CHAPTER 6

RESULTS AND DISCUSSION

6.1 INTRODUCTION

The inherent operational characteristics of the DX VRV-VAV air conditioning unit with genetic based FLC is analyzed for summer and winter design conditions by the scale model developed. The results of the simulation and experimental were compared and analysed for various operating conditions. Several experimental tests have also been conducted to explain the thermal comfort, IAQ and energy saving and compared with those of the conventional system. To simulate the real working conditions of the air-conditioning, various types of cooling loads have been considered.

6.2 THERMAL COMFORT IN THE VRV-VAV AIR CONDITIONING SYSTEM USING GENETIC BASED FLC

6.2.1 Controllers performance at transient and steady state

The simulation of the VRV-VAV air conditioning system with the building thermal model was carried out in MATLAB-SIMULINK environment. The performance comparison of the proposed fuzzy controller with that of the PID and ON/OFF controllers at transient state is shown in Figure 6.1. The tuning of PID controller was done using the classical control theory and the values for proportional gain, integral gain and derivative gain were taken to be $K_p = 2$, $K_i = 0.01$, and $K_d = 0.001$ respectively. The simulation results project the comparison of the controller performance and the system performance of the VRV-VAV A/C system equipped with FLC.
a) Transient response of the system with Two-Position controller

b) Transient response of the system with PID controller

Figure 6.1 Controllers performance of the VRV-VAV A/C system for room temperature variation at transient state
c) Transient response of the system with Fuzzy controller

From Table 6.1 it is clear that the rise time of the fuzzy controller is 25s. It means that to attain the set point temperature inside the room, the fuzzy controller acquired 25s, whereas the Two-Position and PID controllers took 85s and 55s respectively. Peak overshoot for the fuzzy controller was 0.83% which is low while compared with the Two-Position and PID controllers of 5.42% and 4.58% respectively. Hence, the proposed fuzzy controller possessed excellent transient performance compared to the other two conventional controllers.

Table 6.1 Transient performance of Two-Position, PID and FLC

<table>
<thead>
<tr>
<th>Control</th>
<th>Rise time (s)</th>
<th>Peak overshoot (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Two-Position</td>
<td>85</td>
<td>5.42</td>
</tr>
<tr>
<td>PID</td>
<td>55</td>
<td>4.58</td>
</tr>
<tr>
<td>Fuzzy</td>
<td>25</td>
<td>0.83</td>
</tr>
</tbody>
</table>
The steady-state analysis was carried out for the system and the results obtained from the simulation of the HVAC system are shown in Figure 6.2.

a) Steady state temperature variation inside a room with the Two-Position controller

b) Steady state temperature variation inside a room with the PID controller

Figure 6.2 Controllers performance of VRV-VAV A/C system for room temperature variation at steady state
c) Steady state temperature variation inside room with the Fuzzy controller

Figure 6.2 (Continued)

Table 6.2 shows the steady-state response of the HVAC system with Two-Position, PID and Fuzzy controllers. It was shown that the steady-state error for the fuzzy controller was zero. It means that even if the load inside the room changes the fuzzy controller maintains constant temperature, whereas the conventional controllers were unable to maintain the desired temperature inside the room. The values of the Integrated Area Error (IAE) that is inferred that a measure of controller error for the three controllers. It was shown that fuzzy controller has less error compared to conventional controllers.

Table 6.2 Steady-state performance of the Two-Position, PID and FLC

<table>
<thead>
<tr>
<th>Control</th>
<th>Steady-state error</th>
<th>IAE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Two-Position</td>
<td>Not settled</td>
<td>23.84</td>
</tr>
<tr>
<td>PID</td>
<td>Not settled</td>
<td>11.68</td>
</tr>
<tr>
<td>Fuzzy</td>
<td>0</td>
<td>2.32</td>
</tr>
</tbody>
</table>
6.2.2 System performance in achieving thermal comfort using FLC

The performance of the system can be observed by considering the space temperature in the conditioned space in the simulated building during all working hours. The set point temperature inside the conditioned room is 24°C. The set point temperature is based on ASHRAE standard 55-1992. The onsets and offset temperatures are 22.5°C and 25.5°C respectively. This selection is based on the ASHRAE recommended comfort zone for summer. Table 6.3 shows the variation of the space temperature with variation in internal load and outside weather temperature variation. Based on Figure 6.2, it is inferred that the space temperatures with two-position controller fluctuated from 22.5°C to 25.5°C, whereas for the PID controller the variation of space temperatures were observed to be between 23.5°C and 24.5°C. The proposed fuzzy controller maintained the conditioned space temperature precisely around 24°C with faster and better response compared to the other two controllers.

Table 6.3 Variation of space temperature with Two-Position, PID and FLC

<table>
<thead>
<tr>
<th>Control</th>
<th>Temperature variation (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Two-Position</td>
<td>22.5-25.5</td>
</tr>
<tr>
<td>PID</td>
<td>23.5-24.5</td>
</tr>
<tr>
<td>Fuzzy</td>
<td>24.0-24.05</td>
</tr>
</tbody>
</table>

6.2.3 The influence of the GA optimization on thermal comfort

Thermal comfort problems in VAV air-conditioning systems, due to over cooling (or) under cooling of the space, lack of air movement and improper controls. The GA optimization results were given as the set points to
the FLC to minimize thermal comfort problems. The GA optimization variables for thermal comfort are supply air temperature (SAT), zone temperature (ZT) for summer and winter seasons, supply air velocity (Vs) and duct static pressure (Ps) which are shown in Figures 6.3 and 6.4.

**Figure 6.3 Variation of zone temperatures based on GA optimization**

Figure 6.3 shows the Genetic Algorithm optimization variables like Supply Air Temperature (SAT) and occupancy Zone Temperature (ZT) in summer and winter conditions. The Genetic Algorithm optimization results show that the supply air temperature ranged from 13°C to 17°C in summer. In winter, the supply air temperature ranged from 13°C to 15°C from 7.00 a.m to 7.00 p.m and for the remaining hours the economizer cycle was activated during which the compressor was switched off and 100% fresh air was taken. The GA results show that the supply air temperature is very close to the outdoor temperature. The occupancy zone temperature is around 22°C in winter and around 24°C in summer conditions inspite of load variations.
Figure 6.4 Variation of supply air velocity and duct static pressure based on GA optimization

Figure 6.4 shows the Genetic Algorithm optimization result of duct static pressure and supply air velocity for load variations. The GA results are within the upper and lower limit of GA constraints. The duct static pressure ranged from 400 Pa to 675 Pa and the supply air velocity is from 2.6 m/s to 4.0 m/s.
Figure 6.5 Comparison of the simulation and experimental results for duct static pressure variation

Figure 6.6 Comparison of simulation and experimental results for supply air fan speed variations
The variation of duct static pressure and supply air fan speed for simulation and experiment are shown in Figures 6.5 and 6.6. It is shown that, for the fluctuated cooling load, the duct static pressure ranged from 400 Pa to 690 Pa for the simulated model and 410 Pa to 680 Pa for the experimentation. The fan speed was altered according to duct static pressure and supply air velocity. Similarly the fan speed ranged from 1450 rpm to 2960 rpm for the simulation model but for the experiment the value ranged from 1690 rpm to 2890 rpm respectively. At peak load the fan was operated at the maximum speed for maintaining the thermal comfort in the zone. The FLC receives the sensed signal from the static pressure sensor and velocity sensor the inputs, and the fan speed variations were obtained as per the requirement. The supply air velocity and volume flow rate of air were varied depending upon the fan speed.

Figure 6.7 shows the variations in the occupant zone VAV box damper opening. The damper opening ranged from 47 % to 92 % for the entire period of operation of the system depending upon various loads existing at different times. The occupant zone VAV box damper opening is used to maintain the thermal comfort in occupancy zone. The error in temperature, changes in error in temperature and local air velocity were considered as the input to the FLC and the output arrived at in the damper opening. There are three servers under operation in the software laboratory building to satisfy different loads. When the computer’s usage was less than 40 % only two servers were needed, while four servers were required for 40 – 65 % of computer usage. At maximum load (or) above 65 % of computer usage, six servers were under operation. Figure 6.8 shows the VAV box damper opening the server zone and it is varied from 37 % to 98 %.
Figure 6.7 Variations in occupant zone VAV box damper opening

Figure 6.8 Variations in server zone VAV box damper opening
The main problem in the VAV air conditioning system is poor air distribution in part load conditions. For proper air distribution, the supply air velocity should be maintained at a particular range. Very low velocity results in poor air distribution and high velocity produces noise in the duct. Figure 6.9 shows supply air velocity sensed by the velocity sensor for the entire system operation for 24 hours. The supply air velocity varied from a minimum value of 2.5 m/s to 4.0 m/s. This velocity variation provided the required air flow rate for the entire period of operation.

Figures 6.10 and 6.11 show the experimental result of supply air temperature variation for summer and winter with variation of thermal load conditions. The results revealed that, the supply air temperature was almost maintained by the fuzzy controller between 13°C to 17°C for the entire load conditions in summer. In winter, the refrigerant temperature ranged from 13°C to 15°C from 7.00 a.m to 7.00 p.m. For the remaining hours the outdoor air temperature was less than return air temperature. The economizer mode was activated and the compressor was switched off and 100% fresh air was taken because the supply air temperature was very close to outdoor temperature.
Figure 6.9 Variation of supply air velocity

Figure 6.10 Variation of supply air temperature in summer
Figure 6.11 Variation of supply air temperature in winter

The GA optimization results are the set points for the FLC to control the supply fan and VAV box. The zone temperature, local air velocity, RH and duct static pressure were maintained as per ASHRAE standard 55 (1992) for thermal comfort. Figures 6.12 and 6.13 show the temperature maintained in the occupancy zone in summer and winter conditions for the entire period of the operation of the system. The sensed temperature varied from 23.8°C to 24.2°C in summer and from 21.8°C to 22.2°C in winter. The temperature in the server zone is maintained at 18°C as per the manufacturer’s catalogue (Figure 6.14). Figure 6.15 shows the variation of local air velocity inside the occupant zone. The local air velocity varies within the limit of ASHRAE std. 55 (1992) from 0.150 m/s to 0.24 m/s. The value ensures comfort for the occupants in the zone.
Figure 6.12 Variations in occupant zone temperature in summer

Figure 6.13 Variations in occupant zone temperature in winter
Figure 6.14 Variations in server zone temperature

Figure 6.15 Variations in local air velocity in occupant zone
Figure 6.16 Variations in relative humidity

Figure 6.16 shows the variation of relative humidity from 48% to 70% for the entire period of operation of the system. This ensures comfort for the occupants. Therefore, thermal comfort inside the zone is maintained according to ASHRAE standards and it is up to a satisfactory level. In the case of a server zone, as there is almost no occupancy, the temperature inside the zone is maintained according to the manufacturer’s catalogue.

6.2.4 Air distribution analysis results for the VAV A/C system

The air distribution analysis is done using FLUENT 6.1. The distribution patterns for temperature and velocity in occupancy zone are obtained. The direction of flow can be clearly seen in this analysis. The air distribution to every corner of the zone can be seen by this analysis. In particular, the air distribution in a particular plane of a zone can also be visualized. This ensures the level of comfort that is required in an occupant zone as per ASHRAE standards. Figure 6.17 shows the scale model for occupancy zone with four return air and four supply air diffusers.
Figure 6.17  Scale model occupant zone

Figure 6.18  Contours of static temperature in occupancy zone
Figure 6.19  Static temperatures in comfort zone at 0.9 m and 1.2 m

Figure 6.20  Velocity vectors by magnitude
Figure 6.18 shows the temperature distribution inside the occupant zone. The supply inlet temperature is 13°C and in most parts of the zone it is maintained around 24°C. Though the temperature is a little too high in the corners of the zone the temperature in the comfort zone is properly maintained. Figure 6.19 shows the variation of temperature in the comfort zone at a height of 0.9 m and 1.2 m respectively. In that zone the temperature is maintained around 24°C. Figure 6.20 shows the magnitude of the velocity vector inside the occupancy zone in the range from 0.15 to 0.24 m/s.

### 6.3 IAQ AND ENERGY SAVING ANALYSIS

An experimental study was conducted on a multi-zone VAV air conditioning system with the genetic based Fuzzy Logic Controller in a software laboratory building. The readings are taken from an experimental set up for varying outdoor temperature and CO\textsubscript{2} concentration. The outdoor temperature variation was taken using the temperature sensor, which is connected to the fresh air damper inlet. The maximum and minimum temperatures recorded in the month of December ranged from 21.2°C to 29°C in winter and in the month of May, it ranged from 28.5°C to 37°C in summer as shown in Figure 6.21.

The CO\textsubscript{2} concentration of Genetic Algorithms optimization result for summer and winter is shown in Figure 6.22. The DCV for summer and the combined DCV and economizer cycle for winter reduces the IAQ problem. The CO\textsubscript{2} concentration in summer and winter are 850 ppm to 1050 ppm and 410 ppm to 1000 ppm respectively.

The variation of fresh air and return air damper opening for experimental study in winter season are shown in Figures 6.23 and 6.24. It shows that the fresh air damper opening is below 20% from 7 a.m to 7 p.m, when it is in the DCV mode. In this mode, depending upon the CO\textsubscript{2}
Figure 6.21 Variation of outdoor temperature in summer and winter season

Figure 6.22 Variation of CO$_2$ concentration based on GA optimization
Figure 6.23  Fresh air damper opening (Experimental)

Figure 6.24  Return air damper opening (Experimental)
concentration level the damper opens and hence over ventilation is avoided and thus the energy savings are possible. During the remaining periods the outdoor temperature was between the supply air temperature and return air temperature. When the time economizer cycle was activated, the fresh air damper was fully open and 100% fresh air was taken. At the same time, the exhaust damper opens proportionally to the fresh air damper.

The fresh air flow rate, return airflow rate and exhaust flow rate variations from the experiments for summer and winter in the VAV air conditioning system are plotted in Figures 6.25 and 6.26. In summer, only the CO$_2$ based DCV technique was activated with respect to building load, and the supply airflow rate was varied in VAV A/C system. At the same time, the exhaust damper and the fresh air damper were opened proportionally. In winter, both the DCV and economizer cycle were considered. When the outdoor temperature was less than the return air temperature, 100% fresh air was taken and the intake of return air was fully closed. When the outdoor temperature was higher than the return air temperature, the return airflow rate was more than 80% and the fresh airflow rate was found to be less than 20%.

Figure 6.27 shows the CO$_2$ contaminant concentration present in the conditioned space at different occupancy levels for both simulation and experiment. The simulated result exhibited that the CO$_2$ concentration was maintained between 980 ppm to 1080 ppm in the DCV mode whereas under the combined DCV and economizer cycle scheme, the CO$_2$ concentration varied from 380 ppm to 1060 ppm respectively. The test result shows that the CO$_2$ concentration was maintained between 950 ppm and 1040 ppm under the demand controlled ventilation technique and the CO$_2$ concentration was maintained between 350 ppm and 960 ppm under the DCV combined with the economizer cycle ventilation scheme. Both the test results suggested that, the
Figure 6.25 Variation of air flow rate in summer

Figure 6.26 Variation of air flow rate in winter
Figure 6.27  Comparison of the results of the simulation and experiment for CO₂ concentration variation

CO₂ concentration was under permissible levels for the corresponding change in occupancy levels. The CO₂ level in the outdoor air was 350 ppm, as measured by the duct mounted CO₂ sensor. The genetic based fuzzy logic controller maintained the CO₂ concentration within permissible limits by introducing the corresponding ventilation air quantity required to flow into the conditioned space through the control action based on the supply air fan.

The humidity problem in minimum fresh air in the DCV mode was due to the low cooling load and high outdoor air humidity. This showed that the mechanical cooling did not provide sufficient dehumidification. The RH problem is of concern when free cooling is used in summer. In the coldest month (December), the outdoor air dry-bulb temperature ranged generally from 21º C to 25º C and RH was about 70 % in the operating hours, from 7.0 p.m to 7.0 a.m and during this season the economical cycle mode can be activated.
6.4 PERFORMANCE OF EEV WITH FLC

In this study, the refrigerant flow rate has been varied to determine the performance of the system by using two expansion devices. For different cooling loads, the overall performance of VRV system with EEV using FLC was better than that of the VRV system with TXV.

In Figure 6.28 the superheating obtained with TXV, EEV with PID and EEV with FLC are shown. The improved performance concerns the EEV with FLC because after 70 s it fixes the superheating at ± 0.2°C with respect to the selected value, while the EEV with PID and TXV maintains the swing in a range of ± 0.5°C and ± 1°C respectively around the set value of 10°C. Table 6.4 shows that the EEV with FLC has performed better than the EEV with PID and TXV to maintain the same degree of superheating temperature. Since the peak overshoot, settling time and rise time for EEV with FLC are 0.9 %, 55 and 12 sec and these values compared to those of the EEV with PID and TXV are higher.

Figure 6.28 Variation of degree of superheating temperature
Table 6.4  Transient response comparison of TXV, EEV with PID and EEV with FLC for superheat temperature

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Peak overshoot (%)</th>
<th>Settling time (sec)</th>
<th>Rise time (sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>TXV</td>
<td>2.72</td>
<td>Not settled</td>
<td>24</td>
</tr>
<tr>
<td>EEV with PID</td>
<td>1.1</td>
<td>95</td>
<td>17</td>
</tr>
<tr>
<td>EEV with FLC</td>
<td>0.9</td>
<td>55</td>
<td>12</td>
</tr>
</tbody>
</table>

The mass flow rate through the EEV with FLC is more precisely controlled when compared with the EEV with PID and TXV at transient state as shown in Figure 6.29. The improved performance concerns the EEV with FLC because after 93 s it fixes the mass flow rate around the selected value, while the EEV with PID and TXV has taken 138 and 397 sec respectively.
Table 6.5 shows that the EEV with FLC has performed better than the EEV with PID and TXV to regulate the mass flow rate of the refrigerant. Since the peak overshoot, settling time and rise time for EEV with FLC are 14, 93 and 12 sec and these values are higher compared to those of the EEV with PID and TXV.

Table 6.5  
Transient response comparison of TXV, EEV with PID and EEV with FLC for mass flow rate

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Peak overshoot (%)</th>
<th>Settling time (s)</th>
<th>Rise time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>TXV</td>
<td>19</td>
<td>397</td>
<td>18</td>
</tr>
<tr>
<td>EEV with PID</td>
<td>17</td>
<td>138</td>
<td>15</td>
</tr>
<tr>
<td>EEV with FLC</td>
<td>14</td>
<td>93</td>
<td>12</td>
</tr>
</tbody>
</table>

Figure 6.30 Dynamic response of system pressure using EEV with FLC
In Figures 6.30 and 6.31 the dynamic response of the evaporator inlet pressure ($P_{ei}$), evaporator outlet pressure ($P_{eo}$), condenser inlet pressure ($P_{ci}$) and condenser outlet pressure ($P_{co}$) of the EEV with FLC and TXV is shown. It can be seen that the EEV opens more slowly than the TXV. As a result, the initial drop in the evaporating and suction pressure is lower, and the rise in the discharge pressure is higher, for the EEV than for the TXV. This creates an overshoot in power consumption, which reduces as the degree of superheat at the evaporator outlet increases. The evaporating and suction pressures for the thermostatic valve fall sharply and linearly together, indicating that for the first few seconds there is no refrigerant flow through the valve. The superheat temperature of the results of the experiment for the EEV with FLC at steady state is shown in Figure 6.32. It is seen that the superheat temperature is maintained around the set point value.
Figure 6.32 Variation of degree of superheating temperature

Figure 6.33 Simulation result of variation of refrigerant mass flow rate
Figures 6.33 and 6.34 show the results of the simulation and experiment of refrigerant mass flow rate using the TXV and EEV with FLC for the summer and winter season. The results of the experiment shows that the variation of mass flow rate, using the TXV ranges from 0.0191 to 0.0432 kg/s in summer and from 0.0161 to 0.0422 kg/s in winter inspite of load variations. When using the EEV with FLC the mass flow rate ranged from 0.0141 to 0.0374 kg/s in summer and from 0.0125 to 0.0367 kg/s in winter inspite of load variations. The mass flow rate through the TXV was higher during any load variations compared to the EEV with FLC. In addition, as the degree of superheating at the compressor discharge decreases, the life of the compressor could increase. Under all test conditions, the mass flow measured through the mass flow meter with an accuracy of ± 2 % in full-scale reading. The maximum deviation from the result of experiment is only ± 8 % compared to the simulation result. A distinct variation is observed in the mass flow rate of refrigerant through the TXV and EEV with FLC.
6.4.1 The influence of refrigerant mass flow rate on compressor speed

The mass flow rate of the refrigerant has a direct relation to the speed of the compressor. The result of the simulation and experiment for the variation of compressor speed with the modulated refrigerant mass flow rate is represented in Figures 6.35 and 6.36 respectively for summer conditions.

Figure 6.35 Compressor speed variations with respect to time in summer
Figure 6.36 Compressor speed variations with respect to time in summer

The result of the experiment of compressor speed for a fixed ventilation scheme is observed to vary from 3120 rpm to 6790 rpm and the variation of mass flow rate is ranged from 0.018 to 0.040 kg/s through EEV inspite of load variations. The DCV scheme when applied to summer conditions shows that the compressor speed varies between 2520 rpm and 6620 rpm for the mass flow rate ranged from 0.016 to 0.038 kg/s inspire of load variations. The maximum deviation from the result of experiment is only ±10 % compared to the simulation result.

The result of the simulation and experiment of variation of compressor speed with the modulated refrigerant mass flow rate is represented in Figures 6.37 and 6.38 for winter conditions. The compressor speed for a fixed ventilation scheme is observed to vary from 2400 rpm to 5920 rpm when the variation of mass flow rate ranged from 0.015 to 0.037 kg/s through the EEV with FLC inspire of load variations. The DCV scheme when applied to winter conditions infers that the compressor speed
varies between 1960 rpm and 5850 rpm for the variation of mass flow rate ranged from 0.013 to 0.035 kg/s through the EEV with FLC inspite of load variations. The combined DCV and economizer cycle energy conservation technique applied to winter conditions shows that the compressor speed varies between 2660 rpm and 5850 rpm and the variation of mass flow rate ranged from 0.021 to 0.034 kg/s from 7 a.m to 7 p.m, when the DCV mode was activated. During the remaining hours the economizer cycle was activated when the compressor was switched off and there was no mass flow rate through EEV. The maximum deviation from the result of experiment is only ±9 % compared to the simulation result.

Figure 6.37 Compressor speed variations with respect to time in winter
Figure 6.38 Compressor speed variations with respect to time in winter

6.5 PERFORMANCE COMPARISON OF VRV-VAV A/C SYSTEM USING GENETIC BASED FLC

6.5.1 The influence of GA optimization on FLC set points

Figure 6.39 shows the Genetic Algorithm optimization variables like suction pressure and refrigerant temperature (RT) under summer and winter conditions. The Genetic Algorithm (GA) optimization results show that the refrigerant temperature ranged from 8°C to 12°C in summer. In winter, the refrigerant temperature ranged from 7°C to 10°C from 7.00 am to 7.00 pm and for the remaining hours the economizer cycle is activated during which the compressor is switched off. The GA results show that the refrigerant temperature is very close to the outdoor temperature. The coil temperature is around 8°C to 13°C and the suction pressure is in the range of 512 kPa to 520 kPa and 412 to 418 kPa for summer and winter conditions respectively inspite of load variations.
Figure 6.39  Variation of temperatures and suction pressure based on GA optimization

Figures 6.40 and 6.41 shows the results of a variation in refrigerant temperature corresponding to a variation in thermal load conditions during summer and winter. The results show that, the refrigerant temperature was almost maintained by fuzzy controller from 6°C to 8°C for the all load conditions in summer. In winter, the refrigerant temperature ranged from 7°C to 10°C from 7.00 am to 7.00 pm. For the remaining hours the economizer cycle was activated during which the compressor was switched off and 100% fresh air was taken and the refrigerant temperature was related to outdoor temperature. By providing control over refrigerant temperature, the VRV-VAV system is capable of meeting the variable cooling load demand.
Figure 6.40  Variation of refrigerant temperature in summer

Figure 6.41 Variation of refrigerant temperature in winter
Figure 6.42  Variation of suction pressure in summer

Figure 6.43  Variation of suction pressure in winter
The effect of variation of suction pressure of the variable speed rotary compressor utilized in the experiment is shown in Figures 6.42 and 6.43 in summer and winter conditions. The test result shows that when the cooling load variation occurs the suction pressure variation was observed to be between 510 kPa and 520 kPa in summer conditions. In winter conditions, the suction pressure ranged from 408 kPa to 414 kPa during the operating hours of 7.00 am to 7.00 pm. For the remaining hours the economizer cycle was activated during which the compressor was switched off and the suction pressure was close to the system ideal pressure.

6.5.2 The influence of genetic algorithm on fan power

The results of the simulation and experiment of the variation of fan power consumption is represented in Figures 6.44 and 6.45 respectively for summer conditions. In the VRV-VAV system, the speed of the supply air fan is varied by the effective utilization of the genetic algorithm optimum variable, which is the set point for the fuzzy logic controller. Based on the static pressure present in the supply air duct and supply air velocity, the fan speed is considerably modulated to offset the cooling load. It is a well-known fact that in the DCV technique the fan power is considerably reduced in the VAV system. The results of the simulation show that the range of power consumed by the supply fan under DCV with GA optimisation was 120 W to 540 W. The results of the experiment show that the range of power consumed by the supply air fan under DCV in summer conditions was 90 W to 440 W. In the CAV system, the supply air fan operated at a fixed speed and consumed a constant power of around 460 W irrespective of load variations. The energy consumption of the CAV system is higher when compared to that of the VAV system under part load conditions. It is known that, when the genetic algorithm is applied to optimize the supply air fan characteristics, the result will be a substantial reduction of power consumption on the fan side.
The results of the simulation and experiment for the winter design conditions based on GA optimization for the supply air fan are shown in Figures 6.46 and 6.47. It is observed that, the ratio of fan power consumed during part-load and full-load conditions ranges from 70 W to 390 W when using the DCV scheme. The influence of genetic algorithm optimisation result on the supply fan power decrease of the power consumption ranged from 55 W to 380 W. The fan power was more in the economizer cycle mode compared to the DCV mode in the VAV air conditioning system.

![Graph showing fan power variation over time](image)

**Figure 6.44** Simulation result of variation of fan power in summer
Figure 6.45 Experimental result of variation of fan power in summer

Figure 6.46 Simulation result of variation of fan power in winter
6.5.3 The effect of genetic algorithm on compressor power

The influence of the genetic algorithm optimization scheme on compressor power is presented. In the present system, the genetic algorithm optimum variable is the fuzzy logic controller set points to modulate the power consumed by the variable speed compressor (VSC). The results of the simulation and experiment for the summer design conditions for variable speed compressor power consumption is shown in figure 6.48 and 6.49. The results of the experiment show that, under the fixed ventilation and DCV schemes, the variable speed compressor operates from 1.23 kW to 3.95 kW and 0.98 kW to 3.75 kW inspire of load variations. Based on the optimization process done on compressor power consumption, the result shows that, compressor power varies from 1.18 kW to a maximum of 3.70 kW during summer conditions. The maximum deviation from the result of the experiment is in $\pm 12\%$ compare to the simulation result.
The result of the simulation and experiment of variable speed compressor power consumption are shown in figures 6.50 and 6.51 for the winter conditions. It is observed that the power consumed by the VSC varies from 0.87 kW to 3.65 kW, 0.71 kW to 3.20 kW and 0 to 3.20 kW under the fixed ventilation, the DCV and combined DCV & EC techniques respectively, which is less than it consumed in summer conditions. The influence of the genetic algorithm optimization process performed on compressor power consumption, the result shows that, compressor power is less, when combined DCV and economizer cycle during operating design conditions. In the DCV mode, the compressor power consumption ranged from 0.81 kW to 3.57 kW in the operating hours of 7.00 a.m to 7.00 p.m. For the remaining hours the economizer cycle was activated during which the compressor was switched off and 100% fresh air was taken when no power was consumed. The power consumed by the variable speed compressor is less than that of the Constant Speed Compressor (CSC). The GA optimization helps to reduce the power consumed by the variable speed compressor utilized in the VRV-VAV A/C system. The maximum deviation from the result of the experiment is in ± 9% compare to the simulation result.

Figure 6.48 Simulation result of variation of compressor power in summer
Figure 6.49 Experimental result of variation of compressor power in summer

Figure 6.50 Simulation result of variation of compressor power in winter
6.5.4 Energy saving potential in VRV-VAV Air conditioning

The energy saving characteristics of the combined VRV-VAV A/C system is evaluated on the basis of the results of the simulation and experiment as depicted in the Figures 6.52 and 6.53 respectively. The results of the experiment illustrate that, in summer conditions, the energy saving potential of the VRV-VAV A/C system utilizing fixed ventilation, is only 8% at peak load and 50% at part load ratio. When the DCV mode is applied, the energy saving achieved was 18% at peak load and 62% at part load compared to that of the constant speed compressor with fixed ventilation. The maximum deviation from the result of the experiment is in ± 11% compare to the simulation result.

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 CO₂ concentration. The energy saving was evaluated on the
basis of the result simulation and experimental as depicted in the Figures 6.54 and 6.55 respectively. The results of the experiment show that, when operated under winter and part-load conditions, the system yields a maximum potential energy saving of 62%, 68% and 86% for fixed, DCV and combined DCV and economizer cycle schemes respectively. An improvement of energy saving was observed in the case of the combined DCV and economizer cycle using genetic based FLC in the VRV-VAV air conditioning system. This 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.

![Graph showing energy savings comparison](image)

**Figure 6.52** Simulation result of energy saving comparison of the VRV-VAV A/C system in summer conditions
Figure 6.53  Experimental results of energy saving comparison of the VRV-VAV A/C system in summer conditions

Figure 6.54  Simulation results of energy saving comparison of the VRV-VAV A/C system in winter conditions
Figure 6.55  Experimental result of energy saving comparison of the VRV-VAV A/C system in winter conditions

Figure 6.56  Simulation result of per day energy saving comparison of the VRV-VAV A/C system for the summer and winter conditions
The simulation approach of the VRV-VAV air conditioning system energy saving potential is observed in Figure 6.56. While the genetic algorithm was applied, the energy conservation per day was increased to 50% and 59% for fixed and DCV techniques adopted under summer design conditions. For winter design conditions, the results inferred that there exists still more energy saving potential of 60%, 70% and 86% per day for fixed, DCV and combined DCV and economizer techniques respectively. Based on the simulation result it is observed that, the combined VRV-VAV A/C system can be considered to be high-energy efficient air conditioning.

The genetic algorithm approach still increases the system energy saving potential in the result of the experiment is observed in Figure 6.57. The energy conservation per day is increased to 41% and 49% for the fixed and the DCV techniques adopted under summer design conditions using the
genetic algorithm approach. Generally, the weather in December, which can be classified as winter season; the month of November, December, January, and February have plenty of the sunshine and comfortable temperature. In January, the economizer is activated frequency because the average outdoor temperature is around 24°C and never falls below 20°C. The indoor temperature set-point was 22°C. For winter design conditions, the results inferred that there exists greater energy savings potential of 46%, 55% and 68% per day for the fixed, the DCV and the combined DCV and EC techniques respectively. Based on the result of the experiment it is observed that, the combined VRV-VAV A/C system with combined DCV and economizer cycle using genetic based FLC can be considered to be a high-energy efficient air conditioning technique. It has an enormous energy savings potential compared to the conventional CAV A/C systems with fixed ventilation.