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Abstract

Two economizer style compressors are explored, where the high pressure liquid is sub-cooled with a heat exchanger, while the evaporation side in the heat exchanger is connected to an intermediate compressor port. The intermediate port has a higher pressure than the suction pressure and a lower pressure than the discharge pressure. The two compressors being studied are of the scroll and screw type. One of the compressors is an open shaft compressor and the other one is a hermetic compressor. Both compressors have been modeled based on test data with R404A and at conditions that correspond to TRU (Transport Refrigeration Unit) applications. The compressor models have been used to explore lower GWP refrigerants such as R407A, R407F, N-40a, DR7 and DR33. For both compressors, each alternative refrigerant is compared against the baseline refrigerant R404A. This paper examines discharge temperature, capacity and “capacity/input power” (or COP, coefficient of performance) relative to the R404A baseline.

1. Introduction

There is a proposed need for using lower global warming potential (GWP) refrigerants in transport refrigeration applications. The most common refrigerant today for transport refrigeration is R404A, which has a GWP of almost 4,000. The last time there was a major refrigerant change, the transport refrigeration industry, together with the commercial refrigeration industry changed from R502 to R404A. One of the primary motives of the change was to reduce ozone depletion potential (ODP). Among others, some of the refrigerants considered were: R407A, R407B, R407C, R134a, R22, R507, R404A and 69L. Of these candidates, R404A and R507 demonstrated very similar behavior. These two refrigerants were pursued mainly because they have very similar vapor pressure, capacity and discharge temperature. In addition, they have zero ozone depletion potential. These similarities meant that compressor type, compressor displacement and refrigeration systems could remain relatively unchanged. However, there were issues surrounding lubricants, COP, and relatively high GWP.

The lubricant issue was solved with a switch from alkylated benzene or mineral oil based lubricants to Polyolester (POE) based lubricants. Maintaining high COP was not as important at the time, because energy prices were relatively low. Today, high COP is much more important for two reasons. Overall energy prices are considerably higher than during the last refrigerant change and COP is affecting indirect GWP. The high direct GWP of R404A and R507 is primarily due to the R143a component. All potential replacements for R404A do not include R143a. This paper mainly deals with non-flammable refrigerants (class A1), which typically have a GWP in the range of 1,000 to 2,000. One example of a mildly flammable refrigerant (class A2L) is also included. Mildly flammable refrigerants typically have a GWP in the range of 4 to 700. Flammable refrigerants such as hydrocarbons (class A2) are not discussed. Carbon dioxide (CO2) is not discussed because of its very different fluid properties. See Table 1 for the refrigerants being simulated.

Table 1: Basic Characteristics of Simulated Refrigerants

Vendor	Refrigerant Name(s)	Flammability Class	GWP	MW	R32	R125	R143a	R134a	R1234yf	R1234ze	Temperature Glide at 1 atm. (°F)	Critical Temperature (°F)
Baseline	R404A	A1	3922	97.606	0.0%	44.0%	52.0%	4.0%	0.0%	0.0%	1.4	161.70
Common	R407A	A1	2107	90.113	20.0%	40.0%	0.0%	40.0%	0.0%	0.0%	11.5	180.10
Common	R407F	A1	1825	82.06	30.0%	30.0%	0.0%	40.0%	0.0%	0.0%	11.5	180.80
Honeywell	HDR-47; N-40a	A1	1346	87.037	25.0%	25.0%	0.0%	21.0%	9.0%	20.0%	13.7	186.00
DuPont	DR-33; XP40; R449A	A1	1410	87.44	24.0%	25.0%	0.0%	26.0%	25.0%	0.0%	10.9	183.00
DuPont	DR-7	A2L	246	79.799	36.0%	0.0%	0.0%	0.0%	64.0%	0.0%	10.7	186.00

2. Compressor Type and Refrigeration System

Traditionally, reciprocating compressors have been used in transport refrigeration applications. These compressors are single-stage and typically have two, four or six pistons. The refrigeration system for cooling is relatively simple, having one expansion valve and a suction gas to condensed liquid heat exchanger (See Figure 1). This type of compressor and refrigeration system is well proven, but also has some disadvantages. Due to the suction to liquid heat exchanger, the discharge temperature becomes rather high, which limits the choice of refrigerants. The compressor and system also limit the choice of basic vapor pressure, especially for lower vapor pressure refrigerants such as R134a or R1234yf. To accommodate lower vapor pressure refrigerants, reciprocating compressor footprints must become rather large for the transport application. Other factors that affect compressor footprint are oil sump size, oil sump placement and the capacity control system. This paper deals with economizer or vapor injection rotary compressors of the scroll and screw type. This paper also compares the economizer refrigeration system shown in Figure 2, using alternative refrigerants.

In the economizer system, the suction gas and economizer gas can be maintained with relatively low superheat. The economizer gas is introduced to a flute or pocket in the compression process that is between suction and discharge pressure. This “extra” economizer gas provides the benefit of lowering the compressor discharge temperature. In extreme operating conditions, such as low cargo box and high ambient temperatures, additional techniques must be used to regulate compressor discharge temperature. One technique is liquid injection, which affects the compressor’s COP. However, liquid injection has a minimal effect on capacity if liquid is introduced to a flute or pocket that is isolated from suction pressure. Another technique to control discharge temperature is compressor oil cooling, which can improve COP. Both liquid injection and oil cooling add cost and complexity to the system. Economizer systems with both screw and scroll compressors have been in operation for quite some time in transport refrigeration applications [Sjoholm, 1996].

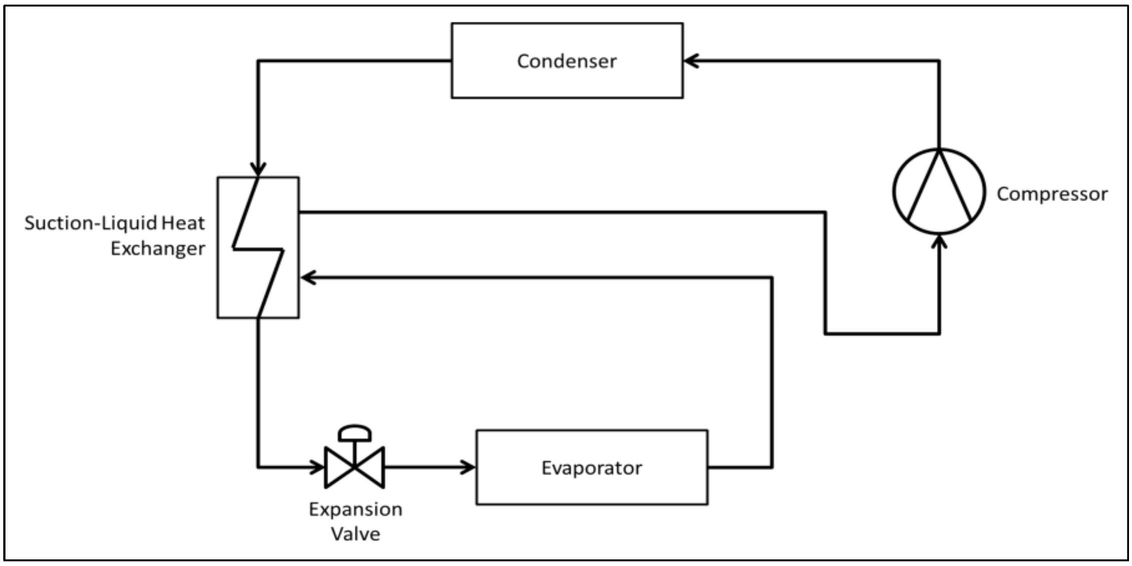

Figure 1: Vapor Compression Cycle with Suction to Liquid Heat Exchanger

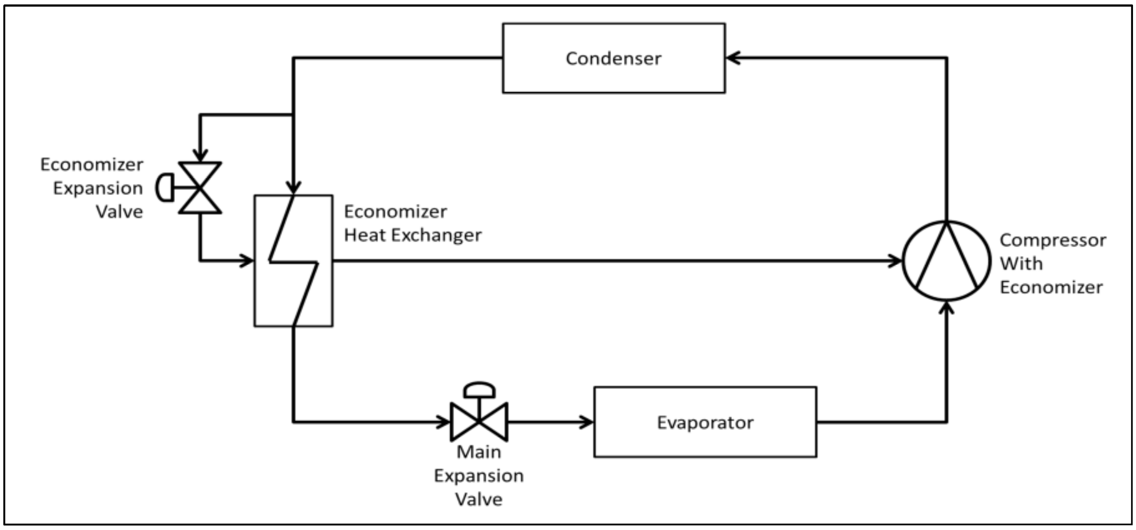

Figure 2: Vapor Compression Cycle with Economizer Heat Exchanger & Compressor

Two economizer type compressors have been tested with R404A. One compressor has an open drive shaft (Compressor A) and the other compressor is driven by a refrigerant gas cooled motor (Compressor B). One of the compressors is a screw type and the other one is a scroll type. The basic test configuration for the compressors is shown in Figure 3. This system makes it possible to test compressor performance, using different economizer mass flow and economizer pressure for a given discharge and suction pressure. In this case, the total isentropic efficiency is defined as "strict isentropic efficiency" [Sjoholm, 1986]. Compressor characteristic curves are generated from test data using numerical methods [Erickson, 1998]. Using this approach, compressor performance can be evaluated for different economizer pressures or balance points. In addition, refrigerants with different thermodynamic behavior and properties can be simulated. All the simulation data is compared at conditions corresponding to -10°F cargo box temperature and 80°F ambient temperature.

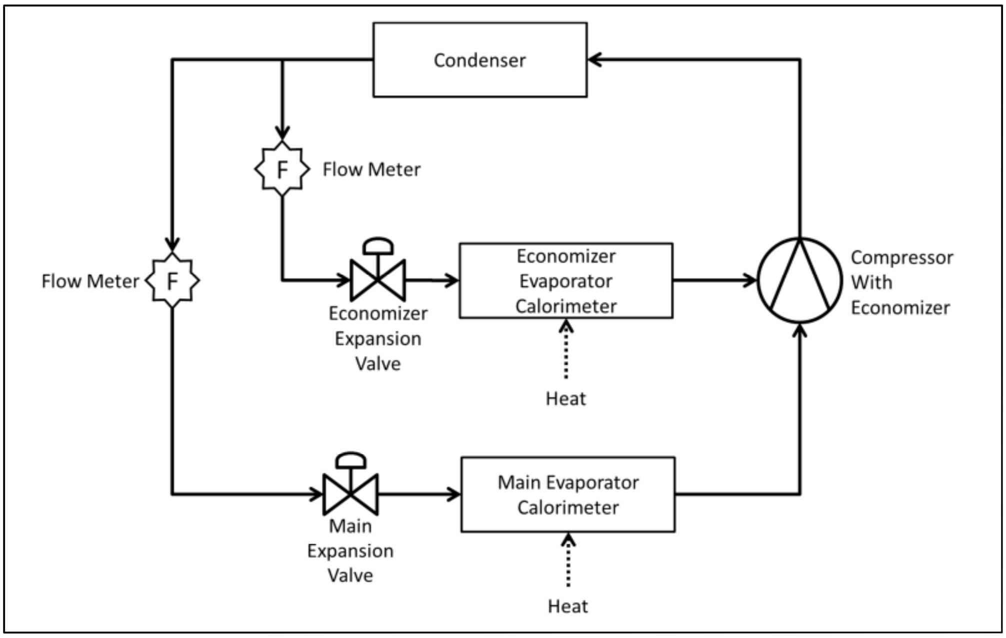

Figure 3: Test Stand Configuration for Economizer Compressors

In the comparison, the refrigerants being simulated have various amounts of glide. To account for this, the following assumptions have been made: The average condensing temperature is based on the 1) average bubble temperature and 2) average dew point temperature. The average evaporating temperature is based on an 1) economizer sub-cooled temperature from the bubble point at condensing pressure and 2) a dew point at the evaporating pressure.

An ideal economizer heat exchanger is assumed. In an ideal economizer heat exchanger, the vapor and liquid refrigerant leave the heat exchanger at the same temperature. For the main suction gas and the economizer suction gas, a superheat of 40°F is used in the simulation. The isentropic and volumetric efficiency equations used in the simulation are the same for all refrigerants. The calculated efficiencies will be different for each refrigerant depending on differences in pressures, temperatures and enthalpies. All simulation results are compared with R404A as the baseline refrigerant.

3. Simulation Results

Figure 4 shows discharge temperature simulation results for Compressors A and B, relative to R404A for refrigerants DR33, R407F, DR7, N-40a and R407A. All refrigerants being simulated show higher discharge temperatures relative to R404A.

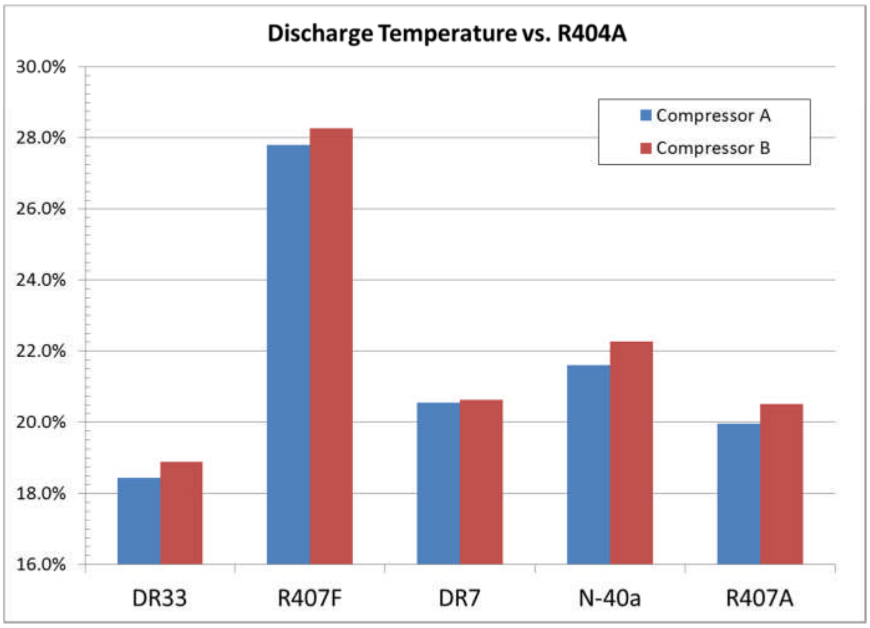

Figure 4: Discharge Temperature versus R404A Baseline

Figure 5 shows capacity simulation results for Compressors A and B, relative to R404A. Refrigerants DR7 and R407F show higher capacity relative to R404A. DR33, N-40a and R407A show lower capacity relative to R404A.

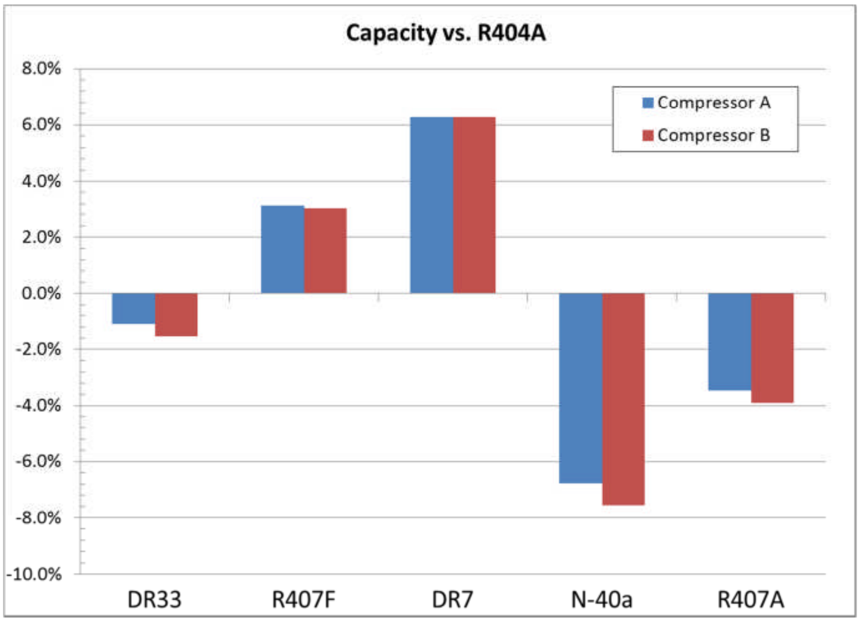

Figure 5: Capacity versus R404A Baseline

Figure 6 shows "capacity/input power" simulation results for Compressor A and B, relative to R404A. All refrigerants show higher "capacity/ input power" relative to R404A.

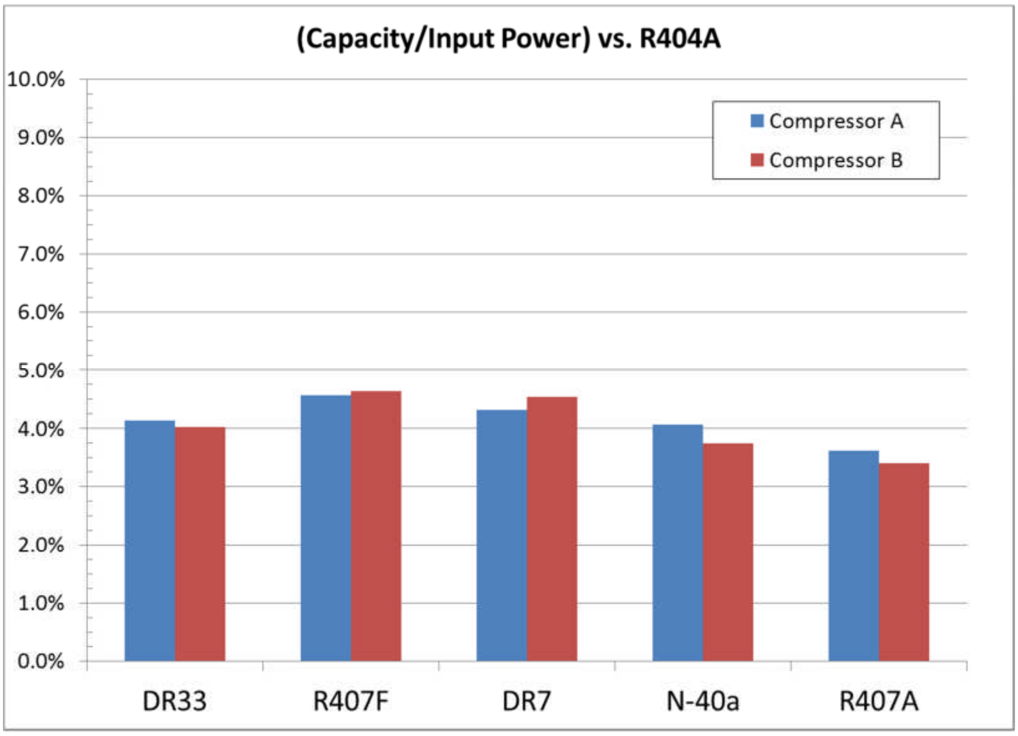

Figure 6: Capacity/Input Power versus R404A Baseline

4. Conclusion

All refrigerants being simulated show higher discharge temperatures than R404A. In practice, higher discharge temperatures can be managed using liquid refrigerant injection or compressor oil cooling for a portion of the operational envelope. The capacity changes range from -7% to +6% relative to R404A. To account for changes in capacity, different compressor displacements or speeds can be implemented. All refrigerants show approximately 4% higher "capacity/input power" relative to R404A. The trends appear to be similar regardless of whether the compressor is hermetic or open drive.

Of the refrigerants being simulated, DR7 and DR33 appear to be the most suitable candidates for use in an ideal economizer system. If the desire is to use a low GWP refrigerant that behaves similarly to R404A (or R502), none of the simulated refrigerants are suitable.

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