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Evaluation of Performance for Heat Pump System using Low GWP Refrigerants in High Ambient Temperature

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Abstract

The refrigerant R32 has a relatively low GWP that contributes to the prevention of global warming, also has a better COP than conventional R410A in air conditioning equipments. R32 was selected as the alternative for R410A for mini-split air-conditioners/heat pumps. After being introduced by all manufacturers in Japan, it started to be introduced by major manufacturers in Asia, Europe, Australia, and so on.

Recently several new alternatives for R410A have been proposed in order to achieve similar capacity to that of R410A and mitigate high discharge temperature issue of R32 as well as to reduce energy consumption. We selected R32 and brought our products to the market since 2012. However, it is important to continue the search for new and better candidates.

We carried out performance comparison for the lower GWP refrigerant R452B and R32. We also conducted the experiments in cooling operation in high ambient temperature. As the result, we found that the COP of R32 is superior to R452B because of its latent heat characteristic. In particular, as condensing temperature increases such as in high ambient temperature operation, latent heat of R452B becomes smaller. Therefore, advantage of R32 in COP becomes more significant in high ambient condition.

From the above, we consider that R32 is still the best refrigerant at present. However, we will continue investigation in search for a better refrigerant.

Key words: GWP; COP; Refrigerant; Heat pump system; R410A; R32/R1234yf; R32; Zeotropic; High ambient temperature

1. Introduction

In recent years, international discussions have been made to prevent global warming, and the Montreal Protocol has phase out CFC and HCFC refrigerants from the viewpoint of protecting the ozone layer. Recently, from the viewpoint of further reducing GWP, the Kigali agreement to phase down HFC in terms of CO2 conversion was done. Selection of refrigerant should consider not only ODP and GWP, but also the effect on global warming in various ways such as safety, energy efficiency, affordability, resource efficiency, recyclability and recoverability. The reasons why R32 was chosen at that time, its GWP is 1/3 of R410A, required refrigerant charge is smaller, and it has excellent thermo-physical properties. There is no issue of fractionation, R32 is easy to manage, in charge and recharge processes. Furthermore, it is attractive from the viewpoints of recyclability and recoverability. Based on these evaluations, we determined that R32 is suitable for air conditioners such as mini split and multi split.

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. Based on this, many researchers of the air conditioning industry and academia continue searching for new refrigerants.

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

Recently, new refrigerant R452B was reported as high efficiency refrigerant. [4]

It contains three component refrigerants which are based on 67wt% of R32. We conducted the experiments on this new refrigerant using a mini split air conditioner, and compared the results with R32.

1.1 Properties of the Refrigerants

Table 1 shows the property of refrigerants which were evaluated in this study. Refrigerant properties are computed using REFPROP version 9.1 (Lemmon et al., 2013). We evaluated two refrigerants. One is R32 as the base refrigerant, and the others is mixed refrigerant of R452B as candidate alternatives of which composition is a 67wt% of R32, 7wt% of R125, and 26wt% of R1234yf. The refrigerant R452B is zeotropic and has temperature glide.

Next, the calculated COP (Coefficient of Performance) was compared in a cooling operation. The evaluation conditions were condensing temperature T_c=46°C, evaporating temperature T_e=12.5°C, suction pipe temperature T_S =15°C, condenser outlet temperature $T_{c.out}$ =38°C, and compressor efficiency η=70%. The results are shown in the Table 1 below. For these calculations the pressure values that have the same temperature at the mean point to the saturation temperature of R32 was used. The result shows that R452B has the maximum COP of 4.98 (99.4% compared to R32): an excellent value. Also, this has a temperature glide (ΔT_{GL}) of 0.9K.

Compared to the R32, R452B has 77.8% of refrigerating effect and 78.2% of compressor work. It means that, using R452B requires 28% greater refrigerant mass flow to achieve the same refrigerating capacity as R32. Pressure drop of HFO-mix is larger and discharge temperature is 10K lower than that of R32

Table 1. Calculated properties of refrigerants charged to the test system

Refrigerant	R452B	R32
Temperature glide T_{GL} (K) @ 12.5°C	0.90	0.0
Discharge / Suction pressure P_d / P_s (MPa abs)	2.667 / 1.118	2.862 / 1.191
Refrigerating effect w_r (kJ / kg)	193.9 (77.8%)	249.3 (100%)
Compressor work w_s (kJ / kg)	39.0 (78.2%)	49.8 (100%)
Coefficient of performance: COP = w_r / w_s	4.98 (99.4%)	5.01 (100%)
Specific volume in suction v_s (m^3 / kg)	0.02427 (86.2%)	0.02814 (100%)
Volume capacity = w_r / v_s (kJ / m^3)	7989 (90.2%)	8858 (100%)
Condenser outlet density $\rho_{c.out}$ (kg / m^3)	928.12 (102.3%)	907.57 (100%)
Condenser outlet isobaric heat capacity $Cp_{c.out}$ (KJ/kg·K)	1.936 (92.6%)	2.091 (100%)
Pressure loss at constant capacity P_{loss} (% of kPa)	(160%)	(100%)
Discharge temperature T_d (°C)	73.3	83.4

*Calculation conditions: T_c=46°C, T_e=12.5°C, Suction line temp.: T_s =15°C, Condenser outlet: $T_{c.out}$=38°C, Compressor efficiency: η_{comp} =70%, in Cooling operation. Saturation temperature of mixed refrigerant is midpoint temperature of two-phase region under constant pressure.

2. Test system and conditions

Figure 1 shows the outline of the system used for the experiments. It is a mini-split type air-conditioner system with a nominal cooling capacity of 7.1 kW. The indoor unit and the outdoor unit are connected with two 7.5 meter 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 rotating speed with a variable frequency drive (V.F.D.). The expansion valve (Exp. Valve) is electrically controlled to change the opening to adjust the mass flow rate entering the evaporator from the condenser, and to adjust the degree of superheat at the compressor suction point. The four-way valve (4-Way Valve) enables to switch cooling and heating operation by switching the function of condenser and evaporator. In this diagram, the solid lines inside of the four-way valve indicate the flow directions in the cooling operation. The compressed gas discharged from the compressor flows into the outdoor heat exchanger, where the gas is cooled down and condenses into liquid phase. Then, the liquid is expanded, and the temperature is reduced 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 test, system capacity measurement was conducted with a facility using the air-enthalpy method (psychrometric type) which is described by ISO 5151-2010. Also, we measured temperature by T-type thermocouples and measured pressure by pressure gauges at six points; discharge pipe and suction pipe of the compressor, the inlet and outlet of the heat exchangers. At the midpoints of the heat exchangers, only temperature was measured.

Table 3 shows three test conditions based on ISO 5151-2010. Operating mode was cooling.

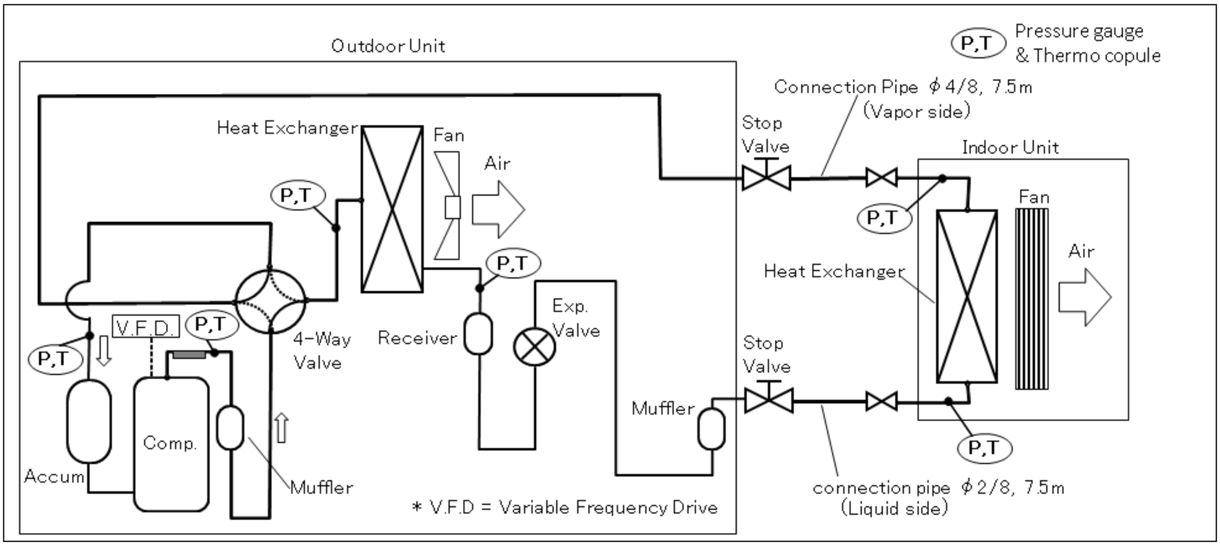

Figure 1. Schematic diagram of test system

Table 2. Charged amount of refrigerant in the test system

Refrigerant	R452B	R32
Optimized refrigerant charge (kg)	1.7	1.55

Table 3. Air temperature conditions at cooling operation

Condition	T2 (°C)	T1 (°C)	T3(H) (°C)
Indoor unit side (evaporator)	DB:21 / WB:15	DB:27 / WB:19	DB:32 / WB:23
Outdoor unit side (condenser)	DB:27 / WB:19	DB:35 / WB:24	DB:52 / WB:38

3. Test results

3.1. Experiment One: Selection of R452B Refrigerant Charge

On T1 condition, we conducted experiments by changing the amount of R452B, compressor speed, and adjusting the opening ratio of the expansion valve. Then, we tried to find the most suitable refrigerant charge amount for the maximum COP at the nominal rated cooling capacity (7100W). As a result, we found it to be 1.7kg amount of the refrigerant R452B as shown in Table 2. While changing the amount of refrigerant, we estimated how much refrigerant is needed considering the ratio density to the original refrigerant (R32). However, we needed more than our estimation for some reasons.

Then, we charged 1.7kg of the R452B and conducted the experiments on T1, T2, T3(H) conditions.

3.2. Experiment Two: System Capacity, Power consumption and COP on Each Conditions

We changed the compressor speed on T1, T2, T3(H) conditions in order to measure the system performance in the wide capacity range. The control algorithm for changing the opening ratio of the electronic expansion valve was the same with the R32 product system. At each refrigerant, expansion valve adjusted the opening ratio to obtain the optimal degree of superheat at the compressor suction point. Other configurations, such as heat exchanger, air flow rate, machine oil, electrical equipment, remained the same.

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

Figure 2 and Figure 3 show the performance capacity depending on the compressor speed on T1, T2, T3(H) conditions. In terms of compressor speed, or cylinder volume, both of the cooling capacity and power consumption of R32 were larger than those of R452B. However, Figure 4 shows the power consumption of R32 was smaller than R452B in terms of cooling capacity.

The result at 50% of the nominal rated (3550W) on T3(H) condition, shows the COP of the R452B decreases 6.3% relative to R32, and the power consumption was 72W larger than that of R32.

From the results of evaluation, R452B at large cooling capacity, and high ambient temperature had worse COP than R32. It is necessary to pursue the cause.

Figure 5 shows the compressor discharge pipe temperature of R32 and R452B. Temperature of R32 was 3.5 to 5.5K higher than that of R452B. This difference of temperature is smaller than calculated in the theoretical refrigeration cycle which mentioned in the sub-section 1.1. This time, T_c rose to 60°C on T3(H) condition, however, discharge temperature rose to 90°C. Therefore, it is possible to operate R32 without shortening the life of a compressor. There is a difference in compressor efficiency as a main cause the difference in discharge temperature differs between calculation and actual measurement. In other words, if the compressor can be driven with less power consumption, the problem of the discharge temperature of R32 will diminish.

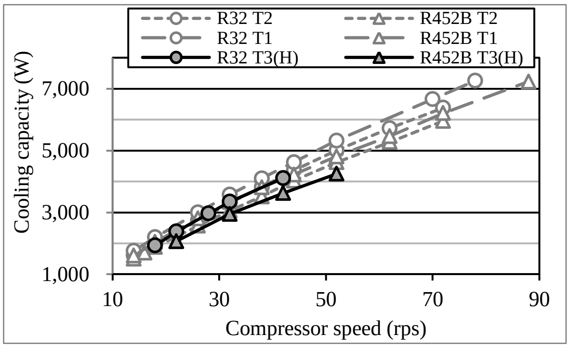

Figure 2. Cooling capacity

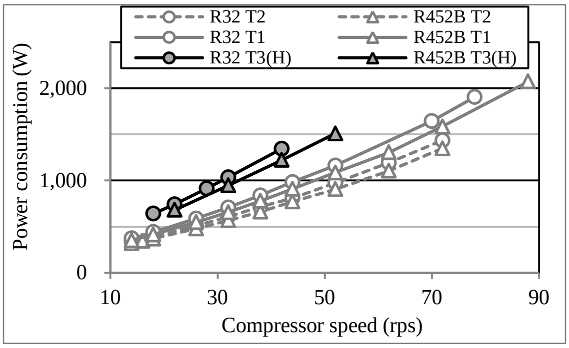

Figure 3. Compressor power consumption

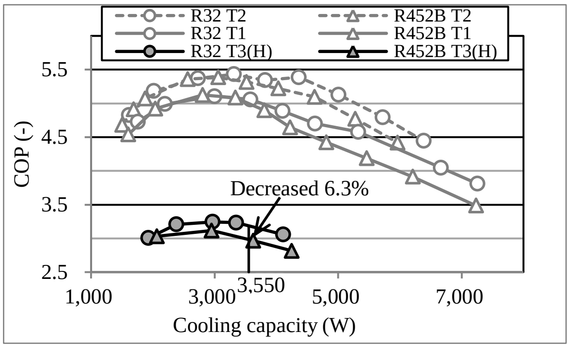

Figure 4. COP by cooling capacity

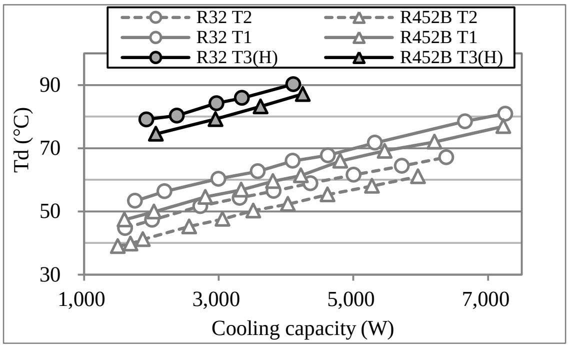

Figure 5. Compressor discharge pipe temperature

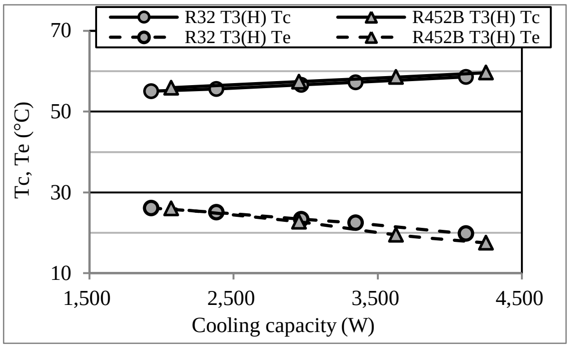

Figure 6. T_c, T_e on T3(H) condition

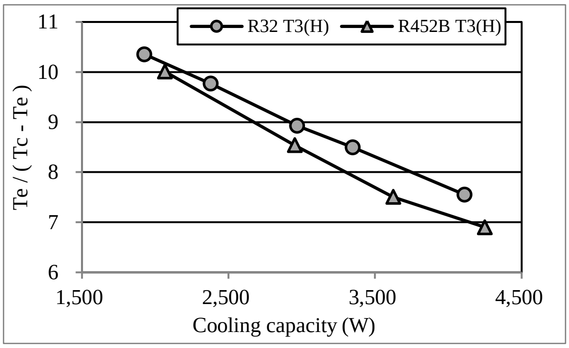

Figure 7. $T_e/(T_c-T_e)$ on T3(H) condition

4. Discussion

4.1. Policy of Analysis from Experiment Results

First, we looked into the theory to understand the experiment results.

Theoretical COP of the reverse Carnot cycle is calculated by $COP_{th} = T_e / (T_c-T_e)$. Figure 6 and Figure 7 show the measured T_c and T_e by changing the compressor speed on T3 (H) condition. T_c and T_e were used as a dew point pressure equivalent temperature of P_d and P_s, respectively for comparing the theoretical power consumption easily which will be described later. As for R452B, T_c became slightly higher and T_e became lower than in case of R32 with increase of the capacity, COP_{th} became lower than that of R32, because of expanding difference between T_c and T_e increases. As a result, in case of R452B more power is required to obtain the same capacity of R32. At 50% of the nominal rated on T3(H) condition, T_c was 0.4K higher and T_e was 2.1K lower than R32.

T_c and *Te* differ even if the capacity of R452B is the same as that of R32 in the performance evaluation.

It indicates that the actual system has distinctive three phenomena.

- Refrigerant mass flow rate increases when latent heat is decreased.

  Refrigerant side capacity (ϕ_0) is calculated by the $\phi_0 = q_{mr} \cdot \Delta h$. When the refrigerant enthalpy difference Δh of the inlet and that of outlet of the heat exchanger is small, in order to obtain the capacity ϕ, the mass flow rate (q_{mr}) is increased. The temperature difference between air and heat exchanger increases in order to maintain the Δh, which means that T_c becomes high and T_e becomes low.
- Even while using the optimum refrigerant charge, the effective heat transfer area for the latent heat is decreased because the amount of subcooled liquid is increased.

During refrigerant passing through the condenser, it is necessary to have a certain degree of subcool to avoid the shortage of the latent cooling capacity. Also, even the refrigerant mass flow rate is increased, degree of subcool will be difficult to obtain, because the specific heat is not changed significantly. In other words, mass flow rate is increased, subcooled part of the condenser increase and it leads to the reduction of the effective heat transfer area in the condenser for the latent heat exchange. In addition, the heat conductance (K), expressed in $\phi_k = K{\cdot}A\ (T_c - (T_{a.in}+T_{a.out})\ /2)$. As the effective surface area is decreased, T_c is increased to obtain the same capacity of R32. It is noted that the effect of the air side on the coefficient of K is higher than that of the refrigerant side in air-conditioner. Therefore, the difference of the heat transfer rate of both refrigerants on the coefficient of K is not considered

• A large pressure loss in the heat transfer tubes and pipes.

Pressure loss is proportional to the square of the mass flow rate and inversely proportional to the density. As to the density, for example condenser outlet density, as shown in Table 1, at T_c = 45°C, $T_{c.out}$ = 40°C, the density of R452B is 1.9% greater than that of R32. It can be said that it does not have a big impact on pressure loss.

However, the increment of the mass flow rate has more impact on it as described above. In general, the pressure loss is remarkably increased in the low pressure gas. Further, the pressure loss in the connecting piping between the compressor suction and the evaporator leads to the increment of the specific volume at the compressor suction and the decrement of the refrigerant mass flow rate. In other words, it is required to increase the compressor speed to overcome the decrement of the refrigerant mass flow rate, and it leads to the increment of the power consumption.

What is more, these influences mentioned above resulted in additional rise of discharge temperature.

These three phenomena are verified in the next section.

4.2. Refrigerating Effect and Mass Flow Rate in the Evaporator

Figure 8 shows the refrigerating effect in the evaporator described as $\Delta h_{eva,}$ is calculated based on the saturated pressure and temperature of the condenser outlet and evaporator inlet. Figure 9 shows the mass flow rate (q_{mr}), which is calculated from $q_{mr}=\phi_0\ /\Delta h_{eva}$. Δh_{eva} of R452B was from 75 to 78% of that of R32, and the refrigerant mass flow rate was increased accordingly. In particular, on the T3(H) condition where T_c shows the highest value than any other conditions, R452B can be seen that the decrement of the Δh_{eva} and the increment of mass flow rate becomes remarkably than R32. The mass flow rate of R452B was 1.32 times on T3(H) condition and 1.27 times on T2 condition respectively compared with that of R32.

Thus, the mass flow rate of R452B is increased by the decrement of the latent heat. It was found to be an increment in high ambient condition. The reason why the mass flow rate of R452B is increased remarkably in high ambient condition is discussed in the sub-section 4.6.

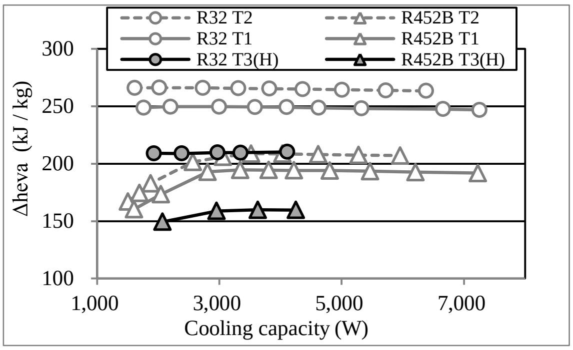

Figure 8. Refrigerating effect in the evaporator Δh_{eva}

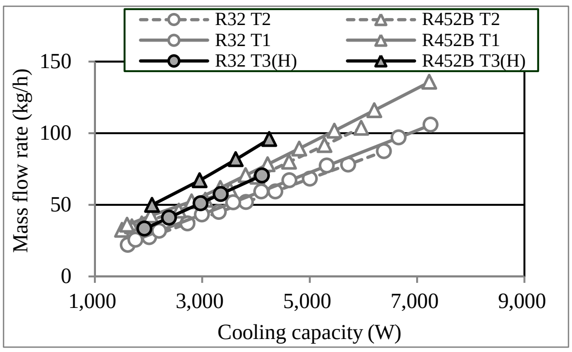

Figure 9. Mass flow rate

4.3. Influence of Subcooled Liquid in the Condenser

Figure 10 shows the evaluation result about the characteristics of the degree of subcool (*SC*). As for *SC*, R452B was 0.5K lower than the R32. We estimated how much this subcooled liquid occupied in condenser. Based on the evaluation results at 50% of the nominal rated (3550W) on T3(H) condition, we estimated: T_c=58.0°C, *SC*=2.5K at R452B, T_c=57.6°C, *SC*=2.9K at R32. The specific heat and density under each condition are shown Table 4. When the refrigerant mass flow rate is the same, R452B is slightly easier to obtain degree of subcool than R32. However, when the refrigerant mass flow rate increases for R452B, its advantage over R32 is reversed. That is, as we considered in the sub-section 4.2, if the mass flow rate of R452B was 1.32 times as much as R32, heat radiation amount of R452B requires 1.28 times as large in order to obtain same degree of subcool.

Figure 11 shows the ratio of the sensible heat capacity (ϕ_{ksc}) and the total capacity (ϕ_k) of the condenser. The ratio of liquid in the condenser of R452B was larger than that of R32 as the load increases or ambient increases. The difference is 0.5% at 50% of the nominal rated (3550W) on T3(H) condition. It can be rephrased that latent heat transfer area of R452B is more decreased by subcooled liquid than the R32. Therefore, P_d and T_c will rise to radiate the refrigerant. T_c of R452B was 0.4K higher than that of R32, which caused power consumption difference; we will discuss it in the sub-section 4.5.

4.4. Influence of the Pressure Loss due to Mass Flow Rate

Figure 12 shows the pressure loss from the evaporator inlet to the compressor suction. It can be seen that the pressure loss of R452B is larger because the mass flow rate of R452B is larger than that of R32. We can see there is a tendency that the pressure loss is particularly large on the T3(H) condition. As mentioned in the sub-section 4.2, one of the reasons is that mass flow rate is increased because Δh_{eva} is particularly small in high ambient temperature. Difference between the pressure loss of R32 and R452B at 50% of the nominal rated (3550W) on T3(H) condition was 0.026MPa. This is equivalent to approximately 0.7K at the R452B temperature. This is not enough to fill the gap between the T_e of R452B and that of R32 as described in the sub-section 4.1. Pressure loss of R452B was 0.026MPa larger than that of R32, which caused power consumption difference; we will discuss it in the sub-section 4.5.

Table 4. Calculated properties of refrigerants charged to the test system on T3(H) condition

Refrigerant	R452B	R32
Condenser outlet isobaric specific heat $C_{pc.out}$ (KJ/kg・K)	2.505 (96.9%)	2.584 (100%)
Condenser outlet density $\rho_{c.out}$ (kg/m^3)	819.05 (100.5%)	814.81 (100%)

*Calculation conditions: HFO-mix T_c=58°C, Condenser outlet: $T_{c.out}$=55.5°C, R32 T_c=57.6°C, Condenser outlet: $T_{c.out}$=54.7°C

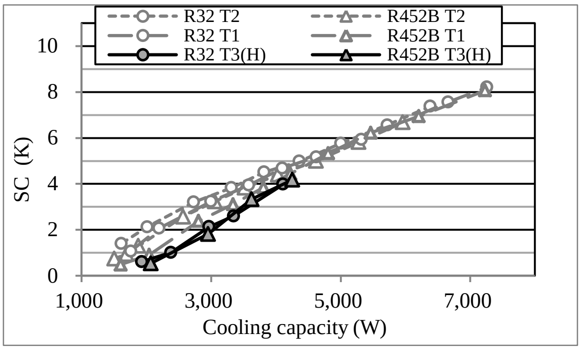

Figure 10. Characteristics of the degree of subcool

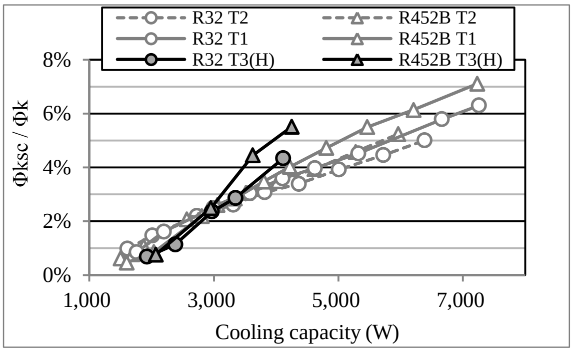

Figure 11. Ratio of ϕ_{ksc} and ϕ_k of the condenser.

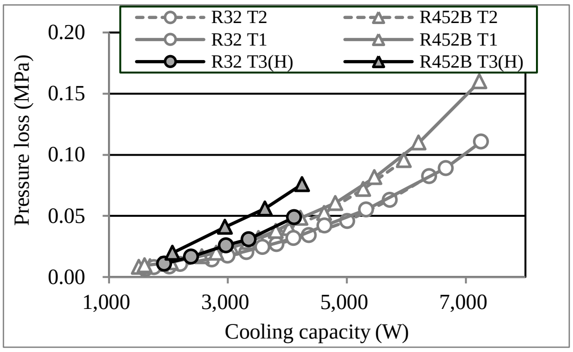

Figure 12. Pressure loss from the evaporator inlet to the compressor suction

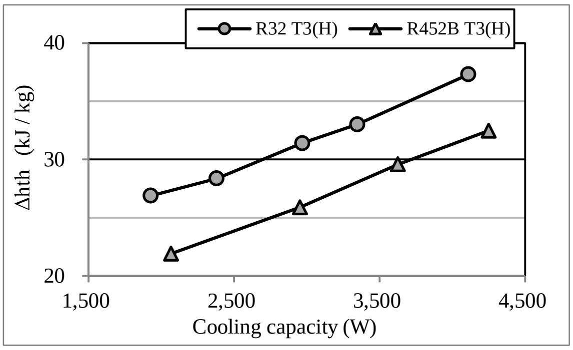

Figure 13. Theoretical enthalpy of compressor work Δh_{th}

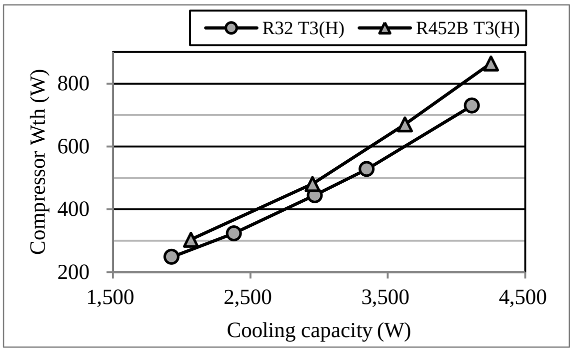

Figure 14. Compressor theoretical power consumption W_{th}

4.5. Analysis of Power consumption Difference

In the sub-section 4.2 to 4.4, we compared R452B with R32, and mentioned the causes that make T_c higher and T_e lower particularly at the high ambient temperature of R452B. Then we verified how much power consumption is increased by those causes. Figure 13 shows the theoretical enthalpy of compressor work (Δh_{th}). It is calculated from the following actual measurement value: pressure and temperature of compressor suction and discharge pressure, and under the condition of adiabatic compression. Figure 14 shows the theoretical power consumption (W_{th}) calculated from $W_{th} = q_{mr} \cdot \Delta h_{th}$. The influence of refrigerant mass flow rate, the subcooled liquid, and the pressure loss shown in the sub-sections 4.2 to 4.4 is contained in the Δh_{th} calculation, because it uses the actual discharge pressure and suction pressure of compressor. The theoretical power consumption of R452B is larger than that of R32.

We made the following calculation at 50% of the nominal rated (3550W) on T3(H) condition.

When estimating the theoretical power consumption in properties, R452B is 73W larger than R32. As shown in the sub-section 3.2, however, the actual difference of power consumption was 72W. One of the possible causes of this calculation error of 1W is that various efficiency, discharge temperature, and refrigerator oil at the compressor operation may cause the difference of loss.

However, we found the difference between the power consumptions of R452B and R32 is caused almost by the mass flow rate, T_c and T_e.

The sub-section 4.1 showed that T_c of R452B was 0.4K higher than that of R32, which caused the difference of 8W to the theoretical power consumption when calculated in the same procedure as above. This corresponds to 11% between the COP difference of R32 and R452B. The sub-section 4.4 revealed that the pressure loss of

R452B was 0.026MPa larger than that of R32, which caused the difference of 15W to the theoretical power consumption when calculated in the same procedure as above. This corresponds to 20% of the COP difference between R32 and R452B.

We considered the influence of mass flow rate mentioned in the sub-section 4.2. We compared the theoretical power consumption of R452B after removing the effect of T_c and pressure loss mentioned above with the theoretical power consumption calculated using the mass flow rate equivalent to R32 after converting the P_d and P_s into the refrigerant properties of R32. And the power consumption of R452B was 42W larger than that of R32. We also calculated another way. R452B's theoretical compressor work was 0.82 times of R32's, and R452B's mass flow rate was 1.32 times of R32's. Furthermore, by multiplying these two numbers, we can get the theoretical power consumption of R452B, 1.08 times. The theoretical power consumption of R32 was 579W and 8% of that was 48W. Thus, the effect of mass flow rate is equal to approximately 67% of the difference between R452B's COP and R32's.

4.6. Latent Heat in High Ambient Condition

Our analysis has shown that R452B has lower refrigerating effect in higher ambient temperature and higher capacity than R32. We have carried out verification from the perspective of refrigerant properties in order to identify the cause. The P-h diagrams of various refrigerants are shown in Figure 15 and Figure 16. Since R452B has lower critical point and smaller latent heat than R32, as the pressure rises, the saturation lines of liquid and gas approach each other.

Similarly, the latent heat of R32 declines as it approaches the critical point, but the decline is relatively small. T_c becomes higher in accordance with the outside air, and it can be seen that the latent heat of the R452B declined more greatly in high ambient condition. Figure 17 shows the ratio of refrigerating effect of the each refrigerant on the basis of the R32. The latent heat of R452B declines as T_c rises, and a remarkable decline can be seen, especially at T_c=50°C or more, as the T3 (H) condition.

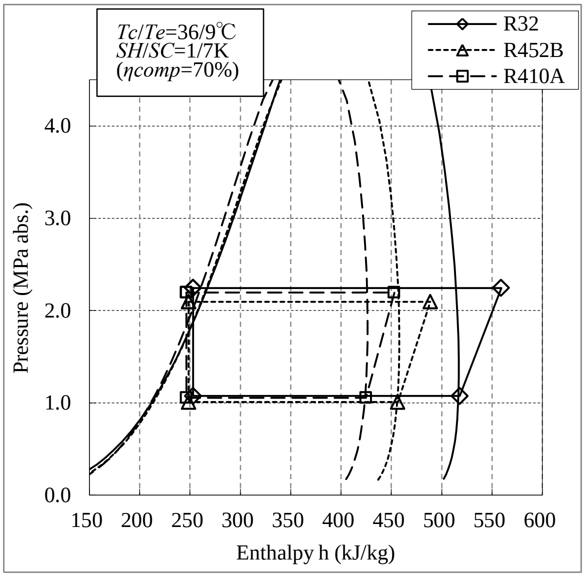

Figure 15. Cooling capacity 6000W on T2 condition

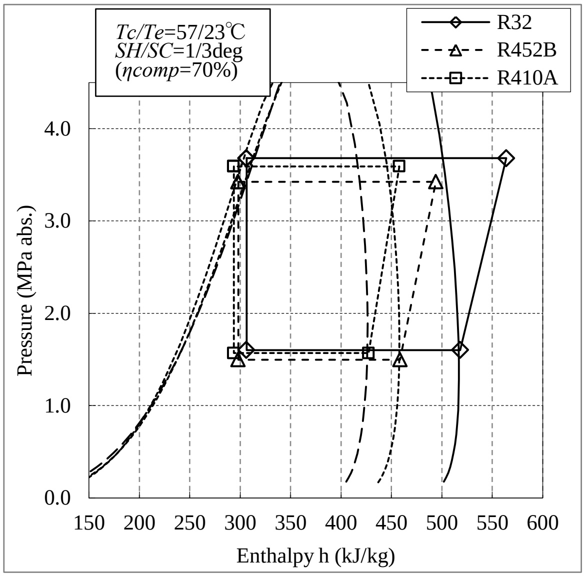

Figure 16. Cooling capacity 3550W on T3(H) condition

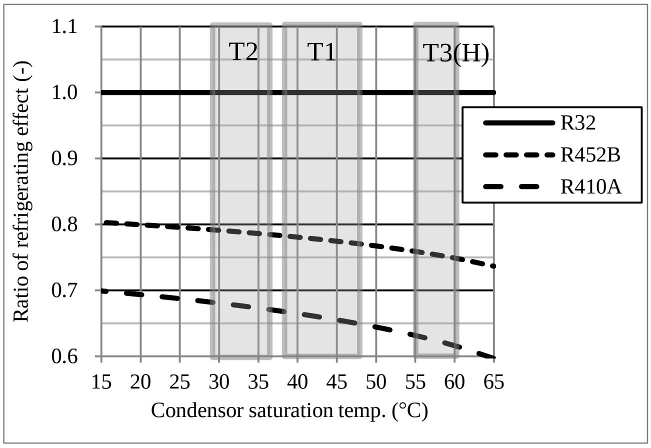

Figure 17. Ratio of refrigerating effect of the each refrigerant on the basis of R32 (T_e =15°C, *SC*=3K, *SH*=1K)

5. Conclusions

The following results were obtained:

- As factors that affect the discharge temperature, compressor efficiency also should be taken into consideration. This is because as the compressor efficiency improves for saving energy, the actual discharge temperature approaches adiabatic compression and the difference in refrigerant comparison becomes smaller.
- In the actual operation in a refrigeration system, effective heat transfer area in the condenser for latent heat exchange or pressure loss in the evaporator heat transfer tubes and suction pipes, affect T_c and T_e, which affects the capacity and the COP.
- The latent heat of R452B is smaller than that of R32. 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.
- Pressure loss in the evaporator heat transfer tubes and suction pipes of R452B is larger than R32. T_e becomes lower by pressure loss, and specific volume at the compressor suction increases. Therefore, compressor load becomes larger to maintain the refrigerant mass flow rate which obtains the cooling capacity.
- In the high heat load such as high capacity and high ambient temperature, the latent heat of R452B declines remarkably, especially at T_c=50°C or more. Thus, the difference between the COP of R452B and that of R32 becomes much higher.
- The influences mentioned above resulted in additional rise of discharge temperature when using R452B. The difference between R32 and R452B is smaller than calculated in the theoretical refrigeration cycle.

Nomenclature

ΔT_{GL}	Temperature glide	(K)
SC	Degree of subcool	(K)
SH	Degree of superheat	(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)	(–)
COP_{th}	Coefficient of Performance (=T_e /(T_c-T_e))	(–)
V_s	Specific volume in suction	(m^3/ kg)
$\rho_{c.out}$	Condenser outlet density	(kg/ m^3)
$C_{pc.out}$	Condenser outlet isobaric heat capacity	(KJ/ kg·K)
P_{loss}	Pressure loss at constant capacity	(% of kPa)
ϕ_0	Refrigerating effect	(kJ/ kg)
ϕ_k	Condensing effect	(kJ/ kg)
T_d	Discharge temperature	(°C)
T_c	Condensing temperature	(°C)
T_e	Evaporating temperature	(°C)
T_s	Suction temperature	(°C)
$T_{c.out}$	Condenser outlet temperature	(°C)
$T_{a.in}$	Air inlet temperature	(°C)
$T_{a.out}$	Air outlet temperature	(°C)
DB	Dry Bulb temperature	(°C)
WB	Wet Bulb temperature	(°C)

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