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Research Article

Experimental Investigation for the Magnetic-Caloric Effect on the Refrigeration Cycle Performance

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Abstract: The present research aims to investigate experimentally applying of the magnetic-caloric effect on the performance of vapor compression cycle. Three levels of magnetic fields of 1300, 2400, and 3600 Gauss are used and two different locations for the magnetic field; condensate and suction lines are compared. The experimental results are analyzed and verified from thermodynamic point of view. The test results showed that; applying magnetic field at the condensate line or suction line improves the performance of the vapor compression cycle. For magnetic field applying at the condensate line, the coefficient of performance increases by 21.8 %, 29.6 %, and 33.2 % at Gauss levels of 1300, 2400, and 3600 respectively. For magnetic field applying at the suction line, the coefficient of performance increases by 3.6 %, 8.6 %, and 10 % at Gauss levels of 1300, 2400, and 3600 respectively. The magnetic field at the condensate line enhances the coefficient of performance than the suction line.
Keywords: Vapor Compression Cycle Magnetic – Magnetic Field – Magnetic-caloric effect

INTRODUCTION

The most frequently used refrigeration cycle is the vapor compression cycle; the beginning of this refrigeration system can be traced back to Professor Williams Cullen of Glasgow University in 1748 that produced refrigeration by creating partial vacuum over ethyl ether. In 1834, Jacob Perkins proposed a hand operated compressor machine working on ether. This system is commonly used in most household refrigerators as well as in much large commercial and industrial refrigeration system.

One of the current improvements of the vapor compression cycle is based upon the magnetic-caloric effect discovered by Warburg in 1881.The magnetic-caloric effect is the response of a magnetic solid to a changing field which is evident as a change in its temperature. When a magnetic field is applied to a magnetic material, the unpaired spins partially comprising the material's magnetic moment are aligned parallel to the magnetic field. This spin ordering lowers the entropy of the system since disorder has decreased. To compensate for the aligned spins, the atoms of the material begin to vibrate, perhaps in an attempt to randomize the spins and lower the entropy of the system again. In doing so, the temperature of the material increases. Conversely, outside the presence of a file, the spins can return to their more chaotic, higher entropy states, and a decrease in the material temperature is observed. The warming and cooling process can be likened to a standard refrigerator which implements compressing and expanding gases for variations in heat exchange and surrounding temperature. Gadolinium and its alloys are the best material available for magnetic refrigeration near room temperature since they undergo second-order phase transitions which have no magnetic or thermal hysteresis involved. Also, crystalline electric fields and pressure can have a substantial influence on magnetic entropy and adiabatic temperature change.

The effect of the magnetic field is not still considered as a well-known subject. 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.

Brown reported that the magnetic heat pump using ferromagnetic material with a curie point can approach the Carnot cycle efficiency [1].

Samuel and Shawn used magnetic field up to 12000 Gauss placed at the condenser outlet (full liquid line) and studied the effect of magnetic field on the performance of refrigeration cycle using refrigerant mixtures such as R-410A, R-507, R-407C, and R-404A according to the ARI/ASHRAE standards [2]. The results showed that; the magnetic field increases, compressor head pressure and discharge temperature slightly increase as well as less liquid refrigerant is boiling in the compressor shell. Furthermore, usage of magnetic field has a positive influence on the system COP as well as thermal capacities of condenser and evaporator. Samuel Sami used ENERGIZER MHD technology [3], for new alternative refrigerant mixtures under various conditions of magnetic field. The test results demonstrated that as magnetic force increases evaporator performance increases as well as coefficient of performance. Kolandavel and Velappan used magnetic field up to 24000 Gauss [4] placed at the condenser outlet (full liquid line) and reported that; for R290/600a, the magnetic field reduce the compressor energy consumption by 1.5 % to 2.5 %, while the coefficient of performance increased by 1.5 % to 2.4 %.To the authors knowledge none has been reported a comparison between applying the magnetic-caloric effect on the condensate line (liquid line) and suction line (vapor line).

The work presented in this paper is concerned with an experimental study of the influence of the magnetic-caloric effect in the performance of the vapor compression cycle when applying magnetic field on the condensate line and suction line.

EXPERIMENTAL SETUP

Figures 1 and 2 illustrate an image and schematic for the test rig respectively. R12 (CF_2Cl_2) is used as a working fluid in the cycle, the reader has to know that; R12 is not used nowadays as a refrigerant due to its Ozone Depletion Potential (ODP) of 1and Global Warming Potential (GWP) of 8100, and toxic group of 6 for these reasons, the manufacturing was discontinued in 1995as recorded by ASHRAE 2001[5].

The test rig is a conventional vapor compression cycle composed mainly of compressor, condenser, evaporator, oil separator and expansion device. The hermetic reciprocating compressor used is Copeland # JRL4-0050-IAV – 1.55” bore – 0.625” stroke – 1.178 cu.” displacement – 20 fluid OZ (Sun ISO 3GS) oil capacity – ½ Hp motor rating (220 V, 50 Hz, 4.9 A, 3500 rpm) – 233 to 280 µF and 220 Volt AC start capacitor.

The condenser and the evaporator consist of 3/8” diameter, 12” length, and 20 passes copper tubes. Each tube has 0.01” thickness, 14 fins/inch aluminum fins. They equipped with 12” diameter, 27 degree pitch 3 propeller fans. Each fan has 1/30 Hp, shaded poles, motor rating (220 V, 50 Hz, 0.7 A, High-1500 rpm, Medium-1025 rpm, Low-800 rpm).
The metering devices used in this work are ½ TR capillary tube, 2 strainers or ½ TR, FF ½ C, 50 F: -40 F range, 3/8” inlet and ½” outlet diameters thermostatic expansion valve. The filter drier capacity is 5 cu” consists of silica gel and brass/copper.

Fig-1: Image for the Test Rig

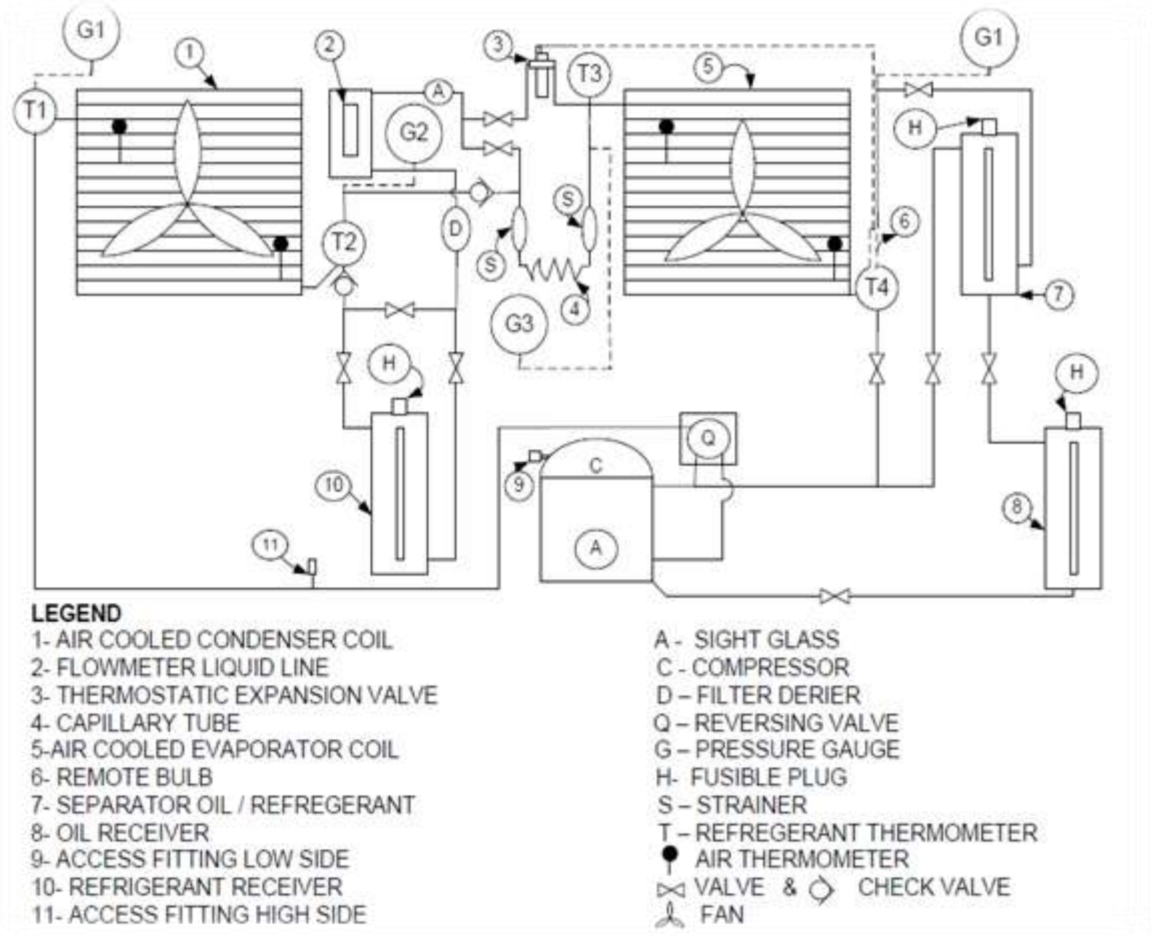

Fig-2: Test Rig

The main objective of this study is to compare the performance of this vapor compression cycle when it is applied to different magnetic field on the condensate line and suction line. To compare the cycle performance the compressor power, inlet and discharge temperatures and pressure ratio, refrigeration effect and the condenser capacity was measured.

Flow meter is used to measure the refrigerantR-12 flow rate with 0.5 to 3 Ib/min operating range and a precision of ±3%.The refrigerant pressure and temperature were measured in different locations in the cycle as shown in image 1 and figure 2. A Bourdon tube pressure gauge of 3-1/2" dial diameter, 0:200 psig scales with 2 % accuracy was used to measure the refrigerant pressure and the temperatures were measured using temperature gauges: BI-metal, 2" dial diameter, 0:200 ^{o}F with 2 % accuracy.

In addition to measure the condenser and the evaporator upstream and downstream cooling air temperature a digital thermometer is used. The digital thermometer type is TPM-30 and has a range of -50 ^{o} C: +70 ^{o} C with a resolution >-20 ^{o}C, 0.1 ^{o}C; ≤ -20 ^{o} C, 1 ^{o}C and an accuracy ± 1 ^{o}C.

The surface temperature of the evaporator and the condenser tubes are measured using an infrared thermometer. The infrared thermometer Type is MIR300 and has a range of -20 ^{o}C: 320 ^{o}C, Resolution 0.1 ^{o}C, 1 ^{o}C, and Accuracy ± 2 ^{o}C.

HVAC analyzer was used to measure the air flow rate, air humidity across both the evaporator and condenser sections with the following specifications: operating temperature 0 ^{o}C:65 ^{o}C, temperature range -5 ^{o}C:65 ^{o}C, humidity range 10: 95% RH, velocity range 0.8:15 m/sec, temperature precision ± 1 ^{o}C, humidity precision± 3% and velocity precision± 3% .

The compressor power consumption was measured by measuring both the input current and the input voltage to the compressor motor. Digital clamp meter was used to measure the electric current drawn by the compressor with the following specifications: Type KYORITSU, measuring range 0:399.9 A with an accuracy ±1.5% rdg ± 4dgt (50/60Hz) and measuring range of 0:599.9 A with an accuracy ±2% rdg ± 5dgt (40~400Hz).

The magnetic field in each identical pairs of magnets was measured by F.W. Bell Model 5080

Gauss/Tesla Meter, division of Bell Technology, a Sypris Company, USA [8].

The test objective is divided into two sections, depending on the location where the magnets are clamped. Firstly the magnets are placed on the condensate line (liquid line); secondly the magnets are places on the suction line (vapor line). The exit air velocity over the outer face area of the evaporator and the condenser as well as the inlet and exit air temperatures are measured at 9 distributed locations while the average value is taken. As the evaporator and condenser fan's speed is kept constant during all experiments, the air flow rate is kept constant at 797 and 851 cfm respectively.

The readings were taken after the system had reached the steady state conditions. The absolute error in refrigerating capacity, compressor energy and coefficient of performance estimated by the single sample analysis represented in ASHRAE guideline 2 [6].

RESULTS AND DISCUSSION

The surface temperature of the evaporator is measured and it was found to be greater than the cooled air dew point temperature. The air temperatures across the evaporator and the condenser are measured, and the refrigeration capacity based on the sensible cooling process is calculated from Equation (1);

$$\dot{Q}_e = \dot{m}_{ae} \int_{Tae,i}^{Tae,o} Cp_a dT \qquad (1)$$

The consumed power by the compressor is measured by its drawn electrical current, the applied voltage. While the coefficient of performance is calculated from Equation (2);

$$COP = \frac{\dot{Q}_e}{\dot{P}_c} \qquad (2)$$

The pressures at the suction and the discharge of the compressor are measured, and the operating pressure ratio is calculated from Equation (3);

$$PR = \frac{P_d}{P_s} \qquad (3)$$

Three different categories of permanent magnets are used in this research; the first, second and third category are consists of three pairs of permanent magnets have a dimensions of (18x3.2x3.2 mm), (32x3.2x1.5 mm) and (58x11x5 mm) respectively. The magnetic field within each pair is measured and it was found to be 1300, 2400 and 3600 Gauss respectively.

It should be highlighted that, each test result represents the average for 5 consecutive measurements for each parameter after 30 minutes interval for cycle stabilization.

The results of the vapor compression cycle's performance without applying a magnetic field are used as a baseline data and declared through the figures of the results at zero number of magnetic pairs. Upon the completion of the baseline results under aforementioned conditions, the first category of magnets is used as follows; one pair of magnets (1300 Gauss) is clamped over the condensate line(1/4indiameter) and far0.45 m from the condenser outlet. This location is chosen to be insuring that the liquid observed through the sight glass in the condenser outlet line and the measurements are taken to account the coefficient of performance and the pressure ratio. After that, additional pair of the same magnets is clamped over the condensate line by 1 cm apart from the first pair, and the same procedure is followed. Finally the third pair is clamped over the condensate line by 1 cm apart from the second pair and the same procedure is repeated. The second and third categories of magnets are used with the same procedures as the first category.

In order to study the influence of the number of magnets and the magnetic field on the performance of the vapor compression cycle when applied on the condensate line, the test results were plotted at various conditions in Figures 3 and 4.

Figure 3 illustrates the coefficient of performance versus the number of pairs of magnets for the different three magnetic fields 1300 Gauss, 2400 Gauss, and 3600 Gauss. The results showed that; for magnetic field of 1300 Gauss, increasing the number of magnetic pairs to 3 increases the coefficient of performance by 21.8 % referred to the case without magnetic field. This is because the energy consumed by the compressor of the vapor compression cycle is often limited by incomplete or insufficient evaporation and condensation of the refrigerant. The compressor consumes additional power to evaporate the liquid refrigerant that enters the compressor when the evaporation of refrigerant is incomplete. The magnetic field is applied to the working fluid in the liquid phase disrupt intermolecular forces in the working fluid and enhance vaporization of the fluid. This reduces the amount of residual liquid that is boiled in the compressor shell, lowering the power consumption of the compressor and improving the performance of the system as reported by Kolandavelet. Al[4]. For example Samuel et. al [2] reported that the compressor power consumed decreased by 8 % by applying the magnetic field.

In addition for magnetic field of 2400 Gauss, increasing the number of magnetic pairs to 3 increases the coefficient of performance by 29.6 % referred to the case without magnetic field while for magnetic field of

3600 Gauss the coefficient of performance increased by 33.2 % referred to the case without magnetic field. It is observed that the magnetic field enhances the refrigeration coefficient of performance and the enhancement value increased as the Gauss levels increased.

Figure 4 illustrates the compressor pressure ratio versus the number of pairs of magnets for the different three magnetic fields 1300 Gauss, 2400 Gauss, and 3600 Gauss. The results showed that for magnetic field of 1300 Gauss, increasing the number of magnetic pairs to 3 increases the pressure ratio by 6.5 % referred to the case without magnetic field. This value increases by 6.8 % and 7.7% referred to the case without magnetic field for magnetic field of 2400 and 3600 Gauss respectively. The increase in the compressor pressure ratio is expected since less liquid refrigerant is being boiled in the compressor shell.

It appears from the previous results that; the magnetic field at the condensate line increases the refrigeration effect and slightly increases the compressor power, while the combined effect is increasing the cycle coefficient of performance. Previous studies explain this result as shown that oil entertained in the refrigerant flow results in degrading the heat transfer rates. The magnetic field changes the polarity of hydrocarbon oil from negative to positive charges. This polarization results in entertaining the oil and being carried away from the evaporator surface, and enhancing the heat transfer. Thus, the consequent effect on the coefficient of performance has been caused by this phenomenon. It was also shown that higher Gauss levels increase thermal capacities which is coincide with Sami et al. [7].

To the author knowledge, no available explanation based on a mathematical thermodynamic relations is found in the literature, the explanation needs more detailed study on the thermodynamics of the refrigeration cycle by aid of the measured pressure and temperature at each state in the cycle. Then, the variation in the coefficient of performance is calculated from Equation (4);

$$\Delta\,COP = \frac{COP^{\prime}}{COP} - 1 \qquad (4)$$

After substitution and mathematical manipulation, the variation in the coefficient of performance is simplified to Equation (5);

$$\Delta\,COP = \frac{\left(\dot{Q}_e + \Delta\,\dot{Q}_e\right)\dot{P}_c}{\left(\dot{P}_c + \Delta\,\dot{P}c\right)\,\dot{Q}_e} - 1 \qquad (5)$$

The condition of positive variation or increase in the coefficient of performance is declared as in Equation (6);

$$\frac{\left(\dot{Q}_e + \Delta\,\dot{Q}_e\right)\dot{P}_c}{\left(\dot{P}_c + \Delta\,\dot{P}c\right)\,\dot{Q}_e} \succ 1 \qquad (6)$$

Applying the first law of thermodynamics for steady-state, steady-flow open system for the vapor compression cycle; the previous condition is re-formulated to be as in Equation (7);

$$\left(h_3^{\prime}\right) \succ \left(h_2^{\prime} + 2h_3 - 2h_1\right) \qquad (7)$$

To verify the experimental results from thermodynamics point of view, it should be noted that; from the measured pressure and temperature at each state in the test rig, one can verify the mathematical thermodynamically constrain. The verification is performed for the case of using 3 pairs of magnetic with the different studied Gauss levels. Table (1) illustrates the pressure, temperature, and corresponding enthalpy.

Table-1: Thermodynamic Properties for some states in the refrigeration cycle (Magnetic Field Applied over the Condensate Line)

Gauss Level	h_1	h_2	$h_2^{\prime}$	h_3	$\left(h_2^{\prime} + 2h_3 - 2h_1\right)$	$h_3^{\prime}$	Verification
1300	359.9	378.4	383.4	181.4	26.4	175.9	Verified
2400	359.9	378.4	386.6	181.4	29.6	173.1	Verified
3600	359.9	378.4	387.1	181.4	30.1	171.2	Verified

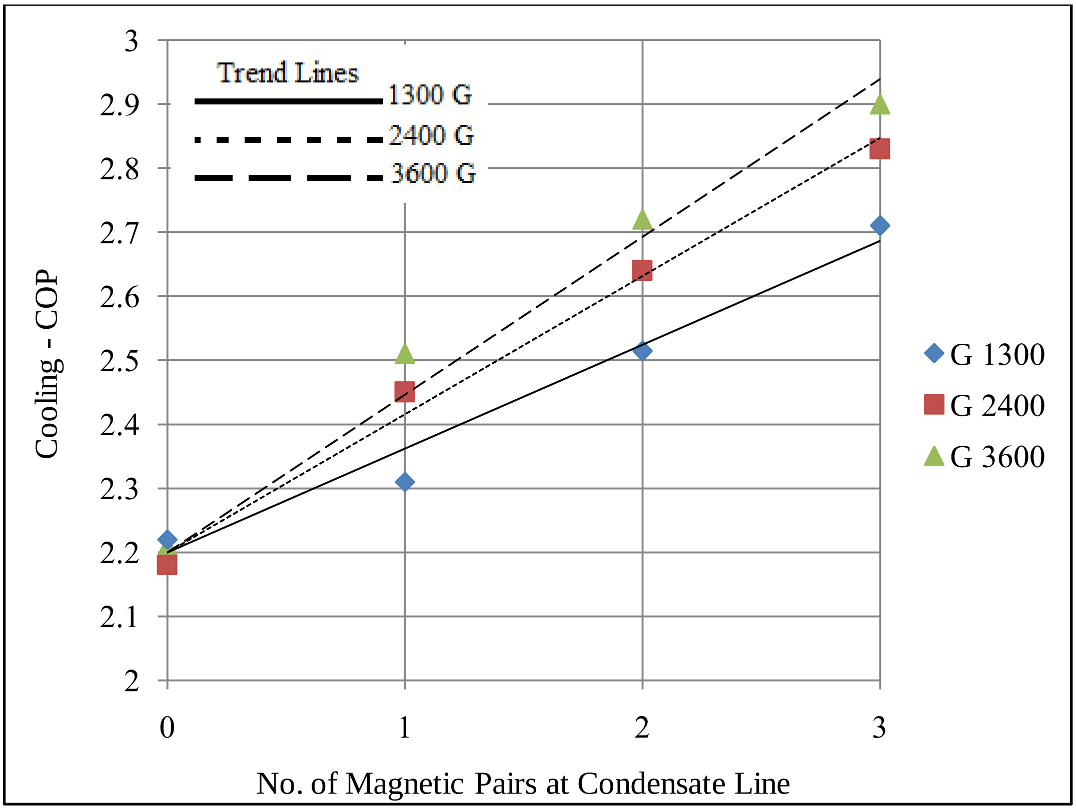

Fig-3: Coefficient of Performance for Various Numbers of Magnetic Pairs at Condensate Line

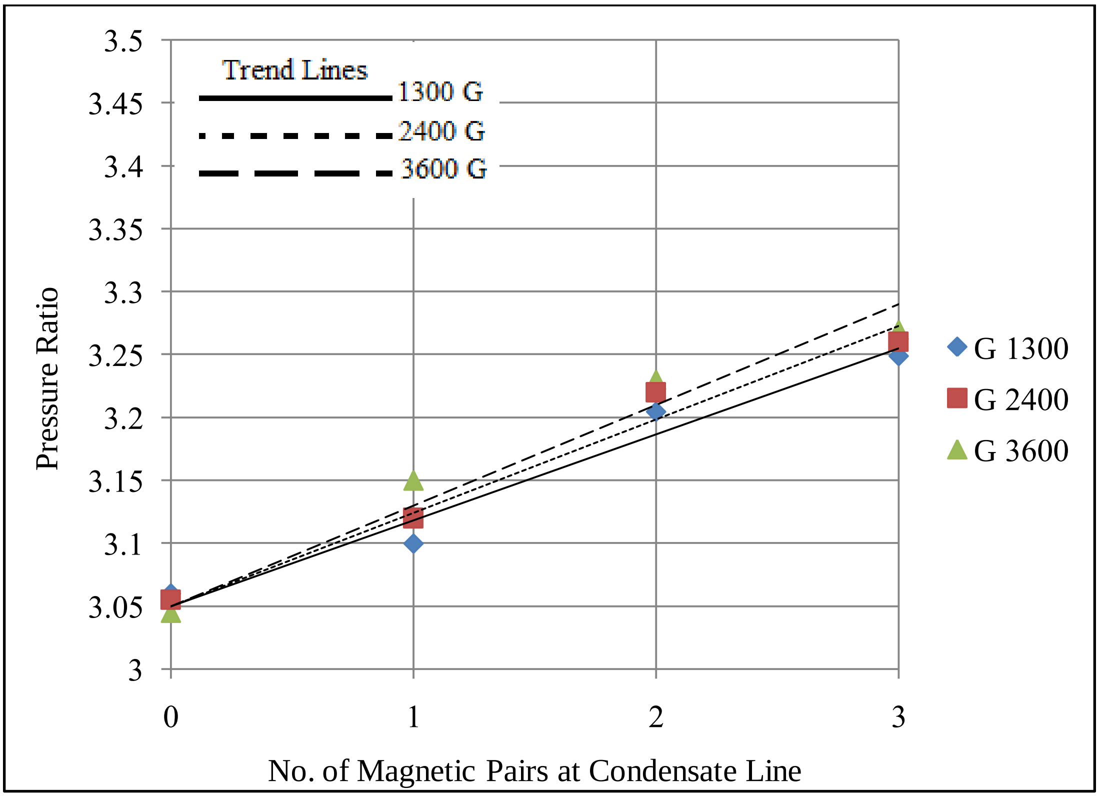

Fig-4: Compressor Pressure Ratio for Various Numbers of Magnetic Pairs at Condensate Line

The new vision for the present study is to investigate the effect of applying the magnetic field on the compressor suction line also. For the same base line data obtained previously; the same magnetic pairs in the same procedure and spacing are applied over the compressor suction line at a distance of 0.35 m from the evaporator outlet, samples of the test results were plotted at various conditions in Figures 5 and 6.

Figure 5 illustrates the coefficient of performance versus the number of pairs of magnets for the different three magnetic fields 1300 Gauss, 2400 Gauss, and 3600 Gauss. The results showed that the coefficient of performance increased by 3.6 %, 8.6% and 10% for magnetic field of 1300, 2400 and 3600 Gauss respectively referred to the case without magnetic field. It appears as discussed before that the refrigeration effect increases as a result of magnetic field, also higher Gauss levels results in increasing the refrigeration effect.

On the other hand figure 6 illustrates the compressor pressure ratio versus the number of pairs of magnets for the different three magnetic fields 1300 Gauss, 2400 Gauss, and 3600 Gauss. The results showed that the pressure ratio has been slightly increased with the increases in the number of magnetic elements. For magnetic field of 1300 Gauss, increasing the number of magnetic pairs to 3 pairs increases the pressure ratio by 2.6 % referred to the case without magnetic field and this value increased by 3.2 % and 3.5 % referred to the case without magnetic field for magnetic field of 2400 and 3600 Gauss respectively. The increase in the compressor pressure ratio is expected since refrigerant is being overheated due to magnetization before the compressor entrance.

It appears from the previous results that; the magnetic field at the suction line increases the refrigeration effect and slightly increases the compressor power, while the combined effect is increasing the coefficient of performance.

On the same way the verification is performed for the case of using 3 pairs of magnetic with the different studied Gauss field. Table (2) illustrates the pressure, temperature, and corresponding enthalpy.

Table-2:Thermodynamic Properties for some states in the refrigeration cycle (Magnetic Field Applied over the Suction Line)

Gauss Level	h_1	h_2	$h_2^{\backslash}$	h_3	$\left(h_2^{\backslash} + 2h_3 - 2h_1\right)$	$h_3^{\backslash}$	Verification
1300	359.9	378.4	386.1	181.4	29.1	309	Verified
2400	359.9	378.4	385.7	181.4	28.7	310	Verified
3600	359.9	378.4	385.6	181.4	28.6	312	Verified

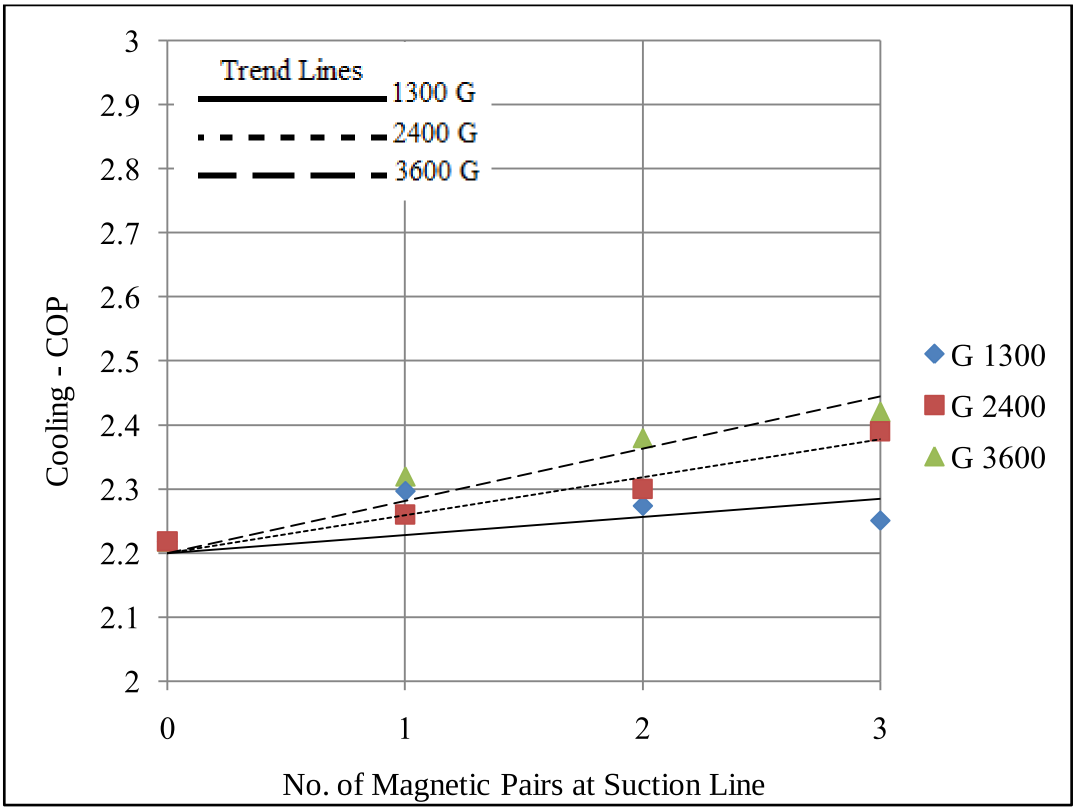

Fig-5: Coefficient of Performance for Various Numbers of Magnetic Pairs at Suction Line

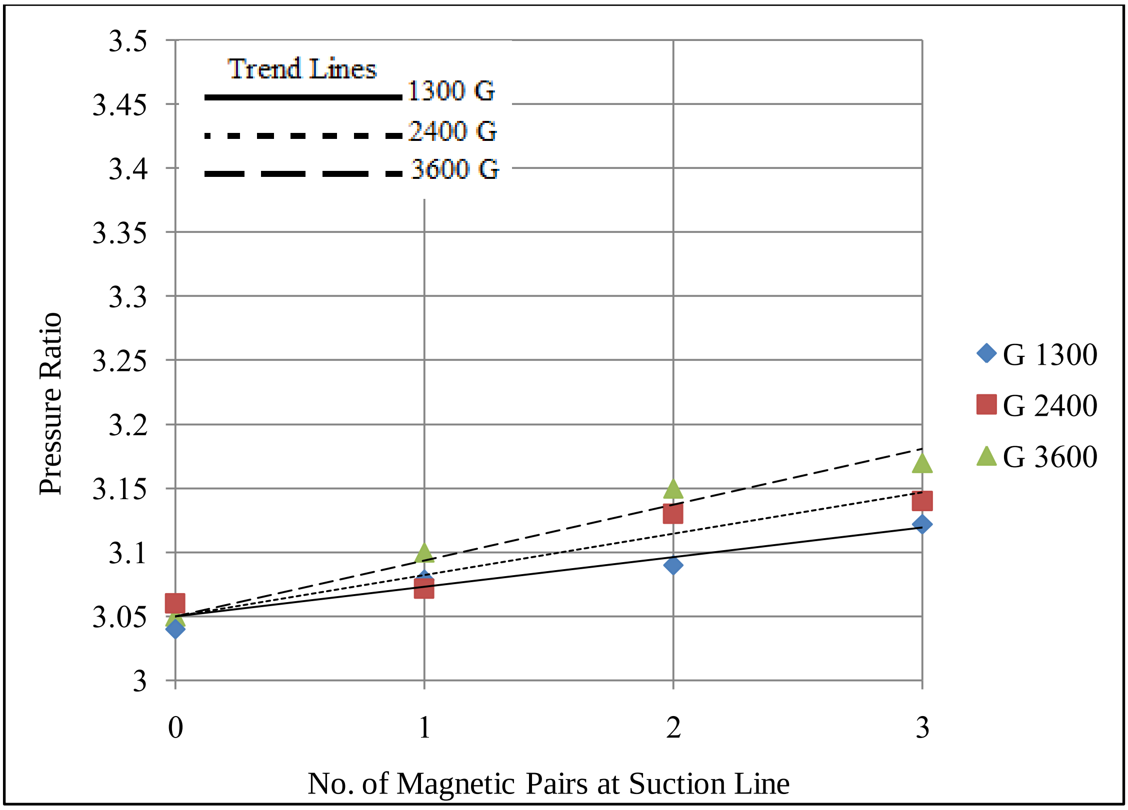

Fig-6: Compressor Pressure Ratio for Various Numbers of Magnetic Pairs at Suction Line

Based on the above results it appears that; for magnetic field of 1300 Gauss in the liquid line increasing the number of magnetic pairs to 3 increase the coefficient of performance by 21.8 % while this value is 3.6 % for the magnetic field in suction line. This means that the magnetic field at the liquid line accelerates the increase of the coefficient of performance than the suction line.

CONCLUSION

It is concluded from the present study;

- Applying magnetic field at the condensate line or suction line improve the performance of the vapor compression cycle.
- For magnetic field applying at the condensate line, the coefficient of performance increases by 21.8 %, 29.6 %, and 33.2 % at Gauss levels of 1300, 2400, and 3600 respectively.
- For magnetic field applying at the suction line, the coefficient of performance increases by 3.6 %, 8.6 %, and 10 % at Gauss levels of 1300, 2400, and 3600 respectively.
- The magnetic field at the condensate line accelerates the increase of the coefficient of performance than the suction line.

NOMENCLATURE

Symbol	Definition	Units
$\dot{Q}_e$	Rate of heat transfer absorbed by the evaporator	W
$\dot{m}_{ae}$	Mass flow rate of air flowing through the evaporator	kg/s
Cp_a	Constant pressure specific heat of air	J/kg.K
Tae, o	Temperature of air outflow the evaporator	K
Tae,i	Temperature of air inflow the evaporator	K
$\dot{P}_c$	Power consumed by the compressor	W
I	Electric current drawn by the compressor	A
V	Voltage supply to the compressor	V
$\cos\theta$	Electric power factor	
COP	Coefficient of performance	
PR	Compressor pressure ratio	

P_d	Compressor discharge pressure	Pa
P_s	Compressor suction pressure	Pa
ΔCOP	Variation in the coefficient of performance	
$COP^{\backslash}$	Coefficient of performance of the refrigeration cycle in case of applying magnetic field	
$\Delta \dot{Q}_e$	Variation in the evaporator capacity due to magnetic field	W
$\Delta \dot{P}c$	Variation in the compressor consumed power due to magnetic field	
h_1	Enthalpy of refrigerant at evaporator outlet	J/kg
h_3	Enthalpy of refrigerant at condenser outlet	J/kg
$h_2^{\backslash}$	Enthalpy of refrigerant at compressor outlet	J/kg
$h_3^{\backslash}$	Enthalpy of refrigerant at condenser outlet	J/kg

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