# Impact of Elevated Ambient Temperatures on Capacity and Energy Input to a Vapor Compression System – Literature Review

Letter report for ARTI 21-CR Research Project: 605-50010/605-50015

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## Theoretical Background

Operation of a system 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 refrigerant performance in the vapor compression cycle are 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 higher volumetric capacity ($Q_{vol}$) and a lower coefficient of performance (COP). The difference between COPs is related to different levels of irreversibility because of the superheated vapor horn and the throttling process, as shown conceptually in Figure 1. The levels of irreversibility vary with operating temperatures because the slopes of the saturated liquid and vapor lines change, particularly when approaching the critical point.

Refrigerants with a low critical temperature have a high pressure, a low drop of saturation temperature for a given pressure drop, and a low condenser-to-evaporator pressure ratio. These properties offer some advantages, which can be exploited in a real system for the betterment of its performance. Some researchers reported that a low pressure ratio promotes an improved compressor isentropic efficiency (e.g., Rieberer and Halozan, 1998). The low drop of refrigerant saturation temperature for a given pressure drop ($dT/dP|_{sat}$) allows designing heat exchangers with a high refrigerant mass flux, which promotes an improved refrigerant-side heat transfer coefficient.

The condenser temperature increases at elevated ambient temperatures, which causes changes in refrigerant transport properties. These changes do not override the thermodynamic consideration, but they should be noted to foster complete understanding of the phenomena involved. The changes of liquid viscosity, conductivity, and heat capacity are smooth and favorable while approching the critical temperature (viscosity decreases, conductivity and heat capacity increase). In the supercritical region, density has a smooth transition above the critical point, but specific heat has a pronounced peak, as Figure 2 shows for R-410A (Bullock, 1999). This trend in the neighborhood of the critical point is typical for all fluids as has been recently presented for carbon dioxide in several studies (e.g., Olson, 1999 who showed that conductivity and viscosity have a smooth transition as well).

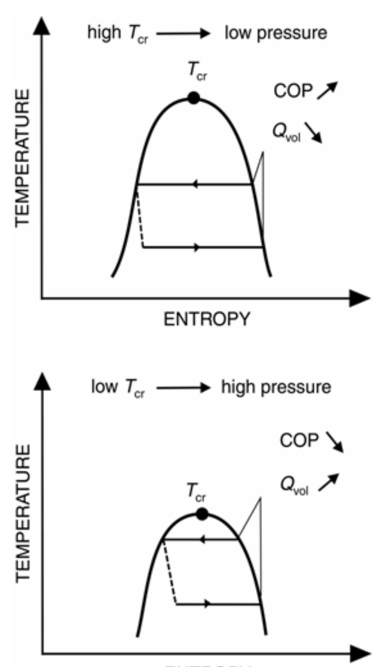


Figure 1. Impact of critical temperature of system performance

Because of the abrupt change in specific heat (Figure 2), the heat transfer coefficient at constant pressure (Figure 3) has a peak while approaching the critical temperature.

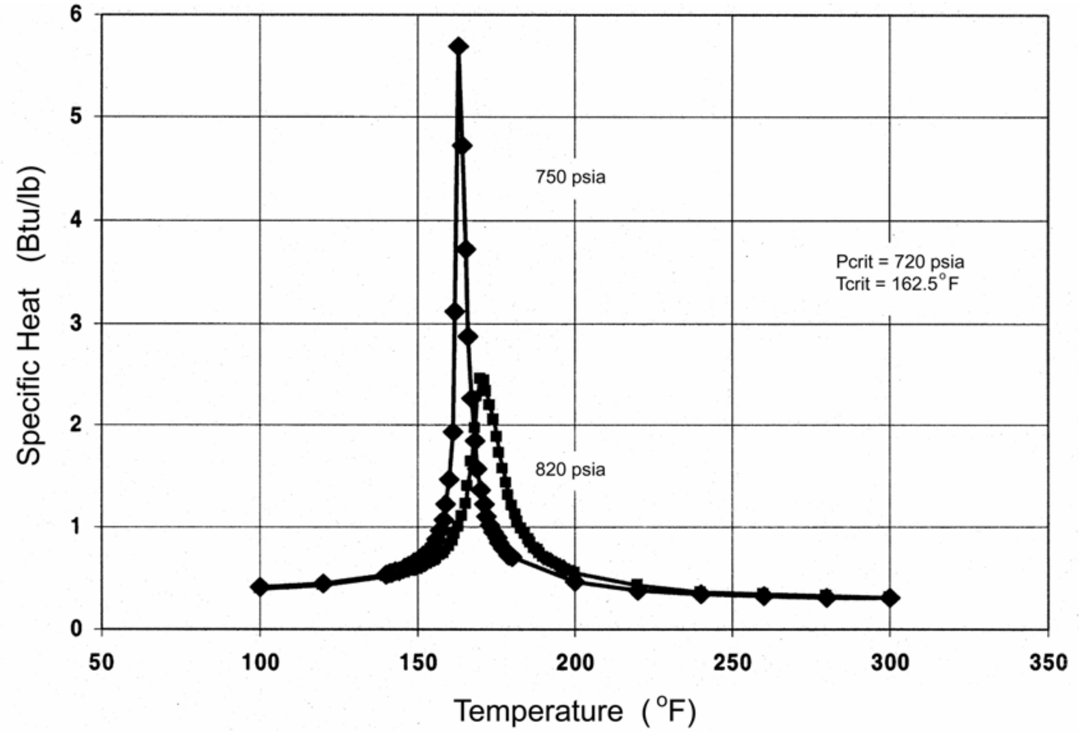


Figure 2. Refrigerant Specific Heat versus Temperature and Pressure: R-410A (Bullock, 1999).

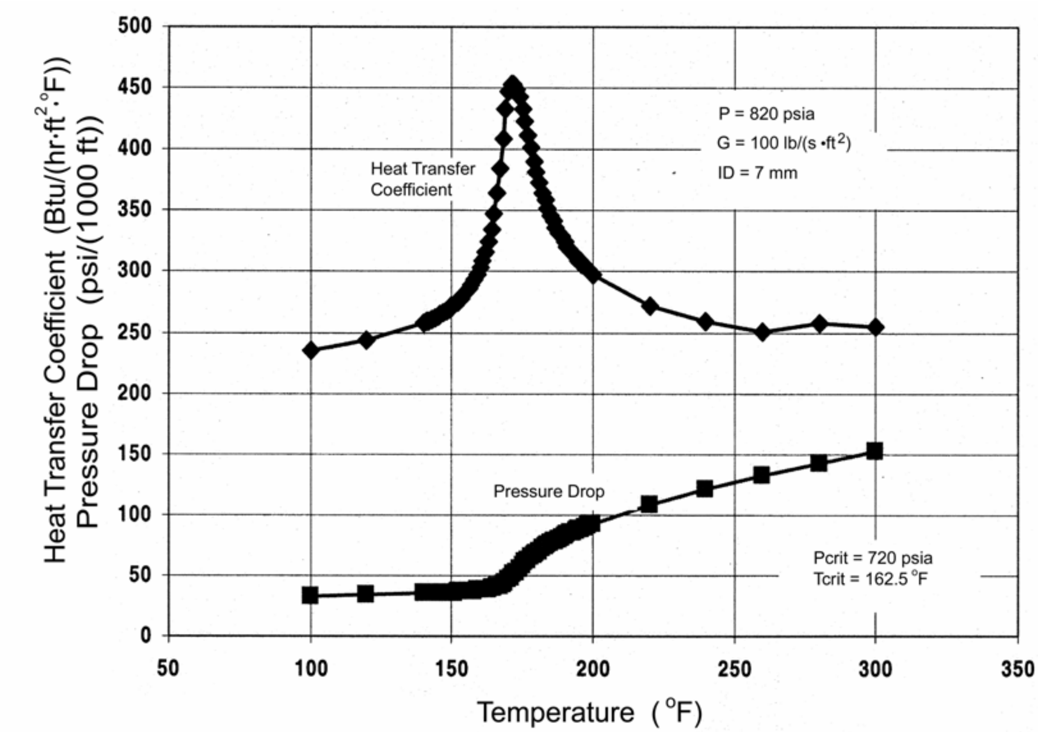


Figure 3. Refrigerant Pressure Drop and Convection Heat-Transfer Coefficient for Supercritical Flow of R-410A (Bullock, 1999).

## Literature review

We were able to locate only a few publications concerned with air conditioner operation at elevated temperatures. They are reported here along with two seminar presentations made during the ASHRAE summer meeting in 1999. LeRoy et al. (1997) investigated capacity and power demands of R-22 unitary systems under extreme operating conditions. The main goal of the study was to validate performance predictions of three public-domain heat pump simulation models. The authors used data of ten systems from tests at the 35 °C (95 °F) rating point and at higher outdoor temperatures. Three of these systems were tested at 46.1 °C (115 °F) and another three at 51.7 °C (125 °F) with the same indoor conditions of 26.7 °C (80 °F) dry-bulb and 19.4 °C (67 °F) wet-bulb temperature. The reported decrease in capacity at 46.1 °C (115 °F) was in the 14 % to 19 % range while the decrease in the energy efficiency ratio (EER) was in the 24 % to 41 % range. At 48.8 °C (120 °F), the capacity and EER decreases were within the 11 % to 20 % range and 34 % to 39 % range, respectively. These data indicate that performance degradation at high ambient temperature varies significantly from one system to another.

Chin and Spatz (1999) explored some of the advantages and disadvantages of R-410A use in air conditioning systems. They used compressor performance data and a heat pump simulation model to compare R-22 and R-410A. In this study, they also performed heat exchanger optimization to exploit the favorable thermophysical properties of R-410A. The authors reviewed experimental heat transfer and pressure drop data for R-22 and R-410A in evaporation and condensation processes. Figure 4 helps to explain the authors' findings. As a reference, they used the R-22 pressure drop and saturation temperature drop at a mass flux of 200 kg/s·m$^2$ (147158 lb/(h·ft$^2$)). For these conditions, R-410A requires a mass flux of 280 kg/s·m$^2$ (206022 lb/(h·ft$^2$)) to match the R-22 pressure drop and a mass flux of 340 kg/s.m$^2$ (250170 lb/(h·ft$^2$)) to match the R-22 drop in saturation temperature. If the R-410A mass flux is selected to match the R-22 drop in saturation temperature, R-410A will have a 55 % higher heat transfer coefficient than R-22.

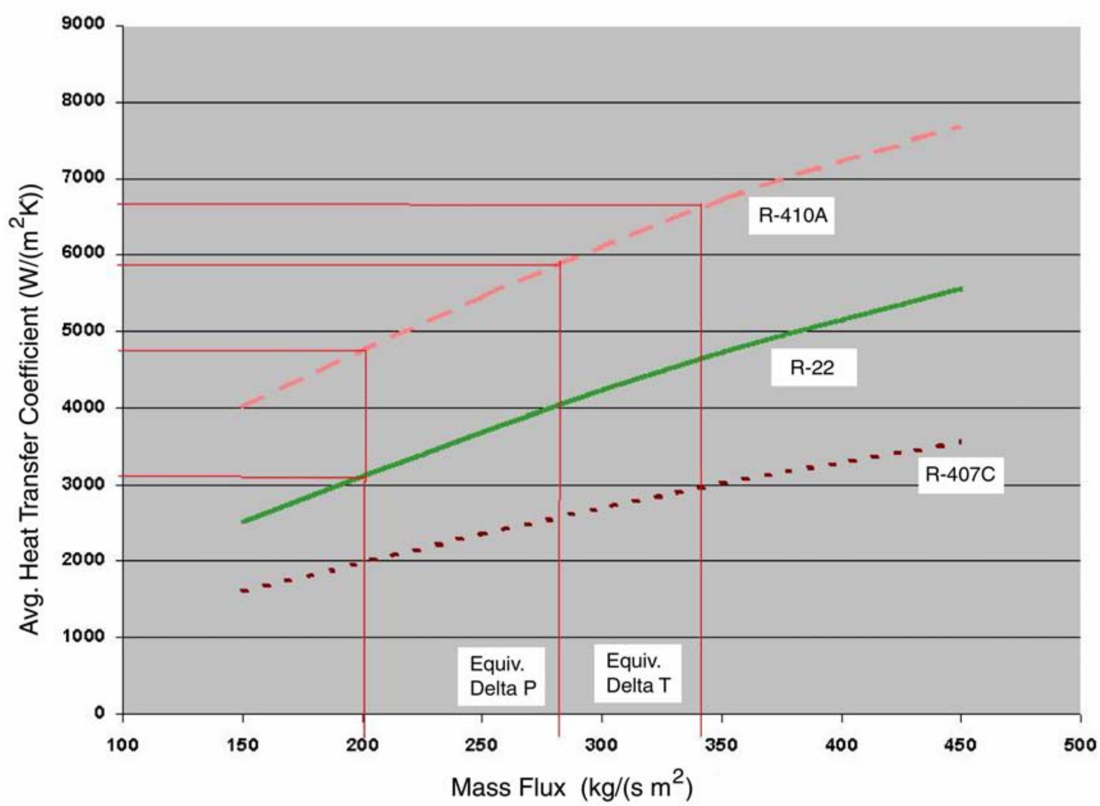


Figure 4. Heat Transfer – Evaporation (Spatz, 2000)

Table 1. Capacity and COP of R-22 and R-410A Systems as Function of Outdoor Temperature (Chin and Spatz, 1999).

| Ambient Air Temp. | | 28 °C (82 °F) | 35 °C (95 °F) | 46 °C (115 °F) | 52 °C (125 °F) | 57 °C (135 °F) |
|---|---|---|---|---|---|---|
| Capacity (kW) | R-22 | 12.84 | 11.98 | 10.63 | 9.95 | 9.32 |
| | R-410A | 13.01 | 11.92 | 10.20 | 9.32 | 8.50 |
| | Rel* (%) | 1.3% | -0.5% | -4.0% | -6.3% | -8.8% |
| COP | R-22 | 3.79 | 3.11 | 2.26 | 1.92 | 1.64 |
| | R-410A | 3.99 | 3.19 | 2.19 | 1.79 | 1.47 |
| | Rel* (%) | 5.3% | 2.4% | -3.4% | -6.9% | -10.7% |

*Rel = 100% ([R-410A value] – [R-22 value])/[R-22 value]

After the evaporator and condenser were optimized, Chin and Spatz performed system simulations for R-22 and R-410A. Table 1 shows their capacity and COP results. The authors concluded that the superior performance of the R-410A compressor and optimized heat exchangers compensated for the lower thermodynamic efficiency of R-410A relative to R-22 at low and moderate condensing temperatures. However, the R-410A optimized-system experienced a loss in COP relative to the R-22 system at condensing temperatures exceeding 47 °C (116.6 °F).

Meurer et al. (1999) compared the performances of R-22 and R-410A working at elevated condensing temperatures up to 60 °C (140 °F) in a breadboard apparatus. The components of the system were an open reciprocating compressor, a water-cooled condenser, a methanol-heated evaporator, a thermostatic expansion valve, and a liquid-line accumulator. The authors reported the R-410A compressor having higher isentropic (+14 %) and volumetric (+22 %) efficiencies than R-22. For a typical evaporation temperature of 9 °C (48.2 °F), the COP of R-410A was higher by 16 % at a condensing temperature of 27 °C (80.6 °F), but it was lower by 1 % at a 57 °C (134.6 °F) condensing temperature. The authors stated that a lower compressor speed accounted for part of the benefits measured with R-410A, but the use of equal rotational speed would negatively affect the R-410A compressor and system performance.

Wells et al. (1999) compared the performance of R-410A and R-22 in split and window-type air conditioners. Their study included theoretical simulations, laboratory testing of split systems, laboratory testing of window units with several hardware modifications, and simulations using the ORNL heat pump model. Figures 5 and 6 show the capacity and EER trends obtained from the R-22 and R-410A split system tests. At an ambient temperature of 51.6 °C (125 °F), the capacity and EER ratios of R-410A fell 12 % below that of R-22. Similar results (within the data scatter) were obtained for the window units. Increased subcooling benefited performance at high ambient temperatures. The study also concluded that using a TXV versus a short tube restrictor or capillary tube results in less performance loss.

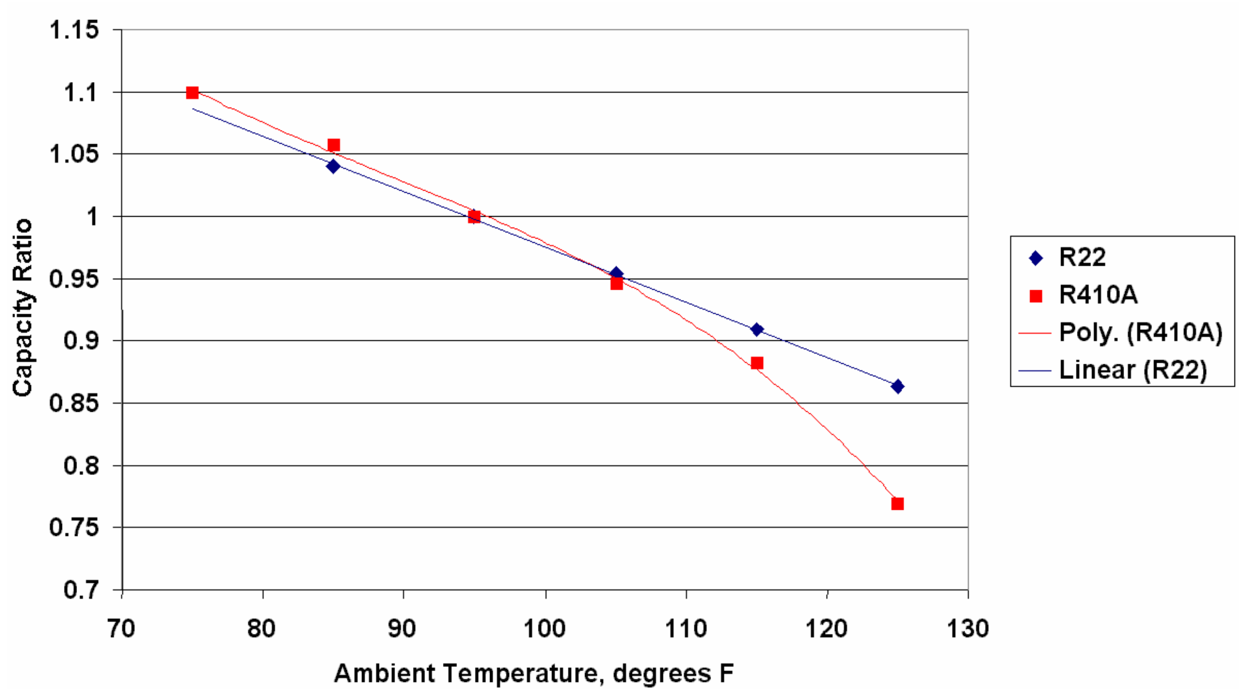


Figure 5. Comparison of capacity loss versus ambient temperature, split system A/C, 12-13 SEER (Wells et al., 1999).

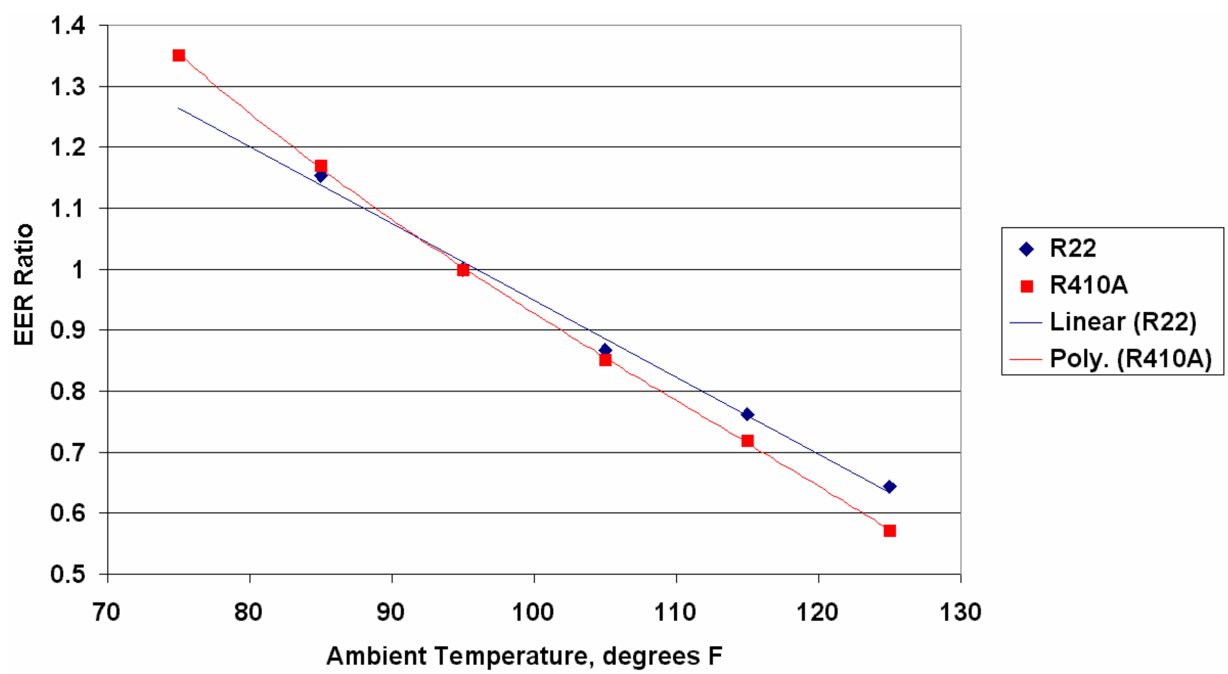


Figure 6. Comparison of EER loss versus ambient temperature, split system A/C, 12-13 SEER (Wells et al., 1999).

Bullock (2000) investigated the performance of HVAC systems working with two low-critical temperature refrigerants: R-404A and R-410A. The study included theoretical analysis of the refrigerant properties, simulations of the basic thermodynamic cycle, and simulations of three split systems: two using R-410A and one using R-404A. The main difference between the systems studied was the condenser and blower size. In Bullock's A/C simulation model, the compressor, expansion device, and condenser/gas cooler models were modified to accommodate transcritical system operation.

Figure 7 presents simulation results for one of the systems studied by Bullock. The vertical arrow indicates the outdoor temperature at which the outdoor coil pressure exceeded that of the critical point. The simulations show that the capacity degradation and compressor power increase become more significant with an increase of outdoor temperature when the condenser pressure is above the critical point. Based on simulation results from the three systems, Bullock offered the following key conclusions: a typical unitary system will cross over to transcritical operation at about 57 °C to 60 °C (135 °F to 140 °F). At the ambient temperature when the critical point is reached, the cooling capacity will be about 60 % to 70 % of the capacity at the 35 °C (95 °F) rating point, and the compressor power will be about 110 % to 160 % of the power at the 35 °C (95 °F) rating point (depends greatly on the compressor type). The system performance at high ambient temperatures can be improved by providing a high capacity condensing unit.

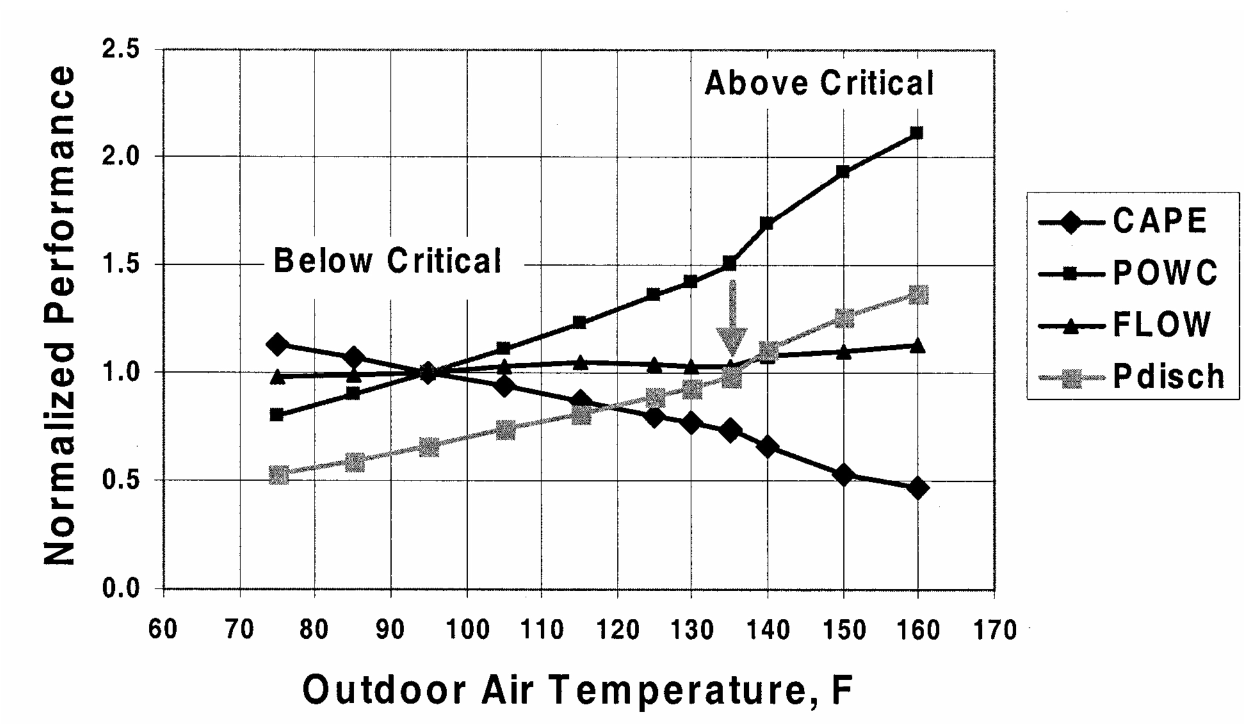


Figure 7. Performance map for R-410A unit with a high performance NTU(0.9) and a low condenser cfm/tom (640) . CAPE = capacity of evaporator; POWC = power of compressor; FLOW = refrigerant flow rate; Pdisch = compressor discharge pressure (all normalized to their values at 35 °C (95 °F), except for the compressor discharge pressure, which is related to the critical pressure); (Bullock, 1999).

Yana Motta and Domanski (2000) performed a simulation study to evaluate capacity and COP of an air conditioner working with R-22, R-134a, R-290, R-410A, and R-407C. Figures 8 and 9 present two-phase domes of the studied refrigerants with the horizontal axes using non-dimensional entropy, $s^*$, and enthalpy, $h^*$, respectively. (where $s^*=(s-s^0_l)/(s^0_v-s^0_l)$, $h^*=(h-h^0_l)/(h^0_v=h^0_l)$, s, h = entropy and enthalpy, $s^0_v$, $h^0_v$ = entropy and enthalpy of saturated vapor at 0 °C (32 °F), and $s^0_l$, $h^0_l$ = entropy and enthalpy of saturated liquid at 0 °C (32 °F)). These figures are suitable for qualitative analyses of the impact of the shape of the two-phase dome on the COP because the width of the two-phase

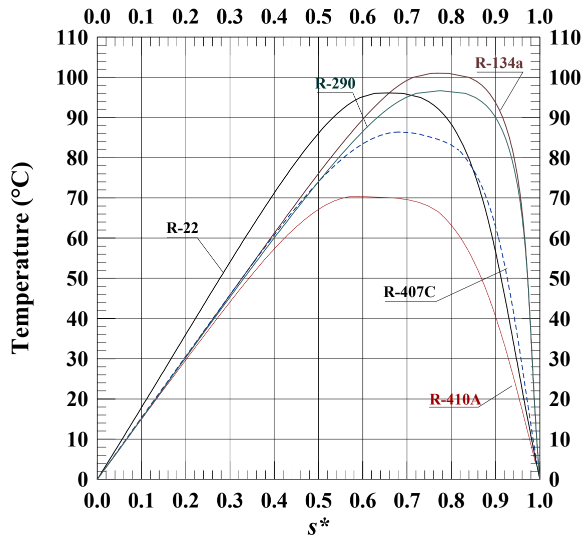


Figure 8. Temperature-dimensionless entropy diagram (Yana Motta and Domanski, 2000).

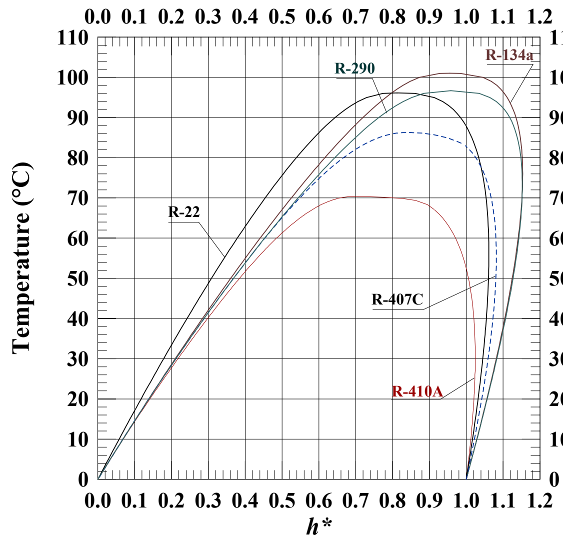


Figure 9. Temperature-dimensionless enthalpy diagram (Yana Motta and Domanski, 2000).

dome is normalized. If we envision vapor-compression cycles with their corresponding Carnot cycles drawn for each refrigerant with the same condensing and evaporating temperatures, we can conclude that the superheated-vapor horn irreversibilities (Figure 8) and throttling-induced capacity losses (Figure 9) will be greater for R-410A than for R-22 due to R-410A's smaller dome.

Yana Motta and Domanski simulated performance of different refrigerants using the UA version of NIST's semi-theoretical vapor-compression model CYCLE-11 (Domanski and McLinden, 1992). All system components were the same for the five fluids, except the compressor for which the swept volume was adjusted to obtain the same capacity at the 35 °C (95 °F) rating point for each fluid. A reference scheme was used to account for different transport properties and their impact on heat transfer coefficient and pressure drop for the different refrigerants.

Figure 10 shows changes of COP for each refrigerant for different outdoor temperatures. The COP values are normalized by the COP at 35 °C for each fluid. R-410A has the highest degradation in COP and R-134a has the lowest. The lines representing performance of R-410A (the lowest-critical-temperature fluid) and R-134a (the highest-critical-temperature fluid) bracket the performance of the remaining refrigerants. The change of COP for R-22, R-290, and R-407C is very similar, because their critical temperatures are within 10 °C of each other.

Figure 11 presents the COP of the four alternatives normalized by the COP of the R-22 system. R-134a, the fluid with the highest critical temperature, improves its performance in relation to R-22. On the other hand, the COP of R-410A drops dramatically at increasing outdoor temperature. Regarding the fluids with critical temperatures similar to R-22 (R-407C and R-290), the small COP differences are caused by the different shapes of the two-phase domes of these fluids rather than their different critical temperatures.

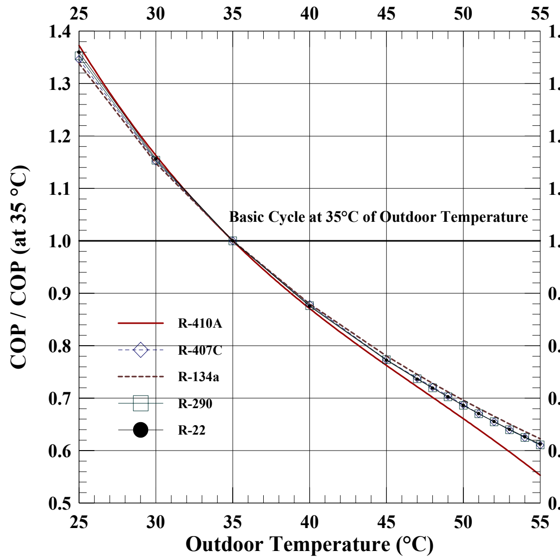


Figure 10. COP referenced to COP at 35 °C (95 °F) (Yana Motta and Domanski, 2000).

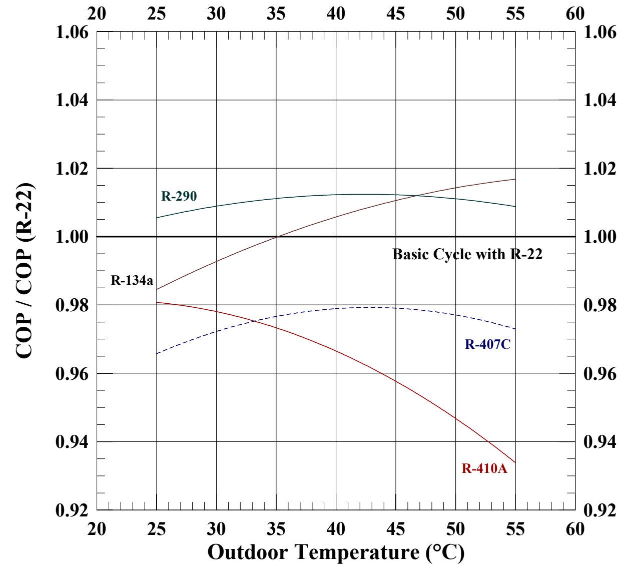


Figure 11. COP referenced to COP of R-22 system (Yana Motta and Domanski, 2000).

Yana Motta and Domanski also evaluated the impact of using liquid-line/suction-line heat exchanger (llsl-hx). As Figure 12 shows, the use of llsl-hx provided COP improvement for all fluids. Refrigerants having high molar capacity benefited more with the llsl-hx application. The benefit of llsl-hx for R-410A increased slightly at high ambient temperatures due to a change in the slope of the saturated liquid line while approaching the critical point; however, the overall impact of approaching the critical point was not significant. At an outdoor temperature of 55 °C (131 °F), the COP increase due to the llsl-hx was 1.9 % higher for R-410A than for R-22.

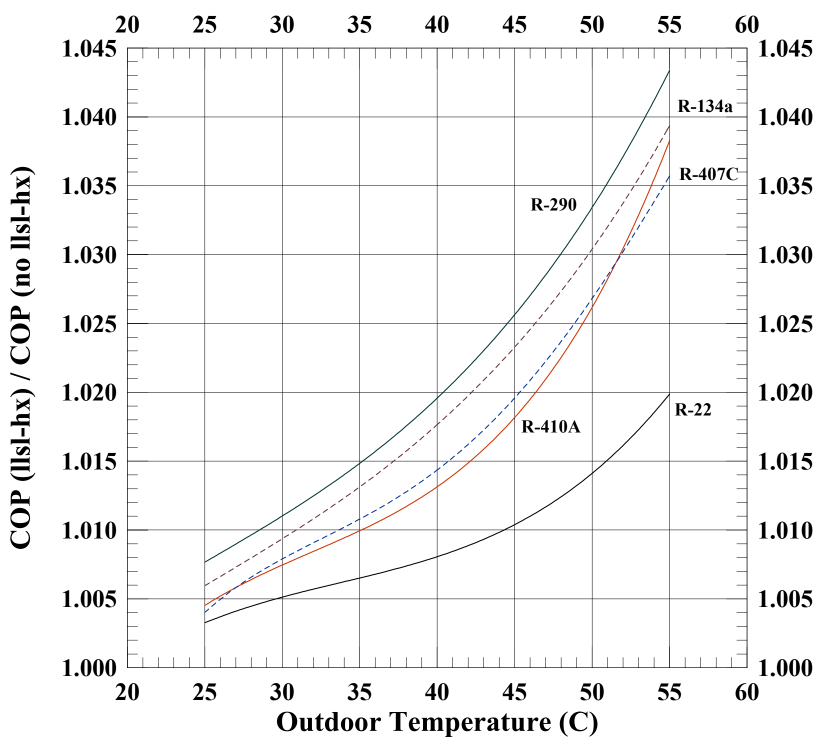


Figure 12. COP for llsl-hx cycle referenced to COP for basic cycle (Yana Motta and Domanski, 2000).

## Concluding Remarks

1. Operation of a vapor compression system at elevated ambient temperatures inherently results in a lower coefficient of performance (COP). For refrigerants operating in the vapor compression cycle, the COP degradation is greater than that for the Carnot cycle and varies between fluids. The refrigerant-related factors that most influence the degradation are the critical temperature and the shape of the two-phase dome.
2. Degradation of capacity and COP at high outdoor temperatures can vary significantly between systems. The system design (size of the condenser, refrigerant charge, refrigerant expansion device) influences performance degradation.
3. All experimental and simulation studies reported a loss of performance for R-410A systems at elevated ambient temperatures by approximately -10 % as compared to R-22.
4. Simulation results indicate that the use of llsl-hx provides slightly better improvement of COP for R-410A than for R-22. At an outdoor temperature of 55 °C (131 °F), the COP increase for R-410A was 1.9 % higher than for R-22.
5. The thermodynamic loop of a typical unitary R-410A A/C will cross above the critical point at an outdoor temperature of approximately 57.2 °C (135 °F).

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## Acknowledgements

The ARTI 21-CR Project Monitoring Committee for this study consists of Donald Bivens, Richard Ernst, Siva Gopalnarayanan, Lawrence R. Grzyl, and Mark Spatz with Project Manager Steven Szymurski. Partial support for this project is provided by the US Department of Energy (project no. DE-AI01-97EE23775) under Manager Esher Kweller. The authors thank C. Bullock, M. Spatz, and W. Wells for providing figures from their presentations to be used in this report. V. Payne and S. Brown provided comments of the draft manuscript.