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Postprint: Study on Optimal Phase Change Temperature for Phase-Change Energy Storage Cogeneration Systems

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Abstract

Gas turbine-driven combined cooling, heating and power (CCHP) systems are small-scale energy supply systems that have been vigorously encouraged for development in China in recent years. Integrating energy storage devices with CCHP systems can achieve capacity reduction and efficiency improvement for the system' s energy supply equipment. This paper establishes a simplified mathematical model for a phase-change energy storage CCHP system and proposes a method for determining the optimal phase change temperature with the optimization objective of minimizing primary energy consumption under given actual user loads. Typical case study analysis demonstrates that the optimal phase change temperature of the phase-change flue gas heat accumulator is related to user load and equipment performance. Compared with infinite NTU, when NTU is finite, the optimal phase change temperature increases and the minimum primary energy consumption increases.

Full Text

Preamble

Optimal Phase Change Temperature for Energy Storage Based on Fluctuating Loads in Building Cooling Heating and Power System

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Abstract

Natural gas-driven Building Cooling Heating and Power (BCHP) systems have been actively promoted in China in recent years as small-scale energy supply systems. Integrating energy storage devices with BCHP systems can reduce equipment capacity and improve efficiency. This paper establishes a simplified mathematical model for a phase-change energy storage BCHP system and proposes a method to determine the optimal phase change temperature that minimizes primary energy consumption under given actual user loads. Analysis of a typical case study demonstrates that the optimal phase change temperature of a phase-change flue gas heat storage unit depends on user loads and equipment performance. Compared with infinite NTU, finite NTU results in a higher optimal phase change temperature and increased minimum primary energy consumption.

Keywords: Tri-generation; Energy storage; Phase change temperature; Primary energy consumption

Introduction

Distributed energy systems offer advantages including high energy efficiency, low pollutant emissions, small capacity, and proximity to users. Building Cooling Heating and Power (BCHP) systems based on the principle of cascade energy utilization represent the primary form of distributed systems [1]. Most BCHP systems operate under variable conditions, with performance degradation in all subsystems as loads decrease. Integrating energy storage devices with BCHP systems can improve energy savings compared to systems without storage [2]. However, the impact of energy storage on BCHP system efficiency depends on numerous factors, and the benefits are not always significant [3]. For the typical configuration of "gas turbine + absorption chiller,"the primary purpose of energy storage is to improve system performance under variable operating conditions and enable equipment to operate as efficiently as possible. Phase-change energy storage represents an important storage method, yet many questions remain regarding its application in BCHP systems: how should the system operate, when should energy be stored and released, and how should the optimal phase change temperature be determined? The first question concerns selecting an appropriate boundary between peak and off-peak periods based on typical daily hourly load profiles. The selection of phase change temperature affects not only flue gas flow rates but also the absorption chiller' s COP and total storage capacity. This paper establishes a simplified mathematical model for a phase-change energy storage BCHP system and proposes a method to determine the optimal phase change temperature that minimizes primary energy consumption under given actual user loads, providing valuable reference for practical system design.

1.1 System Flow

Taking summer operating conditions as an example, the operating strategies are illustrated in [Figure 1: see original paper] and [Figure 2: see original paper]. During off-peak periods when user loads are below the average load, flue gas from the gas turbine prioritizes meeting the absorption chiller' s cooling demand (user-side cooling load). Excess heat enters the thermal storage unit for energy storage. The flue gas flow rate entering the absorption chiller is determined hourly based on the user-side load, while the flow rate for storage is then determined accordingly.

During peak periods when user-side loads exceed the average load, the gas turbine exhaust alone cannot satisfy user demand, requiring discharge of heat stored during off-peak periods. The discharge fluid uses flue gas exiting the absorption chiller at temperature $T_{a,o}$. This discharge fluid mixes with gas turbine exhaust before entering the absorption chiller for cooling. The discharge fluid flow rate is determined by user-side loads and affects both the heat consumption of absorption cooling and the absorption chiller' s COP.

1.3 Absorption Chiller Model

From thermodynamic principles, an absorption refrigeration machine can be viewed as a "heat engine + heat pump" combination. The absorption chiller' s COP is primarily influenced by generator temperature T_g, evaporation temperature T_e, and condensation temperature T_c. For simplified analysis, gas turbine exhaust temperature replaces generator temperature, chilled water supply temperature replaces evaporation temperature, and ambient temperature replaces condensation temperature, giving the absorption chiller COP as:

$$\mathrm{COP} = \mu \frac{T_g - T_c}{T_g} \frac{T_e}{T_c - T_e}$$

where μ is the thermodynamic perfection factor. When the gas turbine operates at full load with exhaust temperature at its rated value, the absorption chiller reaches its highest generator temperature and maximum COP. Assuming $T_g = T = 773\,\mathrm{K}$, $T_0 = T_c = 303\,\mathrm{K}$, $T_e = 280\,\mathrm{K}$, and a rated COP of 1.2, the thermodynamic perfection factor μ is calculated to be 0.163. Assuming μ remains constant under variable operating conditions, the actual COP can be determined from flue gas temperature [5].

1.4 Energy Storage Device Model

During charging, heat exchange between gas turbine exhaust and storage medium satisfies:

$$\varepsilon = 1 - \exp(-\mathrm{NTU})$$

The energy supplied to the user side is:

$$Q = \dot{m} c_p (T - T_{c,o}) \cdot \mathrm{COP}$$

During discharging, heat exchange between discharge fluid and storage medium satisfies:

$$\varepsilon = 1 - \exp(-\mathrm{NTU})$$

where T is gas turbine exhaust temperature, T_m is the phase change temperature to be optimized, $T_{c,o}$ and $T_{d,o}$ are outlet temperatures of charging and discharging fluids respectively, and $T_{a,o}$ is flue gas temperature at the absorption chiller outlet. Total heat stored and discharged over one cycle are equal.

1.2 Gas Turbine Model

Natural gas with heating value Q_g enters the gas turbine (power generation efficiency η), producing electricity P supplied directly to users. Simultaneously, exhaust gas at temperature T and flow rate $\dot{m}$ enters the absorption chiller generator as a high-temperature heat source to drive the absorption refrigeration cycle. For the gas turbine, energy balance gives:

$$P = \eta Q_g$$

2 Calculation Method

During off-peak periods, all flue gas entering the absorption chiller comes from the high-temperature gas turbine exhaust, stabilizing the absorption chiller COP at 1.2. During peak periods, the gas entering the absorption chiller is a mixture of gas turbine exhaust and discharge fluid from the storage unit, resulting in temperatures lower than gas turbine exhaust temperature and COP values below 1.2 (calculated using Equation (6)). The average value of typical daily hourly loads is first selected to divide peak and off-peak periods, yielding corresponding gas turbine flue gas flow rates. Under these conditions, the absorption chiller' s generator temperature during peak periods is lower than during off-peak periods, resulting in lower COP and creating an imbalance between storage and discharge. By optimizing the gas turbine flue gas flow rate $\dot{m}$ and storage unit phase change temperature T_m, the storage-discharge difference can be eliminated. At this point, the gas turbine flue gas flow rate represents the minimum flow rate, and the phase change temperature is optimal.

3 Case Study

A Beijing hotel under summer operating conditions is selected as an example, with typical daily hourly loads shown in [Figure 3: see original paper]. Parameters are set as: gas turbine rated power generation efficiency $\eta_e = 35\%$, flue gas temperature after absorption chiller $T_{a,o} = 453\,\mathrm{K}$, flue gas specific heat $c_{p,a} = 1.2\,\mathrm{kJ/(kg \cdot K)}$, and chilled water supply/return temperatures of 280 K and 285 K.

With infinite NTU, the minimum gas turbine flue gas flow rate is calculated as $\dot{m} = 1.575\,\mathrm{kg/s}$, with an optimal phase change temperature of $T_m = 586\,\mathrm{K}$. Based on hourly cooling loads, the gas turbine' s power generation efficiency, electricity output, and absorption chiller COP can be calculated hourly. According to electricity demand, power purchase/sale quantities are determined to calculate total daily primary energy consumption (natural gas heating value). Surplus electricity can be sold to the grid, with corresponding primary energy consumption deducted from the revenue generated. For comparison, primary energy is uniformly converted to natural gas heating value, assuming the grid uses natural gas combined cycle generation (55% efficiency). The minimum primary energy consumption is 1.84×10^4 kWh of natural gas, achieving a maximum energy savings rate of 12.7% compared to separate production systems. The relationship between primary energy consumption and phase change temperature is shown in [Figure 4: see original paper].

For finite NTU, the optimal phase change temperature increases and minimum primary energy consumption rises. At NTU = 2, the optimal phase change temperature is $T_m = 596\,\mathrm{K}$, minimum primary energy consumption is 1.85×10^4 kWh, and maximum energy savings rate is 11.9%.

Conclusion

This paper proposes a method to determine the optimal phase change temperature that minimizes primary energy consumption under given actual user loads. For a given typical daily hourly load profile, the minimum primary energy consumption and optimal phase change temperature are obtained and compared with separate production systems. With infinite NTU, maximum energy savings of 12.7% are achieved; with NTU = 2, maximum energy savings are 11.9%. Compared with infinite NTU, finite NTU values result in increased optimal phase change temperature and higher minimum primary energy consumption. While determining the optimal phase change temperature, the relationship curve between primary energy consumption and phase change temperature is obtained. When primary energy consumption exceeds the minimum, two phase change temperatures on either side of the optimal value correspond to the same primary energy consumption.

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