The Effect of Reduced Evaporator Air Flow on the Performance of a Residential Central Air Conditioner

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ABSTRACT

This paper discusses the measured degradation in performance of a residential air conditioning system operating under reduced evaporator air flow. Experiments were conducted using a R-22 three-ton split-type cooling system with a short-tube orifice expansion device. Results are presented here for a series of tests in which the evaporator air flow was reduced from 25 to 90% below what is normally recommended for this air conditioner.

At present, very little information is available which quantifies the performance of a residential cooling system operating under degraded conditions such as reduced evaporator air flow. Degraded performance measurements can provide information which could help electric utilities evaluate the potential impact of system-wide maintenance programs.

INTRODUCTION

Energy use in heating and cooling systems represents 20% of the energy use in the residential sector[1]. Approximately 30.7 million central system air conditioners are used in residences[2]. Each year, 4 million new units are installed[3]. The average life of a residential unit is about 15 years[4]. Equipment service and replacement costs represent 40% of residential HVAC trade. In 1988, alone $2.75 billion was spent on residential HVAC maintenance and repair. A typical hourly rate in the residential cooling market costs $40 per hour[5]. One of the major problems facing the residential cooling service industry is a lack of adequately trained service technicians[6]. At the root of the problem is the fact that diagnostic techniques are typically based on the experience of the service personnel and remedial measures vary widely with each individual's background.

The demand for more efficient air conditioners is currently increasing, prompted by the fact that residential energy costs have increased by 200% in the last 15 years[7]. This trend was accelerated by the Appliance Minimum Efficiency Standard, which became effective in 1992[8]. Utilities are also being asked to increase and measure their demand-side savings. However, simply installing an high efficient air conditioner does not always mean reduced energy use. The net effect is that air conditioning installers and servicemen are being asked to look at much more than just installing and charging the system.

Air Conditioning Installation and Service

The benefits from a high efficiency air conditioner will be reduced unless it is properly installed and maintained. Field studies show that many of the existing installations are of poor quality and require immediate service to save the equipment or to restore it to rated efficiency[9,10]. Experts from air conditioning service industry also point out that manufacturers are not addressing the difficulties that arise in field installations[11].

Need for Preventive Maintenance and Diagnosis System

Current trends in the air conditioning service industry are based on corrective maintenance procedures which are initiated only after a failure occurs. Preventive maintenance is rarely used. Some possible reasons include: (1) customers' reluctance to pay, and (2) contractors' lack of the necessary knowledge to provide an effective preventive maintenance procedure.

Preventive maintenance can be primarily classified as [12]: (1) On Condition Maintenance (OCM) where degradation prior to functional failure can be detected by periodic inspections and evaluations, or (2) Condition Monitoring (CM) where degradation prior to functional failure can be detected in sufficient time by instrumentation(e.g., temperature or pressure).

Regardless of which approach is followed preventive maintenance is only effective when potential failures can be ascertained reliably and inexpensively and where the prevention of the failure more than pays for the diagnosis. To predict the shortfall between performance and standards, it is necessary to develop a criteria which are capable of recognizing the degraded conditions.

Objective of Current Investigation

Currently, very little information is available which describes the expected performance of air conditioners under degraded conditions. The air conditioning industry has shown a concern regarding this and several trade journal articles address this issue [13,14]. This problem is further compounded by a lack of consensus for developing proper service procedures[15].

The primary focus of this work is to measure energy consumption, cooling capacity and energy efficiency ratio (EER) during degraded conditions. A detailed discussion on the reduction in evaporator air flow is illustrated here as an example. The long term goal of this work is to classify degraded conditions for the most widely experienced problems on a test bench so that eventually field diagnosis procedures can be developed.

BACKGROUND

Service and Failure Patterns

Several studies have investigated HVAC service problems. Karger and Carpenter discussed failure patterns of residential air conditioning units based on their survey of 531 failed units [16]. This is one of the first studies conducted on failed air

conditioning systems. Their study indicated that failure rates for electrical controls and miscellaneous electrical devices represented 31.4% of the total number of failures and refrigerant leaks constituted 17.2% of the total failure. The failure rates of compressors was 13.5% and outdoor fans was about 11.5%.

Lewis surveyed 492 large HVAC dealers to compile information on heat pump service life [5]. He discovered that refrigerant leaks were the major cause for failure, totaling 19% of failed units. Compressor motor circuits and mechanical part failures were 16% and 12% respectively.

Neal investigated the quality of residential air conditioning system installation and service in North Carolina [9]. His random survey of 10 units indicated that inadequate evaporator air flow was present in 3 of the 10 units(30%) and improper charging prevailed in 7 units(70%). He concluded that a very high percentage of air conditioning units have installation or service problems that affect homeowner's energy bills and comfort.

Proctor tested fifteen homes in Fresno, CA which reported very high summertime energy consumption levels in the Appliance Doctor Pilot Program, a project designed to investigate the causes for the high energy bills [10]. Proctor's results showed low evaporator air supply existed in 10 out of 15 sites(67%). Overcharging and undercharging was found in 8 locations(53%). Refrigerant leaks and kinked lines were present in 6 houses(40%). Remedial measures reduced cooling energy costs from 10 to 30%. He concluded through interviews and field tests that the HVAC contractors who maintained these systems were not identifying or solving the problems that lead to high energy bills.

Degradation Studies

Houcek studied the effect of improper charging by conducting experiments on a 2-ton split system with 37 feet of interconnecting refrigerant lines [17]. Supply air entering the indoor coil was maintained at 80° F DBT and 67° F WBT. Air entering the outdoor unit was maintained at different conditions: 70° F, 82° F, 95° F and 100° F. Houcek's experiments showed that at 95° F outdoor conditions, overcharging by 23% decreases the operating cost by 0.5%. However, this was observed to cause a floodback condition (Floodback describes a situation when liquid refrigerant enters the compressor. Compression of liquid refrigerant will cause mechanical failure of compressor). For undercharging, it was estimated that the operating cost increased by as much as 52%. He recommended a new device (a Visual Accumulator-Charger) for charging air conditioners effectively.

Farzad studied the effect of undercharging and overcharging for three expansion devices: (i) a capillary tube, (ii) a short-tube orifice, and (iii) a thermostatic expansion valve [18]. He conducted his experiments under controlled conditions specified by DOE/ARI testing procedures for Unitary Air Conditioners. He varied the system charge from +20% to -20% for different outdoor temperatures. His study showed that capillary tube expansion is more sensitive to off-charge conditions than other types of expansion valves. For a 20% undercharge the Seasonal Energy Efficiency Ratio (SEER) was reduced by 20%, while overcharging by 20% produced an 11% reduction in SEER. He found that the SEER was not very sensitive to changes from

COMPONENT	SPECIFICATION
COMPRESSOR	Single Speed, 230 V, S-P FLA 17.1 and LRA 79.0 amps Employs internal line breaker(IOL) Internal Pressure Relief (IPR) opens at 350 psig difference
STRAINER DRIER DRIER	Discharge line with sweat fitting 8 cubic inch, 3/8 X 3/8" Liquid line after expansion valve 30 cubic inch, 1/4 X 1/4"
OUTDOOR FAN	3 Blade, Propeller type 22 inch Diameter 2735 CFM @ 0 in. w.g. 1/4 HP motor at 825 RPM
CONDENSER COIL	Spine-fin type, 20 fins per inch Face area of the coil 20.94 sq. ft. Refrigerant tube size is 3/8 inch OD Fan located at top of coil
EXPANSION VALVE	Short-tube orifice restrictor 0.071 inch internal diameter
INDOOR BLOWER	Centrifugal type 1200 CFM delivered at 0.7 in. w.g. Power Consumption 320 Watts
COOLING COIL	Four row, four circuit vertical coil Refrigerant tube size is 3/8 inch Wavy Fin type, 12 fins per inch Coil frontal area is 3.33 sq. ft. Design Cooling Capacity is 36000 Btu per hour

Table 1. List of Components

-20% to +20% of the correct charge when an orifice tube is used for an expansion device.

EXPERIMENTAL APPARATUS AND PROCEDURE

A review of the references showed us that the following degraded tests represented the majority of degraded field conditions:

    1. Insufficient evaporator air flow
    2. Insufficient condensing unit air flow
    3. System undercharging and overcharging
    4. Non-condensable gases (such as air) in the system
    5. Restrictions in the refrigerant lines.

Hence, a test bench was set up to study the effects of these conditions. This paper discusses the effect of insufficient evaporator air flow on the performance of an air conditioner.

Test Bench

In this project, the air conditioning system considered for analysis is a standard split system air conditioner which contains a short-tube orifice expansion device. The cooling coil has a maximum three ton capacity. A schematic diagram and the outline of the experimental setup are shown in Figure 1. A detailed description of each component is described in Table 1.
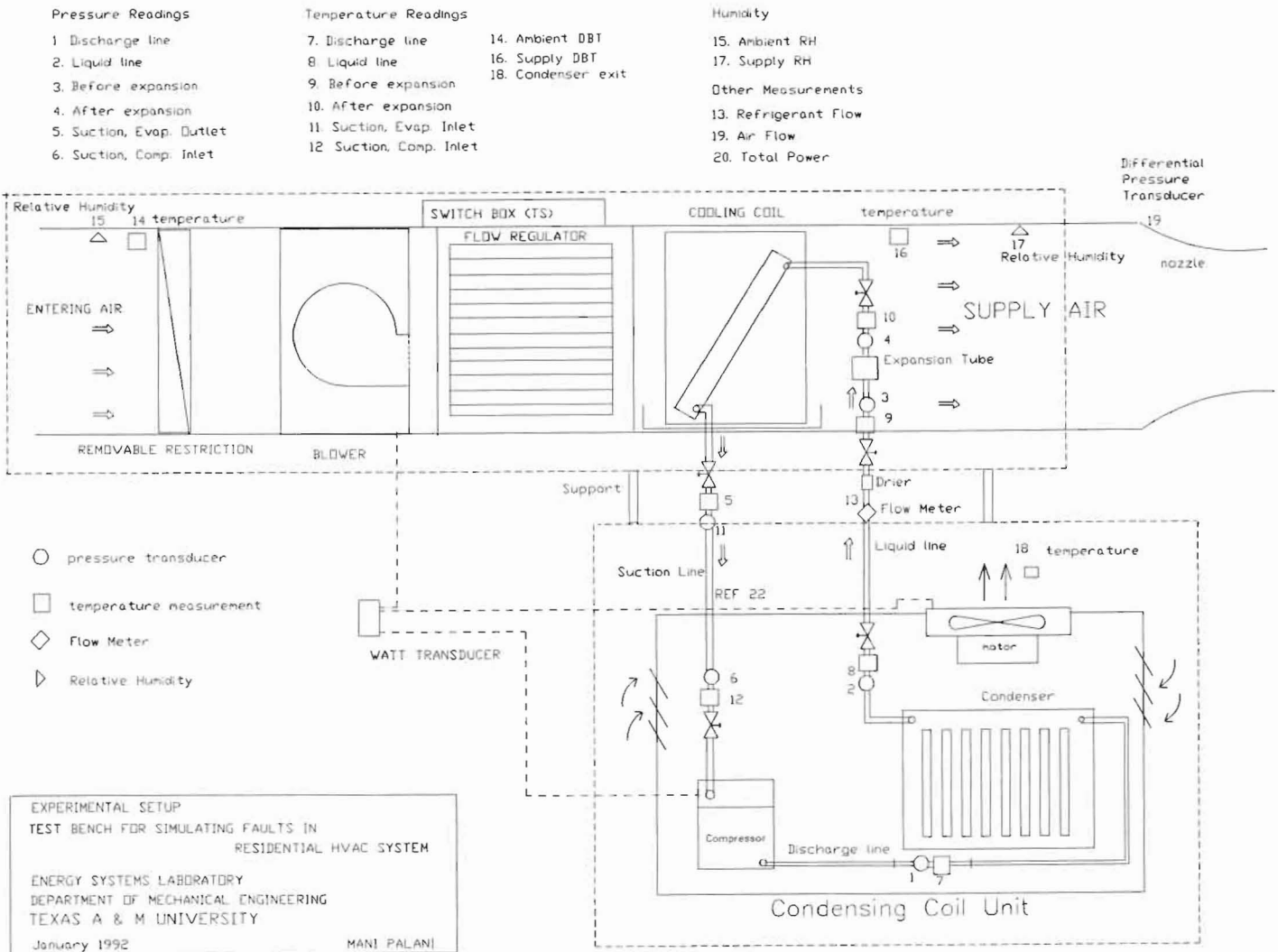
Pressure Readings

1 Discharge line
2 Liquid line
3 Before expansion
4 After expansion
5 Suction, Evap. Outlet
6 Suction, Comp. Inlet

Temperature Readings

7 Discharge line
8 Liquid line
9 Before expansion
10 After expansion
11 Suction, Evap. Inlet
12 Suction, Comp. Inlet

14 Ambient DBT
16 Supply DBT
18 Condenser exit

Humidity

15 Ambient RH
17 Supply RH

Other Measurements

13 Refrigerant Flow
19 Air Flow
20 Total Power

○ pressure transducer
□ temperature measurement
◇ Flow Meter
▷ Relative Humidity

EXPERIMENTAL SETUP
TEST BENCH FOR SIMULATING FAULTS IN
RESIDENTIAL HVAC SYSTEM

ENERGY SYSTEMS LABORATORY
DEPARTMENT OF MECHANICAL ENGINEERING
TEXAS A & M UNIVERSITY

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Figure 1. Experimental Setup

LOCATION	QUANTITY	UNITS	INSTRUMENT	RANGE AND TYPE
1 Discharge line	Pressure	PSIG	Pressure Transducer	0 - 500 PSIG
2 Liquid line	Pressure	PSIG	Pressure Transducer	0 - 300 PSIG
3 Before expansion	Pressure	PSIG	Pressure Transducer	0 - 300 PSIG
4 After expansion	Pressure	PSIG	Pressure Transducer	0 - 300 PSIG
5 Suction line, after evaporator	Pressure	PSIG	Pressure Transducer	0 - 300 PSIG
6 Suction line, before compressor	Pressure	PSIG	Pressure Transducer	0 - 300 PSIG
7 Discharge line	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
8 Liquid line	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
9 Before expansion	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
10 After expansion	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
11 Suction line, after evaporator	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
12 Suction line, before compressor	Temperature	DEG F	Thermocouple	TYPE T (Copper -Constantan)
13 Liquid line	Refrigerant flow	LB/MIN	Mass flow meter	0 - 60 lb/min, Coriolis-type
14 Return air (ambient)	Dry bulb temperature	DEG F	Thermocouple grid	TYPE T (Copper -Constantan)
15 Return air (ambient)	Relative humidity	PER CENT	RH sensor	Thin film capacitive element
16 Supply air	Dry bulb temperature	DEG F	Thermocouple grid	TYPE T (Copper -Constantan)
17 Supply air	Relative humidity	PER CENT	RH sensor	Thin film capacitive element
18 Condenser discharge air	Dry bulb temperature	DEG F	Thermocouple grid	TYPE T (Copper -Constantan)
19 Nozzle	Air supply	CFM	Diff. pressure transducer	0 - 1 inch w.g.
20 Power supply	Total power	WATTS	Watt Transducer	0 - 4 kW, 4 - 20 mA OUTPUT

Table 2. List of Properties Measured

Measurement

A list of properties which were measured at the test bench is shown in Table 2. A total of 20 quantities were measured with a data acquisition system. Refrigerant pressures and temperatures were measured at six locations, as shown in Figure 1. Properties measured on the air side included dry bulb temperature and relative humidity at outdoor conditions and supply conditions. Air temperature at the discharge of the condensing fan was also measured. Copper-constantan thermocouples were used to measure refrigerant and air temperatures. The refrigerant mass flow rate was measured at the liquid line before the expansion valve. A differential pressure transducer was used to measure the static pressure gain before the supply nozzle, which was converted into an air flow rate. The total amount of power to run the compressor, blower and condenser fan was measured with a watt transducer.

Test Operating Procedure

The DOE/ARI specifications indicate that steady-state and cyclic tests for Unitary Air Conditioners should be conducted in controlled environment for 30 minutes [19]. We found that the test bench reached steady-state within 4 minutes after switching on the unit. The coefficient of variation for the standard test was less than 1% which was satisfactory for use in these tests. Another reason for using reduced test duration was to minimize damage to components during certain degraded tests.

Degradation Simulation

Reduction in evaporator air flow was simulated using a plywood restriction board to cover the supply air duct. The plywood board was pre-drilled at several places to allow air flow from 100 CFM to 1000 CFM. Supply air flow was varied by covering the appropriate holes.

Data Reduction and Performance Calculations

For each test 3 second time series data were taken for refrigerant pressure, refrigerant temperature, air temperature and relative humidity, as shown in Figure 2. To calculate the performance factors, values measured during steady-state conditions were averaged for the last five minutes of the test. Refrigerant enthalpies at six locations were calculated using the refrigerant property calculation program developed by Kartsounes[20]. Air-side enthalpy, humidity and specific volume were calculated by a psychrometric program developed at the Energy Systems Laboratory, Texas A&M University[21].

RESULTS AND DISCUSSION

A summary of results is presented in Tables 3 and 4. Pressure-enthalpy and psychrometric charts for the 90% reduction in evaporator air supply are shown in Figure 3.

Power, Cooling Capacity and EER

The total power consumption for running the compressor, blower and outdoor fan during the standard run was 3675 watts. Results shows that as evaporator air flow is reduced from a normal amount the electricity demand, cooling capacity and EER decrease. Power consumption decrease in a near linear fashion, from 3.54% at 25% reduction in evaporator air flow to 17% at 90% reduction in evaporator air flow. This may imply
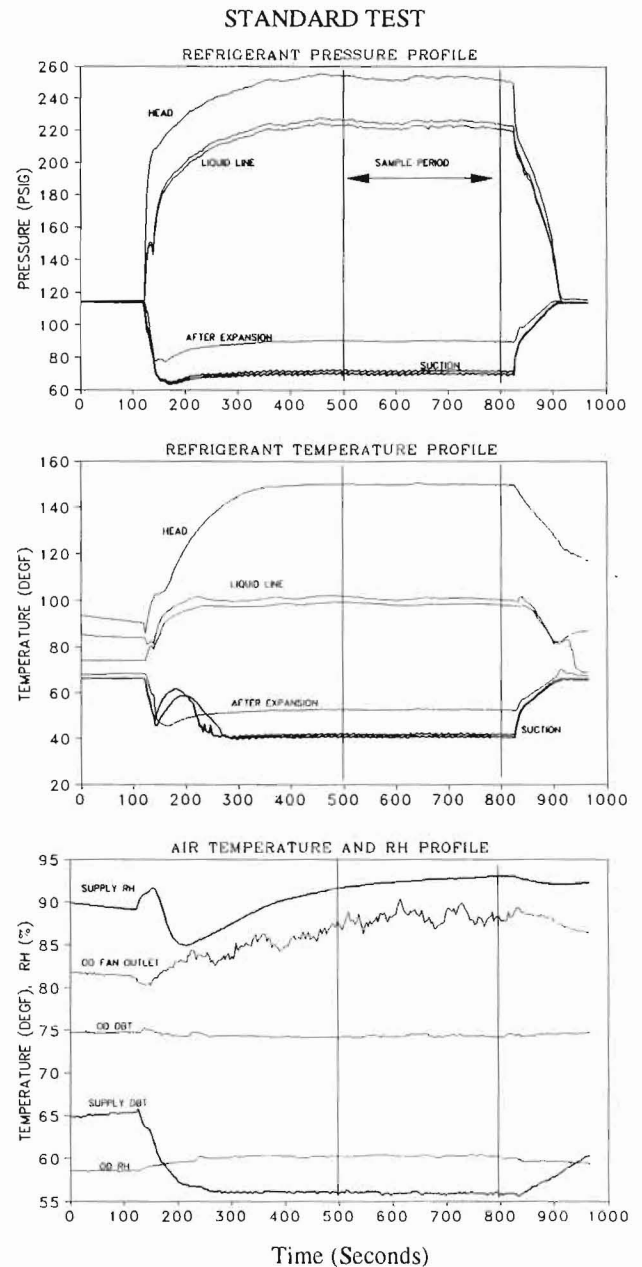
STANDARD TEST

Figure 2. Time Series for 90% Reduction in Evaporator air flow

that as utilities fix degraded air conditioners the demand may go up by 3-17% while usage goes down.

Cooling capacity decreases linearly until about 50% evaporator air flow then dropped suddenly. Measurements showed that cooling capacity reduced by 7.5% at 25% reduction in evaporator air flow to 14.5% for 50% reduction in air flow. Cooling capacity decreased by 45% for 75% reduction in evaporator air flow. For 90% reduction in evaporator air flow, the degradation of performance was the highest, as expected; cooling capacity decreased by 76%.

The sensible heat ratio (SHR - ratio of the sensible capacity to total capacity) for the standard run at 75° F DBT and

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SIMULATED FAULTS	STANDARD AND DEGRADED TEST RESULTS																				
	AMBIENT DBT(F)/RH(%)		EVAPORATOR SUPPLY AIR TEMP (DEG F)		EVAPORATOR TEMP DROP DEG F		CONDENSER TEMP RISE DEG F		DISCHARGE PRESSURE PSIA		SUCTION PRESSURE PSIA		DISCHARGE TEMP DEG F		SUPPLY AIR FLOW CFM		POWER WATTS		EER	
	STANDARD	DEGRADED	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR
DROP IN EVAP. AIR FLOW																				
1. 25% REDUCTION	74.3/60	74.7/60	56	54	18	21	14	10	267	256	85	81	150	142	1135	843	3676	3546	9.53	9.13
(Coeff. of Var., %)	(0.18/0.14)	(0.13/0.15)	(0.22)	(0.21)	(0.28)	(0.25)	(0.9)	(0.86)	(0.35)	(0.38)	(0.62)	(0.78)	(0.12)	(0.46)	(0.37)	(1.01)	(0.21)	(0.31)	(0.96)	(0.86)
2. 50% REDUCTION	74.3/60	75/60	56	48	18	27	14	10	267	242	85	74	150	134	1136	644	3675	3347	9.53	8.93
(Coeff. of Var., %)	(0.18/0.14)	(0.17/0.21)	(0.22)	(0.31)	(0.28)	(0.35)	(0.9)	(0.7)	(0.35)	(0.31)	(0.62)	(1.0)	(0.12)	(0.31)	(0.37)	(1.0)	(0.21)	(0.28)	(0.96)	(0 7)
3. 75% REDUCTION	74.3/60	76.2/59	56	40	18	35	14	8	267	223	85	64	150	125	1135	254	3675	3108	9.53	6.23
(Coeff. of Var., %)	(0.18/0 14)	(0.12/0.28)	(0 22)	(0.36)	(0.28)	(0.38)	(0.9)	(0.49)	(0.35)	(0.27)	(0.62)	(0.55)	(0.12)	(0.61)	(0.37)	(2.87)	(0.21)	(0.29)	(0.96)	(3.08)
4. 90% REDUCTION	74.3/60	75.2/60	56	40	18	35	14	8	267	214	85	58	150	118	1135	109	3675	3042	9.53	2.75
(Coeff. of Var., %)	(0.18/0.14)	(0.12/0.21)	(0.22)	(3.67)	(0.28)	(3.67)	(0.9)	(0.52)	(0.35)	(1.58)	(0.62)	(3.78)	(0.12)	(1.75)	(0.37)	17.51	(0.21)	(1.02)	(0.96)	(19.39)

Note : (1) CV (A) = (SDEV/MEAN)*100
(2) CV (A, B) = [CV (A)^2 + CV (B)^2]^1/2

Table 3. Summary of Results

SIMULATED FAULTS	PERCENT OF VARIATION		
	CAPACITY KBtu/hr	POWER WATTS	EER
REDUCTION IN EVAPORATOR AIR FLOW			
1. 25% REDUCTION	-7.51	-3.54	-4.20
2. 50% REDUCTION	-14.71	-8.93	-6.51
3. 75% REDUCTION	-44.71	-15.43	-34.63
4. 90% REDUCTION	-76.11	-17.22	-71.14

Table 4. Summary of Degraded Tests

60% RH was 0.64. SHR decreases to 0.60, 0.54, 0.51 and 0.51 for 25%, 50%, 75% and 90% reductions in evaporator air flow respectively.

Reductions in the energy efficiency ratio(EER) are similar to reductions in cooling capacity(Figure 4). Performance is linear until about 50% reduction in evaporator air flow, and non-linear afterwards. EER decreases by 4.2%, 6.5%, 34.6% and 71.1% for 25%, 50%, 75% and 90% reductions in evaporator air flow respectively[Table 4]. Figure 4 shows the trends of cooling capacity, power and EER for reduced evaporator air flow.

Temperature Difference Across Coils

Reducing the evaporator air supply increased the temperature drop across the cooling coil from 18° F at normal flow to 35° F at 30% of normal flow [Figure 4]. However, air temperature rise across the condenser decreased from 14° F at normal flow to 8° F at 30 % of evaporator air flow.
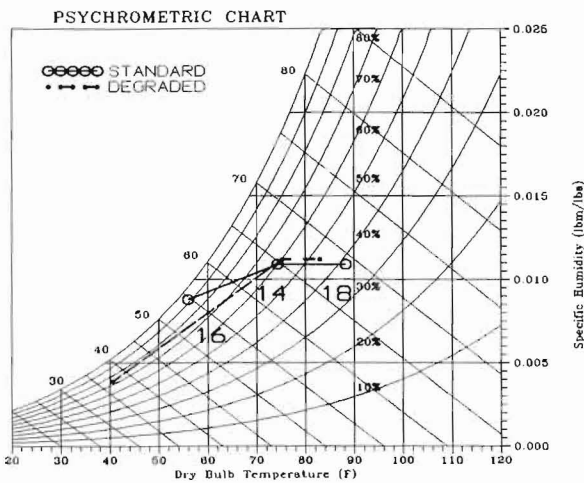
For 90% reduction in evaporator air flow, compressor discharge pressure and temperature decrease by 20%. The drop in suction pressure is considerable (30%) for a 90% reduced air flow rate [Table 3]. The corresponding drop in suction temperature was from 40° F at standard test conditions to 20° F at 90% reduced evaporator air flow rate. This near freezing temperature level may be one of the primary reasons for a considerable reduction in cooling capacity at reduced evaporator air flow. There is no appreciable change in degree of superheat (0 to 1° F) during degraded tests. However, the degree of subcooling dropped from 8° F at standard conditions to 1° F at reduced evaporator air flow.
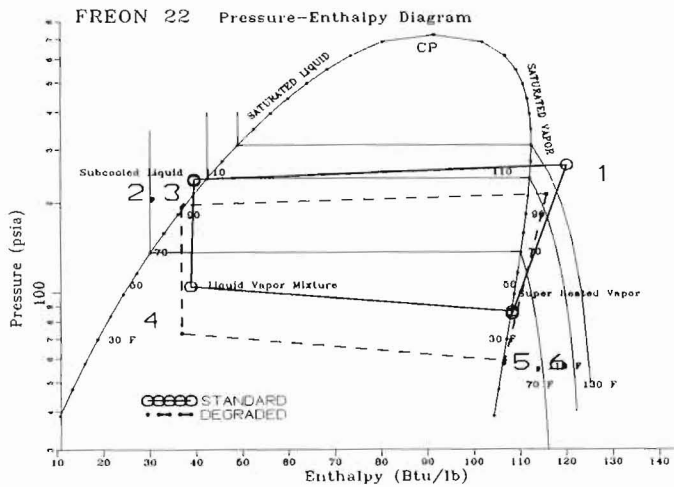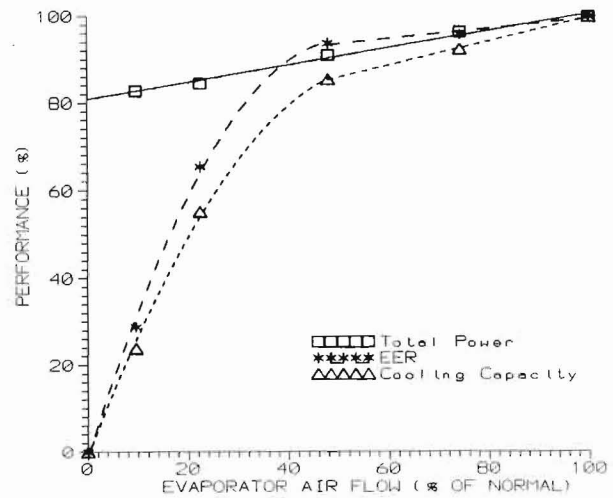
CONCLUSIONS

Degraded field conditions of an air conditioner has been measured on a test bench. At 90% reduced evaporator air flow rate, the total power consumption decreases by 17% and the EER decreases by 71%. The degraded condition test results

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90% REDUCTION IN EVAPORATOR AIR FLOW

PSYCHROMETRIC CHART

FREON 22   Pressure-Enthalpy Diagram

Figure 3. P-H and Psychrometric chart

REDUCTION IN EVAPORATOR AIR FLOW
(TOTAL POWER AND EER)

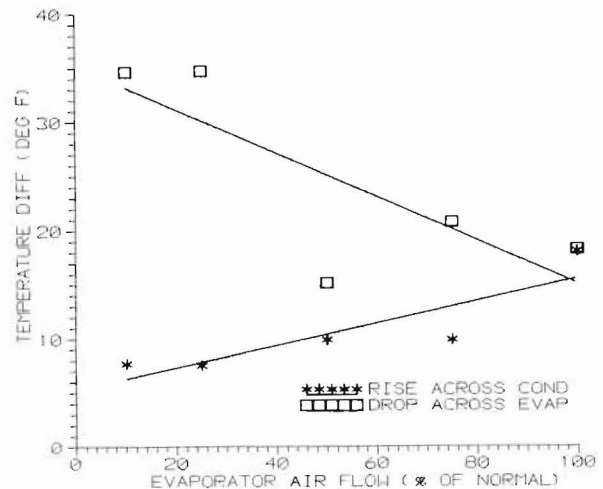REDUCTION IN EVAPORATOR AIR FLOW
(TEMP DIFFERENCE ACROSS COILS)

Figure 4. Performance of Evaporator under Reduced Air Flow

indicated that to maintain sufficient cooling, one definitely must have at least 50% of rated air flow. Low evaporator flow conditions might be inexpensively detected in the field through large temperature difference across cooling coils, reduced condenser discharge temperatures and reduced head temperatures.

FUTURE DIRECTIONS

The preliminary results from the first phase of this on-going project are encouraging. Experience with degraded tests can provide a better procedure to simulate faults in the future. As performance factors for air conditioners vary with different indoor and outdoor conditions, further tests will be conducted at different outdoor temperatures.

The ultimate goal of this project is to develop a temperature based diagnosis procedure. Experience with reduced evaporator flow tests indicated that easily identifiable

temperature patterns exist for each degraded condition which can be utilized for developing an automated diagnosis procedure.

Some commercially available automatic diagnostic systems are already beginning to appear. Kaler developed one such monitoring device which continuously monitors the HVAC system for selected system malfunctions [22]. This device continuously tracks the temperatures of an HVAC system and evaluates this value with an embedded knowledge base, and warns the owner in advance of any potential trouble. Further testing of such systems on a test bench can accelerate their acceptance into the marketplace.

ACKNOWLEDGMENTS

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APPENDIX

Performance Parameters (ASHRAE Std. 116-1983)[23]

Air side enthalpy difference, Δh (Btu/lb.) = (Enthalpy of air at OD - Enthalpy of air at supply)   (1)

Cooling capacity (Btu/hr).

Cooling capacity = (60 X Δh X Evaporator air flow rate (CFM)) / (Sp. vol. of air at supply conditions X (1 + Sp. Humidity))   (2)

Energy efficiency ratio (EER).

EER = Cooling capacity / Total power   (3)

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