The Influence of Heat Transfer Augmentor on the Performance of Window Air Conditioners

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Abstract. This study explored the effects of different designs of a heat transfer augmentor (HTA) on the performance of window air conditioners (WACs). A HTA was fabricated from perforated aluminum plates with different pore diameters according to the surface temperature distribution of the evaporator. The WAC was tested under the T1 condition of CNS14464, and the performance before and after the installation of the HTA was compared. The experimental results showed that the pore diameters of the HTA in proportion to the surface temperature of the evaporator could effectively improve the performance of the WAC. The dehumidification capacity and the energy efficiency ratio (EER) were increased by 22.05% and 3.49%, respectively, compared with the original WAC.

1. Introduction

Air conditioners have almost become home necessities for the pursuit of a comfortable life. In developed countries, 20% of total electricity is consumed for air conditioning [1]. Atmospheric temperature exhibits a gradual upward trend because of environmental changes and the greenhouse effect. A 1 °C increase in summer temperatures has been correlated with a 3.8% increase in peak demand load for air conditioning [2]. An energy efficiency ratio (EER) enhancement of 0.1 for an air conditioning system can save approximately 4% of power consumption. Therefore, the performance of existing air conditioners should be studied and improved to enhance their EER.

Air-cooled heat exchangers (ACHEs) with non-uniform air distribution reduce the effective area of heat exchange and thus lower the efficiency of the heat exchanger [3-8], prevent the normal operation of the thermostatic expansion valve of the evaporator [9], increase the air pressure loss, increase the fan power consumption, and reduce the average air velocity [7, 8]. All these factors affect the overall performance of air conditioners. Improving the air distribution unevenness mainly involves changing the piping configuration of the heat exchanger [10-13] and installing air guide plates [6]. These methods adjust the temperature distribution of the heat exchanger to match the uneven air distribution or improve the uniformity of the air distribution to improve the efficiency of ACHEs. These methods can effectively improve the performance of an ACHE or an air conditioner. However, they also have disadvantages such as the complexity of the modification procedure, high costs, and increased equipment volume. In this study, a heat transfer augmentor (HTA) with different pore diameters were installed directly on the return air (RA) side of the evaporator in a window air conditioner (WAC) to adjust the distribution relationship between the air flow and the surface temperature of the evaporator. This HTA improved the efficiency of the ACHE and the EER of the WAC without changing the original design to achieve the purpose of conserving energy.
2. Principle and Calculations

WACs use a vapor compression refrigeration system to achieve cooling. The enthalpies ($h_{ea,o}$ and $h_{ea,i}$) of the supply air (SA) and RA of the WAC can be obtained using the psychrometric chart from measurements such as the dry-bulb temperature ($T_{DB}$), wet-bulb temperature ($T_{WB}$), and relative humidity ($\psi$) of the SA and RA. The cooling capacity ($Q_{WAC}$) of the WAC is calculated as shown in Eq. 1. The mass flow rate ($m_{ea,o}$) of the SA can be converted using the volume flow rate ($G_{ea,o}$) with the air density ($\rho_{ea,o}$) of the SA in Eq. 2.

$$Q_{WAC} = m_{ea,o} \times (h_{ea,i} - h_{ea,o})$$  

$$m_{ea,o} = \rho_{ea,o} \times G_{ea,o}$$  

The dehumidification capacity of the WAC ($W_{WAC}$) can be obtained from the humidity ratios ($w_{ea,o}$ and $w_{ea,i}$) of the SA and RA. $W_{WAC}$ is calculated as shown in Eq. 3.

$$W_{WAC} = m_{ea,o} \times (w_{ea,i} - w_{ea,o})$$  

The EER of the WAC can be expressed as:

$$EER = \frac{Q_{WAC}}{P_{WAC}}$$  

Where $P_{WAC}$ is the total power consumption of the WAC and usually includes the power consumptions of the compressor, evaporator fan, condenser fan, and related electronic control elements.

3. Experimental Design and Procedure

3.1. Heat Transfer Augmentor Design

Fig. 1 shows a thermal image of the surface temperature of the evaporator of a R410A WAC (SRA-22b, SANYO, Taiwan) measured using an infrared thermal imager (Testo 881, Testo, Germany). The analysis software of the infrared thermal imager was used to determine temperatures at each end of the pipe, which are presented in Fig. 2. The evaporator of the WAC comprised the upper and lower components of a sub-evaporator, assembled in parallel to form a double-loop configuration. Fig. 2 reveals that the surface temperatures of the upper (loop 1, M6–M12) and lower (loop 2, M1–M5) evaporators tended to gradually rise from bottom to top. However, the lower evaporator was affected by the temperature of the upper evaporator, and the temperature of M5 on loop 2 decreased. In theory, the refrigerant passes through the capillary into the bottom of the evaporator to absorb heat through vaporization, and the temperature of the refrigerant gradually increases. Thus, the temperature of each evaporator circuit (loop 1 and loop 2) should gradually increase from bottom to top.

In this study, two HTAs (S1 and S2) were designed with perforated aluminum plates with pore diameters of 5 and 8 mm, respectively, and the HTA was installed between the evaporator and the filter of WAC (Fig. 3). A larger pore size reduces the air resistance, increasing the volume flow rate of air (G), and vice versa. Therefore, the design principles of the two HTAs were that the lower surface temperature of an evaporator corresponds to a larger G (S1) and that the lower surface temperature of an evaporator corresponds to a smaller G (S2). Each HTA was fabricated from four perforated aluminum plates with the same size and different pore diameters. The HTA was installed between the evaporator and the filter to reduce the contamination of the HTA and not introduce additional space requirements.

3.2. Test Standard and Method

The performance test of the WAC was based on the CNS14464 standard with T1 environmental conditions [14]. The $T_{DB}/T_{WB}$ was 35/24 ± 1 °C ($\psi \approx 40\%$ RH) and 27/19 ± 1 °C ($\psi \approx 47\%$ RH) for outdoors and indoors, respectively. In this study, the original WAC (control group) was denoted S0, and WACs with HTA (experimental group named St) were denoted S1 and S2. The thermocouples (T type: accuracy: ±0.75%) were installed on the SA and RA sides to measure the $T_{DB}$ and $T_{WB}$, the pressure sensors (JPT-131S, Jetec, Taiwan; accuracy: ±0.25%) were installed on the refrigerant system to measure the high and low side pressure, and the temperature and pressure data were recorded continuously by using a data logger (TRM-20, TOHO, Japan). $P_{WAC}$ was measured and recorded using a digital power meter (CW121, YOKOGAWA, Japan; accuracy: ±0.4%). The $G_{ea,o}$ of the WAC was
measured and calculated using an anemometer (Testo 445, Testo, Germany) with an air hood and its cross-sectional area. The $G_{a,o}$ for S0, S1, and S2 was 0.1145, 0.1093, and 0.1101 m$^3$/s, respectively. The $G_{a,o}$ of the S1 and S2 was reduced by approximately 4.54% and 3.84% to compare with the S0, respectively.

3.3. Data Analysis
The total time of each test was 2 hours, and the final hour of experimental data was averaged and analyzed. The data from the air side of the evaporator were analyzed using ASHRAE Psychrometric Analysis [15]. Then, the $Q_{WAC}$ and $W_{WAC}$ of the WAC were calculated using Eqs. 1 to 3 The EER of the WAC was then calculated using Eq. 4 with the $Q_{WAC}$ and the $P_{WAC}$ of the WAC. The test results of S1 and S2 in the experimental group (St) were compared with those of S0 to identify the percentage difference as shown in Eq. 5.

$$CS\% = [(St - S0)/S0] \times 100\%$$  

4. Results and Discussion
Fig. 4 shows the $Q_{WAC}$ values of S0, S1, and S2. The $Q_{WAC}$ of S1 was lower than that of S0, whereas the $Q_{WAC}$ of S2 was higher than that of S0. Compared with S0, the rate change in $Q_{WAC}$ for S1 and S2 was $-2.62\%$ and $2.98\%$, respectively. Although the G values of S1 and S2 were lower than 4.54% and 3.84% of that of S0, the configuration of S2 provided a higher enthalpy difference (total heat) between the SA and RA, and the $Q_{WAC}$ was improved. S1 greatly reduced G values, preventing improvements in the heat exchanger efficiency.

Figure 1. Infrared thermal image of the evaporator of an original WAC.

![Figure 1](image1)

Figure 2. Temperature of the measurement point for the evaporator of an original WAC.

![Figure 2](image2)

Figure 3. Design and installation of the HTA.

![Figure 3](image3)

Figure 4. Cooling capacity of the WAC with various experimental parameter configurations.

![Figure 4](image4)
condensation efficiency. Experimental results showed that the $W_{\text{WAC}}$ of S2 was better than that of S1; therefore, S2 should have an air distribution superior to that of S1. Fig. 6 shows the EER values of S0, S1, and S2. The installation of an HTA for the WAC had no significant effect on the $P_{\text{WAC}}$. Compared with S0, the rate change of EER for S1 and S2 was $-2.53\%$ and $3.49\%$, respectively. The EER incorporates both the $Q_{\text{WAC}}$ and the $P_{\text{WAC}}$ of the WAC and is an important index of system performance. The experimental results indicated that the HTA of S2 improved the performance of the WAC and was the optimal configuration in this study.

![Figure 5](image1.png)  
**Figure 5.** Dehumidification capacity of the WAC with various experimental parameter configurations.

![Figure 6](image2.png)  
**Figure 6.** Energy efficiency ratio of the WAC with various experimental parameter configurations.

5. Conclusions
In this study, a HTA for the performance improvement of WACs was successfully developed using perforated aluminum plates with different pore diameters. The HTA did not require changes to the original design of the evaporator of the WAC, was introduced at a very low cost, and increased the EER to achieve the purpose of energy conservation. The results showed that the configuration of S2 exhibited the optimal performance, and the $W_{\text{WAC}}$ and EER increased by 22.05\% and 3.49\%, respectively, compared with the original WAC. The main factor of performance enhancement of WAC was the high temperature difference between the heat transfer and the effective surface area of the evaporator caused by a more uniform air distribution.

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