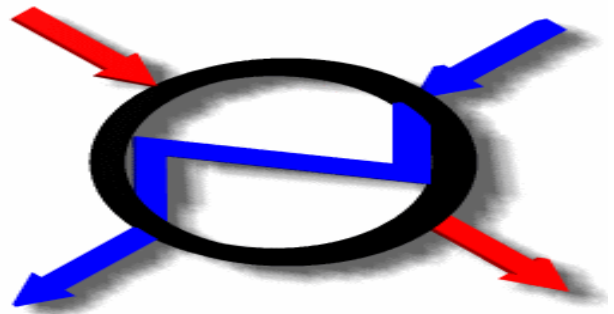




**Research Center for Energy Conversion
Centre de recherche en conversion d'énergie
RCEC/CRCE**

**University of Moncton
Moncton, NB, E1A 3E9**

Analysis of the Test Results

The RCEC has examined the test results obtained in Puerto Rico on two six ton Air Conditioning A/C equipment from McQuay and A&P equipped with the Magnetizer Energizer.

The units were monitored by a StowAway data logger. The following parameters have been measured; air temperatures and humidity across the evaporator and condenser, pressures, as well as amperage and voltage.

The baseline results were obtained under R-22 with the refrigerant Magnetizer Energizer not installed. The baseline testing was carried out for A&P from October 13, 2003 to October 19, 2003 and for McQuay from November 15, 2003 to November 23, 2003.

The baseline test results were compared to the results obtained under similar degree-days with the Magnetizer Energizer installed on both pieces of equipment according to the manufacturer's guidelines. The comparative study based upon the field testing revealed the following;

1. The McQuay A/C equipment with the Magnetizer Energizer installed tested during the periods November 13 to November 21, 2003 and January 13 to January 21, 2004 showed a **saving in the power consumption on the average of 19% compared to the baseline data**. Other benefits such as shorter refrigeration cycles, lower head pressures were observed too.
2. The A&P A/C equipment was also monitored with the Energizer installed from October 27, 2003 to November 1, 2003 and January 03, 2004 to January 11, 2004. The results showed **on the average a reduction of the power consumption of 13%**. Other benefits were also noticed as with the McQuay system.

ANEXO 1

PRUEBAS DE CONSUMO DE ENERGÍA DE REFRIGERADORES DOMÉSTICOS CON EL DISPOSITIVO MAGNETIZER EN EL LABORATORIO DE AMBIENTE CONTROLADO.

1. Características de los refrigeradores domésticos.

Se probarán los siguientes refrigeradores existentes en la Universidad Autónoma Metropolitana-Iztapalapa:

	Marca	Deshielo	Tipo	Año	Refrigerante
1. Refrigerador nuevo	Whirlpool	Automático	Refrigerador-congelador	2007	R 134a
2. Refrigerador usado	Mabe	Semiautomático	Refrigerador convencional	1980	R-12

2. Pruebas a realizar:

Se realizarán 4 pruebas en total, bajo las condiciones definidas por la NOM-015-ENER-2002. Para cada refrigerador, se hará una prueba en condiciones de operación normal del equipo y, otra incorporando los dispositivos *Magnetizer*.

La prueba de los refrigeradores bajo condiciones de operación normal durará entre 12 y 16 horas, una vez que se establece la condición de estabilidad en el laboratorio y se producen al menos 2 ciclos de encendido y apagado.

En la segunda prueba, se colocarán el número de dispositivos *Magnetizer* definido por la **la empresa**, así como en las posiciones que indique. Los investigadores de la Universidad Autónoma Metropolitana Unidad Iztapalapa (UAMI), reportarán esas características en el anexo técnico del Informe con los resultados de la prueba. Se reportarán los resultados del consumo de energía del refrigerador al final del primer, segundo, tercer, quinto y decimocuarto día.

3. Descripción de las pruebas bajo condiciones de Norma.

Las pruebas a refrigeradores domésticos se harán de acuerdo con la norma oficial mexicana NOM-015-ENER-2002: “Eficiencia energética de refrigeradores y congeladores electrodomésticos. Límites, métodos de prueba y etiquetado”, en el Laboratorio de Ambiente Controlado (LAC), de la UAMI.

Esta Norma Oficial Mexicana fija los límites máximos de consumo de energía de los refrigeradores y congeladores electrodomésticos operados por motocompresor hermético y establece los métodos de prueba para determinar dicho consumo de energía. La Norma aplica a los refrigeradores electrodomésticos, refrigeradores-congeladores electrodomésticos de hasta 1104 dm³ (39 pies³) y congeladores electrodomésticos de hasta 850 dm³ (30 pies³) operados por motocompresor hermético comercializados en México.

3.1. Especificaciones del cuarto de prueba.

- **Temperatura ambiente del cuarto de prueba.**

La temperatura ambiente del cuarto de prueba medida a una distancia de 25,4 cm (10 pulgadas) del centro de las paredes laterales del aparato y a una altura de 91,5 cm (3 pies) de la base del aparato debe ser de $32,2^{\circ}\text{C} \pm 0,6^{\circ}\text{C}$. Esta temperatura debe mantenerse dentro de la tolerancia señalada durante el periodo de estabilización al igual que durante las pruebas. El gradiente vertical de temperatura en cualquier punto arriba de 5,1 cm (2 pulgadas) del piso o de la plataforma de soporte y hasta 30,5 cm (1 pie) arriba de la parte superior del gabinete no debe ser mayor de $0,9^{\circ}\text{C}$ por metro de distancia vertical ($0,5^{\circ}\text{F}$ por pie).

- **Circulación del aire.**

El gabinete en prueba debe estar protegido de corrientes de aire cuya velocidad sea mayor de 0,254 m/s (50 pies/minuto).

- **Radiación.**

El gabinete en prueba debe estar protegido de radiación directa de cualquier superficie enfriada o calentada cuya temperatura tenga una diferencia de más de $5,6^{\circ}\text{C}$ (10°F) con la temperatura ambiente del cuarto de prueba.

- **Alimentación eléctrica.**

El suministro eléctrico debe ser de 60 Hz y la tensión de $115\text{ V} \pm 1\text{ V}$. La tensión debe ser medida en la alimentación del producto sometido a prueba mientras el motocompresor esté operando.

- **Instrumentación para Temperatura.**

☞ ① Termómetro de vidrio (únicamente mediciones ambientales).

Ω ① Sensores de temperatura. La masa para elevar la capacidad de calor de un sensor de temperatura, debe tener dimensiones de diámetro y altura de $2,9\text{ cm} \pm 0,6\text{ cm}$ ($1,12\text{ pulgadas} \pm 0,25\text{ pulgadas}$), y debe estar hecha de bronce o cobre u otro material con capacidad térmica total no mayor que la de 20 g de agua.

℞ ① Termómetros de resistencia eléctrica y/o termistores

- **Instrumentación para Electricidad.**

☞ ① Watthorímetros: Los watthorímetros analógicos deben tener una resolución de 0,01 kWh o mejor. Los instrumentos digitales deben tener una resolución de 0,001 kWh o mejor.

Ω ① Voltímetros: Para los voltímetros analógicos deben de tener una resolución de 1 V o mejor. Para mediciones con instrumentos digitales la resolución de 0,1 V o mejor.

- **Tiempo.**

Las mediciones de tiempo se hacen con un reloj eléctrico síncrono de arranque automático o un integrador de tiempo semejante.

3.2. **Preparación del aparato de prueba.**

Se debe de verificar que se cumplan los siguientes puntos:

- No deben funcionar durante la prueba los dispositivos automáticos generadores de hielo.
- Los controles de temperatura para los compartimientos de mantequilla deben funcionar en el nivel de consumo energético mínimo.
- Las luces de cortesía, radios, relojes, lámparas higiénicas y similares deben desconectarse cuando exista la posibilidad mediante un interruptor.
- Los Accesorios operados eléctricamente, de inicio manual y terminación automática deben operar en su nivel de consumo energético mínimo.

- ***Instalación del aparato.***

El aparato se debe instalar con sus paredes laterales a una distancia igual o mayor a 250 mm de cualquier superficie para asegurar la libre circulación del aire. El espacio entre la pared trasera del aparato y la pared del cuarto de pruebas debe ser la que indica el fabricante, o la determinada por topes mecánicos del gabinete; de no existir alguna de estas dos condiciones el aparato se coloca con 50 mm de separación entre la pared y el punto más cercano del fondo del aparato.

- ***Sensores de temperatura.***

Independientemente del tipo refrigerador que se vaya a utilizar es necesario que los sensores de temperatura se distribuyan de tal forma que cada compartimiento del aparato se divide en tres alturas iguales y los sensores se colocan de la parte mas interna al aparato a la puerta correspondiendo con las alturas del mas alto al menos alto.

- ***Carga simulada.***

En las pruebas a refrigeradores solos y refrigeradores-congeladores con deshielo automático no se utiliza carga simulada.

Los refrigeradores convencionales y refrigeradores-congeladores con deshielo manual, deshielo semiautomático y deshielo parcialmente automático y congeladores, con un compartimiento congelador cuyo volumen sea mayor a 14,5 dm³ deben llevar carga simulada en el compartimiento congelador.

La carga simulada consiste en paquetes que miden: 130 mm x 100 mm x 40 mm, $\pm 15\%$. Los paquetes deben sellarse o cubrirse con alguna envoltura que evite se escape la humedad que tienen. Los paquetes de carga deben llenarse con aserrín de maderas duras humedecidas con agua de tal manera que la densidad de los paquetes sea 560 kg/m³ ± 80 kg/m³, o como alternativa a lo anterior, paquetes equivalentes de alimento congelado tales como espinaca picada. El compartimiento congelador debe llenarse con paquetes de carga que ocupen como máximo el 75% de su volumen.

Temperaturas de referencia normalizadas

Aparato	Compartimiento de referencia	Temperatura de referencia
Refrigerador solo	De alimentos	3,3°C
Refrigerador convencional	Congelador	-9,4°C
Refrigerador congelador	Congelador	-15,0°C
Congelador	Congelador	-17,8°C

3.3. Secuencia de las pruebas.

Las pruebas se realizan, después de obtener la condición térmica estable del aparato. La primera prueba requiere que se coloquen los controles de temperatura de los compartimientos en su posición media, entre las posiciones más fría y más caliente del control de temperatura.

La segunda prueba requiere que se coloquen todos los controles de temperatura de los compartimientos en su posición más fría o más caliente que se indique en el control, la que sea apropiada para obtener una temperatura mayor y otra temperatura menor que la temperatura de referencia normalizada que corresponda al tipo de refrigerador que se está probando.

Si las temperaturas obtenidas durante estas dos pruebas son mayor y menor que la temperatura de referencia normalizada, entonces los resultados de las dos pruebas se utilizan para determinar el consumo de energía.

Si la temperatura del compartimiento de referencia, medida con los controles de temperatura en la posición de más caliente, es menor que la temperatura de referencia normalizada y además la temperatura del compartimiento de alimentos es menor a 7,2°C en refrigeradores convencionales y en refrigeradores-congeladores, excluyendo refrigeradores solos, entonces el resultado de esta prueba es el único empleado para determinar el consumo de energía.

3.4. Tiempo de prueba.

Una vez lograda la condición térmica estable del aparato, el tiempo de prueba está determinado por el tipo de deshielo.

Para deshielo manual, semiautomático y parcialmente automático la prueba empieza en un arranque de motocompresor y dura por lo menos 3 h; durante el periodo de prueba el motocompresor debe completar dos o más ciclos. Si no ocurre paro del motocompresor, determinado previamente durante el periodo de estabilización, el periodo de prueba debe ser por lo menos de 3 h.

Si el motocompresor presenta ciclos de motocompresor incompletos (menos de dos ciclos de motocompresor) durante un periodo de 24 h, los resultados del periodo de 24 h deben utilizarse de todas maneras para determinar el consumo de energía.

3.5. Cálculos de los consumos de energía.

- **Consumo de energía durante tiempo de prueba.**

El consumo de energía durante el tiempo de prueba es la que indique el wattímetro desde el inicio hasta el final de la prueba. El consumo de energía de un ciclo durante el tiempo de prueba se ajusta a un periodo de un ciclo, expresándose en kilowatts-hora por día (kWh/día).

En los aparatos con deshielo manual, semiautomático, parcialmente automático y automático el cálculo del consumo energético durante el periodo de prueba se define por:

Donde:

$$EC = \frac{(EP \times 1440 \times K)}{t}$$

EC = Consumo de energía durante un ciclo, en kWh/día

EP = Consumo de energía durante el periodo de prueba, en kWh

1 440 = Factor de conversión para ajustar el tiempo de prueba a un periodo de 24 h

K= Factor de corrección adimensional; 0,70 para congeladores horizontales, 0,85 para congeladores verticales, y 1,00 para refrigeradores electrodomésticos.

t = Tiempo total de la prueba en minutos

- **Consumo de energía en un ciclo.**

La determinación del consumo de energía promedio de un ciclo para un refrigerador convencional y un refrigerador-congelador el consumo de energía se debe expresar en kWh/día, al más cercano centésimo de kWh/día.

Si para la posición de más caliente del control de temperatura; en el caso de un refrigerador convencional, la temperatura del compartimiento de alimentos es menor o igual que 7,2°C y la temperatura del compartimiento congelador es menor o igual que -9,4°C, o es menor o igual que -15°C en el caso de un refrigerador-congelador, el consumo de energía se define como:

$$E = EC2$$

Donde:

E = Consumo de energía promedio del aparato, en kWh/día

EC2 = Consumo de energía durante un ciclo, en kWh/día

Si las condiciones anteriores no se dan, el consumo de energía promedio del aparato se define con el valor más alto calculado por las dos fórmulas siguientes:

$$E = EC1 + (EC2 - EC1) \times \frac{7,2 - TR1}{TR2 - TR1}$$

y

$$E = EC1 + (EC2 - EC1) \times \frac{K - TC1}{TC2 - TC1}$$

Donde:

E = Consumo de energía promedio del aparato, en kWh/día

EC = Consumo de energía durante un ciclo, en kWh/día

TR = Temperatura del compartimiento de alimentos, determinado como se indica en el inciso 9.9, en °C

TC = Temperatura del compartimiento congelador, en °C

7,2 = Temperatura de referencia del compartimiento de comida fresca, en °C

K = Valor constante de -9,4°C en el caso de refrigeradores convencionales y de -15°C en los refrigeradores-congeladores, siendo estos valores las temperaturas de referencia normalizada del compartimiento congelador en cada caso.

Los números 1 y 2 indican las mediciones tomadas durante la primera y segunda pruebas, respectivamente.

- **Consumo de energía anual.**

El consumo de energía anual debe considerarse un periodo de uso continuo a lo largo del año, determinándose de acuerdo a la siguiente fórmula:

$$CA = E \times 365$$

Donde:

CA = Consumo de energía anual, en kWh/año.

E = Consumo de energía en un ciclo, en kWh/día.

365 = Factor de conversión de días a año.

ANEXO 2
PROPUESTA ECONÓMICA

1. Costo de las pruebas.

	Marca	Duración total de la prueba	Costo (pesos)
1. Refrigerador nuevo	Whirlpool	1 días	20,000
		14 días	
2. Refrigerador usado	Mabe	1 días	20,000
		14 días	

Al total se le debe añadir el impuesto al valor agregado.

2. Duración de los trabajos.

6 semanas a partir de la firma del contrato y entrega del anticipo.

3. Condiciones de pago.

Anticipo: 50% a la firma del Contrato de Prestación de Servicios.

Remanente 50% a la entrega del Informe de Resultados de las pruebas.



México D.F. a 5 de Agosto de 2008

At'n Ing. Victor Reyes:
Caribbean Magnetic Systems, Inc.
San Juan, Puerto Rico

REPORTE DEL ANALISIS DE MEDICIONES DEL CONSUMO ELECTRICO DE WESTERN BANK, SUCURSAL PLAZA ESCORIAL SIN Y CON DISPOSITIVOS MAGNÉTICOS.

El siguiente reporte muestra los ahorros en el consumo eléctrico de Western Bank, sucursal Plaza Escorial.

Se analizaron mismos períodos de tiempo (4 meses) esto es; mediciones de consumo eléctrico en kw de Abril a Julio de 2007 como referencia y de Abril a Julio de 2008 con mediciones de consumo eléctrico ya con influencia magnética, ambos períodos con 121 días

La instalación de los módulos magnéticos fue realizada el 15 de Abril de 2008 en una Máquina York de 7.5 ton utilizando gas refrigerante R22.

Los módulos magnéticos instalados fueron los siguientes:

- 1 MIC
- 1 MO
- 1 REW

A continuación se muestran los datos registrados siendo la tabla del lado izquierdo las mediciones de referencia y del lado derecho las mediciones con influencia magnética:

DATOS PROPORCIONADOS CONSIDERANDO TODO EL AÑO

FECHA	DIAS	CONSUMO	CONSUMO/DÍA	COSTO
2007		(kwh)		
Abril	29	3,050	105.17	\$667.42
Mayo	30	3,210	107.00	\$658.18
Junio	36	5,130	142.50	\$1,120.34
Julio	26	3,100	119.23	\$666.19
Agosto	31	3,900	125.81	\$829.54
Septiembre	33	4,350	131.82	\$953.60
Octubre	28	3,530	126.07	\$798.78
Noviembre	33	3,270	99.09	\$816.48
Diciembre	29	3,250	112.07	\$800.32
TOTAL	275	32,790		\$7,310.85
PROMEDIO			119.24	

FECHA	DIAS	CONSUMO	CONSUMO/DÍA	COSTO
2008		(kwh)		
Abril	30	2,850	95.00	\$721.93
Mayo	31	3,200	103.23	\$823.43
Junio	30	3,050	101.67	\$935.66
Julio	30	3,400	113.33	\$1,116.19
TOTAL	121	12,500		\$3,597.21
PROMEDIO			103.31	



ANALISIS:

Valores de referencia en período de 4 meses:

Consumo eléctrico: 14,490 kwh
 Promedio de consumo eléctrico por día : 119.75 kwh/día
 Costo: \$ 3,112.13

Valores después de la instalación de los módulos magnéticos:

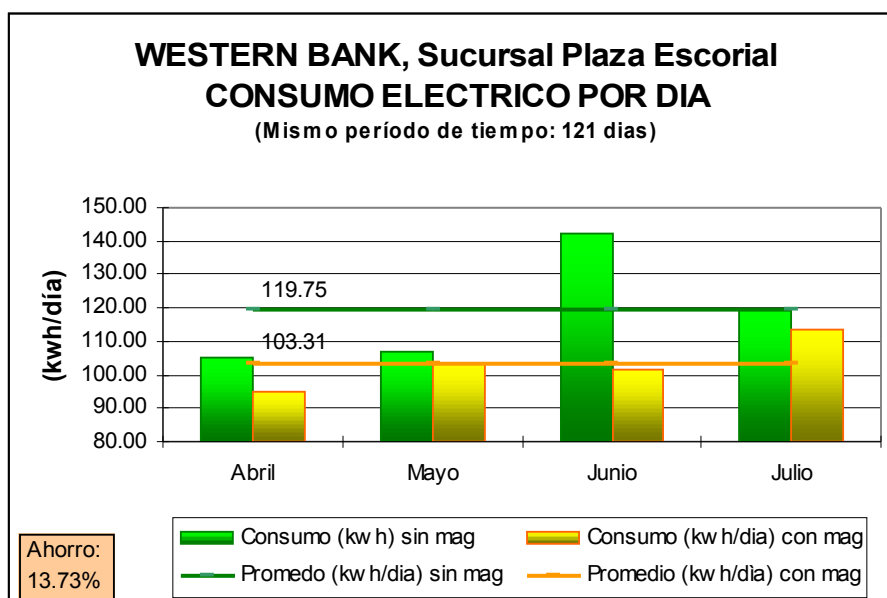
Consumo eléctrico: 12,500 kwh
 Promedio de consumo eléctrico por día : 103.31 kwh/día
 Costo: \$ 3,597.21

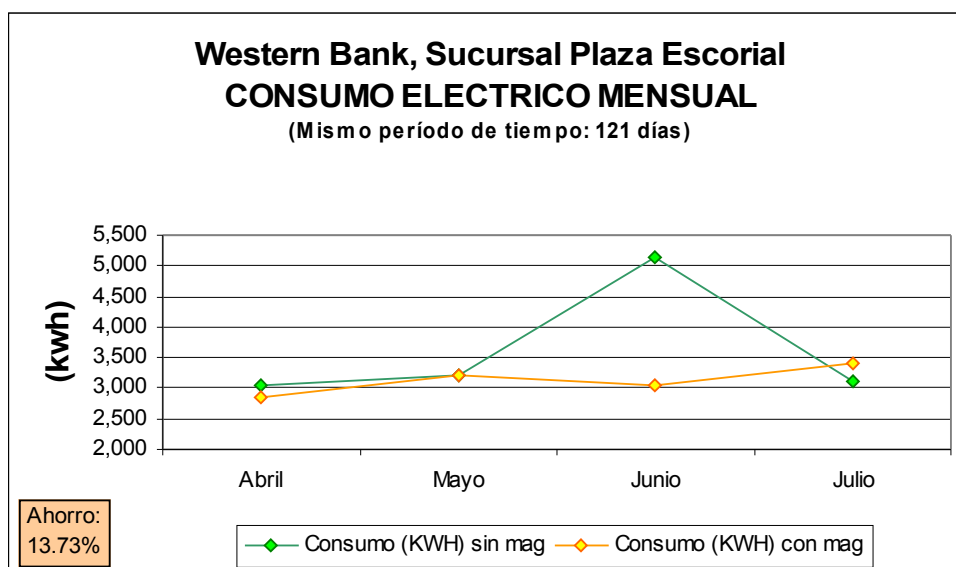
El ahorro obtenido en el consumo eléctrico para el mismo período de tiempo hasta el mes de Julio de 2008 es de **13.73%**

Si consideramos como valores de referencia todo el año de 2007 hasta Marzo de 2008, el ahorro en consumo eléctrico sería del 13.36%

FECHA	DIAS	CONSUMO	CONSUMO/DIA	COSTO	FECHA	DIAS	CONSUMO	CONSUMO/DIA	COSTO
2007		(kwh)			2008		(kwh)	(kwh/día)	
Abril	29	3,050	105.17	\$667.42	Abril	30	2,850	95.00	\$721.93
Mayo	30	3,210	107.00	\$658.18	Mayo	31	3,200	103.23	\$823.43
Junio	36	5,130	142.50	\$1,120.34	Junio	30	3,050	101.67	\$935.66
Julio	26	3,100	119.23	\$666.19	Julio	30	3,400	113.33	\$1,116.19
TOTAL	121	14,490		\$3,112.13	TOTAL	121	12,500		\$3,597.21
PROMEDIO			119.75		PROMEDIO			103.31	
					AHORRO			-13.73%	

A continuación se muestran las gráficas comparativas de los consumos para ambos períodos: sin módulos magnéticos y con módulos magnéticos:





En la siguiente tabla se muestra el precio por kwh correspondiente a cada mes desde Abril de 2007 hasta Julio de 2008. Como se puede observar ha habido un incremento del 50% en el precio por kwh desde Abril de 2007 hasta Julio de 2008, por lo que ahorro obtenido del 13.73% es muy significativo económicamente.

FECHA	PRECIO/KWH
Abril,2007	0.22
Mayo	0.21
Junio	0.22
Julio	0.21
Agosto	0.21
Septiembre	0.22
Octubre	0.23
Noviembre	0.25
Diciembre	0.25
Enero,2008	0.26
Febrero	0.25
Marzo	0.24
Abril	0.25
Mayo	0.26
Junio	0.31
Julio	0.33



Para principios del mes de Septiembre de 2008 queda cubierta la inversión siendo la tasa interna de retorno anual de :

TIR	1er año	140%
	2o año	240%

DISMINUCION DE CONTAMINANTES

1 KWH =0.600 kg de CO2 (disminución)

298.50 kg de CO2

3,582.00 En 1 año

3.58	ton CO2/año
------	-------------

\$15	Precio CERs(USD/Ton CO2)
------	--------------------------

\$53.73	Beneficios por carbono
---------	------------------------

Como se puede ver existe además un incentivo en la disminución de CO2 que se emite a la atmósfera.

CONCLUSIONES:

- El ahorro obtenido en el consumo de energía eléctrica es del 13.73%
- Se obtiene un ahorro mensual promedio de energía de 497.50 kwh
- El ahorro económico promedio mensual es de \$ 164.18
- El tiempo de retorno de la inversión es de 5.21 meses.
- Se tiene una tasa interna de retorno anual del 140% en el primer año.
- Se contribuirá a un mejor ambiente con la disminución de 3.58 ton/año de CO2

CARIBBEAN MAGNETICS

INSTALACIONES DE MAGNETIZER EN P.R.

AAP Air Parts **A/c -----2004**
 AGRO JOAQUÍN PAGÁN – 1999
 A.J. Meat Packing – 1994
 Agua Light **2002**
 AVC Ice Plant – 1999 *****
 Autoridad de Energía Eléctrica – 1997
 Barbería Los Muchachos - 1998
 Baxter Pharmaceutical – 1996
 Beneficiado Abel Velez—**2003**
 Beneficiado Angel Delgado--**2003**
 Beneficiado José Pietri --**2003**
 Beneficiado José Torres--**2003**
 Beneficiado La Isolina -**2002**
 BMF Inc. (planta destiladora) – **2000**
 Burger King **(7) 2003-2004-2005-2008**
 Baxter -----**2007**
 Cutro Elementos Finca Vegetales--**2003**
 Café del Alba – 2000 **A/C 2005**
 Café Max 1998
 Café Cialito – 1997
 Café Sanders – 1996
 Café Lares -----**2003**
 Caribbean Refrescos – 1997
 College Food Service – 1999
 Cold Storage **2005**
 Crespo Ice Plant – 1998
 El Hamburger – 1998
 Escuela Montessori--**2005**
 Empresas Badillo (18 Sup. Grande) – 1998
 Finca Raul Noriega-- **2005**
 Hacienda del Carmen – 1999
 Hidropónicos La Floresta – 1999
 Hidropónico Lares (tomates) **2004**
 Hi-Tech Auto Lavado – 1997
 Hospital Auxilio Mutuo – 1997—**2004-A/C2005**
 Hospital Menonita – 1995
 Hospital Pavía – 1997
 Hospital Previsteriano **2005 A/C**
 Hotel Atlantic Beach – 1991
 Hotel Holiday Inn, Islam Verde – 1996
 Hotel Parador Perished – 1997
 Parador Bahia Salinas ----**2007 A/C**
 J R Manufactory (yate) – **2000**

Hidroponico Hydro-Tec - **2000**
 Hidroponico Tomates **2004**
 Lavadería Principado – 1993
 Lavandería Industrial Valines – 1999
 Luis Noriega (motoras) – 1999
 J.J.C. Industries Service – 1999
 King's Laundry – 1997
 McQuay **A/C----2004 –2005**
 McDonald **(2)-----2007**
 Mario's Pizza (Trinidad) – 1994
 McNeil Pharmaceutical – 1997
 Medic's (Pura Agua) – 1995
 Moca Ice Plant – **2000**
 Mardel Pan Congelado – **2002**
 Panadería Andaluza – 1997
 Panadería Golden Bagel – 1997
 Panadería La Criolla -1996
 Panadeoria Maruxa – 1997
 Panaderia Milenium ---**2007 Refrig.**
 Panaderia La Viequese – **2002**
 Pfizer Pharm. –**2003 ---2004**
 Piazzola Pizza – 1998
 Pizzeria Antonino's (1) - **2001**
 Pizzeria Scoop, Vieques **2002**
 Pollos Picú – 1995
 Ponce Hilton –1999
 Pollos To Rico -----**2007**
 Refresqueria Cupey – **2000----A/C 2004 *******
 Triple S -----1998
 Rest.LongHorn -----**2006-07 (2)**
 Rest.Sizzler -----**2006 (1)**
 Rest.The Red Snapper – **2000**
 Rest. Bounty Steak House --- Gas– 1992
 Rest. Patio de Sam – 1991
 Rest. El Pescador – 1997
 Rest. Ponderosa **(1) ---2007**
 Rest. La Mallorca – 1991
 Rest. Vegetariano El Mezón – 1996
 Rest.Puerto Rico –**Gas-----2008**
 Rexnord – 1994
 SKB Pharmaceutical – 1996
 Sam's Club -----**Gas-----2004**
 Sweet & Cake Bakery – 1999
 San Lorenzo Ice Plant – 1998
 Vaquería Tres Monjitas – 1997
 YMCA -----1991

Unidades Magnetizer han sido instaladas en torres de enfriamiento, auto claves, calderas, calentadores, piscinas, jacuzzis, calderas de saunas, hornos de gas, suavizadores, estiladores, máquinas de hielo, “reverse osmosis”, condensadores, cafeteras, e hidropónicos aire acondicionado y refrigeración.

Effect of magnetic field on the performance of new refrigerant mixtures

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SUMMARY

Performance test results of new alternative refrigerant mixtures such as R-410A, R-507, R-407C, and R-404A under various conditions of magnetic field are discussed, analysed and presented. The test results were obtained using an air-source heat pump set-up with enhanced surface tubing under various magnetic field conditions. Performance tests were conducted according to the ARI/ASHRAE Standards.

The test results demonstrated that as magnetic field force increases, compressor head pressure and discharge temperature slightly increase as well as less liquid refrigerant is boiling in the compressor shell. This has a positive effect in protecting the compressor. The effect of magnetic field on mixture behaviour varies from one mixture to another depending upon the mixture's composition and its boiling point. Furthermore, the use of magnetic field appears to have a positive influence on the system COP as well as thermal capacities of condenser and evaporator. Copyright © 2002 John Wiley & Sons, Ltd.

KEY WORDS: new refrigerant mixtures; magnetic-caloric effect; ARI/ASHRAE standards

1. INTRODUCTION

It is well known that certain materials properties and particularly temperature will increase when placed in a magnetic field and, will likewise decrease when the magnetic field is removed. This mainly is caused by the effect of the magnetic field on the entropy and the heat content of material. This phenomenon is known as magnetic-caloric effect. The effect of magnetism and magnetic field on fluids is still considered as not well-known subject. However, it is well established that there are major changes caused by the passage of fluid through magnetic field.

Several magnetic refrigeration devices under development by Astronautics using convention NbTi magnets have been described by Zimm and De Gregoria (1992). The system advantages of incorporating high-temperature super conducting magnets in designs have been discussed. The authors also explained the nature of active magnetic regenerative (AMR) cycle and its requirements in refrigeration.

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Research on magnetic-caloric effect and its application have been discussed by Gschneidner and Pecharsky (1999) for cooling near-room temperature. The study included the relationship between the nature of magnetic transformation and the temperature dependence of the magnetic-caloric effect and the entropy utilized in the magneto caloric.

The magnetic measurements to evaluate the thermodynamic behaviour of magnetic material have been presented by Foldeaki *et al.* (1995). As reported in this reference, depending on the thermodynamic cycle selected, the isothermal magnetic entropy temperature change or the adiabatic temperature change upon the field application should be preselected as a function of temperature. This paper presented classical magnetic measurements, when evaluated within the framework of the Landau theory.

A magnetic heat pumping can be made according to Brown (1976) using a ferromagnetic material with a curie point and an appropriate thermodynamic cycle. The regenerative magnetic cycle can approach the Carnot cycle efficiency, as reported by Brown.

Furthermore, most refrigeration and air-conditioning systems experience load variation. High efficiency and high performance are greatly in demand. Among techniques employed for improvement, capacity control, optimization of vapour compression systems are the refrigerant liquid and vapour injection. However, it is believed that the magnetic field can be employed as an enhancing technique.

The several studies reported in the literature demonstrated the magnetic field and its capabilities as well as its impact on the thermodynamic characteristics. However, as the EHD technique has shown an improvement of the heat transfer on refrigerant side (Muraki *et al.*, 2001), it is believed that magnetic field could have an enhancement effect on heat transfer properties. Several studies have been reported on the use of magnetic elements in enhancing the performance in many applications such as oil, natural gas furnaces, diesel engines, fuel lines and also in water treatment. To the authors knowledge none has been reported on the use of magnets as a performance enhancer in the refrigeration industry.

Therefore, this paper is concerned with the study and analysis of some refrigerant mixtures behaviour inside enhanced surface tubing air finned heat exchangers under magnetic field at various forces of the magnetic field (Gauss levels). The blends under consideration in this study are; R-507 (R-125/R-143a:50/50%), R-404A (R-125/R-143a/R-134a:44/52/4%), R-410A (R-32/R-125:50/50%), and R-407C (R-32/R-125/R-134a:23/25/52%). All percentages of the aforementioned blends are based on weight. The main thrust of this study is to study the enhancement of the heat transfer rates and system coefficient of performance and optimize the use of magnets in refrigeration systems.

2. EXPERIMENTAL APPARATUS AND MEASUREMENTS

Figure 1 shows a schematic diagram of the experimental set-up, which is an air-source vapour compression heat pump, composed mainly of a 3 kW compressor, oil separator, condenser, precondenser, pre-evaporator, adjustable expansion device, capillary tubes and evaporator.

Three magnetic elements with Gauss level of 4000 each have been employed in this study. These magnets were intended for gasoline fuel line of 1/4 in diameter, they were clamped at the refrigerant line of same diameter. The units were single-type with two brackets strapped around the pipe. They were clamped on the refrigerant liquid line at the post-condenser outlet at various distances, before the capillary tube/thermal expansion valve used as flow control device. During

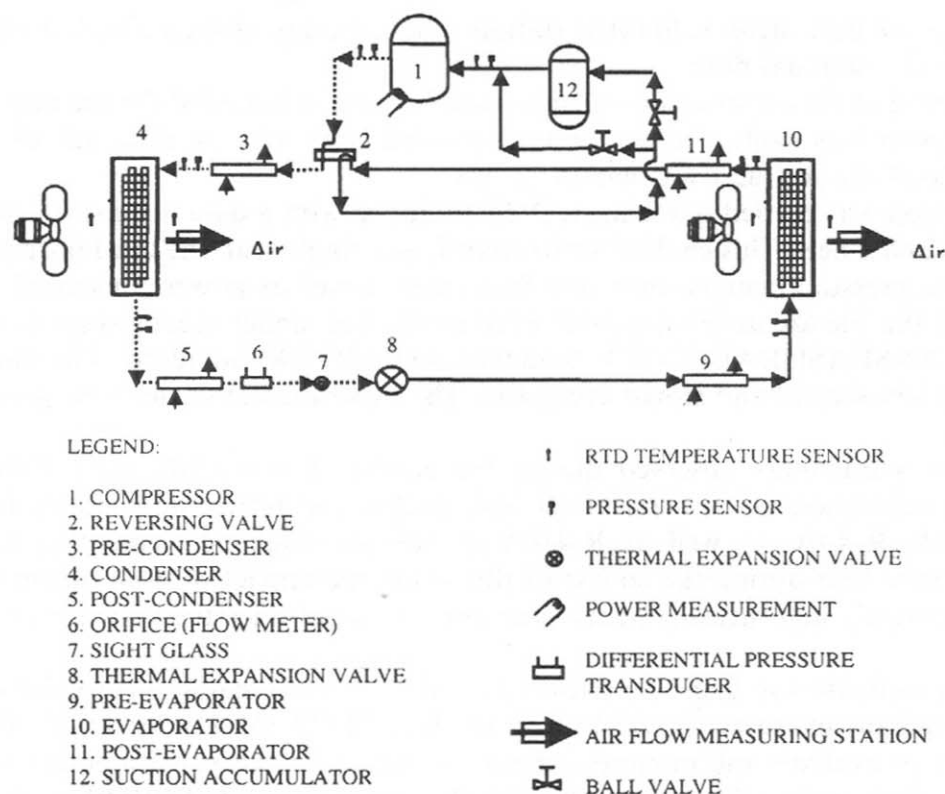


Figure 1. Schematic diagram of the air/air heat pump test facility.

the course of this experimental study, series of three magnets were used totalling 12 000 Gauss levels. The magnets were placed at three locations of the post-condenser outlet at a distance of 0.13, 0.47 and 0.71 m, respectively. This was necessary to ensure that the magnets are placed on the refrigerant full liquid line. This is confirmed by observation through the sight glass. At each position, experiments were conducted with one, two and three magnets placed at 0.01 m spacing between each other. Figure 1 depicts the set-up flow diagram and magnet positions.

The oil content in the refrigerant loop was estimated to be about 1% using gas chromatography. Pressure, temperature and flow rate measuring stations are shown in Figure 1. All pressures were measured using calibrated pressure transducers (0–800 kPa). The accuracy of the pressure transducers was $\pm 2.5\%$. Differential pressure transducers were employed to measure the refrigerant flow rate. Temperatures were measured by RTD temperature sensors which have an accuracy of $\pm 0.5\%$. Humidity measurements were obtained through the accurate recording of dry and wet bulb temperatures.

All recorded measurements were obtained at a variable sink and source air temperature of 21°C entering the condenser. On the other hand, the capillary tubes were adjusted to optimize the system's performance with every tested refrigerant mixture. This simulates the system's performance at different thermal loads and using a variable thermal expansion valve.

A calibrated orifice, installed in the liquid refrigerant line after a liquid receiver, was used to measure the refrigerant mass flow rate. Pressure taps on both sides of the orifice were connected to a differential pressure transducer (0–250 kPa). Air mass flow rate was also measured by a

Pitot tube-type air flow meter calibrated station. The accuracy of the mass flow measurements was $\pm 3\%$ of the nominal flow.

Power supplied to the compressor was measured because it is needed for the heat balance. An AC/DC clamp-on was calibrated for power measurements with an accuracy of $\pm 3\%$. The energy balance of the test unit was within $\pm 3\%$.

Data collection was carried out using a P150 equipped with a data acquisition system with a capacity of 112 channels. This enabled us to record, at a single scan the local properties such as: pressure drops, pressure, temperature, and flow rates as well as power consumption.

All tests of the blends under question were performed under steady-state conditions and according to ANSI/ASHRAE 37-1978 Standard and ARI-240 Standard. The data collection was scanned every second and stored every 10 s. The experimental values were averaged over a period of 10 s.

The primary parameters observed during the course of this study were: mass flux, heat flux, thermal capacities, power consumed and quality for refrigerants under investigation; R-507, R-404A, R-407C as well as R-410A at various magnetic element conditions. It is important to note that during the course of this study the impact of each magnetic condition was tested separately and measurements were only recorded once the system reached a stable condition.

In order to evaluate the blend's performance, the thermodynamic properties of pure and zeotropic refrigerant mixtures should be known. REFPROP (McLinden *et al.*, 1998) version 6.01 was used to evaluate the mixture's characteristics. Interaction parameters were selected with caution, since their values may influence the outcome of REFPROP prediction of the thermodynamic and transport properties. Interaction parameters are the mixing parameters of refrigerant mixtures.

Test conditions and coil specifications of the heat exchangers used employed in this study are given in Tables I and II. The geometrical parameters of the micro-fin tubes are also presented in Table III.

Table I. Air coils specifications.

Tube outer diameter	3/8"
Rows deep	4
Fin per inch	12
Fin depth	3.46"
Fin height	20"
Fin length	30"
Fin thickness	0.0045"
Rifled tubes	Microfins

Table II. Test conditions.

Temperature of air of the condenser inlet	21°C
Temperature of air at the evaporator inlet	-15 to +8°C
Air flow rate	7.07×10^{-2} – $9.4 \times 10^{-2} \text{ m}^3 \text{ s}^{-1}$
Refrigerant mass flow rate	8–40 g s ⁻¹
Condenser pressure	600–1800 kPa
Evaporator pressure	170–450 kPa
Standard relative humidity at condenser inlet	45%

Table III. Geometry of micro-fin tubes, (round tip geometry).

Outside diameter	0.375"
Root diameter	0.344"
Tip diameter	0.331"
Fin height	0.0074"
Pitch	0.016"

3. RESULTS AND DISCUSSION

In the following sections, samples of the system's performance with some new alternative refrigerants under various magnetic fields conditions will be presented, discussed and analysed. The test conditions were: condenser pressure varied between 600 and 1800 kPa, and the condenser refrigerant temperature was between 28 and 38°C. The evaporator pressure ranged from 170 to 450 kPa, and the refrigerant evaporator temperature was between -20 and -6°C. Under these conditions, and at each test, the following parameters have been measured: thermal capacities at evaporator and condenser sides, power consumed by compressor, refrigerant flow rates, coolant flow rates and refrigerant quality at both evaporator and condenser sides.

The aforementioned measured/calculated parameters, such as power consumed and thermal capacities at the evaporator and condenser, are necessary for evaluating the coefficient of performance (COP) under heating and cooling modes. However, only the heating mode was considered in this study.

The COP and heat absorbed/released at system heat exchangers are calculated as follows:

$$\text{COP} = \frac{\text{Heat absorbed/released}}{\text{Compressor power}} \quad (1)$$

and

$$Q_{a/r} = \dot{m}_f C_{p,f} \Delta T \quad (2)$$

$$Q_{a/r} = \dot{m}_f \Delta H \quad (3)$$

where $\dot{m}_f$ and ΔT represent the air mass flow rate, and air temperature difference across the evaporator/condenser coils. $C_{p,f}$ is the specific heat for airflow and ΔH gives the total air enthalpy difference across the heat exchangers.

Equations (2) and (3) represent the heat exchanger sensible and latent heats, respectively. Equation (3) is employed particularly during cooling load calculation.

As mentioned during the course of this study, only heating tests were conducted. Therefore, only the sensible heat was considered in calculating the thermal capacities as shown in Equation (2).

The results of the various refrigerant mixtures with no magnets were used as a baseline for this study. Upon completion of the baseline results of each refrigerant mixture, under the aforementioned conditions, the compressor and the system was drained and evacuated. Following this step, the system was then recharged with the preferred refrigerant mixture. This procedure was repeated before conducting the series of tests for every single alternative mixture.

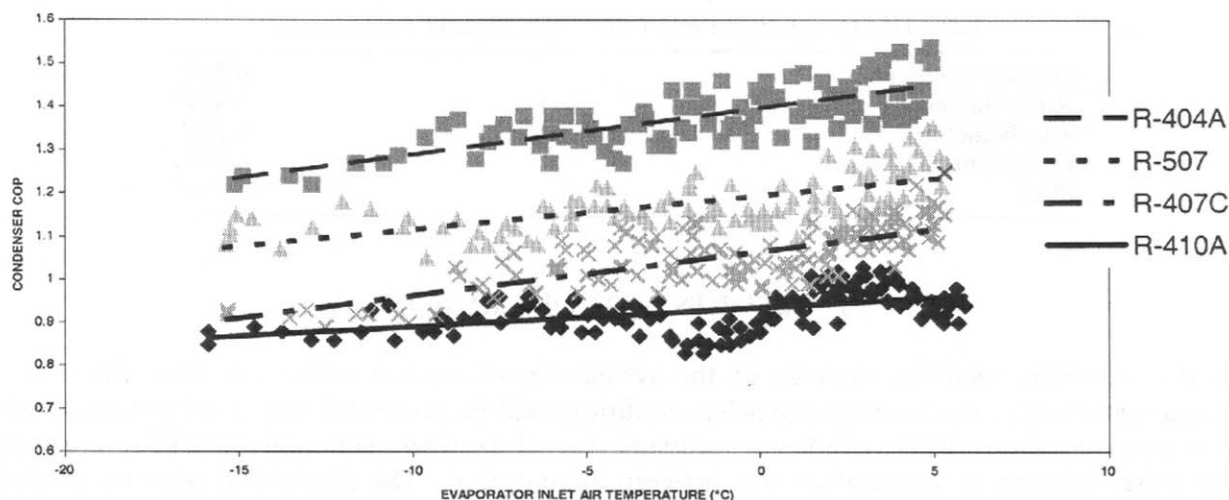


Figure 2. Condenser COP vs evaporator inlet air temperature with no magnets.

In the following, the functional dependence of the following parameters on the refrigerant temperature entering the evaporator will be outlined; input power, thermal heating capacity at the condenser side, pressure ratio, condenser pressure, evaporator thermal capacity and COP.

During the experimentation, the sink's air temperature was kept constant at 21°C, and relative humidity was also kept at 45%. The source coolant temperature varied from -15 to 5°C at the evaporator inlet. The dry and wet bulb temperatures of the source and sink were within the ASHRAE and ARI Standards.

Samples of the results obtained during these runs and used as baseline data with no magnets were plotted in Figure 2 at various entering air temperatures to the evaporator. As expected, the results plotted in this figure show that the COP heating, increases at higher entering air temperature at the evaporator side. The results also demonstrate that R-404A has the highest COP among the mixtures under investigation.

In order to study the influence of the number of magnets on the behaviour of the refrigerant mixtures and the system performance samples of the test, results were plotted at various conditions in Figures 3–12. On the other hand, it appears from the sample results displayed that the magnets accelerated the increase of the COP compared to the no magnets results. Furthermore, R-404A appears to show the highest performance at lower evaporation temperatures and this is mainly due to its significant latent heat at low temperatures.

It is quite clear from Figure 4 that the evaporator COP was enhanced with different percentages depending on the type of mixture and its boiling point. This has a positive and significant impact on the cooling capacity as well as the efficiency of the system performance. It is quite clear from the data in Figure 4 that R-507 behaviour is significantly influenced by the magnetic field force (Gauss levels) and power of the magnets. On the average, it appears that higher Gauss levels enhanced the evaporator COP by 20% depending upon the refrigerant mixture boiling point. It is believed that the magnetic effect or field changes the polarity of oil which is a hydrocarbon from negative charge to positive charge. This results in entraining the oil and being carried away from the heat transfer surface, thus enhancing the heat transfer rate and coefficient of heat transfer. Thus, the consequent effect on the COP has been caused by this

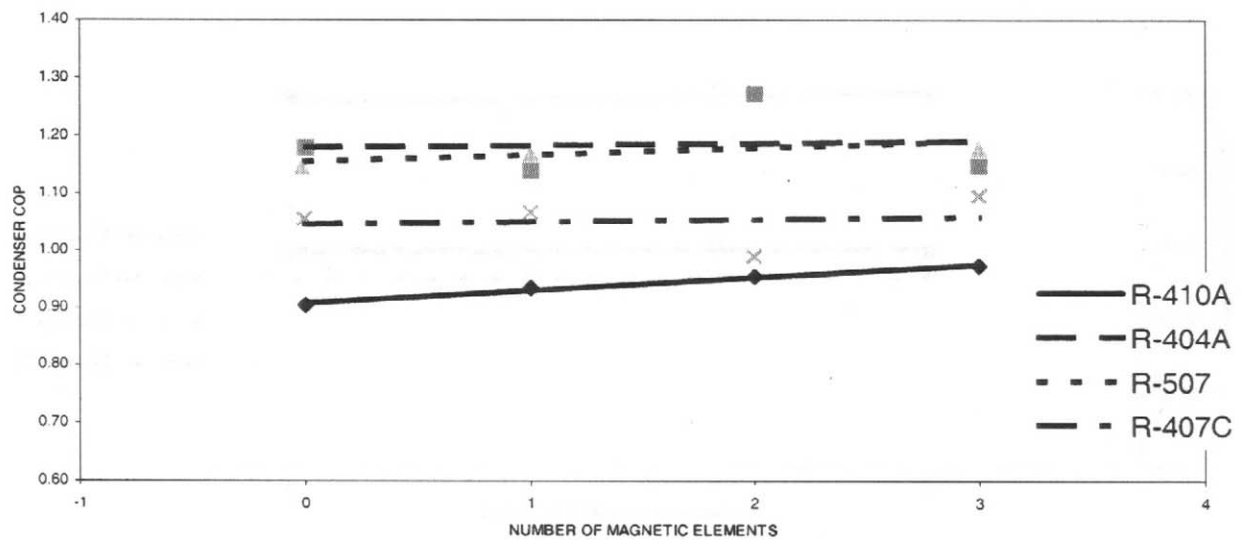


Figure 3. Condenser COP vs number of magnetic elements at constant temperature (0°C).

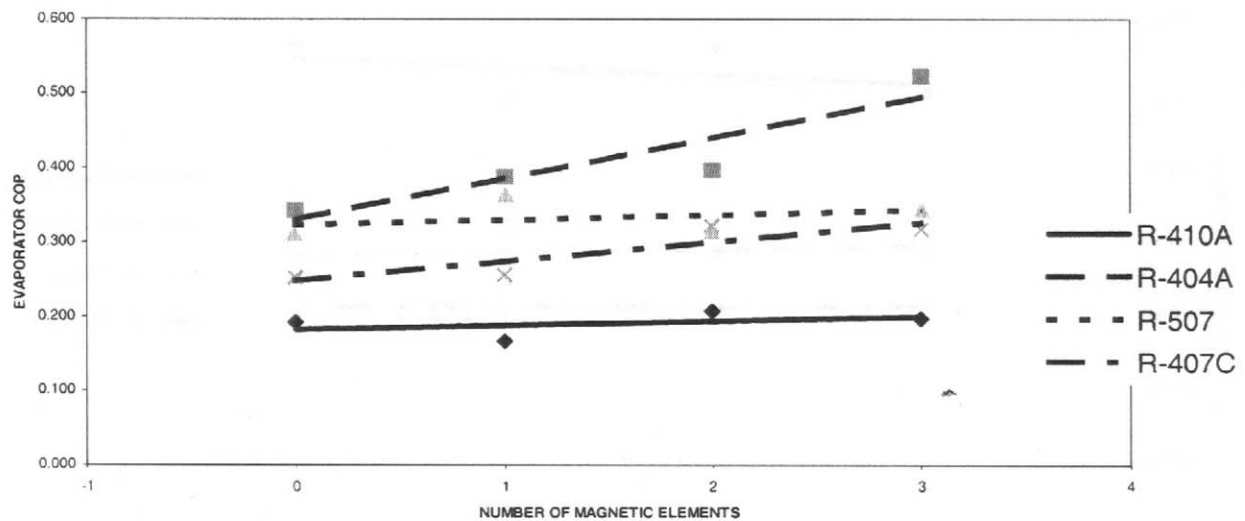


Figure 4. Evaporator COP vs number of magnetic elements at constant temperature (0°C).

phenomenon. It was also shown that higher Gauss levels increase thermal capacities. Previous studies have shown that oil entertained in the refrigerant flow results in degrading the heat transfer rates (Sami *et al.*, 1993).

Furthermore, since the condenser's COP is an important parameter in evaluating the cycle performance, and is calculated as function of the compressor power, Figure 5 has been constructed to show the impact of the magnet Gauss levels on the power consumed. It appears that the magnets reduce slightly the power consumption of the compressor. Also higher Gauss levels results in decreasing the power consumption. The decrease of the power consumption appear to be around 8%. It is believed that increasing Gauss levels decreases the compressor

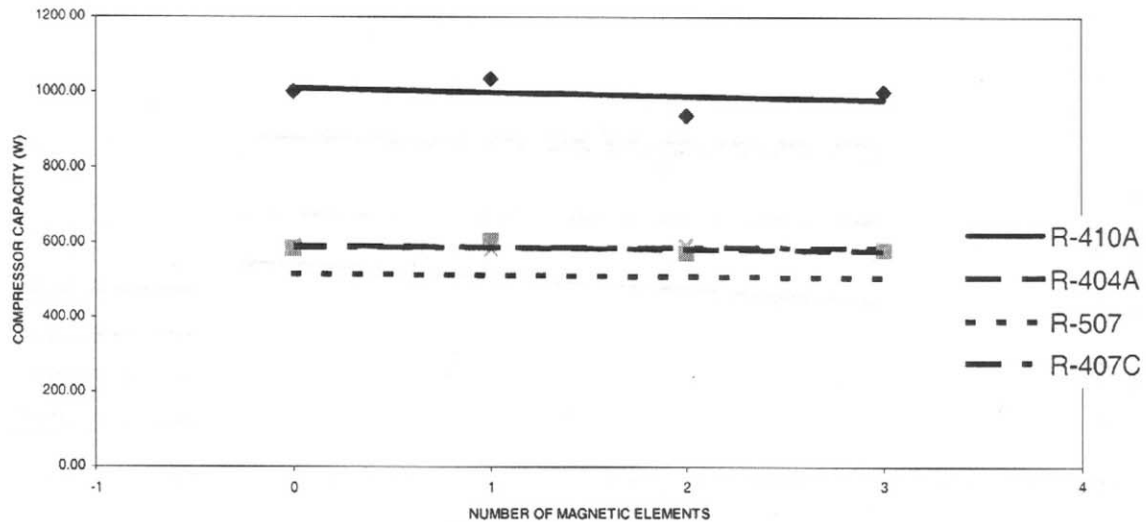


Figure 5. Compressor capacity vs number of magnetic elements at constant temperature (0°C).

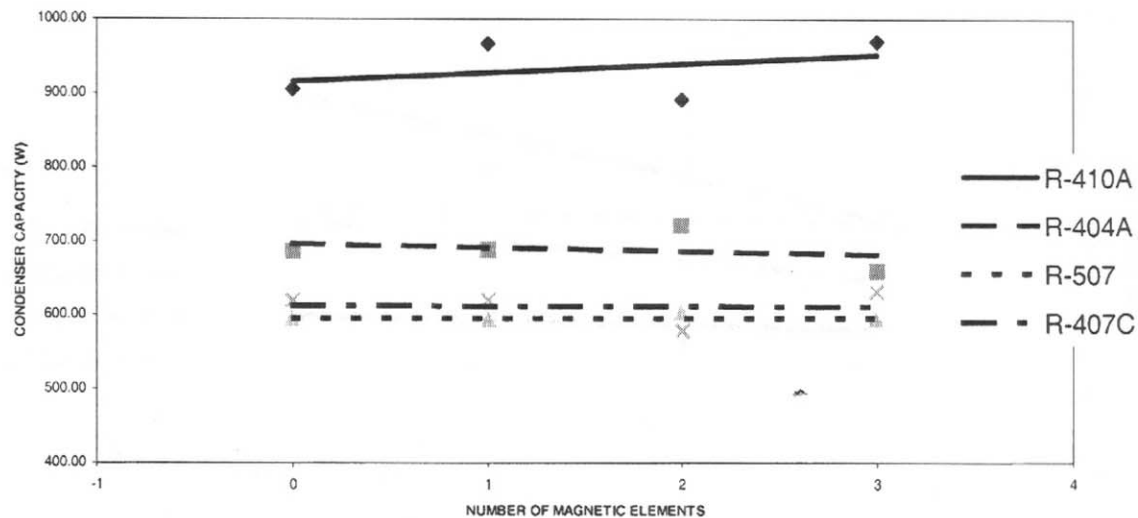


Figure 6. Condenser capacity vs number of magnetic elements at constant temperature (0°C).

power and therefore enhance the COP. This is a result of efficient boiling and condensation and less liquid being boiled at the compressor shell. It is interesting to note that the mixtures with higher latent heat seem to exhibit slower decrease in the power consumption.

In order to understand the impact of the magnetic field on the condenser capacity, Figure 6 has been plotted for the refrigerant mixture in question. Slight increase in the condenser capacity was observed. However, R 410-A showed the highest increase in capacity with the increase of Gauss levels.

Figure 7 shows the evaporator capacity under various Gauss levels. The data presented in this figure clearly demonstrated the enhancement of the evaporator capacity with the increase in the

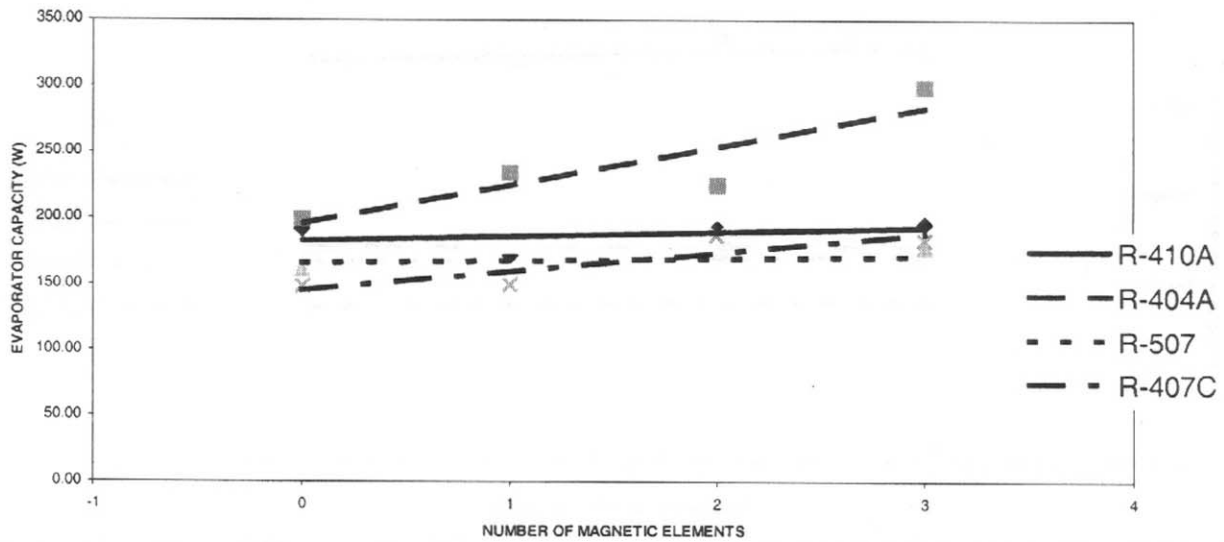


Figure 7. Evaporator capacity vs number of magnetic elements at constant temperature (0°C).

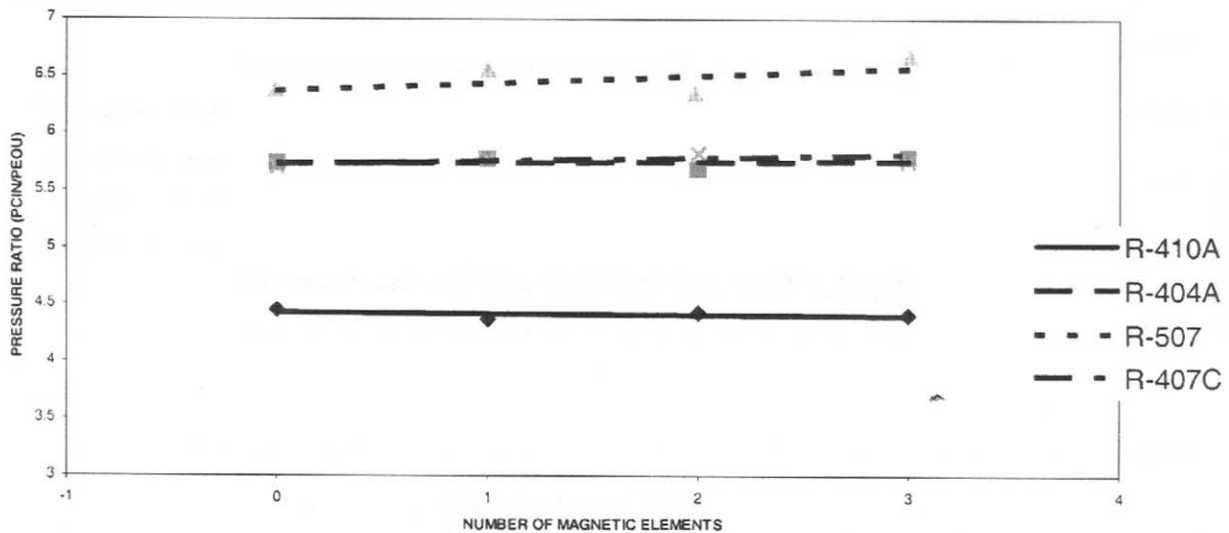


Figure 8. Pressure ratio vs number of magnetic elements at constant temperature (5°C).

number of magnetic elements used. It also appears that the R-404A experiences the highest enhancement among the refrigerant mixtures under investigation. This is mainly due to the low boiling temperature of the mixture and the high latent heat at the test temperature.

The pressure ratio represents the ratio between the discharge and the suction pressure across the compressor. Figure 8 displays results observed with various magnetic element. The results show that the pressure ratio has been slightly increased with the increase in the number of magnetic elements. This is expected since less liquid refrigerant is being boiled in the compressor shell and therefore, this results in increasing the pressure ratio. This trend has been observed with other refrigerant mixtures under investigation.

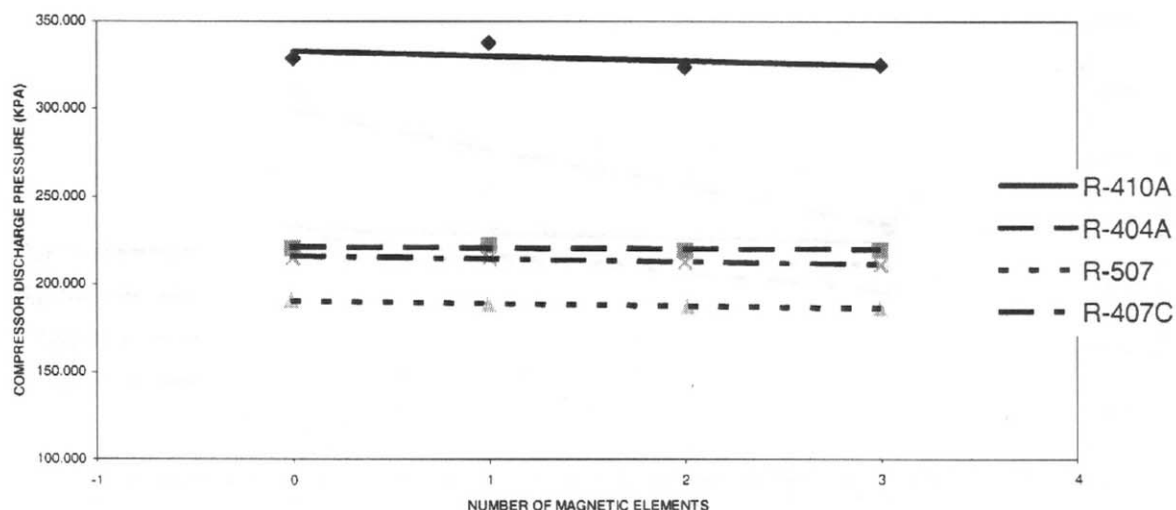


Figure 9. Compressor discharge pressure vs number of magnetic elements at constant temperature (0°C).

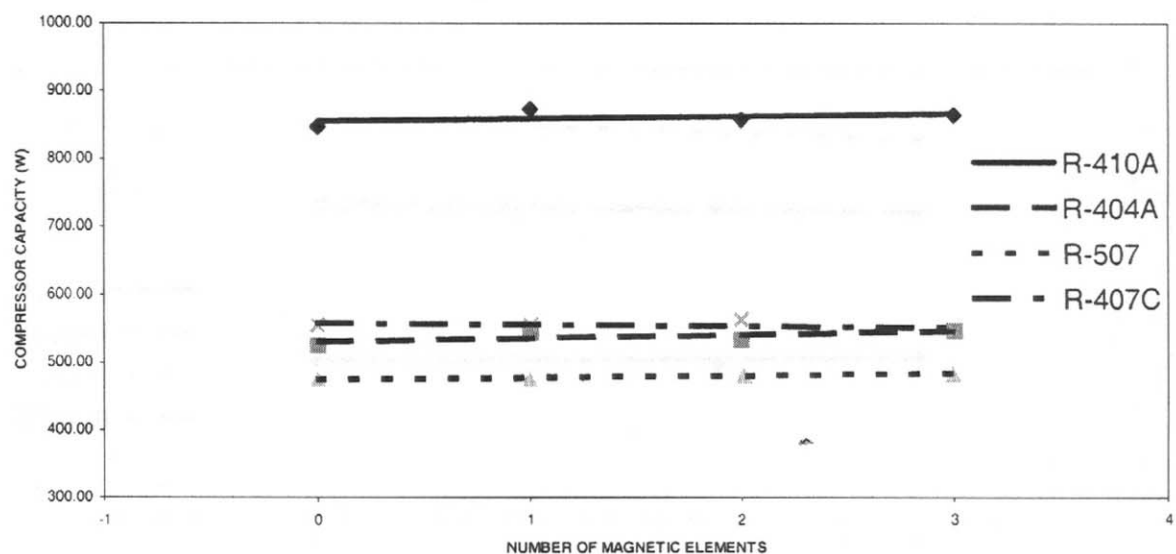


Figure 10. Compressor capacity vs number of magnetic elements at constant temperature (-15°C).

The compressor head pressure and discharge temperature are important parameters to be considered when selecting an alternative mixture, therefore, Figure 9 has been constructed to study these parameters. Figures 4–6 gives clear evidence that as the magnetic field force increases COP of the condenser and evaporator increase. However, it also appears that higher Gauss levels slightly decrease the discharge pressure. It is suggested that this is a result of less liquid refrigerant being carried into the compressor chamber. Furthermore, these results clearly indicate that R-410A has superior compression properties compared to the other blends under investigation.

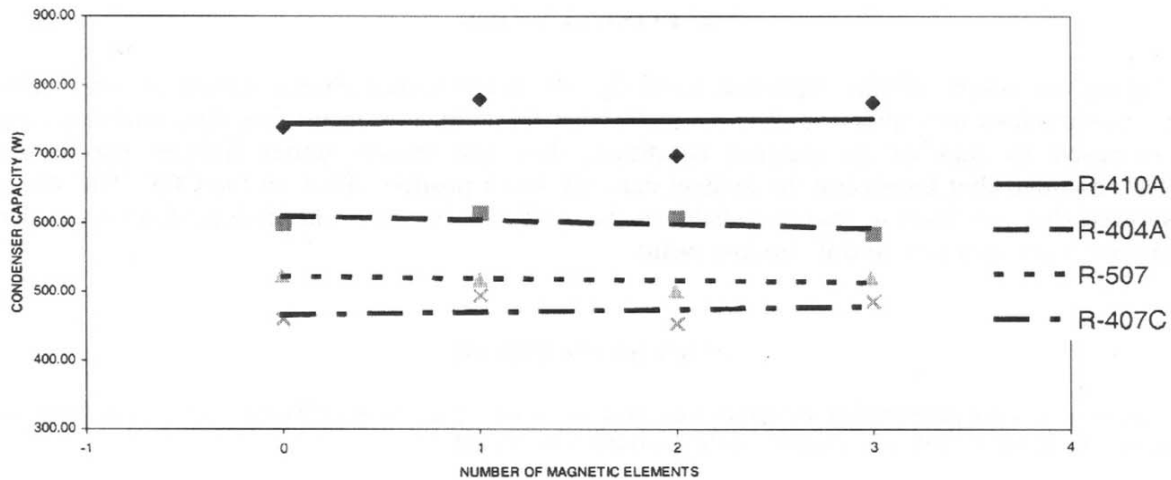


Figure 11. Condenser capacity vs number of magnetic elements of constant temperature (-15°C).

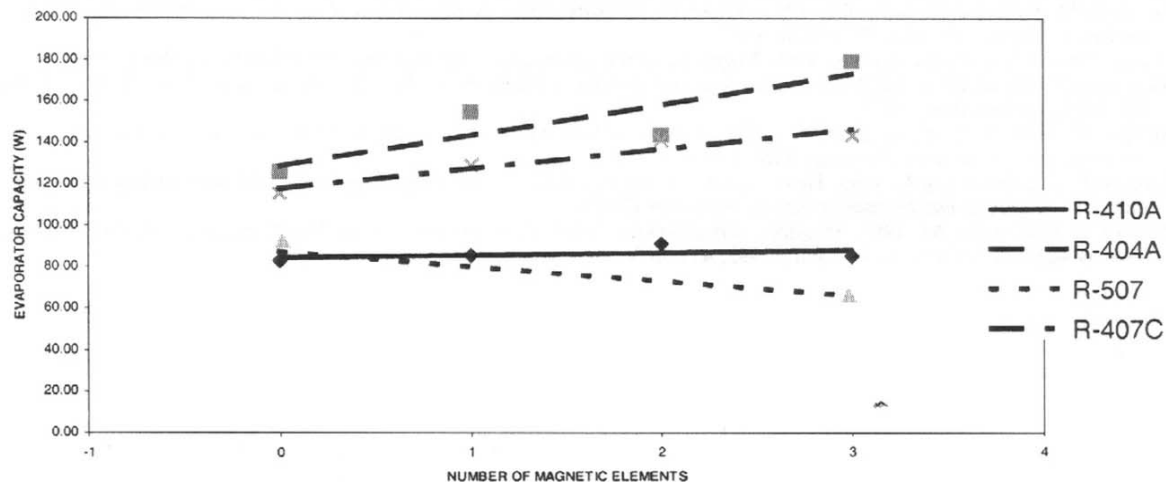


Figure 12. Evaporator capacity vs number of magnetic elements of constant temperature (-10°C).

Based on the above results and the additional information presented in Figures 4–6, it appears from Figure 10 that lower evaporation temperatures seem to slightly increase the compressor capacity with higher Gauss levels. This is quite expected since higher Gauss levels enhance the evaporator capacity and therefore, reduce refrigerant liquid boiling in the compressor shell. Figure 11 also demonstrates that higher magnetic fields results in enhancing the heating capacity of the system even at low evaporation temperatures. This figure also shows that the refrigerant mixture in question respond similarly to the increase of the magnetic field force. Figure 12 displays sample results on the impact of magnetic field force on the cooling capacity of the evaporator. Refrigerant mixture exhibit almost the same behaviour with increase of magnetic field levels however, it appears that R-507 experienced slight decrease in the evaporator capacity.

4. CONCLUSIONS

During the course of this experimental study, the performance characteristics of some new proposed substitutes under various magnetic field levels have been investigated, analysed and compared to that of no magnet condition. The test results under heating conditions demonstrated that increasing the magnet capacity has a positive effect on the COP. The study showed that the effect of magnetic field on the mixture behaviour varied depending upon the mixture's composition and its boiling point.

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