# Capacity Modulation of an Inverter Driven Heat Pump with Expansion Devices

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**Key words** : Inverter heat pump, Expansion device, Capillary tube, TXV, EEV, Capacity modulation

## Abstract

An experimental study was performed to investigate characteristics of an inverter driven heat pump system with a variation of compressor frequency and expansion device. The compressor frequency varied from 30 Hz to 75 Hz, and the performance of the system applying three different expansion devices such as capillary tube, thermostatic expansion valve (TXV), and electronic expansion valve (EEV) was measured. The load conditions were altered by varying the temperatures of the secondary fluid entering condenser and evaporator with a constant flow rate. When the test condition was deviated from the standard value (rated value), TXV and EEV showed better performance than capillary tube due to optimal control of mass flow rate and superheat. In the present study, it was observed that the variable area expansion device had better performance than constant area expansion device in the inverter heat pump system due to active control of flow area with a change of compressor frequency and load conditions.

## 1. Introduction

Since a single-speed heat pump system operates at a fixed frequency, it is not able to correspond efficiently to the change of heating/cooling loads. It has a limitation on providing adequate cooling capacity at high load conditions, and it has disadvantages in comfort and efficiency due to on/off control at low load conditions.

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observed at the film Reynolds number of 90 in a vertical falling film tube. The absorption mass fluxes under the subcooled condition were higher than those under the superheated condition.

(3) The heat transfer coefficients under the subcooled condition were higher than those under the superheated condition. The heat transfer coefficients were much influenced by the inlet temperature of solution rather than by the temperature of coolant.

(4) The maximum absorption effectiveness approached to 23 % at the coolant temperature of 30℃, 31% at 35℃ under the subcooled condition.

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An inverter driven heat pump has advantages over the single-speed heat pump system in energy saving[1,2], easy capacity control[3,4], and improvement in comfort[5]. The inverter heat pump modulates capacity of the system through frequency control. When the cooling load of the system increased, the compressor frequency increased to supply enough capacity to conditioning space. On the contrary, the frequency reduced instead of on/off control of the system when outdoor temperature decreased.

The expansion devices used in refrigeration systems include TXV, manual expansion valve, constant pressure expansion valve, capillary tube, float valve and EEV. The expansion device controls refrigerant mass flow rate and pressure balance in the system. The selection of optimum expansion device having capabilities of controlling flow rate in the wide operating range is essential because the inverter driven heat pump system operates at a wide range of mass flow rate. Due to low price and high reliability, the capillary tube was widely used in air conditioning systems. However, the capillary tube covers the limited range of mass flow rate because its flow area is constant. It is necessary to study on the expansion device that can maintain optimum cycle at all frequencies in the inverter heat pump.

Choi et al.[6] studied the performance of the inverter heat pump as a function of compressor frequency, outdoor temperature and length of capillary tube. They showed that the optimal capillary length for the system was inversely proportional to the frequency. Hewitt et al.[7] compared the superheat characteristics of TXV and EEV, and showed that EEV had better superheat control characteristics than TXV. Farzad[8] studied optimum charge with expansion devices in a refrigeration system. The charge amount was changed from -20% to +20% of the suggested value by the manufacturer. The expansion device was replaced with capillary tube, orifice tube or TXV. The capacity and COP was the most sensitive to the charge amount in the capillary tube system among them. As the outdoor temperature varied, the TXV was more sensitive to the charge amount than the capillary or orifice tube. The seasonal energy efficiency ratio (SEER) of the capillary tube system was reduced by about 21% and 11%, and that of the TXV system was reduced by 2% and 8%, when the charge amount varied from -20% to 20% of the rated value. However, the SEER of the orifice tube system was constant.

In the present study, the characteristics of the inverter driven heat pump were investigated according to outdoor/indoor conditions, expansion devices and driving frequency of the compressor. Capillary tubes were used as constant flow area expansion device. In addition, TXV and EEV were used as variable flow area expansion devices.

Because the optimum capillary tube length was altered with a variation of frequency, the relationship between capillary tube length and system performance according to frequency was measured. In addition, the performances of the system with TXV and EEV were surveyed and compared with a variation of environmental condition and compressor frequency.

## 2. Experimental Setup

### 2.1 Experimental facility

Figure 1 shows the experimental apparatus. The facility consists of two parts: the refrigerant circulation cycle and the secondary fluid circulation cycle. The refrigerant cycle included compressor, condenser, expansion device, and evaporator. Two additional expansion devices, oil separator and accumulator were installed on the refrigerant cycle. The secondary fluid cycle consisted of constant temperature baths, chillers, and heaters. Double tube heat exchangers were used as the condenser and the evaporator for heat transfer between refrigerant and secondary fluid. The secondary fluid for the condenser was water, and that for the evaporator was ethyleneglycol/water (60/40 wt %) mixture. The constant temperature bath maintained the temperature of the secondary fluid at the setting point. Outdoor and indoor conditions were controlled by altering temperature of the secondary fluid while flow rate was maintained at a specified value. The rotary compressor having rated capacity of 1 RT was used in this study.

## 2.2 Experimental methods

The major parameters to be measured were refrigerant mass flow rate, compressor power input, pressure and temperature at several points in the refrigerant cycle, temperature and flow rate of the secondary fluids, and refrigerant charge amount. Pressures and temperatures in the refrigeration cycle were measured using pressure transducer and T-type thermocouple, respectively. Refrigerant mass flow rate was measured by the mass flow meter. Power consumption in the compressor was monitored using the power meter installed between the inverter and the compressor. The flow rate of the secondary fluid was measured by the turbine flow meter.

Compressor discharge temperature and refrigerant mass flow rate were the major parameters for the criterion of steady state. Data

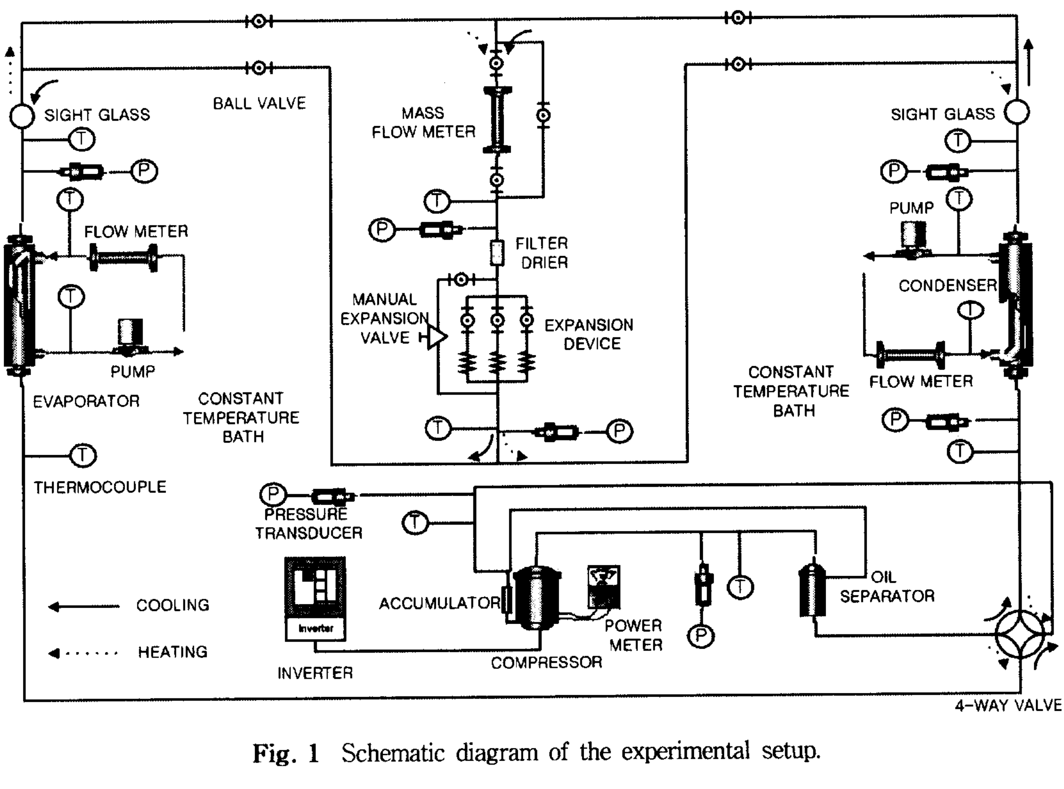


**Fig. 1** Schematic diagram of the experimental setup.

were stored with 10 second interval after the compressor discharge temperature and the refrigerant flow rate vary within ±0.5℃ and ±0.5 kg/hr, respectively.

Table 1 shows test conditions. The flow rate of the secondary fluid was maintained at 7 l/min for the condenser and 17 l/min for the evaporator. The system performance with the various expansion devices (capillary tubes of various lengths, TXV and EEV) was observed with a variation of the secondary fluid temperature in the condenser and the evaporator at a constant compressor frequency of 60 Hz. The variation of the cycle was also observed as to compressor frequency ranging from 30 to 75 Hz with a secondary fluid temperatures of 30℃ and 28.5℃ entering the condenser and evaporator, respectively.

## 3. Results and Discussion

Capacity can be obtained by equation (1) with temperature difference ($\Delta T$) and volume flow rate of the secondary fluid ($Q^{\&}$). Density ($\rho$) and heat capacity (C) of 60% ethylenglycol solution are calculated with curve fitted data from manufacturer.

$$\dot{q} = \rho \dot{Q} C \Delta T \tag{1}$$

Experimental uncertainties of capacity and COP in this study were 3.4% and 3.5%, respectively.

### 3.1 System performance with secondary fluid temperature

The characteristics of the water-to-water heat pump system for the various expansion devices were investigated while the inlet temperature of the secondary fluid to the condenser was altered. To decide the optimum capillary tube length and the optimum refrigerant charge amount, the preliminary tests were performed. The secondary fluid temperature entering the condenser in this preliminary test was maintained at 34℃ and that for evaporator was kept at 28.5℃ with the compressor frequency of 60 Hz. The range of charge amount was from 800 g to 1150 g, and the length of capillary tube varied from 600 mm to 1200 mm with an inner diameter of 1.7 mm. As results, the optimum values for refrigerant charge and capillary length were selected as 920 g and 700 mm, respectively. At the same operating conditions, optimum refrigerant charge for TXV system was 910 g and that for the EEV system was 800 g. With the optimum charge amount, the experiment was performed with test conditions given in Table 1.

Figures 2 and 3 show the variations of cooling capacity temperature of the condenser at a compressor frequency of 60 Hz. The secondary fluid temperature at the condenser inlet varied from 30℃ to 39℃ with a condenser inlet varied from 30℃ to 39℃ with a constant secondary fluid temperature of 26.1℃ at the evaporator inlet. As the secondary fluid temperature en-

**Table 1** Test conditions

| Inlet temperature of secondary fluid (℃) | | Compressor frequency (Hz) |
|---|---|---|
| Evaporator* | Condenser** | |
| 26.1 | 30.0 | 30, 60, 75 |
| | 34.0 | 60 |
| | 39.0 | 60 |
| 28.5 | 30.0 | 30, 60, 75 |
| | 34.0 | 60 |
| | 39.0 | 60 |

* Volumetric flow rate = 7 (l/min)
** Volumetric flow rate = 17 (l/min)

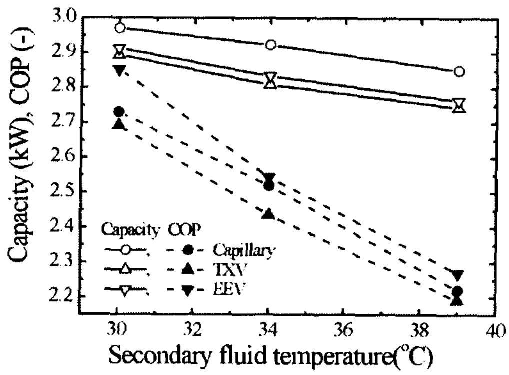


**Fig. 2** Variation of capacity and COP with secondary fluid temperature entering condenser (evaporator secondary fluid temperature = 26.1 ℃).

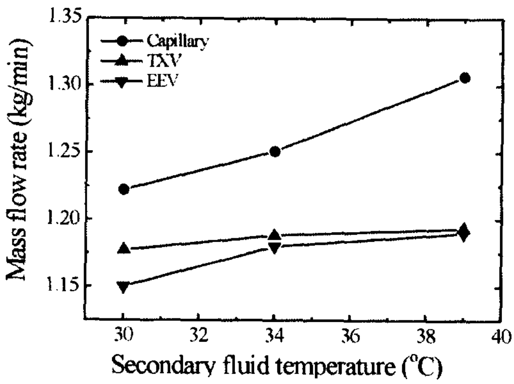


**Fig. 3** Variation of mass flow rate with secondary fluid temperature entering condenser (evaporator secondary fluid temperature = 26.1 ℃).

tering the condenser increased, cooling capacity and COP diminished.

Both the condensing pressure and the mass flow rate through the capillary tube were augmented with an increase of the inlet temperature of the secondary fluid at the condenser. The evaporating pressure also slightly increased, and the mass flow rate provided by the compressor was enhanced due to the rise of refrigerant density at the inlet of the compressor. Thus, the total mass flow rate of the system increased with an increase of the secondary fluid temperature at the condenser inlet. However, the enthalpy difference between saturated vapor and saturated liquid state became smaller due to an increase of evaporating pressure. In addition, the decrease of subcooling and superheat led to reduction of cooling capacity.

For the system with TXV or EEV, as the secondary fluid temperature at the condenser inlet increased, the drop of cooling capacity was larger than that with capillary tubes. The slope of change of mass flow rate with respect to the secondary fluid temperature was smaller than that with capillary tubes as shown in Fig. 3. For the capillary tube, the mass flow rate increased corresponding to the rise of upstream pressure, whereas mass flow rates of TXV and EEV were proportional to the difference between upstream and downstream pressure. Thus, for the system with TXV or EEV, it was difficult to obtain higher flow rate with a slight increase in the pressure difference between condenser and evaporator.

Figure 4 shows the variation of subcooling and superheat as a function of secondary fluid temperature of the condenser. Although subcooling at inlet of TXV and EEV and superheat at inlet of the compressor increased slightly with secondary fluid temperature of the condenser, the decrease of cooling capacity was significant due to an increase of mass flow rate. However, additional power input to compressor was also less than that of capillary tube system, while the drop of COP with an application of TXV was similar to that for the system with capillary tubes.

As the secondary fluid temperature entering the condenser increased from 30℃ to 39℃, cooling capacity and COP of the system decreased by 3.4% and 18.0% for the system with the capillary tube, 5.2% and 18.3% for the TXV system, and 5.2% and 20.7% for the EEV system, respectively.

The characteristics of the heat pump system were also studied according to the secondary fluid temperature entering the evaporator as shown in Fig. 5. The temperature of the secondary fluid at the evaporator inlet varied from 26.1℃ to 28.5℃ with a constant secondary fluid temperature of 34℃ entering the condenser, and a compressor frequency of 60 Hz. As shown in Fig. 5, the cooling capacity and COP increased as the secondary fluid temperature entering the evaporator increased. TXV showed higher slopes of cooling capacity and COP as a function of the secondary fluid temperature of the evaporator as compared to those of EEV and capillary tube.

Figure 6 shows variation of subcooling and superheat as a function of evaporator secondary fluid temperature. Capillary system represented much more increase of superheat than that for TXV and EEV. Refrigerant density at the compressor inlet was dependent on the evaporating pressure and superheat. For the system with TXV, the evaporating pressure increased more than that with capillary tube or EEV. Therefore, the increment of mass flow rate and capacity was more significant in TXV system.

For the EEV system, there was small increment of the evaporating pressure and small pressure difference between the condensing and evaporating pressure, that led to slight increase of the mass flow rate. The capacity and COP for the system with capillary were enhanced by 6.8% and 4.6%, whereas 10.2% and 7.1% for TXV and 7.1% and 5.2% for EEV, respectively, as the secondary fluid temperature entering the evaporator increased from 26℃ to 28.5℃.

## 3.2 System performance with frequency

The effects of compressor frequency on system performance were analyzed when the secondary fluid temperatures entering condenser and evaporator were kept constant at 30℃ and

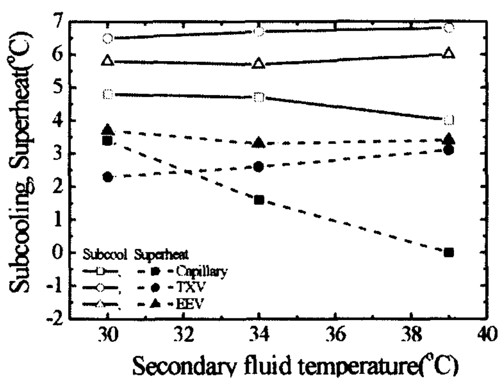


**Fig. 4** Variation of subcooling and superheat with secondary fluid temperature entering condenser (evaporator secondary fluid temperature = 26.1℃).

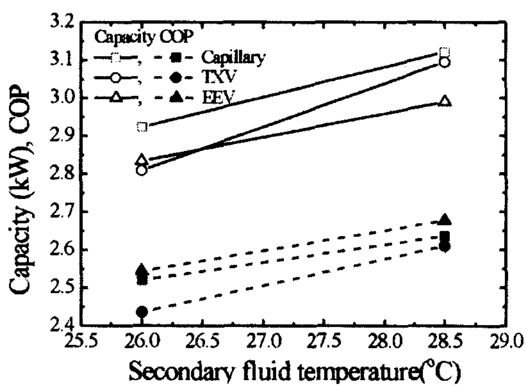


**Fig. 5** Variation of capacity and COP with secondary fluid temperature entering evaporator (condenser secondary fluid temperature = 34.0 ℃).

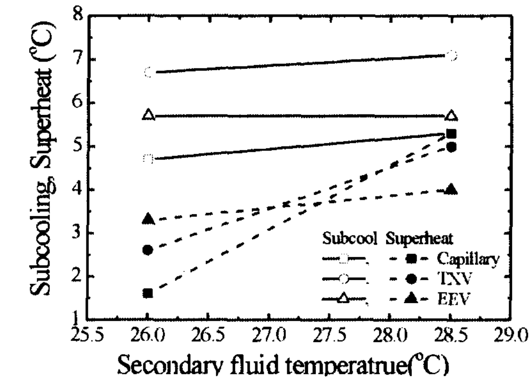


**Fig. 6** Variation of subcooling and superheat with secondary fluid temperature entering evaporator (condenser secondary fluid temperature = 34.0℃).

28.5℃, respectively. The cooling capacity generally increased with the compressor frequency, as shown in Fig. 7.

For the system with capillary tube, as the compressor frequency increased, the condensing pressure and pressure drop in the capillary tube increased. It led to a rise of refrigerant mass flow rate and compression ratio. As results, more compressor power was consumed. The total enthalpy difference between inlet and exit of evaporator reduced due to an increase of the condensing pressure. However, the capacity of the system was enhanced due to the reduction of the evaporating pressure and increment of the mass flow rate.

For the TXV system, the capacity of the system also increased with the frequency. The increment of the capacity was much higher than that of the capillary system due to higher mass flow rate.

When the driving frequency increased, the evaporating pressure decreased and saturation pressure of the refrigerant in the sensing bulb of the TXV dropped. The superheat deviated from the desired value due to the reduction of saturation pressure in the sensing bulb and the mass flow rate increased corresponding to valve opening.

Generally, the pressure difference between the condensing and evaporating pressure increased with a rise of driving frequency. The mass flow rate in the capillary tube depended on upstream pressure (condensing pressure), however mass flow rates through TXV or EEV relied on the pressure difference between the condensing and evaporating pressure. There-

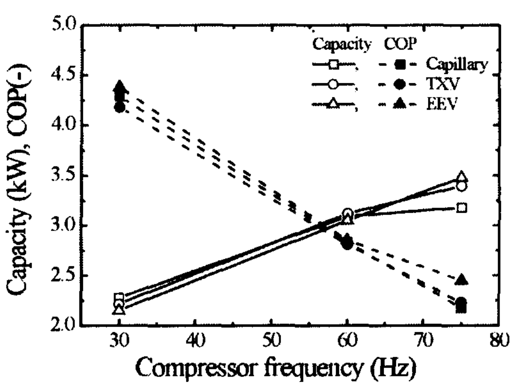


**Fig. 7** Variation of capacity and COP with compressor frequency.

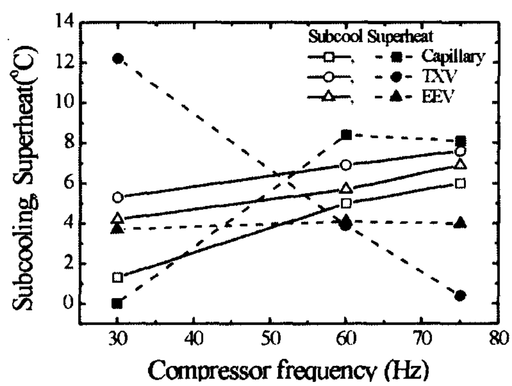


**Fig. 8** Variation of subcooling and superheat with compressor frequency.

fore, the increases of mass flow rate in TXV or EEV system were higher than the system with the capillary tube. However the COP of the both system were similar each other, because the TXV shows sharp increase in compressor power consumption as compared to the capillary system.

The system with EEV showed higher COP and capacity at 75 Hz than that of the capillary tube system. The superheat and subcooling of the system with compressor frequency were maintained at 4~5℃ in EEV system as shown in Fig. 8. Therefore, the COP of the EEV system was higher than that of the capillary system.

Generally, for low frequencies, low superheat was observed in the capillary tube system, while the superheat in the EEV system was maintained at a specified value. The higher superheat led to an increase of the compressor power consumption and reduction of the refrigerant mass flow rate. Based on the experimental results, it can be concluded that the EEV is the best expansion device among the expansion devices considered in this study because the EEV controls superheat actively to get an optimal cycle in the inverter driven system.

## 4. Conclusions

The expansion devices used in the inverter driven refrigeration system must cover wide range of mass flow rate according to the variation of compressor speed. In the present study, the characteristics of the inverter heat pump system according to the variation of expansion devices were investigated as a function of outdoor and indoor condition and compressor frequency. The results of the present study can be summarized as follows:

(1) As the secondary fluid temperature at the inlet of condenser increased, the heat pump system with EEV showed higher performance than the capillary system.

(2) The change of system performance in TXV and EEV system according to the secondary fluid temperature entering the evaporator was higher than that of the capillary tube system.

(3) When the compressor frequency increased, the heat pump system with TXV and EEV showed higher enhancement in cooling capacity and mass flow rate as compared to the system with the capillary tube.

(4) Expansion devices with variable flow area showed better performance than constant area expansion devices due to active control of the refrigerant mass flow rate with the compressor frequency.

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