HVAC Optimisation

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Project Overview

Heating, Ventilation, and Air Conditioning (HVAC) systems are critical components in ensuring thermal comfort in institutional buildings. However, they also contribute significantly to energy consumption and operational costs. This project focuses on simulating and optimizing the HVAC loads in lecture halls to improve energy efficiency without compromising comfort.

IIT Kanpur's lecture halls exhibit high and uneven HVAC load profiles, particularly during peak daytime hours.There is a lack of intelligent scheduling and predictive control, leading to energy inefficiency and potential overloading.

Understanding System:

The simplification of system is as follows:- the model assumes that there is a one cooling unit attached to 5 Lecture Halls , each Lecture Hall unit can have upto a certain amount of load assigned to it.

Theory

for the dynamic simulation of building-plant system,besides the dynamic procedure to calculate the building thermal load we normally find a method toevaluate the average performance of the plant in steady-state conditions on a time interval equal to thetime step of the simulation (usually one hour). As regards the inverse cycle machine, the procedure is based on the use of working curves from the manufacturer or from literature and in details it consists of two stages. First the calculation of the capacity and of the coefficient of performance (COP) at full load operating conditions normally as a functions of the thermal levels of the external fluidsexchanging heat fluxes with the machine at the condenser and evaporator

The indoor design conditions are directly related to human comfort. Current comfort standards, ASHRAE Standard 55- 1992 [4] and ISO Standard 7730 [5], specify a "comfort zone," representing the optimal range and combinations of thermal factors (air temperature, radiant temperature, air velocity, humidity) and personal factors (clothing and activity level) with which at least 80% of the building occupants are expected to express satisfaction. As a general guideline for summer air-conditioning design, the thermal comfort chapter of the ASHRAE fundamentals handbook (Chapter 8, 2001) provides a snapshot of the psychrometric chart for the summer and winter comfort zones.

For most of the comfort systems, the recommended indoor temperature and relative humidity are:

1) Summer: 73 to 79°F; The load calculations are usually based at 75oF dry bulb temperatures & 50% relative humidity

2) Winter: 70 to 72°F dry bulb temperatures, 20 - 30 % relative humidity

The standards were developed for mechanically conditioned buildings typically having overhead air distribution systems designed to maintain uniform temperature and ventilation conditions throughout the occupied space. The *Psychrometric* chapter of the *Fundamentals Handbook* (Chapter 6, 2001) provides more details on this aspect.

Outdoor design conditions are determined from published data for the specific location, based on weather bureau or airport records. Basic climatic and HVAC "design condition" data can be obtained from ASHRAE handbook, which provides climatic conditions for 1459 locations in the United States, Canada and around the world. The information includes values of dry-bulb, wet-bulb and dew-point temperature and wind speed with direction on percentage occurrence basis.

Design conditions for the United States appear in Table 1a and 1b, for Canada in Tables 2a and 2b, and the international locations in Tables 3a and 3b of 1997, ASHRAE fundamentals handbook chapter 26.

The information provided in table 1a, 2a and 3a are for heating design conditions that include:

1) Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,

2) Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence,

3) Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and

4) Average of annual extreme maximum and minimum dry-bulb temperatures and standard deviations.

1) Dry bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm). These conditions appear in sets of dry bulb (DB) temperature and the mean coincident wet bulb (MWB) temperature since both values are needed to determine the sensible and latent (dehumidification) loads in the cooling mode.

2) Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature

3) Dew-point temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and humidity ratio (calculated for the dew-point temperature at the standard atmospheric pressure at the elevation of the station).

4) Mean daily range (DR) of the dry bulb temperature, which is the mean of the temperature difference between daily maximum and minimum temperatures for the warmest month (highest average dry-bulb temperature). These are used to correct CLTD values.

In choosing the HVAC outdoor design conditions, it is neither economical nor practical to design equipment either for the annual hottest temperature or annual minimum temperature, since the peak or the lowest temperatures might occur only for a few hours over the span of several years. Economically speaking short duration peaks above the system capacity might be tolerated at significant reductions in first cost; this is a simple risk – benefit decision for each building design. Therefore, as a practice, the 'design temperature and humidity' conditions are based on frequency of occurrence. The summer design conditions have been presented for annual percentile values of 0.4, 1 and 2% and winter month conditions are based on annual percentiles of 99.6 and 99%.

The term "design condition" refers to the %age of time in a year (8760 hours), the values of dry-bulb, dew-point and wet-bulb temperature exceed by the indicated percentage. The 0.4%, 1.0%, 2.0% and 5.0% values are exceeded on average by 35, 88, 175 and 438 hours.

The 99% and 99.6% cold values are defined in the same way but are viewed as the values for which the corresponding weather element are less than the design condition 88 and 35 hours, respectively. 99.6% value suggests that the outdoor temperature is equal to or lower than design data 0.4% of the time.

Design condition is used to calculate maximum heat gain and maximum heat loss of the building. For comfort cooling, use of the 2.5% occurrence and for heating use of 99% values is recommended.

The 2.5% design condition means that the outside summer temperature and coincident air moisture content will be exceeded only 2.5% of hours from June to September or 73 out of 2928 hours (of these summer months) or 2.5% of the time in a year, the outdoor air temperature will be above the design condition.

Interior temperatures of some bodies remain essentially uniform at all times during a heat transfer process. The temperature of such bodies are only a function of time, $T = T(t)$. The heat transfer analysis based on this idealization is called *lumped system analysis*.

Consider a body of arbitrary shape of mass m, volume V, surface area A, density ρ and specific heat C_p initially at a uniform temperature T_i.

At time $t = 0$, the body is placed into a medium at temperature T_∞ ($T_\infty > T_i$) with a heat transfer coefficient h. An energy balance of the solid for a time interval dt can be expressed as:

heat transfer into the body during dt = the increase in the energy of during dt
With $m = \rho V$ and change of variable $dT = d(T - T_\infty)$, we find:

Using above equation, we can determine the temperature *T(t)* of a body at time *t*, or

alternatively, the time *t* required for the temperature to reach a specified value *T(t)*.
Note that the temperature of a body approaches the ambient temperature T_∞

exponentially.

A large value of *b* indicates that the body will approach the environment temperature in a short time.

b is proportional to the surface area, but inversely proportional to the mass and the specific heat of the body.

The total amount of heat transfer between a body and its surroundings over a time interval is:

$Q=mC_p[T(t)-T_i]$

heat transfer into the body during *dt* = **the increase in the energy of the body during *dt***

$$h A (T_\infty - T)\, dt = m\, C_p\, dT$$

With $m = \rho V$ and change of variable $dT = d(T - T_\infty)$, we find:

$$\frac{d(T - T_\infty)}{T - T_\infty} = -\frac{hA}{\rho V C_p} dt$$

Integrating from $t = 0$ to $T = T_i$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

$$b = \frac{hA}{\rho V C_p} \quad (1/s)$$

Simplified calculation model of space air temperature distribution in under-floor air-conditioning system was developed through a number of experiments. Mixing process of space air, suggested through measurements of space air/temperature distribution, could result in simple equations which express vertical temperature profile. The vertical temperature distributions calculated using these equations and the func- tion agreed with the measured distributions with sufficient accuracies for practical application in HVAC design.

Objective

- To simulate room-wise HVAC loads with high temporal resolution.
- To develop a control strategy that ensures room temperature stays within the 23–26 °C comfort band.
- To analyze energy consumption trends and identify peak load periods.
- To reduce energy usage through optimized HVAC scheduling.

Sets Used

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i belong to I= Room of Lecture Hall in Use or under analysis j belong J = Time step under analysis from j to j+dt

Use of Ideal gas Equation

350	1.0000	1.0002	1.0004	1.0014	1.0038	1.0075	1.0121
400	1.0002	1.0012	1.0025	1.0046	1.0100	1.0159	1.0229
450	1.0003	1.0016	1.0034	1.0063	1.0133	1.0210	1.0287
500	1.0003	1.0020	1.0034	1.0074	1.0151	1.0234	1.0323

The following compressibility table shows us that compressibility factor of air at 1 bar at temperature range 350K to 500K is 1.0003 which is the operating range of temperature and pressure and compressibility factor being closer to 1 implies that ideal gas law is valid.

Modelling a Room

Needed information :

V → Volume of the Room

Tinf → Outside Temperature

t → time taken to reach from one known temperature from another(experimental)

h → coefficient of convection

Modelling effective area :

Why effective area is better than effective h :

Both come as a product so change in one translated to another, now h eff or A off also take into account uneven insulation different , structure ,conduction , radiation ,etc; One advantage Effective area has on effective h is scalability. Using equation A= c* $V^{(2/3)}$. we can scale one experimental result to multiple rooms .

Equation used to model effective area :

From lumped body analysis -> (Valid due to assumption that air is well mixed inside room)

$$A_{\text{eff}} = -\frac{\rho V c_p}{hT} \cdot \ln\left(\frac{T_f - T_\infty}{T_i - T_\infty}\right)$$

T: must be in **seconds** if h is in W/m²·K

Ti,Tf,T∞Ti,Tf,T∞: in °C or K (as long as consistent)

cp: specific heat of air (typically ≈ 1005 J/kg·K)

ρ: density of air (≈ 1.2 kg/m³)

Modelling Temperature change:

we use lumped body analysis with egen to model Temperature profile of Room

egen = q_rest**no_of_people**aij −Lij*3516

aij : indicates if room is occupied or not

using conservation of energy and simple lumped system analysis we arrive at the following intergration with step size 900 sec

$$T_{i,j+1} = T_{i,j} + \frac{900}{C_p \cdot \mathrm{vol}_i} \left(q_{\mathrm{rest}} \cdot \mathrm{size}_i \cdot a_{i,j} - \mathrm{ton} \cdot L_{i,j} - h \cdot A_i (T_{i,j} - T_{\mathrm{out},j}) \right)$$

This equation is used for modelling temperature change in a room.

Input data

- **Number of Rooms**: 5
- **Time Steps**: 96 (15-minute intervals over 24 hours)
- **Room Sizes**: [600, 200, 100, 100, 100] people
- **Room Volumes**: [5000, 1000, 500, 500, 500] m³
- **HVAC Capacities (tons)**: [66, 22, 11, 11, 11]
- **Ambient Temperature (T_out)**: 89 readings interpolated to 96 time steps
- **Comfort Range**: 22°C to 26°C
- **Initial Indoor Temperature**: 35°C
- **Heat Gain per Person**: 80 W
- **Convection Coefficient (h)**: 20 W/m²K
- **Specific Heat Capacity (Cp)**: 1200 J/m³K
- **Cooling Power per Ton**: 3516 W
- **a[i,j]**: Binary schedule matrix indicating whether class is ongoing in room i at time j
- **Tout[j]: T**

Variables

- **T[i,j]**: Temperature in room i at time j
- **L[i,j]**: Integer load (in tons) assigned to HVAC in room i at time j
- **O[j]**: Binary variable indicating whether HVAC is ON at time j

Constraints

- **Temperature Evolution**:

The temperature change is governed by heat gained from occupants, cooling load, and heat lost through convection:

$$T_{i,j+1} = T_{i,j} + \frac{900}{C_p \cdot \mathrm{vol}_i} \left(q_{\mathrm{rest}} \cdot \mathrm{size}_i \cdot a_{i,j} - \mathrm{ton} \cdot L_{i,j} - h \cdot A_i (T_{i,j} - T_{\mathrm{out},j}) \right)$$

- **Thermal Comfort**:

This ensures temperature remains between 22°C and 26°C when class is ongoing.

$$a_{i,j} \cdot 22 \leq T_{i,j} \leq 26 + (1 - a_{i,j}) \cdot 100$$

- **Load Constraints**:

$$0 \leq L_{i,j} \leq \mathrm{AC}_i \cdot O_j$$

- **Initial Condition**:

$$T_{i,0} = 35$$

Modelling HVAC

Heating, Ventilation, and Air Conditioning (HVAC) systems are critical for maintaining indoor thermal comfort. In large buildings like lecture halls, HVAC systems consume significant energy. Mathematical modelling helps simulate and optimize HVAC operation under dynamic external and internal conditions.

Objective Function

To minimize total energy usage while maintaining comfort:

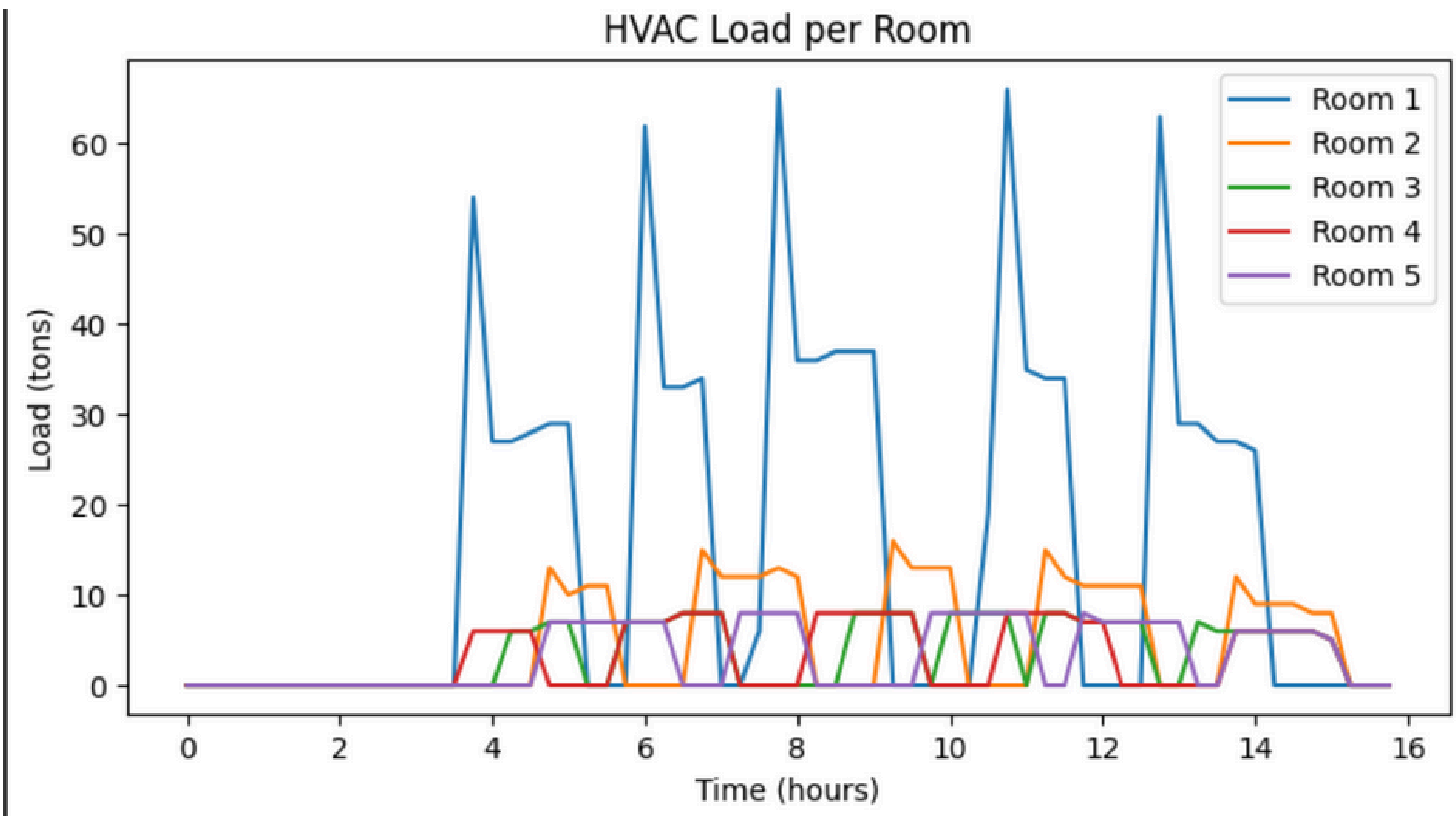
$$\min \sum_j \left[O_j \cdot \mathrm{FixedCost} + \sum_i L_{i,j} \cdot \mathrm{VariableCost} \right]$$

This encourages:

- Turning off HVAC when not needed
- Assigning minimal effective cooling load during class

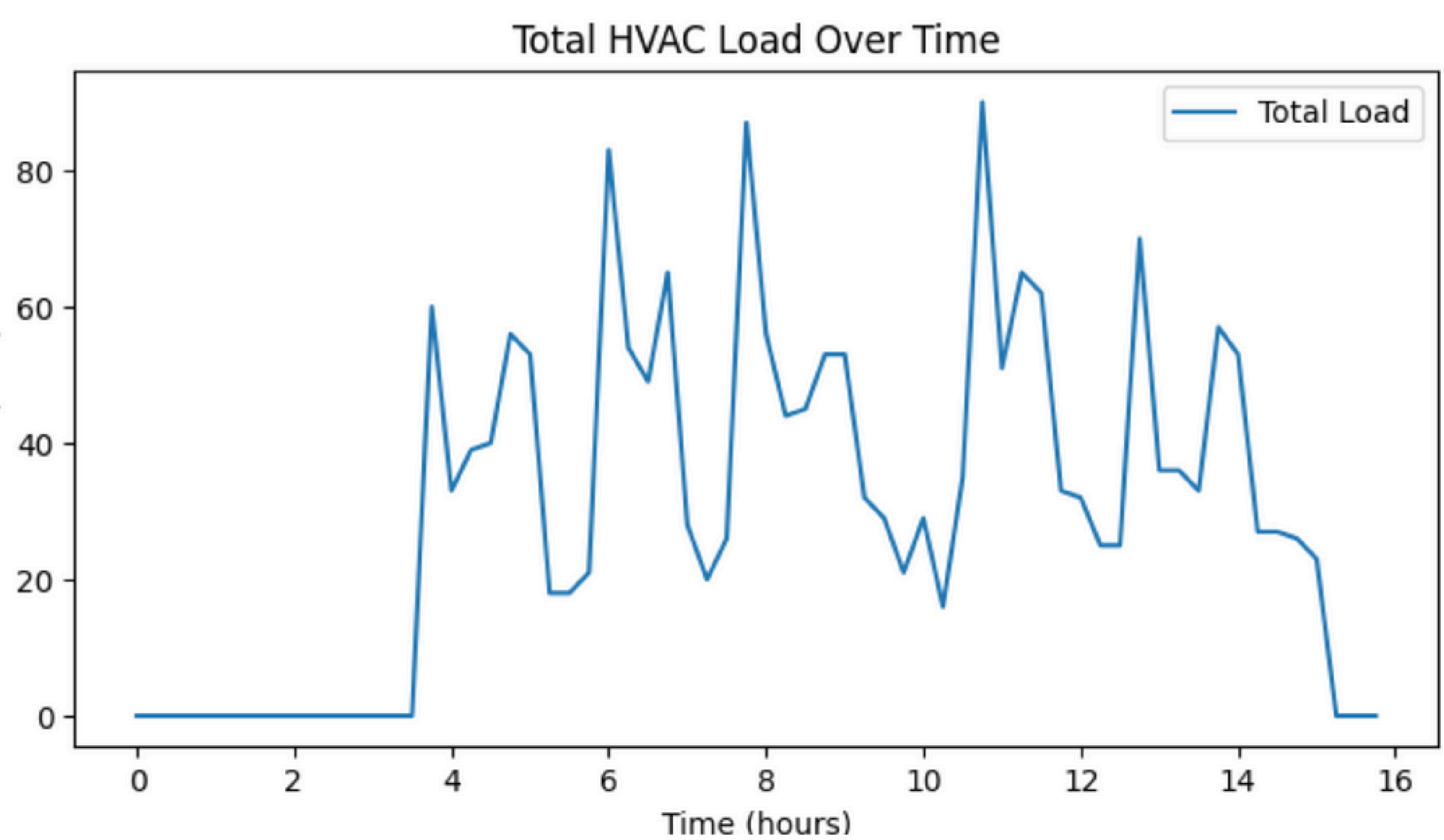
In the model used Fixed cost is 0.2 of total HVAC Costa full load , while Variable is rest 0.8 * Load Utilised / Total Capacity

Results



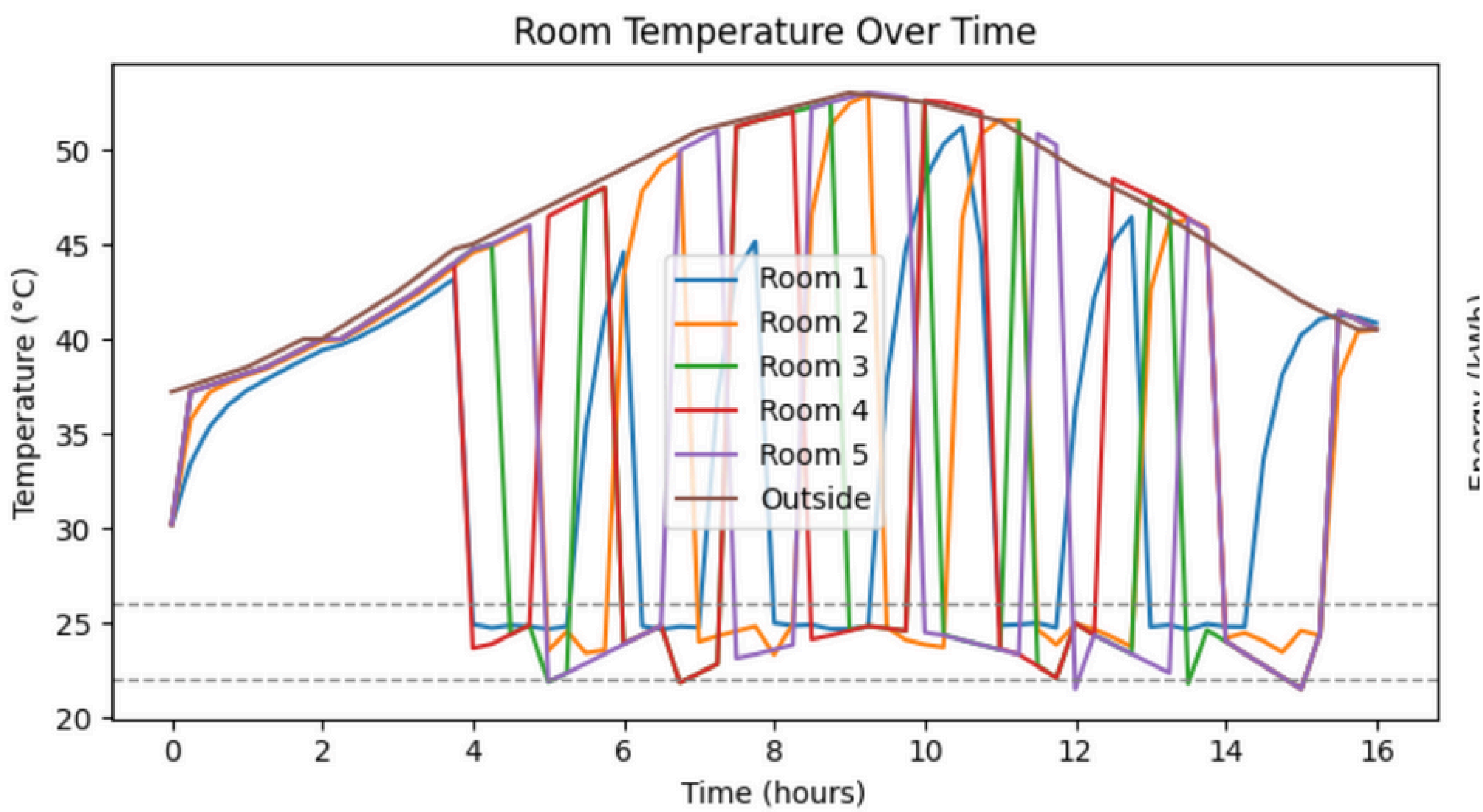
Analysis of load distribution

- **Room 1** dominates the HVAC load with **sharp spikes** reaching **60–70 tons**, indicating high cooling demand during specific periods.
- **Room 2** has consistent medium loads (up to ~15 tons) spread intermittently.
- **Rooms 3 to 5** exhibit **low, periodic loads**, peaking around **10 tons**, likely corresponding to their occupancy schedule.



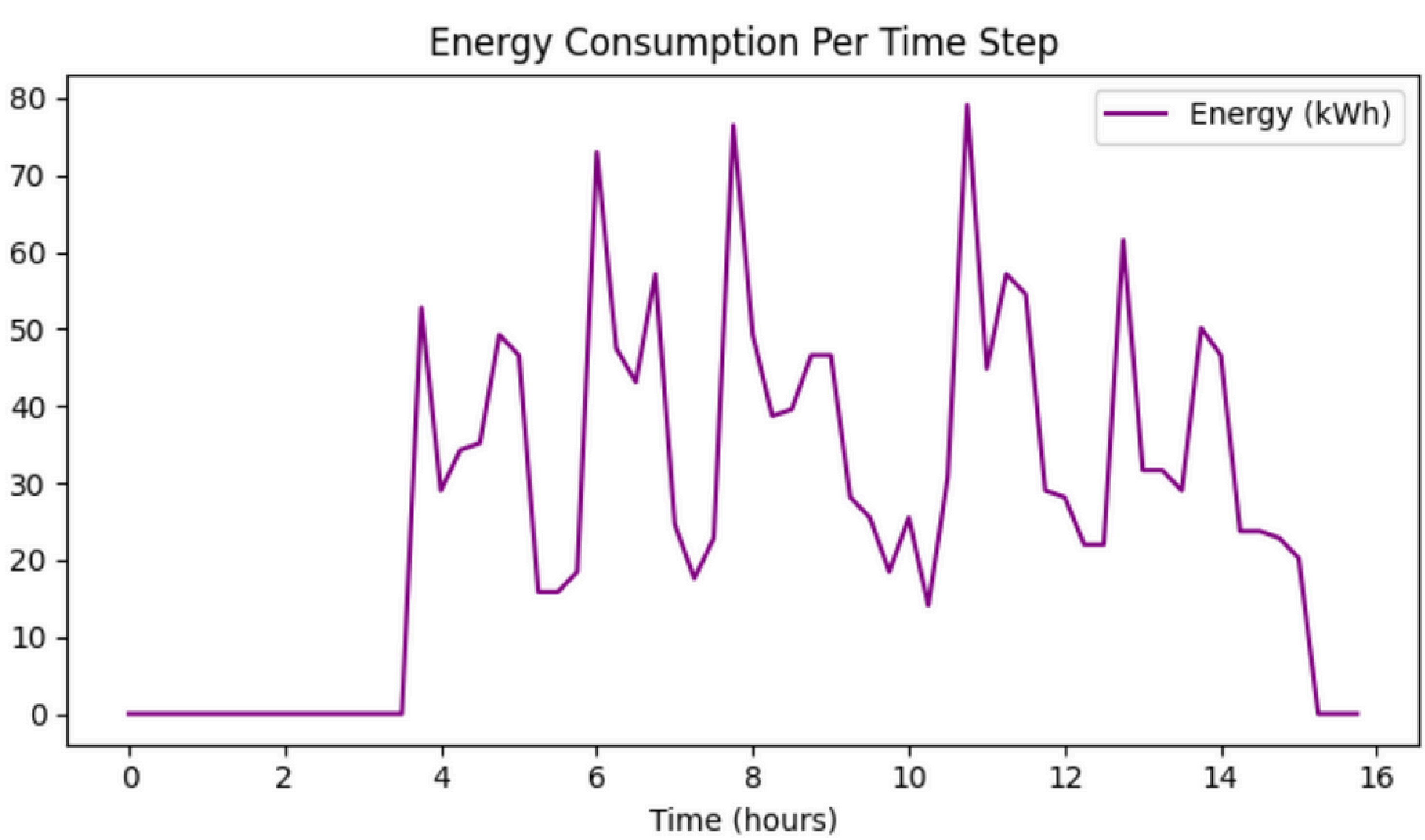
Analysis of total load

- The total HVAC load **peaks above 90 tons**, with frequent fluctuations.
- Load sharply increases around **3.5 to 4 hours** and persists till ~15 hours.
- There are **periods of rapid variation**, indicating load toggling likely due to ON/OFF HVAC switching based on room conditions.



Temperature Profile Analysis

- The **outside temperature** follows a natural diurnal profile, peaking near **52–54°C** at midday.
- Room temperatures rise quickly when HVAC is off and drop steeply when it turns on, indicating **aggressive cooling**when active.
- **Control bands at ~23°C and 26°C** indicate a **desired temperature range**.



Energy Consumption

- Energy consumption aligns well with **total HVAC load**: both spike and dip in similar patterns.
- Max consumption reaches **~80 kWh per time step**.
- There is **no consumption during initial and final hours**, aligning with zero HVAC load.

Future Improvements

Reduction of time intervals for better integration

Scaling up the system to rooms such that scheduling of room do not coincide

Better data for effective area

Improved Integration methods such as RK4

Individual h and effective area for each rooms

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