INTERNATIONAL JOURNAL OF ENERGY RESEARCH
Int. J. Energy Res. 2003; **27**: 203–214 (DOI: 10.1002/er.868)

Effect of magnetic field on the performance of new refrigerant mixtures

Samuel M. Sami*,† and Shawn Aucoin‡

Mechanical Engineering, School of Engineering, University of Moncton, Moncton, NB, E1A 3E9, Canada

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.

*Correspondence to: S.M. Sami, Mechanical Engineering, School of Engineering, University of Moncton, Moncton, NB, E1A 3E9, Canada.
† Professor and Director of the Research Centre for Energy Conversion.
‡ Research Assistant.

Contract/grant sponsor: NSERC.
Contract/grant sponsor: University of Moncton.

Received 9 April 2002
Accepted 5 June 2002

Research on magnetic–caloric effect and its application have been discussed by Gschneidddner 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 fined 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

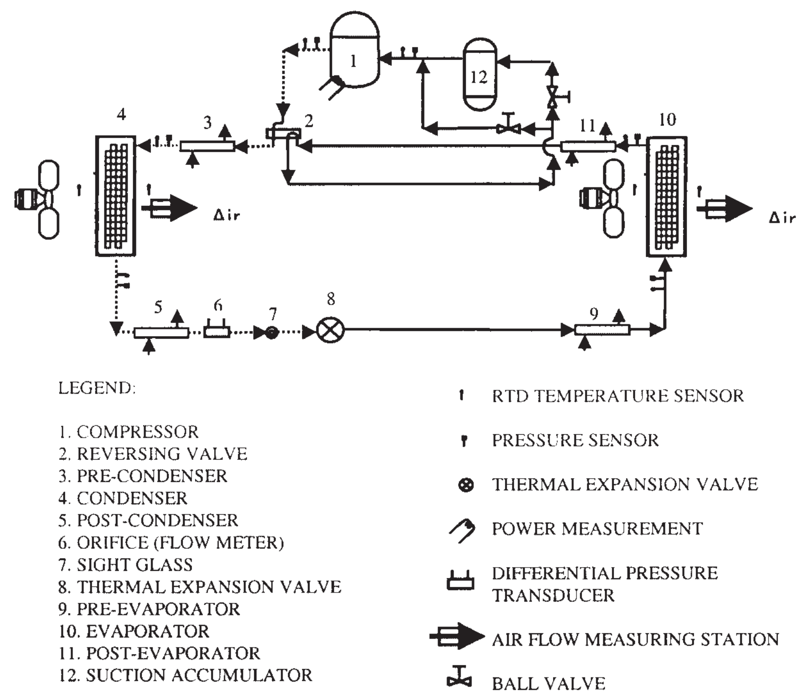

Figure 1. Schematic diagram of the air/air heat pump test facility.

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 preformed 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}–$9.4 \times 10^{-2}\,m^3\,s^{-1}$
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:

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

and

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

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

where $\dot{m}_\mathrm{f}$ and ΔT represent the air mass flow rate, and air temperature difference across the evaporator/condenser coils. $C_{p,\mathrm{f}}$ 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.

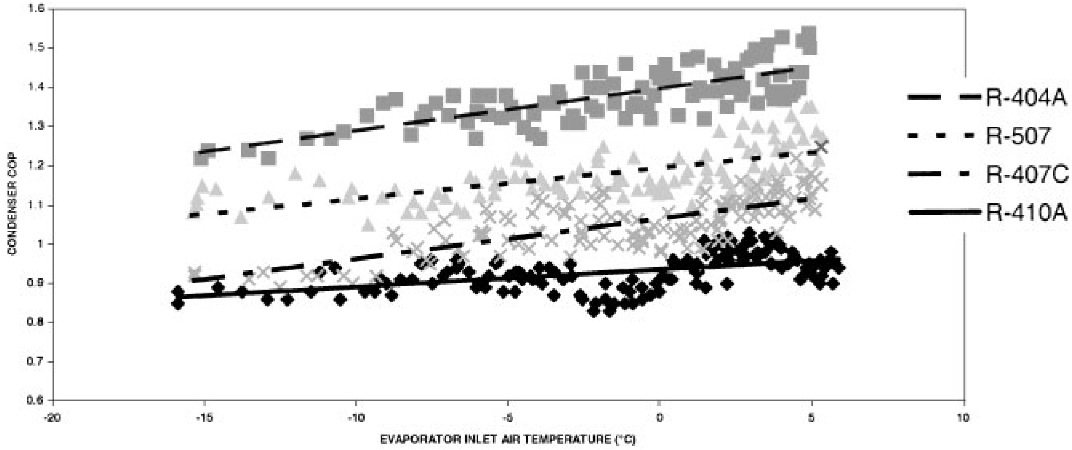

Figure 2. Condenser COP vs evaporator inlet air temperature with no magnets.

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 entertaining 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

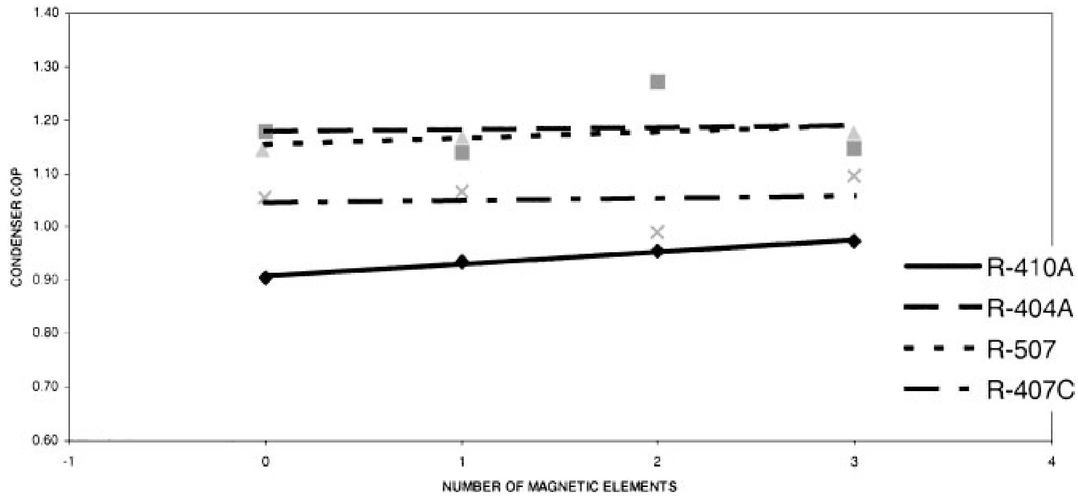

Figure 3. Condenser COP vs number of magnetic elements at constant temperature (0°C).

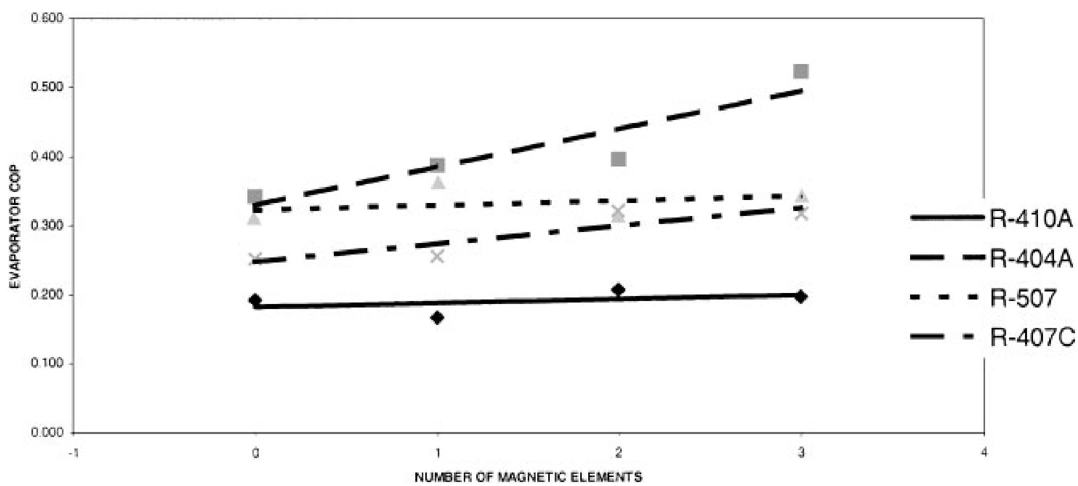

Figure 4. Evaporator COP vs number of magnetic elements at constant temperature (0°C).

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

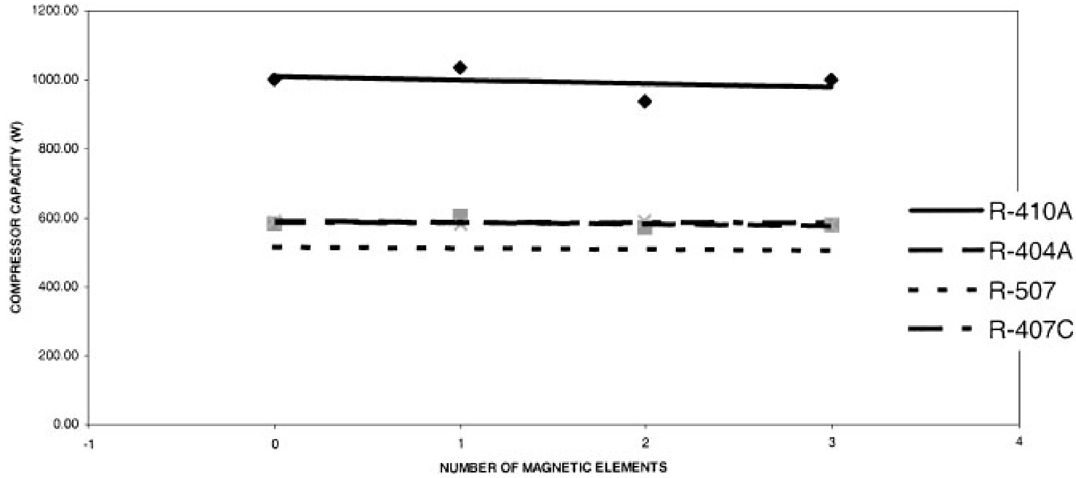

Figure 5. Compressor capacity vs number of magnetic elements at constant temperature (0°C).

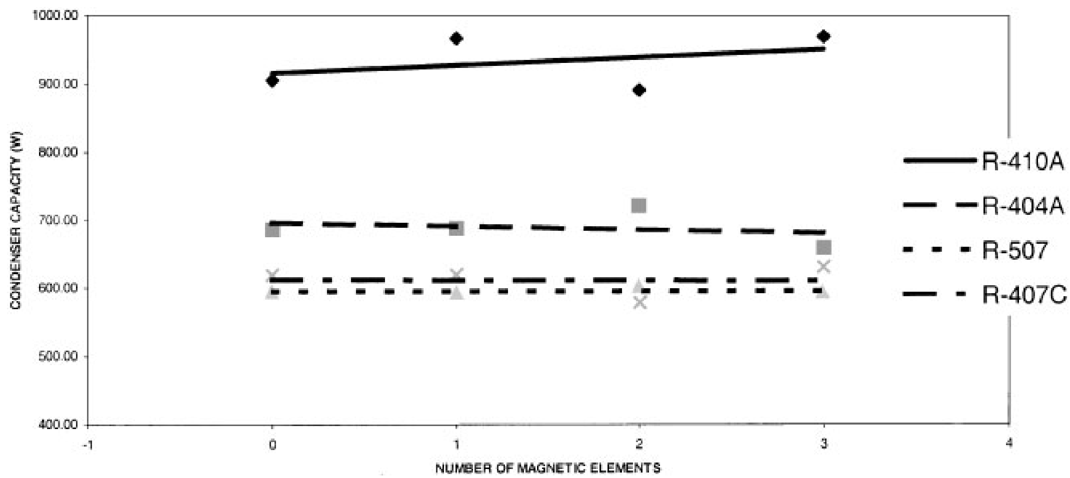

Figure 6. Condenser capacity vs number of magnetic elements at constant temperature (0°C).

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

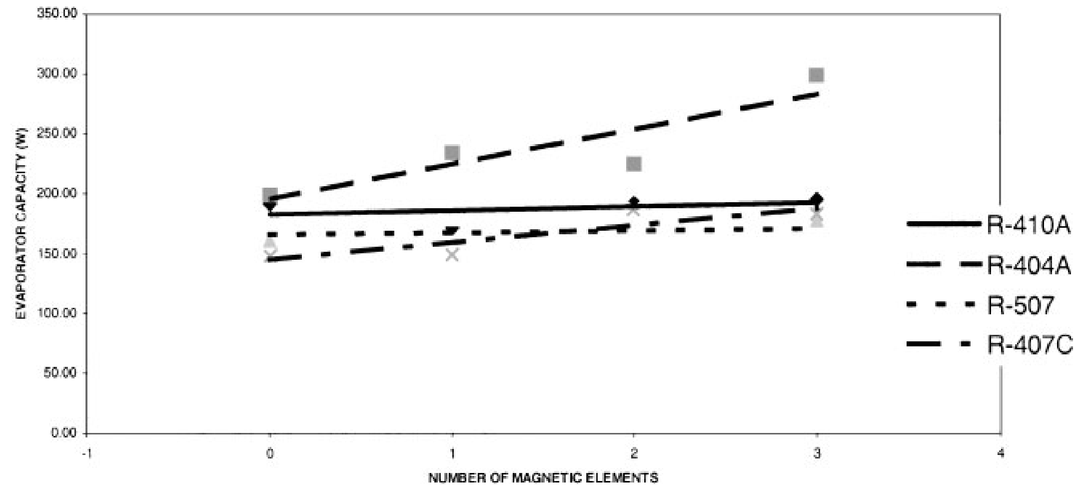

Figure 7. Evaporator capacity vs number of magnetic elements at constant temperature (0°C).

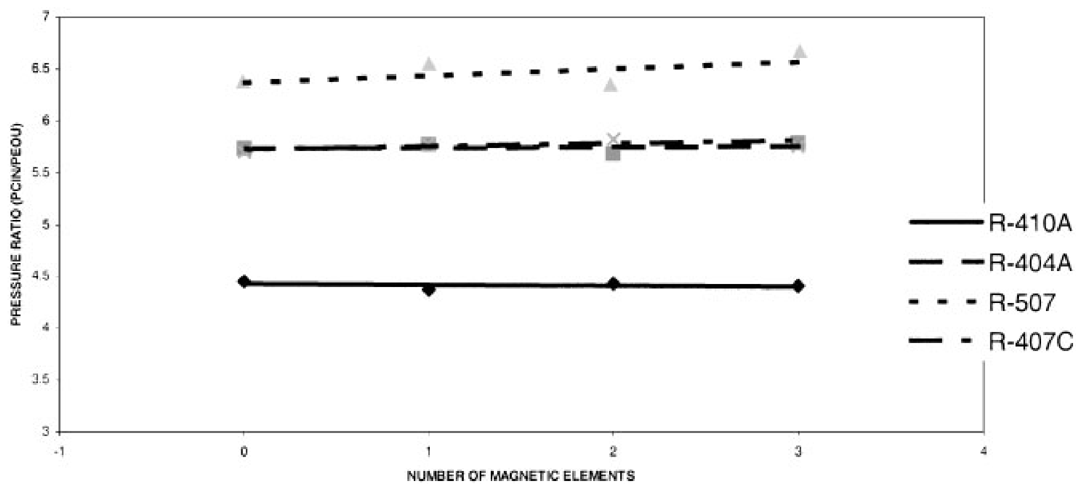

Figure 8. Pressure ratio vs number of magnetic elements at constant temperature (5°C).

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.

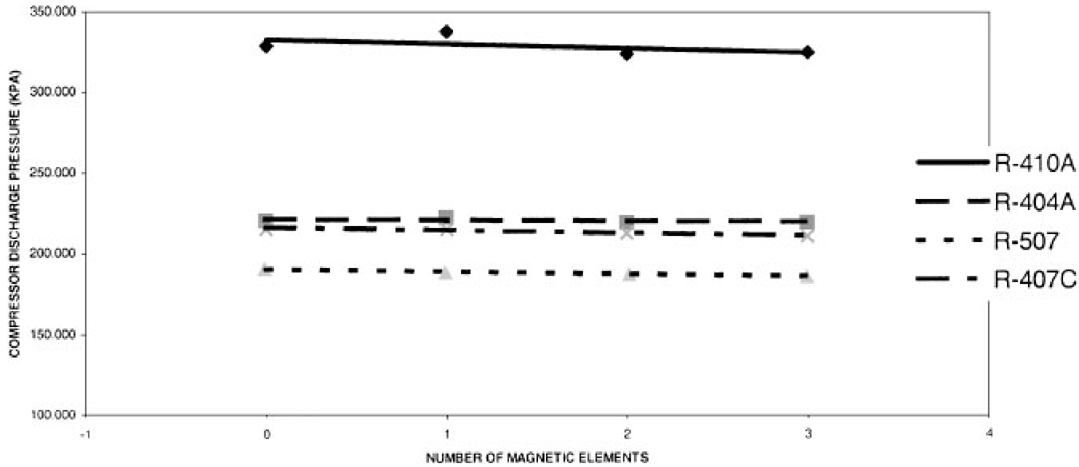

Figure 9. Compressor discharge pressure vs number of magnetic elements at constant temperature (0°C).

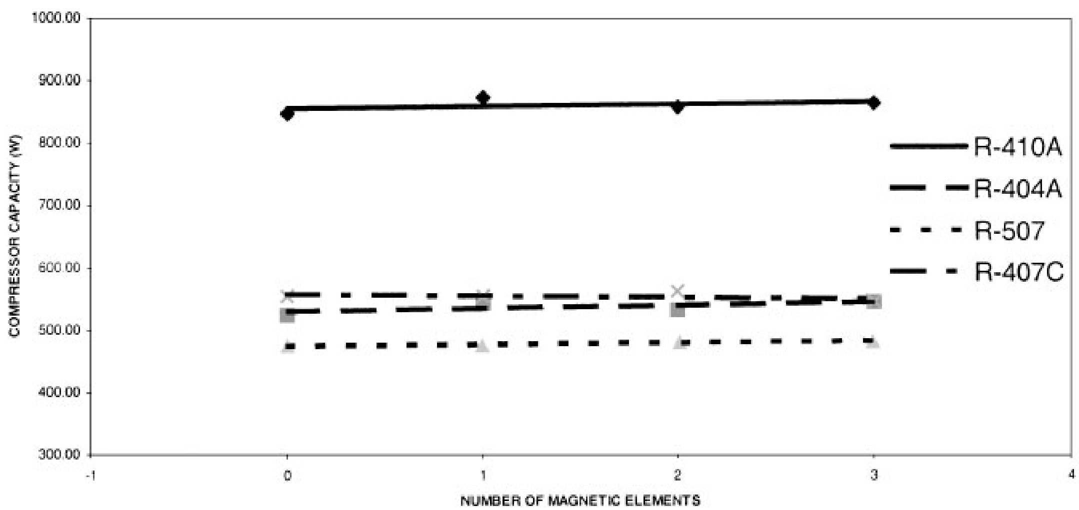

Figure 10. Compressor capacity vs number of magnetic elements of constant temperature (−15°C).

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.

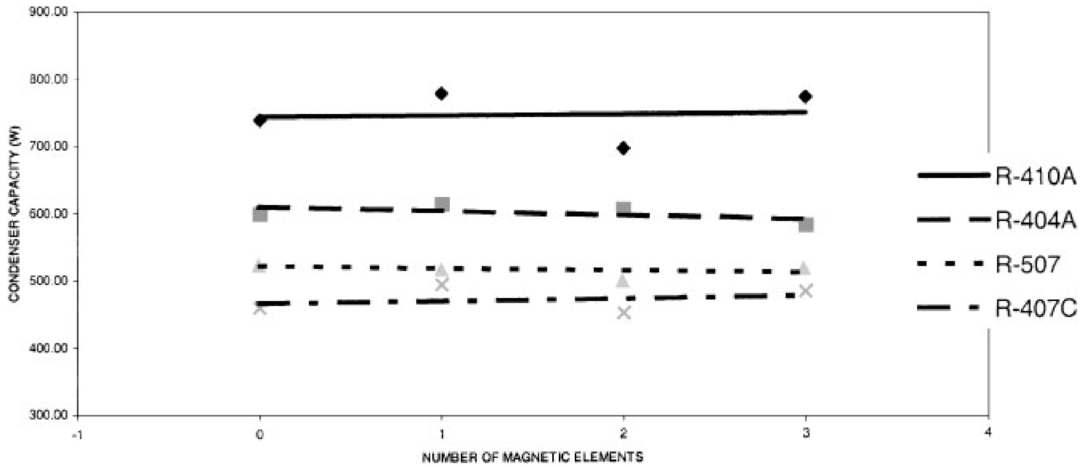

Figure 11. Condenser capacity vs number of magnetic elements of constant temperature (−15°C).

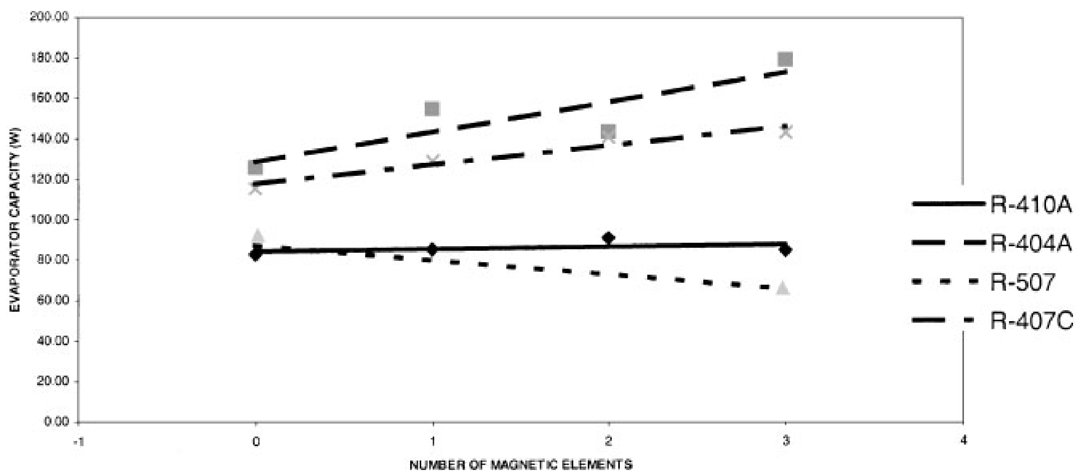

Figure 12. Evaporator capacity vs number of magnetic elements of constant temperature (−10°C).

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.

ACKNOWLEDGEMENTS

The research work presented in this paper was possible through grants from NSERC. The authors wish to acknowledge the continuous support of the University of Moncton.

REFERENCES

Brown GV. 1976. Magnetic heat pumping near room temperature. *Journal of Applied Physics* **47**(8):3673–3680.

Foldeaki M, Chahine R, Bose TK. 1995. Magnetic measurements: A powerful tool in magnetic refrigerator design. *Journal of Applied Physics* **77**(7):3528–3537.

Gschneidddner KA, Pecharsky VK. 1999. Magnetic refrigeration materials. *Journal of Applied Physics* **85**(8):5365–5368.

McLinden MO *et al*. 1998. *NIST Thermodynamic and Transport Properties of Refrigerants Mixtures REFPROP*, Version 6.0. NBS: Gaithersburg, MD.

Muraki M, Sano T, Dong D. 2001. Rheological properties of polyolester under an EHD contact in some refrigerant environments. *Journal of Tribology* **123**(1):54–60.

Sami SM, Tulej PJ, Fang L. 1993. Heat transfer in forced convection boiling of oil–non-azeotropic binary refrigerant mixtures. *International Journal of Energy Research* **17**:903.

Zimm CB, DeGrgoria AJ. 1992. Magnetic refrigeration: Application and enabler for HSTC magnets. *AIP Conference Proceedings*, vol. 273(1), 10 February 1992, 471–480.