EXPERIMENTAL STUDIES ON IMPROVEMENT OF COEFFICIENT OF PERFORMANCE OF WINDOW AIR CONDITIONING UNIT

by

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Original scientific paper
https://doi.org/10.2298/TSCI141215139T

This paper presents the performance analysis of a window air conditioner unit incorporated with wick less loop heat pipes (WLHP). The WLHP are located on the evaporator side of the air conditioning unit. The working medium for the WLHP is R134a refrigerant gas, an alternate refrigerant. The supply and return humidity of room air, the heat removal rat, and the coefficient of performance of the unit are analyzed for various ambient and room temperatures before and after incorporation of WLHP. The performance curves are drawn by comparing the power consumption and humidity collection rates for various room and ambient temperatures. The results show that coefficient of performance of the unit is improved by 18% to 20% after incorporation of WLHP due to pre-cooling of return air by WLHP, which reduces the thermal load on compressor. Similarly, the energy consumption is reduced by 20% to 25% due to higher thermostat setting and the humidity collection is improved by 35% due to pre-cooling effect of WLHP. The results are tabulated and conclusion drawn is presented based on the performance.

Key words: *wick less loop heat pipes, coefficient of performance, pre-cooling, reheating, return air, supply air, dry bulb temperature*

Introduction

The vapor-compression refrigeration system uses a circulating liquid refrigerant as the working medium which absorbs and removes the heat from the space to be cooled and subsequently rejects that heat to ambient. The performance of vapor compression refrigeration (VCR) system is measured as coefficient of performance (COP). When COP is higher, for a given work input the system extracts more heat and hence it is more efficient. Several studies on improvement of COP and hence energy conservation in heating, ventilating, and air conditioning (HVAC) system are reported in literature. In this study, with a view to improve COP, WLHP are incorporated in a small capacity window air conditioning (AC) system to experimentally validate it.

The WLHP is a device that allows transfer of very substantial quantities of heat through small surface areas over long distances, with small temperature differences and with no external pumping power. For instance, a heat pipe can transfer ten times the heat of pure rod with a 100 °C temperature difference (Holman, J. P., 1981, Heat Transfer, 5th ed. McGraw-Hill, Auckland).

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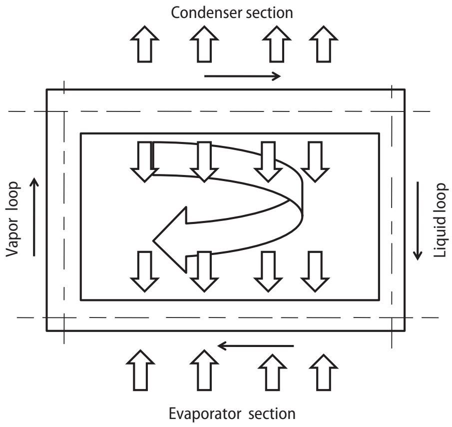


Figure 1. Wick less loop heat pipe

The WLHP is usually made up of high thermal conductivity materials like copper, aluminum, and brass depending upon the compatibility of working medium and the temperature range of application. A typical WLHP is shown in the fig. 1. Condenser section is larger in the dimension compared to evaporator for faster heat rejection. Similarly, the vapor loop diameter is more than that of liquid loop for easy flow lighter vapor than the denser liquid working medium. The vapor which flows through the vapor loop due to thermosiphon effect reach the condenser section of the WLHP, where it rejects the heat and become saturated liquid at the same pressure and temperature and flows down to the evaporator section through the liquid loop. The WLHP are designed without wicks, which offer less resistance to flow unlike in the conventional heat pipes with wicks.

Literature review

Several research works have been reported on incorporation of heat pipe heat exchangers (HPHX) in HVAC systems to reduce energy consumption, to control humidity and to improve indoor air quality. Xiao *et al*. [1] investigated the application of a HPHX to control humidity in air-conditioning system. They have experimentally investigated the incorporation of HPHX to pre-cool the return air and reheat the supply air in the normal air-conditioning system to save the precious reheat energy and also to maintain the required levels of relative humidity (RH). They concluded that with the incorporation of HPHX the cooling capacity of the system is increased by 20% to 32.7% and the RH level maintained below 70%. Etheridge *et al.* [2] have conducted a research on phase change material heat pipe cooling system for reducing AC in buildings. They report that the objective of their research is to retrofit the existing (AC) units in the buildings with previous cooling system on commercial scale and to reduce the carbon emission to save the environment.

Abd El-Baky and Mohamed [3] carried out an experimental study on HPHX for heat recovery in air-conditioning. They have reported the application of HPHX for pre-cooling the incoming fresh air in a HVAC system. Yau [4] carried out an experimental thermal performance study of an inclined HPHX operating in high humid tropical HVAC systems. He reports that the experimental results suggest that the influence of condensate formation on the fins of the inclined HPHX was negligible. Hussam and Meskimmon [5] conducted an experimental investigation on wrap around loop HPHX, used in energy efficient air handling units. Their findings show that pre-cooling and dehumidifying functions of HPHX in hot and humid climates are major contributors to reduce the running costs of the HVAC system.

Suprirattanakul *et al*. [6] carried out an experimental analysis on application of a closed-loop oscillating heat pipes with check valves for performance enhancement in air-conditioning system. The results have shown that the cooling capacity of the test AC unit had increased by 3.6%, the COP by 14.9% and the energy efficiency ratio by 17.6%. Ahmadzadehtalatapeh and Yau [7] carried out an experimental study and prediction on energy conservation options of the HPHX in AC chamber. Based on the performance characteristics and the empirical equations, the energy conservation potential of the HPHX for the years of 2000, 2020, and 2050 for Kuala Lampur, Malaysia, were predicted. Hussam and Hatem [8] carried out an investigation on thermal performance characteristics of a wraparound loop heat pipe charged with R134a refrigerant. They reported an overall thermal resistance of as low as 0.048 C/W and the same is decreasing with increasing in power input due to boiling heat transfer characteristics.

Yau and Foo [9] reported that the enhanced return of condensate in the rotating heat pipes (RHP) due to centrifugal force results in high heat transfer rate. A comparison of heat transfer characteristics of working fluids R134a, R22, and R410a was reported using the RHP with various radial displacements. Wan *et al*. [10] carried out a study on the effect of heat pipe air handling coil on energy consumption in central AC system. They reported that for a typical 20-26ºC, 50% RH indoor condition, the rate of energy saving for the heat pipe equipped AC is 23.5% to 25.7%. Yau [11] carried out a theoretical investigation on potential of HPHX on coolness recovery in tropical buildings.

Yau [12] carried out a full year energy consumption model simulation on a double HPHX system for reducing energy consumption of treating ventilation air in an operating theatre. His report consists of a case study on energy consumption of an operation theatre in Kuala Lampur, Malasiya. Alklaibi [13] carried out a theoretical investigations on evaluating the possible configurations of incorporating the loop heat pipe into the AC system. He reports that due to precooling and reheating effect of the loop heat pipe, the loop heat pipes incorporated AC system has 2.1 times more COP than the conventional AC and consumes lesser compressor work.

All the previous papers explore, study or experimental investigation on energy conservation and humidity control of HVAC system in one way or other by incorporating HPHX. In this experimental study, with a view to optimize the performance of window air conditioners, loop heat pipes are incorporated with proper instrumentations to record the performance parameters like COP, energy consumption, and humidity control.

Methodology

Construction of WLHP and selection of working medium

The selected window AC unit for experimentation is fitted with four numbers of WLHP, made up of Cu (99% pure) pipes. The evaporator sections of WLHP are of 450 mm long and 10 mm internal diameter. The condenser sections of WLHP are of 450 mm long and 16 mm internal diameter. The loops are made up of 3 mm and 1.5 mm Cu tubes for vapor side and liquid side, respectively. The WLHP are fitted with hand shut off valves and they are tested against any leak at 10 bar pressure of dry nitrogen. The WLHP are evacuated with the help of a double stage vacuum pump to the level of 20 Pa.

Selection of working medium was done based on the thermophysical properties, liquid transport factor, and temperature application range. The application temperature range in our research is 10-26 °C. The maximum return air temperature is 26 °C and the minimum supply air temperature is 9 °C. As R134a refrigerant gas fulfills the criteria the same was chosen and filled to the pressure of 3 bar, the required saturation pressure from the R134a application chart for the temperature application of 10 °C. The orientations of WLHP are such that the thermosiphon effect and gravity effect assist vapor and liquid flow of working medium, respectively.

Experiment description

A schematic diagram of the experimental apparatus is shown in fig. 2. The test loop consists of 3350 W cooling capacity window AC unit equipped with WLHP on evaporator side, humidity collection system, and data acquisition system. The humidity collecting system mea-

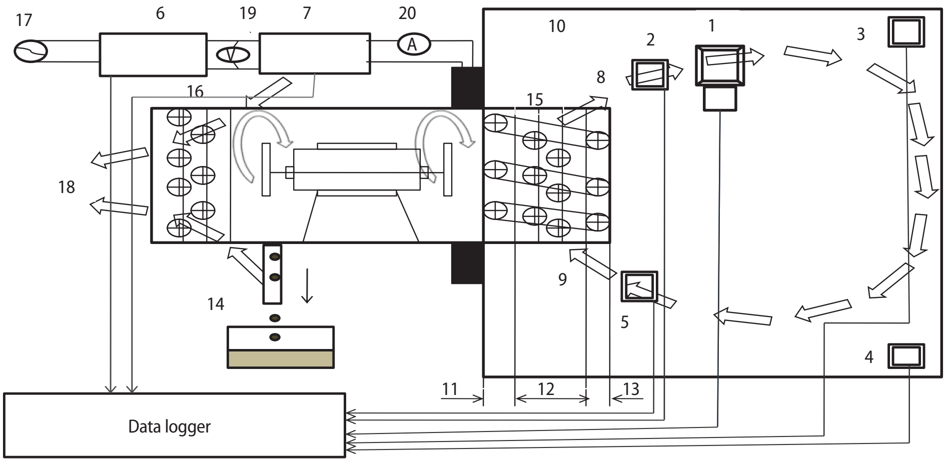


Figure 2. Schematic diagram of the experimental apparatus;
1 – anemometer (0.4-30 m/s), 2 – thermometer (0-100 °C) and hygrometer (0-99%), 3 – air conditioner and humidifier, 4 – hygrometer (0-100%) and thermometer (0-100 °C), 5 – thermometer (0-100 °C) and hygrometer (0-99%), 6 – watt meter (0-3000 W), 7 – energy meter (5-20 A), 8 – supply air, 9 – return air, 10 – standard test chamber, 11 – reheating section of WLHP, 12 – cooling coil, 13 – pre-cooling section of WLHP, 14 – measuring jar, 15 – evaporator coil, 16 – condenser coil, 17 – supply mains, 18 – ambient, 19 – voltmeter (0-240 VAC), 20 – ammeter (0-20 A)

sures the amount of water vapor condensed at the evaporator due to pre-cooling of return air by the evaporator section of WLHP and further cooling at evaporator coil of the AC unit.

Operating conditions

Experiments were conducted on the previous set-up before and after activating the WLHP. For the given ambient conditions (*DBT, RH*) keeping the supply air velocity constant, the return air temperature, return humidity, supply humidity, and power consumed were recorded after bringing the room to steady-state condition. The data were obtained using the digital data acquisition system. The enthalpy of return air before and after activating the WLHP (h_{rb}, h_{ra}), the enthalpy of supply air before and after activating the WLHP (h_{sb}, h_{sa}), the COP before and after activating the WLHP (COP_b, COP_a), difference in COP (*dCOP*) and the improvement in COP (*iCOP*) were computed using C language program for the same return and supply air temperature. The data and the results were tabulated. The uncertainty and accuracy of measurements are given in tab. 1.

Table 1. Accuracy and uncertainty of measurements

Instruments	Manufacturer	Model	Accuracy	Uncertainty
Anemometer	CE marked, Taiwan make	Digital wane probe	±2%	1%
Temperature and humidity controller	A. S. Controls, Mumbai, India (ISO 9001-2008)	Digital probe type	±2%	1%
Kilowatt hr. meter	Bentex Eletronics, New Delhi (ISO 9001-2008)	Digital	±0.11%	±0.17
Watt meter	Bentex Electronics, New Delhi (ISO 9001-2008)	Digital	±0.2%	±0.1%
Hygrometer	A. S. Controls, Mumbai, India (ISO 9001-2008)	Digital	±1%	±0.1%

Test conditions

Test equipment: Window type air conditioner unit (Samsung, window mountable, cooling power input = 1.040 kW, cooling capacity = 3.350 kW, air-flow rate = 0.3403 kg/s).

- The outside air of dry bulb temperature of 29 °C and RH of 72% was considered, which is a typical humid and tropical climate of countries like India, Sri Lanka, Myanmar, *etc.*
- The design average dry bulb temperature of 25 °C and RH of 50% was considered for indoor condition.
- The split air conditioner mounted in the test chamber was used to restore the initial indoor condition after each and every trial.
- The cooling capacity of the test air conditioner is 3350 W and the mass flow rate of air is assumed constant.
- The return air *DBT, RH* and the supply air *DBT, RH* were taken at the same point of supply and return grills, respectively. The effects of dead air pockets in the test chamber are ignored.
- The real time data were used for computation using thermodynamic equations using MATLAB and C language.

Data reduction and analysis

From the measured data the improvement of COP due to incorporation of WLHP is calculated. First the enthalpy of the humid air computed.

Enthalpy of moist air is calculated from the following relation:

$$h = [1.006T_{dbt} + w(2501 + 1.805T_{dbt})] \tag{1}$$

[2001 ASHRAE Fundamentals Handbook (SI)], where 1.006 is the specific heat of humid air at constant pressure (c_{pa}), T_{dbt} – the dry bulb temperature of the humid air, and w – the humidity ratio.

$$h = [1.006T_{dbt} + w(2501 + 1.805T_{dbt})]\dot{m}_a \tag{2}$$

where $\dot{m}_a$ is the mass flow rate of air in [kgs^{-1}].

Calculation of COP before installation of WLHP

Enthalpy of return air before installation of heat pipes:

$$h_{rb} = [1.006T_{dbr} + w_{rb}(2501 + 1.805T_{dbr})]\dot{m}_a \tag{3}$$

where T_{dbr} is the given dry bulb temperature of return air before installation of WLHP and w_{rb} is the specific humidity of return air before installation of WLHP.

Enthalpy of supply air before installation of heat pipes:

$$h_{sb} = [1.006T_{dbs} + w_{sb}(2501 + 1.805T_{dbs})]\dot{m}_a \tag{4}$$

where T_{dbs} is the given dry bulb temperature of supply air before installation of WLHP and w_{sb} being the specific humidity of supply air before installation of WLHP.

Then the difference in enthalpy of return and supply air (refrigerating effect):

$$dh_b = h_{rb} - h_{sb} \tag{5}$$

The COP of the test unit for given operating condition before incorporation of WLHP is:

$$COP_b = \frac{dh_b}{P_b} \tag{6}$$

where P_b [kW] is the power supplied before installation of heat pipes.

Calculation of COP after installation of WLHP

As previously calculated the enthalpy of return, h_{ra}, and supply, h_{sa}, air for the same *DBT* after installation of WLHP are computed. Then:

$$COP_a = \frac{dh_a}{P_a} \tag{7}$$

where P_a [kW] is the power supplied after installation of heat pipes and dh_a is the difference enthalpy ($h_{ra} - h_{sa}$).

Then difference in COP of the test apparatus after and before installation of WLHP is:

$$dCOP = [COP_a - COP_b] \tag{8}$$

Then the percentage improvement of COP is deduced:

$$iCOP = \frac{dCOP}{COP_b}100 \tag{9}$$

Results and discussion

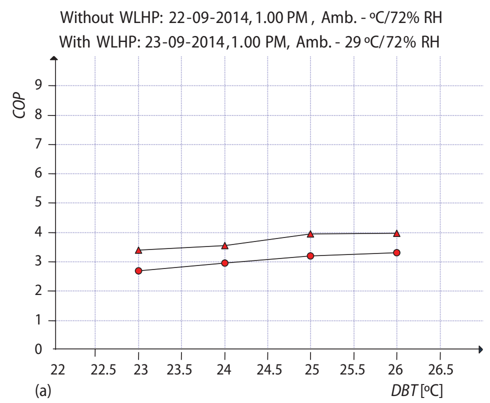
Effect of WLHP on COP

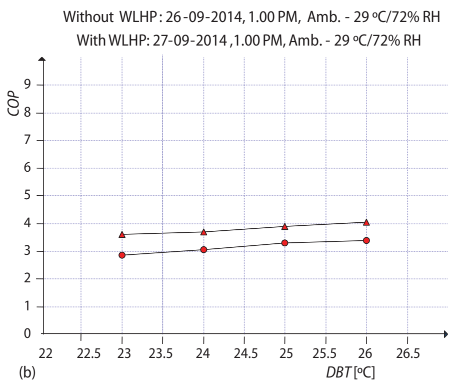
The variables which govern the performance of window air conditioner like the supply and return air humidity, power consumption are recorded with and without WLHP for different conditions of experimentation by varying the return air *DBT*. The COP is calculated before and after incorporation of WLHP and the results are plotted as graphs as shown in fig. 3(a)-(c).

Without heat pipes, at constant return air-flow rate, when the room temperature is increased from 23 °C to 26 °C the COP is found to improve. At a constant return air-flow rate and room set temperature of 25 °C when wickless loop heat pipes are incorporated with window air conditioner unit the COP is improved from 3.18 to 3.95 as evident from the graph fig. 4. This results in enhancement of COP by 24.2 % than that of conventional one. The average improvement in COP is found to be from 18% to 20% from the trials.

Effect of WLHP on energy consumption

Without WLHP, at constant air-flow rate, when *DBT* is increased from 23 °C to 26 °C the energy consumption decreases due to decreasing thermal load on compressor. With WLHP, when *DBT* is increased from 23 °C to 26 °C the power consumption further decreases due to pre-cooling function of the heat pipes. The test air conditioner was subjected to trial run for eight hours a day on different dates with different room and ambient conditions and the energy consumptions were recorded. It is observed from the bar chart fig. 4, for the optimum room set temperature of 25 °C the energy consumption is decreased from 6 kWh to 4.5 kWh (for trials on September 26 and 27, 2014). This results in 25% reduced energy consumption. The average reduction in energy consumption is found to be 20-25% from the trials.





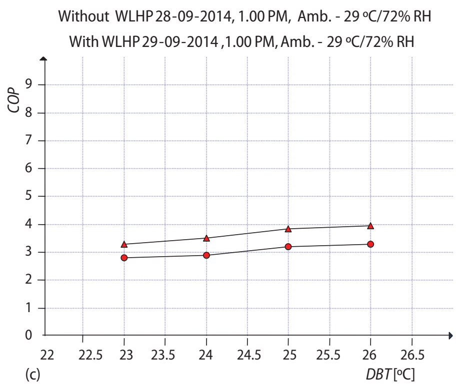


Figure 3. Effects of WLHP on COP
—●—: before installation
—▲—: after installation
(a) trial conducted on 22nd and 23rd of Sept. 2014, (b) trial conducted on 26th and 27th of Sept. 2014, and (c) trial conducted on 28th and 29th of Sept. 2014 under the stated ambient conditions

Effect of WLHP on humidity collection

The test air conditioner was subjected to trial run for eight hours a day on different dates with different room and ambient conditions and the humidity collections were recorded. As evident from the bar chart fig. 5, for the set room temperature of 25 °C the humidity collection increases from 220 gm to 310 gm, improving by 40%. The average improvement in humidity collection is found to be 35% from the trial data.

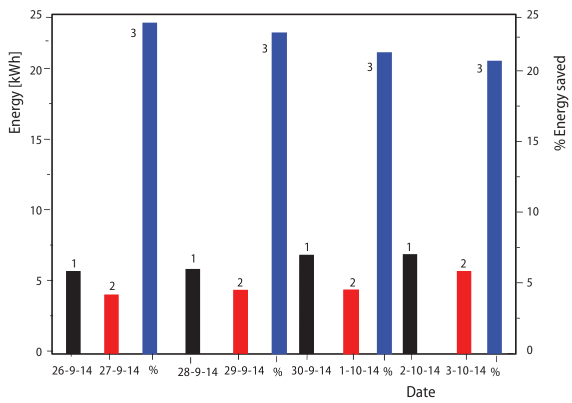


Figure 4. Effects of WLHP on energy consumption: 1 – before installation, 2 – after installation, 3 – % improved

Behavior of heat pipes

At start up, the temperature of the WLHP evaporator wall, T_W, the cooling coil of AC unit and the condenser wall of WLHP are at the same temperature. For set room temperature of 25 °C under steady-state condition, the wall temperature of WLHP evaporator is 28 °C (return air temperature), air outlet temperature from the cooling coil is 9 °C, and the WLHP condenser wall temperature is 10 °C. The saturation temperature of WLHP working medium, T_{sat}, be-

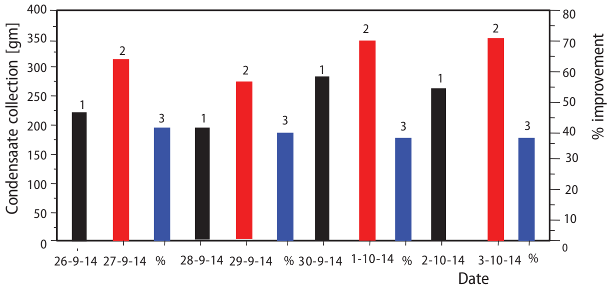


Figure 5. The effect of WLHP on humidity collection: 1 – before installation, 2 – after installation, and 3 – % improved

ing 10 °C, then the degree of super heat $T_w - T_{sat}$ is 28 – 10 = 18 °C. This slightly super-heated vapor reaches the condenser section of WLHP through the vapor loop and condenses (at 10 °C) and then returns back to the evaporator of WLHP due to gravity and the cycle continues. As the gravity assists the quick and continuous return of liquid, the WLHP evaporator will not starve for liquid and hence reliability of the WLHP is ensured.

The psychrometric effect of WLHP

The psychrometric effect of incorporation of WLHP is shown in fig. 6(a)-(c). The WLHP pre-cool section sensibly cools the return air and the same is further cooled in the cooling coil of the AC unit below its saturation temperature at that pressure and hence more humidity is collected.

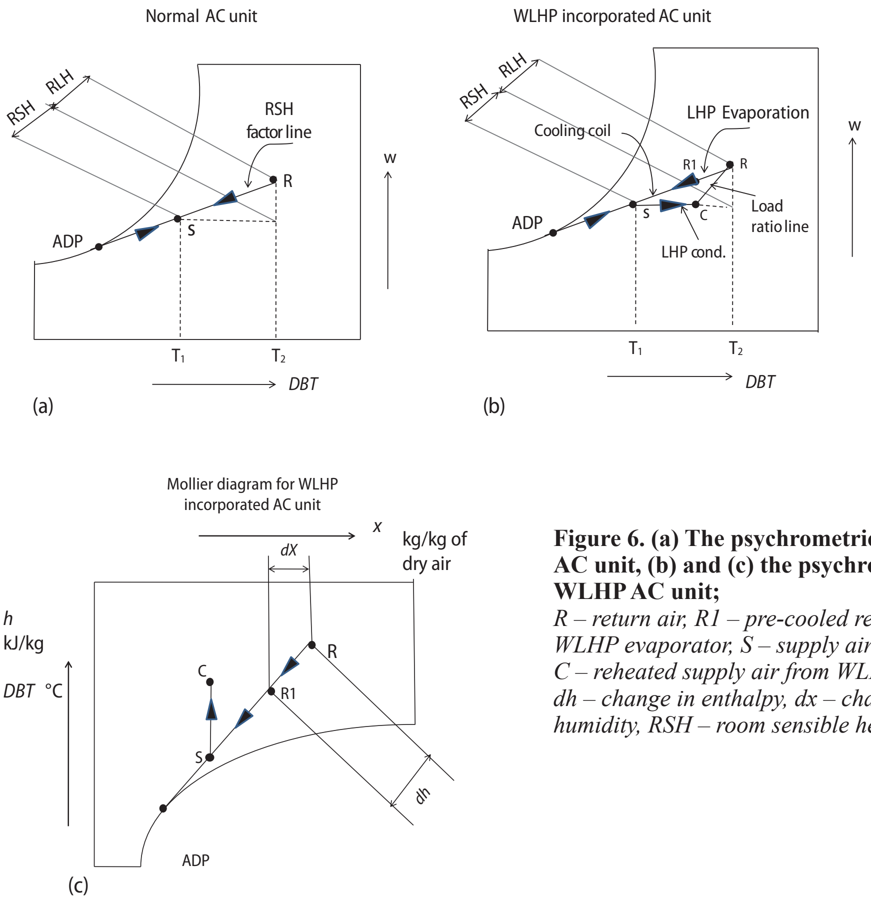


Figure 6. (a) The psychrometric effects of normal AC unit, (b) and (c) the psychrometric effects of WLHP AC unit;
R – return air, R1 – pre-cooled return air from WLHP evaporator, S – supply air from cooling coil, C – reheated supply air from WLHP condenser, dh – change in enthalpy, dx – change in specific humidity, RSH – room sensible heat

Conclusions

- The incorporation of WLHP in small capacity VCR system (domestic window AC units) is done and its performance is evaluated experimentally.
- The results show the COP of the system for the room set temperature of 23-26 °C is improved by 18-20%. This is due to pre-cool and reheat function of WLHP and the consequent reduction in thermal load on the compressor. The energy consumption is found to be reduced by 20-25% and the humidity collection improved by 35%.
- Also proved, the designed (3 bar, 10 °C – R134 a) WLHP are effective in temp range of 23-26 °C.
- The ASHRAE standard for optimum human comfort level is approached.

Nomenclature

COP – coefficient of performance, [–]
c_p – specific heat, [kJkg^{-1}K^{-1}]
DBT – dry bulb temperature, [°C]
h – specific enthalpy, [kJkg^{-1}]
$\dot{m}$ – mass flow rate of air, [kgs^{-1}]
P – power supplied to AC unit, [kW]
p – pressure, [bar]
RH – relative humidity [%]
T_{dbr} – return dry bulb temperature, [°C]
T_{dbs} – supply dry bulb temperature, [°C]
T_{dbt} – dry bulb temperature, [°C]
T_{sat} – saturation temperature, [°C]
v – the volume flow rate, [m^{3}s^{-1}]
w – specific humidity, [kgkg^{-1} of dry air]

Subscripts

a – after installation of WLHP
b – before installation of WLHP
r – return conditions of the air
s – supply condition of air
w – wall

Prefixes

d – difference
i – increase

Abbreviations

AC – air conditionig
ADP – apparatus dew point
HPHX – heat pipe heat exchangers
HVAC – heating, ventilating and air conditioning
LHP – loop heat pipes
RHP – rotating heat pipe
RLH – room latent heat
RSH – room sensible heat
VCR – vapor compression refrigeration
WLHP – wick loop heat pipes

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Paper submitted: December 15, 2014
Paper revised: August 8, 2015
Paper accepted: August 10, 2015