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ABSTRACT

R32 was selected as the alternative for R410A for mini-split air conditioners/heat pumps in Japan and have already phased out R410A in residential market. Recently several new alternatives for R410A have been proposed in order to achieve more close capacity to R410A and mitigate high discharge temperature issue of R32 as well as to reduce energy consumption. We selected R32 from among the candidates a few years ago; however it is important to continue comparing it to new candidates in search for an even better choice.

We carried out the drop-in tests of the new refrigerant R32/R125/R1234yf (67/7/26) by comparing its COP and especially pressure loss to those of R410A, R32/R1234ze(E)(70/30), and R32, using residential mini-split type air conditioner. We conducted the drop-in tests by two types of method, actual measuring and especially simulated calculation. Adjusting compressor suction superheat and the amount of refrigerant charged into the system. Moreover, we measured the performance of the system with changing compressor speed in the wide range by variable frequency drive. Furthermore, we compared electricity consumption at the constant capacities, and analyzed the results precisely by loss analysis.

As a result, we found that the COP of R32/R125/R1234yf (67/7/26) is better than R410A in many conditions and it achieves very close capacity to R410A, but COP at the same capacity is not as high as R32. The reason is that pressure losses are increased by adding R125 and R1234yf to R32. It was clarified, comparing losses with R410A, R32/1234ze(E) (70/30), and R32.

From the above, we confirmed that R32 is still the best choice at present. However, we will continue searching for a better alternative.

Key words: GWP, COP, Refrigerant, Heat pump system, R410A, R32/R1234ze, R32/R125/R1234yf, R32

1. INTRODUCTION

In late years the demand for mitigating global warming impact and energy conservation increased significantly, and 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 CO_2 equivalent in the whole lifecycle of an appliance is essential. Based on this, many researchers of 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, it has excellent thermo-physical properties to achieve better performance of the reversible heat-pump systems, and .We judged at that time it was the best refrigerant among 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]

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

In this paper, we defined those HFO mixed refrigerants as follows.

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

1.1 Properties of the Refrigerants

Table 1-1 shows the properties of four refrigerants which were charged into the test system in this experiment. They are HFC refrigerants, R32 and R410A, while HFO mixed refrigerants R32/R1234ze(E) (70/30) (called as Blend A here) and R32/R125/R1234yf (67/7/26) (called as Blend B here). Showing in Table 1-1, Blend A has temperature glide of 4.4K during phase transition between vapor and liquid. Though Blend B also has temperature glide, it's not as large as Blend A's, and it remains 0.9K. Temperature glide affects the system performance, because the temperature gap between refrigerant and air shrinks. It is expected that Blend B may have better performance than Blend A.

Following to the temperature glide evaluation, we compared the theoretical COP (Coefficient of Performance) a cooling operation cycle. Calculation conditions were Condensing Temperature T_c=45°C, Evaporating Temperature T_e=10°C, Suction pipe Temperature T_s =15°C, Condenser outlet Temperature $T_{c.out}$ =40°C, and Compressor Adiabatic Efficiency η=70%. The results are shown in Table 1-1 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 theory COP requires Refrigerating effect. On the other hand, the larger refrigerating effect per this unit mass w_r, the larger refrigeration capacity tends to become in case of constant compressor speed. In actual, since a compressor sucks gas of amount equivalent to the cylinder volume, system cooling capacity are affected by volume capacity ϕ which is refrigerating effect per suction volume.

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 actual 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 to the work $\Delta W_{P.loss}$. It can be calculated as required work to recover the pressure loss by compressing vapor adiabatically.

In Table 1-1, comparing Pressure Loss ΔP_{loss} of each refrigerant, one with R410A is the largest prominently

Table 1-1: 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℃	0.0	0.1	4.4	0.9
Discharge / Suction Pressure: P_d / P_s [MPa abs]	2.795 / 1.107	2.730 / 1.445	2.366 / 1.233	2.605 / 1.0393
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 [℃]	84.1	69.5	69.1	73.3

*Calculation Conditions: T_c = 45℃, T_e =10℃, Suction line Temp.: T_s =15℃, Condenser Outlet: $T_{c.out}$ =40℃,
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.

among these refrigerants. Pressure Loss ΔP_{loss} of Blend A and Blend B exceeds 140% vs R32 in pressure value. But when focusing onto $\Delta W_{P.loss}$ [W]; Work equivalent to Pressure Loss, that of Blend A and Blend B reaches than 160% vs R32. All of blends have larger pressure loss than R32. It is because refrigerant effect w_r of R32 is significantly larger than R125, R1234ze(E) and R1234yf.

Regarding discharge temperature, R32 has the highest value in this property table. However, since R32 has superior performance to the others in pressure loss and other aspects, discharge temperature in the actual operation does not relatively rise as high as the other refrigerants. And it would not be a significant issue in case of proper system design. Thus R32 is expected to be the best performance refrigerant from these thermo-physical properties and other properties in the Table 1-1.

1.2 Calculation of Pressure Loss in each Refrigerant

As mentioned in foregoing subsections, pressure loss is the one of the key factors which dominate the system efficiency. Moreover, the Work $\Delta W_{P.loss}$ required to recover Pressure Loss ΔP_{loss} at a constant capacity is the most important factor which indicates how small the pressure loss of refrigerant would be. In this section, the way how to calculate the work to compensate pressure loss is explained.

At first, Pressure Loss ΔP_{loss} by fluid flowing inside of pipe, of which length is L[m] and diameter is d[m], is written as bellow (1). Equation about capacity, mass flow rate, and specific volume are also given in (2) and (3).

$$\Delta P_{loss}=f\cdot\frac{L}{d}\cdot\frac{q_{mr}^{\ 2}}{2\rho} \tag{1}$$

$$q_{mr}=\frac{\Phi_0}{w_r} \tag{2}$$

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

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

$$\Delta P_{loss}=\left(f\cdot\frac{L}{2d}\cdot\Phi_0^{\ 2}\right)\cdot\left(\frac{v_s}{\mathrm{w_r}^2}\right) \tag{4}$$

For this pressure loss ΔP_{loss} , estimating the value of work $\Delta W_{P.loss}$ to raise pressure to the pressure before loss by compressing adiabatically.

$$\begin{aligned}\Delta W_{P.loss}&=q_{mr}\cdot\int_{P_1}^{P_2}dh\\&=q_{mr}\cdot\int_{P_1}^{P_2}v\cdot dP\end{aligned} \tag{5}$$

When the pressure change width from P_1 to P_2 is sufficiently small in adiabatic compression, the change width in the compression work due to the change in the specific volume v can be negligible as shown in Figure 1-1. And as here the specific volume v is assumed to be equal to the specific volume at suction side pressure v_s of compressor, the equation (5) simply can be converted into equation (6).

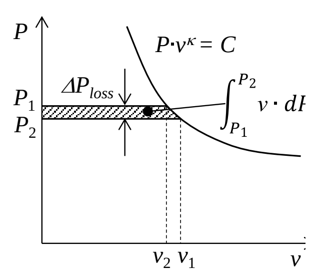

Figure 1-1: Adiabatic compression on *p-v* diagram

$$\Delta W_{P.loss}=q_{mr}\cdot v_s\cdot\Delta P_{loss} \tag{6}$$

Thus this equation (6) can be written as follows equation (7) substituting the equation (2) and (4)

$$\Delta W_{P.loss}=\left(f\cdot\frac{L}{2d}\cdot\Phi_0^{\ 3}\right)\cdot\left(\frac{v_s^{\ 2}}{w_r^{\ 3}}\right) \tag{7}$$

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.

2. TEST SYSTEM FOR ACTUAL MEASUREMENT

2.1 System Outline

Figure 2-1 shows the outline of the system used for the series of the testing. 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 with 7.5 m standard length pipes. This system requires 1.55 kg amount of R32 refrigerant as indicated in Table 2-1.

The Compressor (Comp.) is capable of changing the revolution speed with a Variable Frequency Drive (V.F.D.). The Expansion Valve (Exp. Valve) employed is electrically controlled to change the opening to adjust the mass flow rate entering the evaporator from the condenser, and to adjust the superheat at the compressor suction. The Four-way Valve (4-Way Valve) enables to switch cooling and heating operation by switching condensation/evaporation in the indoor and outdoor heat exchangers. In the diagram, the solid lines inside of the Four-way Valve indicate the flow directions in cooling operation. The gas discharged from the compressor flows into the outdoor heat exchanger, where the gas is cooled down and condenses into liquid state. Then, the liquid is expanded, and lower the temperature of itself at the Expansion Valve. After that, the liquid is heated up and vaporized into gas in the indoor heat exchanger, and the gas from the indoor heat exchanger returns to the compressor to be compressed again.

During the tests, measuring capacity of this system was conducted with 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 gages 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 temperature was measured.

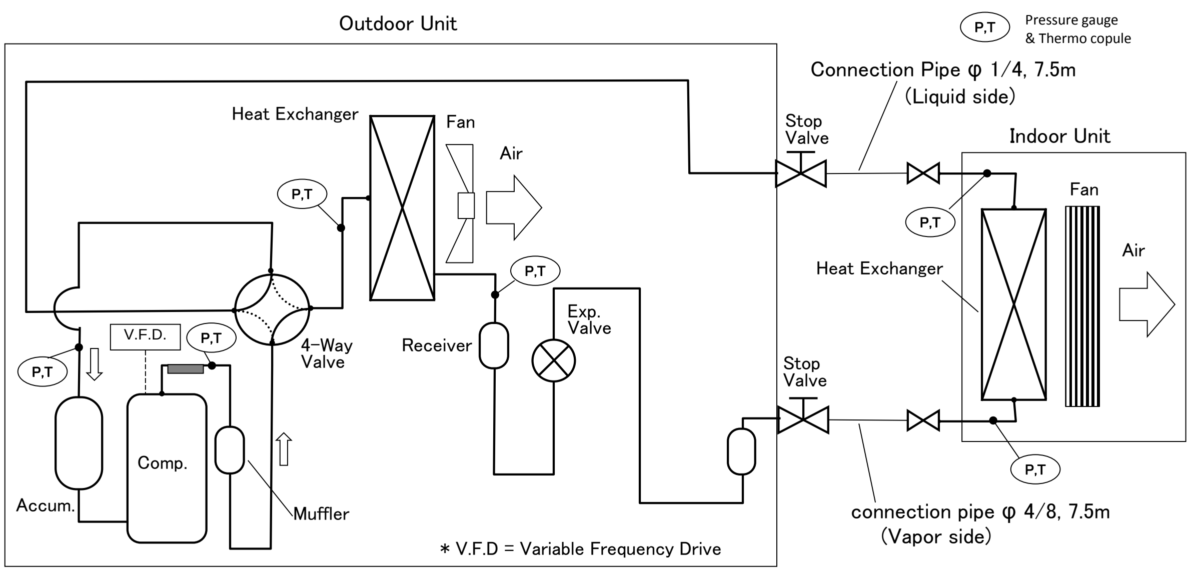

Figure 2-1: Test System Diagram

Table 2-1: 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 2-2: Test Conditions

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

2.2 Test Conditions

Table 2-2 shows the test conditions on cooling more based on ISO 5151-2010. Before measuring the performance of the test system with each refrigerant, we adjusted amount of refrigerant at the optimum for the condition. Several times of trials with different charge were made to find the optimum amount of refrigerant for the COP. The results of refrigerant's amount are described in Table 2-1. Moreover, we adjusted the opening ratio of the expansion valve to achieve the highest COP. In this way, we could compare the systems optimized for each refrigerant.

3. DROP-IN TEST RESULTS FOR WIDE CAPACITY RANGE

As described previously, the tests were conducted in cooling mode for four refrigerants, and the test data were acquired and compared each other.

3.1 COP trend comparison in the wide capacity range

Figure 3-1 shows the COP comparison of each refrigerant in capacity trend. This is the system performance measured with changing the compressor speed, to enable the comparison of COP values at various capacities.

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 increase, COPs tend to relatively lower vs R32. The capacity increases more, refrigerant mass flow rate increases more, and the loss originated from pressure loss also increases. Thus the superiority of R32 is apparent in higher capacity as much as refrigerant properties indicate.

Meanwhile, because the pressure loss gets 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 did not shrink as much as others. This is the result that the temperature gap between refrigerant and air couldn't be smaller, and this is considered to be caused by the larger temperature glide of Blend A, 4.4K.

From the above, it could be mentioned that pressure loss becomes serious issue when in larger capacity, and temperature glide becomes issue too when capacity is smaller. Therefore, generally adding lower capacity component to base components negatively affects in COP in refrigeration cycle due to larger pressure loss. If it causes glide, the negative impact appears significantly in low load condition in heat exchanger such as low capacity operation or high COP operation.

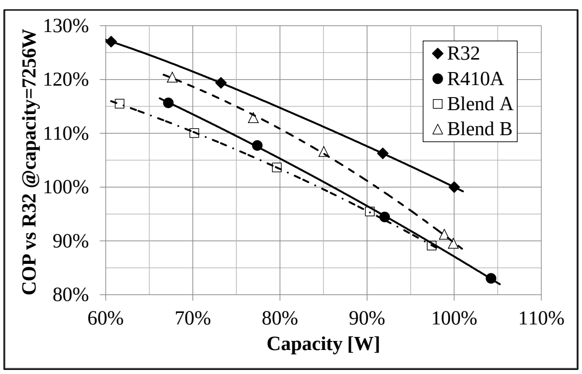

Figure 3-1: COP trend comparison of each refrigerant for Capacity.

4. DROP-IN TEST RESULTS AROUND THE RATED CAPACIY

In this chapter, we explain how the drop-in tests were conducted and how the test results were analyzed. In addition, we conducted the drop-in tests using the system simulation software, and compared two types of "drop-in" tests by actual measurement and by simulation, in which we clarified the differential between constant speed and variable speed about compressor.

4.1 The method of the Drop-in Test

At first, the definition of the conducted "drop-in" test is necessary before the explanation of the results. In case of conducting the drop-in test to full optimized test of a refrigerant, following are listed as adjustable items.

a) Refrigerant amount (Sub-cool degree)
b) Expansion valve open ratio (Superheat degree)
c) Compressor speed
d) Fan speed
e) Heat exchanger preferences(Heat conducting pipe's diameter、a number of Paths, etc.…)
f) Compressor preferences (size of cylinder, motor,)
g) Diameter of connection pipes
h) etc.

However, even b) Expansion valve open ratio and c) Compressor speed were unadjustable in the most drop-in tests performed in the past, since they employ capillary tube or a fixed orifice as expansion device, and a fixed speed compressor. In the worst case, even charge amount was not optimized. Such drop-in tests do not give correct results about potentials of refrigerants.

When comparing the performances of the system using vapor compression cycle, a proper subcool at inlet of expansion valve and a proper superheat at the compressor suction are required to expand refrigerating effect w_r and the temperature differential between a condenser and an evaporator is kept narrow sufficiently. If the amount of refrigerant becomes excessive, a heat transfer area for saturated vapor in a condenser decreases and condensation pressure rises. This makes the compressor consumption rise. Meanwhile, shortage of refrigerant makes refrigerating effect shrink, and the system capacity decreases.

In addition, the refrigerant should be completely vaporized by reaching to the suction for reliability of compressor and also for the performance. In order to maximize the performance of the system, the liquid refrigerant should be evaporated into vapor in the evaporator sufficiently as far as evaporator performance would not fall.

After having replaced a refrigerant to another refrigerant, we carried out a performance comparison tests as maximizing COP by adjusting a) Refrigerant's amount and b) Expansion valve open ratio. Furthermore, we adjusted c) Compressor speed to match its capacity to the rated capacity, and compared COP. And we call it “Drop-in test” here.

4.2 Outline of Simulation

Figure 4-1 is the outline of refrigeration cycle on the simulation.

This is the system simulation software which was developed by Dr. Kiyoshi Saito laboratory of Waseda University. Its software's name is “Energy Flow +M Core System.” [5]

As shown in Figure 4-1, Heat exchangers are divided into two rows of windward row and leeward row. This helps calculating clearly about the wind flow direction for the refrigerant flow. To the outlet of evaporator, we connected connecting pipe and suction pipe in order to calculate the influence of pressure loss accurately.

And We Placed PID controller to adjust the superheat at the outlet of evaporator in order to give target open ratio to expansion valve. Moreover, if it is necessary to adjust the capacity, it can be controlled to the target capacity automatically by PID controller.

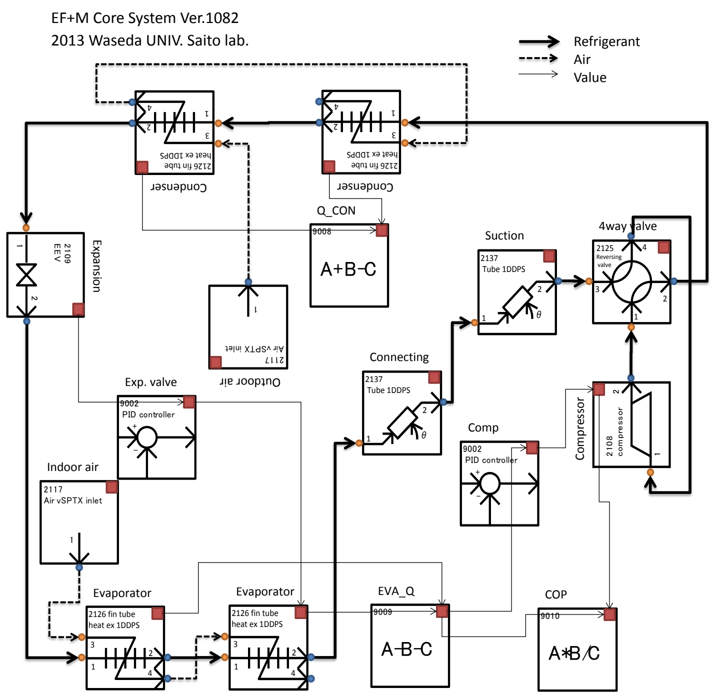

Figure 4-1: Outline of Refrigeration cycle on the simulation

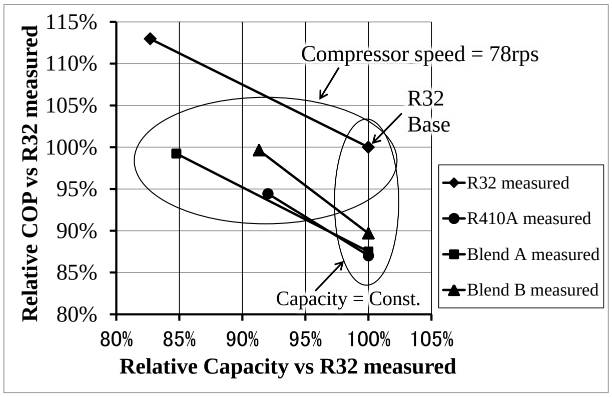

Figure 4-2: COP trend comparison of each refrigerant for Capacity by actual measurement.

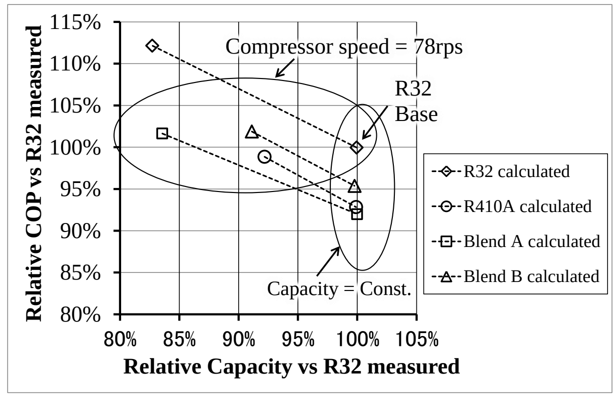

Figure 4-3: COP trend comparison of each refrigerant for Capacity by simulation

4.3 Results of Drop-in test by Actual Measurement

Figure 4-2 shows the results in case that the compressor speed was constant as 78rps, and in case that the system cooling capacity was equal by adjusting the compressor speed.

As a result, the comparison in constant compressor speed 78rps gives the data that the refrigerants except for R410A have COPs near R32.

Although vapor compression refrigeration cycle has the characteristics that condense pressure rise and evaporation pressure lower as capacity increases. This expansion of pressure gap results in increase of compression work and decline of COP. Thus a comparison with the equal capacity is necessary for a correct comparison of refrigerant.

4.4 Results of Drop-in test by Simulation

Figure 4-3 shows the comparison of each refrigerant for capacity by simulation. It can be found that the simulation gave the result approximately as same as actual measurement. The plotted data surrounded by the circle has the same compressor speed 78rps.

At first, we conducted inverse simulation at the R32 base point. This made both of COPs by measurement and by simulation fitted each other by adjusting the coefficients of pressure loss in evaporator and suction pipe, the wind volumes through the condenser and the evaporator, and the volumetric efficiency and adiabatic efficiency of compressor.

Next, we checked the COP when changing the capacity widely, for example, at 83% capacity using R32. From both of the COPs at 83% capacity, we found that the simulation could give the proper result.

The data plotted in Figure 4-3 were calculated in the condition equal to the condition in which Figure 4-2 were actually measured. The results of calculation seem to be approximately equal to the results of measurement.

The performance of other refrigerants seem to become around 2% higher than actual measurement when regarding COPs of other refrigerants to R32 basement data 78rps. In this simulation, as we did not change the compressor efficiency, it is speculated that the influence resulted in residuals. When raising compressor speed in the refrigerant except R32 in particular, a difference of COP in the case of actual measurement grows larger.

For example, in Blend B, relative COP which was at 90% in the case of actual measurement remains in 95% by the simulation for R32, and simulation gives better performance than the actual measurement. This is assumed to be caused by, for example, influences of compressor oil or differences of distribution of air volume in evaporator, etc.

5. LOSS ANALYSIS IN EACH REFRIGERANT

5.1 Calculation Method

Figure 5-1 and Figure 5-2 shows 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; the one necessary for "Theoretical Adiabatic Compression Work" and the other for "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 for the evaporator's intake air temperature to the saturation vapor pressure for the condenser's intake air temperature. This is inevitable work as far as the temperature gap between air and refrigerant exists.
- "Condenser Loss" is from saturation vapor pressure for condenser's intake air to condensing pressure.
- "Others" is the unanalyzed factors

Second, regarding the latter "Compression Loss",

- "Compression Loss" is the loss for the whole compression, and determines compressor efficiency, including Motor Loss here. h in this box means compressor efficiency ,which contains electric loss of V.F.D.

Last, as mentioned above, system input consists of three types of input;

- "Indoor Fan Input", "Outdoor Fan Input"

The above seven categories add up as the whole input of the system in this analysis.

5.2 Result of Analysis

Figure 5-1 and Figure 5-2 shows the loss analysis for actual measurement and simulation in each refrigerant. It can be recognized that evaporator loss and compressor loss increased 3-5% when using the refrigerants except R32. This phenomenon takes place both in actual measurement and in simulation. Meanwhile there was less influence onto the condenser.

For example, in case of actual measurement of Figure 5-1, in terms of Blend B, suction pipe loss increased as 1.8%, evaporator loss increased 4.9%, and condenser loss increased 1.3%. These losses affected compressor loss and it increased as 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 overturning it.

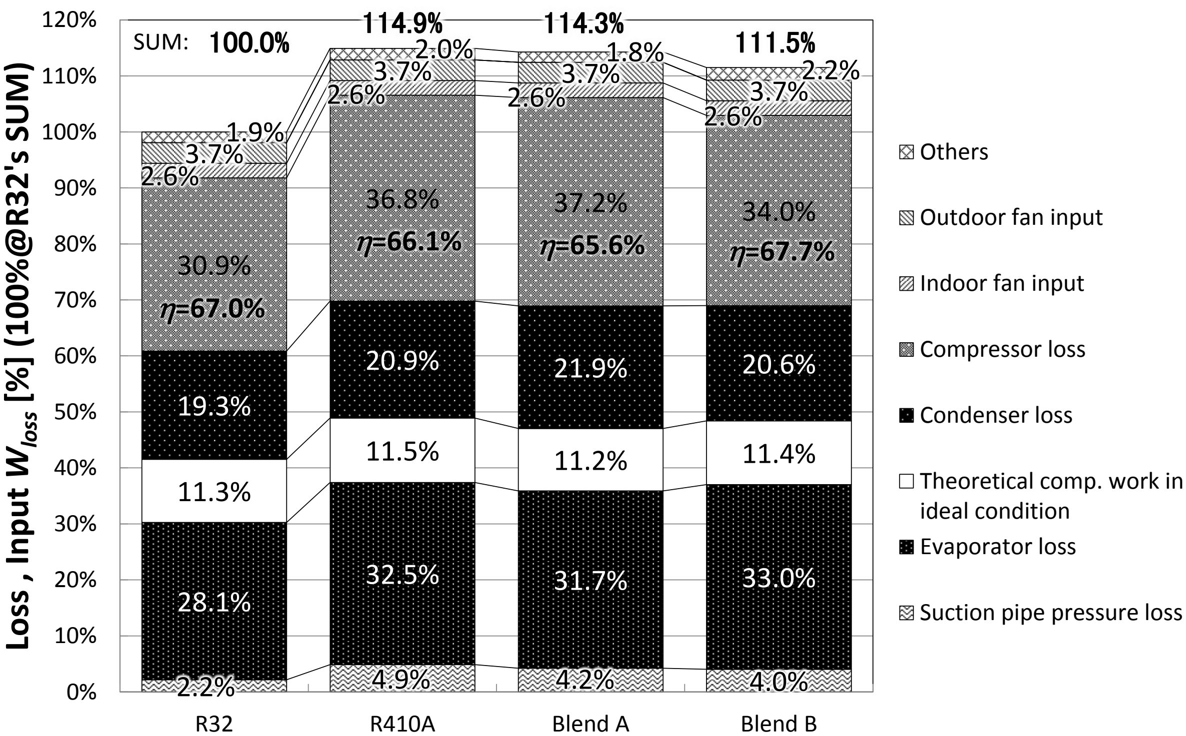

Figure 5-1: Comparison of Loss Analysis in case of each Refrigerant during Cooling Nominal Capacity (7256 W) Operation by actual measurement (Indoor DB/WB: 27℃/19℃, Outdoor DB: 35℃)

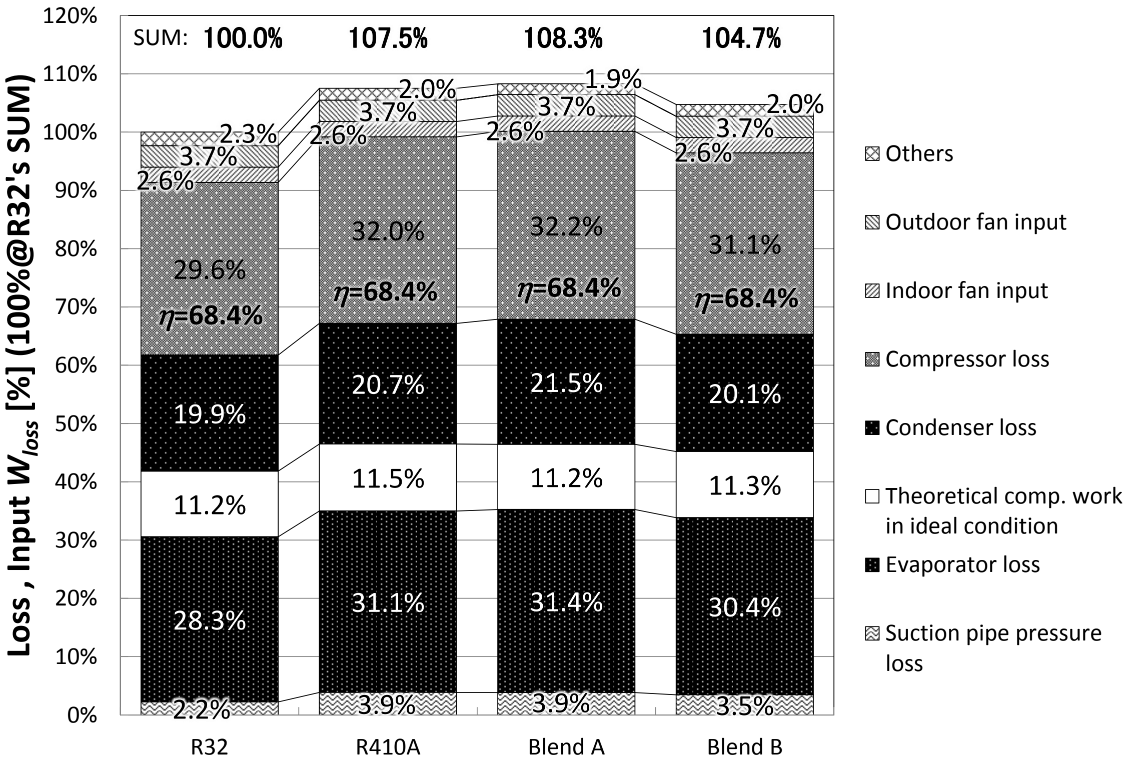

Figure 5-2: Comparison of Loss Analysis in case of each Refrigerant during Cooling Nominal Capacity (7256 W) Operation by simulation (Indoor DB/WB: 27℃/19℃, Outdoor DB: 35℃)

Though the reason of this rise is now known at this time, this rise could have concerning with condenser loss. Condenser loss in actual measurement increased 1.3% from 19.3% of R32 to 20.6% of Blend B in Figure 5-1. On the other hand, Condenser loss in simulation increased only 0.2% from 19.9% of R32 to 20.1% of Blend B in Figure 5-2. About other refrigerants, this phenomenon could not be found.

By the way, in spite of this advantage for blend B, as the ratio of whole compressor input exceeded 90% of whole system input, the decline of compression efficiency which was caused by drop of suction pressure affect to system input seriously. It is assumed that degradation of evaporator performance was caused by the influence of pressure loss and temperature glide.

Moreover, it is speculated that the decline of compressor suction pressure made compressor efficiency lower.

6. CONCLUSIONS

The following results were revealed during our examination:

- COP of Blend B has increased 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 which the refrigeration cycle system requires increases approximately +10%, the actual climate impact of refrigerant in CO2 equivalent increases +10%.
- In case of judging the performance of refrigerant in system from refrigerant property, it is effective to consider "work" equivalent to "pressure loss". And it is proportional to a square of specific volume v_s and inversely proportional to a cube of refrigerating effect w_r.
- When it comes to comparing performance of refrigerant in system, the simulation of refrigeration cycle is very effective to evaluate it.
- Evaluation of Refrigerant's performance in system should be carried out at a constant capacity, not at a constant compressor speed. This is because of the series of evaluations both by actual measurement and by simulation.

・This is because as the kind of refrigerant and compressor speed affects compressor efficiency in actual measurement, compressor performance have to be evaluate properly for accurate comparison. Moreover, the influence of compressor oil for compression efficiency and heat conductance of heat exchanger is necessary to be considered due to more accurate simulation.
・In search for the refrigerant of which GWP is lower or of which efficiency is better than R32 , we will conduct evaluation of refrigerant performance.

NOMENCLATURE

GWP	Global Warming Potential	($kgCO_2$)
ΔT_{GL}	Temperature Glide	(K)
P_d	Discharge Pressure	(MPa abs)
P_s	Suction Pressure	(MPa abs)
w_r	Refrigerating Effect	(kJ/kg)
w_s	Compressor Work	(kJ/kg)
COP	Coefficient of Performance ($= w_r \ / \ w_s$)	(–)
v_s	Specific Volume in suction	(m3/kg)
ϕ	Volumetric Capacity ($= w_r \ / \ v_s$)	(kJ/ m3)
ΔP_{loss}	Refrigerant Pressure loss	(% of kPa)
$\Delta W_{P.loss}$	Work equivalent to Pressure Loss	(% of Watt)
M_{ref}	Amount of refrigerant charge	(kg)
T_d	Discharge Temperature	(℃)
T_c	Condensing Temperature	(℃)
T_e	Evaporating Temperature	(℃)
T_s	Suction Temperature	(℃)
$T_{c.out}$	Condenser outlet Temperature	(℃)
η_{comp}	Compressor Efficiency	(℃)
DB	Dry Bulb Temperature	(℃)
WB	Wet Bulb Temperature	(℃)
W_{loss}	Inputs for Works or Losses required to operate system	(% of Watt)

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