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EVALUATION OF AN EXPERIMENTAL REFRIGERATOR USING R404A BY REGULATING THE COMPRESSOR SPEED

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ABSTRACT

This study outlines the experimental performance of a refrigeration system working with reciprocating compressor by regulating its speed by means of a frequency inverter using an alternative refrigerant, R404a. Regulating the compressor speed is one of the best methods to vary the capacity of refrigeration system. For this aim, an experimental traditional refrigeration system was built up and a frequency inverter driver mounted on compressor. The experiments were made for determining the system performance also including compressor performance when compared to traditional system that is working at a nominal frequency of 50 Hz. The results showed that using variable speed compressor in a refrigeration system affects system performance in terms of thermodynamic aspects.

INTRODUCTION

The vapour compression refrigeration systems, though designed to satisfy the maximum load, work at part-load for much of their life generally regulated by on-off cycles of the compressor, working at the nominal frequency of 50 Hz, imposed by a thermostatic control which determines high energy consumption. Moreover, the inefficient use of electricity to supply the refrigeration and air-conditioning compressors is considered as an indirect contribution to the greenhouse gases emitted in the atmosphere; these emissions can be reduced by improving the energy conversion efficiency of the above mentioned systems [1]. Because the refrigeration systems are vastly used in a wide variety of commercial activities, ranging from temperature control of installations for human comfort to strict storage conditions for perishable food products, any efficiency improvement of refrigeration systems would represent a significant energy economy [2]. Theoretically the most efficient method of refrigeration capacity control is the variable speed compressor which continuously matches the compressor refrigeration capacity to the load [3]. This method of refrigeration capacity control, which consists in regulating the compressor speed to continuously match the compressor refrigeration capacity to the load, has been analyzed during the last years [2-14]. Most of these studies are concerning about CFC of HCFC types of refrigerants.

NOMENCLATURE

D	[m]	Outer diameter
h	[kJ/kg]	Specific enthalpy
$\dot{m}$	[kg/s]	Mass flow rate
P	[Bar]	Pressure
Q	[kW]	Heat transfer rate
T	[°C]	Temperature
n	[-]	Revolution per minute
N	[-]	Number of pistons in compressor
S		Stroke
V_H	[m^3/h]	Volumetric flow rate
W_C	[kW]	Compressor capacity

Special characters

f	[Hz]	Frequency
η	[-]	Efficiency

Subscripts

a	Air
A	Actual
C	Condenser
E	Evaporator
EL	Electricity
is	Isentropic
me	Mechanical
R	Refrigerant
SYS	System
w	Water
1,2,..	Reference conditions

In this study, the performance of an experimental refrigeration system is investigated by regulating the compressor speed. An experimental traditional refrigeration system was designed and adapted with a frequency inverter driver mounted on compressor. The system is designed for alternative refrigerant R404a (44% R125, 52% R143a, 4% R134a). R404a is a hydrofluorocarbon (HFC) that is a non-ozone depleting compound designed to serve as a long-term alternative to R502 and R22 in low and medium temperature commercial refrigeration applications. The experiments were made for different compressor frequencies for investigating the effects of compressor speed on system performance.

EXPERIMENTAL SETUP AND PROCEDURE

Vapour compression experimental system consists of a semi hermetic compressor driven by frequency inverter for regulating compressor speed, an air cooled evaporator, an air cooled condenser and an externally equalized thermostatic expansion valve. Semi hermetic compressor, as declared by the manufacturer, can work with R22, R134a R404a, R407c and R507a and is lubricated with polyester oil that is suitable for R404a. Evaporator is located in a specially designed cold room that is heated by means of electrical heaters for simulating refrigeration load. An isolated air channel is built for mounting condenser in it so it is not affected by outside conditions. To fix the air temperature constant that is entering the condenser channel, electrical heaters are located at the entrance of the channel. Additionally some auxiliary equipments are used; an oil separator located after the compressor, a liquid receiver located after the condenser, suction line accumulator located after evaporator.

Cold room is 2.1 m in height, 1.2 m in width and 1.35 m in length. It is also isolated with styrofoam and having a volume of 2.87 m3. For simulating the refrigeration load, 18 flat electrical heaters are used for homogeny heat distribution. Each heater is 0.045 kW in power and the power of heaters is controlled by means of a variable transformer and the consumed power monitored with the help of a wattmeter.

The regulation of compressor speed is controlled with a frequency inverter mounted on semi hermetic compressor. The device is 0.75 kW and the efficiency of the device is 95.1 % according to the manufacturer’s catalogs. The frequency of compressor’s electric motor can be adjusted very easily by means of a small button between 0-50 Hz. All electrical parts of the refrigeration system are placed into a control panel (Figure 1).

For determining the effects of compressor speed on system performance, the measurements of temperature and pressure were made from specified points of the experimental system. The pressure sensors are specially designed types for refrigerants. Measuring range of pressure sensors is between 0 - 30 bar. Refrigerant mass flow rate is measured using a flow meter that is designed for refrigerant R404a and mounted between condenser and expansion valve. This is because the flow meter is designed for measuring liquid refrigerant mass flow rate. Also air humidity at the inlet and outlet of the condenser channel were measured using compact humidity measuring instrument. Temperature measurements were collected from 12 points of the system, pressure measurements were collected from 7 points and refrigerant mass flow rate measured after the condenser. All the measurement devices are connected to a data logger which has 20 channels for collecting the data. Also the data logger was connected to a computer. The specifications of measuring devices are given in Table 1.

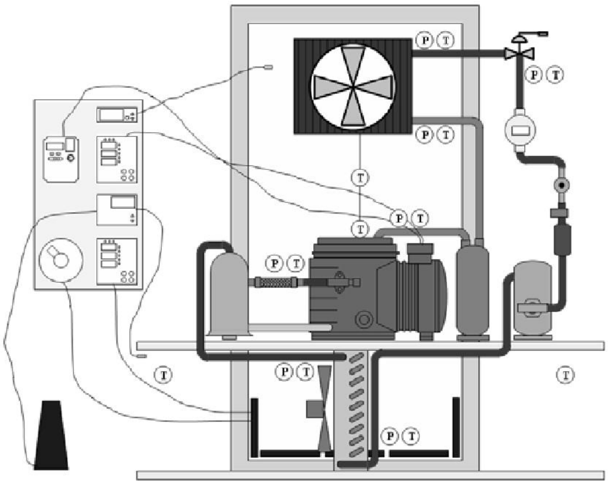

Figure 1 Sketch of experimental setup and measuring points

Table 1 Specifications of measuring devices

Measurement Device	Range	Accuracy
Thermocouple (K type	-180 °C / 1350 °C	± 1.5 °C
Pressure sensor	0 / 30 Bar	± 0.5 %
Mass flow meter	0.05 / 2.5 kg/s	± 3 %
Wattmeter	0 / 1.5 kW	± 0.3 %

Experiments were made up for different compressor electrical motor frequencies. For the different compressor speeds, the adjusted inverter values was 35 Hz, 40 Hz, 45 Hz and 50 Hz. Minimum frequency range was selected to be 35 Hz for avoiding problems for the compressor lubricating by splash. Additionally, compressor vibrations increase at lower frequencies. At each adjusted frequency, cold room refrigeration duty simulated by electrical heaters. Experimental setup was operated for each adjusted frequency and cooling loads. All measurements were made for every 5 seconds and the data was collected to computer by means of data logger.

NUMERICAL METHOD

For the mathematical modeling of refrigeration system thermodynamic balance equations are written. All the equations were written according to reference points of the system given in Figure 2.

For the compressor, energy balance equation can be written as:

$$\dot{m}_R\, h_1 + W_C = \dot{m}_R\, h_2 \qquad (1)$$

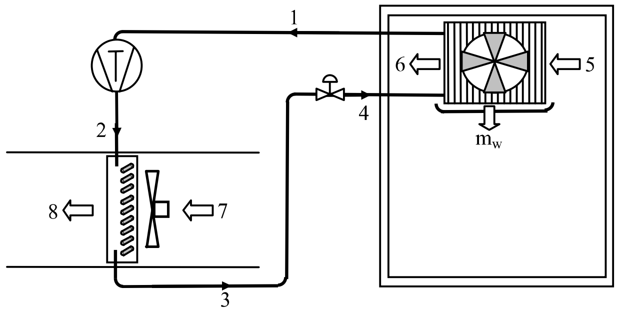

Figure 2 Schematic diagram of experimental setup

From Equation (1), compressor capacity can be determined as:

$$W_C = \dot{m}_R \left(h_2 - h_1\right) \tag{2}$$

It must be noted that capacity in Equation (2) is the actual compressor capacity of experimental system. For determining the isentropic compressor capacity the equation below can be used [15]:

$$W_{C,is} = \frac{m_R\left(h_{2'} - h_1\right)}{\eta_{is}} \tag{3}$$

where η_{is} is the isentropic compressor efficiency and $h_{2'}$ is the enthalpy after isentropic compression process. For the condenser energy balance is given below:

$$\dot{m}_R h_2 + \dot{m}_a h_7 = \dot{m}_R h_3 + \dot{m}_a h_8 \tag{4}$$

For the condenser capacity the equation below is used.

$$Q_C = m_R\left(h_3 - h_2\right) \tag{5}$$

The evaporator energy balance can be defined as:

$$\dot{m}_R h_4 + \dot{m}_a h_5 = \dot{m}_R h_1 + \dot{m}_a h_{H6} + \dot{m}_w h_w \tag{6}$$

Evaporator (refrigeration) capacity is given below.

$$Q_E = m_R\left(h_1 - h_4\right) \tag{7}$$

The coefficient of performance (COP) of a refrigeration system is defined as [16]

$$COP = \frac{Q_E}{W_C} \tag{8}$$

In above equation, W_C is the theoretical compressor capacity of the experimental system. So this equation can be thought as theoretical COP of the system. In many cases this equation demonstrates the system performance. In fact in the definition of COP, the consumed energy must be the electric consumption of the compressor. Thus, the actual COP of the refrigeration can be written as:

$$COP_A = \frac{Q_E}{W_{C,EL}} \tag{9}$$

where $W_{C,EL}$ is the electric consumption of the compressor. For the COP of the whole system, the energy consumed of condenser and evaporator fans must be included. In this case, the COP of the whole system is given below [17, 18]:

$$COP_{SYS} = \frac{Q_E}{W_{C,EL} + W_{FAN,E} + W_{FAN,C}} \tag{10}$$

Volumetric flow rate of compressor is defined as [19]:

$$V_H = \frac{\pi D^2}{4}\frac{n}{60} S\ N \tag{11}$$

Finally the mechanical efficiency of compressor can be expressed using equation below [20, 21]:

$$W_{C,EL} = \frac{W_C}{\eta_{me}} \tag{12}$$

TRENDS AND RESULTS

Performance analysis in terms of thermodynamic aspects of experimental refrigeration system by regulating its speed has been carried out. In the experiments an alternative refrigerant R404A was used. In Figure 3, the variations of suction and discharge pressures with compressor frequency are given. By increasing the compressor frequency, discharge pressure increases and suction pressure decreases. The increase percent of discharge pressure is a bit higher than decreases percent of suction pressure. So the pressure ratio increases with compressor frequency (Figure 4). Generally when the speed compressor decreases the plant performances improve because of the compression ratio decrease and the compressor global efficiency improvement [22]. Also with the increase of compressor frequency discharge temperature increases as parallel to discharge pressure. One of the reasons of this is that the superheating temperature increases with frequency. Suction temperature shows the same trend with the increase of compressor frequency, but the increase ratio is smaller. Additionally subcooling temperature increases a bit with frequency (Figures 5 and 6).

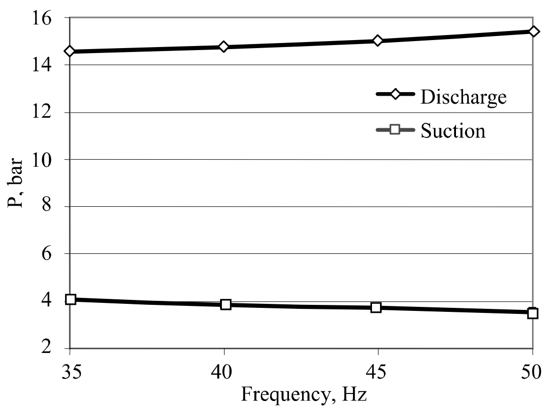

Figure 3 Variation of discharge and suction pressures with frequency

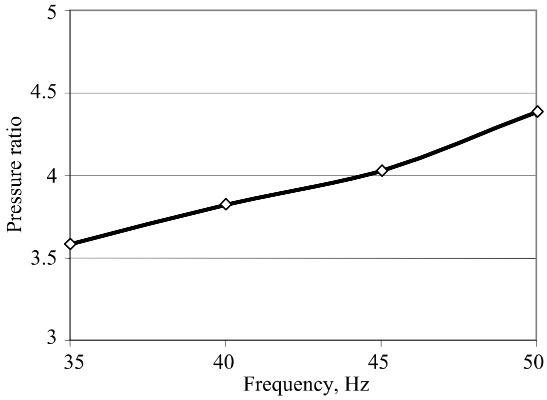

Figure 4 Variation of pressure ratio with frequency

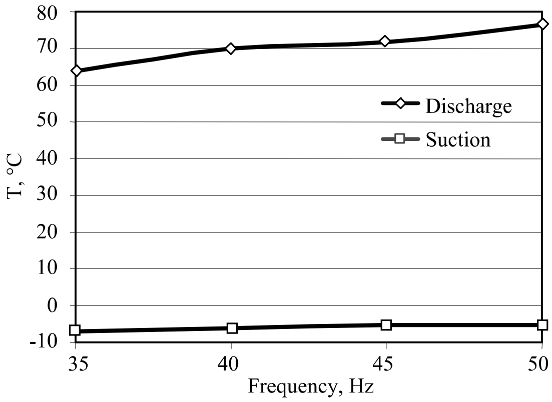

Figure 5 Variation of discharge and suction temperatures with frequency

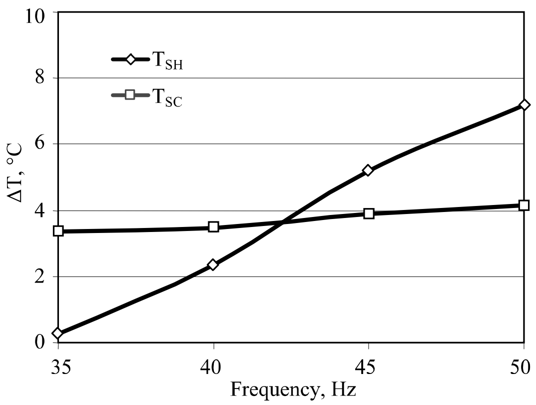

Figure 6 Variation of subcooling and superheating temperatures with frequency

With rise of compressor speed by means of frequency, mass flow rate also increases. So also the volumetric flow rate (Figures 7 and 8).

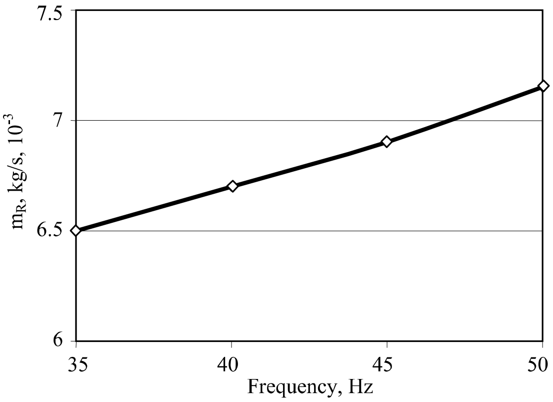

Figure 7 Variation of mass flow rate with frequency

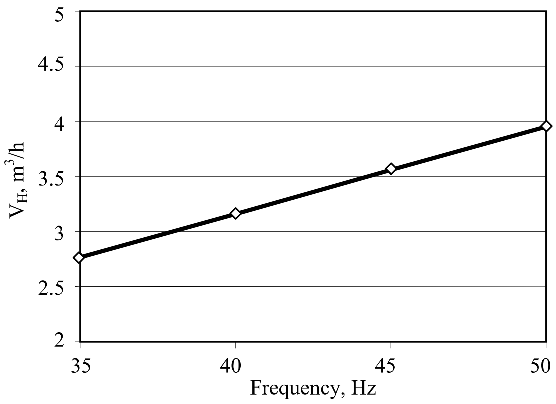

Figure 8 Variation of volumetric flow rate with frequency

In Figure 9, isentropic efficiency of compressor variation with frequency is given. To evaluate the isentropic efficiency Equation (3) is used. The isentropic efficiency increases when the speed decreases because the discharge superheat reduces on diminishing the compressor speed. Rising of the mechanical efficiency at low frequency rates increases in low frequency rates due to the smaller effect of the friction [1] (Figure 10).

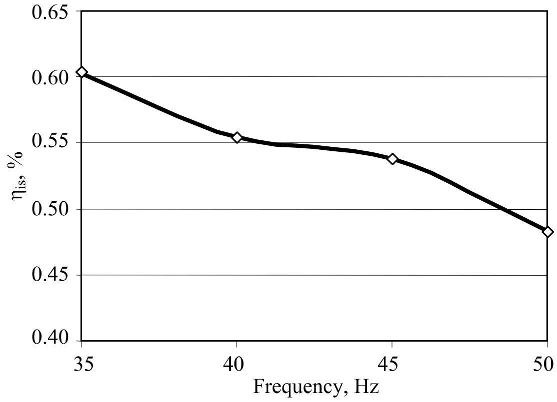

Figure 9 Variation of isentropic efficiency with frequency

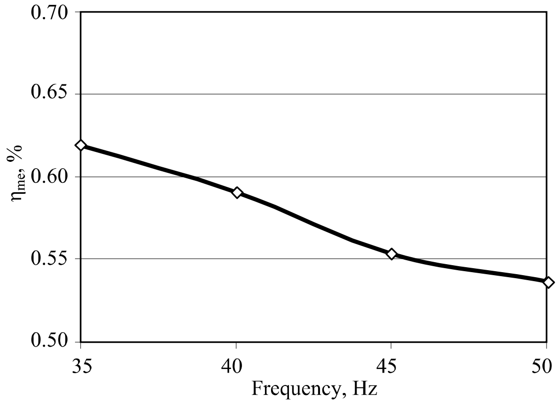

Figure 10 Variation of mechanical efficiency with frequency

In Figure 11, COP versus the compressor frequency is reported. It should be noted that on decreasing the compressor speed the COP increases. On decreasing the compressor speed the refrigerant mass flow rate diminishes and the condensation pressure too while the evaporation pressure presents a small increase; it follows that the compression ratio diminishes on decreasing the compressor speed. In Figure 12 compressor, evaporator and condenser capacities are given. It is necessary to observe that when the compressor velocity decreases the refrigerant mass flow rate decreases and so also the cooling, condensing and compressor capacities [1].

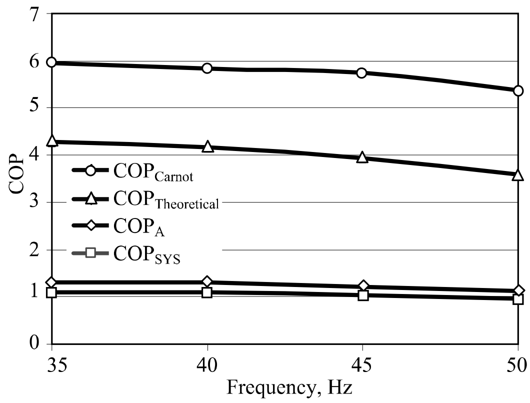

Figure 11 Variation of COP with frequency

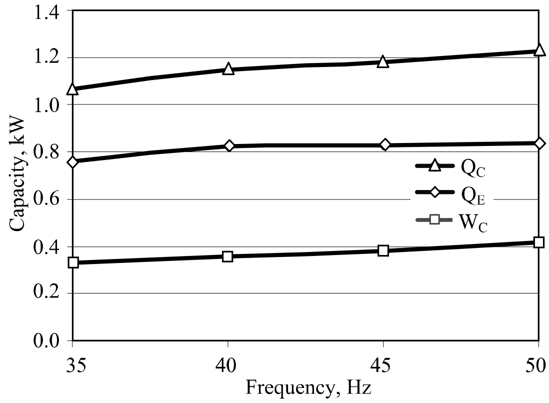

Figure 12 Variation of capacities with frequency

CONCLUSION

An experimental analysis was carried out to determine the performance of refrigeration system by varying its speed by means of an inverter. An alternative refrigerant, R404A was used in the experiments. Analyses were made to examine the effects compressor speed regulation in terms of thermodynamic aspects. Experiments were made for 35, 40, 45 and 50 Hz compressor electric motor frequencies. The results were given in terms of discharge and suction pressures and temperatures, compression ratio, evaporator, condenser and compressor capacities, and efficiencies. Four different COP values were determined. It was seen that all the parameters above affected by compressor speed. Especially, COP values were found to be higher in low frequency ranges. By the increase of compressor speed, COP values were decreased. Generally, it can be pointed out that, when the compressor speed decreases the plant performances are getting better.

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