EXPERIMENTAL ANALYSIS OF FUZZY CONTROLLED ENERGY EFFICIENT DEMAND CONTROLLED VENTILATION ECONOMIZER CYCLE VARIABLE AIR VOLUME AIR CONDITIONING SYSTEM

by

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In the quest for energy conservative building design, there is now a great opportunity for a flexible and sophisticated air conditioning system capable of addressing better thermal comfort, indoor air quality, and energy efficiency, that are strongly desired. The variable refrigerant volume air conditioning system provides considerable energy savings, cost effectiveness and reduced space requirements. Applications of intelligent control like fuzzy logic controller, especially adapted to variable air volume air conditioning systems, have drawn more interest in recent years than classical control systems. An experimental analysis was performed to investigate the inherent operational characteristics of the combined variable refrigerant volume and variable air volume air conditioning systems under fixed ventilation, demand controlled ventilation, and combined demand controlled ventilation and economizer cycle techniques for two seasonal conditions. The test results of the variable refrigerant volume and variable air volume air conditioning system for each techniques are presented. The test results infer that the system controlled by fuzzy logic methodology and operated under the CO_2 based mechanical ventilation scheme, effectively yields 37% and 56% per day of average energy-saving in summer and winter conditions, respectively. Based on the experimental results, the fuzzy based combined system can be considered to be an alternative energy efficient air conditioning scheme, having significant energy-saving potential compared to the conventional constant air volume air conditioning system.

Key words: *energy conservation, thermal comfort, indoor air quality, fuzzy logic, ventilation, inverter air conditioning*

Introduction

The variable refrigerant volume (VRV) systems that vary refrigerant volume are basically large-capacity versions of ductless multisplit air conditioning systems. The VRV systems circulate the refrigerant directly to multiple evaporator units, rather than use water, in contrast to the conventional heating, ventilation, and air conditioning (HVAC) systems, to achieve heat transfer to the conditioning space. Energy-saving can be achieved with a VRV system and it can be attributed to the system's high part-load efficiency. It is fast replacing the traditional chilled water systems owing to its waterless operation, absolute flexibility and energy-saving features. In recent years the HVAC systems for building environments are primarily designed to achieve better thermal comfort and indoor air quality for the indoor occupants with substantial energy-saving to be obtained from the system components, based on the overall performance of the HVAC system. Thermal comfort and indoor air quality inside the building envelope can be obtained by properly monitoring the supply air temperature and supply airflow rate based on the fluctuated thermal load conditions. Variable air volume (VAV) air conditioning (A/C) systems perform better for varying thermal load conditions, and modulate the flow rate of supply air into the conditioned space. An increased number of high rise apartment buildings and rapid increase in land cost paved the way for VRV technology to become increasingly attractive. The revolutionary VRV systems first appeared in Japan in 1982 and are now used throughout the world. The VRV system modulates refrigerant volume according to capacity requirements and this helps reduce the overall system energy consumption. Many research works in HVAC systems involving VRV and VAV A/C systems and their performance in different applications [1-15] have been performed. In order to acquire better thermal comfort, indoor air quality and energy-saving, proper control schemes with respect to the HVAC system, capable of maintaining the supply and indoor temperature set points and indoor humidity within the permissible level, are most essential.

The intelligent fuzzy logic controllers (FLC) are gaining popularity in the field of HVAC to fulfil the increasing demand for control actions posed over the HVAC system components that help accomplish the tasks related to the building A/C requirements. Researchers have discussed the concept of FLC [16-21] dedicated to the control of refrigerant flow and compressor speed in different heating, ventilation, air conditioning and refrigeration (HVAC&R) applications. The distribution of the refrigerant in an A/C system is most significant, as it is associated with the system overall performance in terms of operating efficiency and evaporator capacity. In modern air conditioning applications, electronic expansion valves (EEV) are considered to be an alternative to the conventionally adopted thermostatic expansion valve (TXV). Studies on the refrigerant flow characteristics [22, 23] of refrigerants flowing through EEVs have been reported.

This research work reports on the simulation and experimental analysis of the proposed combined VRV-VAV A/C system using FLC. The proposed A/C system is studied and tested under the fixed ventilation, demand controlled ventilation (DCV) and combined demand controlled ventilation-economizer cycle (DCV-EC) schemes to achieve better thermal comfort, indor air quality (IAQ) and energy-saving for summer and winter design conditions. The comparative results of the proposed A/C system with the conventional constant air volume (CAV) air conditioning system are discussed.

Experimental methodology

The experiment has been carried out to determine the inherent operational characteristics of the combined VRV-VAV multi-zone centralized A/C system using FLC. The schematic representation of the fuzzy logic based A/C control system utilizing the energy efficient VRV-VAV A/C system is shown in fig. 1. VAV A/C software laboratory building was considered for the simulation situated at Anna University, Chennai, India. The building zone was decided to be 33 × 8.5 × 3 m in dimension. The building has seven glazed windows on the east facade and four glazed windows on the west wall, each having dimensions of 0.91 × 1.83 m and a door with a dimension of 0.91 × 2.13 m. The construction materials and properties were selected according to the ASHRAE (American Society of Heating Refrigeration and Air Conditioning Engineers) handbook [24].

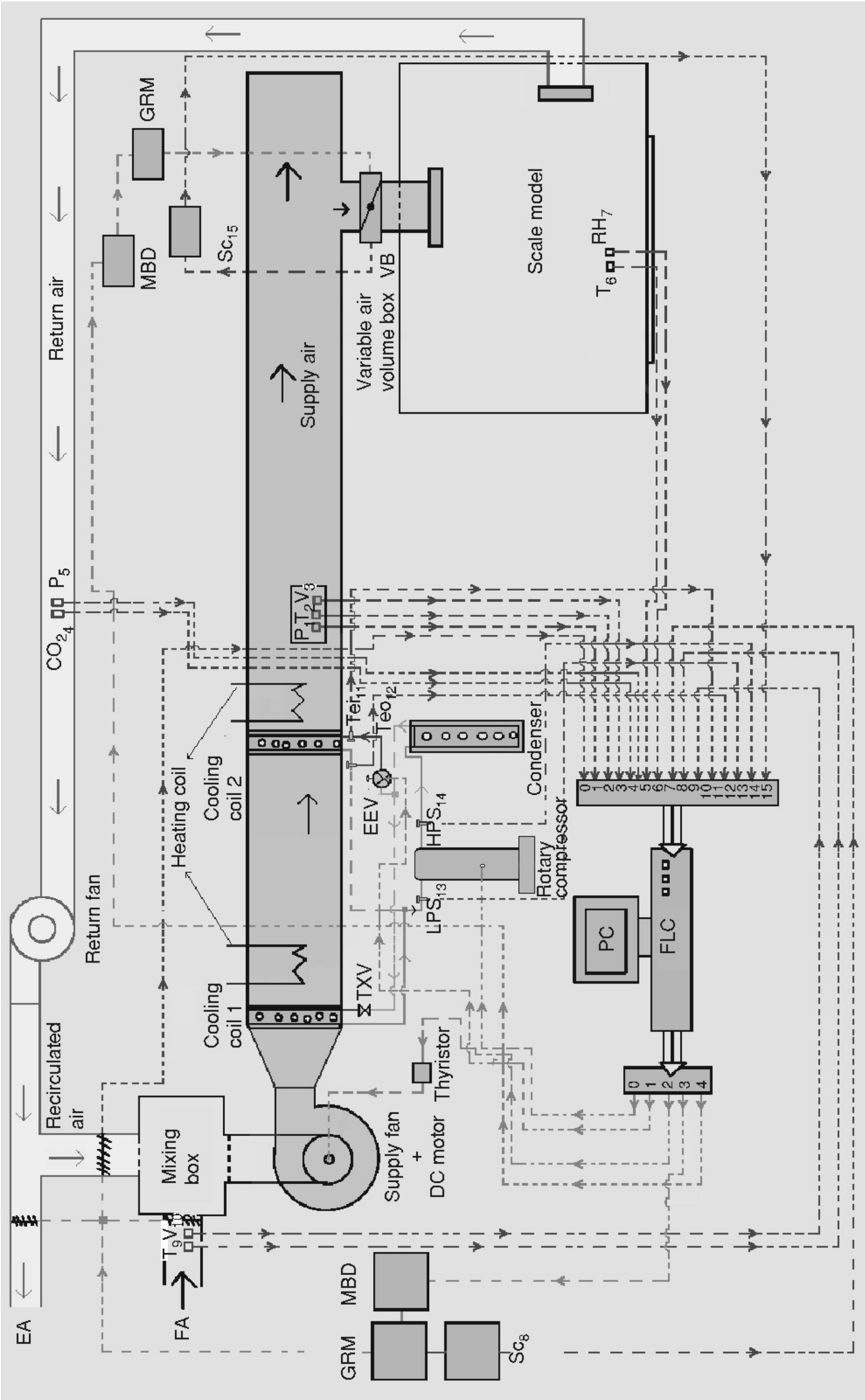


Figure 1. Schematic representation of FLC based VRV-VAV A/C system

The zone has 45 computers on each side and total occupancy of 95 people. A scale model having a dimension of 1.48 × 1.75 × 0.6 m for the building and an air handling system with a FLC unit has been constructed in the refrigeration and A/C laboratory at Anna University, that confines to the numerical values obtained for the building.

This model is geometrically similar to the building in all details that are important for the volume flow, the energy flow, and the contaminant flow. The key components present in the scale model are, a thermally insulated air conditioned room model equipped with temperature and relative humidity sensors, inverter driven variable speed rotary compressor, EEV, cooling coil, supply air fan, return air fan, velocity sensor, silicon-based non dispersive infrared ray (NDIR) CO_2 sensor fixed in the return duct, temperature sensor, pressure sensors for both air side and refrigerant side, FLC, geared motor (GRM), MOSFET based driver (MBO), signal conditioner (SC), fresh air damper, return air damper, exhaust damper, actuator and mixing box. An EEV was used to control the degree of superheat at the evaporator outlet. The working fluid used in the experiment was R22. The experimental combined VRV-VAV A/C system has been fully instrumented. All measurements were computerized, in order that all the measured data can be recorded for a subsequent analysis.

The objective of the experiment was to establish the inherent operational characteristics of the VRV-VAV A/C system for a year-round application based on seasonal variations, when the compressor speed can be modulated for varying conditions of the suction pressure and mass flow rate of the refrigerant. For a fluctuation in thermal load observed in the space to be cooled, the refrigerant suction enthalpy was modulated and for this variation in enthalpy the corresponding mass flow rate of the refrigerant supplied to the evaporator through EEV was varied and by utilizing a FLC, the temperature of the supply air was maintained around the set point precisely. The temperature set point was kept at 13 ºC and the room set points were 24 ºC and 50% relative humidity (RH). The experiment was performed for both summer and winter weather conditions and the outdoor temperature variations were selected according to the Indian Society of Heating Refrigeration and Air Conditioning Engineers (ISHRAE) standards for Chennai, India. The layout of the selected building is shown in fig. 2.

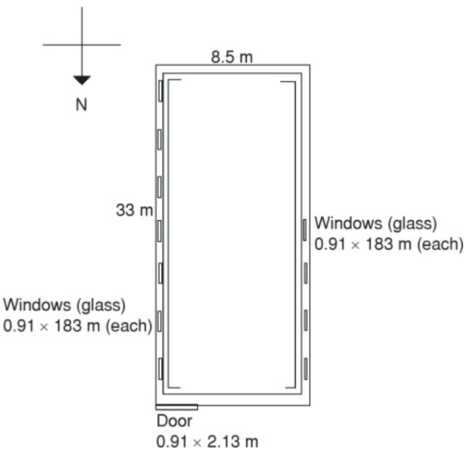


Figure 2. Layout of the selected test building

In the experiment, the refrigerant mass flow rate was modulated by varying the speed of the compressor according to the supply air flow rate required to offset the cooling load which prevailed inside the conditioned space. The DCV technique is incorporated in the experiment adjusts the outside ventilation air, based on CO_2 concentration the occupants generate on hourly basis. The fresh air damper opening and the supply air fan are suitably controlled by FLC. In the economizer cycle (EC) as the outdoor air temperature was lesser than the return air temperature; 100% fresh air was taken into the conditioned space to achieve the desired energy-saving and IAQ. During the EC, the refrigerant plant was turned off and the fresh air damper was set at its full opening position. This helped in conserving a substantial quantity of the total energy consumed. The combined action of the DCV and EC was also investigated and the tested results are presented. Table 1 indicates the transducers' specification and the un-

Table 1. Transducer specifications

Transducers	Range	Uncertainty
Pressure sensor (LP) [psig]	50-80	±0.03
Pressure sensor (HP) [psig]	227-275	±0.15
Thermistor (SP) [ºC]	0-14	±0.03
Thermistor (DP) [ºC]	45-95	±0.15
Velocity sensor [m/s]	0.15-10	±0.05
Static pressure sensor [bar]	0-1	±0.1
CO_2 sensor (GMD20) with display and relay [ppm]	0-2000	±0.2
Relative humidity sensor [%]	0-100	±0.03
Temperature sensor with signal conditioner [ºC]	0-100	±0.5

Table 2. Input parameters pertaining to the analysis

Input parameter	Value
Building data	
Area [m^2]	281
Volume [m^3]	845
Glazing area [m^2]	11.7
Occupancy	95
Room conditions	
– Summer – DBT [ºC]	24
RH [%]	50
– Winter – DBT [ºC]	22
Internal heat gains	
Lighting [W/m^2]	20
Small power load [W/m^2]	30
Occupancy	
– Sensible [W/person]	70
– Latent [W/person]	45
Overall heat transfer coefficient [W/m^{2}K]	
Walls and partitions	1.31
Floor and ceiling	1.1
Glazing	2.2

certainties and tab. 2 represents the input parameters considered for the analysis. The photographic view of the experimental set up is shown in fig. 3.

Fuzzy logic controller design

The FLC is a kind of fuzzy rule-based system composed of a knowledge base (KB) that contains the information used by the proficient operator in the form of linguistic control rules. In the fuzzification process, the crisp values of the input variables are transformed into fuzzy sets that will be used in the fuzzy inference process. The inference system uses the fuzzy values from the fuzzification interface and the information from the KB to perform the reasoning process. The fuzzy inference process essentially operates on the IF-THEN rules (which are a conditional statement) that define the system behav-

Figure 3. Photographic view of FLC based VRV-VAV A/C system

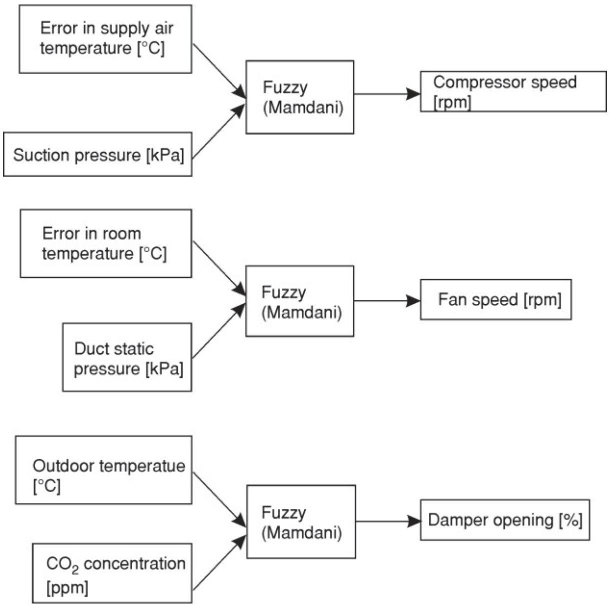


Figure 4. Structure of FLC

ior. Defuzzification takes the fuzzy action from the inference process and translates it into crisp values for the control variables. The structure of the FLC is depicted in fig. 4. The FLC design utilized in this work included multi-input and multi-output parameters to control the VRV-VAV A/C system effectively. The error is expressed as the difference between the set point value and the actual/present value sensed. The error in the supply air temperature and the suction pressure of the compressor were considered to be input variables that constitute the output variable in the form of modulated compressor speed.

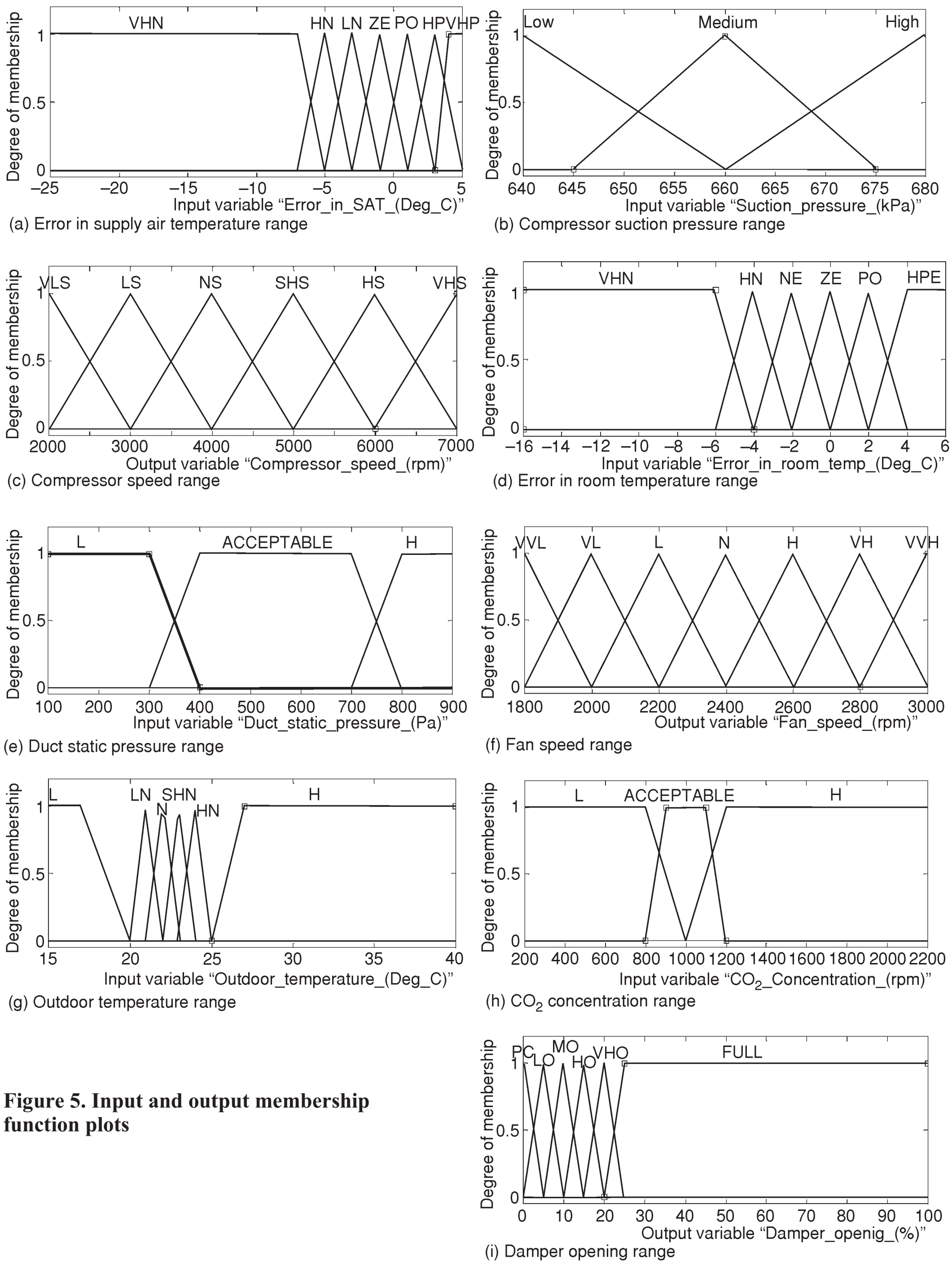
Based on the descriptions of the input and output variables defined, fuzzy rule statements were generated to achieve better speed control of the compressor. For instance, the first fuzzy set for compressor speed control, supply fan speed control, and the damper position control are given by:

– *for compressor speed control*
IF (Error in supply air temperature is low negative [LN] and Suction pressure is medium), THEN (Compressor speed is high speed [HS])
Similarly, IF (Error in supply air temperature is high positive [HP] and Suction pressure is low)
THEN (Compressor speed is low speed [LS])
The input and output variables are divided into ten (a = 1-7), three (b = 1-3) and six (c = 1-6) terms, respectively. Simple triangular and trapezoidal membership functions were used in this paper because of simplicity and their inherent feature that matched well with the control strategy of the proposed VRV-VAV A/C system.

– *for supply fan speed control*
IF (Error in the room temperature is positive [PO] and duct static pressure is ACCEPTABLE)
THEN (Fan speed is normal [NOR])
Similarly, IF (Error in the room temperature is high negative [HN] and duct static pressure is LOW), THEN (Fan speed is very high [VH])
The input and output variables of the second fuzzy set are divided into six (p = 1-6), three (q = = 1-3) and seven (s = 1-7) terms, respectively.

– *for damper position control*
IF (Outdoor temperature is high [HIGH] and CO_2 concentration is high [HG]),
THEN (Damper opening is very high opening [VHO])
Similarly, IF (Outdoor temperature is slightly high negative [SHN] and CO_2 concentration is ACCEPTABLE), THEN (Damper opening is FULL).

A similar set of rules based on the procedure explained above were framed for each membership function terms in the input and output variables involved in the proposed control strategy, that enabled the investigation of the inherent operational characteristics of the proposed VRV-VAV A/C system based on the fuzzy logic controller. A graphical illustration of the membership functions and their ranges for the input and output variables are shown in fig. 5.



(a) Error in supply air temperature range
(b) Compressor suction pressure range
(c) Compressor speed range
(d) Error in room temperature range
(e) Duct static pressure range
(f) Fan speed range
(g) Outdoor temperature range
(h) CO_2 concentration range
(i) Damper opening range

Figure 5. Input and output membership function plots

These ranges were defined in the interval of –25 to 5 ºC for error in supply air temperature, 640 to 680 kPa for suction pressure, 2000 to 7000 rpm for compressor speed, –16 to 6 ºC for error in room air temperature, 100 to 900 Pa for duct static pressure, 1800 to 3000 rpm for fan speed, 15 to 40 ºC for outdoor air temperature, 200 to 2200 ppm for CO_2 concentration, and 0 to 100% for damper opening position. The summary of fuzzy linguistic terms used in this work is represented in the appendix. The fuzzy logic control methodology was designed, based on how the system will respond to a corresponding change in input variables. By using the MATLAB-Simulink environment, the designed FLC can be linked with the simulated model to evaluate the system performance. In this work, the centroid method was used in the defuzzification stage to convert the fuzzy variable back to the output variable that can be varied according to the rules. The mathematical relation behind the centroid method is given as:

$$Z^* \quad \frac{\mu_\mathrm{c}(Z)Z\mathrm{d}Z}{\mu_\mathrm{c}(Z)\mathrm{d}Z} \tag{1}$$

Based on the fuzzy rules generated, the FLC utilized in this research work was able to control the speed of the variable speed rotary compressor and the supply air fan for varying thermal loads inside the conditioned space, with the supply air temperature being maintained at 13 ºC and relative humidity around 50%. During summer and winter conditions, the outdoor air damper position was altered accordingly based on the CO_2 contamination present inside the conditioned space and the outdoor air temperature as well. Altogether, the proposed system acquired good thermal comfort, indoor air quality and energy-saving with the use of fuzzy logic methodology being developed in this research work.

HVAC system simulation

The proposed VRV-VAV A/C system was simulated using the MATLAB-Simulink [25] and the outdoor temperature variation is taken as per the meteorological department for the month of May and December, since May and December record the maximum average and minimum average temperatures throughout the year.

The summer and winter outdoor air temperature variation for 24 hours and the occupancy load pattern for the software laboratory considered, are represented in figs. 6 and 7, respectively.

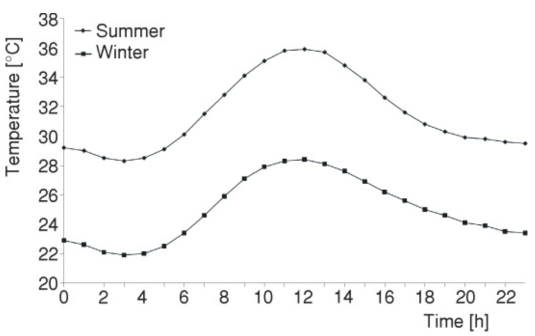


Figure 6. Variation of outdoor temperature for summer and winter

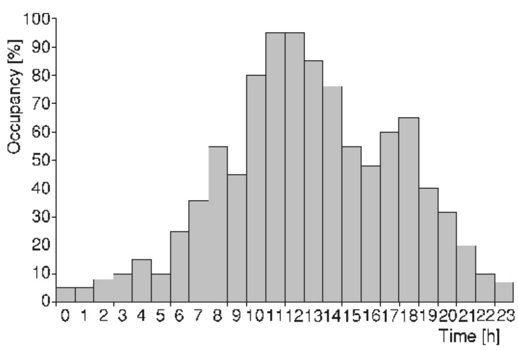


Figure 7. Occupancy load pattern

Mathematical models

The mathematical models considered for the simulation work are given below.

Variable speed rotary compressor (VSC)

The mass flow rate of the refrigerant entering compressor can be calculated as given by eq. (2):

$$m_{\mathrm{r}} \quad \frac{\eta_{\mathrm{V}} n V_{\mathrm{th}}}{v_{\mathrm{in}}} \tag{2}$$

where v_{in} is the specific volume of refrigerant, η_{V} – the volumetric efficiency, and the volume flow rate of refrigerant is given by:

$$V_{\mathrm{th}} \quad 60\pi R^2 l \frac{e}{R} \; 2 \quad \frac{e}{R} \tag{3}$$

The compressor work can be represented by the equation:

$$W_{\mathrm{com}} = h_{\mathrm{d}} - h_{\mathrm{s}} \tag{4}$$

where h_{s} and h_{d} are the specific enthalpy of refrigerant at compressor suction and discharge.

Electronic expansion valve

The refrigerant mass flow rate through the EEV is denoted by:

$$m_{\mathrm{e}} \quad Aev\xi\sqrt{\frac{(P_{\mathrm{con}} \quad P_{\mathrm{ev}})}{V_{\mathrm{c}}}} \tag{5}$$

Evaporator model

The mixed enthalpy of the refrigerant can be found by:

$$h_{\mathrm{s}} \quad \frac{h_{\mathrm{v}} m_{\mathrm{e\,1}} \quad h_{\mathrm{v}} m_{\mathrm{e\,2}}}{m_{\mathrm{e\,1}} \quad m_{\mathrm{e\,2}}} \tag{6}$$

$$m_{\mathrm{com}} = m_{\mathrm{e\,1}} + m_{\mathrm{e\,2}} \tag{7}$$

where $m_{\mathrm{e\,1}}$, and $m_{\mathrm{e\,2}}$ are the mass flow rates of the refrigerant flowing through the corresponding evaporator.

Fan model

The power consumed by the fan can be calculated by using the polynomial equation given by:

$$W_{\mathrm{fan}} = FMP\;\{a + b[PLR(t)] + c[PLR(t)]^2 + [PLR(t)]^3\} \tag{8}$$

where a, b, c, and d are constants, PLR is the part load flow ratio, and FMP – the fan motor power.

Building model

Mathematical energy balance equations were framed for all the heat load components in the zones. The overall heat transfer coefficient U and thermal capacitance C were taken from the building standards as per ASHRAE. All the energy balance equations were simplified in order to make the equations in matrix form under the State-Space notation. The notation for the State-Space model is presented by:

$$\frac{\mathrm{d}T}{\mathrm{d}t} \quad A1\mathbf{T}(t) \quad B1\bar{u}(t) \tag{9}$$

$$\begin{bmatrix} \dfrac{\mathrm{d}T_\mathrm{w}}{\mathrm{d}t} \\ \dfrac{\mathrm{d}T_\mathrm{f}}{\mathrm{d}t} \\ \dfrac{\mathrm{d}T_\mathrm{c}}{\mathrm{d}t} \\ \dfrac{\mathrm{d}T_\mathrm{ai}}{\mathrm{d}t} \end{bmatrix} \quad \begin{bmatrix} \dfrac{A_\mathrm{w}(U_\mathrm{wi} \quad U_\mathrm{wo})}{C_\mathrm{w}} & 0 & 0 & \dfrac{A_\mathrm{w}U_\mathrm{wi}}{C_\mathrm{w}} \\ 0 & \dfrac{A_\mathrm{f}U_\mathrm{f}}{C_\mathrm{f}} & 0 & \dfrac{A_\mathrm{f}U_\mathrm{f}}{C_\mathrm{f}} \\ 0 & 0 & \dfrac{A_\mathrm{c}U_\mathrm{C}}{C_\mathrm{c}} & \dfrac{A_\mathrm{c}U_\mathrm{c}}{C_\mathrm{c}} \\ \dfrac{A_\mathrm{w}U_\mathrm{wi}}{C_\mathrm{a}} & \dfrac{A_\mathrm{f}U_\mathrm{f}}{C_\mathrm{a}} & \dfrac{A_\mathrm{c}U_\mathrm{c}}{C_\mathrm{a}} & \dfrac{1}{C_\mathrm{a}}(A_\mathrm{g}U_\mathrm{g} \quad A_\mathrm{w}U_\mathrm{wi} \quad A_\mathrm{f}U_\mathrm{f} \quad A_\mathrm{c}U_\mathrm{c}) \end{bmatrix} \begin{bmatrix} T_\mathrm{w} \\ T_\mathrm{f} \\ T_\mathrm{c} \\ T_\mathrm{ai} \end{bmatrix}$$

$$\begin{bmatrix} 0 & 0 & 0 & \dfrac{A_\mathrm{w}U_\mathrm{wo}}{C_\mathrm{w}} \\ 0 & 0 & \dfrac{p}{C_\mathrm{f}} & 0 \\ 0 & 0 & 0 & 0 \\ \dfrac{1}{C_\mathrm{a}} & \dfrac{1}{C_\mathrm{a}} & 0 & \dfrac{A_\mathrm{g}U_\mathrm{g}}{C_\mathrm{a}} \end{bmatrix} \begin{bmatrix} Q_\mathrm{p} \\ Q_\mathrm{i} \\ Q_\mathrm{s} \\ T_\mathrm{ao} \end{bmatrix} \tag{10}$$

where A1, and B1 are the coefficients of matrices, ū is the input vector, and **T** is the matrix of temperatures.

Temperatures of walls (T_w), ceiling (T_c), floor (T_f), and room air (T_ai) were selected as the state variables in **T**(t) and input variables in ū(t) include cooling energy supplied by the plant (Q_p), internal heat gains (Q_i), solar radiation through windows (Q_s), and outside air temperature (T_ao).

Well mixed model

To dilute sources from both the building and its occupants, the design ventilation rate (DVR) equation that contains people and floor area components is given by:

$$DVR = V_\mathrm{P} + V_\mathrm{B} \tag{11}$$

Based on the well mixed condition and applying the mass balance on the contaminant, the differential equation relating contaminant concentration and time is given by:

$$N(t) = GP(t) \tag{12}$$

$$\frac{\mathrm{d}c}{\mathrm{d}t} \quad \frac{(c_\mathrm{S} \quad c)Q_\mathrm{S}}{v} \quad \frac{GP(t)}{v} \tag{13}$$

where v is the volume of the room, Q_S – the ventilation quantity, N – the contaminant concentration rate, P – the occupancy, G – the CO_2 generation rate per person, and c – the contaminant concentration.

Damper model

Dampers regulate the flow of air inside the room which is controlled by the FLC controllers. The single blade type dampers are used for simulation. The quantity of air flow rate is given by:

$$Q_{out} = C_{d(\theta)} \mathrm{tg}\theta A \left[2\frac{P_i - P_o}{\rho} \right]^{1/2} \tag{14}$$

The area of the damper is given by:

$$A = DW \tag{15}$$

where, θ is the damper angle, D – the depth, and W – the width

Experimental results and discussion

The inherent operational characteristics of the combined VRV-VAV centralized A/C system for the scale model that was developed based on the FLC, with an inverter driven compressor and variable speed supply air fan, are presented in this section. A variety of tests were performed to determine the operational characteristics of the VRV-VAV A/C system. Under different operating conditions, the performance of the existing system was compared with that of the conventional CAV A/C system under three categories:
– thermal comfort,
– IAQ, and
– energy conservation.

The results presented involve the parameters that have a greater influence on the operating conditions of the system.

The influence of the supply air flow rate on the refrigerant mass flow rate

The variation of the refrigerant mass flow rate (MFR) corresponding to the variation of the supply air flow rate (AFR) for summer and winter design conditions is presented in fig. 8. The supply AFR requirement in winter is observed to be lesser than that in summer. The test result infers that in summer, the supply AFR varies between 9.9 and 19.6 m^3/min. whereas during winter design conditions it varies between 8.4 and 19.2 m^3/min. The reduced supply AFR contributes to energy-saving on the supply air fan. Similarly, the refrigerant MFR in the fixed ventilation and DCV mode is found to modulate between 0.020 and 0.043 and 0.016 to 0.042 kg/s, respectively, for summer and while operating under winter design conditions it yields a minimum and maximum refrigerant MFR of 0.015 and 0.038, and 0.013 and 0.037 kg/s for fixed ventilation and DCV schemes, respectively.

The test result also infers that the refrigerant MFR under the DCV mode is much less than that of the fixed ventilation mode. This trend is exhibited by the proposed system as it is designed using fuzzy logic wherein the supply AFR requirement is determined based on the CO_2 based ventilation method, and the compressor pumps the refrigerant for the corresponding sup-

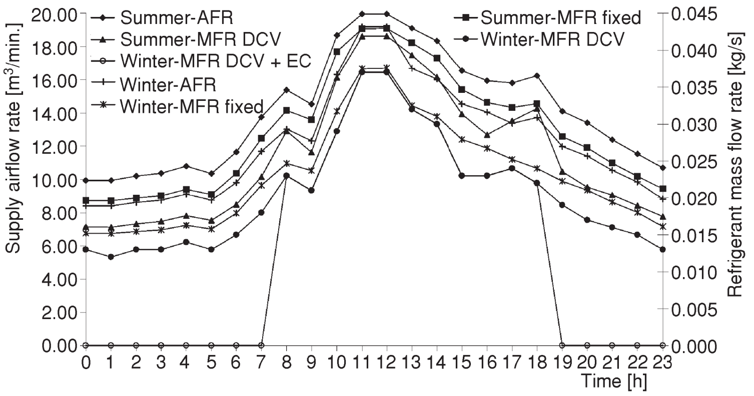


Figure 8. Variation of supply airflow rate and refrigerant mass flow rate for summer and winter design conditions

ply AFR to offset the thermal load from the conditioned space. Under the combined DCV-EC cooling mode operation, the proposed system modulates the refrigerant flow that vary from 0 to 0.037 kg/s , and since the compressor is turned off during the EC, the overall energy consumed by the proposed system is reduced extensively. Thermal comfort and IAQ are well achieved in the present VRV-VAV A/C system as the supply AFR is continuously altered depending on the thermal load existing inside the conditioned space.

The influence of the refrigerant mass flow rate on compressor speed

The MFR of the refrigerant has a direct relation to the speed of the compressor. Figures 9 and 10 refer to the variation of compressor speed with the modulated refrigerant MFR for summer and winter conditions, respectively. The compressor speed for the fixed ventilation scheme is observed to vary from 3100 to 6760 rpm during summer and from 2400 to 5915 rpm during winter conditions, respectively. The DCV scheme when applied to summer and winter conditions, infers that the compressor speed varies from 2505 to 6605 rpm, and from 1975 to 5825 rpm, respectively. In the combined DCV-EC scheme, the compressor speed modulates from 0 to 5825 rpm. Since a distinct variation is observed in the mass flow rates of the refrigerant between fixed and demand controlled ventilation techniques utilized in the present system, the energy consumed by the compressor is substantially reduced and accounts for total energy conservation.

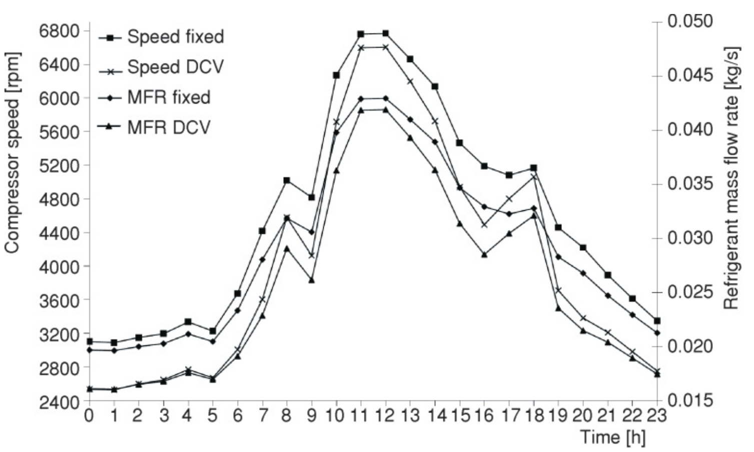


Figure 9. Variation of compressor speed for summer

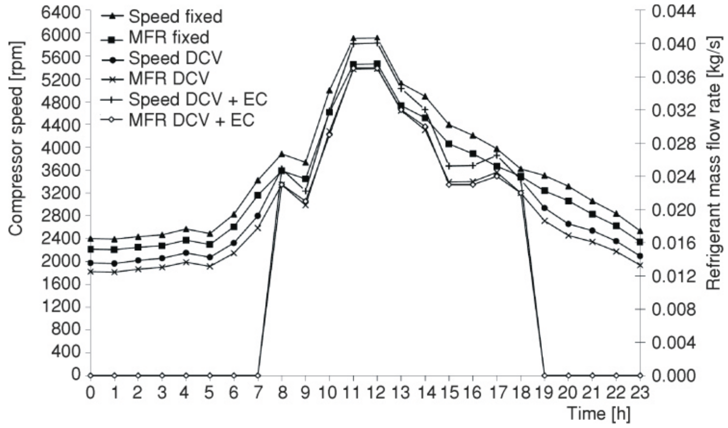


Figure 10. Variation of compressor speed for winter

The variation of supply air temperature, relative humidity and static pressure

The fuzzy controller effectively maintained the supply air temperature around 13 °C and the relative humidity averaged to 50%. Figures 11 and 12 show the supply air temperature and relative humidity varying with respect to time and that is directly related to the cooling load prevailing on the cooling coil for the respective time interval considered. Based on the experimental result, as the supply air temperature and relative humidity is maintained almost constant, it inevitably means that, the cooling load prevailing in the building model is also controlled satisfactorily.

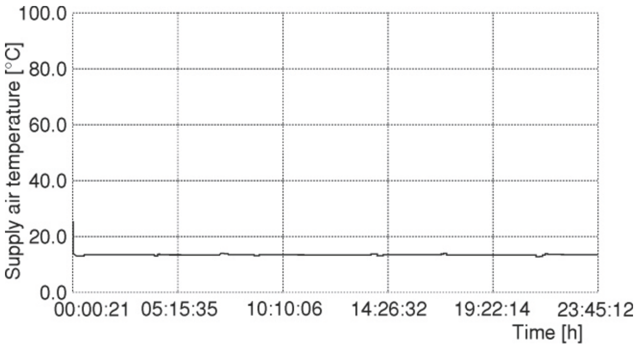


Figure 11. Variation of supply air temperature

The variation of duct static pressure is depicted in fig. 13. The duct static pressure is observed to fluctuate under varying thermal load conditions. As the static pressure in the duct tends to increase or decrease, the fan speed is modulated and controlled accordingly, using the FLC. The experimental result infers that the duct static pressure is observed to vary from 410 to 680 Pa. By maintaining the duct static pressure within this range, the proposed system achieved better speed control and energy-saving in the supply air fan that lead to proper air distribution inside the conditioned space being considered.

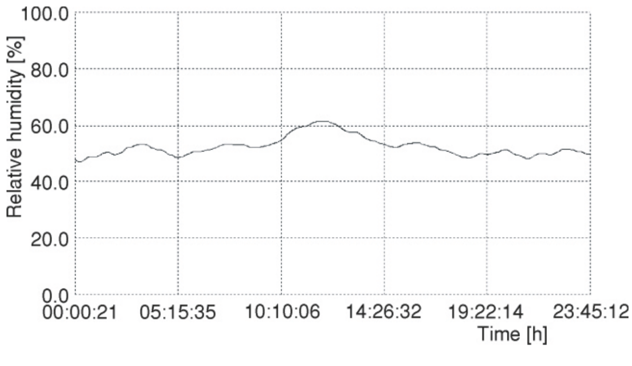


Figure 12. Variation of relative humidity

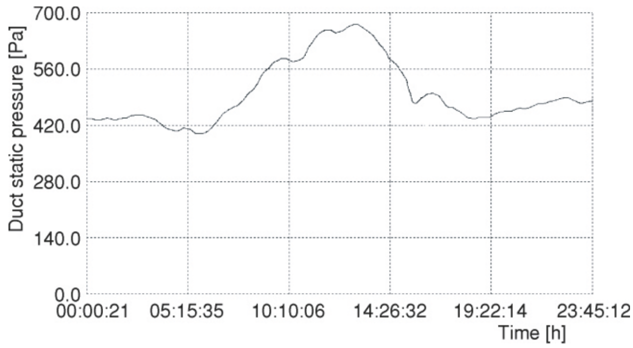


Figure 13. Variation of duct static pressure

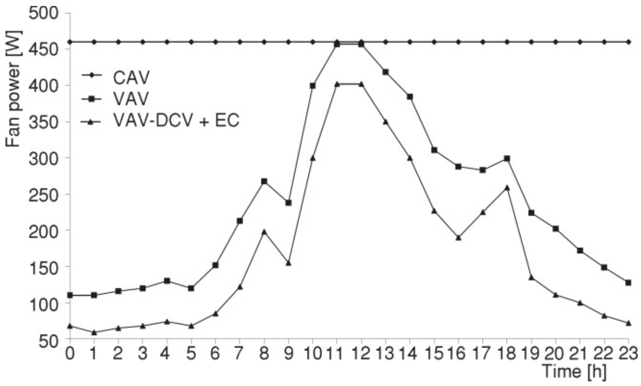


Figure 14. Variation of fan power for summer and winter design conditions

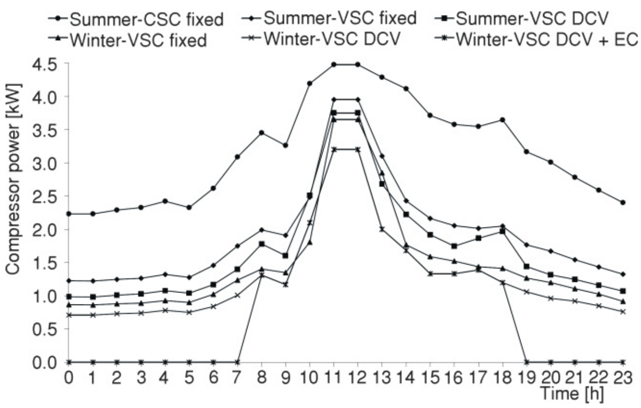


Figure 15. Variation of compressor power for summer and winter design conditions

The effect of fan power on energy conservation

The variation of supply air fan power for summer and winter design conditions is shown in fig. 14. Based on the static pressure present in the supply air duct, the fan speed is considerably modulated to offset the cooling load. In the present system, for summer design conditions, the supply air fan works over a wide range of power inputs that vary from 110 to 456 W. The test result infers that the supply fan incorporated in the proposed system shows significant reduction in power consumption since the supply fan delivers modulated air volume into the conditioned space. The same trend is observed for winter design conditions as well, where the range of power consumed by the supply air fan under the DCV-EC mode of ventilation is found to vary between 68 and 402 W. As the working of the supply fan is governed by fuzzy linked CO_2 based mechanical ventilation scheme, the supply fan is capable of delivering the required airflow rate into the conditioned space that helps achieve thermal comfort and energy-saving, compared to the conventional CAV system operation.

The effect of compressor power on energy-saving

The variation of compressor power for summer and winter design conditions is depicted in fig. 15. In the VRV-VAV system, the speed of the variable speed compressor (VSC) varies according to load fluctuations and the power input to the compressor also varies considerably. The power consumed by the VSC is lesser than that of the constant speed compressor (CSC).

In the present system, the FLC modulates the power consumed by the compressor based on the cooling load prevailing inside the conditioned space. Figure 16 infers that during summer design conditions under the fixed and DCV schemes, the VSC operates from 1.23 to 3.95 kW and from 0.98 to 3.75 kW for fixed ventilation and DCV techniques, respectively. Similarly, for winter design conditions it is observed that the power consumed by the VSC varies between 0.87 and 3.65 kW under fixed ventilation, and from 0.7 to 3.2 kW and from 0 to 3.2 kW for DCV combined DCV and EC modes of ventilation, respectively. The power consumed by the compressor is found to be less during winter due to the reduced building thermal load conditions, and the compressor is turned off in the economizer cooling cycle.

Energy-saving potential of VRV-VAV A/C system

The energy savings characteristics of the combined VRV-VAV A/C system is evaluated based on the test result as depicted in fig. 16. The test result illustrates that in summer conditions, the per day average energy-saving potential of the proposed system utilizing fixed ventilation and DCV, is expected to achieve 30 and 37%, respectively.

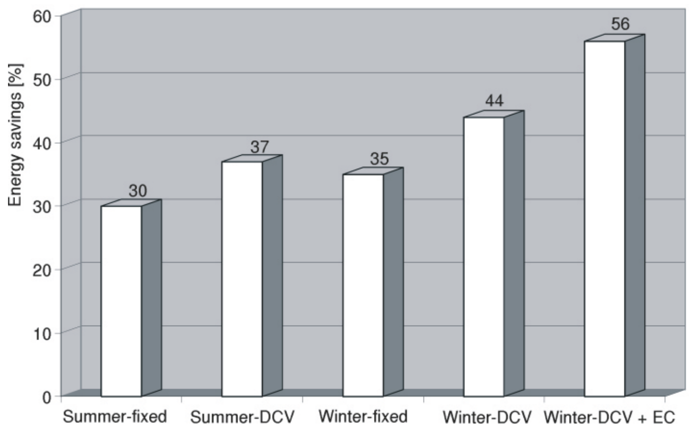


Figure 16. Energy-saving potential of VRV-VAV A/C system

The increase in energy-saving is because of the influence of the required fresh air quantity intake based on the occupancy level and the corresponding CO_2 concentration. The system, when operated under winter design conditions, yields the per day average energy-saving potential of 35, 44, and 56% for the fixed DCV and DCV-EC schemes, respectively. For the proposed system, a significant increase of energy-saving is observed in the case of the combined DCV-EC mode of ventilation scheme, which is because of turning off the compressor and operating only the supply air fan to deliver the required quantity of fresh air while the outdoor temperatures are low enough to achieve better thermal comfort and IAQ. Based on the experimental result it is observed that, the fuzzy based combined VRV-VAV A/C system can be considered to be an alternative energy efficient air conditioning scheme having significant energy savings potential compared to that of the conventional CAV A/C systems.

Conclusions

Compact and flexible A/C systems are being installed in applications ranging from domestic to commercial outlets. However, a viable energy efficient technology is needed to conserve energy as well as to achieve better human comfort. The VRV system is considered to be one of the most promising energy-saving technologies gaining popularity in recent years. In this study, the inherent operational characteristics of the combined VRV-VAV A/C system utilizing the FLC methodology were investigated. In order to assess the benefits of the VRV-VAV A/C system under the cooling mode in terms of energy conservation for both summer and winter design conditions, the system was tested under three different ventilation techniques. The test result infers that the VRV-VAV A/C system controlled by the fuzzy logic methodology effectively maintained the supply air temperature close to 13 °C as well as the room air temperature around 24 °C precisely, and the relative humidity that averaged around 50% which can be expected to achieve better thermal comfort inside the conditioned space. The test results project that for varying occupancy levels the system was capable of maintaining a better IAQ in the sense that, by detecting the CO_2 concentration present inside the conditioned space the required fresh air was delivered and monitored as well.

Based on the experimental results on energy conservation it is obvious that for summer design conditions, the proposed system controlled by fuzzy logic was capable of conserving 30 and 37% of per day average energy in fixed and DCV modes, respectively. Similarly, for winter design conditions the proposed system operated under fixed ventilation, DCV, and combined DCV-EC scheme yields 35, 44, and 56% of enhanced per day average energy savings, respectively. Based on the experimental result it is concluded that the proposed fuzzy based combined VRV-VAV A/C system can be considered to be an alternative energy efficient A/C scheme having significant energy-saving potential compared to that of the conventional CAV A/C system. Also the test results prove that the VRV-VAV A/C system is well suited for seasonal variations. Although conventional controllers are preferred widely, the advantageous and intelligent FLC methodology was utilized in this study for controlling the system parameters precisely for various operational conditions. The test results show that the fuzzy control methodology and algorithm developed are feasible.

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Nomenclature

A – heat transfer area, [m^2]
C_p – specific heat, [kJkg^{-1}K^{-1}]
C_d – coefficient of airflow, [–]
e – eccentricity, [m]
h – enthalpy, [kJkg^{-1}]
l – axial length of cylinder, [m]
m – refrigerant mass flow rate, [kgs^{-1}]
n – rotational speed of compressor, [rps]
P_{con} – condenser pressure, [Pa]
P_{ev} – evaporator pressure, [Pa]
R – radius of cylinder, [m]
U – overall heat transfer coefficient, [kWm^{-2}K^{-1}]
V_{th} – volume flow rate, [m^{3}s^{-1}]
v – specific volume, [m^{3}kg^{-1}]
W_{com} – specific work, [kJkg^{-1}]

Greek symbols

ρ – density, [kgm^{-3}]
ξ – valve flow coefficient, [–]
η_V – volumetric efficiency, [–]

Subscripts

c – condenser
com – compressor
e – evaporator
p – pressure
r – refrigerant
th – theoretical

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Abbreviations

AFR – air flow rate
A/C – air conditioning
CAV – constant air volume
CSC – constant speed compressor
DBT – dry bulb temperature
DCV – demand controled ventilation
DCV-EC – demand controled economizer cycle
DVR – design ventilator rate
EA – exit air
EC – economizer cycle
EEV – electronic expansion valve
FA – fresh air
FC – fully closed
FLC – fuzzy logic controler
GRM – geared motor
H – high
HN – high negative
HO – high open
HP – high positive
HPE – high positive error
HS – high speed
HVAC – heating, ventilation, and air conditioning
HVAC&R – heating, ventilation, air conditioning and refrigeration
IAQ – indor air quality
KB – knowledge base
L – low
LN – low negative
LO – low open
LS – low speed
MBD – MOSFET based driver
MFR – mass flow rate
MO – medium open
N – normal
NDIR – non dispersive infrared ray
NE – negative error
NS – normal speed
PO – positive
RH – relative humidity
SC – signal conditioner
SHN – slightly high negative
SHS – slightly high speed
TXV – thermostatic expansion valve
VAV – variable air volume
VB – variable air valume box
VH – very high
VHN – very high negative
VHO – very high open
VHP – very high positive
VHS – very high speed
VL – very low
VLS – very low speed
VRV – variable refrigerant volume
VSC – variable speed compresor
VVH – very very high
VVL – very very low
ZE – zero error

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