (condenser and evaporator). The distinctive temperature glide requires special design for the main system components such as condenser and evaporator. As shown in table (1), R22 has no temperature glide while R407c has temperature glide of 7 K, which needs some unit modifications to offset this effect.

- **Miscibility with oil**

Immiscible oil can settle out in the heat exchanger to prevent heat transfer to such an extent that the plant can no longer be operated. Oil should have good solubility with the refrigerant. The polyester oil is suitable to be used with R407c due to its good solubility while mineral oil is suitable for R22.

- **Nonflammability**

Non-flammability is an essential requirement for refrigerant used in air conditioning industry. Although R32 is flammable compound, R407c is formulated such that it remains nonflammable during use. The refrigerant, R407c has been confirmed as practically nonflammable, the same as R22.

- **Toxicity**

Table (1) shows that the toxicity for both refrigerants, R407c and R22 is the same (1000ppm). Both refrigerants are classified as class A. Refrigerants with no identified toxicity at concentrations greater than 400 ppm are considered class A. Refrigerants that show evidence of toxicity at concentrations below 400 ppm are classified as class B.

- **Ozone depletion potential**

R22 has ozone depletion potential of 0.055 while R407c has zero ozone depletion potential as indicated in table (1). This advantage of R407c over R22 makes it suitable replacement to R22.

- **Global warming potential**

Table (1) shows approximated equal values of (GWP) of the two refrigerants, R22 and R407c.

3.4 Refrigeration Cycle analyses

The refrigeration cycle can be discussed through the following sections:

3.4.1 Introduction

The actual refrigeration cycle deviates slightly from the ideal refrigeration cycle. This is due to the assumptions that are taken for the ideal cycle, which cannot be considered true for the actual cycle. In this research the deviation effects due to superheating and subcooling will be considered. The effects of superheating and subcooling and the combined superheating and subcooling on the performance parameters of a window type air conditioning unit that uses R407c as a refrigerant instead of R22 will be investigated separately.

3.4.2 Ideal vapor compression refrigeration cycle

The vapor compression refrigeration cycle schematic is shown in figure (3.2). Cooling effect is accomplished in the evaporator by evaporation of a liquid refrigerant under reduced pressure and temperature. The pressure-enthalpy diagram and the temperature entropy diagram of vapor compression cycle are shown in figures (3.3.a) and (3.3.b), respectively. Saturated vapor at low pressure enters the compressor and undergoes a reversible adiabatic compression, process (1-2). Heat is then rejected at constant pressure in process, (2-3) and the working fluid leaves the condenser as saturated liquid. An adiabatic throttling process follows, process (3-4), and the working fluid is then evaporated at constant pressure, process (4-1), to complete the cycle.

3.4.3 Superheating and subcooling

A schematic diagram of refrigeration cycle with superheating and subcooling is shown in figure (3.4.a). Compressor sucks the cold and low pressure vapor through the suction line. This line is so arranged that the refrigerant vapor is wormed and is superheated a few degrees as it picks up heat through the walls of the tubings. This causes a rise in temperature. The absorption of heat in the suction line is necessary so that no liquid gets back to the compressor during operation, for the safety of the equipment. Subcooling is a process of cooling the refrigerant below the condensing temperature for a given pressure. This subcooling is done by installing a heat exchanger or with soldering the liquid line with the suction line. In figure (3.4.b) the zones of superheating and subcooling are shown in the (P-h) diagram for the refrigeration cycle.

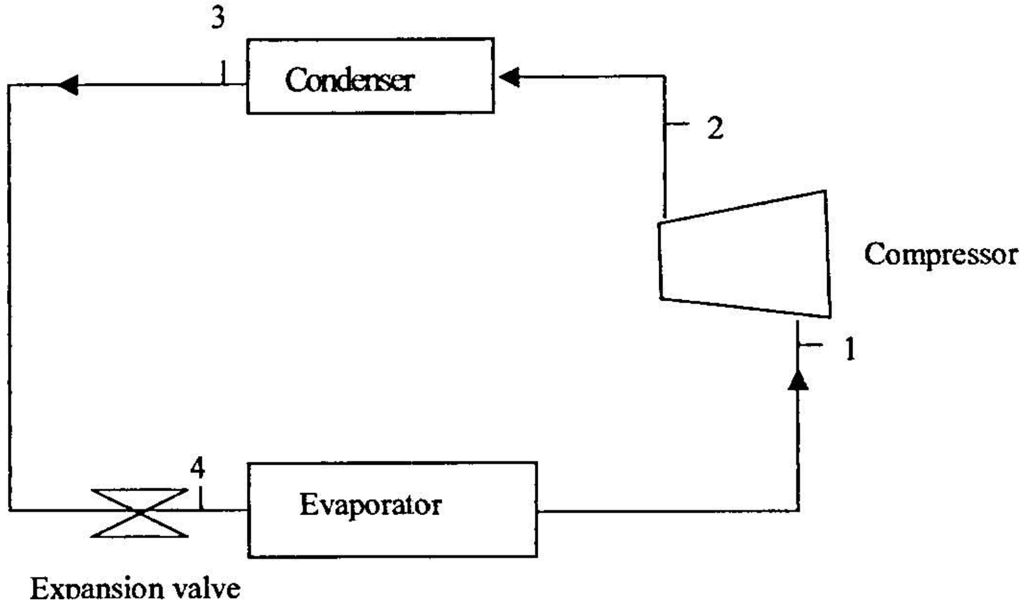

Figure (3.2): Schematic diagram of vapor compression refrigeration cycle

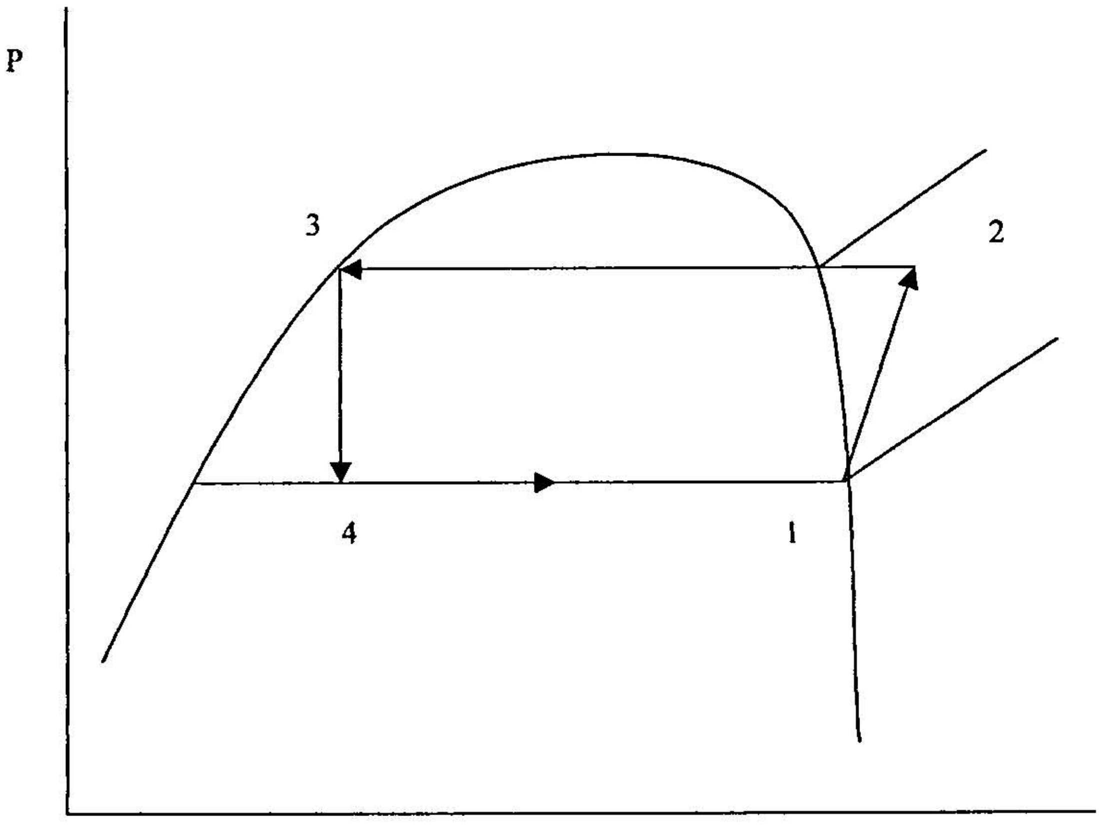

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Figure (3.3.a): P-h diagram for ideal vapor compression cycle

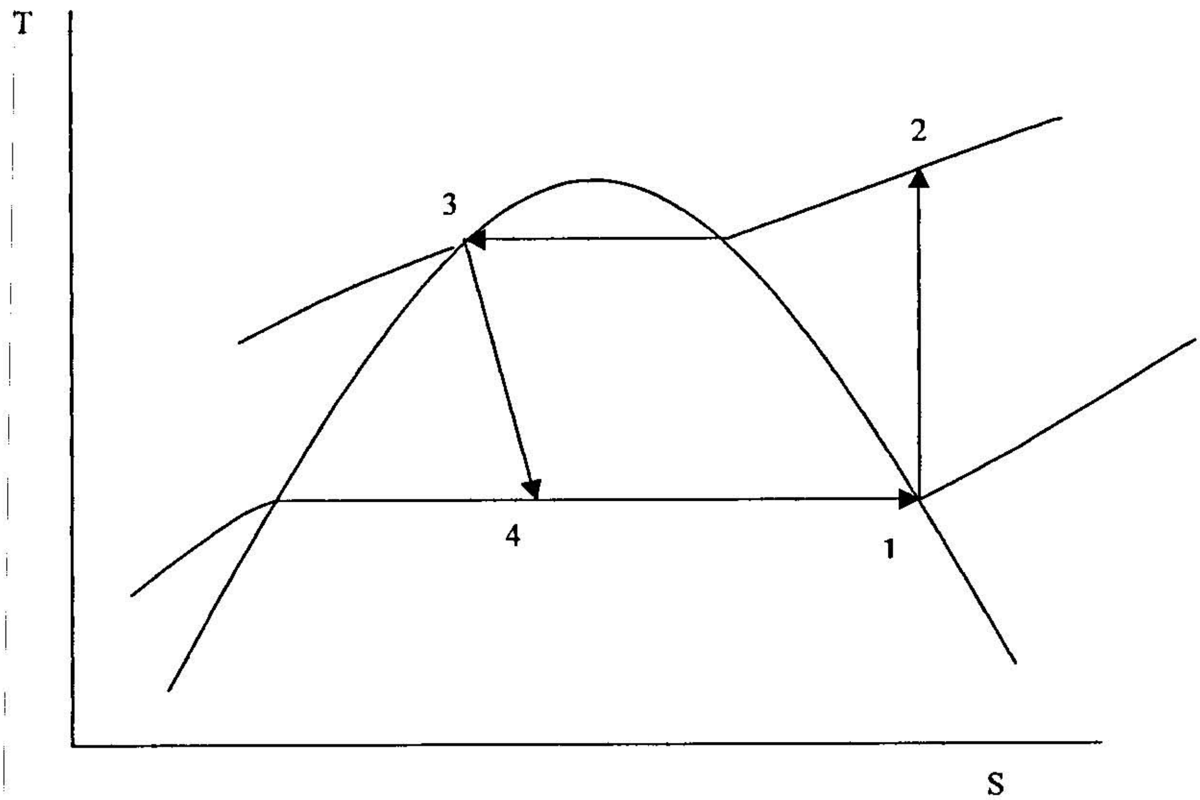


Figure (3.3.b): T-S diagram for ideal vapor compression refrigeration cycle

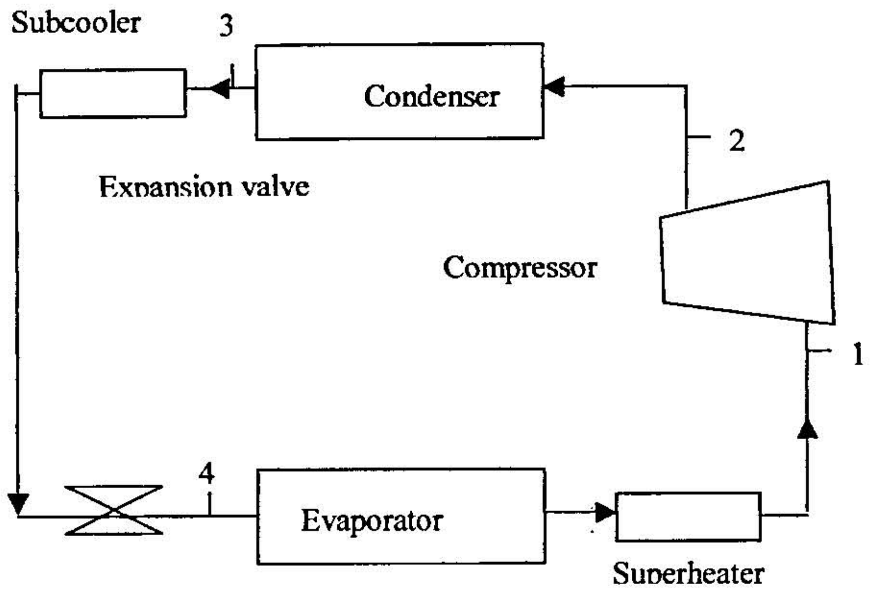


Figure (3.4.a): Schematic diagram of vapor compression refrigeration cycle with superheating and subcooling

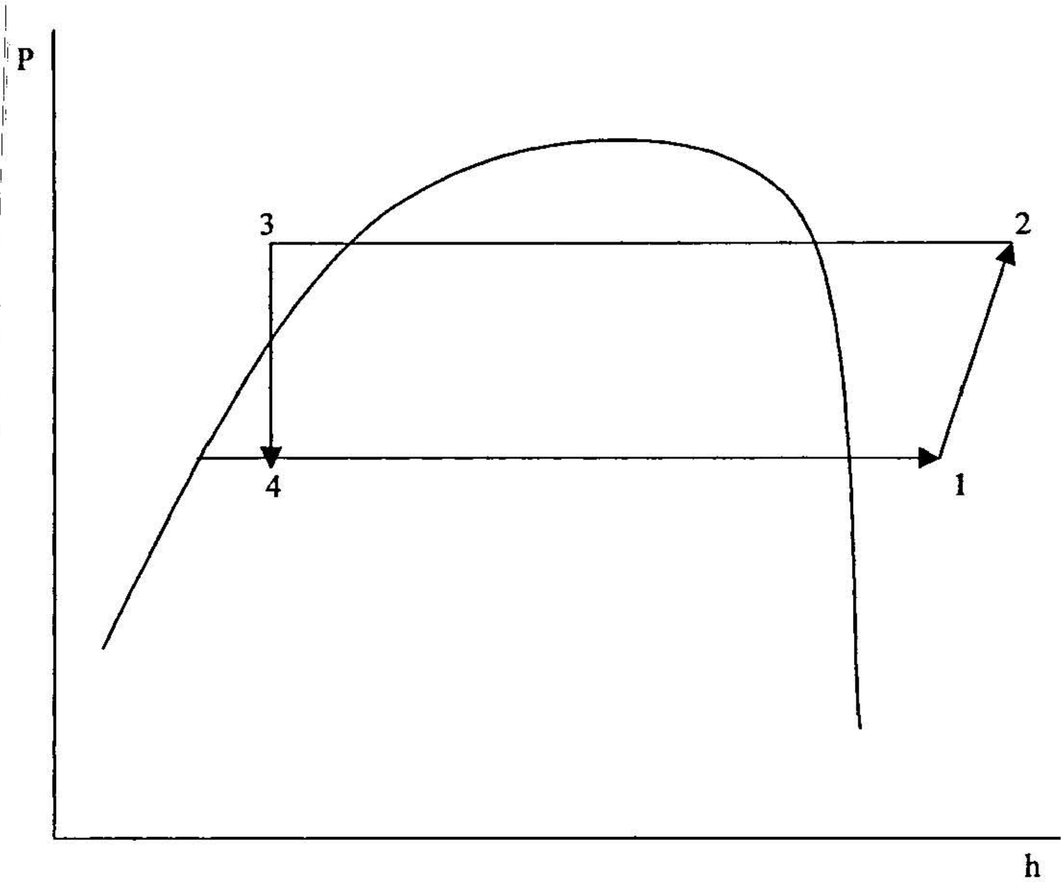


Figure (3.4.b): P-h diagram of vapor compression refrigeration cycle with superheating and subcooling

3.4.4 Refrigeration cycle with superheating and subcooling

As shown in figure (3.4.b) the refrigerant has picked up some superheat in the final evaporator circuit to heat the refrigerant after sufficient latent heat has been added to vaporize all the liquid. So the refrigerant enters the compressor as low temperature, low pressure superheated vapor, (point 1) in figure (3.4.b). The refrigerant leaves the compressor as a high pressure, high temperature superheated vapor, (point 2) in figure (3.4.b). As the refrigerant enters the condenser, heat is removed and the temperature of the refrigerant falls to the saturation temperature. As additional latent heat is removed, the vapor condenses and the condition of the refrigerant is a mixture of high pressure saturated liquid and vapor. At the lower portion of the condenser the refrigerant is completely condensed and is a high pressure liquid. As heat continued to be removed the liquid becomes subcooled and the temperature of the liquid is reduced below its boiling point, (point 3) in figure (3.4.b). In passing through the expansion device, the refrigerant pressure is reduced and the refrigerant is a mixture at this point, (point 4) in figure (3.4.b). Heat from the air or the product being cooled in the evaporator is absorbed by the liquid and causes the refrigerant to vaporize before entering the superheater. The vapor is then superheated above the saturation temperature. The compressor draws the superheated vapor from the superheater to continue the cycle.
In this work the superheating process is taking place inside the refrigerated space in order to obtain a useful refrigeration effect.

3.4.5 Performance parameters investigated

The following performance parameters were investigated throughout this work:

Refrigeration effect: It is the quantity of heat per unit mass of refrigerant absorbed from the refrigerated space. Referring to figure (3.4.b) the refrigeration effect, q_{ref} is expressed as:

$$q_{ref} = h_1 - h_4 \tag{3.1}$$

where h_1 is the specific enthalpy of refrigerant leaving the evaporator and entering the compressor and h_4 is the enthalpy of refrigerant entering the evaporator.

Mass flow rate: The mass flow rate of refrigerant, m, is calculated by using the following formula:

$$m = \eta_0 * P/(h_2 - h_1) \tag{3.2}$$

where η_0 is the overall efficiency that is assumed to equal to (75%), h_2 is the enthalpy of the superheated vapor at the end of the compression state and P is the net electrical compressor power consumed which is expressed as follows:

$$P = P_1 - P_2 \tag{3.3}$$

where P_1 is the total electrical power consumed by the condenser and evaporator fans and the compressor and P_2 is the electrical power consumed by the two fans and equal to 1/3 of the total consumed power.

Refrigeration capacity: The refrigeration capacity, Q_{ref}, is the rate of heat removed from the refrigerated space. It is expressed as:

$$Q_{ref} = m * q_{ref} \tag{3.4}$$

Work of compression: The work of compression, w_c, consumed by the compressor is given by:

$$w_c = h_2 - h_1 \tag{3.5}$$

Heat rejection: The heat rejection, q_c, is the amount of heat rejected in the condenser to the condensing medium. It is expressed as follows:

$$q_c = h_2 - h_3 \tag{3.6}$$

where h_3 is the enthalpy of refrigerant leaving the condenser.

Heat rejection rate: The rate of heat rejected in the condenser to the condensing medium is expressed as follows:

$$Q_c = m \ (h_2 - h_3) \tag{3.7}$$

Coefficient of performance: The coefficient of performance, COP, is an expression of the efficiency of the system and it is given by:

$$COP = q_{ref}/w_c \tag{3.8}$$

Isentropic efficiency: Isentropic efficiency, η_s, is the ratio of the isentropic work to the actual work of the compressor. It is given by:

$$\eta_s = h_{2s}-h_1/h_2-h_1 \tag{3.9}$$

where h_{2s} is the isentropic enthalpy of the superheated vapor leaving the compressor.

Volumetric efficiency: The volumetric efficiency, η_v, is used to predict the performance of the reciprocating compressors and it depends on the re-expansion of the gas trapped in the clearance volume of the compressor cylinder. It is calculated by:

$$\eta_v = 100-M\,((v_s/v_d)-1) \tag{3.10}$$

Where M is the percent clearance, v_s is the specific volume of refrigerant at the compressor suction and v_d is the specific volume at the compressor discharge.

4-EXPERIMENTAL WORK PROCEDURE

4.1 Introduction

Experiments of this research were conducted by varying the evaporating temperature while keeping the condensing temperature constant. For each condensing temperature investigated, the amount of degree of superheat was varied while keeping the amount of degree of subcool constant. Then the experiments were repeated for different values of degree of subcooling. All experiments were repeated by varying the condenser temperature while keeping the evaporator temperature constant with various amounts of degrees of superheating and subcooling.

4.2 The air conditioning unit specifications

Table (2) shows the specifications of the air conditioning unit used as supplied by the manufacturer.

Table (2): specifications of the air conditioning unit

Type	Window-Type
Model	AD917WIGI, SER NO: G241265
Voltage	220-240 Volts
Starting current	11.2 Amperes
Cooling capacity	5 kW
Refrigerant and charge	R22: 965g, R407c: 900g
Lubricant capacity	650cc
Pressure Limits	2.413 MPa (high), 1.034 MPa (low)

power consumed by the compressor can be calculated by subtracting the power consumed by the two fans from the total power consumption.

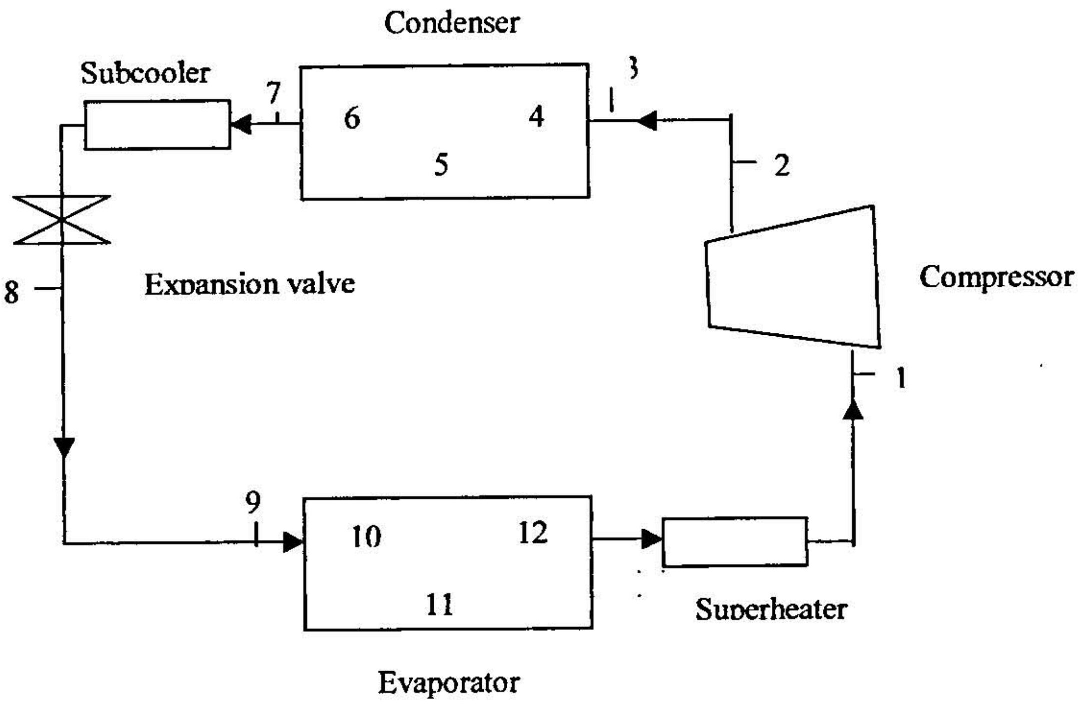


Figure (4.1): Locations of thermocouples

4.4 Experimental work steps

The experimental work can be summarized as follows:

4.4.1 Retrofit procedure

The following procedure is used to charge the refrigeration unit :

1. Recover the existing R22 charge
2. Drain the original oil from the system and recharge using an approved POE lubricant.
3. Recharge the system using R22 and run the system to circulate the new lubricant.
4. Recover the R22 charge again and check the residual oil content of the lubricant.
5. Repeat steps 2, 3 and 4 until the lubricant charge is greater than 95% (POE). At this point, perform standard maintenance on the system such as replacing the filter–drier and fixing any leaks which have been located.
6. Evacuate the system using a deep vacuum.
7. Charge the system with R407c.Be sure to remove the refrigerant from the cylinder as a liquid.
8. Start the system and adjust controls and / or charge until you achieve desired operation.

4.4.2 Evacuation process

The purpose of the evacuation of the system is to ensure removing the air from the system and to ensure also removing the moisture and the R22 refrigerant in order to charge the system with pure R407c. The evacuation process was performed as follows:

1. Connect the hose of the gauge manifold between the vacuum pump and the suction line of the compressor.
2. Start the vacuum pump and continue the evacuation process until the desired vacuum pressure is obtained.
3. Charge the unit with R407c.

The following methods are used to obtain the optimum charge of refrigerants in air conditioning units:

1. Using a sight glass: During the charging process the optimum charge can be observed by viewing a clean stream of liquid refrigerant through the site glass. Flow of bubbles is an indication of having a refrigerant shortage.
2. The formation of dew on the inlet of the evaporator. This will indicate a good performance of the system and an indication of possible optimum charge.
3. Monitoring the difference of the air entering temperature and the air leaving temperature.
4. Calculating the coefficient of performance (COP) at different charges.

In this work the following two methods were used to obtain the optimum charge of R22 and R407c:

1. Calculating the coefficient of performance at different charges.
2. The formation of dew point at the inlet of the evaporator.

5-RESULTS AND DISCUSSION

5.1 Introduction

The performance parameters obtained for refrigerant, R22, and alternative refrigerant, R407c, were plotted versus variable values of the degrees of superheating and the degrees of subcooling at different evaporating and condensing temperatures.

The performance parameters investigated in this work (as shown in the appendix) are refrigeration effect, mass flow rate, refrigeration capacity, work of compression, heat rejection, heat rejection rate, coefficient of performance, electrical power consumption, compressor discharge temperature, isentropic efficiency and volumetric efficiency.

5.2 Superheating variation tests

Superheating variation tests were carried out for both refrigerants R22 and R407c at constant degree of subcooling of 9°C and constant evaporating temperature of 5°C and degrees of super heating of 3, 6, 10, 16°C and condensing temperatures of 38, 40, 41°C. The above parameters were obtained as follows:

5.2.1 Refrigeration effect

Equation (3.1) is used to calculate the refrigeration effect, q_{ref}, for each value of ΔT (sup) investigated. Figures (5.1) and (5.2) show the variation of q_{ref} with ΔT (sup) for R22 and R407c, respectively for T_c=38, 40 and 41°C. It can be seen from these figures that q_{ref} increases linearly with ΔT (sup) for both refrigerants. Figures (5.1) and (5.2) also show that when the condensing temperature increases, the refrigeration effect decreases since, at the same degree of superheating, same degree of subcooling, same while the enthalpy at evaporator exit, h_1, keeps constant. According to equation (3.1) this means that the value of the refrigeration effect decreases as the condensing temperature increases while it increases as ΔT (sup) increases at T_e=constant and T_c=constant. Figures (5.1) and (5.2) indicate that the refrigeration effect, q_{ref}, for R407c reached a value of 101% of that of R22 at T_c = 38°C and ΔT (sup) = 16°C. Figure (5.3) compares q_{ref} of R22 and R407c for T_c=40°C. It shows that the refrigeration effect for both refrigerants increases as ΔT (sup) increases due to the increase of the enthalpy of the superheated vapor that leaves the evaporator as ΔT (sup) increases. Figure (5.3) also shows that the refrigeration effect of R407c reached a value of 99% of that of R22 at different ΔT (sup) from 3°C to 10°C and increased slightly till it reached a value of 101% of that of R22 at ΔT (sup) = 16°C.

5.2.2 Mass flow rate

Equation (3.2) was used to calculate the mass flow rate, m, values at each ΔT (sup) investigated. Figures (5.4) and (5.5) show the variation of m with ΔT (sup) for R22 and R407c, respectively for T_c=38, 40, 41°C. Both figures indicate that m decreases as ΔT (sup) increases for all values of T_c investigated. Figures (5.4) and (5.5) show that the mass flow rate decreases as T_c increases due to the increase of enthalpy across the compressor and due to the decrease in the specific volume at the exit of the compressor. And also these figures show that the mass flow rate of R407c is lower than that of R22 for all values of T_c investigated. As indicated in figure (5.5) the mass flow rate of R407c at T_c = 38°C, ΔT (sup) = 6°C is greater by 5.6% than that at T_c = 41°C, ΔT (sup) = 6°C. As ΔT (sup) increases the mass flow rate decreases until a point is reached at which it tends to be constant. This is because, at constant T_c and constant T_e, the rate of increase of the specific volume at the inlet of the compressor is nearly equal to the rate of increase of the specific volume at the exit of the compressor which makes the volumetric efficiency constant which directly proportional to the mass flow rate. Figure (5.6) shows that the mass flow rate decreases as ΔT (sup) increases due to the increase of the specific volume of the superheated vapor at the inlet of the compressor. Figure (5.6) also shows that the mass flow rate of R407c is lower than that of R22 for all values of ΔT (sup) investigated. It indicates that the mass flow rate of R407c is less by 18.9% than that of R22 at ΔT (sup)=3°C.

5.2.3 Refrigeration Capacity

The refrigeration capacity at each value of ΔT (sup) investigated was calculated using equation (5.4). Figures (5.7) and (5.8) represent the relation between the refrigeration capacity, Q_{ref}, and the degrees of superheating, ΔT (sup) at different condensing temperatures, T_c, for both refrigerants R22 and R407c respectively. Both figures indicate that the refrigeration capacity increases as ΔT (sup) increases while it decreases as T_c increases for both refrigerants. Increasing T_c will increase the saturated liquid enthalpy at the condensing temperature, thus causing a decrease in the enthalpy difference across the evaporator. Also the mass flow rate decreases as T_c increases, thus according to equation (5.4) the refrigeration capacity decreases. Figures (5.7) and (5.8) also show that the refrigeration capacity of R407c reached a value of 80% of that of R22 at T_c=38°C, ΔT (sup)=16°C. Figure (5.9) shows that Q_{ref} of R407c is less than that of R22 at different degrees of superheating. It also shows that as the degree of superheating increases Q_{ref} increases slightly for both refrigerants. This is because the rate of increase of the refrigeration effect is greater than the rate of decrease of the mass flow rate. Figure (5.9) also shows that Q_{ref} of R407c decreased by 18.8% as compared to that of R22 at ΔT (sup) = 16°C.

5.2.4 Work of Compression

The work of compression, w_c, is plotted against ΔT (sup) at different values of T_c for both refrigerants, R22 and R407c as shown in figures (5.10) and (5.11) respectively. Equation (3.5) was used to calculate the data of these figures. As T_c increases w_c increases. This is due to the increase in the enthalpy difference across the compressor. The work of compression of R407c at T_c = 41°C, ΔT (sup) = 16°C is decreased by a value of 7.5% of that at T_c = 38°C, ΔT (sup) = 16°C as shown in figure (5.11) while a percentage decrease of the work of compression of 3.2% is obtained at the same conditions for R22 as shown in figure (5.10). The work of compression is plotted against ΔT (sup) for both refrigerants in figure (5.12). This figure compares w_c of R22 and R407c for T_c = 40°C. It shows that as ΔT (sup) increases w_c increases for both refrigerants. This is due to the increase of the enthalpy difference across the compressor. Figure (5.12) indicates that w_c of R407c is larger than that of R22 for all values of ΔT (sup) investigated. It is increased by a value of 23.2% of that of R22 at ΔT (sup) = 16°C.

5.2.5 Heat rejection

Equation (3.6) is used to determine the value of heat rejection, q_c, for each value of ΔT (sup) investigated. Figures (5.13) and (5.14) show the variation of q_c with ΔT (sup) for R22 and R407c, respectively for T_c =38, 40 and 41°C. It can be seen from these figures that q_c increases linearly with ΔT (sup) for both refrigerants. This is because of the increase of the enthalpy of the refrigerant leaving the compressor while keeping the enthalpy leaving the condenser constant. Figures (5.13) and (5.14) indicate that q_c increases as T_c decreases for each ΔT (sup) investigated. This is because the enthalpies at inlet and exit of the condenser increase as T_c increases, but the increase in the exit enthalpy (saturated liquid enthalpy at condensing temperature) is higher than the increase in the inlet enthalpy causing the enthalpy difference across the condenser to decrease.

Figure (5.15) compares q_c of R22 and R407c for T_c = 40°C. It shows that q_c of R407c is greater than that of R22 for all values of ΔT (sup) investigated. The value of q_c of R407c is greater than that of R22 by 5.2% at ΔT (sup) = 16 °C.

5.2.6 Heat rejection rate

Equation (3.7) is used to determine the heat rejection rate, Q_c, for each value of ΔT (sup) investigated. Figures (5.16) and (5.17) show the variation of Q_c with ΔT (sup) for R22 and R407c, respectively for T_c = 38, 40 and 41°C.

These figures show that Q_c increases as ΔT (sup) increases for both refrigerants. This is due to the fact that the mass flow rate decreases at lower rate than the increase in the enthalpy difference across the condenser. Also these figures show that Q_c decreases as T_c increases for both refrigerants. This is because the mass flow rate of refrigerants decreases and the enthalpies at inlet and exit of the condenser increase as T_c increases but the increase in the exit enthalpy is higher than the increase in the inlet enthalpy causing the enthalpy difference across the condenser to decrease. Figures (5.16) and (5.17) indicate that Q_c of R407c is lower than that of R22 by 11.7% for T_c = 38 °C, ΔT (sup) = 16 °C.

In comparing Q_c of the two refrigerants, figure (5.18) shows that Q_c of R407c is lower than that of R22 for all degrees of superheating investigated. It indicates that Q_c of R407c is lower than that of R22 by 15.1% for T_c=40 °C and ΔT (sup) = 16 °C.

5.2.7 Coefficient of performance

Equation (3.8) is used to determine the value of the coefficient of performance, COP, for each value of ΔT (sup) investigated. Figures (5.19) and (5.20) show the variation of COP with ΔT (sup) for R22 and R407c, respectively for T_c = 38, 40, 41 °C. It can be seen from these figures that COP increases as ΔT (sup) increases for both refrigerants. This is due to the fact that the rate of increase of the enthalpy difference across the evaporator is

greater than the rate of increase of the enthalpy difference across the compressor. Also these figures show that COP decreases as T_c increases for both refrigerants. This is because of the decrease in the enthalpy difference across the evaporator and the increase in the enthalpy difference across the compressor. Figures (5.19) and (5.20) indicates that the COP of R407c reached a value of 87.4% of that of R22 for T_c = 38 °C, ΔT (sup) = 16 °C. Figure (5.21) compares COP of R22 and R407c for T_c = 40 °C. It indicates that COP of R407c is lower than that of R22 and it reached a value of 83% of that of R22 at ΔT (sup) = 16 °C.

5.2.8 Electrical power consumption

The variation of the electrical power consumption for both refrigerants, R22 and R407c with ΔT (sup) is plotted in figures (5.22) and (5.23), respectively. These figures show that the electrical power consumption increases as ΔT (sup) increases for both refrigerants due to the increase of the work of compression in a rate greater than the rate of decrease of the mass flow rate. Figures (5.22) and (5.23) also show that the compressor power increases as T_c increases at constant T_e. This is because increasing T_c causes increase in the enthalpy difference and a slight decrease in mass flow rate. Figure (5.24) shows the variation of electrical power consumption with ΔT (sup) for $T_c = 40$ °C, for both refrigerants. This figure indicates that the electrical power consumption when using R407c is slightly less than that when using R22 and it reached a value of 98.7% of that of R22 at ΔT (sup) = 10 °C.

5.2.9 Isentropic efficiency

Equation (3.9) is used to determine the isentropic efficiency, η_{isent}, for each value of ΔT (sup) investigated. Figure (5.25) shows the variation of η_{isent} with ΔT (sup) for both refrigerants. This figure shows that η_{isent} tends to be constant at different ΔT (sup) for both refrigerants. Figure (5.25) also shows that η_{isent} of R407c is less than that of R22 at constant T_c and constant T_e and different values of ΔT (sup). This figure indicates that η_{isent} of R407c reached a value of about 71.5% of that of R22.

5.2.10 Volumetric efficiency

The volumetric efficiency, η_v, as a function of ΔT (sup) is calculated using equation (3.10). Figures (5.26) and (5.27) show the relation between η_v and ΔT (sup) for R22 and R407c, respectively for T_c =38, 40 and 41 °C. It can be seen from these figures that η_v tends to be constant as increasing ΔT (sup) for both refrigerants. This is because the rate of increase of the specific volume of the superheated vapor at the exit of the compressor is nearly equal to the rate of increase of the specific volume of the super heated vapor at the inlet of the compressor. Figures (5.26) and (5.27) also show that η_v decreases as T_c increases for constant T_e, since increasing T_c will decrease the specific volume of the refrigerant at compressor exit. Figure (5.28) compares η_v of R22 and R407c for T_c = 40 °C. It can be seen from this figure that η_v of R407c is slightly greater than that of R22 and it reached a value of about 101% of that of R22. This means that same compressor can be used for both refrigerants with out any modifications.

5.2.11 Compressor discharge temperature

Figures (5.29) and (5.30) show the variation of compressor discharge temperature, T_s, with ΔT (sup) for R22 and R407c, respectively for T_c =38, 40, 41 °C. It can be seen from these figures that T_s increases linearly with ΔT (sup) for both refrigerants. Also these figures show that as T_c increases the exit temperature from the compressor increases since condensing pressure increases. For R407c T_s increased from 92 to 95 °C when increasing T_c from 38 °C to 41 °C for ΔT (sup) = 16°C. Figure (5.31) compares T_s of R22 and R407c for T_c = 40 °C. This figure indicates that T_s of R407c is less than that of R22 and it is decreased from 86.5 to 82 °C when R407c is used instead of R22 for ΔT (sup) = 3 °C.

It is worthfull to mention that high temperature at the exit of compressor could result in oil break down, causing excessive wear or reducing life of discharge valves and over heating of compressor. To achieve a certain exit temperature the condensing temperature and the degree of super heating should be controlled.

5.3 Subcooling variation tests

Subcooling variation tests were carried out for both refrigerants, R22 and R407c at constant degree of superheating, ΔT (sup) of 6 °C and constant condensing temperature of 40 °C and degrees of subcooling, ΔT (sub) of 9, 11.5, 14, 17 °C. The tests were repeated for evaporating temperatures of 5, 7, 13 °C and the following Parameters were obtained:

5.3.1 Refrigeration effect

Figures (5.32) and (5.33) show the variation of q_{ref} with ΔT (sub) for R22 and R407c, respectively for T_e = 5, 7 and 13 °C. It can be seen from these figures that q_{ref} increases linearly with ΔT (sub) for both refrigerants. This is due to the fact that the enthalpy of the liquid vapor mixture that enters the evaporator as a cause of subcooling was decreased while the enthalpy of the vapor that enters the compressor was kept constant. Figures (5.32) and (5.33) also show that q_{ref} increases as T_e increases for both refrigerants. The increase of q_{ref} is due to slightly higher enthalpy value of the superheated vapor at higher evaporating temperatures. Figure (5.33) indicates that q_{ref} of R407c at T_e = 13 °C, ΔT (sub) = 17 °C is greater than that at T_e = 5 °C by about 5%. This makes R407c suitable to be used in moderate and high temperature systems. Figure (5.34) compares the variation of q_{ref} with ΔT (sub) for R22 and R407c for T_e=5 °C. This figure shows that q_{ref} of R407c is greater than that of R22 and it increased by a value of about 3% of that of R22 at ΔT (sub) = 17 °C. This means that R407c has a larger subcooling effect than R22.

5.3.2 Refrigeration Capacity

Figures (5.35) and (5.36) show the variation of the refrigeration capacity, Q_{ref}, with ΔT (sub) for R22 and R407c, respectively for T_e = 5, 7 and 13 °C. These figures show that Q_{ref} increases as ΔT (sub) increases for both refrigerants. This is due to the fact that the refrigeration effect increases while the mass flow rate keeps constant with the change of ΔT (sub). It can be seen also from these figures that Q_{ref} increases as T_e increases for both refrigerants R22, R407c at a rate of 6% and 4% respectively. Figure (5.37) compares Q_{ref} of R22 and R407c for T_e = 5 °C. It indicates that Q_{ref} of R407c is less than that of R22 and it reached a value of 84% of that of R22.

5.3.3 Heat rejection

Figures (5.38) and (5.39) show the variation of heat rejection, q_c, with ΔT (sub) for R22 and R407c, respectively for T_e = 5, 7 and 13 °C. Both figures indicate that q_c increases as ΔT (sub) increases for both refrigerants. This is due to the decrease of the enthalpy of the saturated liquid at the exit of the condenser as ΔT (sub) increases while the enthalpy at the exit of the compressor is kept constant.

As shown in these figures the heat rejection increases as T_e decreases for both refrigerants. This is because of the increase of the enthalpy at the exit of the compressor as T_e decreases. Figures (5.38) and (5.39) indicate that q_c of R407c for T_e = 13 °C is lower than that for T_e = 5 °C by 2.2% while a value of 3% is obtained for R22 at the same conditions. The heat rejection of R407c is greater than that of R22 as indicated in figure (5.40). It can be seen from this figure that q_c of R407c is greater than that of R22 by 5.3% for T_e=5 °C.

5.3.4 Heat rejection rate

Figure (5.41) and (5.42) show the variation of the heat rejection rate, Q_c, with ΔT (sub) for R22 and R407c, respectively for T_e = 5, 7 and 13 °C. These figures show that Q_c increases as ΔT (sub) increases for both refrigerants. This is due to the increase of the enthalpy difference across the condenser as ΔT (sub) increases while the mass flow rate remains constant. Figures (5.41) and (5.42) indicate that Q_c increases as T_e increases for both refrigerants. This is due to the increase of the mass flow rate at higher rate than the decrease in the enthalpy difference across the condenser. The heat rejection rate of R407c is less than that of R22 as indicated in figure (5.43). It can be seen from this figure that Q_c of R407c is less than that of R22 by 15.3% for T_e=5 °C.

5.3.5 Coefficient of Performance

Figures (5.44) and (5.45) show the variation of COP with ΔT (sub) for R22 and R407c respectively for T_e = 5, 7 and 13°C. It can be seen from these figures that COP increases as ΔT (sub) increases for both refrigerants. This is because of the increase of q_{ref} as ΔT (sub) increases while w_c remains constant. Figures (5.44) and (5.45) also show that in addition to the increase of COP with ΔT (sub) it also increases as T_e increases for both refrigerants. This is due to the increase of the refrigeration effect and the decrease in work of compression. These figures indicate that COP of R407c is increased by a value of 6% as T_e increased from 5 to 13 °C while a percentage increase of 6.5% is obtained for R22 at the same conditions. Figure (5.46) compares COP of R22 and R407c with ΔT (sub) for T_e = 5 °C. This figure indicates that COP of R407c is less than that of R22 and it reached a value of 84% of that of R22 at ΔT (sub) = 17 °C.

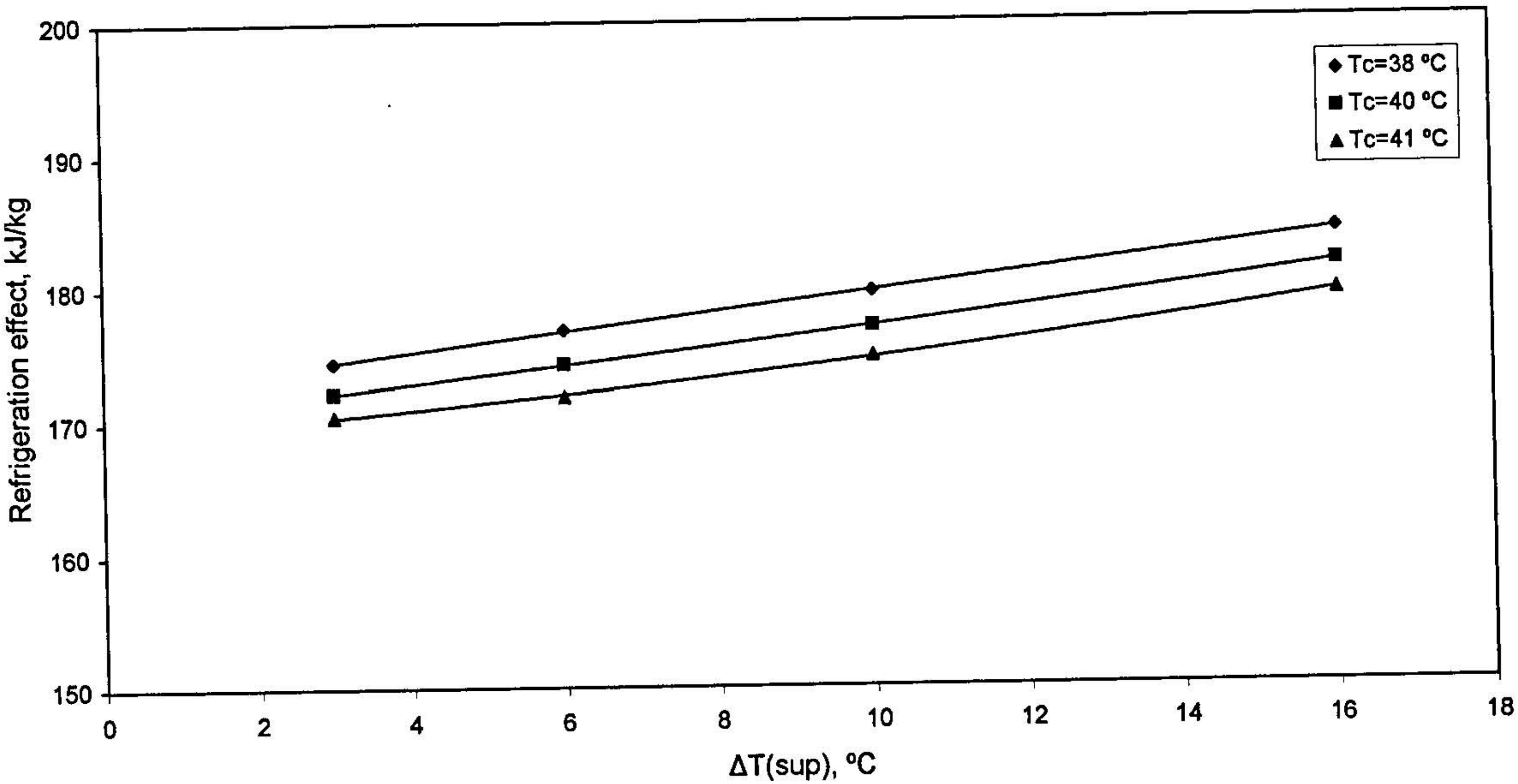


Figure 5.1:Variation of refregeration effect with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9°C for R22

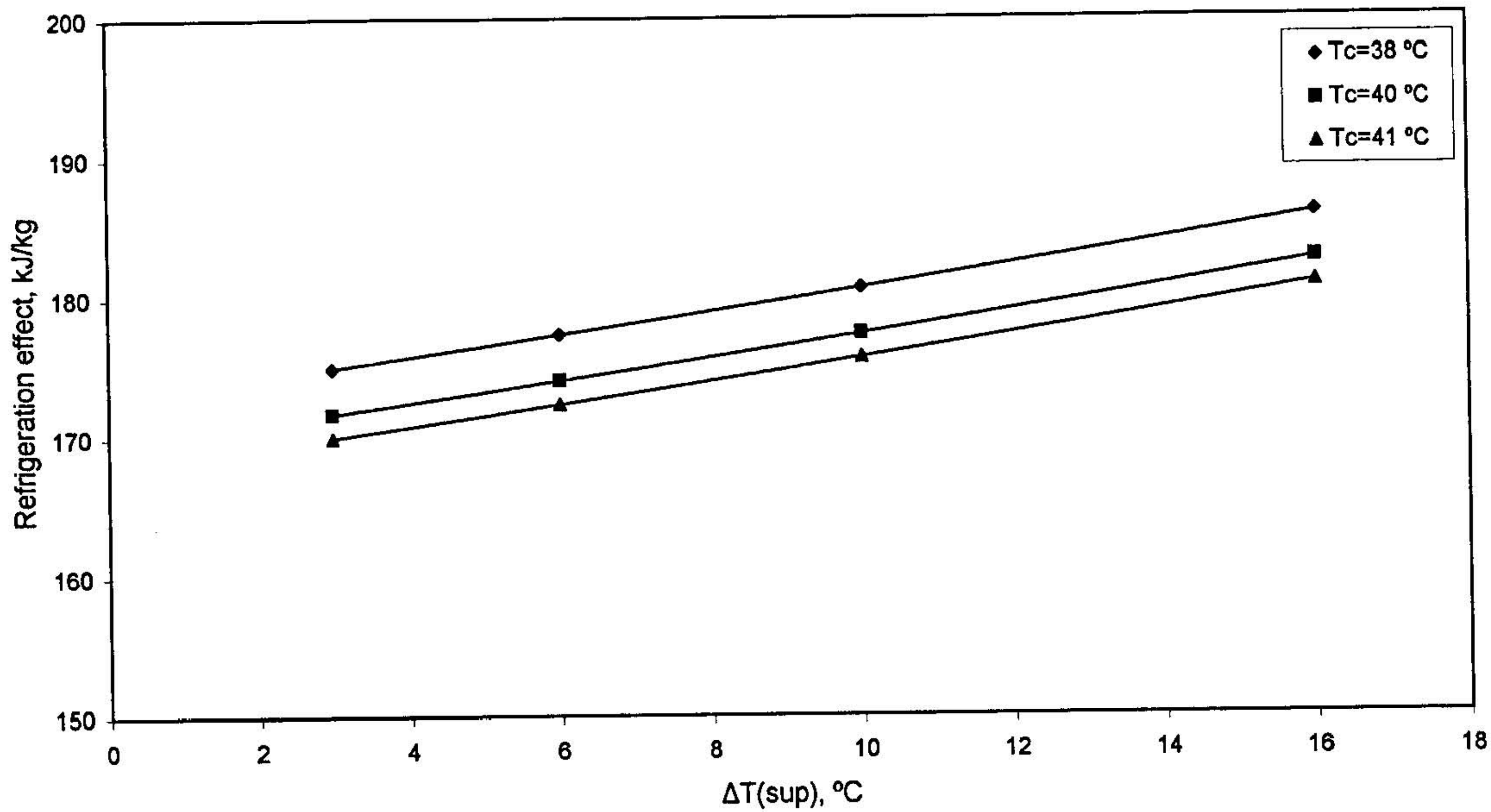


Figure 5.2:Variation of refrigeration effect with $\Delta T(sup)$ at T_e=5 °C, $\Delta T(sub)$=9 °C for R407c

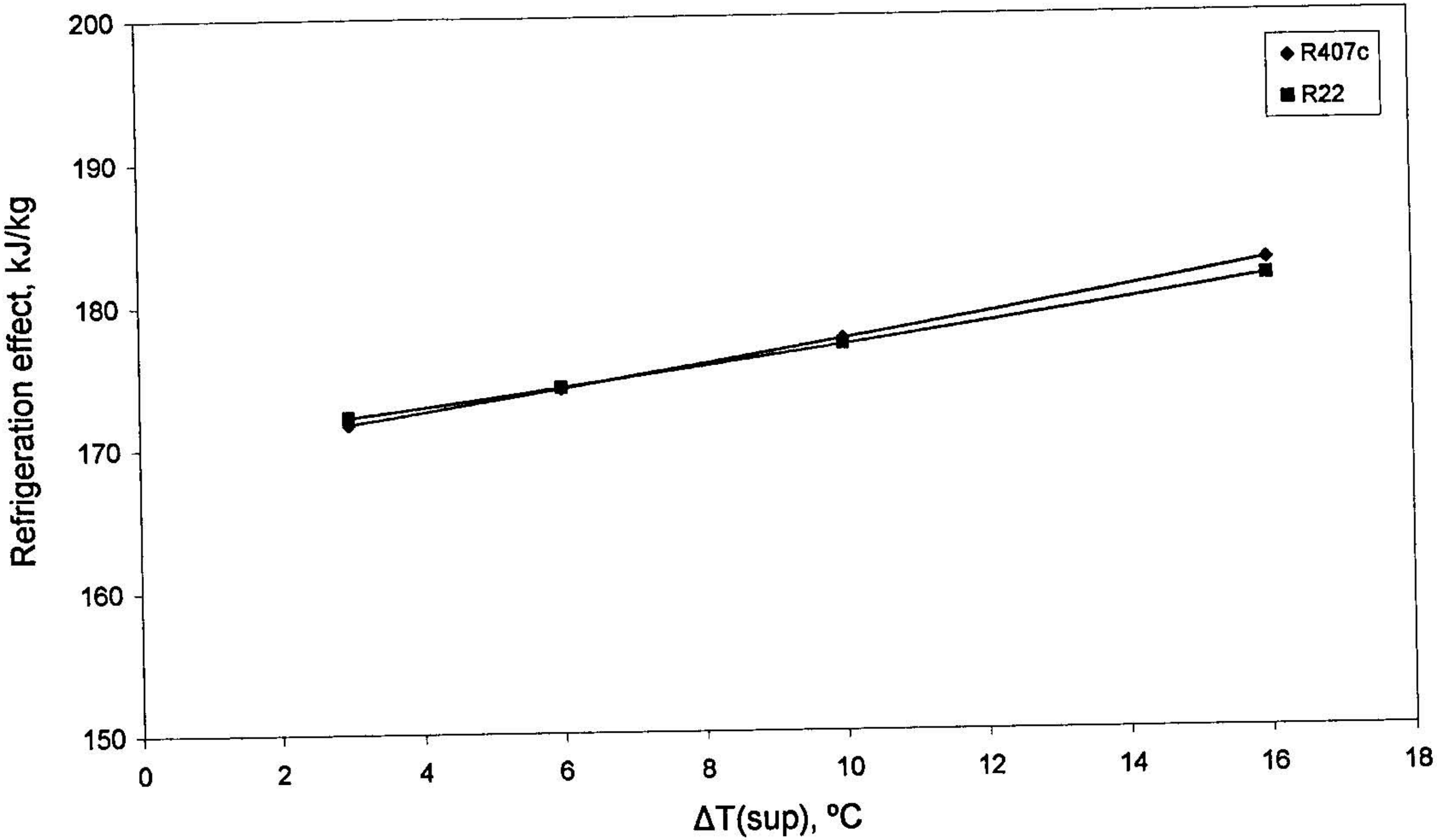


Figure 5.3:Variation of refrigeration effect with ΔT(sup) at T_e=5 °C, T_c=40 °C, ΔT(sub)=9 °C

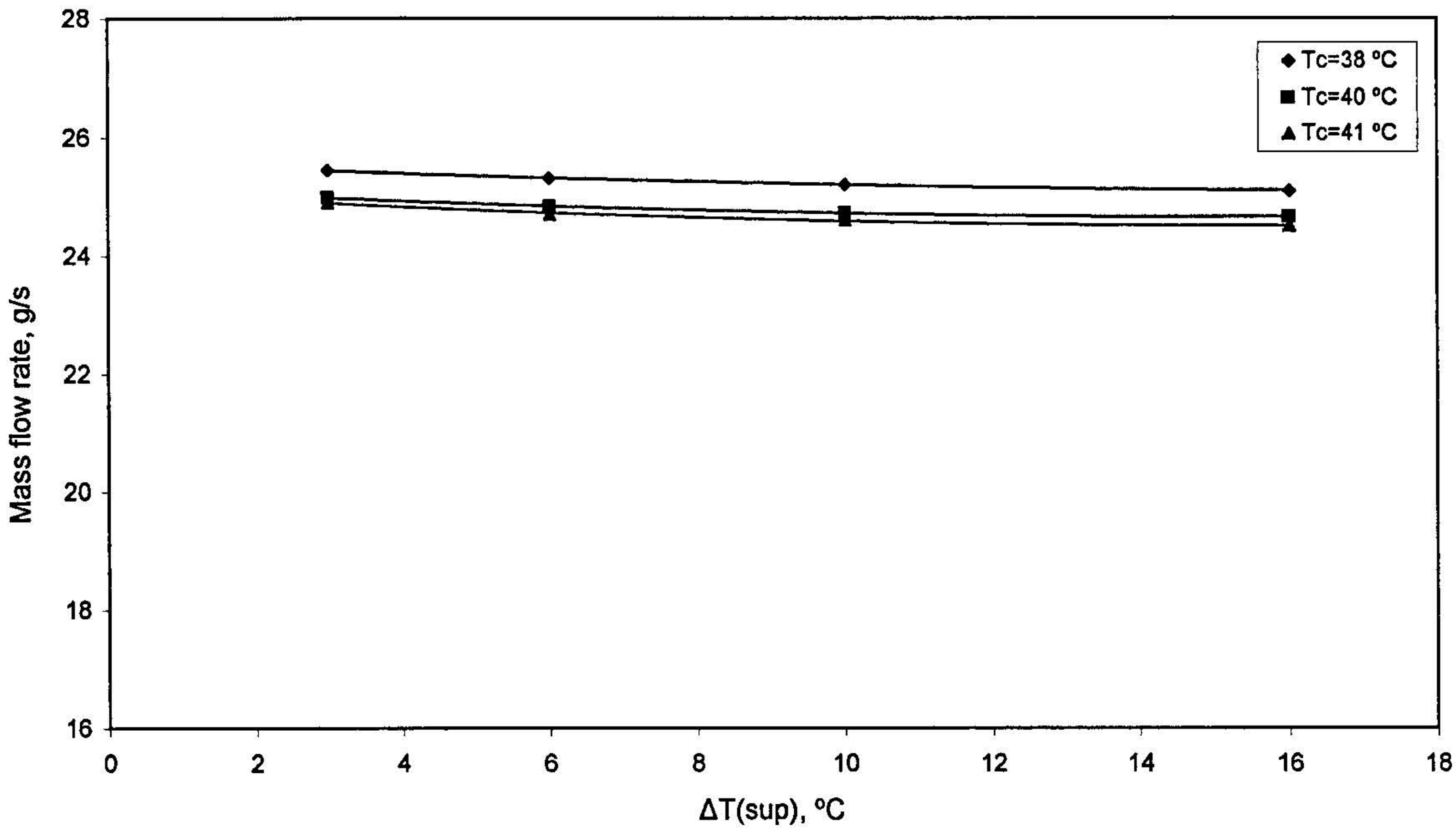


Figure 5.4:Variation of mass flow rate with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9°C for R22

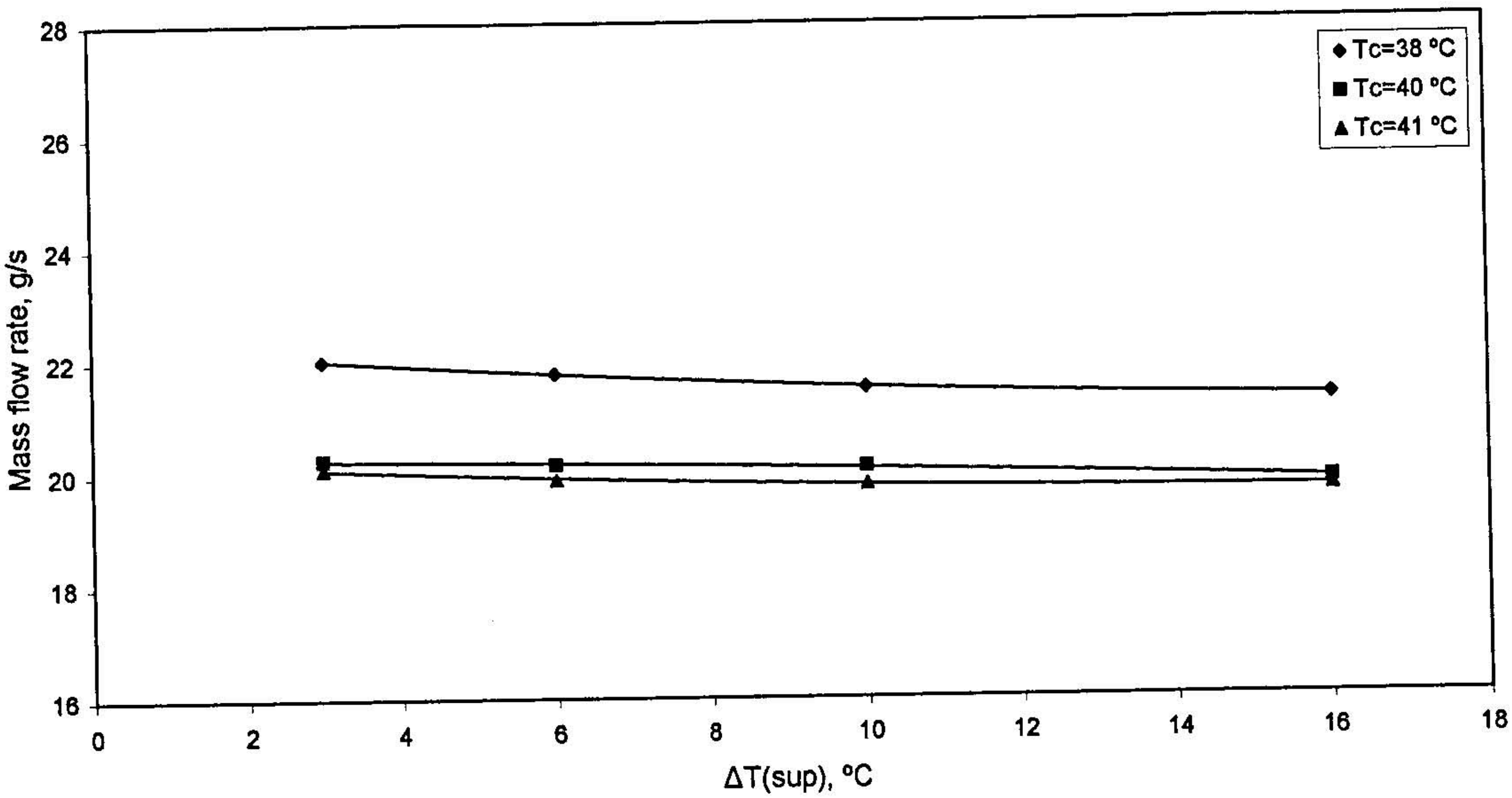


Figure 5.5:Variation of mass flow rate with ΔT (sup) at T_e=5 °C, ΔT (sub)=9 °C for R407c

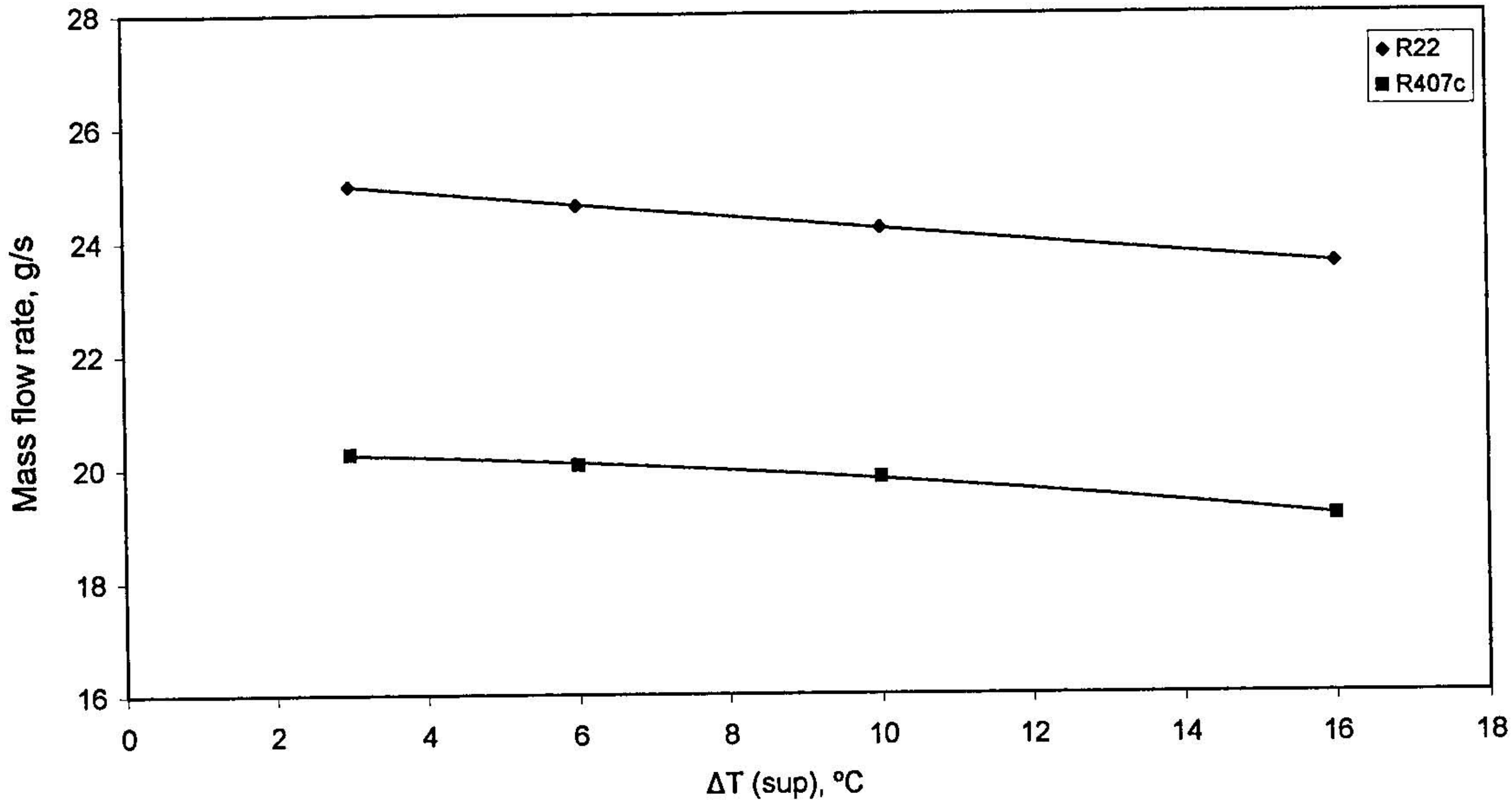


Figure 5.6:Variation of mass flow rate with ΔT (sup) at T_e=5 °C, T_c=40 °C, ΔT (sub)=9 °C

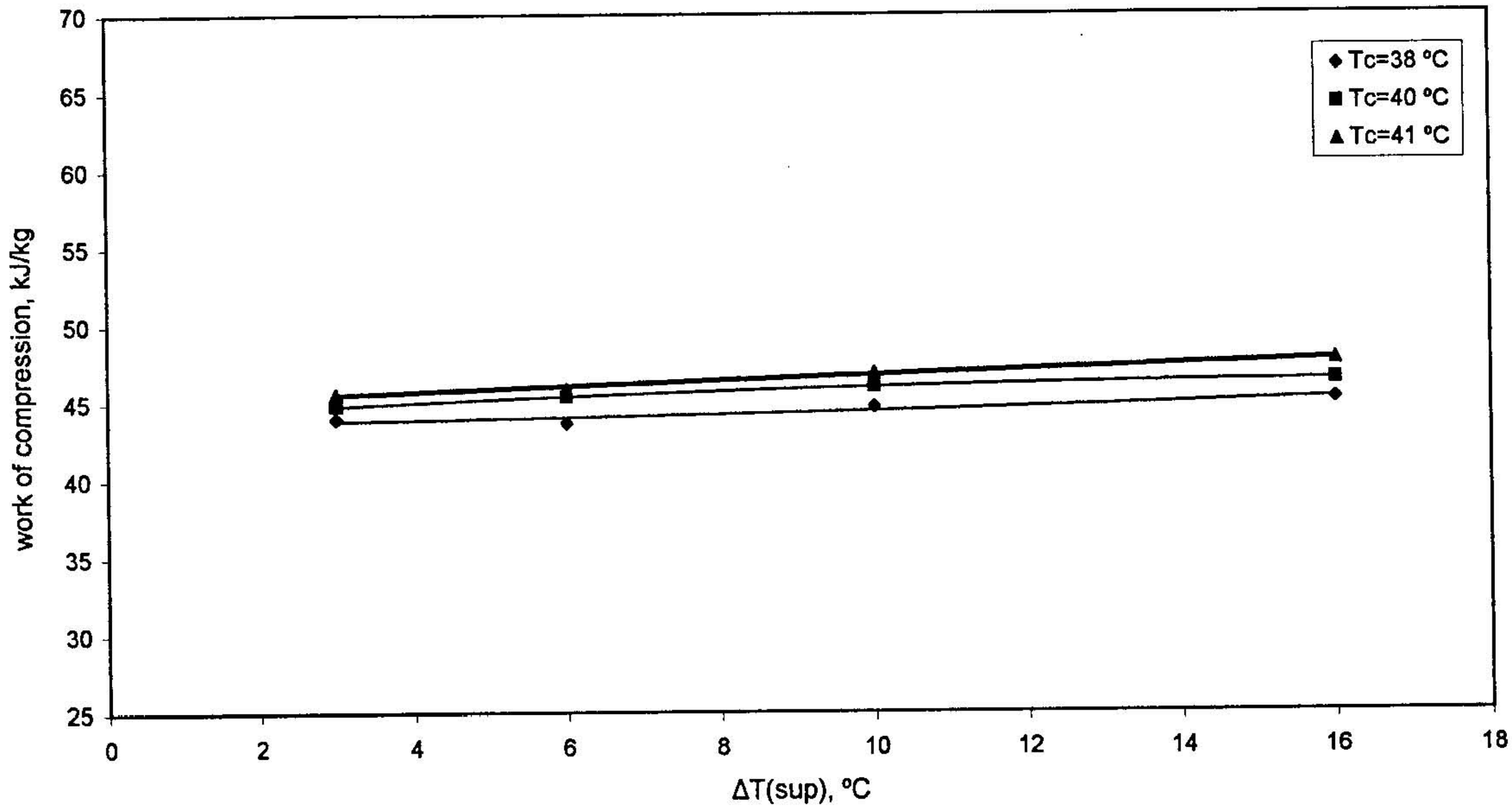


Figure 5.10:Variation of work of compression with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22

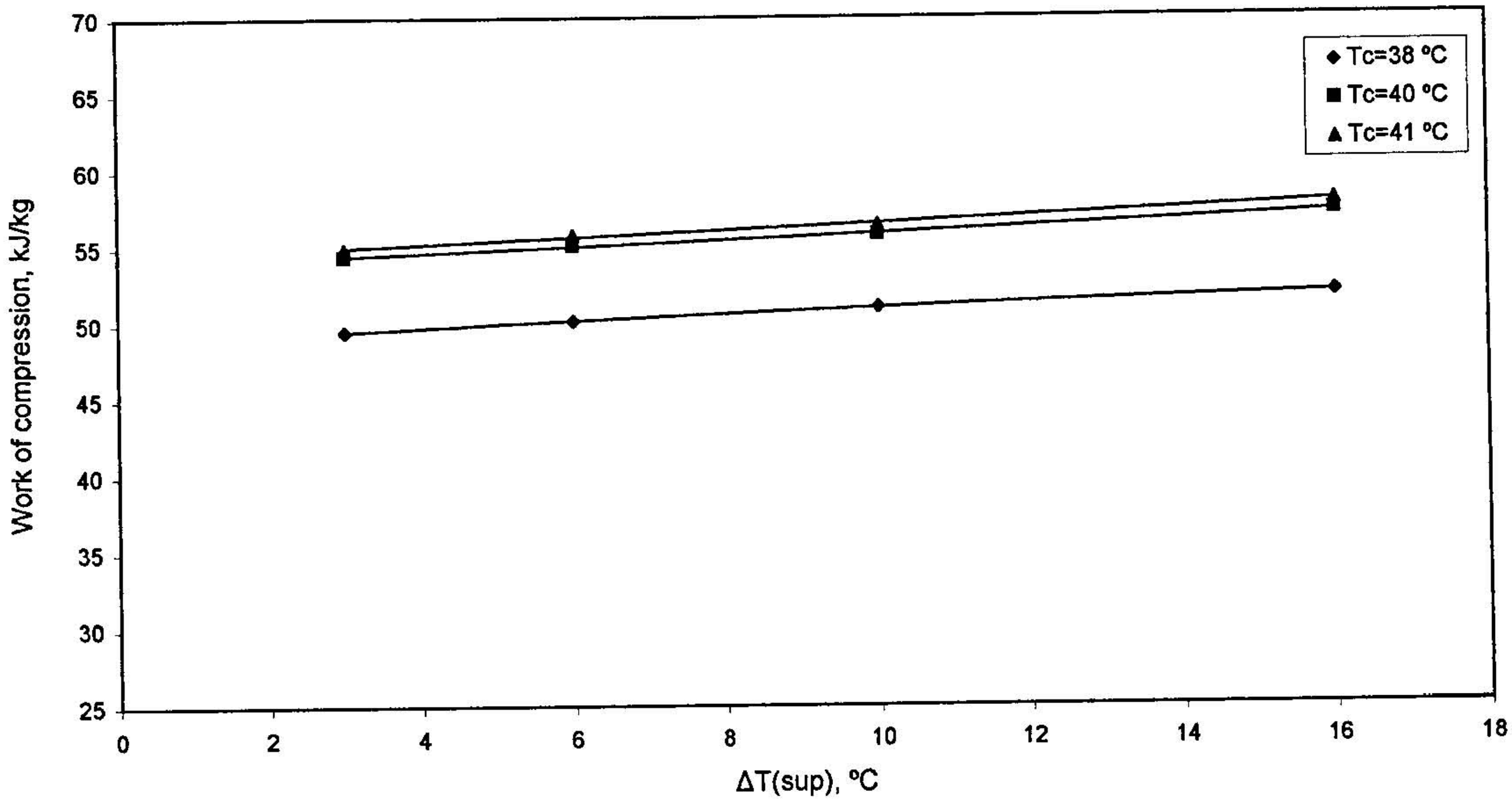


Figure 5.11 : Variation of work of compression with ΔT(sup) at T_e=5 °C , ΔT(sub)=9 °C for R407c

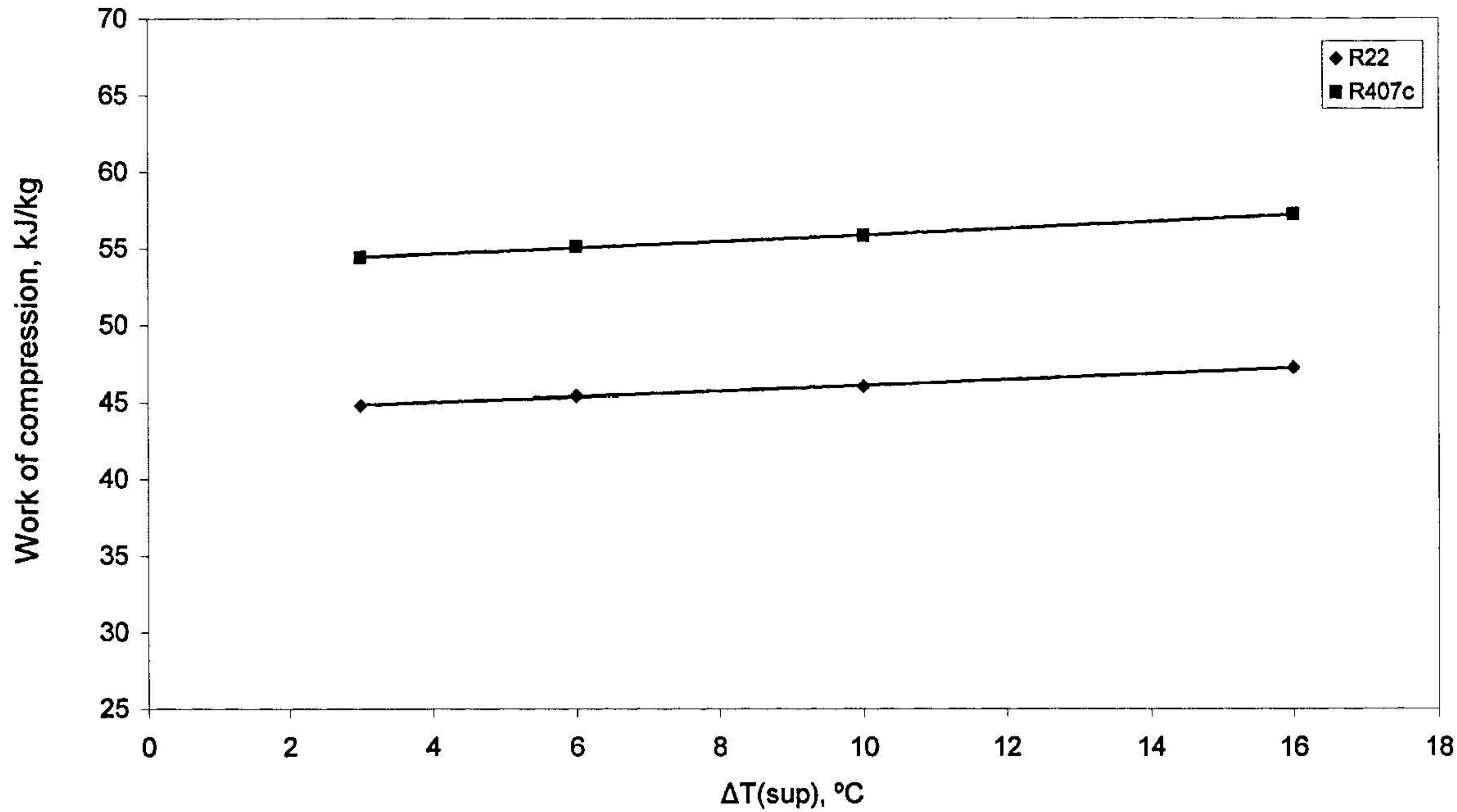


Figure 5.12 : Variation of work of compression with ΔT(sup) at T_e=5 °C ,T_c =40 °C, ΔT(sub)=9°C

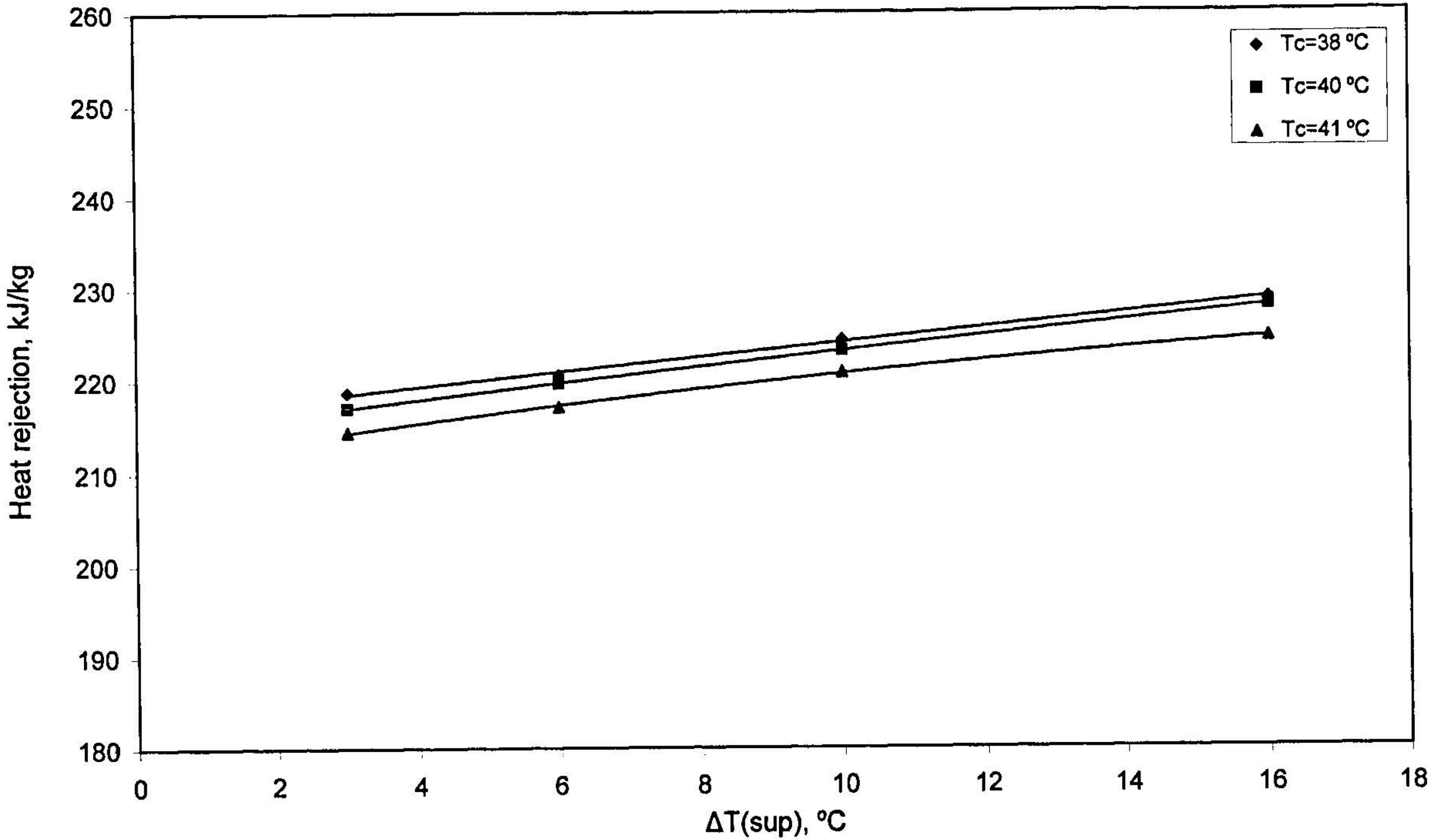


Figure 5.13:Variation of heat rejection with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22

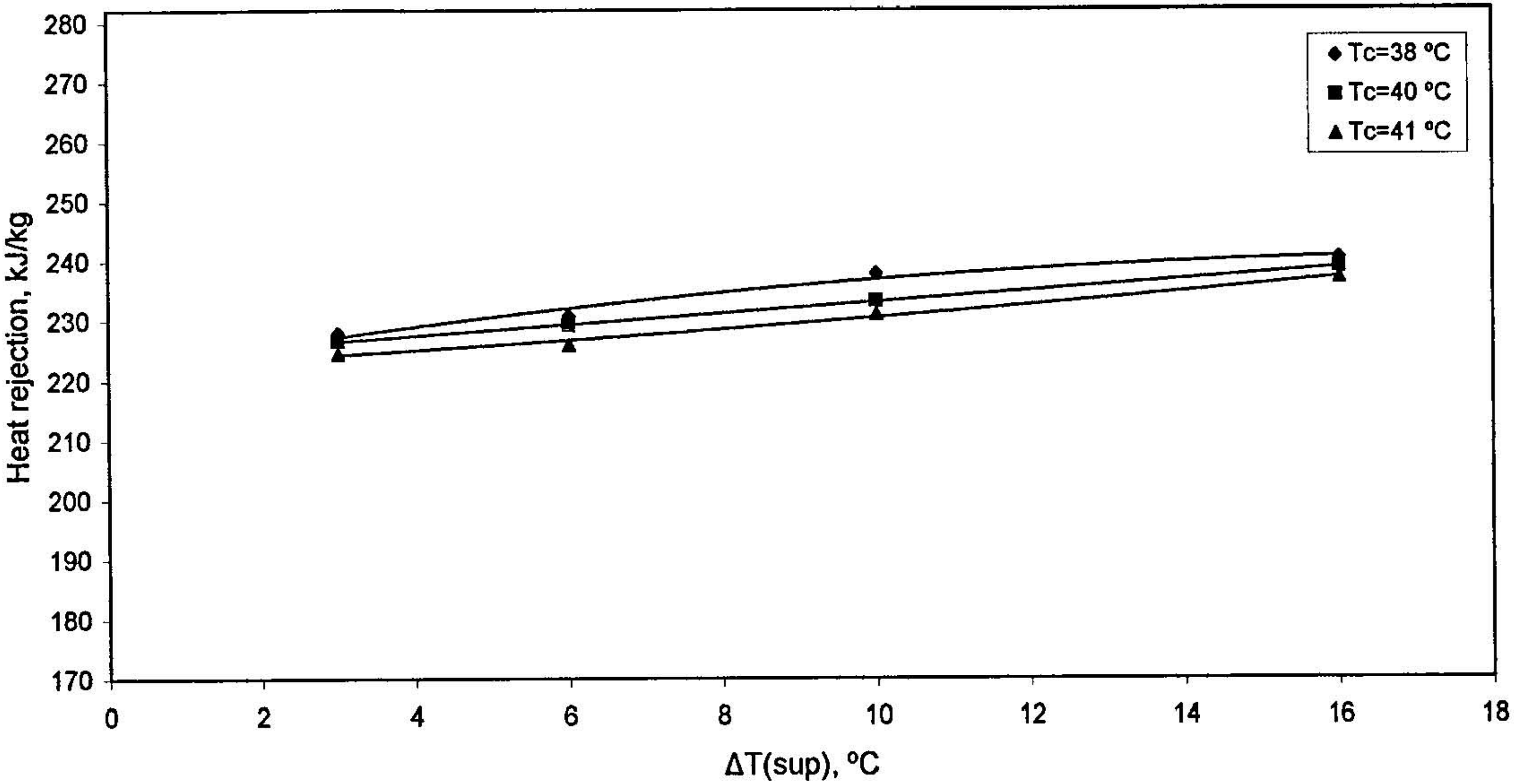


Figure 5.14 : Variation of heat rejection with ΔT (sup) at T_e=5 °C, ΔT(sub)=9 °C for R407c

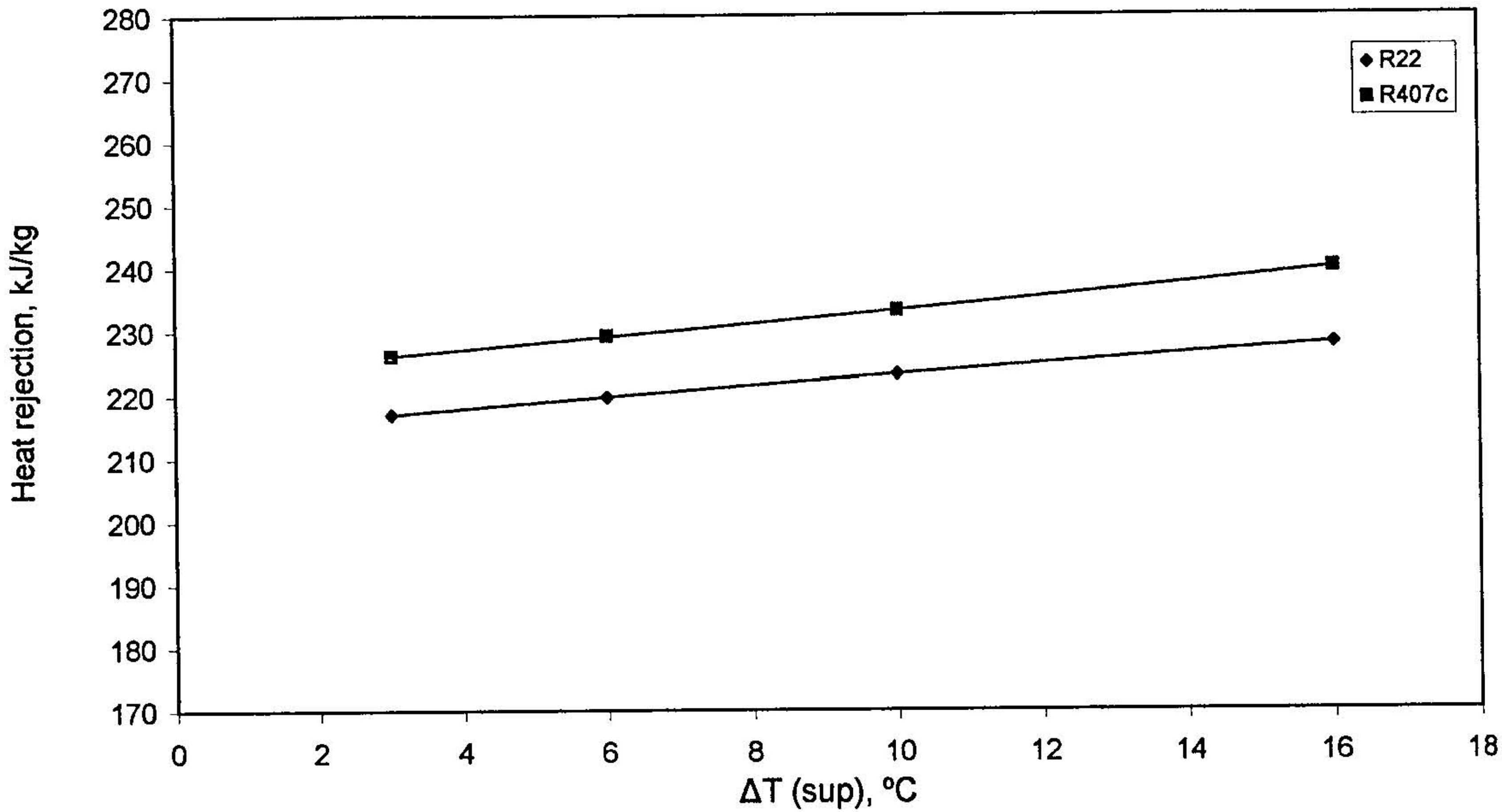


Figure 5.15 : Variation of heat rejection with ΔT (sup) at T_e=5 °C, T_c=40 °C , ΔT(sub)=9 °C

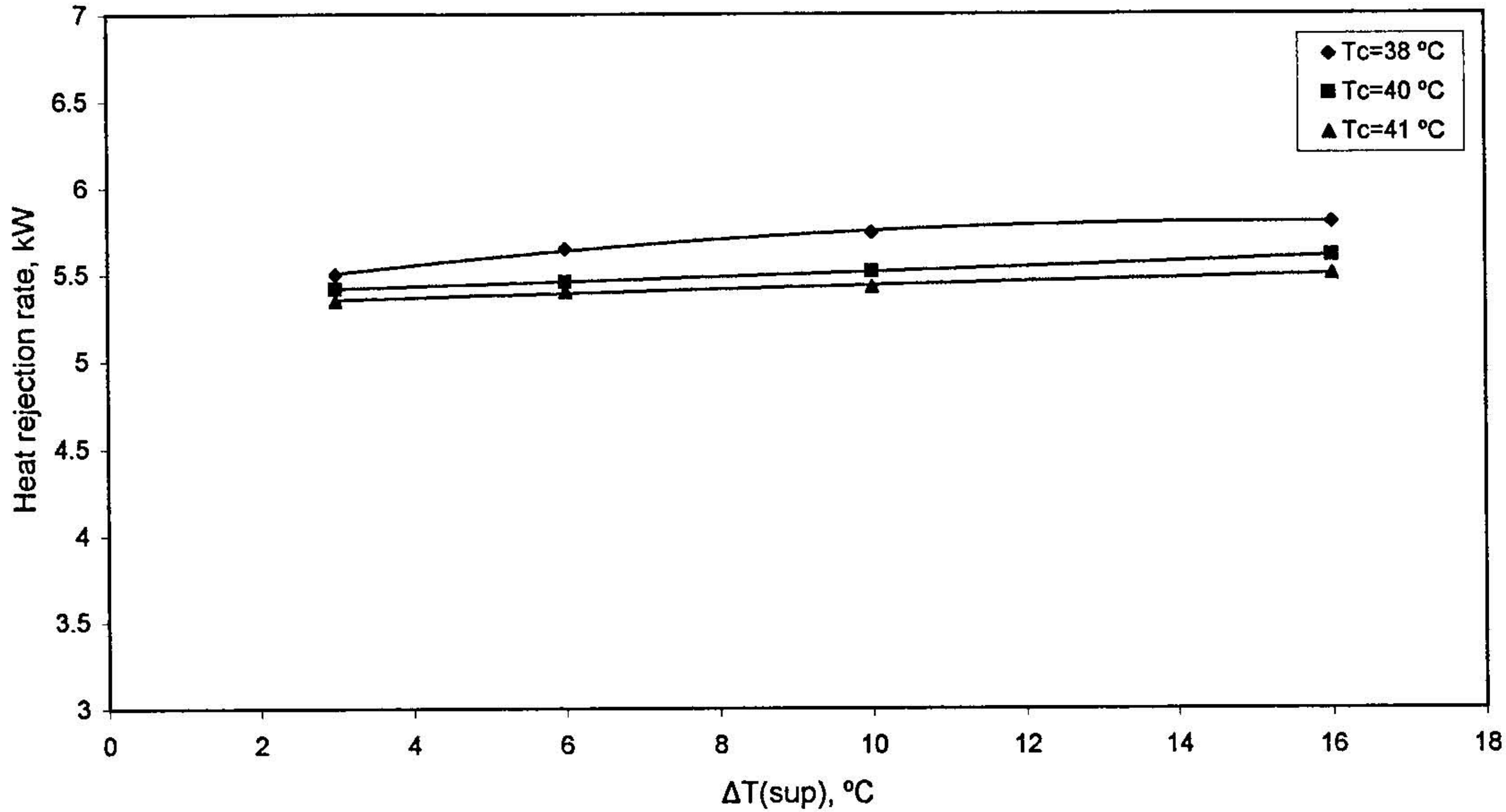


Figure 5.16:Variation of heat rejection rate with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22

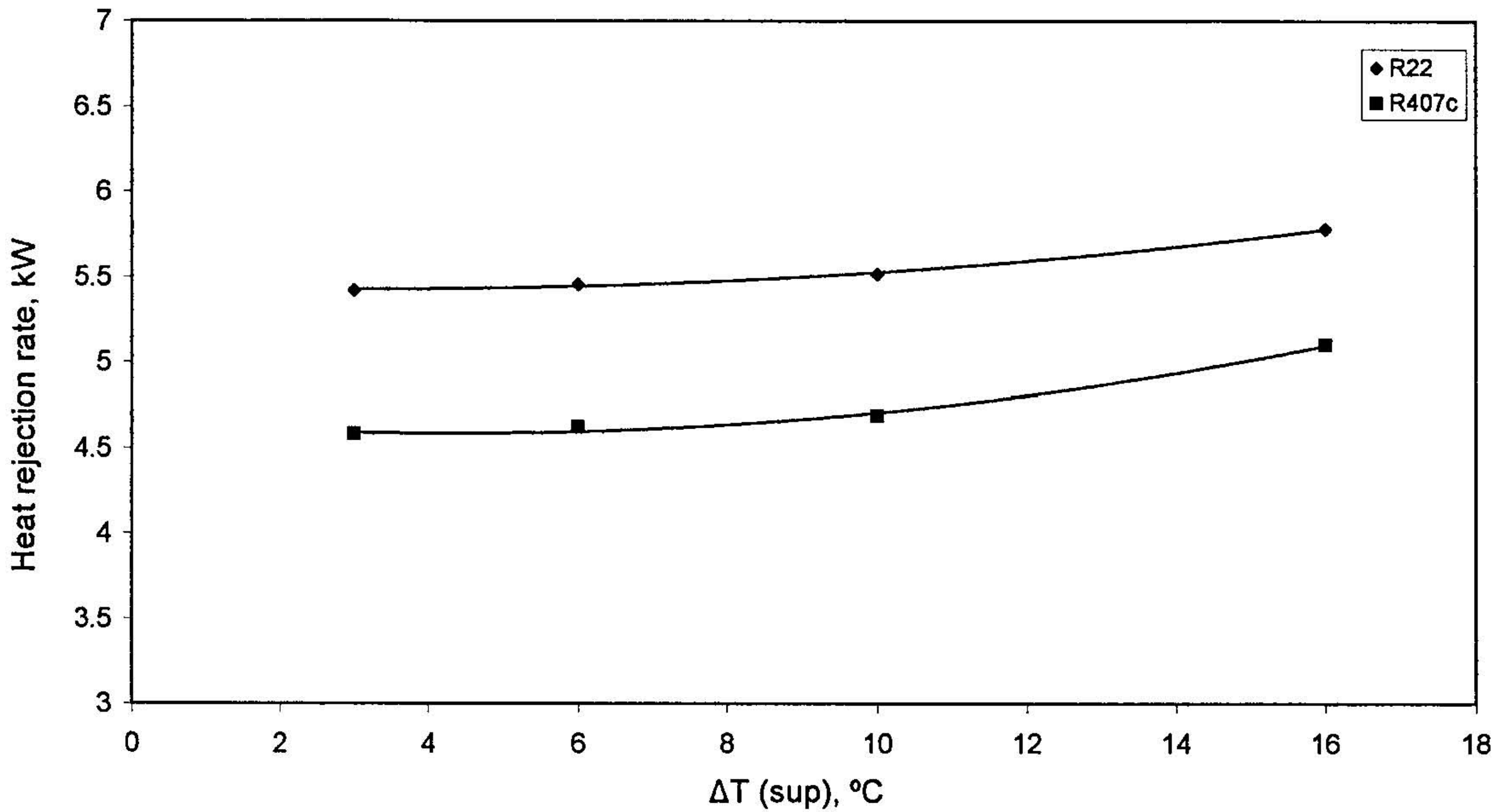


Figure 5.18 : Variation of heat rejection rate with ΔT (sup) at T_e=5 °C,T_c =40 °C , ΔT(sub) =9°C

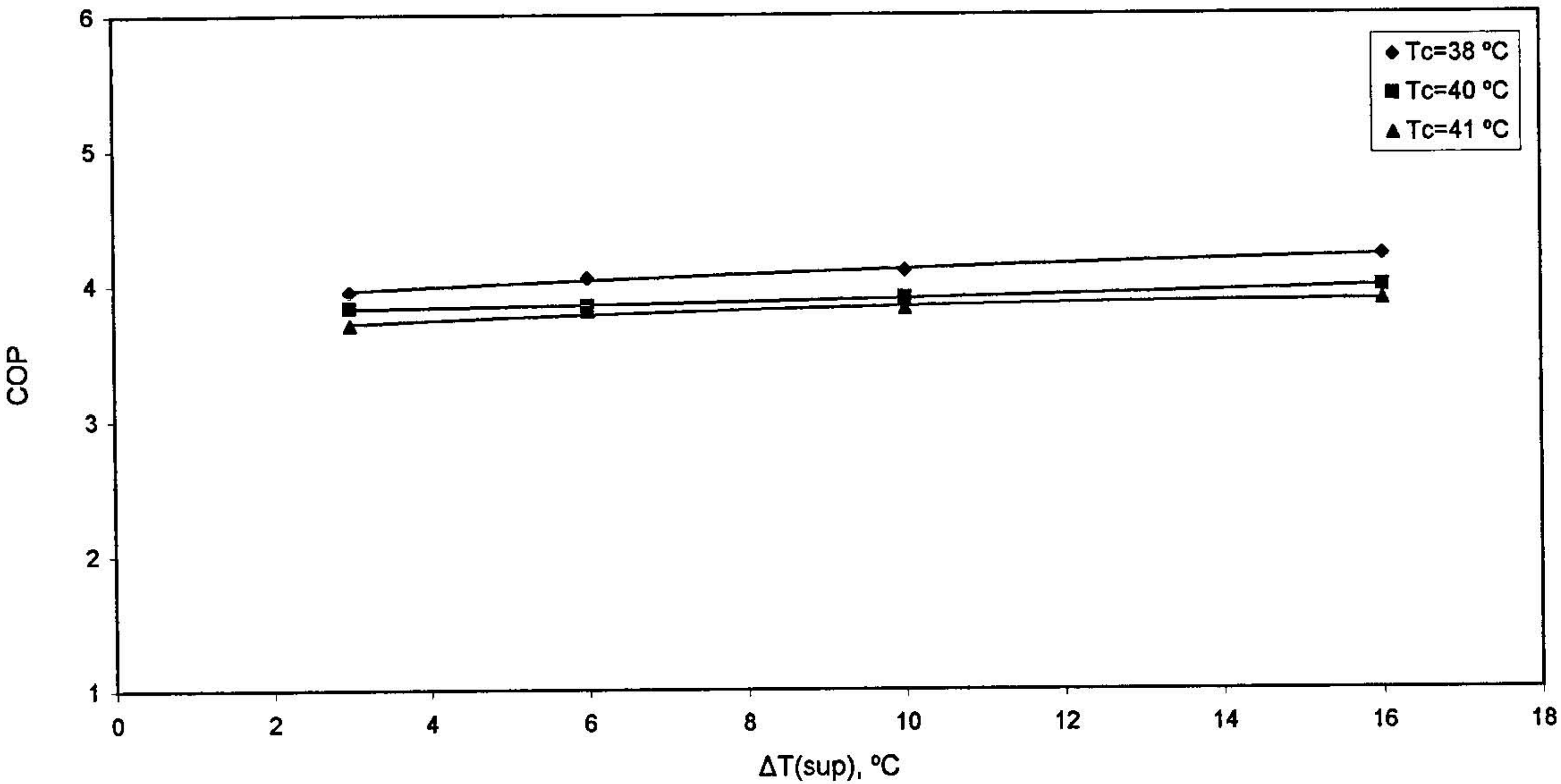


Figure 5.19:Variation of coefficient of performance with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22

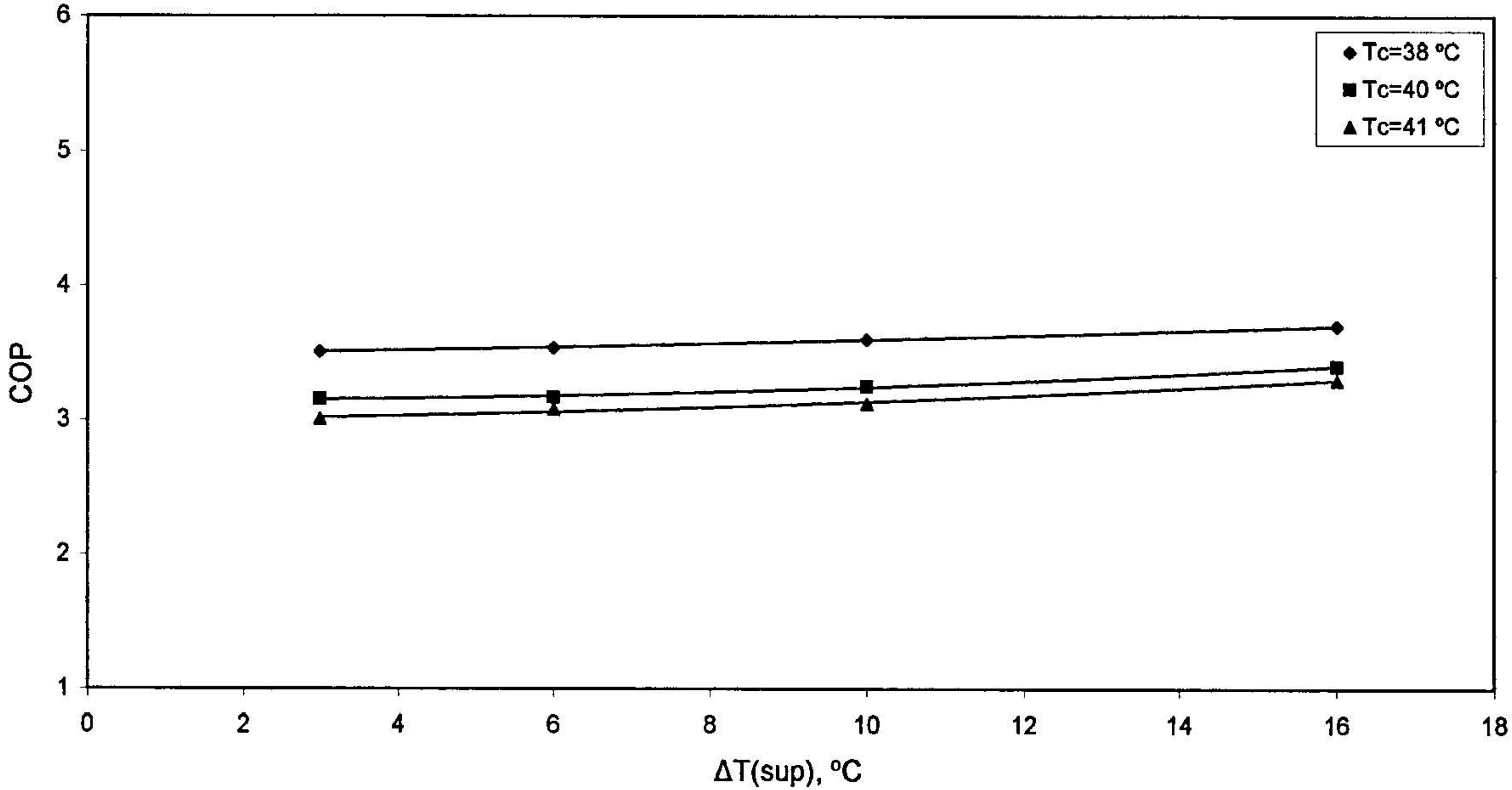


Figure 5.20:Variation of coefficient of performance with ΔT (sup) at T_e=5 °C, ΔT(sub) =9 °C for R407c

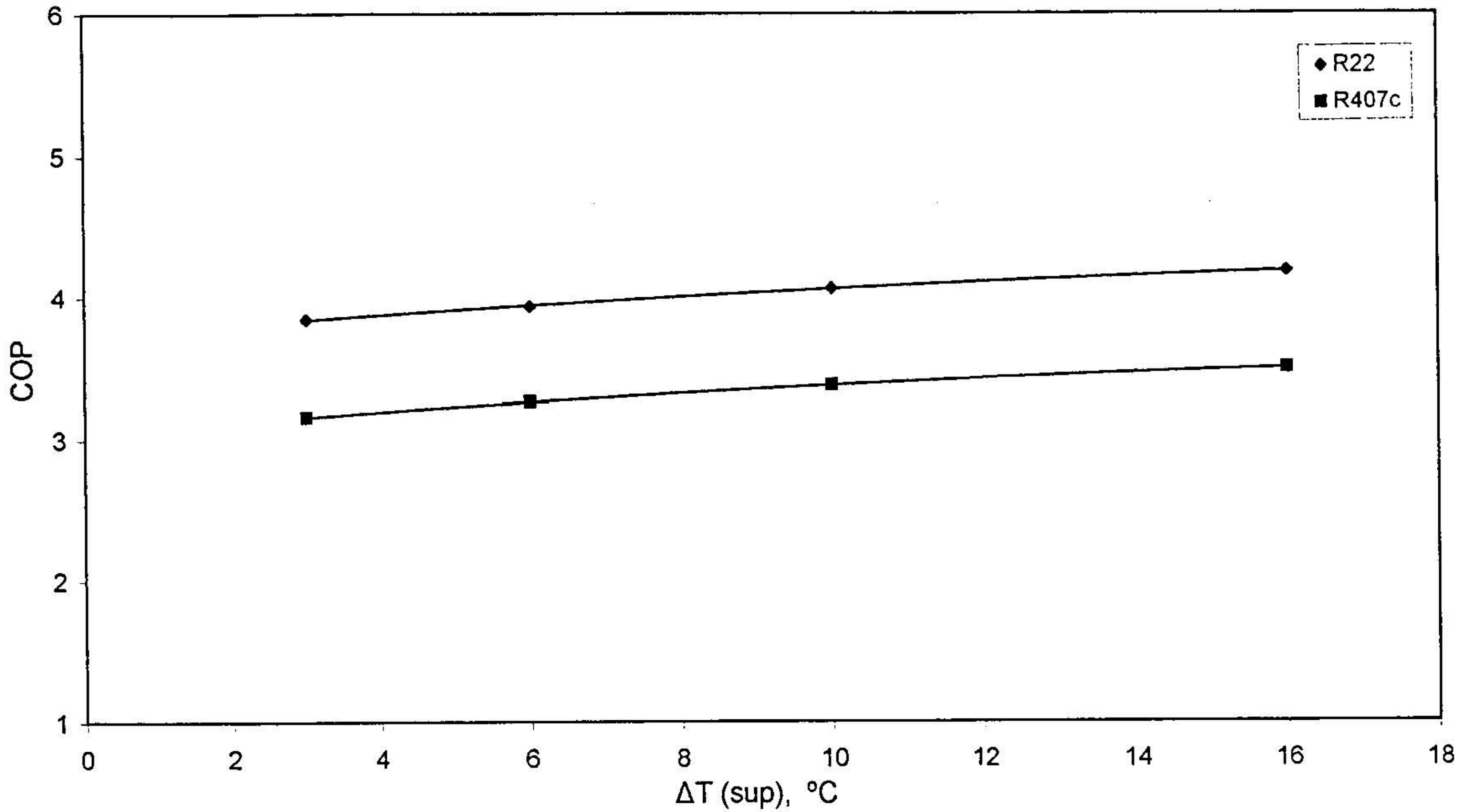


Figure 5.21:Variation of coefficient of performance with ΔT (sup) at T_e=5 °C, T_c=40 °C, ΔT(sub) =9 °C

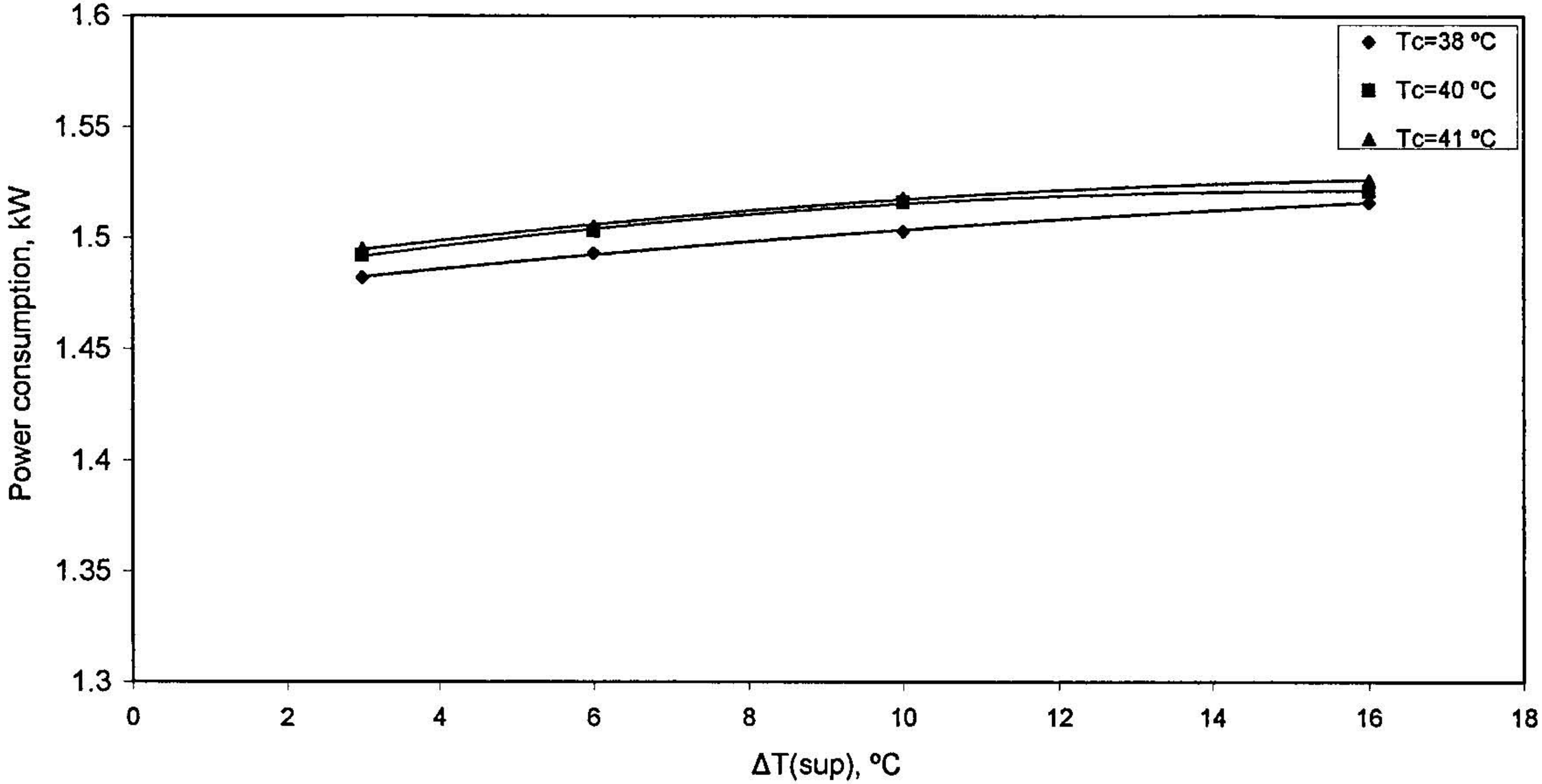


Figure 5.22:Variation of power consumption with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22

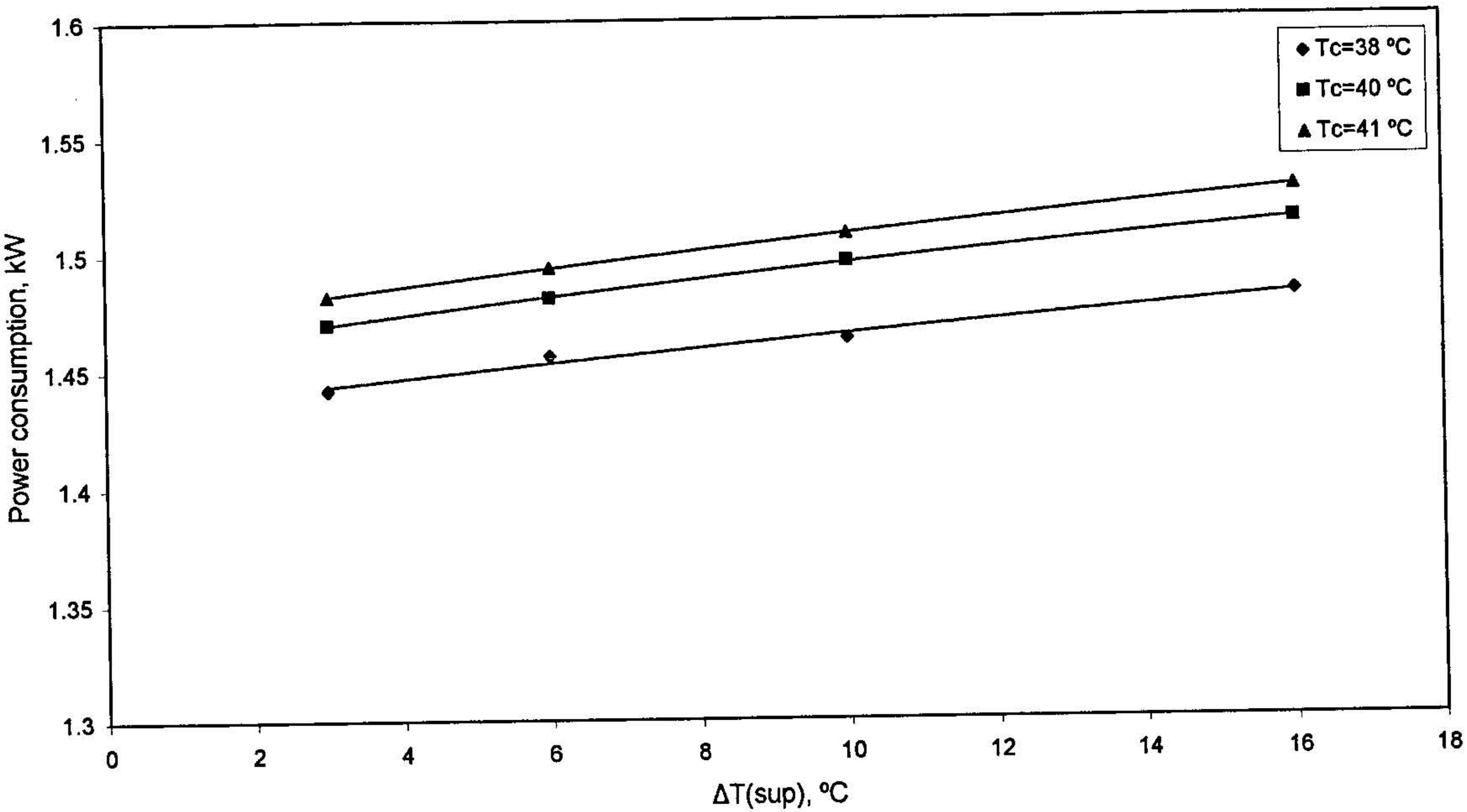


Figure 5.23 :Variation of power consumption with ΔT (sup) at T_e=5 °C, ΔT(sub) = 9 °C for R407c

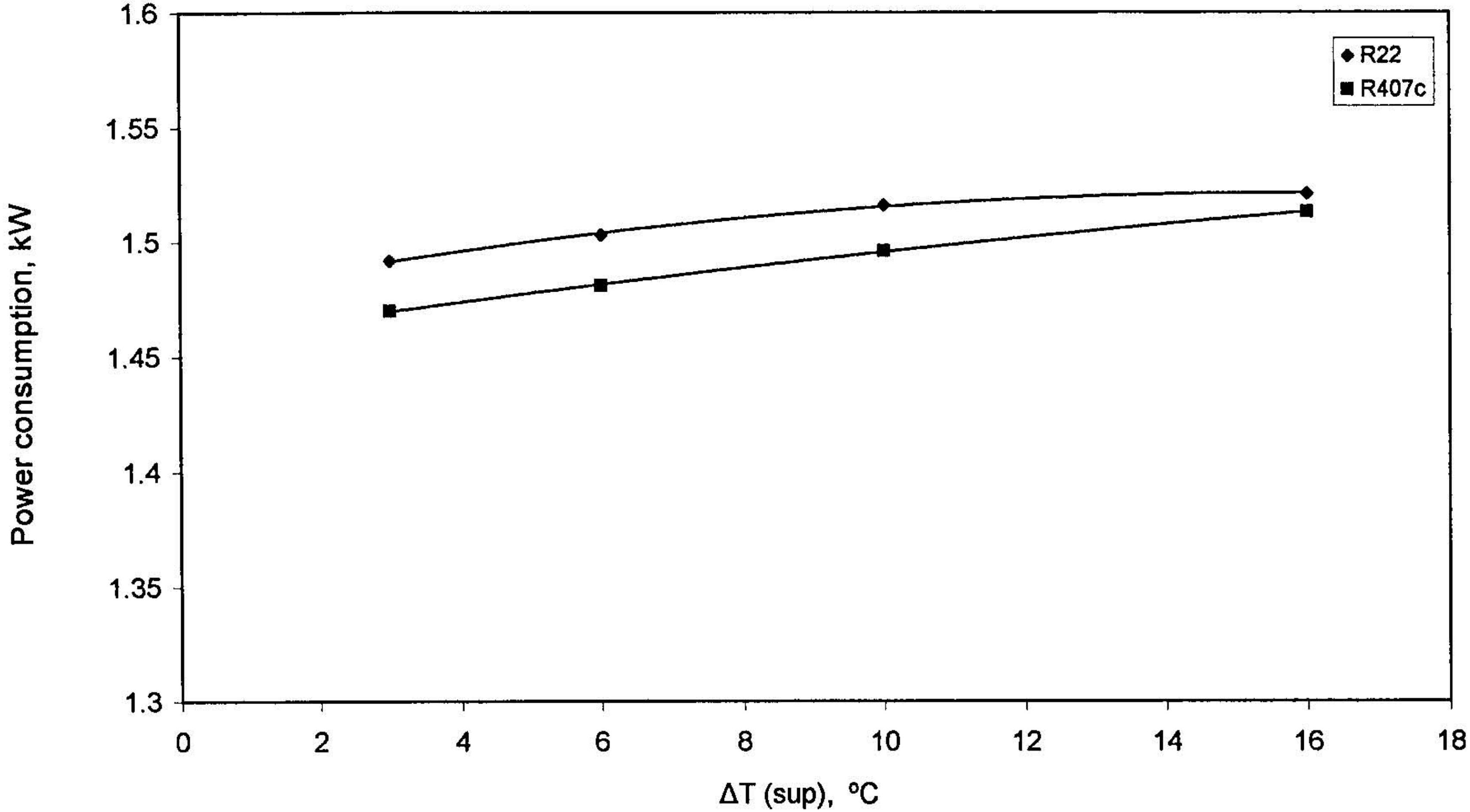


Figure 5.24 :Variation of power consumption with ΔT (sup) at T_e=5 °C, T_c=40 °C, ΔT(sub) = 9 °C

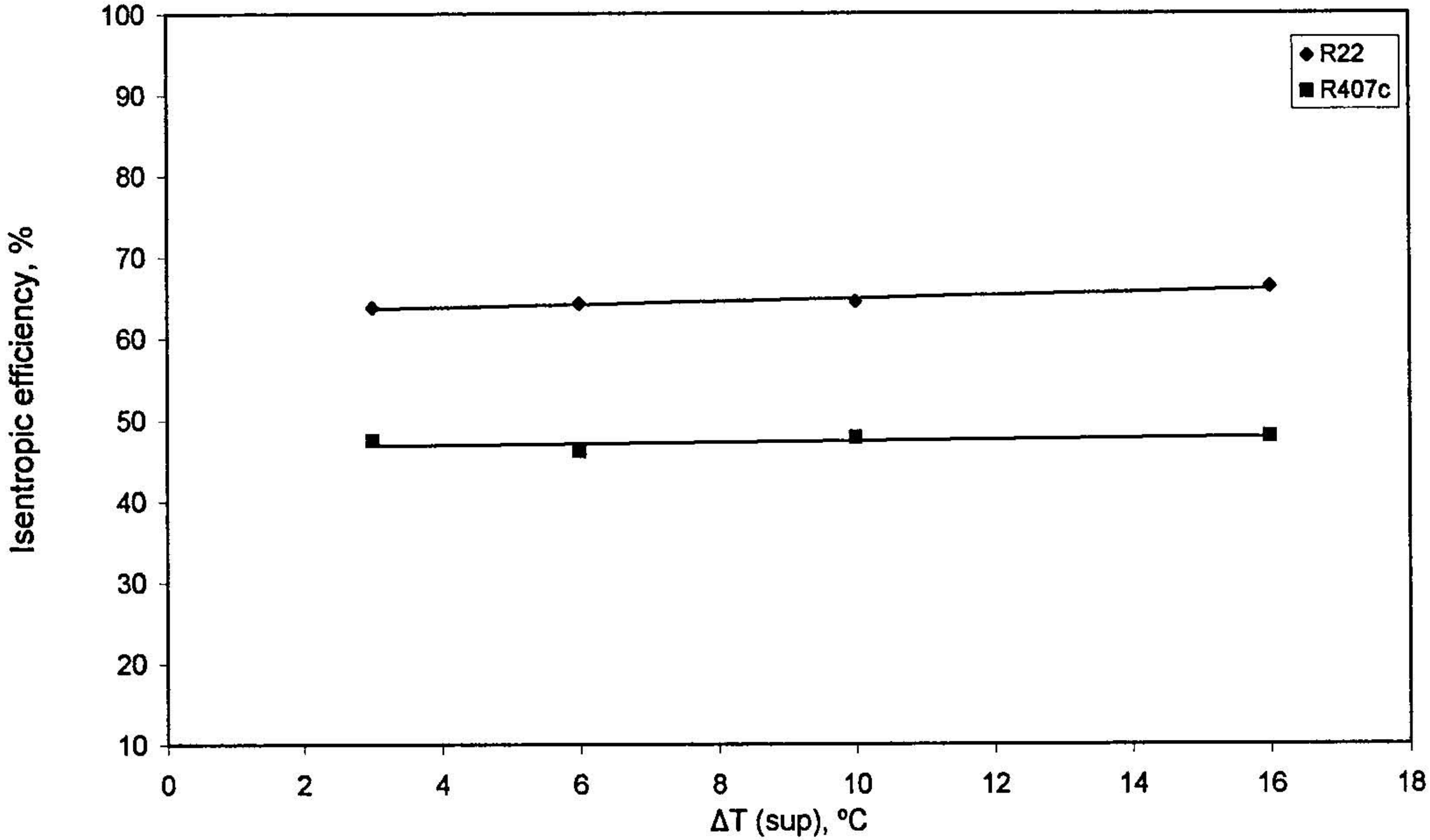


Figure 5.25:Variation of isentropic efficiency with ΔT (sup) at T_e=5 °C,T_c=40 °C, ΔT(sub)=9 °C

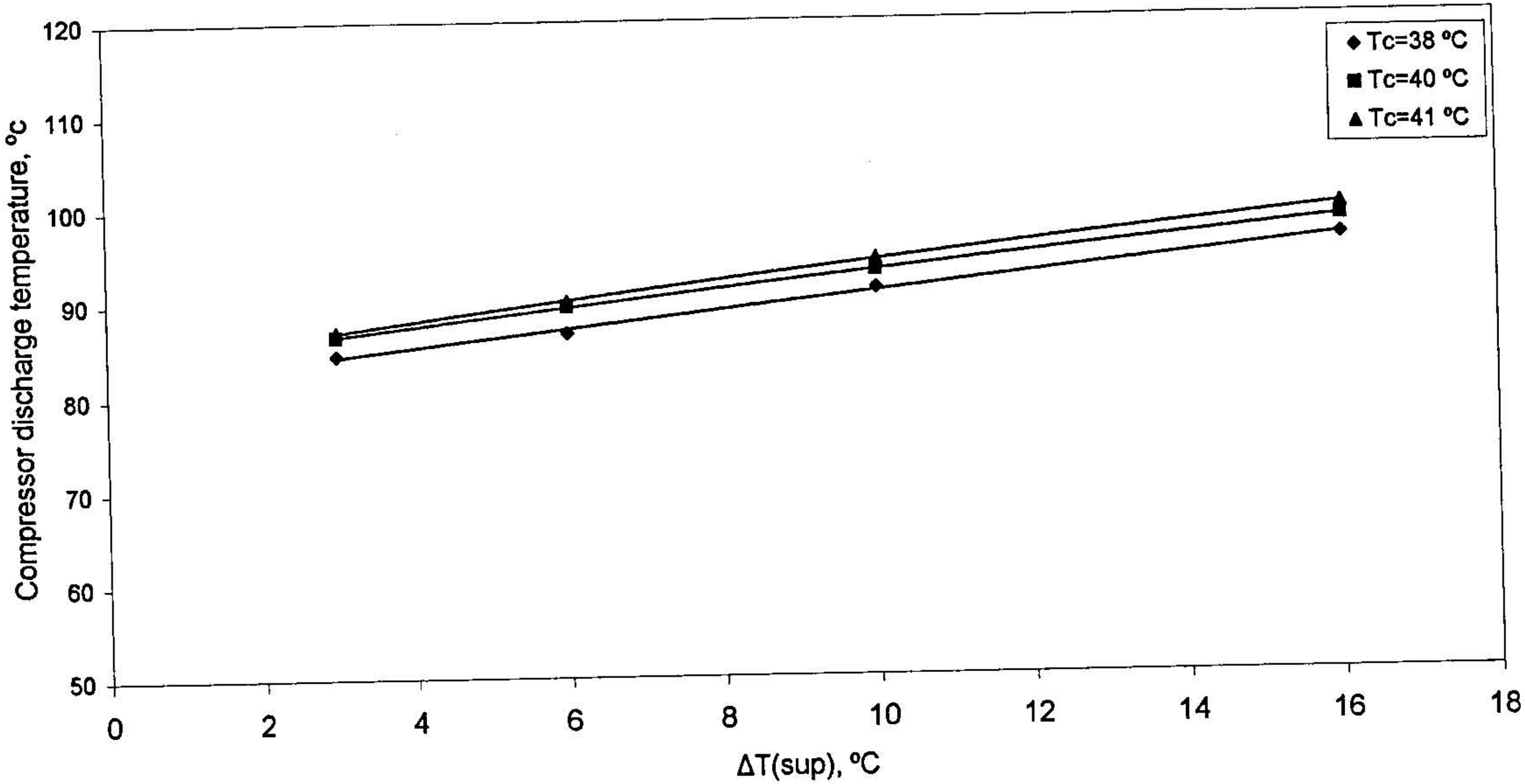
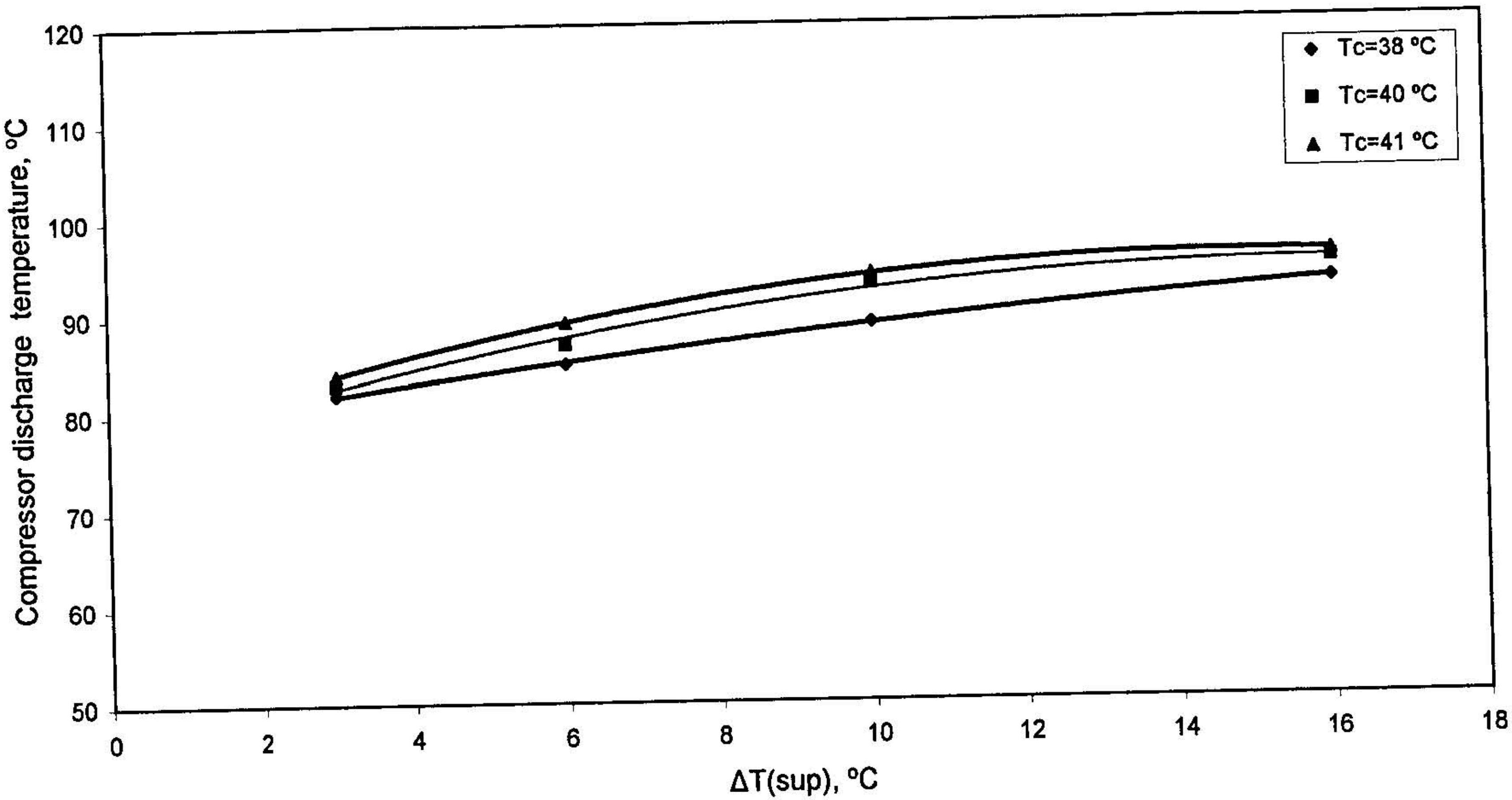
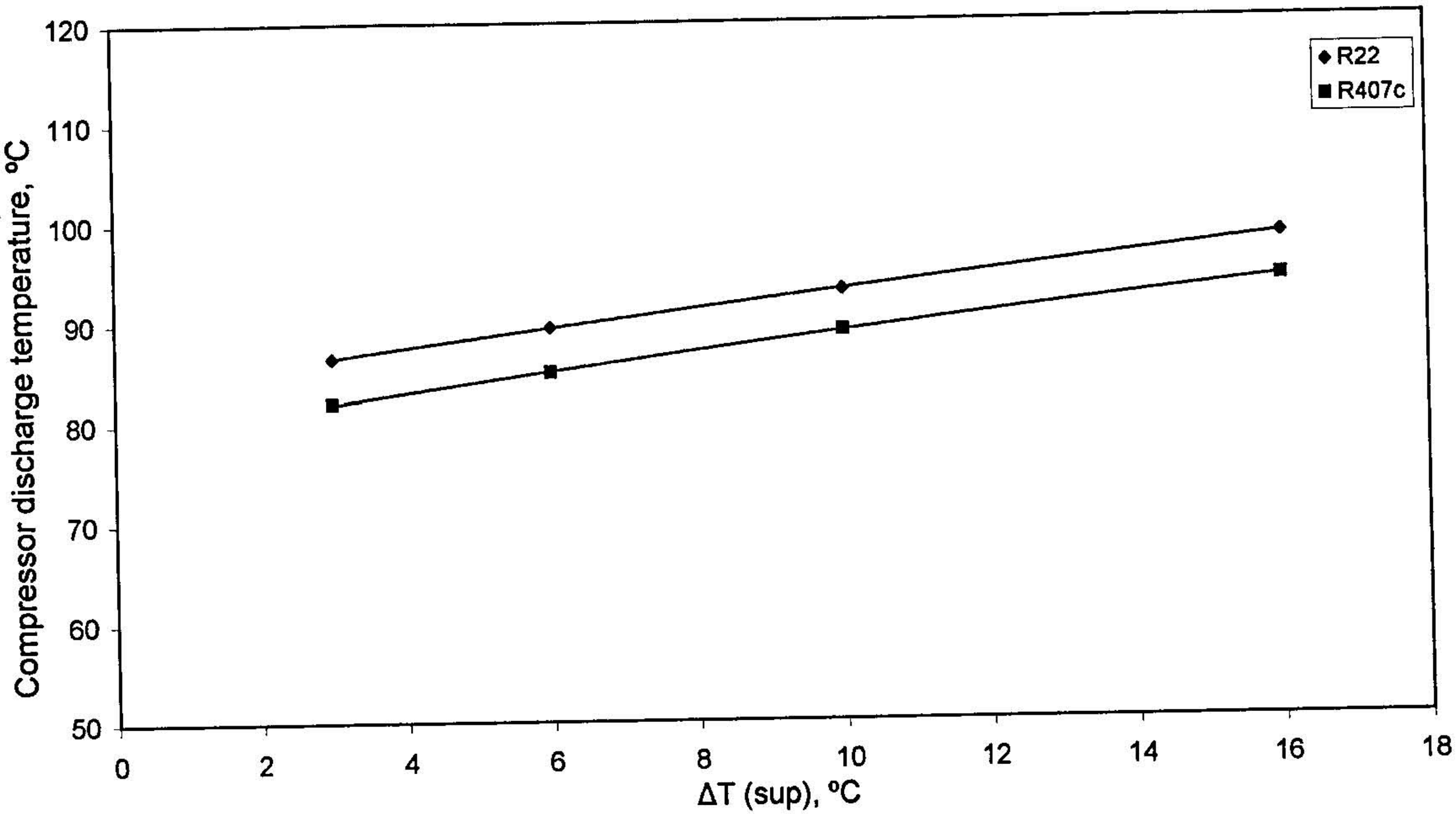


Figure 5.29:Variation of compressor discharge temperature with ΔT(sup) at T_e=5 °C, ΔT(sub)= 9 °C for R22



Fugure 5.30:Variation of compressor discharge temperature with ΔT (sup) at T_e=5 °C, ΔT(sub) =9°C for R407c



Fugure 5.31:Variation of compressor discharge temperature with ΔT (sup) at T_e=5 °C,T_c =40 °C , ΔT(sub) =9°C

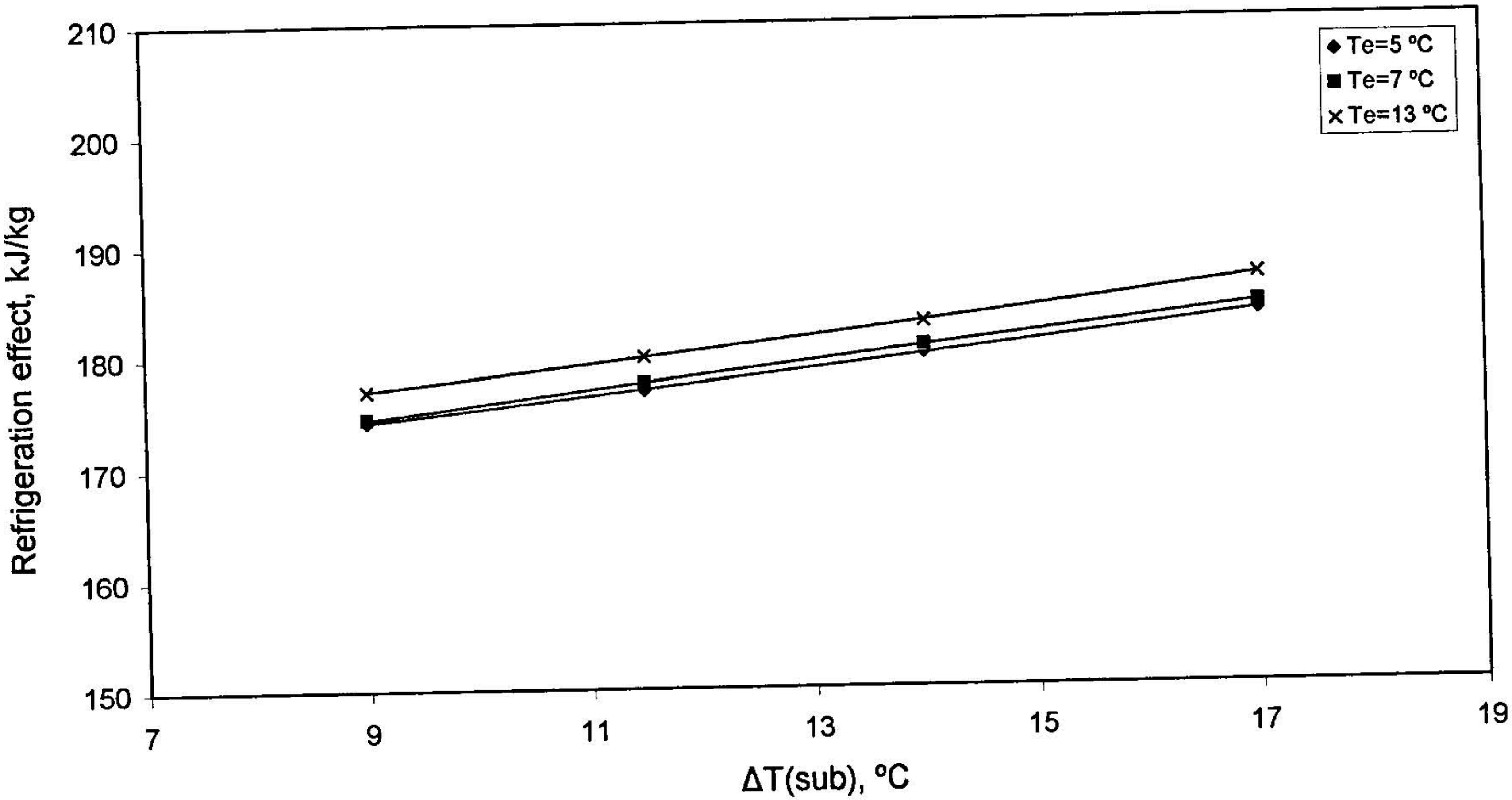


Figure 5.32: Variation of refrigeration effect with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R22

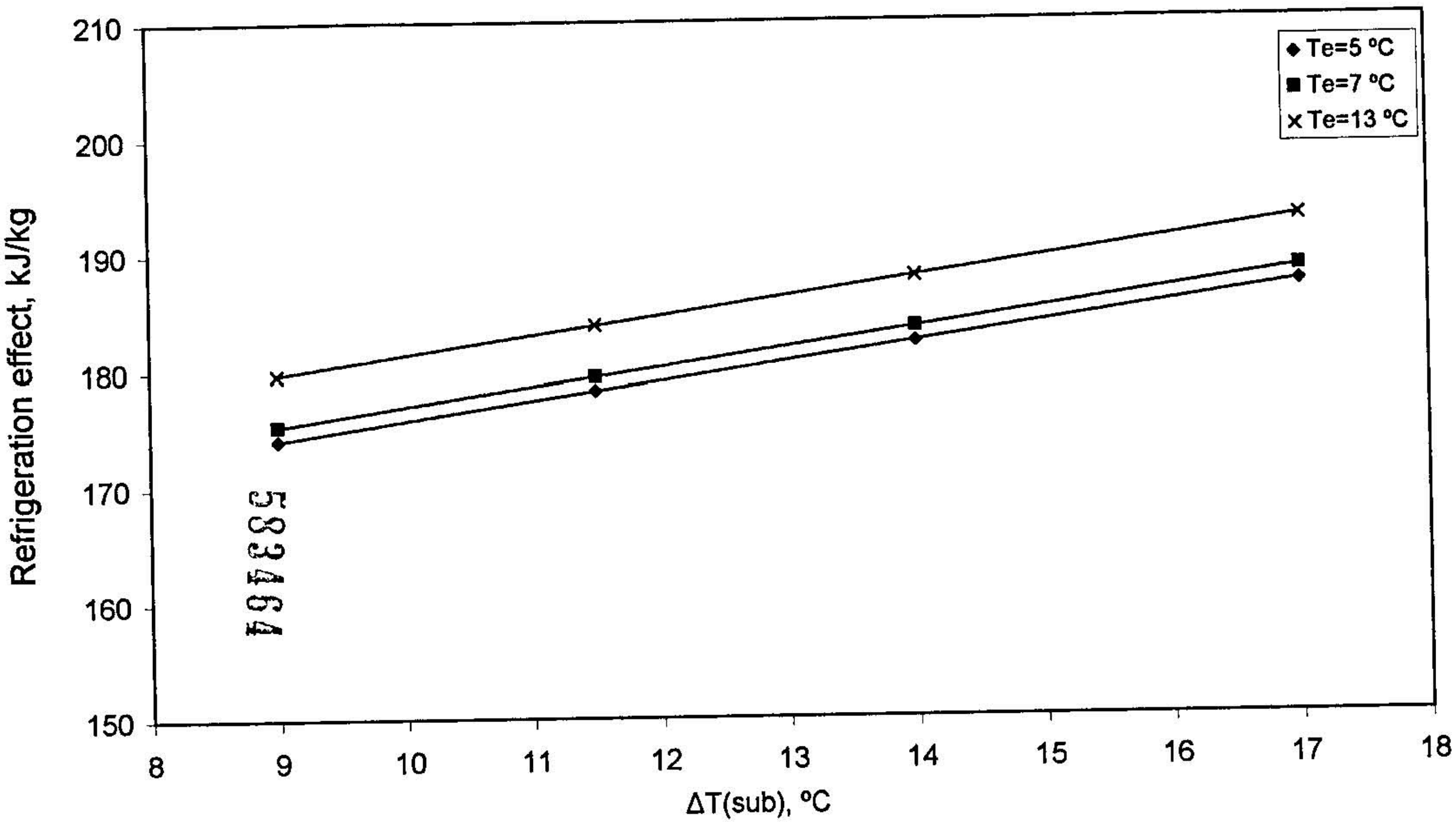


Figure 5.33: Variation of refrigeration effect with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R407c

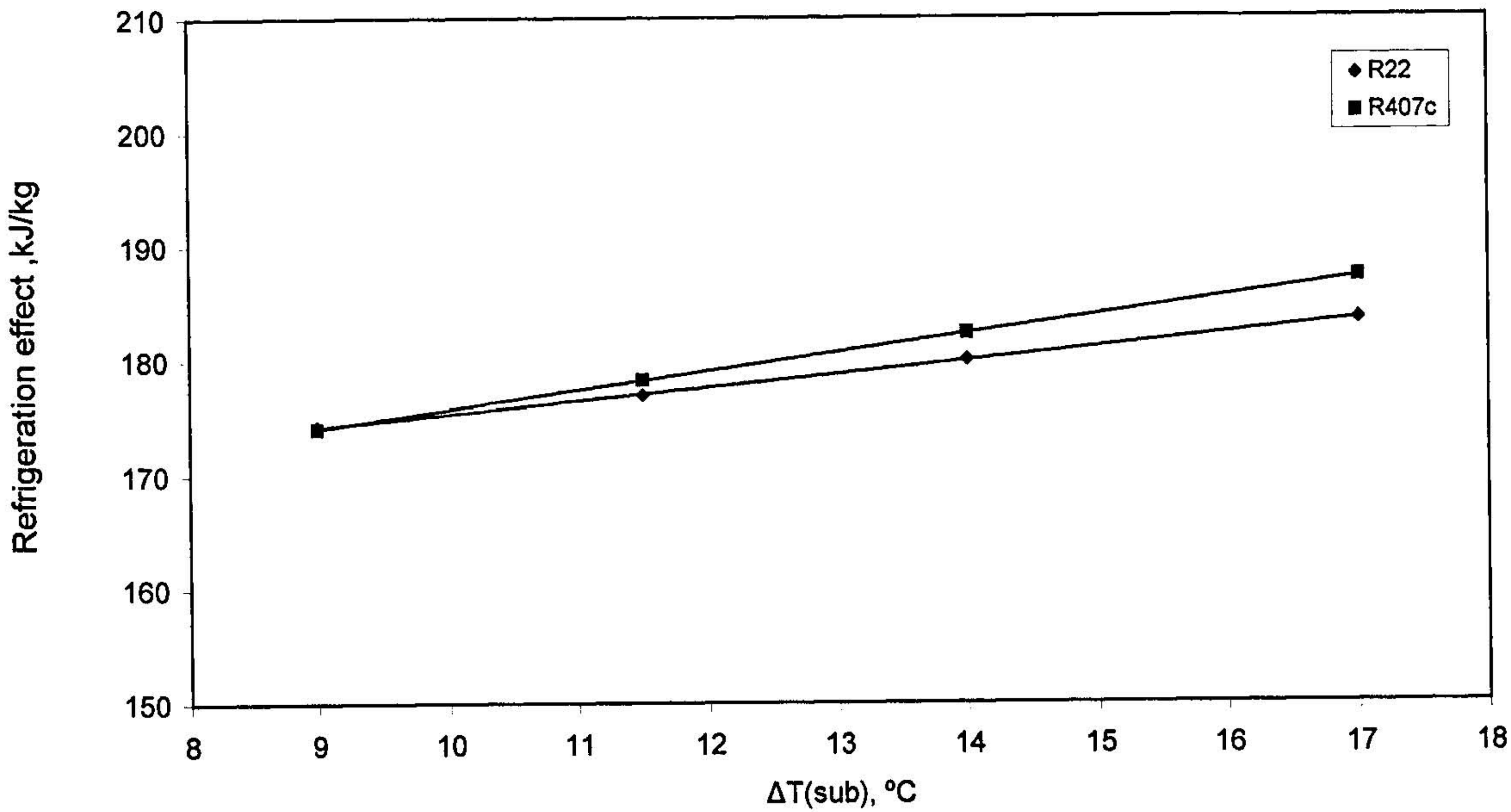


Figure 5.34:Variation of refrigeration effect with ΔT(sub) at T_e=5 °C, T_c=40 °C, ΔT(sup)=6 °C

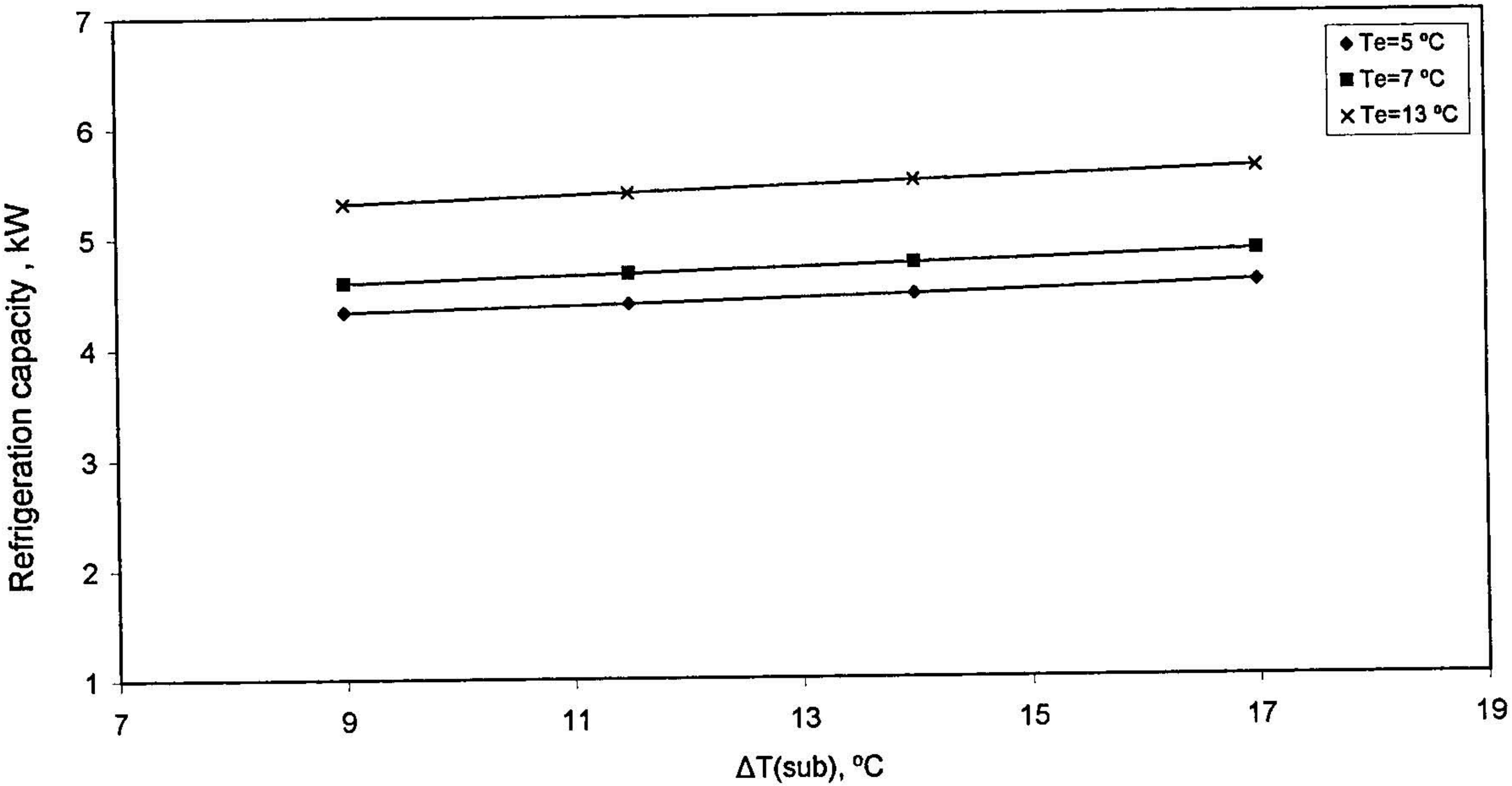


Figure 5.35: Variation of refrigeration capacity with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R22

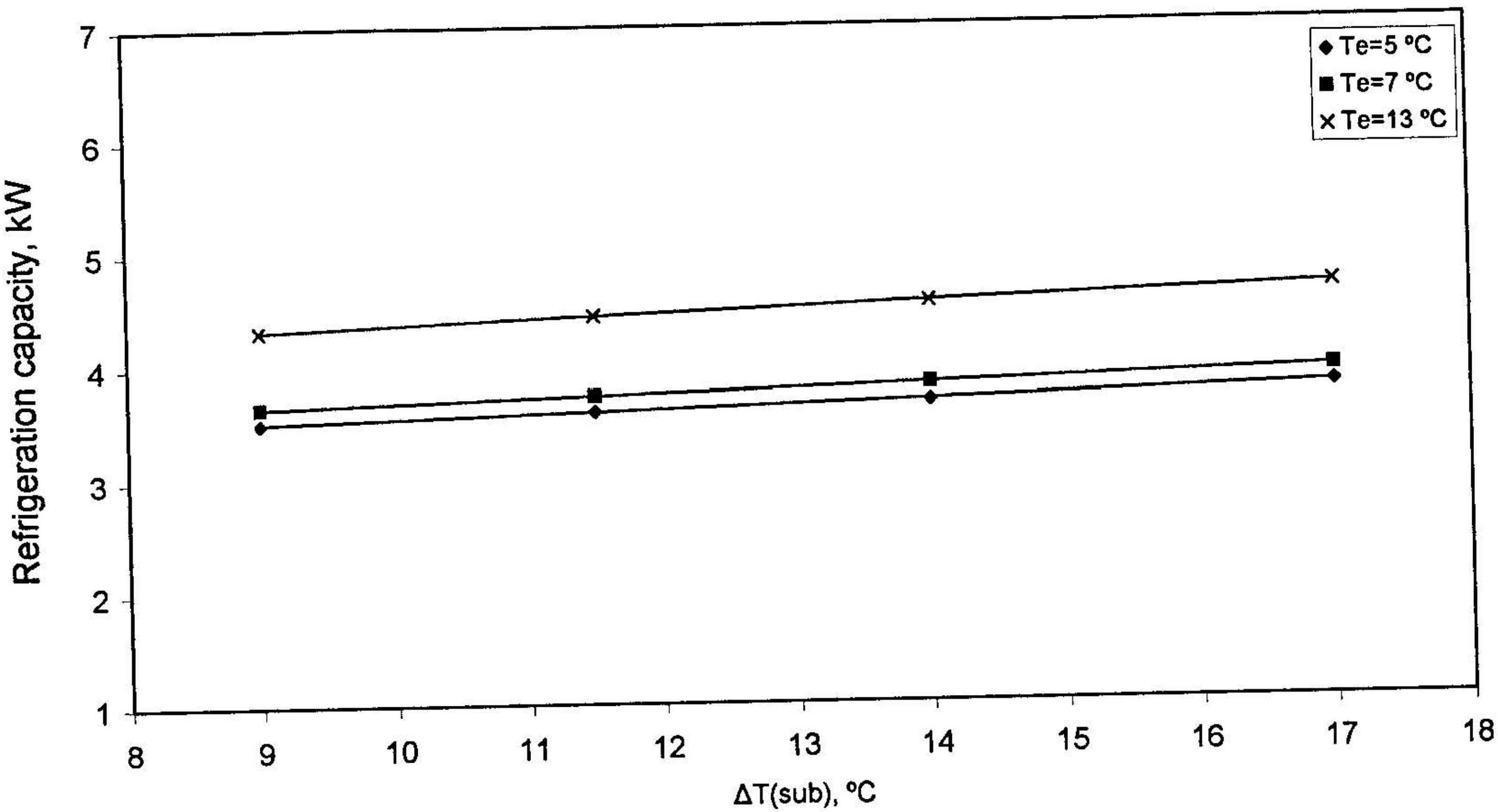


Figure 5.36: Variation of refrigeration capacity with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R407c

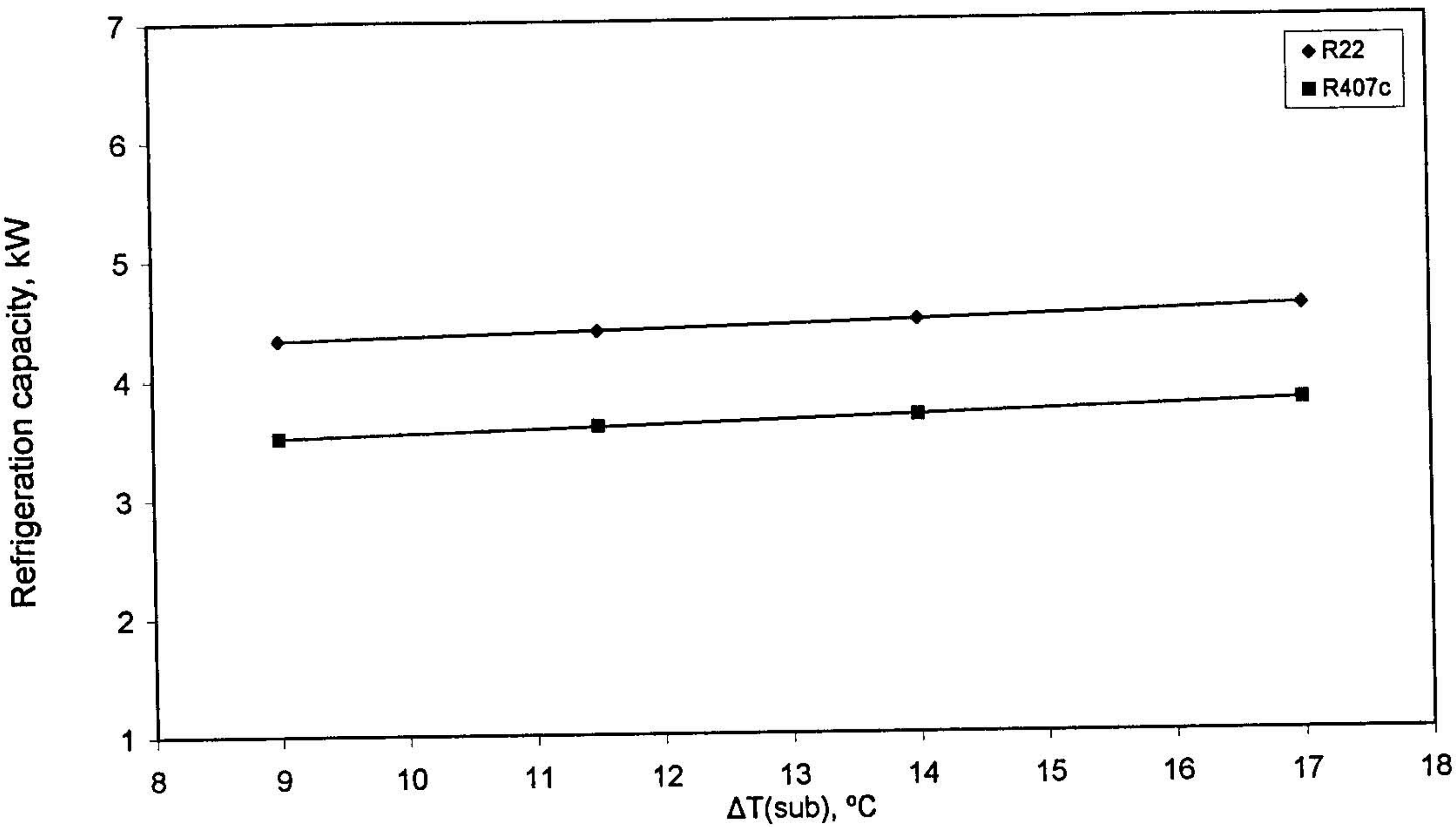


Figure 5.37 : Variation of refrigeration capacity with ΔT(sub) at T_e=5 °C, T_c=40 °C, ΔT(sup) =6 °C

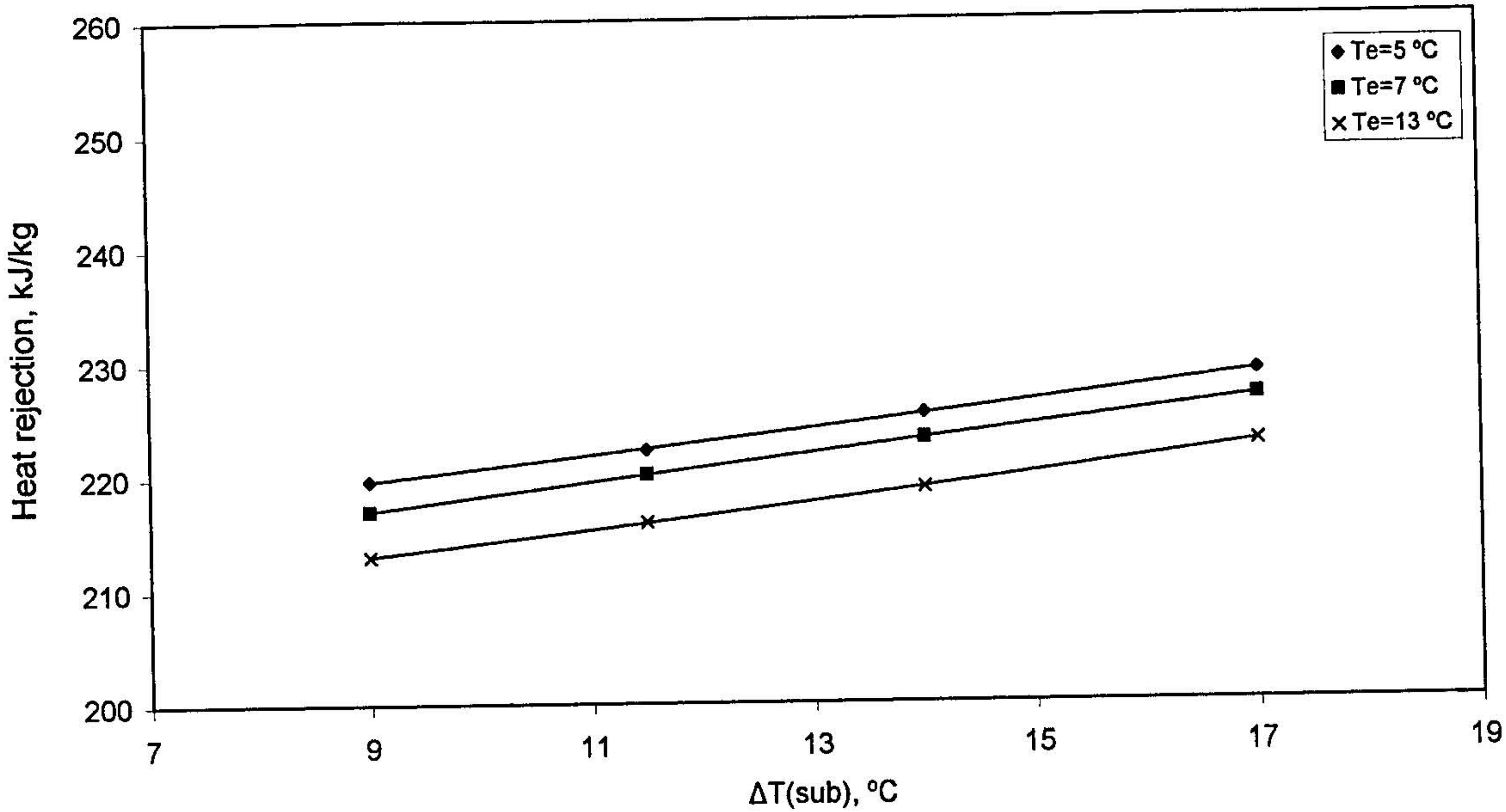


Figure 5.38: Variation of heat rejection with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R22

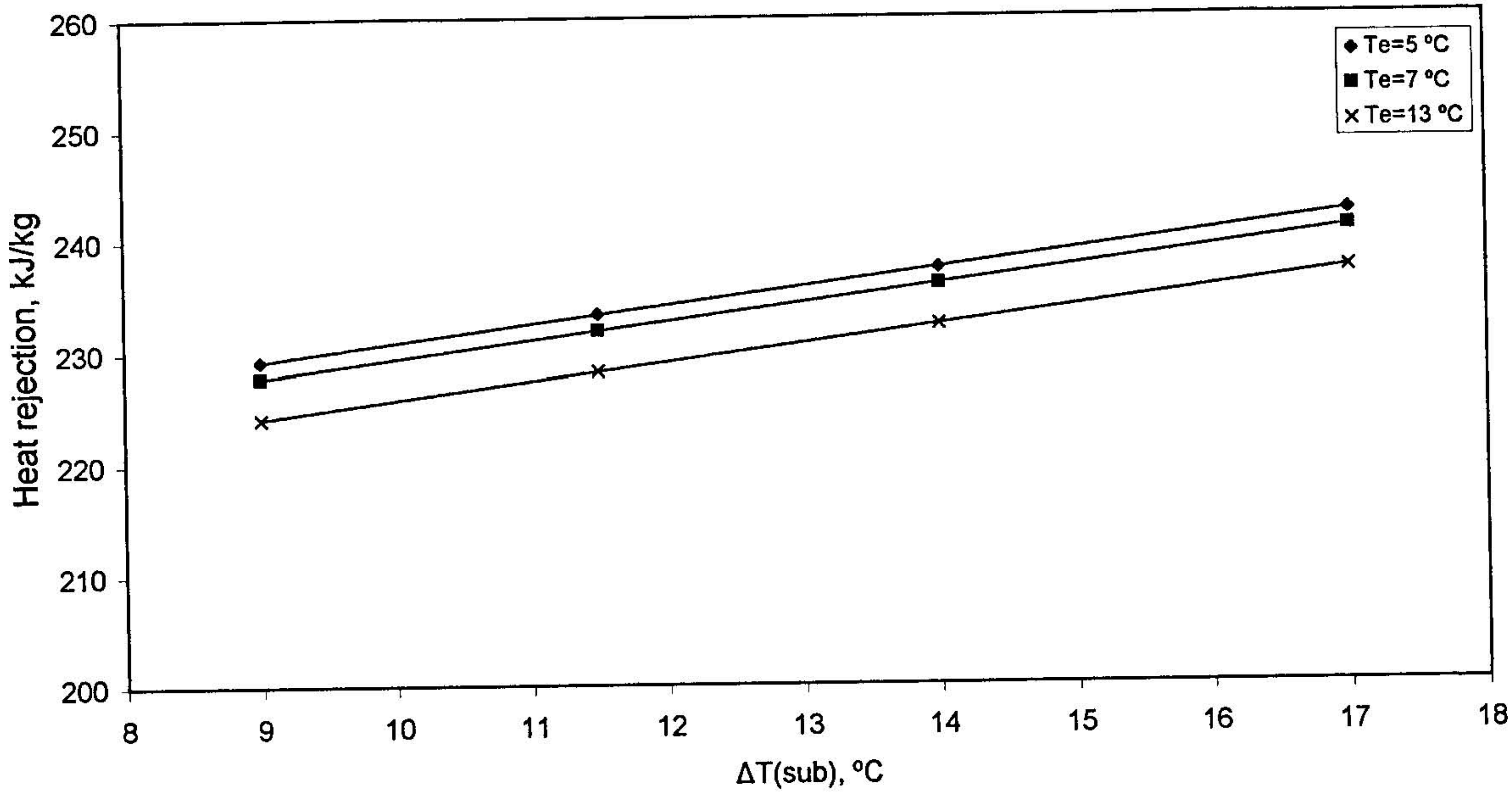


Figure 5.39: Variation of heat rejection with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R407c

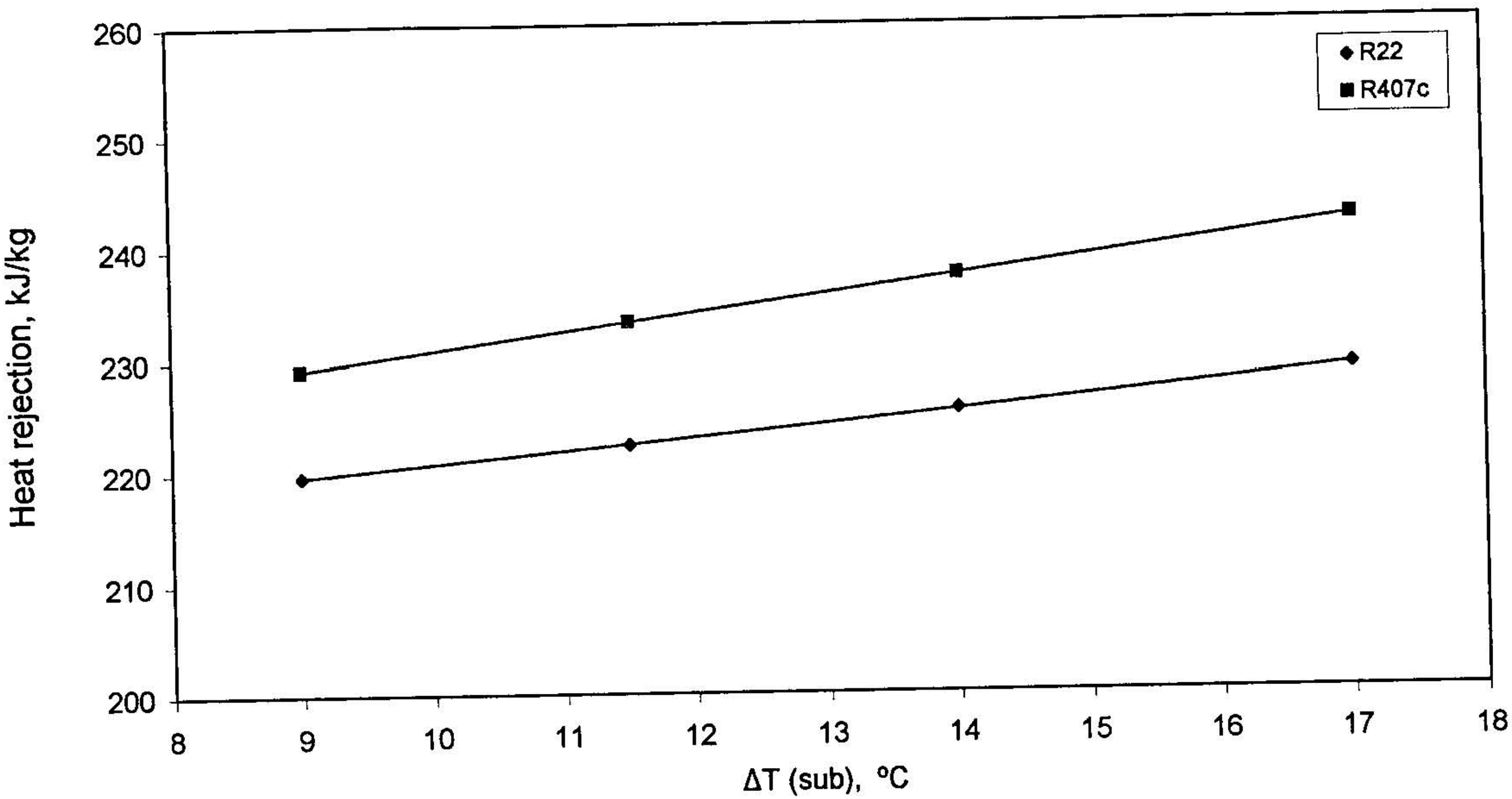


Figure 5.40 : Variation of heat rejection with ΔT (sub) at T_e=5 °C,T_c =40 °C , ΔT(sup) =6°C

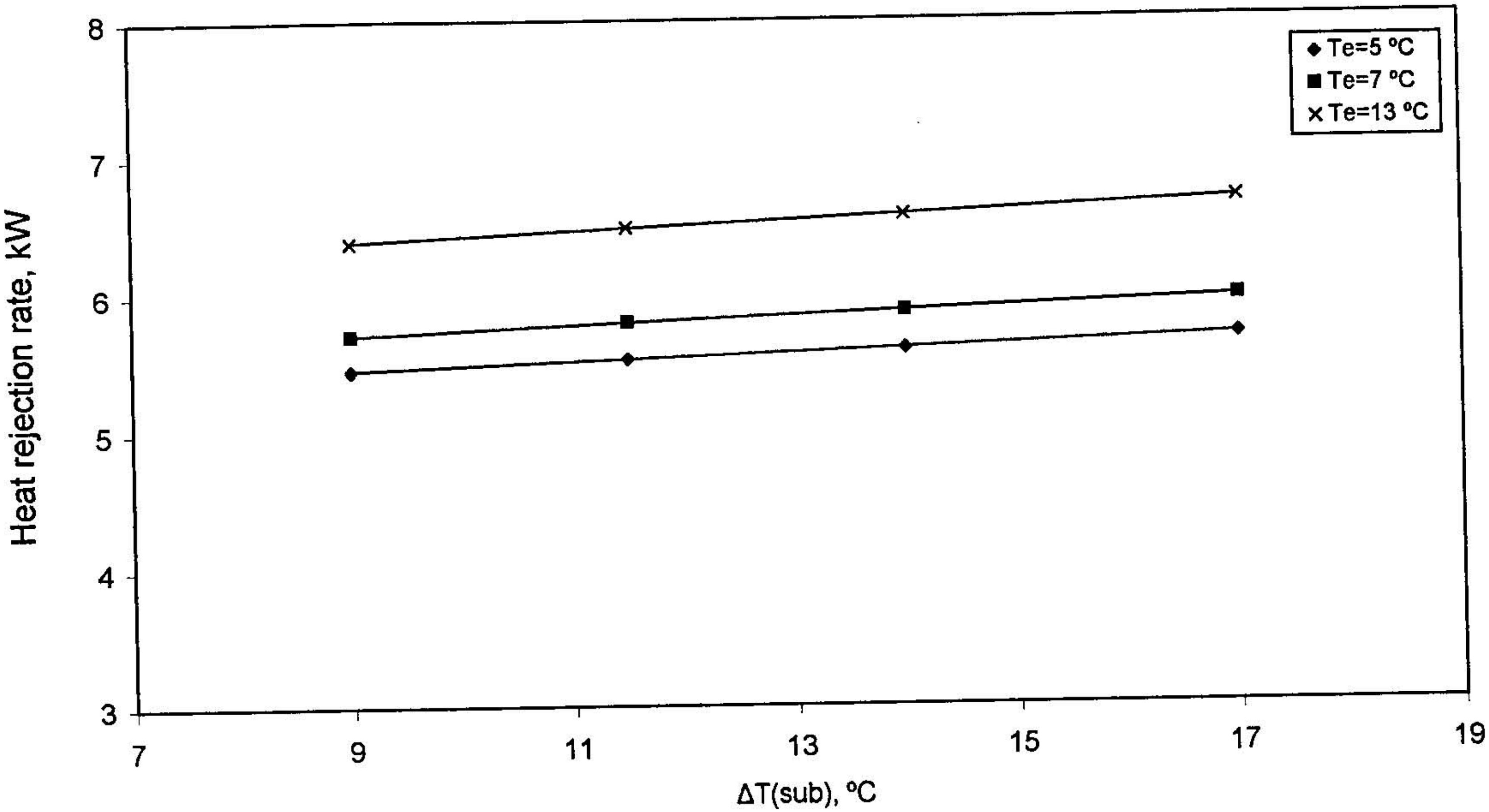


Figure 5.41: Variation of heat rejection rate with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R22

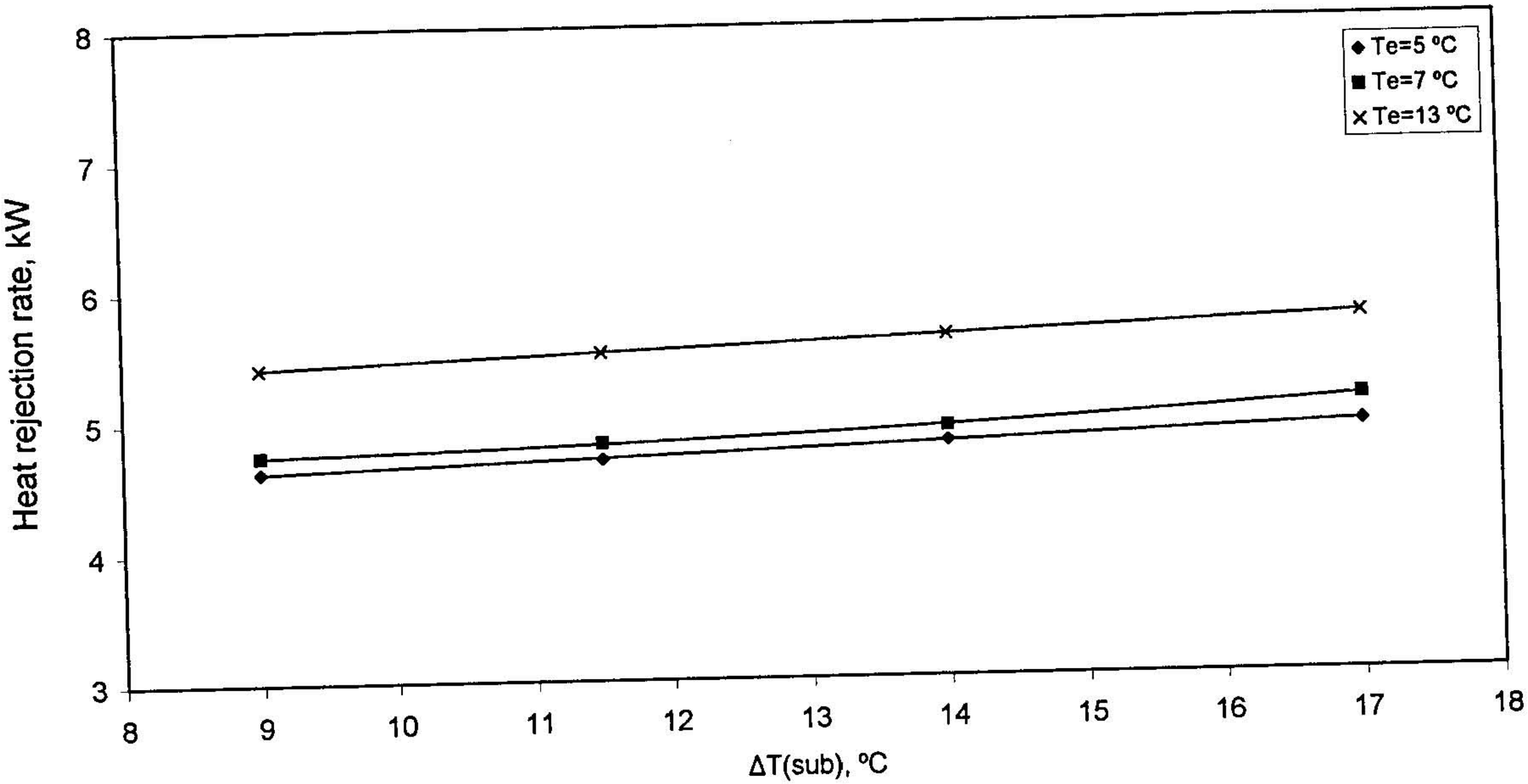


Figure 5.42: Variation of heat rejection rate with ΔT(sub) at T_c=40 °C, ΔT(sup)=6 °C for R407c

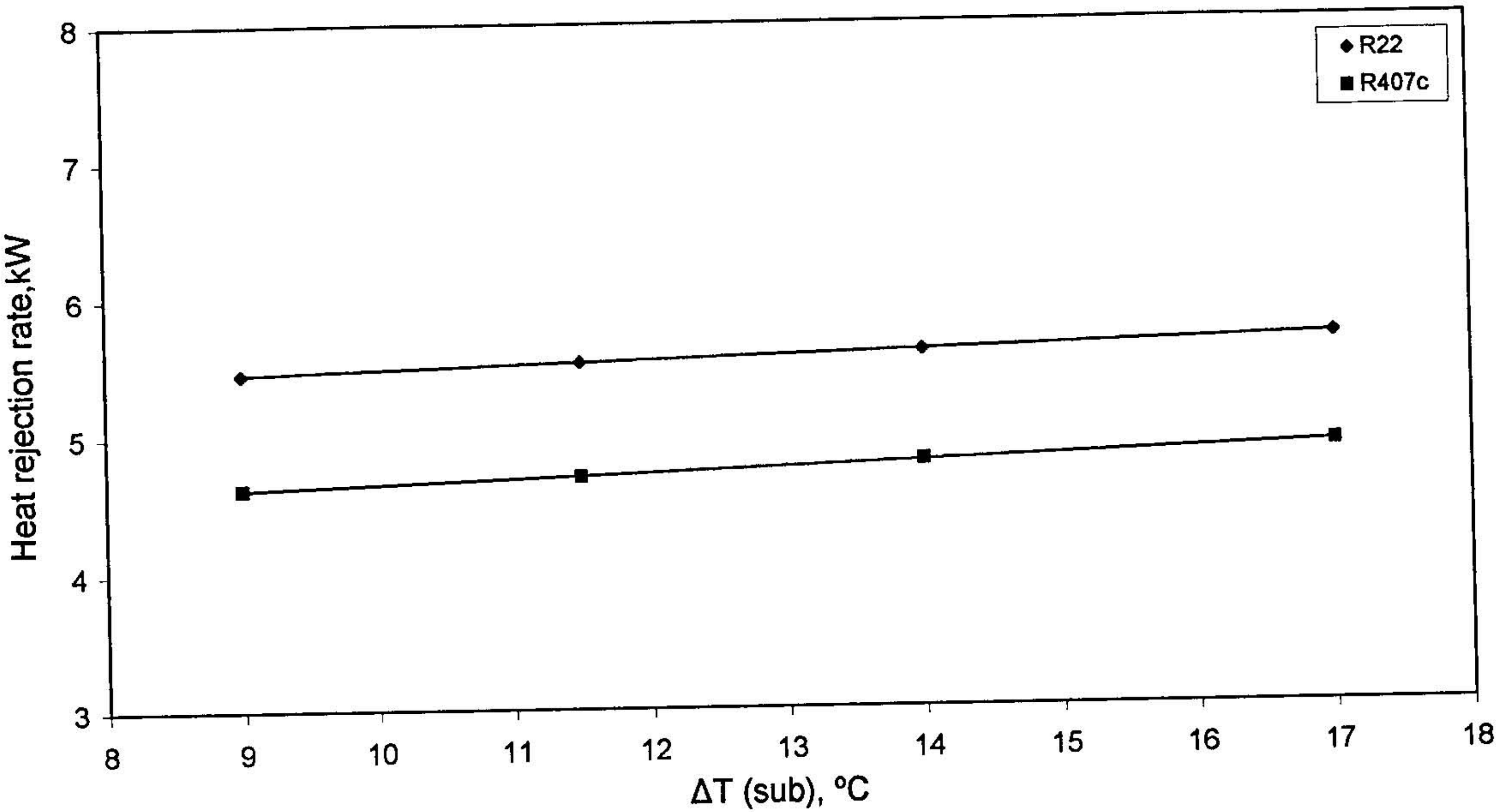


Figure 5.43 : Variation of heat rejection rate with ΔT (sub) at T_e=5 °C, T_c =40 °C , ΔT(sup) =6°C

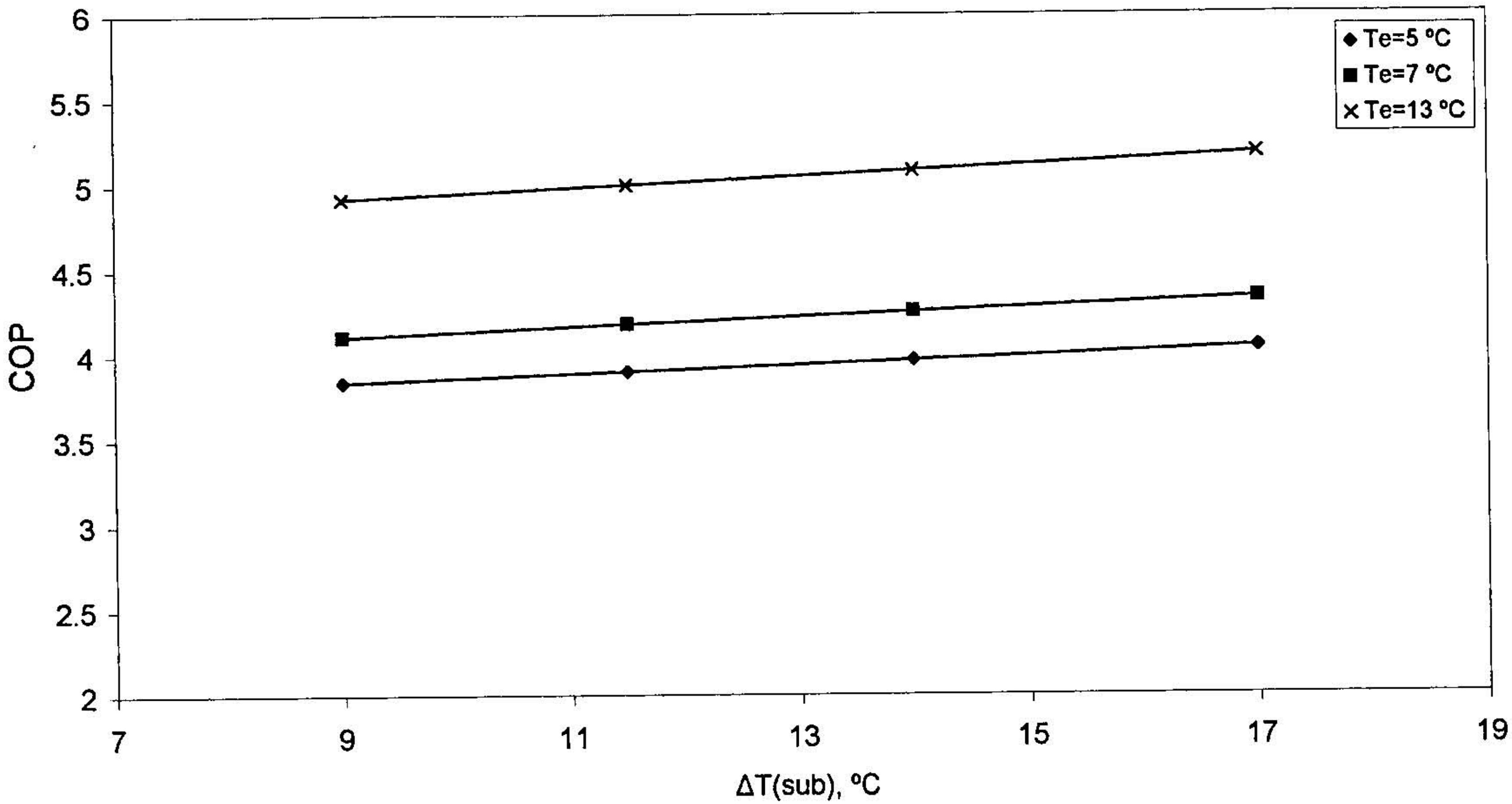


Figure 5.44: Variation of coefficient of performance with ΔT(sub) at T_c=40 ºC, ΔT(sup)=6 ºC for R22

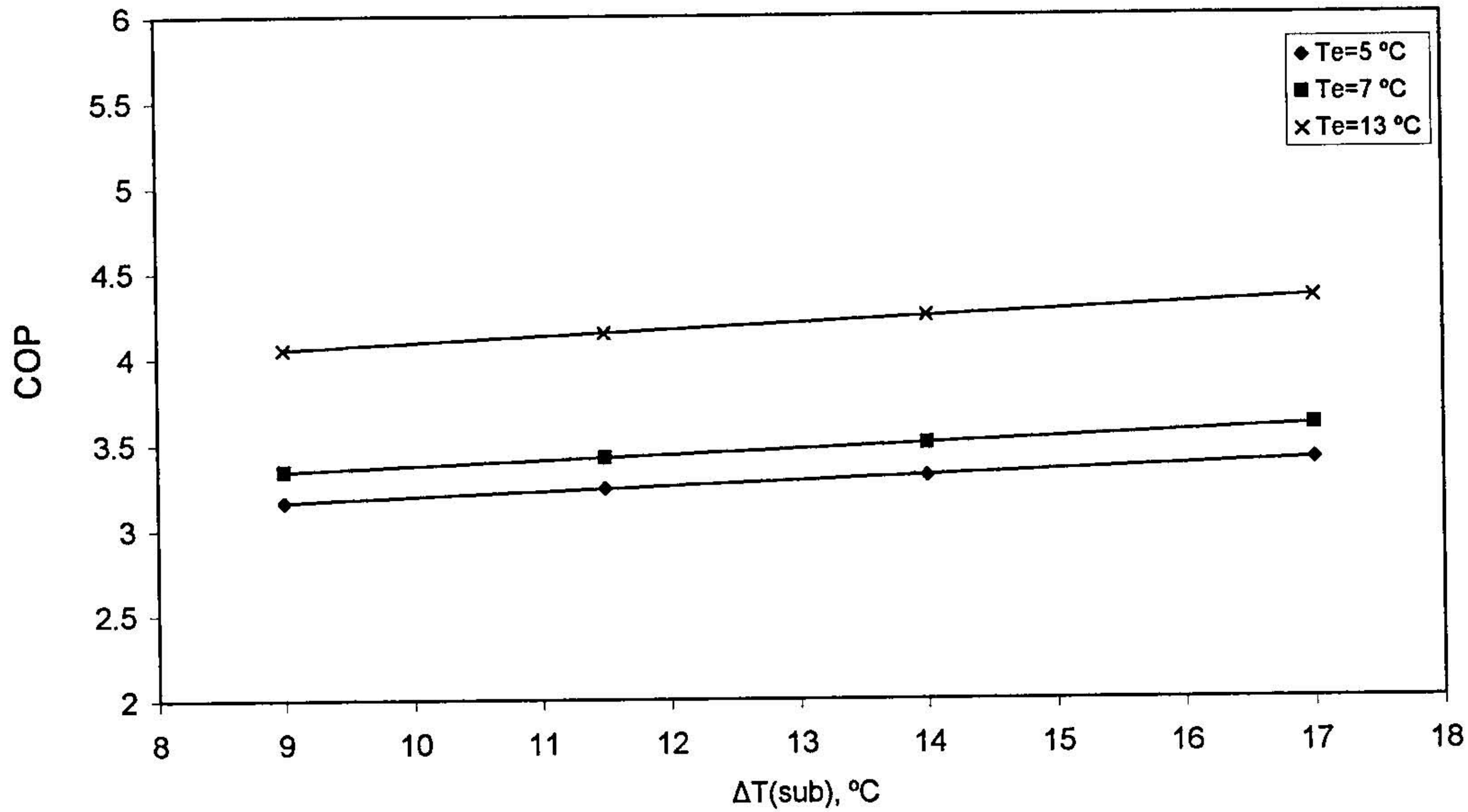


Figure 5.45: Variation of coefficient of performance with ΔT(sub) at T_e=5 °C, ΔT(sup)= 6 °C for R407c

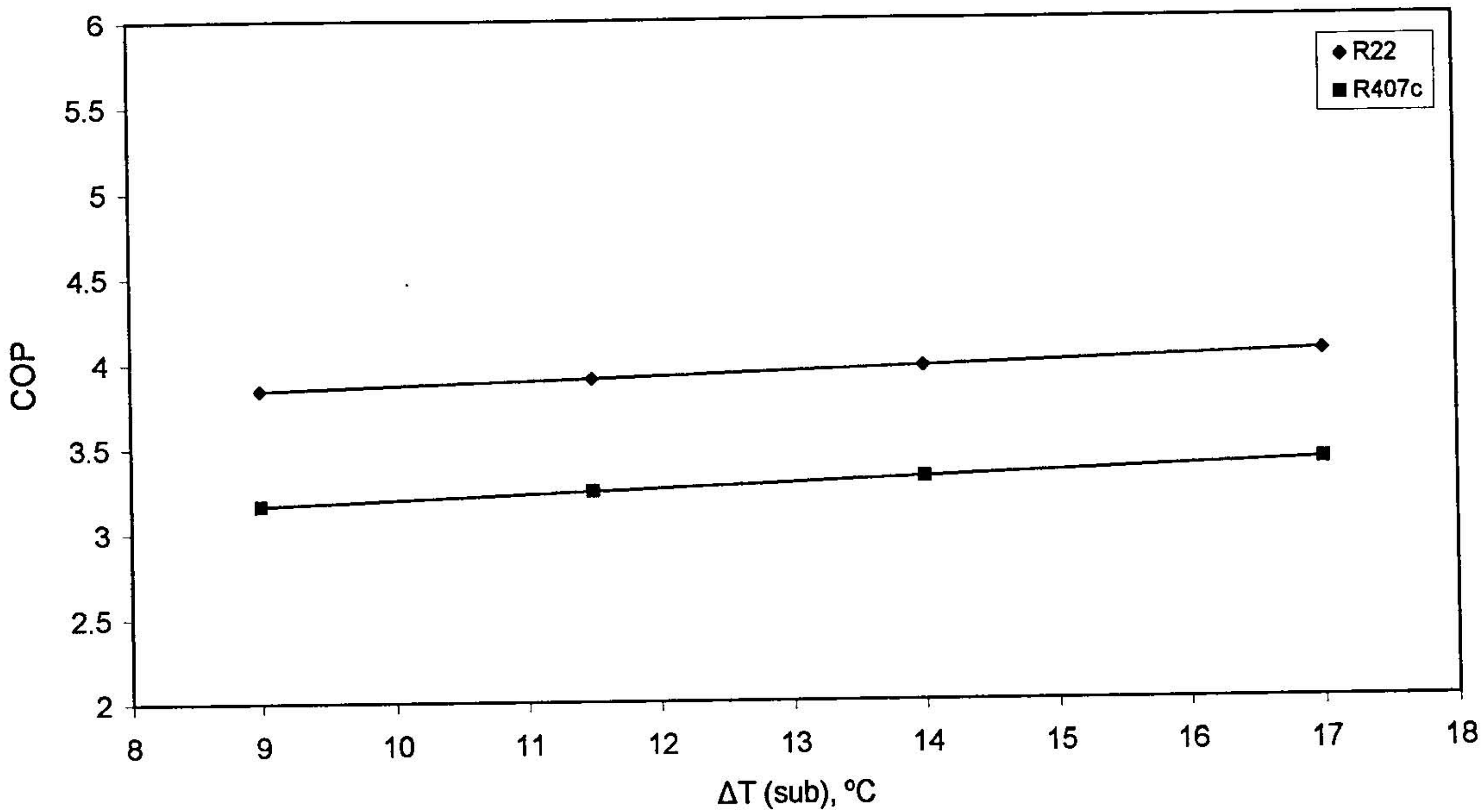


Figure 5.46:Variation of coefficient of performance with ΔT (sub) at T_e=5 °C, T_c=40 °C, ΔT(sup) =6 °C

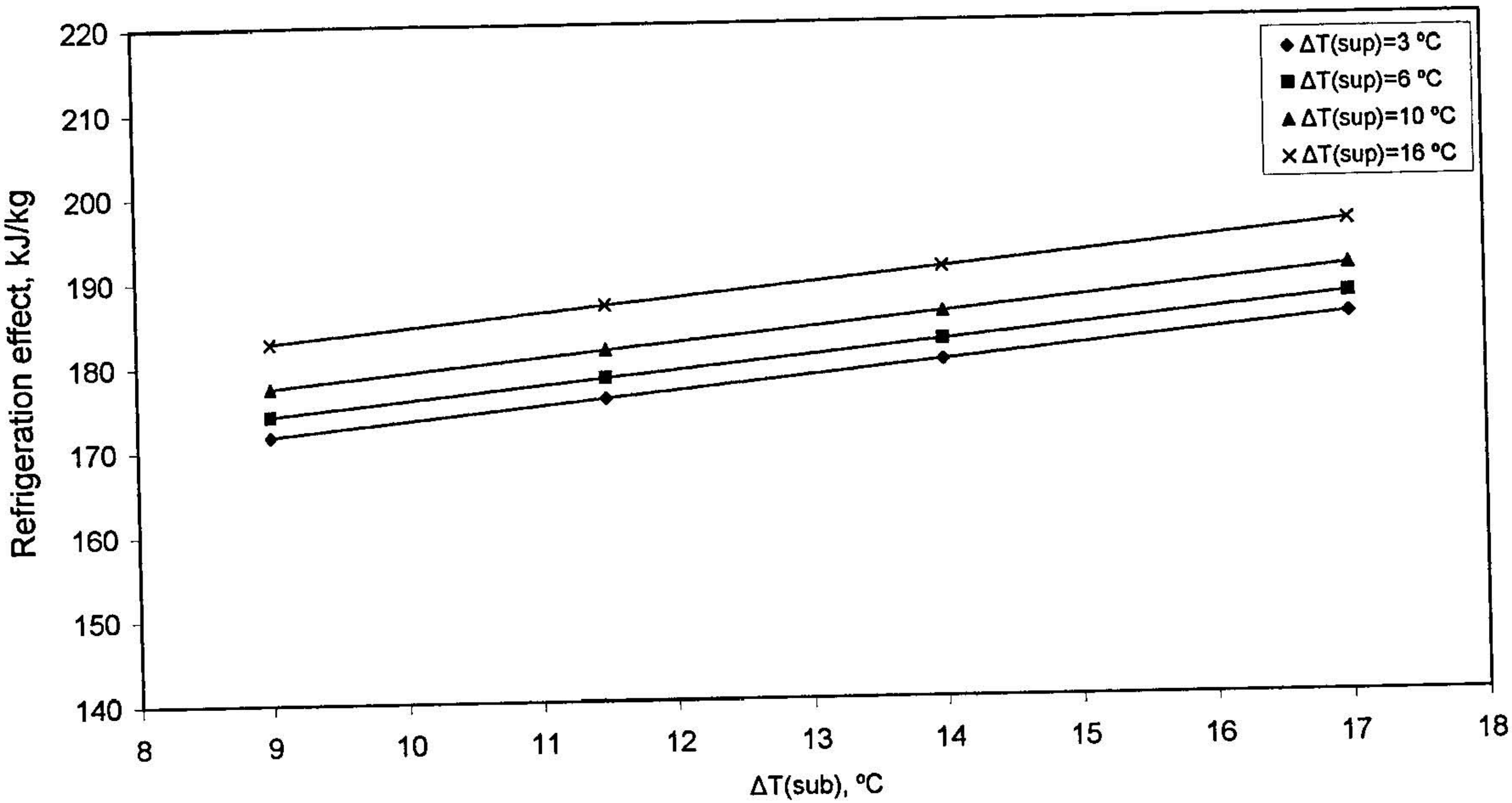


Figure 5.48: Variation of refrigeration effect with ΔT(sub) at T_e=5 °C, T_c=40 °C for R407c

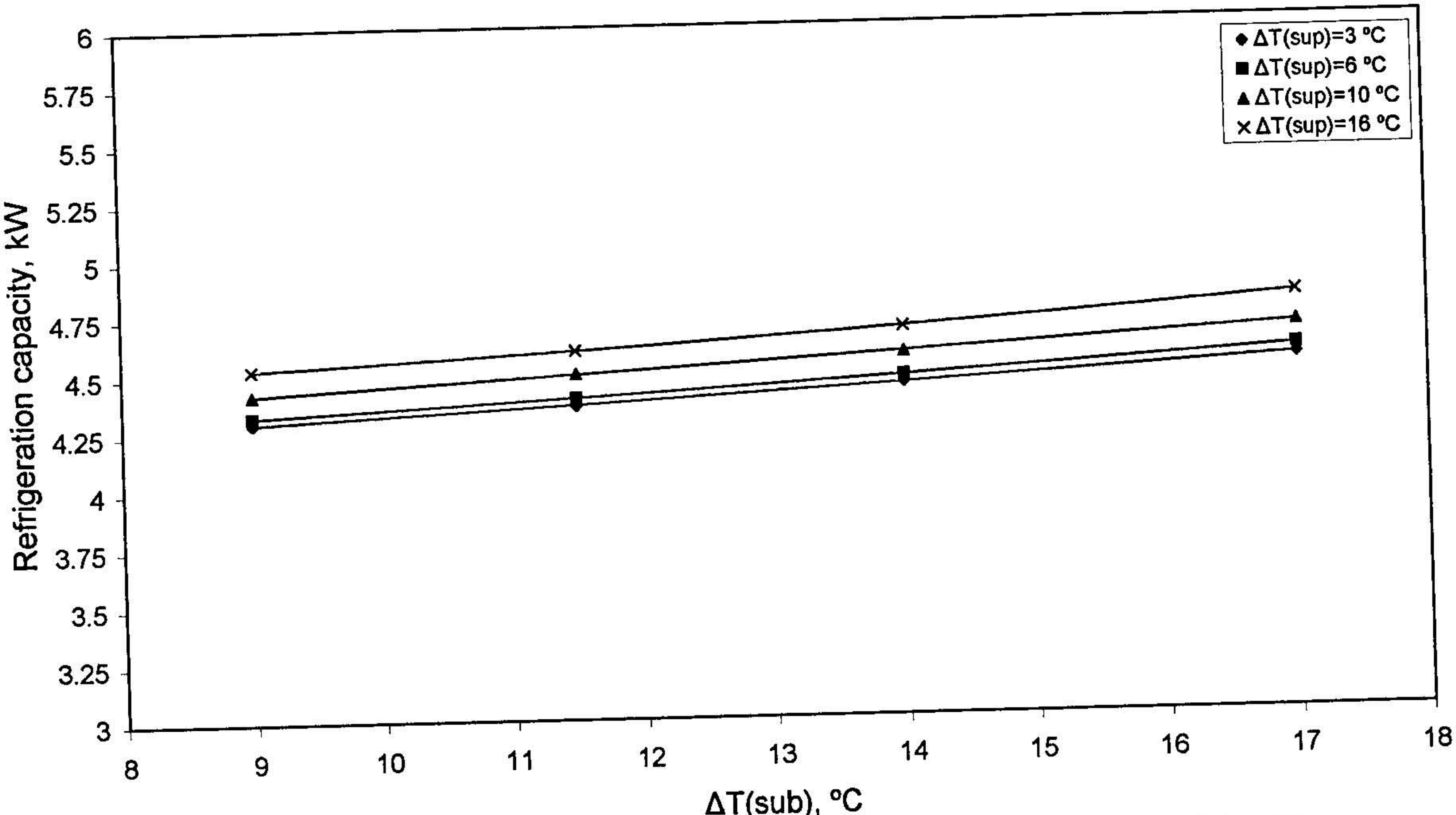


Figure 5.49: Variation of refrigeration capacity with ΔT(sub) at T_e=5 °C, T_c=40 °C for R22

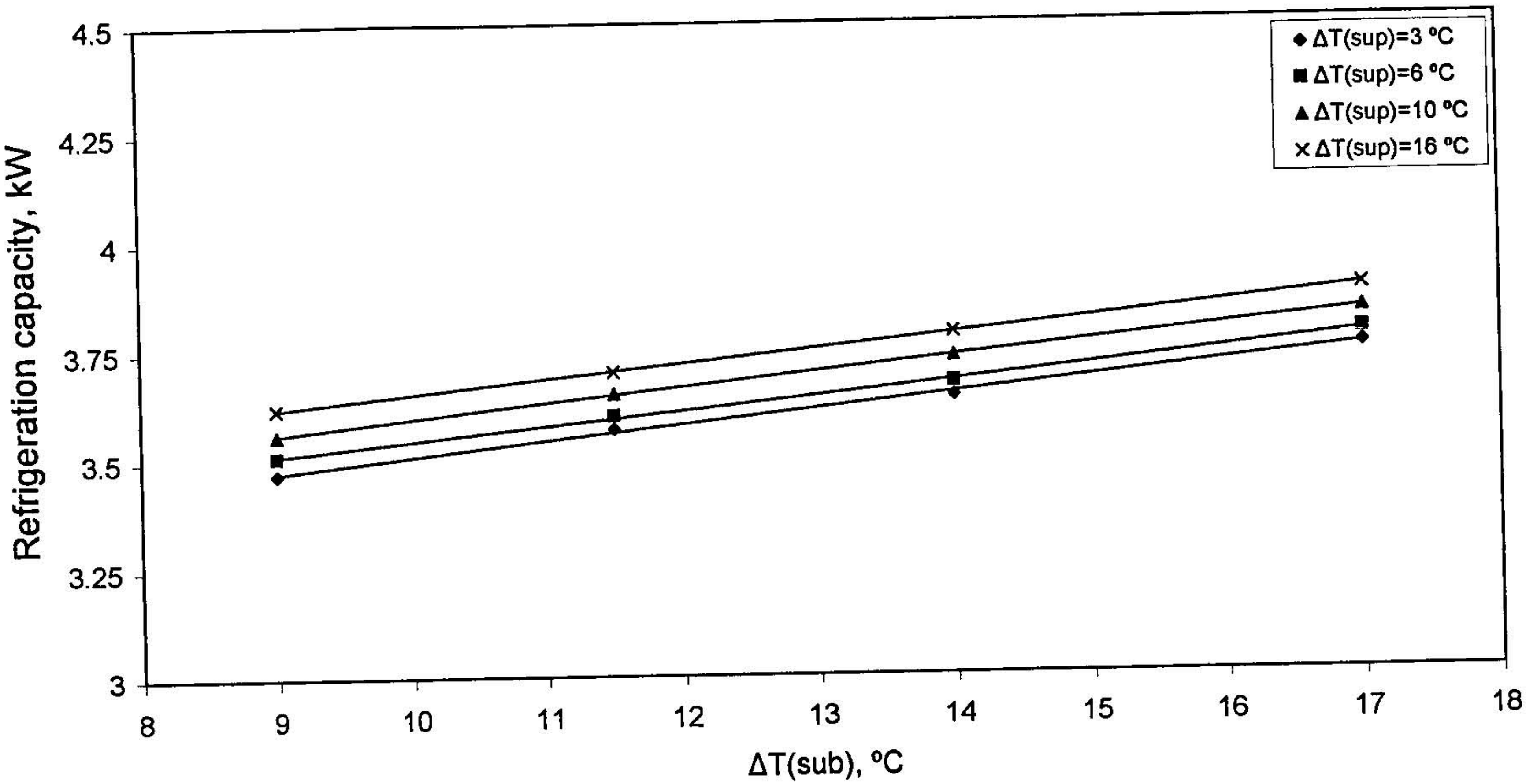


Figure 5.50: Variation of refrigeration capacity with ΔT(sub) at T_e=5 °C, T_c=40 °C for R407c

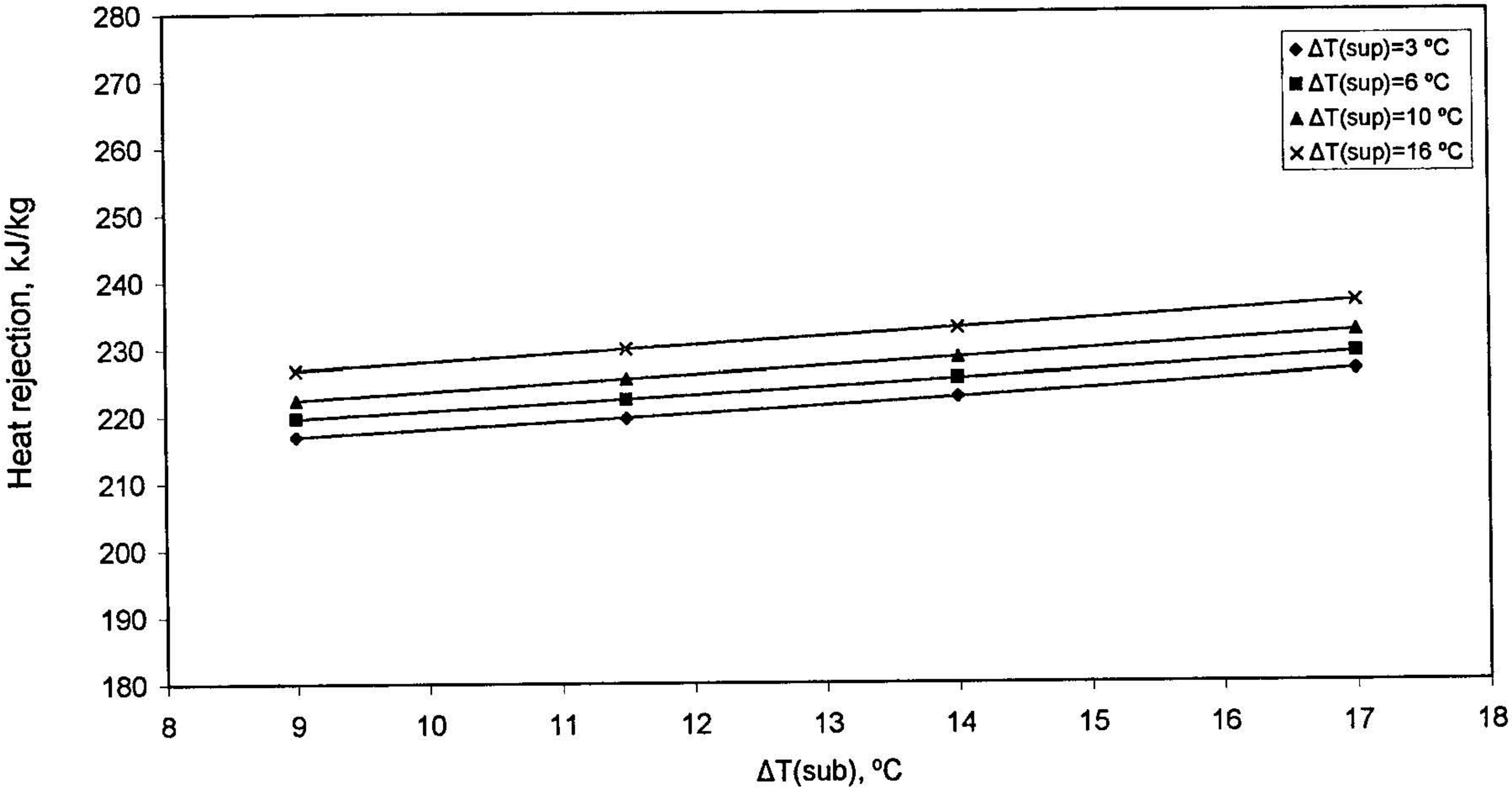


Figure 5.51: Variation of heat rejection with ΔT(sub) at T_e=5 °C, T_c=40 °C for R22

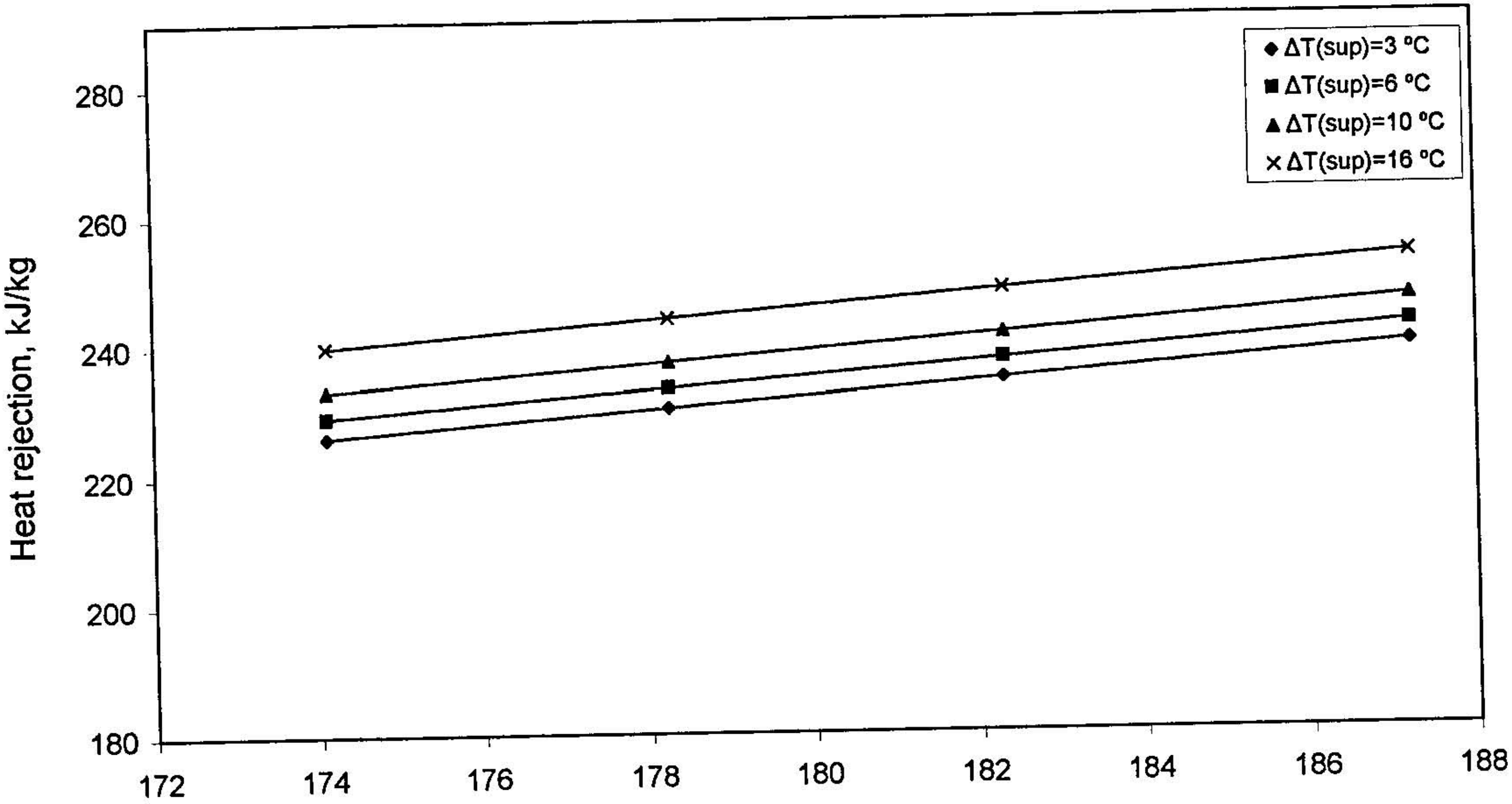


Figure 5.52: Variation of heat rejection with ΔT(sub) at T_e=5 °C, T_c=40 °C for R407c

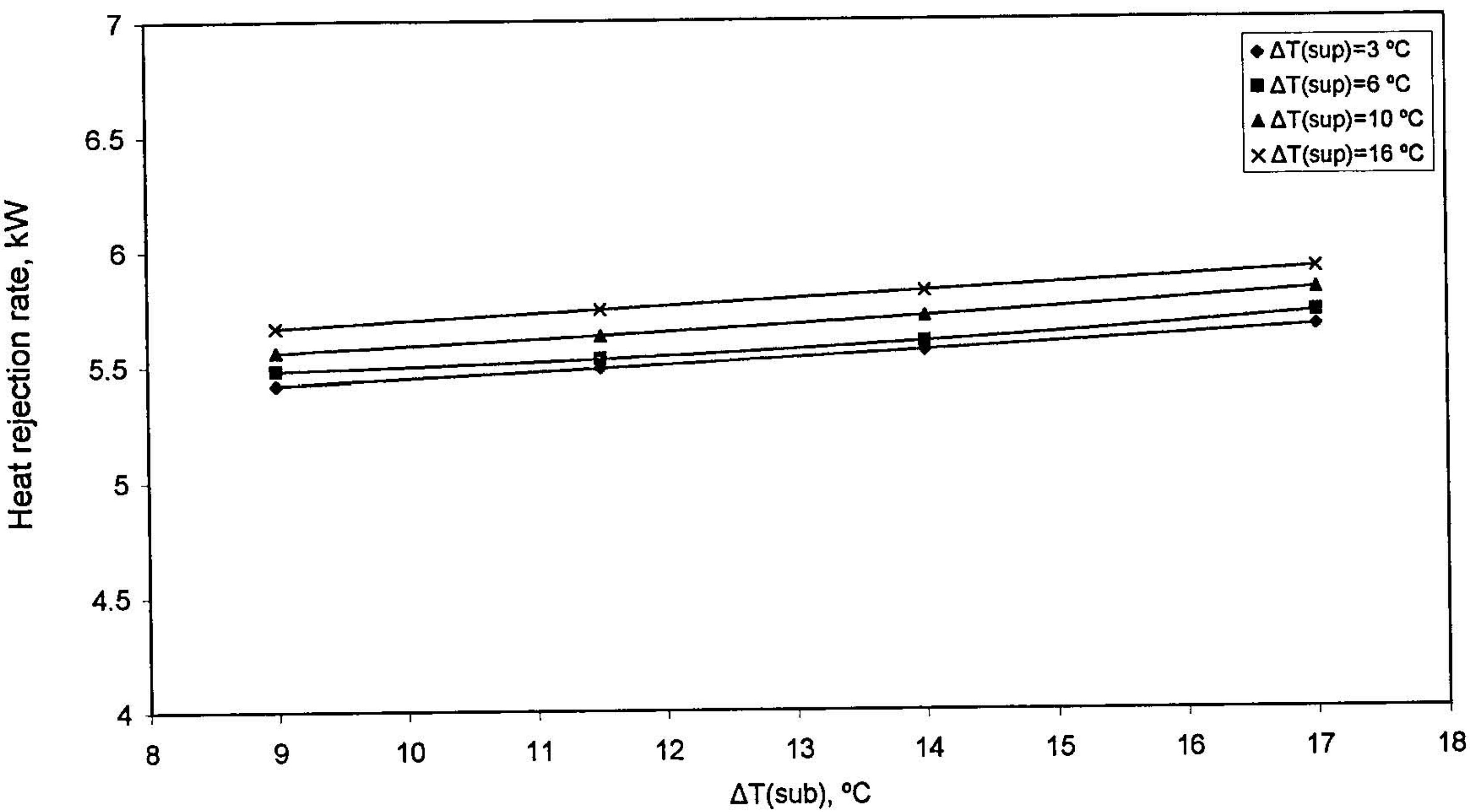


Figure 5.53: Variation of heat rejection rate with ΔT(sub) at T_e=5 °C, T_c=40 °C for R22

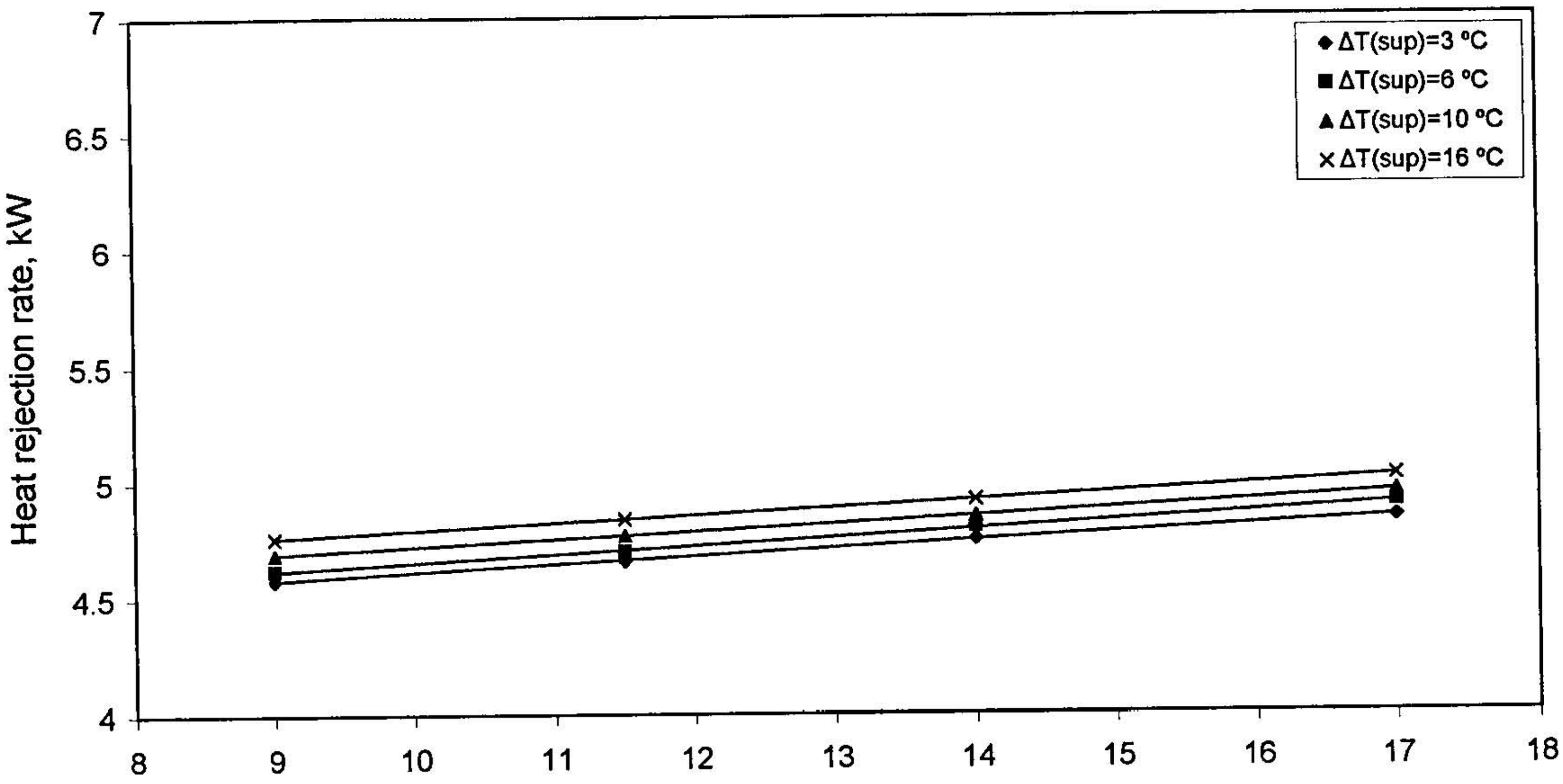


Figure 5.54: Variation of heat rejection rate with ΔT(sub) at T_e=5 °C, T_c=40 °C for R407c

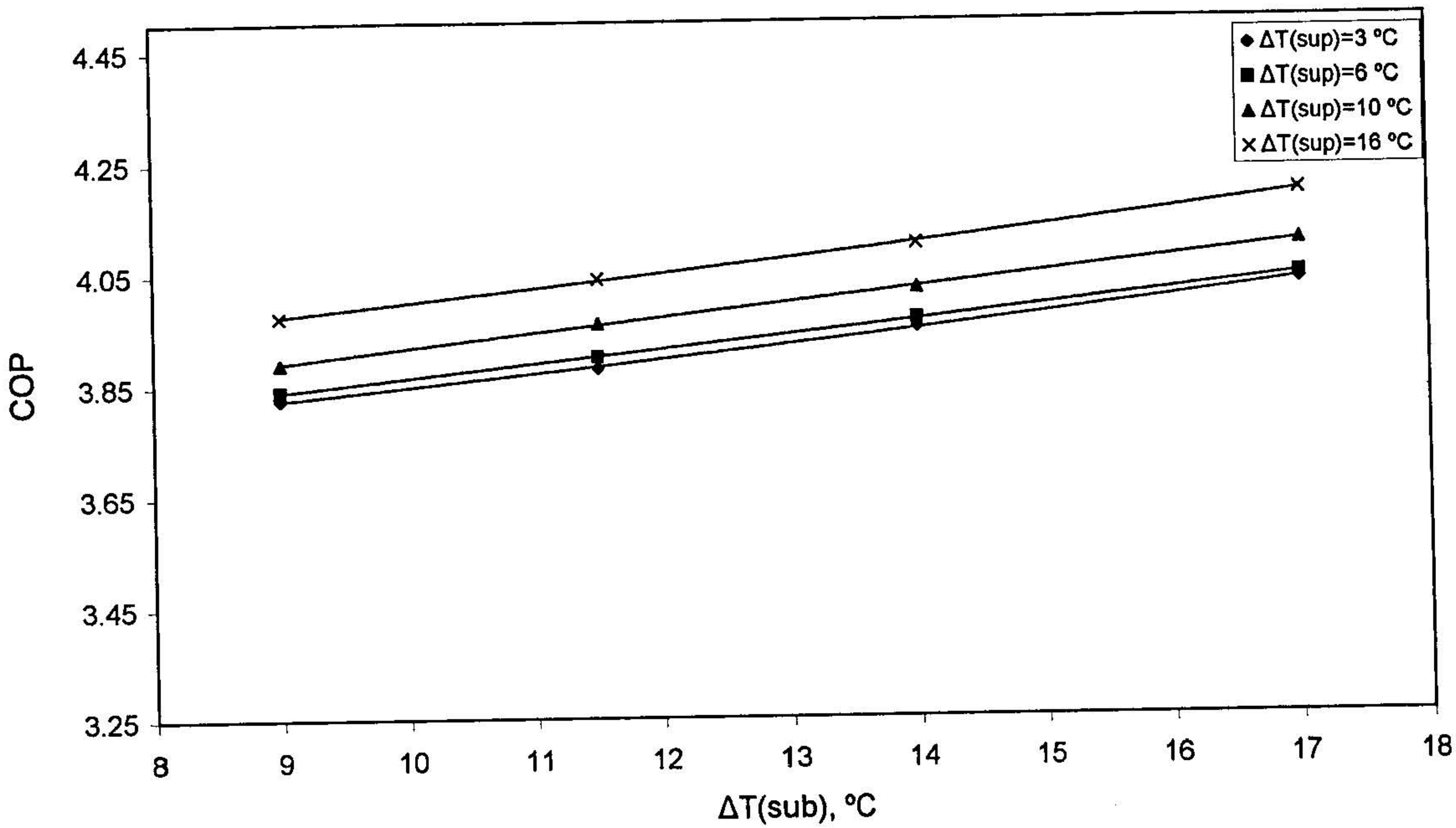


Figure 5.55: Variation of coefficient of performance with ΔT (sub) at T_e=5 °C, T_c=40 °C for R22

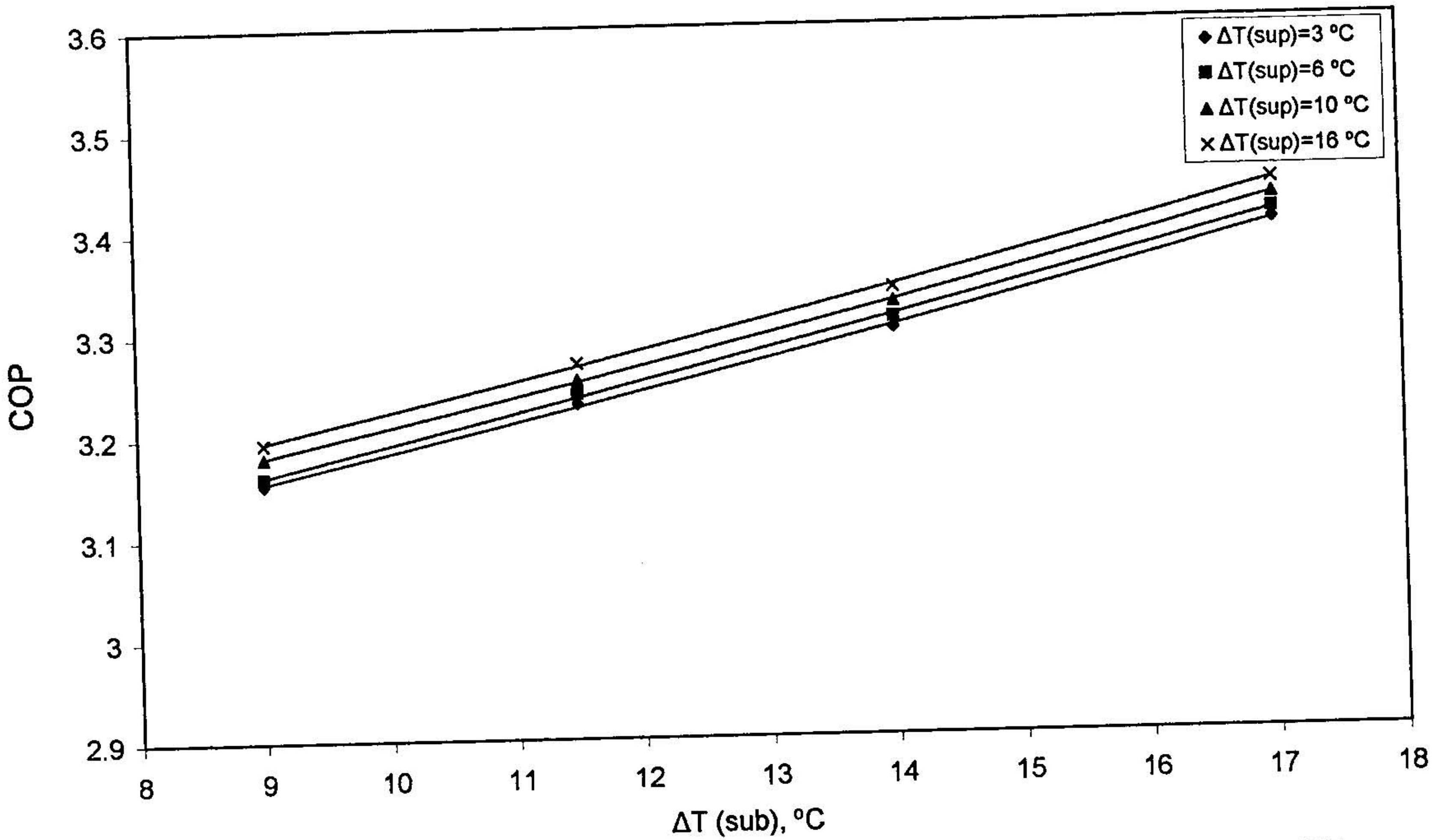


Figure 5.56: Variation of coefficient of performance with ΔT (sub) at T_e=5 °C, T_c=40 °C for R407c

6-CONCLUSIONS AND RECOMMENDATIONS

6.1 Introduction

A window type air conditioning unit was used to determine experimentally the performance parameters when replacing the refrigerant R407c instead of R22 under the effect of superheating, subcooling and combined superheating and subcooling. The performance parameters obtained were compared with that of R22.

6.2 Conclusions

1. The coefficient of performance of R407c for the superheating variation test reached a value of about 83% of that of R22 at T_e=5 °C, T_c = 40 °C, ΔT (sup) = 16 °C, ΔT (sub) = 9 °C and a value of 87% of that of R22 at T_c=38 °C. On the other hand, the COP reached a value of 84% of that of R22 for the subcooling variation test and a value of 82% of that of R22 for the combined variation test.
2. The refrigeration capacity of R407c when it replaces R22 decreased from a value of 5.6 kW to 4.66 kW at T_e = 13 °C, T_c = 40 °C, ΔT (sub) = 17 °C, ΔT (sup) = 6 °C which means that the refrigeration capacity of R407c reached a value of 83.2 % of that of R22. This is achieved during the subcooling test.
3. The refrigeration capacity and the coefficient of performance increase slightly as ΔT (sup) increases and it begin to become constant as ΔT (sup) close to 16 °C.
4. The electrical power consumed when R407c is used is slightly less than that of R22 for all tests and it reached an average value of 98% of that of R22.
5. Results of the present work indicated that refrigerant R407c is a suitable replacement for R22 in a 5 kW window type air conditioning units. R407c is

also considered as a primarily substitute for R22 in medium temperature refrigeration systems.

6. The refrigerant R407c has a subcooling effect greater than that of R22.

6.3 Recommendations

1. Further experimental studies are recommended to be carried out on R407c for a wider range of working and environmental conditions other than the conditions used in this research.
2. Due to the temperature glide of R407c, further study on the design of the condenser and evaporator is recommended in order to offset this effect.
3. More research and studies on other promising and environmentally safe alternative refrigerants, which will be useful for the industry when phasing out existing CFCs and HCFCs.
4. Further study on other air conditioning systems is recommended in order to generalize the results obtained from this study.

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APPENDIX

Table A1.1 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=9 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8
T2	86.5
T3	75.3
T4	40.65
T5	39.81
T6	39.13
T7	31
T8	4.91
T9	4.987
T10	5.1
T11	5.14
T12	5.18
T13	20.6
T14	32.4

2 . Pressure Reading (psi)

P1	71.35
P2	247.8

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temperature °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temperature °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8	0.04133	254.2	0.9285
Compressor outlet	86.5	0.01627	299	0.9748
Condenser outlet	31	0.0008549	82.53	0.3045
Evaporator inlet	4.987	0.007094	82.53	0.3115

Refrigeration effect (kJ/kg)	172.18
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.3
Work of compression (kJ/kg)	44.8
Heat rejection (kJ/kg)	216.98
Heat rejection rate (kW)	5.42
Coefficient of performance	3.826
Power consumption (kW)	1.492
Isentropic efficiency %	63.84
Volumetric efficiency %	87.2

Table A1.2 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.2
T2	89.5
T3	78.26
T4	43.26
T5	39.1
T6	38.2
T7	30.6
T8	4.86
T9	4.987
T10	4.99
T11	5.05
T12	5.1
T13	21.3
T14	34.3

2 . Pressure Reading (psi)

P1	71.15
P2	247.55

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RE

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	s(kJ/kg.k)	h(KJ/Kg)	s(KJ/Kg.k)
Compressor inlet	11.2	0.04198	256.3	0.9364
Compressor outlet	89.5	0.01661	301.7	0.9851
Condenser outlet	30.6	0.0008549	82.53	0.3045
Evaporator inlet	4.93	0.007094	82.53	0.3115

Refrigeration effect (kJ/kg)	174.28
Mas flow rate (gram/sec.)	24.83
Refrigeration capacity (kW)	4.327
Work of compression (kJ/kg)	45.4
Heat rejection (kJ/kg)	219.68
Heat rejection rate (kW)	5.455
Coefficient of performance	3.839
Power consumption (kW)	1.503
Isentropic efficiency %	64.5
Volumetric efficiency %	87.3

Table A1.3 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=9 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.1
T2	93.12
T3	79.91
T4	43.21
T5	39.12
T6	38.2
T7	30.9
T8	4.966
T9	4.978
T10	5.012
T11	5.016
T12	5.12
T13	21.7
T14	34.3

2 . Pressure Reading (psi)

P1	71.2
P2	247.1

3 . Power Reading (kW)

P	2.274

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.1	0.04289	259.3	0.9469
Compressor outlet	93.12	0.01679	304.8	0.9906
Condenser outlet	30.9	0.0008546	82.4	0.3041
Evaporator inlet	5.012	0.007059	82.4	0.311

Refrigeration effect (kJ/kg)	177.03
Mas flow rate (gram/sec.)	24.717
Refrigeration capacity (kW)	4.375
Work of compression (kJ/kg)	46
Heat rejection (kJ/kg)	223.18
Heat rejection rate (kW)	5.516
Coefficient of performance	3.85
Power consumption (kW)	1.516
Isentropic efficiency %	71.5
Volumetric efficiency %	87.12

Table A1.5 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=11.5 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8.1
T2	86.5
T3	75.3
T4	40.65
T5	39.81
T6	39.13
T7	28.52
T8	4.794
T9	4.96
T10	4.982
T11	5.01
T12	5.12
T13	21.53
T14	34

2 . Pressure Reading (psi)

P1	71.2
P2	246.9

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8.1	0.04133	256.2	0.9285
Compressor outlet	86.5	0.01627	299	0.9748
Condenser outlet	28.52	0.0008476	79.36	0.2943
Evaporator inlet	4.96	0.006481	79.36	0.3001

Refrigeration effect (kJ/kg)	174.64
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.362
Work of compression (kJ/kg)	44.8
Heat rejection (kJ/kg)	219.64
Heat rejection rate (kW)	5.49
Coefficient of performance	3.88
Power consumption (kW)	1.492
Isentropic efficiency %	63.84
Volumetric efficiency %	89.2

Table A1.6 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=14 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8
T2	86.5
T3	75.61
T4	40.66
T5	39.83
T6	39.12
T7	26.12
T8	4.83
T9	4.97
T10	4.982
T11	5.03
T12	5.13
T13	21.42
T14	34.16

2 . Pressure Reading (psi)

P1	71.3
P2	247.1

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8	0.04133	254.2	0.9285
Compressor outlet	86.5	0.01627	299	0.9748
Condenser outlet	26.12	0.0008408	76.32	0.2844
Evaporator inlet	4.96	0.005879	76.32	0.2891

Refrigeration effect (kJ/kg)	177.68
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.44
Work of compression (kJ/kg)	44.8
Heat rejection (kJ/kg)	222.68
Heat rejection rate (kW)	5.56
Coefficient of performance	3.95
Power consumption (kW)	1.492
Isentropic efficiency %	63.84
Volumetric efficiency %	89.2

Table A1.7 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C,ΔT(sub)=17 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8
T2	86.5
T3	75.62
T4	40.64
T5	39.82
T6	39.14
T7	23.13
T8	4.896
T9	4.931
T10	4.972
T11	5.01
T12	5.12
T13	21.31
T14	34

2 . Pressure Reading (psi)

P1	70.9
P2	246.8

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8	0.04133	254.2	0.9285
Compressor outlet	86.5	0.01627	299	0.9748
Condenser outlet	23.13	0.0008326	72.57	0.272
Evaporator inlet	4.931	0.005141	72.57	0.2757

Refrigeration effect (kJ/kg)	181.63
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.53
Work of compression (kJ/kg)	44.8
Heat rejection (kJ/kg)	226.43
Heat rejection rate (kW)	5.66
Coefficient of performance	4.05
Power consumption (kW)	1.492
Isentropic efficiency %	63.84
Volumetric efficiency %	89.2

Table A1.8 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C ,ΔT(sub)=11.5 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.1
T2	89.42
T3	79.16
T4	43.13
T5	39.21
T6	38.18
T7	28.48
T8	4.72
T9	4.79
T10	4.8
T11	4.92
T12	5.08
T13	20.92
T14	34.2

2 . Pressure Reading (psi)

P1	71.25
P2	247.41

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.1	0.04198	256.3	0.9364
Compressor outlet	89.42	0.0161	301.7	0.9851
Condenser outlet	28.48	0.0008475	79.34	0.2942
Evaporator inlet	4.79	0.006474	79.34	0.3

Refrigeration effect (kJ/kg)	176.96
Mas flow rate (gram/sec.)	24.83
Refrigeration capacity (kW)	4.394
Work of compression (kJ/kg)	45.4
Heat rejection (kJ/kg)	222.36
Heat rejection rate (kW)	5.53
Coefficient of performance	3.9
Power consumption (kW)	1.503
Isentropic efficiency %	64.5
Volumetric efficiency %	89.3

Table A1. 9 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C , ΔT(sub)=14 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.31
T2	89.52
T3	79.28
T4	43.23
T5	39.31
T6	38.21
T7	26.1
T8	4.71
T9	5.012
T10	5.02
T11	5.06
T12	5.1
T13	20.5
T14	34.25

2 . Pressure Reading (psi)

P1	71.2
P2	247.5

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.31	0.04198	256.3	0.9364
Compressor outlet	89.52	0.0161	301.7	0.9851
Condenser outlet	26.1	0.0008408	76.32	0.2844
Evaporator inlet	5.012	0.005864	76.32	0.2891

Refrigeration effect (kJ/kg)	180
Mas flow rate (gram/sec.)	24.83
Refrigeration capacity (kW)	4.47
Work of compression (kJ/kg)	45.4
Heat rejection (kJ/kg)	225.4
Heat rejection rate (kW)	5.597
Coefficient of performance	3.965
Power consumption (kW)	1.503
Isentropic efficiency %	64.5
Volumetric efficiency %	89.3

Table A1.10 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C , ΔT(sub)=17 °C , ΔT(sup)=6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.1
T2	89.49
T3	79.18
T4	43.31
T5	39.25
T6	38.17
T7	23.12
T8	4.72
T9	5.013
T10	5.02
T11	5.07
T12	5.1
T13	20.2
T14	34

2 . Pressure Reading (psi)

P1	71.2
P2	247.6

3 . Power Reading (kW)

P	2.238

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.1	0.04287	259.3	0.9469
Compressor outlet	89.49	0.01656	299.9	0.9783
Condenser outlet	23.12	0.0008326	72.57	0.272
Evaporator inlet	5.013	0.005138	72.57	0.2759

Refrigeration effect (kJ/kg)	183.52
Mas flow rate (gram/sec.)	24.83
Refrigeration capacity (kW)	4.57
Work of compression (kJ/kg)	45.4
Heat rejection (kJ/kg)	228.92
Heat rejection rate (kW)	5.684
Coefficient of performance	4.04
Power consumption (kW)	1.503
Isentropic efficiency %	64.5
Volumetric efficiency %	89.3

Table A1.11 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C, ΔT(sub)=9 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.1
T2	93.12
T3	79.91
T4	43.21
T5	39.12
T6	38.2
T7	30.9
T8	4.966
T9	4.978
T10	5.012
T11	5.016
T12	5.12
T13	21.7
T14	34.3

2 . Pressure Reading (psi)

P1	71.2
P2	247.1

3 . Power Reading (kW)

P	2.274

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.1	0.04287	259.3	0.9469
Compressor outlet	93.12	0.01679	304.8	0.9906
Condenser outlet	30.9	0.0008546	82.4	0.3041
Evaporator inlet	4.978	0.007059	82.4	0.311

Refrigeration effect (kJ/kg)	176.9
Mas flow rate (gram/sec.)	24.99
Refrigeration capacity (kW)	4.42
Work of compression (kJ/kg)	45.5
Heat rejection (kJ/kg)	222.4
Heat rejection rate (kW)	5.56
Coefficient of performance	3.89
Power consumption (kW)	1.516
Isentropic efficiency %	65.5
Volumetric efficiency %	89.13

Table A1.12 : Data for 960 g of R22 at T_c=40 °C,T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.13
T2	93.2
T3	79.9
T4	43.31
T5	39.1
T6	38.18
T7	28.58
T8	4.97
T9	4.983
T10	5.014
T11	5.016
T12	5.13
T13	21.61
T14	34.1

2 . Pressure Reading (psi)

P1	71.2
P2	247.3

3 . Power Reading (kW)

P	2.274

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.13	0.04287	259.3	0.9469
Compressor outlet	93.2	0.01679	304.8	0.9906
Condenser outlet	28.58	0.0008475	79.34	0.2942
Evaporator inlet	5.014	0.006457	79.34	0.3

Refrigeration effect (kJ/kg)	179.96
Mas flow rate (gram/sec.)	24.99
Refrigeration capacity (kW)	4.5
Work of compression (kJ/kg)	45.5
Heat rejection (kJ/kg)	225.46
Heat rejection rate (kW)	5.63
Coefficient of performance	3.96
Power consumption (kW)	1.516
Isentropic efficiency %	65.5
Volumetric efficiency %	89.13

Table A1.13 : Data for 960 g of R22 at T_c=40 °C,T_e=5 °C , ΔT(sub)=14 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.21
T2	93.22
T3	79.31
T4	43.12
T5	39.14
T6	38.17
T7	26.12
T8	4.983
T9	4.991
T10	5.012
T11	5.013
T12	5.14
T13	21.32
T14	34.2

2 . Pressure Reading (psi)

P1	71.25
P2	247.2

3 . Power Reading (kW)

P	2.274

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.21	0.04287	259.3	0.9469
Compressor outlet	93.22	0.01679	304.8	0.9906
Condenser outlet	26.12	0.0008408	76.32	0.2844
Evaporator inlet	4.991	0.005864	76.32	0.2891

Refrigeration effect (kJ/kg)	182.98
Mas flow rate (gram/sec.)	24.99
Refrigeration capacity (kW)	4.57
Work of compression (kJ/kg)	45.5
Heat rejection (kJ/kg)	228.48
Heat rejection rate (kW)	5.71
Coefficient of performance	4.02
Power consumption (kW)	1.516
Isentropic efficiency %	65.5
Volumetric efficiency %	89.13

Table A1.14 : Data for 960 g of R22 at T_c=40 °C , T_e=5 °C, ΔT(sub)=17 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15
T2	93.1
T3	79.46
T4	43.26
T5	39.16
T6	38.11
T7	23.18
T8	4.978
T9	5.01
T10	5.014
T11	5.016
T12	5.13
T13	21.26
T14	33.2

2 . Pressure Reading (psi)

P1	71.3
P2	247.3

3 . Power Reading (kW)

P	2.253

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.58
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15	0.04287	259.3	0.9469
Compressor outlet	93.1	0.01679	304.8	0.9906
Condenser outlet	23.18	0.0008327	72.63	0.2722
Evaporator inlet	5.01	0.005138	72.63	0.2759

Refrigeration effect (kJ/kg)	186.67
Mas flow rate (gram/sec.)	24.99
Refrigeration capacity (kW)	4.67
Work of compression (kJ/kg)	45.5
Heat rejection (kJ/kg)	232.17
Heat rejection rate (kW)	5.82
Coefficient of performance	4.1
Power consumption (kW)	1.516
Isentropic efficiency %	65.5
Volumetric efficiency %	89.12

Table A1.15 : Data for 960 g of R22 at T_c=40 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.12
T2	98.3
T3	81.21
T4	43.28
T5	39.2
T6	38.16
T7	30.89
T8	4.87
T9	4.916
T10	4.965
T11	5.02
T12	5.03
T13	21.22
T14	33.23

2 . Pressure Reading (psi)

P1	71.2
P2	247.1

3 . Power Reading (kW)

P	2.2815

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.5786
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.12	0.04413	263.6	0.9618
Compressor outlet	98.3	0.01718	309.9	1.003
Condenser outlet	30.89	0.0008546	82.39	0.3041
Evaporator inlet	4.916	0.007076	82.39	0.311

Refrigeration effect (kJ/kg)	181.21
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.53
Work of compression (kJ/kg)	45.6
Heat rejection (kJ/kg)	226.81
Heat rejection rate (kW)	5.67
Coefficient of performance	3.974
Power consumption (kW)	1.519
Isentropic efficiency %	66.9
Volumetric efficiency %	89

Table A1.16 : Data for 960 g of R22 at T_c=40 °C,T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.06
T2	98.31
T3	81.43
T4	43.27
T5	39.15
T6	38.14
T7	28.51
T8	4.86
T9	4.921
T10	4.969
T11	5.013
T12	5.02
T13	21.12
T14	34.2

2 . Pressure Reading (psi)

P1	71.26
P2	247.35

3 . Power Reading (kW)

P	2.2815

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.5807
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.06	0.04413	263.6	0.9618
Compressor outlet	98.31	0.01718	309.9	1.003
Condenser outlet	28.51	0.0008475	79.34	0.2942
Evaporator inlet	4.969	0.006474	79.34	0.3

Refrigeration effect (kJ/kg)	184.26
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.67
Work of compression (kJ/kg)	45.6
Heat rejection (kJ/kg)	229.86
Heat rejection rate (kW)	5.743
Coefficient of performance	4.04
Power consumption (kW)	1.519
Isentropic efficiency %	66.9
Volumetric efficiency %	89

Table A1.17 : Data for 960 g of R22 at T_c=40 °C , T_e=5 °C , $\Delta T(sub)$=14 °C , $\Delta T(sup)$ =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.1
T2	98.16
T3	81.39
T4	43.19
T5	39.16
T6	38.09
T7	26.1
T8	4.896
T9	4.912
T10	4.967
T11	5.01
T12	5.015
T13	21
T14	33.26

2 . Pressure Reading (psi)

P1	71.35
P2	247.1

3 . Power Reading (kW)

P	2.2815

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.581
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.78

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.1	0.04413	263.6	0.9618
Compressor outlet	98.16	0.01718	309.9	1.003
Condenser outlet	26.1	0.0008407	76.3	0.2843
Evaporator inlet	4.967	0.005876	76.3	0.2891

Refrigeration effect (kJ/kg)	187.3
Mas flow rate (gram/sec.)	24.98
Refrigeration capacity (kW)	4.68
Work of compression (kJ/kg)	45.6
Heat rejection (kJ/kg)	225.23
Heat rejection rate (kW)	5.82
Coefficient of performance	4.1
Power consumption (kW)	1.519
Isentropic efficiency %	66.9
Volumetric efficiency %	89

Table A1.18 : Data for 960 g of R22 at T_c=40 °C ,T_e=7 °C , ΔT(sub)=9 °C, ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	13.1
T2	88.1
T3	78.2
T4	41.1
T5	40.1
T6	39
T7	30.9
T8	6.937
T9	6.91
T10	6.986
T11	7.051
T12	7.21
T13	21.8
T14	33.85

2 . Pressure Reading (psi)

P1	76.6
P2	247.8

3 . Power Reading (kW)

P	2.235

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	7	Suction Pressure (Mpa)	0.62
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.8

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	13.1	0.03943	257	0.9334
Compressor outlet	88.1	0.01625	299.5	0.9741
Condenser outlet	30.9	0.0008548	82.47	0.3043
Evaporator inlet	6.91	0.006328	82.47	0.3104

Refrigeration effect (kJ/kg)	174.53
Mas flow rate (gram/sec.)	26.29
Refrigeration capacity (kW)	4.59
Work of compression (kJ/kg)	42.5
Heat rejection (kJ/kg)	217.03
Heat rejection rate (kW)	5.706
Coefficient of performance	4.106
Power consumption (kW)	1.49
Isentropic efficiency %	64.7
Volumetric efficiency %	89.7

Table A1.19 : Data for 960 g of R22 at T_c=40 °C , T_e=10 °C , ΔT(sub)=9 °C, ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	16.1
T2	90
T3	73.78
T4	41.36
T5	40.43
T6	39.15
T7	30.87
T8	9.731
T9	9.846
T10	10.01
T11	10.33
T12	10.41
T13	21.52
T14	34.35

2 . Pressure Reading (psi)

P1	84.6
P2	255.1

3 . Power Reading (kW)

P	2.208

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	10	Suction Pressure (Mpa)	0.675
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.88

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	16.1	0.03625	258.2	0.9302
Compressor outlet	90	0.0157	301.1	0.9771
Condenser outlet	30.87	0.0008545	82.36	0.304
Evaporator inlet	9.846	0.005091	82.36	0.3085

Refrigeration effect (kJ/kg)	175.84
Mas flow rate (gram/sec.)	25.73
Refrigeration capacity (kW)	4.524
Work of compression (kJ/kg)	42.9
Heat rejection (kJ/kg)	218.74
Heat rejection rate (kW)	5.63
Coefficient of performance	4.1
Power consumption (kW)	1.472
Isentropic efficiency %	61.3
Volumetric efficiency %	90.83

Table A1.20 : Data for 960 g of R22 at T_c=40 °C,T_e=13 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	19.15
T2	84
T3	75.6
T4	39.97
T5	40.1
T6	40.07
T7	30.78
T8	12.81
T9	12.85
T10	12.97
T11	13
T12	13.13
T13	22.16
T14	35

2 . Pressure Reading (psi)

P1	94.1
P2	268.1

3 . Power Reading (kW)

P	2.16

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	13	Suction Pressure (Mpa)	0.736
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.92

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	19.15	0.03333	259.3	0.9268
Compressor outlet	84	0.01485	295.3	0.9593
Condenser outlet	30.78	0.0008542	82.25	0.3036
Evaporator inlet	12.97	0.004348	82.25	0.3073

Refrigeration effect (kJ/kg)	177.05
Mas flow rate (gram/sec.)	30
Refrigeration capacity (kW)	5.31
Work of compression (kJ/kg)	36
Heat rejection (kJ/kg)	213.05
Heat rejection rate (kW)	6.39
Coefficient of performance	4.918
Power consumption (kW)	1.44
Isentropic efficiency %	68.3
Volumetric efficiency %	91.26

Table A1.21 : Data for 960 g of R22 at T_c=41 °C , T_e=5 °C , ΔT(sub)=9 °C, ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.2
T2	90.1
T3	79.31
T4	41.9
T5	41.2
T6	40.23
T7	31.76
T8	4.96
T9	5.03
T10	5.11
T11	5.13
T12	5.17
T13	21.5
T14	32.43

2 . Pressure Reading (psi)

P1	76.2
P2	265.5

3 . Power Reading (kW)

P	2.256

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.614
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.902

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.2	0.03938	255.7	0.9296
Compressor outlet	90.1	0.01549	301.2	0.9757
Condenser outlet	31.76	0.0008579	83.82	0.3087
Evaporator inlet	5.03	0.00733	83.82	0.3161

Refrigeration effect (kJ/kg)	171.88
Mas flow rate (gram/sec.)	24.79
Refrigeration capacity (kW)	4.26
Work of compression (kJ/kg)	45.5
Heat rejection (kJ/kg)	217.08
Heat rejection rate (kW)	5.38
Coefficient of performance	3.8
Power consumption (kW)	1.504
Isentropic efficiency %	63.5
Volumetric efficiency %	89.2

Table A1.22 : Data for 960 g of R22 at T_c=41 °C,T_e=5 °C, ΔT(sub)=9 °C, ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8
T2	87
T3	78.12
T4	41.81
T5	41.12
T6	40.56
T7	31.81
T8	4.99
T9	5.03
T10	5.02
T11	5.12
T12	5.16
T13	21.6
T14	32.6

2 . Pressure Reading (psi)

P1	76.2
P2	265.5

3 . Power Reading (kW)

P	2.2425

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.614
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.902

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8	0.03871	253.3	0.9212
Compressor outlet	87	0.01526	298.2	0.9681
Condenser outlet	31.81	0.0008579	83.82	0.3087
Evaporator inlet	5.03	0.00733	83.82	0.3161

Refrigeration effect (kJ/kg)	170.48
Mas flow rate (gram/sec.)	24.95
Refrigeration capacity (kW)	4.25
Work of compression (kJ/kg)	44.9
Heat rejection (kJ/kg)	214.38
Heat rejection rate (kW)	3.35
Coefficient of performance	3.7
Power consumption (kW)	1.495
Isentropic efficiency %	63.25
Volumetric efficiency %	89.24

Table A1.23 : Data for 960 g of R22 at T_c=41 °C ,T_e=5 °C , ΔT(sub)=9 °C, ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.1
T2	94.3
T3	80.23
T4	42.13
T5	41.1
T6	40.36
T7	31.62
T8	4.94
T9	5.09
T10	5.09
T11	5.1
T12	5.14
T13	21.23
T14	32.56

2 . Pressure Reading (psi)

P1	76.2
P2	265.5

3 . Power Reading (kW)

P	2.277

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.614
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.902

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.1	0.0402	258.6	0.9398
Compressor outlet	94.3	0.0158	304.9	0.9858
Condenser outlet	31.62	0.0008579	83.82	0.3087
Evaporator inlet	5.09	0.00733	83.82	0.3161

Refrigeration effect (kJ/kg)	174.78
Mas flow rate (gram/sec.)	24.6
Refrigeration capacity (kW)	4.33
Work of compression (kJ/kg)	46.3
Heat rejection (kJ/kg)	220.78
Heat rejection rate (kW)	5.43
Coefficient of performance	3.8
Power consumption (kW)	1.518
Isentropic efficiency %	64.3
Volumetric efficiency %	89.19

Table A1.24 : Data for 960 g of R22 at T_c=41 °C ,T_e=5 °C , ΔT(sub)=9 °C, ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.1
T2	99.6
T3	81.16
T4	42.2
T5	41.12
T6	40.39
T7	32.1
T8	4.98
T9	5.078
T10	5.08
T11	5.1
T12	5.16
T13	20.65
T14	32.71

2 . Pressure Reading (psi)

P1	76.2
P2	265.5

3 . Power Reading (kW)

P	2.289

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.614
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.902

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.1	0.04144	263.2	0.9553
Compressor outlet	99.6	0.01619	309.9	0.9983
Condenser outlet	32.1	0.0008579	83.82	0.3087
Evaporator inlet	5.078	0.00733	83.82	0.3161

Refrigeration effect (kJ/kg)	179.38
Mas flow rate (gram/sec.)	24.5
Refrigeration capacity (kW)	4.42
Work of compression (kJ/kg)	46.7
Heat rejection (kJ/kg)	224.48
Heat rejection rate (kW)	5.5
Coefficient of performance	4.1
Power consumption (kW)	1.526
Isentropic efficiency %	64.88
Volumetric efficiency %	89.08

Table A1.25 : Data for 960 g of R22 at T_c=38 °C ,T_e=5 C° , ΔT(sub)=9 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8
T2	84.5
T3	74.2
T4	38.82
T5	38.12
T6	37.25
T7	28.87
T8	4.87
T9	4.98
T10	5.02
T11	5.06
T12	5.1
T13	20.43
T14	32.2

2 . Pressure Reading (psi)

P1	69.16
P2	236.2

3 . Power Reading (kW)

P	2.223

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.566
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8	0.04249	254.3	0.9317
Compressor outlet	84.5	0.01726	298.5	0.978
Condenser outlet	28.87	0.0008487	79.85	0.2959
Evaporator inlet	4.98	0.00657	79.85	0.3018

Refrigeration effect (kJ/kg)	174.45
Mas flow rate (gram/sec.)	25.14
Refrigeration capacity (kW)	4.38
Work of compression (kJ/kg)	44.2
Heat rejection (kJ/kg)	218.65
Heat rejection rate (kW)	5.5
Coefficient of performance	3.947
Power consumption (kW)	1.482
Isentropic efficiency %	63.33
Volumetric efficiency %	89.7

Table A1.26 : Data for 960 g of R22 at T_c=38 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.2
T2	86.75
T3	76.62
T4	38.91
T5	38.06
T6	37.23
T7	28.83
T8	4.91
T9	4.96
T10	4.99
T11	5.12
T12	5.16
T13	20.32
T14	32.5

2 . Pressure Reading (psi)

P1	69.16
P2	236.2

3 . Power Reading (kW)

P	2.2395

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.566
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.2	0.04321	256.7	0.94
Compressor outlet	86.75	0.01745	300.4	0.9834
Condenser outlet	28.9	0.0008487	79.85	0.2959
Evaporator inlet	4.96	0.006578	79.85	0.3018

Refrigeration effect (kJ/kg)	176.85
Mas flow rate (gram/sec.)	25.6
Refrigeration capacity (kW)	4.52
Work of compression (kJ/kg)	43.7
Heat rejection (kJ/kg)	220.55
Heat rejection rate (kW)	5.65
Coefficient of performance	4.05
Power consumption (kW)	1.493
Isentropic efficiency %	65.2
Volumetric efficiency %	89.67

Table A1.27 : Data for 960 g of R22 at T_c=38 °C ,T_e=5 C° , ΔT(sub)=9 °C, ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.1
T2	91.2
T3	79.61
T4	38.98
T5	38.06
T6	37.31
T7	28.92
T8	4.94
T9	4.98
T10	5.02
T11	5.12
T12	5.13
T13	20.05
T14	33.1

2 . Pressure Reading (psi)

P1	69.2
P2	236.2

3 . Power Reading (kW)

P	2.2545

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.566
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.1	0.04407	259.5	0.95
Compressor outlet	91.2	0.0178	304.2	0.9939
Condenser outlet	28.92	0.0008487	79.85	0.2959
Evaporator inlet	4.98	0.00657	79.85	0.3018

Refrigeration effect (kJ/kg)	179.65
Mas flow rate (gram/sec.)	25.2
Refrigeration capacity (kW)	4.53
Work of compression (kJ/kg)	44.7
Heat rejection (kJ/kg)	224.35
Heat rejection rate (kW)	5.7
Coefficient of performance	4.02
Power consumption (kW)	1.503
Isentropic efficiency %	65.1
Volumetric efficiency %	89.7

Table A1.28 : Data for 960 g of R22 at T_c=38 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.2
T2	96.3
T3	80.11
T4	38.98
T5	38.09
T6	37.42
T7	29
T8	4.96
T9	4.98
T10	5.07
T11	5.15
T12	5.17
T13	20
T14	33.25

2 . Pressure Reading (psi)

P1	69
P2	236

3 . Power Reading (kW)

P	2.274

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.566
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.2	0.04536	263.9	0.9651
Compressor outlet	96.3	0.0182	309.1	1.006
Condenser outlet	29	0.0008487	79.85	0.2959
Evaporator inlet	4.98	0.00657	79.85	0.3018

Refrigeration effect (kJ/kg)	184.05
Mas flow rate (gram/sec.)	25.1
Refrigeration capacity (kW)	4.62
Work of compression (kJ/kg)	45.2
Heat rejection (kJ/kg)	228.75
Heat rejection rate (kW)	5.74
Coefficient of performance	4.12
Power consumption (kW)	1.516
Isentropic efficiency %	66.37
Volumetric efficiency %	89.56

Table A2.2 : Data for 900 g of R407c at T_c=40 °C , T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.2
T2	85.1
T3	76.72
T4	43.16
T5	39.1
T6	38.1
T7	31.1
T8	0.6
T9	1.3
T10	3.65
T11	4.4
T12	6.63
T13	21.8
T14	34.33

2 . Pressure Reading (psi)

P1	63.5
P2	210

3 . Power Reading (kW)

P	2.2215

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.2	0.04472	422	1.80848
Compressor outlet	85	0.01915	477.1	1.8888
Condenser outlet	31.1	0.0009	247.9	1.1634
Evaporator inlet	0.6	0.0171	247.9	1.2861

Refrigeration effect (kJ/kg)	174.1
Mas flow rate (gram/sec.)	20.16
Refrigeration capacity (kW)	3.51
Work of compression (kJ/kg)	55.1
Heat rejection (kJ/kg)	229.2
Heat rejection rate (kW)	4.65
Coefficient of performance	3.17
Power consumption (kW)	1.481
Isentropic efficiency %	46.18
Volumetric efficiency %	90.65

Table A2.3 : Data for 900 g of R407c at T_c=40 °C , T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15
T2	89
T3	79.56
T4	43.62
T5	39.16
T6	38
T7	30.91
T8	0.57
T9	1.29
T10	3.63
T11	4.39
T12	6.59
T13	20.7
T14	34

2 . Pressure Reading (psi)

P1	63.6
P2	210

3 . Power Reading (kW)

P	2.247

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15	0.0456	425.3	1.8204
Compressor outlet	93	0.01945	477.1	1.8888
Condenser outlet	30.91	0.0009	247.9	1.1634
Evaporator inlet	0.57	0.0171	247.9	1.28611

Refrigeration effect (kJ/kg)	177.4
Mas flow rate (gram/sec.)	20.1
Refrigeration capacity (kW)	3.566
Work of compression (kJ/kg)	55.8
Heat rejection (kJ/kg)	233.2
Heat rejection rate (kW)	4.73
Coefficient of performance	3.18
Power consumption (kW)	1.496
Isentropic efficiency %	47.85
Volumetric efficiency %	90.6

Table A2.4 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.05
T2	94
T3	82.1
T4	43.81
T5	39.24
T6	38.15
T7	31.1
T8	0.61
T9	1.28
T10	3.59
T11	4.41
T12	6.61
T13	20.5
T14	33.85

2 . Pressure Reading (psi)

P1	63.5
P2	210.6

3 . Power Reading (kW)

P	2.2695

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.05	0.04812	430.6	1.84637
Compressor outlet	95	0.01985	487.8	1.9169
Condenser outlet	31.1	0.0009	247.9	1.1634
Evaporator inlet	0.61	0.0171	247.9	1.2861

Refrigeration effect (kJ/kg)	182.7
Mas flow rate (gram/sec.)	19.83
Refrigeration capacity (kW)	3.623
Work of compression (kJ/kg)	57.2
Heat rejection (kJ/kg)	239.9
Heat rejection rate (kW)	4.757
Coefficient of performance	3.194
Power consumption (kW)	1.513
Isentropic efficiency %	47.9
Volumetric efficiency %	90.03

Table A2.5 : Data for 900 g of R407c at T_c=40 °C , T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8.12
T2	82.2
T3	76.16
T4	43
T5	39.18
T6	38
T7	28.5
T8	0.61
T9	1.32
T10	3.62
T11	4.391
T12	6.61
T13	21.6
T14	34.28

2 . Pressure Reading (psi)

P1	63.5
P2	210

3 . Power Reading (kW)

P	2.205

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8.12	0.04646	419.6	1.7993
Compressor outlet	82.2	0.01885	474	1.889
Condenser outlet	28.5	0.0009	243.75	1.15
Evaporator inlet	0.61	0.0107	243.75	1.1836

Refrigeration effect (kJ/kg)	175.85
Mas flow rate (gram/sec.)	20.25
Refrigeration capacity (kW)	3.57
Work of compression (kJ/kg)	54.4
Heat rejection (kJ/kg)	230.29
Heat rejection rate (kW)	4.66
Coefficient of performance	3.25
Power consumption (kW)	1.47
Isentropic efficiency %	47.5
Volumetric efficiency %	89.75

Table A2.6 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=14 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8.25
T2	82.18
T3	76.31
T4	43.26
T5	39
T6	38.16
T7	26.1
T8	0.59
T9	1.33
T10	3.64
T11	4.42
T12	6.52
T13	21.2
T14	34.28

2 . Pressure Reading (psi)

P1	63.5
P2	210.2

3 . Power Reading (kW)

P	2.205

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8.25	0.04646	419.6	1.7993
Compressor outlet	82.18	0.01885	474	1.889
Condenser outlet	26.1	0.0009	239.7	1.1366
Evaporator inlet	0.59	0.0007	239.7	1.0811

Refrigeration effect (kJ/kg)	179.9
Mas flow rate (gram/sec.)	20.25
Refrigeration capacity (kW)	3.64
Work of compression (kJ/kg)	54.4
Heat rejection (kJ/kg)	234.34
Heat rejection rate (kW)	4.75
Coefficient of performance	3.3
Power consumption (kW)	1.47
Isentropic efficiency %	47.5
Volumetric efficiency %	89.75

Table A2.7 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=17 °C , ΔT(sup) =3 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	8.09
T2	82.21
T3	76.15
T4	43.18
T5	39.21
T6	38.09
T7	23.18
T8	0.571
T9	1.29
T10	3.62
T11	4.43
T12	6.58
T13	20.8
T14	34.31

2 . Pressure Reading (psi)

P1	63.55
P2	210.3

3 . Power Reading (kW)

P	2.205

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	8.09	0.04646	419.6	1.7993
Compressor outlet	82.21	0.01885	474	1.889
Condenser outlet	23.18	0.0009	234.8	1.1206
Evaporator inlet	0.571	0.00943	234.8	0.9797

Refrigeration effect (kJ/kg)	184.8
Mas flow rate (gram/sec.)	20.25
Refrigeration capacity (kW)	3.74
Work of compression (kJ/kg)	54.4
Heat rejection (kJ/kg)	239.24
Heat rejection rate (kW)	3.84
Coefficient of performance	3.4
Power consumption (kW)	1.47
Isentropic efficiency %	47.5
Volumetric efficiency %	89.75

Table A2.8 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.1
T2	85.21
T3	76.35
T4	43.23
T5	39.21
T6	38.09
T7	28.5
T8	0.61
T9	1.29
T10	3.64
T11	4.5
T12	6.61
T13	21.6
T14	34.32

2 . Pressure Reading (psi)

P1	63.6
P2	210

3 . Power Reading (kW)

P	2.2215

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.1	0.04472	422	1.80848
Compressor outlet	85.21	0.01915	477.1	1.8888
Condenser outlet	28.5	0.0009	243.75	1.15
Evaporator inlet	0.61	0.0107	243.75	1.1836

Refrigeration effect (kJ/kg)	178.25
Mas flow rate (gram/sec.)	20.16
Refrigeration capacity (kW)	3.6
Work of compression (kJ/kg)	55.1
Heat rejection (kJ/kg)	233.35
Heat rejection rate (kW)	4.7
Coefficient of performance	3.24
Power consumption (kW)	1.481
Isentropic efficiency %	46.18
Volumetric efficiency %	90.65

Table A2.9 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=14 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11
T2	85
T3	76.39
T4	43.18
T5	39.25
T6	38.25
T7	26.1
T8	0.59
T9	1.31
T10	3.63
T11	4.52
T12	6.62
T13	21.25
T14	34.3

2 . Pressure Reading (psi)

P1	63.8
P2	210.5

3 . Power Reading (kW)

P	2.2215

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11	0.04472	422	1.80848
Compressor outlet	85	0.01915	477.1	1.8888
Condenser outlet	26.1	0.0009	239.7	1.1366
Evaporator inlet	0.59	0.01007	239.7	1.0811

Refrigeration effect (kJ/kg)	182.3
Mas flow rate (gram/sec.)	20.16
Refrigeration capacity (kW)	3.67
Work of compression (kJ/kg)	55.1
Heat rejection (kJ/kg)	237.4
Heat rejection rate (kW)	4.8
Coefficient of performance	3.31
Power consumption (kW)	1.481
Isentropic efficiency %	46.18
Volumetric efficiency %	90.65

Table A2.10 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=17 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.25
T2	85.23
T3	76.46
T4	43.31
T5	39.13
T6	38.12
T7	23.21
T8	0.58
T9	1.29
T10	3.63
T11	4.56
T12	6.6
T13	21.1
T14	34.4

2 . Pressure Reading (psi)

P1	63.5
P2	210.5

3 . Power Reading (kW)

P	2.2215

CALCULATIONS AND RESULTS

Process Description

0	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.25	0.04472	422	1.80848
Compressor outlet	85.23	0.01915	477.1	1.8888
Condenser outlet	23.21	0.0009	234.8	1.1206
Evaporator inlet	0.58	0.00943	234.8	0.9797

Refrigeration effect (kJ/kg)	187.2
Mas flow rate (gram/sec.)	20.16
Refrigeration capacity (kW)	3.78
Work of compression (kJ/kg)	55.1
Heat rejection (kJ/kg)	242.3
Heat rejection rate (kW)	4.9
Coefficient of performance	3.4
Power consumption (kW)	1.481
Isentropic efficiency %	46.18
Volumetric efficiency %	90.65

Table A2.11 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.19
T2	89.21
T3	79.62
T4	43.46
T5	39
T6	38.12
T7	28.49
T8	0.56
T9	1.27
T10	3.62
T11	4.41
T12	6.61
T13	20.52
T14	33.9

2 . Pressure Reading (psi)

P1	63.6
P2	211.5

3 . Power Reading (kW)

P	2.247

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.19	0.048	425.3	1.8204
Compressor outlet	89.21	0.01876	481.1	1.90295
Condenser outlet	28.49	0.0009	243.75	1.14995
Evaporator inlet	0.56	0.0107	243.73	1.1836

Refrigeration effect (kJ/kg)	181.55
Mas flow rate (gram/sec.)	20.1
Refrigeration capacity (kW)	3.65
Work of compression (kJ/kg)	55.8
Heat rejection (kJ/kg)	237.35
Heat rejection rate (kW)	4.77
Coefficient of performance	3.254
Power consumption (kW)	1.496
Isentropic efficiency %	47.85
Volumetric efficiency %	90.6

Table A2.12 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=14 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.21
T2	89.31
T3	79.39
T4	43.52
T5	39.14
T6	38.09
T7	26.21
T8	0.6
T9	1.3
T10	3.59
T11	4.44
T12	6.63
T13	20.43
T14	33.86

2 . Pressure Reading (psi)

P1	63.6
P2	210.2

3 . Power Reading (kW)

P	2.247

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.21	0.048	425.3	1.8204
Compressor outlet	89.31	0.01876	481.1	1.90295
Condenser outlet	26.21	0.0009	239.7	1.1366
Evaporator inlet	0.6	0.01007	239.7	1.0811

Refrigeration effect (kJ/kg)	185.6
Mas flow rate (gram/sec.)	20.1
Refrigeration capacity (kW)	3.73
Work of compression (kJ/kg)	55.8
Heat rejection (kJ/kg)	241.4
Heat rejection rate (kW)	4.85
Coefficient of performance	3.326
Power consumption (kW)	1.496
Isentropic efficiency %	47.85
Volumetric efficiency %	90.6

Table A2.13 : Data for 900 g of R407c at T_c=40 °C ,T_e=5 °C , ΔT(sub)=11.5 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.31
T2	95
T3	82.26
T4	43.83
T5	39.31
T6	38.09
T7	28.6
T8	0.591
T9	1.293
T10	3.62
T11	4.38
T12	6.58
T13	20.32
T14	33.63

2 . Pressure Reading (psi)

P1	63.6
P2	210

3 . Power Reading (kW)

P	2.2695

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.54

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.31	0.04932	430.6	1.8686
Compressor outlet	95	0.01995	487.8	1.9169
Condenser outlet	28.6	0.0009	243.75	1.14995
Evaporator inlet	0.591	0.0107	243.75	1.1836

Refrigeration effect (kJ/kg)	186.85
Mas flow rate (gram/sec.)	19.83
Refrigeration capacity (kW)	3.7
Work of compression (kJ/kg)	57.2
Heat rejection (kJ/kg)	244.05
Heat rejection rate (kW)	4.84
Coefficient of performance	3.27
Power consumption (kW)	1.513
Isentropic efficiency %	47.9
Volumetric efficiency %	90.03

Table A2.14 : Data for 900 g of R407c at T_c=40 °C ,T_e=7 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	13.1
T2	83.2
T3	76.1
T4	43.44
T5	38.65
T6	39.86
T7	31.1
T8	1.4
T9	2.23
T10	3.1
T11	8.7
T12	9.5
T13	21.27
T14	32

2 . Pressure Reading (psi)

P1	66.5
P2	204

3 . Power Reading (kW)

P	2.1825

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	7	Suction Pressure (Mpa)	0.55
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.5

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	13.1	0.04516	423.22	1.81444
Compressor outlet	83.2	0.0197	475.66	1.88736
Condenser outlet	31.1	0.0009	247.9	1.1634
Evaporator inlet	1.4	0.01069	247.9	1.1836

Refrigeration effect (kJ/kg)	175.32
Mas flow rate (gram/sec.)	20.8
Refrigeration capacity (kW)	3.647
Work of compression (kJ/kg)	52.44
Heat rejection (kJ/kg)	227.76
Heat rejection rate (kW)	4.74
Coefficient of performance	3.34
Power consumption (kW)	1.455
Isentropic efficiency %	49.73
Volumetric efficiency %	89.7

Table A2.15 : Data for 900 g of R407c at T_c=40 °C ,T_e=7 °C , ΔT(sub)=11.5 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	13.2
T2	83.25
T3	76.21
T4	43.5
T5	38.53
T6	39.78
T7	28.46
T8	1.43
T9	2.2
T10	3.15
T11	8.68
T12	9.57
T13	21.1
T14	32.21

2 . Pressure Reading (psi)

P1	66.5
P2	204.2

3 . Power Reading (kW)

P	2.1825

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	7	Suction Pressure (Mpa)	0.55
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.5

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	13.2	0.04516	423.22	1.81444
Compressor outlet	83.25	0.0197	475.66	1.88736
Condenser outlet	28.46	0.0009	243.75	1.14995
Evaporator inlet	1.43	0.01005	243.75	1.1815

Refrigeration effect (kJ/kg)	179.47
Mas flow rate (gram/sec.)	20.8
Refrigeration capacity (kW)	3.733
Work of compression (kJ/kg)	52.44
Heat rejection (kJ/kg)	231.91
Heat rejection rate (kW)	4.82
Coefficient of performance	3.42
Power consumption (kW)	1.455
Isentropic efficiency %	49.73
Volumetric efficiency %	89.7

Table A2.16 : Data for 900 g of R407c at T_c=40 °C ,T_e=7 °C , ΔT(sub)=14 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	13.3
T2	83.31
T3	76.32
T4	43.46
T5	38.52
T6	39.63
T7	26
T8	1.42
T9	2.13
T10	3.16
T11	8.61
T12	9.48
T13	20.93
T14	32

2 . Pressure Reading (psi)

P1	66.4
P2	204.25

3 . Power Reading (kW)

P	2.1823

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	7	Suction Pressure (Mpa)	0.55
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.5

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	13.3	0.04516	423.22	1.81444
Compressor outlet	83.31	0.0197	475.66	1.88736
Condenser outlet	26.21	0.0009	239.7	1.1366
Evaporator inlet	1.42	0.00941	239.7	1.1794

Refrigeration effect (kJ/kg)	183.53
Mas flow rate (gram/sec.)	20.8
Refrigeration capacity (kW)	3.817
Work of compression (kJ/kg)	52.44
Heat rejection (kJ/kg)	235.96
Heat rejection rate (kW)	4.91
Coefficient of performance	3.5
Power consumption (kW)	1.455
Isentropic efficiency %	49.73
Volumetric efficiency %	89.7

Table A2.17 : Data for 900 g of R407c at T_c=40 °C , T_e=7 °C , ΔT(sub)=17 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	13.25
T2	83.26
T3	76.16
T4	43.51
T5	38.36
T6	39.52
T7	23.28
T8	1.38
T9	2.22
T10	3.21
T11	8.58
T12	9.61
T13	20.62
T14	32

2 . Pressure Reading (psi)

P1	66.6
P2	204.3

3 . Power Reading (kW)

P	2.1825

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	7	Suction Pressure (Mpa)	0.55
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.5

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	13.25	0.04516	423.22	1.81444
Compressor outlet	83.26	0.0197	475.66	1.88736
Condenser outlet	23.18	0.0009	234.8	1.1206
Evaporator inlet	1.38	0.00877	234.8	1.1773

Refrigeration effect (kJ/kg)	188.42
Mas flow rate (gram/sec.)	20.8
Refrigeration capacity (kW)	3.92
Work of compression (kJ/kg)	52.44
Heat rejection (kJ/kg)	240.86
Heat rejection rate (kW)	5.1
Coefficient of performance	3.6
Power consumption (kW)	1.455
Isentropic efficiency %	49.73
Volumetric efficiency %	89.7

Table A2.18 : Data for 900 g of R407c at T_c=40 °C ,T_e=13 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	19.06
T2	80
T3	67.25
T4	40.4
T5	39.67
T6	39.21
T7	30.92
T8	3.5
T9	6.6
T10	11.3
T11	12.97
T12	14.67
T13	21.66
T14	33

2 . Pressure Reading (psi)

P1	73.7
P2	211.5

3 . Power Reading (kW)

P	2.145

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	13	Suction Pressure (Mpa)	0.6
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.55

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	19.06	0.0422	427.62	1.81662
Compressor outlet	80	0.0187	472	1.8745
Condenser outlet	30.92	0.0009	247.9	1.1634
Evaporator inlet	3.5	0.0092	247.9	1.1762

Refrigeration effect (kJ/kg)	179.72
Mas flow rate (gram/sec.)	24.16
Refrigeration capacity (kW)	4.34
Work of compression (kJ/kg)	44.38
Heat rejection (kJ/kg)	224.1
Heat rejection rate (kW)	5.414
Coefficient of performance	4.05
Power consumption (kW)	1.43
Isentropic efficiency %	55.3
Volumetric efficiency %	91.2

Table A2.19 : Data for 900 g of R407c at T_c=40 °C ,T_e=13 °C , ΔT(sub)=11.5 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	19.1
T2	80.23
T3	67.31
T4	40.38
T5	39.62
T6	39.56
T7	28.53
T8	3.491
T9	6.59
T10	11.31
T11	13.03
T12	14.7
T13	21.52
T14	33.13

2 . Pressure Reading (psi)

P1	73.7
P2	211.25

3 . Power Reading (kW)

P	2.145

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	13	Suction Pressure (Mpa)	0.6
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.55

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	19.1	0.0422	427.62	1.81662
Compressor outlet	80.23	0.0187	472	1.8745
Condenser outlet	28.53	0.0009	243.75	1.14995
Evaporator inlet	3.491	0.00856	243.75	1.1741

Refrigeration effect (kJ/kg)	183.82
Mas flow rate (gram/sec.)	24.16
Refrigeration capacity (kW)	4.44
Work of compression (kJ/kg)	44.38
Heat rejection (kJ/kg)	228.25
Heat rejection rate (kW)	5.515
Coefficient of performance	4.143
Power consumption (kW)	1.43
Isentropic efficiency %	55.3
Volumetric efficiency %	91.2

Table A2.20 : Data for 900 g of R407c at T_c=40 °C ,T_e=13 °C , ΔT(sub)=14 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	19.21
T2	80.42
T3	67.42
T4	40.41
T5	39.53
T6	39.62
T7	26.23
T8	3.51
T9	6.62
T10	11.13
T11	13.09
T12	14.69
T13	21.13
T14	33.43

2 . Pressure Reading (psi)

P1	73.5
P2	212

3 . Power Reading (kW)

P	2.145

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	13	Suction Pressure (Mpa)	0.6
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.55

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	19.21	0.0422	427.62	1.81662
Compressor outlet	80.42	0.0187	472	1.8745
Condenser outlet	26.23	0.0009	239.7	1.1366
Evaporator inlet	3.51	0.00792	239.7	1.172

Refrigeration effect (kJ/kg)	187.92
Mas flow rate (gram/sec.)	24.16
Refrigeration capacity (kW)	4.54
Work of compression (kJ/kg)	44.38
Heat rejection (kJ/kg)	232.3
Heat rejection rate (kW)	5.612
Coefficient of performance	4.24
Power consumption (kW)	1.429
Isentropic efficiency %	55.3
Volumetric efficiency %	91.2

Table A2.21 : Data for 900 g of R407c at T_c=40 °C ,T_e=13 °C , ΔT(sub)=17 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	19.25
T2	80.41
T3	67.15
T4	40.36
T5	39.68
T6	39.71
T7	23.16
T8	3.468
T9	6.58
T10	11.29
T11	13.12
T12	14.73
T13	20.62
T14	33.42

2 . Pressure Reading (psi)

P1	73.5
P2	211.7

3 . Power Reading (kW)

P	2.145

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	13	Suction Pressure (Mpa)	0.6
Average Condensing Temp. °C	40	Discharge Pressure (Mpa)	1.55

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	19.25	0.0422	427.62	1.81662
Compressor outlet	80.41	0.0187	472	1.8745
Condenser outlet	23.16	0.0009	234.8	1.1206
Evaporator inlet	3.468	0.00728	234.8	1.1694

Refrigeration effect (kJ/kg)	192.82
Mas flow rate (gram/sec.)	24.16
Refrigeration capacity (kW)	4.66
Work of compression (kJ/kg)	44.38
Heat rejection (kJ/kg)	237.2
Heat rejection rate (kW)	5.73
Coefficient of performance	4.345
Power consumption (kW)	1.43
Isentropic efficiency %	55.3
Volumetric efficiency %	91.2

Table A2.22 : Data for 900 g of R407c at T_c=41 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.16
T2	85.1
T3	76.3
T4	41.9
T5	41.09
T6	40.32
T7	31.89
T8	0.576
T9	1.306
T10	3.55
T11	4.53
T12	6.75
T13	22.31
T14	34.32

2 . Pressure Reading (psi)

P1	63.7
P2	222

3 . Power Reading (kW)

P	2.241

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.6

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.16	0.04712	422	1.8085
Compressor outlet	84	0.0184	476.5	1.8845
Condenser outlet	31.89	0.0009	249.6	1.1688
Evaporator inlet	0.576	0.01143	249.6	1.1827

Refrigeration effect (kJ/kg)	172.4
Mas flow rate (gram/sec.)	20.56
Refrigeration capacity (kW)	3.55
Work of compression (kJ/kg)	54.5
Heat rejection (kJ/kg)	226.9
Heat rejection rate (kW)	4.665
Coefficient of performance	3.164
Power consumption (kW)	1.494
Isentropic efficiency %	53
Volumetric efficiency %	89.62

Table A2.23 : Data for 900 g of R407c at T_c=41 °C , T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.23
T2	89.2
T3	79.91
T4	42.15
T5	41.21
T6	40.21
T7	31.78
T8	0.59
T9	1.31
T10	3.57
T11	4.52
T12	6.78
T13	22.21
T14	34.43

2 . Pressure Reading (psi)

P1	0.53
P2	1.6

3 . Power Reading (kW)

P	2.262

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.6

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.23	0.048	425.3	1.8204
Compressor outlet	89.2	0.0187	480.6	1.8977
Condenser outlet	31.78	0.0009	249.6	1.1688
Evaporator inlet	0.59	0.01143	249.6	1.1827

Refrigeration effect (kJ/kg)	175.7
Mas flow rate (gram/sec.)	20.46
Refrigeration capacity (kW)	3.6
Work of compression (kJ/kg)	55.28
Heat rejection (kJ/kg)	230.98
Heat rejection rate (kW)	4.73
Coefficient of performance	3.18
Power consumption (kW)	1.508
Isentropic efficiency %	50.3
Volumetric efficiency %	89.09

Table A2.24 : Data for 900 g of R407c at T_c=41 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.18
T2	95
T3	82.56
T4	42.2
T5	41.18
T6	40.33
T7	32.1
T8	0.61
T9	1.34
T10	3.56
T11	4.5
T12	7.1
T13	22.06
T14	34.36

2 . Pressure Reading (psi)

P1	63.7
P2	220

3 . Power Reading (kW)

P	2.2905

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	41	Discharge Pressure (Mpa)	1.61

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.18	0.052	430.6	1.8381
Compressor outlet	95	0.0192	486.7	1.9127
Condenser outlet	32.1	0.0009	249.6	1.1688
Evaporator inlet	0.61	0.01143	249.6	1.1827

Refrigeration effect (kJ/kg)	181
Mas flow rate (gram/sec.)	20.4
Refrigeration capacity (kW)	3.7
Work of compression (kJ/kg)	56.1
Heat rejection (kJ/kg)	237.1
Heat rejection rate (kW)	4.84
Coefficient of performance	3.25
Power consumption (kW)	1.527
Isentropic efficiency %	54.5
Volumetric efficiency %	89

Table A2.25 : Data for 900 g of R407c at T_c=38 °C ,T_e=5 °C , $\Delta T(sub)$=9 °C , $\Delta T(sup)$ =6 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	11.05
T2	82.1
T3	76.3
T4	39.78
T5	38.13
T6	37.1
T7	28.92
T8	0.6
T9	1.3
T10	3.66
T11	4.43
T12	6.65
T13	21.8
T14	34.26

2 . Pressure Reading (psi)

P1	63.6
P2	236.2

3 . Power Reading (kW)

P	2.184

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	11.05	0.0426	422	1.80848
Compressor outlet	84	0.01686	472.16	1.8759
Condenser outlet	28.92	0.0009	244.6	1.1526
Evaporator inlet	0.6	0.0112	244.6	1.1834

Refrigeration effect (kJ/kg)	177.4
Mas flow rate (gram/sec.)	21.77
Refrigeration capacity (kW)	3.86
Work of compression (kJ/kg)	50.16
Heat rejection (kJ/kg)	227.56
Heat rejection rate (kW)	4.954
Coefficient of performance	3.536
Power consumption (kW)	1.456
Isentropic efficiency %	58.1
Volumetric efficiency %	92.6

Table A2.26 : Data for 900 g of R407c at T_c=38 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =10 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	15.12
T2	86.2
T3	79.5
T4	40.06
T5	38.2
T6	37
T7	28.78
T8	0.6
T9	1.32
T10	3.67
T11	4.45
T12	6.66
T13	21.61
T14	34.34

2 . Pressure Reading (psi)

P1	63.6
P2	236.2

3 . Power Reading (kW)

P	2.1945

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	15.12	0.048	425.3	1.826
Compressor outlet	89	0.01718	476.3	1.879
Condenser outlet	28.78	0.0009	244.6	1.1526
Evaporator inlet	0.6	0.0112	244.6	1.1634

Refrigeration effect (kJ/kg)	180.7
Mas flow rate (gram/sec.)	21.5
Refrigeration capacity (kW)	3.89
Work of compression (kJ/kg)	51
Heat rejection (kJ/kg)	231.7
Heat rejection rate (kW)	4.98
Coefficient of performance	3.54
Power consumption (kW)	1.463
Isentropic efficiency %	58.7
Volumetric efficiency %	92.43

Table A2.27 : Data for 900 g of R407c at T_c=38 °C ,T_e=5 °C , ΔT(sub)=9 °C , ΔT(sup) =16 °C

EXPERIMENTAL RESULTS

1 . Thermocouple Reading

NO	Temp. °C
T1	21.2
T2	92
T3	84.6
T4	40.13
T5	38.1
T6	37.02
T7	28.91
T8	0.58
T9	1.33
T10	3.7
T11	4.42
T12	6.63
T13	21.43
T14	34.14

2 . Pressure Reading (psi)

P1	63.6
P2	236

3 . Power Reading (kW)

P	2.2125

CALCULATIONS AND RESULTS

Process Description

Average Evaporating Temp. °C	5	Suction Pressure (Mpa)	0.53
Average Condensing Temp. °C	38	Discharge Pressure (Mpa)	1.7

Location	Temp. °C	v(m^3/kg)	h(kJ/kg)	s(kJ/kg.k)
Compressor inlet	21.2	0.04932	429.8	1.8381
Compressor outlet	94	0.01766	482.48	1.9043
Condenser outlet	28.91	0.0009	244.6	1.1526
Evaporator inlet	0.58	0.0112	244.6	1.1834

Refrigeration effect (kJ/kg)	186
Mas flow rate (gram/sec.)	21.3
Refrigeration capacity (kW)	3.96
Work of compression (kJ/kg)	51.88
Heat rejection (kJ/kg)	237.88
Heat rejection rate (kW)	5.07
Coefficient of performance	3.6
Power consumption (kW)	1.475
Isentropic efficiency %	58.2
Volumetric efficiency %	92.1

دراسة أداء وحدة تبريد شباك لتكييف الهواء تستخدم R407c كغاز تبريد بديل لغاز R22 بوجود التسخين الفائق والتبريد الدوني

إعداد

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الملخص

يهدف هذا البحث إلى دراسة تأثيرالتسخين الفائق والتبريد الدوني عند استبدال غازR22بغاز R407c في وحدة شباك لتكييف الهواء قدرة (٥) كيلو واط والحاجة إلى هذا التبديل تأتي من حقيقة أن غازR22 يؤثر على طبقة الأوزون.

دلت نتائج هذا البحث على إمكانية استخدام غاز R407c كغاز تبريد بديل عن غاز R22 في هذا النوع من وحدات التكييف.

وأيضا أشارت نتائج هذا البحث إلى أن معامل الأداء لغاز R407c وغاز R22 يزيد بزيادة درجة التبريد الدوني وزيادة درجة التسخين الفائق أو زيادتهما معا حيث وصلت قيم معامل الأداء إلى 4.35، 5.18 على التوالي عند درجة حرارة مبرد 13°C ودرجة تبريد دوني 17°C ودرجة تسخين فائق 6°C ودرجة حرارة مكثف 40°C.

وهذا يشير إلى أن معامل الأداء لغاز R407c أقل من معامل الأداء لغاز R22 بمقدار 16%. وأيضاً كنتيجة لهذا العمل فإن قدرة التبريد لغاز R407c عند استخدامه بدلا من غاز R22 تنقص من KW (5.6) إلى KW (4.66) عند درجة حرارة مبرد 13°C ودرجة حرارة مكثف 40°C ودرجة تبريد دوني 17°C ودرجة تسخين فائق 6°C.