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Space conditioning using evaporative cooling for summers in Delhi

Krishan Kant, Ashvini Kumar [1], S.C. Mullick*

Centre for Energy Studies, Indian Institute of Technology New Delhi 110 016, India

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Abstract

The article examines the possibility of space conditioning the interiors of a multistorey office building in Delhi using evaporative cooling in the summer months of April, May and June. The temperature and humidity conditions obtained in a room of the building with direct evaporative cooling are studied by simulation. In this case study, the room is assumed to have a south-facing wall with a window and all other walls, ceiling and floor are interior partitions. The effect of number of air-changes per hour (ACH) from 1 to 40 and fresh-air bypass factor (BPF) 0% to 100% on performance is studied by simulation. The aim is to find whether some combination of ACH and BPF succeeds in keeping room conditions below 80% RH and temperatures between 27 and 31°C, depending on RH. It is found that the desired results are achieved by keeping the ACH and the BPF within certain limits depending on weather conditions. If the temperature and relative humidity of the ambient air are too high then a direct evaporative cooler cannot achieve comfort in the room. Appropriate combinations of ACH and BPF have to be selected to obtain the best results.

1. Introduction

Evaporative air-cooling is an easily available method of achieving a comfortable indoor climate, especially in the arid regions of the world. It is economical, energy-efficient and pollution free. With the onset of the energy crisis and ecological problems posed by halogen-based air conditioners, the importance of evaporative cooling has grown further. However, it has certain limitations and its performance has to be rigorously analyzed.

Mathews et al. [1] have brought out many positive features of evaporative cooling. They have developed an improved, user-friendly computer program called EASY for computing the indoor conditions of a building with evaporative cooling. They have hence studied the possibility of achieving comfort in a well-designed building using direct or regenerative evaporative cooling in various parts of South Africa. The results of the computer simulation have also been validated experimentally.

Reference [1] provides a good description of evaporative cooling. Air can be cooled and under ideal conditions saturated with water vapor by passing it over an appropriate wetted surface. If there is no heat transfer from the surroundings, the process is adiabatic, i.e., air loses a certain amount of sensible heat but gains an equal amount of latent heat of water vapor.

Fig. 1 represents an ideal adiabatic saturator, which is completely insulated from the surroundings. Outdoor air at temperature T_1 and specific humidity W_1 enters the saturator and leaves it at saturation conditions T_2 and W_2. In the mixing chamber 3, the saturated air mixes with a stream of bypassed air which is still at the outdoor conditions T_1 and W_1. The temperature and humidity of the resulting mixture depend on air bypassed (BPF = x%), i.e.,

* Corresponding author. Tel. : +91-11-659-1254; fax: +91-11-686-2208.

E-mail address: smullick@ces.iitd.ernet.in (S.C. Mullick).

[1] Director, Solar Energy Centre, Ministry of Non-Conventional Energy Sources, New Delhi.

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Nomenclature

ACH number of air changes per hour (ac/h)
BPF bypass factor (%)
DBT dry bulb temperature (°C)
RH relative humidity (%)

$$T_3 = \frac{\{xT_1 + (100 - x)T_2\}}{100}$$

and

$$W_3 = \frac{\{xW_1 + (100 - x)W_2\}}{100}$$

This is the condition of the supply air in case of a direct evaporative cooler. It is supplied to the room where it takes sensible and latent loads, thereby raising its temperature and specific humidity to T_4 and W_4 as shown in the psychrometric chart in Fig. 1(a).

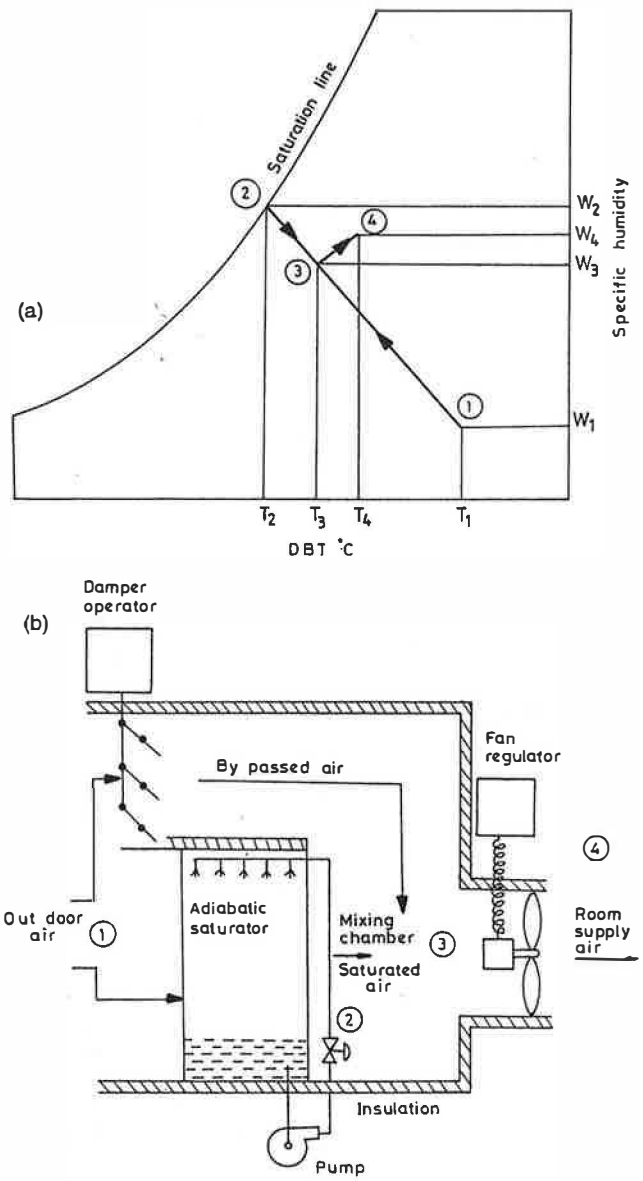

Fig. 1. (a and b) Direct evaporative cooling process.

A non-ideal direct evaporative cooler may be represented as a combination of an ideal saturator and a bypass system. It may have heat gains from the surroundings, the fan and pump motors etc. The bypass factor of the practical cooler will then represent its actual performance as a saturator. The heat gained by it will be added to the room heat load.

The conventional 'desert cooler', i.e., a direct evaporative cooler, can provide adiabatic cooling but beyond a certain limit it renders the air uncomfortable for the human body. Besides, the consumption of electrical energy and water by the increasing number of coolers puts considerable strain on the already depleted resources in a country like ours. It is imperative, therefore, to conduct appropriate studies on more effective utilization of evaporative coolers.

The problem of space conditioning with ventilation-air or evaporatively cooled air is associated with time-varying indoor conditions: the indoor condition is strongly dependent on outdoor temperature and humidity. In the present work, TRNSYS [2], a computer program for transient simulation, has been used to find the thermal response of the room.

2. The present study

In the present study, computer simulations have been carried out for Delhi for the months of April, May and June (In July and August, the monsoon is active and the humidity is high. People not using vapor-compression type air-conditioners must depend on ventilation, especially on rainy days. Evaporative cooling is not possible). Weather data has been obtained from the handbook by Mani [3]. The handbook provides mean hourly values of solar radiation, ambient air temperature, relative humidity and wind speeds based on ten years of data.

For the purpose of this case study, a thermally efficient typical room, 4 m × 4 m and 3.6 m high consisting of insulated walls, ceiling and floor as specified by Wall Number 32, Table 29, Chapter 26 of the ASHRAE Handbook of Fundamentals 1981 [4], has been chosen as a repetitive unit in a multistorey office building. The walls consist of 101.6 mm concrete with 50.8 mm insulation and 12.7 mm finish. The U value of the

structure is 0.693 W/m^2 K and its weight is 260 kg/m^2. It is assumed that the room has one 1.2 m × 1.2 m window in the south-facing exterior wall; all other walls, ceiling and floor are interior partitions. Absorptance of exterior surface to solar radiation is 0.8 and reflectance of inner surface 0.7. It is assumed that the room is occupied by three people at an activity level 5 (standing or doing light work). It is also assumed that the radiative energy due to lights, equipment etc is 139 W and capacitance of room air and furnishings is 1000 kJ/K. A simulation of indoor conditions of the zone is made by using type-19 of TRNSYS [2]. The walls are modeled using the ASHRAE transfer function approach [5]. Transfer function coefficients are taken directly from the data file, ASHRAE.COF, provided with TRNSYS. Simulations have been initialized with a room temperature 30°C and a humidity ratio 0.01, and the values get corrected by iteration. The final values depend on the actual data input to TRNSYS program.

It is assumed that the direct evaporative air cooler draws 100% fresh air and supplies it to the room after adiabatic evaporation (without recirculation) and provides a regulated mass flow of supply air for the room. It is also assumed that there is an arrangement to regulate the bypass factor (BPF) of the supply air. We can have any value of BPF starting from BPF = 0 representing supply air that is saturated, up to BPF = 100 representing ventilation with 100% ambient air (i.e., without any evaporative cooling). Thus BPF = x represents x% ambient air mixed with (100 − x)% saturated air.

Simulations have been carried out for several sets of weather data, parameters and input conditions of supply air, and hourly values of room air temperatures and humidity recorded as outputs. The simulation results have been represented in relation to the extended comfort zone described in Section 3.

3. The extended comfort zone for evaporative cooling

Watt [6] has pointed out that the official ASHRAE Comfort Chart which is bounded by the 17°C dew-point line (corresponding to 12 gm water vapor per kg air) and is centered around 23°C effective temperature on the psychrometric chart, is not pertinent to evaporative cooling. He has suggested constant relative humidity lines of 80% and 20% as upper and lower limits of the extended comfort zone, respectively, for evaporative cooling, with the base depending on air motion in the room. The Extended Comfort Zone 'C' of Fig. IV-3, page 36 of the handbook [6] is applicable for the higher air speeds of 3.5 m/s. (In the present study it is assumed that the room has a ceiling fan of appropriate rating which provides air movement of over 3.5 m/s, whenever required, so that this zone is applicable). The boundaries of the Extended Comfort Zone (ECZ) are shown hatched in Figs. 2–13. Room temperatures and relative humidities higher than this are considered as uncomfortable. The departure of prevailing room air conditions from this zone in terms of degree-hours or gram-hours (considering gms of water vapor per kg of air) is taken as a measure of discomfort. In the present work, the number of uncomfortable hours per day as well as the extent of discomfort have been studied.

4. Results and discussion

Results of simulation study for Delhi in April, May and June are depicted in Figs. 2–13 which show the variation of indoor conditions over the day. The conditions of room air from 1 to 24 h are plotted with dry bulb temperature on the x-axis and relative humidity on the y-axis to provide a good visual representation in relation to ambient air conditions (chained line) and the extended comfort zone, ECZ (hatched boundary). The room air conditions during office hours are represented by firm lines, whereas the dotted lines represent unoccupied periods. The effect of the number of air changes per hour (ACH) and the by-pass factor (BPF) on the resulting indoor conditions is clearly visible in these diagrams.

The aim is to find combinations of ACH and BPF that succeed in keeping room conditions within the extended comfort zone (below 80% RH and temperatures between 27 and 31°C, depending on RH), or close to it. The main features of the diagrams can be illustrated with the help of Fig. 8. It shows room air temperature and humidity over a 24-hour cycle for BPF = 0, 10, 20, 40 and 100% for air change rates of ACH = 10 ac/h. It is seen that a zero air bypass rate leads to excessive humidity in the room, rendering the conditions uncomfortable. A BPF of 10% provides some relief and a BPF of 20% provides comfortable conditions. However, a BPF of 40% leads to discomfort due to higher room temperatures in the afternoon and evening. Finally, a BPF of 100% (i.e., no evaporation), implying only ventilation, cannot provide comfort at any time of the day in May. Results for different combinations of ACH and BPF are given in Figs. 2–13. *It is evident that very low bypass factors are neither required nor desirable.*

Figs. 2–5 depict results for Delhi in April for each air change rate (ACH = 1, 5, 10, 40). They show room air temperature and humidity for different values of bypass factor, BPF = 0, 10, 20, 40 and 100%. It is very clear from these figures that 1 ac/h (Fig. 2) is too low, but there is a marked improvement in performance as air change rates increase to 5 or 10 ac/h (Figs. 3 and

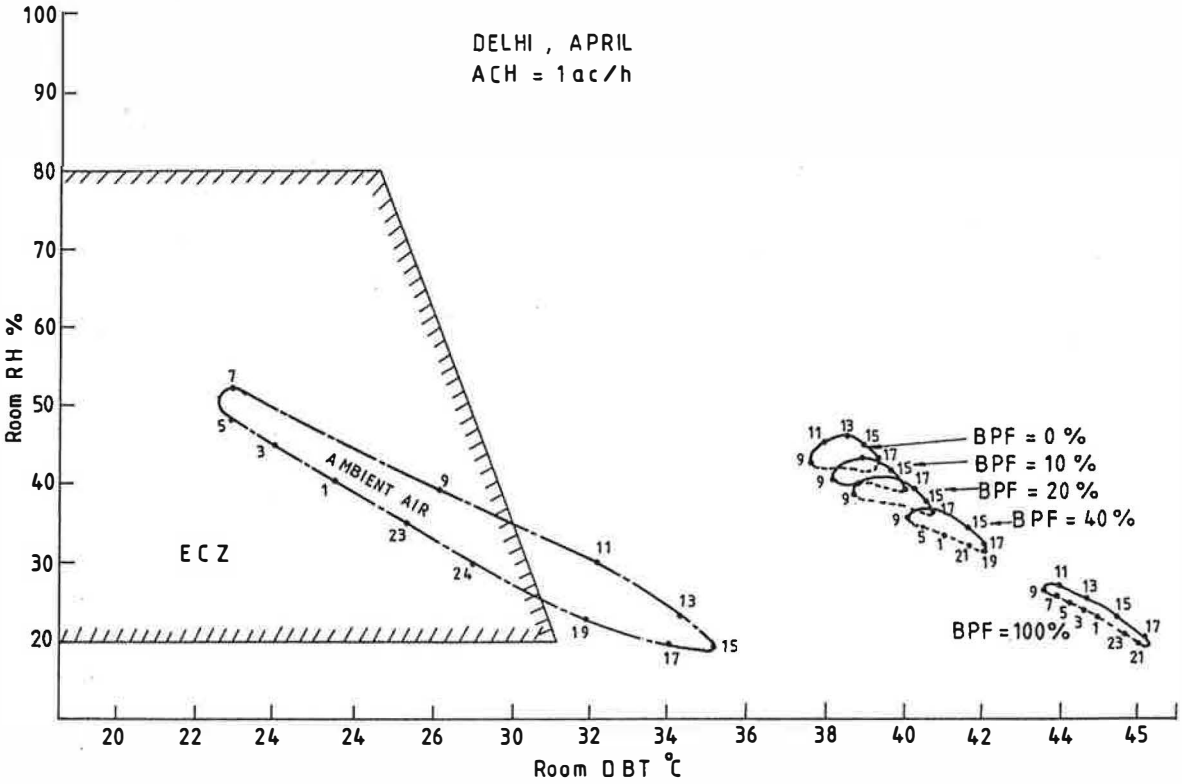

Fig. 2. Hourly values of room relative humidity and dry bulb temperature for ACH = 1 and BPF = 0, 10, 20, 40, 100 for the month of April.

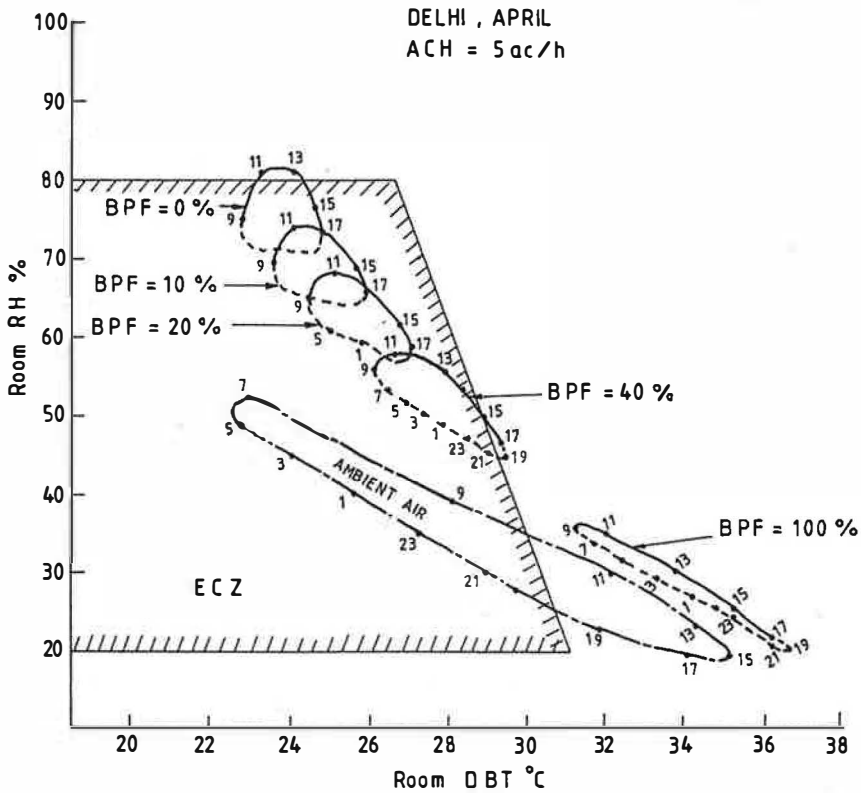

Fig. 3. Hourly values of room relative humidity and dry bulb temperature for ACH = 5 and BPF = 0, 10, 20, 40, 100 for the month of April.

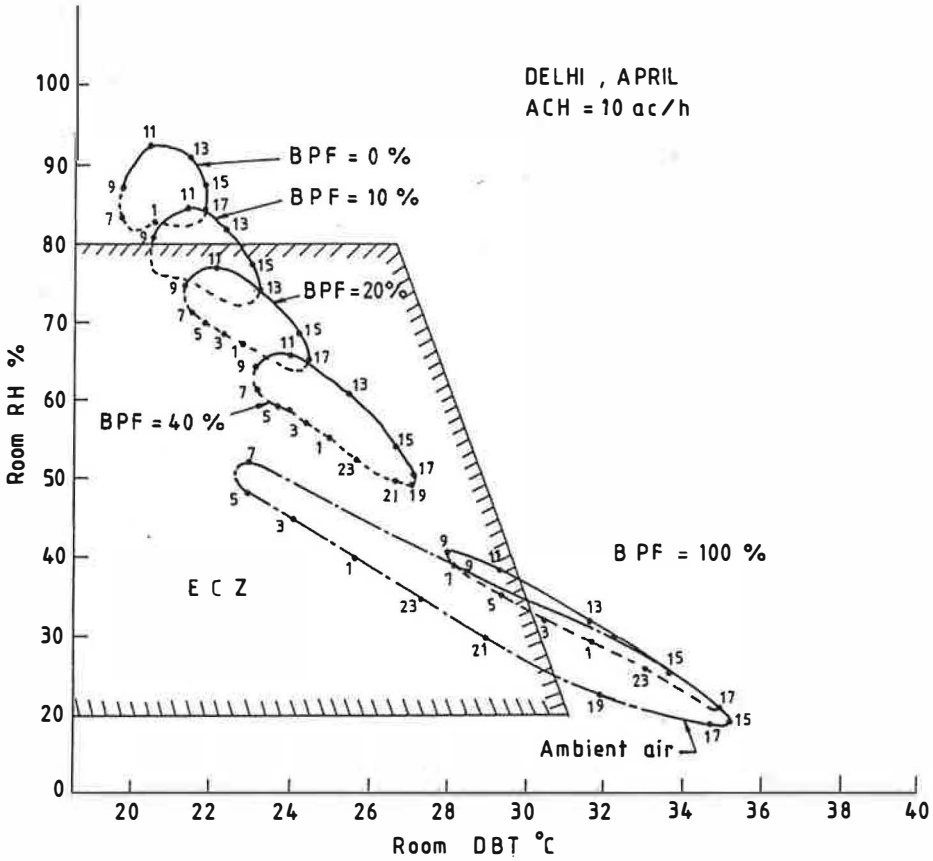

Fig. 4. Hourly values of room relative humidity and dry bulb temperature for ACH = 10 and BPF = 0, 10, 20, 40, 100 for the month of April.

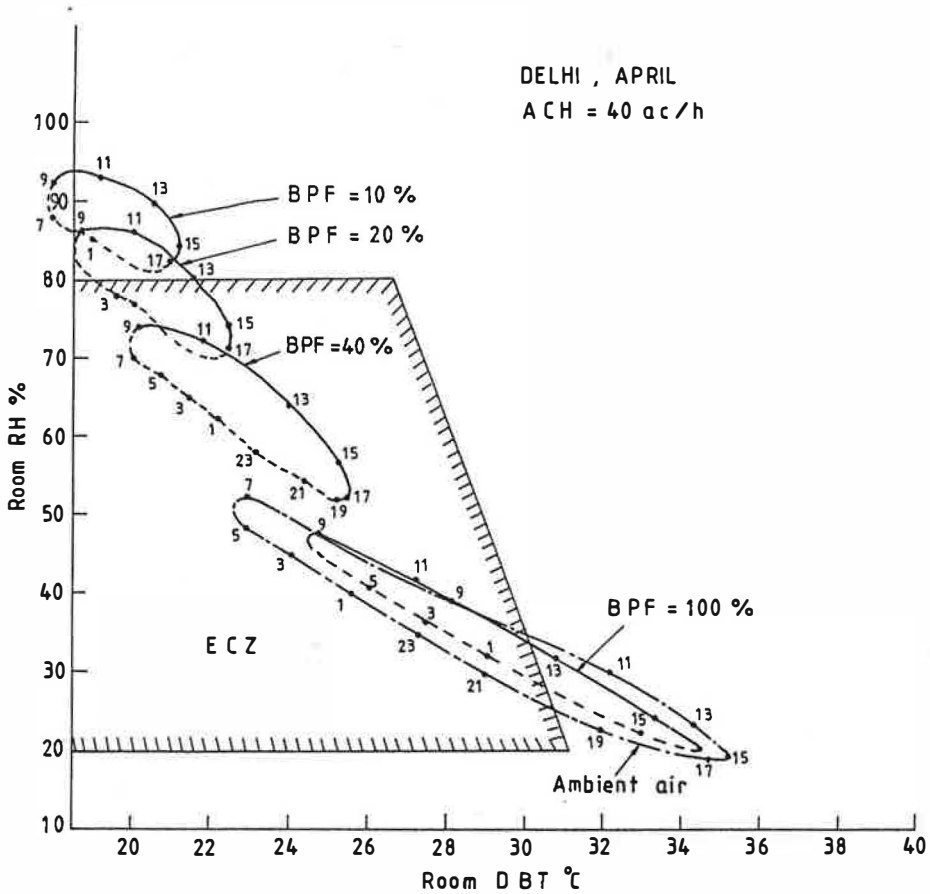

Fig. 5. Hourly values of room relative humidity and dry bulb temperature for ACH = 40 and BPF = 0, 10, 20, 40, 100 for the month of April.

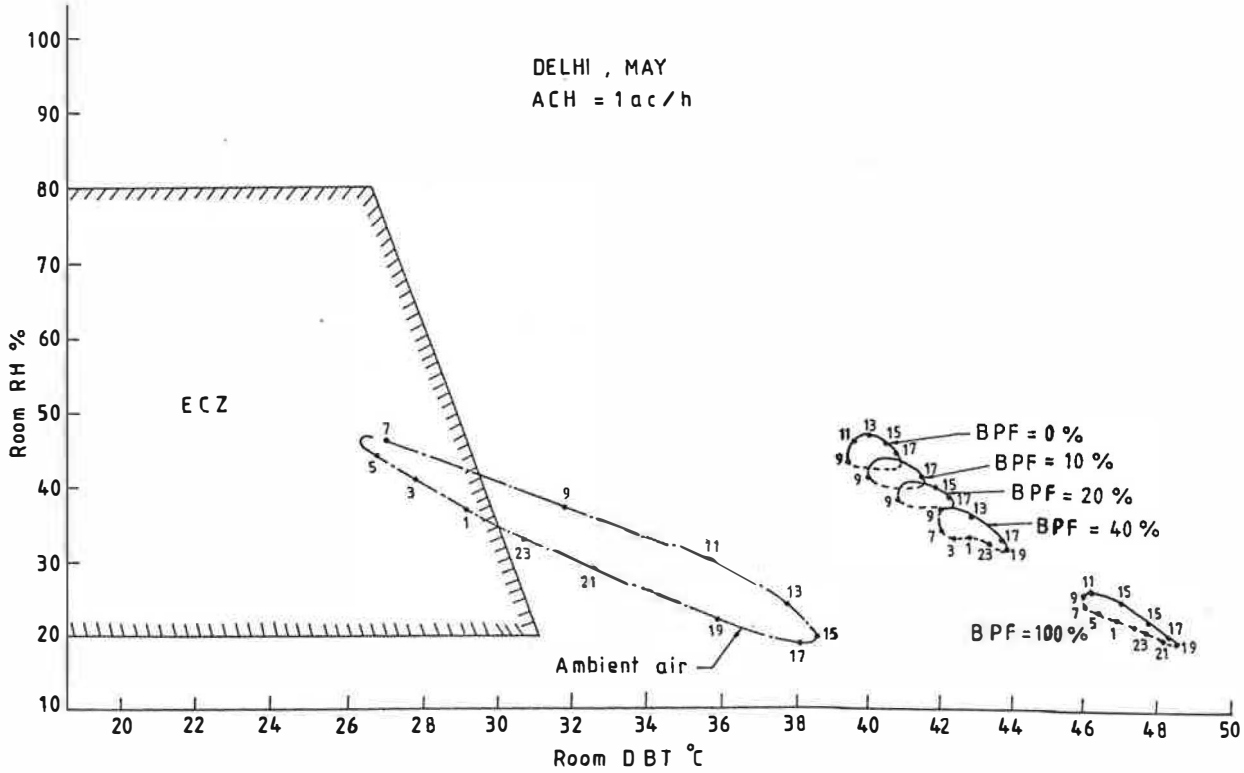

Fig. 6. Hourly values of room relative humidity and dry bulb temperature for ACH = 1 and BPF = 0, 10, 20, 40, 100 for the month of May.

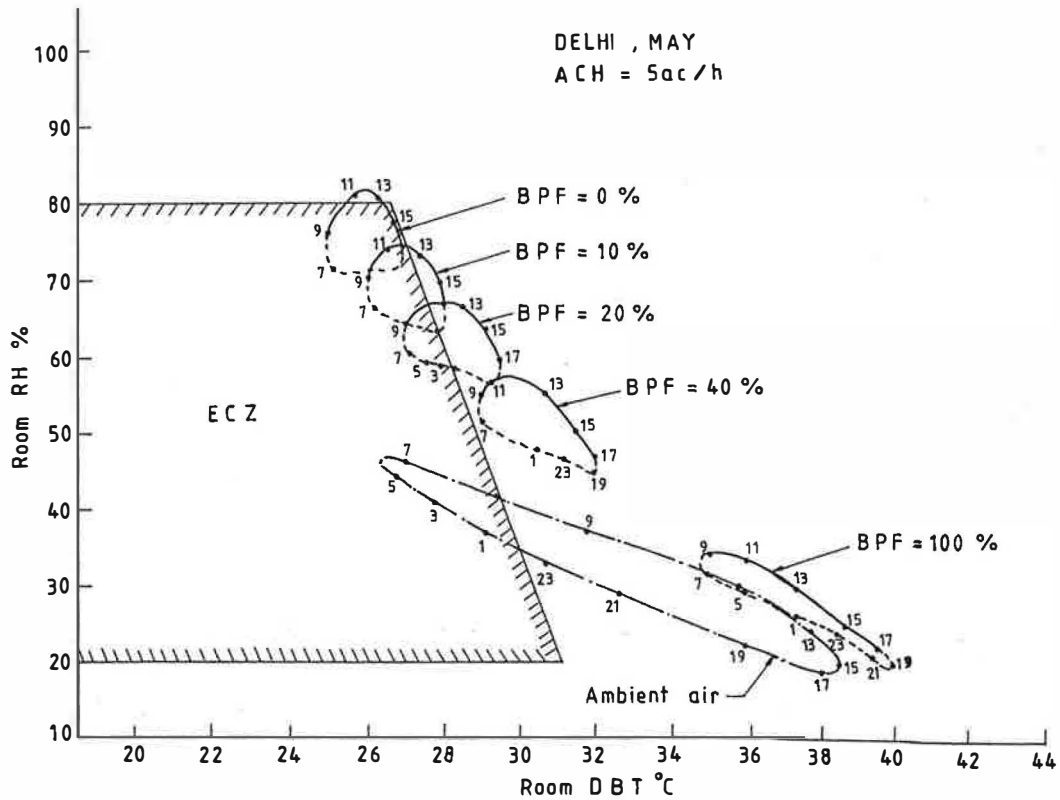

Fig. 7. Hourly values of room relative humidity and dry bulb temperature for ACH = 5 and BPF = 0, 10, 20, 40, 100 for the month of May.

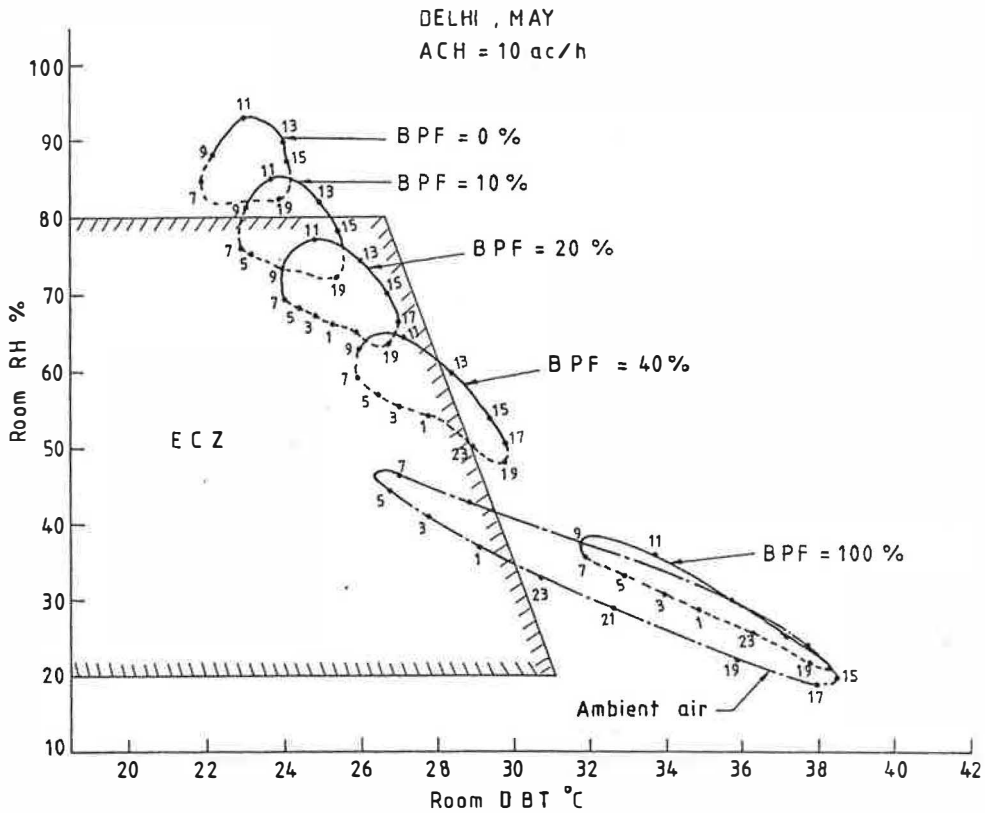

Fig. 8. Hourly values of room relative humidity and dry bulb temperature for ACH = 10 and BPF = 0, 10, 20, 40, 100 for the month of May.

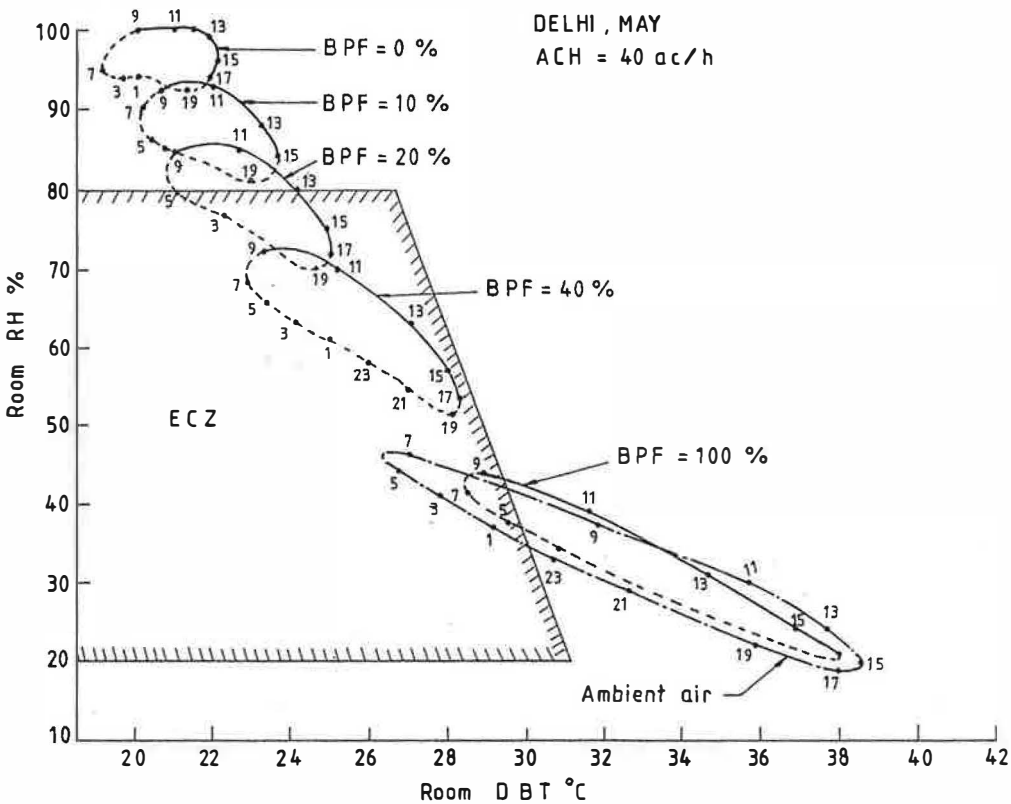

Fig. 9. Hourly values of room relative humidity and dry bulb temperature for ACH = 40 and BPF = 0, 10, 20, 40, 100 for the month of May.

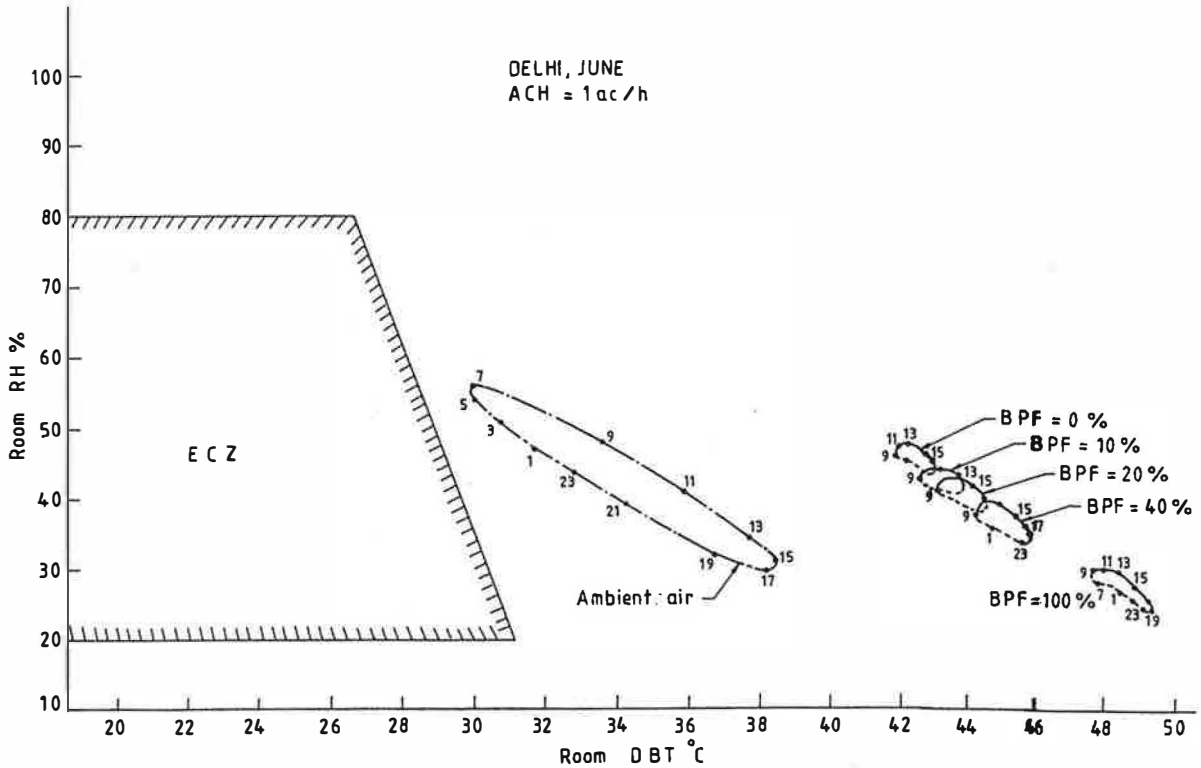

Fig. 10. Hourly values of room relative humidity and dry bulb temperature for ACH = 1 and BPF = 0, 10, 20, 40, 100 for the month of June.

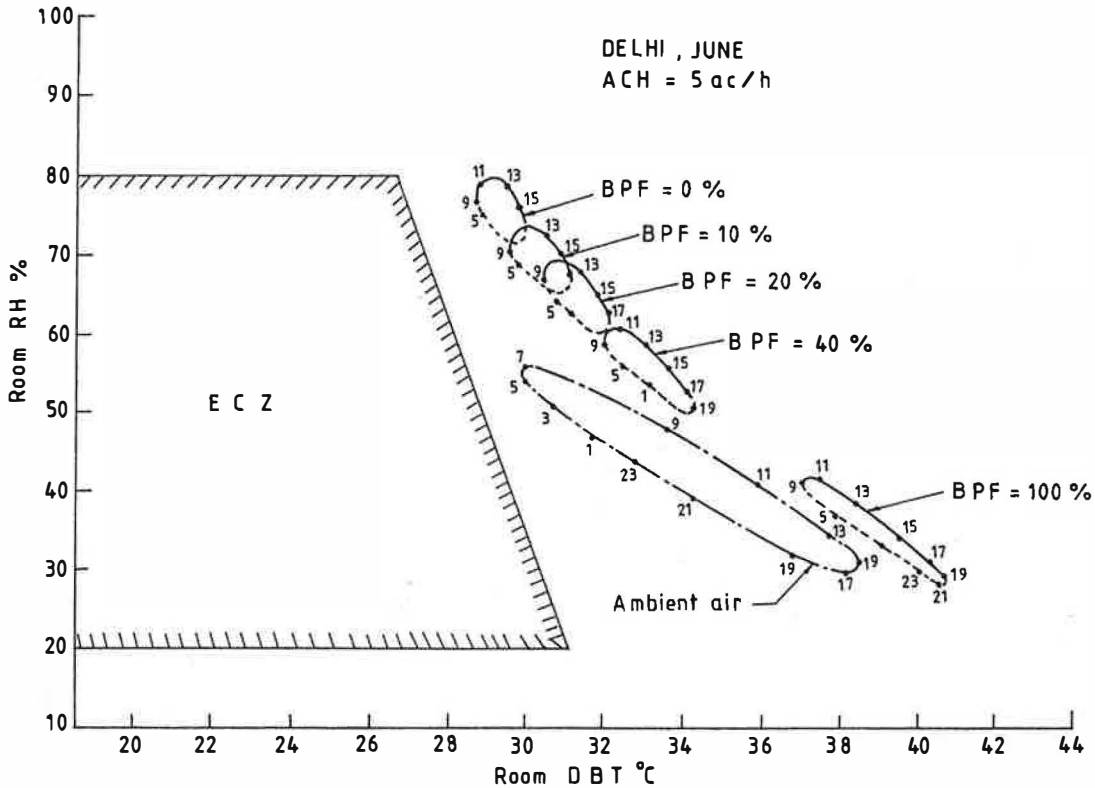

Fig. 11. Hourly values of room relative humidity and dry bulb temperature for ACH = 5 and BPF = 0, 10, 20, 40, 100 for the month of June.

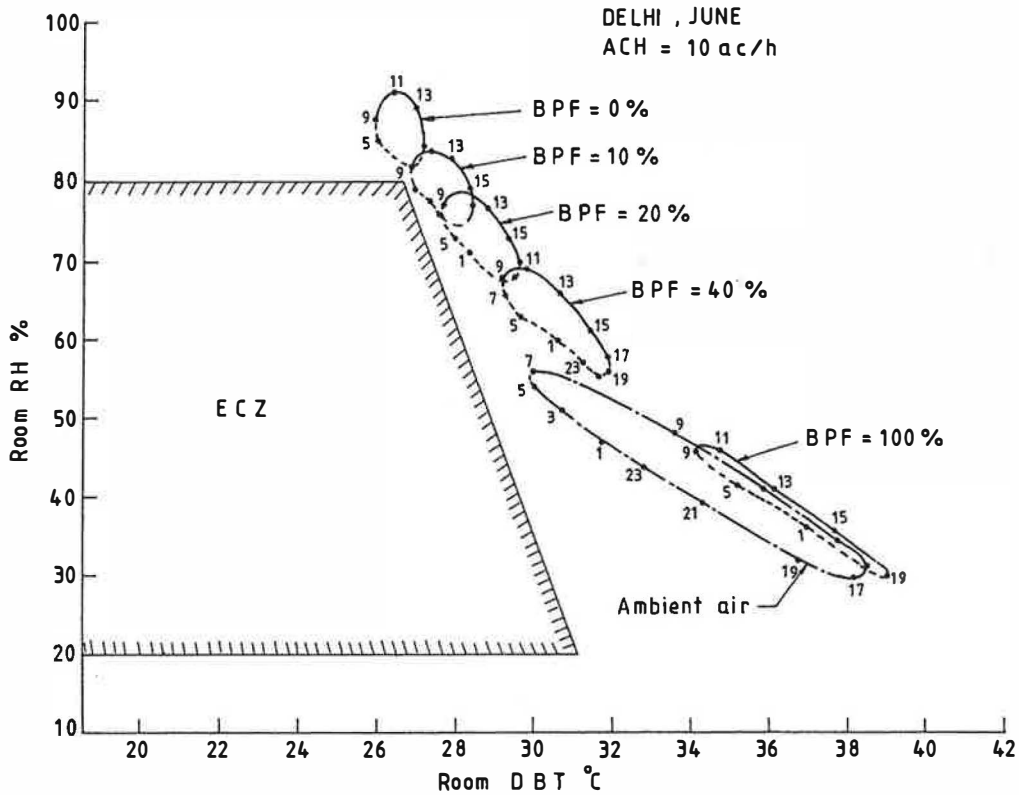

Fig. 12. Hourly values of room relative humidity and dry bulb temperature for ACH = 10 and BPF = 0, 10, 20, 40, 100 for the month of June.

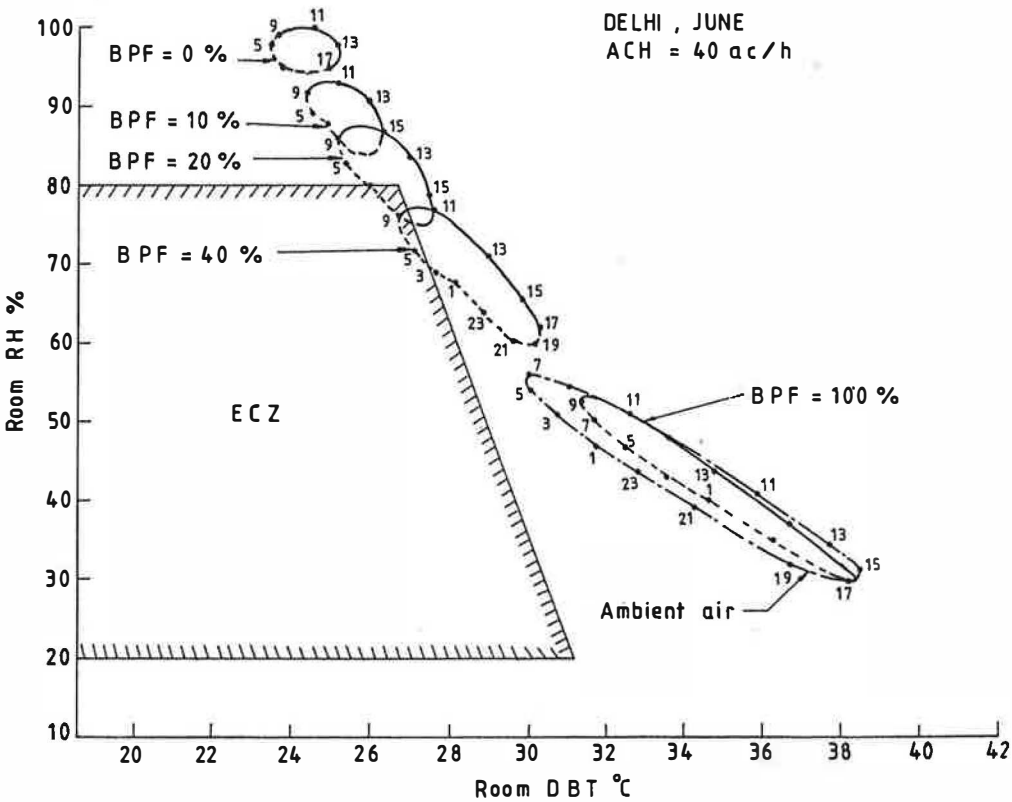

Fig. 13. Hourly values of room relative humidity and dry bulb temperature for ACH = 40 and BPF = 0, 10, 20, 40, 100 for the month of June.

Table 1
Uncomfortable hours and extent of discomfort for different combinations of ACH and BPF in the months of April, May and June

		April				May				June			
ACH	BPF	Degree hours	Uncomfortable hours of day	Gram hours	Uncomfortable hours of day	Degree hours	Uncomfortable hours of day	Gram hours	Uncomfortable hours of day	Degree hours	Uncomfortable hours of day	Gram hours	Uncomfortable hours of day
1	0	252.50	1–24	17.08	10–24	308.50	1–24	61.24	1–24	362.00	1–24	144.80	1–24
1	10	251.30	1–24	9.60	11–21	325.00	1–24	49.58	1–24	380.90	1–24	135.10	1–24
1	20	254.50	1–24	4.30	11–18	343.10	1–24	37.88	1–24	394.80	1–24	125.40	1–24
1	40	252.80	1–24	0		357.40	1–24	15.15	8–2	422.50	1–24	106.00	1–24
1	100	323.80	1–24	0		376.40	1–24	0		505.50	1–24	47.77	1–24
5	0	0		0.19	11–13	0		0.74	11–13	60.41	1–24	35.49	1–24
5	10	0		0		2.87	12–20	0		83.97	1–24	25.77	1–24
5	20	0		0		10.95	11–1	0		102.90	1–24	16.182	4–21
5	40	1.04	15–20	0		38.23	1–24	0		121.90	1–24	5.82	9–17
5	100	80.90	1–24	0		155.70	1–24	0		203.80	1–24	0	
10	0	0		17.26	1–24	0		21.04	1–24	3.66	12–20	31.18	1–24
10	10	0		2.48	9–13	0		3.07	9–13	18.51	1–24	13.67	9–20
10	20	0		0		0		0		35.15	1–24	8.47	9–18
10	40	0		0		5.72	13–21	0		64.06	1–24	1.99	10–14
10	100	38.19	12–24	0		108.70	1–24	0		158.40	1–24	0	
40	0	0		48.70	1–24	0		56.95	1–24	0		71.66	1–24
40	10	0		19.78	1–24	0		21.72	1–24	0		38.08	1–24
40	20	0		4.04	6–13	0		3.96	6–12	4.42	13–20	12.44	1–17
40	40	0		0		0		0		23.60	10–3	0.40	11–13
40	100	24.70	13–23	0		74.33	10–4	0		121.30	1–24	0	

4). Although 40 ac/h (Fig. 5) can provide comfort, it is neither required nor recommended as further lowering of temperature is at the cost of large electrical loads and noise levels, and it also implies a larger water demand.

Similar results for May are shown in Figs. 6–9. Once again, it is seen that 1 ac/h (Fig. 6) is too low. As the air change rate is increased to ACH = 5 ac/h, conditions improve but this ACH is not sufficient. However, ACH = 10 ac/h is satisfactory.

At 1 ac/h, values of the room temperature are very high as: (i) the rate of supply air is too low for meeting the internal heat loads and solar gain through the south-facing wall and window, and (ii) the remaining three walls, floor and ceiling are interior partitions and they do not contribute to heat dissipation. This also explains the large phase lag of about 4 h. (The only wall dissipating heat after sunshine hours, the south wall, is insulated and the heat capacity of the room is large, and air change rate is too low). As the air change rate increases, the phase lag also decreases.

June is the hottest month for Delhi. The results for June are shown in Figs. 10–13. It is seen from these figures that comfort conditions cannot be achieved in June. Air change rates of ACH = 1 and 5 ac/h are inadequate. However, ACH = 10 or more can substantially improve the room conditions: *reducing the extent of discomfort.*

The number of uncomfortable hours per day and the extent of discomfort for the three summer months are presented in Table 1. The extent of discomfort due to higher temperatures for any hour is quantified by considering degrees in excess of comfortable temperatures. The extent of discomfort per day is found by summing the hour-by-hour departure in degrees from the comfort zone (during uncomfortable hours) and results are tabulated in degree-hours. A similar procedure is followed for the hours when the humidity exceeds comfortable values, and the results of summation are tabulated in gram-hours.

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