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Improving energy efficiency of split-type air conditioners by cooling the cooling air

S. Theerakulpisut, Ph.D.
Associate Professor

S. Priprem, Ph.D.
Associate Professor
Energy Management and Conservation Office (EMCO),
Faculty of Engineering, Khon Kaen University.

Ruangrit Lammayot
Automechanics Division
Mahasarakham Technical College.

Abstract

This study deals with an experiment to improve the energy efficiency of small split-type air conditioners by cooling the condenser air using water condensate resulting from dehumidification of air in the evaporator. The condensate was sprayed into the condenser air stream of an experimental air conditioning unit via a nozzle and a small water pump driven by the condenser fan shaft. Energy consumption of the air conditoner was compared between two cases, namely running the air conditoner with and without condenser air cooling. It was found in this study that cooling the condenser air could reduce the power consumption by about 4%. It was also discovered that reducing the temperature of the condenser air by 1°C would reduce the energy consumption of the unit by 2.1%.

Introduction

For hot and humid countries such as Thailand comfort air conditioning is a common practice in houses, offices and commercial complexes. For houses and small offices, split-type air conditioners are most common. Air conditioning contributes very significantly to the energy bills of such establishments. Due to the limited supply of fossil fuels and the escalating prices of energy, attempts have been made to cut energy costs of air conditioning. Energy conservation measures such as regular maintenance of air conditioners, use of electronic thermostats and energy-efficient air conditioners, building insulation as well as energy-awareness campaign have been put to practices in Thailand, especially in government buildings.

This study aims to look into another possible way of reducing energy consumption of small air conditioning units by improving their energy efficiency. Since air which goes through the evaporator of the air conditioner is cooled and dehumidified, water condensate resulting from dehumidification of the air is collected in the water pan and discharged to the outside. The condensate can be used to cool the cooling air before entering the condenser. By this, the air conditioner is expected to operate with a lower condenser pressure and less power consumption.

Vapour compression refrigeration cycle

Most split-type air conditioning units employ the vapour compression refrigeration cycle as shown on the pressure-enthalpy diagram of Figure 1.

1-2 = compression process
2-3 = heat rejection process
3-4 = throttling process
4-1 = cooling and dehumidifying process

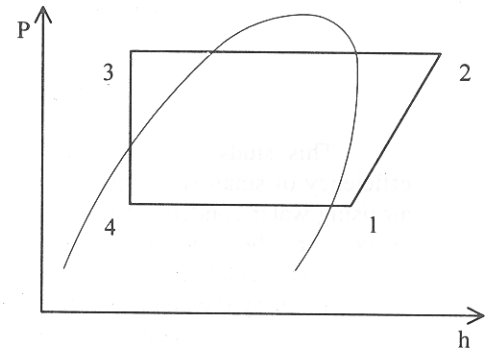

Figure 1 : simple vapour compression refrigeration cycle

From this figure it is seen that the difference between the refrigerant enthalpy of state 2 and state 1 represents the work input of the cycle. Obviously if the refrigerant pressure in the condenser is reduced, the cycle work input will be less. It is the objective of this study to investigate the potential energy savings resulting from the reduction of the pressure. One way to accomplish this is by pre-cooling the condenser air using the water condensate obtained from the cooling and dehumidifying process in the evaporator.

Note that the cycle of Figure 1 is an approximate one. The pressures in the condenser and the evaporator are assumed constant. For the purpose of this study, this simple vapour compression cycle is judged sufficient.

Energy efficiency ratio

The energy efficiency ratio (EER) is used in the air-conditioning industry to indicate the "efficiency" of a small split-type air conditioning unit. Normally, EER is determined at standard conditions and given as a rated value by the manufacturer. For air conditioners operating at other than the standard conditions, the values of EER may also be calculated. By definition, EER has the same meaning as the coefficient of performance (COP), except that COP is dimensionless and defined as,

$$COP = \frac{Q_e}{W} \quad (1)$$

where Q_e = cooling capacity of the evaporator (W)
W = power input of the system (W)

Since EER is expressed in the unit of Btu/h/W, Equation (1) can be rewritten to define EER as

$$EER = 3.4118 \frac{Q_e}{W} \quad (2)$$

The cooling rate of the evaporator can be calculated from

$$Q_e = m_{ae}(h_{aei} - h_{aeo}) \quad (3)$$

where m_{ae} = mass flow rate of air through the evaporator (kg/s)
h_{aei} = enthalpy of air at the inlet of the evaporator (J/kg)
h_{aeo} = enthalpy at the outlet of the evaporator (J/kg)

Note that Equation (3) is an approximate equation since the effect of water condensate which flows out of the system is not taken into account.

Enthalpy of air in Equation (3) can be obtained from ASHRAE (1993)

$$h = 1000\left[t + w(2501 + 1.805t)\right] \quad (4)$$

where t = air temperature (°C)
w = air humidity ratio (kg/kg dry air)
h = enthalpy of moist air (J/kg)

Calculation of h in Equation (4) requires t and w. The value of t can be measured but w cannot be directly measured. However, w may be calculated from the knowledge of relative humidity ϕ and t. The procedure for doing this follows.

$$ln\,P_{ws} = \frac{C_8}{T} + C_9 + C_{10}T + C_{11}T^2 + C_{12}T^3 + C_{13}\,ln\,T \qquad (5)$$

where P_{ws} = partial pressure of water vapour in saturated air (Pa)
T = air temperature (K)

The constants of Equation (5) are $C_8 = -5800.2206$, $C_9 = 1.3914993$, $C_{10} = -0.04860239$, $C_{11} = 4.17614768 \times 10^{-5}$, $C_{12} = -1.4452093 \times 10^{-8}$ and $C_{13} = 6.5459673$.

$$P_w = P_{ws}\phi \qquad (6)$$

Where P_w = partial pressure of water vapour in the air (Pa)
ϕ = relative humidity (dimensionless and decimal)

$$w = 0.62198\frac{P_w}{P - P_w} \qquad (7)$$

where P = atmospheric pressure (Pa)

It is clear that T gives the value of P_{ws} and ϕ will give P_w which will yield w. Similarly, the heat rejection rate at the condenser is given by

$$Q_c = m_{ac}\left(h_{aco} - h_{aci}\right) \qquad (8)$$

where Q_c = heat rejection rate at the condenser (W)
m_{ac} = air mass flow rate through the condenser (kg/s)
h_{aco} = air enthalpy at the condenser exit (J/kg)
h_{aci} i= air enthalpy at the condenser inlet (J/kg)

The values of air enthalpy in Equation (8) can be obtained by using Equation (4). Alternatively the heat rejection rate of the condenser can be written in terms of refrigerant enthalpy values as

$$Q_c = m_r\ \left(h_2 - h_3\right) \qquad (9)$$

where m_r = refrigerant mass flow rate (kg/s)
h_2 = refrigerant enthalpy at the condenser inlet (J/kg)
h_3 = refrigerant enthalpy at the condenser outlet (J/kg)

Similarly, the cooling effect of the evaporator is represented by

$$Q_e = m_r \ (h_1 - h_4) \tag{10}$$

where h_1 = refrigerant enthalpy at the evaporator outlet
h_4 = refrigerant enthalpy at the evaporator inlet

From Equation (9) and Equation (10), the following equation is obtained

$$\frac{Q_c}{h_2 - h_3} = \frac{Q_e}{h_1 - h_4} \tag{11}$$

Since h_3 is equal to h_4, Equation (11) becomes

$$h_3 = \frac{h_2 Q_e - h_1 Q_c}{Q_c - Q_e} \tag{12}$$

Note that Equation (12) is useful for determining h_3 if Q_e, Q_c, state 1 and state 2 are known.

Experimental air conditioning unit

A split-type air conditioning unit with a rated capacity of 12500 Btu/h was used in this study. The unit was modified as follows. Two Bourdon pressure gauges and two thermocouples were installed on the suction and discharge sides of the compressor. The pressures and temperatures were intended to determine state 1 and state 2 of Figure 1. A small water pump was installed on the same shaft of the condenser fan as shown in Figure 2. This pump was therefore driven by the condenser fan motor. The suction side of the pump was connected to the condensate pan of the evaporator via a flexible hosé whereas the discharge side of the pump was also connected to a flexible hose which was installed with a plastic nozzle at the exit. The size of the nozzle was selected so that it effectively delivered about 0.9 kg/h of water into the condenser air stream. The effective rate of water delivered was determined by calculating the increase in humidity ratio across the condenser. The rate of water flow ensured that there was sufficient water to continuously feed the condenser air stream during the experiment.

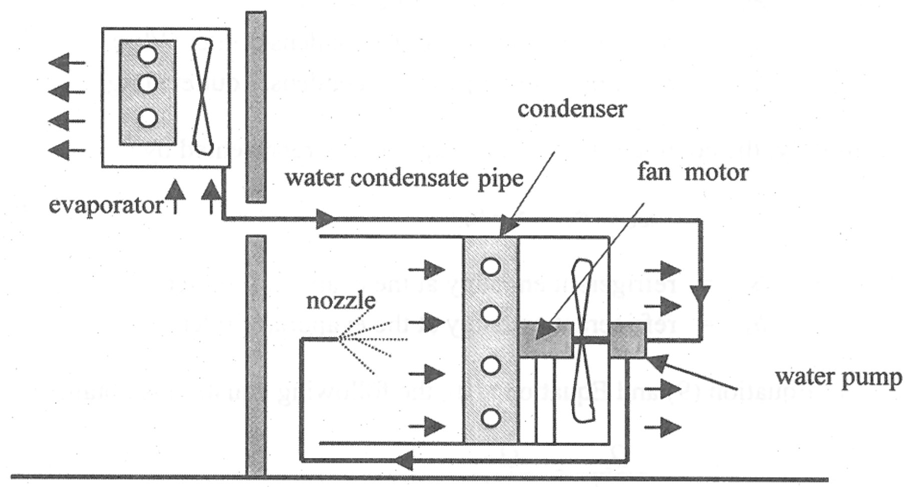

Figure 2 : Air conditioning experimental unit

Experimental procedure

For the purpose of analyzing the vapor compression cycle of Figure 1, the mass flow rates of air through the evaporator and the condenser must be known. Since, these parameters, for a given air conditioner unit, are constant, they need to be determined only once. A hot-wire anemometer was used to determine the velocity distributions over the inlet areas of the evaporator and the condenser. The velocity distributions were then integrated over the areas and multiplied by the density to give the respective air mass flow rates through the evaporator and the condenser.

The experimental air conditioning unit was turned on for about an hour to ensure that it operated steadily. Measurement of the power requirement of the system was then taken by a kilowatt-mater, pressure and temperature readings of the refrigerant at state 1 and state 2 were also taken. Temperature and relative humidity of the air entering and leaving the evaporator and the condenser were also measured. After this, water from the water pan of the evaporator was fed to the air stream at the inlet of the condenser via a nozzle. The experimental unit was left running until steady-state condition was reached. Measurements of the aforementioned parameters were then again taken. Note that steady-state conditions were observed when the pressures and temperatures at state 1 and state 2 did not change with time. Six test runs were conducted on six different days with ambient temperature ranging from 30°C to 35° C and relative humidity from 45% to 65%.

Experimental results

From the data obtained it was possible to plot state points of the refrigeration cycle before and after spraying water into the condenser air stream. Figure 3 represents typical cycles obtained from the experiment.

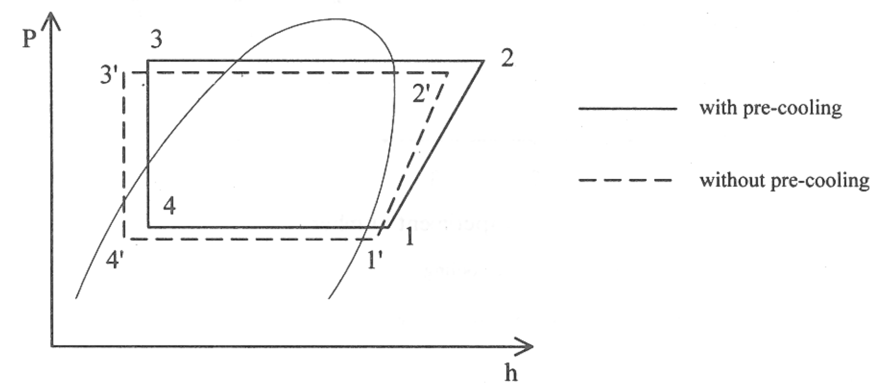

Figure 3 : Typical cycles with and without pre-cooling of condenser air

It should be noted that states 1 and 2 of the cycle 1-2-3-4 were determined by the corresponding measured pressures and temperatures. Evaluation of the cooling capacity of the evaporator coil by Equation (3) and the condenser heat rejection rate by Equation (8) permitted the determination of state 3 by Equaion (12). Once state 3 was known, state 4 was fixed since h_4 is equal to h_3. It should also be noted that constant pressures were assumed in the evaporator and the condenser. Similar procedures also apply in establishing the cycle 1′-2′-3′-4′.

From Figure 3, it was obvious that the effect of cooling the air stream before it entered the condenser depressed the refrigerant pressure in the condenser as well as the pressure in the evaporator with the pressure depression in the condenser being greater than that in the evaporator. Furthermore the cycle operated with slightly less degrees of superheat, higher degrees of subcooling and smaller refrigerant quality. It is also obvious from this figure that the power input of the cycle is also smaller than the case with no pre-cooling of the air stream.

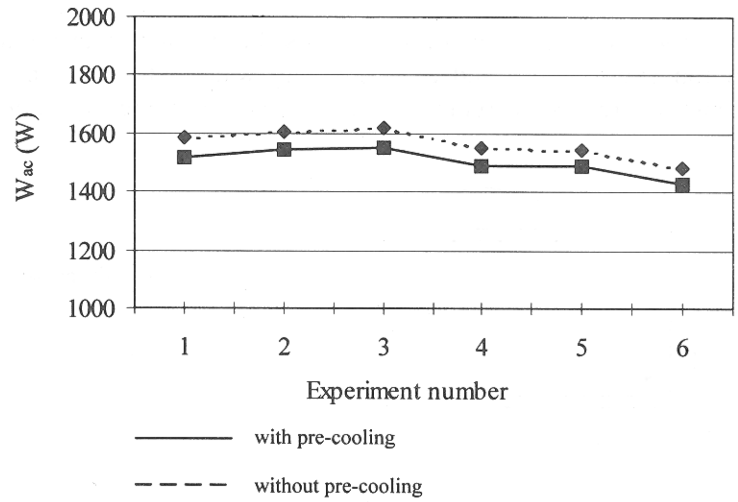

Figure 4 : System power consumption

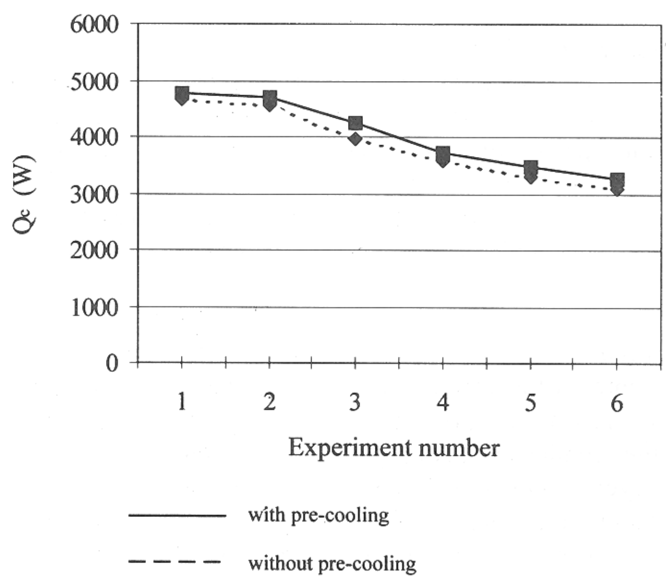

Figure 5 : Heat rejection rate at the condenser

Figures 4, 5 and 6 graphically illustrate the power consumption, the cooling rate of the evaporator and the rate of heat rejection at the condenser for six test runs conducted on six different days. Based an these experimental results, it can be concluded that the air conditioning experimental unit operated with less power consumption. The power consumption, on the average, was reduced by 61.65 W which was 3.94% of the power consumed with no pre-cooling of the condenser air. The average increase of the cooling capacity of the evaporator was also found to be 142.03 W or 4.3%. Furthermore, the rate of heat rejection at the condenser increased by 180.67 W or 4.7%. It should be mentioned, based on the experimental values of air temperature entering the condenser which are not presented here, it was observed that pre-cooling of the condenser air on the average reduced the temperature of the cooling air by 1.9°C. Therefore the reduction of the temperature of the condenser air by 1°C yielded energy savings of 2.1%.

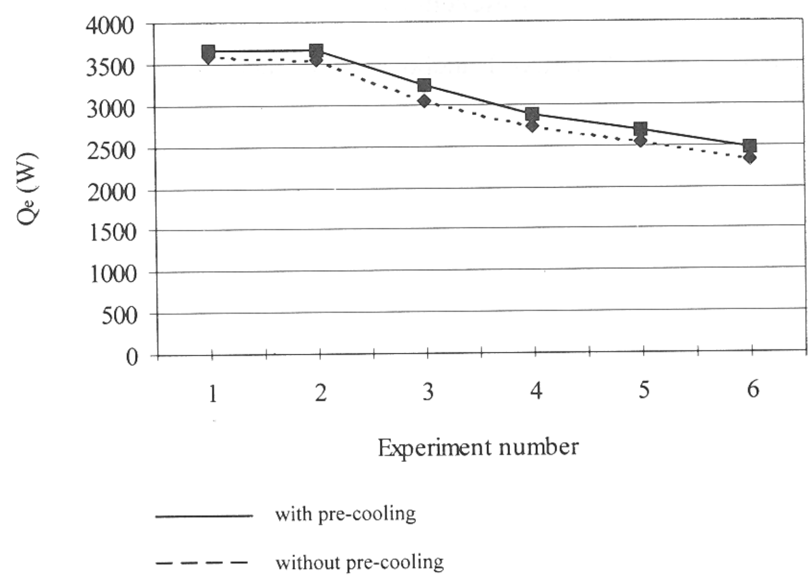

Figure 6 : Evaporator load.

Discussions and conclusions

This study was conducted to demonstrate that it is technically possible to save energy in air condition systems by reducing the temperature of the condenser air. The method employed in this study was to use the water condensate resulting from the dehumidification of the evaporator air which was otherwise wasted. Based on the results of this study it can be concluded that reduction of energy consumption in the order of 4% for such a small air conditioning unit as used in this study is possible. The results of this study also suggest that the condenser coil of the split-type air conditioning units should be placed in a cooler area outside the building. Approximately 2.1% of energy consumption can be reduced if the air surrounding the condenser is cooler than elsewhere by 1° C.

It should be noted that the findings of this study were based on a small air conditioning unit which was not tested throughout the year. However, it is believed that the findings should provide some useful indication as to how much energy can be saved when the condenser air is pre-cooled. It should also be pointed out that placement of the condenser should be in a cooler area whenever possible, since this will result in energy savings without much effort and cost.

References

1. ASHRAE (1993), ASHRAE Handbook (Fundamentals)