IBAL Test Settings

This document summarizes the detailed settings for each test scenario for the IBAL testing. The settings of the **Default** cases are shown in detail. For variation cases, settings that are different from the **Default** cases are highlighted.

1 Test Settings Overview

1.1 Default

1.1.1 Test Day Selection

The selected test days below represent a **typical summer** day in each location, respectively.

Location	**Date (TMY3)**	**Peak (standard time)**	**Non-peak (standard time)**
Atlanta	8/26	13:00-18:00	0:00-13:00, 18:00-24:00
Buffalo	7/2	10:00-16:00	0:00-10:00, 16:00-24:00
New York	6/26	11:00-19:00	0:00-11:00, 19:00-24:00
Tucson	8/28	13:00-19:00	0:00-13:00, 19:00-24:00

1.1.2 Simulation Settings

Item	**Detail**
EnergyPlus model standard	ASHRAE 90.1-2004
Occupancy	Conference_Room_Mid_2: 23 Enclosed_Office_Mid_2: 2 Enclosed_Office_Mid_5: 2 Open_Office_Mid_1: 10
Occupant behaviors	20 (54 %) occupants care energy saving with 80 % restriction probability.
Other modifications	Temperature Capacity Multiplier of the selected zones in the EnergyPlus model are set to 8 to better represent the damping effect of internal thermal mass in typical buildings.

1.1.3 Control Strategy

1.1.3.1 Setpoint Schedule

Item	**Default setpoint**	**Reset strategy**
Zone Temperature	Occupied (6am – 8pm): 68 °F – 78 °F; Unoccupied/setback (12am – 6am, 8pm – 12am): 55 °F – 90 °F	**Eff**: No reset **Shed**: 70 °F – 80 °F during peak **Shift**: 65 °F – 75 °F 3 hours prior to peak, 70 °F – 80 °F during peak **Mod**: No reset
AHU Supply Air Temperature	55 °F (12.8 °C)	No reset

AHU Supply Air Static Pressure	398.5 Pa (1.6 inH_2O)	**Eff, Shed, Shift**: Trim & Respond (see 2.2). **Mod**: Trim & Respond (see 2.2) with Fan Power Modulation (see 2.6).
Chilled Water Supply Temperature	40 °F (4.4 °C)	Outdoor air temperature-based reset (see 2.1).
Chilled Water Secondary Loop Differential Pressure	551.6 kPa (80 psi)	No reset
VAV Minimum Air Flow*	Conference_Room_Mid_2: 460 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 200 CFM	No reset
Outdoor Air Flow Rate†	AHU1: 500 CFM AHU2: 240 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62-1989. The calculated minimum ventilation rates for the four zones are 460 CFM, 40 CFM, 40 CFM, and 200 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.

† The outdoor air flow rate is the sum of the minimum ventilation requirements of the downstream zones.

1.1.3.2 Equipment Availability Schedule

Equipment	Availability
Chiller 1	ON if any AHU is ON as indicated by the cooling coil valve being open for at least 90 seconds. Will switch to Chiller 2: - If average power exceeds 8500 W and the chilled water temperature is 0.3 °F (0.17 °C) above the setpoint for 100 seconds. - Or if average power exceeds 8000 W and the chilled water temperature is 1 °F (0.56 °C) above the setpoint for 100 seconds. - Or if average power exceeds 7500 W for 16.7 minutes.
Chiller 2	ON if a Chiller 2 switch ON rule as shown above is met. Will switch to Chiller 1: - If average power is below 3000 W for 100 seconds.
Ice Tank	OFF.
AHU 1	Occupied period: always ON. Setback period: ON if there is a zone demand (i.e., zone temperature setpoint is violated).
AHU 2	Occupied period: always ON. Setback period: ON if there is a zone demand (i.e., zone temperature setpoint is violated).

1.2 Variation: Extreme Summer (ExtrmSum)

1.2.1 Test Day Selection

The selected test days below represent an **extreme summer** day in each location, respectively.

Location	Date (TMY3)	Peak (standard time)	Non-peak (standard time)
Atlanta	7/8	13:00-18:00	0:00-13:00, 18:00-24:00
Buffalo	7/16	10:00-16:00	0:00-10:00, 16:00-24:00

New York	7/10	11:00-19:00	0:00-11:00, 19:00-24:00
Tucson	8/16	13:00-19:00	0:00-13:00, 19:00-24:00

1.2.2 Simulation Settings

See 1.1.2.

1.2.3 Control Strategy

See 1.1.3.

1.3 Variation: Typical Shoulder (TypShldr)

1.3.1 Test Day Selection

The selected test days below represent a **typical shoulder** day in each location, respectively.

Location	Date (TMY3)	Peak (standard time)	Non-peak (standard time)
Atlanta	4/29	NA	0:00-24:00
Buffalo	3/12	NA	0:00-24:00
New York	10/16	11:00-19:00	0:00-11:00, 19:00-24:00
Tucson	10/7	5:00-9:00, 16:00-20:00	0:00-5:00, 9:00-16:00, 20:00-24:00

Due to the absence of peak hours, shedding and shifting cases are not tested under the typical shoulder conditions in Atlanta and Buffalo.

1.3.2 Simulation Settings

See 1.1.2.

1.3.3 Control Strategy

See 1.1.3.

1.4 Variation: Model Predicted Control (MPC)

1.4.1 Test Day Selection

See 1.1.1.

1.4.2 Simulation Settings

See 1.1.2.

1.4.3 Control Strategy

1.4.3.1 Setpoint Schedule

Item	Default setpoint	Reset strategy
Zone Temperature	Based on MPC decisions (see 2.7)	NA
AHU Supply Air Temperature	Based on MPC decisions (see 2.7)	NA
AHU Supply Air Flow Rate	Based on MPC decisions (see 2.7)	NA

Chilled Water Supply Temperature	Based on MPC decisions (see 2.7)	NA
Chilled Water Secondary Loop Flow Rate	Based on MPC decisions (see 2.7)	NA
VAV Minimum Air Flow*	Conference_Room_Mid_2: 460 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 200 CFM	No reset
Outdoor Air Flow Rate†	AHU1: 500 CFM AHU2: 240 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62-1989. The calculated minimum ventilation rates for the four zones are 460 CFM, 40 CFM, 40 CFM, and 200 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.
† The outdoor air flow rate is the sum of the minimum ventilation requirements of the downstream zones.

1.4.3.2 Equipment Availability Schedule

Equipment	Availability
Chiller 1	Based on MPC decisions (see 2.7)
Chiller 2	OFF.
Ice Tank	OFF.
AHU 1	Occupied period: always ON. Setback period: Based on MPC decisions (see 2.7)
AHU 2	Occupied period: always ON. Setback period: Based on MPC decisions (see 2.7)

1.5 Variation: High Performance Building (STD2019)

1.5.1 Test Day Selection

See 1.1.1.

1.5.2 Simulation Settings

Item	Detail
EnergyPlus model standard	ASHRAE 90.1-2019
Occupancy	Conference_Room_Mid_2: 23 Enclosed_Office_Mid_2: 2 Enclosed_Office_Mid_5: 2 Open_Office_Mid_1: 10
Occupant behaviors	20 (54%) occupants care energy saving with 80% restriction probability
Other modifications	Temperature Capacity Multiplier of the selected zones in the EnergyPlus model are set to 8 to better represent the damping effect of internal thermal mass in typical buildings.

1.5.3 Control Strategy

1.5.3.1 Setpoint Schedule

Item	Default setpoint	Reset strategy

Zone Temperature	Occupied (6am – 8pm): 68 °F – 78 °F; Unoccupied/setback (12am – 6am, 8pm – 12am): 55 °F – 90 °F	**Eff**: No reset **Shed**: 70 °F – 80 °F during peak **Shift**: 65 °F – 75 °F 3 hours prior to peak, 70 °F – 80 °F during peak **Mod**: No reset
AHU Supply Air Temperature	55 °F (12.8 °C)	Trim & Respond (see 2.3).
AHU Supply Air Static Pressure	398.5 Pa (1.6 inH_2O)	**Eff, Shed, Shift**: Trim & Respond (see 2.2). **Mod**: Trim & Respond (see 2.2) with Fan Power Modulation (see 2.6).
Chilled Water Supply Temperature	40 °F (4.4 °C)	Trim & Respond (see 2.4).
Chilled Water Secondary Loop Differential Pressure	551.6 kPa (80 psi)	Trim & Respond (see 2.4).
VAV Minimum Air Flow*	Conference_Room_Mid_2: 215 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 215 CFM	Demand-based reset (see 2.5).
Outdoor Air Flow Rate†	AHU1: 224 CFM AHU2: 231 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62.1-2019. The calculated minimum ventilation rates for the four zones are 215 CFM, 45 CFM, 45 CFM, and 215 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.
† The outdoor air flow rate is determined in accordance with ASHRAE 62.1-2019.

1.5.3.2 Equipment Availability Schedule

See 1.1.3.2.

1.6 Variation: Dense Occupancy (DenseOcc)

1.6.1 Test Day Selection

See 1.1.1.

1.6.2 Simulation Settings

Item	**Detail**
EnergyPlus model standard	ASHRAE 90.1-2004
Occupancy	Conference_Room_Mid_2: 35 Enclosed_Office_Mid_2: 3 Enclosed_Office_Mid_5: 3 Open_Office_Mid_1: 15
Occupant behaviors	30 (54%) occupants care energy saving with 80% restriction probability
Other modifications	Temperature Capacity Multiplier of the selected zones in the EnergyPlus model are set to 8 to better represent the damping effect of internal thermal mass in typical buildings.

1.6.3 Control Strategy

1.6.3.1 Setpoint Schedule

Item	Default setpoint	Reset strategy
Zone Temperature	Occupied (6am – 8pm): 68 °F – 78 °F; Unoccupied/setback (12am – 6am, 8pm – 12am): 55 °F – 90 °F	**Eff**: No reset **Shed**: 70 °F – 80 °F during peak **Shift**: 65 °F – 75 °F 3 hours prior to peak, 70 °F – 80 °F during peak **Mod**: No reset
AHU Supply Air Temperature	55 °F (12.8 °C)	No reset
AHU Supply Air Static Pressure	398.5 Pa (1.6 inH_2O)	**Eff, Shed, Shift**: Trim & Respond (see 2.2). **Mod**: Trim & Respond (see 2.2) with Fan Power Modulation (see 2.6).
Chilled Water Supply Temperature	40 °F (4.4 °C)	Outdoor air temperature-based reset (see 2.1).
Chilled Water Secondary Loop Differential Pressure	551.6 kPa (80 psi)	No reset
VAV Minimum Air Flow*	Conference_Room_Mid_2: 700 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 300 CFM	No reset
Outdoor Air Flow Rate†	AHU1: 760 CFM AHU2: 360 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62-1989. The calculated minimum ventilation rates for the four zones are 700 CFM, 40 CFM, 40 CFM, and 300 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.
† The outdoor air flow rate is the sum of the minimum ventilation requirements of the downstream zones.

1.6.3.2 Equipment Availability Schedule

See 1.1.3.2.

1.7 Variation: Energy Saving Behaviors (EnergySave)

1.7.1 Test Day Selection

See 1.1.1.

1.7.2 Simulation Settings

Item	Detail
EnergyPlus model standard	ASHRAE 90.1-2004
Occupancy	Conference_Room_Mid_2: 23 Enclosed_Office_Mid_2: 2

	Enclosed_Office_Mid_5: 2 Open_Office_Mid_1: 10
Occupant behaviors	30 (81 %) occupants care energy saving with 90 % restriction probability
Other modifications	Temperature Capacity Multiplier of the selected zones in the EnergyPlus model is set to 8 to better represent the damping effect of internal thermal mass in typical buildings.

1.7.3 Control Strategy

See 1.1.3.

1.8 Variation: Thermal Energy Storage (TES)

The TES is only tested in load shifting scenario.

1.8.1 Test Day Selection

See 1.1.1.

1.8.2 Simulation Settings

See 1.1.2.

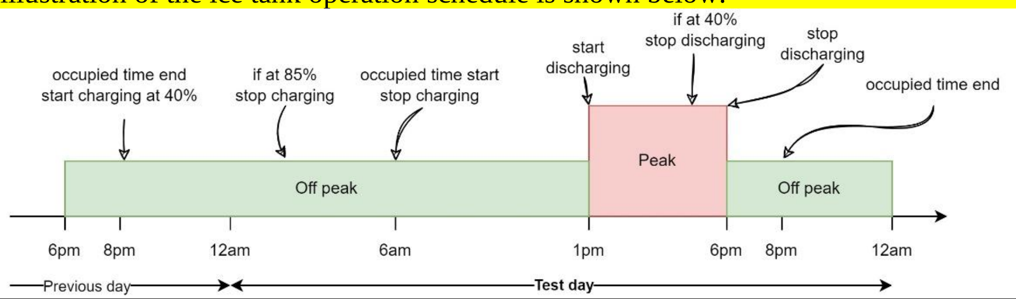
1.8.3 Control Strategy

1.8.3.1 Setpoint Schedule

Item	Default setpoint	Reset strategy
Zone Temperature	Occupied (6am – 8pm): 68 °F – 78 °F; Unoccupied/setback (12am – 6am, 8pm – 12am): 55 °F – 90 °F	No reset
AHU Supply Air Temperature	55 °F (12.8 °C)	No reset
AHU Supply Air Static Pressure	398.5 Pa (1.6 inH$_2$O)	**Shift**: Trim & Respond (see 2.2).
Chilled Water Supply Temperature	40 °F (4.4 °C)	Outdoor air temperature-based reset (see 2.1).
Chilled Water Secondary Loop Differential Pressure	551.6 kPa (80 psi)	No reset
VAV Minimum Air Flow*	Conference_Room_Mid_2: 460 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 200 CFM	No reset
Outdoor Air Flow Rate†	AHU1: 500 CFM AHU2: 240 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62-1989. The calculated minimum ventilation rates for the four zones are 460 CFM, 40 CFM, 40 CFM, and 200 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.
† The outdoor air flow rate is the sum of the minimum ventilation requirements of the downstream zones.

1.8.3.2 Equipment Availability Schedule

<table>
<tr><th>Equipment</th><th>Availability</th></tr>
<tr><td>Chiller 1</td><td>ON if any AHU is ON as indicated by the cooling coil valve being open for at least 90 seconds.
Will switch to Chiller 2 (during occupied period):
- If average power exceeds 8500 W and the chilled water temperature is 0.3 °F (0.17 °C) above the setpoint for 100 seconds.
- Or if average power exceeds 8000 W and the chilled water temperature is 1 °F (0.56 °C) above the setpoint for 100 seconds.
- Or if average power exceeds 7500 W for 16.7 minutes.</td></tr>
<tr><td>Chiller 2</td><td>ON if Ice Tank is in charging mode.
ON if a Chiller 2 switch ON rule as shown above is met.
Will switch to Chiller 1:
- If average power is below 3000 W for 100 seconds.</td></tr>
<tr><td>Ice Tank</td><td>The ice tank starts charging at 8pm of the previous day from approximately 40% inventory until 85% inventory or the beginning of the occupied time, whichever comes first, and starts discharging at the start of the peak hours until approximately 40% inventory or the end of the peak hours, whichever comes first. The graphical illustration of the ice tank operation schedule is shown below.

</td></tr>
<tr><td>AHU 1</td><td>Occupied period: always ON. Setback period: ON if there is a zone demand (i.e., zone temperature setpoint is violated).</td></tr>
<tr><td>AHU 2</td><td>Occupied period: always ON. Setback period: ON if there is a zone demand (i.e., zone temperature setpoint is violated).</td></tr>
</table>

1.9 Variation: MPC with Thermal Energy Storage (MPC&TES)

1.9.1 Test Day Selection

See 1.1.1.

1.9.2 Simulation Settings

See 1.1.2.

1.9.3 Control Strategy

1.9.3.1 Setpoint Schedule

Item	Default setpoint	Reset strategy
Zone Temperature	Based on MPC decisions (see 2.7)	NA
AHU Supply Air Temperature	Based on MPC decisions (see 2.7)	NA
AHU Supply Air Flow Rate	Based on MPC decisions (see 2.7)	NA

Chilled Water Supply Temperature	Based on MPC decisions (see 2.7)	NA
Chilled Water Secondary Loop Flow Rate	Based on MPC decisions (see 2.7)	NA
VAV Minimum Air Flow*	Conference_Room_Mid_2: 460 CFM Enclosed_Office_Mid_2: 200 CFM Enclosed_Office_Mid_5: 200 CFM Open_Office_Mid_1: 200 CFM	No reset
Outdoor Air Flow Rate†	AHU1: 500 CFM AHU2: 240 CFM	No reset

* The minimum air flow rate is determined to comply with ASHRAE 62-1989. The calculated minimum ventilation rates for the four zones are 460 CFM, 40 CFM, 40 CFM, and 200 CFM, respectively. Additionally, a 200 CFM hardware constraint imposed by the emulation setup limits the flow rate. Consequently, the minimum flow rate is set at the higher of these two values. This approach may result in higher than necessary airflow in certain zones (Enclosed_Office_Mid_2 and Enclosed_Office_Mid_5), which could potentially affect system efficiency, flexibility, and operational costs.
† The outdoor air flow rate is the sum of the minimum ventilation requirements of the downstream zones.

1.9.3.2 Equipment Availability Schedule

Equipment	**Availability**
Chiller 1	Based on MPC decision (see 2.7).
Chiller 2	Dedicated chiller for Ice Tank charging.
Ice Tank	Based on MPC decision (see 2.7). Not allowed to discharge during nonpeak period.
AHU 1	Occupied period: always ON. Setback period: Based on MPC decision (see 2.7).
AHU 2	Occupied period: always ON. Setback period: Based on MPC decision (see 2.7).

2 Control Algorithms

2.1 Chilled Water Supply Temperature ($T_{CHW,spt}$) Reset based on Outdoor Air Temperature

According to ASHRAE 90.1-2004 6.5.4.3, the chilled water supply temperature is reset based on the outdoor air temperature. Every 15 minutes, the chilled water supply temperature setpoint is reset based on the equation below, where $T_{CHW,spt}$ is the chilled water supply temperature and T_{out} is the outdoor air temperature, and their units are °F. The relationship between $T_{CHW,spt}$ and T_{out} is derived by employing a Trim and Respond algorithm (see 2.3) to identify optimal setpoints across different outdoor air temperatures, after which a bounded linear curve is fitted to these data points to model the setpoint adjustment.

$$T_{CHW,spt} = \min\left(\max(-0.267 \times T_{out} + 64, 40), 48\right)$$

2.2 AHU Supply Air Static Pressure Setpoint ($SP_{SA,spt}$) Reset based on Trim & Respond

The $SP_{SA,spt}$ controller determines how to reset $SP_{SA,spt}$ based on the mean position of the two VAV dampers. If the mean position of either damper exceeds a threshold, it is an indication that the zone needs more cooling, so the setpoint increases by Δ. If the mean positions of both dampers are below a threshold, this indicates that both zones are over-cooled, so the setpoint decreases by Δ. The controller logic is outlined in Table 2-1 and the parameters are defined in Table 2-2.

Table 2-1 Pseudocode describing the static pressure reset logic (high-performance control)

AHU $SP_{SA,spt}$ Reset Determine the duct static pressure used to set the fan speed (high-performance control)

```
ahuPR( )
first call: N = 0; dSumA = 0; dSumB = 0; SP_SA,spt = P_design
    if W > 50          // the fan is on
        N = N + 1
        dSumA = dSumA + dA        // sum of the damper position in VAV A
        dSumB = dSumB + dB        // sum of the damper position in VAV B
        if N > n
            if ((dSumA/N) or (dSumB/N)) > high      // more air is needed
                SP_SA,spt = SP_SA,spt + Δ
            else if ((dSumA/N) and (dSumB/N)) < low
                SP_SA,spt = SP_SA,spt – Δ
            endif
            N = 0
            dSumA = 0
            dSumB = 0
        endif
    return SP_SA,spt = max(min(SP_SA,spt,P_max),P_min)
```

Table 2-2 Tunable variables in the static pressure reset controller (high-performance control)

Variable	Description	Current Value
n	Number of timesteps between SP changes	12
high	Threshold to indicate that the dampers are open, indicating cooling is needed	7 V
low	Threshold to indicate that the dampers are closed, indicating less cooling is needed	5 V
Δ	The amount by which to change the SP	50 Pa (0.2 inH2O)
P_{max}	The maximum value of the SP	996 Pa (4 inH2O)
P_{min}	The minimum value of the SP	0 Pa (0 inH2O)
SP0	Default setpoint; when the system is off it will go to this SP and when it turns on it will start from this setpoint	398.5 Pa (1.6 inH2O)

2.3 AHU Supply Air Temperature Setpoint ($T_{SA,spt}$)Reset based on Trim & Respond

The $T_{SA,spt}$ controller has two parts: 1) determining if there is a cooling request and 2) adjusting $T_{SA,spt}$. More requests for cooling occur if the VAV dampers for a given AHU are both sufficiently open, the fan speed is sufficiently high, and there is no reheat. If there have been no requests for cooling, this may indicate that the air temperature is too low, and therefore $T_{SA,spt}$ increases by Δ. If there have been enough requests for cooling, this is an indication that the air temperature is too high, and it therefore decreases by Δ. If neither of these situations is true, $T_{SA,spt}$ remains unchanged. The controller logic is outlined in Table 2-3 and the parameters are defined in Table 2-4.

Table 2-3 Pseudocode for determining the supply air temperature setpoint for the AHU (high-performance control)

AHU $T_{SA,spt}$ Reset Determine the supply air temperature for the AHU (high-performance control)

ahuSat( )
first call: $T_{SA,spt} = T_{design}$; $N = 0$; $N_{req} = 0$
$N = N + 1$ // increment the reset counter
$rh = rh_1 + rh_2$ // if either VAV served by the AHU has reheat // enabled, rh > 0
$d = d_1 + d_2$ // sum of the VAV damper positions
$\overline{V}_{CC} = (V_{CC,sum} + V_{CC})/N$
if rh < 1 **and** d > V **and** vfd > Hz
$N_{req} = N_{req} + 1$ // there is a request for cooling
endif
if N > n // time to determine if the setpoint should be reset
N = 0
if $N_{req} < 1$ // no request for cooling
$T_0 = T_{SA,spt} + \Delta$
else if $N_{req} > n_{req}$ // requests for cooling
$T_0 = T_{SA,spt} - \Delta$
else
$T_0 = T_{SA,spt}$
endif

$N_{req} = 0$
if ($\overline{V}_{CC}$< 1.05 Low **and** $T_0 > T_{SA,\ spt}$) **or**
($\overline{V}_{CC}$ > 0.98 High **and** $T_0 < T_{SA,\ spt}$) // do not change the setpoint if the cooling // coil valve is saturated
$T_0 = T_{SA,spt}$
endif
endif
return $T_{SA,spt} = \max(\min(T_0, T_{max}), T_{min})$

Table 2-4 Tunable variables in the supply air temperature reset controller (high-performance control)

Variable	Description	Value
n	Number of timesteps between SP changes	12
n_{req}	Number of requests for cooling required for a SP decrease	2
V	Threshold to indicate that the dampers are open, indicating cooling is needed	15 V
Hz	Threshold to indicate that the fan is delivering a large volume of	55 Hz
Δ	The amount by which to change the SP	0.1 °C (0.2 °F)
T_{max}	The maximum value of the SP	18.3 °C (65 °F)
T_{min}	The minimum value of the SP	10 °C (50 °F)
T_{design}	The design value of the SP	12.8 °C (55 °F)
N	Count of timesteps between SP changes	≥ 0
N_{req}	Number of requests for cooling during the interval	≥ 0
rh_1	Status of reheat for the first VAV	0 (off) or 1 (on)
rh_2	Status of reheat for the second VAV	0 (off) or 1 (on)
rh	Status of reheat for both VAVs served by the AHU	≥ 0
d_1	Position of the damper in the first VAV	$2\ V \leq d_1 \leq 10\ V$
d_2	Position of the damper in the first VAV	$2\ V \leq d_2 \leq 10\ V$
d	Sum of the damper positions	
vfd	Fan speed	$15\ Hz \leq vfd \leq 60\ Hz$
V_{CC}	Cooling coil valve position	$Low \leq V_{CC} \leq High$
Low	Minimum position of the cooling coil valve	2 V
High	Maximum position of the cooling coil valve	6.5 V

2.4 Chilled Water Supply Temperature Setpoint ($T_{CHW,spt}$) Reset and Secondary Loop Differential Pressure Setpoint ($DP_{CHW,spt}$) Reset based on Trim & Respond

2.4.1 Reset Mode Determination

Two supervisory level controllers in the hydronic system make decisions when the zone cooling load changes: 1) reset the chilled water temperature setpoint ($T_{CHW,spt}$), and 2) reset the differential pressure setpoint in the secondary loop ($DP_{CHW,spt}$). Since these two controllers act on the same information, they cannot act at the same time without interfering with each other, so this section describes how to determine which one will be reset first. If the zones need less cooling, the controllers will increase $T_{CHW,spt}$ or decrease $DP_{CHW,spt}$. In general, increasing $T_{CHW,spt}$ will lead to a greater reduction in energy consumption than decreasing $DP_{CHW,spt}$, so the first step is to increase $T_{CHW,spt}$ to its maximum value and then start decreasing $DP_{CHW,spt}$.

Figure 2-1 is a pictorial depiction of the coordinated operation of the two controllers. When more cooling is needed, as indicated by at least one cooling coil valve being open beyond a threshold, $T_{CHW,spt}$ is at its minimum and $DP_{CHW,spt}$ is at its maximum (maximum pump speed). As both cooling coil valves close to a point below a threshold, $T_{CHW,spt}$ increases. Once it reaches its maximum value, and the cooling coil valves are still closed below the threshold, $DP_{CHW,spt}$ starts to decrease. At some point $T_{CHW,spt}$ is at its maximum and $DP_{CHW,spt}$ is at its minimum. In other words, for the high-performance controllers in the hydronic system, it will always reset $T_{CHW,spt}$ first. If $T_{CHW,spt}$ is at its maximum or minimum value, $DP_{CHW,spt}$ will be reset.

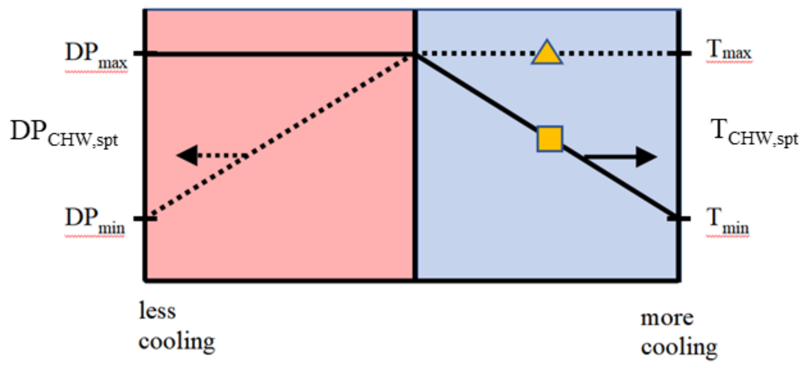


Figure 2-1 Sketch of the coordinated operation of the $DP_{CHW,spt}$ and $T_{CHW,spt}$ controllers

Table 2-5 shows the pseudocode for determining if the mode is $T_{CHW,spt}$ or $DP_{CHW,spt}$. When the system first turns on, $T_{CHW,spt}$ = 10 °C (50 °F), which is between T_{max} and T_{min}, and $DP_{CHW,spt}$ = DP_{max}. These points are show in Figure 2-1 as a square and a triangle, respectively. So, when the system turns on it starts out in $DP_{CHW,spt}$ reset mode. Table 2-6 defines the variables used in this controller. The following sections describe the logic of each controller when in that mode.

Table 2-5 Pseudocode for determining if $T_{CHW,spt}$ or $DP_{CHW,spt}$ controller is primary (high-performance control)

Mode Determination: Determine if the CHWST or DP controller should be enabled	
<u>mode( )</u>	
if chMode == Auto **and** dpMode == Auto	// both controllers are in auto // mode and the chiller and // secondary loop pump are on
if $T_{CHW,spt} > T_{min}$ **and** $T_{CHW,spt} < T_{max}$	
mode = CHWST	// the mode is CHWST
else	
mode = DP	// the mode is DP
endif	
else if chMode == Auto **and** dpMode == Manual	

```
        mode = CHWST
    else
        mode = DP
    endif
    return mode
```

Table 2-6 Variables used to determine the CHWST or DP mode (high-performance control)

Variable	Description	Current Value
chMode	Manual – manually set the chiller setpoint; Auto – automatically set the chiller setpoint; requires that the chiller is on	Auto
slMode	Manual – manually set the secondary loop DP setpoint; Auto – automatically set the secondary loop DP setpoint; requires that the pump is on	Auto
T_{min}	Minimum chiller setpoint	4.4 °C (40 °F)
T_{max}	Maximum chiller setpoint	21.1 °C (70 °F)
mode	CHWST – vary the chiller setpoint; DP – vary the DP setpoint	DP

2.4.2 Chilled Water Supply Temperature (T_{chwst}) Reset

The $T_{CHW,spt}$ controller determines how to reset $T_{CHW,spt}$ based on the mean position of the two cooling coil valves, v12 and v13. If the mean position of either valve exceeds a threshold, it is an indication that the zone cooling demand is high, so the setpoint decreases by Δ. If the mean positions of both valves are below a threshold, this indicates that the zone cooling demand is low, so the setpoint increases by Δ. The controller logic is outlined in Table 2-7 and the parameters are defined in Table 2-8.

Table 2-7 Pseudocode for calculating $T_{CHW,spt}$ (high-performance control)

Chilled Water Setpoint Temperature: Calculate the chiller setpoint temperature.

```
chwst( )
first call: N = 0; v12Sum = 0; v13Sum = 0; T_CHW,spt = SP0
    mode = mode()
    if W > 100 and mode == CHWST        // The chiller is on and the mode is CHWST
        N = N + 1
        v12Sum = v12Sum + v12    // sum of the cooling coil valve position
                                 // in AHU1 during the N timesteps
        v13Sum = v13Sum + v13    // sum of the cooling coil valve position
                                 // in AHU2 during the N timesteps
        if N > n
            if ((v12Sum/N) or (v13Sum/N)) > high    // more cooling is needed
                T_CHW,spt = T_CHW,spt - Δ
            else if ((v12Sum /N) and (v13Sum /N)) < low
                T_CHW,spt = T_CHW,spt + Δ
            endif
            N = 0
```

v12Sum = 0
v13Sum = 0
endif
endif
return $T_{CHW,spt} = \max(\min(T_{CHW,spt}, T_{max}), T_{min})$

Table 2-8 Variables in the $T_{CHW,spt}$ controller (high-performance control)

Variable	Description	Current Value
n	Number of timesteps between SP changes	18
high	Threshold to indicate that the valves are open, indicating more cooling is needed	6 V
low	Threshold to indicate that the valves are closed, indicating less cooling is needed	3.1 V
Δ	The amount by which to change the SP	0.28 °C (0.5 °F)
T_{min}	Minimum chiller setpoint	4.4 °C (40 °F)
T_{max}	Maximum chiller setpoint	21.1 °C (70 °F)
SP0	Default setpoint; when the system is off it will go to this SP and when it turns on it will start from this setpoint	10 °C (50 °F)

2.4.3 Secondary Loop Differential Pressure ($DP_{CHW,spt}$) Reset

The $DP_{CHW,spt}$ controller determines how to reset $DP_{CHW,spt}$ based on the mean position of the two cooling coil valves, v12 and v13. If the mean position of either valve exceeds a threshold, it is an indication that the zone cooling demand is high, so the setpoint increases by Δ. If the mean positions of both valves are below a threshold, this indicates that the zone cooling demand is low, so the setpoint decreases by Δ. The controller logic is outlined in Table 2-9 and the parameters are defined in Table 2-10.

Table 2-9 Pseudocode for determining the differential pressure setpoint for the secondary loop pump (high-performance control)

Secondary Loop Differential Pressure Setpoint: Determine $DP_{CHW,spt}$ for the secondary loop pump.

slDPR( )
first call: N = 0; v12Sum = 0; v13Sum = 0; $DP_{CHW,spt}$ = SP0
if W > 50 **and** mode == DP // the pump is on and the mode is DP
N = N + 1
v12Sum = v12Sum + v12 // sum of the cooling coil valve position
// in AHU1 during the N timesteps
v13Sum = v13Sum + v13 // sum of the cooling coil valve position
// in AHU2 during the N timesteps
if N > n
if ((v12Sum / N) **or** (v13Sum / N)) > high // more cooling is needed
$DP_{CHW,spt} = DP_{CHW,spt} + \Delta$
else if ((v12Sum / N) **and** (v13Sum / N)) < low
$DP_{CHW,spt} = DP_{CHW,spt} - \Delta$
endif
N = 0

```
        v12Sum = 0
        v13Sum = 0
    endif
    return DP_CHW,spt = max(min(DP_CHW,spt, DP_max), DP_min)
```

Table 2-10 Variables in the differential pressure reset controller in the secondary loop (high-performance control)

Variable	Description	Current Value
n	Number of timesteps between SP changes	12
high	Threshold to indicate that the valves are open, indicating more cooling is needed	6 V
low	Threshold to indicate that the valves are closed, indicating less cooling is needed	3.1 V
Δ	The amount by which to change the SP	13.8 kPa (2 psi)
DPmax	The maximum value of the SP	379 kPa (55 psi)
DPmin	The minimum value of the SP	103.4 kPa (15 psi)
SP0	Default setpoint; when the system is off it will go to this SP and when it turns on it will start from this setpoint	379 kPa (55 psi)

2.5 VAV Minimum Airflow Rate ($V_{VAV,minspt}$) Reset

According to Section 6.5.3.8 of ASHARE Standard 90.1-2019 and Section 6.2.6.1.4 of ASHARE Standard 62.1-2019, the minimum airflow rate is permitted to be reduced to zero when the zone is scheduled to be occupied and the occupancy sensor indicates zero population within the zone. This is known as the occupied-standby mode. For IBAL testing, it is assumed that each zone has an occupancy sensor. Notice that in the simulation, the sensed occupancy status is equivalent to the scheduled occupancy status. The controller logic is outlined in Table 2-11 and the parameters are defined in Figure 2-1.

Table 2-11 Pseudocode for determining minimum ventilation rate (high-performance control)

Zone Minimum Ventilation Rate Reset Determine the minimum ventilation of VAV damper (high-performance control)

```
ahuSat( )
first call: V = V_design; N = 0; N_occ = 0
    N = N + 1                  // increment the reset counter
    if Occ = 1
        N_occ=N_occ+1          // Sensor detects there are occupants in the zone
    endif
    if N > n                   // time to determine if the setpoint should be reset
        if N_occ < 1           // No occupant in the zone (occupied-standby mode)
            V_0 = 0
        else
            V_0 = V_design
        endif
        N = 0
```

```
        N_occ = 0
    endif
    return V = V_0
```

2.6 Fan Power Modulation

For load modulating cases (i.e., **Mod**), from 8:00 am to 8:40 am the static pressure (DSP) setpoints of both AHU fans are adjusted so that the fan power tracks the reference fan power signal. The reference fan power signal is determined by the PJM signal, the bid amount, and the baseline fan power, as shown below. The fan power of the corresponding **Efficiency (Eff)** case is treated as the baseline.

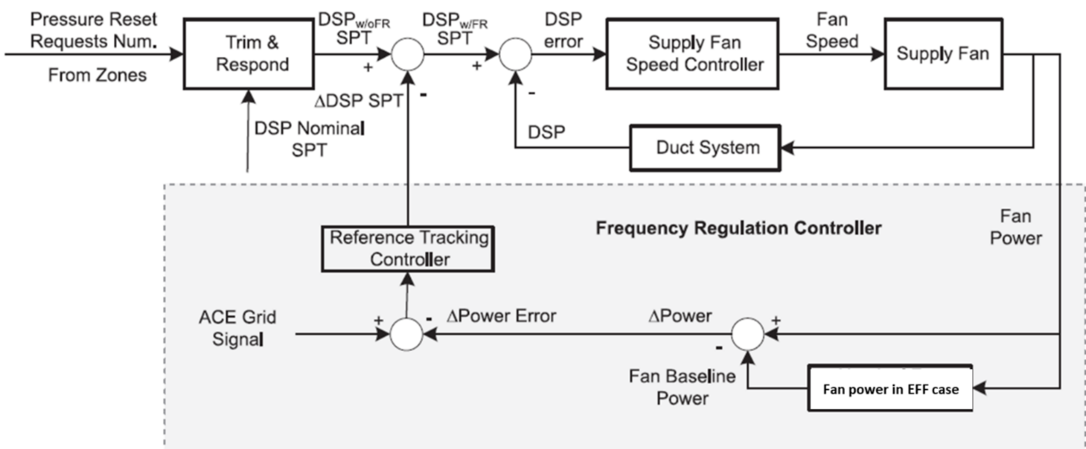


Figure 2-2 Static pressure reset with fan power modulation

2.7 Model Predictive Control

MPC takes the grid signals, disturbance predictions, and system measurements as inputs and generates optimal setpoints for the system. The setpoints include the chilled water supply temperature and flow rate setpoints, the zone supply air temperature and flow rate setpoints, and the zone thermostat setpoint. A two-level hierarchical predictive controller is designed, shown in Figure 2-3.

- The high-level predictive controller adopts an MPC scheme by utilizing the slow dynamics of active storage, such as an ice tank, on a slow time (i.e., 1 hour) basis. It is designed as an open-loop control for planning the system operation mode, chiller cooling rate, storage heat transfer (e.g., charging and discharging) rate, and zone air temperature setpoints. The controller minimizes the operation costs for a long future, while maintaining thermal comfort in the zones and satisfying the physical constraints for individual equipment.
- With respect to the optimal schedules from the high-level controller, the low-level controller further generates control policies for different equipment on a fast time (i.e., 15 minutes) basis. For each operation mode other than M0, a separate nonlinear programming problem is formulated to coordinate different equipment in that mode to track the optimal

plans. The goal is to minimize the control errors between the schedules and their corresponding measurements, while respecting the HVAC system dynamics that occur faster.

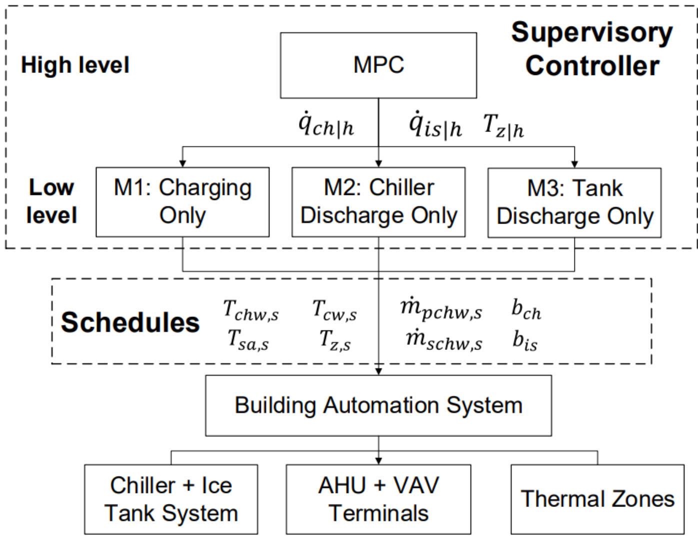


Figure 2-3 Two-level hierarchical predictive controller

The following content describes the details of high-level MPC, low-level MPC, zone temperature state space model used in the MPC, and the MPC implementation with the IBAL testbed.

2.7.1 High-level MPC

The main purpose of the high-level MPC is to decide on which cooling sources should be on and how much cooling energy they should produce to meet the control goals. Eq. (1) to Eq. (4) denote the decision variables for the MPC optimization problem.

$$u_h = [\dot{q}_{ch,c}, \dot{q}_{ch,d}, \dot{q}_{is,d}, \dot{q}_{oa}, \dot{q}_{hvac}, b_{ch,c}, b_{ch,d}, b_{is,d}, z_{P_{ch,d}}, z_{\dot{q}_{ch,d}}, z_{\dot{q}_{is,d}}, \epsilon_T] \quad (1)$$

$$\underline{u}_h = [\underline{\dot{q}}_{ch,c}, \underline{\dot{q}}_{ch,d}, \underline{\dot{q}}_{is,c}, \underline{\dot{q}}_{oa}, \underline{\dot{q}}_{hvac}, 0,0,0, \underline{P}_{ch,d}, \underline{\dot{q}}_{ch,d}, \underline{\dot{q}}_{is,d}, \underline{\epsilon}_T] \quad (2)$$

$$\overline{u}_h = [\overline{\dot{q}}_{ch,c}, \overline{\dot{q}}_{ch,d}, \overline{\dot{q}}_{is,c}, \overline{\dot{q}}_{oa}, \overline{\dot{q}}_{hvac}, 1,1,1, \overline{P}_{ch,d}, \overline{\dot{q}}_{ch,d}, \overline{\dot{q}}_{is,d}, \overline{\epsilon}_T] \quad (3)$$

$$U_h^t = [u_h^t, u_h^{t+1}, \dots, u_h^{t+N_h-1}] \quad (4)$$

where u_h is the decision variable vector for one step, $\underline{u}_h$ and $\overline{u}_h$ are the lower bound and upper bound, respectively, and U_h^t is the decision variable vector for the MPC problem at time t. $\dot{q}_{ch,c}$ is charging energy rate by the chiller when charging the ice tank, $\dot{q}_{ch,d}$ is the discharging energy rate by the chiller for cooling the building, $\dot{q}_{is,d}$ is the ice storage discharging rate, $\dot{q}_{oa}$ is the thermal load introduced by maintaining an outdoor air ratio, $\dot{q}_{hvac}$ is cooling energy provided by the HVAC system. The lower bounds for the above four variables are greater than 0, representing the minimum available energy due to physical constraints such as minimum compressor speed in chillers etc. $b_{ch,c}, b_{ch,d}$ and $b_{is,d}$ are binary variables that represent chiller in charging mode, chiller in discharging mode, and ice storage in discharging mode respectively. $z_{p_{ch,d}}$, $z_{\dot{q}_{ch,d}}$ and $z_{\dot{q}_{is,d}}$ are ancillary variables representing the power consumption of the chiller to cool the building, the chiller discharging energy rate and the ice tank discharging rate, respectively, which intends to linearize the otherwise bilinear terms. ϵ_T is a slack variable introduced for zone temperature control, and a small positive value.

The high-level formulation is shown below:

$$\min_{U_h^t} \sum_{k=1}^{N_h} [\omega_{h,1} p^{t+k} (P_{cp}^{t+k} + P_{ahu}^{t+k}) \triangle t_h + \omega_{h,2} \epsilon_T^{t+k} + \omega_{h,3} | \triangle u_h^{t+k} |^2] \tag{5}$$

$$\min_{U_h^t} \sum_{k=1}^{N_h} [\omega_{h,1} (P_{cp}^{t+k} + P_{ahu}^{t+k}) \triangle t_h + \omega_{h,2} \epsilon_T^{t+k} + \omega_{h,3} | \triangle u_h^{t+k} |^2] \tag{6}$$

$$\dot{q}_{hvac}^{t+k} = b_{ch,d}^{t+k} \dot{q}_{ch,d}^{t+k} + b_{is,d}^{t+k} \dot{q}_{is,d}^{t+k} + \dot{q}_{oa}^{t+k} \tag{7}$$

$$x^t = x(t) \tag{8}$$

$$T_z^{t+k} = SSM_{z,h}(x^{t+k-1}, u_h^{t+k-1}, d_h^{t+k-1}) \tag{9}$$

$$\underline{T}_z^{t+k} - \epsilon_T^{t+k} \leq T_z^{t+k} \leq \overline{T}_z^{t+k} + \epsilon_T^{t+k} \tag{10}$$

$$\chi^t = \chi(t) \tag{11}$$

$$\chi^{t+k} = \chi^{t+k-1} + \frac{b_{ch,c}^{t+k} \dot{q}_{ch,c}^{t+k} - b_{is,d}^{t+k} \dot{q}_{is,d}^{t+k}}{E_{is,max}} \triangle t_h \tag{12}$$

$$\underline{\chi} \leq \chi^{t+k} \leq \overline{\chi} \tag{13}$$

$$\dot{q}_{oa}^{t+k} = \dot{m}_{oa}^{t+k} C_{pa} (T_{oa}^{t+k} - T_z^{t+k}) \tag{14}$$

$$P_{ch,d}^{t+k} = a_0 + a_1 \dot{q}_{ch,d}^{t+k} \tag{15}$$

$$P_{ch}^{t+k} = b_{ch,c}^{t+k} P_{ch,c}^{t+k} + b_{ch,d}^{t+k} P_{ch,d}^{t+k} \tag{16}$$

$$P_{cp}^{t+k} = P_{ch}^{t+k} + P_{pchw}^{t+k} + (1 - b_{ch,c}^{t+k}) P_{schw}^{t+k} \tag{17}$$

$$P_{ahu}^{t+k} = f_{ahu}(\dot{q}_{hvac}^{t+k}) = \sum_{i=0}^{3} \alpha_i (\dot{q}_{hvac}^{t+k})^i \tag{18}$$

$$b_{ch,c}^{t+k} + b_{ch,d}^{t+k} + b_{is,d}^{t+k} \leq 1 \tag{19}$$

$$\triangle u_h^{t+k} = u_h^{t+k} - u_h^{t+k-1} \tag{20}$$

The objective of load shifting cases is shown as Eq. (5), and the objective of the energy efficiency and load shedding cases is shown as Eq. (6). The objective described has three terms: energy cost or consumption, temperature violation and skew rates of the control actions. In the energy cost term, p is the energy price, $P_{cp}$is the power consumed by the chiller plant, consisting of chillers and pumps. P_{ahu} is the power consumed by the AHU fan and $\triangle t_h$ is the control interval of the high-level predictive controller, i.e., 1 hour. In the temperature violation term, the slack variable

ϵ_T is used to estimate the temperature violations from given bounds. The skew rate of the control actions $\triangle u_h$ measures how fast the control actions change between adjacent steps and is defined as Eq. (20). The tradeoff between these three terms is tuned by weighting factors $\omega_{h,1}$, $\omega_{h,2}$ and $\omega_{h,3}$.

Eq. (7) enforces energy balance between the cooling system and the thermal zones, which contains two bilinear terms that represent cooling energy provided by discharging the chiller and that provided by discharging the ice tank. This bilinear formulation means the cooling energy provided to the zone is equal to the total amount of cooling energy provided by the chiller and ice tank depending on the operation mode. Eq. (8 - 10) describe zone-level equality and inequality constraints, including a zone dynamics model and thermal comfort bounds. Eq. (8) defines the initial states at time t, which are estimated from a linear Kalman Filter by adding Gaussian modeling errors and measurement errors to the discrete time SSM. Eq. (9) describes the discrete-time state-space model (SSM) for zone air temperature dynamics. $\underline{T}_z$ and $\overline{T}_z$ are the allowable lower and upper bounds for zone air temperature. Similarly, Eq. (11 - 13) models the storage charging and discharging dynamics in terms of state of charge (SOC) and its physical constraints, where $E_{is,max}$ is the maximum energy that can be stored in the storage.

Eq. (14) models the thermal load introduced by outdoor air. Eq. (15) is a chiller power model during the discharging mode as a linear function of the chiller's cooling load, which can be identified from experimental data. Eq. (16) predicts the total power for the chiller by summing up two bilinear terms that represent the power for charging the ice tank and that for providing cooling to the building, respectively. The total power consumed by the whole chiller plant is presented in Eq. (17), which includes the power from the chiller, primary pump and secondary pumps. The AHU power is represented as a polynomial equation depending on the total cooling energy provided to the building as shown in Eq. (18). Eq. (19) adds a constraint to represent the physical constraints on the selection of operation modes.

2.7.2 Low-level MPC

The low-level predictive controller aims to optimally operate the system under given operation modes while respecting the schedules from the high-level MPC. The low-level predictive controller operates at a faster time step (e.g., 15 min) to respect the system states (i.e., zone temperature and tank SOC) and schedules (i.e. energy transfer rate from different cooling sources) optimized from the high-level controller. Except for M0, where the system is off, and M1, where the system is under charging mode with a predefined fixed schedule, we established an optimal control problem for each operation mode.

<u>M2: Chiller Discharging Mode</u>

When $b_{ch,d}$ = 1 and $b_{is,d}$ = 0, mode M2 is activated so that only the chiller is on to provide cooling to the building. To respect the high-level controller, the following problem is formulated:

$$\min_{U_l^t} [\omega_1 |\epsilon_T^{t+N_t}|^2 + \sum_{k=1}^{N_t} \omega_2 |\epsilon_q^{t+k}|^2] \tag{21}$$

$$u_l = [T_{cws}, T_{chws}, T_{ma}, \triangle T_{cc}, \dot{m}_{schw}, \dot{m}_{sa}, \dot{q}_{ch}, \epsilon_T, \epsilon_q] \tag{22}$$

$$U_l^t = [u_l^t, u_l^{t+1}, \dots, u_l^{t+N_l-1}] \tag{23}$$

$$x^t = x(t) \tag{24}$$

$$T_z^{t+k} = SSM_{z,l}(x^{t+k-1}, u_l^{t+k-1}, d_l^{t+k-1}) \tag{25}$$

$$P_{ch}^{t+k} = \alpha_0 + \alpha_1 T_{chw}^{t+k} + \alpha_2 (T_{chw}^{t+k})^2 + \alpha_3 T_{cw}^{t+k} + \alpha_4 (T_{cw}^{t+k})^2 + \alpha_5 T_{chw}^{t+k} T_{cw}^{t+k} + \alpha_6 (\dot{q}_{ch}^{t+k}) + \alpha_7 (\dot{q}_{ch}^{t+k})^2 \tag{26}$$

$$P_{pchw}^{t+k} = \overline{P}_{pchw} \tag{27}$$

$$P_{schw}^{t+k} = \sum_{i=0}^{3} \beta_i (\dot{m}_{schw}^{t+k})^i \tag{28}$$

$$P_{cp}^{t+k} = P_{ch}^{t+k} + P_{pchw}^{t+k} + P_{schw}^{t+k} \tag{29}$$

$$P_{ahu}^{t+k} = \sum_{i=0}^{3} \gamma_i (\dot{m}_{sa}^{t+k})^i \tag{30}$$

$$\dot{q}_{ch}^{t+k} = \dot{m}_{sa}^{t+k} C_{pa} \triangle T_{cc}^{t+k} \tag{31}$$

$$\dot{q}_{ch}^{t+k} = \dot{m}_{schw}^{t+k} C_{pw} (T_{chwr,coi}^{t+k} - T_{chws}^{t+k}) \tag{32}$$

$$\dot{m}_{oa}^{t+k} T_{oa}^{t+k} + (\dot{m}_{sa}^{t+k} - \dot{m}_{oa}^{t+k}) T_z^{t+k} - \dot{m}_{sa}^{t+k} T_{ma}^{t+k} = 0 \tag{33}$$

$$T_{sa}^{t+k} = T_{ma}^{t+k} - \triangle T_{cc}^{t+k} \tag{34}$$

$$T_{chwr,coi}^{t+k} \le T_{sa}^{t+k} - \zeta_1 \tag{35}$$

$$T_{chws}^{t+k} \le T_{sa}^{t+k} - \zeta_2 \tag{36}$$

$$\frac{\dot{q}_{ch,cap}^{t+k}}{\bar{q}_{ch}} = \delta_0 + \delta_1 T_{chws}^{t+k} + \delta_2 (T_{chws}^{t+k})^2 + \delta_3 T_{cws}^{t+k} + \delta_4 (T_{cws}^{t+k})^2 + \delta_5 T_{chws}^{t+k} T_{cws}^{t+k} \tag{37}$$

$$PLR_{min} \dot{q}_{ch,cap}^{t+k} \le \dot{q}_{ch}^{t+k} \le PLR_{max} \dot{q}_{ch,cap}^{t+k} \tag{38}$$

$$-\epsilon_T \le T_{z \leftarrow h}^t - T_z^{t+N_t} \le \epsilon_T \tag{39}$$

$$-\epsilon_q^{t+k} \le \dot{q}_{ch,d \leftarrow h}^t - \dot{q}_{ch}^{t+N_t} \le \epsilon_q^{t+k} \tag{40}$$

$$\sum_{k=1}^{M} (P_{cp}^{t+k} + P_{ahu}^{t+k}) \triangle t_l \le (P_{cp \leftarrow h}^t + P_{ahu \leftarrow h}^t) \triangle t_h \tag{41}$$

The control objective as shown in Eq. (21) is to respect the optimal schedules generated from the high-level controller and thus to minimize the zone temperature differences and the chiller cooling rate differences between the high-level schedules and the low-level fulfillment. The minimal differences are achieved by searching the decision variables u_l over the future time span $[t, t + N_l]$. For each time step, the decision vector ul defined as Eq. (22) includes the condenser water supply temperature T_{cws}, chilled water supply temperature T_{chws}, air-side temperature difference across the cooling coil $\triangle T_{cc}$, secondary pump mass flow rate $\dot{m}_{schw}$, supply air mass flow rate $\dot{m}_{sa}$, chiller cooling rate $\dot{q}_{ch}$, slack tolerance ϵ_T for zone temperature control, and ϵ_q for chiller cooling rate balance.

Eq. (24) and Eq. (25) describes the low-level discrete-time SSM for zone air temperature dynamics. Eq. (26-30) models the power of major cooling system device, such as chiller, primary chilled water pump and secondary chilled water pump, and AHU, where $\overline{P}_{pchw}$ is the nominal power by the CHWP, and the Greek symbols α, β, γ are the identified model parameters from experiment data. Eq. (31-36) add constraints for the steady-state energy balance between the air-side and water-side of the cooling coil, where ζ_1 is the allowable temperature differences between the chilled water return temperature and the supply air temperature, ζ_2 is the approach temperature of the cooling coil.

Eq. (37-38) bounds the chiller cooling rate based on operational conditions such as CHWST and allowable part load ratio (PLR), where $\dot{q}_{ch,cap}$ is the chiller's available cooling capacity, $\overline{\dot{q}}_{ch}$ is the nominal capacity, and the Greek symbol δ represents the curve coefficients calibrated from experiment data. Eq. (39 - 41) are inequality constraints that force the low-level controller to respect to the high-level schedules in terms of zone air temperature, chiller cooling rate and total power consumption, where $T^t_{z\leftarrow h}$ $\dot{q}^t_{ch,d\leftarrow h}$, $P^t_{cp\leftarrow h}$ and $P^t_{ahu\leftarrow h}$ are the zone temperature schedule, chiller cooling rate schedule, chiller plant power schedule and AHU power schedule from the high-level controller generated at time t, respectively.

<u>M3: Ice Tank Discharging Mode</u>

This mode is activated when $b_{is,d}$ = 1 and $b_{ch,d}$ = 0. To respect the high-level controller, the following problem is established with the objective of minimizing the termination cost of zone air temperature and ice tank SOC and tracking errors for ice tank discharging rate schedules. The decision variables are listed as Eq. (43) and (23), where $\dot{q}_{is}$ is the low-level ice tank discharging rate, and ϵ_χ is the slack variable for the terminal tank SOC.

$$\min_{U_l^t}[\omega_1|\epsilon_T^{t+N_t}|^2 + \omega_2|\epsilon_\chi^{t+N_t}|^2 + \sum_{k=1}^{N_t}\omega_3|\epsilon_q^{t+k}|^2] \tag{42}$$

$$u_l = [T_{chws}, \triangle T_{cc}, \dot{m}_{schw}, \dot{m}_{sa}, \dot{q}_{is}, \epsilon_T, \epsilon_\chi, \epsilon_q] \tag{43}$$

$$\chi^t = \chi(t) \tag{44}$$

$$\chi^t = \chi^{t+k-1} - \frac{\dot{q}_{is,d}^{t+k}\triangle t_l}{E_{is,max}} \tag{45}$$

$$P_{cp}^{t+k} = P_{pchw}^{t+k} + P_{schw}^{t+k} \tag{46}$$

$$\dot{q}_{is}^{t+k} = \dot{m}_{sa}^{t+k}C_{pa} \triangle T_{cc}^{t+k} \tag{47}$$

$$\dot{q}_{is}^{t+k} = \dot{m}_{schw}^{t+k}C_{pw}(T_{chwr,coi}^{t+k} - T_{chws}^{t+k}) \tag{48}$$

$$\dot{q}_{is,cap}^{t+k} = \sum_{i=0}^{3}\eta_i(\dot{\chi}^{t+k})^i \tag{49}$$

$$PLR_{min}\dot{q}_{is,cap}^{t+k} \leq \dot{q}_{is}^{t+k} \leq PLR_{max}\dot{q}_{is,cap}^{t+k} \tag{50}$$

$$-\epsilon_{\chi}^{t+k} \leq \chi_{\leftarrow h}^{t} - \chi^{t+N_l} \leq \epsilon_{\chi}^{t+N_l} \tag{51}$$

$$-\epsilon_{q}^{t+k} \leq \dot{q}_{is,d\leftarrow h}^{t} - \dot{q}_{is}^{t+\mathrm{k}} \leq \epsilon_{q}^{t+k} \tag{52}$$

Eq. (44-45) are the ice tank SOC dynamics, and Eq. (46) is the total power for the hydraulic system. Eq. (47-48) are steady-state heat balance equation for the cooling coil under M3. The ice tank time-variant discharging capacity is modeled as a polynomial equation of its current SOC as described in Eq. (49). This controller should also respect the high-level optimal terminal conditions for the zone air temperature (Eq.(39)) and ice tank SOC (Eq.(51)), and the optimal schedules for ice tank discharging rate (Eq. (52)) and system power (Eq. (41)).

2.7.3 Zone Temperature State Space Model

A dynamic zone air temperature predictor for the aggregated zone is needed when the zone air temperature is used for the high-level and low-level MPC. A lumped RC thermal network model is used to predict the average zone temperature for each of the two zones that are served by the same AHU. The building thermal load model is sketched in Figure . The resulting state space model (SSM) based on energy conservation are shown in Eq. (53-55).

$$C_z \frac{dT_z}{dt} = \frac{T_{oa}-T_z}{R_g} + \frac{T_{w,i}-T_z}{R_i} + \dot{q}_{conv,i} + \dot{q}_{hvac} \tag{53}$$

$$C_{w,e} \frac{dT_{w,e}}{dt} = \frac{T_{oa}-T_{w,e}}{R_e} + \frac{T_{w,i}-T_{w,e}}{R_w} + \dot{q}_{sol,e} \tag{54}$$

$$C_{w,i} \frac{dT_{w,i}}{dt} = \frac{T_{w,e}-T_{w,i}}{R_w} + \frac{T_z-T_{w,i}}{R_i} + \dot{q}_{rad,i} \tag{55}$$

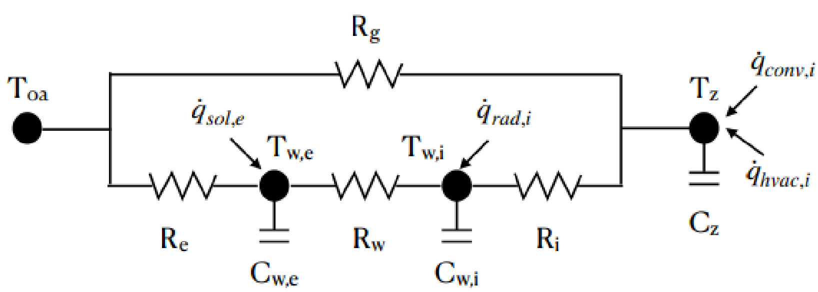


Figure 2-4 RC-based thermal zone model

R_g represents the thermal resistance of windows. The walls are separated into two layers: $C_{w,e}$ and $C_{w,i}$ are thermal capacitance of the exterior layer and interior layer respectively. The thermal resistance between $C_{w,e}$ and $C_{w,i}$ is modeled by R_w, while R_e and R_i capture the thermal resistance associated with heat convection across the exterior and interior wall surfaces respectively. C_z is the thermal capacitance in the zone. The model inputs are outside air temperature T_{oa}, external solar load $\dot{q}_{sol,e}$, internal radiative heat gain $\dot{q}_{rad,i}$, internal convective heat gain $\dot{q}_{conv,i}$, and HVAC heat rate $\dot{q}_{hvac,i}$. The model output is the zone air temperature T_z.

2.7.4 Implementation with IBAL

The two-stage hierarchical predictive controller proposed above is a generalized control method for AHU-VAV systems. There is a gap between this control strategy and the actual IBAL system, specifically that the controller is considering the case of a single zone, whereas the IBAL system has four zones controlled by two AHUs. In this regard, when applying the MPC in practice, the weighted average zone air temperature is used as the measurement value input to the MPC. Then the MPC will perform subsequent operations based on this weighted average zone air temperature. The weighting ratio is determined based on the maximum cooling load of the four zones during the peak period. Specifically, the four zones in IBAL are Conference Room (serviced by AHU1, minimum flow rate = 460 CFM), Enclosed Office 1 (serviced by AHU1, minimum flow rate = 200 CFM), Open Office (serviced by AHU2, minimum flow rate = 200 CFM), and Enclosed Office 2 (serviced by AHU2, minimum flow rate = 200 CFM). By simulation, the weighting ratio of the four zones in peak period is [0.31, 0.18, 0.33, 0.18]. In addition to this, since the MPC only considers one zone, the supply air mass flow rate setpoint output from the MPC is an aggregated mass flow rate setpoint. This aggregated setpoint would be assigned to both AHUs based on the same weighting ratio. The lower bound of the $\dot{m}_{sa}$ is set to 1500 CFM to account for the fact that after assigning, the air supply also needs to meet the minimum air supply requirement.