A Comparison of an R22 and an R410A Air Conditioner Operating at High Ambient Temperatures

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ABSTRACT

R22 and R410A split air-conditioning systems were tested and compared as outdoor temperature ranged from 27.8 °C (82.0 °F) to 54.4 °F (130 °F). The R410A system tests were extended to 68.3 °C (155.0 °F) ambient temperature with a customized compressor. Capacity and efficiency of both systems decreased linearly with increasing outdoor temperature, but the R410A system performance degraded more than the R22 system performance. Operation of the R410A system was stable during all tests, including those with the customized compressor extending up to the 68.3 °F (155.0 °F) outdoor temperature and resulting in a supercritical condition at the condenser inlet. No noticeable changes in noise level or operation of the system was noted.

NOMENCLATURE

Comp compressor
COP   coefficient of performance, W/W
EER   energy efficiency ratio, Btu/Wh
SEER seasonal energy efficiency ratio, Btu/Wh
T ambient temperature, °C (°F)

T_{cond} condenser temperature, K
scfm   standard cubic feet per minute (defined by ASHRAE Standard 37)
T_{evap} evaporator temperature, K
TXV   thermostatic expansion valve

INTRODUCTION

Operation of an air conditioner at elevated ambient temperatures inherently results in a lower coefficient of performance (COP). This conclusion comes directly from examining the Carnot cycle. The COP relation, COP=T_{evap}/(T_{cond}-T_{evap}), indicates that the COP decreases when the condenser temperature increases at a constant evaporation temperature. This theoretical indication derived from the reversible cycle is valid for all refrigerants. For refrigerants operating in the vapor compression cycle, the COP degradation is greater than that for the Carnot cycle and varies among fluids. The two most influential fundamental thermodynamic properties affecting this degradation are a refrigerant's critical temperature and molar heat capacity. (e.g., McLinden 1987, Domanski 1999). For a given application, a fluid with a lower critical temperature will tend to have a lower COP.

The lower critical temperature of R410A versus that of R22 (70.1 °C (158.1 °F) vs. 96.2 °C (205.1 °F)) indicates that degradation of performance at high ambient temperature should be greater for R410A than R22.

Wells et al. (1999) investigated split system air-conditioning units in the 12 to 13 SEER range. Their test results were normalized with respect to cooling capacity at the 35.0 °C (95.0 °F) outdoor condition. The R22 system cooling capacity decreased by 14 % at an outdoor temperature of 51.7 °C (125.0 °F). The R410A system cooling capacity decreased nonlinearly by 22 % at the same condition. EER at 51.7 °C (125.0 °F) decreased by 35 % and 42 % for the R22 and R410A systems, respectively. Their study showed that performance varied between units equipped with a TXV, short tube, or capillary as the expansion device.

Motta and Domanski (2000) simulated the performance of R22 and four of its alternatives in an air conditioner as the outdoor air temperature ramped from 25.0 °C (77.0 °F) to 55.0 °C (131.0 °F). When the performance of the R410A system was normalized with respect to the performance of the R22 system over the entire temperature range, the R410A system EER was approximately 2 % lower at 25.0 °C (77.0 °F) and 6.5 % lower at 55.0 °C (131.0 °F). Their simulations also included the addition of a liquid line to suction line internal heat exchanger. It was shown in the simulations that the 6.5 % loss in COP (EER) could be reduced to 3.2 % by the addition of an internal heat exchanger.

EXPERIMENTAL SET UP

R22 and R410A Systems Selected

Manufacturer data listed the R22 system with a SEER of 12.5 and the R410A system with a SEER of 13. The two systems comprised the same evaporator and condenser. The main differences between the compared systems were the compressor, lubricant, and refrigerant. The piping arrangement included different thermostatic expansion valves (TXVs) and liquid line filters.

Figure 1 below shows the circuiting of the condenser finned tube coil. The condenser was 71.1 cm (28.0 in) x 204.5 cm (80.5 in) finned length, 8.7 fins/cm (22.0 fins/in).
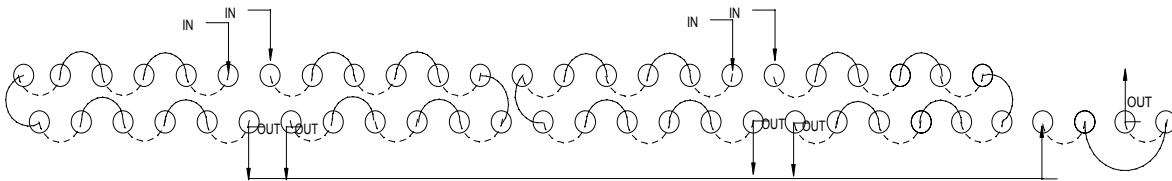

Figure 1: Circuiting of R22 and R410A condenser coil (airflow from bottom to top)

Figure 2 shows the circuiting of the evaporator. The evaporator was 55.9 cm (22.0 in) x 66.0 cm (26.0 in) finned length with 4.7 fins/cm (12.0 fins/in). The evaporator and condenser had a fin thickness of 0.1143 mm (0.0045 in).
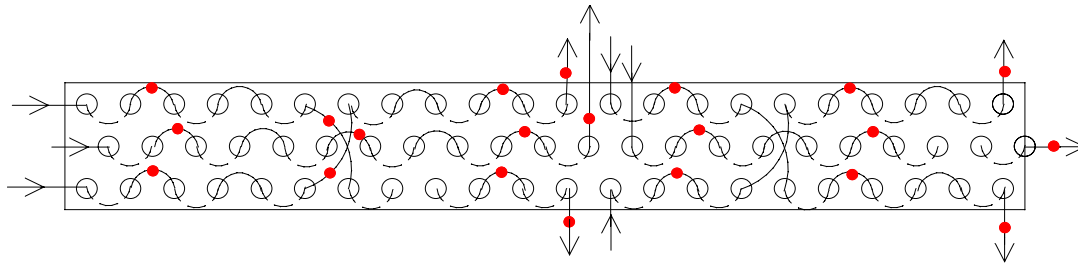
Figure 2: Circuiting of R22 and R410A evaporator coil (airflow from bottom to top)

R410A system tests were performed using two compressors designated as compressor #1 and compressor #2. The original R410A system was supplied with compressor #1. This system was tested at outdoor temperatures up to 54.4 °C (130.0 °F) above which the internal safeties for the compressor prevented the system from operating continuously. Since one of the objectives of the study was to measure performance of the R410A system when the condenser operated above the critical point, and this did not occur during the 54.4 °C (130.0 °F) test, the second compressor, designated as compressor #2, was customized for testing at higher outdoor temperatures. Compressor #2 had the internal safeties removed and had a more powerful electric motor than compressor #1.

Measurement Uncertainty

Measurement points (Table 1) consisted of temperatures, pressures, pressure difference, temperature difference, dew-point temperature, and power. Refrigerant properties were obtained from REFPROP 6.0 (McLinden et al. 1998). Table 1 lists measured and calculated quantities

Table 1: Measurement uncertainty at 95 % confidence level

Quantity	Range	Uncertainty
Pressure	0 –6895 kPa (0 – 1000 psia)	±10.3 kPa (±1.5 psia)
Temperature	-26.1 °C - 93.3 °C (-15 °F – 200 °F)	±0.25 °C (±0.45 °F)
Temperature Difference	0 °C – 27.8 °C (0 °F – 50 °F)	±0.25 °C (±0.45 °F)
Barometric Pressure	0 - 1270 mmHg (0 - 50 inHg)	±0.34 mmHg (±0.0135 inHg)
Dew-point Temperature	0 °C – 50 °C (32 °F – 122 °F)	±0.2 °C (±0.36 °F)
Air Pressure Difference	0 – 622 Pa (0 – 2.5 inH$_2$O)	±1.56 Pa (±0.00625 inH$_2$O)
Power	0 – 5000 Watts	±0.5 %

and their uncertainties for the 95 % confidence limits. The 95 % relative uncertainty in capacity and COP (EER) varied from 2.9 % to 3.5 % and 3.5 % to 5.4 %, respectively.

EXPERIMENTAL PROCEDURE

The tests were performed according to ASHRAE Standard 37 (1988). The same indoor conditions of 26.7 °C (80.0 °F) dry-bulb and 19.4 °C (67.0 °F) wet-bulb were used for all tests. The outdoor conditions varied according to Table 2. Before any testing began the system charge was set according to manufacturer's recommendations. The charge for both units was set by measuring the difference between the liquid line temperature exiting the condensing unit and the outdoor air temperature. For the R22 system this temperature difference was 2.8 °C (5.0 °F). For the R410A system this temperature difference was 3.3 °C (6.0 °F). A steady flow of condensate from the evaporator was present for 1 hour for all tests. The air enthalpy method was used as the primary measurement of air-side capacity. The refrigerant enthalpy method was used as the secondary measurement of capacity. The two measurements always agreed within 3.5 %. The source report, Domanski and Payne 2002, contains a full description of the experimental setup and test procedure.

Table 2: Test conditions

Location	Cooling		Tolerance °C (°F)
	Setpoint °C (°F)		
Indoor Dry-bulb Temperature	26.7 (80.0)		±0.28 (±0.5)
Indoor Dew-point Temperature	15.8 (60.4)		±0.28 (±0.5)
Outdoor Dry-bulb Temperature	27.8 (82.0)[1] 35.0 (95.0)[2] 46.1 (115.0)[2] 51.7 (125.0)[1] 54.4 (130.0)[2]	57.2 (135.0)[3] 60.0 (140.0)[4] 65.6 (150.0)[4] 66.7 (152.0)[4] 68.3 (155.0)[4]	±0.28 (±0.5)
Evaporator Airflow, L/s (scfm)	566.3 (1200.0)		

1) R22 and R410A Compressor #1   2) R22 and both R410A Compressors   3) Only R22   4) Only R410A Compressor #2

EXPERIMENTAL RESULTS

R22 System Tests

Figure 3 shows the cooling capacity and COP as a function of the outdoor temperature. Capacity ranged from 11.78 kW (40201 Btu/h) to 8.71 kW (29711 Btu/h) over the range of outdoor temperatures. This was a decrease of 26.1 % from the high value at 27.8 °C (82 °F) to the low value at 57.2 °C (135 °F). For the same outdoor temperature range, the COP (EER) decreased by 60.3 % from 5.36 (18.3 Btu/Wh) to 2.14 (7.3 Btu/Wh). The sensible heat ratio remained at 0.8 ± 0.05 for all tests. The TXV kept the evaporator superheat between 5.1 °C (9.2 °F) and 9.1 °C (16.3 °F), and the condenser subcooling between 3.2 °C (5.7 °F) and 8.3 °C (15.0 °F).
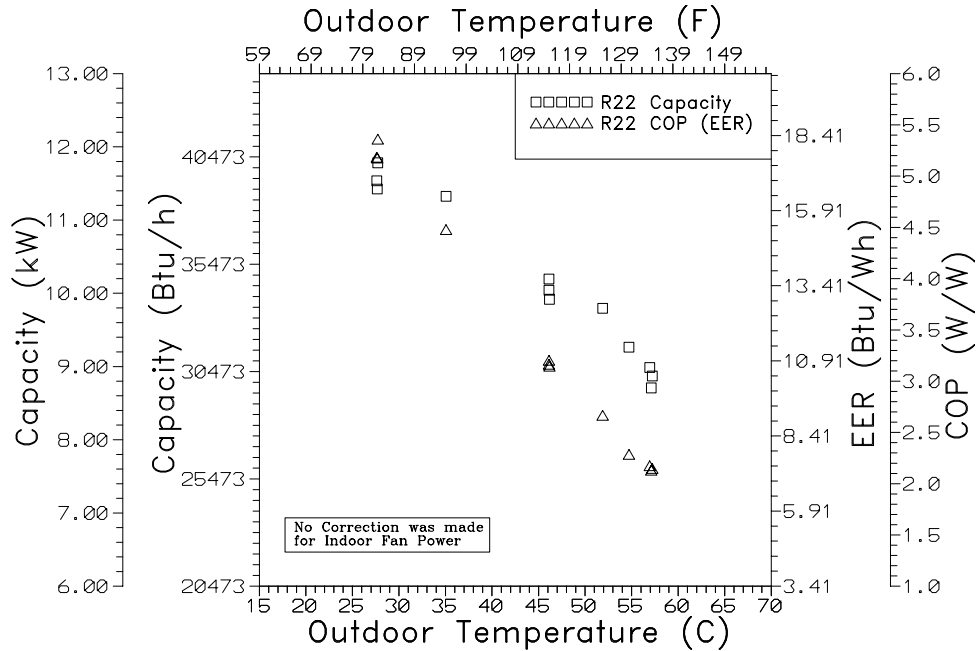
Figure 3: R22 cooling capacity and COP (EER) as a function of outdoor temperature

R410A System Tests

Figure 4 shows the cooling capacity and COP (EER) of the R410A system tested with two compressors. Since the housing and displacement for both compressors were the same, the relative differences in measured capacity and efficiency are mostly reflective of the different sizes of the electric motor used in compressor #1 and compressor #2, and their electric motor efficiency and RPM vs. load characteristics. If we evaluate the combined data obtained with both compressors, the air-side capacity decreased in a nearly linear manner as the outdoor temperature increased from 27.7 °C (82.0 °F) to 68.3 °C (155.0 °F). Over this temperature range, air-side capacity decreased from 11.8 kW (40345 Btu/h) to 6.7 kW (22699 Btu/h); a decrease of 43.7 %. The COP (EER) also decreased linearly by 80.3 % as it dropped from 5.36 (18.3 Btu/Wh) to 1.06 (3.6 Btu/Wh). Similar to the R22 tests, the sensible heat ratio was 0.8 ± 0.05 for all tests.

The R410A system with compressor #2 produced compressor discharge conditions above the critical point during tests at outdoor temperatures of 65.6 °C (150.0 °F), 66.7 °C (152.0 °F), and 68.3 °C (155.0 °F), which was a limit for our chambers. However, the condenser outlet pressure was still below the critical pressure and subcooled refrigerant entered the TXV. Over the whole range of the outdoor temperatures, the evaporator superheat and condenser subcooling were maintained within 1.8 – 2.5 °C (3.3 – 4.5 °F) and 4.4 – 6.4 °C (8.0 – 11.5 °F), respectively. No noticeable changes in noise level or operation of the system was noted. The good control of superheat and subcooling by the TXV inlet may be the main factor contributing to the stability of the system.
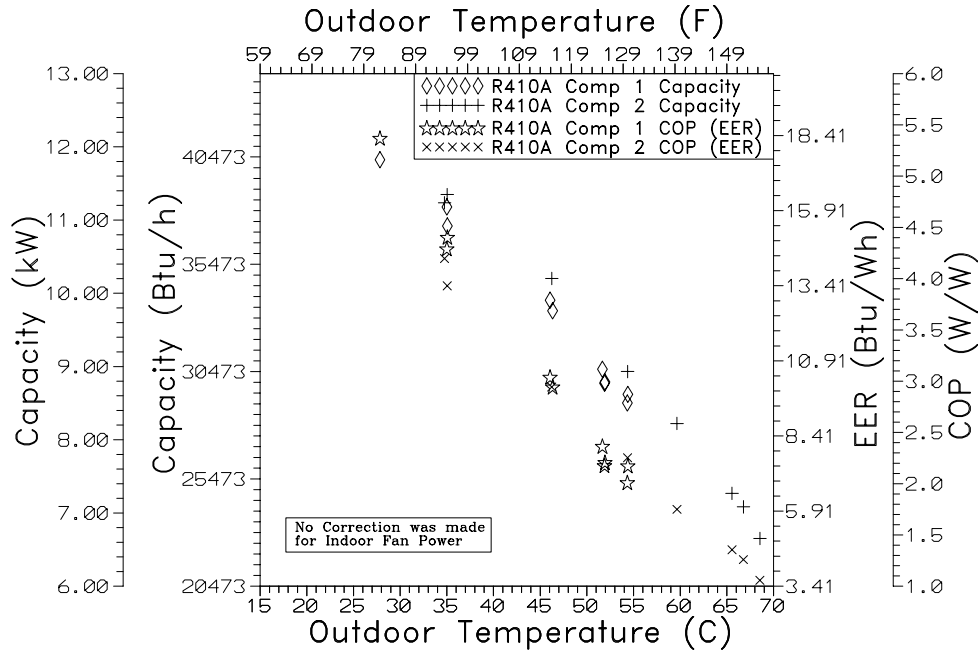
Figure 4: R410A cooling capacity and COP (EER) as a function of outdoor temperature

Comparison of Performance of R22 and R410A Systems

To compare the performance of the R410A systems to the R22 system, capacity and COP (EER) data for the R22 system were fit to a function of the outdoor temperature. Figure 5 presents ratios of the measured R410A/compressor #1 capacity to the R22 capacity calculated from equation 1.

$$Q(\text{kW}) = 11.877 - 1.638 \times 10^{-5} \cdot T(\text{C})^3 \tag{1a}$$

$$Q(\text{Btu/h}) = 42196.9 - 4.8705 \times 10^{-3} \cdot T(\text{F})^3 \tag{1b}$$

The cubic polynomial above fit the R22 capacity data with a Pearson correlation coefficient (R^2) of 0.978 and a fit standard error of 173.2 W (591.1 Btu/h).

The capacities of the R22 system and R410A system were equal at the 35.0 °C (95.0 °F) outdoor temperature. At the 27.8 °C (82.0 °F) rating point, the R410A system capacity was approximately 2 % greater than that of the R22 system. As the outdoor temperature increased, the capacity of the R410A system decreased more rapidly than the R22 system capacity, and at the 54.4 °C (130.0 °F) test point was 9 % below the R22 value.

The power law function presented as equation 2 fits the R22 COP (EER) data with a Pearson correlation coefficient (R^2) of 0.998 and a fit standard error of 0.081 (0.2749 Btu/Wh).

$$\text{COP} = 9.459 - 0.3323 \cdot T(\text{C})^{0.7654} \tag{2a}$$

$$\text{EER} = 36.692 - 0.3611 \cdot T(\text{F})^{0.8981} \tag{2b}$$

The efficiency trend was similar to the trend for capacity; however, the performance degradation of R410A was more pronounced, as shown in Figure 6. At the 27.8 °C (82.0 °F) rating point, the COP (EER) of the R410A system was a few percent higher than that for the R22 system. At the 35.0 °C (95.0 °F) rating point, at which the capacities were equal, the R410A COP (EER) was approximately 4 % below the R22 COP (EER). At the highest ambient temperature of 54.4 °C (130.0 °F), the R410A COP (EER) was about 15 % lower than the COP (EER) of the R22 system. In addition to typical measurement uncertainties, we attribute the scatter shown in the COP (EER) ratios for a given outdoor temperature to day-to-day variations of voltage at the testing facility. However, these voltage variations were always within the range allowed by the ARI 210/240 (1994).
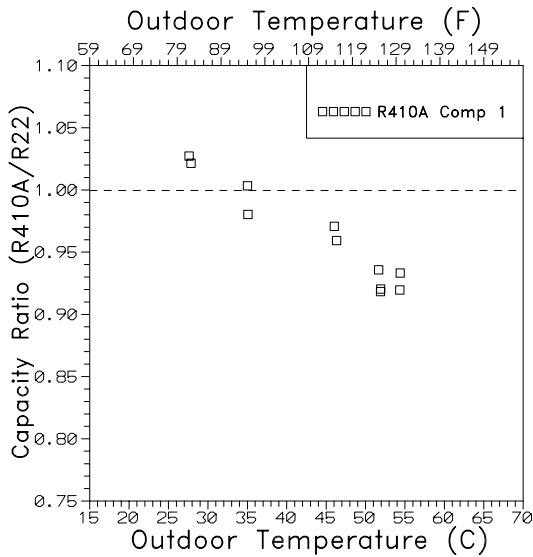

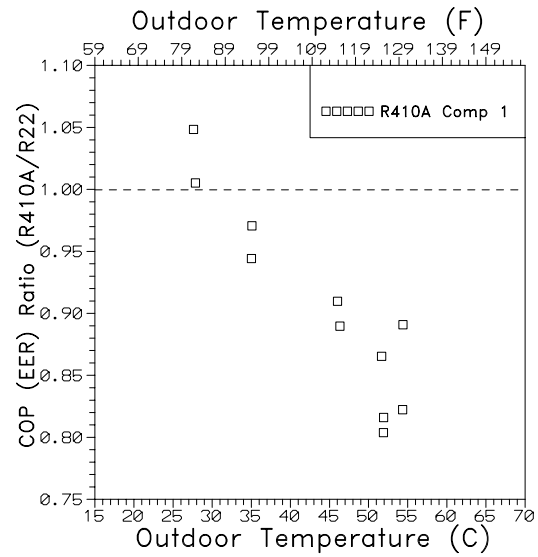Figure 5: Cooling capacity of R410A
system relative to R22 system

Figure 6: Cooling EER of R410A system
relative to R22 system

CONCLUSIONS

R22 and R410A split air-conditioning systems were tested and compared as outdoor temperature ranged from 27.8 °C (82.0 °F) to 54.4 °F (130 °F). The R410A system tests were extended to 68.3 °C (155.0 °F) ambient temperature with a customized compressor. When outdoor temperature increased, the R410A system performance degraded more than the R22 system performance. While capacities of both systems were approximately equal at the 35.0 °C (95.0 °F) rating point, at the 54.4 °C (130.0 °F) outdoor temperature the R410A capacity was 9 % below that of R22. For the same test points, the R410A COP (EER) was 4 % and 15 % lower than the R22 COP (EER), respectively. The degradation trend was linear. Since both systems employed identical heat exchangers and similar design (scroll) compressors, the refrigerant and lubricant used in each system had the dominant effect on the measured results.

Operation of the R410A system was stable during all tests, including those with the customized compressors extending up to the 68.3 °F (155.0 °F) outdoor temperature and

resulting in a supercritical condition at the condenser inlet. No noticeable changes in noise level or operation of the system was noted.

ACKNOWLEDGMENT

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