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ABSTRACT

One hermetic scroll compressor with an economizer port that is designed for R404A has been tested with R404A in a compressor calorimeter. The testing has been completed on several running conditions to create a compressor operating map. Based on this map, a compressor efficiency model has been created. This model has been used to calculate capacities, input power, COPs and discharge temperatures for frozen and fresh conditions in the transport refrigeration application with R452A, R454A and R454C. The capacities, COPs and discharge temperatures for those frozen and fresh conditions are compared with R404A's. Since some of the refrigerants have glide characteristics, calculations based on dew-point and mid-point are compared.

Another hermetic scroll compressor with an economizer port that is designed for R134a has been tested with R134a in a compressor calorimeter. The testing has been completed on several running conditions to create a compressor operating map. Based on this map, a compressor efficiency model has been created. This model has been used to calculate capacities, input power, COPs and discharge temperatures for frozen and fresh conditions in the transport refrigeration application with R513A and R1234yf. The capacities, COPs and discharge temperatures for those frozen and fresh conditions are compared with R134a's.

Furthermore, a general comparison is made between the two different compressors and the two different refrigerants, where one is tested on R404A and the other is tested on R134a. In this case, COPs and discharge temperatures are compared.

1. INTRODUCTION

Today's refrigerants used in transport refrigeration are mainly R404A and R452A. The GWP (Global Warming Potential) for R404A is about 4000 and for R452A, about 2000. There is a need for further reduction of GWP. The challenge the industry is facing is that there does not appear to be a good candidate to stay with similar capacity, vapor pressures and discharge temperature and avoid flammable refrigerants and large glide. Therefore, this paper is looking at both non-flammable group A1 and light flammable group A2L (ANSI/ASHRAE 34, 2016). Considered are also refrigerants with glide, see the definition in section 3 below. Only non-toxic chemical refrigerants are considered in this paper. The studied refrigerants are divided into three groups based on vapor pressure: R404A like refrigerants, R134a like refrigerants and R410A like refrigerants. A similar paper was presented with known refrigerants at that time and limited to R404A like refrigerants. (Sjoholm, 2014)

2. STUDIED REFRIGERANTS

Table 1 for studied refrigerants could have been made much larger but we had to narrow the task with some good representative refrigerants. We prefer as low GWP as possible, low normal boiling and dew point temperatures, no glide or low glide, high critical temperature and pressure as well as a pressure above one atmosphere or as little vacuum as possible at -40 F. At this point, testing compressors with R410A like refrigerant with economizer has not been completed. However, our calculations for those refrigerants indicate very high discharge temperatures in some areas of the operating envelope. Therefore additional compressor cooling has to be applied. Furthermore, a compressor model created for R404A cannot be used properly for R410A due to excessive pressure extrapolations. Therefore, no further calculations are presented in this paper regarding R410A like refrigerants.

Table 1: Basic Characteristics of Studied Refrigerants

ASHRAE Designation	ASHRAE Class	GWP AR5	Normal Boiling Point	Normal Dew Point	Normal Glide	Critical Temperature	Critical Pressure	Pressure at -40 F Dew Temperature	R32	R125	R143a	R134a	R1234yf
			F	F	F	F	psia	psia	mass fraction	mass fraction	mass fraction	mass fraction	mass fraction
R404A	A1	3943	-51.20	-49.85	1.35	161.68	540.81	19.00		0.44	0.52	0.04	
R452A	A1	1945	-52.47	-45.65	6.82	168.17	589.82	17.06	0.11	0.59			0.30
R454A	A2L	238	-54.12	-43.90	10.24	186.29	724.06	16.29	0.35				0.65
R454C	A2L	146	-50.01	-35.96	14.05	191.25	606.42	13.21	0.215				0.785
R134a	A1	1300	-14.93	-14.93	0	213.91	588.75	7.43				1.00	
R513A	A1	572	-21.24	-21.05	0.19	207.81	559.16	8.91				0.44	0.56
R1234yf	A2L	<1	-21.07	-21.07	0	202.50	490.55	9.05					1.00
R410A	A1	1924	-60.60	-60.46	0.14	160.42	710.87	25.36	0.50	0.50			
R32	A2L	677	-60.97	-60.97	0	172.59	838.61	25.73	1.00				

3. REFRIGERATION CYCLE AND GLIDE

Compressors with an economizer utilizing temperature glide refrigerants are being studied. For refrigerants with no or practically no glide, like the refrigerants in the R134a like category, this is done by comparing them at pressures corresponding to saturated suction and saturated discharge temperature. An economizer heat exchanger is assumed because a flash tank would be impractical for transport refrigeration. An ideal economizer heat exchanger is also assumed to make the economizer heat exchanger independent of refrigeration capacity.

We have compressor models for variable economizer pressure, the pressure-enthalpy diagram for each refrigerant together with the compressor modeling gives the balanced economizer pressure. When the economizer pressure is known, the cooling capacities, as well as power consumptions, can be calculated. It becomes a bit more complicated regarding refrigerants with glide, like the refrigerants in the R404A like category. If there is a temperature difference between saturated liquid (bubble point) and saturated gas (dew point) for the same pressure, than the refrigerant has glide.

The amount of glide will change depending on pressure level. It is recommended to compare two different compressors for the same refrigerant by comparing them at the same dew-point temperatures. When comparing two different refrigerants in the same system, it is generally recommended to compare them at the mid-point for the condensing pressure and the evaporating pressure. The challenge with this approach is that the mid-point for the evaporator is very dependent on the amount of sub-cooling achieved by the economizer circuit. To show the difference between a dew-point and a mid-point approach, we calculate them both, see Figure 1.

$$T1mid = (T1dew - T1evapinlet) / 2 \quad (1)$$

$$T2mid = (T2dew - T2bubble) / 2 \quad (2)$$

Where
T1mid: Evaporating mid-point temperature
T1dew: Evaporating dew-point temperature
T1evapinlet: Evaporator inlet temperature
T2mid: Condensing mid-point temperature
T2dew: Condensing dew-point temperature
T2bubble: Evaporating bubble point temperature

The compressor modeling equations in section 4 need to be solved using equation 1 to find the mid-point for the evaporator. The evaporator inlet temperature is solved by the equation 3 below (Hundy, 2000):

$$(T1evapinlet - T1bubble) / (T1dew-T1bubble) = (he-hb) / (hd-hb) \quad (3)$$

Where
he: enthalpy at dew point temperature for economizer or auxiliary pressure P3
hb: bubble point temperature at evaporating pressure P1
hd: dew point temperature at evaporating pressure P1

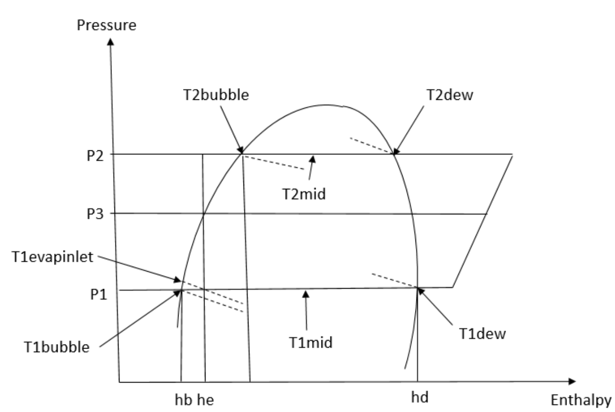

Figure 1: P-h diagram for temperature glide (Hundy, 2000)

4. COMPRESSOR MODELING

The compressor simulation modeling is one of the major challenges to predict the TRU (Transport Refrigeration Unit) performance without testing at each different Box and Ambient conditions.

4.1 The 10-coefficient method

Most of compressor manufacturers provide the estimated Capacity, Input Power, and Mass flow rate at saturated suction temperature and saturated discharge temperature according to 10-coefficient method (ANSI/AHRI Standard 540, 2015). The limitations of this10-coefficient are 1) saturate temperatures are based on dew-point temperature and no mid-point temperature for temperature glide is considered, 2) assume that sub-cooling and super-heat are constant at given compressor rating conditions, 3) no compressor speed is included, and 4) no vapor injection (economizer) modeling at an auxiliary port is included.

$$X = C_1 + C_2\,TS + C_3\,TD + C_4\,TS^2 + C_5\,TS\,TD + C_6\,TD^2 + C_7\,TS^3 + C_8\,TS^2\,TD + C_9\,TS\,TD^2 + C_{10}\,TD^3 \tag{4}$$

Where

C_1 through C_{10}: Regression coefficients,

TS: suction dew-point temperature,

TD: discharge dew-point temperature,

X can be input power, mass flow rate, and capacity (ANSI/AHRI Standard 540, 2015)

4.2 The 23-coefficient method

To improve AHRI 10-coefficient method for vapor injection compressor model, 23-coefficient method was proposed (Cambio, 2016). However this method has also a few limitations in applying to TRU modeling such as 1) no compressor speed is included, 2) no discharge pressure effect is included, 3) the temperatures at the suction port and auxiliary port are not included, and 4) still need 10-coefficient method to calculate the conditions without an economizer.

$$\begin{aligned} X = {} & C_1 + C_2\,TS + C_3\,TD + C_4\,TS^2 + C_5\,TS\,TD + C_6\,TD^2 + C_7\,TS^3 + C_8\,TS^2\,TD + C_9\,TS\,TD^2 + C_{10}\,TD^3 \\ & + C_{11}\,TE + C_{12}\,TE^2 + C_{13}\,TS\,TE + C_{14}\,TE\,TD + C_{15}\,TE^3 + C_{16}\,TS\,TE^2 + C_{17}\,TE\,TS^2 + C_{18}\,TD\,TE^2 \\ & + C_{19}\,TE\,TD^2 + C_{20}\,TS\,TE\,TD + C_{21}\,TE\,TD\,TS^2 + C_{22}\,TS\,TE\,TD^2 + C_{23}\,TS\,TD\,TE^2 \end{aligned} \tag{5}$$

Where

C_1 through C_{23}: Regression coefficients,

TS: suction dew point temperature,

TE: economizer dew point temperature,

TD: discharge dew point temperature,

X can be input power, mass flow rate, and capacity (Cambio, 2016)

4.3 The 12-coefficient method

For more practical modeling of TRU analysis, 12-coefficient method was proposed (Erickson, 1998). The characteristics of this 12-coefficent method shown in the equation (6) are 1) the variable speed can be considered as a speed term is included, 2) the effect of discharge pressure can be considered as a discharge pressure term is included, 3) the temperatures at the suction port and auxiliary port are included, 4) the volumetric efficiency at the compressor suction port is calculated, 5) the auxiliary volumetric efficiency at the auxiliary port is calculated, 6) total isentropic efficiency is calculated, and 7) alternative refrigerants can be easily considered as it includes a discharge pressure term and volume ratio effect.

$$X = C_1 + C_2 N + C_3 N^2 + C_4 MVR + C_5 MVR^2 + C_6 N\, MVR + C_7 AVRD + C_8 MVR\, AVRD + C_9 N\, AVRD + C_{10} N\, AVRD^2 + C_{11} N^2 AVRD + C_{12} Pd \quad (6)$$

Where
C_1 through C_{12}: Regression coefficients,
N: speed,
MVR: volume ratio,
AVRD: volume ratio difference of the auxiliary port between designed volume and working conditions volume (Sulc, 2011),
Pd: discharge pressure,
X can be the volumetric efficiency of main port, the volumetric efficiency of auxiliary port, and the total isentropic efficiency

5. COMPRESSOR TESTING

For the testing, two scroll compressors with an economizer were used. Compressor **A** is initially designed for R404A and compressor **B** is initially designed for R134a.

The basic test configuration for the compressors is shown in Figure 2. This system makes it possible to test compressor performance, using different economizer mass flows and economizer pressures at a given discharge and suction pressure. Compressor characteristic curves are generated from the test data using equation 6 above. Test data includes the main (evaporator) mass flow rate, auxiliary (economizer) mass flow rate, compressor input power, etc. based on one test condition without an economizer, at least two test conditions with different economizer pressures, and multiple compressor speeds at the same suction and discharge pressure condition to cover the whole compressor operating envelope.

Using the compressor model generated with the test data, compressor performance can be calculated for different economizer pressures.

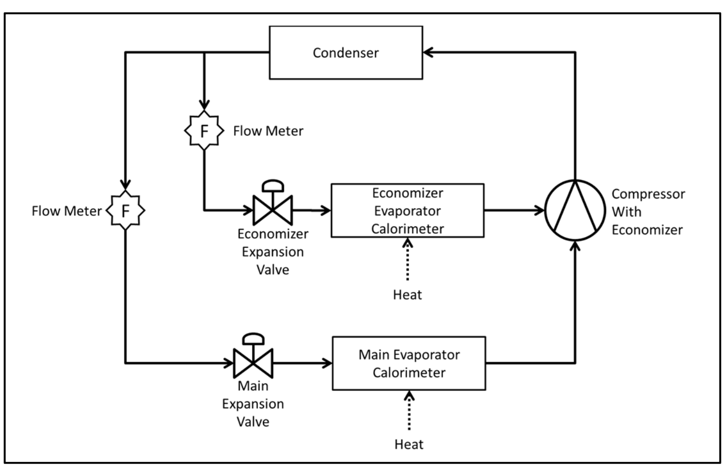

Figure 2: Test Stand Configuration for Economizer Compressors

6. REFRIGERANT COMPARISON

6.1 R404A like Refrigerants

The R404A like refrigerants (R404A, R452A, R454A and R454C) are compared at dew-point and mid-point temperatures. The refrigerants are compared regarding relative discharge temperature (DT), relative COP, relative capacity (Q), relative power (P) for both fresh and frozen conditions, shown in Figure 3 through Figure 6. Mid-point temperature evaluations generally show lower COP but higher discharge temperature, higher power and higher capacity, than dew-point temperature evaluations.

All newer refrigerants (R452A, R454A, and R454C) generally show improved COP, similar capacity, except a capacity drop for R454C, compared to R404A. In general R454C shows different thermodynamic characteristics that probably come from the relatively large glide and the relatively narrow dome (small difference between saturated gas enthalpy and saturated liquid enthalpy).

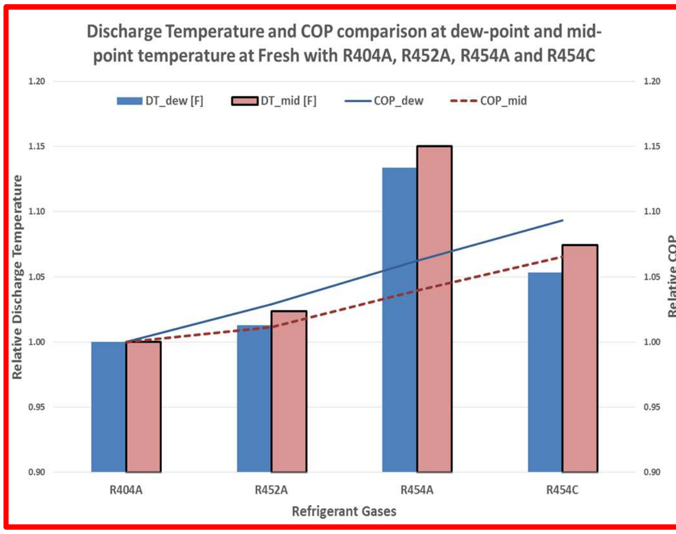

Figure 3: Discharge temp. and COP of R404A, R452A, R454A and R454C at dew-temp. and mid-temp. at fresh condition

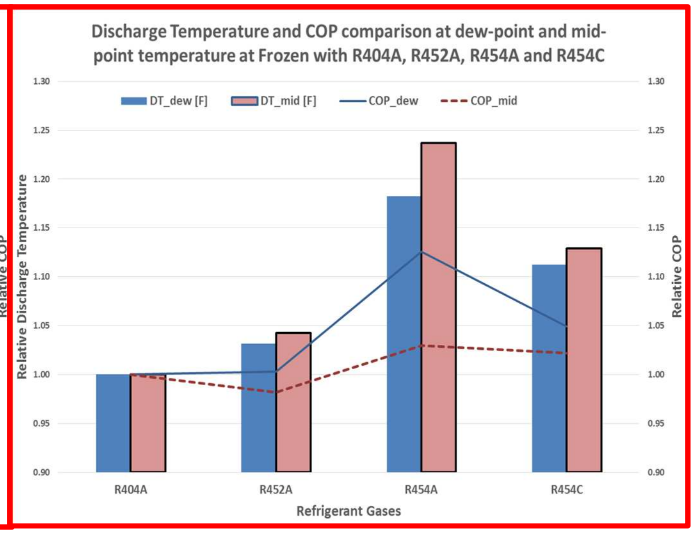

Figure 4: Discharge temp. and COP of R404A, R452A, R454A and R454C at dew-temp. and mid-temp. at frozen condition

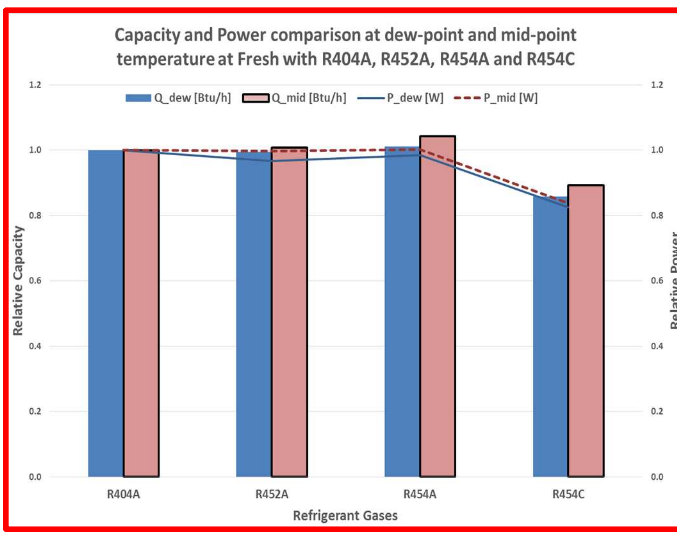

Figure 5: Capacity and Power of R404A, R452A, R454A and R454C at dew-temp. and mid-temp. at fresh condition

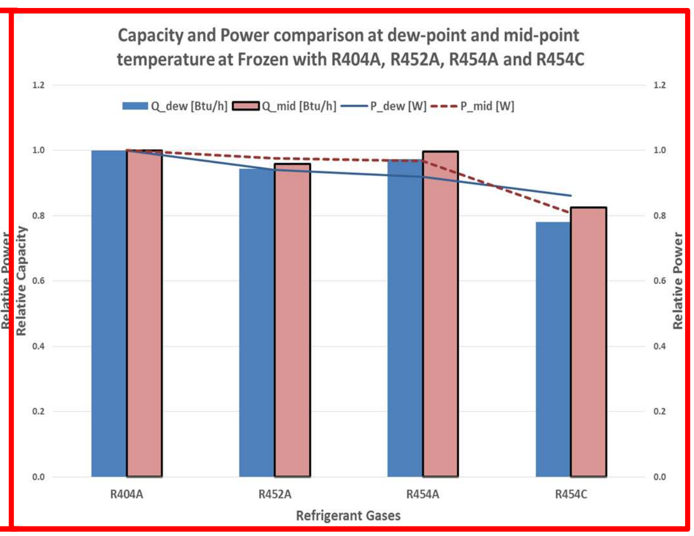

Figure 6: Capacity and Power of R404A, R452A, R454A and R454C at dew-temp. and mid-temp. at frozen condition

6.2 R134a like Refrigerants

The R134a like refrigerants (R134a, R513A and R1234yf) are compared at dew-point temperature because the refrigerants in this category have no or very small glide. The refrigerants are compared regarding relative discharge temperature (DT), relative COP, relative capacity (Q) and relative power (P), shown in Figure 7 through Figure 10.

At fresh and frozen conditions, both R513A and R1234yf show lower discharge temperature compared to R134a. At fresh, R513A and R1234yf show slightly lower COP compared to R134a, while at frozen, R513A and R1234yf show slightly higher COP compared to R134a.

At fresh, R513A show slightly higher capacity compared to both R134a and R1234yf, while at frozen, both R513A and R1234yf show a significant capacity increase over R134a. The power consumption of R134a, R513A and R1234yf follow capacity fairly well.

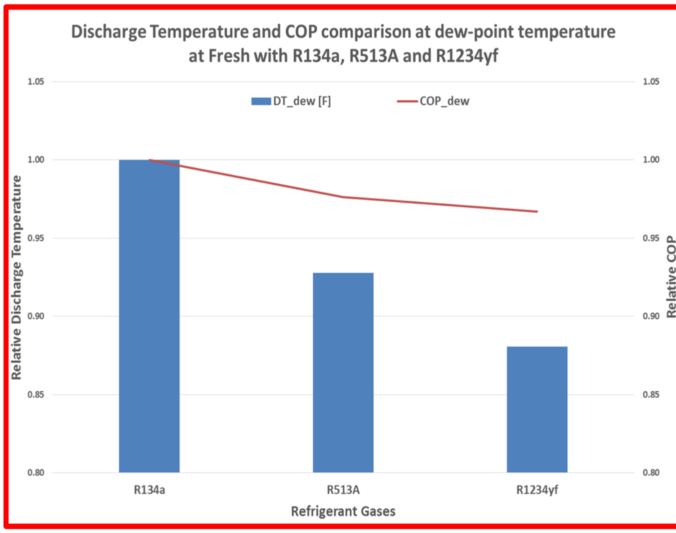

Figure 7: Discharge temp. and COP of R134a, R513A and R1234yf at fresh condition

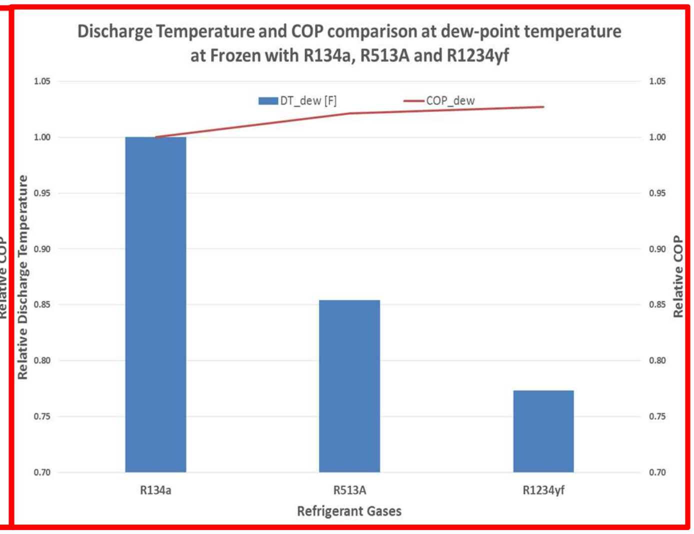

Figure 8: Discharge temp. and COP of R134a, R513A and R1234yf at frozen condition

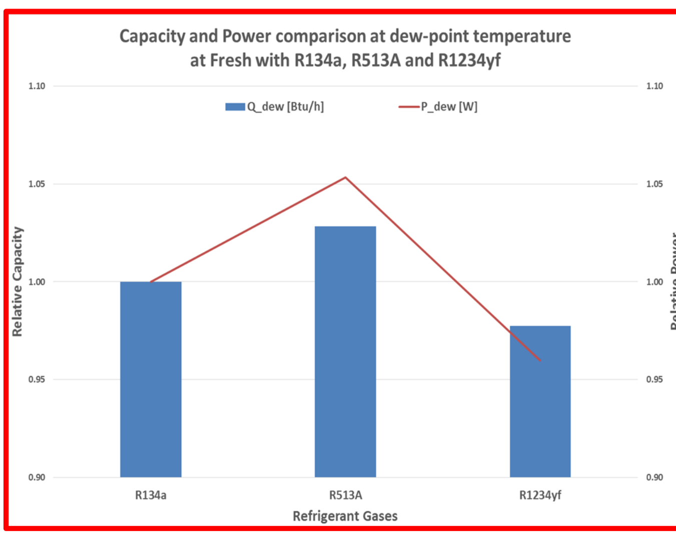

Figure 9: Capacity and power of R134a, R513A and R1234yf at fresh condition

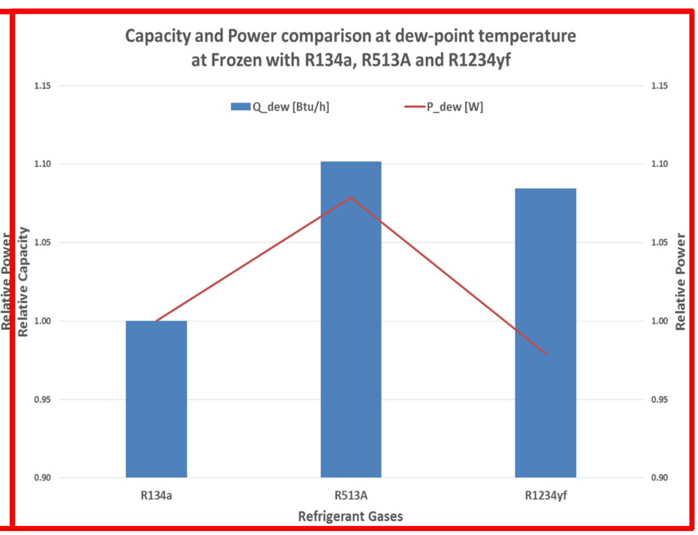

Figure 10: Capacity and power of R134a, R513A and R1234yf at frozen condition

6.3 R404A vs R134a

Comparing R404A with R134a for a hermetic compressor A and compressor B with an economizer is very limited because they have different displacements and motors. In addition, the design intent for the compressor A and compressor B are not fully known. Therefore comparing capacity and power are not discussed here. The COP of R134a significantly increases at fresh condition versus R404A as shown in Figure 11, while it slightly decreases at frozen condition as shown in Figure 12. The discharge temperature of R134a increases versus R404A at both fresh and frozen conditions.

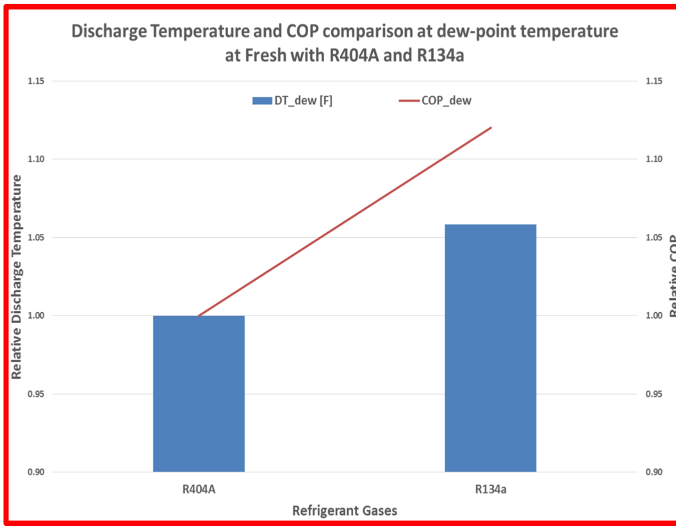

Figure 11: Discharge temp. and COP of R404A and R134a at fresh condition

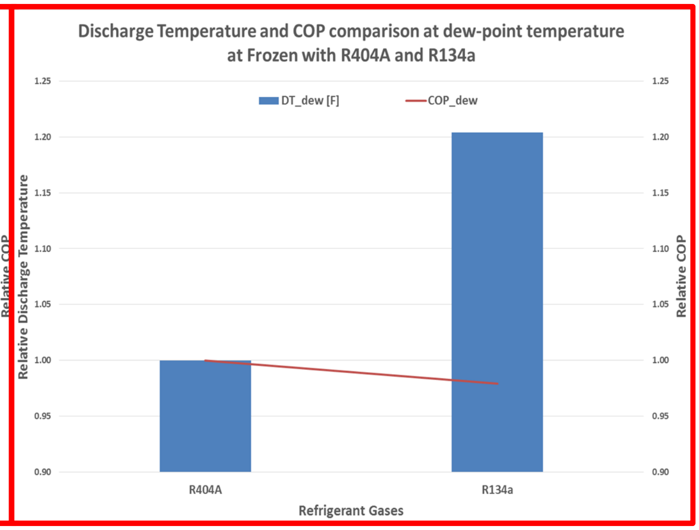

Figure 12: Discharge temp. and COP of R404A and R134a at frozen condition

7. CONCLUSION

All newer refrigerants i.e. R452A, R454A, R454C, R513A and R1234yf show promising thermodynamic behavior for hermetic scroll compressors with economizers in the studied fresh and frozen conditions.

However, for R404A like refrigerants with relative large glide such as R454A and R454C it is difficult to evaluate only the performance of compressors with economizers, without evaluating the entire refrigeration system.

For newer R134a like refrigerants such as R513A and R1234yf, it is important to understand compressor performance at relatively low suction pressure due to high sensitivity to pressure drop. Even in this case, it is important to include the entire refrigeration system.

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