Performance evaluation of the variable refrigerant flow (VRF) air-conditioning system subjected to partial loadings at different outdoor air temperatures

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Abstract
This paper shows the performance evaluation of the variable refrigerant flow (VRF) air-conditioning system at different outdoor air temperatures during cooling and heating operations. The purpose of this study is to determine the real performance of the system when operating at different outdoor temperatures. The one outdoor unit’s and two indoor units’ VRF system was used as a test specimen in the controlled testing chambers in which temperature and humidity were controlled. Several test cases were done covering from partial to full thermal loadings for the different outdoor air temperatures. The results showed that the outdoor air temperature much affected the performance of the VRF air-conditioning system. It showed that during the cooling operation, as the outdoor air dry bulb temperature increases, the coefficient of the performance ratio decreases. In the heating operation, as the outdoor air dry bulb with corresponding wet bulb temperature decreases, the coefficient of the performance ratio decreases. The defrosting mode, which occurs during the heating operation, affects the indoor chamber’s temperature as the air temperature becomes very low due to the indoor units having become an evaporator. The results of the study showed the importance of the VRF air-conditioning system performances at different outdoor air dry bulb and wet bulb temperatures. Hence, it is important to consider the outdoor air temperatures when calculating the energy consumption of the VRF air-conditioning system to be installed in new and/or retrofitted buildings as they affect the electric energy consumption.

Key words: Air-conditioning system, Variable refrigerant flow, Thermal loading, Energy consumption

1. Introduction

The maintenance of the buildings’ indoor comfortable thermal environment and air quality at any time of the day and any month of the year is a very important and an energy intensive operation. For example, in the hot climate of the Middle East, the operation of the air-conditioning system to provide a comfortable indoor thermal environment consumes a large percentage of the energy consumption of the buildings. In hot and humid climates, such as in South and Southeast Asia, the application of an air-conditioning system to maintain a comfortable indoor thermal environment is always needed to avoid the effect of outdoor air high temperature.
and humidity. In a temperate climate, such as in Japan, the application of an air-conditioning system to provide a thermally comfortable indoor environment both summer and winter seasons is very important.

The VRF air-conditioning system is becoming popular globally due to its flexibility of operation to cater to the different indoor thermal environment situations – partial loading, unbalanced loading and so on (Goetzler, 2007). Generally, the performance rating of the VRF air-conditioning system is based on the evaluation standards such as the Japan Industrial Standards (JIS, 2015). In the evaluation standards, the system is operated in steady and controlled conditions with specified loadings and air conditions. In reality, the operations of the VRF air-conditioning systems installed in buildings may be at partial thermal loadings according to the changing outdoor air temperatures since the system is operating from either early morning to late afternoon and from either beginning to end of either summer or winter.

The VRF air-conditioning system utilizes the variable speed electric motor using an inverter to drive the compressor. The advantage of using the variable speed drive compared to a constant drive is that the refrigerant flow rate varies depending on the thermal loading. Coupled with the electronic expansion valve (EEV) to regulate the refrigerant flow, the indoor temperature setting can be uniformly maintained. As the compressor speed varies due to the change of the electric motor speed, the electric power consumption varies as well. Hence, by using the variable speed operated compressor, the electric power consumption is lowered compared to the constant speed, particularly for a partial thermal load. As the application and usage of the VRF air-conditioning system increases, the more accurate estimate of the actual electric energy consumption becomes important during the design phase of new buildings or retrofitted buildings to estimate the building energy consumption (Building Energy Standard of Japan, 2013). Hence, a performance evaluation of the VRF air-conditioning system subjected to different outdoor air temperatures while operating in different thermal loading ratios is needed. Based on the review of previous studies, the evaluation of the system based on different outdoor air temperatures and loading ratios is not yet fully completed (Enteria et al., 2016; Horie and Hihara, 2012; Watanabe et al., 2009; Watanabe et al., 2007).

This paper shows the evaluation of the VRF air-conditioning system with one outdoor unit and two indoor units operated under different outdoor air temperatures for both cooling and heating operations. The system is operated from partial to full thermal loadings under the cooling and heating operations. The results of this study are important in the determination of the actual VRF air-conditioning system performances. Also, the results of the study are needed for a more accurate determination of the electric consumption of the system in the evaluation of the energy consumption of the new or retrofitted buildings.

2. Methodology

2.1 Test facilities

Figure 1a shows the technical diagram of the testing facilities used to make the performance evaluation of the VRF air-conditioning system shown in this paper. Figure 1b shows the actual view of the testing facilities. The testing facilities consist of one outdoor chamber and three indoor chambers. The outdoor chamber can support the system with an outdoor capacity of from 0.33kW to 56kW. The outdoor chamber dry bulb temperature can be varied from -20°C to 45°C. The relative humidity range can be varied from 30% to 80% controlled by using a wet bulb temperature sensor for the outdoor dry bulb temperature when it is above water’s freezing point. The three indoor chambers can simulate both cooling and heating loads. They can support a heating load up to 12 kW with humidification capability. The indoor chamber can support a cooling load up to 12.5kW. The indoor chambers 1 and 2 are separated. In the indoor chamber 3, the two indoor units can be installed in tandem. In this study, indoor chambers 1 and 2 are used. Control and monitoring devices are installed to change the independent parameters, and to monitor these parameters and store them.

2.2 Test specimen

Figure 2 shows the schematic diagram of the VRF air-conditioning system using R410A as the refrigerant and subjected to measurement in controlled chambers. The outdoor unit is installed in the outdoor controlled
Fig. 1.  Testing controlled chambers:  a) schematic diagram, and;  b) actual view.

chamber (See Fig. 1) in which the air temperature and humidity are set to prescribed values. The indoor units are installed in the indoor chambers (See Fig. 1) in which air temperature and humidity are controlled based on the required values. The VRF air-conditioning system’s outdoor and indoor units are connected using the specified refrigerant piping and insulation. There are several sensors attached to the system, such as the electric power meter to measure the electric power consumption of the outdoor and indoor units. Refrigerant flow meters are installed in the refrigerant piping to measure the refrigerant flow rates as presented in the diagram. Refrigerant absolute pressure sensors are installed in the different locations in the refrigerant piping network to measure the pressure of the refrigerant. Refrigerant temperature sensors attached to the surface of the tube covered with insulation measure the different temperatures of the refrigerant as shown in the diagram. Air temperatures and humidity sensors are installed both in the outdoor unit and indoor units to measure the air properties. Rotation sensors are installed for the measurement of rotation of the compressor, outdoor air fan and indoor air fans. Table 1 shows the rated values of the VRF air-conditioning system used in this study. The presented values are based on the catalogue values for the system provided by the manufacturer.

2.3 Test cases

Table 2 shows the conditions for test cases used to measure the performance of the VRF air-conditioning
Enteria, Yamaguchi, Miyata, Sawachi and Kuwasawa, Journal of Thermal Science and Technology, Vol.11, No.2 (2016)

Table 1 Specified values of the variable refrigerant flow air-conditioning system in the study.

| Operation Mode | Rated Electric Consumption, kW | Rated Capacity, kW |
|----------------|-------------------------------|-------------------|
| Cooling        | 6.61                          | 22.4              |
| Heating        | 6.43                          | 25                |

Table 2 Test cases for the measurement of the single outdoor unit and double indoor units’ variable flow air-conditioning system.

| Operation Mode | Outdoor Conditions, °C (Outdoor Chamber) | Indoor Conditions, °C (Indoor Chamber) | Percentage Loading, % (Respect to Rated Load, Table 1) | Load Balance Ratio, $\alpha$ (Ratio of Indoor Units Load) |
|----------------|-------------------------------------------|---------------------------------------|--------------------------------------------------------|--------------------------------------------------------|
| Cooling        | 35 °C DBT                                 | 27 °C DBT, 19 °C WBT                  | 100, 75, 50, 40, 30, 20, 10, 5                          | 1                                                      |
|                | 40 °C DBT                                 |                                       | 100, 50, 35, 20, 10                                     | 1                                                      |
|                | 45 °C DBT                                 |                                       | 100, 50, 35, 20, 10                                     | 1                                                      |
| Heating        | 7 °C DBT & 6 °C WBT                       |                                       | 100, 75, 50, 40, 30, 20, 10, 5                          | 1                                                      |
|                | 2 °C DBT & 1 °C WBT                       | 20 °C DBT                             | 100, 75, 50, 40, 30, 20, 10, 5                          | 1                                                      |
|                | -7 °C DBT & -8 °C WBT                     |                                       | 100, 75, 50, 40, 30, 20, 10, 5                          | 1                                                      |

system when subjected to different outdoor air temperatures. The outdoor and indoor conditions are based on the JIS measurement standard (JIS, 1999). Aside from the JIS measurement standard for the outdoor temperatures, additional outdoor air temperatures are added as presented in the table. The 100% loading is the rated cooling or heating capacity shown in Table 1. For example, the 50% loading is half of the rated thermal capacity. The load balance ratio ($\alpha$) proposed and introduced is shown in Eq. (1) (Enteria et al., 2016). In this measurement evaluation, $\alpha$ is equal to 1, the indoor chamber 1 load is equal to the indoor chamber 2 load.

$$\alpha = \frac{Q_{ic_1}}{Q_{ic_2}} = 1 ; \quad Q_{ic_1} = Q_{ic_2}$$

Fig. 2. Schematic diagram of the single outdoor unit and double indoor units’ variable flow air-conditioning system.
2.4 Test evaluation

Figure 3 shows the representative different compressor operating scenarios observed based on the VRF air-conditioning system operation. Figure 3a shows the steady on operation in which case the compressor is operating continuously. In this case, the compressor might be operating at a constant speed or at a sinusoidal or variable speed depending on the thermal load conditions and the system. Figure 3b is the on-off or intermittent operation which occurs at a very low thermal loading. The on and off operation span much upon what the loading amount will be. Figure 3c represents the on-defrosting during the heating operation of the system. The defrosting mode kicks in from the signal of the system when a large accumulation of ice occurs in the outdoor unit heat exchanger, which hampers the operation of the system. As observed in the system operation, the time span for the defrosting mode is the same for different heating load ratios when the defrosting mode kicks in. However, the heating mode time is different, as it depends on the heating load ratio and the amount of moisture in the air that causes the accumulation of ice in the outdoor heat exchanger.

The calculation of the different thermal capacity is based on the moist air enthalpy difference between the return air and the supply air in the indoor unit, multiplied by the air flow rate. The air flow rate calculation is based on the obtained correlation of the indoor units’ air fan speed to the air volumetric flow rates from a separate measurement. The electric energy consumption is based on the outdoor unit’s power consumption. The indoor units’ electric power consumption is not considered in the calculation of the coefficient of performance, which is intended to compare the actual power consumption with respect to the rated electric power consumption shown in Table 1. It is noted here that the electric power consumption of the indoor unit is minimal, less than 40W each. Eq. (2) shows the calculation of the coefficient of performances for cases shown in Fig. 3: \( \text{COP}_a \) for the case shown in Fig. 3a, \( \text{COP}_b \) for the case shown in Fig. 3b and \( \text{COP}_c \) for the case shown in Fig. 3c. It is noted that for \( \text{COP}_c \), the \( \dot{Q}_{\text{Defrosting}} \) is zero as the indoor unit air fans are stopped.

\[
\text{COP}_a = \frac{\frac{1}{n} \sum_{\text{Time}} \dot{Q}_{\text{Cooling/Heating}}}{\frac{1}{n} \sum_{\text{Time}} \dot{E}_{\text{Cooling/Heating}}} ; \quad \text{COP}_b = \frac{\frac{1}{n} \sum_{\text{Time}_{\text{Cycle}}} \dot{Q}_{\text{Cooling/Heating}}}{\frac{1}{n} \sum_{\text{Time}_{\text{Cycle}}} \dot{E}_{\text{Cooling/Heating}}} , \quad \text{and;} \quad \text{COP}_c = \frac{\frac{1}{n} \sum_{\text{Time}_{\text{Cycle}}} \dot{Q}_{\text{Heating}} + \dot{Q}_{\text{Defrosting}}}{\frac{1}{n} \sum_{\text{Time}_{\text{Cycle}}} \dot{E}_{\text{Heating}} + \dot{E}_{\text{Defrosting}}} 
\]  

(2)

3. Results and discussion

3.1. Instantaneous data

Figure 4 shows the instantaneous results with specified times shown for the electric power consumption, cooling capacity and return air of the two indoor units with the conditions presented in Table 2. Figures 4a
Fig. 4. Instantaneous results of electric power consumption, cooling capacity and indoor units’ return air at 35°C outdoor DB temperature: a-b) steady on mode at 100% load during cooling operation, and; c-d) on-off mode at 20% load during cooling operation.

Fig. 5. Instantaneous results of electric power consumption, heating capacity and indoor units’ return air at 7°C outdoor DB temperature: a-b) steady on mode at 100% load during heating operation, and; c-d) on-off mode at 20% load during heating operation.

and 4b show the results for the case of a 100% load. It is shown that the steady on operation of the compressor is, as expected, at full capacity. The steady on operation of the compressor resulted in the almost stable conditions of the return air dry bulb temperature and humidity ratio. Figures 4c and 4d show the instantaneous results for the case of a 20% partial load. It is shown in the results that the compressor is operating in the on-off mode. The on-off operation of the compressor resulted in the fluctuation of the return air dry bulb temperature and humidity ratio. Based on the observations gathered during the measurement of the system,
the system steady on compressor operation can be attained above a 35% partial load in this test specimen. Below the 35% partial loading, the compressor is operating in the on-off mode.

Figure 5 shows the instantaneous operation of the system with the specified time shown in the graphs during the heating operation with the conditions presented in Table 2. Figures 5a and 5b show the instantaneous results for the 100% load. As shown and as expected, the compressor operation is steady on to support the heating load at full capacity. With the steady on operation of the compressor, the return air dry bulb temperature becomes stable. Figure 5c and 5d show the results for the case of a 20% partial load. It is shown that the compressor is operating in the on-off mode. As presented, the return air dry bulb temperature is fluctuating as it depends on the compressor operation. It is noted here that during the heating operation, the indoor humidity is not controlled. Based on the observation of the full range of heating loads, the compressor steady on operation occurred at above 50% of the partial load compared to above 35% for the case of cooling operation. This is due to several factors affecting the operation of the compressor when comparing the cooling and heating operations, such as the electric motor, compressor and the control function.

Figure 6 shows the instantaneous operation of the system with specified times shown in the graphs during the heating operation at very low outdoor air temperature with the conditions presented in Table 2. The figures shown in Figs. 6a and 6b are for the 100% thermal load. As shown in the results, the defrosting operation took place after the specified heating-mode time to melt the accumulation of ice attached to the outdoor unit heat exchanger. After a specified time of defrosting mode, the heating mode started again. Comparing the electric power consumption during the heating mode and the defrosting mode, the power consumption is much less during the defrosting mode even with the compressor speed at maximum. Based on the system evaluation, the pressure difference between the high side and low side during the defrosting mode is much lower, as compared to the heating mode with almost the same ratio of electric consumption shown in the graph. The much lower pressure difference shown during the defrosting mode is due to the effective heat transfer in the outdoor unit heat exchanger as it is covered with ice which gradually melts even though the fans in both the outdoor unit and indoor units are stopped. However, the results of the defrosting mode decrease the indoor units’ return air temperature as shown in the graph, Fig. 6b. As the indoor units become a refrigerant evaporator, the air temperature around the indoor unit heat exchangers tends to lower as it is very cold (high density), which results in a very low return air temperature as registered in the temperature sensors. In addition, the cold air circulates the indoor chambers as no air heating occurred during the defrosting mode.

3.2 Compressor speed

Figure 7 shows the results of the compressor speed at different load ratios with different outdoor air temperatures. Figure 7a shows the results during the cooling operation which indicate that the compressor speed increases as the cooling load ratio increases. It also shows that when the outdoor air dry bulb temperature increases, the compressor speed increases. Based on this result, the increase of the compressor speed as the load ratio increases is to support the required cooling of air; hence, it needs a high refrigerant flow rate vis-à-vis a high compressor speed. The increase of compressor speed as the outdoor air temperature increases is to
support the amount of heat dissipation in the outdoor unit from the heat absorbed in the indoor units. As the outdoor temperature increases, the temperature difference between the refrigerant and the air decreases; hence, the compressor speed increases to increase the flow rate of refrigerant and at the same time increases the pressure and results in the condensation of refrigerant at higher temperature than the outdoor air temperature. Figure 7b shows the result during the heating operation. It is the same case for the cooling operation: when the heating load ratio increases, the compressor speed increases to increase the refrigerant flow rate needed for the higher heat transfer needed in the higher heating load requirements. It also shows that the compressor speed increases as the outdoor air temperature decreases. As the outdoor air temperature decreases, the amount of heat to be absorbed and transferred in the indoor chambers to maintain the temperature is needed. In order to attain this, a high flow rate of refrigerant is needed and at the same time, a reduction in the low side pressure of the refrigerant so as to result in the lowering of the refrigerant boiling point to less than the outdoor air temperature and absorb the heat from the air.

3.3 Refrigerant high pressure side

Figure 8 shows the results for the high refrigerant pressure of the system at different load ratios. It also shows the different refrigerant pressures at different outdoor air temperatures. As presented in Fig. 8a, the high pressure side of the refrigerant increases gradually as the cooling load ratio increases. The increase of the high pressure side is due to the increase of the compressor speed to provide higher refrigerant flow rate as the cooling load increases. As shown in the figure, the high pressure side of the refrigerant increases as the outdoor air temperature increases, and it is the same case at the different cooling load ratio, where the higher pressure also increases. The increase of the high pressure of the refrigerant is needed for the required heat transfer between the refrigerant and the outdoor air as stated in the previous section. As the outdoor air temperature increases, the temperature difference between the refrigerant and the air decreases. In order for that to take place, a higher refrigerant flow rate at a higher pressure is needed. It is noted here that at the higher refrigerant pressure, its temperature is also higher as shown in P-h diagram, results to the condensation of refrigerant at higher pressure. Figure 8b shows that result for the case of the heating operation. As shown in the graph, the high pressure side of the refrigerant increases as the heating load ratio increases. There is the same explanation
in the case of the cooling operation: the required refrigerant flow at the higher heating load ratio is also higher which results in higher pressure as the heating load ratio increases. However, as presented in the results, when the outdoor air temperature decreases, the trend of the high pressure refrigerant is the same for all. It is noted here that the high pressure side during the heating operation is in the indoor chambers. The amount of heat transfer in the indoor chamber is the same together with the resistance as the load and surface area of the indoor units are the same. For the case of higher heating load ratios for the lower outdoor air temperatures at 2°C and -7°C, the high pressure side tends to lower. Analyses during this operation show that the defrosting cycle operates in these heating load ratios. During the defrosting operation, the high pressure side is in the outdoor unit, as opposed to during the heating operation in which the high pressure side is in the indoor units. Also, during the defrosting operation, the high pressure side of the refrigerant is not as high as when it is compared to the heating operation due to the effectiveness of the heat transfer between the refrigerant and ice in the outdoor heat exchanger when covered with ice. This results in a reduction of the refrigerant pressure. Hence, the average pressure is reduced as shown in these results.

3.4 Refrigerant low pressure side

Figure 9 shows the low pressure of the refrigeration cycle at different load ratios and outdoor air temperatures. Figure 9a represents the cooling operation of the VRF air-conditioning system. As presented in the graph, the low pressure side of the refrigeration cycle decreases as the cooling load ratio increases. This happens even when the outdoor air temperature increases, which decreases the low pressure of the refrigerant by almost the same trend. The decrease of the low pressure side of the refrigeration cycle as the cooling load ratio increases is due to the increase of the compressor speed, which translates to a higher refrigerant flow rate. As the compressor speed increases, a much lower pressure is created in the low side of the refrigeration cycle and translated to lower boiling point of the refrigerant, thus creating a higher temperature difference between the air and the refrigerant. At a higher temperature difference, a higher heat transfer occurs. As shown in the graph, for higher outdoor temperature and at higher cooling load ratio, the low pressure side of the refrigerant tends to increase. As presented in Fig. 8a, it is shown that the high refrigerant pressure side increases as the load ratio increases. It is also shown that when the outdoor temperature increases, the high pressure side increases. As the refrigerant high pressure side increases, the low pressure side increases as well, as the outdoor temperature increases due to the increase of the compressor speed to provide a higher refrigerant flow rate. It is noted here that as mentioned in Fig. 8a, there is always a temperature difference between the refrigerant and air to provide for effective heat transfer. Figure 9b is the result of the case of the heating operation, showing that the low pressure side of the refrigerant decreases as the heating load ratio increases. It also shows that the low pressure refrigerant further decreases as the outdoor temperature decreases. It is noted here that the low pressure side is in the outdoor unit of the VRF air-conditioning system. When the heating load ratio increases, it is shown that the compressor speed increases to deliver a higher refrigerant flow rate to support the higher heating load. This results in the decrease of the low pressure side of the refrigerant in the outdoor unit as the difference between the high pressure side and the low pressure side increases. In addition, at a lower refrigerant pressure, the temperature of the refrigerant becomes lower, and

![Figure 9](image_url)

**Fig. 9.** Refrigerant low pressure side at different load ratios: a) cooling operation at different outdoor air dry bulb temperatures, and; b) heating operation at different outdoor air wet bulb temperatures.
this results in a higher temperature difference between the air the refrigerant for a higher heat transfer. As shown in the results, the low pressure side of the refrigerant further decreases as the outdoor temperature decreases. As mentioned earlier, when the outdoor temperature decreases, the compressor speed increases to support the heating load requirement. In this situation, it results in a large decrease of the low pressure side of the refrigerant cycle.

### 3.5 Coefficient of performance

Figure 10 shows the results of the coefficient of performance of the VRF system and different load ratios and outdoor air temperatures. Figure 10a shows the coefficient of performance of the system during the cooling operation at different cooling load ratios and outdoor temperatures. It shows that the system coefficient of performance decreases as the outdoor air temperature increases. The decrease of the system coefficient of performance as the outdoor air temperature increases is due to the capability in heat dissipation of the refrigerant to the air in the outdoor unit. As the outdoor air temperature increases, the system control increases the compressor speed to provide a higher refrigerant flow rate and increase the pressure of the refrigerant in the outdoor heat exchanger. As the refrigerant pressure increases, its temperature also increases. This results in having a temperature difference between the refrigerant and the outdoor air which in turn results in the heat transfer. This situation is explained for high and low refrigerant pressures (Figs. 8a and 9a). Figure 10b shows the system operation during heating. It shows that the system coefficient of performance decreases as the outdoor air temperature decreases. As presented previously, the compressor speed increases to provide a higher refrigerant flow as the outdoor air temperature decreases. As the refrigerant flow increases, it creates a much lower pressure in the lower pressure side of the refrigerant which is in the outdoor unit. As the lower pressure side of the refrigerant decreases, it creates a lower boiling point of the refrigerant and results in the absorption of the heat from the air to be passed to the indoor units. As presented in Fig. 8b, the low pressure side decreases as the outdoor air temperature increases and the compressor speed increases as shown in Fig. 7b.
3.6 P-h diagram

Figure 11 shows the P-h diagram of the VRF system under evaluation. It shows the diagram both for cooling and heating operations for the load ratios of 50% at different outdoor air temperatures. Figure 11a shows the diagram for the cooling operation. As shown in the diagram, the high pressure side of the refrigerant increases as the outdoor air temperature increases. As discussed in the previous section, as the outdoor air temperature increases, the compressor speed increases to provide higher refrigerant flow rate which results in an increase of the high pressure side pressure and temperature. When the refrigerant is at a higher pressure and temperature, it can provide a temperature difference between the air and refrigerant for the dissipation of the heat absorbed in the indoor units. As shown in the figure, the temperature of the refrigerant is higher than the outdoor air temperature. The low pressure sides are all almost the same as explained in the previous section. In the case of the heating operation, the results are shown in Fig. 11b, which shows that the low pressure side of the refrigerant becomes lower as the outdoor air temperature decreases. As explained in the previous section, as the outdoor air temperature decreases, the compressor speed increases to provide a higher refrigerant flow which results in the further decrease of the lower refrigerant pressure in the low side. As shown in the diagram, the refrigerant temperature in the lower pressure side is lower than the air temperature so as to absorb heat from the air.

5. Conclusion

This paper shows the performance evaluation of the one outdoor unit and two indoor units VRF air-conditioning system at different outdoor air temperatures which would be experienced in the actual operation of the system, such as from the start to end of the summer season, from the start to end of the winter season and during the early part of the day, noon time and later in the day. The system was operated from the partial to full thermal loadings to know the system behavior and the performances. With the system operations mentioned above, important conclusions are drawn as follows:

- During the cooling operation, the compressor speed increases as the cooling load ratio increases. In addition, the compressor speed increases as the outdoor air temperature increases. The increase of the compressor speed as the cooling load ratio increases is to support the higher refrigerant flow rate needed to absorb the high thermal load in the indoor chambers to be dumped into the outdoor chamber. The increase of the compressor speed as the outdoor air temperature increases is to support the high refrigerant pressure and temperature in the outdoor unit for the effective heat dissipation of the heat absorbed in the indoor units.

- In the heating operation, it is observed that the compressor speed increases as the heating load ratio increases. It also shows that the compressor speed increases as the outdoor air temperature decreases. The same is true for the case during the cooling operation, whereas the indoor thermal load increases, it needs a higher refrigerant flow rate to absorb the heat in the outdoor air to be released in the indoor chambers. In addition, as the outdoor air temperature decreases, it needs a lower pressure and temperature refrigerant passing the outdoor unit to absorb the heat in the outdoor air to be released in the indoor chambers. This results in the compressor operating at a higher rate as the outdoor air temperature decreases.

- As the outdoor air temperature further decreases, it is observed that the system operates in heating and defrosting modes. The defrosting mode is an important cycle for melting the accumulation of the ice in the outdoor heat exchanger in order to effectively utilize the outdoor heat exchanger for heat absorption in the outdoor air to be transferred to the indoor chambers during the heating mode. There are several control operations, depending on the different manufacturers in the activation of the defrosting mode and its stoppage. However, as shown in the result, during the defrosting mode, the indoor chambers’ temperatures were affected due to the decrease of the return air temperatures in the indoor units as cold air is circulating when the heating mode is stopped for the defrosting mode. The decrease of the indoor chamber air temperature might affect the thermal
comfort of the occupants in cases where the system is installed in real buildings with the assumption that no back-up air heater is to be installed to heat up the indoor environment air during the defrosting mode.

Based on this study, it is shown that the performance evaluation of the VRF air-conditioning system at different outdoor air temperatures is very important in the determination of the system’s behavior, performance and electric consumption. As presented in the results, it is shown that with the operation of the system in real climatic or weather conditions, the changing outdoor air temperature and the performance of the system deviated much from the specified performance shown in rated values. In terms of the accurate evaluation of the new buildings or retrofitted buildings in energy consumption, it is very important to consider the outdoor air temperatures as they much affect the electric energy consumption of the system in delivering the needed indoor thermal environmental conditions. Moreover, when the VRF air-conditioning system is operating at different outdoor temperatures at unbalanced loading, the performance of the VRF air-conditioning system is expected to be affected further as compressor speed changes to support both the effect of outdoor temperature and unbalanced loading (Enteria et al., 2016). Hence, the results of this study and of previous study on unbalanced loading are important in the realistic and actual electric energy consumption determination of the system in actual building installation and operation.

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Nomenclatures

| Symbol | Definition                        |
|--------|----------------------------------|
| C      | Cooler                            |
| Ė      | Electric consumption (kW)         |
| h      | Specific enthalpy (kJ/kg)         |
| H      | Heater                            |
| I      | Inlet Air                         |
| n      | Specified time for data           |
| O      | Outlet Air                        |
| P      | Pressure (MPa)                    |
| Q      | Heating / Cooling capacity (kW)   |
| R      | Return Air                        |
| S      | Supply Air                        |
| T      | Temperature (°C)                  |
| IC     | Indoor chamber                    |
| IU     | Indoor unit                       |
| OC     | outdoor chamber                   |
| OU     | outdoor unit                      |
| COP    | Coefficient of performance        |
| DBT    | Dry Bulb Temperature (°C)         |
| EEV    | Electronic expansion valve        |
| JIS    | Japan Industrial Standards        |
| WBT    | Wet Bulb Temperature (°C)         |
| VRF    | Variable refrigerant flow         |

Greek letter

α Load balance ratio (α = 1, Balanced; α < 1, Unbalanced)

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