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PERFORMANCE OF TRANSCRITICAL AND SUBCRITICAL R410A RESIDENTIAL A/C AND H/P SYSTEM

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ABSTRACT

A well-instrumented R410A residential a/c split system was operated over subcritical and transcritical conditions. The experimental results showed no advantage going to transcritical mode when the ambient temperature is much lower than the critical temperature of the refrigerant. System operated well in both modes.

INTRODUCTION

R410A, a 50/50 mixture of R32 and R125 in mass, has critical temperature and pressure of 70.2°C, and 4770 kPa (Refprop, NIST, 1998), or 70.22°C, and 4852 kPa (ASHRAE, 1997). The critical temperature of R410A is lower than other commonly used refrigerants such as R134a and R22. At high ambient temperatures, the condensing temperature of this refrigerant may be close to critical, and the cycle may operate in transcritical mode. The question is, it is beneficial to operate in transcritical mode or not?

Recent experience with carbon dioxide (R744) has revealed distinct operating advantages associated with the transcritical cycle. In normal air-conditioning operations, the COP has a maximum at a particular high-side pressure, so it is possible to trade efficiency for capacity when high pulldown rates are desired (Pettersen, 1994; McEnaney et al., 1998). This has led to exploration of novel cycle configurations and control schemes for R744 systems, some of which are described by Beaver, et al. 1999.

For the case of R410A, on the other hand, such benefits have not been claimed or reported. In fact, most experimental and theoretical investigations of the R744 cycle have not addressed the more general issue of whether such benefits are a property of transcritical cycle itself, or attributable to more complex relationships between the actual and critical temperatures and pressures.

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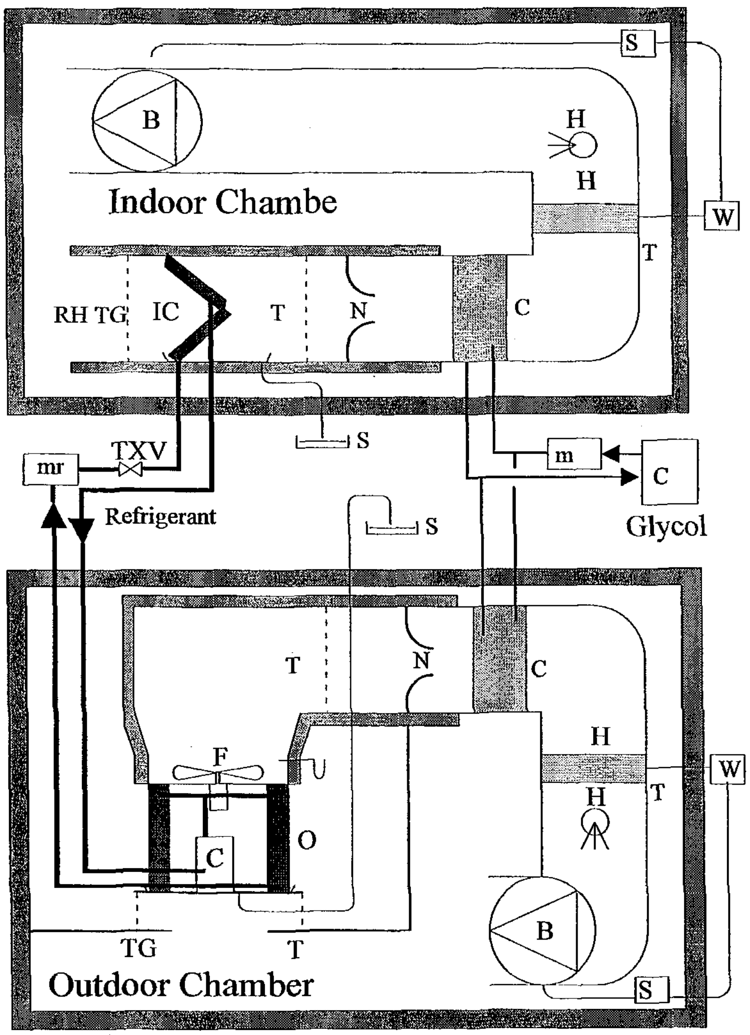
EXPERIMENTAL FACILITY

The test facility consists of two well-insulated environmental chambers that can maintain simulated outdoor and indoor temperature within ± 0.5°C and absolute humidity ±2%. A variable speed wind tunnel in each chamber simulates the range of operating conditions encountered in real applications. A schematic of the test facility is shown in Figure 1. Three independent methods for determining capacity are utilized in both the indoor and outdoor chambers: environmental chamber calorimeter balance, airside energy balance, and refrigerant side energy balance. Three independent methods are used instead of the two, as required by all applicable standards, in order to increase our ability to accurately determine system capacities even in some case when one method can not be used. In some conditions, systems with constant



B – Blower, **C** – Compressor, **CC** – Cooling Coil, **CH** – Glycol Chiller, **F** – Fan, **H** – Heater, **Hu** – Humidifier, **IC** – Residential Indoor Coil, **mg** – Glycol Mass Flow Meter, **mr** – Refrigerant Mass Flow Meter, **N** – Nozzle, **OC** – Residential Outdoor Coil, **RH** – Relative Humidity Probe, **Sc** – Condensate Scale, **Sp** – Speed Controller, **TC** – Temperature Controller, **TG** – Thermocouple Grid, **TXV**-Thermal Expansion Valve **W** – Watt Transducer

Figure 1 Schematic of the experimental test facility for the R410A A/C system area expansion devices have refrigerant liquid carry over at the evaporator exit. This makes the refrigerant side energy balance useless without special procedures. Additionally, in some test runs, the temperature profile at the exit of the heat exchanger is non-uniform which make the air side energy balance difficult to calculate accurately. Having three methods to determine capacity ensures that even when one method is less reliable, we still have two to compare to each other. Three independent procedures also helped to improve our ability to troubleshoot early runs.

Coriolis mass flow meters, together with immersion thermocouples and pressure transducers upstream and downstream of every component allow for the calculation of the refrigerant side cooling capacity. Due to the accuracy with which the electric inputs to each chamber are measured as well as the ability to accurately measure transmission losses, we believe that chamber calorimetry is the most accurate method to determine capacity. The walls of each chamber are constructed of 30cm thick polyurethane. Five thermocouples on both sides of each wall, floor, and ceiling of each environmental chamber provide temperature differences across the environmental chamber walls. The transmission losses are a function of that temperature difference and are both very small and accurately measured. All dry energy inputs (electric) are measured with watt transducers within $\pm$ 0.2%. This testing apparatus was designed to meet or exceed ANSI/ASHRAE 37-1988 and 116-1995 standards.

The data reduction and calculation procedures for these experiments were set up to run without human intervention. During the test run, the most important parameters were monitored in real time graphically and numerically. This allowed the operator to monitor the system's approach to steady state operation. Once this was achieved, data recording was started and the raw data was transferred from the data acquisition program to an Excel file, which then produced a set of averaged data points over approximately a ten-minute test interval. This set of averaged data was then read into an EES program, which proceeded to calculate all important parameters for the cycle calling upon its own internal thermodynamic equations for R410A properties (Klein and Alvarado, 1995).

SYSTEM DESCRIPTION

The R410A system used for these experiments is a Carrier R410A a/c and h/p system with an indoor coil of model FX4ANF042 and an outdoor unit model 38YXA036300. It uses a scroll compressor, with a TXV for air conditioning and an orifice for heat pump operation. The outdoor unit was mounted directly onto our outdoor chamber air duct. The speed-controlled blowers provided zero static pressure difference between the ambient and the discharge of the outdoor fan, while the airflow rate of the outdoor coil was controlled. The indoor heat exchanger is a "A" shaped coil. This coil was placed inside the indoor chamber air duct as shown in Figure 1. The airflow rate of indoor coil was controlled directly by the indoor blower speed controller. Airflow rates for both chambers are measured by using the nozzles according to the ASHRAE standard.

Table 1 gives the specifications of the tested unit.

Table 1 Summary of R410A a/c system components.

System	Type	Commercial
	Nominal capacity at 27/35°C, 50%RH	10.5 kW
Compressor:		Hermetic, scroll
Expansion device		TXV and orifice tube: i.d. = 1.78 mm
Outdoor coil	Description	Two row, three circuits, fin pitch 1.3 mm (20 fpi), louvered wavy fins
	Face area	1.16 m^2
	Core depth	0.038 m
	Core volume	0.043 m^3
	Airside area	66.9 m^2
	Ref. side area	2.38 m^2
	Material	Aluminum plate fins Cu tubes, o.d. = 9.8 mm
Indoor coil	Description	Plate fins, three rows, six circuits, fin pitch 1.7 mm (15 fpi)
	Face area	0.42 m^2
	Core depth	0.057 m
	Core volume	0.024 m^3
	Air side area	27.5 m^2
	Ref. side area	1.31 m^2
	Material	Al. wavy plate fins, Cu tubes, o.d. = 9.5 mm

TEST RESULTS

The tests were conducted at a relatively high ambient temperature (50°C), the highest obtainable in our environmental chamber without damaging the blower motor and variable-speed drive. In order to operate the system in transcritical region, the system was over-charged in regular increments, and air- and refrigerant-side data were recorded for each charge. Overcharging in this manner is essentially equivalent to reducing the condenser size by filling much of it with subcooled liquid. This had the effect of forcing the compressor discharge pressure above the critical point.

Table 2 shows the results for four different discharge pressures from subcritical to supercritical. In order to operate the system in transcritical mode, a cooling coil was wrapped around the compressor shell to protect the compressor. One disadvantage of this approach was the loss of the redundant energy balance measurement for outdoor air side. All the tests were performed in dry mode, with no steam (latent load) added to the indoor chamber. The cooling capacity given in Table 2 is based on the chamber energy balance. Indoor and outdoor fan powers are not included in the calculation of COP.

Table 2 Test results from R410A system as heat rejection moves from subcritical to supercritical region

Test	Teai [°C]	Teao [°C]	mea [kg/s]	Teri [°C]	DTsup [°C]	Pero [kPa]	Tcai [°C]	mca [kg/s]	Tcro [°C]	Pcro [kPa]
A	32.7	19.2	0.67	17.2	2.7	1250	50.8	1.51	54.1	4589
B	32.6	19.1	0.67	17.3	2.3	1258	50.7	1.51	54.2	4767
C	32.6	19.3	0.67	17.4	2.1	1263	50.7	1.51	54.2	4888
D	32.6	19.4	0.67	17.5	2.0	1268	50.6	1.51	54.1	4998

Table 2 (cont)

Test	Tsuc [°C]	Psuc [kPa]	Tdis [°C]	Pdis [kPa]	mr [g/s]	Wc [kW]	Q [kW]	COP
A	17.4	1208	103	4629	69.03	4.60	9.50	2.11
B	17.2	1217	106	4805	69.03	4.85	9.41	2.00
C	17.2	1222	108	4925	68.44	5.06	9.30	1.90
D	17.2	1228	113	5034	67.66	5.28	9.17	1.80

Nomenclature for the Table 2:

Teai - evaporator air inlet temperature; Teao - air outlet temperature; mea - evaporator air flow rate; Teri - evaporator refrigerant inlet temperature; Dtsup - superheat; Pero - refrigerant pressure at evaporator exit; Tcai - condenser air inlet temperature; mca - air flow rate through condenser; Tcro and Pcro - condenser refrigerant exit temperature and pressure; Tsuc and Psuc - suction temperature and pressure; Tdis and Pdis - discharge temperature and pressure; mr - refrigerant flow rate; Wc - compressor power; Q - evaporator cooling capacity and COP - cycle coefficient of performance, Q/Wc.

Figure 2 shows graphically how the performance changes with discharge pressure. Any increase in the discharge pressure will gradually raise the compressor power and decrease the capacity and therefore reduce the COP. This behavior differs from that of R744 at similar operating conditions, where Q increased with discharge pressure, giving rise to a maximum in the COP curve. In the case of R410A the decrease in COP is monotonic, and relatively small. The increase in compressor power can be attributed to compressing the same amount of refrigerant to a higher pressure, while the capacity drop was caused by increase in evaporation temperature as shown in Table 2, and the resulting loss of ΔT.

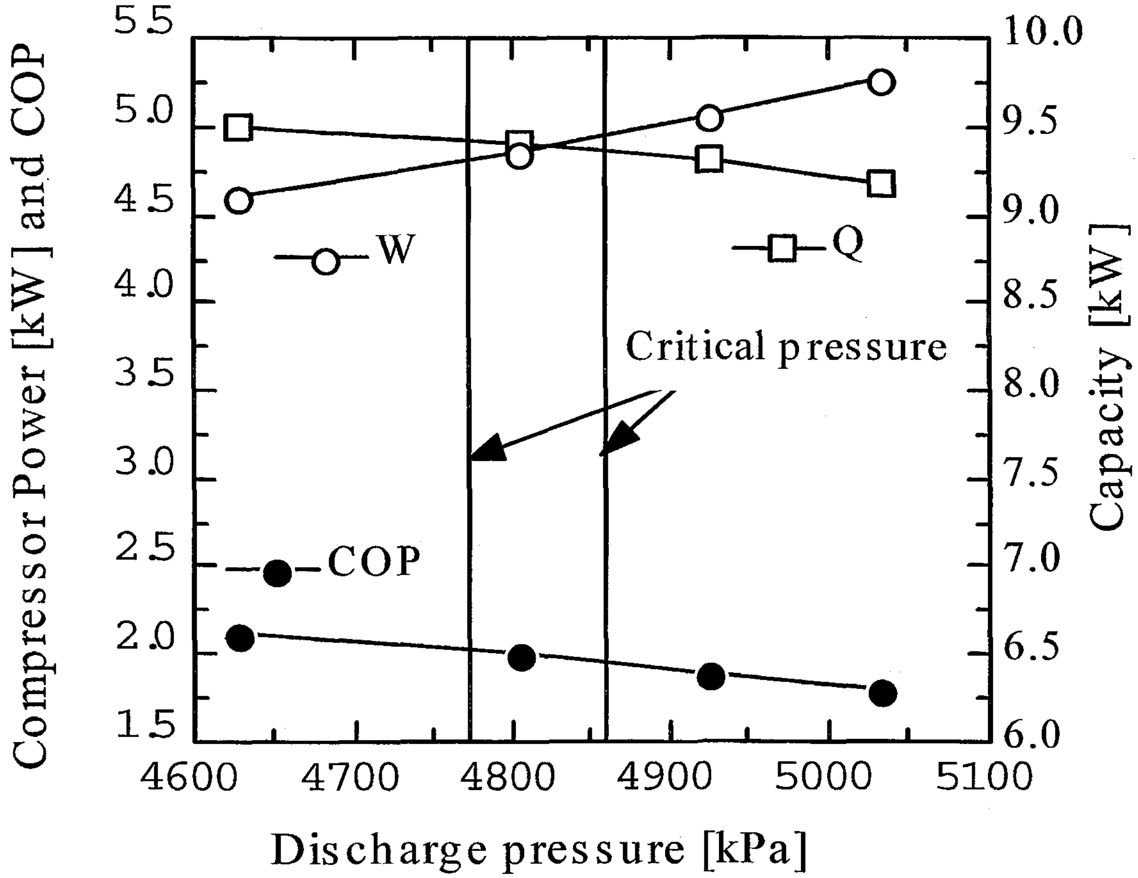


Figure 2 Performance of R410A system in the subcritical and transcritical mode

An obvious question is whether the difference in system behavior is in any way related to the expansion device. Most R744 experiments have been conducted using a manual or backpressure valve, while this R410A system was equipped with a TXV. To explore this question experimentally one would ideally like to raise the discharge pressure farther above critical, and the ambient temperature as well. However given the compressor discharge temperature cutoff (around 113°C) and the high side pressure limit 5000 kPa, we were unable to extend the range upwards. The outdoor ambient was held at 50.6°C for all tests in order to protect the power electronics associated with the variable-speed drives and other equipment in the outdoor chamber.

Fortunately tests at higher pressures are not necessary because most of the remaining questions can be answered by inspecting Figure 3, which displays the thermodynamic cycles of the four test point on temperature-enthalpy diagram. With each increase of discharge pressure, the two-phase region of condenser is reduced, the evaporation temperature is increased slightly, and the capacity reduced.

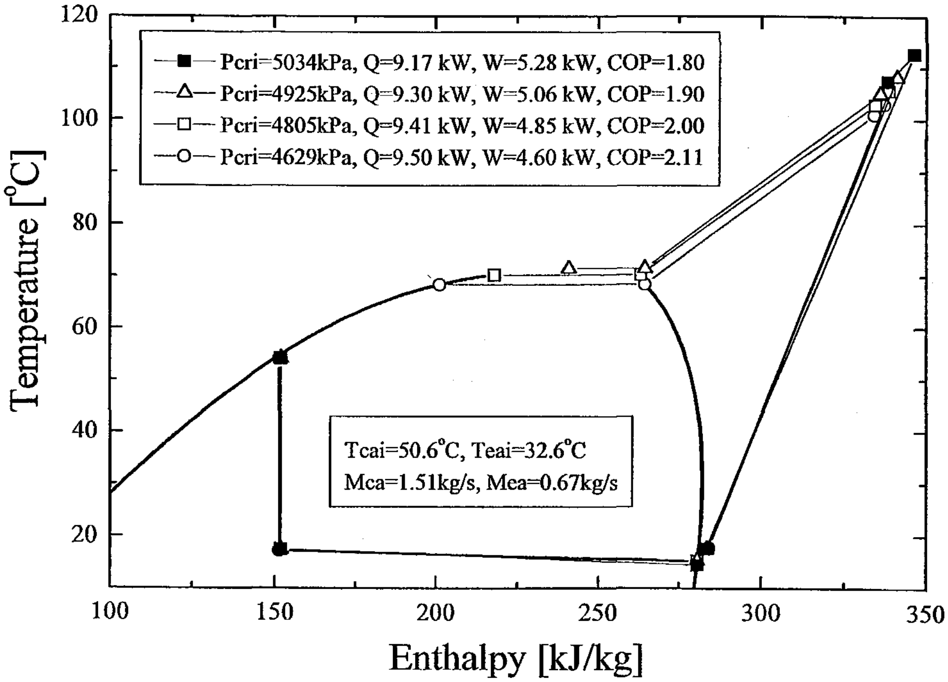


Figure 3 Cycles for four conditions shown in Table 2

At the transition between subcritical and transcritical operation, the thermophysical properties of the refrigerant in heat rejecting coil change sharply. Figure 4 shows how refrigerant side heat transfer coefficient (HTC) and Reynolds number change with temperature. The heat transfer

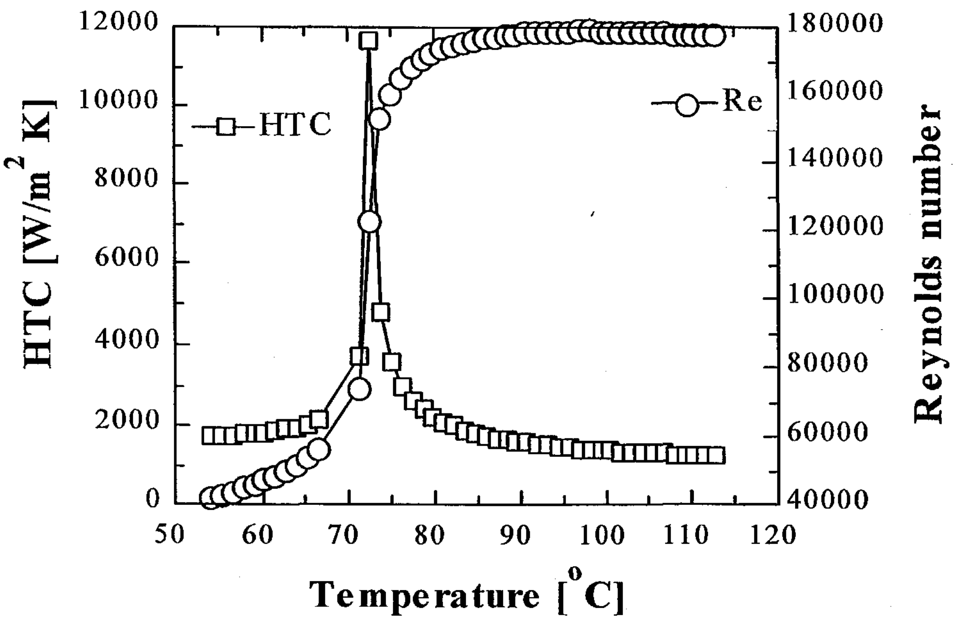


Figure 4 Refrigerant side heat transfer coefficient and Reynolds number

coefficient was calculated from Gnielinski's correlation (1976). From the performance data, the high heat transfer coefficient in the refrigerant side has no apparent effect on the total performance, because the air side resistance was dominant.

CONCLUSIONS

A commercially available R410A heat pump system was operated in both subcritical and transcitical cycles. Detailed air- and refrigerant-side data were recorded. To enable transcritical operation, the high-pressure cutout switch had to be re-set, and additional cooling was provided to the compressor shell.

These experimental results showed that COP was reduced, although not significantly when moving to transcritical operation. There appears to be no advantage obtainable from increasing high side pressure to reject heat in the supercritical region when air inlet temperature to heat rejecting coil is far below critical point.

When the heat rejection is shifted from the subcritical to the supercritical region, the refrigerant side heat transfer coefficient increases due to favorable thermophysical properties, but it can not overcome the dominant heat transfer resistance on the air side.

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