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The Study on Efficiency Improvement of Air Conditioning System by Using Condensate Water

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Abstract

This paper presents numerical and experimental study on efficiency improvement of split type air conditioning unit by using condensate water. Condensate water is fed through the annular ribbed tube double pipe heat exchanger (ADHX) in order to decrease the refrigerant temperature that exits from the condenser. Performances of the proposed system with ADHX and the conventional air conditioning unit without ADHX are tested and compared experimentally. Pull down times of both systems were obtained from transient state experiments. The indoor temperature was initially set at 32 °C while the pull down temperature and the outdoor temperature were set variously. The proposed system with ADHX yielded 22.55% faster pull down time compared with the conventional system when the outdoor temperature was 37 °C, and the indoor temperature was pulled down to 27 °C. COP of both systems were obtained from steady state experiments conducted at various values of indoor and outdoor temperatures. The best performance was found at the 27 °C indoor temperature and 37 °C outdoor temperature, with the COP of the proposed system being 1.5% higher than one of the conventional system.

Keywords: Split type air conditioning, COP, Condensate water, Double pipe heat exchanger.

Nomenclature

ACU	Air Conditioning Unit
ADHX	Annular Double pipe Heat Exchanger
ADP	Apparatus Dew Point
CDW	Condensate Water
COP	Coefficient of Performance
DHX	Double Pipe Heat Exchanger
IHX	Internal Heat Exchanger
VCS	Vapor Compression System

1. Introduction

Air conditioning unit (ACU) is now one of the must-have household electrical appliances due to increasing of world temperature. Since ACU is the household appliance that consumed most power, therefore ACU performance improvement has always been one of the important research topics. This paper presents the study on ACU performance improvement by using condensate water. Condensate water is fed through the annular ribbed tube double pipe heat exchanger (ADHX) in order to decrease the refrigerant temperature that exits from the condense, hence the range of refrigerant subcooling could be increased. Both transient state and steady state experiments are conducted in order to compare the performances of the proposed ACU system with ADHX and the conventional system without ADHX. The performance of the proposed system is also numerically computed and the results are verified with ones obtained from experiments.

Farayedhi et al. [1] analyzed the amount of condensate water created by a 1.5 ton ACU and stated that condensate water was affected by relative humidity. Dusan et al. [2] utilized condensate water to pre-cool fresh air and found that power consumption was decreased approximately 10%. Khan et al. [3] utilized condensate water to pre-cool the condenser coil of a 2 ton ACU and found that power input was decreased by 18%. Pottker et al. [4] studied the effect of condenser subcooling on the performance of vapor compression system (VCS) comparing the system with and without internal heat exchanger. The result showed that condenser subcooling increased the system COP. Pottker et al [5] studied the effects of condenser subcooling of VCS operated with R1234yf, R410A, R134a and R717. The VCS operated with R1234yf achieved the maximum COP improvement of 8.4 %. Linton et al. [6] studied the effect of condenser subcooling of VCS operated with R134a, R12 and R152a. The temperature of condenser subcooling was adjusted at 6-18 °C. The result showed that increasing of condenser subcooling improved COP, VCS operated with R134a achieved the maximum COP improvement of 12.5%. Kang et al. [7] investigated the effect of refrigerant flash at the expansion device inlet of ACU, and found that the high value of flash gas ratio would decrease the system COP as high as 36%. Pachegaonkar et al. [8] studied the performance of double pipe heat exchanger (DHX) with annular twisted tape insert at 45° and 60° angle, and compared with smooth tube DHX. It was found that DHX with 45° insert gave the best performance due to high increase heat transfer and less loss pressure drop. Tumna et al. [9] showed the heat transfer on laminar flow in an isothermal circular tube with helical tape inserts was at higher rate compared with the smooth tube. Li et al. [10] and Mohammet et al. [11] showed that the heat transfer of internal ribbed tube was at higher rate compared with the smooth tube.

2. Theoretical Model

2.1 COP Calculation

2.1.1 Numerical Method

To find out COP numerically, firstly find out the temperature of refrigerant subcooling by numerical finite volume method as will be explained in chapter 2.4. Substitute the temperature of refrigerant subcooling in P-h diagram, and calculate for the value of COP.

The cycle of vapor compression system can be expressed on the P-h diagram as shown in figure 1.

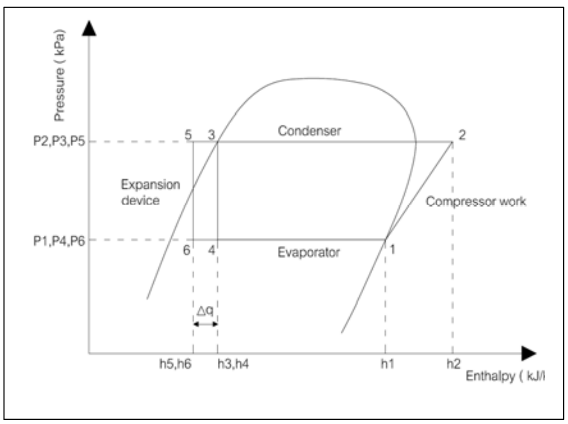

Fig. 1 – P-h diagram showing VCS cycle.

The cycle consists of processes of four components: compressor, condenser, expansion device and evaporator. The conventional VCS cycle has the state change from 1→2→3→4→1 as shown in figure 2.

Four state operations of the cycle are as following;

1) Refrigerating effect by $h_1 - h_4$.
2) Compressor work by $h_2 - h_1$.
3) System heat rejection by $h_2 - h_3$.
4) Expansion process by $h_3 - h_4$.

COP of conventional ACU can be calculated as following.

$$COP = \frac{Q_e}{W_c} = \frac{h_1 - h_4}{h_2 - h_1} \tag{1}$$

COP is coefficient of performance.
Q_e is cooling capacity (W).
W_c is power input of compressor (W).

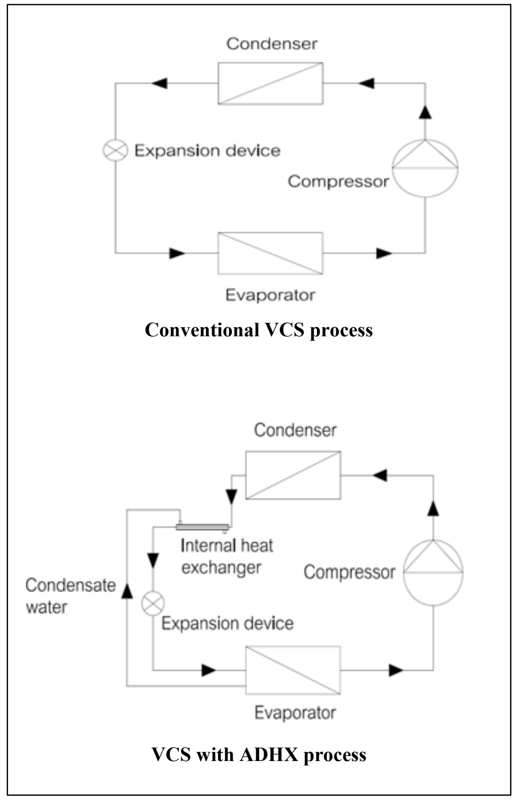

Fig. 2 – Process of VCS with and without ADHX.

The annular ribbed tube double pipe heat exchanger (ADHX) is installed between the condenser and expansion device in the proposed ACU (figure 2). ADHX allow the heat transfer between the condensate water from the evaporator and the saturated refrigerant from the condenser. The saturated refrigerant in the proposed ACU with ADHX would then be more subcooled than conventional ACU. The subcooling process of the refrigerant is shown in the P-h diagram as 3→5 (figure 1). Refrigerating effect is also increased because of lower evaporator inlet enthalpy as seen in the P-h diagram as 6→1 instead of point 4→1.

COP of ACU with ADHX can be calculated as following.

$$COP' = \frac{Q_e}{W_c} = \frac{h_1 - h_6}{h_2 - h_1} \tag{2}$$

2.1.2 Experimental Method

To find out COP experimentally, firstly cooling capacity is calculated from equation3. The calculation is based on the document Thai Industrial Standard 1155 (TIS 1155).

$$qtc_i = \sum E_r + (h_{w1} - h_{w2})W_r + q_p + q_r \qquad (3)$$

qtc_i is cooling capacity of ACU in indoor room (W).
$\sum E_r$ is total power input of equipment in indoor room (W).
h_{w1} is enthalpy of water which used to increase air - humidity (J/kg).
h_{w2} is enthalpy of condensate water (J/kg).
W_r is amount of condensate water (kg/s).
q_p is heat transferred from outdoor room through the wall into indoor room (W).
q_r is heat transferred from surroundings through the wall and ceiling into indoor room (W).

COP can be carried out as equation 4.

$$COP = \frac{q_{tci}}{E} \qquad (4)$$

E is power input of air conditioning (W).

2.2 Condensate water

There are two types of heat loads, sensible heat and latent heat. To reduce temperature in the room we need to reject heats to surrounding. Sensible heat is rejected during air flows through the evaporator, only reduction of air temperature is observed. Latent heat deals with water vapor in the air which is affected from many sources, i.e. human activity, moisture from surrounding, etc. Vapor is rejected when air flows through the evaporator. When air contacts to coil surface whose temperature is lower than air dew point temperature, the water vapor in air condenses. Condensate water can be calculated according to psychrometric chart (figure 3) with the equation 5.

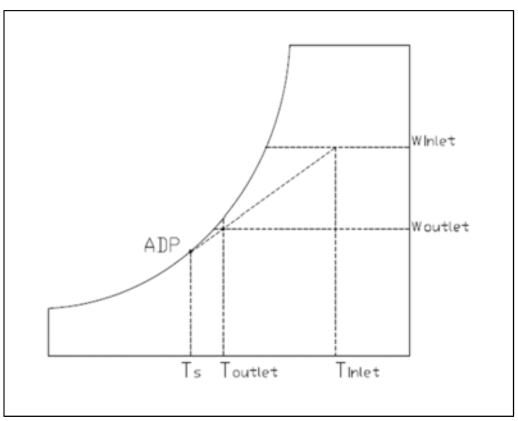

Fig. 3 – Condensation process on psychrometric chart.

Amount of condensate water can be calculated as equation 5 below.

$$M_W = \frac{CFM \times 60}{S_P V_i} \times \frac{W_i - W_o}{7000} \qquad (5)$$

M_W is amount of condensate water (lb/hr).
CFM is air volume flow rate (ft^3/min).
$S_P V_i$ is inlet air specific volume (ft3/lb).
W_i is inlet air specific humidity (gr.w/lb d.a).
W_o is outlet air specific humidity (gr.w/lb d.a).
7000 is derivation of unit changing.

2.3 Annular ribbed tube double pipe heat exchanger (ADHX)

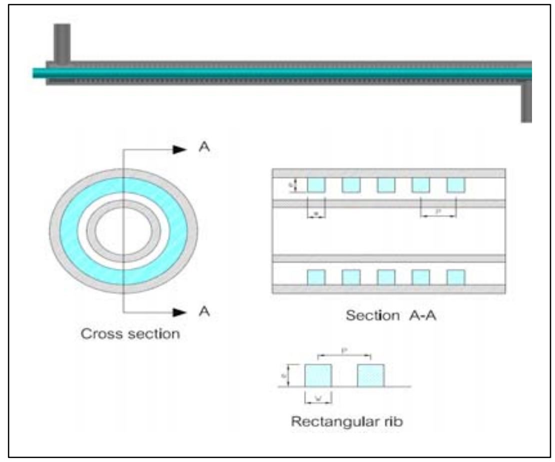

Fig. 4 – ADHX cross section.

This paper studies the capability of double pipe heat exchanger with annular ribbed tube (figure 4) on heat transfer between condensate water and refrigerant. Mohammet [11] have compared the performances of various-dimensioned copper ribbed tubes. It was reported that decreasing of pitch distance led to increasing of heat transfer rate, and increasing of pitch depth led to increasing of heat transfer performance. Decreasing of pitch distance was found to be more effective than reducing of pitch depth. Finite volume numerical was conducted on the design of ADHX in this study. The performances of various size of ADHX were computed and compared. Table 1 shows the size of the ADHX selected from the computation results based on the compromise of the heat transfer rate and the size compactness. Since the evaporator of the experiment setup located above the condenser and the ADHX is installed between these two components closed to the condenser, therefore no pump is needed for the condensate water.

Table 1 – Geometry of annular ribbed tube.

Parameter	Dimension
1. Inner and outer of ribbed DHX	6.35/12.07 mm.
2. Length of ribbed DHX, l	300 mm.
3. Height of ribbed DHX, h	1.5 mm.
4. Ribbed root width, w	1.5 mm.
5. Ribbed pitch, p	3 mm.
6. Pitch ratio, PR	0.24
7. Blockage ratio, BR	0.12

2.4 Simulation by CFD

Computational Fluid Dynamic (CFD), ANSYS.16, was used to solve and analyze for fluid flow. This paper use finite volume method to calculate fluid heat transfer. There are three basic equations leading to find out temperature difference: the continuity, the Navier-Stokes, and the energy equations.

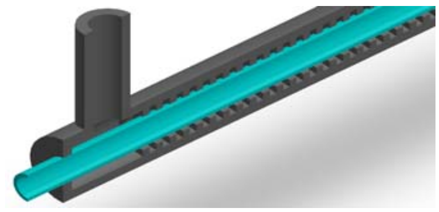

Fig. 5 – Model of ADHX.

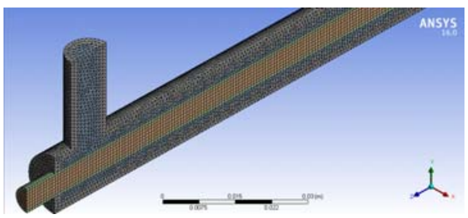

Fig. 6 – Grid system for ADHX.

The continuity equation is described as following.

$$\frac{\partial U}{\partial x}+\frac{\partial V}{\partial x}=0 \qquad (6)$$

The momentum equation is described as following.

$$\rho\left(\mu\frac{\partial \mu}{\partial x}+v\frac{\partial v}{\partial x}\right)=\rho g-\frac{\partial p}{\partial x}+u\frac{\partial 2y}{\partial x2} \qquad (7)$$

The energy equation is described as following.

$$\rho C_p\left(\mu\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\mathrm{k}\frac{\partial^2 T}{\partial y^2} \qquad (8)$$

The standard $(k-\varepsilon)$ model was used to calculate for Reynolds Stress τ (equation 9) in order to cope with turbulence caused by annular ribbed tube.

$$\tau=\frac{k}{\varepsilon} \qquad (9)$$

k (turbulent kinetic energy) = $\frac{3}{2}(u.I)^2$

ε (turbulent dissipation rate) = $C_\mu^{1/4}\frac{k^{3/2}}{L}$

I (turbulent intensity) = $0.16xR_e^{1/8}$

u (mean flow velocity).

3. Experiment

3.1 Experimental Set up and Tools

1.5 ton VCS experimental setup is used to compare the performance of the conventional ACU without ADHX and the proposed ACU with ADHX. The experiment was set up in calibrated room type calorimeter which consists of two testing rooms: simulated outdoor room and simulated indoor room. The dimensions of testing rooms are 5.45L x4.2D x 2.2H m^3 for simulated outdoor room and 3.75L x 4.2D x 2.2H m^3 for simulated indoor room. Table 2 shows the list of equipment and tools installed in simulated outdoor and simulated indoor rooms. All the measurements were fed to the desktop computer via data acquisition system.

Table 2 – Equipment list of testing room.

Outdoor room	Indoor room
1. condensing unit	1. fan coil unit
2. Refrigeration unit	2. thermocouple type K
3. thermocouple type K	3. Boiler and dry heater
4. Boiler and dry heater	4. pressure transmitter
5. pressure transmitter	5. circulate fan
6. annular double pipe with ribbed tube heat exchanger (ADHX) by 300 mm. of length	
7. circulate fan	
8. cooler tank	
9. Water flow meter	

Performances of the conventional ACU and the proposed ACU are tested and compared under both steady state and transient state tests. COP of both systems are obtained under steady state tests. Table 3 shows the steady state test conditions. The proposed ACU makes use of condensate water which is usually drained out as waste. It was found from experiments that condensate water was obtained at an average rate of 1.6 L/h from an ordinary 1.5 ton ACU. The condensate water was mixed with makeup water in an insulate container. The volume and temperature of mixing water is 12.6 Liters and 25 °C, respectively. Therefore, for the steady state tests, the water inlet condition into ADHX were set at 25 °C at a flow rate of 12.6 L/h. The 300 mm ADH was installed at the exit of the condenser coil and wrapped with insulation to prevent heat loss (figure 7).

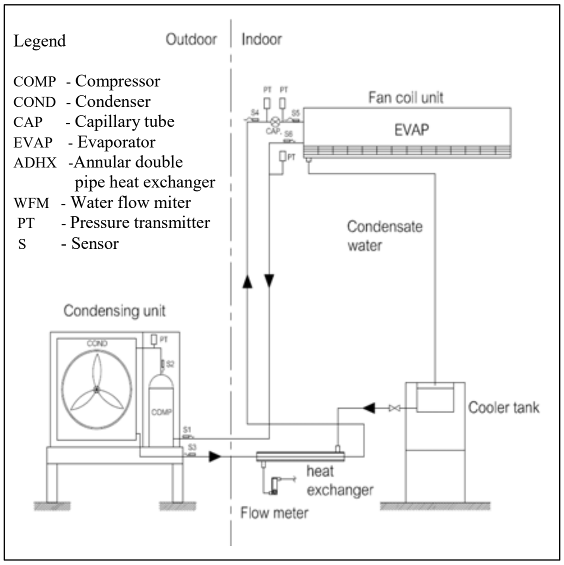

Figure 7 – Schematic of experiment.

Table 3 – Testing steady state condition.

Case	Indoor room		Outdoor room	
	TDB (°C)	TWB (°C)	TDB (°C)	TWB (°C)
1	27	19	35	24
2	27	20	30	23
3	27	20	33	26
4	27	20	35	28
5	27	20	37	30
6	25	18.7	30	23
7	25	18.7	33	26
8	25	18.7	35	28
9	25	18.7	37	30
10	23	17	30	23
11	23	17	33	26
12	23	17	35	28
13	23	17	37	30

For transient state tests, indoor room is initially set at32 °CDB and 25 °CWB for all the tests. The pull down time that each system spends in order that the air is pulled down to the final condition of indoor room is recorded. Table 4 shows the transient state test conditions. Under this initial transient state test condition, condensate water obtained from 1.5 ton ACU could be calculated (equation 5) to be 2.5 L/h at 15 °C. Therefore for the transient state tests, the condensate water is set at 15 °C and 2.5 L/h flow rate. Heat load is constantly set at 1000 watt. Table 5 explains the test procedure for both steady state and transient state tests.

Table 4 – Test condition on transient state.

Case	Initial condition of indoor room		Final condition of indoor room		Outdoor room	
	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)
1	32	25	27	20	33	26
2	32	25	27	20	35	28
3	32	25	27	20	37	30
4	32	25	25	18.7	33	26
5	32	25	25	18.7	35	28
6	32	25	25	18.7	37	30
7	32	25	23	17	33	26
8	32	25	23	17	35	28
9	32	25	23	17	37	30

Table 5 – Experimental procedure**.**

State	Processing
Steady State	1) operate air conditioning *without ADHX* with conditions in table 3 and record data one hour after both rooms are on stable temperature
	2) operate air conditioning *with ADHX* with conditions in table 3 and record data one hour after both rooms are on stable temperature
Transient State	3) operate air conditioning *without ADHX* , record data during starting until gets set point temperature as shown in table 4
	4) operate air conditioning *with ADHX* , record data during starting until gets set point temperature as shown in table 4

4. Result

4.1 Experimental Result

The steady state tests showed that ACU with ADHX could increase the range of refrigerant subcooling which leads to increasing of refrigerating effect and the reduction of ACU power input as shown in table 6. The best performance was found at the test condition of 27 °C indoor and 37 °C outdoor at which the power input was reduced by 1.5% and COP was increased by 1.5% (table 7).

Increasing of subcooling range leads to increasing of refrigerating effect, therefore the pull down time could be decreased. Table 8 shows the comparison of pull down period obtained from the transient state tests. The best percentage reduction of pull down period was found at the test condition of 37 °CDB outdoor and the final indoor temperature of 27 °CDB

Table 6 – ACU power input obtained from steady state tests.

Case	Indoor room		Outdoor room		W/O ADHX	With ADHX	Decresing of power input
	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)	Watts	Watts	(%)
1	27	19	35	24	1532	1516	1.044
2	27	20	30	23	1380	1373	0.507
3	27	20	33	26	1460	1448	0.822
4	27	20	35	28	1532	1516	1.044
5	27	20	37	30	1568	1544	1.531
6	25	18.7	30	23	1372	1365	0.510
7	25	18.7	33	26	1452	1440	0.826
8	25	18.7	35	28	1530	1516	0.915
9	25	18.7	37	30	1564	1544	1.279
10	23	17	30	23	1376	1370	0.436
11	23	17	33	26	1450	1438	0.828
12	23	17	35	28	1524	1508	1.050
13	23	17	37	30	1560	1540	1.282

Table 7 – COP obtained from steady state tests.

Case	Indoor room		Outdoor room		Experiment COP		Increased COP
	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)	W/O ADHX	With ADHX	(%)
1	27	19	35	24	3.560	3.607	1.32
2	27	20	30	23	3.948	3.968	0.50
3	27	20	33	26	3.717	3.758	1.10
4	27	20	35	28	3.560	3.607	1.32
5	27	20	37	30	3.477	3.531	1.55
6	25	18.7	30	23	3.719	3.736	0.46
7	25	18.7	33	26	3.517	3.546	0.82
8	25	18.7	35	28	3.338	3.368	0.90
9	25	18.7	37	30	3.266	3.309	1.31
10	23	17	30	23	3.482	3.497	0.43
11	23	17	33	26	3.267	3.295	0.86
12	23	17	35	28	3.139	3.172	1.05
13	23	17	37	30	3.066	3.106	1.30

Table 8 – Pull down period obtained from transient state tests.

Case	Initial condi-tion of indoor room		Final condition of indoor room		Outdoor room		Pull down time (minute-second)		
	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)	w/o ADHX	With ADHX	Saving time (%)
1	32	25	27	20	33	26	7:25	6:05	17.98
2	32	25	27	20	35	28	8:15	6:35	20.20
3	32	25	27	20	37	30	8:30	6:35	22.55
4	32	25	25	18.7	33	26	11:55	10:15	13.99
5	32	25	25	18.7	35	28	13:00	11:10	14.10
6	32	25	25	18.7	37	30	13:05	10:50	17.20
7	32	25	23	17	33	26	17:55	16:25	8.37
8	32	25	23	17	35	28	19:30	17:50	8.55
9	32	25	23	17	37	30	19:50	17:40	10.92

4.2 Comparative result between experiment and simulation

Values of degree of subcooling obtained from experiments and numerical computation were found to be in the same trend as shown in table 9. The most increasing of subcooling obtained from experiments and computation were found at the same test condition of 27 °C indoor room and 37 °C outdoor. The reason is the temperature difference between the inlet refrigerant and the compensate water is highest at this test condition.

Table 9 – Degree of subcooling obtained from experiments and computation.

Case	Indoor room		Outdoor room		Mass flow rate (Kg/h)	Water inlet temp(°C)	Experiment	Simulation
	T_{DB} (°C)	T_{WB} (°C)	T_{DB} (°C)	T_{WB} (°C)			Subcooling degree(°C)	
1	27	19	35	24	12.6	25	1.7	1.77
2	27	20	30	23	12.6	25	0.93	0.99
3	27	20	33	26	12.6	25	1.42	1.50
4	27	20	35	28	12.6	25	1.70	1.77
5	27	20	37	30	12.6	25	2.10	2.19
6	25	18.7	30	23	12.6	25	0.92	0.97
7	25	18.7	33	26	12.6	25	1.45	1.51
8	25	18.7	35	28	12.6	25	1.65	1.81
9	25	18.7	37	30	12.6	25	1.95	2.19
10	23	17	30	23	12.6	25	0.80	0.90
11	23	17	33	26	12.6	25	1.31	1.43
12	23	17	35	28	12.6	25	1.58	1.73
13	23	17	37	30	12.6	25	1.92	2.10

The value of degree if subcooling of each test condition was then substituted into the P-h diagram of R22, and COP of each test condition was calculated according to equation 1. The values input power reduction and COP obtained from computation are compared with ones from the experiments and shown in tables 10 and 11. The difference of the power input reduction and COP of values obtained from experiments and computation of all cases are below 10%.

Table 10 – ACU power input obtained from experiments and computation.

Case	Indoor room		Outdoor room		Experiment	Simulation	Difference of power input
	TDB (°C)	TWB (°C)	TDB (°C)	TWB (°C)	Watts	Watts	(%)
1	27	19	35	24	1516	1435	5.37
2	27	20	30	23	1373	1290	6.03
3	27	20	33	26	1448	1366	5.64
4	27	20	35	28	1516	1435	5.37
5	27	20	37	30	1544	1468	4.94
6	25	18.7	30	23	1365	1262	7.52
7	25	18.7	33	26	1440	1331	7.60
8	25	18.7	35	28	1516	1419	6.37
9	25	18.7	37	30	1544	1449	6.14
10	23	17	30	23	1370	1244	9.17
11	23	17	33	26	1438	1330	7.52
12	23	17	35	28	1508	1404	6.90
13	23	17	37	30	1540	1416	8.04

Table 11 – COP obtained from experiments and computation.

Case	Indoor room		Outdoor room		COP		Difference of COP
	TDB (°C)	TWB (°C)	TDB (°C)	TWB (°C)	Experiment	Simulation	(%)
1	27	19	35	24	3.607	3.774	4.62
2	27	20	30	23	3.968	4.252	7.17
3	27	20	33	26	3.758	4.023	7.04
4	27	20	35	28	3.607	3.774	4.62
5	27	20	37	30	3.531	3.697	4.70
6	25	18.7	30	23	3.736	4.213	7.67
7	25	18.7	33	26	3.546	4.000	7.63
8	25	18.7	35	28	3.368	3.817	6.22
9	25	18.7	37	30	3.309	3.578	5.89
10	23	17	30	23	3.497	3.815	9.11
11	23	17	33	26	3.295	3.570	8.35
12	23	17	35	28	3.172	3.382	6.61
13	23	17	37	30	3.106	3.353	7.94

5. Conclusion

Experimental and numerical studies showed that the proposed ACU using condensate water with ADHX could increase the degree of refrigerant subcooling, hence increases COP. In transient state test, the proposed ACU with ADHX could decrease pull down time with the highest reduction rate of 22.59% at the test condition of 27 °CDB indoor and 37 °CDB outdoor. In steady state test, the proposed ACU with ADHX achieved better COP. The highest increase of COP of 1.5% was found at the test condition of 27 °CDB indoor and 37 °CDB outdoor. Numerical computation results of the degree of refrigerant subcooling, the input power and COP were all agreed with the experimental results. The difference between the results obtained from computation and experiments are below 10% of all cases.

6. References

[1] Farayedhi,A., Ibrahim,N. and Gandhiasan.P. (2014). Condensate as a water source from vapor compression system and humid regions, *Desalination*, vol.349, July 2014, pp. 60 – 67.

[2] Licina,D. and Sekhar,C. (2012). Energy and water conservation from air handling condensate in hot and humid climates, *Energy and buildings*, vol.45, 2012, pp. 257-263.

[3] Khan,S.,Wang,F., Zubaidy,A. and Sarim (2014). Energy recovery Through the Condensate of air conditioning machine's condensate, *International Journal of Engineering Research & Technology(IJERT)*, vol.3, June 2014, pp. 547 – 555.

[4] Pottker,G. and Hrnjak,P (2015). Effect of the condenser subcooling on the performance of vapor compression systems, *International Journal of refrigeration* , vol.50, November 2015, pp. 156 – 164.

[5] Pottker,G. and Hrnjak,P (2015). Experimental investigation of the effect of condenser subcooling in R134a and R1234yf air-conditioning systems with and without internal heat exchanger, *International Journal of refrigeration* , vol.50, October 2015, pp. 104 – 113.

[6] Linton,W., Snelson,K. and Hearty.F. (1992). Effect of condenser liquid subcooling on system performance for refrigerants, *ASHRAE Trans*, vol.98, July 1992, pp. 140-146.

[7] Kang,H., Joo,Y. and Kim.Y. (2008). Effect of flash gas generation at the expansion device inlet on the dynamic characteristic of a refrigeration system, *International Journal of refrigeration*, vol.31, August 2008, pp. 396-403.

[8] Pachegaonkar,S., Taji,S. and Sane.N. (2014). Performance Analysis of Double Piipe Heat Exchanger with Annular Twisted Tape Insert, *International Journal of engineering and advanced technology (IJEAT)*, vol.3, February 2014, pp. 402-406.

[9] Tamna,S. and Promvonge P. (2014). Numerical simulation of heat transfer in a circular tube with helical tapes in *the28 Conference of the Mechanical Engineering Network of Thailand 2014*, Khonkaen, Thailand.

[10] Li,Z., Lu,J.,Tang.G,,Liu.Q and Wu.Y and Kim.Y. (2015). Effect of rib geometries and property variation on heat transfer to supercritical water in internally ribbed tubes, *Applied Thermal Engineering*, vol. 78, October 2015.pp. 303-314.

[11] Mohammed.A, Che Sidik.N, Mamat.R., Azami.W and Hamzah.W (2014). Experimental and numerical study of thermos-hydraulic performance of circumferentilally ribbed tube with AI2O3 nano fluid, *Journal of heat and mass transfer*, vol. 69, October 2014,pp. 34-40.

[12] Li.Z, Wu.Y, Tang.G, Lu.J and Zhang.D, (2015). Comparison between heat transfer to supercritical water in smooth tube and in an internally ribbed tube, *Journal of heat and mass transfer*, vol. 84, 2015 pp. 529-541.

[13] Pachegaonkar.S, Santosh.T and Sane.N, (2014). Performance analysis of double pipe heat exchanger with annular twisted tape insert, *Journal of engineering and advanced technology*, vol. 3.2014

[14] Islam.R, Jahangeer.K and Chua.K (2015). Experimental and numerical study of an evaporatively –cooled condenser of air-conditioning system, *Applied Thermal Engineering*, vol. 87, 2015,pp. 390-399.