Application of Genetic Algorithms for Optimization of Condenser Water Loop in HVAC Systems

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ABSTRACT
This paper presents an engineering application of optimization method for energy conservation in condenser water loop operation in HVAC (Heating, Ventilation and Air Conditioning) systems. Based on the analysis of component model characteristics and interactions within the cooling tower and between towers and chillers, an objective function, which minimizes the operating costs through optimal set point changes of controlled variables along with changes of outdoor environment and indoor cooling load demands, is proposed. A modified genetic algorithm is employed to optimize the condenser water loop in HVAC systems. The approach has the capability of solving the combinatorial optimization problem with both discrete variables and continuous variables. Compared with the conventional method, this method has potential to substantially reduce the operating costs and is suitable to deal with both single chiller and multi-chiller central plants. An application example is given to show the advantages of this method.

Keywords: Engineering Application, Genetic Algorithms, Centralized HVAC System, Condenser Water Loop.

1. INTRODUCTION
A typical centralized Heating, Ventilation and Air-conditioning (HVAC) system comprises of condenser water loop, together with chillers, indoor air loops and chilled water loop to provide comfort environment for the conditioned space. The process of a condenser water loop consists of condensers, condenser water piping and pumps, and cooling towers. Condensers transfer indoor cooling load and heat generated by compressors into condenser water pipe where condenser water pumps provide the energy to overcome the friction loss and deliver condenser water to cooling towers. Cooling towers reject heat to the environment through heat transfer and evaporation to the ambient air.

The control and optimization has been considered as one of the most difficult problems for practice control engineers. Condenser water loop is one of the five main function loops and its operation has significant effect to the overall system performance. The main issues for efficient operation of condenser water loop are the performance of cooling towers as well as the interactions between cooling towers, condenser water pumps and chillers. For cooling towers, Cassidy and Stack’s work [1] showed that varying the speed of cooling tower fans is a good way to reduce energy consumption. Braun and Doderrich [2] proposed Near-optimal control of cooling towers based on parameter estimated from design data. This method was further adopted by Cascia [3] to simplify the component model and provide equations for determining the set points of near-optimal control. For interactions between chillers and cooling towers, Shelton and Joyce’s work [4] recommended 1.5 gpm/Ton condenser water flow rate as an optimal solution. Later, Kirsner [5] pointed out that high condenser water flow rate (3 gpm/Ton) has good performance at full load condition, while low condenser water flow rate (1.5 gpm/Ton) has advantages at part load conditions. Michael and Emery [6] analyzed the cost-optimal selection of the cooling tower range & approach and provided the design information for hermetic centrifugal and reciprocating chillers. Schwedler [7] studied the variation of condenser water supply temperature and used several examples to demonstrate his main idea that the lowest possible leaving tower water temperature does not always conserve system energy. However, his research only considered the fans with half-speed and full-speed conditions, which was not conclusive. In spite of the progress made in recent years on individual component modeling, control, optimization and operation rules, there is still lack of a systematic approach which consider the condenser water loop as a whole to optimize its operation.

In this paper, we present a novel real time optimization strategy for condenser water loops. By considering characteristics of cooling towers, effects of different ambient environment, interactions between chillers and cooling towers, the energy efficiency of condenser water loop can be formulated as an standard optimization problem. The total energy cost can be minimized by optimizing set points of manipulated variables, such as the flow rate of condenser water, airflow rate in cooling towers and condenser water supply temperature, etc. A modified genetic algorithm consists of binary-coded string population and three basic genetic operators: selection, crossover and mutation is used to search the optimal solutions. This method is suitable to deal with both single chiller and multi-chiller central plants and has potential to substantially reduce the operating costs.

2. ANALYSIS OF CONDENSER WATER LOOPS
2.1 Optimal Operating Range of Cooling Towers
The Braun’s model [8] of the cooling tower is adopted here to calculate the cooling capacity.

\[ Q_{\text{op}} = \varepsilon \cdot m \cdot (h_{\text{w,op}} - h_{\text{w,in}}) \]  

(1)

where,

\[ \varepsilon = \frac{1}{1 - e^{-NTU \cdot (1 + \varepsilon)}} \quad \text{and} \quad NTU = C \left( \frac{m}{m_w} \right)^{1-\varepsilon} \]

\[ m^* = \frac{m}{m_w} \cdot \frac{C_w}{C_{pw}} \quad \text{and} \quad C_w = \left( \frac{dh}{dT} \right) T = \frac{h_{\text{w,op}} - h_{\text{w,in}}}{T_{\text{CWRI}} - T_{\text{CWRI}}} \]

To illustrate the relationships between condenser water flow rate and airflow rate inside a cooling tower, five equal heat rejection rate curves are constructed as shown in Figure 1, where the x-axes is water flow rate and y-axes is airflow rate inside the tower.

Figure 1. The Performance of a Cooling Tower

To analysis the performance, we divide each curve into three portions as indicated in Figure 1.

- In Portion (1), the airflow rate is very small and water flow rate must be very large to achieve given heat rejection rate. In this case, the airflow rate is too small to exchange heat efficiently with condenser water.
- In Portion (2), the airflow rate is very large while the water flow rate is very small. In this portion, the heat exchange is saturated and the outlet water temperature is nearly equal to ambient wet-bulb temperature.
- Portion (3) is in the middle of Portions (1) and (2). In this portion, the heat rejection rate of cooling tower increases with either the increased airflow rate or increased water flow rate, and vice versa.

Apparently, the operating point of cooling tower must lie in the region of Portion (3) for energy efficiency. In this portion, reduced airflow rate leads to lower tower fan power consumption. However, in order to achieve the given heat rejection rate, the water flow rate has to be increased and that results in increased pump power consumption. Similarly, reduced water flow rate will cause increased fan power consumption. Therefore, there exists an optimal operating point within the cooling tower for the given heat rejection rate.

2.2 Effect of \( T_{\text{wb}} \) to Cooling Tower Performance

Discussion in Section 2.1 is based on a fixed ambient wet-bulb temperature. When the ambient wet-bulb temperature changes, the optimal operating point of cooling towers will change accordingly. In addition to cooling tower model of Eq. (1), the power consumption of pumps and fans, \( P_{\text{pump}} \) and \( P_{\text{fan}} \), with VSD proposed by Kreider and Rabl [9]

\[ P_{\text{pump}} = \sum \frac{P_{\text{pump},1}}{T} \left( d_0 + d_1 \frac{m_{\text{w,op}}}{m_{\text{w,op}}} + d_2 \frac{m_{\text{w,op}}}{m_{\text{w,op}}}^2 + d_3 \frac{m_{\text{w,op}}}{m_{\text{w,op}}}^3 \right) \]

and

\[ P_{\text{fan}} = \sum \frac{P_{\text{fan},1}}{T} \left( e_0 + e_1 \frac{m_{\text{w,op}}}{m_{\text{w,op}}} + e_2 \frac{m_{\text{w,op}}}{m_{\text{w,op}}}^2 + e_3 \frac{m_{\text{w,op}}}{m_{\text{w,op}}}^3 \right) \]

(2a)

are also adopted here to illustrate different optimal points.

For illustration purpose, it is assumed that the heat rejection rate of the cooling tower is 400 kW under two different ambient wet-bulb temperatures, 25 °C and 30 °C, respectively. The optimal operating points are labeled in Figure 2 to indicate the corresponding airflow rate and power consumption of fan and pump. The optimal airflow rate is 13 kg/s at 25 °C and 9 kg/s at 30 °C. If the airflow rate is kept at 13 kg/s at 25 °C instead of 9 kg/s, the power consumption of fan and pump is 6.11 kW. Compared with power consumption of 5.26 kW at 9 kg/s at optimal point, almost 16% energy can be saved when the ambient wet-bulb temperature is changed from 25 °C to 30 °C. This implies that the optimal operating points are different for the same heat rejection rate under different wet-bulb temperature.

Figure 2. The Optimal Operating Point at Different \( T_{\text{wb}} \)

2.3 Effect of \( T_{\text{CWS}} \) to Chiller COP

In order to determine the interaction between chillers and cooling towers, it is necessary to find the relationship between condenser water supply temperature, \( T_{\text{CWS}} \), and coefficient of performance (COP) of chiller. Figure 3 illustrates the effect of condenser water supply temperature to chiller COP using manufacturer’s data [10].

The condenser water temperature entering the chiller affects the chiller capacity and hence the chiller power consumption. The higher \( T_{\text{CWS}} \) leads to lower chiller efficiency, whereas, lower \( T_{\text{CWS}} \) results in higher chiller efficiency. Therefore, reducing condenser water supply temperature can reduce chiller energy consumption. However, this will be penalized by increased power usage in cooling tower side in order to achieve the lower \( T_{\text{CWS}} \) condition.
2.4 Interaction between Chillers and Cooling Towers

As the airflow and condenser water flow increase, the fan power and condenser water pump power increase cubically. It has also been generally acknowledged [2, 4, 5, 7, 11, 12], \( T_{CWS} \) decreases resulting in higher COP and lower energy consumption of chillers. On the other hand, however, condenser water return temperature in turn affects the heat exchange efficiencies in cooling towers. When condenser water supply temperature decreases, condenser water return temperature also decreases for the same cooling load. This results in the lower efficiencies of the cooling tower under the same ambient wet-bulb temperature as the enthalpy difference between ambient air and condenser water becomes smaller. The optimal operating point occurs at a point where the rate of power increment in fans and pumps is equal to the rate of power reduction in chillers.

To illustrate the relationship between total power consumption and condenser water supply temperature, let’s look at the total power consumption of a condenser water loop with ambient wet-bulb temperature and cooling load are at 77 °F (25 °C) and 400 Ton, respectively. The total energy consumption for different condenser water supply temperature is given in Table 1.

Here, it clearly shows that the power consumption of chiller increases with the increased condenser water supply temperature for the fixed condenser water flow rate. Meanwhile, fans consume less energy with the increased temperature set points of condenser water supply and the best operating point is 85 °F rather than 82 °F (the lowest condenser water supply temperature).

### Table 1. Relationship between \( T_{CWS} \) and \( P_{\text{total}} \)

<table>
<thead>
<tr>
<th>( T_{CWS} ) (°F)</th>
<th>( m_a ) (kg/s)</th>
<th>( m_{a,k} ) (kg/s)</th>
<th>( P_{\text{chiller}} ) (kw)</th>
<th>( P_{\text{fan}} ) (kw)</th>
<th>( P_{\text{pump}} ) (kw)</th>
<th>( P_{\text{total}} ) (kw)</th>
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</tbody>
</table>

3. PROBLEM FORMULATION

The schematic block diagram of a condenser water loop is shown as in Figure 4.

Without loss of generality, it is assumed that both chilled water supply temperature, \( T_{CHW, S} \) and chilled water flow rate inside chiller, \( m_{CHW,S} \), are kept as constants, whereas the chilled water return temperature, \( T_{CHW, R} \), varies with the cooling load in the conditioned space. The manipulated variables of the system for control and optimization are as follows:

- The numbers of operating chillers \( i \), condenser water pumps \( j \), and cooling tower fans \( k \).
- The condenser water flow rate \( m_{a,i} \), controlled by the \( i \)th Variable-Speed Drive (VSD) condenser water pump.
- The airflow rate in cooling towers \( m_{i,j} \), controlled by the \( j \)th VSD cooling tower fan.
- The condenser water supply temperature, \( T_{CWS} \), controlled by the combination of total condenser water flow rate \( m _a \) and airflow rate \( m _a \) in cooling towers.

3.1 Objective Function

The objective is to minimize total power consumption of the whole condenser water loop, and is given as

\[
\text{min } P_{\text{total}} = P_{\text{chiller}} + P_{\text{pump}} + P_{\text{fan}}
\]  

In Eq. (3), the expression for power consumption of chillers \( P_{\text{chiller}} \), is given as Stoecker [13]:

\[
P_{\text{chiller}} = \sum_i Q_{b,i} \cdot \text{COP}_{\text{chiller}} \cdot \left(\text{PLR}_{\text{adj,i}} \cdot \left(\text{Temp}_{\text{adj,i}}\right)\right)
\]

\[
\text{PLR}_{\text{adj,i}} = b_k + b_k \left(\frac{Q_{\text{b,i}}}{Q_{\text{cap,i}}}\right) + b_k \left(\frac{Q_{\text{b,i}}}{Q_{\text{cap,i}}}\right)^2
\]

\[
\text{Temp}_{\text{adj,i}} = c_0 + c_1 T_{CHWS} + c_2 T_{CHWS}^2 + c_3 T_{CRS}^2 + c_4 T_{CHWS} T_{CRS}
\]

and the power consumption of pumps and fans, \( P_{\text{pump}} \) and \( P_{\text{fan}} \), with VSD are expressed as in Eq. (2a,2b).

3.2 Constraints

The objective function of Eq. (3) subjects to the following constraints:

Constraint (1): \( m_{v,i,\text{min}} \leq m_{v,i} \leq m_{v,i,\text{max}} \);

Constraint (2): \( Q = m_{CHW,S} \cdot C_{p} \cdot (T_{CHW,R} - T_{CHW,S}) \);

Constraint (3): \( Q + P_{\text{chiller}} = m_{CHW,S} \cdot C_{p} \cdot (T_{CRS} - T_{CHW,R}) \);

\( m_a = \sum_i m_{a,i} \)
Constraint (4): \( Q + P_{\text{chiller}} = Q_{\text{req}} \)

Constraint (1) limits all variables in their operating range. The cooling load is calculated by the Constraint (2) and used in Constraints (3) and (4). Constraint (3) demands the condenser transfer the corresponding heat to cooling towers. Constraint (4) implies that the heat is completely rejected to the environment by cooling towers.

For engineering application, we adopt following Equations to calculate the heat rejection rate \( Q_{\text{req}} \) [14]:

\[
Q_{\text{req}} = \sum_{i} Q_{\text{req},i} \tag{5a}
\]

\[
Q_{\text{req},i} = e_{i,a} m_{a,i} (h_{s,a,i} - h_{s,a}) \tag{5b}
\]

\[
e_{i,a} = a_{i} + a_{i} \left( \frac{m_{a,i}}{m_{a}} \right) + a_{i} (T_{\text{CWR}} - T_{\text{wb}}) + a_{i} \left( \frac{m_{a,i}}{m_{a}} \right)^{2} \tag{5c}
\]

\[
+ a_{i} (T_{\text{CWR}} - T_{\text{wb}})^{2} + a_{i} \left( \frac{m_{a,i}}{m_{a}} \right) (T_{\text{CWR}} - T_{\text{wb}}) \tag{5d}
\]

\[
h_{s,a,i} = f(T_{\text{CWR}}) \text{ and } h_{s,a} = f(T_{\text{wb}}, T_{\text{wb}}) \tag{5e}
\]

The single-chiller systems can be considered as a special case of multi-chiller installations, in which there is only one chiller, one condenser water pump and one cooling tower fan. The combination of multiple chillers, pumps, and fans is no longer required in the problem solving process.

4. OPTIMIZATION ALGORITHM

As this optimization problem is a combinatorial optimization problem with nonlinear constraints and contains both continuous and discrete variables, conventional gradient-based optimization methods cannot be applied directly. The exhaustive search methods or combined exhaustive search with conventional gradient-based methods, even though, can find the optimal solutions, is impractical for real time applications of such a complicated problem due to its time consuming nature. Genetic algorithm, on the other hand, is a function value directed random search technique and does not require gradient information of the function. Its ability to encode different kinds of variables into binary code strings and combine them into a chromosome without loss the engineering accuracy make it an ideal tool for such applications [15]. To apply genetic algorithms to this particular problem, following steps are required:

(a) Encoding

The variables \((m_{a,i}, m_{a}, T_{\text{CWS}})\) are changed into binary strings and are connected together to form a chromosome. The lower and upper bounds of the binary variables are the minimum and maximum values of set points shown in constraints (1). The upper bounds of the binary variables, for example all “1”, can be set as the maximum values of set points. The lower bounds of the binary variables, for example all “0”, are used to indicate the components are staged off. The lower bounds plus one are set as the minimum values of set points. Therefore, the constraint (1) is fulfilled in encoding phase.

(b) Constructing Fitness Function

In this step, a penalty function is added if any constraint cannot be fulfilled. The fitness function is expressed in the following equation,

\[
\text{fitness} = P_{\text{total}} + P_{1} + P_{2} + P_{3} \tag{6}
\]

with

\[
P_{1} = v_{1} \cdot (Q - m_{\text{CHWS}} C_{p,s}(T_{\text{CHWS}} - T_{\text{CHWR}}))^2
\]

\[
P_{2} = v_{2} \cdot (Q + P_{\text{chiller}} - m_{c} C_{p}(T_{\text{CWR}} - T_{\text{CRS}}))^2
\]

\[
P_{3} = v_{3} \cdot (Q + P_{\text{chiller}} - Q_{\text{req}})^2
\]

where \(v_{1}, v_{2}, \text{ and } v_{3}\) are the penalty multipliers.

(c) Evolution Operating

After the first two steps, some random binary numbers are used to form the initial population. With these initial values, genetic operators (such as: selection, crossover, and mutation) can be used to compute the optimal results iteratively. In order to implement the genetic algorithms in real time, the original algorithm is modified to reduce the searching space and consequently the computing time. The following is the algorithm in a pseudo code format.

Initialize the parameters of genetic algorithms;

Input the measured variables \((T_{\text{CHWS}}, T_{\text{CHWR}}, T_{\text{wb}}, \text{ etc.});\)

Restrict the searching space by input variables;

Randomly generate the old_population in restricted searching space;

For generation=1: max_generation;

Compute the fitness of each individual in the old_population;

Store the highest fitness of individual;

Use “roulette wheel” selection method to form mating_pool;

While individual_number<population_size do;

Select two parents from the mating_pool randomly;

Perform crossover and mutation operation to produce offsprings;

Place the offspring to new_population;

Endwhile

Replace the old_population by the new_population;

Replace the least fitness of individual in new_population by stored highest one;

Endfor

Output the individual of the highest fitness based on input variables in real time.

(d) Termination

After the optimal operating points have been determined by the modified genetic algorithm, they are compared with the modified genetic algorithm, they are compared with the current operating point before being put in force. This is a safety measure to prevent the uncertainties of the genetic algorithm due to insufficient evolution time. If such a condition occurs, the system will operate at present set points without any changes until the next sampling period.

5. AN APPLICATION EXAMPLE
To illustrate the advantages of the optimization methods for searching the optimal operating points, an application example is given below:

- A centralized air conditioning system has four chillers, four condenser water pumps, and four cooling towers.
- Four chillers have the same nominal cooling load capacity, 500 Ton.
- The chilled water supply temperature is 45 °F (7.2 °C).
- The nominal condenser water supply temperature is 85 °F (29.4 °C).
- The energy consumption of each chiller is 0.6 kW/Ton at the full load condition.
- Each chiller has one condenser water pump and its nominal water flow rate is 100 kg/s.
- Each cooling tower is equipped with a fan of 30 kW at full load and can provide 30 kg/s airflow rate.
- All pumps and fans are equipped with VSDs and their speed ranging from 50% to 150% of full load.

To compare with the optimal operating point control method, a commonly used increment/decrement chiller staging method with fixed condenser water flow rate and airflow rate control strategy is used with following operating conditions:

- The condenser water flow rate is fixed at 100 kg/s per condenser water pump and the pump is staging on with the corresponding chiller.
- The airflow rate in cooling tower is fixed at 30 kg/s when the cooling tower fan operating under the rated speed.

The cooling load and ambient wet-bulb temperature profiles are given in Figures 5 and 6, respectively.

![Figure 5. The Measured Cooling Load](image)

![Figure 6. The Ambient Wet-bulb Temperature](image)

The power consumption of the whole system with optimal operating point control is plotted in the same figure as that with commonly used method. The results are shown in Figure 7.

![Figure 7. Power Consumption of two methods](image)

It can be seen from Figure 7, the average energy consumption by using optimization method is nearly 10% less than that of commonly used control strategy. In practice, the energy savings resulted from operating points optimization may be different from system to system. In general, however, it can be expected that the proposed optimization method can substantially reduce the energy consumption of condenser water loops.

### 6. CONCLUSIONS

This paper proposed a new energy saving strategy for condenser water loop through system optimization. To clearly identify optimization opportunities, detailed characteristic analysis of system components showed the interactions of control variables inside the cooling tower and between the cooling towers and the chillers. Based on the analysis, a model-based optimization method was proposed and a modified genetic algorithm was employed to find the optimal solutions. An application example showed the effectiveness of the proposed method. As other loops in HVAC systems have similar characteristics, the methodology developed in this paper can be easily extended to the other function loops.

### Nomenclature

- $a_0$ ... $a_5$ the constant coefficients to determine $\varepsilon_{ack}$
- $b_0, b_1, b_2$ the constant coefficients to determine $PLR_{adj,i}$
- $c$ the empirical constant in Braun’s model
- $c_0$ ... $c_5$ the constant coefficients to determine $Temp_{adj,i}$
- $C_{pw}$ the specific heat of water under constant pressure
- $C_s$ the derivative of saturation air enthalpy with respect to temperature
- $COP_{nom,i}$ the nominal coefficient of performance of the $i$th chiller
- $d_0$ ... $d_3$ the coefficients of pumps for part-load condition
- $e_0$ ... $e_3$ the coefficients of fans for part-load condition
- $f(.)$ the function of the enthalpy of the moisture air
- $h_{a,in}$ the inlet air enthalpy of cooling tower
- $h_{s,w,i}$ the saturation air enthalpy at $T_{CWR}$
- $h_{s,w,o}$ the saturation air enthalpy at $T_{CHW}$
- $m_a$ the total airflow rate of all the cooling towers
\( m_{\text{air,k}} \) the nominal \( k \)th fan airflow rate at full-load
\( m_{\text{a,k}} \) the exact airflow rate provided by the \( k \)th operating tower fan
\( m_{\text{CHW}} \) the water flow rate in the chilled water loop
\( m_w \) the total water flow rate in condenser water loop
\( m_{\text{w,i}} \) the water flow rate through the \( k \)th cooling tower
\( m_{\text{vadj,i}} \) the nominal \( i \)th pump water flow rate at full-load
\( m_{\text{w,i}} \) the exact water flow rate provided by the \( i \)th operating pump
\( m^* \) the ratio of air to water effective capacitance rate in Braun’s model
\( n \) the empirical constant in Braun’s model
\( P_{\text{chiller}} \) the power consumption of the chillers
\( P_{\text{fan},j} \) the power consumption of the fans
\( P_{\text{pump}} \) the power of all pumps
\( P_{\text{fn},k} \) the \( j \)th nominal fan power at full-load
\( P_{\text{pump}} \) the \( j \)th nominal pump power at full-load
\( P_{\text{total}} \) the total power consumption of the system
\( \text{PLR}_{\text{adj},i} \) the adjustment factor of part-load ratio of \( i \)th the chiller
\( \text{PLR}_i \) the part-load ratio of the \( i \)th operating chiller
\( Q_{\text{cap},i} \) the nominal capacity of the \( i \)th chiller
\( Q \) the real cooling load provided by all the chillers
\( Q_i \) the real cooling load provided by the \( i \)th chiller
\( Q_{\text{euf}} \) the heat rejection rate of all the cooling towers
\( Q_{\text{euf},k} \) the heat rejection rate of the \( k \)th cooling tower
\( T_{\text{CHWS}} \) the chilled water supply temperature
\( T_{\text{CHWR}} \) the chilled water return temperature
\( T_{\text{CWS}} \) the condenser water supply temperature
\( T_{\text{CWR}} \) the condenser water return temperature
\( T_{\text{db}} \) the ambient dry-bulb temperature
\( T_{\text{wb}} \) the ambient wet-bulb temperature
\( \text{Temp}_{\text{adj},i} \) the adjustment factor of temperature of the \( i \)th chiller
\( v_0, v_1, v_2 \) the multipliers of penalty functions
\( \epsilon_{\text{euf},k} \) the heat transfer effectiveness in Braun’s model (1989) of the \( k \)th tower

References