Next Energy 8 (2025) 100363

Contents lists available at ScienceDirect

Next Energy

journal homepage: www.sciencedirect.com/journal/next-energy

Research article

Cold storage systems for electricity management: Performance analysis in office and power plant applications

Ramin Mehdipour[a,*], Behnam Feizollah Beigi[b], Romina Fathiraboki[b], Hasan Asgari[b], Zahra Baniamerian[a]

[a] Mechanical and Aerospace Systems Research Group, University of Nottingham, Nottingham NG7 2TU, UK
[b] Department of Mechanical Engineering, Tafresh University, Tafresh, Iran

ARTICLE INFO

Keywords:
Peak shifting
Electricity consumption
Cold storage system
Partial and full storage system

ABSTRACT

In hot seasons, residential areas consume significant amounts of electricity for refrigeration and air conditioning, leading to peak power consumption. This simultaneous increase in cooling load, combined with reduced performance of gas turbines, places considerable stress on the power grid, particularly during specific periods each year. Cold storage systems offer an effective solution by shifting electricity consumption from peak daytime hours to off-peak nighttime periods. This study evaluates and compares the economic and thermal performance of cold storage systems implemented in both power plants and office buildings for peak demand management. Tailored cold storage systems were designed for each application, with a focus on ensuring reliable performance during peak cooling demand based on load analysis. The study utilized real-world case studies, including modeling for an office building in Arak, Iran, and a nearby power plant, to understand the impact of different climatic conditions on system performance. The results indicate that, during peak hours, the turbine's net power output improved by 15.98% and 17.97% with partial and full storage methods, respectively, compared to scenarios without cooling. Additionally, the economic analysis revealed substantial cost savings, with partial and full storage systems resulting in reductions of 97.36% and 95.54%, respectively, in power plant units compared to similar office buildings with equivalent power consumption. The analysis also highlights that full storage systems in both office and power plant contexts deliver better peak shaving performance but at a higher cost due to the larger size of tanks and equipment required for operation. These findings underscore the potential of cold storage systems as an effective strategy for enhancing electricity management and reducing operational costs.

1. Introduction

Peak load management, particularly for cooling demands, is of critical importance given the rising energy needs due to improved living standards and global warming. In 2022, nearly 9% of total electricity consumption was dedicated to cooling loads, highlighting the significant share this sector occupies in the energy landscape. This becomes even more pronounced in hot regions during peak summer periods, where electricity consumption for cooling can surpass 60%. Such high demands not only strain power generation and distribution networks but also pose a risk of widespread blackouts, with severe economic, social, and health impacts. Moreover, thermal energy storage systems like ice thermal energy storage offer a viable solution by enabling the storage of thermal energy during off-peak hours. This stored energy can then be used to reduce peak loads during high-demand periods, thus playing a pivotal role in managing cooling loads, optimizing air conditioning system operational costs, and enhancing grid flexibility and resilience [1]. In the context of depleting fossil fuel resources and stringent environmental regulations, developing effective strategies for electricity consumption management is becoming increasingly crucial. Fluctuations in electricity generation, especially from renewable energy sources, along with variations in demand, underscore the necessity for energy storage systems [2]. In regions with hot climates, elevated electricity demand during peak afternoon hours presents a significant challenge to power generation and distribution networks, driven predominantly by cooling requirements. Electricity consumption generally follows 2 major peaks annually. The first peak, related to lighting, occurs at night throughout the year. The second, known as the cooling load peak, occurs during the summer months due to increased air conditioning use. This period spans from mid-May to late September. As outdoor temperatures rise, the efficiency of gas turbines declines, further straining the power grid between 11 a.m. and

* Corresponding author.
E-mail address: ramin.mehdipour@nottingham.ac.uk (R. Mehdipour).

https://doi.org/10.1016/j.nxener.2025.100363
Received 30 April 2025; Received in revised form 25 June 2025; Accepted 26 June 2025

Nomenclatures

a	Real amount
C_{pa}	Average heat capacity of the air at constant pressure
C_{pg}	Average heat capacity of turbine exhaust gas at constant pressure (kJ/kg.k)
C_p	Specific heat capacity of moist air
g	Water vapor
$\dot{m}_a$	Air flow rate (kg/s)
$\dot{m}_f$	Fuel flow rate (kg/s)
$\dot{m}_t$	Outlet gas flow rate (kg/s)
NTU	number of transfer units NTU
P1	Ambient pressure (kPa)
$\dot{Q}$	The time rate of heat transfer (kW)
Q_{add}	Added heat (kW)
R	Gas constant
r_p	Compression ratio
ST	Storage tank
SFC	Fuel consumption rate (kg/kW.h)
T1	Ambient temperature (K)
V	Velocity flow rate (m^3/s)
V_{ST}	Storage tank volume (m^3)
W_C	Compressor work (kW)
W_{net}	Net work
Wp	Power consumed by water pump
$\dot{W}_{Pump}$	The time rate of energy transfer by pump work (kW)
W_T	Turbine work (kW)
Z	Capital cost (US$)
t	Temperature (°C)
p	Plate surface
γ	Special heat ratio
η_{cc}	Combustion efficiency
$\eta_{C,isen}$	Isotropic compressor efficiency
$\eta_{c,m}$	Mechanical Compressor Efficiency
η_{th}	Thermal efficiency
$\eta_{T,isen}$	Isotropic turbine efficiency
$\eta_{T,m}$	Mechanical turbine efficiency
η_{pump}	Pump efficiency
ρ	Density (kg/m^3)
Δp_{cc}	Pressure drop of combustion chamber
Δp_{inlet}	Inlet pressure drop (N/m^2)Subscripts and superscripts
El_ch	Electricity consumed by chiller
HR	Heat transfer rate (kJ/kW.h)
ILP	Internal losses of the power plant
LHV	Lowest thermal value of fuel (kJ/kg)
MGT_AUX	Gas turbine miscellaneous consumables
MP_AUX	Miscellaneous power plant costs
Power	Output power
$power_{net}$	Net power output
T_LOS	Transformer losses

5 p.m. The simultaneous increase in cooling demand and decreased turbine efficiency can result in severe consequences, including power outages and blackouts.

To address the challenges associated with balancing energy production and consumption, various solutions have been proposed. These include increasing generation capacity (e.g., constructing new power plants or enhancing the performance of existing plants through inlet air cooling of gas turbines), deploying energy storage systems, implementing demand-side management strategies, utilizing smart grid technologies, and exploiting distributed energy resources such as solar and wind power. The optimal solution strongly depends on climatic conditions, grid architecture, types of generation resources, and consumer usage patterns.

Among these, 2 main strategies have garnered the most attention: first, increasing supply by developing power plants or using local renewable sources such as wind and solar; second, reducing and managing consumption. However, simultaneous fluctuations in energy consumption and production remain a fundamental challenge in grid operation. Hence, any solution that can simultaneously manage supply and demand—without requiring significant expansion of transmission infrastructure or imposing severe restrictions on consumers—can greatly enhance grid performance and reduce peak demand. This is where energy storage plays a vital role. Energy storage is considered a key and critical element in improving grid reliability, optimizing energy consumption, reducing production and operational costs, and facilitating the integration of renewable energy sources. Given the inherent variability in both energy supply and demand, storage technologies act as an effective tool to maintain the supply–demand balance. Various energy storage methods exist, including:

- Mechanical storage, such as

o Pumped hydro storage,
o Compressed air energy storage [3,4],
o Flywheel storage.

- Electrochemical storage, including batteries such as lithium-ion, lead-acid, and flow batteries [5,6];
- Thermal storage, in the forms of sensible heat, latent heat, and thermochemical reactions.

Battery systems are among the most widely used and powerful storage solutions [7]; however, challenges such as limitations in long-duration storage, efficiency degradation at high temperatures, and relatively high costs hinder their broader application. In contrast, thermal energy storage (TES) has gained attention as a reliable and cost-effective approach—especially for managing cooling peak loads. Nevertheless, the need for significant installation space, particularly in urban buildings, poses a barrier to widespread implementation.

It should be noted that building new power plants or relying on distributed energy resources such as solar and wind power involves significant costs and may even exacerbate fluctuations in energy production and consumption. These fluctuations necessitate the design of larger grid infrastructures, which are fully utilized only during short peak demand periods. In contrast, energy storage aims to synchronize production and consumption, promoting a more uniform and stable operation of the power grid.

Generally, thermal storage is used to respond to heating loads, while cold (and especially ice) storage is specifically applied for cooling demands. TES systems are highly effective in managing cooling demand by storing cooling energy during off-peak hours and utilizing it during peak periods. This strategy reduces stress on the power grid, lowers operating costs, and improves Heating, Ventilation, and Air Conditioning system performance. In scenarios with high cooling demand, cold thermal storage presents a highly efficient solution.

By storing cooling energy during off-peak hours, such systems allow chillers to operate under optimal conditions and reduce peak-time loads. The stored cold energy can then be used to meet space cooling demands during peak hours.

Another important application of cold thermal storage lies in increasing power generation capacity during periods of high temperature and demand. Specifically, stored cold energy can be used for inlet air precooling of gas turbines during peak operation hours. This strategy mitigates the drop in turbine performance caused by high ambient temperatures. The cold energy required is stored during off-peak hours when excess electricity is available at the power plant.

Cold storage in the form of ice, which utilizes the latent heat of fusion, offers a unique solution by significantly reducing the storage volume required for energy [7]. When integrated with heating and cooling systems [8], this approach becomes more efficient and contributes to energy consumption management.

Numerous studies have investigated diverse approaches to improving gas turbine performance, highlighting their importance in enhancing efficiency and reliability [9,10]. Simultaneously, substantial research has focused on the critical role of thermal energy storage in power plants, emphasizing its effectiveness as a proven energy management strategy [11]. Cooling turbine inlets is also a widely explored method for improving power plant efficiency [12,13]. Shirazi et al. [14] demonstrated that a cold storage system enhances power output and efficiency, while Sanaye and Shirazi [15,16] highlighted reductions in power consumption and CO_2 emissions through optimized air conditioning and cold storage integration. Similarly, Sanaye et al. [17] reported significant increases in power output and efficiency through turbine inlet cooling, particularly for turbines with capacities between 25 and 100 MW. Ameri and Hejazi [18] showed improvements in output power using absorption chillers, while Najjar et al. [19] achieved notable efficiency enhancements with vapor-compression chillers across various climates. Popli et al. [20] and El-Shazly et al. [21] found evaporative cooling and absorption chillers enhance power output and efficiency, with absorption chillers showing higher gains. Mohapatra [22] highlighted the effectiveness of media cooling in low-humidity areas. Renzi et al. [23] and Ehyaei et al. [24] reported significant efficiency and power increases using fog cooling.

Athari et al. [25] showed that combining steam injection and fog cooling achieves superior exergy efficiency. Beghi et al. [26], Han et al. [27], and Yan et al. [28] emphasized cold storage systems' efficiency and environmental benefits, supported by tariff optimization. Beigi and Mehdipour [29] identified cold storage as a key method for gas turbine performance management.

Table 1 presents a selection of previous studies in this field. A review of these works reveals that cold energy storage is commonly implemented either at the consumer end or at the power generation side. This raises a fundamental question: Where does cold storage offer superior performance—at the power plant or at the consumer site?

In the existing literature, this question has not been directly addressed. Most studies focus on either the consumer-side or the generation-side storage system in isolation, examining or optimizing one without comparing both under the same environmental conditions. To obtain a meaningful answer, it is essential to model and compare both systems under identical climatic scenarios.

This study aims to bridge that gap by evaluating and comparing the thermal and economic performance of cold storage systems implemented both at the power plant and at the consumer site. Another key research question addressed in this study is: How do these systems perform under varying climatic conditions? Unlike previous studies, this work considers the influence of climate on the selection and effectiveness of cold storage technologies, as well as their impact on the overall power grid performance.

To ensure the validity of the modeling, a real office building located in Arak, Iran, along with a nearby power plant, was selected as the case study. The current operational conditions of both systems were modeled, followed by the integration of appropriately designed ice or chilled water storage methods to manage peak electricity demand. Their performance was then compared and further examined under 3 additional climatic conditions represented by the cities of Isfahan, Yazd, and Bandar Abbas.

This paper introduces the modeling approach and the criteria for selecting the appropriate storage system. It includes the simulation of both partial and full energy storage methods, applied to both the production and consumption sides. Finally, the study assesses the impact of these storage systems on electricity production and consumption, particularly in terms of peak demand reduction or management. The results and methodology presented herein can serve as a valuable foundation for policymakers and energy planners to make informed decisions about the implementation of cold energy storage systems, their placement, and their potential contribution to peak demand mitigation and overall energy efficiency.

2. Cold Storage

2.1. In the Power Plant

The necessity of implementing cold storage systems becomes evident when increased gas power efficiency is required only during peak cooling load hours on the grid. During nonpeak hours, ice or cold water produced by compression chillers is stored in large containers. This stored cold energy is then utilized to cool the inlet air of gas turbines when electricity demand reaches its peak. This method leverages the

Table 1
Research in the field of ice thermal energy storage (ITES)

	Reference	Summary
1	Ghaith and Dag [30]	A solar-assisted cooling system integrated with ITES was evaluated for a single-story office building in Abu Dhabi and a 4-story residential complex in Dubai. The system demonstrated significant reductions in annual energy consumption and CO_2 emissions, with payback periods of 8.8 years (office) and 7.8 years (residential).
2	Tevis et al. [31]	A life cycle assessment (LCA) was performed for an office building in Thailand considering several energy supply configurations, including ITES and photovoltaic systems. ITES contributed to a 38% reduction in grid dependency and exhibited the lowest environmental impact across the system's lifetime.
3	Tang et al. [32]	An optimization framework was developed to minimize operational costs in district cooling systems using ITES. The approach yielded an 8% cost reduction. The study was limited to district cooling applications.
4	Rahgozar et al. [33]	The influence of climatic conditions on the economic viability of ITES in office buildings was assessed. While the importance of storage strategy was emphasized, the analysis was confined to office building typologies.
5	Erdemir et al. [34]	The study focused on ITES deployment in commercial buildings to reduce peak cooling demand and operational costs. Notable cost savings were achieved; however, the scope was restricted to commercial settings.
6	Griesbach et al. [35]	A numerical model was designed to assess ITES integration in research facilities across Germany, France, and the EU27 average. The study reported environmental benefits and reduced demand-related expenses, though the analysis did not extend beyond research buildings or compare storage strategies.
7	Shirazi et al. [14]	A thermo-economic and environmental assessment of ITES was conducted for inlet cooling in gas turbines. The findings indicated improvements in power output and thermal efficiency.
8	Sanaye and Shirazi [15]	Thermo-economic optimization of ITES applied to air-conditioning systems revealed 8–10% reductions in operational expenses. The research focused primarily on commercial-scale applications.
9	Beigi and Mehdipour [29]	The performance of ITES in improving gas turbine operations at Iranian power plants was investigated. The study demonstrated increased efficiency and reduced energy usage, with a detailed comparison of various cooling strategies.
10	Mehdipour et al. [36]	In this research, an ice production system based on a heat pump has been investigated.
11	Mehdipour et al. [37]	Seasonal ice storage for a cooling system for domestic applications has been studied.

specific heat capacity of water to effectively store energy. To enhance the thermal capacity of the solution, ethylene glycol is added to the water, which reduces the required tank volume and increases the temperature differential between the inlet and outlet.

Three strategies are commonly employed for cold storage:

1. Partial cooling load strategy with constant cooling power
2. Partial cooling load strategy with variable cooling power
3. Full cooling load strategy.

In the partial storage strategy, the chiller continuously contributes to the cooling process. During certain hours of the day, a portion of the chiller's cooling power is saved, allowing the system to meet the cooling load with the help of the storage system in the following hours. This method is particularly suitable when the cooling load during peak grid hours significantly exceeds the average grid load. The full storage strategy entails turning off the chiller during specific hours to avoid electricity consumption peaks, heat peaks, and to manage chiller capacity and initial investment costs. In this approach, only the stored cool energy in the tank is used to supply cooling energy. At other times, the chiller operates at constant power, with part or all of its coolant output directed to the storage tank, while the remainder is utilized for direct cooling through a heat exchanger.

In comparing the 2 strategies of full and partial thermal energy storage, it is important to examine the physical and operational differences between the 2 systems. In a full storage system, the objective is to meet the entire cooling demand during peak hours using the storage tank, which requires larger system capacity and results in higher initial capital cost. On the other hand, a partial storage system is designed to supply part of the peak demand from the tank and the remainder directly through the chiller, allowing for smaller tank size and greater operational flexibility.

During off-peak hours, partial storage systems have the advantage of meeting part of the cooling demand directly via the chiller. This often results in higher energy efficiency and lower operating costs, as the system can better respond to fluctuations in cooling demand and environmental conditions. In contrast, full storage systems prioritize charging the tank during off-peak hours, which may lead to lower overall energy efficiency under certain conditions.

Nevertheless, a key advantage of full storage systems is that during peak hours, the chiller can be completely shut down, and the entire cooling load is supplied from the storage tank. This enables more effective peak shaving and can significantly reduce the electrical demand on the grid during critical periods.

2.2. In Power Consumption Units (Residential, Commercial, Official)

During the hot months of the year, approximately 30% of all energy produced is consumed for cooling and air conditioning in public and residential areas [38]. Consequently, effectively managing the cooling load in these sectors can significantly contribute to reducing peak electricity demand. One critical factor influencing the economics of cold storage systems is the load profile of the power grid, which directly affects the profitability of these systems and their ability to mitigate peak demand. The optimal utilization of cold storage systems occurs when the peak load of the power grid coincides with the cooling load of buildings [39]. When these 2 peak demands align, the implementation of cold storage systems can alleviate pressure on the grid during peak hours. The operation of storage systems in official unit's mirrors that of those in power plants, following a similar process to manage cooling loads effectively.

3. Problem Definition

As previously mentioned, cold storage systems are highly effective in reducing peak load consumption in power plants and urban units, thereby decreasing overall energy demand. These systems facilitate better energy management through peak shaving and by modifying the load profile during peak consumption hours. This study compares the performance and investment costs of cold storage systems in both power plants and official units. The objective is to determine which type of unit (power plant or official) is more suitable for implementing a cold storage system under identical conditions, including geographical location, temperature, and relative humidity. To evaluate the performance of the storage system in a consumer unit, the research is based on modeling a real official unit.

3.1. Specifications of the Research Location

The primary role of cold storage systems lies in shifting peak energy demand rather than directly reducing overall energy consumption. This capability is particularly valuable in regions where electricity tariffs can be leveraged as a tool for effective energy management, making these solutions economically viable and practical for grid stability.

This study specifically focuses on peak cooling demand, a critical factor in warm climates such as those in certain regions of Iran. Conversely, colder or cold-humid climates face fewer challenges related to cooling loads and were therefore not the primary focus of this research. However, regions characterized by warmer climates and significant diurnal temperature variations offer favorable conditions for the effective design and economic justification of cold storage systems. By targeting these conditions, the systems can significantly enhance energy efficiency and reduce peak loads, thereby contributing to a more sustainable energy infrastructure. Fig. 1 illustrates 5 distinct climatic zones in Iran based on the classification introduced in ref [40], highlighting the particular significance of cold storage in hot regions. Consequently, this research focuses on 4 cities characterized by warm climates and 1 city with a mild climate. Arak represents a mild-arid climate, while Isfahan and Yazd are examples of warm and dry climates, and Bandar Abbas represents a warm and humid climate. Fig. 2 presents temperature and humidity data for these 4 cities as of August 15. The geographical, climatic, and temperature characteristics of these cities are summarized in Table 2.

3.2. Specifications of the Official Building Under Study

To evaluate the performance of the cold storage system in electricity consumption units, this study focuses on an official building. The building under consideration, depicted in Fig. 3, is designed for official purposes. Its dimensions and specifications are based on the Arak Gas

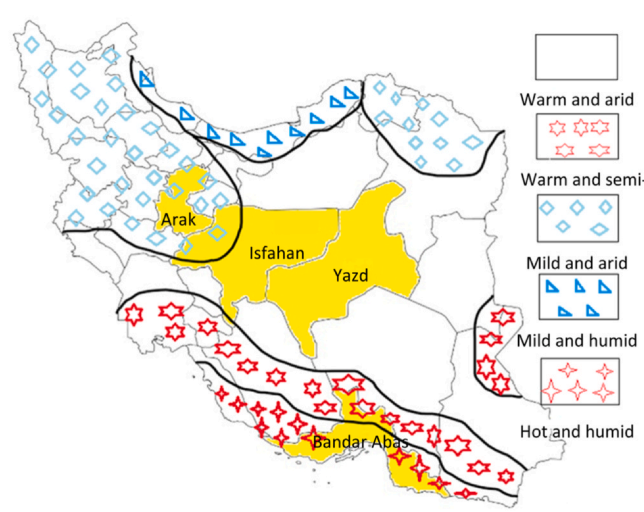
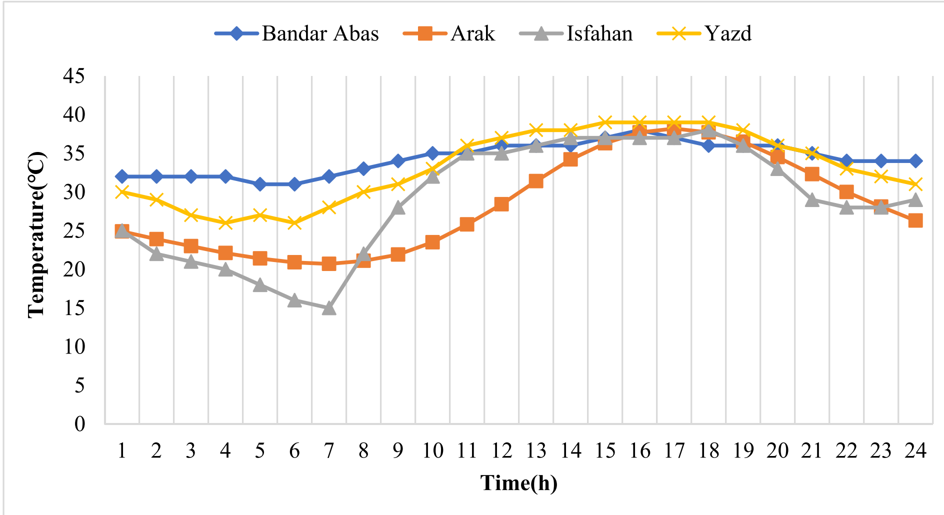
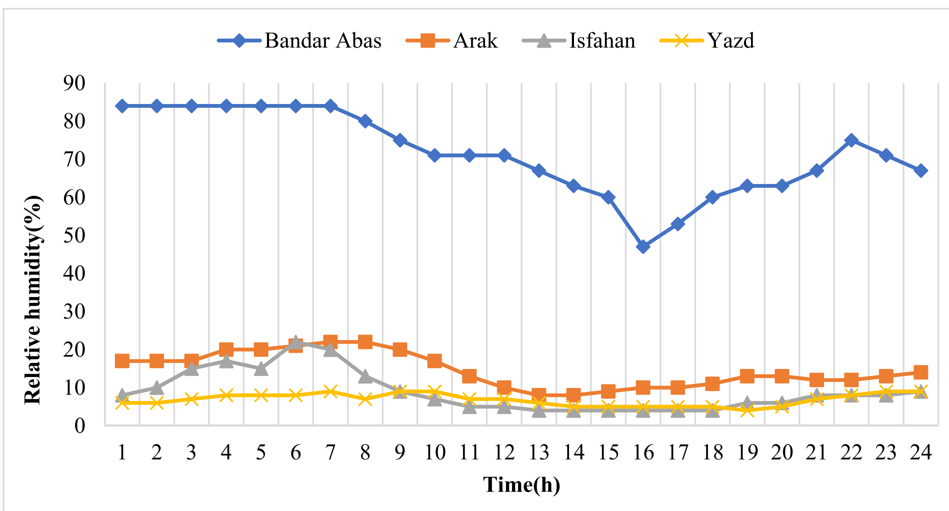


Fig. 1. Climatic zones of Iran according to the classification system introduced in reference [40].



a: Temperature Chart



b: Relative humidity chart

Fig. 2. Temperature and humidity in the studied cities on August 15.[41].

Table 2
Characteristics of the 4 cities under study

City name	Arak	Yazd	Bandar Abbas	Isfahan
Climate type	Moderate-dry	Warm-dry	Warm-humid	Warm-dry
Longitude	−49	−54	−56	−51
Latitude	34	31	27	32
Above sea level	1708	1230	10	1600

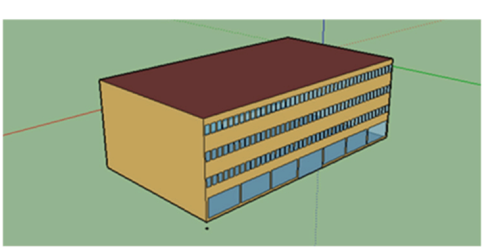


Fig. 3. Characteristics of the studied building.

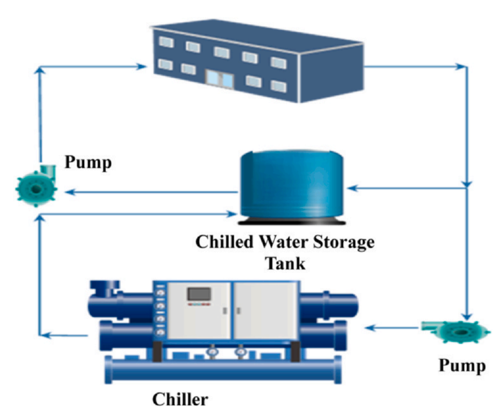


Fig. 4. Schematic of the cold storage cycle in an office building.

Department building, for which thermal and cooling load data is available through experimental measurements. Fig. 4 illustrates the cold storage system configuration, while Table 3 outlines the building's specifications. In this official unit, the cold water supplied by the chiller and stored in the storage tank is maintained at a temperature of 5 °C.

Table 3
Specifications of the studied building and storage equipment

Specifications of the studied building					
Length (m)	Width (m)	Height of each floor (m)	Exterior shell wall light passage area (m^2)	Ratio of side wall light passage to total area of building wall (%)	Number of floors
36	20	3	2959.3	28.9	4
Specifications of equipment in simulated circuit					
Pump	Maximum flow rate ($\frac{m^3}{s}$)		Pump head (pa)	Pump motor efficiency	
Chiller	0.0013509		179352	0.9	
Storage tanks	0.0013509		179352	0.9	
Exterior walls specifications of the building					
Exterior wall orientation	Number of windows		Height and width of the window	The name of the window structure	
North	126		0.65*1.1	Dbl Clr 3 mm/13 mm Air	
South	126		0.65*1.1	Dbl Clr 3 mm/13 mm Air	
East	-		-	-	
West	-		-	-	
Characteristics of the layers forming the outer walls of the building in question					
The name of the layers that form the wall	Thickness (m)	Guidance $\left(\frac{w}{mK}\right)$	Density $\left(\frac{kg}{m^3}\right)$	Heat capacity ($\frac{J}{kgK}$)	R-value ($m^2{\cdot}K/W$) -U-value ($W/m^2{\cdot}K$)
WD01	0.01909	0.115	513	1381	0.17–6.01
PW03	0.0127	0.115	545	1213	0.11–9.05
IN02	0.0099	0.0430	10	837	0.23–4.36
GP01	0.0127	0.16	801	837	0.08–12.6
Specifications refrigeration					
Refrigerant		R290			
Design flow temperature		40 °C			
Outdoor temperature		0 °C			
η_c		%80			
Design flow temperature		40 °C			
Outside design temperature		−3 °C			
Tank		The tank is fully insulated and isolated.			

It is assumed that the cooling system has sufficient capacity to simultaneously meet the cooling demand of the target unit and the load required for ice production during peak hours. Depending on the selected cold storage scenario, the cooling load may vary accordingly. It is also assumed that the storage tanks are well-insulated and that thermal losses from the tanks are negligible.

3.3. Characteristics of the Power Plant Under Study

This study evaluates a power plant equipped with a gas turbine, using the specifications detailed in Table 4. The modeled parameters are based on the Qom power plant, which shares similar technical characteristics and weather conditions with the Arak area. Fig. 5 presents a schematic of the cold storage cycle within the power plant system. In this power plant unit, the cold water supplied by the chiller and stored in the tank is maintained at a temperature of 5 °C.

4. Modeling

4.1. Solution Algorithm

In this study, 2 distinct scenarios for cold thermal energy storage are independently modeled:

1. Cold storage at the consumer site
2. Cold storage at the production site (power plant).

To examine the first scenario, an office building is selected as the consumer, and all components of its cooling system are comprehensively modeled. Subsequently, an ice production and storage system is integrated into the building's cooling infrastructure. Determining the appropriate volume of the storage tank involves ensuring that its capacity allows for full operation of the ice production system during off-peak hours and provides sufficient stored ice to meet peak cooling demands. By the end of this section, the cooling system is designed to ensure effective ice production during low-demand periods and adequate cooling supply during peak hours using the stored ice. Accordingly, the system performance is analyzed under 3 operational scenarios:

1. Without any cold storage
2. Partial storage (ice is both produced and consumed continuously throughout the day)
3. Full storage (ice is produced exclusively during off-peak hours and consumed during peak hours).

Modeling the building's thermal load is a critical step in this section and is carried out using the specifications provided in Section 3.2 and Table 3. In the second scenario, that is, cold storage at the power plant, the thermodynamic cycle of the power plant is modeled using the data presented in Table 4 and the assumptions detailed in reference [29]. Additionally, a cooling cycle is designed and integrated into the main system to pre-cool the intake air to the gas turbine. The size of the required cooling equipment is determined based on the volume of the storage tank and the cooling demand of the turbine. At the end of this

Table 4
Modeling gas turbine specifications [29]

Feature	Value
Gas turbine model	V94.2
Manufacturer	Siemens
Power generated turbine (MW)	164.7
Turbine fuel	Natural gas
Height above sea level (m)	1020
Adiabatic efficiency of the compressor (%)	88.3
Adiabatic efficiency of the turbine (%)	88.3
Pressure ratio	11.65

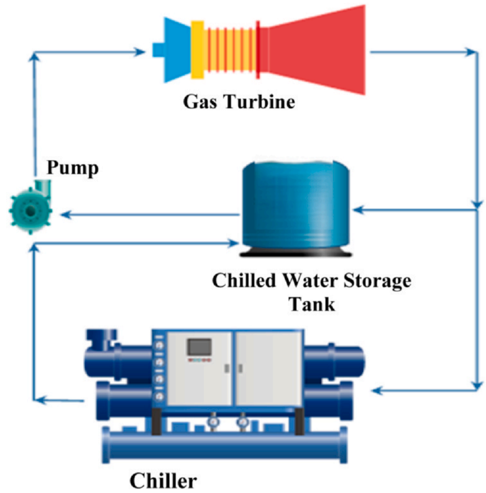


Fig. 5. Schematic of the storage system at a power plant.

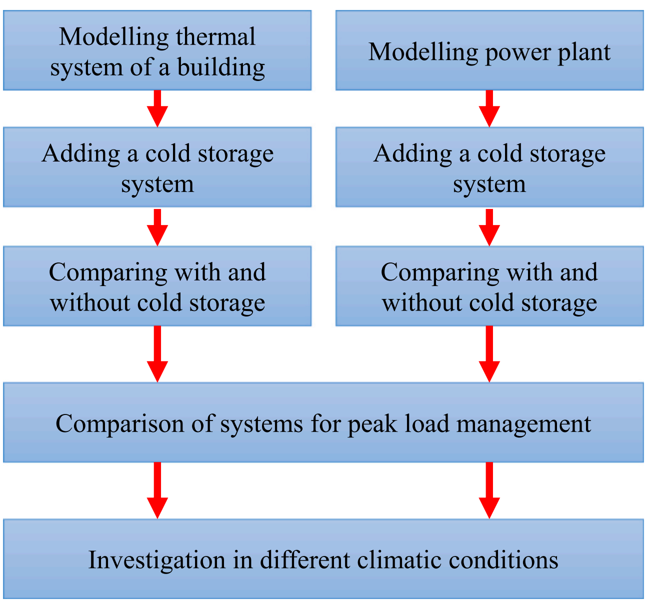


Fig. 6. Research methodology flowchart.

section, the power plant's performance is evaluated under both storage and non-storage conditions across various storage scenarios. Finally, the effectiveness of both storage systems—at the consumer and production sites—in mitigating peak electricity demand is evaluated under different climatic conditions. The overall modeling algorithm used in this study is illustrated in Fig. 6.

In this system, it is assumed that the gas turbine operates under continuous conditions with an isentropic efficiency of 88.3% for both the turbine and the compressor, using natural gas as the fuel. For the building, the current thermal conditions and actual building specifications, as outlined in Table 3, were considered. The temperature and solar radiation conditions specific to each city were applied to calculate the building's cooling load. For heating, a gas-fired boiler system was used, while cooling relied on systems such as air conditioners or heat pumps.

In the section concerning the modeling of the cold storage system and ice or cold-water production cycles using water or air-source heat pumps, assumptions from references [36,37] were adopted. It was assumed that all the ice or cold water produced was fully stored in a well-insulated reservoir, minimizing thermal losses. Since the real building and power plant, located in Arak, already have performance data that align closely with real-world results, it was assumed that replicating the same building and power plant in other cities with varying climatic conditions would allow for an analysis of performance changes. Based on these new conditions, an updated cooling storage system was designed, and its performance was evaluated.

4.2. Analysis of Cold Storage in Official Facilities

The cooling load of the building is modeled to achieve accurate results by defining a schedule of activities for occupants and electrical equipment. The attendance schedule for both people and equipment is established from 7 AM to 5 PM. Two strategies for cold water storage—full and partial—are considered, with the full storage strategy encompassing a 16-hour cycle. The building experiences peak load from 1 PM to 4 PM, allowing for the utilization of a cold storage system to either reduce peak demand or achieve peak shaving. The full storage strategy is effective for shifting peak power consumption, while the partial storage strategy focuses on peak shaving. To design a cost-effective system, the capacity of the chiller is calculated first, followed by the sizing of the cold storage tank based on the hourly cooling load of the building. After developing a suitable storage system, its performance at peak load will be examined.

4.3. Evaluation of Cold Storage in Power Generation Facilities

For the design and simulation of a gas fuel power plant, experts consider various equipment and manufacturer specifications while assessing access to different equipment sources. Once the required power, cycle type, and cooling system have been determined, the power plant cycle is designed. The process begins by selecting the cycle type, turbine specifications, fuel type, site characteristics (such as altitude, air temperature, and relative humidity), and the type of turbine inlet air cooling system. Following this, the storage system is integrated into the cycle, and an hourly simulation of the gas turbine cycle along with the storage system is performed. Environmental factors such as altitude, air temperature, and relative humidity are kept consistent with those of the official unit.

4.4. Economic Viability Assessment

The economic section compares the cold-water storage system under both partial and full 16-hour storage strategies, focusing on energy consumption and the power plant unit. The analysis examines peak grid load in the afternoon, calculating electricity saved in the official unit for each strategy. Subsequently, the surplus electricity generated by the power plant at the same time is evaluated. Next, the saved electricity in the official unit is compared with the surplus produced by the power plant. Finally, investment costs for both units are calculated and compared, using economic metrics to determine the more cost-effective investment per kilowatt-hour of electricity.

5. The Governing Equations

In this study, the cooling load requirements of an office unit are modeled, and in addition to the cooling system of the mentioned unit, the cold storage system is also modeled. The modeling of the cold storage system used in the power plant is carried out alongside the modeling of a gas turbine unit.

5.1. Equations for the Refrigeration and Storage Systems

Modeling the performance of a cooling system is one of the steps in modeling the performance of the proposed system. The compression Refrigeration cycle consists of 4 main components: a compressor, a condenser, an expansion valve, and an evaporator. The compressor work can be calculated by

$$W_c = \dot{m}(h_{c,o} - h_{c,i}) \tag{1}$$

where $\dot{m}$ is the mass flow rate of the refrigerant, $h_{c,o}$ is the enthalpy of the refrigerant at the outlet of the compressor, and $h_{c,i}$ is the enthalpy of the refrigerant before being compressed. For nonisentropic compression, the efficiency of the compressor can be calculated by

$$\eta_c = \frac{h_{c,os} - h_{c,i}}{h_{c,o} - h_{c,i}} \tag{2}$$

$h_{c,o\ s}$ represents the enthalpy at the compressor outlet, assuming the use of an isentropic compressor The energy equation for the evaporator and the condenser are as following:

$$Q_{eva} = \dot{m}_r (h_{e,o} - h_{e,i}) \tag{3}$$

$$Q_{cond} = \dot{m}_r (h_{con,o} - h_{con,i}) \tag{4}$$

The expansion valve operates under a constant enthalpy process.

The coefficient of performance (COP) is a suitable metric for evaluating Refrigeration performance and can be calculated for the heating and cooling processes individually by means of Eqs. 6 and 7, respectively:

$$COP_{heating} = \frac{\dot{Q}_{heating}}{\dot{W}_{compressor} + \dot{W}_{auxiliaries}} \tag{5}$$

$$COP_{cooling} = \frac{\dot{Q}_{cooling}}{\dot{W}_{compressor} + \dot{W}_{auxiliaries}} \tag{6}$$

In Eqs. 3 and 4, the energy balance in the refrigerant section was introduced. If water or air is used as the heat source, the energy balance is as follows:

$$\dot{Q}_c = \dot{m}_h c_{p,h} (T_{h,out} - T_{h,in}) \tag{7}$$

If a phase change occurs, for example, in a latent heat pump, the energy transferred to the water is calculated from the following equation:

$$\dot{Q}_c = \dot{m}_h c_{p,c} (T_{h,out} - T_{h,in}) + x\dot{m}_h h_f \tag{8}$$

This section details the equations used to model the capacity of the refrigeration unit, taking into account the varying time frames and functions associated with cold water and ice storage technologies, as expressed in the following relationship [42]:

$$\text{NominalChillerSize} = \frac{TCL}{H_{charg} CR_{charg} - H_{DC} CR_{DC}} \tag{9}$$

In this equation, TCL is total daily cooling load of the building from 8 to 19, which is for the designing day. CR_{charg} is the chiller capacity ratio (chiller capacity correction coefficient) when charging and CR_{DC} is chiller capacity ratio when direct cooling.H_{charg} is a period of time which refrigerator operates for charging the storage.H_{DC} is a period of time that the chiller operates for direct cooling. Subsequent Eqs. 10 and 11 are employed to calculate the volumes of the ice storage tank and cold-water storage tank, respectively [42]:

$$Vtank = Estorage \cdot \gamma \tag{10}$$

whereV_tank is the required volume of the storage tank (m^3), E_storage is the cooling energy demand during storage period (kWh), and γ is the conversion factor from energy to volume, assumed to be 0.020 m^3/kWh:

$$V_{tank} = (E_{storage} \times 3600) / (\Delta T \times \rho \times c_p \times FOM) \tag{11}$$

whereV_{tank} is the volume of chilled water or ice storage tank (m^3); $E_{storage}$ is the required cooling energy (kWh); ΔT is the temperature difference during discharge period (°C); ρ is the density of water (kg/m^3), typically 998 kg/m^3; c_p is the specific heat capacity of water (kJ/kg·°C), typically 4.2; FOM is the figure of merit of the storage system (dimensionless); and 3600 is the conversion from kWh to kJ

In this equation, Δk is the tank inlet and outlet temperature difference. FOM is the tank's performance coefficient or load recovery coefficient stored in the tank. The full capacity characteristics of the cold storage system are described by the charge and discharge rates in the following equation [43]:

$$\dot{Q}_{ice} = u \times \frac{Q_{TES}}{\Delta t} \tag{12}$$

where $\dot{Q}_{ice}$ is charge (+) or discharge (-) rate of cold storage system (kW), Q_{TES} is storage capacity (kWh), u is charge or discharge rate (fractional), and Δt is simulation period (h). Three operating modes can be considered for the entire continuous storage or discharge rate, which are: rest mode (u = 0); charging mode (u > 0); discharging mode (u < 0).

5.1.1. Charge Mode

In this case, the chiller dedicated to the cold storage system produces enough amount of ice. After each periodΔt, the ice level formed in the cold storage system (x) increases by the following equation [43]:

$$x_t = u\Delta t + x_{(t-\Delta t)} \tag{13}$$

where x_t is ice surface formed at present and $x_{(t-\Delta t)}$ is ice surface formed in the past.

5.1.2. Discharge Mode

The cooling capacity of the cold storage system is determined by the discharge rate (u < 0). Mass flow rate of water in the cold storage system, using Eq. 11 of the cooling load supplied by cold storage system and temperature difference inlet and outlet water are obtained [43]:

$$\dot{m}_{ice} = \frac{\dot{Q}_{ice}}{C_{p,water}(T_{inlet} - T_{Loopsetpoint})} \tag{14}$$

where $\dot{m}_{ice}$ is the mass flow rate of water in the cold storage system (kg/s), $\dot{Q}_{ice}$ is cooling load (W), $C_{p,water}$ is specific heat capacity of water (j/Kg°C), T_{inlet} is inlet water temperature (°C), and $T_{loopset\ point}$ is outlet water temperature set (°C).

5.2. Modeling of the Power Plant Storage System

Fig. 7 provides an overview of the modeling layout, with each number representing different components of the cooling cycle. As depicted, when the cooling process is activated, the inlet air first passes through the air filter (step 2) and is subsequently cooled by the coil, resulting in a temperature reduction to 15 °C. The cooled air then enters the compressor at this temperature, setting the stage for efficient energy conversion.

The isentropic process is assumed for simulation of the turbine and the compressor in this study. Therefore, the exhaust air temperature from compressor and turbine can be calculated by Beghi and Mehdipour [29]:

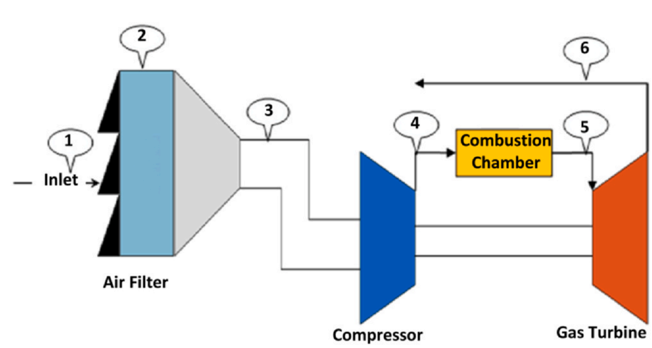


Fig. 7. Schematic of a simple gas turbine cycle. 1) Gas turbine air intake, 2) gas turbine filter, 3) compressor inlet, 4) compressor outlet, 5) turbine inlet, and 6) turbine outlet [29].

$$T_4 = T_3 + \frac{T_3}{\eta_C}\left[(R_{PC})^{\frac{\gamma-1}{\gamma}} - 1\right] \quad (15)$$

$$T_6 = T_5 + T_5 \quad \eta_t [1 - \left(R_{pt}^{\frac{\gamma-1}{\gamma}}\right)] \quad (16)$$

The efficiency of adiabatic expansion ht defines the ratio of the work extracted by the turbine to the ideal work extracted in an isentropic expansion. In a manner similar to that discussed for the compressor (η_C). Here, R_{pc} and R_{pt} denote pressure ratio of the compressor and the turbine, respectively, and γ accounts for the ratio of the specific heats. The works of the compressor and the turbine along with the network are calculated by [29]:

$$W_C = C_p(T_4 - T_3) \quad (17)$$

$$W_t = \quad C_p(T_5 - T_6) \quad (18)$$

The network equals the difference in turbine work and compressor work [29]:

$$W_{net} = W_t - W_C \quad (19)$$

The efficiency of the turbine must be expressed in theoretical and real values, as given by Beghi and Mehdipour [29]:

$$\eta_{th,theory} = 1 - \frac{1}{\gamma - 1/\gamma} \quad (20)$$

$$\eta_{th,theory} = 1 - \frac{\frac{T_5}{T_{amb}}\left\{1 - \eta_t(1 - (R_{pt})^{\frac{\gamma-1}{\gamma}})\right\} - 1}{\frac{T_5}{T_{amb}} - \left\{1 + \frac{1}{\eta_c}((R_{pc})^{\gamma - \frac{1}{\gamma}} - 1)\right\}} \quad (21)$$

Heat added to the combustion chamber [29]:

$$Q_{add} = \dot{m}_t \times C_{pg} \times (T_5 - T_4)/\eta_c = \dot{m}_f \times LHV \quad (22)$$

where LHV denotes the low heat value of the fuel. This relation depicts the transferred heat to the cycle for which η_c, is the efficiency of the combustion chamber. The net output power is equal to the difference in power output and the internal losses of the power plant [29]:

$$power_{net} = power - \quad p_{IL} \quad (23)$$

The internal losses of the power plant include the energy consumed by the various units of the power plant. It should be noted that for the non-cooled and fog modes, the electricity consumed by the electric chiller is considered zero. Gas turbine heat efficiency [29]:

$$\eta_{th} = \frac{power}{Q_{add}} = \frac{power}{\dot{m}_t \times C_{pg} \times (T_5 - T_4) \times \eta_{cc}} \quad (24)$$

5.3. Economic Analysis

The storage system, whether in official units or power plants, comprises 3 primary components: chillers, cold water storage tanks, and water pumps. These components account for a significant portion of the total investment cost. Additional investment costs include peripheral items such as foundations and plumbing. The following equations outline the methods for calculating the costs associated with the main equipment:

Chiller cost calculation [29]:

$$Z_{Chiller} = C_1 \times \dot{Q}_{Chiller} \quad (25)$$

where $Z_{Chiller}$ is the cost of the chiller (USD), $\dot{Q}_{Chiller}$ is the cooling capacity of the chiller (kW), and $C_1 = 150.2$ USD/kW is the unit cost coefficient based on Beghi and Mehdipour [29].

Calculation of storage tank cost [29]:

$$Z_{ST} = 8.67 \times 10^{[2.9211\exp(0.1416\times\log V_{ST})]} \quad (26)$$

where Z_{ST} is the cost of the storage tank (USD) and V_{ST} is the volume of the storage tank (m^3). The formula is based on empirical cost correlations reported in [29].

Calculation of storage tank cost [29]:

$$Z_{pump} = 705.48 \times \dot{W}_{pump}^{0.71}\left(1 + \frac{0.2}{1 - \eta_{pump}}\right) \quad (27)$$

where Z_{pump} is the cost of the water pump (USD), $\dot{W}_{Pump}$ is the rated power input of the pump (kW), and η_{Pump} is the pump efficiency.

6. Verification

6.1. Validation of the Residential Unit Mode

To analyze the thermal load modeling of the building and the ice storage system, references [8,42] can be utilized. This system involves daily ice production and storage. The authors developed a module for ice-cold energy storage systems within the EnergyPlus software. This module uses the building load and thermodynamic system module to model storage systems. The cold storage system is integrated into EnergyPlus as part of the cooling system and is capable of operating at any charge and discharge rate according to the input parameters. The simulation results presented in this paper are compared with those from a similar study by Ihm [43], as shown in Fig. 8. The results of our simulation were compared with those of Pyongyang's study in Figs. 9 and 10. Fig. 9 presents the power consumption of the chiller, while Fig. 10 illustrates ice levels during charging and discharging, where charging corresponds to the positive slope and discharging to the negative slope. Accuracy in calculating the volume of ice produced and stored reflects the precision in determining the cooling and heating load of the building as well as the accuracy of the ice production system modeling. The comparison indicates a 7.35% error between our model and the original research results, as shown in Fig. 10.

6.2. Validation of the Power Plant Model

The validation process began with a simulation of the gas unit at Qom Power Plant No. 1, using data from July 19, 2018, as presented in Table 5. The complete specifications of the power plant, along with the climatic conditions under which the actual performance data were

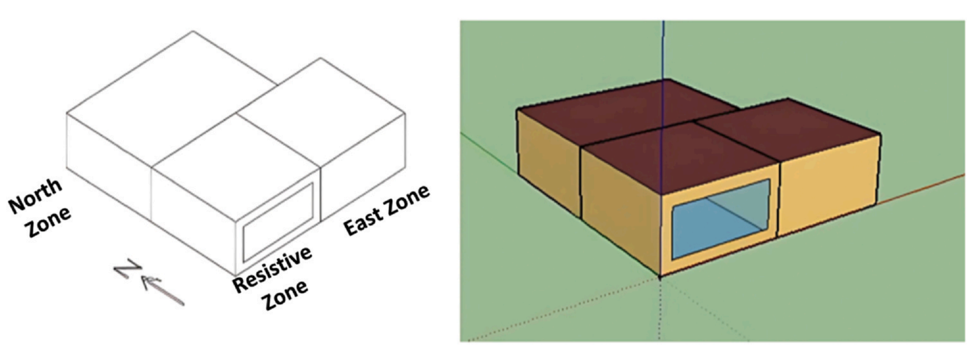


Fig. 8. Right: Building modeled for this study; Left: Building in Ihm's study [43].

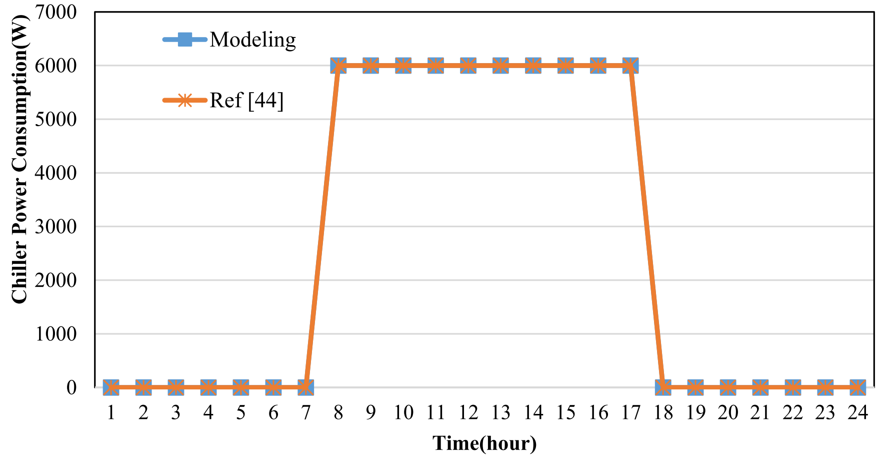


Fig. 9. Comparison of chiller power consumption between Iham's study [43] and current model results.

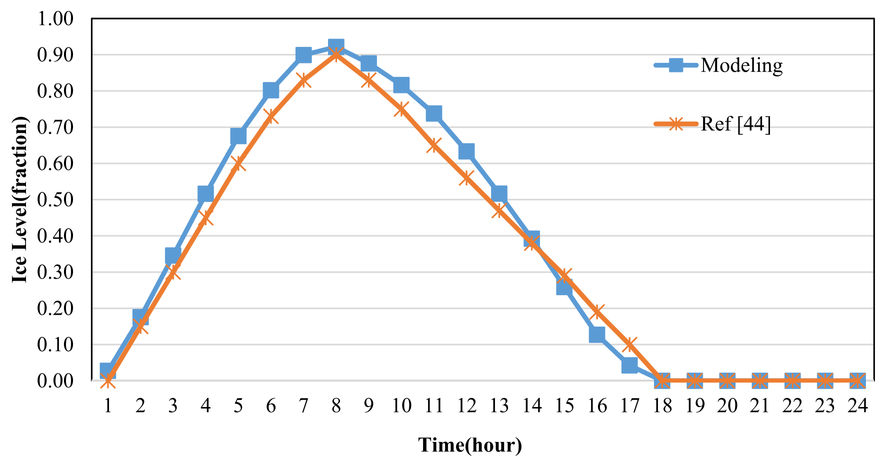


Fig. 10. Comparison of ice storage tank volume between Iham's study [43] and current model results.

recorded, are thoroughly detailed in reference [29]. The simulation was carried out using thermodynamic modeling software, enabling a comparison between the turbine's net power output under real and simulated conditions, as shown in Fig. 11.

During the cooling system's inactive hours (2 a.m. to 10 a.m.), the average deviation between actual and simulated results was 0.62%; while active (11 a.m. to 1 p.m.), this deviation averaged 0.45%.

The second step of validation compared our modeling results to those reported in studies by Sanaye and Tahani [44] and by Bhargava and Homji [45]. Bahar Gawa's research simulated the impact of inlet air cooling on gas turbines in 4 conditions: 1) no cooling, 2) inlet fogging, 3) inlet fogging with a 1% surplus spray, and 4) inlet fogging with a 2% surplus spray. The turbine specifications and environmental settings are detailed in Table 6.

Sanaye's study modeled the effects of inlet air cooling across 16 different gas turbines, applying the same 4 cooling modes and environmental settings as described in [45] and shown in Table 6. Water injection into the combustion chamber was also analyzed. In our study, we validated 3 turbine parameters—net power output, compressor inlet air temperature, and compressor outlet air temperature—against the findings of these studies [44,45]. The modeling accurately replicated the gas turbine's performance, with the greatest deviation occurring in net power output at 6.6%, as shown in Table 7.

6.3. *Sensitivity Analysis*

One of the main foundations of this study is the increase in cooling load demand and the reduction in gas turbine power plant performance due to rising temperature and humidity during the hot months of the year. While the first topic—seasonal variations in building cooling load—is discussed separately, it is essential to evaluate the extent to which ambient temperature and humidity impact power plant performance. To this end, the performance of a gas turbine power plant was analyzed for 3 hot months—(a) June, (b) July, and (c) August—under the following 3 scenarios:

Case 1. Gas turbine without any inlet air cooling

Case 2. Gas turbine with fogging system (evaporative cooling)

Case 3. Gas turbine with air-cooled chiller system without thermal storage (Direct Cooling).

Table 5
Gas turbine specifications and Qom power plant site details

Turbine fuel	Ambient relative humidity (%)	Ambient air temperature (C)	Type of cooling system	Sea level (m)	Manufacturer	Turbine model
Natural gas	5.54 up to 19.78	31.24 up to 41.69	Fog	1020	Mitsubishi	W701D

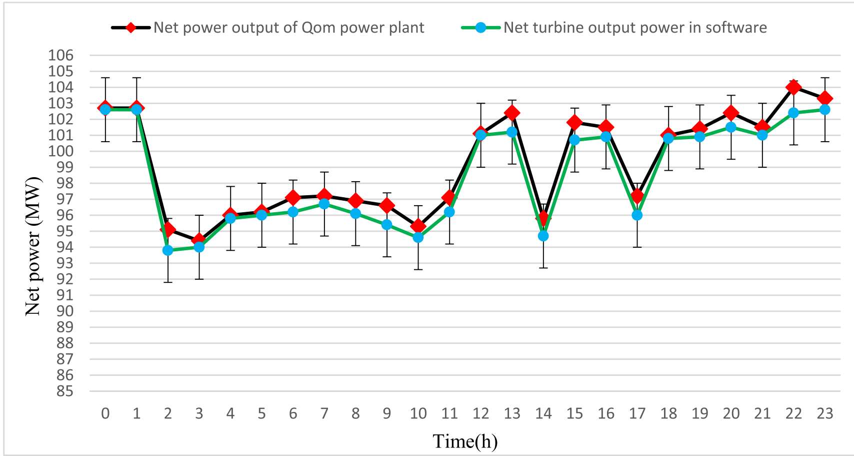


Fig. 11. Comparison of Qom power plant data [29] and simulation results on July 19, 2018.

Table 6
Turbine specifications and site conditions from study [44]

Turbine fuel	Ambient relative humidity (%)	Ambient air temperature (C)	Ambient pressure (bar)	Output power (MW)	Manufacturer	Turbine model
Natural gas	40	43	1.013	124.7	General electric	GE9171E

The performance results of these 3 cases are presented in Fig. 12. Among various inlet air cooling methods, the fogging system is one of the most widely used and cost-effective techniques. However, its effectiveness is highly dependent on the ambient humidity; in more humid climates, the cooling capacity of the fogging system significantly decreases. In contrast, the air-cooled chiller system demonstrates minimal dependency on ambient humidity levels.

By comparing these two cooling approaches, the impact of humidity on turbine performance can be inferred indirectly. Although the COP of chillers typically depends on the temperature difference between heat sources, in gas turbine cooling system design, the inlet air temperature is generally maintained within a fixed range (typically 15–25 °C, depending on the turbine model), and the condenser temperature is designed based on the highest expected ambient temperature at the plant's location. As a result, system performance remains relatively stable throughout the year despite changes in ambient conditions.

The results indicate that improvements in turbine performance during hot months are considerably greater than during cooler months. In fact, performance enhancement of over 20% can be achieved in the hottest conditions using inlet air cooling systems—effectively preventing significant performance degradation.

It is important to note that the chiller system consumes electricity from the power plant itself, whereas the fogging system requires minimal electrical energy. In the following sections, it will be demonstrated that if the required cooling power can be supplied during off-peak hours, the power plant can operate more consistently and respond more effectively to end-user demand.

On the other hand, if the performance of the gas turbine without any cooling system is evaluated over a 24-hour period with a temperature variation of approximately 18 °C, it can be observed that the output

Table 7
Validation results

Compressor outlet air temperature (°c)	Inlet air temperature to the compressor (°c)	Net power output (Mw)		
387	43	101	Article answer [44]	First case without cooling
386	43	100.381	Article answer [45]	
385.7	43	100.3	Modeling results	
0.33%	0%	0.69%	Percentage of disagreement with article [44]	
0.07%	0%	0.08%	Percentage of disagreement with article [45]	
380.3	30.6	110.7	Article answer [44]	Second case spraying in saturated conditions
371	30	110.868	Article answer [45]	
373.1	30	112.5	Modeling results	
1.9%	2%	1.62%	Percentage of disagreement with article [44]	
0.5%	0%	1.4%	Percentage of disagreement with article [45]	
334.7	29.99	127.2	Article answer [44]	Third case spraying with 1% surplus spray
330	30	121.216	Article answer [45]	
334.9	30	126.3	Modeling results	
0.05%	0.03%	0.71%	Percentage of disagreement with article [44]	
1.4%	0%	4.1%	Percentage of disagreement with article [45]	
311.7	29.91	139.26	Article answer [44]	Third case spraying with 2% surplus spray
293	30	130.363	Article answer [45]	
300	30	139.03	Modeling results	
3.9%	0.3%	0.16%	Percentage of disagreement with article [44]	
2.3%	0%	6.6%	Percentage of disagreement with article [45]	

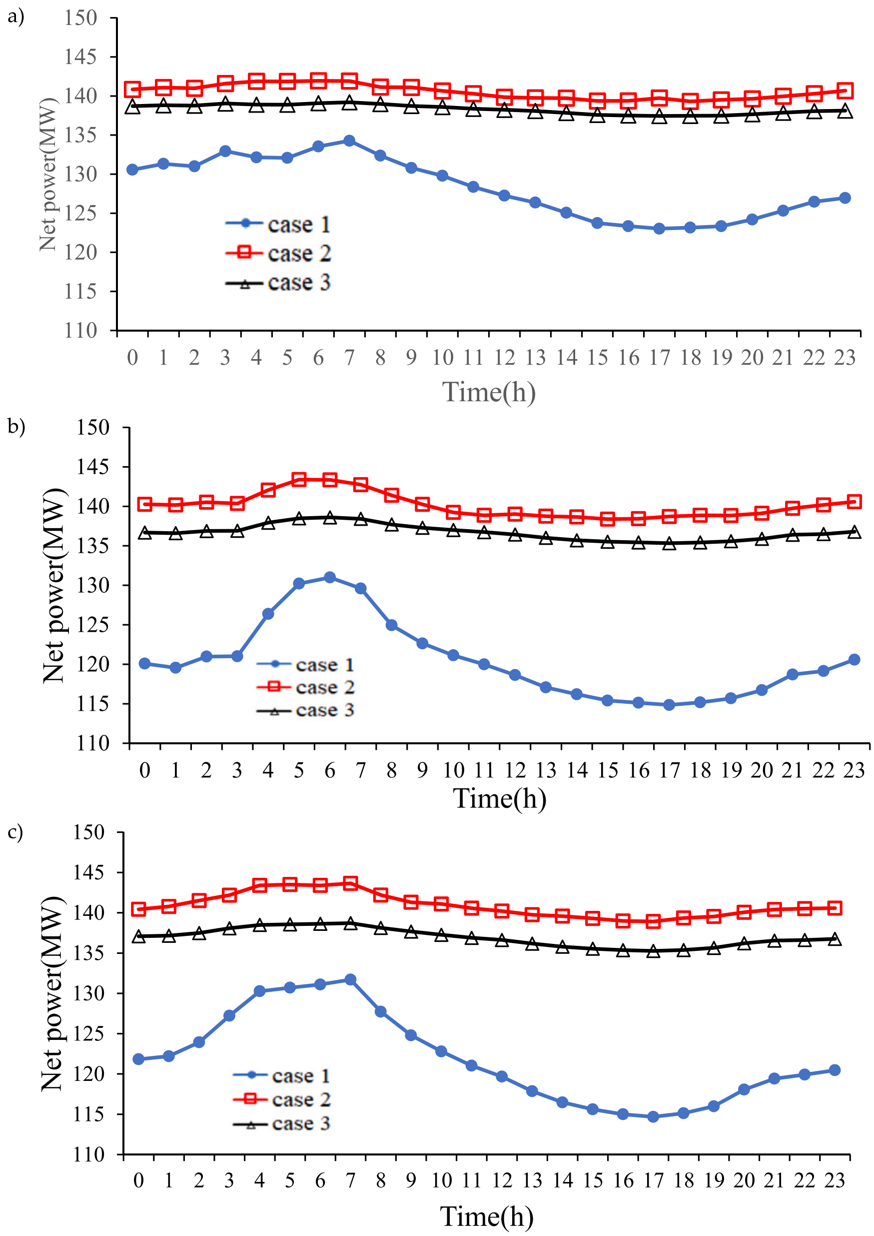


Fig. 12. Comparison of net efficiency gas turbine output in different modes, related to a) June, b) July, and c) August.

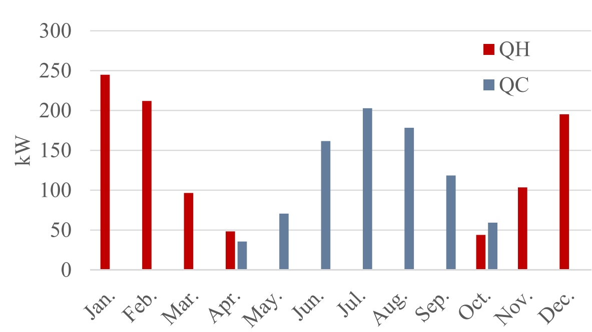


Fig. 13. Average thermal and cooling load of the building throughout the year.

power decreases from 131 MW to 114 MW. This significant drop clearly demonstrates the strong influence of ambient temperature on gas turbine performance. While humidity has a relatively minor direct effect on turbine output, it can significantly impact the effectiveness of the cooling system—particularly when a fogging system is used. In contrast, if an air-cooled chiller system is employed, ambient humidity has a negligible effect on cooling performance.

7. Results

7.1. Performance of Cold Storage System in Office Unit

In the initial step, the required cooling load is calculated, followed by determining the building's electricity consumption for the cooling system without storage (i.e., direct cooling). In Fig. 13, the cooling and heating load of this building throughout the year is shown. Fig. 14 illustrates the required cooling load for the designated office building in Arak. Since the building is an office, 24-hour cooling is unnecessary; therefore, operational hours are set from 7:00 a.m. to 5:00 p.m. According to meteorological data (Fig. 2), during the peak demand period from 10:00 a.m. to 5:00 p.m., the ambient temperature around the chiller rises, reaching its maximum at 4:00 p.m., as anticipated.

Fig. 15 displays the electricity consumption of the chiller throughout the day in a direct cooling setup across 4 cities. Energy

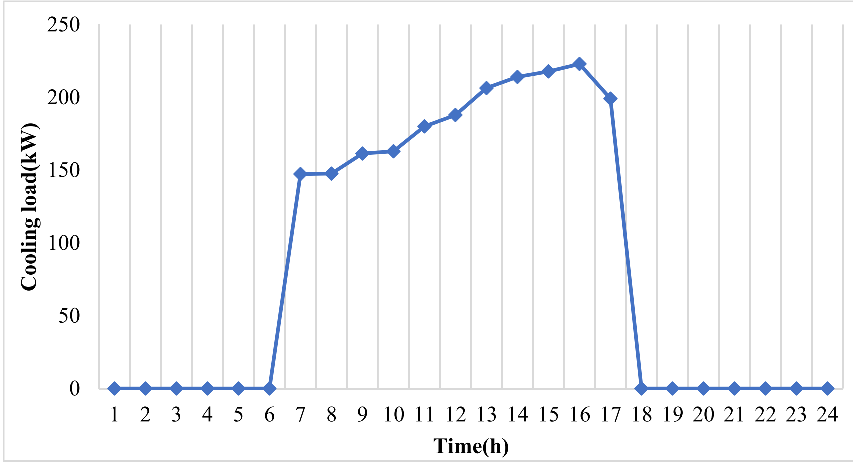


Fig. 14. Required cooling load in Arak.

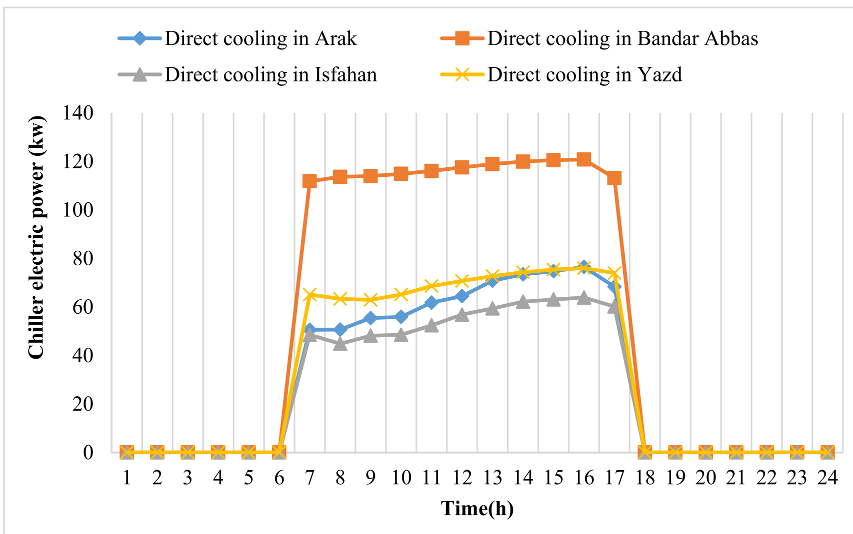


Fig. 15. Chiller electricity consumption in direct cooling for 4 cities in Iran.

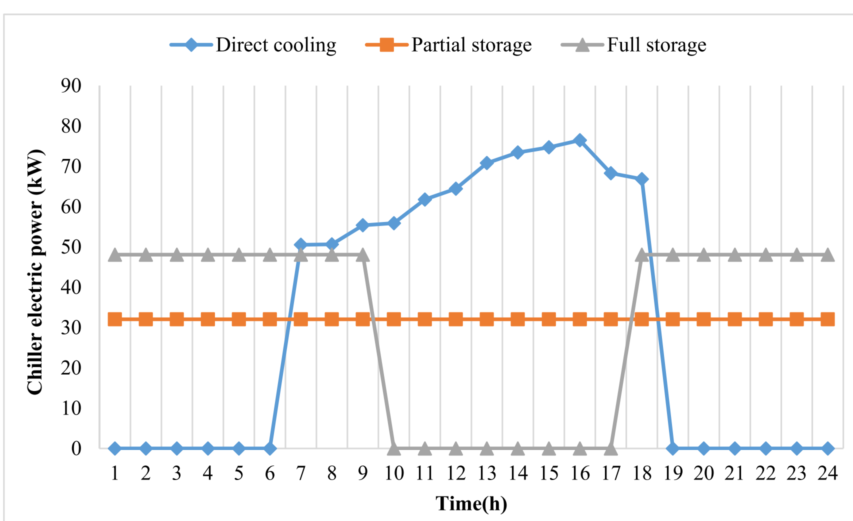


Fig. 16. Chiller power consumption in direct cooling, partial storage, and full storage strategies for Arak.

Table 8
Storage tank volume specifications for office unit

Cold storage strategy	Cold water storage technology
	Tank volume (m^3)
Full 16-hour storage	109.4
Partial storage	97.24

consumption calculations for each city show that, due to heat loads stored in the building's walls from the previous day, there is an increase in demand during the early hours. Based on the temperature data in Fig. 2, the peak demand from 10:00 a.m. to 5:00 p.m. results in a rise in ambient temperature around the chiller, reaching its highest point at 4:00 p.m., as expected.

Fig. 16 compares power consumption across direct cooling, partial storage, and full storage strategies for the office building during peak demand hours. With partial storage activated at 4:00 p.m., electricity consumption is reduced by 58.08% compared to direct cooling.

Table 8 outlines the tank characteristics under both consumption strategies, while Fig. 17 depicts the energy discharged from the storage tank. Due to its climate, Bandar Abbas experiences a notable decrease in tank discharge at 2:00 p.m. Despite higher dry air temperatures than in other cities, high humidity after 1:00 p.m. reduces solar exposure on the building, resulting in lower discharge energy requirements compared to other cities. However, at peak global consumption, Bandar Abbas, situated in a hot and humid climate, still consumes more energy than other cities.

It can be observed that the partial storage system requires a smaller tank and device, but due to electricity usage during peak consumption, it creates a lesser peak-shaving effect. In contrast, the full storage system requires a larger tank and device, which means higher initial costs and more space requirements, but it performs better in terms of peak-shaving. Depending on the region and supporting policies, the most suitable method may vary. If the primary criterion is peak shaving, it can be said that the full storage method is preferred.

7.2. Results of Cold Storage System Operation in Power Plant Unit

The power plant's performance was analyzed under 3 conditions: a) no inlet air cooling, b) inlet air cooling with partial cold storage, and c) inlet air cooling with a full 16-hour cold storage system. Fig. 18 illustrates the net power output of the turbine across 4 modes—full 16-hour storage, partial storage, direct cooling, and no cooling—based on the ambient temperature and environmental conditions in Arak. The temperature fluctuations, as shown in Fig. 2, help explain the variations in the turbine's net power output. In the early morning (around 10:00 a.m.), Arak's ambient temperature is relatively low, but it rises steadily throughout the day.

These temperature fluctuations directly affect the gas turbine's performance. In the scenario without cooling, the turbine's net power output is highest between midnight and 10:00 a.m. but declines as temperatures increase, reaching its lowest point around 4:00 p.m. In the direct cooling method, the power output rises from midnight to 10:00 a.m. when demand is minimal. However, during peak hours (10:00 a.m. to 5:00 p.m.)—when demand is at its highest—the turbine's performance declines.

In the partial storage method, the turbine's net power output remains more stable throughout the day. Here, part of the turbine's output is allocated to cooling energy storage, resulting in a relatively consistent power output, albeit slightly lower than with direct cooling. During peak demand hours, however, turbine performance improves compared to direct cooling, helping to prevent significant power degradation and output reduction.

In the full storage strategy, turbine power output is lower than in the direct cooling and partial storage methods from midnight to 10:00 a.m., a period of low demand. More turbine power is used for cooling storage during this time compared to other strategies. However, during peak demand hours, the chiller is turned off, allowing maximum turbine output. This strategy shifts chiller power consumption from peak hours to low-load hours, maximizing power output when demand is highest. Overall, the partial storage strategy is effective for peak shaving, while the full storage strategy is better for peak shifting.

Fig. 19 illustrates the variation in turbine power output for partial and full storage modes compared to the non-cooled state. The findings reveal that partial storage performs more effectively than full storage during the early hours, from midnight to 10:00 a.m. However, during peak hours (10:00 a.m. to 5:00 p.m.), the full storage strategy surpasses partial storage in performance, as the chiller shuts off by 4:00 p.m. At this point, the power output improvement relative to the non-cooled state is 17.97% for full storage and 15.98% for partial storage.

This difference can be attributed to the midday rise in air temperature, which typically decreases gas turbine efficiency. The cold storage system, by cooling the turbine inlet without consuming additional energy, effectively mitigates this performance loss. In contrast, the partial storage system requires a small amount of energy for cooling, leading to the observed difference in performance levels between 17.97% and 15.98%.

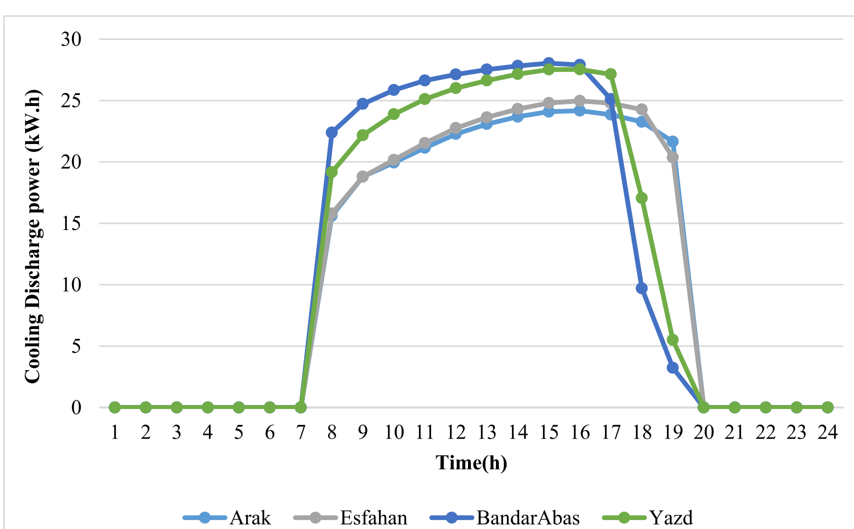


Fig. 17. Power consumption for storage tank discharge across different cities in Iran.

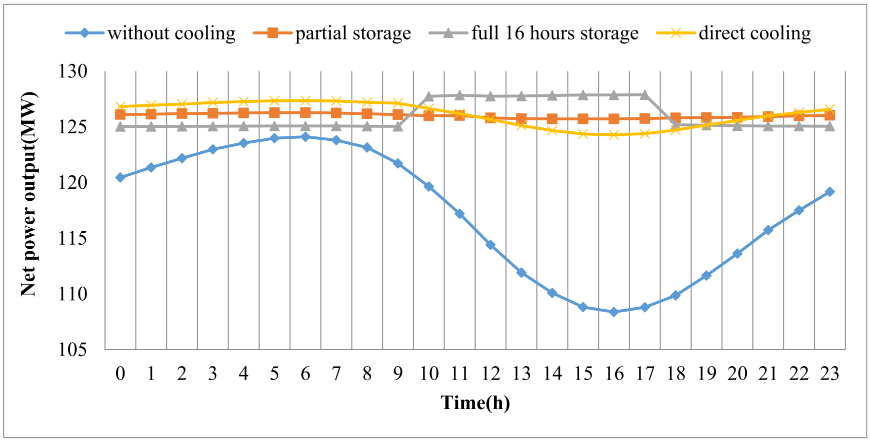


Fig. 18. Net power output of gas turbine under various air-cooling strategies during a July day in Arak.

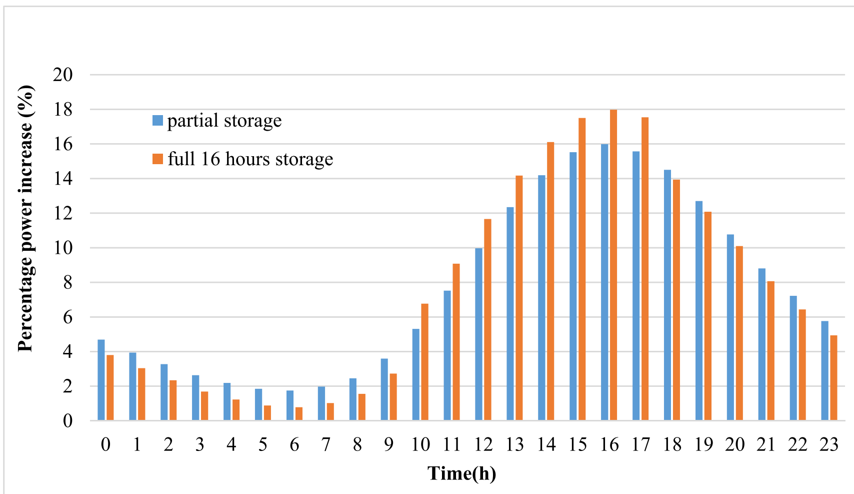


Fig. 19. Percentage change in turbine output power for partial and full storage strategies compared to no-cooling conditions.

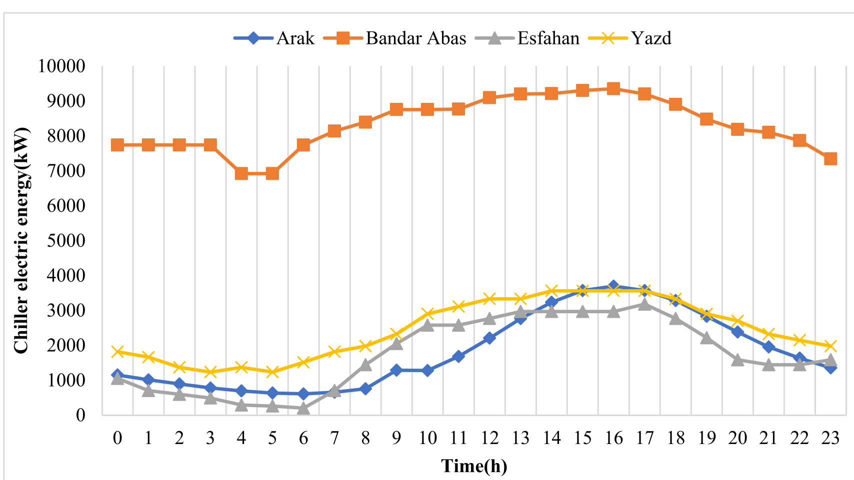


Fig. 20. Chiller power consumption for power plant cooling in 4 Iranian cities.

Fig. 20 illustrates the power consumption of the chiller in direct cooling mode at the power plant unit across 4 cities. The results indicate that Bandar Abbas experiences the highest electricity consumption, especially during peak hours, due to its warmer climate compared to cities with cooler weather. In tropical regions like Bandar Abbas, the cold storage system has a significant impact on shifting electricity consumption from peak hours, aiding in power management.

Figs. 21 and 22 compare chiller power consumption in direct cooling, partial storage, and full storage modes for Bandar Abbas and Arak. The results show that, under all cooling conditions, the chiller's power consumption in Bandar Abbas is higher than in Arak due to their respective warm-humid and moderate-dry climates. This confirms that chiller power consumption is directly related to air temperature. Table 9 provides the cold-water storage tank volumes for both full and

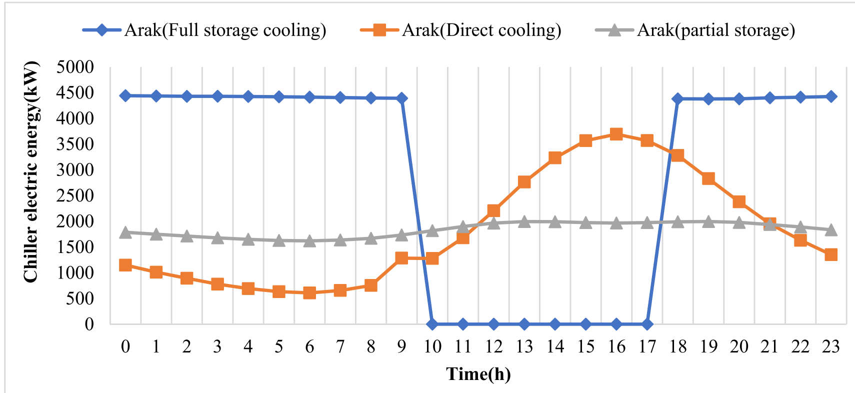


Fig. 21. Chiller power consumption in direct cooling, full, and partial storage modes in Arak.

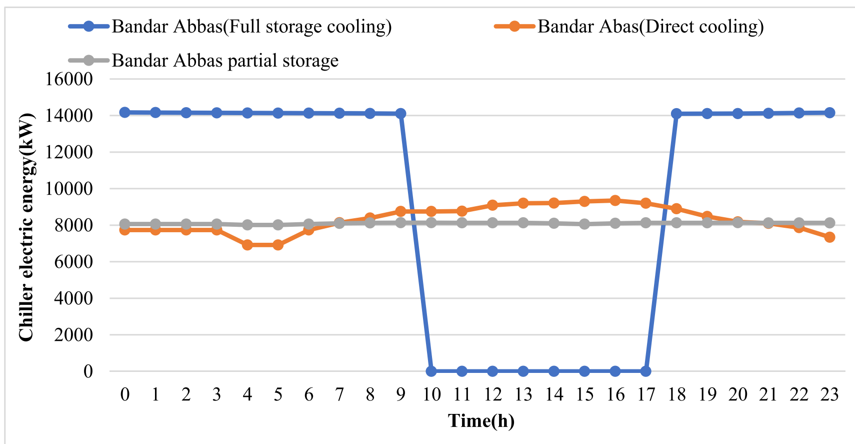


Fig. 22. Chiller power consumption in direct cooling and full and partial storage in Bandar Abbas.

Table 9
Storage tank volume in power section for full and partial storage strategies

Cold storage strategy	Cold water storage technology
	Tank volume (m^3)
Full 16-hour storage	2900
Partial storage	2581.8

partial storage strategies.

It should be noted that Bandar Abbas has a hot and humid climate, and the cooling system operates continuously throughout the day and night. Therefore, the midday peak electricity demand is not significantly higher than nighttime consumption. As a result, the partial storage system does not lead to a considerable reduction in peak demand. On the other hand, although the full storage system increases nighttime electricity consumption, it effectively reduces midday demand. From the perspective of urban-scale electricity management—-where reducing midday peak consumption is a priority—the full storage method proves to be more effective in Bandar Abbas.

In Arak, which has a hot, dry, and mountainous climate, full storage also results in a better peak-shaving effect. However, partial storage can still be considered effective. The use of smaller equipment and smaller storage tanks in the partial storage system may be a deciding factor in specific cases, depending on the design priorities.

7.3. Economic Analysis

The previous sections analyzed the thermal performance of 2 storage systems: one in an electricity-consuming unit (a large office building) and the other in an electricity-generating unit (a power plant). It is important to emphasize that these systems differ significantly in size and scale. Therefore, a fair comparison requires determining the equivalent scale at which both systems achieve similar levels of peak shaving.

Cold storage systems, due to their large reservoirs, are generally more practical and economical when implemented on a larger scale or as part of interconnected energy systems. In the earlier sections, the analyzed unit was a relatively large office building, which benefits from better conditions compared to smaller residential units. However, as the scale of a cold storage system increases and its implementation becomes more centralized, the costs of storage and finding suitable space for the reservoir become more manageable.

In the case of cold storage at power plants, the reservoir size is significantly larger, equivalent to multiple reservoirs used in office building systems. Consequently, the power plant approach is expected to offer better feasibility and cost-effectiveness in terms of implementation and economic viability.

In this study, a simplified economic analysis has been conducted, focusing on a comparison between 2 cold thermal energy storage strategies: one implemented at the power plant and the other at the building level (e.g., office units). These 2 strategies are assumed to have equal capacity and similar effectiveness in peak demand reduction. Therefore, the comparison is limited to initial capital investment without considering life cycle cost (LCC) or operational expenditures.

The underlying assumption is that, for example, the amount of peak load reduction achieved by a centralized storage system at the power plant could be matched by several distributed storage systems in office buildings. As a result, the operational cost savings of both approaches would likely be of the same order of magnitude. Thus, comparing the

Table 10
Comparison of electricity savings and surplus electricity generation in office and power plant applications

	Amount of electricity saved by the storage (at 16:00 pm) in the official unit (kW)	Amount of electricity added to the network by storage cooling at the power plant per hour (kW)	The number of official units that need to install cold storage to reduce consumption equivalent to a power plant
Partial cold-water storage	44.41	17327	390.15
Full 16-hour cold water storage	76.45	19485	254.87

Table 11
Parameters for economic evaluation

	Power plant		official	
	Partial storage	Full 16-hour storage	Partial storage	Full 16-hour storage
$\dot{Q}_{Chiller}$(kW)	1827.39	2741.09	32.04	48.06
V_{ST}(m^3)	2581	2904	97	109
$\dot{W}_{Pump}$(kW)	6	9.8	1	1.64
η_{pump}(%)	82	82	90	90

initial capital costs of the 2 options provides a practical and insightful basis for decision-making. The initial investment cost is calculated based on the required storage capacities derived in the previous sections, using Eqs. 25 to 27.

The results indicate that 4:00 p.m. is the most critical time for peak electricity demand. Consequently, the economic comparison was based on this peak period, as shown in Table 10. The findings reveal that, in partial cold water storage mode, 390.15 office units would be required to achieve the same peak shaving as 1 power plant. In full 16-hour cold water storage mode, the equivalent is 254.87 office units. These values, referred to as the equivalence coefficient, allow us to calculate initial investment costs in the next steps.

In this study, we compare the initial investment costs of the office and power plant storage systems, as peak shaving rates are standardized for both systems, rendering ongoing operational costs similar and, therefore, irrelevant to this economic analysis. By comparing initial costs, we can identify which system offers greater economic efficiency for public policy. Table 11 presents the parameters needed to calculate the cost of acquiring storage systems for both office and power plant units. Table 12 summarizes the initial investment costs, including the chiller, storage tank construction, pump, and ancillary expenses for both types of units.

In a full storage strategy, chiller costs are higher than in partial storage since, with full storage, the chiller is inactive for several hours daily, requiring a larger chiller to meet cooling needs when it is operational. Consequently, the storage tank in a full storage strategy is also larger than in partial storage. Table 13 details the costs for each component of the storage systems in both office and power plant units. The calculation of the equivalence coefficient shows that the initial investment cost for a power plant storage system is considerably lower than for an office storage system.

A key factor in cost reduction is implementing the cold storage system on a larger, more centralized scale within power plants. This approach significantly lowers equipment costs. For example, the cost of cold storage in full storage and partial storage (for the office building) modes is $81,576.2 and $72,948.0, respectively. However, achieving the same level of peak shaving in a power plant costs only $3635.1 and $1925.4 for full and partial storage modes, respectively.

This remarkable cost reduction—95.54% for full storage and 97.36% for partial storage compared to equivalent office systems—results from economies of scale and the reduced impact of fixed and overhead costs in large-scale operations. Consequently, partial storage in power plants stands out as the most cost-effective solution.

It is also worth noting that while full storage in both office and power plant systems offers better peak shaving performance, it entails higher costs due to the larger size of tanks and associated equipment.

Overall, cold thermal storage systems are more favorable at the power plant level. If peak load reduction is the main decision-making criterion, full storage is the optimal choice due to its superior effectiveness—particularly since space constraints and initial investment concerns are less significant in power plant applications. However, if

Table 12
Initial investment cost calculation for storage system equipment

		Chiller ($)	Water tank ($)	Pump ($)	Side costs
Office storage device	Partial storage	4812.40	63902.68	2 × 2116.44	11088.08
	Full 16-hour storage	7218.61	68346.94	2 × 3005.34	12399.58
Power Plant Storage	Partial storage	27473.97	471432.04	5299.28	111178.38
	Full 16-hour storage	411711.71	507246.54	7517.24	137118.3

Table 13
Comparison of initial investment costs in power plant and office units

	Office storage		Power Plant Storage	
	Partial storage	Full storage	Partial storage	Full storage
Chiller ($)	4812.4	7218.6	274474.0	411711.7
Storage tank ($)	63902.7	68346.9	471432.0	507246.5
Pump ($)	4232.9	6010.7	5299.3	7517.2
Total unit cost ($)	72948.0	81576.2	751205.3	926475.5
Equivalence coefficient	390.15	254.87	—	—
The equivalent cost per Office storage unit. (by applying the equivalence factor) ($)	72948.0	81576.2	1925.4	3635.1
Cost reduction (%)	—	—	97.36	95.54

minimizing initial costs is a priority, partial storage—requiring smaller equipment and tanks—may be the preferred option.

The primary role of cold storage systems lies in shifting peak energy demand rather than directly reducing overall energy consumption. This capability is particularly valuable in regions where electricity tariffs can be leveraged as a tool for effective energy management, making these solutions economically viable and practical for grid stability.

This study specifically focuses on peak cooling demand, a critical factor in warm climates such as those in certain regions of Iran. Conversely, colder or cold-humid climates face fewer challenges related to cooling loads, and therefore, these conditions were not the primary focus of this research. However, regions with warm climates and notable day-night temperature variations offer favorable conditions for enhancing the effectiveness and cost-efficiency of cold storage system designs. By targeting these conditions, the systems can significantly enhance energy efficiency and reduce peak loads, thereby contributing to a more sustainable energy infrastructure.

7.4. Challenges of Cold Thermal Energy Storage Deployment

Although cold thermal energy storage (CTES), including chilled water and ice storage systems, is recognized as an effective solution for reducing peak electricity demand and improving cooling system efficiency, its large-scale deployment—particularly in urban environments—faces several practical challenges.

One of the main limitations is the space requirement for installing storage tanks. In densely populated urban areas, especially in residential buildings, allocating sufficient space for large thermal storage units can be highly challenging. These constraints are further complicated by esthetic considerations, building regulations, and safety requirements.

Administrative and commercial buildings generally offer slightly more favorable conditions. They are often equipped with more advanced mechanical systems, and rooftop or basement spaces may be available for equipment installation. Nevertheless, these applications still face difficulties such as maintenance costs, integration with existing Heating, Ventilation, and Air Conditioning systems, and specific technical limitations.

In contrast, power plants present a more suitable environment for the implementation of CTES systems. They typically have access to larger, more affordable land areas and allow for long-term infrastructure planning. Consequently, managing the aforementioned challenges becomes significantly easier, enabling more efficient and cost-effective deployment of chilled water or ice storage systems.

8. Conclusion

This study presents a comprehensive analysis of cold storage systems applied in both electricity consumption (office units) and electricity generation (power plants). By simulating various operational scenarios, including direct cooling and different levels of cold storage, the research highlights how these systems can effectively manage peak electricity demands. The results indicate that the peak demand period, specifically at 4:00 p.m., is critical for both systems, with significant variations in power output and energy consumption based on cooling strategies. Notably, the 16-hour full storage method significantly enhances the turbine's net power and efficiency, demonstrating its potential in peak demand management. This finding underscores the viability of equipping existing power plants with cold storage systems during warmer months, which could avert the need for constructing new facilities.

In terms of economic feasibility, the study demonstrates that using partial cold storage can lead to substantial cost reductions, achieving up to 97.36% savings compared to similar office systems. Furthermore, the analysis shows that the storage system in the power plant unit is more economical than in the administrative sector, with partial storage being the most cost-effective choice. The findings provide valuable insights for power plant designers, guiding the selection of optimal cooling methods and condensers based on consumption patterns and local climate conditions. Overall, this research contributes to improving energy management practices through the strategic implementation of cold storage solutions, promoting more sustainable energy consumption in both residential and commercial applications. Integrating renewable energy sources with chilled water or ice production systems can significantly enhance their efficiency and sustainability. This synergy not only offsets peak energy demand but also supports cooling systems like absorption cycles. The alignment of peak energy demand with maximum solar irradiation presents a unique opportunity to optimize energy systems by combining energy consumers, renewable sources, and storage systems. Future research on their combined operation can pave the way for innovative strategies in sustainable energy management and peak load reduction. While the current economic analysis highlights the superiority of implementing cold thermal storage at the power plant level, it is based solely on initial capital expenditure (CAPEX) and does not account for full life cycle cost (LCC). Therefore, for a more comprehensive evaluation and to support future development, it is recommended that further studies incorporate detailed LCC-based economic models.

Author Contributions

Ramin Mehdipour: Conceived the idea, designed the system, and contributed to the initial draft of the manuscript. Behnam Feizollah Beigi: Led the modeling and simulation aspects of the study and contributed significantly to manuscript writing. Romina Fathiraboki: Assisted in manuscript preparation, including literature review and structuring of the content. Hasan Asgari: Provided support in developing the modeling approach and refining the computational analysis. Zahra Baniamerian: Contributed to the final revision of the manuscript, ensured clarity and coherence, and assisted in the evaluation and interpretation of results. All authors have read and approved the final version of the manuscript.

Funding

The authors declare that no funding was received for conducting this research.

Data availability statement

The data that support the findings of this study are available from the corresponding author, R. Mehdipour, upon reasonable request.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

References

[1] S. Rahgozar, A. Pourrajabian, M. Dehghan, On the role of building use and operational strategy in integrating ice storage systems: an economic perspective, J. Energy Storage 98 (2024) 113025, https://doi.org/10.1016/j.est.2024.113025.

[2] B. Cardenas, S. Garvey, Z. Baniamerian, R. Mehdipour, Heat pumps and the requirement for grid-scale energy storage in the UK, IET Conf. Proc. 2024 (32) (2025) 14–19, https://doi.org/10.1049/icp.2024.4499.

[3] Z. Baniamerian, S. Garvey, J. Rouse, B. Cárdenas, D.L. Pottie, E.R. Barbour, A. Bagdanavicius, How pressure affects costs of power conversion machinery in compressed air energy storage; part II: heat exchangers, J. Energy Storage 86 (Part A) (2024) 111138.

[4] Z. Baniamerian, S. Garvey, J. Rouse, B. Cárdenas, D.L. Pottie, E.R. Barbour, A. Bagdanavicius, How pressure affects costs of power conversion machinery in compressed air energy storage; part I: compressors and expanders, J. Energy Storage 89 (2024) 111791.

[5] J. Hossain, H. Shareef, M.A. Hossain, R. Manojkumar, Q. Abdullah, A.H.H. Al-Masoodi, Optimizing battery energy storage and solar photovoltaic systems for commercial buildings in Malaysia: a case study, 2023 IEEE International Conference on Energy Technologies for Future Grids (ETFG), Wollongong, Australia, 2023, pp. 1–6, https://doi.org/10.1109/ETFG55873.2023.10408468.
[6] J. Hossain, N. Saeed, R. Manojkumar, M. Marzband, K. Sedraoui, Y. Al-Turki, Optimal peak-shaving for dynamic demand response in smart Malaysian commercial buildings utilizing an efficient PV-BES system, Sustain. Cities Soc. 101 (2023) 105107, https://doi.org/10.1016/j.scs.2023.105107.
[7] B. Arcuri, C. Spataru, M. Barrett, Evaluation of ice thermal energy storage (ITES) for commercial buildings in cities in Brazil, Sustain. Cities Soc. 29 (2016) 178–192, https://doi.org/10.1016/j.scs.2016.12.011.
[8] R. Mehdipour, S. Garvey, B. Cardenas, Z. Baniamerian, C.J. Wood, Ice-Source Heat Pump for residential heating: A case study on energy storage and pipeline repurposing in the UK, Case Stud. Therm. Eng 73 (2025) 106579, https://doi.org/10.1016/j.csite.2025.106579.
[9] M. Hosseinpour, H.A. Ozgoli, S.A. Haji Seyed Mirza Hosseini, A.H. Hemmasi, R. Mehdipour, Energy analysis of utilizing biomass gasification to partially substitute fossil fuels in an IBG-GT-ST-Kalina cycle (Materials and Energy Research Center (MERC), Iranian Association of Chemical Engineers (IAChE)), J. Renew. Energy Environ. 9 (3) (2022) 75–86, https://doi.org/10.30501/jree.2022.321835.1307.
[10] M. Hosseinpour, H.A. Ozgoli, S.A. Haji Seyed Mirza Hosseini, A.H. Hemmasi, R. Mehdipour, Evaluation of performance improvement of the combined biomass gasifier power cycle using low-temperature bottoming cycles: organic Rankine cycle, Kalina, and Goswami, Environ. Prog. Sustain. Energy 41 (5) (2022) 1–14, https://doi.org/10.1002/ep.13855.
[11] V. Gkoutzamanis, A. Chatziangelidou, T. Efstathiadis, A. Kalfas, A. Traverso, J. Chiu, Thermal energy storage for gas turbine power augmentation, J. Glob. Power Propuls. Soc. 3 (2019) 592–608, https://doi.org/10.33737/jgpps/110254.
[12] O. Zeitoun, Two-stage evaporative inlet air gas turbine cooling, Energies 14 (5) (2021) 1382, https://doi.org/10.3390/en14051382.
[13] M. Xie, L. Chen, K. Wu, Z. Liu, J. Lin, C. Jiang, S. Xie, Y. Zhao, A novel peak shaving approach to improving load flexibility of the Allam cycle by integrating cold energy storage, J. Clean. Prod. 386 (2022) 135769, https://doi.org/10.1016/j.jclepro.2022.135769.
[14] A. Shirazi, B. Najafi, M. Aminyavari, F. Rinaldi, R.A. Taylor, Thermal–economic–environmental analysis and multi-objective optimization of an ice thermal energy storage system for gas turbine cycle inlet air cooling, Energy 69 (2014) 212–226.
[15] S. Sanaye, A. Shirazi, Thermo-economic optimization of an ice thermal energy storage system for air-conditioning applications, Energy Build. 60 (2013) 100–109.
[16] S. Sanaye, A. Shirazi, Four E analysis and multi-objective optimization of an ice thermal energy storage for air-conditioning applications, Int. J. Refrig. 36 (3) (2013) 828–841.
[17] S. Sanaye, A. Fardad, M. Mostakhdemi, Thermoeconomic optimization of an ice thermal storage system for gas turbine inlet cooling, Energy 36 (2) (2011) 1057–1067.
[18] M. Ameri, S.H. Hejazi, The study of capacity enhancement of the Chabahar gas turbine installation using an absorption chiller, Appl. Therm. Eng. 24 (1) (2004) 59–68.
[19] Y.S.H. Najjar, A.M. Abubaker, A.F.S. El-Khalil, Novel inlet air cooling with gas turbine engines using cascaded waste-heat recovery for green sustainable energy, Energy 93 (2015) 770–785.
[20] S. Popli, P. Rodgers, V. Eveloy, Gas turbine efficiency enhancement using waste heat powered absorption chillers in the oil and gas industry, Appl. Therm. Eng. 50 (1) (2013) 918–931.
[21] A.A. El-Shazly, M. Elhelw, M.M. Sorour, W.M. El-Maghlany, Gas turbine performance enhancement via utilizing different integrated turbine inlet cooling techniques, Alex. Eng. J. 55 (3) (2016) 1903–1914.
[22] A.K. Mohapatra, Comparative analysis of inlet air cooling techniques integrated to cooled gas turbine plant, J. Energy Inst. 88 (3) (2015) 344–358.
[23] M. Renzi, F. Caresana, L. Pelagalli, G. Comodi, Enhancing micro gas turbine performance through fogging technique: experimental analysis, Appl. Energy 135 (2014) 165–173.
[24] M.A. Ehyaei, M. Tahani, P. Ahmadi, M. Esfandiari, Optimization of fog inlet air cooling system for combined cycle power plants using genetic algorithm, Appl. Therm. Eng. 76 (2015) 449–461.
[25] H. Athari, S. Soltani, A. Bölükbaşi, M.A. Rosen, T. Morosuk, Comparative exergoeconomic analyses of the integration of biomass gasification and a gas turbine power plant with and without fogging inlet cooling, Renew. Energy 76 (2015) 394–400.
[26] A. Beghi, L. Cecchinato, M. Rampazzo, F. Simmini, Energy efficient control of HVAC systems with ice cold thermal energy storage, J. Process Control 24 (6) (2014) 773–781.
[27] Y. Han, B. Shen, H. Hu, F. Fan, Optimizing the performance of ice-storage systems in electricity load management through a credit mechanism: an analytical work for Jiangsu, China, Energy Procedia 61 (2014) 2876–2879.
[28] C. Yan, W. Shi, X. Li, S. Wang, A seasonal cold storage system based on separate type heat pipe for sustainable building cooling, Renew. Energy 85 (2016) 880–889.
[29] F. Beigi, R. Mehdipour, Investigation of cold storage performance to improve management of power generation in thermal power plants in Iran, Energy 213 (2020) 118843.
[30] F.A. Ghaith, R.O. Dag, Performance and feasibility of utilizing solar powered ice storage system for space cooling applications, Energy Convers. Manag. X 16 (2022) 100319.
[31] R. Tevis, N. Schuster, F. Evans, R. Himmler, S.H. Gheewala, A multi-scenario life cycle impact comparison of operational energy supply techniques for an office building in Thailand, Energy Build. 190 (2019) 172–182.
[32] H. Tang, J. Yu, Y. Geng, X. Liu, B. Lin, Optimization of operational strategy for ice thermal energy storage in a district cooling system based on model predictive control, J. Energy Storage 62 (2023) 106872, https://doi.org/10.1016/j.est.2023.106872.
[33] S. Rahgozar, M. Dehghan, A. Pourrajabian, H. Haghgou, Economic feasibility of ice storage systems for office building applications: a climate sensitivity analysis, J. Energy Storage 45 (2022) 103712.
[34] D. Erdemir, N. Altuntop, Y.A. Çengel, Experimental investigation on the effect of ice storage system on electricity consumption cost for a hypermarket, Energy Build. 251 (2021) 111368.
[35] M. Griesbach, A. König-Haagen, F. Heberle, D. Brüggemann, Multi-criteria assessment and optimization of ice-energy storage systems in combined heat and cold supply networks of a campus building, Energy Convers. Manag. 287 (2023) 117118, https://doi.org/10.1016/j.enconman.2023.117118.
[36] R. Mehdipour, S. Garvey, Z. Baniamerian, B. Cardenas, Ice source heat pump system for energy supply via gas pipelines – part 1: performance analysis in residential units, Energy (2024) 132974, https://doi.org/10.1016/j.energy.2024.132974.
[37] R. Mehdipour, B. Zeinali, R. Fathiraboki, H.A. Ozgoli, Z. Baniamerian, S. Sadr, Investigating the performance of a seasonal cold storage system for dry and mountainous climates, Contributions of Science and Technology for Engineering 1 University of Mazandaran, 2024, pp. 51–59, https://doi.org/10.22080/cste.2024.27808.1003.
[38] Ministry of Energy of Iran. (2019). Electricity Tariff Schedule. Retrieved October 2019 from http://tariff.moe.gov.ir.
[39] U. Zafer, PCM thermal energy storage systems, ASHRAE 2004 Conference Paper, (2004).
[40] M.S. Najafi, O. Alizadeh, Climate zones in Iran, Meteorol. Appl. 30 (5) (2023), https://doi.org/10.1002/met.2147.
[41] Iran Meteorological Organization, Meteorological data for multiple cities including Arak, Isfahan, Yazd, and Bandar Abbas, Obtained through direct request to the Iran Meteorological Organization, 2020.
[42] C.E. Dorgan, J.S. Elleson, Design guide for cool thermal storage, ASHRAE, Atlanta, USA, 1993.
[43] P. Ihm, M. Krarti, G.P. Henze, Development of a thermal energy storage model for EnergyPlus, Energy Build. 36 (8) (2004) 807–814.
[44] S. Sanaye, M. Tahani, Analysis of gas turbine operating parameters with inlet fogging and wet compression processes, Appl. Therm. Eng. 30 (2-3) (2010) 234–244.
[45] Bhargava, R., & Meher-Homji, C.B. (2002). Parametric analysis of existing gas turbines with inlet evaporative and overspray fogging. In Turbo Expo: Power for Land, Sea, and Air (Vol. 36096, pp. 387-401).