

ISTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY

**EXPERIMENTAL AND SIMULATION EVALUATION OF A
MULTI-SPLIT TYPE AIR CONDITIONING SYSTEM
UNDER STEADY-STATE AND TRANSIENT CONDITIONS**

**Ph.D. Thesis by
Tolga Nurettin AYNUR, M.Sc.**

**Department: Mechanical Engineering
Programme: Mechanical Engineering**

FEBRUARY 2008

**EXPERIMENTAL AND SIMULATION EVALUATION OF A
MULTI-SPLIT TYPE AIR CONDITIONING SYSTEM
UNDER STEADY-STATE AND TRANSIENT CONDITIONS**

**Ph.D. Thesis by
Tolga Nurettin AYNUR, M.Sc.
(503022003)**

Date of Submission:	18 December 2007
Date of Defense Examination:	22 February 2008

Supervisor (Chairman):	Prof. Dr. Nilüfer EĞRİCAN
Members of the Examining Committee:	Prof.Dr. Kadir KIRKKÖPRÜ (İ.T.Ü.)
	Prof.Dr. İsmail TEKE (Y.T.Ü.)
	Prof.Dr. Ahmet ARISOY (İ.T.Ü.)
	Prof.Dr. Sibel ÖZDOĞAN (M.Ü.)

FEBRUARY 2008

**ÇOKLU-SPLİT TİP BİR İKLİMLENDİRME SİSTEMİNİN
SÜREKLİ VE GEÇİCİ REJİM ŞARTLARI ALTINDA
DENEYSEL VE SAYISAL OLARAK İNCELENMESİ**

**DOKTORA TEZİ
Y. Müh. Tolga Nurettin AYNUR
(503022003)**

**Tezin Enstitüye Verildiği Tarih : 18 Aralık 2007
Tezin Savunulduğu Tarih : 22 Şubat 2008**

Tez Danışmanı : Prof.Dr. Nilüfer EĞRİCAN
Diğer Jüri Üyeleri Prof.Dr. Kadir KIRKKÖPRÜ (İ.T.Ü.)
Prof.Dr. İsmail TEKE (Y.T.Ü.)
Prof.Dr. Ahmet ARISOY (İ.T.Ü.)
Prof.Dr. Sibel ÖZDOĞAN (M.Ü.)

ŞUBAT 2008

TABLE OF CONTENTS

ABBREVIATIONS	vii
LIST OF TABLES	viii
LIST OF FIGURES	x
LIST OF SYMBOLS	xxii
SUMMARY	xxv
ÖZET	xxvi
1. INTRODUCTION	1
2. LITERATURE REVIEW	4
2.1 Multi-Split VRF System Studies	4
2.2 Thermal Comfort Studies	13
2.3 Summary of the Literature Review	20
3. MOTIVATION AND OBJECTIVES	23
3.1 Motivation	23
3.2 Research Objectives	24
3.2.1 Experimental objectives	24
3.2.2 Modeling objectives	25
4. EXPERIMENTAL SETUP AND INSTRUMENTATION	26
4.1 Floor Layout	26
4.2 Multi-Split VRF System	27
4.2.1 Outdoor units	28
4.2.2 Indoor units	30
4.2.3 Connection between the outdoor unit and the indoor units	31
4.2.4 Indoor air temperature control	33
4.2.4.2 Individual control mode	34
4.2.4.3 Master control mode	35
4.2.4.4 Changing of the control modes	36
4.3 Ventilation Units	38
4.4 Existing Cooling and Heating System	40
4.5 Scheduler	42
4.6 Measurement Instrumentation	43
4.6.1 Airside measurements	43
4.6.2 Refrigerant side measurements	49
4.6.3 Power consumption measurements	51
4.6.4 Data acquisition system	52
4.6.5 Uncertainty Analysis	53
5. EVALUATION METHODOLOGY	55

5.1	Efficiency of the Multi-Split VRF System	55
5.2	Provided Thermal Comfort	60
6.	EXPERIMENTAL RESULTS	62
6.1	Effect of the Control Modes in the Cooling Performance	64
6.1.1	Daily comparison at 25 °C set temperature	64
6.1.2	Seasonal comparison at 25 °C set temperature	107
6.1.3	Seasonal comparison at 20 °C set temperature	121
6.2	Effect of the Ventilation in the Cooling Performance	138
6.2.1	Seasonal comparison	140
6.3	Effect of the Control Modes in the Heating Performance	153
6.3.2	Daily comparison at 23.3 °C set temperature	154
6.3.3	Seasonal comparison at 23.3 °C set temperature	163
6.3.4	Seasonal comparison at 26.1 °C set temperature	178
6.4	Effect of the Ventilation in the Heating Performance	192
6.4.1	Seasonal comparison	192
7.	SIMULATION	205
7.1	Introduction	205
7.2	Building Model	205
7.3	Multi-Split VRF and HRV Models	208
7.3.1	Multi-Split VRF Model	208
7.3.2	HRV Model	212
7.4	Scheduling	214
7.5	Weather Data	214
7.6	Validation of the Model with Experimental Results	216
7.7	Discussion of Simulation Results	221
7.7.1	Ventilation assisted system	230
7.7.2	Non-ventilated system	260
8.	CONCLUSIONS	286
8.1	Conclusions	286
8.2	Future Work	291
	REFERENCES	293
	APPENDICES	296
	CURRICULUM VITEA	316

ACKNOWLEDGEMENTS

I would like to express my sincere appreciation and thanks for my advisor, Dr. Nilüfer EĞRİCAN.

Without her help and collaboration with University of Maryland, College Park, this work would not have been complete.

I would also like to thank to Dr. Reinhard Radermacher for giving me the opportunity to conduct my research at University of Maryland, College Park.

Special thanks go to all my colleagues at Center for Environmental Energy Engineering, CEEE and to my colleagues in Japan; Mr. Michio Moriwaki, Mr. Takeshi Hikawa, Mr. Nobuki Matsui and Mr. Sumio Shiochi.

Finally I would like to express my deepest gratitude to my wife and my parents whose encouragement and love always motivated me.

December 2007

Tolga Nurettin AYNUR
Mechanical Engineering

ABBREVIATIONS

AHU	: Air handling unit
ARI	: Air conditioning and refrigeration institute
ASHRAE	: American society of heating, refrigeration and air-conditioning engineers
BLAST	: Building loads analysis and system thermodynamics
COP	: Coefficient of performance
CPF	: Cooling performance factor
DB	: Dry bulb
DX	: Direct expansion
EER	: Energy efficiency ratio
EES	: Engineering equation solver
EEV	: Electronic expansion valve
EIR	: Energy input ratio
ERV	: Energy recovery ventilation
FSC	: Fixed speed compressor
HPF	: Heating performance factor
HRV	: Heat recovery ventilation
HVAC	: Heating, ventilation and air-conditioning
IDC	: Inverter driven compressor
IUAis	: Indoor unit located in aisle
IUEle	: Indoor unit located in elevator zone
IURA	: Indoor unit located in room A zone
IURB	: Indoor unit located in room B zone
IURC	: Indoor unit located in room C zone
IURD	: Indoor unit located in room D zone
IURE	: Indoor unit located in room E zone
IURec	: Indoor unit located in receptionist area
MRT	: Mean radiant temperature
NOAA	: National oceanic and atmospheric administration
OU1	: Outdoor unit 1
OU2	: Outdoor unit 2
PLF	: Part load fraction
PLR	: Part load ratio
PMV	: Predicted mean vote
PPD	: Predicted percentage of dissatisfied
TSS	: Thermal sensation scale
U.S.	: United States
VAV	: Variable air volume
VRF	: Variable refrigerant flow
WB	: Wet bulb

LIST OF TABLES

	<u>Page No</u>
Table 2.1 : Multiple regression equations.....	17
Table 2.2 : Equations for predicting thermal sensation of men, women, and men and women combined.....	18
Table 4.1 : Volume of each zone.....	26
Table 4.2 : Catalogue information of the outdoor unit.....	30
Table 4.3 : Type, location and the name of the indoor units.....	31
Table 4.4 : HRV units and the locations.....	39
Table 6.1 : Weekly schedule of the cooling tests in 2006.....	62
Table 6.2 : Weekly schedule of the cooling tests in 2007.....	63
Table 6.3 : Weekly schedule of the heating tests in 2006.....	63
Table 6.4 : Window areas of System1 and System2 zones.....	69
Table 6.5 : Number of the internal load sources of System1 and System2 zones.....	70
Table 6.6 : Comparison of the energy consumption of the outdoor units for the individual and master control modes.....	95
Table 6.7 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone on June 01, 2007 and May 31, 2007....	97
Table 6.8 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone on June 01, 2007 and May 31, 2007....	99
Table 6.9 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes at 25°C set temperature based on the ASHRAE summer thermal comfort zone.....	109
Table 6.10 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes at 25°C set temperature based on the ASHRAE summer thermal comfort zone.....	111
Table 6.11 : Comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems at 25°C set temperature based on the ASHRAE summer thermal comfort zone...	144
Table 6.12 : Comparison of the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems at 25°C set temperature based on the ASHRAE summer thermal comfort zone...	144
Table 6.13 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature.....	167
Table 6.14 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature.....	167
Table 6.15 : Comparison of the indoor condition data of System2 zones provided by continuous and synchronized indoor unit fan operations based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature...	178
Table 6.16 : Comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature...	193

Table 6.17	: Comparison of the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature...	195
Table 6.18	: Comparison of the indoor condition data of System2 zones provided by the non-ventilated system for the continuous and synchronized indoor unit fan operations based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature.....	204
Table 7.1	: Material list and the associated thermal conductivity.....	207
Table 7.2	: Light fixtures used in the office suite.....	207
Table 7.3	: Recommended heat gain from office equipments.....	208
Table 7.4	: Default weather file of EnergyPlus for Sterling, VA.....	215
Table A.1	: Catalogue data for IURA.....	296
Table A.2	: Catalogue data for IURB.....	296
Table A.3	: Catalogue data for IURC, IURE and IUREc.....	297
Table A.4	: Catalogue data for IURD, and IUEle.....	297
Table A.5	: Catalogue data for IUAs.....	298
Table L.1	: Individual power consumptions of the HRV units.....	314
Table L.2	: Individual power consumptions of the indoor units.....	315

LIST OF FIGURES

	Page No
Figure 1.1 : Schematic diagram of a multi-split VRF system with four indoor units.....	1
Figure 2.1 : Schematic diagram of the multi-split air VRF conditioning system.....	4
Figure 2.2 : Schematic diagram of the multi-split three-pipe VRF system.....	5
Figure 2.3 : Schematic diagram of the multi-split three-pipe VRF system.....	7
Figure 2.4 : Schematic diagram of a multi-split VRF system with two indoor units.....	8
Figure 2.5 : Capacity control of the compressor.....	9
Figure 2.6 : System configuration.....	9
Figure 2.7 : Test system for the multi-split VRF system.....	11
Figure 2.8 : Comfort parameters.....	14
Figure 2.9 : Variation of the ASHRAE thermal sensation vote with the indoor air temperature.....	19
Figure 4.1 : Floor layout.....	27
Figure 4.2 : Layout of the multi-split VRF systems.....	28
Figure 4.3 : Outdoor units.....	28
Figure 4.4 : Inside of the outdoor unit.....	29
Figure 4.5 : Ceiling mounted cassette type indoor unit.....	30
Figure 4.6 : Wall mounted type indoor unit.....	30
Figure 4.7 : Branch for the refrigerant pipe.....	32
Figure 4.8 : Multi-split VRF system.....	32
Figure 4.9 : Thermostat.....	33
Figure 4.10 : Thermostat and indoor unit.....	34
Figure 4.11 : Locations of the individual thermostats.....	35
Figure 4.12 : Location of the master thermostat.....	36
Figure 4.13 : Relay box.....	36
Figure 4.14 : Schematic drawing of the relay boxes for the individual control mode.....	37
Figure 4.15 : Schematic drawing of the relay boxes for the master control mode.....	38
Figure 4.16 : Schematic drawing of an HRV unit.....	39
Figure 4.17 : Layout of the HRV units.....	40
Figure 4.18 : Layout of the existing cooling and heating system.....	41
Figure 4.19 : Air damper for supply duct of the existing cooling system.....	42
Figure 4.20 : Shut off valve for the existing heating system.....	42
Figure 4.21 : Scheduler.....	42
Figure 4.22 : Measurement location under the indoor unit.....	43
Figure 4.23 : Measurement locations of the indoor air temperature and relative humidity.....	44
Figure 4.24 : Measurement locations of the occupant location: 0.1m and 0.6m above the floor.....	45
Figure 4.25 : Measurement locations: 1m away from the walls.....	45
Figure 4.26 : Measurement location of the outdoor air temperature and relative humidity, (a) shielding, (b) location.....	46
Figure 4.27 : Thermocouple grid around the outdoor unit heat exchanger.....	47

Figure 4.28	: Measurement location of the indoor air discharge temperature...	47
Figure 4.29	: Measurement location of the outdoor unit air outlet temperature.	48
Figure 4.30	: Measurement locations of HRV1	48
Figure 4.31	: Measurement locations for the compressor refrigerant discharge and suction temperature.....	49
Figure 4.32	: Measurement locations for the compressor refrigerant discharge and suction temperature and pressure.....	50
Figure 4.33	: Measurement locations of the indoor unit temperature for the cooling mode.....	51
Figure 4.34	: Measurement locations of the indoor unit temperature for the heating mode.....	51
Figure 4.35	: Power meter.....	52
Figure 4.36	: Data acquisition system.....	53
Figure 4.37	: Data acquisition system monitoring window.....	53
Figure 5.1	: Variation of the refrigerant mass flow rate with respect to the evaporating temperature for different inverter frequencies and the condensing temperatures when FSC is OFF	56
Figure 5.2	: Variation of the refrigerant mass flow rate with respect to evaporating temperature for different inverter frequencies and condensing temperatures when FSC is ON.....	56
Figure 5.3	: ASHRAE summer and winter comfort zones.....	60
Figure 6.1	: Outdoor temperature variations for the individual control mode tests at 25°C set temperature.....	64
Figure 6.2	: Outdoor relative humidity variations for the individual control mode tests at 25°C set temperature.....	65
Figure 6.3	: Outdoor temperature variations for the master control mode tests at 25°C set temperature.....	65
Figure 6.4	: Outdoor relative humidity variations for the master control mode tests at 25°C set temperature.....	66
Figure 6.5	: Outdoor temperature variations on June 01, 2007 and May 31, 2007.....	67
Figure 6.6	: Outdoor relative humidity variations on June 01, 2007 and May 31, 2007.....	67
Figure 6.7	: Indoor temperatures of System1 zones and outdoor temperature on June 01, 2007.....	68
Figure 6.8	: Indoor temperatures of System2 zones and outdoor temperature on June 01, 2007.....	69
Figure 6.9	: TSS variation of System1 zones on June 01, 2007.....	71
Figure 6.10	: Indoor temperature and the TSS variation of Room A on June 01, 2007.....	71
Figure 6.11	: TSS variation of System2 zones on June 01, 2007.....	72
Figure 6.12	: Indoor units operation of System1 zones in individual control mode on June 01, 2007.....	73
Figure 6.13	: Indoor unit air discharge temperatures of System1 on June 01, 2007.....	73
Figure 6.14	: Relationship between the operation and the EEV opening of IURA on June 01, 2007.....	74
Figure 6.15	: Indoor units operation of System2 zones in individual control mode on June 01, 2007.....	75
Figure 6.16	: Indoor unit air discharge temperatures of System2 on June 01, 2007.....	75
Figure 6.17	: Relationship between the operation and the EEV opening of IURec on June 01, 2007.....	76
Figure 6.18	: Power consumption of OU1 and OU2 (June 01, 2007).....	77

Figure 6.19	: ON time ratios of IDC and FSC of OU1 and OU2 on June 01, 2007.....	77
Figure 6.20	: Relationship between the operation of System1 indoor units and OU1 power consumption on June 01, 2007.....	78
Figure 6.21	: Relationship between the operation of System1 indoor units and OU1 power consumption during the period of 13:00-16:30 on June 01, 2007.....	78
Figure 6.22	: Relationship between the inverter frequency of IDC and OU1 power consumption during the period of 13:00-16:30 on June 01, 2007.....	79
Figure 6.23	: Condensing and evaporating pressures of OU1 on June 01, 2007.....	80
Figure 6.24	: Refrigerant temperatures of IDC suction, condenser inlet, condenser outlet, and IDC discharge for OU1 on June 01, 2007...	80
Figure 6.25	: Relationship between the inverter frequency of IDC and the total refrigerant mass flow rate for OU1 on June 01, 2007.....	81
Figure 6.26	: Total and the individual refrigerant mass flow rates of System1 indoor units (June 01, 2007).....	81
Figure 6.27	: Relationship between the operation of System2 indoor units and OU2 power consumption on June 01, 2007.....	82
Figure 6.28	: Relationship between the operation of System2 indoor units and OU2 power consumption during the period of 14:30-17:30 on June 01, 2007.....	82
Figure 6.29	: Relationship between the inverter frequency of IDC and OU2 power consumption during the period of 14:30-17:30 on June 01, 2007.....	83
Figure 6.30	: Condensing and evaporating pressures of OU2 on June 01, 2007.....	83
Figure 6.31	: Refrigerant temperatures of IDC suction, condenser inlet, condenser outlet, and IDC discharge of OU2 on June 01, 2007....	84
Figure 6.32	: Relationship between the total refrigerant mass flow rate and the inverter frequency of IDC for OU2 on June 01, 2007.....	84
Figure 6.33	: Total and the individual refrigerant mass flow rates of System2 indoor units on June 01, 2007.....	85
Figure 6.34	: Indoor temperature of System1 zones and outdoor temperature variations for master control mode on May 31, 2007.....	86
Figure 6.35	: Indoor temperatures of System2 zones and outdoor temperature on May 31, 2007.....	86
Figure 6.36	: TSS variation of System1 zones on May 31, 2007.....	87
Figure 6.37	: TSS variation of System2 zones on May 31, 2007.....	87
Figure 6.38	: Indoor units operation of System1 and System2 zones in master control mode on May 31, 2007, (a) System1, (b) System2.....	88
Figure 6.39	: Indoor unit air discharge temperatures of System1 and System2 zones in master control mode on May 31, 2007, (a) System1, (b) System2.....	89
Figure 6.40	: Relationship between the operation and the EEV opening of System2 indoor units on May 31, 2007.....	90
Figure 6.41	: Power consumption of OU1 and OU2 on May 31, 2007.....	91
Figure 6.42	: Comparison of the inverter frequencies of IDC of OU1 and OU2.	91
Figure 6.43	: Comparison of the total refrigerant mass flow rates of OU1 and OU2.....	92
Figure 6.44	: Relationship between the operation and the EEV openings of System1 indoor units on May 31, 2007.....	92
Figure 6.45	: Relationship between the operation and the EEV openings of System2 indoor units on May 31, 2007.....	93

Figure 6.46	: Condensing and evaporating pressures of OU1 and OU2 on May 31, 2007.....	93
Figure 6.47	: Comparison of power consumption for the individual and master control modes, (a) OU1, (b) OU2.....	94
Figure 6.48	: Comparison of the total power consumption of OU1 and OU2 for the individual and master control modes.....	95
Figure 6.49	: Indoor conditions of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	96
Figure 6.50	: Indoor conditions of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	98
Figure 6.51	: Indoor temperature variations of the ASHRAE thermal comfort locations for the individual and master control modes, (a) individual control mode, (b) master control mode.....	100
Figure 6.52	: TSS variations of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	101
Figure 6.53	: TSS variations of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	102
Figure 6.54	: TSS variations of System1 and System2 combined zones provided by the individual and master control modes, (a) System1, (b) System2.....	103
Figure 6.55	: Variation of the total cooling capacity, outdoor unit power consumption, total power consumption of the indoor units and CPF of System1 and System2 for the individual control mode on June 01, 2007, (a) System1, (b) System2.....	104
Figure 6.56	: Variation of the cooling capacity, outdoor unit power consumption, total power consumption of the indoor units and CPF of System1 and System2 for the master control mode on May 31, 2007, (a) System1, (b) System2.....	106
Figure 6.57	: Comparison of the variations of CPF for the individual and master control modes.....	107
Figure 6.58	: Number of data points for 25°C set temperature tests.....	108
Figure 6.59	: Outdoor conditions for 25°C set temperature tests.....	108
Figure 6.60	: Indoor conditions of System1 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode.....	110
Figure 6.61	: Indoor conditions of System2 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode.....	112
Figure 6.62	: TSS variations of System1 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode.....	113
Figure 6.63	: TSS variations of System2 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode.....	114
Figure 6.64	: TSS variations of System1 and System2 combined zones provided by the individual and master control modes at 25°C set temperature.....	115
Figure 6.65	: Comparison of the daily averaged energy consumption of the individual and master control modes at 25°C set temperature.....	115

Figure 6.66	: Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 25°C set temperature, (a) OU1, (b) OU2.....	117
Figure 6.67	: Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 25°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF.....	120
Figure 6.68	: Seasonal ON/OFF time ratio of OU2 for the individual and master control modes at 25°C set temperature.....	121
Figure 6.69	: Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes at 25°C set temperature.....	121
Figure 6.70	: Number of data points for 20°C set temperature tests.....	122
Figure 6.71	: Outdoor conditions for 20°C set temperature tests.....	122
Figure 6.72	: Indoor conditions of System1 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode.....	124
Figure 6.73	: Indoor conditions of System2 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode.....	125
Figure 6.74	: TSS variations of System1 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode.....	126
Figure 6.75	: TSS variations of System2 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode.....	127
Figure 6.76	: TSS variations of System1 and System2 combined zones provided by the individual and master control modes at 20°C set temperature.....	128
Figure 6.77	: Comparison of the daily averaged energy consumption of the individual and master control modes at 20°C set temperature.....	128
Figure 6.78	: Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 20°C set temperature, (a) OU1, (b) OU2.....	130
Figure 6.79	: Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 20°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF.....	132
Figure 6.80	: Seasonal ON/OFF time ratio of OU2 for the individual and master control modes at 20°C set temperature.....	133
Figure 6.81	: Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes at 20°C set temperature.....	133
Figure 6.82	: Variation of the inverter frequency, refrigerant pressures and outdoor unit power consumption.....	134
Figure 6.83	: Variation of the indoor unit refrigerant inlet and outlet.....	135
Figure 6.84	: Variation of the indoor unit air blowing temperature.....	135
Figure 6.85	: Variation of the indoor unit air inlet temperature.....	136
Figure 6.86	: Variation of the indoor and outdoor temperature.....	136
Figure 6.87	: Variation of the indoor unit capacities and the COP of the system.....	137
Figure 6.88	: Variation of the power consumption, total cooling capacity and the COP of the system with respect to the outdoor temperature...	137

Figure 6.89	: Comparison of the CPF and COP of the system.....	138
Figure 6.90	: Variation of the supply air temperature with respect to the outdoor air temperature.....	139
Figure 6.91	: Variation of the supply air humidity ratio with respect to the outdoor air humidity ratio.....	140
Figure 6.92	: Outdoor temperatures variations for the non-ventilated tests.....	140
Figure 6.93	: Outdoor relative humidity variations for the non-ventilated tests..	141
Figure 6.94	: Number of data points for the ventilation assisted and non-ventilated tests in cooling mode.....	141
Figure 6.95	: Outdoor conditions for the ventilation assisted and non-ventilated tests in cooling mode.....	142
Figure 6.96	: Indoor conditions of System1 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated.....	143
Figure 6.97	: Indoor conditions of OU2 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated.....	145
Figure 6.98	: TSS variations of System1 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated.....	146
Figure 6.99	: TSS variations of System2 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated.....	147
Figure 6.100	: TSS variations of System1 and System2 combined zones provided by the ventilation assisted and non-ventilated systems...	148
Figure 6.101	: Comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated systems.....	148
Figure 6.102	: Percentage ON time ratio of OU1 and OU2 for the ventilation assisted and non-ventilated systems, (a) OU1, (b) OU2.....	150
Figure 6.103	: Variation of cooling energy, energy consumption of OU2 and CPF of the ventilation assisted and non-ventilated system with respect to the daily averaged outdoor temperature, (a) ventilation assisted, (b) non-ventilated, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF.....	152
Figure 6.104	: Seasonal ON/OFF time ratio of OU2 for the ventilation assisted and non-ventilated systems.....	152
Figure 6.105	: Seasonal ON/OFF time ratio of FSC of OU2 for the ventilation assisted and non-ventilated systems.....	153
Figure 6.106	: Refrigerant path in the heating mode.....	154
Figure 6.107	: Outdoor temperature variations on February 19, 2007 and March 08, 2007.....	155
Figure 6.108	: Outdoor relative humidity variations on February 19, 2007 and March 08, 2007.....	155
Figure 6.109	: Outdoor temperature and indoor temperatures of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	157
Figure 6.110	: Operation of the indoor units of System1 zones for the individual and master control modes, (a) individual control mode, (b) master control mode.....	158
Figure 6.111	: Outdoor temperature and indoor temperatures of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	159
Figure 6.112	: Operation of the indoor units of System2 zones for the individual and master control modes, (a) individual control mode, (b) master control mode.....	160

Figure 6.113	: Comparison of the power consumptions of OU1 and OU2 for the individual and master control modes, (a) individual control mode, (b) master control mode.....	161
Figure 6.114	: Variation of the heating capacity, outdoor unit power consumption, total power consumption of the indoor units and HPF of OU2 for the individual and master control modes, (a) individual control mode (top), (b) master control mode (bottom)...	162
Figure 6.115	: Indoor unit air discharge temperatures of System2 zones on February 19, 2007.....	163
Figure 6.116	: Number of data points for 23.3°C set temperature.....	164
Figure 6.117	: Outdoor conditions for 23.3°C set temperature tests.....	164
Figure 6.118	: Sub regions of the ASHRAE winter thermal comfort zones.....	165
Figure 6.119	: Indoor conditions of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	166
Figure 6.120	: Indoor conditions of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode.....	168
Figure 6.121	: TSS variations of System1 zones provided by the individual and master control modes at 23.3°C set temperature, (a) individual control mode, (b) master control mode.....	169
Figure 6.122	: TSS variations of System2 zones provided by the individual and master control modes at 23.3°C set temperature, (a) individual control mode, (b) master control mode.....	170
Figure 6.123	: TSS variations of System1 and System2 combined zones provided by the individual and master control modes.....	171
Figure 6.124	: Comparison of the daily averaged energy consumption of the individual and master control modes at 23.3°C set temperature...	171
Figure 6.125	: Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 23.3°C set temperature, (a) OU1, (b) OU2.....	172
Figure 6.126	: Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 23.3°C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF.....	175
Figure 6.127	: Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes.....	175
Figure 6.128	: Variation of the outdoor unit power consumption with respect to the inverter frequency in the cooling and heating modes.....	176
Figure 6.129	: Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operations at 23.3°C set temperature, (a) continuous fan operation, (b) synchronized fan operation.....	177
Figure 6.130	: Number of data points for 26.1 °C set temperature tests.....	178
Figure 6.131	: Outdoor conditions for 26.1 °C set temperature tests.....	179
Figure 6.132	: Indoor conditions of System2 zones provided by the individual and master control modes at 26.1 °C set temperature, (a) individual control mode, (b) master control mode.....	180
Figure 6.133	: TSS variations of System2 zones provided by the individual and master control modes at 26.1 °C set temperature, (a) individual control mode, (b) master control mode.....	181
Figure 6.134	: TSS variations of System2 combined zones provided by the individual and master control modes.....	182

Figure 6.135	: Comparison of the daily averaged energy consumption of the individual and master control modes at 26.1 °C set temperature...	183
Figure 6.136	: Percentage ON time ratio of OU2 for the individual and master control modes at 26.1 °C set temperature.....	183
Figure 6.137	: Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF.....	185
Figure 6.138	: Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operation at 26.1 °C set temperature, (a) continuous fan operation, (b) synchronized fan operation.....	186
Figure 6.139	: Variation of the inverter frequency, refrigerant pressures and outdoor unit power consumption.....	187
Figure 6.140	: Variation of the indoor unit refrigerant inlet and outlet.....	188
Figure 6.141	: Variation of the indoor unit air blowing temperature.....	188
Figure 6.142	: Variation of the indoor unit air inlet temperature.....	189
Figure 6.143	: Variation of the indoor and outdoor temperature.....	189
Figure 6.144	: Variation of the total heating capacity and the COP of the system.....	190
Figure 6.145	: Variation of the power consumption, total heating capacity and the COP of the system with respect to the outdoor temperature...	191
Figure 6.146	: Comparison of the HPF and COP of the system.....	191
Figure 6.147	: Number of data points for ventilation assisted and non-ventilated tests in the heating mode.....	192
Figure 6.148	: Outdoor conditions for ventilation assisted and non-ventilated tests in the heating mode.....	193
Figure 6.149	: Indoor conditions of System1 zones provided by ventilation assisted and non-ventilated systems at 23.3 °C set temperature, (a) ventilation assisted, (b) non-ventilated.....	194
Figure 6.150	: Indoor conditions of System2 zones provided by ventilation assisted and non-ventilated systems at 23.3 °C set temperature, (a) ventilation assisted, (b) non-ventilated.....	196
Figure 6.151	: TSS variations of System1 zones provided by the ventilation assisted and non-ventilated systems at 23.3 °C set temperature, (a) ventilation assisted, (b) non-ventilated system.....	197
Figure 6.152	: TSS variations of System2 zones provided by the ventilation assisted and non-ventilated systems at 23.3 °C set temperature, (a) ventilation assisted, (b) non-ventilated system.....	198
Figure 6.153	: Comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated systems at 23.3 °C set temperature.....	199
Figure 6.154	: Percentage ON time ratio of OU1 and OU2 for the ventilation assisted and non-ventilated systems at 23.3 °C set temperature, (a) OU1, (b) OU2.....	200
Figure 6.155	: Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF.....	202

Figure 6.156	: Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operations at 23.3°C set temperature, (a) continuous fan operation, (b) synchronized fan operation.....	203
Figure 7.1	: Office suite.....	206
Figure 7.2	: Flowchart of the developed multi-split VRF model for the EnergyPlus.....	209
Figure 7.3	: Flowchart of the main program for energy-use computation of the multi-split VRF system.....	210
Figure 7.4	: Schematic diagram of a stand alone ERV.....	213
Figure 7.5	: Comparison of the outdoor conditions, (a) temperature, (b) relative humidity.....	216
Figure 7.6	: Comparison of indoor temperature between simulation and experiment, (a) System1 zones, (b) System2 zones.....	217
Figure 7.7	: Comparison of indoor relative humidity between simulation and experiment, (a) System1 zones, (b) System2 zones.....	218
Figure 7.8	: Comparison of total power consumption of HRV units and indoor units between simulation and experiment.....	219
Figure 7.9	: Variations of the experiment and simulation outdoor unit power consumptions.....	220
Figure 7.10	: Comparison of OU1 and OU2 power consumptions between simulation and experiment.....	220
Figure 7.11	: Comparison of daily CPF and PLR between simulation and experiment.....	221
Figure 7.12	: Maryland outdoor conditions during the period of June-August 2007.....	222
Figure 7.13	: Weekly energy consumptions of System1 and System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2.....	223
Figure 7.14	: Percentages of the outdoor, indoor and HRV units in the total energy consumption of System1 and System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2.....	224
Figure 7.15	: Indoor conditions of System1 and System2 zones at 25°C indoor set temperature for MD, (a) System1 zones, (b) System2 zones.....	225
Figure 7.16	: Weekly energy consumptions of the non-ventilated System1 and non-ventilated System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2.....	226
Figure 7.17	: Percentages of the outdoor and indoor units in the total energy consumption of the non-ventilated System1 and non-ventilated System2, (a) System1, (b) System2.....	227
Figure 7.18	: Indoor conditions of the non-ventilated System1 and System2 zones at 25°C indoor set temperature for MD, (a) System1 zones, (b) System2 zones.....	228
Figure 7.19	: Outdoor conditions of Phoenix, AZ and Los Angeles, CA during the period of June-August 2007, (a) Phoenix, AZ, (b) Los Angeles, CA.....	230
Figure 7.20	: Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	232
Figure 7.21	: Energy savings of outdoor and indoor units of System1 for MD, (a) outdoor unit, (b) indoor units.....	234
Figure 7.22	: Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	236

Figure 7.23	: Energy savings of outdoor and indoor units of System2 for MD, (a) outdoor unit, (b) indoor unit.....	237
Figure 7.24	: Energy savings of System1 and System2 for MD, (a) System1, (b) System2.....	238
Figure 7.25	: Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	239
Figure 7.26	: Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	240
Figure 7.27	: Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C	242
Figure 7.28	: Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	243
Figure 7.29	: Energy savings of outdoor units of System1 for AZ and CA, (a) AZ, (b) CA.....	245
Figure 7.30	: Energy savings of indoor units of System1 for AZ and CA, (a) AZ, (b) CA.....	246
Figure 7.31	: Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	247
Figure 7.32	: Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	248
Figure 7.33	: Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C	250
Figure 7.34	: Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	251
Figure 7.35	: Energy savings of outdoor units of System2 for AZ and CA, (a) AZ, (b) CA.....	252
Figure 7.36	: Energy savings of indoor units of System2 for AZ and CA, (a) AZ, (b) CA.....	253
Figure 7.37	: Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	254
Figure 7.38	: Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	255
Figure 7.39	: Seasonal energy consumption of System1 and System2 for MD, AZ and CA.....	256
Figure 7.40	: Seasonal energy savings of System1 and System2 for MD, AZ and CA.....	257
Figure 7.41	: CPF of System1 and System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA.....	258
Figure 7.42	: PLR of System1 and System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA.....	259
Figure 7.43	: Weekly energy consumptions of non-ventilated System1 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	261
Figure 7.44	: Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	263

Figure 7.45	: Energy savings of OU1 and OU2 for non-ventilated System1 and non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD.....	264
Figure 7.46	: Energy savings of non-ventilated System1 and non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD	265
Figure 7.47	: Indoor conditions of non-ventilated System1 zones with cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	266
Figure 7.48	: Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C.....	267
Figure 7.49	: Weekly energy consumptions of non-ventilated System1 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	269
Figure 7.50	: Weekly energy consumptions of non-ventilated System1 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	270
Figure 7.51	: Energy savings of non-ventilated System1 outdoor unit for AZ and CA, (a) AZ, (b) CA.....	271
Figure 7.52	: Energy savings of non-ventilated System1 indoor units for AZ and CA, (a) AZ, (b) CA.....	272
Figure 7.53	: Indoor conditions of non-ventilated System1 zones with cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	273
Figure 7.54	: Indoor conditions of System1 zones with cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	274
Figure 7.55	: Energy consumption of ventilation assisted and non-ventilated OU1 and OU2 for CA, (a) OU1, (b) OU2.....	275
Figure 7.56	: Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	277
Figure 7.57	: Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	278
Figure 7.58	: Energy savings of non-ventilated System2 outdoor unit for AZ and CA, (a) AZ, (b) CA.....	279
Figure 7.59	: Energy savings of non-ventilated System2 indoor units for AZ and CA, (a) AZ, (b) CA.....	280
Figure 7.60	: Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C.....	281
Figure 7.61	: Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C.....	282
Figure 7.62	: CPF of non-ventilated System1 and non-ventilated System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA.....	284
Figure 7.63	: PLR of non-ventilated System1 and non-ventilated System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA.....	285
Figure B.1	: Wind tunnel.....	299
Figure B.2	: Duct taping for the air leakage.....	299
Figure B.3	: Inside of the wind tunnel.....	300

Figure B.4	: Air flow rate and blowing, return, indoor and outdoor air temperatures.....	300
Figure B.5	: Refrigerant inlet and outlet temperatures, condensing and evaporating pressures and the EEV opening.....	301
Figure B.6	: Air and refrigerant side of the capacities and the error.....	301
Figure C.1	: Indoor unit tube circuits, (a) left view, (b) right view.....	302
Figure C.2	: CoilDesigner.....	303
Figure C.3	: Variation of the pressure drop of the indoor unit with respect to the inlet refrigerant mass flow rate.....	303
Figure C.4	: Variation of the pressure drop of the indoor unit with respect to the inlet pressure.....	304
Figure C.5	: Outdoor unit tube circuits.....	305
Figure D.1	: Indoor conditions of System1 zones provided by the multi-split VRF system in master control mode and existing VAV system in central control mode, (a) multi-split VRF system in master control mode, (b) existing VAV system in central control mode.....	305
Figure D.2	: Indoor conditions of System2 zones provided by the multi-split VRF system in master control mode and existing VAV system in central control mode, (a) multi-split VRF system in master control mode, (b) existing VAV system in central control mode.....	306
Figure E.1	: Relationship between the indoor unit operation and the EEV opening on June 01, 2007, (a) IURB, (b) IURC, (c) IURD.....	307
Figure F.1	: Relationship between the indoor unit operation and the EEV opening on June 01, 2007, (a) IUEle, (b) IUAI, (c) IURE.....	308
Figure G.1	: Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 25°C set temperature, (a) OU1, (b) OU2, (c) total.....	309
Figure H.1	: Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 20°C set temperature, (a) OU1, (b) OU2, (c) total.....	310
Figure I.1	: Variation of the daily energy consumption for the ventilation assisted and non-ventilated multi-split VRF system with respect to the daily averaged outdoor temperature, (a) OU1, (b) OU2, (c) total.....	311
Figure J.1	: Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 23.3°C set temperature, (a) OU1, (b) OU2, (c) total.....	312
Figure K.1	: Variation of the daily energy consumption for the ventilation assisted and non-ventilated multi-split VRF system with respect to the daily averaged outdoor temperature, (a) OU1, (b) OU2, (c) total.....	313
Figure L.1	: Fan power consumption of IURA for low and high fan speed configurations.....	314
Figure L.2	: Total power consumption of the eight indoor units and four HRV units throughout a test day.....	315

LIST OF SYMBOLS

A	: The area of EEV
A_w	: Cross section area of the wind tunnel
a, b, c, d, e, f	: Coefficients for the cooling capacity modifier curve
a', b', c', d', e', f'	: Coefficients for the energy input ratio modifier curve
a'', b'', c''	: Coefficients for the part-load fraction correlation
c_D	: The flow coefficient
ΔP	: The pressure drop across EEV
EIR	: Reverse of COP curve-fitted from the catalogue data
$EIRTempModFac$	: Energy input ratio modifier with respect to temperatures
f	: Inverter frequency
F	: Function
H	: Relative humidity
$h_{c,i,o}$	: The outlet enthalpy of the i^{th} indoor unit in the cooling mode
$h_{c,i,i}$	: The inlet enthalpy of the i^{th} indoor unit in the cooling mode
$h_{Exhaust, Inlet}$	: Enthalpy of the air being exhaust from the zone through the unit
$h_{h,i,i}$	: The inlet enthalpy of the i^{th} indoor unit in the heating mode
$h_{h,i,o}$	: The outlet enthalpy of the i^{th} indoor unit in the heating mode
HR_{min}	: Enthalpies evaluated at a constant humidity ratio, the minimum humidity ratio of the supply air outlet or the exhaust air inlet
$h_{Supply, Outlet}$	: Enthalpy of the air being supplied to the zone
MF	: Modifier at part-load ratio
$\dot{m}_{Supply}$	: Mass flow rate of the supply air stream
$\dot{m}_T$	: Refrigerant mass flow rate
$\dot{m}_i$	: The individual refrigerant mass flow rate of the i^{th} indoor unit
n	: Number of variables
n_{IU}	: Total number of indoor units
P	: Vapor pressure
PLF	: Part load fraction
PLR	: Part load ratio
$\dot{Q}_{C,i}$	: The cooling capacity of the i^{th} indoor unit
$\dot{Q}_{Cool}$	: The total cooling capacity
$\dot{Q}_{C,OU,Cat}$	: The catalog cooling capacity of the outdoor unit

$\dot{Q}_{C,T}$	: The total cooling capacity of the multi-split VRF system
$\dot{Q}_{Latent}$	: Latent energy transfer rate to the zone
$\dot{Q}_{Heat}$	: The total heating capacity
$\dot{Q}_{H,i}$	: The heating capacity of the i^{th} indoor unit
$\dot{Q}_{H,T}$	: The total heating capacity of the multi-split VRF system
$\dot{Q}_{Sensible}$	: Sensible energy transfer rate to the zone
$\dot{Q}_{Total,v}$	: Total energy transfer rate to the zone
$\dot{Q}_{total}$	: Operating cooling capacity of the single DX coil
R	: Correlation coefficient
R_i	: The coefficient for the i^{th} indoor unit
RTF	: Runtime fraction of the cooling coil
ρ	: The refrigerant density at the inlet of EEV
STN_{sta}	: Status of the fixed speed compressor
t	: Time
T	: Dry bulb temperature
T_a	: Indoor air temperature
T_c	: Condensing temperature
$T_{c,i}$	: Ambient dry bulb temperature
T_e	: Evaporating temperature
T_{ia}	: Indoor air temperature
$TotCapTempModFac$	: Total cooling capacity modifier with respect to temperatures
TS	: ASHRAE thermal sensation vote
T_{suc}	: Suction temperature
$T_{wb,i}$	: Wet bulb temperature of the air entering the cooling coil
u_F	: Uncertainty of the function
u_n	: Uncertainty of the sensor
v_n	: Uncertainty of the measurement
$\dot{V}$	: Air flow rate
V_{ave}	: The averaged air velocity in the wind tunnel
$\dot{W}$	: The total power consumption
$\dot{W}_{comp}$	: The total power consumption of the compressors
$\dot{W}_{comp,inv}$	: The power consumption of IDC
$\dot{W}_{comp,fix}$	: The power consumption of FSC
$\dot{W}_{contr}$	: The power consumption of the controller in the outdoor unit
$\dot{W}_{IU}$	: The power consumption of the indoor unit

$\dot{W}_{ou}$	: The power consumption of the outdoor unit
$\dot{W}_{oufan}$	: The power consumption of the outdoor unit fan
Y	: Thermal sensation scale

EXPERIMENTAL AND SIMULATION EVALUATION OF A MULTI-SPLIT TYPE AIR CONDITIONING SYSTEM UNDER STEADY-STATE AND TRANSIENT CONDITIONS

SUMMARY

Air conditioning and ventilation for residential and commercial buildings are the necessities of live due to the large demand for thermal comfort and healthy environment.

Multi-split air conditioning system, featuring variable refrigerant flow (VRF) technology, is finding its way in residential and commercial buildings due to the precise capacity control and individualized thermal comfort capability.

Multi-split VRF system can control the capacity of the system by varying the individual refrigerant mass flow rates of each indoor unit with the help of the variable speed compressor and individual electronic expansion valves (EEVs) resulting in a comfortable indoor environment.

This research focuses on the performance evaluations of a multi-split VRF system integrated with heat recovery ventilation (HRV) units under varying outdoor conditions for both cooling and heating seasons in an actual office suite.

Two different control modes, individual and master, are applied to the multi-split VRF system. The individual control mode is the common indoor set temperature control strategy of the multi-split VRF system; on the other hand, the master control mode is the common indoor set temperature control strategy of the ducted type direct exchange systems which are widely used in the United States (US).

Besides, the effect of the ventilation on the performance of the multi-split VRF system is evaluated.

Extensive experimental data analysis was conducted to characterize the effects of the control modes and the ventilation on the multi-split VRF system's performance which covers the indoor thermal comfort, energy consumption and the efficiency of the system.

The energy saving options for the multi-split VRF system integrated with HRV units were investigated through simulation under varying outdoor conditions of different representative states in the US.

ÇOKLU-SPLIT TİP BİR İKLİMLENDİRME SİSTEMİNİN SÜREKLİ VE GEÇİCİ REJİM ŞARTLARI ALTINDA DENEYSEL VE SAYISAL OLARAK İNCELENMESİ

ÖZET

Hem evsel, hem de ticari binalarda sağlıklı ve konforlu yaşam alanı isteğinden dolayı, iklimlendirme ve havalandırma sistemleri yaşamın gereğidir.

Değişken soğutkan akış (DSA) teknolojisi özelliği gösteren, çoklu-split iklimlendirme sistemleri, hassas kapasite kontrol ve bireysel ısı konfor kabiliyetlerinden dolayı, hem evsel hem de ticari binalarda kendine yer bulmaktadır.

Çoklu-split DSA sistemi, değişken hızlı kompresör ve bireysel elektronik kısılma vanaları yardımıyla, her iç üniteden geçen soğutkan kütleli debisini değiştirerek konforlu bir iç ortam sağlamak üzere sistem kapasitesini ayarlayabilmektedir.

Bu araştırma, değişken hava koşulları altında, hem soğutma hem de ısıtma dönemleri için, gerçek bir ofis ortamında, ısı geri kazanım havalandırma üniteleri ile birleştirilmiş bir çoklu-split DSA sisteminin performans değerlendirmelerine odaklanmaktadır.

Çoklu-split DSA sistemine iki farklı iç ortam sıcaklık kontrol şekli; bireysel ve tek nokta, uygulanmıştır. Bireysel kontrol, çoklu-split DSA sistemlerinde genel olarak kullanılan iç ortam sıcaklık kontrol çeşididir. Diğer taraftan, tek nokta kontrolü, Amerika Birleşik Devletleri'nde (ABD) sıklıkla tercih edilen kanal tip iklimlendirme sistemlerinde kullanılan iç ortam sıcaklık kontrol çeşididir.

Ek olarak, havalandırmanın, çoklu-split DSA sisteminin performansına etkisi de incelenmiştir.

Kontrol şeklinin ve havalandırmanın, çoklu-split DSA sisteminin performansına (iç ortam konforu, enerji tüketimi ve sistem verimi) etkisinin belirlenmesi için geniş deneysel bilgi analizi yapılmıştır.

Isı geri kazanım havalandırma üniteleri ile birleştirilmiş olan çoklu-split DSA sistemi için enerji tasarruf seçenekleri, değişken hava koşullarında (farklı dış ortam sıcaklık ve bağıl nemine sahip ABD eyaletleri için) simulasyon ile incelenmiştir.

1. INTRODUCTION

Air conditioning for residential and commercial buildings is the necessities of life due to the large demand for thermal comfort and healthy environment of the living space in modern society.

The conception of the air conditioning has gradually developed from one unit for one house to independent units for separate zones in the same house.

Multi-split air conditioning system, featuring variable refrigerant flow (VRF) technology, so-called multi-split VRF system can satisfy the same needs for the installation of several individual units with less space, because this system consists of one outdoor unit and multiple indoor units.

A typical multi-split VRF system with four indoor units is provided in Figure 1.1.

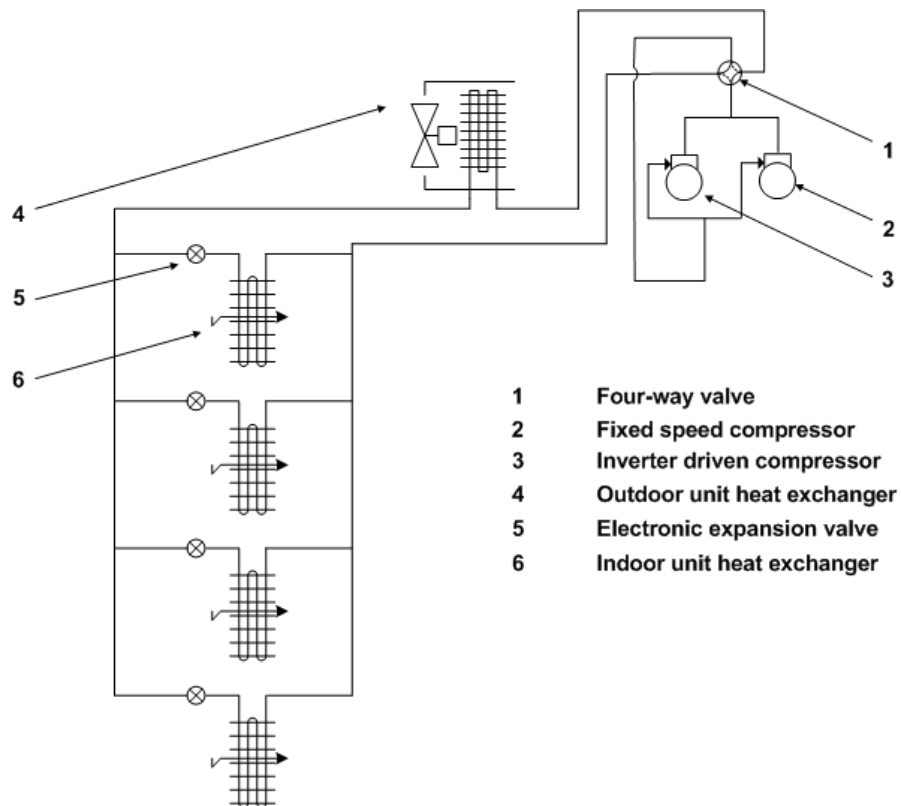


Figure 1.1 : Schematic diagram of a multi-split VRF system with four indoor units

Mainly, each indoor unit consists of a heat exchanger, an electronic expansion valve (EEV), and a fan. On the other hand, outdoor unit consists of two or three

compressors, one of which is variable speed, a four-way valve, a heat exchanger and a fan. The variable speed compressor and individual EEVs control the capacity of the system by varying the refrigerant mass flow rate passing through each indoor unit according to the cooling and heating loads of the corresponding zones. By adjusting the four-way valve, the refrigerant path can be reversed, so that the multi-split VRF system can be used for both air conditioning and heat pumping according to the season.

Multi-split VRF systems were first introduced in Japan around 25 years ago, and after that they have become popular in many countries. However, they are relatively unknown in the United States (US). The ducted direct exchange systems are the common air conditioning systems in the US. The long history of these systems in the US and the construction differences between the ducted type air conditioning systems and the multi-split VRF systems are the main reasons of the limited market of multi-split VRF systems in the US (Goetzler, 2007).

Multi-split VRF systems have several advantages compared to the conventional ducted type air conditioning systems.

Depending on the manufacturer, for time being, up to 20 indoor units with different capacities and configurations can be connected to one outdoor unit in multi-split VRF systems. Since indoor units are equipped with individual EEVs, many zones are possible with individual set temperatures. That's why; the multi-split VRF systems are generally installed to buildings which require multiple zoning such as office buildings, hospitals, hotels and schools.

Outdoor units of multi-split VRF systems are equipped with inverter driven compressors which enable wide capacity modulation with high part-load efficiency.

In addition, multi-split three-pipe VRF systems can provide cooling and heating simultaneously by using only one outdoor unit. Thus, multi-split three-pipe VRF systems can be operated in five different modes:

- a)Cooling-only mode: All indoor units are in cooling operation.
- b)Heating-only mode: All indoor units are in heating operation.
- c)Cooling-principal mode: Cooling is the principal mode in the concurrent heating and cooling operation.
- d)Heating-principal mode: Heating is the principal mode in the concurrent heating and cooling operation.

e)Heat recovery mode: Heat is balanced between indoor units and the outdoor unit heat exchanger is closed.

In addition to the system advantageous, they have installation advantageous due to the light weight and modularity. Since they are much lighter than the ducted type air conditioning systems, they can be transported modular basis (indoor and outdoor units separately) by standard elevators (Goetzler, 2007).

On the other hand, multi-split VRF systems have disadvantageous compared to conventional ducted type air conditioning systems.

The initial cost of the multi-split VRF system is one of the main disadvantageous of these systems (Goetzler, 2007). Besides, multi-split VRF systems do not have any ventilation capability, that's why additional ventilation systems are necessary, which also increases the initial cost.

The second main disadvantageous of the multi-split VRF system is that, there is not any Air Conditioning and Refrigeration Institute (ARI) certified rating system for measuring the efficiency of these systems (Goetzler, 2007).

2. LITERATURE REVIEW

Literature survey focuses on the experimental and modeling studies related to the multi-split VRF systems and the thermal comfort studies related to the residential and commercial buildings.

2.1 Multi-Split VRF System Studies

Wu et al. (2005) proposed a control strategy for a multi-split VRF system. The schematic drawing of the system is provided in Figure 2.1. The system was composed of three identical evaporators with three EEVs, a condenser, and a variable speed rotary compressor. The suction pressure and the room air temperature were taken as the control parameters to modulate the compressor speed and the EEV opening, respectively. A self tuning fuzzy control algorithm with a modifying factor was also input in the controller.

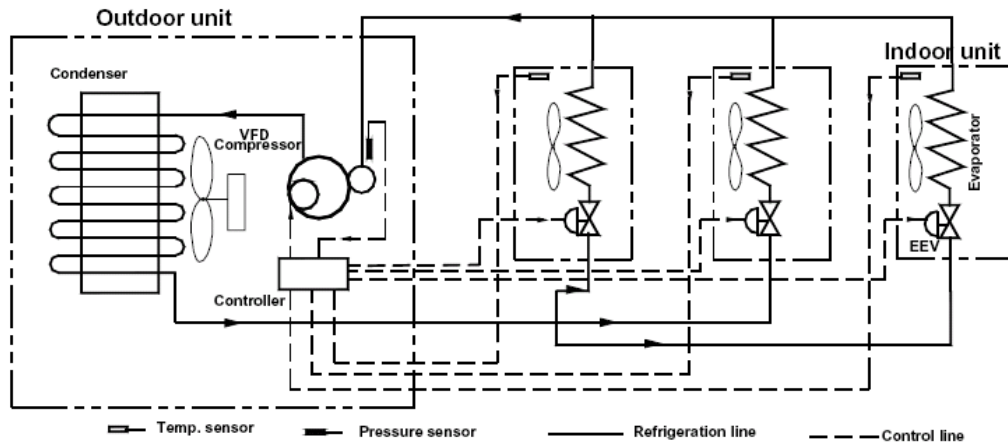


Figure 2.1 : Schematic diagram of the multi-split air VRF conditioning system

A simplified lump parametric model was developed for the controlling of the system. The parametric tests showed that the proposed control strategy with fuzzy control algorithm could achieve the desired control accuracy of the controlled parameters.

Xia et al. (2002) applied a testing methodology to a multi-split three-pipe VRF system. The schematic drawing of the test bench is shown in Figure 2.2. The test bench was composed of six calorimeters. A multi-split three-pipe VRF system with five indoor units was tested. The outdoor unit had two hermetic scroll compressors (one variable speed, one fixed speed), two air-cooled heat exchangers, two variable

speed outdoor unit fans, two EEVs, an accumulator and two four-way valves. The frequency of the variable speed compressor was modulated by the frequency controller with a range of 20-115Hz. On the other hand, each indoor unit had an air-cooled heat exchanger, a three-speed fan and an EEV.

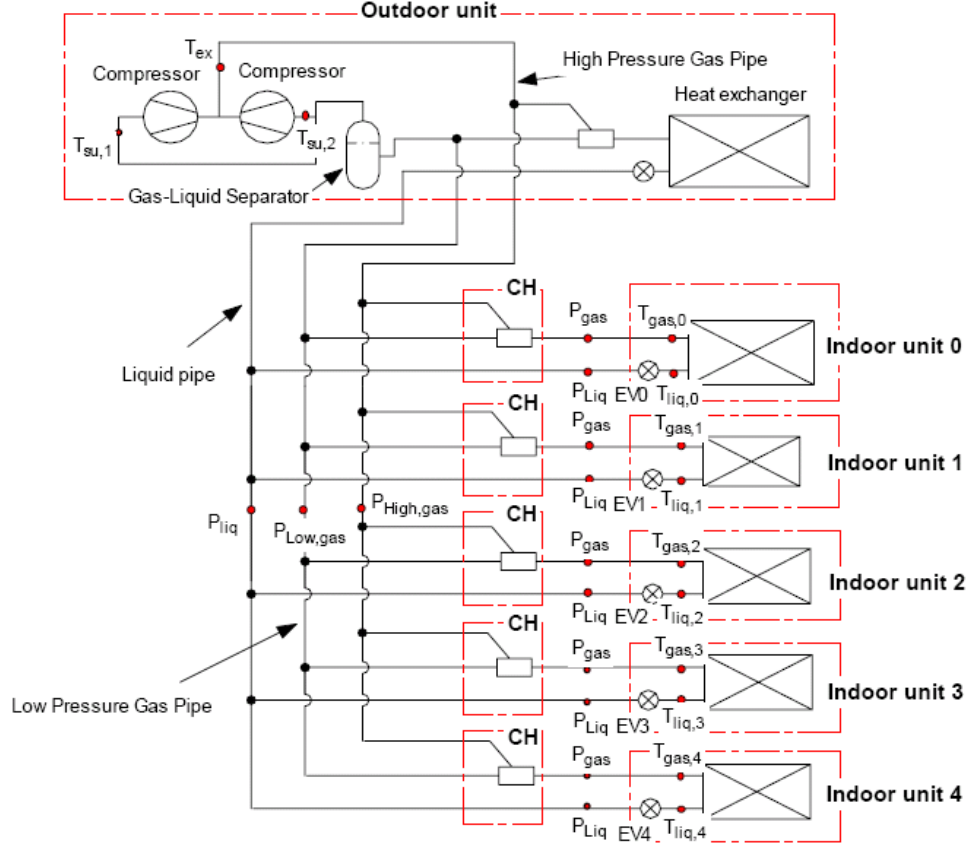


Figure 2.2 : Schematic diagram of the multi-split three-pipe VRF system

Outdoor unit and the indoor units were placed in each calorimeter. The coefficient of performance (COP) of the system was defined as the ratio of the total thermal load to the total electric consumption of the system. All the tests were performed in “cooling all” mode and without latent load. It was found that COP of the system did not vary too much according to the part load ratio (PLR). This was explained by the use of two compressors in “tandem”, which yielded good part load performance. On the other hand, it was also found that COP of the system decreased slightly with the increasing of the outdoor air temperature. COP of the system was obtained within 1.9~2.4 during the sensible cooling regime. Besides, a constant evaporating pressure was found during all the tests, due to the control strategy of the system.

Xia et al. (2004) studied the energy performance of a multi-split three-pipe VRF system numerically. The continuous adaptation of the heat transfer coefficient

method for indoor unit was applied, instead of “ON/OFF” operation of each indoor unit to maintain the same superheating in “ON” periods. In that control strategy, each EEV was adjusted to distribute the suitable refrigerant mass flow rate to each indoor unit in order to maintain the constant indoor room temperature. It was found that the predicted electrical consumption of the system under “cooling-all” mode was within -8% to 6.7% agreement with the experimental data. Besides, the isentropic efficiency of the compressor was found to be within 0.4 to 0.5.

Shi et al. (2003) developed a fluid network model to simulate the performance of a multi-split three-pipe VRF system with two indoor units. It was found that the energy efficiency ratio (EER) of the system in heat recovery mode was about two times higher than EER in “cooling-only” or “heating-only” mode, due to the usage of both cooling and heating capacities.

Masuda et al. (1991) developed a control method for a multi-split VRF system with two indoor units. A rotary type compressor modulated by a frequency controller with a range of 30-120Hz was used in the experiments which were performed in calorimeters under steady-state conditions. The capacity of the indoor units and the refrigerant flow rate was measured in each calorimeter in order to find an optimum control. Each EEV was operated manually, and the rate of the valve opening was varied under the same frequency for the compressor. A relationship between the refrigerant flow rate of indoor units and the frequency of the compressor was found and this relationship was input to the microprocessor to control the compressor and the EEV openings. The new control method showed that, the refrigerant flow rate for the indoor unit installed to a room with higher cooling load was much more than the other indoor unit. It was also obtained that the compressor frequency decreased if each room temperature reached to the setting temperature, and increased in the opposite case. It was concluded that the new control method could control the refrigerant flow rate of indoor units individually and respond to the cooling loads.

Hai et al. (2006) studied a multi-split three-pipe VRF system with a nominal capacity of 30kW. The system was charged with R22 and consisted of five indoor units with different capacities. The outdoor unit was equipped with one inverter driven scroll compressor and one fixed speed compressor. The schematic drawing of the system is provided in Figure 2.3.

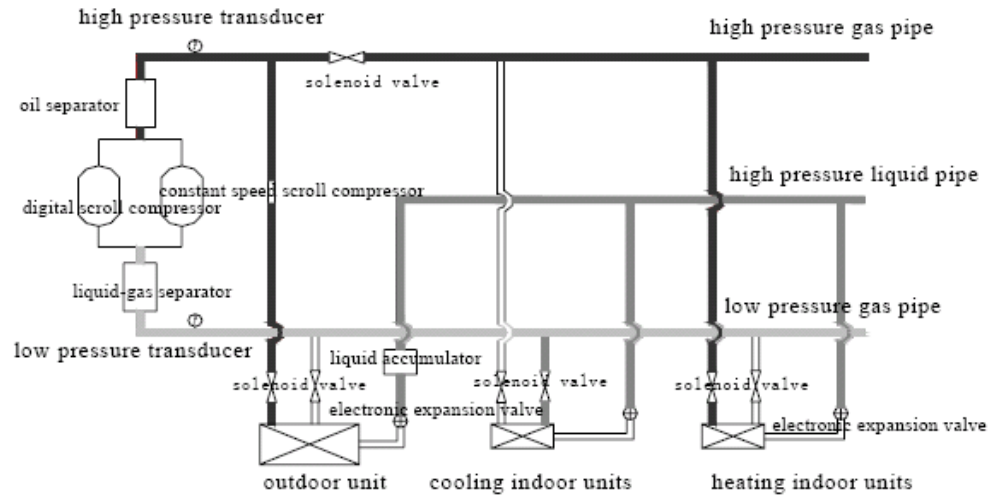


Figure 2.3 : Schematic diagram of the multi-split three-pipe VRF system

Experiments were performed in three climate chambers. The outdoor unit was placed into one chamber, and five indoor units were placed into the other two chambers. Experiments were done under steady state conditions with the following indoor unit combinations;

- Indoor units with the capacities of 12kW and 7kW were in cooling mode, while indoor units with the capacities of 7kW, 5kW and 2.5kW were in heating mode.
- Indoor units with the capacities of 12kW and 7kW were in cooling mode, while the indoor unit with the capacity of 2.5kW was in heating mode.
- Indoor units with the capacities of 12kW and 7kW were in heating mode, while indoor units with the capacities of 7kW, 5kW and 2.5kW were in cooling mode.
- Indoor units with the capacities of 12kW and 7kW were in heating mode, while the indoor unit with the capacity of 2.5kW was in cooling mode.

Similar to the findings of Shi et al. (2003), it was found that COP of the system increased in the “cooling-principal” and the “heating-principal mode”, because both the condensing heat and evaporating heat were used.

Park et al. (2001) studied a multi-split VRF system. A linear electronic valve was used for the expansion device. The system performance was analyzed according to the compressor frequency, the total cooling load and the cooling load fraction between rooms (defined as the ratio of the cooling load of the first room to the total cooling load). The schematic drawing of the test system is provided in Figure 2.4.

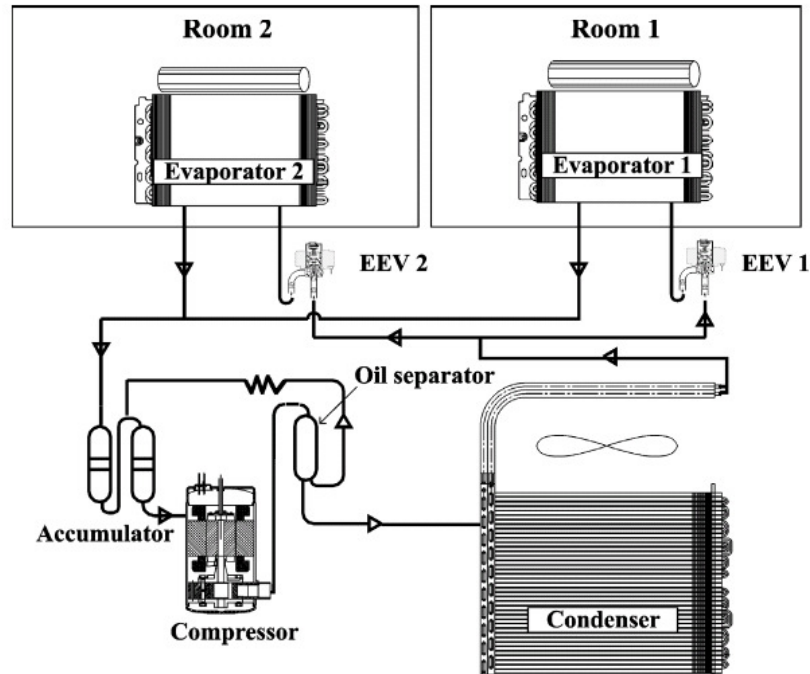


Figure 2.4 : Schematic diagram of a multi-split VRF system with two indoor units

The compressor performance map obtained from the manufacturer was used for the modeling of the compressor which was a rolling piston type rotary compressor with 36.55cm^3 piston displacement. The refrigerant mass flow rate and the compressor power were defined as functions of condensing and evaporating temperatures, and EEV correlation was used for the individual refrigerant mass flow rates of each indoor unit. It was found that COP varied parabolically with the EEV opening, and the compressor power increased with the second-order of the compressor frequency with a reduction in the COP. By fixing the total cooling load of the system at 6kW, it was obtained that the power consumption increased with an increase of the load difference between each room with a reduction in the COP. The reason of the increasing of the power consumption was due to the increasing of the compressor operating frequency. The operating frequency increased with load ratio when the total cooling load was constant. It was observed that when the load ratio was changed from 50% to 100%, the compressor frequency changed only 30%, but the EEV opening changed about 92%. It was concluded that the major control parameter was the EEV opening in a multi-split VRF system rather than the compressor operating frequency when the load ratio was changed.

Zhou et al. (2007a) investigated a VRF system in EnergyPlus dynamic building energy simulation program. They developed a module for the multi-split VRF system and imported that into the EnergyPlus simulation program. The module which was

validated in steady state experimental results by Zhou et al. (2007b) and Zhou et al. (2007c), could give the power and energy consumptions of the indoor and outdoor units, as well as COP and PLR. It was noticed that COP of the multi-split VRF system increased when the system worked in part load conditions due to the high PLR efficiency.

Hu and Yang (2005) developed a cost effective, energy efficient, multi-split VRF system with five indoor units. A variable refrigerant volume scroll compressor was used instead of an inverter aided one. The capacity control of the screw compressor was performed by an “ON/OFF” switching of the solenoid valves which changed the position of a static scroll to provide variable refrigerant flow. The schematic drawing of the compressor control and the test system are provided in Figure 2.5 and Figure 2.6, respectively.

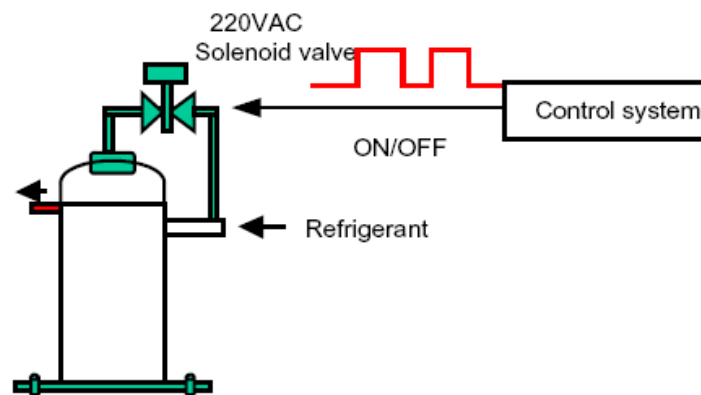


Figure 2.5 : Capacity control of the compressor

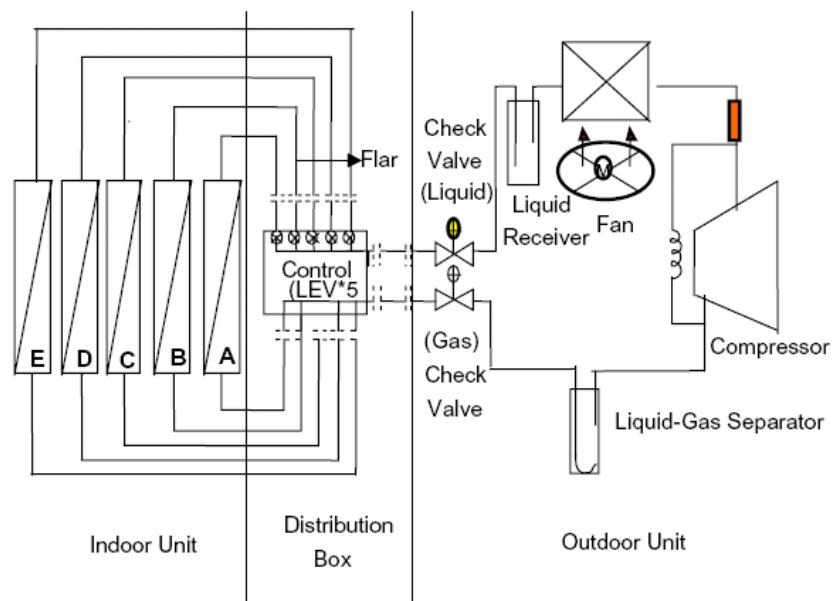


Figure 2.6 : System configuration

The system determined the required load of indoor units from the difference between the room and set temperatures, and regulated the degrees of each EEV opening to control the refrigerant flow and evaporation temperature of each indoor unit. Meantime, outdoor unit determined the running cycle and output time of the refrigerant in the compressor according to the requirement of the indoor units to control the "ON/OFF" cycle time of the solenoid valves, which controlled the refrigerant volume of the compressor. Experiments were performed in two environmental chambers; one for the indoor units, and another for the outdoor unit. The temperature was kept at 35°C dry bulb (DB)/24°C wet bulb (WB) for the outdoor unit chamber and 27°C DB/19°C WB for the indoor units' chamber. The temperature of superheat and sub-cool degree were maintained at 5°C. The EEV correlation was used for the individual refrigerant mass flow rates of each indoor unit. It was found that the developed system could adjust the capacity within 17-100% with a power input of 1.3-4.8kW, on the other hand, the inverter system adjusted the capacity within 48-104% with a power input of 2.5-6.1kW.

Watanabe et al. (2004) compared the annual energy consumptions of two packaged air conditioners in environmental chambers under steady state conditions. One of the systems was based on a fixed speed compressor with a cooling capacity of 12.5kW, and the other was based on a variable speed compressor with a cooling capacity of 5-11.2kW. Cyclic operation tests were performed, and it was found that the inverter system consumed much electricity at each start-up resulting in high efficiency degradation at cyclic operations.

Cho and Kim (2003) studied the capacity modulation characteristics of a multi-split VRF system with a simultaneous control of compressor speed and the EEV opening. The test unit, shown in Figure 2.7, had two indoor units with individual EEVs and one outdoor unit including a vertical type variable speed scroll compressor.

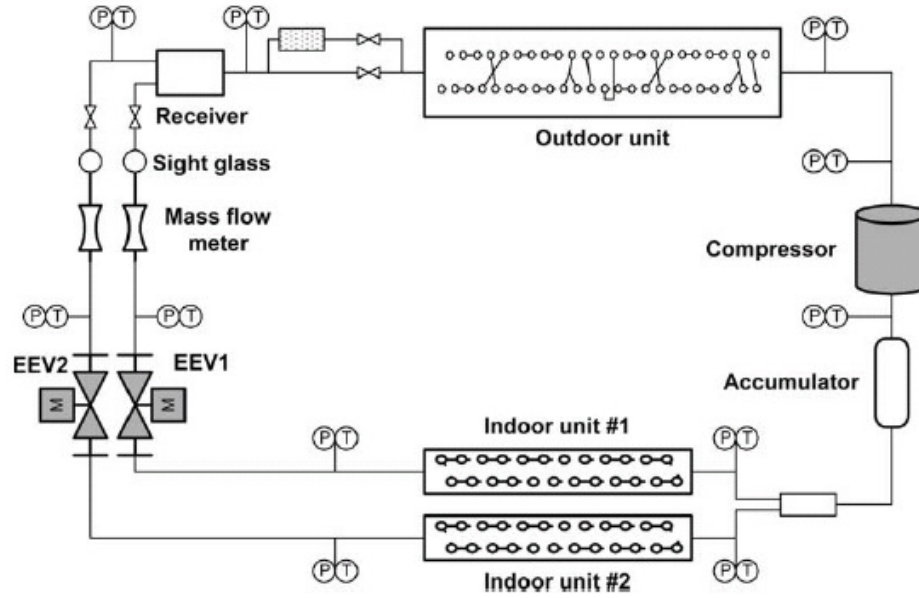


Figure 2.7 : Test system for the multi-split VRF system

The performance of the system was measured by varying load conditions of indoor unit number 1, the EEV opening, and compressor speed at a fixed load condition of indoor unit 2. Experiments were done under steady-state conditions. For at a fixed EEV opening and compressor speed, it was found that the total cooling capacity increased with the increasing of the air temperature entering indoor unit 1, due to a larger temperature difference between the refrigerant and air temperature of indoor unit 1, while power consumption increased due to an increase in compressor suction temperature. COP was enhanced with an addition of air temperature entering indoor unit 1, because the increase of cooling capacity was higher than that of power consumption. On the other hand, it was obtained that as the air temperature entering indoor unit 1 increased at a constant air temperature entering indoor unit 2, the evaporating temperatures of both indoor units increased because indoor unit 1 and indoor unit 2 merge at the same outlet. As the air temperature entering the indoor unit increased, the subcooling became higher because the increase of the evaporating temperature was slightly higher than that of the condensing temperature. However, the pressure difference between condenser and evaporator was reduced with an increase in the air temperature entering the indoor unit. The former tended to increase mass flow rate, while the latter led to a reduction of mass flow rate. This opposite trend resulted in a small variation of flow rate as a function of air temperature entering indoor unit 1. As the air temperature entering indoor unit 1 increased, the cooling capacity of indoor unit 1 was significantly enhanced, while that of indoor unit 2 was slightly reduced. The increment of cooling

capacity of indoor unit 1 was due to an increase in temperature difference between air and refrigerant. The major reason for capacity drop of indoor unit 2 was that increasing evaporating temperature lowered the temperature difference between refrigerant and air of indoor unit 2. The ratio of EEV1 opening to full opening was also varied from 0.30 to 0.46 at the fixed EEV2 opening, and indoor temperatures of indoor unit 1 and indoor unit 2 are maintained at 32°C and 27°C DB, respectively. The mass flow rate of indoor unit 1 linearly increased with an increasing of EEV1 opening, while that of indoor unit 2 slightly decreased.

Shah et al. (2004) developed a new methodology for the dynamic modeling for a multi-split VRF system. The expansion valve was modeled as an isenthalpic orifice, and compressor was defined mainly as a function of the compressor speed, volume of the compressor and pressure ratio. In order to find out the individual refrigerant mass flow rates, the pressure drops in the refrigerant suction lines from each evaporator to the compressor was taken into account. The predicted refrigerant mass flow rate was compared with the experimental data. It was obtained that the dynamic characteristics were well captured by the model. It was also found that the compressor could be used as a control actuator for a multi-split VRF system because of its direct influence on refrigerant flow rate, resulting in effect on charge distribution. The effect of the step input of the opening area of the expansion valve was also investigated. Expansion valve for the first evaporator changed while the second one was kept constant, and it was found that the pressure in the second evaporator also changed due to the coupled system dynamics. It was concluded that a control action from the valve should be included to the whole control algorithm, in order to maintain the pressure of the second evaporator at the desired value, namely its valve opening should be modulated, which will further affect the pressure in first evaporator.

Xia et al (2003) studied the operating characteristics of a multi-split VRF system with three indoor units numerically. The EEV correlation was used for the refrigerant mass flow rate for the indoor units and the refrigerant mass flow rate of the compressor was defined as a function of rotation speed of the compressor, discharge volume per compressor revolution, displacement coefficient and specific volume of the suction gas of the compressor. It was found that the greater the EEV opening, the greater the mass flow rate through the indoor unit. It was also found that the pressure drop across the indoor unit heat exchanger was negligible. It was obtained that for the same compressor speed when the EEV opening of one indoor unit increased while the other two kept unchanged, the cooling capacity of the first

indoor unit increased, while the cooling capacities of the rest decreased because of the distribution of the refrigerant mass flow rate. It was also concluded that by adjusting the compressor rotation speed while keeping the suction pressure unchanged, the cooling capacity of the individual evaporators could be changed without affecting others.

Shao et al. (2007) studied the prediction of the performance of complex refrigeration systems with a two-phase distributed-parameter fluid network model. The model results were validated with the experimental of Shi et al. (2007), and it was found that the error of the model was around 10%.

Lin and Yeh (2007) studied a three-evaporator air conditioner for a feedback controller design through the experimental investigation. For the proposed control structure, the three evaporating temperatures were controlled by the EEV openings in order to keep the indoor temperatures at the set points with no steady-state errors. Besides, the compressor speed was used to control the three superheat temperatures associated with the three evaporators. The proposed control method was experimentally validated.

2.2 Thermal Comfort Studies

Riberon and Kelly (2000) developed a method to investigate the comfort in the office buildings. The method was based on measurement and questionnaire. An apparatus, so-called ambiometre, was used for recording the air temperature at the head and ankle levels of the seated occupant, operative temperature, relative humidity, equivalent sound level, concentration of carbon dioxide and carbon monoxide. Besides, the ambiometre was equipped with a keyboard, so that the occupant could import his own overall comfort feeling. 50 offices in nine buildings were evaluated during the winter and summer conditions in France. According to the questionnaire results, the related comfort parameters was classified by a scale from 0 to 5 which corresponds to “not important” and “very important”, respectively. The results are shown in Figure 2.8.

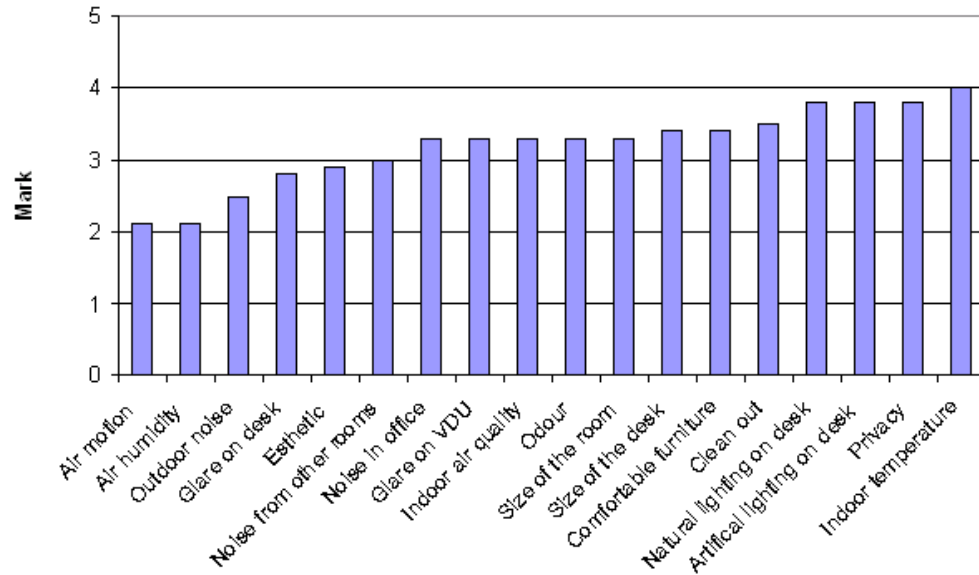


Figure 2.8 : Comfort parameters

As seen from Figure 2.8, temperature, privacy and lighting were considered as the most important parameters for comfort in office environment.

Aizlewood et al. (2000) investigated the effects of relative humidity in the range of 30-40% in winter season. The aim of the study was to evaluate whether changes in relative humidity in that range would affect acute symptoms and comfort in occupants of an office building. The experiments were carried out seven weeks in an air handling unit based air-conditioned office building in a city center. During the tests, indoor temperature and relative humidity were recorded, as well as occupants filled out questionnaire to report their health and comfort feelings. After three weeks, the humidity set point was reduced to achieve 5% decrease in relative humidity, and after three more weeks, the humidity set point was increased to the original level. According to the questionnaire, it was found that relative humidity had a little effect on the office thermal comfort.

Han et al. (2006) studied the resident thermal comfort of three cities in the hot-humid climate of central southern China. 110 subjects from 26 different residences were evaluated with a questionnaire which covered the traditional thermal sensation scale (TSS) which is American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) seven-point scale of ranging from cold (-3) to hot (+3) and with a neutral (0), thermal preference ("want warmer", "no change" and "want cooler"), current clothing garment and metabolic activity. For the indoor thermal environment, the indoor temperature, relative humidity and the air velocity at 1.1m

height were recorded. It was found that the occupant thermal sensation differed in homes with and without air conditioning. The thermal neutral operative temperature was found higher than other studies related to relatively lower humid areas. The following equation was used for the mean radiant temperature

$$MRT = 0.99T_a - 0.01 \quad 2.1$$

where

MRT : Mean radiant temperature(°C)

T_a : Indoor air temperature (°C)

Kwok and Chun (2003) examined the application of the thermal comfort standard to Japanese schools which were naturally ventilated or air conditioned. A measurement station was used to record the ambient and globe temperatures, relative humidity and air velocity at a height of 1.1m. In addition to the measurements, a questionnaire which covered the current status of thermal comfort by using ASHRAE TSS, clothing and demographic information was provided. The three middle scales (-1, 0 and +1) from ASHRAE TSS was assumed to be the thermal satisfaction. It was found that the indoor climatic conditions in naturally ventilated classrooms did not fall within the ASHRAE 55-92 summer comfort boundaries, on the other hand, as expected, the air conditioned classrooms did fall within the comfort zone boundaries.

McCartney and Humphreys (2002) investigate the potential link between actual temperature measurement, subjective response to temperature and perceived productivity. Field experiments from 25 buildings around Europe were performed. ASHRAE TSS (1 to 7, cold to hot, 4 neutral) and 5-point preference scale (1 to 5, much warmer to much cooler, 3 no change) to analyze the subjective temperature response were used. As for the perceived productivity, a 5-point scale was used and the occupants were asked the following question in the questionnaire;

“At present, how is your productivity being affected by the surrounding environmental conditions?”

- Much higher than normal 1
- Slightly higher than normal 2
- Normal 3
- Slightly lower than normal 4

According to the results, it was found that perceived productivity did not vary with indoor air temperature. On the other hand, as thermal preference moved away from “no change (3)”, perceived productivity fell. By considering both findings, it was concluded that even though perceived productivity was not influenced by actual temperature itself, it was influenced by “perception” of temperature which means thermal comfort.

Rohles and Nevins (1971) evaluated the range of thermal conditions at which men and women feel comfortable. In their evaluations, totally 1600 subjects; 800 male and 800 female with groups of 5 male and 5 female, which were exposed to 160 test conditions under sedentary conditions were evaluated. The insulating value for the clothing of the subjects was 0.6 clo. Test conditions covered a wide range; from 15.56°C (60°F) to 36.67°C (98°F) in 1.1°C (2°F) increments at each of eight relative humidity; 15, 25, 35, 45, 55, 65, 75 and 85%. Tests were held in environmental chambers. The subjects were first exposed to the pre-test room environment with an indoor temperature from 23.89°C (75°F) to 25.56°C (78°F). After 30 minutes, they walked into the main chamber and were exposed to the test environment for 3 hours. The questionnaires were recorded at 1.0, 1.5, 2.0, 2.5, and 3.0 hours of the tests. According to the results, it was found that when the temperature and relative humidity variables were equal, temperature was almost 7 and 9 times important than the relative humidity for men and women, respectively.

A multiple regression analysis for each of the five exposure periods for men, women and men and women combined was also conducted. The regression analysis is given in Table 2.1.

Table 2.1 : Multiple regression equations

Gender and exposure time (hour)	Regression Equation	R
Male, 1.0	$Y = 0.135T + 0.01H - 6.724$	0.882
Female, 1.0	$Y = 0.165T + 0.11H - 9.313$	0.908
Combined, 1.0	$Y = 0.15T + 0.01H - 8.018$	0.892
Male, 1.5	$Y = 0.139T + 0.009H - 7.136$	0.891
Female, 1.5	$Y = 0.169T + 0.10H - 9.694$	0.917
Combined, 1.5	$Y = 0.154T + 0.009H - 8.414$	0.901
Male, 2.0	$Y = 0.138T + 0.010H - 7.167$	0.895
Female, 2.0	$Y = 0.169T + 0.008H - 9.712$	0.914
Combined, 2.0	$Y = 0.154T + 0.009H - 8.440$	0.901
Male, 2.5	$Y = 0.134T + 0.011H - 6.987$	0.881
Female, 2.5	$Y = 0.170T + 0.009H - 9.865$	0.919
Combined, 2.5	$Y = 0.152T + 0.01H - 8.426$	0.897
Male, 3.0	$Y = 0.134T + 0.01H - 7.044$	0.879
Female, 3.0	$Y = 0.167T + 0.01H - 9.698$	0.912
Combined, 3.0	$Y = 0.151T + 0.01H - 8.371$	0.893

where

T : Dry bulb temperature (°F)

H : Relative humidity (%)

Y : TSS (from 1 to 7 where 1 is cold, 2-cool, 3-slightly cool, 4-comfortable, 5-slightly warm, 6-warm, 7-hot)

R : Correlation coefficient

This study was also referred in “Fundamentals Handbook, ASHRAE 2001”. The modified regression equations in “Fundamentals Handbook, ASHRAE 2001” are provided in Table 2.2.

Table 2.2 : Equations for predicting thermal sensation of men, women, and men and women combined

Exposure Time (hour)	Subjects	Regression Equations
1.0	Men	$Y = 0.22T + 0.233p - 5.673$
	Women	$Y = 0.272T + 0.248p - 7.245$
	Both	$Y = 0.245T + 0.248p - 6.475$
2.0	Men	$Y = 0.221T + 0.270p - 6.024$
	Women	$Y = 0.283T + 0.210p - 7.694$
	Both	$Y = 0.252T + 0.240p - 6.859$
3.0	Men	$Y = 0.212T + 0.293p - 5.949$
	Women	$Y = 0.275T + 0.255p - 8.622$
	Both	$Y = 0.243T + 0.278p - 6.802$

where

T : Dry bulb temperature (°C)

p : Vapor pressure (kPa)

Y : TSS (from -3 to 3 where -3 is cold, -2-cool, -1-slightly cool, 0-neutral, +1-slightly warm, +2-warm, +3-hot)

Newsham et al (1997) conducted a field test in four different buildings in Canada from October 1994 to January 1995. Software to administer questionnaires on occupants computer screen at predetermined dates and times was developed and used. In addition to the questionnaires, the outdoor and indoor air temperature and relative humidity and also total solar radiation on a horizontal plane was recorded. By using the questionnaires and the recorded data, the ASHRAE thermal sensation votes was grouped according to the corresponding indoor air temperature bin. In Figure 2.9, the variation of the mean vote of each bin with the mean temperature of the bin is given. The number of the data points (n) in each bin and the standard deviations in the ASHRAE votes for each bin shown with the error bars can also be seen from Figure 2.9.

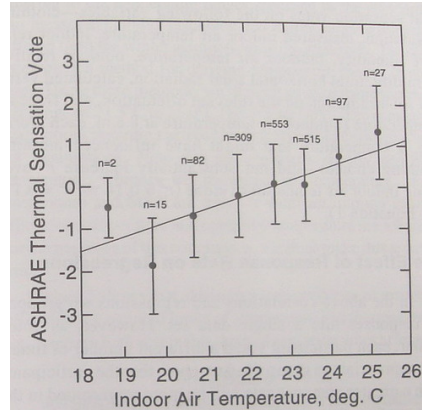


Figure 2.9 : Variation of the ASHRAE thermal sensation vote with the indoor air temperature

From the Figure 2.9, the following regression was found

$$TS = -7.56 + 0.33T_{ia} \quad (n = 1600, \quad r^2 = 0.77) \quad 2.2$$

where

TS : ASHRAE thermal sensation vote

T_{ia} : Indoor air temperature (°C)

During the evaluation, the air temperature and mean radiant temperature were assumed to be equal, which was assumed to be a valid assumption in most offices.

Wang (2006) evaluated 120 subjects in 66 residential units in a field test in order to investigate the thermal environment and thermal comfort in Harbin, northeast China in winter season. An indoor climate station was used to record the air temperature, relative humidity, radiant asymmetry and air velocity and a thermal comfort meter was used to measure the predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD). Air temperature and air velocity were measured at 0.1m, 0.6m, and 1.1m levels. Relative humidity and PMV-PPD indices were measured at 1.1m level and radiant asymmetry was measured at 0.6m level for seated occupants. Questionnaires related to the clothing, activity checklists, thermal sensation, and thermal preference for the occupants were also provided. According to the data and the results from the questionnaires, it was found that the neutral operative temperatures for the Harbin males and females were 20.9 and 21.9°C, respectively. By comparing the current results with the previous studies, it was also found that the operative temperature was very close to the mean air temperature in office buildings.

Lee and Strand (2001) evaluated the effect of exterior surface temperatures on the thermal comfort of a person in residence, office and gymnasium environment numerically. It was found that the placement of individuals was much less critical in the summer when the difference air and surface temperatures were expected to be smaller than in winter.

Murakami et al. (2007) proposed a new system to control the air-conditioning system through occupants' requests. This system collected occupants' requests from their own personal computers and controlled the air-conditioning system with a logic that balanced the thermal comfort of the occupants and the energy consumption. A series of cooling season tests were conducted in a typical open-plan office building with 45 people in Osaka, Japan. The cooling system was based on the fan coil and variable air volume (VAV) boxes. It was found that the proposed system could save around 25% energy compared with the baseline system which was based on the fixed 26°C set temperature.

2.3 Summary of the Literature Review

The multi-split VRF system studies are summarized as follows:

- Studies can be divided into two categories: experimental studies (Xia et al., 2002), (Masuda et al., 1991), (Hai et al., 2006), (Hu and Yang, 2005), (Cho and Kim, 2003), (Zhou et al., 2007b), (Zhou et al., 2007c), (Watanabe et al., 2004) and the numerical studies (Wu et al., 2005), (Xia et al., 2004), (Shi et al., 2003), (Park et al., 2001), (Zhou et al., 2007a), (Shah et al., 2004), (Xia et al., 2003), (Shao et al., 2007)
- It was found that all the experimental studies were performed in the environmental chambers under steady state conditions.
- The previous studies mostly focused on the cooling mode. Only two studies performed by multi-split three-pipe VRF systems were found for the heating mode (Hai et al., 2006), (Shao et al., 2007).
- Most of the studies were related to the control strategies of the multi-split VRF systems. Simultaneous controls of the compressor frequency and the EEV openings of the indoor units was the common control strategy (Wu et al., 2005), (Cho and Kim, 2003), (Park et al., 2001), (Shah et al., 2004), (Masuda et al., 1991), (Hu and Yang, 2005), (Lin and Yeh, 2007). Besides, Xia et al. (2002) monitored constant evaporating pressure due to system

control during the cooling test. Park et al. (2001) claimed that the major control parameter was the EEV opening.

The thermal comfort studies are summarized as follows:

- Studies can be divided into two groups: field tests (Riberon and Kelly, 2000), (Aizlewood et al., 2000), (Han et al., 2006), (Kwok and Chun, 2003), (McCartney and Humphreys, 2002), (Newsham et al., 1997), (Wang, 2006), (Murakami et al., 2007) and environmental chamber test (Rohles and Nevins, 1971)
- Indoor and outdoor environment was recorded and questionnaires were provided to the subjects (Riberon and Kelly, 2000), (Aizlewood et al., 2000), (Han et al., 2006), (Kwok and Chun, 2003), (Newsham et al., 1997).
- Questionnaires mostly covered ASHRAE TSS (Han et al., 2006), (Kwok and Chun, 2003), (McCartney and Humphreys, 2002), (Rohles and Nevins, 1971), (Newsham et al., 1997).
- Indoor environment measurements took place mostly at occupant height level; 0.1m, 0.6m and 1.1m above from the floor which represented the sedentary occupants' ankles, mid-body and neck (Riberon and Kelly, 2000), (Han et al., 2006), (Kwok and Chun, 2003), (Wang, 2006), (Murakami et al., 2007). However, for a regular air conditioning system, mostly stationary thermostats are used and placed on the walls which are relatively far away from the occupant locations, and the height of the thermostats from the floor is likely higher than the occupant level.
- Indoor temperature was found more important than the humidity for the thermal comfort of the occupants (Riberon and Kelly, 2000), (Aizlewood et al., 2000), (Rohles and Nevins, 1971).
- The mean radiant temperature was found very close to the indoor air temperature. Thus, the operative temperature can be assumed to equal to the indoor air temperature (Han et al, 2006), (Newsham et al., 1997), (Wang, 2006).
- In two studies, the predicted ASHRAE TSS was obtained by multiple regression analysis for cooling and heating seasons (Rohles and Nevins, 1971), (Newsham et al., 1997). For the cooling season, ASHRAE TSS was defined as a function of dry bulb temperature and the humidity, but for the

heating season, ASHRAE TSS was defined only as a function of the dry bulb temperature.

- The type of the air conditioning systems were not mentioned in the studies. Only two of them indicated that the air conditioning systems were based on VAV (Aizlewood et al., 2000), (Murakami et al., 2007).

3. MOTIVATION AND OBJECTIVES

3.1 Motivation

The research motivations of this study are summarized as follow:

- The first multi-split VRF system was installed more than 25 years ago in Japan, and that's why; they are well-known systems in Asian countries. However this technology is very new for the US. The multi-split VRF manufacturers have been in the US market for around four years.
- Even though the first system was installed more than 25 years ago in Japan; there is not any available actual operational study and data in the open literature. The only available studies in the literature are related to the steady state environmental chamber tests which do not cover the actual operation of a multi-split VRF system in the actual life (Xia et al., 2002), (Masuda et al., 1991), (Hai et al., 2006), (Hu and Yang, 2005), (Cho and Kim, 2003).
- There is not any ARI certified rating systems for the multi-split VRF systems (Goetzler, 2007). That's why; it is not easy to compare their performance results with the other air conditioning systems such as VAV based air conditioning systems. Because of this drawback, field tests are encouraged for the performance evaluations.
- The multi-split VRF systems are claimed to be powerful about their individualized thermal comfort due to their precise indoor temperature control. However, according to the literature survey, there is not any available thermal comfort study related to the multi-split VRF systems.
- As explained previously, the ducted type direct exchange systems are the common air conditioning systems in the US. The indoor temperature control strategy of these systems is mainly based on central control. 65.8 out of 101.5 million residential houses in the US are controlled by a stationary central thermostat located in the living room which controls the indoor temperature of the entire house (Burd and Burd, 2000), (Burd and Burd, 2002). Even though the multi-split VRF systems are claimed to be powerful about individualize indoor temperature control capability, they can be

controlled by central. Thus; the differences between the individual and central control modes can be evaluated.

- According to the literature survey, it was found that the main focus was on the cooling mode for the multi-split VRF systems. Only two studies related to the heating mode were found for a multi-split three-pipe VRF system, which were performed in environmental chamber tests (Hai et al, 2006), (Shao et al., 2007).
- One of the drawbacks of the multi-split VRF systems is that they can not provide ventilation to the conditioned spaces which is required by ASHRAE regulations. That's why; additional ventilation systems should be installed to the zones. Thus, the integration of the multi-split VRF system and the ventilation system gains importance in actual applications. Since there is not any available actual application study in the literature, there is not any study related to the effect of the ventilation system on the multi-split VRF system.

3.2 Research Objectives

The research objectives of this study are grouped into two parts; experimental and modeling objectives.

3.2.1 Experimental objectives

- Installation of a multi-split VRF system and a ventilation system in an office suite and instrumentation of systems including the indoor and outdoor environments of the office suite for the measurements and comparisons
- Controlling of the multi-split VRF system with the individual and master (central) control modes and assessment of the effect of the control strategies on the system
- Evaluation of the cooling and heating performances of the multi-split VRF system for both individual and master (central) control modes through the performance measurements
- Evaluation of the effect of the ventilation system on the multi-split VRF system for both cooling and heating modes
- Evaluation of the indoor thermal comfort provided by the individual and master (central) control modes for both cooling and heating seasons and the assessment of the effect of the ventilation system on the thermal comfort

3.2.2 Modeling objectives

- Evaluation of the performance of the multi-split VRF system under varying climate conditions with an experimentally validated building energy simulation program
- Assessment of the integration of the multi-split VRF system and the ventilation system under varying climate conditions
- Evaluation of the energy saving options for the multi-split VRF system integrated with the ventilation system

4. EXPERIMENTAL SETUP AND INSTRUMENTATION

4.1 Floor Layout

The performance of the multi-split VRF system was investigated in an actual office suite with a total floor area of around 167.3m², which was the top floor of the “Glenn Martin Hall” on the campus of the “University of Maryland, College Park”.

The floor layout is shown in Figure 4.1. As seen from Figure 4.1, the office suite has four zones on the south side; Rooms A, B, C and D and four zones on the north side; elevator, receptionist, aisle and Room E. There was not any physical separator between aisle and receptionist zones; therefore a dashed line was drawn in order to specify the fictitious boundaries of the aisle and receptionist zones. Stairs and elevator cabin were not taken into consideration in this study.

Table 4.1 shows the internal volume of each zone. The floor-ceiling height was 2.55m.

Table 4.1 : Volume of each zone

Zone Name	Volume (m ³)
Room A	55.1
Room B	73.4
Room C	62.4
Room D	29.4
Elevator	31.3
Receptionist	74.4
Aisle	80.7
Room E	51.4

Rooms A, D, E and receptionist zone mostly occupied from 9:00 to 17:30, and since Room B was used for meeting room, it was mostly unoccupied. In addition, Room C was also mostly unoccupied.

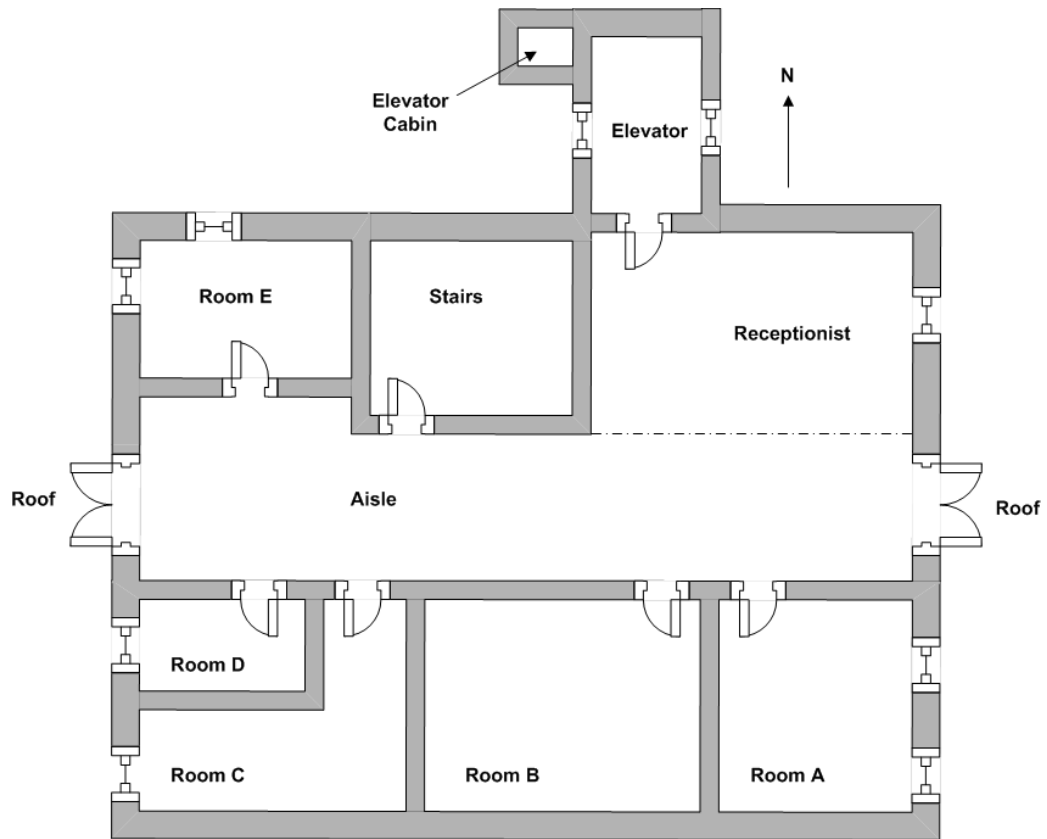


Figure 4.1 : Floor layout

4.2 Multi-Split VRF System

Two multi-split VRF systems charged with R410A were installed in the office suite and tested under varying operating conditions. The outdoor units were located on the roof and the indoor units were installed in each zone.

Figure 4.2 depicts the layout of the multi-split VRF systems on the floor. As seen from Figure 4.2, the outdoor unit 1 (OU1) was connected to the indoor units installed in the south zones; Rooms A, B, C and D. The outdoor unit 2 (OU2) was connected to the indoor units installed in the north zones; elevator, receptionist, aisle and Room E.

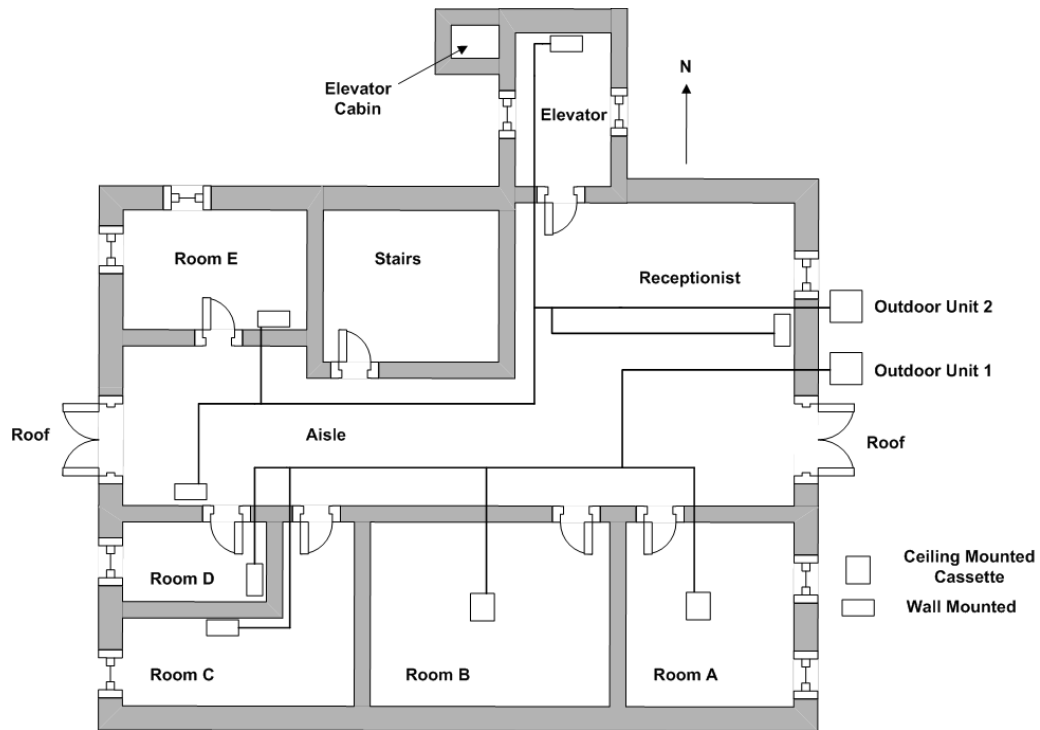


Figure 4.2 : Layout of the multi-split VRF systems

4.2.1 Outdoor units

Both multi-split VRF systems had two outdoor units with the same cooling and heating capacities. Figure 4.3 shows the outdoor units on the roof.



Figure 4.3 : Outdoor units

Basically, each outdoor unit consisted of two compressors, one heat exchanger, one fan and one four-way valve.

One of the compressors was inverter driven compressor (IDC), and the other was fixed speed compressor (FSC). They were connected parallel to the refrigerant

circuit. IDC provided the variable refrigerant mass flow rate to the system depending on the cooling or heating load of the indoors by changing of the inverter frequency within the range of 26-105Hz.

The outdoor unit heat exchanger was a fin-and-tube type heat exchanger. It was used as a condenser in the cooling season and as an evaporator in the heating season. In order to enhance the heat transfer from and to the ambient, a fan was located on top of the outdoor unit.

A four-way valve was located inside the outdoor unit in order to reverse the refrigerant path to change the operation mode from cooling to or from heating to cooling.

IDC (on the right bottom), the heat exchanger (at the back), and the four-way valve (in the middle) and the pipe connections can be seen from Figure 4.4. FSC was placed on the right side of IDC (not seen in the photo).



Figure 4.4 : Inside of the outdoor unit

Specifications, standard cooling and heating capacities and the power consumption of the outdoor unit based on the catalogue data are given in Table 4.2.

Table 4.2 : Catalogue information of the outdoor unit

Capacity	Cooling (kW)	28.13
	Heating (kW)	31.65
Input	Cooling (kW)	8.67
	Heating (kW)	9.19
Compressor	Type	Hermetically sealed scroll type
Capacity Control	%	14 ~ 100
Refrigerant	Type	R410A
	Charge (kg)	11.39

4.2.2 Indoor units

As seen from Figure 4.2, each outdoor unit was connected to four indoor units through the refrigerant pipes. Totally eight indoor units were installed in the office suite. Two out of eight indoor units were ceiling mounted types, and the rest were wall mounted types.

The ceiling and the wall mounted type indoor units are shown in Figure 4.5 and Figure 4.6, respectively.



Figure 4.5 : Ceiling mounted cassette type indoor unit



Figure 4.6 : Wall mounted type indoor unit

Each indoor unit was equipped with a fin-and-tube type heat exchanger, an EEV and a fan. EEV was used to control the refrigerant mass flow rate through the heat exchanger. The fan was used to force the air through the heat exchanger to cool it down in the cooling mode and heat it up in the heating mode.

For the ceiling mounted cassette type indoor units shown in Figure 4.5, the return air was taken into the indoor unit through the grille located in the middle of the unit and supplied to the zone from the four discharge grilles located at each side of the unit. For the wall mounted type indoor units shown in Figure 4.6, the return air was taken into the indoor unit through the grilles located at the front and top of the unit and supplied to the zone from the discharge grille located at the bottom of the unit.

Type, location and the name of the indoor units are summarized in Table 4.3 and the catalogue data of the corresponding indoor units are provided in App. [A].

Table 4.3 : Type, location and the name of the indoor units

Type of Indoor Unit	Connected to	Zone	Indoor Unit
Ceiling Mounted Cassette	OU1	Room A	IURA
Ceiling Mounted Cassette	OU1	Room B	IURB
Wall Mounted	OU1	Room C	IURC
Wall Mounted	OU1	Room D	IURD
Wall Mounted	OU2	Elevator	IUEle
Wall Mounted	OU2	Receptionist	IURec
Wall Mounted	OU2	Aisle	IUAis
Wall Mounted	OU2	Room E	IURE

4.2.3 Connection between the outdoor unit and the indoor units

Outdoor units were connected to the indoor units with the refrigerant pipes. Since there were four indoor units connected to one outdoor unit, the pipe branches, shown in Figure 4.7, were used for the piping. The main refrigerant stream was divided into two sub streams at this branch. The lower pipe shown in Figure 4.7 was connected directly to the corresponding indoor unit, while the upper one carried the rest of the refrigerant to the remaining indoor units.



Figure 4.7 : Branch for the refrigerant pipe

The schematic drawing of the multi-split VRF system is shown in Figure 4.8. The junctions can be seen in the upstream and downstream of each indoor unit.

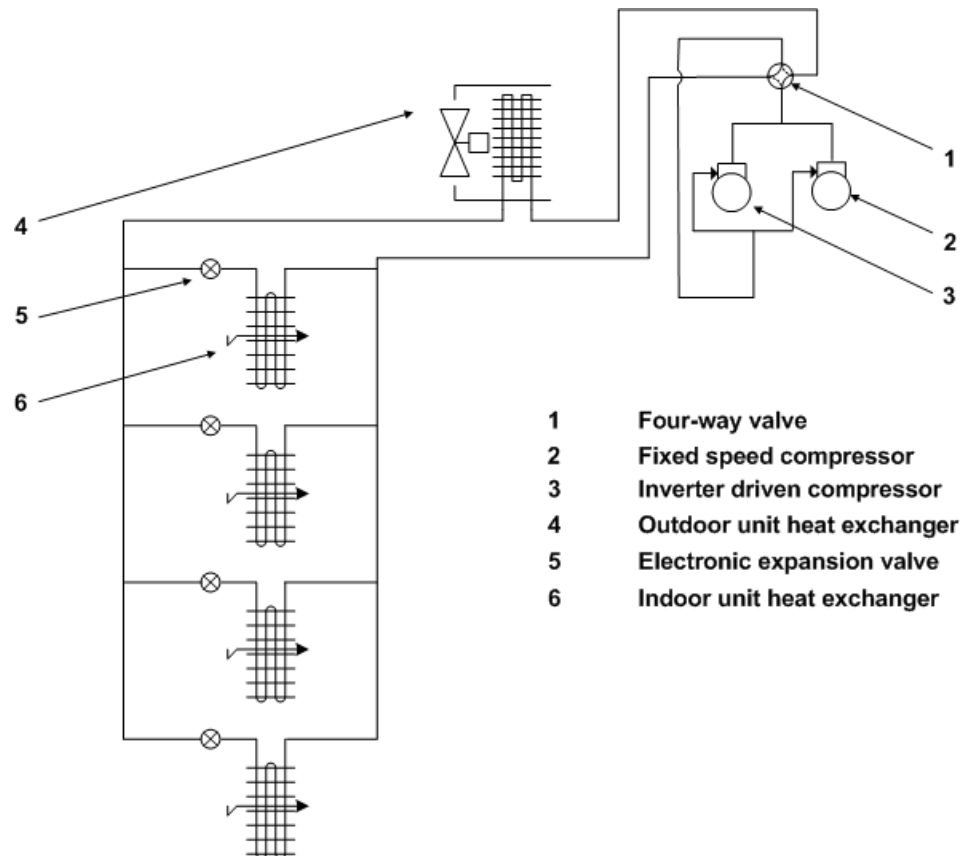


Figure 4.8 : Multi-split VRF system

In the cooling mode, the refrigerant discharged by the compressors goes into the outdoor unit heat exchanger, which is used as a condenser, through the four-way valve. Due to the heat rejection, the superheated refrigerant cools down and condenses in the heat exchanger. High pressure, low temperature refrigerant, is then throttled to the low pressure and enters into the indoor units through the EEVs which require cooling. Indoor units are used as evaporators in the cooling mode. Thus heat transfer occurs from the indoor air to the refrigerant inside the indoor unit heat exchanger, which cools down the indoor air. Then, the low pressure superheated refrigerant returns back to the compressors, and finishes the cycle.

In the heating mode, the four-way valve, shown in Figure 4.8, reverses the refrigerant path. The refrigerant discharged by the compressors goes into the indoor units which are used as condensers. Thus, the high pressure, high temperature refrigerant rejects the heat to the indoor air and heats it up. The high pressure, low temperature refrigerant leaves the indoor units and throttled to the low pressure through the expansion valve. Then, the low pressure, low temperature refrigerant enters into the outdoor unit heat exchanger which is used as an evaporator. The low pressure superheated refrigerant returns back to the compressors, and finishes the cycle.

4.2.4 Indoor air temperature control

Nine identical thermostats, provided by the manufacturer were used for the indoor air temperature control. One of the thermostats is shown in Figure 4.9.

Basically, the desired indoor air temperature and the indoor unit fan speed such as low or high speed could be set by these thermostats.

Thermostats had 0.5°C dead band, namely, if the set temperature was 25°C, then the indoor unit operated within the range of 24.5°C to 25.5°C indoor air temperature. When the indoor air temperature was above 25.5°C, the thermostat sent a signal to the indoor unit and opened EEV and allowed the refrigerant pass through the indoor unit heat exchanger to cool down the indoor air temperature. When the indoor air temperature became lower than 24.5°C, the thermostat sent the signal and closed EEV, thus the refrigerant could not enter to the indoor unit heat exchanger.



Figure 4.9 : Thermostat

One of the nine thermostats is shown under the wall mounted type indoor unit in Figure 4.10.



Figure 4.10 : Thermostat and indoor unit

Eight out of the nine thermostats were located into each zone. The last one was located almost in the center of the office floor.

Two different control modes; the individual and master control modes were applied to the multi-split VRF system for the indoor air temperature control.

4.2.4.2 Individual control mode

In this control mode; each individual indoor unit was controlled by the corresponding individual thermostat. Namely, IURA was controlled by the thermostat located in Room A, IURB was controlled by the thermostat located in Room B, and IURC was controlled by the thermostat located in Room C, and so on.

Totally eight thermostats, referred as individual thermostats, were used during the individual control mode to control the indoor air temperatures.

The locations of the individual thermostats in each zone can be seen from Figure 4.11.

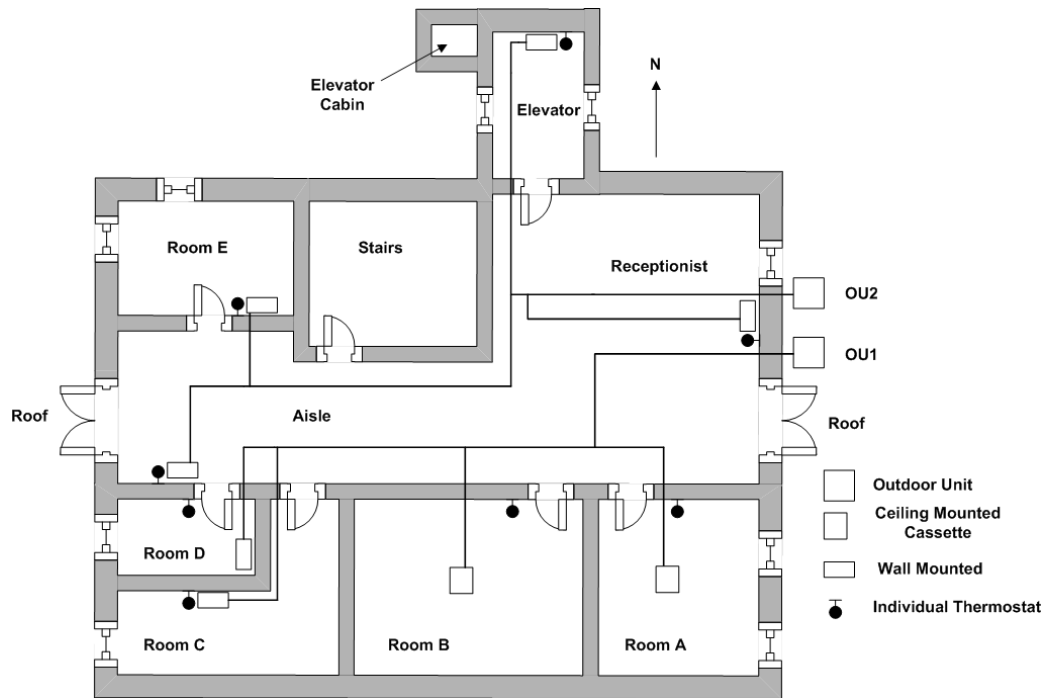


Figure 4.11 : Locations of the individual thermostats

4.2.4.3 Master control mode

The existing cooling system was based on a VAV system and the heating system was based on a convective water heating system. The existing system was controlled by central control, namely one thermostat was located in the center of the office suite to control the indoor temperature of the office suite. The detailed information about the existing system will be provided in Section 4.5.

In order to simulate the operational characteristics of the VAV central control system, one thermostat was installed close to the existing system's thermostat in the center of the office floor. Thus, all indoor units were controlled by only one thermostat, referred as the master thermostat, located in the center of the office floor.

The locations of the master thermostat in the office suite can be seen from Figure 4.12.

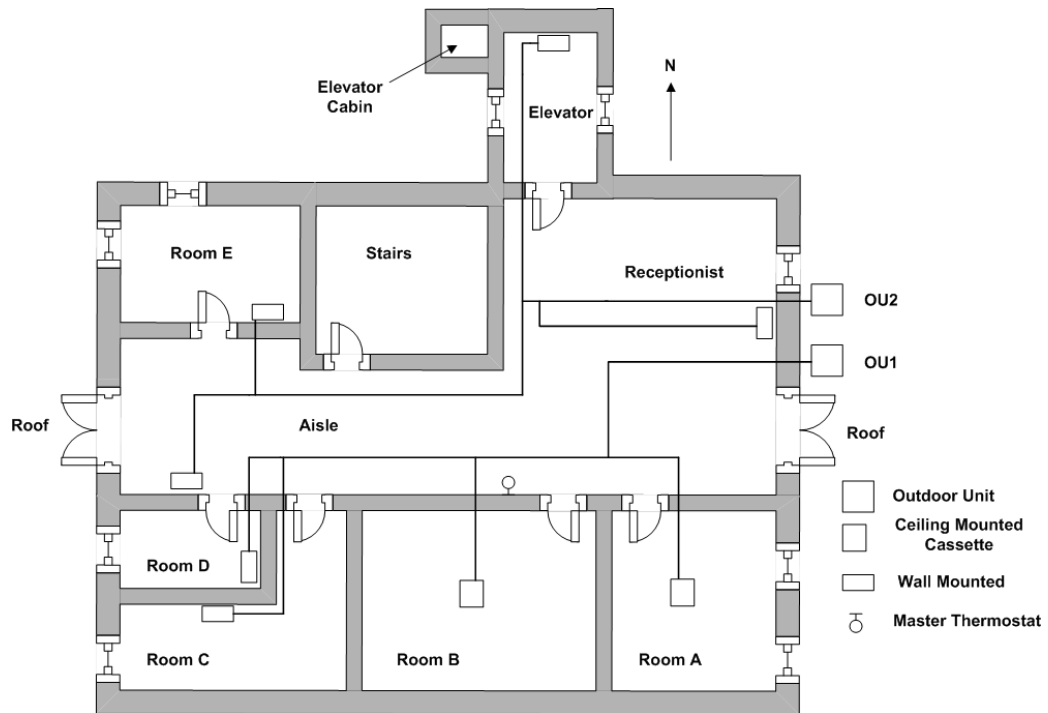


Figure 4.12 : Location of the master thermostat

4.2.4.4 Changing of the control modes

Each indoor unit had a relay box which consisted of the relay connections for the master and the individual thermostats. One of the relay boxes is shown in Figure 4.13.

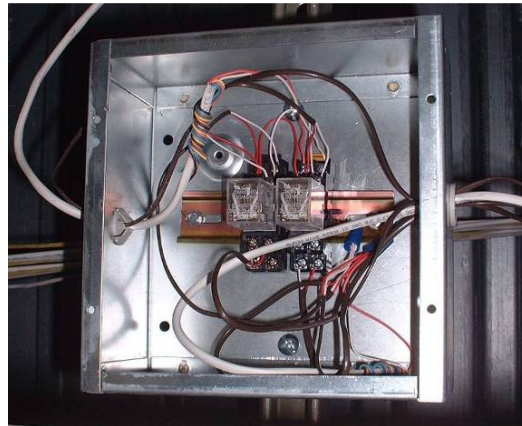


Figure 4.13 : Relay box

When the individual control mode was used for the temperature control, the circuit of the relay connection of the individual thermostat in each relay box was closed and the circuit of the relay connection of the master thermostat in each relay box was opened. Thus, the individual thermostats became active.

Likewise, when the master control mode was used for the temperature control, the circuit of the relay connection of the individual thermostat in each relay box was opened and the circuit of the relay connection of the master thermostat in each relay box was closed. Thus, the master thermostat became active.

The schematic drawings for the relay boxes for the individual and master control modes are shown in Figure 4.14 and Figure 4.15, respectively.

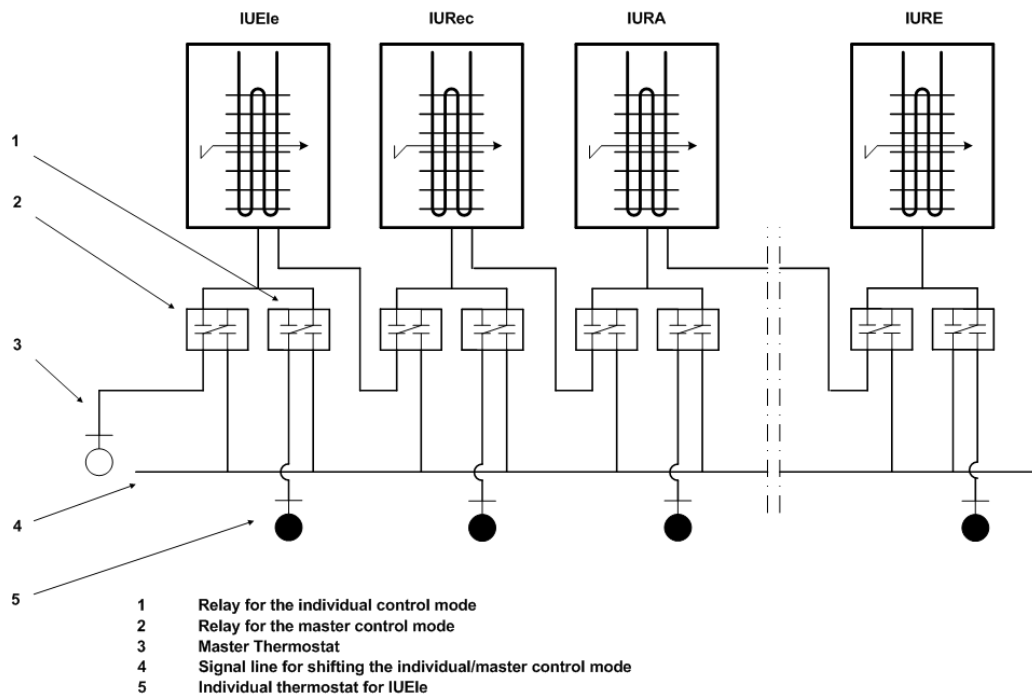


Figure 4.14 : Schematic drawing of the relay boxes for the individual control mode

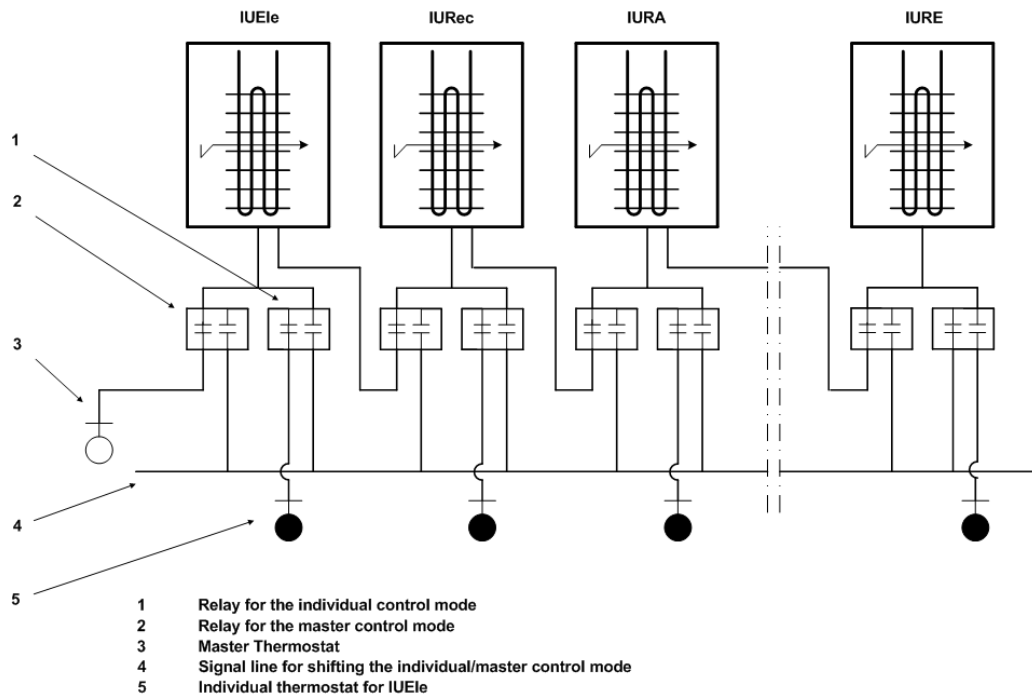


Figure 4.15 : Schematic drawing of the relay boxes for the master control mode

4.3 Ventilation Units

The office suites require ventilation due to the indoor air quality problems and the occupancy thermal comfort. However, the multi-split VRF systems can not provide any ventilation. That's why, additional ventilation systems are necessary to ensure the required ventilation.

In addition to the multi-split VRF systems, four heat recovery ventilation (HRV) units were installed in the ceiling.

HRV units were composed of a paper based heat exchanger and exhaust and supply fans. The schematic drawing of a HRV unit is shown in Figure 4.16.

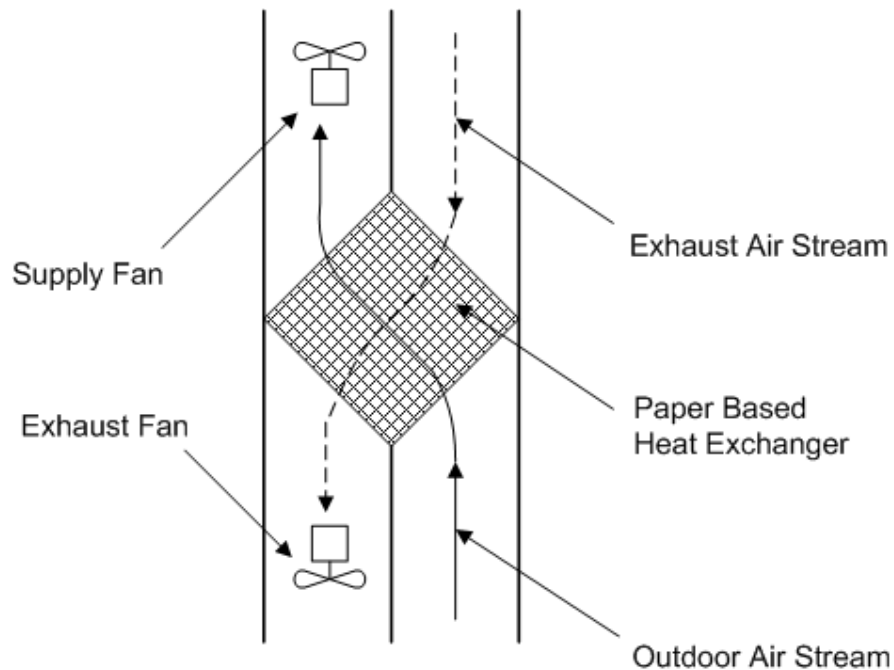


Figure 4.16 : Schematic drawing of an HRV unit

The return air was taken from the indoors by the exhaust fan and exhausted to the outdoor, in the meantime, the outdoor air was supplied to the indoors by the supply fan. During this process, the supply and exhaust air streams did not mix, however the paper based heat exchanger provided heat exchange between the supply air stream and the exhaust air stream.

One-time measurement was performed for the air flow rate of each HRV unit. The location of the four HRV units and the measured supply air flow rates are shown in Table 4.4, and the layout of the HRV units can be seen from Figure 4.17.

Table 4.4 : HRV units and the locations

HRV Unit No	Ventilation for	Air Flow Rate (m ³ /s)
HRV 1	Rooms A and B	0.124
HRV 2	Rooms C and D	0.057
HRV 3	Room E	0.065
HRV 4	Elevator, Receptionist and Aisle	0.143

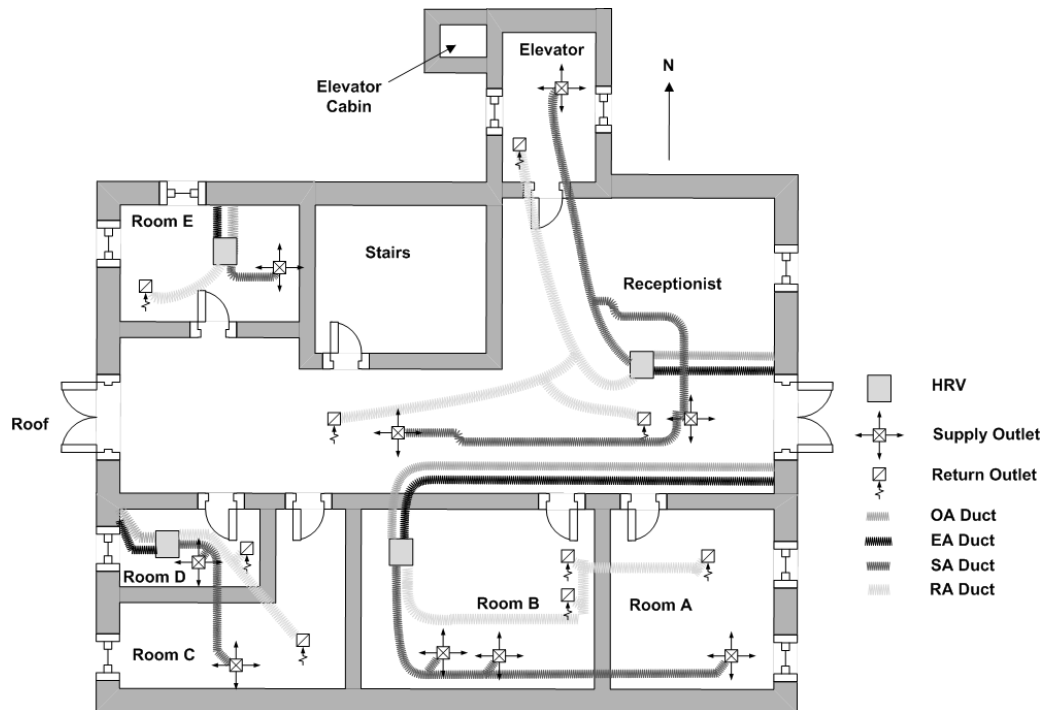


Figure 4.17 : Layout of the HRV units

4.4 Existing Cooling and Heating System

The existing VAV cooling system and the convective water heating system were preinstalled to the office suite.

An air handling unit (AHU) was preinstalled on the third floor of the same building and operated according to a separate prescheduled. Basically, the building return air and the outdoor air were mixed in the AHU and the mixed air was conditioned with a chilled water coil in the AHU and then supplied to the office suite by a supply duct and distributed to the floor through four no-reheat VAV boxes with 21 supply outlets. The design air flow rates of the VAV boxes were $0.39\text{m}^3/\text{s}$.

As for the heating season, the hot water was supplied to the office suite and passed through 11 coils placed in various locations of the floor.

The indoor temperature control of both cooling and heating seasons was performed by a central thermostat located in the center of the office suite.

The layout of the existing cooling and heating system can be seen from Figure 4.18.

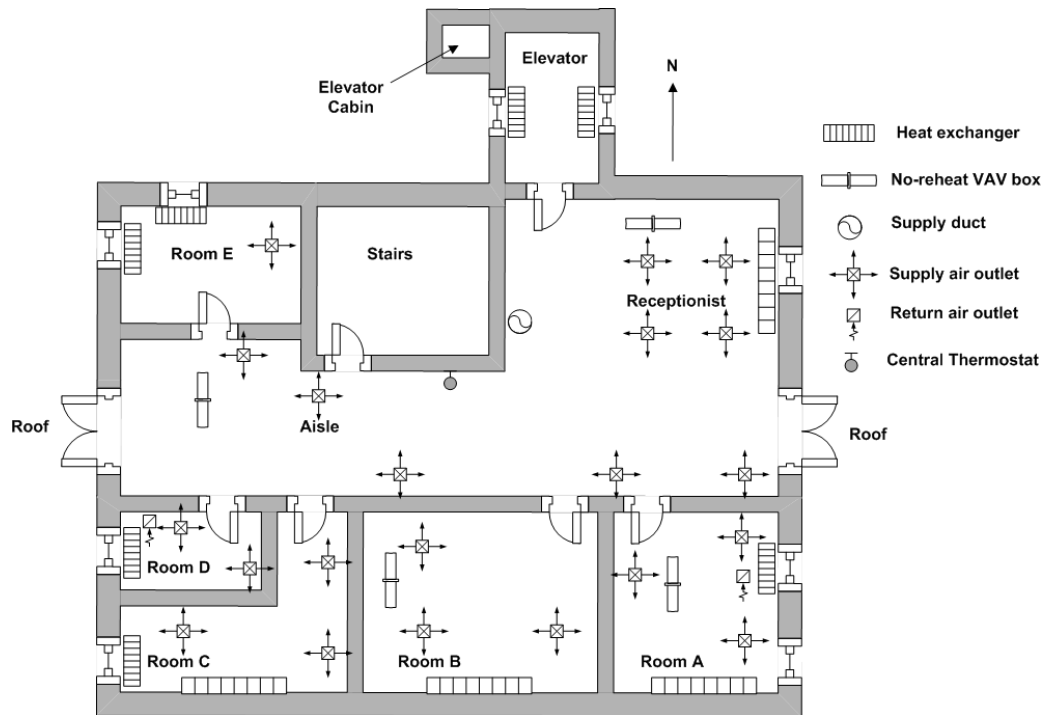


Figure 4.18 : Layout of the existing cooling and heating system

In order to eliminate the effect of the existing system on the multi-split VRF system tests and to evaluate the multi-split VRF system alone, the office suite should be isolated from the existing system during the tests. That's why; a damper was installed into the supply duct to prevent the conditioned air being disturbed to the office suite and two additional dampers were installed into the return ducts located in Rooms A and D.

On the other hand, a shut off valve was installed into the heating system to prevent the hot water being circulated in the office suite.

Figure 4.19 and Figure 4.20 depict the damper for the supply duct, and the shut off valve for the heating system, respectively.



Figure 4.19 : Air damper for supply duct of the existing cooling system

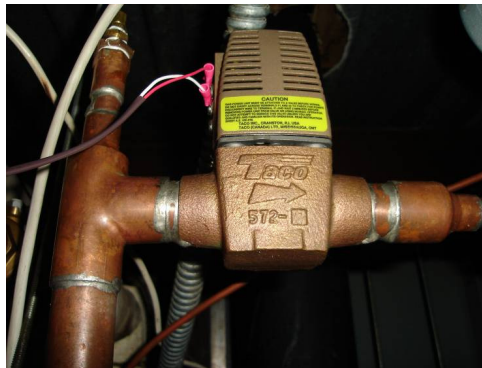


Figure 4.20 : Shut off valve for the existing heating system

4.5 Scheduler

Scheduling of the whole system was performed through the scheduler unit, shown in Figure 4.21, which had the capability of sending signals to the outdoor units, the indoor units, thermostats, the ventilation units, dampers and the shut-off valve according to the input schedule such as turning “ON” or “OFF” the whole system or specific units, operating of the individual or master control modes on specified days and hours. The scheduler was also used for inputting the configuration data to the system such as the set temperature of each thermostat and indoor unit fan speeds.



Figure 4.21 : Scheduler

4.6 Measurement Instrumentation

Indoor and outdoor environment of the office suite and the multi-split VRF systems and the HRV units were instrumented with the sensors and monitored during the tests.

Air and refrigerant temperatures were measured by T-type thermocouples and the relative humidity sensors were used for the air relative humidity measurements.

Pressure transducers were used for the refrigerant pressure measurements.

Power consumptions of the outdoor and indoor units of the multi-split VRF systems and the HRV units were measured by watt meters.

The measurement locations are summarized in the following sections.

4.6.1 Airside measurements

Nine T-type thermocouples with an accuracy of $\pm 0.5^{\circ}\text{C}$ and nine relative humidity sensors with an accuracy of $\pm 2\%$ were placed very close to each thermostat to measure the indoor conditions.

One of the indoor air temperature and relative humidity measurement locations can be seen in Figure 4.22 under the thermostat.



Figure 4.22 : Measurement location under the indoor unit

The layout of the indoor air temperature and relative humidity measurement locations in the office suite is provided in Figure 4.23.

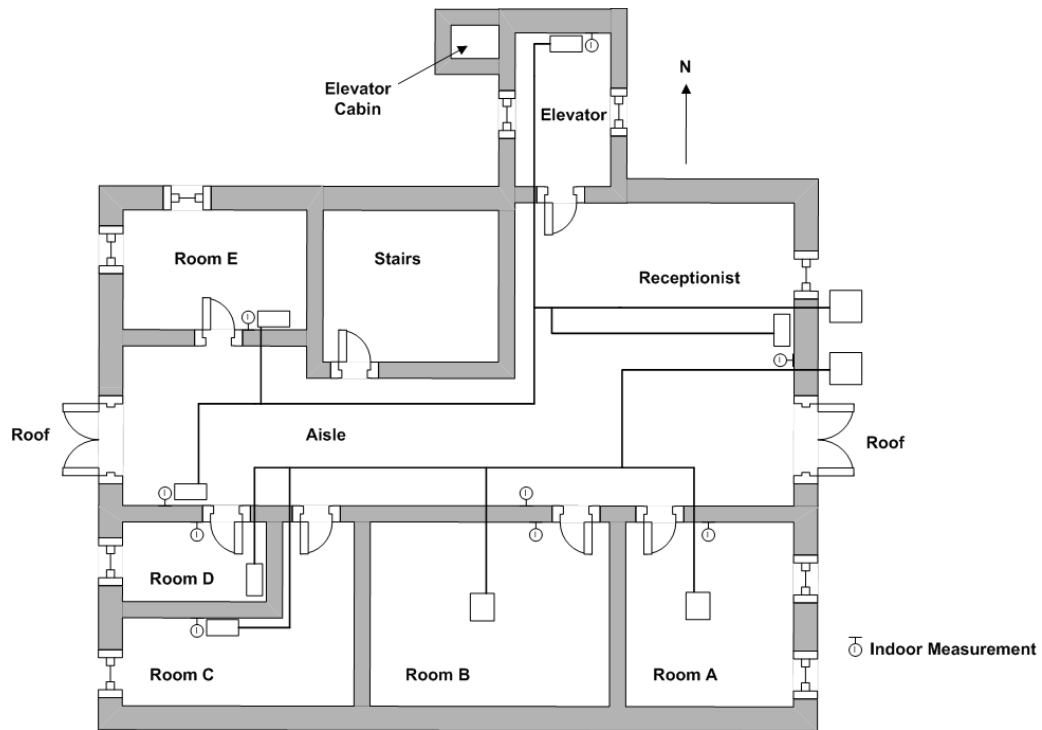


Figure 4.23 : Measurement locations of the indoor air temperature and relative humidity

In addition to these measurements, five additional T-type thermocouples were installed into Room C according to the thermal comfort measurement locations defined in BSR/ASHRAE Standard 55P.

Three out of five thermocouples were placed 1m away from the walls. The rest were placed from 0.1m and 0.6m above the floor in order to measure the temperature of the occupant location. The measurement locations are provided in Figure 4.24 and Figure 4.25.



Figure 4.24 : Measurement locations of the occupant location: 0.1m and 0.6m above the floor



Figure 4.25 : Measurement locations: 1m away from the walls

By using these five additional thermocouples, the indoor air temperature variation inside the zone can be found out and the deviation from the thermostat location can be obtained.

As can be seen from Figure 4.26a and Figure 4.26b, one T-type thermocouple with an accuracy of $\pm 0.5^{\circ}\text{C}$ and one relative humidity sensor with an accuracy of $\pm 2\%$

were installed on the roof to measure the outdoor air temperature and relative humidity. The thermocouple was shielded against the radiation and the rain.

a)



b)

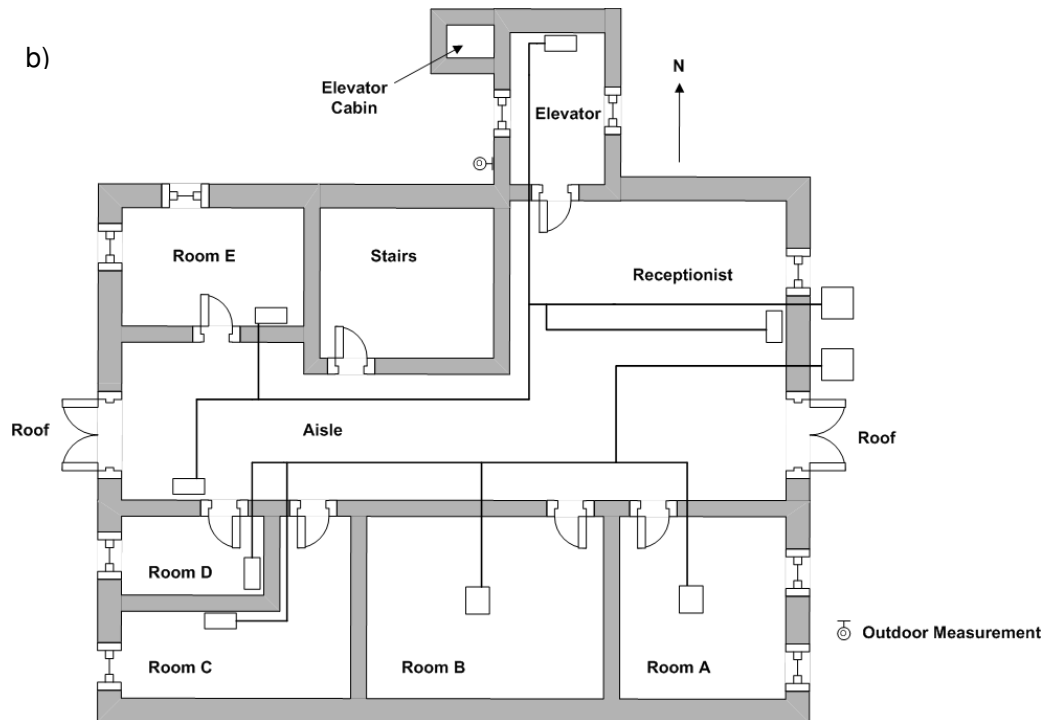


Figure 4.26 : Measurement location of the outdoor air temperature and relative humidity, (a) shielding, (b) location

In addition to the outdoor measurement location provided in Figure 4.26, a thermocouple grid consists of 16 T-type thermocouples, shown in Figure 4.27, were installed around the outdoor unit heat exchanger. Thermocouples were placed around 15cm away from the heat exchanger and shielded against the radiation and the rain. Since both outdoor units were placed side by side, only OU1 was equipped with the thermocouple grid. This thermocouple grid was used for additional outdoor temperature and outdoor unit air inlet temperature measurement.



Figure 4.27 : Thermocouple grid around the outdoor unit heat exchanger

T-type thermocouples were installed to each air discharge and suction grille of each indoor unit to measure the discharge and suction air temperature. Figure 4.28 depicts the location of the thermocouple at the discharge grill.

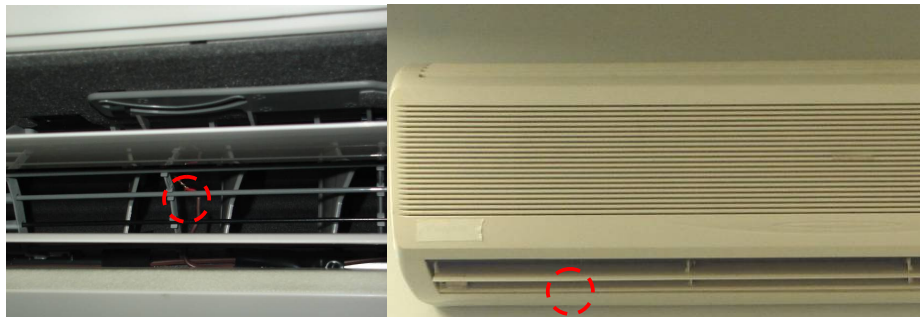


Figure 4.28 : Measurement location of the indoor air discharge temperature

Two T-type thermocouples were installed inside each outdoor unit to measure the air outlet temperature. Figure 4.29 depicts the location of the thermocouples. Thermocouples were not placed on top of the outdoor unit fan, because it was not easy to shield the thermocouples against the radiation and the rain without disturbing the air flow. That's why; the thermocouples were installed into the outdoor unit, just below the outdoor unit fan.



Figure 4.29 : Measurement location of the outdoor unit air outlet temperature

Four HRV units were also installed into the floor for the ventilation. Totally three T-type thermocouples with accuracy of $\pm 0.5^{\circ}\text{C}$ and three duct-type relative humidity sensors with an accuracy of $\pm 2\%$ were installed to each outdoor, supply and exhaust ducts, in order to measure the air temperature and relative humidity. The locations of the sensors are provided in Figure 4.30.

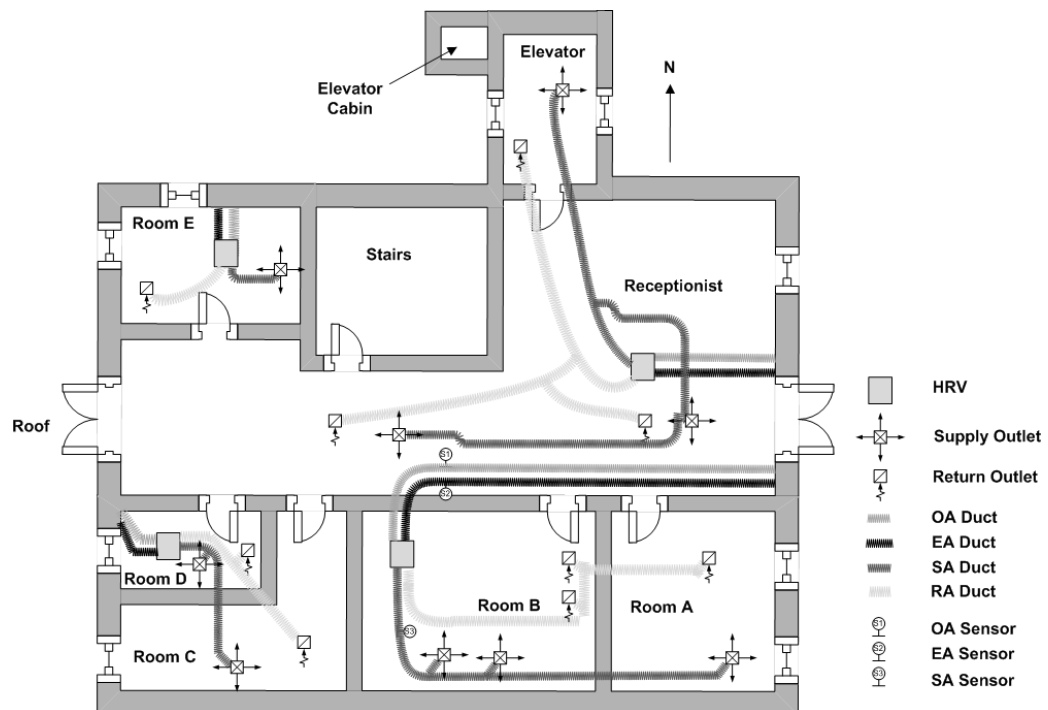


Figure 4.30 : Measurement locations of HRV1

4.6.2 Refrigerant side measurements

T-type thermocouples were attached to the refrigerant pipes for temperature measurement and several layers of insulation material were applied on to the thermocouples to increase the measurement accuracy.

Compressor refrigerant discharge and suction temperatures were measured with T-type thermocouples with an accuracy of $\pm 0.5^{\circ}\text{C}$. One thermocouple was attached to the suction line and another to the discharge line of IDC, likewise, one thermocouple was attached to the suction line and another one to the discharge line of FSC.

On the other hand, the refrigerant pressures were measured from the discharge and suction lines with two pressure sensors. The accuracies of the pressure sensors on the discharge and suction lines are $\pm 83\text{kPa}$ and $\pm 34\text{kPa}$, respectively.

The locations of the thermocouples and the pressure sensors are shown in Figure 4.31 and Figure 4.32.



Figure 4.31 : Measurement locations for the compressor refrigerant discharge and suction temperature

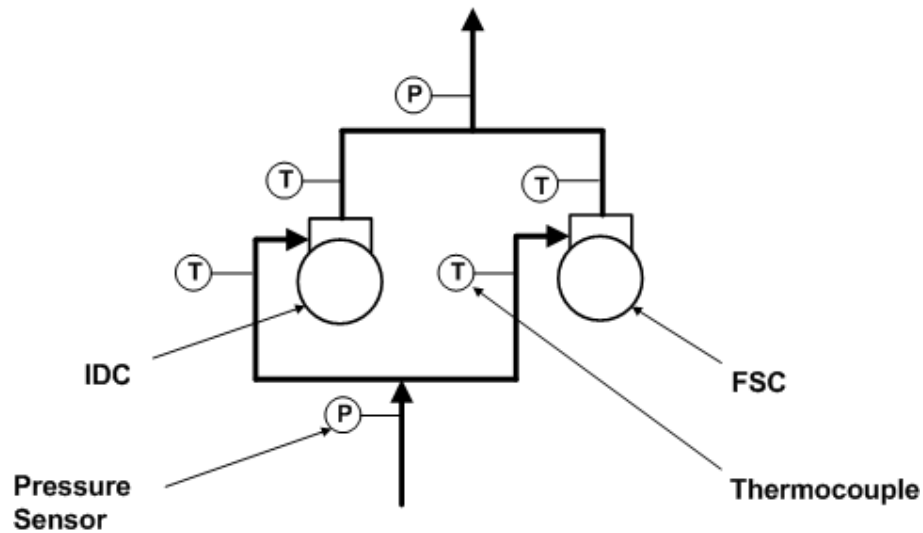


Figure 4.32 : Measurement locations for the compressor refrigerant discharge and suction temperature and pressure

Two T-type thermocouples with an accuracy of $\pm 0.5^{\circ}\text{C}$ were attached to the inlet and outlet of each outdoor unit heat exchanger.

The refrigerant paths of the cooling and heating modes were reverse. In the cooling mode, EEVs were in the downstream of the refrigerant flow and in the heating mode, EEVs were in the upstream of the refrigerant flow. Considering this fact, three T-type thermocouples with an accuracy of $\pm 0.5^{\circ}\text{C}$ were used for the refrigerant temperature measurements.

One thermocouple was attached to the refrigerant pipe just before EEV and another one was attached after EEV and the third one was attached to the refrigerant pipe after the heat exchanger.

The thermocouples used for the temperature measurement in the cooling mode and in the heating mode are shown in Figure 4.33 and Figure 4.34, respectively.

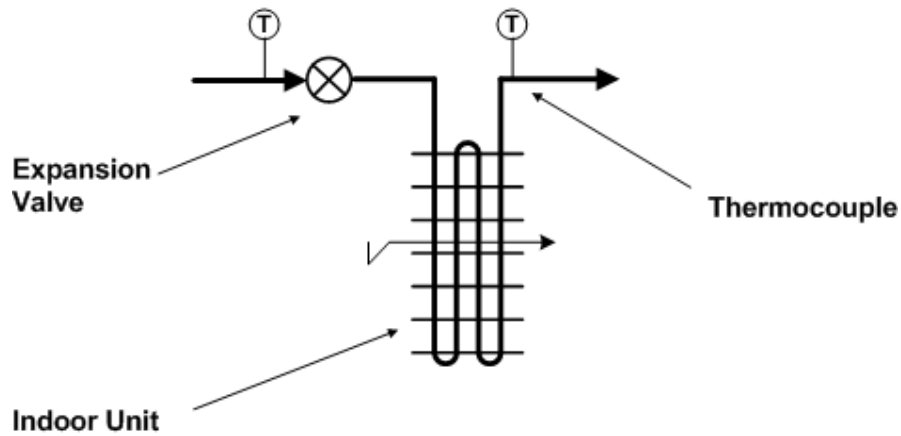


Figure 4.33 : Measurement locations of the indoor unit temperature for the cooling mode

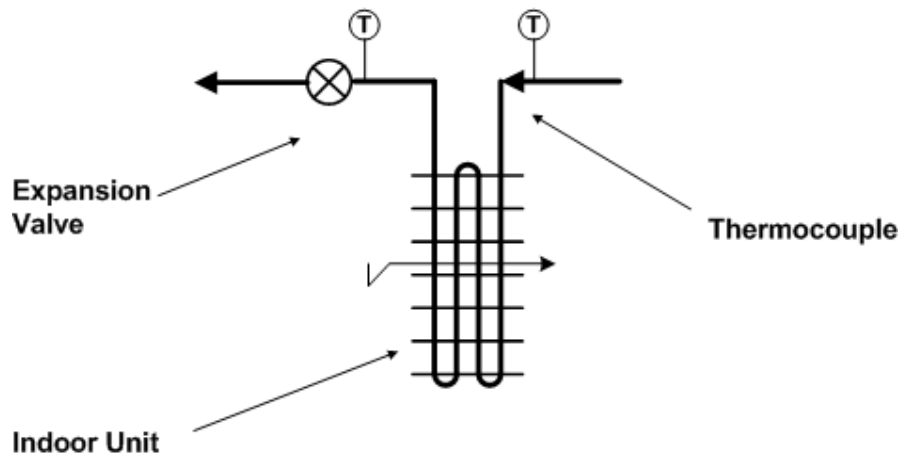


Figure 4.34 : Measurement locations of the indoor unit temperature for the heating mode

4.6.3 Power consumption measurements

Two separate watt meters with accuracy of $\pm 0.5\%$ of full scale were used to measure the power consumption of the outdoor units. The power measurement was the total of the compressors power, fan power and the controller unit power.

In addition to outdoor units, the total power consumption of the eight indoor units and four HRV units were measured by one power meter with an accuracy of $\pm 0.5\%$ of full scale.

One of the power meters is shown in Figure 4.35.

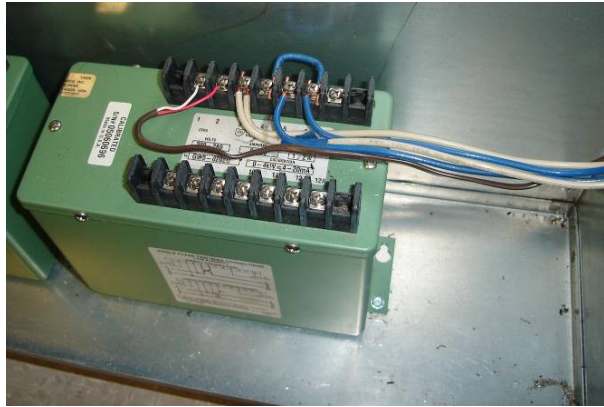


Figure 4.35 : Power meter

4.6.4 Data acquisition system

Thermocouples, relative humidity sensors and power meters were connected to nine thermocouple and three analog fieldpoint modules of National Instrument. These fieldpoint modules were connected to the data acquisition computer via a network fieldpoint module and all the data were collected, recorded and monitored by LabView 7.2 software.

In addition to these measurements, the real-time system information, such as the EEV opening, thermostat “ON/OFF” etc, were recorded through the local multi-split VRF system network.

The three fieldpoint banks with the sensor connection are shown in Figure 4.36 and the screen capture of the data acquisition system is shown in Figure 4.37.

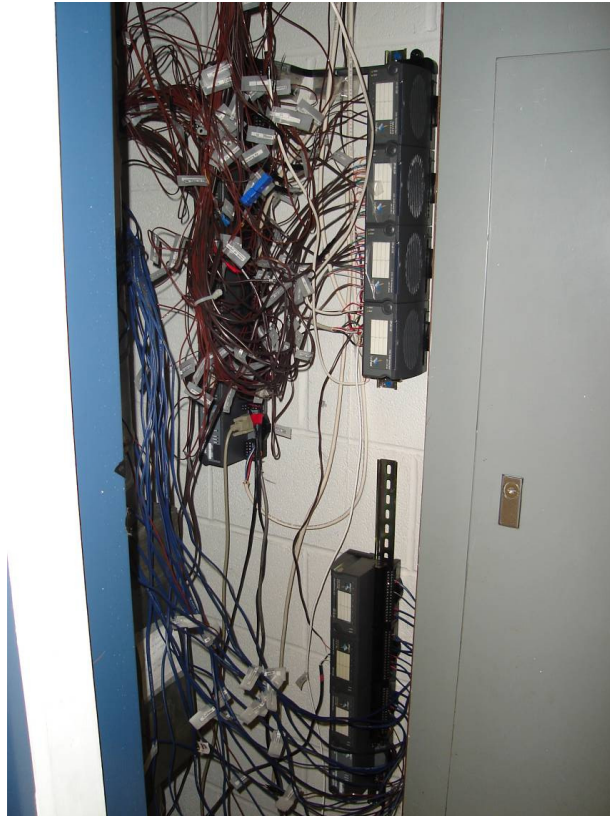


Figure 4.36 : Data acquisition system

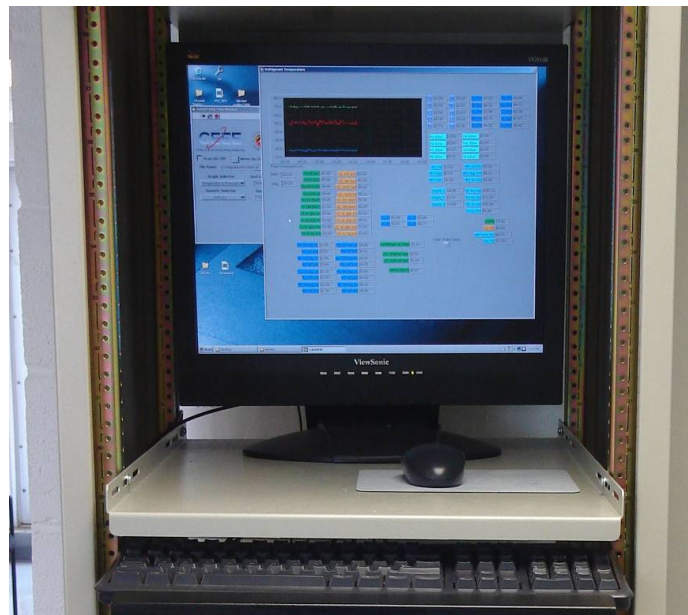


Figure 4.37 : Data acquisition system monitoring window

4.6.5 Uncertainty Analysis

Uncertainty of the experimental result occurs due to the uncertainty in the sensor and the uncertainties of the individual sensors propagate and form the total uncertainty of the measurement.

The total uncertainty of any function can be calculated using the Pythagorean summation of uncertainties which is defined by Kline and McClintoc (1953)

$$u_F = \sqrt{\left(\frac{\partial F}{\partial v_1} \cdot u_1\right)^2 + \left(\frac{\partial F}{\partial v_2} \cdot u_2\right)^2 + \dots + \left(\frac{\partial F}{\partial v_n} \cdot u_n\right)^2} \quad 4.1$$

where

u_F : Uncertainty of the function

u_n : Uncertainty of the sensor

F : Function

v_n : Parameter of measurement

n : Number of variables

Eq. (4.1) was calculated by using the uncertainty propagation function in the Engineering Equation Solver (EES) software.

5. EVALUATION METHODOLOGY

Evaluation of the performance of the multi-split VRF system is divided into two parts; the efficiency of the system and the provided thermal comfort.

5.1 Efficiency of the Multi-Split VRF System

In order to evaluate the efficiency of the multi-split VRF system in cooling and heating seasons, the cooling and heating capacities of each indoor unit should be calculated. That's why; the individual refrigerant mass flow rate of each indoor unit should be obtained.

The refrigerant mass flow rate sensors are relatively expensive devices, because of this situation, compressor performance maps have been extensively used in various studies (Shao et al., 2004), (Park et al., 2001), (Wu et al., 2005). Since this study consisted of two outdoor and eight indoor units, the compressor performance map provided by the manufacturer was used to calculate the total refrigerant mass flow rate discharged by the compressors.

The refrigerant mass flow rate is defined as

$$\dot{m}_T = \dot{m}_T(T_c, T_e, T_{suc}, f, STN_{sta}) \quad 5.1$$

where

$\dot{m}_T$: Refrigerant mass flow rate (kg/s)

T_c : Condensing temperature (°C)

T_e : Evaporating temperature (°C)

T_{suc} : Compressor refrigerant suction temperature (°C)

f : Frequency of IDC (Hz)

STN_{sta} : Status of fixed speed compressor: (ON or OFF)

The refrigerant mass flow rate variation with respect to the evaporating temperature at different operating inverter frequencies and condensing temperatures with a

constant compressor suction temperature (20°C) when FSC is “OFF” and “ON” are provided in Figure 5.1 and Figure 5.2, respectively.

The results show similar trends with Shao et al. (2004).

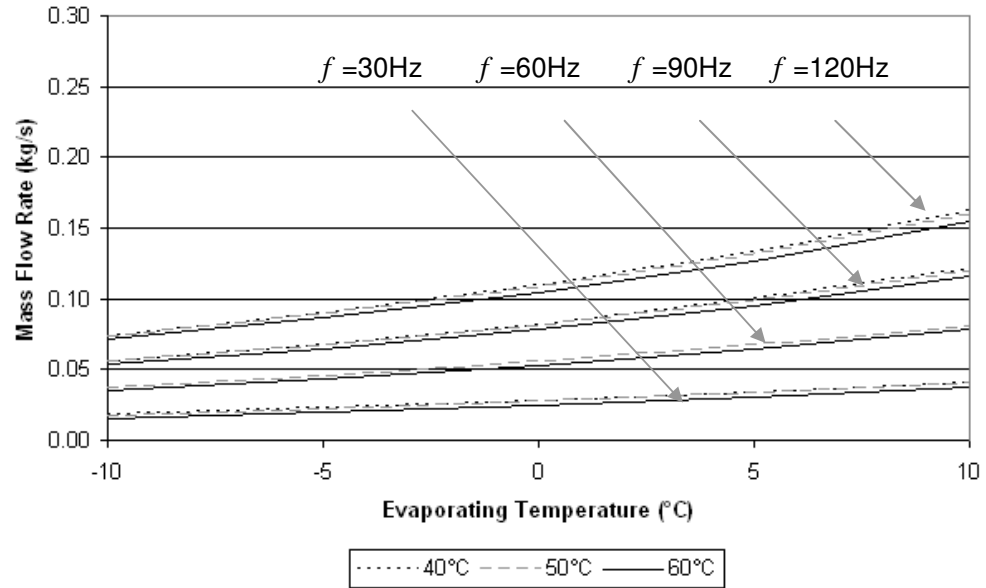


Figure 5.1 : Variation of the refrigerant mass flow rate with respect to the evaporating temperature for different inverter frequencies and the condensing temperatures when FSC is OFF

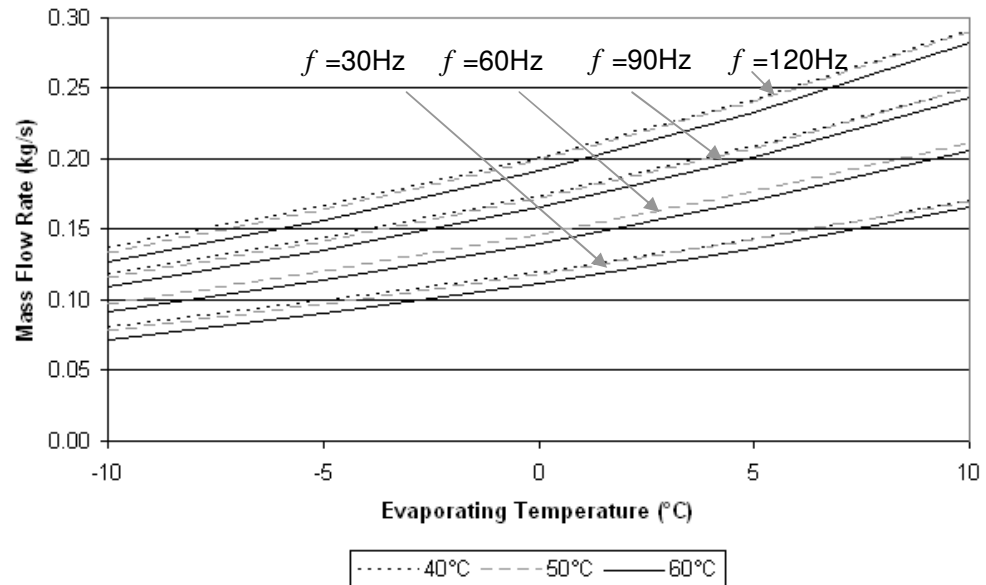


Figure 5.2 : Variation of the refrigerant mass flow rate with respect to evaporating temperature for different inverter frequencies and condensing temperatures when FSC is ON

The total refrigerant mass flow rate was divided into each indoor unit according to the EEV correlations. This approach is widely used in the literature (Hu and Yang, 2005), (Park et al., 2001), (Shah et al., 2004), (Wu et al., 2005), (Xia et al., 2003), (Lee et al., 2006).

The individual refrigerant mass flow rate of the indoor unit is defined as

$$R_i = c_D \cdot A \cdot \sqrt{2 \cdot \Delta P \cdot \rho} \quad 5.2$$

$$\dot{m}_i = \frac{R_i}{\sum_{i=1}^4 R_i} \cdot \dot{m}_T \quad 5.3$$

where

R_i : The coefficient for the i^{th} indoor unit ($i=1$ to 4)

c_D : The flow coefficient

A : The area of EEV (m^2)

ΔP : The pressure drop across EEV (Pa)

ρ : The refrigerant density at the inlet of EEV (kg/m^3)

$\dot{m}_i$: The individual refrigerant mass flow rate of the i^{th} indoor unit (kg/s)

The approach for the individual refrigerant mass flow rate defined in Eq. (5.2) and Eq. (5.3) was verified in App. [B] with an energy balance between the air side and the refrigerant side. During the calculations, the pressure drops of each indoor unit heat exchanger and the outdoor unit heat exchanger are neglected. This assumption was verified in App. [C].

The individual cooling capacity of each indoor unit is defined as

$$\dot{Q}_{C,i} = \dot{m}_i \cdot (h_{c,i,o} - h_{c,i,i}) \quad 5.4$$

where

$\dot{Q}_{C,i}$: The cooling capacity of the i^{th} indoor unit (W)

$h_{c,i,o}, h_{c,i,i}$: The outlet and inlet enthalpies of the i^{th} indoor unit, respectively (J/kg)

As can be seen from Figure 4.33, the refrigerant enters the indoor unit through EEV. That's why, the refrigerant enthalpy before EEV is assumed to be the same as the inlet enthalpy of the indoor unit heat exchanger. Refrigerant enthalpies were calculated by RefProp 7.0.

The total cooling capacity of each multi-split VRF system is defined as

$$\dot{Q}_{C,T} = \sum_{i=1}^4 \dot{Q}_{C,i} \quad 5.5$$

$$\dot{Q}_{C,T} = \sum_{i=1}^4 \dot{m}_i \cdot (h_{c,i,o} - h_{c,i,i}) \quad 5.6$$

where

$\dot{Q}_{C,T}$: The total cooling capacity of the multi-split VRF system (W)

Likewise, the individual heating capacity of each indoor unit is defined as

$$\dot{Q}_{H,i} = \dot{m}_i \cdot (h_{h,i,i} - h_{h,i,o}) \quad 5.7$$

where

$\dot{Q}_{H,i}$: The heating capacity of the i^{th} indoor unit (W)

$h_{h,i,i}, h_{h,i,o}$: The inlet and outlet enthalpies of the i^{th} indoor unit, respectively (J/kg)

The total heating capacity of each multi-split VRF system is defined as

$$\dot{Q}_{H,T} = \sum_{i=1}^4 \dot{Q}_{H,i} \quad 5.8$$

$$\dot{Q}_{H,T} = \sum_{i=1}^4 \dot{m}_i \cdot (h_{h,i,o} - h_{h,i,i}) \quad 5.9$$

where

$\dot{Q}_{H,T}$: The total heating capacity of the multi-split VRF system (W)

The cooling performance factor (CPF) and heating performance factor (HPF) are used for the evaluation of the performance of the multi-split VRF system in cooling and heating modes, respectively, and defined as

$$CPF = \frac{\sum \dot{Q}_{C,T} \cdot t}{\sum \dot{W} \cdot t} \quad 5.10$$

$$HPF = \frac{\sum \dot{Q}_{H,T} \cdot t}{\sum \dot{W} \cdot t} \quad 5.11$$

$$\dot{W} = \dot{W}_{OU} + \sum \dot{W}_{IU} \quad 5.12$$

$$\dot{W}_{OU} = \dot{W}_{comp} + \dot{W}_{OUfan} \quad 5.13$$

$$\dot{W}_{comp} = \dot{W}_{comp,inv} + \dot{W}_{comp,fix} + \dot{W}_{contr} \quad 5.14$$

where

CPF : Cooling performance factor

HPF : Heating performance factor

$\dot{W}$: The total power consumption (W)

$\dot{W}_{OU}$: The power consumption of the outdoor unit (W)

$\dot{W}_{IU}$: The power consumption of the indoor unit (W)

$\dot{W}_{comp}$: The power consumption of the compressors (W)

$\dot{W}_{OUfan}$: The power consumption of the outdoor unit fan (W)

$\dot{W}_{comp,inv}$: The power consumption of IDC (W)

$\dot{W}_{comp,fix}$: The power consumption of FSC (W)

$\dot{W}_{contr}$: The power consumption of the controller in the outdoor unit (W)

t : Time (h)

CPF and HPF can be evaluated daily and seasonally.

Since the multi-split VRF systems were evaluated under actual operating conditions, they worked at part load conditions most of the test time. PLR is defined as

$$PLR = \frac{\dot{Q}_{C,T}}{\dot{Q}_{C,OU,Cat}} \quad 5.15$$

where

PLR : Part load ratio

$\dot{Q}_{C,OU,Cat}$: The catalog cooling capacity of the outdoor unit (W)

5.2 Provided Thermal Comfort

ASHRAE summer and winter comfort zones are used for the thermal comfort comparisons. The comfort zones are provided in Figure 5.3. The zone with the solid lines depicts the summer comfort zone, and the one with the dashed line represents the winter comfort zone.

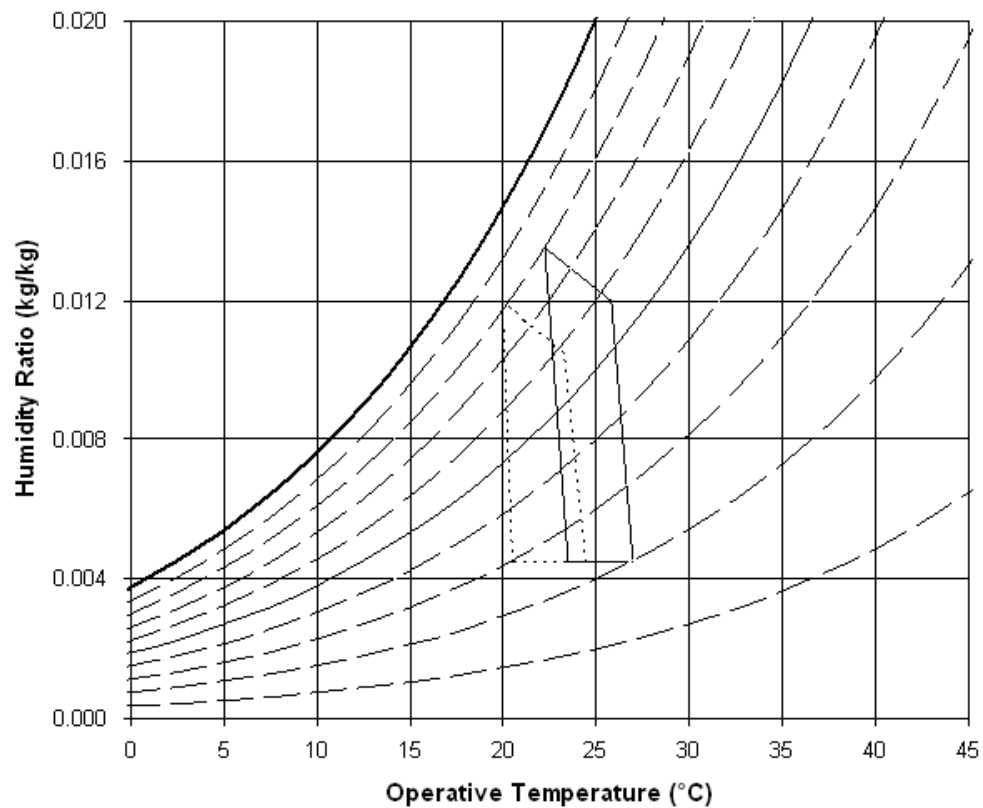


Figure 5.3 : ASHRAE summer and winter comfort zones

The measured indoor dry bulb temperature can be assumed to be equal to the operative temperature (Han et al., 2006), (Newsham et al., 1997), (Chamra et al., 2002), (Amai et al., 2007).

Inside of the comfort zones are claimed to be comfortable. Outside of the ASHRAE summer thermal comfort zone was divided into four regions; cold, hot, humid and dry regions. Cold and hot regions were defined as the left and the right side of the thermal zone, respectively. Humid region was defined as the upper part of the thermal zone with a humidity ratio higher than 0.012kg/kg. Dry region was defined as the humidity ratio lower than 0.0045kg/kg.

In addition to the ASHRAE comfort zones, the thermal sensation scales defined in Rohles and Nevins (1971) and Newsham et al. (1997) are used for the cooling and heating seasons, respectively.

Rohles and Nevins (1971) tabulated the multiple regression analysis for each of the five exposure periods for men, women and men and women combined. It was assumed that the occupants stayed in their office rooms around three hours and gave a break and then stayed three more hours. That's why; three-hour exposure time and both men and women combined equation was chosen for the cooling season evaluations from Table 2.2. The equation is as follows

$$Y = 0.243T + 0.278p - 6.802 \quad 5.16$$

where

Y : ASHRAE TSS (from -3 to 3 where -3 is cold, -2-cool, -1-slightly cool, 0-neutral, +1-slightly warm, +2-warm, +3-hot)

T : Dry bulb temperature (°C)

P : Vapor pressure (kPa)

On the other hand, only one equation was defined in the study of Newsham et al. (1997). That equation was used for the heating season evaluations. The equation is as follows

$$TS = -7.56 + 0.33T_{ia} \quad 5.17$$

where

TS : ASHRAE thermal sensation vote

T_{ia} : Indoor air temperature (°C)

6. EXPERIMENTAL RESULTS

After the installation and the instrumentation of the multi-split VRF systems and the HRV units, they were operated and tested over a wide range of outdoor temperature and humidity ratio in the cooling and heating modes.

The preliminary tests were done in 2005 cooling and heating seasons.

Table 6.1 and Table 6.2 show the weekly schedule of the cooling tests in 2006 and 2007, respectively.

Table 6.3 shows the weekly schedule of the heating tests in 2006.

Table 6.1 : Weekly schedule of the cooling tests in 2006

DAY	SYSTEM CONFIGURATION				
	System	Control Mode	Set Temperature (°C/°F)	Number of Indoor Units	Ventilation
Monday	VRF	Individual	25/77	4, 4	w/
Tuesday	VRF	Individual	25/77	4, 4	w/o
Wednesday	VRF	Individual	25/77	0, 4	w/
Thursday	VRF	Master	25/77	4, 4	w/
Friday	VAV	Central	25/77	X	existing
Saturday	VRF	Master	25/77	4, 4	w/
Sunday	VRF	Individual	25/77	0, 4	w/

Table 6.2 : Weekly schedule of the cooling tests in 2007

DAY	SYSTEM CONFIGURATION				
	System	Control Mode	Set Temperature (°C/°F)	Number of Indoor Units	Ventilation
Monday	VRF	Individual	25/77	4, 4	w/
Tuesday	VRF	Individual	25/77	4, 4	w/o
Wednesday	VRF	Individual	20/68, 25/77	2, 4	w/o
Thursday	VRF	Master	25/77	4, 4	w/
Friday	VRF	Individual	25/77	4, 4	w/
Saturday	VRF	Master	20/68	4, 4	w/
Sunday	VRF	Individual	20/68	4, 4	w/

Table 6.3 : Weekly schedule of the heating tests in 2006

DAY	SYSTEM CONFIGURATION				
	System	Control Mode	Set Temperature (°C/°F)	Number of Indoor Units	Ventilation
Monday	VRF	Individual	23.3/74	4, 4	w/
Tuesday	VRF	Individual	23.3/74	4, 4	w/o
Wednesday	VRF	Individual	23.3/74	2, 4	w/o
Thursday	VRF	Master	23.3/74	4, 4	w/
Friday	VRF	Master	26.1/79	4, 4	w/
Saturday	VRF	Individual	23.3/74	4, 4	w/
Sunday	VRF	Individual	26.1/79	0, 4	w/

The indoor condition comparisons between the multi-split VRF system in master control mode and the existing VAV system in central control mode are presented in App. [D]. According to the comparison, it can be assumed that the master control represents the operation of the central control mode.

6.1 Effect of the Control Modes in the Cooling Performance

In order to find out the effect of the individual and master control modes on the indoor temperature control, thermal comfort, power consumption of the outdoor units and the efficiency of the system, parametric tests were performed with two different indoor set temperatures; 25°C and 20°C over a wide range of outdoor temperature and humidity ratio.

The multi-split VRF systems in conjunction with the HRV units were operated from 7am to midnight, and the fans of the indoor units were operated continuously regardless of the indoor unit operation.

Evaluations were performed daily basis for similar outdoor conditions and seasonally.

6.1.1 Daily comparison at 25°C set temperature

Similar outdoor conditions were tried to be found for a fair daily basis comparison for the individual and master control modes.

Figure 6.1 and Figure 6.2 show the outdoor temperature and the relative humidity variations for the individual control mode tests from 2006 and 2007 cooling seasons, respectively.

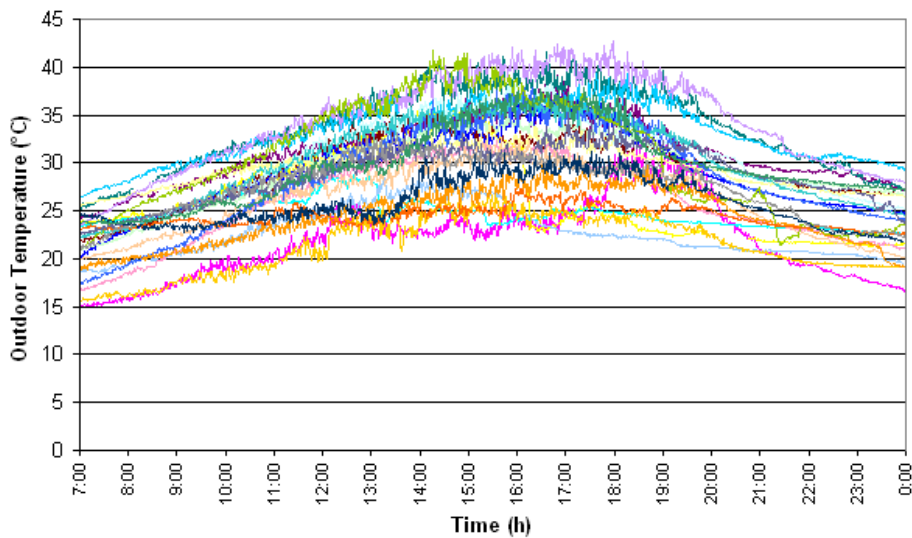


Figure 6.1 : Outdoor temperature variations for the individual control mode tests at 25°C set temperature

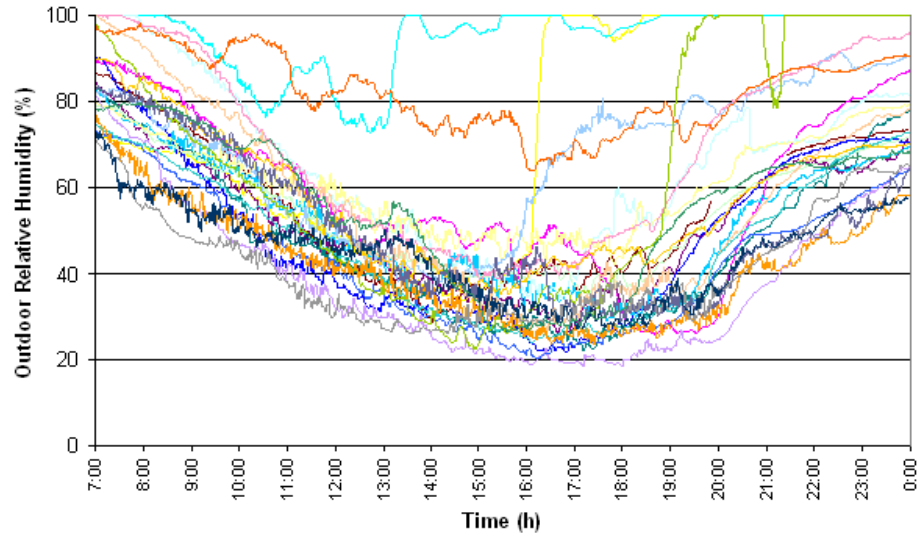


Figure 6.2 : Outdoor relative humidity variations for the individual control mode tests at 25°C set temperature

Figure 6.3 and Figure 6.4 show the outdoor temperature and relative humidity variations for the master control mode tests from 2006 and 2007 cooling seasons, respectively.

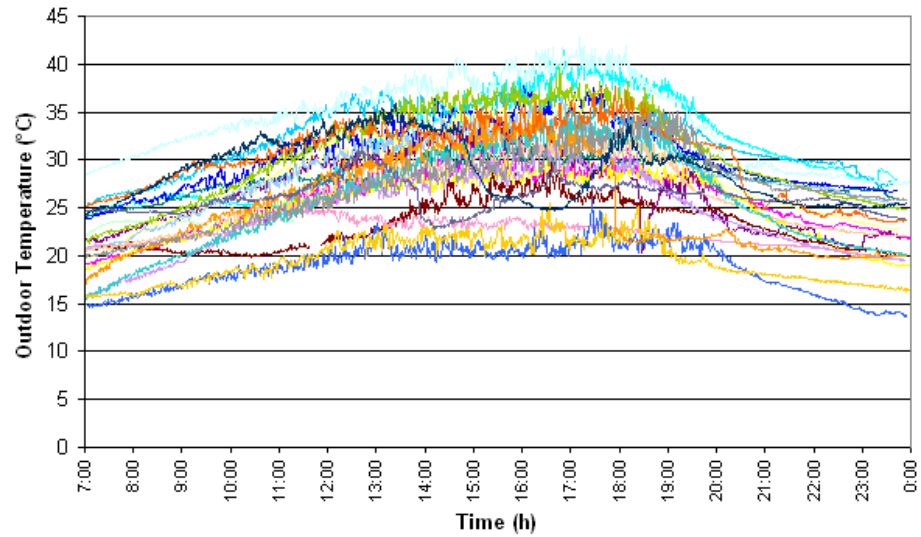


Figure 6.3 : Outdoor temperature variations for the master control mode tests at 25°C set temperature

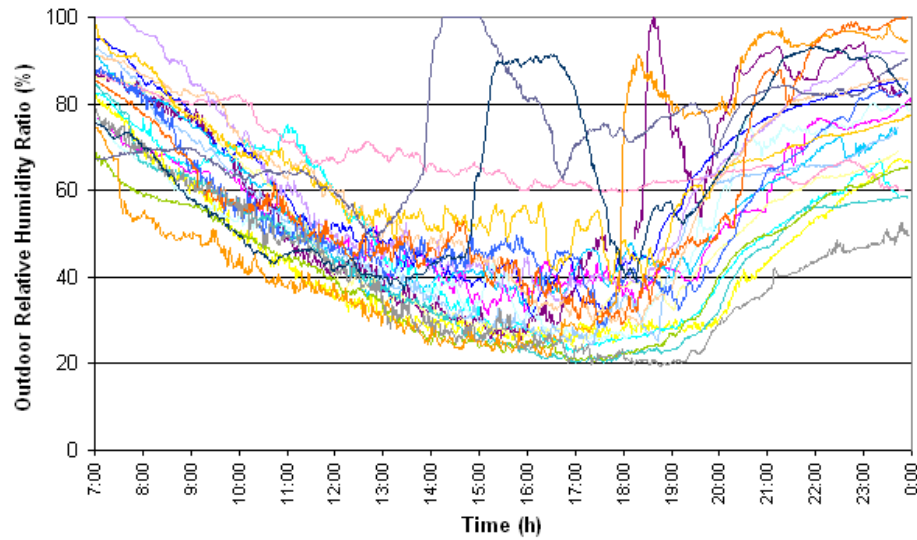


Figure 6.4 : Outdoor relative humidity variations for the master control mode tests at 25°C set temperature

Two weekdays; June 01, 2007 and May 31, 2007 with similar outdoor conditions were chosen for daily basis comparison of the individual and master control modes. The multi-split VRF systems were operated in individual control mode on June 01, 2007 and in master control mode on May 31, 2007.

Figure 6.5 shows the outdoor temperature variations on May 31, 2007 and June 01, 2007 and the outdoor temperature difference during the test period. Data were sampled every minute.

As seen from Figure 6.5 that the variations of the outdoor temperatures were very similar and according to daily average of the temperature difference, the outdoor temperature for the master control mode was 0.45°C higher than the individual control mode one.

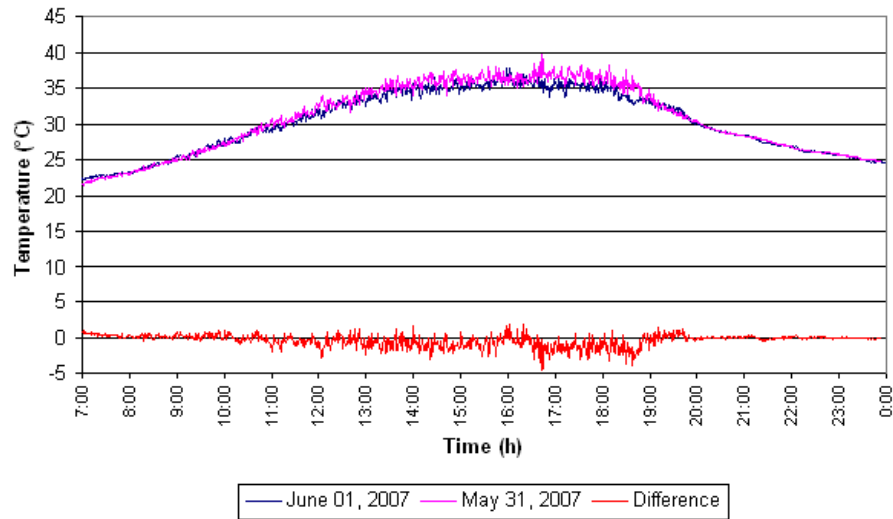


Figure 6.5 : Outdoor temperature variations on June 01, 2007 and May 31, 2007

Figure 6.6 shows the outdoor relative humidity variations and the relative humidity difference for the same days during the test period.

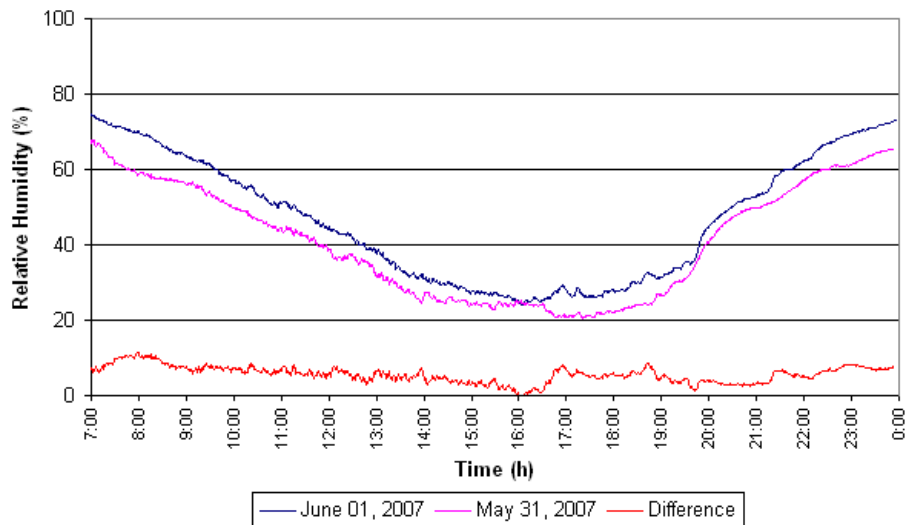


Figure 6.6 : Outdoor relative humidity variations on June 01, 2007 and May 31, 2007

It can be seen from Figure 6.6 that the relative humidity variations were similar and according to daily average of the relative humidity difference, the relative humidity for the individual control mode day was 5.6% higher than the master control mode one.

Since the outdoor conditions of these two weekdays were very similar, they were used for daily comparison of the individual and master control modes.

For simplicity, the total multi-split VRF system composed of OU1 and the associated indoor units (IURA, IURB, IURC and IURD) were referred as System1 and the associated HRV units (HRV1, HRV2) were referred as System1 HRV units, and the total multi-split VRF system composed of OU2 and the associated indoor units (IUEle, IUREc, IUAls and IURE) were referred as System2 and the associated HRV units (HRV3 and HRV4) were referred as System2 HRV units.

Figure 6.7 shows the indoor temperature variations of System1 zones on June 01, 2007. The multi-split VRF system was operated in individual control mode on this day.

It can be seen from Figure 6.7 that the initial zone temperatures were almost 26°C. The multi-split VRF system started working at 7:00 and cooled down the zone temperatures and then tried to maintain the indoor temperatures at the desired indoor set temperature (25°C) throughout the day.

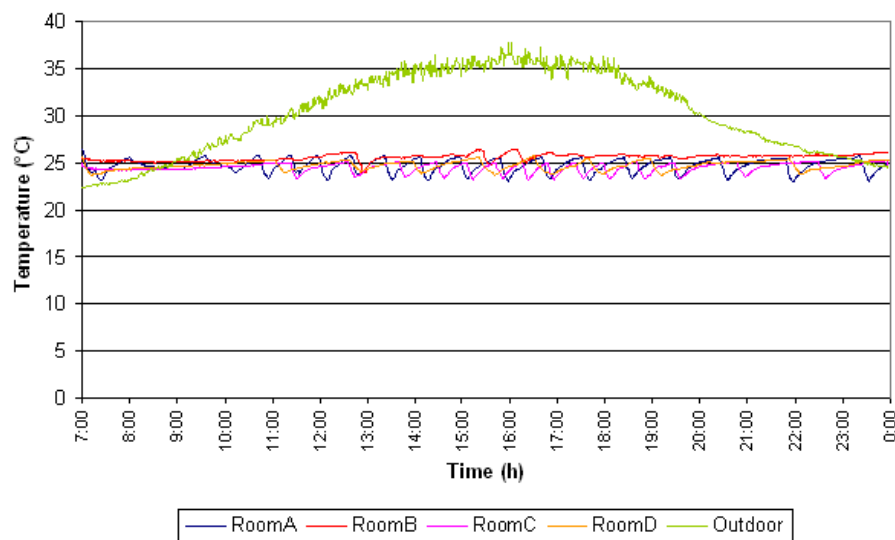


Figure 6.7 : Indoor temperatures of System1 zones and outdoor temperature on June 01, 2007

Figure 6.8 shows the indoor temperature variations of System2 zones on the same day.

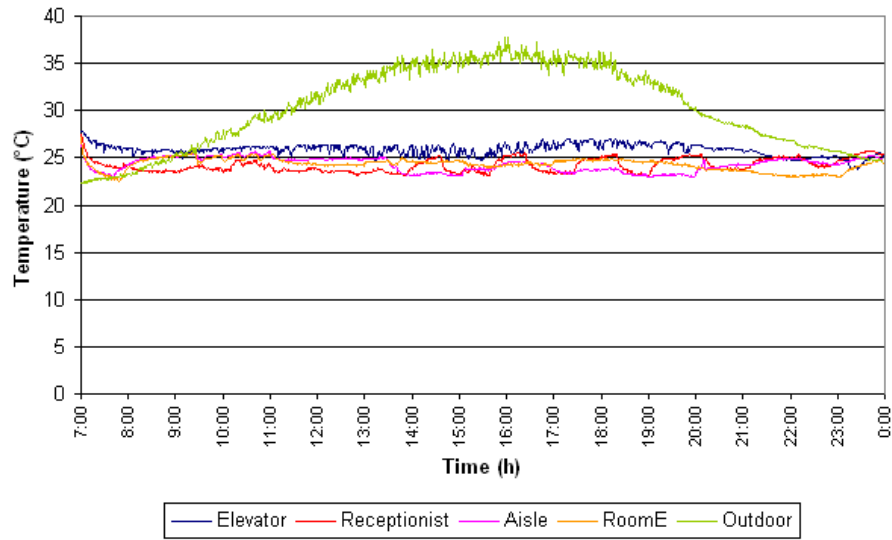


Figure 6.8 : Indoor temperatures of System2 zones and outdoor temperature on June 01, 2007

It can be seen from Figure 6.8 that the indoor initial temperatures of System2 zones were slightly higher than that of System1 zones.

It is attributed to the total window area and the total internal load sources. The window areas of the zones and the number of the internal load sources are given in Table 6.4 and Table 6.5, respectively. The total window area of System2 zones is 2.38 times bigger than System1 zones, and the total internal load source of System2 zones is higher than that of System1 zones.

Table 6.4 : Window areas of System1 and System2 zones

	System1 Zones				System2 Zones			
	Room A	Room B	Room C	Room D	Elevator	Receptionist	Aisle	Room E
Window Area (m ²)	3.54	0	1.77	1.77	6.87	1.77	4.66	3.54
Total Window Area (m ²)	7.08				16.84			

Table 6.5 : Number of the internal load sources of System1 and System2 zones

	System1 Zones				System2 Zones			
	Room A	Room B	Room C	Room D	Elevator	Receptionist	Aisle	Room E
Number of Occupancy	1	0	0	1	0	1	0	1
Number of Computer	1	0	0	1	0	4	0	1
Number of Copy, Fax Machines	0	0	1	1	0	4	0	1
Number of Appliance Machines	0	0	1	0	0	2	1	0
Number of Lights	4x5	4x6	4x4	4x2	2x2	4x5	4x7	4x3

Comparing Figure 6.7 and Figure 6.8 shows that elevator zone which had the bigger window area had the higher initial indoor temperature than the other zones. On the other hand, Room B which had no windows had the lowest initial indoor temperature than the other zones and the indoor temperature barely exceeded the higher limit of the thermostat dead band throughout the test time.

Figure 6.9 depicts the TSS variation of System1 zones on June 01, 2007 according to Eq. (5.16).

As seen from Figure 6.9 with 10-minute averaged data, the TSS variations were almost within the neutral and slightly cool region and the minimum and the maximum of the scale throughout the test time were -0.72 and 0.15, respectively.

Figure 6.10 shows the TSS and the indoor air temperature variations for Room A for the same test day. It can be seen from Figure 6.10 that the indoor temperature and the TSS variations showed similar trends, namely when the indoor temperature decreased the corresponding TSS decreased, and when the indoor temperature increased, the corresponding TSS increased.

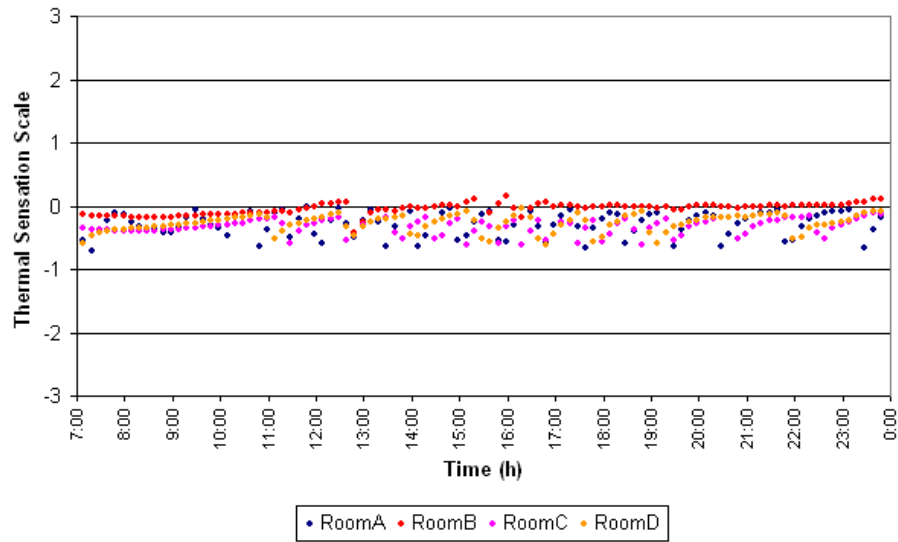


Figure 6.9 : TSS variation of System1 zones on June 01, 2007

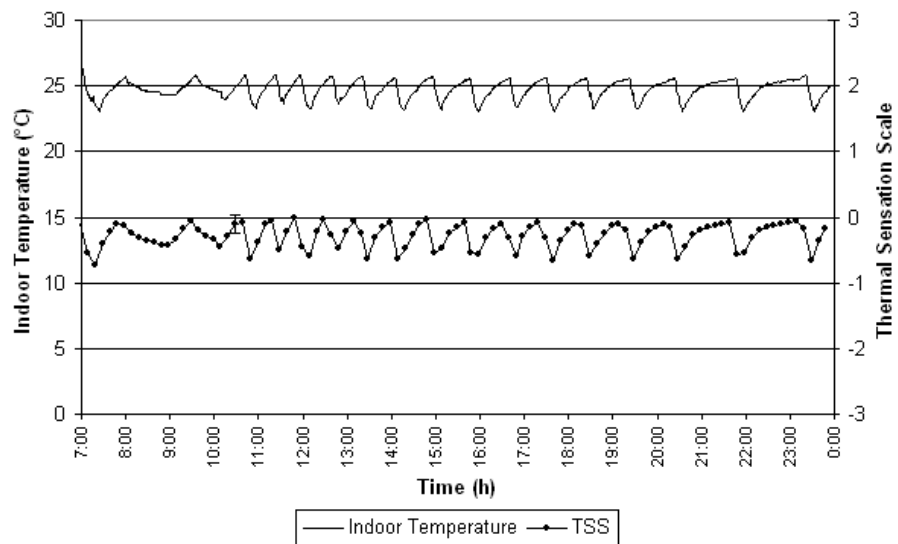


Figure 6.10 : Indoor temperature and the TSS variation of Room A on June 01, 2007

Figure 6.11 shows the TSS variation of System2 zones. As seen from Figure 6.11, similar to System1 zones, TSS varied within the neutral and slightly cool region. The minimum and the maximum of the scale throughout the test time for System2 zones were -0.85 and 0.40, respectively.

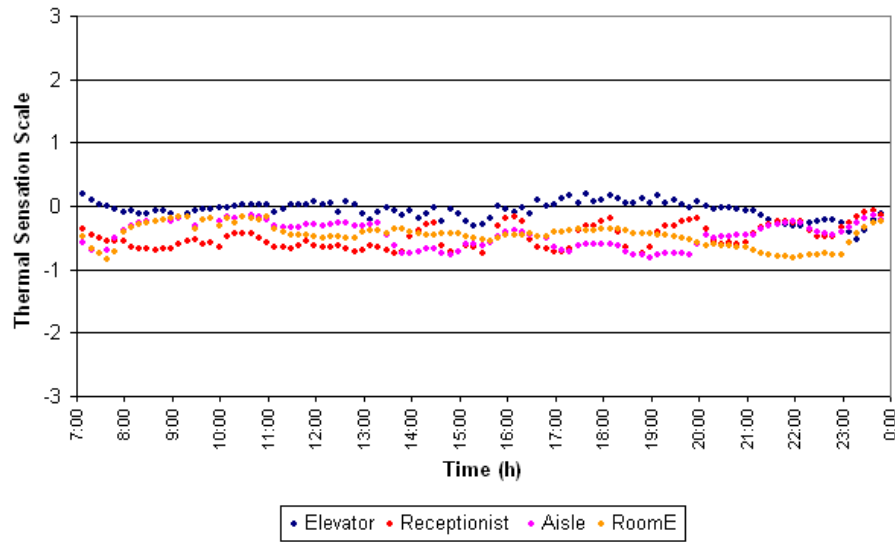


Figure 6.11 : TSS variation of System2 zones on June 01, 2007

Figure 6.12 shows the operation of System1 indoor units in terms of “ON” (working) and “OFF” (not working) on the same day.

Since System1 had four indoor units, four different numbers were assigned for “ON” time; such as 1 for Room A, 2 for Room B, etc. instead of making operations of the indoor units distinctive and zero values were assigned to the “OFF” time of the indoor unit.

During the “ON” time, EEV was kept open, thus the refrigerant coming from the outdoor unit could pass through the heat exchanger of the indoor unit, and indoor unit could provide the requested cooling. On the other hand, during the “OFF” time, EEV was kept close, so that the refrigerant was not allowed to pass through the heat exchanger of the indoor unit. Therefore, the indoor unit could not provide cooling for this period. The air discharge temperature of System1 indoor units are show in Figure 6.13.

As seen from Figure 6.7 and Figure 6.12, when IURA was “ON”, the indoor temperature of Room A decreased and when it was “OFF”, the indoor temperature increased because of the internal load, the ventilation effect and heat transfer from the ambient. Since the indoor set temperature for this case was 25°C, the individual control mode was trying to maintain 25°C by turning “ON” and “OFF” of the corresponding indoor units and this situation ended up with a cycling temperature variation as shown in Figure 6.7. Similar variations were also valid for the other System1 zones; such as Rooms C and D. However, as for Room B, since Room B did not have any window and the internal load for this zone was very low, the indoor

temperature did not increase a lot during the day. That's why; IURB was "OFF" until 13:00.

As seen from Figure 6.12 that when the outdoor temperature increased, the operation time of the indoor units increased accordingly.

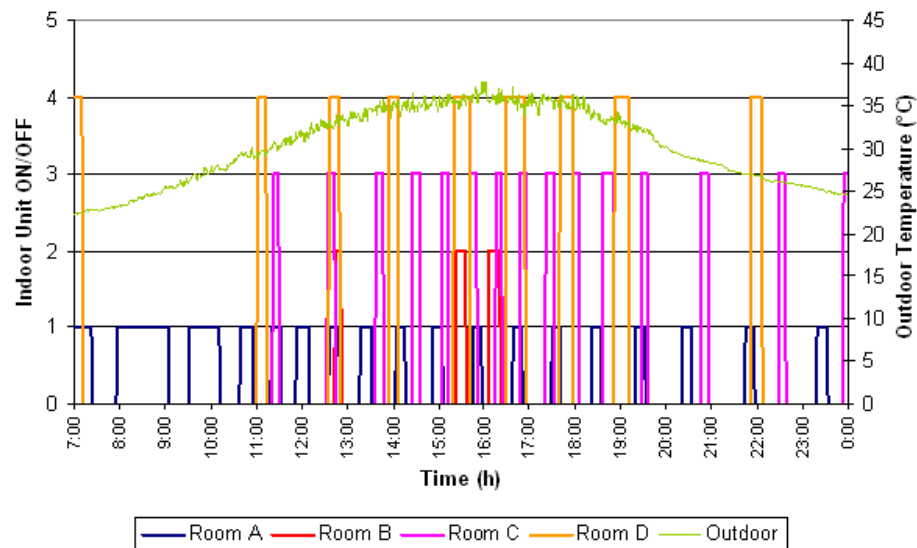


Figure 6.12 : Indoor units operation of System1 zones in individual control mode on June 01, 2007

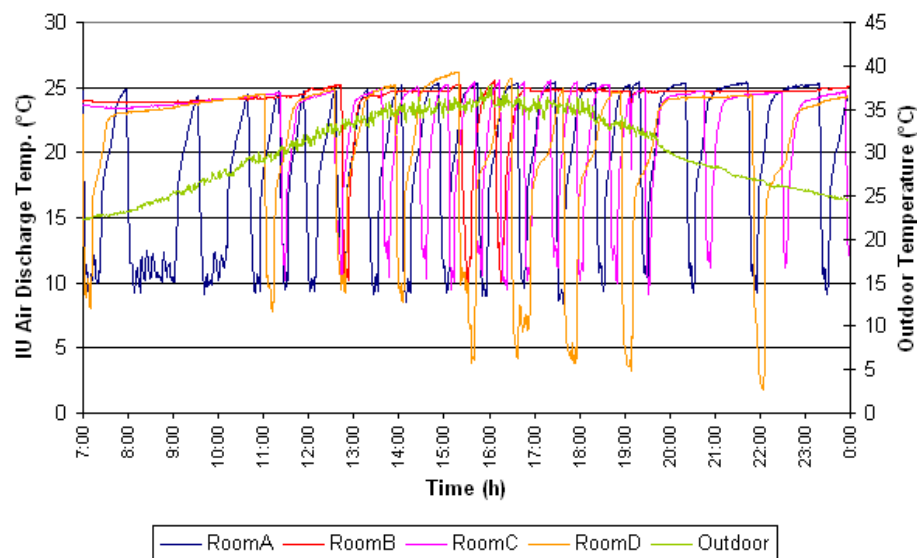


Figure 6.13 : Indoor unit air discharge temperatures of System1 on June 01, 2007

Figure 6.14 shows the relationship between IURA operation and the EEV opening of IURA. As seen from Figure 6.14, when IURA was "ON", the EEV of IURA was opened and allowed the refrigerant pass through the indoor unit heat exchanger to

cool down the indoor air. On the other hand, when IURA was “OFF”, the EEV of IURA was closed. The corresponding Room A indoor temperature fluctuations can be seen from Figure 6.7.

The relationship between the indoor unit operation and the EEV opening for IURB, IURC and IURD are provided in App. [E].

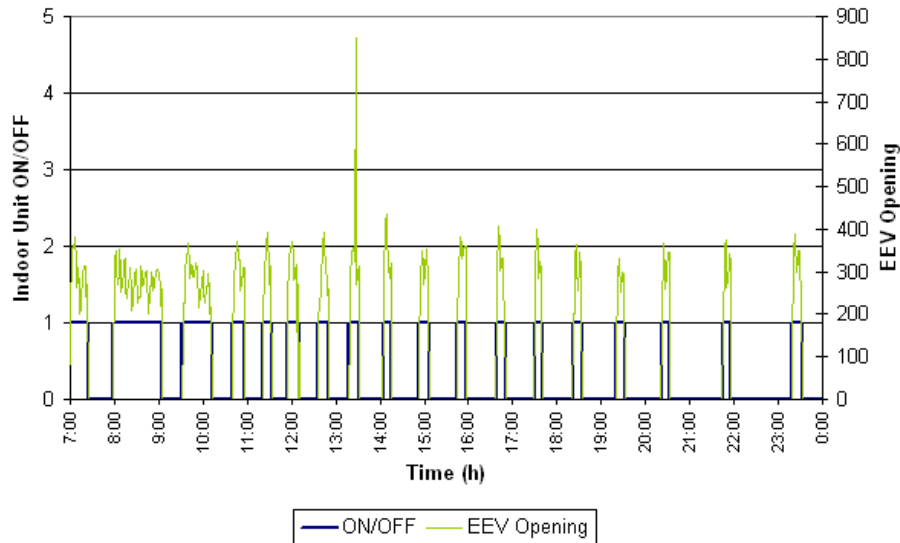


Figure 6.14 : Relationship between the operation and the EEV opening of IURA on June 01, 2007

Figure 6.15 shows the indoor units operation of System2 zones in terms of “ON” or “OFF” time.

It can be seen from Figure 6.15 that the “ON” times of the indoor units of System2 zones were much longer than System1 zones due to the higher internal load and relatively larger window area.

As seen from Figure 6.15, IUEle was “ON” almost all time of the test; it was because the indoor air temperature never went down the lower limit of the thermostat dead band. However, IURec showed cycling variations, similar to temperature variations of System1 zones. The corresponding indoor unit air discharge temperatures of System2 are plotted in Figure 6.16.



Figure 6.15 : Indoor units operation of System2 zones in individual control mode on June 01, 2007

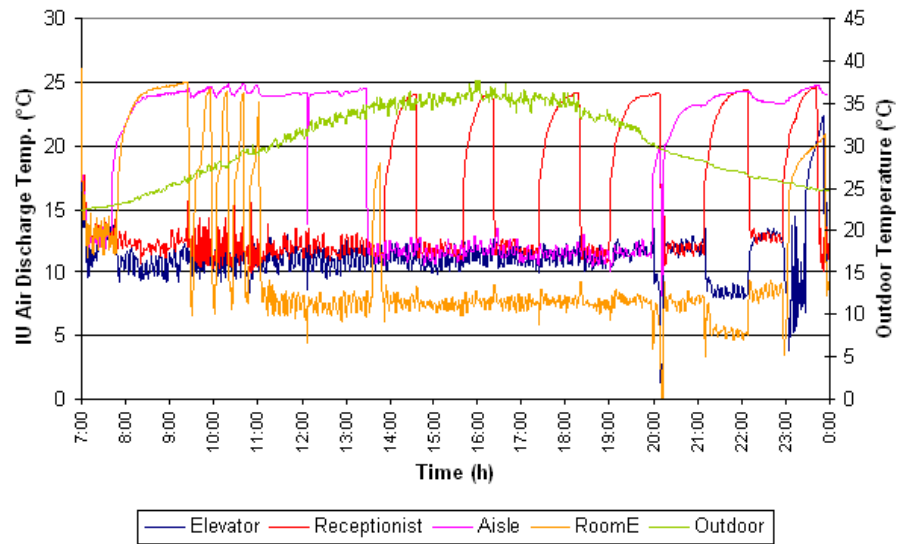


Figure 6.16 : Indoor unit air discharge temperatures of System2 on June 01, 2007

Figure 6.17 shows the relationship between IUREc operation and the EEV opening of IUREc, which was similar to IURA.

The relationship between the indoor unit operation and the EEV opening for IUEle, IUAI and IURE are provided in App. [F].

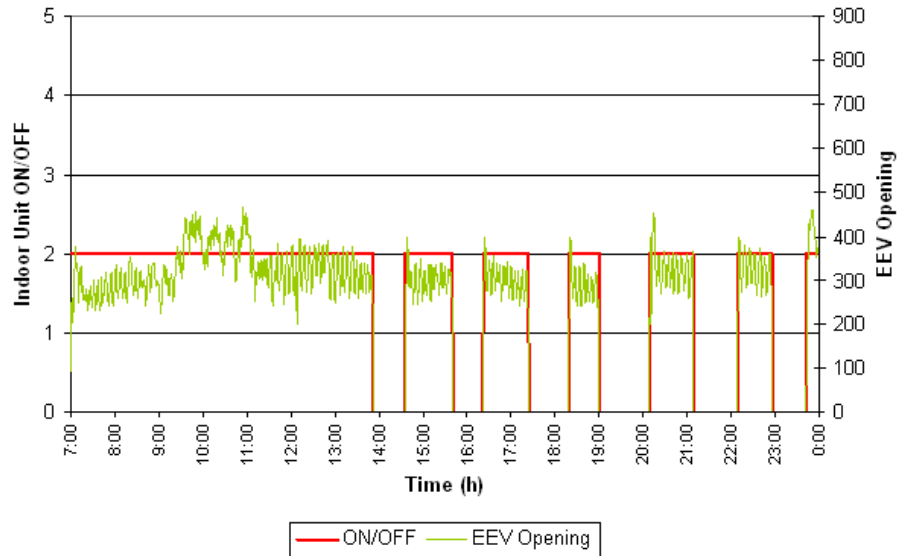


Figure 6.17 : Relationship between the operation and the EEV opening of IURec on June 01, 2007

Figure 6.18 shows the power consumptions of OU1 and OU2 during the same day (June 01, 2007).

As seen from Figure 6.18, the power consumption of OU2 was higher than that of OU1, which had cycling operation throughout the day. The difference came from the demand of the longer indoor unit operation for OU2 zones compared to the OU1 zones.

It can be seen from Figure 6.18 that when the outdoor unit was not working, the outdoor unit still consumed around 70W. It was attributed to the power consumption of the controller inside the outdoor unit.

Figure 6.19 shows the “ON” time ratios of IDC and FSC for OU1 and OU2, and also the “OFF” time of the multi-split VRF system.

As can be seen from Figure 6.19, even though each outdoor unit was equipped with two compressors; IDC and FSC, both outdoor units were operated with IDC almost all time. That’s why; it can be assumed that the power consumption of the outdoor unit was due to the IDC operation.

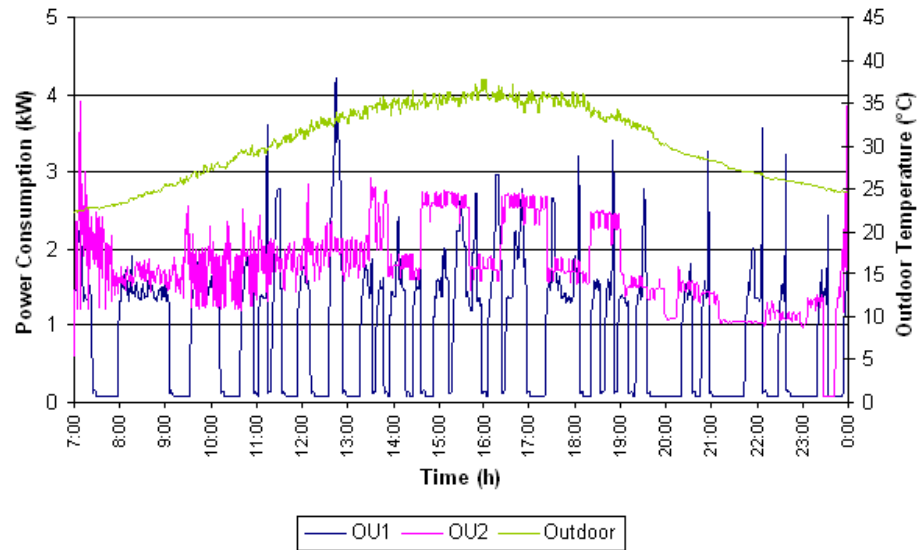


Figure 6.18 : Power consumption of OU1 and OU2 (June 01, 2007)

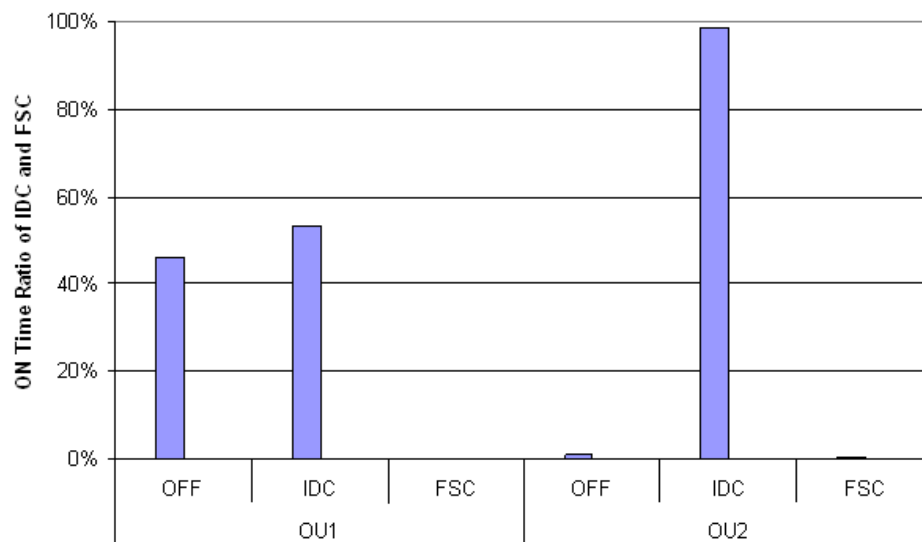


Figure 6.19 : ON time ratios of IDC and FSC of OU1 and OU2 on June 01, 2007

Figure 6.20 shows the relationship between the power consumption of OU1 and the indoor unit operation for the OU1 zones. As expected, when the indoor unit or units were “ON”, the outdoor unit started working and when all indoor units were “OFF”, the outdoor unit stopped. This situation can be seen from the period of 8:00–11:00.

Figure 6.21 shows the period of 13:00–16:30 for the relationship between the power consumption of OU1 and the indoor unit operation for System1 zones which is given in Figure 6.20. As seen from Figure 6.21, when there was only one indoor unit working, OU1 power consumption was almost 1.5kW. However, as seen at 14:05 or

15:45, when second indoor unit started working, OU1 power consumption increased to around 2.5kW because of the increasing demand of the refrigerant mass flow rate.

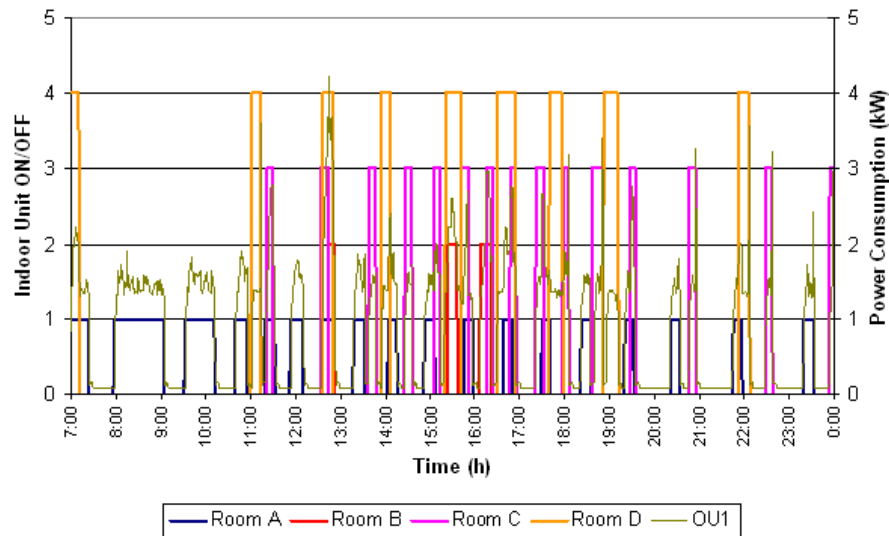


Figure 6.20 : Relationship between the operation of System1 indoor units and OU1 power consumption on June 01, 2007

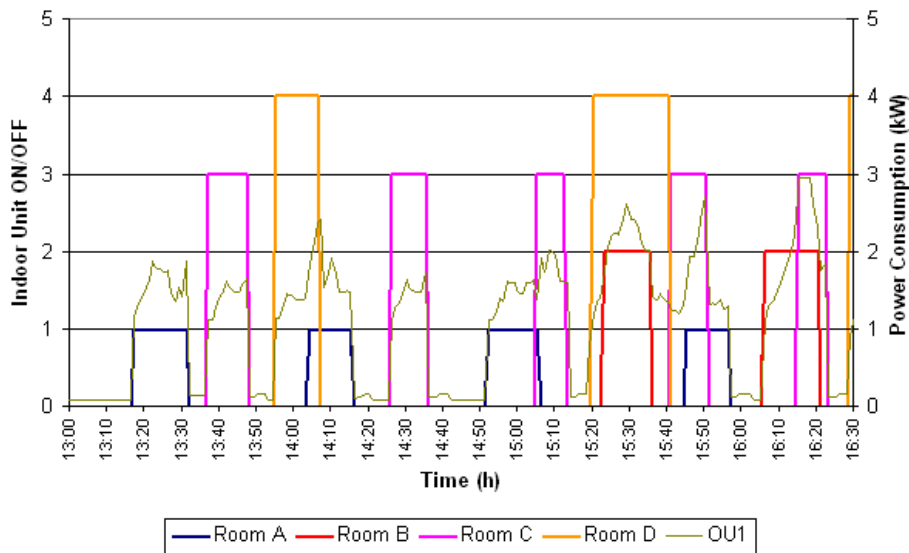


Figure 6.21 : Relationship between the operation of System1 indoor units and OU1 power consumption during the period of 13:00-16:30 on June 01, 2007

Figure 6.22 shows the period of 13:00-16:30 for the relationship between OU1 power consumption and the inverter frequency of IDC.

As seen from Figure 6.21 and Figure 6.22, at 14:05 or 15:45, the inverter frequency increased when the second indoor unit started working and this resulted higher power consumption for the outdoor unit.

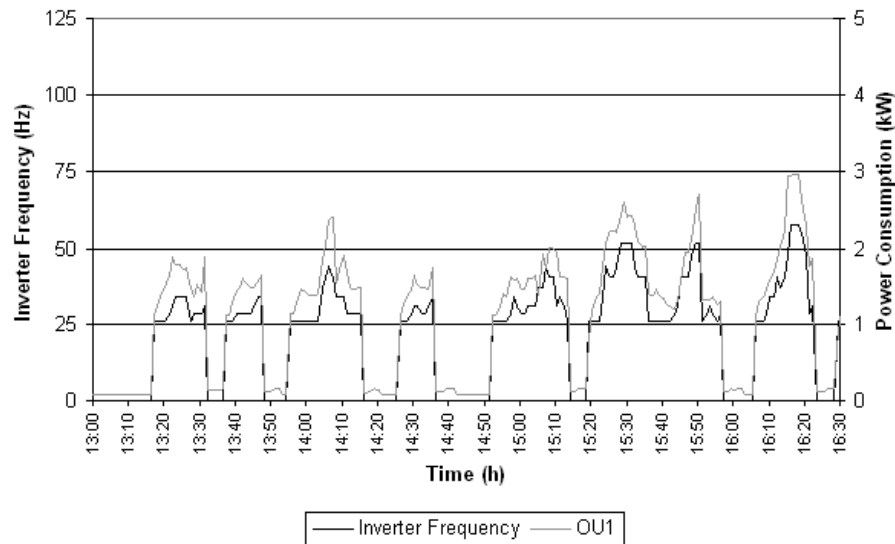


Figure 6.22 : Relationship between the inverter frequency of IDC and OU1 power consumption during the period of 13:00-16:30 on June 01, 2007

Figure 6.23 shows the condensing and evaporating pressures and the power consumption of OU1. As can be seen from Figure 6.23, since the outdoor unit operated cyclically, the condensing and the evaporating pressures fluctuated accordingly. During the “ON” time of the outdoor unit, the evaporating pressure was tried to be kept constant around 1MPa, which is a similar result found from (Xia et al., 2002).

Figure 6.24 shows the refrigerant temperature of IDC suction, condenser inlet, condenser outlet, and IDC discharge of OU1. Similar to the refrigerant pressures, the refrigerant temperatures fluctuated according to the cyclical operation.

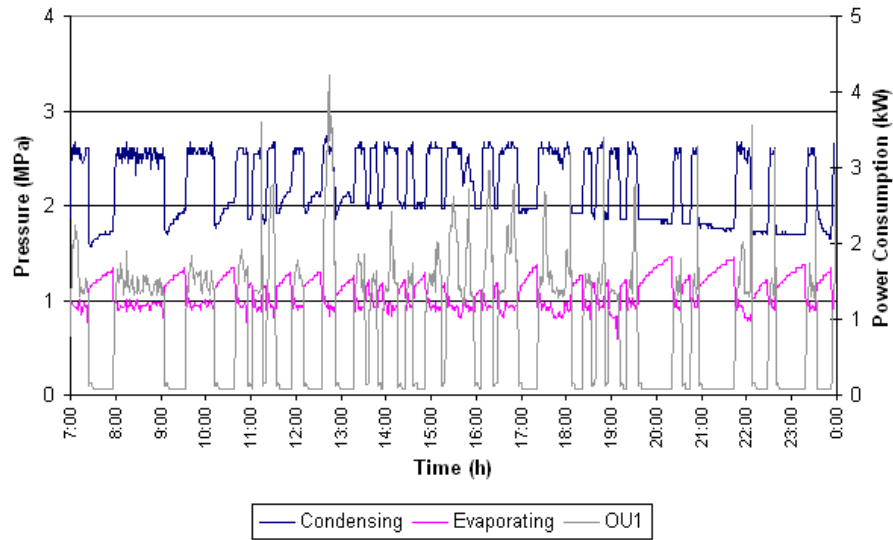


Figure 6.23 : Condensing and evaporating pressures of OU1 on June 01, 2007

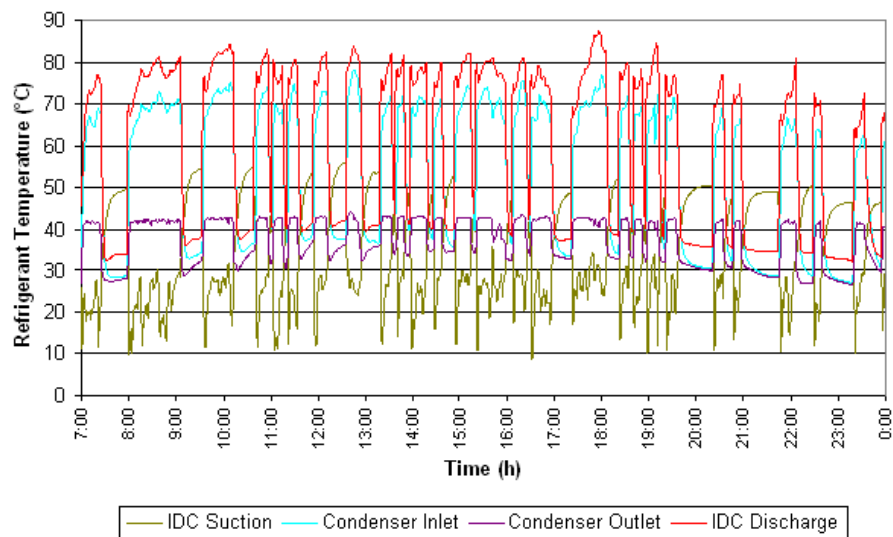


Figure 6.24 : Refrigerant temperatures of IDC suction, condenser inlet, condenser outlet, and IDC discharge for OU1 on June 01, 2007

Figure 6.25 shows the relationship between the refrigerant mass flow rate discharged from the compressor and the inverter frequency of IDC for OU1. As can be seen from Figure 6.25, as expected, refrigerant flow rate and the inverter frequency of IDC had similar trends. Since IDC operated more than 99.5% of all time during the tests, it was the main source for the refrigerant mass flow rate.

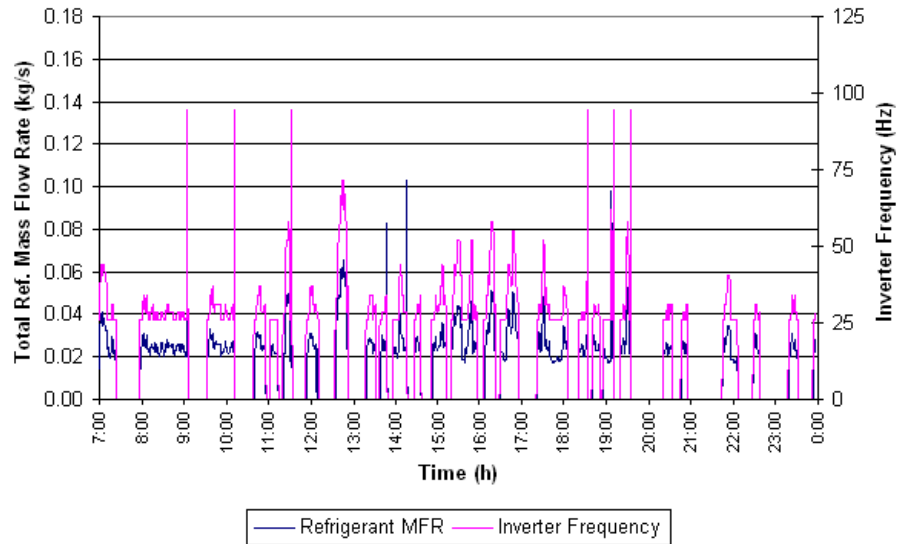


Figure 6.25 : Relationship between the inverter frequency of IDC and the total refrigerant mass flow rate for OU1 on June 01, 2007

Figure 6.26 depicts the total refrigerant mass flow rate discharged from the outdoor unit and the individual refrigerant mass flow rates of OU1 indoor units.

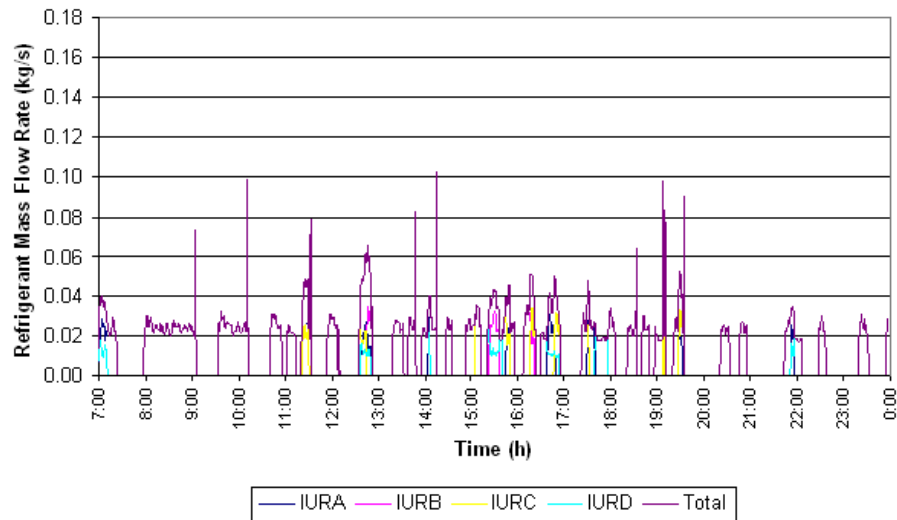


Figure 6.26 : Total and the individual refrigerant mass flow rates of System1 indoor units (June 01, 2007)

Figure 6.27 shows the relationship between OU2 power consumption and the indoor unit operation. As expected, since System2 indoor unit operation time was longer than System1 indoor unit operation time, OU2 power consumption was higher than OU1. Besides, there was not any cyclic operation except the one at around 23:45.

Similar to OU1 results, it can be seen from Figure 6.28, when three indoor units were working, the power consumption increased to around 2.5kW, and when one of indoor units was turned off, the power consumption decreased to around 1.5kW.

Figure 6.29 shows the relationship between OU2 power consumption and the inverter frequency of IDC. As seen, the power consumption of OU2 followed the same trend of inverter frequency of IDC.

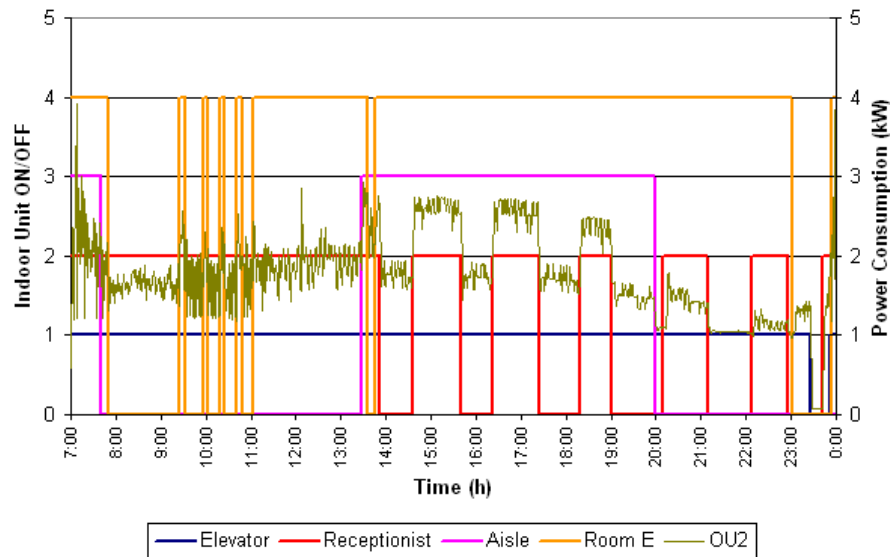


Figure 6.27 : Relationship between the operation of System2 indoor units and OU2 power consumption on June 01, 2007

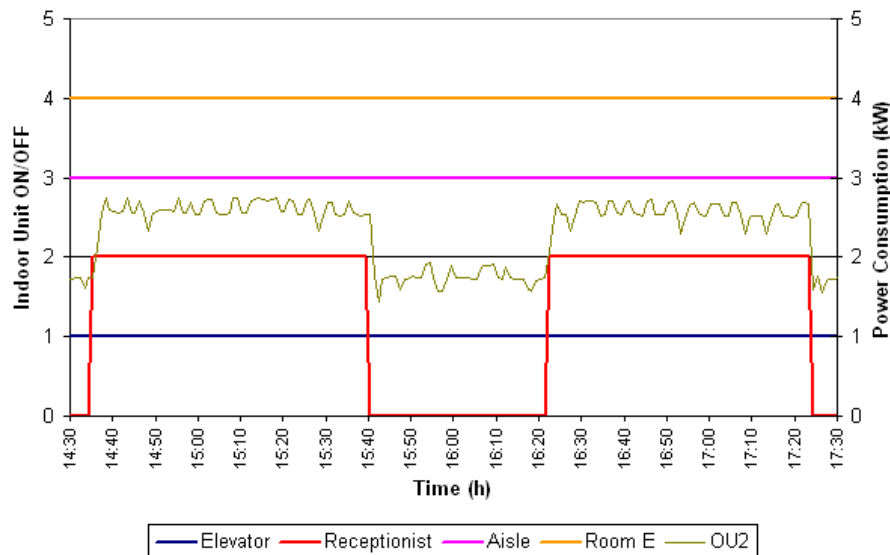


Figure 6.28 : Relationship between the operation of System2 indoor units and OU2 power consumption during the period of 14:30-17:30 on June 01, 2007

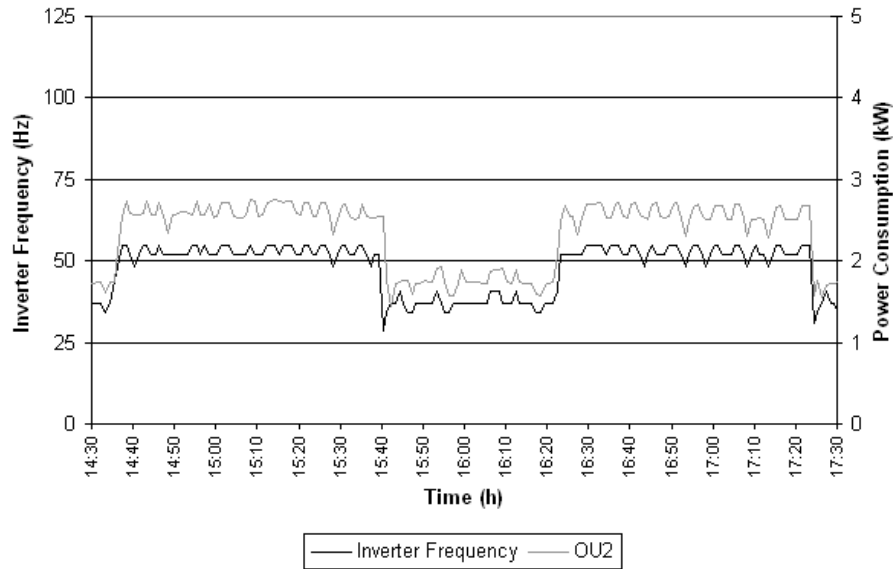


Figure 6.29 : Relationship between the inverter frequency of IDC and OU2 power consumption during the period of 14:30-17:30 on June 01, 2007

Figure 6.30 shows the condensing and evaporating pressures and the power consumption of OU2. As can be seen from Figure 6.30, since the outdoor unit operated continuously almost all the test time, the evaporating pressure was kept constant at around 1MPa. Since the multi-split VRF system operated in the cooling mode, the evaporating pressure was important, and that's why it was tried to be kept constant for both outdoor units.

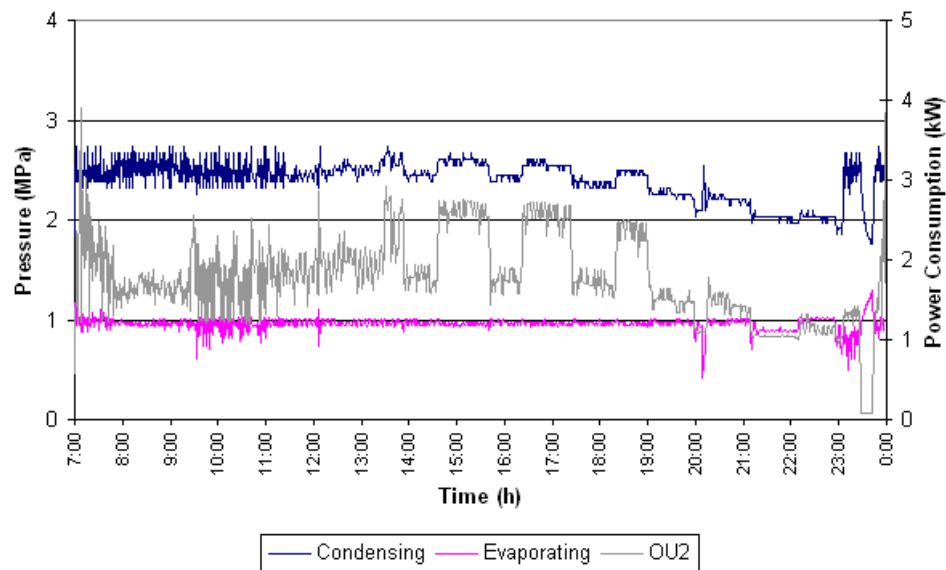


Figure 6.30 : Condensing and evaporating pressures of OU2 on June 01, 2007

Figure 6.31 shows the refrigerant temperature of IDC suction, condenser inlet, condenser outlet, and IDC discharge of OU2.

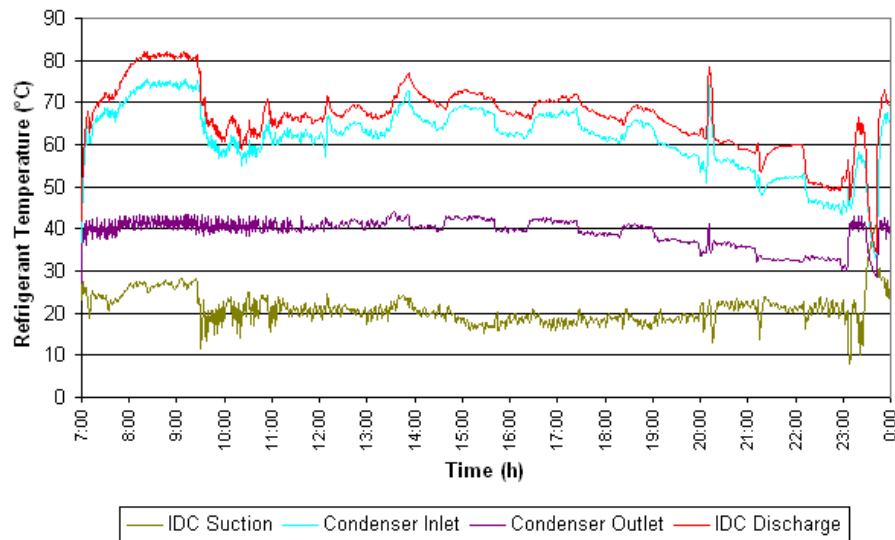


Figure 6.31 : Refrigerant temperatures of IDC suction, condenser inlet, condenser outlet, and IDC discharge of OU2 on June 01, 2007

Figure 6.32 shows the relationship between the refrigerant mass flow rate discharged from the outdoor unit and the inverter frequency of IDC for OU2. Similar to OU1, the refrigerant flow rate and the inverter frequency had similar trends.

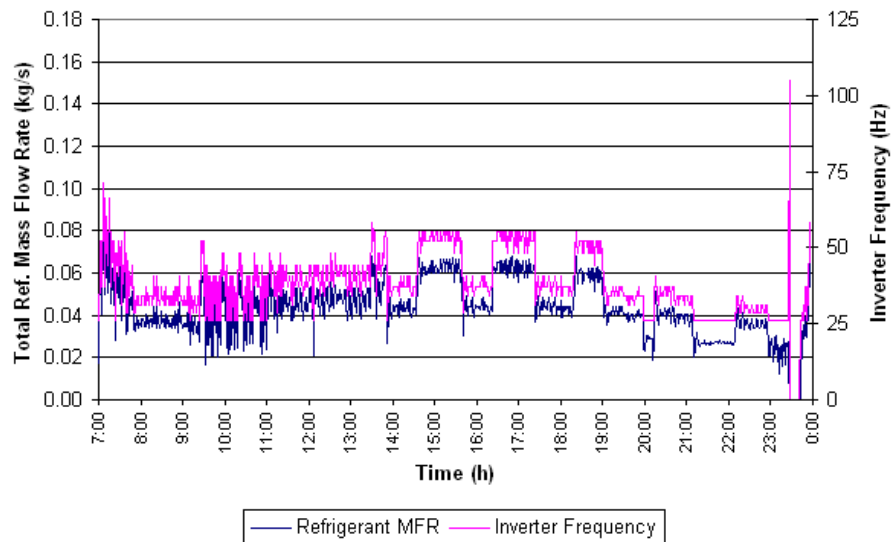


Figure 6.32 : Relationship between the total refrigerant mass flow rate and the inverter frequency of IDC for OU2 on June 01, 2007

Figure 6.33 depicts the total and the individual refrigerant mass flow rates of System2 indoor units. Since System2 zones required more cooling than System1

zones, the individual refrigerant mass flow rates of System2 indoor units were higher than that of System1 zones.

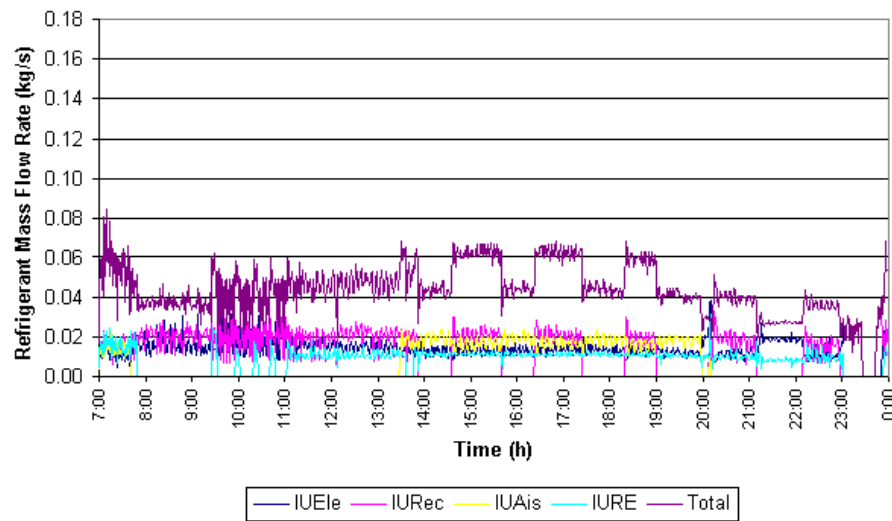


Figure 6.33 : Total and the individual refrigerant mass flow rates of System2 indoor units on June 01, 2007

Figure 6.34 and Figure 6.35 depict the indoor temperature variations of System1 and System2 zones on May 31, 2007, respectively. Multi-split VRF system was operated in master control mode.

Since the master control mode controlled all indoor units with one thermostat, all of indoor units worked and stopped at the same time. It can be seen from Figure 49 and Figure 50 that, the indoor temperatures of System1 and System2 zones decreased or increased simultaneously.

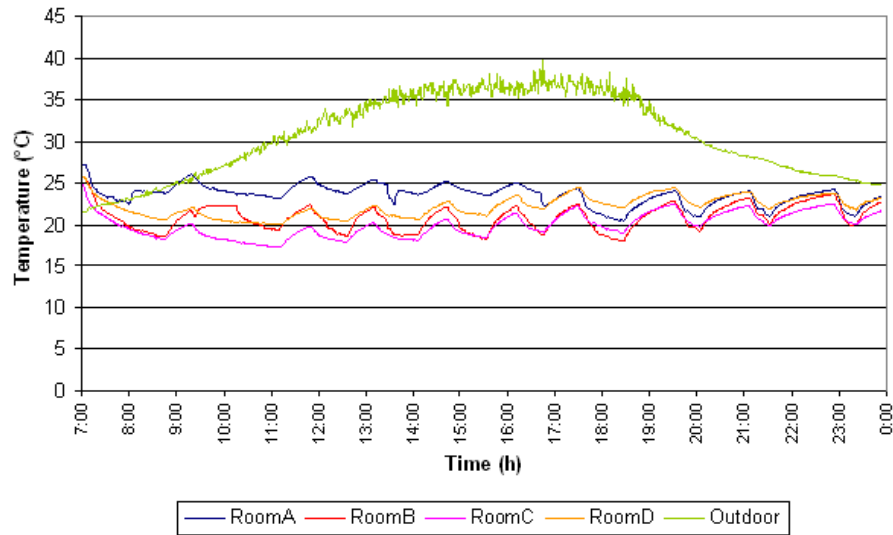


Figure 6.34 : Indoor temperature of System1 zones and outdoor temperature variations for master control mode on May 31, 2007

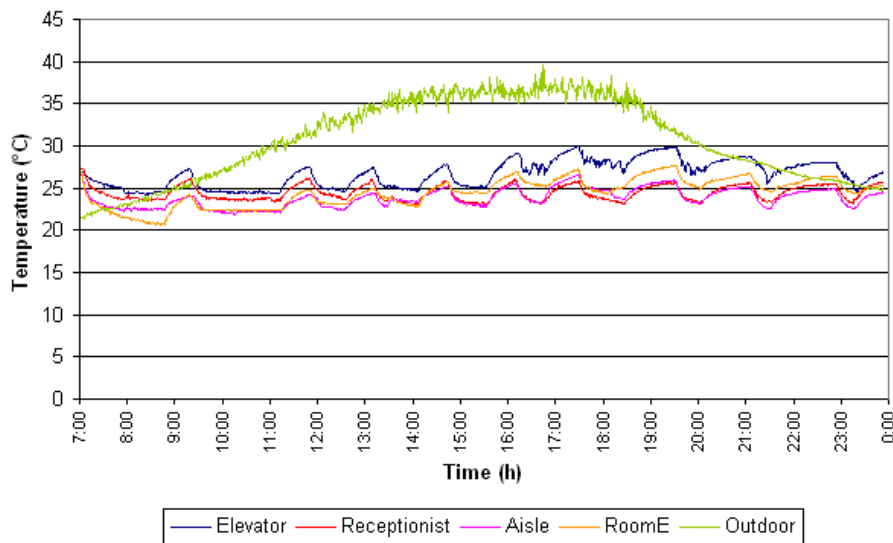


Figure 6.35 : Indoor temperatures of System2 zones and outdoor temperature on May 31, 2007

Figure 6.36 and Figure 6.37 show the corresponding TSS variations. As seen, TSS varied within neutral and cool region for System1 zones and varied within slightly cool and slightly warm region for System2 zones. Unlike individual control mode, shown in Figure 6.9 and Figure 6.11, TSS fluctuated a lot due to the master control mode.

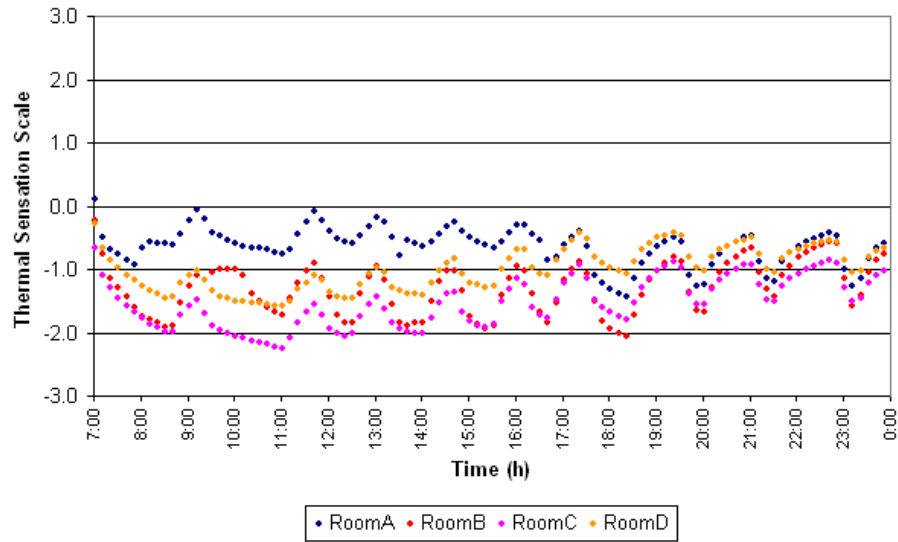


Figure 6.36 : TSS variation of System1 zones on May 31, 2007

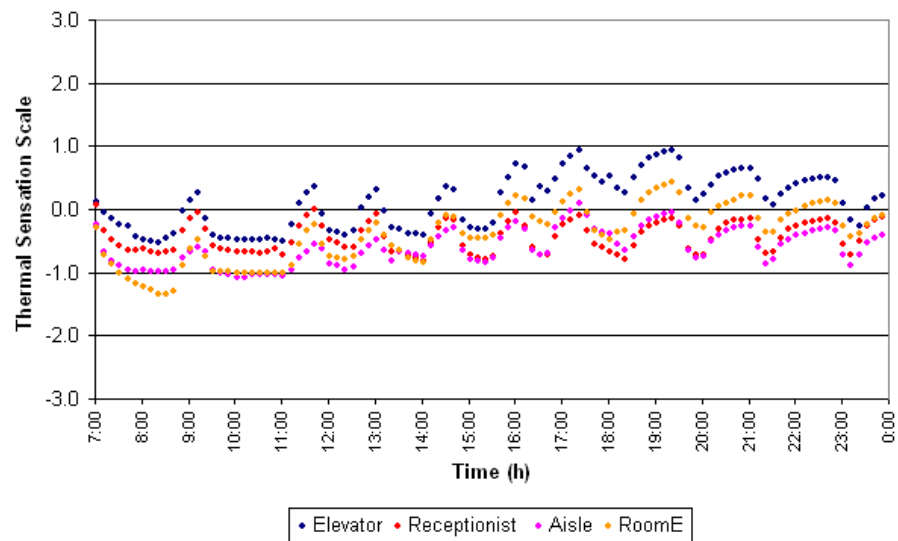


Figure 6.37 : TSS variation of System2 zones on May 31, 2007

Figure 6.38 shows System1 and System2 indoor units operation, respectively. As can be seen from Figure 6.38, all eight indoor units started and stopped working simultaneously. The corresponding indoor unit air discharge temperatures of System1 and System2 are shown in Figure 6.39.

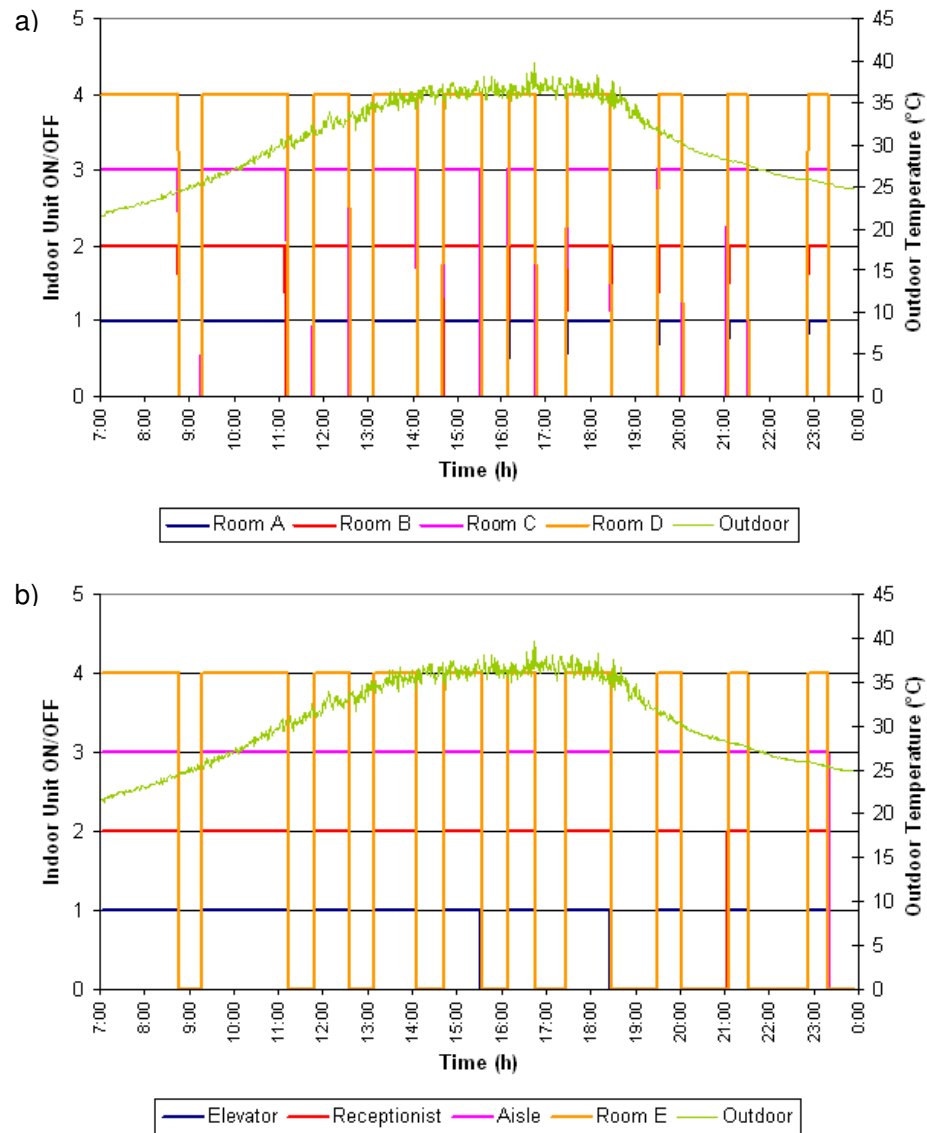


Figure 6.38 : Indoor units operation of System1 and System2 zones in master control mode on May 31, 2007, (a) System1, (b) System2

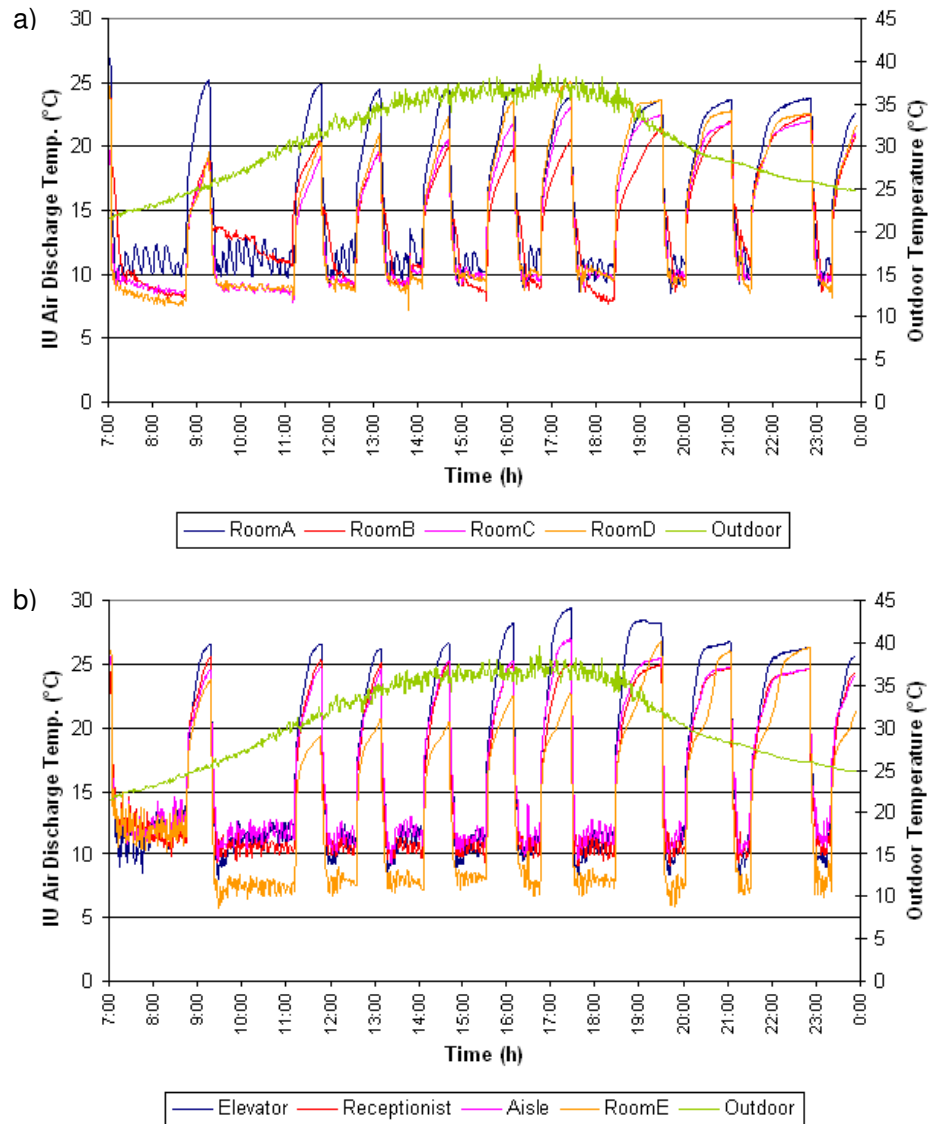


Figure 6.39 : Indoor unit air discharge temperatures of System1 and System2 zones in master control mode on May 31, 2007, (a) System1, (b) System2

Figure 6.40 shows the relationship between the operation of System2 indoor units and the EEV opening of System2 indoor units. As can be seen, EEVs of all System2 indoor units opened and closed simultaneously.

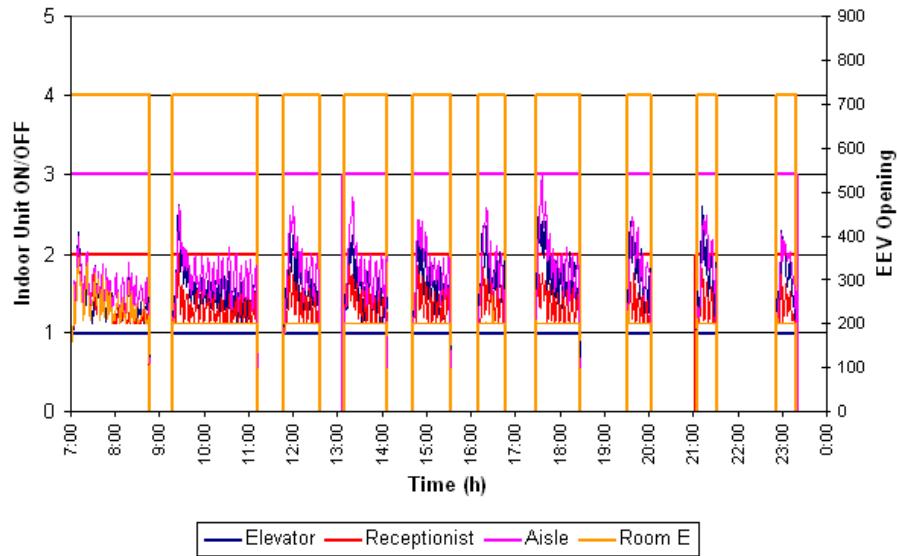


Figure 6.40 : Relationship between the operation and the EEV opening of System2 indoor units on May 31, 2007

Figure 6.41 shows the power consumptions of OU1 and OU2 during the test (May 31, 2007). As seen, the power consumption of OU1 was higher than OU2 at any time.

Since cooling capacities of System1 zones such as IURA and IURB were higher than that of System2 zones, the refrigerant demand for System1 indoor units was higher than System2 indoor units. The comparison of the refrigerant mass flow rate of System1 and System2 can be seen from Figure 6.43.

In order to satisfy the refrigerant demand, IDC of OU1 operated at higher frequencies than that of OU2. Because of these higher operating inverter frequencies, OU1 power consumption was higher than OU2 power consumption. The comparison of the inverter frequencies is shown in Figure 6.42. The peak values of the power consumption (when the system started working) were due to the peak operating inverter frequencies at the start-ups.

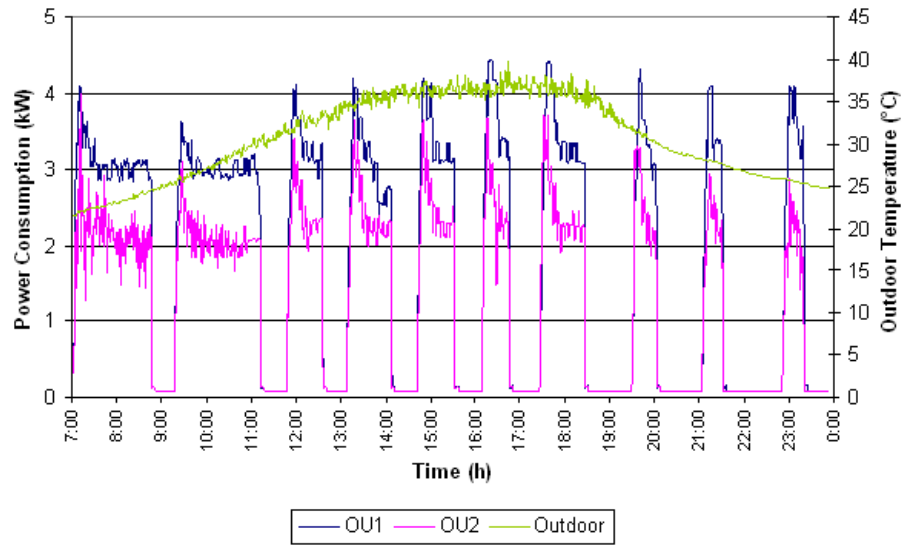


Figure 6.41 : Power consumption of OU1 and OU2 on May 31, 2007

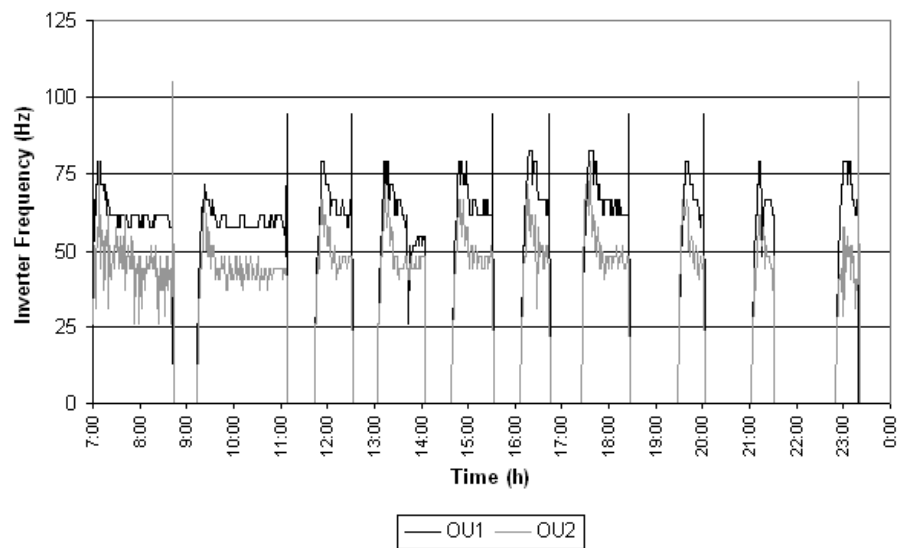


Figure 6.42 : Comparison of the inverter frequencies of IDC of OU1 and OU2

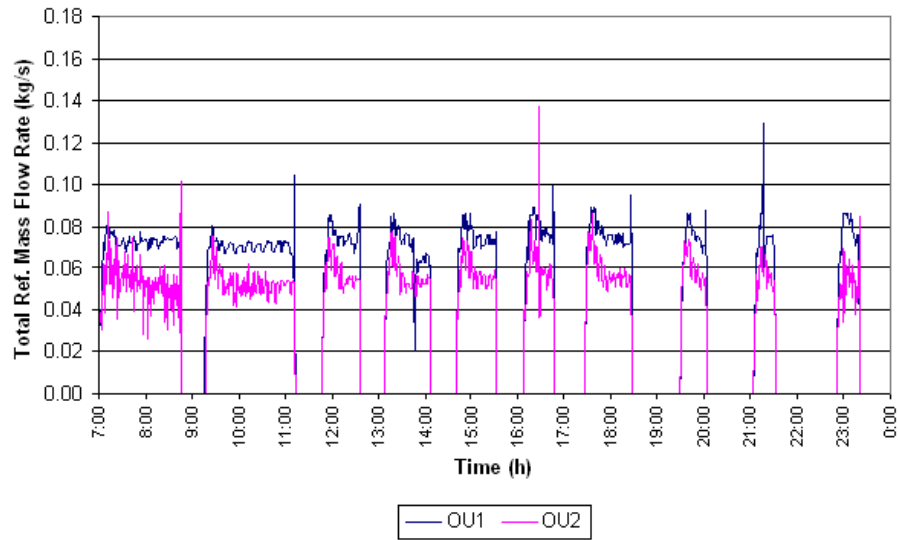


Figure 6.43 : Comparison of the total refrigerant mass flow rates of OU1 and OU2

Figure 6.44 and Figure 6.45 show the relationship between outdoor unit power consumptions and the operation of indoor units for System1 and System2, respectively.

Since four indoor units were operating simultaneously, the power consumption of the outdoor unit for the “ON” time was higher than the individual control mode shown in Figure 6.20 and Figure 6.27.

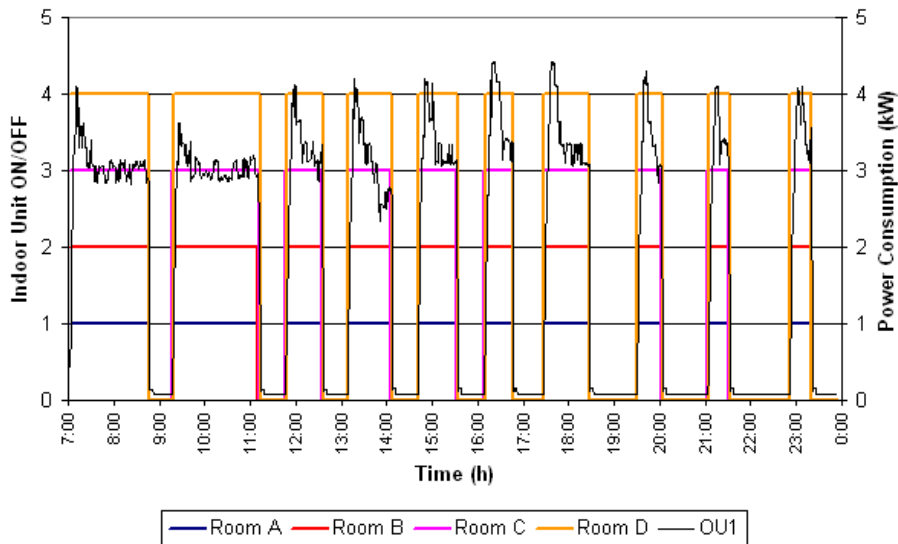


Figure 6.44 : Relationship between the operation and the EEV openings of System1 indoor units on May 31, 2007

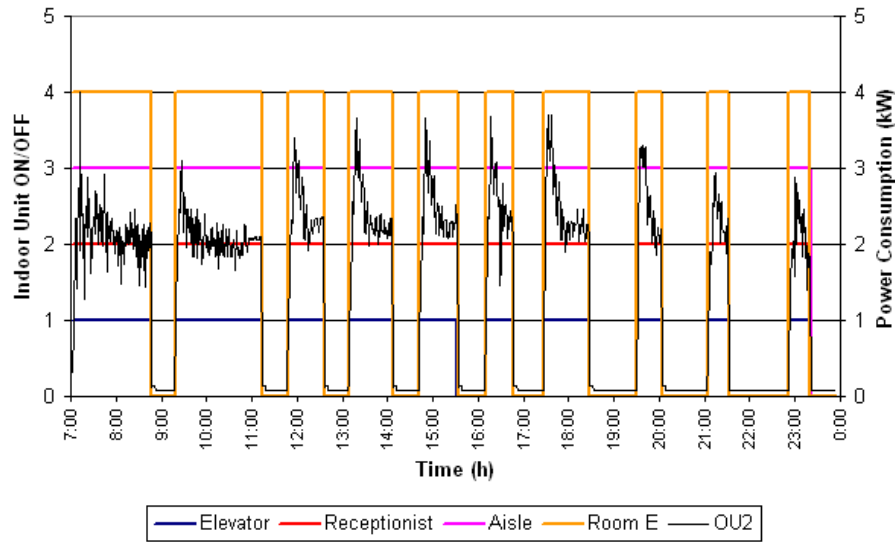


Figure 6.45 : Relationship between the operation and the EEV openings of System2 indoor units on May 31, 2007

Figure 6.46 shows the comparison of the condensing and evaporating pressures of OU1 and OU2. As can be seen, since master thermostat controlled both units, OU1 and OU2 operated simultaneously and during the “ON” time, similar to the individual control mode, the evaporating pressure was tried to be kept constant around 1MPa.

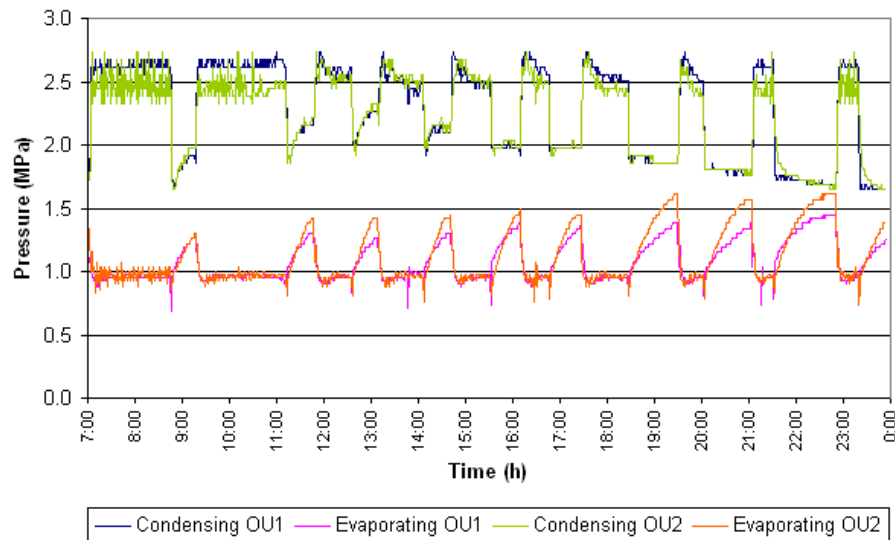


Figure 6.46 : Condensing and evaporating pressures of OU1 and OU2 on May 31, 2007

As described previously, the daily basis comparison of the individual and master control modes was performed for May 31, 2007 and June 01, 2007.

Figure 6.47 shows the effect of the control mode on the outdoor unit power consumption. The significant difference can be seen from Figure 6.47a for OU1.

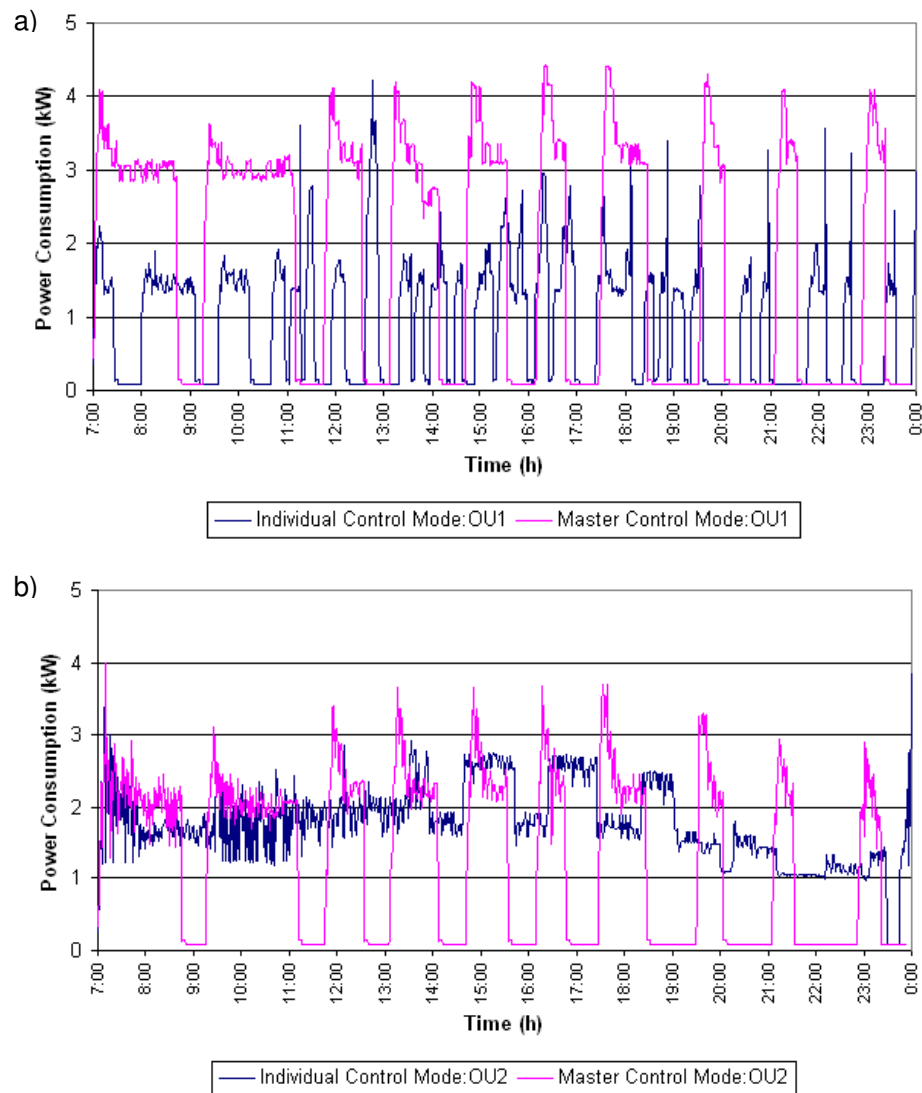


Figure 6.47 : Comparison of power consumption for the individual and master control modes, (a) OU1, (b) OU2

Figure 6.48 shows the comparison of the total power consumption of OU1 and OU2 for the individual and master control modes.

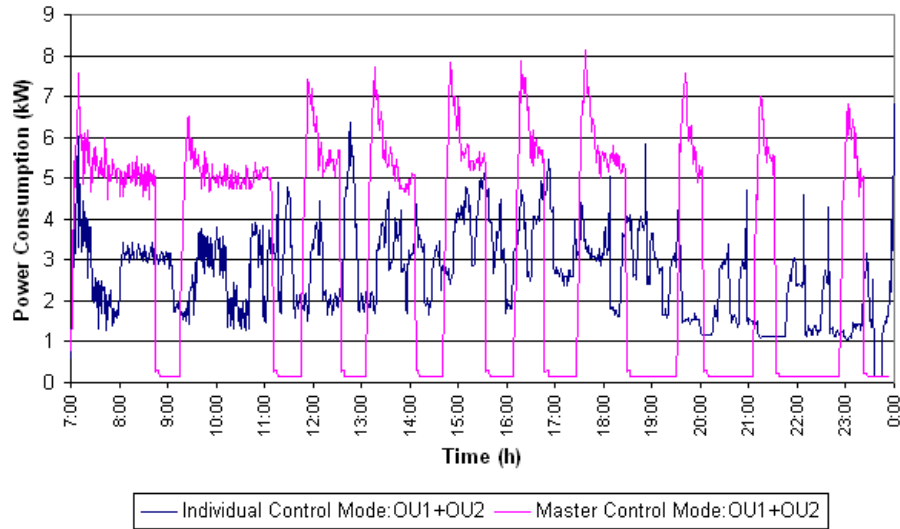


Figure 6.48 : Comparison of the total power consumption of OU1 and OU2 for the individual and master control modes

Table 6.6 summarizes the effect of control modes on the energy consumption of the outdoor units in terms of kWh.

Table 6.6 : Comparison of the energy consumption of the outdoor units for the individual and master control modes

Control Mode	Energy Consumption of OU1 (kWh)	Energy Consumption of OU2 (kWh)	Total Energy Consumption (OU1+OU2) (kWh)
Individual	15.37	30.26	45.63
Master	30.45	21.97	52.42

Considering OU1, it was found that the individual control mode consumed 49.5% lower energy than the master control mode, on the other hand, considering OU2, it was obtained that the master control mode consumed 27.4% lower energy than the individual control mode. Overall, the individual control mode consumed 12.9% lower energy than the master control mode.

Figure 6.49 shows the indoor condition data of System1 zones provided by the individual and master control modes based on the ASHRAE summer comfort zone with 10-minute averaged data. Different colors correspond to different zones, such as dark blue for Room A, red for Room B, etc.

As can be seen, the indoor temperatures provided by the individual control mode were much closer to the set temperature of 25°C than the indoor temperatures provided by the master control mode.

As can be seen from Figure 6.49b, the indoor temperatures of System1 zones provided by the master control mode were even lower than 20°C.

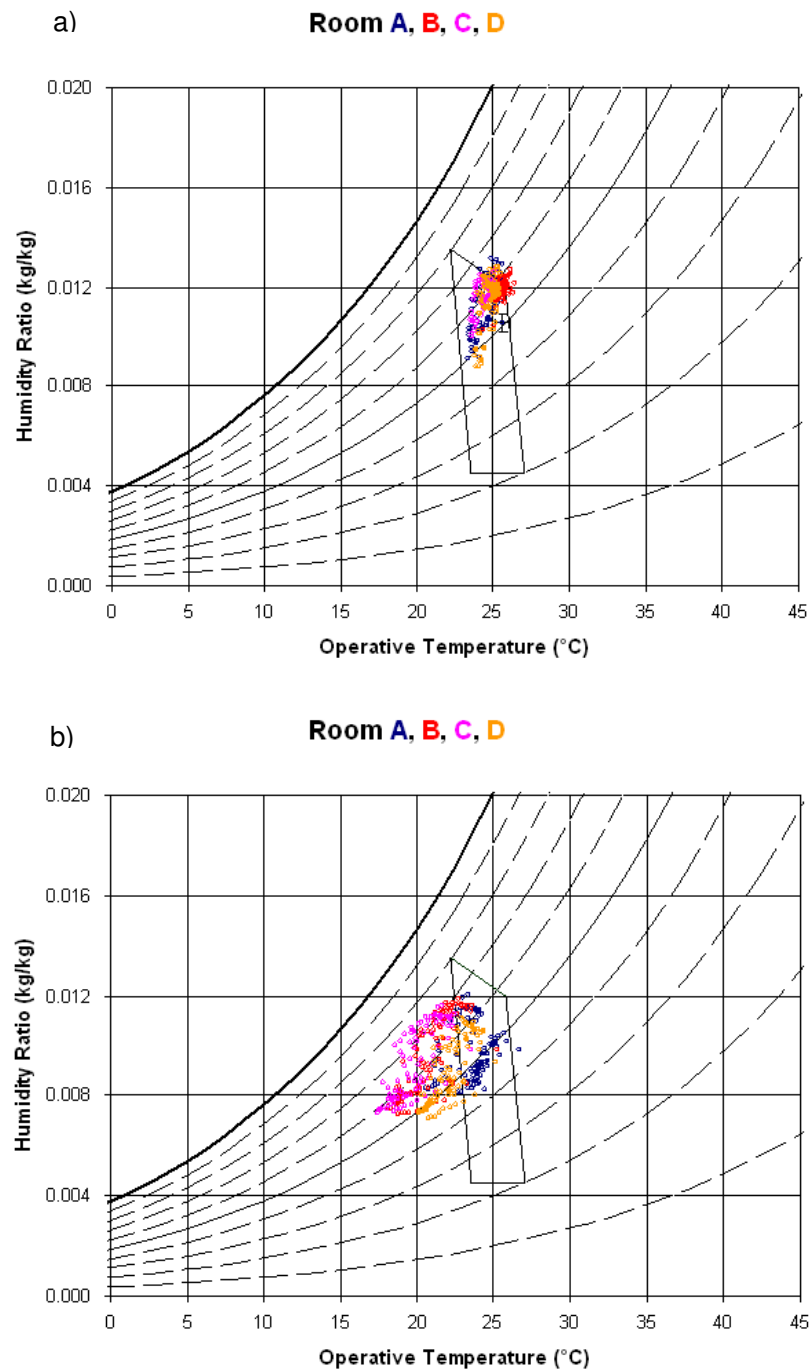


Figure 6.49 : Indoor conditions of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Table 6.7 shows the comparison of the indoor condition data of System1 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone.

Table 6.7 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone on June 01, 2007 and May 31, 2007

Region	Percentage of the indoor condition data for the individual Control Mode (June 01, 2007)	Percentage of the indoor condition data for the master Control Mode (May 31, 2007)
In	83.1%	31.8%
Cold	0.0%	67.9%
Hot	14.5%	0.3%
Humid	2.4%	0.0%
Dry	0.0%	0.0%

As can be seen from Table 6.7, the individual control mode provided better thermal comfort than the master control mode according to the ASHRAE summer comfort zone. Besides, it was observed that the master control mode provided a colder indoor environment than the individual control mode, which also caused an uncomfortable environment to the occupants.

Figure 6.50 shows the indoor conditions data of System2 zones provided by the individual and master control modes based on the ASHRAE summer comfort zone. As can be seen, similar to System1 zones, the indoor temperatures provided by the individual control mode were much closer to the set temperature of 25°C than the indoor temperatures provided by the master control mode.

As can be seen from Figure 6.50b, the indoor temperatures provided by the master control mode even went up to 30°C for System2 elevator zone.

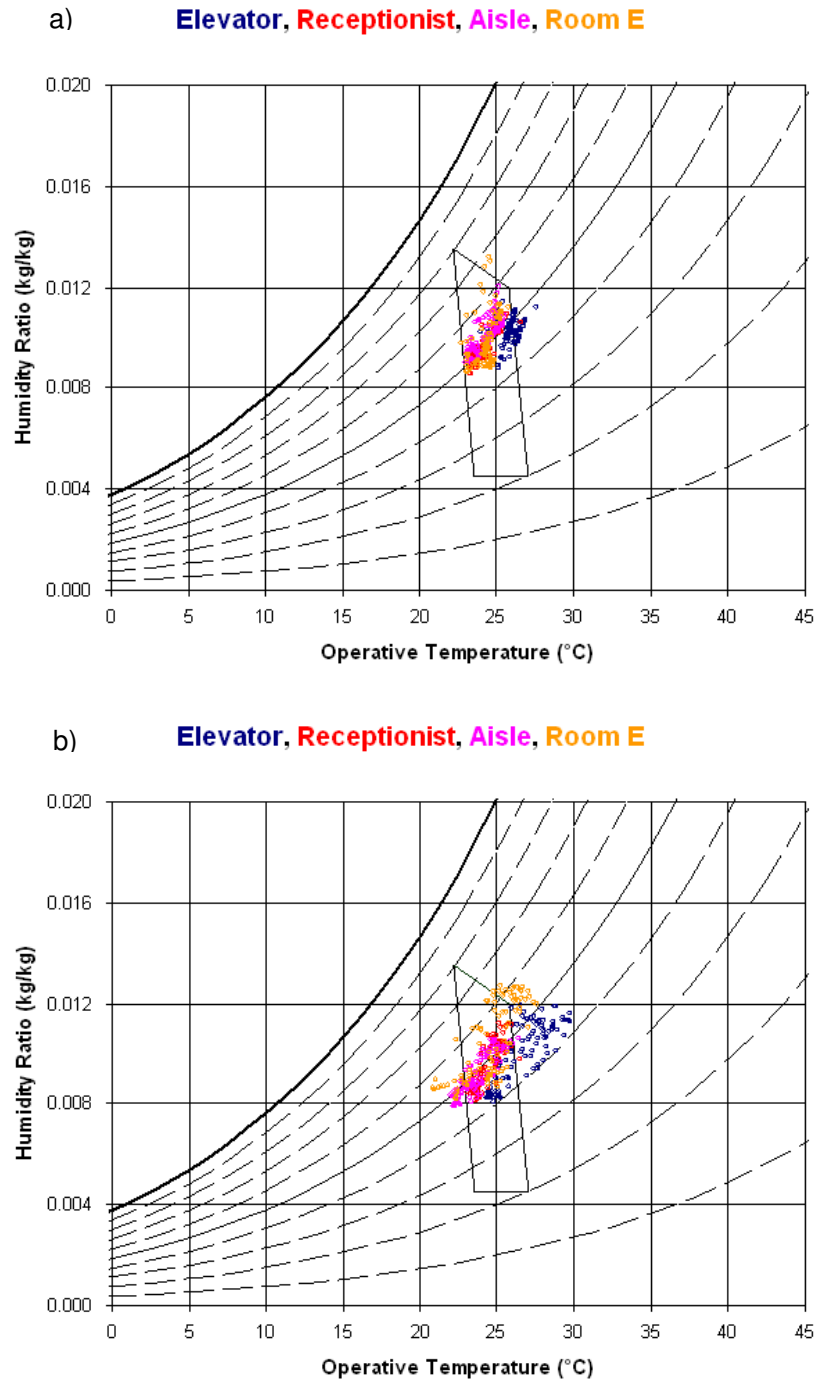


Figure 6.50 : Indoor conditions of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Table 6.8 shows the comparison of the indoor condition data of System2 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone.

Table 6.8 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone on June 01, 2007 and May 31, 2007

Region	Percentage of the indoor condition data for the individual Control Mode (June 01, 2007)	Percentage of the indoor condition data for the master Control Mode (May 31, 2007)
In	89.3%	65.7%
Cold	0.2%	12.0%
Hot	9.8%	16.7%
Humid	0.7%	5.6%
Dry	0.0%	0.0%

As can be seen from Table 6.8, similar to System1 zones, the individual control mode provided better thermal comfort than the master control mode for System2 zone and the master control mode provided a colder indoor environment than the individual control mode.

Thermostat temperature measurement locations were used in the previous evaluations, shown in Figure 6.49 and Figure 6.50. Since the occupant and the thermostat locations were different, the temperature of the occupant location was investigated for the same tests.

Figure 6.51 shows the indoor temperature variations for the ASHRAE thermal comfort measurement locations in Room C for the individual and master control modes. As can be seen for both control modes, during the “ON” time of the indoor unit, as expected, the temperature of the zone was decreasing; however it was found that the temperature of the thermostat location was higher than that of the occupant location (0.1m and 0.6m) at all time. The temperature fluctuations of location 0.6m were due to the operation of the mini refrigerator shown in Figure 4.24.

In spite of the temperature deviation between the thermostat and occupant locations, the temperature of the occupant location provided by the individual control mode was much closer to the set temperature than that of the occupant location provided by the master control mode. That’s why; it can be said that the individual control mode provided much better thermal comfort than the master control mode based on both thermostat and occupant locations.

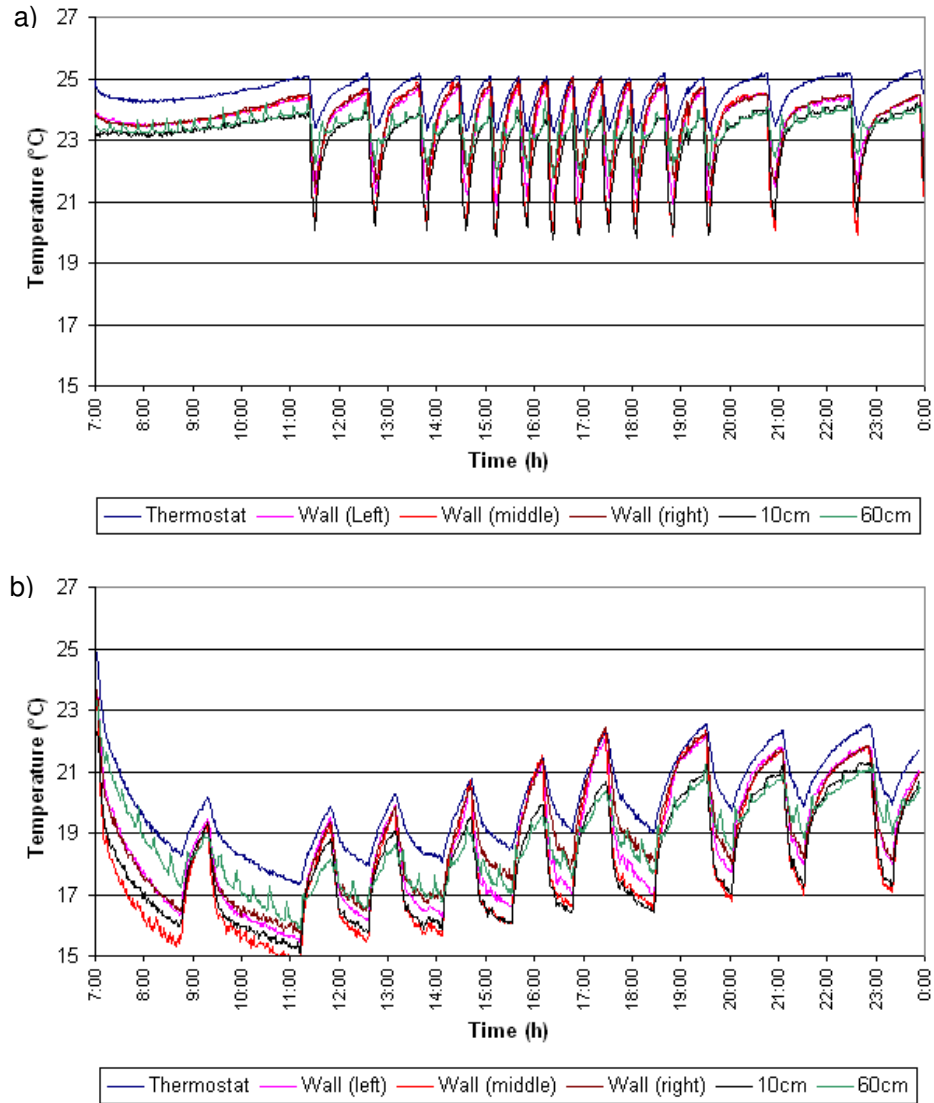


Figure 6.51 : Indoor temperature variations of the ASHRAE thermal comfort locations for the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.52a and Figure 6.52b show the TSS variations provided by the individual and master control modes for System1 zones, respectively.

As can be seen from Figure 6.52a, the TSS variations provided by the individual control mode were much closer to the neutral point than the TSS variations provided by the master control mode.

Not only the absolute value of TSS, but also the percentage is important. The absolute value of TSS totally depends on the temperature as shown in Figure 6.10 and accordingly to the set temperature, and indicates the indoor environment condition; such as slightly cool or cold etc. On the other hand, the percentage shows

whether the system can maintain the indoor temperature at the set temperature. The higher the percentage of TSS means the better the indoor temperature control and the less the indoor temperature fluctuations.

As seen from Figure 6.52a, more than 60% of all data for Rooms B and C and more than 70% of all data for Rooms A and D for the individual control mode were found around neutral point, which means that the indoor temperature can be maintained with less fluctuations throughout the test time. On the other hand, as seen from Figure 6.52b, the percentage of the TSS variations provided by the master control mode were lower than the individual control mode, which indicated that the indoor temperature can not be maintained by the master control mode.

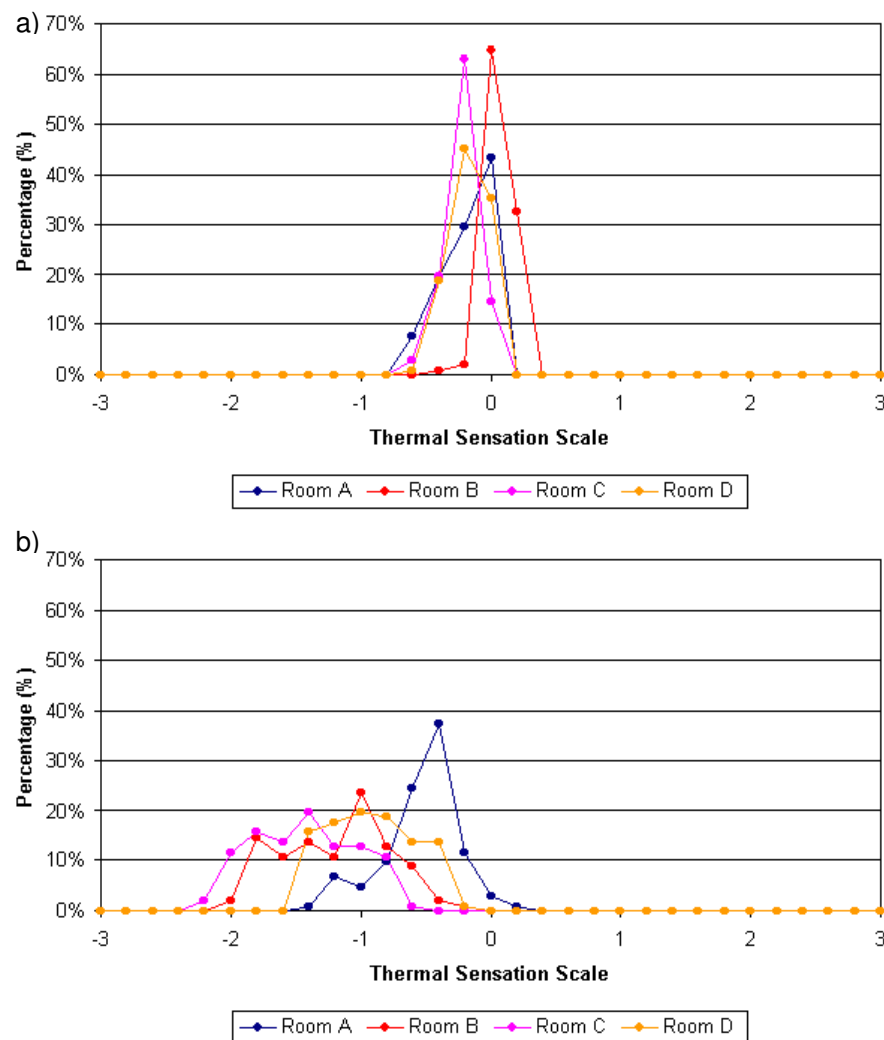


Figure 6.52 : TSS variations of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.53a and Figure 6.53b show the TSS variations provided by the individual and master control modes for System2 zones, respectively.

Similar to System1 results, the TSS variations provided by the individual control mode were much closer to the neutral point than the master control mode. Besides, the percentages of the TSS provided by the individual control mode around the neutral point were higher than that of the TSS provided by the master control mode.

Since the window area for elevator zone was larger than the other zones and had the smallest capacity of indoor unit, the indoor temperature and corresponding TSS variation was slightly higher than the other zones. On the other hand, as seen from Figure 6.53b, TSS varied mostly within slightly cool and slightly warm region for the master control mode.

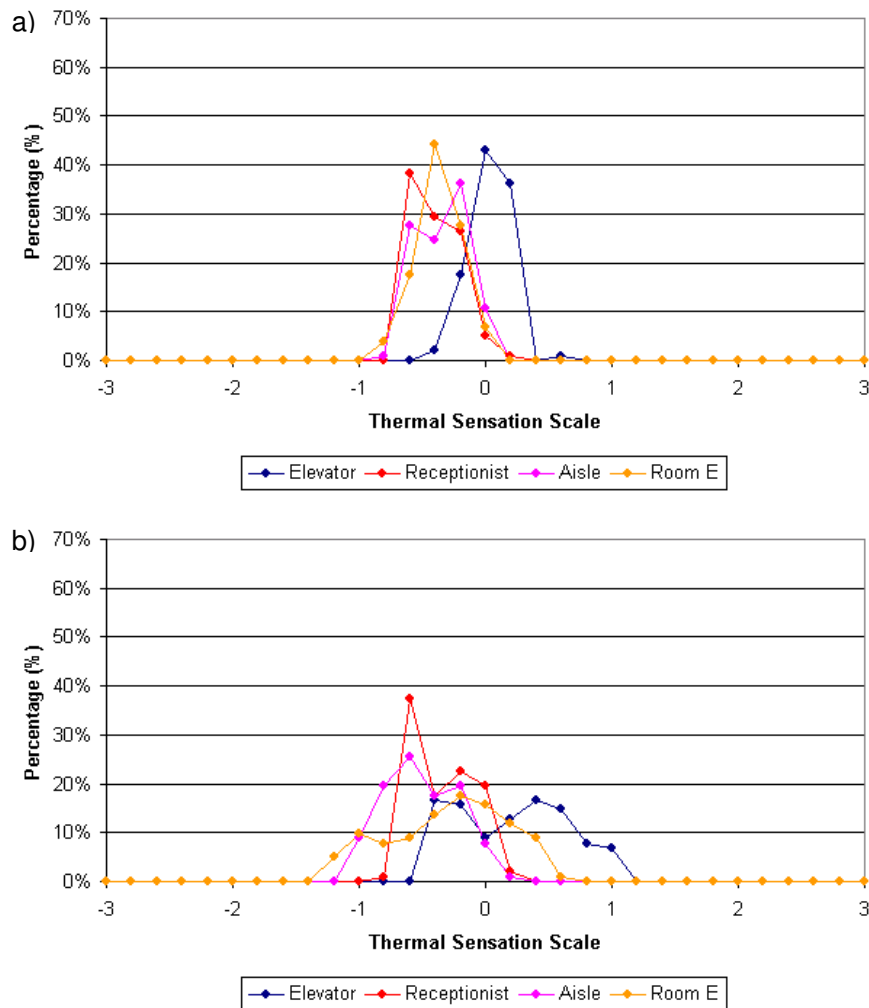


Figure 6.53 : TSS variations of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.54a and Figure 6.54b show the TSS variation of the combined zones (considering all four zones as a group) for System1 and System2 for the individual and master control modes, respectively.

As can be seen, the individual control mode provided much better thermal comfort environment than the master control mode.

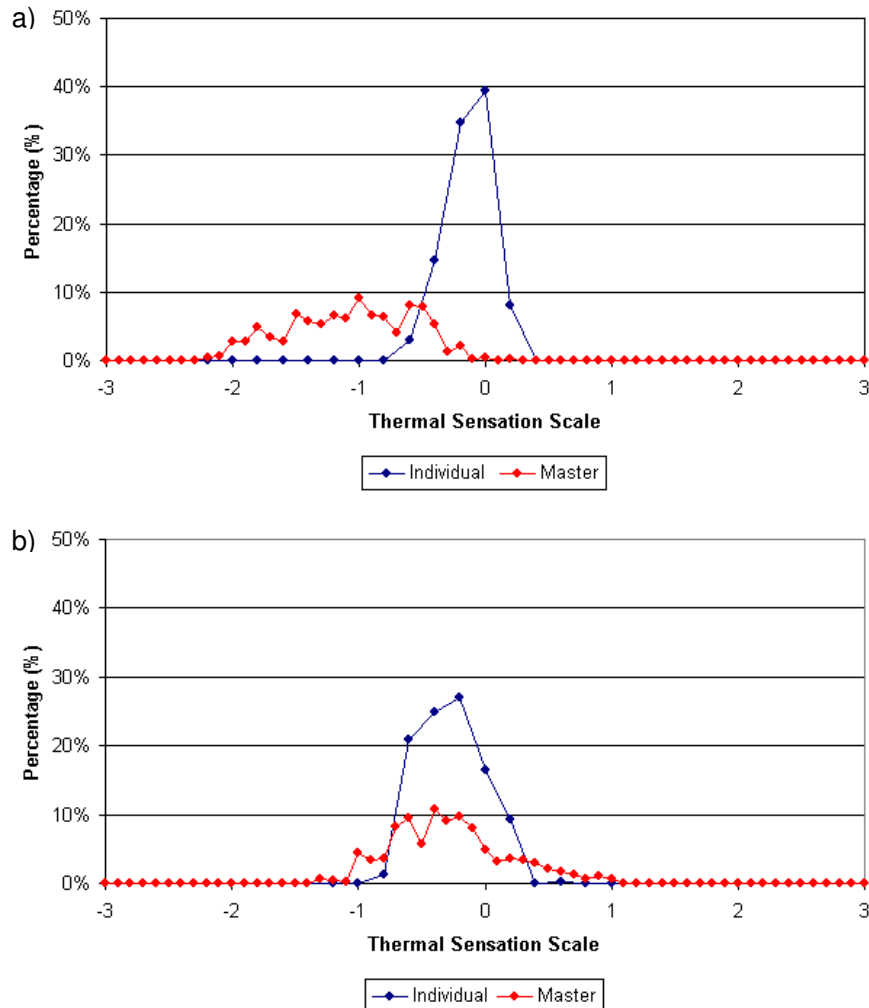


Figure 6.54 : TSS variations of System1 and System2 combined zones provided by the individual and master control modes, (a) System1, (b) System2

The variation of the cooling capacity, the power consumption of the outdoor unit, the total power consumption of the indoor units and the CPF of System1 and System2 for the individual control mode test on June 01, 2007 can be seen from Figure 6.55a and Figure 6.55b, respectively. The data was provided with 10-minute averaged.

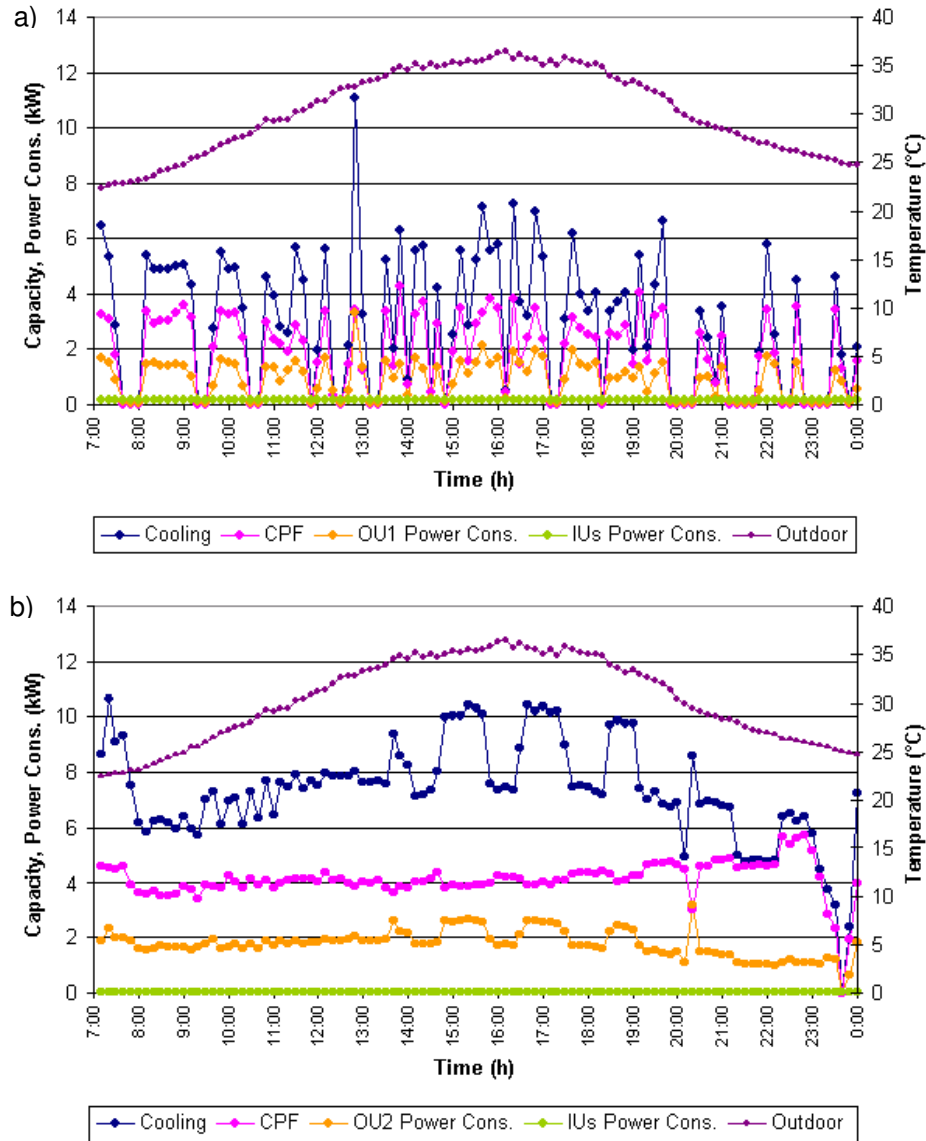


Figure 6.55 : Variation of the total cooling capacity, outdoor unit power consumption, total power consumption of the indoor units and CPF of System1 and System2 for the individual control mode on June 01, 2007, (a) System1, (b) System2

As can be seen from Figure 6.55a and Figure 6.55b, variations of the total cooling capacity and the power consumption of the outdoor unit followed similar trends. Since the IDC operated almost all of the test time, shown in Figure 6.19, the total refrigerant mass flow rate totally depended on the frequency of the IDC. That's why; the cooling capacity mostly depended on the frequency of the IDC.

The cyclic operation of System1 was due to the relatively low cooling load and oversized multi-split VRF system. Since OU1 had cyclic operation throughout the test time, efficiency comparisons were performed based on OU2.

As can be seen from Figure 6.55b, the variation of the total cooling capacity followed the variation of the outdoor temperature profile, except the first hour of the test time. The reason of the high cooling capacity for the first hour was due to the relatively high initial indoor temperatures. After reaching the indoor temperatures to the set temperature, the cooling capacity followed the outdoor temperature profile. As can be seen, the higher cooling capacities were provided within 15:00–19:00 for System2 zones.

The total power consumption of System1 and System2 indoor units were around 190W and 52W, respectively. Since the indoor units were operated continuously throughout the test time with the same air flow rate, the power consumptions were constant.

The variation of the total cooling capacity, the power consumption of the outdoor unit, the total power consumption of the indoor units and CPF of OU1 and OU2 zones for the master control mode test on May 31, 2007 can be seen from Figure 6.56a and Figure 6.56b, respectively. The data was provided with 10-minute averaged.

As can be seen from Figure 6.56a and Figure 6.56b, similar to the individual control mode, the variation of the total cooling capacity and the power consumption of the outdoor unit of the master control mode followed similar trends.

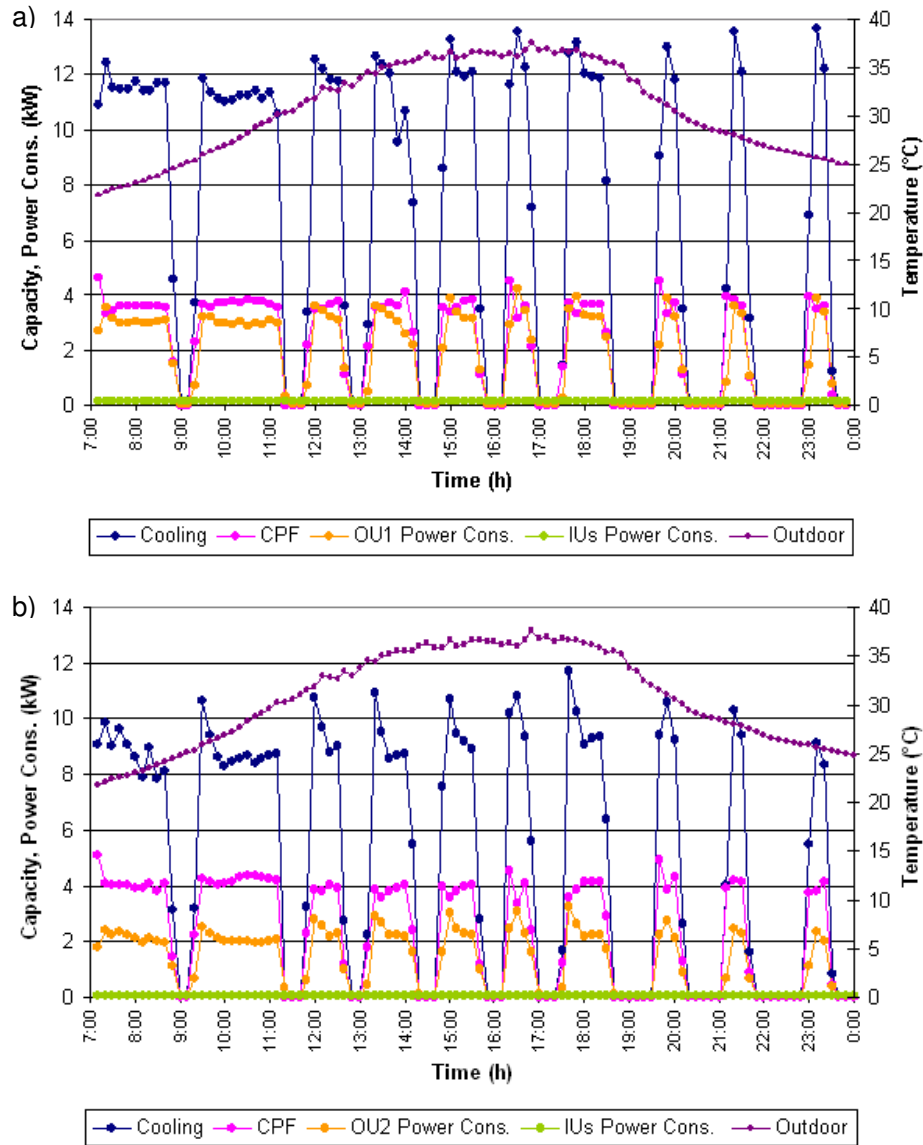


Figure 6.56 : Variation of the cooling capacity, outdoor unit power consumption, total power consumption of the indoor units and CPF of System1 and System2 for the master control mode on May 31, 2007, (a) System1, (b) System2

As can be seen from Figure 6.57, considering only “ON” times of the system, the system CPFs of the individual and master control modes were almost same at around 4 except the last period of the tests.

This shows that the temperature control mode affected the indoor thermal comfort and the power consumption of the outdoor units, but did not affect the efficiency of the system much.

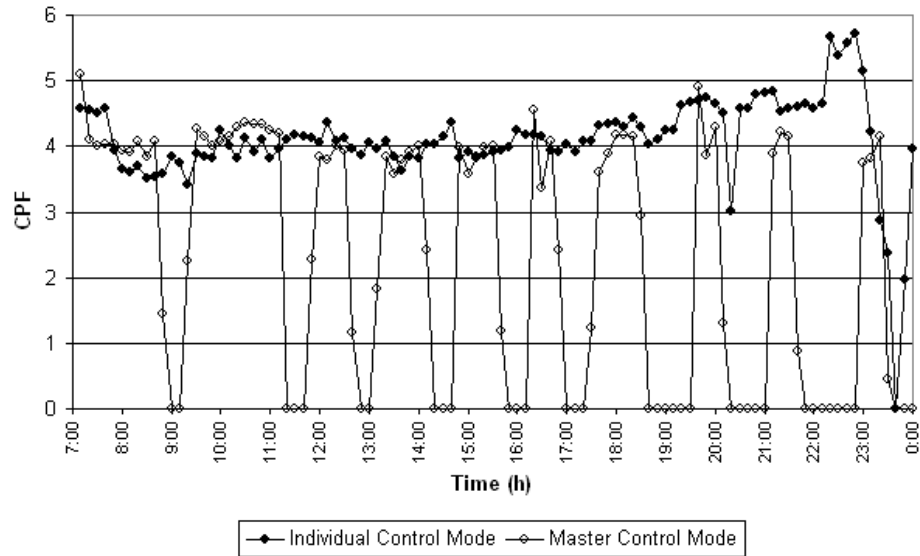


Figure 6.57 : Comparison of the variations of CPF for the individual and master control modes

6.1.2 Seasonal comparison at 25°C set temperature

In addition to the daily based comparison, seasonal comparisons were done for individual and master control mode operations.

The number of data points of the outdoor temperatures for the individual and master control mode tests, and the comparison of the outdoor conditions are provided in Figure 6.58 and Figure 6.59, respectively with 10-minute averaged data.

As can be seen from Figure 6.58 and Figure 6.59, the outdoor conditions of the individual and master control modes days were very close to each other. That's why; the effect of the control mode can be evaluated seasonally.

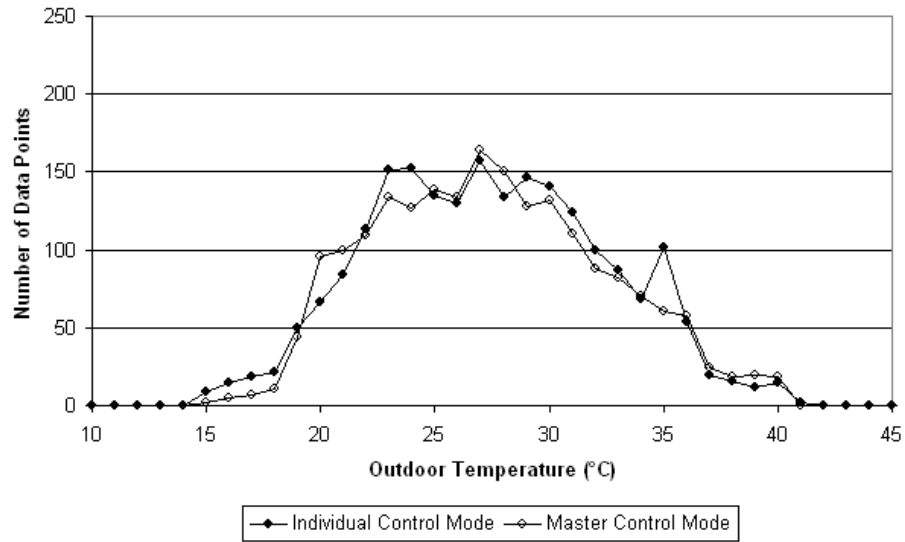


Figure 6.58 : Number of data points for 25°C set temperature tests

Individual Control Mode vs Master Control Mode

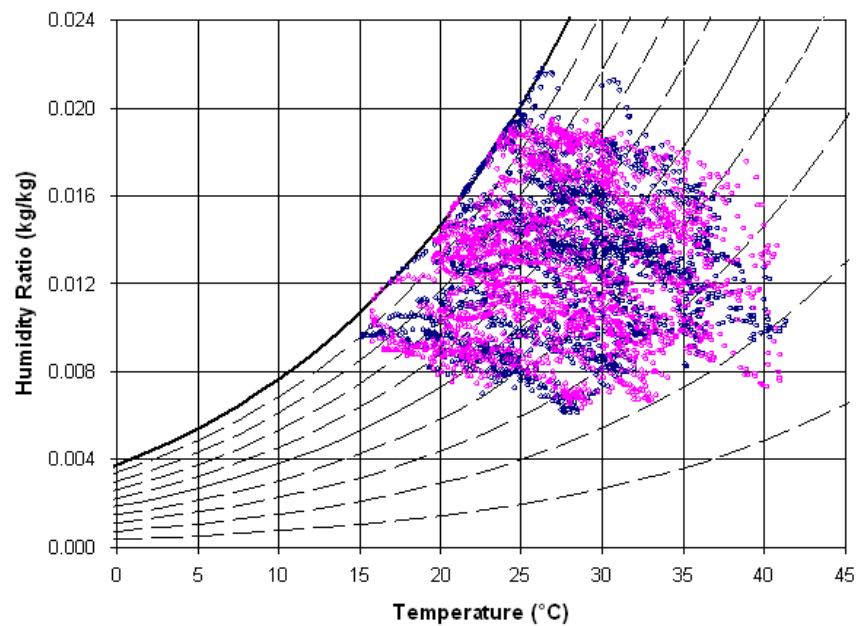


Figure 6.59 : Outdoor conditions for 25°C set temperature tests

Figure 6.60 shows the indoor conditions of System1 zones provided by the individual and master control modes with 10-minute averaged data. Similar to the daily basis evaluation, the different colors correspond to different zones, such as red for Room B, pink for Room C, etc. The ASHRAE summer thermal comfort zone was superimposed to the charts.

As seen from Figure 6.60a and Figure 6.60b, similar to the daily basis comparison, the individual control mode controlled the indoor temperature much better than the master control mode. Since the set temperature was 25 °C, the indoor temperatures for the individual control mode were very close to the set temperature 25 °C. The data corresponded on the right side of the indoor condition data cloud in Figure 6.60 were the initial indoor conditions of the zones.

On the other hand, for the master control mode, shown in Figure 6.60b, the indoor temperatures were mostly lower than 25 °C, and even lower than 20 °C.

Table 6.9 shows the comparison of the indoor condition data of System1 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone.

Similar to the daily basis comparison, in the seasonal basis comparison, it was found that 68.0% of all the indoor condition data of System1 zones provided by the individual control mode data and 48.6% of all indoor condition data of System1 zones provided by the master control mode data were in the ASHRAE summer thermal comfort zone. Thus, it can be said that the multi-split VRF system controlled by the individual control mode provided better thermal comfort than the master control mode. Besides, the master control mode provided colder indoor environment than the individual control mode, because 46.6% of all indoor condition data of System1 zones provided by the master control mode was found in the cold region, on the other than, only 0.1% of all indoor condition data of System1 zones provided by the individual control mode was found in the cold region.

Table 6.9 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes at 25 °C set temperature based on the ASHRAE summer thermal comfort zone

Region	Percentage of the indoor condition data for the individual control mode	Percentage of the indoor condition data for the master control mode
In	68.0%	48.6%
Cold	0.1%	46.6%
Hot	1.4%	0.2%
Humid	30.5%	4.6%
Dry	0.00%	0.0%

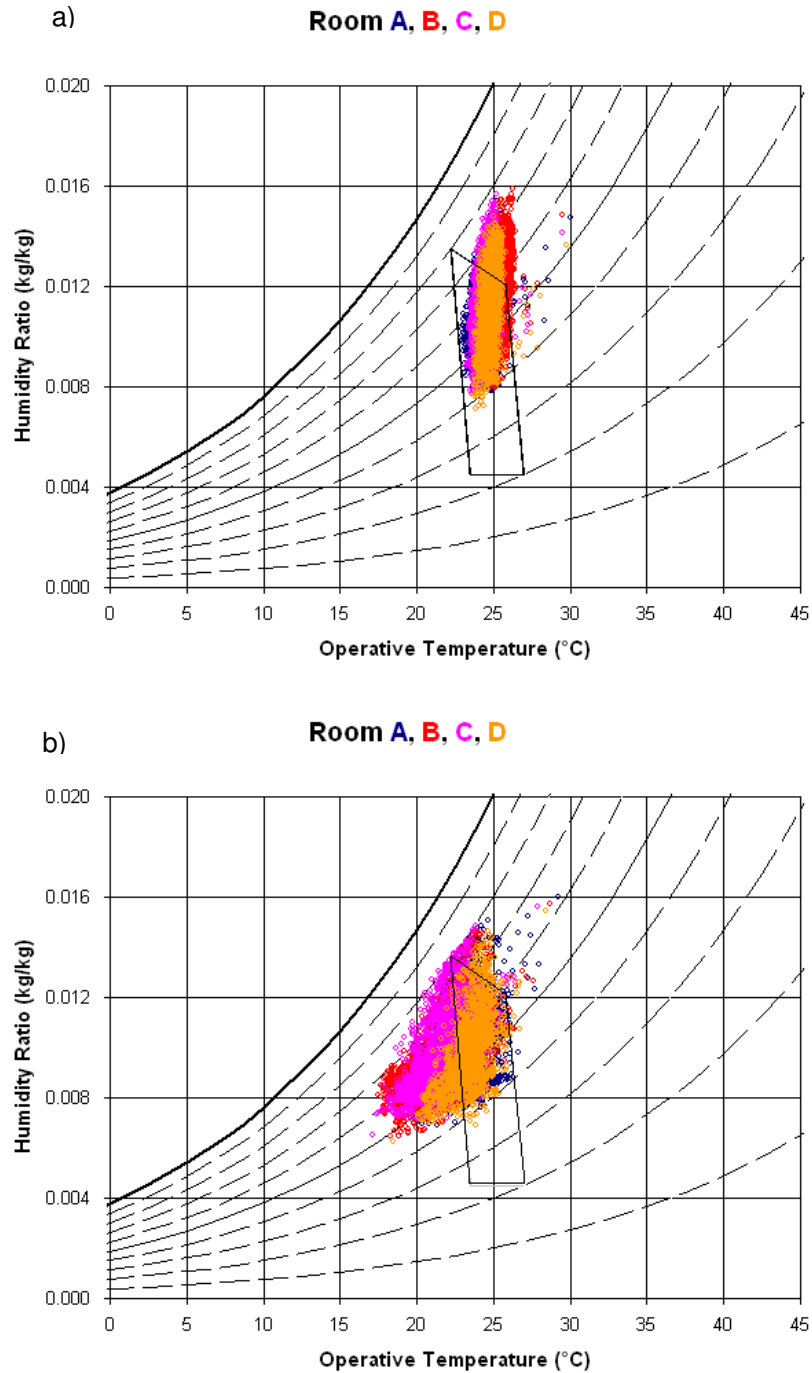


Figure 6.60 : Indoor conditions of System1 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode

The indoor condition data of System2 zones provided by the individual and master control modes can be seen in Figure 6.61.

As seen from Figure 6.60a and Figure 6.60b, similar to the daily basis comparison, the individual control mode controlled the indoor temperature much better than the master control mode.

Table 6.10 shows the comparison of the indoor condition data of System2 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone.

It was found that 87.3% of all the indoor condition data of System2 zones provided by the individual control mode data and 68.5% of all indoor condition data of System2 zones provided by the master control mode data were in the ASHRAE summer thermal comfort zone.

The reason of 3.3% data in the hot region provided by the individual control mode was mostly due to the elevator zone. Because IUEI was the smallest capacity indoor unit and the cooling load of the elevator zone was relatively high. That's why; in some relatively warm days, the indoor temperature could not be kept at the set temperature.

Table 6.10 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes at 25°C set temperature based on the ASHRAE summer thermal comfort zone

Region	Percentage of the indoor condition data for the individual control mode	Percentage of the indoor condition data for the master control mode
In	87.3%	68.5%
Cold	1.9%	10.2%
Hot	3.3%	11.2%
Humid	7.5%	10.1%
Dry	0.0%	0.0%

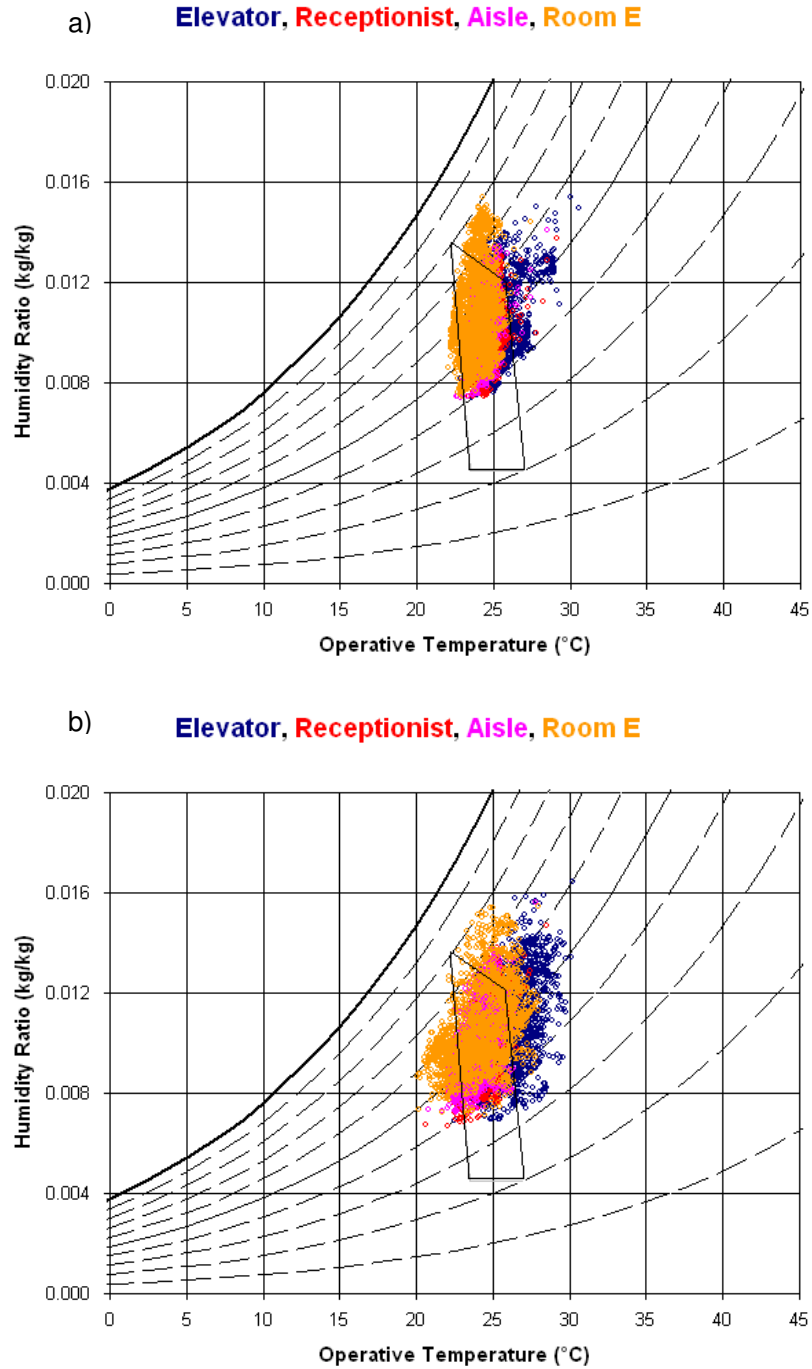


Figure 6.61 : Indoor conditions of System2 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.62 shows the TSS variations of System1 zones for the individual and master control modes.

As can be seen, the TSS variations of System1 zones for the individual control mode were much closer to the neutral point than the master control mode.

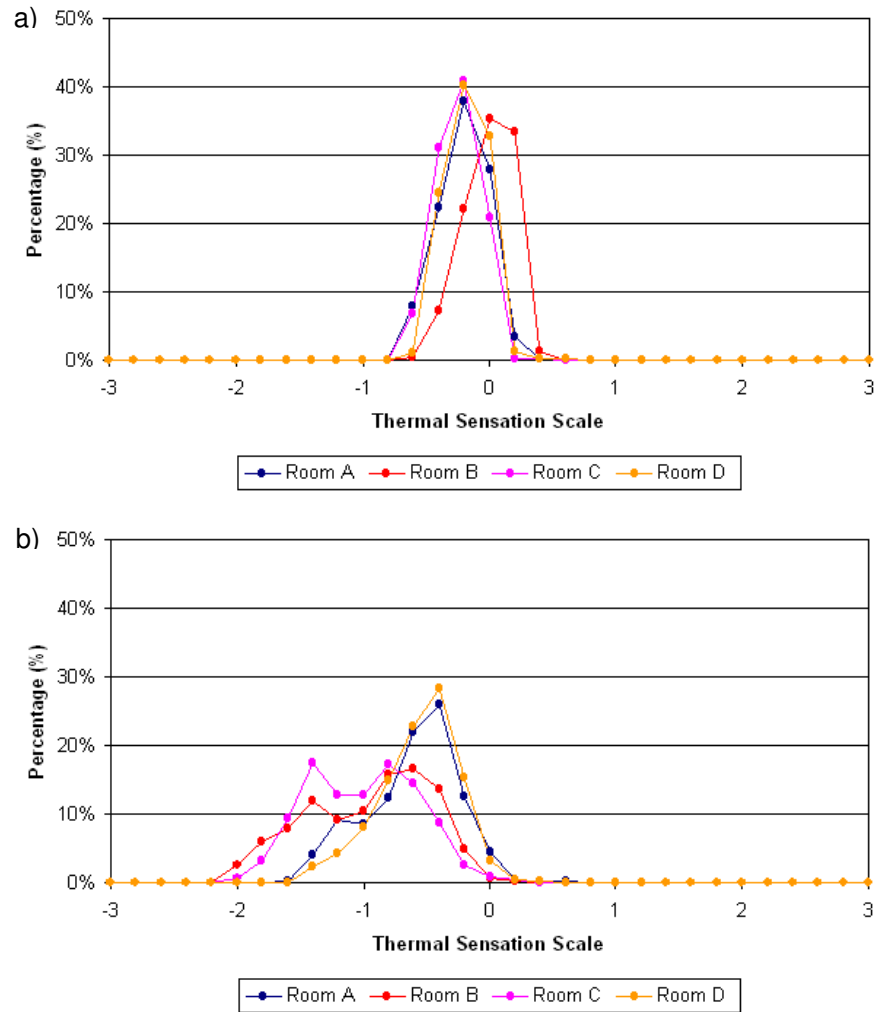


Figure 6.62 : TSS variations of System1 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.63 shows the TSS variations for System2 zones for the individual and master control modes

Similar to System1 zones, TSS variations for System2 zones for the individual control mode were much closer to the neutral point than the master control mode.

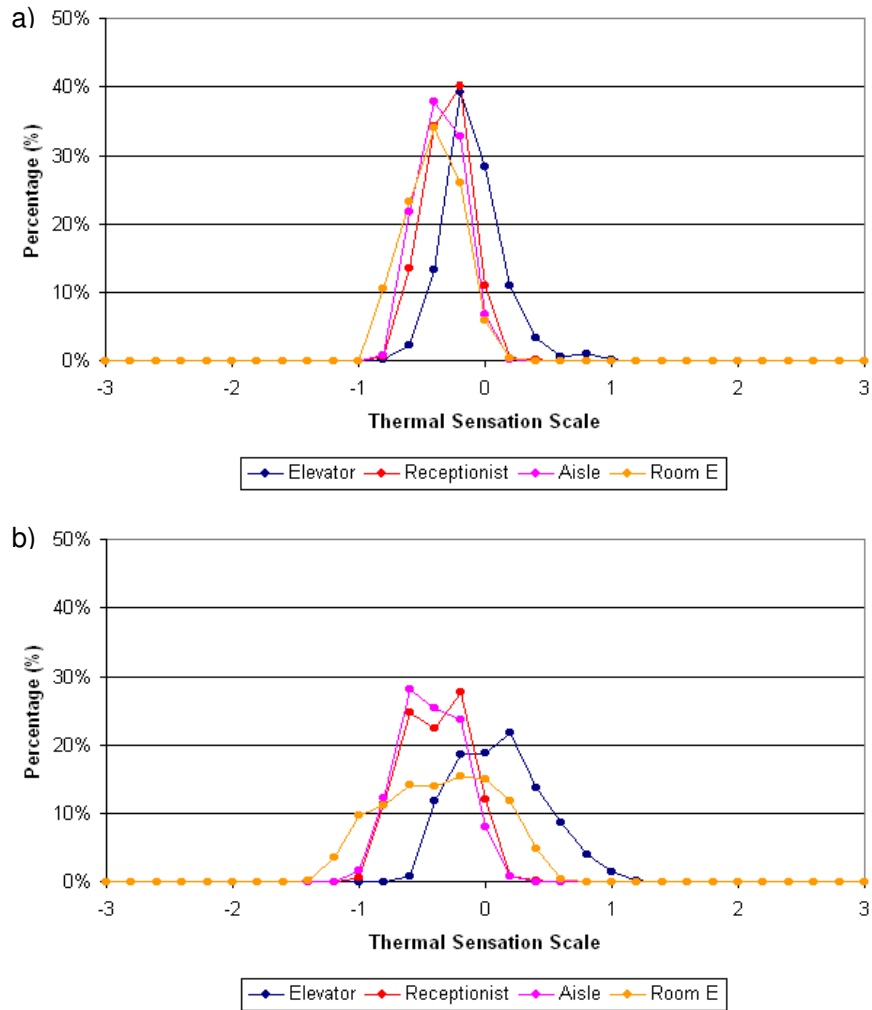


Figure 6.63 : TSS variations of System2 zones provided by the individual and master control modes at 25°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.64 shows the TSS variations based on the combined zones for the individual and master control mode. As expected, TSS for individual control mode was much closer to the neutral point than the master control mode with higher percentage ratio.

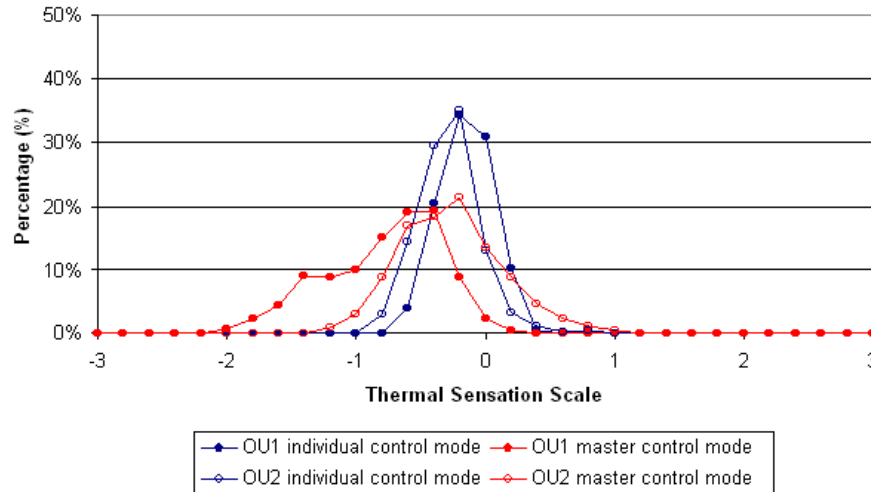


Figure 6.64 : TSS variations of System1 and System2 combined zones provided by the individual and master control modes at 25°C set temperature

Figure 6.65 shows the comparison of the daily averaged energy consumption of the multi-split type VRF system for the individual and master control modes. Since the average daily energy consumption is taken into account, the difference of the number of data points between the individual and the master control modes, shown in Figure 6.58, will be canceled out.

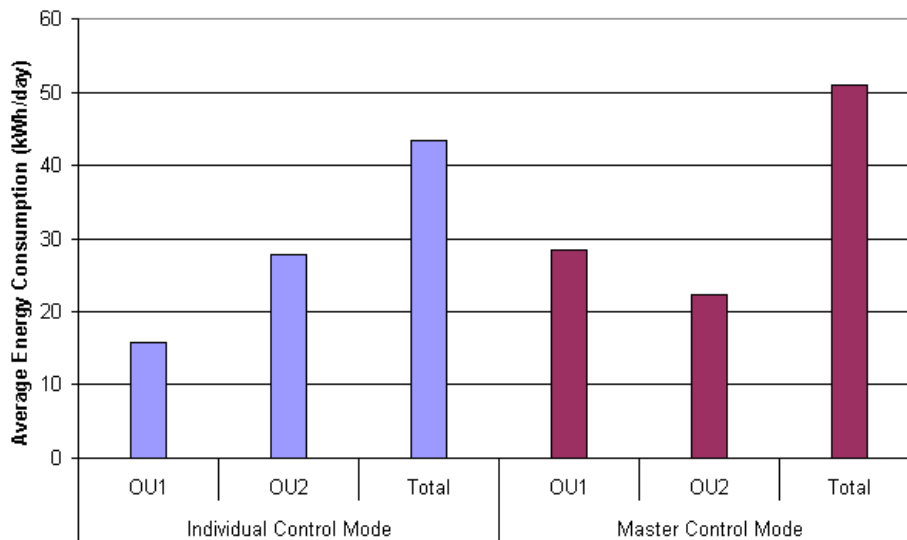


Figure 6.65 : Comparison of the daily averaged energy consumption of the individual and master control modes at 25°C set temperature

Similar to the daily based comparison, the seasonal basis comparison shows that for OU1, the master control mode consumed around 82.9% higher energy than the individual control mode, and for OU2, the individual control mode consumed around

25.2% higher energy than the master control mode. Overall, it was found that the master control mode consumed around 16.9% higher energy than the individual control mode.

Figure 6.66 shows the variations of the percentage “ON” time of OU1 and OU2 for the individual and master control modes with respect to 5°C-outdoor temperature bins within an outdoor temperature range of 10-45°C. As expected, the percentage “ON” time increased for both control modes with the increasing outdoor temperature in order to keep the indoor temperatures at the set temperature. Since both outdoor units operated simultaneously, the percentage “ON” times of OU1 and OU2 for the master control mode were same.

As explained in Figure 6.44, four indoor units were operated simultaneously in the master control mode which caused higher the power consumption compared to the individual control mode. That’s why; even though the percentage “ON” time of OU1 in individual control mode was higher than that of OU1 in master control mode for most of the outdoor temperature bins, the master control mode consumed more power than the individual control mode. However, for OU2, since the difference between the percentage “ON” times for the individual and master control mode was relatively high compared to OU1, the individual control mode consumed more power than the master control mode.

The difference between the percentage “ON” times of OU1 and OU2 for the individual control mode was due to the difference between the internal loads and the effect of the outdoor conditions on the zones. They were summarized in Table 6.4 and Table 6.5.

Variation of daily energy consumptions of the outdoor units with respect to the daily averaged outdoor temperatures for the individual and master control modes are provided in App. [G].

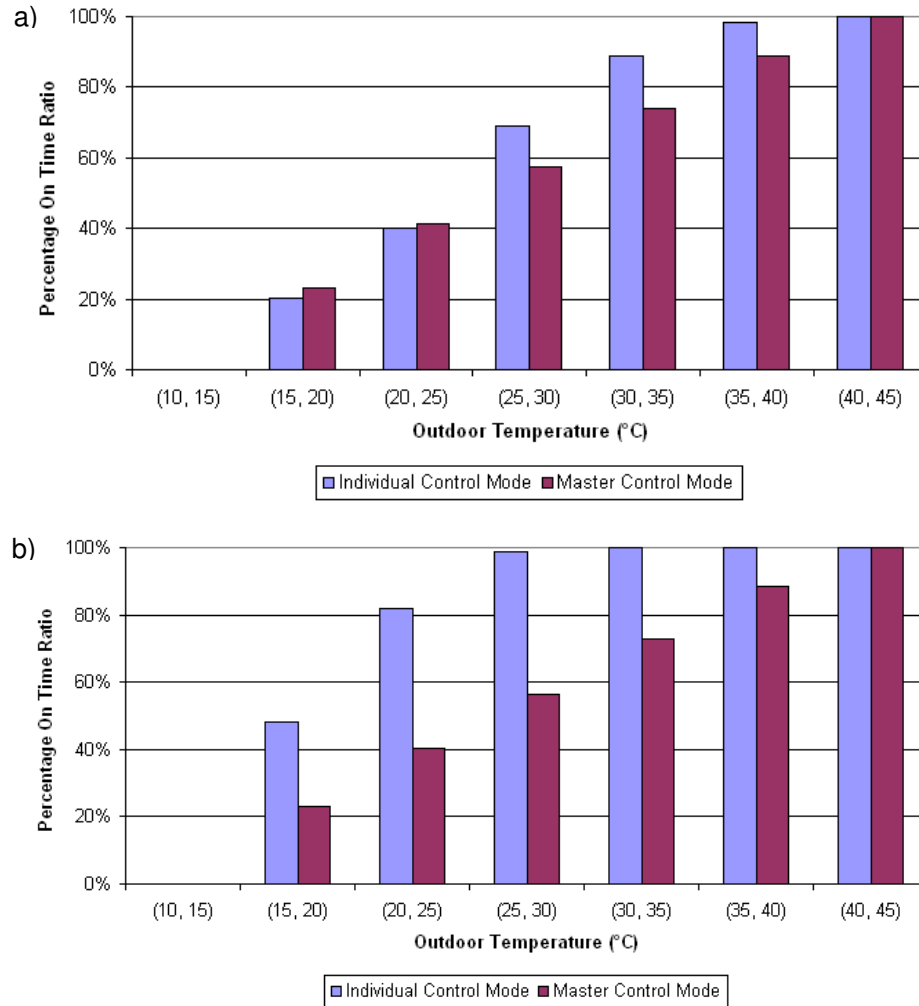


Figure 6.66 : Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 25°C set temperature, (a) OU1, (b) OU2

The provided daily cooling energy, daily energy consumption of OU2 and corresponding CPF and PLR are plotted in Figure 6.67 with respect to the daily averaged outdoor temperature for both individual and master control modes. The trends are also shown with least square regression lines.

As can be seen from Figure 6.67a and Figure 6.67b, similar to the daily basis comparison, the trend of the provided cooling energy followed the trend of the energy consumption of OU2 for both individual and master control modes. It is observed that the energy consumption of OU2 and the corresponding provided cooling energy varied linearly with the outdoor temperature for the individual control mode, on the other hand, they varied exponentially with the outdoor temperature for the master control mode varied.

As seen from Figure 6.67c that, the individual control mode provided more cooling than the master control mode by consuming higher energy, mainly due to the elevator zone, which was explained previously.

As seen from Figure 6.67d that, the variations of the daily CPF of the individual and master control modes with the outdoor temperature were similar which agrees with the daily basis comparison. Besides, it is observed that the efficiency trends are similar to the findings of Watanabe et al. (2004). The reason of the low efficiency at the low outdoor temperature is explained by the cyclic operation of the system by Watanabe et al. (2004). The system consumes high energy while providing less cooling during each start-up, and the system stops with high inverter frequency resulting in high energy consumption shown in Figure 6.18 for OU1. In addition, during the “OFF” times, the system still consumes energy due to the controllers of the outdoor units and due to the indoor units’ fans without providing any cooling.

As can be seen from Figure 6.67d, CPF of the individual control mode is relatively higher than that of the master control mode for high outdoor temperature. This is again because of the cyclic operation of the master control mode. The individual control mode operated continuously, however the master control mode operated cyclically. However, for low outdoor temperature, CPF of the individual and master control modes became identical, since the cyclic operation occurred for both systems.

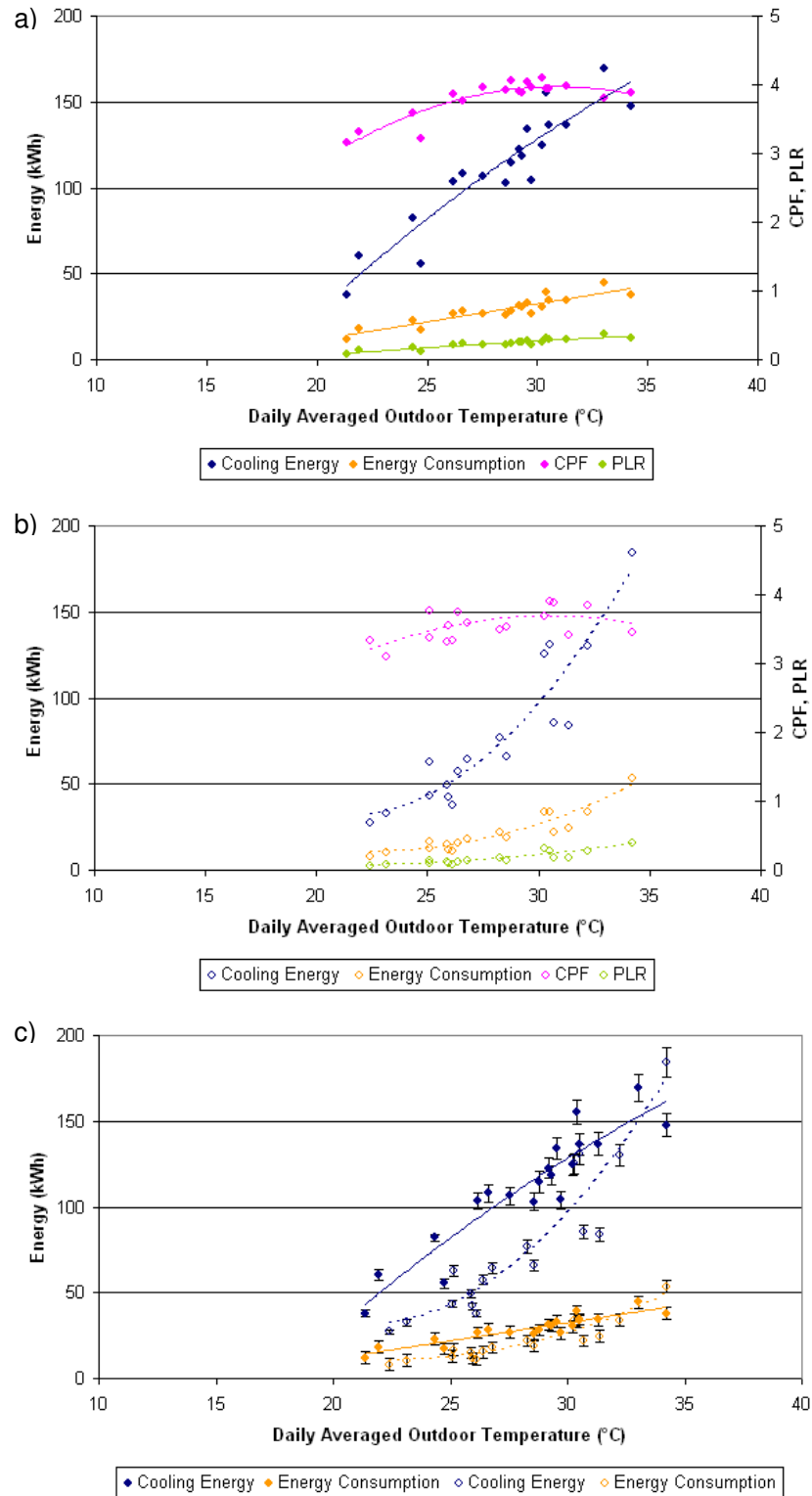


Figure 6.67 : Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 25°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF (continued)

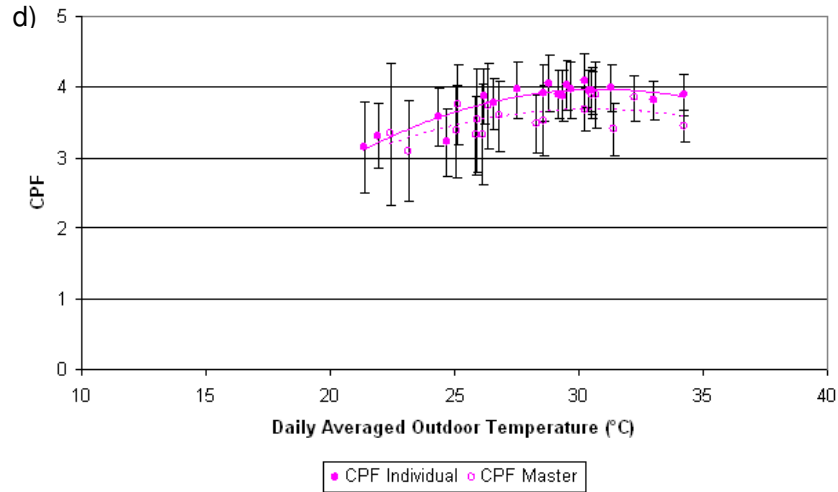


Figure 6.67 : Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 25°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF

It is also observed from Figure 6.67a and Figure 6.67b that, even for high outdoor temperature, PLR was even lower than 0.5. This is because the system was oversized.

Figure 6.68 shows the seasonal “ON/OFF” time ratio of OU2 for the individual and master control modes. As can be seen, OU2 for the individual control mode operated 92.3% of all time during the season, while for the master control mode; OU2 operated 50.3% of all time during the season. During the operation of OU2 for the individual control mode and for the master control mode, it was found that the fixed speed compressor operated in conjunction with the inverter driven compressor only 0.6% of all operation, as shown in Figure 6.69, which shows that the system was oversized.

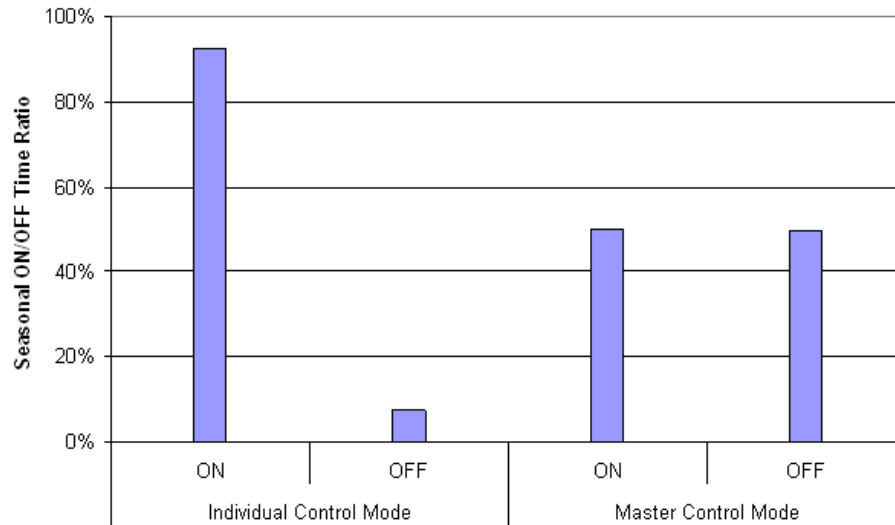


Figure 6.68 : Seasonal ON/OFF time ratio of OU2 for the individual and master control modes at 25°C set temperature

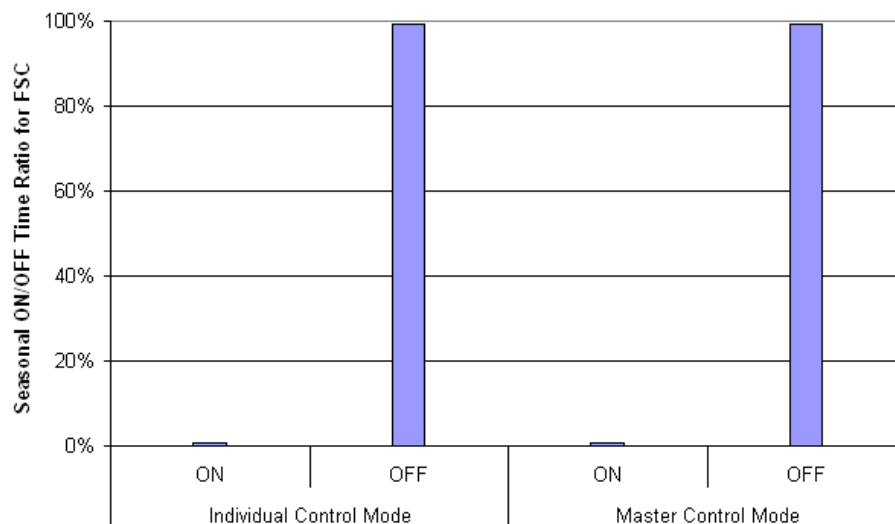


Figure 6.69 : Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes at 25°C set temperature

6.1.3 Seasonal comparison at 20°C set temperature

In addition to 25°C set temperature, 20°C set temperature tests were performed in order to find out the effect of a lower set temperature. Since the set temperature was 20°C, all the tests were held on weekends. Similar to 25°C set temperature tests, HRV units were used in conjunction with the multi-split VRF systems for the ventilations, and the indoor units were operated continuously regardless of the operation.

The number of data points of the outdoor temperatures for the individual and master control mode tests, and the comparison of the outdoor conditions are provided in Figure 6.70 and Figure 6.71, respectively.

As can be seen from Figure 6.70 and Figure 6.71, the outdoor conditions of the individual and master control modes days are very close to each other.

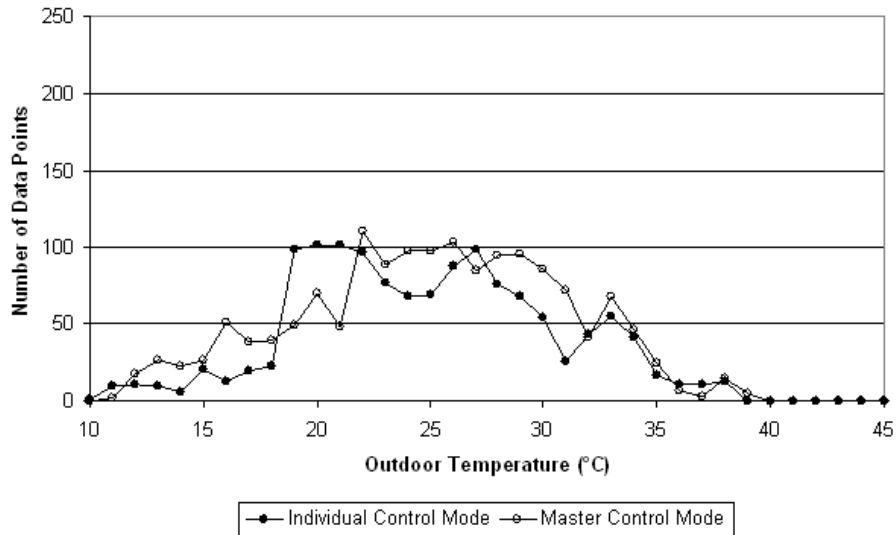


Figure 6.70 : Number of data points for 20°C set temperature tests

Individual Control Mode vs Master Control Mode

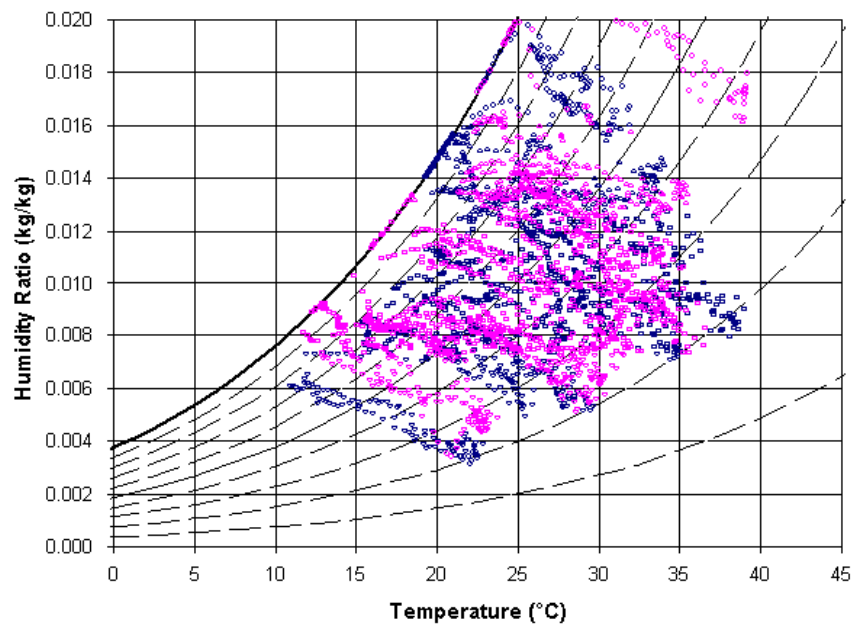


Figure 6.71 : Outdoor conditions for 20°C set temperature tests

Figure 6.72 shows the indoor conditions data of System1 zones provided by the individual and master control modes. Since the set temperature was out of the ASHRAE summer comfort zone, the zone was not superimposed to the charts.

As can be seen, similar to high set temperature tests (25°C), the indoor condition data of System1 zones provided by the individual control mode were much closer to the set temperature of 20°C than that of System1 zones provided by the master control mode. The indoor temperatures provided by the master control mode even reached to 15°C.

Figure 6.73 shows the indoor conditions data of System2 zones provided by the individual and master control modes. Excluding the elevator zone, it can be seen that individual control mode controlled the indoor temperature better than the master control mode. As explained in the previously, the elevator zone was equipped with the smallest capacity indoor unit; however the cooling load of the elevator zone was relatively high. That's why; the indoor temperature could not be kept at the set temperature even for the individual control mode.

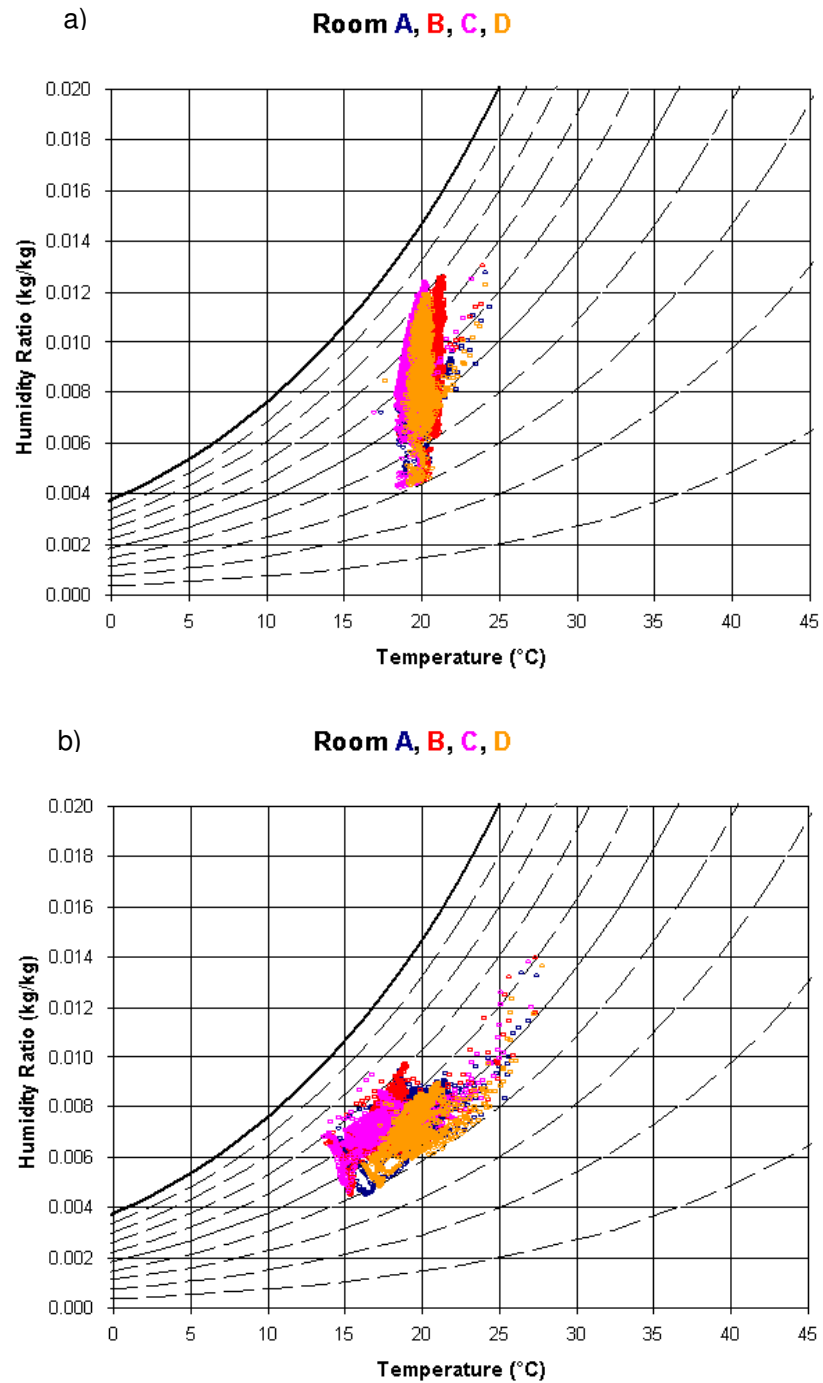


Figure 6.72 : Indoor conditions of System1 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode

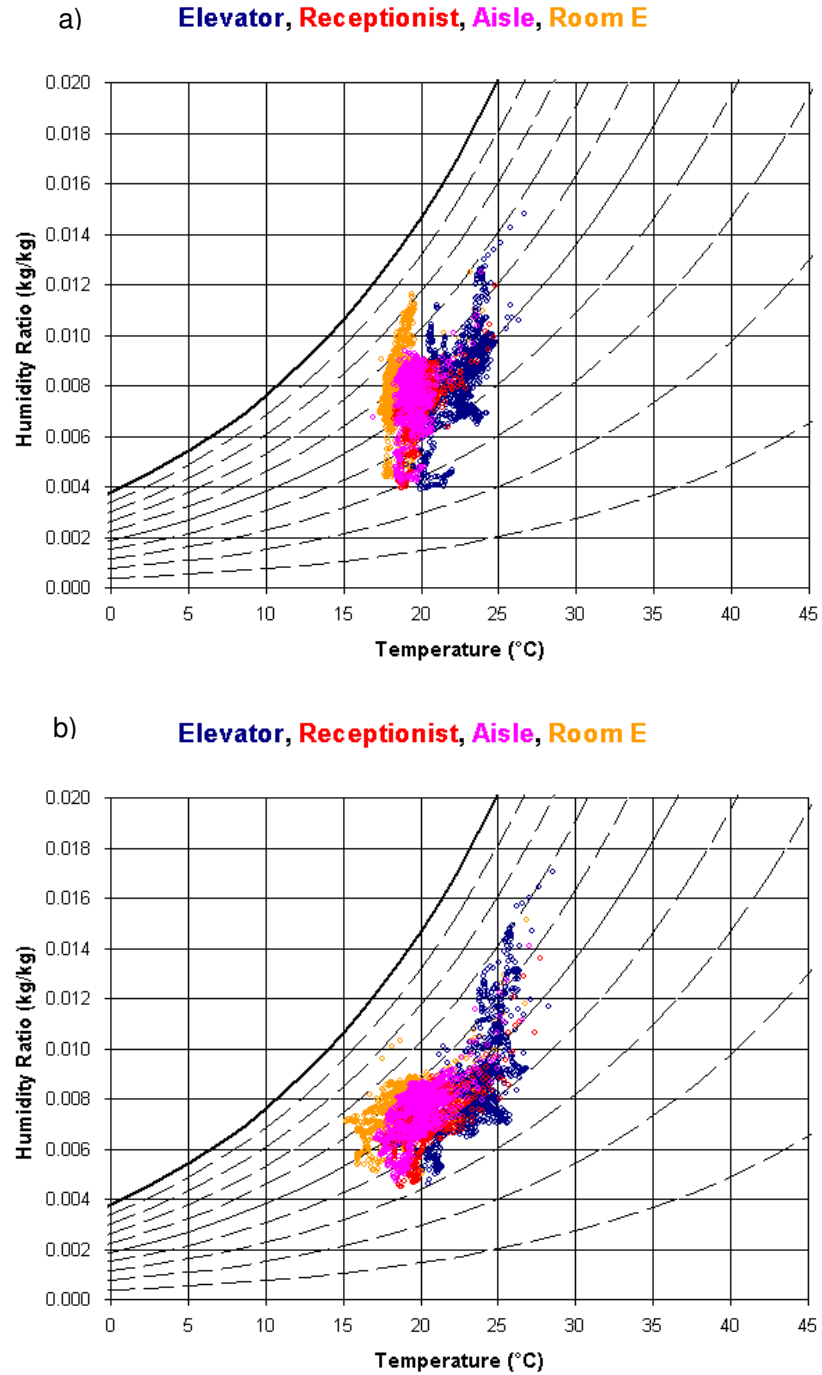


Figure 6.73 : Indoor conditions of System2 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.74 shows the TSS variation of System1 zones provided by the individual and master control modes.

As can be seen, since the set temperature was 20°C, TSS values were no longer around neutral point. TSS for the individual control mode varied between cool and slightly cool region, however, TSS for the master control mode was found within cold and slightly cool region. Besides, the percentages of the individual control mode data were higher than that of the master control mode data which shows that the indoor temperature fluctuations for the individual control mode were lower than the master control mode.

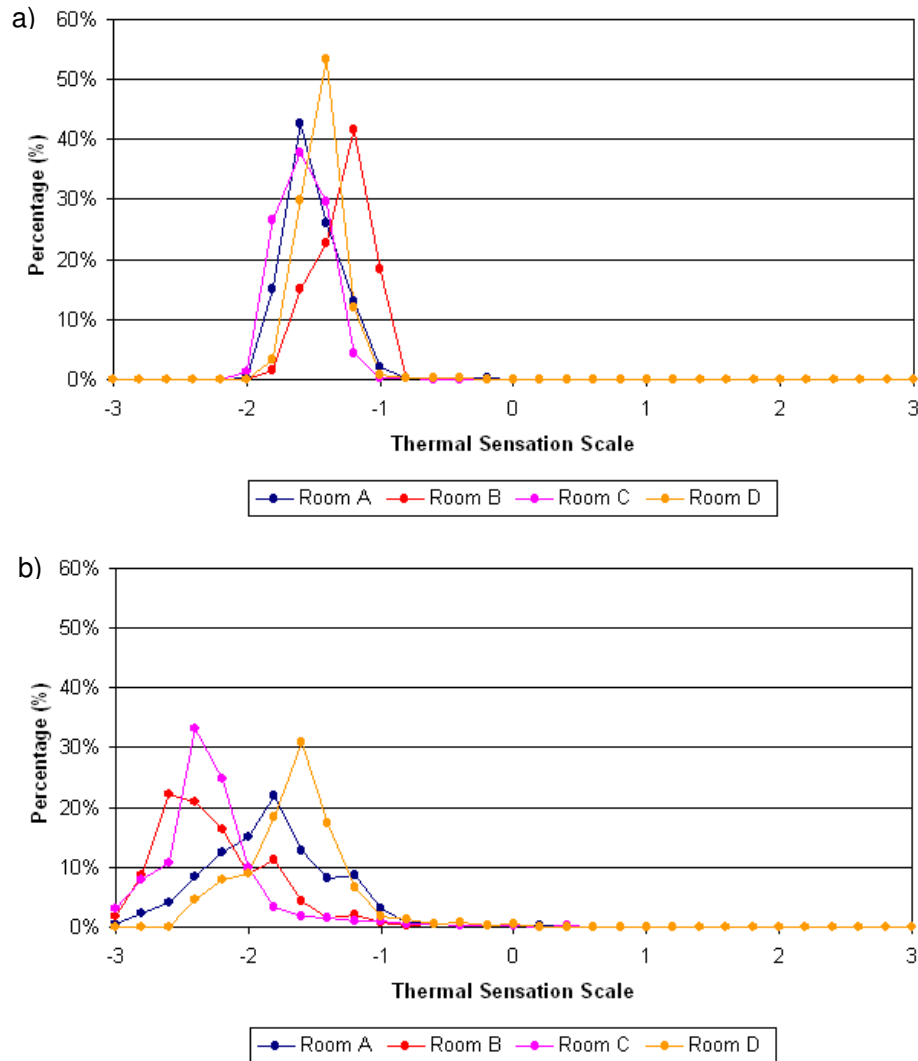


Figure 6.74 : TSS variations of System1 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.75 shows the TSS variation of System2 zones provided by the individual and master control modes.

As can be seen, excluding the elevator zone, it was found that the percentages of the individual control mode data were higher than that of the master control mode data which shows that the indoor temperature fluctuations for the individual control mode were lower than the master control mode.

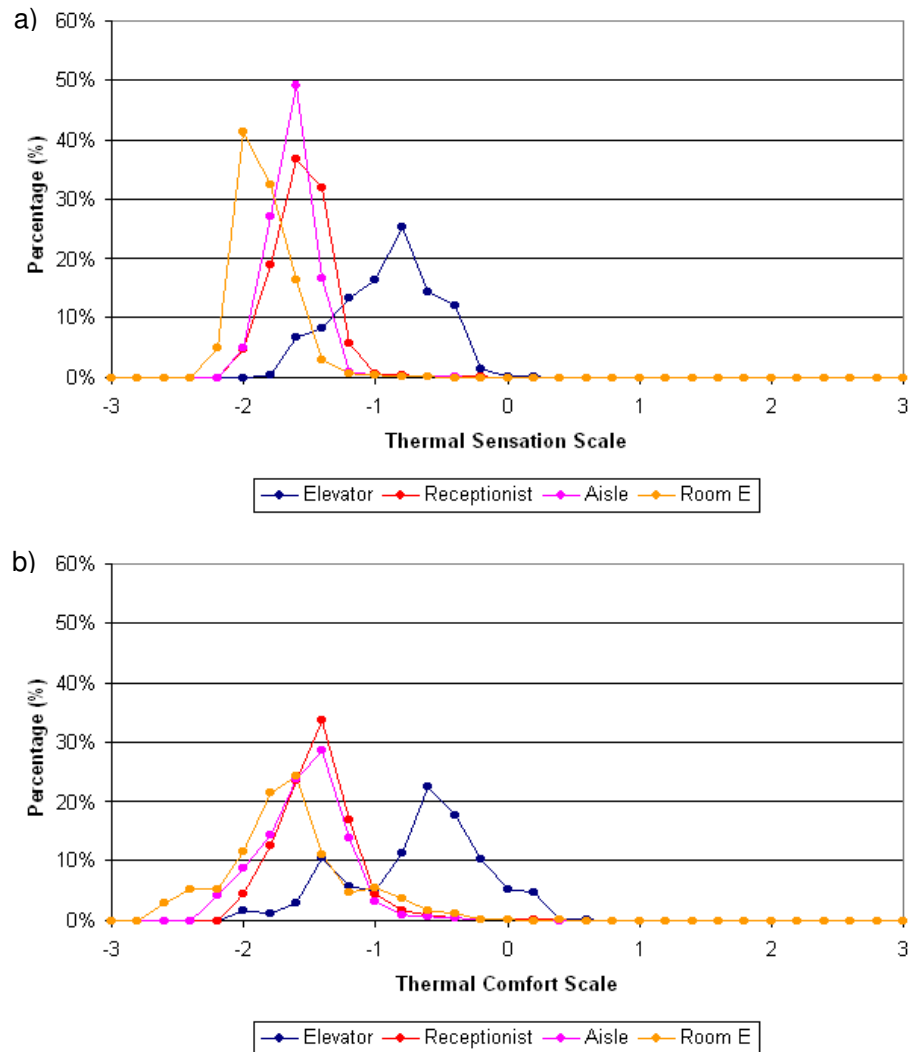


Figure 6.75 : TSS variations of System2 zones provided by the individual and master control modes at 20°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.76 shows the TSS variations based on the combined zones for the individual and master control modes. As seen, the TSS variation of System1 for the individual control mode was better than that of System1 for the master control mode with higher percentages.

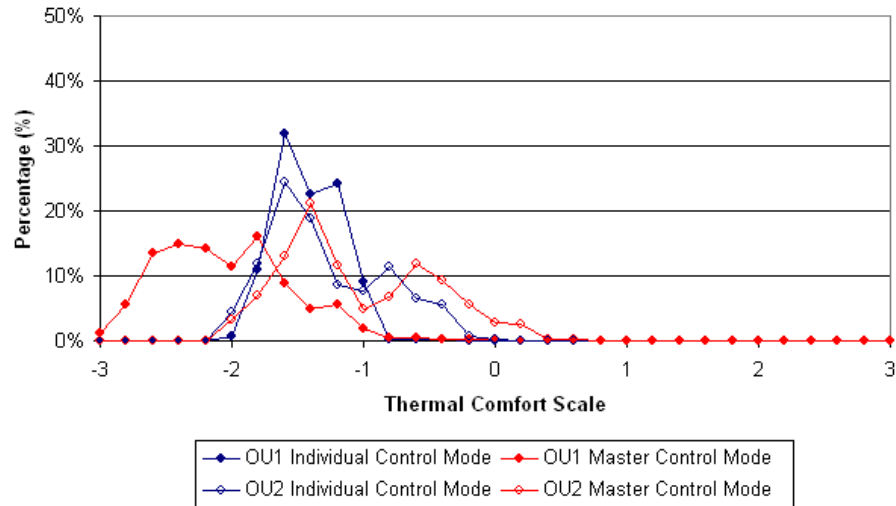


Figure 6.76 : TSS variations of System1 and System2 combined zones provided by the individual and master control modes at 20°C set temperature

Figure 6.77 shows the comparison of the daily averaged energy consumption for the individual and master control modes. It was found that for OU1, the master control mode consumed around 76% higher energy than the individual control mode. On the other hand, for OU2, it was found that the individual and master control modes consumed around same amount of energy. Overall, it was found that the master control mode consumed around 31.7% higher energy than the individual control mode.

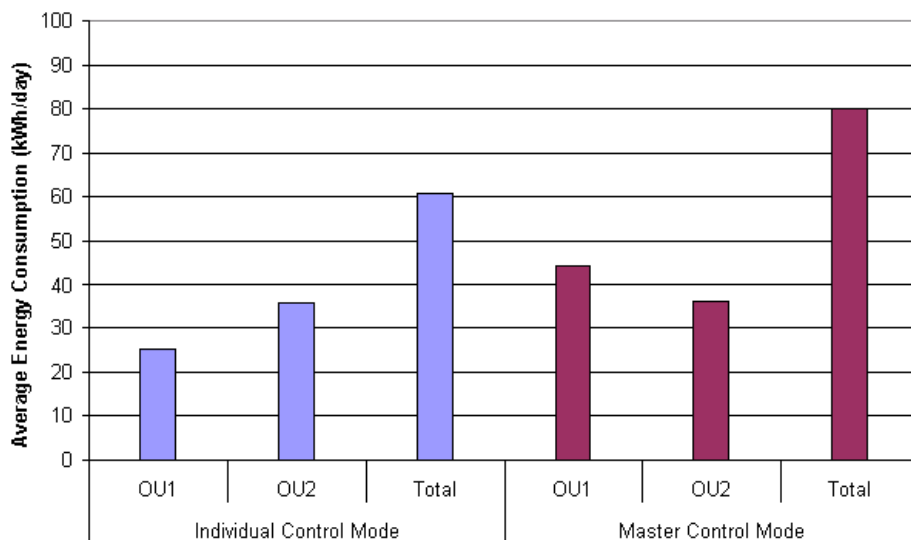


Figure 6.77 : Comparison of the daily averaged energy consumption of the individual and master control modes at 20°C set temperature

Figure 6.78 shows the variations of the percentage “ON” time ratios of OU1 and OU2 for the individual and master control modes.

As can be seen from Figure 6.78b, the percentage “ON” time ratios of OU2 for the individual and master control modes were very close, and since the set temperature was 20°C, the effect of the number of indoor units in operation was no longer important which was dominant for the high set temperature application, shown in Figure 6.66. That’s why; the power consumptions of the individual and master control modes were very close for OU2.

On the other hand, since the percentage “ON” time ratio of the master control mode was higher than that of the individual control mode for even relatively low outdoor temperature, shown in Figure 6.78a, the power consumption of the master control mode was higher than that of the individual control mode, which also caused colder indoor environment, shown in Figure 6.72.

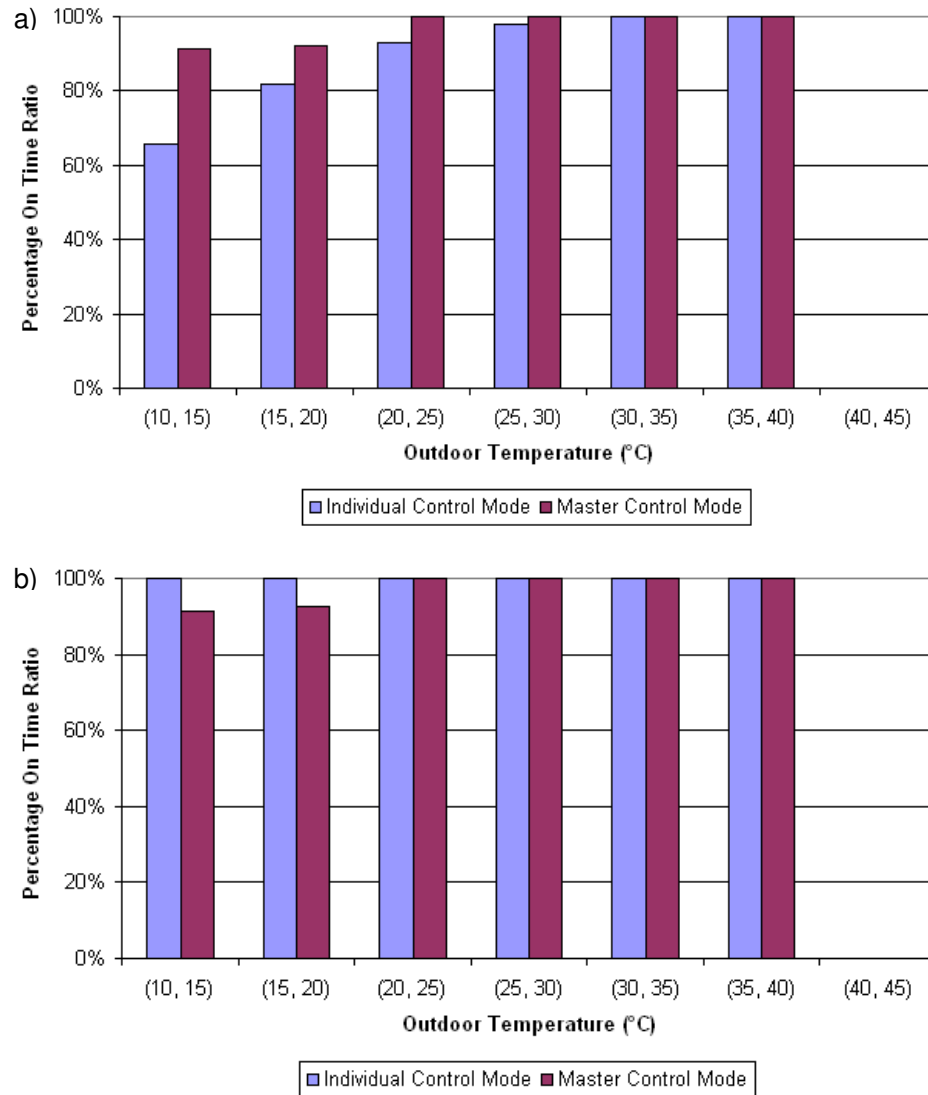


Figure 6.78 : Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 20°C set temperature, (a) OU1, (b) OU2

Variation of daily energy consumptions with respect to the daily averaged outdoor temperatures for the individual and master control modes are provided in App. [H].

The provided daily cooling energy, daily energy consumption of OU2 and corresponding CPF and PLR are plotted in Figure 6.79 with respect to the daily averaged outdoor temperature for both individual and master control modes. The trends are also shown with least square regression lines.

As can be seen from Figure 6.79a and Figure 6.79b, similar to the high indoor set temperature tests (25°C), the trend of the provided cooling energy followed the trend of the energy consumption of OU2 for both individual and master control modes.

It can be observed from Figure 6.67c and Figure 6.79c that, the system with the low indoor set temperature provided more cooling energy at even low outdoor temperature compared to the high indoor set temperature case. The high cooling demand for the low indoor set temperature reduced the cyclic operation, which improved the efficiency at low outdoor temperature as shown in Figure 6.79d.

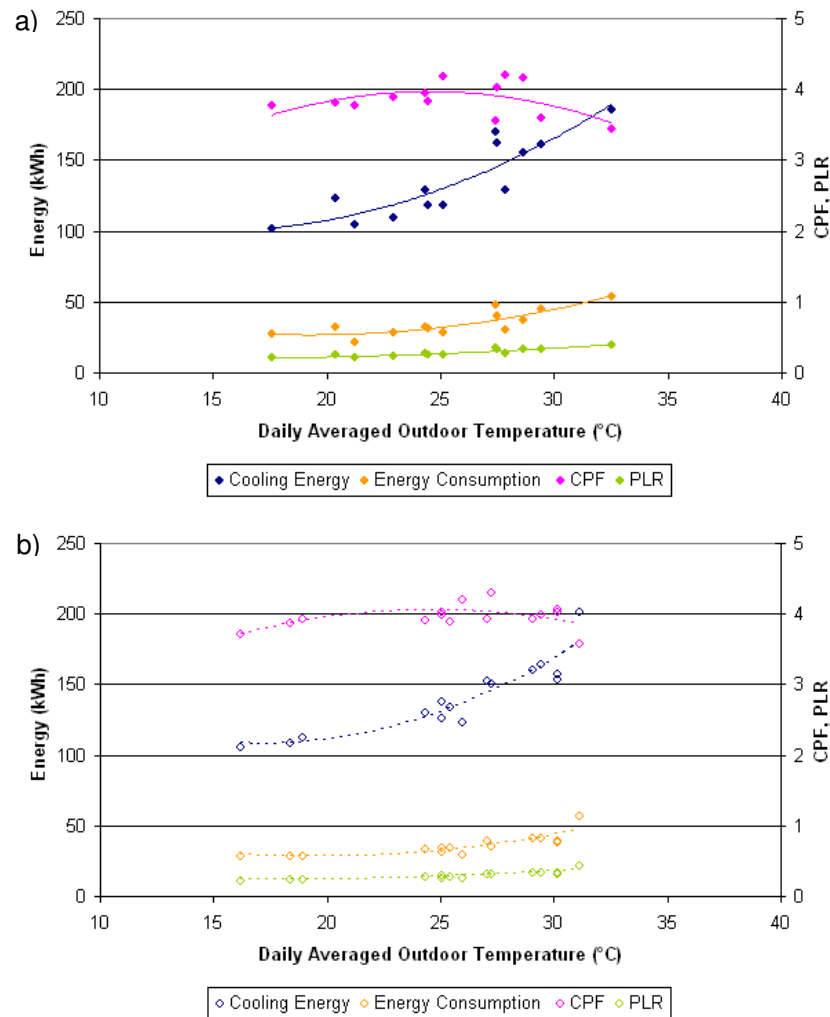


Figure 6.79...: Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 20°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF (continued)

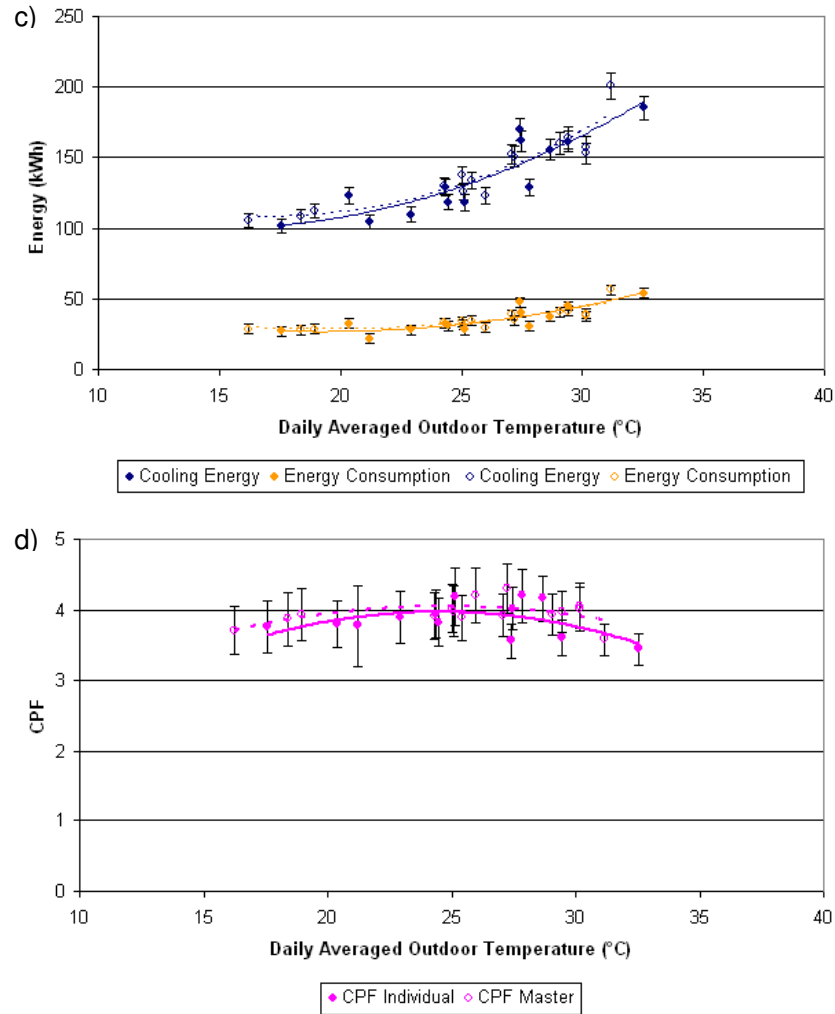


Figure 6.79 : Variation of cooling energy, energy consumption of OU2 and CPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 20°C indoor set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF

As can be seen from Figure 6.80, the seasonal “ON” time ratio increased for the low indoor set temperature. The system worked all time in individual control mode, and 98.3% of all time in master control mode. However, similar to the high indoor set temperature cases, for both individual and master control modes, FSC did not work almost 99.3% of all operation time, shown in Figure 6.81, which explains why PLR was lower even for the low set temperature case.

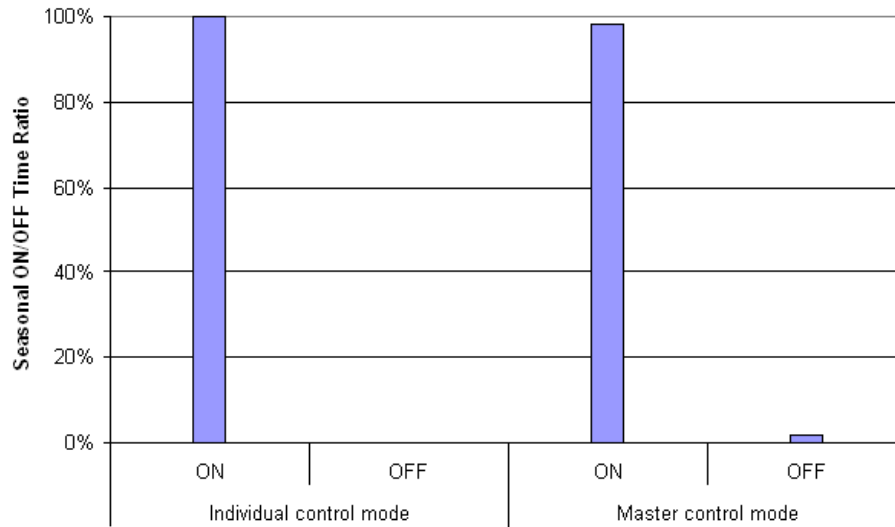


Figure 6.80 : Seasonal ON/OFF time ratio of OU2 for the individual and master control modes at 20°C set temperature

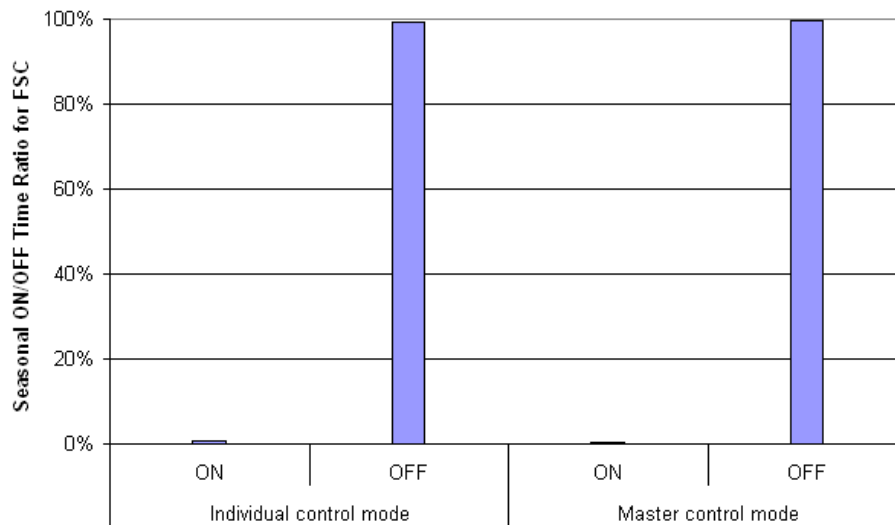


Figure 6.81 : Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes at 20°C set temperature

As explained, the multi-split VRF systems worked almost all of the operation time at the low indoor set temperature case. That's why; compared to the high indoor set temperature case, the possibility of finding operations under steady state conditions at low indoor temperatures was more likely.

Since the refrigerant mass flow rate mainly defines the indoor unit capacity, the operation under constant inverter frequency can be assumed to be steady state. Because the inverter frequency is the main parameter defines the refrigerant mass

flow rate. That's why, at least 20 minute-operations with constant inverter frequencies were tried to be found. In addition to the inverter frequencies, the indoor and outdoor air temperatures, as well as refrigerant temperatures and pressures were checked whether or not they violated the steady state conditions.

A typical 60-minute steady state operation, extracted from a master control mode experiment performed on September 01, 2007, Saturday, was given in Figure 6.82, Figure 6.83, Figure 6.84, Figure 6.85, Figure 6.86 and Figure 6.87 with one-minute intervals.

As can be seen from Figure 6.82, the inverter frequency, the corresponding outdoor unit power consumption, and the discharge and suction pressures were constant throughout the 60-minute period.

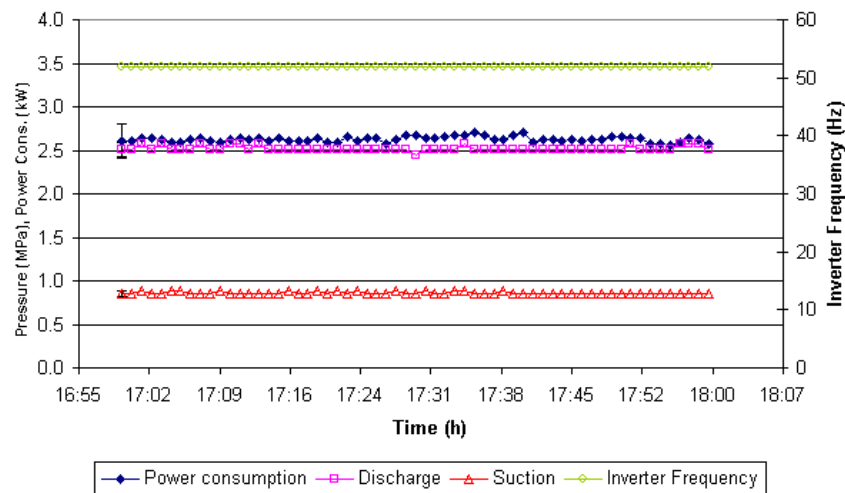


Figure 6.82 : Variation of the inverter frequency, refrigerant pressures and outdoor unit power consumption

As can be seen from Figure 6.83, since the discharge and suction temperatures, as well as the inverter frequency were constant, the indoor unit refrigerant inlet and outlet temperatures were constant.

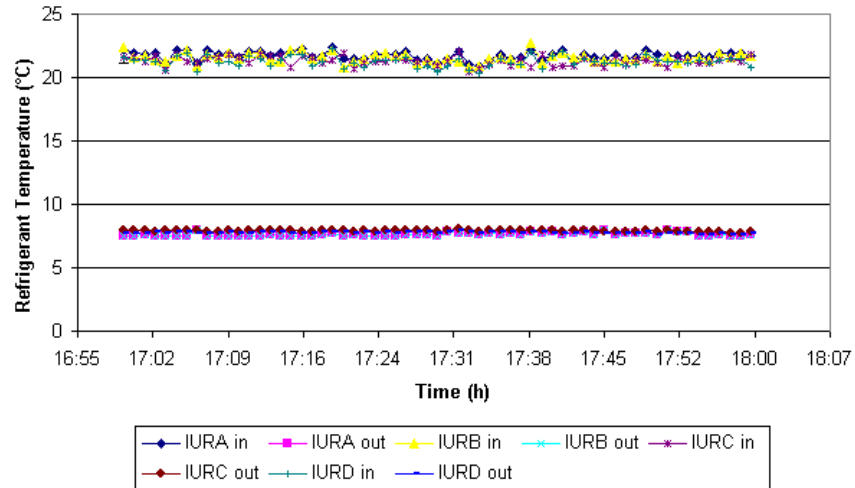


Figure 6.83 : Variation of the indoor unit refrigerant inlet and outlet

Figure 6.84 shows the indoor air blowing temperatures. As can be seen, the blowing temperatures were almost constant throughout the period.

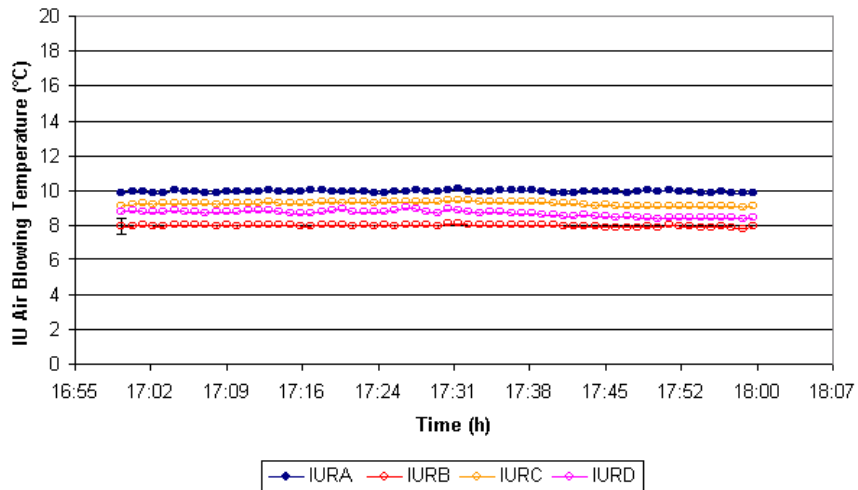


Figure 6.84 : Variation of the indoor unit air blowing temperature

In addition to the air blowing temperatures, Figure 6.85 shows the indoor air inlet temperatures. As can be seen, they can be assumed to be constant during the period.

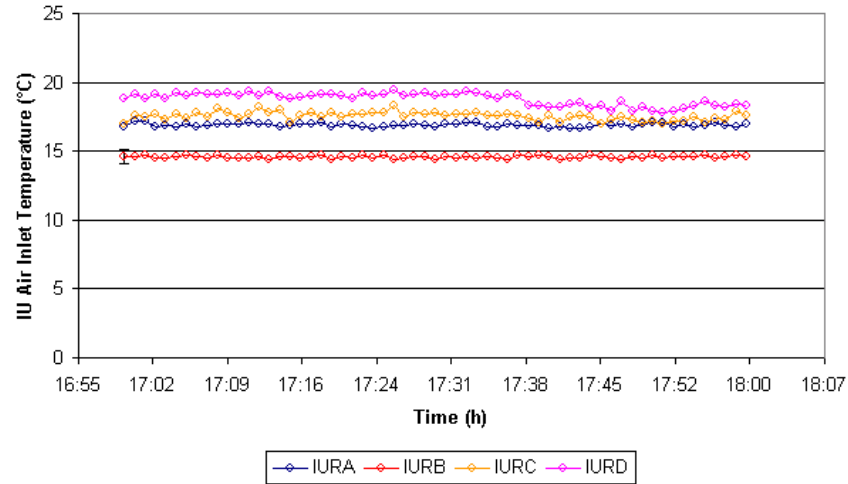


Figure 6.85 : Variation of the indoor unit air inlet temperature

Figure 6.86 shows the indoor and outdoor temperatures for the corresponding period, respectively. As can be seen, they can be assumed to be constant throughout the period.

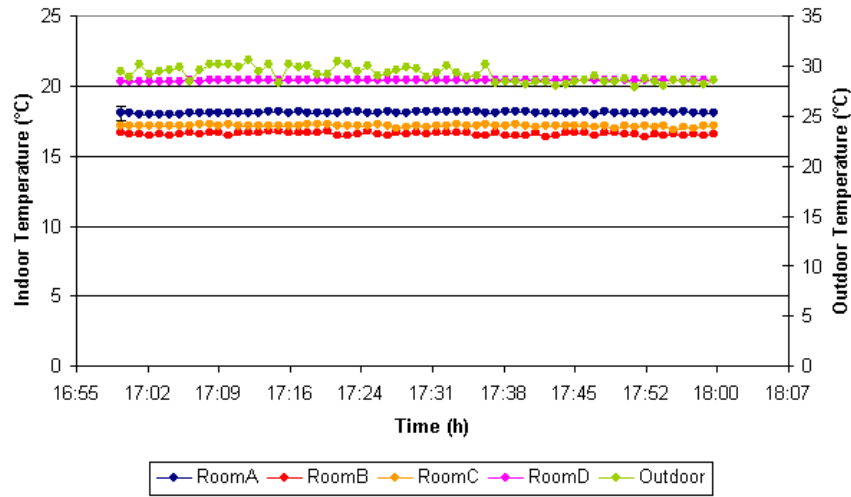


Figure 6.86 : Variation of the indoor and outdoor temperature

The corresponding individual indoor unit capacities and the COP of the system are given in Figure 6.87.

The COP of the system is defined as

$$COP = \frac{\dot{Q}_{cool}}{\dot{W}} \quad 6.1$$

where

$\dot{Q}_{cool}$: Total cooling capacity (kW)

As can be seen, they can be assumed to be constant throughout the 60-minute period.

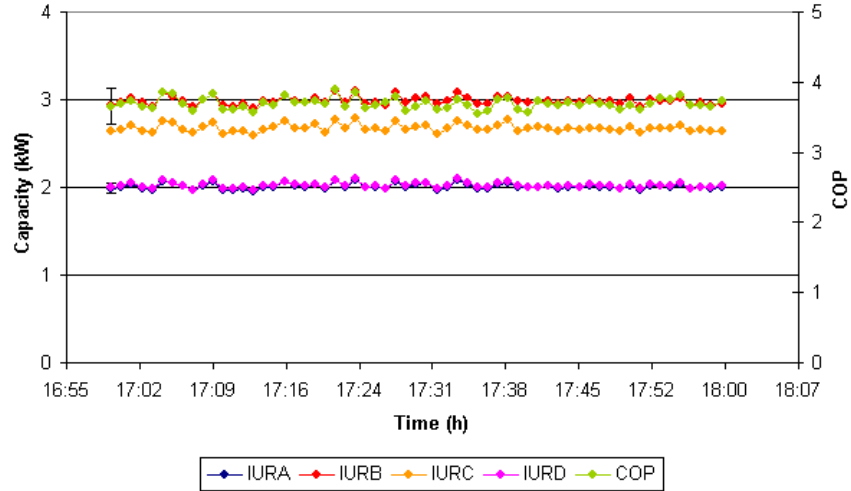


Figure 6.87 : Variation of the indoor unit capacities and the COP of the system

Figure 6.88 shows variations of the power consumption, total cooling capacity and the COP of the system with respect to the outdoor temperature under steady state operations, which were extracted from the master control mode experiments performed at 20°C indoor set temperature.

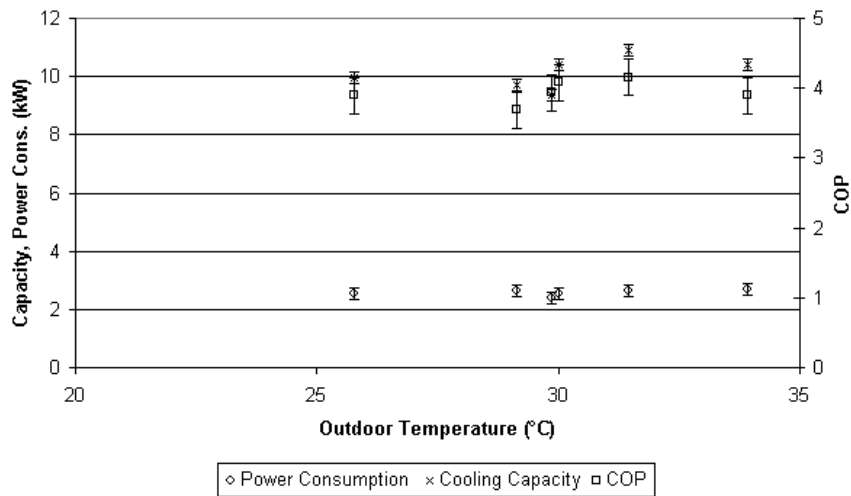


Figure 6.88 : Variation of the power consumption, total cooling capacity and the COP of the system with respect to the outdoor temperature

The comparison of the CPF and COP of the system is provided in Figure 6.89.

It can be observed from Figure 6.89 that the COP and CPF of the system are in the same order of magnitude.

This shows that the steady state tests can be used for the efficiency evaluations of the multi-split VRF system, however, the field tests should be performed for the evaluations of the indoor thermal comfort, and the seasonal energy consumption.

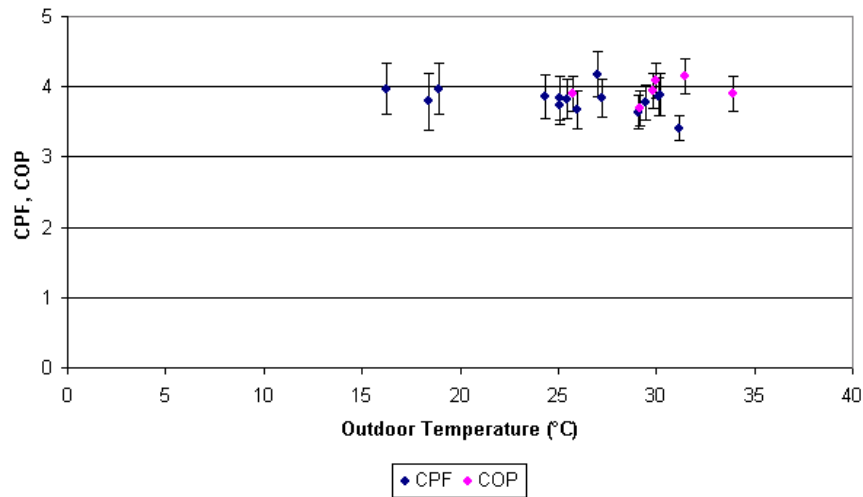


Figure 6.89 : Comparison of the CPF and COP of the system

6.2 Effect of the Ventilation in the Cooling Performance

In order to find out the effect of the ventilation on the indoor temperature control, thermal comfort, power consumption of the outdoor units and the efficiency of the multi-split type VRF system; parametric tests were performed with and without ventilation in the cooling mode.

During the ventilation assisted tests, the HRV units were operated in conjunction with the multi-split VRF systems throughout the test period. On the other hand, during the non-ventilated tests, the HRV units were not operated and the multi-split VRF systems were operated stand-alone. For both comparisons, the indoor set temperatures were kept at 25°C, and the fans of the indoor units were operated continuously regardless of the operation of the indoor unit.

Figure 6.90 and Figure 6.91 show the variation of the supply air temperature and supply air humidity ratio with respect to the outdoor air temperature and outdoor air humidity ratio, respectively. The data was recorded during the ventilation assisted tests from HRV1, and provided with 10-minute averaged data.

If the HRV unit does not have any heat recovery capacity which means 0% efficient, the variation of the supply and the outdoor air temperature follows the dashed line,

shown in Figure 6.90, which corresponds to no heat transfer between the outdoor and the exhaust air streams. On the other hand, for an ideal HRV unit with a 100% efficiency, the variation follows the solid red line, shown in Figure 6.90, which means that the outdoor air temperature is cooled down or heated up to the return air temperature. Since the indoor set temperature was kept at 25°C during the ventilation assisted tests, for an ideal HRV unit, the supply air temperature is expected to be at 25°C. However, as can be seen Figure 6.90, the real HRV unit operated around 40% efficiency.

As can be seen from Figure 6.91, the HRV unit has a slight dehumidification capacity of within 0.0005-0.001kg/kg for relatively low outdoor air humidity ratio and increases to around 0.002kg/kg with the increasing outdoor air humidity ratio.

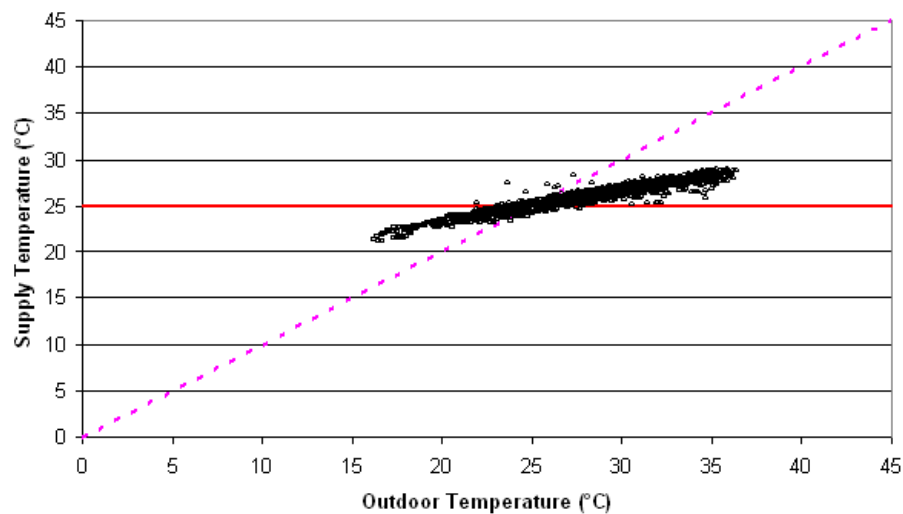


Figure 6.90 : Variation of the supply air temperature with respect to the outdoor air temperature

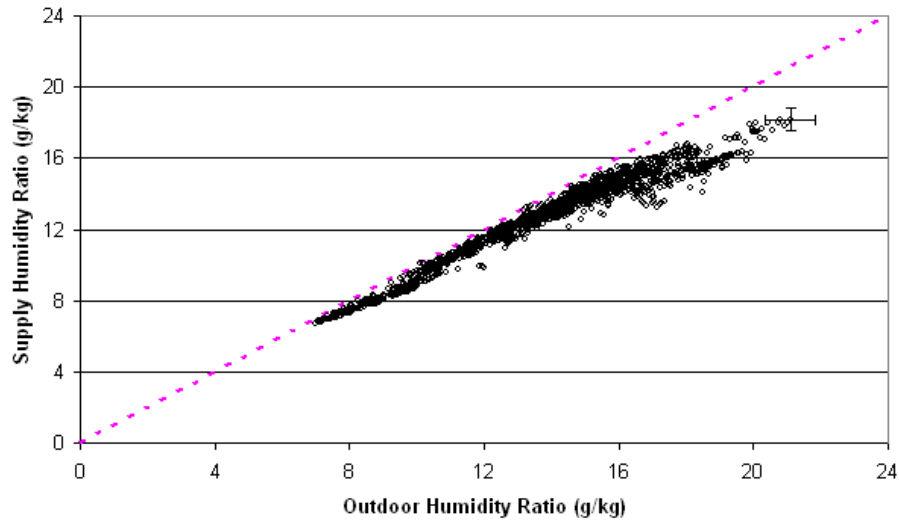


Figure 6.91 : Variation of the supply air humidity ratio with respect to the outdoor air humidity ratio

6.2.1 Seasonal comparison

Figure 6.92 and Figure 6.93 show the outdoor temperature and the relative humidity variations for the non-ventilated tests from 2006 and 2007 cooling seasons, respectively. The outdoor temperature and the relative humidity variations for the ventilation assisted tests were provided in Figure 6.1 and Figure 6.2, respectively.

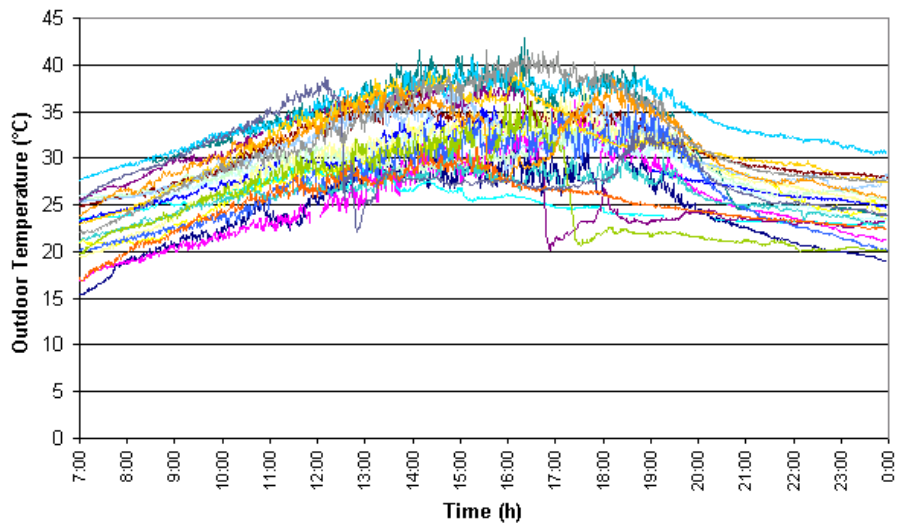


Figure 6.92 : Outdoor temperatures variations for the non-ventilated tests

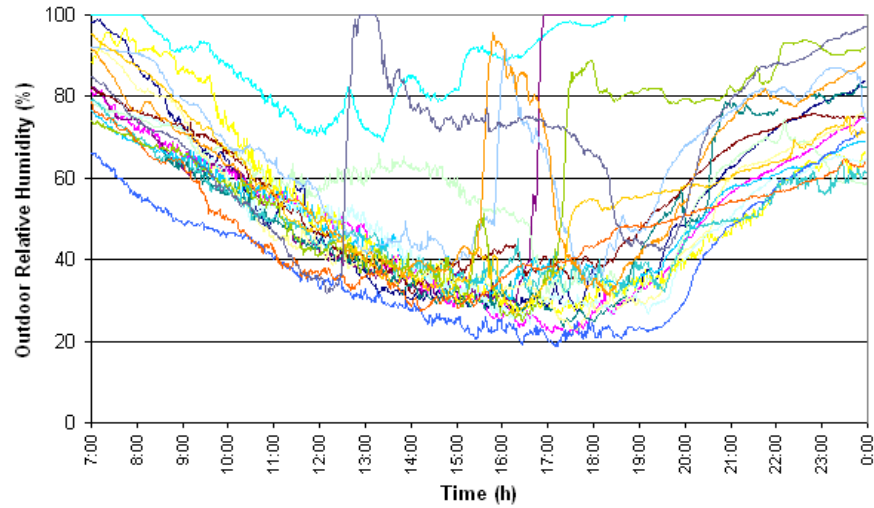


Figure 6.93 : Outdoor relative humidity variations for the non-ventilated tests

For a daily basis comparison, similar outdoor conditions between the ventilation assisted and the non-ventilated tests could not find. That's why; seasonal basis comparison was taken into account.

The number of data points of the outdoor temperatures for the ventilation assisted and non-ventilated multi-split VRF systems test and the comparison of the outdoor conditions are provided in Figure 6.94 and Figure 6.95, respectively.

As can be seen from Figure 6.94 and Figure 6.95, the outdoor conditions of the ventilation assisted and non-ventilated multi-split VRF systems days were fairly close, that's why; the effect of the ventilation can be evaluated seasonally.

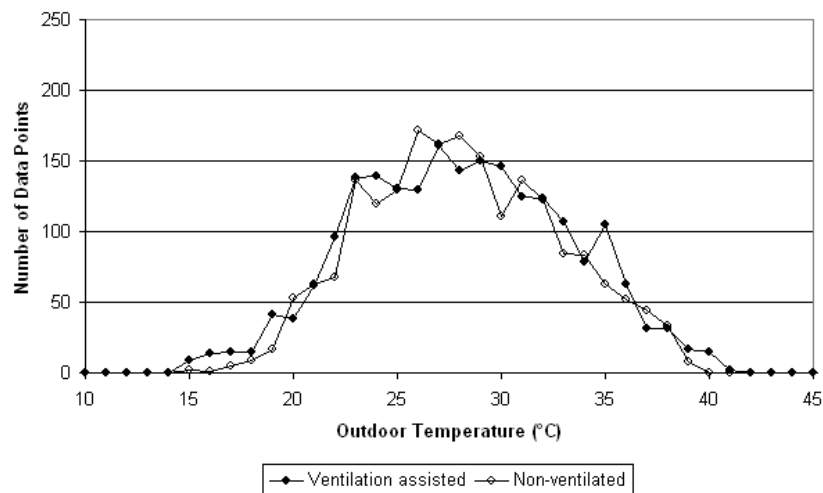


Figure 6.94 : Number of data points for the ventilation assisted and non-ventilated tests in cooling mode

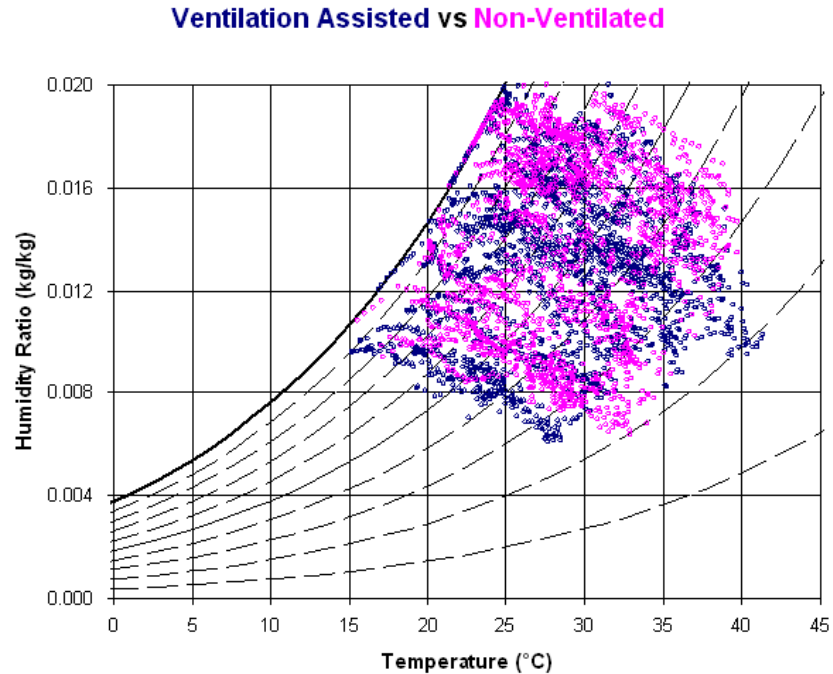


Figure 6.95 : Outdoor conditions for the ventilation assisted and non-ventilated tests in cooling mode

Figure 6.96a and Figure 6.96b show the indoor conditions of System1 zones provided by the ventilation assisted and non-ventilated systems, respectively.

As can be seen from Figure 6.96a and Figure 6.96b, since in both conditions, the multi-split VRF systems were controlled by the individual control mode with 25°C set temperature, the indoor temperatures were found at 25°C, which shows that ventilation did not affect the indoor temperature control. However, the main difference between the ventilation assisted and the non-ventilated systems comes from the indoor humidity ratio.

As can be seen from Figure 6.96a and Figure 6.96b, the indoor humidity ratio of System1 zones reached to 0.016kg/kg for the ventilation assisted operation which is around 0.002kg/kg higher than the non-ventilated operation.

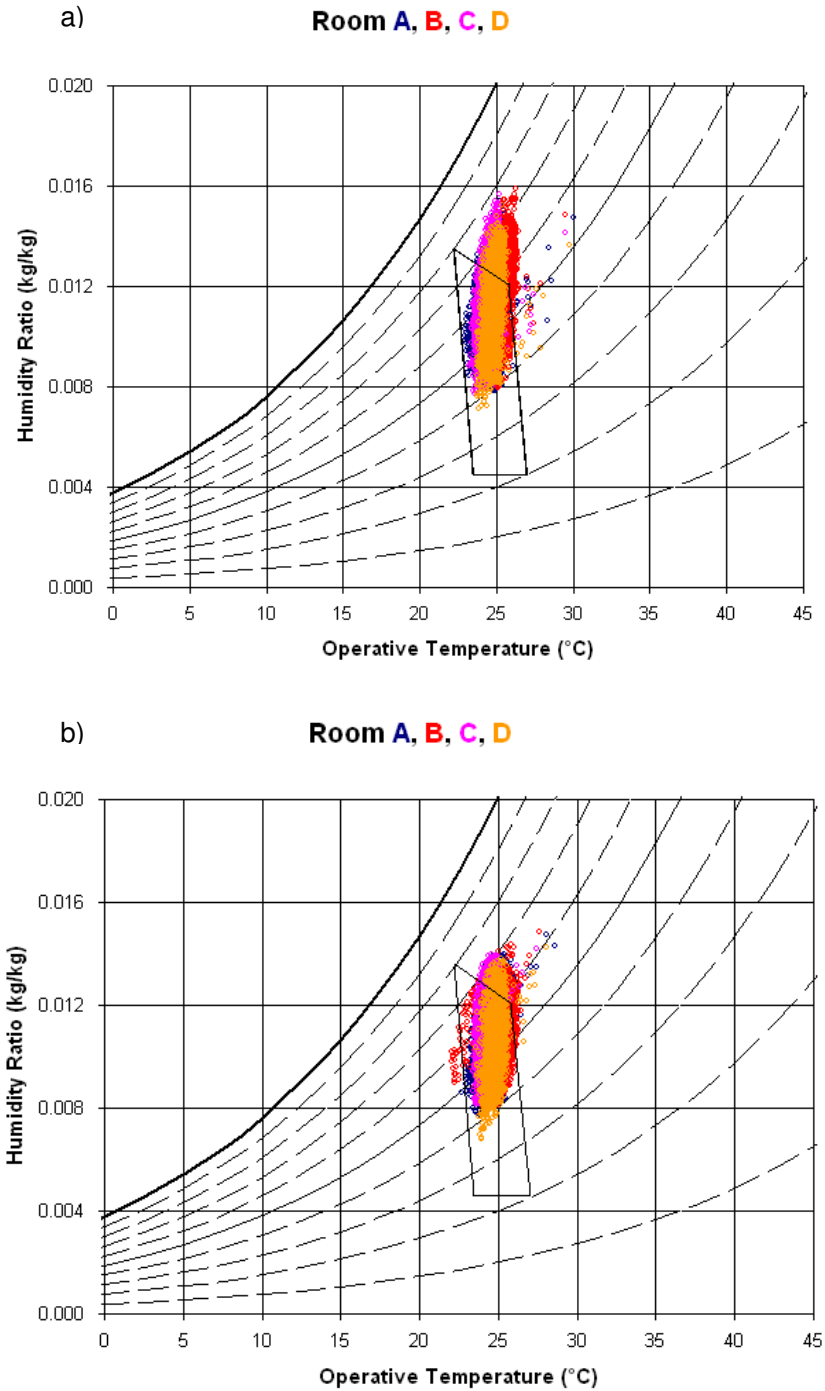


Figure 6.96 : Indoor conditions of System1 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated

Table 6.11 shows the comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone. As can be seen from, the non-ventilated system decreased the percentage of the data corresponded to the humid region,

and increased the percentage of the data in the ASHRAE summer comfort zone. That is, according to the ASHRAE summer comfort zone, the thermal comfort of System1 zones for the non-ventilated system was better than the ventilation assisted one.

Table 6.11 : Comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems at 25 °C set temperature based on the ASHRAE summer thermal comfort zone

Region	Percentage of the indoor condition data for ventilation assisted system	Percentage of the indoor condition data for non-ventilated system
In	66.8%	86.3%
Cold	0.1%	0.3%
Hot	1.7%	1.1%
Humid	31.4%	12.3%
Dry	0.0%	0.0%

Figure 6.97a and Figure 6.97b show the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems, respectively.

Table 6.12 shows the comparison of the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone. Similar to System1 zones, the non-ventilated system decreased the percentage of the data corresponded to the humid region, and increased the percentage of the data in the ASHRAE summer comfort zone. That is, according to the ASHRAE summer comfort zone, the thermal comfort of System2 zones for the non-ventilated system was better than the ventilation assisted one.

Table 6.12 : Comparison of the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems at 25 °C set temperature based on the ASHRAE summer thermal comfort zone

Region	Percentage of the indoor condition data for ventilation assisted system	Percentage of the indoor condition data for non-ventilated system
In	87.8%	94.9%
Cold	1.8%	1.4%
Hot	3.5%	1.2%
Humid	6.9%	2.5%
Dry	0.0%	0.0%

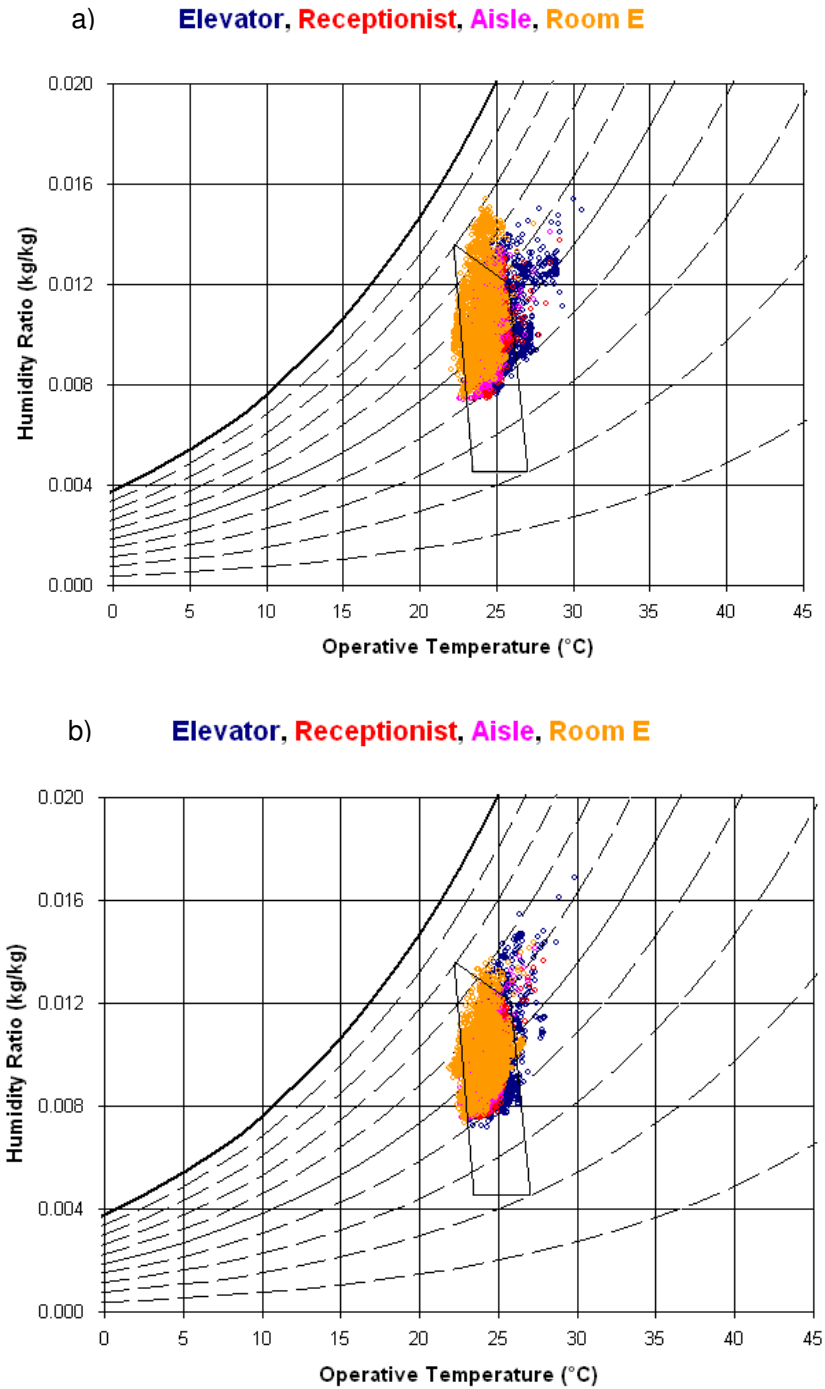


Figure 6.97 : Indoor conditions of OU2 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated

Figure 6.98 shows the TSS variations of System1 zones provided by the ventilation assisted and non-ventilated systems. As can be seen, the percentages of the TSS variations of System1 zones for the non-ventilated system were higher than that of

the ventilation assisted system, which shows that the non-ventilated assisted system provided better thermal comfort than the ventilation assisted system.

Figure 6.99 shows the TSS variations of System2 zones for the ventilation assisted and non-ventilated systems. As can be seen, there was not any significant difference between these two systems.

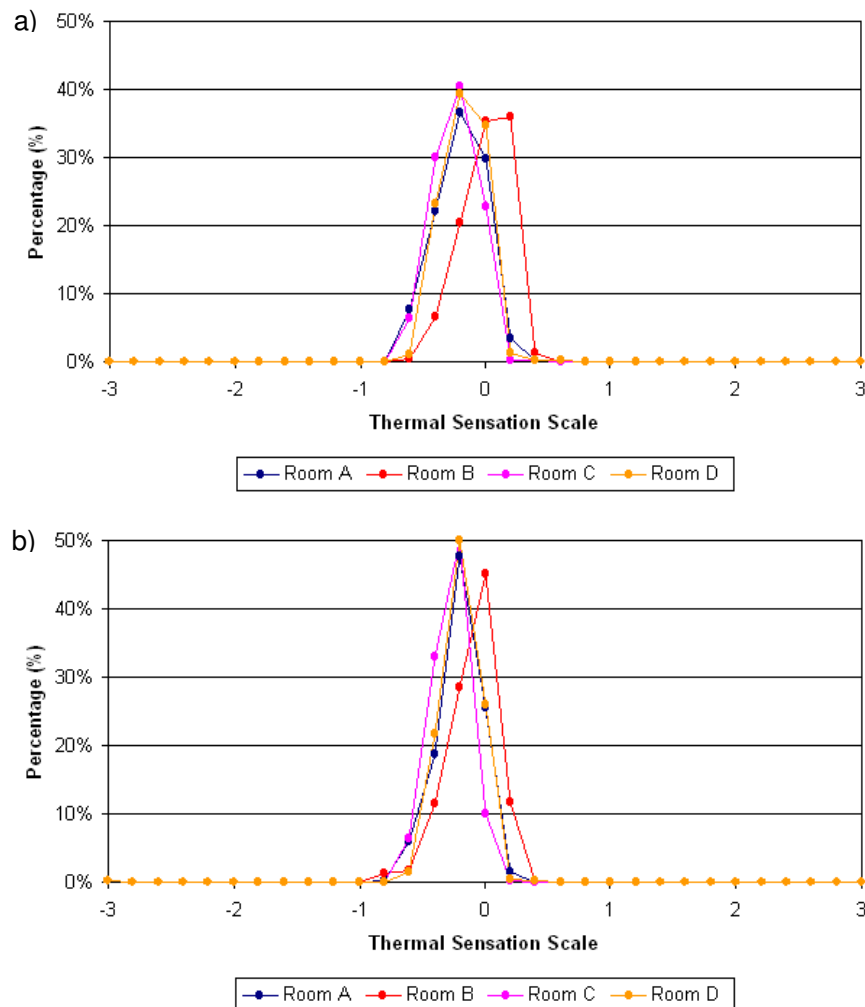


Figure 6.98 : TSS variations of System1 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated

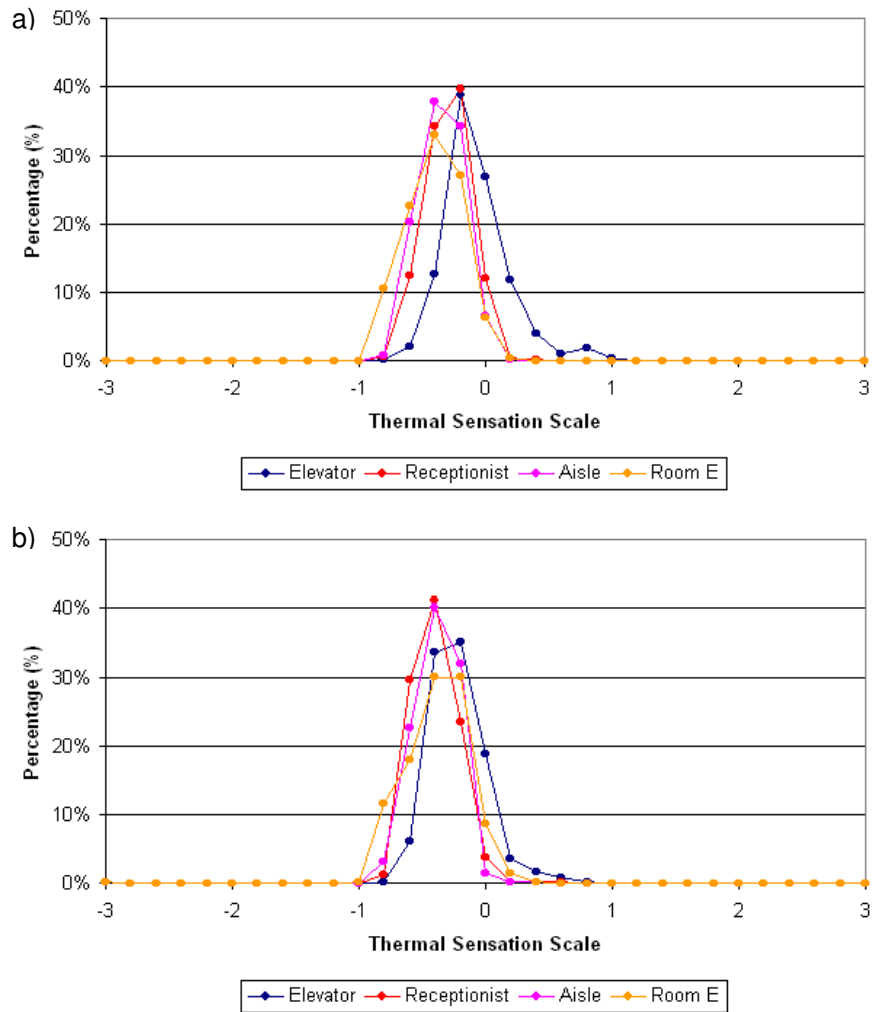


Figure 6.99 : TSS variations of System2 zones provided by the ventilation assisted and non-ventilated systems, (a) ventilation assisted, (b) non-ventilated

Figure 6.100 shows the TSS variations based on the combined zones for the ventilation assisted and non-ventilated systems. As can be seen, the TSS variations of both units for both systems were similar.

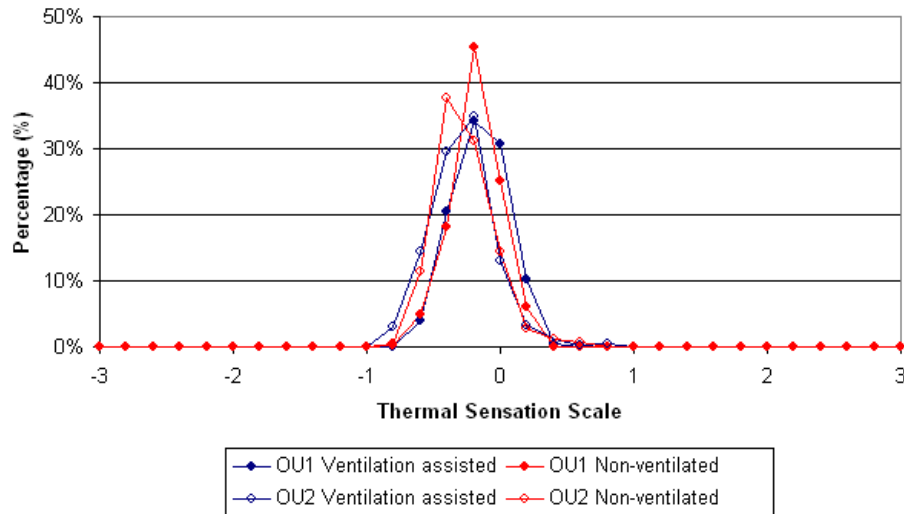


Figure 6.100 : TSS variations of System1 and System2 combined zones provided by the ventilation assisted and non-ventilated systems

Figure 6.101 shows the comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated multi-split VRF systems. Since the ventilation increased the indoor humidity ratio, shown in Figure 6.96a and Figure 6.97a, and introduced additional cooling load, it was expected that the ventilation assisted multi-split VRF system required more energy to keep the indoor temperatures at the set temperature. It was found from Figure 6.101 that the ventilation assisted system consumed around 24%, 15% and 18% higher energy for OU1, OU2 and total (OU1+OU2) compared to the non-ventilated case, respectively.

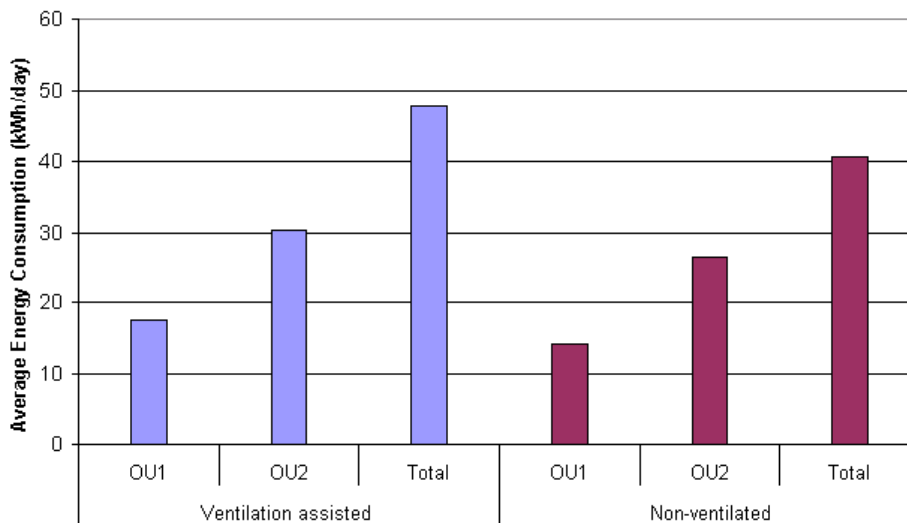


Figure 6.101 : Comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated systems

Figure 6.102 shows the variations of the percentage “ON” time ratios of OU1 and OU2 for the ventilation assisted and non-ventilated systems with respect to the outdoor temperature bins.

As can be seen from Figure 6.102a, even though the percentage of “ON” time ratio for the ventilation assisted system increased with the increasing outdoor temperature, the non-ventilated system did not follow that trend. This is because; the ventilated assisted system used the relatively low outdoor temperature to cool down the indoors, on the other hand, the non-ventilated system needs to take care of the internal load. This causes higher “ON” time ratio for the non-ventilated system compared to ventilation assisted system at low outdoor temperature, shown in Figure 6.102.

On the other hand, for outdoor temperatures higher than the set temperature of 25°C, the ventilation introduced more cooling load to the indoors, which ended with higher “ON” time ratios compared to the non-ventilated system.

Since the outdoor temperature was mostly higher than the set temperature for Maryland conditions, the benefit of the ventilation on the power consumption of the multi-split VRF system was little and that’s why; the ventilation assisted system required more energy to keep the indoor temperatures at the set temperature.

This case is valid for OU2, as shown in Figure 6.102b.

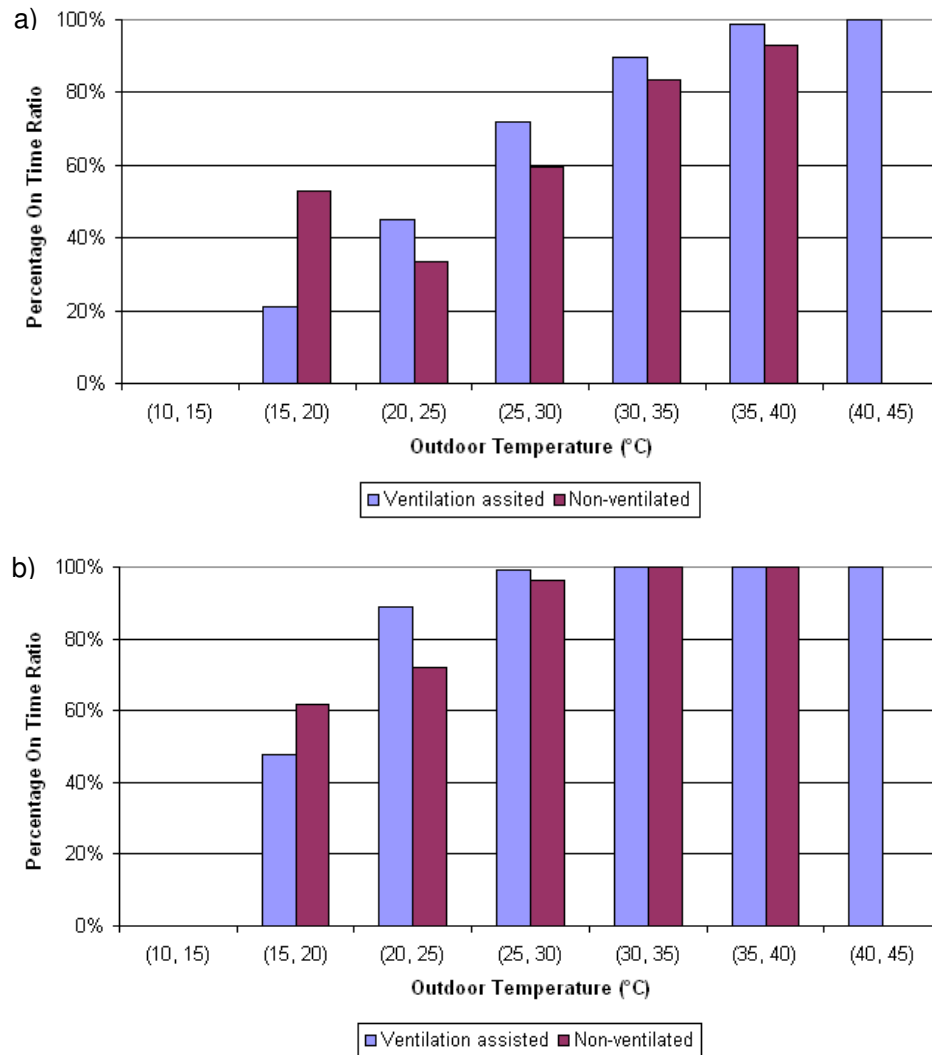


Figure 6.102 : Percentage ON time ratio of OU1 and OU2 for the ventilation assisted and non-ventilated systems, (a) OU1, (b) OU2

Variation of daily energy consumptions with respect to the daily averaged outdoor temperatures for the ventilation assisted and non-ventilated multi-split VRF systems are provided in App. [J].

The provided daily cooling energy, daily energy consumption of OU2 and corresponding CPF and PLR are plotted in Figure 6.103 with respect to the daily averaged outdoor temperature for both ventilation assisted and non-ventilated systems. The trends are also shown with least square regression lines.

As can be seen from Figure 6.103c, the ventilation assisted system required more cooling compared to the non-ventilated system. Besides, it was observed that the ventilation did not affect the efficiency of the system, as shown in Figure 6.103d.

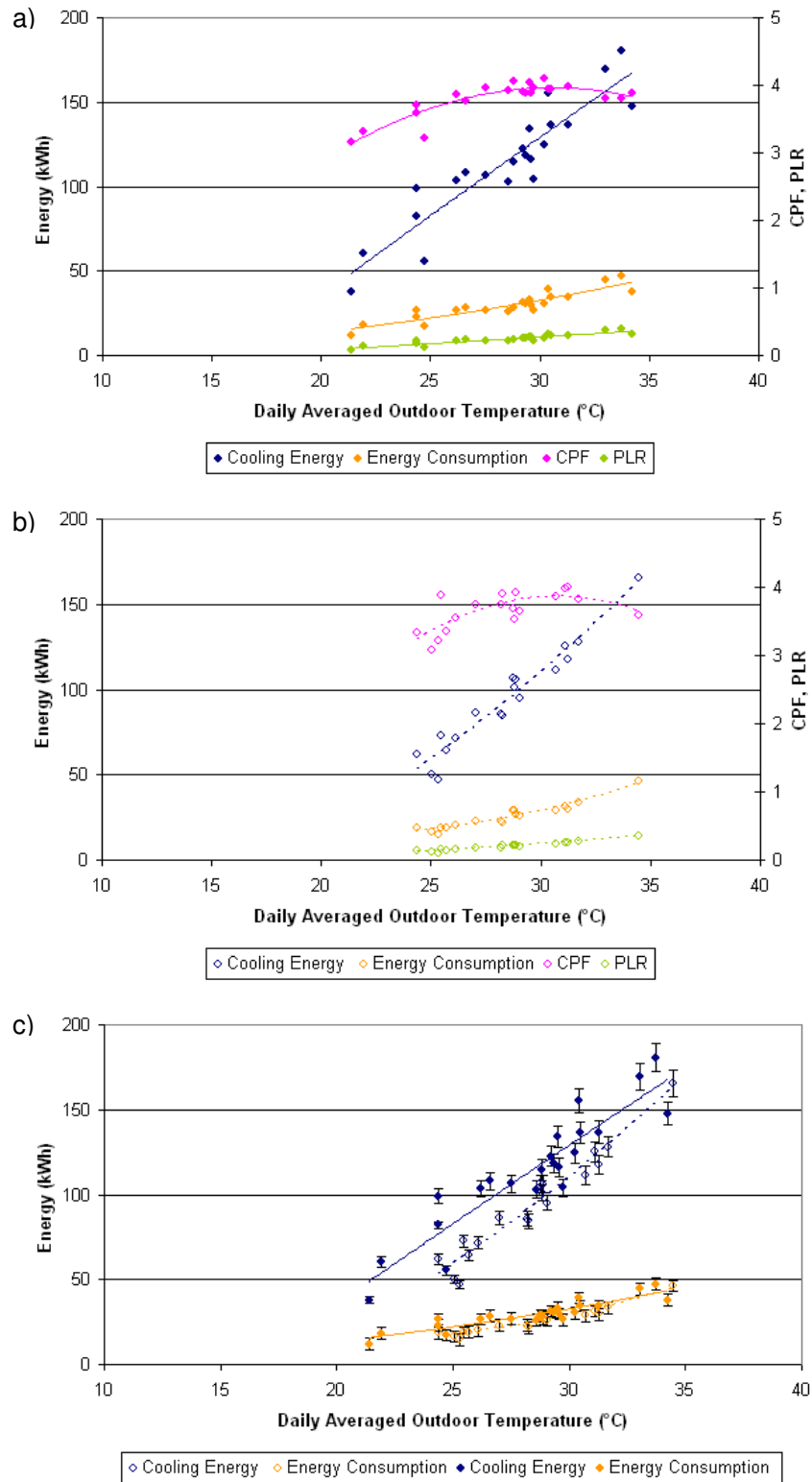


Figure 6.103 : Variation of cooling energy, energy consumption of OU2 and CPF of the ventilation assisted and non-ventilated system with respect to the daily averaged outdoor temperature, (a) ventilation assisted, (b) non-ventilated, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF (continued)

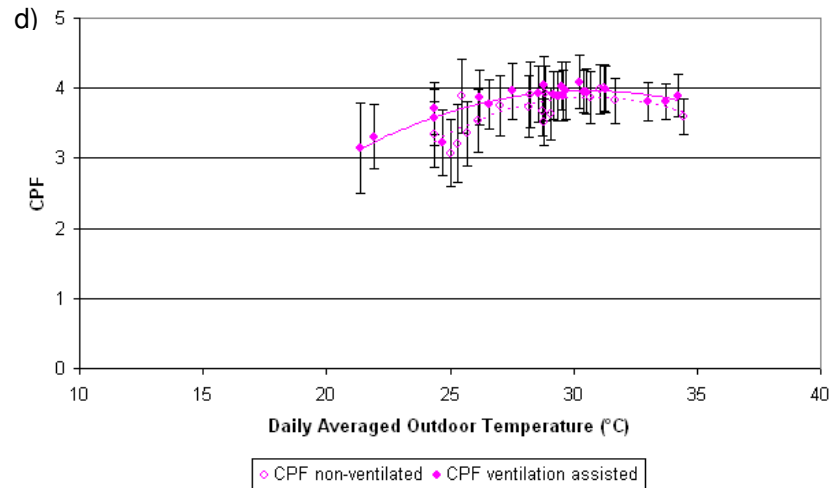


Figure 6.103 : Variation of cooling energy, energy consumption of OU2 and CPF of the ventilation assisted and non-ventilated system with respect to the daily averaged outdoor temperature, (a) ventilation assisted, (b) non-ventilated, (c) comparison of the cooling energy and the energy consumption, (d) comparison of CPF

As shown in Figure 6.104, the seasonal “ON” time for the non-ventilation system was lower than the ventilation assisted system, which increased the cyclic operation resulting lower efficiency. Since the non-ventilated system required less cooling, the fixed speed compressor did not work during the operation. As shown in Figure 6.105, 99.5% of all operational time, the fixed speed compressor was OFF.

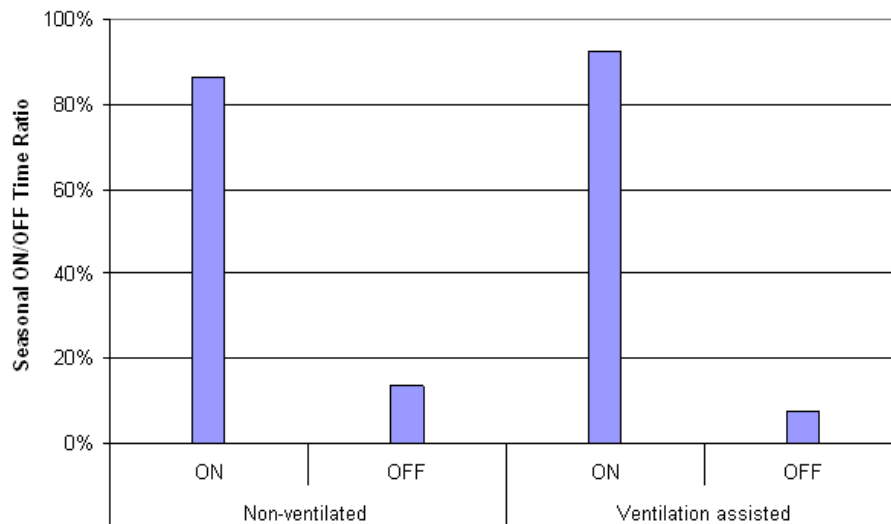


Figure 6.104 : Seasonal ON/OFF time ratio of OU2 for the ventilation assisted and non-ventilated systems

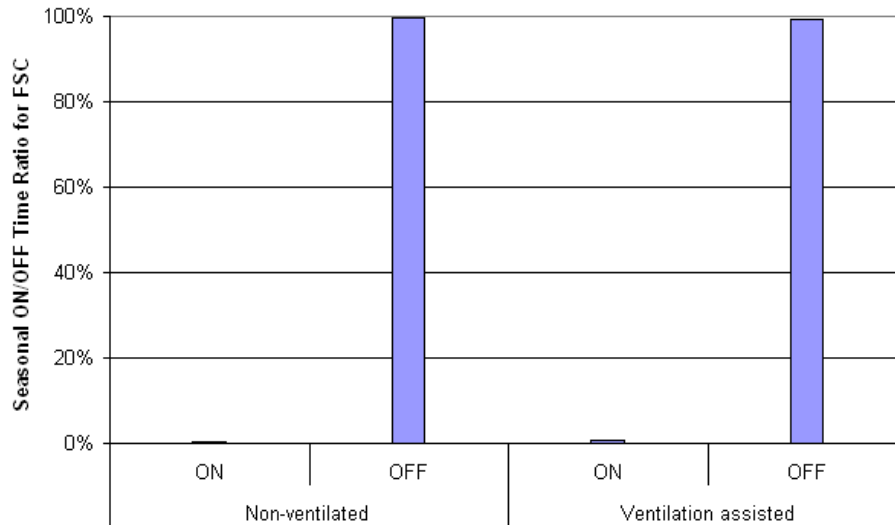


Figure 6.105 : Seasonal ON/OFF time ratio of FSC of OU2 for the ventilation assisted and non-ventilated systems

6.3 Effect of the Control Modes in the Heating Performance

The multi-split VRF system described in Chapter 4.2 can be operated in the heating mode to provide warm air to the indoors by changing the refrigerant path with the four-way valve located in the outdoor unit.

Figure 6.106 shows the refrigerant path in the heating mode. As can be seen, after shifting to the heating mode, EEVs were no longer in the upstream of the indoor units. This causes a potential risk of refrigerant and/or oil accumulation in the indoor units. In order to prevent this risk, each EEV was kept open at the minimum opening during the “ON” time of the outdoor unit. This is the main difference between the cooling and heating operation of the multi-split VRF system.

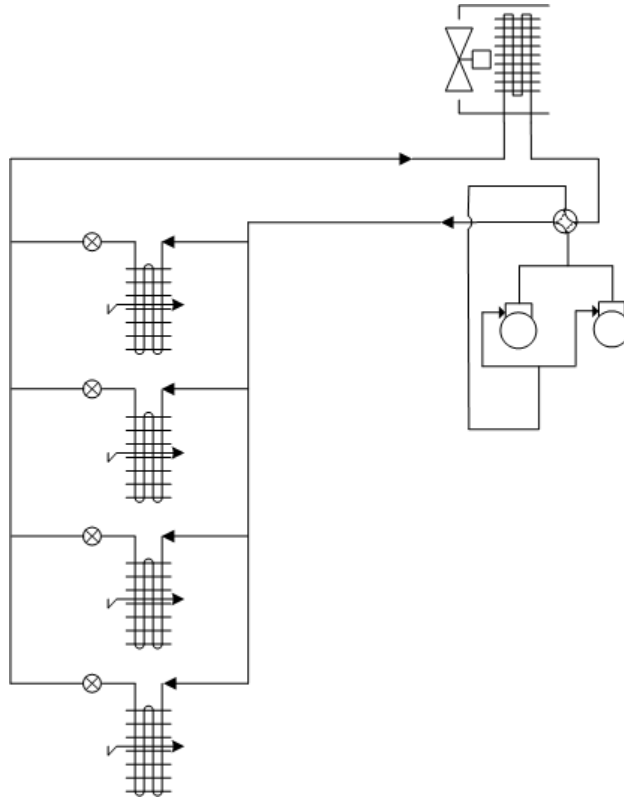


Figure 6.106 : Refrigerant path in the heating mode

Two different indoor unit fan operations; continuous and synchronized were tested in the heating mode.

In the continuous indoor fan operation, similar to the cooling mode operation, the indoor unit fans were kept working continuously throughout the test regardless of the indoor unit operation. On the other hand, in the synchronized indoor unit fan operation, the fans were operated only when the thermostats of the indoor units sent “ON” signal to the indoor unit which means the indoor temperature was lower than the set temperature, and the zone required heating.

Since EEVs were kept open at least at the minimum opening during the “ON” time of the outdoor unit, the continuous indoor unit fan operation provided heating to all zones which had the potential of overheating of the zones. On the other hand, for the synchronized indoor unit fan operation, the potential of the overheating of the zones would be eliminated.

6.3.2 Daily comparison at 23.3°C set temperature

Similar outdoor conditions were tried to be found for the daily comparison. Two weekdays; February 19, 2007 and March 08, 2007 were found with similar outdoor conditions. The multi-split VRF systems were operated in individual control mode on

February 19, 2007 and in master control mode on March 08, 2007 with same 23.3°C indoor set temperatures.

Figure 6.107 and Figure 6.108 show the outdoor temperature and relative humidity variations on May 31, 2007 and June 01, 2007. Data were sampled every minute. As seen from Figure 6.107 and Figure 6.108, the temperature variations of both days were similar until 21:00, and the relative humidity trends were similar after 9:00.

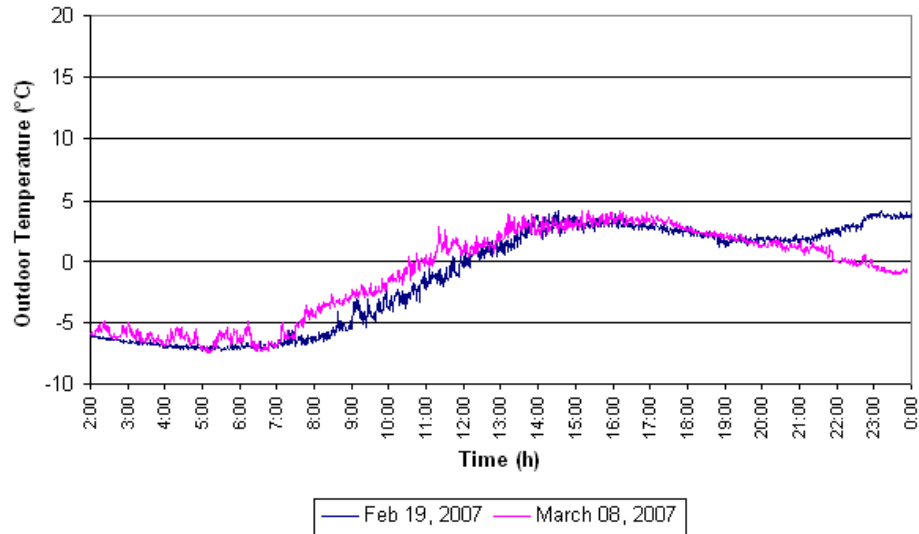


Figure 6.107 : Outdoor temperature variations on February 19, 2007 and March 08, 2007

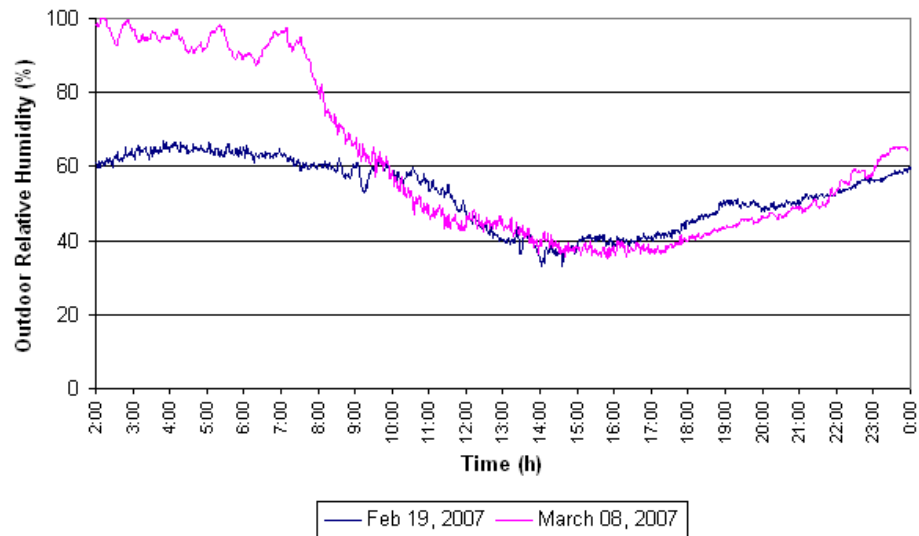


Figure 6.108 : Outdoor relative humidity variations on February 19, 2007 and March 08, 2007

Figure 6.109 and Figure 6.110 depict the indoor temperatures of System1 zones and the operation of indoor units of System1 zones for the individual and master control modes, respectively.

As seen from Figure 6.109a and Figure 6.110a, since the indoor unit fans were operated continuously, each indoor temperature was affected by the other indoor unit operation. That is, even though, only IURA was working from 15:00 to 20:00 cyclically, indoor temperatures of the other zones were affected from the operation of IURA.

On the other hand, as seen from Figure 6.109b and Figure 6.110b, since all indoor units operated simultaneously in the master control mode, similar to the cooling performance tests, all the indoor temperatures increased and decreased simultaneously.

As can be from Figure 6.109a and Figure 6.109b, the magnitude of the indoor temperature fluctuation of the master control mode was higher than that of the individual control mode.

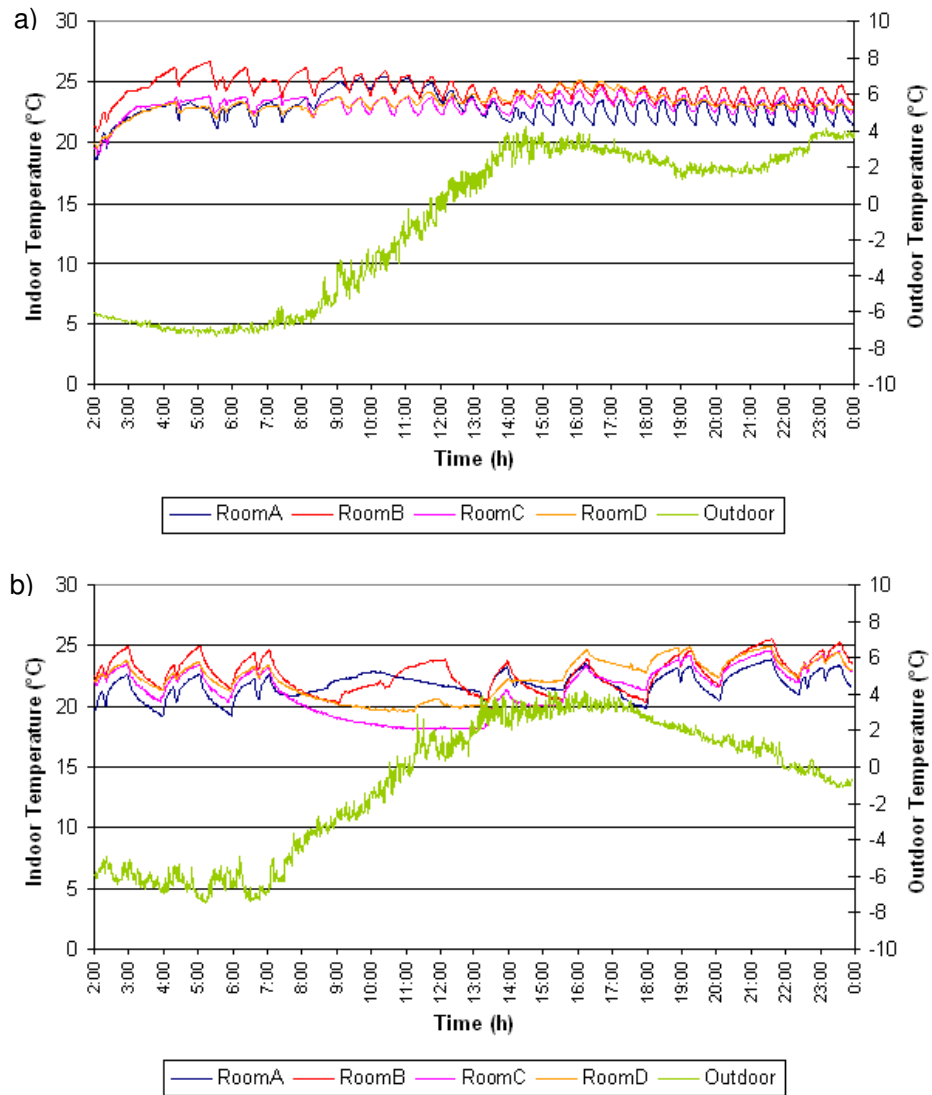


Figure 6.109 : Outdoor temperature and indoor temperatures of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

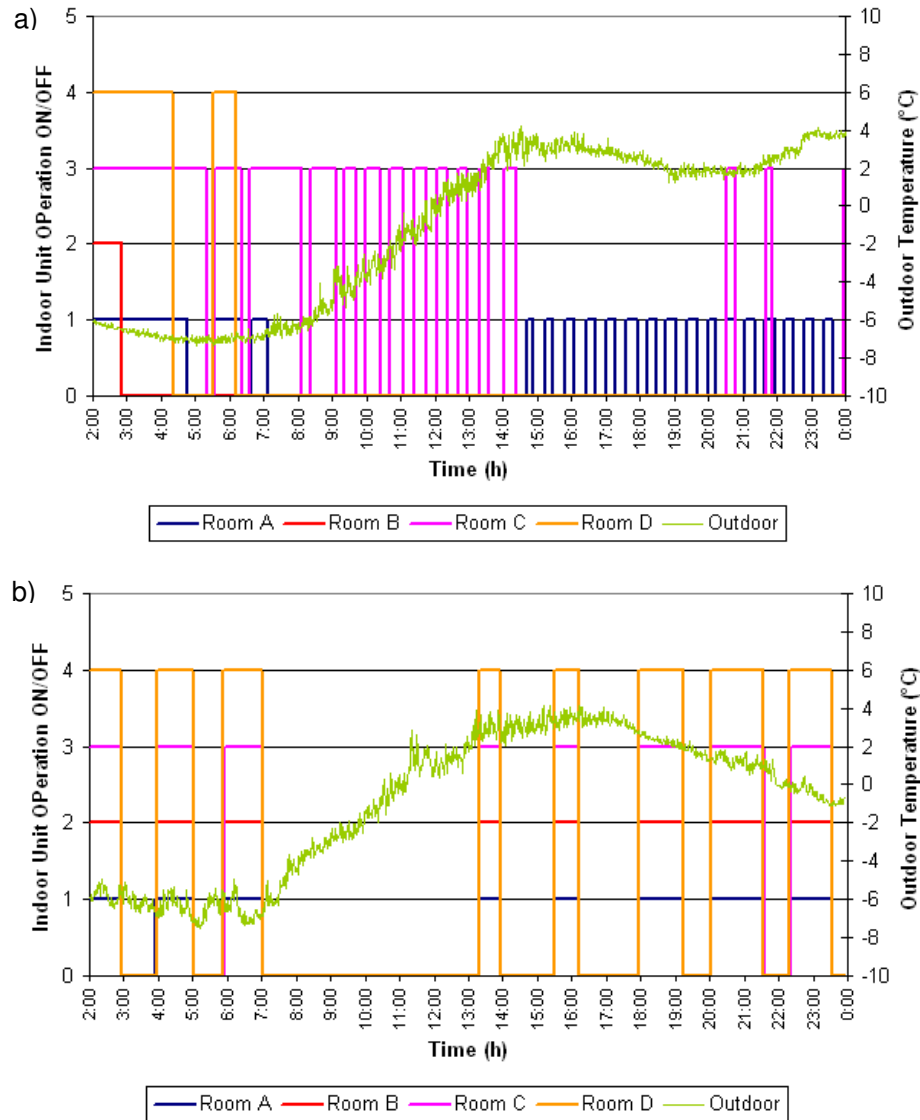


Figure 6.110 : Operation of the indoor units of System1 zones for the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.111 and Figure 6.112 depict the indoor temperatures of System2 zones and the operation of indoor units of System2 zones for the individual and master control modes, respectively.

As can be seen from Figure 6.111a and Figure 6.112a, the indoor temperature of the elevator zone was the lowest among System1 zones and required heating all day, and after 7:30am, due to the continuous fan operation, the indoor temperatures of the other zones increased.

Since the multi-split VRF system in the individual control mode tried to keep the indoor temperature of the elevator zone at the set temperature, it worked much longer than the master control mode, which caused higher energy consumption.

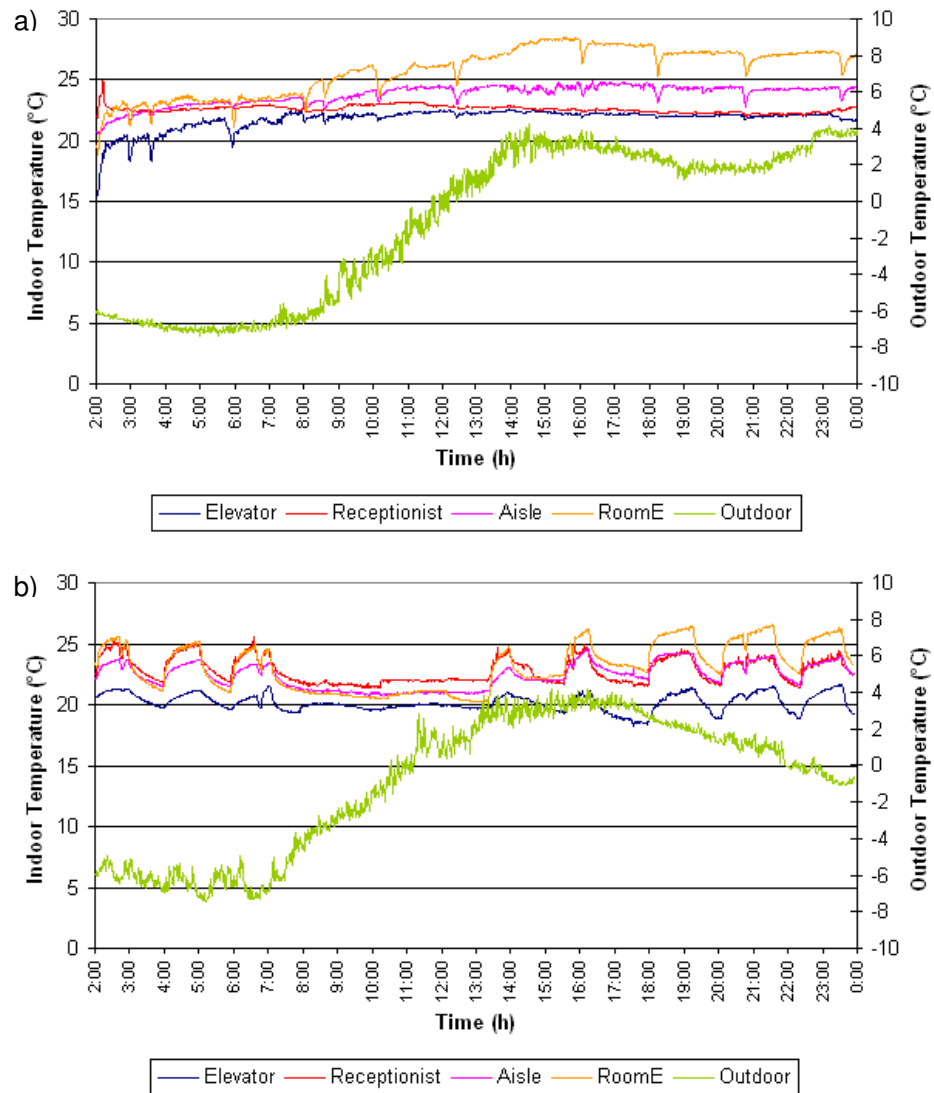


Figure 6.111 : Outdoor temperature and indoor temperatures of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

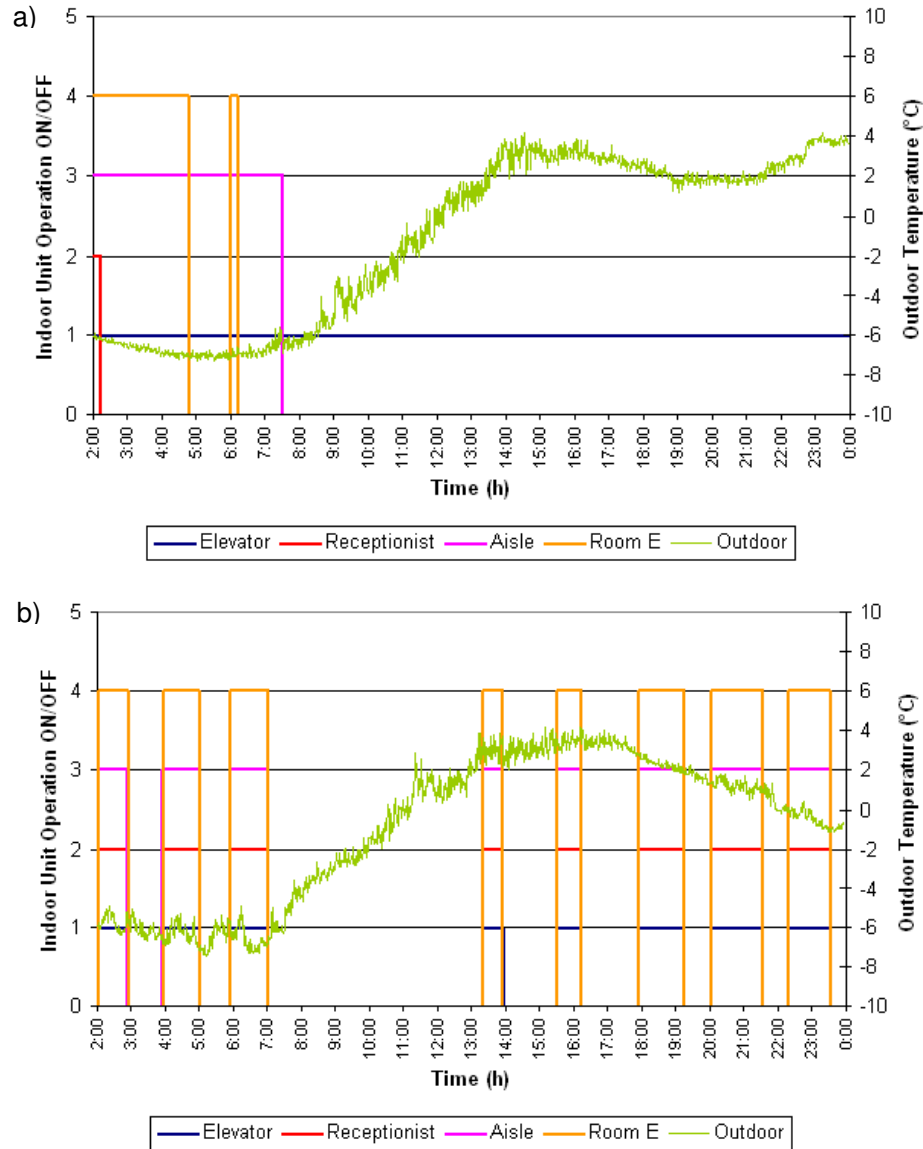


Figure 6.112 : Operation of the indoor units of System2 zones for the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.113 shows the comparison of the power consumption of OU1 and OU2 for the individual and master control modes, respectively.

As can be seen, in order to keep the indoor temperatures at the set temperature, the individual control mode consumed more energy than the master control mode.

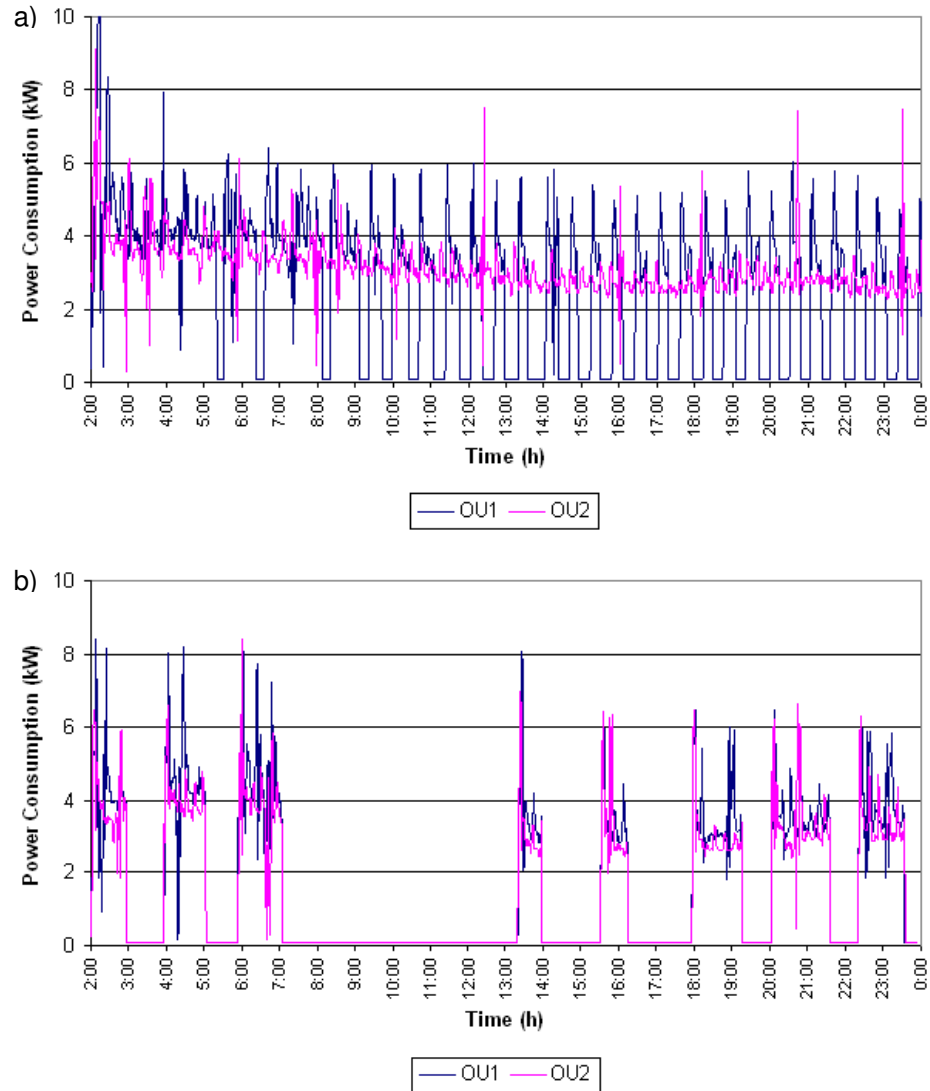


Figure 6.113 : Comparison of the power consumptions of OU1 and OU2 for the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.114 shows the comparison of the heating capacity, power consumption of OU2 and HPF of OU2 for the individual and master control modes, respectively.

As can be seen, the individual control mode provided more heating than the master control mode while consuming more power, in order to keep the indoor temperature of the elevator zone at the set temperature, shown in Figure 6.111a.

Since the initial indoor temperatures were much lower than the set temperature, the individual control mode provided more heating until 8:00, and since the indoor temperature of elevator zone was reaching to the set temperature, the provided heating capacity decreased to around 7kW.

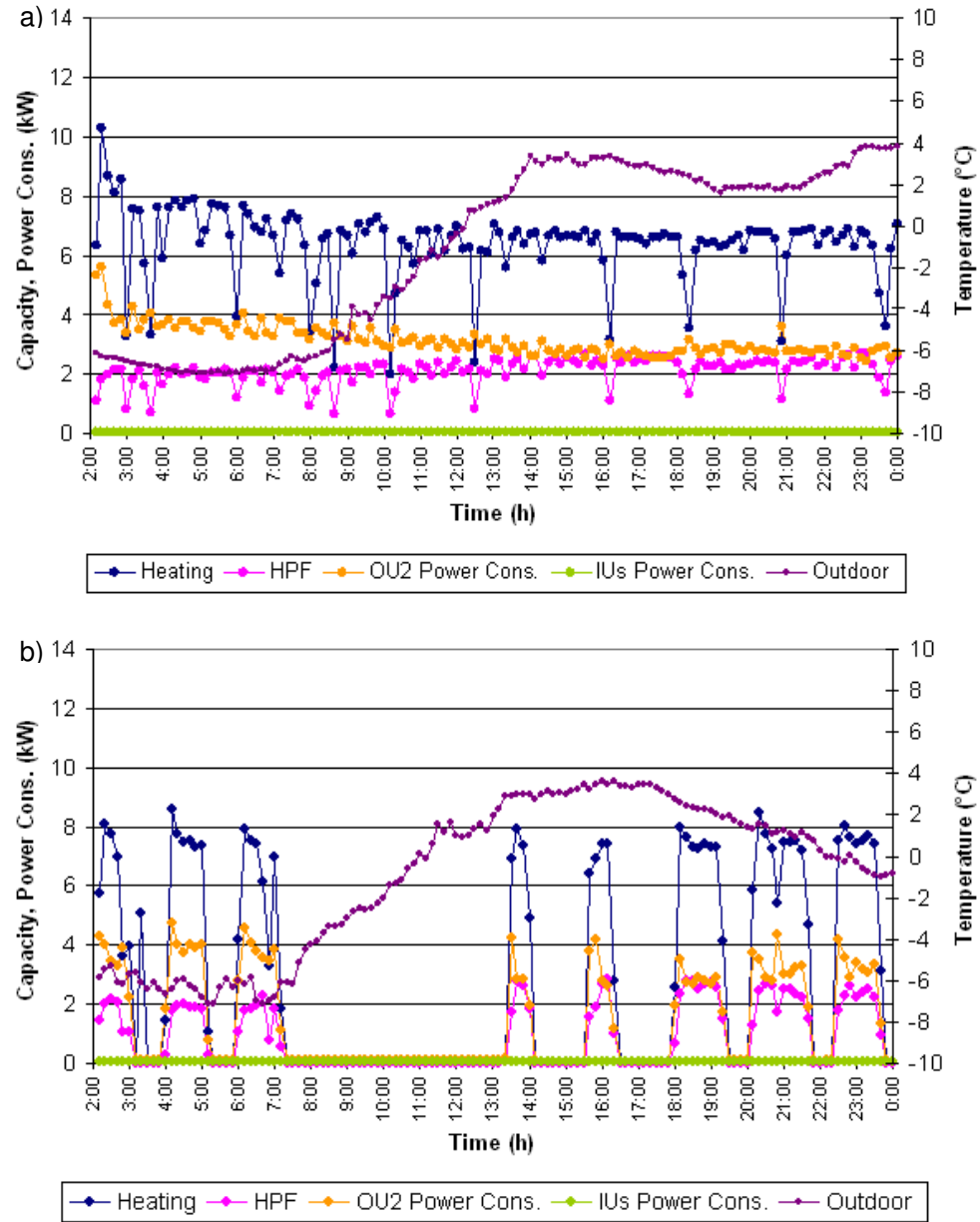


Figure 6.114 : Variation of the heating capacity, outdoor unit power consumption, total power consumption of the indoor units and HPF of OU2 for the individual and master control modes, (a) individual control mode (top), (b) master control mode (bottom)

The drastic reductions of the heating capacity at some points during the test, shown in Figure 6.114a, were due to the defrosting, which required shifting to the cooling mode. The indoor unit air discharge temperatures for the individual control mode are plotted in Figure 6.115, which also shows the sharp decrease of the air temperatures during the defrosting process. Even though the detected air blowing temperature in the indoor unit showed sharp decrease during the defrosting process, it was not supplied to the indoors, because all the indoor fans were

stopped and the louvers of the indoor units were closed during the defrosting process. Thus, the cold air, trapped in each indoor unit, was not discharged to the corresponding zones. That's why; even though the heating capacity degraded during the defrosting, it did not affect the indoor comfort.

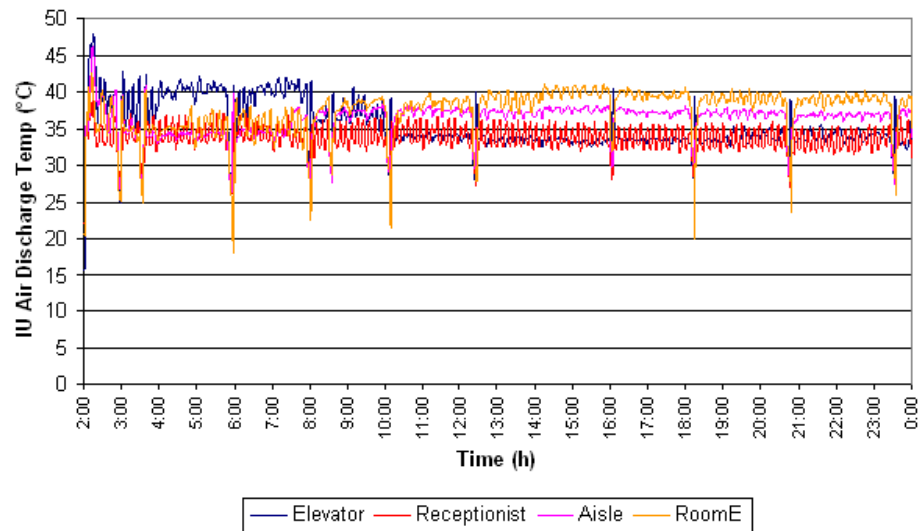


Figure 6.115 : Indoor unit air discharge temperatures of System2 zones on February 19, 2007

6.3.3 Seasonal comparison at 23.3°C set temperature

Figure 6.116 and Figure 6.117 show the number of data points of the outdoor temperature and the outdoor conditions for the individual and master control modes in the heating performance tests at 23.3°C set temperature, respectively. As seen, the individual and master control mode tests were performed under similar outdoor conditions.

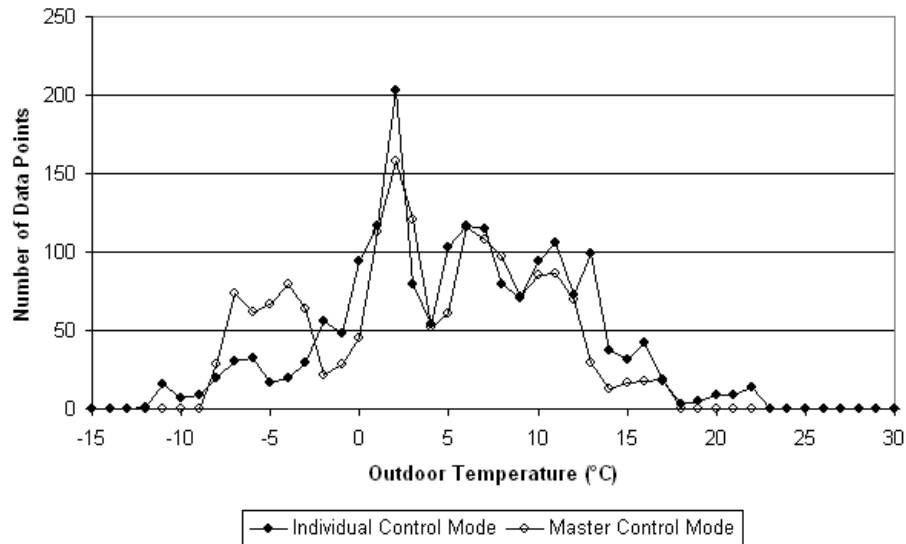


Figure 6.116 : Number of data points for 23.3°C set temperature

Individual Control Mode vs Master Control Mode

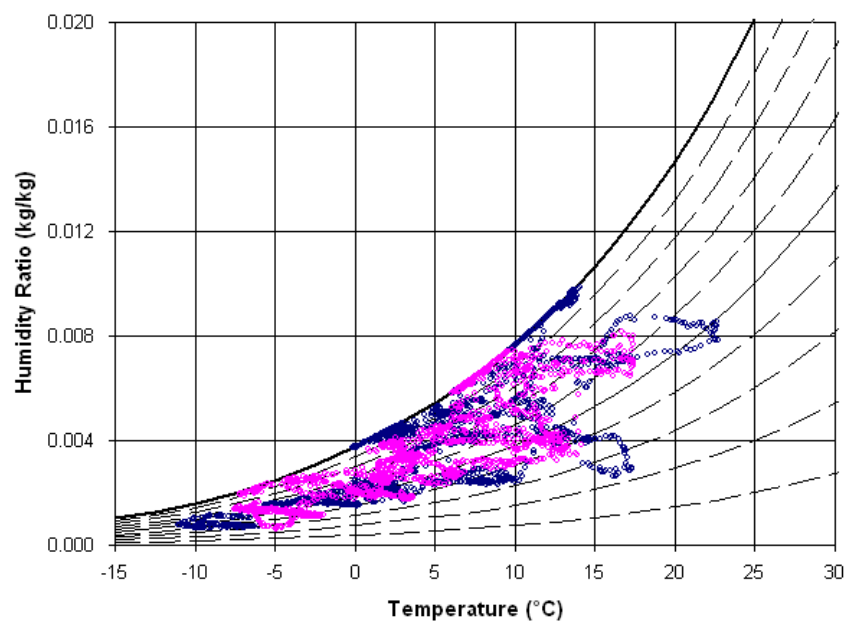


Figure 6.117 : Outdoor conditions for 23.3°C set temperature tests

As can be seen from Figure 6.117, most of the outdoor humidity ratios were lower than 0.0045kg /kg which is the lower limit of the humidity ratio defined in the ASHRAE winter thermal comfort zone, provided in Figure 5.3. Since the multi-split VRF system or the HRV system did not have any humidification capacity, the expected indoor humidity ratio will be around the outdoor humidity ratio range. That's why; the dry zone of the ASHRAE winter thermal comfort zone (region lower

than 0.0045kg/kg humidity ratio) was divided into three sub regions; cold dry, dry and hot dry; shown in Figure 6.118, in order to evaluate the percentage of the indoor condition data lower than 0.0045kg /kg humidity ratio.

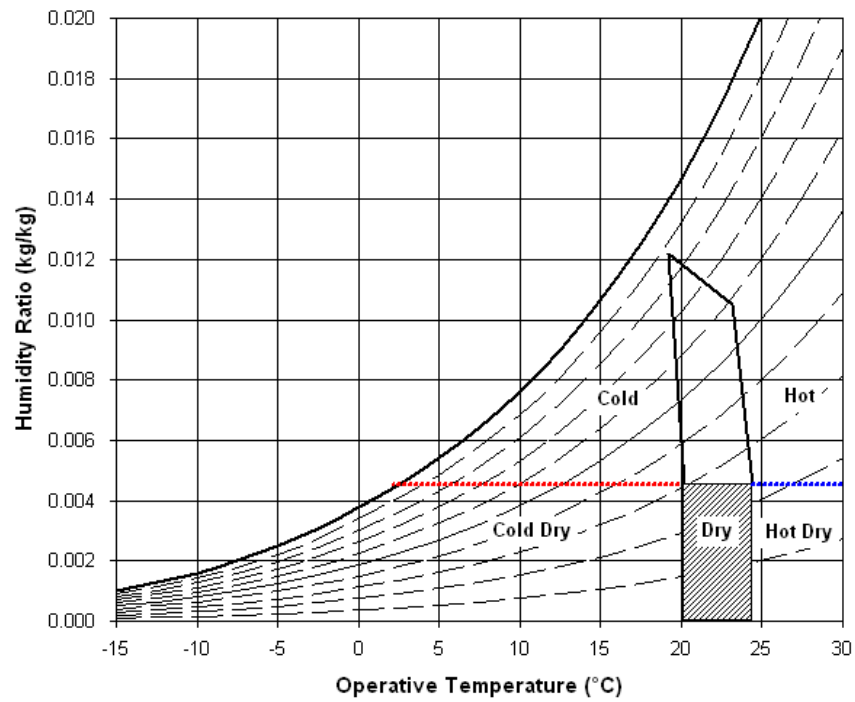


Figure 6.118 : Sub regions of the ASHRAE winter thermal comfort zones

Figure 6.119 shows the indoor condition data of System1 zones provided by the individual and master control modes at 23.3°C set temperature under the continuous indoor fan operation.

As can be seen from Figure 6.119a and Figure 6.119b, as expected, most of the indoor condition data were found in the dry region due to the lower outdoor humidity ratio. On the other hand, it was found the individual control mode provided relatively warmer indoor environment than the master control mode. However, due to the continuous indoor fan operation, some of the indoor condition data were found even in the hot region.

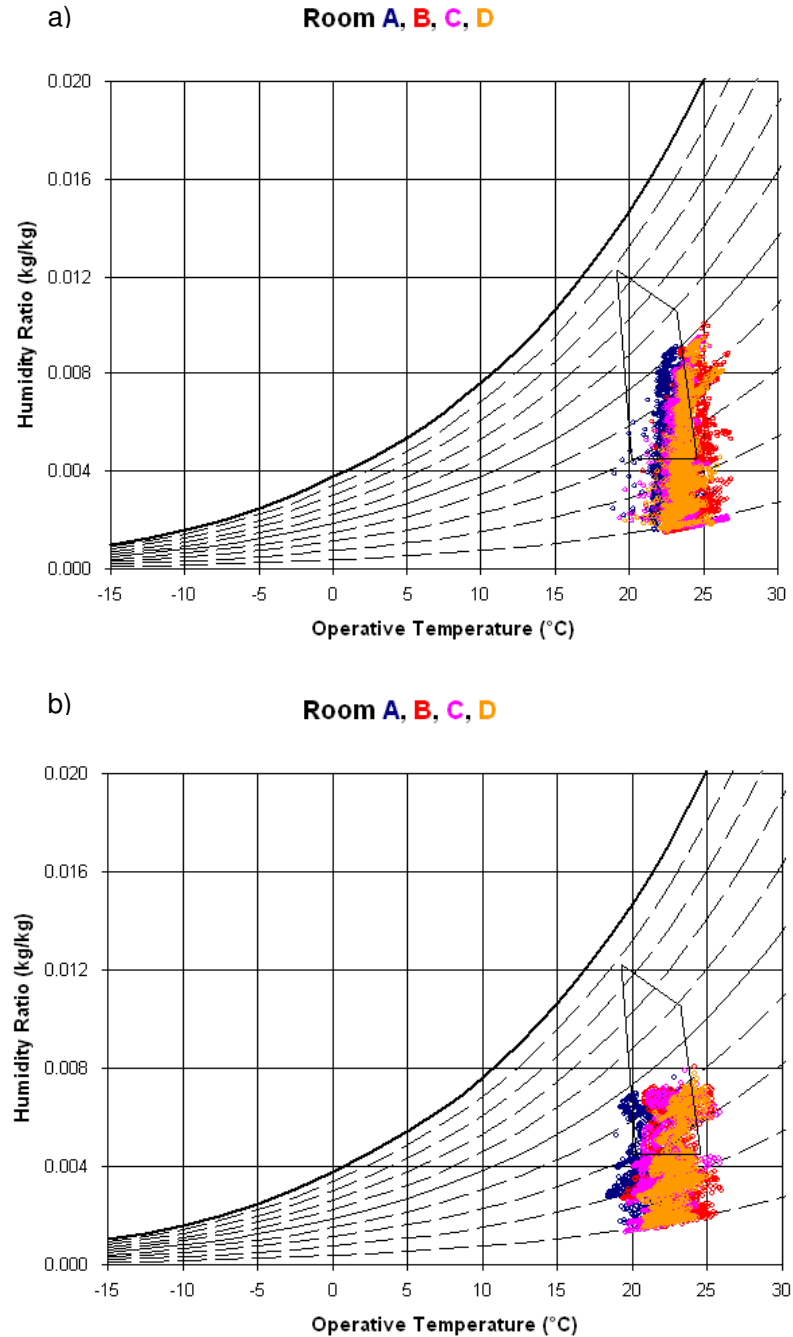


Figure 6.119 : Indoor conditions of System1 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Table 6.13 shows the comparison of the indoor condition data of System1 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature. It was found that, the percentage of the indoor condition data for the individual control mode in the ASHRAE winter thermal comfort zone was higher than that of the indoor condition

data for the master control mode, and besides, the master control mode provided much colder indoor environment than the individual control mode.

Table 6.13 : Comparison of the indoor condition data of System1 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the individual control mode	Percentage of the indoor condition data for the master control mode
In	30.4%	23.4%
Cold	0.1%	2.6%
Hot	9.7%	2.5%
Cold Dry	0.1%	4.5%
Dry	52.2%	65.0%
Hot Dry	7.5%	2.0%
Humid	0.0%	0.0%

Figure 6.120 shows the indoor condition data of System2 zones provided by the individual and master control modes at 23.3°C set temperature under the continuous indoor fan operation. As can be seen from Figure 6.120a and Figure 6.120b, similar to System1 zones, the master control mode provided relatively colder indoor environment than the individual control mode, and due to the continuous indoor unit fan operation, some of the indoor condition data, especially Room E, provided by the individual control mode were found in the hot region.

Table 6.14 shows the comparison of the indoor condition data of System2 zones provided by the individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature. As can be seen, the percentage of the indoor condition data for the individual control mode in the ASHRAE winter thermal comfort zone was slightly higher than that of the indoor condition data for the master control mode, while the master control mode provided colder indoor environment.

Table 6.14 : Comparison of the indoor condition data of System2 zones provided by individual and master control modes based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the individual control mode	Percentage of the indoor condition data for the master control mode
In	21.9%	21.6%
Cold	0.2%	2.6%
Hot	19.2%	4.4%
Cold Dry	0.4%	10.6%
Dry	44.3%	58.0%
Hot Dry	14.0%	2.8%
Humid	0.0%	0.0%

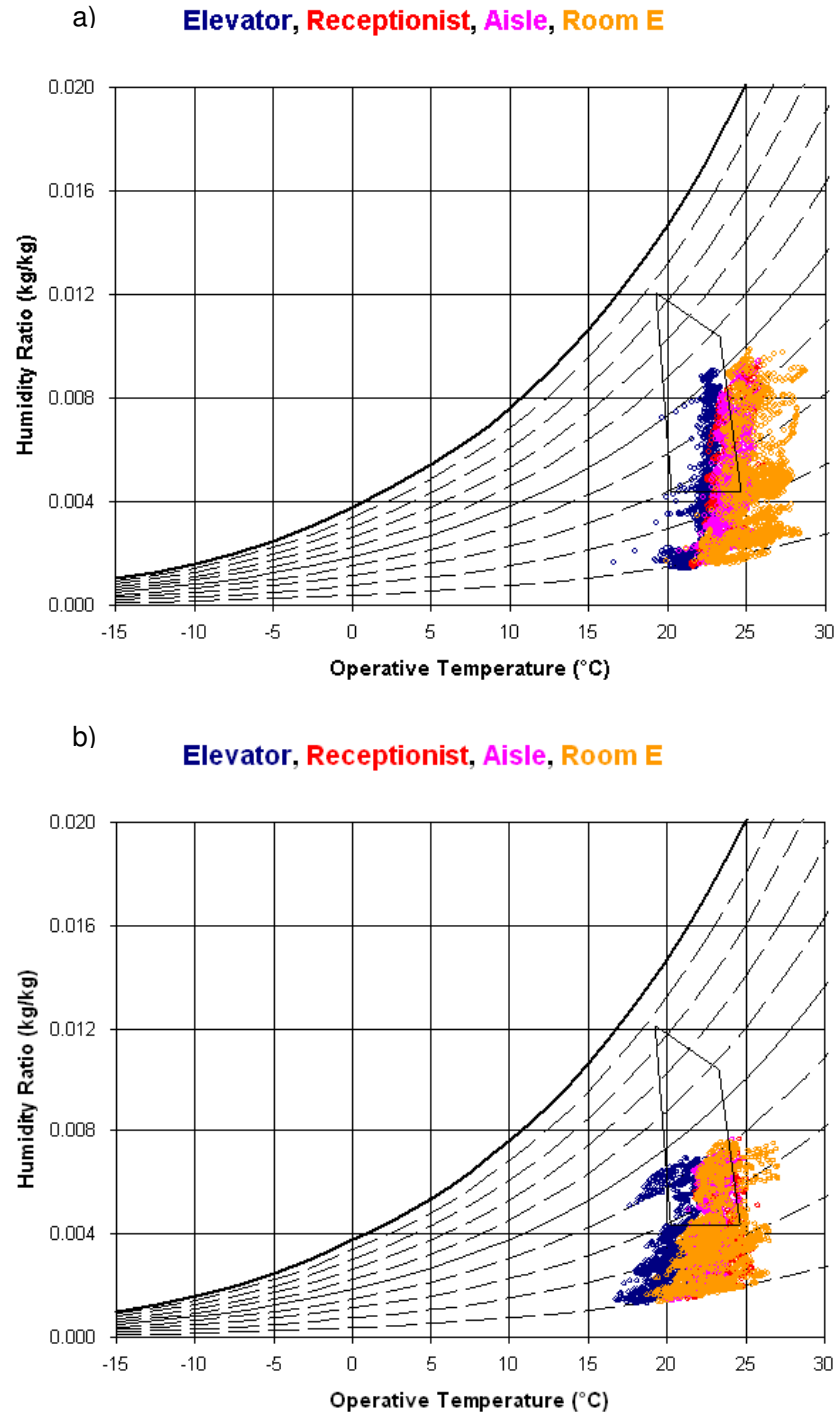


Figure 6.120 : Indoor conditions of System2 zones provided by the individual and master control modes, (a) individual control mode, (b) master control mode

Figure 6.121 and Figure 6.122 show the TSS variations of System1 and System2 zones provided by the individual and master control modes, respectively.

As can be seen, similar to the cooling performance results, the percentages of TSS of the individual control mode were higher than that of the master control mode which means that the indoor temperature fluctuations due to the individual control mode were much less than the master control mode.

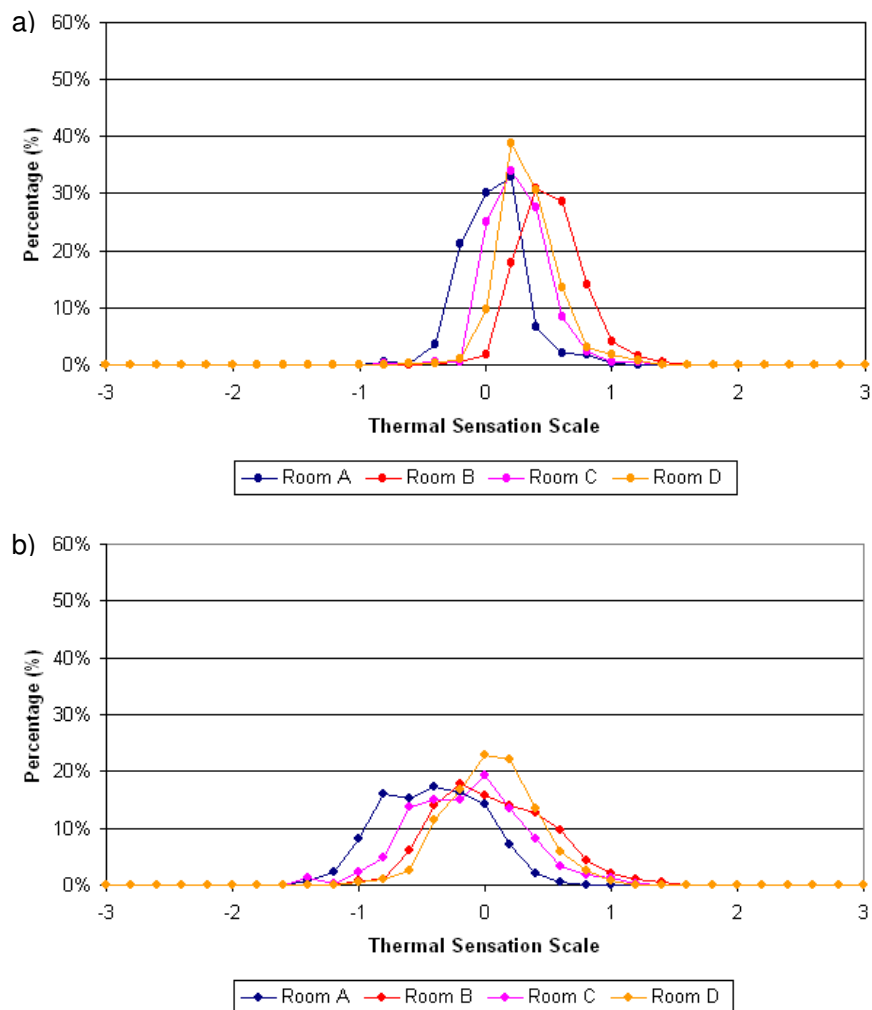


Figure 6.121 : TSS variations of System1 zones provided by the individual and master control modes at 23.3°C set temperature, (a) individual control mode, (b) master control mode

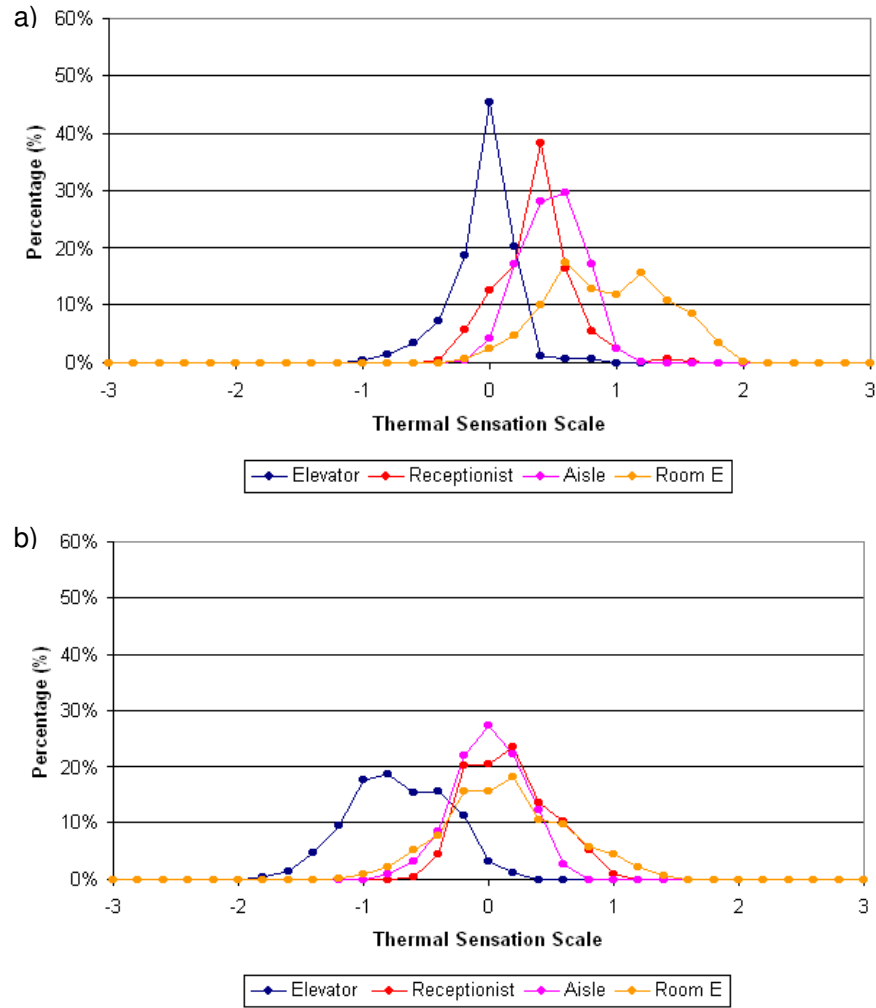


Figure 6.122 : TSS variations of System2 zones provided by the individual and master control modes at 23.3°C set temperature, (a) individual control mode, (b) master control mode

Figure 6.123 shows the TSS variation based on the combined zones for the individual and master control modes. As can be seen, for both combined zones, the master control mode provided much colder indoor environment than the individual control mode with low percentages.

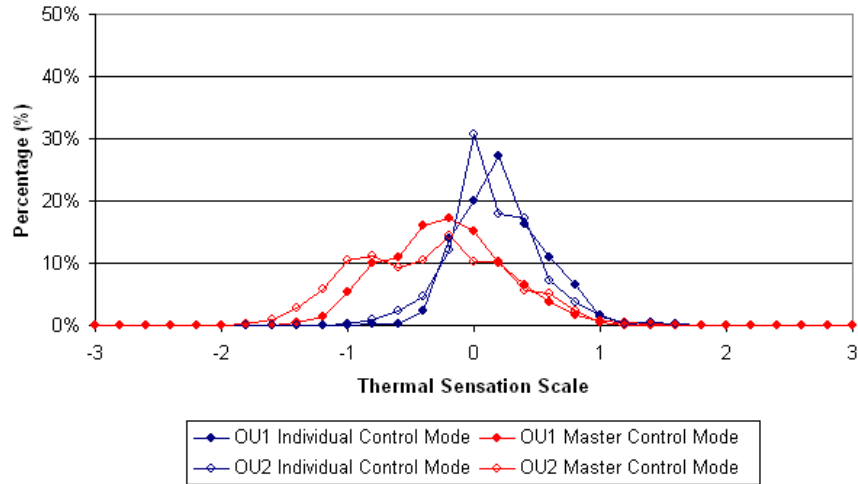


Figure 6.123 : TSS variations of System1 and System2 combined zones provided by the individual and master control modes

Figure 6.124 shows the comparison of the daily averaged energy consumption of the individual and master control modes at 23.3°C set temperature. Unlike to the cooling performance tests, it was found that the individual control mode consumed around 40%, 71% and 55% higher energy for OU1, OU2 and total (OU1+OU2) compared to the master control mode, respectively.

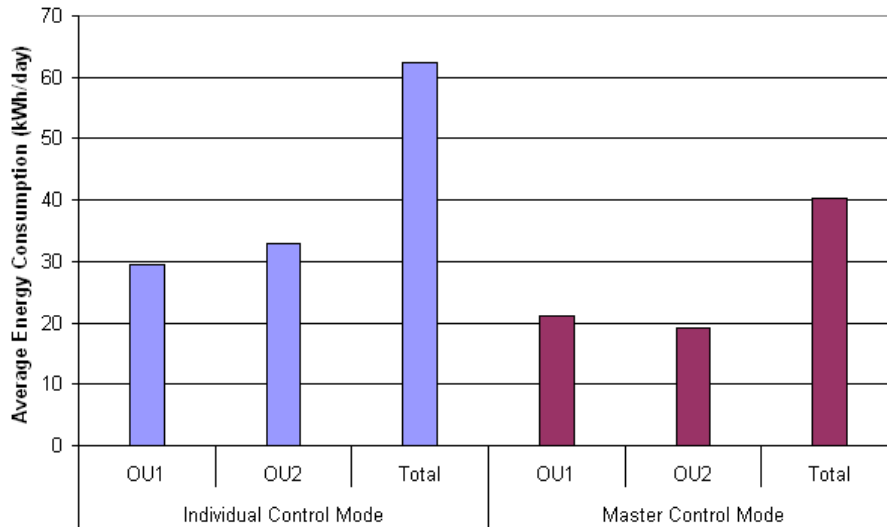


Figure 6.124 : Comparison of the daily averaged energy consumption of the individual and master control modes at 23.3°C set temperature

Variation of daily energy consumptions of the outdoor units with respect to the daily averaged outdoor temperatures for the individual and master control modes of the multi-split VRF systems are provided in App. [K].

Figure 6.125 shows the variation of the percentage “ON” time ratio of OU1 and OU2 for the individual and master control modes with respect to the outdoor temperature bin. As expected, the percentage “ON” time ratios increased with the decreasing outdoor temperature in order to keep the indoor temperatures at the set temperature.

As seen, at any outdoor temperature bin, the percentage “ON” time ratio of the individual control mode was higher than that of the master control mode. Since the power consumptions of the outdoor units for the individual and master control modes were almost same during the “ON” time, shown in Figure 6.113, the higher percentage “ON” time ratio of the individual control caused higher power consumption.

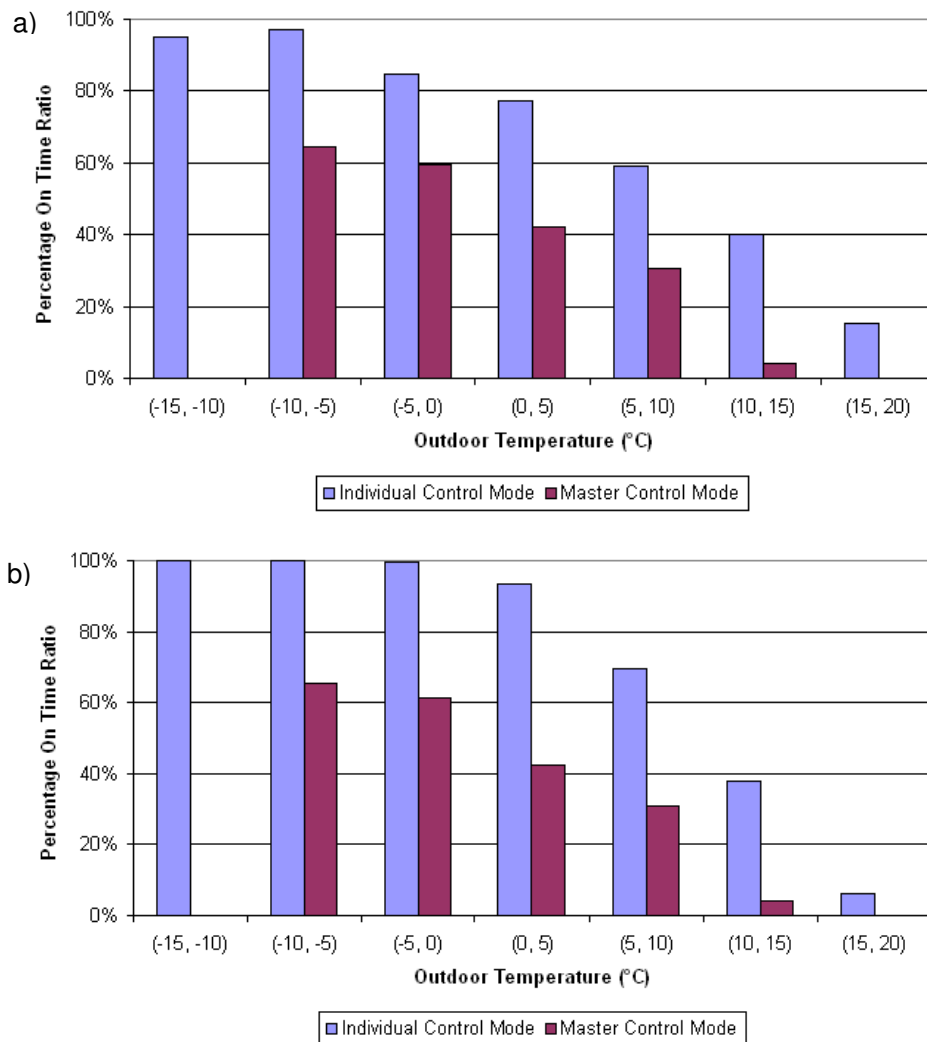


Figure 6.125 : Percentage ON time ratio of OU1 and OU2 for the individual and master control modes at 23.3°C set temperature, (a) OU1, (b) OU2

The provided daily heating energy, daily energy consumption of OU2 and corresponding HPF and PLR are plotted in Figure 6.126 with respect to the daily averaged outdoor temperature for both individual and master control modes. The trends are also shown with least square regression lines.

As expected, the required energy for the outdoor unit increased with the decreasing outdoor temperature and associated provided heating energy increased for both individual and master control modes. However, as explained previously, due to the elevator zone, the individual control mode consumed higher energy to keep that zone's temperature at the desired set temperature.

Similar to the cooling performance tests, the efficiency trend is similar to the findings of Watanabe et al. (2004).

As can be seen from Figure 6.126d, even though the error bars are relatively high, it was found that the performance factor of the individual control mode was higher than that of the master control mode. This is because the individual control mode operated almost continuously even for the high outdoor temperature due to the elevator zone.

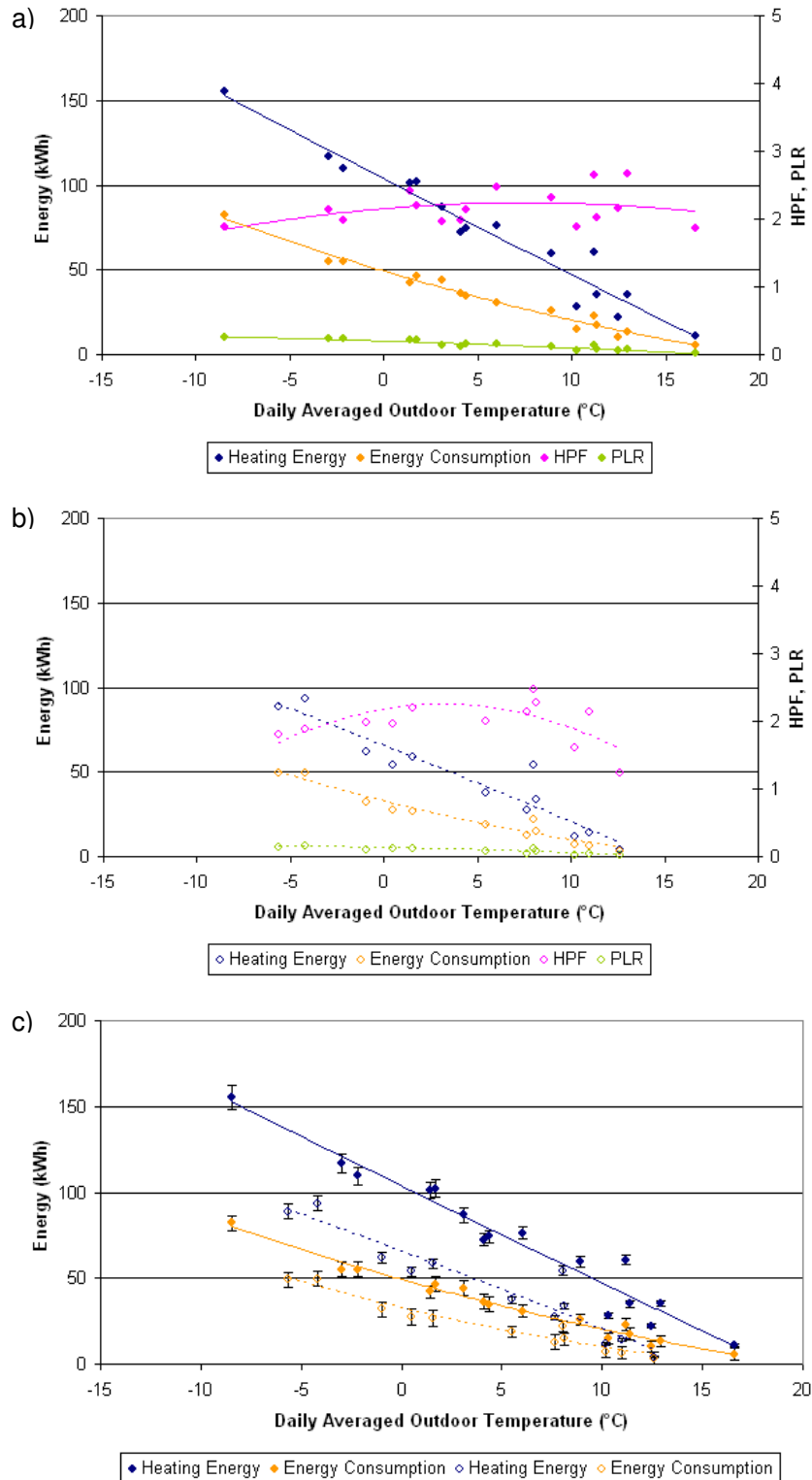


Figure 6.126 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 23.3°C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF (continued)

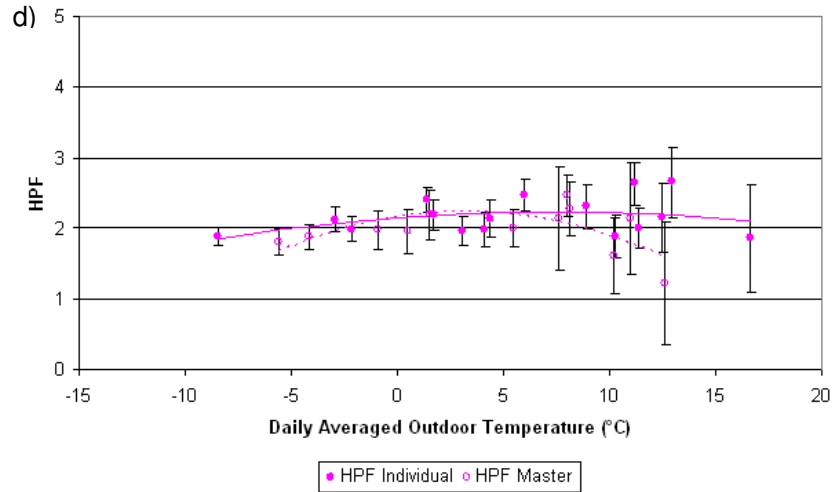


Figure 6.126 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 23.3°C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF

As shown in Figure 6.127, similar to the cooling mode operation, the fixed speed compressor operated only around 2% of all operational time.

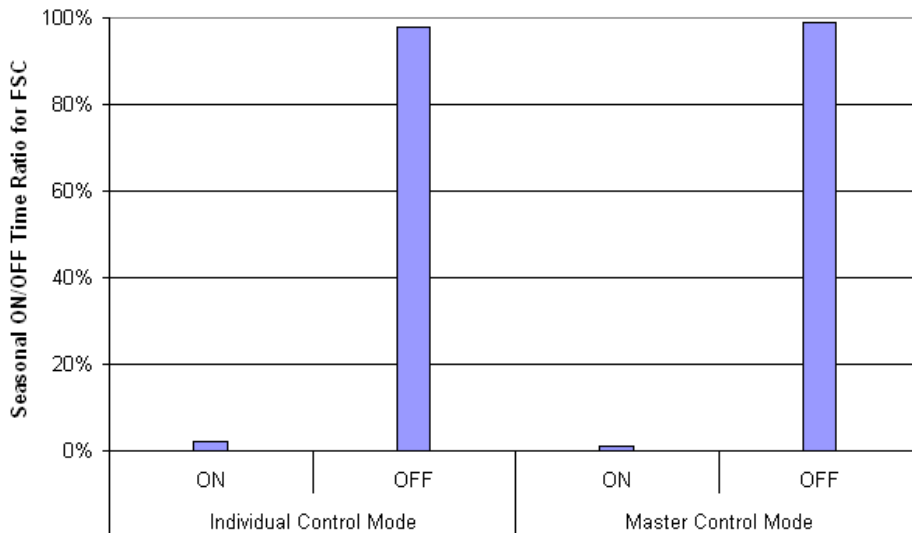


Figure 6.127 : Seasonal ON/OFF time ratio of FSC of OU2 for the individual and master control modes

The reason of why the performance factor in heating mode is much lower than the cooling mode is due to the operation difference between the cooling and heating modes.

Figure 6.128 shows the variation of the outdoor unit power consumption with respect to the inverter frequency for both cooling and heating modes with hourly averaged data. The indoor set temperatures were 25°C and 23.3°C for the cooling and heating modes, respectively.

As can be seen, the required power in the heating mode was higher than the cooling mode for the same inverter frequency. Namely, roughly, the heating mode required more power to provide same amount of refrigerant mass flow rate than the cooling mode, which ended up with lower performance factor.

In addition to the power consumption difference, the defrost requirement during the heating mode also affected the heating performance of the system. Since the multi-split VRF system utilized the reverse cycle for defrosting, the provided heating energy decreased during the defrosting, as shown in Figure 6.114 and Figure 6.115, which ended up with lower performance factor.

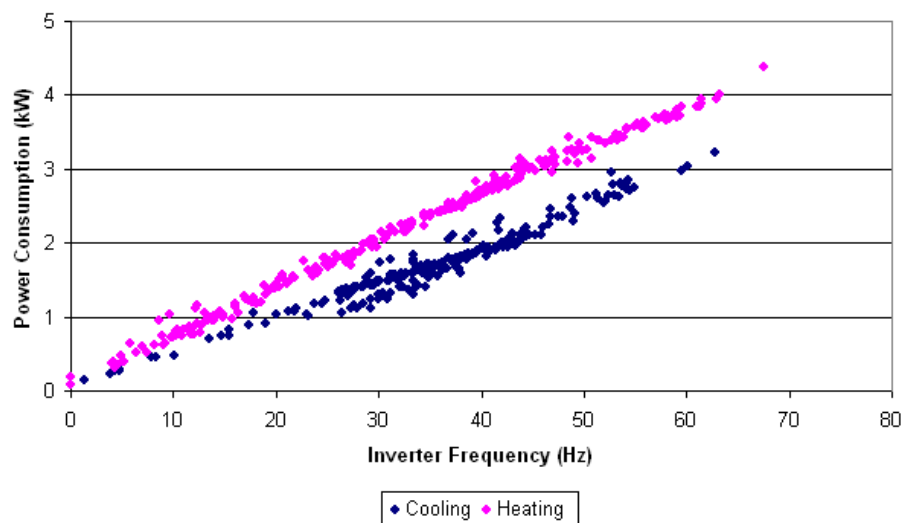
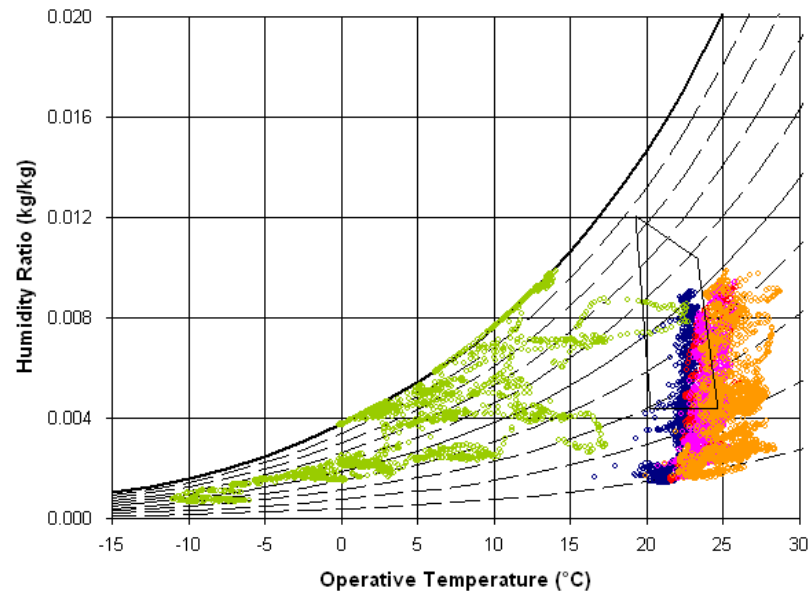


Figure 6.128 : Variation of the outdoor unit power consumption with respect to the inverter frequency in the cooling and heating modes

Figure 6.129 shows the indoor condition data of System2 zones provided by the individual control mode for the continuous and synchronized indoor unit fan operations.

As can be seen from Figure 6.129a and Figure 6.129b, considering the similar outdoor conditions, as expected, the synchronized indoor unit fan operation increased the number of indoor condition data in the comfort zone by eliminating the potential of the overheating of the zones.

a) Elevator, Receptionist, Aisle, Room E, Outdoor



b) Elevator, Receptionist, Aisle, Room E, Outdoor

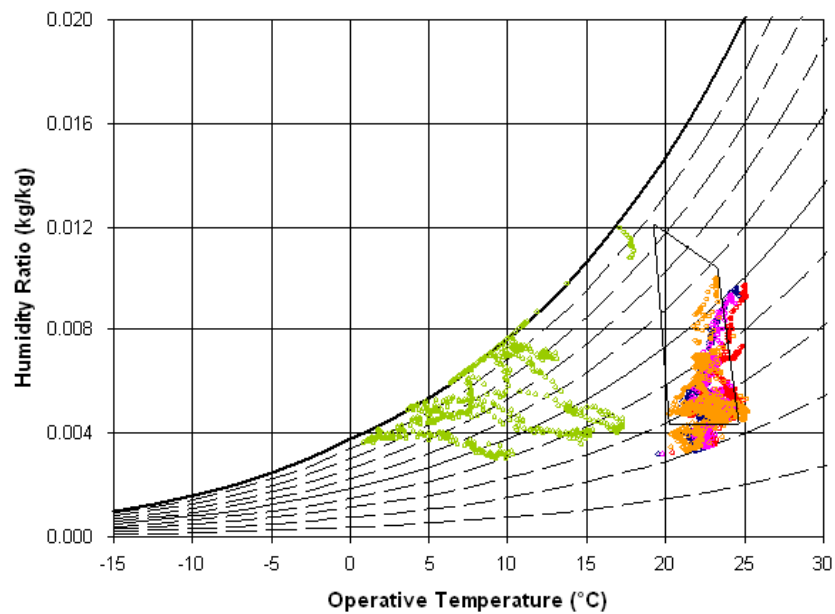


Figure 6.129 : Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operations at 23.3°C set temperature, (a) continuous fan operation, (b) synchronized fan operation

Table 6.15 shows the comparison of the indoor condition data of System2 zones provided by the continuous and synchronized indoor unit fan operation based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature. As can be seen,

the percentage of the indoor condition data for the synchronized indoor unit fan operation in the comfort zone was drastically higher than that of the indoor condition data for the continuous indoor unit fan operation.

Table 6.15 : Comparison of the indoor condition data of System2 zones provided by the continuous and synchronized indoor unit fan operations based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the continuous	Percentage of the indoor condition data for the synchronized
In	21.9%	70.6%
Cold	0.2%	0.0%
Hot	19.2%	0.0%
Cold Dry	0.4%	0.0%
Dry	44.3%	19.8%
Hot Dry	14.0%	9.6%
Humid	0.0%	0.0%

6.3.4 Seasonal comparison at 26.1°C set temperature

In addition to 23.3°C set temperature, higher set temperature tests at 26.1°C, were performed. Only System2 were operated during the tests.

The number of data points for the outdoor temperature and the outdoor conditions are provided in Figure 6.130 and Figure 6.131, respectively.

As can be seen, the outdoor conditions of the individual and master control mode tests were fairly close.

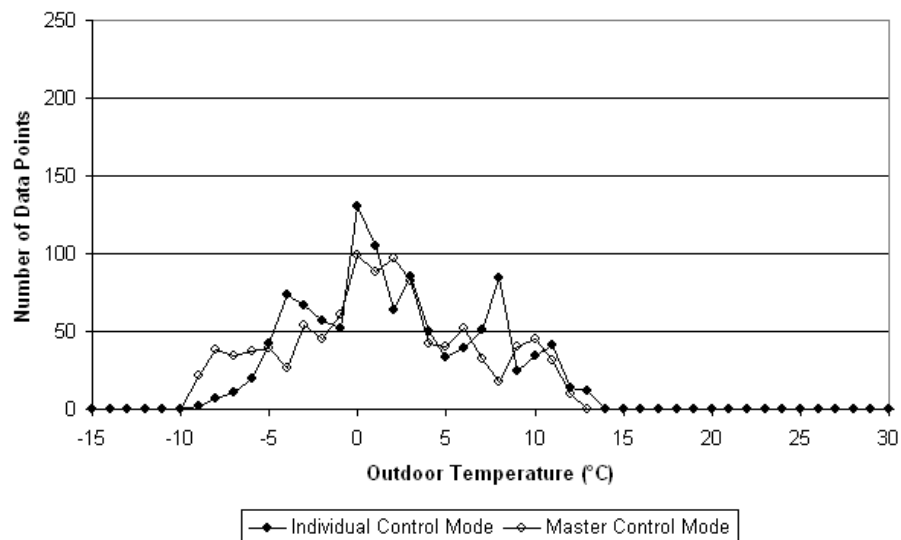


Figure 6.130 : Number of data points for 26.1°C set temperature tests

Individual Control Mode vs Master Control Mode

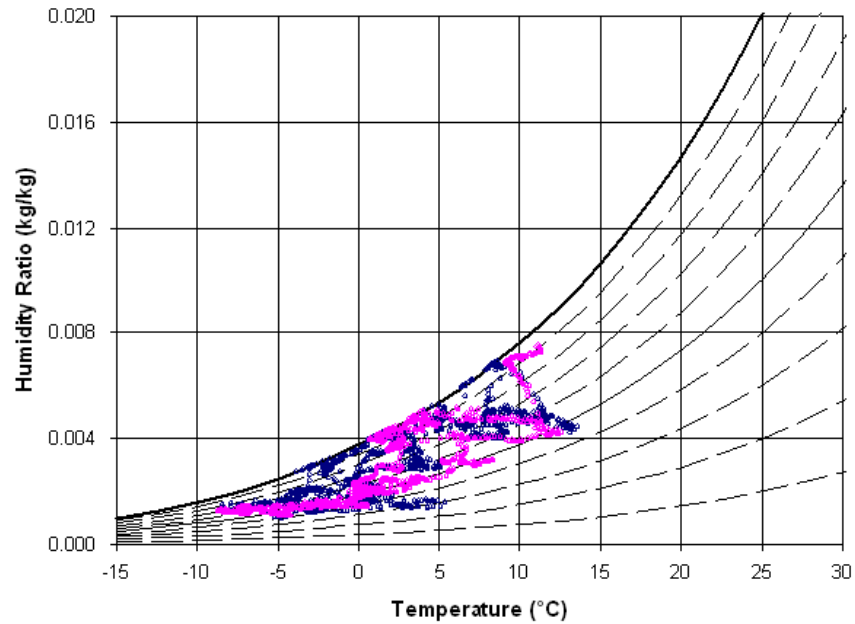
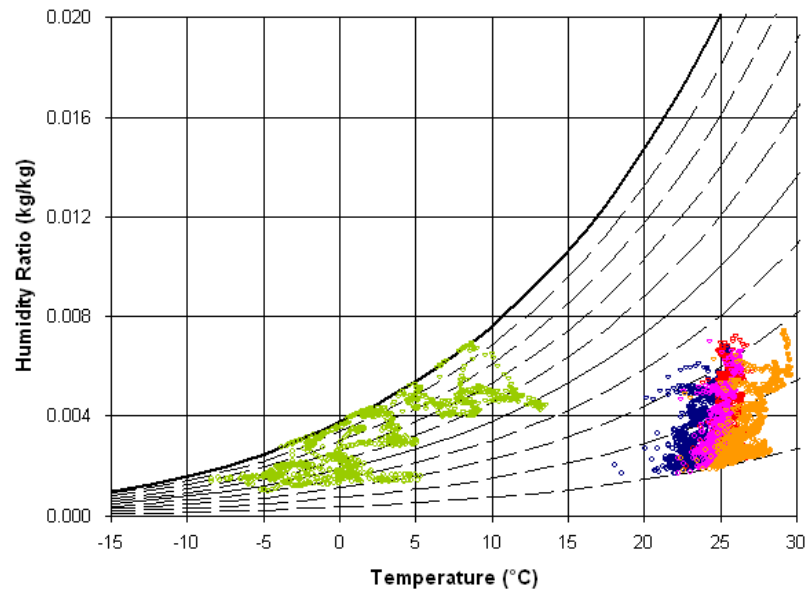


Figure 6.131 : Outdoor conditions for 26.1°C set temperature tests

Figure 6.132 shows the indoor condition data of System2 zones provided by the individual and master control modes at 26.1°C set temperature under continuous indoor fan operation. Since the set temperature of 26.1°C was out of the ASHRAE winter thermal comfort zone, it was not superimposed to the chart. As seen, the indoor conditions provided by the individual control mode were relatively closer to the set temperature than the master control mode. Although the individual and master control modes operated under the similar outdoor conditions, the temperature of the elevator zone controlled by the master control mode was much lower. The high temperature of Room E was due to the continuous fan operation.

a) Elevator, Receptionist, Aisle, Room E & Outdoor



b) Elevator, Receptionist, Aisle, Room E & Outdoor

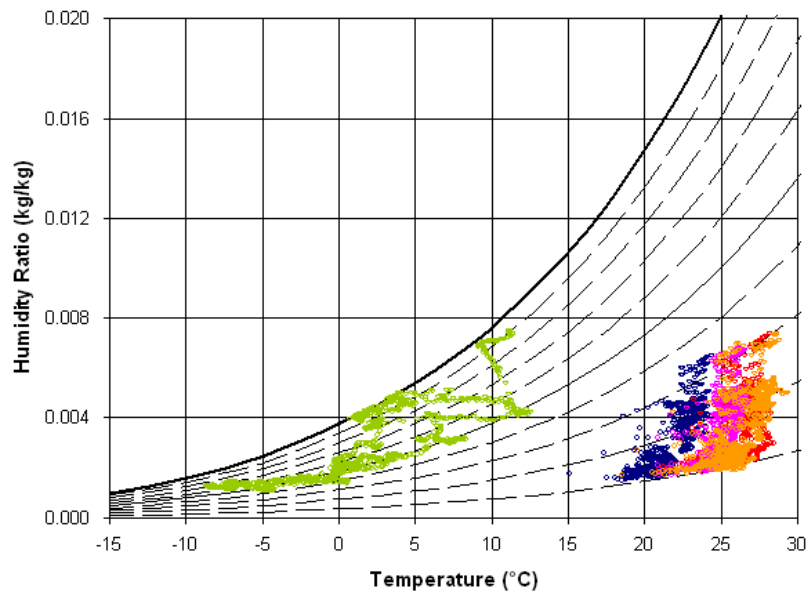


Figure 6.132 : Indoor conditions of System2 zones provided by the individual and master control modes at 26.1 °C set temperature, (a) individual control mode, (b) master control mode

Figure 6.133 shows the TSS variations of System2 zones provided by the individual and master control modes, respectively. As can be seen, percentages of TSS of the receptionist and aisle zones provided by the individual control mode were higher

than that of the master control mode, which indicates that the temperature fluctuations due to the individual control mode were lower the master control mode.

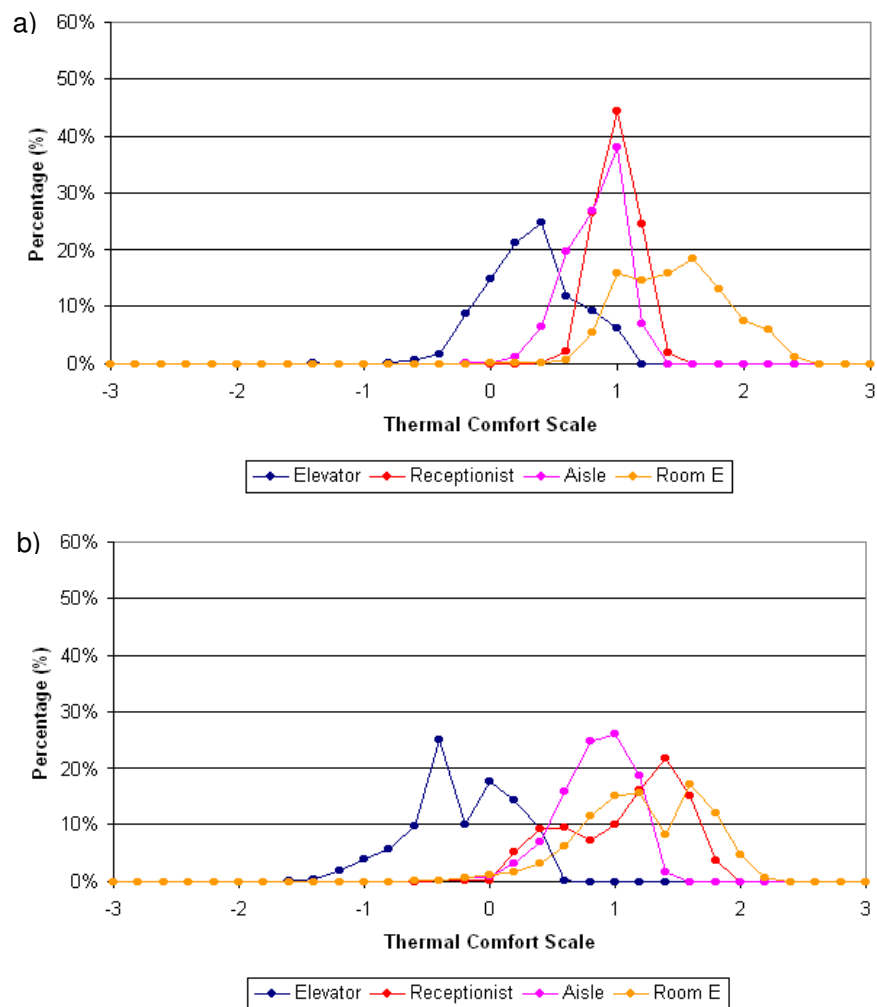


Figure 6.133 : TSS variations of System2 zones provided by the individual and master control modes at 26.1 °C set temperature, (a) individual control mode, (b) master control mode

Figure 6.134 shows the TSS variation based on the combined zones for the individual and master control modes. As can be seen, the TSS variation of the individual control mode was relatively narrower compared to the master control mode with high percentage, which indicates better temperature control with less temperature fluctuation.

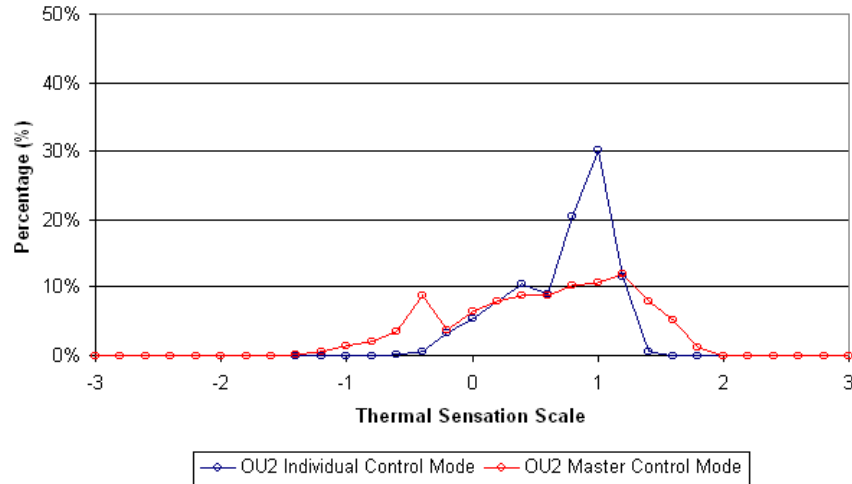


Figure 6.134 : TSS variations of System2 combined zones provided by the individual and master control modes

Figure 6.135 shows the comparison of the daily averaged energy consumption of the individual and master control modes at 26.1°C set temperature. Similar to 23.3°C set temperature, the individual control mode consumed 10.3% higher than the master control mode at 26.1°C set temperature. This is because the individual control mode operated almost 100% of all time, even for relatively high outdoor temperatures, as shown in Figure 6.136.

Figure 6.136 shows the variation of the percentage “ON” time ratio of OU2 for the individual and master control modes with respect to the outdoor temperature bin. As can be seen, at any outdoor temperature bin, the percentage “ON” time ratio of the individual control mode was higher than that of the master control mode, which caused more power consumption with relatively higher indoor temperatures.

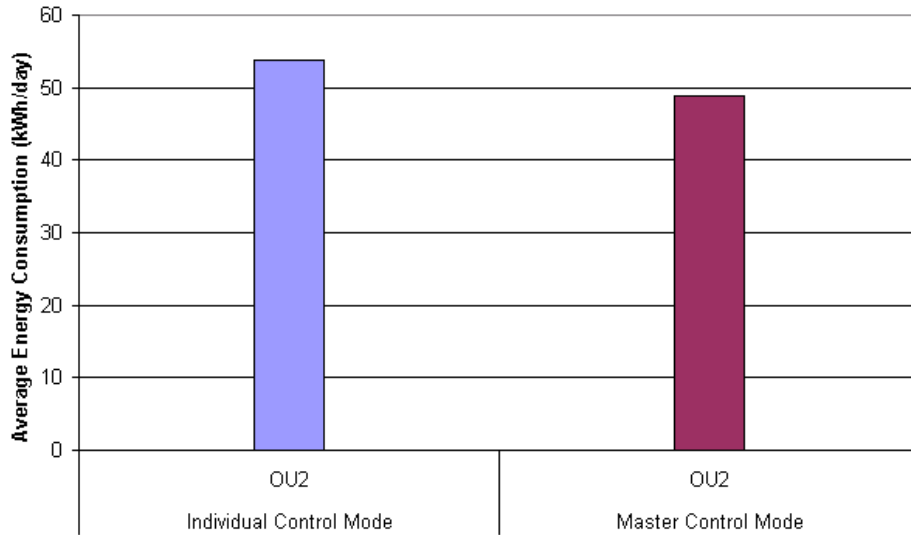


Figure 6.135 : Comparison of the daily averaged energy consumption of the individual and master control modes at 26.1 °C set temperature

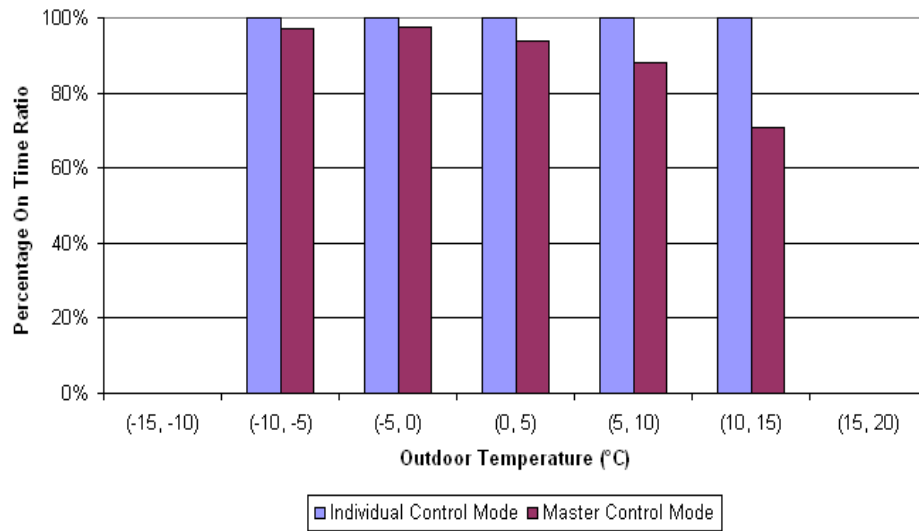


Figure 6.136 : Percentage ON time ratio of OU2 for the individual and master control modes at 26.1 °C set temperature

The provided daily heating energy, daily energy consumption of OU2 and corresponding HPF and PLR are plotted in Figure 6.137 with respect to the daily averaged outdoor temperature for both individual and master control modes. The trends are also shown with least square regression lines.

Similar to the low indoor set temperature cases, the required energy for the outdoor unit increased with the decreasing outdoor temperature and associated provided heating energy increased for both individual and master control modes. Due to the

fact of the elevator zone, the individual control mode consumed higher energy compared to the master control mode, as shown in Figure 6.137b.

As seen from Figure 6.137d, the efficiency values were relatively higher compared to low indoor set temperature tests shown in Figure 6.126 due to the continuous operation.

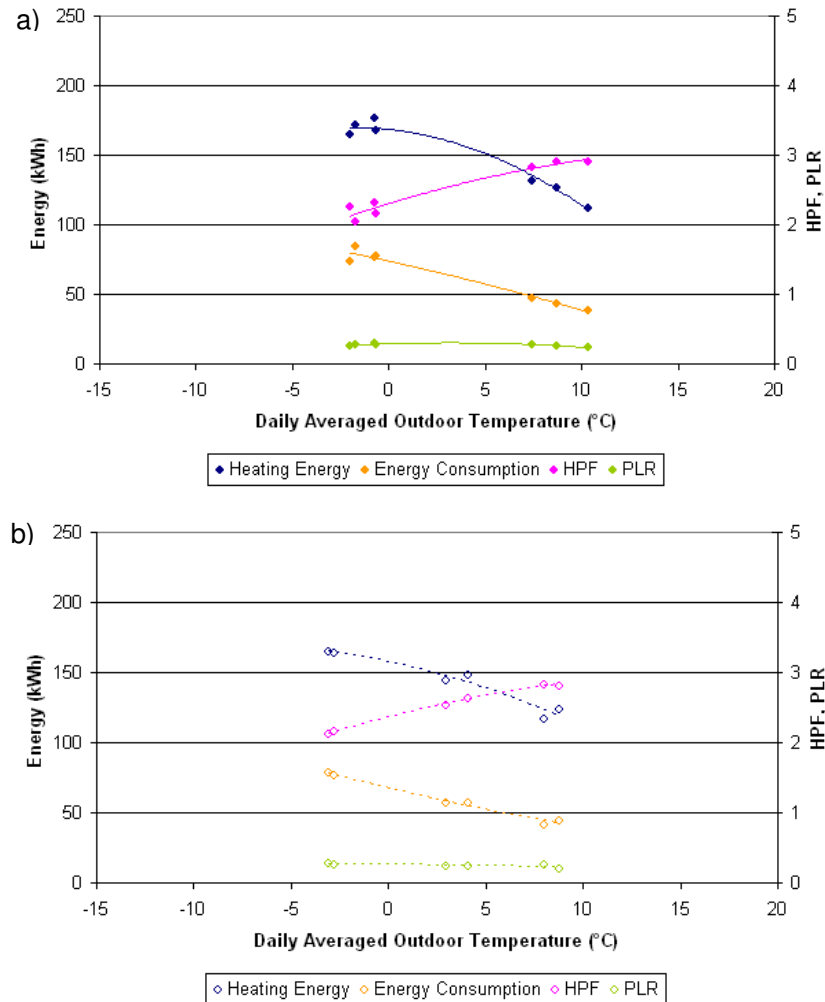


Figure 6.137 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF (continued)

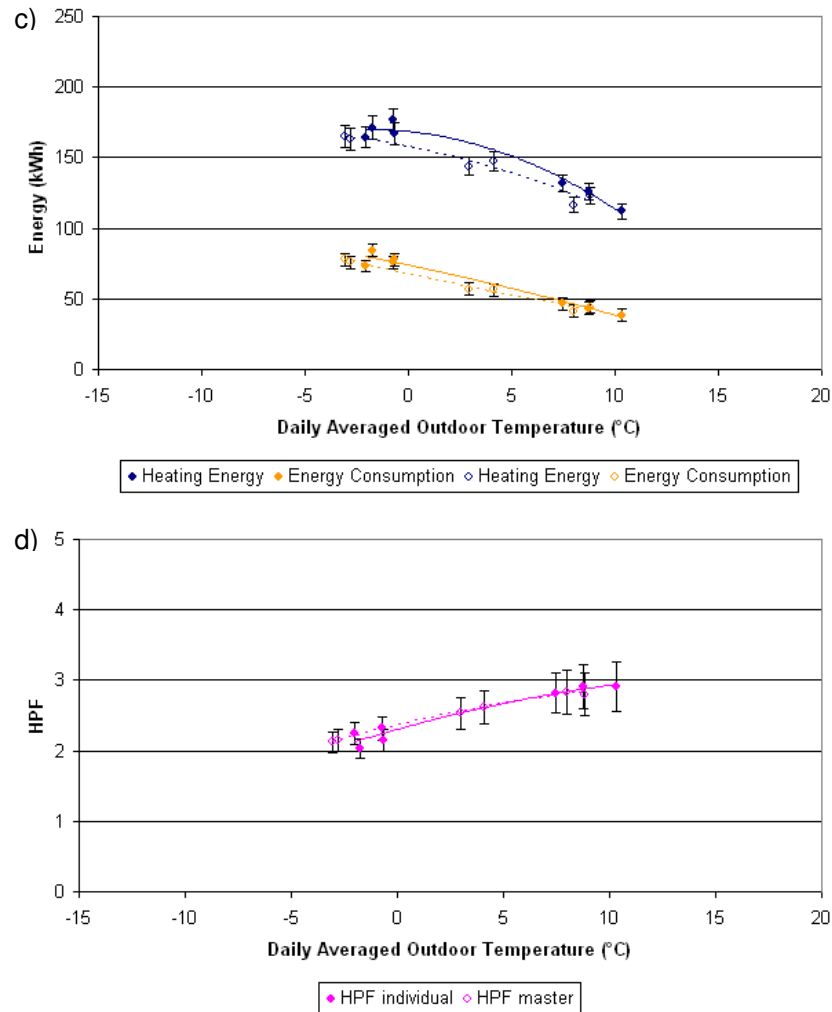


Figure 6.137 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF

Figure 6.138 shows the indoor condition data of System2 zones provided by the individual control mode under the continuous and synchronized indoor unit fan operation.

As can be seen, similar to low indoor set temperature tests, the synchronized indoor unit fan operation eliminated the overheating of Room E, and controlled the indoor temperature much better compared to the continuous indoor unit fan operation.

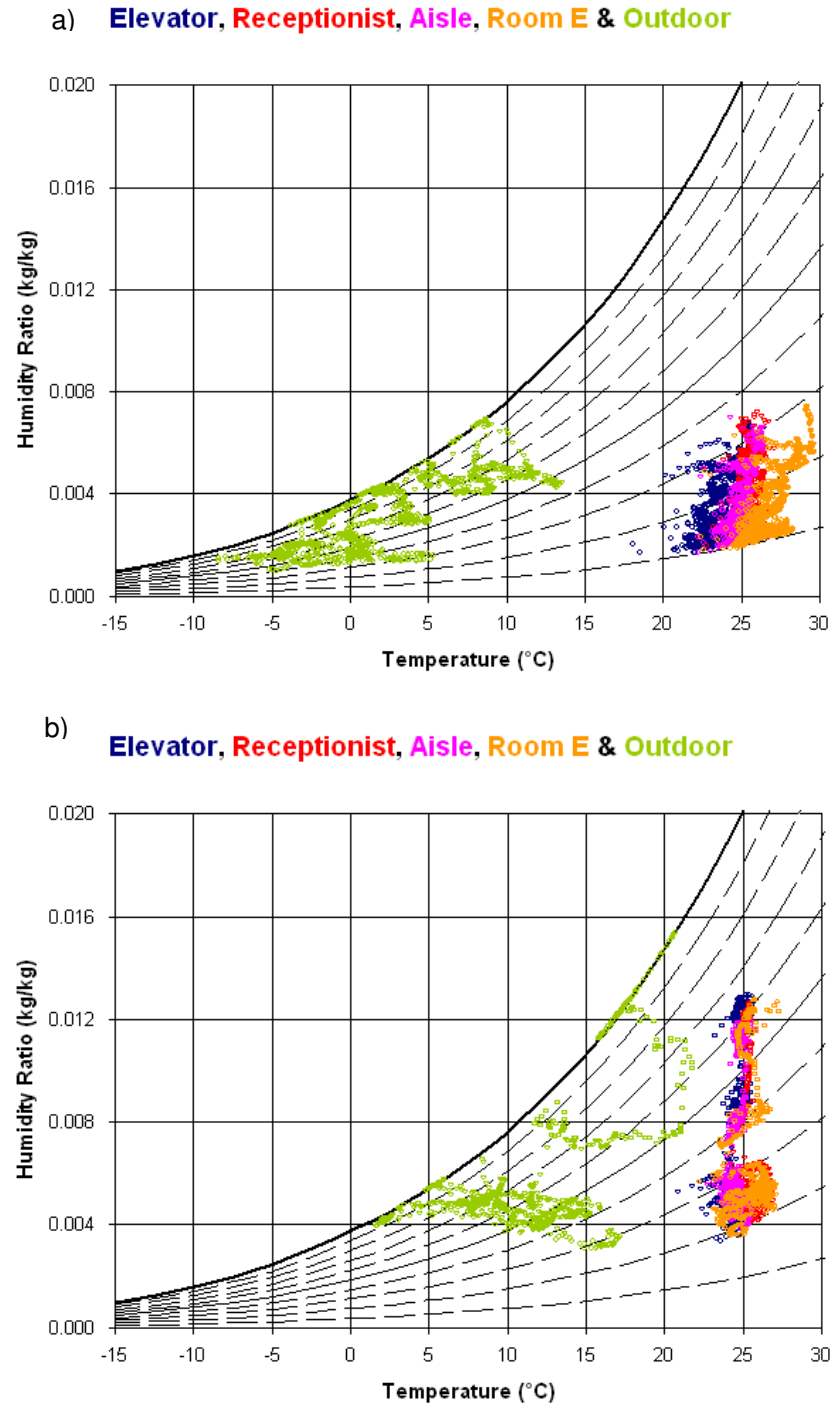


Figure 6.138 : Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operation at 26.1 °C set temperature, (a) continuous fan operation, (b) synchronized fan operation

As described previously, the multi-split VRF systems worked almost all of the operation time at the high indoor set temperature case. That's why; compared to the

low indoor set temperature case, the possibility of finding operations under steady state conditions at high indoor temperatures was more likely.

As explained previously, since the refrigerant mass flow rate mainly defines the indoor unit heating capacity, the operation under constant inverter frequency can assumed to be steady state. Because the inverter frequency is the main parameter defines the refrigerant mass flow rate.

Similar to the cooling mode evaluations, at least 20 minute-operations with constant inverter frequencies were tried to be found. In addition to the inverter frequencies, the indoor and outdoor air temperatures, as well as refrigerant temperatures and pressures were checked whether or not they violated the steady state conditions.

A typical 46-minute steady state operation, extracted from the individual control mode experiment performed on December 10, 2006, Sunday, was given in Figure 6.139, Figure 6.140, Figure 6.141, Figure 6.142, Figure 6.143 and Figure 6.144 with one-minute intervals.

As can be seen from Figure 6.139, the inverter frequency, corresponding outdoor unit power consumption, and the discharge and suction pressures were constant throughout the period.

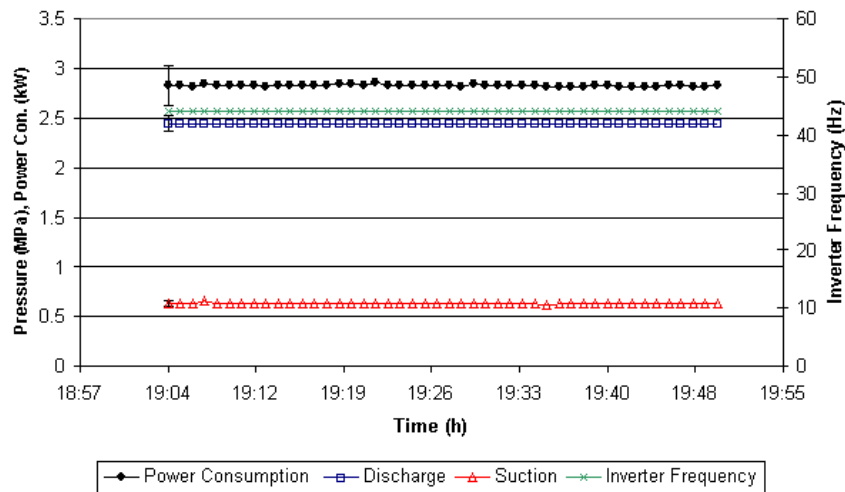


Figure 6.139 : Variation of the inverter frequency, refrigerant pressures and outdoor unit power consumption

As can be seen from Figure 6.140, since the discharge and suction temperatures, as well as the inverter frequency were constant, the indoor unit refrigerant inlet and outlet temperatures were constant.

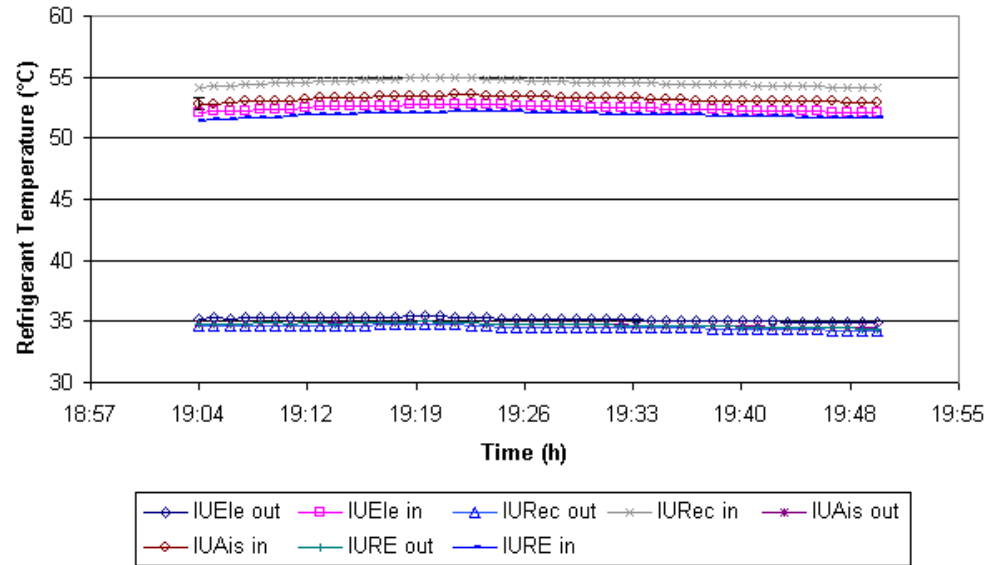


Figure 6.140 : Variation of the indoor unit refrigerant inlet and outlet

Figure 6.141 shows the indoor air blowing temperatures. As can be seen, the blowing temperatures were almost constant throughout the period. Since there was not any drastic temperature drop, there was not any defrosting operation during the period which could violate the steady state operation.

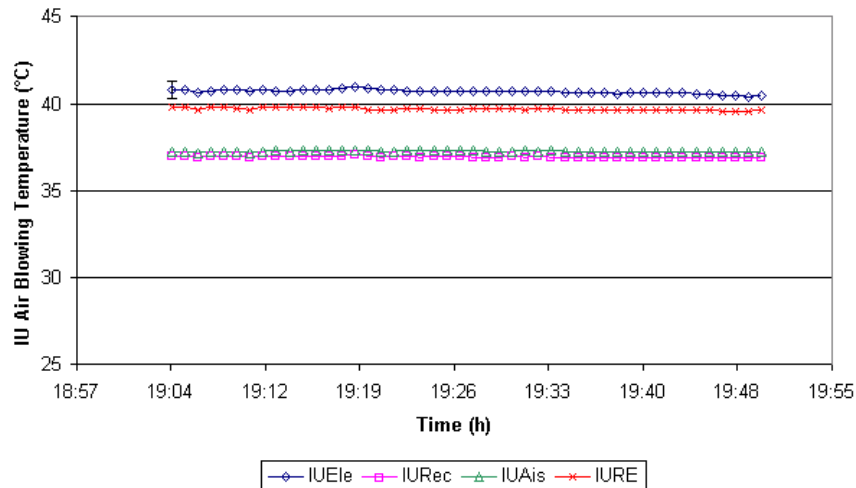


Figure 6.141 : Variation of the indoor unit air blowing temperature

In addition to the blowing air temperatures, Figure 6.142 shows the indoor air inlet temperatures.

As can be seen, they can be assumed to be constant during the period.

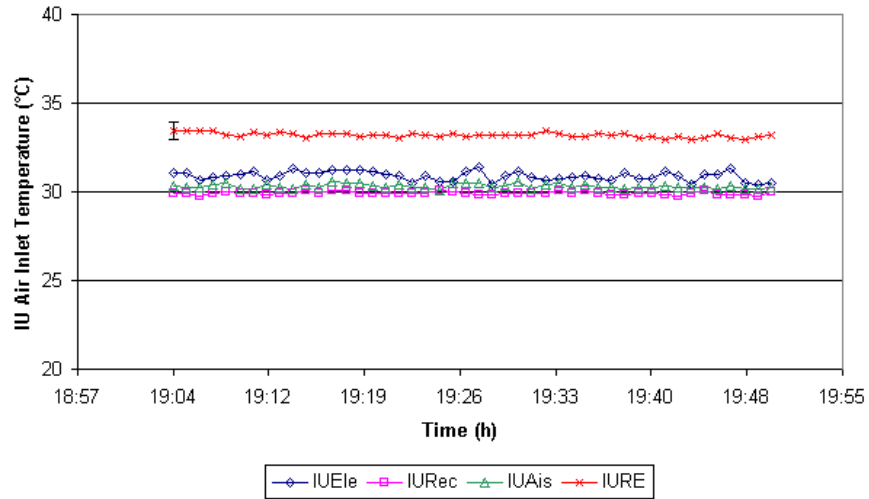


Figure 6.142 : Variation of the indoor unit air inlet temperature

Figure 6.143 depicts the indoor and outdoor temperatures for the corresponding period, respectively.

As can be seen, they can be assumed to be constant throughout the period.

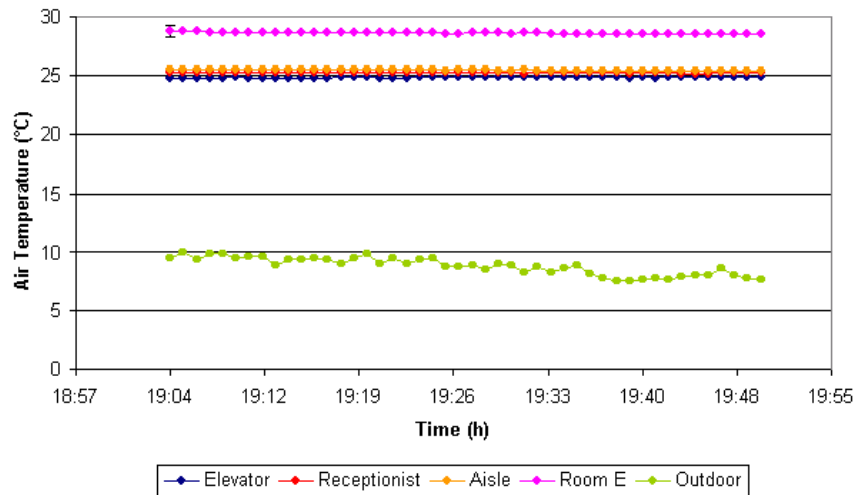


Figure 6.143 : Variation of the indoor and outdoor temperature

The total heating capacity and the COP of the system are given in Figure 6.144.

Except the two slight increase and decrease, they can be assumed to be constant. The discrepancies were due to the slight increase and decrease of the refrigerant pressures.

The COP of the system is defined as

$$COP = \frac{\dot{Q}_{heat}}{\dot{W}}$$

6.1

where

$\dot{Q}_{heat}$: Total heating capacity (kW)

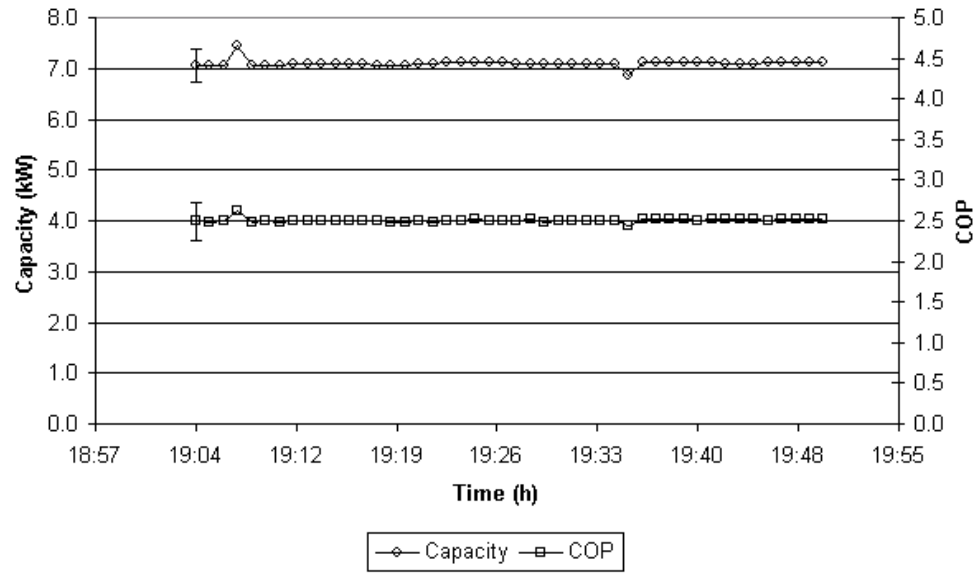


Figure 6.144 : Variation of the total heating capacity and the COP of the system

Figure 6.145 shows the variations of the power consumption, total heating capacity and the COP of the system with respect to the outdoor temperature under steady state operations, which were extracted from the individual control mode experiments performed with 26.1 °C indoor set temperature.

As seen from Figure 6.145, the power consumption increases with the decreasing outdoor temperature, and the corresponding total heating capacity increases with the increasing power consumption, due to the increasing inverter frequency which increases the refrigerant mass flow rate.

This result is similar to what is found for the daily and seasonal basis evaluations, discussed in the previous chapters.

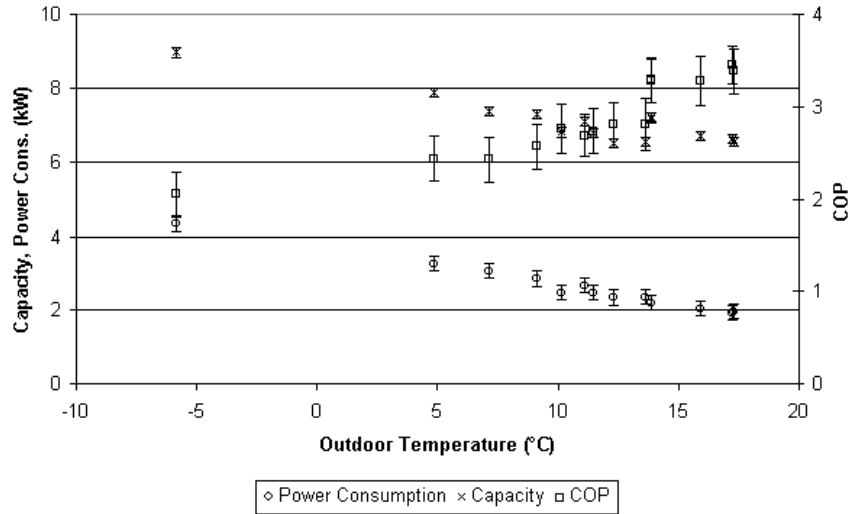


Figure 6.145 : Variation of the power consumption, total heating capacity and the COP of the system with respect to the outdoor temperature

The comparison of the HPF and COP of the system is provided in Figure 6.146.

It can be observed from Figure 6.146 that the COP and the HPF of the system are in the same order of magnitude and they can be assumed to be same.

This shows that the steady state tests can be used for the efficiency evaluations of the multi-split VRF system, however, the field tests should be performed for the evaluations of the indoor thermal comfort, and the seasonal energy consumption.

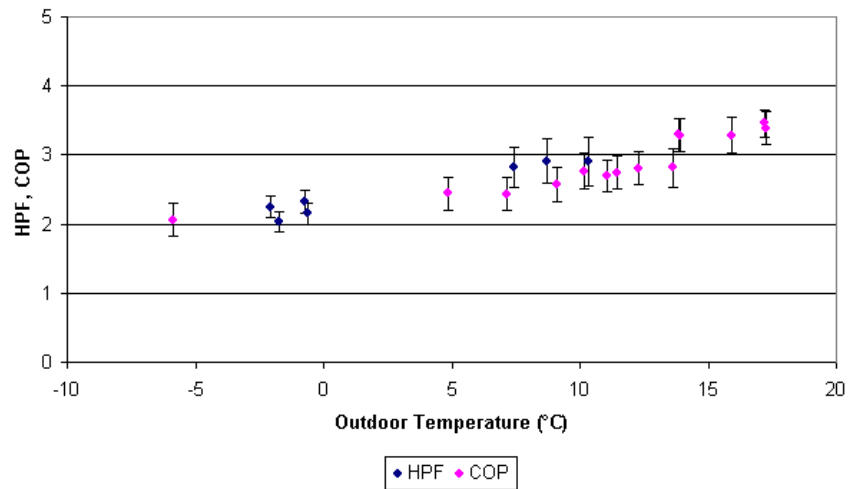


Figure 6.146 : Comparison of the HPF and COP of the system

6.4 Effect of the Ventilation in the Heating Performance

In order to find out the effect of the ventilation on the indoor temperature control, thermal comfort, power consumption of the outdoor units and the efficiency of the multi-split type VRF system; parametric tests were performed with and without ventilation for the heating mode.

During the ventilation assisted tests, the HRV units were operated in conjunction with the multi-split VRF systems throughout the test period. On the other hand, during the non-ventilated tests, the HRV units were not operated and the multi-split VRF systems were operated stand-alone. For both comparisons, the indoor set temperatures were kept at 23.3°C.

6.4.1 Seasonal comparison

The number of data points of the outdoor temperature for the ventilation assisted and non-ventilated systems is provided in Figure 6.147 and the comparison of the outdoor conditions is shown in Figure 6.148. As can be seen, the two operations were performed under similar outdoor conditions.

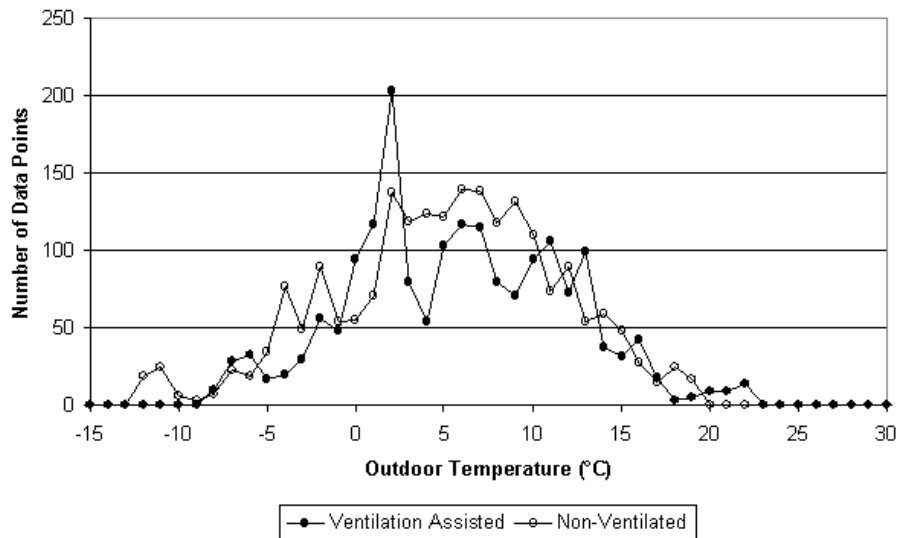


Figure 6.147 : Number of data points for ventilation assisted and non-ventilated tests in the heating mode

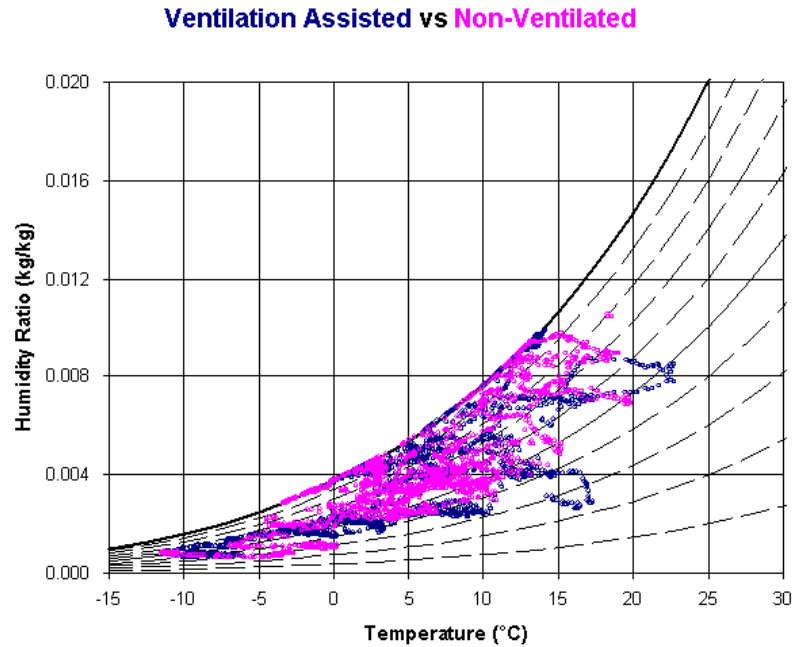


Figure 6.148 : Outdoor conditions for ventilation assisted and non-ventilated tests in the heating mode

Figure 6.149 shows the indoor conditions of System1 zones provided by the ventilation assisted and non-ventilated systems, and Table 6.16 shows the comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE winter thermal comfort zone. As can be seen from Table 6.16, the percentages of the indoor condition data in the ASHRAE winter thermal comfort zone were same for both operations. On the other hand, it was found that the non-ventilated system provided more humid environment than the ventilation assisted system. This is because; dry outdoor air was introduced to indoors during the ventilation assisted tests.

Table 6.16 : Comparison of the indoor condition data of System1 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the ventilation assisted	Percentage of the indoor condition data for the non-ventilated
In	31.9%	31.9%
Cold	0.1%	0.0%
Hot	11.1%	17.7%
Cold Dry	0.1%	0.0%
Dry	50.1%	39.6%
Hot Dry	6.7%	10.8%
Humid	0.0%	0.0%

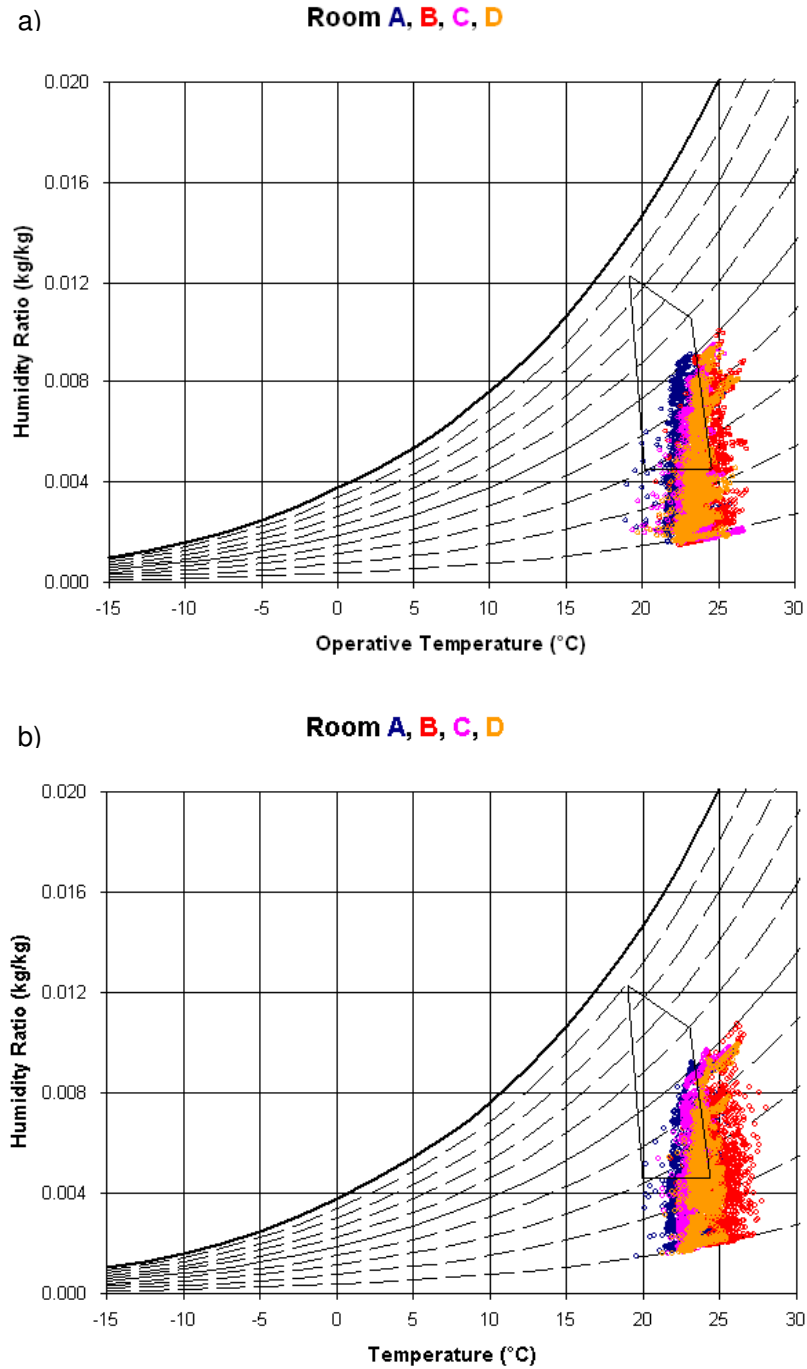


Figure 6.149 : Indoor conditions of System1 zones provided by ventilation assisted and non-ventilated systems at 23.3°C set temperature, (a) ventilation assisted, (b) non-ventilated

Figure 6.150 shows the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems, and Table 6.17 shows the comparison of the indoor condition data of System2 zones provided by the

ventilation assisted and non-ventilated systems based on the ASHRAE winter thermal comfort zone.

As can be seen from Table 6.17, similar to System1 zones, the non-ventilated system provided more humid indoor environment than the ventilated assisted system. However, since the indoor temperatures provided by the non-ventilated system were relatively warmer than the ventilation assisted system, the percentage of the indoor condition data provided by the non-ventilated system in the ASHRAE winter thermal comfort zone was lower than the percentage of the indoor condition data provided by the ventilated assisted system.

Table 6.17 : Comparison of the indoor condition data of System2 zones provided by the ventilation assisted and non-ventilated systems based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the ventilation assisted	Percentage of the indoor condition data for the non-ventilated
In	31.4%	25.4%
Cold	0.1%	0.0%
Hot	6.7%	17.8%
Cold Dry	0.1%	0.0%
Dry	50.1%	44.2%
Hot Dry	11.5%	12.6%
Humid	0.0%	0.0%

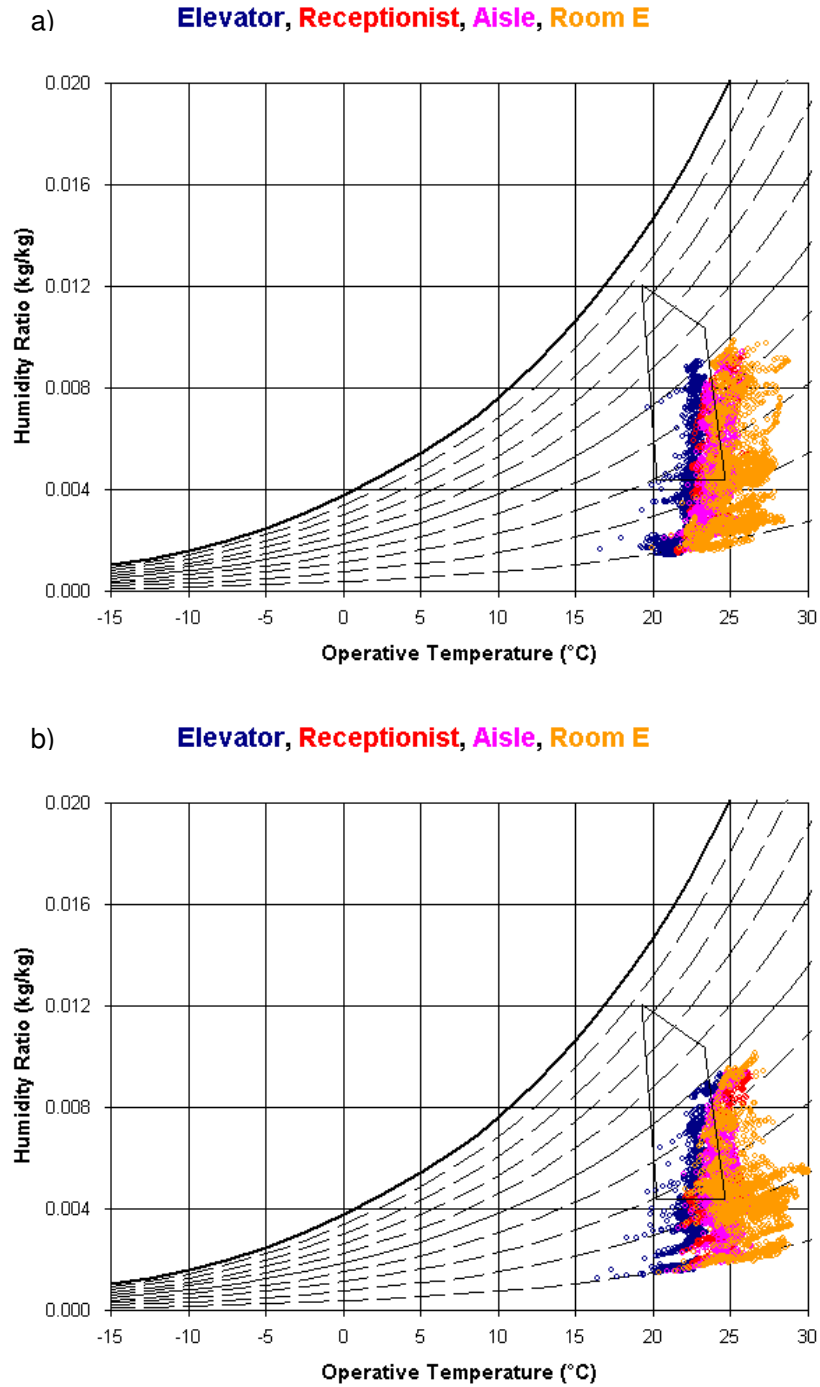


Figure 6.150 : Indoor conditions of System2 zones provided by ventilation assisted and non-ventilated systems at 23.3°C set temperature, (a) ventilation assisted, (b) non-ventilated

Figure 6.151 and Figure 6.152 show the TSS variations of System1 and System2 zones provided by the ventilation assisted and non-ventilated systems, respectively. As can be seen, the TSS percentages and the variations for the ventilation assisted and non-ventilated systems are very similar.

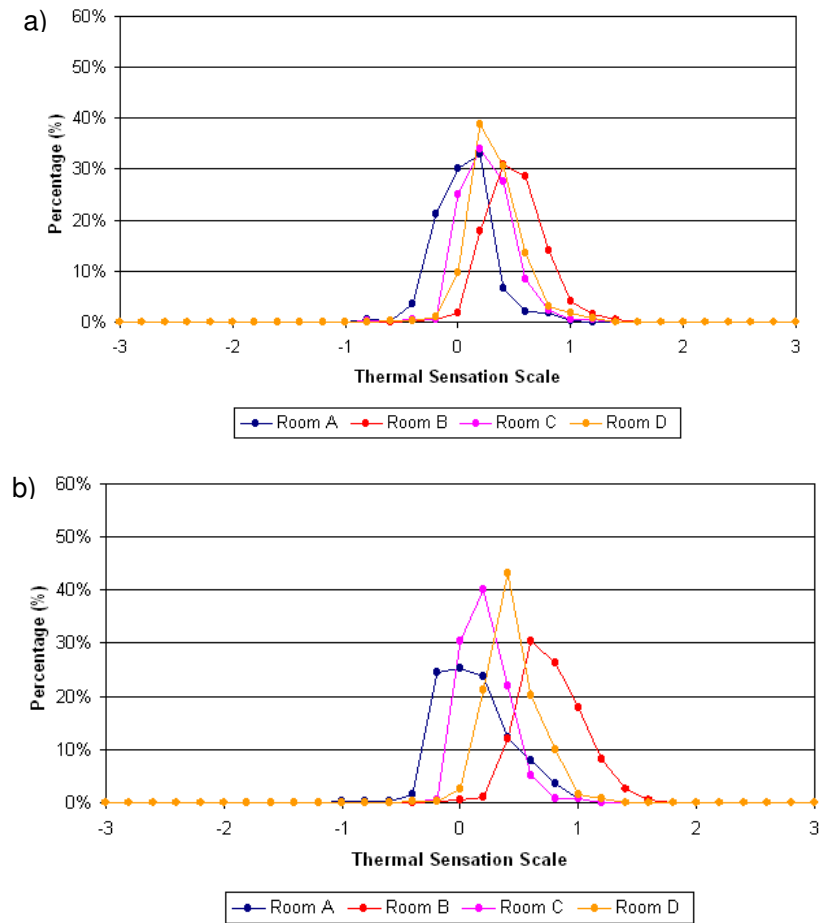


Figure 6.151 : TSS variations of System1 zones provided by the ventilation assisted and non-ventilated systems at 23.3°C set temperature, (a) ventilation assisted, (b) non-ventilated system

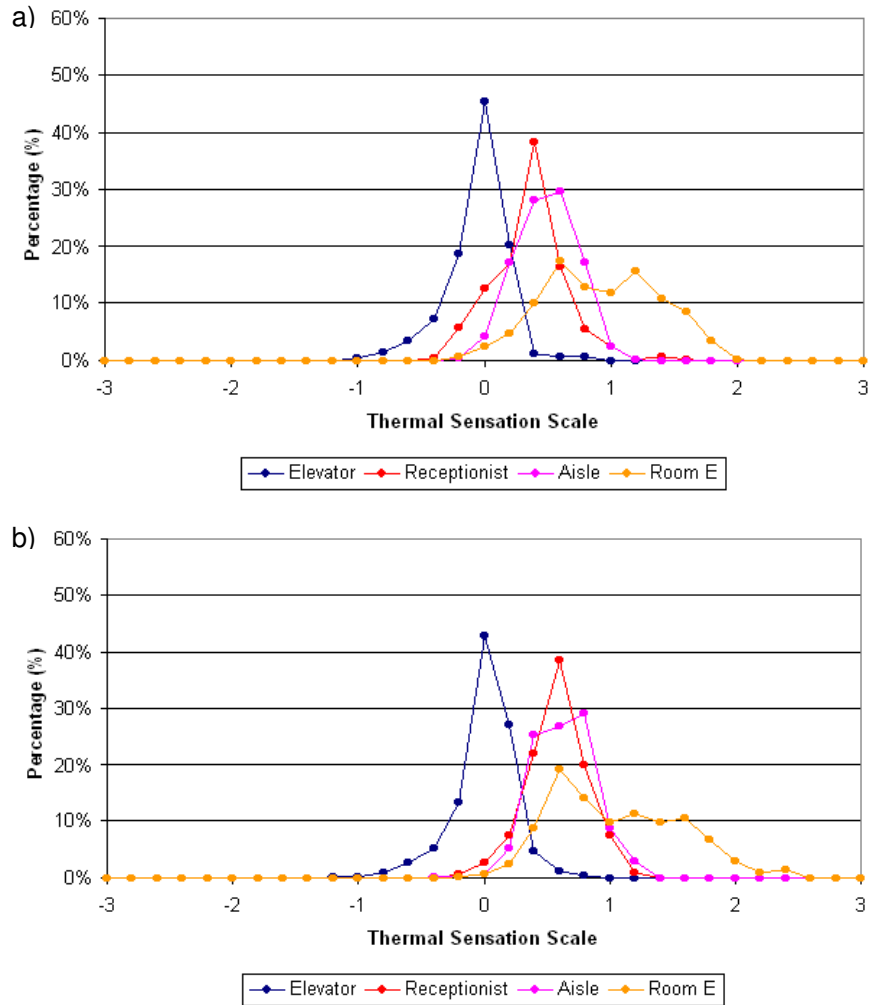


Figure 6.152 : TSS variations of System2 zones provided by the ventilation assisted and non-ventilated systems at 23.3°C set temperature, (a) ventilation assisted, (b) non-ventilated system

Since the TSS variations are quite similar, the TSS evaluation based on the combined zone is not taken into account.

Figure 6.153 shows the comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated systems. As expected, the ventilation assisted system consumed 62%, 9% and 29% higher energy for OU1, OU2 and total (OU1+OU2) compared to the non-ventilated system, respectively. This is because the ventilation increased the heating load for the ventilation assisted system; on the other hand, the internal load helped the non-ventilated system.

Figure 6.154 shows the variation of the percentage “ON” time ratio of OU1 and OU2 for the ventilation assisted and non-ventilated systems with respect to the outdoor temperature bin. As can be seen, at any outdoor temperature bin, the percentage

“ON” time ratio of the ventilation assisted system was higher than that of the non-ventilated system, which resulted in more energy consumption.

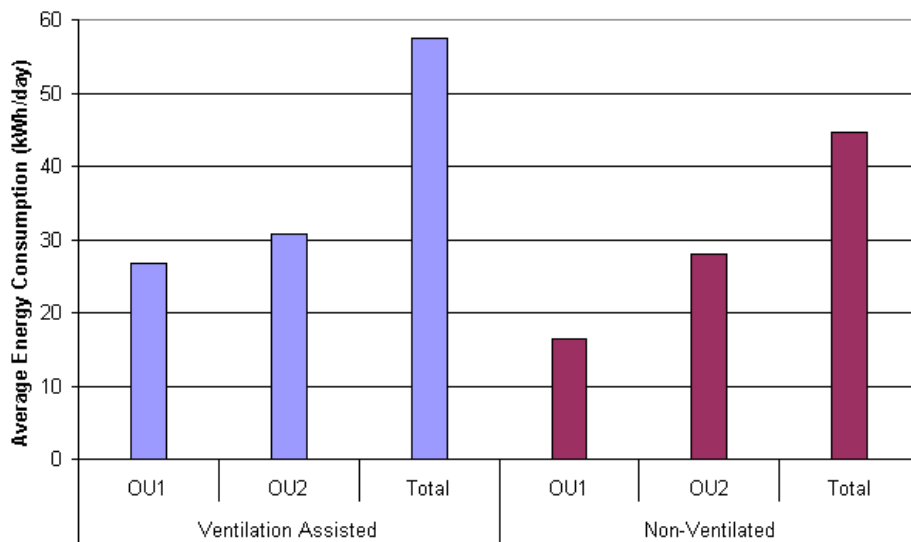


Figure 6.153 : Comparison of the daily averaged energy consumption of the ventilation assisted and non-ventilated systems at 23.3°C set temperature

Variation of daily energy consumptions of the outdoor units with respect to the daily averaged outdoor temperatures for the ventilation assisted and non-ventilated multi-split VRF systems are provided in App. [M].

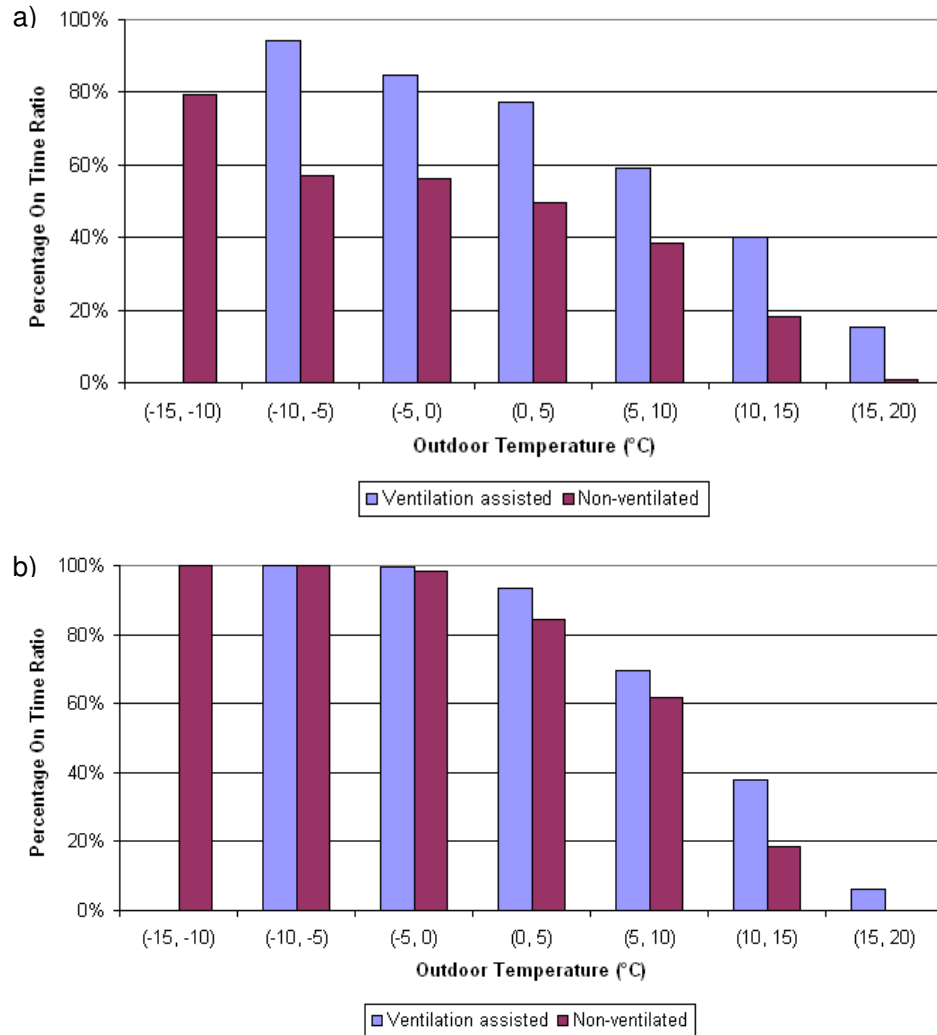


Figure 6.154 : Percentage ON time ratio of OU1 and OU2 for the ventilation assisted and non-ventilated systems at 23.3°C set temperature, (a) OU1, (b) OU2

The provided daily heating energy, daily energy consumption of OU2 and corresponding HPF and PLR are plotted in Figure 6.155 with respect to the daily averaged outdoor temperature for both ventilation assisted and non-ventilated systems. The trends are also shown with least square regression lines.

As can be seen from Figure 6.155c, the energy consumptions of the ventilation assisted and non-ventilated systems and associated provided heating energies were close to each other.

It is also observed from Figure 6.155d that the ventilation did not have any effect on the efficiency of the multi-split VRF system.

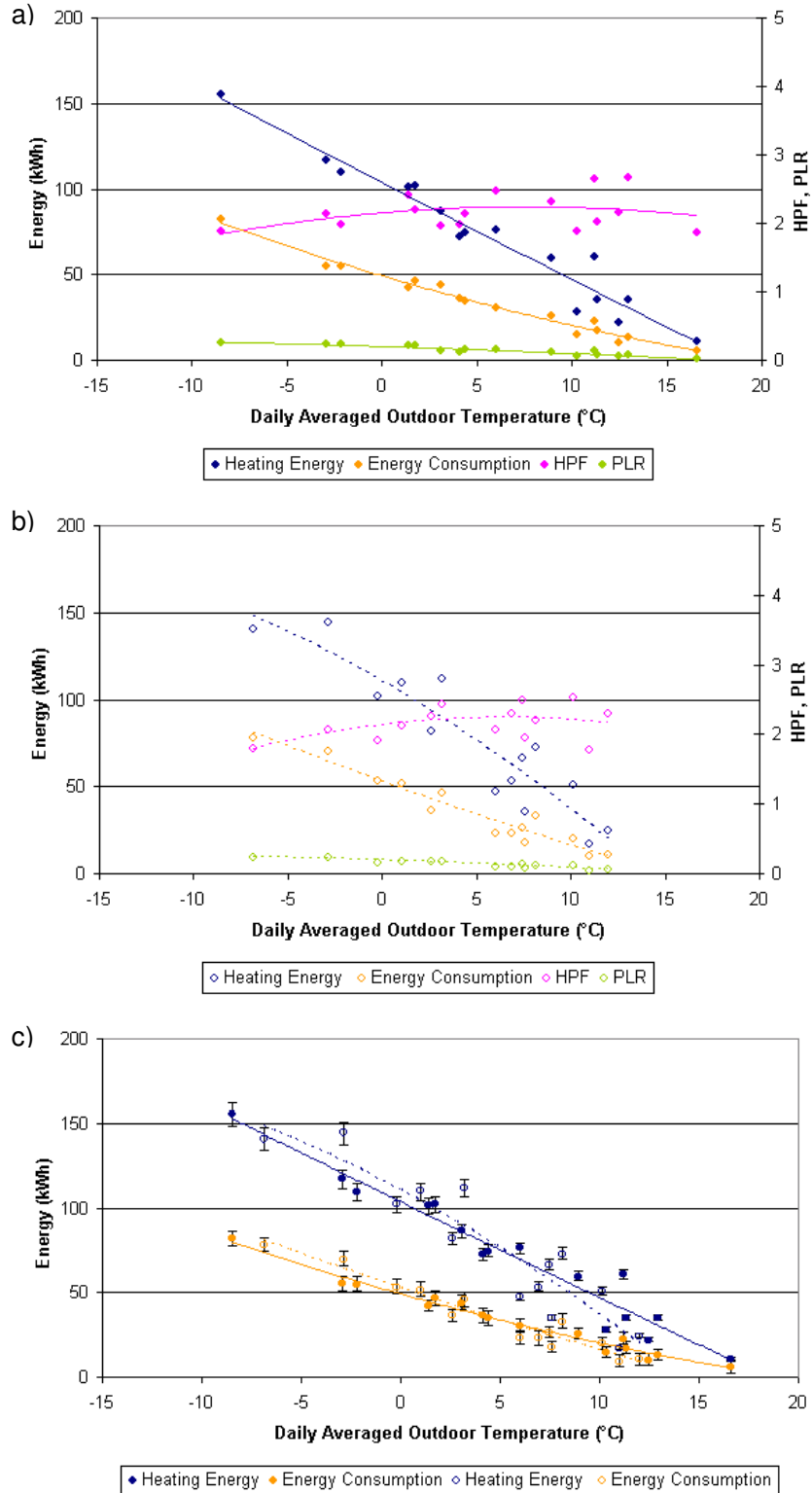


Figure 6.155 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF (continued)

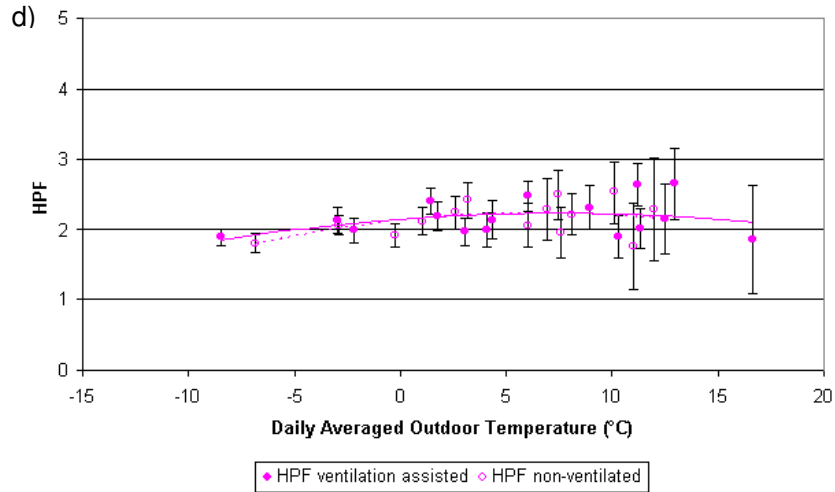
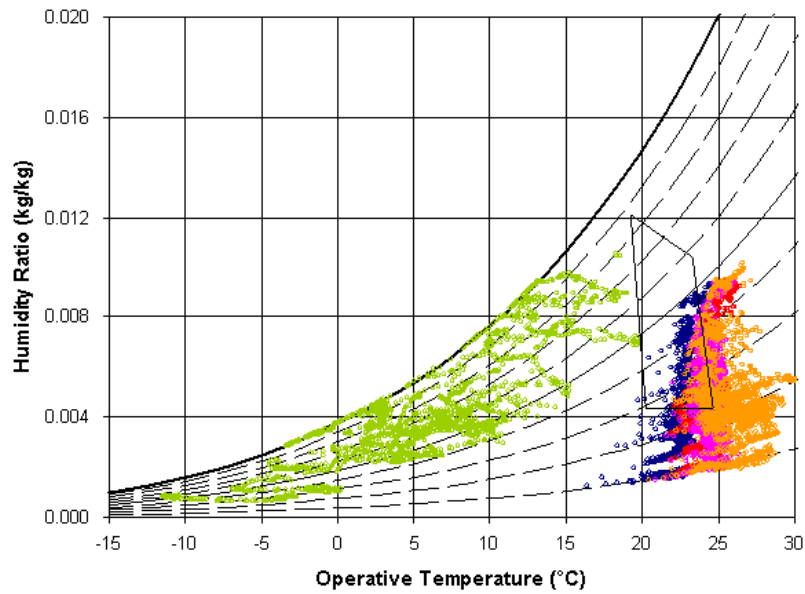


Figure 6.155 : Variation of heating energy, energy consumption of OU2 and HPF of the individual and master control modes with respect to the daily averaged outdoor temperature at 26.1 °C set temperature, (a) individual control mode, (b) master control mode, (c) comparison of the heating energy and the energy consumption, (d) comparison of HPF

Figure 6.156 shows the indoor condition data of System2 zones provided by non-ventilated systems for the continuous and synchronized indoor unit fan operations.

As can be seen from Figure 6.156, considering the similar outdoor conditions, it can be observed that the synchronized indoor unit fan operation increased the number of indoor condition data in the comfort zone by eliminating the potential of the overheating of the zones.

a) Elevator, Receptionist, Aisle, Room E, Outdoor



b) Elevator, Receptionist, Aisle, Room E, Outdoor

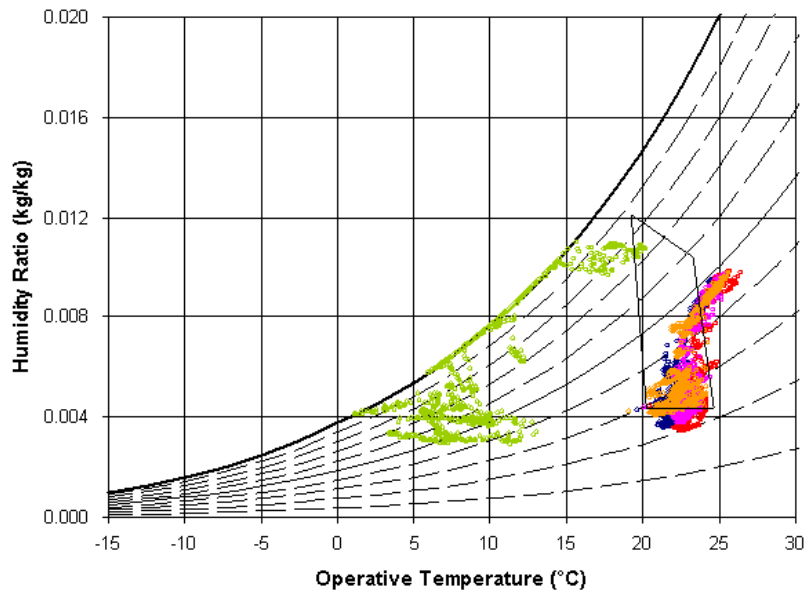


Figure 6.156 : Indoor conditions of System2 zones provided by the continuous and synchronized indoor unit fan operations at 23.3°C set temperature, (a) continuous fan operation, (b) synchronized fan operation

Table 6.18 shows the comparison of the indoor condition data of System2 zones provided by the continuous and synchronized indoor unit fan operation based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature.

It can be observed from Table 6.18 that the synchronized indoor unit fan operation had the potential to increase the indoor thermal comfort.

Table 6.18 : Comparison of the indoor condition data of System2 zones provided by the non-ventilated system for the continuous and synchronized indoor unit fan operations based on the ASHRAE summer thermal comfort zone at 23.3°C set temperature

Region	Percentage of the indoor condition data for the continuous	Percentage of the indoor condition data for the synchronized
In	25.4%	56.1%
Cold	0.0%	0.0%
Hot	17.8%	24.5%
Cold Dry	0.0%	0.0%
Dry	44.2%	18.7%
Hot Dry	12.6%	0.7%
Humid	0.0%	0.0%

7. SIMULATION

7.1 Introduction

As defined in Chapter 4, two multi-split VRF systems with four HRV units were installed in the office suite for the performance evaluation tests. This process including the installation of the units, instrumentation of the test systems and recording long term seasonal data is not cost and time effective.

On the other hand, a simulation model of a multi-split VRF system integrated with HRV system under actual weather information may provide a tool to evaluate the performance of the system. That's why; after evaluating the performance of the multi-split VRF system experimentally, a theoretical evaluation of the system was conducted through simulation. The modeling of the ventilation assisted and non-ventilated multi-split VRF system in cooling mode were performed in EnergyPlus energy simulation software.

Building loads analysis and system thermodynamics (BLAST) and DOE-2, which were developed and released in the late 1970s and early 1980s as energy and load simulation tools, could be assumed to be parents of EnergyPlus which is an energy analysis and thermal load simulation program. Based on a user's description of a building from the perspective of the building's physical envelop, including the heating, ventilation and air conditioning (HVAC) system, EnergyPlus can calculate the cooling loads necessary to maintain the zone set temperatures, HVAC coil loads and the energy consumption of the associated HVAC components as well as many other simulation details that are necessary to verify that the simulation is performing as the actual building would.

EnergyPlus has been extensively validated through analytical and experimental tests. For the calculation of the energy performance of HVAC systems, EnergyPlus uses the performance-curve-oriented way. That's why; detailed performance data are required, and may be obtained from the manufacturer catalogues.

7.2 Building Model

In order to perform the simulation runs, the building information of the office suite, such as the layout of the floor, the dimensions of the building envelop including the

windows, materials of the walls and the windows should be introduced to the EnergyPlus software.

Since EnergyPlus version 1.2.1 is not user-friendly software for inputting the building information, DesignBuilder version 1.1.1, one of the developed interfaces for EnergyPlus, was used to draw the office suite. The building information was obtained from the blueprints of the existing office suite.

After drawing the office floor in DesignBuilder ver.1.1.1, the file was input to the EnergyPlus, so that the office floor could be introduced to the EnergyPlus.

The drawing of the office suite can be seen in Figure 7.1.

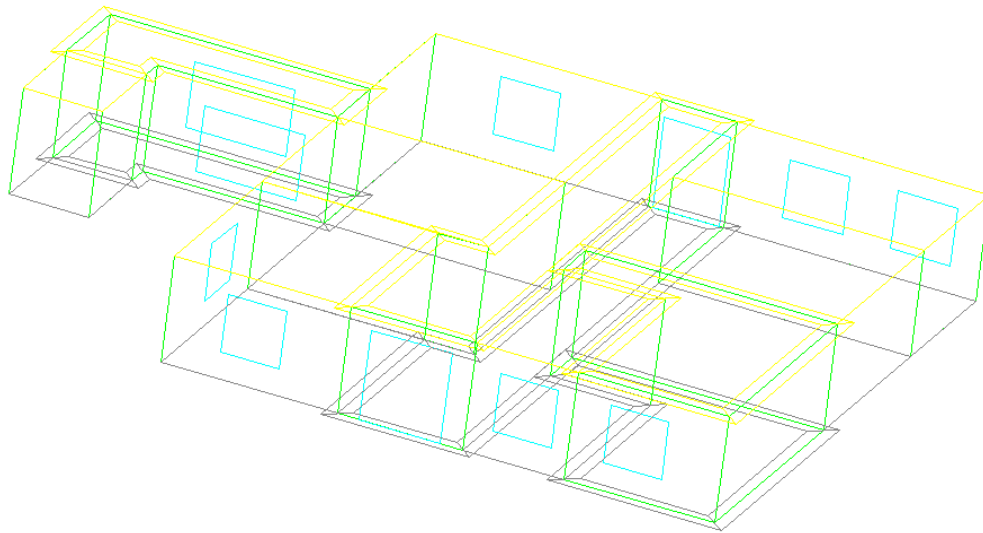


Figure 7.1 : Office suite

The volume of each zone and the dimensions of the windows are provided in Table 4.1 in Chapter 4 and Table 6.4 in Chapter 6, respectively.

Basically, the exterior walls were composed of 2mm steel and 87.5mm glass wool, the inner walls were composed of 2mm steel and 25mm glass wool, and the floor consists of 100mm concrete block. On the other hand, the windows consisted of two 3mm single-pane glazing with a 12mm air gap, and the ceiling was composed of tile, concrete and plaster. In addition, there was an air gap between the zones and the ceiling, where the ventilation units were installed.

Table 7.1 shows the material list used for the inner and outer envelop of the office suite, and the associated thermal conductivities.

Table 7.1 : Material list and the associated thermal conductivity

Material	Conductivity (W/mK)
Steel	45
Glass wool	0.042
Concrete Block	0.54
Concrete	1.4
Plaster	0.79
Tile	1.28
Carpet	0.0805
Air	0.022
Wood	0.175
Glass	0.9

The internal load sources such as occupants, lights and office equipments are crucial for the load calculations and should be input to EnergyPlus. Table 6.5 shows the number of internal load sources of the OU1 and OU2 zones.

The activity of the occupants can be assumed very light office work, and according to the “ASHRAE handbook, Fundamentals 2005”, the associated representative rate of heat emission by the occupants is assumed to be 131W. 33% of the total emission can be classified as the radiant heat gain (Hosni et al., 1998).

48” T8 fluorescent fixtures were used for the lighting, and Table 7.2 shows the fixture watts according to the “ASHRAE handbook, Fundamentals 2005”.

Table 7.2 : Light fixtures used in the office suite

Fluorescent Fixture	Lamps/Fixture	Fixture Watts
48”, T8 lamp	2	71
48”, T8 lamp	4	142

50% of the total emission can be classified as the radiant heat gain (Hosni et al., 1998).

On the other hand, the total heat gain from the office equipment such as desktop computers, monitors, copy machines and printers were significantly less than the name plate ratings and the actual power consumption ranged from 14% to 36% of

the name plate ratings (Hosni et al., 1998). That's why, instead of using the name plate ratings, the following recommended heat gain by the "ASHRAE handbook, Fundamentals 2005", Hosni et al. (1999) and Hosni et al. (1998), shown in Table 7.3, are chosen.

Table 7.3 : Recommended heat gain for the office equipments

Office Equipment	Heat gain (W)
Desktop computer with monitor	150
Small office printer	70
Desktop copier	20

The heat gain from the coffee brewer located in the reception area is assumed to be 170W according to the "ASHRAE handbook, Fundamentals 2005" and the heat gain from the mini refrigerators are also assumed to be 170W.

7.3 Multi-Split VRF and HRV Models

7.3.1 Multi-Split VRF Model

As described in Chapter 7.1, EnergyPlus is capable of calculating the cooling loads of each zone necessary to maintain the zone set temperatures, and associated coil loads and the energy consumption of the related HVAC components. However, the default HVAC modules in EnergyPlus do not include a multi-split VRF system. That's why; the default version of EnergyPlus can not be used for the simulation study of the multi-split VRF systems.

On the other hand, as described in the literature review, Zhou et al. (2007a) developed a module for EnergyPlus to evaluate the multi-split VRF systems, and validated the module with the experimental steady-state study (Zhou et al., 2007b), (Zhou et al., 2007c). This module was used for the current simulation study.

Figure 7.2 shows the flowchart of the schematic process of the modeling of the multi-split VRF system in EnergyPlus, which was defined in the "Input Processor" module to get the required information for the system.

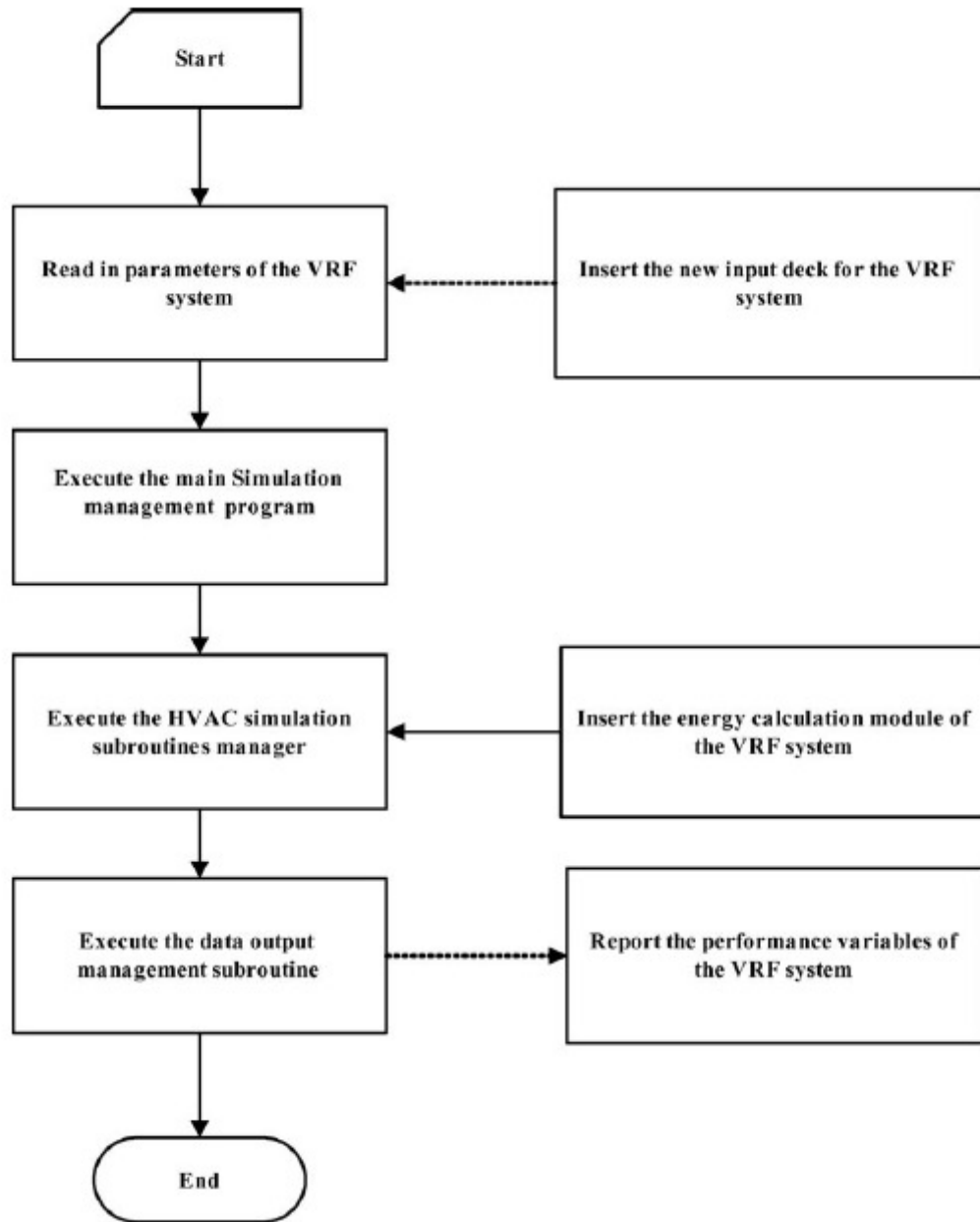


Figure 7.2 : Flowchart of the developed multi-split VRF model for the EnergyPlus

Figure 7.3 shows the schematic flowchart of the main energy consumption calculation program for the multi-split VRF system.

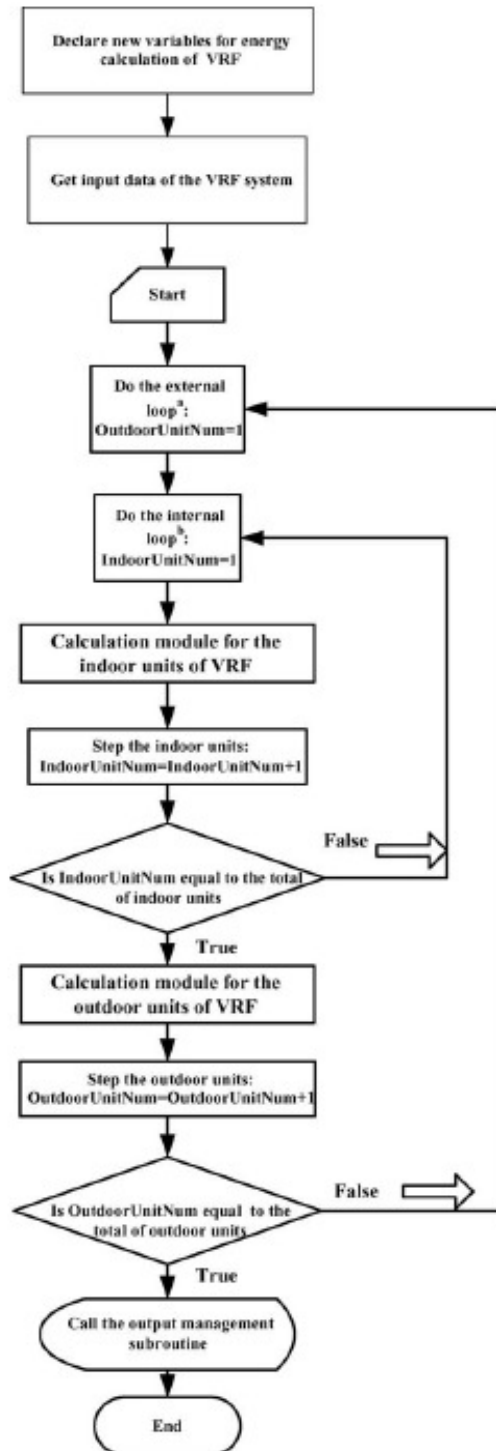


Figure 7.3 : Flowchart of the main program for energy-use computation of the multi-split VRF system

Basically, the developed model uses the existing air-cooled direct expansion (DX) coil in EnergyPlus, which was assumed to be the indoor unit of a multi-split VRF system. This existing model determines the performance of the DX cooling coil at

part-load conditions, through the performance information at rated conditions along with curve fits for variations in total capacity, energy input ratio and part load fraction.

EnergyPlus uses five performance curves for the evaluation of the total cooling capacity and the efficiency at part-load conditions for the DX cooling coil:

- Total cooling capacity modifier curve as a function of the temperature
- Total cooling capacity modifier curve as a function of the flow fraction
- Energy input ratio (EIR) modifier curve as a function of the temperature
- Energy input ratio modifier curve as a function of the flow fraction
- Part-load fraction (PLF) correlation as a function of the part-load ratio (PLR)

Since it was found that the indoor units operated continuously at constant air flow rate during the tests in the cooling mode, the effect of the flow fraction on the total cooling capacity and on the energy input ratio can be neglected.

The rest of the modifier curves are defined in EnergyPlus manual as

$$TotCapTempModFac = a + b \cdot T_{wb,i} + c \cdot T_{wb,i}^2 + d \cdot T_{c,i} + e \cdot T_{c,i}^2 + f \cdot T_{wb,i} \cdot T_{c,i} \quad 7.1$$

$$EIRTempModFac = a' + b' \cdot T_{wb,i} + c' \cdot T_{wb,i}^2 + d' \cdot T_{c,i} + e' \cdot T_{c,i}^2 + f' \cdot T_{wb,i} \cdot T_{c,i} \quad 7.2$$

$$PLF = a'' + b'' \cdot PLR + c'' \cdot PLR^2 \quad 7.3$$

where

TotCapTempModFac : Total cooling capacity modifier with respect to temperatures

a, b, c, d, e, f : Coefficients for the cooling capacity modifier curve

T_{wb,i} : WB temperature of the air entering the cooling coil (°C)

T_{c,i} : Ambient DB temperature (°C)

EIRTempModFac : EIR modifier with respect to temperatures

a', b', c', d', e', f' : Coefficients for the energy input ratio modifier curve

PLF : Part-load fraction

a'', b'', c'' : Coefficients for the part-load fraction correlation

PLR : Part load ratio

The performance of the multi-split VRF system including the power consumption of the outdoor unit and the efficiency are defined by Zhou et al. (2007c) as

$$Power = \left[\sum_{k=1}^{n_{IU}} \left(\dot{Q}_{total} \cdot RTF \cdot MF \right)_k \right] \cdot EIR \quad 7.4$$

$$EIR = \frac{EIRTempModFac}{COP_{rated}} \quad 7.5$$

where

$\dot{Q}_{total}$: Operating cooling capacity of the single DX coil (W)

RTF : Runtime fraction of the cooling coil

MF : Modifier at part-load ratio

n_{IU} : Total number of indoor units

EIR : Reverse of COP curve-fitted from the catalogue data

The coefficients of the total cooling capacity and EIR modifier curves were obtained from the manufacturer catalogue data by curve fitting. The other necessary data such as standard cooling capacities of the indoor and outdoor units, air flow rates of the indoor units were also obtained from the manufacturer catalogue.

Even though, a multi-split VRF module had been developed for EnergyPlus, a real office suite application for a multi-split VRF system integrated with a ventilation system was not performed, and will be performed in this current simulation study.

7.3.2 HRV Model

Since HRV units are well-known systems and have been extensively used for the ventilation purposes, EnergyPlus has already a module for that kind of systems named as stand alone energy recovery ventilator (ERV). This module was used for the simulation part of the HRV units.

Figure 7.4 shows the schematic diagram of a stand alone ERV. As can be seen, basically, a stand alone ERV is composed of supply and exhaust air fans and an air-to-air heat exchanger, and this module models the basic operation of the fans and the heat exchanger. The stand alone ERV module can be used in conjunction with an economizer feature. During the economizer operation, the heat transfer between the supply and the exhaust air streams through the air-to-air heat exchanger is

suspended, and the supply and exhaust air flows are fully bypassed around the heat exchanger, which allows free-cooling.

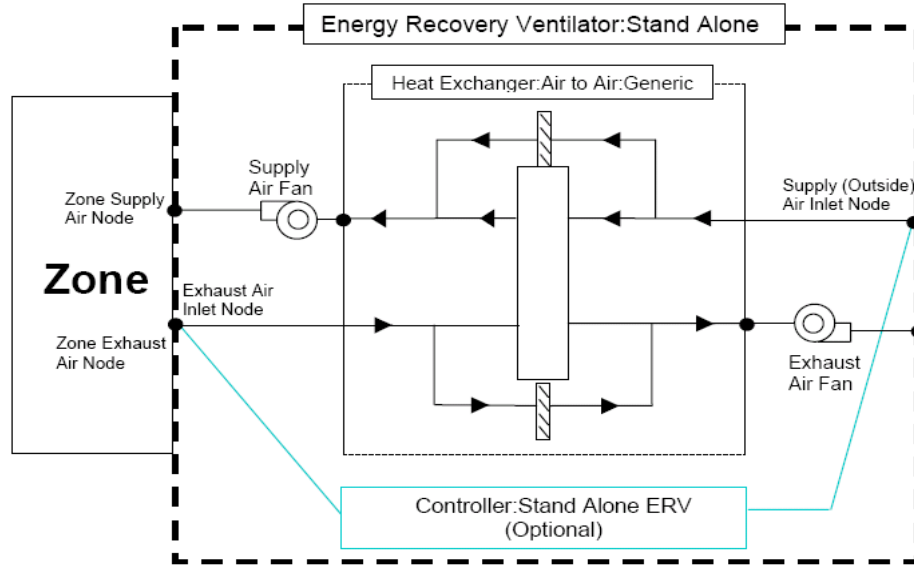


Figure 7.4 : Schematic diagram of a stand alone ERV (EnergyPlus Engineering Document)

The total, sensible and latent heat transfer rate to the zone by the ventilation unit are calculated as

$$\dot{Q}_{Total,v} = \dot{m}_{Supply} \cdot (h_{Supply,Outlet} - h_{Exhaust,Inlet}) \quad 7.6$$

$$\dot{Q}_{Sensible} = \dot{m}_{Supply} \cdot (h_{Supply,Outlet} - h_{Exhaust,Inlet})_{HR\ min} \quad 7.7$$

$$\dot{Q}_{Latent} = \dot{Q}_{Total,v} - \dot{Q}_{Sensible} \quad 7.8$$

where

$\dot{Q}_{Total,v}$: Total energy transfer rate to the zone (W)

$\dot{Q}_{Sensible}$: Sensible energy transfer rate to the zone (W)

$\dot{Q}_{Latent}$: Latent energy transfer rate to the zone (W)

$\dot{m}_{Supply}$: Mass flow rate of the supply air stream (kg/s)

$h_{Supply,Outlet}$: Enthalpy of the air being supplied to the zone (J/kg)

$h_{Exhaust, Inlet}$: Enthalpy of the air being exhaust from the zone through the unit (J/kg)

HR_{min} : Enthalpies evaluated at a constant humidity ratio, the minimum humidity ratio of the supply air outlet or the exhaust air inlet

The resulting energy transfer rate is added to the zone load to be met by cooling system.

7.4 Scheduling

EnergyPlus intends to perform the actual building conditions during the simulation by using the schedule module which consists of three descriptions; daily, weekly and annually. This module allows the user to specify scheduling of many items such as the occupancy density and the occupancy activity in each zone, lighting of each zone, set temperatures of the zones, the operation of the cooling system. They can be introduced to the software hourly by the daily description, daily by the weekly description and monthly by the annually description.

For the existing office suite, the multi-split VRF systems in conjunction with the HRV units were operated within 7am and midnight. The occupants were assumed to be in their rooms from 8am to 5pm on the weekdays, and the associated lights and the electric equipments such as computers, printers and copy machines were assumed to be “ON”, during the same time period.

The above scheduling information was input to EnergyPlus.

7.5 Weather Data

EnergyPlus uses a weather file to define the outdoor environment, which basically consists of the outdoor dry bulb and dew point temperatures, relative humidity, humidity ratio, barometric pressure, wind speed, wind direction, diffuse and direct solar data, as well as the conditions such as rain, snow etc.

The default weather files are prepared by using representative monthly data. Table 7.4 shows the default weather file for Sterling, Virginia (VA) which is around 30 miles away from College Park, Maryland (MD). Since the locations are close, the outdoor conditions are assumed to be same. As can be seen from Table 7.4, after long-term data collection, for instance, January in 1976 and February in 1988 are assumed to

be the representative months for all Januaries and all Februaries for Sterling, VA, respectively.

Table 7.4 : Default weather file of EnergyPlus for Sterling, VA

Year	Month	Year	Month
1976	January	1990	July
1988	February	1986	August
1974	March	1990	September
1972	April	1986	October
1987	May	1962	November
1985	June	1981	December

However, actual weather condition data is crucial for the validation of the simulation model, because simulation and experimental results should be compared under the same outdoor condition for the validation. Since same outdoor conditions can not be found from the representative monthly data, the default weather file can not be used for the evaluations. That's why; actual measured outdoor condition data; such as outdoor temperature and relative humidity, are input to the weather file. On the other hand, since the solar data was not recorded during the experiments, the solar data for Sterling, VA was downloaded from National Oceanic and Atmospheric Administration (NOAA) website, and input to the weather file. The other important weather condition data, such as wind direction and wind speed, rain etc. was found from the website of "wunderground.com", and input to the modified weather file.

Figure 7.5 shows the outdoor temperature and relative humidity variation from August 01 to August 07, 2006 for the existing weather file for Sterling, VA, and for the modified weather file with the measured outdoor condition data.

As can be seen, even though the trends of the variations are similar for some days, there are significant discrepancies between the existing and modified weather files. That's why; modified weather file was used for the current study.

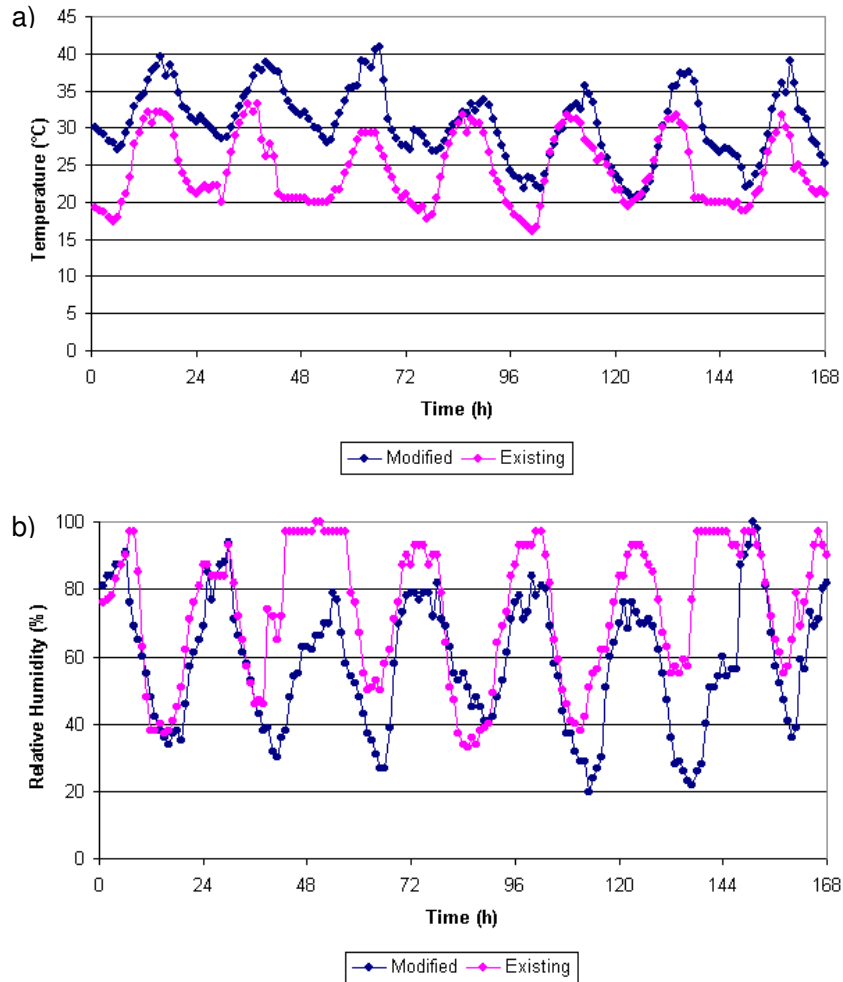


Figure 7.5 : Comparison of the outdoor conditions, (a) temperature, (b) relative humidity

7.6 Validation of the Model with Experimental Results

After introducing the building information and inputting the necessary data of the multi-split VRF system with the HRV units and specifying the schedule information in EnergyPlus, the validation of the model was performed by comparing the simulation runs and the experimental data collected on Mondays in August 2006 (August 07, August 14, August 21 and August 28). As described in Chapter 7.5, the actual outdoor conditions were input to the weather file, and hourly averaged simulation results and experimental data were compared.

Figure 7.6 shows the comparison of the indoor temperature of the System1 and Ssystem2 zones between simulation and experiment.

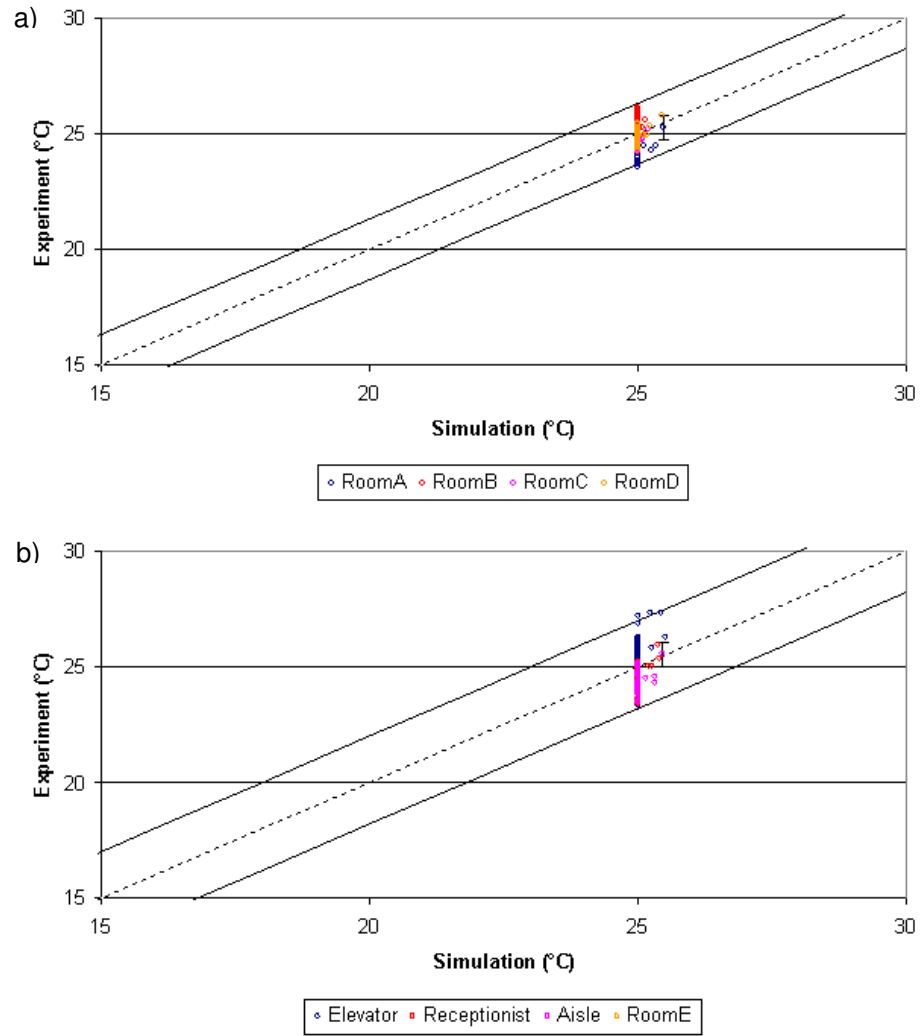


Figure 7.6 : Comparison of indoor temperature between simulation and experiment, (a) System1 zones, (b) System2 zones

It is observed from Figure 7.6 that there is good agreement between the experimental and simulation results for the indoor temperatures. The difference between the experimental and predicted indoor temperature was within $\pm 1.3^{\circ}\text{C}$ and $\pm 1.8^{\circ}\text{C}$ for the System1 and System2 zones, respectively.

Figure 7.7 shows the comparison of indoor relative humidity of the System1 and System2 zones between simulation and experiment.

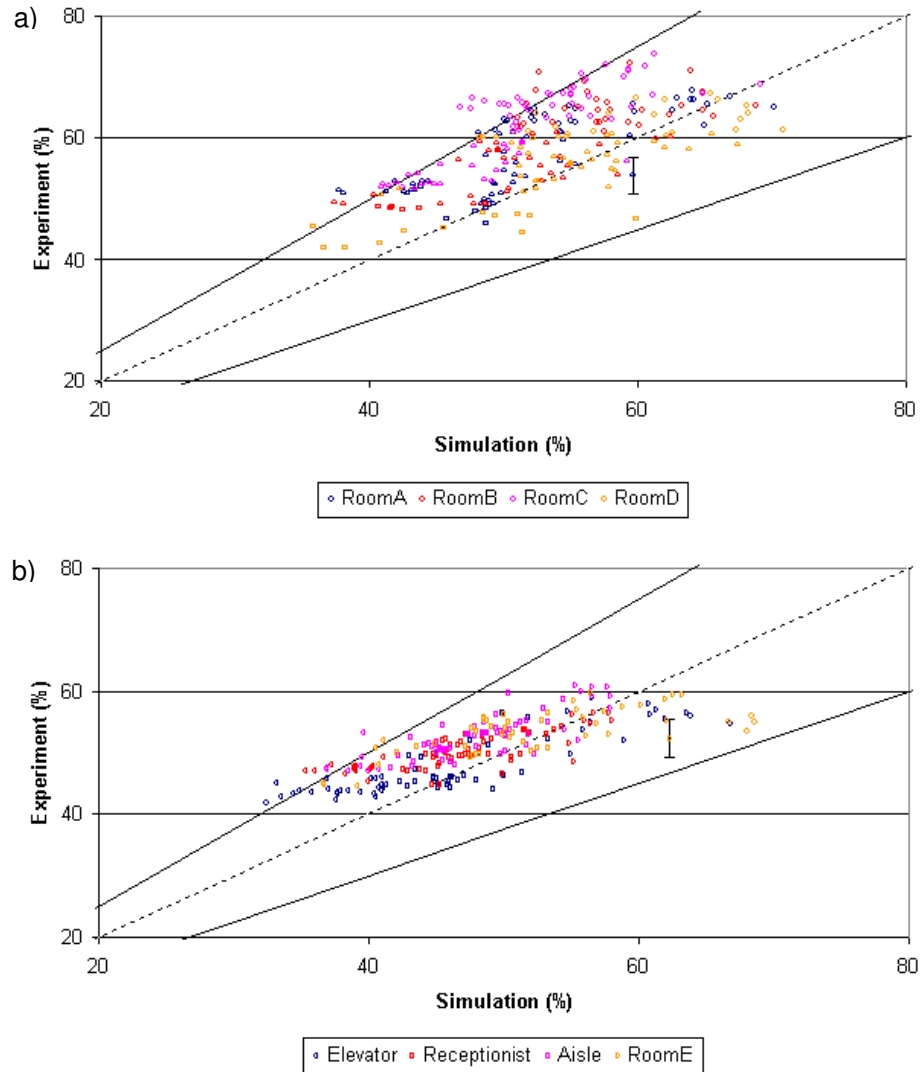


Figure 7.7 : Comparison of indoor relative humidity between simulation and experiment, (a) System1 zones, (b) System2 zones

It is seen from Figure 7.7 that the indoor relative humidity ratio found from experimental results falls within $\pm 25\%$ of the predictions from the simulation.

The comparison between the experimental data and the simulation result for the total power consumption of the HRV units and the indoor units' fans is plotted in Figure 7.8.

The HRV units were operated continuously, and the fans of the indoor units were also operated continuously under "low speed" configuration throughout the tests.

Each fan pressure drop (indoor unit fan, supply and exhaust fans of the HRV unit) in EnergyPlus was adjusted in order to give the individual power consumption, provided in App. [N].

It is observed that the total power consumption found from experiment falls within $\pm 3\%$ of the predictions from the simulation.

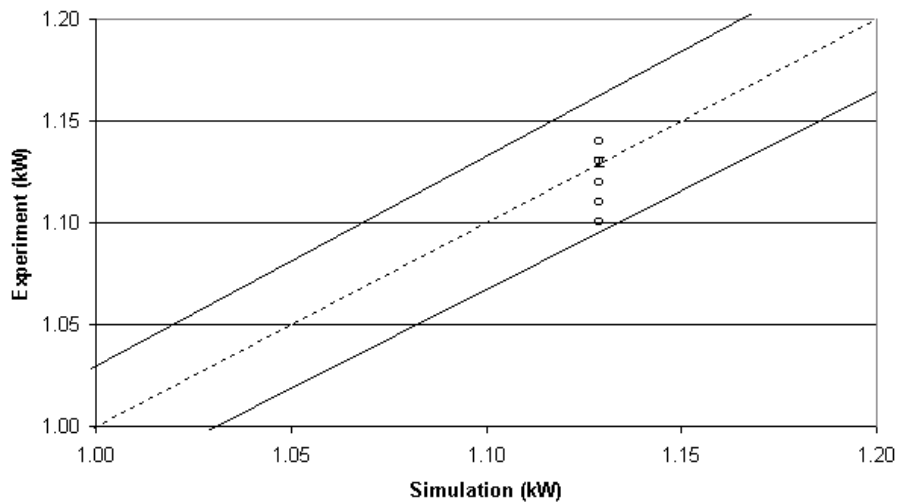


Figure 7.8 : Comparison of total power consumption of HRV units and indoor units between simulation and experiment

The variation of the experimental data and the simulation results for the outdoor unit power consumption is plotted in Figure 7.9 for Mondays August 2006.

The comparison between the experimental data and the simulation results for the power consumption of the outdoor units is shown in Figure 7.10.

It is seen from Figure 7.10 that the outdoor unit power consumptions found from the experimental results mostly falls within $\pm 25\%$, which agrees with Zhou et al. (2007b) and Zhou et al. (2007c). It is found that 66.2% and 86.8% of all data for OU1 and OU2 fall within $\pm 25\%$, respectively. The reason of less OU1 data in the boundary is due to the more fluctuations related to “ON/OFF” cyclic operations.

In addition, the solar data downloaded from NOAA was input and used for the simulation. Since the solar data has a significant role in the cooling load calculations, the accuracy of the data has a significant effect on the power consumption of the outdoor unit.

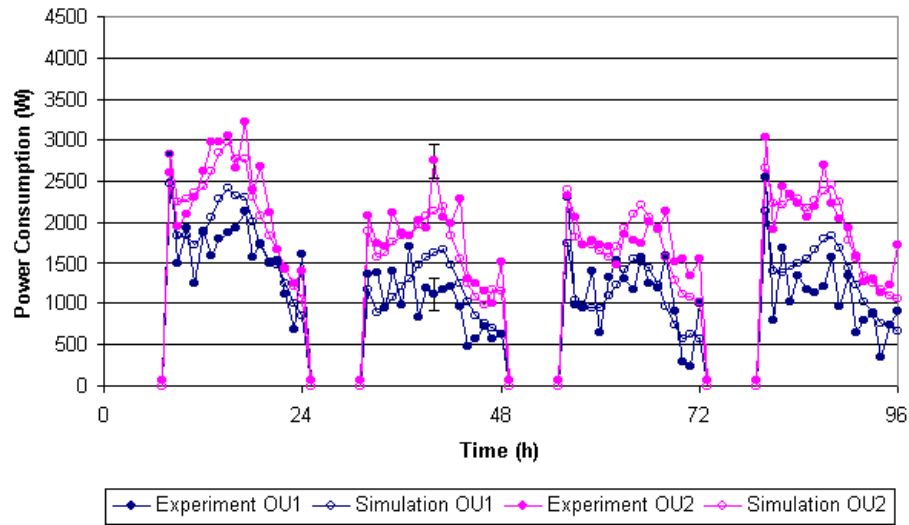


Figure 7.9 : Variations of the experiment and simulation outdoor unit power consumptions

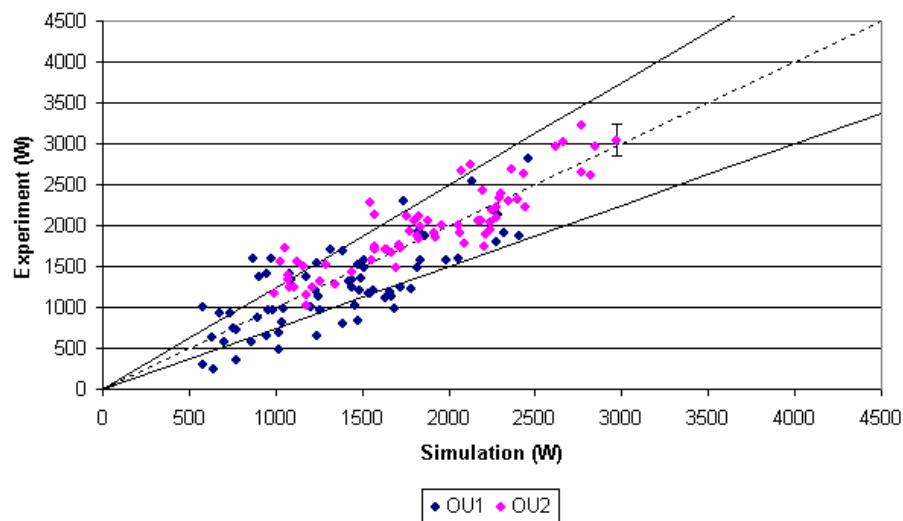


Figure 7.10 : Comparison of OU1 and OU2 power consumptions between simulation and experiment

The comparison between the experimental data and the simulation results for the daily CPF and PLR is shown in Figure 7.11. Due to the problem in the data acquisition system, the refrigerant pressure measurements could not be performed on August 21, 2007. That's why; evaluation on August 21, 2007 could not be performed.

It is observed from Figure 7.11 that there is good agreement between the experimental and simulation results for the daily CPF and PLR.

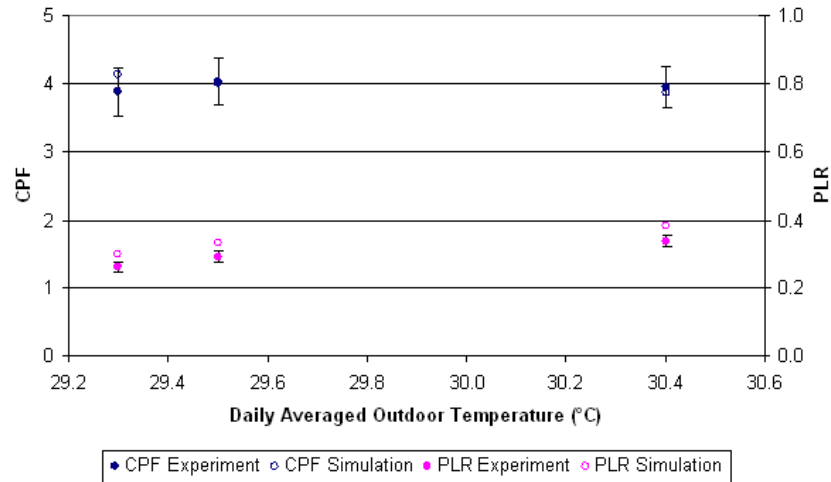


Figure 7.11 : Comparison of daily CPF and PLR between simulation and experiment

7.7 Discussion of Simulation Results

After the validation of the simulation model with the experimental data, the model was run for one cooling season under actual outdoor conditions of 2007.

The cooling season is considered to comprise of three months from June to August. The multi-split VRF system is assumed to be operating 24 hours on weekdays. For the ventilation assisted systems, the HRV units are assumed to be operating 24 hours in conjunction with the multi-split VRF systems.

In order to make the office suite much more generic, four people are added to Room B and one people is added to Room C, because Rooms B and C were mostly unoccupied during the experiments, as explained in Chapter 4.1.

Figure shows Maryland outdoor conditions during the period of June-August 2007.

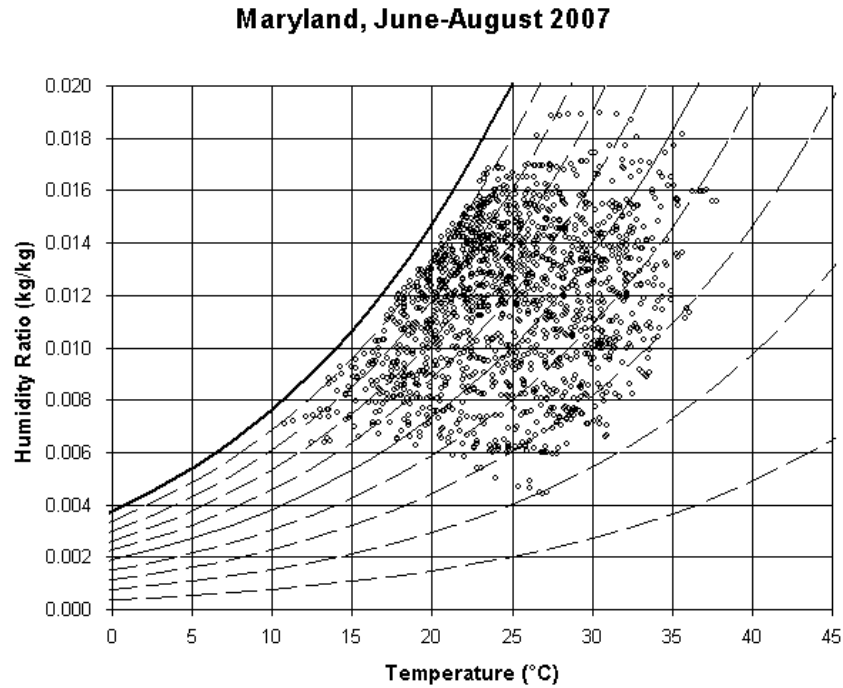


Figure 7.12 : Maryland outdoor conditions during the period of June-August 2007

Figure 7.13 shows the weekly energy consumption of System1 and System2 in terms of kWh. The fans of the indoor units and the HRV units are operated continuously throughout the operation time, and the indoor set temperatures are kept at 25°C. The weekly averaged outdoor temperature with minimum and maximum temperatures for the corresponding week can also be seen from Figure 7.13.

It can be observed from Figure 7.13 that the total energy consumption of System1 indoor units is higher than that of System2 indoor units.

As can be seen from Figure 7.13, the energy consumptions of the outdoor units increase with the increasing outdoor temperature in order to keep the indoor temperatures at the set temperature. When the weekly averaged outdoor temperature is 23°C in the second week of June, the energy consumptions of OU1 and OU2 are 79.1kWh and 97.1kWh, respectively, and when the weekly averaged outdoor temperature increases to 28.2°C in the first week of August, the energy consumptions of OU1 and OU2 increase to 145.4kWh and 157.0kWh, respectively.

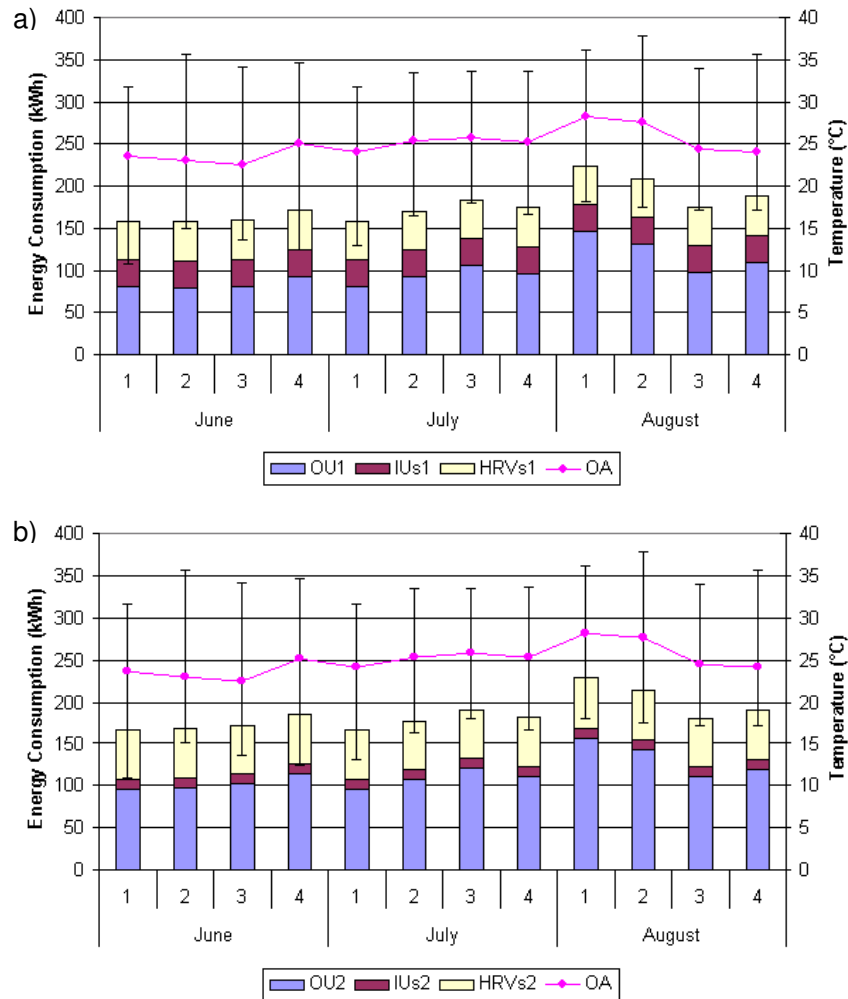


Figure 7.13 : Weekly energy consumptions of System1 and System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2

Figure 7.14 shows the percentages of the outdoor, indoor and HRV units in the total energy consumption of System1 and System2.

As can be seen from Figure 7.14, OU1 and OU2 consume around 50-65% and 51-69% of the total energy consumption of System1 and System2, respectively, throughout the season.

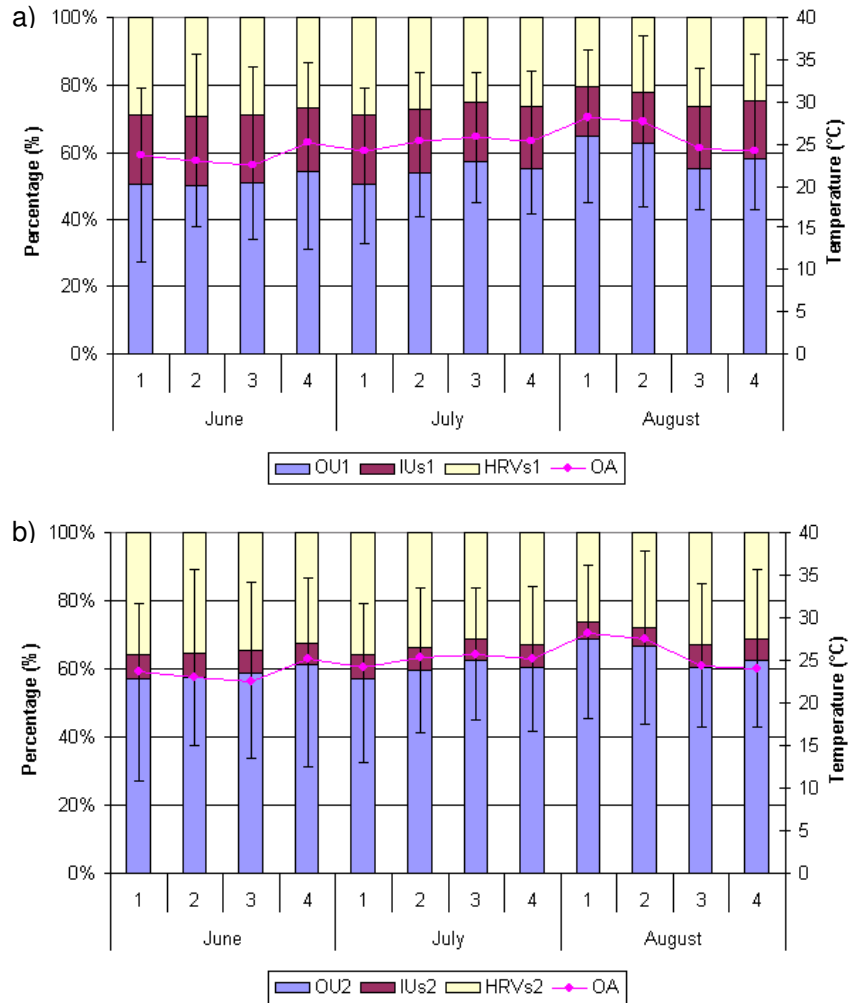


Figure 7.14 : Percentages of the outdoor, indoor and HRV units in the total energy consumption of System1 and System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2

Figure 7.15 shows the indoor conditions of System1 and System2 zones during the period of June-August 2007 with hourly averaged data. As can be seen, the indoor set temperature of 25°C can be maintained almost all of the operation time. The indoor temperatures lower than the set temperatures are due to the very low outdoor temperature. Ventilation cooling effect is higher than the internal heat gain when the outdoor temperature is very low, which causes indoor temperatures become lower than the set temperatures.

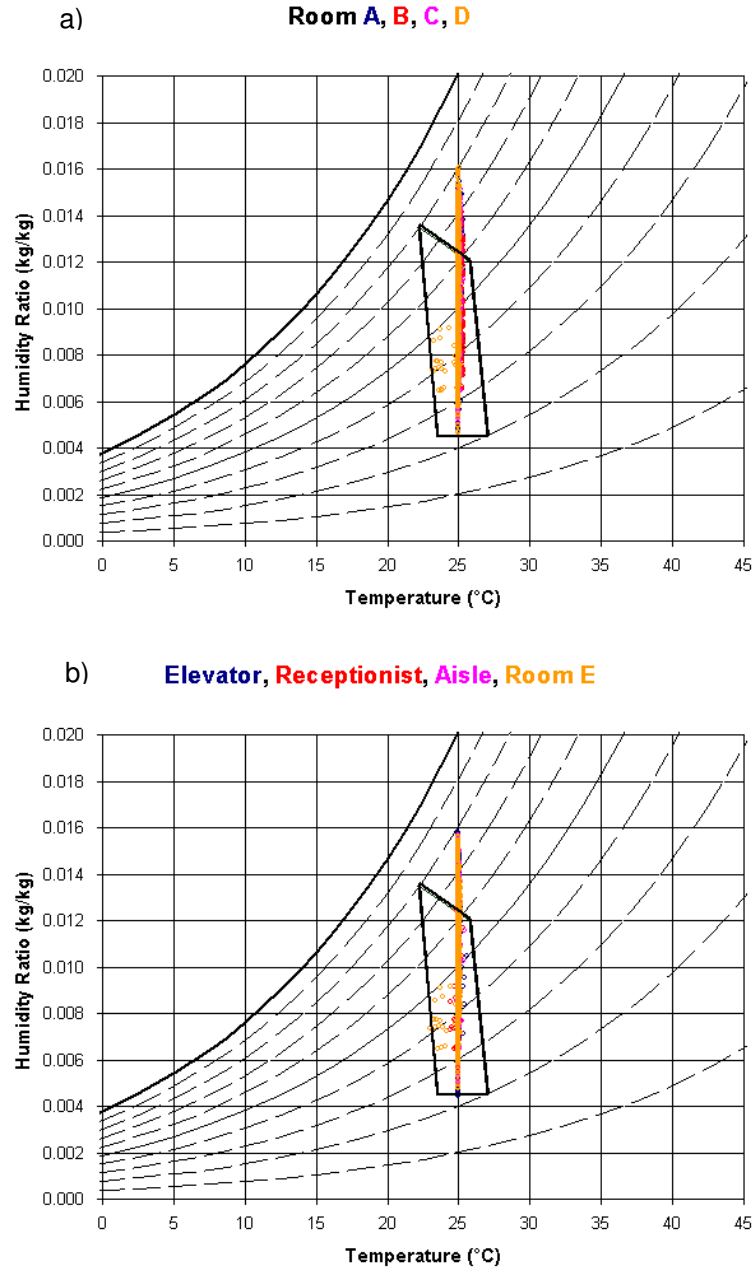


Figure 7.15 : Indoor conditions of System1 and System2 zones at 25°C indoor set temperature for MD, (a) System1 zones, (b) System2 zones

Figure 7.16 shows the weekly energy consumption of non-ventilated System1 and non-ventilated System2 in terms of kWh. Considering Figure 7.13 and Figure 7.16, it can be observed from that since the ventilation load is not introduced to the indoors for the non-ventilated system, the energy consumptions of outdoor units are lower compared to the ventilation assisted systems.

It is found that the energy consumptions of OU1 and OU2 are 59.0kWh and 74.0kWh for the second week of June which are 26% and 24% lower than the ventilation assisted system for the same period, and 93.2kWh and 105.4kWh for the first week of August which are 36% and 33% lower than the ventilation assisted systems for the same period, respectively.

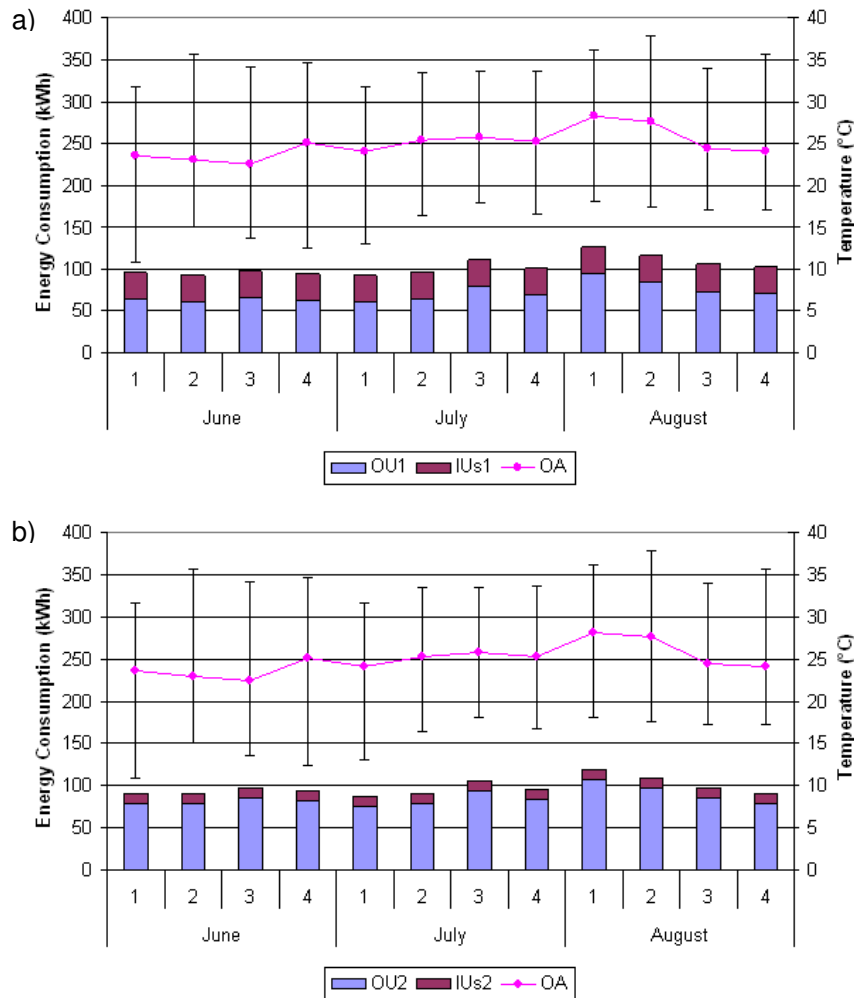


Figure 7.16 : Weekly energy consumptions of the non-ventilated System1 and non-ventilated System2 at 25°C indoor set temperature for MD, (a) System1, (b) System2

Figure 7.17 shows the percentages of the outdoor and indoor units in the total energy consumption of the non-ventilated System1 and non-ventilated System2. Since there is no HRV unit in operation, the percentages of the outdoor units in the total energy consumption increases. As seen, OU1 and OU2 consume 65-74% and 86-90% of the total energy consumption of the systems, respectively, throughout the season.

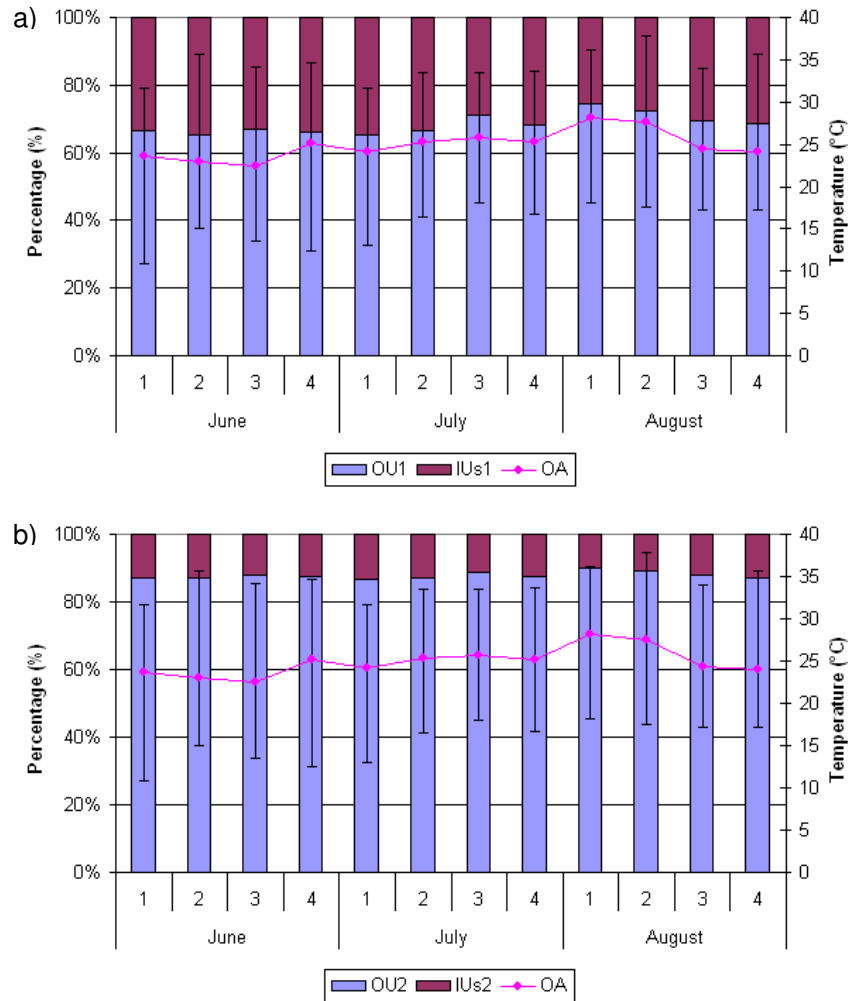


Figure 7.17 : Percentages of the outdoor and indoor units in the total energy consumption of the non-ventilated System1 and non-ventilated System2, (a) System1, (b) System2

Figure 7.18 depicts the indoor conditions of non-ventilated System1 and System2 zones.

As can be seen from Figure 7.18, similar to the ventilation assisted systems, the indoor set temperature of 25°C can be maintained almost all of the operation time by the non-ventilated systems.

The main difference between the ventilation assisted and non-ventilated systems is the indoor humidity ratio, which was also observed during the experiments.

As seen from Figure 7.18, the indoor humidity ratios for the non-ventilated systems are remarkable lower compared to the ventilation assisted systems, shown in Figure 7.15. This is because the ventilation introduces not only sensible load, but also latent load, which increases the indoor humidity ratio.

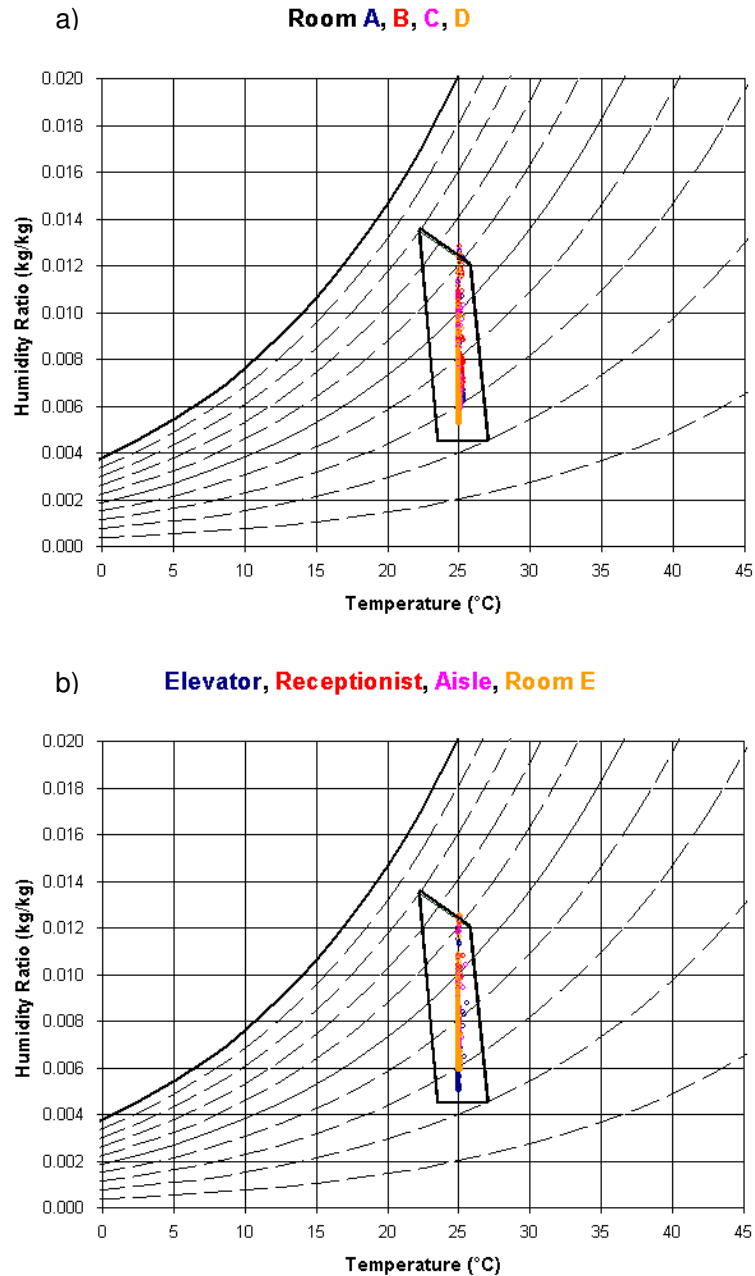


Figure 7.18 : Indoor conditions of the non-ventilated System1 and System2 zones at 25°C indoor set temperature for MD, (a) System1 zones, (b) System2 zones

Two possible energy saving options can be taken into account according to the results of previous evaluations. These options are

- Economizer
- Cyclic indoor unit fan operation

The economizer provides free cooling during the ventilation, when the outdoor and indoor condition criteria are satisfied as described in Chapter 7.3.2. Thus, due to the free cooling possibility, there is an energy saving potential for the outdoor units.

During the experiments, regardless of the indoor unit operation, the fans of the indoor units were operated continuously during the operation time. However, there is an energy saving potential for the indoor units when they are operated cyclically. Namely, the fan can be operated when the indoor unit is working, and can be turned “OFF” when the indoor unit is not working. Since the “ON” time ratios of the fans decrease by shifting from the continuous fan operation to cyclic fan operation, the corresponding energy consumption decreases. Besides, since the fans operate less for the cyclic fan operation, the heat gain coming from the fans reduces accordingly which decreases the cooling load and ends up with an energy saving potential for the outdoor units.

These two possible energy saving options can be applied to the ventilation assisted system, and the cyclic indoor unit fan operation option can be applied to the non-ventilated system.

In order to investigate those options clearly, two additional indoor set temperatures, 23°C and 27°C and two additional climates, hot and cold are taken into account. Phoenix, Arizona (AZ) and Los Angeles, California (CA) are chosen for the hot and cold climate applications, respectively.

Figure 7.19 shows the outdoor conditions of Phoenix, AZ and Los Angeles, CA downloaded from the website of wunderground.com during the period of June-August 2007.

As can be seen from Figure 7.19, the outdoor temperature and humidity ratio vary within 21.5-45°C and 1.7-19.4g/kg for Phoenix, AZ and 13.6-27°C and 7-13.3g/kg for Los Angeles, CA, respectively.

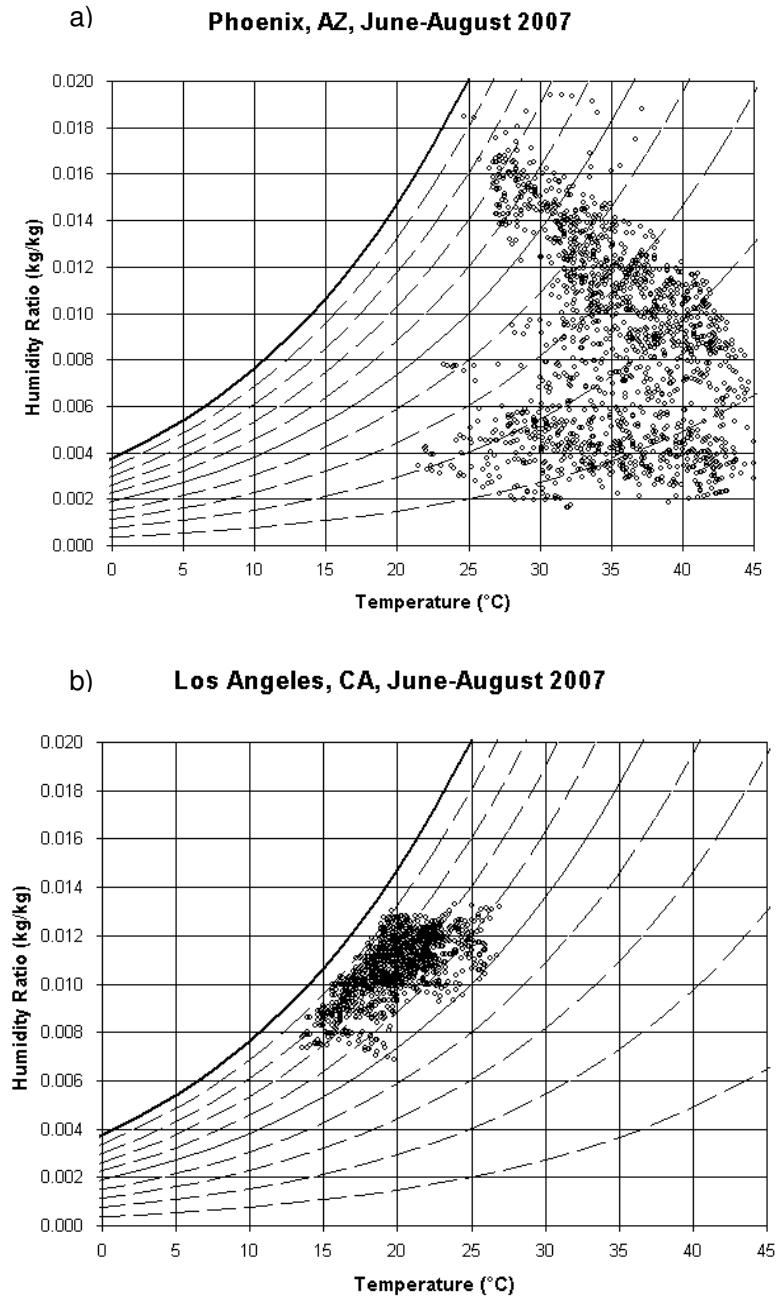


Figure 7.19 : Outdoor conditions of Phoenix, AZ and Los Angeles, CA during the period of June-August 2007, (a) Phoenix, AZ, (b) Los Angeles, CA

7.7.1 Ventilation assisted system

Figure 7.20 shows the weekly energy consumptions of System1 with and without economizer and continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD.

The indoor set temperature was chosen for the high temperature limit, and 15°C was chosen for the low temperature limit for the economizer. In addition, the exhaust

air enthalpy was used as a third parameter. Namely, when the outdoor temperature is within the high and low temperature limits, and the enthalpy of the outdoor air is lower than the exhaust air enthalpy, then the air flow is fully bypassed around the heat exchanger in the HRV unit in order to provide free cooling. The economizer option is terminated under 15°C in order to prevent overcooling which causes uncomfortable indoor environment.

As seen can be seen from Figure 7.20, regardless of the configuration (w/ or w/o economizer, continuous or cyclic fan operation), the outdoor unit energy consumption increases with the decreasing indoor set temperature.

For the first week of August, OU1 consumes 158.6, 145.4 and 130.2kWh energy at 23°C, 25°C and 27°C indoor set temperatures, respectively, when there is no economizer and the fans are operated continuously. On the other hand, it consumes 157.7, 143.4 and 126.9kWh energy for the same period at 23°C, 25°C and 27°C indoor set temperatures, respectively, when the economizer is “ON” and the fans are operated continuously. When the fans are operated cyclically with economizer, it consumes 150.1, 135.4 and 118.6kWh for the same period at 23°C, 25°C and 27°C indoor set temperatures, respectively.

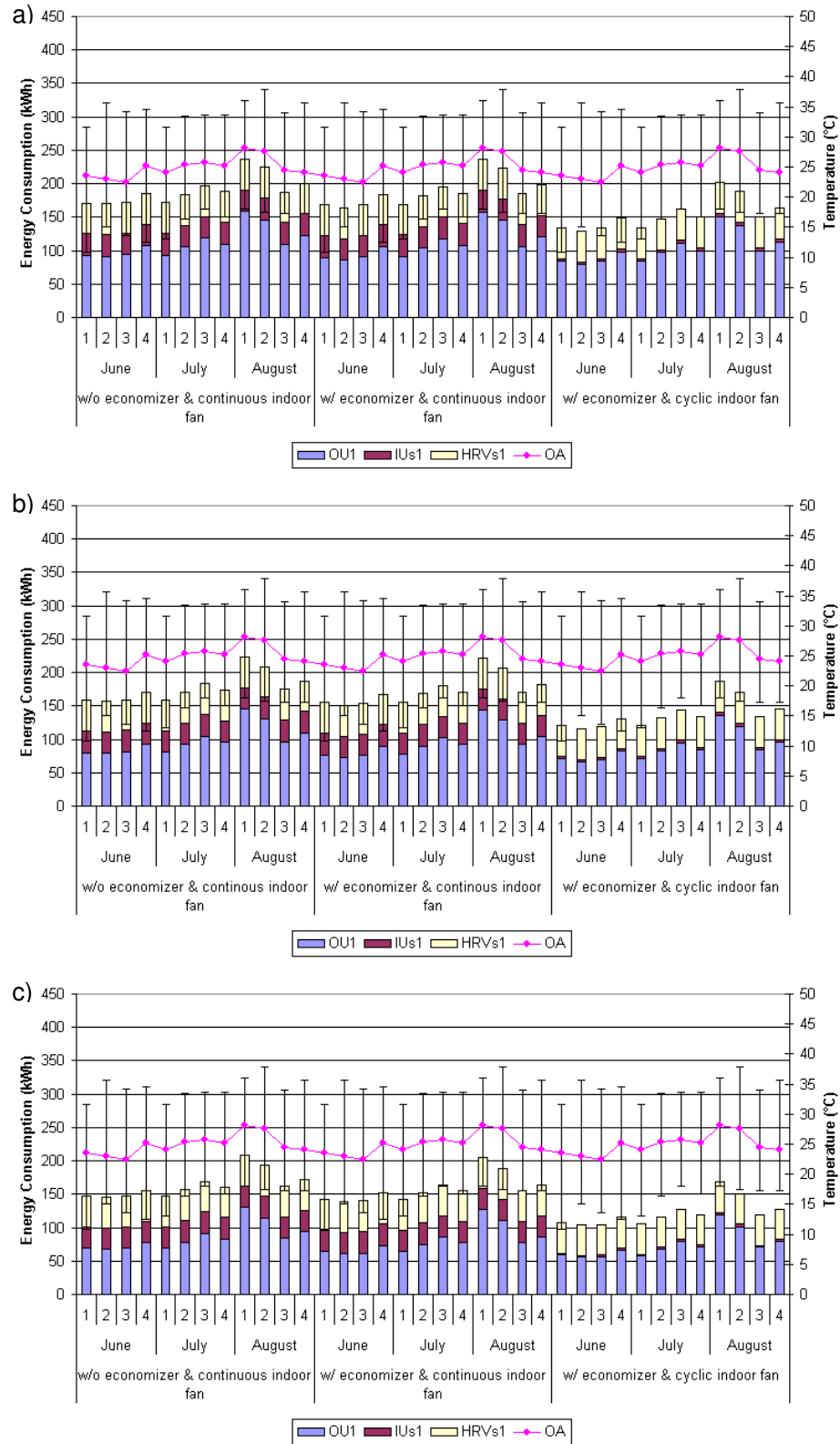


Figure 7.20 : Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.21 shows the energy savings of outdoor and indoor units of System1 with and without economizer and continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD. Since the economizer and/or cyclic indoor fan operation do not have any effect on the energy consumption of the HVR units, it is not shown in the evaluation.

It is observed that economizer has the potential to reduce the outdoor unit energy consumption by providing free cooling. It is found that the energy saving of the outdoor unit is around 0.5-6%, 1.3-8% and 3-11% at 23°C, 25°C and 27°C indoor set temperatures, respectively. Since free cooling possibility increases with the increasing indoor set temperature due to the increasing of the temperature limit for the economizer control criteria, the saving of the outdoor unit energy consumption increases with the increasing indoor set temperature.

It can be observed from Figure 7.21 that the cyclic indoor fan operation has a potential to reduce the indoor units' energy consumption. It is found that when the fans of the indoor units are operated cyclically instead of continuously, the total energy consumption of the fans decreases within 81-89%, 84-92% and 87-94% at 23°C, 25°C and 27°C indoor set temperatures, respectively. The energy saving increases with the increasing indoor set temperature, because for the high indoor set temperature, the multi-split VRF system can maintain the set temperature with less operation time.

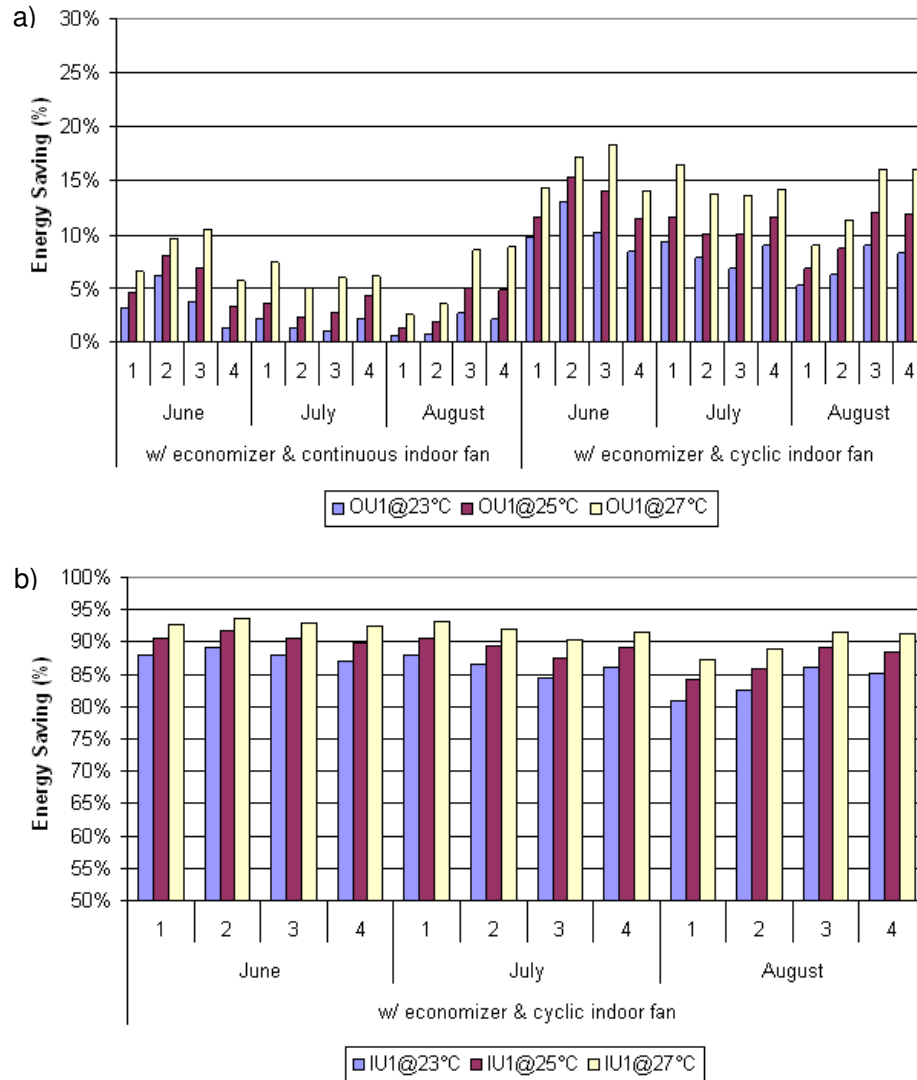


Figure 7.21 : Energy savings of outdoor and indoor units of System1 for MD, (a) outdoor unit, (b) indoor units

Figure 7.22 shows the weekly energy consumption of System2 with and without economizer and continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD.

As can be seen from Figure 7.22, similar to System1 results, OU2 energy consumption increases with the decreasing indoor set temperature. For the first week of August, OU2 consumes 167.7, 157.0 and 144.4kWh energy at 23°C, 25°C and 27°C indoor set temperatures, respectively, when there is no economizer and the fans of the indoor units are operated continuously. On the other hand, it consumes 166.4, 154.5 and 151.4kWh energy for the same period at 23°C, 25°C and 27°C indoor set temperatures, respectively, when the economizer is “ON” and the fans of the indoor units are operated continuously. When the fans of the indoor

units are operated cyclically with economizer, it consumes 163.6, 140.1 and 137.0kWh for the same period at 23°C, 25°C and 27°C indoor set temperatures, respectively.

Figure 7.23 shows the energy savings of the outdoor and indoor units of System2 with and without economizer and continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD.

As can be seen from Figure 7.23, similar to System1 results, it is observed that the economizer promises around 1-7%, 2-9% and 3-12% energy savings for the outdoor unit at 23°C, 25°C and 27°C indoor set temperatures, respectively. Since the total air flow rate of HRV3 and HRV4 which were located on System2 part was higher than that of HRV1 and HRV2 which were located on System1 part, shown in Table 4.4, the economizer effect on the energy saving of OU2 is slightly higher than the economizer effect on the energy saving of OU1.

It can be seen from Figure 7.23 that the total energy consumption of the fans of the indoor units decreases within 75-84%, 78-87% and 82-89% at 23°C, 25°C and 27°C indoor set temperatures, respectively, when the fans of the indoor units are operated cyclically instead of continuously.

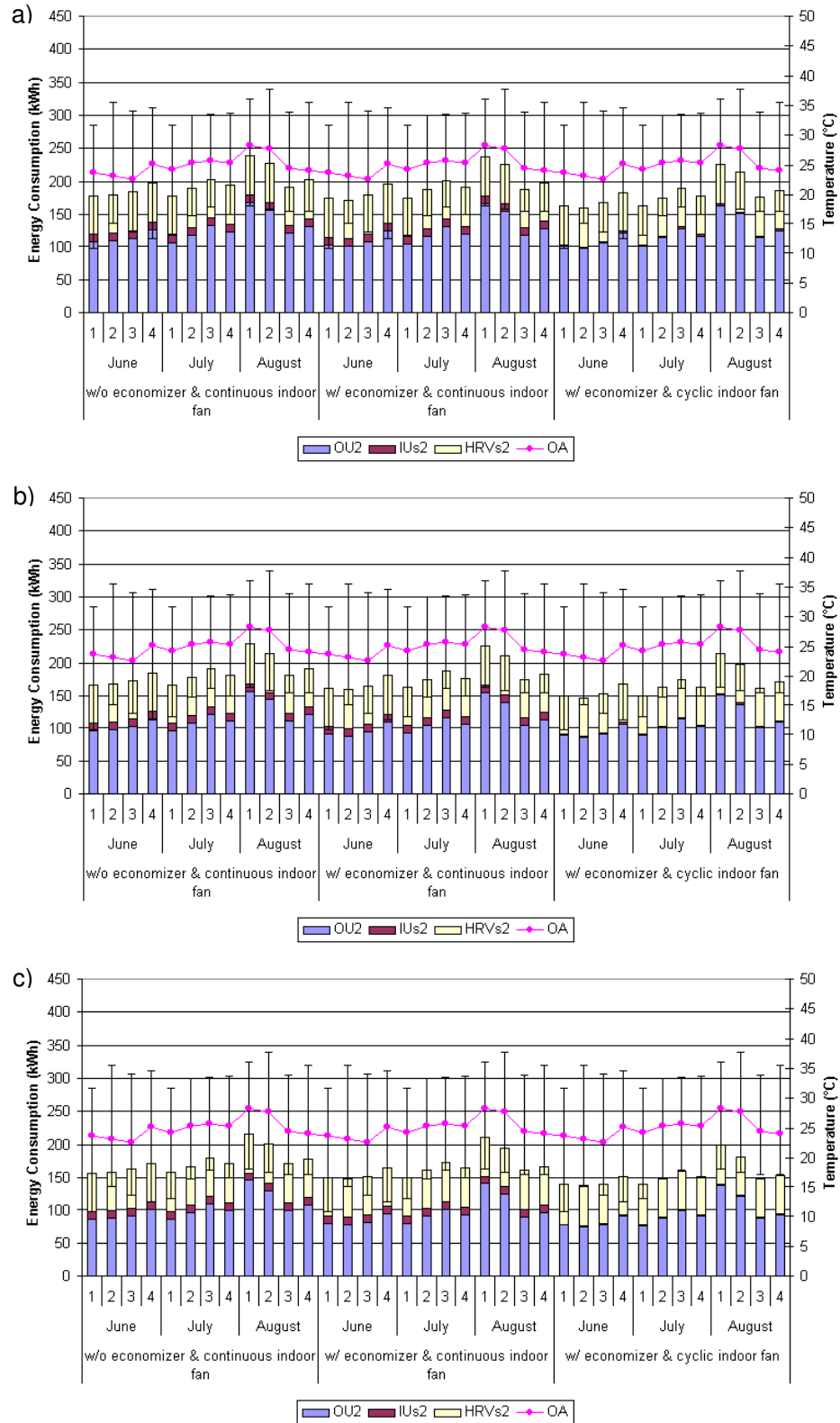


Figure 7.22 : Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C

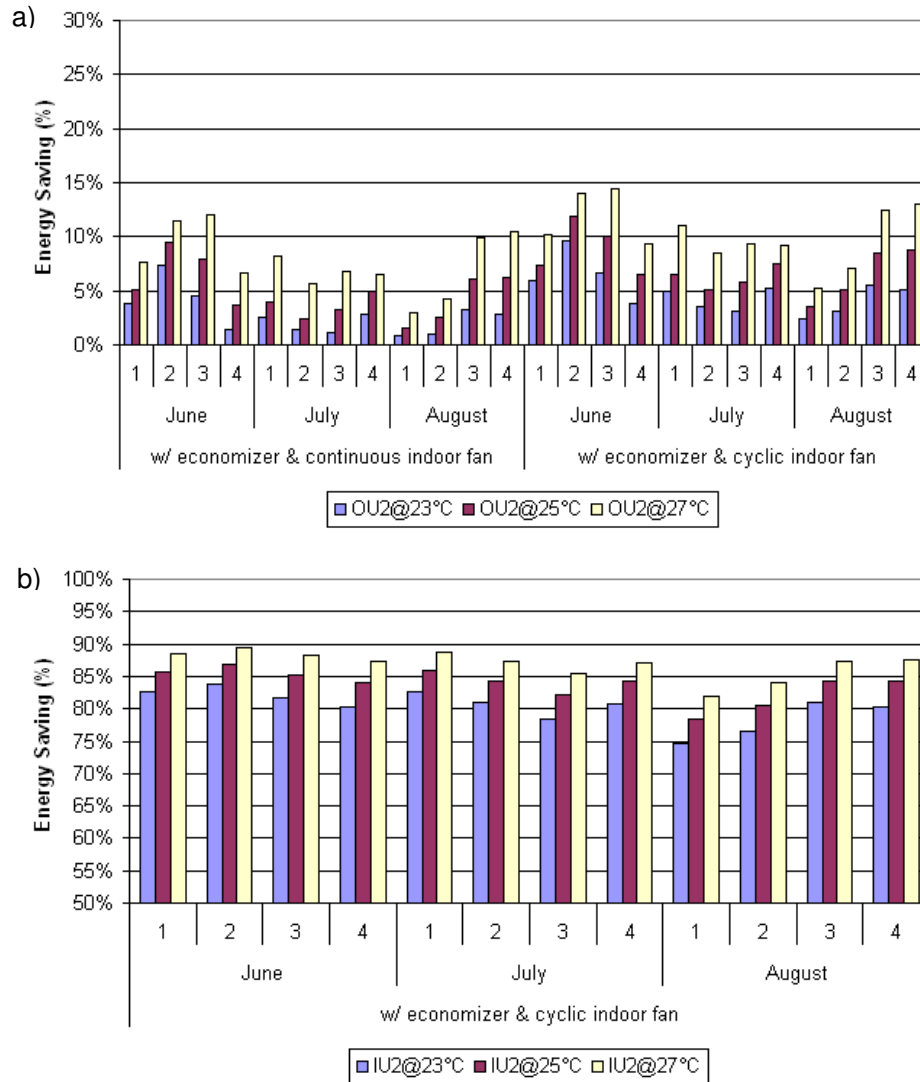


Figure 7.23 : Energy savings of outdoor and indoor units of System2 for MD, (a) outdoor unit, (b) indoor unit

Figure 7.24 shows the energy savings of System1 and System2 for MD.

It can be observed from Figure 7.24 that the cyclic indoor fan operation promises higher energy saving than the economizer.

It is found that the economizer promises around 0.5-3%, 1-4% and 2-5% energy saving for System1 and 1-4%, 1-5% and 2-7% energy saving for System2 at 23°C, 25°C and 27°C indoor set temperatures, respectively.

On the other hand, it can be seen from Figure 7.24 that the cyclic indoor fan operation with the economizer provides 14-24%, 17-26% and 19-29% energy saving for System1 and 5-11%, 6-13% and 8-14% energy saving for System2 at 23°C, 25°C and 27°C indoor set temperatures, respectively.

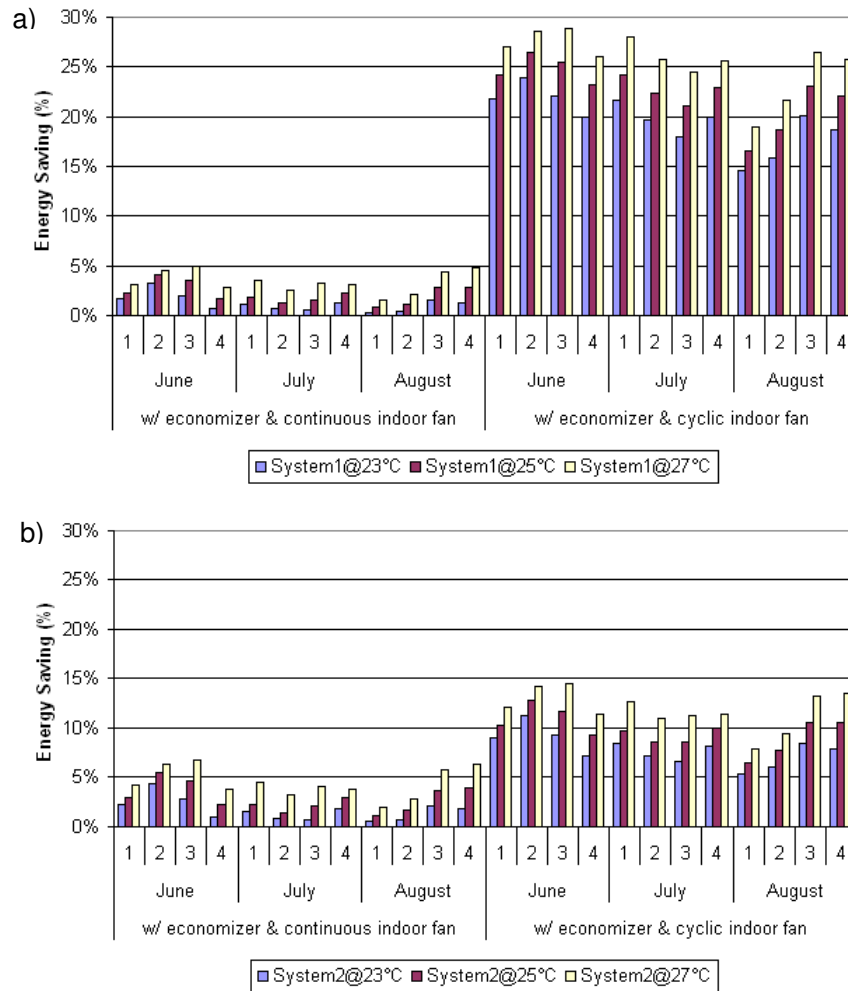


Figure 7.24 : Energy savings of System1 and System2 for MD, (a) System1, (b) System2

Figure 7.25 and Figure 7.26 show the indoor conditions of System1 zones and System2 zones at 23°C, 25°C and 27°C indoor set temperatures, respectively for MD. As can be seen, for both systems, the indoor set temperatures are maintained almost all of the operation time. The indoor temperatures higher than the set temperatures are mostly due to the elevator zone, which agrees with the experimental results. On the other hand, as described previously, the indoor temperatures lower than the indoor set temperatures are due to the ventilation effect. This is seen mostly at 27°C indoor set temperature. It can also be observed that the indoor humidity ratio decreases with the decreasing indoor set temperature, due to the increasing system operation time, which also agrees with the experimental results. Since the multi-split VRF system takes care of not only the sensible load, but also the latent load, the longer system operation reduces the indoor humidity ratio.

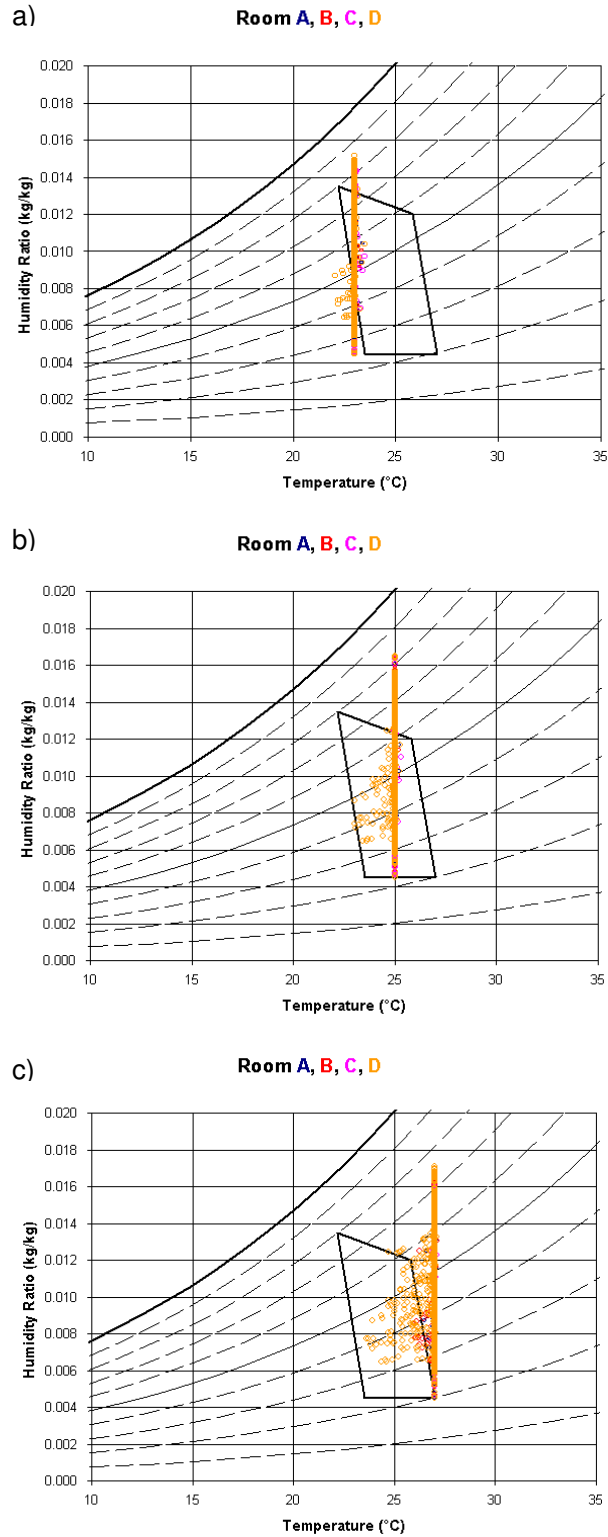


Figure 7.25 : Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23 °C, (b) 25 °C, (c) 27 °C

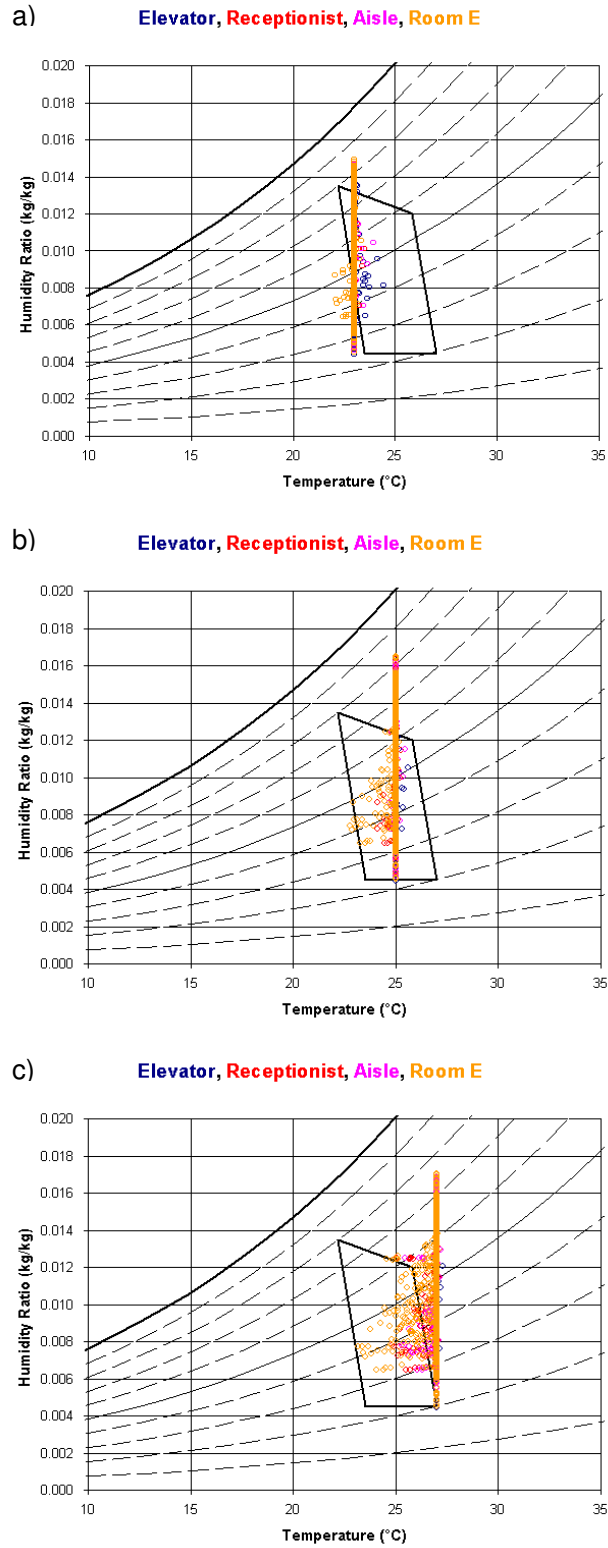


Figure 7.26 : Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23 °C, (b) 25 °C, (c) 27 °C

Figure 7.27 and Figure 7.28 show the weekly energy consumption of System1 for AZ and CA, respectively.

It can be observed from Figure 7.20, Figure 7.27 and Figure 7.28 that since AZ's outdoor temperature is the highest and CA's outdoor temperature is the lowest among the three states for the same period of June-August 2007, the outdoor unit of the multi-split VRF system consumes the highest energy in AZ and the lowest energy in CA.

On the other hand, due to the high outdoor temperature, the economizer does not promise energy savings for AZ. On the contrary, it has a high potential on the energy savings for CA due to the low outdoor temperature.

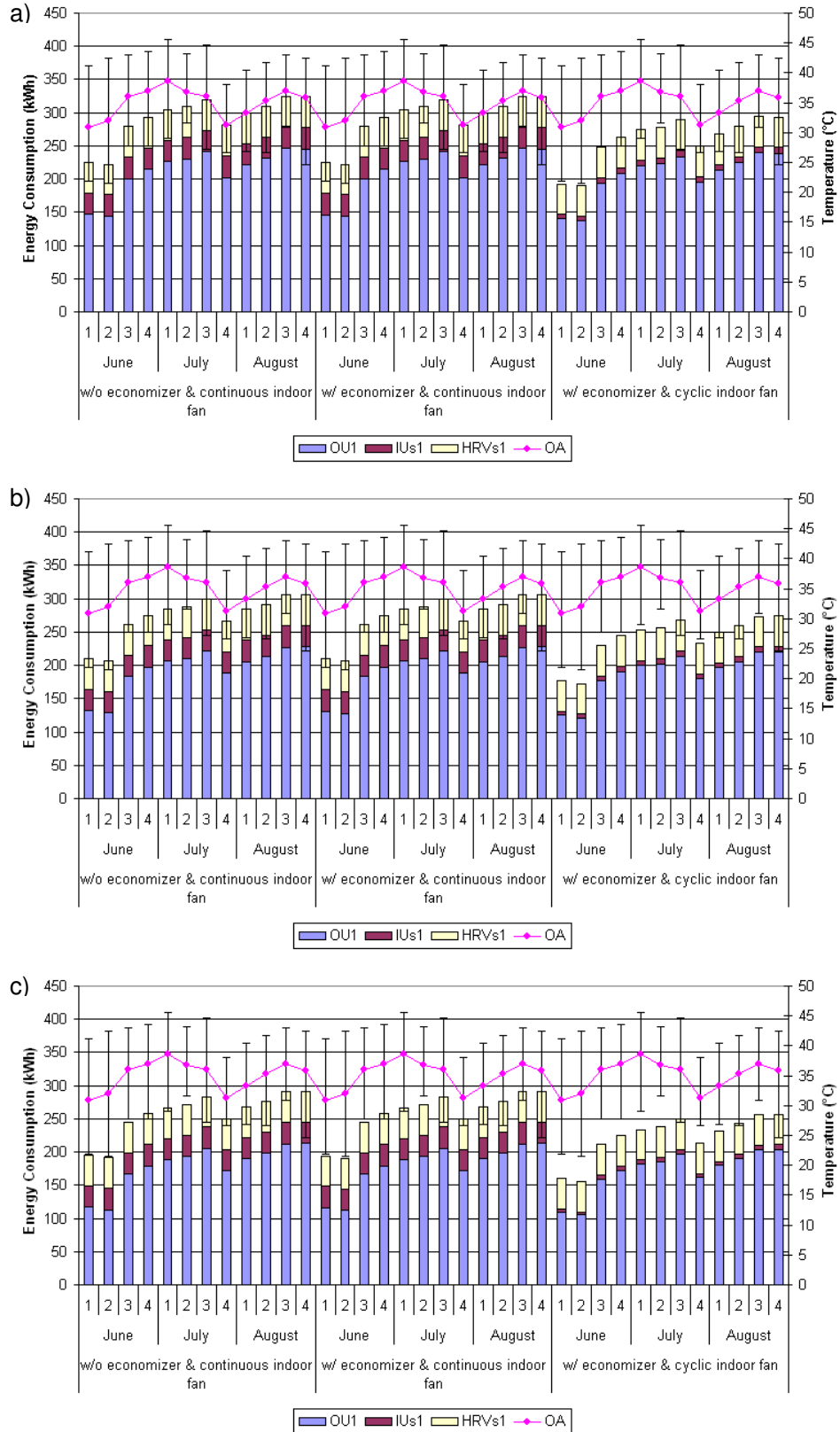


Figure 7.27 : Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

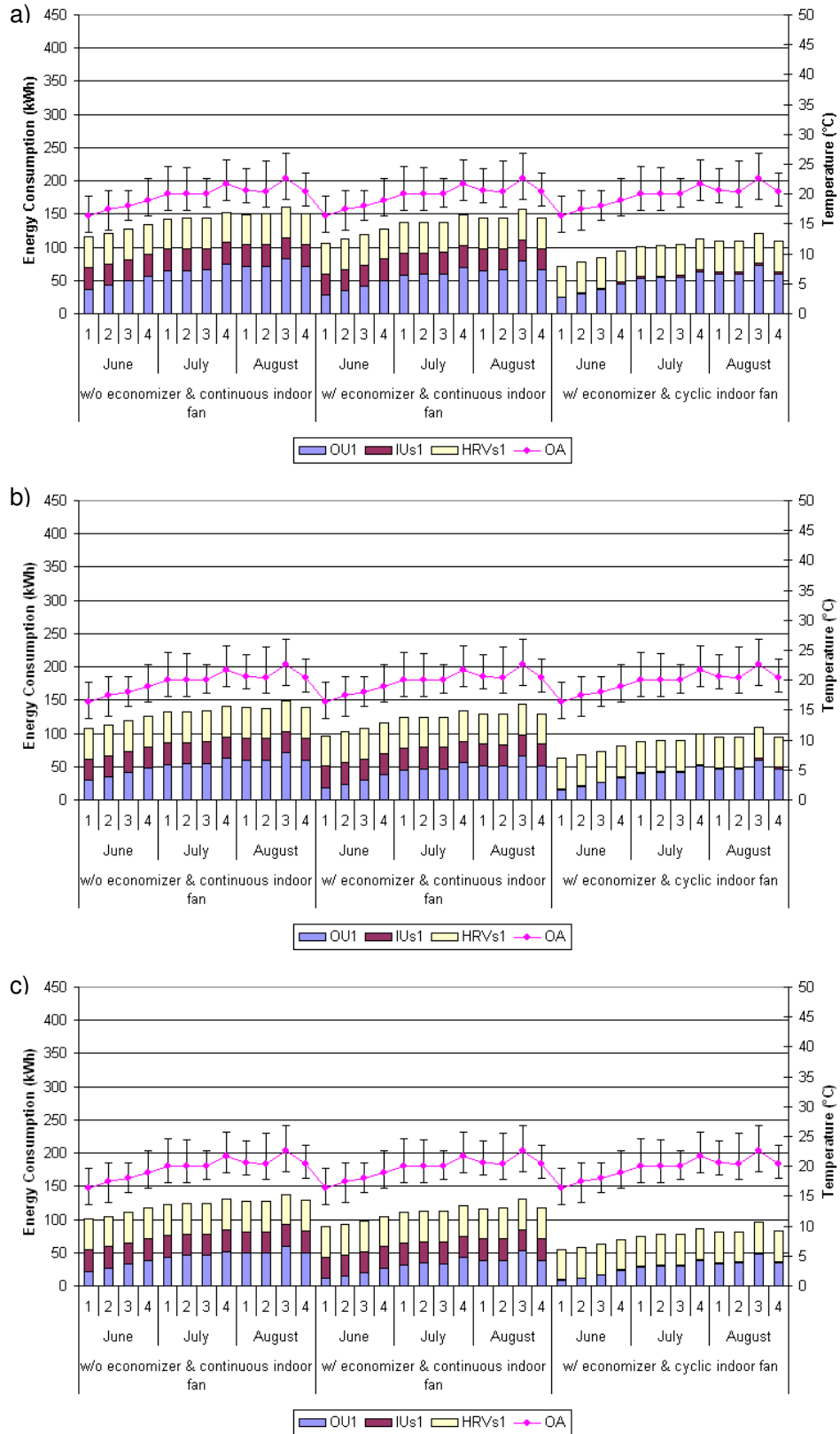


Figure 7.28 : Weekly energy consumptions of System1 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.29 and Figure 7.30 show the energy savings of outdoor and indoor units of System1 for AZ and CA, respectively.

As can be seen from Figure 7.29b, the economizer promises only around 1% energy savings potential for OU1 in AZ. Because AZ's outdoor temperature is mostly higher than the high temperature limit of the economizer which limits the availability of the free-cooling options. That's why; the economizer is not an applicable energy saving option for AZ. On the contrary, since the outdoor temperature is mostly lower than the high temperature limit of the economizer in CA, it promises around 4-25%, 7-37% and 12-51% energy savings at 23°C, 25°C and 27°C indoor set temperatures for the outdoor unit, respectively.

Besides, since System1 operates less in CA compared to AZ, the energy saving potential for CA due to the cyclic indoor fan operation is higher than AZ. The energy saving varies within 89-96%, 92-98% and 94-99% at 23°C, 25°C and 27°C indoor set temperatures for CA and 71-81%, 75-84% and 79-86% at 23°C, 25°C and 27°C indoor set temperatures for AZ.

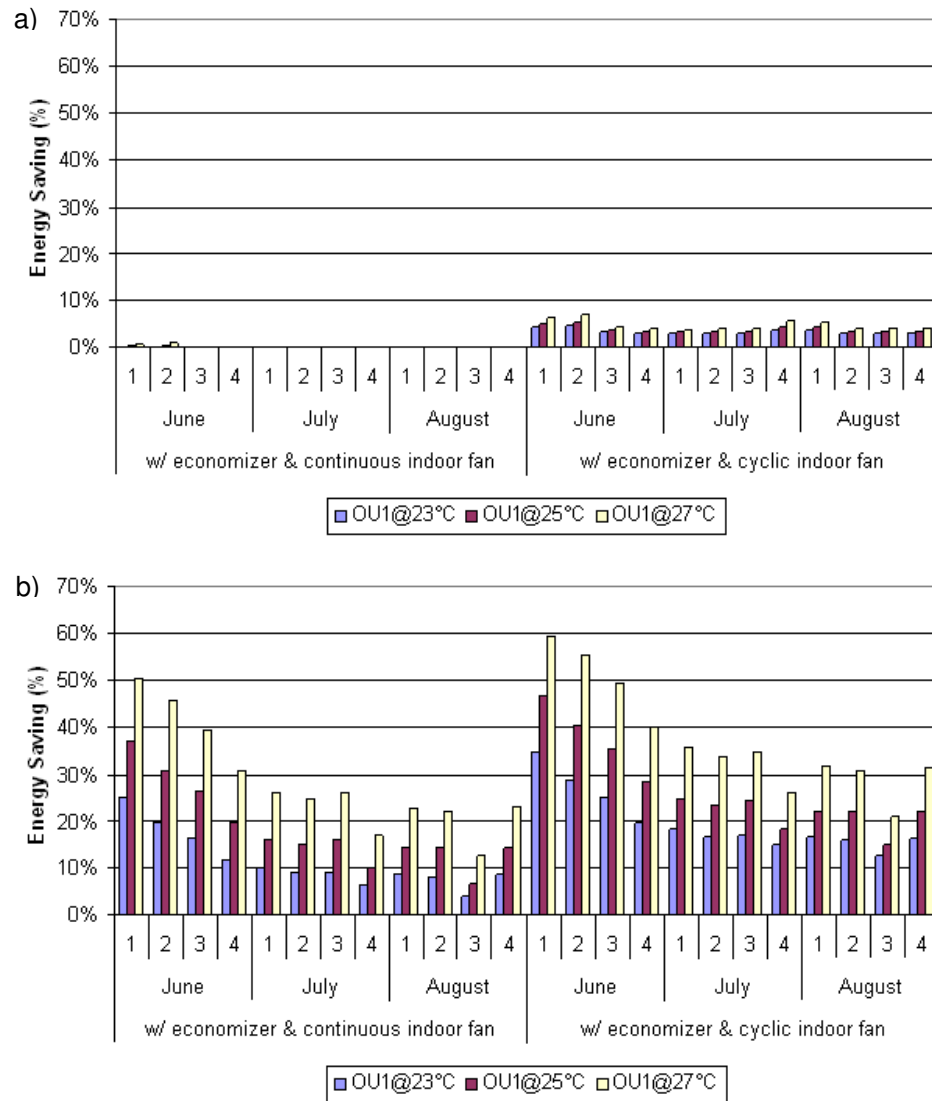


Figure 7.29 : Energy savings of outdoor units of System1 for AZ and CA, (a) AZ, (b) CA

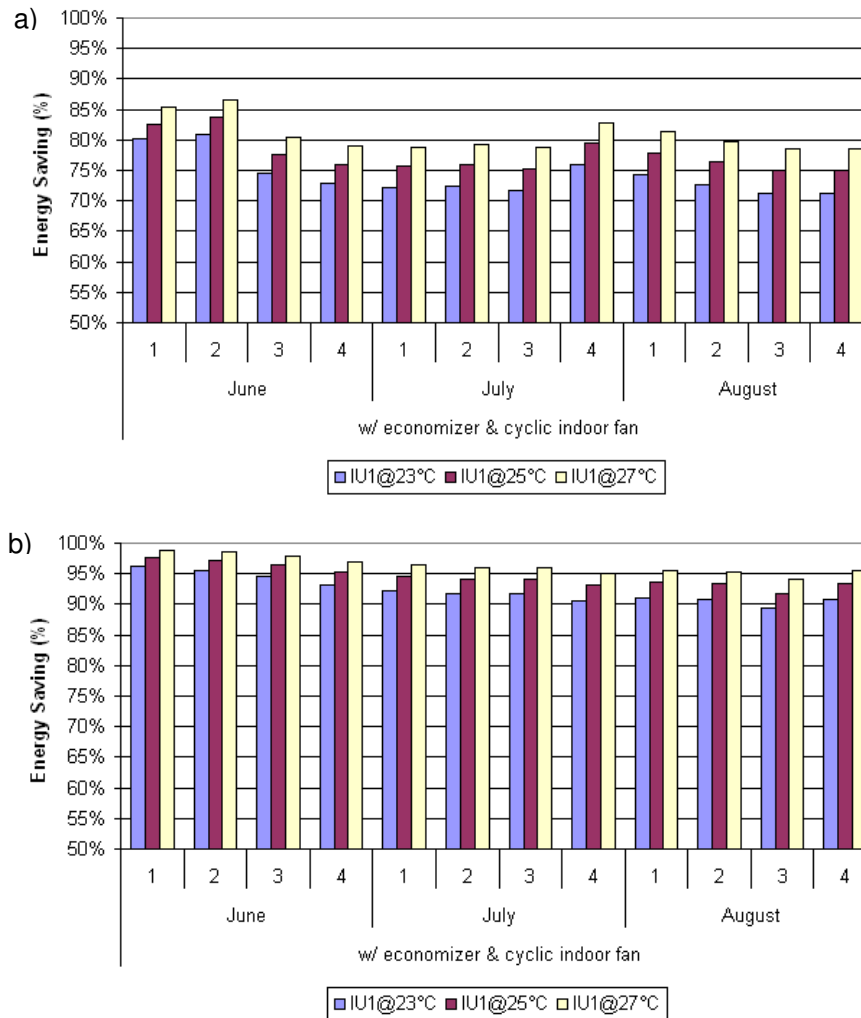


Figure 7.30 : Energy savings of indoor units of System1 for AZ and CA, (a) AZ, (b) CA

Figure 7.31 and Figure 7.32 show the indoor conditions of System1 zones at 23°C, 25°C and 27°C indoor set temperatures for AZ and CA, respectively. The indoor conditions of System1 zones at 23°C, 25°C and 27°C indoor set temperatures for MD is provided in Figure 7.25. As can be seen from Figure 7.31, indoor set temperatures are maintained almost all of the operation time, however, as seen, since the outdoor temperature of AZ is high, sometimes the capacities of the indoor units are not sufficient to maintain the indoor set temperatures. On the other hand, since the cooling load for CA is low, there is not any indoor temperature higher than the set temperature, as seen Figure 7.32.

As shown in Figure 7.19, AZ is relatively dry compared to CA, which causes the lower indoor humidity ratios, shown in Figure 7.31 compared to the indoor humidity ratios shown in Figure 7.32.

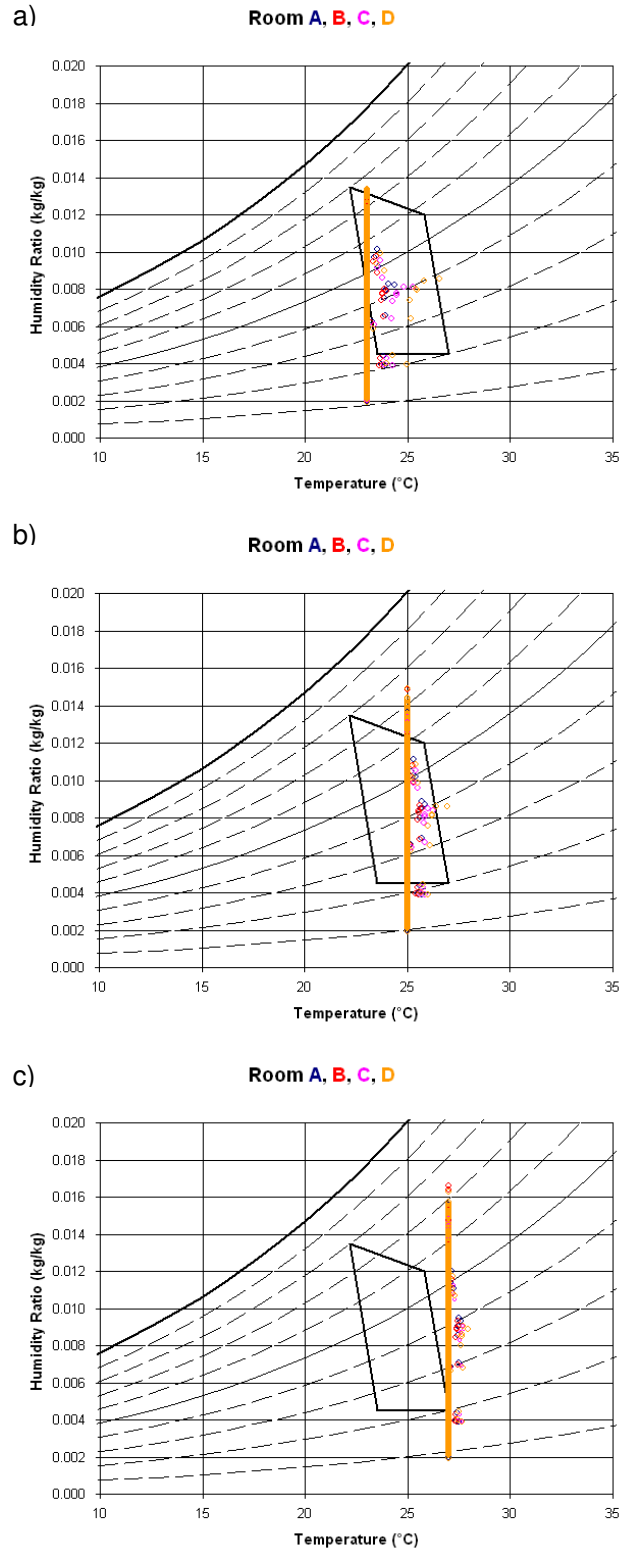


Figure 7.31 : Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

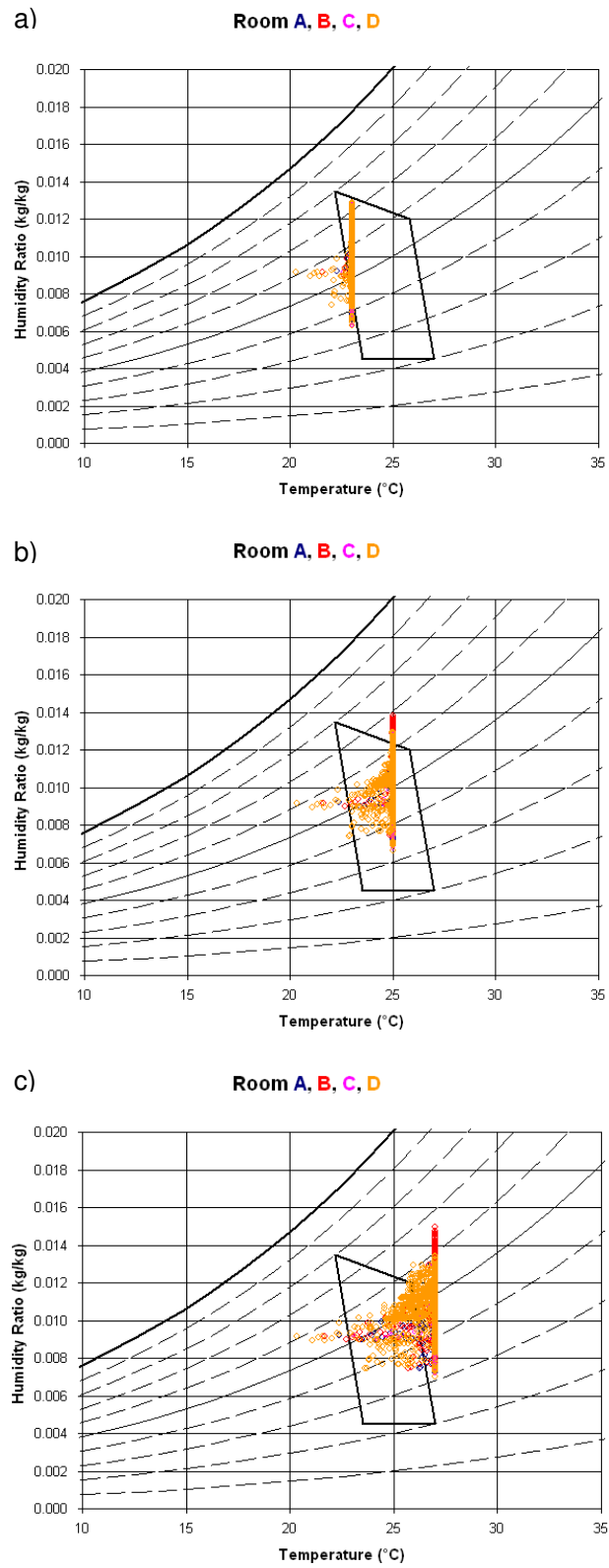


Figure 7.32 : Indoor conditions of System1 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.33 and Figure 7.34 show the weekly energy consumption of System2 for AZ and CA, respectively.

It can be observed from Figure 7.22, Figure 7.33 and Figure 7.34, similar to System1 results, the outdoor unit of the multi-split VRF system consumes the highest energy in AZ and the lowest energy in CA, due to the different outdoor temperature profiles.

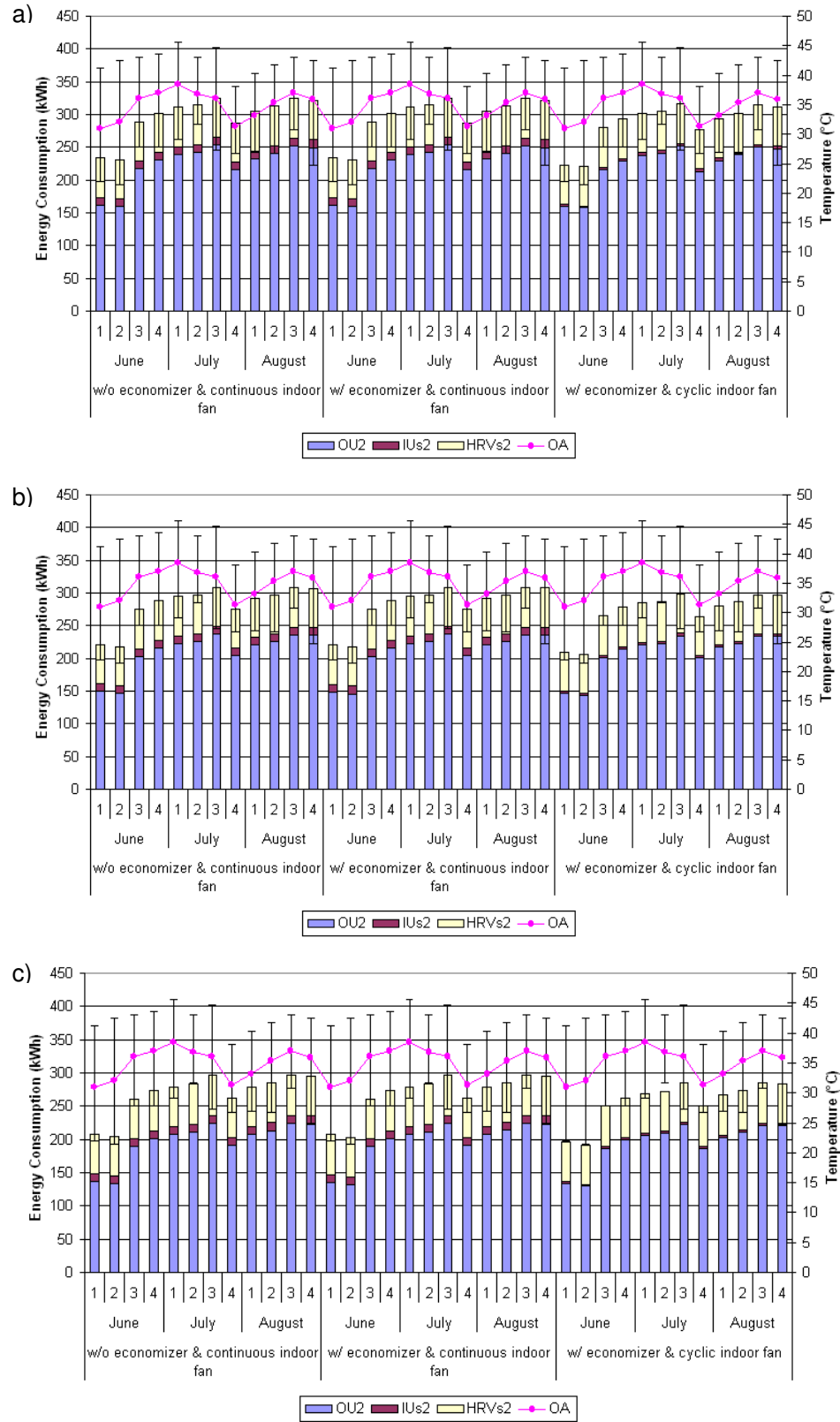


Figure 7.33 : Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

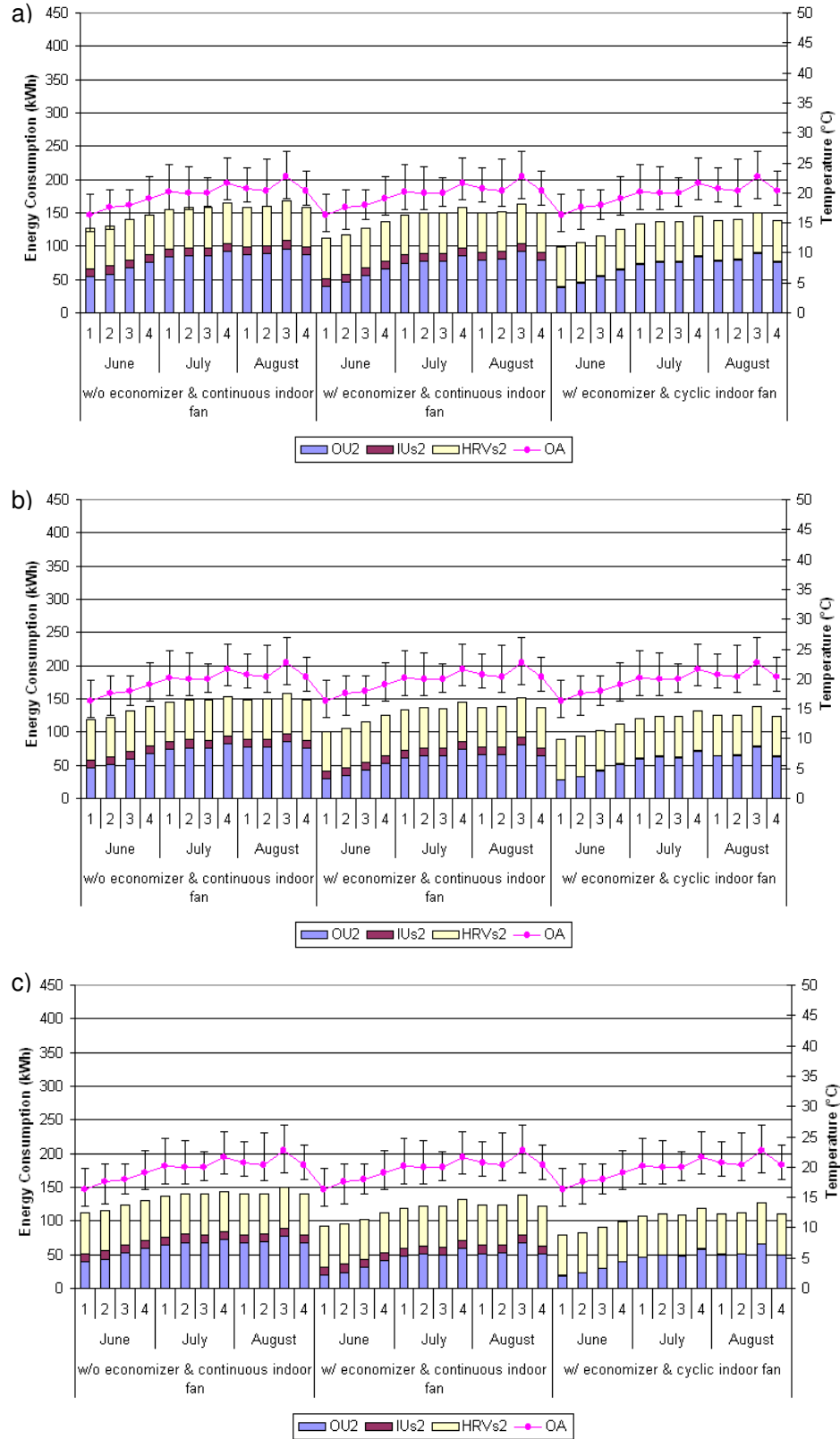


Figure 7.34 : Weekly energy consumptions of System2 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.35 and Figure 7.36 show the energy savings of outdoor and indoor units of System2 for AZ and CA, respectively.

As can be seen from Figure 7.35a, similar to System1 results, the economizer promises only less than 1% energy savings potential for OU2 in AZ. On the other hand, for CA, it promises around 5-27%, 7-37% and 13-50% energy savings at 23°C, 25°C and 27°C indoor set temperatures for OU2, respectively.

Besides, similar to System1 results, since System2 operates less in CA compared to AZ, the energy saving potential for CA due to the cyclic indoor fan operation is higher than AZ. The energy saving varies within 84-93%, 82-95% and 90-97% at 23°C, 25°C and 27°C indoor set temperatures for CA and 63-74%, 67-77% and 71-80% at 23°C, 25°C and 27°C indoor set temperatures for AZ, respectively.

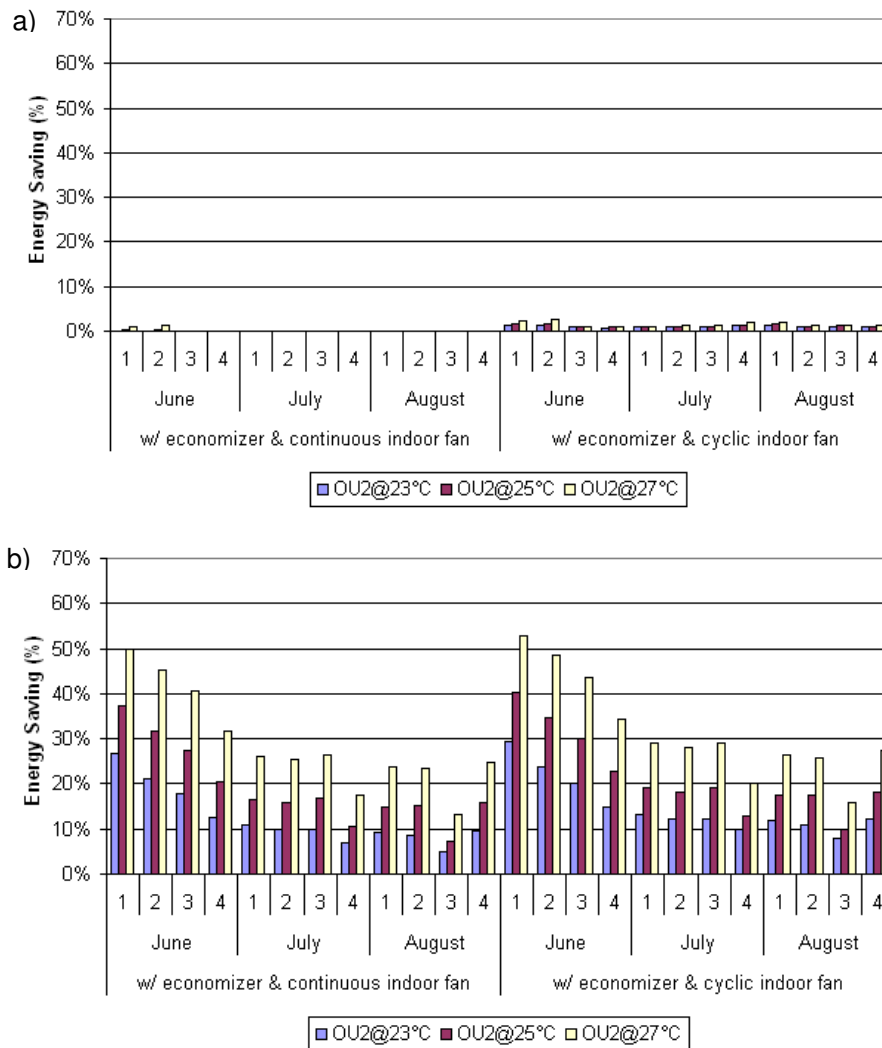


Figure 7.35 : Energy savings of outdoor units of System2 for AZ and CA, (a) AZ, (b) CA

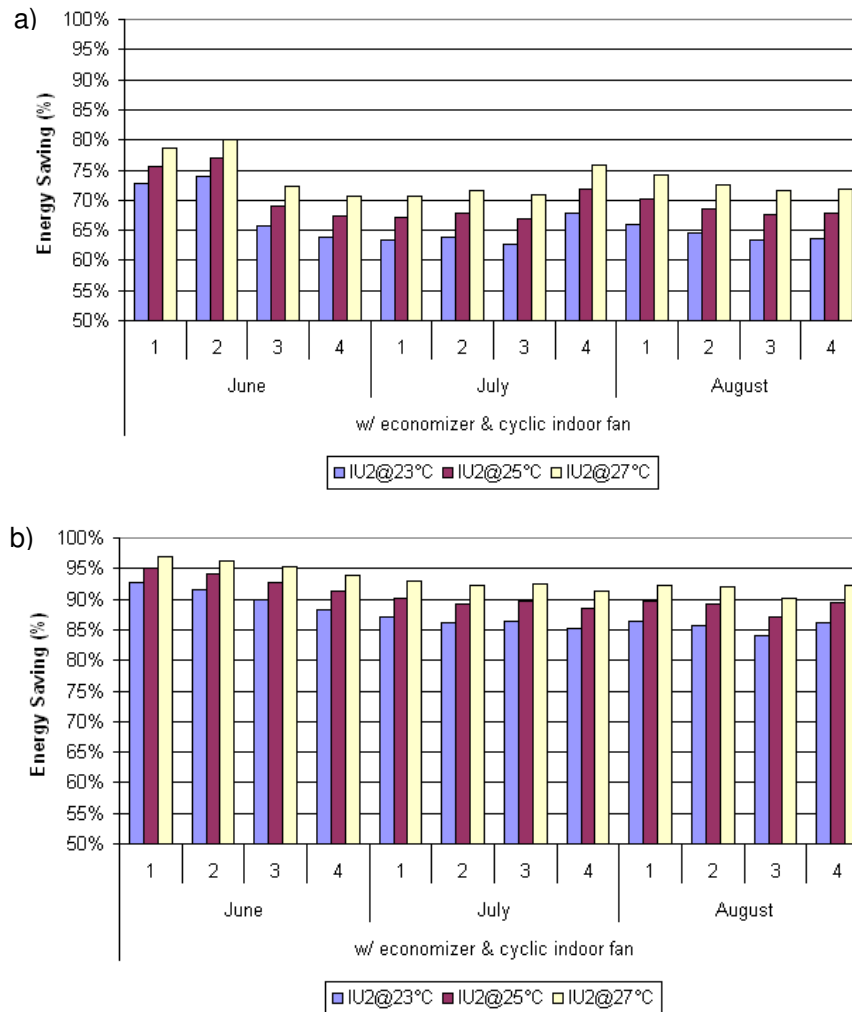


Figure 7.36 : Energy savings of indoor units of System2 for AZ and CA, (a) AZ, (b) CA

Figure 7.37 and Figure 7.38 show the indoor conditions of System2 zones at 23°C, 25°C and 27°C indoor set temperatures for AZ and CA, respectively. The indoor conditions of System2 zones at 23°C, 25°C and 27°C indoor set temperatures for MD is provided in Figure 7.26. As can be seen from Figure 7.37, especially the elevator zone temperature can not be maintained at the set temperatures due to the high outdoor temperature and relatively small capacity indoor unit, which agrees with low indoor set temperature experimental results. Increasing the capacity of the indoor unit helps maintaining the indoor temperature at the desired set temperature. On the other hand, since the cooling load for CA is low, the capacities of the indoor units are sufficient to maintain the indoor temperatures at the desired set temperatures. That's why; there is not any indoor temperature higher than the set temperature, as seen Figure 7.38. Besides, similar to System1 results, the indoor humidity ratios are relatively lower in AZ than in CA.

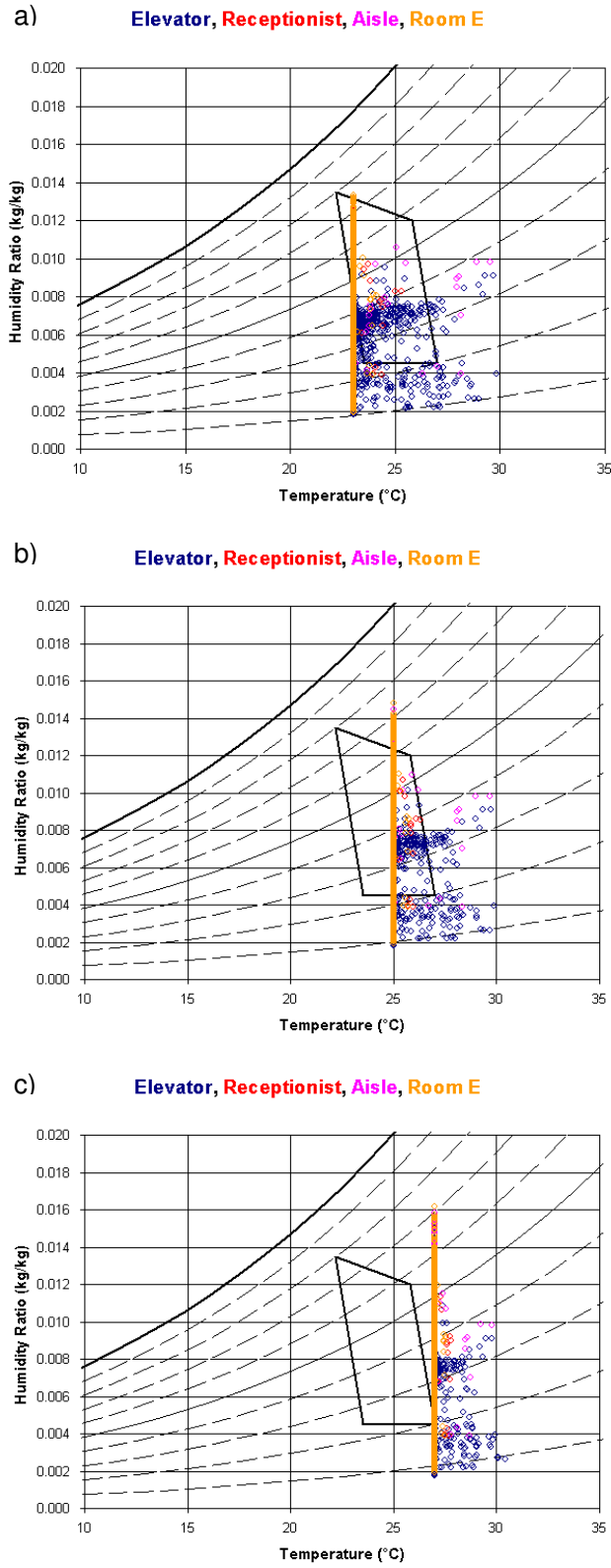


Figure 7.37 : Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

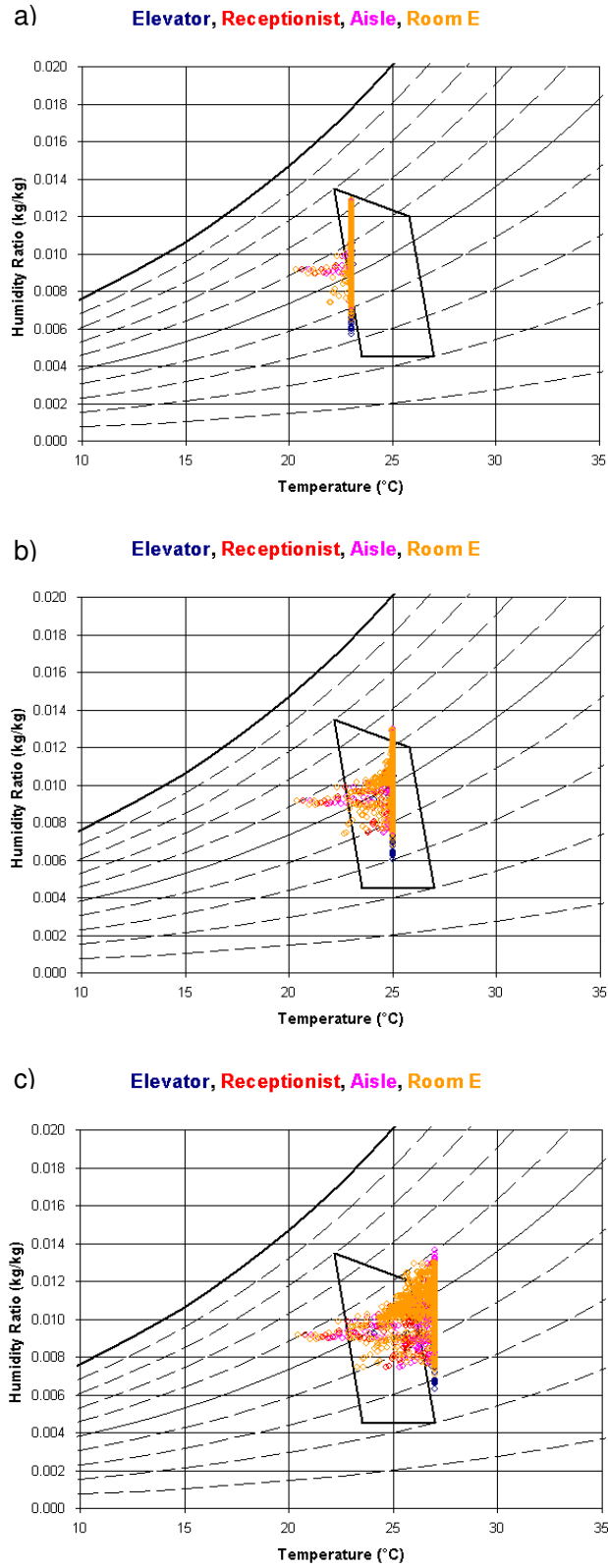


Figure 7.38 : Indoor conditions of System2 zones w/ economizer and cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.39 shows the seasonal energy consumption of System1 and System2 for MD, AZ and CA.

Since the cyclic indoor fan operation with the economizer is found to be the most promising energy saving options for the whole system, the comparison is done based on the cyclic indoor fan operation with the economizer.

It can be observed from Figure 7.39 that, as expected, both systems consume the highest energy in AZ due to the high outdoor temperature throughout the season, and the lowest energy in CA due to the low outdoor temperature throughout the season. Due to the high power consumption of System1 indoor units', the cyclic indoor unit fan operation has higher energy saving potential for System1.

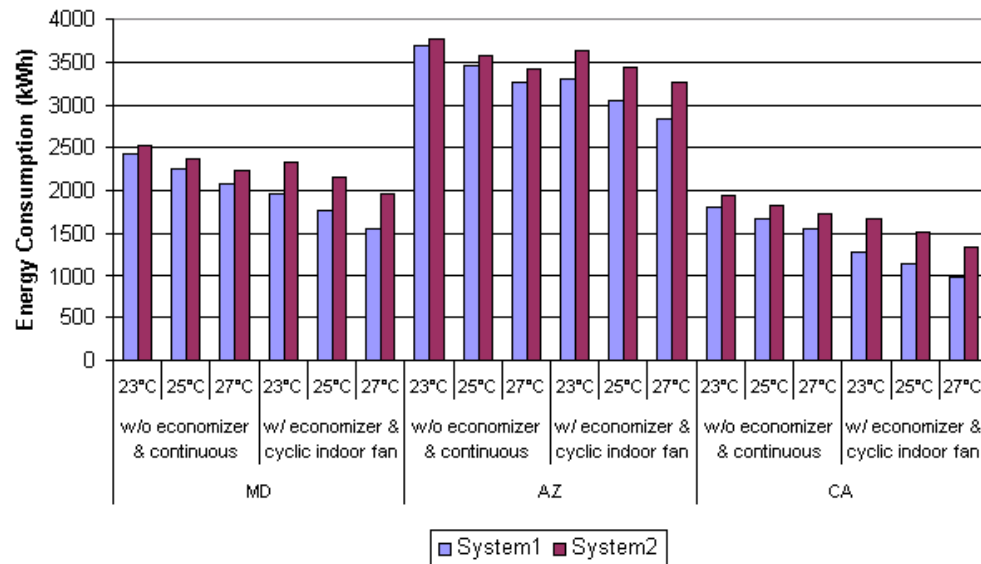


Figure 7.39 : Seasonal energy consumption of System1 and System2 for MD, AZ and CA

Figure 7.40 shows the seasonal energy savings of System1 and System2 for MD, AZ and CA.

As can be seen from Figure 7.40, the economizer and the cyclic indoor fan operation promise higher energy saving in CA and lower in AZ due to the low and high outdoor temperatures, respectively. As explained previously, the outdoor temperature affects not only the outdoor unit energy consumption and accordingly the indoor unit energy consumption due to the cooling load, but also the availability of the free-cooling.

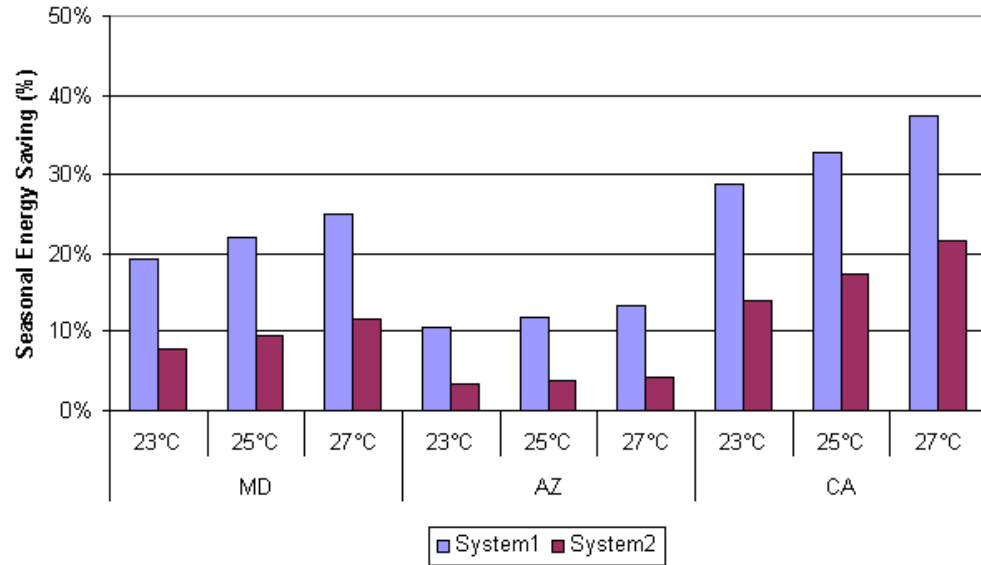


Figure 7.40 : Seasonal energy savings of System1 and System2 for MD, AZ and CA

Figure 7.41 and Figure 7.42 show CPF and PLR of System1 and System2 for MD, AZ and CA at 25°C indoor set temperatures, respectively.

It is observed from Figure 7.41 that the cyclic indoor fan operation with economizer can improve the efficiency of the multi-split VRF system remarkable depending on the outdoor conditions. Considering the first week of July for System1, if cyclic indoor fan operation with economizer is used instead of continuous indoor fan operation without economizer, CPF increases from 3.17, 2.87 and 2.93 to 4.22, 3.19, and 4.57 for MD, AZ and CA, respectively. Considering System2 for the same period, CPF increases from 3.99, 3.15 and 4.36 to 4.22, 3.26, and 4.71 for MD, AZ and CA, respectively. As explained previously, the indoor unit energy saving potential is the highest in CA among the three states. That's why; the increment in CPF in CA is the highest.

It can be observed from Figure 7.42 that PLR increases with the increasing outdoor temperature due to the increasing cooling load, and has the highest in AZ. As can be seen, the cyclic indoor unit fan operation with economizer does not have a big influence on PLR.

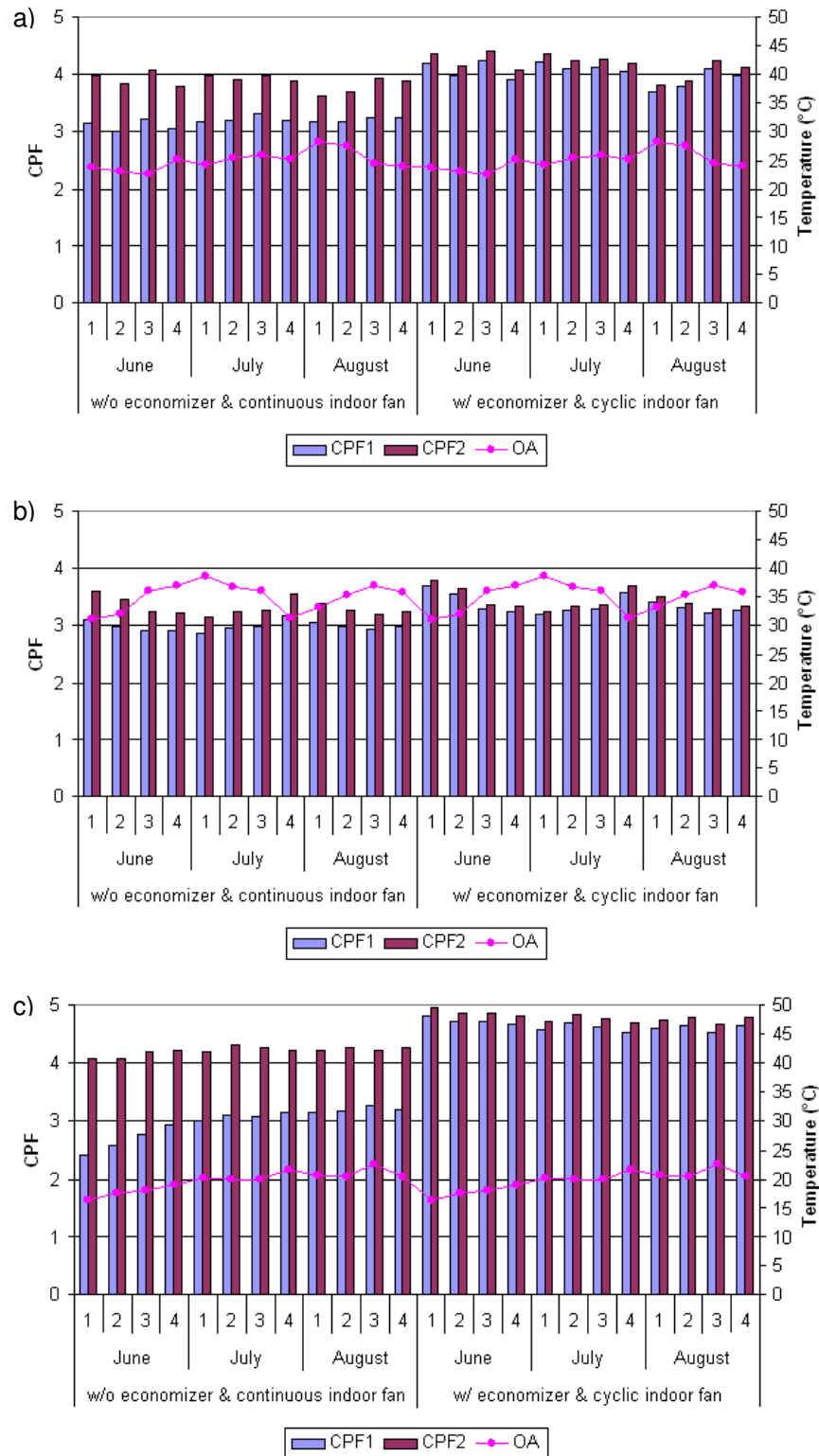


Figure 7.41 : CPF of System1 and System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA

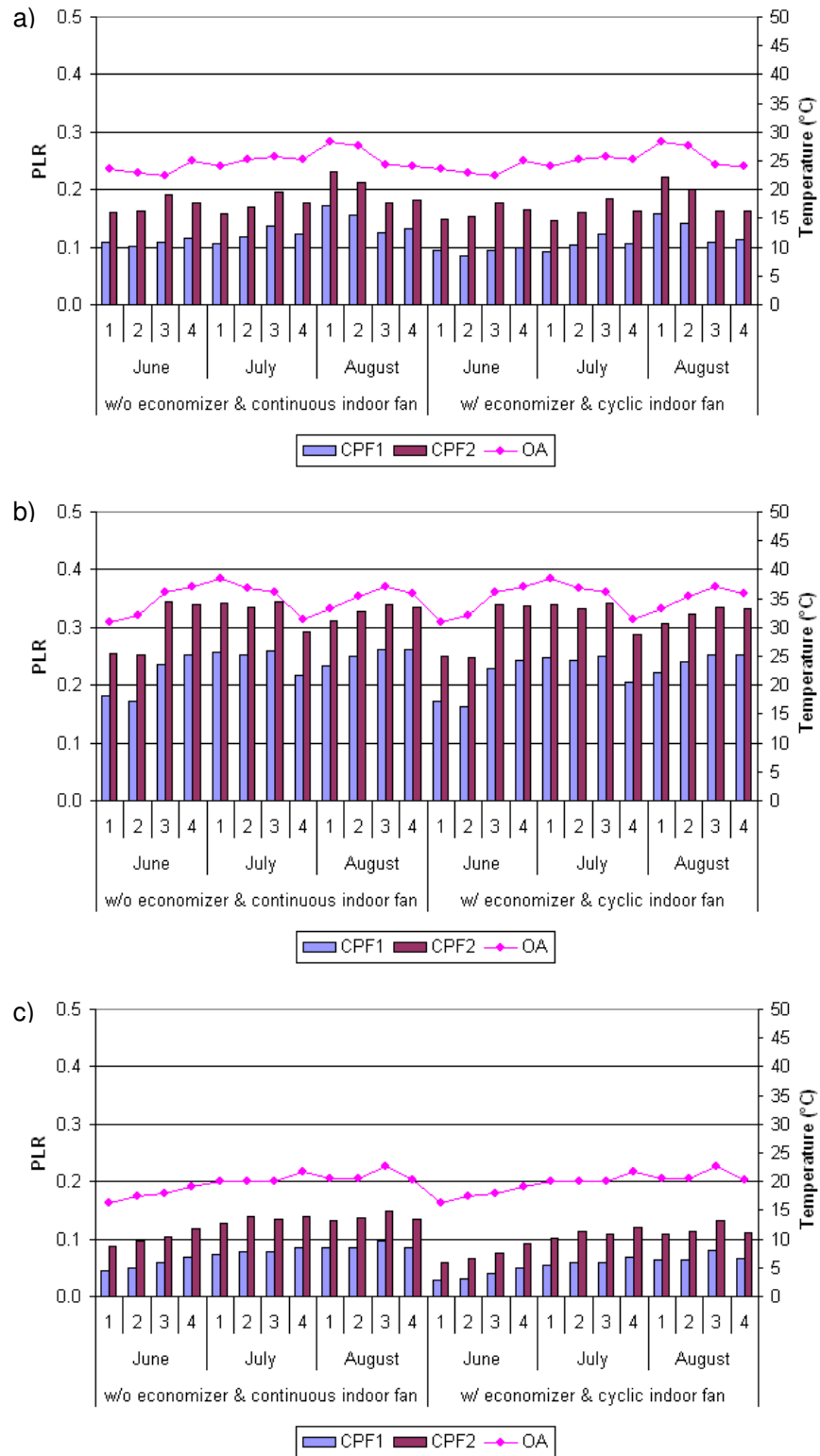


Figure 7.42 : PLR of System1 and System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA

7.7.2 Non-ventilated system

Figure 7.43 shows the weekly energy consumption of non-ventilated System1 for continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD.

It is observed from Figure 7.43 that for the first week of August, OU1 consumes 100.1, 93.2 and 89.3kWh energy for continuous indoor fan operation and consumes 93.8, 86.6 and 82.6kWh energy for cyclic indoor fan operation at 23°C, 25°C and 27°C indoor set temperatures, respectively. Since the heat load coming from the indoor unit fan decreases with the cyclic indoor fan operation, the outdoor unit energy consumption decreases accordingly.

It is also found from Figure 7.43 that the total energy consumption of indoor fans is around 32.2kWh for the continuous indoor fan operation, and it reduces around 86%, 88% and 90% for the cyclic indoor fan operation at 23°C, 25°C and 27°C indoor set temperatures, respectively for the first week of August.

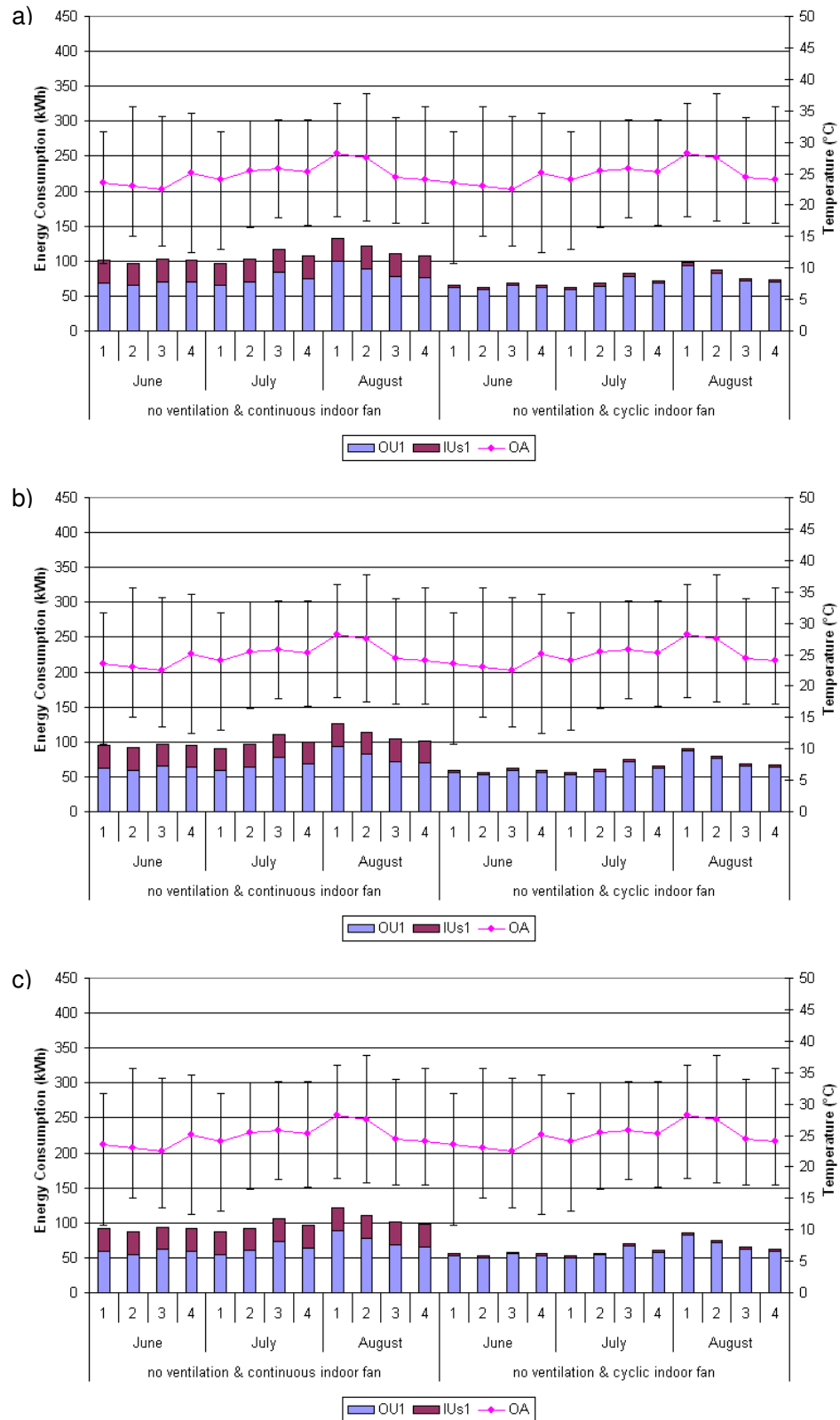


Figure 7.43 : Weekly energy consumptions of non-ventilated System 1 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.44 shows the weekly energy consumption of non-ventilated System2 for continuous and cyclic indoor fan operation options at 23°C, 25°C and 27°C indoor set temperatures for MD.

Similar to non-ventilated System1 results, it can be observed from Figure 7.44 that for the first week of August, OU2 consumes 111.8, 105.4 and 101.9kWh energy for continuous indoor fan operation and consumes 109.5, 102.9 and 99.1kWh energy for cyclic indoor fan operation at 23°C, 25°C and 27°C indoor set temperatures, respectively.

It is also found from Figure 7.44 that the total energy consumption of indoor fans is around 11.7kWh for the continuous indoor fan operation, and it reduces around 81%, 83% and 85% for the cyclic indoor fan operation at 23°C, 25°C and 27°C indoor set temperatures, respectively for the first week of August.

Since the individual and total power consumptions of System1 indoor units are higher than that of System2 indoor units with higher air flow rates, the heat gain coming from System1 indoor units are greater than System2 indoor units. That's why; the effect of the cyclic indoor fan operation on OU1 energy consumption, shown in Figure 7.43 is greater than that of the cyclic indoor fan operation on OU2 energy consumption.

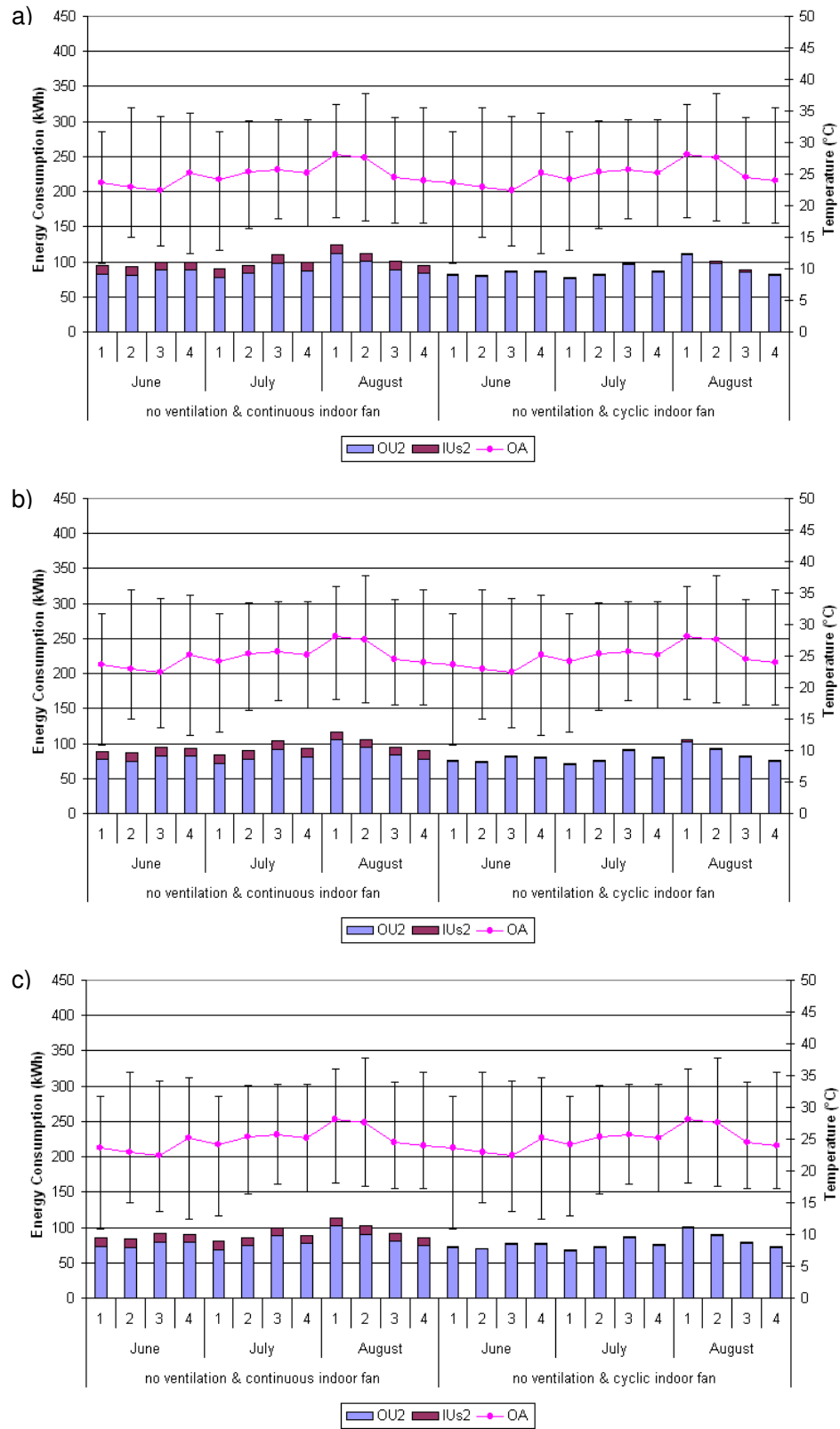


Figure 7.44 : Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.45 and Figure 7.46 show the energy savings of OU1 and OU2 and the energy savings of non-ventilated System1 and non-ventilated System2 due to the cyclic indoor fan operation at 23°C, 25°C and 27°C indoor set temperatures for MD, respectively.

It can be observed from Figure 7.45 and Figure 7.46 that the effect of the cyclic indoor fan operation on non-ventilated System1 is greater than that of the cyclic indoor fan operation on non-ventilated System2, which was described previously.

It is found from Figure 7.46 that the cyclic indoor fan operation promises 26-36%, 28-39% and 29-40% energy saving for non-ventilated System1 and 10-14%, 10-15% and 11-15% energy saving for non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures, respectively.

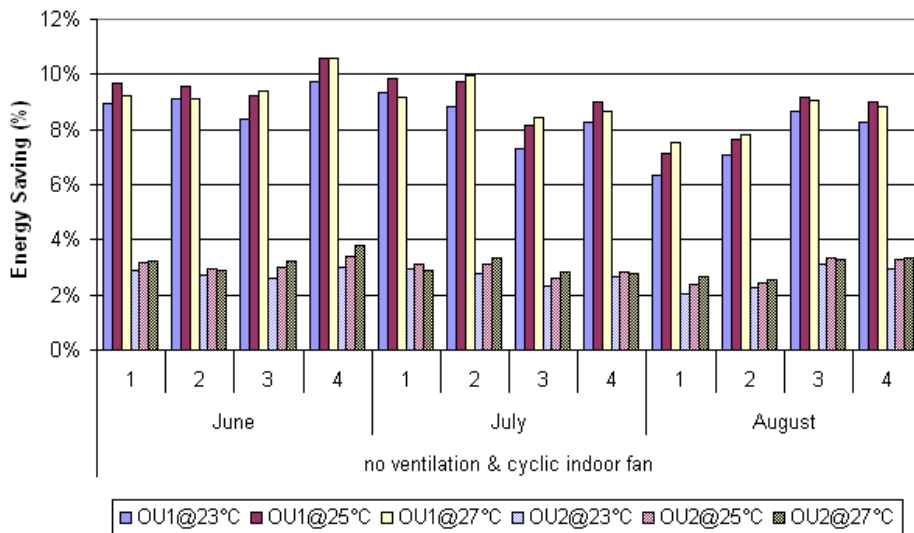


Figure 7.45 : Energy savings of OU1 and OU2 for non-ventilated System1 and non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD

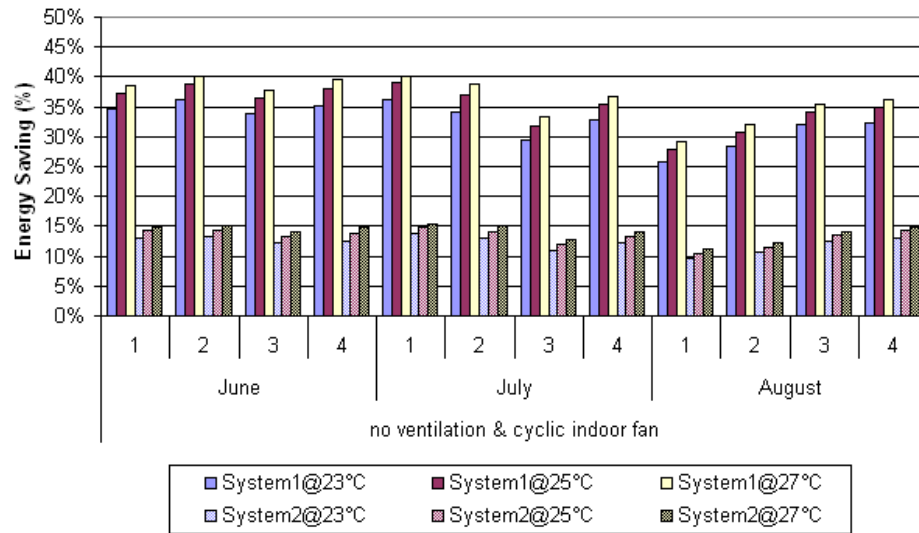


Figure 7.46 : Energy savings of non-ventilated System1 and non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for MD

Figure 7.47 and Figure 7.48 show the indoor conditions of the non-ventilated System1 and non-ventilated System2 zones at 23°C, 25°C and 27°C indoor set temperatures, respectively for MD.

As can be seen from Figure 7.47 and Figure 7.48, similar to System1 and System2 indoor condition results, the indoor temperatures can be maintained almost all of the operation time by the non-ventilated multi-split VRF systems.

Since there is no ventilation load introduced to the indoors during the operation, the occupancy density becomes the main source for the indoor humidity. As can be seen from Figure 7.47c, the indoor humidity ratio of Room B is the maximum among the other zones, because four people were added to Room B, as described in Chapter 7.7.

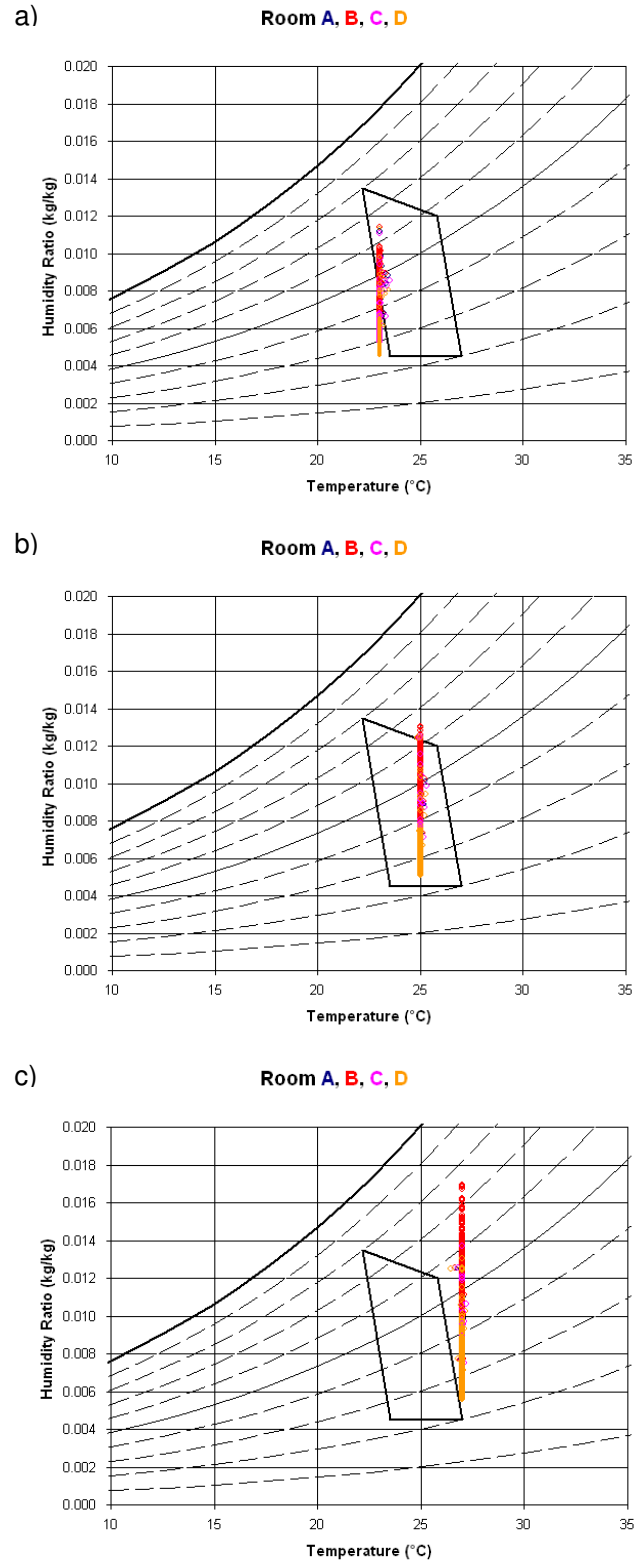


Figure 7.47 : Indoor conditions of non-ventilated System1 zones with cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23°C, (b) 25°C, (c) 27°C

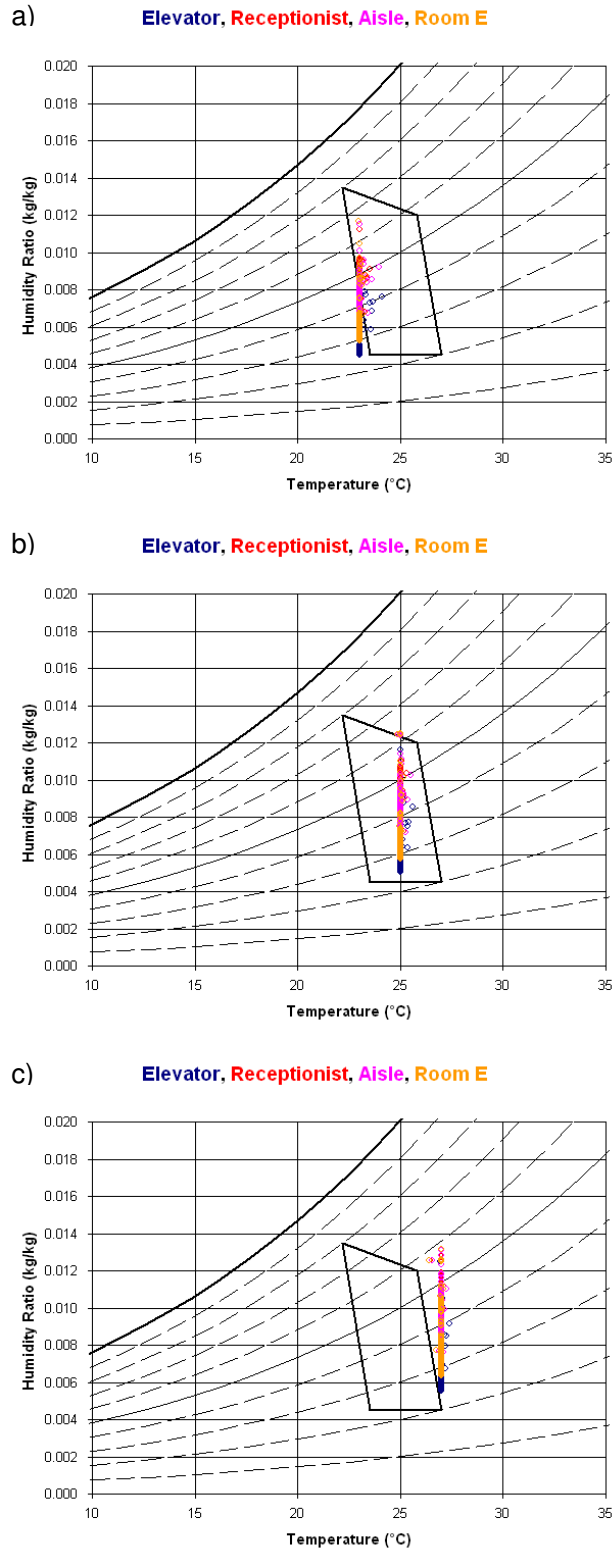


Figure 7.48 : Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for MD, (a) 23 °C, (b) 25 °C, (c) 27 °C

Figure 7.49 and Figure 7.50 show the weekly energy consumption of non-ventilated System1 for AZ and CA, respectively.

Similar to the ventilation assisted System1 results, it can be observed from Figure 7.43, Figure 7.49 and Figure 7.50 that since AZ's outdoor temperature is the highest and CA's outdoor temperature is the lowest among the three states for the same period of June-August 2007, the outdoor unit of the multi-split VRF system consumes the highest energy in AZ and the lowest energy in CA.

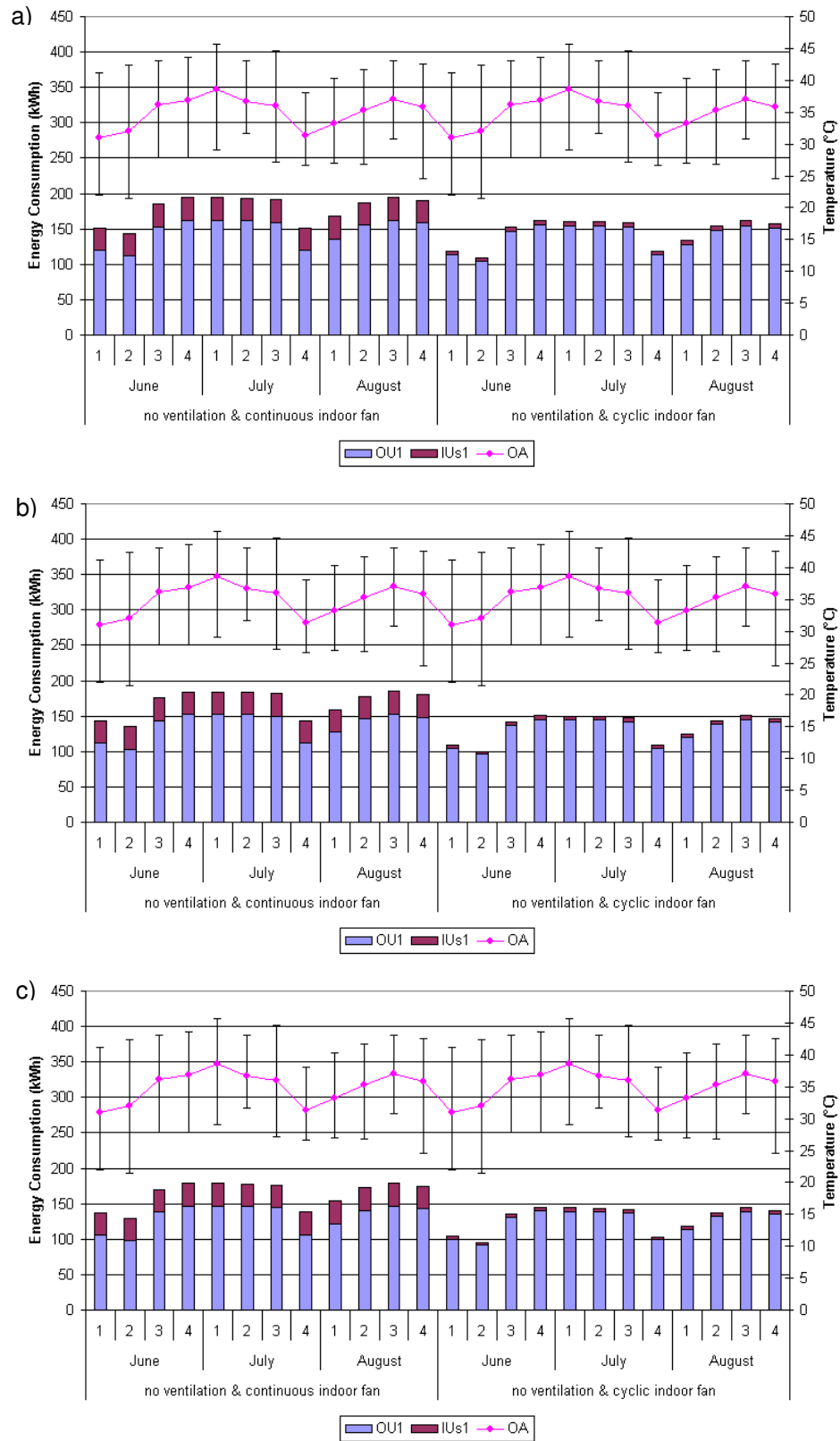


Figure 7.49 : Weekly energy consumptions of non-ventilated System1 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

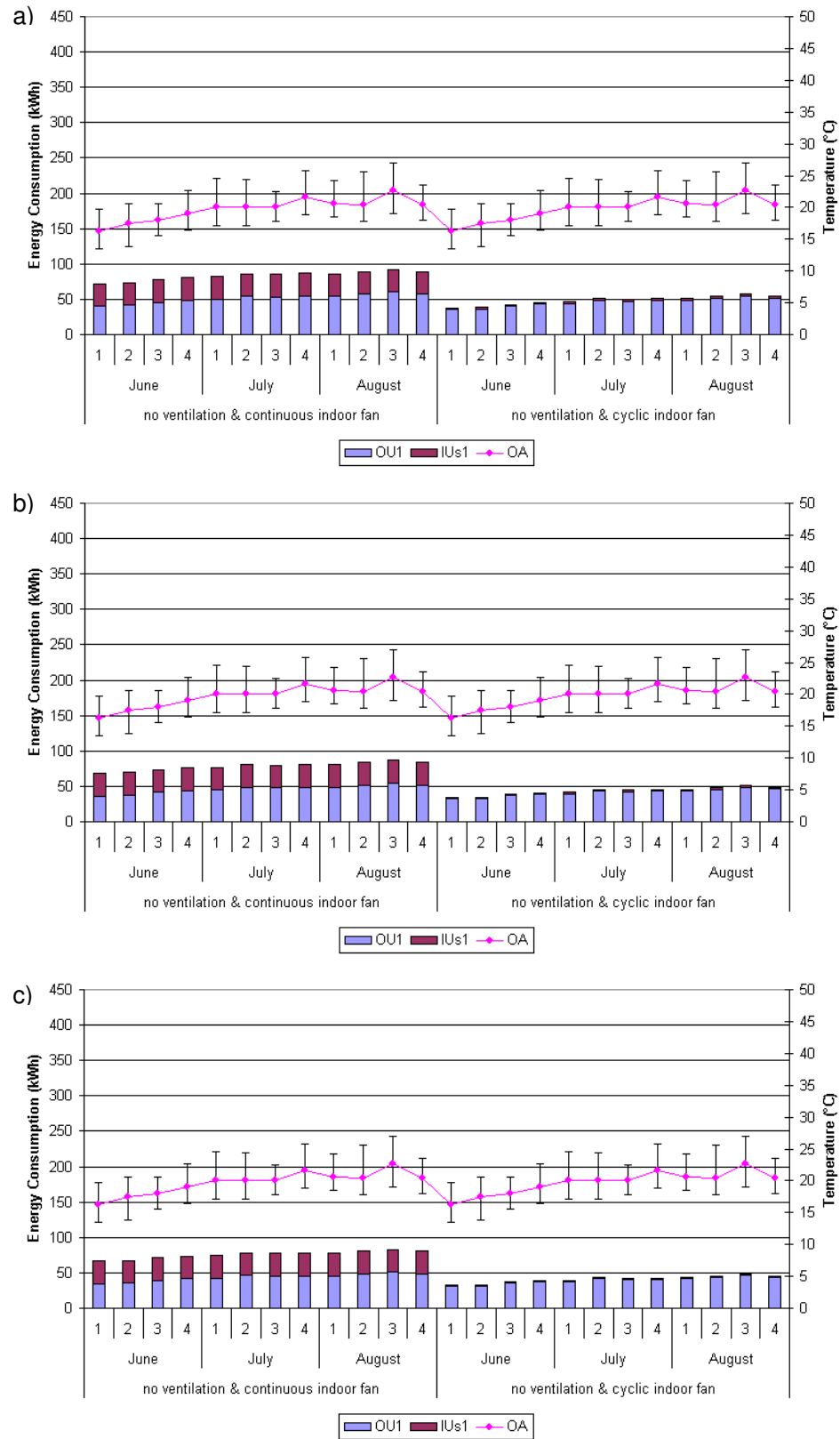


Figure 7.50 : Weekly energy consumptions of non-ventilated System1 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.51 and Figure 7.52 show the energy savings of outdoor and indoor units for non-ventilated System1 for AZ and CA, respectively.

Similar to the results of the ventilation assisted System1, as seen from Figure 7.52, the energy saving potential for CA due to the cyclic indoor fan operation is higher than AZ. The energy saving varies within 91-94%, 92-95% and 93-96% at 23°C, 25°C and 27°C indoor set temperatures for CA and 80-86%, 82-87% and 84-89% at 23°C, 25°C and 27°C indoor set temperatures for AZ.

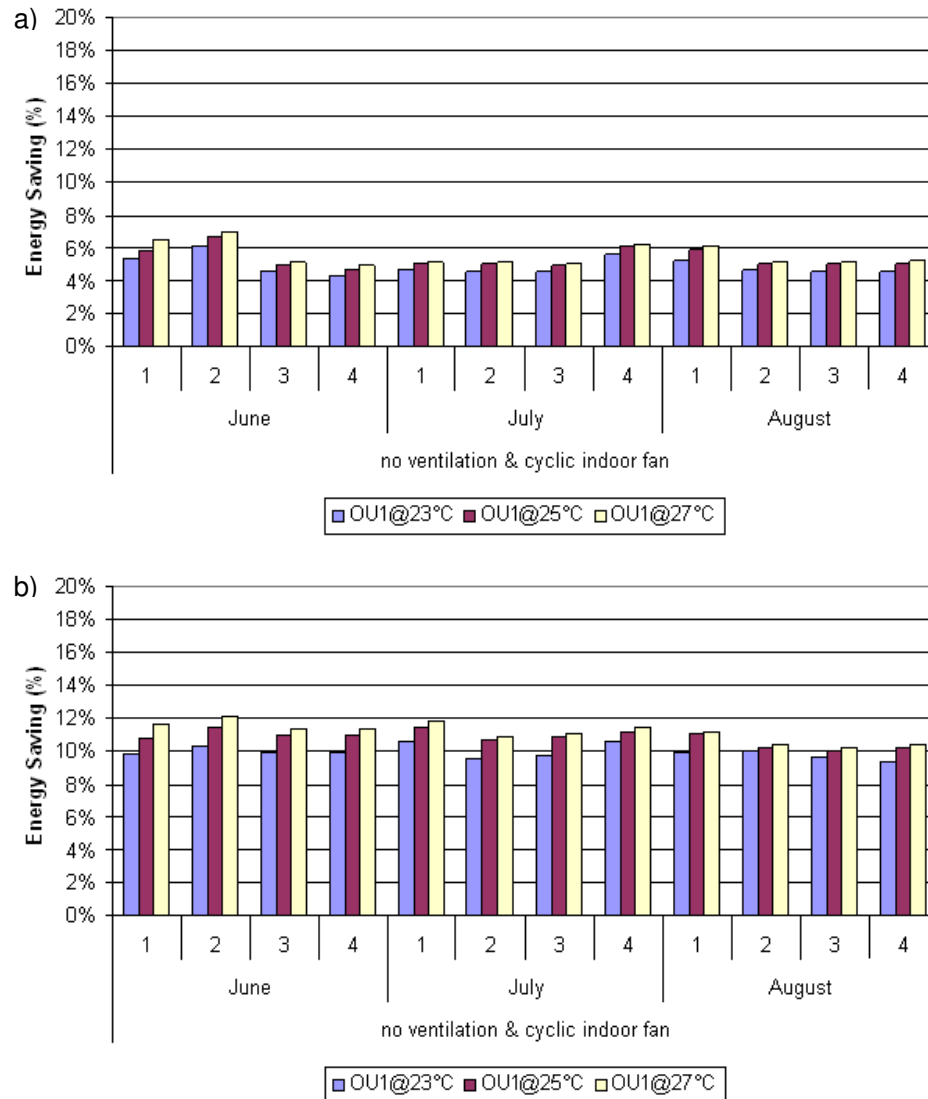


Figure 7.51 : Energy savings of non-ventilated System1 outdoor unit for AZ and CA, (a) AZ, (b) CA

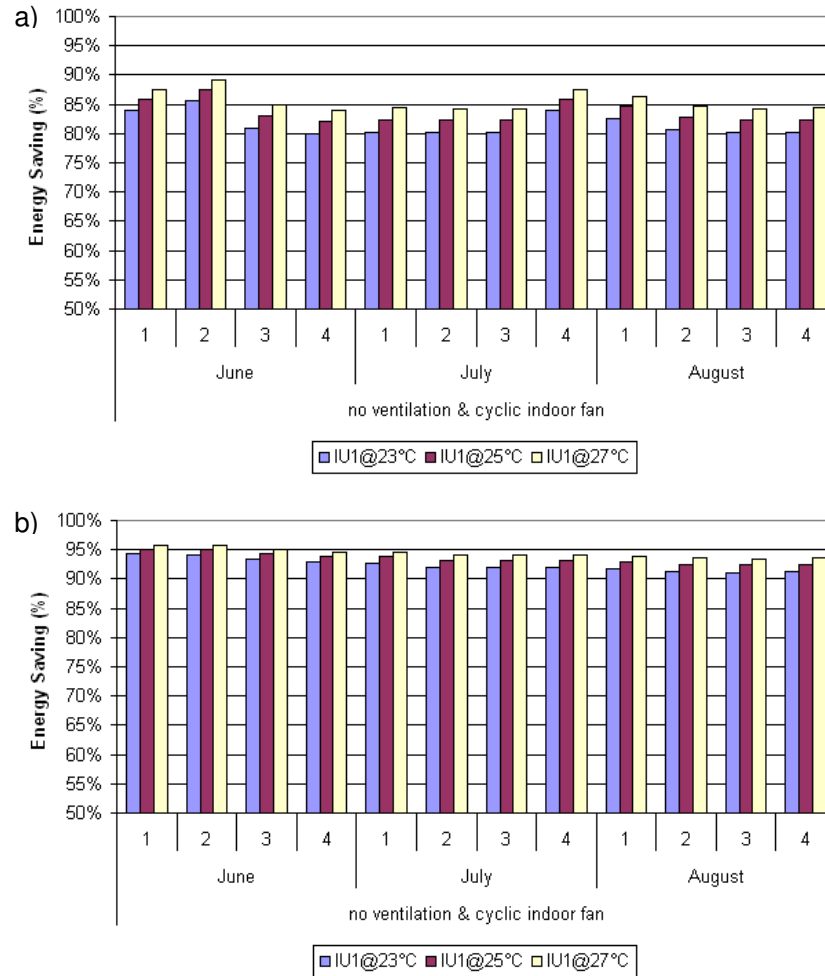


Figure 7.52 : Energy savings of non-ventilated System1 indoor units for AZ and CA, (a) AZ, (b) CA

Figure 7.53 and Figure 7.54 show the indoor conditions of non-ventilated System1 zones at 23°C, 25°C and 27°C indoor set temperatures for AZ and CA, respectively. As can be seen from Figure 7.53, similar to ventilation assisted System1 results shown in Figure 7.31, when the outdoor temperature is high, the capacities of the indoor units are not sufficient to maintain the indoor set temperatures. The main difference between the ventilation assisted and non-ventilated indoor conditions is the indoor humidity ratio. As can be seen from Figure 7.31 and Figure 7.53, the indoor humidity ratio for the non-ventilated System1 is relatively low compared to the ventilation assisted System1. On the other hand, unlike to the ventilation assisted System1 for CA shown in Figure 7.32, there is not any scatter for the indoor conditions, shown in Figure 7.54. Internal heat gain is the reason of this situation, which increases the indoor temperature.

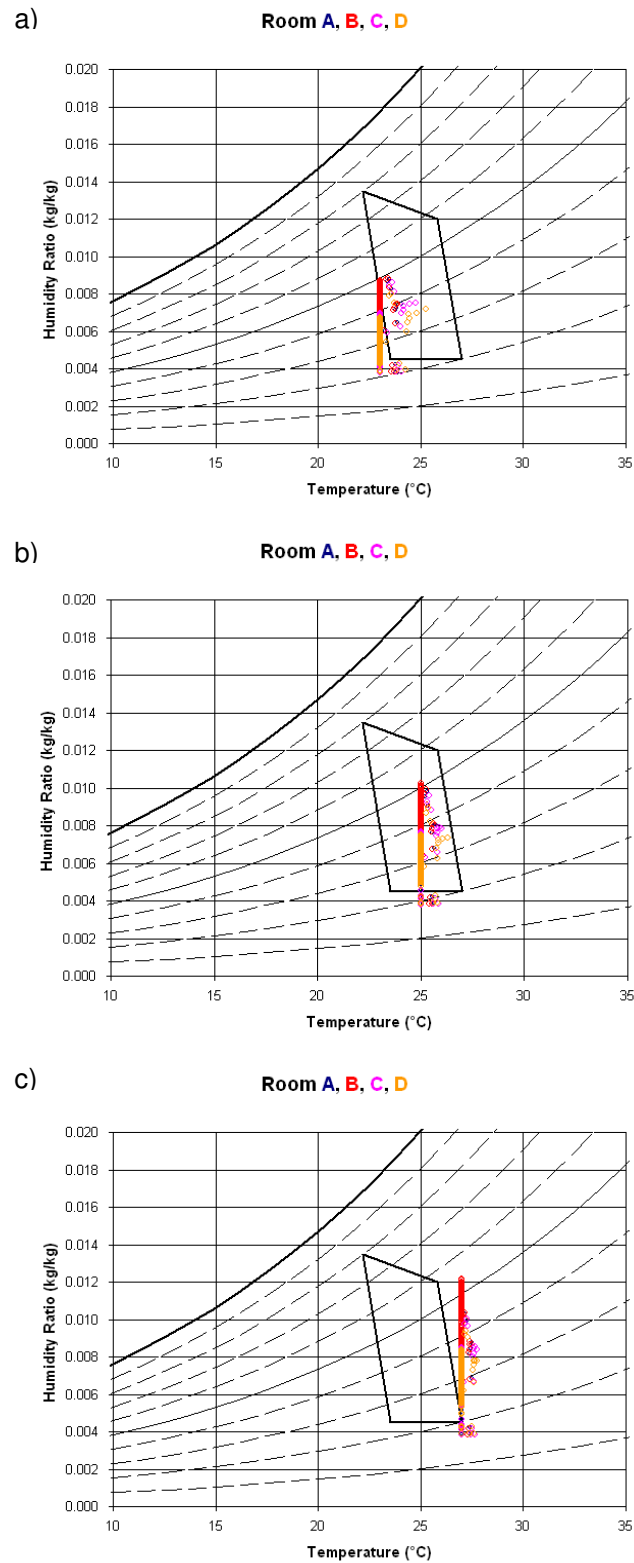


Figure 7.53 : Indoor conditions of non-ventilated System1 zones with cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

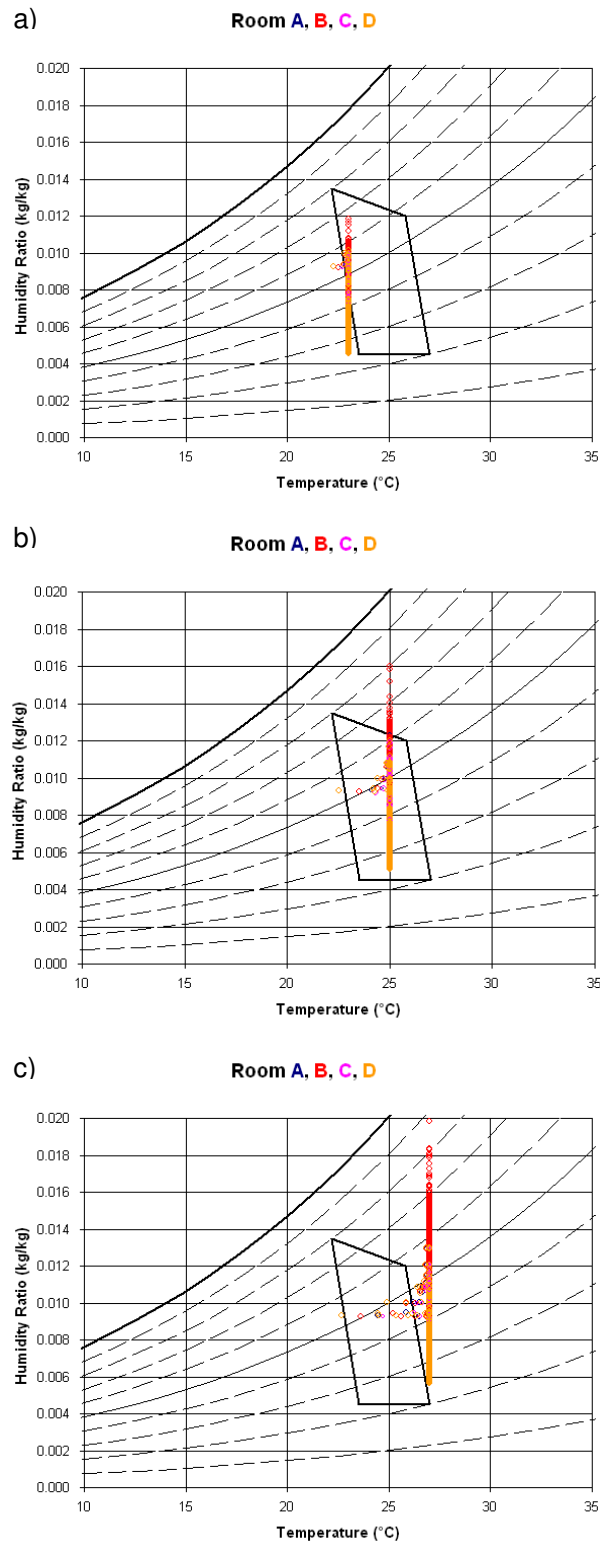


Figure 7.54 : Indoor conditions of System1 zones with cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.55 shows the energy consumptions of ventilation assisted and non-ventilated OU1 and OU2 for CA.

As can be seen, for the lower outdoor temperature cases, the non-ventilated systems consume higher energy than the ventilation assisted systems due to the internal cooling load, which agrees with the experimental results. This situation also explains the difference between the indoor conditions shown in Figure 7.32 and Figure 7.54.

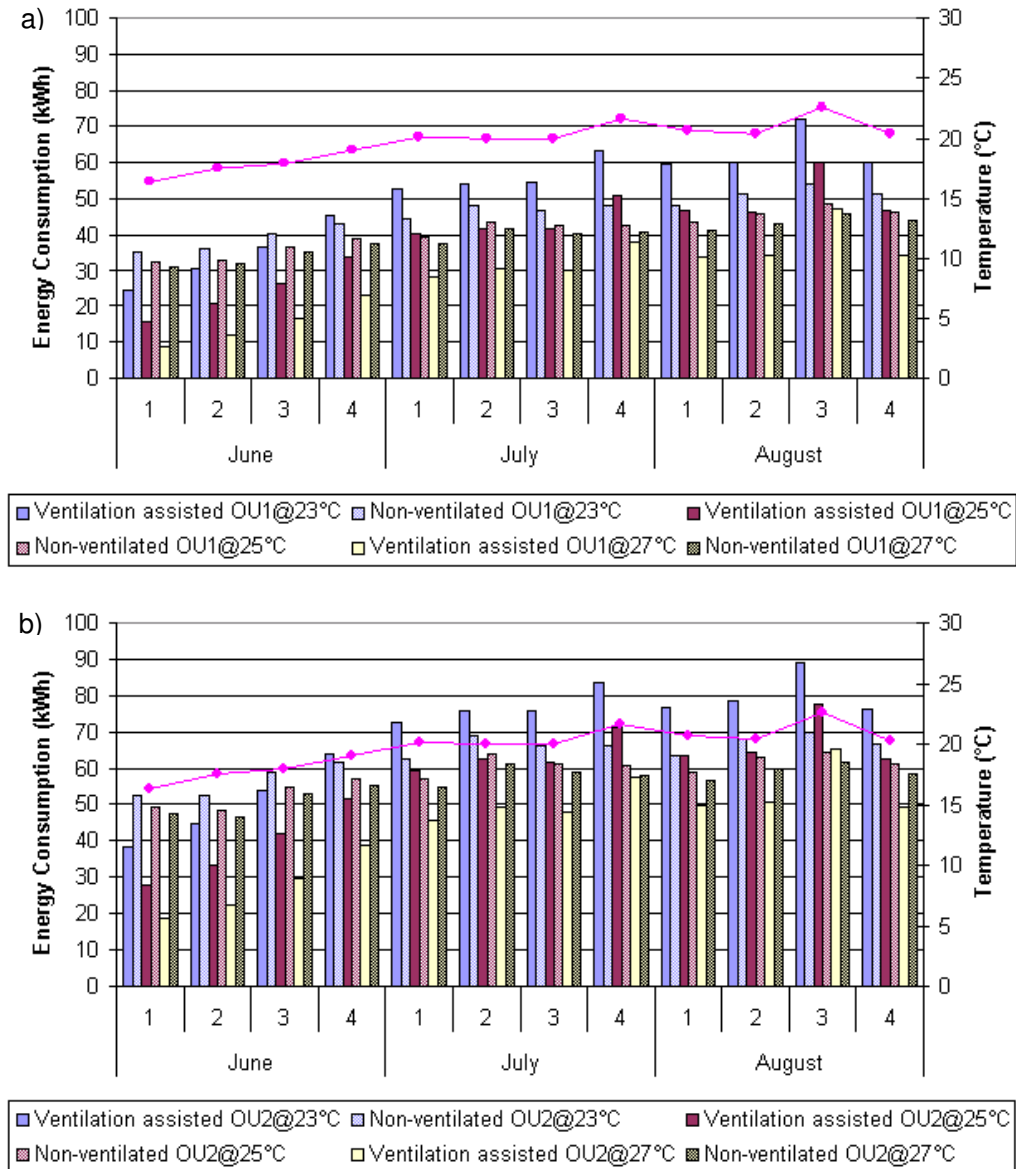


Figure 7.55 : Energy consumption of ventilation assisted and non-ventilated OU1 and OU2 for CA, (a) OU1, (b) OU2

Figure 7.56 and Figure 7.57 depict the weekly energy consumption of non-ventilated System2 for AZ and CA, respectively.

Similar to the results of the ventilation assisted System2, it can be observed from Figure 7.44, Figure 7.56 and Figure 7.57 that since AZ's outdoor temperature is the highest and CA's outdoor temperature is the lowest among the three states for the same period of June-August 2007, the outdoor unit of the multi-split VRF system consumes the highest energy in AZ and the lowest energy in CA.

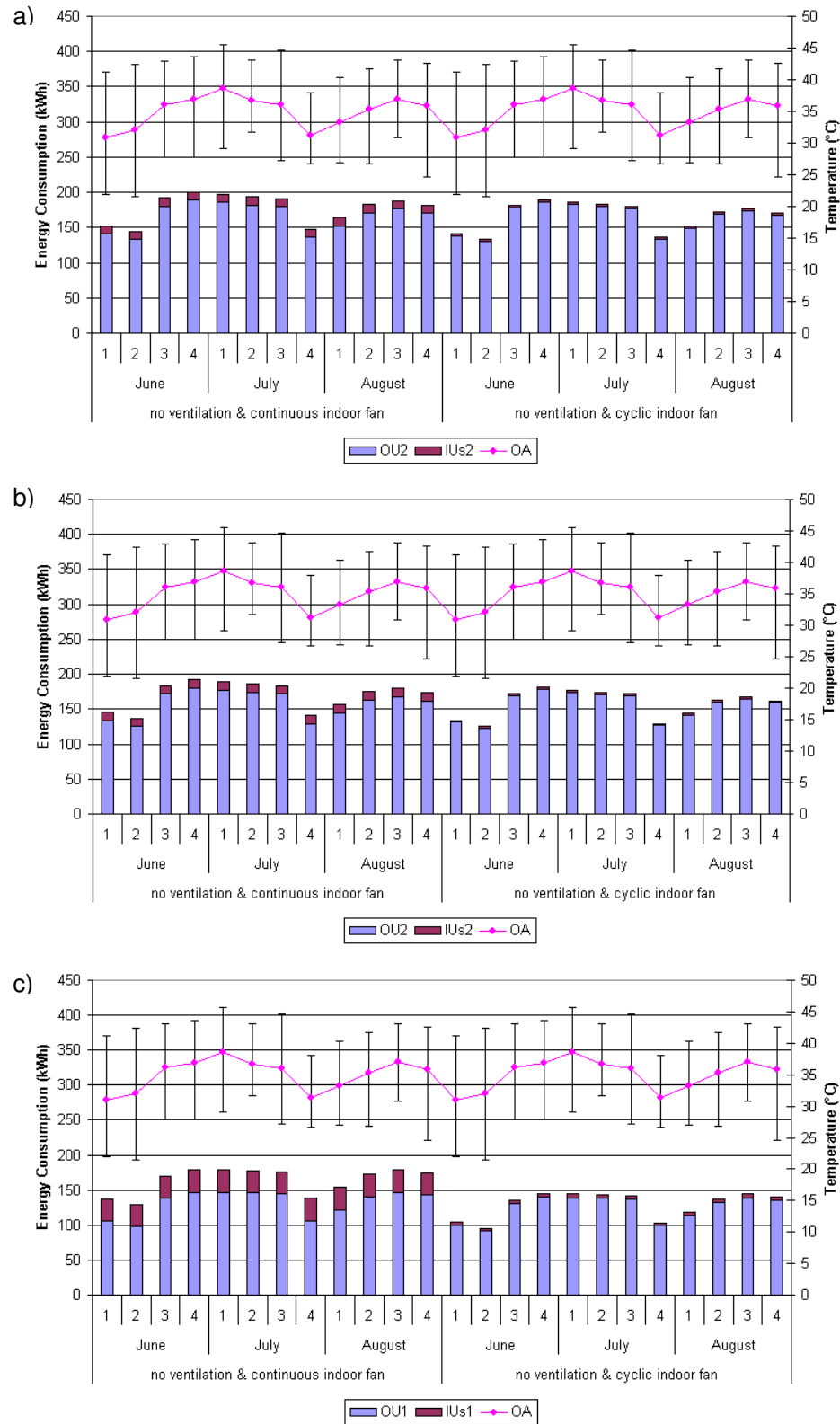


Figure 7.56 : Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

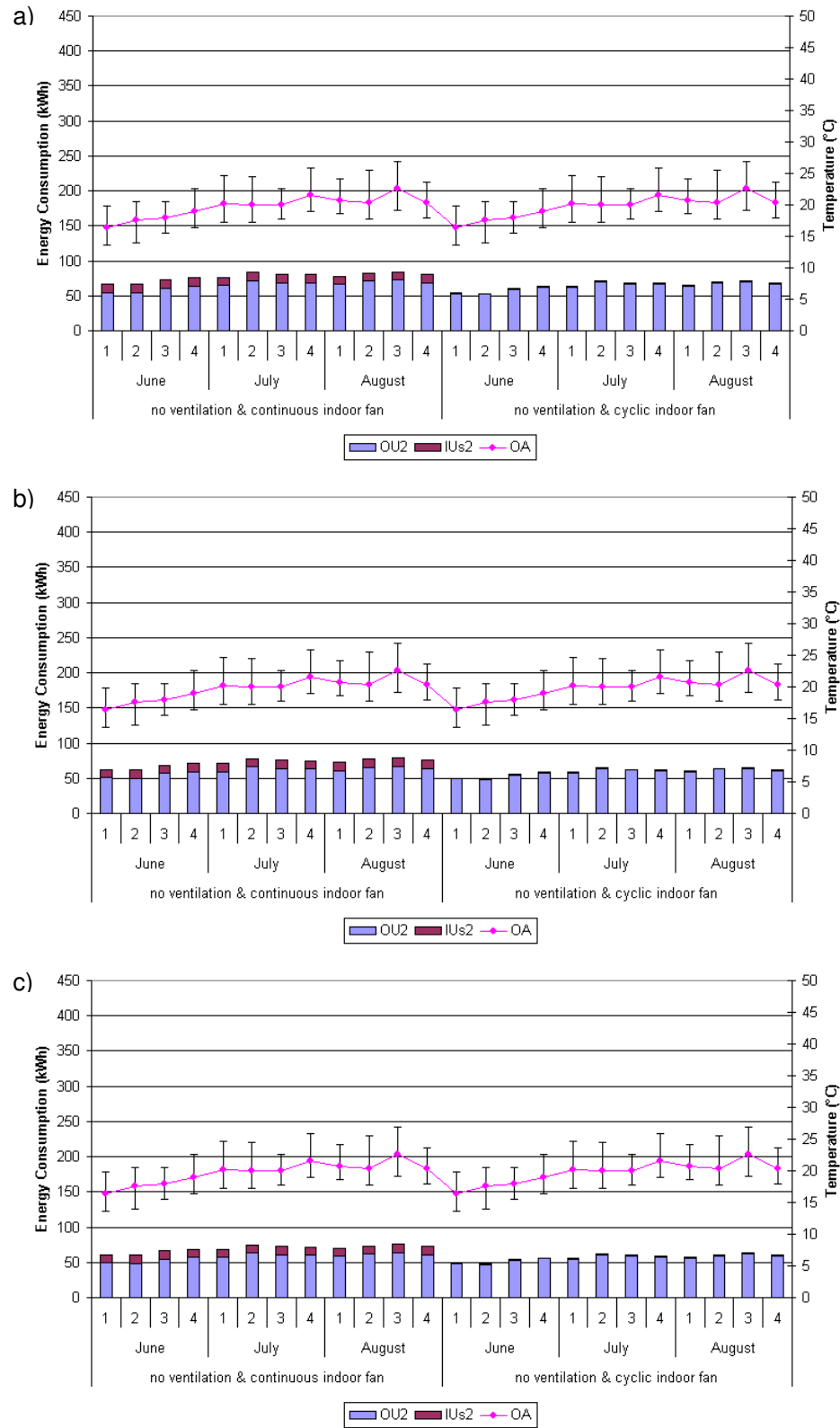


Figure 7.57 : Weekly energy consumptions of non-ventilated System2 at 23°C, 25°C and 27°C indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.58 and Figure 7.59 show the energy savings of outdoor and indoor units for non-ventilated System2 for AZ and CA, respectively.

Similar to results of the non-ventilated System1, as seen from Figure 7.51, the energy saving potential for CA due to the cyclic indoor fan operation is higher than AZ. The energy saving varies within 86-90%, 88-91% and 89-92% at 23°C, 25°C and 27°C indoor set temperatures for CA and 71-79%, 74-81% and 76-83% at 23°C, 25°C and 27°C indoor set temperatures for AZ.

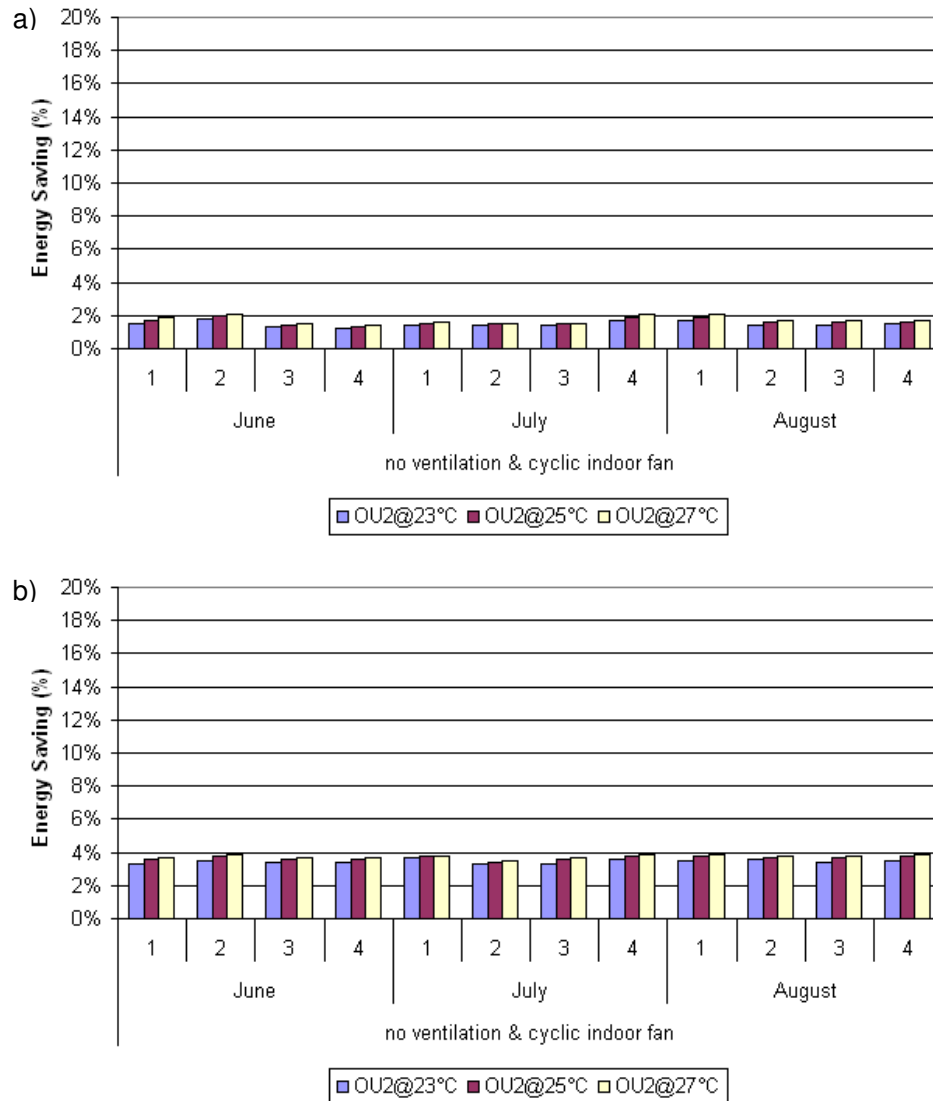


Figure 7.58 : Energy savings of non-ventilated System2 outdoor unit for AZ and CA, (a) AZ, (b) CA

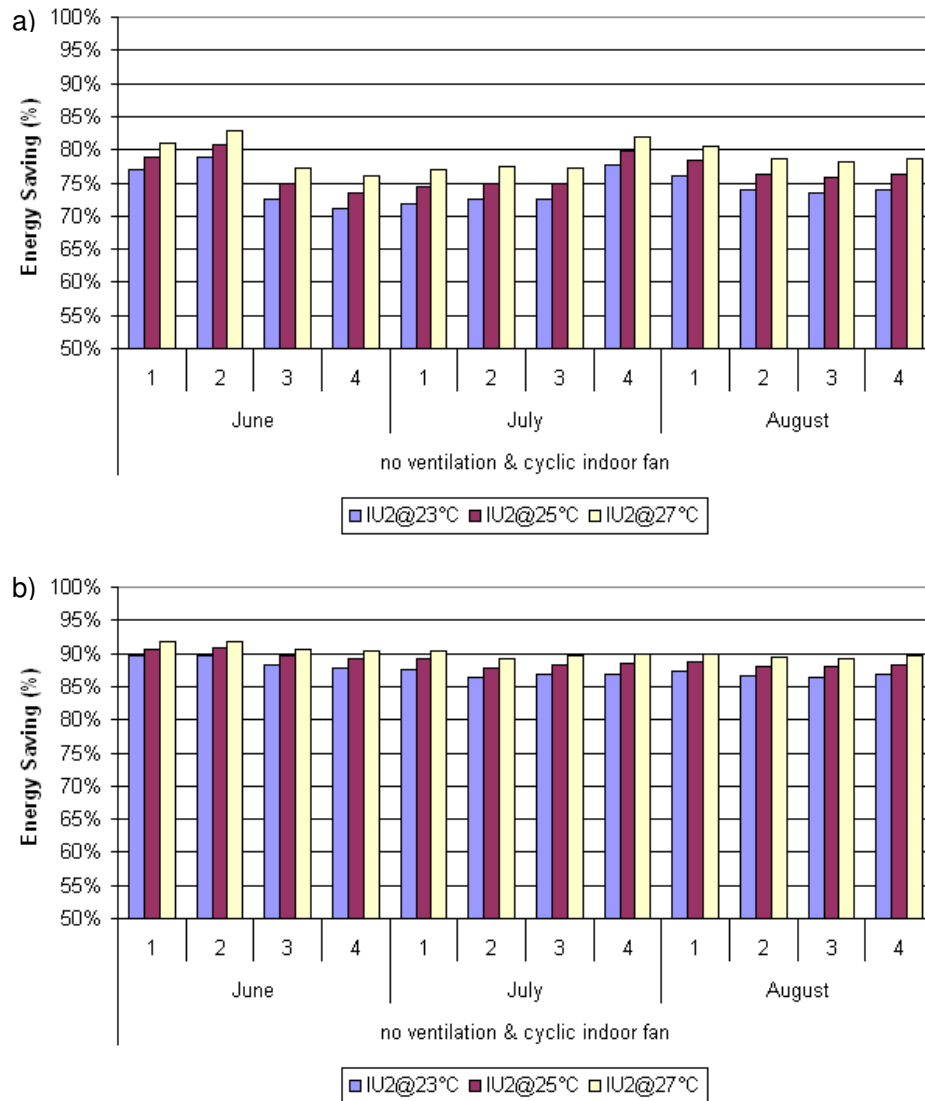


Figure 7.59 : Energy savings of non-ventilated System2 indoor units for AZ and CA, (a) AZ, (b) CA

Figure 7.60 and Figure 7.61 show the indoor conditions of non-ventilated System2 zones at 23°C, 25°C and 27°C indoor set temperatures for AZ and CA, respectively.

Since there is not any ventilation introduced to the indoors, the indoor humidity ratio decreases during the operation of the system, as explained previously. That's why; the data cloud shown in Figure 7.37 appears in the lower humidity ratio levels for the non-ventilation system in AZ. On the other hand, it is observed that the indoor conditions of the non-ventilation System1 zones shown in Figure 7.54 and System2 zones shown in Figure 7.61 are similar for CA, except the humidity levels.

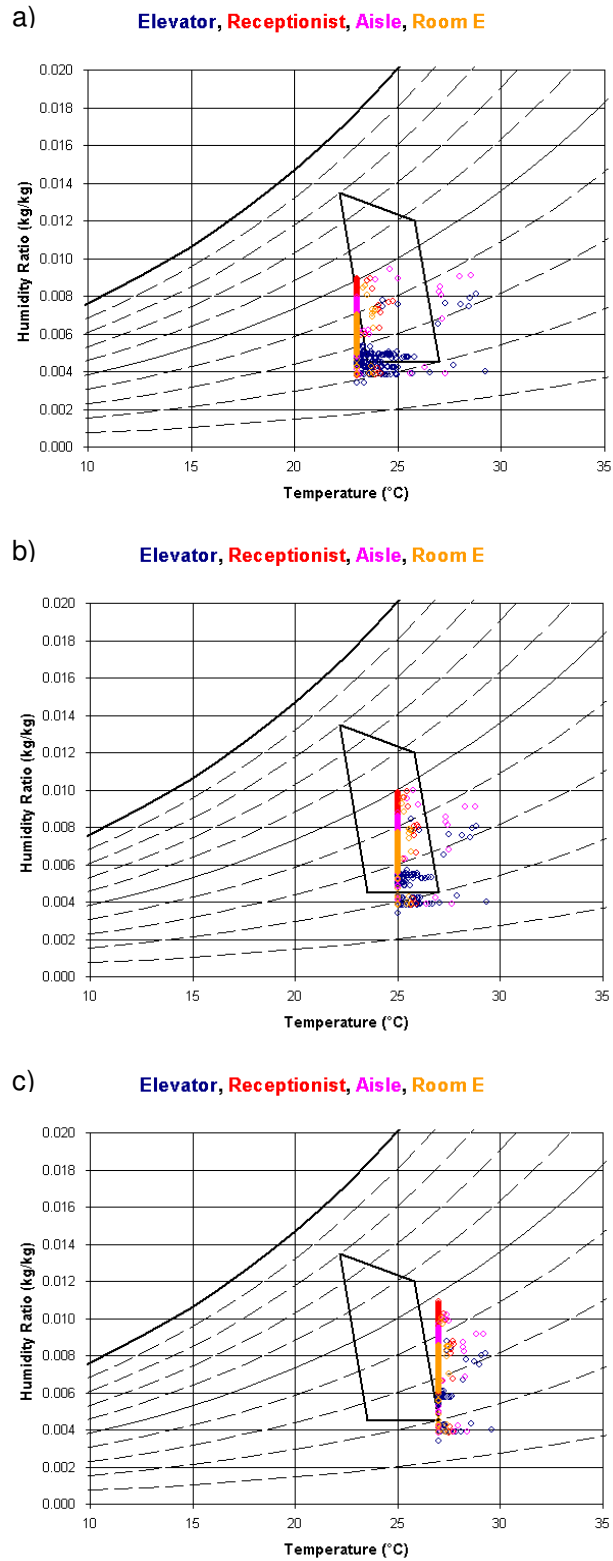


Figure 7.60 : Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for AZ, (a) 23°C, (b) 25°C, (c) 27°C

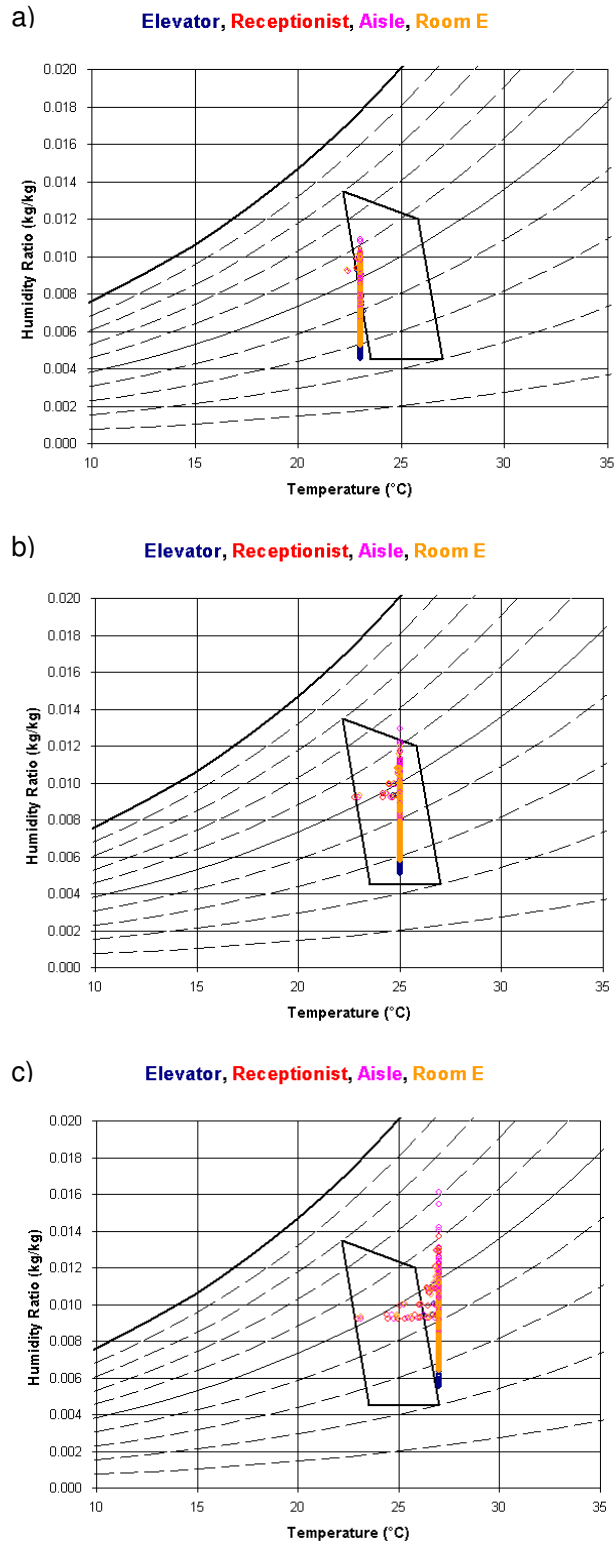


Figure 7.61 : Indoor conditions of non-ventilated System2 zones with cyclic indoor fan operation at different indoor set temperatures for CA, (a) 23°C, (b) 25°C, (c) 27°C

Figure 7.62 and Figure 7.63 show CPF and PLR of non-ventilated System1 and non-ventilated System2 for MD, AZ and CA at 25°C indoor set temperatures, respectively.

It is observed from Figure 7.62 that, similar to the results of the ventilation assisted system, the cyclic indoor fan operation has the potential to improve the efficiency of the multi-split VRF system depending on the outdoor condition. The indoor unit energy saving potential is the highest in CA among the three states.

It can be observed from Figure 7.63 that, similar to the results of the ventilation assisted system, PLR increases with the increasing outdoor temperature due to the increasing cooling load, and has the highest in AZ. The cyclic indoor unit fan operation does not have a big influence on PLR.

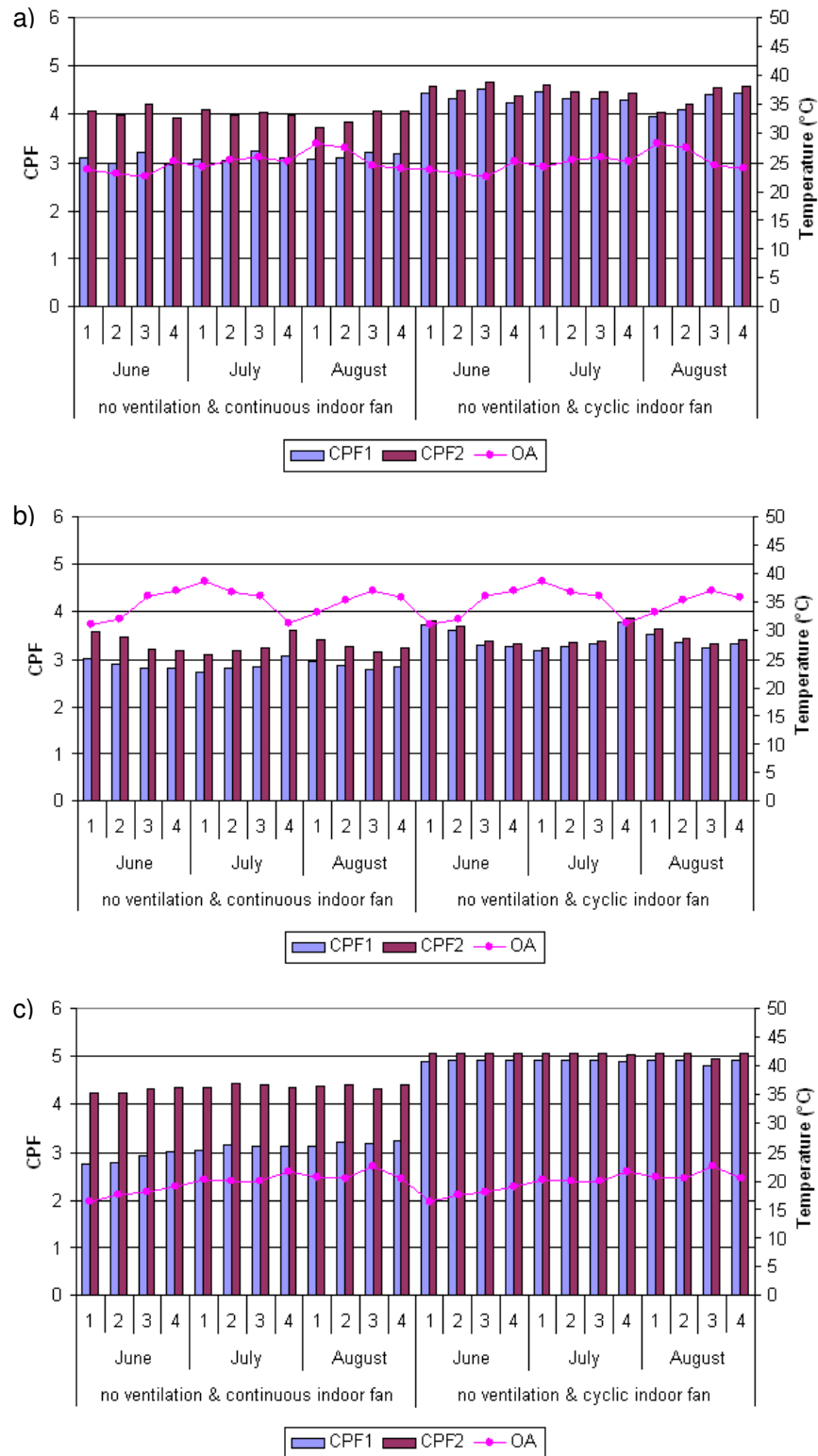


Figure 7.62 : CPF of non-ventilated System1 and non-ventilated System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA

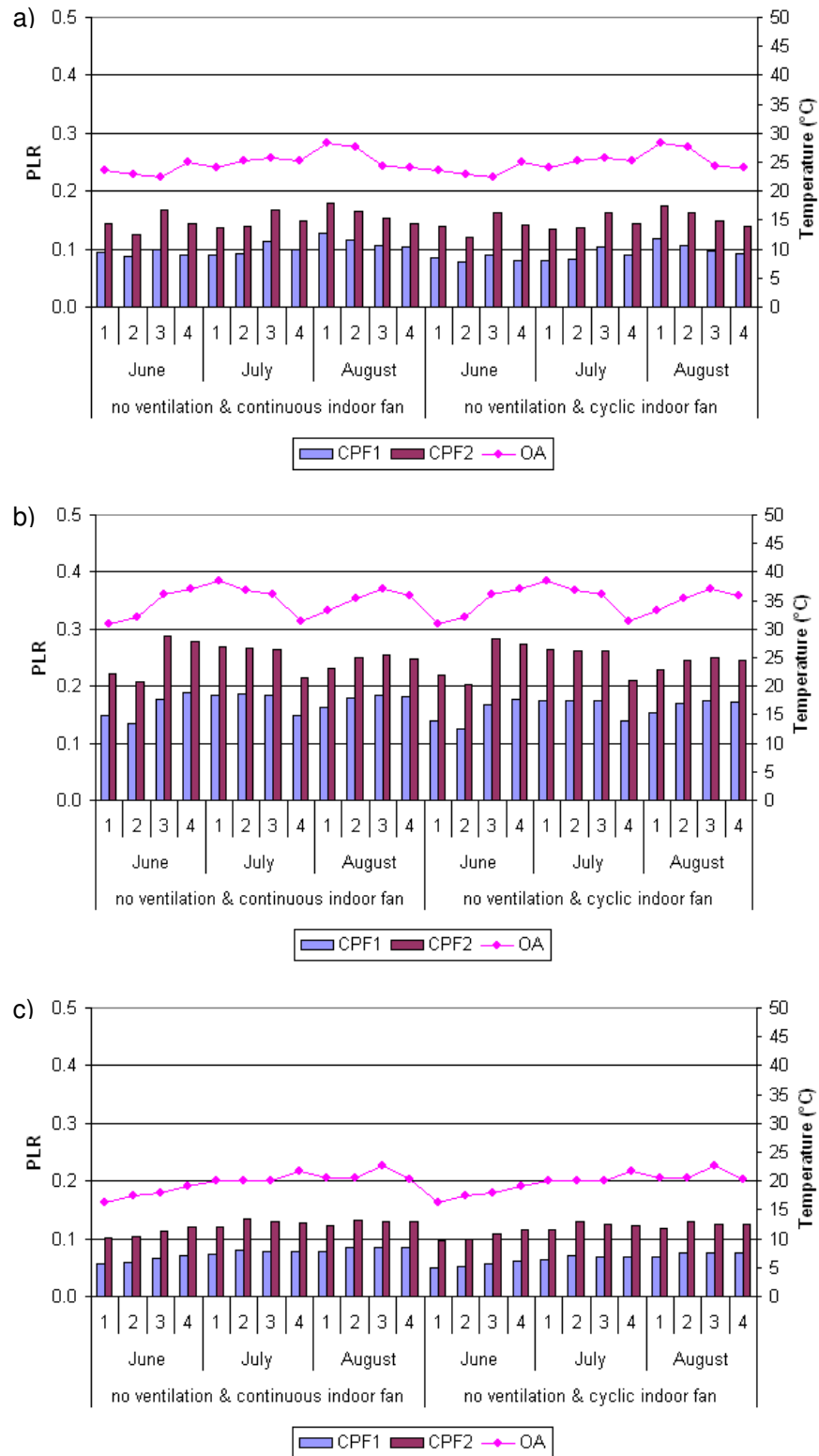


Figure 7.63 : PLR of non-ventilated System1 and non-ventilated System2 for MD, AZ and CA at 25°C indoor set temperature, (a) MD, (b) AZ, (c) CA

8. CONCLUSIONS

8.1 Conclusions

The distinguishing contributions of this research are summarized as follows:

- A field test of a multi-split VRF system, the first time in the open literature, was performed and presented in this research. Two multi-VRF systems and four HRV units were installed in an actual office suite for air conditioning and ventilation purposes, respectively and both systems and indoor and outdoor environments of the office suite were fully instrumented for the measurement and comparison purposes.
- The multi-split VRF systems were operated both in the cooling and heating seasons. The heating season operation of the multi-split VRF system was the first study in the open literature excluding the three-pipe multi-split VRF systems.
- The indoor temperature control of the multi-VRF systems were performed based on both individual and master control modes. Individual control mode is the common control strategy of the multi-split VRF systems, however for the first time; the master control mode, which is a common control strategy of the ducted type direct exchange systems in the US, was applied to a multi-split VRF system for the indoor temperature control.
- The effect of the ventilation system on the performance of the multi-split VRF system was presented for the first time in this research.
- Extensive thermal comfort analyses of the multi-split VRF system based on ASHRAE TSS and ASHRAE thermal comfort zones were performed for the first time in this research. Both individual and master control modes, as well as the ventilation assisted and non-ventilated operations were considered in the thermal comfort evaluations.
- A multi-split VRF module developed for EnergyPlus energy simulation software program obtained from the literature was applied to the current study and validated with experimental results. The effect of the ventilation and the operations of the indoor unit fan on the performance of the multi-split

VRF systems were evaluated through simulation under varying weather conditions for the first time in this research.

The main conclusions of this research are summarized as follows:

- The cooling performance of the multi-split VRF system in both individual and master control modes was experimentally evaluated with two fixed indoor set temperatures, 20°C and 25°C under varying outdoor conditions.
 - It was observed that, inherently, when the multi-split VRF systems were operated in individual control mode, the indoor units were operated individually, however, when the multi-split VRF systems were operated in master control mode, all the indoor units were operated simultaneously.
 - It was found that the number of the indoor units in operation affected the power consumption of the outdoor unit due to varying demand of the refrigerant mass flow rates. The inverter frequency of the inverter driven compressor increased with the increasing number of indoor units in operation in order to provide the required refrigerant mass flow rate to the system resulting in an increase in the power consumption.
 - It was seen from the experimental results that System1 in master control mode consumed higher energy compared to the individual control mode; on the other hand, System2 in individual control mode consumed higher energy compared to the master control mode. Overall, it was found that the multi-split VRF system in master control mode consumed within 12-31.7% higher energy than the individual control mode depending on the indoor set temperature.
 - According to ASHRAE TSS and ASHRAE summer thermal comfort zone evaluations, it was observed that for both indoor set temperatures, 20°C and 25°C, the multi-split VRF systems in individual control mode controlled the indoor temperature much better than the master control mode with less indoor temperature fluctuations. The master control mode provided much colder environment for System1 zones and relatively warmer environment for elevator zone and colder environment for the rest of the System2 zones.
 - It was observed that the control mode did not affect the efficiency of the multi-split VRF system, however affected the energy consumption and accordingly the indoor thermal comfort.

- It was seen from the experimental results that the efficiency of the multi-split VRF system degraded with the increasing cyclic operation. That's why; decreasing the indoor set temperature improved the efficiency by decreasing the cyclic operation especially at low outdoor temperatures.
- The effect of the ventilation system on the performance of the multi-split VRF system in the cooling mode was experimentally evaluated. The multi-split VRF systems were operated in individual control mode with a fixed indoor set temperature of 25°C during the experiments, which were performed with and without ventilation system under varying outdoor conditions.
 - It was obtained that the ventilation did not affect the indoor set temperature control of the multi-split VRF system. Both ventilation assisted and non-ventilated multi-split VRF systems could maintain the indoor set temperature of 25°C under varying outdoor conditions.
 - It was seen that the ventilation increased the indoor humidity ratio around 0.002kg/kg and according to the ASHRAE summer thermal comfort zone, the ventilation assisted system provided less comfortable indoor environment compared to the non-ventilated system.
 - It was observed from the experimental results that the percentage "ON" time ratio of the non-ventilated system was higher than that of the ventilation assisted system for the outdoor temperature lower than the set temperature. This is because, the ventilation assisted system utilized the relatively low outdoor temperature to cool down the indoors, however the non-ventilated system needed to take care of the internal load. On the other hand, for the outdoor temperature higher than the set temperature, the ventilation assisted system faced with higher cooling loads compared to the non-ventilated system, that's why the percentage "ON" time ratio of the ventilation assisted system was higher than that of the non-ventilated system.
 - As expected, it was found that the ventilation assisted multi-split VRF system consumed higher energy compared to the non-ventilated multi-VRF system due to the ventilation loads.
 - It was observed that the ventilation did not have an effect on the efficiency of the multi-split VRF system, however affected the energy consumption and the indoor humidity ratio and consequently the thermal comfort.

- The heating performance of the multi-split VRF system in both individual and master control modes was experimentally evaluated with two fixed indoor set temperatures, 23.3°C and 26.1°C under varying outdoor conditions.
 - It was observed that EEVs were no longer in the upstream of the indoor units in the heating mode. This caused a potential risk of refrigerant and/or oil accumulation in the indoor units. In order to prevent this risk, each EEV was kept open at least at the minimum opening during the “ON” time of the outdoor unit
 - It was found from the experimental results that the multi-split VRF system in individual control mode consumed more energy than the multi-split VRF system in master control mode.
 - According to ASHRAE TSS and ASHRAE winter thermal comfort zone evaluations, it was observed that the multi-split VRF system in individual control mode provided better thermal comfort than the multi-split VRF system in master control mode. It was observed that the multi-split VRF system in master control mode provided much cooler indoor environment, and could not maintain the set temperature for especially critical zones, such as elevator and Room A which had relatively high heating loads. Since the multi-split VRF system in individual control mode could maintain the set temperatures for even elevator and Room A, this caused higher energy consumption compared to the master control mode.
 - Two different indoor unit fan operations; continuous and synchronized fan operation were applied to the multi-split VRF systems for the individual control mode. It was found that the continuous indoor unit fan operation caused overheating of some particular zones resulting in an uncomfortable warm indoor environment. However, the synchronized indoor unit fan operation had the potential of preventing of the overheating and provided much better thermal comfort.
 - Similar to the cooling performance results, it was observed that the control mode did not affect the efficiency of the multi-split VRF system in the heating mode, however affected the energy consumption and accordingly the indoor thermal comfort. It was also found that the efficiency of the heating mode was much lower than that of the cooling mode due to the operational differences. For the same inverter frequency, it was found that the power consumption of the multi-split VRF system in heating mode was

higher than that of the multi-split VRF system in cooling mode, which caused lower efficiency. In addition, it was observed that the defrost requirement also degraded the heating efficiency.

- The effect of the ventilation system on the performance of the multi-split VRF system in the heating mode was experimentally evaluated. The multi-split VRF systems were operated in individual control mode with a fixed indoor set temperature of 23.3°C during the experiments, which were performed with and without ventilation system under varying outdoor conditions.
 - Similar to the cooling performance results, it was obtained that the ventilation did not affect the indoor set temperature control of the multi-split VRF system.
 - According to ASHRAE winter thermal comfort zone, the minimum indoor humidity ratio is around 0.0045kg/kg. However, it was observed from the experimental results that most of the indoor condition data provided by the ventilation assisted multi-split VRF system were below that minimum indoor humidity ratio resulting in a dry indoor environment. On the other hand, it was found that the non-ventilated multi-split VRF system provided more humid indoor environment than the ventilation assisted one.
 - Similar to the cooling performance results, it was found that the ventilation assisted multi-split VRF system consumed higher energy compared to the non-ventilated multi-VRF system due to the ventilation loads.
 - It was observed that the ventilation did not have an effect on the efficiency of the multi-split VRF system, however affected the energy consumption and indoor humidity ratio and consequently the thermal comfort.
- A multi-split VRF module developed for EnergyPlus energy simulation software program obtained from the literature was applied to the current study and validated with experimental results. The effect of the ventilation and the operations of the indoor unit fan on the performance of the multi-split VRF systems were evaluated through simulation for Maryland, Arizona and California outdoor conditions.
 - It was found that the multi-split VRF systems consumed the highest energy in Arizona and the lowest in California due to the outdoor temperature profiles.

- The economizer promised only 1% energy savings for Arizona, however promised within 5-50% energy savings for California depending on the outdoor condition.
- It was observed that the cyclic indoor unit fan operation had higher impact on the energy savings of the multi-split VRF system than the economizer.
- Reduction in the total energy consumption of the multi-split VRF system, mostly due to the cyclic indoor unit fan operation, improved the efficiency of the system.

8.2 Future Work

The following items are suggested to be addressed for possible future work related to the current investigation.

For a complete picture of the energy consumption and the thermal comfort of the multi-split VRF system integrated with the ventilation system, a similar investigation should be performed with allowable indoor set temperatures other than the fixed indoor set temperatures. This will add the occupant desire for the actual comfort conditions.

It was observed from the experimental results that the temperature of the occupant location was every time lower than the thermostat location. That's why; a similar investigation can be performed with mobile thermostats located to occupant locations; such as on the desk, other than stationary thermostats located on the walls. Mobile thermostats may have the potential to control the indoor temperature according to the occupant location resulting in a better thermal comfort and energy saving.

It was also observed that the indoor units provided cooling or heating to the zones whether or not they were occupied. In order to eliminate the unnecessary "ON" times of the multi-split VRF system, occupant sensors can be used in order to control the indoor unit operation. These sensors will have the potential to reduce the energy consumption of the multi-split VRF system by turning "OFF" the indoor units when the corresponding zones are unoccupied.

It was seen that lower indoor humidity ratio in the cooling season and the higher indoor humidity ratio in the heating season compared to the outdoor humidity ratio improves the indoor thermal comfort. However, the multi-split VRF system integrated with the HRV ventilation system could not achieve these desired conditions completely. That's why; a similar investigation related to the multi-split

VRF system integrated with desiccant and humidification systems can be performed.

Extensive simulation studies for the cooling season were carried out in the presented study. A similar simulation study for the heating season can be performed.

REFERENCES

- Aizlewood, C.E., Coward, S.K.D., Hamilton, L., Raw, G.J. and Wilde, D.J.**, 2002. The impact of humidity on health and comfort in an office building, *9th International Conference on Indoor Air Quality and Climate*, Monterey, California, 671-676.
- Amai, H., Tanabe, S., Akimoto, T., and Genma, T.**, 2007. Thermal sensation and comfort with different task conditioning systems, *Building and Environment*, **42**, 3955-3964.
- ANSI/ASHRAE 41.2.1987**, 1987. Standard methods for laboratory air flow measurements, *American National Standard*, Atlanta, Georgia.
- ASHRAE**, 2001. ASHRAE handbook of fundamentals, Atlanta, Georgia.
- ASHRAE**, 2005. ASHRAE handbook of fundamentals, Atlanta, Georgia.
- BSR/ASHRAE Standard 55P**, 2003. Thermal environmental conditions for human occupancy, *American National Standard*, Atlanta, Georgia.
- Burd, A.L. and Burd, G.S.**, 2000. Mobile thermostat for climate control in residential buildings, *Proceedings of CIBSE/ASHRAE Joint Conference*, Dublin, Ireland, H-1562, 1-22.
- Burd, A.L. and Burd, G.S.**, 2002. Application of mobile wireless thermostat and energy conservation and comfort in residential houses, *ASHRAE Transactions*, **108**, 2, 202-211.
- Chamra, L.M., Huynh, K., and Hodge, B.K.**, 2002. Thermal comfort for sedentary and moderate activity levels, *ASHRAE Transactions*, **108**, 1, 428-434.
- Choi, J.M. and Kim, Y.C.**, 2003. Capacity modulation of an inverter-driven multi-air conditioner using electronic expansion valves, *Energy*, **28**, 141-155.
- EnergyPlus Engineering Document**, 2005. Version 1.2.1, U.S. Department of Energy.
- Goetzler W.**, 2007. Variable refrigerant flow systems, *ASHRAE Journal*, April, 24-31.
- Hai, X.H., Jun, S., Hang, Z.Y. and Bin, T.C.**, 2006. Design and research of the digital VRV multi-connected units with three pipes type heat recovery system, *18th International Refrigeration and Air Conditioning Conference at Purdue*, **R018**, 1-6.
- Han, J., Zhang, G., Zhang, Q., Zhang, J., Liu, J., Tian, L., Zheng, C., Hao, J., Lin, J., Liu, Y. and Moschandreas, D.J.**, 2007. Field study on occupants' thermal comfort and residential thermal environment in a hot humid climate of China, *Building and Environment*, **42**, 12, 4043-4050.
- Hosni, M.H., Jones, B.W., and Sipes, J.M.**, 1998. Total heat gain and the split between radiant and convective heat gain from office and laboratory equipment in buildings, *ASHRAE Transactions*, **104**, 1A, 356-365.

- Hosni, M.H., Jones, B.W., and Xu, H.**, 1999. Experimental results for heat gain and radiant/convective split from equipment in buildings, *ASHRAE Transactions*, **105**, 2, 527-539.
- Hu, S.C. and Yang, R.H.**, 2005. Development and testing of a multi-type air conditioner without using AC inverters, *Energy Conversion and Management*, **46**, 373-383.
- Kline, S. and McClintock, F.**, 1953. Describing uncertainties in single-sample experiments, *Mechanical Engineering*, **75**, 3-8.
- Kwok, A.G. and Chun, C.**, 2003. Thermal comfort in Japanese schools, *Solar Energy*, **74**, 245-252.
- Lee, G., Lee, I.H., Yoo, K.J., Kim, M.S. and Ha, S.**, 2006. Analysis of the operational characteristics of multi system through steady state and dynamic simulation, *18th International Refrigeration and Air Conditioning Conference at Purdue*, **R180**, 1-7.
- Lee, J. and Strand, R.K.**, 2001. An analysis of the effect of the building envelope on thermal comfort using the EnergyPlus program, *Proceedings of the 2001 ACSA (Association of Collegiate Schools of Architecture), Technology Conference*, Austin, TX, July.
- Lijima, H., Tanaka, N., Sumida, Y., and Nakamura, T.**, 1991. Development of a new multi-system air conditioner with concurrent heating and cooling operation, *ASHRAE Transactions*, **97**, 2, 309-315.
- Lin J.L., and Yeh, T.J.**, 2007. Identification and control of multi-evaporator air-conditioning systems, *International Journal of Refrigeration*, article in press.
- Masuda, M., Wakahara, K., and Matsuki, K.**, 1991. Development of a multi-system air conditioner for residential use, *ASHRAE Transactions*, **97**, 2, 127-131.
- McCartney, K.J. and Humphreys, M.A.**, 2002. Thermal comfort and productivity, *9th International Conference on Indoor Air Quality and Climate*, Monterey, California, 822-827.
- Murakami, Y., Terano, M., Mizutani, K., Harada, M., and Kuno, S.**, 2007. Field experiments on energy consumption and thermal comfort in the office environment controlled by occupants' requirements from PC terminal, *Building and Environment*, **42**, 4022-4027.
- Newsham, G.R., Tiller, D.K. and Phil, D.**, 1997. A field study of office thermal comfort using questionnaire software, *ASHRAE Transactions*, **103**, 2, 3-16.
- Park, Y.C., Kim, Y.C., and Min, M.K.**, 2001. Performance analysis on a multi-type inverter air conditioner, *Energy Conversion and Management*, **42**, 1607-1621.
- Riberon, J. and O'Kelly, P.**, 2002. Field investigations on comfort in office buildings, *Proceedings: Indoor Air 2002, 9th International Conference on Indoor Air Quality and Climate*, Monterey, California, 665-670.
- Rohles, F.H. and Nevins, R.G.**, 1971. The nature of thermal comfort for sedentary man, *ASHRAE Research report*, RP-43, No: 2191, 239-246.
- Shah, R., Alleyne, A.G., and Bullard, C.W.**, 2004. Dynamic modeling and control of multi-evaporator air-conditioning systems, *ASHRAE Transactions*, **110**, 109-119.

- Shao S., Shi W., Li, X., and Chen, H.,** 2004. Performance representation of variable-speed compressor for inverter air conditioners based on experimental data, *International Journal of Refrigeration*, **27**, 805-815.
- Shao, S., Shi, W., Li, X., and Yan, Q.,** 2007. Simulation model for complex refrigeration systems based on two-phase fluid network – Part I: Model development, *International Journal of Refrigeration*, Article in press.
- Shi, W., Shao, S., Li, X., and Yan, Q.,** 2007. Simulation model for complex refrigeration systems based on two-phase fluid network – Part II: Model application, *International Journal of Refrigeration*, Article in press.
- Shi, W., Shao, S., Li, X., Peng, X., and Yang, X.,** 2003. A network model to simulate performance of variable refrigerant volume refrigerant systems, *ASHRAE Transactions*, **109**, 2, 61-68.
- Wang, Z.,** 2006. A field study of thermal comfort in residential buildings in Harbin, *Building and Engineering*, **41**, 1034-139.
- Watanabe, N., Tanaka, K., and Hihara, E.,** 2004. Research on annual energy consumption of air conditioning equipment, *Transactions of the JSRAE*, **21**, 2, 157-165.
- Wu, C., Xingxi, Z. and Shiming, D.,** 2005. Development of control method and dynamic model for multi-evaporator air conditioners (MEAC), *Energy and Conversion*, **46**, 451-465.
- www.wunderground.com.** Weather conditions.
- Xia, J., Winandy, E., Georges, B. and Lebrun, J.,** 2004. Experimental analysis of the performances of variable refrigerant flow systems, *Building Services Engineering Research and Technology*, **25**, 17-23.
- Xia, J., Winandy, E., Georges, B., and Lebrun, J.,** 2002. Testing methodology for VRF systems, *9th International Refrigeration and Air Conditioning Conference at Purdue*, **R4-2**, 1-8.
- Xia, J., Zhou, X., Jin, X., and Wu, Y.,** 2003. Simulation study on operating characteristics of multi-evaporator VRV air conditioner, *21st IIR International Congress of Refrigeration*, Washington DC, US, ICR0298, 1-7.
- Zhou, Y.P., Wu, J.Y., Wang, R.Z., and Shiochi, S.,** 2007a. Energy simulation in the variable refrigerant flow air-conditioning system under cooling conditions, *Energy and Buildings*, **39**, 212-220.
- Zhou, Y.P., Wu, J.Y., Wang, R.Z., Li, Y.M., and Shiochi, S.,** 2007b. Development and experimental investigation of the variable-refrigerant-flow (VRF) air conditioning system in EnergyPlus, *International Congress of Refrigeration*, Beijing, China, ICR07-E1-296.
- Zhou, Y.P., Wu, J.Y., Wang, R.Z., Shiochi S., and Li, Y.M.,** 2007c. Simulation and experimental validation of the variable-refrigerant-volume (VRV) air-conditioning system in EnergyPlus, *Energy and Buildings*, Article in press.

APPENDICES

Appendix A.

The manufacturer catalogue data for IURA is provided in Table A.1.

Table A.1 : Catalogue data for IURA

Type of the Indoor Unit		Ceiling mounted cassette type unit
Cooling Capacity, Btu/h (W)		24000 (7033.7)
Heating Capacity, Btu/h (W)		27000 (7912.9)
Heat Exchanger	Rows x Stages	2 x 8
	Fin density, fin/inch (fin/cm)	17 (6.7)
	Face area, ft ² (m ²)	3.56 (0.33)
	Air Flow Rate (High/Low), cfm (m ³ /s)	670 (0.316) / 490 (0.231)

The manufacturer catalogue data for IURB is provided in Table A.2.

Table A.2 : Catalogue data for IURB

Type of the Indoor Unit		Ceiling mounted cassette type unit
Cooling Capacity, Btu/h (W)		36000 (10550.6)
Heating Capacity, Btu/h (W)		40000 (1172.8)
Heat Exchanger	Rows x Stages	2 x 12
	Fin density, fin/inch (fin/cm)	17 (6.7)
	Face area, ft ² (m ²)	5.35 (0.497)
	Air Flow Rate (High/Low), cfm (m ³ /s)	990 (0.467) / 740 (0.35)

Identical indoor units were installed in Rooms C, E and Receptionist and the manufacturer catalogue data for these indoor units is provided in Table A.3.

Table A.3 : Catalogue data for IURC, IURE and IUREc

Type of the Indoor Unit		Wall mounted type unit
Cooling Capacity, Btu/h (W)		24000 (7033.7)
Heating Capacity, Btu/h (W)		27000 (7912.9)
Heat Exchanger	Rows x Stages	2 x 14
	Fin density, fin/inch (fin/cm)	17 (6.7)
	Face area, ft ² (m ²)	2.29 (0.213)
	Air Flow Rate (High/Low), cfm (m ³ /s)	635 (0.3) / 470 (0.221)

Identical indoor units were installed in Room D and Elevator and the manufacturer catalogue data for these indoor units is provided in Table A.4.

Table A.4 : Catalogue data for IURD and IUEle

Type of the Indoor Unit		Wall mounted type unit
Cooling Capacity, Btu/h (W)		12000 (3516.8)
Heating Capacity, Btu/h (W)		13500 (3956. 5)
Heat Exchanger	Rows x Stages	2 x 14
	Fin density, fin/inch (fin/cm)	17 (6.7)
	Face area, ft ² (m ²)	1.73 (0.16)
	Air Flow Rate (High/Low), cfm (m ³ /s)	300 (0.142) / 180 (0.085)

The manufacturer catalogue data for IUAI is provided in Table A.5.

Table A.5 : Catalogue data for IUAs

Type of the Indoor Unit		Wall mounted type unit
Cooling Capacity, Btu/h (W)		18000 (5275.3)
Heating Capacity, Btu/h (W)		20000 (5861.4)
Heat Exchanger	Rows x Stages	2 x 14
	Fin density, fin/inch (fin/cm)	17 (6.7)
	Face area, ft ² (m ²)	2.29 (0.213)
	Air Flow Rate (High/Low), cfm (m ³ /s)	500 (0.236) / 400 (0.189)

The cooling capacities are based on 26.6°C DB/19.45°C WB return air and 35°C outdoor temperature conditions.

The heating capacities are based on 21.1°C return air temperature and 8.3°C outdoor temperature.

Appendix B.

The energy balance between air and the refrigerant sides was evaluated in a heating performance test on December 22, 2006.

A wind tunnel was built according to the ANSI/ASHRAE 41.2.1987 Standard. The dimensions of the width, height and length of the wind tunnel were 0.089m, 0.876m, and 1.6m, respectively. As can be seen from Figure B.1, it was placed at the discharge outlet of IURC.



Figure B.1 : Wind tunnel

The inlet of the wind tunnel was sealed with aluminum duct tapes in order to eliminate the air leakage, as seen from Figure B.1 and Figure B.2.



Figure B.2 : Duct taping for the air leakage

A honeycomb air straightener, shown in Figure B.3, was placed in the tunnel. Three air velocity sensors with an accuracy of $\pm 0.5\%$ of reading and $\pm 1.5\%$ full scale were installed in the wind tunnel shown in Figure B.1 and Figure B.3. The sensors could monitor the air velocity within the sensor range of 0-5m/s.

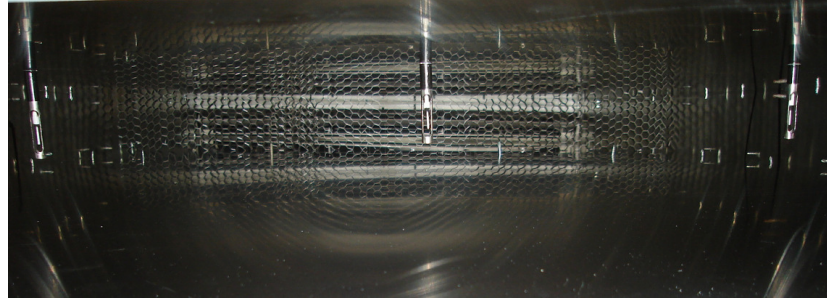


Figure B.3 : Inside of the wind tunnel

The air flow rate was calculated as

$$\dot{V} = V_{ave} \cdot A_w \quad \text{B.1}$$

where

$\dot{V}$: Air flow rate (m^3/s)

V_{ave} : The average air velocity in the wind tunnel (m/s)

A_w : Cross section area of the wind tunnel (m^2)

The indoor unit was operated in high speed configuration and the air flow rate of the indoor unit was found to be $0.314 \text{ m}^3/\text{s}$ which is 4.6% higher than the catalog value of $0.3 \text{ m}^3/\text{s}$.

The air flow rate and blowing, return, indoor and outdoor air temperatures are plotted in Figure B.4. As can be seen from Figure B.4, the last 10 minutes were assumed to be at the steady state conditions.

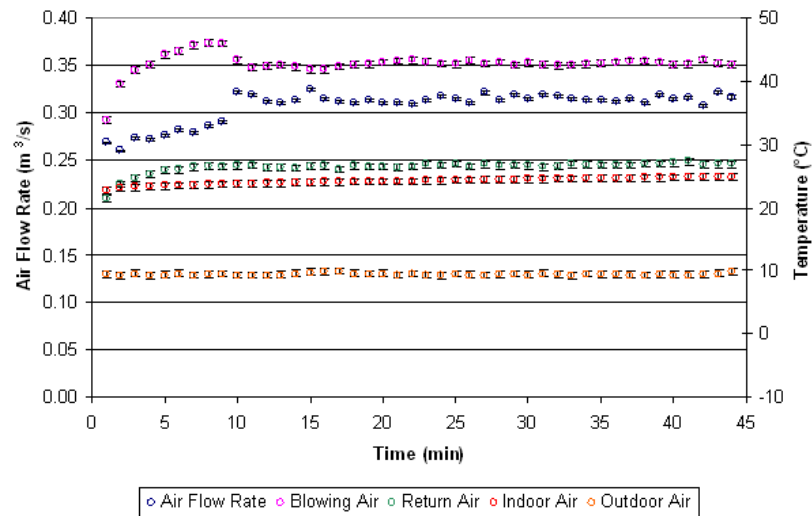


Figure B.4 : Air flow rate and blowing, return, indoor and outdoor air temperatures

The refrigerant inlet and outlet temperatures and the condensing and evaporating pressures are provided in Figure B.5 for the steady-state condition.

The slight increase and the decrease of the temperature, pressure and the EEV opening are due to the control of the system. Since it was a heating test, the evaporating pressure did not have a significant effect on the system, and the rest of the values were assumed in steady state ranges.

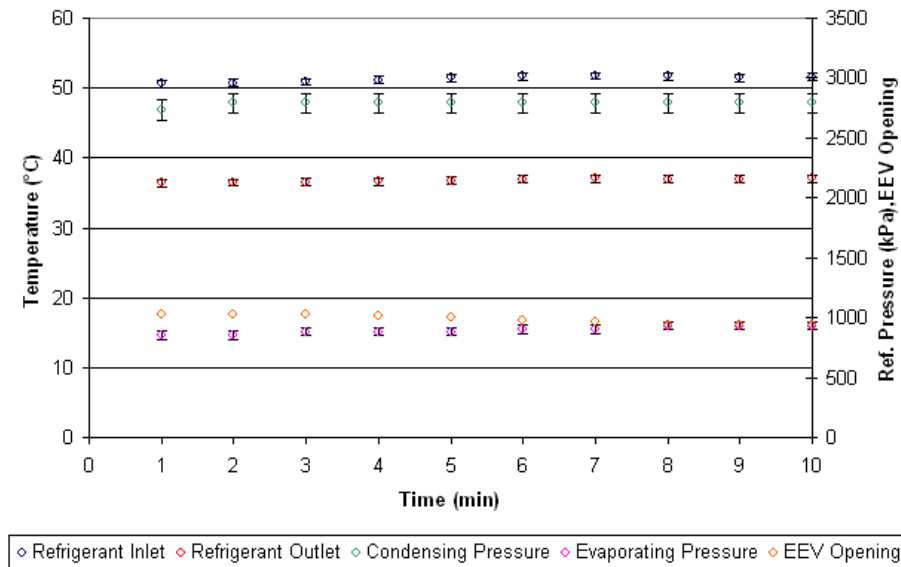


Figure B.5 : Refrigerant inlet and outlet temperatures, condensing and evaporating pressures and the EEV opening

The comparison of the air and refrigerant side of the indoor unit capacity and the error are provided in Figure B.6. It was found that the average error is 6.8%.

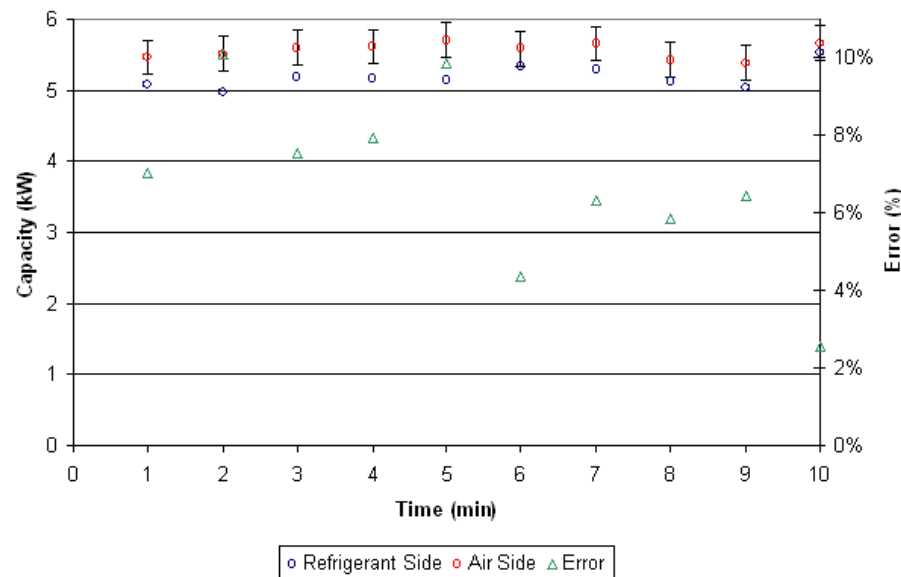


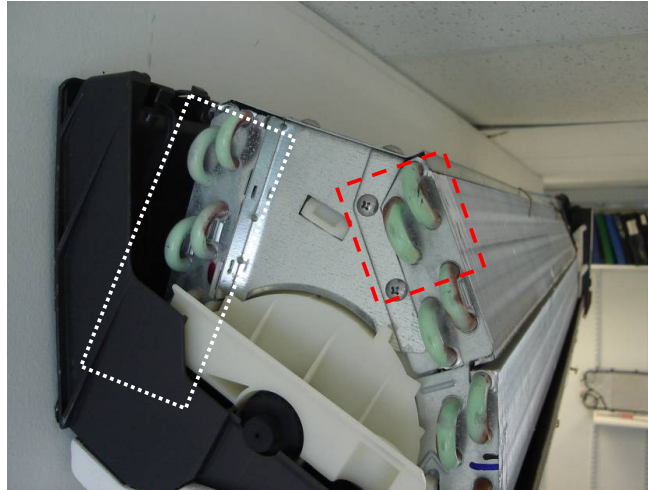
Figure B.6 : Air and refrigerant side of the capacities and the error

Appendix C.

CoilDesigner 3.1 software package was used to calculate the refrigerant side pressure drop of the indoor unit.

The tube banks and number of tubes per bank are provided in Figure C.1 for IURC.

a)



b)



Figure C.1 : Indoor unit tube circuits, a) left view, b) right view

As can be seen from Figure C.1a, the indoor unit had two banks and six tubes per bank for left side (one pair behind the cover) (white rectangular), and two banks and eight tubes per bank for the right side (one pair behind the cover).

Since there was a header in the indoor unit, the total refrigerant mass flow rate was distributed to each tube pairs (red rectangular), totally to seven.

CoilDesigner evaluation for this indoor unit heat exchanger is provided in Figure C.2.

The tube circuit can be seen in the middle of Figure C.2. The refrigerant enters from the red circled tubes and exists from the blue circled tubes. The solid lines represented the tube bends coming out of the paper plane and the dashed lines represent the other tube bends on the other side of the tubes.

The air enters to the coil from the left side and the properties of the air can be seen from the left side of the Figure C.2. The air temperature, relative humidity and the velocity can be input for each segment.

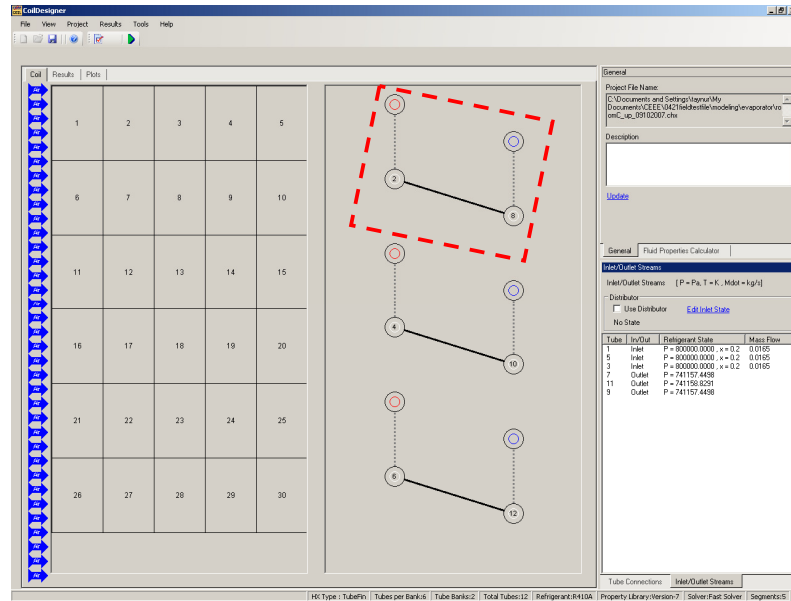


Figure C.2 : CoilDesigner

Figure C.3 shows the variation of the pressure drop of the indoor coil with respect to the inlet refrigerant mass flow rate to each tube pair (red rectangular) at 0.9MPa inlet refrigerant pressure and 0.2 inlet refrigerant quality, and 303.15K, 50% and 0.114m/s air inlet temperature, relative humidity and velocity, respectively. As expected the pressure drop increases with the increasing refrigerant mass flow rate.

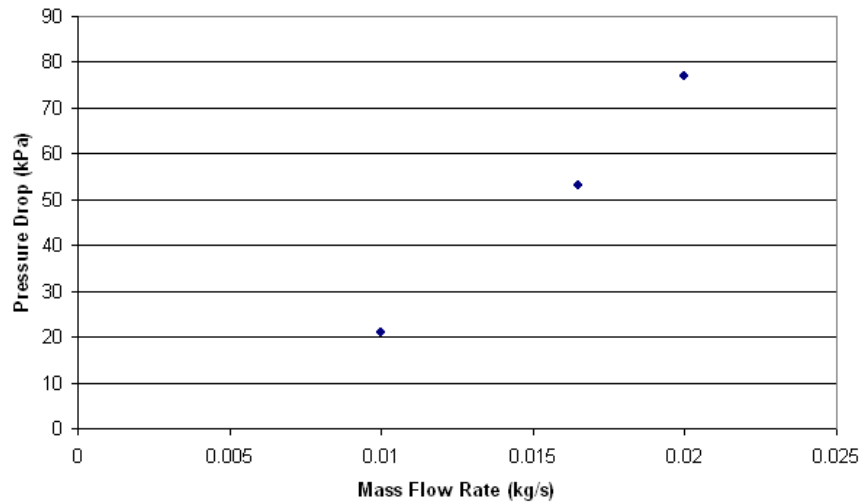


Figure C.3 : Variation of the pressure drop of the indoor unit with respect to the inlet refrigerant mass flow rate

Figure C.4 shows the variation of the pressure drop of the indoor coil with respect to the inlet refrigerant pressure at 0.2 inlet refrigerant quality. As seen, the pressure drop increases with the decreasing inlet refrigerant pressure.

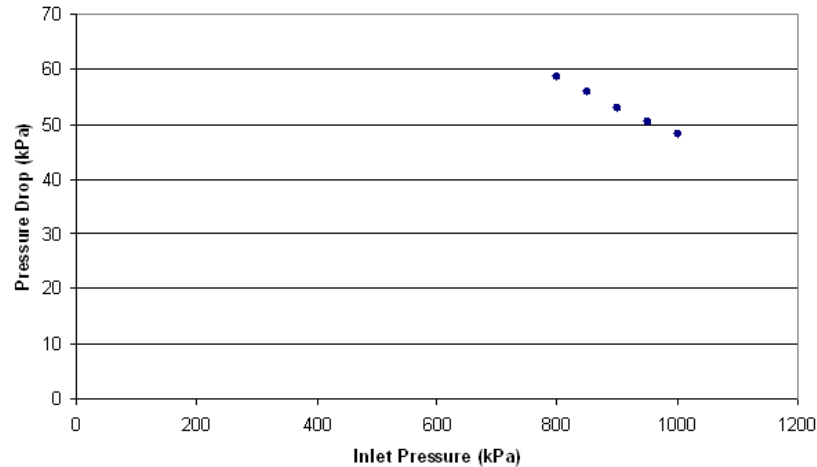


Figure C.4 : Variation of the pressure drop of the indoor unit with respect to the inlet pressure

During the tests, it was found that the multi-split VRF system tried to keep the evaporating pressure at around 1MPa and the total refrigerant mass flow rate was not found more than 0.1kg/s (when divided by seven tube pairs, the refrigerant mass flow rate entering each pair decreases to around 0.014kg/s). Under these conditions, it was assumed that the pressure drop of the indoor coil was lower than 50kPa, which can be neglected.

On the other hand, Figure C.5 shows the outdoor unit tube circuits. Since the total refrigerant mass flow rate provided by the compressor was distributed to 27 with a header at the inlet of the outdoor unit (red rectangular depicts one of the 27 circuits), the pressure drop of the outdoor unit was small. That's why, CoilDesigner evaluation did not perform for the outdoor unit coil.



Figure C.5 : Outdoor unit tube circuits

Appendix D.

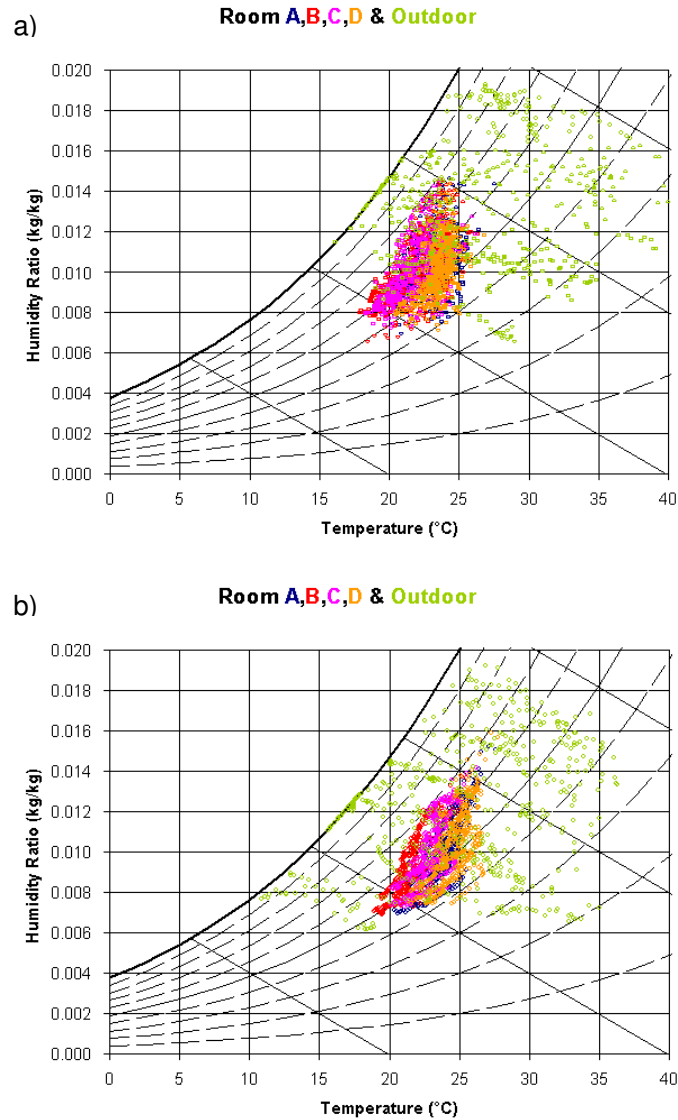


Figure D.1 : Indoor conditions of the System1 zones provided by the multi-split VRF system in master control mode and existing VAV system in central control mode, (a) multi-split VRF system in master control mode, (b) existing VAV system in central control mode

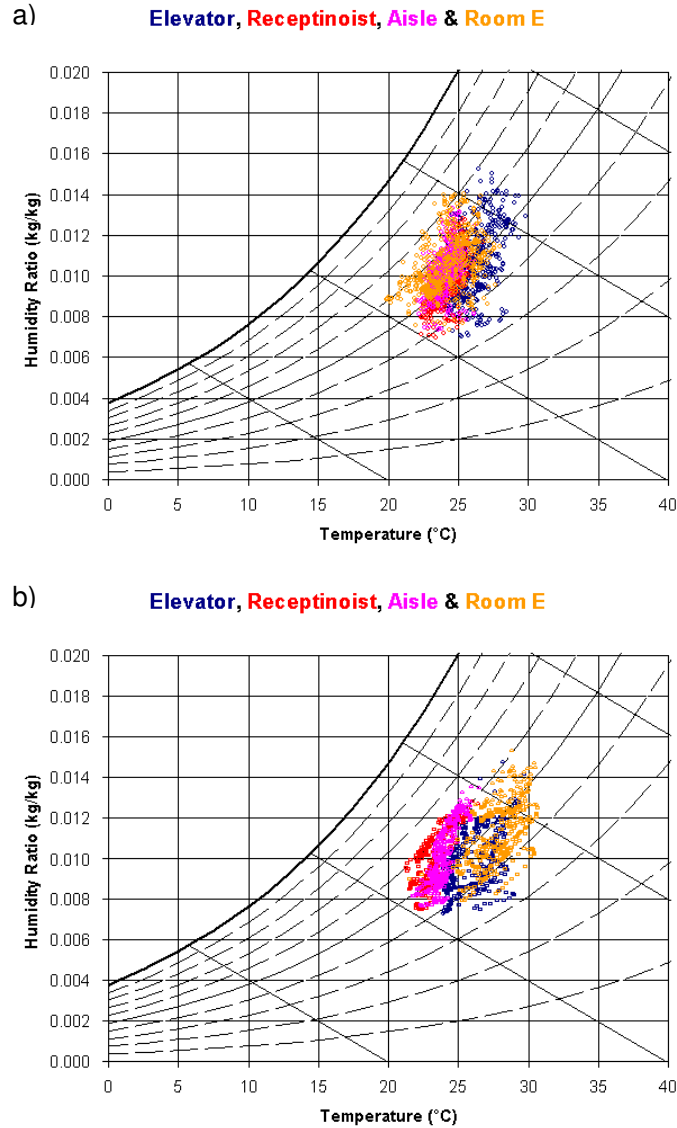


Figure D.2 : Indoor conditions of the System2 zones provided by the multi-split VRF system in master control mode and existing VAV system in central control mode, (a) multi-split VRF system in master control mode, (b) existing VAV system in central control mode

As can be seen from Figure D.1, the indoor conditions of System1 zones were similar for multi-split VRF system in master control mode and existing VAV system in central control mode. Even though, there were some deviations for Room E for System2 zones, it was assumed that the master control mode and the central control mode were similar.

Appendix E.

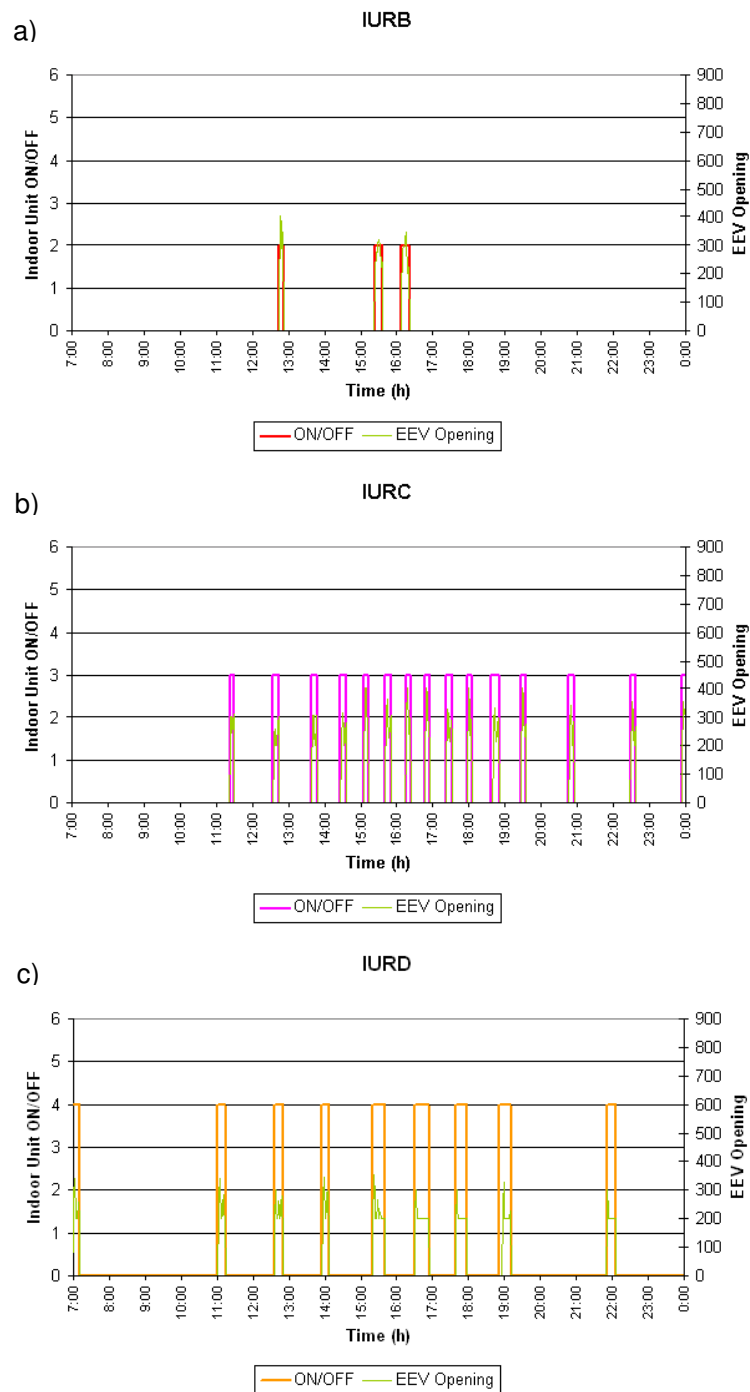


Figure E.1 : Relationship between the indoor unit operation and the EEV opening on June 01, 2007, (a) IURB, (b) IURC, (c) IURD

Appendix F.

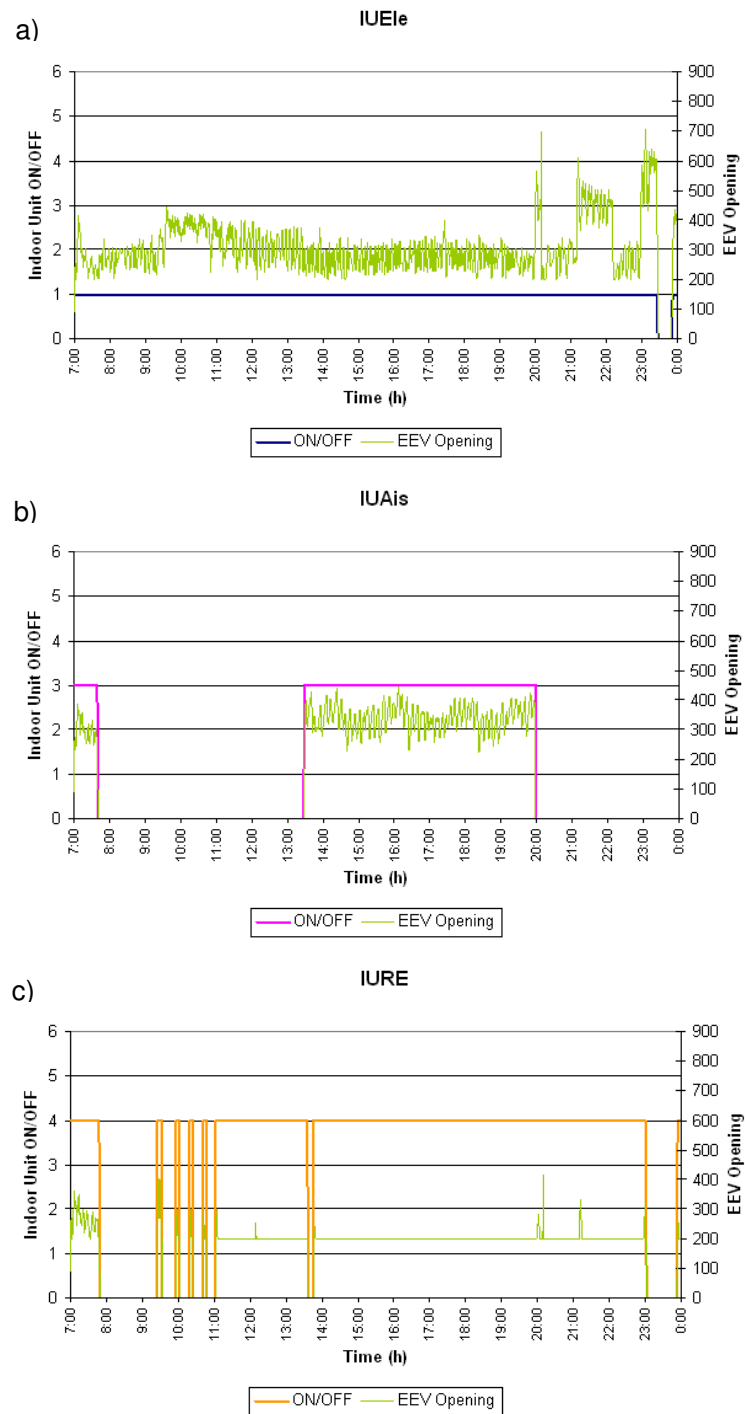


Figure F.1 : Relationship between the indoor unit operation and the EEV opening on June 01, 2007, (a) IUEle, (b) IUAls, (c) IURE

Appendix G.

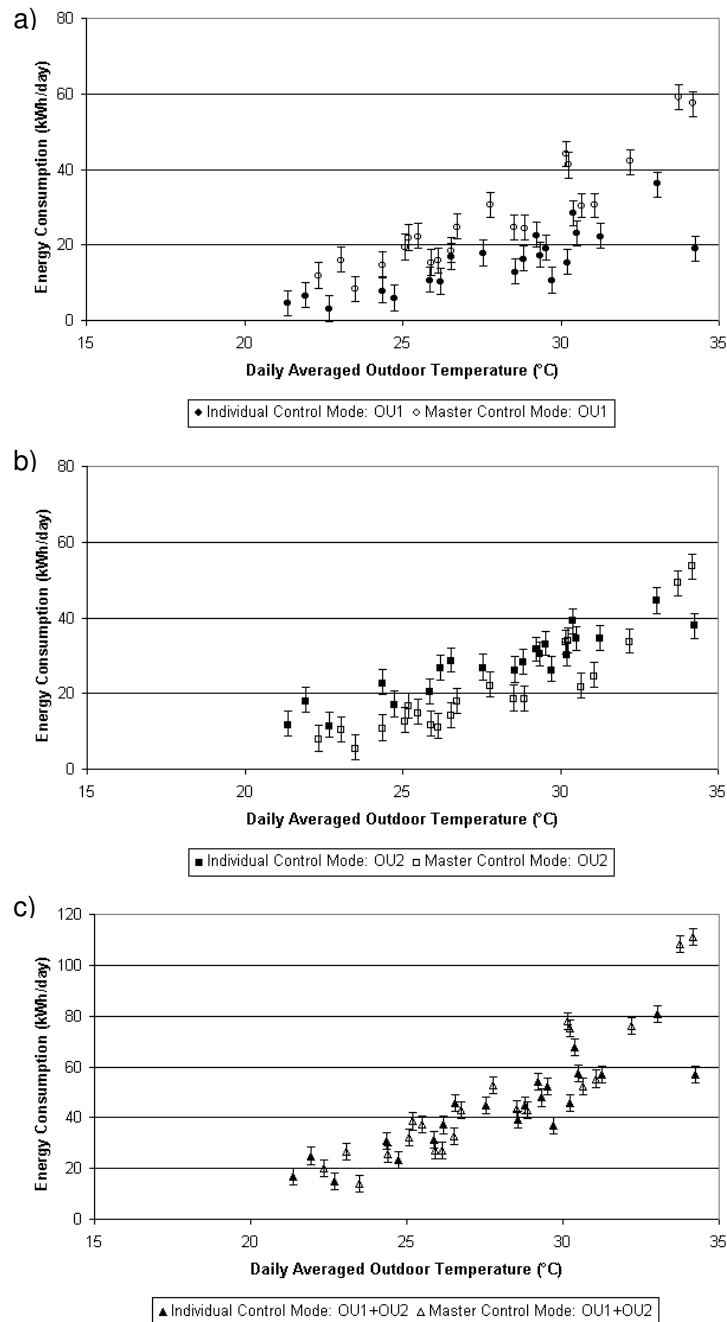


Figure G.1 : Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 25°C set temperature, (a) OU1, (b) OU2, (c) total

Appendix H.

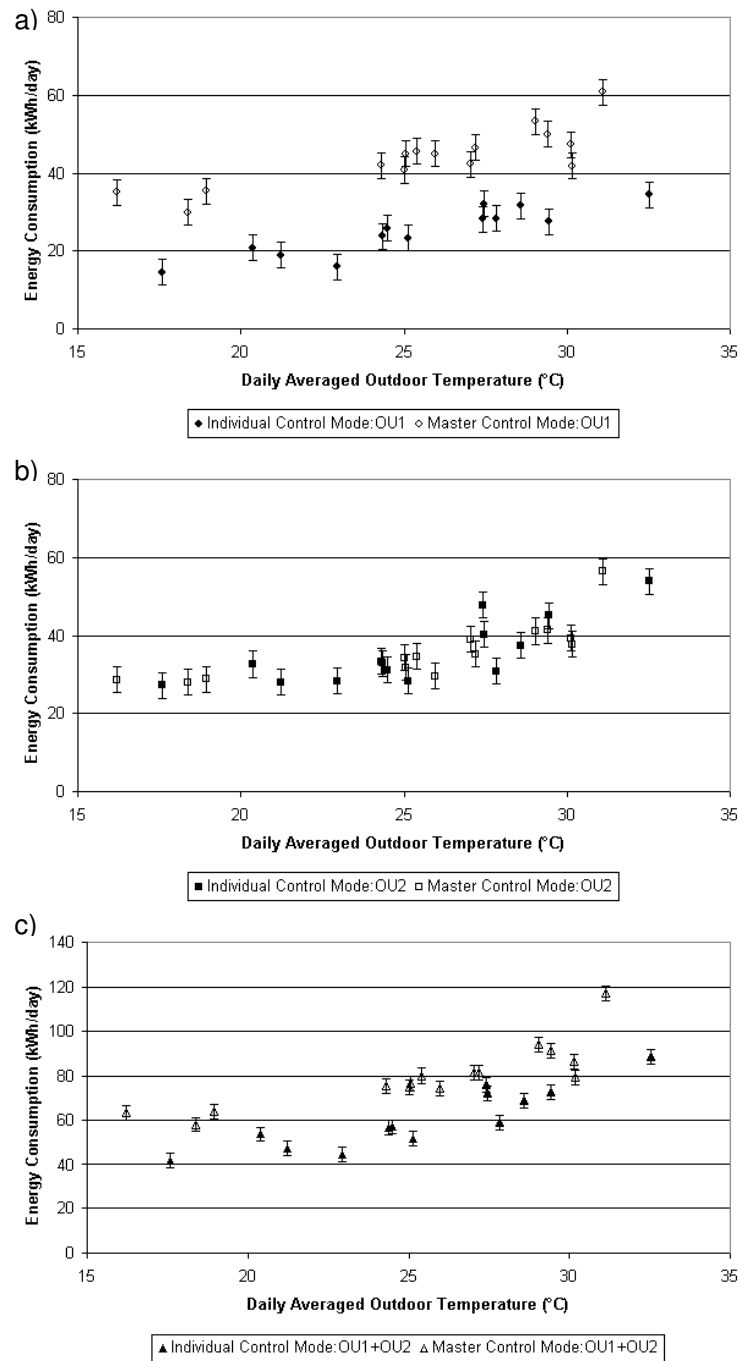


Figure H.1 : Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 20°C set temperature, (a) OU1, (b) OU2, (c) total

Appendix I.

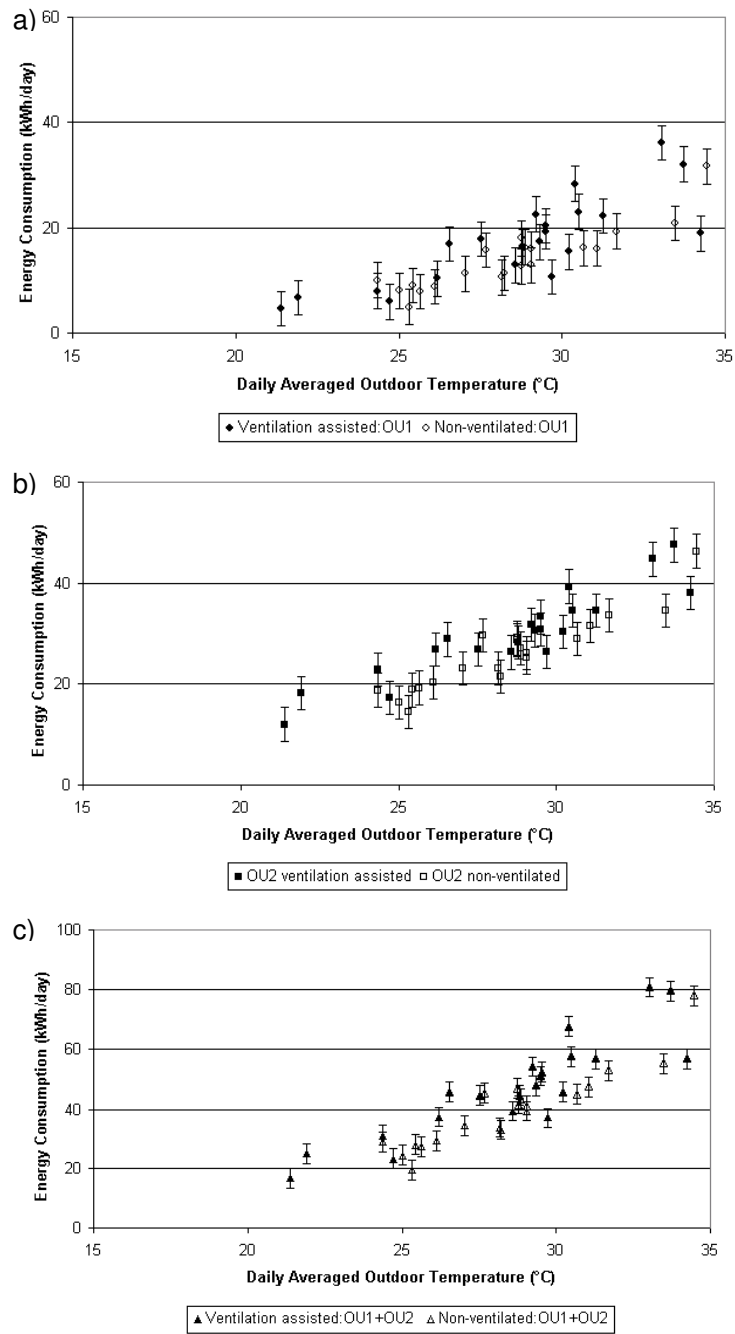


Figure I.1 : Variation of the daily energy consumption for the ventilation assisted and non-ventilated multi-split VRF system with respect to the daily averaged outdoor temperature, (a) OU1, (b) OU2, (c) total

Appendix J.

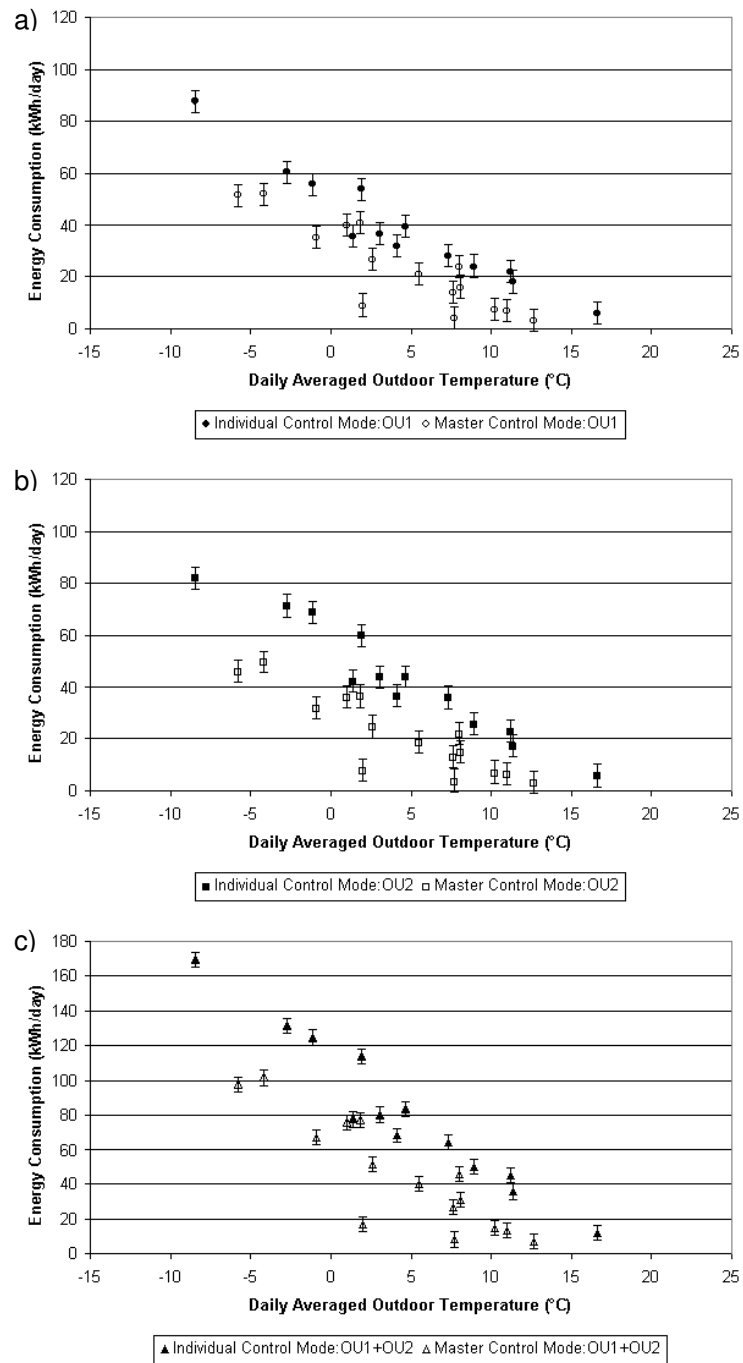


Figure J.1 : Variation of the daily energy consumption of the outdoor units for the individual and master control modes with respect to the daily averaged outdoor temperature at 23.3°C set temperature, (a) OU1, (b) OU2, (c) total

Appendix K.

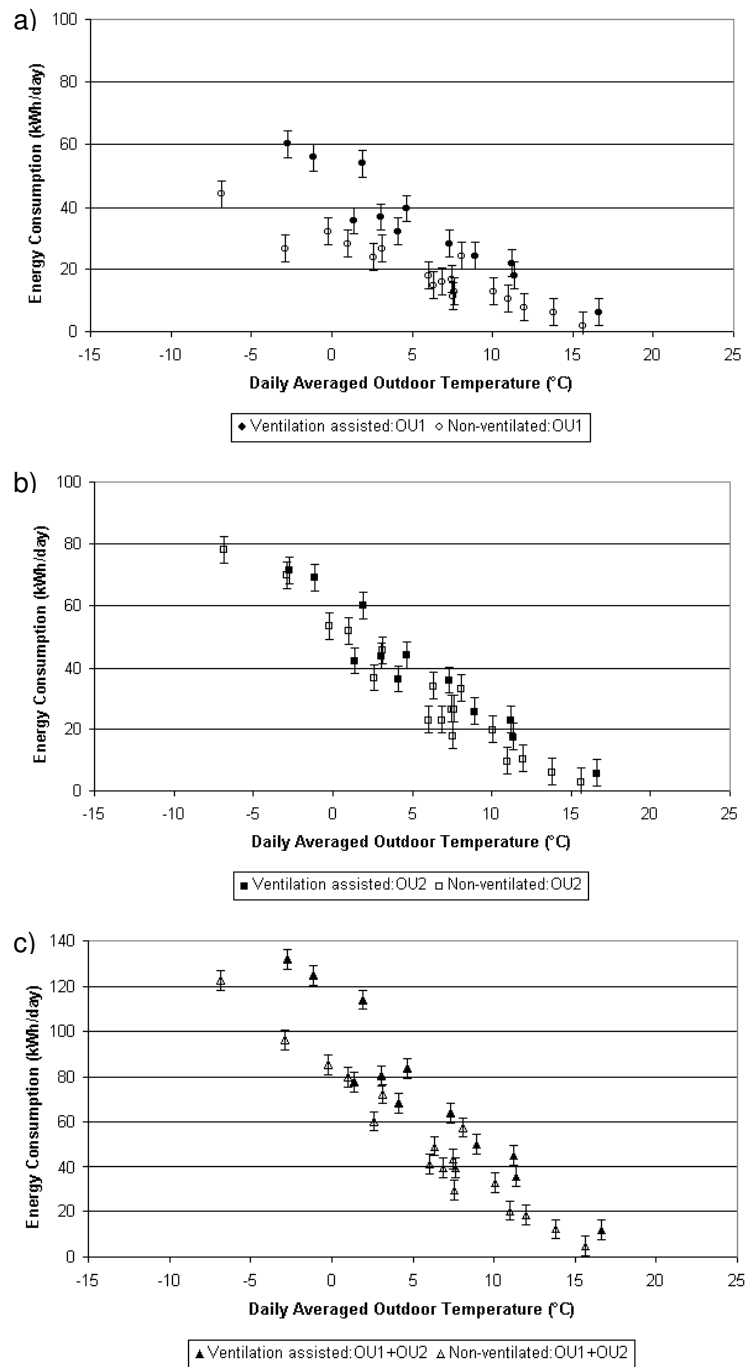


Figure K.1 : Variation of the daily energy consumption for the ventilation assisted and non-ventilated multi-split VRF system with respect to the daily averaged outdoor temperature, (a) OU1, (b) OU2, (c) total

Appendix L.

One power meter was used for the total power measurement of eight indoor units and four HRV units. In order to find out the individual power consumptions of each indoor and HRV unit, one time power measurement was performed by operating the indoor units and HRV units individually.

Figure L.1 shows the individual power consumption of IURA for low and high speed configurations, respectively. As can be seen, when the fan speed configuration changed from “low” to “high”, the power consumption of the indoor unit increased accordingly.

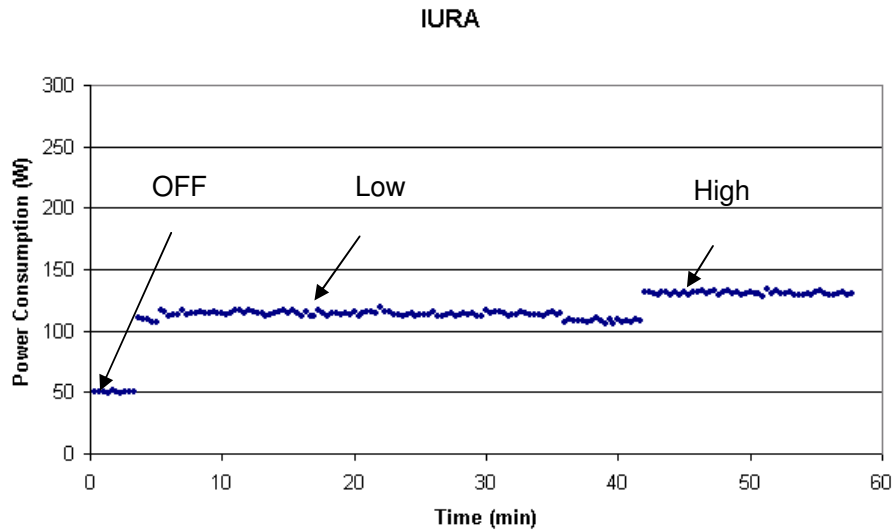


Figure L.1 : Fan power consumption of IURA for low and high fan speed configurations

Around 50W power consumption was monitored from the data acquisition system for the first 5 minutes when IURA was “OFF”. Since all units were connected to one power meter, this power consumption attributed to the total power consumption due to the controllers of the indoor units and HRV units and the wiring losses. This assumption matched with the manufacturer information.

Table L.1 and Table L.2 show the individual power consumption of the indoor units and the HRV units, respectively.

Table L.1 : Individual power consumptions of the HRV units

HRV1	233.6
HRV2	150.9
HRV3	146.7
HRV4	349.6

Table L.2 : Individual power consumptions of the indoor units

	Power Consumption for the Low Speed Configuration (W)	Power Consumption for the High Speed Configuration (W)
IURA	62.6	80.4
IURB	102.5	140.3
IURC	19.6	35.2
IURD	5.2	10.4
IUEle	5.8	10.3
IURec	17.5	30.8
IUAis	10.4	17.3
IURE	18.8	32.1

Figure L.2 shows the total power consumption measurement of the indoor units (operated at the “low” fan speed configuration) and the HRV units throughout a test day.

As can be seen, the total power consumption did not change a lot, and assumed to be constant throughout the test time.

The averaged total power consumption was found 1117.5W, which is 0.5% lower than the summation of the individual power consumption of the eight indoor units (for “low” speed configuration) and four HRV units.

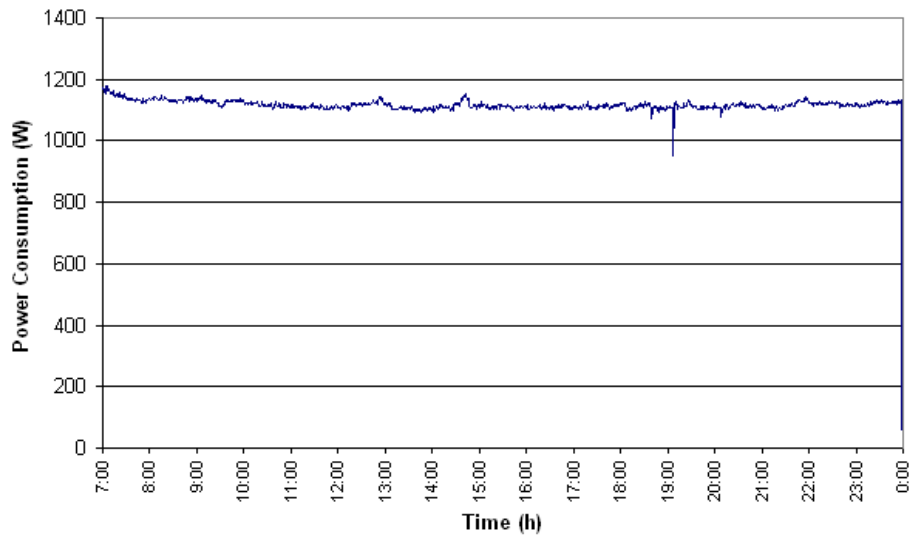


Figure L.2 : Total power consumption of the eight indoor units and four HRV units throughout a test day

CURRICULUM VITEA

Tolga N. Aynur was born in Istanbul in 1978. He graduated from Gazi Anatolia High School in 1996. He received his undergraduate degree from Istanbul Technical University (ITU), Department of Mechanical Engineering in 2000. His graduation rank was 6 out of 150 students. He was accepted for M.Sc. program to the Department of Mechanical Engineering in ITU in 2000. He worked as a researcher at the “Research and Development Center” of ARCELIK A.S. between 2000 and 2002 for his M.Sc. thesis. He earned his M.Sc. degree from ITU in 2002, and from his M.Sc. thesis work, he published an international conference paper and owned a world patent related to the defrosting process. He also earned invention and university-industry cooperation awards from ARCELIK A.S. He was accepted for Ph.D program to the Department of Mechanical Engineering in ITU in 2002. After the qualifying exam, he was accepted as an exchange student to the Department of Mechanical Engineering, Center for Environmental Energy Engineering (CEEE) at the University of Maryland, College Park (UMD) in April 2005, and he conducted research on the multi-split type air conditioning system for his Ph.D. thesis between April 2005 and February 2008. He earned his Ph.D. degree from ITU in February 2008, and from his Ph.D. thesis work, he published three international conference and three journal papers. He is currently participating in another project at CEEE as a faculty research assistant.