Study on energy consumption model of multi-unit air conditioning system with digital scroll compressor

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Abstract
The objectives of this study were to: (i) build simulation model for multi-unit air conditioning (AC) system with digital scroll compressor (DSC) and validate its precision by experiments; (ii) build system energy consumption calculation model by simulation. Lumped parameter model of compressor and electronic expansion valve, and district lumped parameter model of condenser and evaporator were employed in simulation program of multi-unit AC system with DSC. The results indicated that errors between simulated value and experimental data of system hourly energy consumption were within 10%. The simulation model showed good precision. Simulation results indicated that system hourly energy consumption differences caused by indoor unit operating number were less than 15%, which can be neglected. Thus, hourly energy consumption (\(HW\)), hourly energy efficiency ratio (\(HEER\)) and hourly heating performance factor (\(HHPF\)) calculation model of multi-unit AC system with DSC were built based on simulation results. Simulation results indicated that the variation of \(HW\) with part load ratio (\(PLR\)) and outdoor air temperature presented concave surface distribution and the variations of \(HEER\) and \(HHPF\) with \(PLR\) and and outdoor air temperature presented convex surface distribution. The model provides a tool for energy saving optimization and seasonal energy consumption evaluation of multi-unit AC system with DSC.

Keywords: Multi-unit AC, Digital scroll compressor, Model, Simulation, Energy consumption

1 Introduction
Multi-unit air conditioning (AC) system with digital scroll compressor (DSC) is a kind of efficient direct evaporative multi-connected air conditioning system which consists of at least one outdoor unit with digital scroll compressor inside and multiple indoor units. The refrigerant volume is regulated by digital scroll compressor and each electronic expansion valve (EEV) to adapt to air conditioning load of each indoor unit timely. The output of digital scroll compressor is regulated by pulse width modulation valve which regulates compressor’s loaded time and unloaded time. Due to its advantages, such as highly efficient, wide range output, precise capacity control, space saving and easy maintenance (Zhang, 2014; Meng, 2015; Yu,2016; Zhang, 2017), multi-unit AC system with DSC has become popular in commercial buildings and residential buildings (Iijima, 1991; Masuda, 1991; Aynur, 2010).

There have been several relevant studies conducted on simulations for multi-connected air conditioning system. Shao (Shao, 2008; Shao, 2012; Liang, 2010; Liu, 2018) developed simulation model for complex refrigeration systems with several condensers, evaporators and inverter compressor based on two-phase fluid network, and performance study has been conducted based on the above simulation model. Li presented simulation module embedded in the software of EnergyPlus for water-cooled variable refrigerant flow (VRF) system with inverter compressor (Li, 2009). Zhu developed generic simulation models for performance and control analysis of VRF systems with variable speed compressor (Zhu, 2013), and conducted simulation of VRF system with outdoor air processing unit in heating mode (Zhu, 2014). Li presented a model for recombined household multi-split VRF system with split-type air conditioners and demonstrated its high energy efficiency superior to split-type air conditioners by simulation (Li, 2017). Kim [16]...
developed procedures embedded in the software of EnergyPlus for model calibration of VRF combined with dedicated outdoor air system (Kim, 2018). Zhang developed an improved energy model to determine the energy performance of household VRF systems by simulation (Zhang, 2018).

From the above surveyed study, seldom studies were conducted on simulation model for multi-unit AC system with DSC currently. The lack of multi-unit AC system with DSC simulation model is preventing the study on its energy consumption performance and its energy saving optimization. Thus, this paper aims to develop multi-unit AC system with DSC simulation model and validate its precision by experiments. Meanwhile, energy consumption performance and model of multi-unit AC system with DSC is studied by simulation in this paper. The innovative points are introducing simulation model to multi-unit AC system with DSC and developing its energy consumption model.

| Nomenclature |
|--------------|
| $F$ | area [m$^2$] |
| $Pr$ | prandtl number |
| $K$ | heat transfer coefficient [W/(m$^2$K)] |
| $Gr$ | grashof number |
| $L$ | length [m] |
| $Q$ | capacity [W] |
| $R$ | ratio |
| $W$ | input power [W] |
| $d$ | diameter [m] |
| $f$ | friction factor |
| $h$ | enthalpy [J/kg] |
| $k$ | local resistance factor |
| $m$ | mass flow [kg/s] |
| $p$ | pressure [Pa] |
| $r$ | heat transfer resistance [m$^2$K/W] |
| $s$ | spacing [m] |
| $t$ | temperature [°C] |
| $HEER$ | hourly energy efficiency ratio |
| $HHPF$ | hourly heating performance factor |
| $HW$ | hourly energy consumption [kWh] |
| $PLR$ | Part load ratio |
| $\eta$ | efficiency |
| $\alpha$ | convective heat transfer coefficient [W m$^{-2}$ K$^{-1}$] |
| $\lambda$ | conductive heat transfer coefficient [W/(m·K)] |
| $\rho$ | density [kg m$^{-3}$] |
| $\nu$ | velocity [m s$^{-1}$] |
| $\eta$ | efficiency |
| $\chi$ | dryness |
| $\delta$ | thickness [m] |
| $\nu$ | velocity [m s$^{-1}$] |
| $\rho$ | density [kg m$^{-3}$] |
| $\alpha$ | convective heat transfer coefficient [W m$^{-2}$ K$^{-1}$] |
| $\lambda$ | conductive heat transfer coefficient [W/(m·K)] |
| $Nu$ | nusselt number |
| $Re$ | reynolds number |

| Subscripts |
|------------|
| $a$ | air |
| $c$ | cooling |
| $d$ | dirt |
| $e$ | equivalent |
| $f$ | fin |
| $h$ | heating |
| $m$ | mean |
| $p$ | pressure |
| $t$ | tube |
| $w$ | wall |
| $ad$ | air direction |
| $co$ | contact |
| $in$ | inlet |
| $ls$ | local resistance |
| $oa$ | outdoor air |
| $pm$ | per meter |
| $sc$ | supercooled |
| $sh$ | superheated |
| $sl$ | saturated liquid |
| $sp$ | single phase |
| $sv$ | saturated vapor |
| $tp$ | two phase |
| $com$ | compressor |
| $con$ | condenser |
| $eva$ | evaporator |
| $ref$ | refrigerant |
| $out$ | outlet |

2 Experimental System

Multi-unit AC system with DSC experimental system is used to validate the precision of its simulation model in this paper. As depicted in Fig.1, the lab consists of four rooms and adopts multi-unit AC system with DSC which is composed of four indoor units and one outdoor unit. Each unit type and rated capacity are shown in Table1.
Both air side and refrigerant side operating parameters, such as air temperature and velocity, refrigerant temperature and pressure, are measured during experiments. All these measured data are collected automatically every 15s by data module ADAM4000 series. Compressor power consumption and fans power consumption are measured and recorded, too. Testing instruments and their parameters are presented in Table 2.

3 Component model

Multi-unit AC system with DSC consists of digital scroll compressor, condenser, EEV and evaporator. Each model of four components is developed firstly, and then by integrating them together, multi-unit AC system with DSC model is developed. The refrigerant used in multi-unit AC system with DSC is R22, and its thermodynamic properties are calculated by using the simplified Cleland model (Cleland, 1986).

3.1 Digital scroll compressor model

Refrigerant flow and compressor energy consumption are two essential factors that affect system performance, which should be reflected by compressor model. The variation of compressor state parameters is quick and the thermal inertia is small, thus, compressor is in quasi-steady state. And it is assumed that thermodynamic parameters of working fluid is uniform. The compression process is regarded as adiabatic. Therefore, compressor model is built by employing steady-state lumped parameter method, which simplifies thermodynamic parameters of working fluid only related to
According to compressor test data, ten-coefficient equations of mass flow, energy consumption and refrigeration capacity are built as follows.

\[
Q_{\text{con}} = c_0 + c_1 t_{\text{eva}} + c_2 t_{\text{eva}}^2 + c_3 t_{\text{eva}}^3 + c_4 t_{\text{eva}}^4 t_{\text{con}} + c_5 t_{\text{eva}}^5 t_{\text{con}} + c_6 t_{\text{eva}}^6 t_{\text{con}} + c_7 t_{\text{con}} + c_8 t_{\text{con}}^2 + c_9 t_{\text{con}}^3
\]

(1)

\[
W_{\text{con}} = w_0 + w_1 t_{\text{eva}} + w_2 t_{\text{eva}}^2 + w_3 t_{\text{eva}}^3 + w_4 t_{\text{eva}}^4 t_{\text{con}} + w_5 t_{\text{eva}}^5 t_{\text{con}} + w_6 t_{\text{eva}}^6 t_{\text{con}} + w_7 t_{\text{con}} + w_8 t_{\text{con}}^2 + w_9 t_{\text{con}}^3
\]

(2)

\[
m_{\text{con}} = m_0 + m_1 t_{\text{eva}} + m_2 t_{\text{eva}}^2 + m_3 t_{\text{eva}}^3 + m_4 t_{\text{eva}}^4 t_{\text{con}} + m_5 t_{\text{eva}}^5 t_{\text{con}} + m_6 t_{\text{eva}}^6 t_{\text{con}} + m_7 t_{\text{con}} + m_8 t_{\text{con}}^2 + m_9 t_{\text{con}}^3
\]

(3)

3.2 Condenser model

Condenser model is built by employing district lumped parameter method, which is on the basis of the following primary assumptions (Ding, 2001). District lumped parameter method simplifies each zone temperature only related to time, not to coordinates.

1. Both air flow outside the pipes and refrigerant flow inside the pipes are regarded as one-dimensional.
2. Refrigerant pressure drop in the pipes is neglected.
3. Refrigerant flow is regarded as homogeneous.
4. Thermal resistance of pipe wall is neglected.

Based on the above assumptions, condenser model is divided into the following three phase zones as shown in Fig.2, superheated zone, two-phase zone and supercooled zone.

As shown in Fig.3, input initial values and assume outlet refrigerant enthalpy, then the length of supercooled zone, two-phase zone, and superheated zone are calculated. If the error between three zones’ total calculated length and actual length is greater than convergence accuracy, adjust output enthalpy and repeat the above calculation until the calculation converges.
In supercooled zone, heat flux between refrigerant and air equals refrigerant heat loss. Thus, heat transfer area of supercooled zone is calculated by Eq.(4).

\[
F_{\text{con,sc}} = \frac{m_{\text{ref,con}}(h_{\text{ref,con,sl}} - h_{\text{ref,con,out}})}{K_{\text{con,sc}}\Delta t_{\text{con,sc}}} \tag{4}
\]

Refrigerant heat loss equals air heat gain. Thus, outlet air enthalpy of supercooled zone is calculated by Eq.(5).

\[
h_{a,\text{con,ip, in}} = h_{a,\text{con, in}} + \frac{m_{\text{ref,con}}(h_{\text{ref,con,sl}} - h_{\text{ref,con,out}})}{m_{a,\text{con}}} \tag{5}
\]
Outlet air temperature is calculated according to outlet air enthalpy. Thus,

\[
\Delta t_{\text{con,sc}} = \frac{(t_{\text{ref,con,sl}} - t_{a,\text{con,ip,in}}) - (t_{\text{ref,con,ip,in}} - t_{a,\text{con,in}})}{\ln \frac{t_{\text{ref,con,sl}} - t_{a,\text{con,ip,in}}}{t_{\text{ref,con,out}} - t_{a,\text{con,in}}}}
\]

(6)

\[
K_{\text{con,sc}} = \frac{1}{1 + \alpha_{\text{ref,con,sc}} F_m + \frac{\delta_{a}}{\lambda_{i}} F_m + r_{co} + r_{d} + \frac{1}{\alpha_{a,\text{con}} \eta}}
\]

(7)

Similarly, in two-phase zone, corresponding parameters are calculated as follows.

\[
F_{\text{con,sp}} = \frac{m_{\text{ref,con}} (h_{\text{ref,con,sv}} - h_{\text{ref,con,sl}})}{K_{\text{con,sp}} \Delta t_{\text{con,sp}}}
\]

(8)

\[
h_{a,\text{con,ip,out}} = h_{a,\text{con,ip,in}} + \frac{m_{\text{ref,con}} (h_{\text{ref,con,ip,in}} - h_{\text{ref,con,ip,out}})}{m_{a,\text{con}}}
\]

(9)

\[
\Delta t_{\text{con,sp}} = \frac{t_{a,\text{con,ip,in}} - t_{a,\text{con,ip,out}}}{\ln \frac{t_{\text{ref,con}} - t_{a,\text{con,ip,out}}}{t_{\text{ref,con}} - t_{a,\text{con,ip,in}}}}
\]

(10)

\[
K_{\text{con,sp}} = \frac{1}{1 + \alpha_{\text{ref,con,sp}} F_m + \frac{\delta_{a}}{\lambda_{i}} F_m + r_{co} + r_{d} + \frac{1}{\alpha_{a,\text{con}} \eta}}
\]

(11)

Similarly, in superheated zone, corresponding parameters are calculated as follows.

\[
F_{\text{con,sh}} = \frac{m_{\text{ref,con}} (h_{\text{ref,con,in}} - h_{\text{ref,con,sv}})}{K_{\text{con,sh}} \Delta t_{\text{con,sh}}}
\]

(12)

\[
h_{a,\text{con,out}} = h_{a,\text{con,ip,out}} + \frac{m_{\text{ref,con}} (h_{\text{ref,con,in}} - h_{\text{ref,con,sv}})}{m_{a,\text{con}}}
\]

(13)

\[
\Delta t_{\text{con,sh}} = \frac{(t_{\text{ref,conin}} - t_{a,\text{con,out}}) - (t_{\text{ref,conin}} - t_{a,\text{con}})}{\ln \frac{t_{\text{ref,conin}} - t_{a,\text{con,out}}}{t_{\text{ref,conin}} - t_{a,\text{con}}}}
\]

(14)

\[
K_{\text{con,sh}} = \frac{1}{1 + \alpha_{\text{ref,con,sh}} F_m + \frac{\delta_{a}}{\lambda_{i}} F_m + r_{co} + r_{d} + \frac{1}{\alpha_{a,\text{con}} \eta}}
\]

(15)

Convective heat transfer coefficient of air side is calculated by the following heat transfer correlation (Li, 1997).


\[ \alpha_{a,con} = \frac{\lambda_a}{d_e} N u_{a,con} \]  

Where,  

\[ N u_{a,con} = \frac{0.687 R e_{a,con}^{0.518} (S_f^{0.0935} - 0.0935 N \cdot S_{f,ad}) - 0.1990}{d_e} \]  

Convective heat transfer coefficient of refrigerant side is calculated by Dittus-Boelter heat transfer correlation as follows.

\[ \alpha_{ref,con,sp} = \lambda N u_{ref,con,sp} / d_{in} \]  

\[ N u_{ref,con,sp} = 0.023 R e_{ref,con}^{0.8} P r_{ref,con}^{0.3} \]  

Condensate convective heat transfer coefficient of refrigerant inside the tube is calculated by Shah heat transfer correlation as follows (Shah, 1979).

\[ \alpha_{ref,con,sp} = \alpha_{ref,con,sp} \left( 1 - x_{sp} \right)^{0.8} + \frac{3.8 x_{sp}^{0.76} (1 - x_{sp})^{0.04}}{P r_{0.38}} \]  

### 3.3 EEV model

In VRF system, refrigerant flow is mainly affected by compressor operating parameters and evaporating temperature. Compressor operating parameters are not available by experiments. Thus, the ratio of EEV inlet pressure to its outlet pressure is used to replace compressor operating parameters in EEV model. And on the basis of experimental data regression, the functions between indoor unit refrigerant flow and the ratio of EEV inlet pressure to its outlet pressure and evaporating temperature are obtained as follows.

\[ m_{c1} = 0.01735 + 0.04139 R_{p} - 0.14362 R_{p} - 0.00167 t_{eu}^2 + 0.03195 R_{p}^2 \]  

\[ m_{c2} = -0.00326 + 0.02989 R_{p} - 0.12585 R_{p} - 0.00118 t_{eu}^2 + 0.02978 R_{p}^2 \]  

\[ m_{c3} = -0.00180 + 0.02674 R_{p} - 0.11994 R_{p} - 0.00105 t_{eu}^2 + 0.02890 R_{p}^2 \]  

\[ m_{c4} = 0.13312 + 0.0126 R_{p} - 0.08023 R_{p} - 0.00068 t_{eu}^2 + 0.01918 R_{p}^2 \]  

\[ m_{h1} = 0.00368 - 0.00729 R_{p} + 0.03868 R_{p} + 0.00036 t_{eu}^2 - 0.00607 R_{p}^2 \]  

\[ m_{h2} = 0.01559 - 0.00005 t_{eu} + 0.00260 R_{p} - 0.00003 t_{eu}^2 - 0.00008 R_{p}^2 \]  

\[ m_{h3} = 0.00774 - 0.00423 t_{eu} + 0.01872 R_{p} + 0.00022 t_{eu}^2 - 0.00326 R_{p}^2 \]  

\[ m_{h4} = 0.01690 - 0.00493 t_{eu} + 0.02149 R_{p} + 0.00022 t_{eu}^2 - 0.00307 R_{p}^2 \]  

Where, subscript 1, 2, 3 and 4 represent indoor unit 1, 2, 3 and 4, respectively.

### 3.4 Evaporator model

Low dryness gas-liquid two-phase refrigerant enters the evaporator, evaporates gradually, and leaves as superheated gas finally. Therefore, the refrigerant side in evaporator is generally composed of two-phase zone and superheated zone as presented in Fig.4. And distributed parameter method is employed to model evaporator. Because of the existence of...
accelerated pressure drop and frictional pressure drop, evaporator pressure drop is considerable. Thus, it is necessary to introduce pressure drop calculation in evaporator model.

![Evaporator distributed parameter model](image)

Evaporator distributed parameter model is on the basis of the following assumptions [19]:

1. Thermal resistance of pipe wall is neglected.
2. Pressure drop in the superheated zone is neglected due to short superheated zone and small accelerated pressure drop.

As shown in Fig. 5, input initial values, and assume outlet refrigerant enthalpy and two-phase zone pressure drop, then two-phase zone pressure drop and the length of two-phase zone and superheated zone are calculated. If the error between two-phase zone pressure drop and its assumed value is greater than convergence accuracy, adjust two-phase zone pressure drop assumed value and repeat the above calculation until the calculation converges. At last, if the error between calculated length and actual length of two zones is greater than convergence accuracy, adjust output enthalpy and repeat the above calculation until the calculation converges.
Two-phase zone pressure drop is calculated by the following momentum equation.
\[ \frac{p_{in} - p_{out}}{L} = \frac{4fG_i^2}{\rho_{in}d_i} + \frac{G^2}{L} \left( \frac{1}{\rho_{out}} - \frac{1}{\rho_{in}} \right) \]  

(28)

Refrigerant heat gain is calculated by Eq.(29).

\[ Q_{ref} = m_{ref} (h_{ref,in} - h_{ref,out}) = K_{ref} F(T_{i,w} - T_{ref,in}) \]  

(29)

Thus, each zone length is calculated by Eq.(30).

\[ L = \frac{Q_{ref}}{\pi d_i K_{ref} (T_{i,w} - T_{ref,in})} \]  

(30)

Air side convective heat transfer and mass transfer coefficient are calculated by heat transfer correlations referring to (Ma, 2008). Refrigerant side convective heat transfer coefficient is calculated by Dittus-Boeler heat transfer correlation as follows (Kandlikar, 1987).

3.5 Pipeline network model

Connecting pipe network pressure loss consists of on way resistance and local resistance of components such as elbows and valves. Refrigerant in pipe network is single-phase fluid, so friction pressure drop is calculated as follows.

\[ \Delta p_f = f \frac{L \rho \nu^2}{d} \]  

(31)

Where, \( f \) is obtained by Colebrook relation (Moody, 1994).

Pressure loss at elbow and tee joints are corrected as follows.

\[ \Delta p_k = \sum_{i=1}^{k} \frac{k_i \rho \nu^2}{d} \]  

(32)

Where, \( k \) is affected by pipe diameter, joint and elbow shape. It can be found in ASHRAE Handbook (2001).

4 Solution procedure

Based on each component model of multi-unit AC system with DSC, simulation program is developed. Program for calculating thermodynamic properties of air and refrigerant are compiled, too. Flow chart of multi-unit AC system with DSC simulation program under cooling condition and heating condition are presented in Fig.6 and Fig.7, respectively. As shown in Fig.6 and Fig.7, input known parameters, such as condenser inlet air temperature and volume, evaporator inlet air temperature and volume, evaporator superheat, and assume evaporating temperature and condensing temperature, then compressor outlet refrigerant flow and other parameters are calculated by using compressor model. Then, condenser outlet refrigerant parameters are calculated by using condenser model. EEV refrigerant flow is obtained by using EEV model. If the error between EEV refrigerant flow and compressor outlet refrigerant flow is greater than convergence accuracy, adjust condensing temperature assumed value and repeat the above calculation until the calculation converges. Then, calculate evaporator outlet temperature by using evaporator model and if the error between calculated and assumed value of evaporator outlet temperature is greater than convergence accuracy, adjust evaporating temperature assumed value and repeat the above calculation until the calculation converges.
Compressor calculation

Assuming condensing temperature and evaporating temperature

Condenser calculation

Four EEVs parallel connection calculation

Compressor mass flow equals to EEV mass flow

Adjusting condensing temperature

Four evaporators parallel connection calculation

Compressor inlet temperature equals to evaporator outlet temperature

Adjusting evaporating temperature

End

Fig. 6 Flow chart of multi-unit AC system with DSC simulation program under cooling condition
5 Model validation and energy consumption performance simulation

Multi-unit AC system with DSC simulation model was validated under typical cooling and heating conditions experimentally. Indoor air temperature under cooling and heating conditions were 25°C and 22°C, respectively, which were set on the basis of empirical operating parameters in Shanghai city. Outdoor air temperature under cooling and heating conditions were 30°C and 4°C, respectively, which were set according to frequently occurring outdoor air temperature in Shanghai city.

Since this paper mainly study energy consumption model, hourly energy consumption simulated value was validated by experimental data to ensure accuracy of energy consumption model. Hourly energy consumption ($H_W$) comparison results between simulated value and experimental data were shown in Fig.8 and Fig.9. The results showed that errors between simulated value and experimental data were within 10%. The model of multi-unit AC system with
DSC shows good precision.

![Fig.8 HW validation under cooling condition](image1)

![Fig.9 HW validation under heating condition](image2)

Based on multi-unit AC system with DSC model, hourly energy consumption differences caused by indoor unit operating number was studied by simulation under typical cooling and heating conditions, which were set based on typical design parameters as follows: indoor air temperature under cooling and heating conditions were 25°C and 20°C, respectively; outdoor air temperature under cooling and heating conditions were 34°C and 4°C, respectively. And part load ratio (PLR) is defined as the ratio of actual capacity to maximum capacity in this paper, which is convenient for energy consumption analysis.

The model of multi-unit AC system with DSC is used to study energy consumption difference caused by indoor units operating number in case of the same PLR. As presented in Fig.10 and Fig.11, simulation results indicated that hourly energy consumption differences caused by indoor units operating number were less than 15%, which can be neglected.
As depicted in Fig.10 and Fig.11, indoor units operating number has little effect on hourly energy consumption differences of multi-unit AC system with DSC. Thus, $HW$ of multi-unit AC system is mainly affected by outdoor air temperature ($t_{oa}$) and PLR. Therefore, hourly energy consumption values under various $t_{oa}$ and PLR conditions were calculated by simulation when indoor air temperature under cooling and heating conditions were 25℃ and 20℃, respectively. Based on hourly energy consumption simulation results, hourly energy efficiency ratio ($HEER$) and hourly heating performance factor ($HHPF$) defined as follows were calculated.

$$HEER = \frac{Q_{c,\text{max}}}{HW_c}$$  \hspace{1cm} (33)  

$$HHPF = \frac{Q_{h,\text{max}}}{HW_h}$$  \hspace{1cm} (34)

Where, $Q_{c,\text{max}}$ is outdoor unit cooling capacity, kWh; $Q_{h,\text{max}}$ is outdoor unit heating capacity, kWh.

As shown in Fig.12-14, the variation of $HW$ with PLR and outdoor air temperature presented concave surface distribution, the variations of $HEER$ and $HHPF$ with PLR and outdoor air temperature presented convex surface distribution. This concave and convex surface distribution are due to scroll compressor operating characteristics. Scroll compressor output capacity decreases with the decreasing of PLR. And the ratio of condenser pressure to evaporating pressure decreases, too. Since compressor efficiency increases and its increasing ratio decreases with the decreasing of pressure ratio, the decreasing ratio of energy consumption will decrease with the decreasing of PLR.
Fig. 12 $HW_c$ variation with $PLR$ and $t_{oa}$ under cooling condition

Fig. 13 $HW_h$ variation with $PLR$ and $t_{oa}$ under heating condition

Fig. 14 $HEER$ variation with $PLR$ and $t_{oa}$ under cooling condition
By regressing the simulation results, $HW$, $HEER$, and $HHPF$ model were obtained as follows.

$$HW = 0.01001t_{oa}^2 + 0.546t_{oa}PLR + 12.69PLR^2 - 0.6572t_{oa} - 23.66PLR + 14.3$$ (35)

$$HW = 0.01208t_{oa}^2 - 0.6776t_{oa}PLR + 11.73PLR^2 - 0.1003t_{oa} + 3.022PLR + 1.438$$ (36)

$$HEER = 0.003898t_{oa}^2 - 0.1932t_{oa}PLR - 8.387PLR^2 - 0.3115t_{oa} + 17.55PLR + 5.891$$ (37)

$$HHPF = 0.0007281t_{oa}^2 + 0.1199t_{oa}PLR - 7.553PLR^2 + 0.08269t_{oa} + 8.537PLR + 0.5911$$ (38)

The above energy consumption and energy efficiency ratio calculation model of multi-unit AC system with DSC provides a tool for its energy saving optimization and seasonal energy consumption evaluation.

6 Conclusions

In this paper, simulation model for multi-unit AC system with DSC composed of four indoor units and one outdoor unit was developed. And the precision of the model was validated by experiments. Then, energy consumption performance was analyzed and energy consumption model was built based on simulation results.

In summary

1. Lumped parameter model of compressor and EEV, and district lumped parameter model of condenser and evaporator in multi-unit AC system with DSC program ensured its simulation accuracy. The errors between system hourly energy consumption simulated value and experimental data were within 10%. multi-unit AC system with DSC model showed good precision.

2. Simulation results indicated that multi-unit AC system with DSC hourly energy consumption deviations caused by indoor unit operating number were less than 15%, which can be neglected.

3. Simulation results indicated that the variation of $HW$ with $PLR$ and outdoor air temperature presented concave surface distribution, and the variations of $HEER$ and $HHPF$ with $PLR$ and outdoor air temperature presented convex surface distribution.

4. Multi-unit AC system with DSC hourly energy consumption calculation model was built based on simulation results of system hourly energy consumption values under various outdoor air temperature and part load ratio conditions. This model provides a tool for its energy saving optimization and seasonal energy consumption evaluation.
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