UNIVERSITY OF JORDAN
FACULTY OF GRADUATE STUDIES

PERFORMANCE OF PROPANE/BUTANE (LPG) MIXTURE AS A REFRIGERANT FOR USE IN DOMESTIC REFRIGERATORS

BY

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Submitted in Partial fulfillment of the requirements
for the degree of master of science in
Mechanical Engineering
Faculty of Graduate Studies
University of Jordan

June , 1994

This thesis was defended successfully on 29/6/1994

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ACKNOWLEDGMENT

It is a deep pleasure I express my gratitude to all who helped me during the preparation and completion of this work.

I wish gratefully to thank my supervisor, Dr. Mahmoud Hammad for his sincere guidance and encouragement through this work. Special thanks are also indebted to Prof. Mohammed Al-Sa'ad for his helpful advises and supporting.

Also, I would like to thank the management of the Household Appliances Manufacturing Co. (HAMCO), especially Mr. Mahmoud A. Ali, for their favor in providing us with a refrigerator unit for the research.

Thanks are also extended to Mr. Aref Shahin for his appreciable assistance, and to all my friends who supported me during this work.

Finally, I highly appreciate the patience and support of my family and my fiancee during the long period of this research.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ABSTRACT

" The performance of propane/butane (LPG) mixture as a refrigerant for use in domestic refrigerators "

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The aim of this research is to test experimentally an alternative to refrigerant R-12 which can work with domestic refrigerators without changing the design of these refrigerators.

The alternative tested in this research is the **Liquefied Petroleum Gas (LPG)**, which is composed mainly of butane and propane gases. LPG has three main advantages over other alternatives, especially for the Developing Countries, which are : availability, cheapness, and its significant environmentally friendly nature.

A domestic refrigerator unit is used to test and study the performance of LPG as a refrigerant, and compare it with the performance of R-12; the original refrigerant. The same refrigerator unit is used to test the two refrigerants to avoid any performance mismatches between different units.

Seven different LPG charge amounts are tested to determine which amount has the best performance. Then the performance of the best LPG charge quantity is compared with that of R-12, and a number of different performance curves are presented.

The refrigerator worked satisfactorily on the LPG mixture for four months and gave evaporating temperature, T_e, down to -19.6 °C, and coefficient of performance (COP) up to 5.9 at T_e of -5 °C, condensing temperature, T_c, of 28.7 °C and ambient temperature, $T_{amb.}$, of 19.5 °C.

The LPG refrigerant was found to have a slightly higher COP than R-12, slightly lower refrigeration capacity and slightly lower power consumption, all at $T_e \leq -5$ °C.

The results have shown that the LPG (propane/butane mixture) is an attractive alternative to R-12 in domestic refrigerators. Its performance is fairly comparable with that of R-12 without making any changes or replacements in the refrigerator system. No side effects nor any disadvantages were detected during the operation period .

CHAPTER 1

INTRODUCTION

Domestic refrigerators are the most world-wide used home appliances that works with refrigerants. Therefore, the phasing out of chlorofluorocarbon (CFC) refrigerants will cause heavy effect on these appliances and their users.

1.1 Environmental Aspects : [1, 2, 3]

Two aspects of refrigerants effect have recently been much discussed in relation to environmental issues . The first concerns the ozone layer depletion by chlorofluorocarbons (CFCs) and the second relates to global warming. CFCs, notably CFC-12 and CFC-11, find much use in the refrigeration and air-conditioning industries as refrigerants. The former refrigerant is used almost universally in domestic refrigerators and freezers while the later is used as a blowing agent in refrigeration and other applications.

It was accepted in the mid-1980s that CFCs reaching the upper layers of the earth's atmosphere would attack the ozone molecules and destroy them. As ozone protects the earth's surface against ultra-violet light, its destruction has a significant impact to all life forms on the planet. So much so that exposure to sunlight in the southern hemisphere for more than some minutes daily is now considered a health hazard.

Quite recently, In January 1992, dramatic news about high concentrations of chlorine monoxide (which is formed by chlorine atoms released from CFCs) over the northern hemisphere was reported. Large parts of Europe and North America are now threatened by an ozone hole overhead.

When a chlorine monoxide molecule meets a free ozone molecule, an oxygen molecule is formed and the chlorine monoxide molecule is free again to renew its attack on another ozone molecule. This process can be repeated until up to 50,000 ozone molecules have been destroyed. The relative ozone depletion potentials (ODPs) for various fluorocarbon refrigerants are shown in figure (1.1).

The rising of the CFC molecules to the upper layers of the atmosphere is quite slow in that quantities that have been freed since their discovery in the 1930s are still finding their way to the stratosphere and those quantities that have been released now will show the damage within the coming decades [3].

International action was first agreed upon in Montreal, Canada in 1987 and later in London in 1990 and in Copenhagen in 1992. Based on the Montreal Protocol, the European Community (EC) and some European countries had already provided more stringent regulations. In the execution of the Montreal Protocol requiring a 50% CFC reduction by 1998, the European Community (EC) aimed at an approximate phase-out by the turn of the century ; some even more stringent national schedules in Europe stipulated the same goal by 1995 (as Germany) or 1996. These regulations are summarized in Table 1.1 .

Now, with even more stringent United Nations Environment Program (UNEP) regulations ,which are based on the amendments of the Montreal Protocol decided on in London and in Copenhagen , the situation in Europe as well as world-wide is quite different.

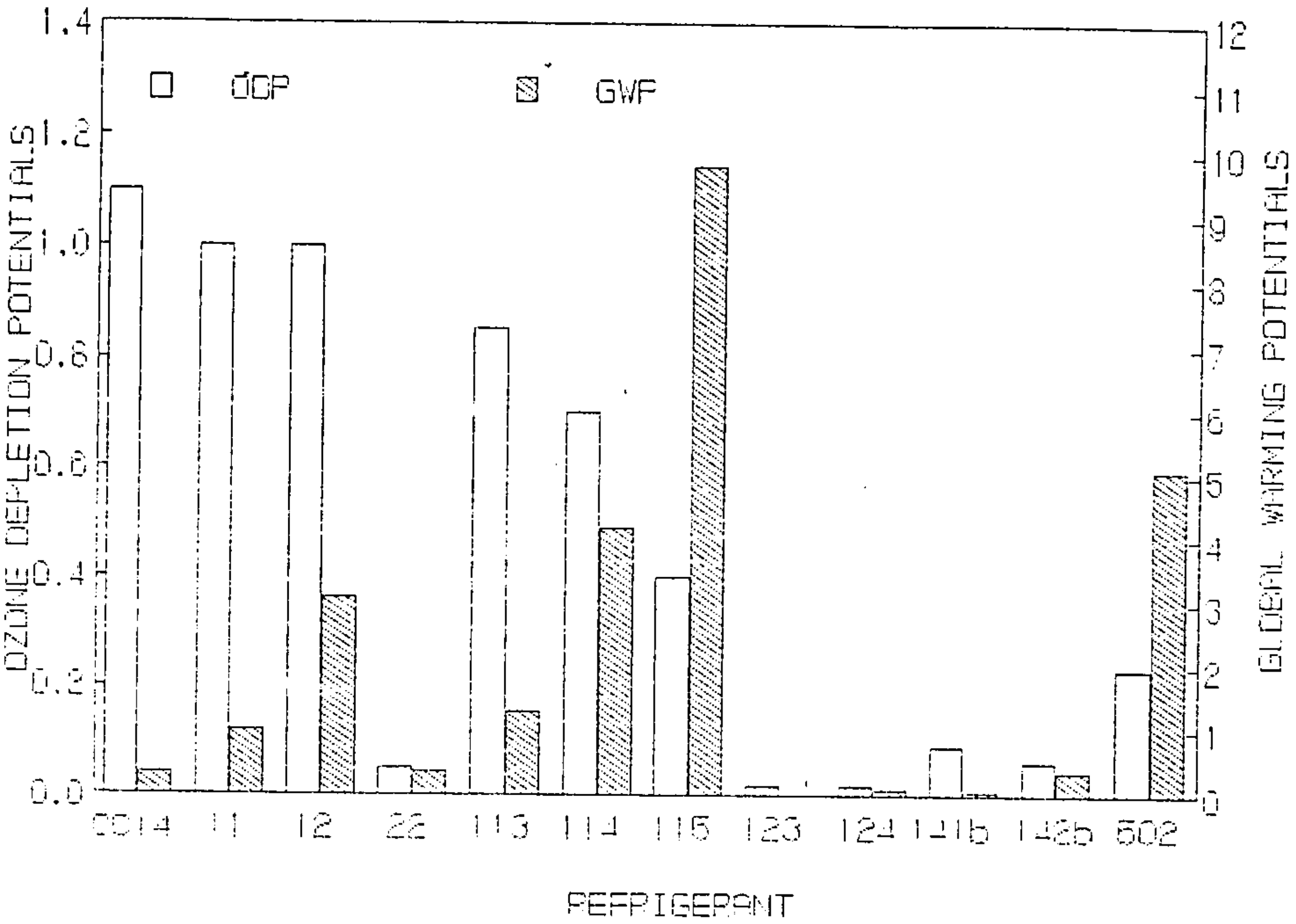

Fig.(1.1) : ODPs and GWPs for some fluorocarbon refrigerants [3].
(Relative to ODP and GWP of HCFC-11 = 1.0)

Table 1.1 : CFC and HCFC phasing-out regulations [2].

Country	CFC regulations		HCFC regulations	
	Date	Regulation	Date	Regulation
World/UNEP	1/1/96	CFC ban	1/1/2030	HCFC ban
EC	1/1/95	all uses ban	1/1/95	HCFC ban
Austria	1/1/92	ban > 1 kg		*
	1/1/94	small units ban (< 1 kg)		
Belgium	1/1/91	ban (> 10 kg)		
	1/1/95	all uses ban		
Denmark	1/1/95	all uses ban		
Finland	1/1/96	all uses ban		
France	1/1/95	all uses ban		
Germany	1/1/92	new equipment ban (> 5 kg)	1/1/2000	R-22 ban of new plants
	1/1/94	new equipment ban (< 5 kg)	1/1/2030	all uses and production ban
	1/1/95	all uses ban		
Great Britain	1/1/95	all uses ban		
Italy	1/1/95	all uses ban	1/1/96	R-22 ban (not essential uses)
			1/1/2003	all uses ban
Netherlands	1/1/94	ban for automotive aircond.		
	1/1/95	all uses ban		
Spain	1/1/95	all uses ban		
Sweden	1/1/94	ban for new units (> 10 kg)	1/1/94	only allowed in refrigeration systems, foams and heat pumps
	1/1/95	ban in new installations		
Switzerland	1/1/94	ban in new equipment		*

* The HCFC phase-out should start before the year 2004 and should be finished before the year 2030.

Whereas in other parts of the world hydrochlorofluorocarbons (HCFCs) were still considered as substitutes for CFCs, in Europe already a discussion about their role concerning ozone depletion and their future was initiated very early. In addition, HCFC phase-out worldwide between 2004 and 2030 has been taken into consideration by UNEP, including the important refrigerant R-22 (see table 1.1).

The other aspect of refrigeration relating to environmental issues is the global warming issue. This issue was stressed due to the considerable global warming potentials (GWP) of fluorocarbon refrigerants, as shown in fig.(1.1). It has particularly affected the discussions of CFC and HCFC substitution technologies in Europe because of uncertainty about the future of hydrofluorocarbons (HFCs), such as HFC-134a and HFC-152a, and their mixtures as substitutes for CFCs and HCFCs.

Because of this uncertainty in Europe and especially in Germany, development has been driven by the global warming issue to draw more and more attention to non-fluorocarbon technologies as substitutes for CFCs and HCFCs.

It was stated that, similar to the process which led to the Montreal Protocol on ozone-depletion substances, a climate convention could lead to a global warming regulation, which, besides CO2 and other greenhouse gases, could also affect HFC fluids, which are not under the control of the Montreal Protocol.

1.2 Finding Alternative Refrigerants :

Since the phasing-out of CFC refrigerants is only a matter of time, it is a vital matter to find environmentally safe alternatives that could replace these refrigerants.

which became a very well investigated refrigerant now. On the other hand, this state-of-the-art refrigerant has a number of problems that can't be ignored. Some of these problems are :

- The cost of this refrigerant is considerably higher than old refrigerants.

- It is almost immiscible with standard mineral refrigeration oils suitable for CFC refrigerants. Therefore, new lubricants are now produced to work with R-134a (as polyglycol or ester oils), which adds more costs to its users [4,5,6].

- Its thermophysical properties make it difficult to use in low temperature refrigeration, and its oil hygroscopicity makes it difficult to be used in warm humid climates [4].

- It is not compatible with fluorinated rubbers which makes it necessary to replace all rubber sealings of this type. Also, because of its different diffusion characteristics, refrigerant hoses have to be equipped with an inner core of polyamide [5].

- It is not suitable for high temperature applications (beyond 70 °C), where its performance deteriorates severely [7, 8].

- High system cleanliness care must be taken before filling R-134a refrigerating units, which can be cumbersome and quite expensive [5, 8].

- Because of its global warming potential (GWP), the whole future of HFC refrigerants is in question. It is well to remember that when the CFCs were discovered in the 1930s they were thought ideal. It has taken decades for their harmful environmental effects to appear. Similarly, some of the characteristics of the HFC alternatives will take a long time to assess.

Another example is hydrocarbons. Hydrocarbons (such as methane, ethane, propane, butane, ...,etc.) have only two reactive characteristics, combustion and halogenation, otherwise they are very stable. The most important thing about these compounds is their environmentally friendly characteristics. Their ozone depletion potential is zero, and their greenhouse effect appears to be insignificant compared to R-12 (about 0.15% of the effect of R-12).

In addition, they have excellent thermodynamical, physical and chemical characteristics (as will be discussed in chapter 2). Their only main disadvantage is their high flammability, and because of this characteristic they have been prohibited from being used in domestic applications as well as many commercial and industrial applications.

Recently, some of these compounds have been put into application and proved successful in replacing R-12 and other CFC refrigerants. For example, propane/butane filled refrigerators have now entered the German market and are proving extremely popular [9]. At the same time, tests are being made to replace R-22 by propane in heat pumps, where propane has possessed better performance than R-22 [9]. Hence, the use of hydrocarbons in applications where small refrigerant amounts are required, is gaining more and more popularity.

1.3 Literature Survey :

The phasing out of CFCs due to their destructive environmental impact is relatively a new issue. This issue began with the Montreal Conference in 1987 and is still reacting worldwide until the date of this research. Hence, this field of research is, compared to other fields, a very new one, and therefore the number of researches concerning it is little compared to other fields.

The researches concerning the issue of replacing CFCs can be divided into two groups. The first group includes researches which deals with the dangerous and destructive effects of CFCs on environment, and emphasizes the necessity of phasing out their production as soon as possible. The second one includes researches on studying and experimenting some refrigerant alternatives which could replace CFC refrigerants. This work is part of the later efforts. In the next two sections, a literature survey for the two groups researches is given.

1.3.1 Researches on CFC's Environmental Impact :

Rowland [1] discussed all the issues regarding CFC impact on ozone layer in the stratosphere. He refutes all the claims which absolves CFCs from the depletion of ozone layer, in a controversy manner. He listed strong scientific evidences with numbers from geophysical, chemical, meteorological and aeronautical points of view. The author concluded that scientific evidences on the destructive effect of CFCs on ozone layer are so strong that they cannot be denied.

Kruse [2] discussed the latest world reactions on the CFCs issue. His research is divided into two parts, the first one dealing with researches and regulations regarding the ozone depletion issue and the second one dealing with the global warming issue. In the first part, the CFCs phasing-out schedule and the HCFCs phasing-out schedule are discussed according to the UNEP regulation, the EC regulation and the individual regulations in European countries. According to these phasing-out schedules, CFC refrigerants will be phased out completely and all its uses banned by the first of January 1996, while HCFC refrigerants will be phased out completely by the first of January 2030.

In the second part of the research, the author concluded that in spite of not having up-to-date regulations on the global warming effect of refrigerant systems and fluids, a dynamic development has taken place in Europe to employ as far as possible more and more natural fluids (such as hydrocarbons, ammonia and carbon dioxide) in refrigeration and air-conditioning systems. Hence, European research and development at this time will concentrate on the application of natural old fluids, which are well known and already available without any necessary toxicity testing.

Mango et al. [3] summarized the problem of CFC environmental impact and CFCs replacement. After describing briefly the mechanism of ozone depletion by released CFCs and CFC phasing-out regulations, they gave some suggested replacements for CFCs in different applications; most of these replacements based on HFC and HCFC refrigerants. They concluded that the number of alternative refrigerants for the moment and for the foreseeable future is rather limited.

1.3.2 Researches on CFC Alternatives :

The International Institute of Refrigeration (IIR) has published recently an informatory note [4] about the replacement of the currently used CFCs and HCFCs. The object of this note is to help refrigerant manufacturers and government officials in making right decisions regarding the problem of phasing out CFCs and HCFCs.

The note presents and discusses all possible alternatives to CFC and HCFC refrigerants including HFCs (such as HFC-134a), ammonia, hydrocarbons, water, air, CO_2 and refrigerant mixtures. The disadvantages of each one are discussed briefly.

According to this note, hydrocarbons (such as propane and butane) are excellent refrigerants for many applications. Their toxicity/flammability generally do not keep them from being used for domestic as well as industrial applications. But, of course, special plant equipment design, suitable precautions and adequate risk assessment are required. Whatever justifications there are, all regulations which inhibit the use of hydrocarbon substitutes affect the CFC phase out process.

The note concluded that when selecting substitutes for CFCs and HCFCs, there will be an inevitable trade-off to some extent between the following : ozone depletion (ODP), global warming (GWP), flammability and toxicity.

Most of the researches regarding the replacement of R-12 have concentrated on R-134a (or 1,1,1,2-tetraflouroethane) as a perfect substitute.

Preisegger and Henrici [5], summarized the requirements for a suitable replacement for R-12 by R-134a, the developmental efforts that have been made and the results of this process. Chemical properties, material compatibility and thermodynamic properties of R-134a are described. Also, special requirements for suitable compressor lubricants and system cleanliness are mentioned. Finally, some explanations of installing closed product cycles for CFCs and their substitutes are given. They concluded that R-134a has excellent chemical and thermal stability.

The authors also compared some thermodynamic properties such as the isentropic component, the speed of sound in refrigerant vapour, COP, specific heat capacity, thermal conductivity and viscosity of R-134a with that of R-12 and they proved acceptable.

Carpenter [6] investigated the retrofitting of R-134a into existing R-12 systems. In his research, he concentrated on the practical problems encountered when retrofitting R-134a into R-12 systems. The items that the retrofitting process involves from replacing the existing mineral oils to the modification made on the system design and the replacement of some system components as expansion valves and driers are discussed.

The retrofitting procedure is summarized in eight steps and five case studies for different applications are given with describing the conversion to R-134a procedure for each one. Carpenter concluded that R-134a and ester lubricants can be retrofitted into many of the existing refrigeration and air-conditioning systems currently running on R-12.

Bansal et al. [7,8] evaluated the performance of HFC-134a when retrofitted into heat pumps. They followed two approaches in their performance evaluation: theoretically, by making a simulation study [7] and experimentally [8].

Their experimental study of HFC-134a on an industrial heat pump showed that HFC-134a has some disadvantages. For example, in order to replace R-12 with R-134a in existing heat pumps, cleaning a heat pump system is cumbersome and can be quite expensive. In addition, R-134a cannot offer the full operating range of R-12, particularly beyond 70 °C, because its performance deteriorates severely at large temperature differences ($\Delta T \geq 60$ °C). Despite its shortcomings, R-134a offers many advantages and may be a potential alternative for new heat pump installations, but only for low to medium temperature applications (i.e. applications with operating temperatures below 70 °C).

Although the impact of HCFC refrigerants (such as R-22) on ozone depletion is significantly less than that of CFCs and they will not be phased out until the beginning of the next century, some recent researches investigated the possible alternatives for HCFCs.

Spartz [10], investigated alternative refrigerants for R-22 chillers. All his alternatives are based on HFC refrigerant mixtures. He summarized the results of his research as that there are already some promising candidates to replace R-22 in chillers, although much work is still to be done. The R-32/R-125/R-134a mixture, R-134a and R-32/R-125 azeotrope are possible alternatives, but each one possesses some problems which need to be solved.

Other authors investigated various alternative refrigerants for retrofitting in existing CFC and HCFC systems and equipment.

Parsnow [11] investigated the retrofitting of R-134a in positive pressure chillers. He discussed the modifications that should be made to the positive pressure chillers for the conversion process to be successful. He concluded his research with an advice to the owners of existing chillers to develop feasibility evaluations to decide if it is better to make conversion for the existing systems or replace them with new systems designed for the new alternative refrigerants.

Madan [12] also selected R-134a as the best alternative for chillers, and discussed the recommended system changes for the conversion process.

Massien and Demke [13] investigated R-134a, R-32, R-125 and R-152a as possible alternatives for chillers, while Smithart and Grawford [14] investigated R-123 as alternative refrigerant for low pressure systems working on R-11.

Ostman [15] made a comparison between R-123, R-22 and R-134a as alternatives to CFC refrigerants in air-conditioning applications. Although non of them is perfect from environmental view point, they are significantly better than CFCs. The author recommended some strategies to retrofit existing CFC systems with the suggested alternatives. He also suggested other refrigeration technologies to face the CFC phasing out problem, such as expanding the use of absorption chillers, using direct natural gas-fired absorption systems and using ammonia-based refrigeration systems for air-conditioning applications.

Most of the prementioned researches had concentrated on HFC refrigerants as substitutes for CFC and HCFC refrigerants. Some had also considered HCFCs (such as R-22) to replace CFCs in the present time, since their environmental effects are significantly less than CFCs, until finding other alternatives in the future. On the other hand, the researches which have considered natural refrigerants, such as hydrocarbons, ammonia or carbon dioxide as possible alternatives to CFCs are rare.

James and Missenden [16], investigated the performance of propane for use in R-12 domestic refrigerators and compared it with the performance of R-12. In order to make this comparison, they used two identical new domestic refrigerators designed to operate with R-12. The two units were located side by side in a large laboratory so that any disturbance was applied to both units. Then one of the units was retrofitted with propane.

Several different tests were done by the researchers to compare the performance of the two refrigerants. The power consumption of the two units was monitored over a period of 45 days. The results showed small differences between the consumption of the two units during that period. Some other small tests are made such as measuring the on/off cycle period and the motor protection period.

The most important tests done in this research are the safety tests. Three main safety issues regarding domestic refrigerators were addressed, which are :

1. Leakage inside a refrigerator cabinet and ignition.
2. Leakage near a flame such as a cooker or boiler.
3. The risks in the event of a fire.

The first of these is relatively easy to overcome. The evaporator can be placed between the insulation and the inner cabinet skin (which is already done by some manufacturers for other reasons). Alternatively, both light switch and thermostat may be placed outside the refrigerated enclosure. In any event the consequences are not catastrophic as was proved by the authors.

A combustion test was done on a domestic refrigerator fitted with propane, where a connection was made to the evaporator that allowed the refrigerant into the cabinet and an internal ignition device was constructed to produce a high-energy spark. The resulting ignition was only partly explosive; a small short-lived flame was observed inside the cabinet and recorded by video. The intensity of the flame was not sufficient to burn the plastic liner even after numerous tests.

The second problem can be overcome by instructing the user not to place the refrigerator next to the cooker or boiler; this is very bad practice anyway and few people would do this. Due to the small amount of propane used, in the case of any leakage, the concentration of propane in the room will not reach its lower explosion limit.

The third point was the subject of a fire test made by the authors in the Fire Services College, UK. Two identical refrigerator units, both originally operated with R-12, were used. The refrigerant in one was replaced by propane. Then the two units were mounted in the fire test room. The fire test showed that the most greater hazard in a fire event are the toxic fumes from the cabinet and its insulation which include cyanides and choking smoke. The lubricant oil materially contributed to the severity of the fire and would cause particular hazard in a house fire.

On the other hand, propane did not noticeably add to the conflagration nor escape catastrophically. The products of combustion from propane are much less dangerous than those of R-12 leaking near a flame, where intensely toxic products such as carbonyl chloride ($COCl_2$) are produced. It should be noted that most fire injuries are due to smoke inhalation, not to burns or explosions.

Finally, some other parameters as refrigerant quantity, environmental effects and costs were compared between R-12 and propane. The authors concluded from their research that propane presents an attractive alternative to current CFCs in small systems such as domestic refrigerators, given correct technical application for operational and safety factors. For the very large third world requirements, propane is well suited because of its wide and multiple sourcing and low cost.

1.4 Importance of This Work :

From the previous discussions and from the fact that the number of alternative refrigerants for the moment is still limited, it is clear that finding and testing more and more alternative refrigerants is a vital matter.

This research tests and studies the thermodynamic performance of LPG in domestic refrigerators, and tries to detect any side effects on the refrigerator system or hazards on humans that may be caused by this refrigerant. Also, any required system design changes, and the cost savings made by using LPGs are discussed. Therefore, this research is part of the efforts of finding new suitable, environmentally safe, alternative refrigerants for current CFCs.

CHAPTER 2

R-12 VS. PROPANE/BUTANE (LPG) MIXTURE

A COMPARATIVE STUDY

2.1 Introduction :

When selecting a refrigerant for a specific application, the properties of this refrigerant have to satisfy a number of application requirements in order to be suitable for that application. The requirements that must be satisfied by the refrigerant may be classified as thermodynamical, physical, and chemical requirements in addition to other factors as cost and availability. Each refrigerant has its own specific properties and characteristics which must be known well before deciding if a refrigerant is suitable for an application or not. Also, these properties and characteristics form a good basis for comparisons among different refrigerants [17].

In this chapter, a comparative study is made between R-12 and the propane/butane (LPG) mixture regarding their thermodynamic, physical and chemical properties in addition to other comparison factors. The aim of this chapter is to come out with a conclusion about if LPG is competent enough to replace R-12 or not. If the properties of the LPG fulfill the domestic refrigeration requirements, then it is worthwhile to make researches regarding this issue.

R-12 is the most famous refrigerant worldwide especially for domestic applications. It was the first fluorocarbon-type refrigerant developed and used commercially and is still a standard for comparing other refrigerants. All its properties and characteristics have been well studied, documented, and represented in the form of tables, charts and mathematical formulas, over the past 60 years.

On the other hand, hydrocarbons as propane and butane did not have that chance in the refrigeration world. Although they are well known, and extensively used, as fuels for domestic, commercial and industrial applications, they are rarely used as refrigerants. Although they possess satisfactory thermodynamic properties, their flammability characteristic has limited their use as refrigerants to few special applications.

Before discussing the characteristics of the two refrigerants, a general background about Liquefied Petroleum Gases (LPGs) is listed in brief.

2.2 Liquefied Petroleum Gases, A General Background : [18]

The liquid products derived from crude petroleum can be very complex mixtures of hydrocarbons, but petroleum gases have a simpler composition. A study of the properties of these gases can be based to a large extent on a determination of their exact composition and a knowledge of the characteristics of the constituents. The term *petroleum gases* is used to describe the gaseous phase and liquid phase mixtures comprised mainly of C1 to C4 hydrocarbons. They are classified into three main groups: natural gases, refinery gases, and liquefied petroleum gases (LPG).

LPGs are composed almost entirely of propane and butanes and are liquefiable under pressure (up to 14 atm max.) at ambient temperature. Therefore, they can be stored and transported as a liquid and then vaporized and used as a gas. In the conversion from liquid phase LPG to the vapor phase the contents of the bottle or tank should be of such quality that it will all evaporate at ambient temperature whilst exerting sufficient pressure to enable the LPG appliance to operate satisfactorily. Also, the LPG should burn satisfactorily without causing any corrosion or producing any deposits in the system.

This involves the supply of propane in cold climates and either propane or butane in hot climates. In temperate climates (as in Jordan), propane/butane mixtures are used with seasonal variation in their constituent proportions.

LPG is usually available in four grades: Commercial Propane, Commercial Butane, Commercial Propane-Butane Mixtures and Special Duty Propane. Typical specifications, shown in Tables A.1 and A.2 in Appendix A, contain clauses covering volatility, vapor pressure, specific gravity, hydrocarbon composition, sulfur and its compounds, corrosion of copper, residues and water content presented by three famous specification standards. These tables list the specifications limits that should be available in the mixtures; they don't give the exact specifications of a given mixture. This is because the exact specifications and characteristics of any mixture can't be determined without knowing the exact composition of that mixture.

In this research, the Commercial Propane-Butane mixture is used. These mixtures are produced to meet particular requirements and find application as fuels in areas and at times where low ambient temperatures are less frequently encountered. Since the proportions of the constituents of these mixtures are different from place to place and from time to time, it is more advantageous to make the comparison between R-12 and the pure constituents of LPG, propane and butanes. Any mixture of these gases will has an intermediate properties between its components, therefore it is reasonable to make this comparison. In the following sections, the various refrigerant properties are discussed comparatively.

2.3 Thermodynamic Properties :

Thermodynamic properties are the most important properties in selecting refrigerants for any application. A refrigerant is not useful for an application unless its properties fulfills the thermodynamic requirements of that application. The thermodynamic properties of R-12, propane, butane and isobutane are listed in Table 2.1, with discussions on each one as follows.

2.3.1 Boiling Point :

Low boiling temperature at atmospheric pressure (Normal Boiling Point, N.B.P.) of the refrigerant is required for an efficient refrigerant. High boiling point of the refrigerant at atmospheric pressure requires to operate the compressor at high vacuums, which reduces the capacity of the system and lowers the operating cost.

Referring to Table 2.1, propane has the lowest N.B.P. while butanes, especially n-butane, have relatively high N.B.P.s compared to propane and R-12. Therefore, a low boiling point can be obtained, as required, by increasing the proportion of propane in the mixture.

Table 2.1 : Thermodynamic properties of refrigerants [17]

	Boiling point [a] (°C)	Freezing point (°C)	Critical temperature (°C)	Critical pressure (bar)	Latent heat of vapor.[b] (kJ/kg)
R-12	-29.79	-158	112	41.14	165.2
Propane	-42.07	-187.7	96.8	42.54	423.3
Butane	-0.5	-138.5	152	37.94	386.0
Isobutane	-11.73	-160	135	36.45	364.4

[a] At standard atmospheric pressure.

[b] At normal boiling points.

2.3.2 Freezing Point :

Low freezing point of the refrigerant is necessary because the refrigerant should not freeze under required evaporator temperature. The refrigerant must have a freezing point well below the operating evaporator temperature. As shown in Table 2.1, the four refrigerants have a sufficient low freezing points.

2.3.3 Critical Temperature and Pressure :

The critical temperature of a gas is the temperature above which the gas cannot be condensed regardless of the amount of pressure applied. Therefore, the critical temperature of the refrigerant used should be higher than the temperature occurring in the condenser for easy condensation of the refrigerant vapor.

As shown in Table 2.1, the critical temperatures of the four refrigerants are well above the temperature occurring in the condenser. Also the critical pressures for the four refrigerants are much higher than any pressure in the system.

2.3.4 Latent Heat of Vaporization :

The latent heat of vaporization of a liquid (in kJ/kg) is the amount of heat in kJ that is required to vaporize one kg of the liquid at atmospheric pressure, the liquid to be at its boiling point when the operation begins.

Any refrigerant, when evaporating in the evaporator, must absorb heat from within the cooled space exactly equal to its latent heat of vaporization. When a refrigerant has a high latent heat, it will absorb more heat per kg of refrigerant than a refrigerant with a lower latent heat of vaporization (i.e. it has higher refrigerating effect). Thus, if a refrigerant with a high latent heat of vaporization is used, lower refrigerant quantity and/or smaller compressor, condenser and evaporator can be used. As shown in Table 2.1, the latent heats of vaporization for propane, butane and isobutane are comparatively higher than that of R-12.

The following thermodynamic properties are performance parameters for a refrigeration cycle. Therefore, for comparison purposes they are calculated for a theoretical simple saturated cycle at evaporating temperature of -15 °C and a condensing temperature of 30 °C. The cycle performance parameters are shown in Table 2.2 for the four refrigerants and discussed below.

2.3.5 Evaporating and Condensing Pressures :

It is always desirable to have positive pressures in evaporator and condenser for the required temperatures, but the pressures should not be too high above

Table 2.2 Comparative Thermodynamic Performance Characteristics of Refrigerants

Operation on a standard saturated vapor-compression cycle with an evaporating temperature of -15 °C and a condensing temperature of 30 °C.

	Evaporating Pressure (bar)	Condensing Pressure (bar)	Compression Ratio	Refrigerating effect (kJ/kg)	Work (kJ/kg)	COP	Mass Flow Rate (g/s.kW)	Specific volume (m^3/kg)	Volume Flow Rate (L/s)	Power required (W/kW)	Discharge temperature (°C)
R - 12	1.827	7.446	4.08	116.6	24.31	4.79	8.578	0.0914	0.7842	208.5	38
Propane	2.906	10.787	3.71	279.6	68.53	4.08	3.576	0.1542	0.5515	245.1	45
Butane	0.562	2.833	5.04	291.9	67.67	4.31	3.425	0.6407	2.079	231.8	32
Isobutane	0.889	4.078	4.59	262.7	66.34	3.96	3.806	0.0894	0.3403	252.5	32

The power requirement per unit refrigeration is an important consideration from the economic point of view. Low power consumption is always desirable. In Table 2.2, it is seen that propane, butane and isobutane have power consumption per kW of refrigeration slightly higher than that with R-12, but are all acceptable.

2.3.7 Other Cycle Performance Parameters :

A low discharge temperature is highly desirable. When combined with a reasonable compression ratio, a low discharge temperature greatly reduces the possibility of overheating of the compressor and contributes measurably to a maintenance-free long life for the compressor. A low discharge temperature is particularly important when hermitic motor-compressors are employed. As shown in Table 2.2 for the theoretical simple saturated cycle, butane and isobutane have lower discharge temperatures than R-12 while propane has a higher one.

2.4 Physical Properties :

The important physical properties of R-12, propane and butane are listed in Table 2.3. A discussion of each property follows.

Table 2.3 : Physical properties of refrigerants[a] [17]

	Specific heat (kJ/kg.K)		Thermal conductivity (mW/m.K)		Dielectric constant[b]		Viscosity (μ Pa. sec.)	
	liquid	vapor	liquid	vapor	liquid	vapor	liquid	vapor
R-12	0.979	0.602	68.7	9.70	1.74	1.012	213.9	12.60
propane	2.76	1.68	90.8	18.44	1.27	1.009	110.0	8.26
butane		1.72		17.5				7.9

[a] At 300 K and 1 atm.

[b] At ambient temperature and saturated vapour pressure.

2.4.1 Specific Volume :

low specific volume of the refrigerant vapor at the suction into the compressor is always desirable, because it reduces the size of the compressor for the same refrigeration capacity. Reciprocating compressors are always used with low specific volume at suction, while centrifugal compressors are desirable with high specific volume of the refrigerant at suction. As shown in table 2.2, propane has a slightly higher specific volume than R-12 while isobutane has a slightly lower one. Butane has a relatively high suction specific volume, but is acceptable.

2.4.2 Specific Heat of Liquid and Vapor :

The specific heat of a substance is the amount of heat required to raise the temperature of 1 g of that substance by 1 °C. Low specific heat of liquid refrigerant tends to increase the subcooling of liquid (since in this case low amount of heat rejection in the condenser is sufficient to lower the liquid temperature considerably). On the other hand, high specific heat of vapor tends to decrease the superheating of vapor (since in this case high amount of heat is required to raise the temperature of the vapor considerably). Thus they are both desirable and tends to increase the refrigerating effect per kg of refrigerant. When both are found in a single fluid, the efficiency of liquid-suction heat exchanger is much improved.

As shown in Table 2.3, both propane and butane have higher vapor specific heats than R-12. But propane has also higher liquid specific heat than that of R-12. Data on the liquid state of butane is not available.

2.4.3 Thermal Conductivity :

A high thermal conductivity of refrigerant in both liquid and gaseous states is desirable for a high heat transfer coefficient, and thus for more efficient heat transfer in the evaporator and the condenser. As shown in Table 2.3, propane and butane have considerable high thermal conductivities in both liquid and gaseous states compared to R-12.

2.4.4 viscosity :

Low viscosities of refrigerants in both liquid and vapor states are desirable for better heat transfer (higher heat transfer coefficient) in the evaporator and condenser, and low pumping power. Also, pressure drops during flow are small for low viscosity. As shown in Table 2.3, propane and butane have considerably lower viscosities in both liquid and vapor states than R-12.

2.4.5 Dielectric Strength :

The electric resistance of a refrigerant is primarily important in hermetically sealed units in which the electric motor is exposed to the refrigerant. High dielectric strengths are desirable for good electrical insulation. Dielectric strengths of refrigerants are either expressed as ratios relative to that of nitrogen (relative dielectric strength), or as a dielectric constant, defined as the ratio of the capacitance of a condenser (capacitor) filled with a gas or liquid to its capacitance when evacuated.

The dielectric constants for R-12 and propane are given for liquid and vapor states in Table 2.3. It is noticed that propane has slightly lower dielectric strengths than R-12. Dielectric constants for butane are not available.

2.4.5 Leak-Tendency and Leak Detection :

The leakage of the refrigerant outside the system is due to the openings in the joints or flaws in the construction material. The leak tendency of refrigerants should be low. A dense fluid has less tendency to leak than lower density fluid.

Also, the detection of a leak should be easy. The greatest drawback of fluorocarbons is the fact that they are odorless. This, some times, results in a complete loss of costly gas from leaks without being detected. On the other hand, hydrocarbons leaks can be very easily detected by their distinct odour.

2.5 Chemical Properties :

The most important chemical properties for the refrigerants are discussed as follows.

2.5.1 Miscibility with Oil :

Oil is needed in refrigeration systems to lubricate compressor bearings, valves, pistons and cylinders, and the properties of the oil must be suitable for that purpose. If the oil stayed in the crankcase where it belongs, many problems in refrigeration would be eliminated. But it does not. Oil leaves the crankcase by slipping past the piston rings in reciprocating compressors, by entrainment with the refrigerant, and by excessive foaming as refrigerant is released from solution in the oil. A small amount of oil circulating with the refrigerant would be beneficial by lubricating valves, controls, and so on. A large amount, however, may cause problems.

2.5.2 Toxicity :

The effect of refrigerant on the human body is one of the major considerations in the selection of the refrigerant and that is because of the possibility of the leakage of refrigerant from the refrigeration system. Toxic nature of the refrigerant may cause the injury to the human body or death depending upon its percentage in air. It increases suffocation and poisons the air used for breathing.

In Table 2.4, the Underwriters Laboratories Classification of Comparative Hazard to Life of Gases and Vapors is shown for R-12 and the three hydrocarbon refrigerants (with the description of this classification given in appendix B). According to this table, R-12 is considered as non-toxic (Group 6) refrigerant. Propane, butane and Isobutane are classified as either non-toxic or slightly more toxic than Group 6 (Group 5a). Therefore, propane and butanes are safe from this point especially in domestic refrigerators, where small refrigerant charges are used.

2.5.3 Flammability :

Ideal refrigerant should not have any danger of explosion in the presence of air or in association with lubricating oil. R-12 is a non-flammable refrigerant. The main drawback in the refrigerants of hydrocarbon family is that they are highly flammable.

For a domestic refrigeration system, the quantity of refrigerant being considered is small, typically about 1.6% of the hydrocarbons contained in the average butane cylinder; an item which will be found in most houses and which is potentially more dangerous. Such a vessel is fitted with a valve whereas the refrigerator holds the compound in a sealed system.

Table 2.4 : Toxicity and flammability [17]

	Underwriters Laboratories Group Classification	Explosive limits in air (% by volume)	
		Lower limit	Upper limit
R-12	6	Nonflammable	
Propane	5 b	2.3	7.3
Butane	5 b	1.6	6.5
Isobutane	5 b	1.8	8.4

In addition, a large proportion of the hydrocarbon mixture will not be released by the compressor oil in the short term. Hence, any leakage in the domestic refrigerator will not release gas quantity more than that may be released by one oven eye when it is opened for few minutes.

In Table 2.4, the explosive limits of propane, butane and Isobutane in air are given as % by volume. Even a very small kitchen would have a volume of about 20 m^3, which requires much more quantity of hydrocarbons than the one contained in a refrigerator to reach the lower limit listed in the table.

2.5.4 Corrosivity and Action with Materials :

The chemical reaction of the refrigerant on the materials used in refrigeration system is not the prime consideration in the selection of the refrigerant, but the selected refrigerant decides the material to be used for the construction of the system. But if a refrigerant is seeked to replace another used refrigerant, then it should has similar effect on materials as the replaced refrigerant in order to be a successful alternative.

The refrigerant must be non-corrosive in order to use more common materials. Refrigerants must be chemically inert with their system construction materials as well as they must also remain inert in the presence of water or air.

The Freon refrigerants are non-corrosive with all metals, but they become acidic in the presence of air and water. Magnesium and aluminum should not be used with Freon refrigerants as they are readily attacked by acids. Also, Freons have high solvent action on natural rubber so that synthetic rubber is used for gaskets and other sealing purposes to avoid this difficulty.

Propane and butanes, as well as most of the hydrocarbons, are non-corrosive with all metals even in the presence of air and water. On the other hand, they are known to affect rubber but the size of their effect and the types of rubbers affected by them are not specified yet. In this research, a domestic refrigerator was operated on LPG continuously for a period of four months. No rubber damage or any leakage was detected until the date of writing this report.

2.5.5 Chemical Stability :

An ideal refrigerant should not decompose at temperatures normally encountered in the system and should not polymerize. Some refrigerants decompose into gases which do not condense in the condenser and cause high condensing pressures and vapor lock. Sometimes, disintegration of the refrigerant may happen due to the catalytic action of some metal. To avoid this, the refrigerant must be inert with all materials used in refrigeration system.

Freon group is very stable up to 500 °C. But they are unstable above 600 °C and forms corrosive and poisonous products. But such high temperatures are never reached under normal conditions in refrigeration systems.

2.5.6 Action with Water :

Water is a most undesirable contaminant in refrigeration systems. By itself or in combination with air it may cause rusting, corrosion, copper plating, refrigerant decomposition, valve damage, oil sludging, and general deterioration of the system. If water solubility in the refrigerant is exceeded at low temperatures, ice may form in the expansion valve or capillary tube and restrict the flow of refrigerant, or in sever cases, stop it altogether.

Therefore, the solubility of water in the refrigerant should be as low as possible. The solubility of water in fluorocarbons is, in general, low. It ranges from poor, as for R-12, to fairly good, as for R-22. Water is not soluble in hydrocarbons as propane and butane and they don't absorb any moisture from air.

2.5.7 Effect on Stored Products :

The refrigerants used in cold storage plants and in domestic refrigerators should not affect the quality (color, test, etc.) of the material when it comes in contact with stored products. Freons have no effect on foods, meat, vegetables, and dairy products. There will not be any change in color, test or texture of the material which is exposed to Freon.

Hydrocarbons, on the other hand, have specific odours and may, therefore, change the taste of some products (as butter and meat) if subjected to high concentrations of gases for a long enough time. In domestic refrigerators this issue is not a problem since small refrigerant amounts are used and the refrigerator door is usually opened several times a day providing sufficient ventilation for the stored products.

2.6 Other Properties :

There are some other factors that are not critical in deciding which refrigerant to use, but are fairly important in comparing between alternative refrigerants. These factors include cost and availability.

2.6.1 Cost :

When comparing between alternative refrigerants that have similar performance factors, cost may be a critical factor in deciding which refrigerant to use, especially for Developing Countries. In Table 2.5, the costs of R-12, LPG, and R-134a are compared, not to mention also the expensive costs of the new lubricants (ester oils) needed to work with R-134a.

Table 2.5 : Refrigerant costs

Refrigerant	Estimated Cost (JD/kg)
LPG	0.15
R-12	3.0
R-134a	15.0

2.6.2 Availability :

In addition to the cost, the availability of a refrigerant is an important factor. Most developing countries depend on the producing countries in providing them with the synthetic refrigerants, such as R-12 and R-134a, they need and with the price that producers dictate. This situation may be changed if Developing Countries used locally available refrigerants. LPGs are available in enormous

CHAPTER 3

EXPERIMENTAL RIG AND PROCEDURE

In this research, a locally manufactured refrigerator unit was used to test the performance of the two refrigerants: R-12 and a propane/butane (LPG) mixture. By using a single unit for the test of the two refrigerants, any performance mismatch between different units is avoided. Also, the difficulty of providing two refrigerators is avoided, but with the disadvantage of doubling the time of the research. The specifications of the refrigerator used are given in the next section.

3.1 Refrigerator Unit Specifications :

The refrigerator unit used in the research is a simple refrigerator that contains two separate compartments, a fresh food storage compartment and a frozen food storage compartment, as shown in figure (3.1). It does not include defrosting devices or forced air circulation. The specifications of the refrigerator denoted by the manufacturer, are as follows :

Table (3.1): Specifications of the refrigerator unit used.

Trade mark : MISTRAL
Made by : HAMCO (Household Appliances Manufacturing Co., Jordan)
Class : T
System : K
Gross capacity : 320 l (11.3 cf.)
Freezer storage capacity : 65 l (2.3 cf.)
Freezing capacity : 4.5 kg/24 h (158 kcal/h)
Refrigerant R-12 charge mass: 0.21 kg
Nominal input : 150 w
Nominal current and Voltage: 1.2 A (240 volts)
Evaporator temperature range : -10 to -40 °C
Compressor Design : reciprocating (hermetically-sealed)
Compressor displacement size : 8 cc
Capillary tube diameter : 0.6 mm
Capillary tube length : 2.45 m

3.2 Domestic Refrigerator Components and their Functions : [19, 20]

Before any meaningful analysis can be made, the function of the various refrigerator components must be fully understood. The refrigeration system used in this research is a static or gravity type condenser system. It consists of an evaporator, compressor, condenser, capillary tube, and a filter-drier all connected with tubing through which the refrigerant circulates.

The refrigeration system is divided into two parts; high pressure and low pressure sides. The dividing points are at the compressor and at the outlet of the capillary tube.

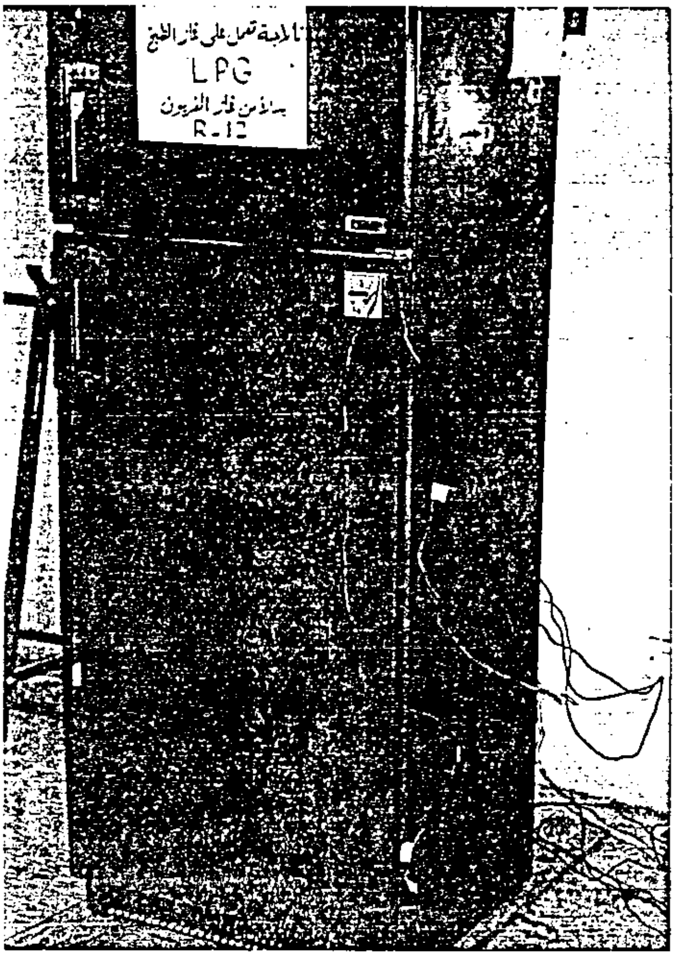

Figure (3.1): The refrigerator unit used in the research.

The low pressure side of the refrigeration system consists of the evaporator, the suction line, and part of the compressor. The high pressure side of the system consists of the condenser, discharge line, filter-drier, capillary tube, and part of the compressor. Each component in the refrigeration system, its function and how it works will be described briefly, as follows:

3.2.1 Evaporator :

The evaporator, interchangeably called the cooling coil, freezer coil, or just plain freezer, is where the action is. The heat in the cabinet is transferred to the refrigerant that is circulating through the evaporator by conduction and convection currents. As the refrigerant absorbs heat from the foods, it begins to boil, changing its state from a low pressure liquid to a low pressure heat-laden vapor. This vapor is then drawn through the accumulator and suction line to the compressor. The plate-surface type evaporator is widely used in household refrigerators and home freezers because it is easily cleaned, economical to manufacture, and can be readily formed into any shape required.

3.2.2 Compressor :

The compressor and motor are combined in a hermetically sealed steel casing with no exposed moving parts. All the parts are continuously lubricated internally, eliminating the need for periodic service lubrication. The motor operates on a single-phase alternating current. It is a split-phase motor having two sets of windings, a start winding and a run winding. The speed of compressor motors ranges from 1725 rpm to 3450 rpm depending on the type of stator used in the motor.

3.2.3 Condenser :

The function of the condenser is to transfer the heat absorbed by the refrigerant to the ambient or room air. The high pressure vapor (gas) is routed to the condenser where, as the vapor temperature is reduced, it condenses into a high pressure liquid state. This heat transfer takes place because the discharged vapor is at a higher temperature than the air that is passing over the condenser. Since the condensers used in domestic refrigerators and freezers are air-cooled, natural convection type condensers, a relatively large condensing surface is required. It is very important that an adequate air flow over the condenser be maintained. Any accumulation of lint or other material covering the condenser should be removed to assure normal air flow.

3.2.4 Capillary tube :

The capillary tube is the simplest type of refrigerant flow control device used in modern refrigeration systems. It consists merely of a fixed length of small diameter tubing installed between the condenser and the evaporator. Because of the high frictional resistance resulting from its length and small bore and because of the throttling effect resulting from the flashing of the liquid refrigerant into vapour as the pressure of the liquid is reduced below its saturation pressure, the capillary tube acts to restrict or meter the flow of liquid from the condenser to the evaporator and also to maintain the required operating pressure differential between those two units.

Other than its simple construction and low cost, the capillary tube has the following additional advantages :

1. Because the high and low pressures equalize through the capillary tube during the off cycle, the compressor starts in an unloaded condition, which allows the use of a low starting torque motor to drive the compressor.

2. The small and critical refrigerant charge required by the capillary tube system results not only in reducing the cost of the refrigerant but also in eliminating the need for receiver tank.

All these factors represent a substantial savings in the manufacturing costs. Thus, capillary tubes are employed almost universally on all types of domestic refrigeration units.

3.2.5 Heat exchanger :

Bonding (soldering) the capillary tube to the suction line for some distance in order to provide a heat transfer relationship between the two is usually desirable. This heat exchange between the warm liquid refrigerant in the capillary and the cold suction vapor tends to minimize the formation of flash gas in the capillary tube, by cooling it, which enhances the flow capacity of the tube. In the same time, this heat exchange helps to evaporate any liquid present in the suction line. The final result is an increase in the overall efficiency of the refrigeration unit.

3.2.6 Filter-drier :

Because of the small bore in the capillary tube, it is essential that the system be kept free from dirt and foreign matter. Usually a high side filter-drier is located at the outlet of the condenser before the capillary tube. It filters foreign matter that might restrict the refrigerant flow through the system and it removes moisture that might freeze in the capillary tube, causing a restriction.

3.3 Measuring Devices and Procedures :

The variables that was measured during the experiments are : temperature, pressure, power consumption, and time intervals.

3.3.1 Temperature measurement :

For taking temperature measurements, copper-constantan (Type T) thermocouples were used. They were connected to nine points in the refrigerator and three outer points. In the refrigerator, the surface temperatures at seven points were measured, which are :

1. The freezer cabinet evaporator, T1.
2. The refrigerator cabinet evaporator, T2.
3. The middle of the condenser, T3.
4. The end of the condenser, T4.
5. The beginning of suction line, before the heat exchanger, T5.
6. The end of suction line, at the inlet of the compressor, T6.
7. The discharge line, T7.

These points are shown on the refrigeration system schematic diagram, in figure (3.2), and the pressure-enthalpy (p-h) diagram, in figure (3.3).

The other two thermocouples, T8 and T9, were connected to measure the space (air) temperature inside the refrigerator compartment and the freezer compartment.

Two of the three outer thermocouples were connected to the load used in the experiments (a container filled with hot water) as follows :

10. The surface of the metal container, T10.
11. Inside the water load, T11.

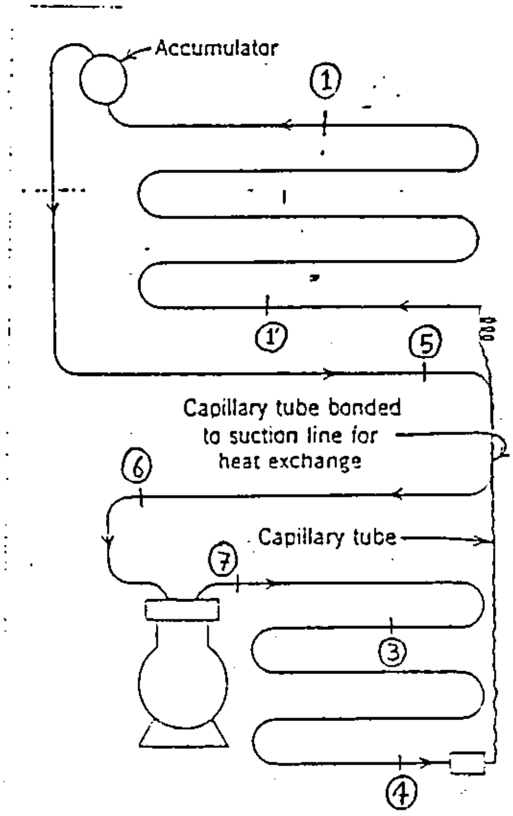

Figure (3.2): Refrigerator system schematic diagram.

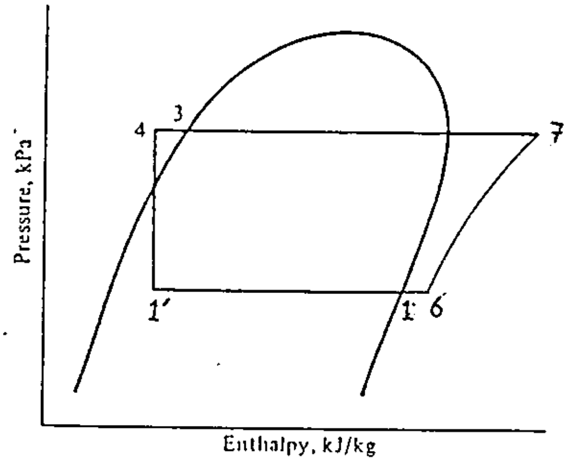

Figure (3.3): p-h diagram of the refrigeration system.

The third outer thermocouple, T12, was used to measure the ambient temperature in the vicinity of the refrigerator unit.

The other ends of the thermocouples were connected to a Data Logging System (3530 ORION Data Logging System, made by Schlumberger Electronics Ltd., UK) with an accuracy of 0.01 °C. This data logger contains a powerful software which provides a comprehensive and flexible logging capability. It has a display with a front panel controls, a built-in printer and a cartridge recorder. A maximum of 200 inputs can be received by the logger.

3.3.2 Pressure measurement :

Pressure was measured using a gauge manifold which is comprised of a compound gauge, a pressure gauge, and the valve manifold, as shown in figure (3.4). The compound gauge is used to measure pressures both above and below atmospheric (vacuum). In practice it is used to determine pressures in the low side of the refrigeration system and was connected to the suction line. The pressure gauge, which is used to determine pressures on the high side of the system, was connected to the discharge line.

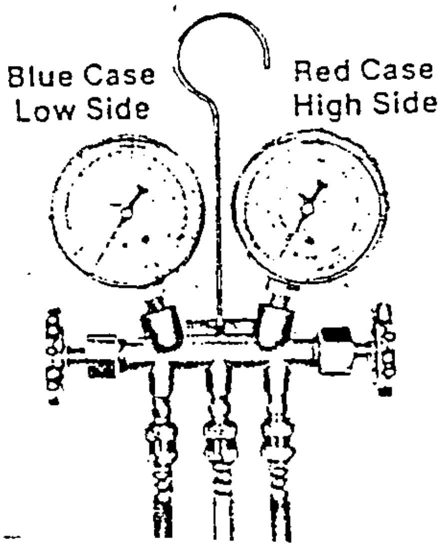

Figure (3.4): Refrigeration gauge manifold

3.3.3 Power consumption measurement :

To measure the actual power consumed by the motor-compressor of the unit, a KW.hr meter was connected to the electric mains. This meter is a single phase power meter with an accuracy of 0.01 KW.hr .

3.3.4 Time :

For the simulated load experiments, readings were taken at specified time intervals (usually 10 minutes) which were controlled by the Data Logging System. Measuring time intervals is important for calculating the rate of heat .removal from the load, as will be discussed in chapter 4.

3.4 Propane/Butane (LPG) Mixture :

The propane/butane mixture used in the tests was obtained from ordinary commercial LPG bottle which is filled in the Jordanian Petroleum Refinery. A chemical analysis was made, in the laboratory of the Jordanian Petroleum Refinery, by the gas chromatography technique to determine the composition of the gas used in the experiments. The composition was as follows :

Table (3.2): LPG composition.

component	% by volume
propane	26.4
n-butane	56.4
Isobutane	17.2

3.5 Experimental Work Procedure :

Since the same refrigerator unit is used in the tests of the two refrigerants, the work was divided into two stages; the first stage for the tests on R-12 and the second one for the tests on the propane/butane mixture. Each stage experiments was performed as follows :

3.5.1 First stage :

In this stage the refrigerator was operated with its original refrigerant R-12. Two types of tests were done, which are :

(1) Evaporator temperature variation test :

In order to make performance curves for the different refrigeration cycle parameters versus the evaporator temperature (Te), Te was varied and measurements were taken at different values of Te. To perform this, a simulated load which consists of a metal container of known specific heat and mass was filled with a specified quantity of hot water. After connecting thermocouples 10 and 11 to it, it was placed in the freezer compartment. Placing this load in the freezer compartment (which contains the main evaporator) will cause Te to increase rapidly to a maximum value then starts to decrease slowly until it reaches its steady state again. During this period of Te variation, temperatures at the nine system locations (T1 through T9) are recorded in addition to the load temperatures at a fixed time intervals. Also during this period the power consumption is taken during the same intervals. From these readings and using R-12 tables and charts, the state at each point in the system can be determined fully, as will be shown in the next chapter.

(2) Condenser temperature variation test :

The condenser temperature (Tc) was varied and measurements were taken, in order to make performance curves versus Tc. The variation of Tc was performed by spraying cold water over the condenser and making a forced air stream over it, using a small fan, in order to cool the condenser by evaporative cooling. Also a heater was used to raise the condenser temperature (only in R-12 tests). During the periods of Tc variation, the same readings as in the evaporator temperature variation test were taken (except thermocouples 10 and 11, since no load was used here).

3.5.2 Second stage :

In this stage, tests were performed on the propane/butane (LPG) mixture. First, the refrigerator was evacuated from R-12 and then charged with the LPG mixture and operated on it for 24 hours. After that, it was evacuated again to insure that no R-12 traces are left in the system, and then charged with a specified quantity of LPG mixture.

The refrigerator was first charged with 100 g of LPG mixture, then the same experiments described above in tests (1) and (2) were performed. Pressure readings were recorded for both low and high pressure sides of the system during all experiments.

After finishing all tests on the 100 g charge, the LPG refrigerant was increased in steps of 25 g until reaching a charge of 250 g. The same experiments were performed for each of the seven LPG charge quantities (i.e. 100, 125, 150, 175, 200, 225 and 250 g). The objective of doing all these experiments is to determine

CHAPTER 4

MATHEMATICAL ANALYSIS

4.1 Data Readings :

As discussed in section 3.3 of the previous chapter, the readings taken during the experimental work are: temperature readings (in °C), pressure readings (in psi), power consumption readings (in kW.hr) and time intervals (in minutes). Temperature readings are taken at the locations mentioned in section 3.3.1.

The data which were used in the calculations are listed in the tables of Appendix C for T_e variation and T_c variation experiments, for both R-12 and LPG refrigerants.

4.2 Calculations :

From the data readings collected, the various performance parameters of the refrigeration system are calculated. The methods and equations used for calculating these parameters are discussed in this section. The results of the various calculations are plotted and discussed in the following chapter.

In the following subsections the different refrigeration quantities and parameters calculated are represented and discussed.

4.2.1 Enthalpy calculations : [21]

In order to calculate any refrigeration cycle parameter, enthalpies at different locations in the cycle have to be calculated first. These locations include: inlet and outlet of the evaporator, compressor and condenser.

For R-12, temperature and pressure readings are enough to calculate the enthalpy at any location in the cycle using R-12 saturated and superheated tables. But for the propane/butane (LPG) mixture, such tables does not exist. Therefore, further calculations have to be made.

The only way available to calculate the enthalpy of a LPG mixture is to calculate first the enthalpies of its main constituents (propane, butane and isobutane) and then calculate the enthalpy of the mixture using these enthalpies and the mass fraction of each constituent in the mixture. The mass fractions of the three constituents are calculated from the volume fractions, listed in section 3.4, as shown in table 4.1.

Table (4.1): Mass fractions of the LPG mixture constituents.

component	percent by volume	Mole fraction	Molecular weight	Mass (kg/kmol)	Mass fraction
propane	26.4	0.264	44.10	11.64	21.39
butane	56.4	0.564	58.13	32.78	60.23
Isobutane	17.2	0.172	58.13	10.00	18.38

Hence, the enthalpy of the LPG mixture at any state may be calculated using the mass fractions shown in the last column of table 4.1, using the following equation:

$$h_{mixture} = 0.2139\, h_P + 0.6023\, h_B + 0.1838\, h_I \qquad (4.1)$$

where h_P, h_B and h_I are the enthalpies of propane, butane and isobutane respectively.

For the saturated state points, the enthalpies of the pure constituents are obtained at the saturated temperature using the propane, butane and isobutane saturated tables. Then the enthalpy of the mixture is calculated from eqn.(4.1).

For the superheated state points (at suction and discharge of compressor), an additional property is needed in order to calculate the enthalpy. Therefore, pressure measurements are taken for the low and high pressure sides of the system, as described earlier in section 3.3.2. By knowing the pressure and temperature of each constituent, the superheated enthalpy is obtained using propane, butane and isobutane superheated charts and the enthalpy of the mixture is calculated from eqn.(4.1).

The partial pressures of the LPG constituents are obtained from the total pressure of the gas mixture (the measured pressure) using the mole fraction of each constituent as follows:

$$\begin{aligned} P_P &= 0.264 \times P_t \\ P_B &= 0.564 \times P_t \qquad (4.2) \\ P_I &= 0.172 \times P_t \end{aligned}$$

where P_t is the total pressure of the mixture, and P_P, P_B and P_I are the partial pressures of propane, butane and isobutane respectively in the mixture.

The enthalpy of LPG in the evaporator inlet is taken to equal that at condenser exit of liquid saturated conditions, due to the adiabatic throttling process.

4.2.2 Refrigerating effect and refrigeration capacity : [22]

The refrigerating effect is the quantity of heat absorbed from the enclosed refrigerated compartment by the evaporator in kJ per kg of refrigerant circulated. It is thus given by:

$$q_e = h_1 - h_4 \tag{4.3}$$

where, h_1 and h_4 are the enthalpies (in kJ/kg) of the refrigerant leaving and entering the evaporator, respectively.

The refrigeration capacity is the rate of heat removal (in kW) from the refrigerated compartment by the evaporator. It is calculated by multiplying the refrigerating effect by the mass flow rate of the refrigerant in the evaporator, thus:

$$Q_e = m\, q_e = m\,(h_1 - h_4) \tag{4.4}$$

where, Q_e is the refrigeration capacity in kW and m is the refrigerant mass flow rate in kg/s.

Usually, the refrigerant mass flow rate is difficult to measure directly; therefore, it is easier to measure the refrigeration capacity and then calculate the mass flow rate from eqn.(4.4) :

$$m = Q_e / q_e \qquad (4.5)$$

The refrigeration capacity is calculated by measuring the rate of heat removal from a simulated load (which consists of a metal container filled with hot water), using the following equation:

$$Q_e = (M_W \, Cp_W \; \Delta T_W + M_c \, Cp_c \; \Delta T_c) / \Delta t \qquad (4.6)$$

where, Q_e is the refrigeration capacity in kW.

M_W and M_c are the masses of the water load and the container in kg.

Cp_W and Cp_c are the specific heats of the water and container in kJ/kg.°C.

ΔT_W and ΔT_c are the temperature differences in °C of water and container during the time period Δt (in seconds).

4.2.3 Work of compression and power consumption : [22]

The work of compression done by the compressor on the refrigerant is represented by the increase in refrigerant enthalpy during the compression process, that is:

$$W = h_7 - h_6 \qquad (4.7)$$

where W is the work of compression in kJ per kg of refrigerant; h_6 and h_7 are the enthalpies of refrigerant at compressor suction and discharge respectively.

The COP is thus a measure of the performance of the refrigeration system, since it indicates the amount of refrigeration capacity provided by the system per unit of power consumption.

4.3 Sample Calculation for the LPG Mixture :

A sample calculation is presented for the 200 g LPG charge using one set of readings from the data tables in appendix C. The readings are :

P_L = low side total mixture pressure = -8 psig (0.462 bar)
P_H = high side total mixture pressure = 45 psig (4.114 bar)
T_1 = evaporator temperature = -6.6 °C (266.6 K)
T_6 = suction vapor temperature = -3.9 °C (269.3 K)
T_7 = discharge vapor temperature = 32 °C (305.2 K)
T_4 = condenser temperature = 26.3 °C (299.5 K)

Assuming that we have a saturated vapor at T_1 (although it is slightly superheated, the error of considering it saturated vapor was found to be 0.8 %, which may be neglected), then the enthalpy of each component is obtained from the saturated vapor tables of propane, butane and isobutane at the temperature of 266.6 K . Or it may be considered superheated and calculated using the components partial pressures, which are calculated using eqn.(4.2), and then using the p-h charts of propane, butane and isobutane. (As was mentioned, the error between the two approaches is about 0.8 % only).

For the two superheated state points at the suction and discharge of the compressor, the suction and discharge temperatures in addition to the suction and discharge components partial pressures are used to calculate the enthalpies of the components in the superheated states using the p-h charts of the components.

At the outlet of the condenser (at T_4), the refrigerant is considered in a subcooled liquid state. In this case, the high side partial pressures of the components and the temperature of $T_4 = 26.3$ °C are used to obtain the enthalpies of the components from their p-h diagrams. Or the refrigerant may be assumed as a saturated liquid at this point, and in this case the saturated liquid tables for propane, butane and isobutane are used to obtain their enthalpies at $T_4 = 26.3$ °C. The two methods gave almost the same results.

After calculating the enthalpies of the propane, butane and isobutane components at the four points (i.e. points 1, 6, 7 and 4) as described above, eqn.(4.1) is used to calculate the enthalpy of the mixture at any of the mentioned points. The calculations and results regarding the data readings mentioned above are summarized in the following table :

Table (4.2): Sample calculation for the 200 g LPG charge.

	$P_{L,i}$ (bar)	h_1 (kJ/kg)	h_6 (kJ/kg)	$P_{H,i}$ (bar)	h_7 (kJ/kg)	h_4 (kJ/kg)
Propane	0.021	891.1	910.6	0.184	965.0	605.5
Butane	0.338	664.3	671.9	3.01	723.1	363.6
Isobutane	0.103	663.9	671.1	0.920	725.8	392.2
Mixture	0.462	712.9	722.9	4.114	775.4	420.6

After calculating the enthalpies of the mixture at each point in the cycle, the system parameters may be obtained from equations (4.3) through (4.10), except eqn.(4.6). From these equations and the results listed in the table above, the following parameters are calculated : q_e = 292.2 kJ/kg , W = 52.52 kJ/kg and COP = 5.564 ; other parameters may be calculated as well, using their equations listed in section 4.2 above.

4.4 Comparison with Other Methods of Analysis :

Until now, all the enthalpy calculations were based on utilizing thermodynamic tables and charts of refrigerants, which are practical and reliable. Nevertheless, other methods of calculating the enthalpy of real gas mixtures exist. One of these methods is the use of the generalized enthalpy chart.

Following is a sample calculation that will be performed on the LPG mixture using the generalized enthalpy chart [21], and the result will be compared to that calculated from refrigerant tables and charts. The enthalpy difference between two points in the system is going to be calculated for a real LPG mixture as follows:

The enthalpy change for a real gas between two points is given as : [21]

$$h_2 - h_1 = (h_1^* - h_1) - (h_2^* - h_2) + (h_2^* - h_1^*) \qquad (4.11)$$

where $(h_2^* - h_1^*)$ is the ideal gas enthalpy change. Hence, the quantity $\{(h_1^* - h_1) - (h_2^* - h_2)\}$ represents the difference between the ideal and the real gas enthalpy changes. This quantity is calculated using Kay's rule and the generalized enthalpy chart as follows:

Kay's rule for pseudo-critical pressure and temperature states that : [21]

$$P'_c = \Sigma\, y_i\, P_{c,i}$$
$$T'_c = \Sigma\, y_i\, T_{c,i} \qquad (4.12)$$

where, P'_c and T'_c are the pseudo-critical pressure and temperature for the gas mixture, respectively. y_i is the mole fraction for component i. $P_{c,i}$ and $T_{c,i}$ are the critical pressure and temperature for component i, respectively.

Applying eqn.(4.12) for our LPG mixture (mole fractions of the components are listed in section 3.4 and their critical values in table 2.1) :

$$P'_c = 0.264\ (4.26) + 0.564\ (3.8) + 0.172\ (3.65) = 3.896\ \text{MPa}$$

$$T'_c = 0.264\ (370) + 0.564\ (425.2) + 0.172\ (408.2) = 407.7\ \text{K}$$

Now, the two points selected are the suction and discharge points of the data readings mentioned in the previous section, and will be denoted by points 1 and 2 respectively. The temperatures and pressures for the two points are listed above. The pseudo-reduced temperatures and pressures for the two points are given as:

$$T'_{R,1} = T_1 / T'_c = 269.3/ 407.7 = 0.66$$

$$P'_{R,1} = P_1 / P'_c = 0.0462/ 3.896 = 0.012$$

$$T'_{R,2} = T_2 / T'_c = 305.2/ 407.7 = 0.75$$

$$P'_{R,2} = P_2 / P'_c = 0.4114/ 3.896 = 0.105$$

These pseudo-reduced pressures and temperatures will be used to obtain $(h_1^* - h_1)$ and $(h_2^* - h_2)$ terms from the generalized enthalpy chart.

It is noticed that $P'_{R,1}$ is very low, and since at low pressures real gases behave as ideal ones (i.e. as $P_R \rightarrow 0$, then $h \rightarrow h^*$). Therefore, at point 1, $(h_1^* - h_1) \approx 0$. This fact is also confirmed by the generalized enthalpy chart. For point 2, and from the generalized enthalpy chart it is found that :

$$(h_2^* - h_2)_{mol} / T_c \approx 1.6$$

$$\therefore \quad (h_2^* - h_2) = 1.6 \times T_c / M_m = 1.6\ (407.7)/ 54.42 = 11.99$$

where, M_m is the molecular weight of the mixture. The ideal gas enthalpy change, $(h_2^* - h_1^*)$, may be calculated as:

$$(h_2^* - h_1^*) = C_{p,m}\ (T_2 - T_1) = 1.70 \times (305.2 - 269.3) = 61.03 \text{ kJ/kg}$$

where, $C_{p,m}$ is the specific heat of the mixture. Substituting the above results into eqn.(4.11) :

$$h_2 - h_1 = 0 - 11.99 + 61.03 = 49.04 \text{ kJ/kg}$$

In the previous sample calculation, it was found that :

$$h_2 - h_1 = 775.4 - 712.9 = 52.52 \text{ kJ/kg}$$

Hence, the difference between the two approaches is :

$$\text{Difference} = (52.52 - 49.04)/\ 52.52 = 6.6\ \%$$

Therefore, since the difference is below 10 %, using the refrigerant gases tables gives the actual enthalpies. Using the generalized enthalpy chart is another way and cannot be considered as an improvement due to the considerably low pressures that exist in the system.

CHAPTER 5

RESULTS AND DISCUSSION

5.1 Introduction :

The discussion of the results of this research is divided into two main parts. The first part discusses the behavior of seven different propane/butane (LPG) mixture charge quantities and compares between their performance to find out which quantity is best appropriate to be charged in the used domestic refrigerator.

The second part discusses and compares the performance of the best suited LPG charge and the original refrigerant for which the unit is designed to work with, R-12. Of course, this part is the most important one, since it proves if the propane/butane (LPG) mixture has a competent performance compared to R-12 or not; shortly speaking, it decides if the LPG refrigerant is an appropriate alternative for R-12 in domestic refrigerators and other small refrigeration systems or not.

Two of the most important performance characteristics of a refrigeration system are its refrigeration capacity and its power requirement. These two characteristics are controlled largely by the suction and discharge pressures, and thus the evaporating and condensing temperatures, of the system. Therefore, it is a major trend in the refrigeration literature and industry to make system performance curves with respect to these two temperatures.

5.2 LPG Performance Parameters Versus T_e and Charge Quantity :

As stated earlier in chapter 3, seven different LPG charge quantities were tested to obtain which quantity has the best performance. The performance parameters for these quantities are plotted with respect to the evaporating temperature at T_c = 28.7 °C and $T_{amb.}$ = 19.5 °C. These parameters will be discussed separately as follows.

Also, some performance parameters are plotted versus the LPG charge quantity for different T_e values to show clearly the effect of charge quantity on these parameters.

5.2.1 Refrigerating effect :

In fig.(5.1) the refrigerating effects for the different charges are plotted with respect to the evaporating temperature, T_e. The refrigerating effect increases slightly with the increase in evaporating temperature for all charge quantities. This is due to the slightly higher saturated vapor enthalpy leaving the evaporator. Therefore, at a fixed condensing temperature, this will tend to increase the refrigerating effect, since $q_e = (h_{out} - h_{in})_{evap.}$.

In comparing between the different charges, the figure shows that as the charge quantity increases, the refrigerating effect decreases. This may be due to that increasing the charge will increase the pressures in the system with the high side pressure being more affected than the low side one (as was noticed in the experiments). Therefore, the enthalpy of the refrigerant entering the evaporator will increase more than the one leaving it, which will cause the refrigerating effect to decrease.

In figure (5.2), the refrigerating effect is plotted versus the LPG charge quantity and the effect of the LPG refrigerant charge quantity on the refrigerating effect is more clear.

5.2.2 Refrigeration capacity :

The result above agrees with the refrigeration capacity curves, shown in figure (5.3), where the same behavior is recognized. That is, the refrigeration capacity decreases as the charge quantity increases. The refrigeration capacity was obtained by calculating the heat removal rate from a simulated load placed in the freezer compartment, as described in chapter 4, which makes it independent from the refrigerating effect calculations.

5.2.3 Work of compression :

The work of compression curves are shown in figure (5.4) for the different LPG charges. As the evaporating temperature increases, at a constant condensing temperature, the work will decrease due to that the suction pressure and temperature (and thus the suction refrigerant enthalpy) will increase while keeping the discharge enthalpy constant. Therefore, this will cause the work to be reduced as T_e increases, since $W = h_{discharge} - h_{suction}$.

The effect of the charge quantity on the compression work can be seen in the figure. From this figure it is noticed that at low evaporating temperatures (< -5 °C), increasing the charge quantity will decrease the work of compression until it reaches a minimum value of 51.5 kJ/kg, at 200 g charge, and then starts to increase again as the charge increases. This behavior is more clear in figure (5.5) where the work is plotted versus the charge quantity.

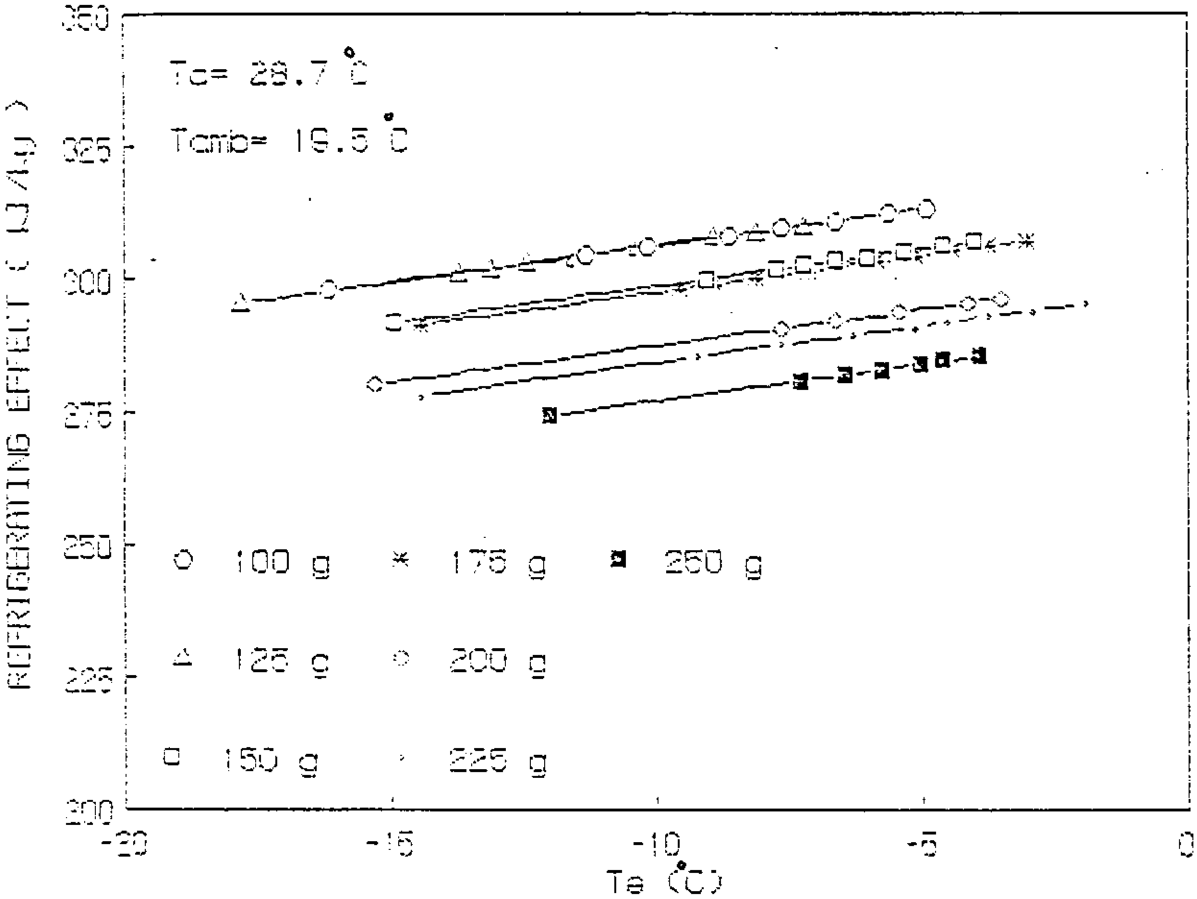

Fig.(5.1) : LPG refrigerating effects vs. Te

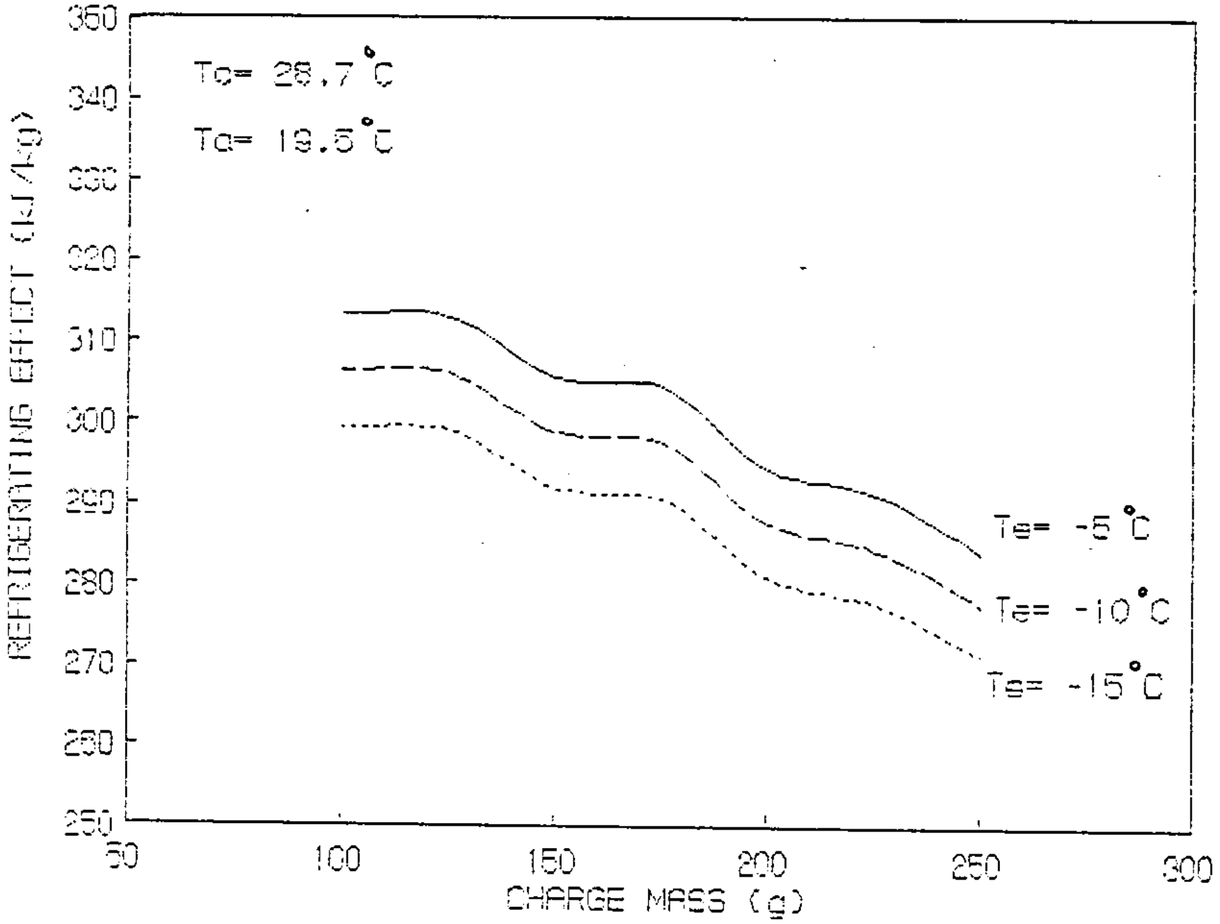

Fig.(5.2) : Refrigerating effect vs. LPG charge quantity

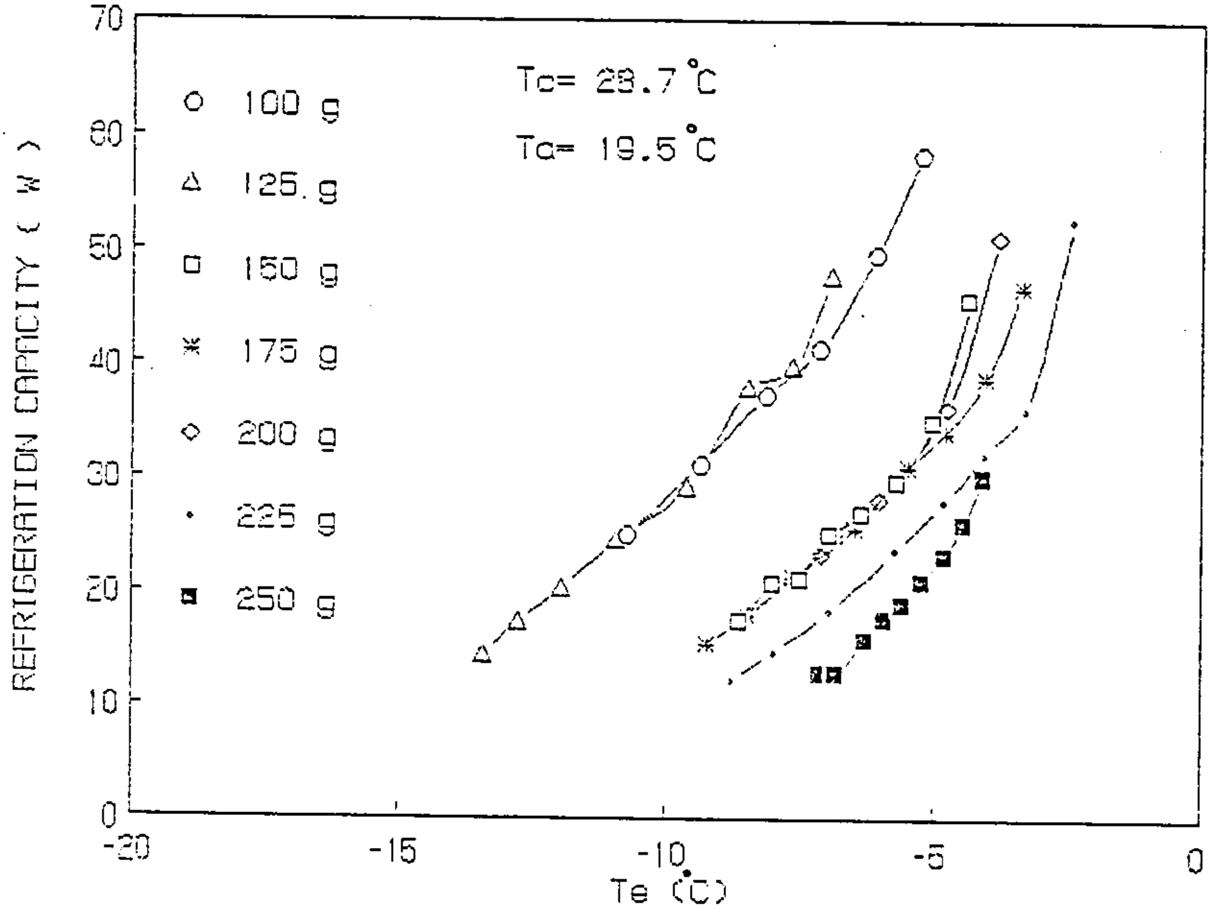

Fig.(5.3) : LPG refrigeration capacity vs. Te

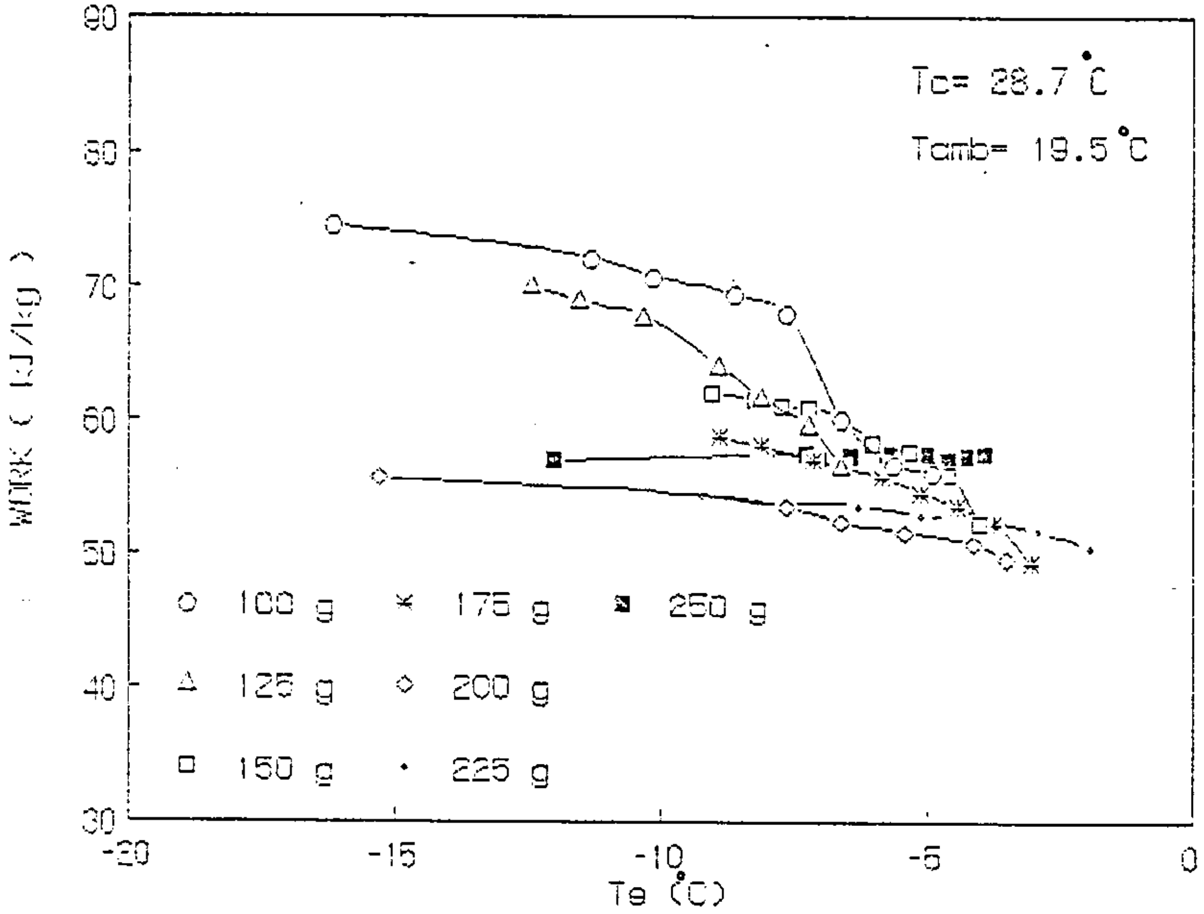

Fig.(5.4) : Compression works of LPG vs. Te

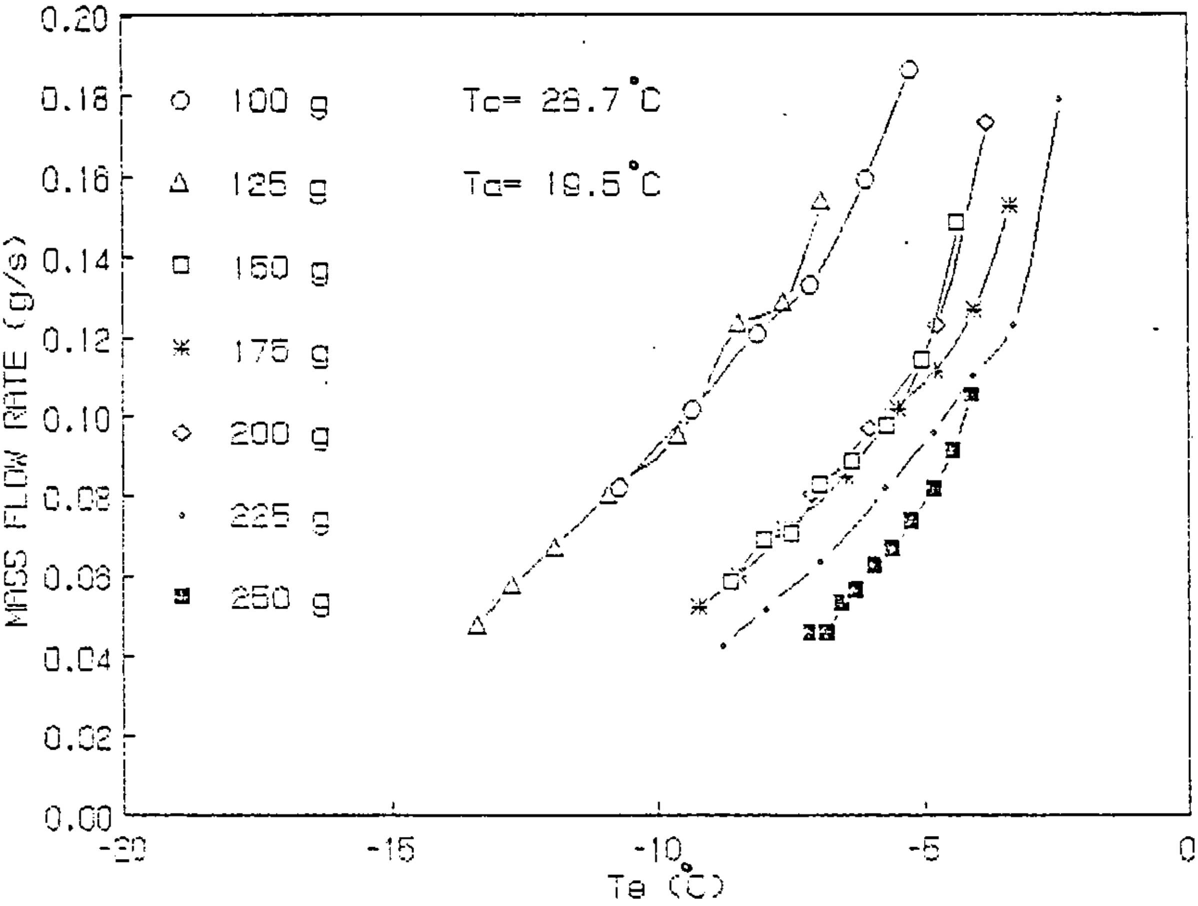

Fig.(5.9) : LPG actual mass flow rates vs. Te

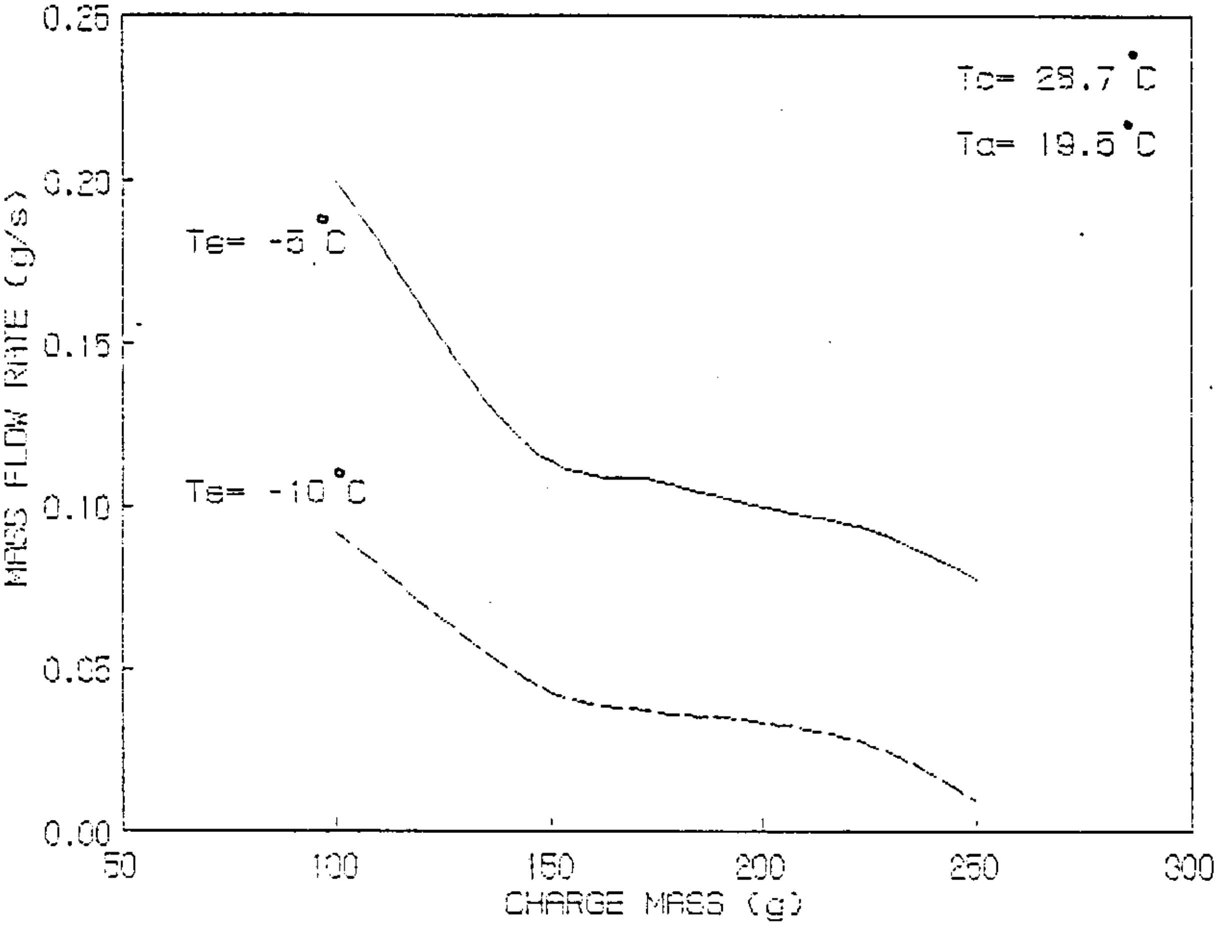

Fig.(5.10) : Actual mass flow rate vs. LPG charge quantity

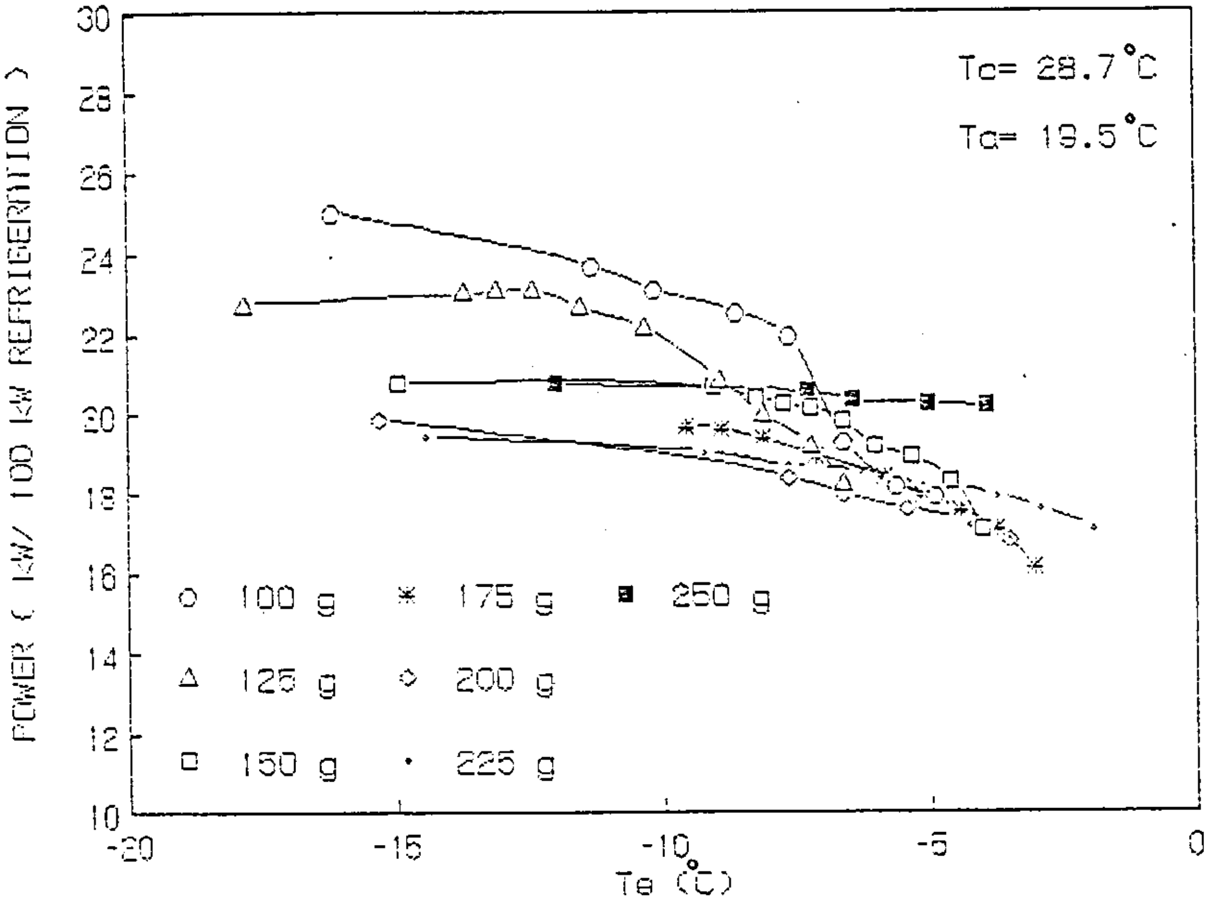

Fig.(5.11) : LPG power consumption per 100 kW refrigeration vs. Te

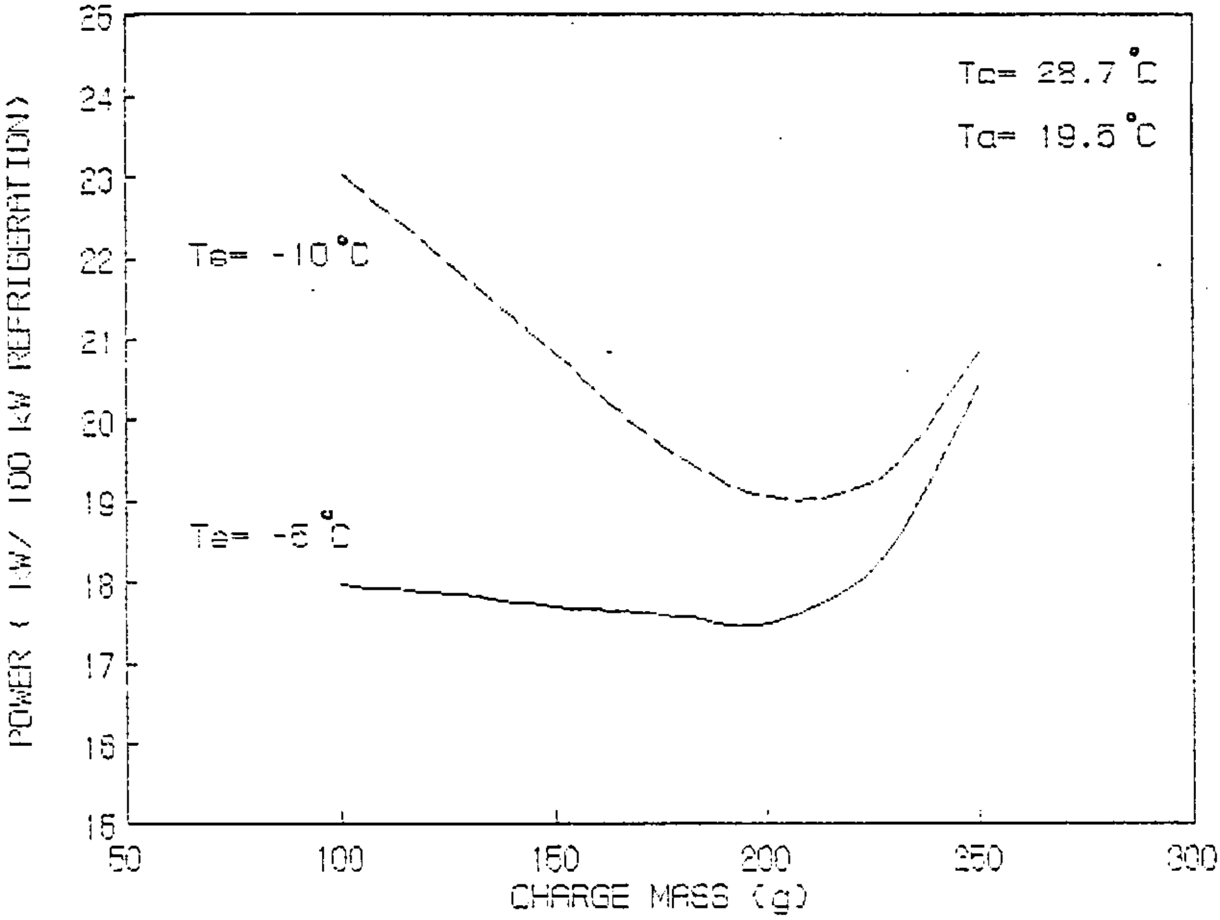

Fig.(5.12) : Power consumption per 100 kW refrigeration vs. LPG charge quantity

5.2.8 Actual power consumption :.

The actual power consumptions (in W) of the different charges are shown in figure (5.13). Theoretically, the power consumption curve with respect to T_e has two zero points (at mass flow rate = 0 and at $T_e=T_c$ points), and between the two, it reaches its peak value. Practically, most refrigeration systems operate on the left side of the peak of the power curve (i.e. where the power increases with T_e). This behavior is noticed in the power consumption curves shown in the figure.

Due to the decrease in the mass flow rate as the charge quantity increases, as was shown in fig.(5.10), the power consumption will also decrease as the charge quantity increases, as shown in figure (5.14).

5.3 LPG Performance parameters versus T_c :

The performance curves for the different LPG charges versus T_c are taken at T_e = -15.6 °C and $T_{amb.}$ = 19.5 °C.

5.3.1 Refrigerating effect :

The other set of curves are plotted with respect to the condensing temperature (which is directly proportional to the discharge pressure). Figure (5.15) shows the refrigerating effects of the different LPG charges versus T_c. It is noticed that increasing the condensing temperature at constant evaporating temperature causes the refrigerating effect to decrease. This is because this results in increasing the enthalpy of the refrigerant entering the evaporator while keeping the one leaving the evaporator constant. Referring to eqn.(4.3), this will cause a decrease in the refrigerating effect.

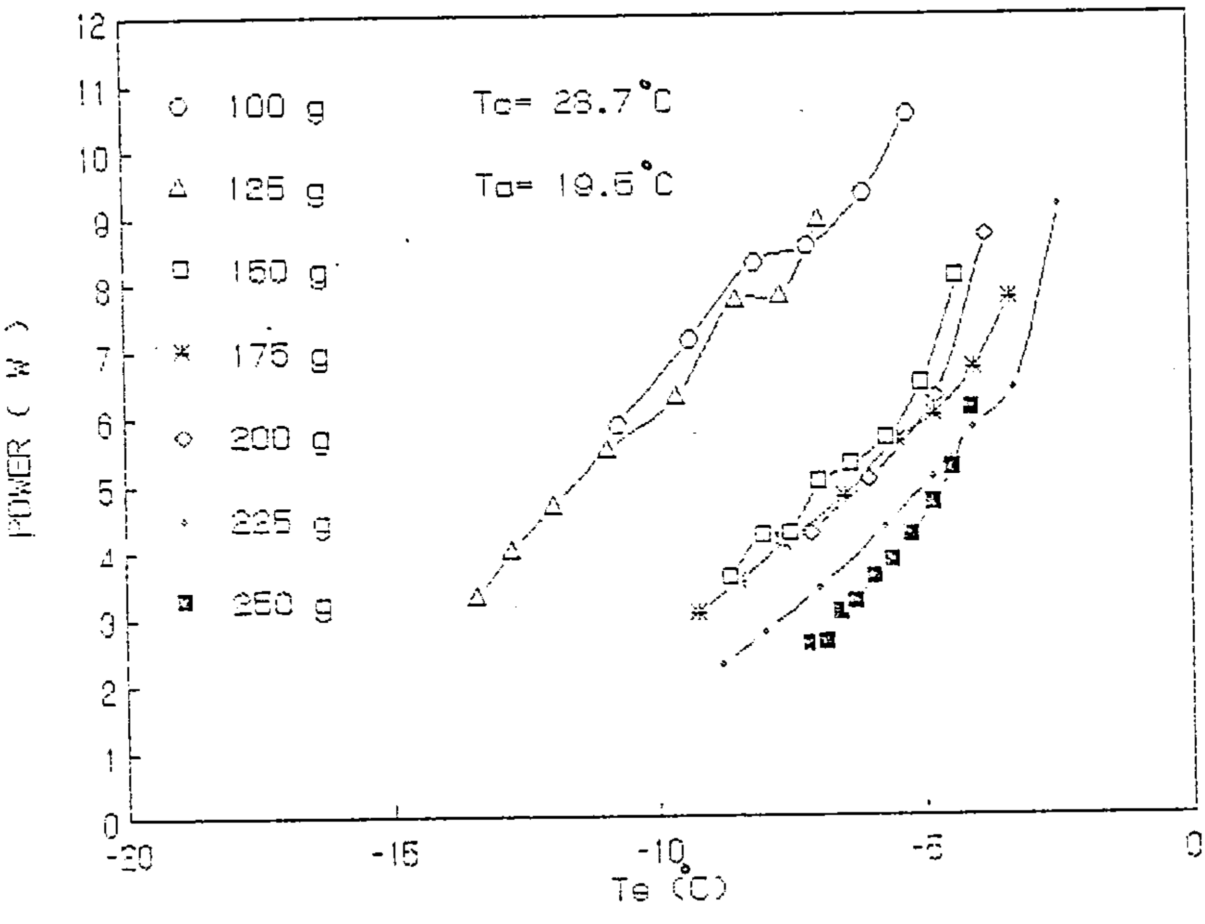

Fig.(5.13) : LPG actual power consumptions vs. Te

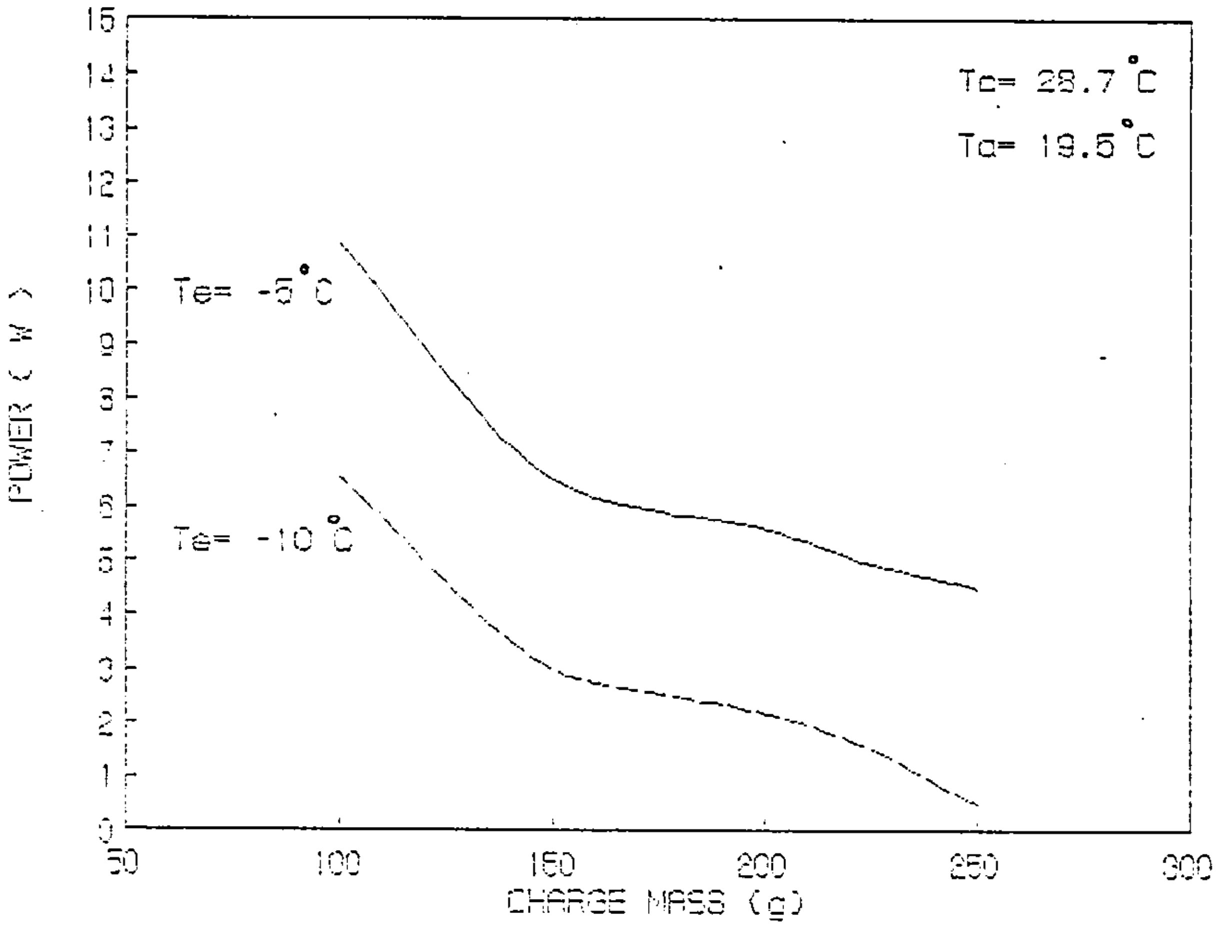

Fig.(5.14) : Actual power consumption vs. LPG charge quantity

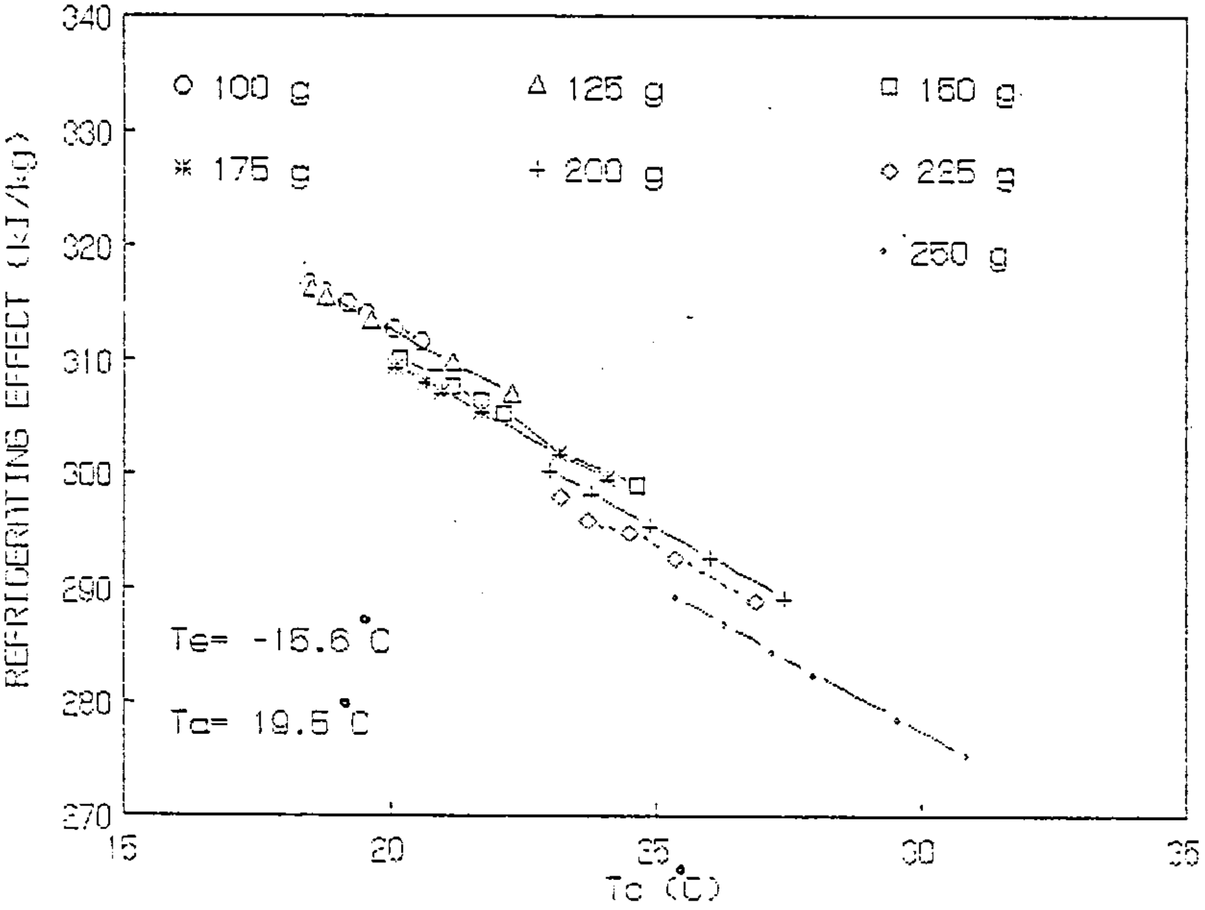

Fig.(5.15) : LPG refrigerating effects vs. Tc

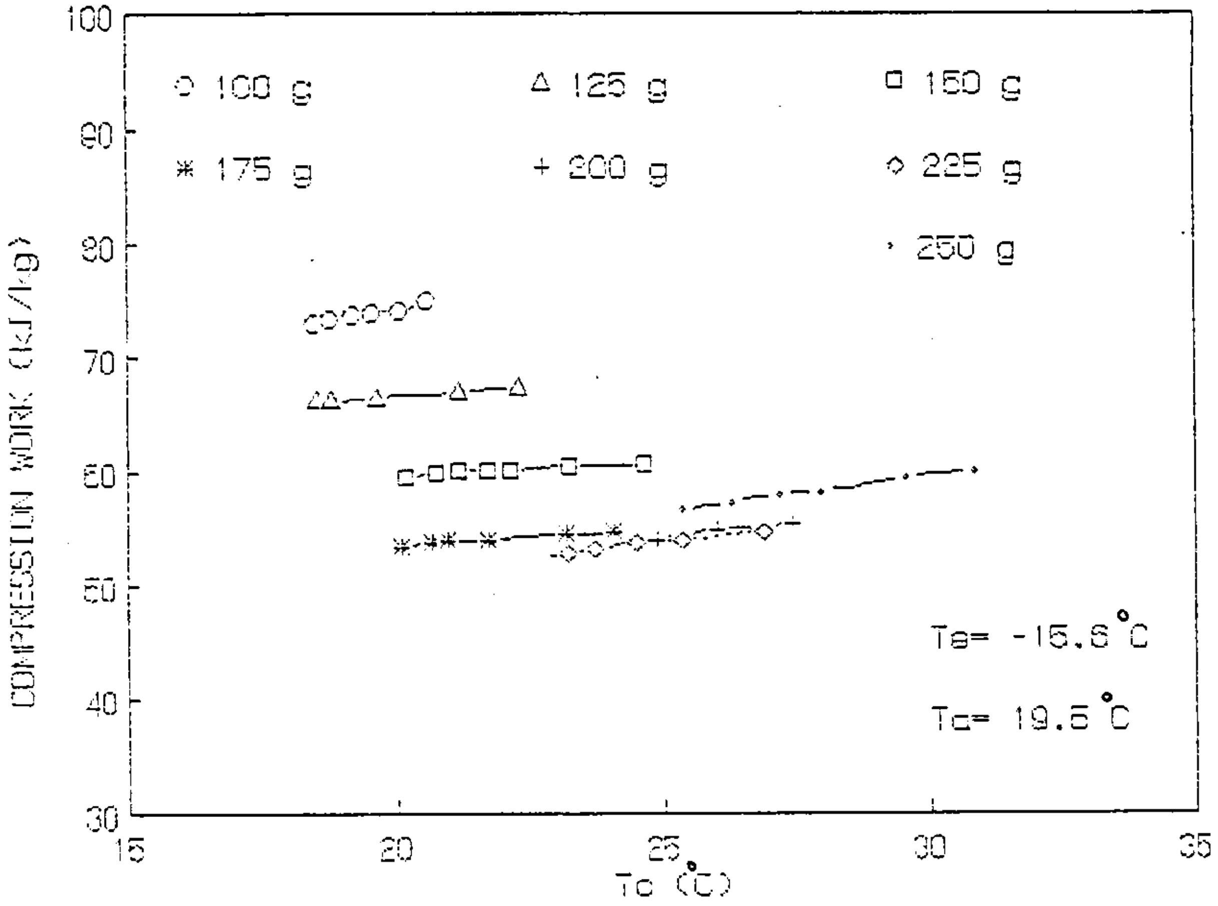

Fig.(5.16) : LPG works of compression vs. Tc

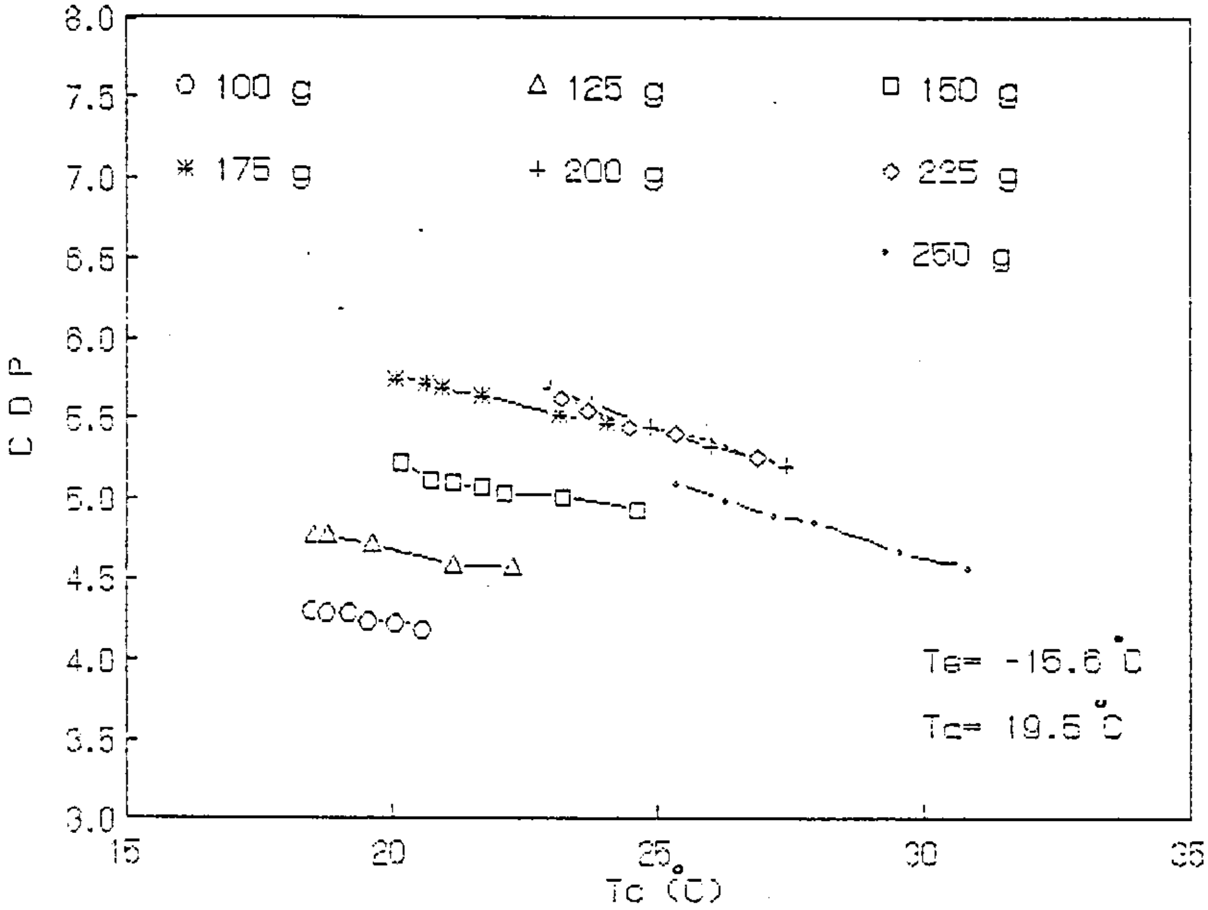

Fig.(5.17) : LPG coefficients of performance vs. Tc

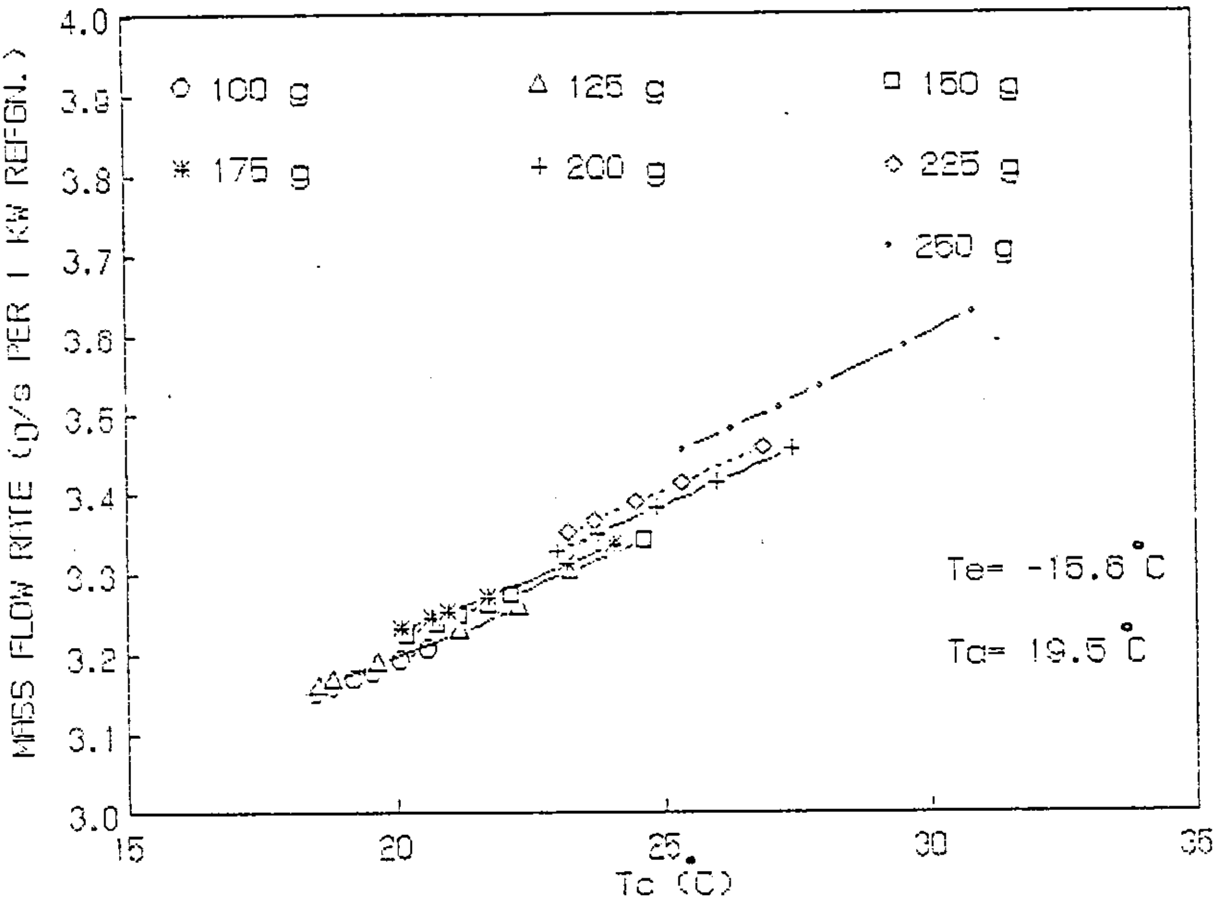

Fig.(5.18) : LPG mass flow rates per 1 kW refrigeration vs. Tc

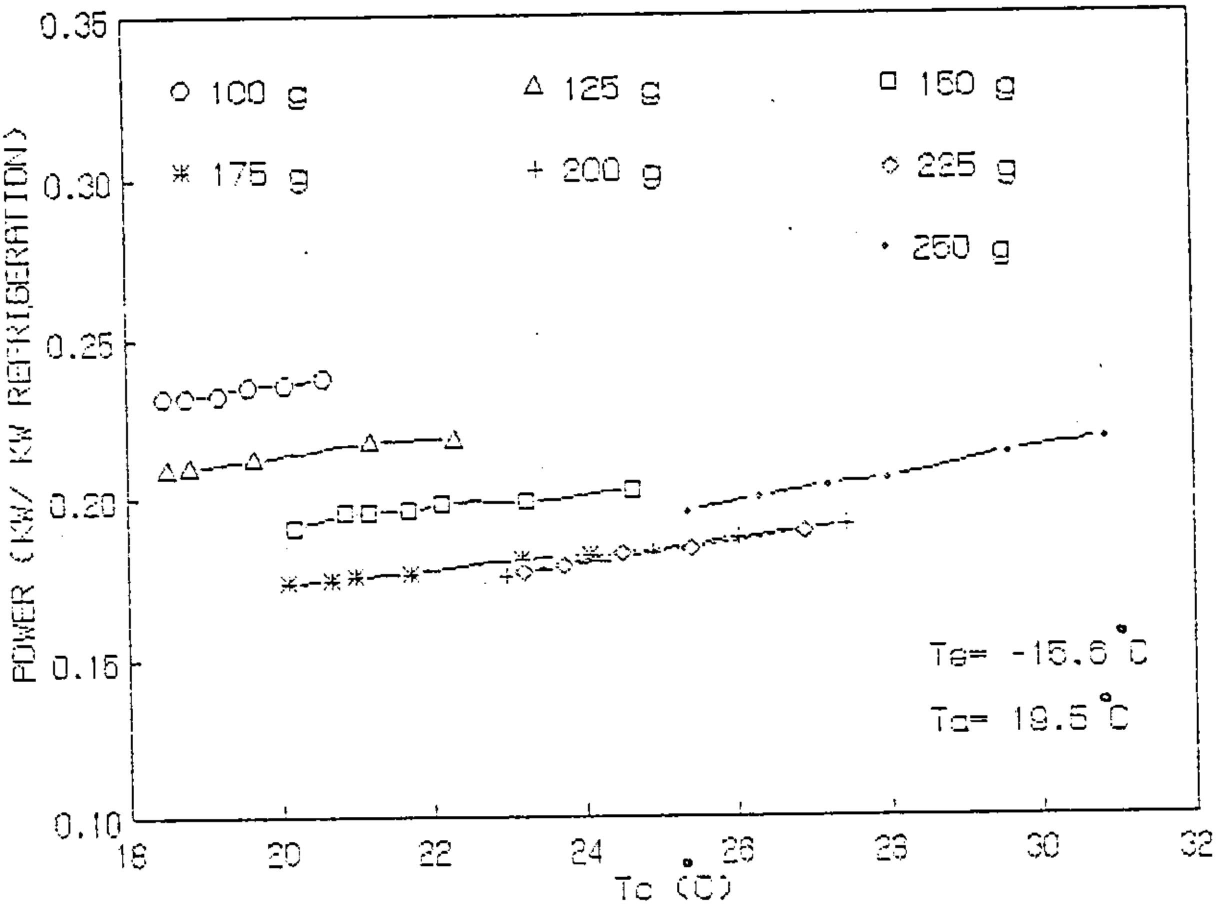

Fig.(5.19) : LPG power consumptions per 1 kW refrigeration vs. Tc

This explains the increase in the refrigerant flow rate per unit capacity as the charge quantity increases, since the refrigerating effect decreases with the increase of charge quantity, as was shown in fig.(5.15).

5.3.5 Power consumption :

The power consumption per unit refrigeration capacity is plotted versus T_c in fig.(5.19). The power consumption per unit capacity is inversely proportional to the COP. That is, the charge with the highest COP will require the lowest power consumption to produce the same capacity as the others, and the opposite is true.

5.4 Propane/Butane Mixture (LPG) versus R-12 :

Now the comparison between the performance of the propane/butane (LPG) mixture refrigerant compared to that of R-12 will be studied through a set of result curves. The first set of performance parameters is plotted with respect to the evaporating temperature, while the other one is plotted with respect to the condensing temperature.

5.4.1 Performance parameters versus T_e :

The performance curves with respect to T_e are taken at $T_c = 25.2$ °C and $T_{amb.} = 18.1$ °C.

In figure (5.20), the refrigerating effect versus T_e is shown for R-12 and the propane/butane (LPG) mixture at a constant condensing temperature . As noticed from the figure, The LPG refrigerant has a refrigerating effect which is about two to three times higher than that of R-12.

This is due to the fact that at the same condensing and evaporating temperatures, the LPG has much higher enthalpies than R-12 in both liquid and vapor states (about double for both states). Therefore, any enthalpy difference between two different LPG enthalpy points in the system will be about double of that for R-12.

The consideration noticed for refrigerating effects applies for the work of compression too, as shown in figure (5.21), where a similar difference is noticed. That is, the work of compression for LPG is about two times higher than that of R-12.

The COP curves for both refrigerants with respect to T_e, at $T_c = 25.2$ °C, are shown in figure (5.22). As noticed in the figure, the two refrigerants have close COP values with LPG refrigerant exhibiting slightly better performance than R-12 at low evaporating temperatures (< -6 °C) and poorer performance at high evaporating temperatures (> -5 °C).

The refrigeration capacities for the two refrigerants (calculated using a simulated load) are shown in figure (5.23). It is noticed that R-12 has a slightly higher refrigeration capacity than the LPG. This may be due to the considerably higher mass flow rate of R-12, as will be shown.

In figure (5.24), the refrigerants mass flow rates per 100 kW of refrigerating capacity are shown for the two refrigerants. In order to accommodate the same refrigeration capacity, considerably higher mass flow rate of R-12, which has considerably lower refrigerating effect than LPG, is required, as shown in the figure.

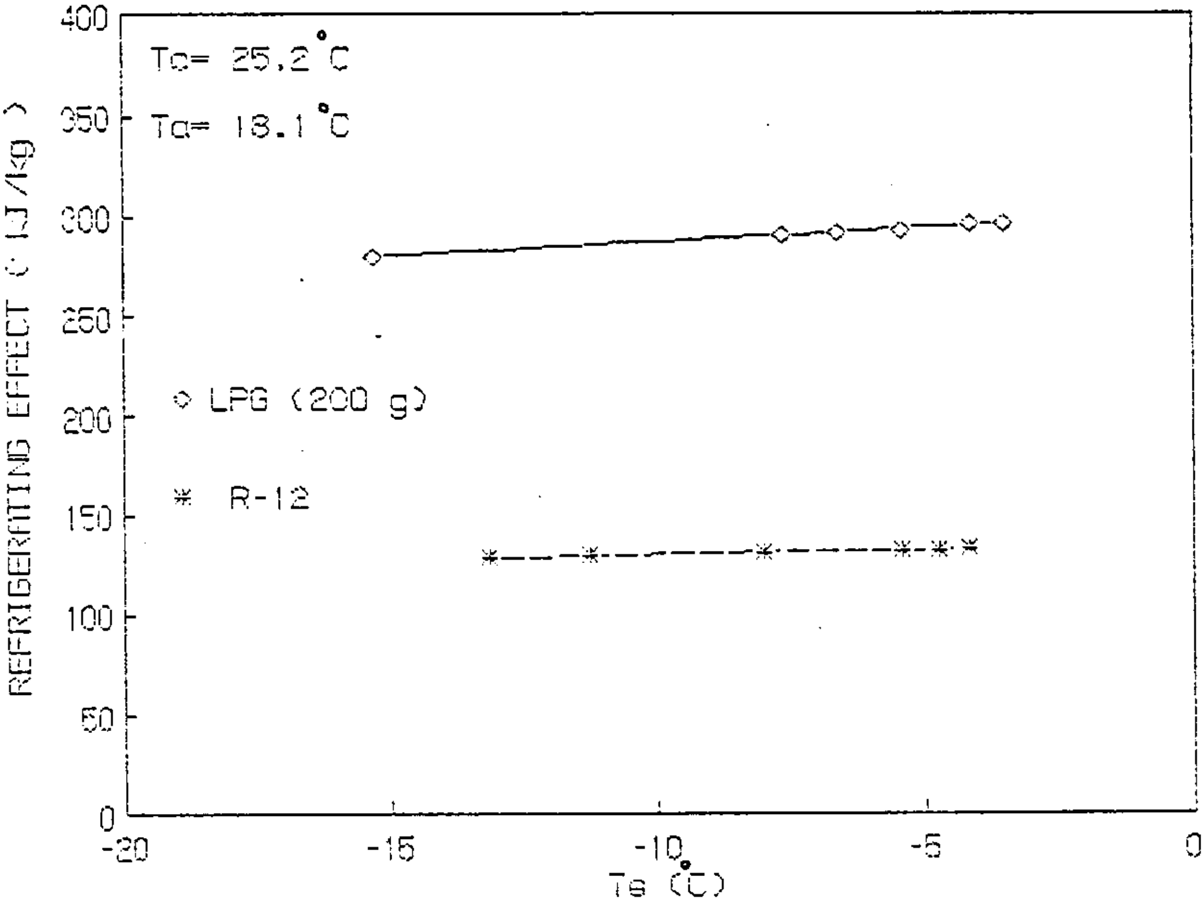

Fig.(5.20) : Refrigerating effects vs. Te

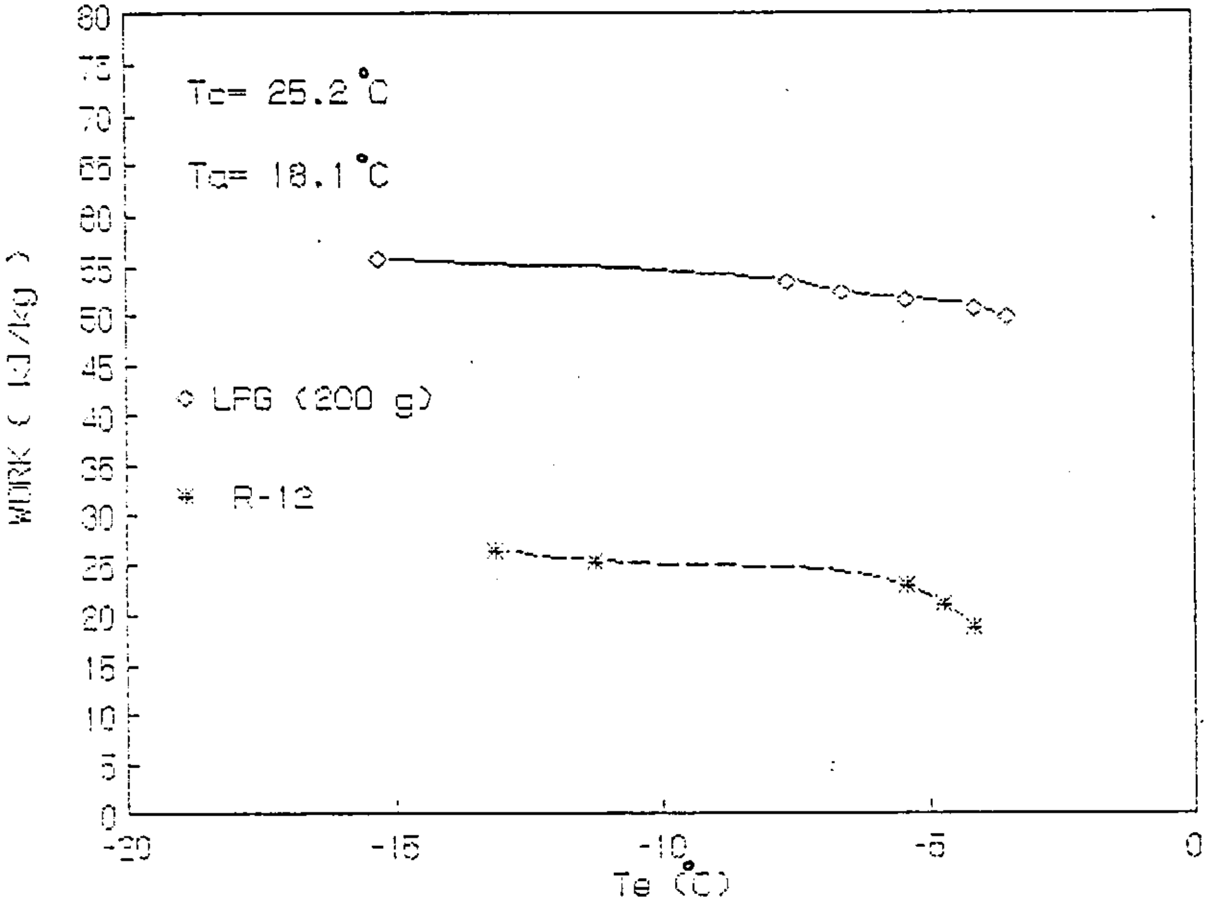

Fig.(5.21) : Works of compression vs. Te

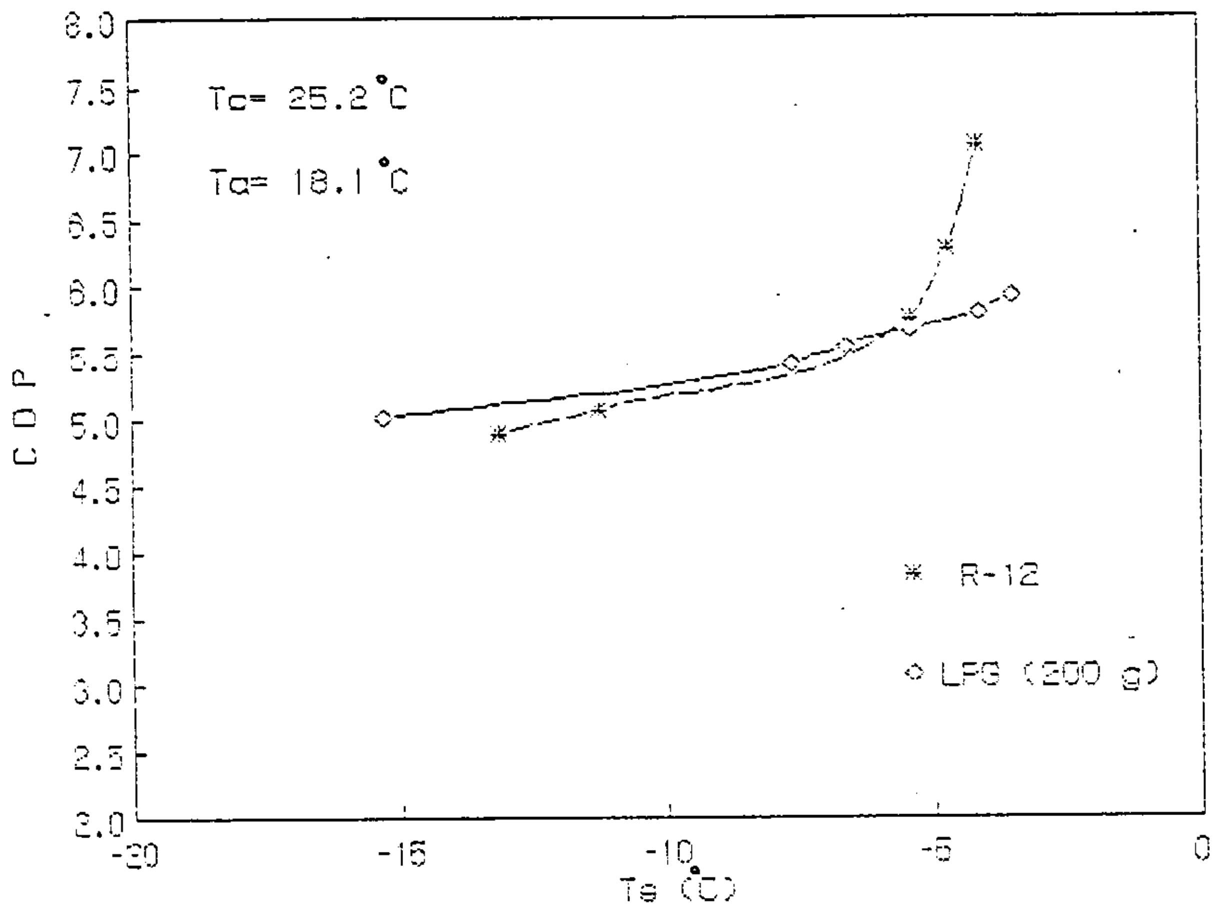

Fig.(5.22) : Coefficients of performance vs. Te

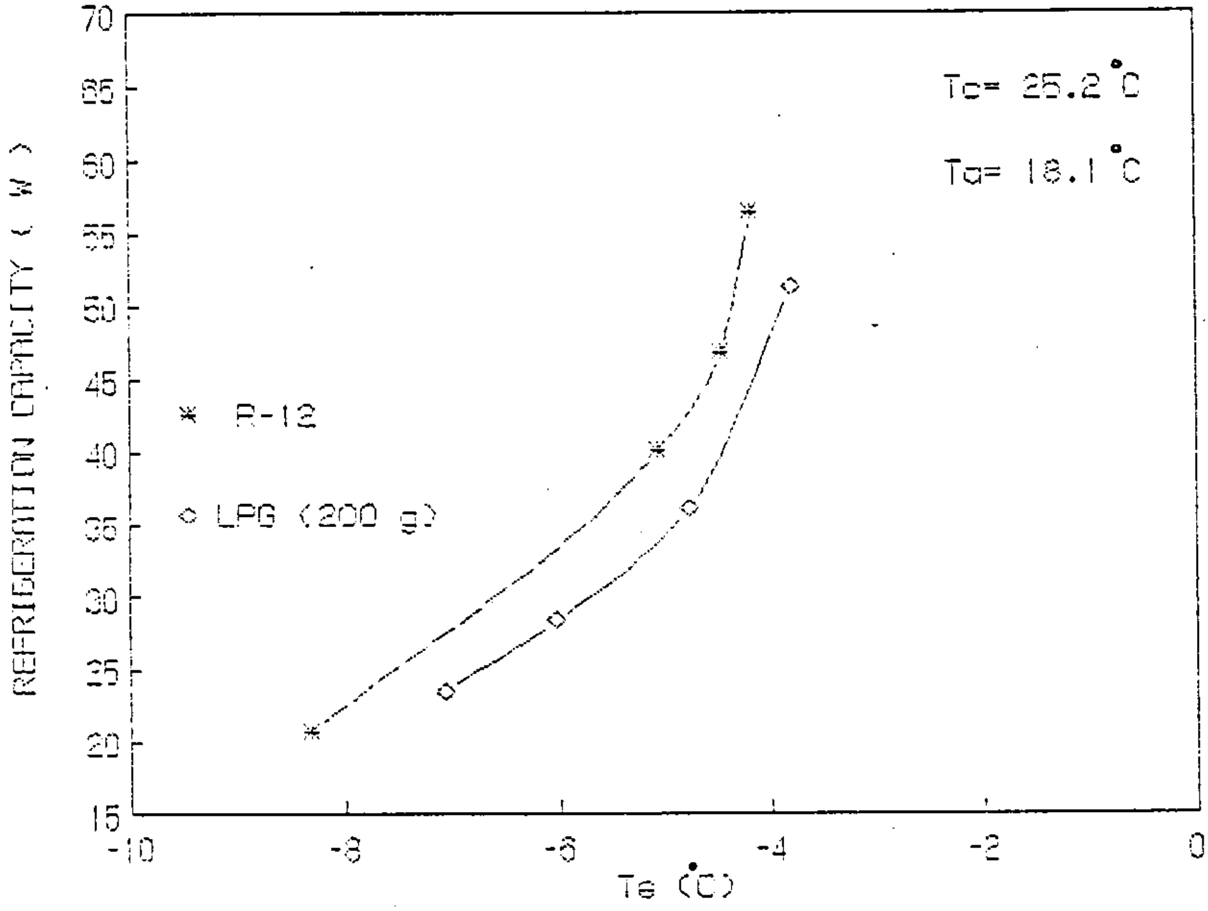

Fig.(5.23) : Refrigeration capacities vs. Te

The actual mass flow rates for the two refrigerants are shown in figure (5.25). As noticed in the figure, R-12 has a considerably higher mass flow rate than the LPG which explains the slightly higher refrigeration capacity in fig.(5.23).

Since the work of compression for the LPG refrigerant is considerably higher than that of R-12 and its mass flow rate is considerably lower, the power consumption required to produce the same refrigeration capacity for the two refrigerants is nearly the same, as shown in figure (5.26). LPG refrigerant has slightly lower power consumption per unit capacity below -5 °C, and higher consumption above that temperature.

The actual power consumptions are shown in figure (5.27). In this figure it is noticed that R-12 has a higher power consumption than the LPG. This is due to its high mass flow rate compared to the LPG.

5.4.2 Performance parameters versus T_c :

The performance curves of R-12 and LPG refrigerants versus T_c are taken at $T_e = -17.7$ °C and $T_{amb.} = 18.1$ °C.

The refrigerating effects with respect to the condensing temperature for the two refrigerants are shown in figure (5.28). The same behavior as that with respect to the evaporating temperature is recognized here, where the LPG refrigerant has a considerably higher refrigerating effect than R-12. As was mentioned in section 5.3.1, the refrigerating effects decrease as T_c increases.

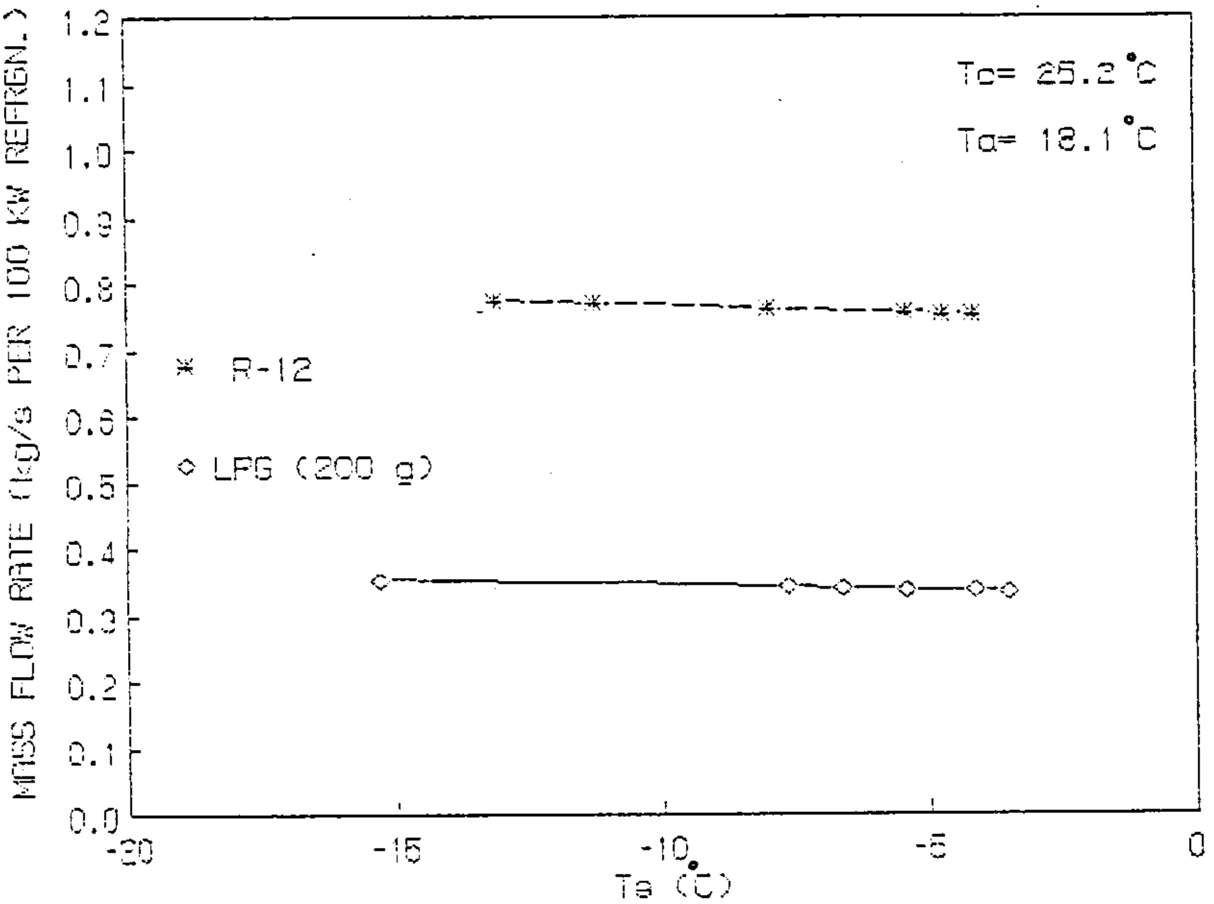

Fig.(5.24) : Mass flow rates per 100 kW refrigeration vs. Te

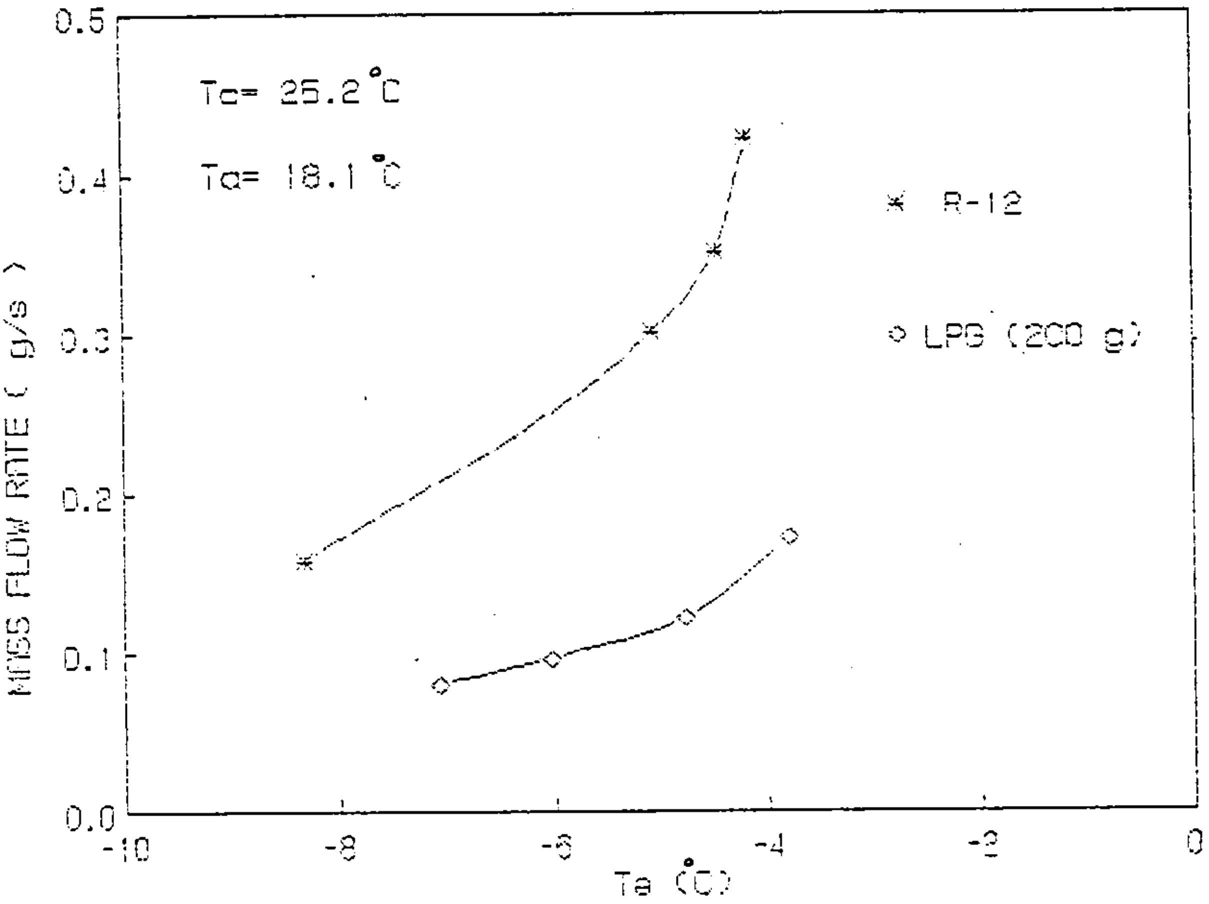

Fig.(5.25) : Actual mass flow rates vs. Te

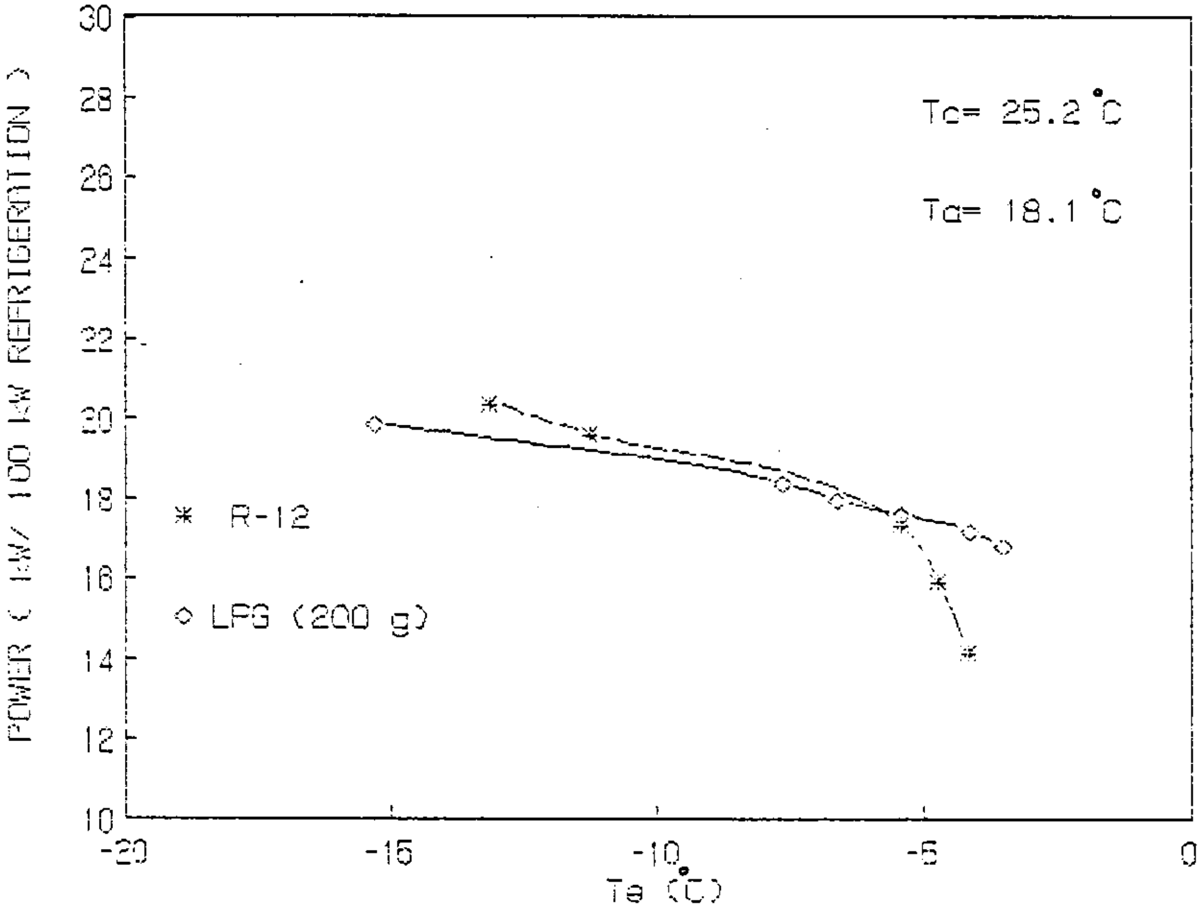

Fig.(5.26) : Power consumptions per 100 kW refrigeration vs. Te

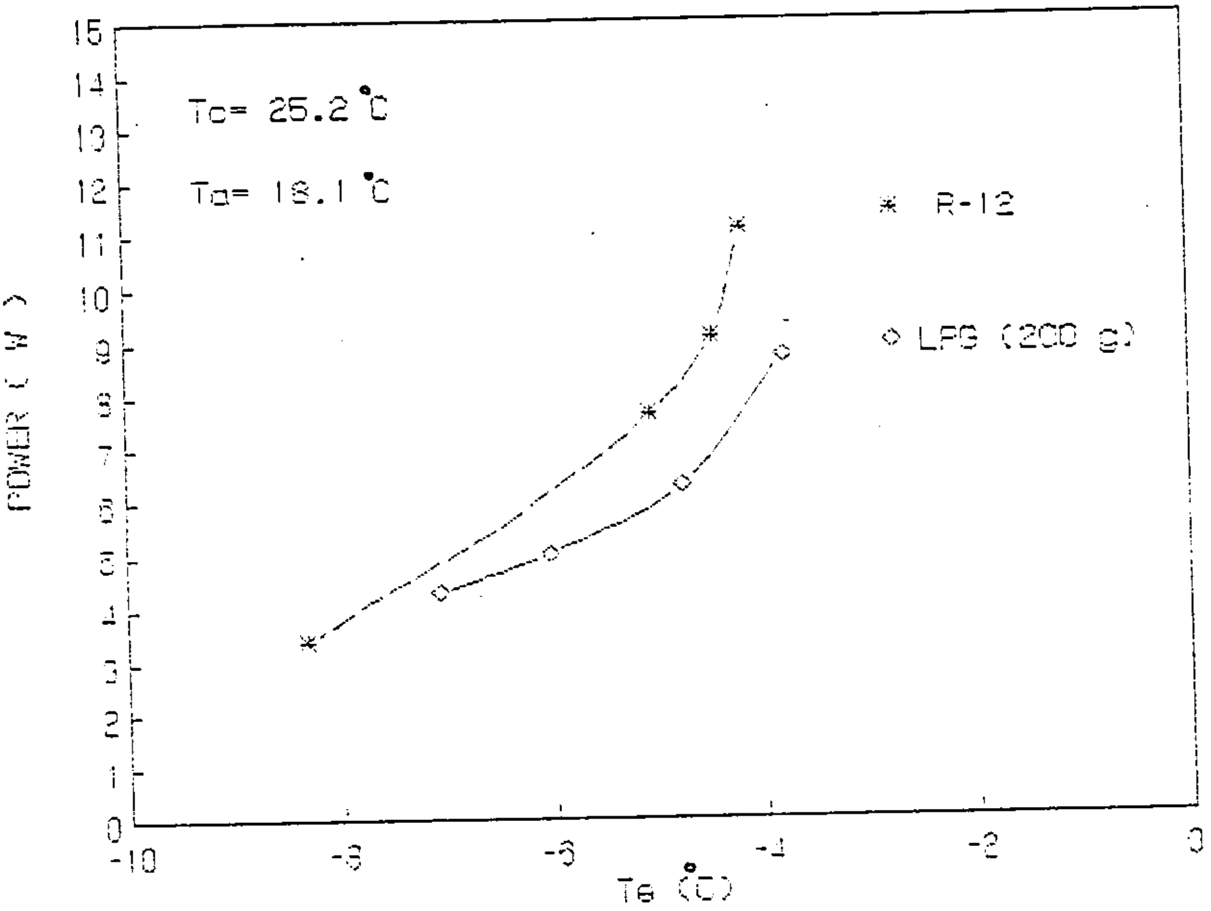

Fig.(5.27) : Actual power consumptions vs. Te

Also the same behavior as that with respect to the evaporating temperature is recognized for the compression works with respect to the condensing temperature, as shown in figure (5.29). That is, the LPG has a higher work of compression than R-12. As was discussed in section 5.3.2, the works of compression increase as T_c increases.

The coefficient of performance for the two refrigerants is shown with respect to the condensing temperature, at T_e = -17.7 °C, in figure (5.30). From this figure, it is noticed that the LPG refrigerant has a higher COP than R-12. This agrees with the result of fig.(5.22) with respect to the evaporating temperature, where it was noticed that at low evaporating temperatures (< -6 °C) the performance of LPG becomes better than R-12. The COP decreases as T_c increases, as was discussed in section 5.3.3.

The mass flow rates per unit refrigeration capacity are shown for the two refrigerants in figure (5.31) with respect to the condensing temperature. In order for the two refrigerants to produce the same refrigeration capacity, R-12 should have higher mass flow rate because of its lower refrigeration effect. That is, the same behavior as that with respect to T_e is recognized here (see fig.5.24). As T_c increases, the refrigerating effects decrease and therefore, the mass flow rates required to produce unit capacity increase, as shown in the figure.

The power consumption for the two refrigerants per unit refrigeration capacity is shown with respect to the condensing temperature, at T_e= -17.7°C, in figure (5.32). It is noticed from this figure that R-12 has a higher power consumption than LPG. This approves the behavior noticed in fig.(5.26), where it was noticed that at evaporating temperatures less than -5 °C, the power consumption of R-12 per unit refrigeration capacity becomes higher than that of LPG. Also, it is noticed that as T_c increases, the power consumptions increase.

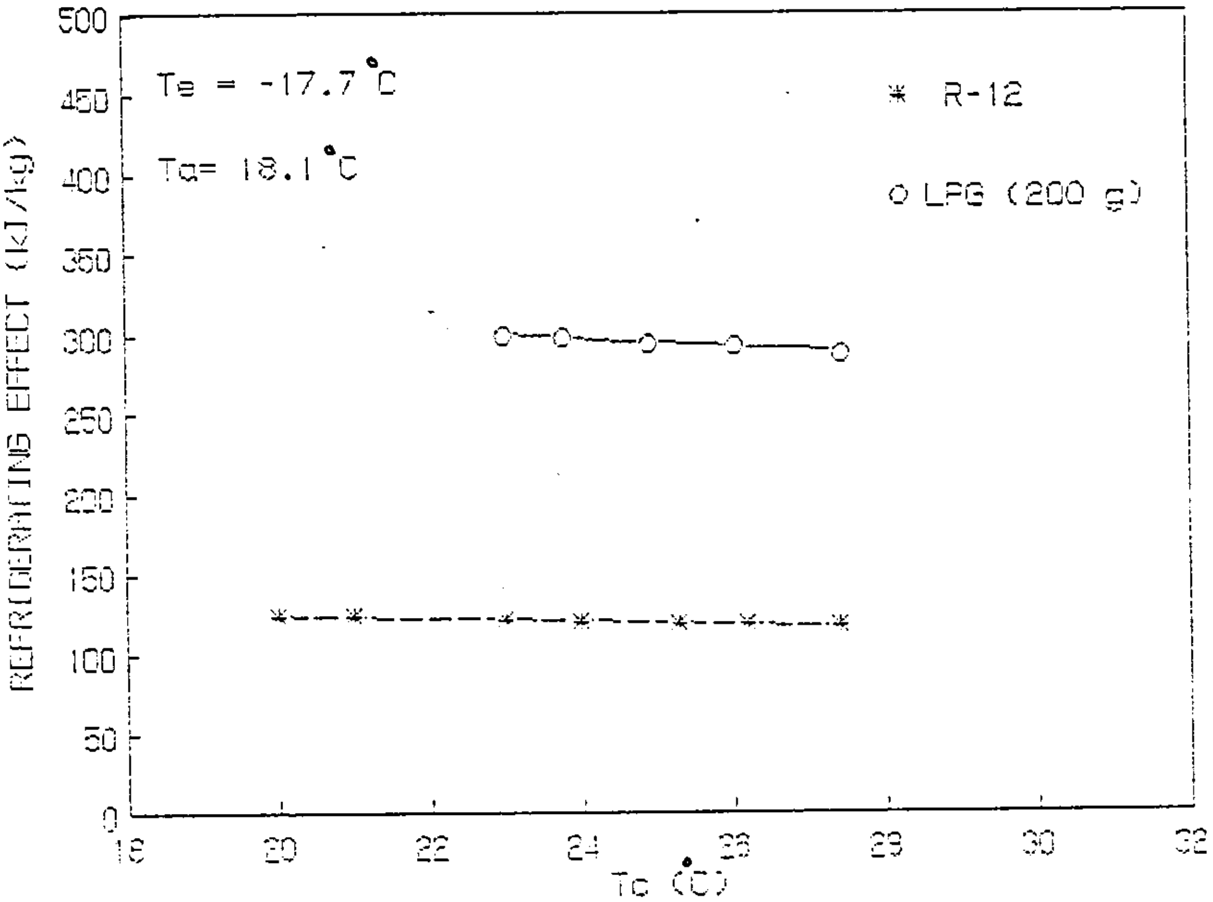

Fig.(5.28) : Refrigerating effects vs. Tc

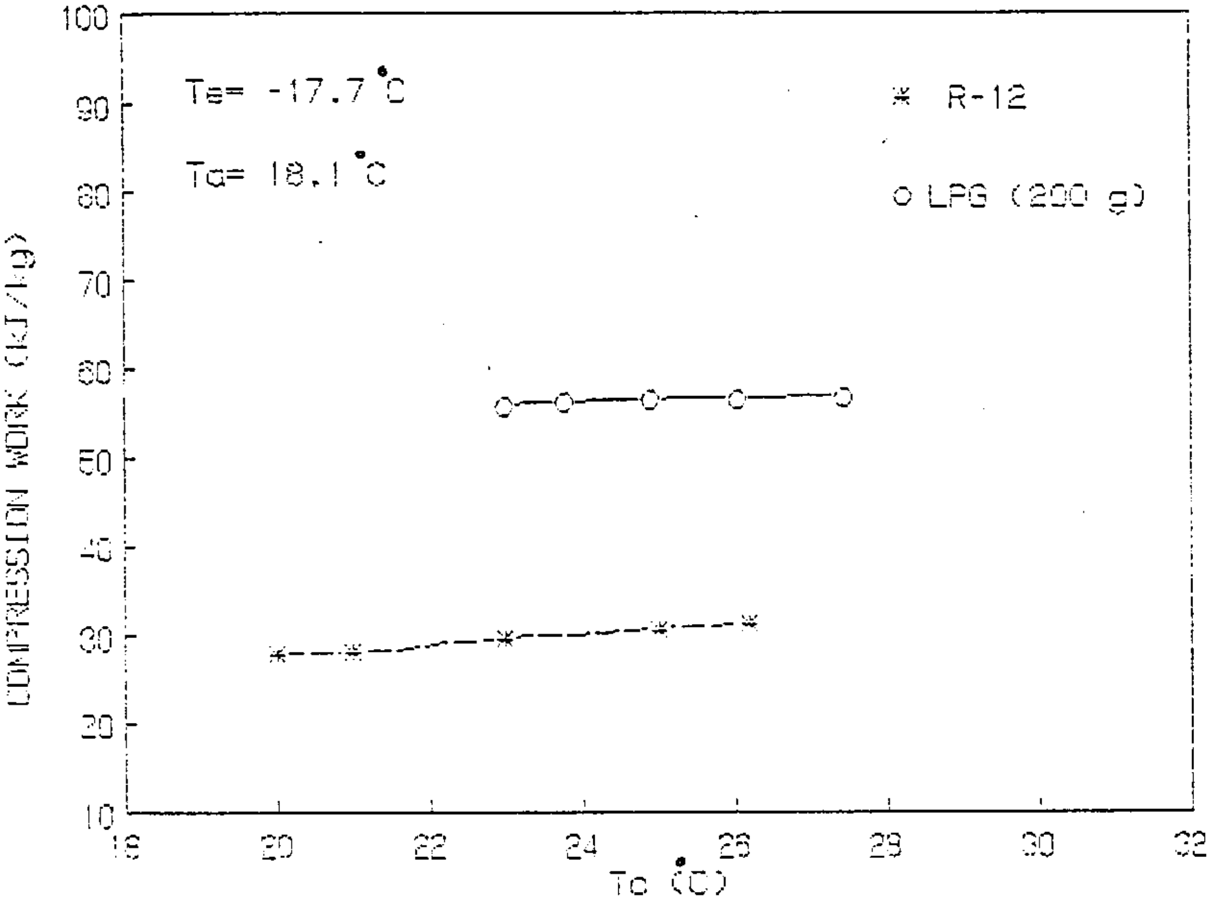

Fig.(5.29) : Works of compression vs. Tc

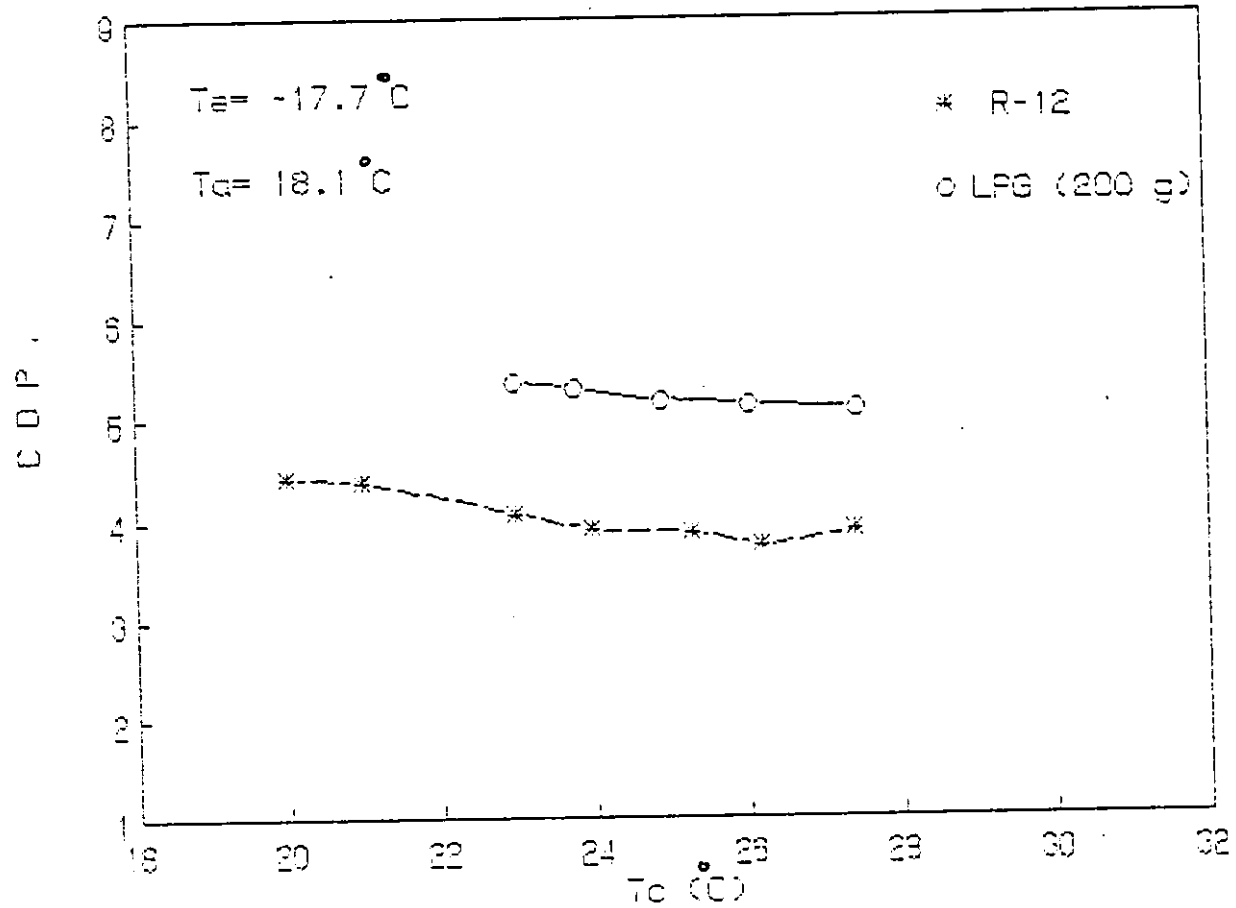

Fig.(5.30) : Coefficients of performance vs. Tc

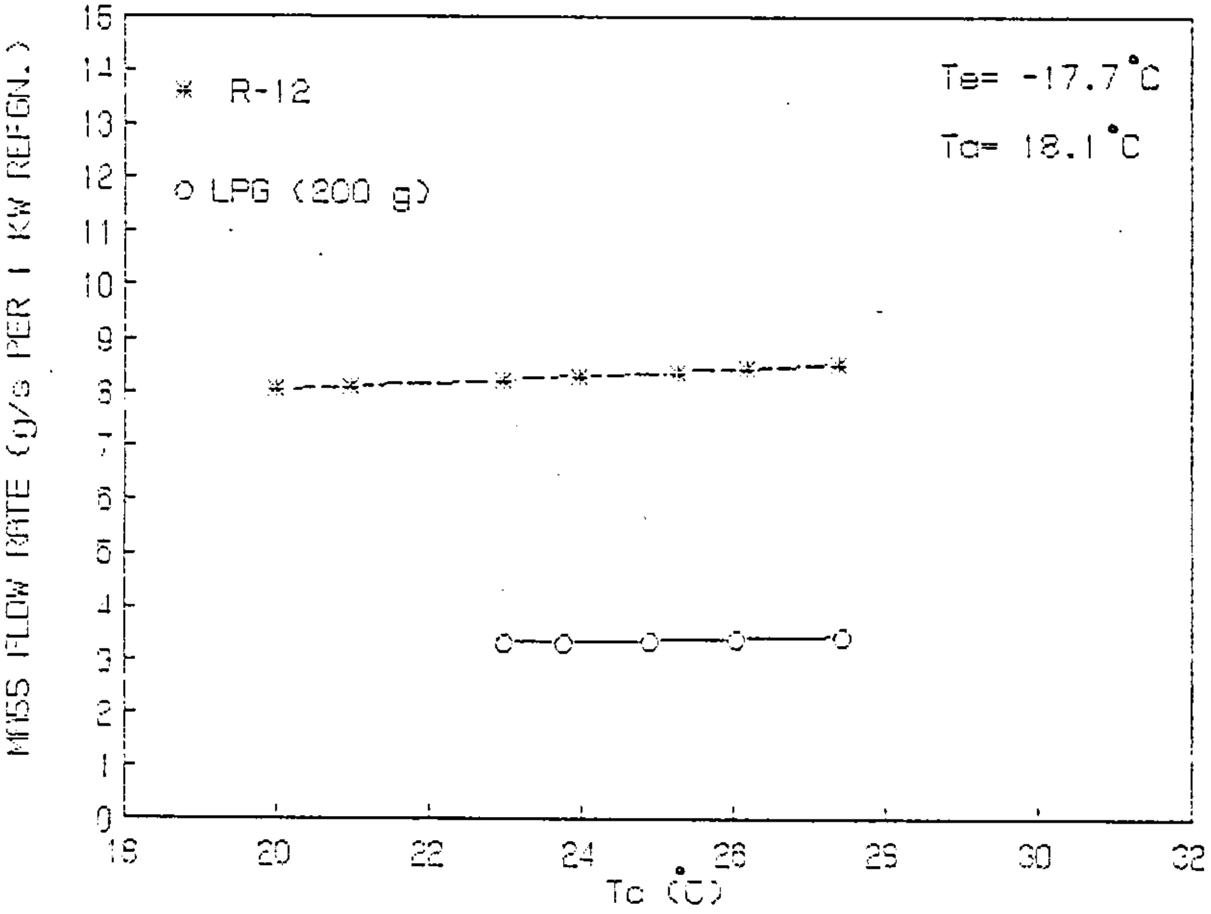

Fig.(5.31) : Mass flow rates per 1 kW refrigeration vs. Tc

5.4.3 Other Comparison Factors :

Other types of comparisons can be made between R-12 and the propane/butane (LPG) mixture refrigerants. Some examples are: the response of the evaporating temperature when subjected to a sudden load, and the long-term power consumption of the refrigerator unit.

The variation of the evaporating temperature with respect to time when placing a load in the frozen food compartment may be useful in comparing between the two refrigerants.

A similar load (container with 1.3 kg of hot water at 80 °C) was placed in the frozen food storage compartment in the two stages of experimental work (i.e. the one with R-12 and the other with LPG) to test the effect of a sudden load on the evaporator temperature for the two refrigerants.

The results are shown in figure (5.33). From the figure, it is noticed that initially the LPG had lower evaporating temperature than R-12. After placing the load, T_e for both refrigerants varied in a similar manner, but with that of LPG having slightly higher temperature values. They both reached their highest T_e value in nearly the same time (after about 25 minutes of placing the load), and then began slowly to decrease again. Hence, except the little higher temperature values of LPG than R-12, they both reacted similarly, as shown in the figure.

The long-term power consumption by the refrigerator unit was monitored by a KW.hr meter over a long period (about 4 months) for R-12 and the LPG refrigerants. The results indicated similar power consumptions for the two refrigerants. They both had a mean consumption of about 2.60 KW.hr per day.

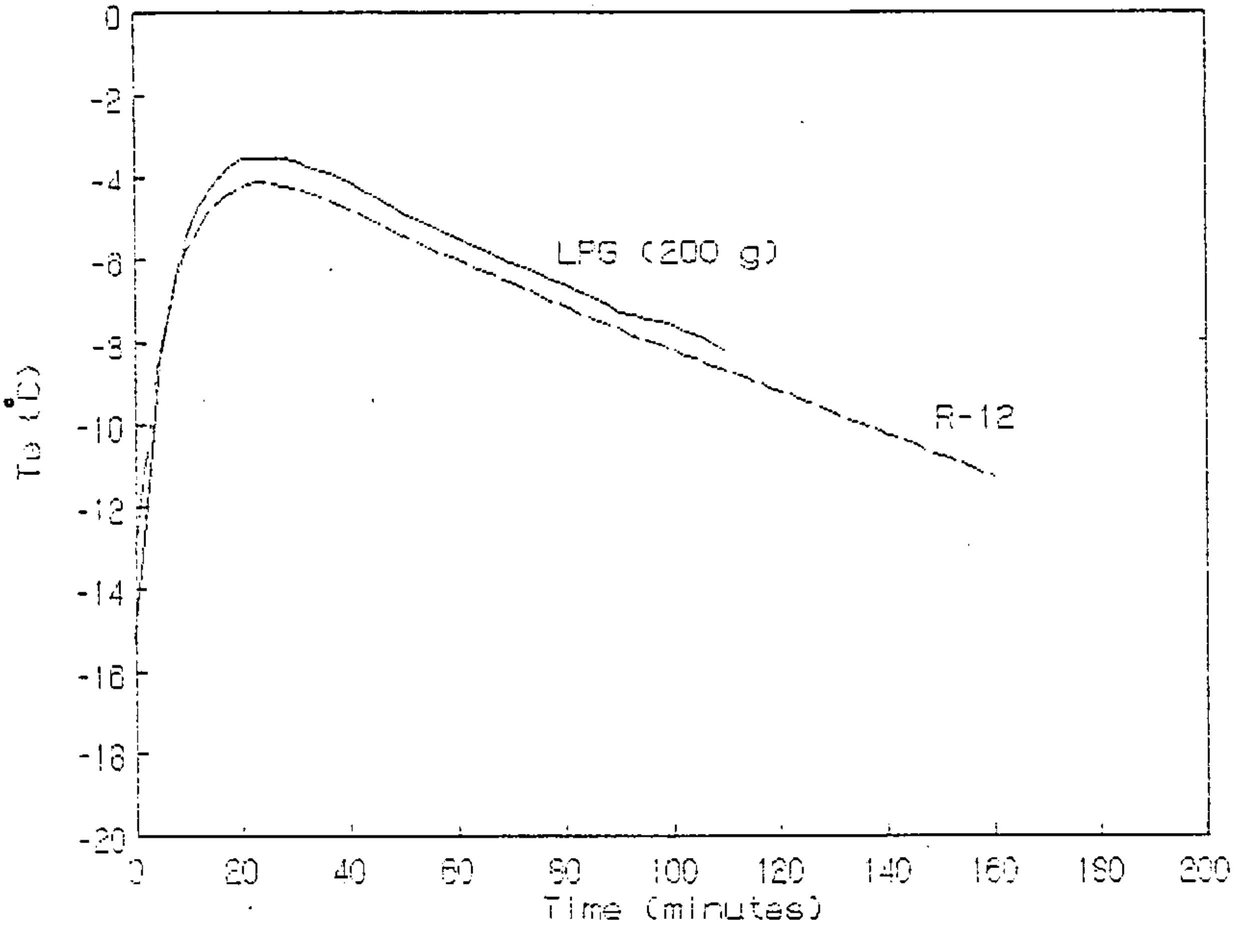

Fig.(5.33) : Evaporating temperatures response to load

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

This research covers an experimental study of the performance of propane/butane (LPG) mixture as an alternative refrigerant to R-12 in domestic refrigerators.

A domestic refrigerator unit was used to test the performance of the LPG mixture and compare it with that of R-12, the original refrigerant. The final conclusions of this research are stated in the next section.

6.1 Conclusions :

1. The thermodynamical performance of LPG refrigerant is fairly competent with that of R-12. Results showed evaporator temperatures down to about -19 °C, C.O.P. up to 5.9 at T_e of -5 °C, T_c of 28 °C and T_{amb} of 19 °C. Comparisons with R-12 indicated a slightly higher COP, lower capacity and lower power consumption.

2. Lower evaporating temperatures are reached with the LPG than with R-12 for the same condensing temperatures.

3. No design changes, system optimization or component replacements are required for the domestic refrigerator unit in order to be retrofitted with the LPG refrigerant.

4. No replacement of the lubricating oil is needed, since LPG is completely miscible with refrigeration mineral oils.

5. Long-term power consumption monitoring indicated acceptable power consumption (similar to that of R-12).

6. No leakage was detected during the period of operation with LPG (about 4 months), which indicates good material compatibility with the existing system materials including the rubber seals.

7. No side effects, such as compressor overheating or frost accumulation was noticed during the period of operation of the LPG.

8. The propane/butane (LPG) mixture is an attractive substitute for R-12 in domestic refrigeration systems. In addition to its satisfactorily performance, its availability, cheapness and environmentally friendly nature makes it one of the best available alternatives.

6.2 Recommendations :

1. According to the results of this research, the LPG is recommended to replace R-12 in domestic refrigerators and other small systems, given correct technical application for operational and safety factors.

2. The performance of the LPG refrigerant in domestic refrigerators may be enhanced by making small and limited changes to some components. For example, the tuning of the capillary tube (i.e. changing its length and diameter) to work with the LPG is recommended.

3. More researches and studies are recommended on different LPG mixtures with different compositions to determine the effects of changing the proportions of the constituents on the overall performance of the mixture.

4. Chemical researches on LPG mixtures are needed to make accurate temperature-composition and enthalpy-composition charts, T-s and p-h diagrams, and superheated and subcooled tables for these mixtures. These items, when available, will be very helpful and benefit in future works and researches regarding LPGs as refrigerants.

5. The performance of LPG when retrofitted into other refrigeration systems (as air conditioners, small heat pumps, chillers,---,etc.) has to be studied.

6. Hydrocarbon refrigerants (such as methane, ethane, ethylene, propylene, etc.), in general, may be possible substitutes for existing fluorinated refrigerants. Therefore, more studies and researches have to be performed on these natural refrigerant gases.

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APPENDICES

APPENDIX A: LPG SPECIFICATION STANDARDS

APPENDIX B: UNDERWRITER'S LABORATORIES GROUP
CLASSIFICATION OF COMPARATIVE HAZARD

APPENDIX C: DATA TABLES

APPENDIX A

LPG SPECIFICATION STANDARDS

Table (A.1)

SUMMARY OF ASTM SPECIFICATIONS FOR LIQUEFIED PETROLEUM (LP) GASES [18]

Product characteristic	Product designation				ASTM test methods
	Commercial propane	Commercial butane	Commercial PB mixtures	Special duty propane	
Vapour pressure at 100°F (37·8°C), psig (kPa), max	210 (1 447)	70	[a]	200 (1 379)	D1267 or D2598
Volatile residue: evaporated temperature, 95 per cent, °F (°C), max	−37 (−38·3)	36 (2·2)	36 (2·2)	−37 (−38·3)	D1837
or					
butane and heavier, per cent, max	2·5	—	—	2·5	D2163
pentane and heavier, per cent, max	—	2·0	2·0	—	D2163
Motor octane number, min	—	—	—	95	D2598 or D2623
Residual matter:					
residue on evaporation of 100 ml, ml, max	0·05	0·05	0·05	0·05	D2158
oil stain observation	Pass[b]	Pass[b]	Pass[b]	Pass[b]	D2158
Sulphur content, grains/100 ft³ (2·8 m³), max	15	15	15	10	D1266
grains/m³, max	—	—	—	2·832	
Specific gravity, 60°F/60°F	[c]	[c]	[c]	—	D1657 or D2598
Corrosion, copper strip, max	No. 1	No. 1	No. 1	No. 1	D1838
Hydrogen sulphide content	—	—	—	Pass[d]	D2420
Moisture content	Pass	—	—	Pass	D1835
Free water content	—	None	None	—	D1835

[a] The permissible vapour pressures of products classified as PB mixtures must not exceed 200 lb and additionally must not exceed that calculated from the following relationship between the observed vapour pressure and the observed specific gravity:

Vapour Pressure, max = 1 167 − 1 880 (sp gr 60°F/60°F)

[b] An acceptable product shall not yield a persistent oil ring when 0·3 ml of solvent–residue mixture is added to a filter paper, in 0·1 ml increments and examined in daylight after two minutes.

[c] Although not a specification requirement, the specific gravity should be reported.

[d] An acceptable product shall not show a distinct coloration.

Table (A.2)

BRITISH STANDARD SPECIFICATIONS FOR COMMERCIAL BUTANE AND PROPANE
BS 4250: 1968

Product characteristics	Product designation		Test method
	Commercial butane	Commercial propane	
Scope	Suitable for general domestic and industrial purposes		
Description	Shall be a hydrocarbon mixture consisting predominantly of butanes and/or butylenes. It shall not contain harmful quantities of toxic or nauseating substances and shall be free from mechanically entrained water.	Shall be a hydrocarbon mixture consisting predominantly of propane and/or propylene. It shall not contain harmful quantities of toxic or nauseating substances and shall be free from mechanically entrained water.	—
Volatility	95 per cent v shall evaporate at 35°F (2·2°C) or lower	—	ASTM D1837
Vapour pressure at 45°C (113°F)	35 psig max (70 psig min for portable containers only)	255 psig max	BS 3324
Total sulphur	0·2 per cent w max		ASTM D1266 IP 107
Mercaptan sulphur (after stenching)	4·0 grains/100 cu ft max (under review)		IP 104A
Hydrogen sulphide	not detectable		BS 3156
Odour	The odour of the gas shall be distinctive, unpleasant, and non-persistent and shall indicate the presence of gas down to concentrations in air of 1/5th of the lower limit of flammability		BS 4250 App. 3
Oil residue			under development
Acetylenes	2 moles per cent max		By gas chromatography mass spectrometry or infra-red spectrometry as appropriate.
Dienes	10 moles per cent max	—	
C_2 hydrocarbons	—	5·0 moles per cent max	
Ethylene	—	1·0 moles per cent max	
C_4 and higher hydrocarbons	—	10 moles per cent max	
C_5 and higher hydrocarbons	—	2 moles per cent max	

APPENDIX B

UNDERWRITERS LABORATORIES GROUP CLASSIFICATION OF COMPARATIVE HAZARD

Group	Definition	Examples
1	Gases or vapors which in concentrations of about 1/2 to 1 % for durations of exposure of about 5 minutes are lethal or produce serious injury.	Sulfur Dioxide
2	Gases or vapors which in concentrations of about 1/2 to 1 % for durations of exposure of about 1/2 hour are lethal or produce serious injury.	Ammonia Methyl Bromide
3	Gases or vapors which in concentrations of about 2 to 2.5 % for durations of exposure of about 1 hour are lethal or produce serious injury.	Carbon Tetrachloride Chloroform Methyl Formate

4	Gases or vapors which in concentrations of about 2 to 2.5 % for durations of exposure of about 2 hours are lethal or produce serious injury.	Dichloroethylene Methyl Chloride Ethyl Bromide
Between 4 & 5	Appear to classify as somewhat less toxic than Group 4.	Methylene Chloride Ethyl Chloride
	Much less toxic than Group 4 but somewhat more toxic than Group 5.	R-113
5a	Gases or vapors much less toxic than Group 4 but more toxic than Group 6.	R-11 R-22 Carbon Dioxide
5b	Gases or vapors which available data indicate would classify as either Group 5a or Group 6	Ethane Propane Butane
6	Gases or vapors which in concentrations up to at least about 20 % by volume for durations of exposure of about 2 hours do not appear to produce injury.	R-12 R-114

APPENDIX C

DATA TABLES

All T_e variation test data are given at $T_c = 28.7$ °C and $T_{amb.} = 19.5$ °C, and all T_c variation test data at $T_e = -15.6$ °C and $T_{amb.} = 19.5$ °C. (Temperatures are given in °C and pressures in psig)

Table (C.1): R-12 refrigerant, T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-13.12	14	5.2	44.5	60.5	69.61	41.34
-7.96	19.5	8.9	47.1	66.5	62.86	35.97
-5.62	22.5	11.2	47.4	67	57.7	33.92
-5.28	23	11.8	47.8	67	56.77	33.56
-4.14	24.5	14	48	66	48.67	29.37
-4.21	24	12	47.8	65.5	42.59	25.82
-4.71	23.5	9.5	47.7	65.5	37.54	22.55
-5.39	23	6	47.5	64.5	33.22	19.67
-11.23	16	0.8	46.3	60	8.53	1.57

Table (C.2): R-12 refrigerant, T_c variation test

T3	T6	P1	T7	P7
27.4	-1.2	86	57.8	13.5
26.2	-0.8	83	57.7	13
25.3	0.4	80.5	57.5	13
24	0.5	77.5	57.3	12.5
23	1.3	75	57	12.5
21.8	2.3	72	56.6	12
21	2.9	70	56.3	12
20	3.2	67.5	55.4	12
18.7	3.4	64.5	53.1	11.5
16.3	3.3	59.5	51.7	11.5
15.2	3.5	57	50	11

Table (C.3): LPG refrigerant (100 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-16.15	-12	-10.86	37.46	25	71	48.2
-7.75	-11.5	-5.73	38.6	27	64.95	46.01
-5.5	-11	-3.87	38.27	28	60.05	43.47
-4.88	-10.5	-1.62	38.32	29	52.97	38.74
-5.57	-10	-1.36	38.83	30	47.05	34.13
-6.57	-10	-0.92	39.88	29	42.02	30.17
-7.62	-10	-3.78	40.06	28.5	37.84	26.76
-8.58	-10.5	-4.62	40.17	28	34.06	23.58
-10.09	-11	-5.06	40.48	27	27.73	18.37
-11.31	-11	-6.67	39.79	26	22.64	14.16

Table (C.4): LPG refrigerant (100 g charge), T_c variation test.

T3	T6	P1	T7	P7
20.6	-12.8	-12	35.77	30
20.1	-12.5	-12	35.76	28
19.6	-12.9	-12	35.45	28
19.2	-12.3	-12	35.66	27.5
18.8	-12.56	-12	35.56	27
18.7	-12.9	-12	35.46	27
18.5	-13.05	-12	35.35	26.5
18.4	-13.76	-12	35.15	26

Table (C.5): LPG refrigerant (125 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-17.83	-12	-12.44	31.78	25	68.24	47.23
-10.6	-11	-8.49	32.26	27	61.29	37.46
-7.66	-11	-6.02	32.1	28	56.69	36.06
-6.73	-10.5	-4.71	32.47	29	52.77	34.16
-6.58	-10	-3.95	33.54	30	46.24	30.4
-7.17	-10	-4.32	34.97	31	41.2	27.09
-8.06	-10	-4.82	35.62	30	37.01	23.87
-8.87	-10.5	-5.57	36.33	29	33.01	20.92
-10.28	-11	-7.01	37.08	28	26.84	16.04
-11.46	-11	-7.77	36.96	27	21.64	11.99
-12.37	-11	-8.57	36.93	26	17.33	8.66
-13.08	-10.5	-8.93	36.24	26	13.64	5.35
-13.67	-10	-9.26	35.67	25	10.59	3.42

Table (C.6): LPG refrigerant (125 g charge), T_c variation test.

T3	T6	P1	T7	P7
22.31	-12.4	-12	31.87	30
22.08	-12.03	-12	31.57	29
21.2	-12.34	-12	32.15	28
19.68	-12.37	-12	31.54	27.5
19.57	-12.55	-12	31.9	27
18.83	-12.16	-12	31.55	27
18.73	-12.66	-12	31.32	26
18.35	-12.58	-12	31.12	26.5

Table (C.7): LPG refrigerant (150 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-14.94	-12	-10.3	30.31	29	67.55	47.91
-7.37	-11	-5.67	30.96	31	60.69	39.58
-5.14	-11	-4.09	30.79	32	56.3	37.68
-4.3	-11	-3.53	31.05	32.5	52.48	35.75
-4.04	-10.5	-3.06	32.19	33	49.39	33.93
-4.63	-10.5	-3.34	34.13	34	39.27	26.77
-5.33	-10	-3.74	34.67	35	35.4	23.81
-6	-10	-3.94	34.82	35	32.1	21.21
-6.63	-10	-4.57	35.26	34	29.1	18.9
-7.19	-10.5	-4.67	35.56	33	26.31	16.78
-7.73	-10.5	-5.14	35.21	32.5	23.93	14.85
-8.19	-11	-5.56	35.09	32	21.63	13.08
-9	-11	-6.14	34.76	31	17.73	10.03

Table (C.8): LPG refrigerant (150 g charge), T_c variation test.

T3	T6	P1	T7	P7
24.66	-10.2	-12	30.3	30
23.25	-10	-12	30.14	28
22.15	-10.35	-12	30.02	28
21.72	-10.32	-12	29.84	29
21.57	-10.25	-12	29.81	28
21.19	-10.43	-12	29.65	28
20.87	-10.65	-12	29.48	27
20.77	-10.53	-12	29.28	26.5
20.21	-10.6	-12	28.82	25.5

Table (C.9): LPG refrigerant (175 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-14.39	-12	-9.82	28.6	35	59.75	51.54
-7.45	-11	-6.13	29.22	38	64.44	49.19
-4.55	-10	-3.83	29.82	39	59.78	46.41
-3.42	-10	-2.71	30.12	40	56.03	43.76
-3	-10	-2.18	31.78	41	49.8	39
-3.56	-9	-2.56	32.88	40	42.47	33.03
-4.4	-9	-2.91	33.3	39.5	38.42	29.66
-5.12	-9	-3.48	33.4	39	34.83	25.72
-5.84	-9.5	-3.67	33.59	38	31.54	23.97
-7.06	-10	-4.55	33.75	37	26.26	19.42
-8.08	-10	-5.26	33.63	36	21.76	15.63
-8.35	-11	-5.57	33.72	36	18	12.21
-9.53	-11	-6.4	32.88	35	14.73	9.46

Table (C.10): LPG refrigerant (175 g charge), T_c variation test.

T3	T6	P1	T7	P7
24.1	-9.67	-12	27.3	30
23.2	-9.57	-12	27.2	28
21.73	-9.8	-12	26.7	28
21	-9.85	-11.5	26.5	27.5
20.68	-9.9	-11	26.3	27
20.24	-9.95	-11.5	26.1	26.5
20.12	-9.98	-12	25.16	26

Table (C.11): LPG refrigerant (200 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-15.27	-9	-8.95	28.82	37	72.76	52.08
-8.15	-8.5	-5.23	30.42	42	66.13	40.66
-5.2	-8	-3.11	31.29	43	61.32	38.6
-4.08	-8	-2.35	31.79	45	57.08	36.57
-3.5	-7.5	-1.75	32.48	46.5	50.57	32.39
-3.57	-7.5	-1.97	32.67	47	47.39	30.48
-4.07	-7.5	-2.24	32.78	49	42.33	26.92
-4.78	-7	-2.6	32.59	48	38.27	23.9
-5.44	-7	-2.98	32.37	46	34.58	21.2
-6.05	-7.5	-3.55	32.24	46	31.26	18.81
-6.58	-8	-3.87	32.01	45	28.5	16.56
-7.2	-8	-4.27	31.95	45	25.75	14.7
-7.56	-8	-4.69	31.18	45	23.45	12.92
-8.11	-8	-5.06	31.13	43	21.29	11.32

Table (C.12): LPG refrigerant (200 g charge), T_c variation test.

T3	T6	P1	T7	P7
27.46	-7.97	-11	30.05	32
26.04	-8.04	-11	29.9	29
24.92	-8.06	-11	29.75	29
24.44	-8.12	-11.5	29.65	29
23.9	-8.25	-11.5	29.45	28
23.72	-8.26	-12	29.39	28
23.51	-8.29	-12	29.34	27
23.1	-8.3	-12	29.18	26
23	-8.36	-12.5	29.08	26

Table (C.13): LPG refrigerant (225 g charge), T_e variation test.

T1	P1	T6	T7	P7	T10	T11
-14.43	-11	-9.46	27.47	40	77.78	54.56
-3.52	-7	-2.82	31.73	49	63.6	42.16
-1.88	-6	-1.4	33.36	51	54.9	34.57
-2.04	-5	-1.25	33.93	57	47.94	29.53
-2.88	-5	-1.49	34.02	53	42.49	26.29
-3.66	-6	-2.03	33.98	51	38.25	23.31
-4.47	-6	-2.48	33.72	50.5	34.46	20.65
-5.12	-6	-2.96	33.28	50	31.16	18.28
-5.69	-7	-3.4	33.02	49.5	28.22	16.17
-6.25	-7	-3.41	33.23	49	25.56	14.3
-6.73	-7	-3.9	32.92	48	23.22	12.48
-7.58	-7.5	-4.45	32.36	47	19.04	9.48
-8.32	-8	-4.8	31.75	45.5	15.54	6.92
-8.94	-8	-5.39	31.68	45	12.55	4.75
-9.21	-8	-5.62	31.52	45	11.21	3.77

Table (C.14): LPG refrigerant (225 g charge), T_c variation test.

T3	T6	P1	T7	P7
26.9	-9.19	-11	28.09	35
25.4	-9.24	-11	27.42	32
24.5	-9.5	-11	27.13	32
23.74	-9.53	-11.5	26.76	32
23.43	-9.56	-12	26.65	31
23.31	-9.66	-12	26.62	30
23.22	-9.34	-12	26.57	30