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ABSTRACT

Capacity control with variable speed compressors in heat pump systems is one of the techniques having a potential for efficiency improvement in heat pump systems. It is anticipated that the compressor and inverter efficiency are influenced by changes of the compressor speed. The present experimental study evaluates these losses in a variable speed heat pump system.

The experimental results show that increasing the compressor speed reduces the heat pump COP up to 30%. The inverter loss increases as the compressor speed is increased, although the inverter loss as the percentage of the total compressor power decreases. Increasing the compressor speed increases the pressure ratio from 2.7 to 5.8, increasing the loss due to the pressure ratio mismatch drastically. Finally, the highest total isentropic efficiency of the compressor is obtained when the compressor frequency is close to 50Hz.

1. INTRODUCTION

Brine to water heat pump systems like Ground Source Heat Pumps (GSHPs) are one of the common and fast growing systems for heating the buildings in USA and European countries. As of 2005, there are over a million units installed worldwide providing about 15 GW of thermal capacity (Rybach 2005) and the annual global increase has been more than 10% over the last ten years (Lund *et al*. 2004). Therefore, any improvement in the efficiency of these systems can save a considerable amount of energy and decrease the greenhouse emissions to a large extent.

Capacity control by introducing variable speed compressors in heat pump systems is found by some researchers as one of the techniques having a potential for efficiency improvement in GSHPs. The speed of the inverter-driven compressor is modulated by the aid of a control algorithm so that the heat output from the heat pump unit offsets the load of the building; consequently, the discrepancy between the heat supply and the heat demand (in the building) can be reduced.

Zhao *et al.*(2003) made a comparison among variable speed capacity controls and three other control methods for GSHPs and concluded that changing the compressor speed is the preferred method; Furthermore, Madani *et al*. (2010) conducted a comparative analysis between on/off controlled and variable capacity heat pump system and they found that variable speed compressors heat pump systems yields a better performance compared to constant speed on/off control when the ambient temperature is below the balance point and the auxiliary heater operates. The annual modeling of both on/off and variable speed systems done by Madani *et al.*(2010) showed that when the on/off controlled HP system is designed to cover about 90% of the annual energy need and the auxiliary heater takes care of the rest, the seasonal performance factor of the system may be improved about 10% by switching to a variable speed HP system.

However, Karlsson and Fahlen (2007) concluded from their research that despite improved performance at part load (also shown by Qureshi and Tassou,1996), the variable-speed controlled heat pump did not improve the annual efficiency compared to the intermittently operated heat pump. Karlsson (2007) suggested that it is mainly due to the inefficiencies of the inverter, the electrical motor of the compressor, and the need for control of pumps used in the heating and ground collector systems (Karlsson 2007). The experimental study done by Cuevas and Lebrun (2009) showed that in the variable speed scroll compressor, the inverter efficiency varies between 95% and 98% when the compressor electrical power varies between 1.5 and 6.5 kW. It is also found that the additional electrical motor losses induced by the presence of the inverter are negligible (Cuevas and Lebrun 2009).

The present study aims at making an experimental analysis about the loss behavior in the variable speed compressor and frequency inverter when the compressor speed varies. The present paper provides an understanding how changing the compressor speed influences the frequency inverter losses and the total isentropic efficiency of the compressor; It would then be possible to evaluate the impact of the inverter and compressor loss on the annual performance of the heap pump system.

2. Methodology

The experimental setup built for the measurements consists of a variable speed heat pump unit, liquid pumps, a storage tank, two plate heat exchangers, valves, and a data acquisition system (see fig.1). In the variable speed heat pump unit, two brazed plate heat exchangers are used as the evaporator and condenser. The scroll compressor is equipped with a frequency inverter in order to facilitate various compressor speeds. An electronic expansion valve maintains the superheat temperature of the heat pump unit at 5ºC. When the speed of the compressor is changed, the following parameters are maintained constant:

- Approach heat sink temperature to condenser called load side temperature
- Approach heat source temperature to evaporator called source side temperature

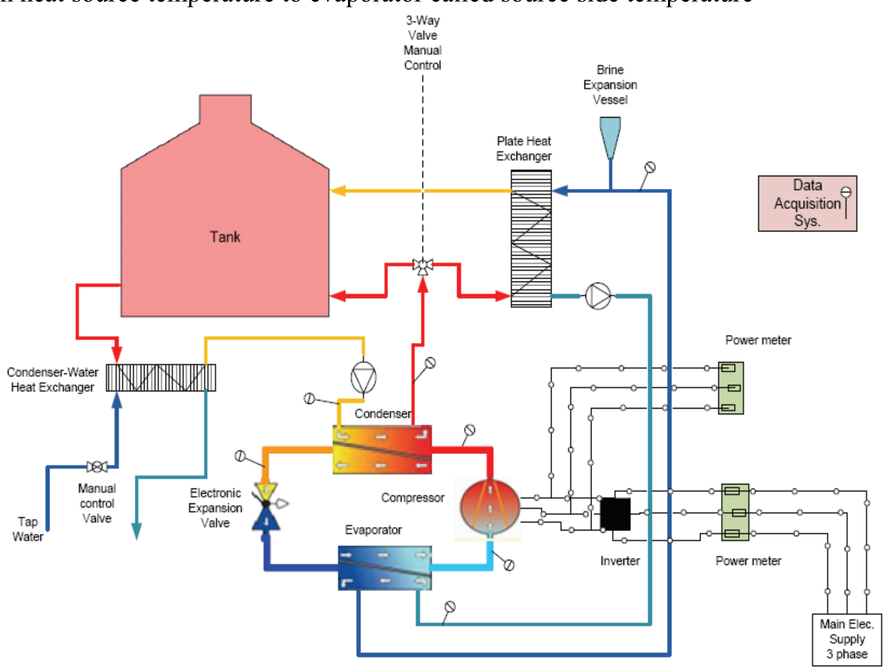


Fig. 1. The schematic of the experimental setup

Parameters which are measured and used in the present study are:

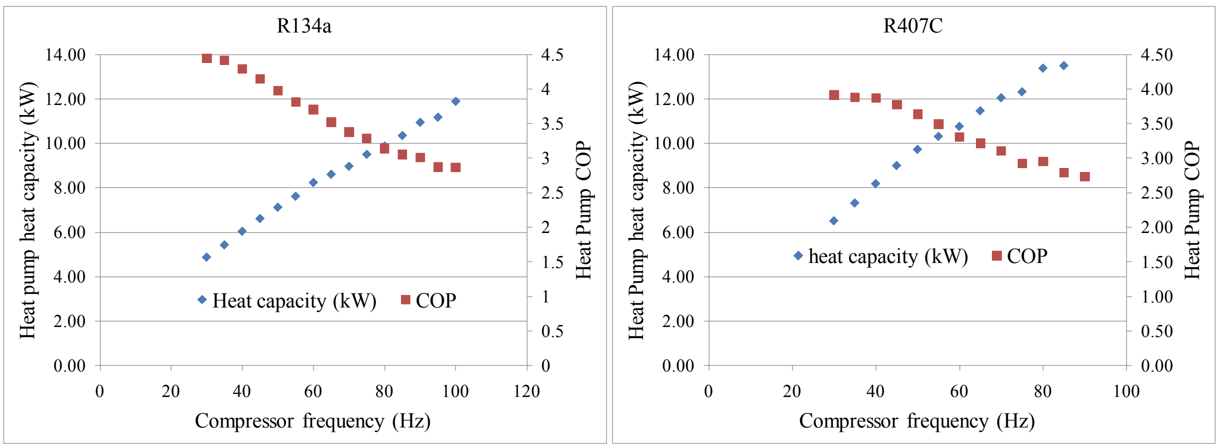
- The compressor power before and after the frequency inverter
- The compressor speed
- The heating and cooling capacity of the heat pump unit
- The condensation and evaporation pressure
- Superheat and sub-cooling temperature

Furthermore, the built-in volume ratio of the compressor together with the constant and variable parts of the electromechanical losses, are estimated by a compressor model (section 3.4). The built-in volume ratio found from the modeling is used to calculate the compression power in the compressor and to evaluate the loss due to the mismatch between the actual and the built-in pressure ratio.

3. Result

3.1 The heat pump heat capacity and Coefficient Of Performance (COP)

Fig. 2-a and 2-b shows the heat pump heat capacity and COP (defined as the ratio of heat capacity to compressor power) when changing the compressor frequency. R134a and R407C is used as the refrigerant in fig. 2-a and2-b, respectively. The source/load side temperatures are held constant at 4.5ºC/26ºC in all the operating points presented in both figures. As may be seen from fig. 2-a, varying compressor speed from 30 Hz to 100 Hz increases the heat capacity from 5 kW to 12 kW and decreases the COP from 4.4 to 2.9.
Furthermore(fig. 2-b) when the R134a is replaced by R407C, the heat pump yields higher heat capacity and lower COP. When the compressor speed increases from 30 Hz to 90 Hz, the heat capacity of the system with R407C increases from 6.5kW to 14kW and system COP decreases from 3.9 to 2.7 (see fig.2-b).

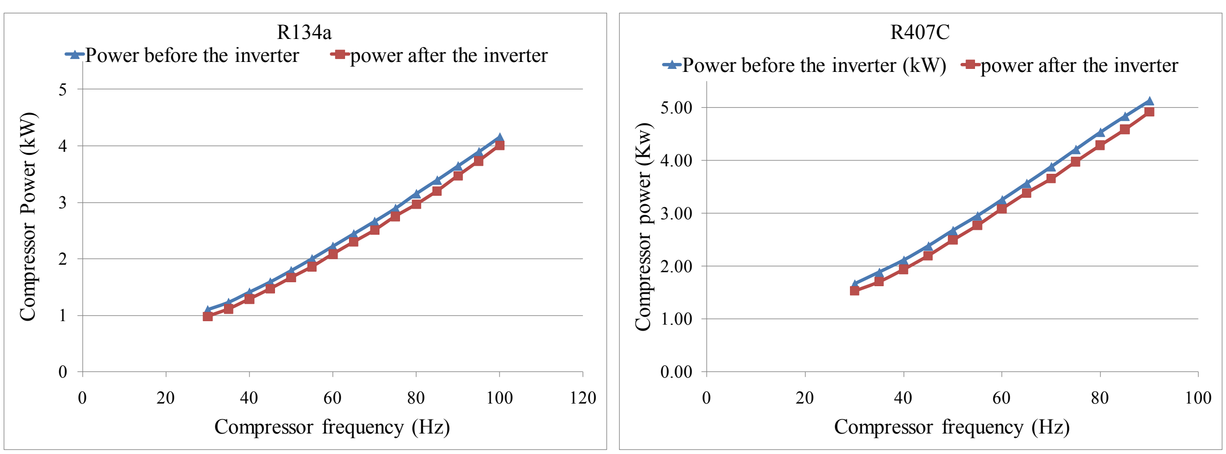


a) b)

Fig. 2. The heat pump heat capacity (kW) and COP versus compressor frequency when R134a (a) and R407C (b) is used as the refrigerant

3.2 The compressor power before and after inverter

Fig. 3-a and 3-b present the compressor power measured before and after the inverter when R134a or R407C is used as the refrigerant, respectively. The source/load side temperatures for both cases are held constant at 4.5ºC/26ºC. As may be seen from fig. 3-a, for the system with R134a, changing the compressor speed from 30Hz to 100Hz increases the compressor power from 1 kW to 4.2 kW. For the system with R407C, the compressor power increases from 1.7kW to 5.1kW by increasing the compressor frequency from 30Hz to 90 Hz (see fig. 3-b).

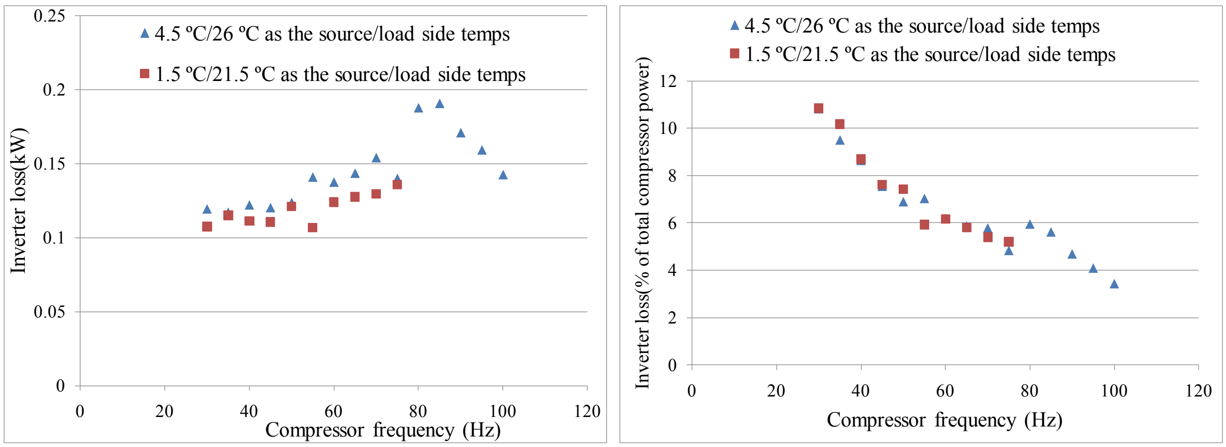


a) b)

Fig. 3. Compressor power (kW) measured before and after inverter when R134a (a) or R407C (b) is used as the refrigerant. The source/load side temperatures are 4.5ºC/26ºC for all the operating points.

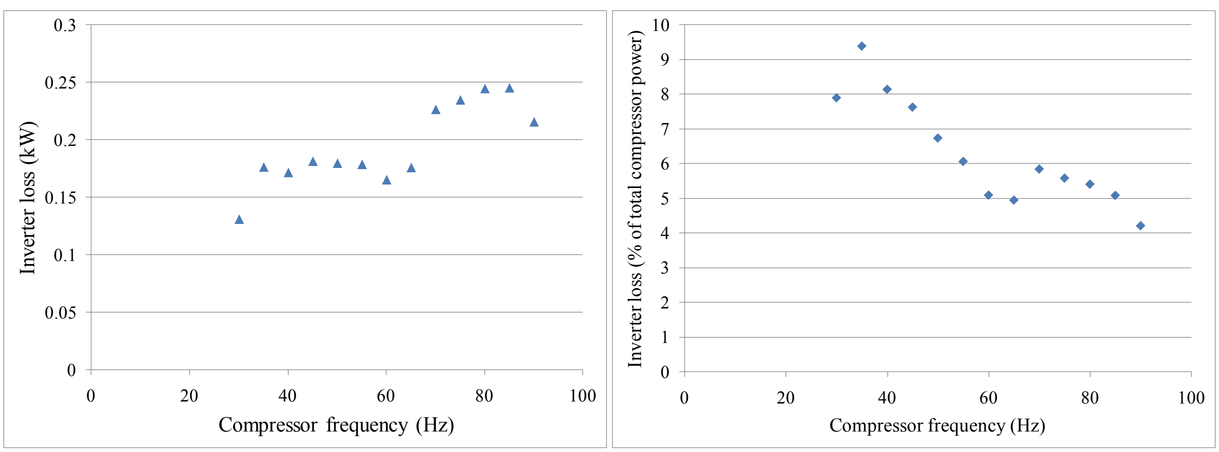
3.3 The inverter loss versus compressor frequency

Fig. 4 and fig.5 shows how the inverter loss changes when the compressor frequency increases from 30Hz to 100 Hz for two different set of source/load side temperatures (square and triangle dots). As it may be observed from fig. 4-a, with R134a the inverter loss varies between 100 W and 200 W with a peak at about 85 Hz. Fig. 4-b shows that the ratio of the inverter loss to the total compressor power (the power measured before the inverter) decreases from 11% to 3% when the frequency changes from 30 to 100 Hz. The trend for the system with R407C is very similar, as it can be seen from fig.5. The inverter loss varies between 130 W and 250 W. Furthermore, as it is presented in fig. 5-b, the higher the compressor speed, the lower the ratio of inverter loss to the total compressor power.



a) b)

Fig. 4. The measured inverter loss in kW (a) and in % of the total compressor power (b) when the frequency varies between 30Hz and 100Hz: the heat pump with R134a



a) b)

Fig. 5. The measured inverter loss in kW (a) and in % of the total compressor power (b) when the frequency varies between 30Hz and 90Hz: the heat pump with R407C

3. Compressor modeling

A semi-empirical of the compressor is made in order to estimate the built-in volume ratio, the compression power of the compressor (at every operating point), and the variable and constant part of the electromechanical losses in the compressor. Winandy (1999) obtained equation (1) to calculate the compression work, assuming an ideal gas:

$$W_{comp}=\frac{P_{sup}\cdot V_s}{\gamma-1}\cdot\left(\nu_i^{\gamma-1}-\gamma\right)+\frac{P_{ex}\cdot V_s}{\nu_i} \qquad (1)$$

ν_i is the built-in volume ratio, i.e. the ratio between the suction and discharge volume. The built-in volume ratio is a compressor characteristic based on the geometry and under idealized condition, it is related to built-in pressure ratio given by equation (2).

$$\pi_i = \nu_i^{\gamma} \tag{2}$$

The difference between the built-in pressure ratio and the actual pressure ratio (the ratio of condensation pressure to evaporation pressure) can influence the compressor efficiency, as it will be discussed later in this section. Furthermore, the swept volume flow can be determined by the equation (3):

$$\dot{V}_s = \frac{V_s . n}{60} \tag{3}$$

n is the compressor speed in rpm. It is possible to estimate the compression power in the compressor by combining Equation (1) and (3). Moreover, as Winandy (1999) and Cuevas *et al*.(2010) suggested, the total compressor power comprises of the internal compression power and the compressor electromechanical losses which also can be divided into two terms: constant part and variable part which varies by the internal compression power, as given in equation (4):

$$\dot{W} = \frac{\dot{W}_{comp}}{\eta} + \dot{W}_{cons} \tag{4}$$

where η is the parameter representing the variable part of electromechanical losses, and $\dot{W}_{cons}$ is the constant part of electromechanical loss (kW). Consequently, η , $\dot{W}_{cons}$, and built-in volume ratio are the three parameters which are found by the semi-empirical parameter estimation model (see fig.6).

Table 1 presents the results of the modeling for the analyzed heat pump system with two different refrigerants: R134a and R407C. The difference between the built-in volume ratios for the heat pump with R134a and R407C can be due to the uncertainties in the compressor model (about 6%).

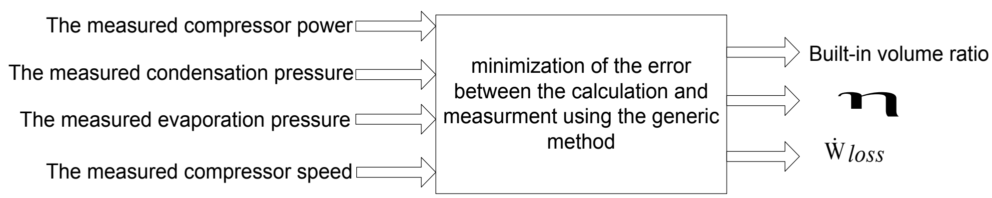


Fig. 6. The flowchart of compressor modeling

Table 1. Results of the modeling for the analyzed heat pump system in two different cases: the heat pump uses either R134a or R407C as the refrigerant

The estimated parameter	The heat pump with R134a	The heat pump with R407C
Built-in volume ratio	2.17	2.04
η	0.72	0.86
$\dot{W}_{cons}$	0.245	0.44

Fig.7 shows the influence of the mismatch between the built-in and actual pressure ratio on the isentropic efficiency when the actual pressure ratio changes by varying the compressor speed. Fig. 7 is made based on equation (5) suggested by Granryd (2005) in order to estimate the losses due to the fact that the actual pressure ratio for compression does not match the built-in pressure ratio. As fig.7 depicts, the compressor tested in the present study is designed in a way that the loss due to the pressure ratio mismatch is very low when the actual pressure ratio is close to 2.7 and it decreases when the pressure ratio increases from 2.7 to 5.8. It is predicted that the

$$\eta_{built\_in} = \frac{\left(\frac{Pex}{Psup}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\pi_i^{\frac{\gamma-1}{\gamma}} - \frac{\gamma-1}{\gamma} . \pi_i^{-\frac{1}{\gamma}} . \left(\pi_i - \frac{Pex}{Psup}\right) - 1} \tag{5}$$

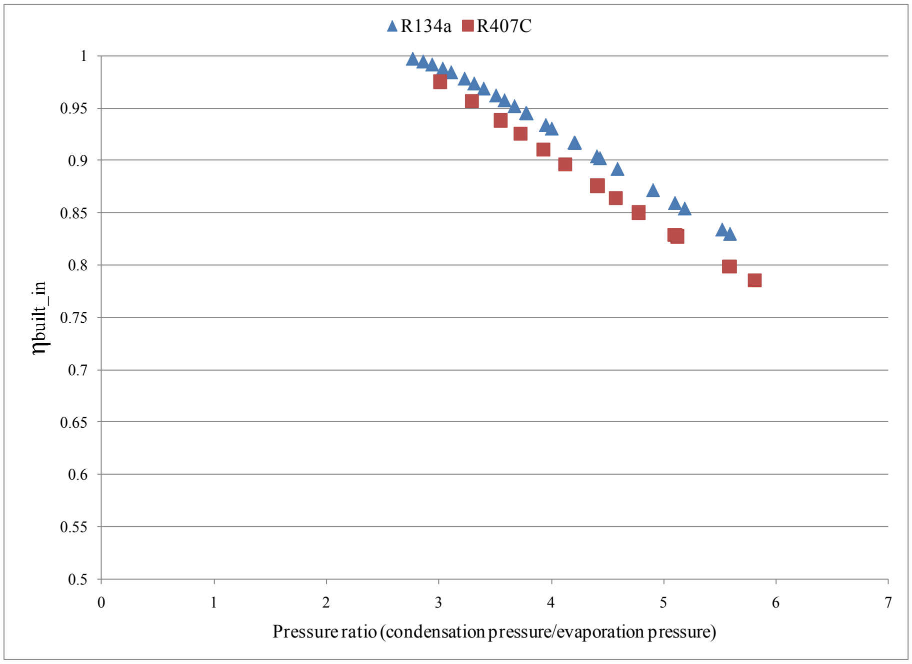


Fig. 7. The influence of built-in volume ratio on isentropic efficiency of compressor

3.5 The compressor total isentropic efficiency

Fig. 8 present the total isentropic efficiency of the compressor versus the compressor frequency for two sets of source/load side temperatures, when R134a (fig.8-a) or R407C (fig.8-b) is used as the refrigerant. This efficiency, η_{isen} in equation (6), covers all kind of losses including both constant and variable parts of electromechanical losses, presented in equation (3), and also the losses due to the pressure ratio mismatch (shown in figure 7).

$$\dot{W}=\frac{\dot{W}_{isen}}{\eta_{isen}} \tag{6}$$

As it is shown in fig.8, the total isentropic efficiency of the compressor changes about 11% when the compressor frequency changes from 30Hz to 100Hz. Furthermore, fig. 8 shows that the highest total isentropic efficiency occurs when the compressor frequency is between 50Hz and 55Hz for all the presented cases.

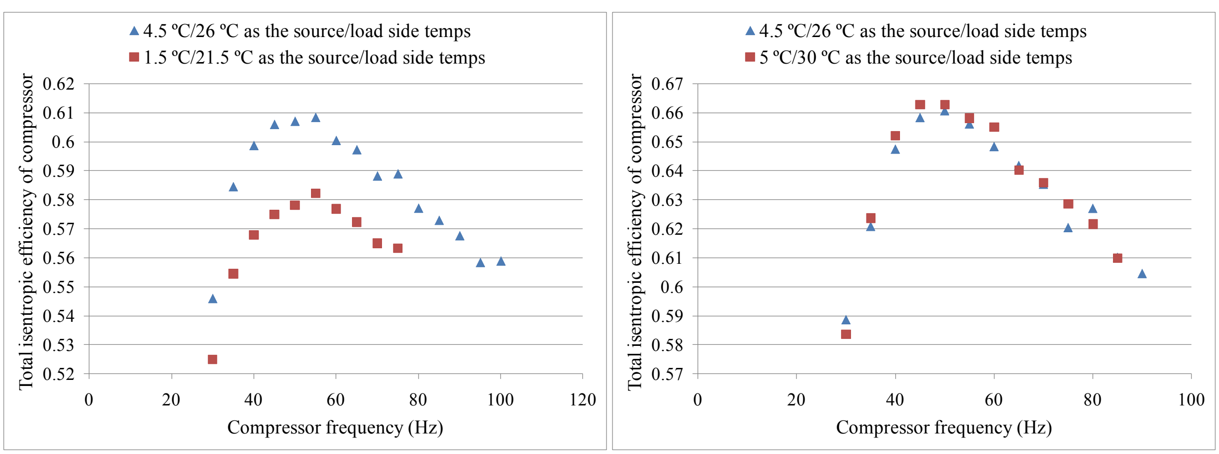


Fig. 8. The variation in the total isentropic efficiency of the compressor based on the changes in the compressor frequency when R134a (a) or R407C (b) is used as the refrigerant

4. Conclusion

The present study mainly aims at analyzing the loss behavior in the variable speed compressor and frequency inverter when the compressor speed varies. To make the analysis, an experimental setup is made and measurements are done for two different cases:

- The heat pump uses R134a as the refrigerant
- The heat pump uses R407C as the refrigerant

When the compressor speed is changed, other boundary parameters such as source/load side temperatures are kept constant.

The measurement results showed that increasing the compressor speed from 30 Hz to100Hz lowers the heat pump COP up to 30% for both refrigerants used. Furthermore, the inverter loss increases quantitatively by increasing the compressor speed. For example, for the heat pump with R407C, increasing the compressor frequency from 30Hz to 90 Hz almost doubles the amount of inverter loss (the inverter loss increases from 130W to 250W) although the inverter loss as the percentage of the total compressor power decreases from 8% to 4%.

The compressor loss due to the mismatch between the actual pressure ratio and the built-in pressure ratio is found based on the compressor built-in volume ratio of the compressor. A semi-empirical model of the compressor is made to obtain the built-in volume ratio for the compressor studied in the present paper. Then, the loss due to the pressure ratio mismatch is studied and it is found that this loss rises from 1% to 20% when the pressure ratio increases from 2.7 to 5.8.

Finally, the total isentropic efficiency of the compressor which represents all kinds of losses in the compressor was analyzed at different compressor speeds. The results showed that the total isentropic efficiency of the compressor shows a maximum value when the compressor frequency is close to 50Hz. This efficiency changes about 11% when the compressor frequency varies between 30Hz and 100Hz.

NOMENCLATURE

n	Speed	(rpm)
P	Pressure	(kPa)
V	Volume	(m^3)
$\dot{V}$	Volume flow rate	(m^3/s)
W	Work	(J)
$\dot{W}$	Power	(kW)
γ	Polytropic exponent	(–)
η	Efficiency	(–)
ν_i	Built-in volume ratio	(-)
π_i	Built-in pressure ratio	(-)

Subscripts

cons	constant
comp	compression
ex	exhaust
isen	isentropic
sup	supply

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