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Development of high efficiency VRF systems under partial heat load for commercial buildings

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Abstract
VRF systems in commercial buildings are mostly operated under less than 50% heat load through a year, even if the capacity is optimally selected. Therefore it is important to improve its performance under low heat load condition. For this purpose, innovative energy-saving A/C controller and compressor were developed in order to enable an optimum operation under low heat load conditions in buildings. These technologies were applied to new developed VRFs "VRV X" and these new VRF products have been launched since March, 2015 in Japan. The partial heat load performance tests of the conventional VRFs and new VRFs were conducted by air enthalpy method. This report shows the operation states and the EER/COP of the new VRFs in comparison with those of the conventional VRFs by the laboratory tests in the test facility. And the seasonal average EER/COP (not SEER/SCOP) of these VRFs was predicted by combining the EER/COP of the VRFs obtained by the laboratory tests and the annual variations of operating conditions in the model building.

Keywords - Multi-split type heat pump; energy saving; partial load; capacity control

1. Introduction

The energy efficiency of conventional VRFs degrades under low heat load conditions. On the other hand, VRF systems in commercial buildings are mostly operated under less than 50% heat load conditions through a year. Fig. 1 shows heat load conditions and the degradation of the EER/COP of VRFs at actual buildings. Therefore it is important to improve its performance under low and middle heat load condition. [1-3]

From 2010 through 2012, New Energy and Industrial Technology Development Organization (NEDO) in Japan have carried out the research and development project on "Next-generation Heat Pump Systems", which aimed for the development of innovative heat pump systems with 150 % higher EER/COP in comparison with conventional ones. As a part of this NEDO project, we have developed an innovative energy-saving A/C controller and a new compressor in order to enable an optimum operation under low and middle heat load conditions in buildings. By these technologies, it was demonstrated that 170 % higher seasonal average EER/COP was achieved by the air conditioning system with the developed new controller. [4]

After that, these technologies have been applied to new VRFs "VRV X" and these new VRF products have been launched since March, 2015 in Japan. This report shows the seasonal average EER/COP in real VRF operations (not SEER/SCOP, the detail will be described in section3) and annual energy consumption of the new VRFs in comparison with those of the conventional VRFs.

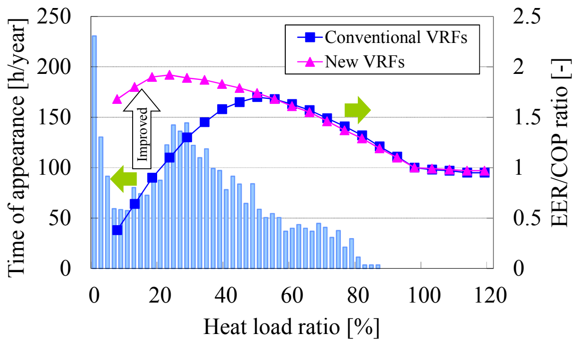

Fig. 1 Typical heat load model and EER/COP ratio in real operations

2. Features of New Developed VRFs

2.1. Product overview

Fig. 2 shows new developed VRFs "VRV X".

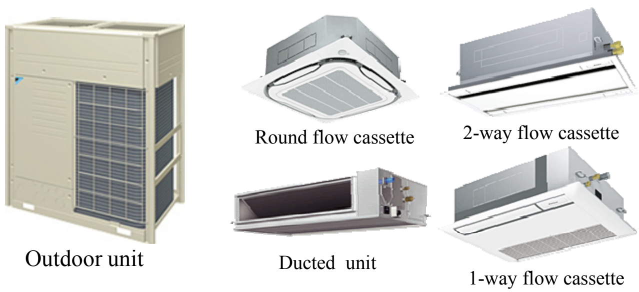

Fig. 2 Schematic of "VRV X" air-conditioning systems

For improvement of the energy efficiency of VRFs, we developed a capacity optimization control, a crank case heater control which reduces energy consumption when the system is standby and a new compressor increasing energy saving amount by capacity optimization control. These technologies were applied to "VRV X"

Each of the technologies is described below.

2.2. Capacity Optimization Control

In the conventional VRF systems, the indoor units have been controlled independently of the outdoor unit. In the outdoor unit, the compressor speed is controlled to keep the evaporation or condensation temperature of the refrigerant, T_e or T_c, at a target value. Although T_e and T_c should be optimized to the heat load, they are usually fixed irrespective of the operating condition of the air conditioner. Thus, under low heat load conditions, the high-low pressure difference of the refrigerant is excessively relative to the heat load conditions, and the energy consumption of the compressor cannot be decreased. In addition, mismatch arises between the controls of the indoor and outdoor units, and consequently the air conditioner repeats the operations with surplus capacities and stops. Such on/off operations of the compressor also cause the degradation of EER/COP.

In order to improve the performance of the air conditioner under low heat load conditions, T_e and T_c must be optimized to the heat load. It results in stabilizing of the compressor speed. Fig. 3 shows the flow of "VRT* smart control" (* stands for "variable refrigerant temperature") to optimize T_e and T_c.

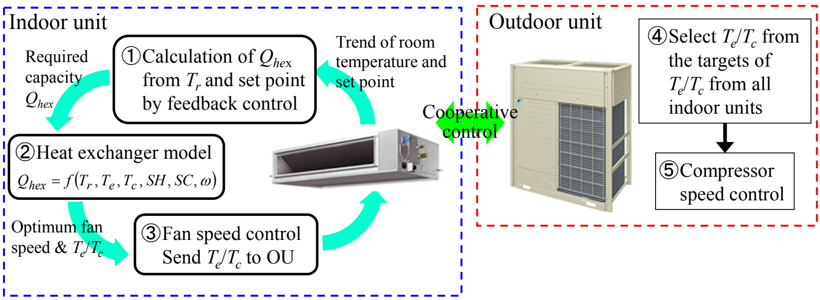

Fig. 3 Flow diagram of new developed controller

[Indoor unit]

1. A required capacity Q_{hex} is calculated from room temperature T_r and set point temperature by feedback control.
2. A fan speed, SH or SC and target T_e or T_c corresponding to renewed Q_{hex} by (1).

$$Q_{hex} = f\left(T_r, T_e, T_c, SH, SC, \omega\right) \qquad (1)$$

SH and SC are superheat and subcool of the refrigerant at the outlet of the indoor heat exchanger, and omega is the fan speed of the indoor unit. f is a function representing capacity characteristics of the indoor units.

There are some flexibility of setting target T_e or T_c and fan speed for the same Q_{hex}. The new A/C control algorithm equips the function to increase the fan speed in order to raise T_e in cooling mode (or lower T_c in heating mode). It must be more energy saving because the reduction of the compressor energy consumption by raising T_e (lowering T_c) is larger than the increase of the indoor units energy consumption with higher fan speed. It is noted that the controlled range of T_e, T_c and omega is limited not only to meet the occupant comfort and demand but also to ensure the reliability of a compressor.

The new A/C controller can maximize the energy savings with the equation (1) taking into account the limitation and user demand.

3. Every target T_e or T_c determined in each operating indoor unit is sent to the outdoor unit.

   [Outdoor unit]

4. The lowest T_e for cooling (or the highest T_c for heating) is selected from among T_e (or T_c) sent from the indoor units. The reason to select the lowest T_e (or the highest T_c) is to satisfy the largest required capacity all over the indoor units in operation.
5. The compressor speed should be changed accordingly to keep the required T_e in cooling mode (or T in heating mode).

By these intelligent controls between the indoor and outdoor units, the high-low pressure difference of the refrigerant is can be minimized and as a consequence, the energy consumption of the compressor decrease. Moreover, the intermittent compressor operations at middle and low cooling/heating load can be suppressed effectively. Consequently, EER/COP with the new control algorithm is expected to be better than with the conventional algorithm.

2.3. Crank Case Heater Control

The heating/cooling capacities and piping length between indoor and outdoor units are oversized and long in general in VRF systems, and it proportionally increases the amount of the refrigerant charge in the system. The refrigerant gradually dissolves into the oil which is charged in the compressor at standby mode according to the outside temperature. Then the oil viscosity will decrease.

As a consequence, the compressor suffers the serious damage on the bearing at start-up, and it causes the breakdown. This phenomenon tends to be more serious in the VRF system rather than in the residential system.

The crankcase heater is equipped with covering the lower part of the compressor and warms the oil indirectly to avoid being damaged at restart. Most of the time during stand-by, crank case heater is activated in the conventional VRF system, then the lots of energy should be consumed in addition to the energy consumption in operation.

In our study, the crank case heater with conventional control algorithm consumed up to 8.5% of the total energy through the year. The prediction for the amount of the dissolved refrigerant into the oil has not been studied and the suitable heater control has not been determined. Especially, the amount of the dissolved refrigerant right after the start up, normally it leads to sudden decrease of the viscosity, have not been determined, then the required heater control which support both of the reliability of the compressor and energy saving have not been implemented.

To overcome this challenge, several experiments to determine the amount of solution have been conducted and it can be determined through the predicted parameters on the calculation model (2). The required oil temperature is determined through by using this model, and then the annual power consumption of the heater can be decreased by 15% compared with the conventional system.

$$Mcond = Qcond\left(Ta, Toil, \kappa, A\right)\,\Delta h \tag{2}$$

2.4. Developed Compressor

As for an our conventional scroll type compressor, the orbiting scroll is pressed against the fixed scroll because of the pressure difference between the center part, which has high pressure, and the outer part, which has low pressure. This pressure difference is only optimal at the rated condition, and at low compression ratio, refrigerant will leak from high pressure side to the low pressure side.

As a result, this leakage spoils the effect of the energy saving of the capacity optimization control. The development of the technology for controlling the orbiting scroll back pressure overcomes this issue. This technology enables to introduce the intermediate pressure refrigerant during the compression into the back face of the outer part on the orbiting scroll. Thus, the outer part of the orbiting scroll is pressed properly by this intermediate pressure even in a low compression ratio operation, and it leads to the minimization of the refrigerant leak between the scrolls with the reduction of the friction of the orbiting scroll on a wide operation range from a low compression ratio to high compression ratio.

Furthermore, introducing the injection mechanism enlarges the capacity with downsizing and improves the efficiency of the product lineup of the compressor.

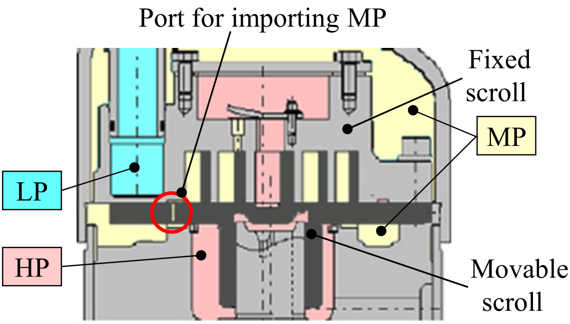

Fig. 4 Schematic of new compressor

3. Evaluation of Energy-saving Effect of New Developed VRFs by Part-load Performance Tests

3.1. Tested VRF Systems

For examination of seasonal average EER/COP and annual energy consumption of VRF systems, we conducted the partial heat load performance tests in the test facility in Chubu Electric Power. In this test facility, the performance of an A/C with a rated cooling capacity from 3.5 kW to 168 kW and a rated heating capacity from 3.5 kW to 200 kW can be measured by the air-enthalpy method, so it can accurately measure the capacity not only at the rated point but also under middle and low heat load conditions. Furthermore these tests simulated real VRF system operations in a real office building, for this purpose the test chamber for indoor units can accurately control heat load at a constant condition and the test chamber for outdoor unit can accurately control the room temperature at a constant condition, and tested indoor units keep the air temperature in the test chamber for indoor units to the set point temperature.

In tests for measuring SEER/SCOP, compressor speed, valve opening and fan speed etc. of VRF are fixed, but the purpose of our tests is to measure the controller performance, therefore these actuators aren't fixed. That's why the EER/COP in our tests has different meanings from SEER/SCOP.

The specifications of the VRV systems provided for this test are shown in Table 1.

Table 1 Specifications of tested VRF systems

Model	Rated cooling capacity	Rated heating capacity	Indoor units
Ve-upIV Hi-COP series and VRV X	45 kW	50 kW	Cassette type 11.2kW x 4

Using this test facility, we could measure the performances of the conventional VRF system "Ve-upIV" and the new VRF system "VRV X" at the same conditions. Fig.5 shows the schematic of the test chambers.

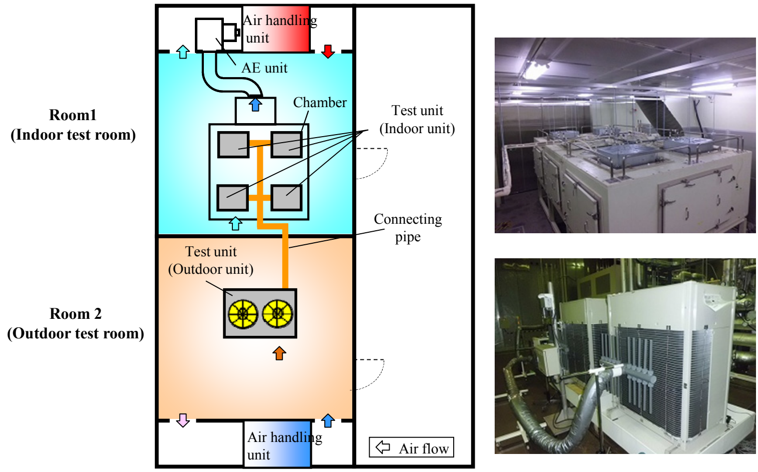

Fig. 5 Schematic of the test chambers

3.2. Test Conditions

From Dec. 2010 to Nov. 2011, we measured the operating performances of the VRF systems installed in the real general office building in Nagoya city. Fig. 6 and Fig. 7 show those operating performances for cooling and heating mode. Energy efficiency of VRF systems changed depending on outdoor temperature t_j and the heat load ratio BL_c/Φ_{cr} or BL_h/Φ_{hr}, where BL_c (BL_h) and Φ_{cr} (Φ_{hr}) denote the indoor cooling (heating) load and rated cooling (heating) capacity of the VRF system, respectively.

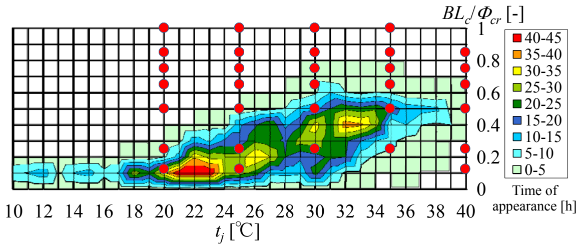

Fig. 6 Operating condition of VRF systems in cooling mode

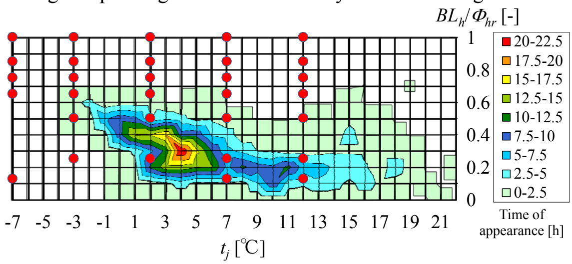

Fig. 7 Operating condition of VRF systems in heating mode

Therefore, the test conditions of the part-load performance tests were determined based on those field data so that they covered almost the complete range of the actual operating conditions measured in the office building. The red symbols in Fig. 6 and Fig. 7 indicate these conditions, which are 37 points in cooling operations and 31 points in heating operations. The detail of the conditions is shown in Table 2. And the set point temperatures of indoor units were 27 degree C in cooling mode and 20 degree C in heating mode from the general set point temperatures in office buildings in Japan.

Table 2 Test conditions for VRF systems

Cooling mode test							
t_j [℃] DB/WB	Cooling load ratio BL_c/Φ_{cr}						
	12.5%	25%	50%	65%	75%	85%	100%
20 / -	○	○	○	○	○	○	○
25 / -	○	○	○	○	○	○	○
30 / -	—	○	○	○	○	○	○
35 / -	—	○	○	○	○	○	○
40 / -	○	○	○	○	○	○	○
45 / -	—	—	○	—	○	—	○
50 / -	—	—	—	—	—	—	○
Heating mode test							
t_j [℃] DB/WB	Heating load ratio BL_h/Φ_{hr}						
	12.5%	25%	50%	65%	75%	85%	100%
-7 / -8	○	—	—	○	○	○	○
-3 / -4	—	○	○	○	○	○	○
2 / 1	—	○	○	○	○	○	○
7 / 6	○	○	○	○	○	○	○
12 / 11	○	○	○	○	○	○	○

3.3. Results of Part-load Performance Tests

EER/COP of the conventional VRF system and new VRF system measured in the part-load cooling performance test and heating performance test are shown in Fig. 8 and Fig. 9.

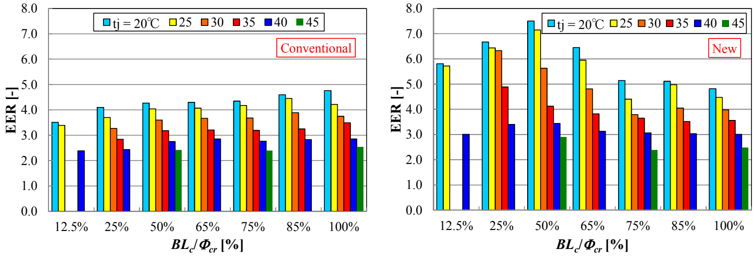

Fig. 8 EER in partial load test (cooling mode)

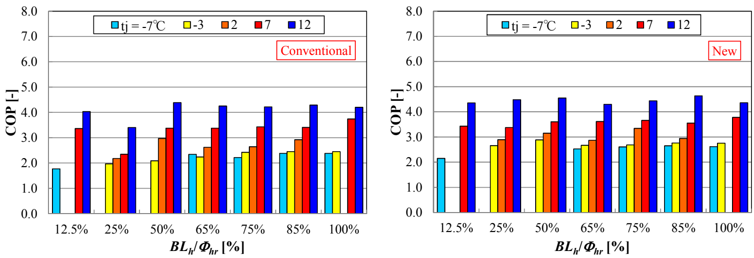

Fig. 9 COP in partial load test (heating mode)

Cooling EER of the new VRF system shows higher values than the conventional one under all test conditions. In particular, the EER is over 1.5 times as high as the conventional VRF system in a relatively middle and low range of cooling load conditions, which are most frequently experienced conditions in real buildings. Heating COP of the new VRF system is also increased, and the maximum increase ratio is about 1.4 times as high as the conventional VRF system. Relatively the increase ratio of the EER in the cooling operation is much higher than the COP in the heating operation.

Fig. 10 shows the trends of the cooling capacity, power consumption, room temperature and heat load measured with the conventional VRF system under the low cooling load of $BL_c / \Phi_{cr} = 25\%$. Fig. 11 shows these trends with new VRF system.

In Fig. 10, the conventional VRF repeats start-up/shut down operations, and then EER gets worse and the room temperature fluctuates. As for the new VRF, the operation is stabilized and the room temperature almost keeps a constant value in Fig. 11.

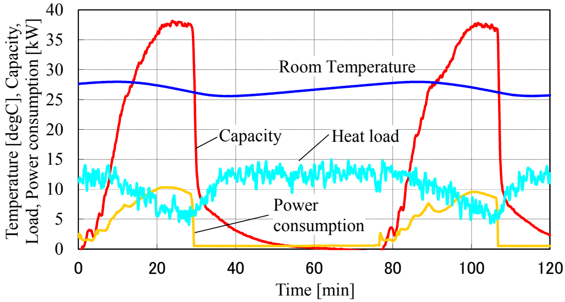

Fig. 10 Trend of operation data (Conventional VRF, cooling mode, $BL_c / \Phi_{cr} = 25\%$)

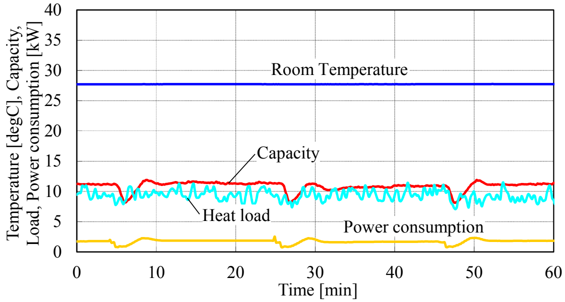

Fig. 11 Trend of operation data (New VRF, cooling mode, $BL_c / \Phi_{cr} = 25\%$)

4. The Prediction of the Energy Saving Effect Based on the Part Load Tests

4.1. Methodology[5]

The energy saving amount of the developed “VRV X” and conventional VRF system are calculated respectively considering the results of the part load tests referred in section3 and the predicted operating conditions of VRF systems against the predicted load variation which is calculated with the ambient air temperature. The detailed calculation method is as follows. (The hourly power consumption is calculated with the capacity which is assumed as equal as the predicted load based on the ambient temperature and EER/COP which is examined in the test room, and then the annual power consumption is determined by integrating the calculated power consumption over occurrence hours of the corresponding load in a year).

Finally, the annual power consumption rate and averaged EER/COP are determined. It should be noted that the weather data is referred from the Expanded AMeDAS Weather Data in Nagoya city in Japan and the load are modeled as the typical office building in Nagoya, and the cooling load ratio is defined as 100% at 40 degree Celsius of ambient air.

4.2. Results

Table 3 shows comparison results of the seasonal averaged EER/COP of the developed “VRV X” and conventional VRF system against the assumed typical building.

Compared to the conventional VRF systems, the developed “VRV X” saved significant amount of energy up to 28.9% and increased the annual averaged COP by 40.6% with the combination of the newly developed technologies.

Table 3 Results of estimation of average COP

	Conventional (Ve-upIV)	Developed (VRV X)
Averaged Cooling EER	3.520	5.434
Averaged Heating COP	2.691	3.237
Averaged Annual COP	3.233	4.545
COP improvement rate	-	140.6%
Power consumption reduction rate	-	28.9%

5. Conclusions

We developed an innovative energy-saving A/C control technology and a new compressor in order to enable an optimum operation under middle and low heat load conditions in buildings, and they were applied to new developed VRF products “VRV X”. We compared the performance of “VRV X” and a conventional VRF by the part-load performance tests in the test facility. It was demonstrated that the annual energy consumption of “VRV X” was remarkably lower than conventional VRFs. We will continue to contribute to energy saving by disseminating products with these technologies.

In the development of this product, we utilized the results of the project on “Next-generation Heat Pump Systems” of NEDO.

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