Performance Study of a Locally Manufactured Domestic Refrigerator When Used as a Heat Pump

By

Mohammad Abdel Rahim Al-Najjar

Supervisor

Prof. Mohammed Al- Sa'ad

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Faculty of Graduate Studies
University of Jordan

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This thesis was successfully defended and approved on May 30th, 2002

Examination committee	**Signature**
Dr. Mohammed Al-Sa'ad, Chairman Prof. Of Mechanical Engineering	
Dr. Mahmoud Hammad, Member Prof. Mechanical Engineering	أ. د. محمود حماد
Dr. Adnan Jradat, Member Assoc. Prof. Mechanical Engineering	Adnan M. Jaradat
Dr. Salem Najmeh, Member Assoc. Prof. Mechanical Engineering	د. سالم نجمة

DEDICATION

TO...

My Father

My Mother

My Sisters and Brothers

My Friends

Who Gave Meaning to
This Effort

With

Love and Respect

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I would like to express my deep sense of gratitude to my supervisor Prof. Mohammed Al-Sa'ad for his valuable guidance, support and encouragement.

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TABLE OF CONTENTS

LIST OF APPENDICES

NOMENCLATURE

COP	Coefficient of Performance
C_p	Specific heat (kJ/kg °C)
h	Enthalpy (kJ/kg)
M	Mass (kg)
m`	Refrigerant mass flow-rate (kg/s)
P	Pressure (bar)
q	Heating effect (kJ/kg)
Q	Heating capacity(Watt)
T	Temperature (°C)
t	Time (s)
w	Compression work (kJ/kg)
W	Compressor power consumption (Watt)

Subscripts

a	Ambient
c	Condenser
e	Evaporator
co	Container
w	Water load
air	Air inside the freezer zone
Al	aluminum

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ABSTRACT

The heating performance of a heat pump system was studied experimentally, by reversing the compressor connections of a locally manufactured domestic refrigerator. This changes the freezer cabinet of the refrigerator into an oven with high coefficient of performance.

Two experimental tests were carried out; the condenser temperature variation test and the evaporator temperature variation test. A number of different performance curves were presented for a range of condensing and evaporating temperatures.

From the experimental results of the performance tests, it was concluded that heat pump is suitable for food heating and keeping it hot. The maximum air temperature attained inside the freezer cabinet was 54.6 °C. This occurred when the condensing temperature reached 77.5 °C and the evaporating temperature was 19.4 °C.

Also the results showed a coefficient of performance of 3.2 at a condensing temperature of 75.4 °C and an evaporating temperature of 13.7 °C.

Since the coefficient of performance of the heat pump is higher than that of the refrigerator by one, thus the heat energy is higher than the electrical input energy. This leads to reduction in energy demand and consequently energy generation.

Chapter One

INTRODUCTION

As fuel prices keep changing and energy reserves keep decreasing, efficient methods of energy conservation and utilization should be used. Heat pump is a good example where thermal energy is extracted from outdoor air in order to raise the temperature of indoor air. It can be likened to a concentration process, in which the existing diluted energy in the outdoor air is collected and concentrated by the heat pump.

The heat pump is a relatively efficient device. It can upgrade low grade heat, e.g. at 5 °C to a useable level (50 °C or more). For every kilowatt of electrical energy used by the compressor 2 to 3 kW are delivered at the condenser. If low associated temperatures are employed, heat pumps can be used in different applications, such as space heating, drying and distillation. If simultaneous heating and cooling are utilized in some applications such as hotels, cold stores computer rooms and laundries. Then an energy conversion factor of around 250% can be achieved. In distillation columns, the heat pump can reduce by 15-30% the energy required to boil a liquid. (Najjar, 2001)

1.1 Vapor – compression cycle

The vapor-compression cycle is the most widely used refrigeration cycle in practice. The thermodynamic cycle of both heat pump and the refrigerator is the same. The processes, which comprise the standard (ideal) vapor-compression cycle, are: -

1. Reversible adiabatic compression from saturated vapor to the condenser pressure.
2. Rejection of heat at constant pressure and condensation to saturated liquid.
3. Irreversible expansion at constant enthalpy from saturated liquid to the evaporator pressure.
4. Addition of heat at constant pressure in the evaporator to saturated vapor.

1.2 Working fluids for vapor – compression system

A large number of different working fluids (refrigerants) are utilized in vapor-compression refrigeration "heat pump" systems. Ammonia and sulfur dioxide were important in the early days of vapor-compression refrigeration, but both are highly toxic and therefore dangerous substances. For many years now, the principal refrigerants have been the halogenated hydrocarbons, which are marketed under the trade names of

Freon and Genatron. For example, dichlorodifluoromethan (CCl_2F_2) is known as Freon-12, and therefore as refrigerant-12 or R-12. This group of substances, known commonly as chlorofluorocarbons (CFCs), are chemically very stable at ambient temperature, especially those lacking any hydrogen atoms. This characteristic is necessary for a refrigerant working fluid. This same characteristic, however, has devastating consequences if the gas, having leaked from an appliance into the atmosphere, spends many years slowly diffusing upward into stratosphere. There it is broken down, releasing chlorine, which destroys the protective ozone layer of the stratosphere. It is therefore of overwhelming importance to us all to eliminate completely the widely used but life-threatening CFCs, particularly R-11 and R-12, and to develop suitable and acceptable replacements. The CFCs containing hydrogen, hydrochlorofluorocarbons (HCFCs), such as R-22, have shorter atmospheric lifetimes, and therefore are not as likely to reach the stratosphere before being broken up and rendered harmless. The most desirable fluids, hydrofluorocarbons (HFCs), contain no chlorine atoms at all.

1.3 Deviation of the actual vapor compression cycle from the ideal cycle

The actual heat pump cycle deviates from the ideal cycle primarily because of pressure drops associated with fluid flow and heat transfer to or from the surroundings.

a- In practice a degree of superheating at the evaporator outlet is required to ensure that minimum amount of liquid droplets are being carried over into the compressor.

b- Actual compression process is not isentropic because of the losses due to friction and heat transfer to refrigerant and surroundings.

c- The pressure of the liquid leaving the condenser will be less than the pressure of the vapor entering, and the temperature of the refrigerant in the condenser will be somewhat higher than that of the surroundings to which heat is being transferred. Usually the temperature of the liquid leaving the condenser is lower than the saturation temperature.

d- Pressure drop of the refrigerant as it flows in the condenser, evaporator, interconnecting piping and through the valves and passages of the compressor is mainly due to friction, momentum change, liquid vapor stratification and by spring-loading of compressor suction and discharge valves during their opening.

Chapter Two

LITERATURE SURVEY

To help solving the problems of global environmental protection by reducing the CO_2 emission and saving the primary energy, it is necessary to achieve a society that is even more fuel efficient. To achieve this goal, in addition to adapt measures on the energy supply side, such as improving the efficiency of power generation facilities, it is essential that measures be adopted on the demand side. Attention is now being directed towards using heat pumps as an energy-saving technology on the demand side. Researchers are currently developing and promoting the technology aspects of heat pumps.

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

Kumar et al. (1984) studied the performance of heat pump with R-22, R-12, and R-11 as working fluids. The R-12 was the most suitable for achieving simultaneous heating and dehumidification of air. Experiments were carried out in an open tunnel on a heat pump using a compressor, hermetically sealed, window air conditioner compressor charged with R-12. The effects of air temperature, air flow rate, and evaporating pressure were studied. Actual performance factors in the range of 1-4 were achieved.

Cox-Smith and Carrington (1990) examined the performance of the heat exchangers of an unsophisticated air – air heat pump with particular emphasis on the evaporator frosting characteristics. The data relate to an operating domestic heat pump in Dunedin. Evaporator frosting reduces the seasonal COP by only a small amount (less than 1.7%) in this example but the defrost rate is shown to be very sensitive to small changes in the evaporator capacity. Thus frost formation generally has an important influence on the evaporator optimization calculation. The effectiveness of the expansion valve and of the defrost thermostat is also analyzed, and particular opportunities for improving the system are identified.

Okken et al. (1993) discussed the opportunities for heat pumps to reduce emission in competition with other CO_2 reducing options. The study was expected costs and efficiencies of electrical, gas-absorption and gas-compression heat pumps in the industrial, residential and commercial sectors. They concluded that for space heating in the residential sector in the Netherlands, gas heating in combination with thermal insulation remains the best option up to the year 2030 and beyond. Only at severe

CO_2 constraints (80% emission reduction) or in case of reduced investment costs, electric heat pumps become competitive.

Silva and Rosa (1993) emphasized the potential of the heat pumps in combined heating and refrigeration, or combined heating and dehumidification applications. As well as in other process integration approaches for efficient energy use in industry and buildings. This work showed that heat pumps are of paramount interest in heat recovery and process integration in several industrial sectors and in the residential/building sector. A broad range of applications are not at all or sufficiently acknowledged and exploited in view of the technical and economical potentialities and promise.

Linton and Snelson (1993) provided an up-to-date review of high temperature heat pump technology from a global perspective. For the purpose of this survey high temperature heat pumps were considered to be those having output temperatures greater than 80 °C. They concluded that high temperature heat pumps are expected to find increased application in the future to help reduce the emission of greenhouse gases from industry.

Hammad and Abo Gharbia (1993) developed a mathematical model to simulate a performance of heat pump. The mathematical model was processed in a microcomputer using "FORTRAN" and results of different working conditions were obtained.

Pelletier et al. (1996) studied the heat-transfer and pressure drop characteristics of propane and R-22 during boiling and condensation in brazed plate heat exchangers and their compressor performances have been studied. Two plate heat exchangers have been tested, both as condensers and as evaporators, in a test rig simulating a small domestic heat pump. The two heat exchangers had the same external geometry and the same number of plates, but different angles of the V-shaped pattern of the plates. The overall heat transfer coefficients were calculated from the mass flow and the enthalpy change of the refrigerant, and from the LMTD (logarithmic mean temperature difference) between the refrigerant and the secondary refrigerant/water. The results show that, at equal heating and cooling load, the overall heat transfer coefficients are slightly higher for propane than for R-22 in the evaporator, but slightly lower in the condenser. However, the pressure drop is about half with propane in both condenser and evaporator. It is therefore believed that with a new design of the heat exchangers, giving equal pressure drop at equal load, the heat transfer coefficients will be higher for propane than for R-22 in both condenser and evaporator. Using the same compressor for the two refrigerants, it was found that the heating capacity was lower and the COP1 higher with propane than with R-22.

Saikawa and Hasegawa (1997) developed commercial two-stage compression-type heat pump with a view to achieving energy efficiency in the field of hot-water production. They concluded that the two-stage compression-type heat pump is superior to the conventional gas-fired system in saving primary energy and reduction of CO_2 emission, and also became evident that it's economic efficiency is competitive.

Rieberer et al. (1999) discussed CO_2 as gained renewed interest as refrigerant. It is a natural substance with no negative environmental impact and it is also non-flammable and non-toxic. The properties of this refrigerant are very favorable for heat pump applications when heating is linked to a certain temperature increase. Thus, it is a good alternative to the synthetic working fluids in many applications. A very promising application for CO_2 heat pumps is tap water heating. System testing with a prototype has shown that for an Oslo climate a Seasonal Performance Factor of 4 can be achieved when water is heated from 8 to 60 °C and ambient air is used as heat source. Further investigations have shown that CO_2 heat pumps are very suitable for space heating Potential applications that have been investigated are heating of sport facilities, low-heating-energy buildings, and commercial buildings. CO_2 allows the combination of space heating and water heating as well.

Gomri (2001) studied the entropic and exergetic analysises based on conbined first and second laws of thermodynamics applied to solar-assisted heat pump systems are presented. A relation between the total production of irreversibilities and exergy destruction in heat pump is derived. Thus this kilometers of pipes designed in loops that will ensure adequate circulation of water. This results in a substantial heat exchanger with great potential. Indirect heat transfer occurs between the refrigerant and ground via the municipality water reticulation system that acts as the water-to-ground heat exchanger. The experimental and simulated comparisons of the ground-source system to the air-source system are conducted in both the cooling and the heating cycles. Climatology statistics are used to calculate the capacities and coefficients of performance of the ground-source and air-source heat pumps. Results obtained from measurements and simulations indicate that the utilization of municipality water reticulation systems as a heat source/sink is a viable method of optimizing energy usage in the air conditioning industry, esp. when used in the heating mode.

Najjar (2001) reviewed three research investigations, carried out by the author and associates. They covered engine-assisted heating and cooling systems both vapor compression and absorption, associated with gas turbine and internal combustion engines.

Primary energy is utilized 100% more efficiently in engine driven heat pumps. The superiority of the gas turbine system is quite clear.

Tetsuya and Nobuhiro (2002) arranged an auxiliary heat exchanger in a heat pump between outlet of an evaporator and inlet of a compressor.

Chapter Three

EXPERIMANTAL WORK PROCEDURE

3.1 Introduction

A locally manufactured domestic refrigerator when used as a heat pump was tested in this research. By reversing the compressor connection in the refrigerator, this changes the refrigerator into an oven. The same refrigerant (R-12) is used and performance tests were carried out.

This chapter describes the test procedure and points out the heat pump preparation and instrumentation.

3.2 Refrigerator unit specifications

The refrigerator used in this research is a simple domestic refrigerator that contains two separate compartments. It dose not include defrosting device or forced air circulation. The specifications of the refrigerator denoted by the manufacturer are listed in table 4.1.

Table 3.1 Specification of the refrigerator used in this research

Trade mark	MISTRAL
Manufacturer	HAMCO (Household Appliances Manufacturing Co., Jordan)
Gross capacity	320 L
Freezer storage capacity	65 L
Refrigerant R-12 charge mass	0.21 kg
Nominal input power	150 watt
Nominal current and voltage	1.2 A and 220 Volts
Evaporator temperature range	- 10 to - 40 °C
Compressor design	Reciprocating (hermetically)
Compressor displacement size	8cc
Capillary tube diameter	0.6 mm
Capillary tube length	2.45m

3.3 Measuring Devices

The variables that were measured during the experiments are temperature, pressure, power consumption, time, and mass of water and refrigerant.

3.3.1 Temperature measurement

Copper-constantan thermocouples were used to measure the temperatures of the system. They were connected to ten points in the refrigerator as follows:

1. The outlet of the compressor (beginning of discharge line), T_1. Since the discharge line is insulated the temperature at the inlet of the condenser is equal to the discharge compressor temperature, T_1.
2. The condenser in the freezer cabinet, T_2 , T_3 , and T_4, the average of these is the condensing temperature, T_c.
3. The outlet of the condenser, T_5.
4. The middle and the end of the capillary tube, T_6 and T_7.
5. The evaporator, T_8 and T_9 , the average of these is the evaporating temperature, T_e.
6. The inlet to the compressor, T_{10}.
7. The ambient temperature, T_a.

These points are shown on the schematic diagram, Figure 3.1.

For the load temperature test:

8. Inside the water load, T_w.
9. Space (air) temperature in the freezer compartment, T_{air}.

3.3.2 Pressure measurement

The pressure was measured using a gage manifold, which is comprised of compound gage, a pressure gage, and the valve manifold. The compound gage is used to measure pressure both above and below atmospheric (vacuum). In practice it is used to determine pressure in the low side of the refrigeration cycle and was connected to the suction line. The pressure gage, which is used to determine pressure in the high side of the system, was connected to the discharge line.

3.3.3 Input power consumption measurement

To measure the actual power consumed by the motor of the compressor, a Clamp-meter and a Voltmeter were used to measure the current and voltage, respectively.

3.3.4 Time

For the simulated load experiments, readings were taken for time. Measuring time intervals is important for calculating the rate of heat added to the load.

3.3.5 Mass

A digital scale was used to weight the charge of the used refrigerant, and to weight the load of ice. The digital scale range (0 – 4 kg) and has an accuracy of one gram was used.

in the system, and then charged with a specified quantity of the refrigerant (0.21 kg of R-12).

3.4.2 Experimental work

The following tests were carried out:

A- Condensing temperature variation test

In order to perform this experiment, a simulated load which consists of a metal container of known specific heat and mass (0.529 kg of steel) filled with a specified quantity of cold water (3.2 kg of water at 3 °C) was placed in the freezer compartment. After connecting thermocouples to the container and the water, it was placed in the freezer compartment – near the condenser. In order to keep the evaporator temperature constant, a cold water was sprayed over the evaporator.

During the period of condensing temperature (T_c) variation test, temperatures at all the locations, pressure, and time were recorded. The state at each point in the heat pump is determined from the measured data (see figure 3.2 and 3.3).

B- Evaporating temperature variation test

In order to decrease the values of the evaporator temperatures, a cold water was sprayed over the evaporator. Also a load of ice and forced air stream using a small fan was introduced over the condenser to keep its temperature constant.

During the period of T_e variation test, the temperature readings, pressure and time were recorded.

C- Heating rate test

Finally the heating rate test was carried out to get the time variation of the load temperature.

In order to do that a simulated load, which consists of a metal container (0.529 kg of steel) with 2.640 kg of ice at 0 °C was placed in the frozen food compartment and all temperature and pressure data were recorded.

For R-12, temperature and pressure readings are needed to specify the state at any location in the cycle and calculate the enthalpy using R-12 tables.

In the first step, the following assumptions were made to determine the enthalpy at each location in the heat pump cycle.

a. At the exit of the condenser the temperature was recorded, and the refrigerant is assumed to be saturated liquid at this state.

b. At the inlet of the evaporator the enthalpy equals to the enthalpy at the exit of the condenser. Because of expansion takes place at constant enthalpy from saturated liquid to the evaporator pressure.

The enthalpies of suction and discharge locations of the compressor were determined by the measured data at the locations.

4.2.2 Heating effect

As the refrigerant flows through the condenser it rejects heat to the surrounding air. The quantity of heat that is rejected in kJ per kg of refrigerant circulated in this process is known as the heating effect, q.

The net heating effect depends upon the temperature and pressure at which the refrigerant liquid leaving the condenser and the temperature and pressure of the refrigerant discharged from the compressor. Therefore:

$$q = h_1 - h_5 \qquad (4.1)$$

Heating capacity is calculated by measuring the rate of heat rejected by the condenser to a simulated load (which consists of a metal container filled with cold water) in the freezer zone, using the following equation:

$$Q=(M_w\, Cp_w\, \Delta T_w+M_{co}\, Cp_{co}\, \Delta T_{co}+M_{Al}\, Cp_{Al}\, \Delta T_{Al}+M_a\, Cp_a\, \Delta T_a)/\, \Delta t \quad (4.4)$$

Where, M_w, M_{co}, M_{Al} and M_a are the masses of the water, container, aluminum freezer and air inside the freezer zone in kg.

Cp_w, Cp_{co}, Cp_{Al} and Cp_a are the specific heats of the water, container, aluminum freezer and air in kJ/kg °C.

ΔT_w, ΔT_{co}, ΔT_{Al} and ΔT_a are the temperature differences in °C of water, container, aluminum freezer and air during the time period Δt (in seconds).

4.2.5 Mass flow rates of the refrigerant

The mass flow rate of refrigerant is defined as the mass of the refrigerant, which must be circulated per second for any operating condition. It is calculated from the following equation:

$$m` = Q\, /\, q \qquad (4.5)$$

4.2.6 Compressor power consumption

The power required by the compressor is the product of refrigerant mass rate of flow and the increase in the enthalpy during the compression process. It is given by:

$$W = \dot{m}\, w \qquad (4.6)$$

Where, W is the compressor power consumption in kW.

4.2.7 Coefficient of performance

The coefficient of performance (COP) of heat pump system is an expression of the efficiency of the system. It is obtained by dividing the heating capacity over the compressor power consumption (or the heating effect over the compression work).

$$COP = Q / W = q / w = (h_1 - h_5) / (h_1 - h_{10}) \qquad (4.7)$$

4.3 Sample of calculation

A sample of calculation is presented using one set of readings from the data tables of Appendix A. The readings needed to perform the sample of calculation are listed in table 4.1.

Table 4.1 Sample data readings

Readings (units)	Symbols	Measured values
Suction pressure (bar)	P_{in}	1.6
Discharge pressure (bar)	P_{out}	16.5
Outlet temperature from the compressor (°C)	T_1	92.3
Average condensing temperature (°C)	T_c	62.3
Condenser temperature at the exit (°C)	T_5	53.9
Average evaporator temperature (°C)	T_e	8.0
Inlet temperature to the compressor (°C)	T_{10}	0.4
The temperature difference of the water load (°C)	ΔT_{co}	1.5
The temperature difference of the container (°C)	ΔT_w	1.8
The temperature difference of aluminum freezer (°C)	ΔT_{Al}	6.3
The temperature difference of the air in freezer (°C)	ΔT_a	4.7
The time period during the test (second)	Δt	45

For the two states at the inlet and outlet of the compressor, the suction and discharge temperatures and pressures are used to get the enthalpies from R-12 tables.

The enthalpy of the refrigerant at the condenser inlet was taken to be equal to that at the outlet of compressor.

At the outlet of the condenser, the refrigerant was assumed as saturated liquid.

The enthalpy of the refrigerant in the evaporator inlet was taken to be equal to that at the outlet of condenser (due to the adiabatic throttling process).

After finding from tables the enthalpies of the refrigerant (R-12) at each state of the heat pump cycle (states 1, 5, 7 and 10). The system performance parameters were obtained using equations (4.1) through (4.7).

From equation (4.1), the heating effect, q, equal to:

$$\mathbf{q = h_1 - h_5 = 233.6 - 89.09 = 144.5\ kJ/kg}$$

From equation (4.2) the compression work, w, equal to:

$$\mathbf{w = h_1 - h_{10} = 233.6 - 188.8 = 44.8\ kJ/kg}$$

To find the heating capacity, Q, equation (4.4) is used as follows:

$$\mathbf{Q = (M_w\ Cp_w\ \Delta T_w + M_{co}\ Cp_{co}\ \Delta T_{co} + M_{Al}\ Cp_{Al}\ \Delta T_{Al} + M_a\ Cp_a\ \Delta T_a) / \Delta t}$$

$$\mathbf{= (3.2*4.18*1.5+0.529*0.434*1.8+0.8*0.903*6.3+0.0755*1.007*4.7)/45}$$

$$\mathbf{= 564\ Watt}$$

It can be noticed that $M_{co}\ Cp_{co}\ \Delta T_{co} / \Delta t$ represent 1.6%, and $M_a\ Cp_a\ \Delta T_a / \Delta t$ represent 1.4% of the heating capacity, and thus can be neglected.

To calculate the refrigerant mass flow rate, m\`, equation (4.5) is used:

$$\mathbf{m` = Q/q = 564.1277/144.51 = 0.0039\ kg/s}$$

From equation (4.6) compression power consumption equal to:

$$\mathbf{W = m`\ w = 0.00390 * 44.8 = 175\ Watt}$$

From equation (4.7) the coefficient of performance equal to:

$$\mathbf{COP = q / w = 144.5 / 44.8 = 3.2}$$

Chapter Five

RESULTS AND DISCUSSION

5.1 Introduction

The discussion of the results of this research is divided into two main parts. First, variations of the performance parameters of the domestic refrigerator when used as a heat pump with condensing temperatures at a constant evaporating temperature.

Second, variations of the performance parameters of the domestic refrigerator when used as a heat pump, with evaporating temperatures at a constant condensing temperature.

The performance parameters studied are the heating effect, compression work, heating capacity, refrigerant mass flow rate, compression power consumption, and coefficient of performance.

5.2 Heat pump

Heat pumps are considered part of energy conservation devices, worldwide. Due to having a coefficient of performance higher than one, the used energy will be higher than the input energy. This leads to reduction in energy demand and consequently energy generation.

The heat pump is more efficient in converting electrical energy to heat energy than the resistance heater. In resistance heating the performance factor is one since the power derived from heating is the same as the electrical power provided to the heater.

5.3 Performance variation with the condensing temperature

Variation of the performance parameters with the condensing temperature T_c, are presented for two values of evaporating temperature 8 and 15 °C, and ambient temperature T_a of 30.7 °C.

The performance parameters studied are the heating effect, compression work, heating capacity, refrigerant mass flow rate, compression power consumption, and coefficient of performance.

All T_c variation test data and results are presented in Appendix A and the results of this testing are presented graphically in figures 5.1 to 5.6.

5.3.1 Heating effect

Figure 5.1 shows the heating effect, which is the heat rejected from the heat pump, versus T_c. For two values of constant evaporating temperature, the heating effect increases with increasing the condensing temperature, because increasing T_c causes increase in the enthalpy difference of the refrigerant across the condenser. This will cause an increase in the heat rejected in the condenser, and thus increasing the heating effect of the heat pump.

5.3.3 Heating capacity

The heating capacity was calculated by measuring the rate of heat rejected in the "freezer compartment".

Variation of the heating capacity with the condensing temperatures is presented in figure 5.3. It is noticed that for constant evaporating temperature, the heating capacity decreases with increasing the condensing temperature.

The heating capacity equal to the refrigerant mass flow rate multiplied by the enthalpy difference across the condenser. Increasing T_c will cause enthalpy difference across the condenser to increase, but the mass flow rate will decrease and have more effect on the heating capacity value, so the heating capacity will decrease with increasing condensing temperature.

Also figure 5.3 shows that the heating capacity at T_e= 8°C decreases than that at T_e = 15°C by 6.456 Watt per 1 °C of T_e. The heating capacity decreases with decreasing T_e, due to the higher rate of decreasing in the mass flow rate than the increase in the heating effect, at a given value of condensing temperature.

of the work compression with increasing T_c, thus the compression power consumption decreases, with increasing T_c.

Figure 5.5 also shows that the compression power at T_e= 8°C and T_e= 15 °C, depend on which have the higher effect on the power value, the work which decrease with increasing T_e, or the mass flow rate which increase with increasing T_e.

5.3.6 Coefficient of performance

Variations of the coefficient of performance with the condensing temperature are presented in figure 5.6.

As the condensing temperature increases, at a constant evaporating temperature, the coefficient of performance (COP) will decrease, since the enthalpy difference across the compressor will increase at a higher rate than the enthalpy difference increase across the condenser.

The figure also shows that at T_e = 8°C, COP slightly lower than that at T_e = 15°C, COP decreases by 0.048 per 1°C of T_e decreasing. That is due to, at a lower value of T_e, the enthalpy difference across the compressor will increase and have more effect than that increase across the condenser.

because the increasing in the enthalpy of the refrigerant at the outlet of the condenser h_5 at T_c=75°C, this decrease the enthalpy difference across the condenser, although the increase of the enthalpy at the inlet of the condenser. Thus decreases the heating effect, with increasing the condensing temperature.

5.4.2 Compression work

The compression work is plotted against T_e for two values of T_c, as shown in figure 5.8. As the evaporating temperature increase, at constant condensing temperature, the work will decrease due to that the pressure and temperature (and thus the compressor suction enthalpy of the refrigerant) will increase while keeping the discharge enthalpy constant. Therefore, this will cause the compression work to decrease as T_e increase, since the compression work equal to the enthalpy difference across the compressor.

The figure also shows that at T_c = 75°C, the compression work increases than that at T_c = 60 °C, by 0.598 kJ/kg per 1°C of Tc, for the same T_e. The compression work increases due to the increase in the enthalpy difference across the compressor by increasing the enthalpy at the outlet of the compressor with increasing the condensing temperature.

5.4.4 Refrigerant mass flow rates

Increasing the evaporating temperature, at a constant condensing temperature, will increase the refrigerant mass flow rate. The refrigerant mass flow rate is inversely proportional to the specific volume, as T_e increase, the pressure and temperature of the compressor suction are increased, thus decrease the specific volume of the refrigerant, which causes the mass flow rate to increase with increasing the evaporating temperature for constant T_c.

The refrigerant mass flow rate at T_c= 75 °C decreases than at T_c = 60 °C, by 0.00006 kg/s per 1°C of T_c. The refrigerant mass flow rate decreases due to the increasing in the specific volume of the refrigerant with increasing T_c for the same T_e, see paragraph 5.3.4.

5.4.5 Compression power consumption

Compressor power variation with evaporating temperature is shown in figure 5.10.

According to equation (4.6) the compressor power equal to the work of compression multiplied by the refrigerant mass flow rate. Since the work of compression decreases at a rate higher than the increase rate of the mass flow rate with increasing T_e, so that the power consumption decreases.

5.5 Load temperature test

All data and results of the load temperature test are presented in Appendix C.

Variation of the load temperature with time when placing a load in the "frozen food compartment which include the condenser" is presented in figure 5.12. In order to perform this a simulated load, which consists of a metal container of known specific heat and mass (0.529 kg of steel) with 2.640 kg of ice at 0 °C. The load was placed in the frozen food compartment. Figure 5.13, shows the heating curve that indicates that 65 minutes is need to convert all the ice to liquid - latent heat fusion for ice 233 kJ/kg -.

Figure 5.13 shows the condensing temperatures response to load during the operation period. After placing the load, Tc increasing with time.

The relation between the ambient temperature air inside the frozen food compartment, T_{air} and the time is shown in figure 5.14.

Figure 5.15 shows the coefficient of performance for the heat pump and the refrigerator, it is clearly that the coefficient of performance of the heat pump equal to the coefficient of performance of the refrigerator plus one.

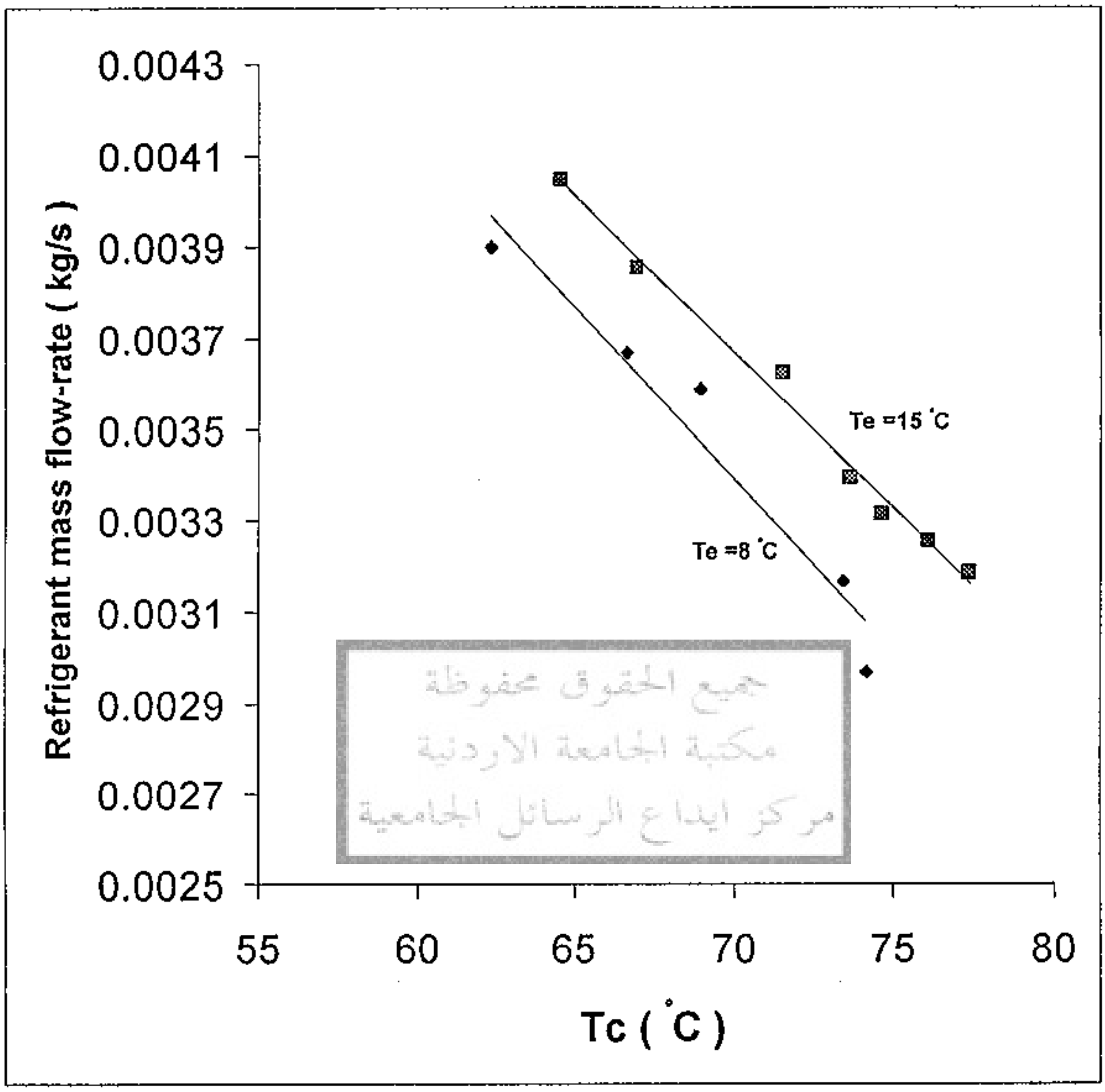

Figure 5.4 Refrigerant mass flow rate versus T_c

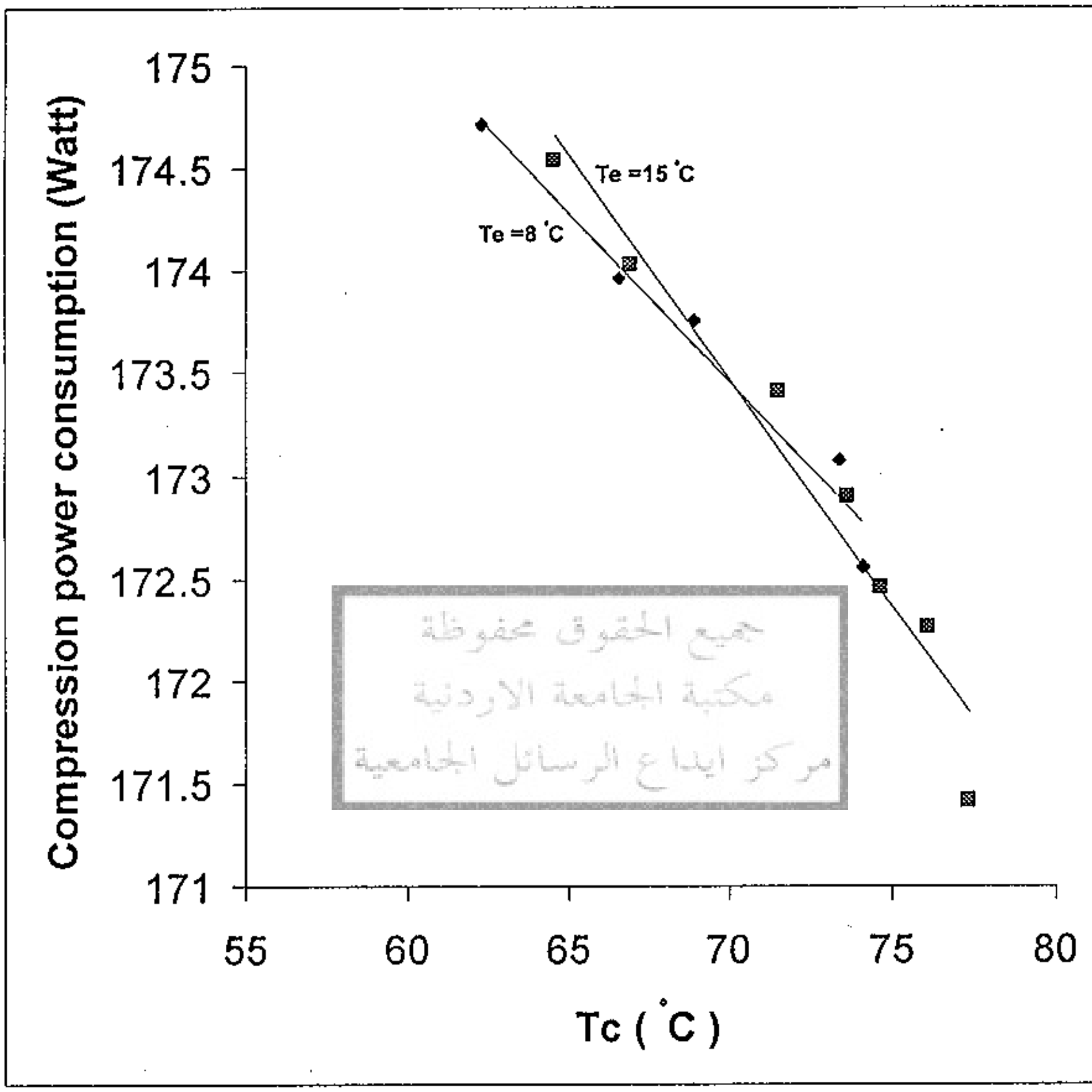

Figure 5.5 Compression power consumption versus T_c

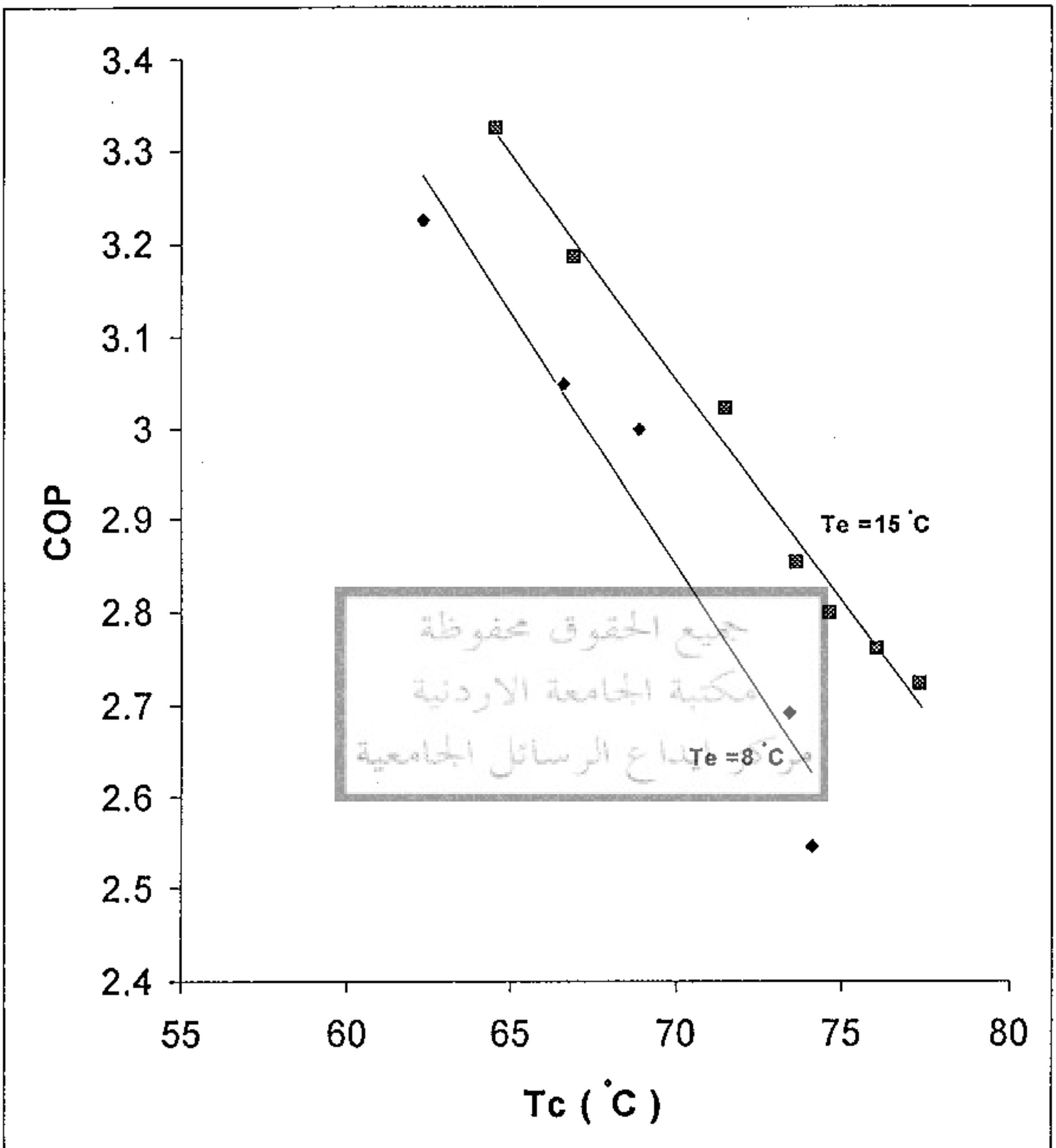

Figure 5.6 Coefficient of performance versus T_c

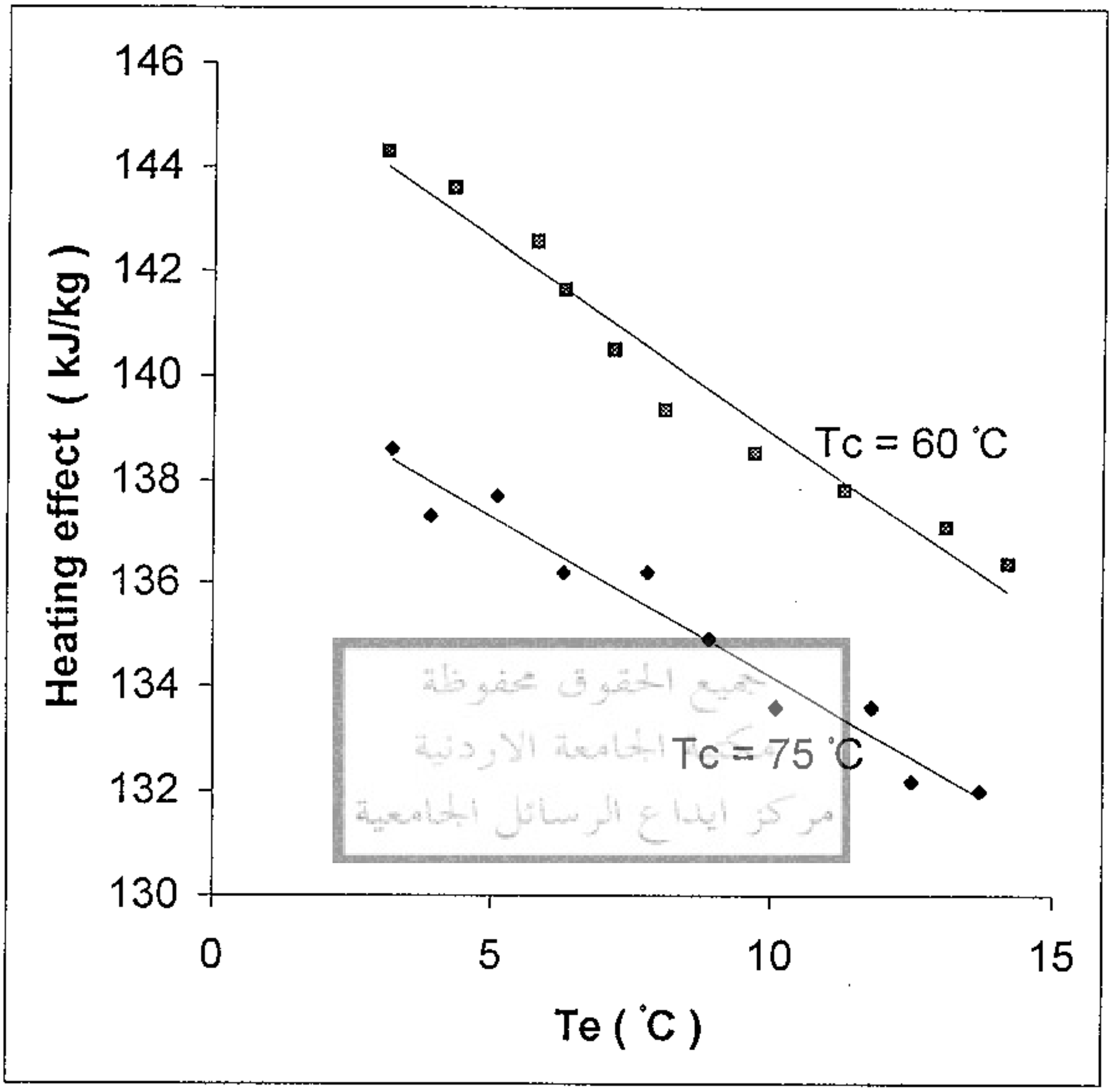

Figure 5.7 Heating effect versus T_e

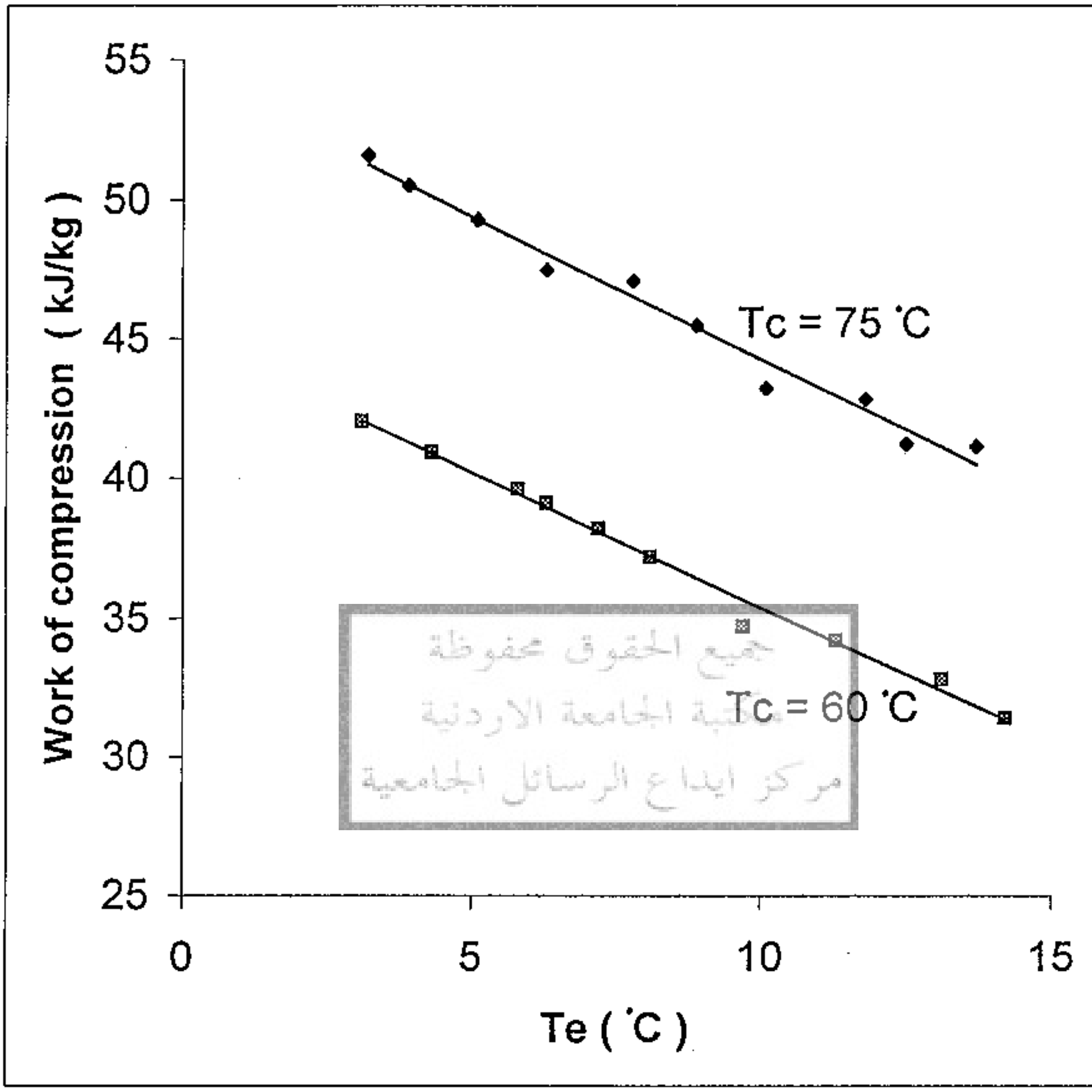

Figure 5.8 Work of compression versus T_e

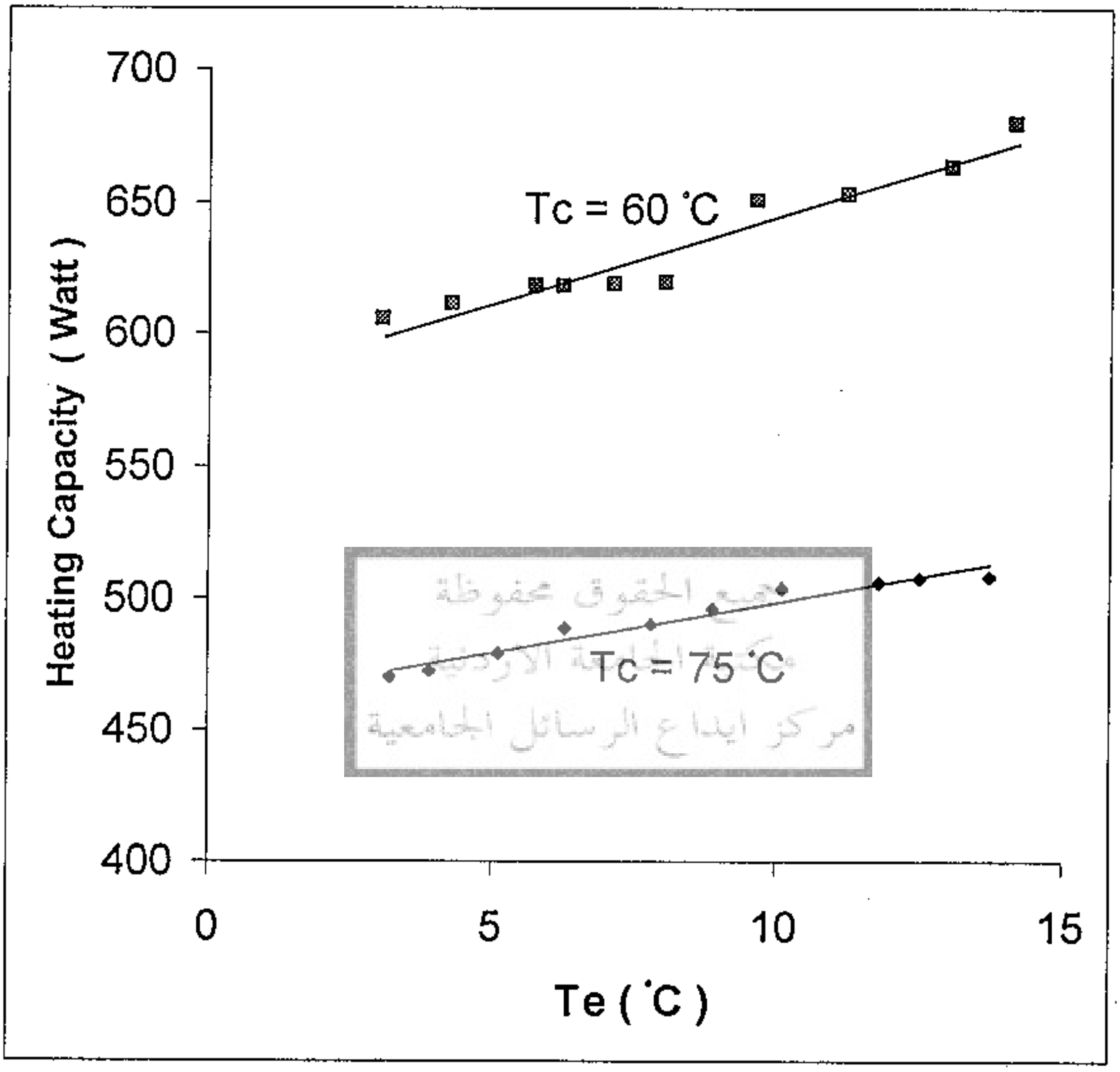

Figure 5.9 Heating capacity versus T_e

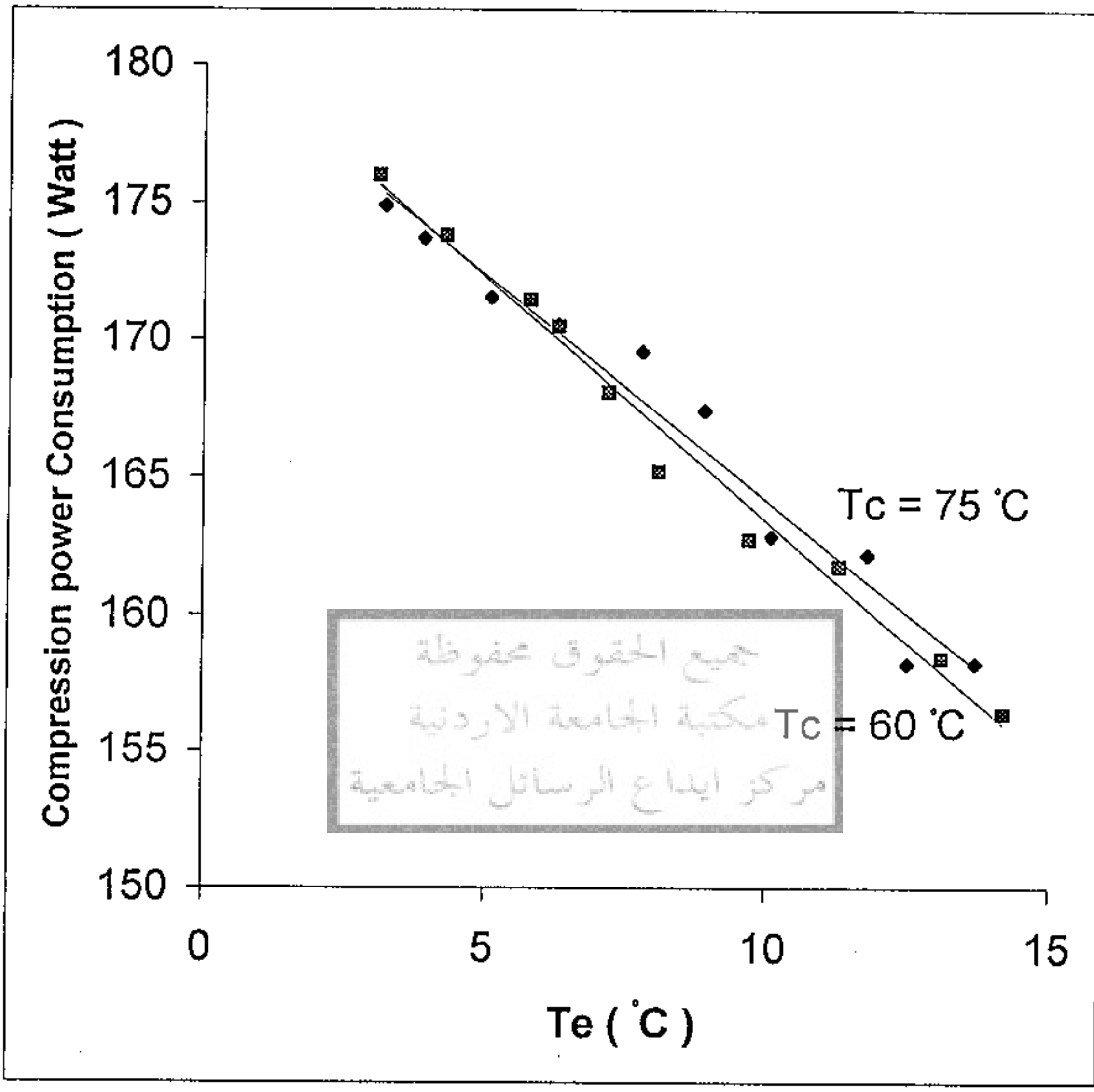

Figure 5.10 Compression power consumption versus T_e

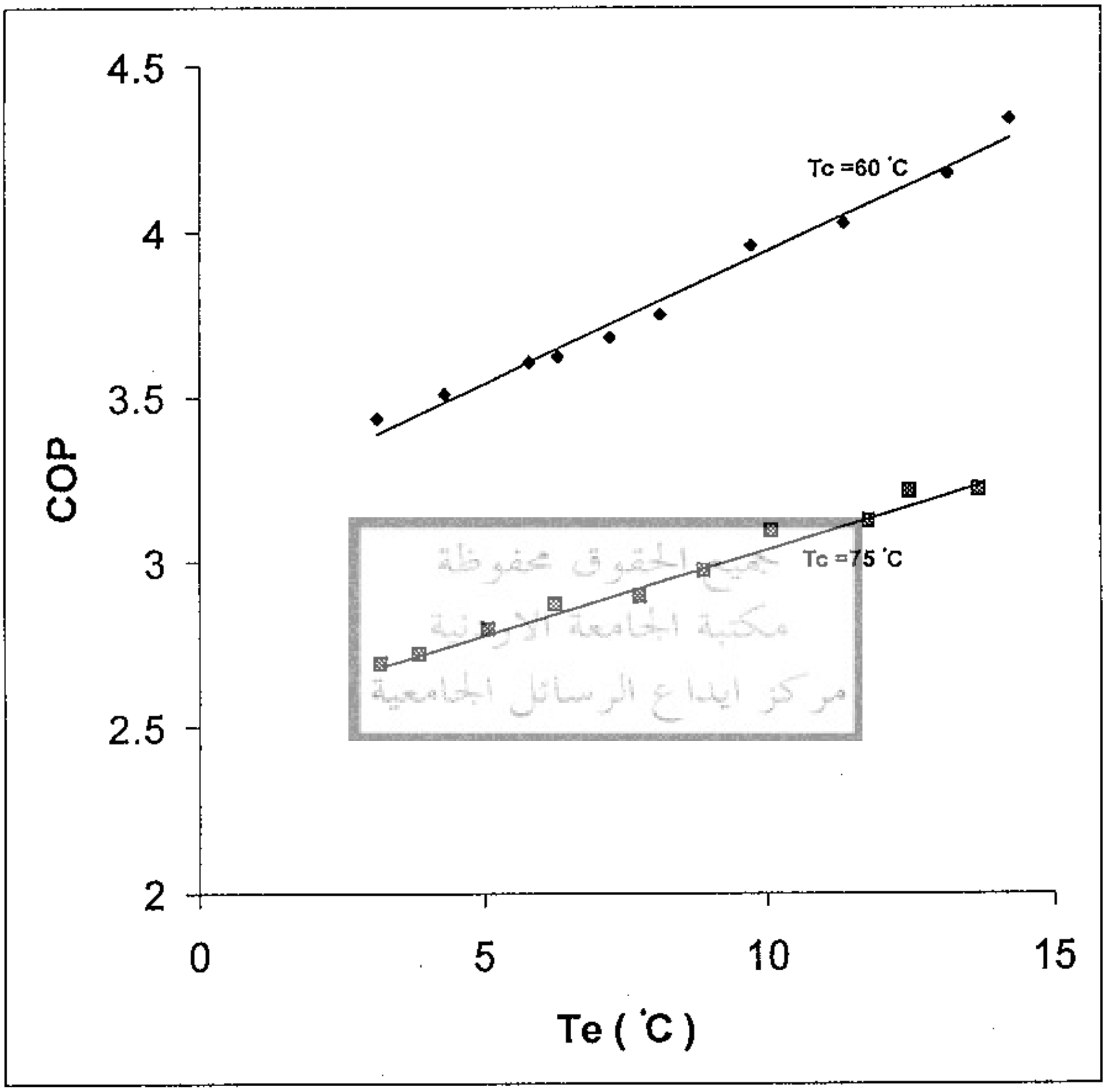

Figure 5.11 Coefficient of performance versus T_e

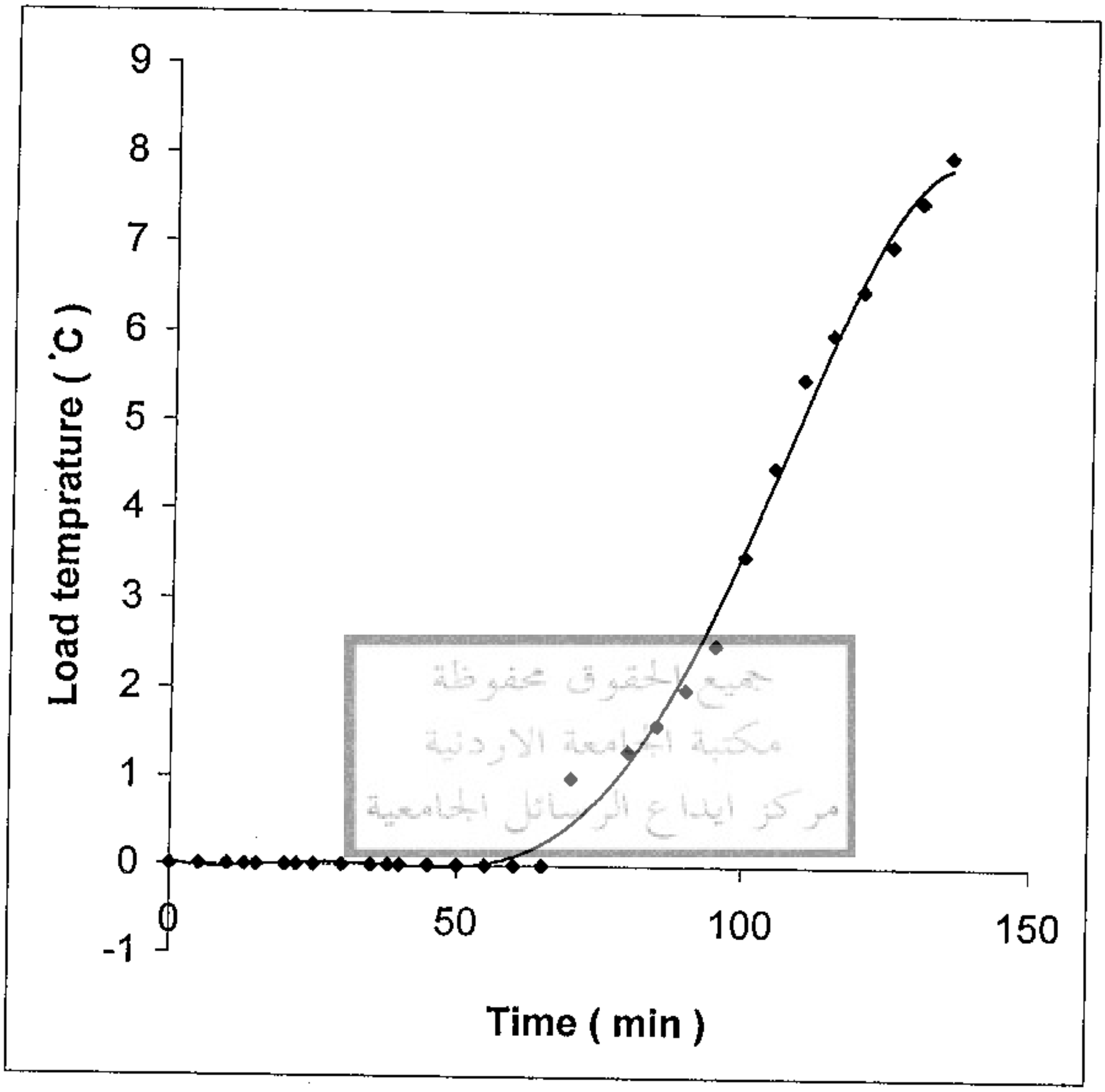

Figure 5.12 Load temperature versus time

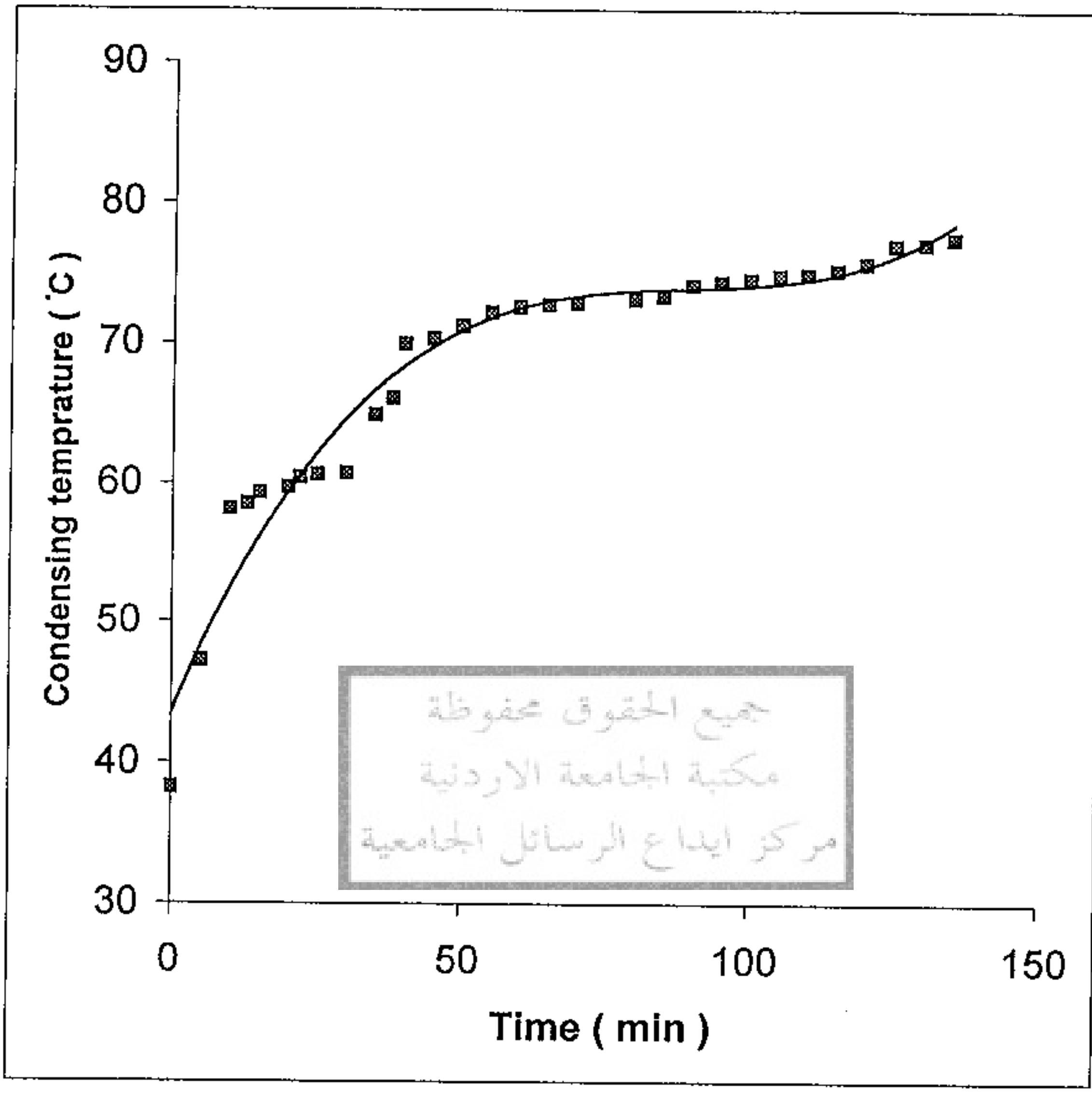

Figure 5.13 Condensing temperature versus time

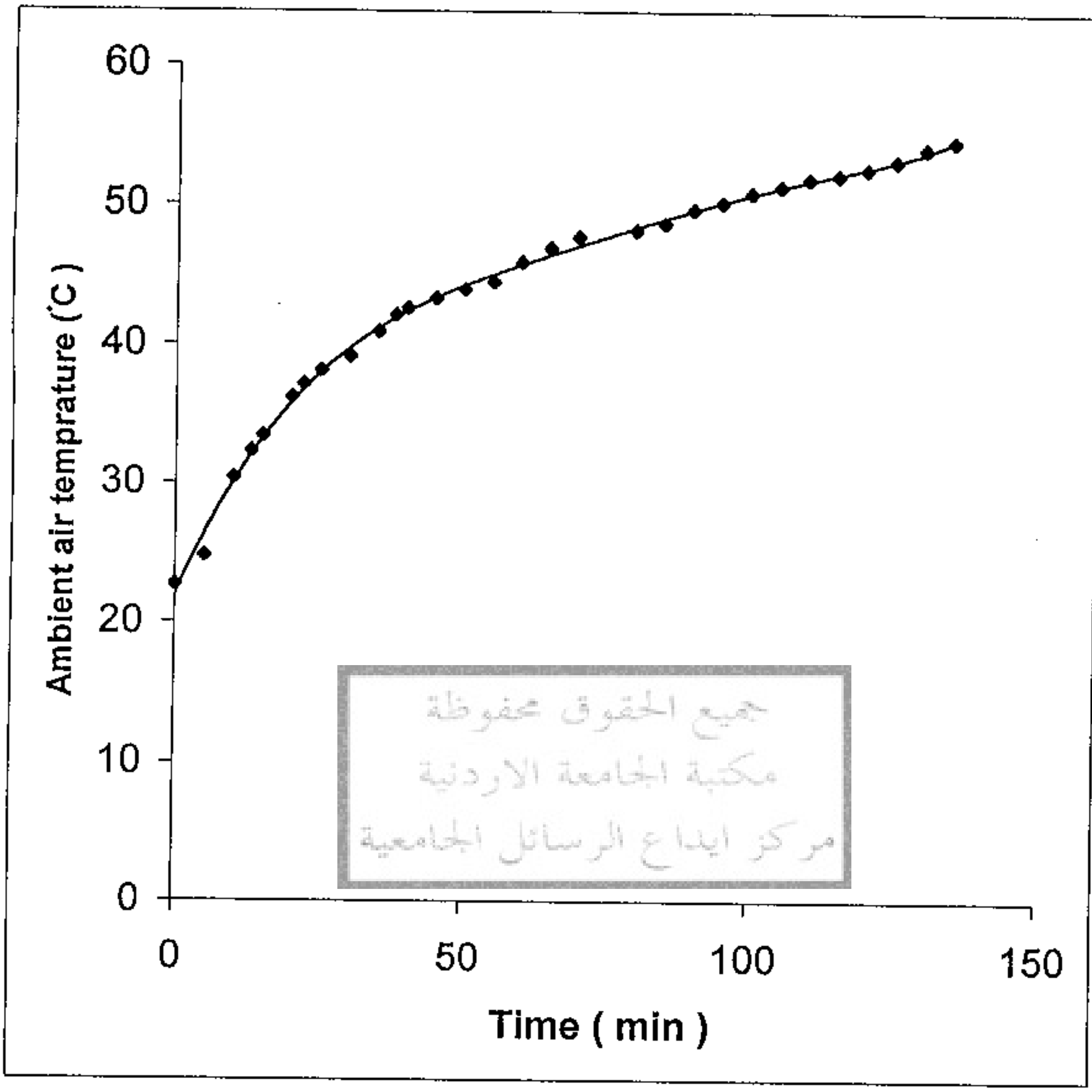

Figure 5.14 Ambient air temperature versus time

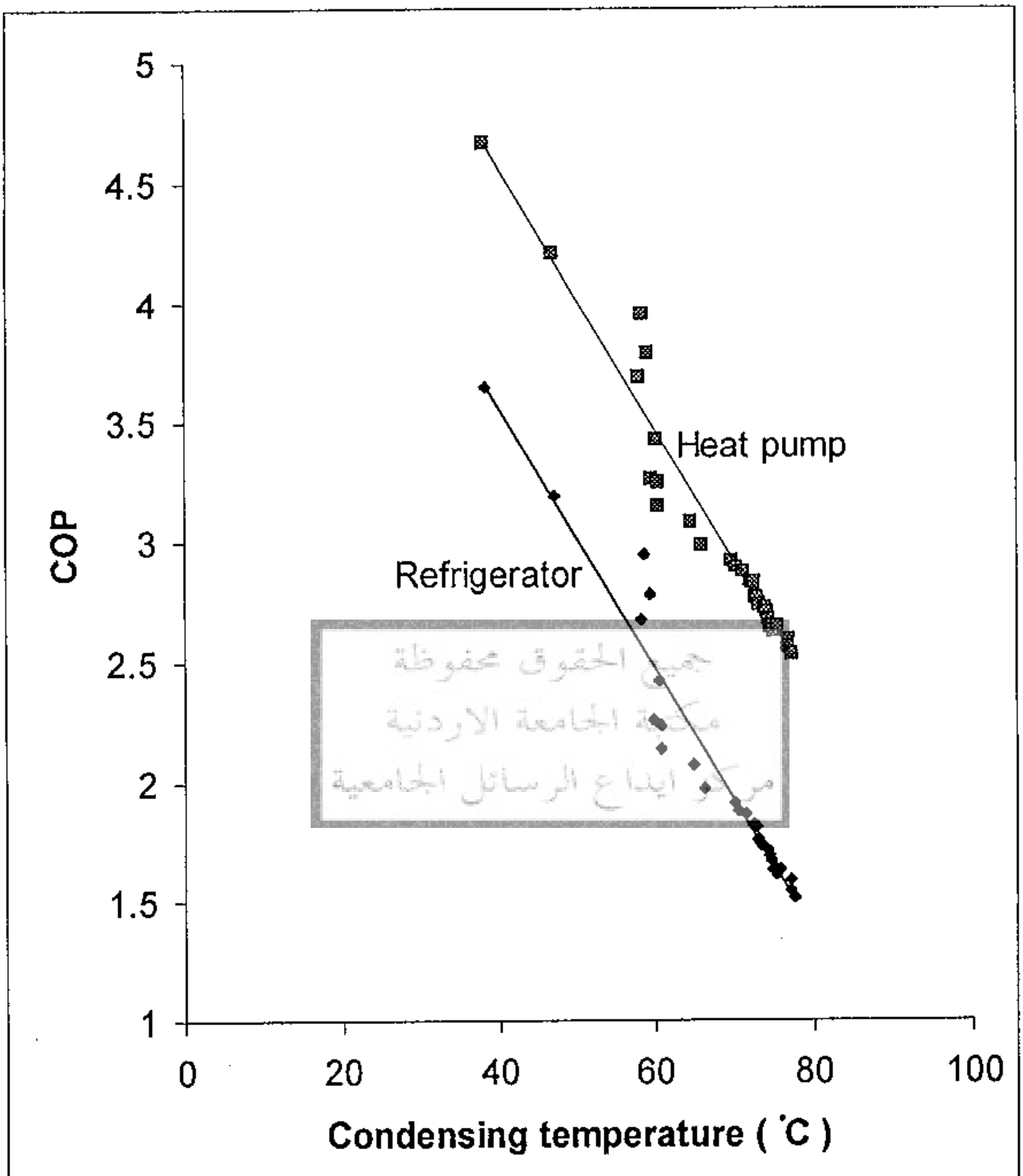

Figure 5.15 Coefficient of performance versus condensing temperature for refrigerator and heat pump

Chapter Six

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

This research covers an experimental study of the performance of a locally manufactured domestic refrigerator when used as a heat pump, by reversing the compressor connections.

The performance of the heat pump was tested and the following conclusions were deduced:

1. From the experimental results of the performance tests, it was concluded that heat pump is suitable for food heating and keeping it hot. The maximum air temperature attained inside the freezer cabinet is 54.6 °C. This occurred when the condensing temperature was 77.5 °C and the evaporating temperature was 19.4 °C.

2. The thermodynamical performance of the heat pump is competent with conventional ovens. Results showed that the coefficient of performance reached 4.7 at T_c of 38.2 °C and T_e of 2.2 °C. The COP decreased to 2.5 at T_c of 77.5 °C and T_e of 19.4 °C.

3. Due to the obtained high values of the coefficient of performance for the used heat pump, the heat pump is more efficient in converting electrical energy to heating energy than the resistance heater. In resistance heating the performance factor is one since the power derived from heating is the same as the electrical power provided to the heater.
4. There is energy saving in energy demand and consequently energy generation when the heat pump is used. In addition, the reduction in CO_2 emission is one of the advantages of this technique.
5. No leakage, noise or side effects were detected during the period of operating as a heat pump.

6.2 Recommendations

1. More work has to be done on calculating the long term performance of the heat pump for a wide range of working and environmental conditions.

2. Performance study has to be made for the heat pump using other refrigerants owing environmentally friendly nature as a replacement to R-12.

3. More experimental studies are recommended to use the domestic refrigerator for cooling and heating at the same time.

4. Some modifications in the domestic refrigerator when used as a heat pump have to be studied, like changing the capillary tube length and its diameter, to enhance the COP and to reduce the evaporating temperature. Also a suitable thermostat is recommended to be designed for the heat pump system to efficiently control the compartment temperature.

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APPENDICES

APPENDIX A: Data and Results (T_c Variation Test)

APPENDIX B: Data and Results (T_e Variation Test)

APPENDIX C: Data and Results (Load Temperature Test)

Data and Result (T_c variation Test)

Table A.2 T_a= 30.7 °C, T_e= 15 °C, Current = 1 Amp, voltage = 220 Volt

Data:

P_{in} bar	P_{out} bar	T_1 °C	T_c °C	T_5 °C	T_e °C	T_{10} °C	h_1 kJ/kg	h_5 kJ/kg	h_7 kJ/kg	h_{10} kJ/kg
2	13	69.3	58.2	51.5	14.2	3.9	219.4	86.52	86.52	190.2
2	14.5	81.6	59.6	53.4	14.3	4.2	227.4	88.55	88.55	190.4
2.1	16.5	91.7	60.4	54.4	14.7	4.3	233.1	89.63	89.63	190.3
2.1	17.5	93.8	64.6	54.8	14.8	4.5	233.6	90.06	90.06	190.4
2.2	18.5	97.3	67.0	56.0	14.9	4.4	235.3	91.36	91.36	190.1
2.3	19.5	101.7	71.6	57.7	15.3	4.3	237.8	93.21	93.21	189.9
2.3	20.5	106.7	73.7	59.8	15.4	4.3	240.9	95.52	95.52	189.9
2.3	20.5	108.3	74.7	60.7	15.6	4.7	242.2	96.52	96.52	190.1
2.3	20.5	109.7	76.1	61.3	15.8	5.1	243.4	97.19	97.19	190.4
2.4	21	111.3	77.4	61.6	15.9	5.2	244.2	97.52	97.52	190.3

Results:

Heating effect kJ/kg	W kJ/ kg	Q Watt	Mass flow rate kg/s	Compression Power Watt	COP
132.88	29.2	801.27	0.00603	176.08	4.5507
138.85	37.1	656.76	0.00473	175.48	3.7527
143.47	42.8	586.79	0.00409	175.05	3.3521
143.54	43.2	579.90	0.00404	174.53	3.3227
143.94	45.2	554.17	0.00385	174.02	3.1845
144.59	47.9	523.42	0.00362	173.40	3.0186
145.38	51.0	492.84	0.00339	172.89	2.8506
145.68	52.1	482.20	0.00331	172.45	2.7962
146.21	53.0	475.18	0.00325	172.25	2.7588
146.68	53.9	466.44	0.00318	171.40	2.7213

APPINDIX B

Data and Result (T_e variation Test)

Table B.1 T_a= 29.6 °C, T_c= 60 °C, Current = 1.1 Amp, voltage = 220 Volt

Data:

P_{in} bar	P_{out} bar	T_1 °C	T_c °C	T_5 °C	T_e °C	T_{10} °C	h_1 kJ/kg	h_5 kJ/kg	h_7 kJ/kg	h_{10} kJ/kg
1.8	11.5	80.5	59.6	51	3.1	0.2	230.3	85.99	85.99	188.3
1.8	12	80.7	59.8	51.3	4.3	1.3	229.9	86.31	86.31	189.0
1.9	12.5	80.8	59.9	51.7	5.8	2.8	229.3	86.74	86.74	189.7
2.1	13	81.1	60.0	52.2	6.3	3.6	228.9	87.27	87.27	189.8
2.1	13.5	81.3	60.2	52.8	7.2	4.2	228.4	87.91	87.91	190.2
2.2	14	81.5	60.3	53.4	8.1	5.3	227.9	88.55	88.55	190.7
2.2	14.5	81.7	60.5	53.7	9.7	7.8	227.4	88.87	88.87	192.4
2.3	15	81.9	60.5	53.9	11.3	8.7	226.9	89.09	89.09	192.7
2.3	15.5	82.2	60.6	54.2	13.1	10.1	226.5	89.41	89.41	193.7
2.4	16	82.3	60.7	54.3	14.2	11.7	225.9	89.52	89.52	194.5

Results:

Heating effect kJ/kg	W kJ/ kg	Q Watt	Mass flow rate kg/s	Compression power Watt	COP
144.31	42.0	604.66	0.00419	175.98	3.4360
143.59	40.9	610.26	0.00425	173.83	3.5108
142.56	39.6	617.29	0.00433	171.47	3.6000
141.63	39.1	617.51	0.00436	170.48	3.6223
140.49	38.2	618.16	0.00440	168.08	3.6778
139.35	37.2	618.71	0.00444	165.17	3.7460
138.53	34.7	649.71	0.00469	162.74	3.9580
137.81	34.2	651.84	0.00473	161.77	4.0295
137.09	32.8	662.15	0.00483	158.42	4.1796
136.38	31.4	679.17	0.00498	156.37	4.3433

Data and Result (T_e variation Test)

Table B.2 T_a= 29.6 °C, T_c= 75 °C, Current = 1.1 Amp, voltage = 220 Volt

Data:

P_{in} bar	P_{out} bar	T_1 °C	T_c °C	T_5 °C	T_e °C	T_{10} °C	h_1 kJ/kg	h_5 kJ/kg	h_7 kJ/kg	h_{10} kJ/kg
1.9	16	99.4	74.6	65.0	3.2	0.8	240	101.4	101.4	188.4
2.0	17	99.6	74.7	65.3	3.9	1.2	239	101.7	101.7	188.5
2.1	17	100.2	74.9	65.4	5.1	4.2	239.5	101.8	101.8	190.2
2.1	18	100.3	75.0	65.7	6.3	5.3	238.4	102.2	102.2	190.9
2.2	18	100.5	75.0	65.9	7.8	6.5	238.6	102.4	102.4	191.5
2.3	19	100.7	75.1	66.1	8.9	7.6	237.5	102.6	102.6	192
2.3	20	101.1	75.2	66.4	10.1	9.7	236.6	103	103	193.4
2.3	20	101.4	75.3	66.7	11.8	10.8	236.9	103.3	103.3	194.1
2.3	21	101.8	75.3	67.1	12.5	11.9	236	103.8	103.8	194.8
2.4	21	101.9	75.4	67.3	13.7	12.3	236	104	104	194.9

Results:

Heating effect kJ/kg	W kJ/ kg	Q Watt	Mass flow rate kg/s	Compression Power Watt	COP
138.6	51.6	469.85	0.00339	174.92	2.6861
137.3	50.5	472.31	0.00344	173.72	2.7188
137.7	49.3	479.20	0.00348	171.56	2.7931
136.2	47.5	489.00	0.00359	170.53	2.8674
136.2	47.1	490.32	0.00360	169.56	2.8917
134.9	45.5	496.43	0.00368	167.44	2.9648
133.6	43.2	503.67	0.00377	162.86	3.0926
133.6	42.8	506.34	0.00379	162.21	3.1215
132.2	41.2	507.65	0.00384	158.21	3.2087
132.0	41.1	508.2	0.00385	158.24	3.2117

APPINDIX C

Data of Load Temperature Test

Table C T_a= 29.6 °C, Current = 1.1 Amp, voltage = 220 Volt

Time min.	T_w °C	T_{air} °C	P_{in} bar	P_{out} bar	T_1 °C	T_c °C	T_5 °C	T_e °C	T_{10} °C
0	0	22.7	1.5	11	62.5	38.2	36.8	2.2	-4.9
5	0	24.8	1.5	13	69.1	47.2	43.5	2.5	-5.1
10	0	30.4	1.7	14.5	79.3	58.1	54.1	6.3	0.2
13	0	32.3	1.8	14.5	80.4	58.5	56.8	7.7	6.6
15	0	33.4	1.7	15	82.5	59.3	58.4	8.6	5.9
20	0	36.2	1.7	15	86.5	59.7	59.5	8.6	1.2
22	0	37.1	2.0	16.5	91.5	60.4	59.4	15.2	7.5
25	0	38.1	1.8	16.5	88.7	60.6	59.9	10.9	1.2
30	0	39.1	2.0	16.5	91	60.7	59	11.7	1.4
35	0	40.9	2.1	17.5	93.8	64.8	61.6	13.1	2.8
38	0	42.1	2.2	17.5	94.7	66.0	62.1	13.5	1.8
40	0	42.6	2.1	18.5	95.9	69.9	68	13.3	3.4
45	0	43.3	2.11	18.5	96.3	70.3	68.5	14.7	3.4
50	0	43.9	2.2	19.5	97.9	71.2	69.8	15.5	4.1
55	0	44.5	2.3	20	98.6	72.1	71.3	15.7	4.3
60	0	45.9	2.3	20.5	99.9	72.5	71.4	16.0	4.3
65	0	46.9	2.3	21	100.3	72.7	71.7	16.1	4.6
70	1	47.7	2.3	21	101.7	72.8	71.9	16.2	4.7
80	1.3	48.2	2.3	21.5	102.8	73.1	72.3	16.7	5
85	1.6	48.7	2.4	22	103.5	73.3	72.8	17.1	5.1
90	2	49.7	2.4	22	104.6	74.1	73.1	17.4	5.8
95	2.5	50.2	2.4	22	105.3	74.3	73.2	17.6	6.1
100	3.5	50.9	2.4	22	105.9	74.5	73.1	17.8	5.7
105	4.5	51.4	2.4	22	106.4	74.8	74.5	17.4	6.2
110	5.5	51.9	2.4	22	107.1	74.9	73.6	17.9	6.3
115	6	52.2	2.4	22	107.8	75.2	73.9	18.3	6.4
120	6.5	52.6	2.4	22.5	108.6	75.7	74.1	18.7	7.3
125	7	53.2	2.4	22.8	109.3	77.0	76	18.7	7.4
130	7.5	54.1	2.5	23.0	110.2	77.1	76.7	19.2	7.5
135	8	54.6	2.5	23.0	111.7	77.5	76.9	19.4	8

Results of Load Temperature Test

Table C T_a= 29.6 °C, Current = 1.1 Amp, voltage = 220 Volt

h_1 kJ/kg	h_5 kJ/kg	Heating effect kJ/kg	h_7 kJ/kg	h_{10} kJ/kg	Work of compression kJ/kg	COP
217	71.35	145.65	71.35	185.7	31.3	4.6534
219.2	78.16	141.04	78.16	185.6	33.6	4.1976
225.5	89.3	136.2	89.3	188.5	37	3.6811
226.4	92.23	134.17	92.23	192.4	34	3.9462
227.4	93.98	133.42	93.98	192.1	35.3	3.7796
230.7	95.19	135.51	95.19	189.1	41.6	3.2575
232.9	95.08	137.82	95.08	192.6	40.3	3.4199
230.6	95.63	134.97	95.63	188.9	41.7	3.2367
232.5	94.64	137.86	94.64	188.6	43.9	3.1403
233.6	97.52	136.08	97.52	189.3	44.3	3.0718
234.3	98.08	136.22	98.08	188.5	45.8	2.9742
234.1	104.8	129.3	104.8	189.7	44.4	2.9121
234.4	105.4	129	105.4	189.7	44.7	2.8859
234.5	106.9	127.6	106.9	190	44.5	2.8674
234.4	108.7	125.7	108.7	189.9	44.5	2.8247
234.9	108.8	126.1	108.8	189.9	44.7	2.8211
234.6	109.2	125.4	109.2	190.1	44.5	2.8180
235.9	109.4	126.5	109.4	190.1	45.8	2.7620
236.2	109.9	126.3	109.9	190.3	45.9	2.7516
236.2	110.5	125.7	110.5	190.2	46	2.7326
237.2	110.8	126.4	110.8	190.7	46.5	2.7183
237.8	111	126.8	111	190.8	47	2.6979
238.3	110.8	127.5	110.8	190.6	47.7	2.6730
238.8	112.5	126.3	112.5	190.9	47.9	2.6367
239.4	111.5	127.9	111.5	191	48.4	2.6426
240	111.8	128.2	111.8	191	49	2.6163
240.1	112.1	128	112.1	191.6	48.5	2.6392
240.3	114.4	125.9	114.4	191.7	48.6	2.5905
240.9	115.2	125.7	115.2	191.6	49.3	2.5497
242.2	115.5	126.7	115.5	191.9	50.3	2.5189

دراسة أداء ثلاجة منزلية مصنعة محليا عند استعمالها كمضخة حرارية

إعداد
محمد عبد الرحيم النجار

إشراف
أ.د. محمد السعد

ملخص

هذا البحث دراسة تجريبية للأداء الحراري لثلاجة منزلية مصنعة محليا عند استعمالها كمضخة حرارية، بوساطة عكس أنابيب السحب والتصريف الموصولة بالضاغط في الثلاجة. هذا يحول الثلاجة إلى فرن ذو عامل أداء عالي.

تم إجراء تجربتين ، الأولى لدراسة أداء المضخة الحرارية عند تغير درجة حرارة المكثف مع ثبات درجة حرارة المبخر والثانية عند تغير درجة حرارة المبخر مع ثبات درجة حرارة المكثف. ورسمت منحنيات أداء المضخة الحرارية بالاعتماد على هاتين التجربتين.

بالاعتماد على نتائج التجارب المذكورة يمكن الاستنتاج أن هذه المضخة الحرارية مناسبة لتسخين الطعام وحفظه ساخنا. بلغت درجة حرارة الهواء داخل الثلاجة ٥٤،٦ درجة

مئوية، عندما كانت درجة حرارة المكثف ٧٧،٥ درجة مئوية و درجة حرارة المبخـــر ١٩،٤ درجة مئوية.

كذلك فإن عامل أداء المضخة الحرارية كان ٣،٢ عندما كانت درجة حرارة المكثـــف ٧٧،٥ درجة مئوية، ودرجة حرارة المبخر ١٣،٧ درجة مئوية.

وبما أن عامل أداء المضخة الحرارية أعلى من عامل أداء الثلاجـــة بواحـــد، أي أن الطاقة الحرارية الناتجة عن المضخة الحرارية تكون دائما أعلى من الطاقة الكهربائية المقدمـــة لها. لذلك نكون قد حققنا وفرا في الطاقة المطلوبة.