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Co-Simulation of a load-shifting system with cold thermal storage in a campus environment

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Abstract

The increasing trend of climate change and its global effects has prompted a call for sustainable energy solutions and decarbonization efforts to protect our planet. Heat source systems, such as compression chillers, significantly contribute to the energy consumption of buildings. In this context, digital twin-based energy management has potential for enhancing efficiency through real-time monitoring and operational optimization. This paper presents part of an ongoing project at Waseda University "Development and Standardization of Fundamental Technologies for Thermal Energy Management Systems" (FUTTEMS), which aims to digitize heat management from various heat utilization systems to improve energy utilization, enhance system performance, and reduce carbon emissions. To accurately represent the physical system, reliable models of system components and appropriate simulation tools are required. In this study, we introduce a co-simulation framework for a cooling system that includes a turbo chiller, and a cold thermal energy storage unit. The chiller and thermal storage components are modeled in Dymola using the Modelica language, while the building cooling load is simulated using EnergyPlus. Data exchange and synchronization between the two platforms was facilitated by the Building Controls Virtual Test Bed (BCVTB). The integrated system functions as a load-shifting mechanism within the central air conditioning system in the Waseda campus. Additionally, this load-shifting model represents an initial step toward developing a comprehensive energy management system (EMS) for the entire central air conditioning plant using digital twin technology. It is expected to enable a realistic evaluation of system performance under dynamic building demand and subsequently open up to optimal operation as an integrated part of the building. The models for the chiller and storage for load-shifting have been developed and tested using performance data from the installed chiller plant.

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Keywords: Chiller; thermal storage; load-shifting; decabonization

1. Introduction

Central air conditioning systems are essential for maintaining comfortable temperatures in medium to large buildings, particularly during peak heating and cooling periods. The demand for these systems is increasing due to rising global temperatures and technological advancements. However, heating and cooling systems also account for the largest share of energy consumption in buildings [1,2]. The ongoing discussions about environmental issues and the global energy crisis highlight the need for practical and sustainable solutions to decarbonize the energy sector. In a campus environment, the operation of central air conditioning systems varies based on demand scenarios and occupancy patterns. Often, the heating or cooling needs of a building are limited to certain hours or fluctuate significantly throughout the day. Over the years, thermal energy storage has been implemented in heat pump systems to enhance operational flexibility and optimize energy use, leading to greater energy savings and improved system reliability. The chiller generates a cooling effect, particularly during off-peak hours, and stores cold energy in the form of chilled water or ice. This stored cool energy can be used to meet the building's cooling needs during peak operating hours, enhancing overall energy efficiency. To achieve optimal performance in an integrated central cooling system that includes various components—such as heat pumps, thermal storage, the building itself, and other auxiliary units—effective control strategies are essential for ensuring efficient and sustainable operation.

Recent advancements in digital transformation, particularly the internet of things (IoT), have created new opportunities for effective energy management through the concept of digital twins in various applications, including heat pump systems and their interactions with buildings [3]. Several case studies [4–11] have successfully implemented the concept of digital twin for energy management systems in heating and cooling systems, highlighting its significant potential. Modeling and simulation remain an important aspect in developing digital twin-based energy management systems. In particular, modeling and simulation of HVAC+R thermal systems, building components, and their interconnections are important requirements for the development of digital twin-based energy management systems [12]. Because of the complex integration of the heat source systems and their interaction with buildings as well as the environment, which are often modeled using different software tools, it is important to explore the power of co-simulation by integrating the models of individual components into an appropriate environment that can capture the energy performance of an entire building envelope [13].

This paper presents part of an ongoing effort at Waseda university of developing and implementing thermal energy management systems (EMS) using digital twin concepts to clarify the energy system technologies that will serve as the foundation for the realization of a carbon neutral campus by reducing CO_2 emissions. An initial case study that involves modeling and co-simulation of a load-shifting system is briefly presented in this paper. The system under consideration is discussed in section 2.

2. System Description

The operation of central air conditioning systems in campus environment varies in time based on demand scenarios or occupancy periods. Often, the heating or cooling demand in the campus building is limited to only several hours or significantly varies over the 24 hours. This variation in cooling or heating demands as well as occupancy periods in buildings makes it important requirement for proper heat source systems configuration and operation schedules in a controlled manner. This enables effective EMS on campus, expected to contribute to reducing carbon emissions and enhancing energy efficiency.

The system in this study is a central air conditioning system installed in an existing building at Waseda University. It is equipped with dual heat source systems consisting of a compression (turbo) chiller, an absorption chiller, thermal energy storage (TES) tank, and several auxiliary equipment [14]. The integration of this kind is intended to provide flexible operating options with different modes. As such, the current system configuration operates in three basic modes; (1) charging mode – where the turbo chiller operates at night (off-peak period) to charge the TES tank and store chilled water, (2) the discharging mode - where the stored chilled water in the TES is discharged to provide cooling to the building during peak cooling demand (usually day time), and (3) absorption chiller mode – here the absorption chiller driven primarily by heat is used to cover part of the cooling need of the building. Certainly, managing the energy flow in such a complex system to ensure efficient energy use requires high degree of technical and engineering commitment. Hence, the main goal is to develop an effective energy management system based on digital twin concept that will enable real-time operation monitoring and control optimization for better energy utilization in the future.

In this paper, we present only a sub-part of the whole system described above that represents a load shifting unit in the integrated system - that is the charging and discharging operation modes, showcasing the turbo chiller, TES tank and the load (building), as shown in Fig. *1*. The turbo chiller has a rated capacity of 200 TR (703.3 kW) and is operated at night, the time when the ambient is cooler and energy consumption is less (off-peak period). In this operation, the chiller produces chilled water that is stored in the TES tank. The chiller is a water-cooled type where heat is rejected through the cooling water loop, as shown in Fig. *1*. The tank has a total volume capacity of 1075 m^3 water. The stored chilled water is discharged during the daytime (peak period) through a heat exchanger (Load HX) to the building, (at a temperature T_{supply}), thereby shifting the thermal load to meet the cooling needs in the building. It should be noted that the load-shifting system presented in this paper does not cover all the cooling needs of the building but rather is contributing to the central cooling system. A co-simulation platform is developed where the cooling system is modeled in Dymola and the building load side in EnergPlus.

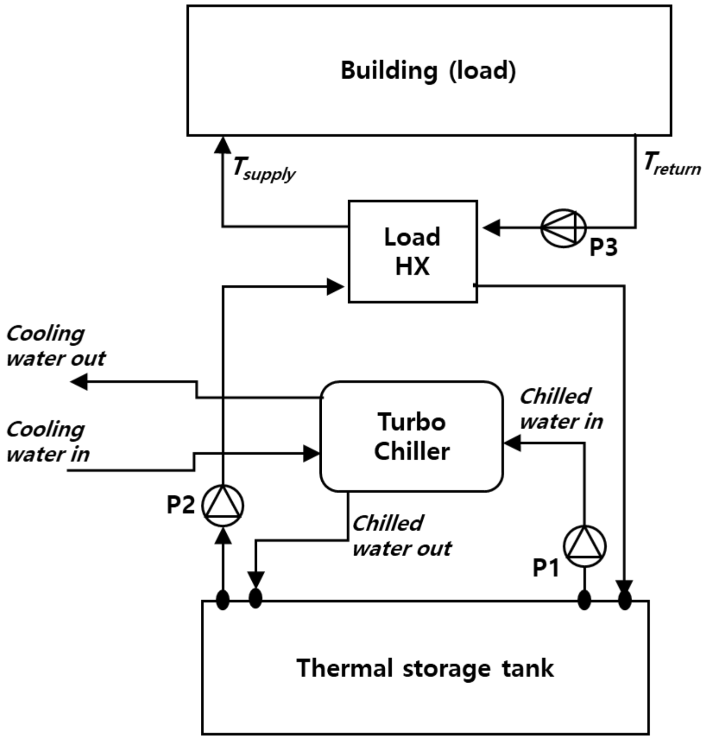


Fig. 1 Schematic of a load-shifting cooling system.

3. Methodology

This section presents the modeling and simulation methodology adopted in this paper. The mathematical models of the main components of the system – thermal storage tank and turbo chiller are presented. The models of these components are integrated in a modular form and simulated in Dymola – a simulation environment based on Modelica language. While the building is characterized in EnegyPlus and then coupled with the Dymola model for co-simulation.

3.1. Model of thermal storage tank

The storage tank is rectangular in shape located under the ground with four ports used for charging and discharging processes, as shown in Fig. *2*(a). The following assumptions are considered in the model development: (i) uniform fluid temperature in each elemental volume, (ii) 1-D model of the storage tank with constant heat loss coefficient (ground model not considered), (iii) equal mass flows in and out during each charging and discharging process. (iv) the tank is fully buried in the shallow ground, therefore heat loss is only to the internal ground environment. A 1-dimensional energy balance approach that has been widely used in dynamic system simulation tools is used to model the storage tank [15,16]. Accordingly, the storage volume is divided into several equal control volumes (fluid layers) along the horizontal direction, forming energy balances across the entire volume.

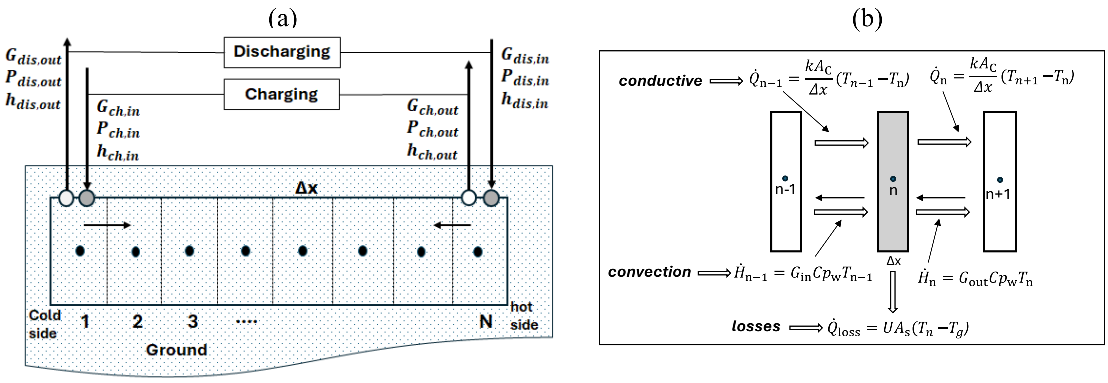


Fig. 2 (a) Discretization of storage tank volume, (b) energy balance for a volume element.

The water temperature in the tank at the various layers is determined from the overall energy equation of the entire storage volume, applied to each layer, given as:

$$\rho_n V_n Cp_n \frac{dT_n}{dt} = \sum_n \dot{H} + \sum_n \dot{Q} \tag{1}$$

n = 1, 2, ……...N

where ρ, V, T are the density, volume and temperature of water at each elemental volume, respectively. Subsequently, for an elemental volume (n) shown in Fig. 2(b), the right-hand side of Eq. (1) is represented in terms of temperature, which is composed of convective terms due to fluid flow (enthalpy flow $\dot{H}$), conductive terms due to heat transfer between fluid layers ($\dot{Q}$) and heat loss terms ($\dot{Q}_{loss}$) due to heat transfer from the tank to the ground in a discretised form into N-layers of fixed thickness (Δx). Hence, from the energy terms in Fig. 2(b), the overall energy balance, Eq. (1) can be written in terms of temperature for each elemental volume as:

$$\rho_n \Delta x\, A_c Cp_{w,n} \frac{T_{[n]}}{dt} = \Delta\dot{H}_n + \frac{kA_C}{\Delta x}(T_{[n+1]} - T_{[n]}) - \frac{kA_C}{\Delta x}(T_{[n]} - T_{[n-1]}) - UA_{s,n}(T_{[n]} - T_g) \tag{2}$$

where A_c is the cross-sectional area of the tank, Cp_w is the specific heat capacity of water, G_{in} is the mass flow rate of water entering the tank, k is the thermal conductivity of water, U is the overall heat lost coefficient, A_s is the surface area of the tank with respect to fluid layer, and T_g is the ground temperature.

The direction of enthalpy flow terms in Fig. 2(b) and subsequently in Eq. (2) depends on whether charging or discharging process. Therefore, the enthalpy flows in and out of the tank (convection) is given by the enthalpy balance or change ($\Delta\dot{H}_n$) at the first, internal, and last layers of the control volumes, as given in Eq. (3) below.

$$\Delta\dot{H}_n = \begin{cases} G_{ch,in}Cp_{w[n]}\left(T_{ch,in} - T_{[n]}\right) & if\ n = 1\ (charing -\ first\ layer) \\ G_{ch,in}Cp_{w[n]}\left(T_{[n-1]} - T_{[n]}\right) & if\ n\ > 1\ (charging - internal\ layers) \\ G_{dis,in}Cp_{w[n]}\left(T_{dis,in} - T_{[n]}\right) & if\ n = N\ (discharging -\ last\ layer) \\ G_{dis,in}Cp_{w[n]}\left(T_{[n+1]} - T_{[n]}\right) & if\ n < N\ (discharging - internal\ layers) \end{cases} \tag{3}$$

3.2. Turbo chiller model

The turbo chiller model was developed based on performance data from the existing system shown in Fig. 3. It utilizes a coefficient performance – (COP) map with load-factor - K and cooling water inlet temperature -

T_{cw} (°C) as independent variables. From the operational profile of the chiller shown in Fig. 3, the COP increases as the inlet cooling water temperature decreases, indicating a typical trend of chiller performance.

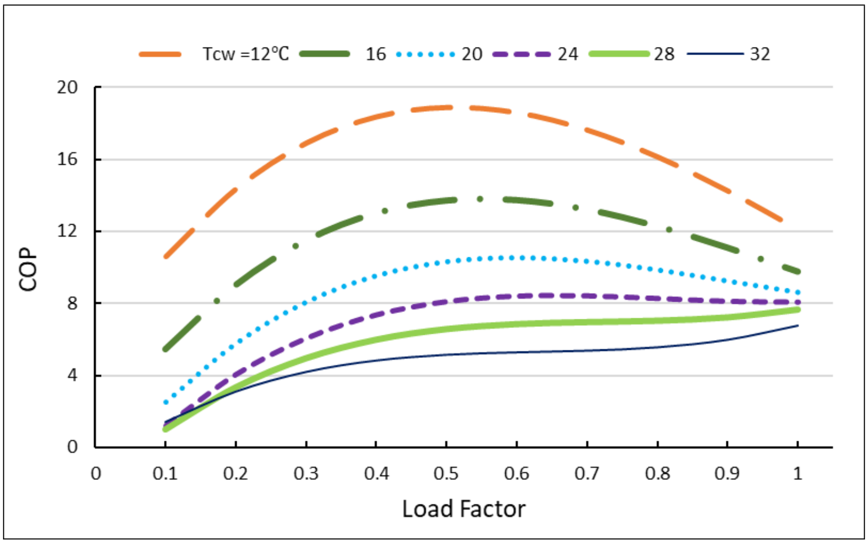


Fig. 3 Performance curves of the turbo chiller: COP, load factor K and cooling water inlet temperature (T_{cw}).

The COP map was created through cubic approximation using the actual operation data presented in Fig. 3. The approximate COP correlation derived from the map approximation is presented in Eq. (4). The letters a – j represents the correlation coefficients obtained from the fitting of the field data.

$$COP = a + bK + cT_{cw} + dK^2 + eKT_{cw} + fT_{cw}{}^2 + gK^3 + hK^2T_{cw} + iKT_{cw}{}^2 + jT_{cw}{}^3 \quad (4)$$

where the load factor K represents the ratio of the actual load Q_c to the rated cooling capacity Q_{rated} of the chiller. The values of the correlation coefficients a – j are given in the Table below:

Table 1 Coefficients values of COP map correlation in Eq. (4).

a	*b*	*c*	*d*	*e*	*f*	*g*	*h*	*i*	*j*
142.74	72.97	-16.71	-44.75	-3.11	0.65	24.12	-0.18	0.062	-0.0085

The cooling rate Q_c and heat rejection rate Q_{cw} are calculated from the energy balance across the evaporator and condenser sides, respectively as:

$$G_{c,out}h_{c,out} = G_{c,in}h_{c,in} - Q_c \quad (5)$$

$$G_{cw,out}h_{cw,out} = G_{cw,in}h_{cw,in} + Q_{cw} \quad (6)$$

where h is the specific enthalpy of the cooling fluid, water.

The outlet chilled water temperature is considered part of the inputs to the model and the required power input to the chiller is calculated from the COP relation as follows:

$$COP = Q_c / W_{com} \quad (7)$$

where W_{com} represents compressor power consumption by the turbo chiller. The model calculates the COP, heat rates, and required compressor power input for the turbo chiller.

3.3. Other components models

As shown in Fig. *1* pumps are used to circulate cold water between the chiller, TES tank, heat exchanger and the load, P1, P2, P3, respectively. To capture the unique characteristics of each water pump, characteristic curves provided by the manufacturers of the installed pumps are utilized to develop the model. These characteristic curves include total head (H), pump efficiency, and motor efficiency. The pump model takes the volumetric flow rate of water as input and computes the electric power consumption. In each hydraulic unit, the pressure change within the piping system is determined using the Darcy-Weisbach equation, which calculates friction losses considering a viscous, incompressible flow. The load HX connects the TES tank to the load and is responsible for transferring cold energy from the tank to the load during discharging operation. The load HX model is developed using the classical approach of logarithmic mean-temperature difference.

The rate of heat transfer in the load HX between primary water and secondary water streams is given by:

$$Q_{HX} = UA_{HX}\Delta T_{lm} \tag{8}$$

where

$$\Delta T_{lm} = \frac{\left(T_{sec,in} - T_{prim,out}\right) - \left(T_{sec,out} - T_{prim,in}\right)}{ln\left(\frac{T_{sec,in} - T_{prim,out}}{T_{sec,out} - T_{prim,in}}\right)} \tag{9}$$

and $T_{sec,in}$ and $T_{sec,out}$ are the inlet and outlet temperature of water, respectively, at the secondary side of the load HX connecting the building, while $T_{prim,in}$ and $T_{prim,out}$ are the inlet and outlet temperature of cold water, respectively, at the primary side of the HX connecting the TES tank.

3.4. Co-simulation and Building characteristics

The models of the thermal components introduced above are integrated into a single model that represents the cooling system (chiller, TES tank, and load HX) in Dymola, one of the popular simulation environments based on Modelica language. A co-simulation platform is developed where the cooling system is modeled in Dymola and the building load side is modeled in EnergyPlus [17]. In this configuration, the Dymola model takes as input the flow rates of chilled water and cold water through pumps P1 and P2, respectively, flow rate and inlet temperature of cooling water, and outlet temperature of chilled water from the Excel data (system operation data), Fig. 4. The Dymola model exposes the return-water temperature *(T_{return})* and water flow rate (*$P3_{flow}$*) from the building as external inputs while exporting two key outputs: (1) the cooling rate (*Q_{supply}*) extracted from the TES through the load HX during discharging, and (2) the cold-water supply temperature (T_{supply}) to be delivered to the building model in EnergyPlus. Data exchange and synchronization between the two platforms are facilitated by the Building Controls Virtual Test Bed (BCVTB) [18], as shown in Fig. 4. Based on the cold-water supply and return temperatures as well as the flow rate of P3, EnergyPlus computes the resulting indoor temperature and building cooling demand.

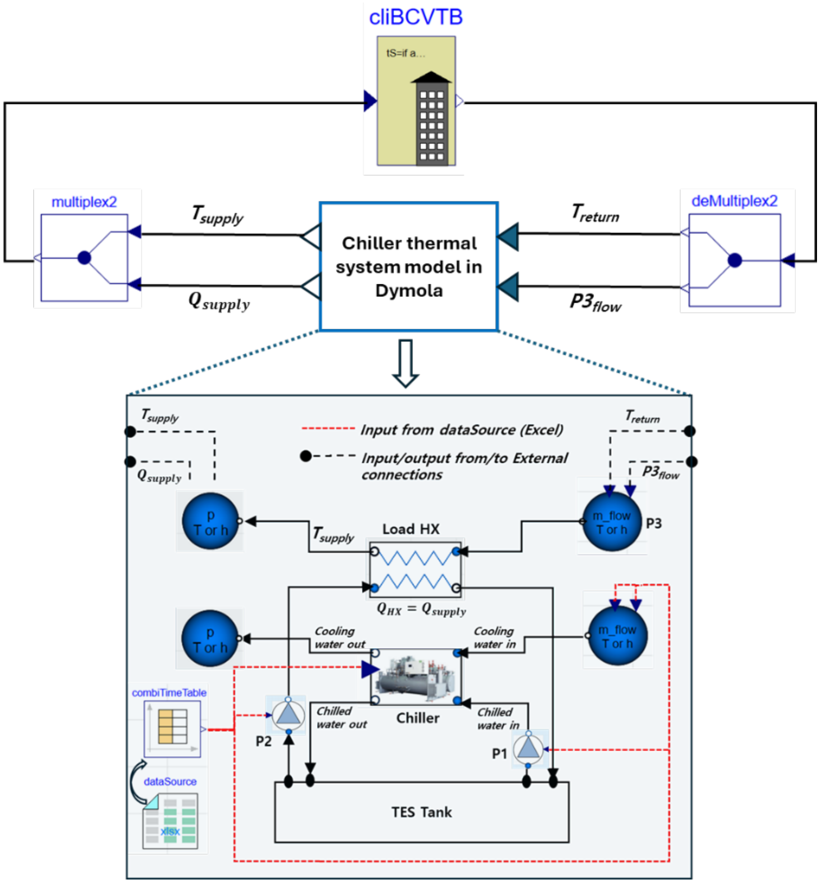


Fig. 4 Dymola-BCVTB co-simulation interface in Dymola GUI.

The building used in the co-simulation is an idealized structure with a floor area of 80 × 80 m^2 and a height of 5 m, designed to produce a cooling demand in the range of 600-900 kW. The building thermal load is simulated using EnergyPlus with a simplified geometry and envelope to allow a clear evaluation of the cooling system performance.

An HVAC system is explicitly included in the EnergyPlus model and is configured as a water-cooled chiller system with a maximum cooling capacity of around 929 kW and a rated coefficient of performance (COP) of 3. In the co-simulation, the cold-water supply temperature from the Dymola model (T_{supply}) stays between 10 - 11 °C, while the return temperature from the building (T_{return}) is maintained at approximately 15 °C during the operation.

By controlling the cold-water mass rate (P3), the cooling capacity available to the building model in EnergyPlus is constrained by the cooling capacity calculated by the cooling system model in Dymola, such that the building cooling demand is met only up to the capacity supplied by the Dymola model. This configuration ensures stable data exchange between EnergyPlus and Dymola. EnergyPlus then computes the resulting building cooling demand and indoor temperature response. The energy equation used to calculate the indoor/room temperature in EnergyPlus is given by:

$$C_z \frac{T_z}{dt} = Q_{load} + Q_{sys} \tag{10}$$

where T_z is the room/indoor temperature, C_z is the thermal capacitance of the room air, Q_{load} is the cooling load of the building, and Q_{sys} is the cooling rate provided by a cooling system, typically a chiller system in EnergyPlus. Note that Q_{load} is a function of several parameters including internal heat gain, surface areas of walls, windows, and ceiling, inner wall temperature, air flow rate, and outdoor air temperature, e.t.c. It should be noted that the Q_{sys} in this co-simulation is supplied by the Dymola model of the chiller-storage system, and a corresponding cooling rate is computed in EnergyPlus chiller system. The next section presents some initial results.

4. Results and discussion

This section presents the initial simulation results of the cooling system model, followed by a co-simulation test linking Dymola and EnergyPlus.

4.1. Cooling system

The cooling system was initially simulated in Dymola to verify its functionality before proceeding with the co-simulation. To simulate the cooling system, several inputs are required as outlined in Section 3. These include chilled water outlet temperature and flow rate from the chiller side, cooling water inlet temperature and flow rate, and inlet secondary water temperature and flow rate of the load HX. The input data is obtained from the field operation of the actual system. Note that the inlet secondary water temperature of the load HX is fixed constant (T_{return}) and the flow rate is adjusted during the co-simulation, as discussed in Section 3. A representation of the input data consists of cooling water inlet temperature of the chiller (*Fig. 3*), and water flow rates of pumps P1, P2, and P3, shown in Fig. 5. Other input to the system model include chilled water outlet temperature, which is around 6 °C from the field data and parameters related to the TES tank and load HX listed in *Table 2*.

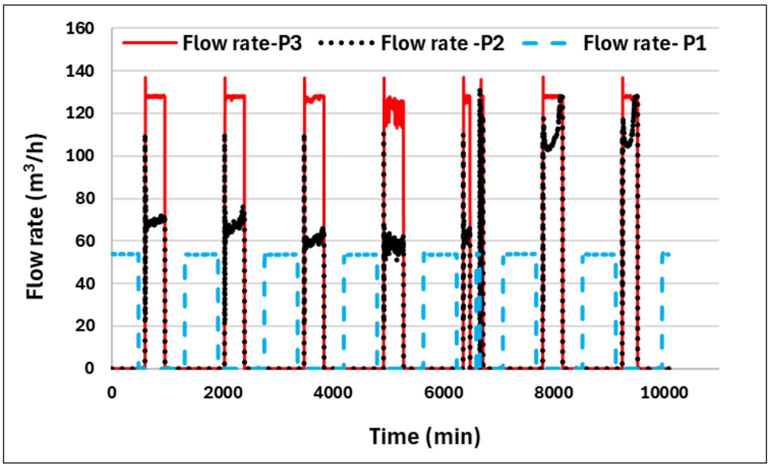


Fig. 5 Flow rate input data.

Table 2 Input parameters for thermal storage tank and load HX

Parameter	Unit	Value
Water volume in tank, V	m^3	1075
Tank length, L	m	24
Tank width, W	m	16
Heat loss coefficient, U	W/m^2K	1.5
Initial water temperature in tank	^oC	~6 - 12 °C at different layers
Ground temperature, T_g	^oC	20
Number of volume elements, N	-	8
Area-overall heat coefficient of load HX, UA_{HX}	W/K	529909

The cooling rates obtained from the chiller during energy charging and discharging operations are illustrated in Fig. 6. The charging operation takes place at night, while the discharging operation lasts for about 4 hours during the day, representing the load-shifting process. The comparison of the cooling output between simulation results and operational data for both charging and discharging shows good alignment, with minimal overestimation and underestimation in the model results for both scenarios. During the charging process, the chiller's performance is represented by the Coefficient of Performance (COP), as depicted in Fig. *7*(a), while the power consumption of the chiller is shown Fig. *7*(b). Both the simulated chiller COP and power consumption align satisfactorily with the field data. Minor discrepancies at certain load conditions are likely due to simplified assumptions in the models and measurement fluctuations in the input data.

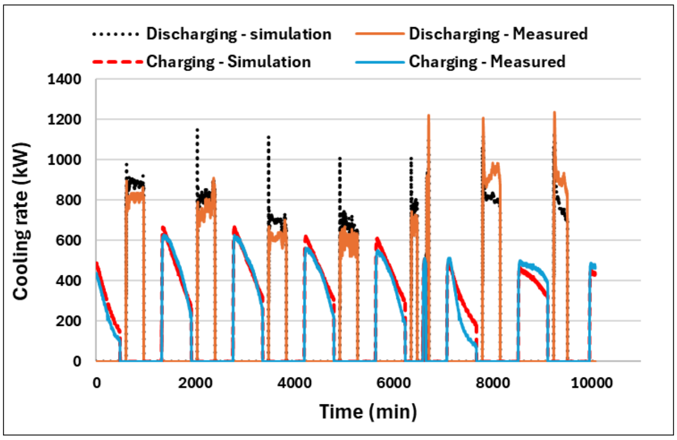
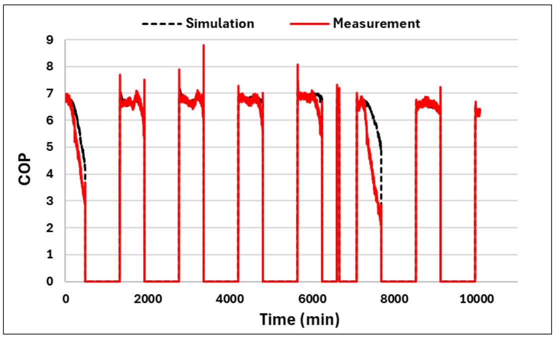


Fig. 6 Cooling rates during charging and load shifting (discharging) operations.

(a) (b)



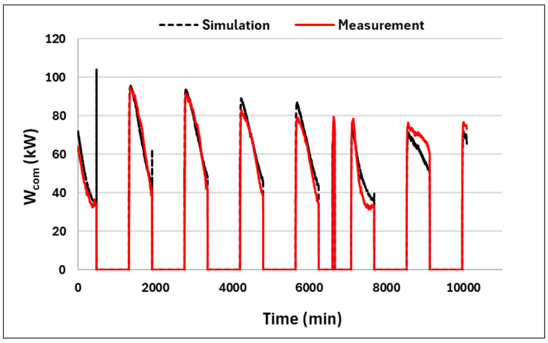


Fig. 7 Chiller COP-(a) and power consumption-(b).

4.2. Co-simulation test

The next step involved establishing a co-simulation link between the Dymola cooling system model and the EnergyPlus building model. The primary objective of this test was to validate data exchange and synchronization between the platforms using the BCVTB interface. This will enable complete performance analysis and control strategies of the system in the subsequent work ahead.

Fig. *8*(a) shows the cooling-rate balance obtained during the co-simulation at fixed return temperature of 15°C and supply temperature of 10 - 11°C, respectively. To match the cooling rate delivered by the Dymola subsystem with the cooling demand computed by the EnergyPlus model, the cold-water flow rate was adjusted and the resulting flow-rate profile and the corresponding room temperature are shown in Fig. 8(b). The room temperature exhibits natural variation during the cooling period, reflecting the simplified building envelope and the absence of active control strategies at this stage.

(a) (b)

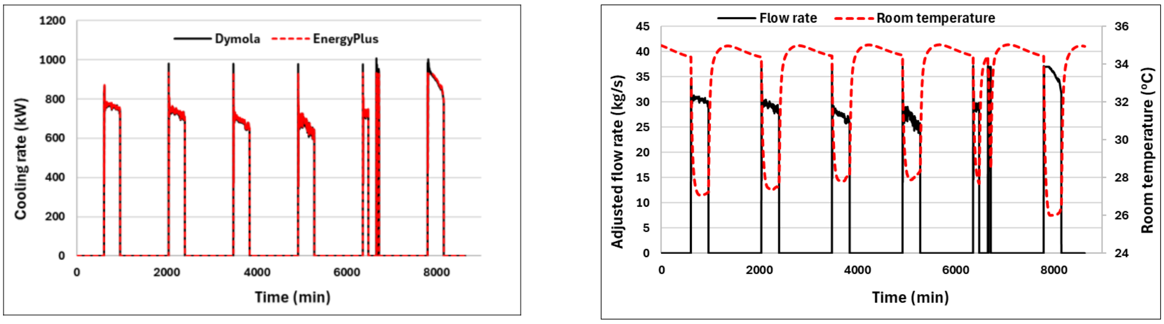


Fig. 8 (a) Cooling capacity from Dymola and EnergyPlus, (b) Cold water mass flow rate *(P3)* in EnergyPlus and resulting room temperature.

5. Conclusion

This work presented an initial step toward developing a digital twin–based energy management system for a central cooling system in Waseda campus. A modular Modelica/Dymola model of the turbo chiller and thermal storage tank was created, and a co-simulation link with an EnergyPlus building model was successfully established using BCVTB interface. The subsystem model showed reasonable agreement with system operational data. The present study focuses only on charging and discharging components, which represent a unit of a larger central air conditioning plant, but the co-simulation framework provides a solid foundation for expanding to the entire plant. Future work will focus on the following: (i) refine the subsystem models as more field operation data from the actual equipment becomes available, (ii) extend the modeling and co-simulation to include all major equipment, and (iii) implement control optimization toward a full digital twin–enabled energy management system for a carbon-neutral campus. These developments are expected to support the university's long-term goal of achieving a carbon-neutral campus through intelligent scheduling and optimized thermal energy management.

Nomenclature

Symbol	
A	Area, m^2
A_C	Cross-sectional area of storage tank, m^2
COP	Coefficient of performance
Cp_w	Specific heat capacity of water, J kg^{-1} K^{-1}
G	Mass flow rate, kg s^{-1}
$\dot{H}$	Enthalpy flow, W
h	Specific enthalpy, J kg^{-1}
k	Thermal conductivity of water, W m^{-1} K^{-1}
K	Load factor
P	Pump
Q	Heat rate, W
T	Temperature, K
U	Overall heat loss coefficient, W m^{-2} K^{-1}
$Wcom$	Compressor power, W
Δx	Horizontal length of fluid layer in storage tank, m
Subscripts	
c	cold side
g	ground
h	hot side
in	inlet
out	outlet
s	surface

Greek symbols
ρ Density, kg m^{-3}

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