Renewable Energy 79 (2015) 66–71

Contents lists available at ScienceDirect

Renewable Energy

journal homepage: www.elsevier.com/locate/renene

# Evaluation of the impacts of high stage refrigerant charge on cascade heat pump performance

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ARTICLE INFO

*Article history:*
Received 25 March 2014
Accepted 22 July 2014
Available online 19 August 2014

*Keywords:*
Cascade heat pump
High stage
Low stage
Refrigerant charge amount
Capacity
COP

ABSTRACT

Heat pump systems offer economical alternatives for recovering heat from different sources for use in domestic and industrial refrigeration, space heating, and air conditioning. Some of these domestic and industrial applications require very low evaporating temperatures and very high condensing temperatures which induce high compressor pressure ratios beyond the practical range for single-stage heat pump cycles. This challenge can be overcome by adopting cascade heat pump cycles. In this study, a water-to-water cascade heat pump is tested to investigate the effects of high stage refrigerant charge amount on the performance in a steady state and heating mode operation. The temperature difference between the condensing temperature of the LS cycle and the evaporating temperature of the HS cycle at cascade heat exchanger was increased by reducing the HS refrigerant charge amount, while the heat transfer rate between HS and LS cycles decreased due to a decreasing of refrigerant flow rate. Finally, COP showed lower value at the undercharged condition than that at the fully charged condition. The slope of the capacity with a HS charge amount was much steeper at undercharged conditions than that at overcharged conditions. For HS undercharged conditions, the heating capacity decreased greatly, because heat transfer rate from LS cycle to HS cycle reduced.



## 1. Introduction

Over the years, the world has seen increasing demand for domestic and industrial refrigeration, space heating, air conditioning and hot water generation. Heating and hot water generation are mainly obtained through the burning of fossil fuel in boilers and the use of electric heaters. However, these processes induce rise in energy consumption, environmental pollution and global warming. Another great challenge is the recent hike in fuel prices all over the world. The development and use of energy saving and new & renewable energy technologies are effective methods to overcome these challenges [1]. The ability of heat pump systems to recirculate environmental and waste heat back into the heat production process makes it highly efficient and energy saving [2]. Also heat pump adopting geothermal or solar energy sources as a heating and cooling system for building and industries is very efficient and has very low impact on the environment [3,4].

Due to the rapid increase of global civilization, very low evaporating temperature and high condensing temperature are required for both industrial and domestic purposes. However heat pumps, like any other system have their own limitations. At high temperature and pressure ranges beyond the practical range for single stage cycle, the system tends to produce very low heating capacity and low COP and in worse cases, becomes impractical. High pressure ratio also exposes the compressor to high discharge temperature, low volumetric efficiency and eventually damages it [5]. These problems are successfully overcome and the performance of the system is enhanced by using cascade cycle [1,6,7].

In other to maximize the reliability and performance of a heat pump system, the compressor should have high efficiency and also be in optimization with other parts. The amount of the refrigerant in the system is another crucial factor to the performance of the heat pump system [2,8,9].

A two-stage $CO_2/C_3H_8$ cascade system for simultaneous refrigeration and heating was studied by Bhattacharyya et al. [10]. The system was analyzed for optimum performance parameters with respect to system COP and exergetic efficiency and validated with experimental result. Lee et al. [11], Dopazo et al. [12] and Dopazo and Fernandez-Seara [13] conducted experiments on $CO_2/NH_3$

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http://dx.doi.org/10.1016/j.renene.2014.07.042
0960-1481/

**Nomenclature**

| | |
|---|---|
| COP | coefficient of performance |
| $C_p$ | specific heat [kJ/kg·C] |
| EEV | electronic expansion valve |
| EWT | entering water temperature [°C] |
| HS | high stage cycle |
| ID HX | indoor heat exchanger |
| LS | low stage cycle |
| LWT | leaving water temperature [°C] |
| $\dot{m}$ | mass flow rate [kg/s] |
| OD HX | outdoor heat exchanger |
| $\dot{Q}$ | heating capacity [kW] |
| $U$ | uncertainty |
| $W$ | power consumption [kW] |
| $x$ | nominal value of variable |

cascade systems. Optimal cascade condenser temperature (also the condensing temperature of the low temperature cycle) and its corresponding maximum COP and exergetic efficiency were determined. Houcek and Thedford [14] showed that the single stage heat pump unit's capacity and COP (Coefficient of Performance) were reduced at outrange of nominal charge conditions. Farzard and O'Neal [9] also reported refrigerant charging effects on the performance of a single stage heat pump unit with a capillary tube, short tube orifice, and TXV. It was found that the TXV heat pump unit showed a small variation of the COP according to refrigerant charge but a strong dependence on the outdoor temperature. Choi and Kim [2,8] investigated the effects of the expansion device on the performance of a single stage heat pump unit under various charging conditions, providing information on the design of an expansion device and refrigerant charge.

Most of the researches on cascade heat pumps were executed to improve performance with the variation of the operating conditions, and previous studies on the effects of refrigerant charge were focused on the single stage heat pump. A comprehensive study on the effects of refrigerant charge in the cascade heat pump is required to optimize system performance and to achieve proper capacity modulation. In this study, a water-to-water cascade heat pump is tested to investigate the effects of high stage refrigerant charge amount on the performance in a steady state and heating mode operation.

## 2. Experimental setup and test procedure

An experimental setup was designed to measure the performance of the water-to-water cascade heat pump according to high stage refrigerant charge amount. Fig 1 shows a schematic diagram of the test rig. The test rig included two heat pump cycles (high stage and low stage cycle) and a secondary fluid flow loop. R410A and R134a were selected as working fluids for low and high stage cycle respectively. Each heat pump system consisted of a scroll compressor, two double tube type heat exchangers and an expansion device. The indoor heat exchanger worked as a condenser in the high stage. The outdoor heat exchanger played a role of evaporator in the low stage. The cascade heat exchanger also worked as an evaporator of the high stage and a condenser of the low stage. All the heat exchangers had counter flow pattern. Electronic expansion valves (EEV) were installed at both the high and low stages of the system to help regulate precisely the flow rate of the refrigerant. A stepping motor using 1–2 excitation method drove the EEV. Control system for EEV driving included an A/D card, stepping motor driver, and computer.

The cascade heat pump was connected to two constant temperature baths for simulating heat source and heat sink. The constant temperature baths were both equipped with an electric heater and a refrigerator to provide exact entering temperature conditions. Ethylene glycol was selected as the secondary fluid for the cascade heat pump system because of its simplicity of capacity measurement. Ethylene glycol loops for the indoor side and outdoor side heat exchanger were closed loop having a magnetic pump. An inverter driven pump and manual needle valve controlled the secondary fluid flow rate supplied to the indoor and outdoor heat exchangers to establish test conditions. Sight glasses were installed at the inlets of compressors and at the outlets of EEVs. The system was operated at flow rates of 12 LPM for the secondary fluid.

Temperatures in the setup were monitored closely at vantage points using thermocouples according to ASHARE standard [15]. Refrigerant pressure was also measured with the use of pressure transducers (RTD), according to the ASHARE standard [16]. Mass flow meters were also installed into the test rig at both high and low stages to measure the mass flow rate of the refrigerants flowing through the system. A volumetric flow meter was installed to measure water flow rate in the secondary flow loop. Each sensor was calibrated to reduce experimental uncertainties. The specifications and uncertainties of sensors are summarized in Table 1.

The first step of the test procedure was to determine full charge under a standard condition, the secondary fluid water temperatures of 40 °C and 5 °C entering the indoor heat exchanger(ID HX) and outdoor heat exchanger(OD HX), respectively. The standard condition was chosen according to ISO 13256-2 [17] and NR GT 101 standards [18]. Both the secondary fluid flow rate through ID HX and OD HX were kept constant at 12 LPM. The full charge condition was selected to have the maximum COP. The refrigerant was added into the heat pump in a 50 g increment until the maximum COP was obtained. The full charge amount was found to be 7.5 kg for high stage (HS) cycle and 5.7 kg for low stage (LS) cycle. Once the full charge was determined, the cascade heat pump was evacuated and then the refrigerant charge to HS cycle was varied from −30% to +10% of full charge. The system was operated in the heating mode. Table 2 shows the test conditions to investigate the performance of the cascade heat pump with the variation of (HS) refrigerant charge amount.

Measured data from the cascade heat pump unit were collected using MX100 data logger of Yokogawa through all the measuring equipments specified above. The collected data was then transmitted from DAQ (Data Acquisition System) system onto a PC via TCP/IP protocol. The test data were recorded continuously for 40 min with 2 s interval. The heating capacity and COP were calculated from Eqs. (1) and (2), respectively. Based on the ASHRAE Guideline 2 [19], the uncertainty analysis of the system performance parameters such as heating capacity and COP was performed by using the Pythagorean summation of discrete uncertainties as shown in Eq. (3). The uncertainties of the heating capacity and the COP were approximately 3.1% and 3.3%, respectively.

$$\dot{Q} = \dot{m}C_p|\mathrm{LWT} - \mathrm{EWT}| \tag{1}$$

$$\mathrm{COP} = \frac{\dot{Q}}{W} \tag{2}$$

$$\mathrm{U} = \sqrt{\sum_{i=1}^{n}\left(\frac{U_i}{x_i}\right)^2} \tag{3}$$