OPTIMAL PHASE CHANGE TEMPERATURE FOR BUILDING COOLING HEATING AND POWER SYSTEM WITH PCM-TES BASED ON ENERGY STORAGE EFFECTIVENESS

by

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Original scientific paper
https://doi.org/10.2298/TSCI170222184Z

Integrating thermal energy storage (TES) equipment with building cooling heating and power (BCHP) system can improve system thermal performance. In this paper, a simplified model of TES-BCHP system composed of a gas turbine, an absorption chiller and TES equipment with PCM is presented. To evaluate the energy saving effect of PCM-TES, a new index, energy storage effectiveness, is proposed and its relationship with primary energy consumption is established. Aimed at maximizing the energy storage effectiveness, the optimal phase change temperature of the PCM-TES-BCHP system is obtained. The results show that the theoretically optimal phase change temperature is just the geometrical average value of ambient temperature and exhaust gas temperature from gas turbine for ideal PCM-TES equipment with infinite number of transfer units. It also indicates that both energy storage effectiveness and optimal phase change temperature increase with increasing number of transfer units. So improving the thermal performance of PCM-TES device is favorable for increasing energy efficiency and saving primary energy consumption accordingly. This work is of great importance in guiding the optimization design of practical phase change temperature of the PCM-TES-BCHP systems.

Key words: *co-generation, energy storage, thermal optimization, phase change material, energy efficiency*

Introduction

With the rapid development over the recent two decades, the global total energy consumption has grown by 49% [1]. Therein, buildings account for about 30% of total energy consumption and the percentage keeps increasing [2]. As a result, the increasing demand for cooling, heating, and power supplies in buildings appeals for resurveying traditional energy systems and stimulates the search for more high efficient and low emission energy production, conservation and utilization methods [3]. The BCHP is a novel kind of building energy supply system which can meet users' different load demands simultaneously with a single primary energy input [4]. Compared to traditional separated generation system, BCHP systems show high energy efficiency, low pollutions emission and good economic benefit [5]. However, the energy supply units in a BCHP system often show poor thermal performance under part load working conditions, due to the non-synchronized and fluctuating thermal and electrical demands [5-7].

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It is found that introducing TES equipment into BCHP systems proves to be an effective way to improve the part load performance of the whole system and saving the primary energy consumption [8]. Many researchers investigated the thermal and economic performance of the co-generation or tri-generation system with different types of TES equipment [9]. Khan *et al.* [10] integrated heat accumulator with combined heating and power system to match the hot water supply and demand through the dynamic charge and discharge processes of the TES equipment. Bogdan and Kopjar [11] and Campos *et al.* [12] conducted similar studies and found that profitability of co-generation system with TES equipment was impacted by various external factors. The results inferred that the water tanks might substantially improve the economic performance when electricity price was governed by the dual-time tariff policy. Furthermore, Katulic *et al.* [13] put forward a new approach to determine the optimal daily heat storage tank capacity for a co-generation system. Fu *et al.* [14] established the dynamic simulation model for stratified water storage and conducted experiment on the energy saving effect of combined cooling heating and power system. Bailey *et al.* [15] applied sensible TES equipment to the co-generation system and optimized its installed capacity based on completely mixing assumption for the water tanks.

On the other hand, compared to sensible heat storage (*e. g.*, water tank), latent heat storage with PCM is of relatively high energy storage density, which makes them increasingly attractive for applications [16]. Pitie *et al.* [17] presented the potential usage of PCM particles for high temperature energy capture and storage in industry fields through a fluidized bed. Zhang *et al.* [18] briefly reviewed the TES development, with special emphasis on the important applications of PCM in both solar energy projects and waste heat recovery from industrial processes. Zeng *et al.* [19] integrated PCM with building envelopes and optimized its thermal physical properties, in order to improve the indoor thermal comfort and reduce the energy consumption for passive buildings. Fiorentini *et al.* [20] applied PCM-TES to HVAC systems and found that the PCM tank can effectively shift the cooling load and increase the overall efficiency of a heat pump system for space cooling. However, even though PCM-TES application in BCHP system has a great potential for energy saving, relevant researches are not enough. Zhang *et al.* [21] proposed a new method to pre-estimate the feasibility of TES-BCHP system before design of practical systems, under ideal assumption that there is no irreversible loss during the heat transfer processes for the TES equipment. Chen *et al.* [22] evaluated the energy saving potential of BCHP system with latent TES equipment based on case study of practical fluctuating user loads.

Nevertheless, few researchers focused on the key parameter optimization, such as the phase change temperature of PCM-TES, even though it had a great influence on the performance of the whole PCM-TES-BCHP system. Therefore, how to determine the optimal phase change temperature for the PCM-TES-BCHP system is an important but unsolved problem. In this paper, a simplified model of PCM-TES-BCHP system is established and the analytical optimal phase change temperature is determined based on the proposed energy storage effectiveness. Moreover, the impact of number of transfer units (*NTU*) of the PCM-TES equipment is analysed to evaluate the energy saving effect of the whole system. This work can provide guidance for PCM-TES-BCHP system design.

Method

The BCHP system

The typical BCHP system under summer working condition is shown in fig. 1. The gas turbine (GT) is driven by natural gas (NG) and the mechanical energy is further changed into electricity power, which is then delivered to the users directly. At the same time, the absorption chiller (AC), activated by the high temperature exhaust gas, produces low temperature water to fulfil the cooling requirement. For the operation strategy, Teng *et al.* [23] found that following thermal load (*FTL*) was more energy-saving than following electrical load (*FEL*) for BCHP systems. Thus the system gives priority to meet cooling demand, and insufficient electricity can be bought from the power grid.

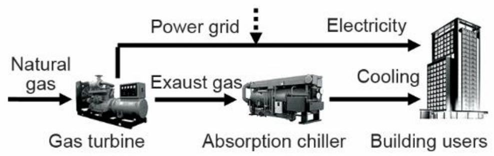

Figure 1. Typical BCHP system in summer working condition

So, for the whole BCHP system, the total primary energy consumption (*PEC*) comprises two parts: the consumed NG by the GT and the imported electricity from the power grid.

$$PEC = Q_{\mathrm{NG}} + \frac{Q_{\mathrm{E,grid}}}{\eta_{\mathrm{grid}}} = \frac{Q_{\mathrm{C}}}{[1-\eta_{\mathrm{GT}}]COP_{\mathrm{AC}}} + \frac{Q_{\mathrm{E}} - Q_{\mathrm{E,GT}}}{\eta_{\mathrm{grid}}} \tag{1}$$

In eq. (1), the bought electricity from the power grid is converted to the equivalent heat value of corresponding primary fuel (*e. g.* coal and NG) through the conversion parameter η_{grid}, the electricity generation efficiency of the power plant [5].

Waste heat utilization subsystem

As fig. 2 shows, in order to improve the energy efficiency under part load working conditions, a TES device with PCM is installed between the GT and the AC. During the charge process (*i. e.* off-peak hours), high temperature exhaust gases (Q_{exhaust}) from the GT flow into the PCM-TES equipment for heat storage. Whereas during the discharge process (*i. e.* peak hours), stored heat, Q_{TES}, is released and flows into the AC to produce cooling water.

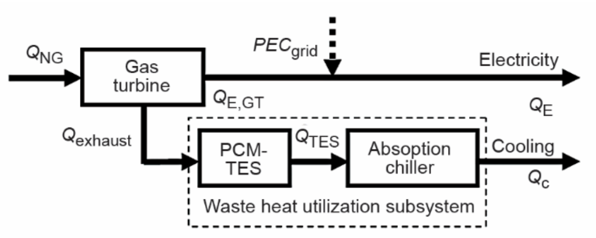

Figure 2. Schematic diagram of a PCM-TES-BCHP system

For the AC, from the perspective of thermodynamics, it can be regarded as a heat engine combined with a heat pump, so that its thermal performance highly depends on generation, evaporation and condensation temperatures [24]. To simplified analysis, the inlet exhaust gas temperature, $T_{\mathrm{AC,i}}$, cooling water temperature, T_{w}, and ambient temperature, T_{a}, are substituted for the generation, evaporation and condensation temperatures of the AC, respectively [5]. Hence, the COP, of the AC can be obtained by using the simplified thermodynamic model:

$$COP = X \frac{T_{\mathrm{AC,i}} - T_{\mathrm{a}}}{T_{\mathrm{AC,i}}} \frac{T_{\mathrm{w}}}{T_{\mathrm{a}} - T_{\mathrm{w}}} \tag{2}$$

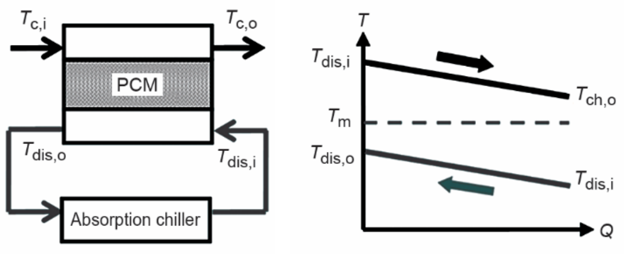

Figure 3. Heat transfer process in the PCM-TES equipment

where X is the thermodynamic perfectness of the AC, which represents the thermal performance difference between practical device and ideal one. As shown in fig. 3, the inlet temperature of the AC just equals the outlet temperature of the PCM-TES equipment during discharge process ($T_{\mathrm{AC,i}} = T_{\mathrm{dis,o}}$). Therefore, COP is highly influenced by the heat transfer performance of the PCM-TES equipment.

It can be seen that the inlet temperature of the PCM-TES equals the exhaust gas temperature ($T_{\mathrm{ch,i}} = T_{\mathrm{exhaust}}$). The outlet temperatures depend on not only the phase change temperature, T_{m}, of the energy storage material but also the heat transfer performance, *NTU*, of the PCM-TES equipment. According to the heat transfer model shown in fig. 3, the outlet temperatures of PCM-TES during charge and discharge processes can be expressed by:

$$T_{\mathrm{ch,o}} = T_{\mathrm{ch,i}} - (T_{\mathrm{ch,i}} - T_{\mathrm{m}})[1 - \exp(-NTU_{\mathrm{ch}})] \tag{3}$$

$$T_{\mathrm{dis,o}} = T_{\mathrm{dis,i}} + (T_{\mathrm{m}} - T_{\mathrm{dis,i}})[1 - \exp(-NTU_{\mathrm{dis}})] \tag{4}$$

After introducing PCM-TES device, the gas turbine can work steadily under rated condition in theory [21]. Therefore, the total primary energy consumption is mainly impacted by the waste heat utilization subsystem (WHUS) (*i. e.* PCM-TES and AC). To evaluate the overall energy conversion effect, the efficiency of WHUS, η_{WHUS}, can be designated by the ratio of output cooling to input heat:

$$\eta_{\mathrm{WHUS}} = \frac{Q_{\mathrm{C}}}{Q_{\mathrm{exhaust}}} \tag{5}$$

Integrating eq. (5) with eq. (1), it can be obtained that:

$$PEC = PEC_{\mathrm{GT}} + PEC_{\mathrm{grid}} = \frac{Q_{\mathrm{C}}}{1 - \eta_{\mathrm{GT}}} \left(1 - \frac{\eta_{\mathrm{GT}}}{\eta_{\mathrm{grid}}}\right) \frac{1}{\eta_{\mathrm{WHUS}}} + \frac{Q_{\mathrm{E}}}{\eta_{\mathrm{grid}}} \tag{6}$$

In practical engineering fields, the cooling and electrical loads (Q_{C}, Q_{E}) are determined by users. The power generation efficiencies (η_{gird}, η_{GT}) are determined by the power grid and chosen GT, respectively. Moreover, in most situations, there is $\eta_{\mathrm{GT}} < \eta_{\mathrm{grid}}$ [23]. So from eq. (6), it can be seen that PEC always decreases monotonically with increasing η_{WHUS}.

Energy storage effectiveness

The WHUS is indeed an energy storage unit, where the stored heat is converted to the cooling water through the AC. The effectiveness of such an energy storage unit can be defined:

$$\varepsilon = \frac{\eta_{\mathrm{WHUS}}}{\eta_{\mathrm{WHUS,max}}} = \frac{\dfrac{Q_{\mathrm{C}}}{Q_{\mathrm{exhaust}}}}{\dfrac{Q_{\mathrm{C,max}}}{Q_{\mathrm{exhaust}}}} = \frac{Q_{\mathrm{C}}}{Q_{\mathrm{C,max}}} = \frac{Q_{\mathrm{TES}} COP}{Q_{\mathrm{TES,max}} COP_{\max}} \tag{7}$$

It is the ratio of the practical provided cooling power to the theoretically maximal one for the energy storage unit. Furthermore, eq. (7) indicates that the defined energy storage effectiveness is indeed the product of two efficiencies:

$$\eta_1 = \frac{Q_{\mathrm{TES}}}{Q_{\mathrm{TES,max}}}, \qquad \eta_2 = \frac{COP}{COP_{\max}} \tag{8}$$

where η_1 is the ratio of stored heating power to its maximal one (*i. e.* the heat storage capacity of ideal TES equipment) and η_2 – the ratio of practical *COP* of the AC to its maximal one (*i. e.* often the rated COP). According to the system process, fig. 2, by combining eqs. (2) to (4) with eq. (8), there are:

$$\eta_1 = \frac{(T_{\mathrm{ch,i}} - T_{\mathrm{m}})[1-\exp(-NTU_{\mathrm{ch}})]}{T_{\mathrm{ch,i}} - T_{\mathrm{AC,o}}} \tag{9}$$

$$\eta_2 = \frac{\dfrac{T_{\mathrm{dis,i}} + (T_{\mathrm{m}} - T_{\mathrm{dis,i}})[(1-\exp(-NTU_{\mathrm{dis}})] - T_{\mathrm{a}}}{T_{\mathrm{dis,i}} + (T_{\mathrm{m}} - T_{\mathrm{dis,i}})[(1-\exp(-NTU_{\mathrm{dis}})]}}{\dfrac{T_{\mathrm{ch,i}} - T_{\mathrm{a}}}{T_{\mathrm{ch,i}}}} \tag{10}$$

So, the defined energy storage effectiveness of energy storage unit can be changed into:

$$\varepsilon = \eta_1 \eta_2 = \frac{(T_{\mathrm{ch,i}} - T_{\mathrm{m}})[1-\exp(-NTU_{\mathrm{ch}}]\dfrac{T_{\mathrm{dis,i}} + (T_{\mathrm{m}} - T_{\mathrm{dis,i}})[1-\exp(-NTU_{\mathrm{dis}})] - T_{\mathrm{a}}}{T_{\mathrm{dis,i}} + (T_{\mathrm{m}} - T_{\mathrm{dis,i}})[1-\exp(-NTU_{\mathrm{dis}})]}}{(T_{\mathrm{ch,i}} - T_{\mathrm{AC,o}})\dfrac{T_{\mathrm{ch,i}} - T_{\mathrm{a}}}{T_{\mathrm{ch,i}}}} \tag{11}$$

It is clear that T_{m} has impacts on both η_1 and η_2. For given energy supply devices (GT and AC), the maximal heat storage capacity and the rated COP are all known and $\eta_{\mathrm{WHUS,max}}$ is a constant value. According to eq. (7), energy storage effectiveness, ε, increases monotonically with increasing η_{WHUS}. As a consequence, there is:

$$\min PEC \Leftrightarrow \max \eta_{\mathrm{WHUS}} \Leftrightarrow \max \varepsilon \tag{12}$$

In other words, for the PCM-TES-BCHP system optimization, minimizing the primary energy consumption is just equivalent to maximizing the overall energy conversion efficiency of the WHUS, also equivalent to maximizing the defined the energy storage effectiveness.

Results

Based on the previous analysis and established model, the energy storage effectiveness, ε, is the function of T_m, T_a, inlet temperatures, $T_{ch,i}$, $T_{dis,i}$, and numbers of heat transfer unit, NTU_{ch}, NTU_{dis}, of PCM-TES. It is assumed that the numbers of transfer unit equal each other during charge and discharge processes ($NTU_{ch} = NTU_{dis} = NTU$). Aimed at maximizing ε and minimizing *PEC* accordingly, the optimal phase change temperature, $T_{m,opt}$, can be deduced out from eq. (11):

$$\frac{\partial\varepsilon}{\partial T_m} = 0 \Rightarrow T_{m,opt} = \frac{-\exp(-NTU)T_{dis,i} + \sqrt{(\exp(-NTU)T_{dis,i}T_a + [1-\exp(-NTU)]T_aT_{ch,i}}}{1-\exp(-NTU)} \tag{13}$$

Equation (13) gives the analytical optimal phase change temperature based on the simplified PCM-TES-BCHP system model. According to eqs. (3) and (4), if the heat exchange area of the PCM-TES equipment is infinite ($NTU \rightarrow +\infty$), the outlet temperatures just equal the phase change temperature ($T_{ch,o} = T_{dis,o} = T_m$). In that ideal situation, $T_{m,opt}$ can be expressed by:

$$T_{m,opt} = \sqrt{T_aT_{ch,i}} = \sqrt{T_aT_{exhaust}} \tag{14}$$

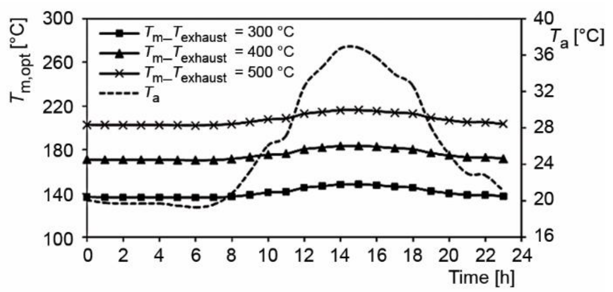

Figure 4. Optimal phase change temperature for BCHP system with ideal PCM-TES equipment (*NTU*→+∞)

For ideal PCM-TES equipment with infinite *NTU*, $T_{m,opt}$ is just the geometrical average value of T_a and $T_{exhaust}$ from the GT. For instance, $T_{exhaust}$ often ranges from about 300 °C to 500 °C for different gas turbines [6]. Whereas the ambient temperature varies widely for different climate zones as well as fluctuates timely in one day. Figure 4 gives the hourly outdoor air temperature in a typical summer day in Beijing, China (from Chinese Architecture-specific Meteorological Data Sets for Thermal Environment Analysis). Then according to eq. (14), the optimal phase change temperature for BCHP system with ideal PCM-TES (*NTU*→+∞) can be obtained, fig. 4. It can be seen that optimal phase change temperature varies slightly with changing ambient temperature in one day, but varies considerably with changing exhaust gas temperature.

On the other hand, for practical PCM-TES equipment with finite *NTU*, $T_{m,opt}$ is impacted by various factors, eq. (12). It is assumed that $T_{exhaust}$ = 300 °C, $T_{ch,i}$ = 140 °C, T_a = 20 °C, *NTU* = 1, the energy storage effectiveness is shown in fig. 5.

It can be seen that with the increasing T_m, η_1 decreases while η_2 increases. On the one hand, higher phase change temperature leads to higher outlet temperature of PCM-TES during discharge process, which is favorable for increasing the COP of the AC. On the other hand, higher phase change temperature also leads to relatively lower temperature difference, $T_{exhaust} - T_m$, during charge process, resulting in low energy storage capacity for the PCM-TES equipment. Thus due to such two counteractive influences, the energy storage effectiveness, ε, increases first and then decreases after the peak value, with increasing T_m. And ε reaches the maximal value, 0.28, only if T_m = 183 °C for this case.

As fig. 6 shows, with the increasing *NTU* of the PCM-TES equipment, both the energy storage effectiveness, ε, and the optimal phase change temperature, $T_{m,opt}$, increases. In other words, improving the thermal performance of the TES device can reduce the heat transfer irreversible losses during the charge and discharge processes, so that the overall energy conversion and usage efficiency of the PCM-TES-BCHP system increases, which is favorable for primary energy consumption saving.

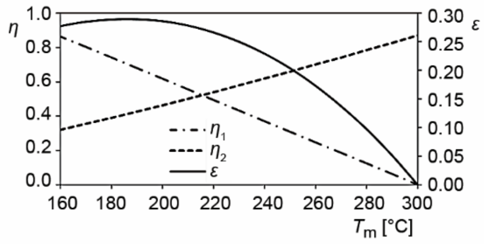

Figure 5. Energy storage effectiveness variations with changing phase change temperature

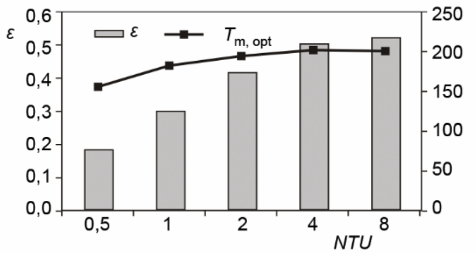

Figure 6. Optimal phase change temperature under different *NTU* of PCM-TES

Conclusion

Integrating PCM-TES equipment with BCHP system can improve the thermal performance and reduce the primary energy consumption. In this paper, a new index, energy storage effectiveness, is proposed to evaluate the energy saving effect of the PCM-TES. Based on the simplified system model, the relationship is established between the energy storage effectiveness and the primary energy consumption for the whole PCM-TES-BCHP system. Aimed at maximizing the energy storage effectiveness, the optimal phase change temperature of the PCM-TES equipment is obtained. The results of an illustrative example show that the theoretically optimal phase change temperature is just the geometrical average value of the ambient temperature and the exhaust gas temperature from the gas turbine for ideal PCM-TES equipment with infinite *NTU*. It also indicates that both energy storage effectiveness and optimal phase change temperature increase with increasing *NTU*. So improving the thermal performance of PCM-TES device is favourable for increasing overall energy conversion and usage efficiency of the PCM-TES-BCHP system and saving the primary energy consumption accordingly. This work is of great importance in guiding the optimization design of PCM-TES-BCHP systems.

Acknowledgment

This research is financed by National Key Research and Development Program of China (2016YFB0901405), National Natural Science Foundation of China (51706148 and 51521005) and Sichuan Science and Technology Program of China (2017JY0333).

Nomenclature

Q – heat energy capacity, [kW]
T – temperature, [°C]

Greek symbols

ε – energy storage effectiveness
η – efficiency

Abbreviations

AC – absorption chiller
AHP – absorption heat pump
BCHP – building cooling heating and power
FEL – following electrical load
FTL – following thermal load

GT – gas turbine
NG – natural gas
NTU – number of transfer units
PEC – primary energy consumption
TES – thermal energy storage
WHUS – waste heat utilization subsystem

Subscripts

a – ambient
C – cooling
ch – charge
dis – discharge
E – electricity
exhaust – exhaust gas
grid – power grid
i – inlet
m – phase change material
max – maximal
o – outlet
opt – optimal
w – water

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Paper submitted: February 22, 2017
Paper revised: August 12, 2017
Paper accepted: August 14, 2017