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Experimental Investigation of Multi-Functional Variable Refrigerant Flow System

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ABSTRACT

The variable refrigerant flow (VRF) systems have a broader flexibility in control and a wider range of capacity modulation. In this study, a multi-functional VRF (MFVRF) system, which is capable of heat recovery operation and water heating, was experimentally investigated. The MFVRF system could supply space cooling and heating simultaneously in multiple zones, as well as provide hot water. The system's performance during the experiment was measured and discussed. We found that the partial load performance of the system was improved with water heating capability, which increased the daily performance factor (DPF) by 17 percent. We also found that the system performance was enhanced by a heat recovery operation, which could increase the hourly performance factor (HPF) by reducing the pressure differences across the compressor.

1. INTRODUCTION

The VRF systems, or the variable refrigerant volume (VRV) systems were first introduced by Japanese companies in 1980s. It was also called multi-split or multi-evaporator systems. A typical VRF system consists of one outdoor unit (OU), one heat recovery unit (HRU) or mode change unit (MCU), and several indoor units (IU). The VRF system could provide precise control of the system by the electronic expansion valves and the compressor. Moreover, the VRF system is a modular system with better installation flexibility over other air conditioning solutions. Due to this fact, the VRF system has been widely used in commercial buildings.

The VRF systems could be divided into two categories: heat pump VRF systems and heat recovery VRF systems. The heat pump VRF system could operate in either cooling or heating mode. In the summer, the refrigerant flow discharged by the compressor would be cooled in the outdoor heat exchanger before being sent to the heat recovery unit. The refrigerant is distributed by the heat recovery unit and sent to indoor units. In the indoor units, the refrigerant expands through electronic expansion valves (EEV). The refrigerant would then absorb heat and cool down the air in the indoor unit heat exchangers, which act as evaporators. The refrigerant would be sent back to the compressor through the heat recovery unit and then the cycle would be complete. In the winter, the direction of the refrigerant flow is changed by the four-way valves, and the behavior of the system is similar to a heat pump system. The heat pump VRF system has been widely investigated experimentally (Aynur *et al.*, 2008 Hu and Yang, 2005; Tu *et al.*, 2011; Zhou *et al.*, 2008; Kwon *et al.*, 2012). However, the heat pump VRF system could only operate in cooling mode or heating mode compared to the heat recovery VRF system, which is capable of heat recovery within the indoor units. In a heat recovery VRF system, the system has an advanced heat recovery unit which could balance the cooling and heating demand in the indoor units by transferring refrigerant from heating indoor units to cooling units. Therefore, the heat recovery VRF systems support more operating modes and could provide space cooling and heating simultaneously. To experiment with heat recovery VRF systems, Kang *et al.* (2009) investigated the performance of a heat recovery VRF system by adjusting the controlling parameters of the system, such as the compressor speed and the opening of the EEV. They obtained a coefficient of performance (COP) of 7.69 at the entire heat recovery mode, which was 146.5 percent higher than the cooling-only mode. Hai *et al.* (2006) designed and tested a heat recovery VRF system with a nominal 30 kW cooling capacity. They found that the system could work in a temperature range from -18 ℃ to 48 ℃. Moreover, the system could achieve a total capacity as high as 110 percent of the nominal capacity. Joo *et al.* (2011) investigated a heat recovery VRF system consisting of one outdoor unit and four indoor units. Each indoor unit had a cooling capacity of 2.15 kW and the outdoor unit had a condensing capacity of 11.34 kW. The controlling parameters were the fan speed of the outdoor unit, the compressor speed, and the EEV opening. They found that an optimal COP of 6.81 under fully operational mode and a COP of 5.98 when two indoor units were turned off. In this study, they also found that the imbalance between the cooling and heating capacity could undermine the performance of the system.

Compared to the heat pump VRF system, the research in the heat recover VRF system was constrained to laboratory testing and the field performance was rarely discussed. Moreover, the effect of part load performance of the system was limited to the ON and OFF of the units instead of the actual partial load performance of the VRF system under various ambient conditions. Also, the effect of the heat recovery operation on the performance of the system was investigated within only a small range of heat recovery operations, such as having only one unit serve as the heater, and there was not enough research focusing on the reasons of the performance improvement of the system behind the heat recovery operation. The effect of heat recovery operation still needs further investigation. Finally, the heat recovery VRF system that had been investigated in previous work did not consider the necessity of providing hot water to the buildings along with space cooling and heating. In this study, a multifunctional heat recovery VRF system capable of providing space cooling, heating, and hot water was installed in an office building and the field performance of the system was investigated.

2. EXPERIMENT SYSTEM

2.1Experimental set-up

In this study, a MFVRF system was installed in an office building. The system consisted of one outdoor unit, two heat recovery units, seven indoor units, and one hydro kit. R410A was used as working fluid. The outdoor unit was installed on the roof top. The heat recovery units were connected to the outdoor units using three refrigerant pipes: a high pressure gas pipe, a liquid pipe and a low pressure gas pipe. The indoor units and the hydro kit were connected to the heat recovery units. All the indoors units were installed on the third floor of the building, while the hydro kit was installed on the fourth floor. Since one heat recovery unit can support four indoor units maximum, in this study, two heat recovery units were used.

The layout of the part of the system located on the third floor can be seen from Figure 1. Three indoor units: IU1, IU2, IU3, and the hydro kit were connected to the first heat recovery unit. The rest four indoor units: IU 4, IU5, IU 6 and IU 7 were connected to the second heat recovery unit. The indoor units were connected to the heat recover unit using two pipes: a low pressure gas pipe and a liquid pipe.

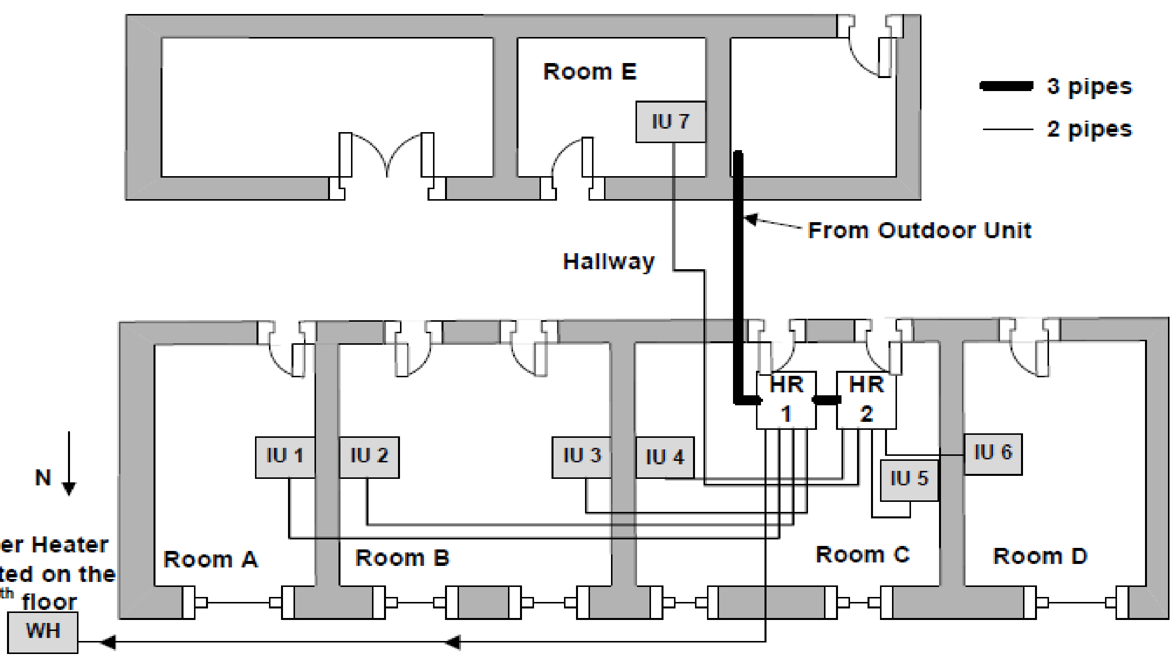


Figure 1: The layout of the system

The specification of the system, including the outdoor units, indoor units, hydro kit, is listed in Table 1.

Table 1: System nominal capacities

Component	Cooling Capacity [kW]	Heating Capacity [kW]
Outdoor Unit	35.2	39.6
Indoor Unit #1,#6,#7	2.2	2.5
Indoor Unit #2,#3	3.6	4.0
Indoor Unit #4,#5	5.6	6.3
Hydro kit	14.1	15.9

The systematic working principle of the MFVRF system in cooling-main mode is shown in Figure 2. In Figure 2, six indoor units were in cooling mode, and only one indoor unit was in heating mode. The plate heat exchanger in the hydro kit was also in heating mode. The outdoor unit consisted of two compressors, two outdoor exchangers, one sub-cooled heat exchanger, two electronic expansion valves, and two four-way valves. The constant speed compressor satisfied the high cooling or heating loads when the invert-driven compressor was not able to satisfy the cooling or heating loads of the system. The four-way valves would adjust the flow direction of the system under different operation modes.

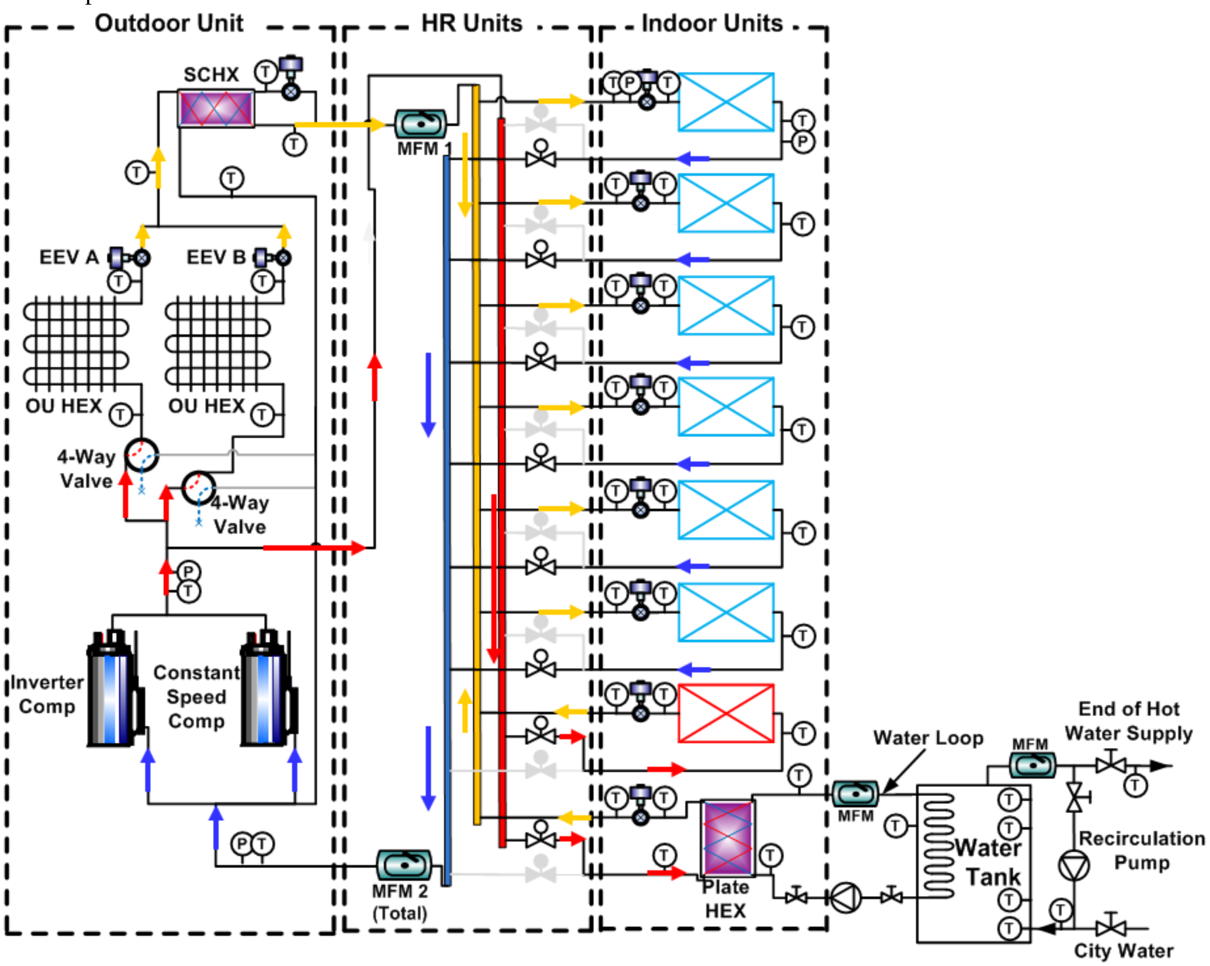


Figure 2: Systematic diagram in cooling-main mode (Kwon, 2013)

In Figure 2, the four-way valves were controlled so part of the refrigerant leaving the system would go to the outdoor heat and part of the refrigerant would bypass the outdoor heat exchangers. The refrigerant was cooled down in the outdoor exchangers by the ambient air. The expansion valves in the outdoor units were opened fully in this case. Therefore, the refrigerant passed through fully-opened expansion valves prior to entering the sub-cooled heat exchanger. In the sub-cooled heat exchanger, part of the refrigerant separated from the mainstream of the flow. The sub-cooling degrees of the refrigerant could be adjusted by opening the expansion valves of separated flow.

The refrigerant leaving the outdoor unit was distributed by the heat recover units. The heat recovery units were made of refrigerant pipes and solenoid valves, which altered the direction of the refrigerant flow entering the indoor heat exchangers. The high pressure gas leaving the compressor passed to the heating units and the hydro kit plate heat exchangers through the high pressure gas pipe. After releasing heat to the indoor air and water, the refrigerant left the heating units, and the plate heat exchanger merged with the sub-cooled refrigerant in the liquid pipe. The refrigerant in the liquid pipe was passed to the cooling indoor units, and the heat exchangers in the cooling indoor units were evaporators for the refrigerant.

In this MFVRF system, the water in the water first absorbed the heat from the refrigerant in the plate heat exchanger. Then, it heated the water in the tank through copper tubes. A recirculation pump was installed to improve the temperature distribution in the tank. Finally, an actuator was used to drain the hot water to the sewage according to the hot water demand.

2.2 Measurement

The refrigerant-side temperature of the system was measured by T-type thermocouples that had been installed on the surface of the refrigerant pipes. The inlet air-side temperature of the outdoor unit was measured by eight T-type thermocouples on the surface of the heat exchangers. The outlet air–side temperature was measured by the thermocouples installed on the surface of the outdoor unit fan. The humidity of the air was measured by relative humidity sensors. In order to avoid the effects of radiation and rain, the exposed thermocouples and relative humidity sensors were all shielded. The inlet air temperature and relative humidity of the indoor unit were measured by a T-type thermocouple and a relative humidity sensor that had been installed at the inlet of the indoor unit. Similarly, the temperature and relative humidity of the air leaving the indoor heat exchanger were measured by a thermocouple and a relative humidity sensor installed at the outlet port. The temperature was measured by in-stream thermocouples in the hydro kit. The water temperature within the tank was measured by six vertically inserted thermocouple probes.

The pressure of the system was measured by both the built-in sensors in the compressor and by the two pressure sensors at the indoor units. The built-in sensors in the compressor measured the suction and discharging pressures. The pressure sensors at the indoor units measured the pressure before the EEV and after the indoor heat exchanger.

As shown in Figure 2, the mass flow rate entering the heat recovery units were measured by two mass flow meters located in the liquid line and the low pressure gas line. Both were Coriolis mass flow meters. The mass flow rates of the indoor units and hydro kit were measured with the correlation equation of EEV. The water flow rate of the plate heat exchanger in the hydro kit was measured by the turbine flow meter installed in the main water loop. Similarly, a turbine flow meter was installed in the draining pipe of the water tank.

The power consumption of the systems were measured by two watt meters: one watt meter for outdoor units, and one watt meter for the rest of the system. The watt meter for the outdoor unit measured the power consumption of the compressor and the fans of the outdoor heat exchangers. The watt meter for the indoor units measured the power consumption of the all indoors units, hydro kit, heat recovery units, and other features of the VRF system.

2.3 Test Conditions

The system was installed in a building in College Park, Maryland. The tests were conducted during the winter. The types of system tests in this study are listed in Table 3. The hot water consumption considered in this study was the typical residential hot water defined in the DOE energy data book (DOE 2011). The setting temperature for the water tank was 50 ℃. The set point for the heating units was 20 ℃, and the set point for the cooling units was 27 ℃.

Table 2: Test conditions

Test mode	Cooling Units	Heating Units	Daily Hot Water Consumption: liters/day	Ratio of Nominal Cooling Capacity to Nominal Heating Capacity
Case 1	IU #7	IU #1 #2 #3 #4 #5 #6	0	0.09
Case 2	IU #7	IU #1 #2 #3 #4 #5 #6	738	0.05
Case 3	IU #4 #5	IU #2 #3 #6 #7	0	0.39
Case 4	None	IU #2 #3 #4 #5 #6 #7	0	0

2.4 Data Reduction

The two mass flow meters measured the mass flow rate of the refrigerant as it entered the heat recovery units. The heat recovery units then distributed the refrigerant among the components according to the operation mode. The total mass flow rate during the heating main mode was calculated in Equation (1) and (2) for the cooling and heating mass flow.

$$\dot{m}_{total,heating} = \dot{m}_2 + \dot{m}_1 \quad (1)$$

$$\dot{m}_{total,cooling} = \dot{m}_2 \quad (2)$$

In Equation (1) and (2), $\dot{m}_1$ is the mass flow rate measured in the liquid line and $\dot{m}_2$ is the mass flow rate measured in the low pressure gas line. $\dot{m}_{total,heating}$ and $\dot{m}_{total,cooling}$ are the total mass flow rates of heating and cooling units, respectively. However, since it was too expensive to install a mass flow meter for each of the indoor units, the total refrigerant flow rate of either cooling or heating needs to be divided to obtain the mass flow rate within each operating units. The method of calculating mass flow rate of each unit was calculated in Equation (3) and (4).

$$\dot{R}_{total,heating} = C_d A\sqrt{2\Delta P\rho} \quad (3)$$

$$\dot{m}_i = \frac{R_i}{\sum_{i=1}^{n} R_i} m_{total} \quad (4)$$

R_i is the calculated mass flow rate by the EEV correlation equation which consists of the flow efficient (C_d), the area of EEV (A), the pressure drop across the EEV (ΔP) and the density at the inlet of EEV (ρ). The individual mass flow rate is calculated by Equation (4) where m_{total} is the total mass flow rate for either cooling or heating obtained from Equation (1) and (2). n is the number of cooling or heating units.

The approach to calculating the individual and total capacity for heating and cooling indoor units is shown in Equation (5), (6) and (7).

$$\dot{Q}_{cooling/heating} = \dot{m}_i \Delta h \quad (5)$$

$$\dot{Q}_{cooling,total} = \sum_{i=1}^{k} \dot{Q}_{cooling,i} \quad (6)$$

$$\dot{Q}_{heating,total} = \sum_{i=1}^{m} \dot{Q}_{heating,i} \quad (7)$$

$\dot{Q}_{cooling/heating}$ is the cooling or heating capacity of the indoor units. Δh is the enthalpy difference of refrigerants across the indoor unit. $\dot{Q}_{cooling,total}$ and $\dot{Q}_{heating,total}$ are the calculated total capacities of cooling and heating indoor units of the system. The capacity of the hydro kit is calculated according to the water side temperature difference rather than the refrigerant side. It is given in Equation (8).

$$\dot{Q}_{hydrokit} = \dot{m}_w C_p (T_{w,out} - T_{w,in}) \tag{8}$$

$\dot{Q}_{hydrokit}$ is the calculated hydro kit capacity. $\dot{m}_w$ and C_p are the flow rate and specific heat capacity of the water, respectively. $T_{w,out}$ and $T_{w,in}$ are the water outlet and inlet temperature of the plate heat exchanger, respectively.

The total cooling capacity, heating capacity, and hydro kit capacity are used to evaluate the PLR of the system. The PLR is defined in Equation (9).

$$PLR = \frac{\dot{Q}_{heating,total} + \dot{Q}_{cooling,total} + \dot{Q}_{waterheater}}{\dot{Q}_{heating,rated}} \tag{9}$$

$\dot{Q}_{heating,rated}$ is the rated heating capacity of the outdoor unit.

The daily performance factor (DPF) and the hourly performance factor (HPF) of the system is evaluated by Equation (10) and (11).

$$DPF = \frac{\sum \dot{Q}_{heating,total} t + \sum \dot{Q}_{cooling,total} t + \sum \dot{Q}_{hydrokit} t}{\sum (P_{OU} + P_{IU} + P_{pump}) t} \tag{10}$$

$$HPF = \frac{\sum \dot{Q}_{heating,total} + \sum \dot{Q}_{cooling,total} + \sum \dot{Q}_{hydrokit}}{\sum (P_{OU} + P_{IU} + P_{pump})} \tag{11}$$

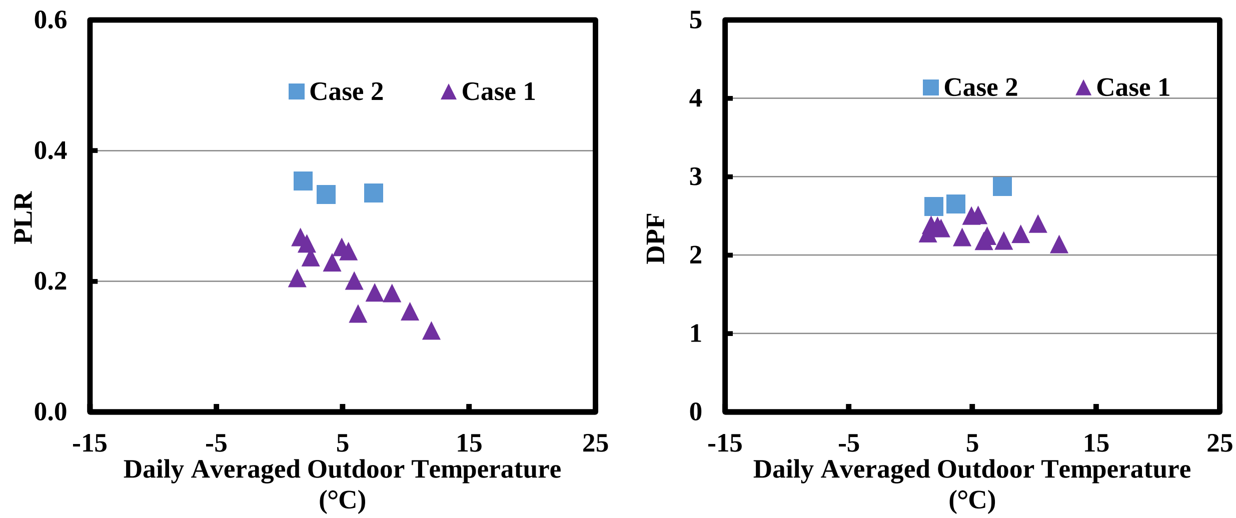
P_{total} is the power consumption of the whole system, which is made up of the power consumption of the outdoor units (P_{OU}), indoor units (P_{IU}), and the pump in the hydro kit (P_{pump}). t is the duration of testing.

3. RESULTS AND DISCUSSION

3.1 Effect of hot water

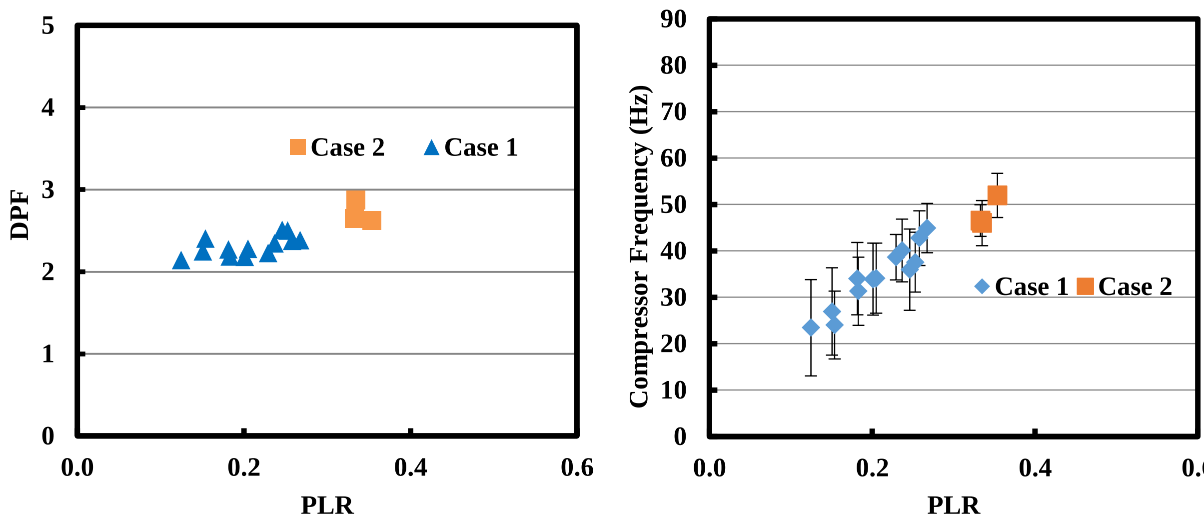
Figure 3 shows the effect of the hot water demand on the operation of the system. Figure 3(a) shows the variation of the PLR and the ambient temperature with and without hot water demand. Similarly, Figure 3(b) shows the change in the system's performance with and without hot water demand.

The PLR of the system decreased when the ambient temperature increased from -5 °to 15 ℃. This is caused by the decreased heating demand of the indoor units. When the ambient temperature was above 5 ℃, the PLR of the system was below 0.2, which decreased the frequency of the compressor and caused unnecessary cycling loss in the compressor. The power consumption of the outdoor unit increased due to the degraded performance of the compressor. With the hot water demand, the influence of the decreased indoor capacity was partially offset by the hydro kit. Therefore, the PLR of the system was considerably more stable. Therefore, in Figure 3(b), it was found that the daily performance of the system improved due to the hot water demand. In Figure 4, the compressor frequency and the daily performance factor in respect to the PLR are shown. Figure 4(a) shows the improvement of daily performance factor with the hydro kit. Figure 4(b) shows both the average compressor frequency and the standard deviation.



(a) PLR (b) DPF

Figure 3: PLR and DPF with respect to the average outdoor temperature



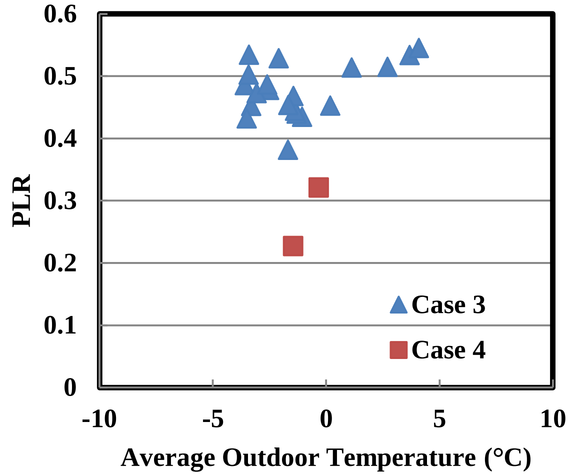
(a) DPF (b) Compressor frequency

Figure 4: Compressor performance and DPF with respect to PLR.

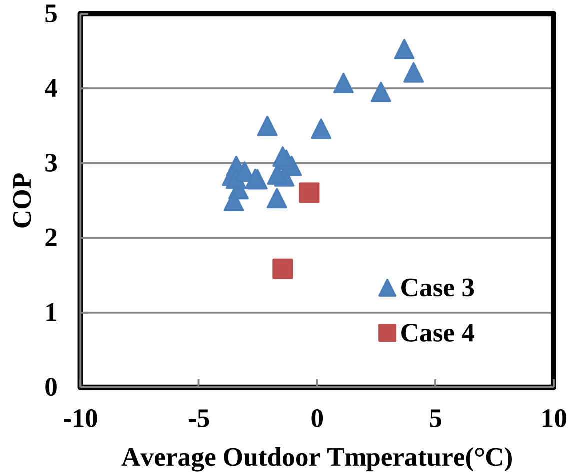
Figure 4(a) shows that without the hot water demand, the PLR of the system was below 0.3 and the daily performance factor was below 2.5. When the PLR of the system was enhanced to 0.4, the daily performance factor also increased. Meanwhile, Figure 4(b) indicates that when the PLR was below 0.3, the standard deviation of the measured compressor frequency increased. A higher fluctuation of the compressor frequency under a low PLR resulted in decreased compressor performance and a lower inverter efficiency. The performance of the system would decline if the power consumption was increased in the outdoor section of the system.

3.2 Effect of heat recovery operation

The effect of the heat recovery operation is shown in Figure (5). Since this study focused on the field testing of an MFVRF system, the ambient conditions, such as temperature, would change during the testing. A daily average performance analysis could not completely show the transient performance of the system under heat recovery mode. Therefore, the hourly based data was used to illustrate the benefit of the heat recovery operation. Figure 5(a) shows the variation of the PLR of the system with the ambient temperature. Figure 5(b) shows the variation of HPF with respect to the ambient temperature. Both the PLR and HPF of the system in Case 3 are higher than the PLR and HPF of the system in Case 4.

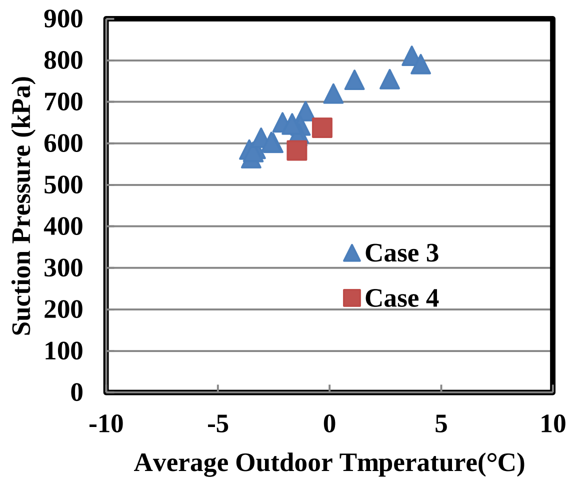


(a) PLR

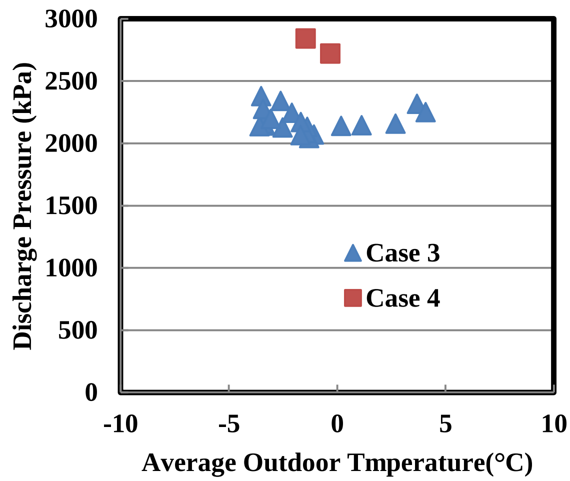


(b) HPF

Figure 5: PLR and HPF of the system with respect to the ambient temperature



(a) Suction Pressure



(b) Discharge Pressure

Figure 6: Suction and discharge pressure of the system with respect to ambient temperature

As can be found from Figure 5(a), when the ambient temperature was around 0 ℃, compared to a pure heating system under Case 4, the PLR of Case 3 was elevated from 0.3 to 0.4. Accordingly, in Figure 5(b), the HPF of the system also increased to 3.5. This is because the room where indoor unit #4 and #5 were installed had a relatively higher internal load compared to the other rooms due to a large amount of electric equipment and frequent human activity. If the units were set to heating mode, then the "ON" period of the units was short. However, when set to cooling mode, the units stayed in the "ON" mode for a longer period of time. Therefore, the system performed better in Case 3. This is further illustrated in Figure 6(a) and 6(b). Figure 6(a) and Figure 6(b) show how the discharge and suction pressure of the system changed in Case 3 and Case 4. The discharge pressure in Case 3 was lower than in Case 4, however the suction pressure was higher.

When the system switched from heating mode to a heating main mode, some of the indoor units acted as evaporators rather than condensers. In the heating main mode, the refrigerant leaving all the heating indoor units was distributed by the heat recovery units before entering the cooling indoor units. When compared to the pure heating mode, the evaporating capacity of the system was separated into two parts: the first part was attributed to the capacity of the outdoor heat exchanger, and the second part was attributed to the capacity of the indoor units in cooling mode. Since the room temperature was higher than the ambient temperature, the evaporating temperature of the system increased. In Figure 6(a), the suction pressure under 0 ℃ increased from 630 to 720 kPa during the mode switch, as did the ambient temperature. Moreover, when compared to the pure heating mode, the compressor frequency dropped to balance the whole mass flow rate of refrigerant of the system, causing a decreased discharge pressure. The discharge pressure of the system dropped to 2100 kPa when the ambient temperature was around 0 ℃. With a decreased discharge pressure and increased suction pressure, the power consumption of the compressor was reduced due to a lower pressure ratio across the compressor.

4. CONCLUSIONS

In this study, a MFVRF with the capability of providing cooling, heating, and hot water simultaneously was installed in an office building. The performance of the system was experimentally measured and investigated to see the impact of heat recovery operations and hot water demand. We found that with the water heating capability, the PLR of the system was more stable, leading to a better performance of the compressor. The performance of the system increased by 17 percent, and the PLR increased from 0.2 to 0.34. The effect of the heat recovery operation was also investigated. We found that with the heat recovery operation, the PLR of the system increased. The HPF of the system increased as well, due to decreased discharge pressure and increased suction pressure, which both contribute to the improvement of the compressor performance. When the ambient temperature was near 0 ℃, the HPF of the system is increased from 2.6 to 3.4.

NOMENCLATURE

VRF	variable refrigerant flow
MFVRF	multifunctional variable refrigerant flow
RH	relative humidity
EEV	electronic expansion valve
PLR	part load ratio
DPF	daily performance factor
IU	indoor units
OU	outdoor unit
HR	heat recovery
HRU	heat recovery unit
HEX	heat exchanger
HPF	hourly performance factor
SCHX	sub-cooled heat exchanger
CER	cooling energy ratio
m_{total}	total mass flow rate of cooling or heating
R_i	calculated mass flow rate according to correlation of expansion valve
ΔP	pressure difference through expansion valve
ρ	density at the inlet of expansion valve
C_d	flow coefficient
Δh	enthalpy difference across the indoor unit
$\dot{Q}_{cooling/heating}$	cooling or heating capacity of indoor unit
$\dot{Q}_{cooling,total}$	total cooling capacity
$\dot{Q}_{cooling,total}$	total heating capacitu
$T_{w,in}$	water inlet temperature of plate heat exchanger

$T_{w,out}$	water outlet of plate heat exchanger
t	testing duration
P_{total}	total power consumption of system
$\dot{Q}_{hydrokit}$	capacity of hydro kit

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