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Using cooling load forecast as the optimal operation scheme for a large multi-chiller system

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ABSTRACT

Energy saving is one of the most important issues in high-tech manufacturing industries, such as semiconductor and electronics, because large chilled water systems are used to satisfy big cooling load requirements. In this paper, a new optimal integrity scheme based on a two-stage strategy, including a scheduling stage and an operating stage, is proposed to minimize the system energy consumption within a future time period. Instead of a lag scheme used in the general method, a forecasting scheme consisting of a series of optimal schemes at each sub-time period is also proposed for the two-stage design. The performance of the proposed method is examined through an industrial case. The cost of the proposed method is much less than that of the conventional method, so the proposed method is cost-efficient in applications of large air-conditioning systems.

Utilisation de la charge thermique pour prévoir le fonctionnement optimal d'un grand système à plusieurs refroidisseurs

Mots clés : Système ; Refroidisseur ; Économies d'énergie ; Modélisation ; Conception optimale

1. Introduction

In the high-technology manufacturing industry, such as semiconductor factories and electronics factories, the cooling load of the air-conditioning is heavy as the operation plant is strict with the cleanliness and the temperature of air. In a typical semiconductor plant, more than ten sets of chilled water units are needed to satisfy heavy-load requirements. The power consumption of the chilled water system accounts for about 60%–70% of the total costs of facility & utility systems in a hi-tech industry. Using the optimization control strategy has tremendous potential to reduce operating costs

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Nomenclature

A_{eva}	heat transfer area of the evaporator (m^2)
A_{con}	heat transfer area of the condenser (m^2)
$h_{a,i}$	enthalpy of air entering the cooling tower (kJ kg^{-1})
$h_{a,o}$	enthalpy of air leaving the cooling tower (kJ kg^{-1})
$h_{w,i,t}$	enthalpy of water entering the cooling tower (kJ kg^{-1})
$h_{w,o,t}$	enthalpy of water leaving the cooling tower (kJ kg^{-1})
$h_{s,w,i}$	enthalpy of saturation air corresponding to the temperature of water entering the cooling tower (kJ kg^{-1})
$h_{s,w,o}$	enthalpy of saturation air corresponding to the temperature of water leaving the cooling tower (kJ kg^{-1})
K_{eva}	heat transfer coefficient of the evaporator ($Wm^{-2}K^{-1}$)
K_{con}	heat transfer coefficient of the condenser ($Wm^{-2}K^{-1}$)
COP	coefficient of performance of the chiller
c_s	state variable
C_w	specific heat capacity of water (kJ $kg^{-1}K^{-1}$)
f_e	evaporator state variable
f_c	condenser state variable
N	number of transfer units of the cooling tower
m_a	mass flowrate of air entering the cooling tower (kg s^{-1})
$m_{a,r}$	rated flowrate of air entering the cooling tower (kg s^{-1})
m_{ew}	mass flowrate of chilled water (kg s^{-1})
m_{cw}	mass flowrate of water entering the condenser (kg s^{-1})
$m_{cw,i}$	mass flowrate of water entering the cooling tower (kg s^{-1})
$m_{cw,o}$	mass flowrate of water leaving the cooling tower (kg s^{-1})
m^*	state variable
P	atmospheric pressure
P_s	saturated water vapor pressure
P_c	power consumption of the chiller (kW)
P_{fan}	power consumption of the cooling tower fan (kW)
$P_{fan,r}$	rated power consumption of the cooling tower fan (kW)
$P_{pump,con}$	power consumption of the constant-speed pump (kW)
$P_{pump,var}$	power consumption of the variable-speed pump (kW)
PLR	partial load ratio of the chiller
Q_{eva}	heat transfer quantity of the evaporator (equivalent to the chiller cooling capacity) (kW)
Q_{cr}	rated cooling capacity (kW)
Q_{con}	heat transfer quantity of the condenser
$T_{a,w}$	wet-bulb temperature of air entering the cooling tower (°C)
T_e	evaporating temperature (°C)
T_c	condensing temperature (°C)
$T_{w,i,eva}$	chilled water returning temperature (°C)
$T_{w,o,eva}$	chilled water supply temperature (°C)
$T_{w,i,con}$	condenser water entering temperature (°C)
$T_{w,o,con}$	condenser water leaving temperature (°C)
$T_{w,i,t}$	temperature of water entering the cooling tower (equivalent to $T_{w,o,con}$) (°C)
$T_{w,o,t}$	temperature of water leaving the cooling tower (equivalent to $T_{w,i,con}$) (°C)
Δt	time interval
Z	binary variable
$\omega_{a,i}$	moisture content of air entering the cooling tower (kg (kg dry air)$^{-1}$)
$\omega_{a,o}$	moisture content of air leaving the cooling tower (kg (kg dry air)$^{-1}$)
ω_{awo}	moisture content of saturation air leaving the cooling tower (kg (kg dry air)$^{-1}$)
ϵ_a	airside heat transfer effectiveness of the cooling tower
γ_w	specific weight of water (kNm^{-3})
η_p	efficiency of the pump
η_f	efficiency of the variable-frequency driver

and increase energy efficiency (Wang and Ma, 2008; Gordon et al., 2000).

Many efforts have been undertaken to develop the optimal operation and control strategies for the heating, ventilation and air-conditioning (HVAC) systems in the residential buildings or office buildings (Wang and Ma, 2008; ASHRAE, 2007). Austin (1991) stated that the true optimum loading point of centrifugal chillers could lead to the increase of the chiller plant efficiency by 20% or more. A significant increase in operation efficiency is possible when a chiller optimum loading point is correctly determined. Later Gordon et al. (2000) developed a simple thermodynamic model for the chiller performance using the measured performance data. The model succeeded in predicting the fundamental relation between coefficient of performance (COP) and the cooling rate for the centrifugal chiller. Yao et al. (2004a) investigated a large cooling system of residential buildings. The relationships among the controlled variables, the uncontrolled variables and the chillers' performance were obtained empirically with the test data. These studies aforementioned are useful to comprehend the process and to improve the chiller performance, but these models were only applied under specified conditions.

Based on the studies of chillers' performance, many optimal operation strategies and solving methods have been developed. Hackner et al. (1984) presented an equal loading rate method to operate chillers. It is a conventional method and is still commonly used in HVAC systems. Chang (2004) used the Lagrangian method to solve the optimal chiller loading problem and to improve the deficiencies of conventional (equal loading rate) methods. Braun and Diderrich (1990) developed optimal and near-optimal control strategies using quadratic relationships for chiller systems. In the system based methodology, an overall empirical cost function of the total power consumption of a chiller plant was developed using a quadratic function. Nassif et al. (2005) used