ADAPTIVE THERMAL COMFORT - AN EXTENDED APPROACH FOR CONDITIONED INDOOR ENVIRONMENT USING COMPUTATIONAL FLUID DYNAMICS

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ABSTRACT

This article leaps a step forth extending the limited applicability of Adaptive Comfort Standard (ACS) ahead from naturally ventilated buildings to conditioned indoor environment, moving ahead of the conventional method of system sizing in India by air temperature and relative humidity alone. Application of ACS is been analyzed and compared for its potential of conditioning system downsizing by adopting the upper limit of comfort band (derived using Running Mean Temperature as basis) and adopting it as set temperature range (as resultant air temperature considered for sizing calculations). Parallel to the conventional analysis methodology, this article highlights an analysis methodology of adopting Computational Fluid Dynamics (CFD) as simulation technique to investigate the fluid flow patterns and behavior of thermal comfort indices in conditioned indoor environment. Computational fluid dynamics (CFD), as the most sophisticated airflow modelling method, can simultaneously predict airflow and heat transfer inside buildings, thereby keeping the emphasis of verification and validation of the study extensively dependent on the outputs from CFD analysis. With effective optimization of Mean Radiant temperature & design of conventional air conditioning system to the present scenario of buildings in India, there are chances to save maximum 20 % in the Energy bill. So there is huge potential, which varies between 40 – 60% saving potential in Energy consumption, if there is combined approach of ACS and low energy cooling techniques in India

1. INTRODUCTION

In a typical office space, centralized air-conditioning system provides cooling to the conditioned space. Irrespective of the orientation and climatic condition, the cooling set point temperature for a conventional conditioned office space is assigned as 23±2 °C. Widely in India, this temperature is the basis for cooling load calculation, system sizing and thermostat controls. The air-conditioning system is sized to meet the set point temperature of space by offsetting all heat load gains to the space. The overall air-conditioning system load is calculated using

$$Q_{system} = m*C_p * \Delta T \quad (1)$$

Where,

m = air flow rate from the system (m^3/s)

C_p = specific heat capacity of air (kJ/kgK)

$\Delta T = T_{set} - T_{supply}$

T_{set} = set point temperature (23 ± 2°C)

T_{supply} = supply temperature from the system (°C)

Q_{system} = air-conditioning system capacity (kW)

An effective system design means the system maintains the set point temperature inside the conditioned space after offsetting all the heat gains associated with the space. The heat loads associated with cooling requirement is the heat gain through conduction, convection, radiation, metabolic rate (human occupancy & work type), equipment load (inclusive of lighting load).

$$Q_{loads} = Q_{conduction} + Q_{convection} + Q_{radiation} + Q_{metabolic} + Q_{equipment} \quad (2)$$

Where,

$Q_{conduction}$ = heat gain through conduction (W)

$Q_{convection}$ = heat gain through convection (W)

$Q_{radiation}$ = heat gain through radiation (W)

$Q_{metabolic}$ = heat gain through metabolic rate of humans (W)
$Q_{equipment}$=heat gain through internal equipment's inclusive of lighting (W)
Q_{loads} = total cooling load of the space (W)

The constraint in the context of using 23 °C as set point temperature is that there is very little interaction with the daily outside environment (i.e.) the outside dry bulb temperature. In Indian climate condition, temperature of 23 °C is attained very rarely during the course of entire year. The average monthly dry bulb temperature indicated in table 1 gives a fair picture of the ΔT i.e. difference between the indoor set point temperature and the outside dry bulb temperature. Specifying a broader temperature band varying on the basis of outdoor air temperature suited to the Indian context has the potential to reduce the use of energy intensive space cooling for Indian buildings.

Table 1. Average monthly outdoor air temperature

Month	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**
DBT (°C)	24.2	25.8	28.2	30.1	31.5	31.1
ΔT	1.2	2.8	5.2	7.1	8.5	8.1
Month	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**
DBT (°C)	30.3	29.4	29.2	27.7	**25.7**	24.9
ΔT	7.3	6.4	6.2	4.7	**2.7**	1.9

There are two primary impacts of lower cooling set point temperature in conditioned space

a) Higher the difference between the cooling set point temperature and the outdoor dry bulb temperature, higher is the chance for thermal shock.

b) The sizing of the air-conditioning system is considerably higher.

Optimizing the cooling set point temperature will increase comfort band for occupants; get accustomed to variable indoor thermal conditions reflective of daily and seasonal climate changes and also provide an opportunity to optimize system design. This optimization is possible by having an interactive cooling set point temperature (i.e.) based upon the history of outdoor air temperature; the indoor set point temperature is varied. For the basis of study the impact of increasing the cooling set point temperature of room on occupant comfort & energy demand is analyzed in detail for two options

Option 1 – Determining cooling set point based on average outdoor DBT for monsoon & winter season.

Option 2 - Determining cooling set point temperature based on EN15251 code i.e. adaptive thermal comfort.

Option 1 keeps tab on the average monthly outdoor dry bulb temperatures. From table 1, it is observed during the monsoon & winter season ranging from October to February the outside average DBT distribution is within 24 to 27°C and this is a general representative range of the comfortable lower band of temperature. Option 1 explores the extent of cooling energy optimization for the entire year by increasing the cooling set point temperature. Fig 1 captures the list of temperatures indicated as case 1, 2, 3 for analysis.

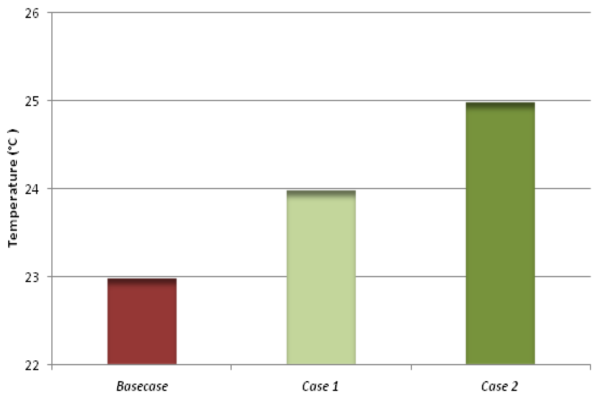


Fig. 1. Cooling set point temperature

In option 2, EN15251 code is the basis for determining the cooling set point temperature. Derived from a European comfort base, EN15251 code, calculates the indoor comfortable temperature based on the running mean of daily outdoor temperature for an entire week. The outdoor air temperature for each day is allotted a specific weightage and the calculated sum of weighted daily dry bulb temperature for the last 7 days is defined as the running mean temperature as indicated in equation (3)

$$T_{rm7} = (T_{-1} + (0.8*T_{-2}) + (0.6*T_{-3}) + (0.5*T_{-4}) + (0.4*T_{-5}) + (0.3*T_{-6}) + (0.2*T_{-7}))$$ (3)

Where,

T_{-n} = average outdoor temperature n days before the day in question.

$T_{setpoint} = 18.8 + (0.33* T_{rm7})$

Where,

$T_{setpoint}$ = cooling set point temperature

T_{rm7} = calculated running mean temperature for the last 7 days.

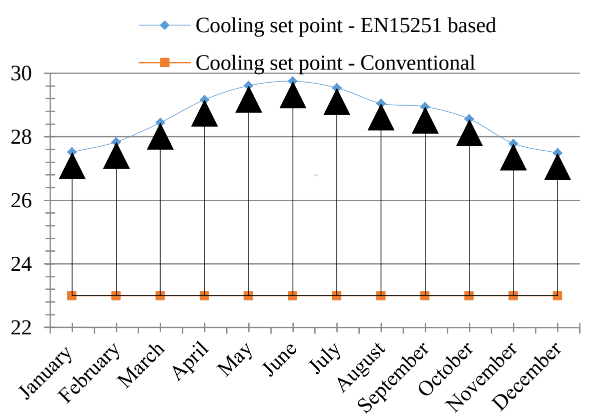
Taking into account the daily outdoor air temperatures of entire year, table 2 represents the average monthly cooling set point temperature as per EN15251.

Table 2. Set point temperature based on en15251

Month	Average outdoor DBT (°C)	Cooling set point temperature (°C) – EN15251
January	32.9	27.52
February	34.8	27.84
March	36.1	28.45
April	37.1	29.17
May	39.5	29.61
June	38.6	29.75
July	37.1	29.54
August	36.9	29.05
September	36.4	28.95
October	34.6	28.56
November	31	27.79
December	30.8	27.49

There is a variation of minimum 4˚C between the cooling set point of 23˚C considered during conventional design and the cooling set point formulated by adopting EN1521 for all the months.

Fig. 2. Cooling set point temperature shift



A graph representing the cooling set point temperature variation between the normally designed building and an outdoor air interactive building is shown in figure 2.

2. METHODOLOGY

The study aims to understand the effect of cooling set point temperature which is revised as per two options on the occupant comfort, system sizing and the overall energy consumption.

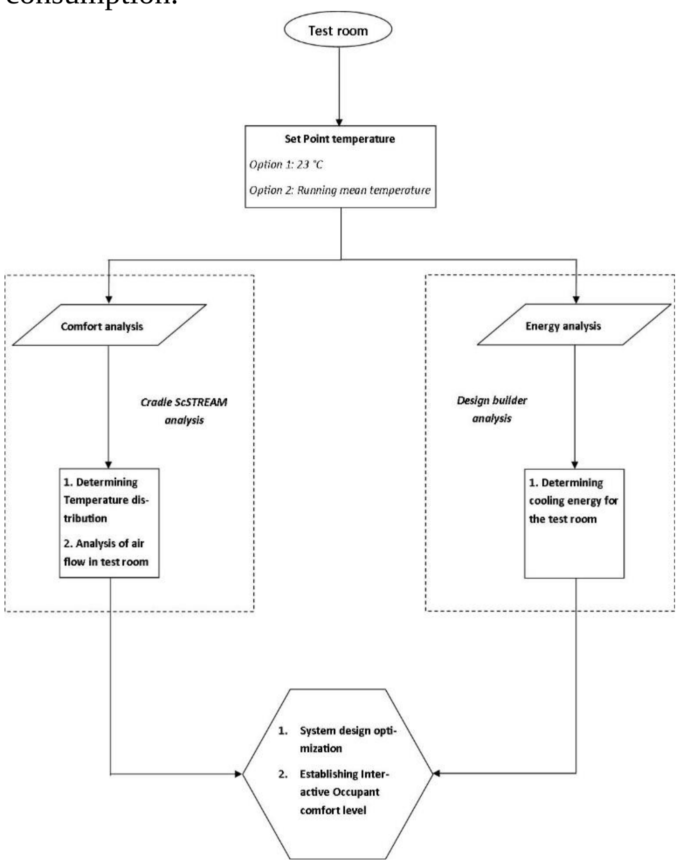


Fig. 3. Methodology of study

Figure 3 represents the analysis methodology of the study. Two modelling & simulation software's namely cradle scSTREAM and Design builder is used for the comfort & energy analysis study respectively. scSTREAM is mainly used to understand the effective temperature field distribution and air flow analysis over the entire test room (termed as comfort analysis in figure 3); whereas Design builder calculates the cooling energy consumption for each month of the year for varying cooling set point temperature (termed as energy analysis in figure 3);. The cooling set point temperature is representative of three options; one being the set point temperature of 23˚C (conventional design condition in use in India) which will be the baseline case for comparative study, the second being the option 1 elaborately detailed in figure 1 and the last being cooling set point temperature as per EN15251 code based on the running mean temperature (reflective of temperature variation in the outside environment).

3. COMFORT ANALYSIS

For this study, a test air-conditioned room is considered for flow and temperature analysis in cradle scSTREAM. The test room is modelled in the preprocessor plug-in of the software using the following design parameters listed in table 3. The air inside the conditioned test room (i.e.) the arbitrary shape of the conditioned test room is defined as computational domain and using various time based solver techniques temperature field in the test room is attained.

Figure 4 is the plan representative of test room model. The study focusses on varying the test room set point temperature which invariably is achieved by varying the supply air temperature and understand the occupant comfort index.

Table 3. Design inputs

Description	Units	Measured values
Type of occupancy	No units	Office (5 days x 8 hours)
Number of occupants	No units	15
Number of inlet supply air grills	No units	8 no's
Height of the room	m	3
Glazing area	sq. m	27.466

The supply air temperature to the test room under normal design circumstances is around 14-16°C to ensure room cooling set point temperature of 23°C is achieved. ASHRAE defines a temperature difference of 11°C between the supply air to the conditioned room air set point (refer Appendix G3.1.2.8 in ASHRAE 90.1.2007). Based on this ASHRAE design air flow approach, the supply air flow rate temperature will be varied for the cooling set point temperatures specified in figure 1 and table 2. Cradle scSTREAM simulates the air velocity and temperature distribution inside the test space for various supply air temperatures considering heat gains as per equation (2). The simulation involves three stages

a) Pre-processing phase which involves modelling and defining simulation conditions.
b) Solver phase where convergence of the temperature and velocity is analyzed by governing equations.
c) Post- processing phase in which visualization of the results within the computational domain.

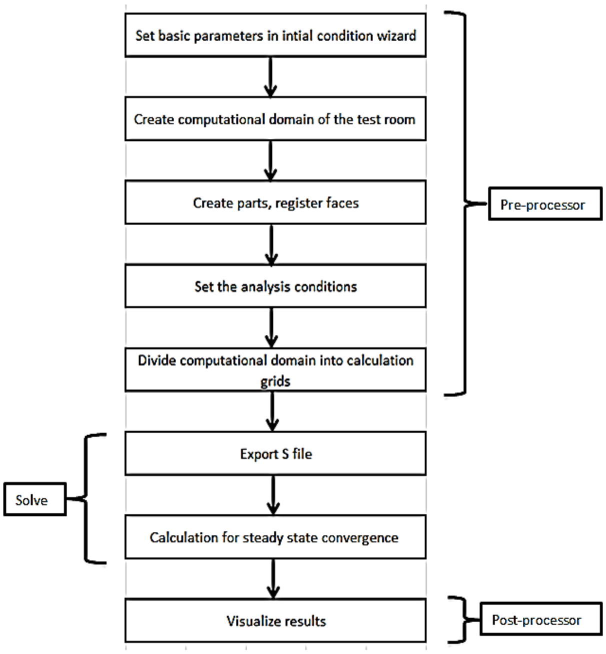


Fig. 4. Simulation methodology in sc-STREAM

The designed air velocity in the room for a cooling set point temperature of 23°C is measured using handheld anemometer and the supply air temperature is measured by non-contact thermometer. There are 8 no's of supply grills for the conditioned inlet air to enter into test room as is visible in figure 6. The measured air flow rate values highlight a considerable difference in the air flow rates at each supply grill. Measurements of air velocity is taken near to the face of supply grill. Table 4 provides an insight of air velocity through each supply grill; with grill 6 & 7 inside the test room receiving very high air velocities of 8.05 m/s and 7.70 m/s respectively.

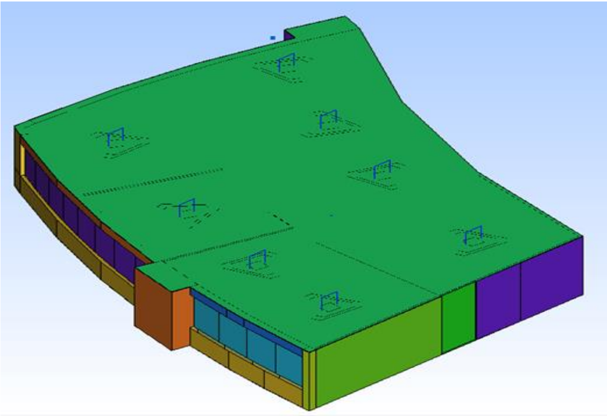


Fig. 5. 3-dimensional plan of test room

Table 4. Supply grill air velocities

Supply grill	Air velocity in m/s
Grill – 1	2.28
Grill – 2	2.01
Grill – 3	2.79
Grill – 4	4.82
Grill – 5	1.93
Grill – 6	8.05
Grill – 7	7.70
Grill – 8	4.30

Air velocities in table 4 are assigned as fixed velocity to the inlets in the model. The temperature in the conditioned air stream at the supply grill face is measured as 19 °C. Return air grill to exhaust the return air is designed in the periphery of ceiling element. The return air grill is maintained a lower pressure and to replicate the same in modelling, it is assigned a static pressure of 0 Pascal to ensure air circulation in the space. The test model seen in figure 5 is modelled as per the actual design and the properties of the envelope are initialized as per table 3. At the end of detailed conditions being applied to elements inside the computational domain, field of study is meshed using tetrahedron mesh elements to understand the variation of physical quantities in space and time. The supply air temperature from each of the 8 no's of supply grill is 19 °C for achieving the cooling set point temperature of 23°C in the test room. The temperature field distribution in the test room is attained by solving turbulence & conservation equations.

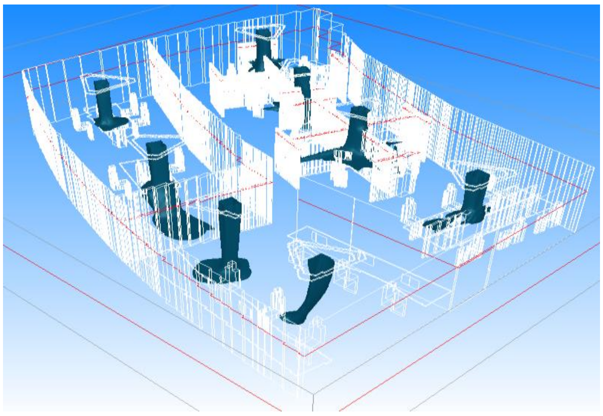


Fig. 6. Supply air temperature at 19˚C

Occupied zone is typically the volume from the floor up to a height of 6 feet (1.8 m) and 1 foot (305 mm) from the walls. Supply air draft captured in figure is supplied at 19 °C and as it enters the occupied zone, the draft temperature is around 19-20 °C (Fig 6). This supply draft offsets all the heat gains inside the test room thus maintaining the cooling set point temperature.

3.1. 23°C as cooling set point temperature

The temperature field for the test room is monitored at a height of 1.5 m from the floor level in figure i.e. in the occupied zone. Across the entire spectrum of test room at 1.5 m, the temperature is 22-23 °C.

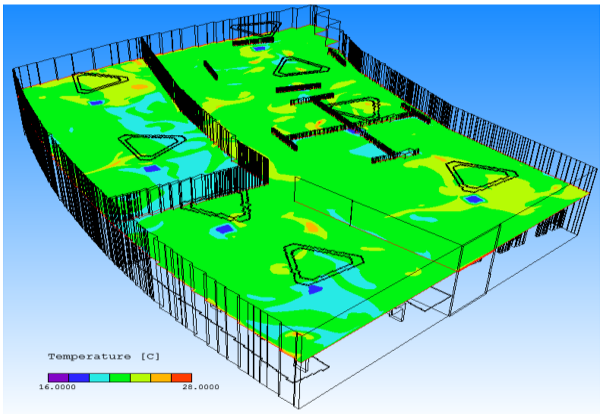


Fig. 7. Indoor air temperature along X-axis at 1.5 m

Another notable feature is the spread of temperature field (22-23°C) is even through the height of the occupied zone indicating effective draft temperature difference is within the -2°C to +1°C. The temperature of air near to the roof (limited to 500 mm from the roof) is higher as a result of conduction gain and the convection phenomena of high temperature air moving upwards.

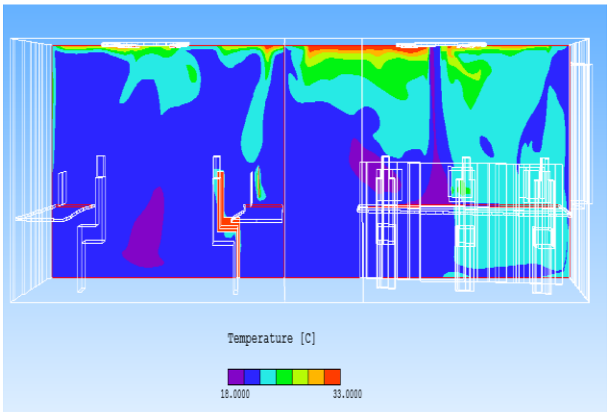


Fig. 8. Indoor air temperature along Y-axis

In addition to the slice analysis, the temperature distribution over the entire volume of the domain is also analyzed as shown in figure. The temperature distribution is analyzed for three ranges 20 to 23°C, 23 to 27°C and 27°C & above. Except for the volume nearer to the roof, temperature field is between 20 to 23°C for the test room volume. Nearer to the roof, where conduction gains are highest the temperature increases beyond 23°C. However, this temperature gradient is concentrated towards the roof are and does not spread into the occupied zone thereby limiting its impact on the comfort experienced by occupants (Fig 7 & 8).

The average temperature and the air flow velocity at a height of 1.5 m from the floor level is analyzed to check the predicted mean vote (PMV) and people dissatisfied rate (PPD) for cooling set point temperatures as per Fig 9.

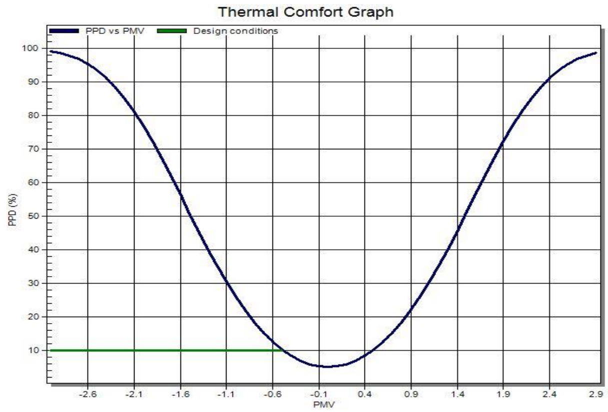


Fig. 9. PMV-PPD for 23˚C

ASHRAE defines for a comfortable environment for occupants inside a space, the seven point PMV scale should fall between +0.5 & -0.5 and the PPD index should be below 10. The PMV-PPD index for the temperature analyzed in the room is shown in Fig 9. The simulated PMV index is -0.48 and the corresponding PPD is 9.90 for the cooling set point temperature of 23°C and air flow rate of 0.32 m/s. It is observed that the PMV index is closer to the slight colder scale (-0.5) encouraging the scope of optimization of cooling set point temperature to a higher scale.

3.2. 24°C as cooling set point temperature

Figure 10 & 11 is indicative of the temperature and air velocity field of distribution at a height of 1.5 m from the floor level of test room.

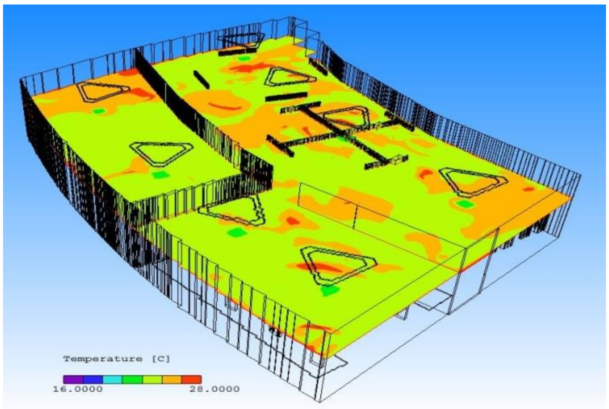


Fig. 10. Air temperature field along X-axis

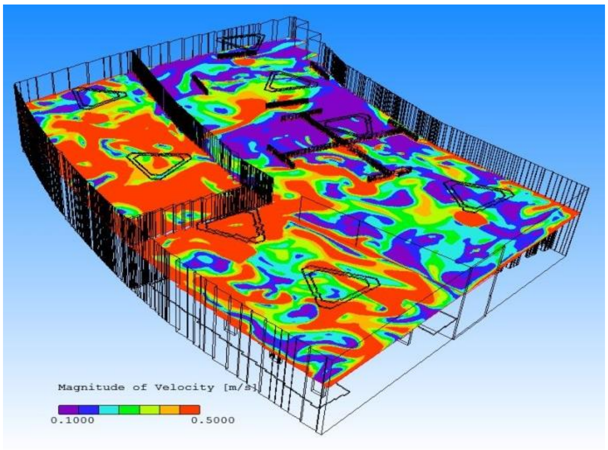


Fig. 11. Velocity field along X-axis

The vertical air temperature field at an instantaneous point is shown in figure 15 and it indicates uniform temperature distribution & the absence of cold pockets. The simulated PMV index is -0.20 and the corresponding PPD is 5.81 (refer Fig 13) for the cooling set point temperature of 23°C and air flow rate of 0.34 m/s.

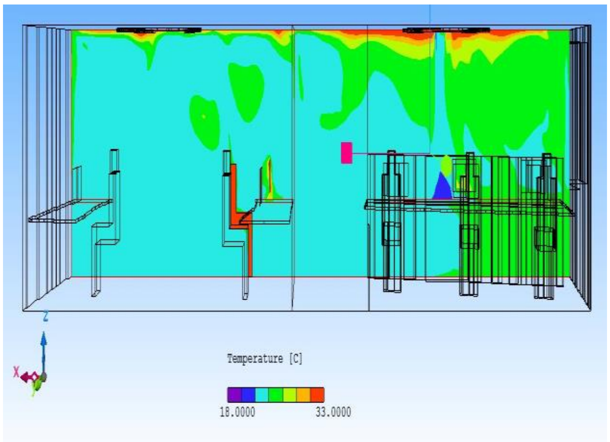


Fig. 12. Air temperature field along Y-axis

The occupant comfort has moved to a higher comfort band with 1°C rise in the cooling set point temperature.

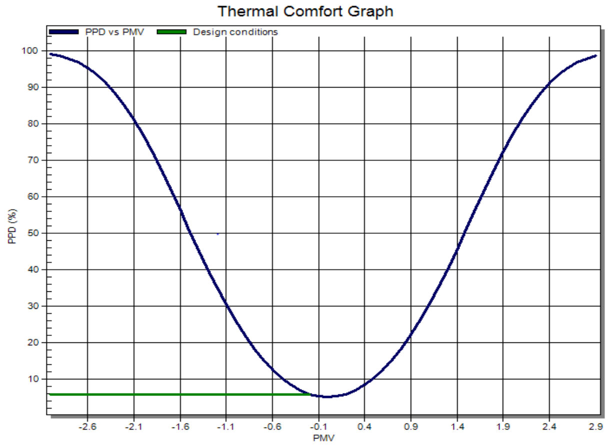


Fig. 13. PMV-PPD for 24˚C

3.3. 25°C as cooling set point temperature

The temperature and air velocity field at a height of 1.5 m from the floor level i.e. within the occupied zone highlights an average temperature value of 25°C and 0.33 m/s respectively. The temperature distribution is even in this case also along the height of room however, there is a marked increase in the temperature range in comparison to the baseline case of 23 °C (Fig 14-16).

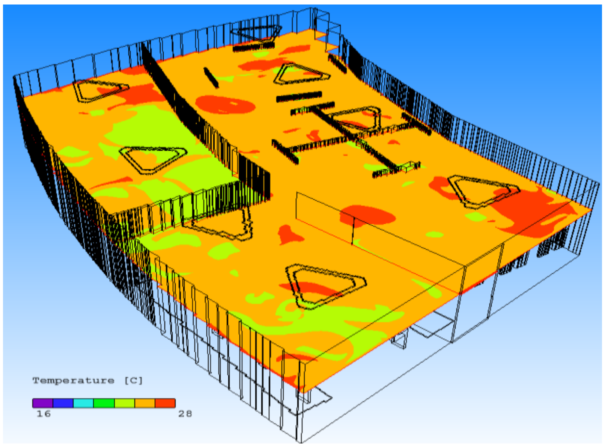


Fig. 14. Air temperature field along X-axis

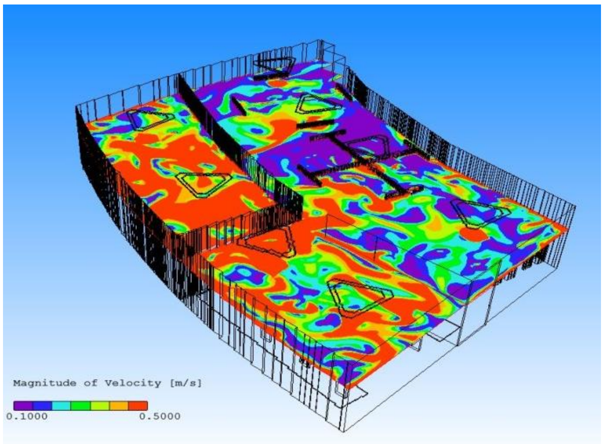


Fig. 15. Velocity field along X-axis

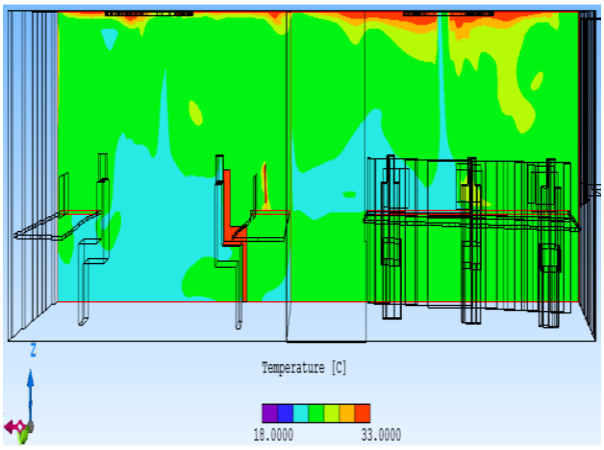


Fig. 16. Air temperature along Y-axis

The PMV-PPD graph for this temperature range & air velocity is simulated and the corresponding PMV index is 0.14 i.e. nearly neutral & PPD rate is 5.39. The PMV index indicates 25°C as the cooling set point temperature range is highly comfortable than the baseline case of 23°C as the index is near to 0.

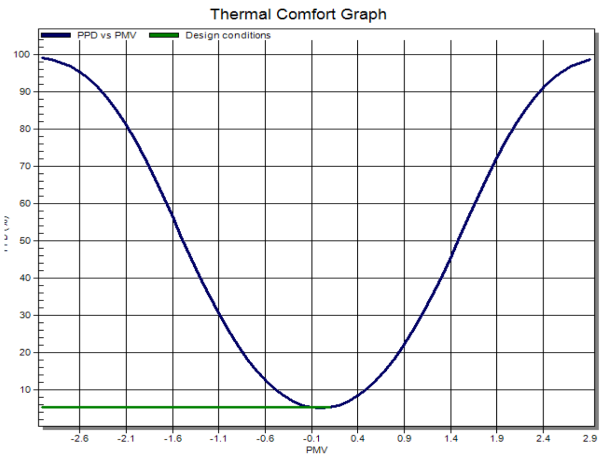


Fig. 17. PMV-PPD for 25˚C

The PMV-PPD index in accordance with ASHRAE 55 is analyzed for temperatures indicated in Fig 17 to understand its effect ion occupants whereas the cooling set point temperature listed in table 2 for each month is the acceptable adaptive comfort temperature as per EN15251 standard.

Table 5. Comfort analysis - summary

Scenario	**PMV Value**	**PPD Index**	**Remark**
OPTION – 1 (As per ASHRAE 55)			
BASE (23°C)	- 0.48	9.90	Shifting to cool
CASE 1 - 24°C	- 0.20	5.81	Nearly Neutral
CASE 2 - 25°C	+ 0.14	5.39	Nearly neutral
CASE 3 – 26°C	+ 0.49*	9.97*	Shifting to hot
OPTION – 2 (Adaptive thermal comfort)			
Cooling set point for each month is the actual comfortable temperature as per adaptive code EN15251			

4. ENERGY ANALYSIS

The intent of modelling the test room in design builder is to simulate the energy consumption patterns for the entire year considering different cooling set point temperature as shown in Fig 18 &19. In addition to the comfort analysis, for an informed and efficient design energy optimization analysis is required as it provides direct benefit in terms of reducing billed energy consumption. The overall energy consumption for option 1 i.e. with set point temperatures of 23, 24 & 25°C is shown in figure and

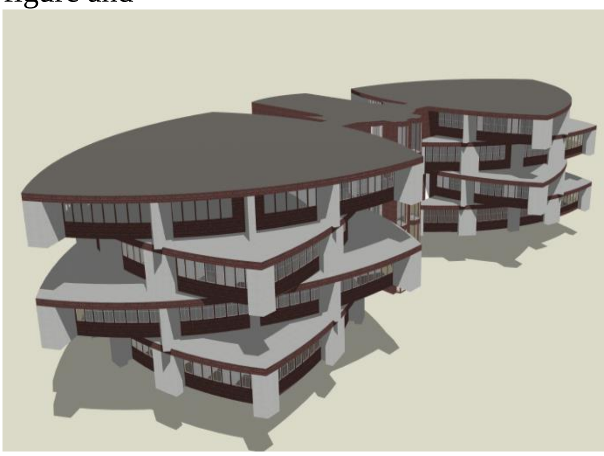


Fig. 18. 3-dimensional model of building

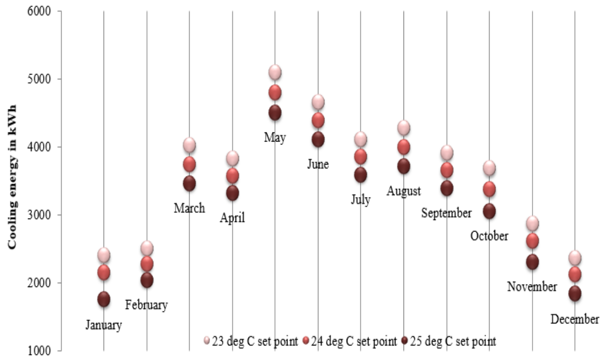


Fig. 19. Cooling energy comparison

Cooling set point temperature of 23°C is considered as the baseline for cooling energy consumption. The cooling energy consumption is highest during the month of May i.e. peak summer condition in Chennai climatic zone and more or less similar cooling consumption pattern (3700 kWh to 4600 kWh) is observed between the months of March to April & July to October. From November to February the cooling energy consumption is comparatively on the lower range i.e. 2400 kWh to 2500 kWh, as this period of year is peak winter thereby negating any requirement for higher cooling requirement by lower conduction, convection and radiation gains through the building envelope & environment.

Similar patterns emerge when the cooling set point temperature of test room is varied to 24 & 25°C. The variation in the cooling energy consumption between the baseline cooling set point temperature of 23°C and case 1 i.e. 24°C is in the range of 250 kWh to 300 kWh translating to an prospective energy saving percentage of 5% to 7% during the months of March to October and 8% to 10% during the months of November to February.

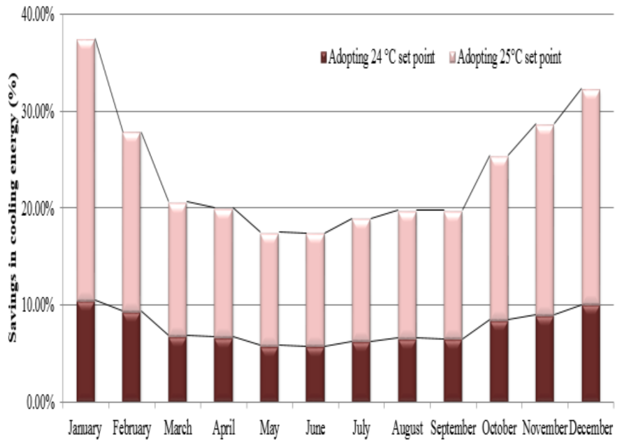


Fig. 20. Percentage savings in cooling energy

Cooling set temp of 23°C is considered as the baseline for cooling energy consumption. The cooling energy consumption is highest during the month of May i.e. peak summer condition in Chennai climatic zone. When the cooling set point temperature is varied by 2°C i.e. shifted from 23°C to 25°C, the band of attainable energy savings is much wider (Fig 20).

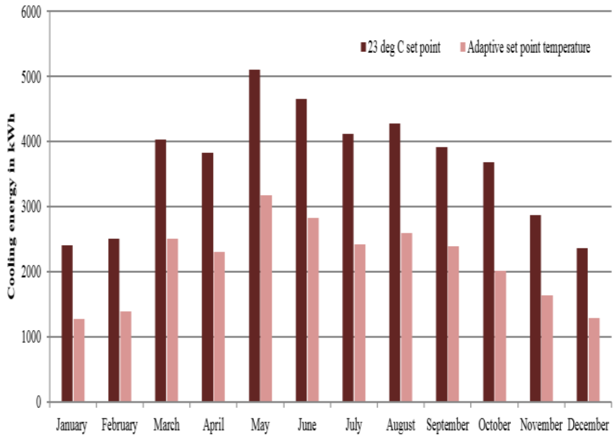


Fig. 21. Energy reduction by using adaptive set point

From the figure 28, energy savings range for 25°C cooling set point temperature is observed to be 500 kWh to 750 kWh which translates to a prospective energy saving percentage range of 11% to 13% during March to October and 19% to 27% during November to February.

When the baseline set point temperature i.e. 23°C is varied by adopting adaptive thermal comfort set point temperature, there is a huge savings in cooling energy i.e. 37% to 45% over the baseline (Fig 21).

Table 6. Cooling energy - summary

Month	CASE 1 - 24°C	CASE 2 - 25°C	CASE 3 - 26°C	Adaptive set point
Jan	10%	26%	30%	47%
Feb	9%	18%	22%	44%
Mar	6%	13%	17%	37%
Apr	6%	13%	17%	39%
May	5%	11%	15%	37%
June	5%	11%	15%	39%
July	6%	12%	16%	41%
Aug	6%	13%	17%	39%
Sep	6%	13%	17%	39%
Oct	8%	16%	20%	45%
Nov	8%	19%	23%	43%
Dec	10%	22%	25%	45%

The variation is energy savings is quantified in table 6

- For 24°C – the cooling energy variation over the baseline is in the range of 5%-6% during summer and 8%-10% during winter
- For 25°C – the cooling energy variation over the baseline is in the range of 11%-13% during summer and 16%-26% during winter
- For 26°C – the cooling energy variation over the baseline is in the range of 15%-17% during summer and 20%-30% during winter
- For Adaptive cooling set point – the cooling energy variation over the baseline is in the range of 37%-41% during summer and 44%-47% during winter

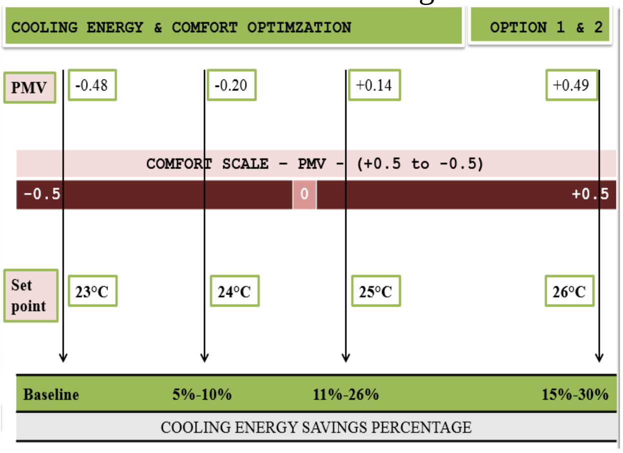


Fig. 22. Energy & comfort scale

Set temperature of 26°C provides comfort as per ASHRAE 55 and also the overall energy savings is higher compared to other cases. For Adaptive cooling set point – the cooling energy variation over the baseline is in the range of 37%-41% during summer and 44%-47% during winter. The cooling set point itself is a comfortable temperature for adaptive option. There is an increased cooling energy saving percentage without compromising the comfort in Adaptive cooling set point (Fig 22).

5. CONCLUSION

Adopting set point temperature of 26˚C or adaptive cooling set point will not affect the occupant comfort drastically rather it will ensure comfort is within prescribed limits and also energy being saving saved in the case of existing buildings. Whereas in the case of new buildings in Indian conditions, the cooling systems design can be optimized hence saving initial cost in addition to offsetting the energy costs.

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