Investigation on annual energy performance of a VWV air-source heat pump system

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Abstract:
This work studies the annual energy performance of a VWV air-source heat pump system by means of experiment and simulation. An advanced VWV R410a air-source heat pump system, integrated with independent normal fan-coil unit, small temperature difference fan-coil unit and floor radiation heating-coil unit indoor terminals, is successfully constructed. Multi-groups of field tests show that system COP varies from 2.0~4.0 for the different operating load under corresponding environment conditions. Moreover, in order to predict the long term energy performance of VWV R410a ASHP system, several system dynamic models are constructed and validated with experimental data. Then the annual energy performance of investigated system is simulated for the same buildings under different climate zones in China. Both experimental and simulation results show that VWV ASHP system can achieve high competitive seasonal energy efficiency, which shows the SPF could be more than 3.0, for the buildings under selected climate zones. At last, several recommendations based on the simulation results are proposed to the existed heat pump system.

Keywords:
VWV system, R410a, Air source heat pump, Energy efficiency.

1. Introduction
In December 2015, 195 countries are committing the world to limit a rise in global temperatures to 2\degree C at the Paris climate conference. This Paris Agreement is due to enter into force in 2020. One of the important actions, committed by different countries, is lowering carbon dioxide intensity to a certain level according to the Intended Nationally Determined Contributions (INDCs). So the energy efficiency improvement must be realized across all sectors of the economy. As well known, the air-source heat pump (ASHP) system is widely used for building thermal energy management system through the world due to its flexibility and efficiency. Moreover, more and more countries, like EU, Japan and China etc., incorporate the air-source heat into the category of renewable energy. There is huge potential to use air-source heat pump as thermal energy management system both for building space heating and cooling. Take China as an example, the energy efficient solutions to building thermal management in the hot summer and cold winter climate zone, is urgently required. Especially, it is of importance to solve to winter heating problems in this area due to lack of central heating systems or the uncomfortable heating by normal split room air conditioning heat pumps.

Looking into different ASHP systems, there are mainly following 3 types of arrangement based on the different indoor terminal installation. The first type is air-to-air heat pump system with refrigerant pipe connecting the outdoor unit and indoor units, for those which have the ability to vary refrigerant flow rate also called VRV or VRF system [1]. The second is air-to-water heat pump system with water pipe carrying heat from or to the indoor terminals for air conditioning, and for those which can optimize the water flow rate regarded as VWV system. The last one is air to air heat pump system with air handling unit, which can fulfil VAV with the help of HVAC box. Some research work on the comparison of energy consumption for the above 3 systems is conducted. They either chose different AC systems for the same building, or employed different operation control strategies, then compare energy consumption and thermal comfort for different cases [2-4].
It is concluded that the VRV system is the most energy efficient, and followed by VWV and VAV systems. However, several research papers published to argue that the system Coefficient of Performance (COP) of VWV system could be improved by integrating small temperature difference indoor terminals.[5,6] Moreover, the VWV system show superiority on the aspect of installation flexibility comparing with VRV and VAV system.

So latest work by Zhang [7] experimentally investigate the system performance of same VWV ASHP integrated with normal fan coil unit (NFCU) or small temperature difference fan coil unit (STDFCU). The test result shows ASHP+STDFCU system can save 15%-22% electricity consumption than ASHP+NFCU system under similar weather conditions. However, it is impossible for field tests to predict the long period annual energy consumption and the system performance of investigated heat pump system, especially under different climate conditions. So this work constructed the dynamic R410a ASHP system models for cooling and heating mode for the purpose of long period simulation and the annual energy performance of a VWV R410a ASHP system is analysed for different climate conditions.

2. Experimental field test results

2.1 Experimental setup

An air source heat pump system is installed in the Sino-Italy green energy laboratory located in Shanghai, China. The total space conditioning area is 292m², including one room on the ground floor and four rooms on the first floor. The nominal cooling capacity of the ASHP unit is 45.8kW, with the double compressors On/Off control strategy. The whole ASHP system consists of an air-cooled heat pump unit, water pump, water pipes and indoor terminals. Figure 1 shows the schematic diagram of the experimental system. Most indoor terminals are NFCU. However, there are three kinds of indoor terminals are installed in the ground floor room, which aims to compare the influence of different indoor terminals on system energy efficiency and indoor thermal comfort.

Moreover, as shown in Figure 1, several different kinds of sensors are used in the field test system to monitor the operation parameters of system. First, in order to analyse the energy consumption of different system components, current transformers are used in the power wires for gathering power consumption data of ASHP unit, pump, and indoor terminals. In addition, flow meters are installed in four locations in the system to get flow of ASHP total supply water and three different indoor terminals. At last, thermal resistance PT1000 are immerged into different locations of water pipe to get the temperature change of ASHP inlet and outlet temperature of water and different indoor terminals. All the signals from different sensors are transformed to Data Acquisition System through signal wire, then to PC.
In order to test the whole day performance of system under variable water volume condition, the number of indoor terminals is changed during test period. The ASHP system includes 11 indoor terminals (2×5.9kW+6×5.4kW+3×3.7kW). To control the cooling and heating load, it is available to adjust the number of opening terminals in which opening 11 terminals or 6 terminals (2×5.9kW+2×5.4kW+2×3.7kW). This is the reason to choose two typical cooling and heating days for the field test investigation.

2.2 Field test result for typical cooling and heating days

In this section, the used experimental data is mainly from short periods of field test under typical cooling and heating days. For example, the experimental data in Aug.1st and Aug.2nd of 2015 are used to represent the system performance for typical cooling days, while system performance of typical heating days utilize the experimental data on Mar.5th and Dec.19th of 2015. The Fig.2 and Fig.3 show daily ambient temperature variation under the selected typical reference summer and winter days.

Figure 2 Ambient temperatures in summer

Figure 3 Ambient temperatures in winter

Figure 4 shows the ASHP inlet and outlet water temperature curve for the typical cooling days. The mean temperature difference is about 5°C both for two test conditions. Both compressors worked under all terminals operation condition, and they switched between On and Off condition for 30 times. However, the two compressors switched more frequently under part terminal condition, and they switched between On and Off condition for 64 times due to lower cooling load, which led to the intensely variation for inlet and outlet water temperature, as Fig.4(b) shows.

(a) All terminals operation condition

(b) Part terminals operation condition

Figure 4 ASHP unit inlet and outlet water temperature curve in summer

Similarly, Fig.5 shows the ASHP inlet and outlet water temperature curve for the typical heating days. The inlet and outlet water temperature difference is about 5.3°C with all terminals operation condition. The frequently switch of On/Off condition for compressor also caused the intensely variation of inlet and outlet water temperature for terminals operation condition.
In Fig.6, the daily COP variation of ASHP system with different indoor terminals is shown for typical cooling and heating days. It is obvious that COP varies with outdoor air temperature. When all terminals are under operation the COP for cooling and heating mode are 2.98 and 3.17 respectively, while the values are 3.80 and 3.74 for the part terminals operation condition.

It can be also concluded that the COP is closely associated with the different operation condition of compressors. The frequently switch On/Off of compressors definitely decrease the system performance, so it is meaningful to optimize the compressor control strategy to increase the system performance. One of the effective solutions is integrating inverter compressor which can be linear controlled by the frequent variation.

3. Long period simulation analysis

Experimental field test could well and truly predict the ASHP system operating characteristic and system performance. However, it is impossible to predict the long period annual energy consumption and the system performance of investigated heat pump system, especially under different climate conditions. So in this section, the dynamic R410a ASHP system models for cooling and heating mode are constructed for the long period simulation purpose.

Fig.7 shows the R410a ASHP system model configurations at cooling and heating mode. The models were successfully constructed in the Modelica/Dymola environment by means of combining the TIL Suite package and Modelica.IDEAS library.
In detail, the R410a refrigerant cycle mainly includes an inverter scroll compressor, finned coil condenser, expansion valve and evaporator, and Table 1 shows the detail component type and specification. The rotary speed of compressor is controlled by PI module based on fixed return or supply water temperature; the opening control of expansion valve is based on the principle of thermal expansion valve with the superheat of 3ºC. In addition, the air side fan and hydraulic side pump can be controlled by PI module using the feedback from the temperature sensor, as showed in Fig.7.

<table>
<thead>
<tr>
<th>Components</th>
<th>Type and specification</th>
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<tbody>
<tr>
<td>Compressor</td>
<td>Copeland_ZP83KCE_TFD_50Hz with variable speed control</td>
</tr>
<tr>
<td>Air-cooled Condenser</td>
<td>Grooved copper tubes and aluminium fins</td>
</tr>
<tr>
<td>Water-cooled evaporator</td>
<td>Plate heat exchanger</td>
</tr>
<tr>
<td>Expansion device</td>
<td>Thermal expansion valve with fixed degree of superheat</td>
</tr>
<tr>
<td>Volumetric flow of fan</td>
<td>Axial Flying Bird IV with rotating shroud, 3800 l/s</td>
</tr>
<tr>
<td>Hydraulic pump</td>
<td>Variable-speed pump with 0.9<del>3.0 l/s and 20</del>120 kPa</td>
</tr>
<tr>
<td>Environmental condition</td>
<td>Weather and underground condition from 3 cities in China</td>
</tr>
<tr>
<td>Load condition</td>
<td>Reference office building from 3 cities in China</td>
</tr>
<tr>
<td>Possible terminals in Summer</td>
<td>ASHP+STDFCU or ASHP+NFCU</td>
</tr>
<tr>
<td>Possible terminals in Winter</td>
<td>ASHP+STDFCU or ASHP+NFCU or ASHP+floor heating unit</td>
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</table>

The validation of the developed models is conducted for one typical cooling day, as showed in Fig.8 and Fig.9 which show the power consumption of compressor and hourly COP variation both for simulation and field test.
Comparing the results of the simulation and field test, it can be found that the daily averaged COP values are 3.99 and 3.80 for simulation and field test, which shows 5% of difference.

At last, the simulation boundary conditions of ASHP system include the cooling and heating load of the reference buildings from Energy Plus calculation [8], the outdoor air temperature and humidity of the reference cities. The simulation results include all of the thermal physical parameters of the heat pump system, and COP for heat pump unit and total system, and supply and return water temperature are mainly showed to evaluate the characteristic performance of the R410a VWV ASHP system.

### 3.1 Simulation for the cooling season

Fig. 10 and Fig.11 show the simulation boundary condition of the R410a ASHP system under cooling season of one year. These data is calculated from Energy Plus for the reference office building based on PNNL’s study [8]. The simulation time is selected from early of May until late of September of reference year, and the climate data in Shanghai, China is used. Actually, there is frequently variation of the cooling load, and this is critical for numerical calculation of the Dymola software. So the cooling load showed in Fig.11 is acquired through 1 day’s moving average of the practical calculated building load.

As described above, the COP of heat pump unit and total system, and supply and return water temperature are mainly showed to evaluate the characteristic performance of the system. So the variation of values for these parameters is showed in Fig.12 and Fig.13. Equation 1 and 2 represent the calculation method of the COP\(_{hp,c}\) and COP\(_{total,c}\), and the difference is whether involve the work of air side fan and hydraulic side pump for the calculation.

\[
\text{COP}_{hp,c} = \frac{\dot{Q}_c}{W_{comp}} \tag{1}
\]
\[ \text{COP}_{\text{total,c}} = \frac{\dot{Q}_c}{W_c + W_{\text{fan}} + W_{\text{pump}}} \]  \hspace{1cm} (2)

The daily ambient temperature variation is the main reason for the frequent variation of the COP curve. Since Fig.12 shows that both values of COP are lower in the hottest summer days, it can be also concluded that the ambient temperature have of importance impact on the system energy performance. However, the difference between COP\textsubscript{hp,c} and COP\textsubscript{total,c} of hottest days is smaller due to the limitation of maximum speed of air side fan. Moreover, water volume flow regulated by PI module between the maximum and minimum value also caused the varying inlet and outlet water temperature difference, as showed in Fig.13.

![Figure 12 COP variation in cooling season](image1)

![Figure 13 Inlet and outlet water temp to building](image2)

**3.2 Simulation for the heating season**

Simulation boundary conditions of the heating season of one year are showed in Fig. 14 and Fig.15. These data is also got from the reference office building based on PNNL’s study. The simulation period is selected from late of October middle of March of reference year.

![Figure 14 Ambient temperatures in heating season](image3)

![Figure 15 Reference building heating load](image4)

Similarly, the variation of the COP of heat pump unit and total system, and supply and return water temperature under heating season are showed in Fig.16 and Fig.17. Equation 3 and 4 represent the calculation method of the COP\textsubscript{hp,h} and COP\textsubscript{total,h}.

\[ \text{COP}_{\text{hp,h}} = \frac{\dot{Q}_h}{W_{\text{comp}}} \]  \hspace{1cm} (3)

\[ \text{COP}_{\text{total,h}} = \frac{\dot{Q}_h}{W_c + W_{\text{fan}} + W_{\text{pump}}} \]  \hspace{1cm} (4)

The daily ambient temperature variation also caused the frequent and monthly variation of the COP curve in heating season. However, there is no PI module for the regulation of air side fan in the heating mode of ASHP system, and the maximum fan speed is used for the model simulation. This is due to the difficulty in finding a proper fan control strategy for the heating mode. So Figure.16...
shows the relatively constant value of the COP\textsubscript{total,h} during the whole heating season. In the same way, water volume flow regulated by PI module between the maximum and minimum value also caused the varying inlet and outlet heating water temperature difference, as showed in Fig.17.

4. Discussion for simulation results under different climate

In this section, the long period annual energy efficiency of investigated VWV R410a air-source heat pump system under different climate conditions is simulated and discussed. So the simulation boundary conditions of Shanghai, Wuhan and Chongqing are selected for ASHP system, and all the 3 cities locate in the hot summer and cold winter climate zone of China. In addition, the cooling and heating load, as showed in Fig.19 and Fig.21, are calculated from the same reference office buildings from Energy Plus under the different outdoor ambient conditions, as Fig.18 and Fig.20 showed. So the difference of building cooling and heating load is mainly due to the various outdoor ambient conditions.

Fig.22 and Fig.23 show the COP variation of the R410a heat pump unit utilized for the 3 different cities under cooling and heating season. Similar with the simulation result of Shanghai climate condition, the daily ambient temperature variation caused the frequent variation of the COP curve.
However, the different COP values for the 3 cities are due to both the different outdoor ambient conditions and cooling and heating load of the reference building.

In order to compare the seasonal energy performance of the VWV system under these 3 different cities, Fig.24 and Fig.25 show the seasonal averaged $COP_{hp}$ and $COP_{total}$ both for the cooling and heating season. It is obvious that all the seasonal COP of the heat pump unit is more than 4.0 except the cooling season in Chongqing, and the seasonal COP of total system is around 3.0 with quite small variation. So it shows that the relatively stable and good energy performance of the VWV R410a ASHP system under hot summer and cold climate zones, which is based on Chinese classification method.

In addition, the cooling and heating water supply temperature is set to be used for normal fan coil unit, which consume much more energy comparing with the terminals using small temperature difference fan coil unit, as demonstrated in the latest research [7-9]. This means that there is still potential to increase the energy performance of the simulation result by setting higher cooling temperature or lower the heating temperature. So, to be concluded, the VWV R410a ASHP system, using advanced hydronic technologies and the variable refrigerant flow heat pump unit, show good indoor comfort, energy efficiency, system flexibility, and ease of installation for buildings under different climate conditions based on both experimental and simulation results.

5. Conclusion

The annual energy performance of a VWV R410a air-source heat pump system was investigated by means of experiment and simulation. So the following conclusions are summarized for this study:

1. Multi-groups of field tests of an advanced VWV R410a air-source heat pump system show that system COP varies from 2.0–4.0 for the different operating load under corresponding environment conditions; 

2. In order to predict the long term energy performance of VWV R410a ASHP system, several dynamic models are constructed for 3 different cities in China;
3. Both experimental and simulation results show that VWV ASHP system can achieve high competitive seasonal energy efficiency, which shows the annual averaged system COP could be around 3.0, for the buildings under selected climate zones.

**Nomenclature**

**Abbreviation**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>AC</td>
<td>air-conditioning</td>
</tr>
<tr>
<td>ASHP</td>
<td>air-source heat pump</td>
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<tr>
<td>COP</td>
<td>coefficient of performance</td>
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<tr>
<td>HVAC</td>
<td>heating ventilation air-conditioning</td>
</tr>
<tr>
<td>NFCU</td>
<td>normal fan coil unit</td>
</tr>
<tr>
<td>STDFCU</td>
<td>small temperature difference fan coil unit</td>
</tr>
<tr>
<td>VRV</td>
<td>variable refrigerant volume</td>
</tr>
<tr>
<td>VRF</td>
<td>variable refrigerant flow</td>
</tr>
<tr>
<td>VWV</td>
<td>variable water volume</td>
</tr>
<tr>
<td>VAV</td>
<td>variable air volume</td>
</tr>
</tbody>
</table>

**Subscripts and superscripts**

- \(hp,c\): heat pump unit, cooling
- \(hp,h\): heat pump unit, heating
- \(total,c\): heat pump system including fan and pump, cooling
- \(total,h\): heat pump system including fan and pump, heating
- \(comp\): compressor

**Reference**


