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Advanced Central Cooling System with Dehumidification Mode

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Armin Rudd

Abstract:

The overall goal of the DOE residential research program is to reduce average whole house energy use in new residential buildings by 30-90 percent by 2020, including homes that achieve zero net energy use on an annual basis. High performance space conditioning and control systems that match the high performance of Building America enclosures are necessary to meet performance targets. Conditioning systems with integrated mechanical ventilation and year-around temperature and humidity control are necessary. The most significant climate-specific need is for system-integrated dehumidification for humidity control without overcooling the space. Cost-effective dehumidification without overcooling will enable continued and further reduction of sensible loads (including high-performance glazing) that would otherwise exacerbate humidity control problems in humid climates.

SYSTEMS ENGINEERING APPROACH TO DEVELOPMENT
OF ADVANCED RESIDENTIAL BUILDINGS

8.C.1 Final Report: Advanced Central Cooling System with Dehumidification Mode

RE: TASK ORDER NO. **KAAX-3-32443-08**
UNDER
TASK ORDERING AGREEMENT NO. **KAAX-3-32443-00**

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Table of Contents

List of Figures

List of Tables

8.C.1 Final Report: Advanced Central Cooling system with Dehumidification Mode

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21 December 2005

INTRODUCTION:

The overall goal of the DOE residential research program is to reduce average whole house energy use in new residential buildings by 30-90 percent by 2020, including homes that achieve zero net energy use on an annual basis. High performance space conditioning and control systems that match the high performance of Building America enclosures are necessary to meet performance targets. Conditioning systems with integrated mechanical ventilation and year-around temperature and humidity control are necessary. The most significant climate-specific need is for system-integrated dehumidification for humidity control without overcooling the space. Cost-effective dehumidification without overcooling will enable continued and further reduction of sensible loads (including high-performance glazing) that would otherwise exacerbate humidity control problems in humid climates.

BACKGROUND:

Standard residential cooling equipment responds primarily to a sensible load as registered by a thermostat to activate a forced air cooling system by transferring interior heat to a direct expansion refrigerant evaporator and rejecting that heat to outdoors via an air cooled refrigerant condensing unit. Some more recent thermostats employ a relative humidity sensor that allows the thermostat to respond to a relative humidity set point as well as a temperature set point. Pertaining to cooling, a switch at the thermostat generally opens on humidity rise, which signal generally results in at least one of: a) the control of lower air flow across the evaporator; and, b) a three degree set point depression. Lower air flow across the evaporator increases the latent cooling capacity by lowering the evaporator temperature and increasing the contact time of air on the coil surfaces. The set point depression forces the system to operate more. While these cooling system enhancements improve indoor humidity control, they do not achieve the ultimate goal of year-around humidity control without over-cooling.

DESIGN APPROACH:

Modification of a standard cooling system to include an additional refrigerant condensing coil in the indoor air stream can yield a dual-mode system providing: a) standard cooling with incidental dehumidification; and, b) dehumidification only.

The most significant system concept included adding a condensing/reheat coil in the process air stream. The control strategy would be such that when a temperature set point was satisfied, but a humidity set point was not yet satisfied, a dedicated dehumidification mode would be activated.

The original concept was such that, when dehumidification mode was activated, while the compressor operated, automatic valves would divert refrigerant away from the outdoor condensing coil to the indoor condensing coil, delivering dry air at room temperature or warmer, until the humidity set point was achieved or until the space temperature rose sufficiently to initiate a reversal to normal cooling mode. This system is generally shown in the schematic of Figure 1. The system concept also incorporated:

a) an electronically commutated motor (ECM) blower for efficient and variable air moving capacity; and
b) a thermal expansion valve (TXV) for efficient and adaptive refrigerant metering.

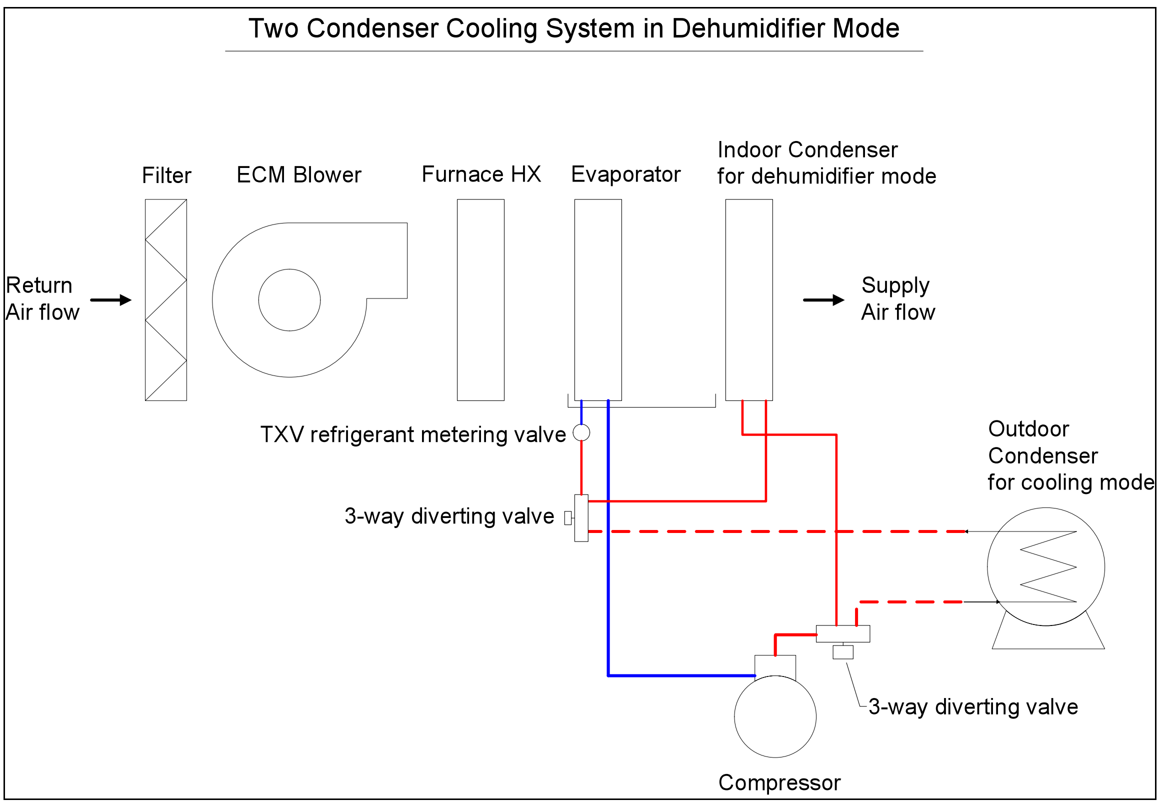

Figure 1: Initial basic concept schematic

A design kickoff meeting was held with a refrigeration consultant and a manufacturing partner. The outcome of the meeting identified more specific design elements, components, and criteria that would frame the prototyping project. The resultant schematic with some notes is shown in Figure 2.

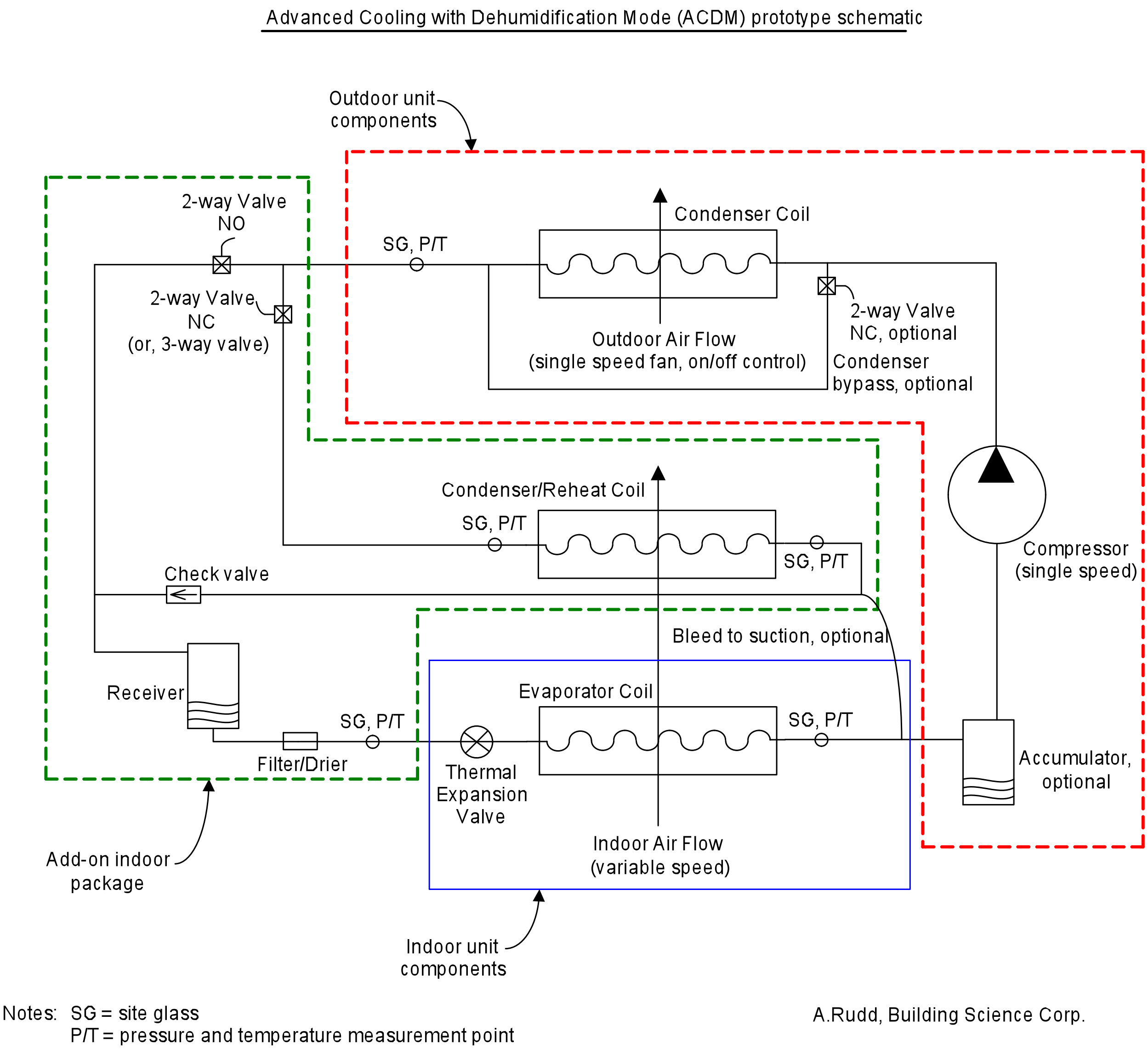

Figure 2: Initial design schematic with notes from the kickoff meeting

As shown in Figure 2, there were multiple possible paths identified that needed to be refined down to one choice for constructing the prototype. The overriding factor in making design decisions was in favor of keeping cost down. It was thought that the equivalent competitor to beat was a separate dehumidifier integrated with the central air handling system, costing in the range of $1,500 to $2,000 installed. Because of the cost factor, it was highly desirable to not have to make any modifications to a standard, off-the-shelf outdoor compressor/condenser unit. It was therefore decided that, rather than cut into the outdoor unit piping to install valves for diverting refrigerant around the outdoor condenser, hot gas from the compressor discharge would be forced through the outdoor condenser with the condenser fan motor shut off to significantly reduce refrigerant cooling and condensing in the outdoor unit. It was thought that, if the compressor did not suffer with too high a pressure drop by forcing gas through the inactive outdoor condenser, then a major cost would be eliminated, and due to some heat loss in the outdoor condenser the final process supply air temperature could be beneficially lowered while remaining above room temperature.

Another choice that needed to be made was whether or not to use an accumulator to protect the compressor from liquid due to large swings in refrigerant flow, and due to dehumidification operation during outdoor temperatures lower than normal cooling conditions. Some thought that the accumulator was unneeded because of the established design criteria of: a) a TXV which, in theory, should assure superheated vapor going to the compressor; and, b) a scroll compressor which is more tolerant of some liquid. However, it was decided that if refrigerant conditions temporarily got outside of the TXV throttling range, the risk of compromising the compressor might be too great. Normal cooling-only outdoor units do not come with accumulators, and it would be too costly to cut into the piping to install one. Thus, it was decided to select a heat pump outdoor unit which came with an accumulator and low pressure switch to aid in compressor protection and low ambient operation.

Manufacturers of refrigerant valves were consulted to determine the right type of valve for activating and deactivating the reheat coil. This resulted in the selection of a 3-way valve with an internal pilot valve that allowed the reheat coil to be pumped down to the system suction pressure when it was not in use. This kept useful refrigerant from being stranded in the reheat coil while the system operated in normal cooling mode.

As the design phase continued with refinements, it was thought to be desirable to be able to operate the system in three modes: a) normal cooling; b) dehumidification only; and, c) heat pump heating. Since the refrigerant flow direction changes between cooling and heating mode, causing the cooling mode evaporator to become a heating mode condenser, a design was developed that required a 4-way reversing valve to feed and leave the receiver in the proper direction regardless of the system mode of operation. This is shown in the schematics of Figures 3 through 5, which illustrate the as-built prototype configuration.

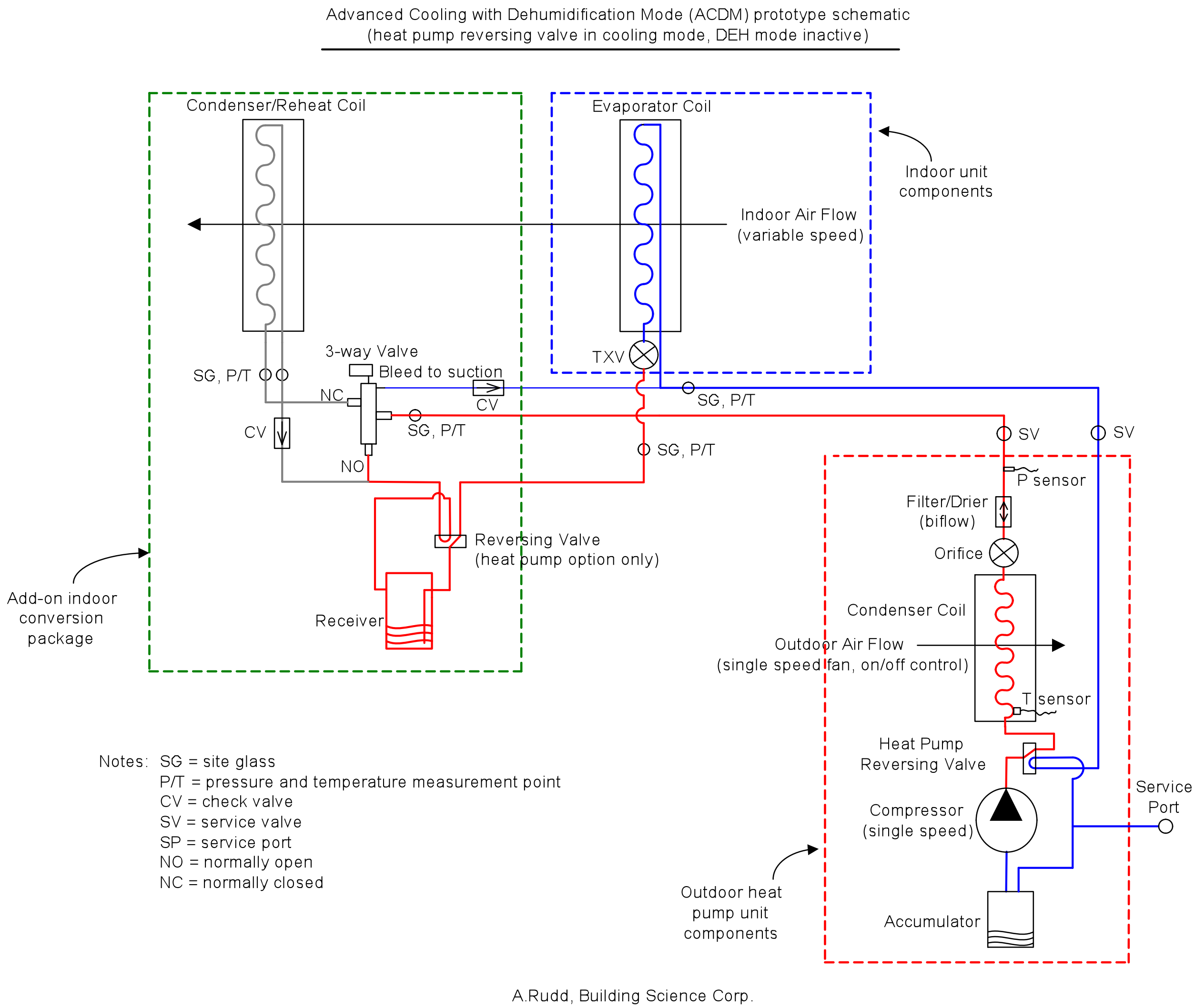

Figure 3: Final design schematic showing cooling operation (greyed-out lines are inactive)

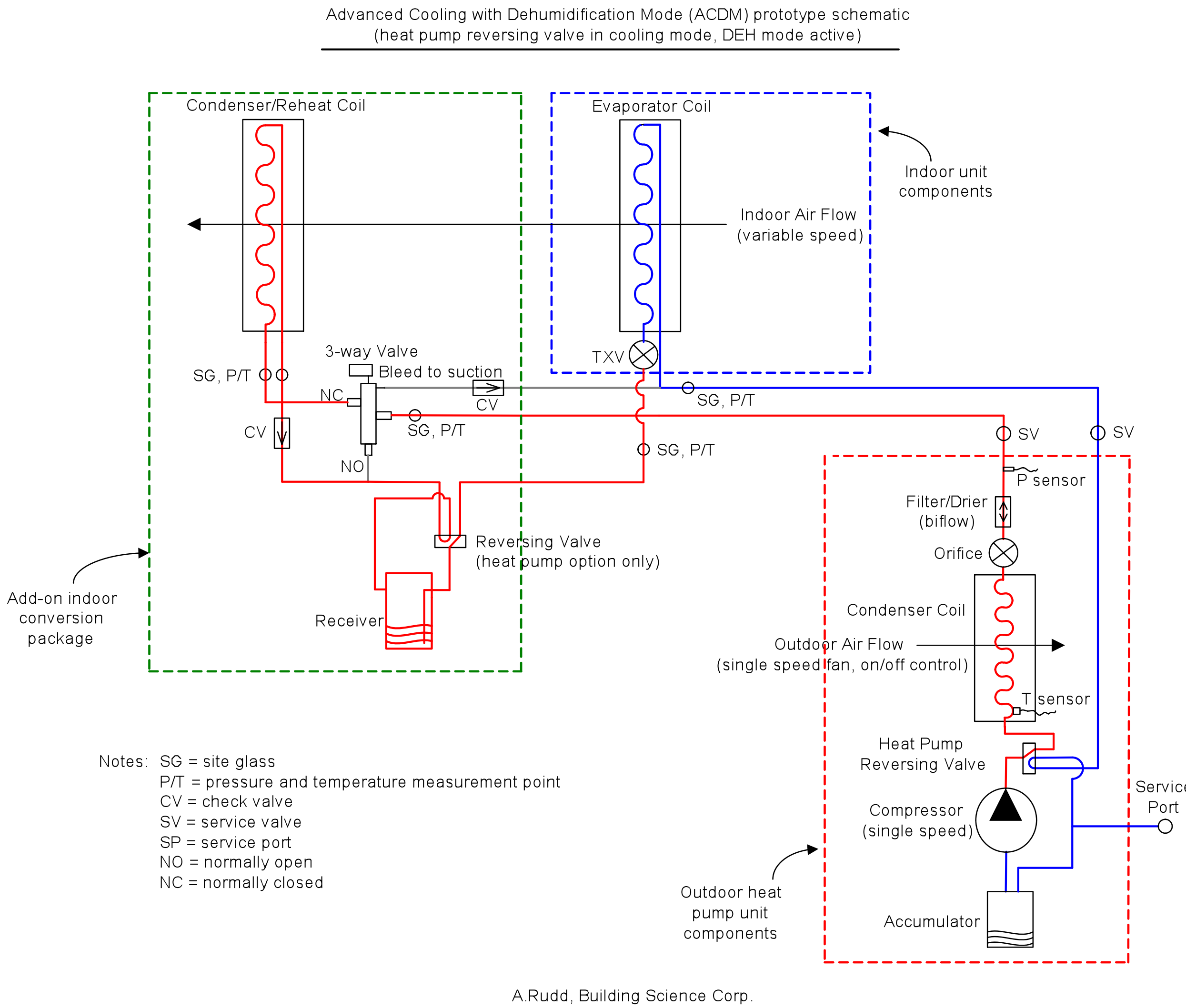

Figure 4: Final design schematic showing dehumidification-only operation (greyed-out lines are inactive)

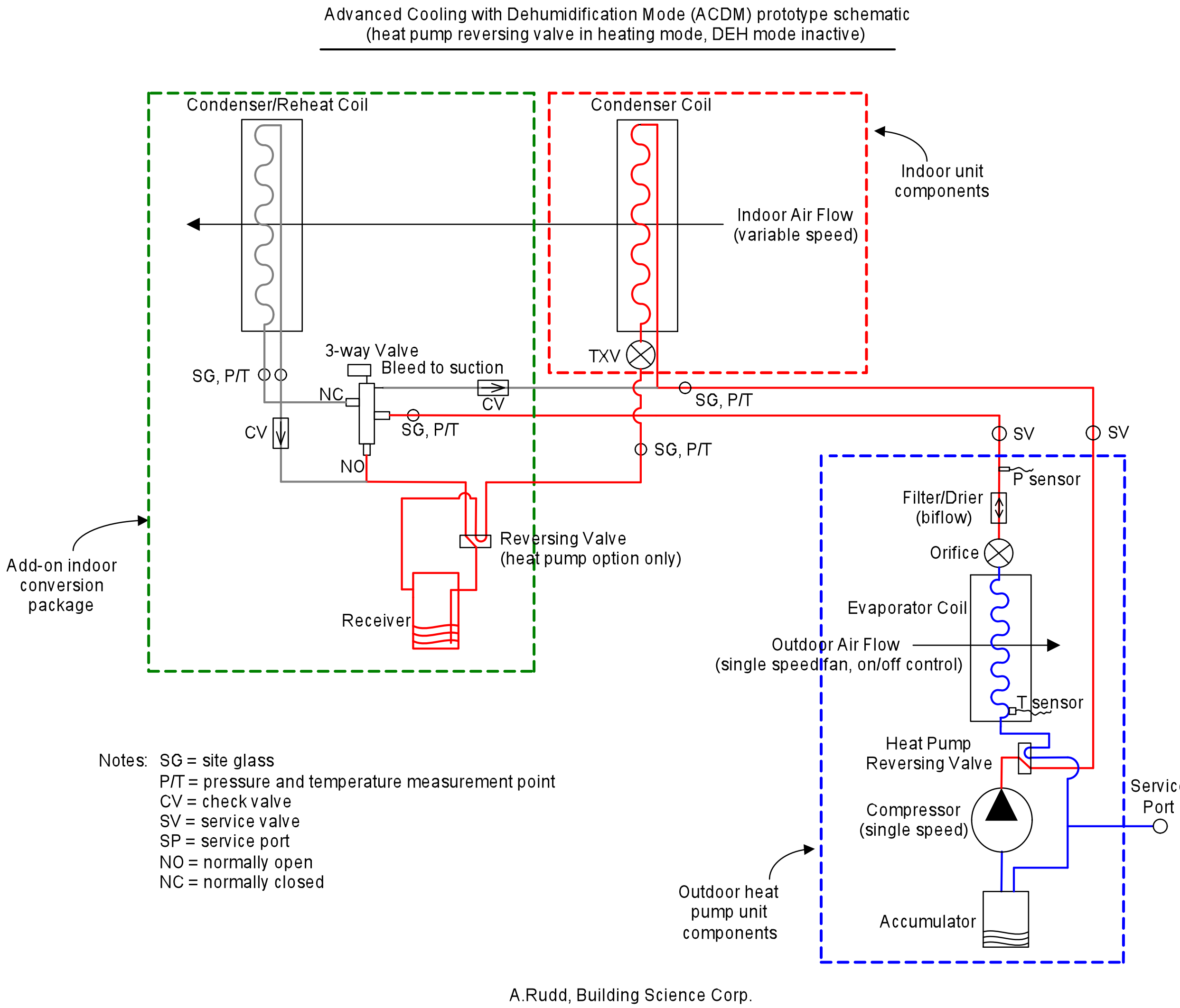

Figure 5: Final design schematic showing heating operation (greyed-out lines are inactive)

Condenser coil manufacturers were consulted to establish the indoor condenser coil design. Based on the manufactures design models, many parameters were parametrically changed to arrive at the desired condensing/reheat capacity within the desired coil cross sectional area and with the least air pressure and refrigerant pressure drop. The final coil specifications are shown in Figure 6.

Construction and Performance Details

Tag	C-3
Coils per bank	1
Total capacity (MBH)	24.1
Air flow (CFM)	800
Face velocity (ft/min)	366
Altitude (ft)	0
Air pressure drop (in w.g.)	0.03
Entering dry bulb (°F)	50.0
Leaving dry bulb (°F)	77.7
Coil type	1/2
Fin height (in)	17.5
Fin length (in)	18.0
Face area (ft²)	2.19
Rows	1
Fin spacing (fins/in)	13
Fin material	Al
Fin type	Cor.
Fin thickness (in)	0.006
Tube wall thickness (in)	0.016
Number of circuits	1
Supply conn. size (in)	0.75
Return conn. size (in)	0.75
Weight (lb)	18
Subcooler circ./face tubes	0 / 0
Subc. capacity (MBH)	0.0
Subc. leaving temp. (°F)	0.0
Subc. pressure drop (psi)	0.00
Suction temp. (°F)	45.0
Condensing temp. (°F)	125.0
Ref. pressure drop (psi)	3.37
Refrigerant	R-22

Figure 6: Condenser/reheat coil specifications

Receiver manufacturers were consulted to determine the correct size receiver in order to assure liquid refrigerant was always fed to the thermal expansion valve, but not to be capable of storing the entire system charge as is often the purpose of a receiver.

PROTOTYPE CONSTRUCTION

A nominal 2.0 ton Goodman heat pump split air conditioning system was purchased as the base, off-the-shelf system. It was rated at 14 SEER with a variable speed air handling unit. The outdoor unit had a scroll compressor, an accumulator, and low pressure controls to aid in low ambient temperature operation. It also came with a factory installed, bi-flow, liquid line drier. Normal installation called for a 3/4 inch vapor line and a 3/8 inch liquid line between the outdoor and indoor units. However, a 5/8 inch line was substituted for the 3/8 inch line because it would carry gas while in dehumidifier mode.

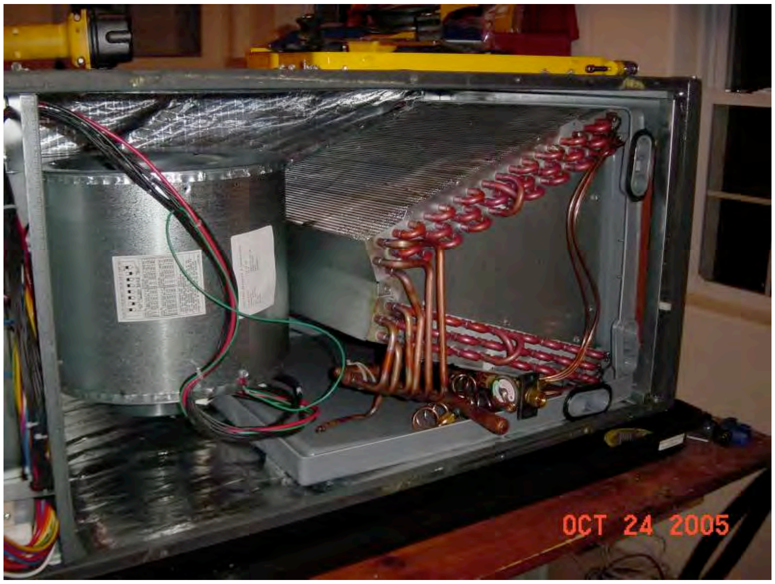

Figure 7: Indoor unit with ECM blower and evaporator with TXV

Figure 8: Outdoor unit with scroll compressor and accumulator being fitted for refrigerant piping

The copper connecting piping was fit to components according to the schematic in Figure 3. All connections were brazed with an 80% copper 5% phosphorus alloy with 15% silver. No flux was used or required, but dry nitrogen gas was flowing to eliminate internal oxidation.

Figure 9: Brazing connections at outdoor unit with nitrogen flowing internally

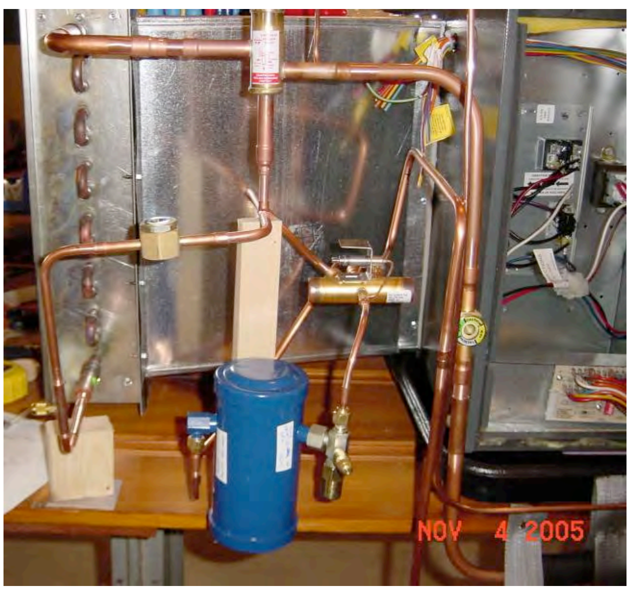

Figure 10: Add-on dehumidifier components fitted (from left to right: cased condenser/reheat coil, 3-way diverting valve with suction bleed, receiver, and reversing valve (required only for heat pump heating operation)

Figure 11: Connections brazed at indoor unit with nitrogen flowing internally; heat sensitive components protected with blaze cloth and wet rags

Figure 12: System pressure tested with dry nitrogen

The system was pressure tested with a 118 psi charge of nitrogen. Passing that test, the system was evacuated to less than 50 microns using a 2-stage vacuum pump and digital vacuum gauge.

Figure 13: System evacuation with 2-stage vacuum pump and digital vacuum gauge

After the vacuum was held for 10 minutes without exceeding 300 microns, R-22 refrigerant was weighed-in to compensate for the larger diameter tubing and the additional length of tubing, as shown in Table 1.

Table 1: Refrigerant charge adjustment for tubing size and length

line size	R-22 liquid line oz/ft	liquid line additional charge needed for 3/8 to 5/8	liquid line additional charge needed for additional 5/8 length	R-22 suction line oz/ft	suction line additional charge needed for additional 3/4 length
1/4	0.22				
3/8	0.58				
1/2	1.14				
5/8	1.86	19.2	9.3	0.04	
3/4				0.06	0.3
7/8				0.08	
1 1/8				0.15	
1 3/8				0.22	
Total additional charge needed:			28.8 oz		

Needing to be protected from heat during the brazing process, the TXV suction line sensing bulb comes temporarily installed from the factory. This was later permanently attached according to the manufacturer's instructions at the 10 o'clock position, and insulated.

Figure 14: TXV bulb permanently strapped and insulated

Figure 15: Compensating refrigerant charge weighed-in using digital scale while system still in a vacuum

The indoor and outdoor units were electrically wired according to the manufacturers instructions. A 24 Vac relay was spliced into the outdoor unit wiring to allow the condenser fan to be turned off during dehumidification mode while the compressor operated.

Figure 16: Indoor and outdoor units electrically wired

Figure 17: 24 Vac relay added to outdoor unit to allow on/off control of condenser fan

Low-voltage controls wiring and selectable setup configurations were completed according to manufactures instructions. Like the reversing valve in the outdoor unit, the reversing valve around the receiver was wired to be energized on cooling and de-energized on heating. This is the more common configuration, since, if the valve fails, it fails to heating mode, which is thought to be the more critical function.

Figure 18: Low-voltage controls wired; thermostat and air handler unit settings configured

An additional switch was installed to manually switch the system into dehumidifier mode. That switch energized the 3-way diverting valve, sending refrigerant to the reheat coil, and de-energized the outdoor condenser fan motor. Switching to dehumidifier mode will be made automatic in future work.

Figure 19: Manual switch to energize dehumidification mode will be made automatic

BENCH-TOP TESTING RESULTS

Performance measurements taken during bench-top testing are listed in Table 2. These measurements were taken by hand-held instruments at this time in order to save time before outdoor weather conditions precluded operation conducive to this testing. More detailed instrumentation and continuous monitoring will be added by Spring of 2006.

All of the indoor and outdoor operating conditions available for the bench-top testing were on the low end of normal for this type of equipment. Standard rating conditions for refrigerant based cooling systems is 95 F outdoors, and 80 F drybulb with 67 F wetbulb indoors which is 57% relative humidity. Outdoor conditions for this bench-top testing ranged from 50 F to 67 F. Indoor conditions ranged from about 60 to 70 F and about 50% relative humidity. The lower indoor conditions could be considered like those encountered in basements or in above grade living spaces in swing or mild winter seasons.

Standard conditions for rating dehumidifiers are 80 F drybulb and 60% relative humidity. While the ACDM system could not be considered a standard dehumidifier, because the compressor and inactive condenser are located outdoors, the moisture removal efficiency was calculated for comparison sake. Moisture removal efficiency is listed as liters of water removed per kilowatt-hour of electrical energy consumed. Energy Star listed dehumidifiers vary depending on their capacity, but are in the range of 2.25 L/kW-h.

The first test result shown in Table 2 is that of standard cooling. While the split cooling system had a manufacturer's SEER rating of 14, under the lower ambient operating conditions for this test, it was shown to have an EER of 17. At those test conditions, the standard cooling system showed a sensible heat ratio of 0.73 and a moisture removal efficiency of 2.06 L/kW-h.

The remaining tests shown in Table 2 are of the dehumidifier mode. There was some concern that forcing the hot gas refrigerant from the compressor through the outdoor condenser coil and factory installed filter/drier may cause too much restriction for vapor flow while in dehumidifier mode. However, compressor current draw and discharge pressure remained in an acceptable range, indicating that the system was not working against excessive head pressure. The RLA (running load amps) for the compressor was rated at 10.9 A, but the measured current draw never exceeded 9 A.

Another concern was the tubing diameter that would carry vapor from the outdoor unit to the indoor condenser coil. In testing, it was found that there was little pressure drop in the 5/8 inch line which carried both liquid and gas in dehumidifier mode, therefore, to reduce refrigerant charge and cost, it could likely be reduced to a 1/2 inch line. The 1/2 inch line set would also match the reheat coil tube size.

While in dehumidifier mode, the outdoor condenser fan was de-energized to allow condensing at the indoor condenser/reheat coil. Turning off the outdoor condenser fan certainly reduced the amount of outdoor condensing, but it did not eliminate it. To try to simulate higher ambient temperatures two of the tests shown in Table 2 list “(blanket)” in the header. In those cases, a thick blanket was laid over the outdoor condenser to see what would happen if less heat was allowed to escape outdoors via natural convection. This raised the supply air temperature to about 6 degrees above the return air temperature, whereas the supply air temperature had been close to the same as the return air temperature without the blanket. In this configuration, the increase in system pressure and compressor power draw reduced the moisture removal efficiency. As shown in Figure 2, a bypass could be installed to avoid the restriction of the inactive outdoor condenser, but the additional cost to retrofit off-the-shelf units would likely be prohibitive compared to available stand-alone dehumidifiers. If the bypass was installed in manufacturing, the additional cost would be relatively small, and would be advantageous. This will be investigated in future bench-top testing.

Moisture removal efficiency during the standard dehumidification mode operation tests ranged from about 1 to 1.5 L/kWh, or between 50% to 75% that of the normal cooling mode test. The lower moisture removal efficiency was expected, but the advantage of not having to overcool the space is considerable. Overcooling the space is often unacceptable for a number of reasons and always reaches a limit which precludes further moisture removal.

The manufacturer’s predicted air pressure drop across the indoor reheat coil was 0.03 inch w.c. or 7.5 Pa at 800 cfm. This proved very realistic in testing, which showed an air pressure drop of 6 Pa. That amount of air flow resistance is small and inconsequential, especially compared to the pressure drop across a wet evaporator coil in the range of 40 to 50 Pa. Based on measurement from cooling systems and dehumidifiers, the initial prediction for refrigerant condensing temperature was 125 F. That turned out to be too high because of heat loss in the inactive outdoor condenser. The actual refrigerant condensing temperature was closer to 100 F which reduced the expected capacity of the reheat coil. As a result of that finding, future modification will result in a reheat coil with smaller tubing and closer fin spacing to increase reheat capacity with an acceptable increase in air pressure drop.

Refrigerant pressure drop as a result of the add-on components was very small, except for the reversing valve which was needed to properly feed the receiver only if heat pump heating mode was desired.

Table 2: Bench-top test measurements

Bench-top testing of ACDM system												
	11/9/2005			11/13/2005		11/16/2005						
	8:10 PM	8:24 PM	8:46 PM	2:00 PM		12:53 PM	12:55 PM	1:08 PM	1:13 PM	1:20 PM	1:48 PM	2:00 PM
					(blanket)						(blanket)	(blanket)
	Cooling mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode	DEH mode
T outdoors	50	50	50	62	62	67	67	66	66	65	65	65
Tdb return air	61	61	61	59	59	68	68	68	68	68	68	70
Tdb after evaporator		41	42	41			50	50	50	49		
Tdb supply air	44	59	59	62	68		74			68	74	76
RH return air	52	49	49	47	46	56	54	52	52	51	49	49
RH after evaporator		84		83			86	87	87	86		
RH supply air	75	44	44	39	31		37			43	38	36
Tdp return air	43	42	42	39	38	52	51	50	48	49	48	50
Tdp after evaporator		37		36			46	45	46	45		
Tdp supply air	37	37	37	37	36		46			45	47	47
Grains return air	42	39	39	35	34	58	56	53	54	53	50	54
Grains after evaporator		32		32			46	47	47	44		
Grains supply air	32	33	33	32	32		46			44	48	49
P high side out of outdoor condenser						200		223		220	275	
P high side into TXV	102	185	187	175	210	190	220	223	220		267	285
P low side out of evaporator	57	62	63	61	63	73	76	72	72		79	80
P low side into accumulator						73		73		72	77	
Tsat, high side out of outdoor condenser						103		109		108	124	
Tsat, high side into TXV	60	96	96	92	105	100	108	109	108		123	127
Tsat, low side out of evaporator	31	35	36	34	36	43	45	42	42		47	47
Tsat, low side into accumulator						43		43		42	46	
Tline, high side out of outdoor condenser				94						105	120.5	
Tline, into reheat coil		90		89						102		
Tline, out of reheat coil		82		86						99		
Tline, high side into TXV	60	86	86	82			103	105	104	105	122	121
Tline, low side out of evaporator	34	36	36	36			46	45	45	44	46	47
Tline, low side into accumulator				43						48	53	
Superheat	3	1	0	2			1	3	3		7	0
Subcooling	0	10	10	10			5	4	4		1	6
Air ΔP across reheat coil (Pa)				6			6					
Refrigerant ΔP across reheat coil (psi)							1		1			
Ref. ΔP between reheat coil and TXV (psi)									7			
Line voltage	241	241	241	239	239		237			237	236	237
Compressor amps	5.2	6.7	6.7	6.7	7.4		7.1			6.8	8.9	7.4
AHU amps	0.60	0.51	0.51	0.48	0.48		0.5			0.51	0.51	0.51
Power factor	0.86	0.86	0.86	0.86	0.86		0.86			0.86	0.86	0.86
Compressor power (W)	1078	1389	1389	1377	1521		1447			1386	1806	1508
AHU power (W)	124	106	106	99	99		102			104	104	104
Total power (W)	1202	1494	1494	1476	1620		1549			1490	1910	1612
Sensible cooling (Btu/h)	14960	1760	1760	-2640	-7920		-5280			0	-5280	-5280
Latent cooling (Btu/h)	5440	3264	3264	1632	1088		5440			4896	1578	2938
Latent cooling (lb/h)	5.4	3.3	3.3	1.6	1.1		5.4			4.9	1.6	2.9
Total cooling	20400	5024	5024							4896		
Sensible Heat Ratio	0.73	0.35	0.35							0.00		
Cooling EER (Btu/h/W)	17.0											
DEH efficiency (L/kwh)	2.06	0.99	0.99	0.50	0.31		1.60			1.49	0.38	0.83

PROBLEMS ENCOUNTERED

Heat Pump Heating Mode

During the later part of the design phase, it was thought to be desirable to be able to operate the system in three modes: a) normal cooling; b) dehumidification only; and, c) heat pump heating. Since the refrigerant flow direction changes between cooling and heating mode, causing the cooling mode evaporator to become a heating mode condenser, a schematic was developed that required a 4-way reversing valve to feed and leave the receiver in the proper direction regardless of the system mode of operation. The reversing valve operation worked as designed to properly feed and leave the receiver, but too much refrigerant remained stranded in the inactive reheat coil in heating mode. The 3-way valve bleed to suction, which recovered refrigerant adequately in cooling mode, did not see adequate pressure drop to accomplish that in heating mode. The system subsequently encountered low system pressure and did not function correctly in heating mode. A good solution to this problem is being investigated. It may require a third line (small bleed line) going between the indoor and outdoor units, along with a 2-way solenoid valve, but this would only be needed if heat pump heating mode was desired.

ORNL Heat Pump Design Model

The original proposal anticipated that ORNL would have made sufficient progress on enhancing their existing computerized heat pump design model to inform the ACDM design process, and for bench-top evaluation of the prototype to inform the design model. However, that did not happen in time, so the prototype was designed, constructed, and bench-top tested without the benefit of an enhanced ORNL heat pump design model.

Field Testing

Due to the amount of work encountered in designing, constructing, and bench-top testing the prototype, and due to time and weather constraints, field testing of the prototype was not accomplished to date.

FUTURE WORK

Bench-top testing and evaluation of the prototype will continue in Spring and Summer of 2006. More detailed instrumentation for monitoring will be added including continuous monitoring of moisture removal and energy consumption at varying indoor and outdoor conditions. Humidification equipment will be added to increase humidity in the indoor lab space.

It is anticipated that field testing may be conducted in the hot-humid climate of Cocoa, Florida in the Spring of 2006. This testing may occur in collaboration with the Florida

Solar Energy Center in their manufactured housing laboratory. This lab building is fitted with automatic equipment to simulate occupancy effects of moisture and heat gain and has existing instrumentation to record the space temperature and humidity conditions in multiple locations. The existing split system cooling equipment will be removed and the ACDM system will be installed and instrumented. Performance will be established under standard cooling and dehumidification-only operation under varying indoor and outdoor environmental conditions, including standard rating conditions. Cooling Energy Efficiency Ratio (EER) will be measured and reported for standard cooling mode, and moisture removal capacity (L/day) and efficiency (L/kWh) will be measured and reported for dehumidification mode.

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About this Report

This report was prepared with the cooperation of the US DOE, Building America Program.

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