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Evaluation and Development of Air-conditioners using Low GWP Refrigerant

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Abstract

It is important to make the refrigeration technologies sustainable to the environment in order to continue to provide people with many technical benefits for a long time. In particular, to cope with global warming by reducing greenhouse gas and saving energy is an immediate task in the world. Characteristics of R32 for environment and high efficiency are effective for this opportunity in residential air-conditioner market, and 20 million R32 air-conditioners have already been sold by the year 2016. In Japanese market in particular, use of R410A have already been phased out.

Introduction of new HFO blended refrigerants, for example, blended with R1234yf or R1234ze(E), are expected to further contribute to the reduction of global warming. To check the reports that adding HFO refrigerant to R32 makes performance superior to R32, we conducted drop-in test to confirm the performances of 2 types of HFO blend refrigerants, R32/R1234ze(E)(70/30) and R32/R125/R1234yf (67/7/26):R452B. We compared the results from experiment and system simulation. As a result, R452B indicated higher COP than R410A, but lower than R32. This may be attributed to the increase in pressure loss that caused the degradation although this investigation using system simulation proved to be effective for prediction. From now on, we will continue to utilize this system simulation to accelerate exploration for next generation refrigerants.

Key words: GWP; Refrigerant; Heat pump system; System simulation; R32

1. Introduction

In recent years, the demand for mitigating global warming impact and energy conservation has increased significantly. We chose R32 as a new refrigerant for reversible heat pump systems. However, it is expected that the demands for air conditioning will continue to increase in the future, thus minimizing climate impact in CO2 equivalent in the whole lifecycle of an appliance is essential. To achieve this, many researchers in the air conditioning industry and academia continue searching for new refrigerants.

The reason why we chose R32 was that its GWP (Global Warming Potential) is 1/3 as small as that of R410A, required refrigerant charge is smaller, and it has excellent thermo-physical properties to achieve better performance of the reversible heat-pump systems.

In addition, it proved to be the best refrigerant among all other candidate refrigerants from the viewpoint of safety and economy. Because there is no concern about fractionation, R32 is easy to manage, furthermore it is attractive even from the viewpoint of recovery and recycle.

On the other hand, various refrigerants mixed with R32 have been born from many studies, and there are some which have been declared to be superior in the aspect of GWP and performance. [1-3]

Recently, new refrigerant R32/R125/R1234yf (67/7/26) was reported as a high efficiency refrigerant. [4] This contains three types of refrigerant which are based by 67wt% R32. We charged the refrigerant into a mini split type air-conditioner and compared the respective performances.

2. Properties of Refrigerants

Table 1 shows the properties of four refrigerants which were charged into the test system in this experiment. They are HFC refrigerants, R32 and R410A, HFO mixed refrigerants R32/R1234ze(E) (70/30) (Blend A) and R32/R125/R1234yf (67/7/26) (Blend B). In this paper, we name those HFO mixed refrigerants as follows.

Blend A: R32/R1234ze(E) (70/30)
Blend B: R32/R125/R1234yf (67/7/26); R452B

Showing in Table 1, Blend A has temperature glide of 4.4K during phase transition between vapor and liquid. Although Blend B also has temperature glide, it's not as large as Blend A's, as B's remains in 0.9K. Temperature glide affects the system performance, as the temperature gap between refrigerant and air shrinks. Blend B can be expected to have better performance than Blend A.

Table1. Calculated properties of refrigerants charged to the test system

Refrigerant	R32 (Pure)	R410A =R32/R125 (50/50)	Blend A =R32/R1234ze(E) (70/30)	Blend B =R32/R125/R1234yf (67/7/26)
Global warming potential: GWP (AR4)	675	2088	<500	698
Temperature glide: ΔT_{GL} (K) @ 10°C	0.0	0.1	4.4	0.9
Discharge / Suction pressure: P_d / P_s (MPa abs)	2.795 / 1.107	2.730 / 1.087	2.366 / 0.929	2.605 / 1.039
Refrigerating effect w_r (kJ/kg)	248.0 (100.0%)	163.9 (66.1%)	210.4 (84.8%)	192.4 (77.6%)
Compressor work: w_s (kJ/kg)	54.0 (100.0%)	36.5 (67.6%)	45.4 (84.2%)	42.2 (78.2%)
Coefficient of Performance: COP = w_r / w_s	4.593 (100.0%)	4.493 (97.8%)	4.629 (100.8%)	4.555 (99.2%)
Specific volume in suction v_s (m^3/kg)	0.0343 (100.0%)	0.0248 (72.1%)	0.0349 (101.7%)	0.0297 (86.5%)
Volume capacity $\phi = w_r / v_s$ (kJ/ m$^{3)}$	7228(100.0%)	6625(91.7%)	6029(83.4%)	6482(89.7%)
Pressure loss at constant capacity: ΔP_{loss} (% of kPa)	(100.0 %)	(165.0 %)	(141.3 %)	(143.7 %)
Work equiv. to pressure loss at constant capacity: $\Delta W_{P.loss}$ (% of W) ($\propto v_s^2 / w_r^3$)	(100.0 %)	(180.0 %)	(169.4 %)	(160.2 %)
Discharge temperature T_d (°C)	88.0	72.9	79.3	76.8

*Calculation conditions: T_c = 45°C, T_e =10°C, Suction line temp.: T_s =15°C, Condenser outlet: $T_{c.out}$ =40°C,
Compressor adiabatic efficiency: η_{comp} =70%, in cooling operation.
Saturation temperature of the blend is mean temperature of bubble point and dew point.
Properties of refrigerants' are calculated with NIST REFPROP Version 9.1.

Following to temperature glide, we compared 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 (η) is 70%.

These results are shown in Table 1 below. Regarding the pressures equivalent to those representative temperatures, we chose the pressure that has the same mean temperature between the bubble point and the dew point for the blends. Calculating theoretical COP requires refrigerating effect (w_r) which is enthalpy change width in evaporation. On the other hand, the larger refrigerating effect tends to give the larger refrigeration capacity in case of constant compressor speed. In fact, since a compressor sucks gas the amount equivalent to the cylinder volume, system cooling capacity is affected by volumetric capacity (ϕ) which is w_r per suction specific volume (v_s).

Meanwhile, there is a very important factor; pressure loss at constant capacity in the next row. Since this is the parameter which reduces the performance of system by raising discharge pressure and reducing suction pressure of compressor, the method how to calculate the factor is very important and is detailed in the following subsection. In addition, when the impact of pressure loss on the performance of a system is considered, it's important to convert the pressure loss at constant capacity (ΔP_{loss}) to the work equivalent to pressure loss at constant capacity ($\Delta W_{P.loss}$). It can be acquired by calculating required work to recover the pressure loss by compressing vapor adiabatically.

In Table 1, comparing ΔP_{loss} of each refrigerant, R410A has the largest loss among these refrigerants. ΔP_{loss} of Blend A and Blend B exceeds 140% when compared to R32. However, $\Delta W_{P.loss}$ (W), of Blend A and Blend B reaches above 160% as of R32. All the blends have larger pressure loss than R32. It is because R32 has larger w_r than R125, R1234ze(E) and R1234yf.

Regarding discharge temperature in the last row, R32 has the highest value in this property table. However, since R32 has superior performance to the others especially from the viewpoint of pressure loss and other aspects, its discharge temperature in the actual operation does not relatively rise as high as the other refrigerants because R32 system has narrower pressure differential (T_e-T_c) in actual operation. Therefore it would not be a significant issue in case of proper system design. From the above, R32 can be expected to have the best performance from these thermo-physical properties and other properties such as in the Table 1.

3. Calculation of Pressure Loss for Comparison

In this section, the way to calculate the equivalent work for compensating the pressure loss is explained.

At first, ΔP_{loss} derived from flow friction inside of pipe, of which friction factor is f (-), length is L (m) and diameter is d (m), is written as below in the equation (1). The equations on capacity, mass flow rate, and specific volume are also given in (2) and (3).

$$= \cdot - \cdot \frac{(\ /\ ^2)^2}{2} \tag{1}$$

$$= \frac{\ _0}{\ } \tag{2}$$

$$= \frac{1}{\ } \tag{3}$$

Uniting these equations (1), (2), and (3),

$$= \left(\cdot \frac{\ }{2\ ^5} \cdot\ _0^2 \right) \cdot \left(\frac{\ _s}{\ ^2} \right) \tag{4}$$

For this *Δ* , estimating the value of $\Delta W_{.}$ by raising the pressure up to the pressure before loss by compressing adiabatically.

$$W_{.} \;=\; \cdot \int_{1}^{2} \;=\; \cdot \int_{1}^{P_2} \cdot \tag{5}$$

If the pressure change width from $_1$ to $_2$ is sufficiently small in adiabatic compression, the specific volume ( ) change width can be considered negligible. As here the is assumed to be equal to the specific volume in suction ( ) of compressor, the equation (5) simply can be converted into equation (6).

$$W_{.} \;=\; \cdot \; \cdot \tag{6}$$

Substituting the equations (2) and (4) to equation (6) .

$$\Delta W_{.} \;=\left(\cdot \frac{}{2 \;\; 5} \cdot \;\; _0{}^{3} \right) \cdot \left(\frac{2}{3} \right) \tag{7}$$

The part in anterior parenthesis in equation (7) is determined by the specifications of an air conditioner, while the part in posterior parenthesis is determined with the properties of refrigerant. When comparing the performances of refrigerants, if the capacity is fixed as constant and specifications of an air conditioner is the same, it is enough only to take the term of the part in posterior parenthesis into account. Thus, a refrigerant with smaller specific volume and larger refrigerating effect per mass has smaller pressure loss and better energy efficiency.

Incidentally, the friction factor can be calculated from inside diameter of pipe, flow velocity in the pipe, density and viscosity of refrigerant. In the condition of the gas side communication pipe (ϕ1/2 inch) and rated nominal capacity, in case of the refrigerant other than R32 are approximately 5% smaller than R32. In this study, since the difference from R32 is fairly small than influence of posterior parenthesis, is put in anterior parenthesis of the air conditioner's specification. However, further calculations are needed for more precise study.

4. Test System for Experiment

4.1. System Outline

Fig. 1 shows the outline of the system used for the series of tests. It is a mini-split type air conditioner with a nominal cooling capacity of 7.1 kW. The indoor unit and the outdoor unit are connected by 7.5 m standard length pipes. This system requires 1.55 kg amount of R32 refrigerant as indicated in Table 2.

The compressor (Comp.) is capable of changing the revolution speed with a variable frequency drive (V.F.D.).

During the tests, measurement of capacity of this system was conducted by a facility using the air-enthalpy method (psychrometric type) which is described by ISO 5151-2010. Also, we measured temperature and pressure by T-type thermocouples and pressure gauges at the discharge and suction of the compressor as well as the inlet and outlet of the heat exchangers. At the midpoints of the heat exchangers, only temperatures were measured.

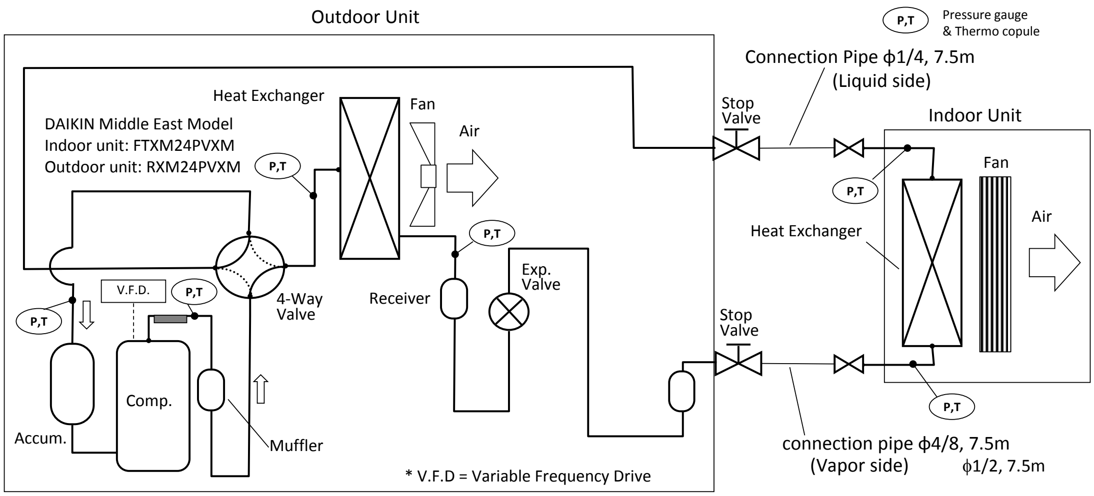


Fig. 1. Test system diagram

Table 2. Charged amount of refrigerant in the test system

	R32	R410A	Blend A	Blend B
Optimized refrigerant charge M_{ref} (kg)	1.55 kg (100%)	1.88 kg (121%)	1.7 kg (110%)	1.7 kg (110%)

Table 3. Test conditions

Operating mode	Capacity	Indoor Ambient		Outdoor Ambient	
		DB(℃)	WB(℃)	DB(℃)	WB(℃)
Cooling	Nominal (7.1 kW)	27	19	35	24

4.2. Test Conditions

Table 3 shows cooling test conditions based on ISO 5151-2010. Before measuring the performance of the test system with each refrigerant, we adjusted the amount of refrigerant to find the optimum amount of refrigerant for the COP. The results of charged amounts are described in Table 2. Moreover, we adjusted compressor suction super heat to achieve the highest COP by changing the opening ratio of the expansion valve. In this way, we could compare the systems optimized for each refrigerant.

5. Drop-In Test Results for Wide Capacity Range

5.1. COP Trend Comparison in the Wide Capacity Range

Fig. 2 shows a COP comparison in which the cooling capacity of each refrigerant is changed by the compressor speed on T1 condition. The vertical axis and the horizontal axis indicate COP ratio and cooling capacity ratio respectively, and each measured values are plotted with R32's capacity 7256(W) on T1 condition as the standard (100%: define as "R32 base point" in Fig. 2). This shows the system performance measured by changing the compressor speed, in order to compare the COP values in wide capacity range.

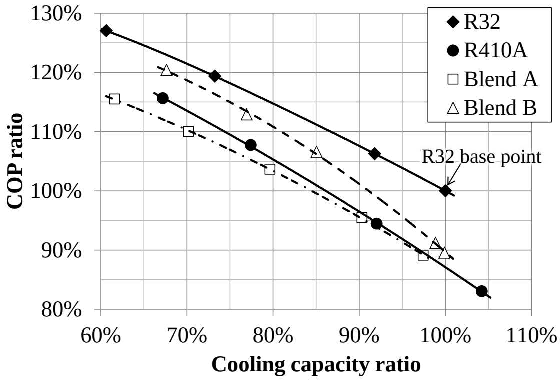


Fig. 2. COP trend comparison of each refrigerant for capacity

At first, in the whole capacity range, it was found that R32 achieved the best COP in this study. Regarding R410A and Blend B, as capacity increases, COPs tend to plunge when compared to R32. When capacity increases, refrigerant mass flow rate increases, and the loss originated from pressure loss also increases. Thus the superiority of R32 is apparent in higher capacity and refrigerant properties indicated in Table 1.

In contrast, because the pressure loss becomes smaller due to smaller mass flow rate, the differentials of COP between refrigerants in case of smaller capacity were assumed to become smaller. However, when operating with Blend A, the differential to R32 did not shrink as much as the other refrigerants. This is the result that the temperature gap between refrigerant and air couldn't be smaller, and this is caused by the larger temperature glide of Blend A i.e. 4.4K.

From the above, it could be noted that pressure loss becomes serious issue in case of larger capacity and temperature glide becomes more important in case of smaller capacity.

6. Drop-In Test Results around the Rated Capacity

In this section, we explain drop-in tests methods and the analysis of test results. In addition, we conducted the drop-in tests using the system simulation software, and compared two types of "drop-in" tests: experiment and by simulation, in which we also included the clarification regarding the difference between constant speed and variable speed compressor.

6.1. Outline of Simulation

Fig. 3 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, tested and acknowledged by "JRAIA Refrigerant evaluation working group."

As shown in Fig. 3, Heat exchangers are divided into two rows namely windward row and leeward row. This helps to calculate the wind flow direction for the refrigerant flow. We set connecting pipe and suction pipe separately in order to calculate the influence of pressure loss accurately.

We placed PID controller to adjust the superheat at the outlet of evaporator with the aim of giving target open ratio to 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.

The cooling operation was conducted, and air conditions are as shown in Table 3.

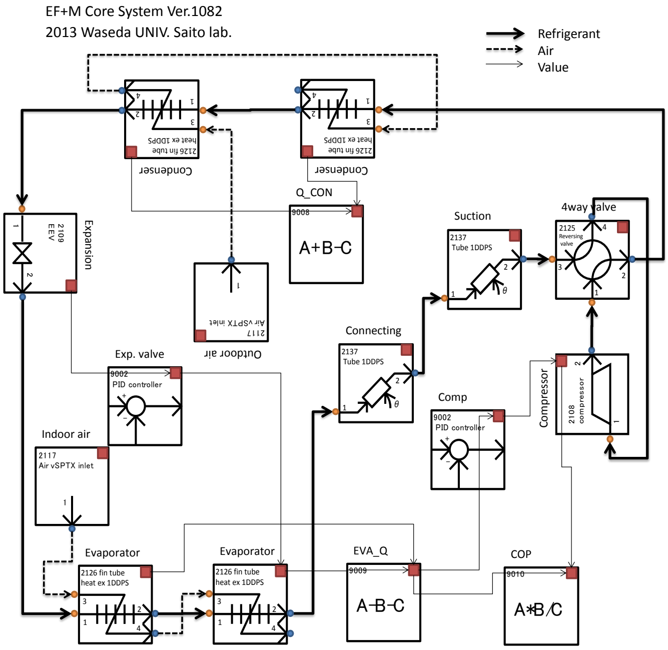


Fig. 3. Outline of refrigeration cycle on the simulation

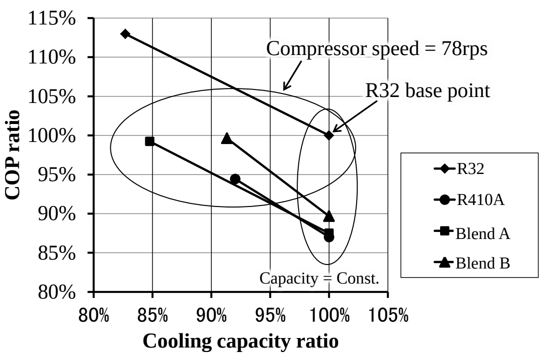


Fig. 4. COP trend comparison of each refrigerant for capacity by experiment

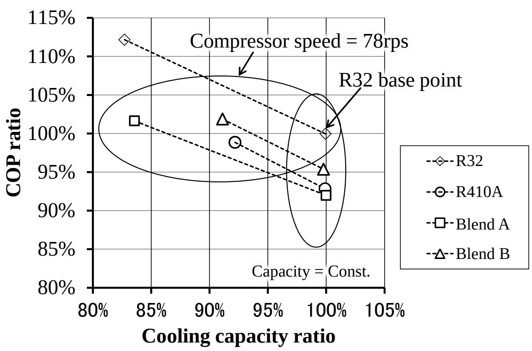


Fig. 5. COP trend comparison of each refrigerant for capacity by simulation

6.2. Result of Drop-In Test by Experiment

Fig. 4 shows the results of the experiment. The vertical axis and the horizontal axis indicate COP ratio and cooling capacity ratio respectively, and each measured values are plotted with R32's capacity 7256(W) on T1 condition as the standard(100%: define as "R32 base point" in Fig. 4). The data in case of the compressor speed of 78rps and the data when the capacity was made constant by adjusting the compressor speed are encircled.

As a result, it can be seen that COP of the refrigerants excluding R410A are almost same to R32 while comparing at a constant compressor speed of 78rps. As the vapor compression refrigeration cycle has characteristics that the COP decreases as the capacity increases, it is obvious that COP of the refrigerants at the constant capacity are as expected and shown in the Fig. 4.

Thus, comparison with equal capacity is necessary for an accurate comparison of refrigerants

6.3. Result of Drop-In Test by Simulation

Fig. 5 shows the same comparison as Fig. 4, however these data are based on simulation. In the simulation, approximately the same results as the experiment Fig. 4 were obtained. These are simulation results matched to the experimental results by adjusting the following values: Pressure loss in evaporator and suction pipes; The wind volumes through the condenser and the evaporator; The volumetric efficiency and adiabatic efficiency of compressor.

Next, we checked the COP when changing the capacity widely in same air condition, for example, at 83% capacity using R32. Comparing it to the Fig. 4, it is found that the results of calculation seem to be approximately equal to the results of experiment, and the simulation can be expected to produce adequate results and worth consideration before drop-in test.

COP of other refrigerants at 78rps seem to be around 2% higher than experiment when compare with “R32 Base point” data. In this simulation, as we fixed the compressor efficiency to the constant value decided at 78rps, it is speculated that this influence resulted in calculation error. When raising compressor speed in the refrigerant except for R32 in particular, a mismatch in COP in the case of experiment grows larger. For example, about Blend B, relative COP which was at 90% in the case of experiment remains in 95% by the simulation for R32, and simulation gives better performance than the experiment. This is assumed to be caused by the influences of compressor oil or differences of distribution of air volume in evaporator, etc.

7. Loss Analysis in each Refrigerant

7.1. Calculation Method

We conducted loss analysis to make the reason clear why the difference between experiment and simulation occurred. Fig. 6 and Fig. 7 show the results of loss analysis in case of each refrigerant during cooling operation at the rated nominal capacity. We measured compressor input, indoor fan input, and outdoor fan input during operation. Regarding compressor input, it can be divided into two types of input; “theoretical adiabatic compression work” and “compression loss”.

First, the former “theoretical adiabatic compression work” is divided into four parts in this analysis. When considering enthalpy transition during compression, the four parts are below, in order of increasing vapor pressure:

- ”Suction pipe pressure loss” is from suction pressure to evaporating pressure.
- ”Evaporator loss” is from evaporating pressure to saturation vapor pressure for evaporator’s intake air.
- ”Theoretical compression input in ideal condition” is from the saturation vapor pressure equivalent to both of the temperatures of evaporator’s intake air and condenser’s intake air. This is inevitable work as far as the temperature gap between indoor air and outdoor air.
- ”Condenser loss” is from saturation vapor pressure for condenser’s intake air to condensing pressure.

Second, regarding the latter "compressor loss",

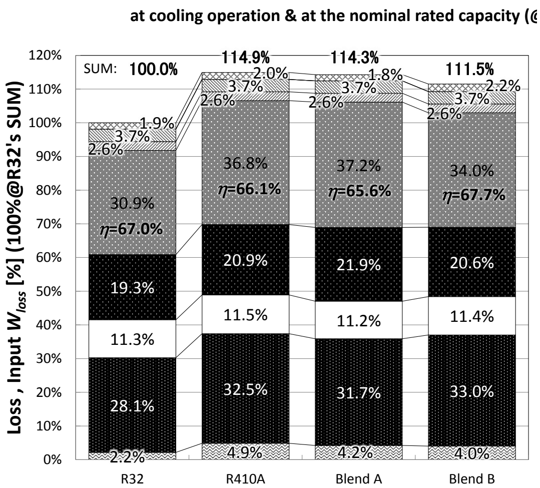
- "Compressor loss" is the loss for the whole compression, which determines compressor efficiency including motor loss here. η in the vertical bars graph mean compressor efficiency, which contain electric loss of V.F.D.
- Last, as mentioned above, system input consists of additional three types of input;
- "Indoor fan input", "Outdoor fan input", and the unanalyzed factors; "others".
- The above seven categories add up as the whole input of the system in this analysis. In addition, "others" is including errors of calculation and measurement.

7.2. Result of Analysis

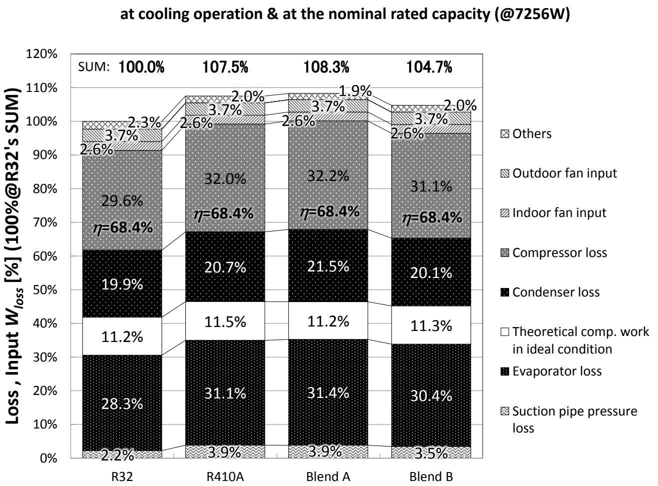
Fig. 6 and Fig. 7 show the loss analysis for experiment and simulation in each refrigerant. It can be recognized that evaporator loss and compressor loss increased 3-5% when using the refrigerants except for R32. This phenomenon takes place both in experiment and in simulation. Meanwhile there was less influence onto the condenser. For example, in case of experiment of Fig. 6, in terms of Blend B, suction pipe loss increased by 1.8%, evaporator loss increased by 4.9%, and condenser loss increased by 1.3%. These losses affected compressor loss and it increased by 3.1% coinciding with increase of whole compression work.

Meanwhile, compression efficiency of blend B was 0.7% better than that of R32. Though compression efficiency generally can be worse by decrease of compressor's suction pressure, compression efficiency of blend B rose with in exception to it.

As the ratio of whole compressor input exceeded 90% of whole system input, compression efficiency declined because of drop in suction pressure affect the system input seriously. It is assumed that degradation of evaporator performance was caused by the influence of pressure loss and temperature glide.



capacity operation by experiment
(Indoor DB/WB: 27°C /19°C, Outdoor DB: 35°C)



capacity operation by simulation
(Indoor DB/WB: 27°C /19°C, Outdoor DB: 35°C)

8. Overview about Safety and Risk Assessment as for R32 Air-conditioners

We discussed about the analysis results in experiment and simulation for each refrigerant. It was reconfirmed that the smaller pressure drop property of R32 results in high performance, particularly in case of mini-split type air-conditioner. On the other hand, R32 requires some cautions because of its A2L flammability, however it is very low compared to highly flammable refrigerants such as propane (R290), so the actual risk while handling it can be assumed to be low. Risk assessment has been carried out for various mini-split types of air-conditioners so far proves that R32 can be used safely enough by establishing safety standards.

In addition, in terms of infrastructure, this action for risk assessment enabled the tools for R32 to be the same as ones for non-flammable refrigerant R410A, as shown in Fig. 8. This indicates that flare type connection can be used also for R32 for piping connection.

However, it is recommended to confirm that the motor of the refrigerant recovery machine does not become an ignition source, and the basic knowledge the operator should know about handling, such as the thermos-physical properties of R32, enlightenment is being promoted by proactively educating workers through the development of manuals.

As a result, in the mini-split type air conditioner, R32 maintains enough high safety and reliability, and in addition to drastic reduction of GWP, as mentioned above, its high performance in actual use compared to conventional refrigerants, makes it an interesting option.

As for mini-split type air conditioners which are for residential use and for business use, Daikin Industries launched the R32 products in Japanese domestic market for the first time in 2012. After that, all other air-conditioner manufacturers in Japan followed it and launched their respective products in the market. Likewise, in Asia, Europe, Australia, and so on, market introductions by major manufacturers are expanding. It is estimated that about 20 million units have already been sold worldwide by many manufacturers. Daikin have sold about 8 million units in 50 countries.

<table>
<tr><th>Tools (*)</th><th>R32</th><th>R410A</th><th>R22</th></tr>
<tr><td>(1) Gauge manifold</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(2) Charging hose</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(3) Scale</td><td colspan="3">Compatible</td></tr>
<tr><td>(4) Pipe bender</td><td colspan="3">Compatible</td></tr>
<tr><td>(5) Flare tool</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(6) Torque wrench</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(7) Pipe cutter</td><td colspan="3">Compatible</td></tr>
<tr><td>(8) Cylinder adaptor</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(9) Vacuum pump</td><td colspan="3">Compatible</td></tr>
<tr><td>(10) Refrigerant recovery unit</td><td colspan="3">Compatible</td></tr>
<tr><td>(11) Refrigerant recovery cylind</td><td colspan="2">Compatible</td><td></td></tr>
<tr><td>(12) Electric gas leak detector</td><td colspan="3">Compatible</td></tr>
</table>

Fig. 8. Service tool compatibility

9. Conclusions

The following results were revealed during our examination:

- COP of Blend B has increased by 4% from that of R410A, however it couldn’t reach as 11% to that of R32.
- As for GWP, Blend B has a similar value to R32, however, as the amount of refrigerant required by the refrigeration cycle system increases approximately by +10%, the actual climate impact of refrigerant in CO2 equivalent may increase by +10%.
- In case of judging the refrigerant’s performance in system from refrigerant property, it is effective to consider “work” equivalent to “pressure loss”. And it is proportional to the square of specific volume in suction and inversely proportional to the cube of refrigerating effect.
- Evaluation of refrigerant’s performance in system should be carried out at a constant capacity, not at a constant compressor speed.
- In case of comparing performance of refrigerant in system by COP especially in various conditions, the system simulation is very effective. The progress of system simulation development will accelerate the evolution of refrigerants.
- We will continue the evaluation of refrigerant performance to find a refrigerant with lower GWP and higher efficiency than R32.

Nomenclature

GWP	Global Warming Potential	$(kg\cdot CO_2)$
ΔT_G	Temperature glide	(K)
	Discharge pressure	(MPa abs)
	Suction pressure	(MPa abs)
	Refrigerating effect	(kJ/kg)
s	Compressor work	(kJ/kg)
COP	Coefficient of performance (= r /)	(–)
	Specific volume	(m^3/kg)
s	Specific volume in suction	(m^3/kg)
ϕ	Volumetric capacity (= /)	(kJ/ m^3)
Δ	Pressure loss at constant capacity	(% of kPa)
ΔW .	Work equivalent to pressure loss at constant capacity	(% of Watt)
	Friction factor	(–)
	Pipe length	(m)
	Pipe diameter	(m)
	Mass flow rate	(kg/s)
ρ	Density	(m)
Φ_0	Refrigeration capacity	(m)
	Pipe diameter	(m)
M_e	Amount of refrigerant charge	(kg)
T	Discharge temperature	(°C)
T_c	Condensing temperature	(°C)
T_e	Evaporating temperature	(°C)

T	Suction temperature	(°C)
$T_{c.\ ut}$	Condenser outlet temperature	(°C)
$\eta_{c\quad p}$	Compressor efficiency	(°C)
DB	Dry bulb temperature	(°C)
WB	Wet bulb temperature	(°C)
W	Inputs for works or losses required to operate system	(% of Watt)
JRAIA	The Japan Refrigeration and Air-conditioning Industry Association	

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