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

FINAL REPORT

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ABSTRACT

The effect of the improper charging on the performance (capacity, EER, power consumption, SEER, and coefficient of degradation) of a residential air conditioner during the steady state (wet and dry coils) and cycling operation was investigated. The fully charged condition was established as a base case. A full charge was obtained charging the unit to the superheat specified by the manufacturer's charging chart for specific set of indoor and outdoor temperatures. Once the full charge was determined, the unit was subjected to 20%, 15%, 10%, and 5% under and overcharging of refrigerant (by mass). The fully charged tests were compared to 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 investigation of improper charging showed that the total capacity, EER, and SEER were more sensitive to undercharging than overcharging conditions. A 20% undercharge resulted in a 21% reduction in SEER while a 20% overcharge produced a 11% reduction in SEER. Other data such as refrigerant flow rate, sensible heat ratio, superheat and subcooling are 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 must be charged with an optimum amount of refrigerant. Over and undercharging the system will result in degraded performance when compared to an optimally charged unit.

From the operational standpoint, an undercharged system will cause a loss in capacity and a reduction in efficiency. Undercharging can create an abnormally high superheat temperature which could adversely affect the cooling of the compressor motor windings. It is expected that the electrical demand of the compressor would decrease, since the compressor is pumping less dense vapor.

An overcharged system should cause a slight increase in system capacity over the nominal, but with a higher electrical demand. Overcharging the system should cause the evaporator superheat to decrease because of the larger refrigerant flow. The compressor would then be forced to pump against a higher head pressure. With a large enough overcharge, liquid refrigerant could be introduced into the compressor which would reduce the life expectancy of the compressor. Thus accurate charging is important for providing best performance and protecting the compressor.

The objective of this study was to quantify the degradation of air conditioner performance (capacity, coefficient of performance(COP), etc.) during under and overcharging of system. A medium efficiency air conditioner was tested under steady state conditions at four outdoor air temperatures: 82^{o}, 90^{o}, 95^{o}, and 100^{o} F. Standard Air Conitioning and Refrigeration Institute(ARI) tests [5] were also run on the unit. Tests were performed for 5, 10, 15, and 20% under and overcharging of refrigerant (by mass).

Chapter 2 provides a literature review of the subject.

In Chapter 3, a description of the test setup and experimental procedure will be discussed. A 3-ton Trane air conditioner with capillary tube expansion (model

TTX736A100A1/BWV736A) was used in this study. The air conditioner was tested in the psychrometric rooms at the Energy System Laboratory. Detailed description of the dry and wet coil steady state tests as well as the cyclic test are provided.

Results and discussion are presented in Chapter 4. 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 pressures are also reported.

Major conclusions from this study and recommendations for further study are provided in Chapter 5. Some of the recommendations center on the relationship between the amount of refrigerant and capacity, EER, SEER, and C_D.

CHAPTER 2

LITERATURE REVIEW

The subject of improper refrigerant charge has been given little attention in the literature. The Trane Company performed a study on an eight year old residential air conditioner [6]. The objective was to determine the improvements in efficiency which could be achieved by proper refrigerant charging and maintenance. Their findings showed that a twenty percent improvement in efficiency by cleaning the coils (evaporator and condenser) and adjusting to the proper refrigerant charge. Trane Co. found that a ten percent drop in refrigerant charge caused efficiency to drop off twenty percent.

The importance of proper refrigerant charge has been investigated by Houcek and Thedford for Texas Power and Light (TP&L) in a laboratory test of a 1.5 ton split system unit[7]. They tested a properly charged unit at 5 outdoor temperatures: 70°, 75°, 82°, 95° and 100°F. The indoor conditions were maintained at 80°F dry bulb and 67°F wet bulb. The unit was then overcharged by 23% (by mass) and tests run at 82°F and 95°F outdoor temperatures. The system was then undercharged by 23% (by mass). All tests were steady state.

Figures 2.1 and 2.2 show the effect of over/undercharging on the capacity and power of the air conditioner in TP&L Study [7]. Undercharging caused a rapid decline in total capacity as the outdoor ambient increased. For instance, at 82^{o}F, a 23% loss in charge translated into a 23% reduction in capacity. At 95^{o}F, the reduction was 38%. The TP&L data would indicate undercharging has its most adverse impact at higher outdoor ambient temperatures.

While overcharging the unit increased its capacity, Houcek and Thedford indicated flooding in the compressor was experienced for outdoor temperatures ranging from 70^{o} to 95^{o}F [7].

The effect of undercharging was dramatic on the EER (Figure 2.3). For instance, at 95^{o}F, the EER dropped from 8.31, for properly charged, to 5.49 for the 23% undercharging. This would result in a 52% increase in energy use. Overcharging had little impact on the EER. At the same time the electrical demand (kw) increased at the a slower rate.

They concluded that: "An undercharged condition will create abnormally high superheat which has an adverse effect on compressor motor winding cooling. The long term effect will be the eventual breakdown of motor winding insulation with premature compressor failure as a result. In extreme cases, due to the repeated opening of the internal motor overload protector, a much faster failure could occur."

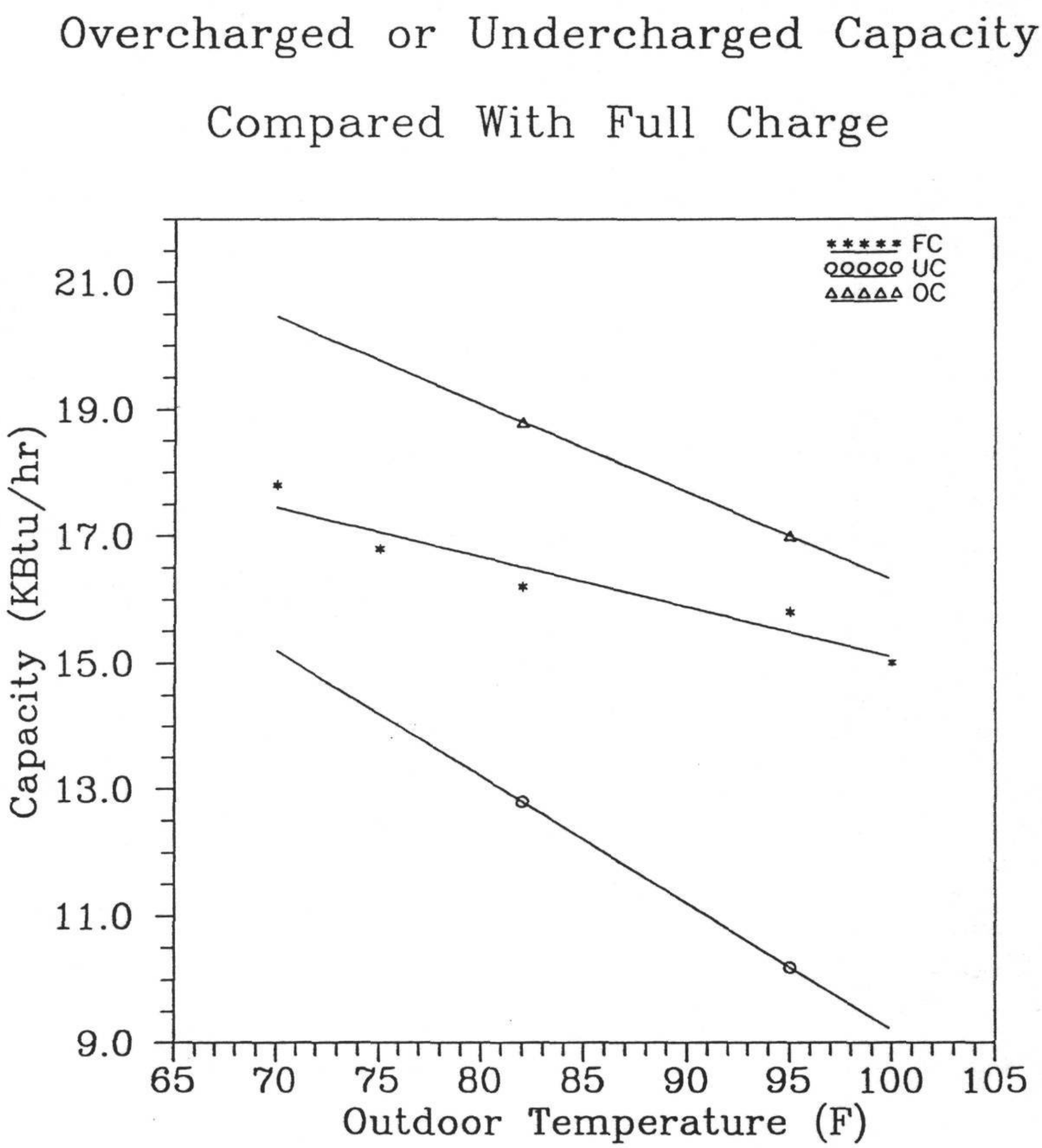

Figure 2.1 - The Effect of Over/undercharging on the Capacity (adapted from ref.7)

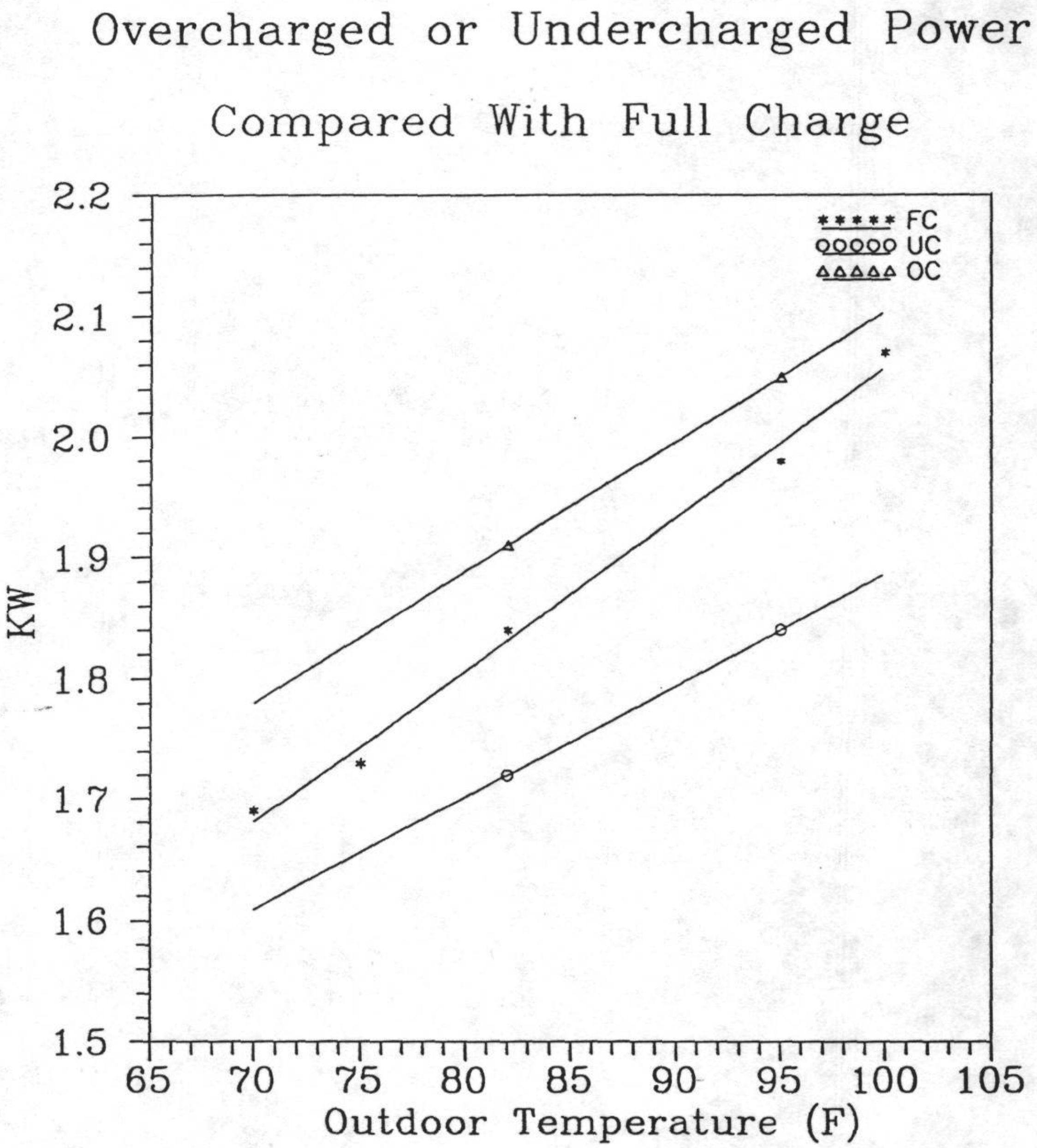

Figure 2.2 - The Effect of Over/undercharging on the Power (adapted from ref.7)

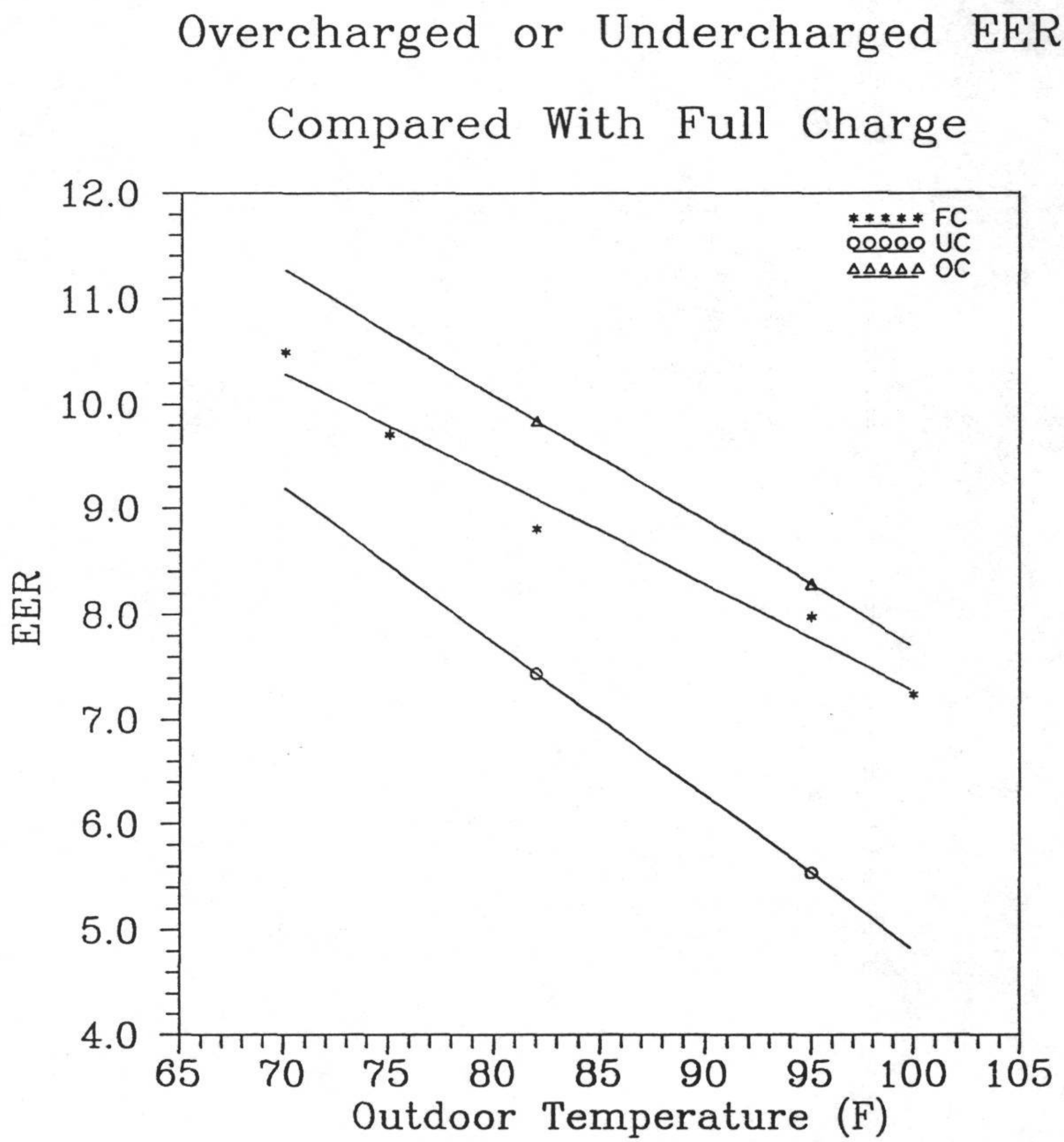

Figure 2.3 - The Effect of Over/undercharging on the EER (adopted from ref.7)

A. A. Domingorena [8] conducted a laboratory test to determine the performance of a 3-ton air-to-air heat pump as a function of R-22 refrigerant charge in the heating mode. The tests were conducted with 50°F outdoor air and 70°F indoor air temperatures. The test unit did not employ a suction-line accumulator. Figures 2.4 and 2.5 show the heating capacity and the COP as a function of refrigerant charge.

The optimum heating capacity was approximately 40 KBtu/hr for the nameplate refrigerant charge of 7.5 lbs. As shown in Figure 2.4, undercharging caused a steep decline in total heating capacity as the refrigerant charge decreased in the unit. For 47% and 20% undercharge by weight, the heating capacity dropped to 13.5 KBtu/hr and 30 KBtu/hr, respectively. The effect of overcharging had little effect in the heating capacity.

The effect of undercharging was dramatic on the COP (Figure 2.5). At 50°F outdoor temperature, the COP dropped from 2.23 for properly charged, to 1.85 for the 20% undercharging.

A. A. Domingorena [8] found the COP was essentially constant in the overcharge range and dropped off in the undercharge range, but less sharply than heating capacity and refrigerant mass flow rate.

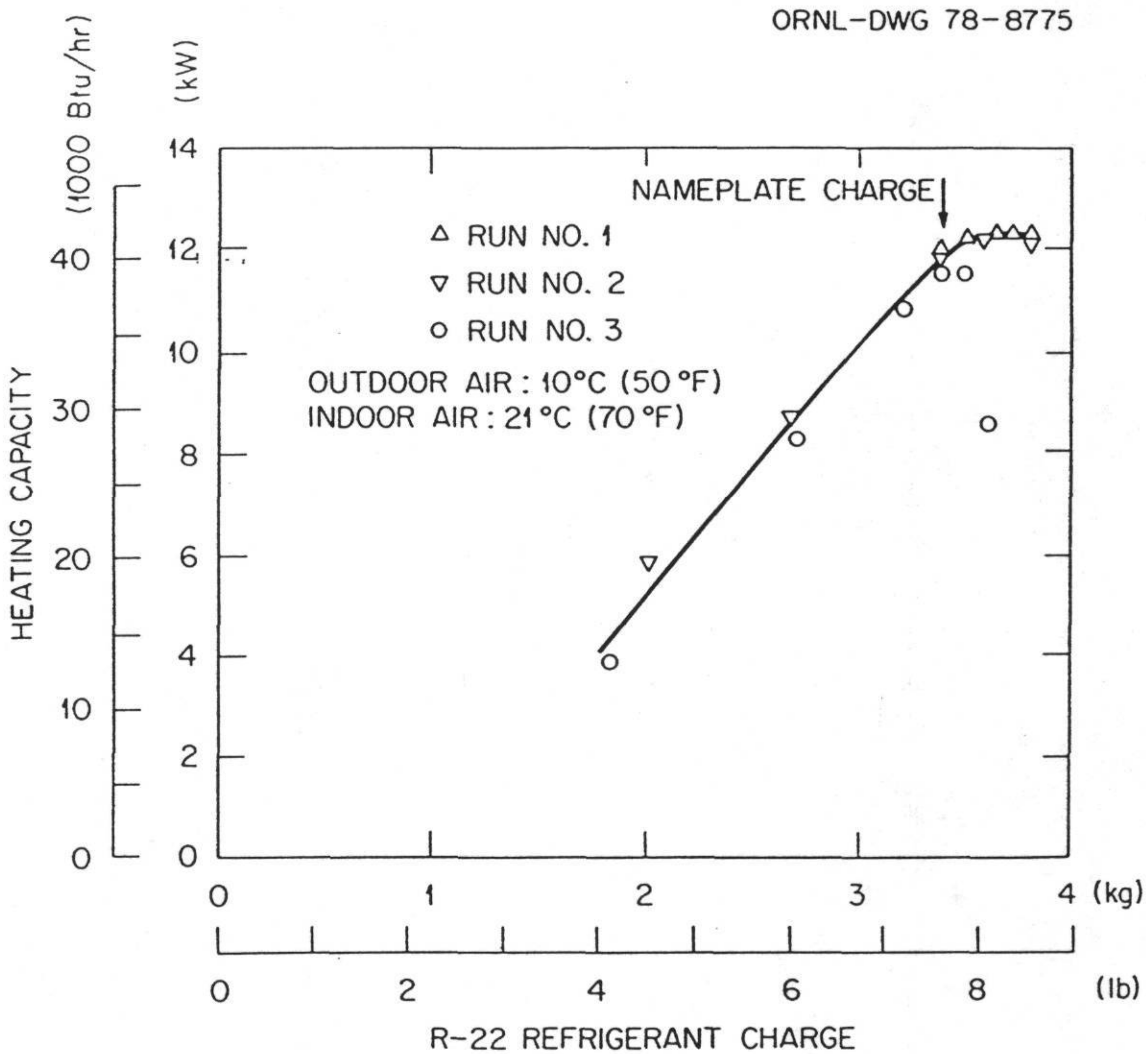

Figure 2.4 - Heating Capacity as a Function of Refrigerant Charge (adopted from ref.8)

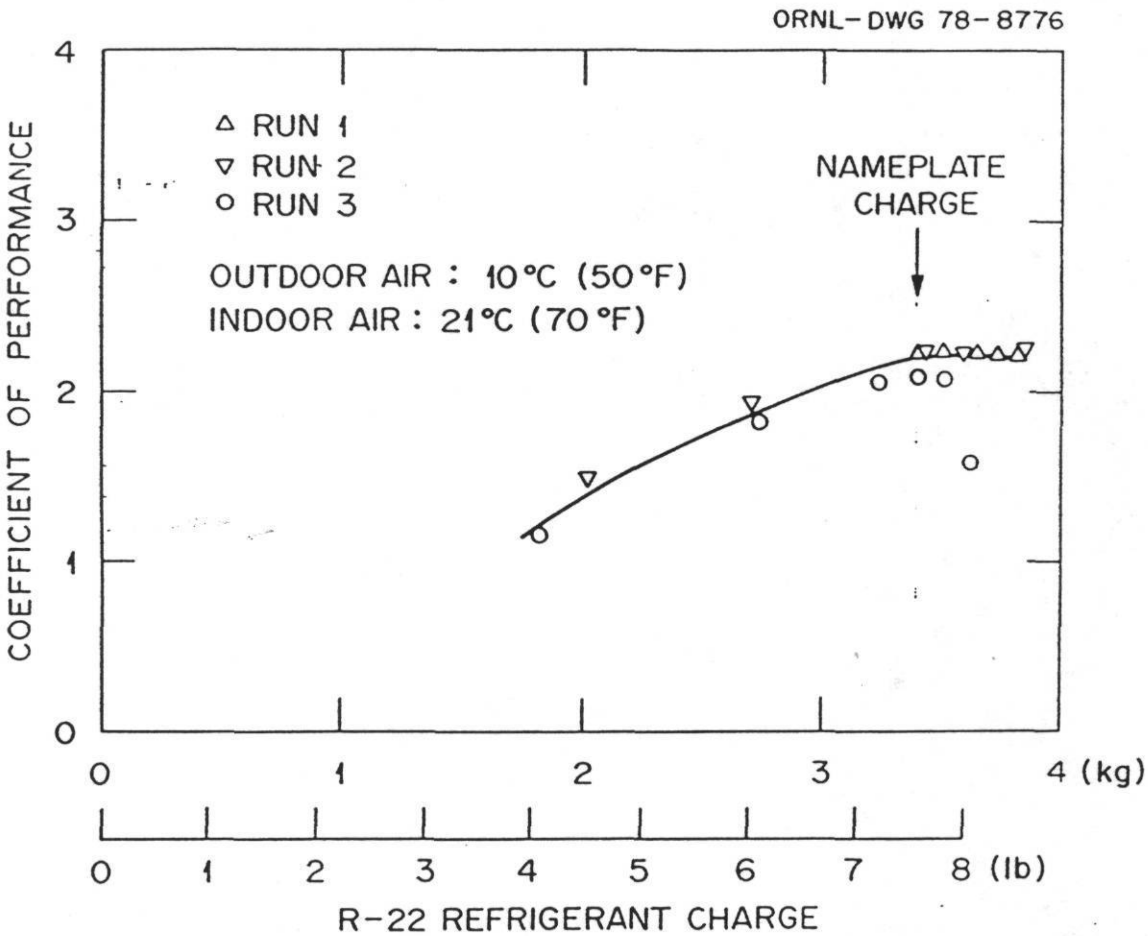

Figure 2.5 - Coefficient of Performance as a Function of Refrigerant Charge (adopted from ref.8)

A. A. Domingorena and S. J. Ball [9] studied the performance of a selected three-ton air-to-air heat pump in the heating mode at Oak Ridge National Laboratory (ORNL). In a field operation of a heat pump, the charge may be reduced by leaks in the system. Attempts to compensate for lost refrigerant after the repair of leaks may result in excess charge.

Tests were conducted with the heat pump with a suction-line accumulator operating in the heating mode. The outdoor and indoor air temperatures were at $50^{o}F$ and $70^{o}F$, respectively. The amount of refrigerant in the system was varied from 4 lbs to 8 lbs. The variation of heating capacity is shown in Figure 2.6. The maximum heating capacity was at full charge (6 lbs, 5 oz). A 36% loss in charge (4 lbs) resulted in a 23% reduction in heating capacity. The capacity remained constant from 5 ounces undercharged to 27 ounces overcharged.

They concluded " The low-first-cost heat pump tested, which lacks a suction-line accumulator, was found to be highly sensitive to charge. The same unit with a suction-line accumulator was tested under similar conditions. The efficiency of the heat pump with a suction-line accumulator for these tests remained essentially constant when the actual charge is within about 20% of the rated proper charge[9]." Figure 2.7 shows the COP of the heat pump with and without suction line accumulator for comparison.

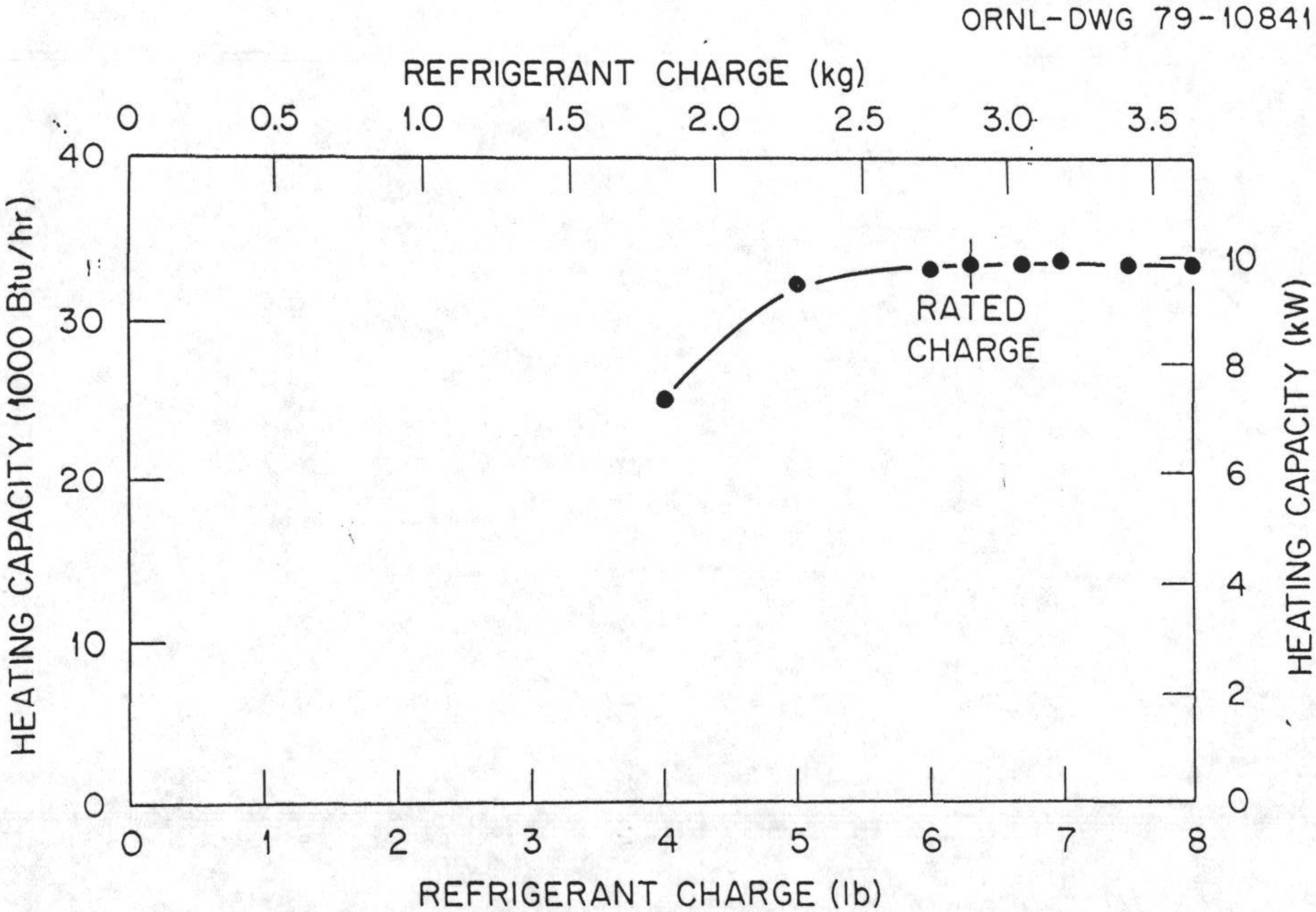

Figure 2.6 - Heating Capacity as a Function of Refrigerant Charge (adopted from ref.9)

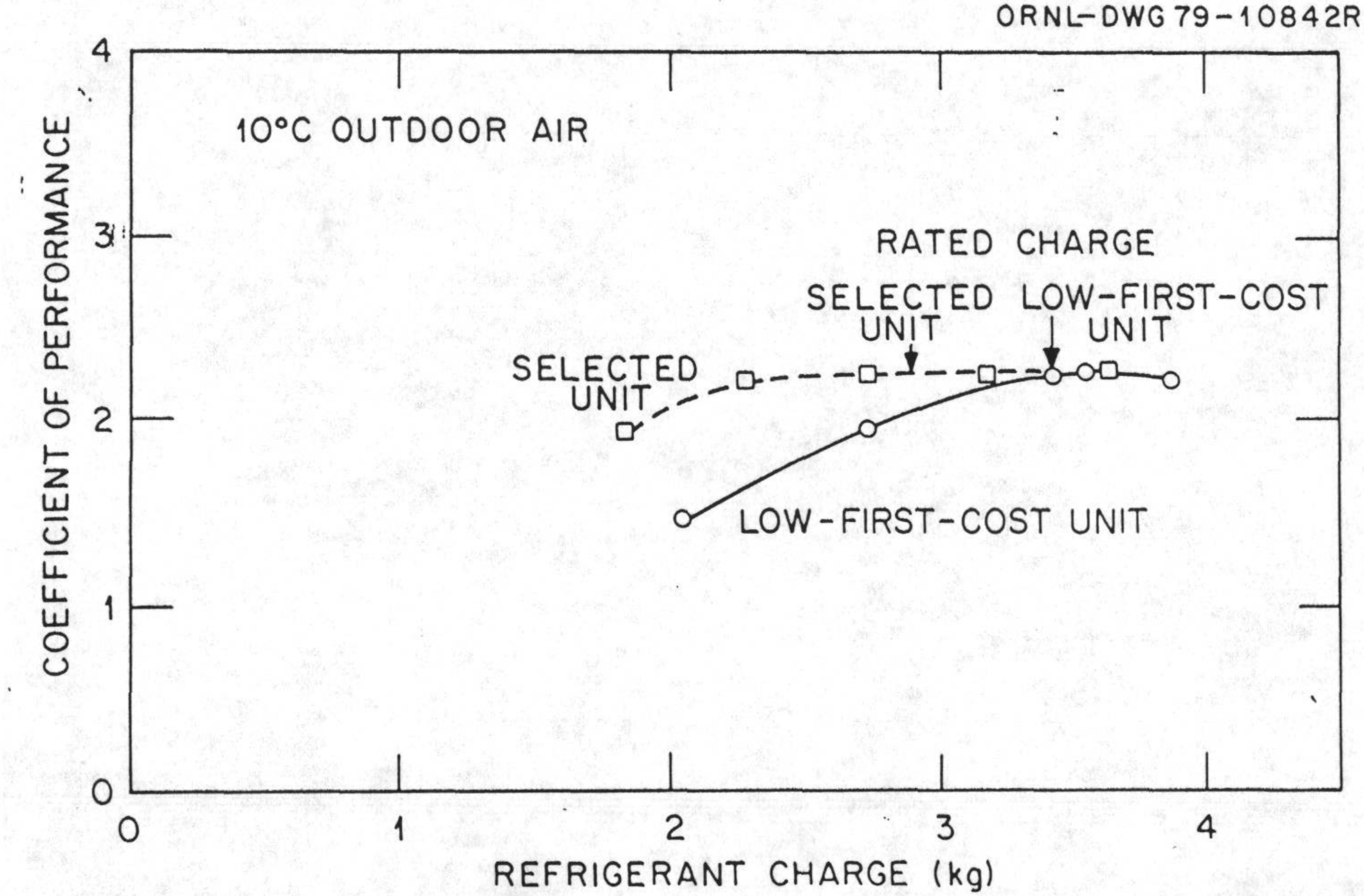

Figure 2.7 - Coefficient of Performance as a Function of Refrigerant Charge (adopted from ref.9)

Proper and accurate charging of split system air conditioning and refrigeration equipment utilizing capillary tube devices at other than optimum conditions appears to be a problem. A survey of the most common charging techniques are presented by John Houcek and Marvin Thedford [7]. They include: (1) Weighing-in the Charge, (2) Charging to Full Load or Nameplate Amps, (3) "Feel the Lines" or Suction Line Sweatback, (4) High and Low Sided Gauges, (5) Manufacturer's Chart and Tables with high & low side gauges, and (6) Superheat Charging [5]. Manufacturer's Chart and Tables and Superheat Charging are recognized as the most accurate field charging techniques.

Summary of Literature Review

The studies reviwed indicate that improper charging is detrimental to air conditioner performance and potentially to life expectancy. The TP&L data indicated undercharging has its most adverse impact at higher outdoor ambient temperatures[6]. While overcharging the unit increased its capacity, Houcek and Thedford indicated flooding in the compressor was experienced over a limited temperature rang. The effect of undercharging was dramatic on the COP and the heating capacity[8,9]. A.A. Domingorena and S.J. Ball found the COP and the heating capacity were essentially constant in the overcharging range.

The effect of outdoor temperature on air conditioner performance under the normal, over, and undercharge have not been adequately addressed in the literature. No study has provided enough quantitve data for developing guidelines for the effects of improper refrigerant charging. Most of references cited above considered only a single amount of under or overcharging. System performance (capacity, power demand, and efficiency) should also be systematically quantified as a function of the amount of under or overcharging.

The effect of improper refrigerant charge on cyclic test, C_D, and SEER have never been addressed in the literature. The studies reviewed above were limited to steady state measurements. Current test procedures require a cyclic test for determination of the SEER. The impact of under and overcharging on SEER would be of more use in evaluating the seasonal energy use.

CHAPTER 3

EXPERIMENTAL APPARATUS

The objective of the experimentation was to quantify the effect of improper refrigerant charge on the performance of a residential air conditioner system during the 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 3.1. 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. The air flowed through the test section. A damper was mounted on the outlet that

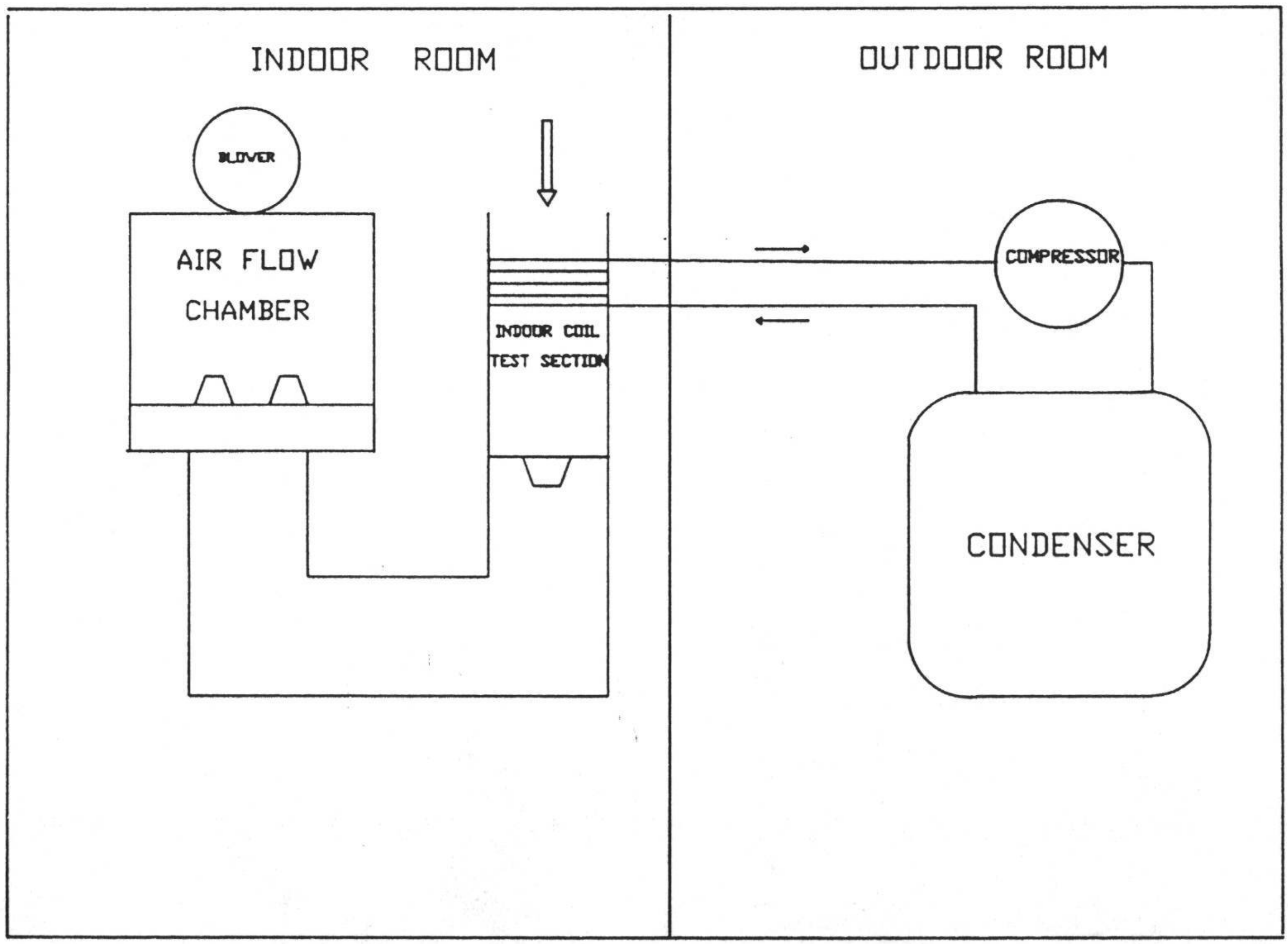

Figure 3.1 - The General Layout of the Test Apparatus

was adjustable and was set to maintain a 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 test 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 can be maintained within +/-0.2 F of the set point. The room temperature was controlled by a Texas Instruments TI-550 controller which was integrated into the control system of the rooms.

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 stabilize chilled water temperature. 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) [5]. 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^{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 3.2. 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

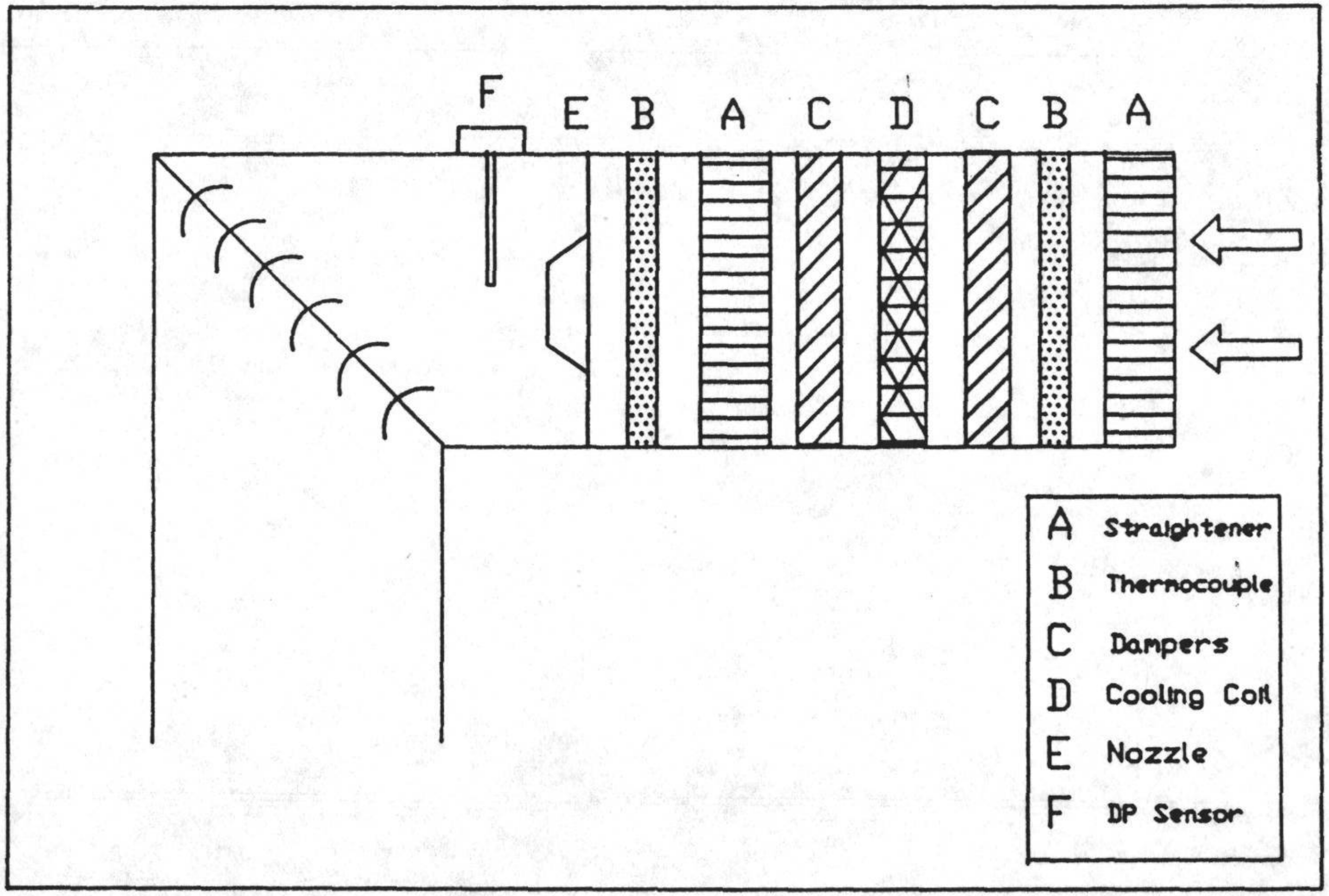

Figure 3.2 - Detail of Indoor Test Section

which were controlled by an "on-off" switch from the control room. After leaving the coil, the air flowed 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 capillary tube expansion (model TTX736A100A1/BWV736A) was used in this experiment. The indoor coil had vertical plate fins at 12 fins per inch with 4 rows of 3/8" copper tubing. The indoor coil had 3.33 ft^2 face area with a rated capacity of three ton.

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 contains 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 [10]. 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 is shown in Figure 3.3. This section included the compressor, the outdoor coil (condenser), and a turbine flow meter. A 3-ton Trane air conditioner with TTX736A100A outdoor unit was used. The outdoor coil was a spine fin type. The coil had one row 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 3.1.

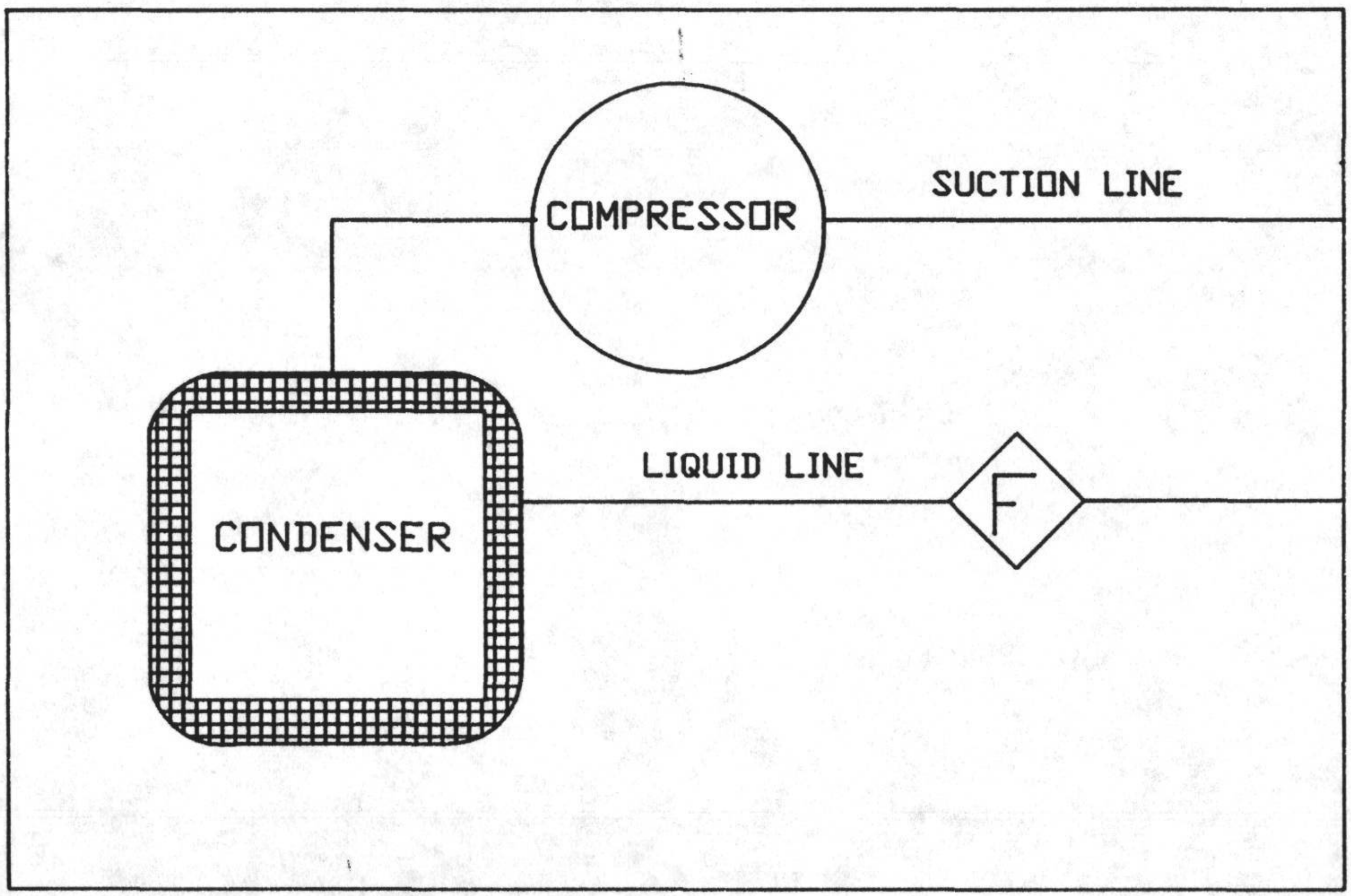

Figure 3.3 - Detail of Outdoor Test Section

Table 3.1 - Fan Specification

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

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 3.4. Refrigerant pressures were monitored at the 5 points shown with the use of 0-300 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

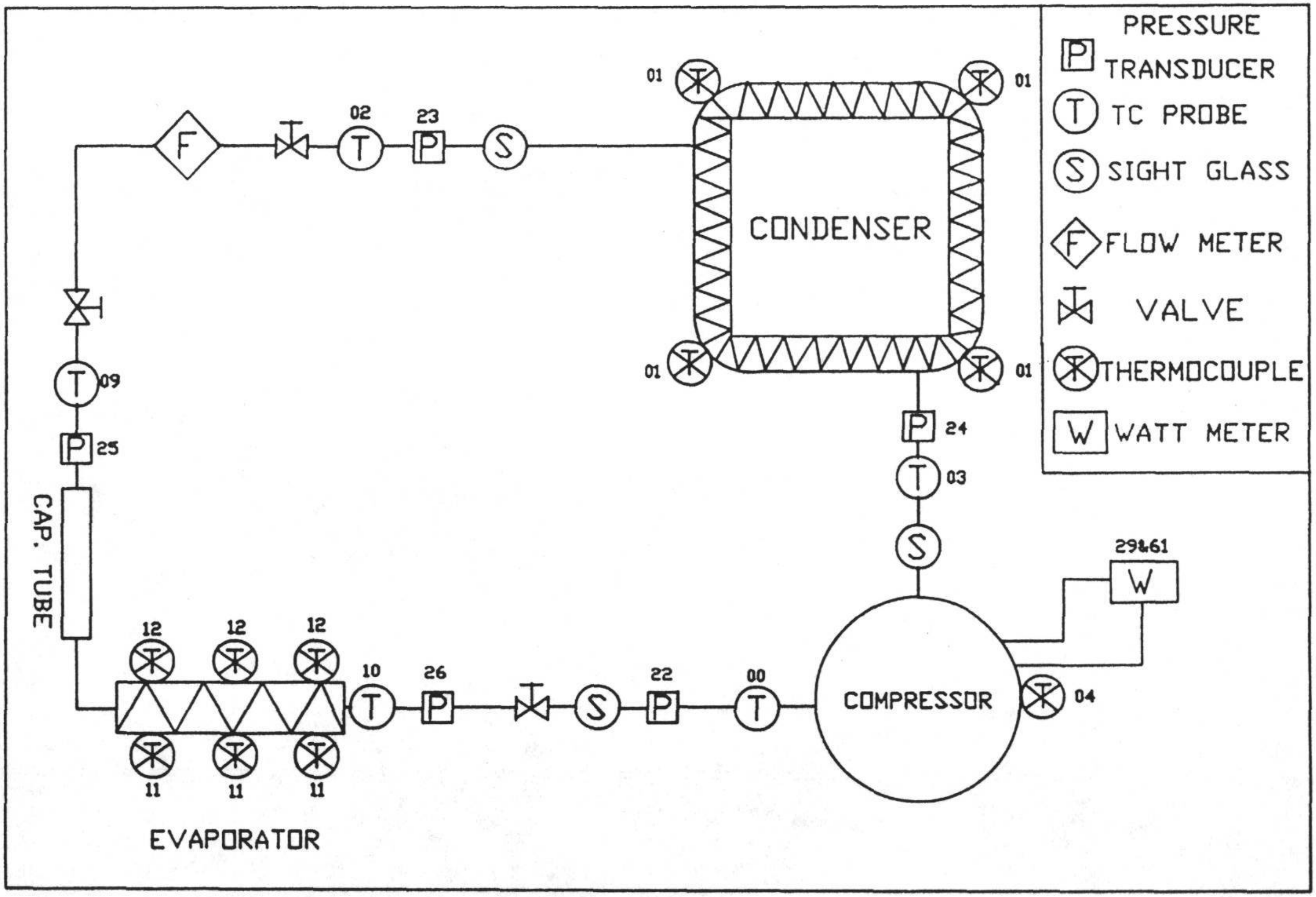

Figure 3.4 - Schematic of the Refrigerant Circuit

minimize the tube conduction effects. Figure 3.5 shows a typical refrigerant temperature probe.

Refrigerant mass flow was measured with a turbine flow meter. As shown from figure 3.4, the turbine flow meter was placed on liquid line after the condenser unit. The pressure drop across the turbine flow meter was 7 psi (for fully charged condition). This pressure drop was less than the 12 psi pressure drop acceptable by ASHRAE Standard 116-83 [11] (12 psi is the equivalent 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-actuate 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 3.2 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

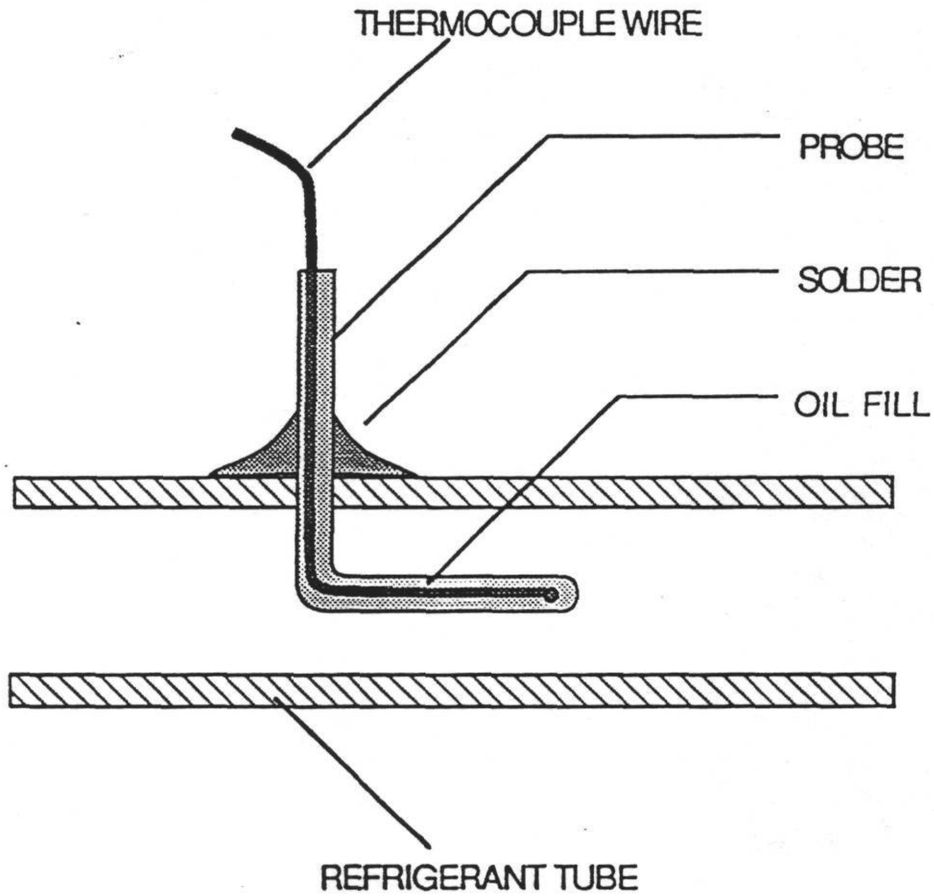

Figure 3.5 - A Typical Refrigerant Temperature Probe

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 mainframe VAX for analysis. Data were backed up on floppy disks. FORTRAN programs were written and used to drive refrigerant and moist air property subroutines. These subroutine 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 3.2 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 Nazzle-Chamber
07	Thermocouple	Cab. Tube Temp.
08	Thermocouple	Cab. Tube Temp.
09	TC-Probe	Cab. Tube Inlet
10	TC-Probe	Evaporator Outlet
11	TC-Grid	DB Indoor Coil Outlet
12	TC-Grid	DB Indoor Coil Inlet
13	Thermocouple	Outdoor Room Temp.
14	Thermocouple	Chilled Water Temp.
15-19		-- 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.	Cap. Tube Inlet
26	Pressure Trans.	Evaporator Outlet
27-28		-- Not Used --
29	Watt Trans.	208 VAC (single phase)
30-60		-- 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, six minutes of "on" time and two minutes longer until it reached zero. Electrical energy was measured for 6 minutes of "on" time. The dampers were shut off after first8 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. The superheat charging chart was based on 400 CFM/ton indoor air flow. The charging chart states "if operating superheat is within 5^{o}F chart value then the charge is OK". The charge was set for outdoor room conditions at 90^{o}F and indoor room conditions of 80^{o}F dry bulb and 67^{o}F wet bulb. Refrigerant was added to the system until the superheat reached 12.6^{o}F which was within 4^{o}F of the value in the charging chart. The superheat was also checked at other outdoor temperatures (Figure 3.6). According to charging chart, the suction and the liquid pressures should be within +/-3 and +/-10 psig of the chart, respectively. The suction pressure was 5 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.

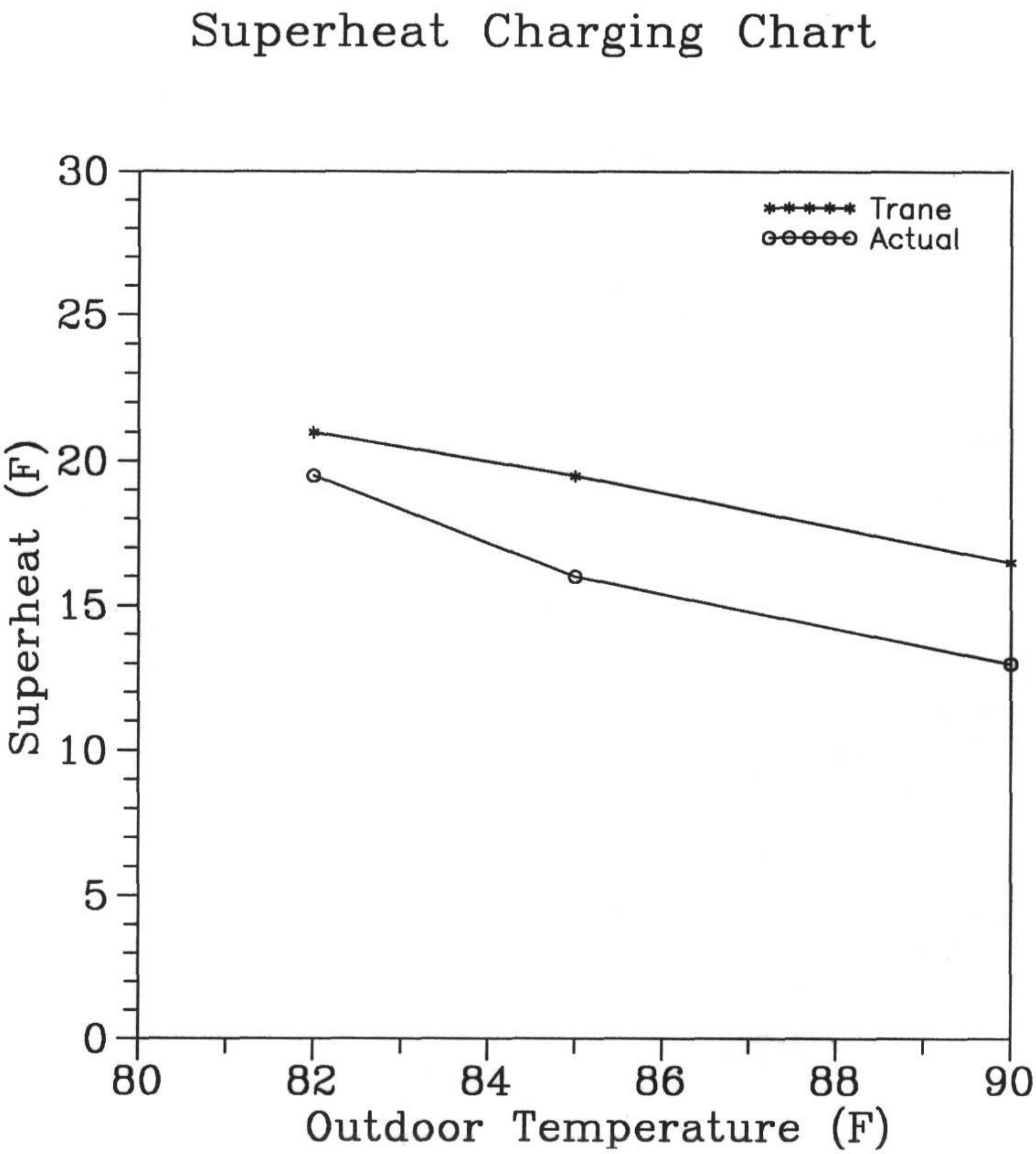

Figure 3.6 - Comparison Between The Test Superheat with the Manufacturer Charging Chart

Once the full charge was determined (140 ounces), the system was vacuumed and it was recharged to 112 ounces of refrigerant which corresponds to 20% undercharge. The under and overcharge amount of refrigerant in the system in increment of 7 ounces are shown in Table 3.3. A scale was used to weigh the amount of refrigerant added to the system.

Table 3.3 - Different Amount of Charge in the System

Charge	Ounces
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 super eat chart at the room conditions mentioned previously. On e the proper charge was determined, the system was vacuumed and recharged to 20% under charge (112 oz).

The steady state tests (wet coil) under 4 differ nt outdoor temperatures: 82°, 90°, 95°, and 100°F were performed while the indoor conditions were set at 80° DB and 67°F WB. Both steady state and cyclic tests (dry oil) were performed with indoor conditions set at 80° DB nd 5 $^{\circ}$ WB F while the outdoor conditions were constantly kep at 82°F and 20% relative humidity. Both the wet and dry oil tests were repeated on the unit for 5, 10, 15, and 2(und r and overcharging of refrigerant (by mass).

CHAPTER 4

RESULTS & ANALYSIS

The performance of residential air conditioners is directly related to the amount of refrigerant in the system. The refrigerant charge in a system was systematically varied to determine its effect on the capacity, EER, SEER, and coefficient of degradation (C_D) of a central air conditioner. The results of improperly charging the air conditioners 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, detailed specifications on the unit were obtained from Trane Co.

The unit was charged according to the procedures specified in the manufacturer's procedure. These procedures included setting of a particular superheat for specific outdoor conditions. A copy of the charging chart is shown in Figure 4.1.

The superheat charging chart was based on 400 CFM/ton indoor air flow. To determine the proper charge, the indoor room temperature was maintained at 80^{O}F DB and 67^{O}F WB. The

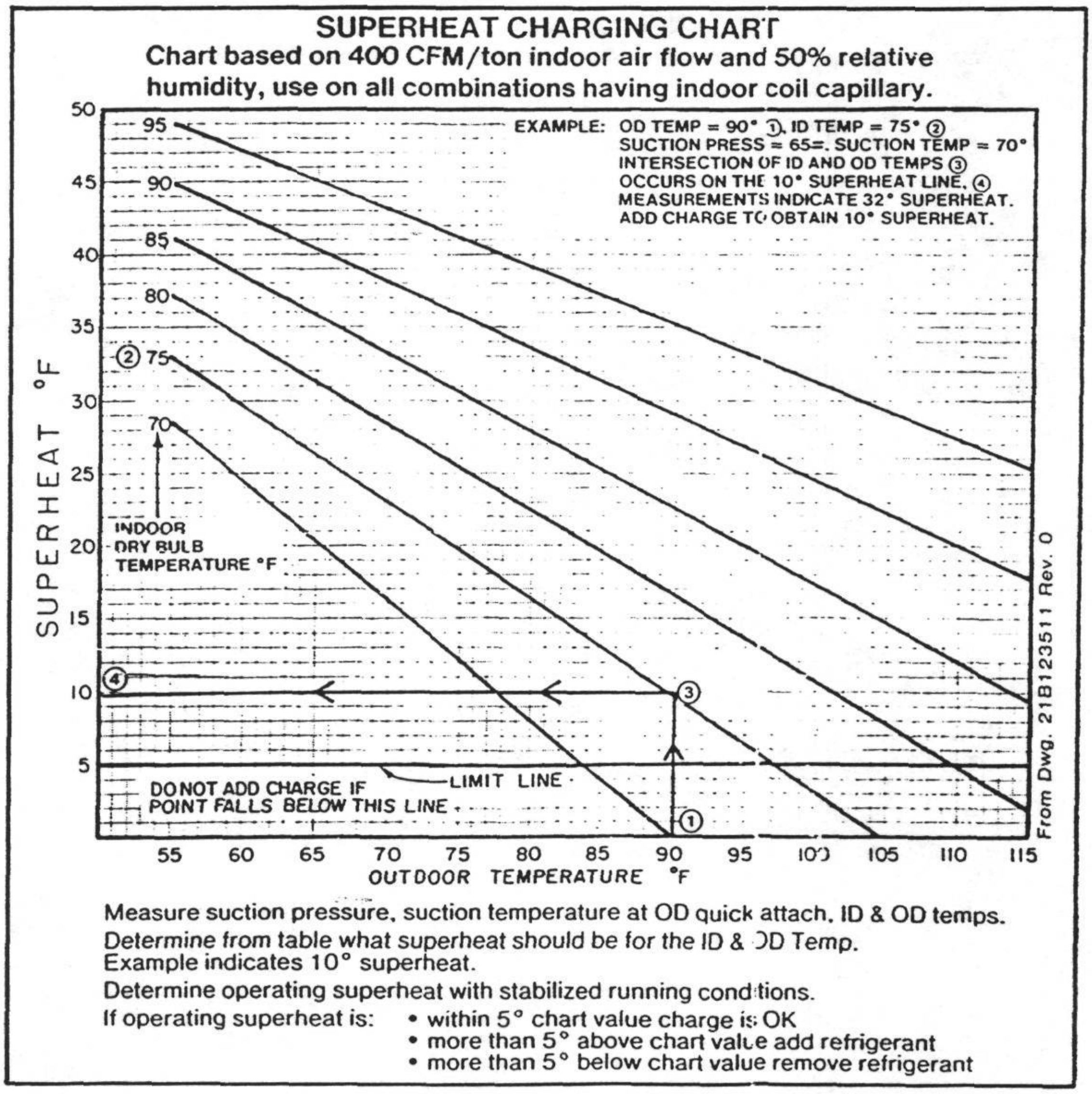

Figure 4.1 - The Manufacturer's Charging Chart

outdoor room temperature was kept at different temperatures 82°F, 85°F, and 90°F DB. At different outdoor temperatures 19.5°F, 16°F, and 12°F of superheat leaving the evaporator were obtained for 140 ounces of R-22 refrigerant in the system, . The obtained superheat temperatures were within the 5°F of the manufacturer chart.

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 of the evaporator or on the refrigerant side. 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[11]. 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 [14]. The airflow calculations were done using a method provided in ANSI/ASHRAE Standard 51-1985[10].

To verify the calculations for the air-side capacity, an energy balance was performed on the indoor coil. Figure 4.2 shows a plot of the refrigerant side and air side capacity for the indoor coil as a function of outdoor temperature for fully charged conditions.

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[15]. Typically, the air-side and refrigerant side were +/-3% to +/-6% 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[10].

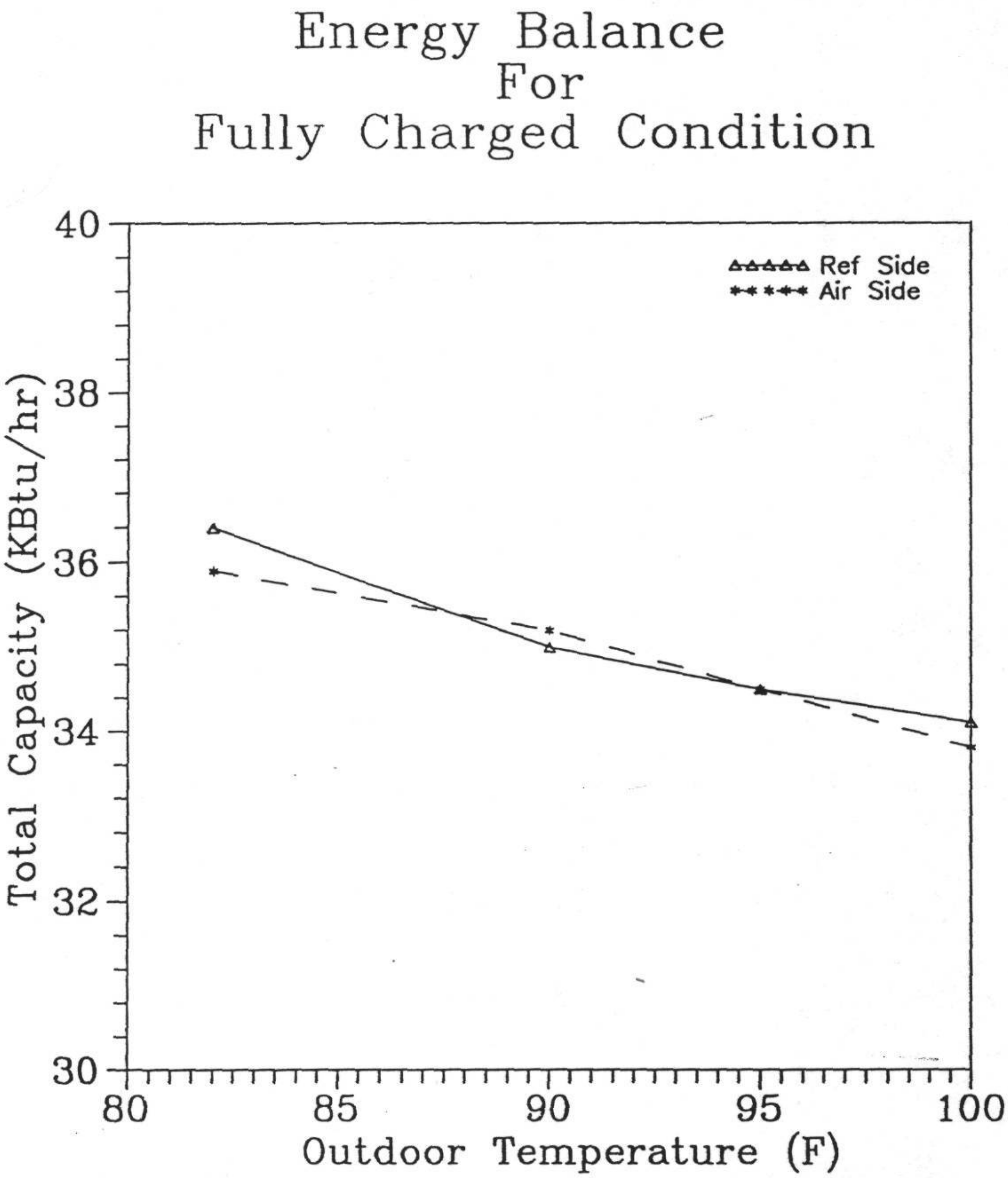

Figure 4.2 - Refrigerant-Side/Air-Side Capacity Comparison

EER is a steady state measure of efficiency. It is calculated by dividing the net cooling capacity in Btu/hr 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).

The comparisons between the manufacturer's performance data and the unit tested at full charge are shown in Table 4.1. One reason for lower tested EER versus manufacturer EER is due to higher total tested kw (by 3.4%) and lower net capacity (by 1.9). The other reason is the Trane data are at 1200 cfm where as the test data at 1130 cfm air flows. The rooms conditions for the comparisons are 80^{o}F DB and 67^{o}F WB indoor and 90^{o}F DB outdoor temperatures.

1. Steady State Tests (Wet Coil)

The DOE test procedure requires two steady state tests of the air conditioner in which dehumidification would occur 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).

Table - 4.1 Performance Data Cooling[†]

Item	Trane Data	Test Data	%Change
Net Cap. (KBtu/hr)	35.3	34.6	-1.9
Sensible Cap.	25.6	26.9	+5.2
CMPR (kw)	2.67	2.70	+1.1
Total Power (kw)	3.45	3.57	+3.4
EER	10.2	9.81	-3.9
SEER	9.7	9.44	-2.6
Refrigerant (oz)	140	140	0.0

† 80°F DB & 67°F WB indoor and 90°F outdoor temperatures

For each outdoor temperature, several 30-minute sets of data under steady state were recorded. Figures 4.3 and 4.4 show the units 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 35.4 KBtu/hr at 82°F to 32.15 Kbtu/hr at 100°F. The EER ranged from 10.65 at 82°F to 8.57 at 100°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°F DB

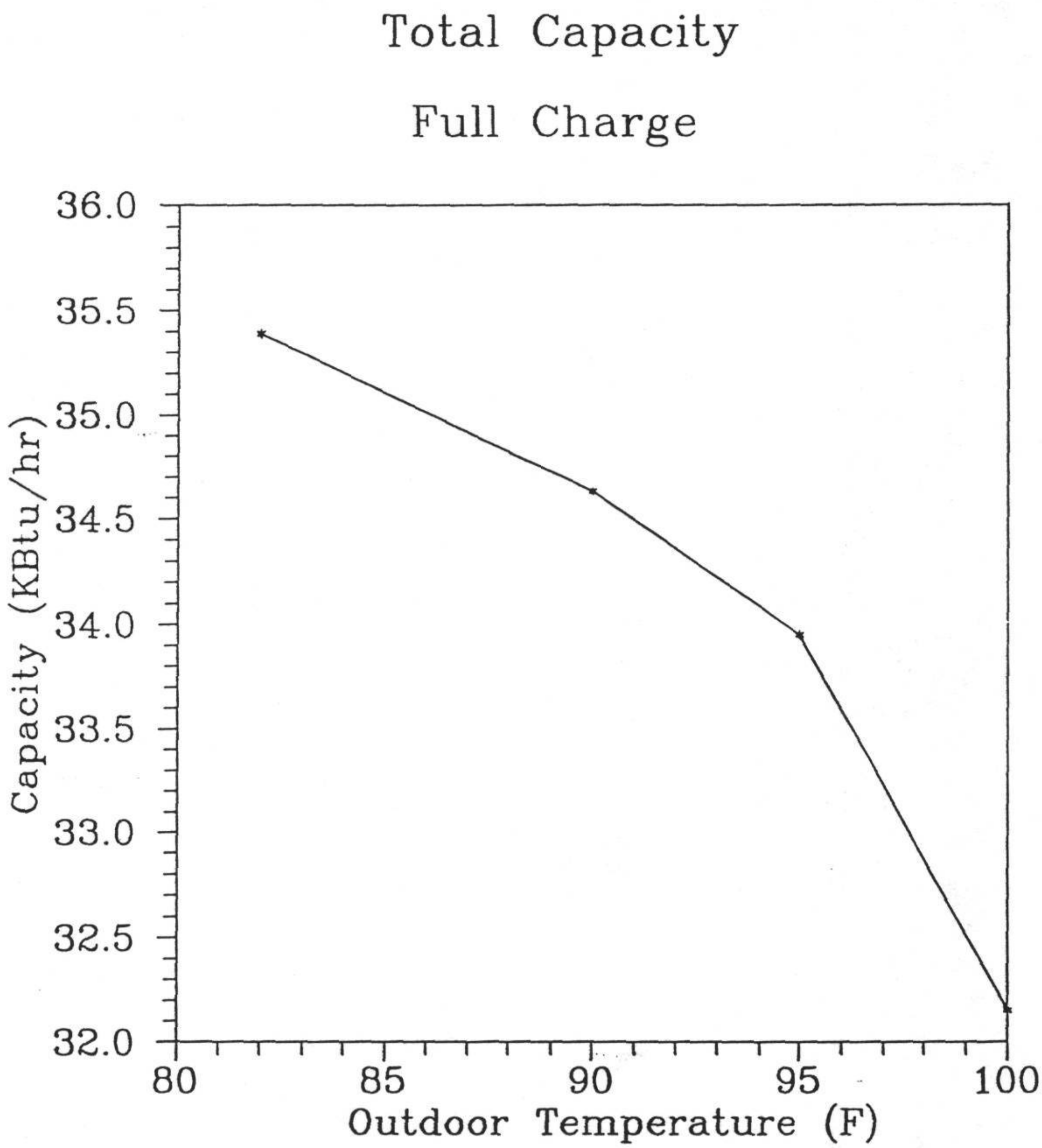

Figure 4.3 - Total Capacity of the Fully Charged Unit

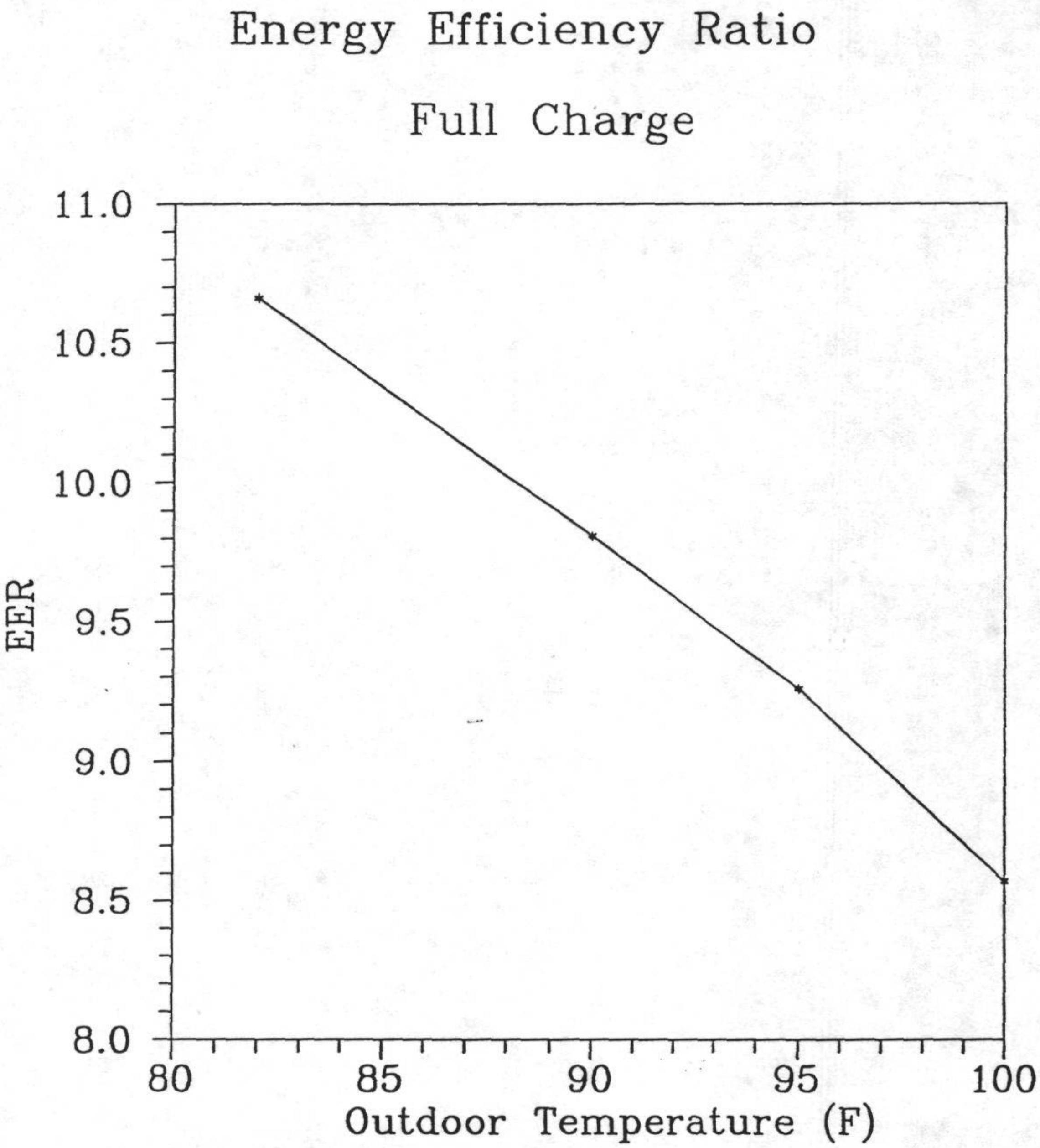

Figure 4.4 - Energy Efficiency Ratio of the Fully Charged Unit

and 57^{o}F WB. The wet bulb temperature was sufficiently low enough so that no condensate formed on evaporator coil. The 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[5], 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 4.5 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

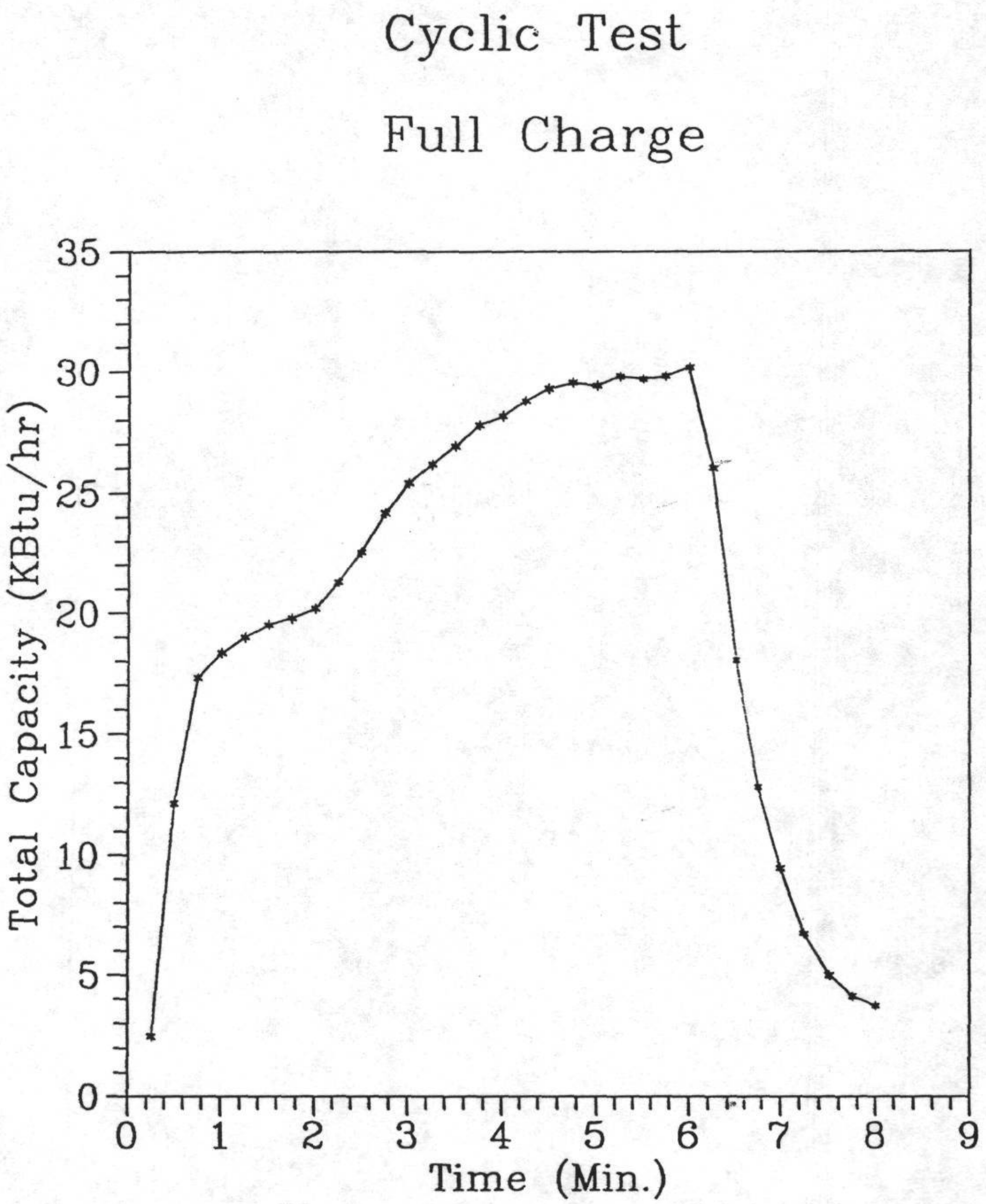

Figure 4.5 - Cyclic Capacity of the Fully Charged Unit

steady state wet coil tests at 95° and 82° F DB outdoor room temperatures, respectively. Test (D) is a steady state dry coil test at 82° F DB outdoor room temperature. The calculation of the unit's SEER with a single-speed compressor and single-speed condenser fan is done in the following way (Federal Register, December 27, 1979). First, a cyclic-cooling-load factor (CLF) is determined from:

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

Q_D is the total cooling capacity of test D and 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:

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

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°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 4.2 shows the unit performance under fully charged condition for steady state and cyclic tests.

Table 4.2 - Dry Coil & Cyclic Tests Performance

EER_C	EER_D	SEER	C_D	CCLF
9.75	7.84	9.44	0.235	0.163

CAPACITY

Once the proper charge was determined , the unit was vacuumed and recharged initially to 20% undercharged condition (112 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 4.6 shows the total capacity as a function of the outdoor temperature and the refrigerant charge. At fully charged conditions, the highest capacity was obtained at 82°F outdoor room

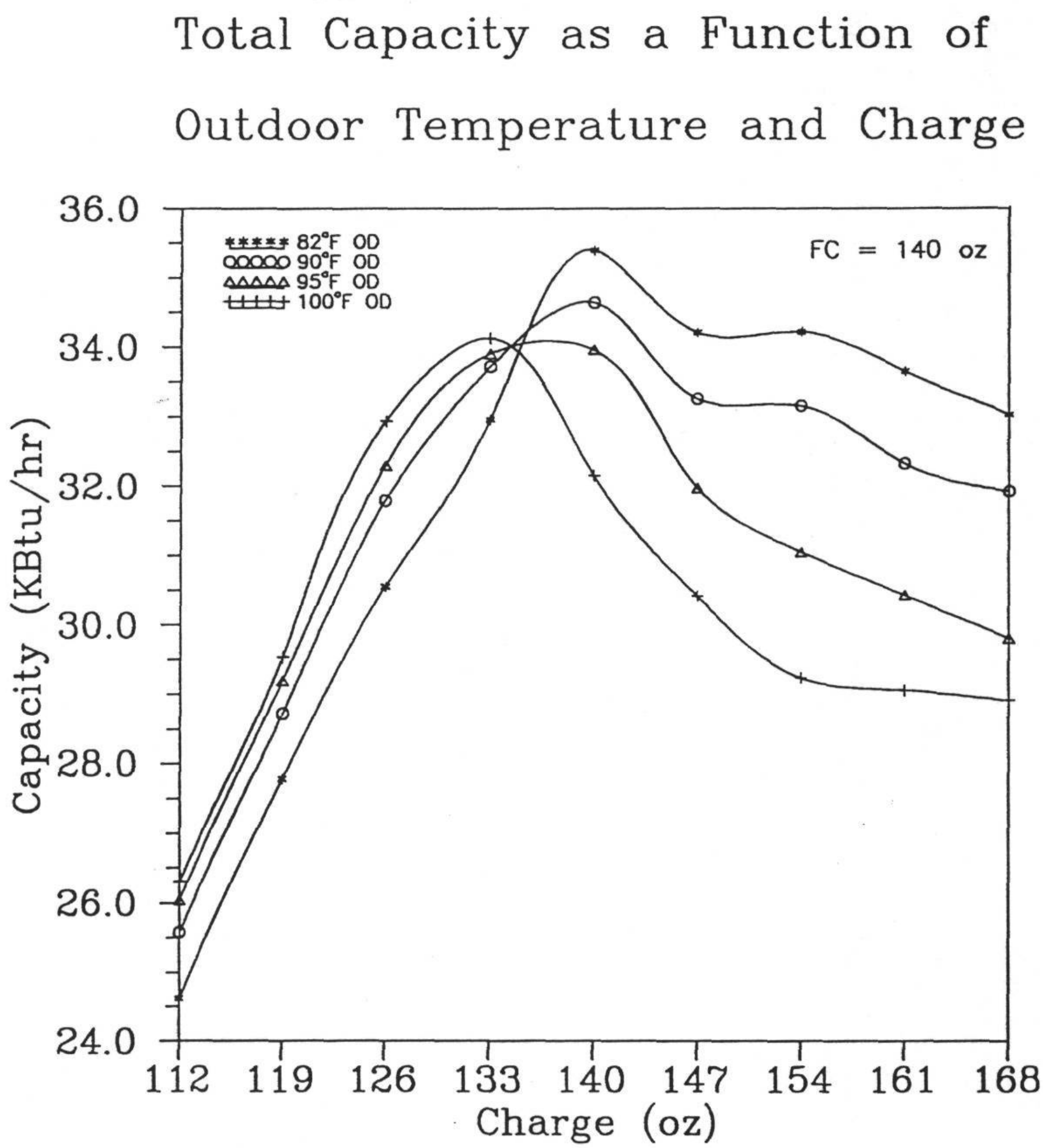

Figure 4.6 - Total Capacity as a Function of Outdoor Temperature and Charge (wet coil)

temperature. The capacity of the unit dropped off by 10% as the outdoor room temperature increased from 82°F to 100°F.

One surprising result shown in Figure 4.6 was the difference in the behavior of the capacity for the undercharged condition as compared to the full or overcharged case. Normally, the capacity of an air conditioner is expected to decrease as the outdoor temperature increases (Figure 4.3). However, for the undercharged tests, the capacity increased as the outdoor temperature increased.

One possible explanation for this behavior for the undercharged condition might be found in the changes of the refrigerant flow rate by the capillary tubes for different charges. Liquid refrigerant enters the capillary tube, and as it flows through the tube, the pressure drops because of friction and acceleration of the refrigerant. Some of the liquid flashes into vapor as the refrigerant flows through the tube.

Numerous combinations of bore and length are available to obtain the desired restriction. The size of capillary tube for the Trane 3-ton air conditioner was 26" long and 0.1" in diameter. Once the capillary tube was selected and installed, the tube is fixed in the adjustments it can make to variations in discharge pressure, suction pressure, load, or amount of charge in the system. Figure 4.7 shows such a generic plot with the flow through the capillary tube [13]

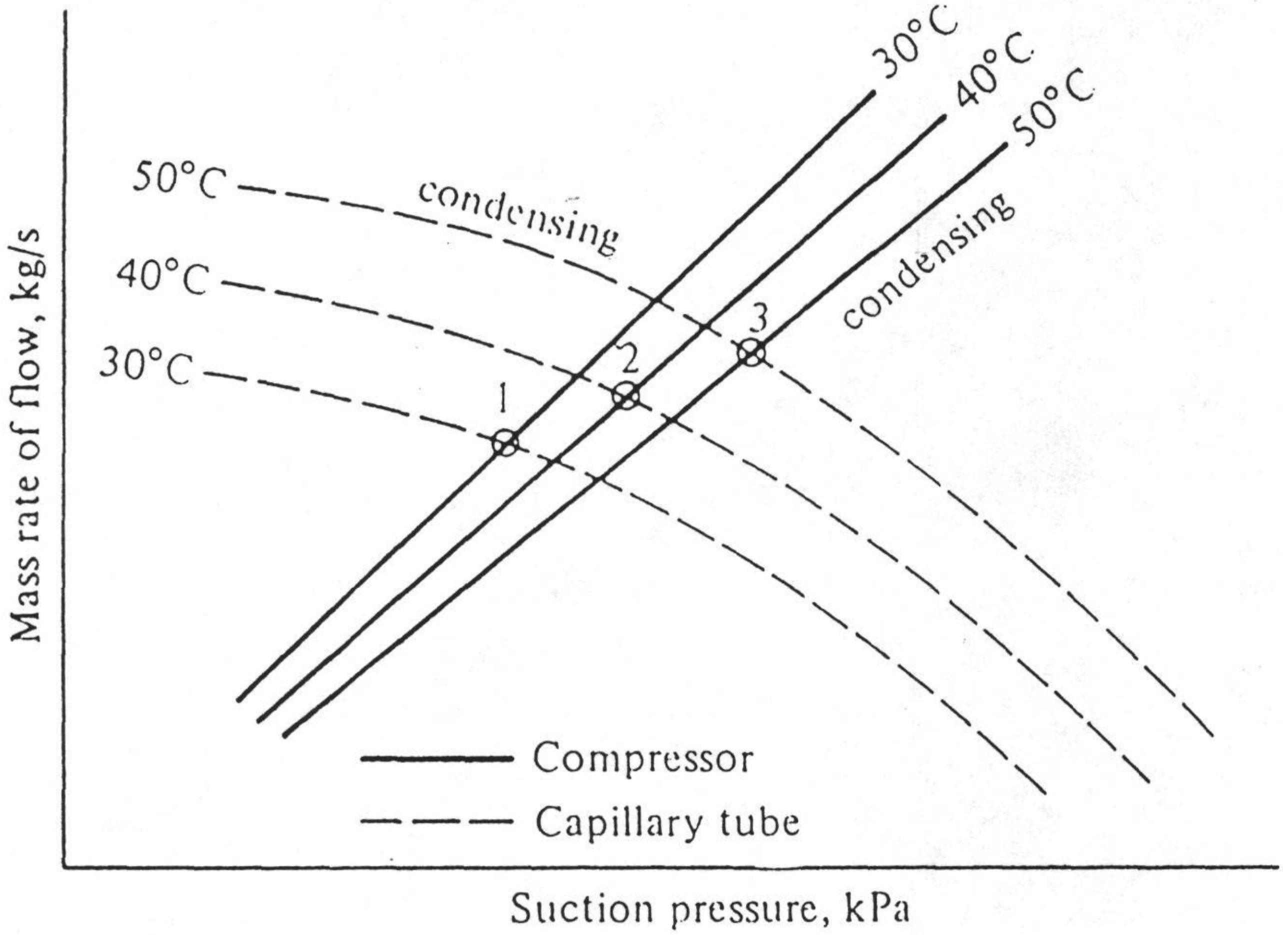

Figure 4.7 - Balance Points with a Reciprocating Compressor and Capillary Tube

as a function of condensing and evaporating temperatures. At high condensing pressures the capillary tube feeds more refrigerant to the evaporator than it does at low condensing pressures, because of the increase in pressure difference across the capillary tube.

According to the ASHRAE handbook[12], refrigerant mass flow rate is calculated as follows:

$$MFR = \grave{o}\ m$$

where,

MFR = actual mass flow rate of refrigerant

$\grave{o}$ = flow factor

m = standard mass flow rate

The flow factor and standard mass flow rate of refrigerant curves are shown in Figure 4.8 and 4.9. The standard mass flow rate is directly proportional to the subcooling temperature and condenser pressure. For the test condenser pressures and subcooling temperatures, the refrigerant mass flow rate was calculated from Figures 4.8 and 4.9 and it is shown in Figure 4.10. This figure shows for the 20 and 10 percent undercharged tests, the refrigerant flow rate increased by 18% when the outdoor temperature increased from 82°F to 100°F. However, for the same increase in outdoor temperature, the refrigerant flow

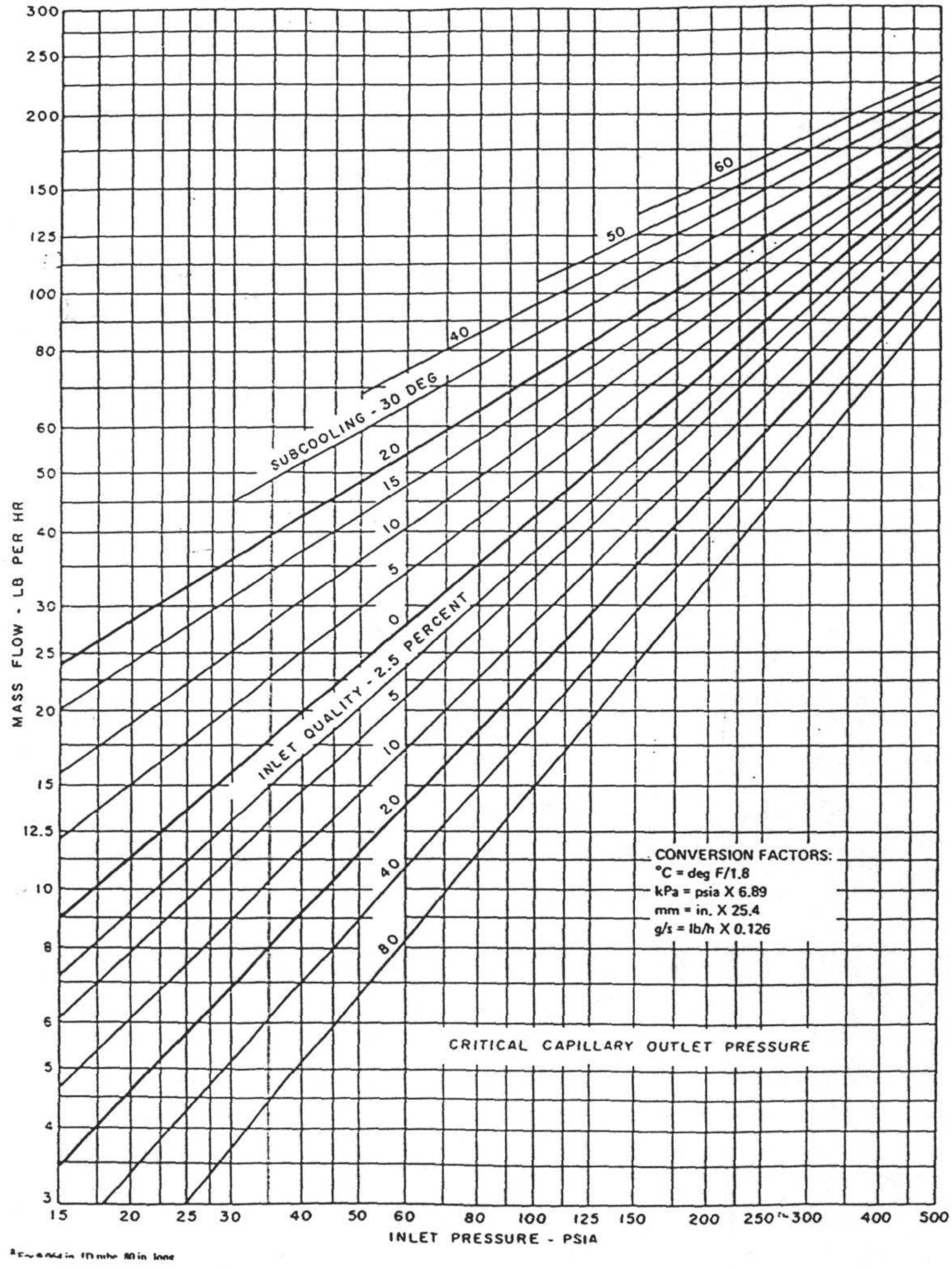

Figure 4.8 - Basic Rating Curves for Condenser-to-Evaporator Capillary (Ref. 12 and 22)

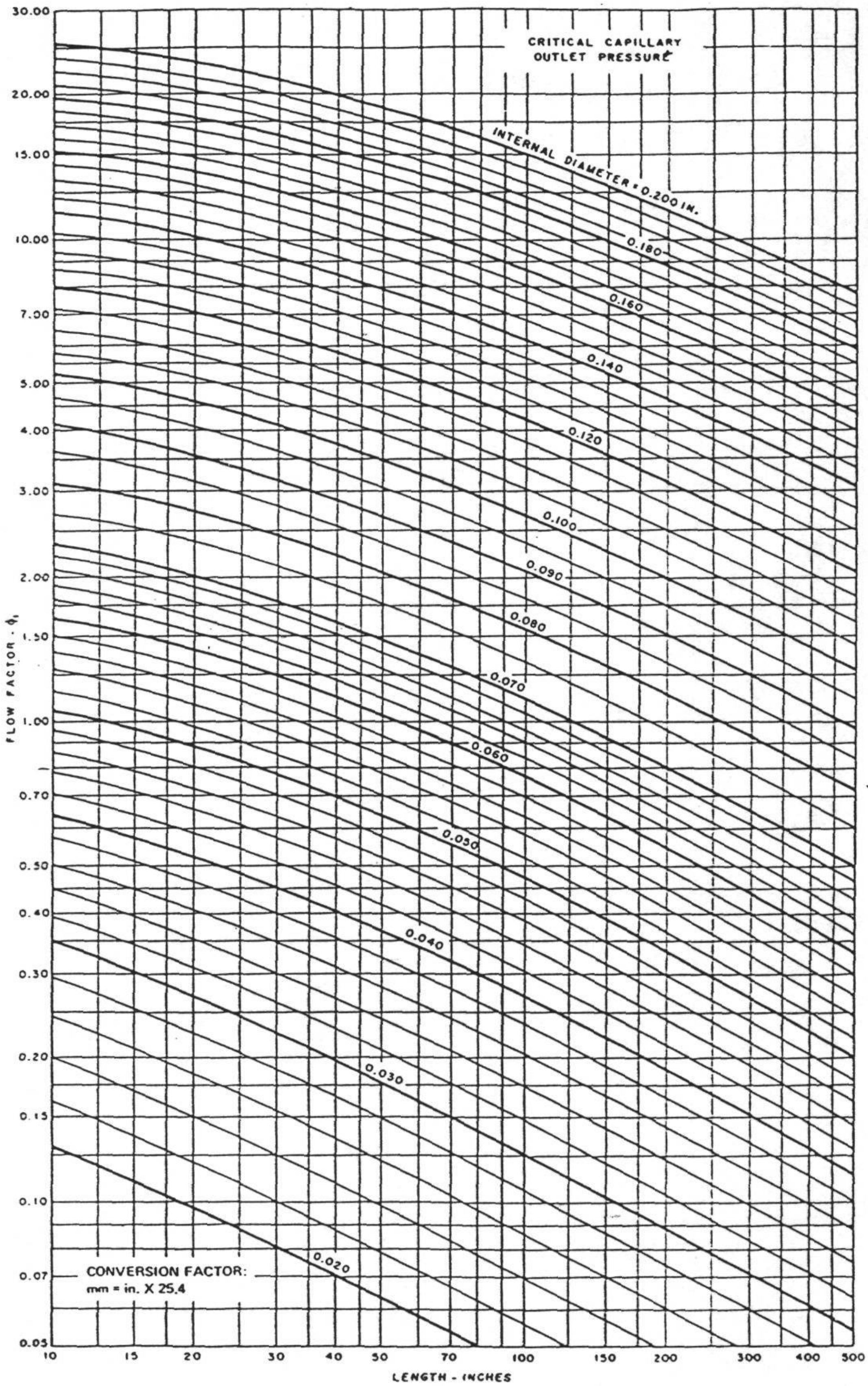

Figure 4.9 - Capillary Flow Factors(Ref. 12 and 22)

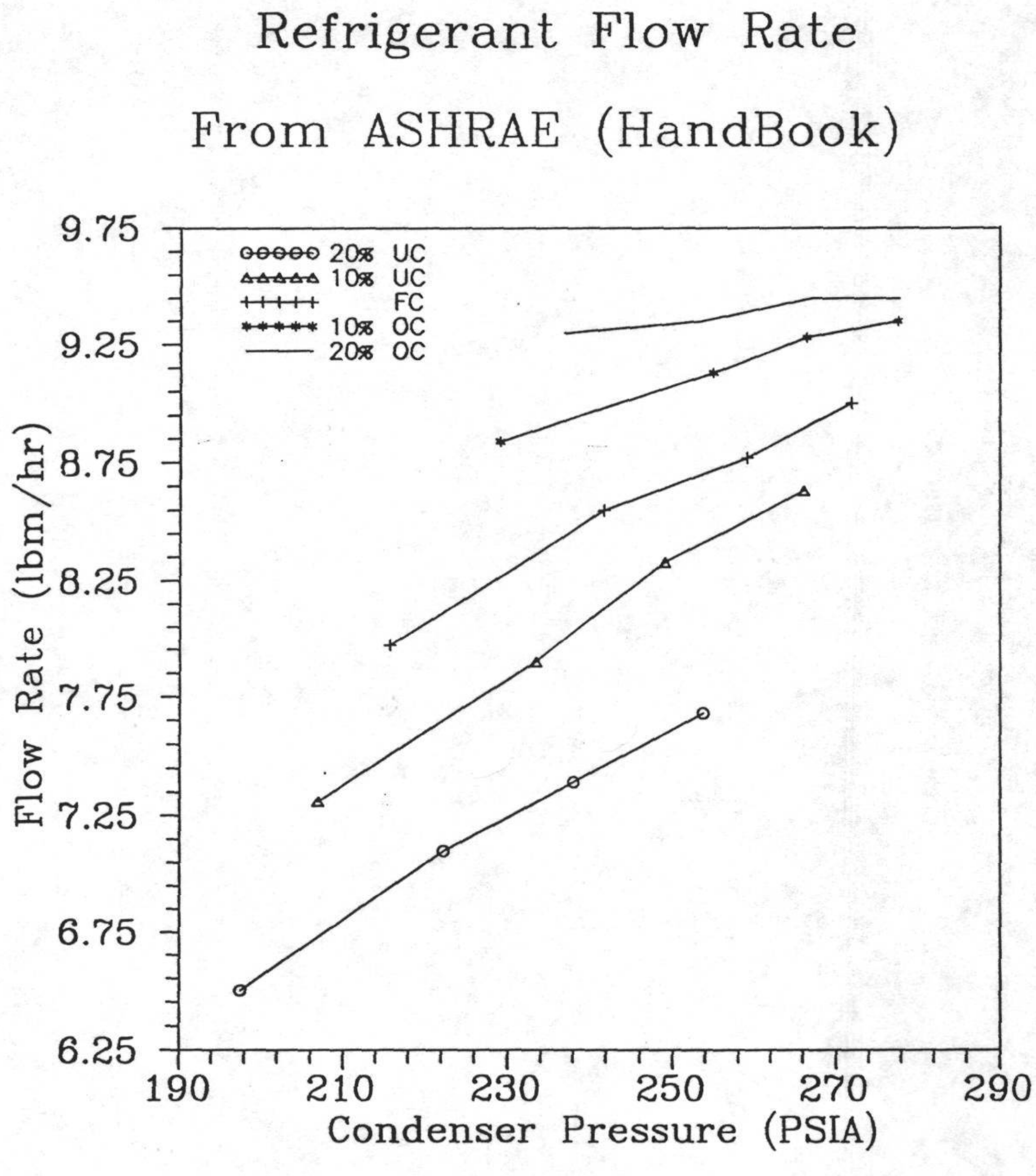

Figure 4.10 - Calculated Refrigerant Mass Flow Rate

rate increased by 5.7% and 1.6% for the 10 and 20 percent overcharging tests, respectively.

As the outdoor temperature increased, the pressure drop across the capillary tube increased too. Figure 4.11 shows the pressure drop across the capillary tube as a function of outdoor temperature. As the pressure drop increased, the capillary tube feeds more refrigerant to the evaporator. Also, at higher outdoor temperatures, the suction pressure increased slightly while the suction temperature decreased as shown (Figures 4.12 and 4.13). A decrease in suction temperature due to the higher pressure drop would result in lower superheat temperature. Thus, the higher refrigerant flow rate is the reason for higher capacity at higher outdoor temperatures for the conditions of undercharge. The refrigerant flow rate as a function of condenser pressures and charge is given in Figure 4.14. The measured refrigerant flow rates as shown in Figure 4.14 showed a similar trend to the calculated flow rates from the ASHRAE handbook of equipment (Figure 4.10). For the 20% undercharged tests, the refrigerant flow rate increased by 13.5% when the outdoor room temperature increased from 82^{O}F to 100^{O}F. A 13.5% increase in refrigerant flow rate resulted 5.7% increase in total capacity of the unit. However, for the same increase in outdoor room temperature, the refrigerant flow rate only increased by 3.7% and 2.6% for the 10 and 20 percent overcharged conditions. The refrigerant flow rate as a function of outdoor temperature is shown in Figure 4.15.

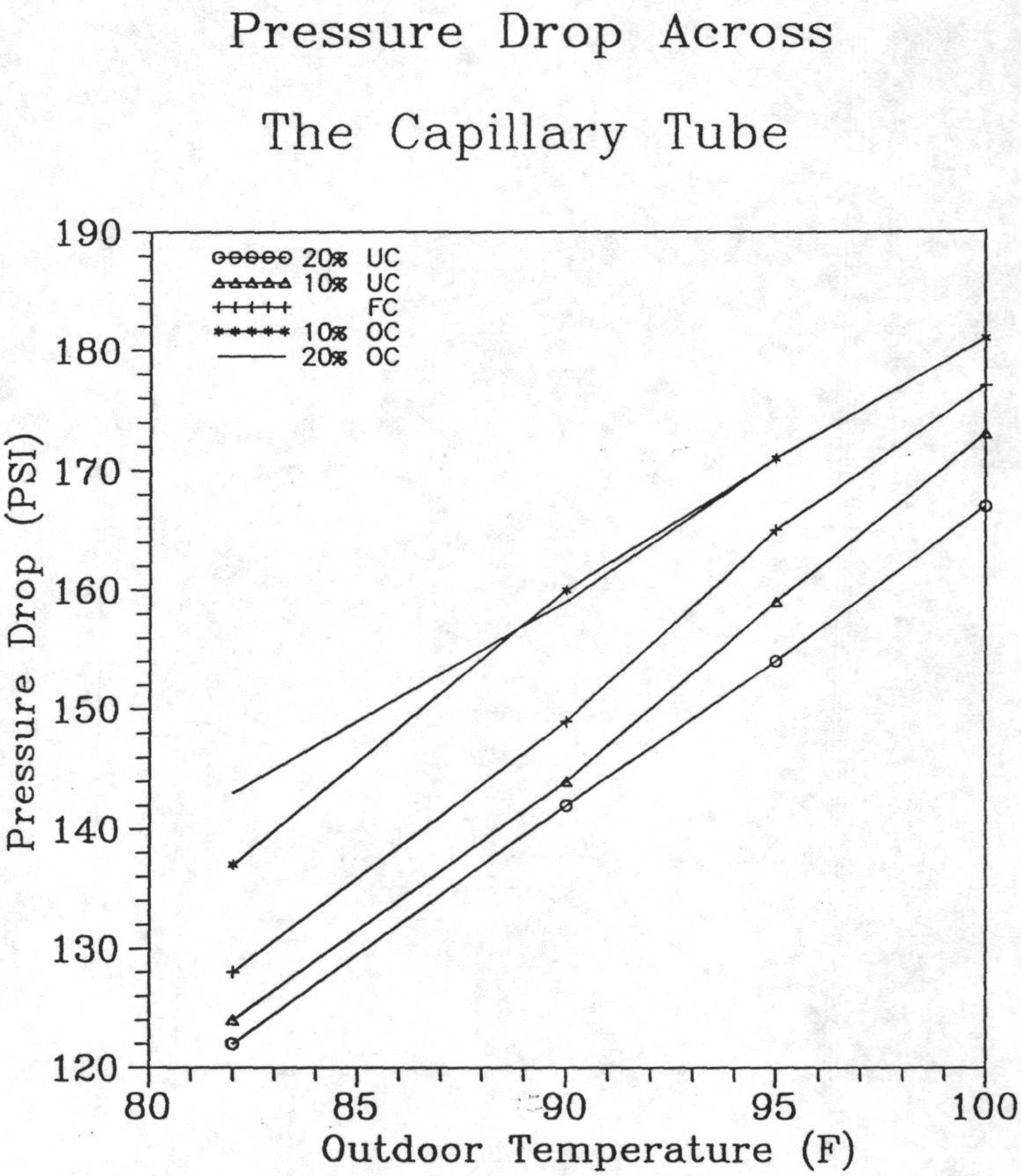

Figure 4.11 - Pressure Drop Across the Capillary Tube

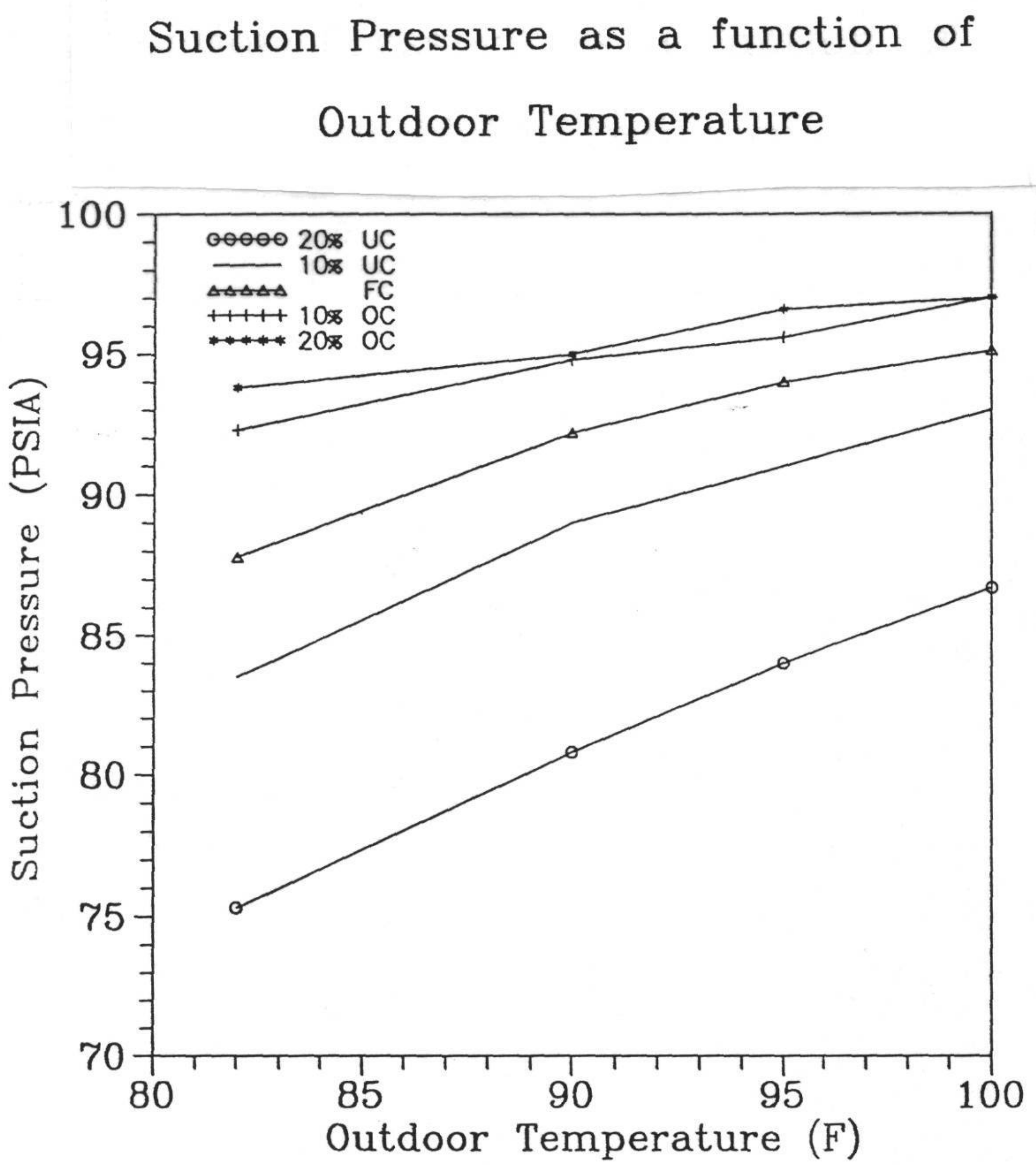

Figure 4.12 - Suction Pressure as a Function of Outdoor Temperature

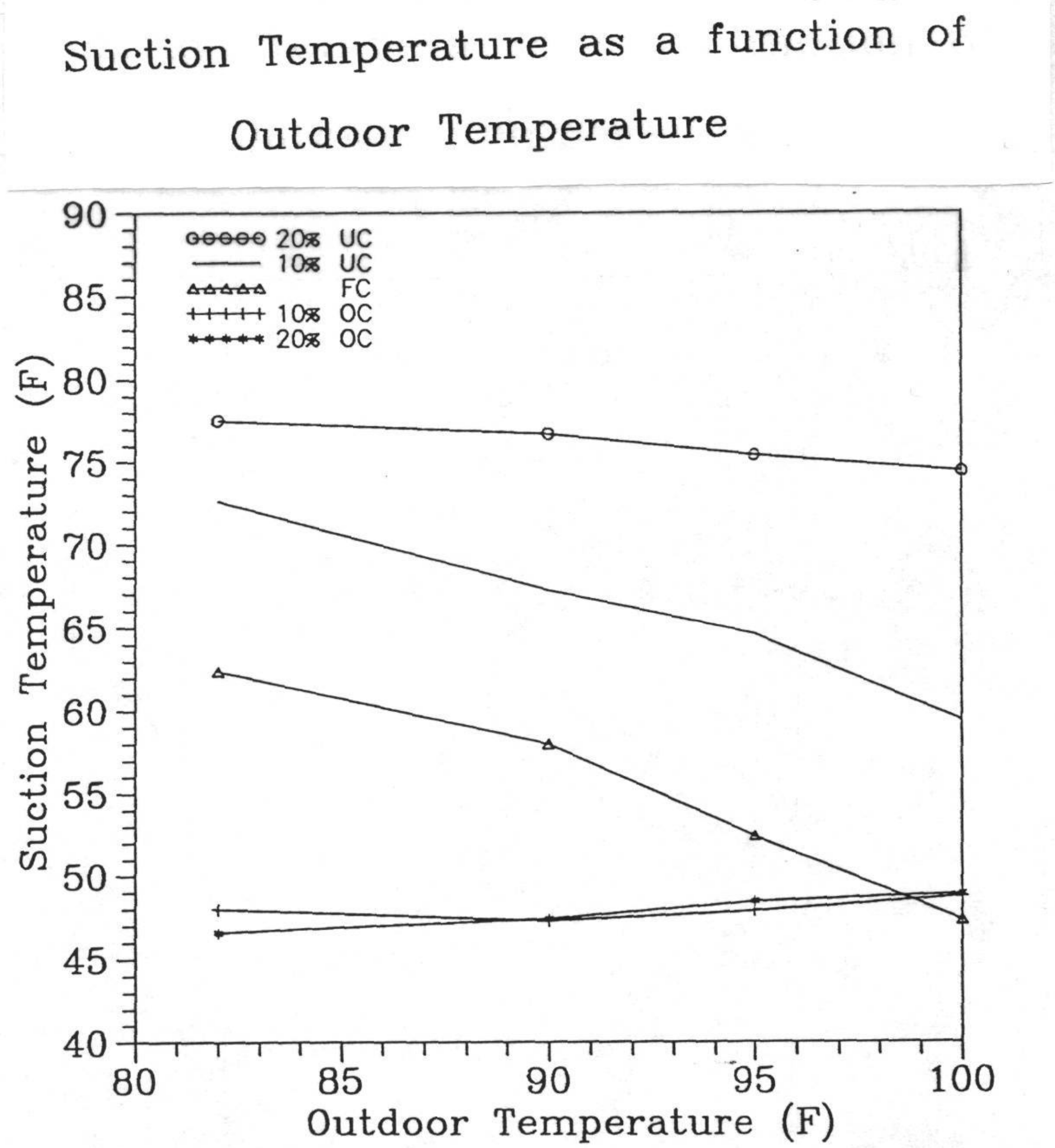

Figure 4.13 - Suction Temperature as a Function of Outdoor Temperature

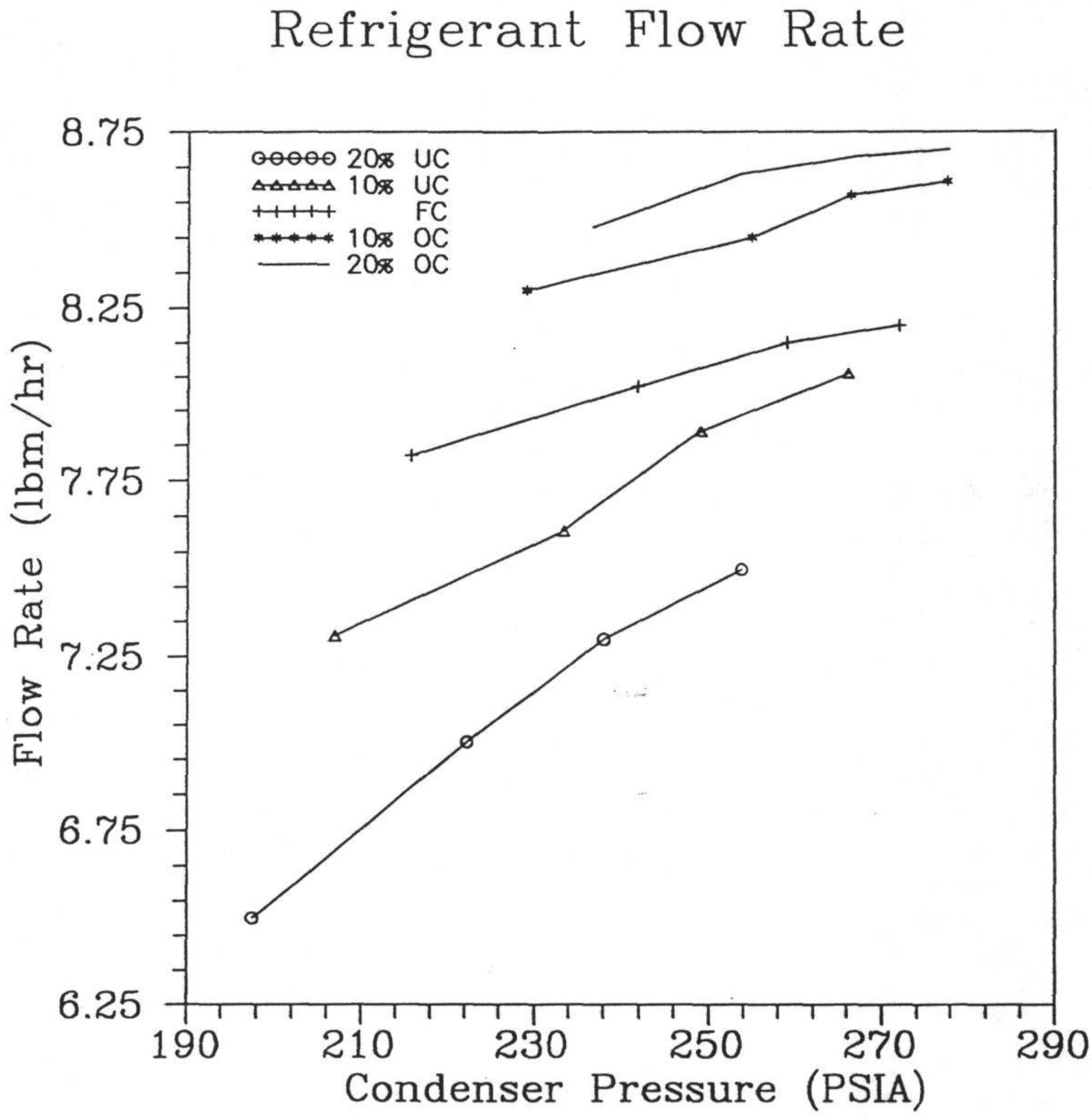

Figure 4.14 - Test Refrigerant Flow Rate as a Function of Condenser Pressure

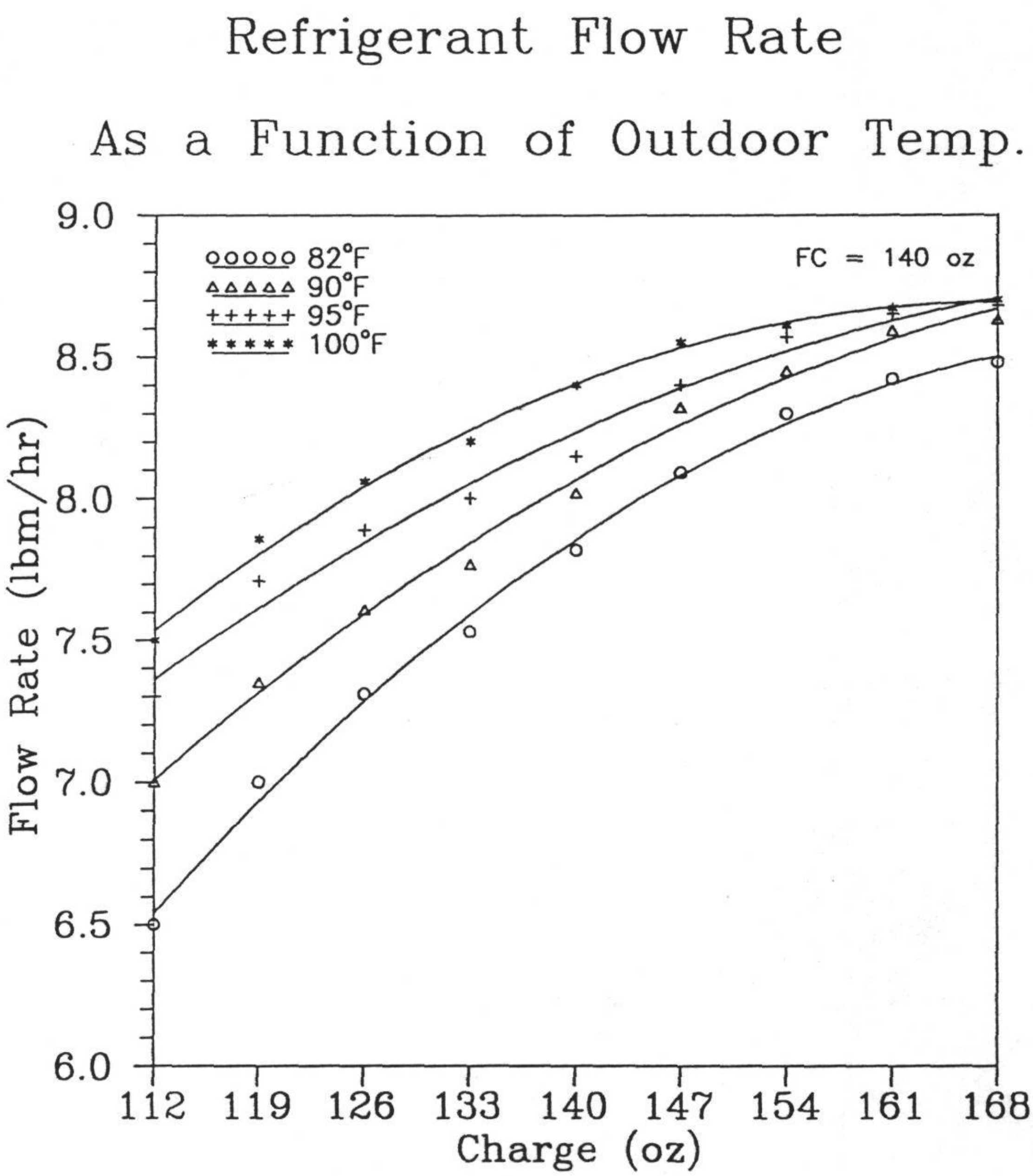

Figure 4.15 - Test refrigerant Flow Rate as a Function of Outdoor Temperature

For different outdoor room temperatures, the subcooled temperature was constant at $10^{\circ}F$ and $14.5^{\circ}F$ for 20% and 10% undercharged conditions, respectively. As shown in Figure 4.16, the subcooled temperature dropped for overcharging conditions when the outdoor room temperature increased from $82^{\circ}F$ to $100^{\circ}F$. For instance, the subcooled temperature dropped from $26.5^{\circ}F$ to $19.6^{\circ}F$ for 20% overcharged conditions.

As the amount of the refrigerant charge in the system increased, the capacity of the unit decreased. Increase in outdoor room temperature was another factor for decreasing the capacity during the overcharging conditions. Figure 4.6 shows at the $82^{\circ}F$ and $95^{\circ}F$ outdoor temperatures, the capacity of the unit dropped by 6.7% and 12.3% for 20 percent overcharged conditions, respectively. The drop in total capacity (KBtu/hr) is higher for higher outdoor room temperatures. As the amount of refrigerant in the system increased, the superheat at the exit of the indoor coil reduced and reached saturation as shown in Figure 4.17 for the overcharged conditions. The temperature difference between the compressor outlet and saturation temperature at obtained pressures is shown in Figure 4.18. This graph shows that there was no saturated vapor coming out the compressor outlet. This means there was no slugging and flooding found for the overcharging.

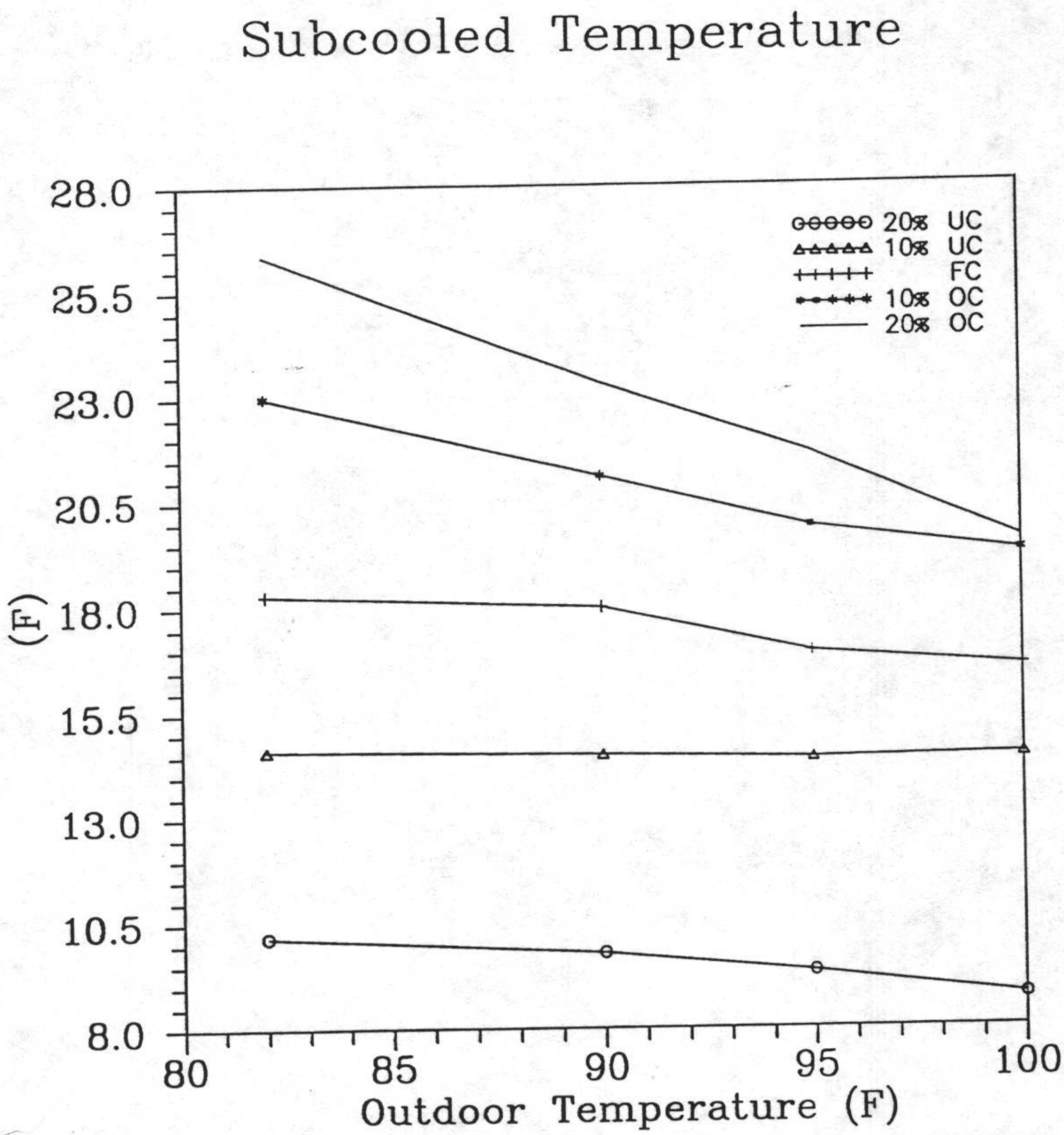

Figure 4.16 - Subcooled Temperature as a Function of Outdoor Temperature and Charge

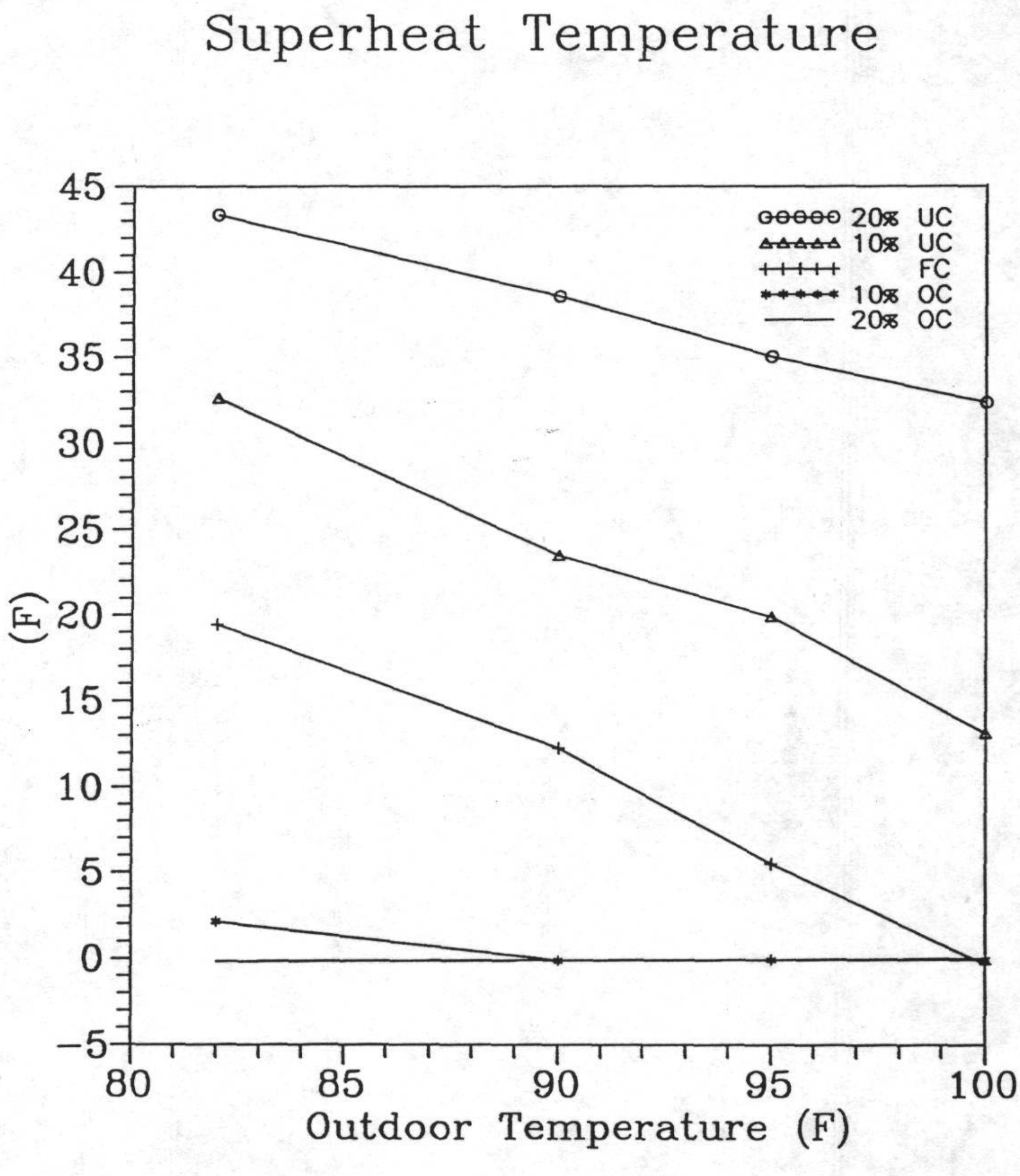

Figure 4.17 - Superheat Temperature as a Function of Outdoor Temperature and Charge

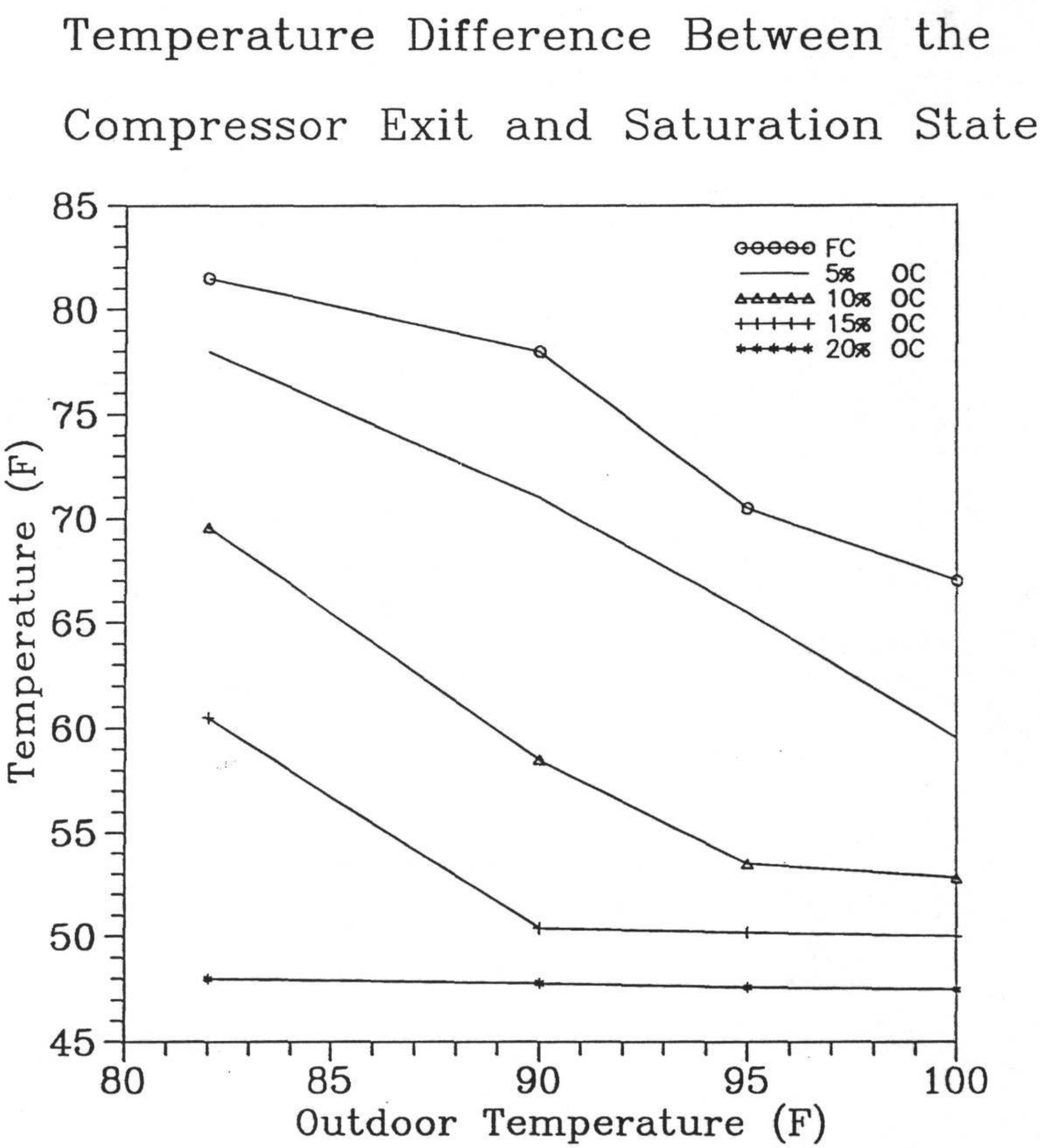

Figure 4.18 - Temperature Difference Between the Compressor Outlet and Saturation State

1.1 Sensible Heat Ratio (SHR)

The sensible heat ratio is defined as the ratio of the sensible capacity to the total capacity of the unit. Figure 4.19 shows the SHR as a function of charge and outdoor temperatures. The SHR increased as the amount of the charge in the system increased systematically. The SHR increased linearly at 82°F, 90°F, 95°F, and 100°F outdoor temperatures. For instance, the SHR was 0.727 for 20% undercharging at 82°F outdoor temperature. It increased linearly to 0.745 and 0.774 for full charged and 20% overcharged tests. As the outdoor temperature increased, the SHR increased too for a given charge. The effect of outdoor temperature was more noticeable on full charged and overcharging than undercharging. For the 20% undercharged tests, the SHR increased by 2.5% when the outdoor temperature increased from 82°F to 100°F. However, for the same increase in outdoor temperature, the SHR increased by 6% for 20% overcharged condition.

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

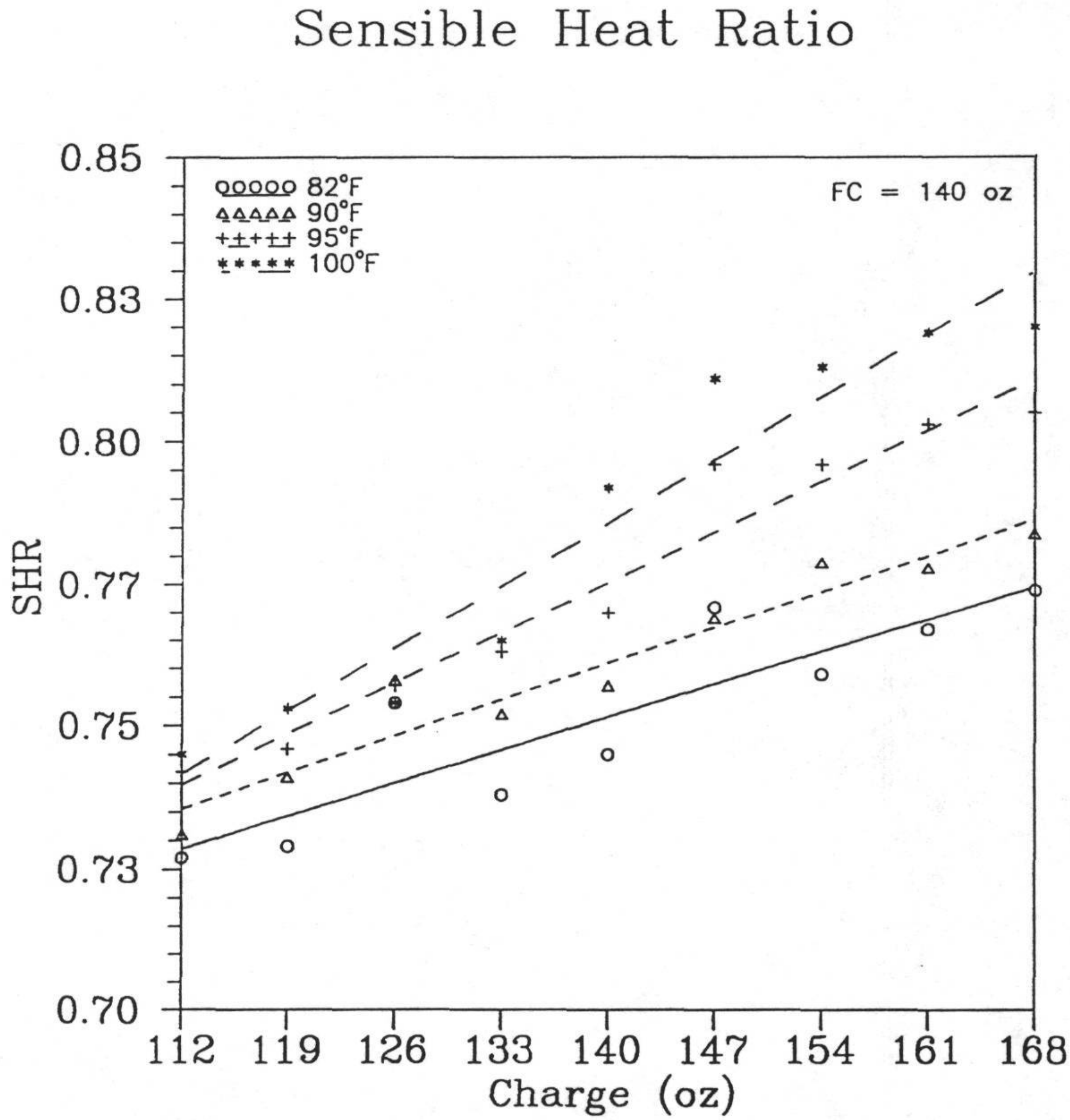

Figure 4.19 - Sensible Heat Ratio as a Function of Outdoor Temperature and Refrigerant Charge

(D) tests were performed on the under and overcharged conditions. Total capacity (test-C) is shown in Figure 4.20. The unit capacity (Btu/hr) peaked at 31.83 KBtu/hr for the fully charged condition. The drop in capacity was more dramatic for undercharging than overcharging. The capacity dropped to 24.84 Kbtu/hr for 20% undercharging and 30.82 KBtu/hr for 20% overcharging.

The cooling cyclic tests were conducted by cycling the compressor 6 minutes "on" and 24 minutes "off" for under and overcharged conditions. Figures 4.21 shows the coefficient of degradation (C_D). This coefficient is a measure of the efficiency loss due to the cycling of the unit. C_D peaked at 0.25 for 5% undercharging. It dropped to 0.15 for 20% overcharging and 0.21 for 20% undercharging.

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. Figures 4.22 through 4.29 show the net capacity for cyclic tests as a function of time for different refrigerant charges. 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 [16].

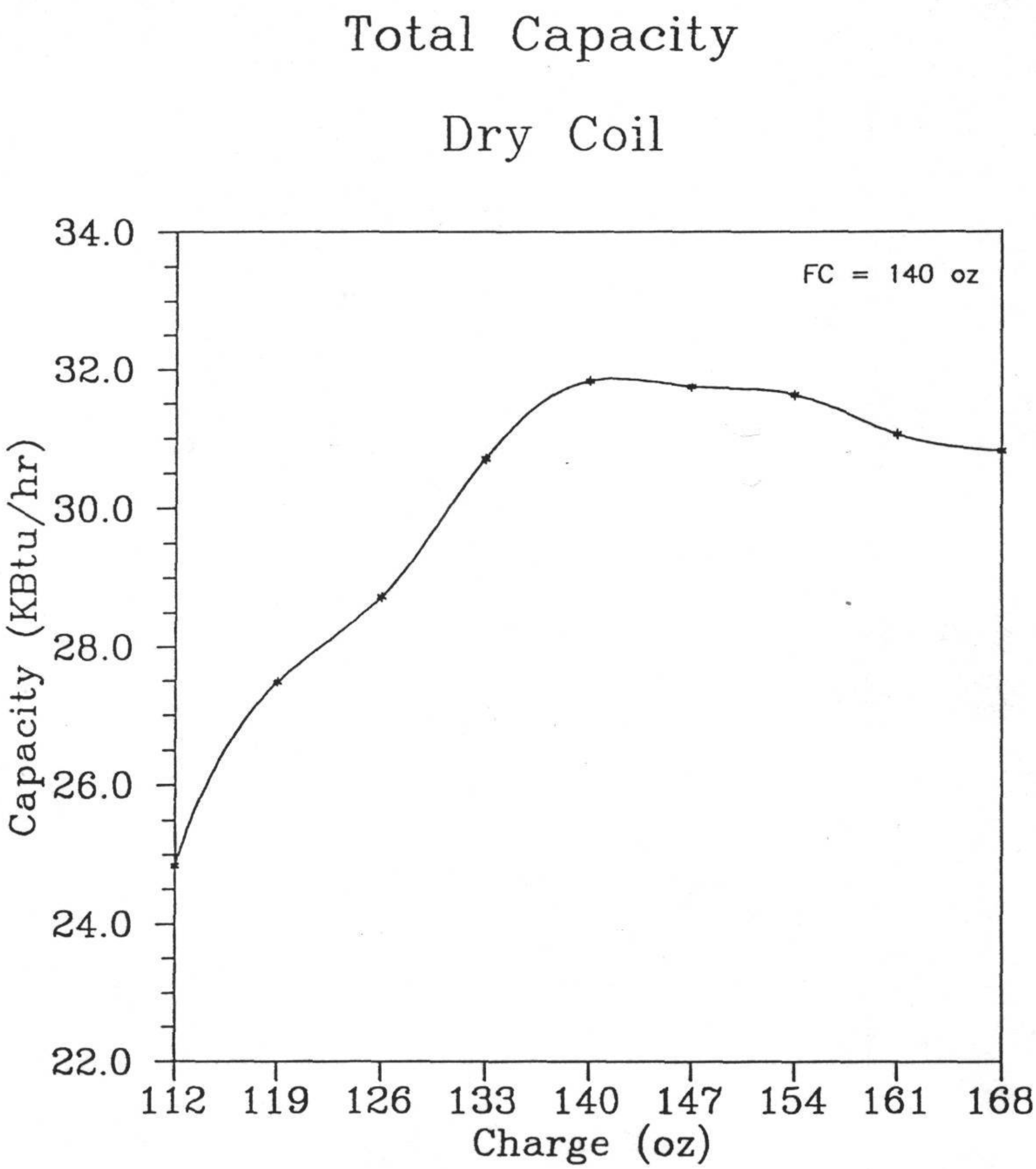

Figure 4.20 - Total Capacity for Dry Coil Test

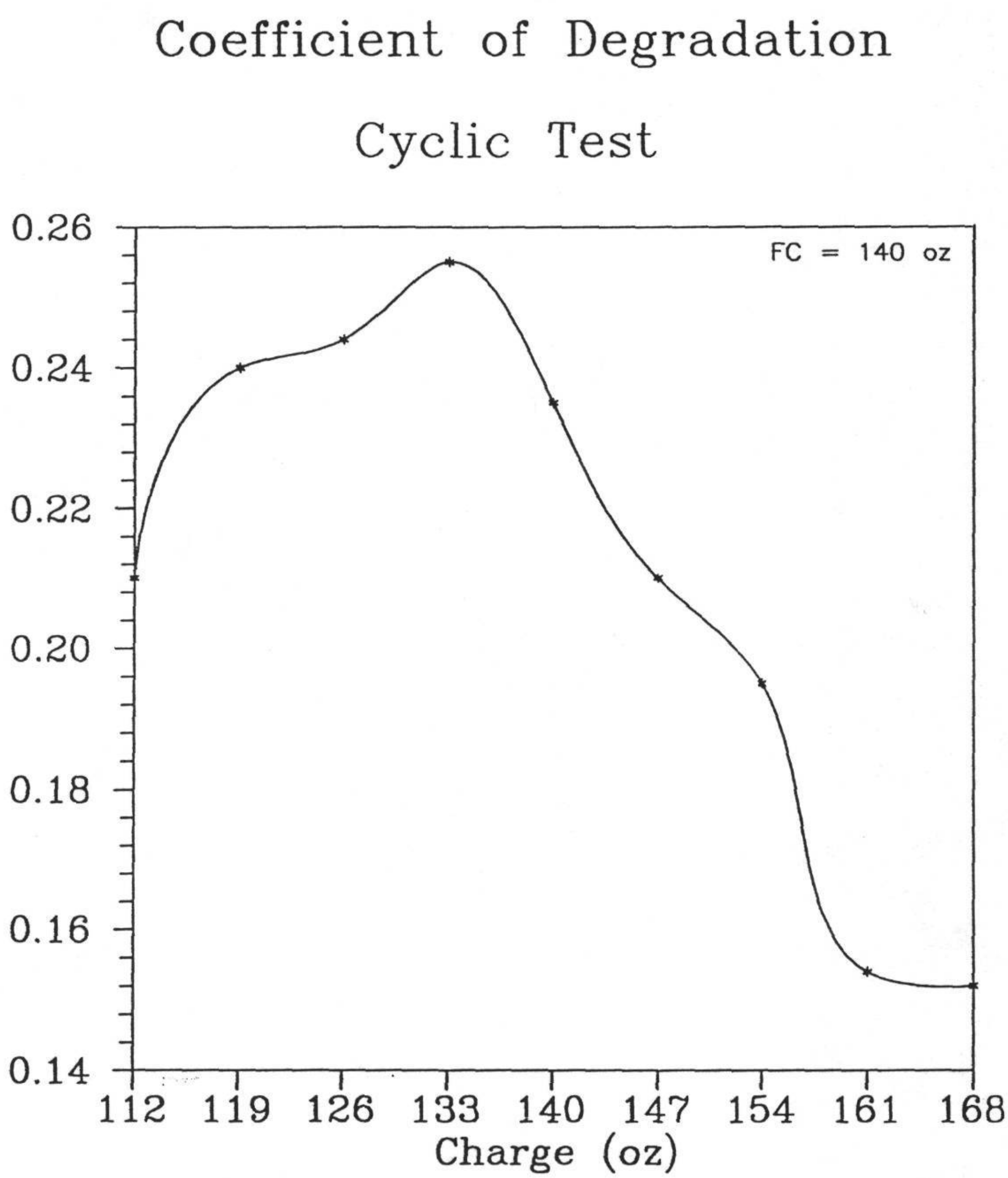

Figure 4.21 - Coefficient of Degradation for Under/
Overcharging Tests

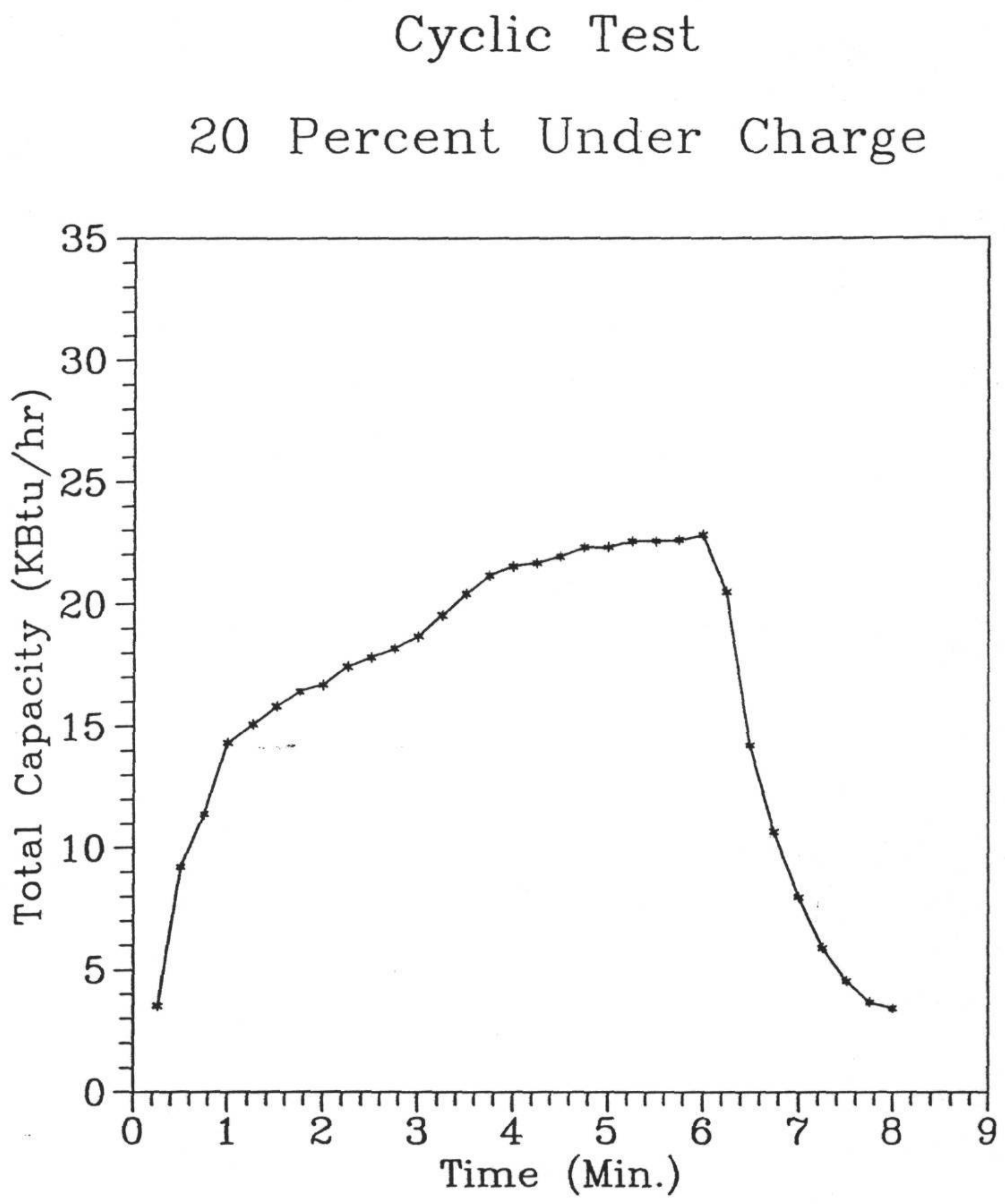

Figure 4.22 - Capacity as a Function of Time for 20% Undercharged Cyclic Test

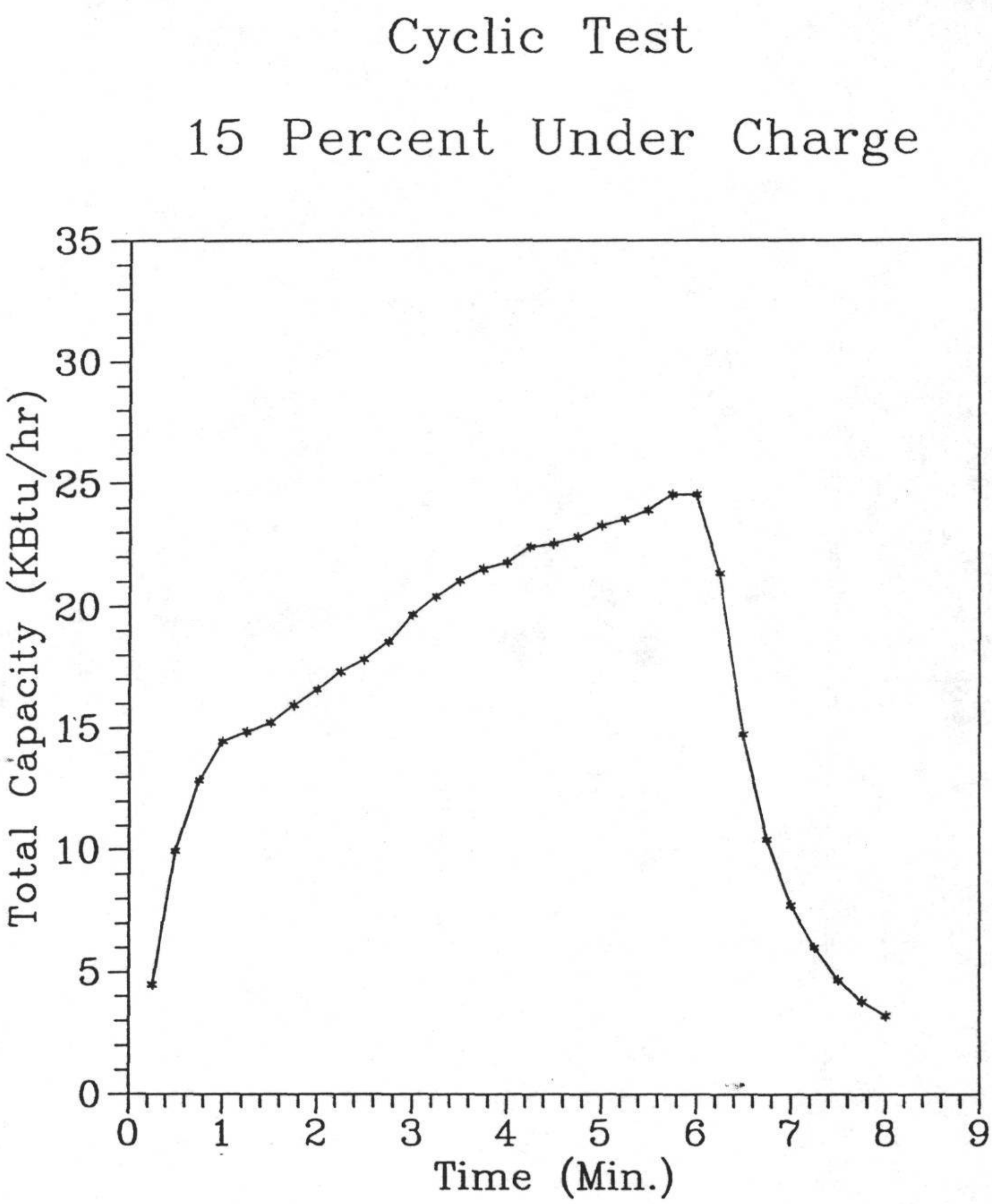

Figure 4.23 - Capacity as a Function of Time for 15% Undercharged Cyclic Test

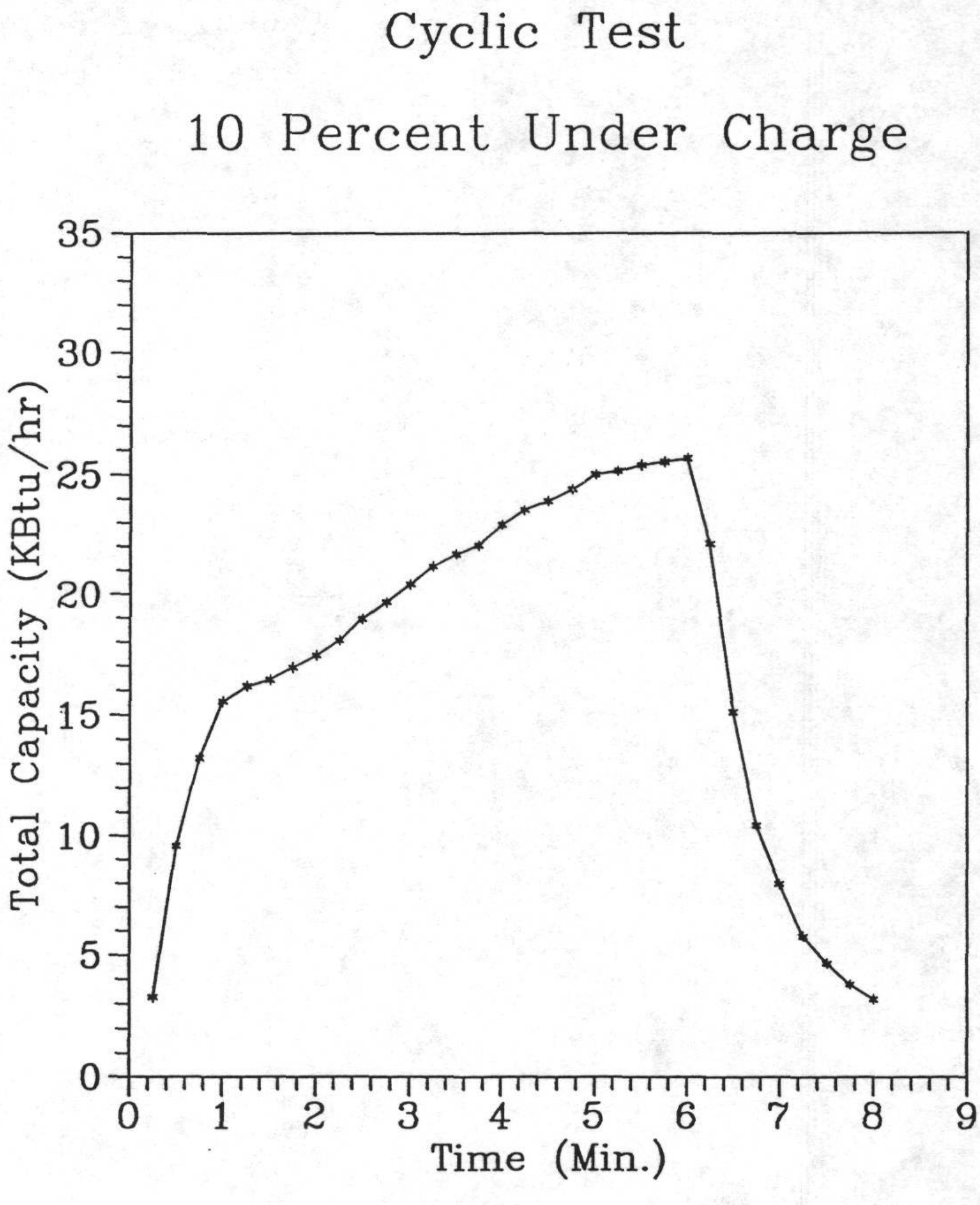

Figure 4.24 - Capacity as a Function of Time for 10% Undercharged Cyclic Test

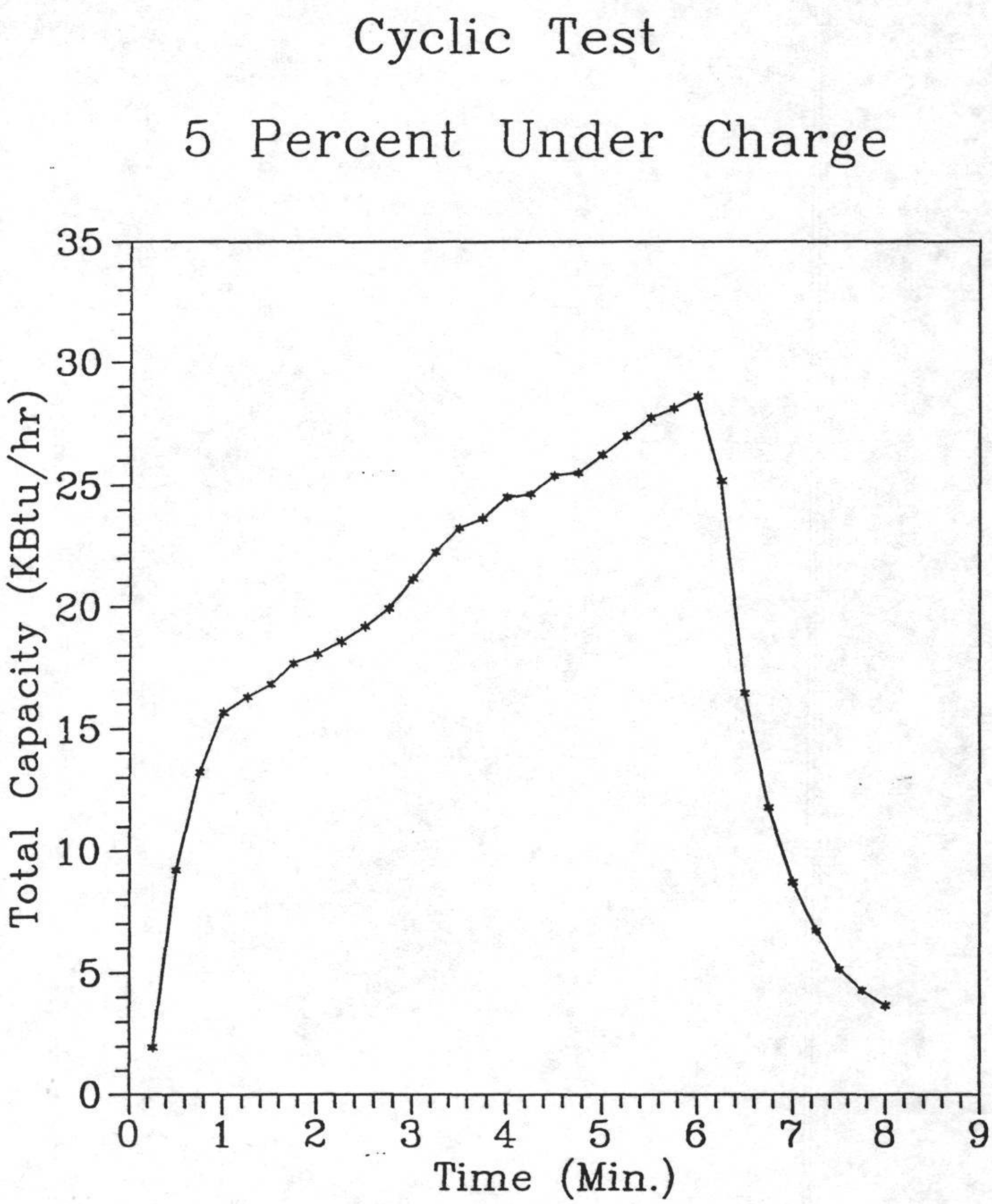

Figure 4.25 - Capacity as a Function of Time for 5% Undercharged Cyclic Test

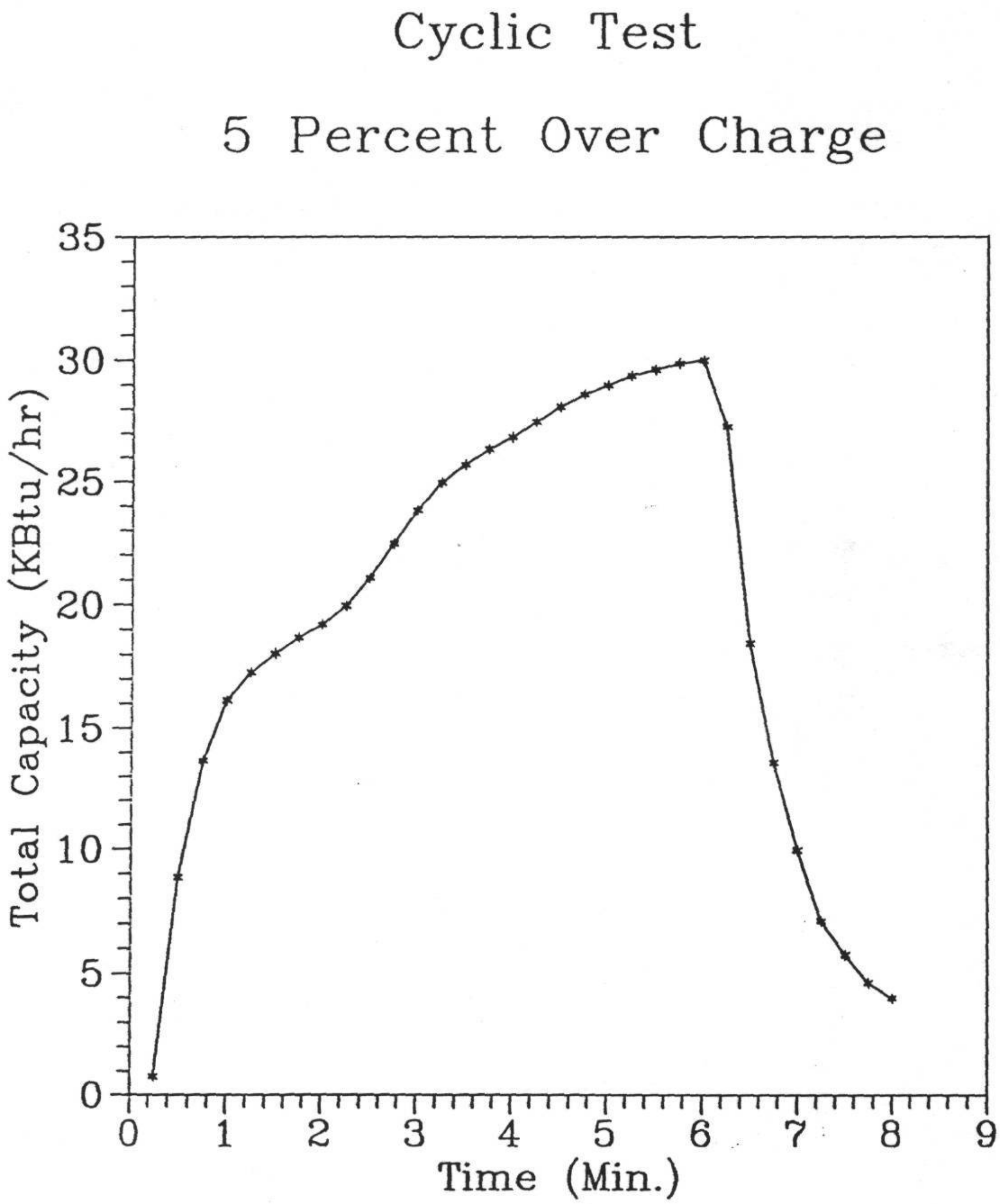

Figure 4.26 - Capacity as a Function of Time for 5% Overcharged Cyclic Test

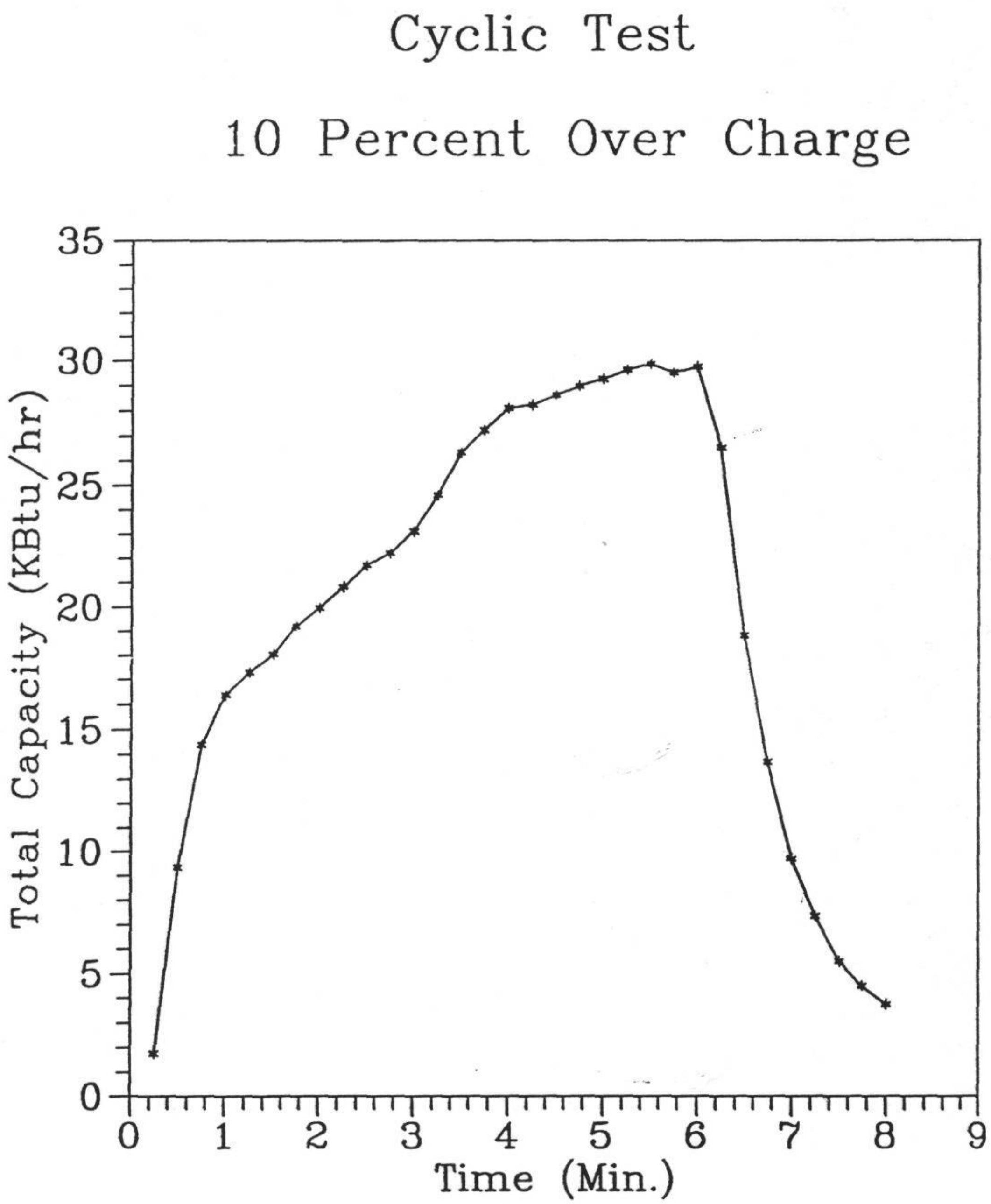

Figure 4.27 - Capacity as a Function of Time for 10% Overcharged Cyclic Test

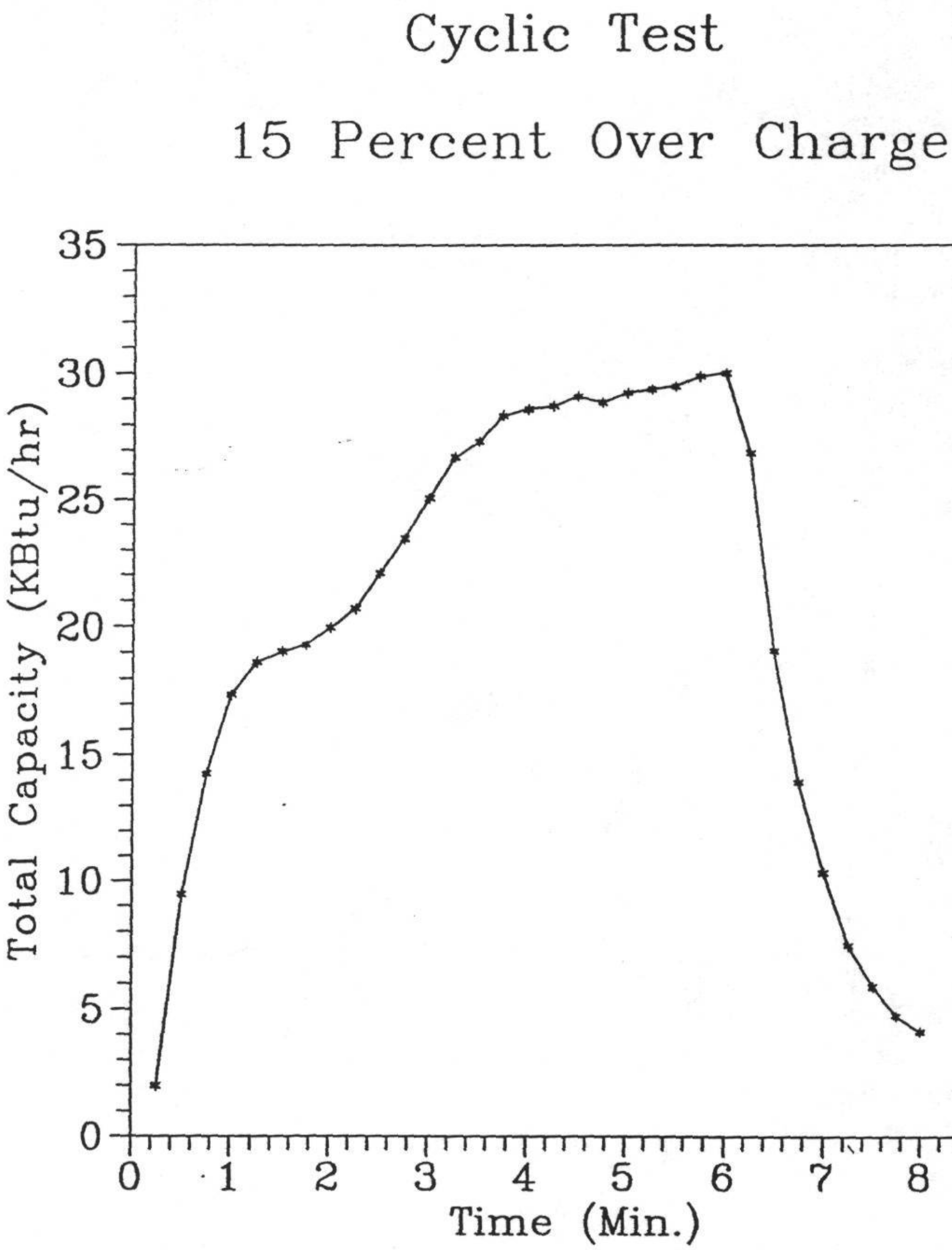

Figure 4.28 - Capacity as a Function of Time for 15% Overcharged Cyclic Test

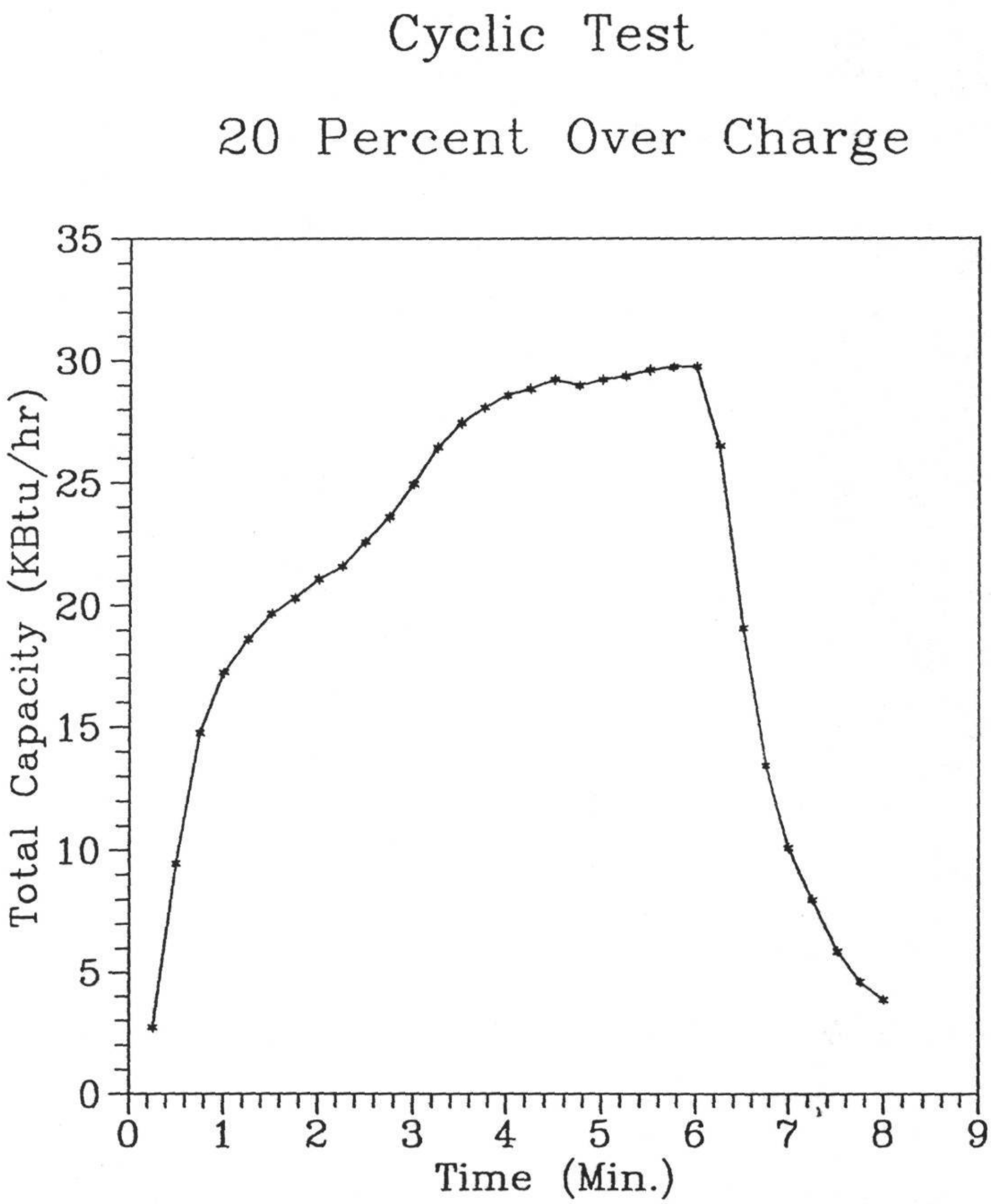

Figure 4.29 - Capacity as a Function of Time for 20% Overcharged Cyclic Test

As the amount of refrigerant charge increased in the system, the unit capacity increased during the first minute of the start-up. During the first minute of the start-up, the unit capacity rose to about 15 KBtu/hr for 20%, 15%, and 10% undercharging tests. It increased to approximately 18 KBtu/hr for the rest of the tests. During the last two minutes of "on" time, the total capacity leveled off for 20% and 15% overcharging and 20% undercharging. The total capacity increased steadily after the first minute of the compressor start-up to its maximum capacity for full charged and 15%, 10%, and 5% undercharged as well as 5%, and 10% overcharged. 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 y the power input in watts (w) at any given set of rating conditions, expressed in Btu/wh.

1. Steady State Tests (Wet Coil)

Figure 4.30 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°F outdoor room temperature for the fully charged condition. As the outdoor room temperature increased, the peak of the curves shifted to the

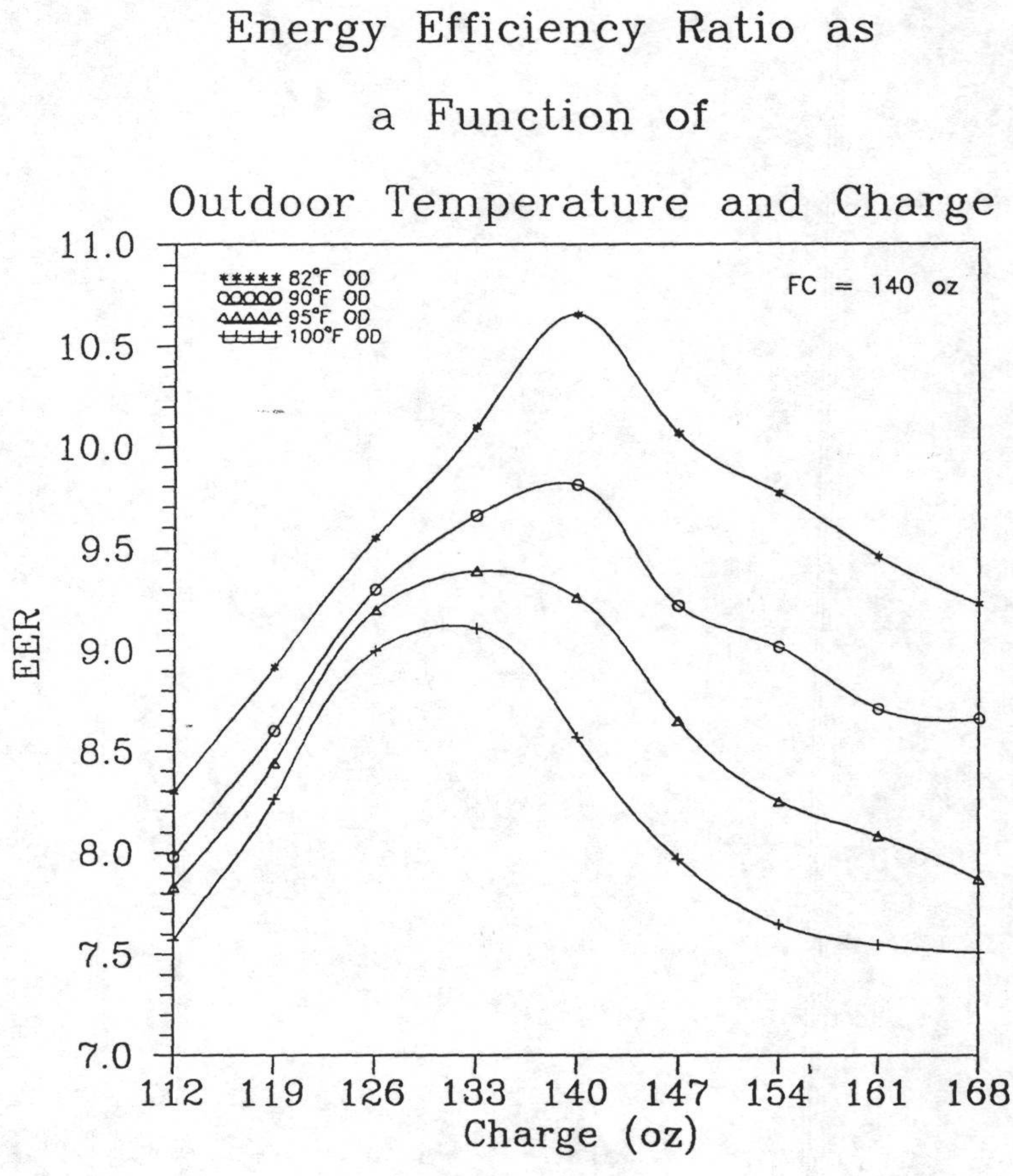

Figure 4.30 - Energy Efficiency Ratio as a Function of Outdoor Temperature and Charge (wet coil)

left (lower charge). For instance, the EER curves for 95^{o}F, and 100^{o}F outdoor temperatures were maximum at 5% under charge rather than full charge. At 95^{o}F, and 100^{o}F outdoor temperatures, the fully charged EER actually dropped by 1.4% and 6.4%, respectively. The drop in EER was more noticeable for the undercharged than overcharged conditions at outdoor room temperature of 82^{o}F. The EER dropped to 8.3 for 20% undercharging and 9.23 for 20% overcharging. 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 shown in Figure 4.31. The condenser outlet temperature decreased as the refrigerant charge in the system increased. This drop was due to the increase in the subcooled temperature. As the condensing temperature increased, the power (kw) to the compressor increased too. Figure 4.32 shows the power consumption of the outdoor unit as a function of outdoor room temperatre and charge. For 20 percent undercharging, the power (kw) increased by 19.6% when the outdoor room temperature increased from 82^{o}F to 100^{o}F. However, for the same increase in outdoor room temperature, the power increased by 8.5% for 20% overcharging.

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.

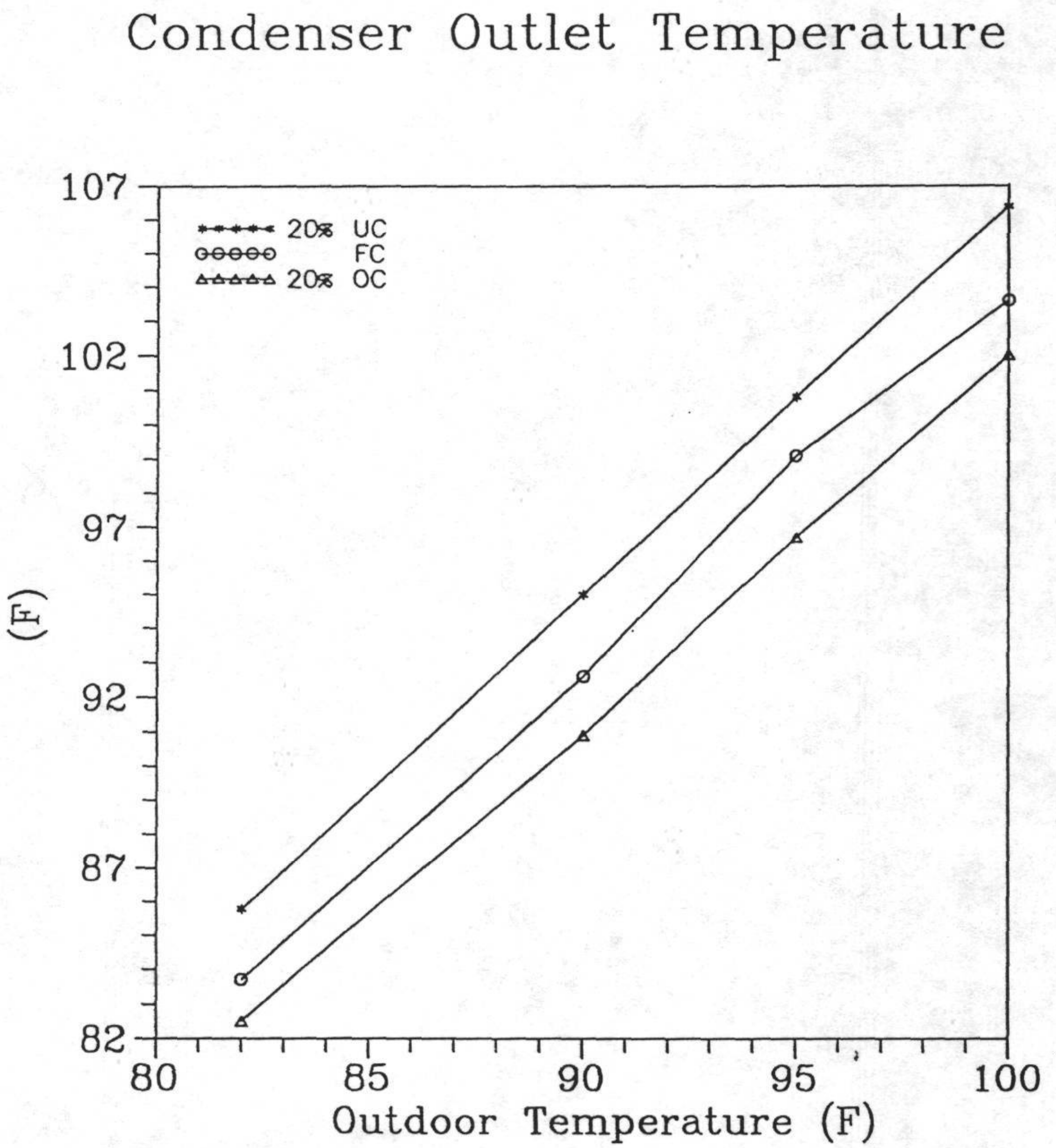

Figure 4.31 - Under/overcharging and Fully Charge Condenser Temperatures

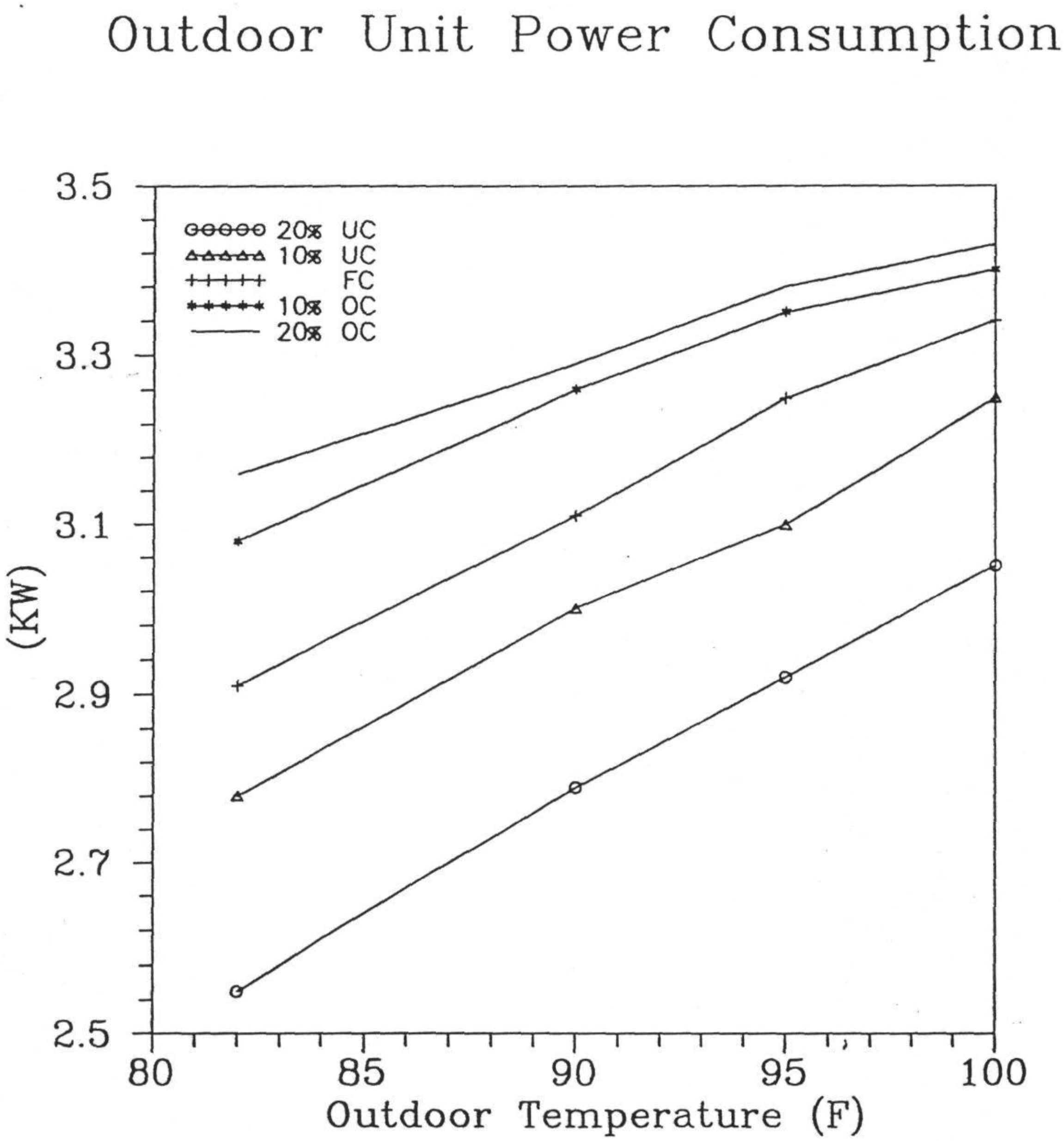

Figure 4.32 - Outdoor Unit Power Consumption as a Function of Outdoor Temperature and Charge

EER_C for the steady state dry coil test is shown in Figure 4.33 as a function of charge. The decrease in EER_C was due to the increase of compressor power and lower capacity. The power is shown in Figure 4.34. For 20% over charging, the power was 3.02 KW. It dropped to 2.52 KW for 20% undercharging condition. The maximum EER_C peaked at 9.77 for fully charged condition. EER_C dropped to 8.45 for 20% undercharging and to 9.03 for 20% overcharging.

The cooling cyclic tests were conducted by cycling the compressor 6 minutes "on" and 24 minutes "off". The Energy Efficiency Ratio for cyclic test (EER_D) is a ratio calculated by dividing the total sensible cooling during first eight minutes in Btu/hr by the total power (kw) during "on" time. Figure 4.35 shows the Energy Efficiency Ratio for cyclic test (EER_D). EER_D was 7 for 20% undercharging. It increased by 12.8% at full charge. During the overcharged conditios, EER_D was approximately constant.

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 4.36. As expected, the SEER curve peak occured at full charge. The drop in SEER was more dramatic for the undercharged than overcharged conditions. It dropped to 7.5 for 20% undercharged and 8.47 for 20% overcharged conditions.

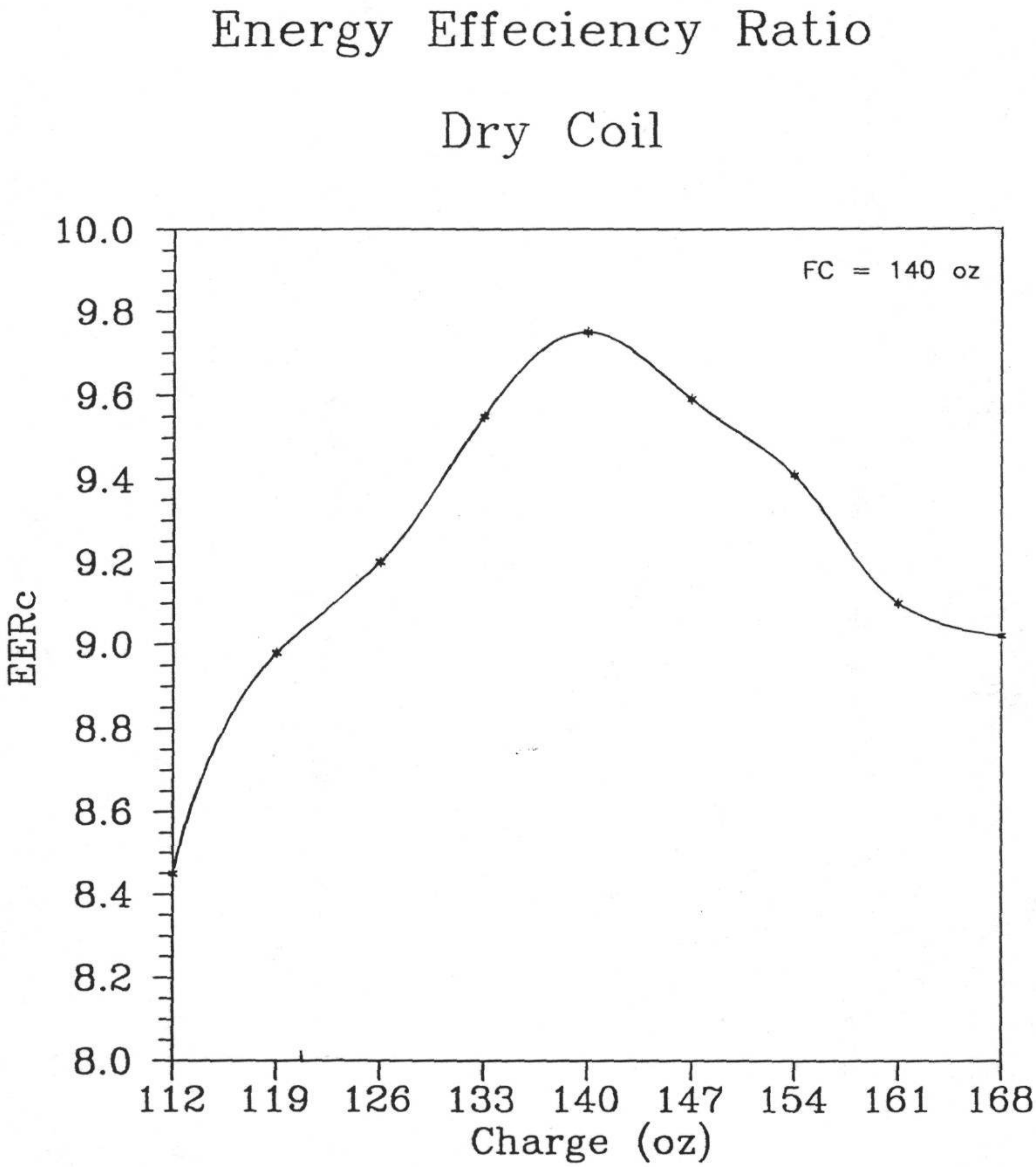

Figure 4.33 - Energy Efficiency Ratio (dry coil test)

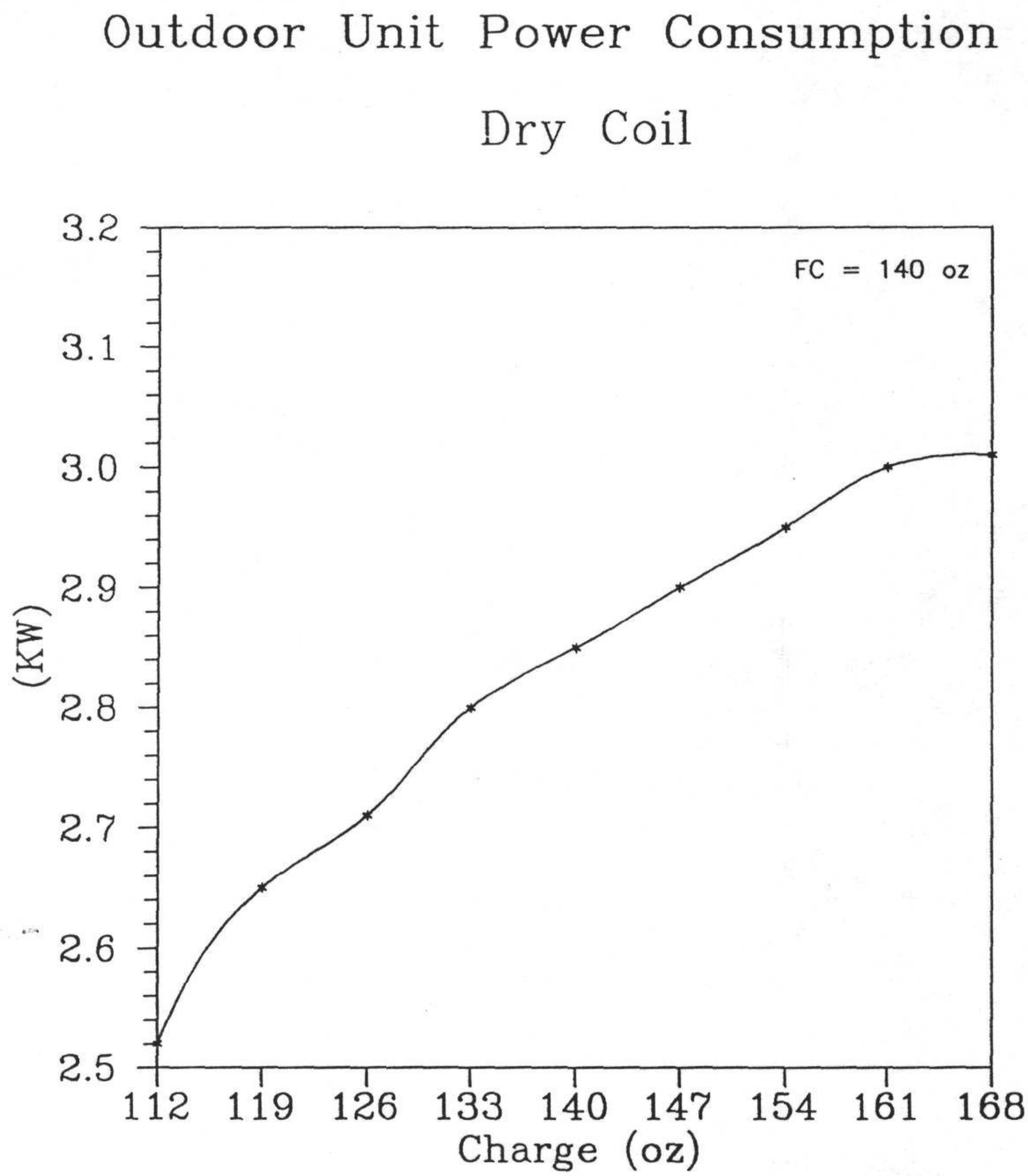

Figure 4.34 - Outdoor Unit Power Consumption as a Function of Refrigerant Charge (dry coil test)

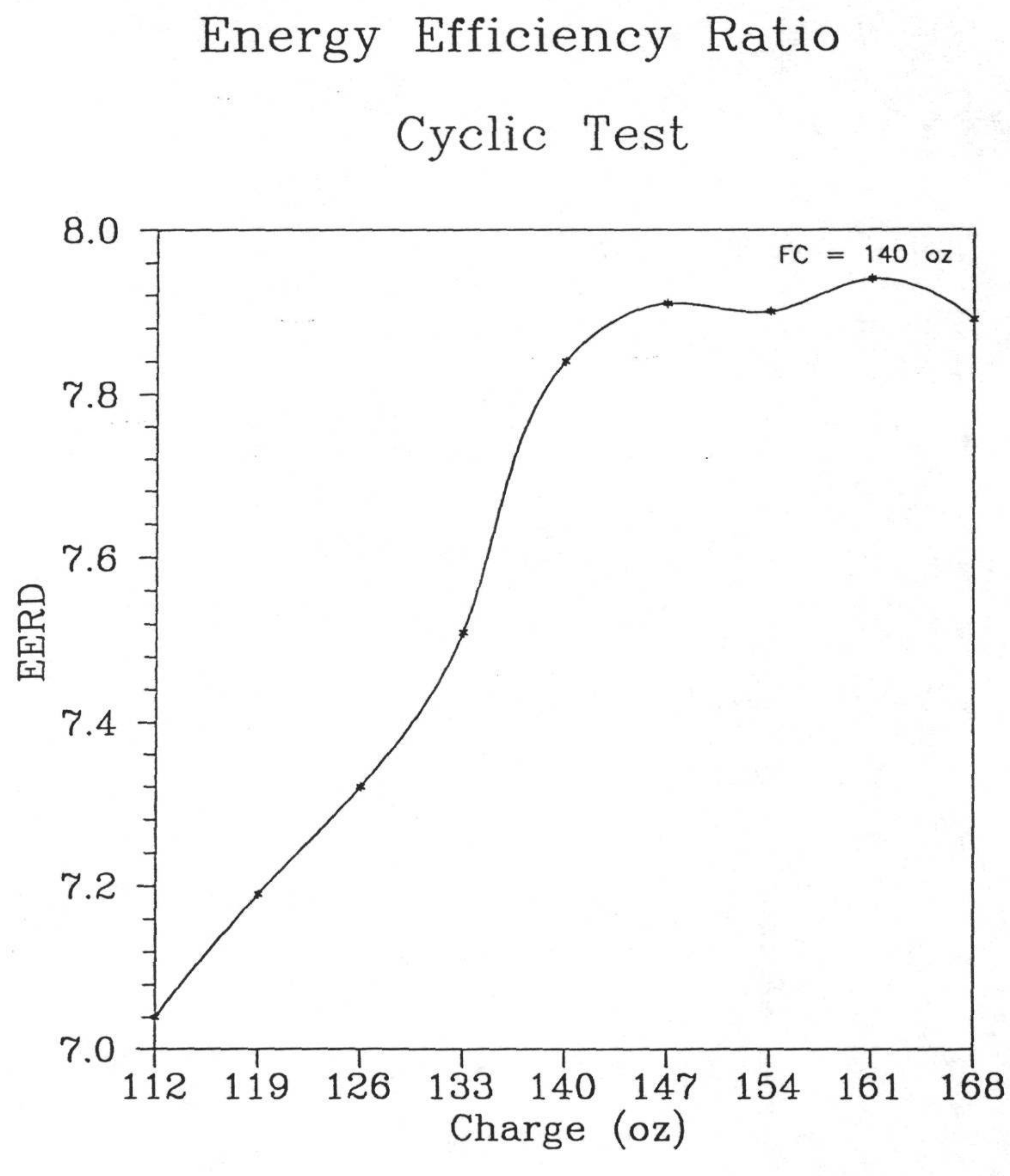

Figure 4.35 - Energy Efficiency Ratio (cyclic test)

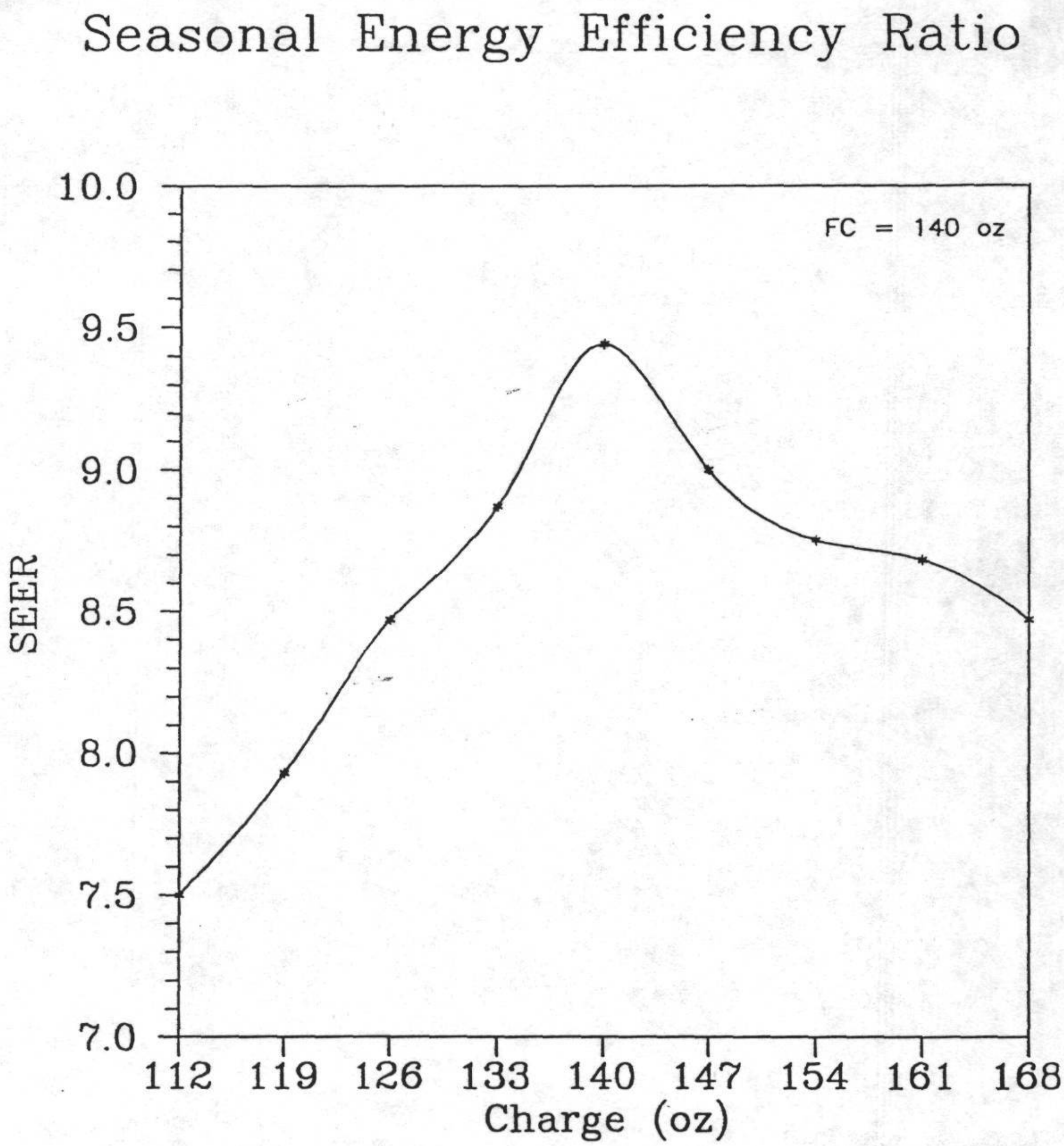

Figure 4.36 - Seasonal Energy Efficiency as a Function of Refrigerant Charge

CHAPTER 5

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 capillary tube expansion. To achieve such an objective, a literature review was conducted, an experimental setup constructed, and data collected and analyzed. The literature review yielded only a handful of papers related to under and overcharging. While some investigators had presented limited steady state data, no systematic study on the effects of improper charging on cyclic or seasonal performance had been reported.

A 3-ton split system air conditioner with capillary tube expansion was instrumented to evaluate the steady state and cyclic performance of the air conditioner during 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), EER, and SEER decreased with increasing outdoor temperature for the fully charged case. The sensible heat ratio increased as the outdoor temperature increased.

One surprising result was the increase of the capacity for the undercharged condition as compared to the full or overcharged case. For the undercharged test, the capacity increased as the outdoor temperature increased. One possible explanation for this behavior was in the changes of the refrigerant flow rate by the capillary tube for different charges.

In general, the degradation of performance was larger for undercharaging than that for overcharging. The measure of seasonal performance, the SEER, dropped from 9.5 to 7.5 for 20% undercharging while only dropping to 8.4 for 20% overcharging. The data for capacity showed similar trends. A measure of performance of interest to electric utilities is the capacity and power during the hottest part of the summer days. The 100°F test would provide some hints at this performance. Capacity peaked for 5% undercharging at 34.0 kBtu/hr. It dropped to 26.3 kBtu/hr for 20% undercharging and 28.9 kBtu/hr for 20% overcharging.

As the outdoor temperature increased, the subcooling was constant for undercharging where it increased for the overcharging tests. The superheat at the outlet of the evaporator decreased as the outdoor temperature increased. For 10%, 15%, and 20% overcharging, the refrigerant at the outlet of the evaporator was saturated. While potentially saturated conditions at the evaporator outlet could indicate possible introduction of wet vapor into the compressor, more detailed measurements at the suction inlet would have to be taken to confirm it.

Because of the limited budget for this study, one of its primary limitations was that data were taken on only one central air conditioner. Some of the trends measured in this system may be a characteristic of this system alone and may not be general trends expected across many of the air conditioners or heat pump systems currently in use. For instance, this system had a capillary tube expansion device, no accumulator, and moderately sized coils. Different expansion devices (thermal expansion valves and orifices) and different sized capillary tubes may react differently to improper charging than did the capillary tubes in this system. Because many heat pumps employ an accumulator which can store excess refrigerant, these systems may have different overcharging characteristics than measured here. The higher efficiency units employ larger coils than those used on this unit. It would seem that a thorough study should include the effect of different expansion devices (types and size), the accumulator, and different coil sizes on the performance of the unit. A comprehensive test plan could help verify those characteristics of the system that are more generic and those that are typical of specific systems.

This study has attempted to answer the question of what the effects of loss of charge or improper charge may have on the performance of a unit. The next logical question to ask would be what fraction of units out in the field have appreciable leaks or are improperly charged. It appears that loss of charge may not be as an important issue as improper charge. Recent conversations with air conditioner manufacturer indicated that if units are installed properly, the amount of refrigerant leakage may be about an ounce over a ten year period. If the charge of the unit is not set properly from the beginning, then the system would potentially operate at less than peak performance. One recommendation would be to study the different charging techniques used in the field and evaluate how effective each is in establishing an acceptable charge. A list of these has already been provided in the earlier Texas Power and Light study[7]. Another related recommendation would be to evaluate the charge on a random sample of units in the field. A large enough sample should provide a good indication of how big a problem improper charging is in residential sized units.

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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	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^{o}F	General Eastern/ DEW-10	0.5^{o}F
Thermocouples	30 AWG	Omega/Type T	0.5^{o}F

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