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Effects of air flow maldistribution on refrigeration system dynamics of an air source heat pump chiller under frosting conditions

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Abstract

The effects of air flow maldistribution on the performance of an air source heat pump chiller under frosting conditions were investigated experimentally. The results indicated that air flow maldistribution was the dominant factor leading to hunting of the thermostatic expansion valve for medium and/or large size finned tube evaporators. With air flow maldistribution degree (AMD) increasing, frost occurred earlier, and the frost layer grew faster. The operating characteristics became lower when AMD was increased. We found such phenomenon seemed to be related to both the difference of refrigerant outlet superheat and the frosting velocity. In the hunting stage, the frost block effect became the main factor degrading the refrigeration system performance. With AMD increasing, the heat pump system pertinent performance data (suction pressure, evaporation temperature, discharge pressure, refrigerant outlet temperature, etc.) were degraded more dramatically.

Keywords: Heat pump; Air source; Frosting; Airflow maldistribution

1. Introduction

Air source heat pumps have been widely used in southern China. However, frost forms on the air cooled heat exchanger because of the cold and humid climate in many Chinese areas. As the frost grows, the performance of the unit degrades to such a degree that the frost has to be removed. Therefore, the characteristic study of frosting and defrosting and their pertinent influence factors on an air source heat pump is very important for the unit's safe and stable operation as well as improving the unit's performance [1]. For medium and/or large size air source heat pump, the air cooled heat exchanger has a large volume and many refrigerant circuitries. Such a heat exchanger's construction inevitably results in air flow maldistribution. Air flow maldistribution is one of the main factors affecting frosting. Therefore, air flow maldistribution has a great effect on the safe and stable operation of medium and/or large size air source heat pumps [2].

The study of air velocity maldistribution on the performance of heat exchangers has been conducted many years ago. Early in 1980, the effect of one dimensional air maldistribution on evaporator performance was studied theoretically by Thomas and Fagan [3]. In his analytical study of an evaporator with one refrigerant circuit, it was shown that capacity loss depends on the degree of deviation from uniformity as well as on the type of air distribution. For the worst case, the capacity degradation was as much as 20%. A plate fin air-to-refrigerant heat exchanger used as an evaporator was simulated by Domanski [4]. The simulation model accounts for non-uniform air distribution. Simulation results indicate that air maldistribution may induce an increase of refrigerant outlet superheat difference and maldistribution of a refrigerant, which leads to performance degradation of the evaporator. A steady state simulation model was established on an evaporator by Agenda et al. [5]. The refrigerants were controlled by one

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thermostatic expansion valve. According to his research, the refrigerant branch outlet may be in liquid state with the least air velocity, which will lead to hunting of the total refrigerant flow because of the thermostatic expansion valve adjustment. Therefore, the evaporator performance will be reduced ultimately, and the heating capacity can be reduced as much as 35%. The effect of non-uniform air flow distribution on the thermal performance of fin tube heat exchangers in typical outdoor units of residential air conditioning and heat pump systems was investigated numerically by Amaged [6]. In his simulation model, the heat exchanger is simplified as a porous plate. A simulation model was developed to predict the airflow distribution over the face of the heat exchanger. The impact of the air flow maldistribution on the air side average heat transfer coefficient and the pressure drop was studied. According to his study, the average mass pressure drop and the average heat transfer coefficient for the non-uniform flow was found to be 8% and 1.5%, respectively, more than for a uniform flow. A simulation model was developed by Chen Nan et al. [7] to predict the performance of a plate fin tube evaporator with different oblique angles between the inlet air velocity and frontal face of the evaporator under frosting conditions. He analyzed the effect of the oblique angle on the performance of the evaporator. His study shows that the oblique angle has effects on the evaporator parameters, refrigerating capacity, frost weight and air side pressure drop to different extents.

The above mentioned papers are limited to residential air to air heat pumps, and most papers used theoretical methods. Almost all the above papers' simulation models are steady state. However, to date, few papers of air flow maldistribution influence on medium and/or large size air to water heat pumps under frosting conditions are found by the authors. The reason may be that they have been widely used only during the past 10 years. It also may be due to the lack of the artificial test environment required for experimental research on medium and/or large size heat pump systems because the cost of an artificial test is too high.

The present study focused on the effects that air flow maldistribution have on the refrigerant system dynamic characteristics under frosting conditions. A unitary air to water heat pump with a nominal 50 kW cooling capacity was investigated experimentally. Because it was expensive to construct an artificial environment to meet the requirements of Chinese standard JB/T 4329–1997 (the counterpart of standard ARI 590–1998) completely for a heat pump of this size, the experiment was performed under natural conditions that approximated those required by the standard. This was achieved by proper selection of climate conditions in the Yellow River region on cold and humid winter nights. The working refrigerant was R22. This research will give a comprehensive understanding of frosting characteristics and defrosting methods that relate tightly to the frosting characteristics. Therefore, this study is meaningful to find more reasonable defrosting methods.

2. Experimental apparatus and method

2.1. Experimental setup

The test facility consisted of three major parts: the test air source heat pump, the cooling water system and the data acquisition system.

As shown in Fig. 1, the test heat pump was a refrigeration system with a 50 kW nominal cooling capacity, including an air side heat exchanger, a reciprocating compressor, a water side heat exchanger, a receiver, two thermostatic expansion valves, a reverse valve, an accumulator and two check valves. The air-to-water heat pump was installed in a test room of 5000 mm long, 5100 mm wide and 5500 mm high. The test room was open to the atmosphere. The air side heat exchanger consists of two air cooled coils. The two air cooled coils were placed in a V type position. Each coil was a four row, 12 circuit finned heat exchanger with 32 tubes per row. The finned tubes were made of smooth copper tubes and wavy aluminum fins. The tubes were vertical to the air flow. The tube had an outer diameter of 9.52 mm. The fin had a thickness of 0.14 mm and a gap of 2.1 mm between neighboring fins. The coil had a surface area of 1.0483 m^2. Two fans were installed on the top of the V type coils to pass 25,000 m^3/h of air across the coil. A drain pan was used to collect the water from the outdoor coil during defrosting. The water side heat exchanger was a plate heat exchanger. It had a surface area of 5.56 m^2 and 67 piece plates. A reciprocating compressor was used with the displacement of 76.37 m^3/h. The accumulator and the receiver could hold approximately 60 and 50% of the system charge, respectively. The two expansion devices were designed to be used for heating mode and cooling mode, respectively. Both expansion devices were nominal 15 ton external equalized thermal expansion valves that were designed to maintain 5.5 °C of superheat.

As denoted by the dashed arrow in Fig. 1, when the air source heat pump operated in the heating mode, the

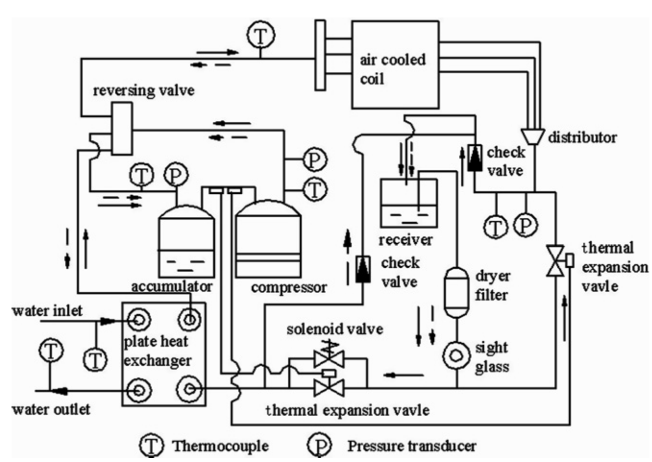

Fig. 1. Schematic diagram of the experimental setup.