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Experimental Analysis of Energy Efficient Building Air Conditioning System Using Fuzzy Logic Controller

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Abstract – The present work is focused on investigating the thermal comfort and indoor air quality (IAQ) in buildings through the use of energy efficient air conditioning (A/C) system. In this context, a combined variable air volume (VAV) and variable refrigerant volume (VRV) system is developed and tested with different ventilation strategies for summer and winter design conditions. The proposed system is controlled by the intelligent fuzzy logic controller that enhanced the overall system performance. The proposed system is tested under fixed ventilation, demand controlled ventilation (DCV) and combined DCV and economizer cycle (EC) ventilation that ensured better indoor thermal comfort and IAQ without compromising on the energy efficiency. The test results infer that the proposed air conditioning system controlled by fuzzy logic methodology yield a maximum of 34% and 52% of per day energy savings in summer and winter design conditions respectively. The test results for each technique in terms of thermal comfort, IAQ and energy savings potential are presented.

Keywords – Energy conservation, fuzzy logic, thermal comfort, VRV-VAV air conditioning.

1. INTRODUCTION

The revolutionary variable refrigerant volume systems first appeared in Japan in 1982 and are now used throughout the world. Variable refrigerant volume (VRV) systems circulate refrigerant directly to multiple evaporator units, rather than using water, as in contrast to conventional heating, ventilation and air conditioning (HVAC) systems, to achieve heat transfer to the conditioning space. It is fast replacing the traditional chilled water systems owing to its waterless operation, absolute flexibility and energy saving features. Energy savings can be obtained with a VRV system and performs better under full load and part load conditions. In modern days due to the increase of building construction with rapid increase in land cost worldwide, the variable VRV technology have become increasingly attractive among the clients, architects and engineers.

The VRV system modulates refrigerant volume according to capacity requirements. Many researchers have discussed the concept of fuzzy logic control dedicated to solve many real world applications [1], [2]. Ko *et al.* [4] expressed the effective utilization of fuzzy logic control for refrigerant distribution for multiple evaporating units of an air conditioning system. This paper well described the control of refrigerant flow by a linear expansion valve (LEV) for the operating conditions using a fuzzy logic control. Aprea *et al.* [5] focused research on the cold store analysis, making use of a variable speed compressor for refrigeration purpose, in which the speed of the compressor was modulated by employing the fuzzy logic control. In this paper, the compressor speed was suitably selected in function of the cold store air temperature. The energy saving capabilities of fuzzy logic control algorithm when applied to the compressor side was concentrated.

Performance prediction of a vapor compression heat pump using different ratios of refrigerant mixtures of R12/R22 was determined in [14] utilizing fuzzy logic instead of expensive experimental study. Georges *et al.* [3] estimated the performances of variable refrigerant flow systems through an experimental analysis in which a calorimetric test methodology was adopted for the VRF equipment to identify the heating or cooling emission of each indoor unit. The thermodynamic properties of refrigerant were analyzed and results were projected.

Wu *et al.* [11] have developed a new variable refrigerant flow (VRF) module based on the energy plus simulation and elucidated about the energy usage of VRF system. This paper evaluated the energy consumption of VRF with that of VAV (variable air volume) and fan-coil plus fresh air (FPFA) system. A generic office building was considered for simulation and the simulation result showed that the energy conservation potential expected for VRV system was reasonably better, compared with VAV and FPFA systems respectively. Brodrick *et al.* [13] described about the variable refrigerant volume system features and its retrofit capability, enabling its integration into virtually any building, old or new, with the minimum of structural alteration. They have also explained about the VRV systems extreme flexibility in the manner that, it can have a single condensing unit connected to multiple indoor units of varying capacity and configuration.

Deng and Chen [15] demonstrated about the dynamic modeling of individual components present in the Direct Expansion (DX) VAV air conditioning system. The dynamic modeling was developed based on the principles of mass and energy conservation and using correlations describing the operational performance of some of the system's components which were either field tested or available from manufacturers. It was an experimental paper which validates the dynamic model developed using the experimental data obtained. Steady

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state and transient response of both model and experimental were compared based on the five major operating parameters. Research works were also done on electronic expansion valve (EEV) operating characteristics that determine the performance of EEV for delivering appropriate quantity of refrigerant to the evaporator to satisfy the thermal load.

Dern [6] provided in-depth information about the performance of electronic expansion valve (EEV) when compared with conventional thermostatic expansion valve (TXV) and capillary tube with suitable instances. Jiangpin *et al.* [7] experimentally analyzed the flow characteristics of electronic expansion valve (EEV) that utilized R22 and R407C and developed a dimensionless correlation based on the experimental data to predict the mass flow rate of the refrigerants flowing through EEV. They showed that the operation condition, flow area and the thermo physical properties of the refrigerant would affect the mass flow rate through the EEV based on the analysis of the experimental results. This paper used Buckingham pi theorem to develop the dimensionless correlation.

The influence of EEV utilized in automobile air conditioning was presented by [12]. The flow rate characteristic of the EEV for automobile air conditioning was presented. A microcontroller is used to receive the input signal and generate the output signal to control the opening of the EEV. Wu *et al.* [8] explained that with the rapid increase in land cost and the increased number of high rise apartment buildings, a multi-evaporator air conditioner (MEAC) featuring variable refrigerant volume (VRV) technology has become increasingly attractive and gaining its momentum towards the occupant comfort conditions. The multi-evaporator air conditioning system basically consists of an outdoor condensing unit equipped with compressor and condenser unit with fan and multiple indoor units. In this paper, the compressor speed was modulated and the electronic expansion valve opening was altered by taking into account of suction pressure and the room air temperatures as the control strategies respectively. A controllability test was also conducted and it infers that the control strategy and algorithm were feasible.

Nasution and Hassan [9] identified and projected the potential usage of variable speed compressor running on a controller that provides enhanced energy conservation, load matching capabilities and thermal comfort for air conditioning application. The article infers that based on the experimental data the energy savings obtained from the air conditioning system was satisfactory. It also explains that the thermal comfort and energy conservation together can be obtained through a proper selection of gain (K) parameter for the controller. The controller utilized in this paper was a proportional, integral and derivative (PID) type. Chen *et al.* [10] illustrated the performance of variable speed compressor utilized for inverter air conditioning applications. This paper best described the analysis of VSC modeling for simulation of inverter air conditioners.

The map-based method is utilized to fit the performance curves of inverter compressor. The model was built at the basic frequency and the map condition as the second-order function of condensation temperature and evaporation temperature. Then it was corrected by the compressor frequency as the second-order function of frequency and by the actual operating condition as the actual specific volume of the suction gas. Based on the experimental data and simulation model, the frequency at zero mass flow rate and power input at zero frequency were discussed and the relation between COP and compressor frequency was also analyzed. Constant air volume systems (CAV) consume more power as it runs through out the life span with out change in air volume even at part load conditions. Variable air volume (VAV) air conditioning system operates by varying the supply air volume delivered into the conditioned space by varying the fan speed and dampers angles there by considerable quantity of energy can be conserved. Conventional PID controllers are used to control the air volume in VAV systems. PID controllers are sluggish in response and are characterized by high overshoot. They are also limited by SISO (single input single output). Fuzzy logic control overcomes the problem of high overshoot and has quicker response. Pan *et al.* [16] demonstrated the studies on VAV air conditioning system applied to office buildings. The paper exposed the utilization of outdoor air to obtain building zone comfort especially during part load conditions.

A possible research on varying the airflow rate to individual zones of a building was illustrated in [17] having a control over diffuser outlet preventing excessive draft that allows selecting wide range of supply air temperatures. The PID controller tuning methods are useful in simulation and a tuning method for PID controller utilizing optimization subject to constraints on control input was developed in [18]. Case study on variable frequency drives installed in office building was presented in [19]. The study analyzed the fan input power consumption as a function of airflow rate supplied into the conditioned space. The study also infers that upon having reduced supply duct static pressure, the energy savings potential can be increased. A notable work was done on VAV system that utilized direct digital control (DDC) terminal boxes to achieve occupant comfort conditions making use of feedback control loop technique [19], [20].

Energy conservation studies in VAV systems incorporated with variable speed drive (VSD) fan units was determined in [21] through modeling fan power as a function of outside temperature. The performance curves obtained was used to estimate the energy savings of VSD over variable inlet vanes for the same air-handling units considered. The energy savings were calculated using least square best-fit model technique. Control functions of energy management for effective control of HVAC systems in buildings has been developed that utilized a dynamic simulation technique [22]. Optimal strategy for outdoor air control using a system approach based on prediction to minimize energy consumption was projected by [23] using ARMA model and the energy-increment equation was formed to involve the real-time variations of air handling unit (AHU) load and energy use of reheaters of VAV terminals.

2. EXPERIMENTAL METHODOLOGY

The experimental work has been carried out to determine the inherent operational characteristics of the combined VRV-VAV multi-zone centralized air conditioning system using fuzzy logic controller. The schematic representation of the fuzzy logic based air conditioning control system utilizing the energy efficient VRV-VAV A/C system is shown in the Figure 1. A variable air volume (VAV) air conditioning software laboratory building was considered for the simulation situated at Anna University, Chennai, India. The building zone was decided to be 33m x 8.5m x 2.9m of dimensions. The building has seven windows at each side and door with dimensions 0.91m x 1.83m and 0.91m x 2.13m. The construction materials and properties were selected according to the ASHRAE (American Society of Heating Refrigeration and Air Conditioning Engineers) handbook. The zone has 50 computers on each side and total occupancy of 95 people and lightning load was taken as per the ASHRAE Standards.

A scale model (2.0 × 1.3 × 0.6 m) for the building and air handling system with fuzzy logic controller unit has been constructed in the refrigeration and air conditioning laboratory at Anna University that confines to the numerical values obtained for the building. Due to the symmetry of the room, only a portion was considered for the analysis. This model is geometrically similar to full scale in all details that are important for the volume flow, the energy flow and the contaminant flow. The total nominal capacity of the experimental scale model system is 5 kW. The key components present in the scale model are thermally insulated air conditioned room model equipped with temperature and relative humidity sensors, inverter driven variable speed rotary compressor of 5 kW capacity, electronic expansion valve (EEV), cooling coil, 220 V DC shunt type supply air fan (speed range 0-3000 rpm), return air fan, velocity sensor, silicon-based NDIR CO_2 sensor fixed in the return duct, temperature sensor, pressure sensors for both air side and refrigerant side, fuzzy logic controller, fresh air damper, return air damper, exhaust damper, actuator and mixing box.

Electronic expansion valve was used to control the degree of superheat at evaporator outlet. The experimental combined VRV-VAV air conditioning system has been fully instrumented. All measurements were computerized, in order that all the measured data can be recorded for subsequent analysis. The main objective of the experiment was to establish the inherent operational characteristics of the proposed system for a year around application based on seasonal variations, when the compressor speed can be modulated for varying condition of the suction pressure and mass flow rate of the refrigerant. For a quite 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 refrigerant supplied to the evaporator through EEV was varied and by utilizing a fuzzy logic controller the temperature of supply air was maintained around the set point precisely. The supply air temperature (the dehumidified air temperature measured at downstream of the cooling coil) set point was kept at 13ºC and the room set points were 24 ºC and 50% 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 condition. 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 prevailed inside the conditioned space. The demand-controlled ventilation technique incorporated in the experiment adjusted the outside ventilation air based on CO_2 concentration and the ventilation demands that those occupants created. The fresh air damper opening is suitably controlled by FLC and the required quantity of fresh air is drawn by supply air fan.

In the economizer cycle the outdoor air (OA) temperature which was found less than the return air temperature was pumped into the conditioned space to achieve the desired thermal comfort and indoor air quality (IAQ). During the economizer cycle, the refrigerant plant was turned off and the fresh air damper is set at its full opening position. This helped in conserving a substantial quantity of total energy consumed. The combined action of DCV and EC were also investigated and the tested results are presented. In Table 1 transducers specification and the uncertainties are indicated. Photographic view of the experimental set up is shown in Figure 2.

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

3. FUZZY LOGIC CONTROLLER DESIGN

The fuzzy logic controller design utilized in this work included multi input and multi output parameters to control the proposed air conditioning system effectively. In 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 IF-THEN rules that define the system behavior.

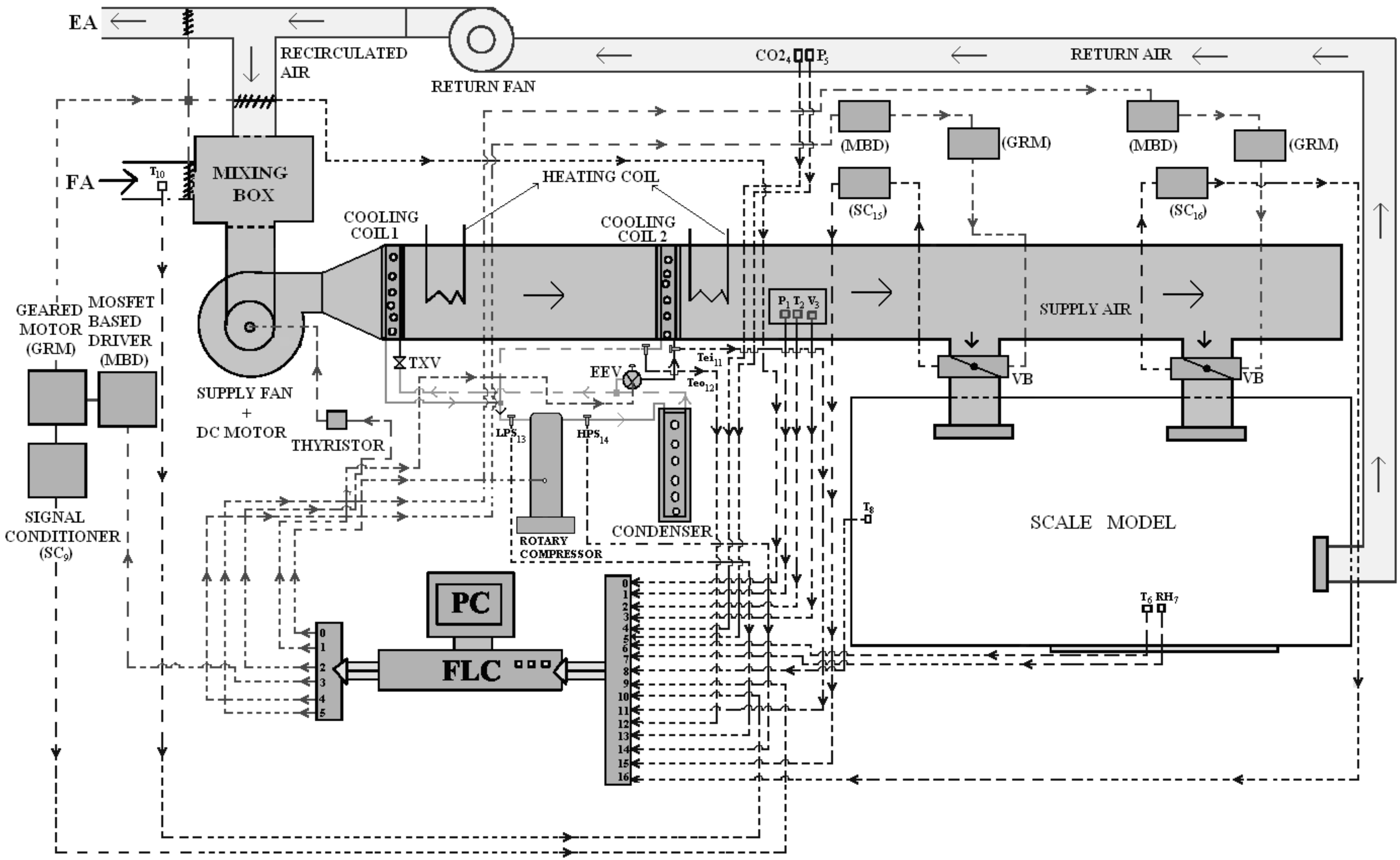


Fig. 1. Schematic representation of FLC based VRV-VAV A/C system.

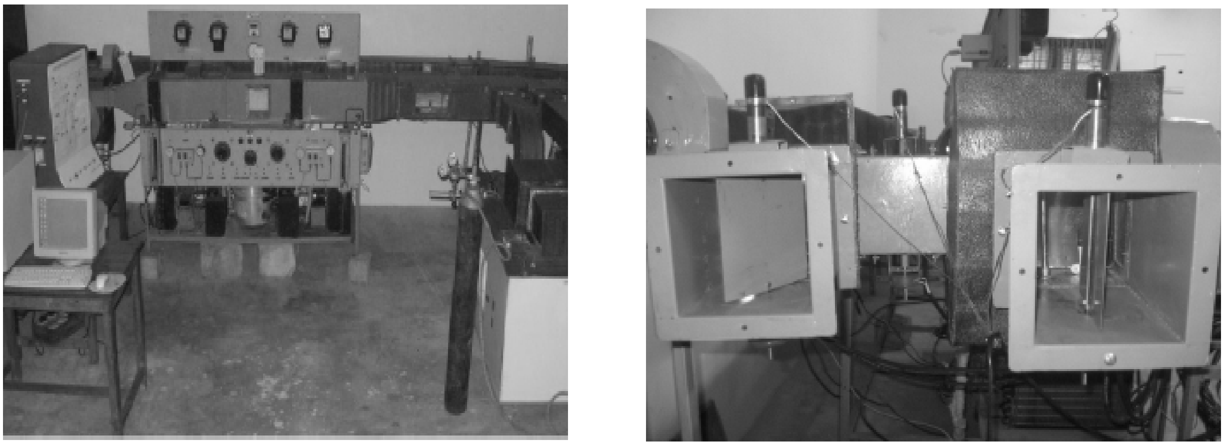


Fig. 2. Photographic view of FLC based VRV-VAV A/C system.

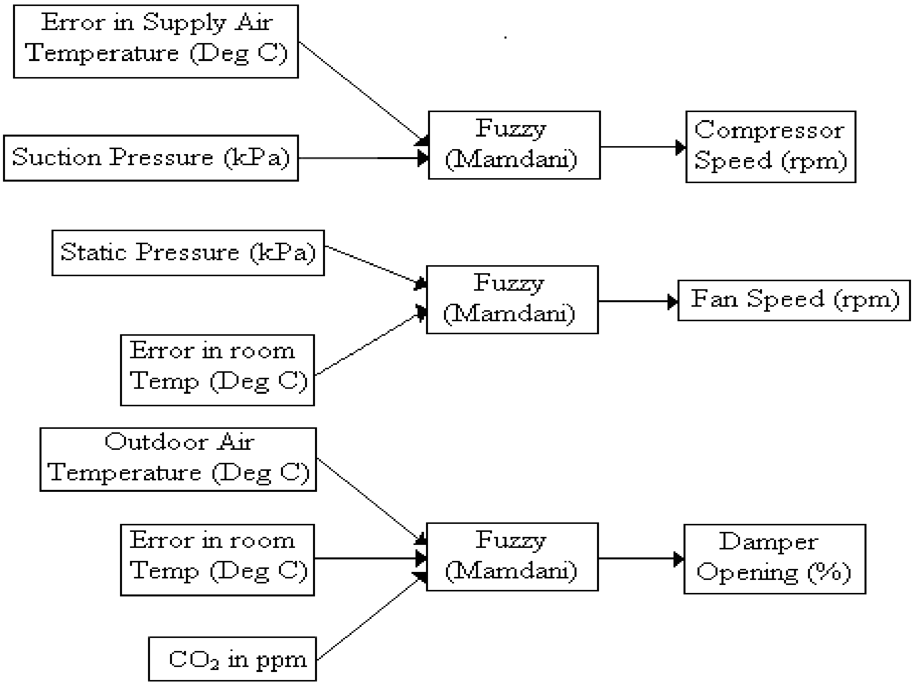


Fig. 3. Structure of fuzzy logic controller.

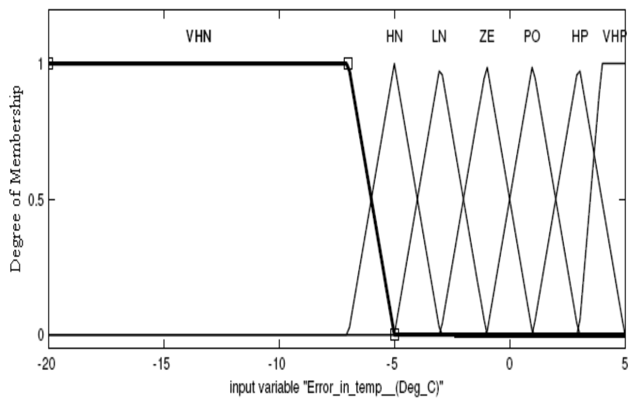
Defuzzification takes the fuzzy action from the inference process and translates it into crisp values for the control variables. The structure of the fuzzy logic controller is shown in Figure 3. The error in supply air temperature and suction pressure of compressor were considered to be input variables that constitute for output variable in the form of modulated compressor speed. Similarly, the duct static pressure and error in room temperature were another set of input parameters considered that corresponds to the varied fan speed that was obtained as the output from the FLC.

In order to have a better control over proper ventilation air requirements, the FLC was tuned effectively by mapping the input variables like outdoor temperature, error in room temperature and CO_2 concentration over the damper opening position and that was the desired output required from FLC. Graphical illustration of the membership functions and their ranges for the input and output variables are shown in Figure 4.

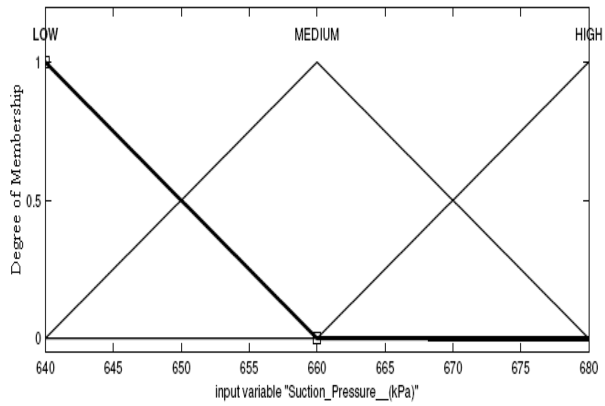
The range of the membership functions were selected according to the minimum and maximum values occurred. These ranges are defined in the interval of -20ºC to 5ºC for error in supply air temperature, 640 kPa to 680 kPa for suction pressure, 2000 rpm to 7000 rpm for compressor speed, 0.20 kPa to 0.80 kPa for duct static pressure, -16ºC to 6ºC for error in room air temperature, 2000 rpm to 3000 rpm for fan speed, 15ºC to 40ºC for outdoor air temperature, 200 ppm to 2000 ppm for CO_2 concentration, and 0% to 100% for damper opening position. FLC was utilized to vary the compressor speed and fan speed according to the room cooling load fluctuations. The process for determining the rules is to track the set point temperature with a minimum steady-state error.

Fuzzy rules are generated for the VRV-VAV air conditioning system. The logic is based in the way, how the system will respond for the corresponding change in input variables. For instance, when error in temperature is very high negative (VHN) and suction pressure is high then the compressor speed is maintained at high speed (HS). By using Matlab-Simulink environment, the designed FLC can be linked with the simulated model to evaluate the system performance. Centroid method is used to convert the fuzzy variable back to the output variable that can be varied according to the rules. Mathematical relation behind the centroid method is given by:

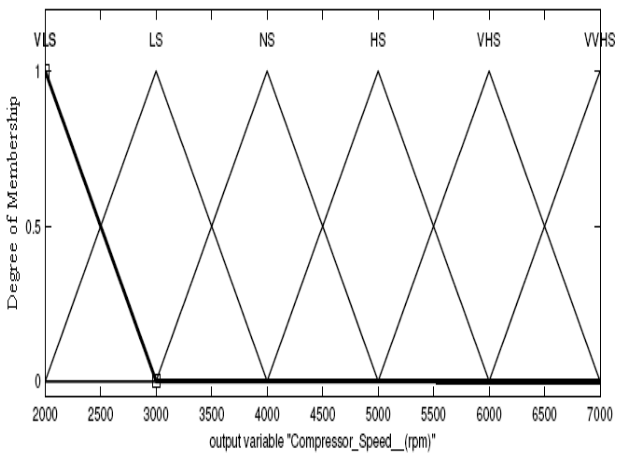
$$Z^* = \frac{\int \mu_c(Z).Z.dZ}{\mu_c(Z)dZ} \tag{1}$$



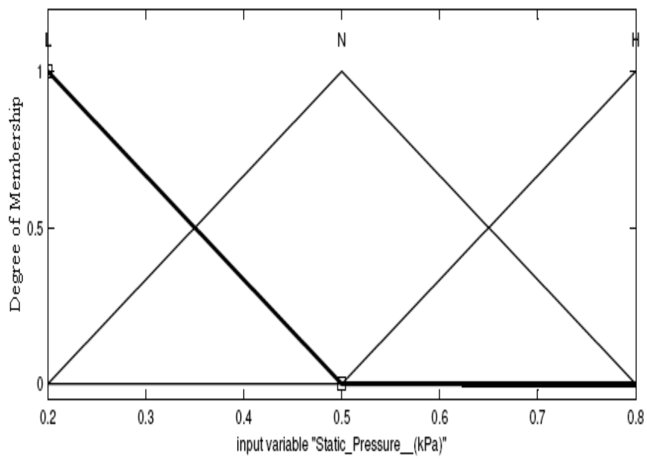
(a) Error in supply air temperature range.



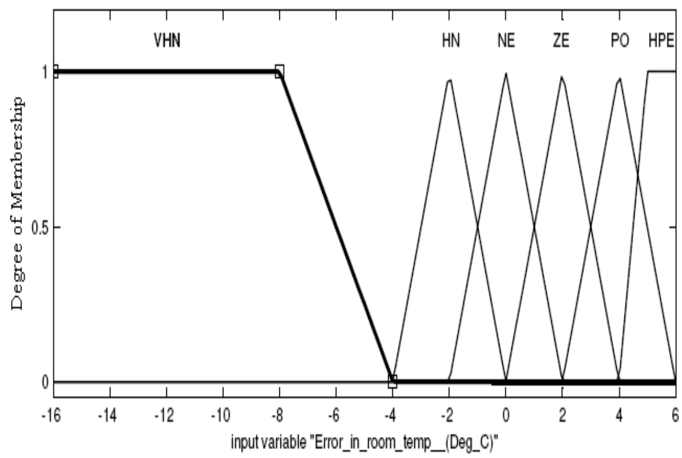
(b) Compressor suction pressure range.



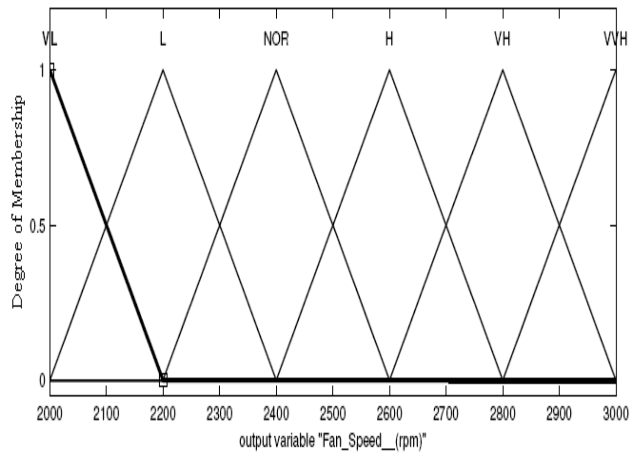
(c) Compressor speed range.



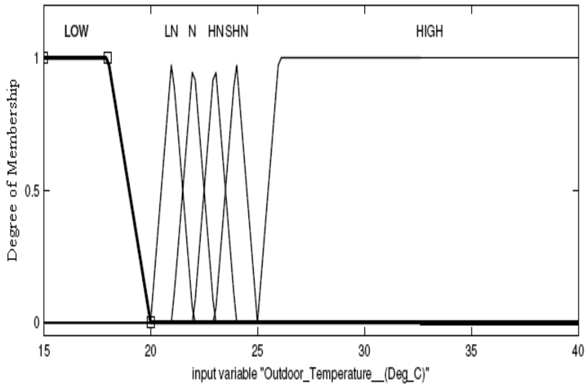
(d) Duct static pressure range.



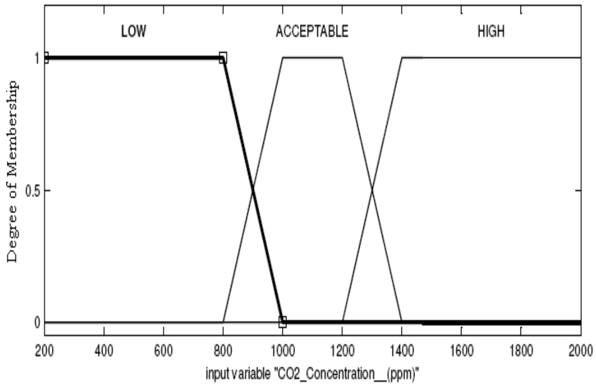
(e) Error in room temperature range.



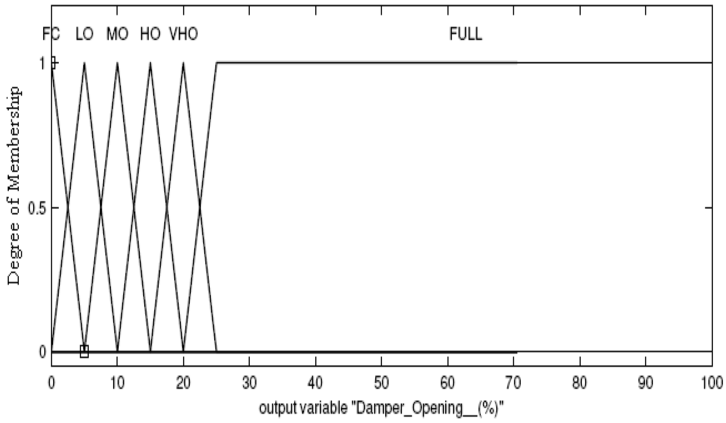
(f) Fan speed range.



(g) Outdoor air temperature range.



(h) CO_2 concentration range.



(i) Damper opening range.

Fig. 4. Input and output membership function plots.

4. HVAC SYSTEM SIMULATION

The MATLAB-SIMULINK software package was used for the simulation purpose. The outdoor temperature variation is taken as per the meteorological department for May and December, since these months record 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 are represented in Figure 5. The load pattern is noted for the real building for the design day and the pattern is drawn in terms of percentage, so that the pattern can be followed for both building and scale model.

Mathematical Models

The mathematical models considered for simulation work are given below:

i. Variable Speed Rotary Compressor (VSC)

The mass flow rate of refrigerant entering compressor can be calculated given by the equation,

$$m_{com} = \left(\lambda \frac{V_{th}}{v_{suc}} \right) \tag{2}$$

where v_{suc} is the pecific volume of refrigerant;

The volume flow rate of refrigerant is given by,

$$V_{th} = 60 * N * \pi * R^2 * L * \frac{e}{R} \left(2 - \frac{e}{R} \right) \tag{3}$$

The compressor work can be represented by the equation,

$$W_{com} = \left(h_{ds} - h_{suc} \right) \tag{4}$$

where, h_{ds}, h_{suc} is the enthalpy of refrigerant

ii. Electronic Expansion Valve

Refrigerant mass flow rate through EEV [8] is denoted by,

$$(m_{ev}) = A_{ev} * \xi * \{ (P_c - P_e) / v_c \}^{0.5} \tag{5}$$

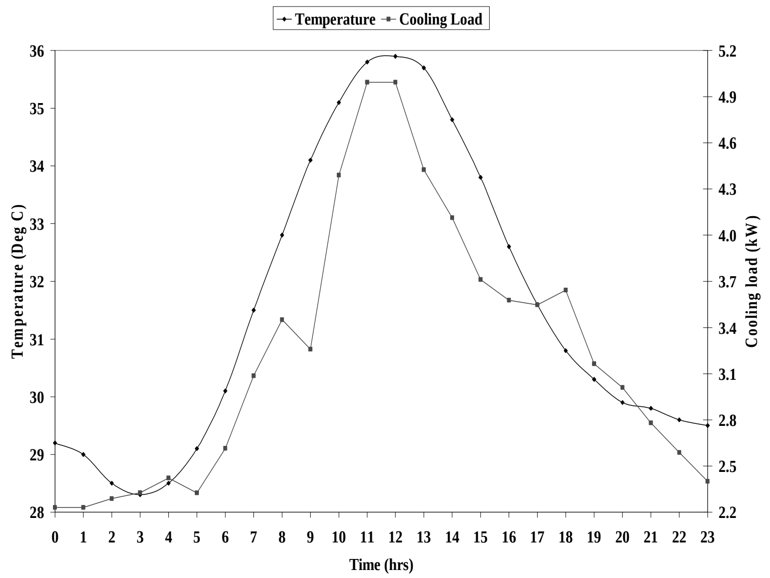
iii. Evaporator Model

The mixed enthalpy of refrigerant can be found by,

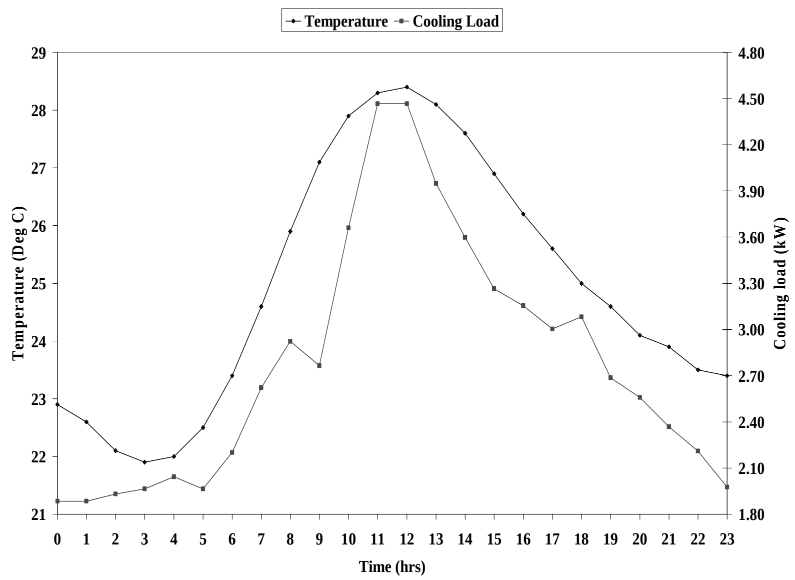
$$h_s = \frac{h_v * m_{ev1} + h_v * m_{ev2}}{m_{ev1} + m_{ev2}} \tag{6}$$

$$m_{com} = m_{ev1} + m_{ev2} \tag{7}$$

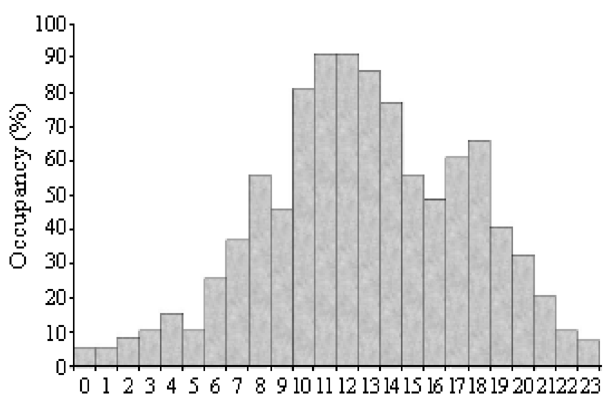
where, m_{ev1}, m_{ev2} are the mass flow rates of refrigerant flowing through the evaporators



(a) Variation of cooling load – Summer.



(b) Variation of cooling load – Winter.



(c) Occupancy load pattern.

Fig. 5. Variation of outdoor temperature, cooling load and occupancy load pattern.

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 state-space notation.

The notation for state-space model is presented by:

$$dT/dt = AT + Bu \tag{8}$$

$$\begin{bmatrix} \frac{dT_w}{dt} \\ \frac{dT_f}{dt} \\ \frac{dT_c}{dt} \\ \frac{dT_{ai}}{dt} \end{bmatrix} = \begin{bmatrix} \frac{-A_w(U_{wi}+U_{wo})}{C_w} & 0 & 0 & \frac{A_w U_{wi}}{C_w} \\ 0 & \frac{-A_f U_f}{C_f} & 0 & \frac{A_f U_f}{C_f} \\ 0 & 0 & \frac{-A_c U_c}{C_c} & \frac{A_o U_o}{C_o} \\ \frac{A_w U_{wi}}{C_a} & \frac{A_f U_f}{C_a} & \frac{A_c U_c}{C_a} & -\left(\frac{1}{C_a}\right)\left(A_g U_g + A_w U_{wi} + A_f U_f + A_o U_c\right) \end{bmatrix} \begin{bmatrix} T_w \\ T_f \\ T_c \\ T_{ai} \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 & \frac{A_w U_{wo}}{C_w} \\ 0 & 0 & \frac{p}{C_f} & 0 \\ 0 & 0 & 0 & 0 \\ \frac{1}{C_a} & \frac{1}{C_a} & 0 & \frac{A_g U_g}{C_g} \end{bmatrix} \begin{bmatrix} Q_p \\ Q_i \\ Q_s \\ T_{ao} \end{bmatrix} \tag{9}$$

where A, B are matrices of coefficients, u is the input vector, T is the matrix of temperatures. Temperatures of walls (T_w), ceiling (T_c), floor (T_f), room air (T_{ai}) were selected as the state variables in T(t) and input variables in u(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}).

Fan Model

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

$$(W_{fan}) = FMP\ \{a + b\,(PLR\,(t)) + c\,(PLR\,(t))^2 + d\,(PLR\,(t))^3\} \tag{10}$$

where, a = 0.00153, b = 0.005208, c = 1.1086, d = 0.11635 are the fan performance coefficients obtained from manufacturer catalogue ; PLR – part load flow ratio; FMP – fan motor power

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,

$$DVR = V_P + V_B \tag{11}$$

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,

$$N(t) = G * P(t) \tag{12}$$

$$dc/dt = (C_S - C) * Q_S / v + G * P(t) / v \tag{13}$$

where, v-volume of the room; Q-ventilation quantity; N-contaminant concentration rate; P-occupancy

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,

$$Q_{out} = C_{d(\theta)} \tan\theta * A * (2 * [p_i - p_o] / \rho)^{½} \tag{14}$$

The area of the damper is given by A= D * W; where, θ - damper angle; D – depth; W – width.

5. EXPERIMENTAL RESULTS AND DISCUSSION

The inherent operational characteristics for the combined VRV-VAV centralized summer air conditioning system for the scale model developed based on FLC enabled with 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 proposed system. For different operating conditions the performance of existing system is compared with the conventional CAV A/C system under three broad classifications i.e., thermal comfort, IAQ and energy conservation The results presented involve the parameters that influence greater on the system operating conditions.

The Influence of Supply Air Flow Rate on Refrigerant Mass Flow Rate

The variation of refrigerant mass flow rate (MFR) corresponding to the variation of supply airflow rate (AFR) is represented in Figures 6 and 7. Figure 6 refers to the summer design conditions in which the airflow rate requirement in demand controlled ventilation is observed to be 18.2 % less than the fixed ventilation. It is so because in a DCV the outdoor airflow requirement is based on the CO_2 concentration level thereby a considerable energy savings on the supply air fan can be obtained.

Similarly the mass flow rate of refrigerant in a DCV technique is 18.4% less than the fixed ventilation condition which enhances the energy conservation on compressor side. Based on the Figure 7 during winter design conditions the similar pattern of variation of airflow rate and refrigerant mass flow rate is observed. The airflow rate requirement in demand controlled ventilation is observed to be 17.4% less than the fixed ventilation. The mass flow rate of refrigerant is 17.1% less in a DCV than the fixed ventilation scheme. Thermal comfort is well achieved in the proposed system as the supply airflow rate is continuously altered depending on the cooling load existing inside the conditioned space. The test results express that the proposed air conditioning system is well suited for seasonal variations that lends it to achieve better thermal comfort and IAQ.

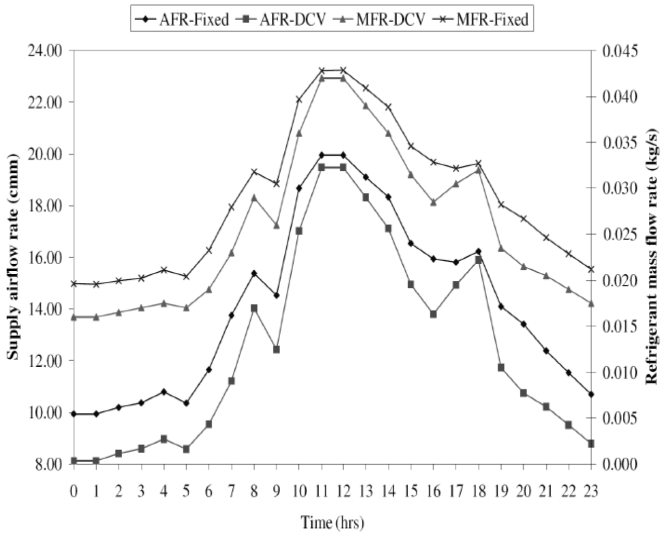


Fig. 6. Variation of supply airflow rate – Summer.

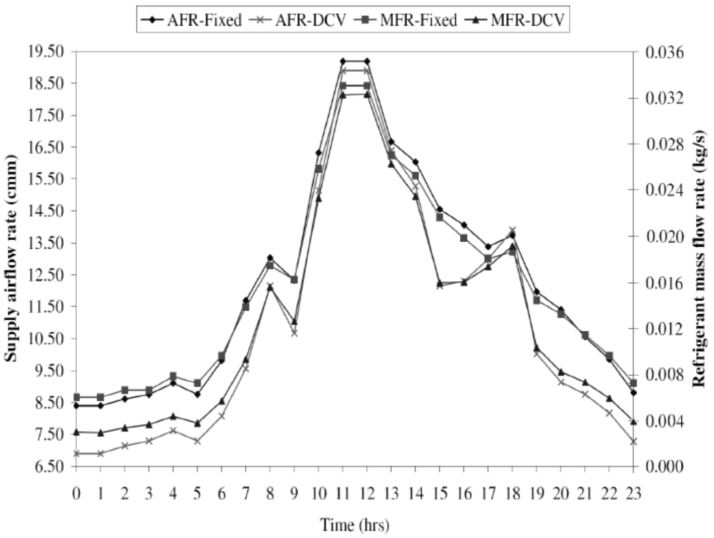


Fig. 7. Variation of supply airflow rate – Winter.

The Influence of Refrigerant Mass Flow Rate on Compressor Speed

The mass flow rate of the refrigerant has a direct relation with the speed of the compressor. The variation of compressor speed with the modulated refrigerant mass flow rate is represented in Figures 8 and 9 for summer and winter conditions, respectively. The compressor speed for a fixed ventilation scheme is observed to vary from 3080 rpm to 6790 rpm during summer and 2400rpm to 5920 rpm during winter conditions respectively. The DCV scheme when applied to summer and winter conditions infers that the compressor speed varies between 2520rpm to 6620rpm and 1960rpm to 5850rpm respectively. Since a distinct variation is observed in the mass flow rates of refrigerant between fixed and demand controlled ventilation techniques utilized in this system, the energy consumed by the compressor is substantially reduced and accounts for total energy conservation.

In case of a CAV system, since the compressor rotates at a constant speed, for modulated refrigerant mass flow rates, the energy savings are expected very minimum. The supply air temperature and relative humidity are effectively maintained around 13°C and 50% by utilizing the fuzzy logic controller. Figures 10 and 11 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.

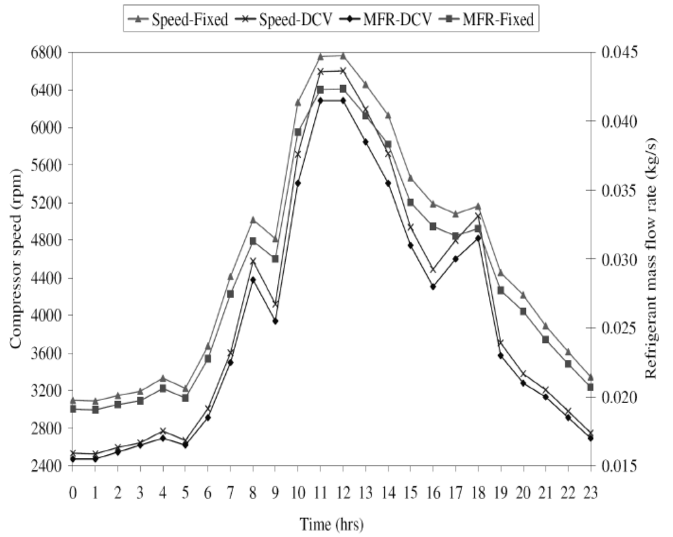


Fig. 8. Variation of compressor speed – Summer.

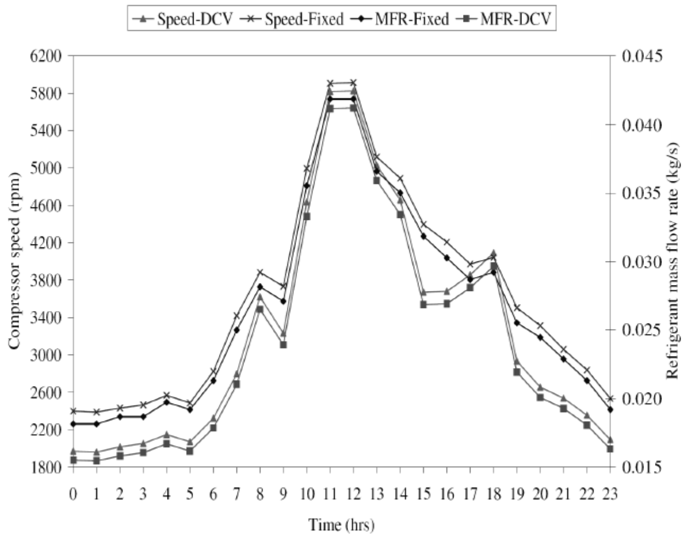


Fig. 9. Variation of compressor speed – Winter.

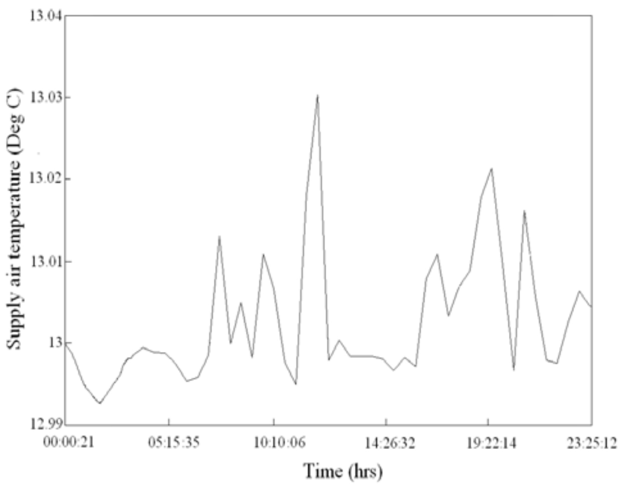


Fig. 10. Variation of supply air temperature.

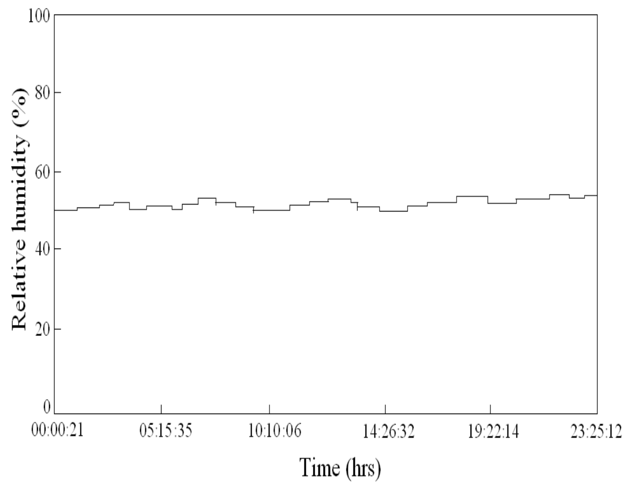


Fig. 11. Variation of relative humidity.

The Variation of Supply Air Temperature, Relative Humidity and Static Pressure

Based on the experimental result, as the supply air temperature and relative humidity is maintained almost constant, it inevitably express that, the cooling load prevailing in the building model is also controlled satisfactorily. Figure 12 depicts the variation of duct static pressure relative to the change in cooling load existing inside the conditioned space respectively. By varying the supply air flow rate for varying load conditions the duct static pressure is observed to vary. The experimental result infers that the static pressure in the duct is observed to be nominal in the range of 325kPa to 750kPa that is expected to provide better thermal comfort, proper air distribution and less fan power consumption.

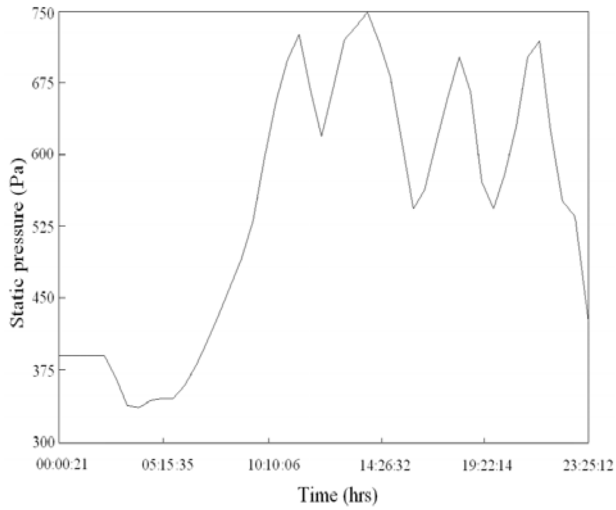
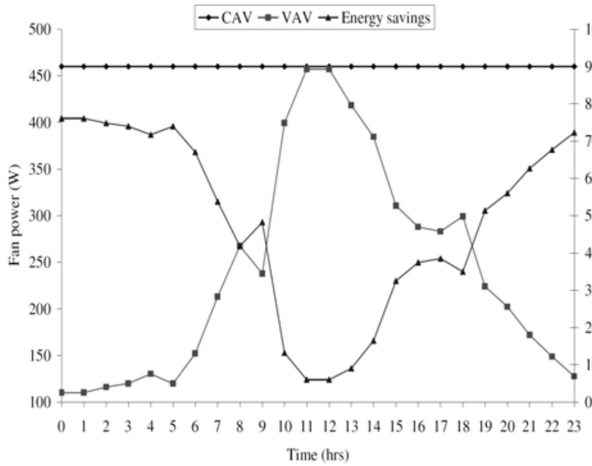
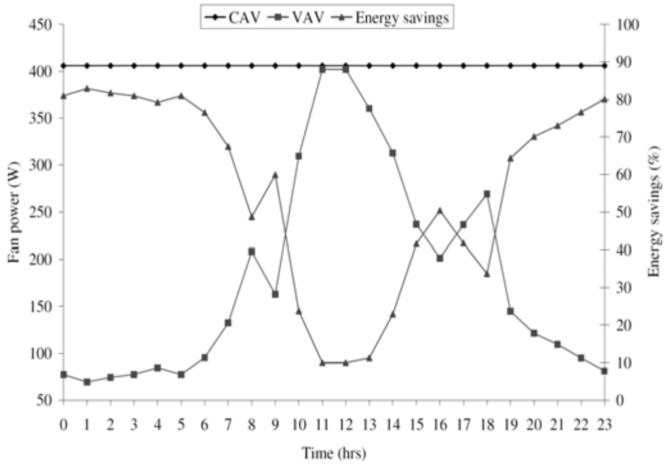


Fig. 12. Variation of duct static pressure.

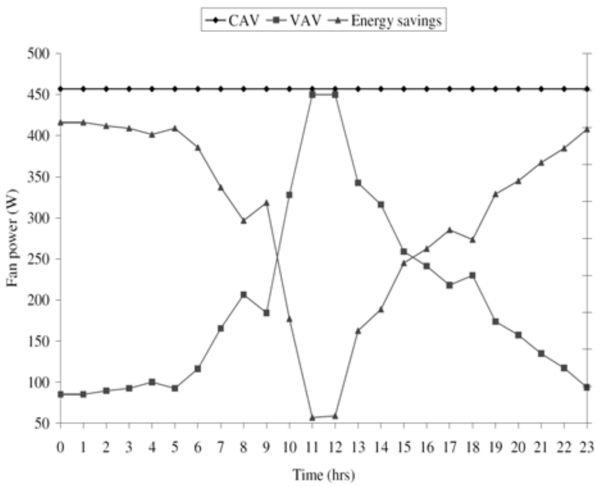


(a) Variation of fan power – Fixed ventilation.

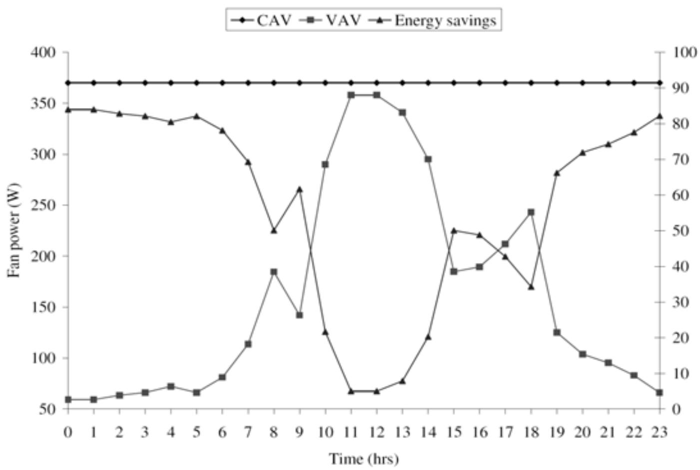


(b) Variation of fan power – DCV.

Fig. 13. Variation of fan power for summer design conditions.



(a) Variation of fan power – Fixed.



(b) Variation of fan power – DCV.

Fig. 14. Variation of fan power for winter design conditions.

The Effect of Fan Power on Energy Conservation

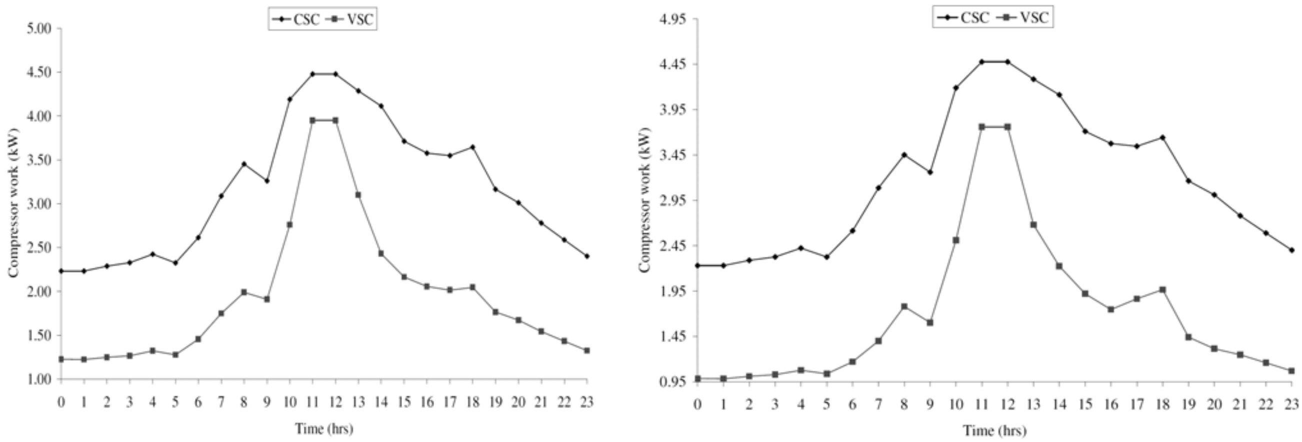
The variation of supply air fan power for summer and winter design conditions is shown in Figures 13 and 14, respectively. In this system, the speed of the supply air fan is varied by the effective utilization of fuzzy logic controller. Based on the static pressure present in the supply air duct, the fan speed is considerably modulated to offset the cooling load. In the present system, the supply air fan works over a wide range of power inputs that vary between 110W to 456W.

It is a well known fact that in a DCV technique the fan power could be considerably reduced. The range of power consumed by the supply air fan under DCV in summer and winter conditions are 77W to 406W and 59W to 367W respectively. In the CAV system, the fan operates at a constant power input and energy consumption is high when compared to the combined VRV-VAV system.

The Effect of Compressor Power on Energy Savings

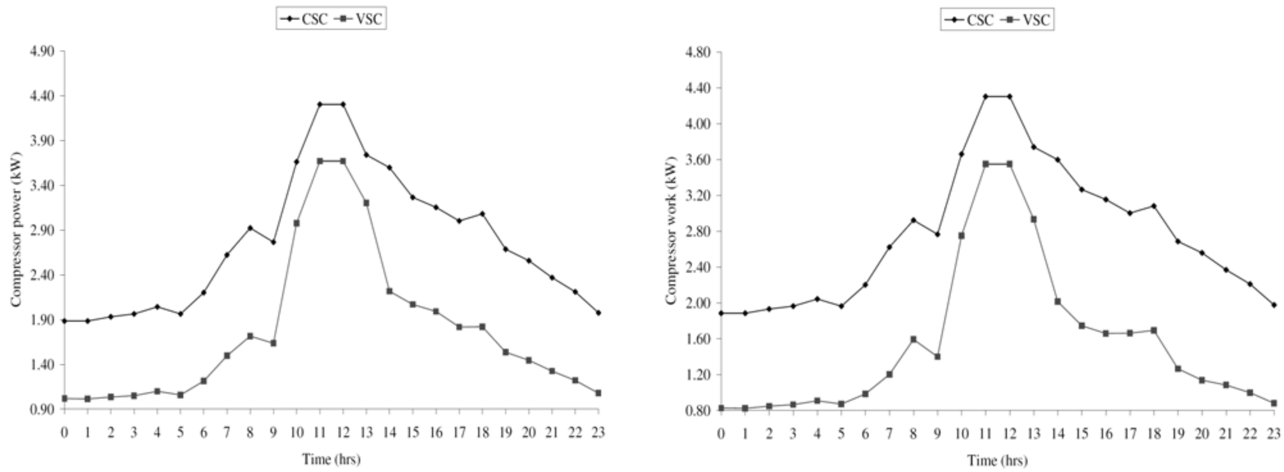
The variation of compressor power for summer and winter design conditions are shown in Figures 15 and 16, respectively. In the proposed system, the speed of the variable speed compressor (VSC) varies according to the load fluctuations and the power input to the compressor also varies considerably.

The power consumed by variable speed compressor is less than that of the constant speed compressor (CSC). In the present system, fuzzy logic controller modulates the power consumed by the compressor based on the cooling load persisting inside the conditioned space. Figure 15 infers that during summer design conditions under fixed and DCV schemes, the variable speed compressor operates between 1.23kW to 3.95kW and 0.98kW to 3.76kW for fixed ventilation and DCV techniques, respectively.



(a) Variation of compressor power – Fixed ventilation. **(b) Variation of compressor power – DCV.**

Fig. 15. Variation of compressor power for summer design conditions.



(a) Variation of compressor power – Fixed ventilation. **(b) Variation of compressor power – DCV+EC.**

Fig. 16. Variation of compressor power for winter design conditions.

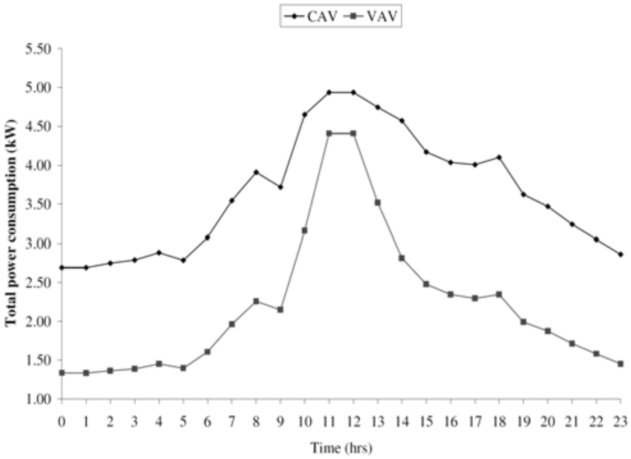
In Figure 16, for winter design conditions it is observed that the power consumed by the VSC varies between 0.93kW to 3.67kW under fixed ventilation scheme and 0kW to 3.67kW for combined DCV and EC technique which is less than it consumed for summer design conditions. This occurs because the power consumed by the compressor is in direct relation with building cooling load.

The Effect of Total Power Consumption

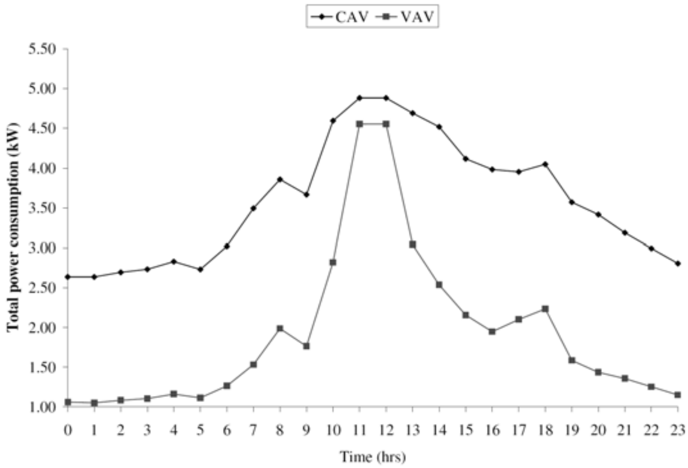
Figures 17 and 18 show the variation of total power consumption for summer and winter design conditions of the VRV-VAV A/C system when compared with the CAV

A/C system, respectively. The system inherent losses are neglected. In the CAV system it is found that the total power consumption for fixed ventilation ranges for summer and winter conditions lies between 2.69kW to 4.94kW and 2.34kW to 4.76kW, respectively.

Experimental result shown in Figures 17 and 18 refers that, in the proposed air conditioning system, the total power consumption for fixed ventilation varies between 1.34kW to 4.41kW for summer condition and 1.02kW to 4.22kW for winter condition. Substantial reduction in the total power consumption is observed when DCV scheme is incorporated with the VAV-VRV system. The DCV power consumption ranges between 1.06kW to 4.56kW for summer and 0.83kW to 4.04kW for winter, respectively. When economizer cycle is integrated with DCV, the combined scheme showed a tendency in minimizing the total energy consumption.

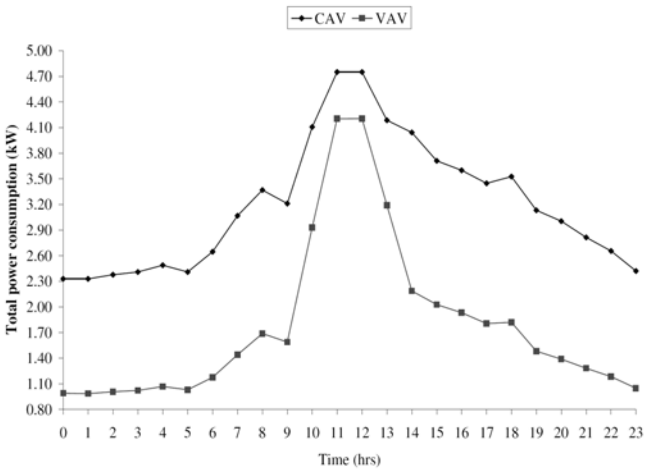


(a) Variation of total power consumption – Fixed ventilation.

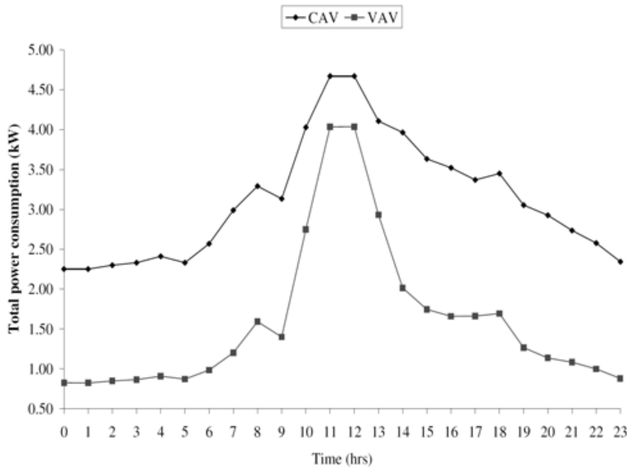


(b) Variation of total power consumption DCV.

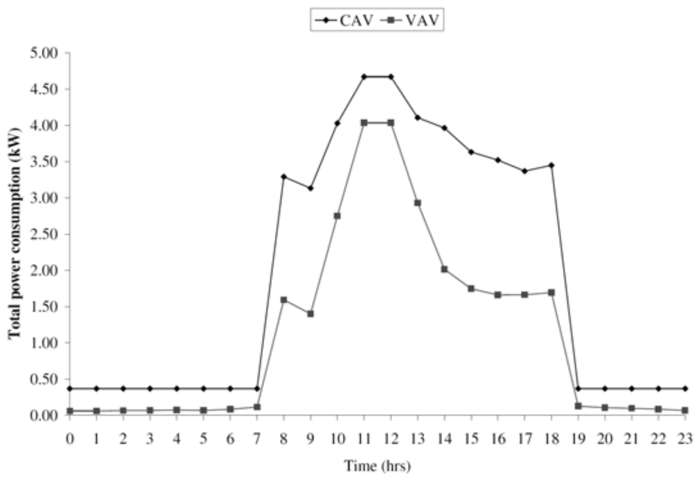
Fig. 17. Variation of total power consumption for summer design conditions.



(a) Variation of total power consumption – Fixed ventilation.



(b) Variation of total power consumption – DCV.



(c) Variation of total power consumption – DCV with economizer.

Fig. 18. Variation of total power consumption for winter design conditions.

Energy Savings Potential of VRV-VAV A/C System

The energy savings characteristics of combined VRV-VAV A/C system while compared with the CAV A/C system is evaluated based on the test result as depicted in the Figure 19. In summer design conditions, the proposed system in spite of having its inherent losses yielded the per day energy savings potential of 27% and 34% when operated under fixed and DCV methods respectively. The increase in energy savings is because of the influence of required fresh air quantity intake based on the occupancy level and the corresponding CO_2 concentration. Similarly, considering the system inherent losses, the proposed system when operated under winter design conditions achieved per day energy savings potential of 33%, 40% and 52% for fixed, DCV and DCV-EC schemes respectively. An increase of energy savings is observed in case of combined DCV-EC is because of operating only the supply air fan to deliver the prescribed quantity of fresh air while the outdoor temperatures are low enough to achieve better thermal comfort and IAQ. During economizer cycle the compressor is turned off completely. Based on the experimental result it is observed that, the combined VRV-VAV A/C system can be considered to be high energy efficient air conditioning scheme that has an enormous energy savings potential compared to the conventional CAV A/C systems.

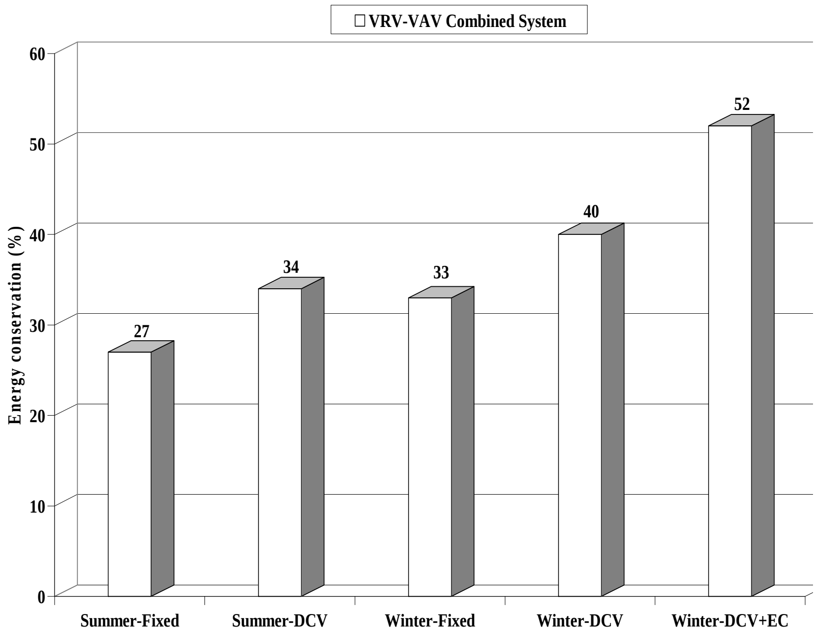


Fig. 19. Energy savings potential of VRV-VAV A/C system.

6. CONCLUSION

Compact and flexible air-conditioning systems are being installed in applications raging 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 variable refrigerant volume system is considered to be one of the most promising energy saving technologies gaining its momentum in recent years. In this study, the inherent operational characteristics of the combined VRV-VAV A/C system utilizing fuzzy logic control methodology for the scale model developed was investigated. In order to assess the benefits of the proposed system under cooling mode in terms of energy conservation for both summer and winter conditions, the system was tested under three different ventilation techniques. The test results infer that the proposed air conditioning system controlled by fuzzy logic methodology effectively maintained the supply air temperature closely to 13ºC as well as the room air temperature around 24ºC precisely and relative humidity around 50% which can be attributed to achieve a better thermal comfort inside the conditioned space. The test results projects that for varying occupancy levels the system was capable of maintaining a better IAQ in the sense that by detecting the CO2 concentration present inside the conditioned space, the required fresh air was delivered into the conditioned space.

Based on the experimental results on energy conservation, it is observed that the VRV-VAV A/C system was capable of conserving energy up to 27% and 34% for summer design conditions utilizing fixed ventilation and DCV techniques. A similar but enhanced energy savings was experienced by the system for winter design conditions that yield 33 %, 40% and 52% for fixed ventilation, DCV and combined DCV-EC schemes. Based on the experimental results, the combined VRV-VAV A/C system can be considered to be high energy efficient air conditioning scheme that has an enormous energy savings potential compared to the conventional constant air volume (CAV) systems. Although conventional controllers are preferred widely, the advantageous fuzzy logic control 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. It is pertinent to note that the variable refrigerant volume system is having abundant research and development opportunities that would shape the system to perform better and contribute it for growing demands in HVAC applications.

ACKNOWLEDGEMENT

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NOMENCLATURE

A	heat transfer area, m^2
e	eccentricity, m
h	enthalpy, kJ/kg
L	axial length of cylinder, m
m	refrigerant mass flow rate, kg/s
N	rotational speed of compressor, r/s
P_c	condenser pressure, Pa
P_e	Evaporator pressure, Pa
R	radius of cylinder, m
U	overall heat transfer coefficient, kW/m^2 K
v	specific volume, m^3/kg
V	volume flow rate, m^3/s
W	specific work, kJ/kg
λ	volumetric efficiency
ξ	valve flow coefficient
Cp	specific heat, kJ/kg K
C_d	coefficient of airflow
ρ	density, kg/m^3

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