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TESTING AND ECONOMIC EVALUATION OF A HIGH EFFICIENCY 10 TON ROOFTOP AIR CONDITIONER

Final Report

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EXECUTIVE SUMMARY

In 1993, the U.S. Environmental Protection Agency initiated a project to design, build and demonstrate a high efficiency commercial rooftop air conditioning unit. The unit was designed by Hibberd Consulting of Westminster, Colorado and was built by LaSalle Manufacturing Corporation of Houston, Texas. The rooftop unit was then tested at Texas A & M University. The economics of manufacturing, purchasing and operating were also evaluated. This report summarizes the testing of the unit and an evaluation of the economics of the system.

This unit was designed to have both a high steady state and part load efficiency. For rooftop air conditioners, the capacity and Energy Efficiency Ratio (EER) is used to rate the unit for steady state operation and the Integrated Part Load Value (IPLV) is used to rate the unit for part load operation. The design values for the EER and IPLV were 12.9 Btu/Wh and 17.7 Btu/Wh, respectively. By contrast, a 10 ton rooftop sold in 1995 had EERs ranging from approximately 9 to 11 Btu/Wh and an IPLV of 9 to 11.7 Btu/Wh. To achieve these high efficiencies, the unit included three high efficiency compressors, larger evaporator and condenser surface areas, high efficiency fans, high efficiency fan motors and variable speed control of fan motors. The compressors were state-of-the-art reciprocating compressors. The use of three compressors allowed for five capacity steps for part-load operation. The high efficiency fans for both the condenser and evaporator used airfoil design blades. The variable speed controllers allowed the fan power to be significantly reduced during part load operation.

The unit was tested with the use of the psychrometric facilities at the Energy Systems Laboratory at Texas A & M University. A set of baseline tests were first run with the unit at fan speeds that would produce design airflow conditions. For this case, the unit measured EER and the IPLV were 11.15 Btu/Wh and 13.39 Btu/Wh respectively. These values were higher than the best 10 ton rooftop unit currently on the market. However, the measured EER and IPLV were 13.6% and 24.7% below their respective design values.

The airflow through both the condenser and evaporator were varied to determine if the performance could be improved. Optimum fan speeds were found for the nominal 2, 4, 8, and 10 ton capacity steps. The EER improved to 11.51 Btu/Wh and the IPLV increased to 13.80 Btu/Wh. Close inspection of the unit and the data collected indicated that there was a combination of at least five different factors that were responsible for the measured performance being less than the design. These included: higher than expected power consumption with compressors, poor refrigerant circuiting for one of the compressors, non-uniform airflow across the evaporator, internal leakage of air from the evaporator section, and high power consumption by the evaporator blower. While none of these sources of efficiency losses individually had more than a 1 to 3% effect on efficiency, when combined they produced over a 10% loss in the measured EER.

An overwhelming success with this unit was the design of the fan systems. This unit demonstrated that a combination of high efficiency fans and motors plus variable speed controllers could be used to provide a substantial drop in the power used by the fans and increase the IPLVs. At full load operation, the evaporator fan used about half the energy used in conventional rooftops. With the VFDs, it was possible to optimize airflow through both the condenser and evaporator to maximize the efficiency at each of the five capacity steps. This resulted in an IPLV that was almost 20% higher than the EER. Correction of some of the problems with the unit and optimization of the 6 ton capacity step could potentially have led to IPLVs even higher. By contrast, several of the 10 ton systems on the market had IPLVs that were only 0 to 7% higher than the EER.

Added costs to manufacture the unit were estimated at $729 over that of a 9 Btu/Wh EER unit. The added retail price for the high efficiency unit was estimated at $2187. A small two-story office building was simulated with the building simulation software ASEAM 3.0 for five different locations: Chicago, Dallas, Los Angeles, Miami, and Washington, D.C. Comparisons of energy use and electrical demand were made between the 9 Btu/Wh EER unit and the high efficiency unit, both as tested and with the problems fixed that were identified during the tests. The high efficiency unit with all problems fixed was estimated to provide annual savings between $752 and $919, which translated into simple paybacks between 2.4 to 2.9 years. With the inclusion of electric utility rebates of $100/ton, the payback was reduced to less than 1.6 years. Besides offering high efficiencies, this unit also could reduce the electrical demand from a typical 10 ton air conditioner by as much as 3.7 kW.

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CHAPTER 1

INTRODUCTION

Rooftop air conditioners are used extensively in commercial building applications in the United States. They are often found in retail store, supermarkets, schools, low rise office building and restaurants. Unlike smaller residential air conditioners which have seen large increases in efficiency over the past 15 years, the efficiency of rooftop air conditioners has changed little during this same period. Only recently have rooftop air conditioners larger than 5 ton capacity had minimum efficiency standard applied to them. Congress passed the National Energy Policy Act (1992) which required a minimum Energy Efficiency Ratio (EER) of 8.9 Btu/Wh @95°F outdoor temperature for all units with capacities between 65,000 and 135,000 Btu/h. For larger air conditioners (135,000 to 240,000 Btu/h), the minimum EER requirement was 8.5 Btu/Wh @95°F. These ratings are steady state ratings which differ from the seasonal ratings required for residential sized air conditioners and heat pumps.

One of the major reasons given for the lack of significant improvements in the efficiency of rooftop air conditioners is the perceived lack of market for a high efficiency rooftop air conditioner (EPA 1993). Many rooftop air conditioners are installed in speculative property where first cost is the highest priority. Retail and office space often fall into this category. For speculative buildings, the owner of the building simply passes the costs of operation to the tenant. Because energy costs represent only a small fraction of total rent on a building, there is little incentive to install high efficiency air conditioners.

High efficiency rooftop air conditioners could benefit a number of parties. Electric utilities have major concerns about energy demand at high outdoor temperatures which may coincide with their system peak demand. Higher efficiency rooftop air conditioners could help reduce their peak demand. For the person occupying a building, high efficiency air conditioners could help reduce their energy and demand charges. Improvements in the efficiency of rooftop air conditioners similar to that experienced with smaller residential equipment could reduce the energy used by rooftop air conditioners by 30 to 40% over the minimum efficiencies required by the National Energy Policy Act (1992). Society could benefit from higher efficiency air conditioners by seeing a reduction in total energy use and emissions of greenhouse gases.

In 1993, the U.S. Environmental Protection Agency initiated a project to design, build and demonstrate a high efficiency commercial rooftop air conditioning unit. The unit was designed by D. Hibberd Consulting of Westminster, Colorado, and built by LaSalle Manufacturing of Houston, Texas. The unit was then to be tested and then an economic analysis performed by Texas A&M University. This report summarizes the testing of the unit and an evaluation of the economics of the system. The tests were conducted with the use of the psychrometric facilities at the Energy Systems Laboratory at Texas A & M University. This report is divided into six chapters. Chapter two provides a brief description of the design of the air conditioner. Chapter three describes the test setup and procedures for testing the unit. The test results are described in chapter four. Chapter five describes the economic evaluation of the unit and analysis of higher efficiency options. Conclusions and recommendation are provided in chapter six.

CHAPTER 2

UNIT DESCRIPTION

The rooftop air conditioner was designed by D. Hibberd Consulting in Westminster, Colorado, and constructed by LaSalle Manufacturing Corporation in Houston, Texas. The unit was delivered to Texas A & M University in late September, 1994. A complete description of the design of the unit can be found in Hibberd (1993). This chapter provides a brief overview of the design.

This unit was designed to have both a high steady state and part load rating. For rooftop air conditioners, the capacity and Energy Efficiency Ratio (EER) is used to rate the unit for steady state operation and the Integrated Part Load Value (IPLV) is used to rate the unit for part load operation. The EER is derived from tests at outdoor temperature of 95°F and indoor conditions of 80°F dry bulb and 67°F wet bulb. The IPLV is derived from a series of tests at an outdoor temperature of 80°F and indoor conditions of 80°F and 67°F wet bulb. The design values for the EER and IPLV were 12.9 Btu/Wh and 17.7 Btu/Wh, respectively. Table 2.1 shows the design performance of the unit at 25, 50, 75, and 100% of design capacity. The high design IPLV comes from the large EER values at 50% and 25% capacity. The ARI 210/240 test procedure weights 64% of the IPLV at these capacities (ARI 1989). Thus, obtaining a high IPLV requires high EERs at the lower capacity steps.

Table 2.1 - Design performance of the 10 ton rooftop unit.

VARIABLE	Unit Capacity			
	100%	75%	50%	40%
Net Capacity (tons)	10.2	7.7	5.2	2.6
Total Power (kW)	9.49	5.50	3.39	1.69
EER (Btu/Wh)	12.9	16.8	18.4	18.4
Evaporator Fan				
Speed (Frequency)	60	45	30	24
Power (kW)	0.49	0.27	0.12	0.08
Condenser Fan				
Speed (Frequency)	60	45	30	30
Power (kW)	1.34	0.73	0.32	0.32

The unit started with the chassis of a 15 ton Trane air conditioner. In order to get to a higher efficiency, a number of design changes were made in this unit compared to the typical rooftop air conditioner. These included: three high efficiency compressors, larger evaporator and condenser surface areas, high efficiency fans, high efficiency fan motors and variable speed control of fan motors. Each is briefly discussed below.

This unit used three high efficiency reciprocating compressors manufactured by Bristol Compressors. A review of 10 ton rooftop designs from three manufacturers indicated that the typical unit used two compressors rather than three. Two of the compressors were four tons nominal capacity with an EER rating of 16.2 Btu/Wh at 45°F evaporating and 110°F condensing temperature. The third compressor had a nominal two ton capacity with an EER rating of 15.5 Btu/Wh. The use of two four ton compressors and one two ton compressors allowed for part load operation at 20, 40, 60, 80, and 100% of full load. Increasing the number of points for part load operation allowed for a better match of the system capacity to the load on a building.

The unit also had a large evaporator. The evaporator had 17.67 ft^2 of face area with three rows and a fin density of 14 fins/inch with internally grooved tubes. In contrast, two 10 ton units from one manufacturer both had 9.46 ft^2 of face area with four rows and 12 fins/inch. One of the units had an EER of 9 Btu/Wh and the other 10 Btu/Wh. The unit with a 10 Btu/Wh EER also had an IPLV of 10.6 Btu/Wh. Thus, the evaporator in the high efficiency unit was significantly larger than those used in systems that meet the current minimum efficiency standards.

The condenser on this unit had 24 ft^2 of face area with two rows of internally grooved tubes and 26 fins/inch. Units available today have face areas ranging from approximately 20 to 30 ft^2. One feature of this unit is the fin density. A typical unit has 17 to 19 fins/inch while this high efficiency design had 26 fins/inch. The higher fin density would give the unit a much larger condenser surface area.

An important feature of the unit was the fans, fan motors and their controllers. The efficiency of the evaporator fan was 60% while that for the condenser was 50%. The evaporator fan was a double width, double inlet, backwardly inclined airfoil fan. Generally, the backward inclined airfoil blade designs should have much higher efficiencies than the forward curved design fans commonly used in package air conditioners (McQuiston and Parker 1993). The motor used to drive the evaporator fan was rated at 1.5 horsepower with a 86% efficiency. The evaporator fan/motor combination was designed to use just 153 watts/1000 cfm at full load, which was 3200 cfm at 0.63 in. external static pressure. Typical evaporator fan powers in 10 ton units from two major manufacturers ranged from 350 to 376 watts/1000 cfm for 0.6 inch external static pressure.

The unit had two condenser fans. Both were direct drive, 26 inch diameter, airfoil, axial propeller type fans. Each fan was driven by a 1 horsepower motor with an efficiency of 85%. The fans were designed to produce 10,500 cfm at 0.45 inches static pressure (Hibberd 1993). The fan power at design conditions was 1.34 kW.

In addition to a high efficiency fan and motor, the unit was design with variable speed drives for both the evaporator and condenser fan motors. The variable speed drives allow control of fan speed to better match the airflow to the capacity of the unit. While fan airflow is a linear function of the speed of the fan, the fan power is a function of the cube of the fan speed. Thus, if the fan speed could be reduced by half, the airflow would drop by 50%, but the fan power would drop by 87.5%.

The use of variable speed drives to control the evaporator and condenser fans also allows for the option of optimizing the efficiency of the unit for a given capacity step. As has been demonstrated by a number of studies (Rice 1992 and Boecker 1987), the optimum efficiency of an air conditioner is a function of both the evaporator and condenser air flow. For a given unit, this optimum point requires some trial and error to find the best airflows. With five capacity steps, the evaporator and condenser fan speeds can be varied to find the fan speeds that provide the optimum efficiency at each step.

CHAPTER 3

TEST SETUP AND PROCEDURE

The 10 ton rooftop unit was tested using the psychrometric facilities at the Energy Systems Laboratory. These facilities include two rooms. One was called the indoor room because it was used to simulate indoor conditions during a test. The other room was called the outdoor room because it was used to simulate outdoor conditions during a test.

Normally during a test of a rooftop unit, it would be placed in the outdoor room and the supply and return ducts would be run to the indoor room. Because of the size of the unit, it was not possible to physically place the unit inside the outdoor room. The unit was placed outside the outdoor psychrometric room. This led to a slightly more complicated test setup than would normally be used with a rooftop unit.

Duct was run from the outdoor psychrometric room to the condenser (Figure 3.1). The duct cross sectional area from the outdoor room to the condenser was approximately 29 ft^2. Air was drawn from the outdoor room and discharged directly into the laboratory. Measurements were made at the entrance to the condenser to determine the pressure loss generated by the addition of the ductwork. For full load operating conditions in the condenser, the total pressure drop through the duct section to the condenser was 0.02 inches of water.

Condenser inlet air temperature and humidity were measured directly in front of the condenser coil with a thermocouple grid and relative humidity sensor connected to an aspirated sampling system which pulled air from the cross section of the duct. The entering condenser air temperature was maintained with a chilled water cooling coil in the room and by 40 kW of electric resistance heaters in the outdoor psychrometric room. The cooling coil was supplied with a water ethylene glycol solution which was cooled using a 150 ton chiller. The chiller was connected to a 1000 gallon storage tank which provided thermal capacity in the chilled water system.

The supply and return air ducts were approximately 16 ft^2 in cross sectional area. All joints of the ducts were taped and caulked to prevent any leakage of air. Figure 3.1 shows a functional representation of the supply and return ducts. The actual ducts were in an over/under configuration with the supply duct above the return duct. The supply air duct was connected to an AMCA nozzle airflow chamber that was capable of flow measurements up to approximately 4000 cfm (AMCA 1985). The flow chamber contained two eight inch nozzles, one five inch nozzle and one three inch nozzle. A damper in the chamber allowed adjustment of flow through the chamber.

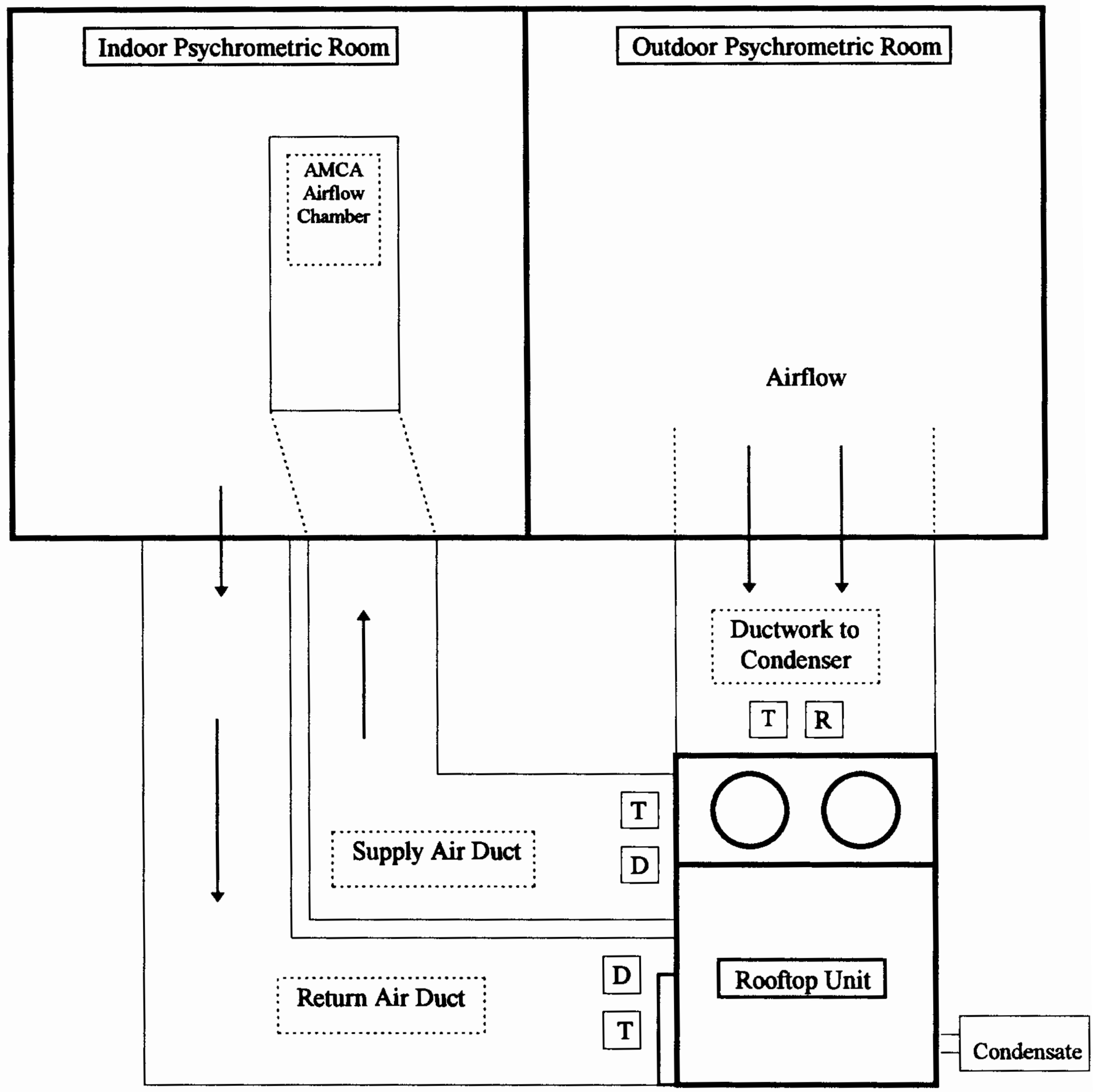

Figure 3.1 - Schematic of the rooftop unit and the psychrometric facilities.

The dry bulb temperature of both the return and supply air were measured with thermocouple grids located immediately before the evaporator (return) and after the supply outlet of the rooftop unit. The temperature measurements are indicated in Figure 3.1 with a boxed "T". The dew points of the return and supply air were measured with chilled mirror dewpoint sensors (indicated with a boxed "D" in Figure 3.1). The sensors were located in aspirated sampling systems which pulled air from the cross section of the airstream.

Temperature in the indoor room was controlled with chilled water coils. Humidity in the room was controlled with steam humidification and a dehumidification coil. Steam from an electric boiler was fed into the airstream in the indoor room to raise humidity. The dehumidification coil received water from the chiller and were used to lower the humidity when necessary.

Besides the air temperatures and humidity measurements indicated in Figure 3.1, refrigerant temperatures and pressures were measured at key locations within each of the three refrigeration circuits. These included: (1) compressor discharge, (2) compressor suction, and (3) TXV inlet. The total power of the unit and power to the two four ton compressors (2 and 3) were monitored with watt transducers. Condensate was also weighed on selected tests to serve as a check on the accuracy of the humidity measurements.

Most of the data collection was handled by a Kaye Instruments Netcom data logger. Table 3.1 lists the data points used in this logger. Because of lack of availability of watt transducers, a Synergistic Controls data logger was used to monitor electrical power for both the evaporator and condenser fans. The channel table for this data logger is shown in Table 3.2.

The Kaye Instruments data logger was attached via serial cable to an IBM compatible personal computer which collected the data at 10 second intervals. The data were stored on the hard drive of the computer. At the conclusion of a test, the data were input into a program which had psychrometric and refrigerant property routines in it. This program was used to calculate airflow, capacity, subcooling, superheat, and average/standard deviation of major variables as well as print out the results of the test in a standard format used at the Energy Systems Laboratory. A sample output from the program is shown in Figure 3.2

Test Procedure

Because this was an experimental prototype air conditioner, there was no past history on the proper amount of charging needed in each circuit. Discussions with several manufacturers indicated that for a new system, there is always a trial-and-error process for getting the charge for each circuit that will produce the best performance in the unit. To complicate the process, each of the TXVs were also adjustable. Thus, both charge and the TXVs could be adjusted. Each of the four ton refrigerant circuits were initially charged with approximately six pounds of refrigerant and the two ton circuit started with approximately four pounds while the system was running. The psychrometric rooms were brought to 95°F outdoor temperature and 80°F indoor dry bulb and 50% indoor relative humidity. Both the charge and thermal expansion valves (TXVs) were then adjusted in an attempt to obtain evaporating temperatures (52°F) in all three circuits.

Table 3.1 - Channel table for the Kaye Netcom data logger.

Channel	Description
0	Evaporator Entering Air Dry bulb Temp (°F)
2	Evaporator Leaving Air Dry bulb Temp (°F)
4	Indoor Nozzle Dry bulb Temperature (°F)
5	Fan Differential Pressure (in. water)
8	Indoor Nozzle Differential Pressure (in. water)
14	Condenser Entering Air Dry bulb Temp (°F)
16	Evaporator Entering Air Dewpoint Temp (°F)
17	Evaporator Leaving Air Dewpoint Temp (°F)
20	Compressor 1 Suction Temperature (°F)
21	Compressor 1 Discharge Temperature (°F)
22	Compressor 2 Suction Temperature (°F)
23	Compressor 2 Discharge Temperature (°F)
24	Compressor 3 Suction Temperature (°F)
25	Compressor 3 Discharge Temperature (°F)
26	TXV 1 Temperature (°F)
27	TXV 2 Temperature (°F)
28	TXV 3 Temperature (°F)
30	Compressor 1 Suction Pressure (psig)
31	Compressor 1 Discharge Pressure (psig)
32	Compressor 2 Suction Pressure (psig)
33	Compressor 2 Discharge Pressure (psig)
34	Compressor 3 Suction Pressure (psig)
35	Compressor 3 Discharge Pressure (psig)
36	TXV 1 Inlet Pressure (psig)
37	TXV 2 Inlet Pressure (psig)
38	TXV 3 Pressure (psig)
40	Total Power (watts)
42	Compressor 2 Power (watts)
43	Compressor 3 Power (watts)

Table 3.2 - Channel table for the Synergistics Data Logger.

Channel	Description
1	Indoor Fan Power (watts)
2	Outdoor Fan Power (watts)

Figure 3.2 - Typical Printout from the program used to reduce data from the tests.

Energy Systems Lab, Texas A&M University - Psychrometric Room A/C Test Data
EPA 10 Ton A/C Unit Performance Tests

TEST DESCRIPTION: 80 F Test/All Compressors
Input File Name: eairall.pra   Barometric Pressure: 29.92 In Hg
Start of Test 01:31:39 04-18-1995   End of Test 01:41:01 04-18-1995

SYSTEM PERFORMANCE DATA	AVERAGE	STD DEV
System EER :	13.33	0.15
Sensible Heat Factor :	0.69	0.01
System Capacity :	121423.60 Btu/hr	1457.73
Total System Power :	9109.75 Watts	8.75
INDOOR COIL CONDITIONS		
AIR PROPERTIES		
Air Side Capacity :	121423.60 Btu/Hr	1457.73
Entering Dry Bulb :	80.39 Deg F	0.25
Entering Dew Point :	61.02 Deg F	0.34
Exiting Dry Bulb :	57.38 Deg F	0.18
Exiting Dew Point :	54.39 Deg F	0.17
Flow Rate :	3231.86 Cfm	12.05
OUTDOOR COIL CONDITIONS		
AIR PROPERTIES		
Air Entering Dry Bulb :	80.09 Deg F	0.10
Air Entering Rel Humidity :	0.00 %RH	0.00
EXPANSION VALVE REFRIGERANT PROPERTIES		
TXV #1		
TXV Inlet Temperature :	91.81 Deg F	13.66
TXV Inlet Pressure :	193.21 Psig	0.80
TXV Inlet -SubC/+SuprH :	-7.26 Deg F	13.61
TXV #2		
TXV Inlet Temperature :	90.77 Deg F	0.48
TXV Inlet Pressure :	199.16 Psig	1.08
TXV Inlet -SubC/+SuprH :	-10.36 Deg F	0.50
TXV #3		
TXV Inlet Temperature :	91.41 Deg F	0.84
TXV Inlet Pressure :	194.74 Psig	1.00
TXV Inlet -SubC/+SuprH :	-8.19 Deg F	0.81
COMPRESSOR PROPERTIES		
COMPRESSOR #1		
Compressor Power :	0.00 Watts	0.00
Suction Pressure :	75.05 Psig	0.91
Suction Temperature :	55.96 Deg F	1.26
Suction Line -SubC/+SuprH :	11.59 Deg F	1.27
Discharge Line Pressure :	196.38 Psig	0.74
Discharge Line Temperature :	154.37 Deg F	0.86
Discharge Line -SubC/+SuprH :	54.20 Deg F	0.90
COMPRESSOR #2		
Compressor Power :	2757.38 Watts	7.92
Suction Pressure :	81.56 Psig	0.85
Suction Temperature :	64.76 Deg F	1.12
Suction Line -SubC/+SuprH :	16.26 Deg F	1.21
Discharge Line Pressure :	206.33 Psig	0.91
Discharge Line Temperature :	165.81 Deg F	0.56
Discharge Line -SubC/+SuprH :	62.26 Deg F	0.62
COMPRESSOR #3		
Compressor Power :	2684.35 Watts	8.33
Suction Pressure :	82.74 Psig	0.72
Suction Temperature :	61.78 Deg F	1.66
Suction Line -SubC/+SuprH :	12.56 Deg F	1.67
Discharge Line Pressure :	204.06 Psig	0.80
Discharge Line Temperature :	159.88 Deg F	0.88
Discharge Line -SubC/+SuprH :	57.10 Deg F	0.88

After approximately two days of attempting to obtain these conditions, it was evident that there might be some circuiting or airflow problems with the evaporator. The evaporating temperature for the first circuit was consistently near 45 to 46°F (76 to 78 psig saturation pressure). The circuits for the two four ton compressors typically had evaporating temperatures 1 to 3°F lower than the design value of 52°F. With these problems, a decision was made to adjust the charge and TXVs until each refrigerant circuit had approximately 8 to 10°F of subcooling at the above conditions and approximately 10 to 12 °F superheat. Once the charge was adjusted to produce these conditions, tests were then completed on the unit.

Tests were then run at the conditions specified in the Air Conditioning and Refrigeration Institutes (ARI) test procedure (ARI 1989). Tests were run at the full load test conditions (95°F outdoor temperature) for 100% capacity and at the part load test conditions (80°F outdoor temperature) for 20, 40, 60, and 80% nominal capacity steps.

Because of the large number of tests run on the unit (over 25), the lengths of the test were abbreviated to 8 to 12 minutes of data collection. Once data were taken for a tests, the required changes in room conditions, or changes in fan speed and compressor on/off made in the rooftop unit and conditions allowed to come to steady state. This typically took anywhere from 30 minutes to two hours, depending on the change. Data were then taken and reduced.

Once the data were reduced, the measured capacity of the system was compared to that predicted from the compressor maps given the measured evaporating and condensing conditions. The maps were provided by the compressor manufacturer. For all the tests taken, the measured capacity averaged 3.9% below the capacity predicted by the compressor maps. The capacity, power, and EER reported in the next chapter are all based on the measured values on the indoor air side of the system.

CHAPTER 4

TEST RESULTS

This chapter is divided into three sections. The first section describes the baseline results with the initial recommended condenser and evaporator fan speeds. The second section discusses the results of the airflow optimization and its impact on overall performance. The last section discusses some of the problems with the unit and their impact on the overall performance.

Baseline Performance

The baseline operating frequencies for the five nominal capacity steps are shown in Table 4.1. These frequencies were recommended by EPA personnel as an initial starting point based on the initial design considerations. This table also shows the fan power for the evaporator and condenser fans at each design capacity step: 2, 4, 6, 8, and 10 tons. For nominal full capacity (10 tons), the fan operated at 60 Hz, and produced approximately 3200 cfm. Figure 4.1 shows a comparison of the total design fan power (condenser plus evaporator) and measured fan power for operation at 30, 45, and 60 Hz. While there were some small differences in fan power between the individual fans--typically, the measured evaporator fan power was higher than the design (by as much as 100 watts at 60 Hz) and the measured condenser fan power was slightly less than the design (by as much as 40 watts at 60 Hz). However, the combined (evaporator and condenser) measured fan power was within 2% of the design at the capacity steps shown in Figure 4.1.

The low evaporator fan power in Table 4.1 indicates what can be achieved with a high efficiency fan and fan motor. The measured power also includes losses through the inverter. One way to characterize fan power is to divide the power (in watts) by the airflow (in 1000 cfm). For the 60 Hz operation, this would be 184 watts/1000 cfm for a 0.63 inch static pressure loss. This value is approximately one half the value (365 watts/1000 cfm) required in the ARI standard test procedure for rating air conditioners that don't have an evaporator fan(ARI 1989). A high efficiency unit (EER rating of 11) that was recently released by a major U.S. manufacturer used 388 watts/1000 cfm. Thus, the fan power used in this unit represents a major improvement over some of the best units on the market.

The ARI standard required a steady state full load (10 ton) test at 95°F and part load tests for each capacity step at an outdoor temperature of 80°F (ARI 1989). Tables 4.2 and 4.3 summarize the full load performance of the unit at 95°F and part load performance at 80°F, respectively. At 95°F outdoor temperature, the unit produced an EER of 11.15 Btu/Wh. The best part load performance for the baseline fan speeds was at 4 tons nominal design capacity with an EER of 14.26 Btu/Wh. The integrated part load value (IPLV) for the baseline was calculated to be 13.39 Btu/Wh.

Table 4.1 - Evaporator and condenser fan speeds for baseline tests.

Nominal Capacity (tons)	Evaporator Fan Speed (Hz)	Evaporator Fan Power (kW)	Condenser Fan Speed (Hz)	Condenser Fan Power (kW)
2	22	0.15	22	0.18
4	33	0.23	33	0.33
6	43	0.33	43	0.57
8	52	0.42	52	0.89
10	60	0.59	60	1.28

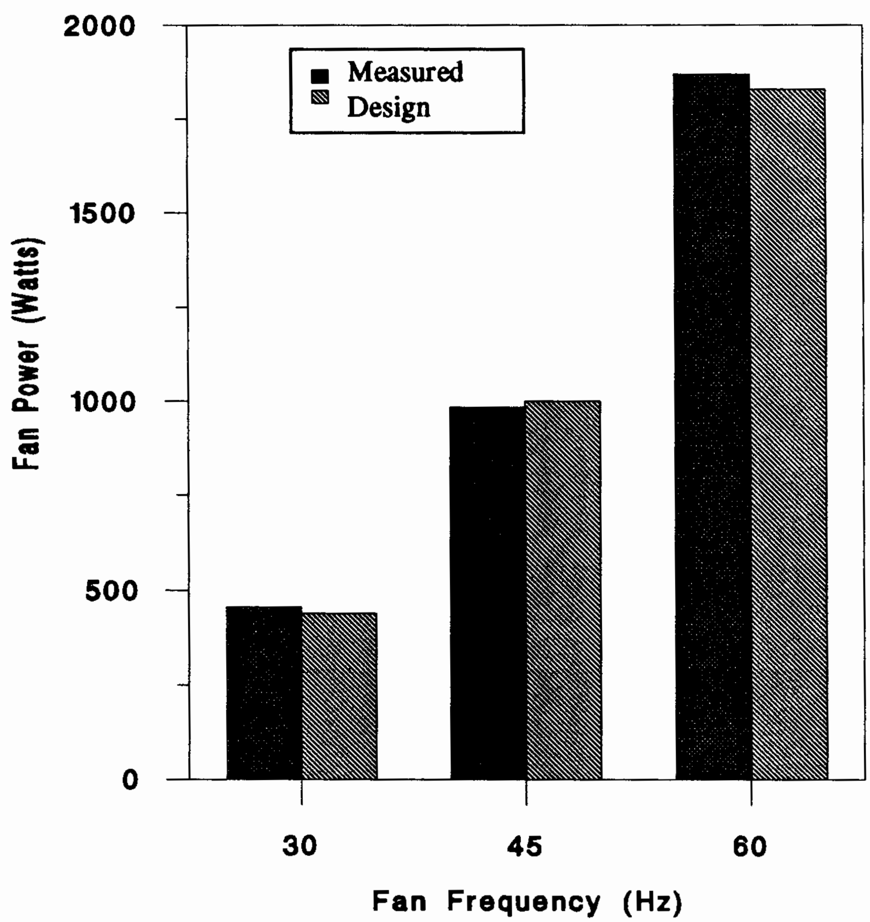

Figure 4.1 - Comparison of total (condenser and evaporator) power at different inverter drive frequencies.

Table 4.2 - Full load baseline performance of the unit for outdoor temperature of 95°F.

Performance Variable	Value
Capacity (Btu/h)	112,630
Power (W)	10,105
EER (Btu/Wh)	11.15

Table 4.3 - Summary of part-load baseline tests for 80°F outdoor temperature.

Nominal Capacity (tons)	Measured Capacity (Btu/h)	Total Power (W)	Energy Eff. Ratio (Btu/Wh)
2	23,205	2,061	11.26
4	48,129	3,374	14.26
6	70,763	5,210	13.58
8	94,437	6,902	13.68
10	121,423	9,109	13.33

The results with the unit were below the design values. For example, the steady state EER (11.15) at 95°F was 13.6% below the design value (12.9). The IPLV was 24.7% below the design value of 17.7 Btu/Wh. However, this unit is better than the best 10 ton rooftop on the market which had a 11 Btu/Wh EER and 11.7 Btu/Wh IPLV.

Airflow Optimization

It was not clear that the airflows of the evaporator and condenser produced optimum performance of the air conditioner. Therefore, it was decided to try to vary airflow through both the condenser and evaporator to determine if higher EERs could be achieved from the unit, both at full load and part load conditions. The airflow was varied by changing the frequency of the variable speed controllers for each of the fan motors. For each change in frequency, data were taken to determine the change in capacity, power, and efficiency. Over 25 tests were run at different fan speeds (driving frequencies) to determine optimum EER for the four of the five nominal capacity steps: 2, 4, 8 and 10 tons. Table 4.4 lists the complete lists of tests, including the baselines. Additional tests beyond these were not conducted because of time and resource constraints.

Figures 4.2 and 4.3 show the results on EER of varying both condenser and evaporator airflow for the nominal 4 ton capacity step with compressor 2 operating. Compressor 2 was used rather than compressor 3 because it produced higher baseline EERs (14.26 versus 13.46 Btu/Wh) than compressor 3. The optimization started with both the condenser and evaporator fan speeds running at 33 Hz. The condenser fan speed was then increased while holding the evaporator fan speed constant. The performance of the unit was evaluated at condenser fan input frequencies of 33, 38, 43, and 48 Hz. As shown in Figure 4.2, the optimum EER was at a condenser fan speed of 38 Hz. Once the optimum condenser fan speed was found, the condenser fan speed was then set at the optimum value (38 Hz) and the evaporator fan speed varied. The optimum evaporator fan motor frequency was 33 Hz (Figure 4.3). The drop in EER was more noticeable for changes in condenser fan speed than evaporator fan speed. A change in condenser fan speed from 38 to 33 Hz produced a 7% in EER compared to less than a 1% change in EER for the same change in evaporator fan speed.

Table 4.4 - Fan speed optimization runs.

Nominal Capacity (Tons)	Outdoor Temperature (Deg. F)	Compressor Used	Condenser Fan Speed (Hz)	Evaporator Fan Speed (Hz)
2	80	1	22	15
2	80	1	22	22
2	80	1	22	26
4	80	2	38	43
4	80	2	38	38
4	80	2	43	33
4	80	2	38	33
4	80	2	33	33
4	80	2	48	33
4	80	3	33	33
6	80	1&2	43	43
8	80	2&3	44	52
8	80	2&3	48	52
8	80	2&3	48	52
8	80	2&3	52	52
8	80	2&3	56	52
8	80	2&3	48	56
8	80	2&3	48	48
10	80	1,2&3	60	66
10	80	1,2&3	60	60
10	80	1,2&3	66	66
10	80	1,2&3	60	66
10	80	1,2&3	56	66
10	95	1,2&3	60	66
10	95	1,2&3	60	60
10	95	1,2&3	60	56
10	95	1,2&3	56	66

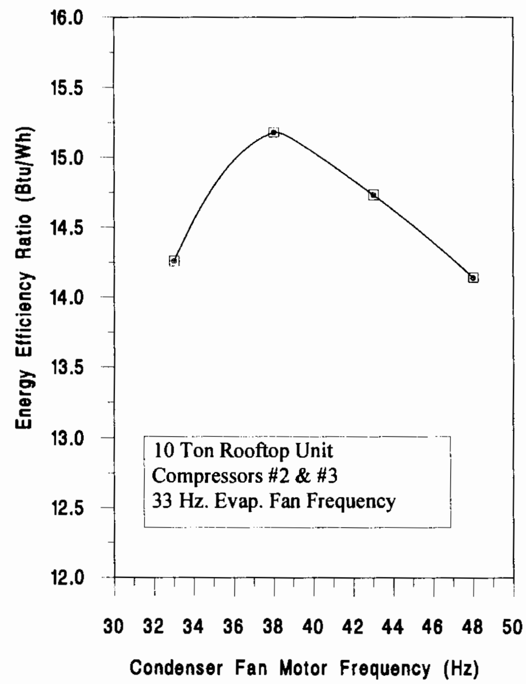

Figure 4.2 - Impact of condenser fan motor frequency on the EER of the unit with compressor #2 (4 ton) running.

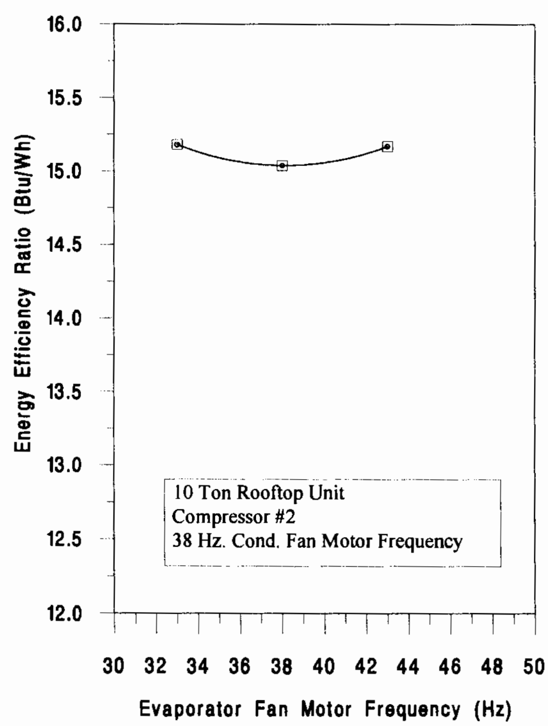

Figure 4.3 - Impact of evaporator fan motor frequency on the EER of the unit with compressor #2 (4 ton) running.

Figures 4.4 and 4.5 show the airflow optimization results for the nominal 8 ton capacity step. Compressors 2 and 3 were used for these tests. For the condenser fan, the best operating frequency was 48 Hz. The EER increased from 13.68 Btu/Wh in the base case (52 Hz) to 13.95 Btu/Wh in the optimized case (Figure 4.4). Because the peak EER occurred at a condenser fan motor frequency of 48 Hz, this speed was fixed and the frequency for the evaporator fan motor varied. The EER decreased as the frequency was increased to 56 Hz or decreased to 48 Hz. Thus, the optimum was at 52 Hz.

Table 4.5 summarizes the optimum EERs and fan speeds that achieved the optimum EERs for each nominal capacity step. Figure 4.6 compares the optimum EERs at part load to the baseline EERs at part load. The biggest improvement with optimization occurred at the 4 ton part load condition, changing from 14.26 to 15.18 Btu/Wh. Overall, the IPLV increased from 13.39 to 13.80 Btu/Wh. At the full load test (95°F outdoor temperature), the capacity increased from 112,630 to 117,317 Btu/hr and the EER increased from 11.15 to 11.51 Btu/Wh. The optimum condenser and evaporator fan speeds were at 60 and 66 Hz, respectively.

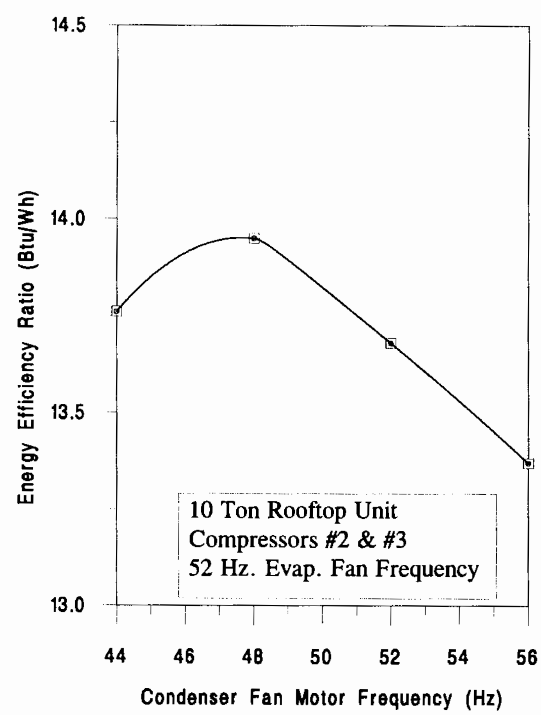

Figure 4.4 - Impact of condenser fan motor frequency on the EER of the unit with compressors #2 and #3 running.

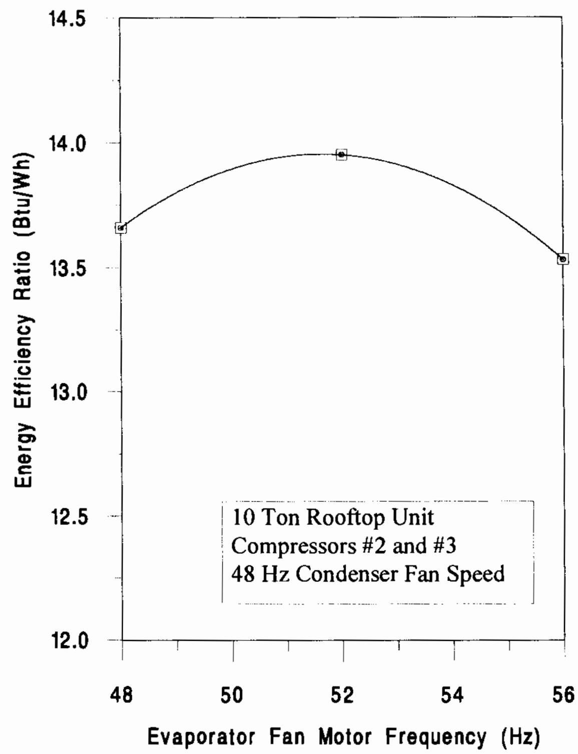

Figure 4.5 - Impact of evaporator fan motor frequency on the EER of the unit with compressors #2 and #3 running.

Table 4.5 - Summary of part-load optimum tests for 80°F outdoor temperature.

Nominal Capacity (tons)	Condenser Fan Speed (Hz)	Evaporator Fan Speed (Hz)	Measured Capacity (Btu/h)	Total Power (W)	Energy Eff. Ratio (Btu/Wh)
2	22	26	25,563	2,115	12.09
4	38	33	52,586	3,283	15.18
6	43	43	70,763	5,210	13.58
8	48	52	94,747	6,790	13.95
10	60	66	127,084	9,216	13.79

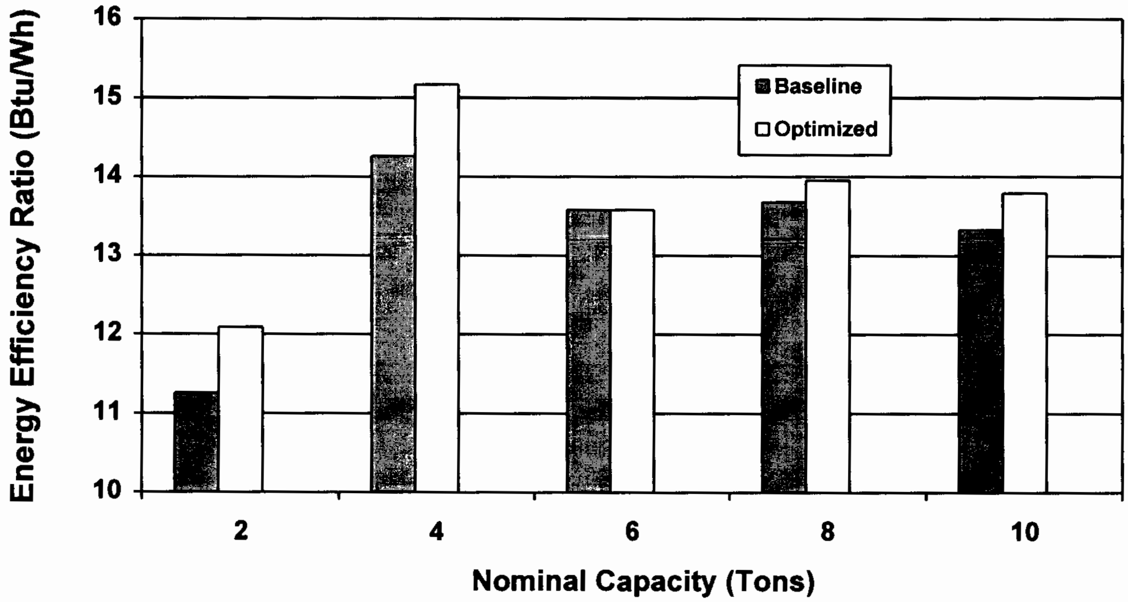

Figure 4.6 - Comparison of baseline and optimized EERs for part load conditions.

Analysis of Performance Shortfall

This unit was designed to have an EER of 12.9 Btu/Wh and an IPLV of 17.7 Btu/Wh. The best test results for the optimized airflow cases were an EER of 11.56 Btu/Wh and an IPLV of 13.8 Btu/Wh. A natural question occurs as to why the measured performance was not as good as the designed. Several potential problems were identified during the testing that could help explain why the unit did not perform as designed. These include: (1) inadequate evaporator surface area for compressor 1 (the two ton unit), (2) potential poor airflow distribution through the evaporator, (3) larger power draw in compressor 2 (four ton) than expected, (4) heat transfer and air flow leakage, and (5) higher than expected evaporator fan power usage.

The design evaporating refrigerant temperatures for each of the evaporator circuits was 52°F, which corresponds to 87 psig for R-22. Both four ton compressors (2 and 3) usually provided evaporating temperatures within 1 to 3°F of this value. However, the evaporating temperatures in the circuits for compressor 1 were consistently down near 45 to 46°F and in some instances as low as 42°F. Ideally, if there were either enough heat exchanger surface area or high enough heat transfer within the refrigerant circuits for this compressor, its evaporating temperatures would have been nearer the design value. To estimate the impact of the reduced evaporating temperature on the performance of the system, the change in capacity and power of the compressor was estimated for the higher evaporating temperature using the compressor maps. The additional capacity and power was added to the measured value. The results are shown in Table 4.6. This one change should have raised the EER from the optimized baseline of 11.51 to 11.81 Btu/Wh. In addition, the capacity would have increased by over 3,400 Btu/h tons. Thus, the circuiting in the section of the evaporator for compressor #1 did have a measurable impact on the performance of this unit.

Table 4.6 - Impact of evaporator circuiting for compressor #1 on system performance.

Run	Capacity (Btu/h)	Power (W)	EER (Btu/Wh)
Optimized Baseline (95°F)	117,317	10,193	11.51
Compressor #1 Evaporator Circuit Operating at Design Temperature (52°F)	120,736	10,227	11.81

Another potential problem that could have contributed to the low evaporating temperature in the evaporator circuit for compressor 1 as well as the circuits for the other two compressors was possible poor airflow distribution through the evaporator. The unit was designed around a Trane 15 ton chassis. The original Trane unit had an evaporator blower with smaller dimensions than the blower used for the high efficiency 10 ton. Because the high efficiency fan was larger, portions of the fan housing extended to within approximately two inches of the evaporator. Potentially, this could have caused poor airflow distribution in the coil. However, because of the confined space around the evaporator, no measurements of airflow distribution were made to determine if this was the problem that explained why the evaporating temperatures for the circuits for compressors 2 and 3 were slightly lower than designed. A potential way to model poor air distribution would be to estimate what system performance would have been if evaporating temperatures were at 52°F for the evaporator circuits for these two compressors as well as compressor 1. As before, this can be done using the compressor maps. With a 52°F evaporating temperature for all evaporating circuits, the capacity would have risen to 121,960 Btu/h, the power increased to 10240 watts, and the EER increased to 11.91 Btu/Wh. Thus, this represents an increase in EER of 0.1 Btu/Wh over the previous improvement related to compressor 1 circuits.

The power for both of the larger two compressors (2 and 3) were both measured separately from the total power of the system. For the full load optimized case at 95°F, the measured power for compressor 2 was 145 watts higher than the manufacturer's compressor maps and the power for compressor 3 was 75 watts higher. Both compressors performed within ±5% of the manufacturer's specifications, but they were both on high side of the power rating. The total (220 watts) represented an unexpected increase in total system power of approximately 2% over those values predicted by the compressor maps. If these compressors had performed at their rated powers, the total system power would have been 9,975 kW and the system EER would have been 11.76 Btu/Wh rather than 11.51 Btu/Wh. When combined with good circuiting in compressor 1 circuit and good air distribution through the evaporator, the EER would have increased to 12.17 Btu/Wh.

Two related problems that were noticed when the system was first being tested were heat loss from the outer shell of the air conditioner and air leakage from the evaporator to condenser sections. While the unit was insulated, there were numerous locations on the outer shell than "sweated" during some of the tests. There were some internal air leaks through portions of the frame that allowed cold air from the discharge of the fan to come into contact with the sheet metal on the cabinet. A number of these leaks were located and sealed before testing began. However, there were some that were not because they could not be reached without disassembling significant portions of the unit. It was felt that such an endeavor could have created more problems and so were not attempted.

Another problem with the unit was internal air leakage from the evaporator to the condenser section. At one point early in the testing, the condenser fan was shut off and tissue paper placed on the discharge of one of the condenser fans. The tissue paper "fluttered" because of the air leaking from the evaporator to condenser sections. Caulk and duct tape were used to seal as many cracks as possible. The types of heat and airflow losses detailed above were difficult to measure or quantify. A 1% leakage rate from the evaporator for 60 Hz operation of the evaporator blower would be 32 cfm. If this air leaks into the condenser side, this creates a pathway around the condenser. Thus, there would be less air going through the condenser, which would raise the condenser refrigerant pressure slightly and potentially increase the power to the compressor. While this power increase would be expected to be small, it would be coupled with a loss from the measured capacity on the evaporator side of the unit. For the tests, a leakage or heat transfer loss would show up in a discrepancy between the air side capacity measurement and the refrigerant side capacity predicted by the compressor maps. For all the tests conducted, the air side capacity measurement averaged 3.9% less than the refrigerant side measurement predicted by the compressor maps. While this difference is within the acceptable limits allowed by the ARI test procedure, they point to the possibility that heat and airflow losses contributed to the lower than expected performance of the unit. If these losses were in the range of as little as 2 to 4% of the total capacity, they would account for a drop of EER of 0.24 to 0.48 Btu/Wh.

Another difference between the measured performance of the unit compared to its design performance was the evaporator fan/motor. At full load (60 Hz), the evaporator fan was expected to consume 490 watts. The measured fan power was 590 watts. This difference in 100 watts was offset by the lower power (60 watts) from the condenser fans. However, this 100 watts also contributed an additional 341 Btu/h of heat energy to the airstream going through the evaporator. This additional heat energy would have reduced the net capacity of the unit by 341 Btu/h. If this additional evaporator fan power had not been present, the capacity would have been 0.3% larger and the power would have been 1%

smaller. The net result on the EER would have been an increase of 1.3%, or a value of 11.66 Btu/Wh rather than 11.51 Btu/Wh.

The total impact of the above losses/problems in the performance of the unit would translate into a steady state EER of between 12.56 to 12.81 Btu/Wh, which would have been very close to the design EER of 12.9 Btu/Wh. An evaluation of the impact of some of these losses on the part load performance is shown in Figure 4.5. The optimized baseline was plotted along with the estimated performance of the unit if the losses were eliminated associated with the circuiting and airflow distribution in the evaporator and the measured power for compressors 2 and 3 equaled their rated values. Thus, the "reduced losses" case represents what the unit could have produced if these losses had not been present in the unit. This "reduced losses" case does not include the heat and airflow leakage or elimination of the higher evaporator fan power usage. The biggest increases in the performance were at the 2, 6, and 8 ton capacities. For the 2 ton case, the performance increase was attributed solely to what would have occurred with improved circuiting with compressor # 1. The six ton capacity step, the increase was both due to improvement in refrigerant circuiting (and airflow distribution) in the evaporator for compressors #1 and #2 to achieve a 52°F evaporating temperature and the power reduction if the actual compressor power had been that reported in the compressor maps. For the eight ton case, the improvement would The IPLV for the "reduced losses" case was calculated to be 15.6 Btu/Wh, which represents a 13% improvement over the IPLV of the optimized case of 13.8 Btu/Wh.

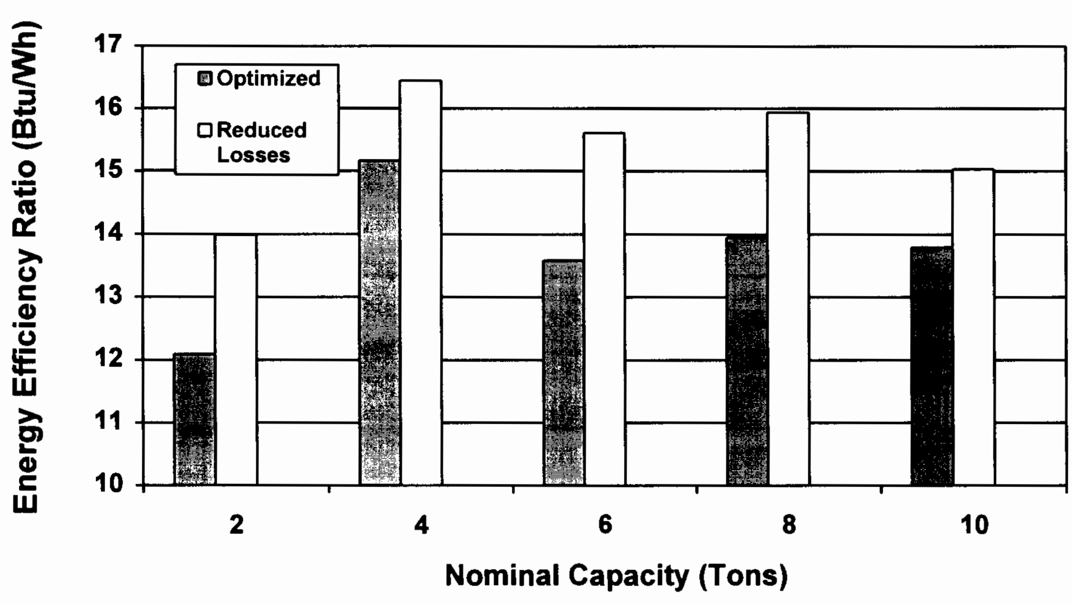

Figure 4.5 - Increase in Energy Efficiency Ratio with elimination of some of the identified problems and losses.

Summary

For the baseline case, the unit was better in steady state efficiency (EER) and in IPLV than the best 10 ton rooftop unit currently on the market. However, the measured baseline EER (11.15 Btu/Wh) and IPLV (13.39 Btu/Wh) were 13.6% and 24.7% below their respective design values. The airflow through both the condenser and evaporator was varied to see if the performance could be improved. Optimum fan speeds were found for the nominal 2, 4, 8, and 10 ton capacity steps. The EER improved to 11.51 Btu/Wh and the IPLV increased to 13.80 Btu/Wh. Close inspection of the unit and the data collected indicated that there was a combination of at least five different factors that were responsible for the measured performance being less than the design. Each of the five items in isolation would have had a minor impact on performance (typically 1 to 3% in EER). However, in combination, they produced the 10.7% measured shortfall in EER for the optimized case.

CHAPTER 5

ECONOMIC EVALUATION OF THE PROTOTYPE UNIT

One important aspect of this project was the evaluation of the costs to purchase the high efficiency unit and the potential operational economics of the unit. This chapter is divided into four sections. The first section discusses the originial cost estimates of both the baseline unit and the costs of the high efficiency unit. The second section describes the procedure used to acquire manufacturing cost of the high efficiency design data and the results of the costing survey. The third section describes some of the comments on the costing data by four major manufacturers of packaged rooftop air-conditioners and provides revised cost data which reflects manufacturers' comments. The last section provides an evaluation of the economics of purchasing and operating the high efficiency rooftop unit.

Original Cost Estimates

When the high efficiency rooftop unit was originally designed, D Hibberd provided cost estimates of both the base case and high efficiency systems (Hibberd 1993). A summary of these cost estimates are provided in Table 5.1. The original cost estimates were developed by first starting with the base chassis of a 15 ton packaged unit manufactured by Trane. The base case cost was determined by phone conversations with the Trane supplier in Denver, Colorado. D. Hibberd identified the major system components for pricing purposes as the base unit, the compressors, the supply fan and motor, the condenser fans and motors, the evaporator and condenser coils, and the control system.

Table 5.1 - Baseline cost data as reported by D. Hibberd (1993).

Component	Cost	Data Source
Base Case 10 ton unit, 8.9 EER	$4,739	Trane, Denver
Base Case 15 ton unit, 8.9 EER	$6,130	Trane, Denver
Standard Trane Compressors	($1,030)	Trane, Denver
Bristol Compressors	$796	Bristol Compressors
High Eff. Supply Fans and Motors	$300	Marginal Cost, Twin City Fan & Blower Co.
Supply Fan VFD	$125	Estimated, Source Not Listed
High Eff. Condenser Fans and Motors	$90	Condenser Coil
Condenser Fans VFD	$125	Estimated, Source Not Listed
Evaporator Coil	$130	Marginal Cost Based on Weight Increase of Coil @ $4/lb
Condenser Coil	($58)	Marginal Cost Based on Weight Increase of Coil @ $4/lb
Controls	$250	Estimated, Source Not Listed
Total Unit Cost	$6,858	
Total Marginal Cost	$2,119	

Verification Of Cost Estimates

D. Hibberd Consulting submitted the cost estimates to several of the air-conditioner manufacturers for their review and comments. None of the manufacturer's were willing to substantiate any of the costing data. In support of Hibberd's efforts, additional data were gathered from manufacturers of the major system components. Hibberd's costing data were focused on the final price to the purchaser of the eqiupment. The effort here attempted to identify manufacturing costs. Once these were estimated, the costs were sent to four major rooftop manufacturers (Lennox, Trane, Carrier, and York) for comment.

Several manufacturers of components (fans, compressors, coils, motors, and controls) were contacted by phone and asked to submit an OEM cost estimates (typically for at least a quantity of 1000) for their product. The estimates varied widely depending on several factors such as the expected annual quantities, various design parameters relevant to the component of interest, and when the manufactured unit was expected to go into production. Some of the component manufacturers were reluctant to help with cost estimates when they were told of the nature the bid request.

Data for the compressors were obtained directly from Bristol Manufacturing. Similar quotes on 5 ton compressors were obtained from Trane. The estimated incremental OEM cost of replacing the two five ton compressors in the 10 ton baseline system with the three high efficiency Bristol compressors was initailly estimated at $75.

The incremental cost of the evaporator and condenser coils was estimated to be $75 and $150, respectively. The data were obtained by phone conversations with several heat exchanger manufacturers including Heat Craft and CP/AAON. Each of the major air-conditioner manufacturers were surveyed for the cost of replacement coils in an effort to get a better estimate of the baseline cost.

Several manufacturers of fans were surveyed for the cost of the evaporator and condenser fans and motor assemblies. The incremental OEM costs of the evaporator fan assembly was estimated to be $100. The incremental cost of the condenser fan assembly was estimated to be $160. Fan cost estimates were submitted by multiple manufacturers including Twin City Fan & Blower Co., Airfoil Impellers (Bryan, Texas), and W.L. Lashley & Associates (Houston, Texas). The four major air-conditioner manufacturers (Lennox, Trane, Carrier, and York) were also surveyed for the replacement fan costs in an effort to determine the baseline fan costs.

Several manufacturers of digital controllers, single board computers, and variable frequency drives (VFD) were surveyed for the cost of the controller, and the condenser and evaporator fan VFD's. Costs for the VFDs ranged from a low of $50 to $300. One manufacturer indicated that in large quantities, the VFDs could cost as little $50 each. The control system was initally estimated to cost $125. Table 5.2 summarizes the estimated OEM (manufacturer) incremental costs for the compressors, fans, coils, and controls.

Table 5.2 - Initial estimate manufacturer's incremental costs to improve 10 ton unit from an EER of 9.0 to 12.9 Btu/Wh.

Component	Cost
Compressors	$75
High Eff. Supply Fans and Motors	$100
Supply Fan VFD	$150
High Eff. Condenser Fans and Motors	$160
Condenser Fans VFD	$300
Evaporator Coil	$75
Condenser Coil	$150
Controls	$125
Total Incremental Cost	$1,135

Manufacturers' Review of Costing Data

After the estimated costs were updated, a letter was written to the air-conditioner manufacturers detailing the updated costing data found in Table 5.2. The manufacturers were asked to review the data for reasonableness and respond with comments as appropriate. The letter was sent to Trane, Carrier, York, and Lennox. There was no written response from Carrier, York, and Lennox. Trane was the only company to respond, but their letter made no attempt to comment on the accuracy of the cost data.

After waiting for a response for two months, it was decided that each company that received the letter should be called. Each company declined to make any "official" comment on the cost data. After it was established that the manufacturers had no official comments to make, the engineers from the companies indicated that they were willing to make some unofficial comments on the data. None of the engineers were willing to share any of their company's cost data. However, they were willing to comment on whether the data were "reasonable", "high", "low", or not within their area of expertise. Table 5.3 below is a summary of the comments about the updated cost data.

Table 5.3 - Manufacturers' review of the estimated incremental manufacturer's cost for building the high efficiency rooftop air conditioner.

Component	Cost	Comments
Compressors	$75	Too Low, Should About $150
High Eff. Supply Fans and Motors	$100	Reasonable
Supply Fan VFD	$150	No Comment, No Experience With VFD's
High Eff. Condenser Fans and Motors	$160	Reasonable
Condenser Fans VFD	$300	No Comment, No Experience With VFD's
Evaporator Coil	$75	Too High
Condenser Coil	$150	Too High
Controls	$125	Too High

Only one of the estimates was judged to be too low and that was the compressor costs. Feedback from the rooftop manufacturers indicated that the estimated costs were probably one-half the costs that were considered "reasonable".

There were several estimated incremental costs that were judged to be too high. These included the coils and the controls. Every air-conditioner manufacturer contacted by phone indicated that coil costs were "very adequate" or that "you can buy a lot of coil" with the reported data. Thus, these data were considered high by the manufacturers. The data presented to the rooftop manufacturers were from several independent coil manufacturers. Potentially, the rooftop manufacturers may be able to set up and produce large quantities of the same coil more cost effectively than the coil manufacturers who were contacted. Even though the coil manufacturers were asked to give OEM estimates for the coils, the coil manufacturers often would ask for additional manufacturing and design specifications for which assumptions they had to make. This may have contributed to the cost estimates being too high because the coil manufacturer included enough markup to cover any contingencies. After discussion with both the rooftop and coil manufacturers, a reduction in the estimated costs by 25% was felt to be a more realistic number. Thus, the incremental manufacturing costs for the evaporator and condenser would costs $56 and $113, respectively. The controls costs were also deemed too high. Further discussions with controls manufacturers yielded a better estimate of approximately $50.

None of the rooftop manufactueres had any comments on the VFD costs. Initially a value of $150 was used for the evaporator and $300 for the codenser VFDs to solicit a response from the rooftop manufacturers. Because there were no comments and because a major VFD manufacturer stated they could produce these for approximately $50 in quantities of a 1000, these estimated costs were reduced to $50.

Part of the reasons why some of the estimated costs were high was due to how the component manufacturers perceived how the data were going to be used. For example, if a manufacturer was told these were "budget" estimates, then the cost would be higher than when the same manufacturer was asked for OEM pricing. When asked for OEM pricing, the cost varied depending on the annual quantity that the consumer was willing to purchase each year.

After dealing with the component manufacturers, it was concluded that the actual pricing would probably be lower than what has been indicated in the tables above but that the final manufactured price can only accurately be determined by a manufacturer actual price cannot be determined until the new design is to actually be put into production.

The costs above were for manufacturing the unit and not the final price to the purchaser of the equipment. The final markup from the factory to the purchaser can be expected to vary by a factor of two to three, depending on the technology, the number of units purchased and the manufacturer. For example, calls to one rooftop dealer yielded costs for the purchase of a 10 ton, 9 Btu/Wh EER rooftop air conditioner of approximately $3100 when purchased in quantities of three or more. The single unit price would have been over 20% more. Thus, the actual markup per unit would not be constant, even though the costs to manufacturer the units would be the same since they were the same model coming off the same assembly line.

With the manfacturer's costs, a three fold markup was used to estimate the final price to the purchaser of the unit (Table 5.4). This probably would represent the price for the purchase of one unit. A factor of two would probably be more representative of a purchase of multiple units. The $2,187 incremental price is very close to the $2119 marginal costs estimated by Hibberd (1993). Even though the incremental prices for individual components varied from Hibberd's estimates, the totals were within $100.

Table 5.4 - Final estimates of manufacturers' Incremental Costs and incremental price paid by a purchaser of the high efficiency rooftop unit over a 9.0 Btu/Wh EER system.

Component	Incremental Manufacturer Cost	Incremental Price to Purchaser
Compressors	$150	$ 450
High Eff. Supply Fans and Motors	$100	$ 300
Supply Fan VFD	$50	$ 150
High Eff. Condenser Fans and Motors	$160	$ 480
Condenser Fans VFD	$50	$ 150
Evaporator Coil	$56	$ 168
Condenser Coil	$113	$ 339
Controls	$50	$ 150
TOTAL	$729	$2187

Economics of Owning and Operating a High Efficiency Rooftop

An important question regarding the high efficiency rooftop is the potential payback to a commercial building operator. In order to estimate this, a small two story office building was simulated using the program ASEAM 3.0, which used the bin method to estimate annual energy use (ASEAM 1991).

The building was assumed to be a one-story, rectangular building with a gross floor area of 4,880 square feet. It was divided into five thermal zones: a lobby (480 square feet) and four office zones (three at 1,220 sq. ft. each and one at 740 sq. ft.). All zones were fully conditioned for heating and cooling. The building's operating hours were assumed to be from 8 AM to 6 PM on weekdays, and from 8 AM to 12 noon on Saturdays. The building was assumed unoccupied on Sunday.

The U-values (all units in Btu/(hr-ft^2-°F)) of important thermal envelope components in the shell of the building include: Roof (U=0.05), Walls (U=0.08), and Doors(U=0.73). The windows were reflective, double pane with a U-value of 0.65.

Infiltration was assumed to be 0.50 air changes/hr during occupied hours and 0.30 air changes/hr during unoccupied hours. Infiltration was 0.30 air changes/hr at all times in the two zones on the upper floor of the building. Lighting capacities were 2.0 W/sf for all five zones. The average peak equipment capacity was 2.0 W/sf. There was a maximum occupancy of 18 people in the building. The loads from the people were distributed evenly throughout the zones. Heating setpoint was 72°F and 62°F during occupied and unoccupied hours, respectively. During summer, the cooling setpoint was 72°F and 90°F during occupied and unoccupied hours, respectively.

The design load on the building wsa approximately 120,000 Btu/hr for a 93°F outdoor temperature. Thus, cooling could be provided by a 10 ton packaged rooftop air conditioner. All units (baseline and high efficiency) had capacities normalized to 120,000 Btu/h. The baseline package system had an overall EER of 9.0 Btu/Wh and the evaporator fan consumed 375 W/1000 cfm. At 95°F, this unit drew 13.3 kW of total power.

Two alternative high efficiency systems were considered. The first high efficiency system performed at the level of the test results in Chapater 4. It was designated as HE I. It had an EER of 11.51 Btu/Wh. It's evaporator fan was assumed to use 205 W/1000 cfm and produce 3500 cfm. This unit had a total power draw of 10.4 kW at a 95°F outdoor temperature. The second high efficiency system (designated as HE II) included the corrections (compressors using rated power, evaporator fan using rated power, proper circuiting, etc.) to unit discussed in Chapter 4. These corrections brought it's efficiency up to an estimated EER of 12.5 Btu/Wh. It's evaporator fan would have used 178 W/1000 cfm at full load. It's estimated power draw was 9.6 kW at a 95°F outdoor temperature.

Because of limitations in ASEAM, the rooftop air conditioner had to be modeled as a single zone system. This assumption would potentially underestimate the savings with the high efficiency rooftop air conditioner because it could operate as a variable air volume system.

ASEAM runs were made for five cities: Chicago, Dallas, Los Angeles, Miami, and Washington, D.C. These cities provided a variety of weather conditions for the building, but also provided some significant cooling loads. Tables 5.5 and 5.6 summarize the total cooling energy use for the baseline and the two high efficiency units.

Table 5.5 - Estimated annual energy savings for the HE I unit.

LOCATION	Annual Cooling Energy Use (kWh)		Cooling Energy Savings (kWh)
	Baseline Unit	HE I Unit	
Chicago	12,747	8,796	3,951
Dallas	17,338	12,421	4,917
Los Angeles	12,560	8,837	3,722
Miami	19,542	14,169	5,373
Washington, D.C.	14,509	10,194	4,314

Table 5.6 - Estimated annual energy savings for the HE II unit.

LOCATION	Annual Cooling Energy Use (kWh)		Cooling Energy Savings (kWh)
	Baseline Unit	HE II Unit	
Chicago	12,747	8,141	4,606
Dallas	17,338	11,454	5,884
Los Angeles	12,560	8,163	4,397
Miami	19,542	13,053	6,489
Washington, D.C.	14,509	9,419	5,090

The savings in Tables 5.5 and 5.6 needed to be converted to dollar savings. While there was a wide range in energy and demand costs throughout the country, for this analysis a value of $0.08/kWh was assumed for electrical energy costs. This was slightly higher than the $0.07/kWh used by Hibberd in his 1993 report. Because the high efficiency units would also save peak demand because of their lower power draw, this savings needed to be included. Demand charges for commercial building also ranged widely throughout the country. A value of $10/kW was used here. Some utilities also include a ratchet clause on demand bills that penalize the customer the whole year for using power once during the utility's peak demand time. If the utility's peak demand time was in the summer months, then the peak charge or a large fraction of it would be applied to the monthly bill each month for the next year. Thus, a saving in power for the peak month would provide savings throughout the year. For this analysis, a ratchet of 85% was assumed for 8 months and the full demand savings was applicable the other four months. For HE I, there was a savings of 2.9 kW over the baseline unit. For on-peak months, this would translate into a $29 demand savings compared to the baseline. For off-peak months with the ratchet in effect, the demand savings would be $25. For the year, the savings would be $316. For HE II, the reduction in demand would be 3.7 kW and the savings per year would be $400..

Table 5.7 summarizes the total savings in energy use and demand with the high efficiency rooftops. Theses savings would provide a 2.4 to 2.9 year paybacks in most locations.

A factor that has helped speed the market acceptance of high efficiency residential air conditioners are rebates from the electric utilities. If electric utilities could be persuaded to provide a $100/ton rebate on the high efficiency rooftop air conditioners, they would speed a $1000 on a 10 ton unit to reduce peak demand by as much as 3.7 kW. This works out to be an investment of only $270/kW for the utility, which is significantly below the cost for building new electrical generating capacity. For the commercial building owner, the payback with a rebate would drop to less than 1.6 years in all five locations. Thus, the high efficiency rooftop would not only benefit the building owner by reducing electric bills, but it would also benefit the utility with reductions in peak demand.

Table 5.7 - Annual energy and demand costs savings with the high efficiency units.

LOCATION	Annual Saving (Dollars)	
	HE I Unit	HE II Unit
Chicago	$632	$768
Dallas	$709	$870
Los Angeles	$614	$752
Miami	$746	$919
Washington, D.C.	$661	$807

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

While the air conditioner tested at a lower efficiency than expected, the unit did test out with an EER of 11.51 Btu/Wh and IPLV of 13.8 Btu/Wh. These values were better than any unit on the market. The discrepancies between the test results and design values were explainable by higher than expected power consumption of the fans and compressors, poor refrigerant circuiting for one of the compressors, non-uniform airflow across the evaporator, and internal leakage of air from the evaporator section. While none of these sources of efficiency losses was individually more than a 1 to 3% effect on efficiency, when combined, they produced over a 10% loss in efficiency over the design EER. Improvements beyond the measured efficiency numbers would have required making physical modifications to the system or reworking the evaporator and replacing two compressor(s).

It is recommended that if another unit is built, that it be built and tested by the manufacturer. This recommendation arises from two considerations. The first focuses on the design of the unit and the second from the difficulties that arose during the testing of the unit. While four major manufacturers (Lennox, Trane, Carrier, and York) were given an opportunity to build the unit, none of them bid on the construction of it. Perhaps, some type of incentive, such as the promise of purchase of a number of units could persuade one of the larger manufacturers to get involved with the design and testing of the high efficiency unit. With a clean sheet of paper, a highly efficient 10 ton system could be design from "the ground up". This unit started with an existing 15 ton design and attempted to design the ten ton unit into the existing cabinet. While this approach produced a high efficiency unit, it may not have produced the most cost effective design.

Some of the problems that arose during the initial testing of the unit, such as getting the control system to function, replacing a bad compressor, faulty service valves, etc. could have been fixed faster if the unit had been tested in facilities in close proximity to where the unit was manufactured. Getting a manufacturer involved in the design and testing could also provide for more accurate costing data.

It appears that one of the problems encountered with this unit was the optimistic values of some of the component data used for the design. These optimistic values led to design efficiencies that could not be achieved in the tests. For example, both of the larger compressors used more power than the compressor maps predicted. While these power numbers were within the ±5% range expected with the maps, they were both off in the same direction (more power). It was possible too, that one of the problems with the evaporator is that is did not perform up to its expected value. It is recommended that in future designs, the component manufacturer be required to provide test results on the actual components used in the unit.

An overwhelming success with this unit was the design of the fan systems. This unit demonstrated that a combination of high efficiency fans and motors plus variable speed controllers could be used to provide a substantial drop in the power used by the fans and increase the IPLVs. With the VFDs, it was possible to optimize airflow through both the condenser and evaporator to maximize the efficiency at each of the five capacity steps. This resulted in an IPLV that was almost 20% higher than the EER. By contrast, several of the 10 ton systems on the market whose data were reviewed had IPLVs that varied from 0 to 7% higher than the EER. Correction of some of the problems with this unit and optimization of the 6 ton capacity step could potentially have led to IPLVs above 15 Btu/Wh.

Significant improvements in design EERs and IPLVs beyond this unit will probably be small. Potentially, improvements can be made in compressor, heat exchanger, and motor technologies that could each provide a few percent improvement in EER or IPLV. For example, two compressor manufacturers (Copeland and Bristol) were contacted about what could be expected in compressor technology in the next few years. The compressors in the system were prototypes of Bristol's new line of high efficiency compressors. The compressors used in this unit were significantly better than standard reciprocating compressors. A representative from Bristol claims that gains of about 3% can be expected over the models used in the tests can be expected with their new models. These gains in performance were comparable to those expected with the best scroll compressors. When consulted on the future of compressor technology, the manufacturers stated they do not expect to see "big" improvements in efficiency. They expect most of the development efforts will be in the area of cost containment, so that prices are reduced while maintaining or improving reliability and performance.

There have been a substantial number of improvements in heat exchanger technology over the past two decades with internally finned tubes and different external fin configurations. One technology that has been extensively used in the automotive industry that may be useful for rooftop applications is brazed tube heat exchangers. The brazed tube technology offers several advantages over conventional designs including high heat transfer capacity, lighter weight for a given capacity, reduced refrigerant charge, reduced air-side pressure drop, reduced potential for galvanic corrosion, and improved transient response time. Over the past several years, a number of air conditioning and heat pump systems have been tested using the new heat exchanger technology. The compact nature of the coils is relevant to packaged rooftop units where weight and size are very important design and installation parameters. Using the compact brazed tube heat exchanger technology, it may be possible to replace the evaporator with new coil that would fit into a 10 ton chassis and still outperform the unit tested. When used in the automotive industry, the brazed tube heat exchangers showed a 30% higher capacity over the standard finned tube coils of the same physical dimensions. That were also more durable, lighter in weight, and used less refrigerant.

A potential item that should be addressed with high efficiency rooftop systems is the impact on performance with alternate refrigerants. Rooftop systems generally use R-22.

There is currently underway an extensive effort to find and test suitable replacements to R-22. Much of the effort has been on a ternary zeotropic blend of R-32/R-125/R-134a and a near azeotropic mixture of R-32/R-125(ARI 1994). Both mixtures would require a significant redesign of the rooftop system. For example, with the zeotropic blend, it may be possible to incorporate heat exchangers designs that could take advantage of temperature glide. With the binary blend, the operating pressures would be much higher than with R-22. This could require thicker tube walls, but could allow for smaller diameter tubing in the condenser.

A vital link to the potential successful introduction of high efficiency rooftop air conditioners may be the electric utilities. Their willingness to offer rebates for high efficiency rooftops would have a dramatic effect on the payback of the units to the end-user. Because of the lower power draw offered by the high efficiency rooftop, the utility would directly benefit by reduction in demand, particularly if the high efficiency unit is used to replace an existing low efficiency system.

REFERENCES

AMCA 1985, AMCA/ANSI Standard 210-85, *Laboratory Methods of Testing Fans for Rating*, Air Movement and Control Association, Arlington Heights, IL.

ARI 1989, *Standard for Unitary Air-Conditioning and Air-Source Heat Pump Equipment*, Standard 210/240, Air Conditioning and Refrigeration Institute, Arlington, Virginia.

ARI 1994, *Results From ARI's R-22 Alternative Refrigerant Evaluation Program (AREP)*, Volumes 2 and 3, Air Conditioning and Refrigeration Institute, Arlington, VA., April

ASEAM 1991, *ASEAM 3.0 A Simplified Energy Analysis Method, U.S. Department of Energy*, Washington, D.C.

Boecker 1987, *Design Optimization of Residential-Sized Air-Source Heat Pumps*, M.S. Thesis, Department of Mechanical Engineering, Texas A & M University, College Station, Texas, May.

EPA 1993, Meeting with representatives from four major rooftop manufacturers, EPA, Texas A & M University, and California Institute for Energy Efficiency, Dallas-Forth Worth Airport, Texas, December 8.

Hibberd 1993, *Final Design Specification for Prototype High Efficiency 10 Ton Rooftop Package Unit*, D. Hibberd Consulting, Denver, Colorado, October 18.

McQuiston, F. C., and Parker, J. D. 1994, *Heating, Ventilating, and Air Conditioning Analysis and Design*, Fourth Edition, Wiley & Sons, Inc., New York.

NEPA 1992, "Conference Report on H.R. 776, Comprehensive National Energy Policy Act", *Congressional Record-House*, H12056-H12169, Washington, D.C., October 5.

Rice, C.K. 1992, "Benchmark Performance Analysis of an ECM-Modulated Air-to-Air Heat Pump with a Reciprocating Compressor", *ASHRAE Transactions: Research*, Vol. 98, Part 1, June.