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COMPARISON OF DIFFERENT REFRIGERANT CIRCUIT LAYOUTS FOR TWO CONDENSERS WITH AIR AND WATER AS HEAT SINKS

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ABSTRACT

HVAC hydronic installations with simultaneous heating and cooling demand are usually projected by using two different energy production systems i.e. a chiller and a boiler or heat pump. However there is a commercial demand for project simplification and plug&play installations. Because of that many air conditioning and heat pump manufacturers are offering units which are able to produce chilled and hot water at the same time performing an actual heat recovery from one zone of the building to another. These are so called water-to-air-to-water heat pumps. Basically they are heat pumps which integrate an evaporator for chilled water production, a condenser for hot water production and an air-coil to be used as alternative heat/sink for the unbalanced load.

Many times, the commercial available products are not fulfilling the expectations in terms of cooling and heating capacities control. It is difficult to design the suitable frigorific circuit and regulation algorithm in a way that variable simultaneous cooling and heating demand can be satisfied at the same time.

Some possible arrangements of the refrigerant circuit are compared by means of simulation. These possible arrangements of the refrigerant circuit are three: serial, parallel and on/off circuit.

The serial configuration consists in two condensers connected in the same refrigerant line, the parallel configuration consists in two condensers connected in parallel and the on/off configuration consists of alternating the operation of the condensers in order to satisfy demand so that when one works the other is in standby.

The three arrangements of the refrigerant circuit are simulated by software “IMST-ART” comparing range of partial load covered and energetic efficiency of the different options as function of partial load ratio using the new concept of Total COP.

1. INTRODUCTION

Commercial buildings, hospitals, and hotels can have heating loads during spring and summer season at mild temperatures. One common solution to solve this problem is to use fan-coils with four tubes providing hot and chilled water simultaneously. Hot and chilled water normally is served by two independent air source units or a chiller and a boiler. This common situation is not efficient, in fact unit serving chilled water is rejecting heat to the ambient that is lost while the unit serving hot water is taking heat from the ambient consuming both electricity. Therefore there is an opportunity of heat recovery and to use only one unit serving both chilled and hot water. Building loads are not balanced, so normally cooling load is lower than heating load or vice versa; this means that the design of a heat recovery unit must provide full capacity in the main mode (cooling or heating) and partial capacity in the auxiliary mode, the rest of the capacity not used in the auxiliary mode has to be exchanged with the ambient used as heat sink/source. Figure 1 describes the situation.

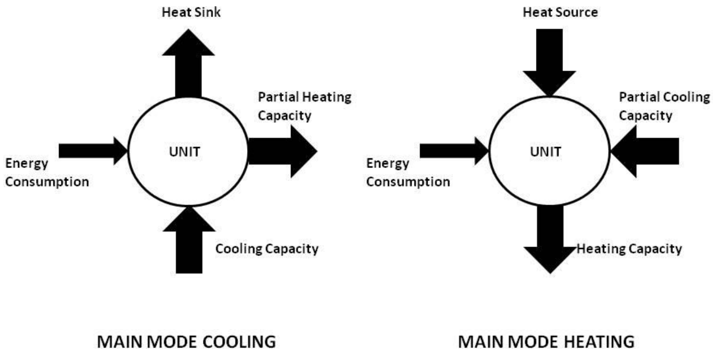


Figure 1 Scheme of a Simultaneous Heating and Cooling Unit

Different solutions are found in the literature for designing units with two heat exchangers, one with air and the other providing the auxiliary capacity working both as evaporators or condensers.

(Kang et al. 2009) presented experimental results of a design patented by LG. The simultaneous heating and cooling heat pump is a variable refrigerant volume type, it consist of four indoor units connected in parallel and fed by individual electronic expansion valves. Unit can operate the indoor units delivering cold or hot air that is accomplished working in parallel with the outdoor unit. Refrigerant flow through outdoor unit is controlled by a bypass valve and electronic expansion valves.

(Byrne, Miriel and Lenat 2009) presented a heat pump for simultaneous heating and cooling using HFC or carbon dioxide. It can provide simultaneously hot and chilled water by using brazed plate heat exchangers and uses an additional air heat exchanger. Hot refrigerant gas from the compressor is passed through the air heat exchanger or trough BPHE by using solenoid valves.

The design described in (Matsuura 1993) can provide hot and chilled water simultaneously using two condensers and two evaporators; water heat exchangers are of shell and tube type and air heat exchangers are of fin and tube type. Refrigerant flows alternatively trough water heat exchangers or air heat exchangers; 3 way valves releases refrigerant to one heat exchanger or the other. This system is designed for using in fish farms.

(Berti and Marcel 1984) describes a unit designed for providing simultaneously hot and chilled water, this is accomplished using two evaporators (BPHE and fin and tube) and two condensers (BPHE and fin and tube) disposed in parallel. Refrigerant flow is circulating continuously through the heat exchangers using 3 way valves to control the flow through BPHE to adapt the capacity to the load.

It is clear from the study of the different solutions proposed that there are two approaches: continuous or non-continuous flow.
Systems designed with the principle of continuous flow split the refrigerant between AHE and WHE providing auxiliary capacity; mass flow rate of refrigerant through WHE is controlled to match the auxiliary load demand. Designs found in the literature disposed heat exchangers in parallel but also it is possible to connect heat exchangers in serial as usually used in heat pump providing sanitary hot water using a desuperheater before the condenser.
Systems designed with the principle of non-continuous flow send all refrigerant alternatively to AHE or WHE/. Thermostat in the indoor unit gives the signal to 3 way valves or solenoid valves to send refrigerant to the right heat exchanger.

The following questions arise from this introduction: what is the best approach in terms of energy efficiency, flow control and demand adaptation; continuous or non-continuous flow?. Simulation software IMST-ART (J.M. Corberán 2010) has been used to calculate COP of three different designs: continuous flow serial, continuous flow parallel and non-continuous.

2. SIMULATION PROCEDURE

The simulation of the unit has been made under conditions listed in Table 1. Evaporation temperature has been maintained constant for all the cases in order that evaporator design does not affect conclusions for the condensation side. Three different typical ambient temperatures in cooling mode has been taken into account for all the studies. Water mass flow rate is also maintained constant in order to supply water at 50 ºC to the building at full load capacity.

Evaporation Temperature	2 ºC
Ambient Temperature	15 ºC, 35 ºC, 44 ºC
Air Std. FlowRate	10 000 m^3/h
Condenser Inlet Water Temperature	45 ºC
Condenser Water Mass flow Rate	1.18 kg/s (4.36 m^3/h)
Superheat	5 K
Subcooling	8.28 K

Table 1 Boundary values used for the simulation.

The components used in the unit are listed in Table 2.

Compressor	Danfoss SH090-4
Water Source Heat Exchanger	BPHE type SWEP V80HP/32
Air Source Heat Exchanger	Tube and Fin Type Core Height 1350 mm Core Depth 44 mm Finned Length 2400 mm Tube Diameter 9.52 mm Tube Thickness 0.813 mm Fin Spacing 16 fpi Fin thickness 0.12 mm

Table 2 Components of the unit used in the simulation

Three different configurations of the unit have been considered: Non-Continuous (NC), Serial Continuous (SC) and Parallel Continuous (PC).

Figure 2 shows the design of the NC model. Condensation takes place in the WHE sending hot water to the building till the heating load is satisfied, then unit changes the refrigerant flow path in order to condense in the AHE.

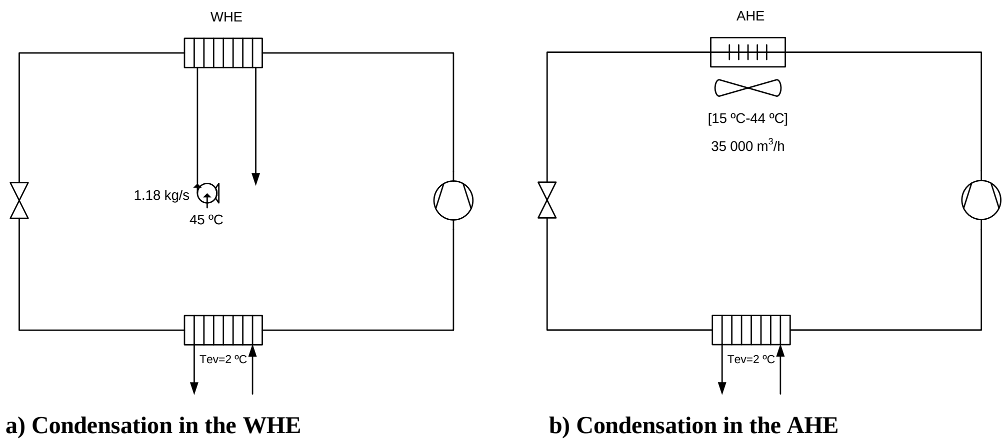
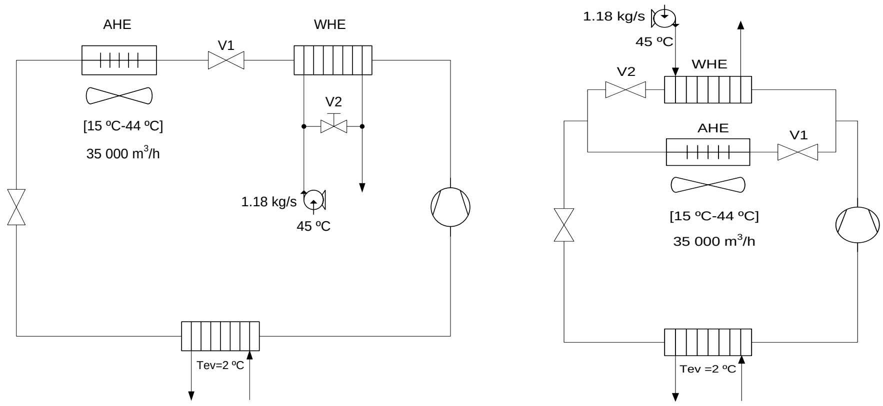


Figure 2 Non-Continuous Designs of the Heat Pump.

Figure 3 shows the design of the continuous models, SC and PC. There are two possibilities to dispose heat exchangers in the SC design, AHE downstream WHE or the opposite; the right election is the one shown in the figure as pressure in the WHE tends to be higher than in AHE as water inlet temperature is higher than air temperature. The control of the heating capacity at WHE is proposed by three different means: fan speed regulation, pressure drop at valve V1 and water bypass at valve V2.
Second continuous design is the parallel configuration. Valve V2 is needed to equalize pressures of WHE and AHE at the expansion valve inlet for the same reason discussed above. Control of the heating capacity at WHE is made with valve V1.



b) Serial Continuous (SC) Design **b) Parallel Continuous (PC) Design**

Figure 3 Continuous Designs of the Heat Pump

Part load ratio in the condenser is defined as the heating load required by the building divided by the nominal heating capacity of the unit working at standard conditions in a water to water mode (Figure 2a):

$$PLR = \frac{Load}{\dot{Q}_{h,nom}} \tag{1}$$

COP of the unit must to be redefined for this type of heat pump due to the reason that now there are two useful energy flows, the heat absorbed in the evaporator and the heat rejected at the WHE condenser. Then COP of the simultaneous heating and cooling unit is defined as:

$$COP_t = \frac{Q_c + Q_h}{E} \tag{2}$$

It is assumed that continuous designs match building load and does not stop so Total COP is calculated as:

$$COP_t = \frac{Q_c + Q_h}{E} = f(PLR) \tag{3}$$

In the case of non-continuous designs, heat rejected at the WHE condenser is not constant and assuming negligible transient effects, unit COP must be calculated as:

$$COP_t = \frac{PLR\ \dot{Q}_{c,ww} + (1-PLR)\dot{Q}_{c,aw} + PLR\ \dot{Q}_{h,ww}}{PLR\ \dot{E}_{ww} + (1-PLR)\dot{E}_{aw}} \tag{4}$$

Where $\dot{Q}_{c,ww}$,$\dot{Q}_{h,ww}$, $\dot{E}_{ww}$, are the cooling capacity, heating capacity and power input of the unit working with WHE as condenser and $\dot{Q}_{c,aw}$, $\dot{E}_{aw}$ are the cooling capacity and power input of the unit working with AHE as condenser.

3. RESULTS AND DISCUSSION

3.1. Regulation of the unit

The first step in the simulation is to investigate if continuous designs are capable to cover the full PLR range with the controls proposed.

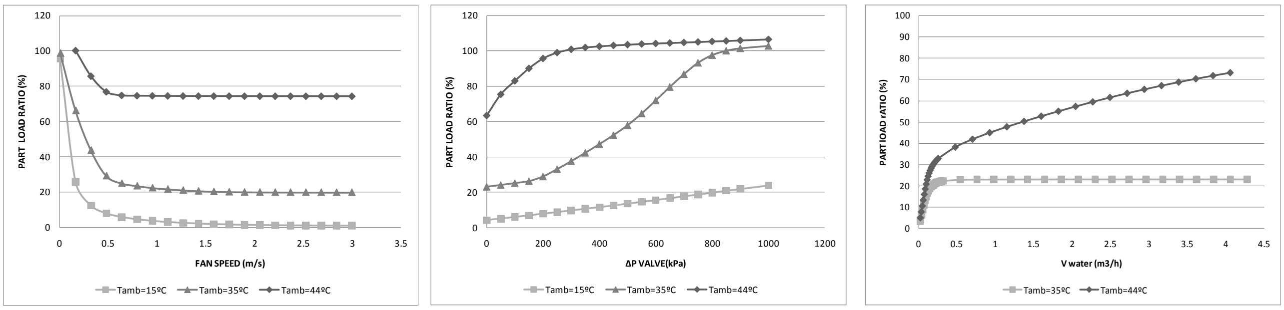
Figure 4 shows the effect of the different control strategies for SC design on the part load ratio.

Figure 4a shows the effect of actuating over the fan speed. Only in the case of having an ambient temperature of 15 ºC is possible to cover all the PLR range. At 35 ºC ambient temperature, the minimum PLR is around 20% and 80% at 45ºC ambient temperature. Also, the region where is it possible the regulation is between 0 an 0.5 m/s fan velocity.

Increasing Fan velocity enhances air heat transfer and therefore increases AHE heating Capacity (and decreasing heating capacity in WHE) and AHE condensation temperature decreases. There is a limit for AHE condensation temperature, given by ambient temperature plus subcooling used in the cycle. This effect means that increasing air velocity results in an asymptotically approach to this minimum condensation temperature and PLR as shown in Figure 4a

Figure 4b shows the effect of valve regulation, range of PLR covered is similar as Fan Speed regulation for 35ºC and 44 ºC ambient temperatures. Working at 15 ºC ambient temperatures, the maximum PLR achieved is around 20% at 1000 kPa pressure drop. The effect of using a valve that introduces a pressure drop between WHE and AHE is not trivial. Detailed results of the study shows that condensation temperature in the AHE remains constant due to its outlet temperature is very close to the ambient and the condensation temperature in the WHE raises as the valve closes. At 15 ºC ambient temperature, it is required a very high condensation temperatures in the WHE to obtain PLR higuer than 20%.

Figure 3b shows the effect of regulating heating capacity using a water side bypass. Results for 15ºC ambient temperature are not shown as passing 100% of the water through WHE gives a PLR near zero and therefore regulation will result in a horizontal line with null PLR. It is interesting to discuss the trend at 35ºC ambient temperature; there is a region where the water flow through the WHE does not affect performance. At this conditions, WHE is working as a single phase heat exchanger (desuperheater) as condensation is made in the AHE; flow having minimum thermal capacity is refrigerant and also thermal resistance of the water is negligible compared with refrigerant vapor. This makes that water flowrate variation does not affect WHE performance until is very low as shown in Figure 4c. A wide range of PLR coverage can be obtained at 44ºC temperature but half of the range is obtained in only 0.4 m^3/h variation of the water flowrate which makes control difficult.



a) Fan Regulation **b)** Valve Regulation **c)** Bypass regulation

Figure 4 Regulation of the Serial Continuous design with different strategies

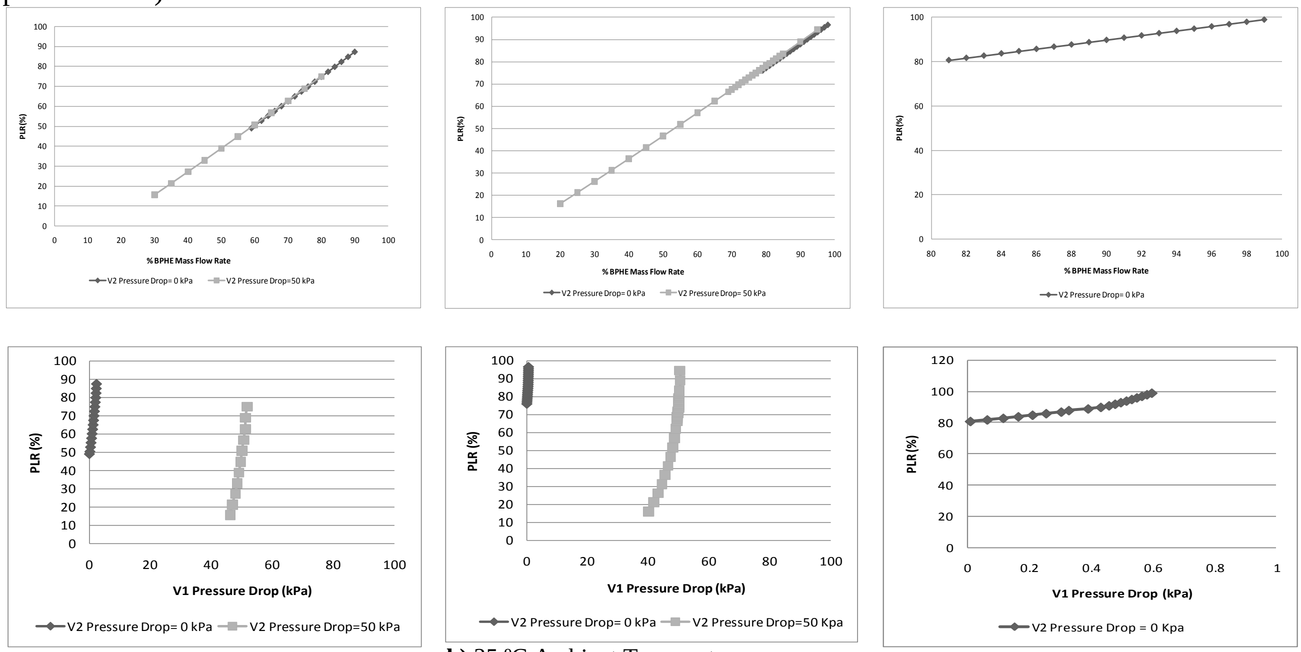
Figure 5 shows regulation results for the PC design. The first observation is that it is not possible to cover all the range without using valve V2 for introducing a pressure drop downstream WHE.

Results calculated with null pressure drop at valve V2 indicates that there is a minimum mass flow rate through WHE (an a minimum PLR therefore) due to the fact that outlet pressure of both heat exchangers (WHE and AHE) are equal; therefore with valve V1 totally opened refrigerant flow is split depending on the pressure drop of each heat exchanger. Minimum WHE mass flow rate is 50.7% at 15ºC, 75% at 35ºC nad and 80.64% at 44 ºC.

The way to cover a wider range of PLRs with the PC design is then to introduce a pressure drop at WHE outlet with valve V2 as shown in the figure 3b. However is not possible to use this strategy working at 44ºC because this is a condition where WHE condensation temperature is near ambient temperature; introducing a pressure drop at WHE outlet is not possible without dramatically increasing WHE condensation temperature.

Figure 5 also shows response of PLR to variations of pressure drop at valve V1. Sensitivity depends on the ambient temperature and pressure drop at valve V2, results vary from requiring 0.59 kPa to cover 80%-100%

of PLR (0.0295 kPa per PLR unit) at 44 ºC to requiring 10.3 kPa to cover 16.9%-95.7% of PLR (0.129 kPa per PLR unit).



a) 15 ºC Ambient Temperature **b)** 35 ºC Ambient Temperature **c)** 44 ºC Ambient Temperature

Figure 5 Regulation of the Parallel Continuous design

Figure 6 shows Total COP of the different unit design options at different ambient temperatures, Non-Continuous design (NC), Serial Continuous design with fan regulation (SC-Fan), Serial Continuous design with valve regulation (SC-Valve), Serial Continuous design with bypass (SC-Bypass) and Parallel Continuous design (PC).

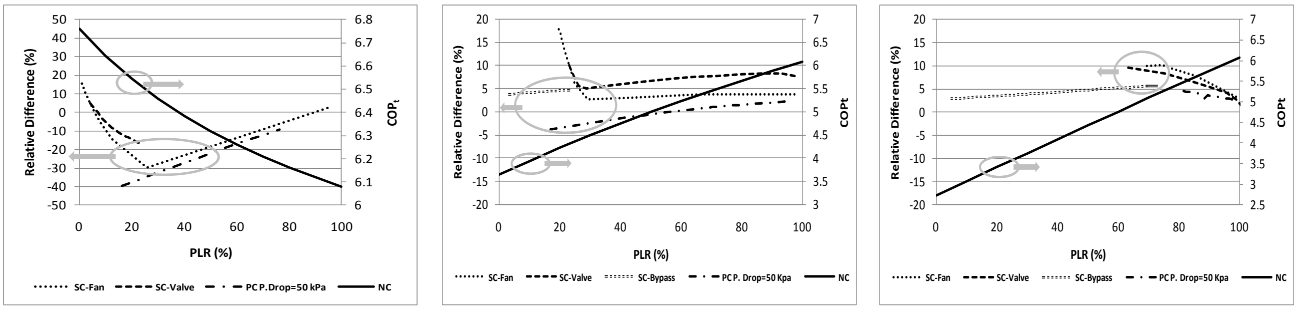
PLR influence on the Total COP defined in Eq. 2 for NC design varies depending on the ambient temperature. Total COP at 100% PLR (COPt=6.08) does not change with ambient temperature as is the case where unit is working all the time with WHE producing water at 50 ºC. Total COP of the unit working at 0% PLR depends on the ambient conditions as all the heat is rejected in the AHE. At 15 ºC ambient temperature, thermodynamic cycle is very efficient and Total COP of the unit working with AHE condenser is higher than the unit working with WHE as condenser; unit working with AHE Condenser has null useful heating capacity (Q_h in Eq. 2) but production of chilled water is more efficient than unit working with WHE condenser. At 35 ºC and 44 ºC ambient temperature Total COP at 0% PLR is lower than 100% PLR as condensation temperature of the unit working with AHE condenser approaches that of the unit working with WHE condenser and also there is not useful heat available.

SC-Fan design has lower COP than NC design at 15 ºC except at low PLR. The reason of COP degradation is that working in continuous mode means that always there is a water sink temperature of 45 ºC in the WHE. Although ambient temperature is favorable for the thermodynamic cycle, condensation temperature must be at least 45 ºC except in the case of low PLR where WHE works as a desuperheater and condensation temperature can be lower than 45 ºC. At 35ºC and 44 ºC SC-Fan design has higuer COP than NC design (around 10%) and there is also an increase in COP at 35 ºC working at low PLR. Ambien temperatures in these cases are not so favorable for the thermodynamic cycle and improvement comes with increase in condensing area.

SC-Valve design behaves in a similar way as SC-Fan design. At 15 ºC COP degradation compared with NC design has the same reason as SC-Fan design; although the valve makes that WHE and AHE can work at different condensation temperatures, COP is influenced by WHE condensation temperature because is at compressor outlet.

SC-Bypass design can only be employed at 35 ºC and 44 ºC and its improvement compared with NC-design is around 5% as a decrease in water flow through WHE tends to increase condensation temperature in the WHE that is opposite to the tendency of decreasing condensation temperature using two heat exchangers for condensation.

PC design at 15ºC also has bad values of COP compared with NC design. Reason of this COP decrease is the same as commented above with SC-Fan, SC-Valve and SC-Bypass, i.e. compressor condensation temperature is imposed mainly by WHE that has the higher sink temperature, therefore losing the advantage of working at low ambient temperature. COP at 35 ºC is slightly higher at low PLR and lower at high PLR. At 44 ºC, regulation range is very narrow as discussed above and COP is slightly higher than NC design.



a) 15 ºC Ambient Temperature **b)** 35 ºC Ambient Temperature **a)** 44 ºC Ambient Temperature

Figure 6 Energy Efficiency comparison of the different options

4. CONCLUSION

Different options for designing a unit capable to produce chilled and hot water at different load ratios in the hot side have been analyzed using simulation software IMST-ART.

Capability of the different designs to adapt to the load has been studied and the following conclusions can be drawn:

- Non-Continuous design is capable to adapt to any PLR required.
- Serial Continuous with fan regulation design is able to cover all the PLR at ambient temperature of 15ºC but at 35 ºC and 44 ºC the minimum PLR that can work are 19.8% and 69.4% respectively.
- Serial Continuous with valve design cover 0%-24% PLR range at 15 ºC, 23%-100% at 35 ºC and 63.4%-100% at 44ºC.
- Serial Continuous with bypass design cannot be employed at ambient temperature of 15 ºC, it can cover 0%-23% PLR at 35 ºC and 0%-63% at 44ºC.
- Parallel Continuous design can cover 16%-73% PLR at 15 ºC and 16.9%-79.8% at 35ºC if a pressure drop of 50 kPa is provoked at WHE outlet. At 44 ºC, this design can only work without any pressure drop at WHE outlet; in this condition the PLR range covered is 81%-100%

Efficiency of the different designs defined as Total COP unit have been calculated, the following conclusions can be drawn from results:

- Continuous designs have lower COP than Non-Continuous design at 15ºC except Serial Continuous with fan and valve regulation at low PLR. Reason is that condensation temperature at compressor outlet is imposed by water temperature at WHE that is very high and the benefit in the thermodynamic cycle to work at low ambient temperatures is lost.
- Serial Continuous designs have higher COP than Non-Continuous design (between 3%-10%) at 35ºC and 44ºC. Parallel Continuous design show higher COP than Non-Continuous design only at 44 ºC and 35 ºC at very high PLR.

Taking into account range of PLR covered, complexity and efficiency, the Non-Continuous design is the best option to make a unit capable to produce chilled and hot water simultaneously at different loads on the hot water. This statement is valid assuming negligible transient effect in the Non-Continuous operation.

NOMENCLATURE

E	Energy Input	(J)
$\dot{E}$	Power Input	(W)
Q	Thermal Energy	(J)
$\dot{Q}$	Capacity	(W)

Subscripts

aw	Ait to Water Unit
c	Cooling
h	Heating
nom	Nominal
ww	Water to Water Unit

Abbreviations

AHE	Air Source Heat Exchanger
BPHE	Brazed Plate Heat Exchanger
COP	Coefficient of Performance
HFC	HydroFluoroCarbon
NC	Non-Continuous Design
SC	Serial Continuous Design
PC	Parallel Continuous Design
PLR	Partial Load Ratio
WHE	Water source Heat Exchanger

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