Experimental Study on Improving Coefficient of Performance for Split Air Conditioning System by Using an Innovative Separated– Vapor Device

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Experimental Study on Improving Coefficient of Performance for Split Air Conditioning System by Using an Innovative Separated–Vapor Device

MinhHung Doan *, TrongTuan NguyenTran, XuanVien Nguyen and Thanhtrung Dang

Abstract - ***This paper presented the results of improvement on Coefficient of Performance (COP) of split air–conditioner which has capacity of 9000 Btu/h. This air–conditioner uses R410A as the refrigerant. COP is improved by using an innovative separated–vapor device (flash chamber) in the system. The system is designed to operate in two cases. The first case, the saturated mixture of liquid–vapor refrigerant at low pressure passes through expansion valve (using capacity tube) and enters flash chamber. The second case is that it directly enters to evaporator. Both cases were experimented in the same heat load and environmental condition. The procedure of investigation is implemented in the same different temperature in outdoor and indoor. The results show that the power consumption reduces from 3.4 % to 4.4 % and actual coefficient of performance (COP) increase from 3.8 % to 8.7 %. Besides that, pressure drop on low pressure side for two cases were also presented. Experimental results were reported to show the feasibility of an innovative separated–vapor device which can be applied in air–conditioning systems.***

Keywords: ***split air–conditioner, heat exchanger, liquid–vapor, flash chamber, separated–vapor device***.

I. INTRODUCTION

One of the most important topics in this century is energy saving and environmental protection. There is extensive evidence of global–warming–related phenomena and energy shortages. We are facing with the increasingly serious challenges in energy security, environmental protection and greenhouse gas control. Thus, many methods of energy saving are being developed.

Many previous researchers studied on many advanced cycles to improve the energy efficiency, including organic flash cycle [1], auto–cascade cycle [2], flash gas bypass cycle [3]. Most of works in the past focused on investigating phase separation and improvement methods in various run type T–junctions. The effect of insert within a horizontal run type T–junction on the phase split was studied [4–8]. Depending on the approaching flow patterns and insert direction, inserts were seen either improve the partial phase separation or promote a more equal flow split between the two outlets. While, altering the angle of cut at the top of the insert from 30 ºC to 45 ºC had little to no effect on the flow split. An experimental study of vapor and liquid refrigerant separation in vertical impact T–junctions using R134a and R410A for application in but not limited to vapor compression systems was implemented [9]. The results showed that liquid separation efficiency depended on the flow pattern right above the impact region. The efficiency deteriorates dramatically when mist turns into churn flow regime, with increasing inlet flow rate and/or quality.

Shikazono studied a series of compact vapor–liquid separators for a R410A heat pump system based on conventional round tube heat exchangers, and claimed separation is done by surface tension. An efficient and compact flash gas tank used as a two–phase refrigerant separator is crucial for implementation of flash gas bypass into real air–conditioning systems [10].

II. EXPERIMENTAL

A. Experimental set up

This study was implemented on a conventional air conditioner model at Thermal–Refrigeration Workshop in HCMC University of Technology and Education. The model has an outdoor and an indoor unit with specification details in the Table 1.

TABLE 1. SPECIFICATIONS OF AIR–CONDITIONER

Model	**Refrigerant: R-410A**	
	Unit	***Values***
Indoor Unit		
Cooling Capacity	kW	2.65
Power Consumption	W	35
Dimensions (H x W x D)	mm	283 x 800 x 195
Outdoor Unit		
Power Consumption	W	784
Maximum Current	A	5.3
Dimension (H x W x D)	mm	418 x 695 x 244

The system was designed to operate in two cases, as shown in Fig 1.

- In first case (N–AC): Liquid refrigerant is throttled by capacity tube to become a mixture vapor–liquid at low pressure after that it enters the indoor unit.
- The second case (F–AC): The mixture of vapor–liquid refrigerant after the capacity tube enters the

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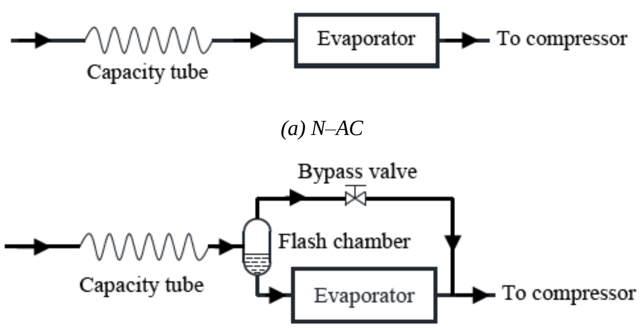
flash chamber where the amount of saturated vapor will be separated and bypassed to suction line of compressor while the residual gas will come indoor unit.



(a) N–AC

(b) F–AC

Figure 1. *Schematic of detail systems for: a) N–AC and b) F–AC*

Equation (1) shows a coefficient of performance (COP) calculation to compare the performance of F–AC and N–AC.

$$COP = \frac{Q_{indoor}}{P_{sys}} \quad (1)$$

Cooling capacity is determined by experimental data and defined as following:

$$Q_{indoor} = m.\Delta h \quad (2)$$

Where m is mass flow rate of the dry air on outlet of indoor unit, $m = A.\omega.\rho$ [kg/s] and Δh is different enthalpy of the air from inlet to outlet, $\Delta h = h_{air\ inlet} - h_{air\ outlet}$ [kJ/kg] and enthalpy of the air on inlet–outlet dependent their temperature and humidity. AC digital meter is shown in Table 2 that is used to determine power consumption of the system; it is calculated by Eq. (3)

$$P_{sys} = U.I.Cos\varphi \quad (3)$$

Where U is working voltage (V), I is running current (A) and $Cos\varphi$ is power factor.

B. Test procedure

To compare the capacity of flash chamber–split air conditioning system with conventional air conditioning system, Valves (V3, V4, V5) were installed with flash chamber as shown in Fig. 2.

In the first case, valve 3 and 4 were closed and valve 5 was opened, the saturated mixture of liquid–vapor refrigerant at low pressure entered to indoor unit. This operation is conventional air conditioner (N–AC) and the split air conditioner is being operated as this case.

In the second case, valve 3 and 4 were opened and valve 5 was closed. The mixture of vapor–liquid refrigerant after the capacity tube entered the flash chamber where the amount of saturated vapor will be separated and bypassed to suction line of compressor while the residual gas will come the indoor unit.

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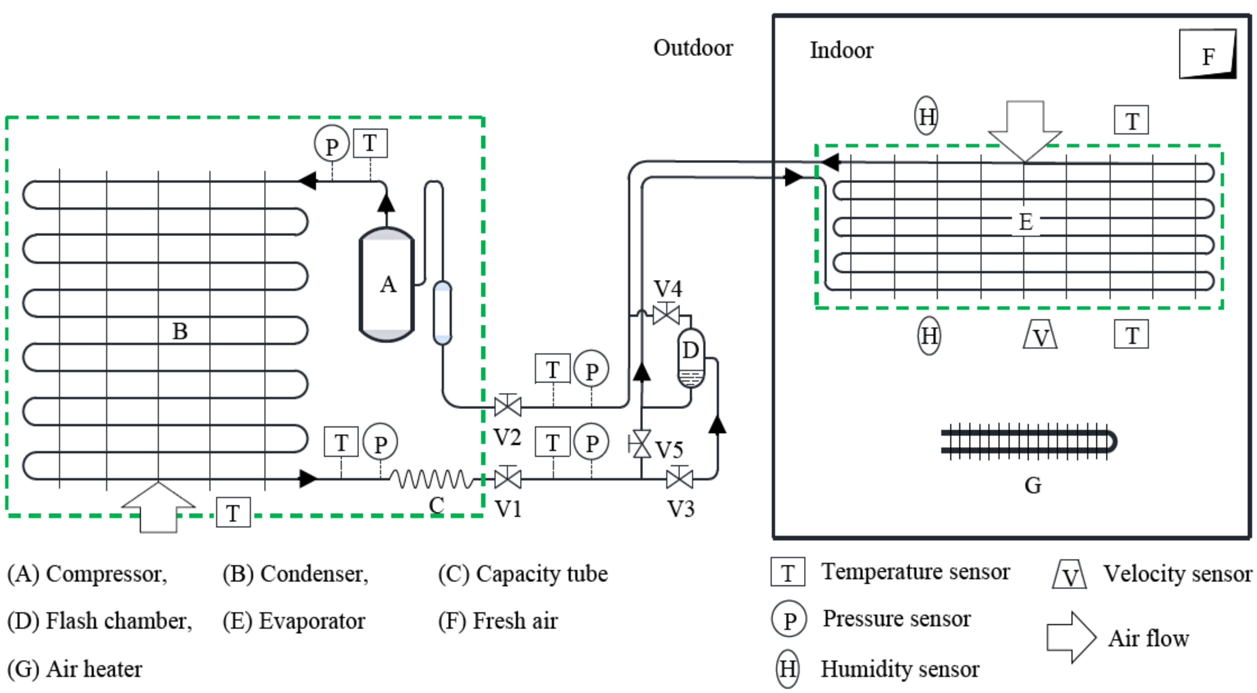

Figure 2. *Schematic of experimental facility*

Fig. 2 showed the position of measurement devices on the model. To determine the thermodynamic properties of refrigerant, pressure and temperature sensors were assembled at four node points. The ambient temperature was determined by temperature sensors which was assembled in condenser air–inlet. To determine evaporator capacity, a velocity sensor was assembled in evaporator air–inlet to specify air flow. Besides that, temperature and humidity detector was installed in both of air–inlet and return air to specify the different enthalpy of the evaporator air–inlet and outlet. The suction fan supplied fresh air which was air–heater, was designed to maintain thermal–humidity regime for inside room.

TABLE 2. MEASUREMENT PARAMETERS AND ACCURACY

Measurement	**Unit**	**Range**	**Accuracy**
AC digital multi–function meter: Peacefair, PZEM–021			
Power consumption	W	0.0–4500	1.0 grade
Current	A	0.00–20	1%
Voltage	V	80–260	1%
Apparatus Accuracy			
Temperature	^{0}C	-50–70	±1
Pressure	Bar	0–50	1%FC
Relative humidity	%RH	1.0– 99.0	±3% (30%-90%)
Velocity	m/s	0.0–45	±3% ± 0.3

The experiment system was designed with two different power sources. The first source was used for condenser and evaporator. The second source was used for measurement devices, suction fan, and air–heater. AC digital multi–function meter was installed to determine the first power source. It meant that only mete the air–conditioner power consumption to determine the power consumption of the system (P_{sys}).

As shown in Fig. 3, the experiment model was fabricated with dimensions of 3.8 × 1.6 × 1.8 m (L×W×H). Condenser was installed in outside of room. Evaporator was installed in inside of room. The room was built by wood to warrant strengthening. Both sides of wall surfaces of room -were assembled with PE–OPP insulation, a thickness of 10mm.

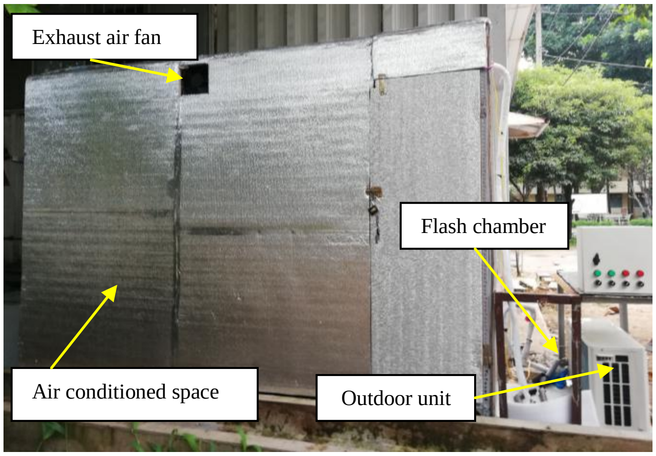

Figure 3. *Split Air Conditioning system for experiments*

This ensured heat and humidity insulation for space in room. The length of refrigerant pipe from condenser to evaporator is 5 m, the different height between condenser and evaporator is 1.7 m.

A suction fan supplied fresh air and 200 W heater to heat air in room to maintain thermal–humidity regime for inside room as real air–conditioning space. Flash chamber device was installed in outside of room at the condenser inlet and outlet.

III. RESULTS AND DISCUSSIONS

The model was independently operated in two case and were started when the temperature in the room and outdoor were equal. Experimental conditions for both of them were showed in the following:

- Outdoor room temperature: from 29 ºC to 32.5 ºC
- Different temperature from outdoor to indoor: from 6ºC to 9 ºC.
- Indoor room relative humidity: 61.9 % (min) to 65.7% (max).

A. *Refrigerant pressure drop*

Fig. 4 showed the relation of refrigerant pressure drop from throttle valve outlet to compressor suction that depended on the different room temperature and ambient in both cases (N–AC and F–AC). This pressure drop included of the refrigerant pressure drop on liquid pipe from throttle valve outlet to evaporator, the boiling refrigerant pressure drop in evaporator and from evaporator to compressor suction.

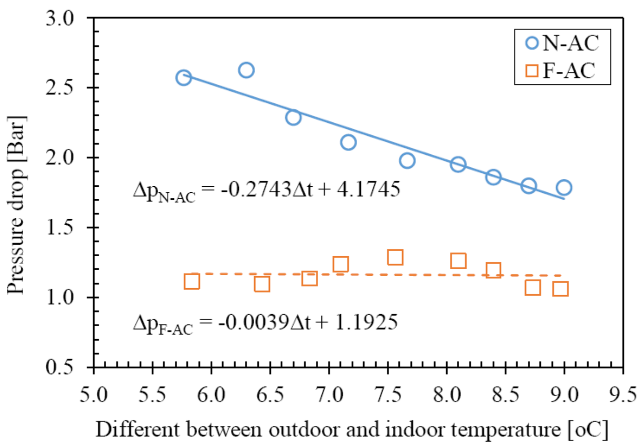

Figure 4. *Schematic* of *pressure drop*

The results presented the pressure drop of F–AC system is always lower than N–AC system. This was caused by amount of refrigerant which was vaporized after throttle valve and then returned to compressor suction directly. It was not through evaporator.

Experimental data also showed the different pressure in F–AC system. It changed from 1.1 to 1.3 bar while the different pressure in N–AC system changed from 1.8 to 2.6 bar.

Based on catalogue, standard parameter is at 35 ºC ambient temperature and 27 ºC room temperature. Thus, the different pressure drop in F–AC system is lower than N–AC system of 0.7 bar. This plays important role. The compression work decreases because the different pressure decreases. Actually, the power consumption is reduced.

B. Power consumption

AC digital meter was used to measure the system power consumption for both condenser and evaporator. The F–AC system power consumption is always lower N–AC system through $P_{N\text{-}AC} = f(\Delta t)$ and $P_{F\text{-}AC} = f(\Delta t)$ functions, as shown in Fig. 5.

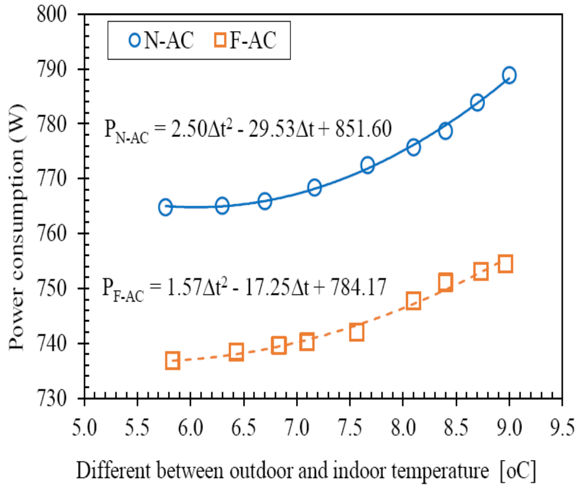

Figure 5. *Schematic of power consumption*

The experimental results showed that the F–AC system power consumption reduced from 3.4% to 4.4% compared with N–AC system.

C. Coefficient of Performance

The real COP was determined by Eq. (1). The cooling capacity was calculated by temperature, humidity and air velocity values of evaporator at the same time, the power consumption is determined as Fig. 5. The Fig. 6 showed the relation of COP following different room temperature condition. The COP of F–AC system is always higher N–AC system from 3.8 % to 8.7 %.

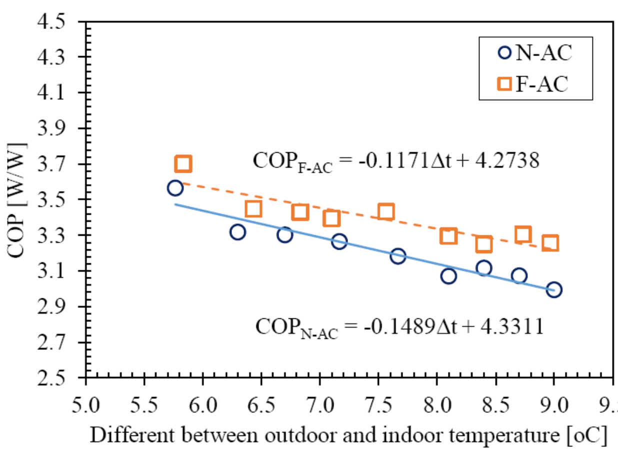

Figure 6. *Schematic of COP for N–AC and F–AC system*

In addition, the catalogue COP is 3.22 at ambient of 35°C dry bulb and 24 °C wet bulb temperature, the room condition is at 27 °C dry–bulb and 19 °C wet–bulb temperature. In this experiment, when the different ambient temperature is 8 °C, the COP of N–AC and F–AC are 3.072, 3.295, respectively. This meant the conventional air–conditioner with F–AC increased 7.3 % of COP at above condition.

IV. CONCLUSION

The experiment was tested on split air–conditioner with capacity of 9000 Btu/h. This air–conditioner uses R410A as the refrigerant. The test was implemented in two cases: F–AC and N–AC. This was implemented in ambient temperature condition from 29 °C to 32.5 °C and the different room temperature compared with ambient from 6 °C to 9 °C. The results showed several conclusions as below:

- System power consumption in F–AC reduced from 3.4% to 4.4% compare with N–AC.
- F-AC always get COP higher than N–AC from 3.8% to 8.7%.
- Refrigerant pressure drop at low pressure side (from behind throttling valve to the compressor suction) in F–AC is always lower than N–AC.

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