Total energy consumption optimization design of surface-water source heat pump systems

Min Liu¹, Ronghua Wu¹ and Qirong Yang¹,c

¹Institute of Mechanic and Electronic Engineering, Qingdao University, 308 Ningxia Road, Qingdao, Shandong, China 266071

a lm664083242@sina.com, b wuronghua18@126.com, c luyingyi125@163.com

Keywords: Transmission and distribution energy consumption(TDEC), Optimization design, Variable Flow(VF), Constant Flow(CF)

Abstract. This paper deals with the study of transmission and distribution energy consumption (TDEC) optimization design about surface-water source heat pump system. Three mathematical model have been established. The fitted polynomials of \( \text{COP} \), \( N_e \), \( N_c \) were obtained by the MATLAB curve fitting toolbox according to datum from product samples. The differences between two operating states which are constant flow and variable flow were analyzed and compared. Under variable flow operating state it was found that there exists an maximum energy conversation rate 12.68% of TDEC; the heat pump unit will consume more than average 8.41% energy, while \( \text{COP} \) will decline average 7.61%, the mean energy conversation rate of \( N_e \) and \( N_c \) are 49.38% and 38.86%, the average declination rate of \( t_{\text{cin}} \) and \( t_{\text{cout}} \) are 17.09% and 5.73% compared with constant flow operating state.

Introduction

With the rapid development of global economy, energy and environment problems are becoming more and more serious. In the whole energy consumption, building energy consumption possess a large proportion, and Heating Ventilation and Air Conditioning(HVAC) part occupies a lot[1]. As a kind of energy consumption and environment protection technology, SWSHP(Surface-water source heat pump) can greatly reduce air conditioning energy consumption, taking into consideration of a tendency that people pay more and more attentions to the indoor environment quality, SWSHP will get more and more applications. However in the current SWSHP systems there exist “ two low and one high ” phenomenons: low efficiency of heat pump units, low water pump efficiency and high operating energy consumption. In order to solve the above problems, it is necessary to optimize TDEC(transmission and distribution energy consumption) of SWSHP system so that it will truly realize energy saving and environment protection goal.

This paper will analyze the energy consumption difference between two operating conditions of SWSHP system: constant flow and variable flow. When heat pump unit keeps operating in a constant flow state, it will maintain high \( \text{COP} \) compared with variable flow state[2,3], however this will lead more pump energy consumption. Reference[4] put forward that under variable flow state it will be more energy efficient compared with constant flow state. It is filled with contradiction between high \( \text{COP} \) and low water pump energy consumption, the purpose of this paper is to solve this contradiction through optimization to TDEC by MATLAB optimization toolbox.

SWSHP system components

A integral SWSHP system contains there subsystems: Refrigerant cycle system, chilled water system, cooling water system, the flow chart of SWSHP system is shown in Fig.1.

- Refrigerant cycle system: it includes the following four components: compressor, condenser, evaporator, expansion value. The compressor needs to consume energy in order to maintain system cycle.
Chilled water system: chilled water pump, evaporator and pipeline are the main components. Chilled water pump provides hydraulic press to overcome resistance of evaporator and pipeline.

Cooling water system: cooling water pump, condenser make a loop through pipeline. The cooling water from surface water with lower temperature can absorb heat from high temperature refrigerant vapor so that the surface water can be regarded as one kind of environment protection heat sink.

Model establishment

**Heat pump model.** In summer air conditioning conditions, heat pump carries through reverse Carnot cycle, COP (Coefficient of performance) is proportional to \( t_0 \) and inverse proportional to \( t_k \)[5], when \( V_e \) and \( V_c \) keep constant, the mathematical model of COP can be described by formula (1):

\[
COP_1 = \alpha_1 \exp (\alpha_2 t_0 + \alpha_3 t_k) + \alpha_4 t_0/t_k + \alpha_5.
\]

When \( t_0 \) and \( t_k \) keep constant, COP can be described by formula (2):

\[
COP_2 = \beta_1 \exp (\beta_2 V_e + \beta_3 V_c) + \beta_4 V_e/V_c + \beta_5.
\]

Where, \( t_0 \) and \( t_k \) are chilled water outlet temperature and cooling water inlet temperature respectively, \( V_e \), \( V_c \) are flow rate of chilled water and cooling water respectively, \( \alpha_1-\alpha_5 \) and \( \beta_1-\beta_5 \) are fitted coefficients.

Under the actual operating conditions, the above parameters \( t_0 \), \( t_k \), \( V_e \), \( V_c \) are not always keep constant, their values can be changed according to load rate of users. Taking into account the variation of users’ load rate–\( R \), the mathematical model of COP can be described by formula (3):

\[
COP = [\alpha_1 \exp (\alpha_2 t_0 + \alpha_3 t_k) + \alpha_4 t_0/t_k + \alpha_5] / \beta_6.
\]

Where \( \alpha_1-\alpha_5 \), \( \beta_1-\beta_6 \) represent fitted coefficients.

**Chilled water pump model.** In theory, energy consumption of chilled water system concludes water pump energy consumption, cooling loss etc. In order to simplify calculation, it can only consider the major energy consumption—chilled water pump energy consumption. In general, chilled water system is closed system. It needs a minimal differential pressure \( \Delta H_{\text{min}} \) to ensure that the most unfavorable terminal device can work normally. The pipeline characteristic curve, chilled water pump energy consumption and volume flow rate can be expressed respectively by formula (4), (5) and (6):

\[
H_e = \Delta H_{\text{min}} + S_e V_c^2.
\]
\[ V_e = \frac{Q_e}{\rho C_p (t_{\text{ein}} - t_i)} . \]  \hfill (5)

\[ N_e = \frac{\rho g V c^2 H_e}{\eta_e} . \]  \hfill (6)

Where, \( H_e, N_e, \eta_e \) are lift, power and total efficient of chilled water pump respectively. \( Q_e \) is cooling load in the evaporator side. \( \rho \) and \( C_p \) are density and specific heat capacity of water. \( g \) is gravitational acceleration, \( S_e \) is impedance of chilled water pipeline. \( t_{\text{ein}} \) is inlet water temperature of evaporator.

For a given actual project, \( \Delta H_{\text{emim}} \), \( Q_e \) and \( S_e \) can be approximately regarded as constant, and it will satisfy the engineering demand. Meanwhile \( \eta_e \) is the function on \( V_e \). Through above simplification it was found that \( N_e \) is only connected with \( V_e \). So it is easy to get a polynomial of \( N_e \) on \( V_e \) through taking advantage of cftool (curve fitting toolbox) of MATLAB. The polynomial can be shown by formula (7):

\[ N_e = \gamma_0 + \gamma_1 V_e + \gamma_2 V_e^2 + \gamma_3 V_e^3 . \]  \hfill (7)

Where \( \gamma_0 - \gamma_3 \) represent fitted coefficients.

**Cooling water pump model.** It is known that cooling water system is a open system, there exists a height difference \( \Delta H_0 \) between water intake surface and water drainage surface. It is feasible that water pump energy consumption is only considered after ignoring energy dissipation and other loss.

According to energy conservation principal the function of \( V_c \) on \( Q_e \) and \( COP \) can be deduced. The pipeline characteristic curve, chilled water pump energy consumption and volume flow rate can be expressed respectively by formula. The following formula (8), (9), (10), (11) can be used to calculate \( N_c \):

\[ H_e = \Delta H_0 + S_c V_c^2 . \]  \hfill (8)

\[ V_c = \frac{Q_e + W_{co}}{\rho C_p (t_{\text{cout}} - t_k)} . \]  \hfill (9)

\[ W_{co} = \frac{Q_e}{COP} . \]  \hfill (10)

\[ N_c = \frac{\rho g V c^2 H_e}{\eta_e} . \]  \hfill (11)

Where, \( H_e, N_c, \eta_c \) are lift, power and total efficient of cooling water pump respectively. \( S_c \) is impedance of cooling water pipeline. \( t_{\text{cout}} \) is outlet water temperature of condenser. \( W_{co} \) is energy consumption of heat pump unit.

For a given actual project, \( \Delta H_0 \), \( Q_e \) and \( S_c \) can be approximately regarded as constant, and it will satisfy the engineering demand, meanwhile \( \eta_c \) is the function on \( V_c \). Through above simplification it was found that \( N_c \) is only connected with \( V_c \). The polynomial of \( N_c \) on \( V_c \) can be obtained by curve fitting toolbox of MATLAB. The polynomial can be shown by formula (12):

\[ N_c = \lambda_0 + \lambda_1 V_c + \lambda_2 V_c^2 + \lambda_3 V_c^3 . \]  \hfill (12)

Where \( \lambda_0 - \lambda_3 \) represent fitted coefficients.

**TDEC model.** Through above three mathematical model TDEC of SWSHP system can be expressed by formula (13):

\[ N_t = W_{co} + N_e + N_c . \]  \hfill (13)
Where $N_t$ is the sum of heat pump unit, chilled water pump and cooling water pump energy consumption.

It is obvious that $N_t$ is determined by six parameters: $V_c$, $V_e$, $\tau_{ein}$, $t_0$, $t_k$, $t_{cout}$, through analyzing formula (13), it is a multi-parameter function which includes some nonlinear and linear constraints such as mass conservation, energy conservation and flow limit of heat pump unit etc. And this optimization problem can be solved by MATLAB optimization toolbox.

**Optimization function analysis.** In the MATLAB optimization toolbox the fmincon function can solve such problems very well[6,7]. It contains four optimization algorithm: Trust region reflective, Active set, Interior point and SQP. It needs to create two M-file which stand for objective function and constraints respectively. The invoking format is shown as formula (14):

$$[x, fval, exitflag, output] = \text{fmincon}(@myfun, x0, A, b, Aeq, beq, lb, ub, @mycon).$$

Where, $x$ represents variations, $fval$ represents function value, $exitflag$ represents the optimizing final state, $output$ represents outputting the optimization message, $myfun$ represents objective function expression, $x0$ is a initial value so that the calculation can begin, $A$ and $b$ represent linear inequality, $Aeq$ and $beq$ represent linear equality, $lb$ and $ub$ represent the lower and upper bound, $mycon$ represents nonlinear inequality.

**Project case analysis.** Here an actual project case was analyzed. This building which is used to office and dormitory is located in Qingdao China. In the northwest of the building there is an artificial lake whose dimensions are 80m long × 30m wide × 6m deep, annual mean water yield is roughly 8600 cubic meter, the maximum and minimum water temperature of the lakebed are 3°C and 28°C.

The total construction area of this project is 3750 square meter. Air conditioning cooling load is 300kW, the water-source heat pump unit is TECKA SRSW-90-1, the cooling water pump is KQB 80-125, and chilled water pump is KQB 80-160B. In the air conditioning water system variable flow primary pumps form is adopted, the terminal devices is fan coil. The design parameters of this SWSHP system are shown in Table.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_e$ [kW]</td>
<td>300</td>
<td>$Q_c$ [kW]</td>
<td>354.4</td>
</tr>
<tr>
<td>$H_e$ [m]</td>
<td>22.17</td>
<td>$H_c$ [m]</td>
<td>17.77</td>
</tr>
<tr>
<td>$S_e$ [h$^2$/m$^5$]</td>
<td>0.0072</td>
<td>$S_c$ [h$^2$/m$^5$]</td>
<td>0.0037</td>
</tr>
<tr>
<td>$\Delta H_{emin}$ [m]</td>
<td>3</td>
<td>$\Delta H_0$ [m]</td>
<td>4</td>
</tr>
<tr>
<td>$t_0$ [°C]</td>
<td>7</td>
<td>$t_k$ [°C]</td>
<td>27</td>
</tr>
<tr>
<td>$t_{ein}$ [°C]</td>
<td>12</td>
<td>$t_{cout}$ [°C]</td>
<td>32</td>
</tr>
<tr>
<td>$V_e$ [m$^3$/h]</td>
<td>51.6</td>
<td>$V_c$ [m$^3$/h]</td>
<td>61</td>
</tr>
</tbody>
</table>

According to formula (3) and correction coefficients Table.2 and Table.3 of heat pump unit sample, fitted values of $\alpha_i$ and $\beta_i$ (i=1~6) were obtained as follow: the row vector of $\alpha=[-0.06178 -0.02769 0.1266 1.98909 5.455514]$, $\beta=[-5.9920 -0.02963 -0.00319 -0.6669 8.05699 5.511]$. The variation tendency of COP$_1$ with $t_0$, $t_k$ and COP$_2$ with $V_e$, $V_c$ can be obtained which were shown in Fig.2 and Fig.3 respectively.

<table>
<thead>
<tr>
<th>$t_{ein}$ [°C]</th>
<th>$t_0$ [°C]</th>
<th>$t_k$ [°C]</th>
<th>$t_{cout}$ [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>14</td>
<td>18</td>
<td>21</td>
</tr>
<tr>
<td>21</td>
<td>25</td>
<td>29</td>
<td>32</td>
</tr>
<tr>
<td>$Q_e$</td>
<td>$W_{co}$</td>
<td>$Q_c$</td>
<td>$W_{co}$</td>
</tr>
</tbody>
</table>
By observing Fig. 2 and Fig. 3 it was found that COP₁ will increase with increase of $t₀$, $tₖ$, and COP₂ will increase with increase of $Vₑ$, $V_c$. It can be explain by that when flow rate increases, it can improve the fluid velocity, thus it will enhance heating transfer effect which is beneficial to improving COP.

According to the samples of chilled water pump and pipeline characteristic curve, the values of $γ₀-γ₃$ can also be calculated, then put the results into formula (7):

$$Nₑ = 0.11 + 0.0066Vₑ + 0.00034Vₑ² + 2.6e−05Vₑ³.$$  (15)

Similarly $λ₀-λ₃$ can be obtained by cooling water pump sample and pipe characteristic curve, then put the results into formula (12):

$$N_c = 0.11 + 0.012V_c + 0.00015V_c² + 1.4e−05V_c³.$$  (16)

The variation tendency of $Nₑ$ and $N_c$ are shown in Fig.4.

**Table.3 Cooling performance correction coefficient table of variable flow**

<table>
<thead>
<tr>
<th>$Vₑ$[$m³/h$]</th>
<th>31.04</th>
<th>43.45</th>
<th>55.87</th>
<th>68.28</th>
<th>80.70</th>
</tr>
</thead>
<tbody>
<tr>
<td>28.96</td>
<td>0.9272</td>
<td>0.9839</td>
<td>1.0180</td>
<td>1.0417</td>
<td>1.0592</td>
</tr>
<tr>
<td>34.74</td>
<td>0.9693</td>
<td>1.0286</td>
<td>1.0642</td>
<td>1.0890</td>
<td>1.1073</td>
</tr>
<tr>
<td>40.25</td>
<td>1.0030</td>
<td>1.0643</td>
<td>1.1012</td>
<td>1.1268</td>
<td>1.1458</td>
</tr>
<tr>
<td>46.30</td>
<td>1.0303</td>
<td>1.0934</td>
<td>1.1312</td>
<td>1.1576</td>
<td>1.1770</td>
</tr>
<tr>
<td>52.00</td>
<td>1.0524</td>
<td>1.1168</td>
<td>1.1555</td>
<td>1.1824</td>
<td>1.2023</td>
</tr>
</tbody>
</table>

**Fig.2 COP₁ with different $t₀$, $tₖ$**

**Fig.3 COP₂ with different $Vₑ$, $V_c$**
It is obvious that both $N_e$ and $N_c$ reduce with the decrease of $V_e$ and $V_c$ respectively, it can illustrate that when variable flow condition was adopted there will save quite lots of energy consumption when terminal load of users begins to decrease.

**Comprehensive optimization.** According to above results, firstly it is necessary to create two M-file, then list various kinds of equality constraints and inequality constraints. The range of the six parameters $V_e$, $V_c$, $t_{ein}$, $t_0$, $t_k$, $t_{cout}$ are shown in Table.4.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_e$[m$^3$/h]</td>
<td>20-62</td>
<td>$t_k$[°C]</td>
<td>20-30</td>
</tr>
<tr>
<td>$V_c$[m$^3$/h]</td>
<td>15-52</td>
<td>$t_{cout}$[°C]</td>
<td>28-40</td>
</tr>
<tr>
<td>$t_{ein}$[°C]</td>
<td>8-15</td>
<td>$t_{ein}$−$t_0$[°C]</td>
<td>0-10</td>
</tr>
<tr>
<td>$t_0$[°C]</td>
<td>5-9</td>
<td>$t_{cout}$−$t_k$[°C]</td>
<td>0-11</td>
</tr>
</tbody>
</table>

$COP$, $W_{co}$, $N_e$, $N_c$ under constant flow rate condition can be obtained by Table.1, and the energy consumption results are shown in Table.5.

<table>
<thead>
<tr>
<th>$R$[%]</th>
<th>$N_e$[kW]</th>
<th>$N_c$[kW]</th>
<th>$W_{co}$[kW]</th>
<th>$N_i$[kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>4.959</td>
<td>4.596</td>
<td>58.254</td>
<td>67.809</td>
</tr>
<tr>
<td>90</td>
<td>4.959</td>
<td>4.596</td>
<td>52.428</td>
<td>61.983</td>
</tr>
<tr>
<td>80</td>
<td>4.959</td>
<td>4.596</td>
<td>46.603</td>
<td>56.158</td>
</tr>
<tr>
<td>70</td>
<td>4.959</td>
<td>4.596</td>
<td>40.777</td>
<td>50.332</td>
</tr>
<tr>
<td>60</td>
<td>4.959</td>
<td>4.596</td>
<td>34.952</td>
<td>44.507</td>
</tr>
<tr>
<td>50</td>
<td>4.959</td>
<td>4.596</td>
<td>29.127</td>
<td>38.682</td>
</tr>
<tr>
<td>40</td>
<td>4.959</td>
<td>4.596</td>
<td>23.301</td>
<td>32.856</td>
</tr>
<tr>
<td>30</td>
<td>4.959</td>
<td>4.596</td>
<td>17.476</td>
<td>27.031</td>
</tr>
</tbody>
</table>

Taking into account of indoor temperature and humidity requirements $t_0$ and $t_k$ should be regarded as constant whose values are 7°C and 27°C respectively, meanwhile the load rate $R$ need to be 30% at least.

The optimization results are shown in Table.6.
The differences between two operating conditions VF and CF are shown in Fig. 5.

![Comparison of NT under constant flow and variable flow](image1)

![The change of Vc and Vc with load rate R](image2)

![Comparison of Wco under constant flow and variable flow](image3)

![The change of COP with load rate R under variable flow](image4)

![The change of Ne and Nc with load rate R](image5)

![The change of tein and tcout with load rate R](image6)

**Fig. 5.** a) Comparison of Ne under constant flow and variable flow; b) The change of \( V_c \) and \( V_c \) with load rate \( R \); c) Comparison of \( W_{co} \) under constant flow and variable flow; d) The change of COP with load rate \( R \) under variable flow; e) The change of \( N_e \) and \( N_c \) with load rate \( R \); f) The change of \( t_{ein} \) and \( t_{cout} \) with load rate \( R \).
**Figure analysis.** a) via optimization the total energy consumption of SWSHP system get a decrease compared with constant flow state, the maximun decrease percentage reaches 12.68%, it is proved that variable flow water system can save energy consumption; b) when load rate $R$ reduces, $V_e$ and $V_c$ also go down which will lead a declination to heat exchange amount of evaporator and condenser so that the system can adapt the change of $R$; c), d) as load rate $R$ reduces, it will lead a declination to $COP$, while $W_{co}$ has a rising tendency, and it will consume more energy of heat pump unit compared with constant flow, it is because that when volume flow reduces, it will weaken heat-exchanging effect which will lead a declination to $COP$; e) as load rate $R$ reduces, energy consumption of water pump will have a huge decrease, the smaller $R$ is, the more energy will save, the reason is that both of $N_e$ and $N_c$ are proportion to $V_e^3$ and $V_c^3$ respectively; f) as load rate $R$ reduces, both of $t_{ein}$ and $t_{cout}$ get reduced, this can be explained by that the temperature difference will reduce with the declination of water flow which is proportion to $R$.

**Conclusions**

In this paper, through the establishment of three mathematical model which are heat pump unit, chilled water pump and cooling water pump respectively, the curve fitting box was used to get fitted coefficients as well as an minimal TDEC through optimization tool box. Under the condition of that $t_0$ and $t_k$ keep constant as 7°C and 27°C respectively, the results were shown as following:

(1) Under the variable flow operating state there exists an maximum energy conversation rate 12.68% of TDEC compared with the constant flow operating state, moreover with the decrease of $R$, it has more scope to save energy.

(2) With the reduction of $R$, under variable flow operating state the heat pump unit will consume more than average 8.41% energy, while $COP$ will decline average 7.61% compared with constant flow.

(3) With the reduction of $R$ both of the $V_e$ and $V_c$ will decrease so that there will have a energy saving of $N_e$ and $N_c$. When $R$ ranges from 30% to 100%, the mean energy conversation rate of $N_e$ and $N_c$ are 49.38% and 38.86%.

(4) With the reduction of $R$, both of $t_{ein}$ and $t_{cout}$ decrease due to the declination of temperature difference. When $R$ ranges from 30% to 100%, the average declination rate of $t_{ein}$ and $t_{cout}$ are 17.09% and 5.73%.

**Acknowledgements**

This work was financially supported by the National Science & Technology Program during the 12th Five-year-plan period of China (sq2011sf14b00503sq) and Science & Technology Benefiting People Program of China (s2013gmc620002).

**References**


