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AN EVALUATION OF IMPROPER REFRIGERANT CHARGE ON THE PERFORMANCE OF A SPLIT SYSTEM AIR CONDITIONER WITH A THERMAL EXPANSION VALVE

FINAL REPORT

Submitted by

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ABSTRACT

The effect of the improper charging on steady state and cyclic performance (capacity, EER, power consumption, SEER, and coefficient of degradation) of a residential air conditioner which utilized a thermal expansion valve was investigated. This study was the continuation of ESL/CON/88-1 performed by Mohsen Farzad and Dennis O'Neal. A fully charged condition was established as a base case. The full charge was obtained by charging the unit to the subcooling specified by the manufacturer for a specific indoor and outdoor temperatures. Once the full charge was determined, the unit was subjected to 40%, 30%, 20%, 15%, 10%, and 5% undercharging and 5%, 10%, 15%, and 20% overcharging of refrigerant by mass. The fully charged tests were compared to those for under and overcharging. The performance of the unit was evaluated as a function of charge as well as at four outdoor room temperatures (82^{o}F, 90^{o}F, 95^{o}F, and 100^{o}F). As the outdoor temperature increased, the total capacity and EER dropped. The maximum total capacity, EER, and SEER were found at 10% undercharging (126 oz). The capacity and efficiency (EER and SEER) of the unit were found to be less sensitive to under/overcharging than the unit with a capillary tube expansion previously studied. Other data such as

refrigerant flow rate, sensible heat ratio, superheat and subcooling were also presented.

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GLOSSARY OF TERMS

AMCA	Air Movement and Control Association
ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating and Air conditioning Engineers
ASME	American Society of Mechanical Engineers
btu	British thermal unit
cfm	Cubic feet per minute
COP	Coefficient of performance
DB	Dry bulb
DP	Dew point
FC	Full charge
fpm	Feet per minute
lbma	Pounds mass of dry air
NBS	National Bureau of Standards
OD	Outdoor room temperature
ORNL	Oak Ridge National Laboratory
pt	point
TC	Thermocouple
Tsat	Saturation temperature
TXV	Thermal expansion valve
WB	Wet bulb
WG	Water gauge

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CHAPTER 1

INTRODUCTION

Central air conditioning is one of the major electrical energy using appliances in residences. Even though only 29% of existing homes in U.S. have central air conditioners, over 57% of new homes constructed in 1984 had central air conditioners [1,2]. In Austin, Texas, air conditioning energy use accounts for over one-third of the energy use and over one-half of the peak electrical demand in the residential sector[3]. According to Energy Information Administration (EIA), an average household electricity consumption for air conditioning during 1984 was 29 Million Btus[4].

To obtain the best performance (capacity and COP) for an air conditioner, the unit should be charged with an optimum amount of refrigerant. A previous study on a system using capillary tube expansion indicated that over or undercharging the system would result in degraded performance (capacity and efficiency) when compared to an optimally charged unit [5].

The objective of this study was to quantify the degradation of performance (capacity, coefficient of performance(COP), etc.) during under and overcharging of a residential sized central air conditioner which utilized a thermal expansion valve (TXV). A medium efficiency (9<SEER<10) air conditioner was tested under steady state conditions at four outdoor air temperatures: 82^{o}, 90^{o}, 95^{o}, and 100^{o} F. The Standard Air Conditioning and Refrigeration Institute(ARI) tests [6] were also run on the unit. The tests were performed for 5, 10, 15, and 20% under and overcharging of refrigerant (by mass). Two additional undercharging conditions (30% and 40%) were also performed to examine the effects of severe undercharging. This study was the continuation of the investigation on the air conditioner performance with capillary tube expansion [5]. The major change in this study was the replacement of the capillary tube expansion with a thermal expansion valve.

In Chapter 2, a description of the test setup and experimental procedure is discussed. A 3-ton Trane air conditioner with thermal expansion valve (model TTX724A100A/TWV742) was used in this study. The air conditioner was tested in the psychrometric rooms at the Energy System Laboratory. A detailed description of the dry and wet coil steady state tests as well as the cyclic test are provided.

Results are presented in Chapter 3. Major variables evaluated included: Capacity, Energy Efficiency Ratio(EER), Seasonal Energy Efficiency Ratio(SEER), coefficient of degradation (C_D), and demand power (kw) for both under and overcharging. Other system variables such as refrigerant mass flow rate, superheat, and subcooling temperatures are also reported.

Major conclusions from this study and recommendations for further study are provided in Chapter 4. Some of the recommendations center on the relationship between the amount of refrigerant and capacity, EER, SEER, and C_D.

CHAPTER 2

EXPERIMENTAL APPARATUS

The objective of the experimentation was to quantify the effect of improper refrigerant charge on the performance of a residential air conditioner during steady state and cyclic operations. The data collected included pressures and temperatures throughout the system, power consumption, capacity, EER, SEER, and refrigerant and air flow rates. A testing apparatus was constructed that would allow measurement of these important performance parameters. The air conditioner testing apparatus and testing procedure are described below.

GENERAL DESCRIPTION

The test apparatus was located in the psychrometric rooms of the Energy System Laboratory at the Texas A & M University Research Annex. The General layout of the test apparatus is given in Figure 2.1. The psychrometric rooms simulated the indoor and outdoor conditions (temperature and humidity) necessary for air conditioner performance testing.

The indoor test section consisted of the indoor coil (evaporator) and the indoor air flow chamber. Conditioned air from the indoor room was drawn through the indoor test section by the air flow chamber fan. A damper was mounted on the outlet that was adjustable and was set to maintain a

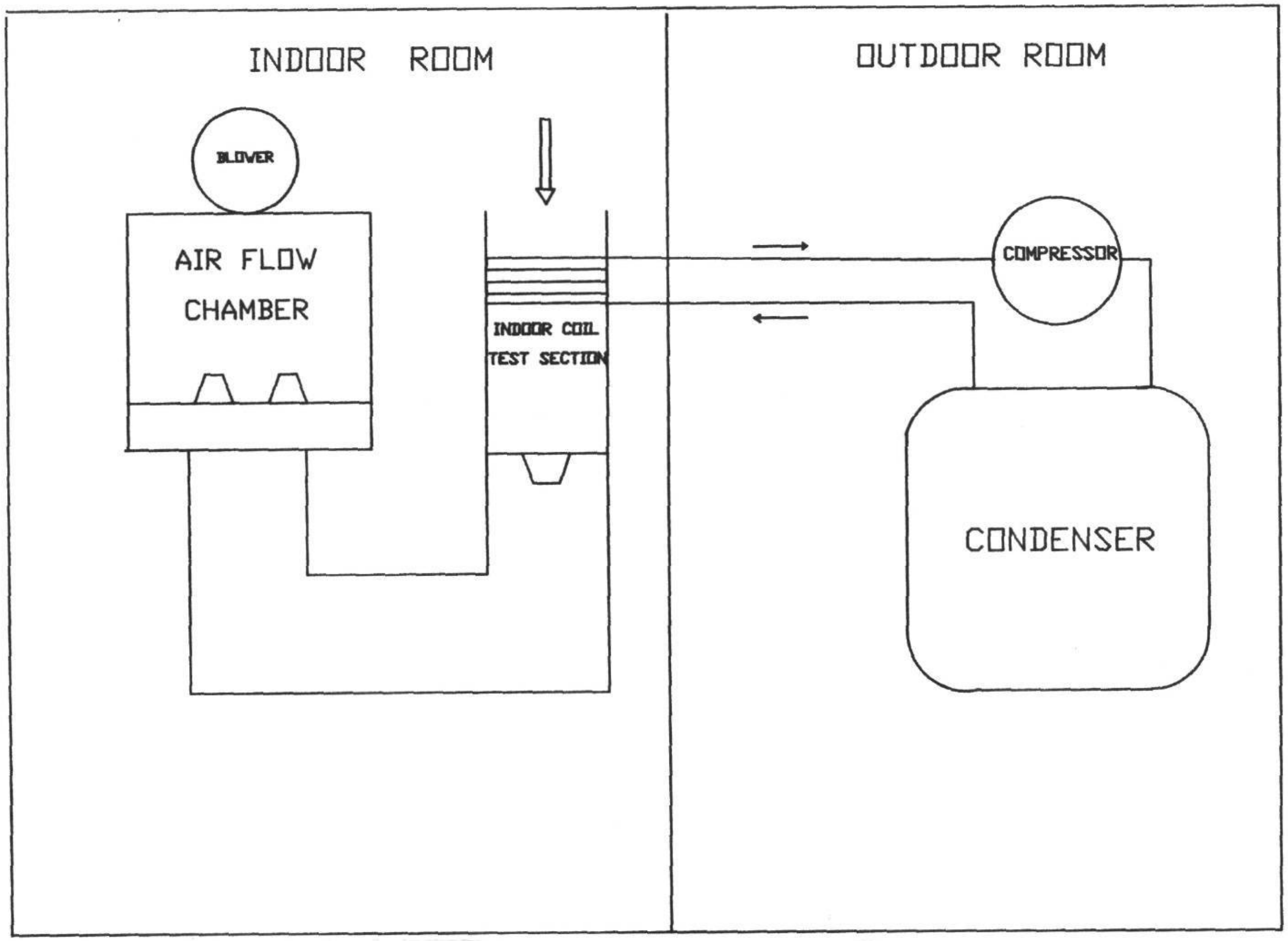

Figure 2.1 - The General Layout of the Test Apparatus

constant air flow of 1200 cubic feet per minute(cfm) through the indoor test coil. The air was routed back into the indoor room after leaving the chamber.

The outdoor room section consisted of the compressor and outdoor coil. The conditioned outdoor air entered the outdoor coil and was exhausted by the unit fan back into the room through the outdoor coil.

PSYCHROMETRIC ROOMS

The psychrometric rooms could simulate all testing conditions required for air conditioning and heat pump performance testing. Dew point and room temperatures could be maintained within +/-0.2 F of the set point. The room temperature was controlled by a Texas Instruments TI-550 controller.

Room temperatures were maintained with chilled water coils and electric resistance heaters. The chilled water coils were fed with an ethylene glycol solution that was chilled by a 105 ton capacity chiller. A 300 gallon chilled water thermal storage tank was mounted in the chilled water system to provide some thermal capacity. There were four banks of electric heaters in each room with 9900 watts per bank.

Humidity levels in the rooms were controlled by electric humidifiers and dehumidification coils. The dehumidification coils were fed from the same circuit as the cooling coils. The humidifiers were mounted in each room and supplied steam directly into the supply air duct.

TESTING CONDITIONS

The testing conditions used for the steady state wet and dry coils and cyclic tests were those prescribed in the Department of Energy (DOE) "Test procedures for Central Air Conditioners, Including Heat Pumps (1979)[7]. The entering dry bulb temperature for the outdoor coil for steady state and cyclic tests was 82^{o} +/-0.3 F DB and 20% relative humidity. The steady state tests were repeated for outdoor temperatures of 90^{o}, 95^{o}, and 100^{o}F. The indoor conditions were set at 80^{o} +/-0.3 F DB and 60.4^{o} +/-0.3 F DP (67^{o}F WB) for the wet coil test (A&B). For dry coil and cyclic tests, the dew point was set at 37^{o} +/-0.3 F DP (57^{o}F WB).

INDOOR TEST SECTION

The indoor test section is shown in Figure 2.2. Conditioned air flowed through a 22"x34", one-inch insulated sheet metal duct. A set of straighteners were used as the air entered this section. The air temperature was measured by a 16-element thermocouple grid before it entered into the coil. There were two dampers installed before and after the coil. The dampers were driven by two hydraulic actuators which were controlled by an "on-off" switch from the control room. After leaving the coil, the air flowed

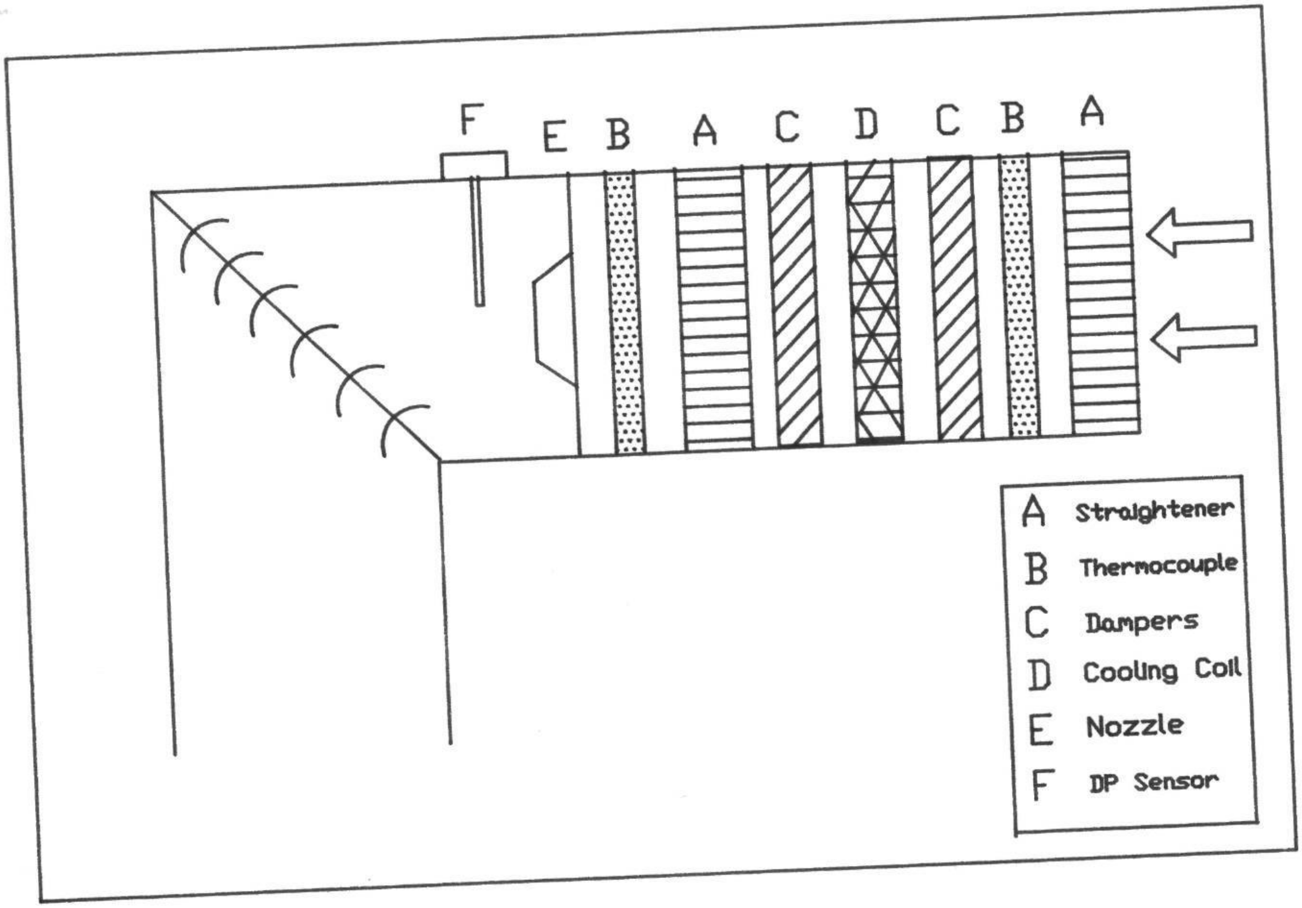

Figure 2.2 - Detail of Indoor Test Section

through another set of straighteners. Its temperature was then measured by a second 16-element thermocouple grid.

To accurately measure the dew point temperature, the dew point sensors had to be mounted in an air stream of 500 to 3000 fpm. An air sampler was constructed to sample the air entering the indoor coil. The sampler was a 4x6 inch duct with a fan at the end of the duct. The fan drew air through the duct where the dew point sensor was mounted. The air flow through the duct was approximately 1700 fpm which was within the operating range of the sensor. A 12-inch nozzle was mounted after the second 16-element thermocouple grid to increase the velocity of air up to 1500 fpm for the down stream dew point sensor.

A 3-ton Trane air conditioner with thermal expansion valve (Model TTX736A100A1/TWV 742) was used in this experiment. The indoor coil was an A-shape coil with a rated capacity of three tons.

After leaving the test section, the air was drawn into an Air Movement and Control Association(AMCA) 210 flow chamber where the air flow was measured. The chamber contained four American Society of Mechanical Engineers(ASME) air flow nozzles (one-8", two-5" and one-3") that could be used in any combination to accurately measure a flow range of 100 to 5000 cfm [8]. A booster fan mounted on the end of the chamber provided the air flow through the setup. The air flow was adjusted by operating a set of dampers mounted on the fan outlet. For the steady state and cyclic tests, two 5" nozzles were used in the chamber to achieve a pressure drop of 1.13" WG which was equivalent to 1150 cfm through the indoor test coil.

OUTDOOR TEST SECTION

The outdoor test section included the compressor, the outdoor coil (condenser), and a turbine flow meter, (Figure 2.3). A 3-ton Trane air conditioner with model TTX736A100A outdoor unit was used. The outdoor coil had one row with spine fins at 20 fins per inch. The face area of the coil was 20.94 ft^2 with refrigerant tube sizes of 3/8". The outdoor fan was located on the top of the outdoor coil. The fan specifications are given in Table 2.1.

Table 2.1 - Fan Specifications.

Fan Type	Propeller
Diameter (in)	22
Drive Type	Direct
CFM @ 0 in. w.g.	2735
Motor HP	1/4
Motor Speed RPM	825
F.L. Amps	1.9

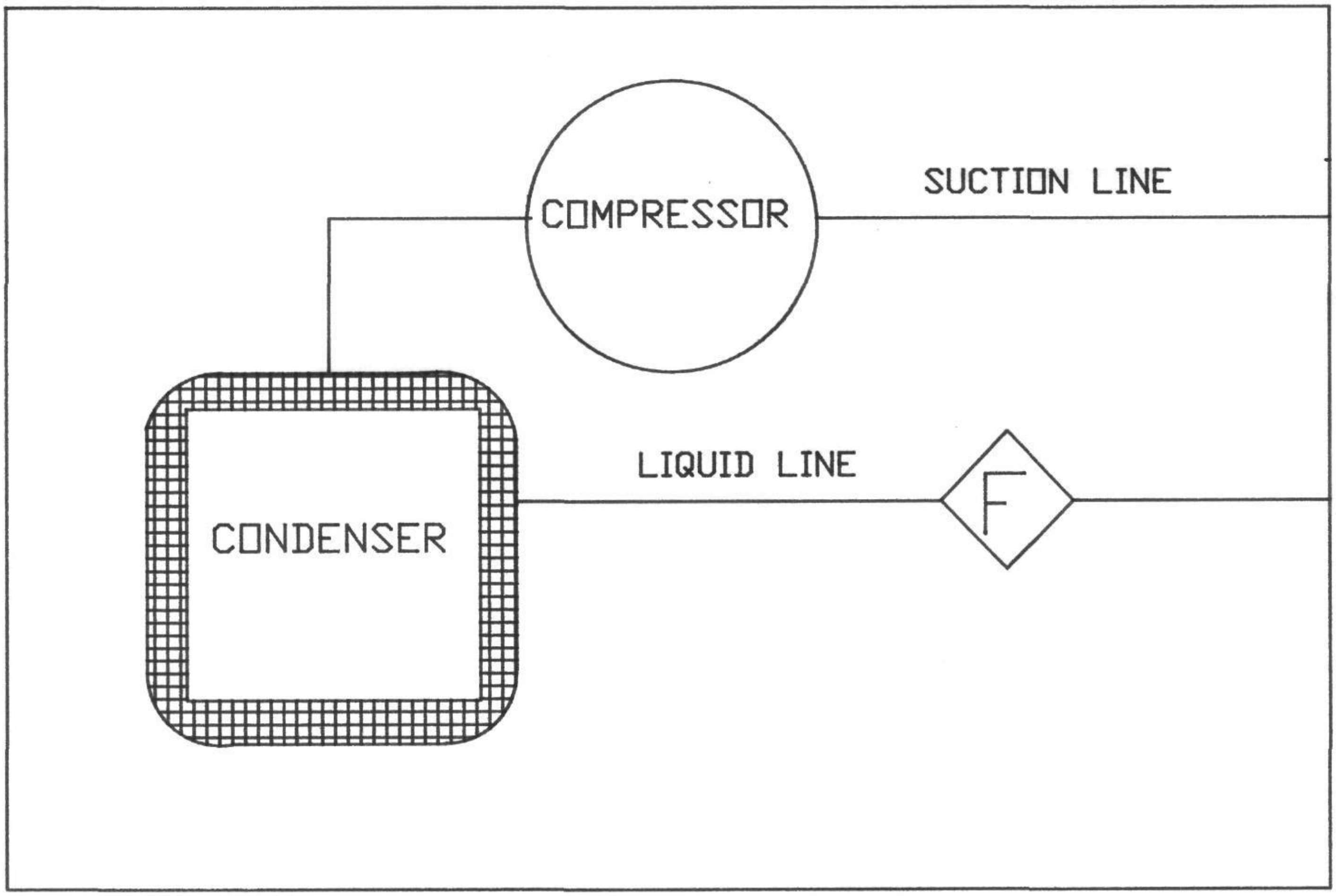

Figure 2.3 - Detail of Outdoor Test Section

The temperature of the air leaving the outdoor coil was measured by a 6 element thermocouple grid. According to ARI standard 210/240-84, the wet bulb temperature condition was not required when testing an air-cooled condenser which did not evaporate condensate.

REFRIGERANT SIDE

A schematic of the refrigerant circuit is shown in Figure 2.4. Refrigerant pressures were monitored at the 6 points shown with the use of 0-300 and 0-500 psig pressure transducers. To accurately measure the refrigerant temperatures and reduce the conduction effects of the copper tubing, seven thermocouple probes were installed in the refrigerant lines. The probes were 1/16"in diameter and mounted far enough into the flow of the refrigerant to minimize the tube conduction effects (Figure 2.5).

Refrigerant mass flow was measured with two mass flow meters mounted in parallel. The flow meters measure mass flow according to the Coriolis principle as the refrigerant flows through a U-shaped tube. As shown in Figure 2.4, the flow meters were placed on the liquid line after the condenser unit. The flow rate through the meters varied from 0 to 10 pounds per minute with a pressure drop at maximum flow rate of approximately 10 psi. This pressure drop was less than the 12 psi pressure drop acceptable by ASHRAE Standard 116-83 [7] (12 psi is the equivalent

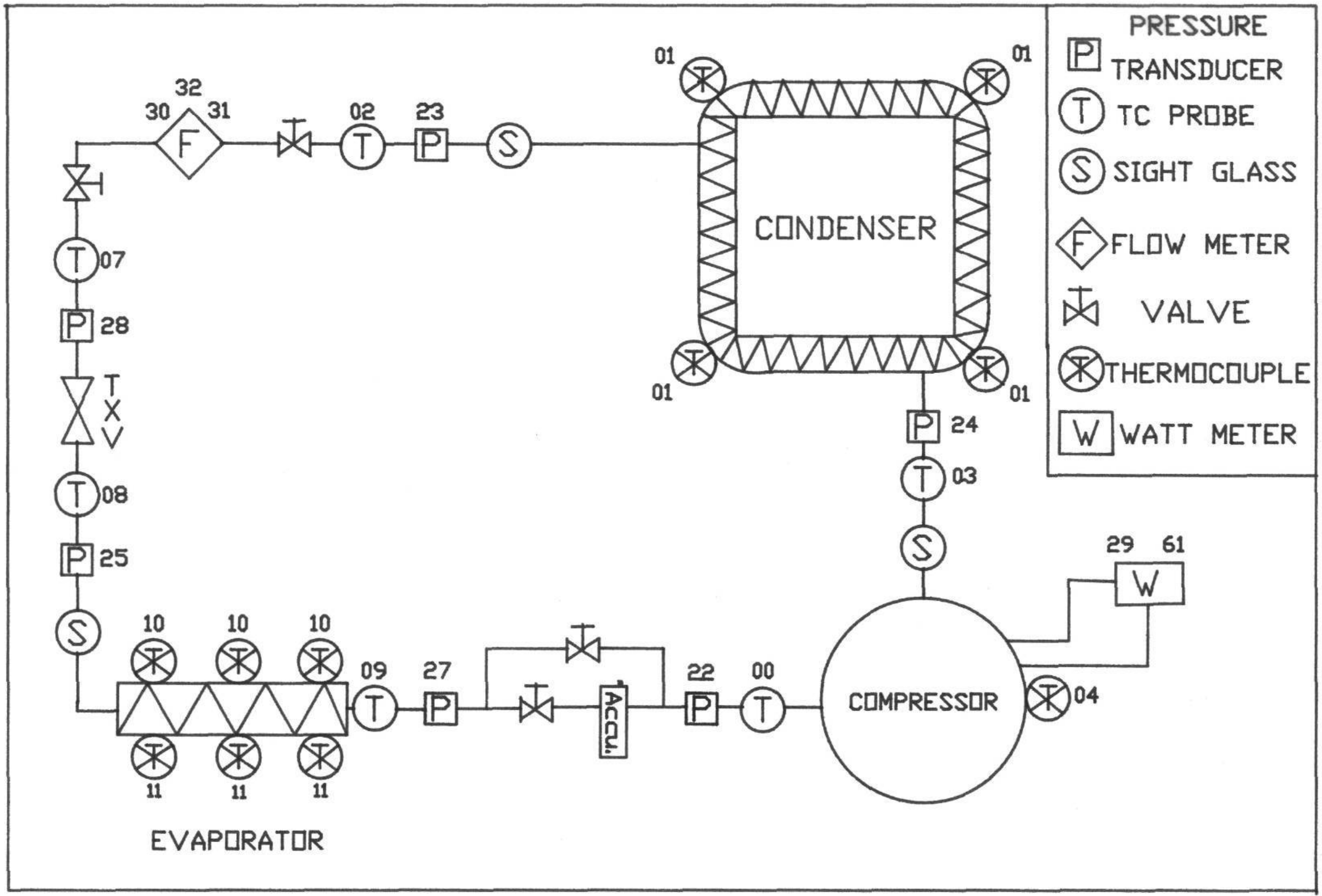

Figure 2.4 - Schematic of the Refrigerant Circuit

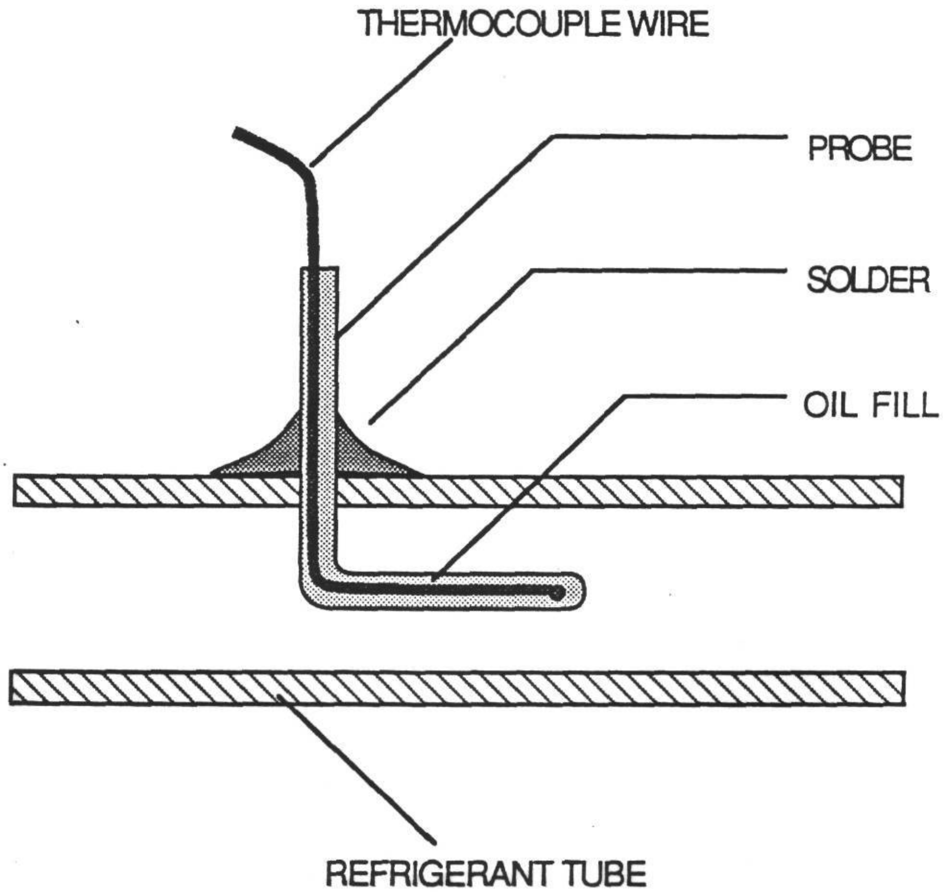

Figure 2.5 - A Typical Refrigerant Temperature Probe

pressure drop for refrigerant at the test conditions experiencing the maximum allowed temperature drop of $3^{o}F$).

The valves shown in the refrigerant circuit diagram were lever-actuated shut-off valves. Several ball valves were mounted around all sections of the refrigerant circuit to allow easy disassembly of the unit without any loss of refrigerant charge. Charging taps in each section of the circuitry allowed purging and charging of each section independently.

DATA ACQUISITION

Sensor signals from the test points listed in Table 2.2 were collected and converted to engineering units by an Acurex (model Autocalc) data logger. The data logger handled millivolt and milliamp signals as well as larger voltages and frequency signals. During each test, the data processed by the data logger was transferred to a portable Compaq personal computer where it was stored on a 10 megabyte hard disk. The maximum collection and storage rate for the set of data channels used in a test was eight seconds per set. The scan rate was adjustable, so data from each test (cyclic and steady state) were collected every 15 seconds.

A feature of the data acquisition set-up was the continual display of run-time data on the screen during testing. After completion of a test series, all data collected on the hard disk were transferred to a VAX minicomputer for analysis. Data were backed up on floppy disks. All data analysis programs were written in FORTAN. Refrigerant and moist air property subroutines were used in calculation of air and refrigerant-side cooling capacities to provide an energy balance for data validation. Additional calculated properties and performance parameters for each test were plotted.

Table 2.2 Description of Test Points Used in the Test Set-Up

Channel	Sensor	Location
00	TC-Probe	Compressor Inlet
01	Thermocouple	Outdoor Room Temp.
02	TC-Probe	Condenser Outlet
03	TC-Probe	Compressor Outlet
04	Thermocouple	Compressor Shell Temp
05	Thermocouple	Before Nozzle-Chamber
06	Thermocouple	After Nozzle-Chamber
07	TC-Probe	TXV Inlet Temp.
08	TC-Probe	Evap.Inlet Temp.
09	TC-Probe	Evap.Outlet Temp.
10	TC-Grid	DB Indoor Coil Outlet
11	TC-Grid	DB Indoor Coil Inlet
12	Thermocouple	Chilled Water Temp.
13-19		-- Not Used --
20	Dew-Point	Downstream Indoor Coil
21	Dew-Point	Upstream Indoor Coil
22	Pressure Trans.	Compressor Inlet
23	Pressure Trans.	Condenser Outlet
24	Pressure Trans.	Compressor Outlet
25	Pressure Trans.	Evaporator Inlet
27	Pressure Trans.	Evaporator Outlet
28	Pressure Trans.	TXV Inlet
29	Watt Trans.	208 VAC (kw)
30	Flow Meter	Ref. Liquid Line
31	Flow Meter	Ref. Liquid Line
32		Sum of 30 & 31
33-60		-- Not Used --
61	WattHr Trans.	Compressor Power

The psychrometric rooms and the unit ran for two hours prior to any data recording. This allowed the rooms time to reach steady state conditions. The data for the steady state tests were recorded continuously for 30 minutes. Several 30-minute sets of data were recorded for each test. The cooling cyclic tests were conducted by cycling the compressor 6 minutes "on" and 24 minutes "off". The capacity was measured for 8 minutes: 6 minutes of "on" time and 2 minutes longer until it reached zero. Electrical energy was measured for 6 minutes of "on" time. The dampers were shut off after first 8 minutes of the cyclic test to isolate the indoor coil.

PROPER AND IMPROPER REFRIGERANT CHARGING PROCEDURES

A procedure was established for testing that would ensure the repeatability and reliability of the test data. The first step of the procedure was to set the system refrigerant charge. Manufacturer recommendations called for 14°F of subcooling leaving the condenser. Initially, the charge was set for outdoor room conditions at 95°F and indoor room conditions of 80°F dry bulb and 67°F wet bulb. Refrigerant was added to the system in one ounce increments until a subcooling of 13.5°F was reached. This was within 0.5°F of the value recommended by the manufacturer. As shown in Figure 2.6, the subcooling was also checked at other outdoor temperatures (82°, 90°, and 100°F). The tests

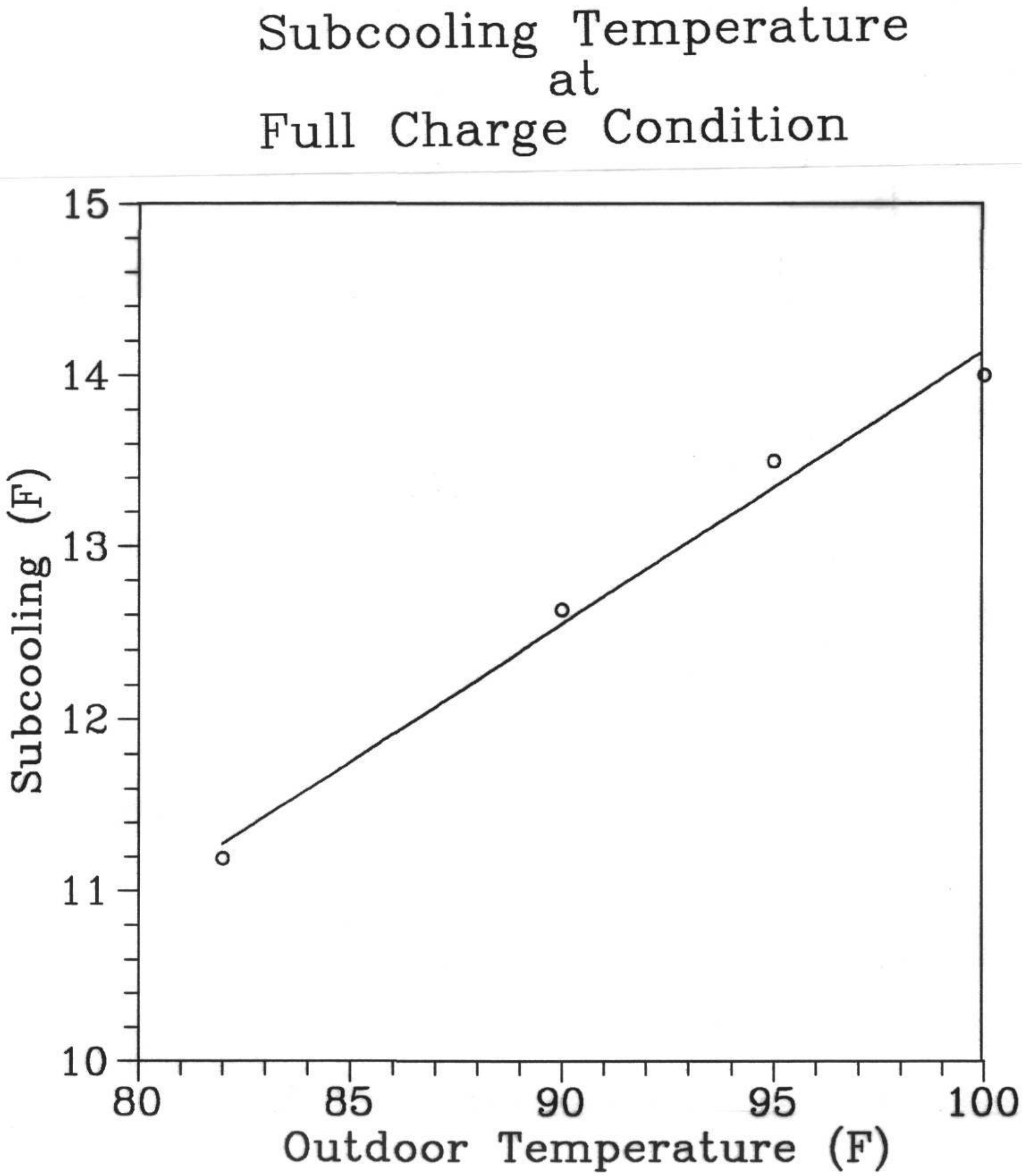

Figure 2.6 - The Fully Charged Subcooling Temperature as a Function of Outdoor Temperature

were conducted based on 400 CFM/ton indoor air flow. According to the factory chart and graphs (provided by Trane), the suction and the liquid pressures should be within +/-3 and +/-10 psig of the chart, respectively. The suction pressure was 3 psig higher than the chart pressure. The liquid pressure was 6 psig lower than chart pressure at 82^{o} F outdoor temperature. At higher outdoor room temperatures, the liquid pressure was within 2 psig of the chart pressure.

Once the full charge was determined (140 ounces), the system was vacuumed and it was recharged to 84 ounces of refrigerant which corresponds to 40% undercharge. The under and overcharge of refrigerant in the system in increments of 7 ounces are shown in Table 2.3. A scale was used to weigh the amount of refrigerant added to the system.

Table 2.3 - Different Amount of Charge in the System

Charge	Ounces
40% U.C.	84
30% U.C.	98
20% U.C.	112
15% U.C.	119
10% U.C.	126
5% U.C.	133
F.C.	140
5% O.C.	147
10% O.C.	154
15% O.C.	161
20% O.C.	168

EQUIPMENT

A complete listing of equipment used in the testing apparatus is given in Appendix A. All testing instrumentation was calibrated prior to data collection and the accuracies are also listed in Appendix A.

TESTING PROCEDURE

The first step in the testing session was to set the unit proper refrigerant charge according to the subcooling temperature at the room conditions mentioned previously. Once the proper charge was determined, the system was vacuumed and recharged to 40% undercharged condition (84 oz).

The steady state tests (wet coil) under 4 different outdoor temperatures: 82°, 90°, 95°, and 100°F were performed while the indoor conditions were set at 80°F DB and 67°F WB. Both steady state and cyclic tests (dry coil) were performed with indoor conditions set at 80°F DB and 57°F WB while the outdoor conditions were constantly kept at 82°F and 20% relative humidity. Both the wet and dry coil tests were repeated for 40 and 30% undercharging as well as 5, 10, 15, and 20% under and overcharging of refrigerant (by mass). All tests were conducted based on 400 CFM/ton indoor air flow.

CHAPTER 3

RESULTS & ANALYSIS

The refrigerant charge in the system was systematically varied to determine its effect on the capacity, EER, SEER, and coefficient of degradation (C_D). The results of under/over charging the air conditioner are presented below.

FULL CHARGE CONDITION

All tests were performed on a split system central air conditioner provided by the Trane company. To determine the proper amount of refrigerant charge needed in the system and the unit's corresponding performance, charging specifications on the unit were obtained from Trane Co.

The unit was charged according to the procedures specified by Trane. These procedures included setting of a particular subcooling for specific outdoor conditions. The amount of refrigerant in the system for full charge was found to be 140 ounces of R-22.

Four variables were used to quantify the overall performance of the unit: total capacity, total electrical power consumption, Energy Efficiency Ratio(EER), and Seasonal Energy Efficiency Ratio(SEER).

The total capacity of the unit is expressed in Btu/hr. It can be measured by either measurements on the air-side or refrigerant side of the evaporator. While both measurements were made, only data from the air-side are presented in this report. The indoor coil capacity was calculated using the air-enthalpy method found in ASHRAE Standard 116-1983 [7]. In the air-enthalpy method, the steady state capacity of the indoor coil was determined from:

$$Capacity = \frac{cfm}{v} \times (h_2 - h_1)$$

where

h_1 = Enthalpy of air entering the indoor coil (Btu/hr),

h_2 = Enthalpy of the air leaving the indoor coil (Btu/hr),

cfm = cubic feet per minute of dry air passing through the indoor coil, and

v = specific volume of the air passing through the coil (ft^3/lb).

Values h_1, h_2, and v were obtained from methods contained in the ASHRAE 1985 Fundamentals Handbook [9]. The airflow calculations were done using a method provided in ANSI/ASHRAE Standard 51-1985[8].

To verify the calculations for the air-side capacity, an energy balance was performed on the indoor coil. Figure 3.1 shows a plot of the refrigerant side and air side

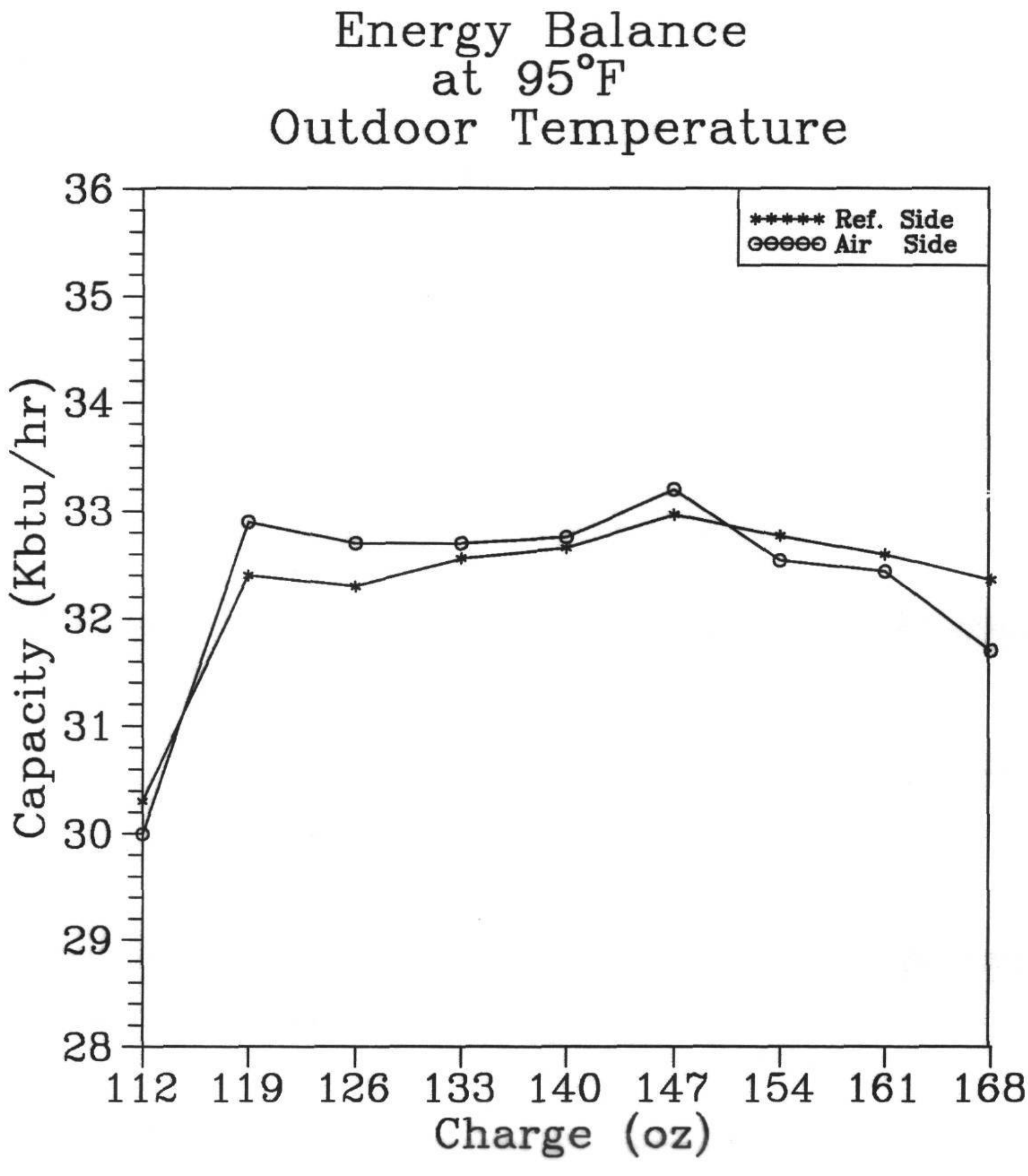

Figure 3.1 - Refrigerant-Side/Air-Side Capacity Comparison

capacity for the indoor coil as a function of charge at 95^{o}F outdoor temperature.

The refrigerant side capacity was calculated by multiplying the refrigerant mass flow rate by the change in enthalpy of the refrigerant entering and leaving the indoor coil. The enthalpy of the refrigerant was calculated using subroutines developed by Kartsounes and Erth [10]. Typically, the air-side and refrigerant side were within +/- 1% to +/-3% agreement for under and overcharging.

The total electricity power consumption by the system is the combination of power consumed by the indoor and outdoor sections. The outdoor section power was measured directly with a watt-hour meter. The indoor fan power was calculated based on 365 watt per 1000 cfm of air because the test unit operated without an indoor unit fan[7].

EER is a steady state measure of efficiency. It is calculated by dividing the net cooling capacity in Btu/h by the power input in watts (w) at a given set of indoor and outdoor conditions. It is expressed in Btu/wh.

SEER is a measure of the seasonal efficiency of the unit. It is calculated from a series of steady state and cycling tests (described in the next section).

1. Steady State Tests (Wet Coil)

The DOE test procedure requires two steady state tests in which dehumidification occurs on the evaporator coil. Both tests are at the same indoor conditions ($80^{o}F$ DB and $67^{o}F$ WB) and at two outdoor temperatures ($82^{o}F$ and $95^{o}F$). In addition to the two outdoor temperatures required by the test procedure, steady state tests were also performed at two more outdoor temperatures ($90^{o}F$ and $100^{o}F$).

For each outdoor temperature, several 30-minute sets of data under steady state were recorded. Figures 3.2 and 3.3 show the net capacity and EER as a function of outdoor temperature under the fully charged condition. Both the capacity and EER decreased with increasing outdoor temperature. The capacity dropped from 34.1 KBtu/hr at $82^{o}F$ to 30.8 Kbtu/hr at $100^{o}F$. The EER ranged from 10.41 at $82^{o}F$ to 8.53 at $100^{o}F$.

2. Steady State and Cyclic Tests (Dry Coil)

The DOE test procedure also requires testing of an air conditioner under conditions in which no condensation would occur on the evaporator coil. Both steady state and cyclic tests were performed with indoor condition set at $80^{o}F$ DB and $57^{o}F$ WB. The wet bulb temperature was sufficiently low enough so that no condensate formed on evaporator coil. The

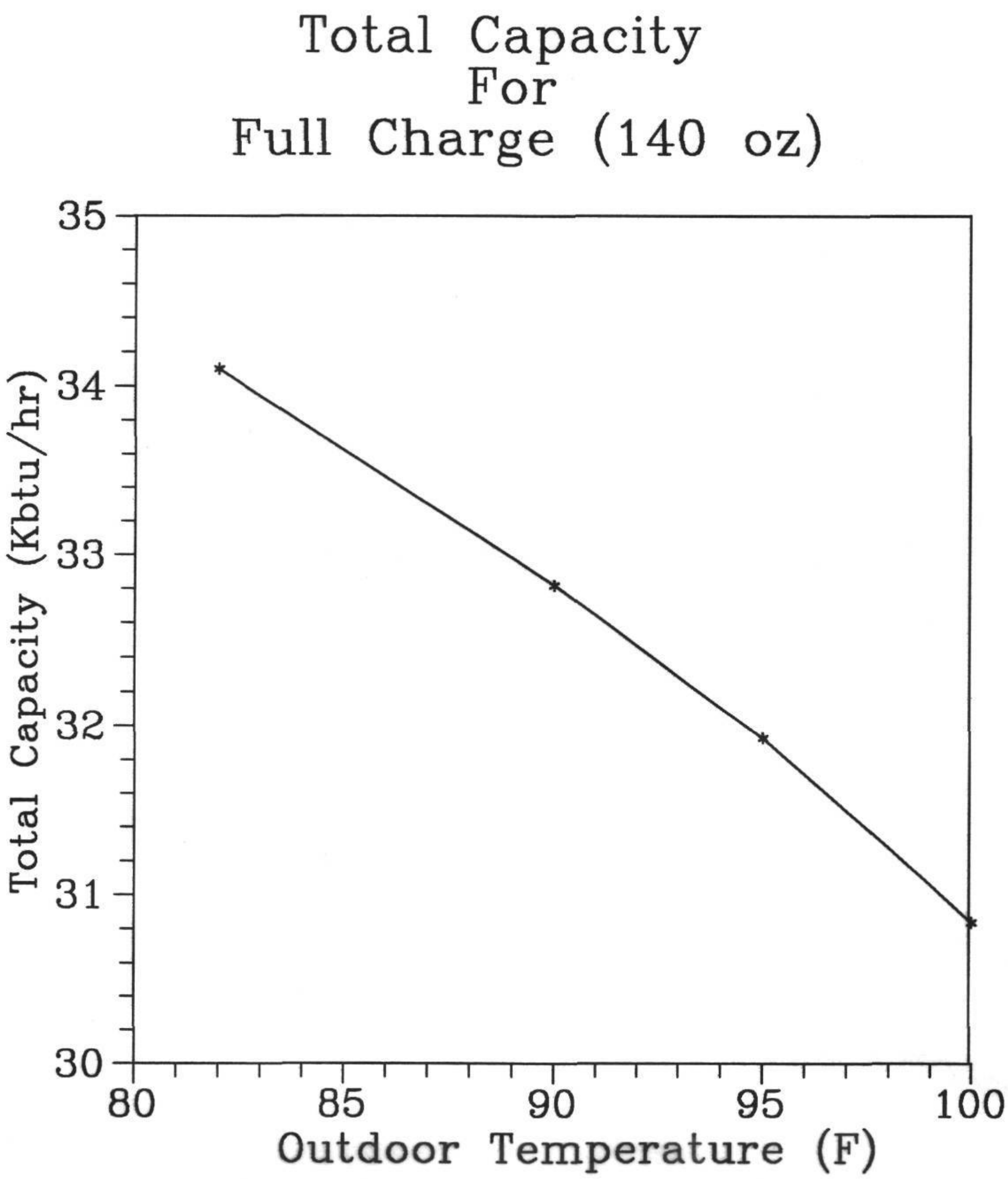

Figure 3.2 - Total Capacity of the Fully Charged Unit

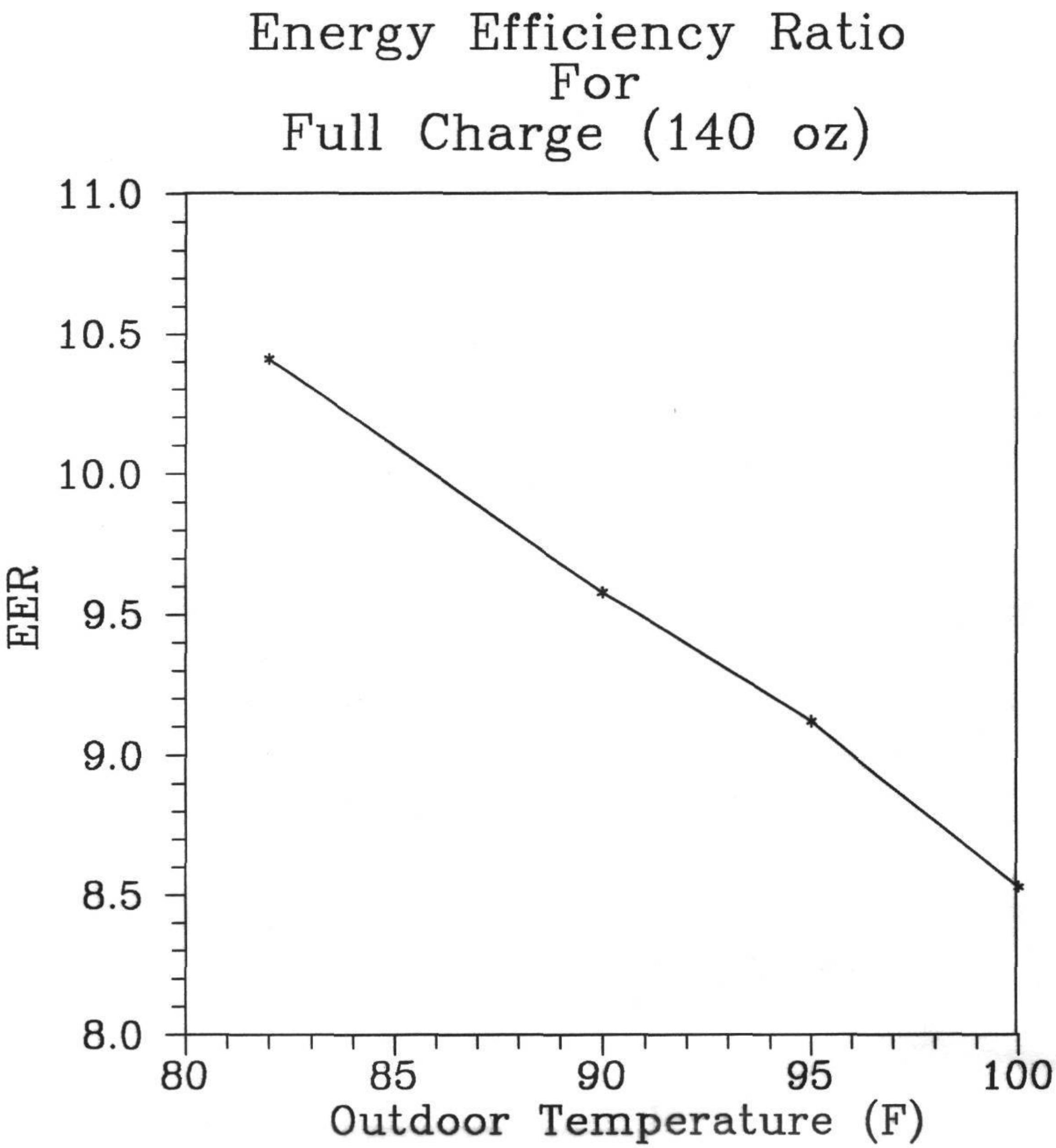

Figure 3.3 - **Energy Efficiency Ratio of the Fully Charged Unit**

outdoor room condition was constantly kept at 82^{o}F DB and 20% relative humidity during these tests.

The cooling cyclic test was conducted by cycling the compressor 6 minutes "on" and 24 minutes "off". During the "on" period, electrical energy and capacity measurements were made. According to the DOE test procedure[6], during the first two minutes of the "off" period, the capacity was also measured. Then the evaporator coil was isolated by shutting off the dampers during "off" time for 22 minutes. The capacity was calculated for the 8 minutes (the six minutes during the "on" cycle and two minutes after). All electrical energy (outdoor fan and compressor) was measured for the "on" time of 6 minutes. The indoor fan power for the time period during 6 and 8 minutes was added to the measured electrical energy. The power was calculated based on 365 watts per 1000 cfm of air. Figure 3.4 shows the net capacity during the cyclic test under the full charge. Due to the change in indoor and outdoor rooms conditions during the start up of cyclic test, the test was repeated four times to obtain more accurate readings. This procedure was established for testing that would ensure the repeatability and reliability of the test data.

The DOE test procedure requires three steady state tests (A,B,C) and one cyclic test (D). Tests A & B are steady state wet coil tests at 95^{o} and 82^{o} F DB outdoor room temperatures, respectively. Test (D) is a steady state dry

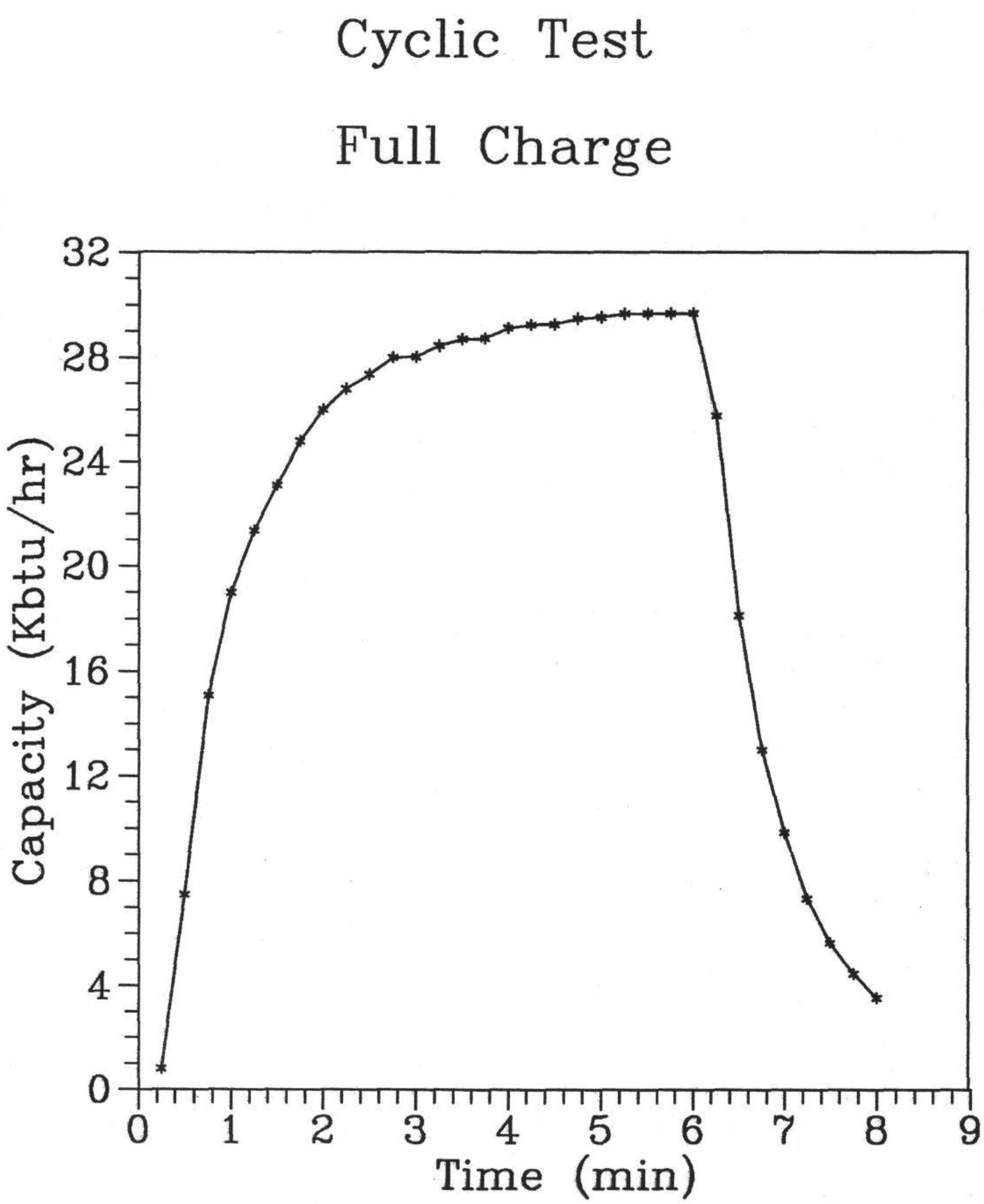

Figure 3.4 - Cyclic Capacity of the Fully Charged Unit

coil test at 82^{o} F DB outdoor room temperature. The calculation of the SEER with a single-speed compressor and single-speed condenser fan is done according to the test procedure (6,7). First, a cyclic-cooling-load factor (CLF) is determined from:

$$CLF = Q_D \ / \ (Q_C \ t_C)$$

where,

Q_D is the total cooling capacity of test D.

Q_C is the steady state cooling capacity of test C.

t_c is duration of time (hours) for one complete cycle consisting of one compressor "on" time and one compressor "off" time.

The degradation coefficient, C_D, is the measure of the efficiency loss due to the cyclic of the unit. C_D is calculated from:

$$C_D = (1 - EER_D/EER_C) \ / \ (1 - CLF)$$

where,

EER_D and EER_C are the energy efficiency ratios of tests D and C , respectively.

The SEER is then determined from a bin hours cooling method calculated based on representative use cycle of 1000 cooling hours per year. A 95^{o}F cooling outdoor design temperature was used. In accordance with ARI test procedure, the cooling building load size factor 1.1 (10% oversizing) was used.

Table 3.1 shows the unit performance under fully charged condition for steady state and cyclic tests.

Table 3.1 - Dry Coil & Cyclic Tests Performance

EER_C	EER_D	SEER	C_D	CCLF
9.46	8.46	9.70	0.131	0.196

CAPACITY

Once the proper charge was determined , the unit was vacuumed and recharged initially to 40% undercharged condition (84 oz). Different under and overcharging were obtained by systematically adding seven ounces of refrigerant to the unit and retesting it.

1. Steady State Tests (Wet Coil)

The steady state wet coil tests were performed at four different outdoor room temperatures 82°, 90°, 95°, and 100°F DB while the indoor conditions were set at 80°F DB and 67°F WB. These tests were repeated on the unit for 20, 15, 10, and 5% under and overcharged conditions. Figure 3.5 shows the total capacity as a function of the outdoor temperature and the refrigerant charge. The highest capacity was obtained at 82°F outdoor temperature and 10% underchanging.

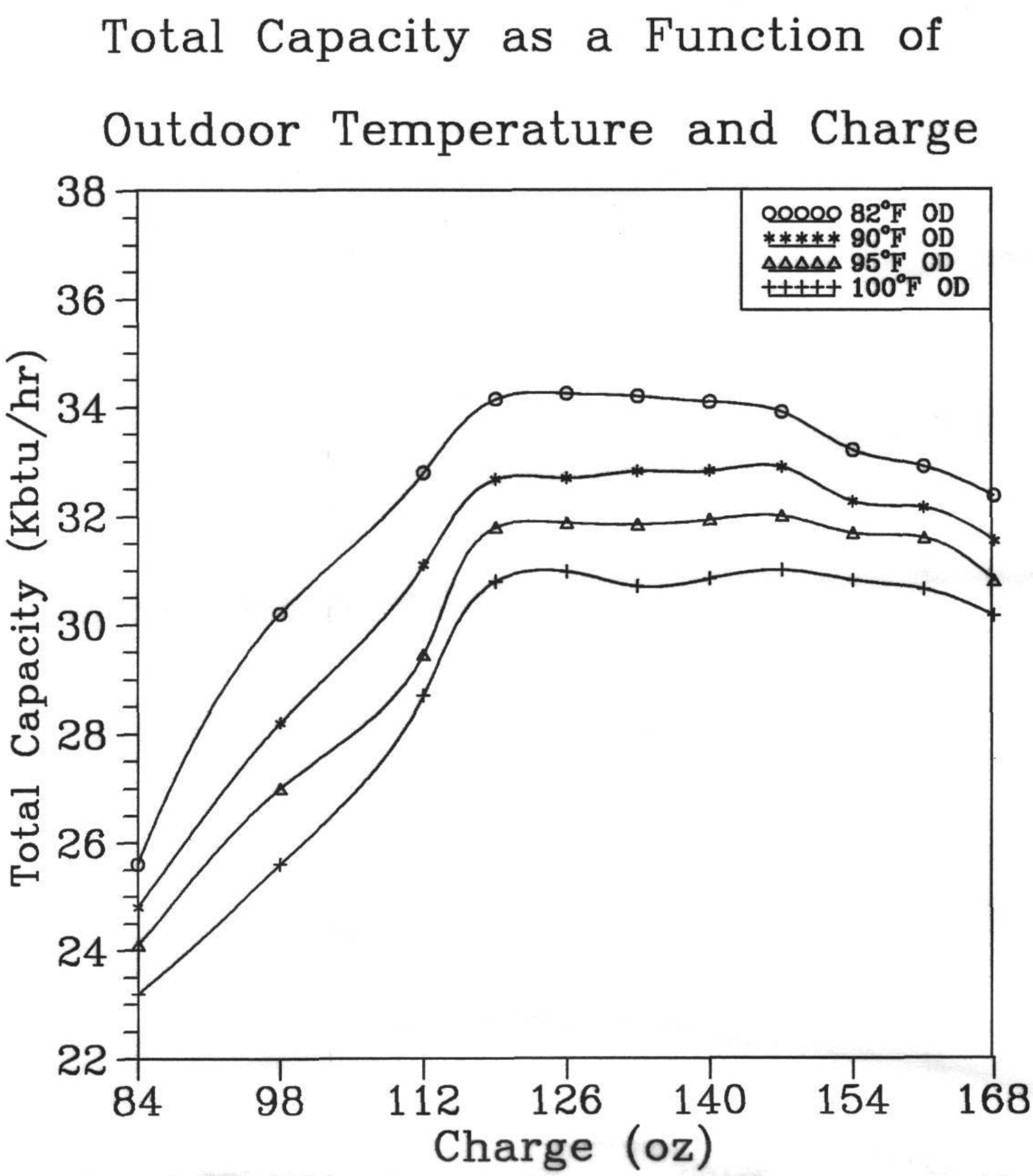

Figure 3.5 - Total Capacity as a Function of Outdoor Temperature and Charge (wet coil)

The capacity was nearly constant from 15% undercharging to 5% overcharging for the four outdoor temperatures. The capacity of the unit dropped off by an average of 10% as the outdoor room temperature increased from 82^{O}F to 100^{O}F for 15%, 10%, 5% undercharged, fully charged, and 5% overcharged conditions. At 95^{O}F, the capacity was 24, 27.9, and 29.1 KBtu/hr for 40%, 30%, and 20% undercharging, respectively, as compared to full charging.

The ability of the TXV to maintain capacity over a wide charge contrasts with the results in the previous study of the unit having capillary tubes. In that study, capacity dropped by as much as 30% for a 20% underchanging at 95^{O}F outdoor temperature.

1.1 Subcooling and Superheat Temperatures

For 40%, 30%, 20%, and 15% undercharging, the refrigerant at the inlet of the TXV was saturated (no subcooling). This subcooling increased slightly as the outdoor temperature increased for 10%, 5% undercharging, fully charged, 5%, 10%, 15%, and 20% overcharging (Figure 3.6). At 95^{O}F outdoor temperature, the subcooling temperature was constant at less than 1^{O}F for 40%, 30%, and 20% undercharging then increased linearly to 27^{O}F for 20% overcharging (Figure 3.7).

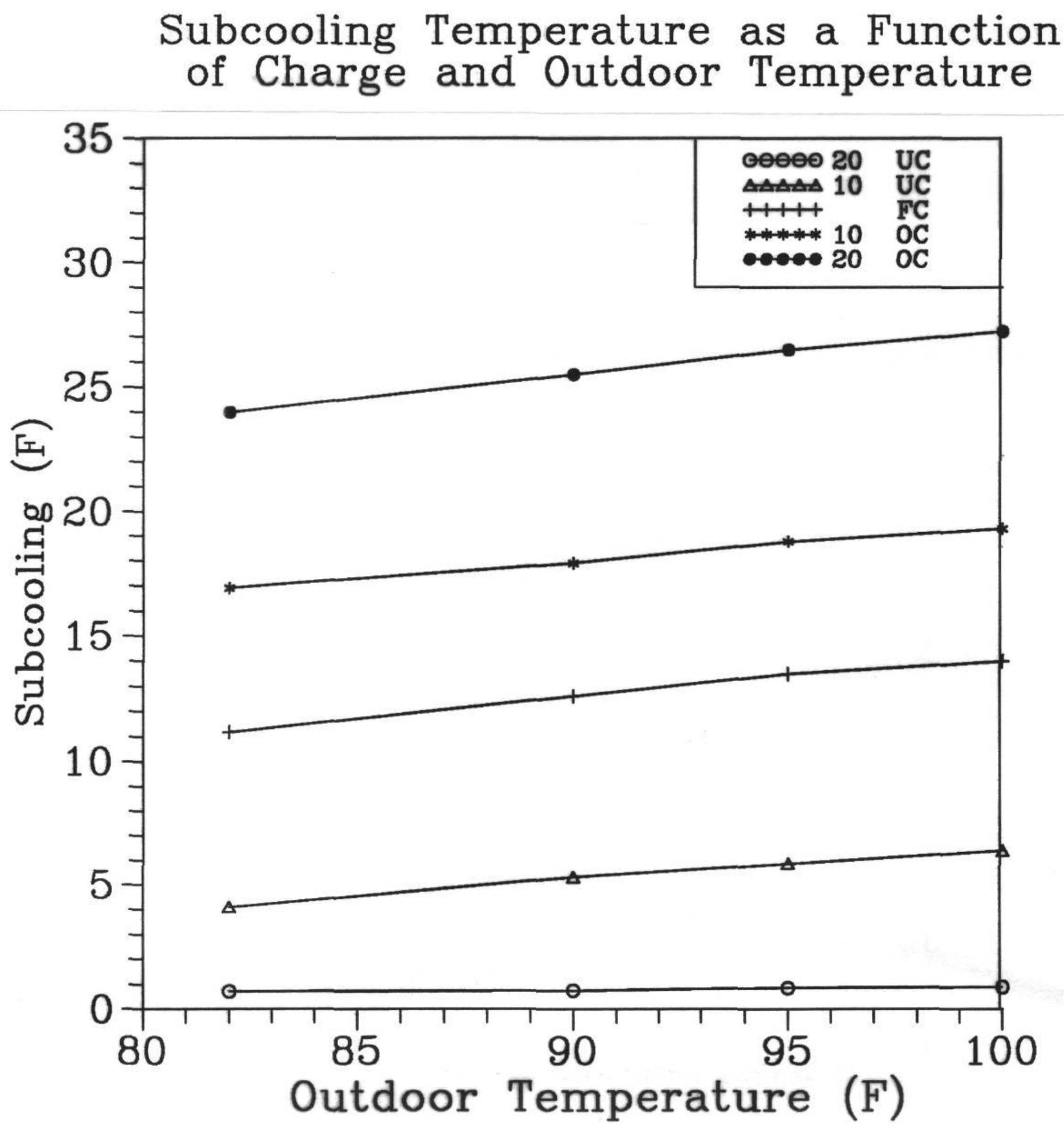

Figure 3.6 - Subcooling Temperature as a Function of Outdoor Temperature and Refrigerant Charge

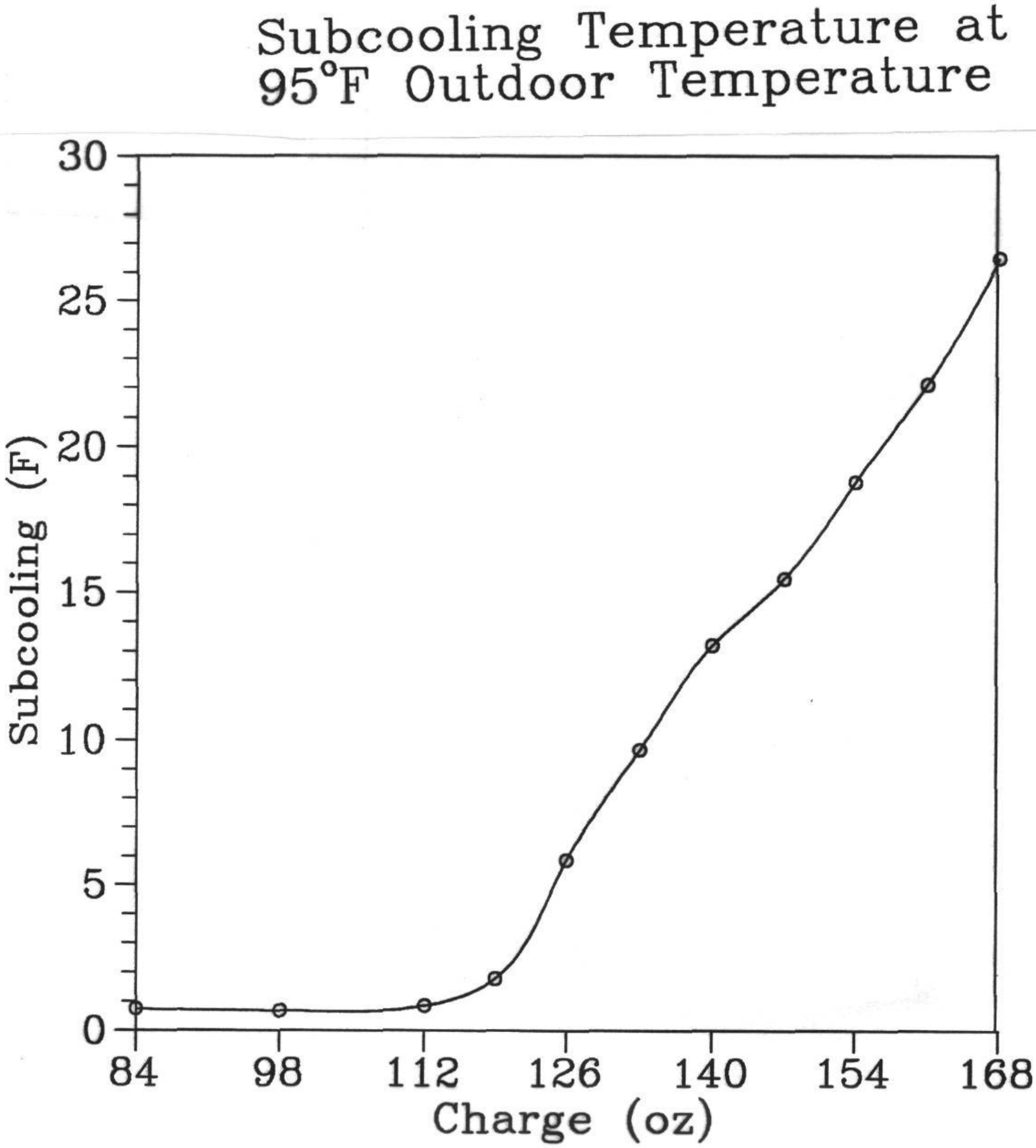

Figure 3.7 - Subcooling Temperature as a Function of Refrigerant Charge

The less than $1^{\circ}F$ subcooling for 84 to 112 oz charging probably corresponds to saturated conditions. This small subcooling could be due to the error associated with the refrigerant temperature and pressure probes.

The superheat temperature decreased as the outdoor temperature increased. As the charge in the system increased, the superheat decreased slightly (Figure 3.8). The superheat temperature dropped from an average of $17.5^{\circ}F$ to an average of $16^{\circ}F$ when the outdoor temperature increased from $82^{\circ}F$ to $100^{\circ}F$ for a charge condition.

A TXV should provide relatively constant superheat over the outdoor temperature tested in this study. The results from Figure 3.8 indicate that this TXV was able to maintain a relatively constant superheat for the range in outdoor temperatures tested as well as charges from 20% under to 20% overcharging.

1.2 Refrigerant Flow Rate

A thermal expansion valve (TXV) regulates the rate of flow of liquid refrigerant to produce a specified superheat at the outlet of the evaporator. A TXV responds to: (1) the temperature of the refrigerant gas leaving the evaporator and (2) the pressure in the evaporator. As the temperature of the refrigerant increases, the pressure in the remote bulb also increases and causes the valve to open more. As the outdoor temperature increases, the superheat temperature

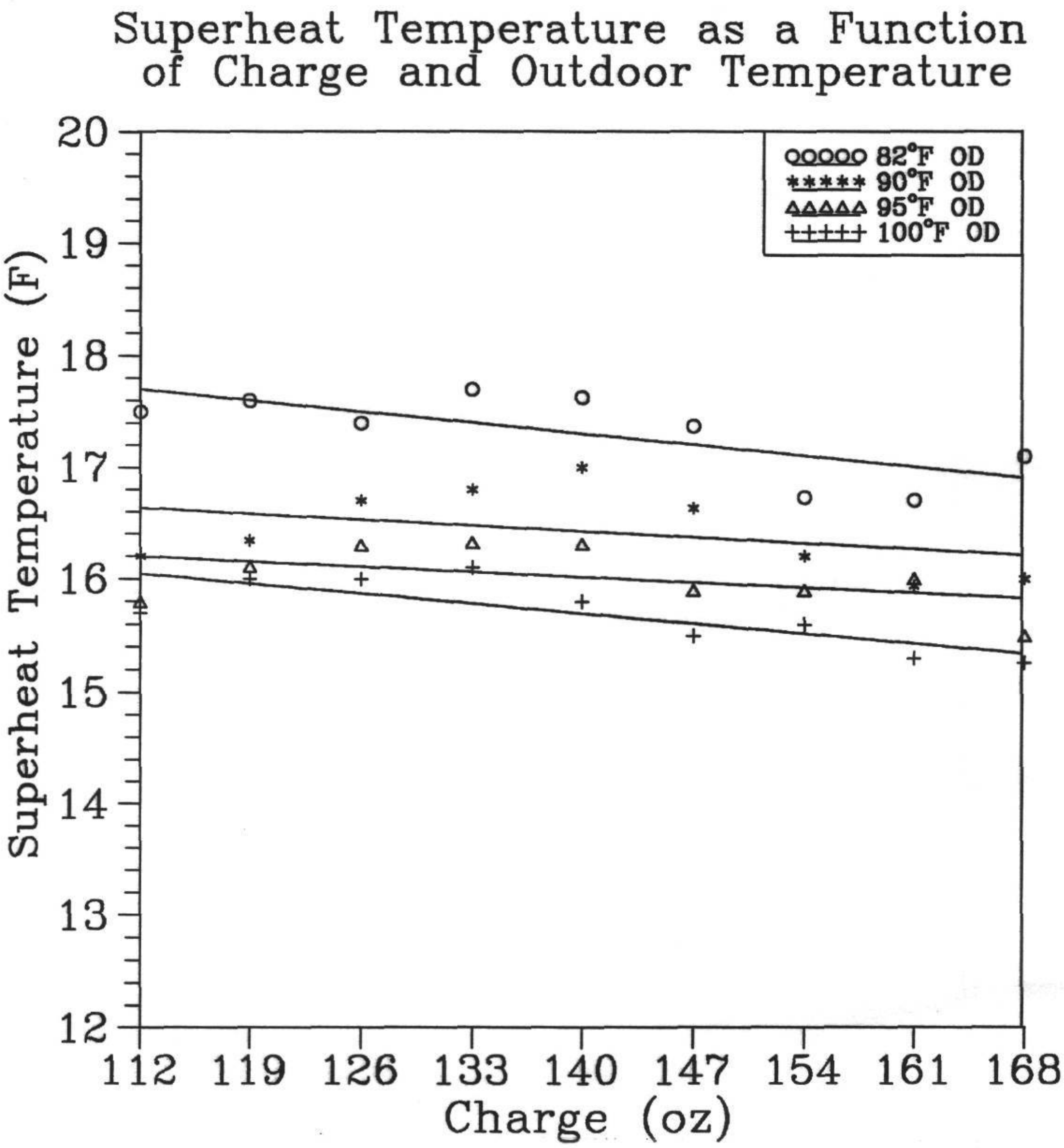

Figure 3.8 - Superheat Temperature as a Function of Refrigerant Charge and Outdoor Temperature

should decreased for a given charge. Higher outdoor temperatures should also increase the condenser pressure. For lower superheat, the TXV constrained the refrigerant flow rate. Figure 3.9 shows the refrigerant flow rate as a function of TXV inlet pressure for different charge. The refrigerant flow rate dropped from an average of 7.7 to 7.5 lbs/min when the outdoor temperature increased from 82^{O}F to 100^{O}F for 15%, 10%, 5% under/overcharging as well as 20% overcharged and fully charged conditions. At any given outdoor temperature, the refrigerant flow rate was relatively constant as the charge in the system increased. The largest increase in refrigerant flow rate was the increase in charge from 20% to 10% undercharging conditions. The refrigerant flow rate increased by 7% at 95^{O}F outdoor temperature.

A relatively constant superheat for the different charge in the system had an effect on the TXV regulating the refrigerant flow rate in the system. The results from Figure 3.9 indicated that the refrigerant flow rate was similiar for all the under and overcharging tests except for 40%, 30%, and 20% undercharging. The similar flow rates would explain why the capacities remained relatively constant from 15% undercharging to 10% overcharging and why they dropped for 20%, 30%, and 40% undercharged conditions.

1.3 Sensible Heat Ratio (SHR)

The sensible heat ratio is defined as the ratio of the

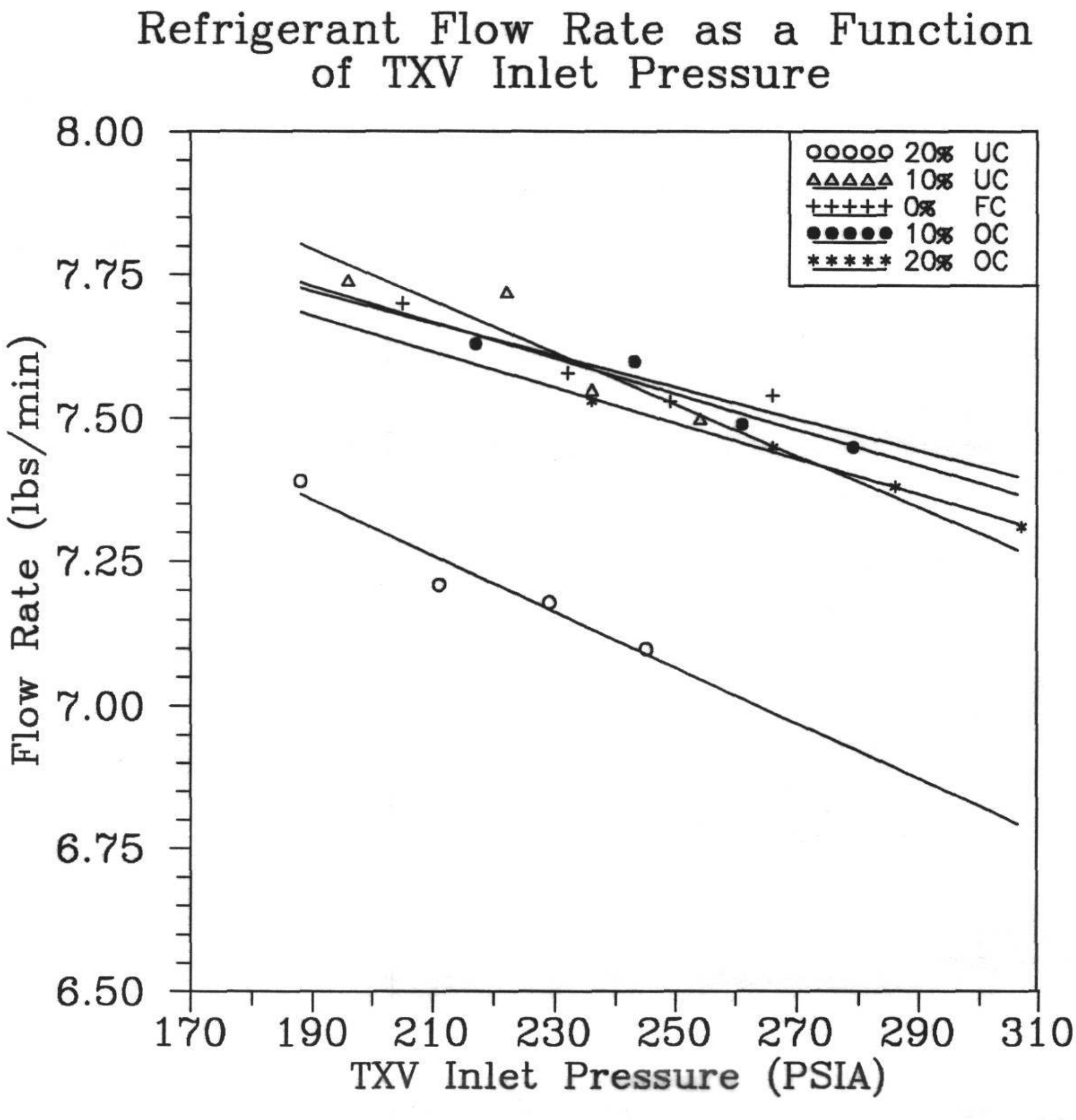

Figure 3.9 - Test Refrigerant Flow Rate as a Function of Refrigerant Charge and TXV Inlet Pressure

sensible capacity to the total capacity of the unit. Figure 3.10 shows the SHR as a function of charge and outdoor temperatures. The SHR decreased as the amount of the charge in the system increased until reaching full charge where the SHR remained approximately constant. At 95°F outdoor temperature, the SHR dropped by 5% when the unit was charged from 20% undercharging to full charging. For 20% undercharging, the SHR increased from 0.79 to 0.85 when the outdoor temperature increased from 82°F to 100°F. In case of overcharged conditions, the SHR was constant for 90°F, 95°F, and 100°F outdoor temperatures. However, the SHR increased slightly for 82°F outdoor temperature. The effect of outdoor temperature was less noticeable for overcharged than undercharged tests. For instance, The SHR increased by less than 2% for overcharged tests when the outdoor temperature increased from 82°F to 100°F.

2. Steady State & Cyclic Tests (Dry Coil)

For the dry coil tests, both steady state and cyclic tests were performed with indoor conditions set at 80°F DB and 57°F WB. The wet bulb temperature was sufficiently low that no condensate formed on evaporator coil. The outdoor room condition was constantly kept at 82°F DB and 20% relative humidity. The steady state dry coil (C) and cyclic (D) tests were performed on the under and overcharged conditions. The unit total capacity (TestC) is shown in Figure 3.11. It peaked at 29.6 KBtu/hr for the 10% under

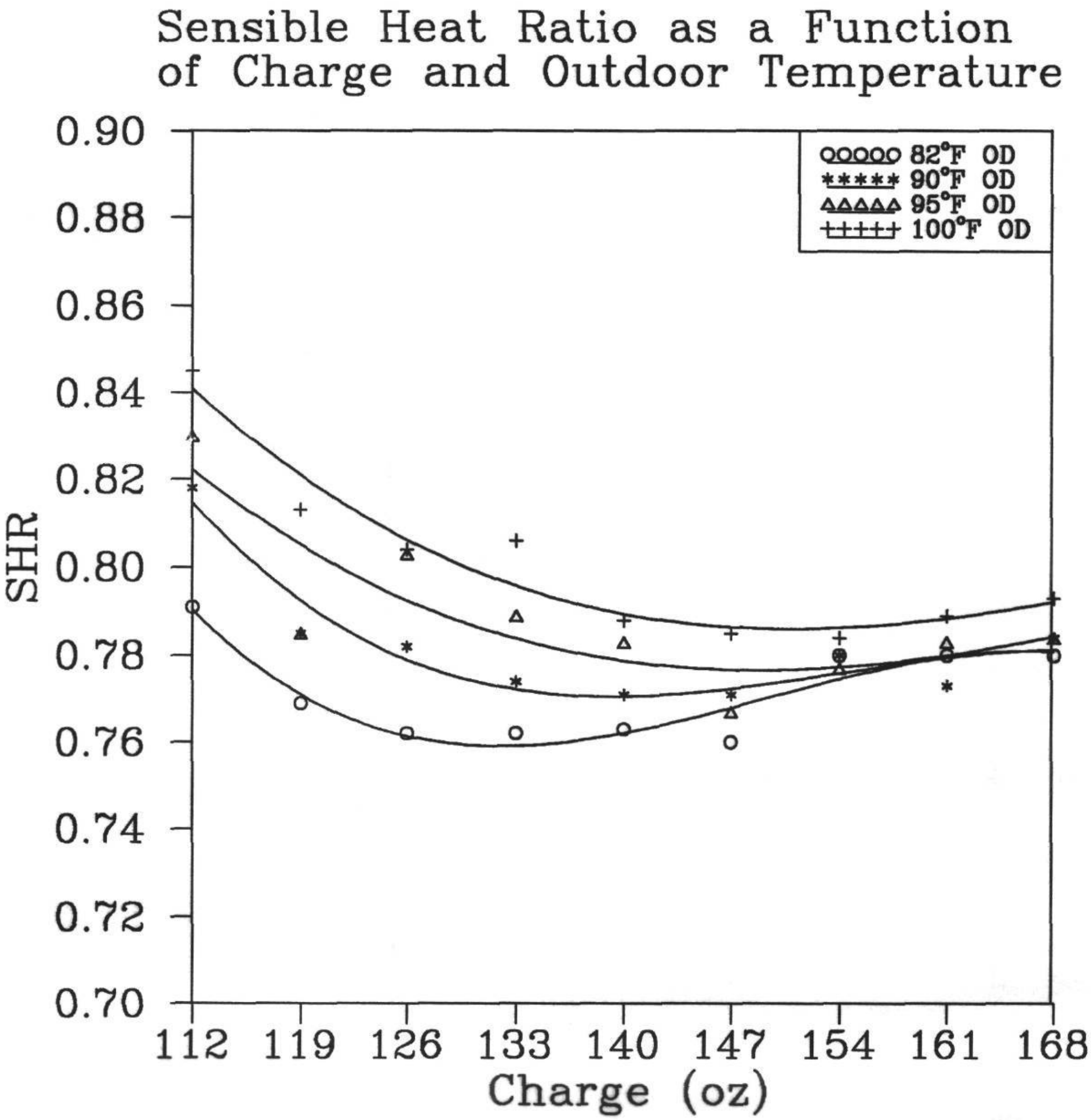

Figure 3.10 - Sensible Heat Ratio as a Function of
Refrigerant Charge and Outdoor Temperature

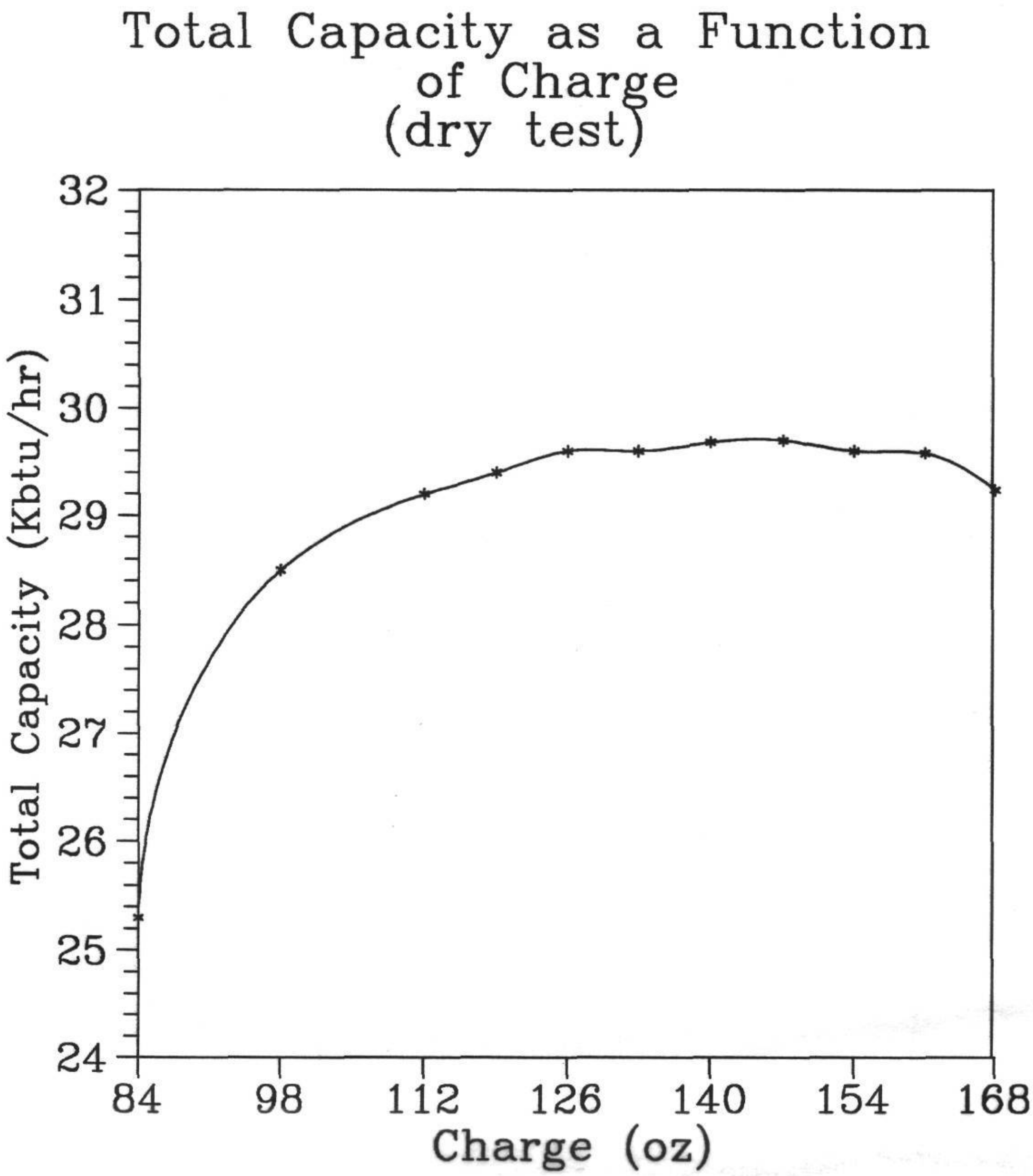

Figure 3.11 - Total Capacity as a Function of Refrigerant Charge (dry coil)

undercharging. It was constant from 10% undercharging to 10% overcharging. The capacity dropped slightly to 29.2 Kbtu/hr for 20% undercharging.

Figure 3.12 shows the coefficient of degradation (C_D), which is a measure of the efficiency loss due to the cycling of the unit. C_D peaked at 0.155 for 40% undercharging. It dropped to 0.13 for 10% undercharging and remained constant through 10% overcharging condition. C_D decreased slightly for 15% and 20% overcharged conditions.

Upon compressor start-up, the cooling capacity of an air conditioner increases to its steady-state value gradually, rather than instantaneously. This lack of an instantaneous response leads to lower average capacities and efficiencies than the respective steady state values. The first few minutes after start-up are the most crucial for a cyclic losses. The start-up losses are results of off-cycle phenomena. One of the major losses is due to the refrigerant migration from the condenser to evaporator [11].

As the amount of refrigerant charge increased in the system, the unit capacity increased during the first minute of the start-up. During the firsttwo minutes of the start-up, the unit capacity rose to about 27 KBtu/hr for 20%, 15%, 10%, and 5% undercharging tests. As shown in Figure 3.13, the unit capacity in first two minutes for 40% and 30% undercharged conditions were 21 and 25 KBtu/hr,

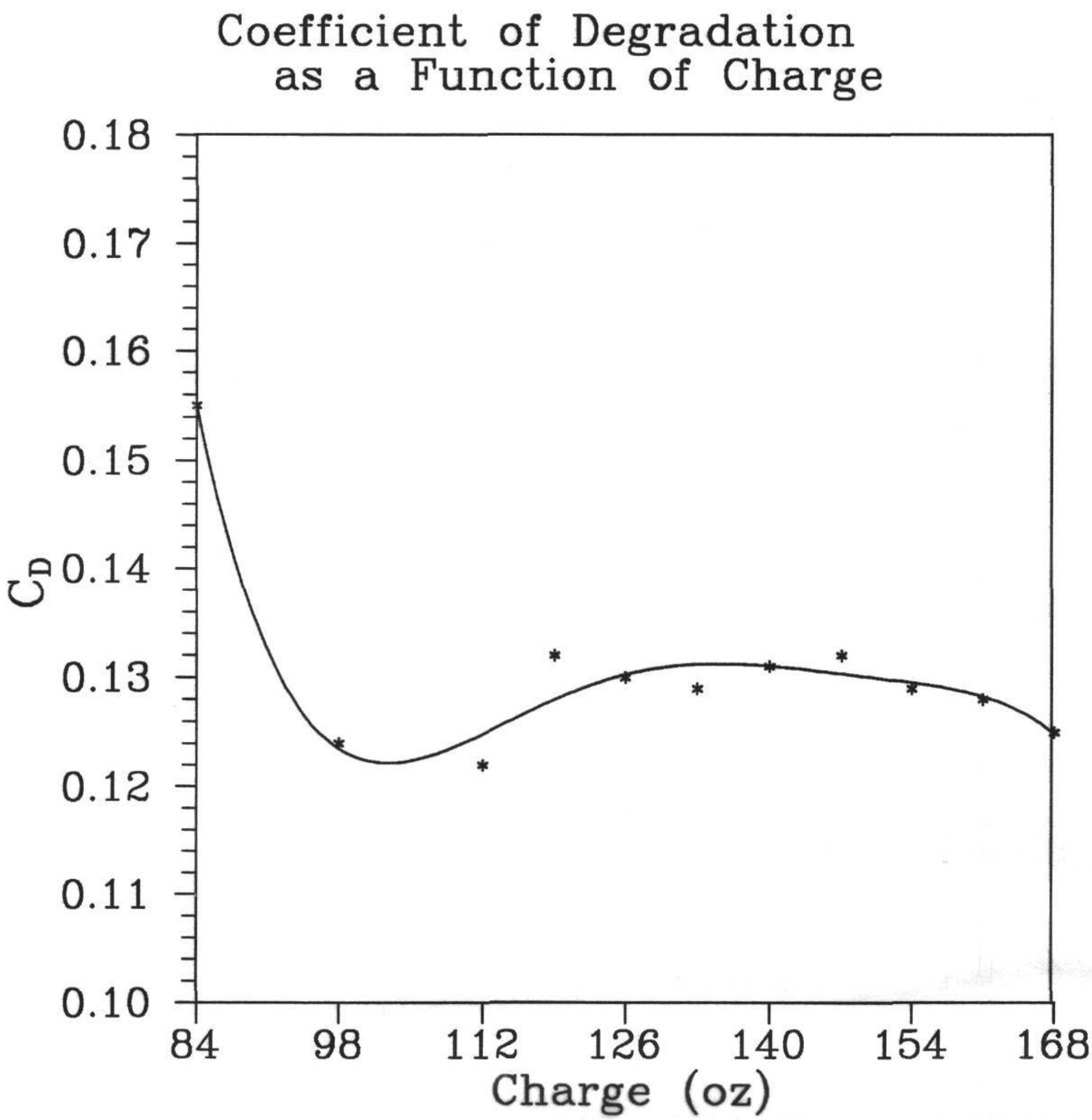

Figure 3.12 - Coefficient of Degradation for Under and Overcharging Tests

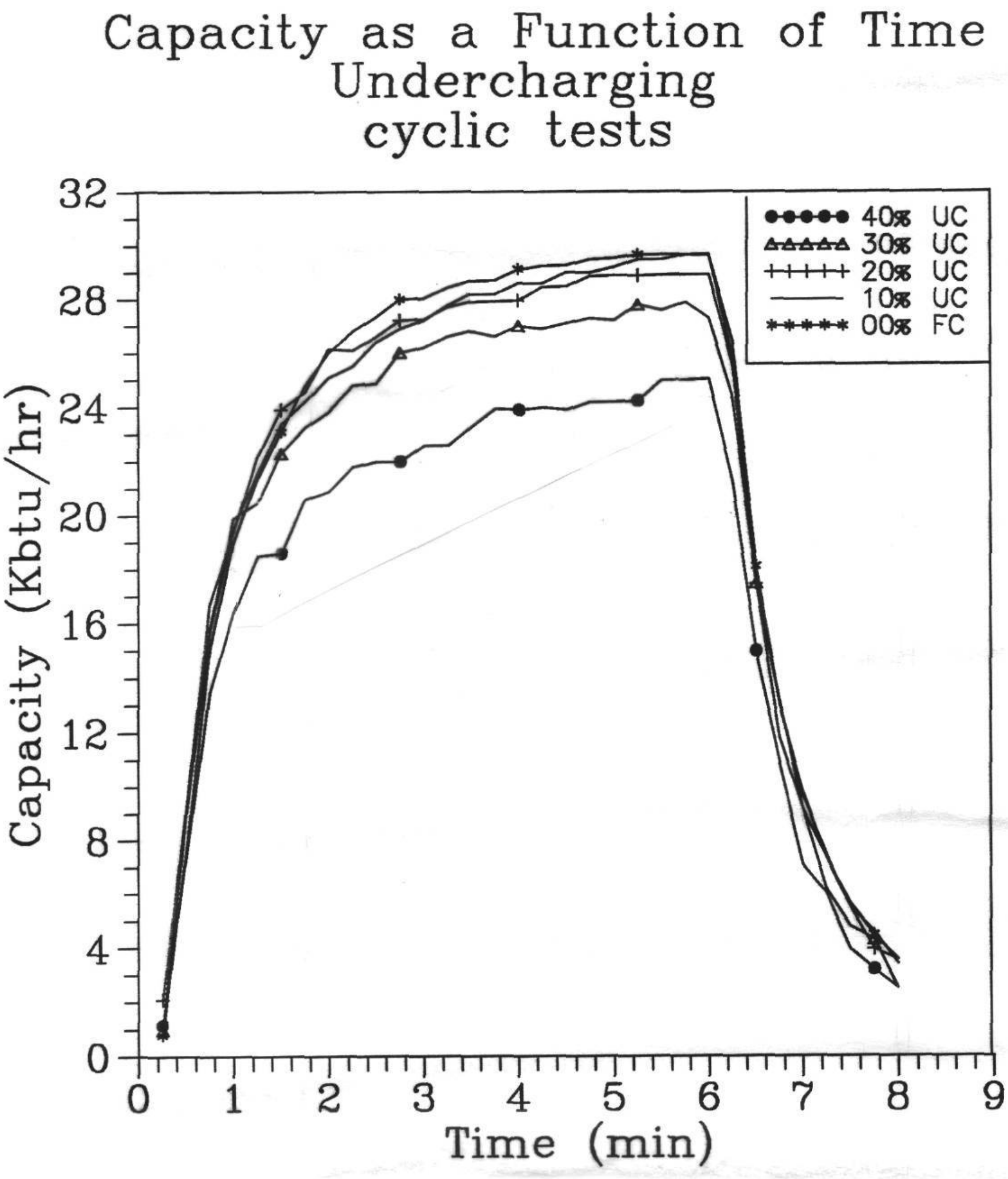

Figure 3.13 - Capacity as a Function of Time for Full Charge and Undercharged Cyclic Tests

repectively. It increased to approximately 28 KBtu/hr for proper charging. The capacities for overcharging were identical to that for the fully charged condition (Figure 3.14). During the last two minutes of "on" time, the total capacity leveled off to about 30 KBtu/hr for all the overcharging and fully charged tests. However, it reached to 25 and 28 KBtu/hr for 40% and 30% undercharged conditions, respectively. During the first two minutes of compressor shut-off, the total capacity dropped off quickly for all the tests.

ENERGY EFFICIENCY RATIO (EER)

EER is a ratio calculated by dividing the net cooling capacity in Btu/hr by the power input in watts (w) at any given set of rating conditions, expressed in Btu/wh.

1. Steady State Tests (Wet Coil)

Figure 3.15 shows the EER as a function of outdoor room temperature and refrigerant charge. As the outdoor room temperature increased, EER decreased for a given charge. The maximum EER occurred at 82^{o}F outdoor room temperature for the 15%, 10%, and 5% undercharged conditions. As the outdoor room temperature increased, the EER curves were approximately constant from 15% undercharging to 5% overcharging conditions. For instance, the EER curves for 95^{o}F outdoor temperature varied from 9.26 to 9.11 over the

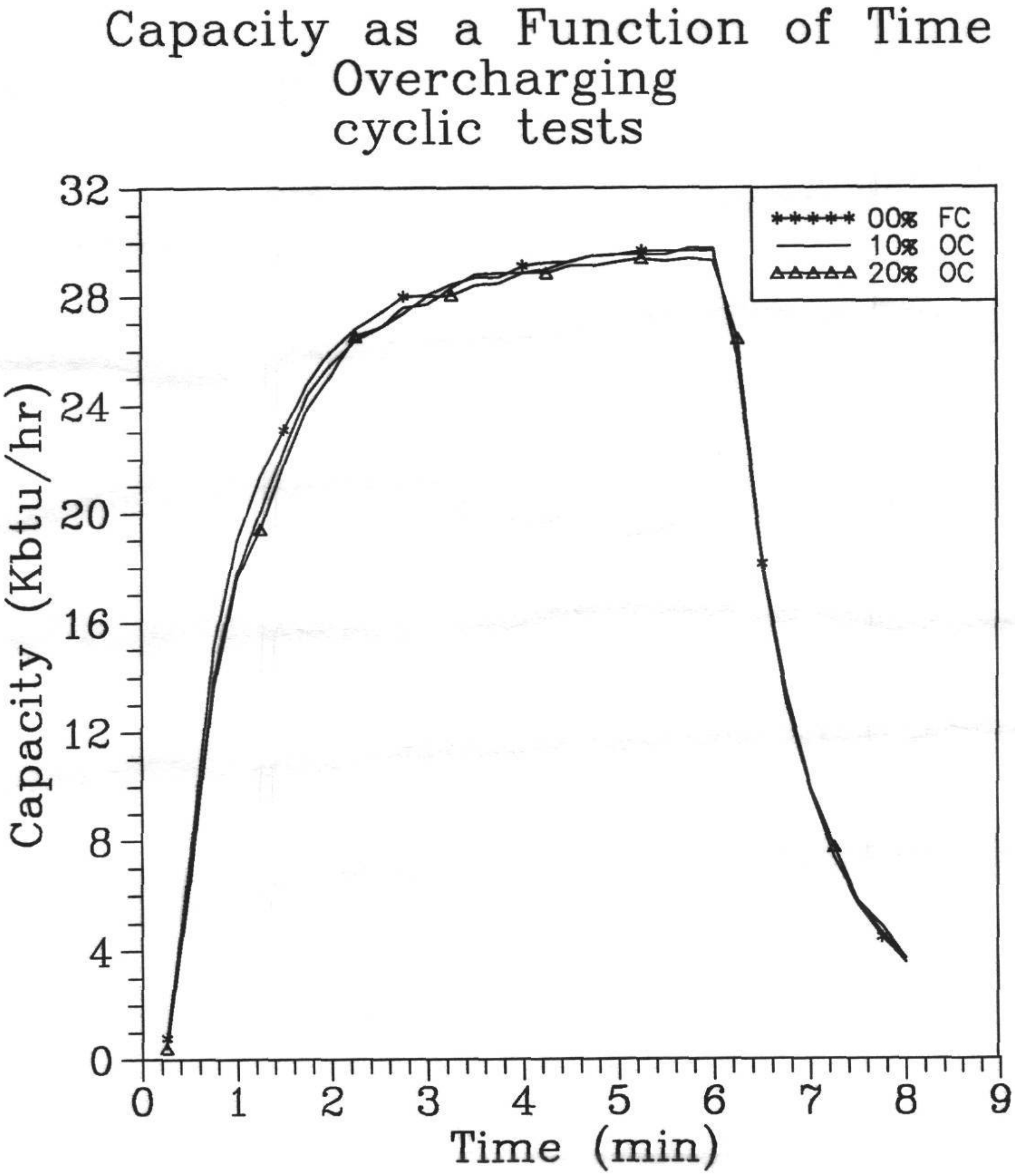

Figure 3.14 - Capacity as a Function of Time for Full Charge and Overcharged Cyclic Tests

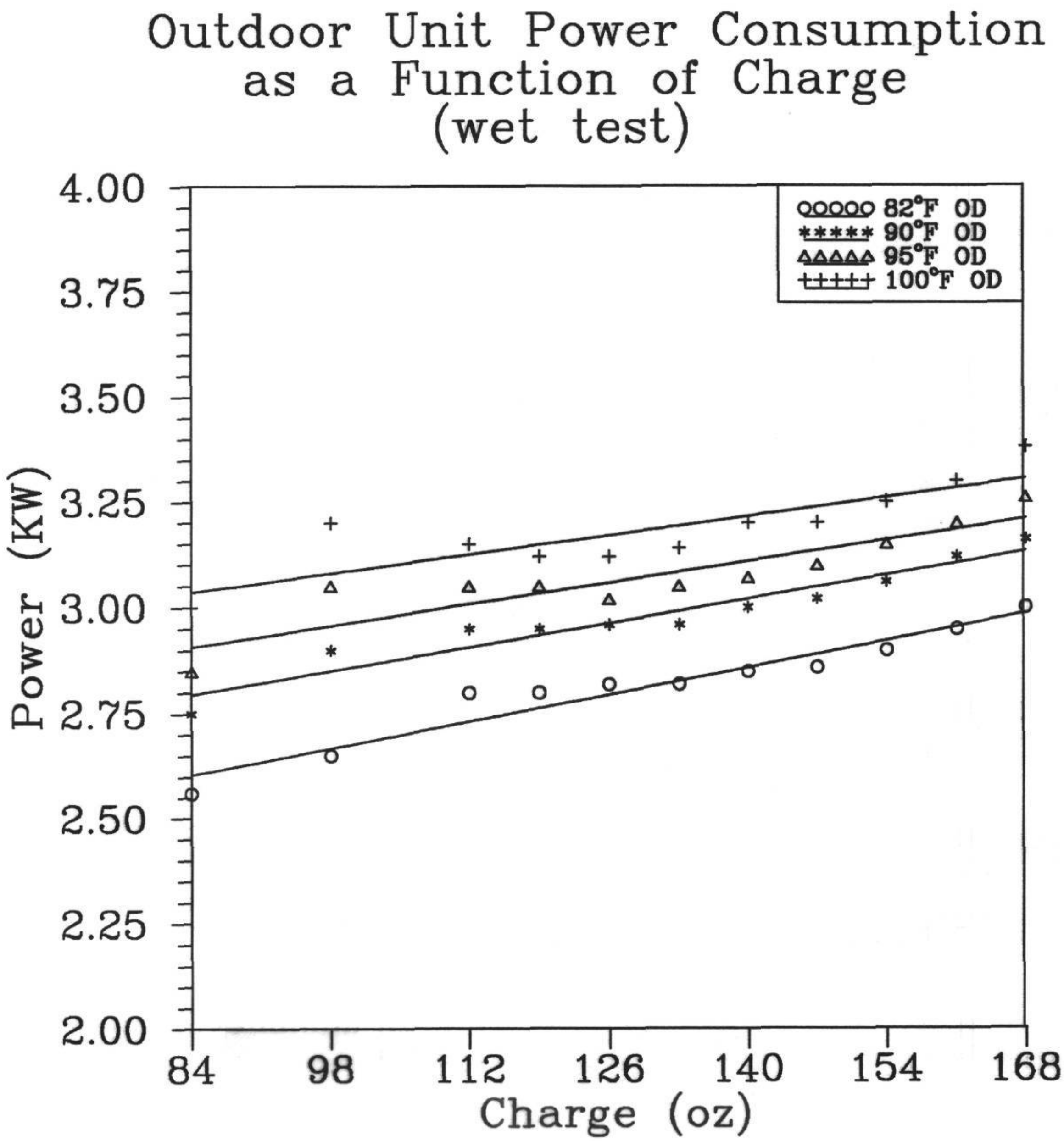

Figure 3.15 - Outdoor Unit Power Consumption as a Function of Charge and Outdoor Temperature (wet coil)

range of 15% undercharged to 5% overcharged conditions. For full charging, EER dropped by 22% when outdoor temperature increased from 82^{o}F to 100^{o}F. The drop in EER was more noticeable for the 40%, 30%, and 20% undercharged than overcharged conditions at the four outdoor temperatures. The reason for lower EER at higher outdoor temperature was due to higher compressor power consumption and lower unit capacity. The increase in power (kw) was due to the higher condensing temperature of the unit. The condenser outlet temperature decreased as the refrigerant charge in the system increased. This drop was due to the increase in the subcooling. As the condensing temperature increased, the power (kw) to the compressor increased too.

Figure 3.16 shows the power consumption of the outdoor unit as a function of outdoor room temperature and charge. For fully charged condition, the power (kw) increased by 12% when the outdoor temperature increased from 82^{o}F to 100^{o}F. However, for the same increase in outdoor room temperature, the power increased by 12.5 for 20% under and overcharging tests. As the charge in the unit was increased systematically, the power (kw) increased linearly. At 82^{o}F outdoor temperature, the power (kw) increased by 5% when the unit's charge was increased from 20% undercharging to 20% overcharging.

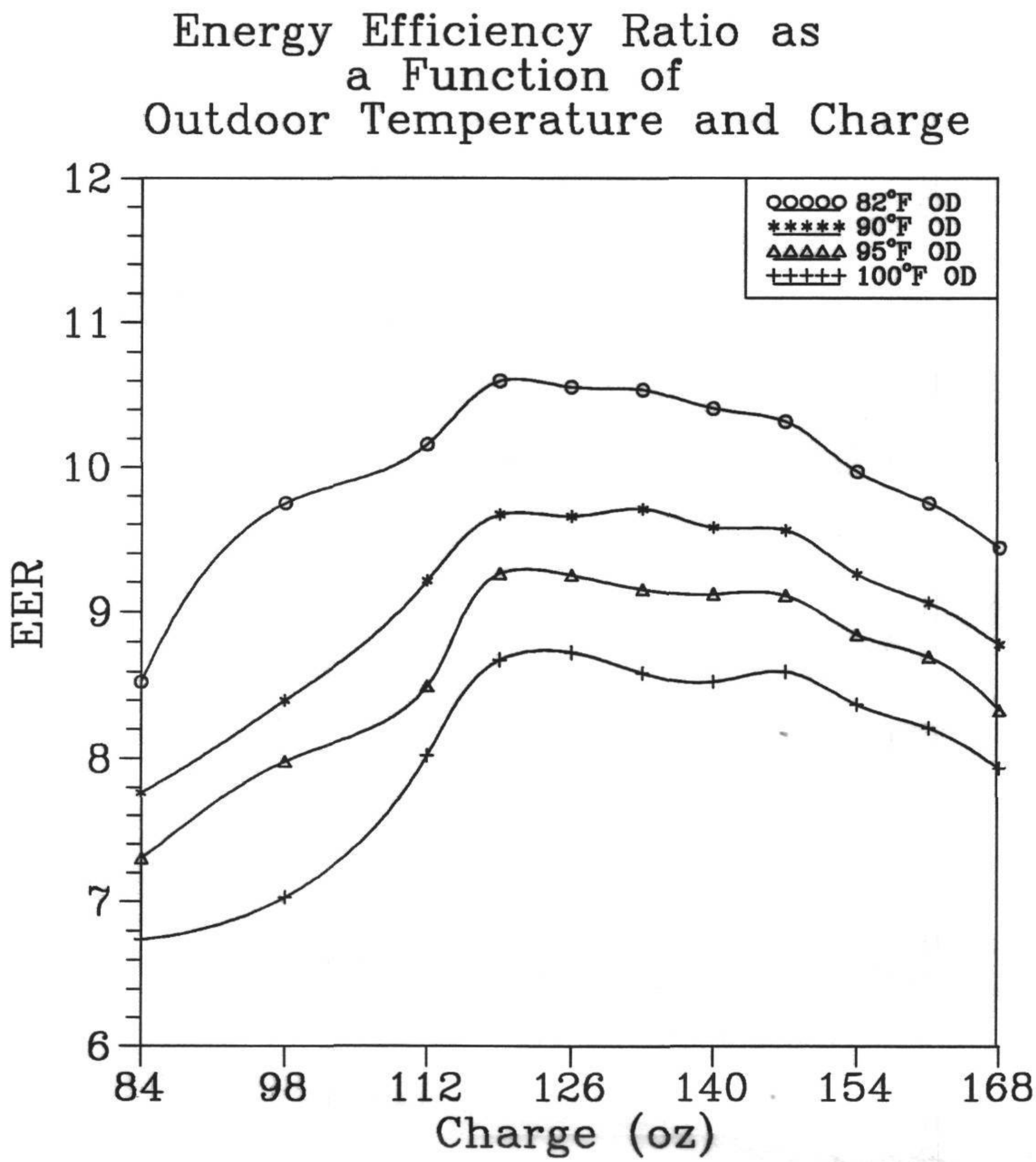

Figure 3.16 - Energy Efficiency Ratio as a Function of Outdoor Temperature and Charge (wet coil)

2. Steady State & Cyclic Tests (Dry Coil)

The steady state dry coil (C) and cyclic (D) tests were performed on the unit for under and overcharged conditions. EER_C for the steady state dry coil test is shown in Figure 3.17 as a function of charge. The maximum EER_C peaked at 9.55 for 10% undercharged test. The EER_C was 9.46 for full charge (140 oz). It dropped by 8% for the 20% overcharge. The drop in EER_C for 20% overcharge was more drastic than 20% undercharge. The decrease in EER_C was due to the increase of compressor power and lower capacity. The unit power consumption for steady state dry coil is shown in Figure 3.18. For 40% undercharging, the power was 2.45 kw. It increased to 2.66 kw for 30% undercharging condition. The power consumption (kw) was constant for 30%, 20%, 15%, 10%, 5% undercharging and fully charged conditions. For instance, the power consumption (kw) increased linearly for overcharged conditions. It increased by 4.5% for 20% overcharged test.

The cooling cyclic tests were conducted by cycling the compressor 6 minutes "on" and 24 minutes "off". The Energy Efficiency Ratio for the cyclic test (EER_D) is a ratio calculated by dividing the total sensible cooling (in Btu/hr) during first eight minutes by the total power (kw) during "on" time. Figure 3.19 shows the Energy Efficiency Ratio for cyclic test (EER_D). The maximum EER_D was at 126 oz (10% undercharging). It dropped to 7.65 for 40%

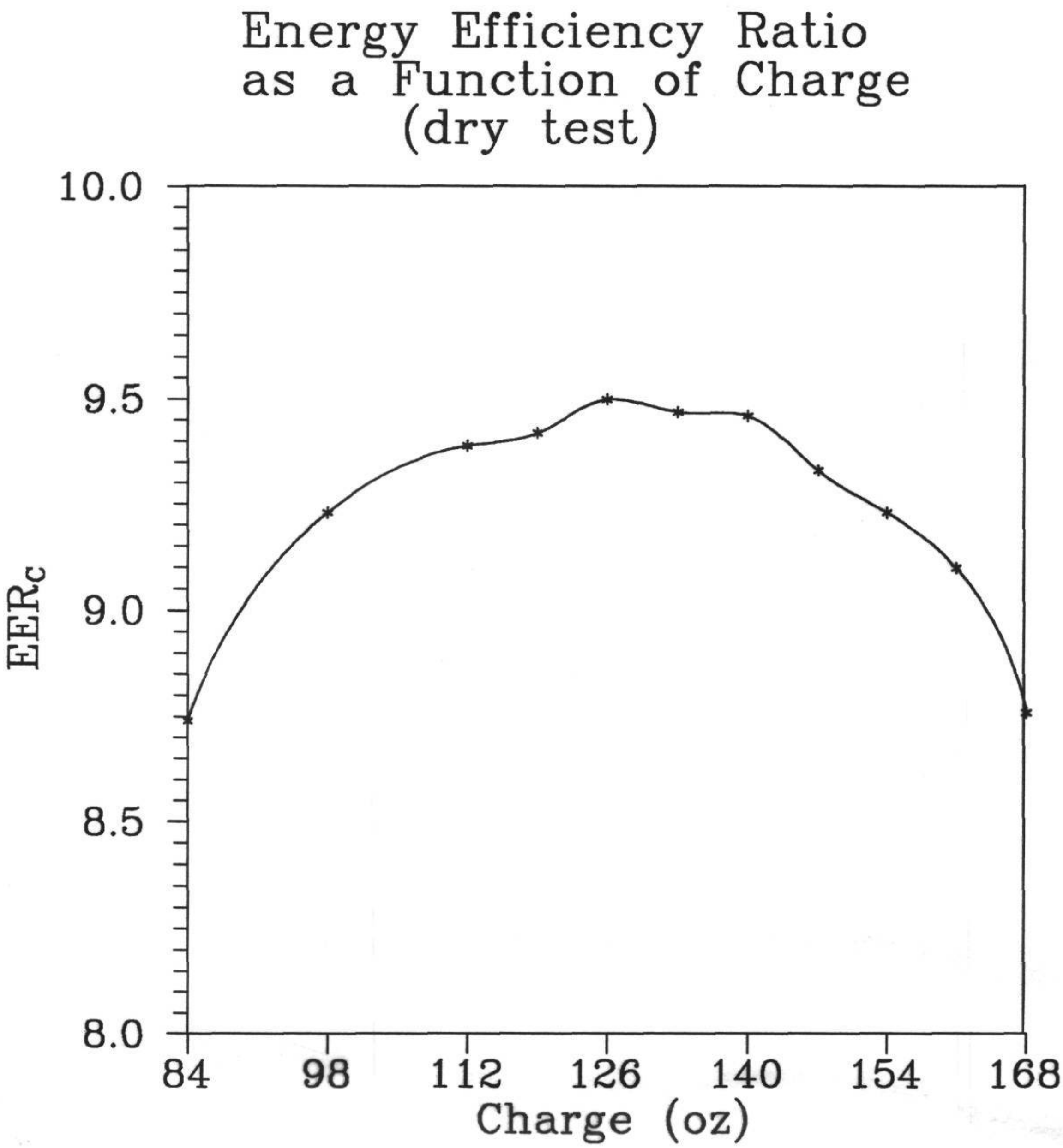

Figure 3.17 - Energy Efficiency Ratio as a Function of Charge (dry coil)

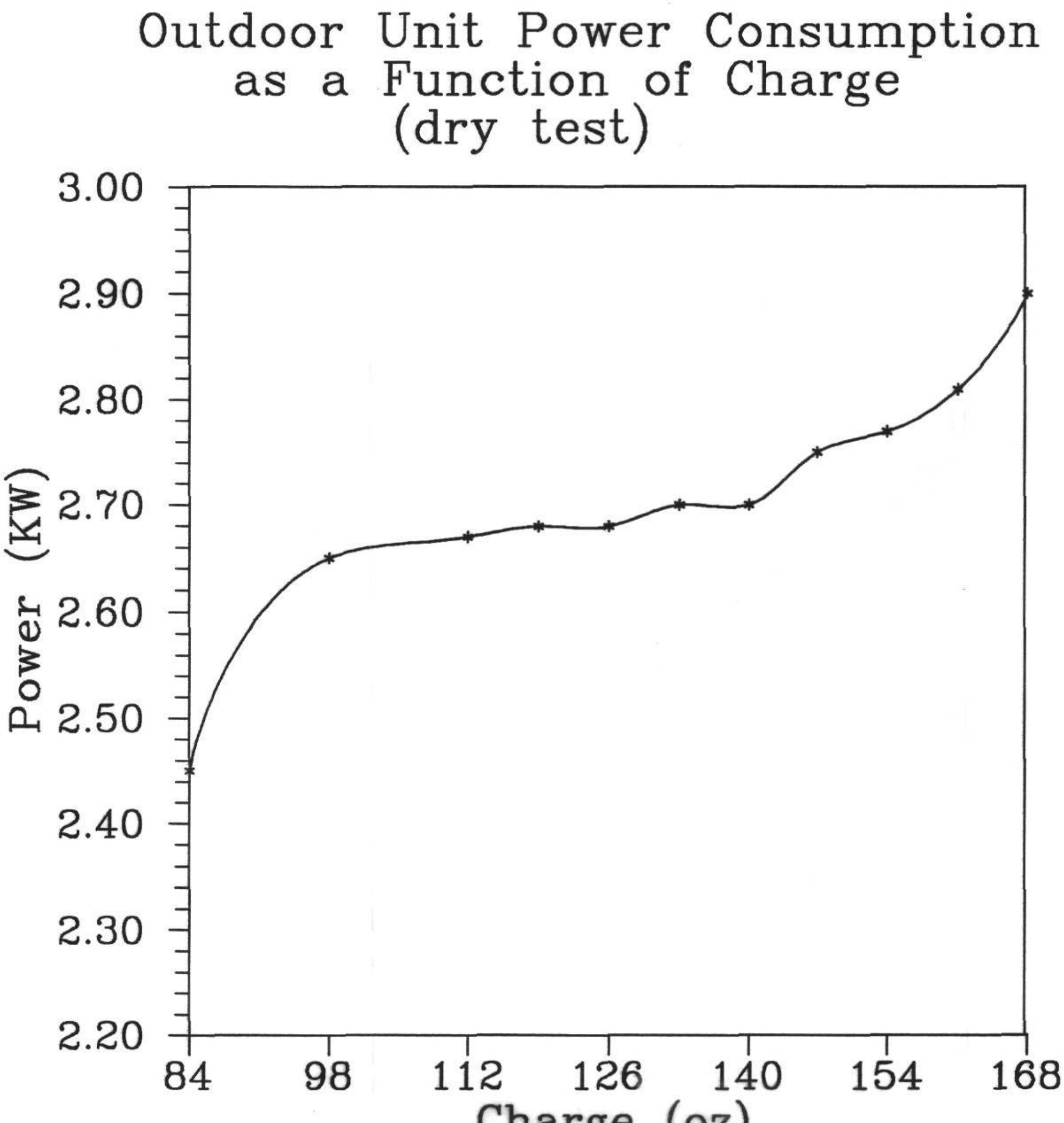

Figure 3.18 - Outdoor Unit Power Consumption as a Function of Charge (dry coil)

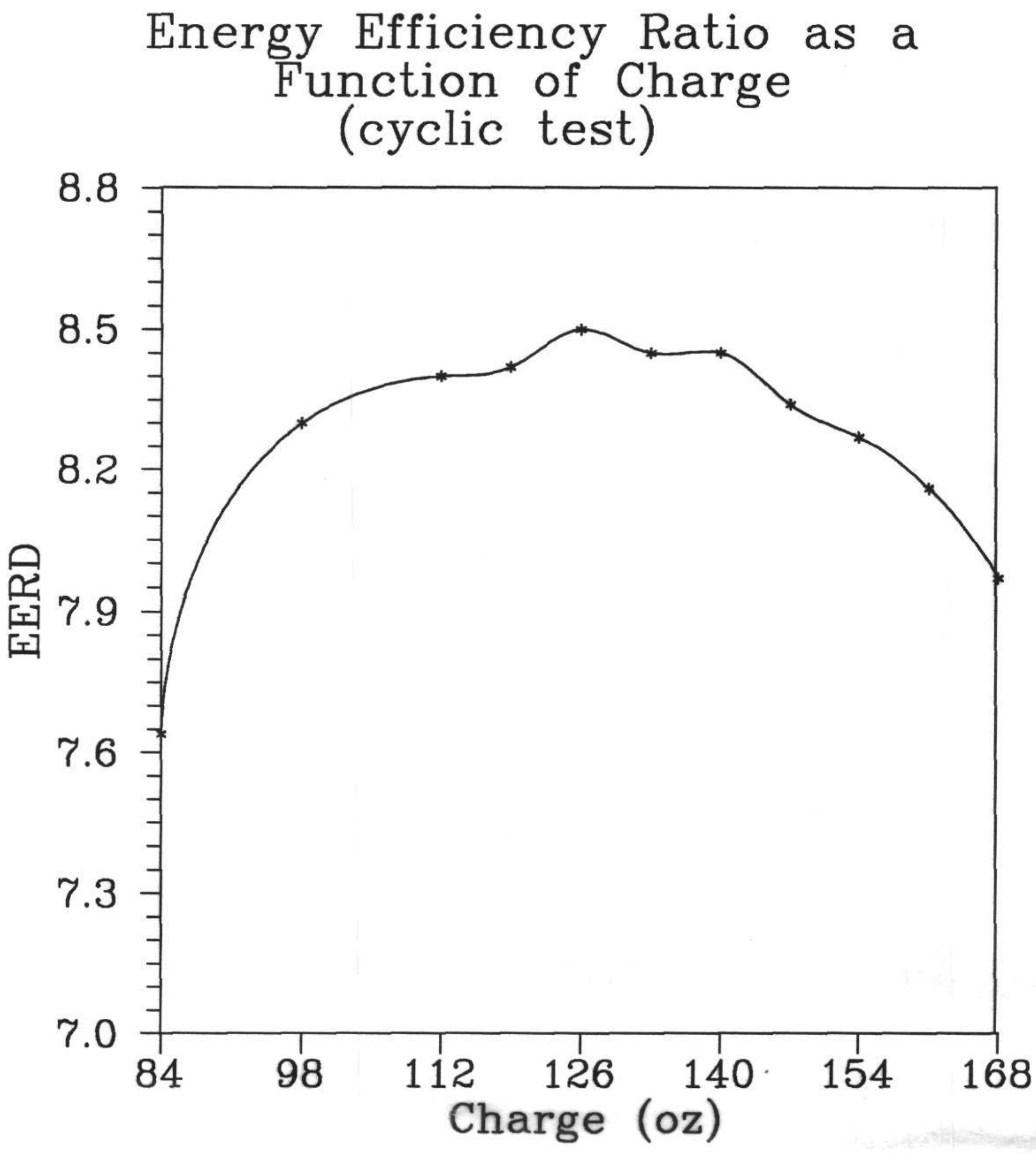

Figure 3.19 - Energy Efficiency Ratio as a Function of Charge (cyclic test)

undercharging. The EER_D dropped to 7.97 for 20% overcharging. The decrease in EER_D was more noticeable for 20% overcharging than 20% undercharging.

SEASONAL ENERGY EFFICIENCY RATIO (SEER)

SEER is a measure of the seasonal efficiency of the unit. The unit SEER as a function of charge is shown in Figure 3.20. At full charge condition, the SEER was 9.7. The SEER curve peak was 9.86. It occurred at 15%, 10%, and 5% undercharged conditions. It dropped to 8.9 for 20% overcharged. The drop in SEER was more dramatic for the 20% overcharged than 20% undercharged conditions. The interesting result is that the maximum SEER occurred from 15% to 5% undercharged than full charge.

A 20% decrease in charge for the capillary tube produced a 27% decrease in SEER. In contrast, the SEER only dropped 3.5% for 20% undercharging for the TXV study.

For the capillary tube system tested earlier, the SEER was found extremely sensitive to changes in refrigerant charge the the TXV study.

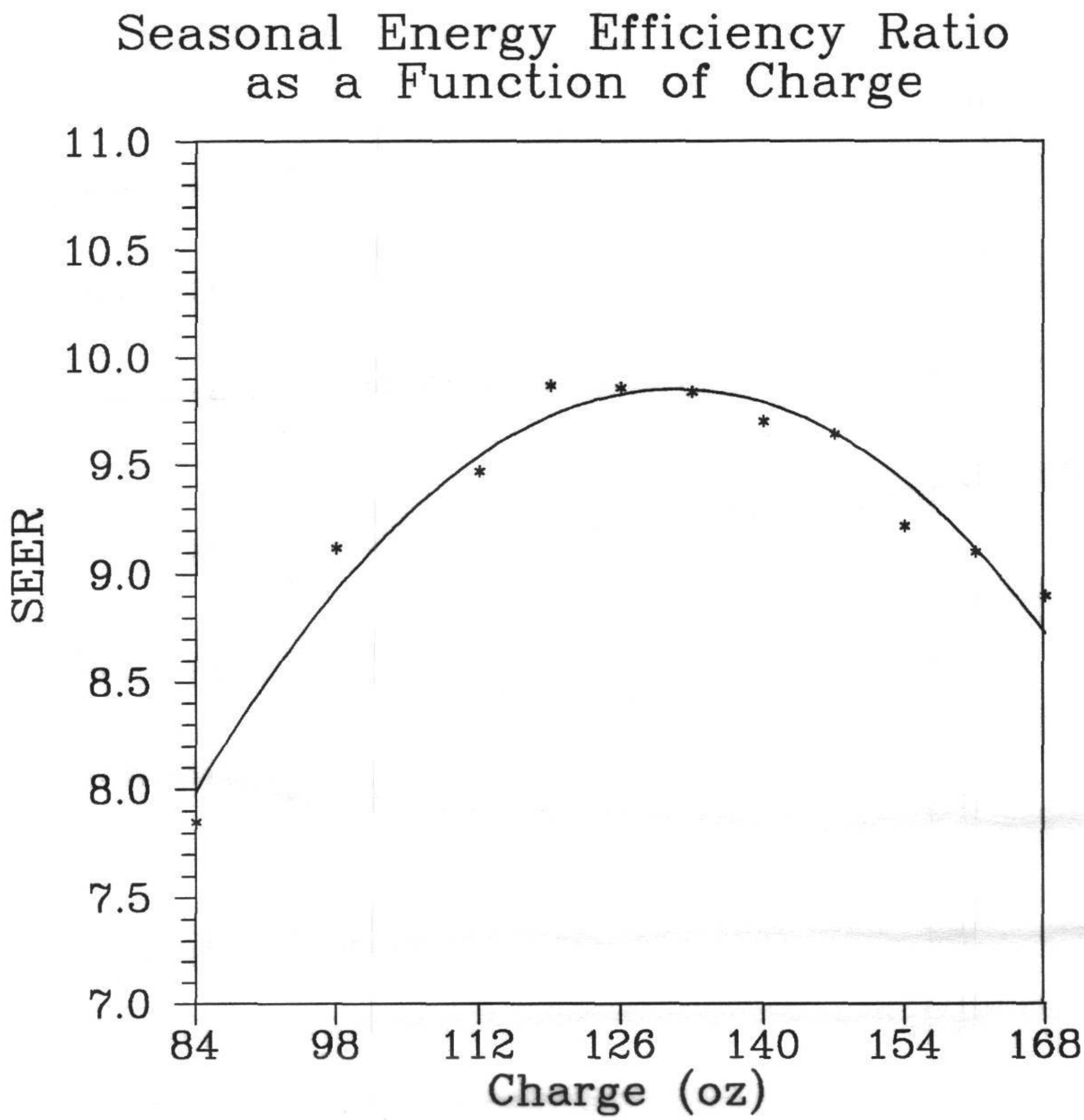

Figure 3.20 - Seasonal Energy Efficiency Ratio as a Function of Refrigerant Charge

CHAPTER 4

CONCLUSIONS AND RECOMMENDATION

The objective of this investigation was to quantify the effect of the improper refrigerant charge on the steady state and cyclic performance of a residential air conditioner system with thermal expansion valve. This study was a continuation of an earlier study on capillary tube systems [5].

A 3-ton split system air conditioner with thermal expansion valve was instrumented to evaluate the steady state and cyclic performance of the air conditioner for under and overcharged conditions. The charge in the system was systematically varied to determine its effect on the capacity, power consumption, EER, cycling capacity, coefficient of degradation, and SEER. The tests were conducted under four different outdoor temperatures.

The results of the experimentation showed that the total capacity (wet and dry coil tests) and EER decreased with increasing outdoor temperature for the under and overcharged cases. The sensible heat ratio and outdoor unit power consumption increased as the outdoor temperature increased.

The effect of outdoor temperature on superheat and subcooling temperatures were investigated. As the outdoor temperature increased, the subcooling rose slightly for the TXV. No subcooling was found for 40%, 30%, 20%, and 15% undercharging. For the capillary tube system, the subcooling was constant for undercharging and decreased for overcharging tests. The superheat at the outlet of the evaporator decreased as the outdoor temperature increased for the capillary tube. For 20%, 15%, and 10% overcharging, the refrigerant at the outlet of the evaporator was saturated[5]. For the TXV system, the superheat dropped by 2^{O}F where outdoor temperature increased from 82^{O}F to 100^{O}F. There was no saturated vapor at the outlet of the evaporator for any of the charging conditions with the TXV.

In general, the degradation of performance was larger for undercharging (40%, 30%, and 20%) than that for overcharging. The measure of seasonal performance, SEER, peaked between 10% and 5% undercharging. It dropped from 9.7 (FC) to 9.4 for 20% undercharging while only dropping to 8.9 for 20% overcharging. The data for capacity and EER showed similar trends. The EER and the total capacity (wet and dry tests) of the unit were approximately constant over a wide range of charge for a given outdoor temperature.

The TXV maintained a relatively constant superheat by regulating the refrigerant flow rate in the system. The data indicated that the refrigerant flow rate were similar for all the under and overcharging tests except for 40%, 30%, and 20% undercharging. The similar flow rates would explain why the capacities remained relatively constant from 15% undercharging to 10% overcharging.

A measure of performance of interest to electric utilities is the capacity and power during the hottest part of the summer days. The 100^{O}F test would provide some hints at this performance. The capacity was constant from 15% undercharging to 15% overcharging at 30.8 kBtu/hr. It dropped to 28.5 kBtu/hr for 20% undercharging and 30.2 kBtu/hr for 20% overcharging. The EER was constant from 15% undercharged to 5% overcharged conditions at 8.73 for 100^{O}F outdoor temperature. It dropped to 8.0 for 20% undercharging and 7.9 for 20% overcharging.

At fully charged condition (140 oz), The SEER was 9.7 and 9.45 for TXV and capillary tube, respectively. With a TXV, the maximum SEER was 9.85 at 15%, 10%, and 5% undercharging. It was 9.7 at full charge for the capillary tube study. For 20% undercharging, the degradation of performance was larger for the capillary tube than the thermal expansion valve case. The SEER dropped to 7 (27% reduction) with capillary tubes and only to 9.5 (3.5% reduction) with the TXV. In case of 20% overcharging, the SEER dropped to 8.55 (11.4% reduction) and 8.9 (9% reduction) for capillary tube and TXV, respectively.

According to the manufacturer's specification, the unit was charged to 14^{O}F subcooling at 95^{O}F outdoor temperature. To obtain 14^{O}F subcooling, the unit was charged to 140 ounces of R-22 refrigerant. The measured data indicated that the maximum was obtained for approximately 10% undercharging (126 oz). If the system were charged with this amount of refrigerant, there would be a saving of 14 ounces of refrigerant in the system over the "proper" charge. At the same time there would be a small increase in SEER.

Another conclusion from the TXV study is the small variation of capacity, EER, and SEER to refrigerant charge. The results from earlier studies on capillary tubes suggested that a fixed expansion device would be extremely sensitive to changes in refrigerant charge. For instance, a 10% undercharging would result in a 11.5% reduction in the SEER for the system with capillary tube in early study. However, the same condition would lead to 1.5% increase in the SEER for the TXV case.

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APPENDIX A

Equipment Used in the Testing Apparatus

EQUIPMENT	SIZE/RANG	MAKE/MODEL	ACC[1]
Air Conditioner	3 Ton	Trane TTX736A100A1 /BWV736A	
Outdoor Fan	1/4 HP	Propeller	
Datalogger	65 Chan.	Acurex/Autodata	
Watt/Watt-hr Transducer	20 kw	Ohio Semitronics/ W-53	0.5%
Pressure Transducer	0-300 Psig 0-500 Psig	Foxboro/1225-12G-K-42	0.5%
Turbine Flow Meter	2-20 GPM	Flow Measurement System, Inc. FM-6-8N5-L142	0.5%
Dew Point Sensor	0-100°F	General Eastern/ DEW-10	0.5°F
Thermocouples	30 AWG	Omega/Type T	0.5°F

[1] Percentages are percent of span or range. Temperatures are deviations (+/-).