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ABSTRACT

Global demand for air conditioning is expected to increase continuously. In order to minimize the impact on the climate as a total amount of CO2 equivalent, it is essential to continue to search for better refrigerants. On the other hand, product size and cost are important factors in providing value to users.

This time, while using the experiment result of the mini-split type air-conditioner reported in the past, in addition, we conducted a new experiment on multi-type VRF air-conditioner. Then, the difference in COP performance between R-410A and alternative candidate refrigerants was evaluated. In addition, we studied the product design using R-452B, R-454B by simulation and compare the size when the performance is made identical by changing the system specifications with alternative refrigerant candidates.

As a result, common results were obtained in both systems. It was found that R-452B and R-454B have better performance than R-410A. On the other hand, for split type air-conditioner with connecting pipes, it was found that these refrigerants need to increase both the indoor heat exchanger and the outdoor heat exchanger to 120% in order to obtain the same performance as R-32.

1. INTRODUCTION

The refrigerant R-32 has a relatively low GWP that contributes to the prevention of global warming, also has a better *COP* than conventional R-410A in air-conditioners. Also, because it is a single refrigerant, it is easier to manage and recover and regenerate easily. R-32 was selected as the alternative for R-410A for mini-split type heat pump air-conditioners. After applied in all Japanese manufacturers, it is started to launch from major manufacturers in Asia, Europe, Australia, and so on.

However, as global demand for air conditioning is expected to increase continuously. In order to minimize the impact on the climate as a total amount of CO2 equivalent, it is essential to continue to search for better refrigerants. On the other hand, product size and cost are important factors in providing value to users.

It is evaluated that the high ambient temperature cooling performance of R-452B and compared with R-32 using mini-split type air-conditioner [3], and evaluated the performance with R-410A and R-32 etc. using system simulation [4]. As a result, it was found that R-452B is superior to conventional R-410A in *COP*, but it is inferior to R-32 and its difference becomes large with high ambient temperature.

Up to now, it had been studying a mini split type air-conditioner using alternative refrigerant of R-410A. This time it is expanded the subjects of the study to multi type VRF air-conditioner and decided to compare what would become if we changed the product size so that the performance would be the same.

2. PROPERTIES OF REFRIGERANTS

Table 1 shows the property of refrigerants which were evaluated. Refrigerant properties are computed using REFPROP version 9.1 (Lemmon et al., 2013) [2]. R-452B and R-454B are calculated with mixing parameters. It is compared that the theoretical *COP* (Coefficient of Performance) in a cooling operation cycle. Calculation conditions were as follows: condensing temperature (T_c) is 45°C, evaporating temperature (T_e) is 10°C, suction pipe temperature (T_s) is 15°C, condenser outlet temperature ($T_{c.out}$) is 40°C, and compressor adiabatic efficiency is 70%. Regarding the pressures equivalent to those representative temperatures, it is selected that the pressure which has the same mean-temperature between the boiling point and the dew point for each blends. Refrigerating effect is obtained by the enthalpy difference in the evaporator, and the theoretical *COP* is obtained by dividing refrigerating effect by the theoretical compressor work. However, actual *COP* requires consideration of volumetric capacity, pressure loss, and so on. Since a compressor sucks gas the amount equivalent to the cylinder volume, if the refrigerant is changed with a constant speed compressor with a fixed cylinder volume, the refrigeration capacity of the system changes in proportion to the volumetric capacity which is refrigerating effect per suction specific volume. Therefore, the refrigeration capacity decrease when a refrigerant with a small volumetric capacity. In order to recover this, if the compressor rotation speed is increased, the mass flow rate increases and the pressure loss in the gas pipes, especially on the low pressure side of system, is increased. And due to the decrease in the suction pressure in the compressor, it leads to increase the compression ratio, and the *COP* of actual refrigeration cycle is lower than the theoretical *COP*. The pressure loss ratio in Table 1 represents the pressure loss per unit length of each refrigerant compared with R-410A. For comparison, it is calculated taking into consideration the influence of volume flow rate and suction density necessary to give the same capability as R-410A. According to this, since the pressure loss of R-32 is smaller than that of R-410A, it is expected that system performance of a split type A/C system and VRF system with long connecting pipes will be advantageous. For R-452B and R-454B, theoretical *COP* is higher and pressure loss are smaller than R-410A, so it is expected that the actual system performances will be higher than R-410A.

3. EXPERIMENT SYSTEM FOR EXPERIMENT

3.1 System Outline

Actual experiment was carried out using the R-32 (R-410A also same specification) mini split type air-conditioner for residential (mini-split system) and the R-410A multi-split type air-conditioner using variable refrigerant flow

Table 1: Physical Properties and Calculated Properties of Refrigerants

Refrigerant	R-410A =R-32/R-125 (50/50wt%)	R-452B =R-32/R-125/ R-1234yf (67/7/26wt%)	R-454B =R-32/R-1234yf (68.9/31.1wt%)	R-32 (100%)
ASHRAE 34 safety Classification	A1	A2L	A2L	A2L
GWP (AR5)	2088	698	466	675
Theoretical *COP* ratio (%)	100%	101.3%	101.4%	102.1%
Refrigerating effect ratio (%)	100%	117.2%	119.2%	150.5%
Compressor Work: ratio (%)	100%	115.7%	117.5%	147.4%
Volumetric Capacity ratio (%)	100%	97.4%	96.6%	107.9%
Specific Volume in Suction (%)	100%	120.3%	123.3%	139.5%
Pressure loss ratio (%)	100%	90%	90%	57%
Discharge / Suction pressure (MPa abs)	2.730/1.256	2.605/1.201	2.579/1.190	2.795/1.281
Temperature glide (K) at 15°C	0.1	0.9	1.0	0
Discharge Temperature T_d (°C)	69.1	72.3	72.6	81.4

*Calculation Conditions: T_c=45°C, T_e=15°C, Suction line Temp.: T_s =20°C, Condenser Outlet: $T_{c.out}$=40°C, Compressor Efficiency: η_{comp} =70%, in Cooling Operation.
Saturation temperature of mixed refrigerant is midpoint temperature of two-phase region under constant pressure.

control for commercial buildings (VRF system). Experiments on the mini-split system are reference from the previous report [3].

Figure 1 and Figure 2 show the outline of system used for this experiment. Figure 1 shows the mini-split system with rated cooling capacity of 7.1 kW. Indoor unit and outdoor unit are connected with two 7.5-meter Standard length pipes. The outer diameter of the heat transfer tube of the heat exchanger is 7 mm for the indoor unit and 8 mm for the outdoor unit. Indoor and outdoor heat exchanger in this product are designed for R-32, however, the specification of heat transfer tubes are the same for R-410A, not a special optimization. The compressor is optimized for R32, but the difference in volumetric capacity between R-32 and R-410A is approximately 8%, that is, the compressor rotation speed is approximately 8% different, and the compressor efficiency is not different greatly. Figure 2 shows the R-410A VRF system with rated cooling capacity of 28kW. The indoor unit connects two Ceiling Mounted Cassette types with rated cooling capacity of 14 kW. Indoor unit and outdoor unit were connected by two pipes of 25 m gas and liquid. In both systems, the compressor is capable of changing the mass flow rate by rotating speed with a variable frequency drive.

During the Experiment, measurement of capacity of this system was conducted by a facility using the air-enthalpy method which is described by ISO 5151-2010.

3.2 Experimental Conditions

Table 2 shows cooling experiment conditions based on ISO 5151-2010. Before measuring the performance of the experiment system with each refrigerant, it is adjusted that the amount of refrigerant to find the optimum amount for the *COP*. In the mini-split system, the degree of suction superheat was set to obtain the maximum COP, and in the

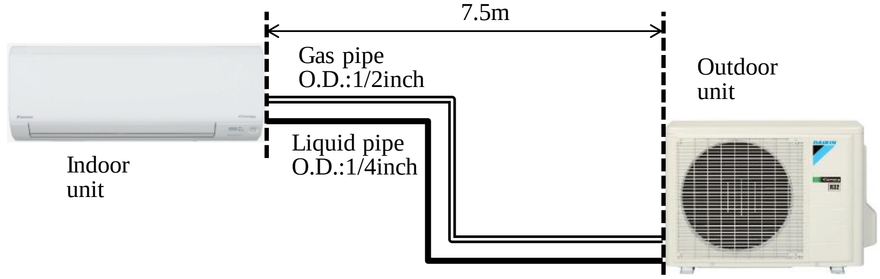

Figure 1: R-32 (R-410A :Heat exchanger and pipes) mini-split system

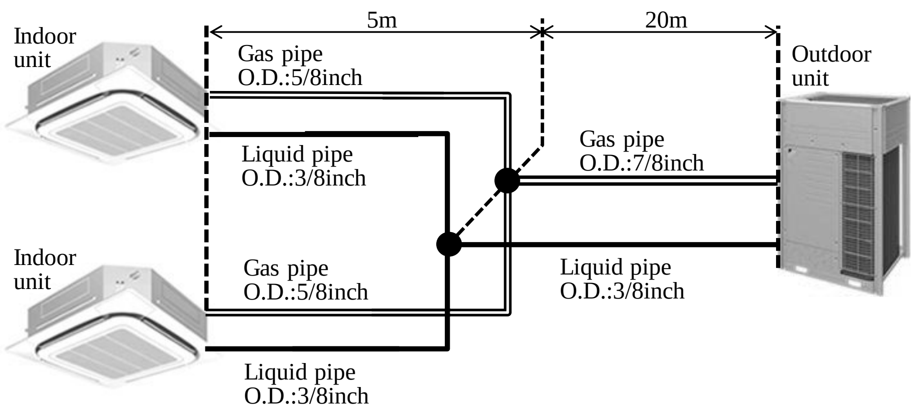

Figure 2: R-410A VRF system

Table 2: Air Temperature Conditions at Cooling Operation

Operation mode	Cooling		Heating
Condition	T1	T3(H)	H1
Indoor Unit (°C)	DB:27 / WB:19	DB:32 / WB:23	DB:20 / WB:15
Outdoor Unit (°C)	DB:35 / WB:24	DB:52 / WB:38	DB:7 / WB:6

VRF system, it was adjusted to be constant at 5 °C. In this way, it could be compared that the system optimized for each refrigerant, it is conducted that a performance evaluation by changing compressor rotation speed and air temperature.

4. DROP-IN EXPERIMENTAL RESULT

4.1 Experimental Results at Mini-Split System

According to the previous report [3], it was conducted that experiments by changing the amount of R-452B, compressor rotation speed, and adjusting the opening ratio of the expansion valve at condition T1. Then, it was tried to find that the most suitable amount of refrigerant for the maximum *COP* at the rated cooling capacity 7.1kW. Then, it is conducted that the experiment on air temperature condition. As a result, the *COP* difference due to the refrigerant was remarkable at rated capacity T1 and half-load T3(H). Both R-452B and R-32 showed that the *COP* is excellent from the viewpoint of R-410A alternative. However, since R-32 was even better than R-452B, we concluded that this performance difference was due to the following reasons by analyzing.

- The evaporation latent heat of R-452B is smaller than that of R-32. This means that in order to obtain the same capacity, higher refrigerant mass flow rate is needed. Thus, the heat transfer area to obtain degree of subcool increases at the high refrigerant mass flow rate, that is, effective heat transfer area for the latent heat exchange decreases. Therefore, T_c becomes higher, the compressor load becomes larger, and *COP* becomes lower.
- In the high heat load such as large capacity and high ambient temperature, the latent heat of R-452B declines remarkably, especially at T_c=50°C or more. Thus, the difference between the *COP* of R-452B and that of R-32 becomes much higher.

The performance result at T1 and T3(H) are shown in Figure 3. The difference in *COP* between R-32 and R-452B was 9.5% at T1 and 6.3% at T3(H).

4.2 Experimental Results at VRF System

As in Section 4.1, the optimal amount of refrigerant was determined for each refrigerant by changing the amount of refrigerant so that *COP* would be maximized at condition T1. As shown in Table 3, the liquid density at the outlet of the condenser with much refrigerant and the trend of the refrigerant charge amount of each refrigerant are in agreement. However, compared with other refrigerants, the amount of R-32 is smaller than the liquid density ratio. This is the same phenomenon in the case of the mini-split system, as reported in the previous study [3], It is considering that R-410A and R-452B hold more liquid in the condenser. Experimental results of both the

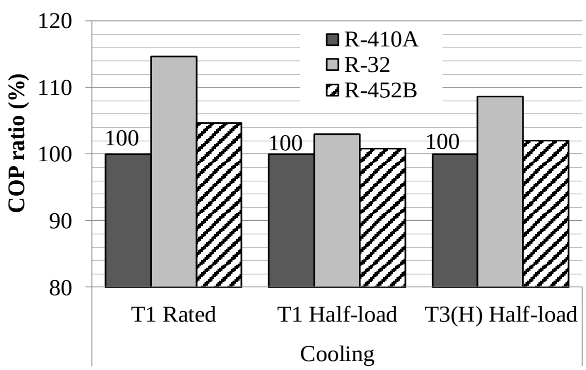

Figure 3: Mini-split *COP* ratio at each capacity

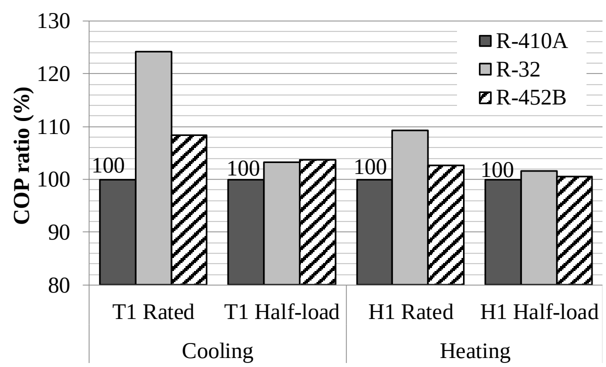

Figure 4: VRF *COP* ratio at each capacity

Table 3: Optimum refrigerant charge

System	Mini-split			VRF		
Refrigerant	R-410A	R-32	R-452B	R-410A	R-32	R-452B
Calculated density at outlet of condenser [kg/m^3]	980	896	916	933	860	875
vs R-410A	100%	91.4%	93.4%	100%	92.1%	93.8%
Optimum refrigerant charge by experiments vs R-410A	100%	82.4%	90.4%	100%	88.2%	98.8%

mini-split system and the VRF system showed that the amount of R-32 can be kept the lowest at the comparison of these three refrigerants and that the amount refrigerant difference between R-452B and R-32 is approximately 9%.
After that, it was conducted the experiments by changing air temperature and compressor rotation speed. The result is shown in Figure 4. Both cooling and heating were noticeably different in rated *COP* by refrigerant difference. R-452B and R-32 showed that *COP* is excellent from the viewpoint of R-410A alternative. And among them, R-32 is even better in these refrigerants. This is because that the characteristic of low pressure loss on the gas phase side, especially because it is remarkable in the cooling condition. On the other hand, the difference of *COP* at half-load condition *COP* becomes smaller. It is thought that the pressure loss of R-410A and R-452B is reduced and at the same time R-32 has a large volume capacity, so that, the compressor is operated at a low speed. And the compressor efficiency drops as compared with other refrigerants to achieve the same capability.

5. STUDY ON SYSTEM PERFORMANCE IMPROVEMENT

5.1 Outline of Optimization Study by Simulation

As mentioned in Chapter 4, there is a difference in *COP* due to refrigerant. From the viewpoint of the alternative refrigerant of R-410A, it is found that R-452B is comparable refrigerant.

However, since R-32 is better in *COP*, it will be compared R-452B and R-32 after this chapter.

It is explained that how R-452B needs to improve performance in order to obtain the same *COP* as R-32. At the same time, R-454B having refrigerant performance characteristics similar to R-452B is also described. In order to know what changes in refrigeration cycle specification are necessary to maintain same performance as R32, it is conducted that the drop-in experiments s using the system simulation software. Figure 5 shows the outline of refrigeration cycle used in the simulation. The system simulation software used is “Energy Flow +M Core System” [5] which has been developed by Professor Kiyoshi Saito laboratory of Waseda University, experimented and acknowledged by “JRAIA Refrigerant evaluation working group”.

In order to adjust the degree of superheat at the outlet of the evaporator, a PID controller was installed to change the open ratio of the expansion valve. Moreover, if it is necessary to adjust the capacity, it can be controlled to achieve the target capacity automatically with another PID controller.

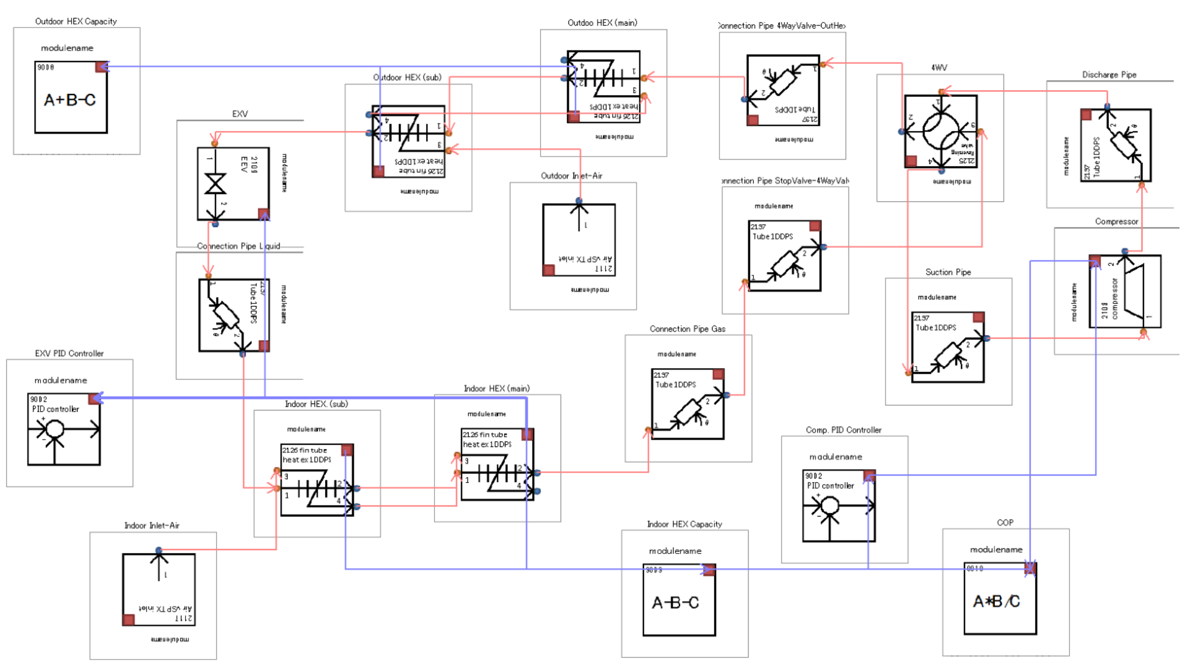

Figure 5: Outline of refrigeration cycle on the simulation

Following study using this system simulation software were conducted.

- Firstly, it was verified that the certainty of system performance simulation. The accuracy verification of actual measurement and simulation has already been carried out based on the actual measurement result of R-32 [4]. However, it was verified the accuracy based on the actual measurement result of R-452B in this time. This is because to investigate the refrigeration cycle specification to raise the *COP* of R-452B and R-454B until the measured *COP* of R-32 to be described later.
- Secondly, it was examined the improvement of *COP* by changing pipes specifications. Specifically, paying attention to the influence of pressure loss at gas pipes in the low-pressure side, it was effective to increase the diameters of suction pipe and gas connection pipe between indoor unit and outdoor unit.
- Thirdly, it was examined improvement of *COP* by changing the specification of the heat exchanger. Specifically, the effect by increasing the size of the indoor heat exchanger and the outdoor heat exchanger was examined.

For the second and the third process, it was compared that how much specifications had to be raised in the case of other refrigerants, based on the actual measurement result of the cooling rated of R-32.

In the actually measured drop-in experiment, the cylinder volume of the compressor was optimized for the original refrigerant, and tuning was not performed for the other refrigerant, but if actually changing the refrigerant, it is necessary to optimize the cylinder volume. As a result of the actual measurement, the difference in compressor rotation speed due to the difference in refrigerant at the same capacity is about 10 rps, and optimizing the cylinder volume by changing the refrigerant may improve the compressor efficiency by about 1% to 2%. However, since the parameters affecting compressor efficiency include complicated phenomena such as pressure loss and heat conduction in the compressor. It is focused on the change of actual system balance and do not consider the optimization of cylinder volume on simulation in this paper.

5.2 Accuracy Verification of System Performance Simulation

Two kinds of results of experiment and simulation were compared. Figure 6 shows mini-split system and Figure 7 shows the result of VRF system. In the simulation, based on the measurement result of condition T1 of R-452B as a baseline, the following values were adjusted to match the experimental result: pressure loss of the gas connecting pipe and suction pipe, flow rate of air passing through the condenser and the evaporator. volume efficiency, adiabatic efficiency, mechanical efficiency of the compressor. Next, the *COP* were checked in changing the capacity widely in same air condition. Compared with the actual measurement results, the simulation results increased errors when the pressure and temperature were adjusted and changed significantly from the adjusted condition T1. On the other hand, in the vicinity of fitting, it was calculated with an error range of approximately 2%.

In the simulation, since the compressor efficiency was fixed to a constant value, it was presumed that this influence becomes a calculation error.

From the above discussion, it is considered that accuracy of system performance prediction can be sufficient by paying attention to the scope of application.

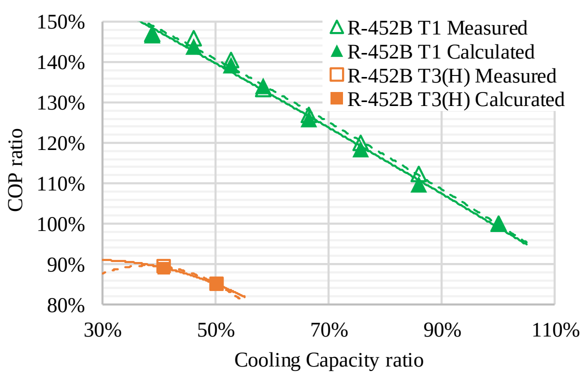

Figure 6: Simulation results at mini-split

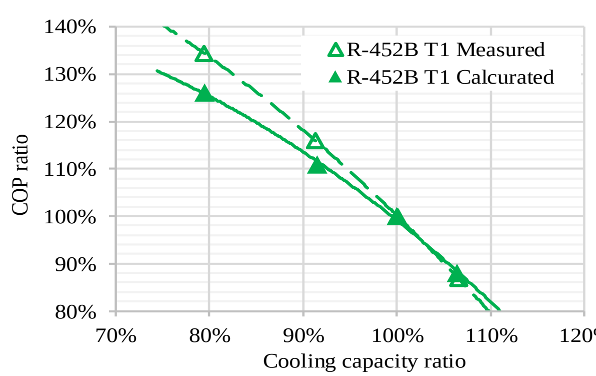

Figure 7: Simulation results at VRF

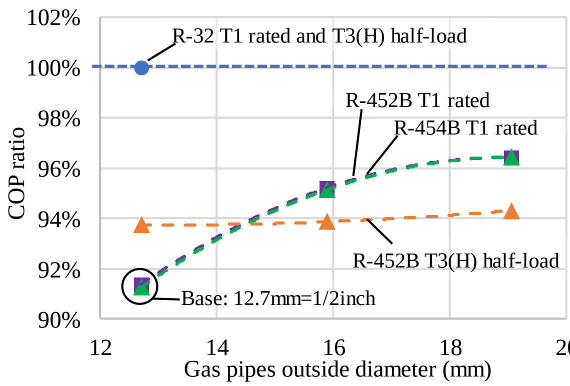

Figure 8: Effect of gas pipes upsize at mini-split

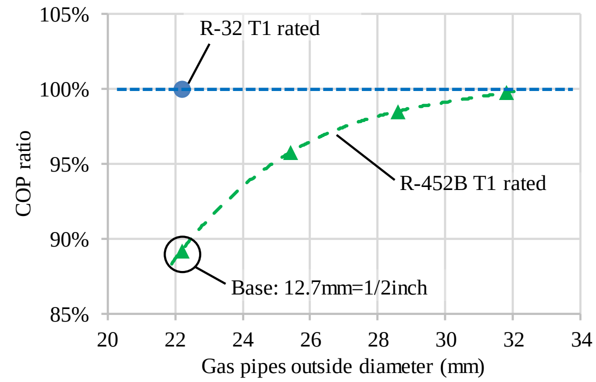

Figure 9: Effect of gas pipes upsize at VRF

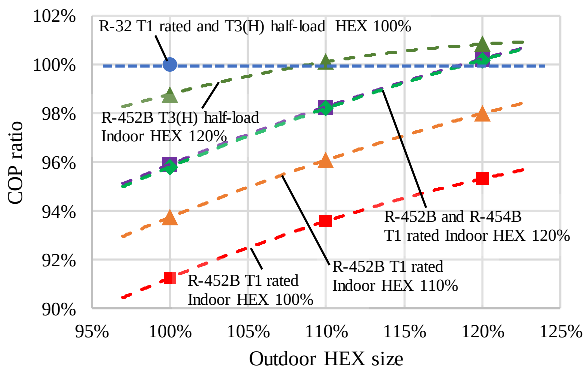

Figure 10: Effect of HEX upsize at mini-split

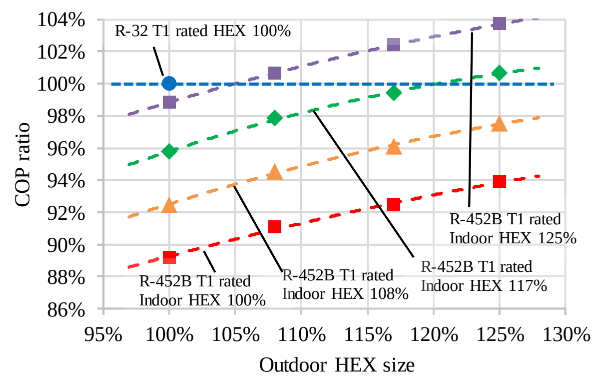

Figure 11: Effect of HEX upsize at VRF

5.3 Effect of Gas Pipes Diameter Upsize

Figure 8 and Figure 9 show the effect of changing gas pipes for R-452B and R-454B. When increasing the size of the pipes from the gas connection pipe to the compressor suction, how much the *COP* improves was calculated.

In the mini-split system, the size of gas pipes in all section from the indoor unit to the compressor of the outdoor unit was increased every 1/8 inch. In the VRF system, as shown in Figure 2, the gas pipes diameter is different between the section from the indoor unit to the pipes confluence and the section from the confluence to the outdoor unit, so they were increased every 1/8 inch at each section. The horizontal axis of Figure 9 expresses the pipe diameter of the section from the confluence point to the outdoor unit as a representative value.

As a result, in the mini-split system, the optimum diameter for the refrigerant flow rate is fully satisfied, so even if the diameter is increased, R-452B and R-454B improvement reaches saturation. On the other hand, in the VRF system, it was found that R-452B improved to the same level as R-32 by increasing the pipe diameter. However, from the viewpoint of workability to bend and to connect pipes, design of vibration suppression and cost, it is not practical to expand the pipe diameter to about 1.5 times the current size.

From the above, in order to obtain the *COP* equivalent to R-32, it was found that it is insufficient to increase the gas pipes size only.

5.4 Effect of Heat Exchanger Upsize

Figure. 10 and Figure. 11 show the effect of changing the size of heat exchanger for R-452B and R-454B. It is examined that *COP* improvement by increasing the size of the heat exchanger without changing the gas pipes diameter.

When the indoor heat exchanger is expanded to 117% to 120%, improved by about 5% to 7%. In the same way, when the outdoor heat exchanger is expanded to 117% to 120%, improved by about 3% to 4%. It is more effective to increase the size of the indoor heat exchanger. In addition, increasing the size to 120% for both indoor and outdoor heat exchangers for R-452B and R-454B, it can be Improve to same *COP* as R-32 at condition T1. In mini-split system, increasing the indoor heat exchanger size to 120% and outdoor heat exchanger size to 110%, *COP* can be improved to the same level as R-32 at condition T3 (H).

5.5 Consideration on Upsize of Gas Pipe Diameter and Heat Exchanger

For R-452B and R-454B, *COP* same as R-32 could not obtained without 120% larger heat exchanger. As gas pipes diameter increase, installation workability can be deteriorated and vibration and noise caused by the compressor are also deteriorated. In addition, it is necessary to use the long pipe to suppress vibration propagation, which is also disadvantageous from the viewpoint of ensuring reliability and size and cost of air-conditioners. It is important factor to install easily particularly in residential air-conditioners, because the total number of sales is much larger than that of commercial air-conditioners. In addition, increasing the size of the heat exchanger as a main task deteriorates the equipment cost. Especially for commercial air-conditioners with high capacity, the original sizes are naturally large, which means that it becomes even larger, and the equipment costs increases proportionally and installation space may become insufficient.

Increasing the size of pipe diameter and increasing the size of the heat exchanger leads to increase in the amount of refrigerant in the system. In spite of using low GWP refrigerant, the effect of mitigating global warming at the product life cycle diminishes.

If R-410A is used as a reference, R-452B and R-454B are good performances and it is thought that the above problem does not arise. However, there is a possibility of obstacles in steadily advancing the user benefits and installer benefits of future air-conditioners. On the other hand, R-32 is superior from the above viewpoint among the refrigerants evaluated this time, and it can be considered that it can contribute to the progress of air-conditioners.

6. CONCLUSIONS

It is found that the following result with these experiments and system simulations for split type air-conditioner with connecting pipes:

- In the drop-in experiment results to the VRF system for R-410A, the performance of R-452B is superior to R-410A in both cooling and heating operation. And it is found to be excellent as an alternative refrigerant for R-410A.
- Furthermore, the performance of R-32 is superior to R-452B under the rated condition of cooling and heating.
- By comparing R-32 and R-452B, the performance of R-32 is dominant under rated conditions, and the difference of *COP* between R-32 and R-452B was 16% in the rated cooling conditions. The difference of *COP* becomes smaller at half-load condition.
- In the mini-split system, the COP of R-452B and R-454B which estimate the performance by changing the system specifications using the system performance simulation is less than that of R-32. It is not enough to just increase the size of the gas pipe diameter. It is necessary to increase the size by 120% for both the indoor heat exchanger and the outdoor heat exchanger.
- In the VRF system, the COP of R-452B which estimate the performance by changing the system specifications using the system performance simulation is less than that of R-32. It is not enough to just increase the size of the gas pipe diameter. It is necessary to increase the size by 120% for both the indoor heat exchanger and the outdoor heat exchanger.
- In current R-410A market, both R-452B and R-454B showed higher *COP* than R-410A. On the other hand, the viewpoint of prevention of global warming potential is an indispensable factor. However, it is need to consider not only this factor but also other factors for further progress in the future. For example, equipment cost, suppression of power fee, installation space, and good installability. Based on reason, R-32 is a refrigerant with better *COP*, and further improvement in the future of air-conditioners can be expected the benefits in the market.

7. NOMENCLATURE

COP	Coefficient of Performance	(–)
GWP	Global Warming Potential	(CO_2-eq.)
SC	Degree of subcool	(K)
SH	Degree of superheat	(K)
P_d	Discharge pressure	(MPa abs)
P_s	Suction pressure	(MPa abs)
T_c	Condensing temperature, (midpoint temperature of two-phase region of P_d)	(°C)
T_e	Evaporating temperature (midpoint temperature of two-phase region of P_s)	(°C)
T_s	Suction temperature	(°C)
$T_{c.out}$	Condenser outlet temperature	(°C)
DB	Dry Bulb temperature	(°C)
WB	Wet Bulb temperature	(°C)

8. REFERENCES

1 Abdelaziz, O., Shrestha, S., et al., Alternative Refrigerant Evaluation for High-Ambient Temperature Environments: R-22 and R-410A Alternatives for Mini-Split Air Conditioners, Oak Ridge National Laboratory, n.ORNR/TM-2015/536, 2015

2 Lemmon, E.W., Huber, M.L., McLinden, M.O., 2013, NIST Standard Reference Database 23: NIST Reference Fluid Thermodynamic and Transport Properties-REFPROP, version 9.1, National Institute of Standards and Technology, Gaithersburg.

3 Minamida, T., Taira, S., Haikawa, T., Performance Evaluation of Heat Pump System using R32 and HFO-mixed Refrigerant in High Ambient Temperature, 16th International Refrigeration and Air Conditioning Conference at Purdue University, USA, 2016, n.ID2408

4 Haikawa, T., Taira, S., Minamida, T., Evaluation of Performance of Heat pump using R32 and HFO-mixed Refrigerant by Refrigeration cycle simulation and Loss analysis, 16th International Refrigeration and Air Conditioning Conference at Purdue University, USA, 2016, n.ID2409

5 Ueno, K., Saito, K., Ohno, K., Unsteady State Simulation of Vapor Compression Heat Pump Systems with Modular Analysis, 14th International Refrigeration and Air Conditioning Conference at Purdue University, USA, 2012, n.ID2421