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EFFECT OF CONTROL OPTIMIZATION OF CO_2 TRANSCRITICAL SPLIT AIR CONDITIONER ON THERMAL COMFORT OF OCCUPANTS IN SINGLE ZONE ROOM

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Energy, environment and economics are considered as very vital parameters for the evaluation of an air conditioning system and associated indoor environment. The cooling performance of an air conditioner has an effect on the thermal comfort of occupants in the room. Transcritical CO_2 air conditioner (System B) with a control for gas cooler pressure has better energy performance than a transcritical CO_2 air conditioner (System A) without any control on the gas cooler pressure. An experimental technique is used for generating performance equations to define transcritical CO_2 air conditioners in the EnergyPlus program. EnergyPlus simulates combined model of a transcritical CO_2 air conditioner and room for known yearly weather data for an effect on thermal comfort in the room. Thermal comfort in the room is evaluated using the Fanger thermal comfort model and the Pierce two node model. The better energy performance of System B results in improved indoor room environment of the room. The total cooling of System B is 15.78–20.2% higher than that of System A. The Fanger thermal comfort model shows that 95% to 133% people are more dissatisfied with an indoor thermal environment during the morning and 85% to 127% people during the afternoon for a room coupled with System A vis-à-vis room with System B.

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Keywords: Transcritical; CO_2; air conditioner; thermal comfort; Fanger model; Pierce model; EnergyPlus; IMST-ART; SEER (Seasonal Energy Efficiency Ratio).

1. Introduction

Thermal comfort is the condition of the mind, which expresses satisfaction with the thermal environment. Due to both physiological and psychological variations from person to person, it is difficult to satisfy every occupant in the thermal environment.[1] The six primary factors: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed and humidity defines the state of thermal comfort. People spend more than 80% of their time in indoor environments like apartments, offices, bungalows and so on. These indoor environments or spaces are usually installed with unitary room air conditioners having no provisions for fresh outside air supply. The rated cooling capacity and power consumption of room air conditioner are worked out as per Indian standard IS 1391 Part I with conditions: indoor 26.9°C (DBT)/18°C (WBT) and outdoor 35°C (DBT)/24°C (WBT). Room air conditioners operate at ambient conditions, which for nearly all cases are different from rated conditions. Many a times, the selection of cooling capacity of a room air conditioner for a known carpet area of the room is worked out based on the approximation. Because of the aforementioned factors, the cooling capacity and power consumption of room air conditioners deviate from their rated performance. At ambient conditions higher than rated conditions, it is likely that thermal comfort conditions of the room or occupied space will not be achieved.

In order to reduce the environmental problems related to the global warming and the ozone depletion, the traditional man-made chemical refrigerants are being phased-out as per worldwide agreed schedules and are being replaced by environmental friendly, low or nontoxic alternative chemical refrigerants and natural refrigerants. Carbon dioxide (CO_2) is an environmental friendly natural refrigerant since its Ozone Depletion Potential (ODP) is zero and Global Warming Potential (GWP) is one. However, the critical temperature of CO_2 (30.9°C) is very close to the observed typical weather dry bulb temperature (DBT) in the subtropical and tropical climatic regions. This imposes the CO_2 vapor compression refrigeration cycle to work in transcritical mode whenever the weather DBT is more than the CO_2 critical temperature. In the transcritical mode of operation, the heat absorption happens in the subcritical region and the heat rejection process takes place in the supercritical region of thermodynamic phase change diagram of CO_2. In the supercritical region, the pressure is independent of the temperature and the isotherms are 'S' shaped, resulting in a nonmonotonic variation of COP with the gas cooler pressure.[2] Hence, in a transcritical CO_2 air conditioner operating at ambient temperatures above the critical temperature of CO_2, there exist an optimum value of the gas cooler pressure, which gives the maximum Coefficient of Performance (COP).

For a fixed geometrical configuration of the CO_2 unitary-split air conditioner, authors have experimentally developed a simple procedure to control the gas cooler pressure as a function of indoor wet bulb temperature (WBT_i) and outdoor dry bulb temperature (DBT_o) to maintain the COP at its maximum level. In this paper, the air conditioner with control on the gas cooler pressure is called a 'controlled' CO_2 air conditioner (System B) and the CO_2 air conditioner with no such control on the gas cooler pressure is called a 'noncontrolled' air conditioner (System A). The energy performance of both systems has been compared.

The present simulated research work aims to evaluate the effects of energy performance of controlled and noncontrolled transcritical CO_2 air conditioning systems on the thermal comfort in the indoor environment of the room for outdoor ambient DBT above the critical temperature of CO_2. The work also includes the evaluation of seasonal performance of both controlled and noncontrolled CO_2 air conditioning systems. For simulation studies, the hourly building energy simulation program, EnergyPlus, has been used.[3,4] The map coefficients of the performance curves to define the controlled and noncontrolled transcritical CO_2 air conditioners in EnergyPlus have been developed experimentally.

2. Literature Review

Zhou *et al.*[5] have simulated Variable Refrigerant Volume (VRV) system using EnergyPlus for two rooms. The simulated results were validated with experimental results. The average deviation for COP between simulated and experimental results, excluding first hour of operation is 5.66%. Further, it was found that the mean relative error in the test week is 28.31% for the VRV system power simulation, while 25.19% for the system cooling capacity simulation. The actual daily average Part Load Ratio (PLR) of the VRV system ranges from 0.497 to 0.779.

Aynur *et al.*[6] have used EnergyPlus for simulation of variable air volume (VAV) system for an existing building in cooling mode. The simulated and experimental data falls within $\pm 15\%$ for power consumption and within $\pm 1.5\%$ for indoor temperatures. The simulated indoor relative humidity (RH) values have a maximum of 18% deviation with experimental values. Authors have recommended the use of EnergyPlus for single- and multi-story building energy simulation considering its accuracy of performance prediction.

Lin and Deng[7] reported a simulation study on the characteristics of night time bedroom cooling load in tropic and subtropic climatic conditions using EnergyPlus for a 30- story building. The authors have mentioned that the thermal mass of the building and furniture contributes to the total cooling load of the room air conditioner. They have further observed that the orientation of the building has a major influence on the day operating mode vis-à-vis the night operating mode of a room air conditioner. The difference between the hourly total cooling load for bedrooms facing west with respect to bedrooms facing the other three orientations at night operating mode ranges from 6.6% to 25.3% as compared to 44% to 57% at day operating mode.

Schiavon and Melikov[8] reported the potential saving of cooling energy for three elevated air velocities (less than 0.2 m/s, 0.5 m/s and 0.8 m/s) and the impact of increased room air temperature on the occupants' comfort. The authors had used EnergyPlus software for simulation. A cooling energy saving in the range 17–48% and a reduction of the maximum cooling power in the range 10–28% have been observed. The results reveal that the input power of the fan is a critical factor for achieving the energy saving at elevated room temperatures.

Eskin and Turkmen[9] have studied variations in the energy demand of office buildings and control strategies for the four climatic zones in Turkey with EnergyPlus and results have been compared with actual site data. The simulated results for the daily variations in the cooling and heating requirements are within 5% of the experimental findings. They have also further studied the effects of parameters like ventilation rate, aspect ratio, window system, outside wall color, thermal mass and shading on the cooling load profile of the building.

Chowdhury *et al.*[10] have evaluated the different low energy cooling technologies like chilled ceiling, economizer, and pre-cooling; for their effect on thermal comfort by using DesignBuilder and EnergyPlus. The chilled ceiling technology offers the best thermal comfort measure over the other two cooling technologies for occupants during summer and winter in a subtropical climate. The difference between the simulated and measured (base case) humidity levels during occupied hours is a maximum of 15% during the summer season and a maximum of 18% during the winter season, which lie within the acceptable range.

The annual energy cost for displacement ventilation and VAV systems that use free cooling is about 20% less than an existing system, which uses fixed minimum supply air rate technology that does not take advantage of free cooling. This is a conclusion of the research done by Olsen and Chen[11] through EnergyPlus for mild climate in the United Kingdom.

The literature review shows that the work is significantly missing on the study of the impact of energy performance of a transcritical CO_2 room air conditioner on the thermal comfort indices of the indoor environment for both cold and warm climatic conditions. In the current work, an EnergyPlus simulation is carried out for the summer design day 8th of May in subtropical climatic conditions typically observed in Pune city (India). A similar procedure extends to other climatic zones, building configurations and load characteristics.

3. Experimentation

The baseline noncontrolled CO_2 air conditioning system (System A) consisted: a reciprocating-hermetic compressor, a fin-tube gas cooler, an internal heat exchanger (IHX), a thermostatic expansion valve (TXV), an oil separator and a fin-tube evaporator. The controlled (System B) is similar to System A except for an electronic expansion valve (EEV) instead of a TXV, and the additional receiver at the compressor inlet (Fig. 1). System A is based

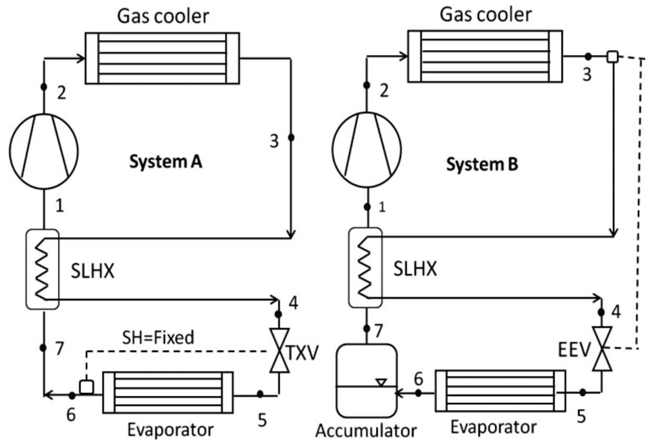


Fig. 1. Schematic diagram of the two systems.[2]

on a TXV, which maintains a fixed superheat of 7°C at the evaporator outlet through modulating the refrigerant flow. This method does not have any control on the gas cooler pressure. However, System B has an EEV, which adjusts the gas cooler pressure to a predetermined optimum gas cooler pressure corresponding to indoor WBT and outdoor DBT by modulating the refrigerant flow from the high pressure gas cooler to low pressure evaporator. System A has a fixed CO_2 charge 144 g which was optimized for rated conditions. However, System B has a variable charge due to a receiver at the evaporator outlet which leads to build up of oil in the receiver. However, having a receiver at the evaporator outlet has two advantages; it helps to maintain saturated vapor at inlet of a Suction Line Heat Exchanger and adds flexibility to operate System B in reversible cycle mode. IMST ART[12,13] program has been used to optimize the geometrical configuration of these systems and determine their rated capacities for test conditions as per IS 1391 Part 1 (1992). The configurations of a split air conditioner including all components are given separately for the rated capacity (Table 1). The system simulation was done using the software IMST-ART. The simulated performance of the system was validated with experimental performance for the same conditions.

The variation in rated cooling capacity was ± 2.5% and the variation in COP was ± 3.87% for System A. For System B, the variation in rated cooling capacity was ± 1.89% and the variation in COP was ± 2.59%.

Detailed experimental work has been performed for both systems A and B to determine their energy

Table 1. Configuration of a split air conditioner.

Evaporator			
Tube length [mm]	550	Height [mm]	180
Depth [mm]	50	Width [mm]	500
Number of tube rows	4	Tube diameter [mm]	4.75
Number of tubes per row	10	Type of fin	Louvered
Number of circuits	5	Fin spacing [mm]	1.37
Gas cooler			
Tube length [mm]	550	Height [mm]	375
Depth [mm]	36	Width [mm]	700
Number of tube rows	2	Tube diameter [mm]	4.75
Number of tubes per row	15	Type of fin	Louvered
Number of circuits	5	Fin spacing [mm]	1.37
Compressor			
Maker and model	Danfoss TN1416	Displacement capacity [cm^3]	2.46
Internal heat exchanger (IHX)			
Overall thermal resistance (1/UA) [W/K]	4		
Connecting tubes/lines			
Pipe lines length [mm]	1000		

performance in terms of polynomial equations for a wide range of indoor and outdoor conditions. However, an interpretation of operating parameters on the performance of the system and thereon thermal comfort in the room is not the scope of this paper. The cooling capacity and suction saturation pressure basically govern the air side parameters off the cooling/evaporator coil of an air conditioner.

The experimental test set-up for the transcritical CO_2 air conditioning system had four fluid circuits: the air-side of the CO_2 evaporator, the air-side of the CO_2 gas cooler, the cold water (CW) chiller circuit and the CO_2 refrigerant circuit (Fig. 2). The test set-up was able to simulate ambient conditions typically observed in Pune (India). During tests, the DBT of the supplied air to the CO_2 evaporator was maintained fixed at 26.9°C (±0.1°C) and the WBT was changed from 18°C to 23°C (±0.3°C) in step of 1°C. On the gas cooler side, the DBT of supplied air was varied from 33°C to 45°C (±0.1°C) in step of 2°C with no control on the WBT. There were total of 42 combination sets of indoor WBTs and outdoor DBTs.

Throughout the experimental tests, the air flow at the evaporator was kept constant at 550 m^3/h (±2%), and that of at the gas cooler was constant at 1100 m^3/h (±2%). All temperatures and pressures of the refrigerant and air were measured and recorded on a real-time basis using a data acquisition system. The refrigerant mass flow rate at the suction of the compressor was also recorded. Using measured quantities, the cooling capacity, power consumption, and COP of the system for all the 42 combination sets of WBTs and DBTs were calculated. The experimental setup was used to simulate both the noncontrolled and controlled CO_2 split air conditioners. A statistical regression analysis method was applied to the experimental data and performance map coefficients were worked out for the polynomial Eqs. (1) to (5). These coefficients are given for both systems (Tables 2 and 3). These coefficients are used to get the performance of the CO_2 transcritical air conditioner in EnergyPlus.

EnergyPlus is a freeware program developed by the Department of Energy, USA, which gives hourly energy analysis of a multi-zone building with

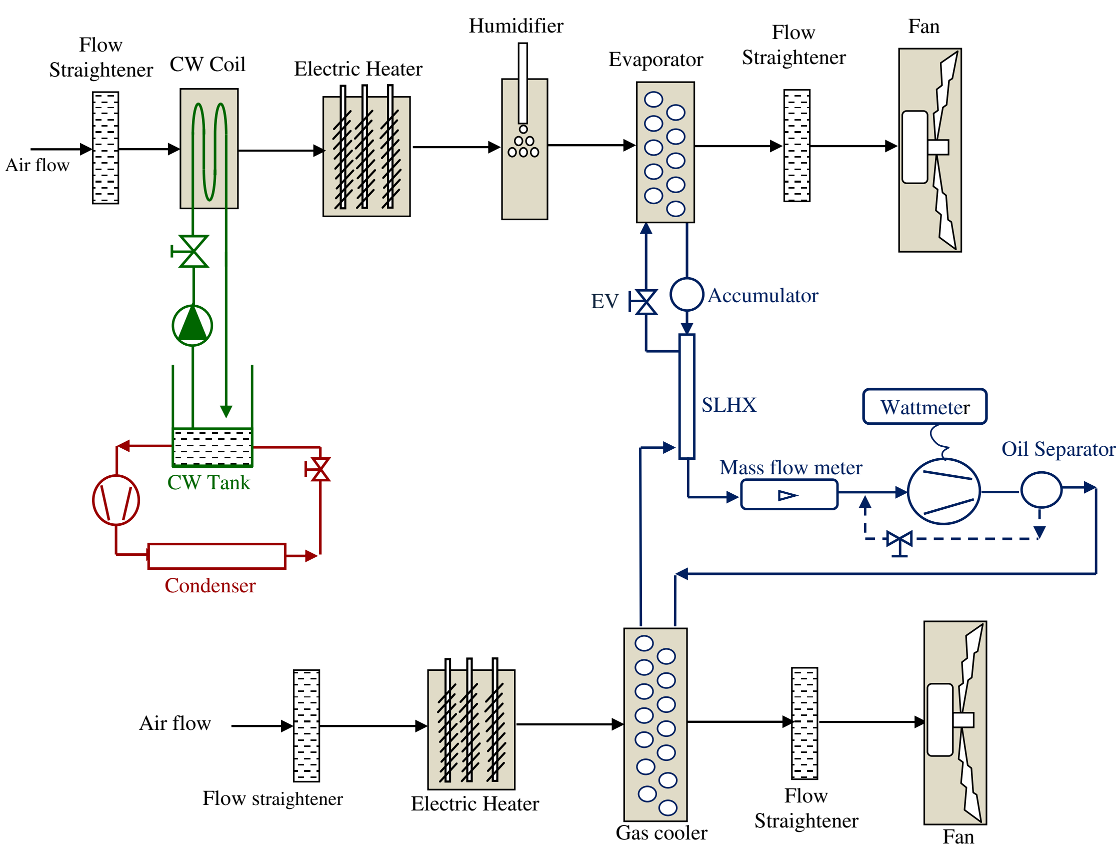


Fig. 2. Experimental setup of a transcritical CO_2 air conditioner.

Table 2. Performance coefficients for a noncontrolled System A.

Performance curve	A	B	C	D	E	F
TotCapModFac	−0.017857907	0.1300720875	−0.001420261	−0.008084856	0.000125054	−0.001236889
TotCapflowModFac	1	—	—	—	—	—
EIR TempModFac	1.53639499832	−0.042002704	0.0010213453	−0.032050573	0.00138529477	−0.0010276839
EIRflowModFac	1	—	—	—	—	—
PartLoadFrac	0.85	0.15	—	—	—	—

Table 3. Performance coefficients for a controlled System B.

Performance curve	A	B	C	D	E	F
TotCapModFac	−3.688666	0.37774805	−0.0064976	0.03839541	−0.000147	−0.0019928
TotCapflowModFac	1	—	—	—	—	—
EIR TempModFac	8.074983	−0.62951	0.014157	−0.047	0.001118	0.00008907375
EIRflowModFac	1	—	—	—	—	—
PartLoadFrac	0.85	0.15	—	—	—	—

heating and cooling systems.[3] It is being considered the most tested and reliable energy simulation program for buildings in all climatic zones.[4]

EnergyPlus uses the total cooling capacity modifier curve function of temperature (Eq. (1)). It is a biquadratic curve with two independent variables: WBT of the air entering the cooling coil and DBT of the air entering the air-cooled condenser coil. This factor helps to correct the rated cooling capacity of a system for various ambient conditions.

$$\begin{aligned} TotCapModFac &= a + b(T_{wb,i}) + c(T_{wb,i})^2 + d(T_{c,i}) \\ &\quad + e(T_{c,i})^2 + f(T_{wb,i})(T_{c,i})\,. \end{aligned} \tag{1}$$

The total cooling capacity modifier function of flow fraction is a quadratic curve with the independent variable being the ratio of the actual air flow rate across the cooling coil to the rated air flow rate (i.e., fraction of full load flow) (Eq. (2)). This factor has been used to correct the rated cooling capacity at a specific temperature and air flow at which the dry expansion (DX) unit is operating.

$$\begin{aligned} &TotCapflowModFac = a + b(\mathrm{ff}) + c(\mathrm{ff})^2 + d(\mathrm{ff})^3\,, \\ &\text{Flow fraction, ff} = \frac{\text{Actual air mass flow rate}}{\text{Rated air mass flow rate}}\,. \end{aligned} \tag{2}$$

The energy input ratio (EIR) modifier curve function of temperature is a biquadratic curve with two independent variables: WBT of the air entering the cooling coil, and DBT of the air entering the air-cooled condenser coil (Eq. (3)). This factor corrects the rated EIR (inverse of the rated COP) to give the EIR at the specific entering air temperatures at which the DX coil unit is operating.

$$\begin{aligned} EIRTempModFac = a &+ b(T_{wb,i}) + c(T_{wb,i})^2 \\ &+ d(T_{c,i}) + e(T_{c,i})^2 \\ &+ f(T_{wb,i})(T_{c,i})\,. \end{aligned} \tag{3}$$

EIR modifier curve function of flow fraction is a quadratic curve with the independent variable being the ratio of the actual air flow rate across the cooling coil to the rated air flow rate (Eq. (4)). The output of this curve is multiplied by the rated EIR (inverse of the rated COP) and the EIR modifier curve (function of temperature) to give the EIR at a specific temperature and air flow conditions at which the DX unit is operating.

$$\begin{aligned} &EIRflowModFac = a + b(\mathrm{ff}) + c(\mathrm{ff})^2 + d(\mathrm{ff})^3\,, \\ &\text{Flow fraction, ff} = \frac{\text{Actual air mass flow rate}}{\text{Rated air mass flow rate}}\,. \end{aligned} \tag{4}$$

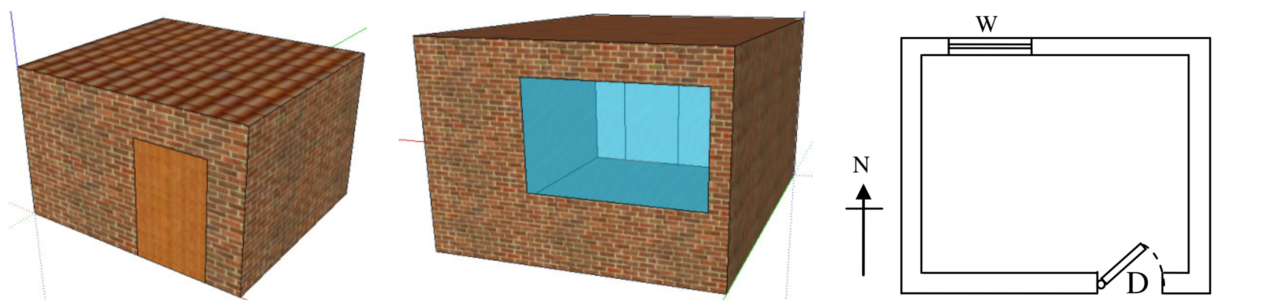
The Part Load Fraction (PLF) correlation is function of PLR (Eq. (5)).

$$\begin{aligned} &EIRflowModFac \\ &\quad = a + b(\mathrm{PLR}) + c(\mathrm{PLR})^2 + d(\mathrm{PLR})^3\,, \\ &\text{Part Load Ratio, PLR} \\ &\quad = \frac{\text{Sensible cooling load}}{\text{Steady state sensible cooling capacity}}\,. \end{aligned} \tag{5}$$

The output of this curve has been used in combination with the rated EIR and EIR modifier curves to give the 'effective' EIR for a given simulation time step. The PLF correlation accounts for efficiency losses due to compressor cycling.

4. Single Zone Room Model

The single zone room model used is having a floor area 12.23 m^2 with ceiling height 3 m (Fig. 3). The room construction details and materials of construction are considered as per latest building construction practices in India (Tables 4 and 5). Electrical appliances (power density 81.73 W/m^2) and occupancy (three people seated quietly) are considered as the heat load for 365 days. The lighting heat load (power density 13.89 W/m^2) has been considered for a period of 18.00 to 23.00 h throughout the year. The simulation is carried out for the typical cooling design day 8th of May in subtropical conditions of Pune city (India).



(a) (b) (c)

Fig. 3. A single zone room model. (a) Asymmetric view: door side, (b) asymmetric view: window side and (c) plan view.

Table 4. Construction materials for the room.

	Layer 1 (outdoor facing)		Layer 2 (middle)		Layer 3 (indoor facing)	
	Material	Thickness (m)	Material	Thickness (m)	Material	Thickness (m)
External wall	Plasterboard-1	0.16	M01 brick	0.2286	Plasterboard-2	0.15
Flooring	Hf-c5	0.1015	—	—	—	—
Roof	Roof deck	0.5	Plasterboard-2	0.15	Plasterboard-1	0.16
Door	Wood siding	0.025	—	—	—	—
Window	Plain glass	0.00635	Air gap	0.0125	Plane glass	0.00635

Table 5. Building material thermal properties.

Material	Density (kg/m3)	Specific heat (J/kg-K)	Thermal conductivity (W/m-K)
Plasterboard-1	950	1000	0.16
Plasterboard-2	950	1000	0.16
M01 brick	1920	790	0.89
HF-C5	2243	837	1.7296
Roof deck	530	1000	0.1
Wood Siding-1	530	900	0.14

5. Thermal Comfort Models

Thermal comfort indices of the room indoor environment are defined using Fanger and Pierce comfort models.

The Fanger model is based upon an energy analysis that takes into account various energy losses from the body in the form of convection and radiation. These heat losses include: the heat loss by water vapor diffusion through the skin, the heat loss by evaporation of sweat from the skin surface, and the dry and latent respiration heat loss. The model considers the heat transfer resistance of clothing.[14] The model assumes the person is in thermal steady state with the environment. The Predicted Mean Vote (PMV) and Predicted Percent of Dissatisfied (PDD) are the indices of the Fanger comfort model. PMV is the expression of people about their satisfaction with available thermal comfort in the space according to the ASHRAE thermal scale.[1] The PMV is extended to determine the percentage of people dissatisfied with the surrounding thermal environment.

The Pierce model assumes the human body as a thermal lump of two concentric compartments. The core simulates the body generating metabolic energy and the outer simulates the skin. The Pierce model presents the Predicted Mean Vote at Effective Temperature (PMVET) as an index of thermal comfort. The Pierce model converts the actual environment into an environment at an Effective Temperature (ET). The ET is the DBT of a hypothetical environment at 50% RH and mean radiant temperature (MRT) where the occupants would experience the same physiological strain as in the real environment. PMVET more than zero indicates warmness and less than zero indicates coolness.

6. Results and Discussion

The simulated results include outdoor DBT, RH, COP and comfort indices. It has been observed that the swings in outdoor DBT and RH are very high throughout 24 h of the day (Fig. 4). In a day, from 12.00 h to 18.00 h, the outside DBT reaches a maximum of 42°C and RH reaches to 16%. In the early morning period, the ambient is at a higher RH and lower DBT.

The MRT, DBT, Clothing Surface Temperature (CST) and percentage RH are varying as function of the time of day (Fig. 5). Fundamentally, the MRT is

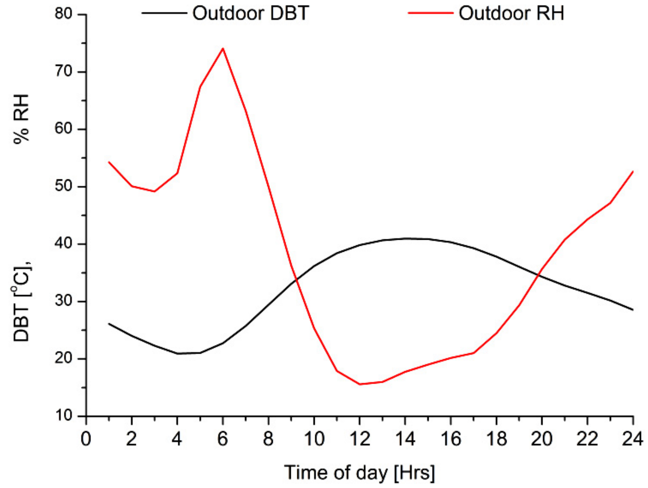


Fig. 4. Variation in outdoor DBT and RH.

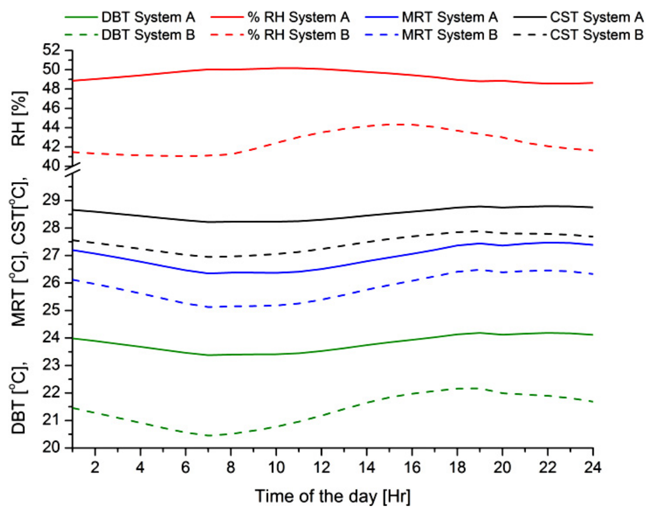


Fig. 5. Temperature and % RH for a single zone room coupled with Systems A and B.

based on a weighted average area-emissivity of all the surfaces in the zone. The DBT for the room coupled with System A is 16.6% and 8.96% more than that of System B at 6.00 h and 18.00 h, respectively. For the same times, CST is also more by 3.64% and 3.21%, respectively when the room is coupled with System A than that of System B. Approximately the same difference is also found with the MRT. The RH is 19.51% more for System A as compared to System B at 6.00 h (Fig. 5).

The effect of warm room environment with System A is shown through the Fanger thermal comfort indices: PMV and PPD and the Pierce PMVET (Fig. 6). The warm temperature with System A is because of lesser cooling capacity. The system

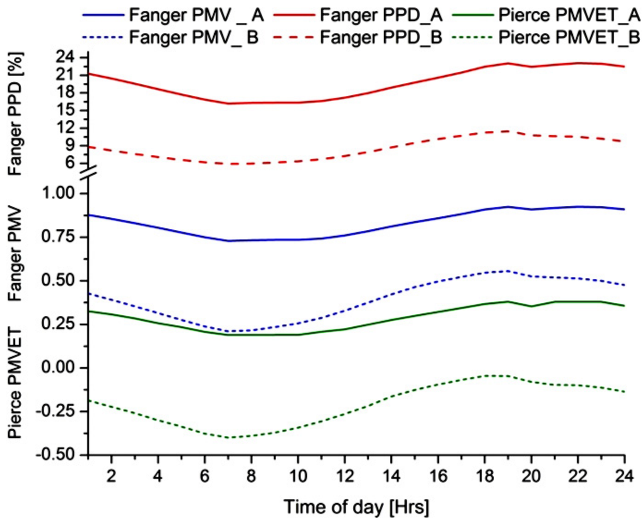


Fig. 6. Thermal comfort indices for a single zone room with Systems A and B.

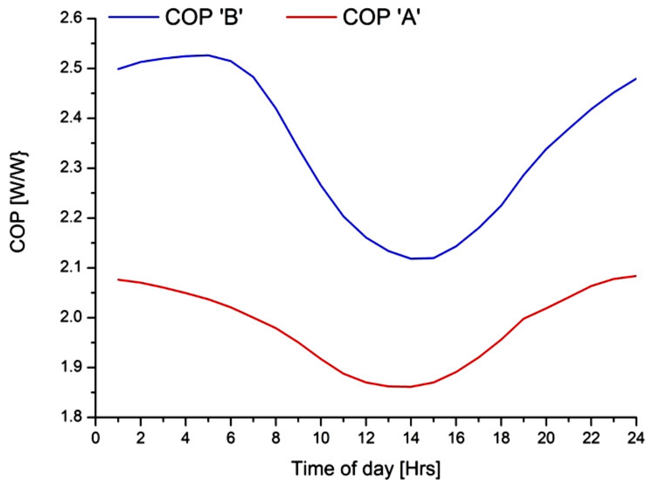


Fig. 8. Power consumption of the CO_2 air conditioner for various timings of the day.

operates at gas cooler pressure different than optimum gas cooler pressure resulting in lower cooling capacity (Fig. 7).

The Fanger PMVs are 200% to 275% more in the morning and 75% to 125% more in the afternoon for the room with System A as compared to the room with System B (Fig. 6). This has resulted in 95% to 133% people being more dissatisfied during the morning and 85% to 127% during the afternoon with System A as compared to System B. The Pierce PMVET values for System A are slightly above '0' indicating warm environment. The room with System B causes slightly cold environment since its Pierce PMEVT is below '0'.

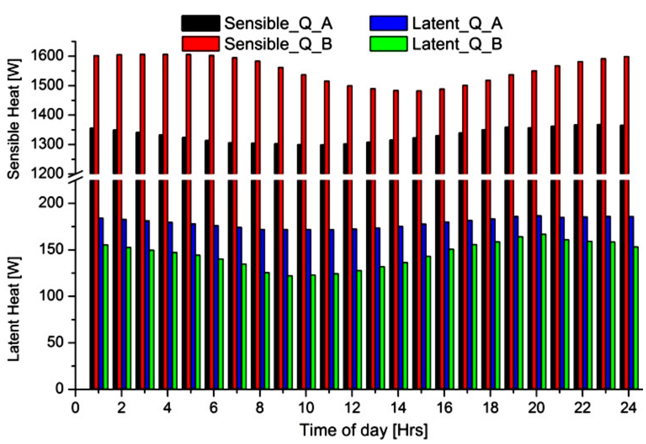


Fig. 7. Cooling capacity of the CO_2 air conditioner for various timings of the day.

System B sets the gas cooler pressure to optimum values resulting in maximum cooling capacity with lower compressor power input. This control strategy has affected the thermal comfort conditions in the single zone room.

The sensible cooling capacity for System B is higher than that of System A in the range 15.62–17.29% in the morning period from 1.00 h to 10.00 h and in the range 11.29–12.69% during afternoon period from 13.00 h to 16.00 h (Fig. 7). However, the latent heat capacity of System A is higher than that of System B in the range 15.56–16.87% and 10.56–12.21% for the morning and afternoon periods, respectively. The higher latent heat for System A is because the suction pressure for System A is lower than that of System B.

The power consumption of System B is higher than that of System A in the range of 0.3–1% for the afternoon period. The higher sensible cooling capacity of System B results in better control on thermal comfort conditions in the room. In the early morning, the COP of System B is higher than that of System A in the range of 20.5–21.56%. During the afternoon, the COP of System B is higher than that of System A in the range of 9–11.34% (Fig. 8). The evaluated Seasonal Energy Efficiency Ratio (SEER) for the DX cooling coil of System A is 2.35 W/W and for System B is 2.85 W/W.

7. Conclusion

The transcritical CO_2 air conditioning systems, System A and System B, coupled with single zone room located in subtropical conditions typically of Pune city (India) in the summer are studied for their energy performance and thermal comfort conditions. The System B controls the gas cooler pressure to achieve the maximum COP and the maximum cooling capacity unlike that achieved by the non-controlled System A. The following conclusions can be drawn from the present simulation studies:

(1) System B has higher total cooling capacity than that of System A in the range 15.78–20.2% for a typical design summer day at tropical conditions.
(2) The Fanger thermal indice PDD shows that 21–24% occupants are dissatisfied with the thermal comfort in a single room with System A. On other hand, 9–11% occupants are dissatisfied with the thermal environment when System B is coupled.
(3) The Pierce two node thermal model parameter PMVET shows that System A is more close to normal condition '0' than System B indicating that the room coupled with System B is slightly more chilled than the room coupled with System A.
(4) For a DX cooling coil, the control of gas cooler pressure in System B results in more SEER than that of System A in the range 8–20%.

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