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Evaluation of Performance of Heat Pump System using R32 and HFO-mixed Refrigerant

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ABSTRACT

Recently several refrigerants have been lined up as candidate alternatives to R410A in order to prevent further global warming and to promote saving energy. We have selected R32 from among these candidates; however it is important to continue comparing it to other refrigerants in search for an even better choice.

Therefore we carried out evaluation of the new R32/R1234ze(50/50) refrigerant by comparing its COP and compressor discharge temperature etc. to those of R410A, and the results of R32. We also conducted analysis regarding the drop of COP during Cooling operation in case of high outside temperature. As a result, we found that the COP of R32 is superior to the other refrigerants due especially to its small amount of pressure loss. As the discharge temperature for both the R32/R1234ze(50/50) refrigerant and R32 was roughly the same, we re-confirmed that there is no problem using R32.

Additionally, for non-azeotropic refrigerants such as R32/R1234ze(50/50), a Temperature Glide occurs during phase change. During the current evaluation, we came across a number of performance degradation phenomena thought to be caused by this glide.

From the above, we confirmed that R32 is indeed the best candidate at present. We will continue to carry out evaluation in search for a better candidate.

Key words: GWP, COP, Refrigerant, Heat pump system, R410A, R32/R1234ze, R32, Non-azeotropic

1. INTRODUCTION

In late years the demand for prevention of global warming and energy preservation increased significantly, and we chose R32 as a new refrigerant for heat pump systems. However, it is expected that the demand for air conditioning will continue to increase in the future, and thus reducing CO_2 discharge in the whole lifecycle of the apparatus is inevitable. Based on this, many researchers of the air conditioning industry and academic researchers continue searching for new refrigerants. The reason why we chose R32 is that its GWP (Global Warming Potential) is 1/3 as small as that of R410A, it is suitable to improve the performance of the Heat-pump system, and at present it is superior to other refrigerants from the viewpoint of safety and economy. Because there is no concern about mixture ratio, R32 is easy to manage, furthermore it is attractive even from the viewpoint of collection and reuse.

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]

Therefore at this time, we chose R1234ze as a candidate among HFO refrigerants. We charged the refrigerant mixed with R32 by the same amount into a mini split air-conditioner and compared which refrigerant enables achieving the highest performance.

1.1 Characteristics of the Refrigerants

Table 1-1 shows the properties of each refrigerant which was charged into the test system in this experiment. We used 3 refrigerants. One is R410A as the base refrigerant, the others are R32 and R32/R1234ze(50/50A) as candidate alternatives. R32/R1234ze(50/50) refrigerant which is a 50% - 50% mix of R32 and R1234ze. This R32/R1234ze(50/50) refrigerant is non-azeotropic and has a Temperature Glide during phase conversion between vapor and liquid. For example, under the saturation pressure where the midpoint temperature of saturated liquid and dry saturated vapor is 20.0℃, the saturated liquid temperature is 17.8℃, the dry saturated vapor temperature is 22.2℃, and the Temperature Glide ΔT_{GL} becomes as large as 4.4 K. Because temperature of non-azeotropic refrigerant decreases during condensing and increases during evaporating, the temperature tends to approach air temperature as approaching outlet of the heat-exchanger, and it is speculated to affects the system performance.

Next, we compared the calculated COP (Coefficient of Performance) during Cooling operation. Calculation conditions were Condensing Temperature T_c=45℃, Evaporating Temperature T_e=20℃, Suction pipe Temperature T_s =25℃, Condenser outlet Temperature $T_{c.out}$ =40℃, and Compressor Efficiency η=70% . The results are shown in Table 1-1 below. Figure 1-1 is a specific enthalpy-pressure diagram comparing each refrigerant under the same conditions. Regarding saturation pressure, using the same midpoint temperature for calculation, R32/R1234ze(50/50) has the best COP of 6.96 (103% compared to R410A): an excellent value. However, this is because midpoint temperature of two-phase region is applied as evaporating or condensing temperature. If dry saturated vapor temperature is applied to evaporating temperature, the evaporator's outlet temperature (dry saturated vapor temperature) is lowered to 20℃ from 22.2℃, the midpoint temperature of evaporator drops to 17.8℃, and the COP of the R32/R1234ze(50/50) refrigerant drops to 6.27 (90% compared to R410A). As the Temperature Glide causes the rise of the refrigerant temperature at the evaporator outlet side, the Temperature Gap from the air shrinks. It is assumed that the evaporator's performance should drop and the practical evaporating pressure should fall. Therefore, regarding a non-azeotropic refrigerant, as calculated COP can easily differ from COP in practical use, judging superiority and inferiority in each refrigerant becomes difficult.

In Figure 1-1, the specific Enthalpy-Pressure diagram for R32/R1234ze(50/50) shows that the Refrigerating Effect w_r is larger than for R410A and near to R32, however its evaporation pressure is clearly lower than that of the other refrigerants, and thus the Volume Capacity ϕ also falls to 91% of that of R410A as shown in Table 1-1. It is

Table 1-1: Calculated Properties of Refrigerants Charged to the Test System

Refrigerant	R410A =R32/R125(50/50)	R32 (Pure)	R32/R1234ze (50/50)
Temperature Glide: ΔT_{GL} [K] @ 20℃	0.1	0.0	4.4
Discharge / Suction Pressure: P_d / P_s [MPa abs]	2.730 / 1.445	2.795 / 1.475	2.369 / 1.233
Refrigerating Effect w_r [kJ/kg]	166.3 (100%)	248.9 (150%)	216.4 (130%)
Compressor Work: w_s [kJ/kg]	24.5 (100%)	36.0 (147%)	31.1 (127%)
Coefficient of Performance: COP = w_r / w_s	6.79 (100%)	6.91 (102%)	6.96 (103%)
Specific Volume in Suction v_s [m^3/kg]	0.0184 (100%)	0.0255 (139%)	0.0262 (142%)
Volume Capacity $\phi = w_r / v_s$ [kJ/ m^3]	9059(100%)	9759(108%)	8257(91%)
Pressure Loss at constant capacity: ΔP_{loss} [% of kPa]	(100%)	(62%)	(84%)
Discharge Temperature T_d [℃]	60.3	67.6(+7.3)	63.1(+1.8)

*Calculation Conditions: T_c=45℃, T_e=20℃, Suction line Temp.: T_s =25℃, Condenser Outlet: $T_{c.out}$=40℃, Compressor Efficiency: η_{comp} =70%, in Cooling Operation.
Saturation temperature of mixed refrigerant is midpoint temperature of two-phase region under constant pressure.

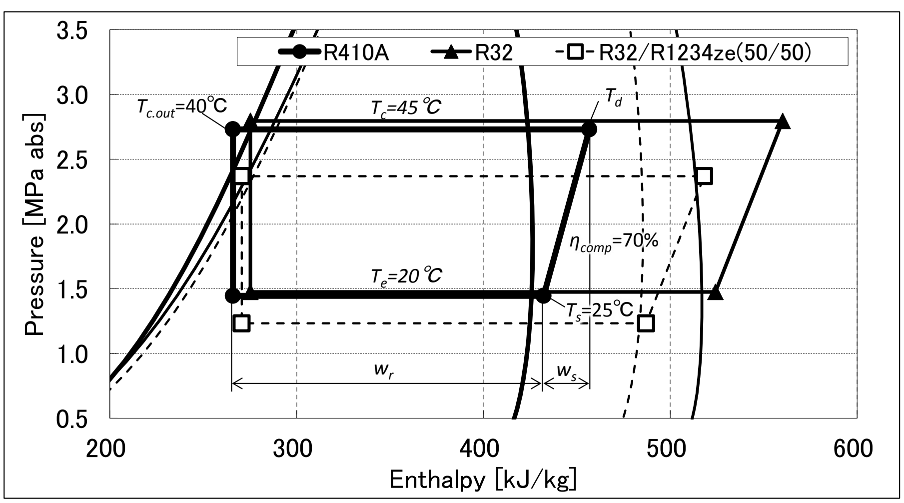

Figure 1-1: Theoretical Refrigeration Cycles with each Refrigerant on Enthalpy-Pressure diagram

speculated that low Suction Pressure requires faster compressor speed for the constant capacity, and the Compressor Efficiency η_{comp} tends to be worse. Meanwhile, the Volume Capacity ϕ of R32 was 108% of that of R410A, and it means that a capacity growth of +8% can be expected at the constant Compressor Speed by charging R32 into a R410A system.

2. TEST SYSTEM

2.1 System Outline

Figure 2-1 shows the outline of the system used for the series of testing. It is a small-sized split-type system with a nominal cooling capacity of 4.0 kW, and the indoor unit and outdoor unit is connected with 5 m length connection pipes. This system requires 1.34 kg amount of R32 refrigerant as also 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) is electrically controlled to change the opening rate, enabling to adjust the mass flow rate entering the evaporator from the condenser, and to adjust the amount of super-heating at the compressor suction side. 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 continuous 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 by 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.

Table 2-1: Charged Amount of Refrigerant in the Test System

	R410A	R32	HFO-mixed
Amount of Refrigerant Charge M_{ref} [kg]	1.67 kg (100%)	1.34 kg (80.2%)	1.41 kg (84.4%)

Table 2-2: Test Conditions

Operating mode	Capacity	**Indoor Ambient**		**Outdoor Ambient**	
		DB(℃)	WB(℃)	DB(℃)	WB(℃)
Cooling	Nominal (4.0 kW) / Half of nominal	27	19	35	24
Heating	Nominal (5.0 kW) / Half of nominal	20	(15 max)	7	6

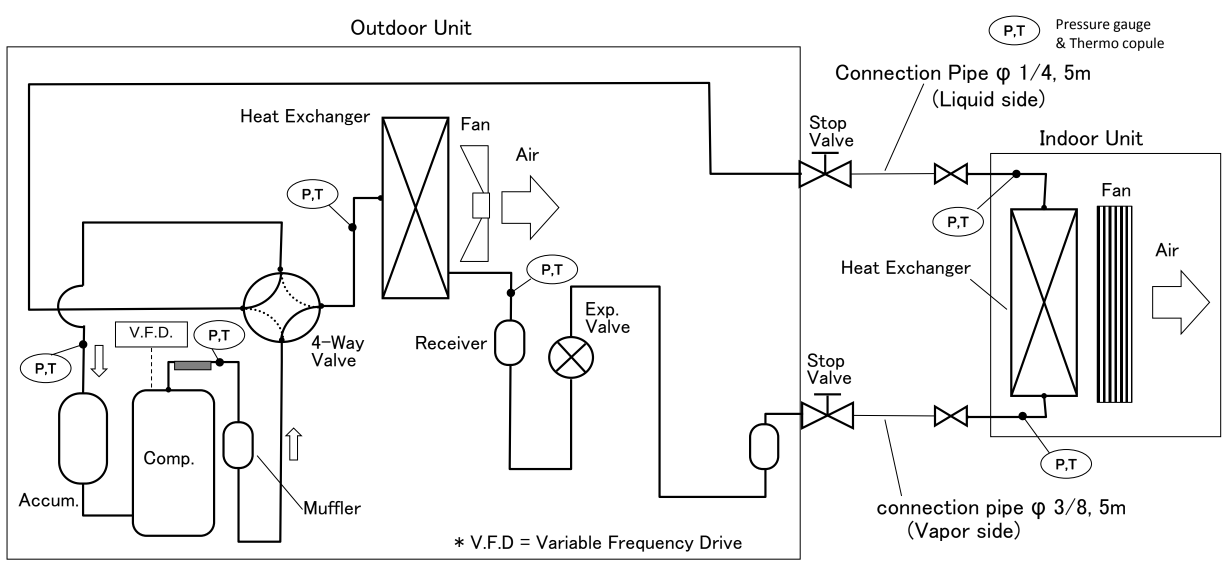

Figure 2-1: Test System Diagram

During the test, measuring the 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 the temperature and the pressure by T-type thermocouples and pressure gages at the discharge pipe and suction pipe of the compressor and the inlet and outlet of the heat exchanger. At the midpoints of the heat exchangers, only temperature was measured.

2.2 Test Conditions

Table 2-2 shows the four test conditions based on ISO 5151-2010. Operating modes are Cooling and Heating, and the capacity in each operating mode is the nominal rated or half the nominal capacity.

Before measuring the performance of the test system with each refrigerant charged, we added a little amount of refrigerant several times to find the optimum amount of refrigerant for the COP. The results are described in Table 2-1. Moreover, we adjusted the opening ratio of the expansion valve to achieve the amount of super-heating in the suction pipe which allows the best COP with each amount of refrigerant. In this way, we could compare the systems optimized for each refrigerant.

3. TEST RESULTS FOR EACH REFRIGERANT

As described previously, the tests were conducted for three kinds of refrigerants, and the results were acquired and compared in each operating mode and with each refrigerant.

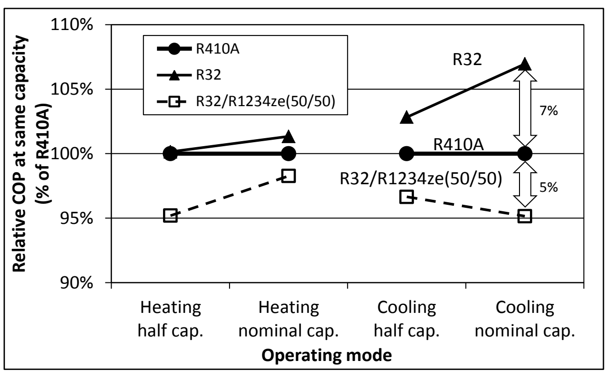

Figure 3-1: Relative COP in each Operating mode

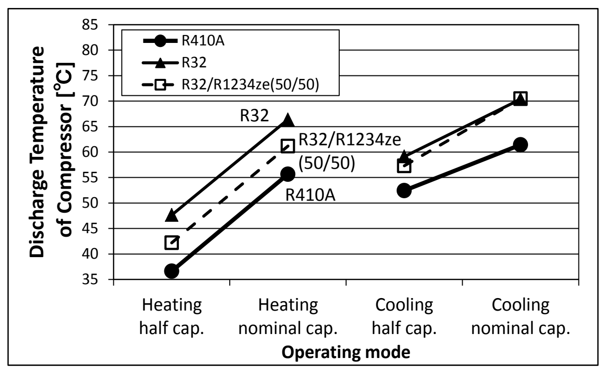

Figure 3-2: Discharge Temperature in each Operating mode

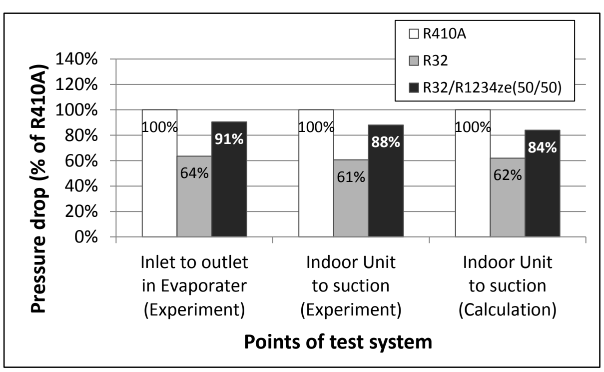

Figure 3-3: Relative Pressure Drop in Low Pressure Side (in Cooling, nominal capacity)

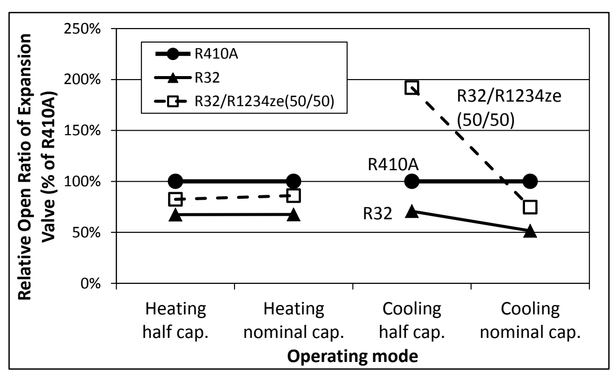

Figure 3-4: Relative Opening Ratio of Expansion Valve

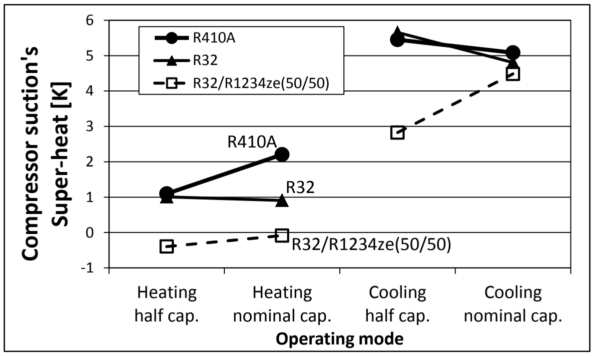

Figure 3-5: Compressor Suction's Super-heat

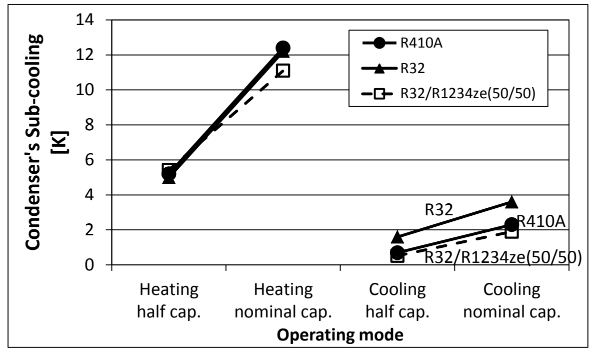

Figure 3-6: Condenser's Sub-cooling

3.1 System COP

Figure 3-1 compares each COP for nominal rated and half the nominal rated capacity in both Cooling and Heating operation, with each refrigerant. Defining R410A as the standard (100%), it can be seen how the COP for R32 and R32/R1234ze(50/50) differs at the same capacity level.

The COP of R32 at Heating at half the nominal capacity was nearly the same as of R410A. However, under other operating conditions, the COP of R32 was higher than that of R410A. It especially improved more at the rated capacity, where the amount of refrigerant flow rate is larger than at half capacity. Also, it improved more in Cooling operation than in Heating operation, thought to be due to the low pressure loss characteristic of R32. In general, the larger the amount of flow rate is the larger the pressure loss gets. Thus the larger the capacity is, the more the COP improvement of R32 gets.

In Cooling operation, the indoor heat exchanger works as an evaporator, and the pressure loss in the piping connecting the indoor unit and outdoor unit lowers the compressor suction pressure directly and causes COP to decrease significantly. That is, COP improvement of R32 in a split-type system is advantageous especially in Cooling operation, and at high capacity.

On the other hand, the COP of R32/R1234ze(50/50) was worse as 5% than R410A. According to Table 1-1, R32/R1234ze(50/50)'s calculated pressure loss is as much as 84% of R410A. This is only about 1/3 of the improvement of R32. Moreover, due to the effects of the Temperature Glide of R32/R1234ze(50/50) itself, the performance of the heat exchanger tends to decline. Especially if the Temperature Gap between air and refrigerant is small (for example, the capacity is small such as under half the nominal capacity, or at the evaporator in Heating operation), this Temperature Glide causes the Temperature Gap to shrink extremely in some places. As heat flux is proportional to Temperature Gap, heat exchanger performance tends to decline. Details on actual heat exchanger temperature, etc. will be discussed later, in paragraph 3.5.

3.2 Discharge Temperature

According to the calculation results in Table 1-1, R32 shows Discharge temperature rise is +7.3℃ compared to R410A. On the other hand, the temperature rise for R32/R1234ze(50/50) remains in +1.8℃, suggesting its superiority. However according to the test results that Figure 3-2 shows, the discharge temperature of R32/R1234ze(50/50) during Cooling operation rose to approximately the same level as of R32. Thus the superiority of R32/R1234ze(50/50) regarding discharge temperature was not shown in all operating modes.

The reason for this could be, as described in 3-1, that the COP of R32/R1234ze(50/50) in rated capacity of Cooling operation was 12% smaller than that of R32. The reason for this will be discussed in paragraph 4.2 in precise, but the fact that its pressure ratio of high pressure and low pressure was larger than for other refrigerants, may have led to a rise in discharge temperature.

3.3 Pressure Drop in the Evaporator and Suction Pipe

Figure 3-3 shows the comparison of experimental pressure drop at the low pressure side in each refrigerant and the comparison of calculated pressure drop listed in Table 1-1. Regarding pressure drop, the calculations corresponded to the test results, and the pressure drop of R32 is apparently the smallest among these refrigerants. That is, it can be expected that the larger the capacity is, the more advantageous R32 can be for system performance.

On the other hand, the pressure drop of R32/R1234ze(50/50) was 84% of R410A by calculation. But, as saturated pressure during evaporation is low by its refrigerant characteristic, the pressure drop amount per evaporating pressure of R32/R1234ze(50/50) becomes larger than that of R32 and R410A. Thus, when using R32/R1234ze(50/50), the pressure ratio between discharge and suction tends to rise; its COP lows, and its discharge temperature rises

3.4 Expansion Valve Opening

Figure 3-4 shows the comparison of the relative opening ratio of the expansion valve to optimize system COP for each refrigerant and each operation mode, and R410A is defined as the standard (100%). The opening ratio of R32/R1234ze(50/50) decreased, but the decrease was not as small as that of R32. Additionally, this was same as in the results for pressure drop, and the opening ratio was also similar, except for Cooling operation at half the nominal capacity.

On the other hand, the opening ratio required at half the nominal capacity in Cooling operation is twice as much as for R410A. This is thought to be because achieving sub-cooling is more difficult due to the effects of the Temperature Glide of R32/R1234ze(50/50). As shown in Figure 3-6, in Cooling operation with half the nominal capacity, the Sub-cooling for any refrigerant was close to 0.0 K, however it was especially low in the case of R32/R1234ze(50/50). It is deduced that this Sub-cooling shortage generated flash gas, moreover the flow rate through the valve dropped.

Compared to R32, R32/R1234ze(50/50) enables refrigerating cycle operation with the expansion valve opening ratio close to that of R410, however if required reliable control or the use of simple expansion devices such as capillary tubes, the risk of capacity drop caused by flow rate insufficiency must be considered.

Also, as shown in Figure 3-5, the decrease of suction super-heat caused by Temperature Glide. The temperature rise of refrigerant during evaporation makes it difficult to obtain super-heat degree. As a result, enthalpy at the evaporator outlet decreases, and the system capacity may decrease. Insufficient suction super-heat degree is by nature easy to occur during operation where the Temperature Gap between air and refrigerant is small, and as shown in Figure 3-5, must be considered as a possible problem in Heating operation or when the load is small.

3.5 Preventing the Outdoor Heat-Exchanger from Freezing-up in Heating Operation

For example, in the winter season; when outdoor temperature is low, the heat-exchanger of the outdoor unit may sometimes freeze up, and the Heating capacity may decrease. This phenomenon is due to the temperature drop of the evaporator (outdoor heat exchanger), and it was seen that this temperature drop is easy to happen especially in systems using non-azeotropic refrigerant.

As seen in Figure 3-7, although the midpoint temperature of the evaporator using R32 or R410A could be kept above 1.5℃, while it dropped as low as 0.0℃ in case of R32/R1234ze(50/50). Accordingly, because in R32/R1234ze(50/50) systems, increasing the capacity results in a high risk of freezing up on the evaporator, it can be said that the rated capacity (5.0 kW) of this system is almost the upper limit of capacity in case that R32/R1234ze(50/50) refrigerant is used.

This also points out that R32/R1234ze(50/50) is not suitable for high capacity Heating operation. As a further assumable problem, in case of using reverse-cycle defrosting in order to melt the frost, temperature drop by Temperature Glide may occur around the refrigerant outlet of the outdoor heat exchanger, which may cause the frost to remain partially without melting the frost.

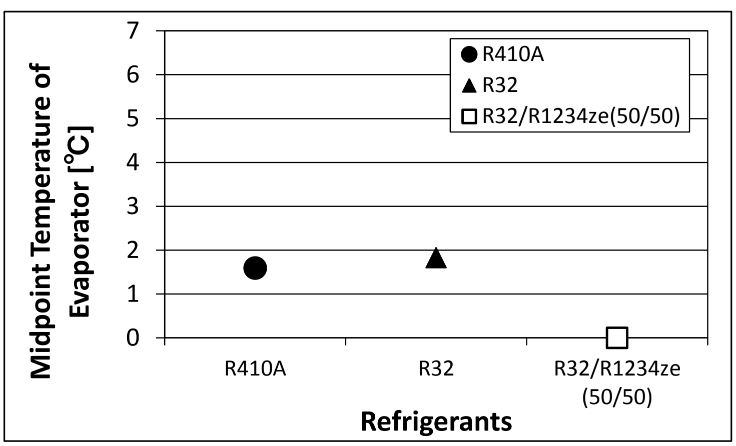

Figure 3-7: Midpoint Temperature of Evaporator
in Heating Operation at the Nominal Capacity

4. LOSS ANALYSIS IN EACH REFRIGERANT

4.1 Calculation Method

Figure 4-1 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, outdoor fan input, and outdoor electrical component 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" consists of five parts. When considering enthalpy transition during compression, the five parts are below, in order of decreasing vapor pressure:

・"Discharge pipe Pressure Loss" is from condensing pressure to discharge pressure.

・"Condenser Loss" is from saturation vapor pressure for condenser's intake air to condensing pressure.

・"Theoretical Compression Work under the 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.

・"Evaporator Loss" is from evaporating pressure to saturation vapor pressure for evaporator's intake air.

・"Suction pipe Pressure Loss" is from suction pressure to evaporating pressure.

Second, regarding the latter "Compression Loss",

・"Compression Loss" is the loss for the whole compression, and determines compressor efficiency, including Motor Loss here.

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

・"Indoor Fan Input", "Outdoor Fan Input", "Electrical Component Loss" (except for compressor input)

The above ten categories add up as the whole input of the system.

4.2 Result of Analysis

Calculation results are shown in Figure 4-1, where the R410A system's input sum is set as the base of 100%.

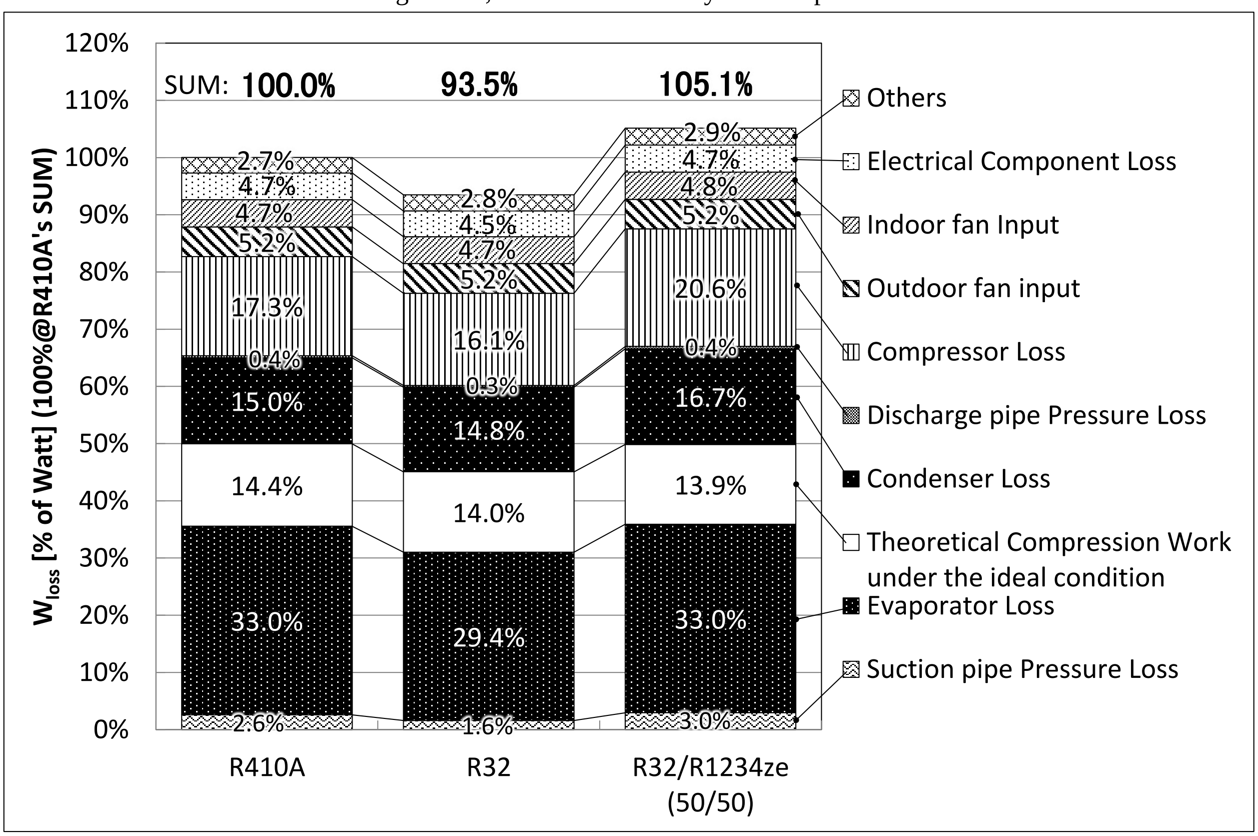

Figure 4-1: Comparison of Loss Analysis in case of each Refrigerant during Cooling Nominal Capacity (4000 W) Operation (Indoor DB/WB: 27℃/19℃, Outdoor DB: 35℃)

First, the differences of "Theoretical Compression Work under the ideal condition" were small and the loss amounts were almost equal to each other. However, as the reciprocal ratio of this work indicates the ratio of Theoretical COP under a certain condition, this category is significant as values indicating the performance of each refrigerant. By the way, although the calculation condition is not equal to this ideal condition, those reciprocal ratios of COPs of the refrigerant are almost equal to the calculation results in Table 1-1.

Next, comparing "Evaporator Loss", "Discharge pipe Pressure Loss" and "Suction pipe Pressure Loss" for each refrigerant, the loss of R32 was smallest, and the loss of R32/R1234ze(50/50) was largest. This difference is mainly due to the pressure drop difference between each refrigerant. As mentioned in paragraph 3.3, the pressure drop of R32 was as much as about 30% smaller than that of R410A or R32/R1234ze(50/50). Moreover, R32/R1234ze(50/50) has a Temperature Glide of about 4K, which reduces the heat-exchanging performance by shrinking the Temperature Gap between air and refrigerant at the outlet of evaporator, and reduces super-heat at the suction pipe of the compressor. Likewise, it is deduced that this is why the evaporator loss of R32/R1234ze(50/50) became worse.

Regarding "Condenser loss", the loss of R32 was smallest. But it was approximately equal to that of R410A. It is thought that the smaller pressure loss characteristic of R32 affected it in the same way as in case of the evaporator, but it is also thought to depend on the sub-cooling characteristic of R32, among other things.

This is mainly because the entire amount of enthalpy change during condensation of R32 is approximately 1.5 times of that of R410A. Thereby the ratio of the enthalpy change amount in sub-cooling region compared to the entire amount in a condenser of R32 is 3/4 of R410A. Accordingly the influence of sub-cooling is relatively smaller. Reaching sub-cooling effectively increases the amount of heat exchange in a condenser and an evaporator per refrigerant flow rate, thereby leading to better COP. For refrigerants such as R410A where the ratio of enthalpy change during sub-cooling is large, the area filled with sub-cooled liquid in heat exchanger grows larger. As the refrigerant temperature descends in the sub-cooling region, the Temperature Gap between air and refrigerant becomes smaller, and simultaneously the heat exchange amount becomes smaller. Therefore it can be stated that R32 is superior compared to R410A.

On the other hand, the "Condenser Loss" for R32/R1234ze(50/50) was 16.7%. As the outdoor heat exchanger is used as a condenser during cooling operation, air and refrigerant are in Counter flow in this test system. Counter flow helps to maintain the Temperature Gap wide between air and refrigerant. however, as condensation of non-azeotropic refrigerant progresses with lowering its temperature even in the two-phase region, its heat exchanging speed(heat flux) slows down and resulting in sub-cooling shortage as shown in Figure 3-6. Moreover, enthalpy change amount decreases, and the amount of "Condenser Loss" grows.

Regarding the heat exchange performance with using non-azeotropic refrigerant, the Temperature Gap between air and refrigerant at the refrigerant side inlet becomes relatively expanded, and the heat exchange performance in condensation or evaporation can be expected to become superior. However, due to the defect that the Temperature Gap at the refrigerant side outlet shrinks and the enthalpy change amount decreases, as shown in this result, the COP of the system which originally achieves sufficiently sub-cooling and super-heating may worsen.

Note that because this test system is a heat pump system for both cooling and Heating operation, if one heat exchanger is a condenser, the other becomes an evaporator. And the evaporator's flow direction between the air and refrigerant becomes Parallel each other. In case of a non-azeotropic refrigerant, although the air temperature becomes the lower with approaching to downstream of the evaporator, the refrigerant temperature gets higher. Thus their Temperature Gap shrinks and the heat exchange amount drops. In case of pure R32 refrigerant or near-azeotropic refrigerant R410A, if the pressure loss of the refrigerant is small, the temperature of the refrigerant is almost constant from the inlet to the outlet, and regardless of whether the flow is Parallel or Counter, the difference of the heat exchanging performance does not exceed the difference of the amount of super-heating, and can become considered small. In addition, even if the refrigerant pressure loss is large, in case of Parallel flow the temperature of the refrigerant drops with heading toward the outlet, heat exchange efficiency is relatively kept. As a result, Counter flow is not necessarily superior to Parallel flow in all cases.

In order to make the most of a non-azeotropic refrigerant, considering the Temperature Glide of the refrigerant itself, using both the condenser and evaporator with Counter flow can be said to be optimal. However, as a non-azeotropic refrigerant is worse than a pure or near-azeotropic refrigerant in making the most of the refrigerant sub-cooling or super-heating, it cannot be said to have superior performance even if in case of Counter flow. Also, limiting both heat exchangers to Counter flow means specializing in either Cooling only or Heating only. On the contrary, in case of a pure or near-azeotropic refrigerant using a condenser with Counter flow and an evaporator with Parallel flow enables stable high capacity on both Cooling and Heating operation.

Finally, regarding the item "Others" in Figure 4-1, for example, absorbing heat during passing the connecting pipes between the indoor unit and outdoor unit is suspected as the reason. However, since they are no more than assumptions, their detailed explanation shall be excluded.

5. SYSTEM PERFORMANCE UNDER HIGH AMBIENT CONDITION

Figure 5-1 shows the results of comparing the change of COP in accordance with the change of the outdoor temperature during Cooling operation with each refrigerant.

As shown in Figure 4-1, under 35℃ outdoor temperature, the COP for R32 is 7% better than that of R410A, and the COP of R32/R1234ze(50/50) is 5% inferior to R410A. This difference does not relevantly change even when the outdoor temperature rises, and the COP of each refrigerant shows the same tendency of decreasing. This decrease in COP is assumed to be due to the rise of condenser intake air temperature, which causes the condenser pressure to rise. Together with the compressor motive power increase the condenser outlet temperature on the refrigerant side rises, and the amount of enthalpy change of the refrigerant decreases, causing the capacity to also decrease.

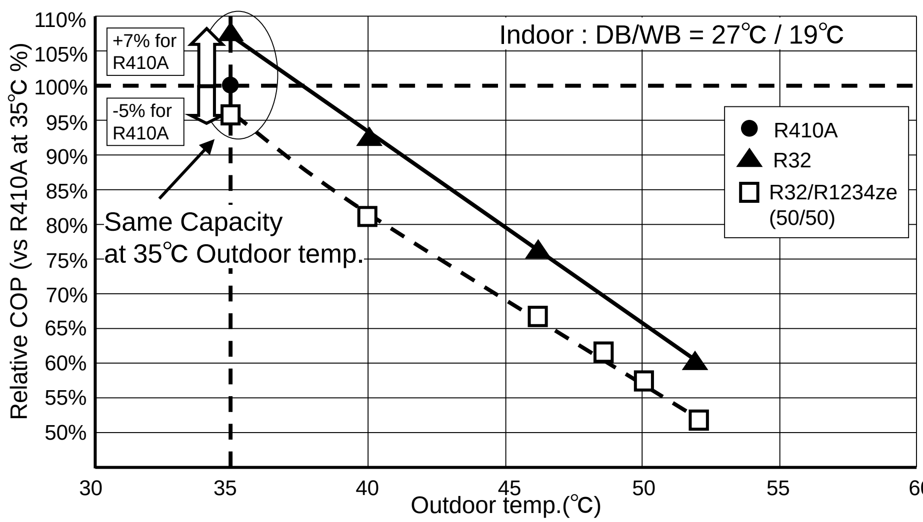

Figure 5-1: COP Comparison Result for High Ambient Conditions

6. CONCLUSIONS

The following results were revealed during our examination:

- Systems using R32 refrigerant performed better than systems using the other refrigerants. R32 extracts high performance from both condenser and evaporator heat exchange.
- While the Discharge temperature of the R32/R1234ze(50/50) refrigerant is lower than that of R32 according to refrigerant properties, in practice was not necessarily lower.
- Given the two points mentioned above, it can be stated that the pressure loss of R32 is remarkably smaller than that of other refrigerants, and in split systems requiring connecting piping, the performance of R32 in Cooling operation is superior. Also, as the Temperature Glide of non-azeotropic refrigerant additionally decreases heat exchanger performance, it leads to worse performance and a rise of Discharge temperature.
- With R32/R1234ze(50/50) refrigerant, the outdoor heat exchanger (evaporator) in Heating operation is easy to freeze up because the Temperature Glide makes ice formation on the evaporator inlet easier. This reduces the maximum capacity in Heating operation.

As a following step for the R32/R1234ze(50/50) refrigerant, risk assessment evaluation and confirming the long-term reliability of the system is necessary. For the former, it is essential to confirm safety considering actual sites of usage, and for the latter, it is important that the refrigerant should not deteriorate the refrigerant oil or compressor on the long term use.

Based on the superior performance of the pure R32 refrigerant, it can be said that the burden of operation on the environment can be reduced. Moreover, as the recyclability of the pure refrigerant is also better than that of the mixed refrigerant, it is safe to state that a total reduction of the burden on the environment can be expected.

NOMENCLATURE

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 Efficiency (= w_r / v_s)	(kJ/ m3)
ΔP_{loss}	Refrigerant Pressure loss	(% of kPa)
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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