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Monitoring the Performance of a Residential Central Air Conditioner under Degraded Conditions on a Test Bench

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DRAFT

ABSTRACT

This report presents the measured degradation in performance of a residential air conditioning system operating under degraded conditions. Experiments were conducted using a R-22 three-ton split-type cooling system with a short-tube orifice expansion device. Results are presented here for a series of tests in which the various commonly occurring degraded conditions were simulated on a test bench.

At present, very little information is available which quantifies the performance of a residential cooling system operating under degraded conditions. Degraded performance measurements can provide information which could help electric utilities evaluate the potential impact of system-wide maintenance programs. This report also discuss the development of a diagnostic procedure based on measurement of refrigerant and air side temperatures.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

C_p	Specific heat at constant pressure, Btu/lb ^oF
h	Enthalpy of fluid, Btu/lb.
m	Mass flow rate, lb./min
Q	Air flow rate, ft^3/min
T	Temperature, ^oF
v	Specific volume, ft^3/lb
w	Specific humidity, lbm/lba

Subscripts

a	air
c	condenser
e	evaporator
i	inlet
n	standard conditions
o	outlet
od	outdoor
p	constant pressure
r	refrigerant
ra	return air
sa	supply conditions

GLOSSARY OF TERMS

ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating and Air Conditioning Engineers
Btu	British thermal unit
CFM	Cubic feet per minute
DBT	Dry Bulb Temperature
DOE	Department of Energy
EER	Energy Efficiency Ratio
floodback	Condition in which liquid enters the compressor
fpm	Feet per minute
HVAC	Heating Ventilating and Air Conditioning
lbm	Pounds mass
lbma	Pounds mass of dry air
NIST	National Institute of Standards and Technology
OD	Outer diameter
O.D.	Outdoor
ORNL	Oak Ridge National Laboratory
RH	Relative Humidity
SHR	Sensible Heat Ratio
Subcooling	the difference between refrigerant saturation temperature and the measured temperature at the outlet of condenser
Superheat	the difference between refrigerant saturation temperature and the measured temperature at the inlet of compressor
WBT	Wet bulb temperature

INTRODUCTION

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

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

AIR CONDITIONING INSTALLATION AND SERVICE

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

NEED FOR PREVENTIVE MAINTENANCE AND DIAGNOSIS SYSTEM

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

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

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

OBJECTIVE OF CURRENT INVESTIGATION

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

The primary focus of this work is to measure energy consumption, cooling capacity and energy efficiency ratio (EER) during degraded conditions. A detailed discussion on performance under degraded conditions are presented in this report. The long term goal of this work is to classify degraded conditions for the most widely experienced problems on a test bench so that eventually field diagnosis procedures can be developed.

LITERATURE REVIEW

SERVICE AND FAILURE PATTERNS

Several studies have investigated HVAC service problems. Karger and Carpenter discussed failure patterns of residential air conditioning units based on their survey of 531 failed units [16]. This is one of the first studies conducted on failed air conditioning systems. Their study indicated that failure rates for electrical controls and miscellaneous electrical devices represented 31.4% of the total number of failures and refrigerant leaks constituted 17.2% of the total failure. The failure rates of compressors was 13.5% and outdoor fans was about 11.5%.

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

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

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

Hewett et al., sought to quantify the energy and demand savings through efficiency tune-ups of commercial unitary cooling equipment in the service territory of a major New England utility [17]. They tuned up and monitored 18 systems (7 dual compressor systems) which range between 4 to 15 tons cooling capacity. Their results show that reduced evaporator air flow condition exist in 3 units(17%), overcharging was found in 10 units (60%) and undercharging was found in 8 units (40%). They also found out that none of the Thermostatic expansion valves(13 systems) provided correct superheat. Efficiency tune-ups results in average energy savings of 9 to 10%. They conclude that "tune-ups do not appear to be cost-effective to the utility except perhaps for larger equipment".

DEGRADATION STUDIES

Houcek studied the effect of improper charging by conducting experiments on a 2-ton split system with 37 feet of interconnecting refrigerant lines [18]. Supply air entering the indoor coil was maintained at 80° F DBT and 67° F WBT. Air entering the outdoor unit was maintained at different conditions: 70° F, 82° F, 95° F and 100° F. Houcek's experiments showed that at 95° F outdoor conditions, overcharging by 23% decreases the operating cost by 0.5%. However, this was observed to cause a floodback condition (Floodback describes a situation when liquid refrigerant enters the compressor. Compression of liquid refrigerant will cause mechanical failure of compressor). For undercharging, it was estimated that the operating cost increased by as much as 52%. He recommended a new device (a Visual Accumulator-Charger) for charging air conditioners effectively.

Farzad studied the effect of undercharging and overcharging for three expansion devices: (i) a capillary tube, (ii) a short-tube orifice, and (iii) a thermostatic expansion valve [19]. He conducted his experiments under controlled conditions specified by DOE/ARI testing procedures for Unitary Air Conditioners. He varied the system charge from +20% to -20% for different outdoor temperatures. His study showed that capillary tube expansion is more sensitive to off-charge conditions than other types of expansion valves. For a 20% undercharge the Seasonal Energy Efficiency Ratio (SEER) was reduced by 20%, while overcharging by 20% produced an 11% reduction in SEER. He found that the SEER was not very sensitive to changes from -20% to +20% of the correct charge when an orifice tube is used for an expansion device.

Authors	Unit failure rate survey	Field performance check	Degraded Experiments	Field Monitoring system	Remote Diagnostics	Service Support	Service Procedure
Karger and Carpenter (16)	* (A/C)						
Lewis (5)	*(Heat pump)						
Neal (13)		*					*
Proctor (10)		*					
Hewett (17)		*					
Houcek (18)			*				
Farzad (19)			*				
Katipamula (24)			*				
Fluke (29)				*			
Kaler (30)				*			
Dencor (31)				*	*		
Danfoss (32)				*	*	*	
Wheeler (11)							*

Table D-1. Index of authors and products

MODEL DEVELOPMENT

This chapter introduce the underlying theory behind predicting the performance of an air conditioner under degraded conditions. Description of governing physical principles are introduced in general manner without involving empirical relations. The purpose of this chapter is to propose the idea that degraded conditions can be modeled physically. Compilation of relevant empirical relations to predict heat and fluid flow to describe the overall modeling of an air conditioner under degraded conditions, is in developing stage. It is attempted here to provide an encompassing idea behind the current investigation. A description of standard cycle, degraded conditions and temperature patterns are presented here. A review of currently available models also discussed here.

THEORY OF OPERATION

A typical arrangement of components which are essential to provide comfort cooling is shown in Figure [1]. Comfort conditions during cooling season is maintained in a conditioned space through supplying sufficient amount of cool air. This cool supply air is a mixture of re-circulated air from space which is cooled and the outside fresh air which is necessary to dilute indoor pollutants. In most of the residential comfort cooling the space is conditioned through re-circulating the indoor air alone. It is assumed that infiltration is sufficient enough to dilute the indoor pollutants. Comfort condition in residences is regulated by a thermostat which controls the air conditioning unit to be on or off to keep the space at desired temperature.

When thermostat is calling for cooling (switching the air conditioning unit on) the indoor air is warmer than the desired temperature levels. Cool air is obtained through circulating warm room air across evaporator which transfers heat from warm air to cold refrigerant circulating through it in a closed circuit. The warm room air also contains moisture which condenses on the surface of evaporator (dehumidification). To maintain comfort levels it is essential that temperature and humidity both should be within the specified limits.

The cold refrigerant which acts as a heat sink for warm return air is obtained by maintaining low pressure levels in the evaporator. The normal ranges of pressure at evaporator for a typical residential central air conditioner using R-22 as refrigerant is between 65 to 75 psig for 80^{o}F outdoor conditions. At these pressure levels the temperature of the refrigerant is around 42^{o}F. The normal return air temperature in residential comfort cooling is around 75-80^{o}F. Due to this temperature difference between air and refrigerant, heat is removed from return air and added to refrigerant. Air circulation across evaporator is maintained by employing a blower. The normal air flow across evaporator in residential air conditioning is ranges between 350 to 400 CFM per ton of cooling. The heat added to refrigerant causes it to boil rapidly and its state is changed to vapor as more heat is added. The heat carrying capacity of refrigerant will diminish if the vapor is not removed continuously to maintain the evaporator at low temperature levels. The circulation of refrigerant is done by a compressor which draw low pressure vapor from evaporator and compress it to hot and high pressure vapor which is passed to condenser. The condenser is a heat exchanging device which acts as medium to transfer heat from hot and high pressure refrigerant to ambient air which is circulated across it coils. The typical pressure ranges of a residential condenser will be 220 psig and it will vary depending on outdoor conditions. When the heat is removed from hot and high pressure vapor it condense as liquid. After leaving the condenser the refrigerant will be completely liquid at high pressure. This high pressure liquid refrigerant is metered through an expansion device to evaporator to maintain the refrigerant circulation. When refrigerant is passed through the metering device its pressure at the outlet of the expansion device is lowered because of pressure drop which occurs at the expansion valve. The high pressure liquid refrigerant is flashed into low pressure liquid vapor mixture which is admitted in to the evaporator. This cycle is continued as long as thermostat is calling for cooling or desired comfort levels are achieved at the conditioned space.

DEGRADED CONDITIONS

The above theory of operating principle for an air conditioner is rudiment and well understood. However the goal of this project is to monitor and quantify the operating environment of an air conditioner when some of the above said criteria's are failed to occur. The essential functions to carry out the comfort cooling can be listed as

1. Prescribed amount of air flow across evaporator which acts as a cooling agent for the space to be conditioned

2. Desired amount of air flow across condenser to carry the heat away from hot and high pressure liquid

3. The amount of refrigerant circulated to provide sufficient level of cooling source

4. The purity of refrigerant to allow various components in the circuit to carry out their duty successfully

From the above list of functions we can anticipate the following conditions which hinders the normal functioning of an air conditioner

1. The amount of air flow across evaporator is reduced by dirty filter or dirty coil or a failed blower

2. The amount of air flow across condenser is reduced by debris, foliage's and dirt

3. Restriction in refrigerant lines which prevents proper circulation

4. More refrigerant is present in the system than the normal amount (overcharge)

5. Less refrigerant is present in the system than the normal amount due to leakage etc. (undercharge)

6. Air and moisture are admitted into the system while charging.

The impact of above mentioned faults are described below.

1. Reduced evaporator air flow

The amount of air flow across evaporator is critical to carry out the conditioning of space successfully. The heat balance for evaporator is given as

$$m_a (h_{ra} - h_{sa}) = m_r (h_{eo} - h_{ei})$$

The components in an air conditioner are selected such that the amount of air flow and the amount of refrigerant flow are maintained within desired limits. Strictly speaking, only air flow is maintained at particular flow rate and the refrigerant flow varies with the indoor and outdoor conditions. When the air flow across evaporator is reduced the heat removed from supply air is increased, which results in lower temperature levels at the surface of evaporator. If the supply air flow is reduced considerably, the refrigerant temperature may reach to a very low temperature level such that frost may form at the surface of evaporator.

2. **Reduced condenser air flow**

The design air flow across condenser is constant for air conditioners. However this quantity is decreased due to leaves, debris and various other materials which reduce the free surface available for air flow thus reducing the amount of air flow. Another common incident which happens with the outdoor unit is, the top of the unit is covered with some heavy material which increase the power necessary to operate the fan. The heat balance across condenser is given as

$$m_{ao} X \; C_p (T_{ci} - T_{co}) = m_r (h_{ic} - h_{oc})$$

The above relation indicates that the air temperature rise across condenser is directly proportional to total heat rejected from refrigerant side. In turn, the total heat rejected from refrigerant side is basically a function of change in enthalpy and the amount of refrigerant circulated.

3. Restriction in refrigerant lines

The amount of refrigerant circulated to carry out the cooling duty will be affected when there is some restriction at the refrigerant flow passage. Common sources for restrictions are 1. Kinked lines or inadvertently crimped lines, 2. Dirt or some other material which blocks the narrow passage of expansion valve and 3. Bent heat exchanging coils. It is very rare to come across restriction in heat exchanger coils. However the common source for restrictions are restriction in liquid lines and at expansion valve. The suction line pipes are relatively sturdy (5/8 inch OD) compared to liquid and discharge lines. The length of discharge line is usually small and it is location is such that the occurrence of restriction in discharge line is not frequent.

Restriction in refrigerant lines produces unnecessary pressure drop which de-stabilizes the normal functioning of components and thus reducing the efficiency. Depending on the degree of restriction the pressure drop will vary and in extreme cases refrigerant flow is obstructed to such a high level that compressor is operating at excessive levels of superheat at compressor inlet.

4. Amount of charge

The amount of refrigerant circulated through the system is a function of indoor and outdoor conditions and the amount of charge initially admitted into the system. The components are balanced such that the variation in circulation due to change in indoor and outdoor conditions does not affect the efficiency significantly. However the change in circulation due to change in the amount of charge present in the system has serious impact on efficiency.

The amount of charge admitted into the system is basically decided on two factors:
1. To optimize the efficiency of the unit throughout the cooling season, 2. to ensure good operating environment for the compressor. The amount of charge present in the system is such that it should cool the compressor to mitigate the friction heat generated due to reciprocating parts. However if wet refrigerant enters the compressor it will wash away the lubrication oil thus reducing component life. Providing a particular amount of superheat for the vapor which is entering the compressor is one of the basic criteria to be satisfied while charging the system.

The normal amount of superheat provided during initial charging is around 8 to 15^{o}F. There will be a small change in superheat will result when indoor and outdoor conditions change [Figure 2]. We can observe from Figure 2, that for the same amount of charge the degree of superheat varies depend on load on the cooling coil and outdoor temperature. However, when the charge present in the system is higher than the recommend value the wet vapor enters the system. If the system is undercharged then excessive superheat will present at the inlet of compressor. Prolonged operations at these situations are detrimental to the system.

5. Presence of non-condensable gases in system

The heat transferring capability of evaporator and the condenser reduces drastically when non-condensable gases are present in the system. Ambient air is introduced into the system when proper precautions are not observed while charging the system. The air will stay at condenser and diminish the capability of condenser to reject heat. The moisture which presents with the air causes acid to form at the compressor which reduces the life of compressor.

NORMAL TEMPERATURE LEVELS

From the above arguments we can predict temperature patterns at some salient points under normal and degraded conditions.

1. Temperature drop across evaporator

The supply air temperature drop across evaporator is a function of 1. Amount of air flow, 2. Inlet relative humidity of the return air. If the air flow is reduced then the enthalpy drop across evaporator increase. Depending on the amount of moisture (RH) in the air, the sensible cooling load will vary (Table 1). Figure 3 shows that at normal operating conditions, when return air is at 80^{O}F, the temperature drop across evaporator will vary between 15^{O}F at 70% RH to 26^{O}F at 30 % RH.

2. Temperature rise across outdoor fan

At normal operating conditions the capacity of the evaporator is 36000 Btu/hr and the power consumed at compressor is 3600 watts. The total heat rejected at condenser is (36000 + 3600 X 3.412) 48283 Btu/hr. At 80^{O}F and 50% RH the specific volume of inlet air 13.75 ft^3/lb. and C_p of air 0.24. Assuming the normal amount of air flow across condenser coil is 2750 CFM then the temperature increase should be 16^{O}F for zero heat flow across compressor shell and in discharge line. However the normal temperature rise ranges between 12 to 14^{O}F depending on indoor and outdoor conditions.

3. Head temperature

The normal range of head temperatures for a R-22 system at $80^{O}F$ outdoor temperature is $150\text{-}170^{O}F$. The compression process is normally considered as an isentropic process. Increase in suction line superheat increase the temperature rise across compressor.

4. Suction temperature

As mentioned previously the suction temperature varies with the evaporator pressure and the amount of charge present in the system. If the system is initially charged for some minimum amount of superheat (8 - $15^{O}F$), as evaporator pressure is lowered the suction line temperature will increase or decrease depending on conditions exist at air and refrigerant side. If the supply air flow is reduced then cooling load on evaporator decrease which reduces superheat and suction line temperature both. However, if the system is circulating less charge than normal design conditions due to undercharge or restriction in refrigerant lines and the supply air flow is normal than high degree of superheat results in suction line which increase the suction line temperature.

5. Liquid line temperature

Normally liquid line temperatures are few degrees higher than ambient temperatures. When some restriction occurs at the refrigerant lines than liquid at the outlet of condenser stagnates and attains equilibrium with ambient dry bulb temperature. If there is not sufficient air flow across condenser then the liquid leaving condenser will be relatively at higher temperature than ambient temperature.

6. Degree of superheat

Superheat can be defined as the difference between refrigerant temperature and refrigeration saturation temperature corresponding to refrigerant pressure at the outlet of evaporator. We reviewed the necessity of maintaining certain amount of superheat at suction line. This minimum amount of superheat is set at one particular indoor and outdoor condition which will vary throughout the cooling season. The degree of superheat also varies depending on the cooling load on the coil and the amount of refrigerant circulated. If the amount of refrigerant circulated is within design limits than decrease in cooling load decrease the degree of superheat. When supply air flow across evaporator is normal the degree of superheat will vary depend the amount of refrigerant circulated. Overcharge conditions decrease the degree of superheat and undercharge conditions increase the degree of superheat.

7. Degree of subcooling

Subcooling can be defined as the difference between refrigerant temperature and saturated refrigerant temperature corresponding to refrigerant pressure at the outlet of condenser. Degree of subcooling increase or decrease depend on the air flow across condenser and the amount of refrigerant circulated. Thus degree of subcooling is somewhat similar to degree of superheat in predicting the behavior of system at high side.

8. Expansion valve outlet temperature

The normal temperature range for expansion valve outlet temperature will be between 45 to 55^{O}F. When there is any restriction occurs at the distributor lines to evaporator or restriction at liquid line after expansion valve then the temperature at the outlet of expansion valve increases.

PREDICTION OF DEGRADED CONDITIONS

There are several steady state simulation programs which are available to predict the design conditions of air conditioners and heat pumps [20, 21]. The ORNL and NIST models are popular and extensively used in industry. Farzad modified ORNL model using NIST subroutines to simulate heat exchangers in a better fashion through tube-by-tube simulation [22]. Katipamula modified TRPUMP [TRansient PUMP, 23] to simulate transient conditions to elucidate about dehumidification and cyclic degradation due to startup [24].

ORNL model requires to specify that a fixed amount of superheat at the outlet of evaporator or the amount of refrigerant in the system to begin the simulation [20]. From Farzad's findings [Figure 2] and current results show that degree of superheat varies depending on cooling load on the evaporator, outdoor conditions and the amount of charge in the system [22]. Fixed amount of superheat can be ensured only to air conditioners which employ thermostatic expansion valves (TXV). Assuming constant superheat for capillary tube and orifice-tube expansion valve will not reflect the true behavior under varying indoor and outdoor conditions. Simulation of degraded conditions can provide a better information on understanding degraded conditions thus temperature patterns to develop a better diagnostic procedure.

With current results it is found that at least three areas which need to be developed further theoretically and quantitatively are

1. Frost formation and heat transfer between refrigerant and air side at very low air flow and high humid conditions.

2. To optimize the degree of superheat set at charging conditions to ensure safe working environment throughout the cooling season.

3. To predict the temperature patterns under degraded conditions throughout the cooling season.

This report presents the preliminary results for an air conditioner under degraded conditions at one particular temperature (80 F). Knowledge from current results encourages that predicting performance under degraded conditions are possible. Benefits of such information on developing diagnostic procedures are discussed on the following chapters.

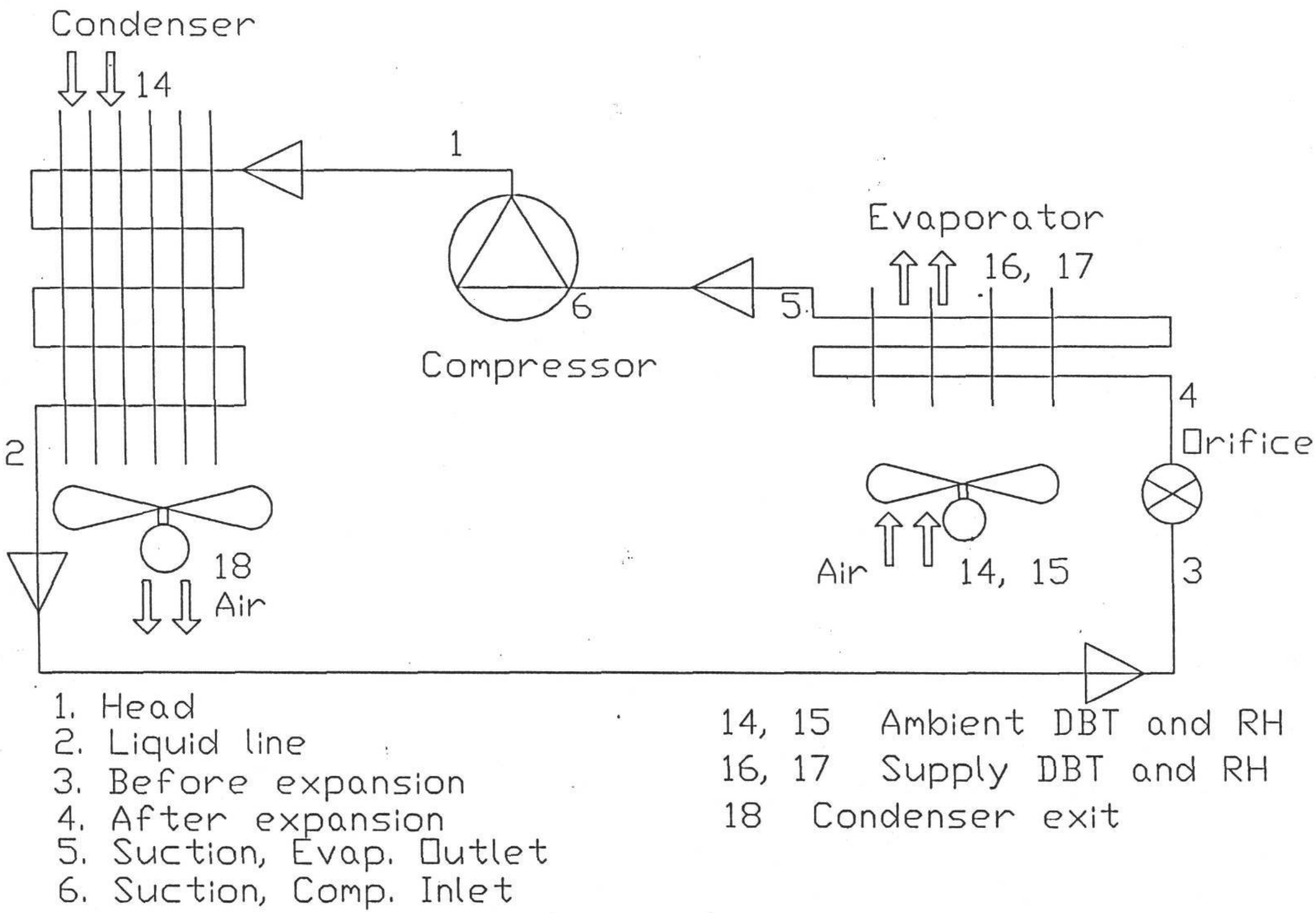
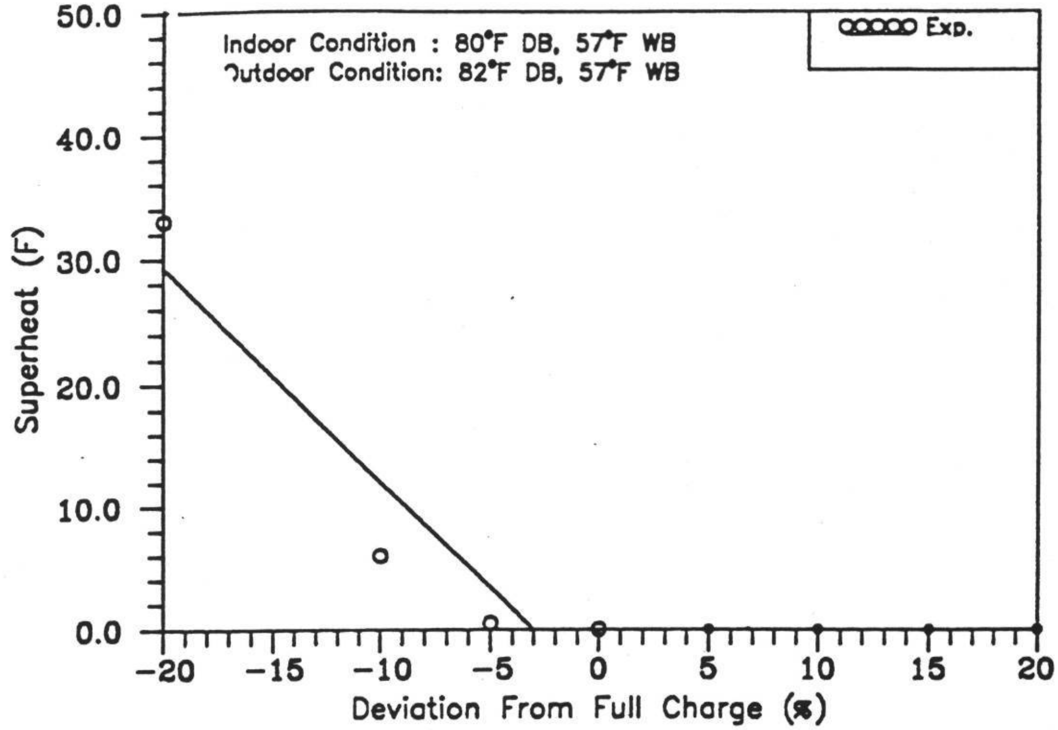
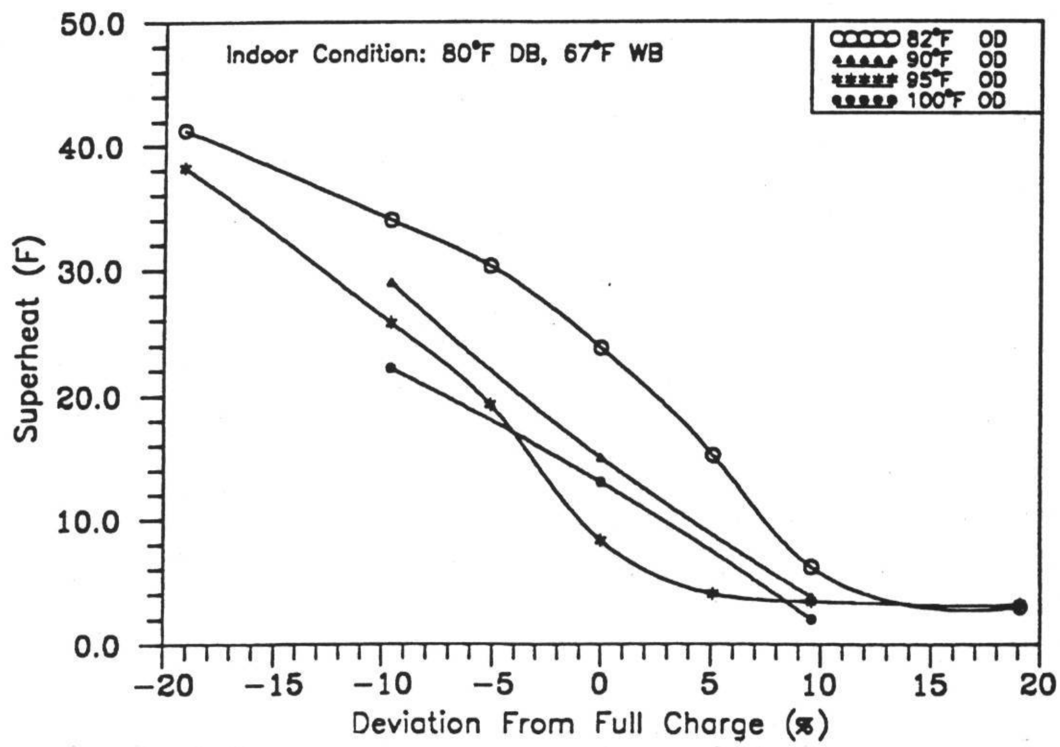


Figure 1. Cooling cycle schematic diagram





Firgure 2. The short-tube orifice system superheat for various outdoor temperatures and charging conditions (Source: Farzad M.; 1990. Ref. 22)

PERFORMANCE DATA COOLING

(CAPACITIES ARE NET IN BTUH/1000-INDOOR FAN HEAT DEDUCTED)

TTD736B WITH TXC736B4 AT 1200 CFM

O.D. D.B.	I.D. W.B.	TOTAL CAP.	SENS. CAP. AT ENTERING D.B. TEMP. 72	74	76	78	80	COMPR. KW	APP.DEW PT.
85	59	31.1	24.1	26.1	28.1	30.1	31.6*	2.98	44.8
	63	33.7	20.3	22.3	24.3	26.3	28.2	3.08	48.6
	67	36.4	16.1	18.1	20.0	22.0	24.0	3.18	52.7
	71	39.2	11.7	13.7	15.7	17.7	19.6	3.28	56.8
90	59	30.6	23.9	25.9	27.8	29.8	31.1*	3.12	45.1
	63	33.1	20.1	22.0	24.0	26.0	28.0	3.21	48.9
	67	35.7	15.8	17.8	19.8	21.7	23.7	3.31	53.0
	71	38.4	11.5	13.4	15.4	17.4	19.4	3.42	57.1
95	59	30.0	23.6	25.6	27.6	29.6	30.7*	3.25	45.4
	63	32.4	19.8	21.8	23.8	25.7	27.7	3.35	49.2
	67	35.0	15.5	17.5	19.5	21.5	23.5	3.45	53.3
	71	37.6	11.2	13.2	15.1	17.1	19.1	3.55	57.5
100	59	29.2	23.3	25.3	27.3	29.3*	30.1*	3.40	45.8
	63	31.6	19.5	21.5	23.4	25.4	27.4	3.49	49.6
	67	34.1	15.2	17.2	19.2	21.1	23.1	3.59	53.7
	71	36.6	10.8	12.8	14.8	16.8	18.8	3.69	57.9
105	59	28.4	23.0	25.0	27.0	28.7*	29.4*	3.55	46.1
	63	30.8	19.1	21.1	23.1	25.1	27.1	3.64	50.0
	67	33.2	14.9	16.8	18.8	20.8	22.8	3.73	54.1
	71	35.6	10.5	12.5	14.4	16.4	18.4	3.83	58.3
115	59	26.9	22.4	24.3	26.3	27.5*	28.2*	3.85	46.8
	63	29.1	18.5	20.5	22.4	24.4	26.4	3.93	50.7
	67	31.3	14.2	16.2	18.2	20.1	22.1	4.01	54.9
	71	33.6	9.8	11.8	13.8	15.7	17.7	4.10	59.1

CORRECTION FACTORS - OTHER AIRFLOWS
(multiply or add as indicated)

AIRFLOW	1050	1350
TOTAL CAP.	X0.98	X1.01
SENS. CAP.	X0.94	X1.05
COMPR. KW	X0.99	X1.01
A.D.P.	-1.4	+1.2

VALUES AT ARI RATING CONDITIONS

TOTAL NET CAPACITY = 35000 BTUH
AIRFLOW = 1200 CFM
APP. DEW PT. = 53.3 DEG. F
COMPRESSOR POWER = 3447 WATTS
I.D. FAN POWER = 440 WATTS
O.D. FAN POWER = 270 WATTS
S.E.E.R. = 9.35 BTUH/WATT

NOTE: RATED WITH 25 FEET OF 7/8 SUCT. AND 5/16 LIQUID LINES

* Dry coil condition (Tot. Cap. = Sens. Cap.)
Tot. Cap., Comp. KW and App. Dew Pt. valid only for wet coil. All temperatures in degrees F.

Table 1. Manufacturers' sample data

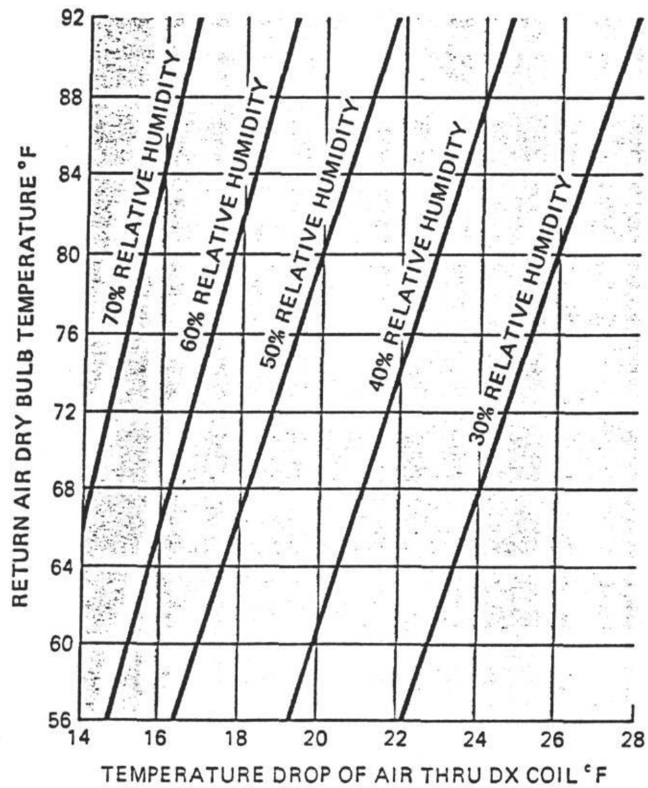


Figure 3 Supply air temperature drop for various load ratios (Source: Refrigeration and Air conditioning, ARI)

EXPERIMENTAL APPARATUS AND PROCEDURE

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

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

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

TEST BENCH

In this project, the air conditioning system considered for analysis is a standard split system air conditioner which contains a short-tube orifice expansion device. The cooling coil has a maximum three ton capacity. A schematic diagram and the outline of the experimental setup are shown in Figure 4. A detailed description of each component is given in Table A-1.

MEASUREMENT

A list of properties which were measured at the test bench is shown in Table A-2. A total of 20 quantities were measured with a data acquisition system. Refrigerant pressures and temperatures were measured at six locations, as shown in Figure 4. Properties measured on the air side included dry bulb temperature and relative humidity at outdoor conditions and supply conditions. Air temperature at the discharge of the condensing fan was also measured. Copper-constantan thermocouples were used to measure refrigerant and air temperatures. The refrigerant mass flow rate was measured at the liquid line before the expansion valve. A differential pressure transducer was used to measure the static pressure gain before the supply nozzle, which was converted into an air flow rate. The total amount of power to run the compressor, blower and condenser fan was measured with a watt transducer.

TEST OPERATING PROCEDURE

The DOE/ARI specifications indicate that steady-state and cyclic tests for Unitary Air Conditioners should be conducted in controlled environment for 30 minutes [25]. We found that the test bench reached steady-state within 4 minutes after switching on the unit. The coefficient of variation for the standard test was less than 1% which was satisfactory for use in these tests. Another reason for using reduced test duration was to minimize damage to components during certain degraded tests.

DEGRADATION SIMULATION

1. Reduced evaporator air flow

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

2. Blocked outdoor fan air flow

A wooden frame was built over outdoor fan to shut off the air flow completely during degraded test. It was observed that when the top of the outdoor fan was completely blocked air enters at the bottom portion of the condenser and leaves at the top portion of the coil.

3. Presence of non-condensable gases

Initially the low pressure section of the system was isolated by closing the valves. Pressure at evaporator was reduced less than atmospheric pressure through a vacuum pump. Air was introduced in to the system by opening the charging valves attached to low pressure section. The total amount of air introduced in to the system was equivalent to the evaporator volume at normal ambient conditions (80 F).

4. Restricting refrigerant lines

Refrigerant line restrictions were simulated by closing the valves partially at particular test section. Valves are closed partially until the pressure drop levels of 50 psig were achieved.

5. Undercharge and Overcharge

The normal amount of refrigerant charge for an orifice-tube expansion device to provide three tons of cooling was found by Farzad, as 136 ounces [22]. Initially the system was charged with 136 ounces. For overcharging tests, refrigerant was admitted in to the system at the increments of 13.5 ounces up to 68 ounces. This amount corresponds to 50% overcharge. After conducting the 50% overcharge degraded test the refrigerant was removed by 64 ounces. A standard test was run before removing the charge at 13.5 ounce increments down to 64 ounces.

DATA REDUCTION AND PERFORMANCE CALCULATIONS

For each test 5 second time series data were taken for refrigerant pressure, refrigerant temperature, air temperature and relative humidity, as shown in Figures B-1.1.1 through B-6.5.3. To calculate the performance factors, values measured during steady-state conditions were averaged for the last ten minutes of the test. Refrigerant enthalpies at six locations were calculated using the refrigerant property calculation program developed by Kartsounes[26]. Air-side enthalpy, humidity and specific volume were calculated by a psychrometric program developed at the Energy Systems Laboratory, Texas A&M University[27].

Pressure-Enthalpy and Psychrometric chart are made by imposing data points for standard and degraded test on templates for these charts. To facilitate review of charts and data the key values are enclosed with each chart. A complete list of time series, Pressure-Enthalpy and Psychrometric charts are presented in the appendix.

Figure 4.2, an example P-H chart and Psychrometric chart for reduction in evaporator air flow conveys the effect of degradation in a nut shell. Pressure-Enthalpy chart shows the reduction in pressure thus temperature at evaporator. There is no significant drop in refrigerant effect per pound of refrigerant. However the refrigerant mass flow rate decrease with decrease in air flow. Very low head temperature and increased pressure ratio for 90% reduction in evaporator air flow can be seen from the diagram. Air side temperature drop increase with reduction in air flow. Decrease in SHR can be seen through the change in slope of the air side cooling curve for degraded tests.

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

Table A-1. List of components

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

Table A-2. List of measuring instruments

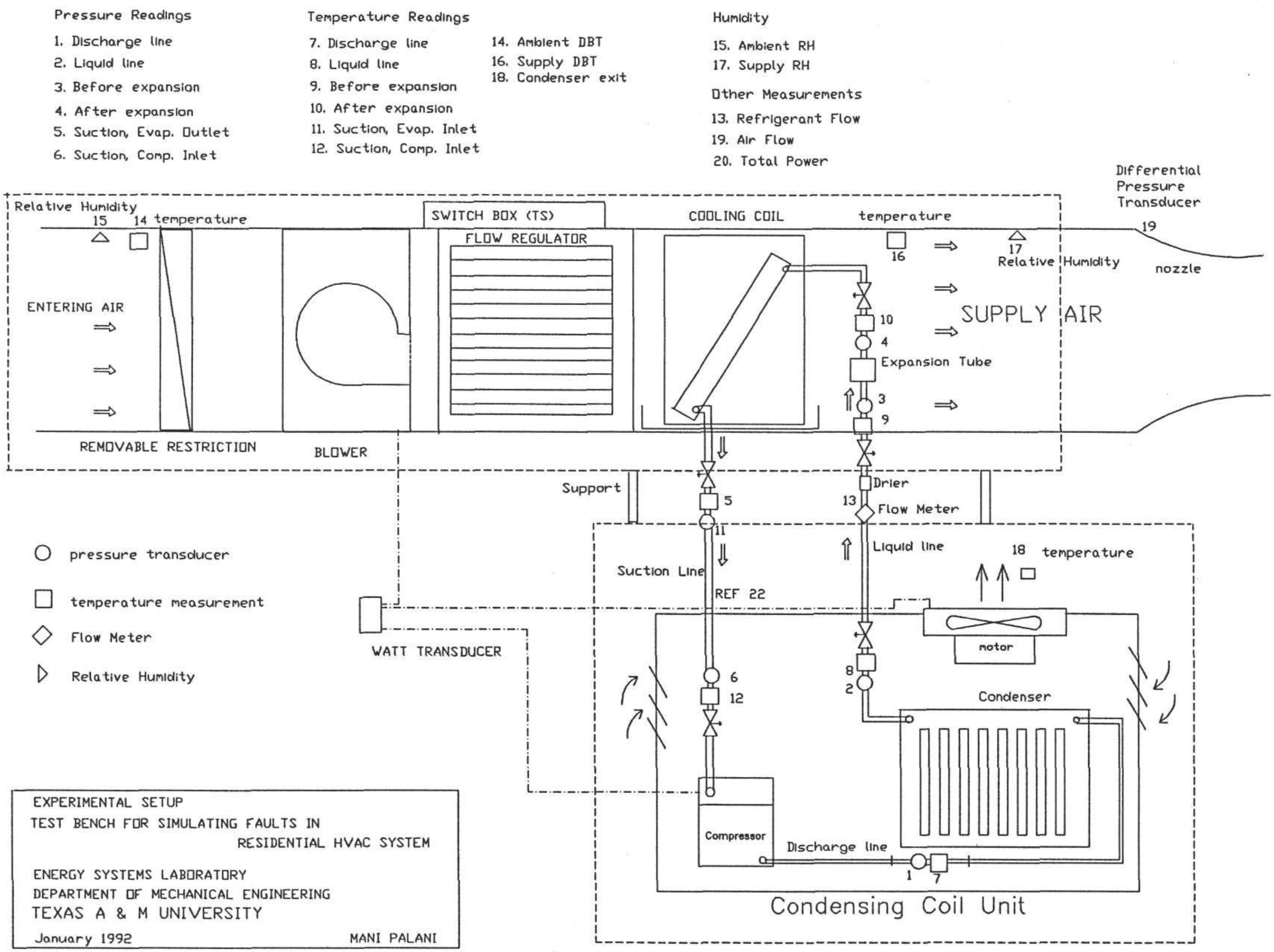


Figure 4. Schematic drawing of Experimental setup

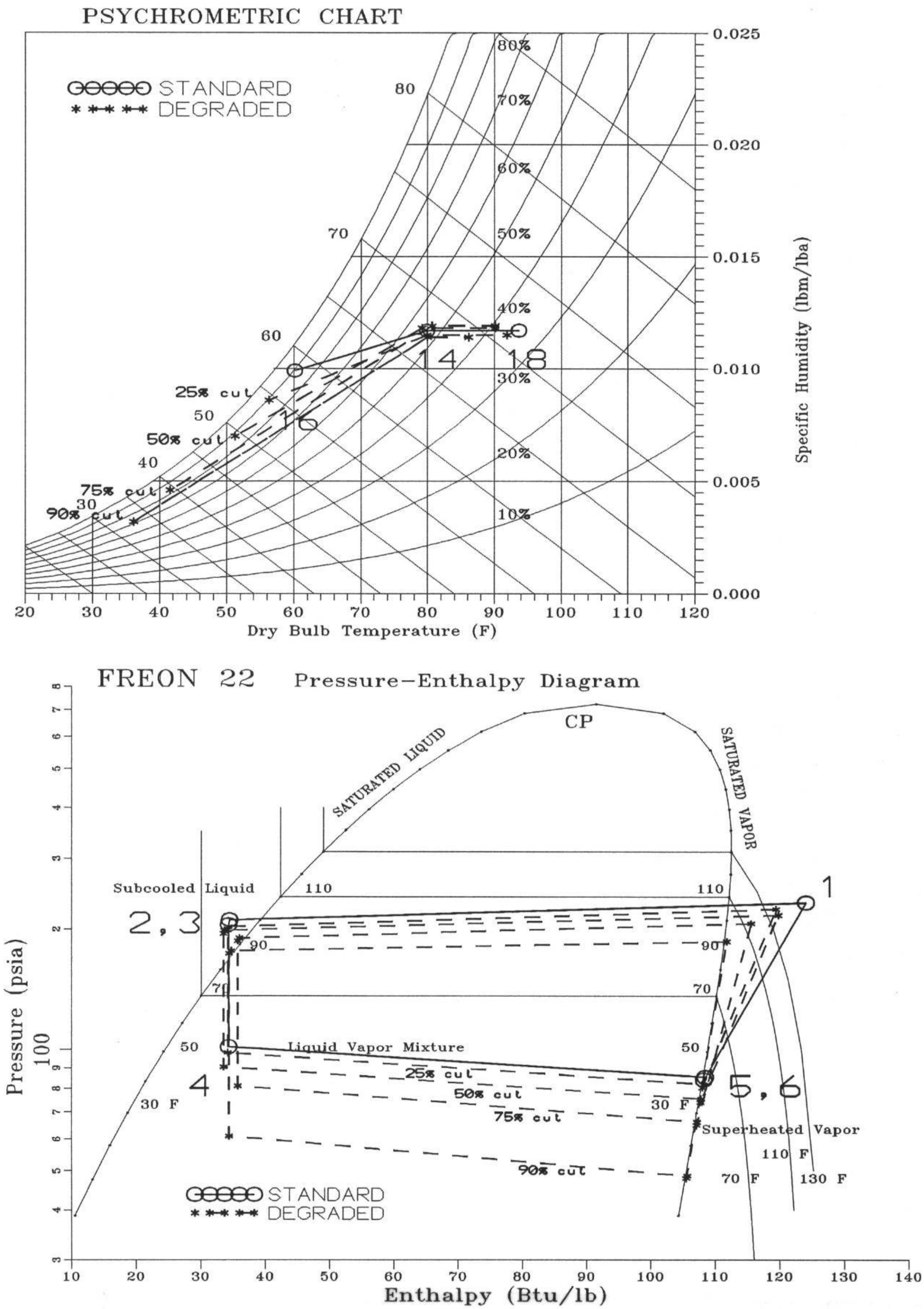


Figure 4.2 Pressure-Enthalpy chart and Psychrometric chart

RESULTS AND DISCUSSION

Results are presented in this report in two major groups. 1. Summary of performance factors (Power, Cooling capacity, and EER), 2. Temperature patterns for diagnostic analysis. Tables 2 describes summary of results in general. Effect of degraded tests on performance factors are described in Table 3. Figures 5.1 to 5.9 show the degraded performance factors. Temperature patterns for diagnostics are shown in Figures 6.1 to 6.9. Results for standard tests and degraded tests are included in the appendix (Tables B-1 and B-2).

POWER, COOLING CAPACITY AND EER

Reduced evaporator air flow

The total power consumption for running the compressor, blower and outdoor fan during the standard run was 3607 watts. Results show that as evaporator air flow was reduced from a normal amount the electricity demand, cooling capacity and EER decrease. Power consumption decrease in a near linear fashion, from 3.01% at 25% reduction in evaporator air flow to 21.2% at 90% reduction in evaporator air flow. This may imply that as utilities fix degraded air conditioners the demand may go up by 3-21% while usage goes down.

Cooling capacity decreases linearly until about 50% evaporator air flow then dropped suddenly. Measurements showed that cooling capacity reduced by 5.6% at 25% reduction in evaporator air flow to 11.5% for 50% reduction in air flow. Cooling capacity decreased by 33% for 75% reduction in evaporator air flow. For 90% reduction in evaporator air flow, the degradation of performance was the highest, as expected; cooling capacity was decreased by 62%. The sensible heat ratio (SHR - ratio of the sensible capacity to total capacity) for the standard run at 80° F DBT and 53.5% RH was 0.73. SHR decreases to 0.66, 0.58, 0.56 and 0.56 for 25%, 50%, 75% and 90% reductions in evaporator air flow respectively.

Reductions in the energy efficiency ratio(EER) are similar to reductions in cooling capacity(Figure 5.1). Performance was linear until about 50% reduction in evaporator air flow, and non-linear afterwards. EER decrease by 1.9%, 4.5%, 23.5% and 52% for 25%, 50%, 75% and 90% reductions in evaporator air flow respectively[Table 3]. Figure 5.1 shows the trends of cooling capacity, power and EER for reduced evaporator air flow.

Blocked outdoor fan air flow

The total demand during standard test was 3458 watts. When outdoor fan air flow was blocked completely, the demand rose to 4010 watts. Cooling capacity during degraded test decreased by 7.25%. Increase in demand and reduction in cooling capacity resulted in 20.1% reductions in EER for the degraded test.

Presence of non-condensable gases

Increase in demand during the presence of air was very negligible (0.68%). Cooling capacity increased by 3.3% and EER increased by 2.54%.

Restriction in Refrigerant lines

Reduced mass flow rate caused by restriction in liquid line which is located before expansion valve reduced the power consumption by 20.4%. Cooling capacity decreased by 65% and EER decreased by 56.1%. There was no appreciable change in performance observed during restriction in refrigerant line located after expansion valve and restriction in discharge line. Restriction in suction line decreased the demand by 4.67% and cooling capacity decreased by 30.4%. Reduction in EER was 27.25%.

Undercharge

There was no significant change in demand and EER was observed up to 20% undercharging. These results are similar in trend with Farzad's observations [22]. There was a point of changeover observed between 20% and 30% undercharge. Demand decreased by 8.4%, 12.82% and 17.18% for 30%, 40% and 50% undercharge. Cooling capacity reduced by 40% at 30% undercharge to 59% at 50% undercharge. Drop in EER was noticeable at 20% to 30% undercharge and EER reduces by 34% at 30% undercharge to 50% at 50%

undercharge. SHR for the standard test was 0.77. For 30% undercharge the SHR increased to 0.92 and for 40 % and 50% undercharge only sensible heat is removed from return air.

Overcharge

Up to 20% overcharge there was no significant change in EER. For 30% overcharge degradation was maximum of about 8.1%. For 40% overcharge reductions in EER was 5.3% and for 50% overcharge it was 4.4%. There was no appreciable change in demand observed during overcharge. The maximum increase was 4.7% at 30% overcharge. There was no appreciable change in SHR occurred.

TEMPERATURE ANALYSIS

Reduced evaporator air flow

Reducing the evaporator air supply increased the temperature drop across the cooling coil from 21° F at normal flow to 45° F at 25% of normal flow [Figure 6.1.]. However, air temperature rise across the condenser decreased from 14^0 F at normal flow to 8^0 F at 30 % of evaporator air flow.

Head temperature (refrigerant temperature at discharge line) drops significantly when evaporator air flow was reduced. The drop in suction pressure was considerable (30%) for a 90% reduced air flow rate [Figure B-1.4.2]. The corresponding drop in suction temperature was from 40° F at standard test conditions to 11° F at 90% reduced evaporator air flow rate. This near freezing temperature level may be one of the primary reasons for a considerable reduction in cooling capacity at reduced evaporator air flow. There was no appreciable change in degree of superheat (0 to 1° F) during degraded tests. However, the degree of subcooling dropped from 17° F at standard conditions to 2° F at reduced evaporator air flow.

Blocked outdoor fan air flow

Under blocked condenser air flow conditions the temperature rise across condenser increased substantially. The normal level of temperature rise was 16 F and it increased to 30F during degraded test. Increase in liquid line temperature was around 27F. There was no significant change in degree of superheat and subcooling.

Presence of non-condensable gases

Introducing air into the system does not show any evidence of its presence while the unit was operating. The only noticeable trend was subcooling increased from 16 F to 23 F. Rise in head temperature was moderate at 80 F outdoor conditions.

Restriction before expansion valve

Reduced refrigerant circulation caused reduced temperature drop across cooling coil and reduced temperature rise across condenser coil. Head temperature rose substantially and the drop in suction temperature was also significant. Degree of superheat increased considerably during degraded conditions. The liquid which was stagnant in the liquid line between expansion valve and condenser coil was near ambient temperature during degraded test.

Restriction after expansion valve

Refrigerant temperature at the outlet of expansion valve increased to 80 F from 51 F during normal test. Drop in suction temperature and increase in degree of superheat are moderate.

Restriction in Suction line

Head temperature and the degree of subcooling both dropped significantly. Drop in suction line temperature from evaporator outlet to compressor inlet was 19 F.

Restriction in Discharge line

A moderate rise in head temperature was observed during the degraded test.

Undercharge

It was observed that a significant change in temperature pattern occurred between 20% and 30% undercharge. Supply air temperature drop across evaporator reduced from 22 F to 14 F at undercharge of 30% and above. There was considerable increase head temperature and similar drop in suction temperature occurred during undercharge conditions. The degree of superheat increased markedly. Degree of subcooling reduced to zero for undercharge of 30% and above.

Overcharge

Drop in head temperature and rise in degree of subcooling observed during overcharge conditions.

REVIEW OF CURRENT RESULTS WITH AVAILABLE LITERATURE

Threlkeld discuss performance of single stage cycle at low evaporating temperatures [28]. He list down the three principal disadvantageous of single-stage operation at low evaporating temperatures. These are 1. Low refrigerating efficiency, 2. Low compressor volumetric efficiency(reciprocating compressors), and 3. High compressor discharge temperature. Figure [6.10] shows the variation of compressor discharge temperature for theoretical single-stage cycle. From Table 2 and Figure 6-1 we can observe that experimentl results show that head temperature decrease with reduction in evaporator temperature. This may be due to an assumption that condensing pressure (temperature) remains constant as evaporator temperature was lowered.

Experimental results show that the pressure drop across evaporator (including evaporator distributor) is around 15-17 psig. Katipamula's findings also support this trend [Figure 6-12]. The temperature drop corresponding to this pressure drop is nearly 10 F. Measuring degree of superheat by measuring pressure and temperature will not be affected by this phenomenon. However one product literature [29] indicates that degree of superheat can be computed by measuring the difference between temperature at the outlet of expansion valve and the temperature at the outlet of evaporator. Unless the temperature drop due to pressure drop is accounted with the measured temperature difference, it will give erroneous values for degree of superheat.

SIMULATED FAULTS	STANDARD AND DEGRADED TEST RESULTS																			
	AMBIENT DBT(F)/RH(%)		EVAPORATOR SUPPLY AIR TEMP (F)		EVAPORATOR TEMP DROP DEG F		CONDENSER TEMP RISE DEG F		DISCHARGE PRESSURE PSIA		SUCTION PRESSURE PSIA		DISCHARGE TEMP DEG F		SUPPLY AIR FLOW CFM		POWER WATTS		EER	
	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR	STD	DGR
REDUCTION IN EVAPORATOR AIR FLOW																				
1. 25% Reduction	79.9/53.8	79.9/52.8	60	56	20	24	14	12	234	225	84	81	161	137	1185	848	3588	3480	10.3	10.06
2. 50% Reduction	79.9/53.8	79.2/55.2	60	51	20	28	14	11	234	217	84	74	161	137	1185	589	3588	3340	10.3	9.84
3. 75% Reduction	79.7/53.5	80.7/53.2	60	42	20	39	15	9	238	207	83	65	176	115	1163	290	3626	3191	9.8	7.5
4. 90% Reduction	79.7/53.5	80/52.3	60	37	20	44	15	6	238	186	83	48	176	92	1163	142	3626	2857	9.8	4.7
BLOCK OD FAN AIR FLOW	80.1/47.7	81.2/47.2	58	60	22	21	13	30	230	291	82	89	170	170	1135	1160	3458	4010	10.1	8.07
PRESENCE OF NON-CONDENSABLE GASES	80/48.3	9.82/49.3	58	58	22	22	13	12	234	244	85	84	139	170	1138	1125	3542	3566	9.85	10.1
RESTRICTING REFRIGERANT LINES																				
1. Before Expansion Valve	82.3/43.7	82/44.1	56	69	26	13	13	7	234	201	82	47	157	227	922	975	3570	2842	9.98	4.38
2. After Expansion Valve	79.8/51.4	81.8/48.3	59	61	21	21	14	14	233	237	83	82	165	186	1142	1127	3503	3452	10.3	9.83
3. Suction Line	80.9/47.8	80.5/49	59	62	22	18	13	10	233	212	83	72	172	128	1130	1170	3487	3324	10.2	7.42
4. Discharge Line	80.3/48.8	80.3/48.8	59	59	21	21	13	13	231	286	82	85	174	190	1132	1133	3483	3800	9.8	8.88
REFRIGERANT UNDERCHARGING																				
1. 10 % Undercharge	80.1/48	80.5/48.2	59	59	21	22	13	14	230	231	84	83	163	166	1137	1136	3486	3490	10.04	10.2
2. 20 % Undercharge	80.1/48	79.6/47.3	59	58	21	21	13	12	230	218	84	80	163	176	1137	1129	3486	3337	10.04	9.97
3. 30 % Undercharge	80.1/48	80.3/46.9	59	65	21	15	13	11	230	210	84	73	163	193	1137	1150	3486	3192	10.04	6.62
4. 40 % Undercharge	80.1/48	1.14/46.4	59	68	21	13	13	9	230	201	84	65	163	204	1137	1159	3486	3039	10.04	5.27
5. 50 % Undercharge	80.1/48	80.3/48.1	59	67	21	13	13	8	230	192	84	57	163	213	1137	1163	3486	2887	10.04	5.03
REFRIGERANT OVERCHARGING																				
1. 10 % Overcharge	80.1/48	80.9/50.1	59	60	21	21	13	14	230	240	84	86	163	144	1137	1133	3486	3586	10.04	9.98
2. 20 % Overcharge	80.1/48	81.1/50.4	59	60	21	21	13	13	230	243	84	87	163	136	1137	1139	3486	3616	10.04	10.1
3. 30 % Overcharge	80.1/48	80/50.9	59	60	21	20	13	14	230	249	84	88	163	132	1137	1156	3486	3650	10.04	9.23
4. 40 % Overcharge	80.1/48	80.6/49.4	59	59	21	21	13	13	230	252	84	88	163	128	1137	1143	3486	3637	10.04	9.51
5. 50 % Overcharge	80.1/48	80.1/49.6	59	59	21	21	13	13	230	250	84	88	163	123	1137	1147	3486	3589	10.04	9.6

STD - Standard Test, DGR - Degraded Test

Table 2. Summary of Results

SIMULATED FAULTS	PERCENT OF VARIATION		
	CAPACITY KBtu/hr	POWER WATTS	EER
REDUCTION IN EVAPORATOR AIR FLOW			
1. 25% Reduction	-5.61	-3.01	-1.94
2. 50% Reduction	-11.46	-6.91	-4.47
3. 75% Reduction	-32.63	-12.00	-23.47
4. 90% Reduction	-62.33	-21.21	-52.04
BLOCK OD FAN AIR FLOW	-7.25	15.96	-20.10
PRESENCE OF NON-CONDENSABLE GASES	3.27	0.68	2.54
RESTRICTING REFRIGERANT LINES			
1. Before Expansion Valve	-65.07	-20.39	-56.11
2. After Expansion Valve	-5.90	-1.46	-4.56
3. Suction Line	-30.40	-4.67	-27.25
4. Discharge Line	-1.17	9.10	-9.39
REFRIGERANT UNDERCHARGING			
1. 10 % Undercharge	1.49	0.11	1.39
2. 20 % Undercharge	-4.97	-4.27	-0.70
3. 30 % Undercharge	-39.62	-8.43	-34.06
4. 40 % Undercharge	-54.24	-12.82	-47.51
5. 50 % Undercharge	-58.53	-17.18	-49.90
REFRIGERANT OVERCHARGING			
1. 10 % Overcharge	2.20	2.87	-0.60
2. 20 % Overcharge	4.31	3.73	0.60
3. 30 % Overcharge	-3.74	4.70	-8.07
4. 40 % Overcharge	-1.17	4.33	-5.28
5. 50 % Overcharge	-1.63	2.95	-4.38

Table 3. Summary of reduction in performance for degraded tests

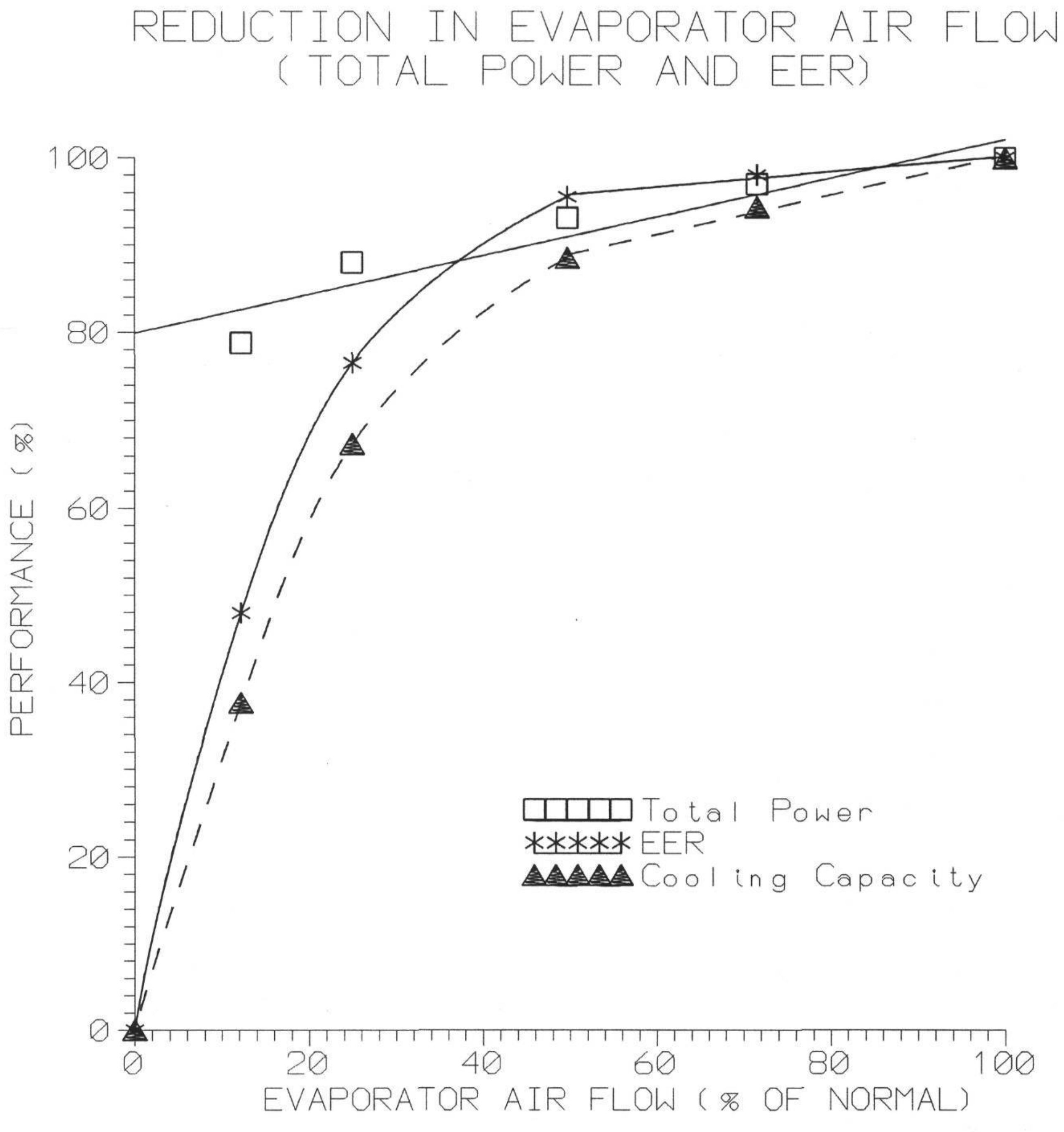


Figure 5.1. Reduced evaporator air flow Power, EER and Cooling capacity curves

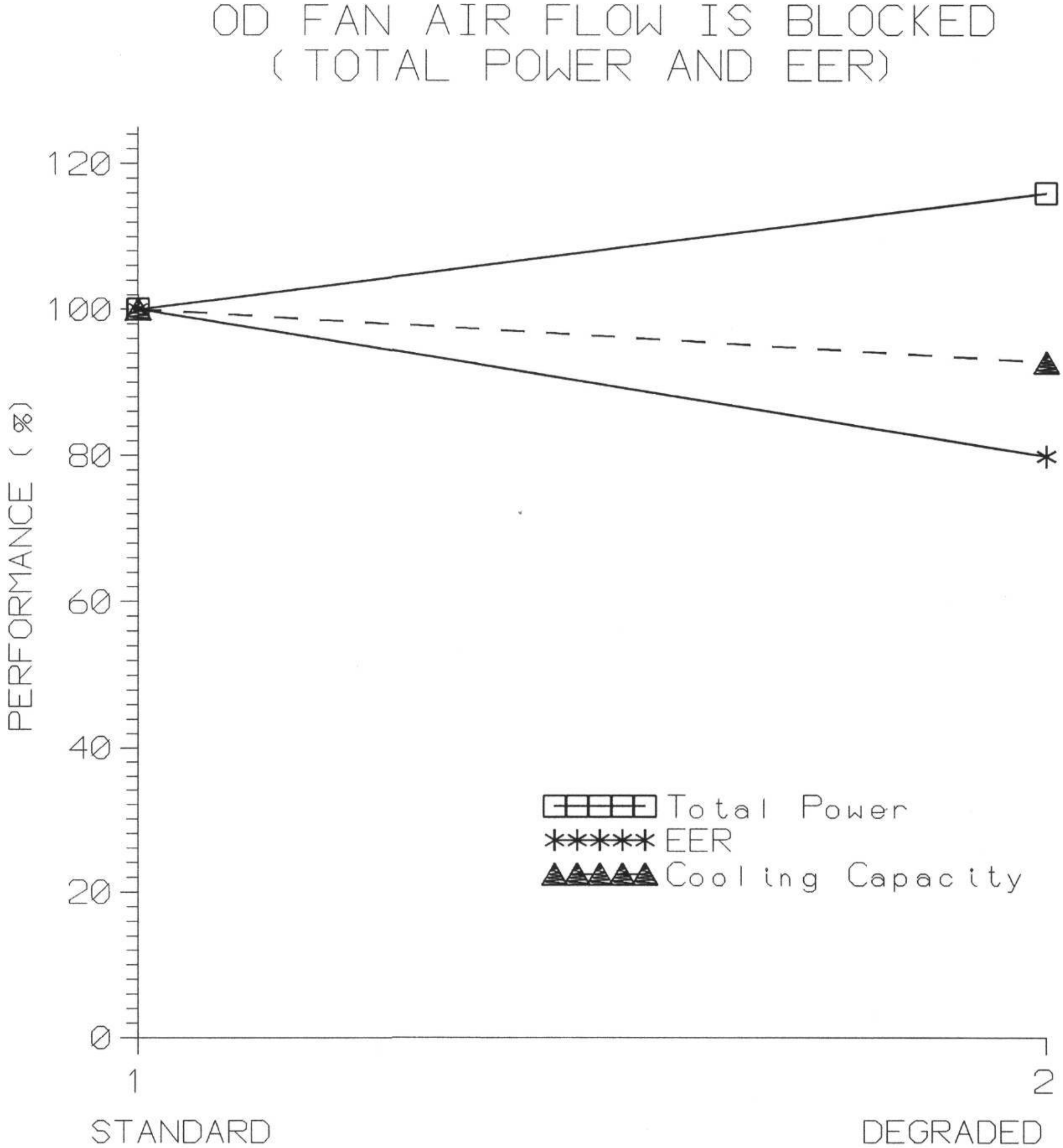


Figure 5.2. Blocked outdoor fan air flow Power, EER and Cooling capacity curves

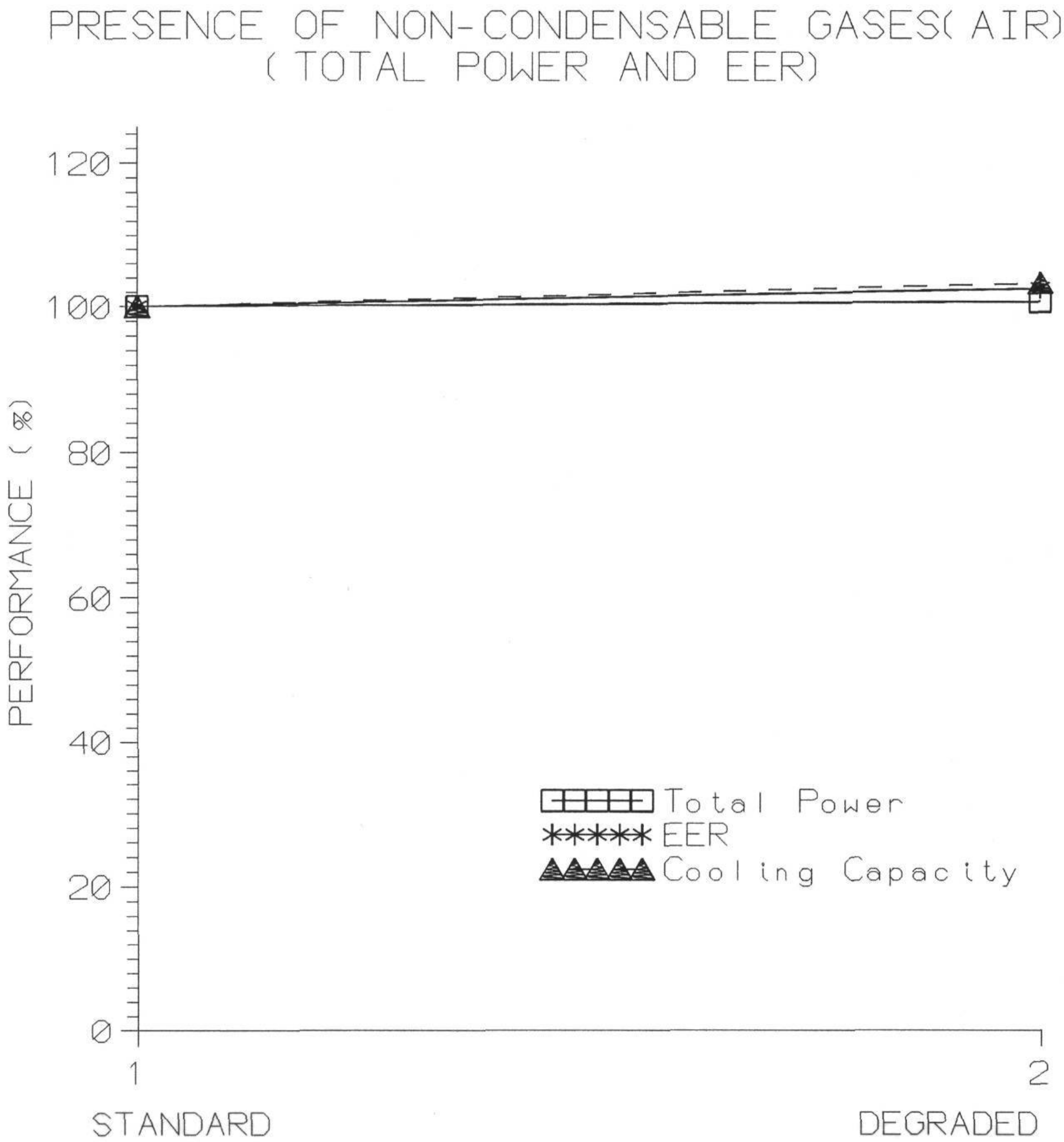


Figure 5.3. Presence of non-condensable gases Power, EER and Cooling capacity curves

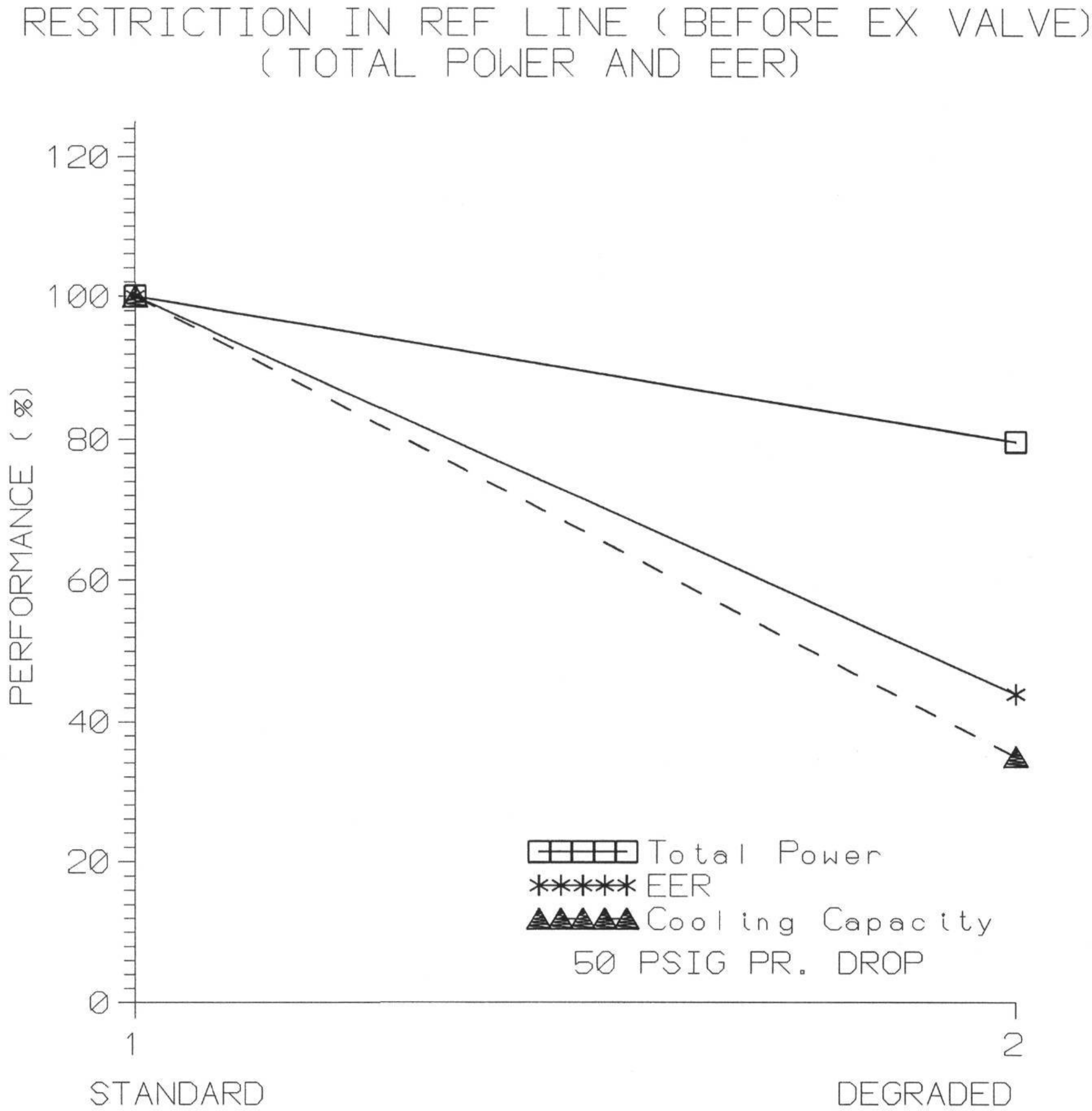


Figure 5.4. Restriction in refrigerant line before expansion valve Power, EER and Cooling capacity curves

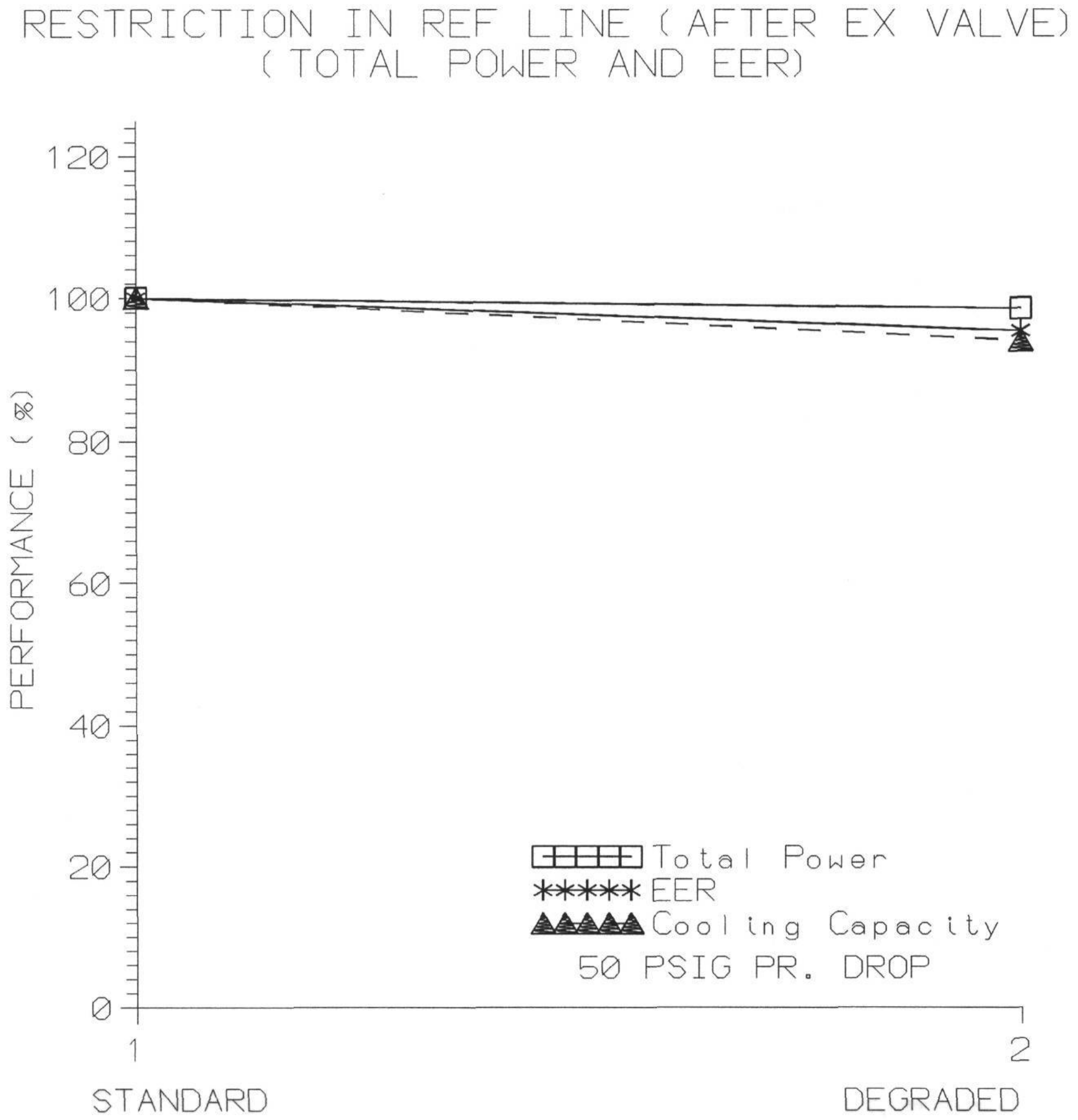


Figure 5.5. Restriction in refrigerant line after expansion valve Power, EER and Cooling capacity curves

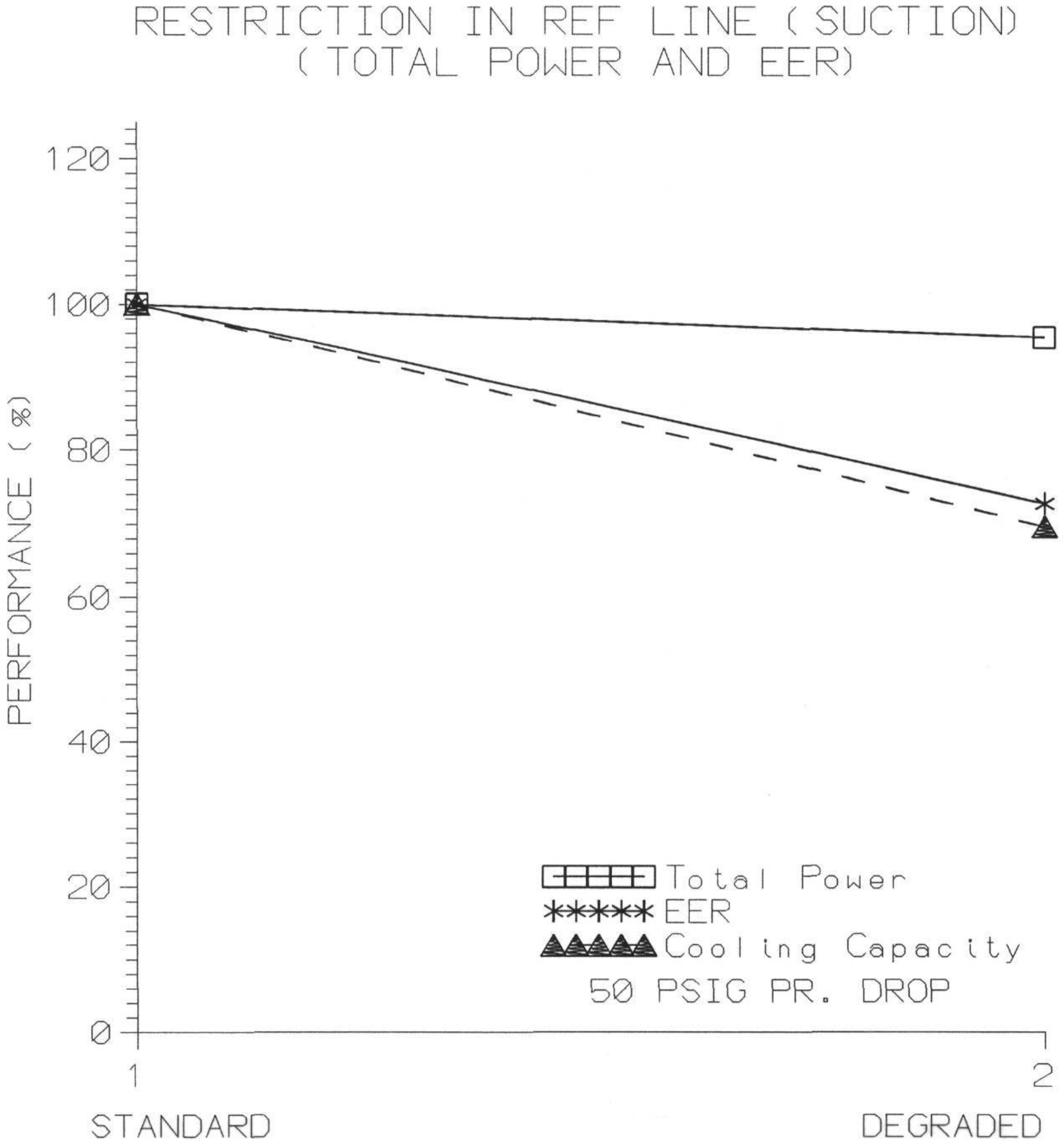


Figure 5.6. Restriction in suction line Power, EER and Cooling capacity curves

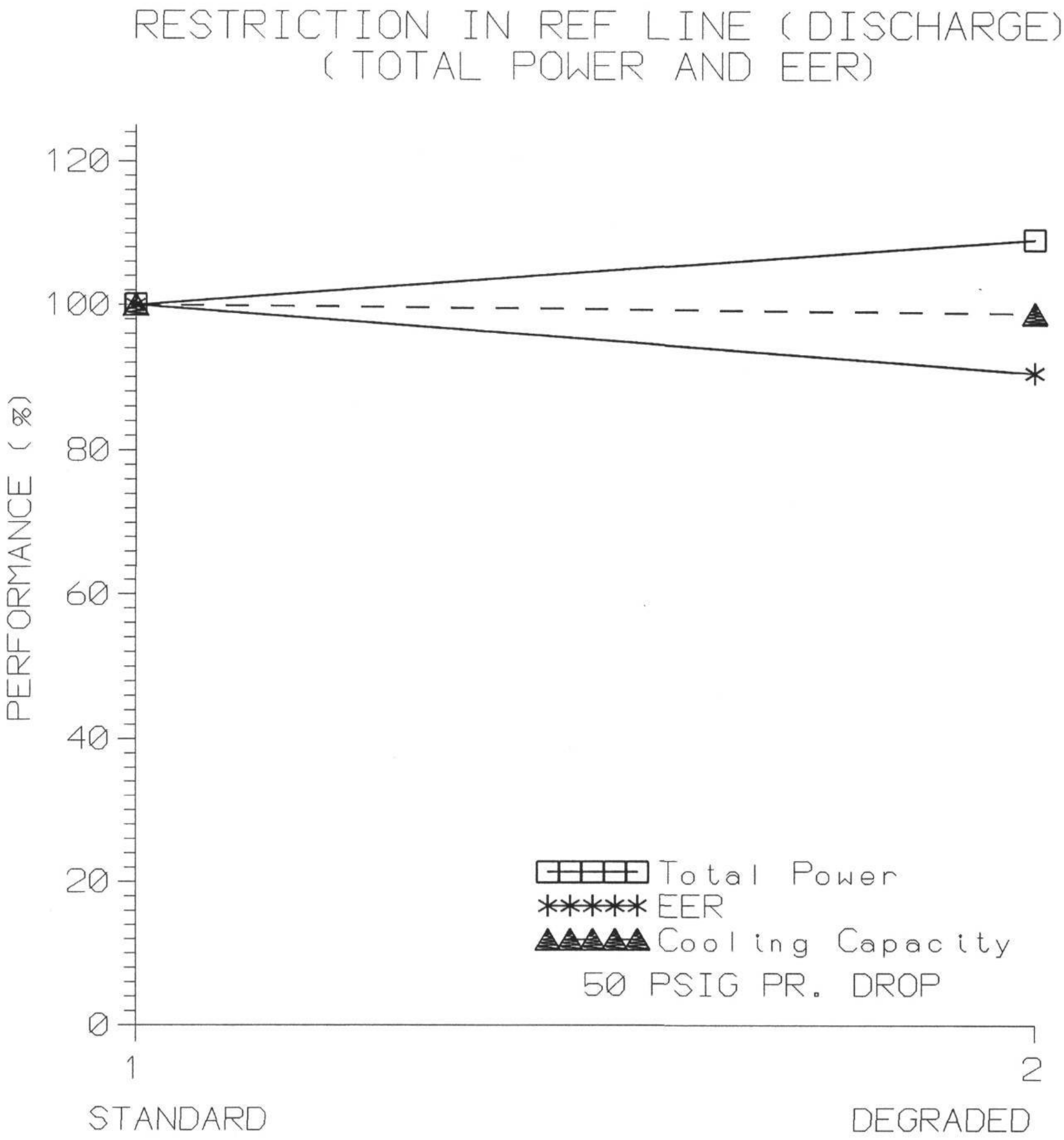


Figure 5.7. Restriction in discharge line Power, EER and Cooling capacity curves

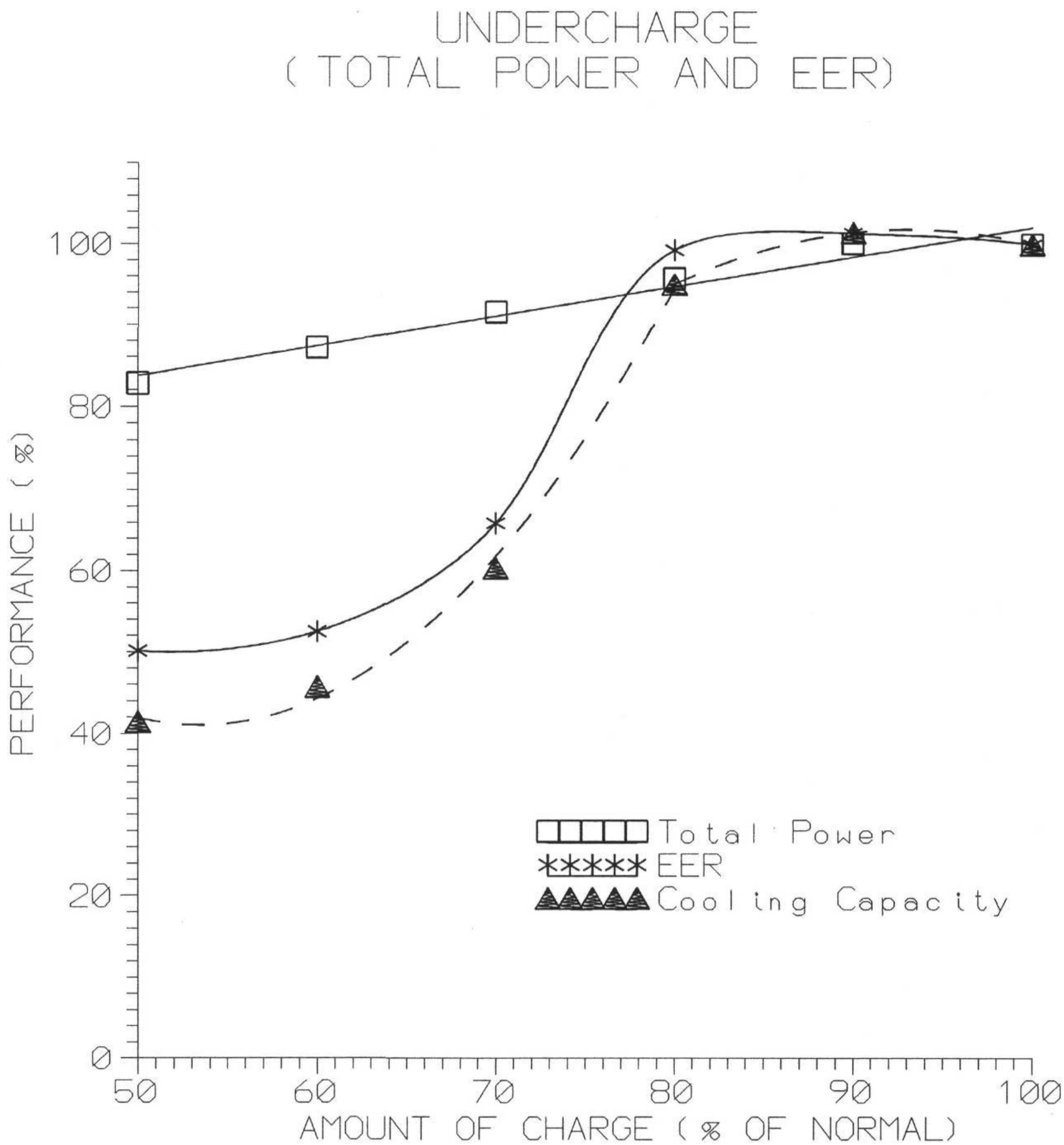


Figure 5.8. Undercharge Power, EER and Cooling capacity curves

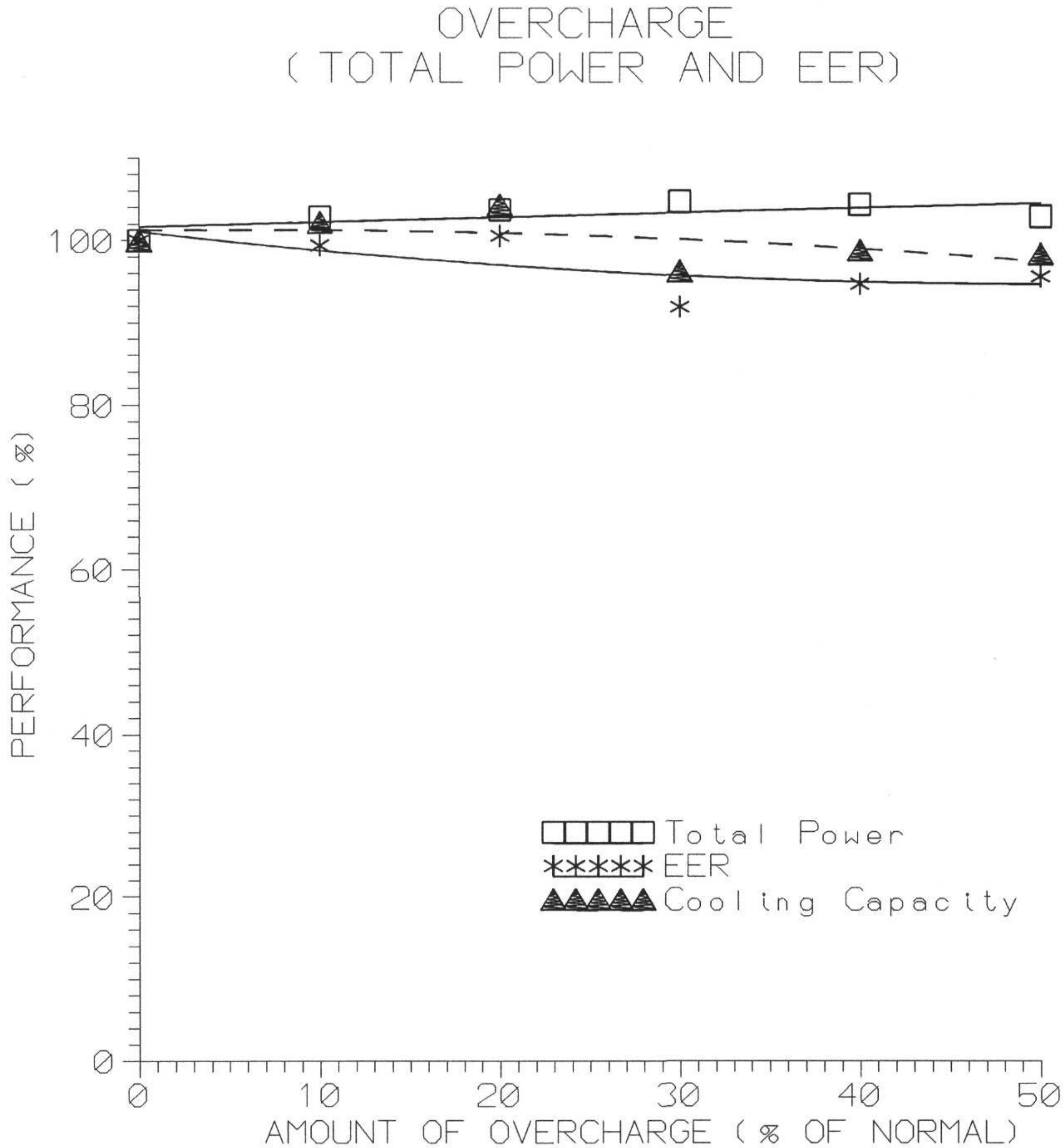


Figure 5.9. Overcharge Power, EER and Cooling capacity curves

SYMPTOM	FAULT	Reduced Evaporator Air Flow	Reduced OD Fan Air Flow	Air in the System	Restriction Before Ex. Valve	Restriction After Ex. Valve	Restriction in Suction Line	Restriction in Discharge Line	Undercharge	Overcharge
Evap. Temp. Drop	High	X								
	Normal		X	X		X	X	X		X
	Low				X				X	
Cond. Temp. Rise	High		X							
	Normal			X		X		X		X
	Low	X			X		X		X	
(Head T - Ambient DBT)	High			X	X			X	X	
	Normal		X			X				
	Low	X					X			X
(Return DBT - Suction T)	High	X					X			
	Normal		X	X				X		X
	Low				X	X			X	
(Liquid T - Ambient DBT)	High		X						X	
	Normal	X		X		X	X	X		
	Low				X					X
Superheat	High				X	X			X	
	Normal	X	X	X			X	X		X
	Low									
Subcool	High			X						X
	Normal	X	X		X	X	X	X		
	Low								X	
(Supply DBT - Ref. T)	High				X				X	
	Normal	X	X	X			X	X		X
	Low					X				

Table 4. Diagnostic decision chart

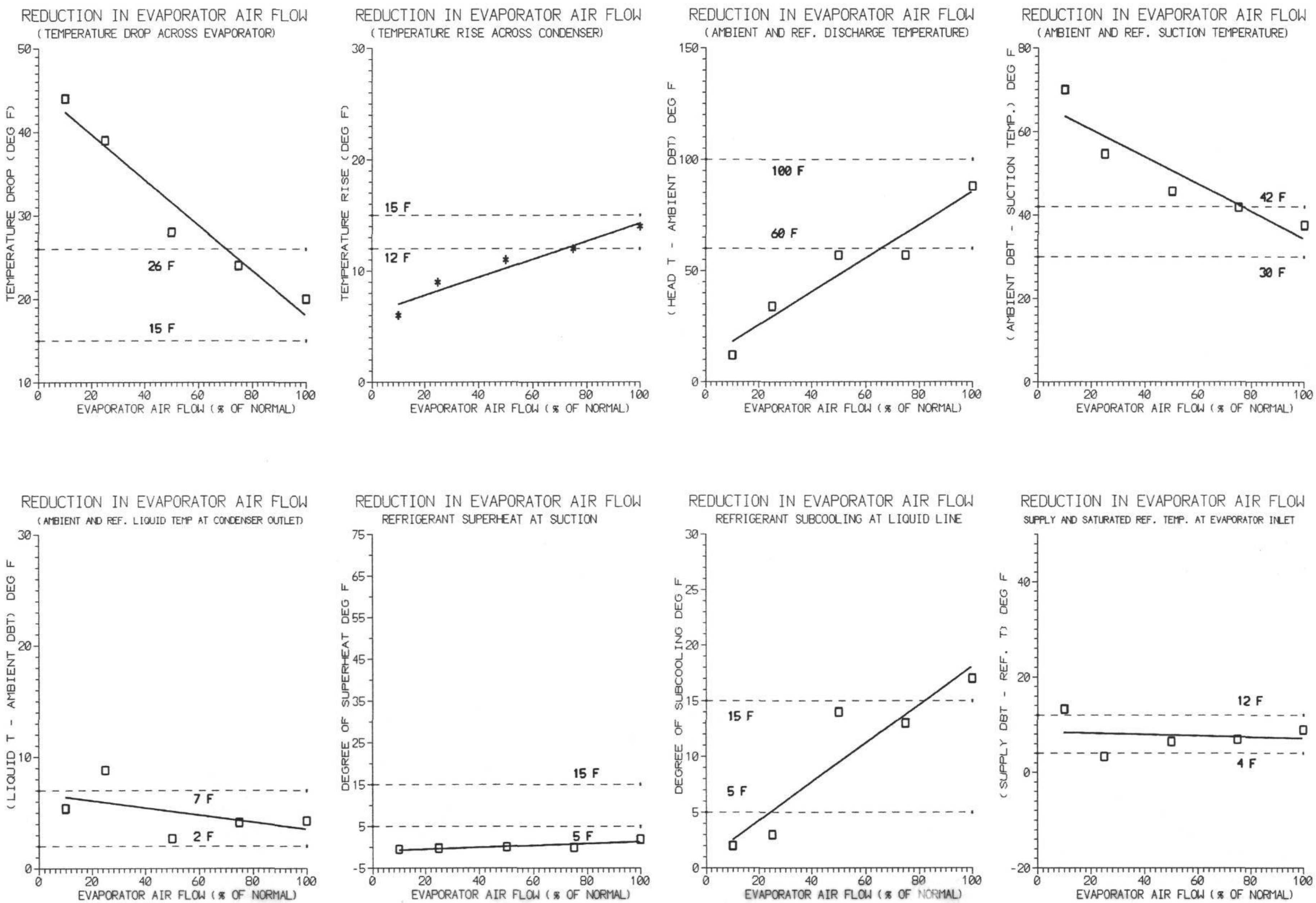


Figure 6.1. Reduced evaporator air flow temperature plots for diagnostic analysis

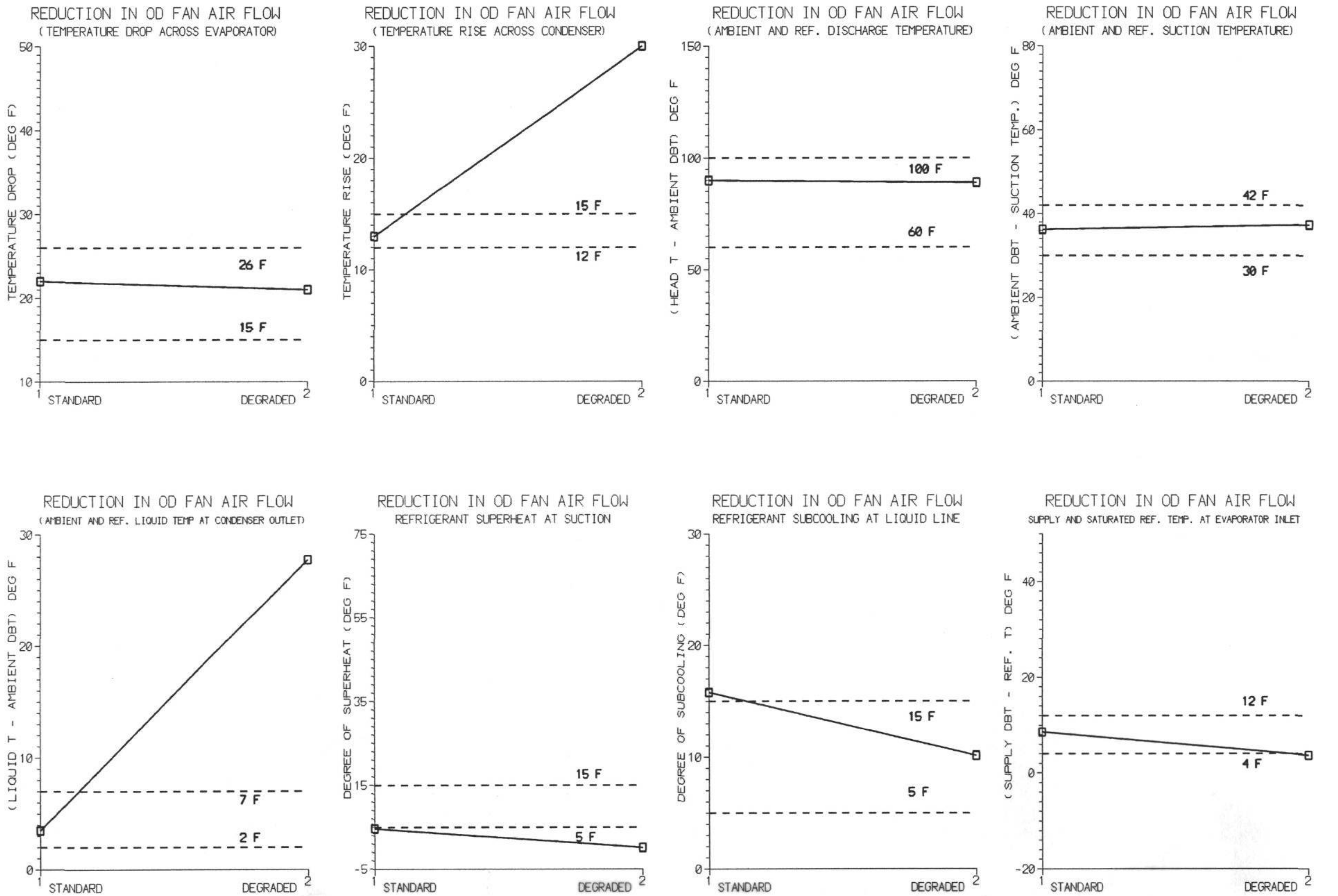


Figure 6.2. Blocked outdoor fan air flow temperature plots for diagnostic analysis

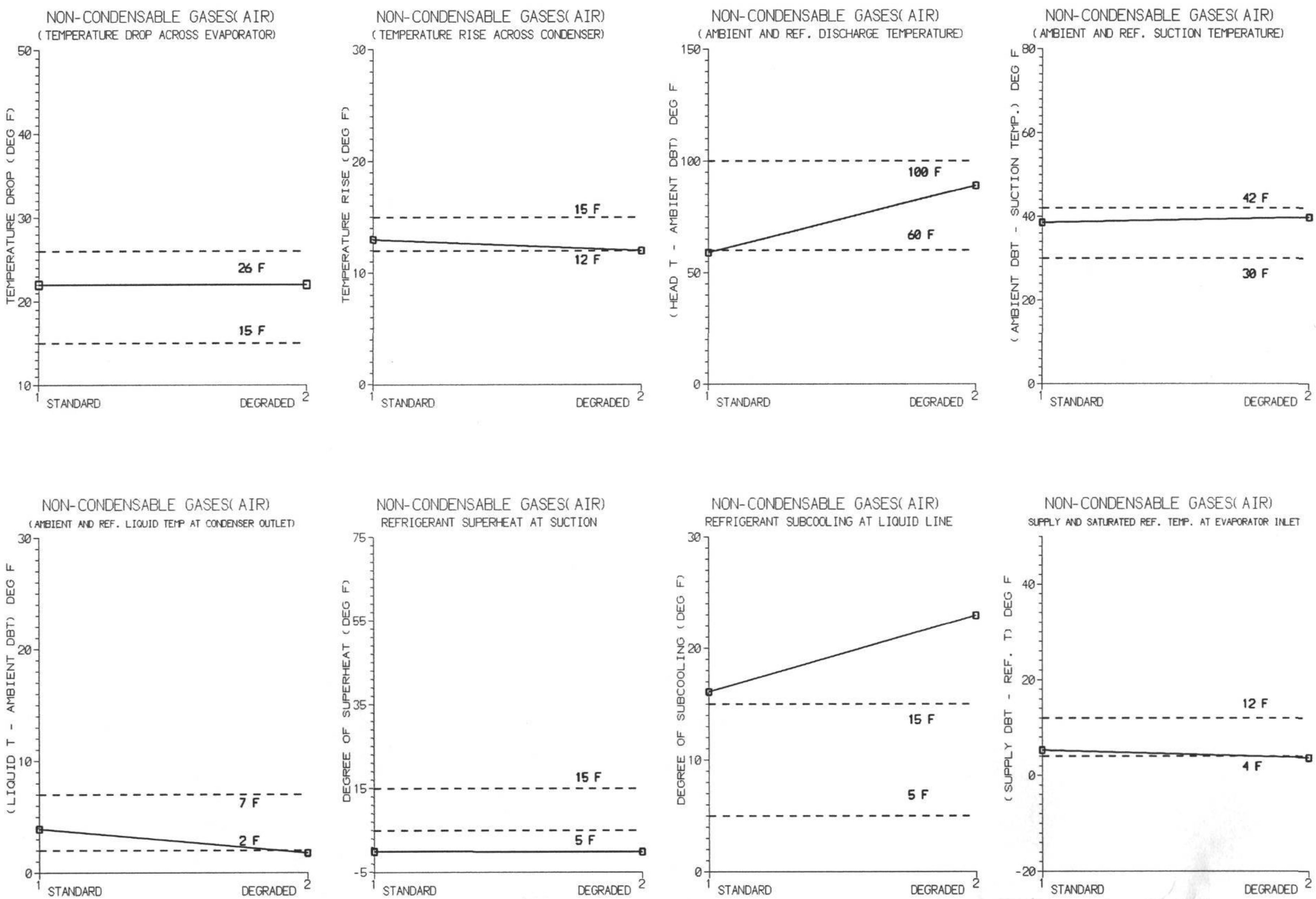


Figure 6.3. Presence of non-condensable gases (air) in system temperature plots for diagnostic analysis

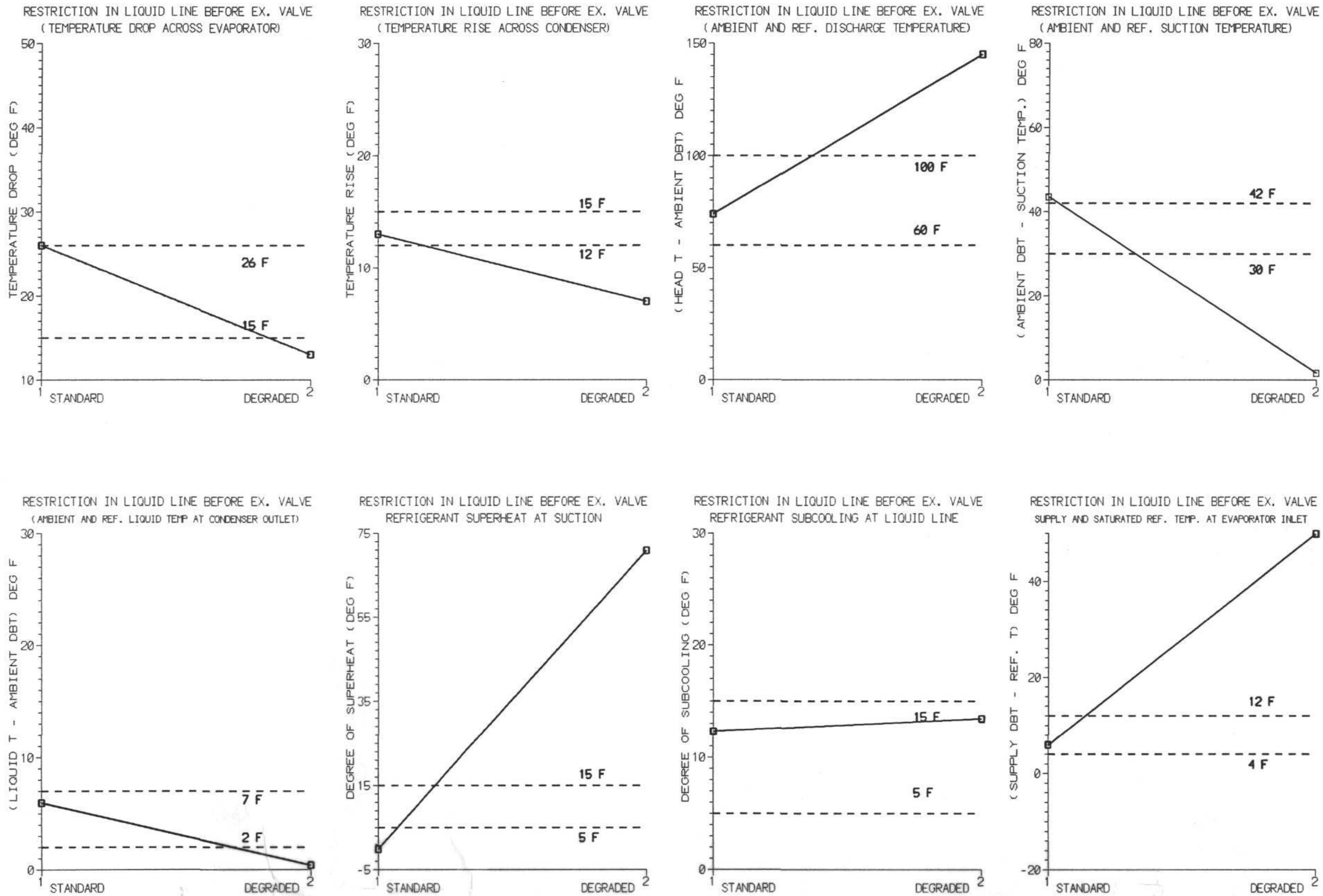


Figure 6.4. Restriction in liquid line before expansion valve temperature plots for diagnostic analysis

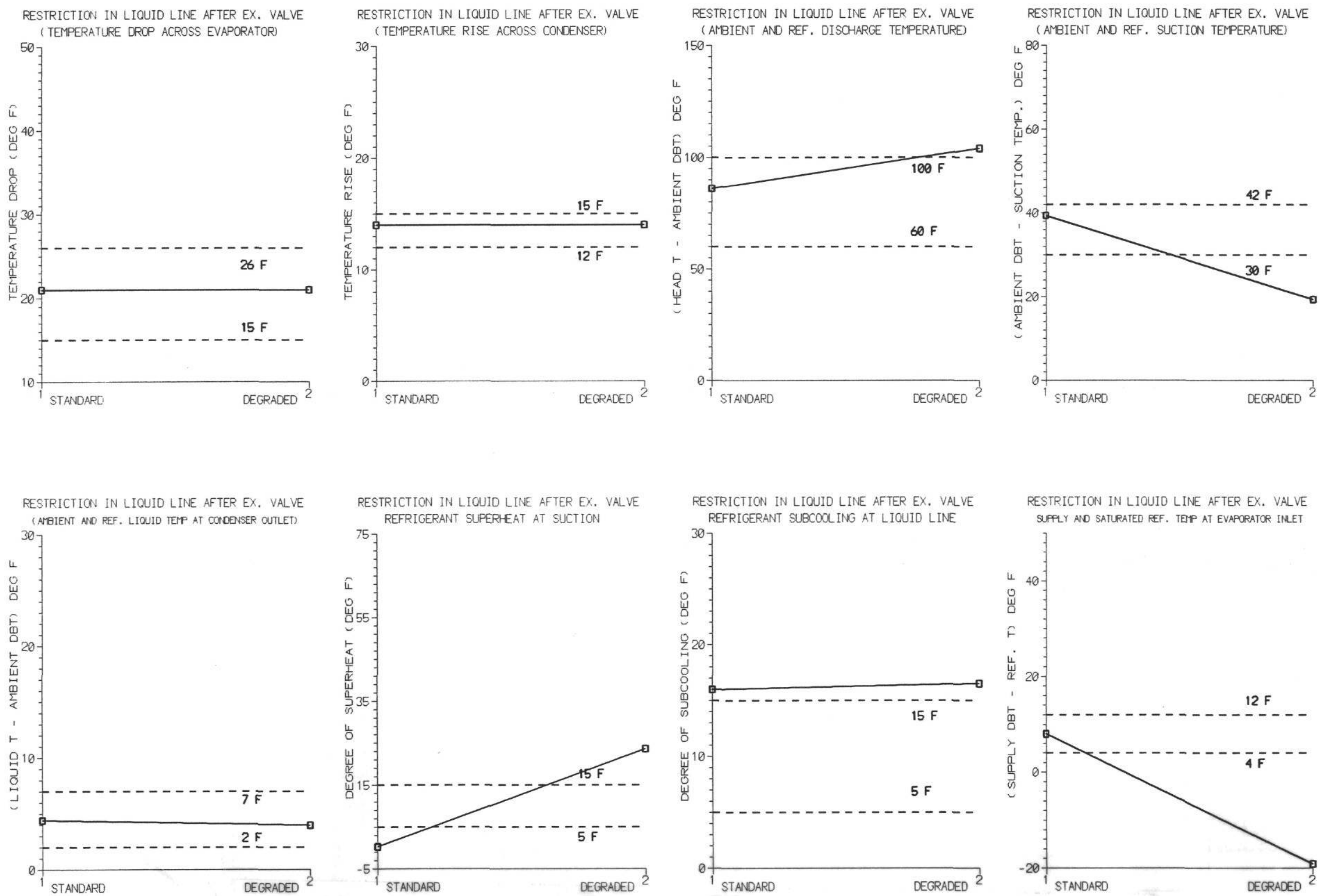


Figure 6.5. Restriction in liquid line after expansion valve temperature plots for diagnostic analysis

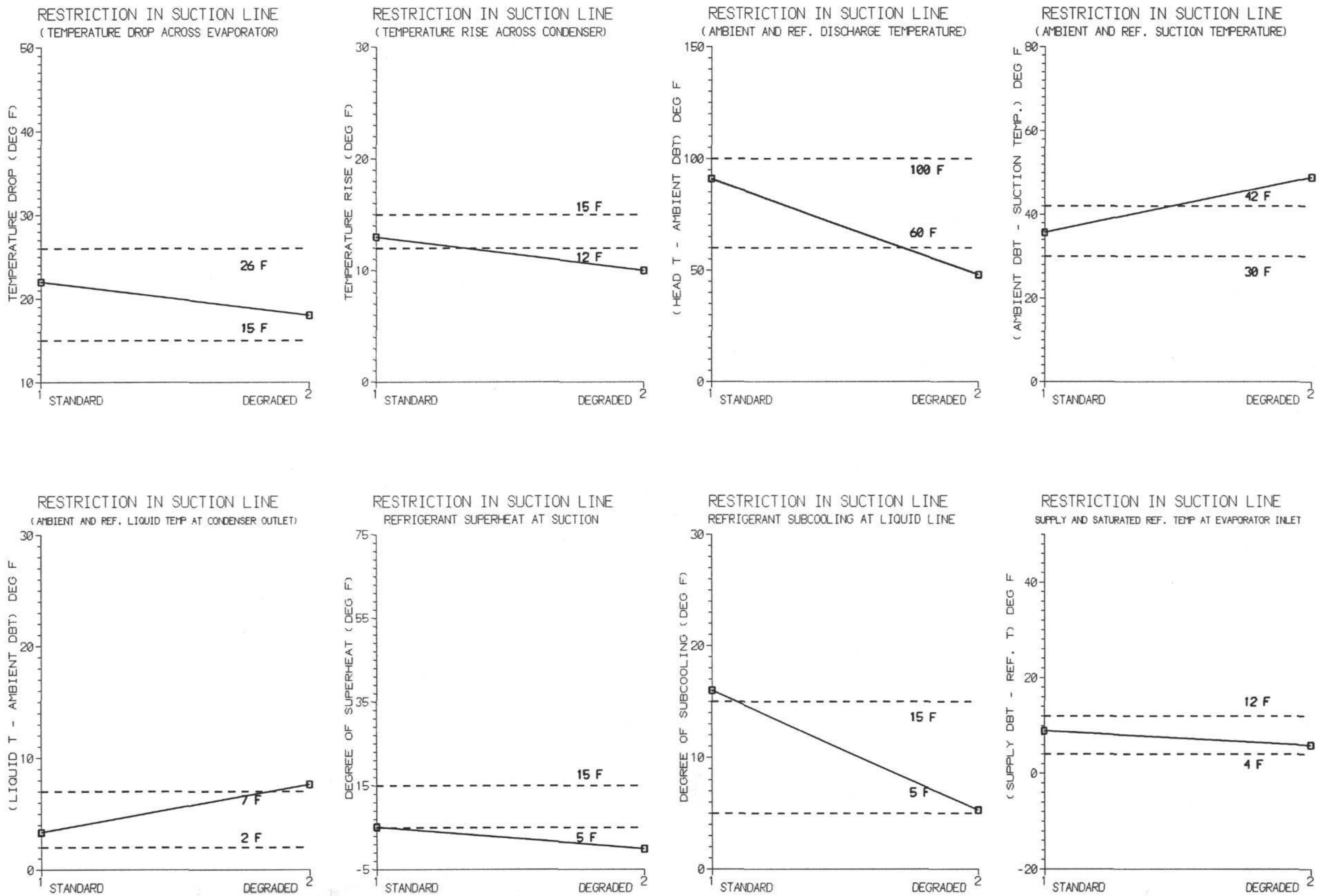


Figure 6.6. Restriction in suction line temperature plots for diagnostic analysis

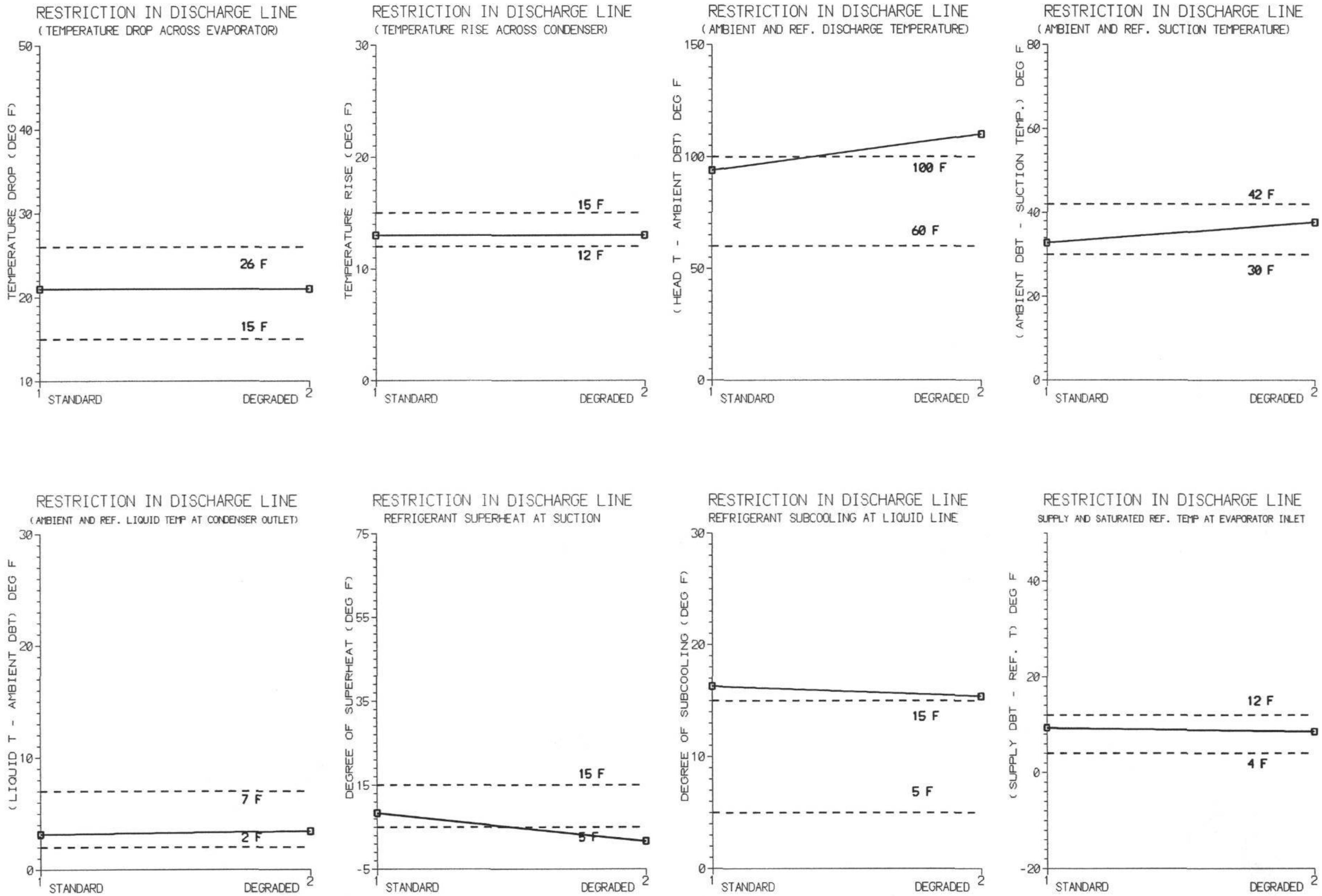


Figure 6.7. Restriction in discharge line temperature plots for diagnostic analysis

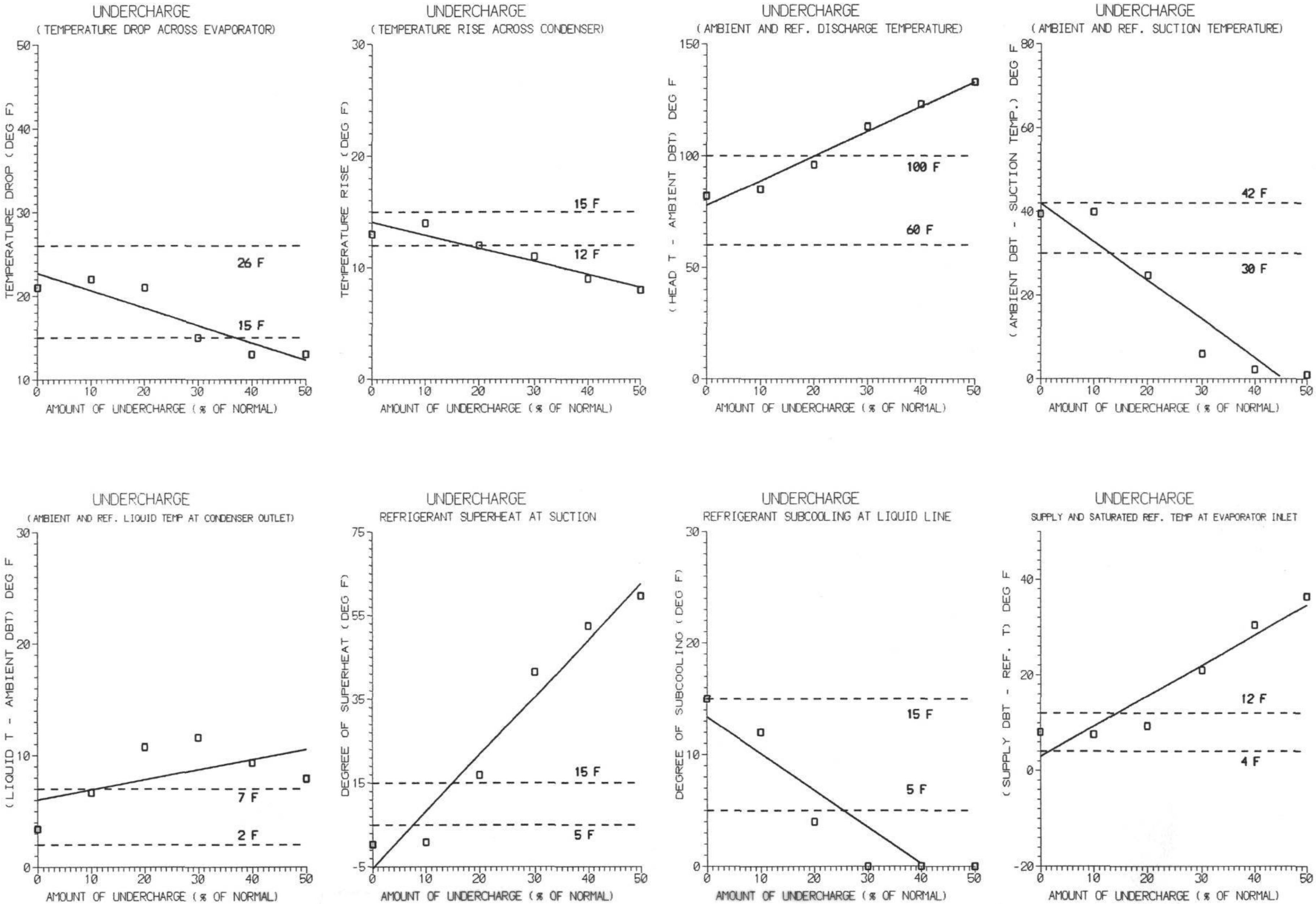


Figure 6.8. Undercharge temperature plots for diagnostic analysis

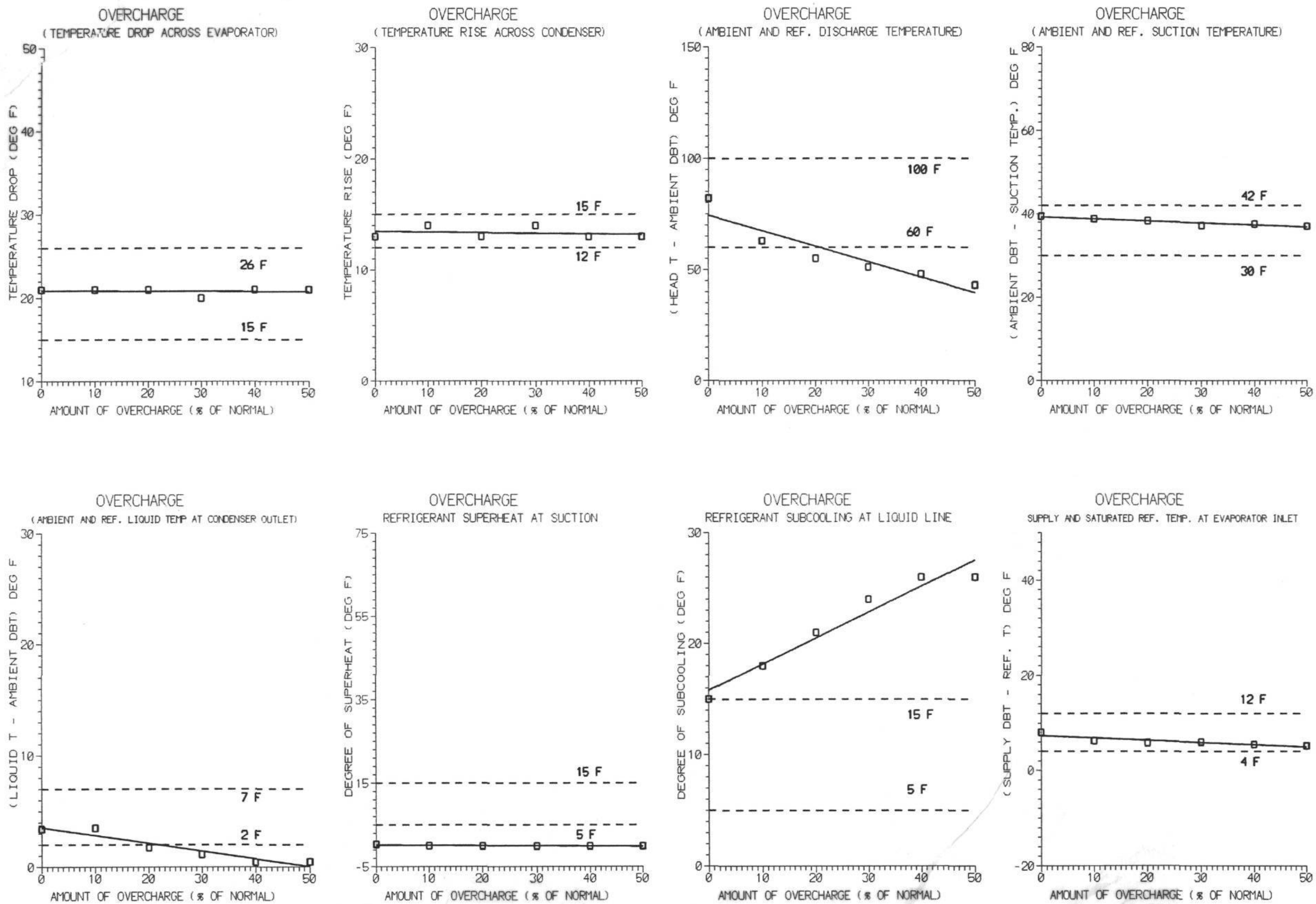


Figure 6.9. Overcharge temperature plots for diagnostic analysis

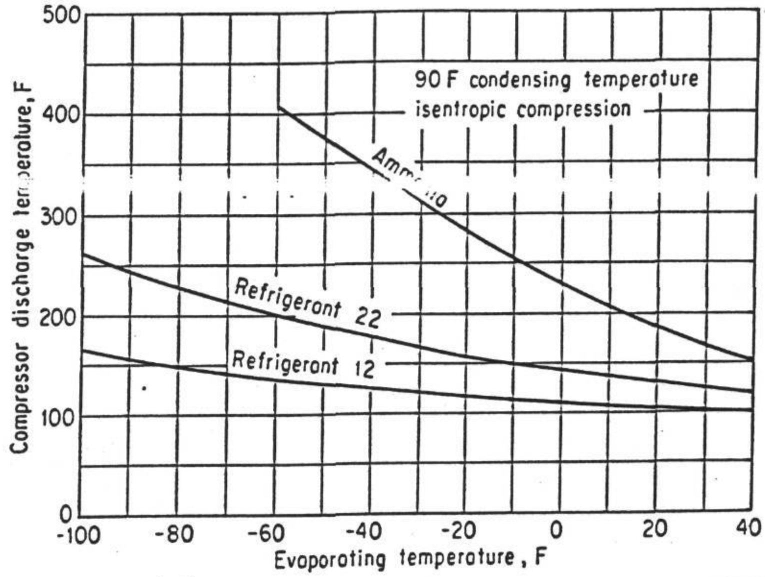


Figure 6.10. Variation of compressor discharge temperature for theoretical single-stage cycle (Source: Threlkeld J.; 1970. Ref. 28)

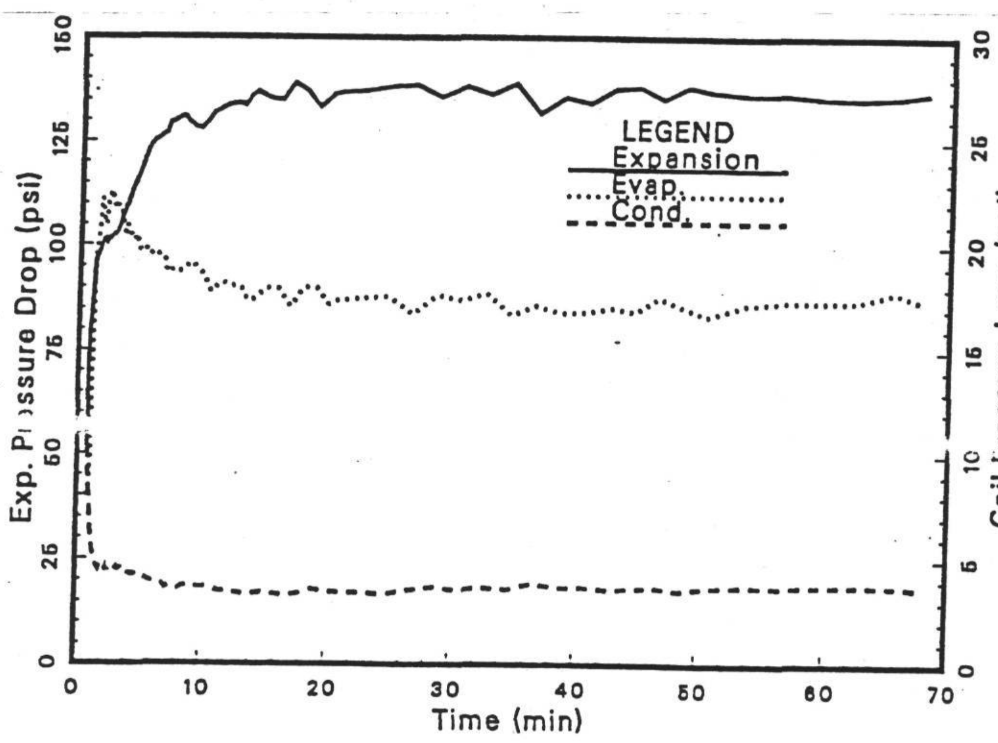


Figure 6.11. Pressure drop across evaporator coil (Source: Katipamula S.; 1989. Ref. 24)

CONCLUSIONS

1. Degraded field conditions of an air conditioner has been measured on a test bench. At 90% reduced evaporator air flow rate, the total power consumption decreases by 17% and the EER decreases by 71%. The degraded condition test results indicated that to maintain sufficient cooling, one definitely must have at least 50% of rated air flow. Low evaporator flow conditions might be inexpensively detected in the field through large temperature difference across cooling coils, reduced condenser discharge temperatures and reduced head temperatures.

2. For 90% reduction in Evaporator airflow, cooling cycle failed to achieve steady state even after considerable amount of period (30 minutes). Frosting due to low evaporator temperature may be the primary reason.

3. Preventing outdoor fan air flow through top of the condensing unit increase the demand and the EER was reduced by 20%.

4. There was no significant change in performance occurred when air was introduced in to the system. Variation in temperature range was also minimum during the degraded test.

5. For 50 psig pressure drop, the maximum degradation occurs for the case of restricting refrigerant flow in the liquid line before expansion valve.

6. There was a considerable change in performance occurs between 20% and 30% undercharge. Demand was reduced from 12.8% at 20% undercharge to 17.2% at 30% undercharge. Up to 20% undercharge there was no significant drop in performance. Undercharge of 30% and above produces considerable levels of degradation.

7. Up to 50% of overcharge there was no significant change in demand and EER.

7. A guideline for diagnosing a degraded air conditioner using temperature measurements was presented.

FUTURE DIRECTIONS

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

The ultimate goal of this project is to develop a temperature based diagnosis procedure. Experience with preliminary tests indicated that easily identifiable temperature patterns exist for each degraded condition which can be utilized for developing an automated diagnosis procedure.

Some commercially available automatic diagnostic systems are already beginning to appear. Kaler developed one such monitoring device which continuously monitors the HVAC system for selected system malfunctions [30]. This device continuously tracks the temperatures of an HVAC system and evaluates this value with an embedded knowledge base, and warns the owner in advance of any potential trouble.

"CoolGuard" developed by Dencor Inc., is an electronic monitoring device that can detect failure symptoms [31]. It monitors return and supply air temperature and outdoor air temperature for abnormal values. Remote monitoring possibility is also available with this system.

"Fluke 52" a digital recording thermometer developed by Fluke Inc. can record minimum and maximum values of a set point in the system, or minimum and maximum values of the temperature difference between two points [29]. Measurements over time (subcooling or superheating) can be recorded using this instrument. Further testing of such systems on a test bench can accelerate their acceptance into the marketplace.

Danfoss-EMC Inc., markets NC-25, a compressor rack control system for supermarket refrigeration racks [32]. As large percentage of energy consumed in supermarket environment is due to refrigeration compressors, the NC-25 system optimizes compressor pressure for saving energy and to maintain trouble free operation. Danfoss also provides remote service through a central monitoring center. Utilities can similarly provide a central service system for residential air conditioners by measuring temperatures.

ACKNOWLEDGMENTS

This work was funded through the Texas A&M Energy Systems Laboratory Research Consortium.

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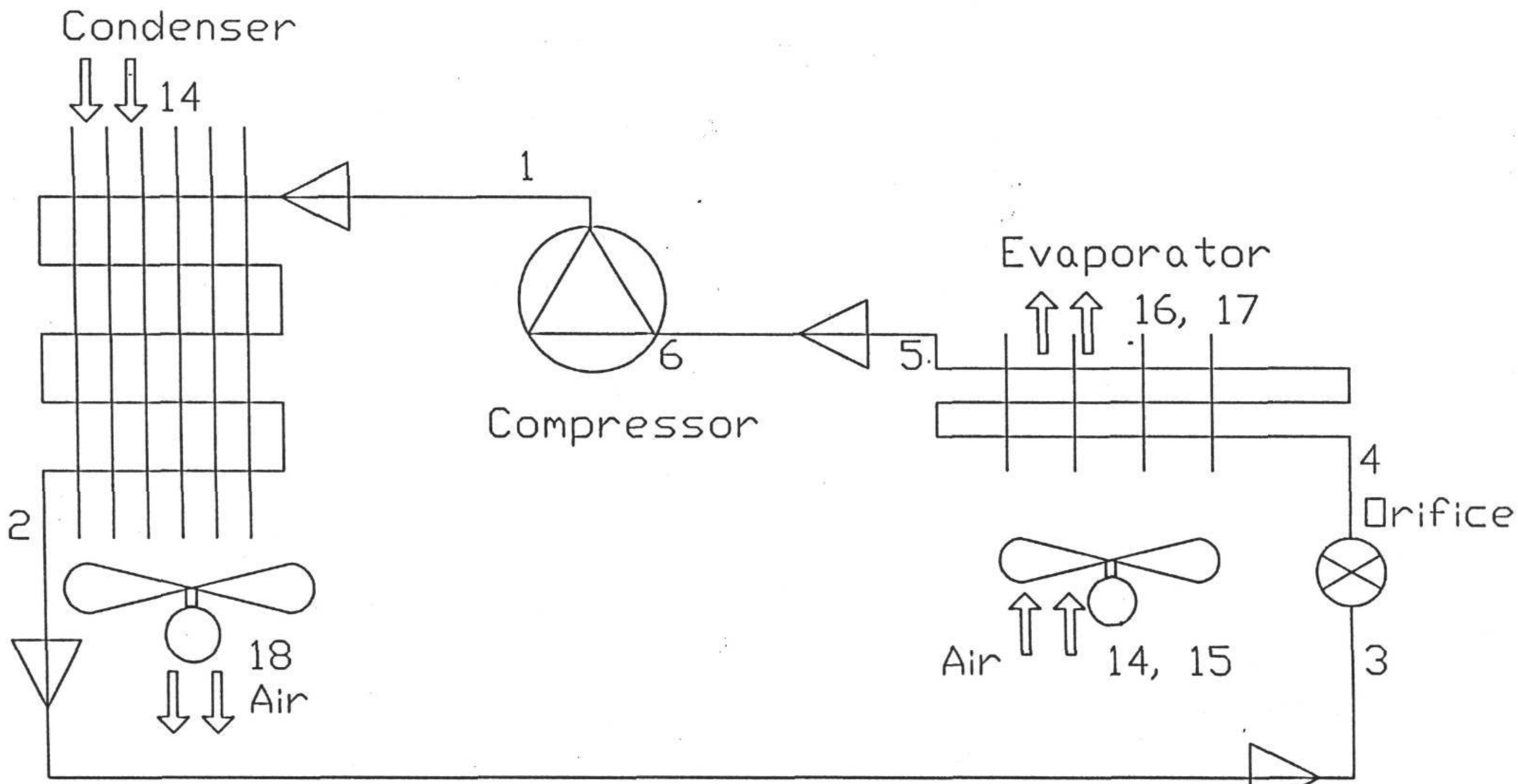
APPENDIX

DATA ACQUISITION AND ANALYSIS

A detailed review of data acquisition and analysis is discusses here. The time series plots for standard and degraded tests are presented exactly as measured except the supply air flow rate. The supply air flow rate was presented as CFM in time series plots assuming standard supply air conditions. Variation in specific humidity and density were considered during the final calculations. Relative humidity values are presented as measured. During performance calculations the values are adjusted with the calibration results of RH sensors.

Airflow across evaporator is reduced when dehumidification started occurring at the surface of evaporator. This caused a small variation in power and air flow rate. Equilibrium conditions during 75% reduced evaporator air flow was achieved slowly and during 90% reduced evaporator air flow rate the steady state conditions were never obtained.

Supply air flow rate across evaporator was low (930 CFM) during restriction in liquid line (before expansion valve) tests. The refrigerant flow rate during 40% and 50% undercharge are spurious since the mass flow meter used to measure the refrigerant flow is designed to measure single phase fluid only. This difficulty can overcome by placing the mass flow sensor in the discharge line.



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

14, 15 Ambient DBT and RH
16, 17 Supply DBT and RH
18 Condenser exit

Figure A-1. Schematic of refrigerant and air flow

PERFORMANCE CALCULATIONS (ASHRAE Std. 116-1983) [Ref. 33]

Air side enthalpy difference, Δh (Btu/lb.) $= (h_{ra} - h_{sa})$ (1)

Cooling capacity (Btu/hr)

Cooling capacity $= (60 \text{ X } \Delta h \text{ X } Q_{sa} \text{ (CFM)}) / (v_{sa} \text{ X } (1 + w_{sa}))$ (2)

Energy efficiency ratio (EER)

EER = Cooling capacity / Total power (3)

SIMULATED FAULTS	STANDARD TEST RESULTS		
	CAPACITY KBtu/hr	POWER WATTS	EER
REDUCTION IN EVAPORATOR AIR FLOW			
1. 25% Reduction	37.10	3588	10.3
2. 50% Reduction	37.10	3588	10.3
3. 75% Reduction	35.49	3626	9.8
4. 90% Reduction	35.49	3626	9.8
BLOCK OD FAN AIR FLOW	34.90	3458	10.1
PRESENCE OF NON-CONDENSABLE GASES	34.91	3542	9.85
RESTRICTING REFRIGERANT LINES			
1. Before Expansion Valve	35.64	3570	9.98
2. After Expansion Valve	36.08	3503	10.3
3. Suction Line	35.43	3487	10.2
4. Discharge Line	34.15	3483	9.8
REFRIGERANT UNDERCHARGING			
1. 10 % Undercharge	35.01	3486	10.04
2. 20 % Undercharge	35.01	3486	10.04
3. 30 % Undercharge	35.01	3486	10.04
4. 40 % Undercharge	35.01	3486	10.04
5. 50 % Undercharge	35.01	3486	10.04
REFRIGERANT OVERCHARGING			
1. 10 % Overcharge	35.01	3486	10.04
2. 20 % Overcharge	35.01	3486	10.04
3. 30 % Overcharge	35.01	3486	10.04
4. 40 % Overcharge	35.01	3486	10.04
5. 50 % Overcharge	35.01	3486	10.04

Table B-1. Summary of performance factors for standard tests

SIMULATED FAULTS	DEGRADED TEST RESULTS		
	CAPACITY KBtu/hr	POWER WATTS	EER
REDUCTION IN EVAPORATOR AIR FLOW			
1. 25% Reduction	35.02	3480	10.1
2. 50% Reduction	32.85	3340	9.84
3. 75% Reduction	23.91	3191	7.5
4. 90% Reduction	13.37	2857	4.7
BLOCK OD FAN AIR FLOW	32.37	4010	8.07
PRESENCE OF NON-CONDENSABLE GASES	36.05	3566	10.1
RESTRICTING REFRIGERANT LINES			
1. Before Expansion Valve	12.45	2842	4.38
2. After Expansion Valve	33.95	3452	9.83
3. Suction Line	24.66	3324	7.42
4. Discharge Line	33.75	3800	8.88
REFRIGERANT UNDERCHARGING			
1. 10 % Undercharge	35.53	3490	10.18
2. 20 % Undercharge	33.27	3337	9.97
3. 30 % Undercharge	21.14	3192	6.62
4. 40 % Undercharge	16.02	3039	5.27
5. 50 % Undercharge	14.52	2887	5.03
REFRIGERANT OVERCHARGING			
1. 10 % Overcharge	35.78	3586	9.98
2. 20 % Overcharge	36.52	3616	10.1
3. 30 % Overcharge	33.70	3650	9.23
4. 40 % Overcharge	34.60	3637	9.51
5. 50 % Overcharge	34.44	3589	9.6

Table B-2. Summary of performance factors for degraded tests

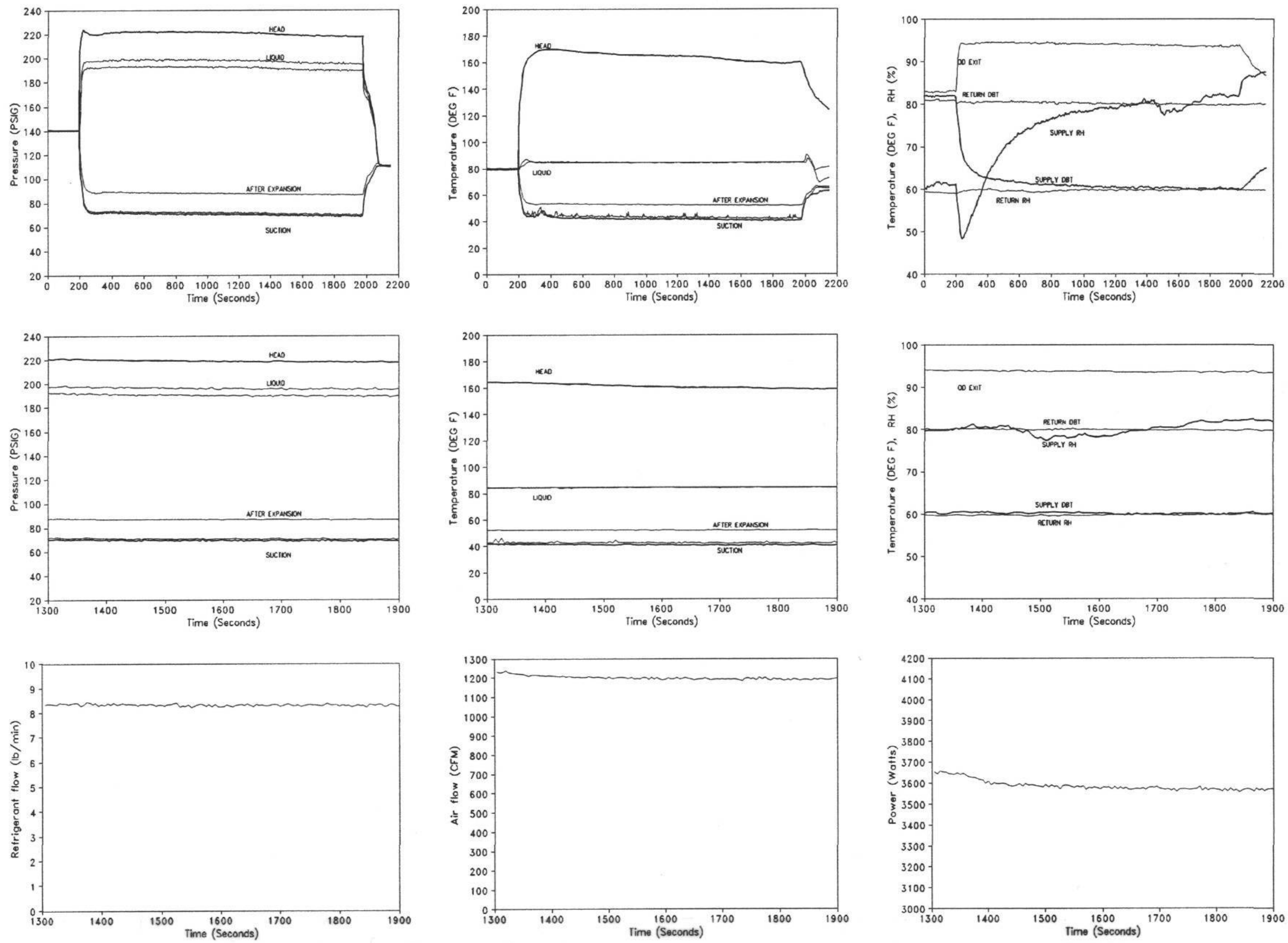


Figure B-1.1. Time series plot for 25% reduced evaporator air flow standard test

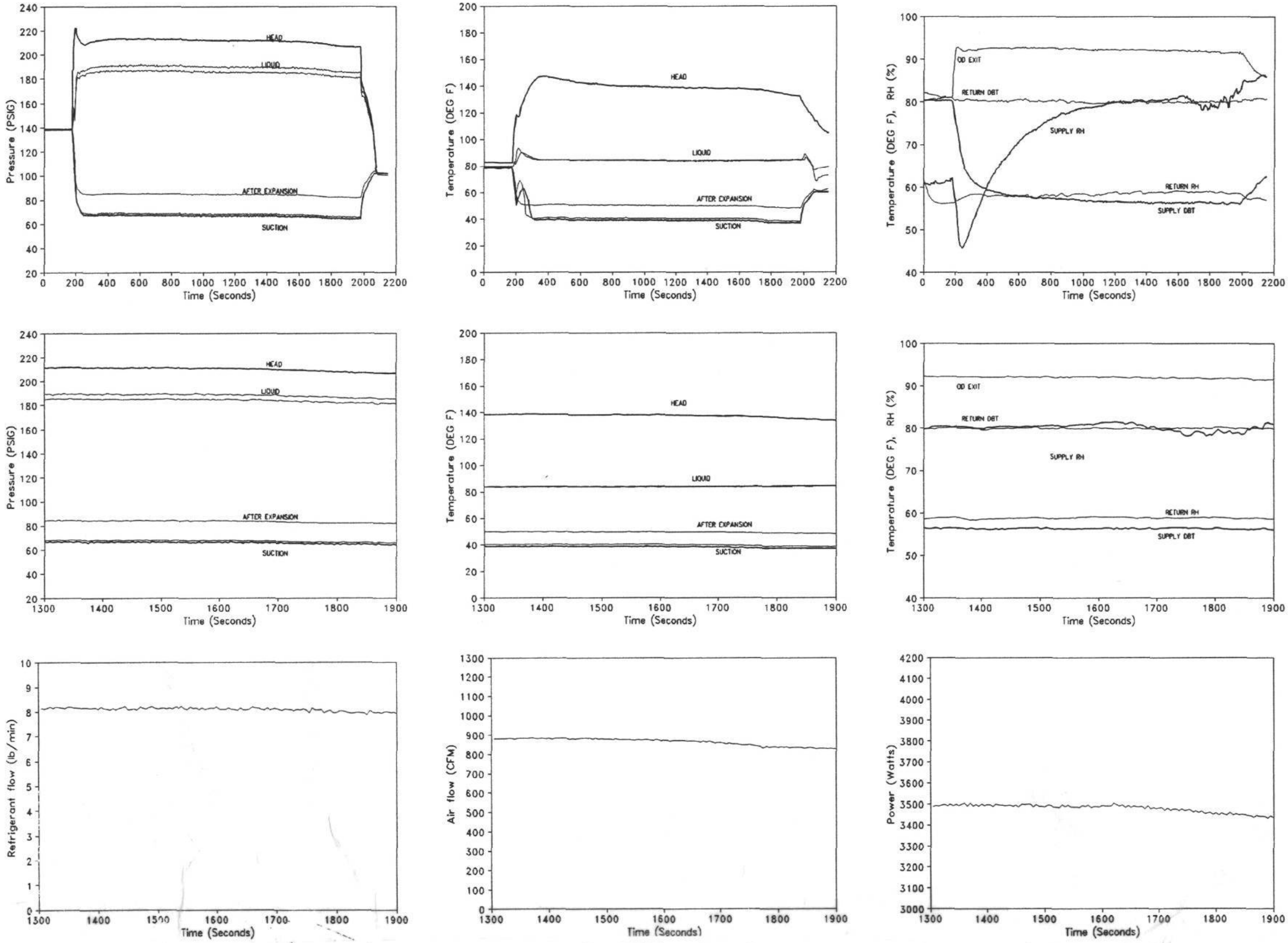


Figure B-1.1.2. Time series plot for 25% reduced evaporator air flow degraded test

STANDARD RUN			25 % CUT IN EVAP AIR		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 234	161	124	1. 225	137	119.4
2. 211	85	34.5	2. 203	84	34.3
3. 205	84	34.3	3. 199	84	34.2
4. 102	52	34.3	4. 98	49	34.2
5. 85	42	108.4	5. 82	40	108.2
6. 84	41	108.2	6. 81	38	108.0
REF. CAP		37.03 KBtu/hr (3.09 TONS)			35.95 KBtu/hr (3.00 TONS)
REF. FLOW		8.33 LB/MIN			8.10 LB/MIN

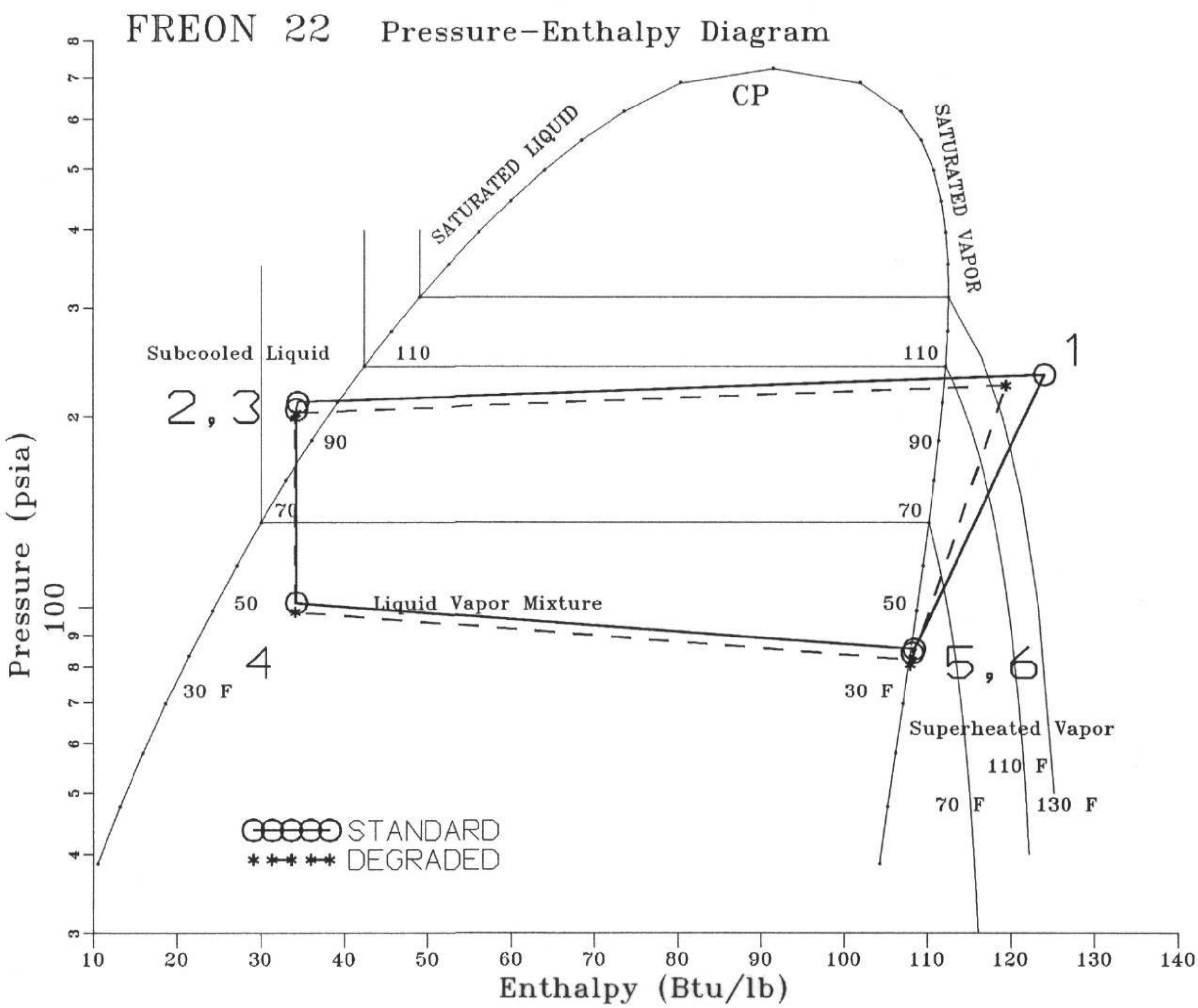


Figure B-1.1.3. P-H Chart for 25% reduced evaporator air flow

STANDARD RUN			
	DBT(F)	RH(%)	W(lbm/lb air)
13.	79.9	54	0.0117
15.	60.1	89	0.0099
16.	93.7		0.0117
EVAP. AIR FLOW	1185		
CAPACITY	37.1		

25 % CUT IN EVAP AIR			
	DBT(F)	RH(%)	W(lbm/lb air)
13.	79.9	53	0.0115
15.	56.3	89	0.0086
16.	91.9		0.0115
848 CFM			
35.02 KBtu/hr			

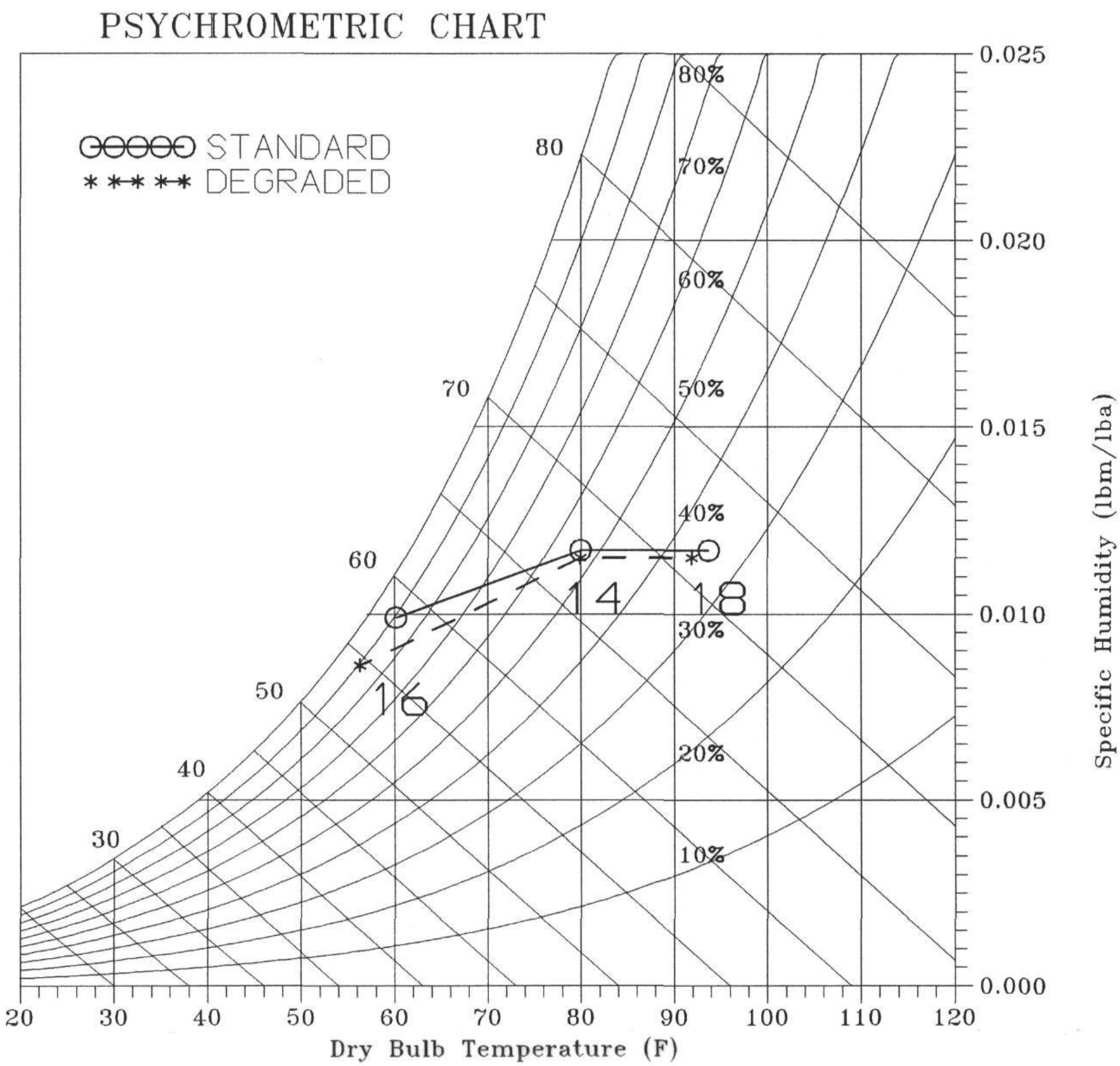


Figure B-1.1.4. Psychrometric Chart for 25% reduced evaporator air flow

	STANDARD RUN			50 % CUT IN EVAP AIR		
	P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1.	234	161	124	217	137	119.8
2.	211	85	34.5	199	82	33.7
3.	205	84	34.3	195	81	33.6
4.	102	52	34.3	90	45	33.6
5.	85	42	108.4	75	35	107.8
6.	84	41	108.2	74	33	107.7
REF. CAP			37.03 KBtu/hr (3.09 TONS)			32.78 KBtu/hr (2.73 TONS)
REF. FLOW			8.33 LB/MIN			7.36 LB/MIN

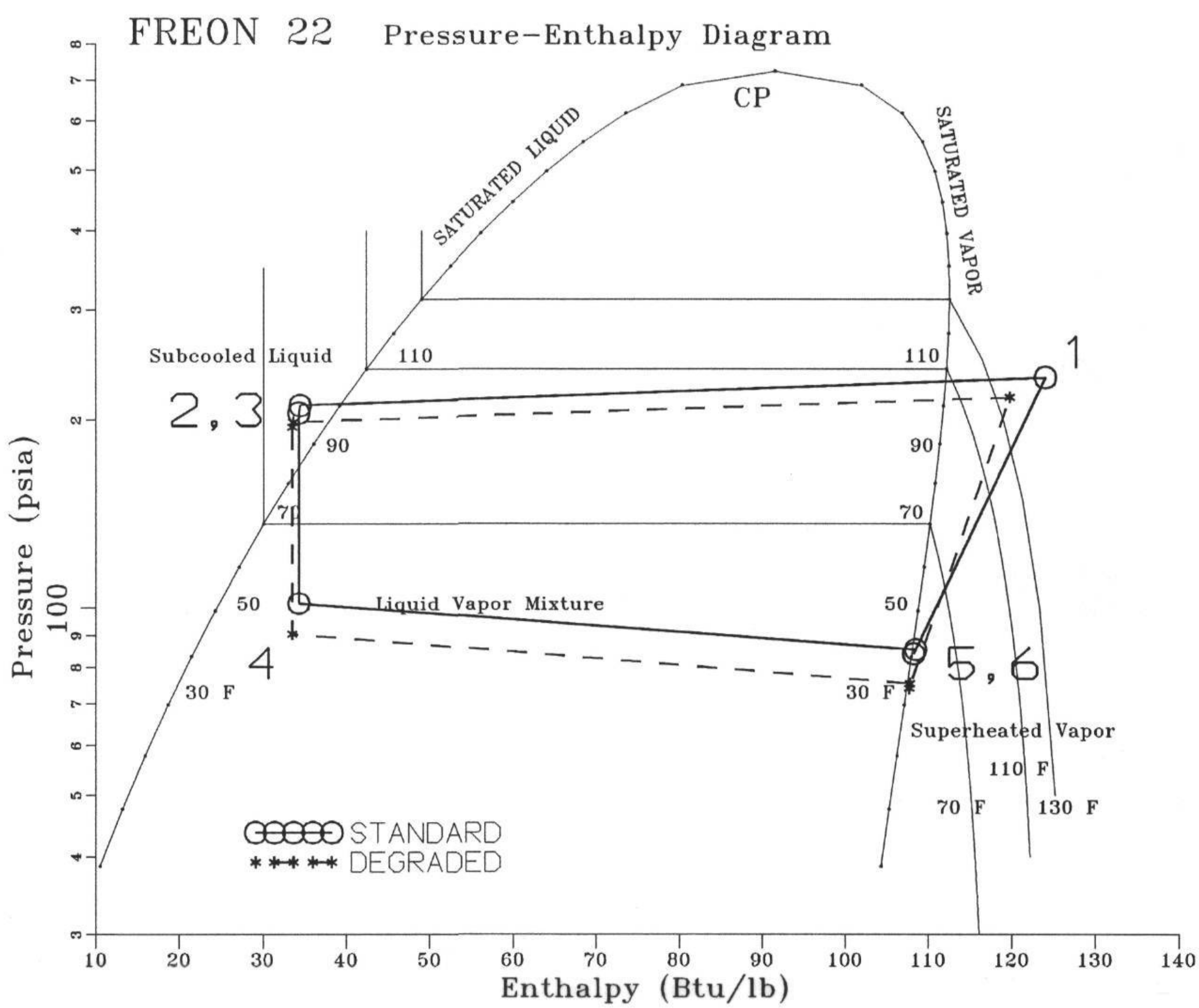


Figure B-1.2.2 P-H Chart for 50% reduced evaporator air flow

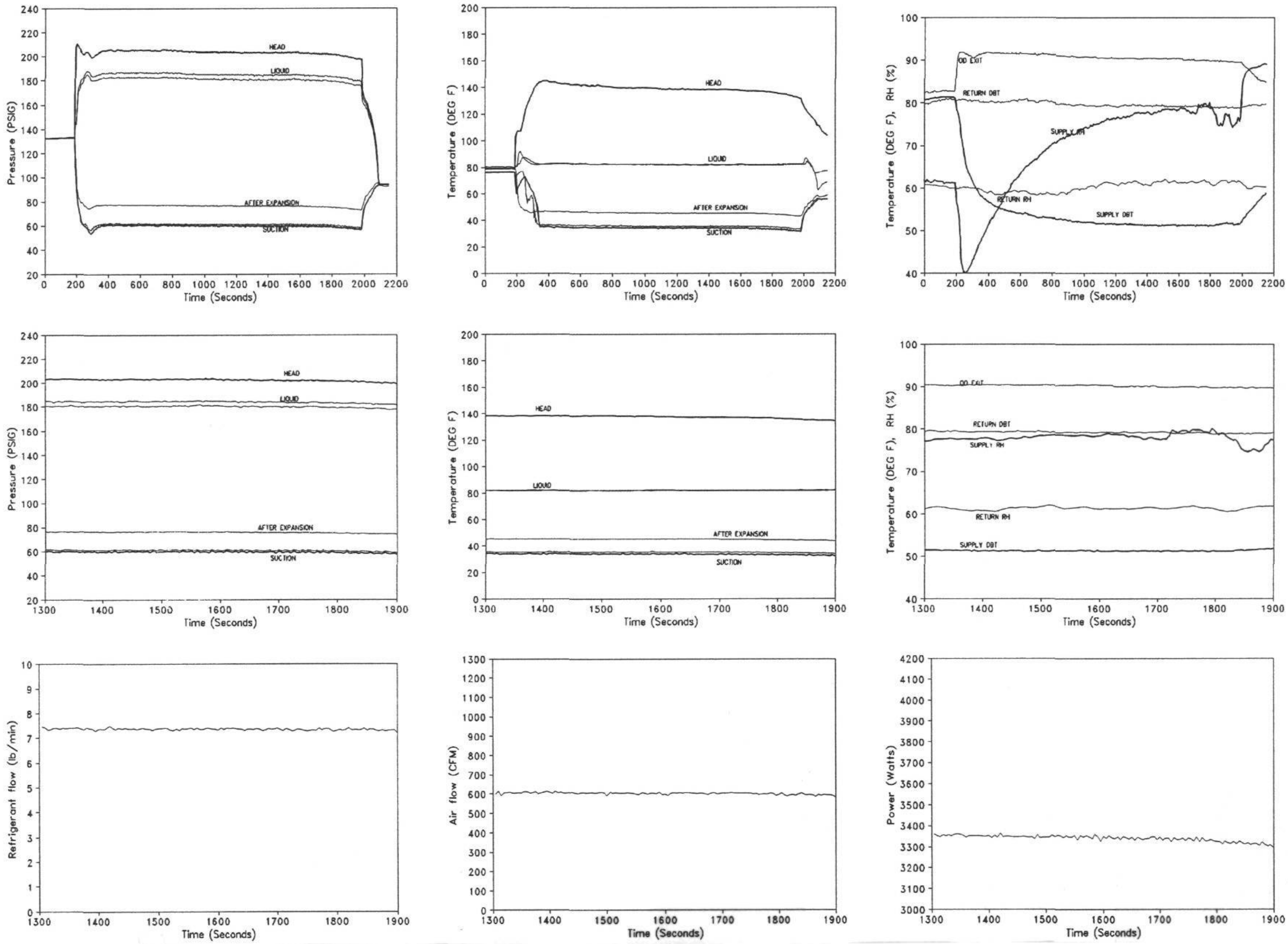


Figure B-1.2.1. Time series plot for 50% reduced evaporator air flow degraded test

STANDARD RUN				50 % CUT IN EVAP AIR			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	79.9	54	0.0117	13.	79.2	55	0.0118
15.	60.1	89	0.0099	15.	51.2	88	0.0070
16.	93.7		0.0117	16.	90.1		0.0118
EVAP. AIR FLOW			1185	589 CFM			
COOLING CAPACITY			37.1	32.85 KBtu/hr			

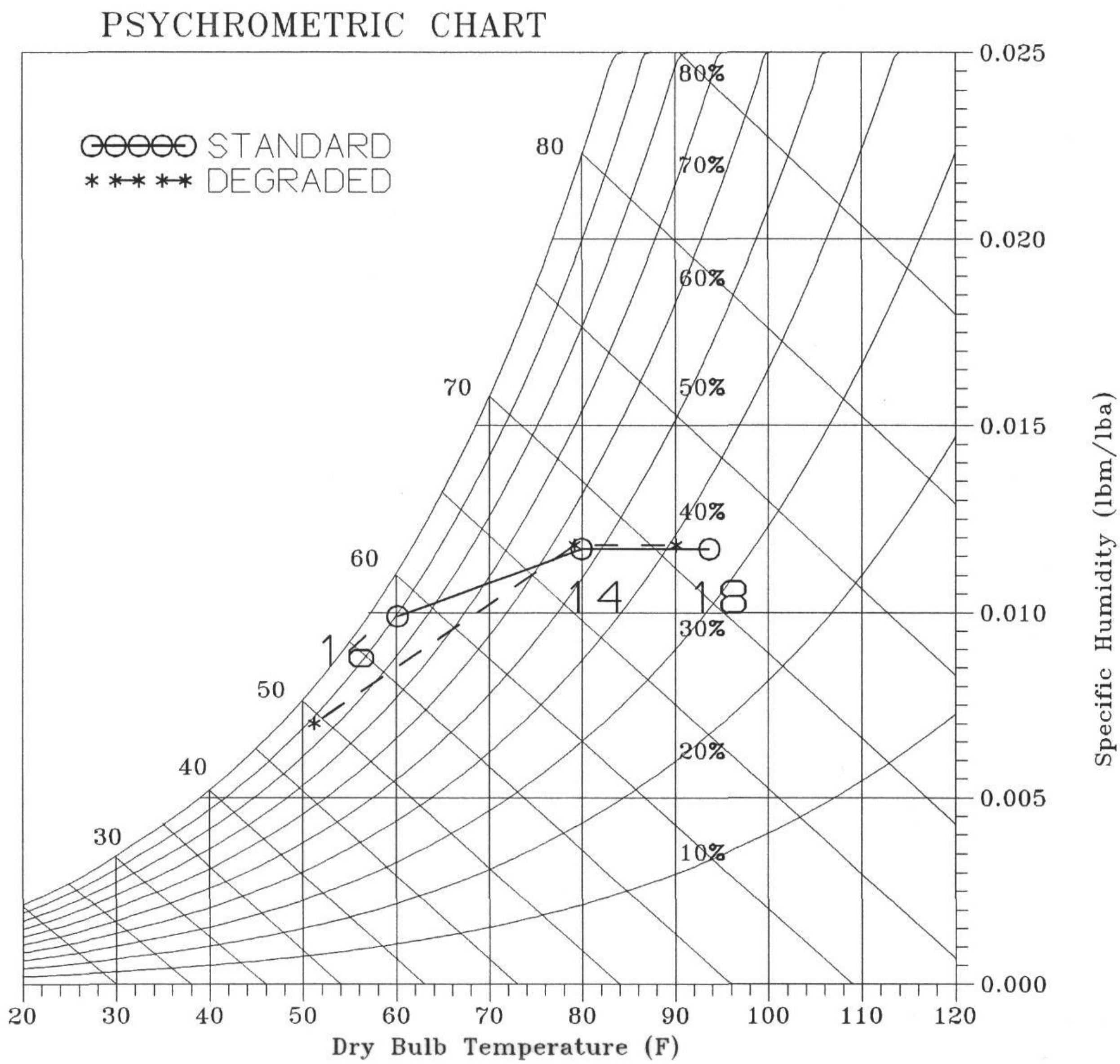


Figure B-1.2.3 Psychrometric Chart for 50% reduced evaporator air flow

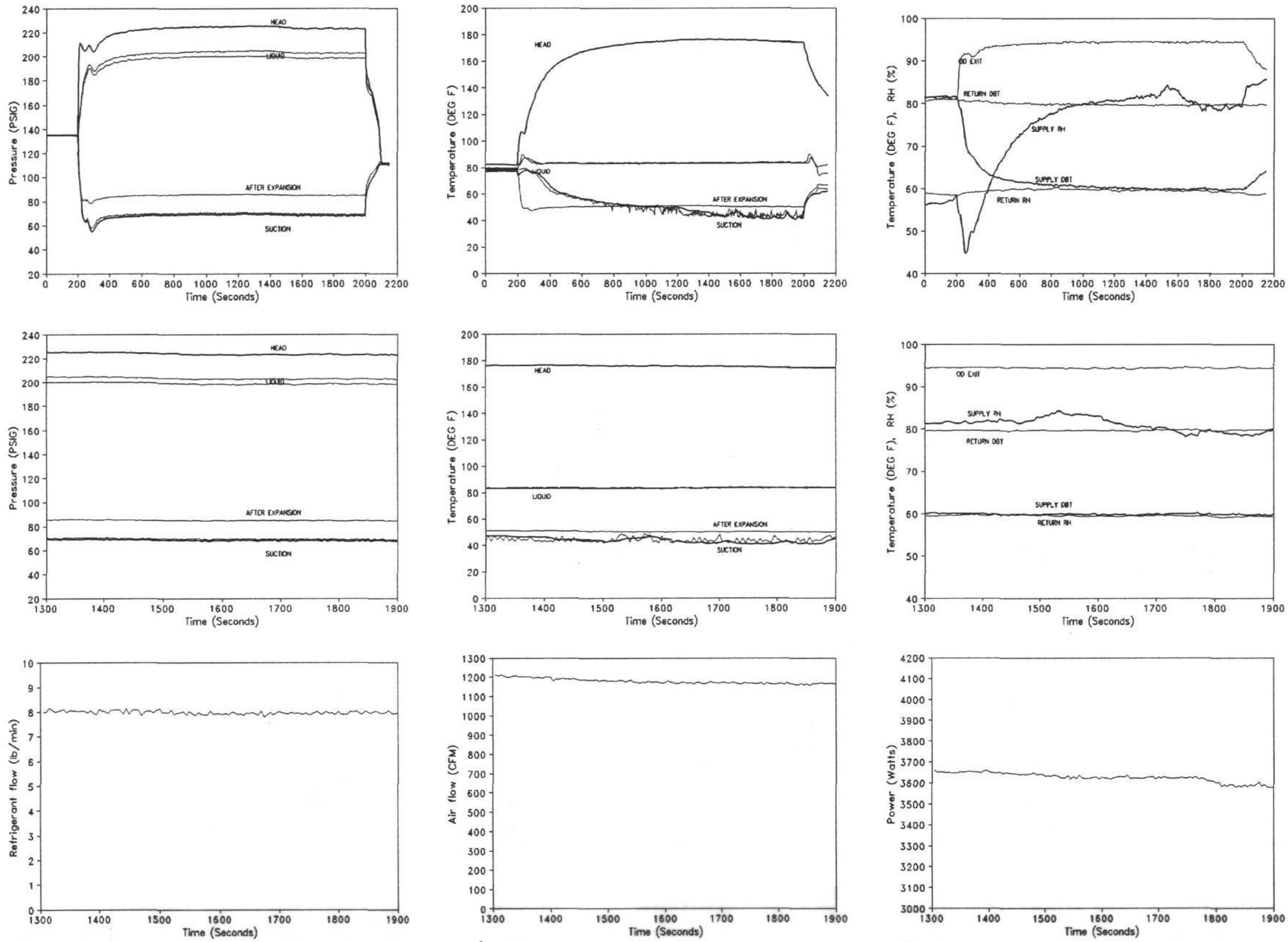


Figure B-1.3.1. Time series plot for 75% reduced evaporator air flow standard test

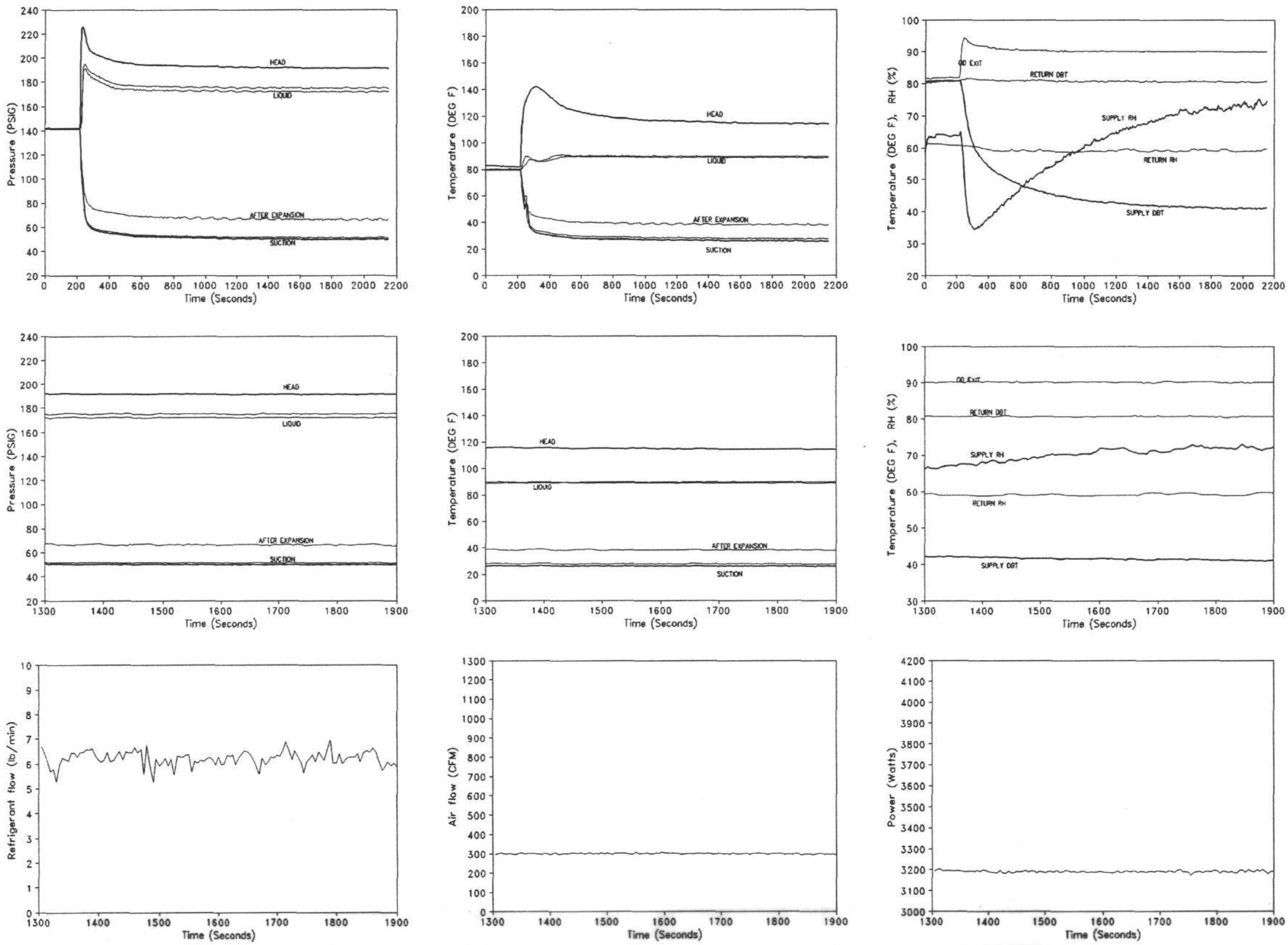


Figure B-1.3.2. Time series plot for 75% reduced evaporator air flow degraded test

STANDARD RUN				75 % CUT IN EVAP AIR		
P(psia)	T(F)	H(Btu/lb)		P(psia)	T(F)	H(Btu/lb)
1. 238	176	126.9		1. 207	115	115.6
2. 218	84	34.2		2. 190	89	36.7
3. 214	83	34.1		3. 187	89	35.8
4. 100	50	34.1		4. 81	38	35.8
5. 84	44	108.8		5. 66	27	107.2
6. 83	44	108.8		6. 65	26	107.1
REF. CAP		35.74 KBtu/hr (2.98 TONS)				26.61 KBtu/hr (2.22 TONS)
REF. FLOW		7.97 LB/MIN				6.21 LB/MIN

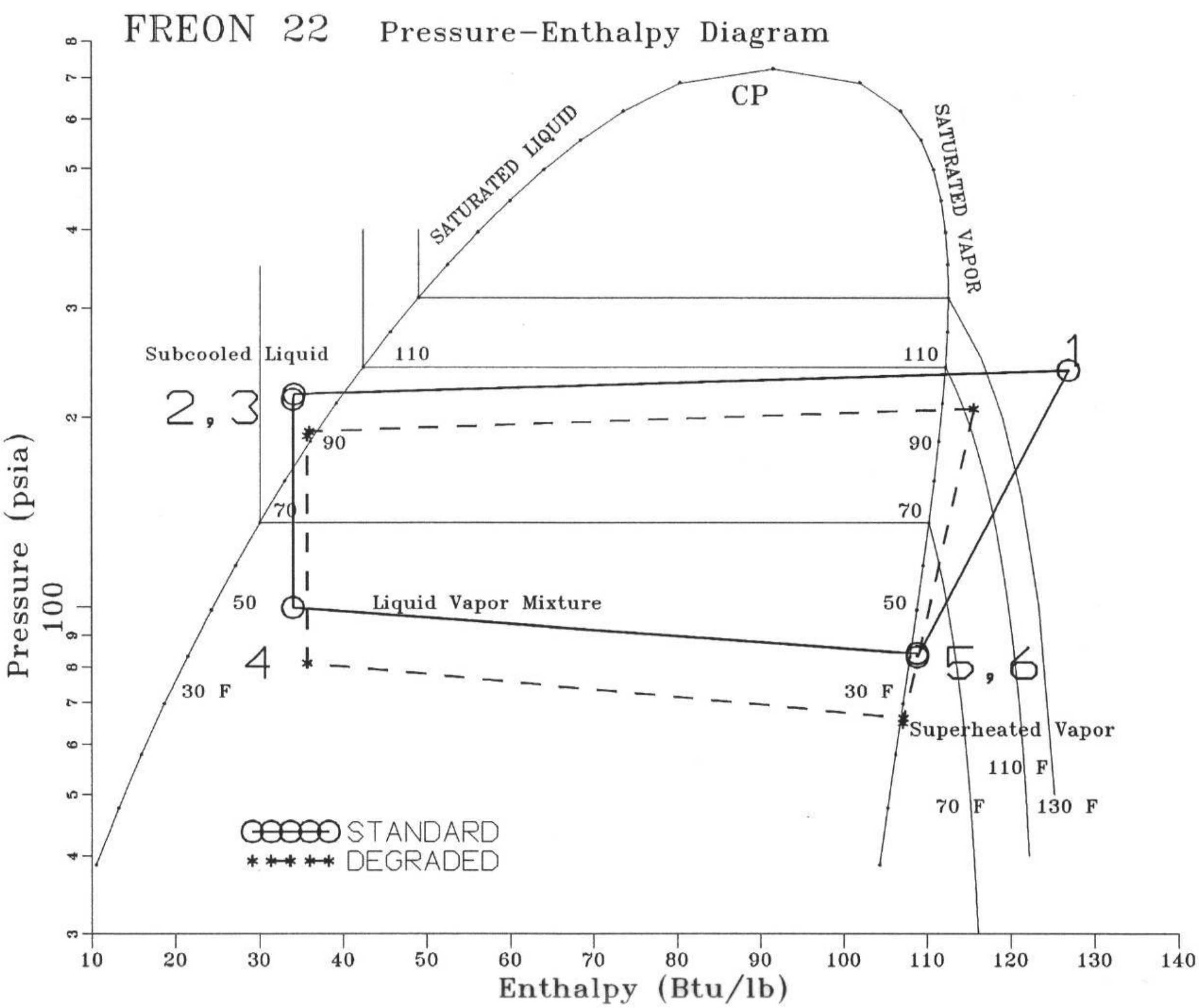


Figure B-1.3.3. P-H Chart for 75% reduced evaporator air flow

STANDARD RUN

	DBT(F)	RH(%)	W(lbm/lb air)
13.	79.7	53	0.0116
15.	59.9	90	0.0099
16.	94.4		0.0116

EVAP. AIR FLOW 1163
COOLING CAPACITY 35.49

75 % CUT IN EVAP AIR

	DBT(F)	RH(%)	W(lbm/lb air)
13.	80.7	53	0.0119
15.	41.6	84	0.0046
16.	90.1		0.0119

290 CFM
23.91 KBtu/hr

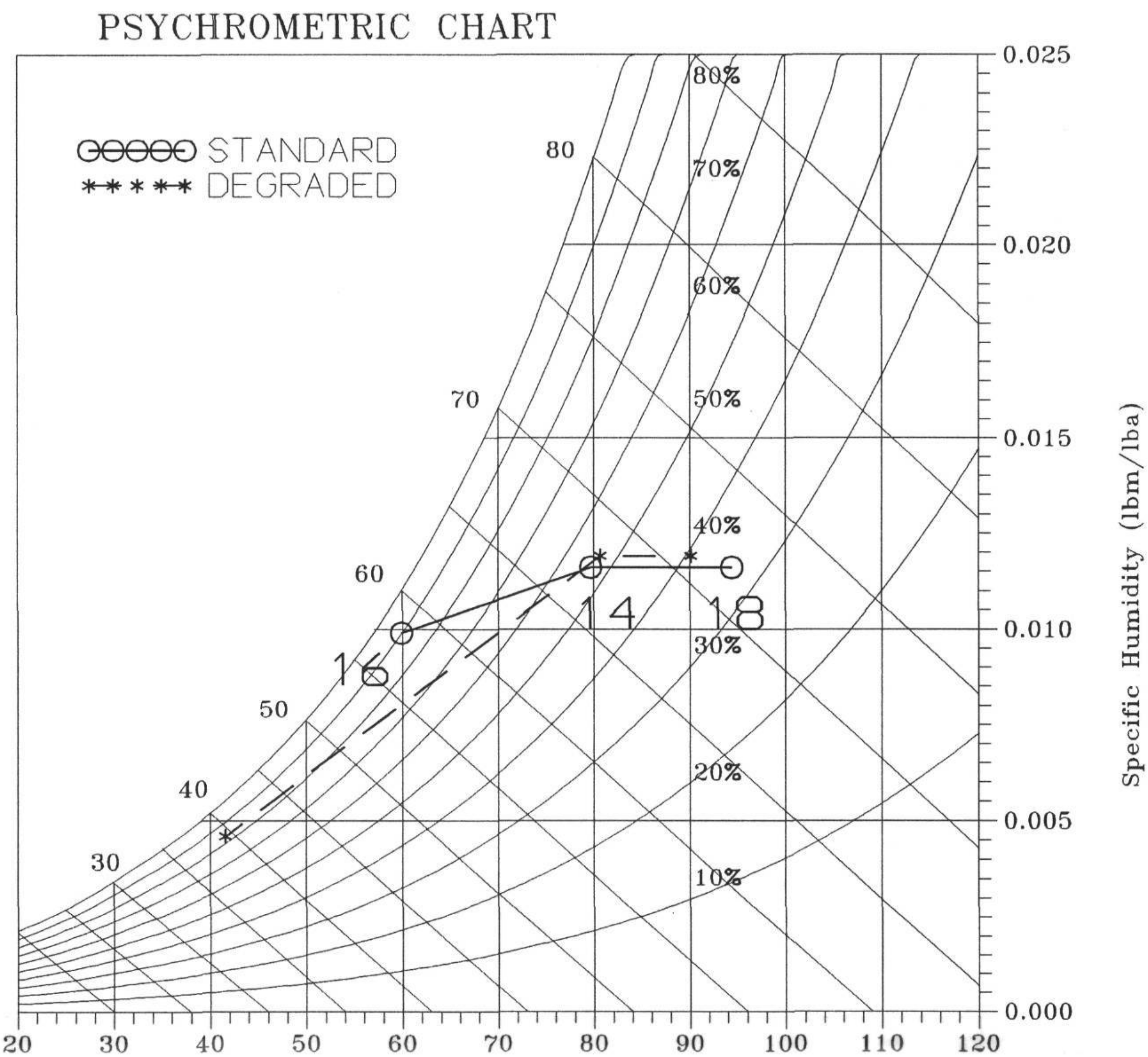


Figure B-1.3.4. Psychrometric Chart for 75% reduced evaporator air flow

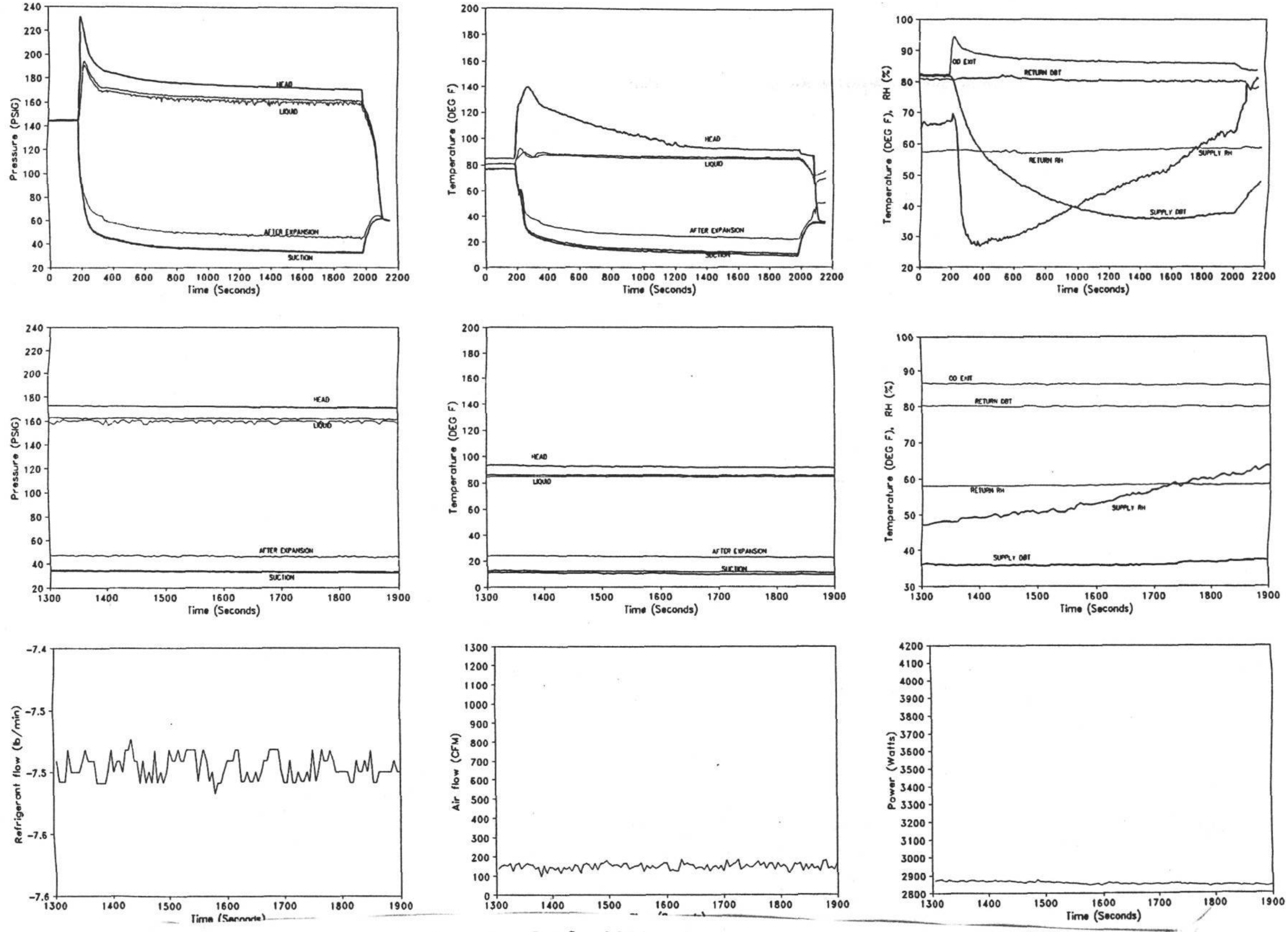


Figure B-1.4.1. Time series plot for 90% reduced evaporator air flow degraded test

STANDARD RUN			90 % CUT IN EVAP AIR		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 238	176	126.9	1. 186	92	111.8
2. 218	84	34.2	2. 177	85	34.7
3. 214	83	34.1	3. 174	84	34.4
4. 100	50	34.1	4. 61	23	34.4
5. 84	44	108.8	5. 49	12	105.6
6. 83	44	108.8	6. 48	10	105.6
REF. CAP	35.74 KBtu/hr (2.98 TONS)			16.72 KBtu/hr (1.39 TONS)	
REF. FLOW	7.97 LB/MIN			3.9 LB/MIN	

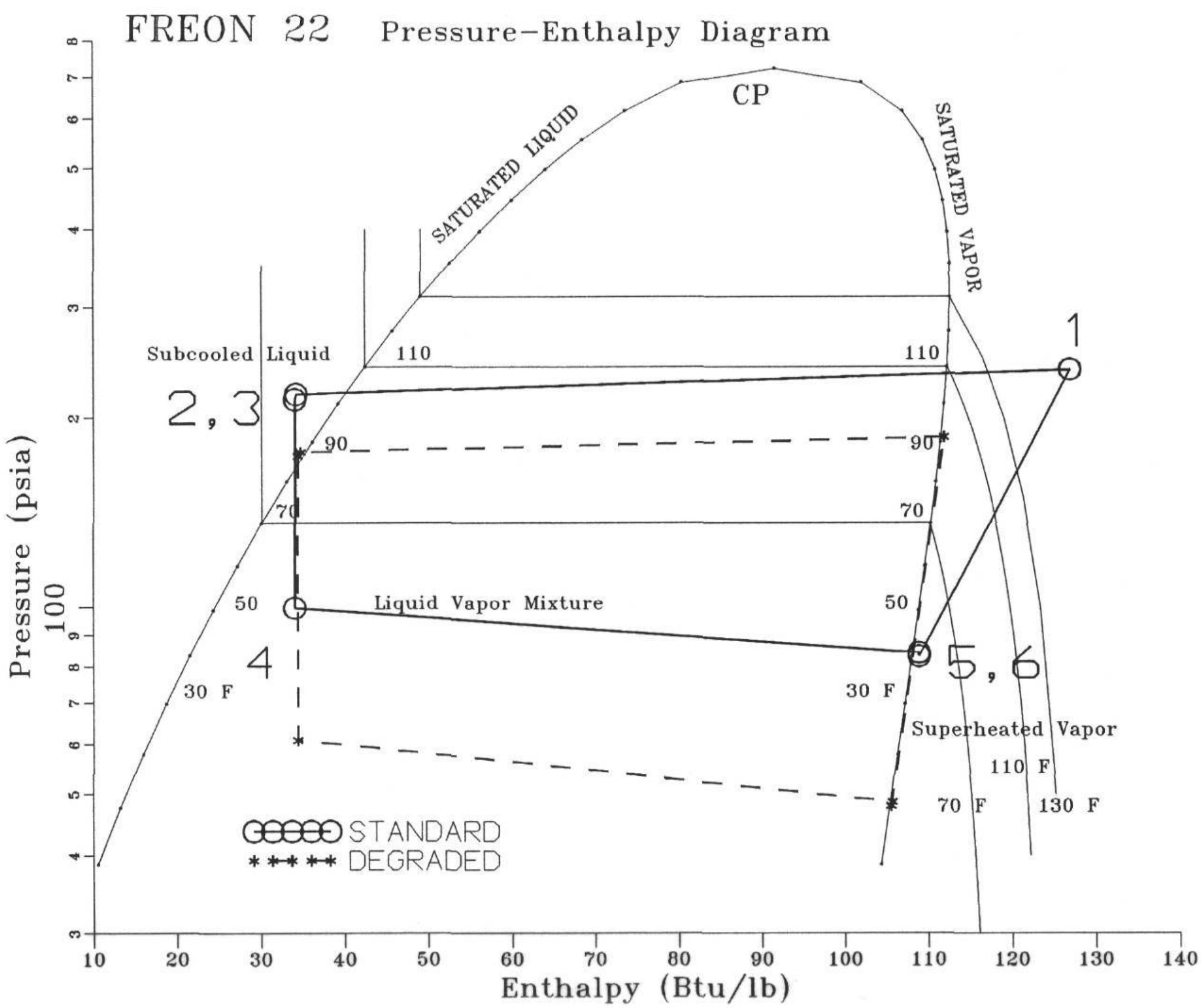


Figure B-1.4.2. P-H Chart for 90% reduced evaporator air flow

STANDARD RUN				90 % CUT IN EVAP AIR			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	79.7	53	0.0116	13.	80.0	52	0.0114
15.	59.9	90	0.0099	15.	36.1	72	0.0032
16.	94.4		0.0116	16.	86.1		0.0114
EVAP. AIR FLOW			1163		142 CFM		
COOLING CAPACITY			35.5		13.37 KBtu/hr		

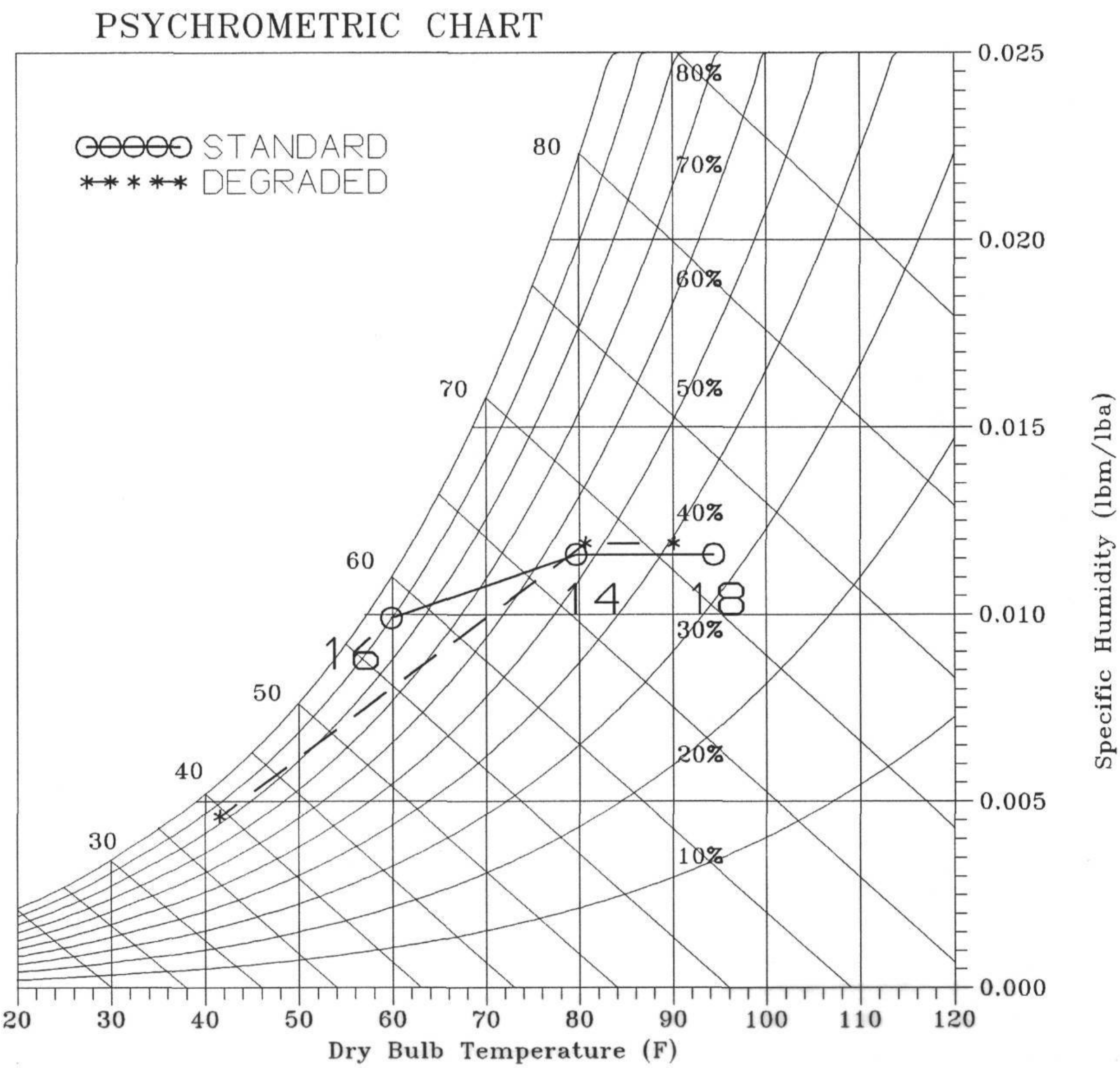


Figure B-1.4.3. Psychrometric Chart for 90% reduced evaporator air flow

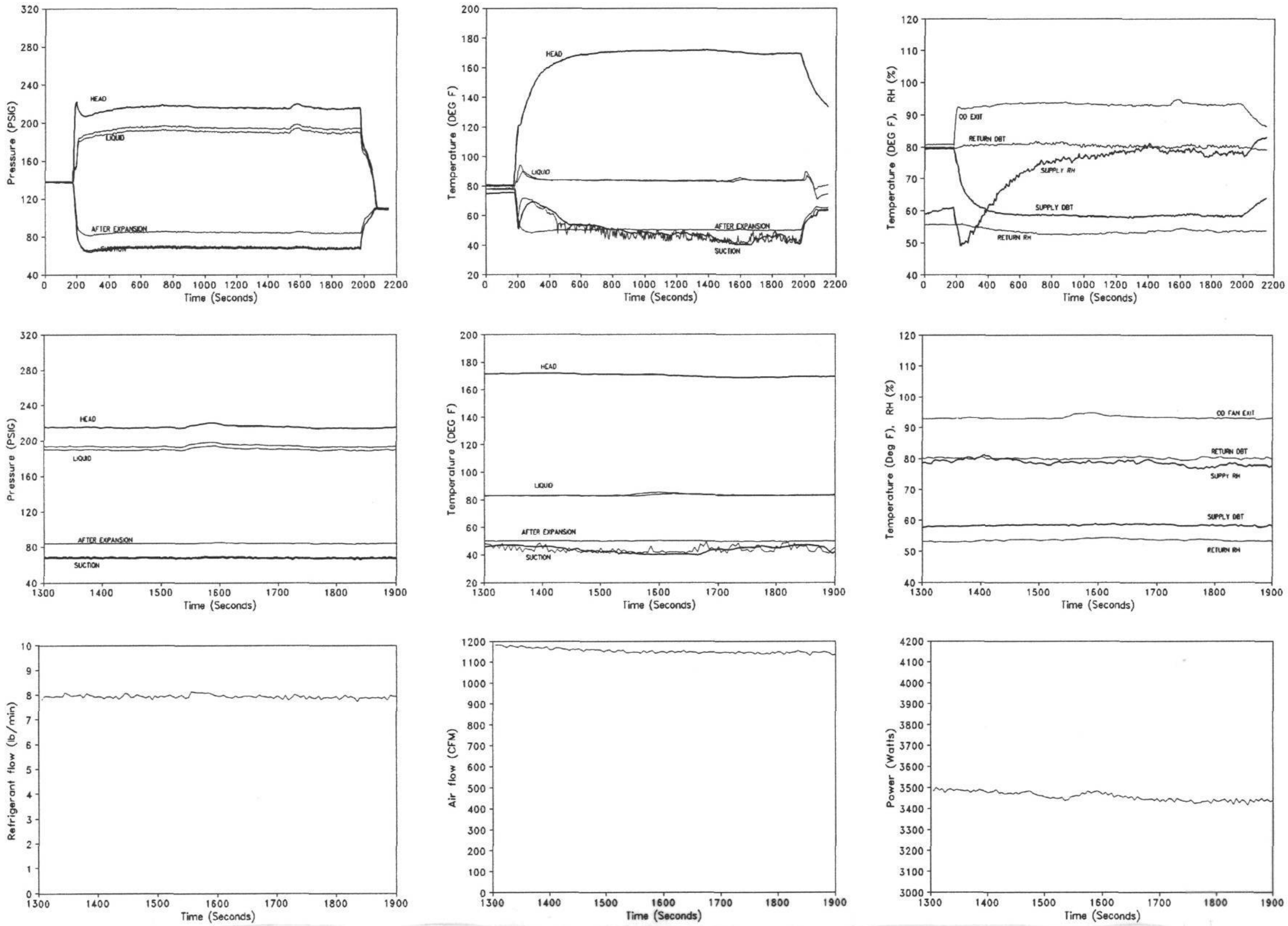


Figure B-2.1.1. Time series plot for blocked outdoor fan air flow standard test

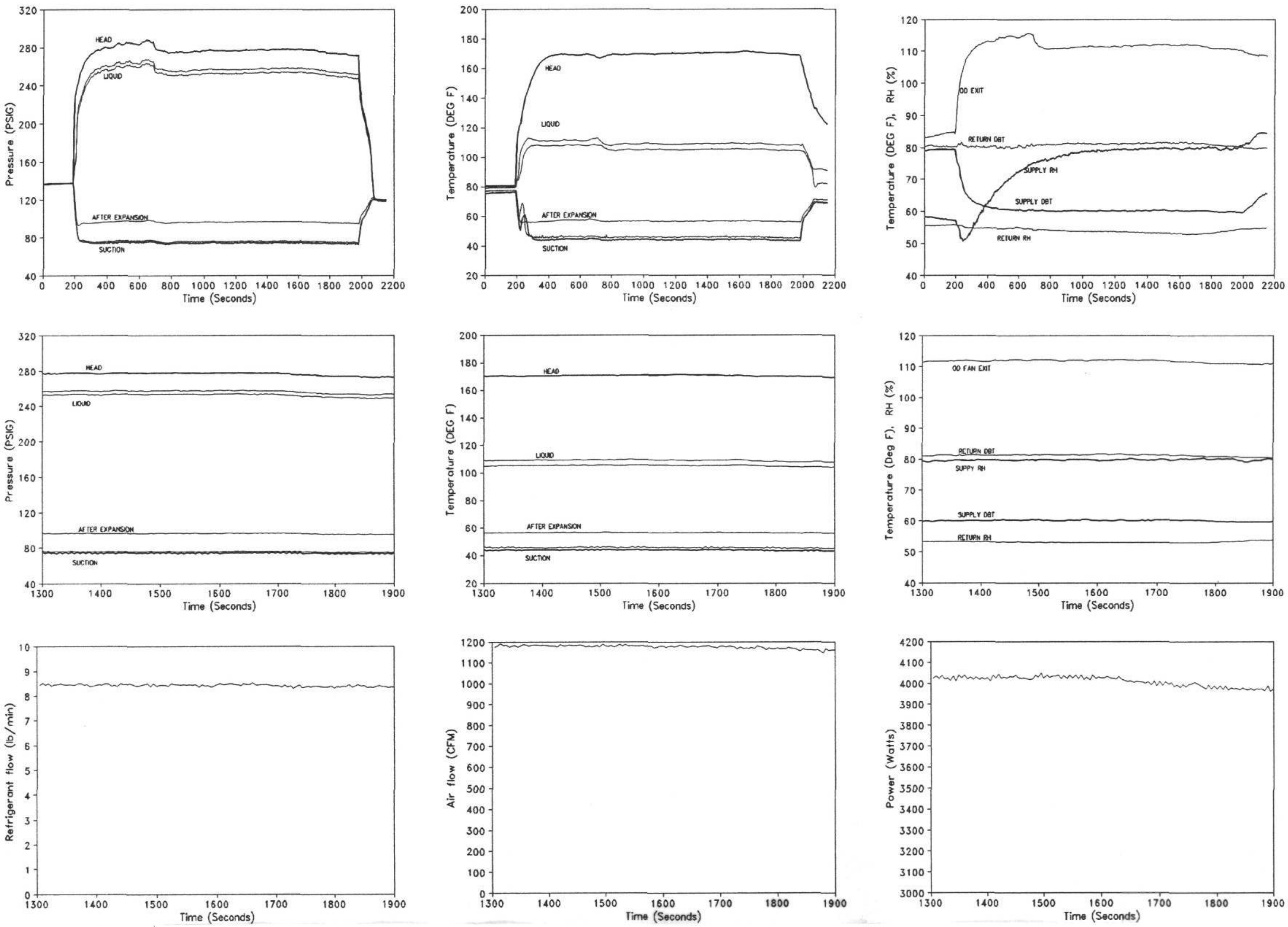


Figure B-2.1.2. Time series plot for blocked outdoor fan air flow degraded test

STANDARD RUN				OD FAN IS BLOCKED			
	P(psia)	T(F)	H(Btu/lb)		P(psia)	T(F)	H(Btu/lb)
1.	230	170	126	1.	291	170	123.7
2.	209	84	34.2	2.	271	109	42.1
3.	205	83	34.1	3.	267	105	40.9
4.	99	50	34.1	4.	111	56	40.9
5.	83	44	108.8	5.	90	46	108.7
6.	82	44	108.9	6.	89	44	108.5
REF. CAP		35.6 KBtu/hr (2.97 TONS)				34.25 KBtu/hr (2.85 TONS)	
REF. FLOW		7.94 LB/MIN				8.42 LB/MIN	

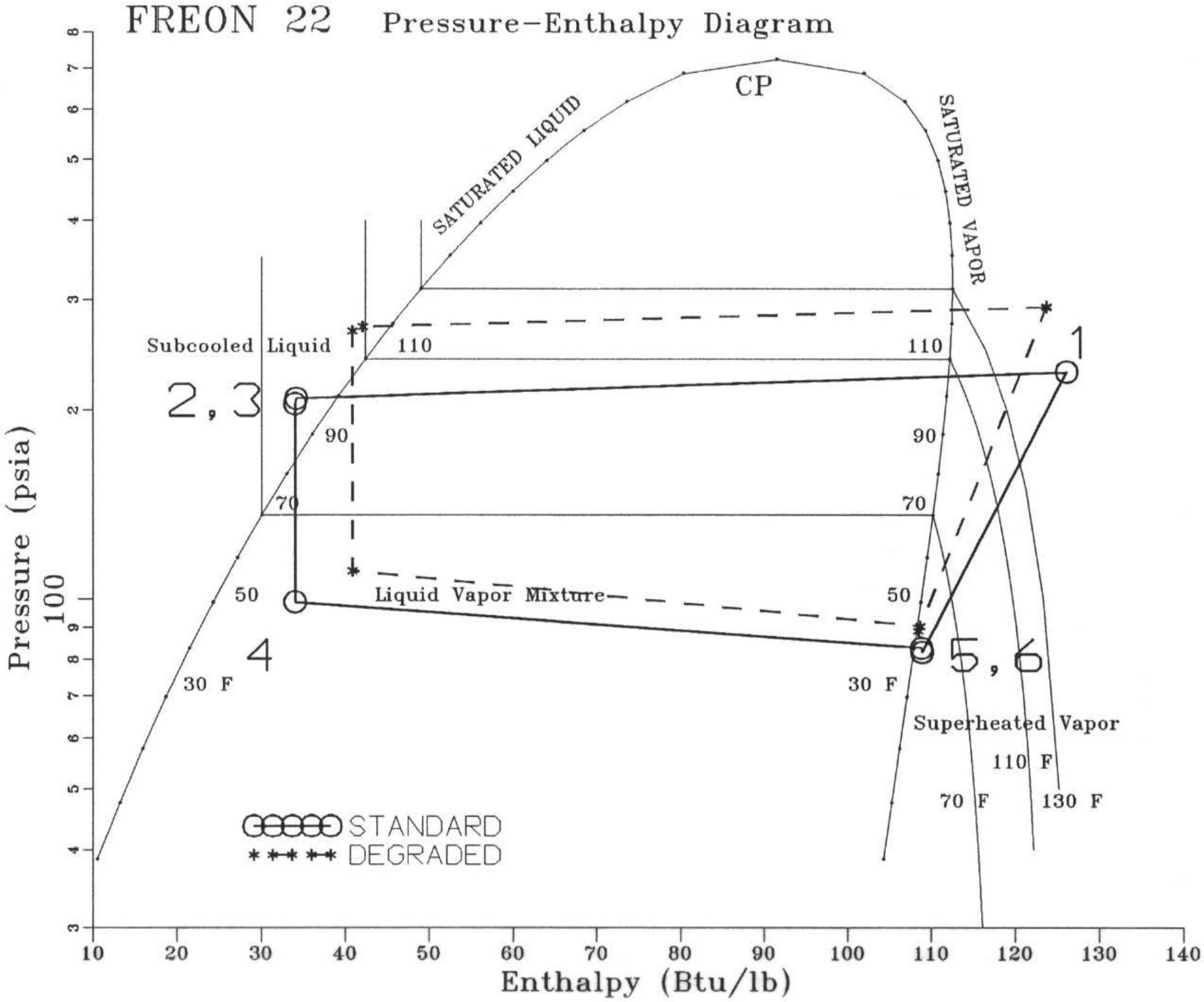


Figure B-2.1.3. P-H chart for blocked outdoor fan air flow

STANDARD RUN				OD FAN IS BLOCKED			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	48	0.0104	13.	81.2	47	0.0107
15.	58.4	88	0.0091	15.	60.1	89	0.0098
16.	93.2		0.0104	16.	111.7		0.0107

EVAP. AIR FLOW 1135 | 1160 CFM

COOLING CAPACITY 34.9 | 32.37 KBtu/hr

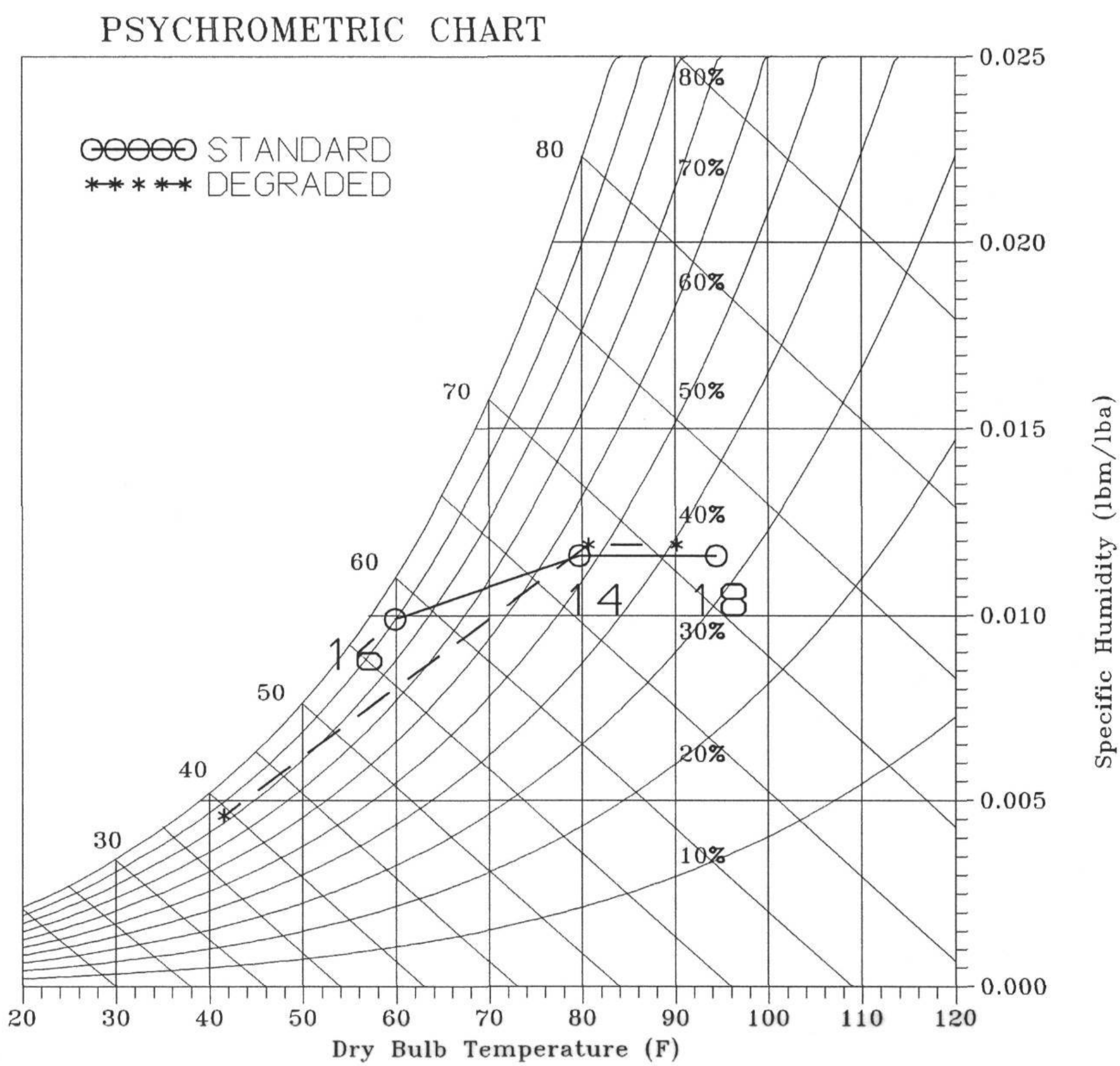


Figure B-2.1.4. Psychrometric chart for blocked outdoor fan air flow

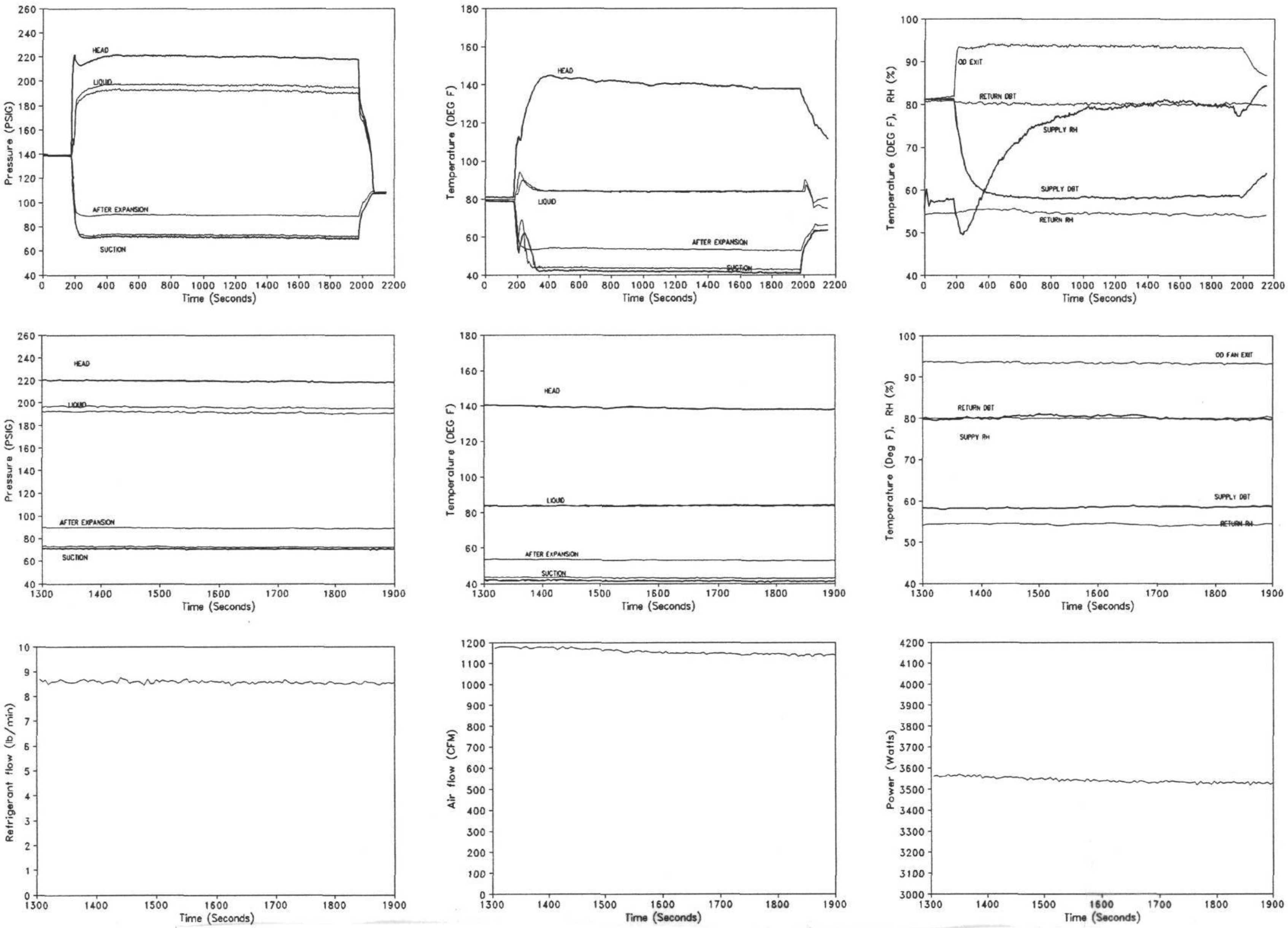


Figure B-3.1.1 Time series plot for non-condensable gases standard test

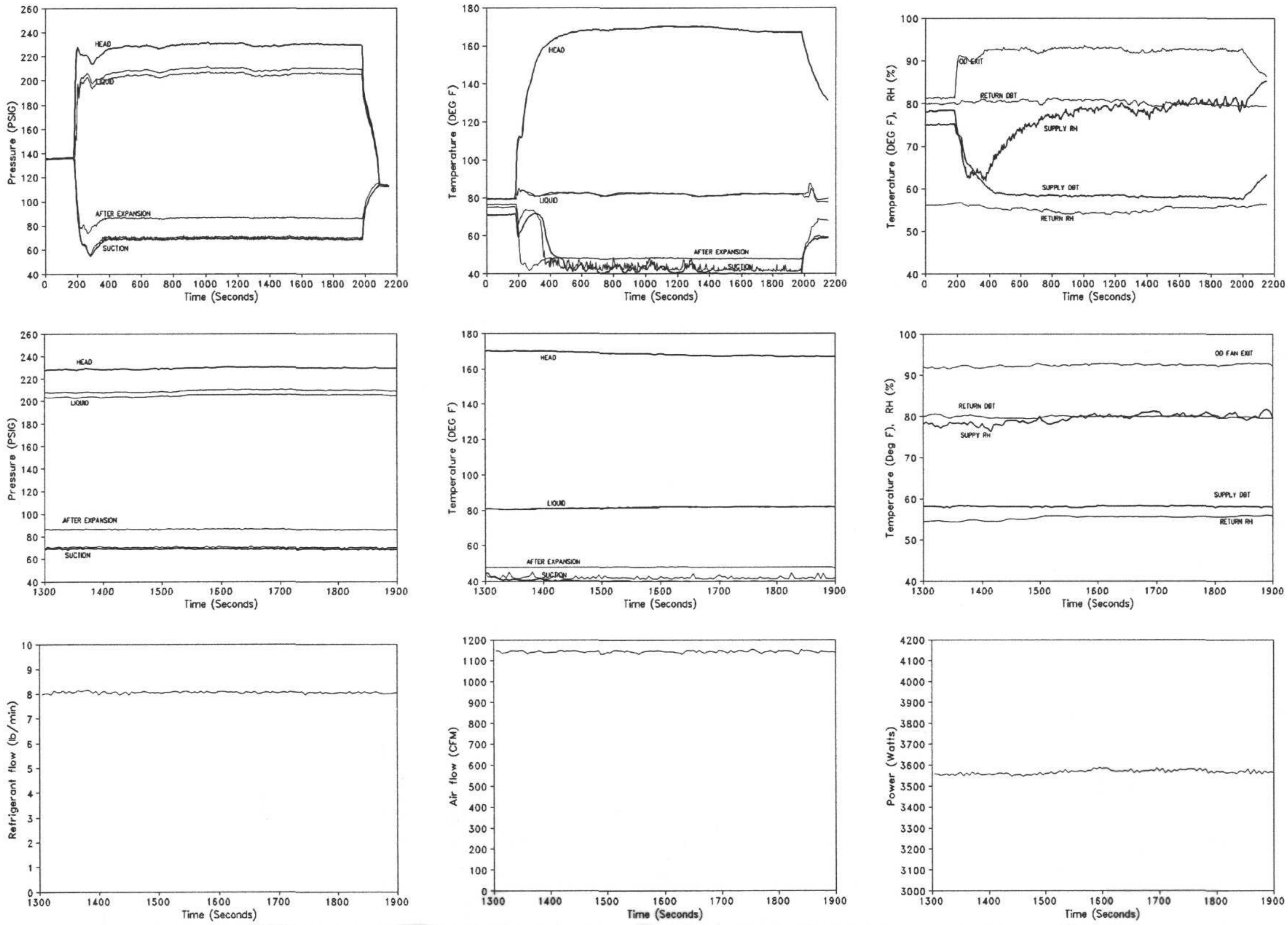


Figure B-3.1.2 Time series plot for non-condensable gases degraded test

STANDARD RUN			AIR IN SYSTEM		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 234	139	119.9	1. 244	168	125.1
2. 211	84	34.3	2. 224	82	33.6
3. 206	84	34.2	3. 220	81	33.5
4. 104	53	34.2	4. 101	48	33.5
5. 87	43	108.5	5. 85	42	108.4
6. 85	41	108.4	6. 84	40	108.2
REF. CAP	38.2 KBtu/hr (3.18 TONS)			36.16 KBtu/hr (3.01 TONS)	
REF. FLOW	8.57 LB/MIN			8.05 LB/MIN	

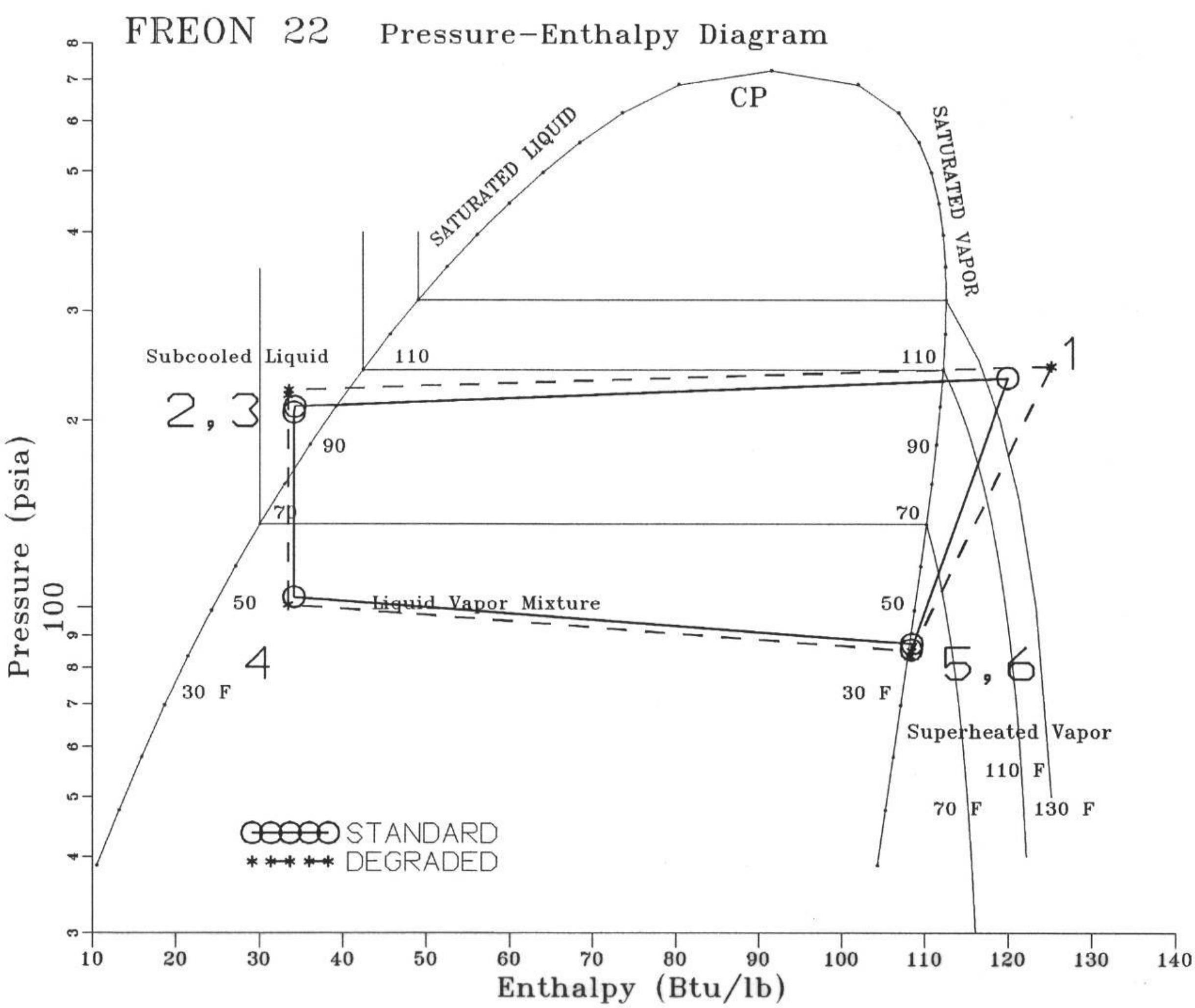


Figure B-3.1.3 P-H chart for non-condensable gases

	STANDARD RUN			AIR IN SYSTEM		
	DBT(F)	RH(%)	W(lbm/lb air)	DBT(F)	RH(%)	W(lbm/lb air)
13.	80.0	48	0.0105	79.9	49	0.0107
15.	58.4	89	0.0092	58.0	89	0.0092
16.	93.4		0.0105	92.4		0.0107
EVAP. AIR FLOW	1138			1125 CFM		
COOLING CAPACITY	34.9			36.05 KBtu/hr		

PSYCHROMETRIC CHART

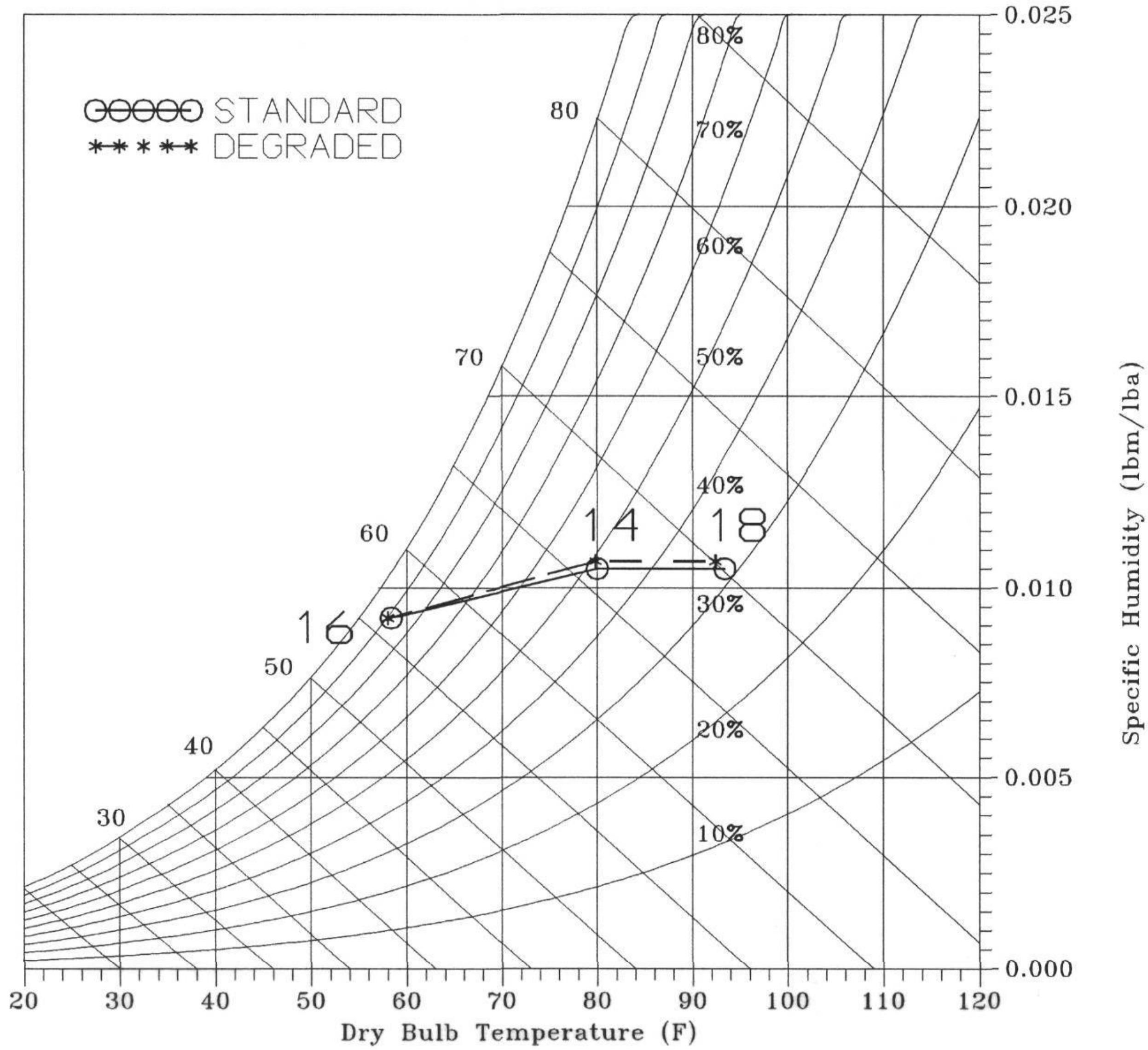


Figure B-3.1.4 Psychrometric chart for non-condensable gases

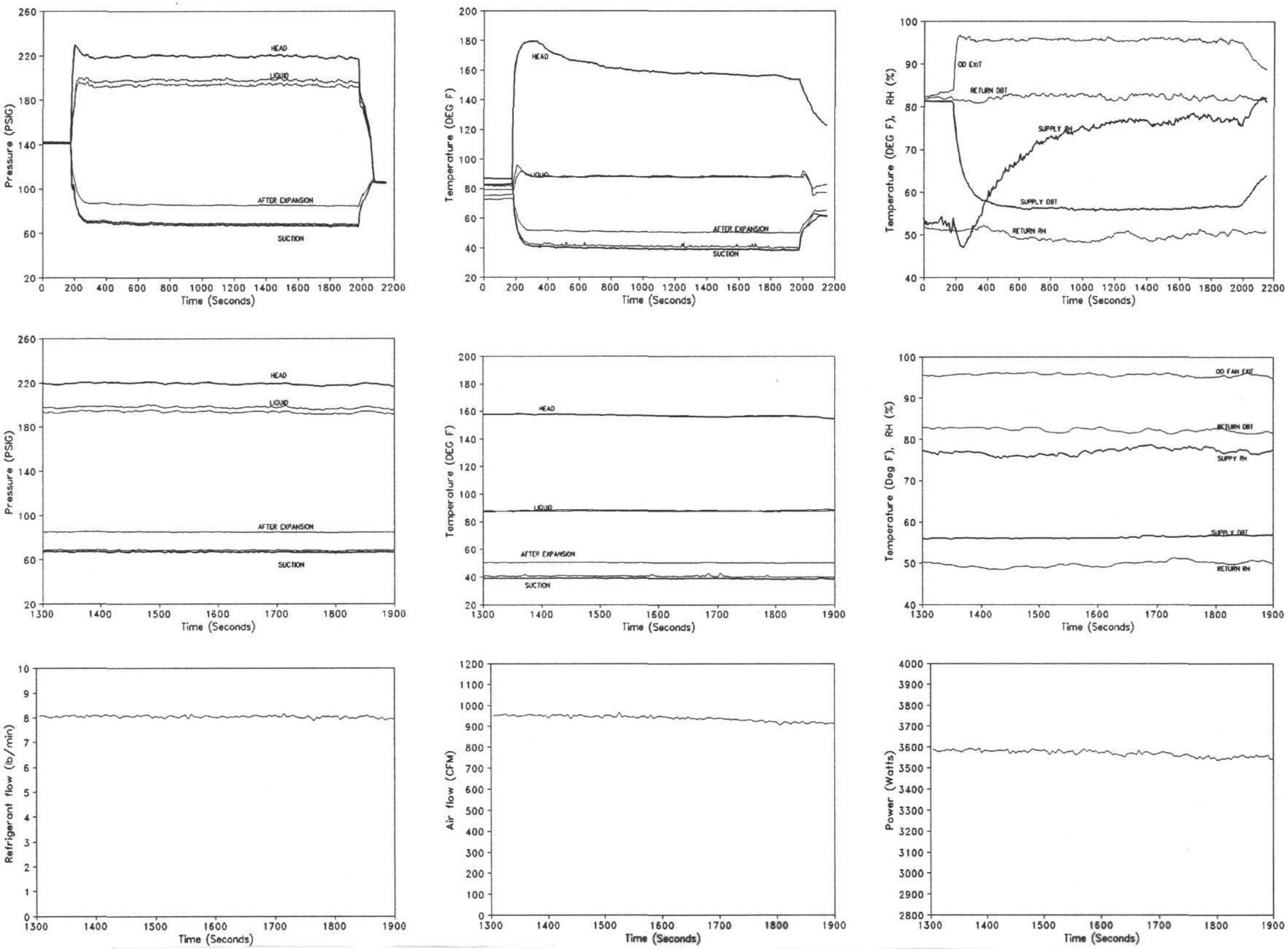


Figure B-4.1.1. Time series plot for restriction before expansion valve standard test

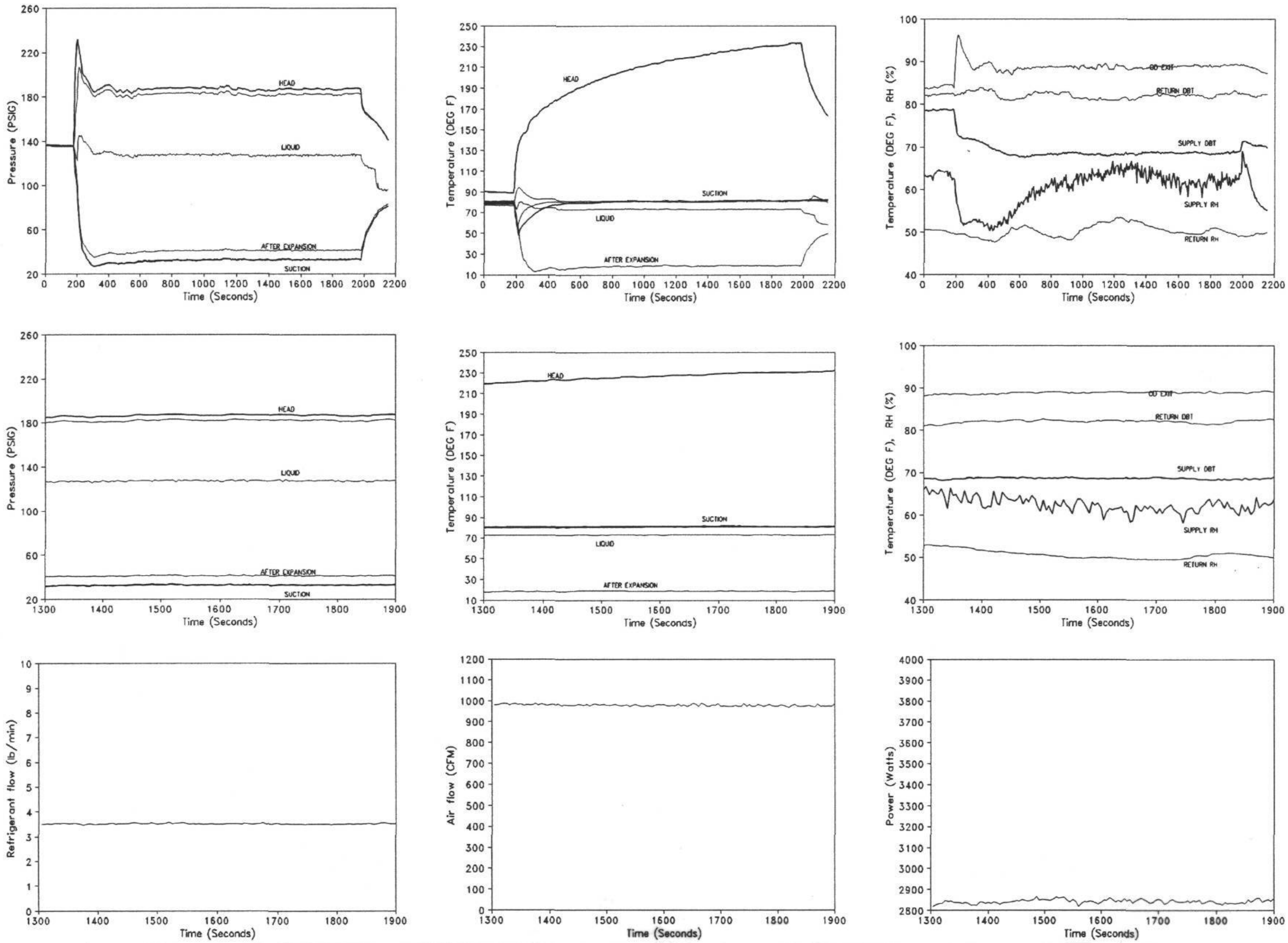


Figure B-4.1.2. Time series plot for restriction before expansion valve degraded test

STANDARD RUN			RESTRICTION IN REFRIGERANT LINE BEFORE EX. VALVE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 234	157	123.1	1. 201	227	138.3
2. 212	88	35.6	2. 196	82	33.6
3. 208	88	35.4	3. 142	73	31.1
4. 100	50	35.4	4. 56	19	31.1
5. 83	41	108.3	5. 48	81	117.0
6. 82	39	108.1	6. 47	81	117.0
REF. CAP	35.14 KBtu/hr (2.93 TONS)			18.15 KBtu/hr (1.51 TONS)	
REF. FLOW	8.04 LB/MIN			3.52 LB/MIN	

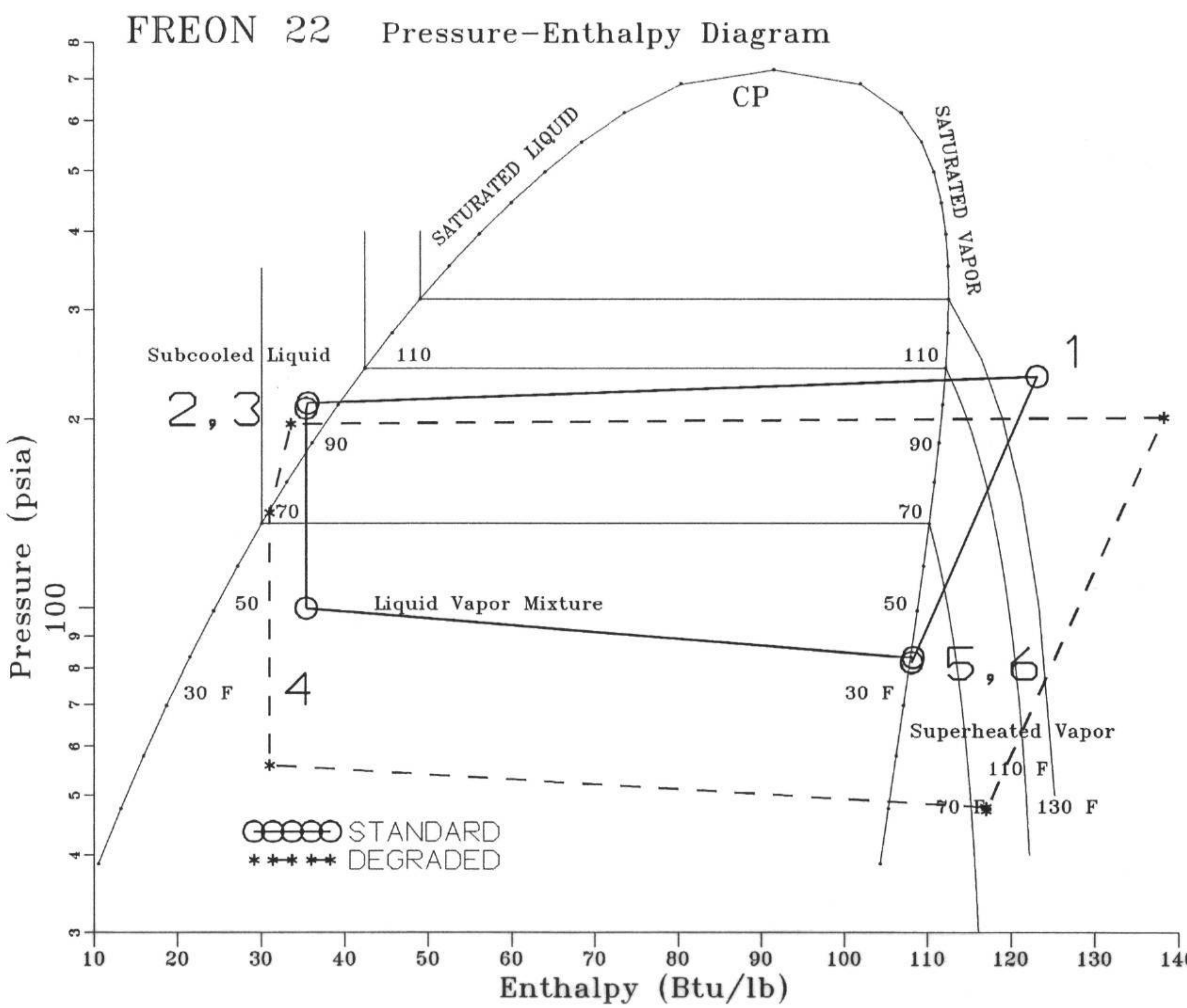


Figure B-4.1.3. P-H chart for restriction before expansion valve

STANDARD RUN	RESTRICTION IN REF LINE BEFORE EX. VALVE
DBT(F) RH(%) W(lbm/lb air)	DBT(F) RH(%) W(lbm/lb air)
13. 82.3 43 0.0102	13. 81.9 44 0.0103
15. 56.3 86 0.0083	15. 68.6 71 0.0106
16. 95.6 0.0102	16. 88.9 0.0103
EVAP. AIR FLOW 922	975 CFM
COOLING CAPACITY 35.6	12.45 KBtu/hr

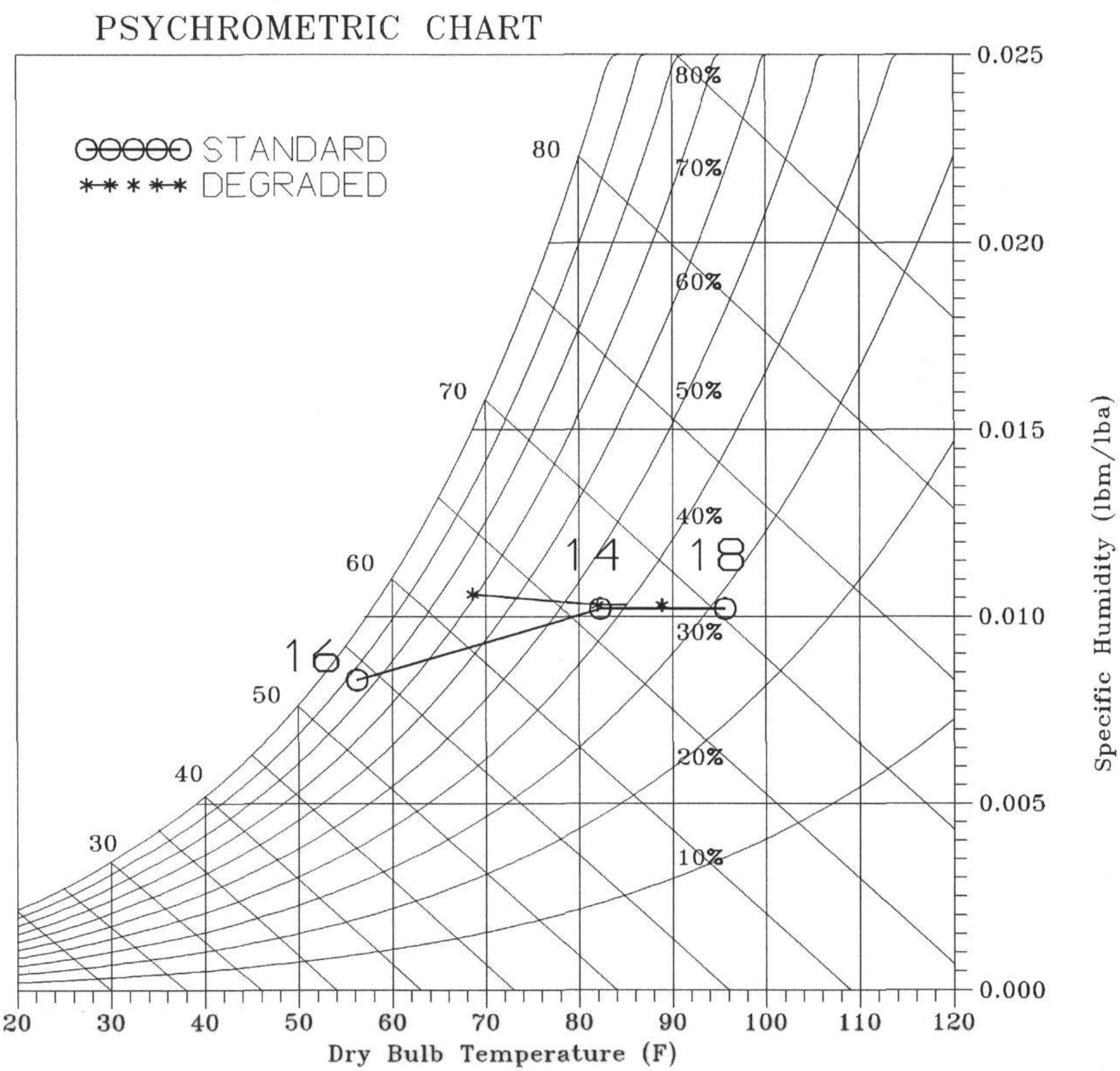


Figure B-4.1.4. Psychrometric chart for restriction before expansion valve

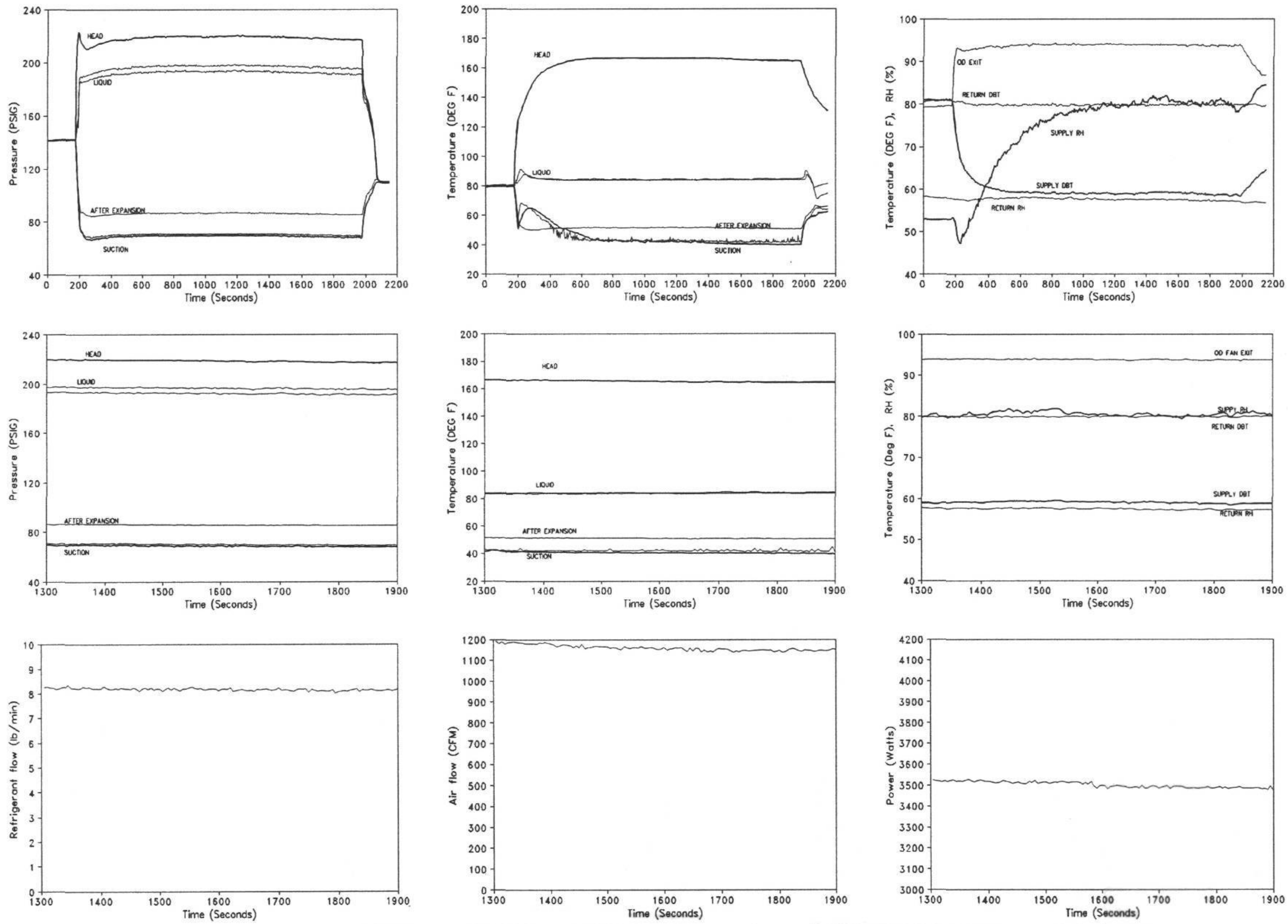


Figure B-4.2.1. Time series plot for restriction after expansion valve standard test

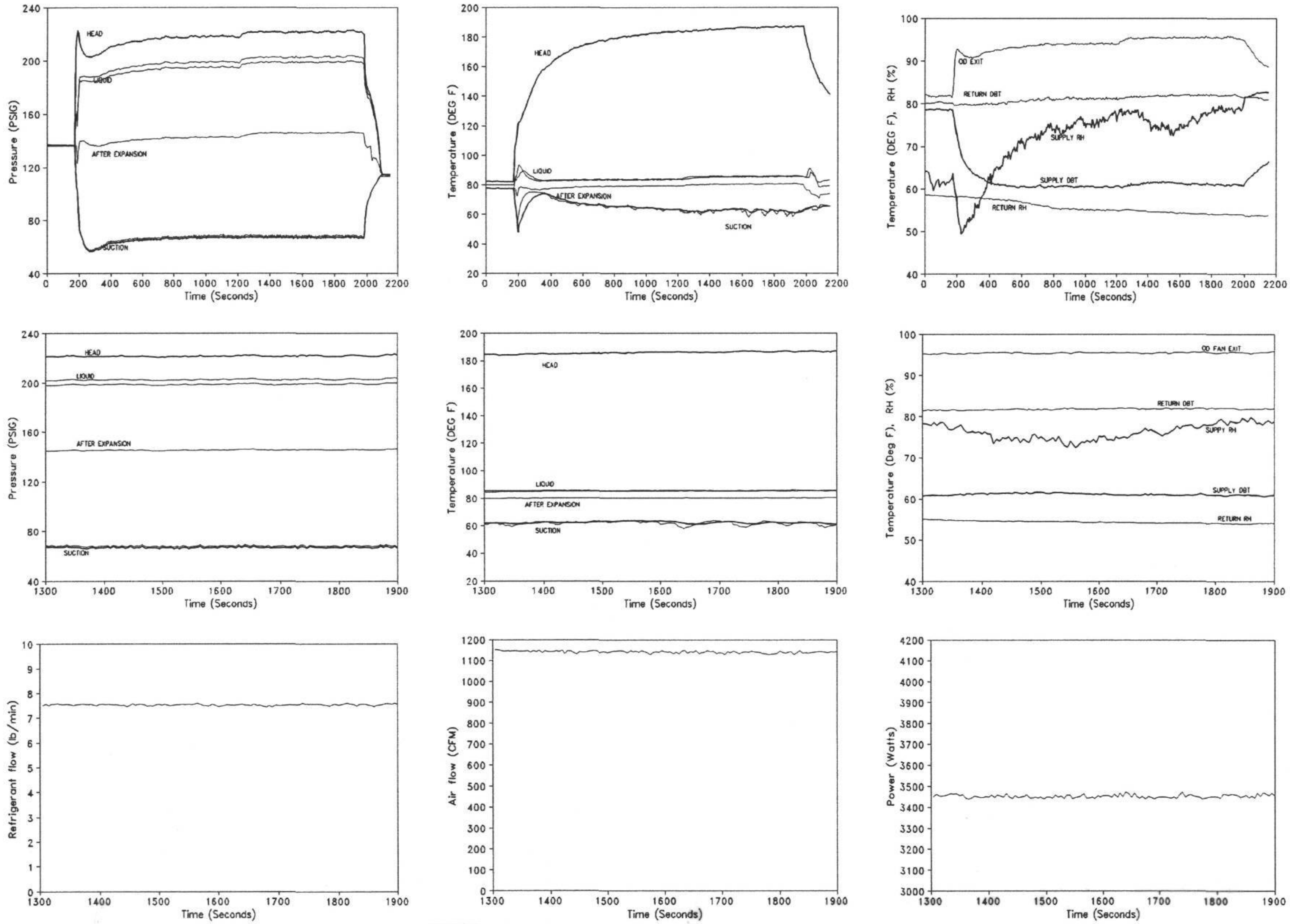


Figure B-4.2.2. Time series plot for restriction after expansion valve degraded test

STANDARD RUN				RESTRICTION IN REFRIGERANT LINE AFTER EX. VALVE		
	P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1.	233	165	125.0	236	186	129.0
2.	211	84	34.4	217	86	34.9
3.	207	84	34.2	213	85	34.6
4.	101	51	34.2	160	80	34.6
5.	85	42	108.4	83	62	112.0
6.	83	40	108.2	82	62	112.2
REF. CAP			36.36 KBtu/hr (3.03 TONS)			35.01 KBtu/hr (2.92 TONS)
REF. FLOW			8.17 LB/MIN			7.54 LB/MIN

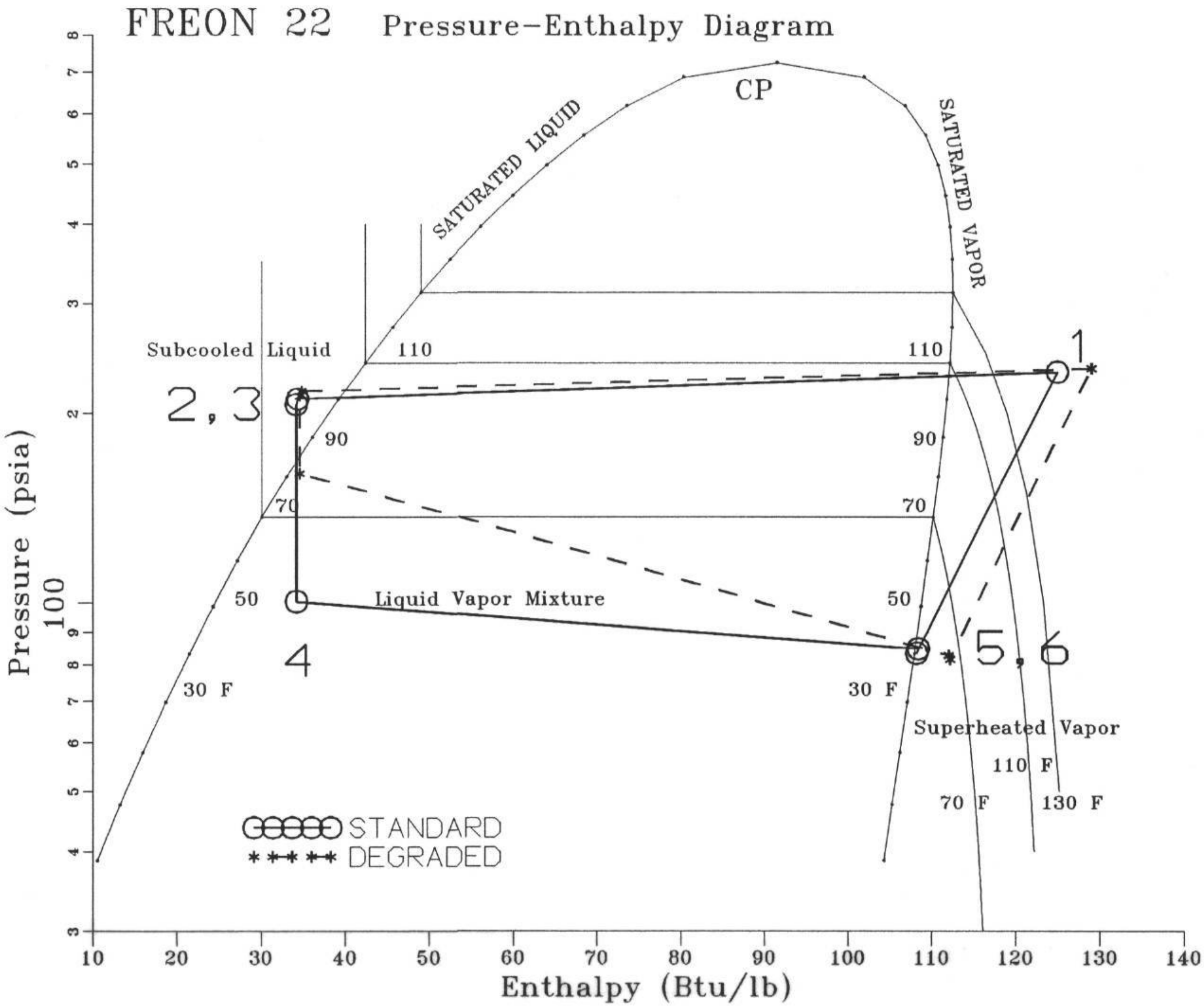


Figure B-4.2.3. P-H chart for restriction after expansion valve

STANDARD RUN				RESTRICTION IN REF LINE AFTER EX. VALVE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	79.8	51	0.0112	13.	81.8	48	0.0112
15.	59.0	90	0.0095	15.	61.1	85	0.0098
16.	93.8		0.0112	16.	95.4		0.0112
EVAP. AIR FLOW			1142	1127 CFM			
COOLING CAPACITY			36.1	33.95 KBtu/hr			

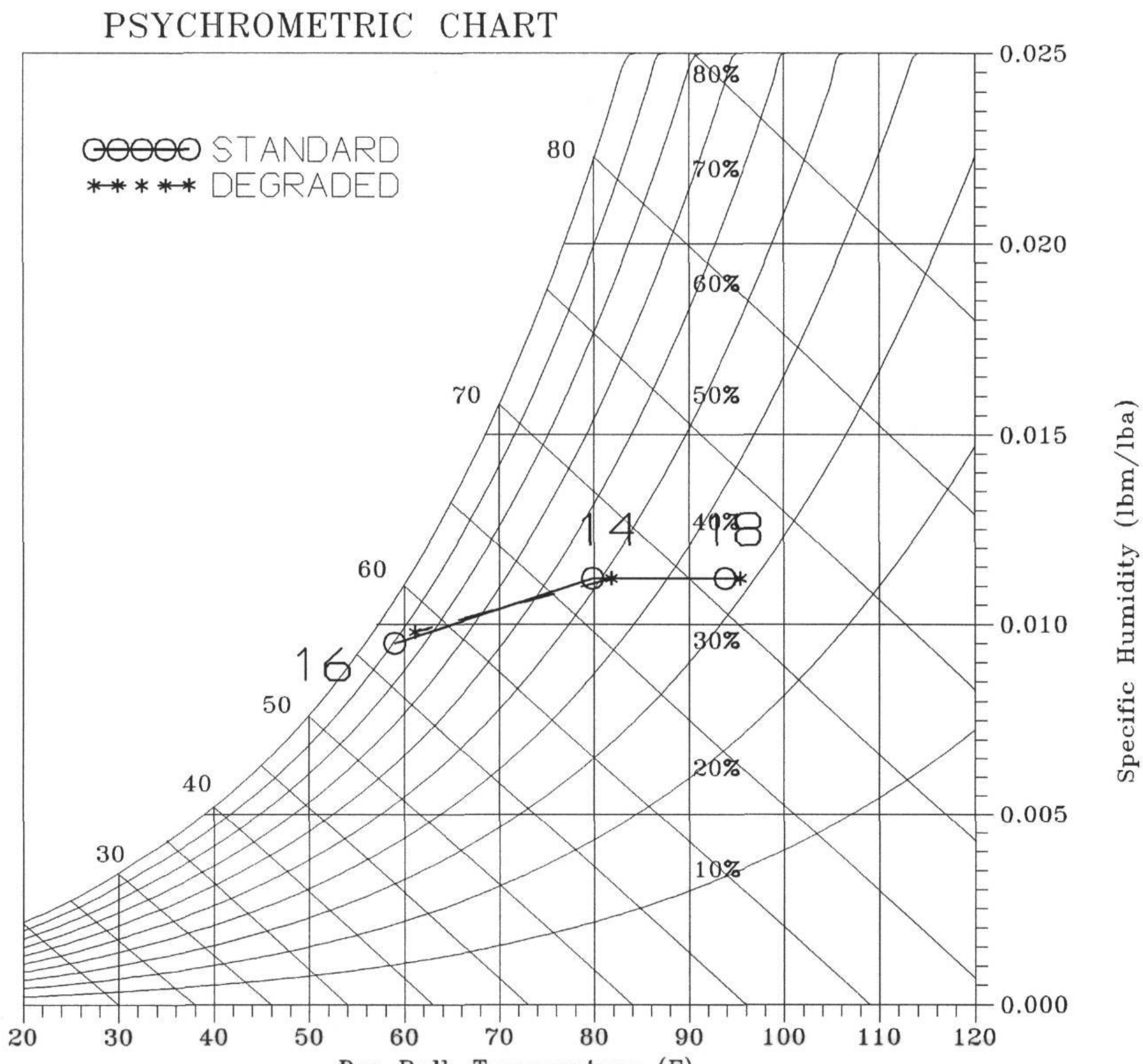


Figure B-4.2.4. Psychrometric chart for restriction after expansion valve

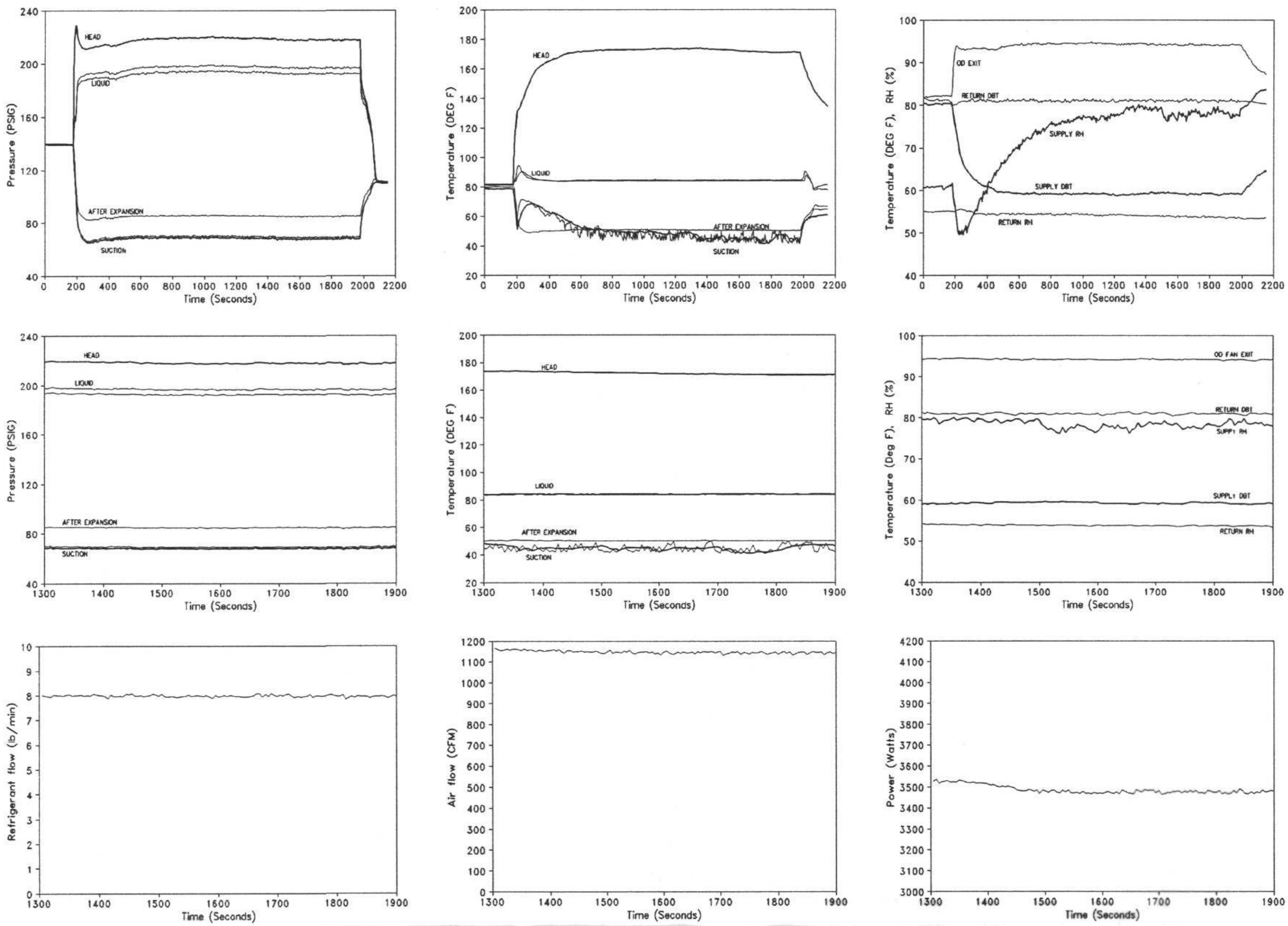


Figure B-4.3.1. Time series plot for restriction in suction line standard test

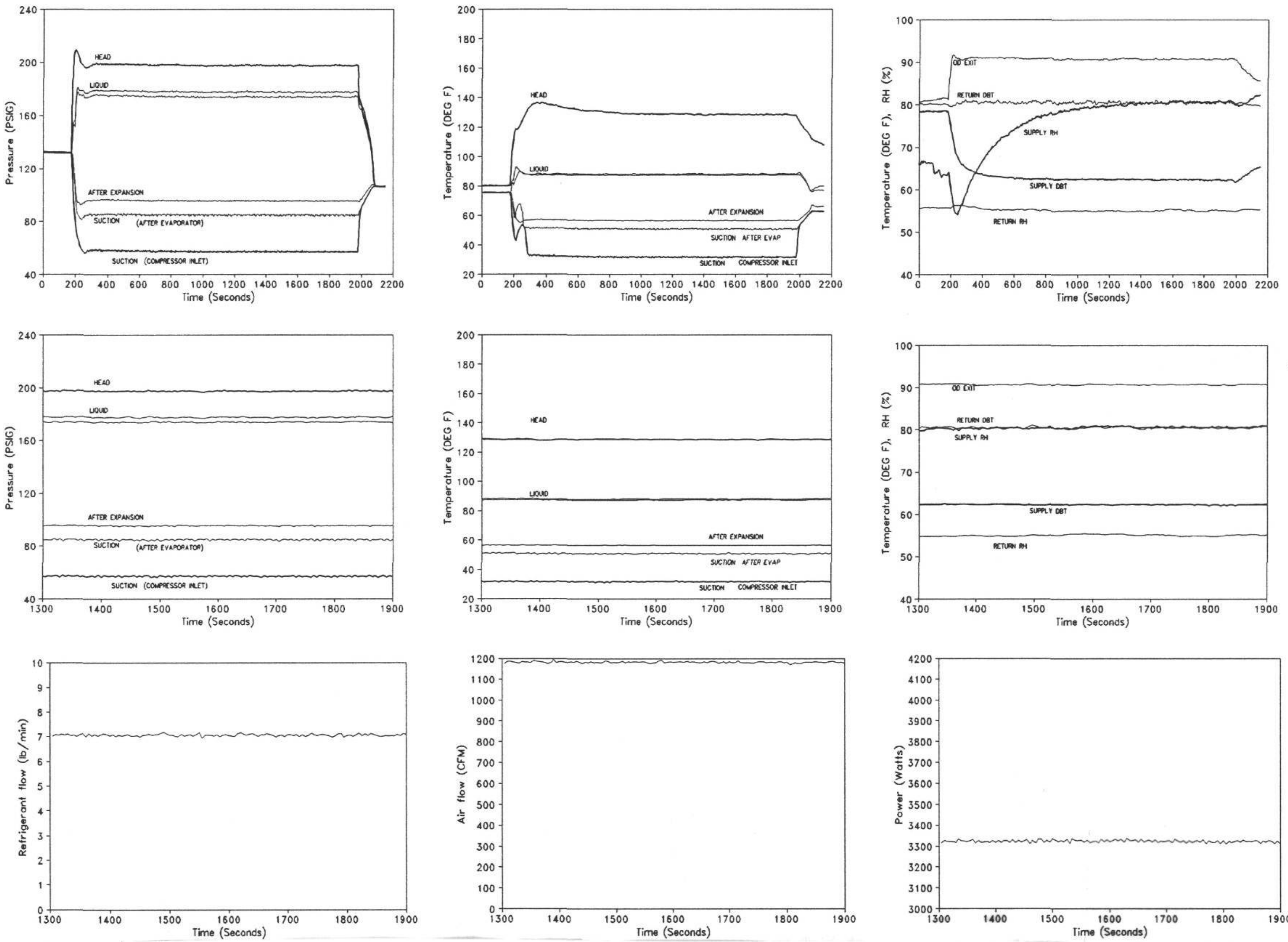


Figure B-4.3.2. Time series plot for restriction in suction line degraded test

STANDARD RUN			RESTRICTION IN SUCTION LINE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 233	172	126.4	1. 212	128	118.2
2. 211	84	34.4	2. 192	88	35.6
3. 207	84	34.3	3. 189	87	35.3
4. 99	50	34.3	4. 110	56	35.3
5. 84	45	108.9	5. 99	51	109.0
6. 83	45	109.0	6. 72	32	109.2
REF. CAP		35.7 KBtu/hr (2.98 TONS)			31.33 KBtu/hr (2.61 TONS)
REF. FLOW		7.98 LB/MIN			7.08 LB/MIN

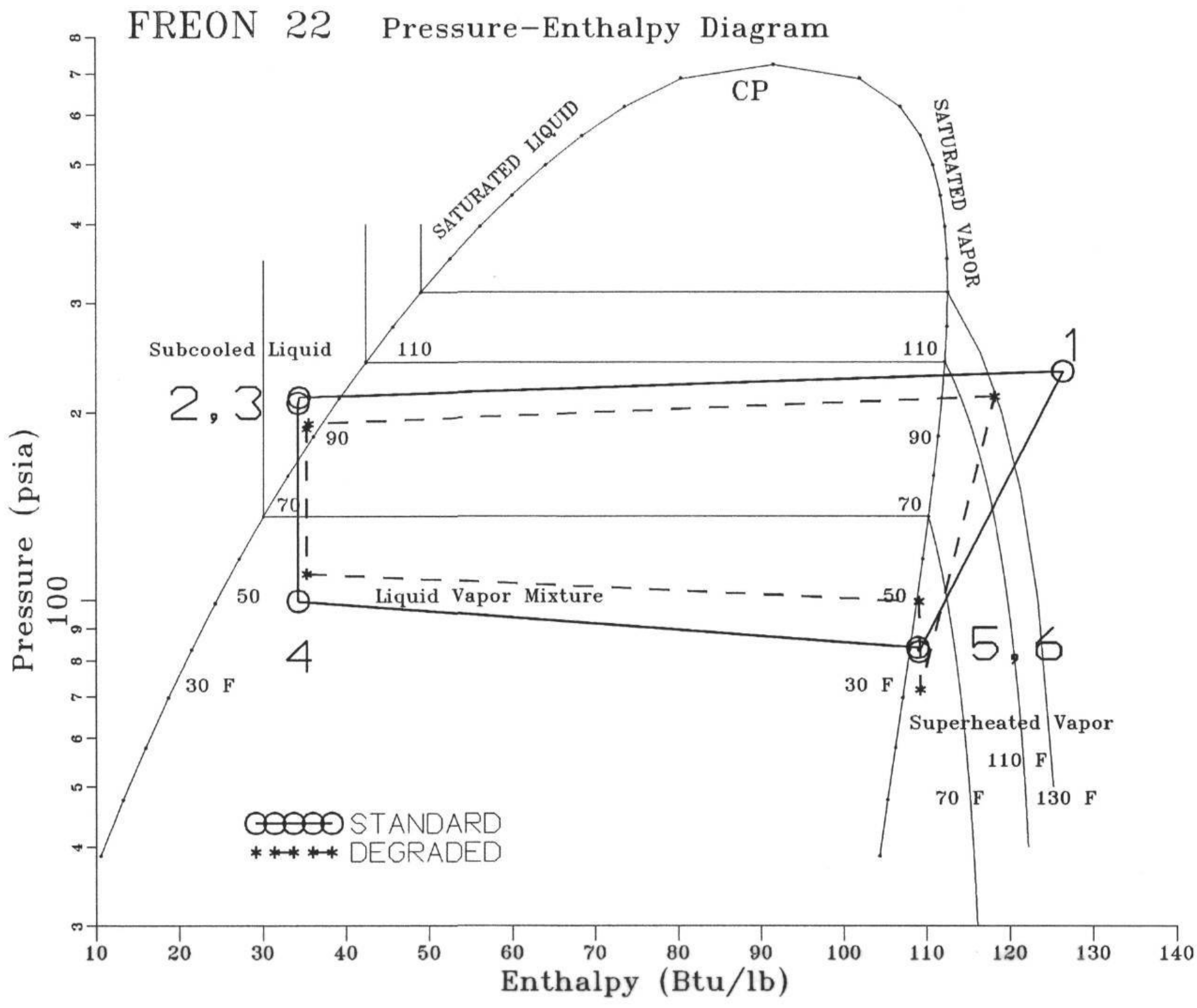


Figure B-4.3.3. P-H chart for restriction in suction line

	STANDARD RUN			RESTRICTION IN SUCTION LINE		
	DBT(F)	RH(%)	W(lbm/lb air)	DBT(F)	RH(%)	W(lbm/lb air)
13.	80.9	48	0.0108	80.5	49	0.0109
15.	59.0	87	0.0093	62.3	90	0.0107
16.	94.2		0.0108	90.6		0.0109
EVAP. AIR FLOW	1130			1170 CFM		
COOLING CAPACITY	35.4			24.66 KBtu/hr		

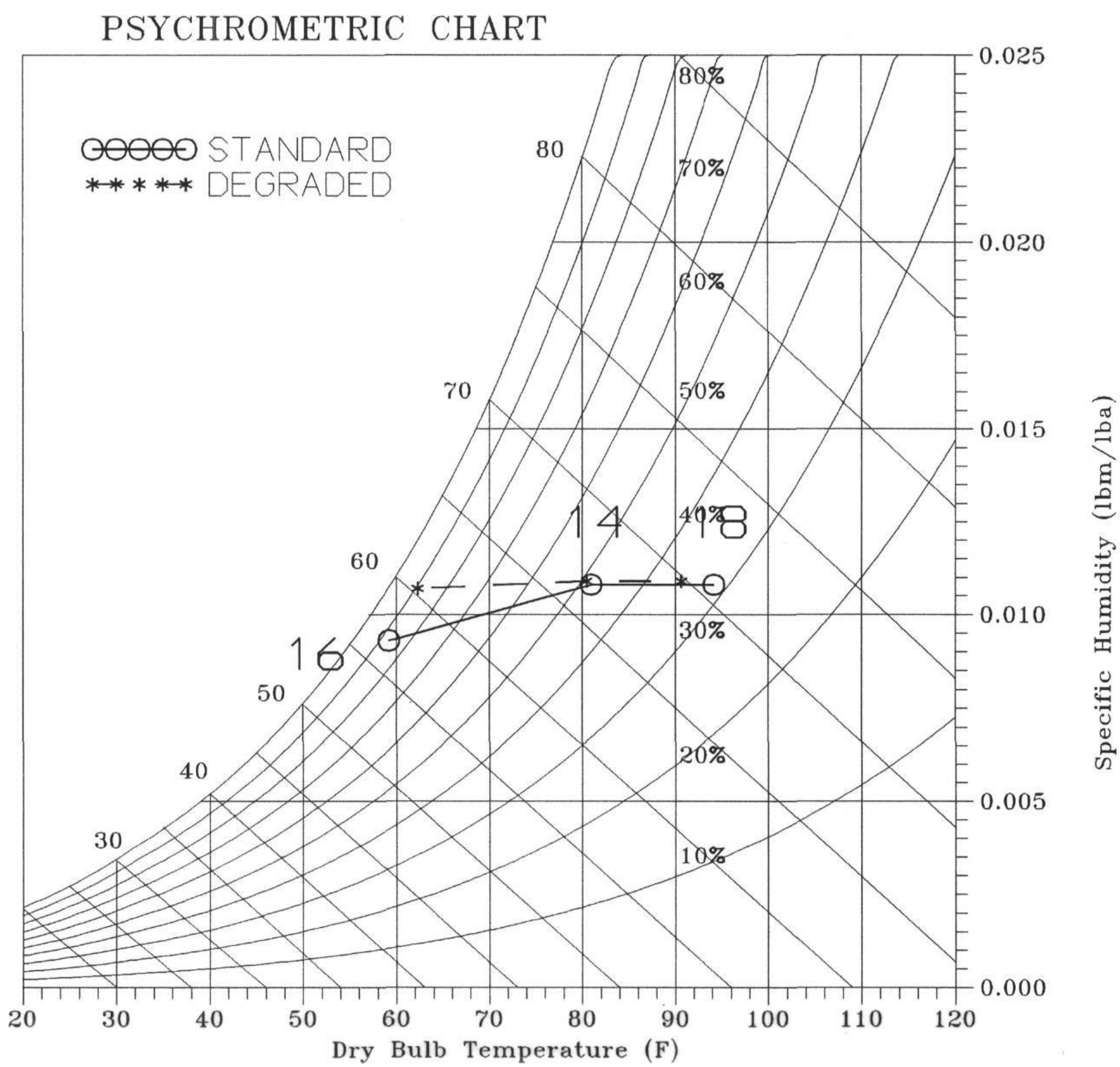


Figure B-4.3.4. Psychrometric chart for restriction in suction line

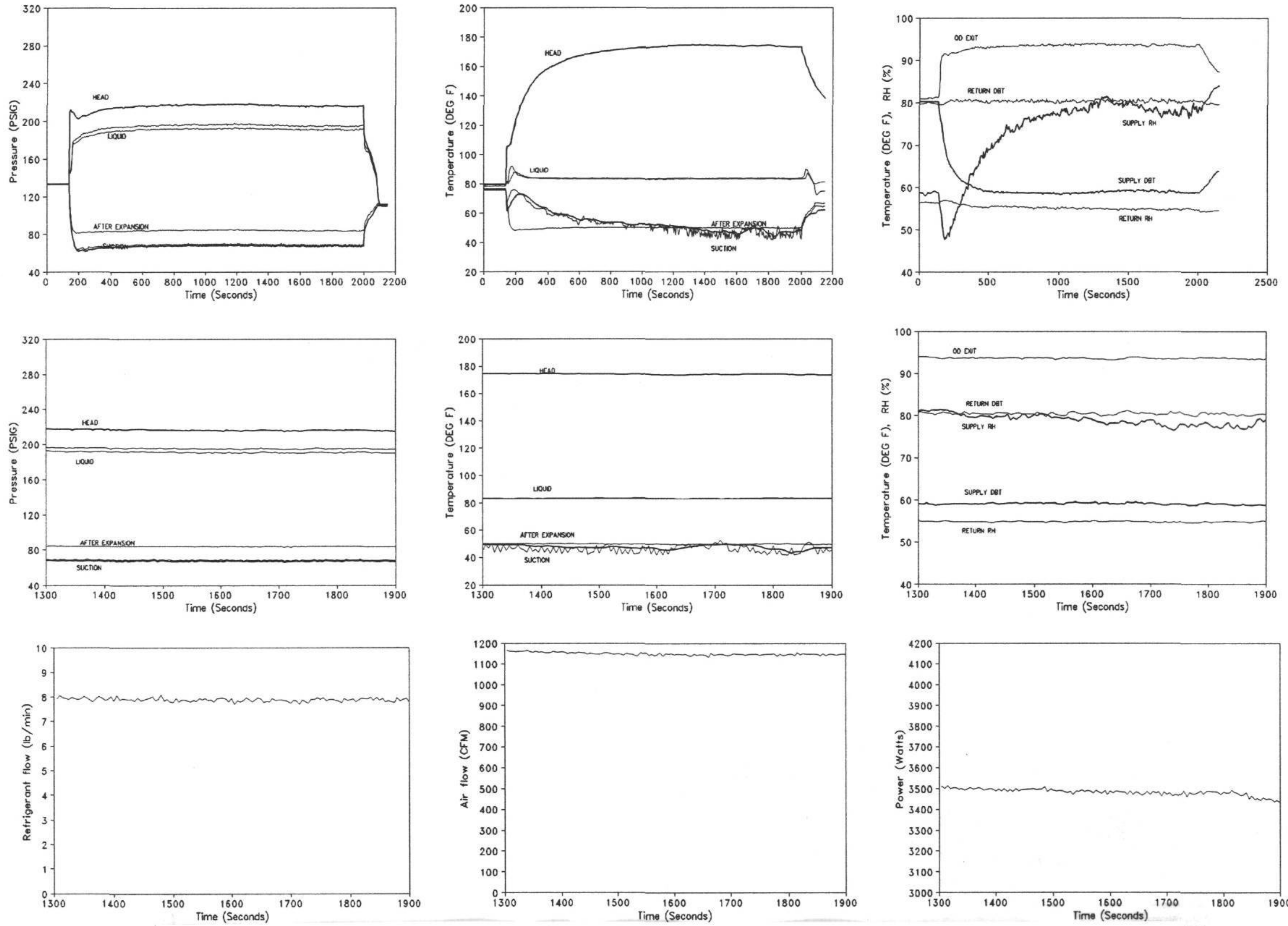


Figure B-4.4.1. Time series plot for restriction in discharge line standard test

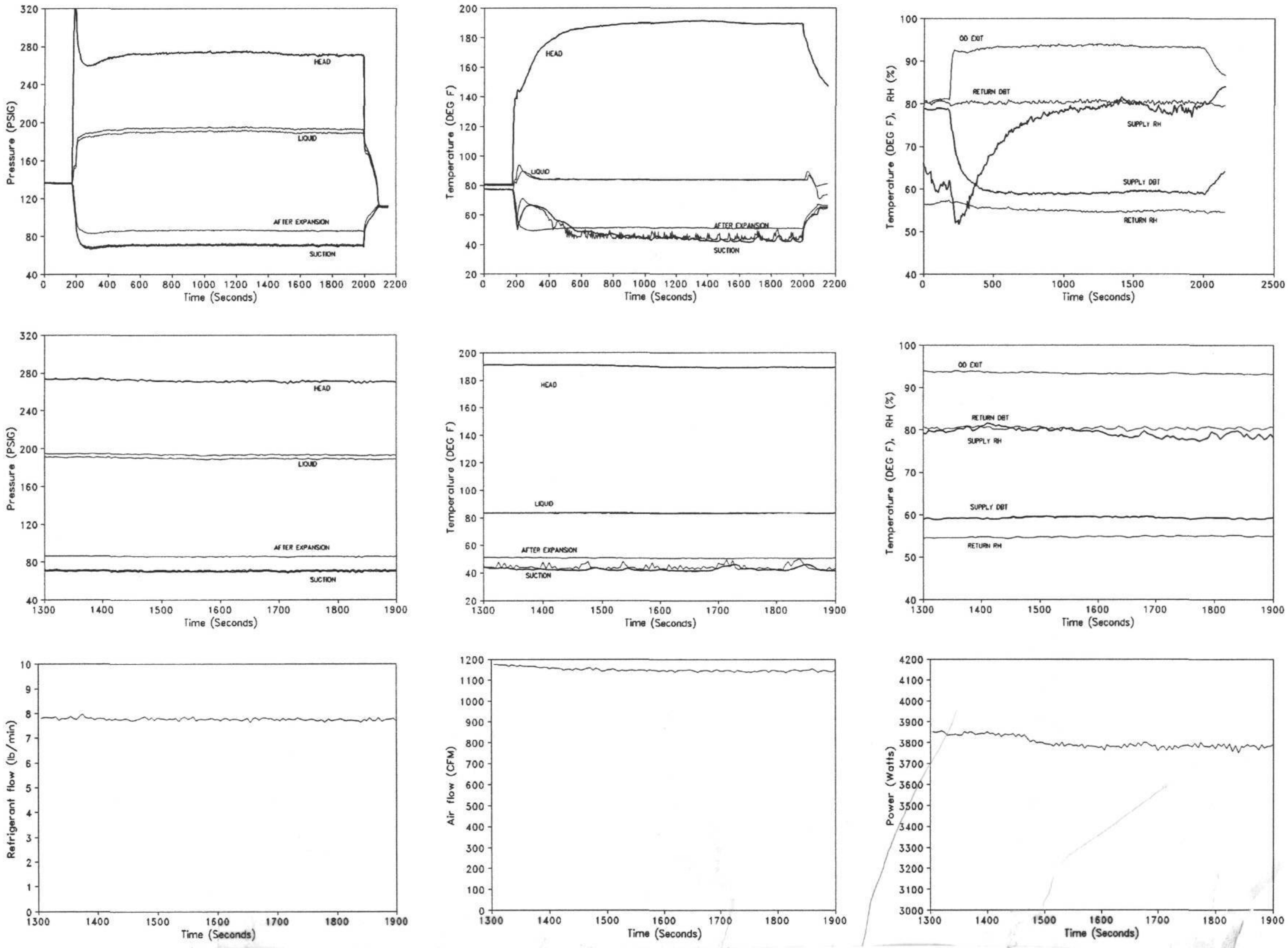


Figure B-4.4.2. Time series plot for restriction in discharge line degraded test

STANDARD RUN				RESTRICTION IN DISCHARGE LINE			
	P(psia)	T(F)	H(Btu/lb)		P(psia)	T(F)	H(Btu/lb)
1.	231	174	126.8	1.	286	190	128.1
2.	210	83	34.2	2.	208	84	34.2
3.	206	83	34.0	3.	205	83	34.1
4.	99	50	34.0	4.	100	51	34.1
5.	83	46	109.2	5.	86	44	108.7
6.	82	48	109.6	6.	85	43	108.5
REF. CAP		35.6 KBtu/hr (2.97 TONS)				34.78 KBtu/hr (2.90 TONS)	
REF. FLOW		7.89 LB/MIN				7.77 LB/MIN	

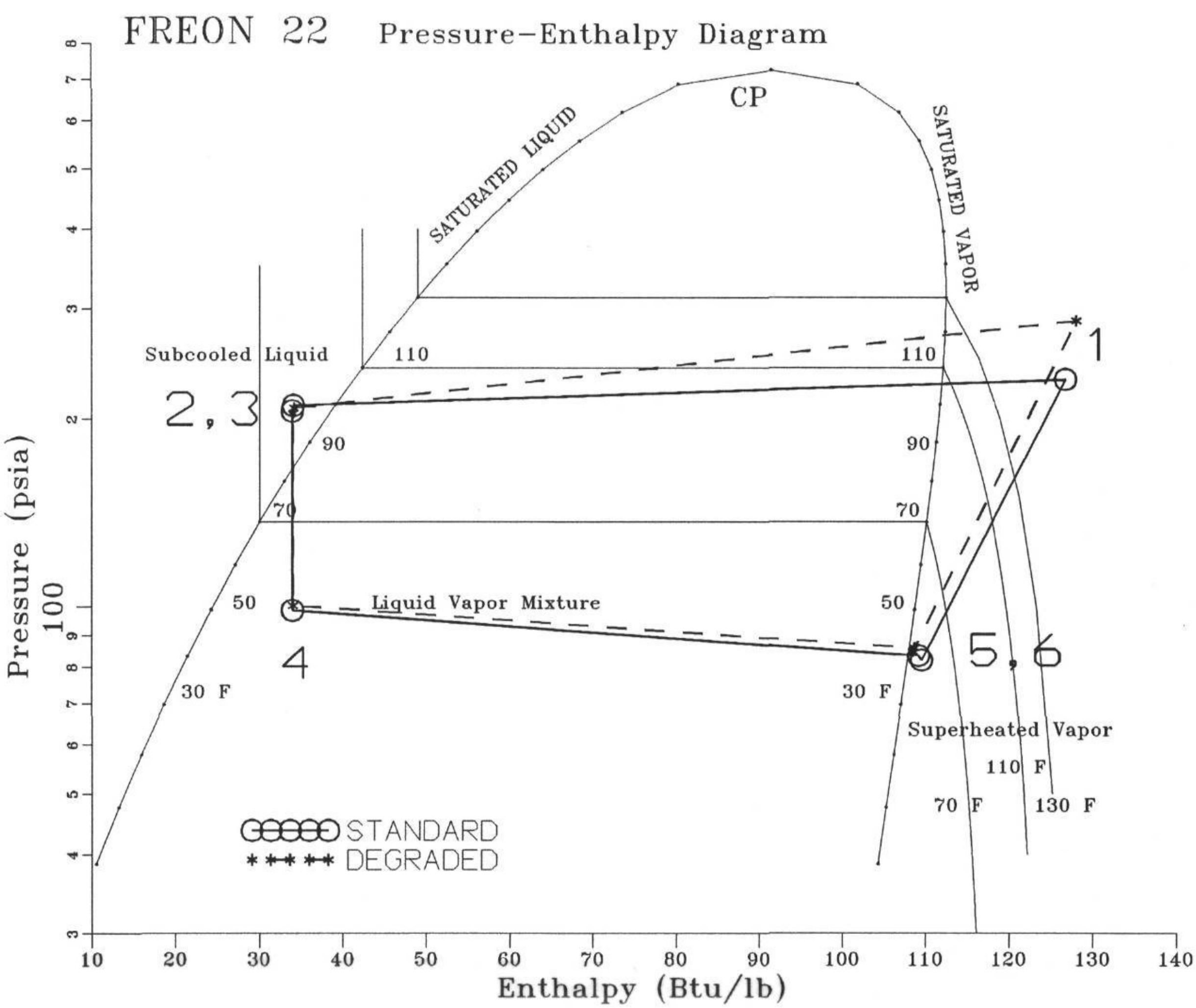


Figure B-4.4.3. P-H chart for restriction in discharge line

STANDARD RUN				RESTRICTION IN DISCHARGE LINE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.3	49	0.0108	13.	80.3	49	0.0107
15.	59.0	89	0.0095	15.	59.3	88	0.0095
16.	93.6		0.0108	16.	93.4		0.0107
EVAP. AIR FLOW			1132	1133 CFM			
COOLING CAPACITY			34.15	33.75 KBtu/hr			

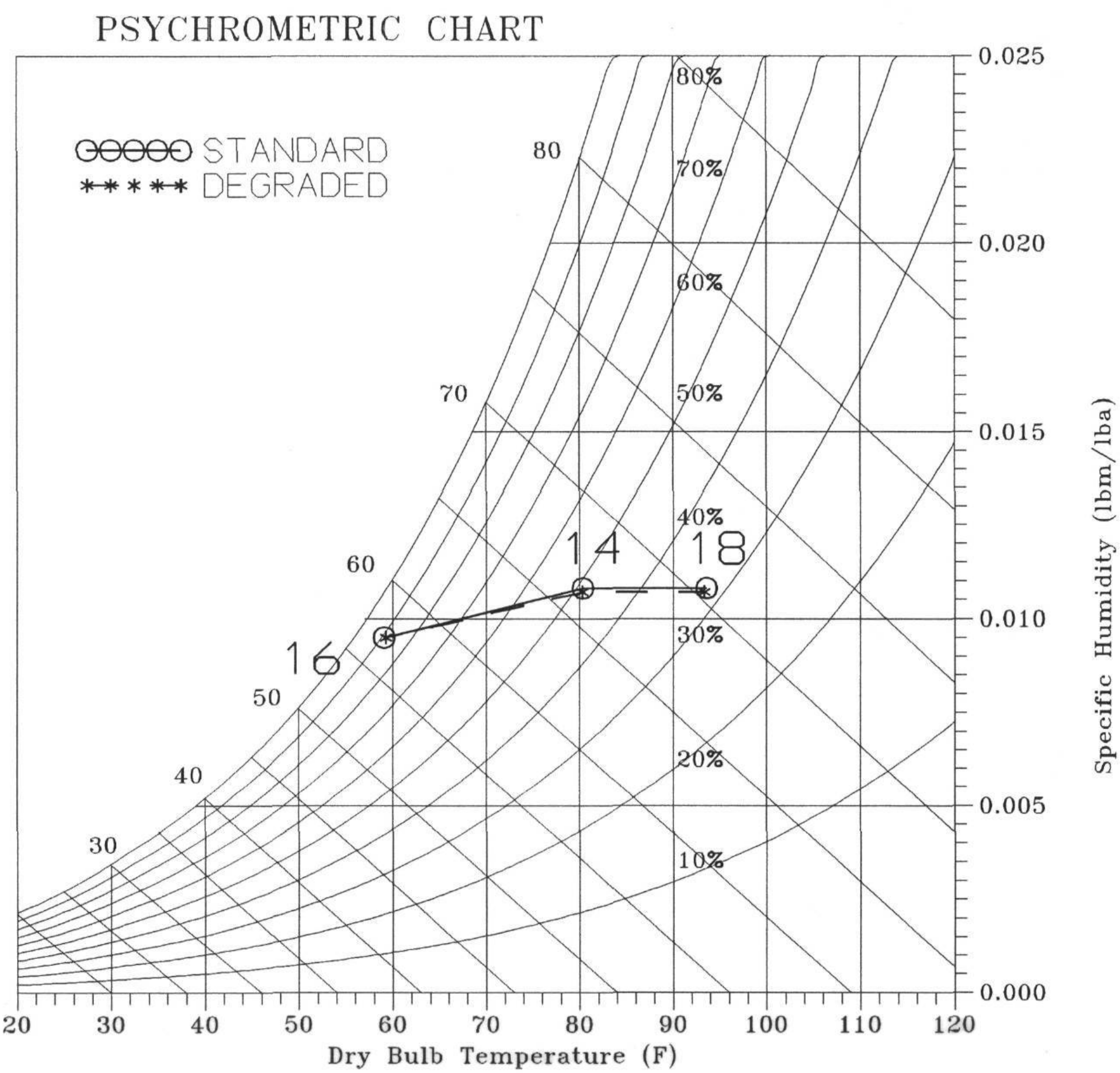


Figure B-4.4.4. Psychrometric chart for restriction in discharge line

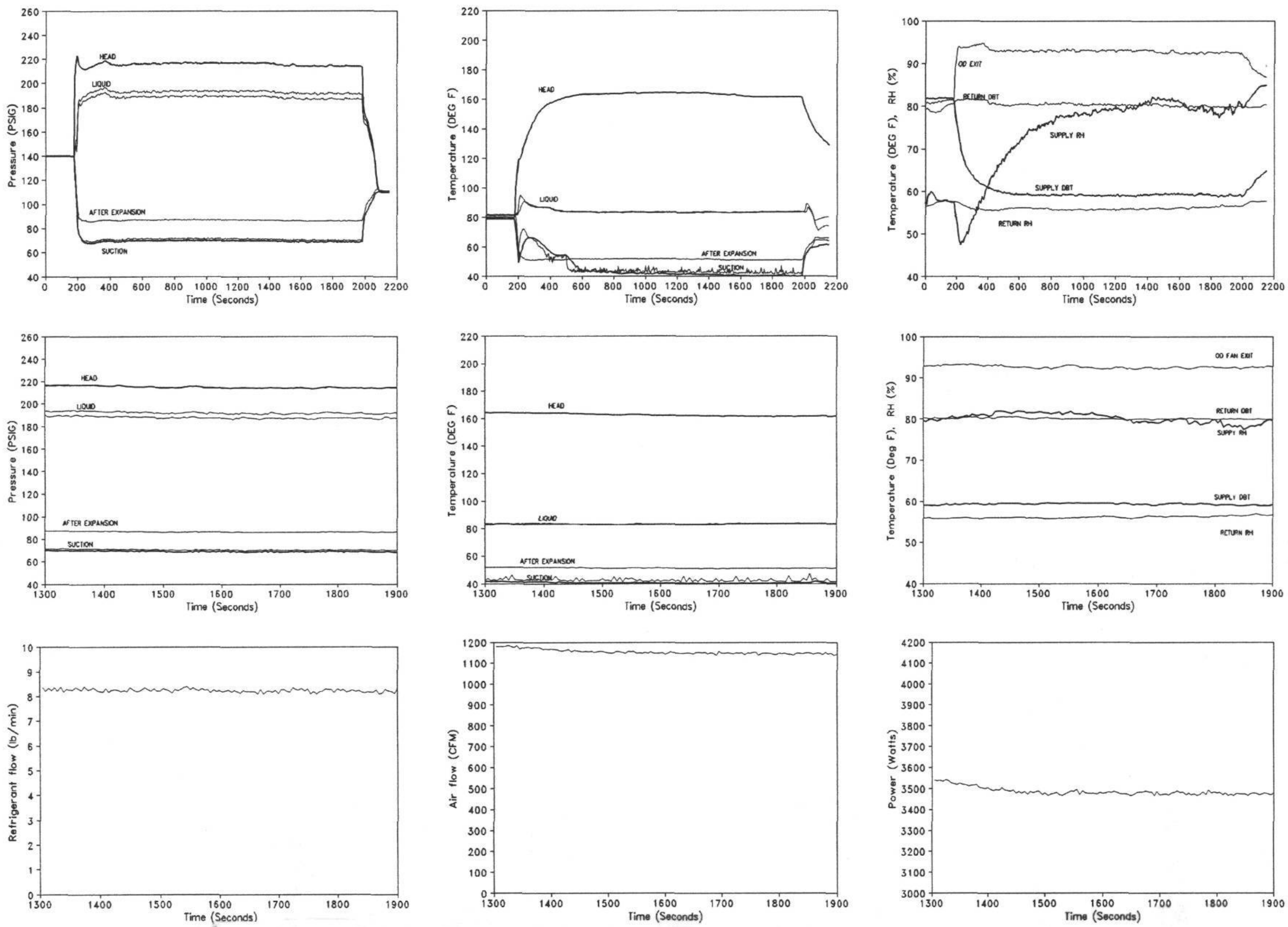


Figure B-5.1.1. Time series plot for undercharge and overcharge standard test

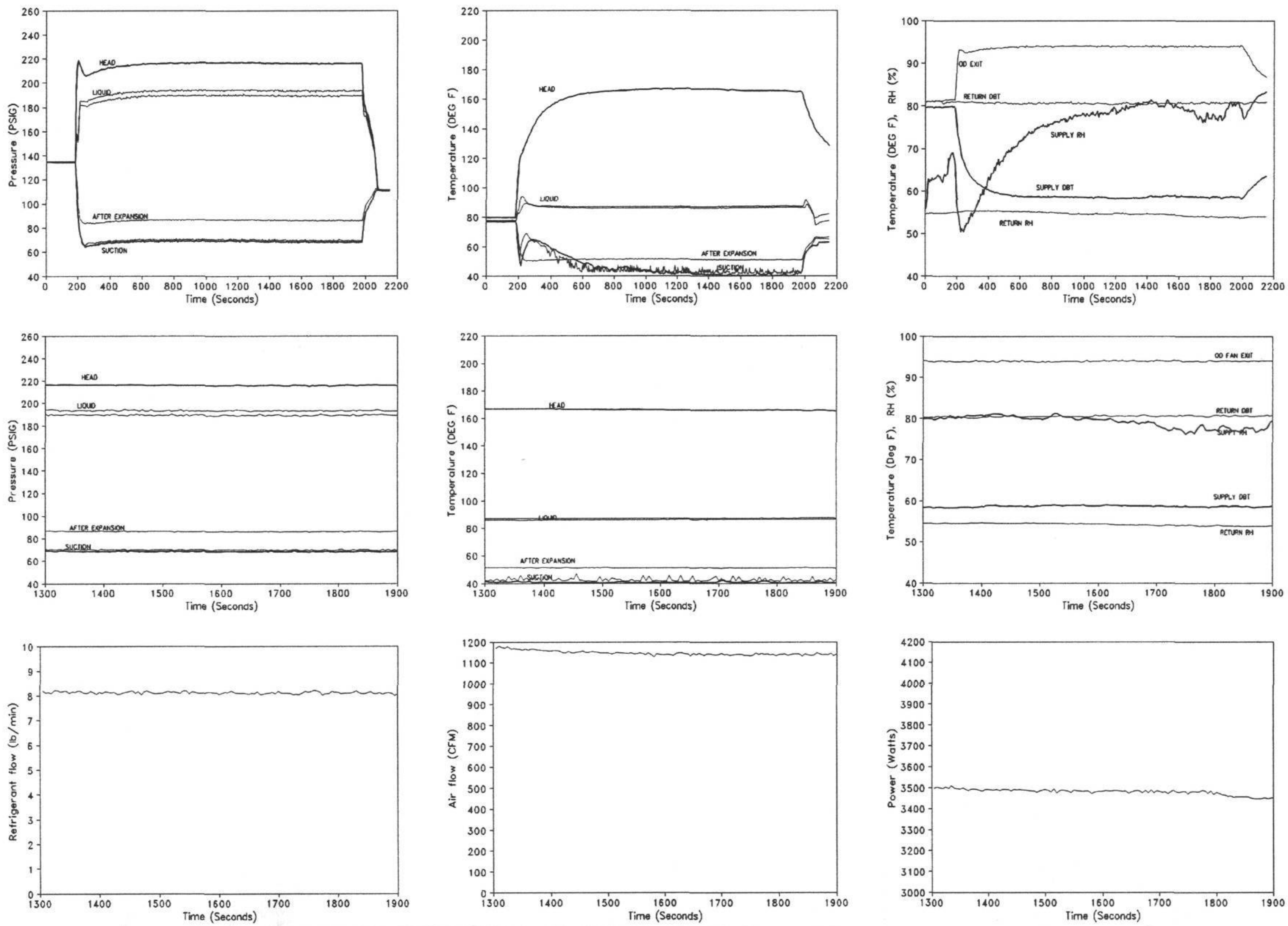


Figure B-5.1.2. Time series plot for 10% undercharge degraded test

STANDARD RUN			10 % UNDERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 231	166	125.2
2. 208	83	34.2	2. 208	87	35.3
3. 202	83	34.1	3. 204	86	35.0
4. 101	51	34.1	4. 101	51	35.0
5. 85	43	108.5	5. 84	43	108.5
6. 84	41	108.2	6. 83	41	108.3
REF. CAP		36.8 KBtu/hr (3.07 TONS)			35.83 KBtu/hr (2.99 TONS)
REF. FLOW		8.24 LB/MIN			8.12 LB/MIN

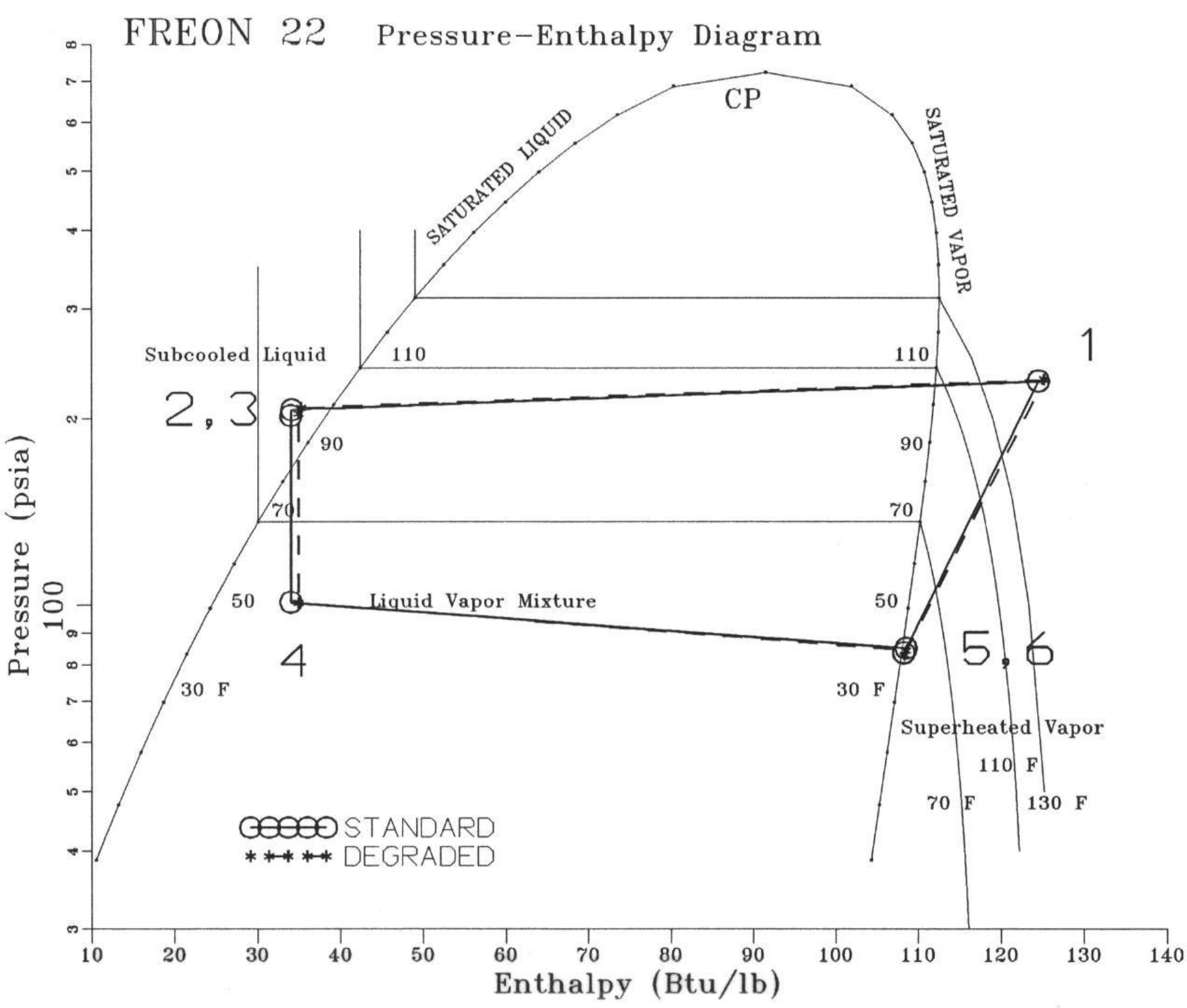


Figure B-5.1.3. P-H chart for 10% undercharge

STANDARD RUN	10 % UNDERCHARGE
DBT(F) RH(%) W(lbm/lb air)	DBT(F) RH(%) W(lbm/lb air)
13. 80.1 50 0.0110	13. 80.5 48 0.0107
15. 59.2 89 0.0095	15. 58.6 89 0.0094
16. 92.6 0.0110	16. 93.9 0.0107
EVAP. AIR FLOW 1137	1136 CFM
COOLING CAPACITY 35.01	35.53 KBtu/hr

PSYCHROMETRIC CHART

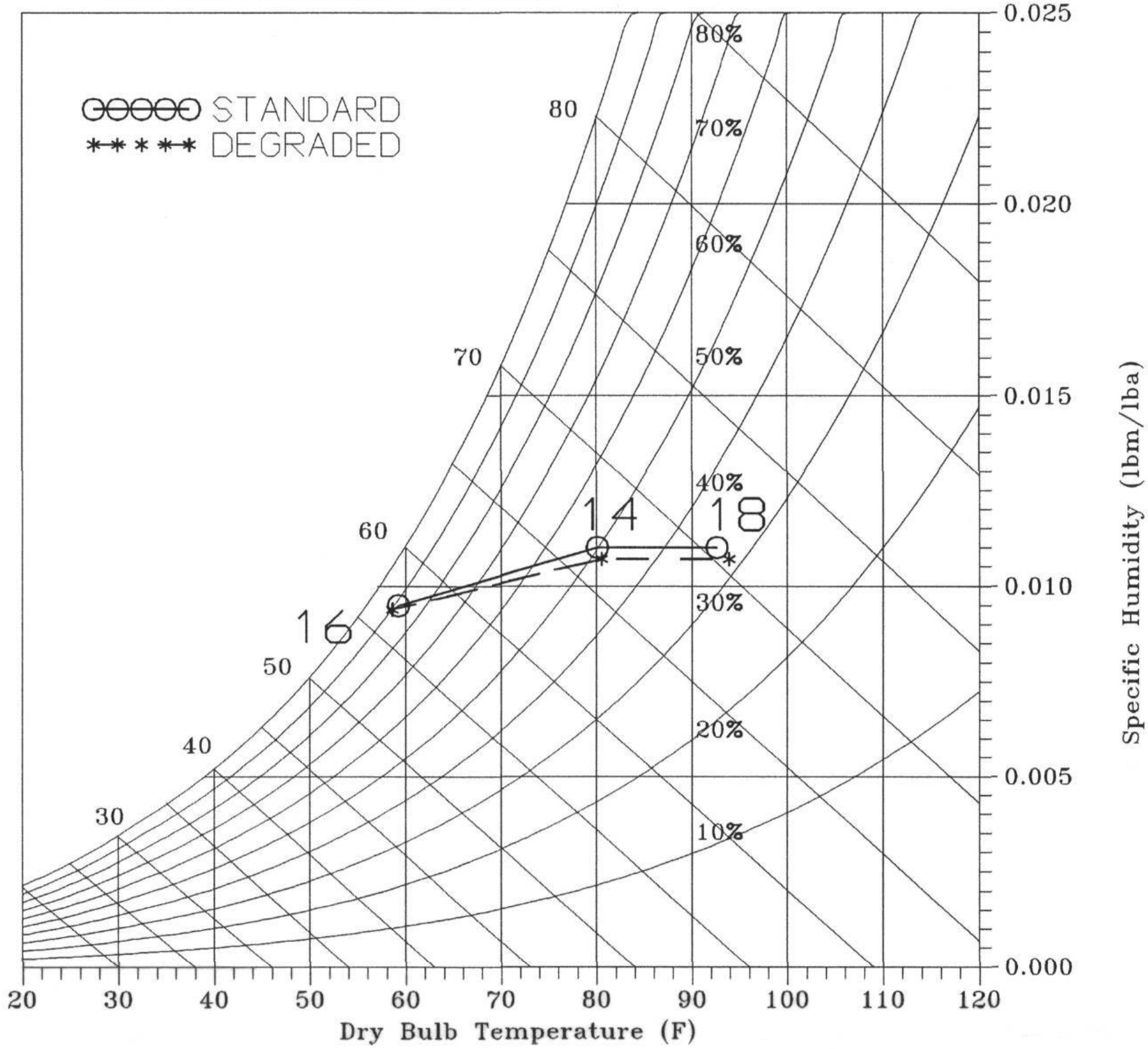


Figure B-5.1.4. Psychrometric chart for 10% undercharge

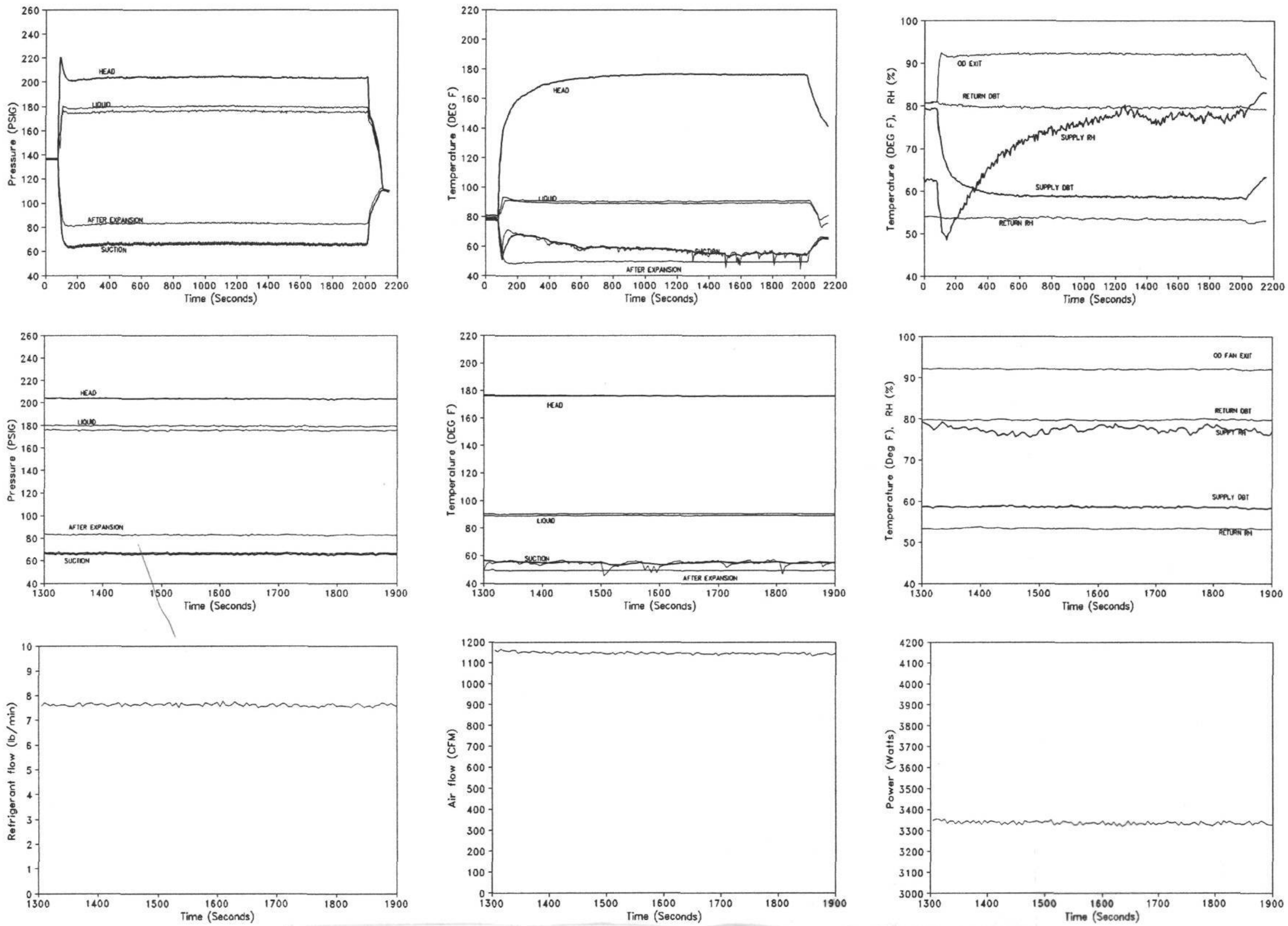


Figure B-5.2.1. Time series plot for 20% undercharge degraded test

STANDARD RUN				20 % UNDERCHARGE			
	P(psia)	T(F)	H(Btu/lb)		P(psia)	T(F)	H(Btu/lb)
1.	230	163	124.5	1.	218	176	127.7
2.	208	83	34.2	2.	194	90	36.3
3.	202	83	34.1	3.	190	89	35.9
4.	101	51	34.1	4.	97	54	35.9
5.	85	43	108.5	5.	81	55	110.8
6.	84	41	108.2	6.	80	41	110.9
REF. CAP		36.8 KBtu/hr (3.07 TONS)				34.23 KBtu/hr (2.86 TONS)	
REF. FLOW		8.24 LB/MIN				7.62 LB/MIN	

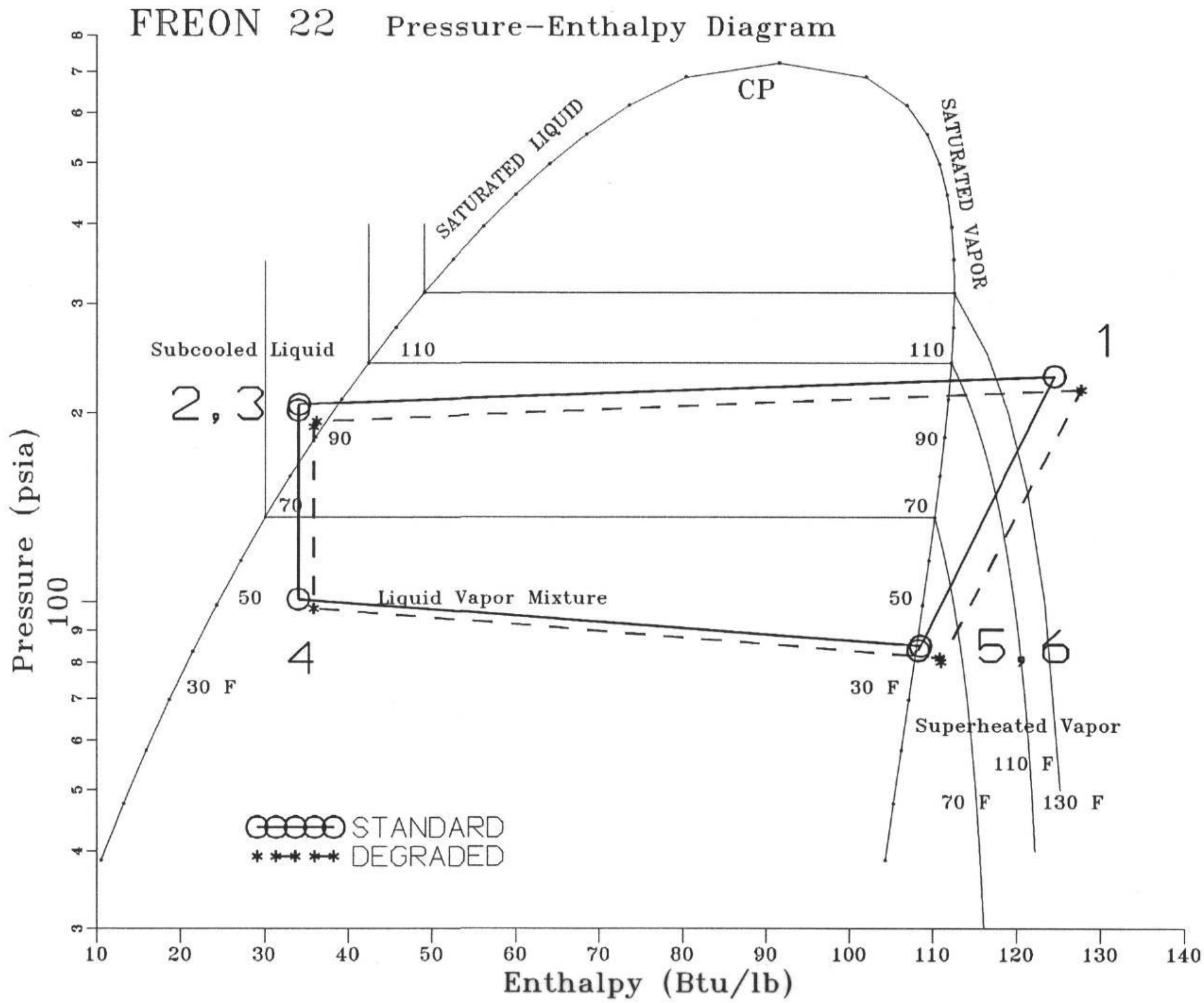


Figure B-5.2.2. P-H chart for 20% undercharge

STANDARD RUN				20 % UNDERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	79.6	47	0.0102
15.	59.2	89	0.0095	15.	58.5	86	0.0090
16.	92.6		0.0110	16.	92.0		0.0102
EVAP. AIR FLOW			1137		1129 CFM		
COOLING CAPACITY			35.01		33.27 KBtu/hr		

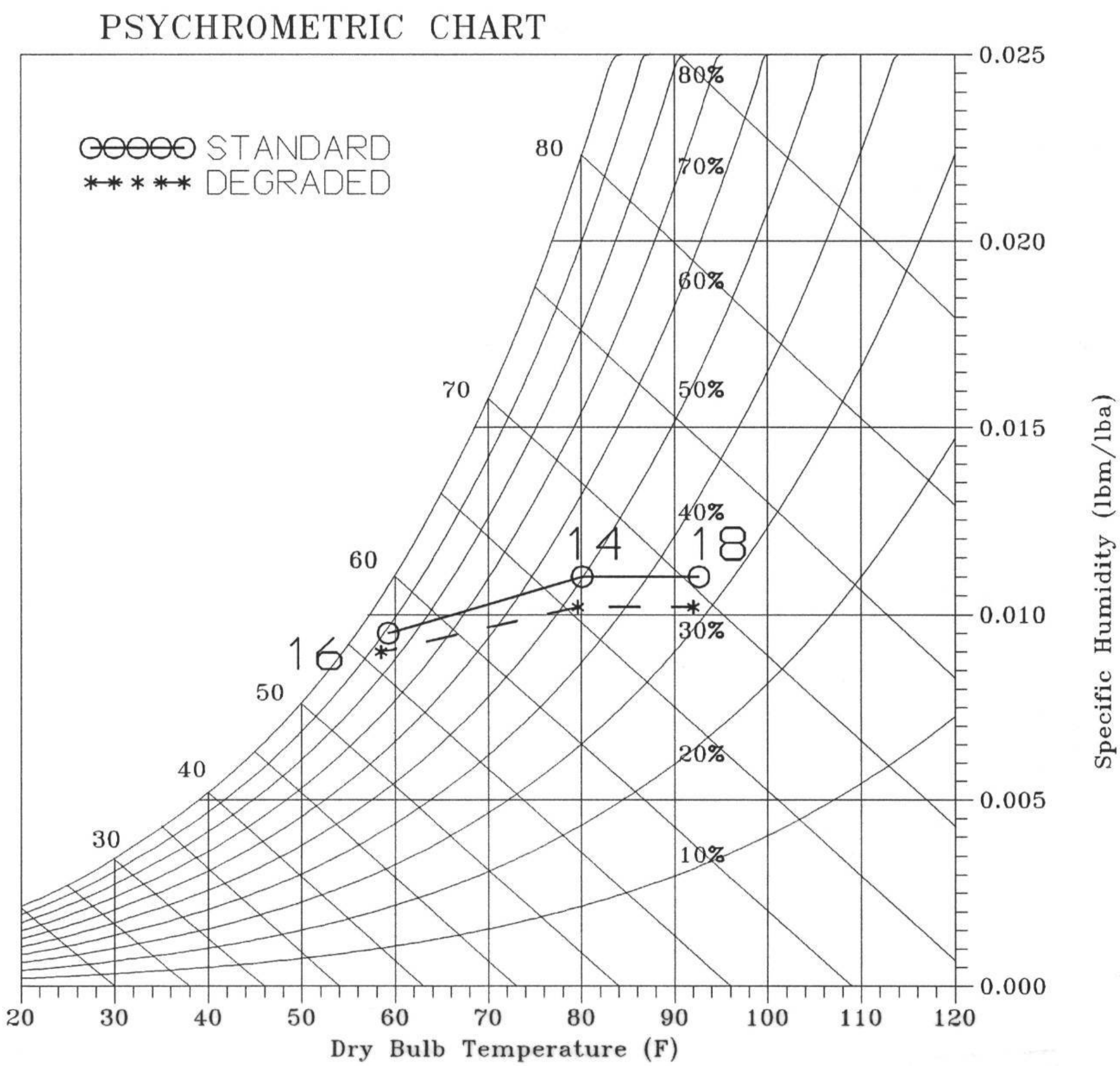


Figure B-5.2.3. Psychrometric chart for 20% undercharge

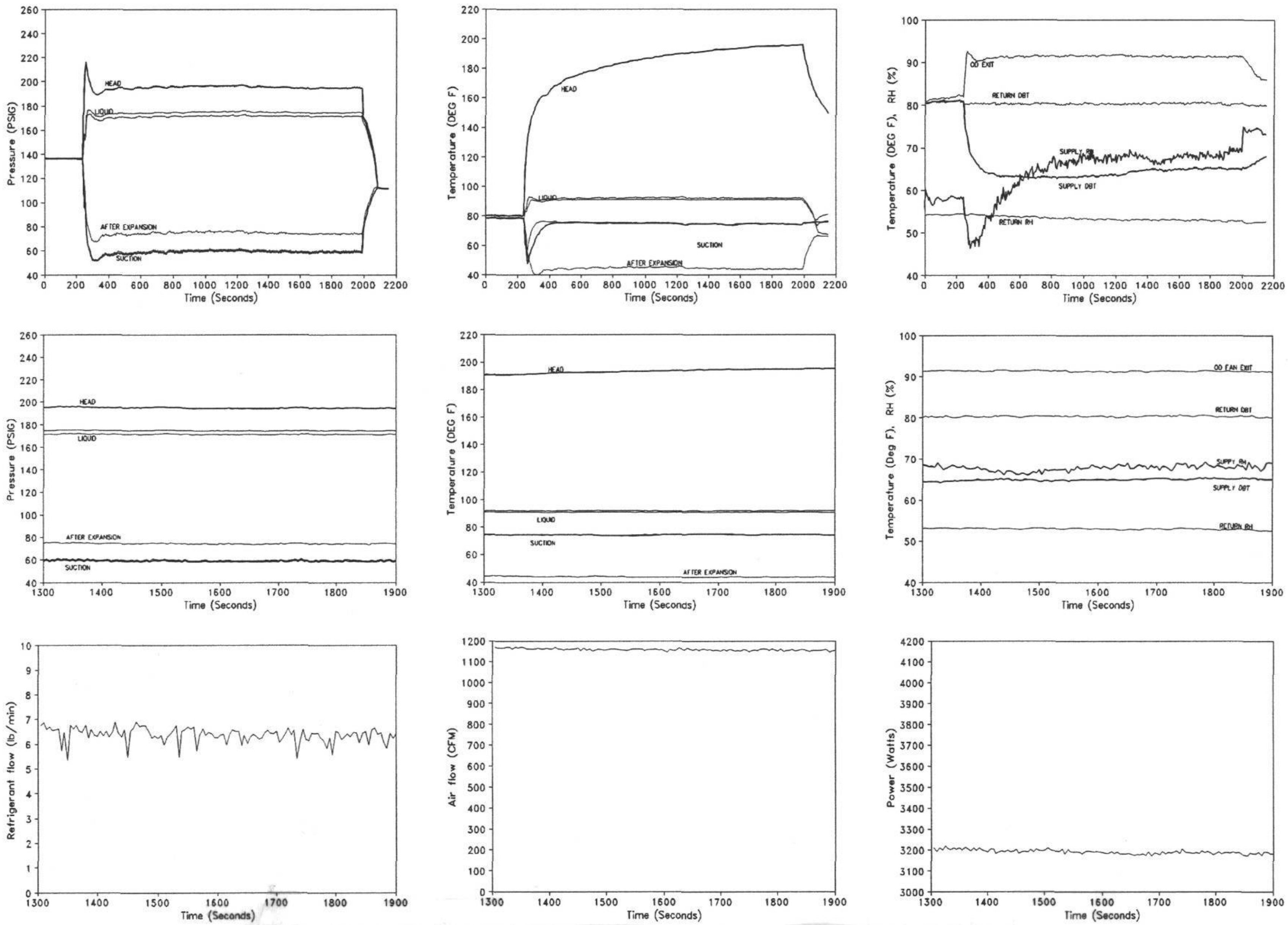


Figure B-5.3.1. Time series plot for 30% undercharge degraded test

STANDARD RUN			30 % UNDERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 210	193	131.5
2. 208	83	34.2	2. 189	92	36.8
3. 202	83	34.1	3. 186	91	36.4
4. 101	51	34.1	4. 89	44	36.4
5. 85	43	108.5	5. 74	74	114.6
6. 84	41	108.2	6. 73	74	114.7
REF. CAP		36.8 KBtu/hr (3.07 TONS)			29.94 KBtu/hr (2.50 TONS)
REF. FLOW		8.24 LB/MIN			6.38 LB/MIN

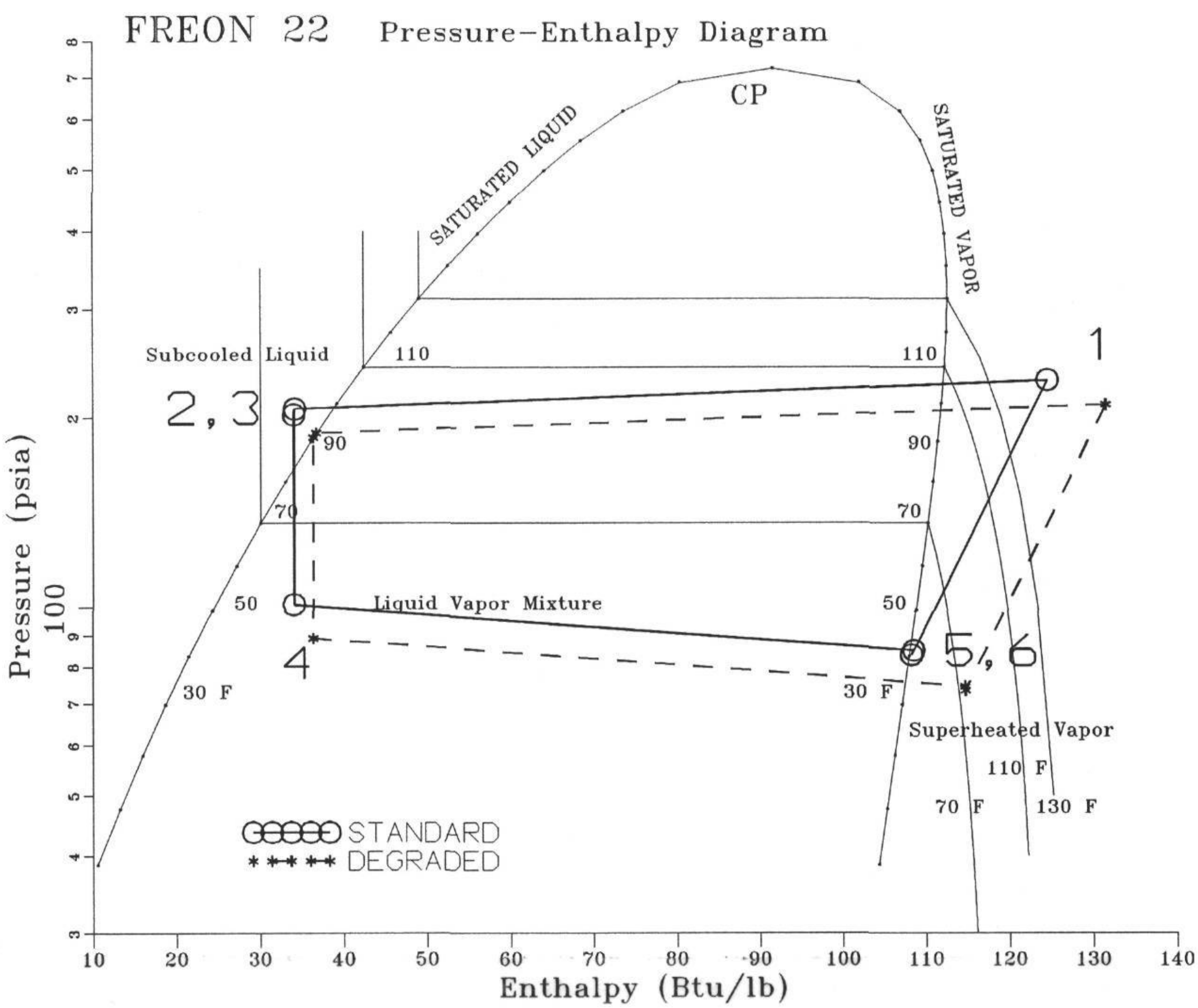


Figure B-5.3.2. P-H chart for 30% undercharge

	STANDARD RUN			30 % UNDERCHARGE		
	DBT(F)	RH(%)	W(lbm/lb air)	DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	80.4	47	0.0104
15.	59.2	89	0.0095	64.9	77	0.0101
16.	92.6		0.0110	91.3		0.0104
EVAP. AIR FLOW	1137			1149 CFM		
COOLING CAPACITY	35.01			21.14 KBtu/hr		

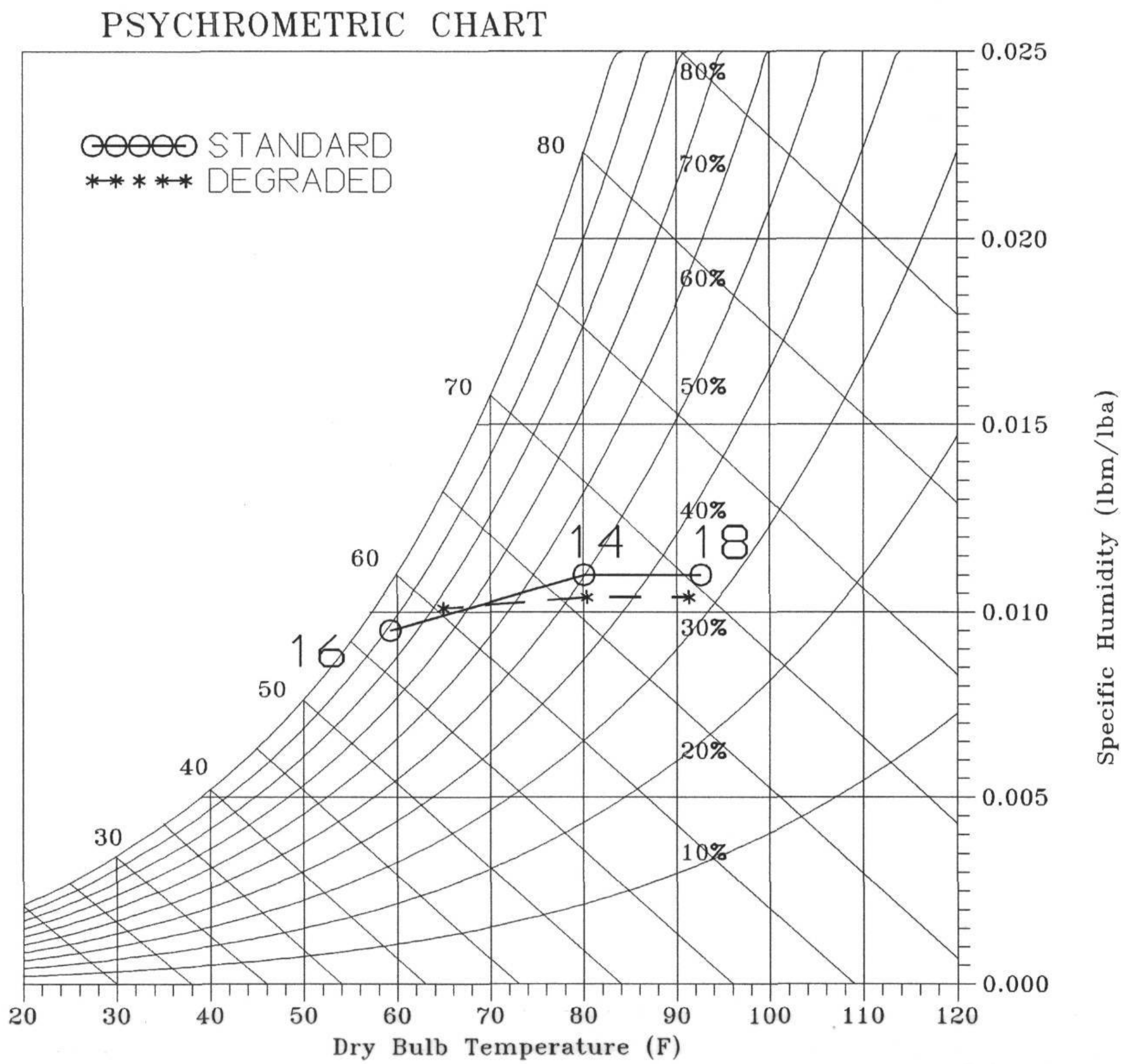


Figure B-5.3.3. Psychrometric chart for 30% undercharge

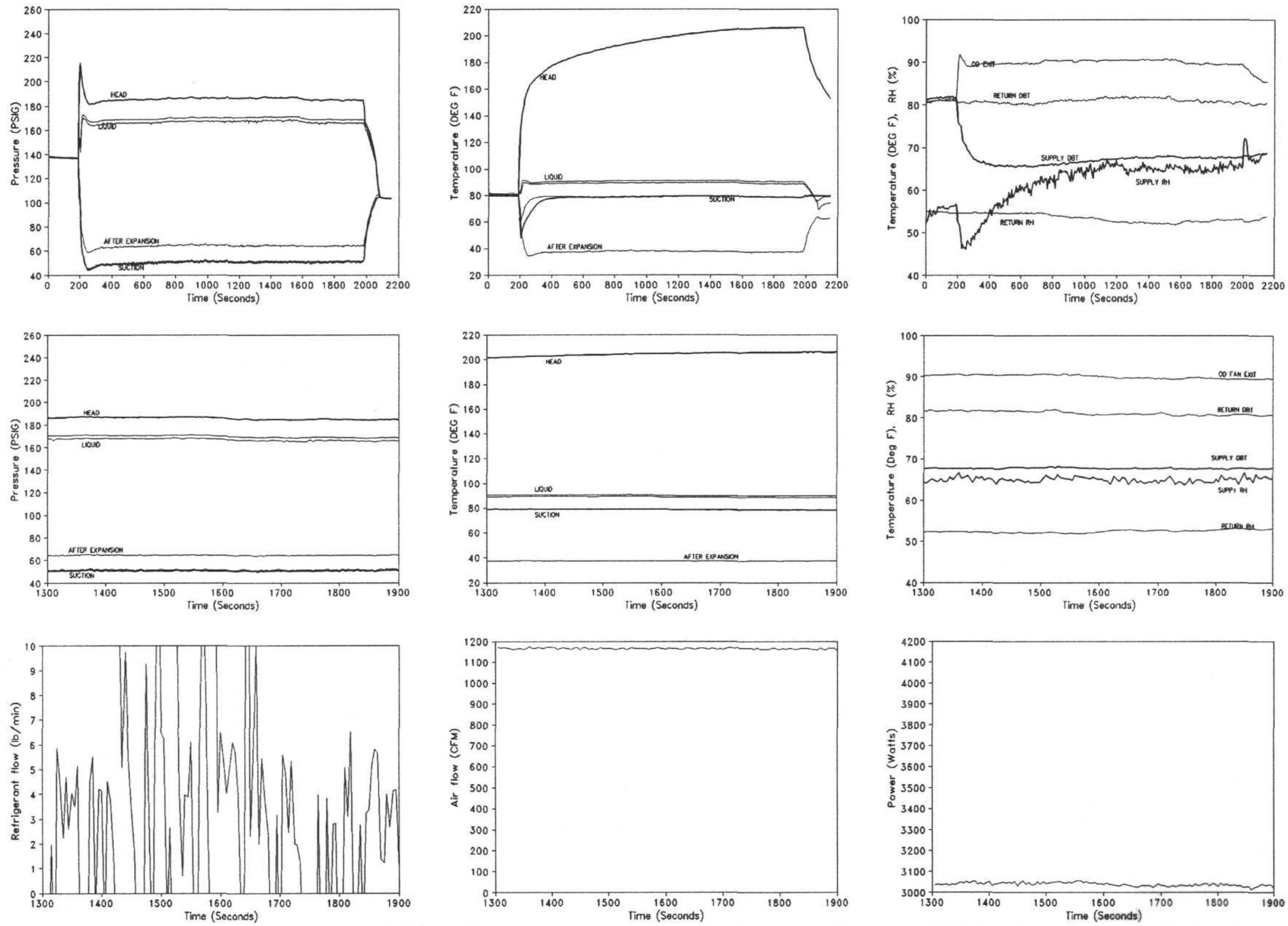


Figure B-5.4.1. Time series plot for 40% undercharge degraded test

	STANDARD RUN			40 % UNDERCHARGE		
	P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1.	230	163	124.5	201	204	133.9
2.	208	83	34.2	185	90	36.3
3.	202	83	34.1	182	89	35.9
4.	101	51	34.1	79	37	35.9
5.	85	43	108.5	66	79	115.8
6.	84	41	108.2	65	79	115.8
REF. CAP		36.8 KBtu/hr (3.07 TONS)			N.A.	
REF. FLOW		8.24 LB/MIN			N.A.	

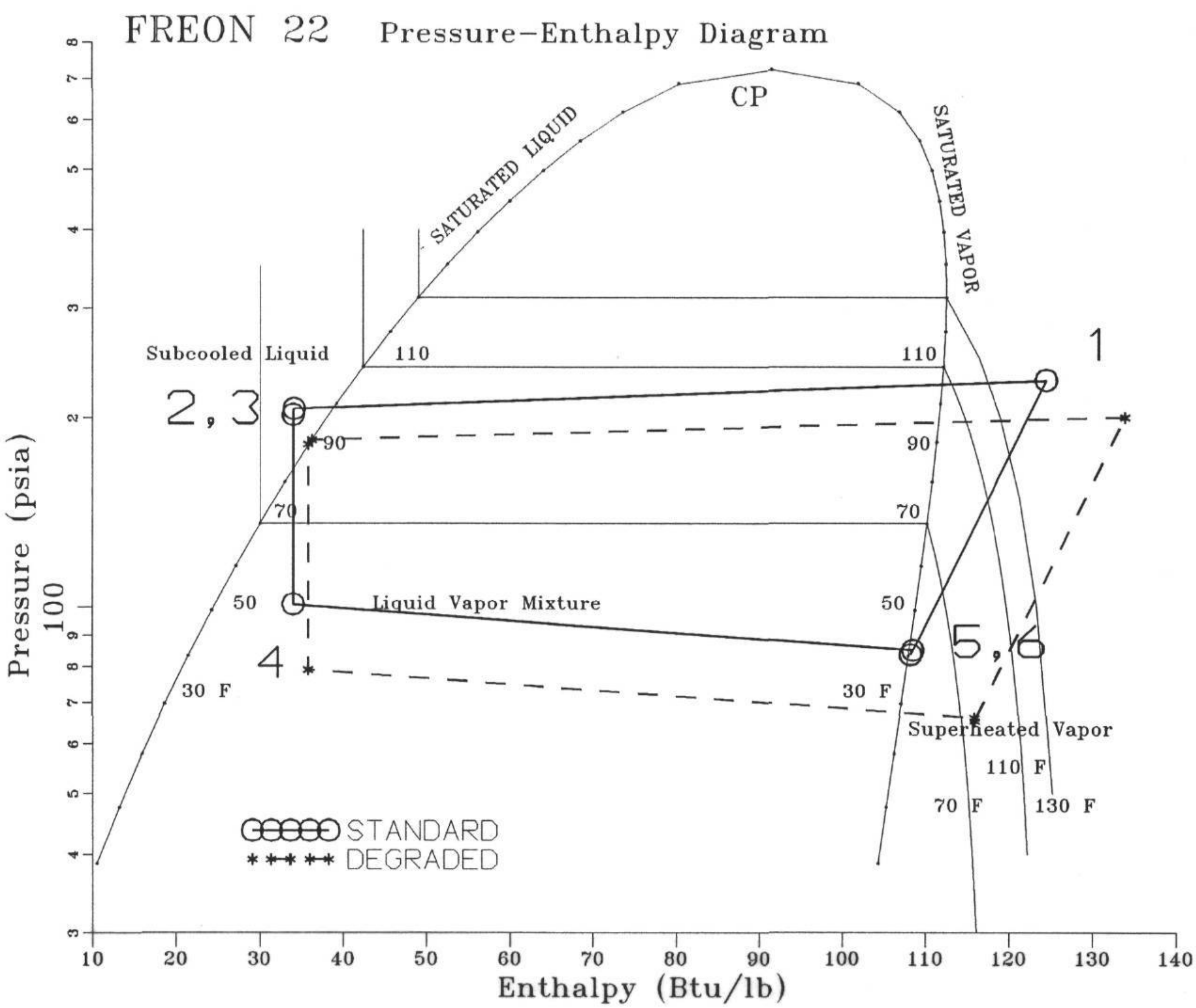


Figure B-5.4.2. P-H chart for 40% undercharge

STANDARD RUN				40 % UNDERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	81.1	46	0.0105
15.	59.2	89	0.0095	15.	67.7	74	0.0107
16.	92.6		0.0110	16.	90.0		0.0105
EVAP. AIR FLOW			1137	1159 CFM			
COOLING CAPACITY			35.01	16.02 KBtu/hr			

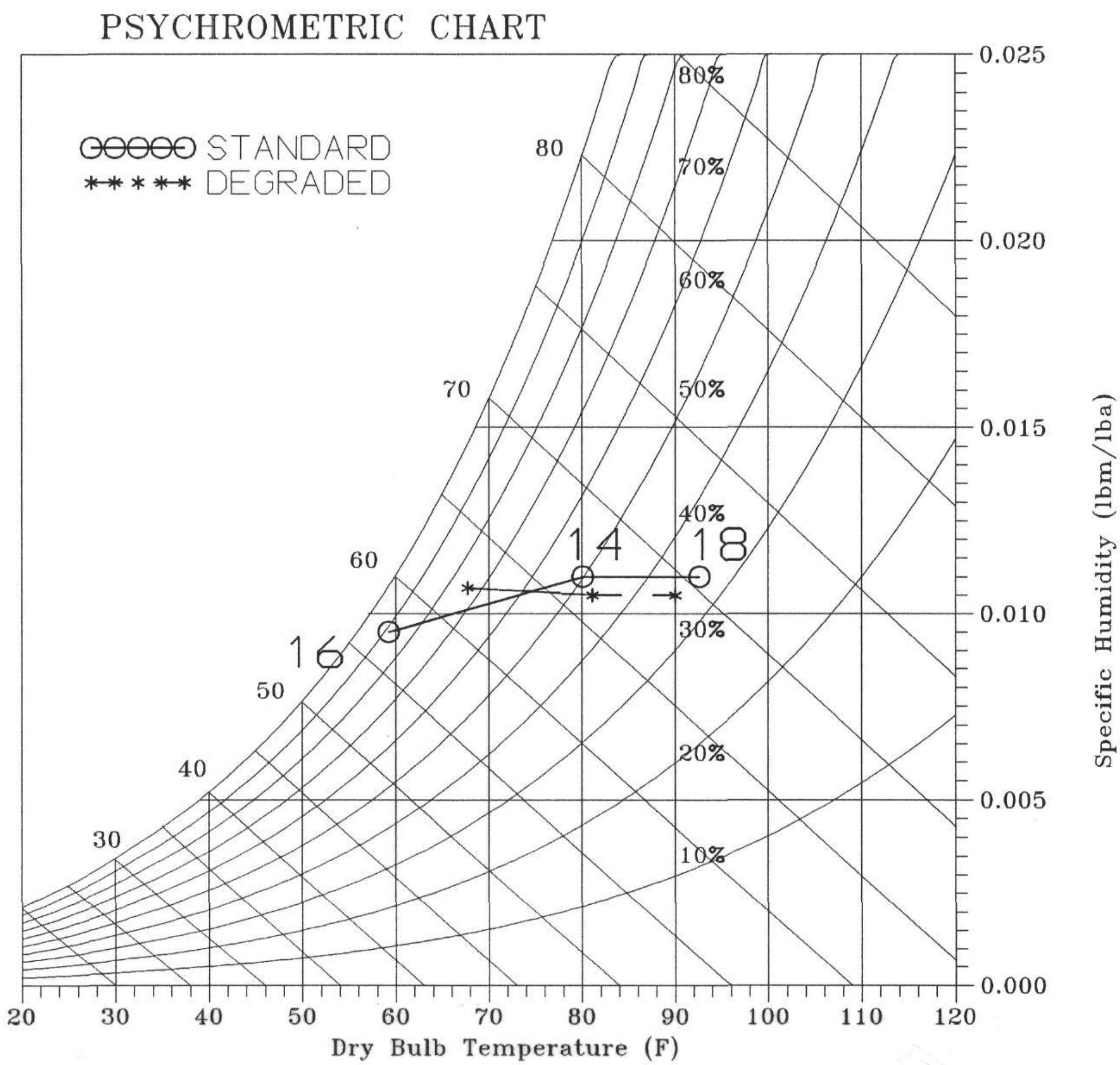


Figure B-5.4.3. Psychrometric chart for 40% undercharge

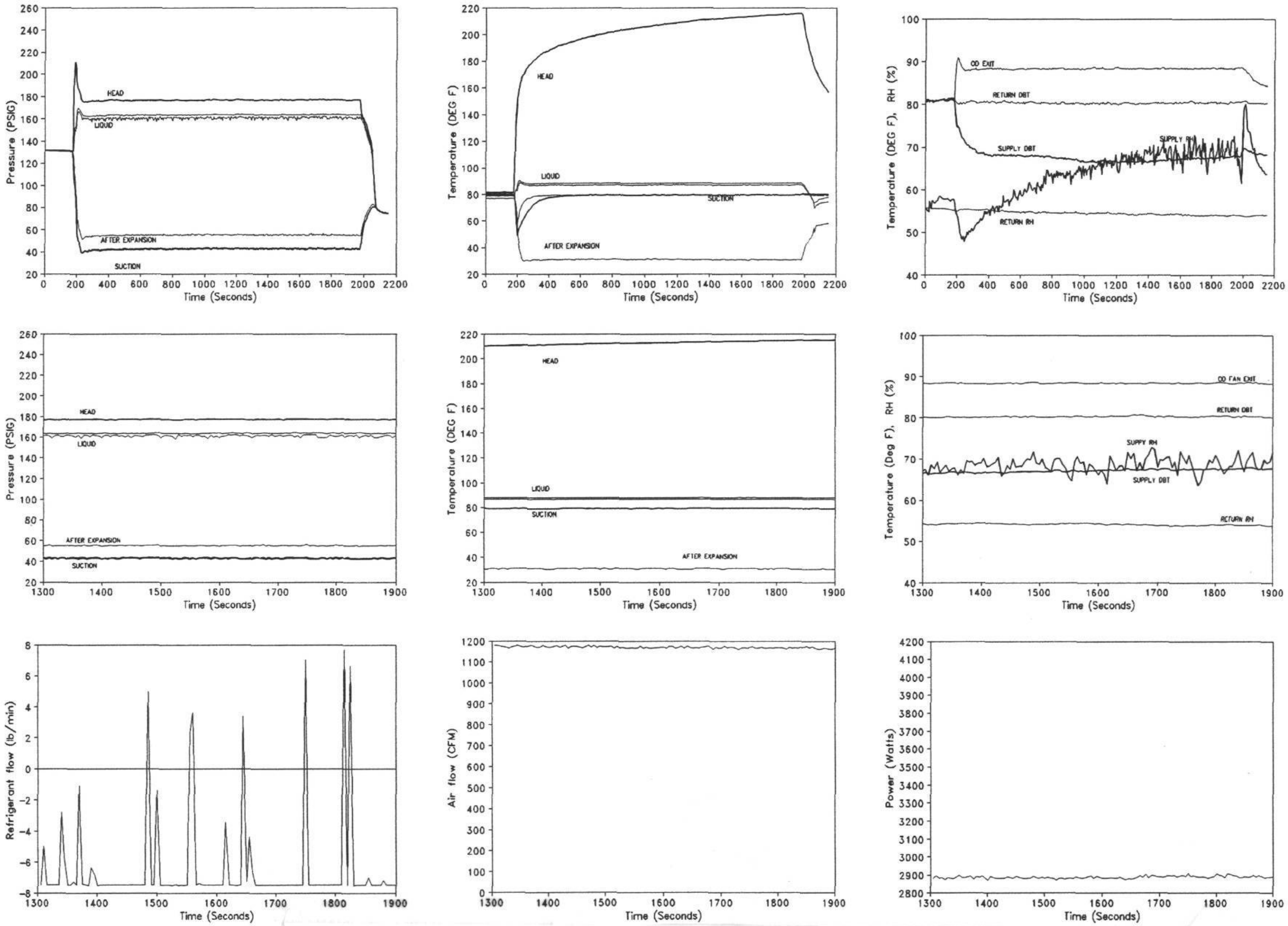


Figure B-5.5.1. Time series plot for 50% undercharge degraded test

STANDARD RUN			50 % UNDERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 192	213	135.8
2. 208	83	34.2	2. 178	88	35.6
3. 202	83	34.1	3. 176	87	35.2
4. 101	51	34.1	4. 70	31	35.2
5. 85	43	108.5	5. 58	79	116.3
6. 84	41	108.2	6. 57	79	116.3
REF. CAP	36.8 KBtu/hr (3.07 TONS)			N.A.	
REF. FLOW	8.24 LB/MIN			N.A.	

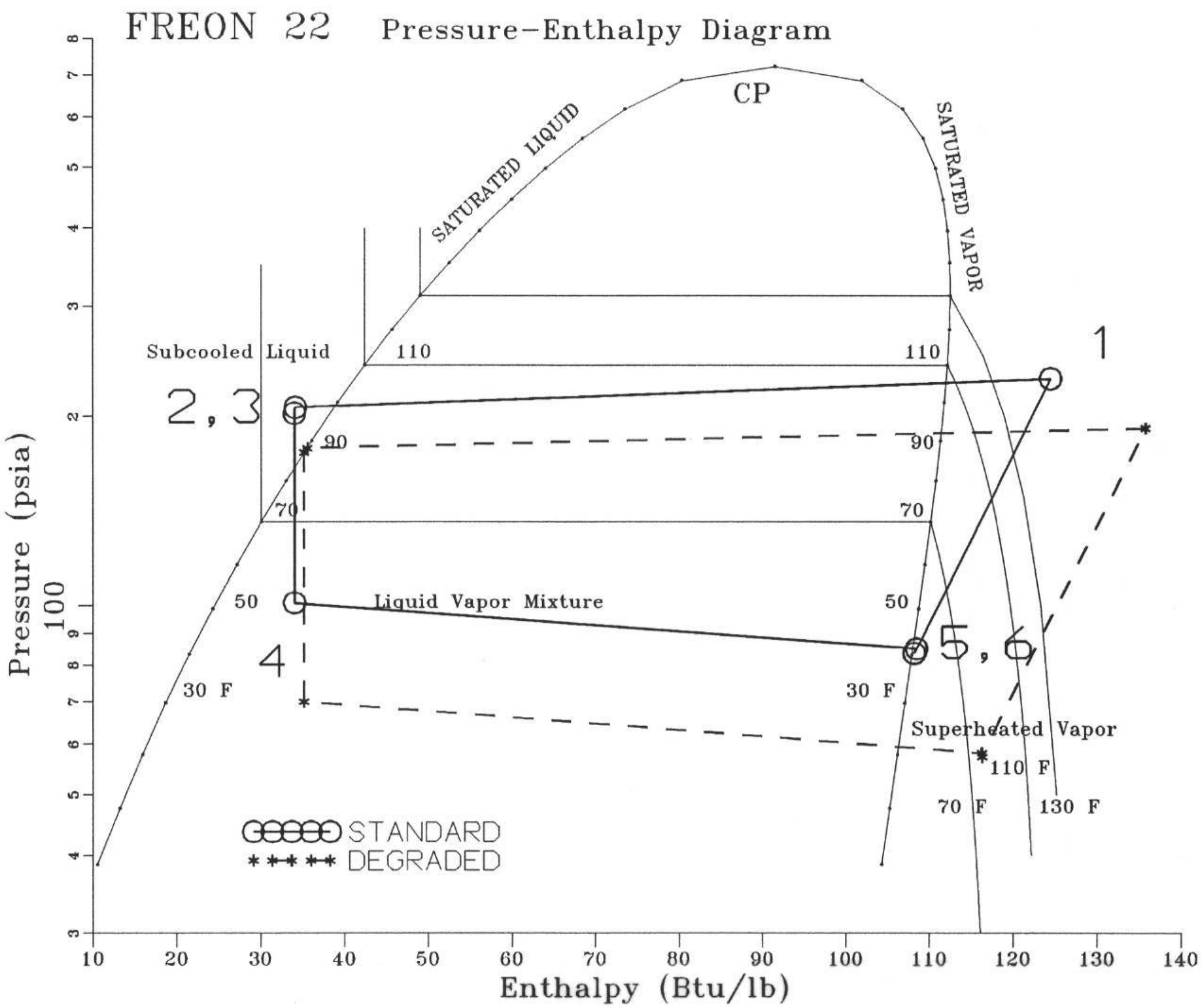


Figure B-5.5.2. P-H chart for 50% undercharge

STANDARD RUN				50 % UNDERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	80.3	48	0.0106
15.	59.2	89	0.0095	15.	67.2	78	0.0110
16.	92.6		0.0110	16.	88.4		0.0106
EVAP. AIR FLOW			1137		1163 CFM		
COOLING CAPACITY			35.01		14.52 KBtu/hr		

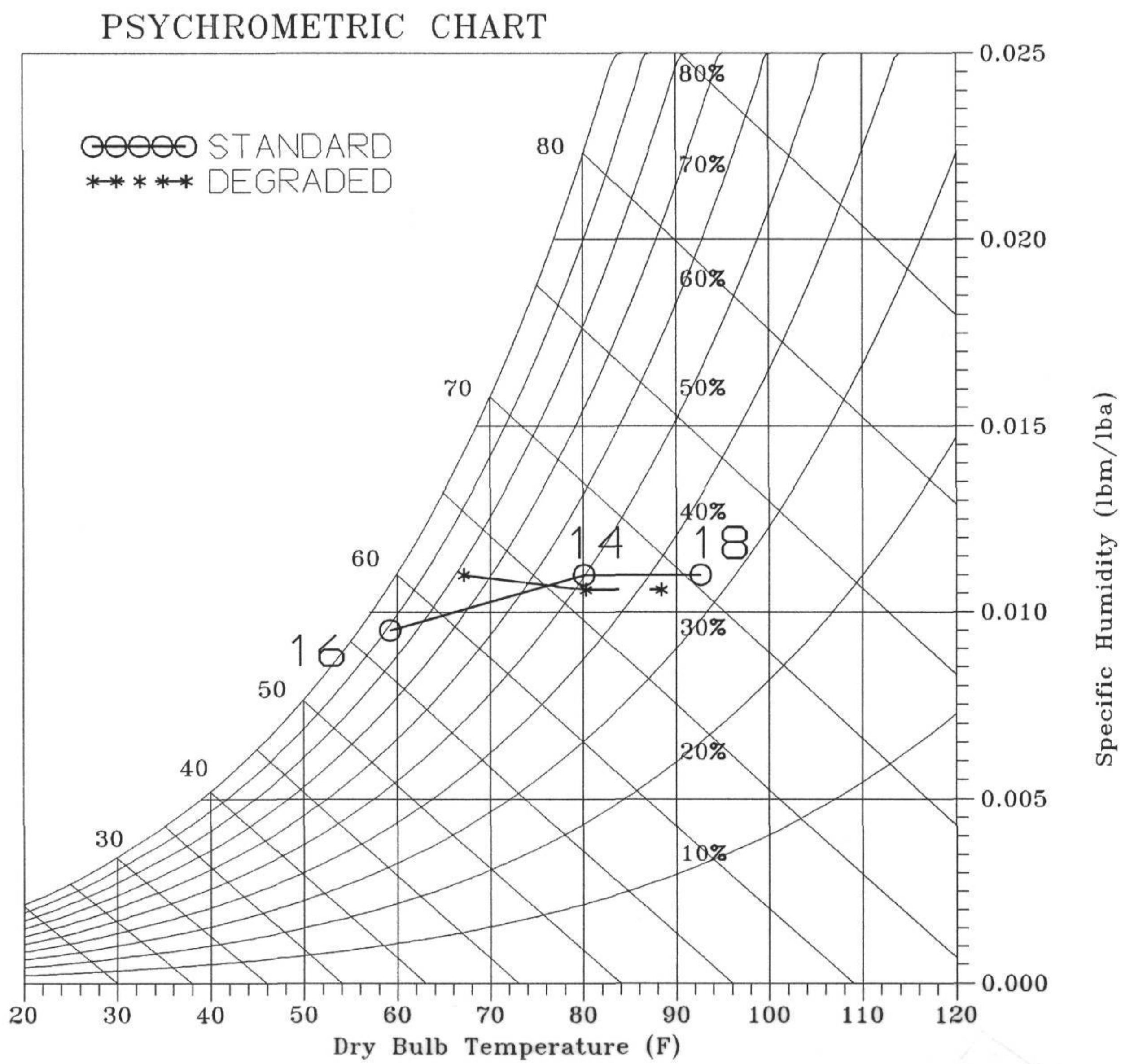


Figure B-5.5.3. Psychrometric chart for 50% undercharge

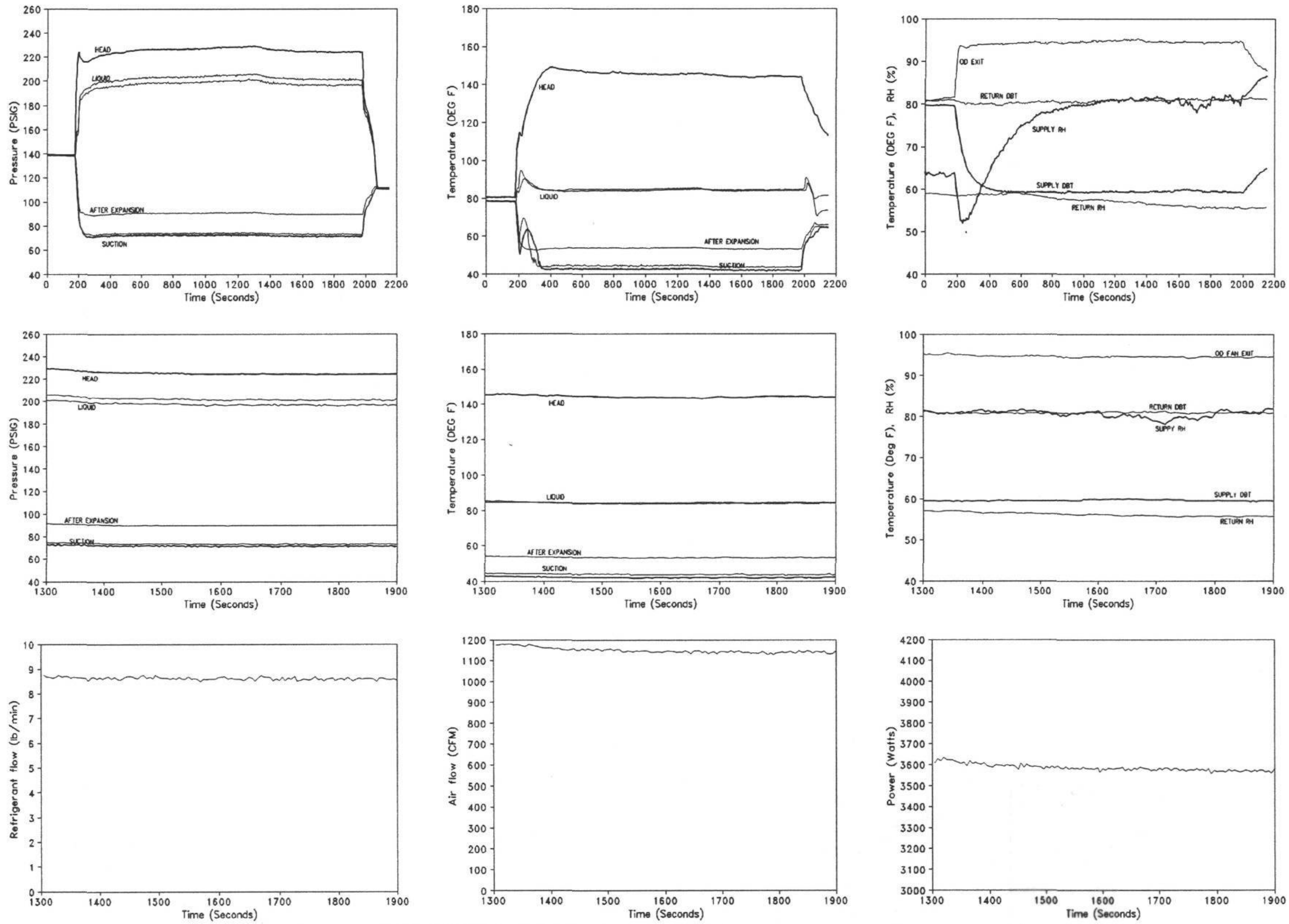


Figure B-6.1.1. Time series plot for 10% overcharge degraded test

STANDARD RUN			10 % OVERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 240	144	120.2
2. 208	83	34.2	2. 217	84	34.4
3. 202	83	34.1	3. 212	84	34.3
4. 101	51	34.1	4. 105	53	34.3
5. 85	43	108.5	5. 88	44	108.5
6. 84	41	108.2	6. 86	42	108.5
REF. CAP		36.8 KBtu/hr (3.07 TONS)			38.39 KBtu/hr (3.2 TONS)
REF. FLOW		8.24 LB/MIN			8.62 LB/MIN

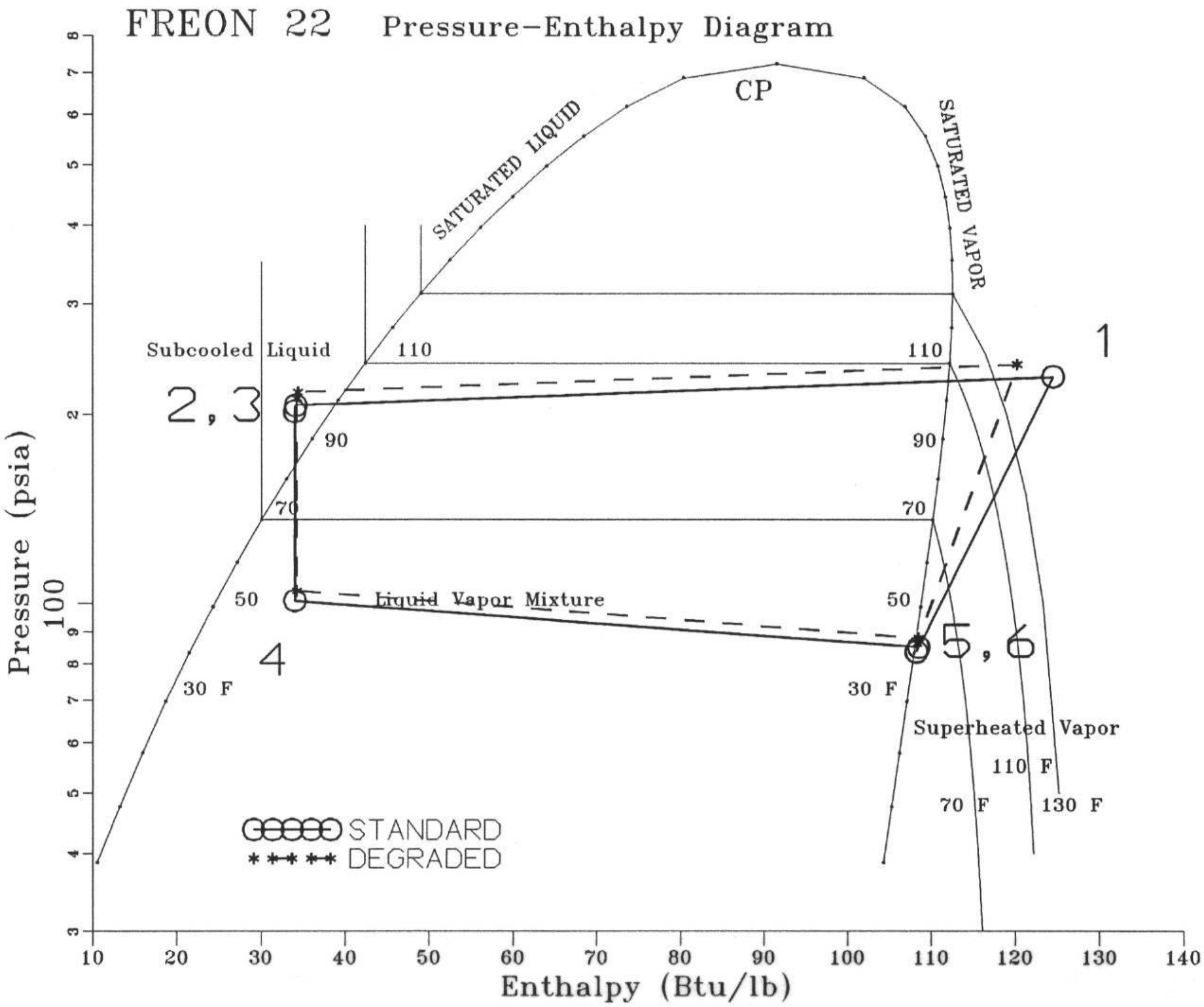


Figure B-6.1.2. P-H chart for 10% overcharge

STANDARD RUN				10 % OVERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	80.9	50	0.0113
15.	59.2	89	0.0095	15.	59.6	90	0.0097
16.	92.6		0.0110	16.	94.6		0.0113
EVAP. AIR FLOW			1137	1133 CFM			
COOLING CAPACITY			35.01	35.78 KBtu/hr			

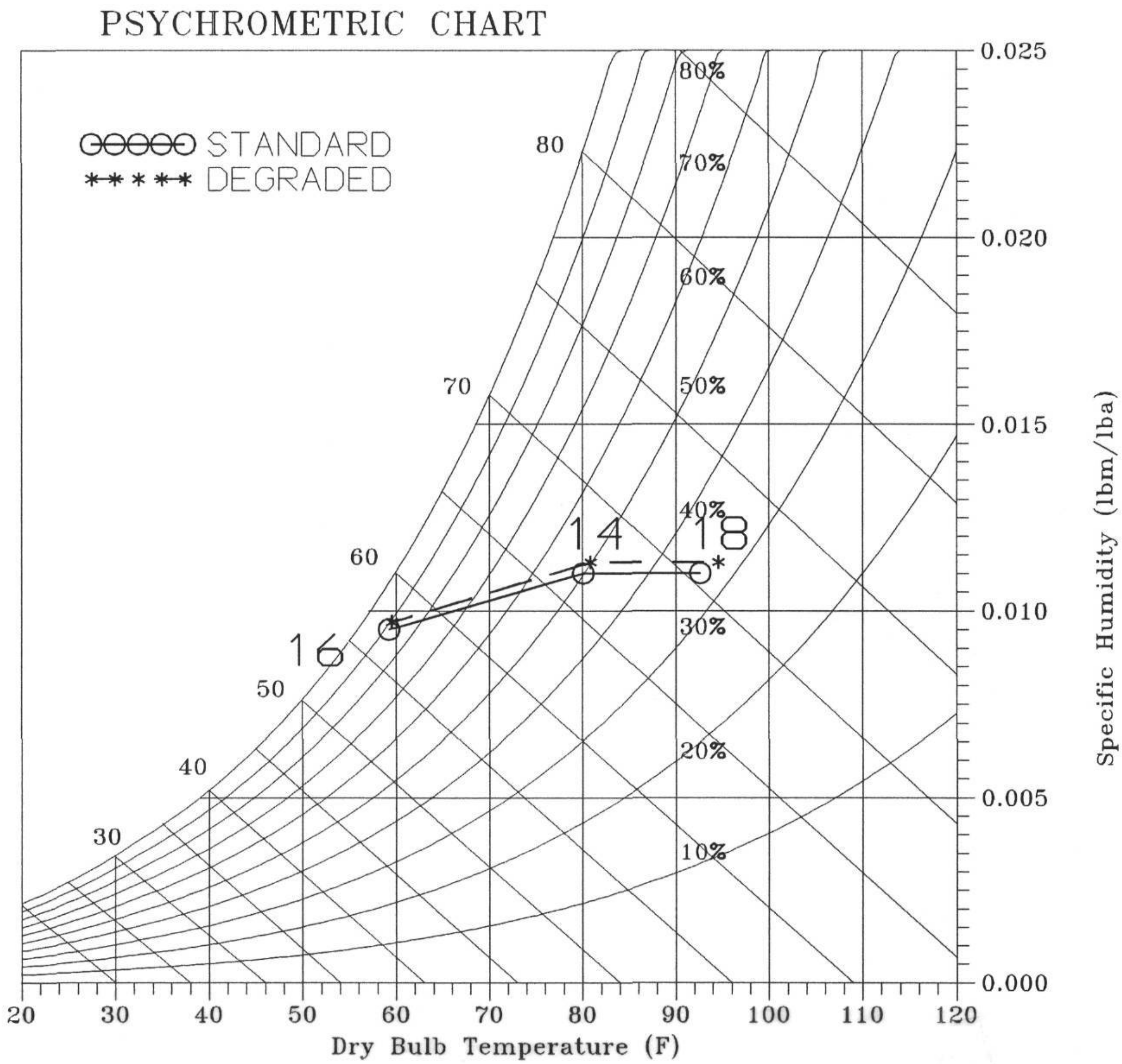


Figure B-6.1.3. Psychrometric chart for 10% overcharge

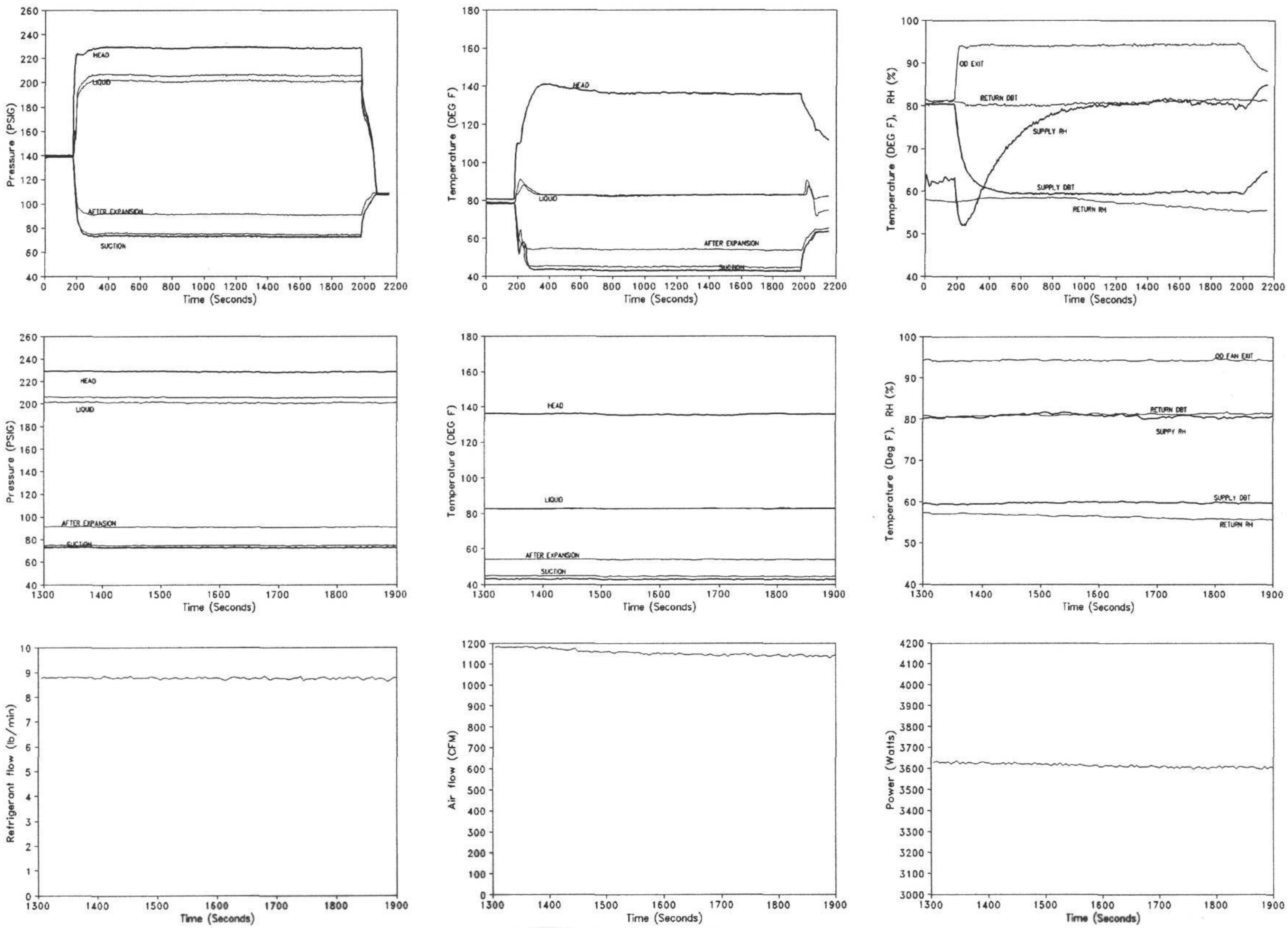


Figure B-6.2.1. Time series plot for 20% overcharge degraded test

STANDARD RUN			20 % OVERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 243	136	118.2
2. 208	83	34.2	2. 221	83	34.0
3. 202	83	34.1	3. 216	83	33.9
4. 101	51	34.1	4. 106	54	33.9
5. 85	43	108.5	5. 89	45	108.6
6. 84	41	108.2	6. 87	43	108.4
REF. CAP		36.8 KBtu/hr (3.07 TONS)			39.27 KBtu/hr (3.27 TONS)
REF. FLOW		8.24 LB/MIN			8.76 LB/MIN

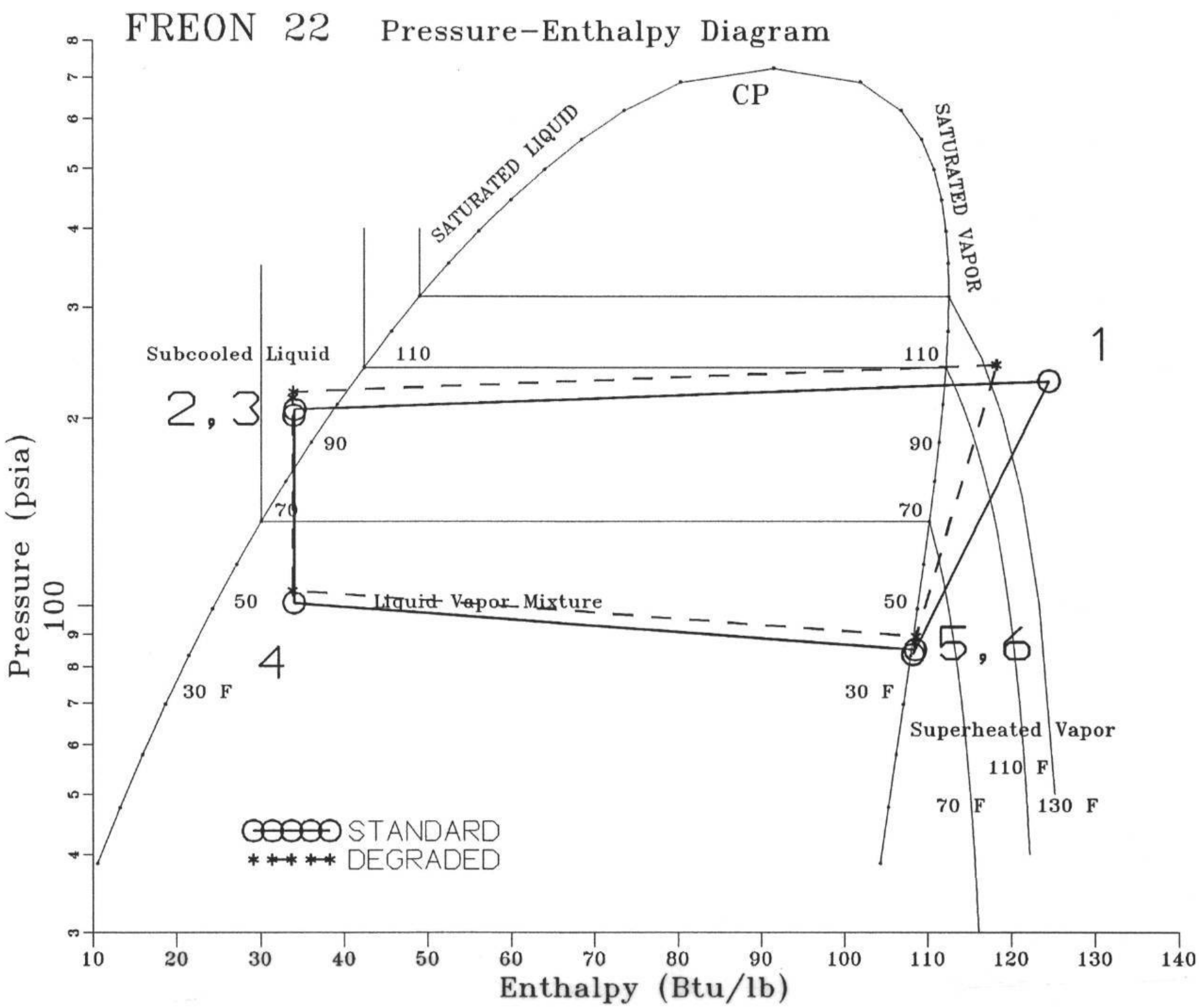


Figure B-6.2.2. P-H chart for 20% overcharge

STANDARD RUN				20 % OVERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	81.1	50	0.0114
15.	59.2	89	0.0095	15.	59.7	90	0.0098
16.	92.6		0.0110	16.	94.3		0.0114
EVAP. AIR FLOW			1137	1139 CFM			
COOLING CAPACITY			35.01	36.51 KBtu/hr			

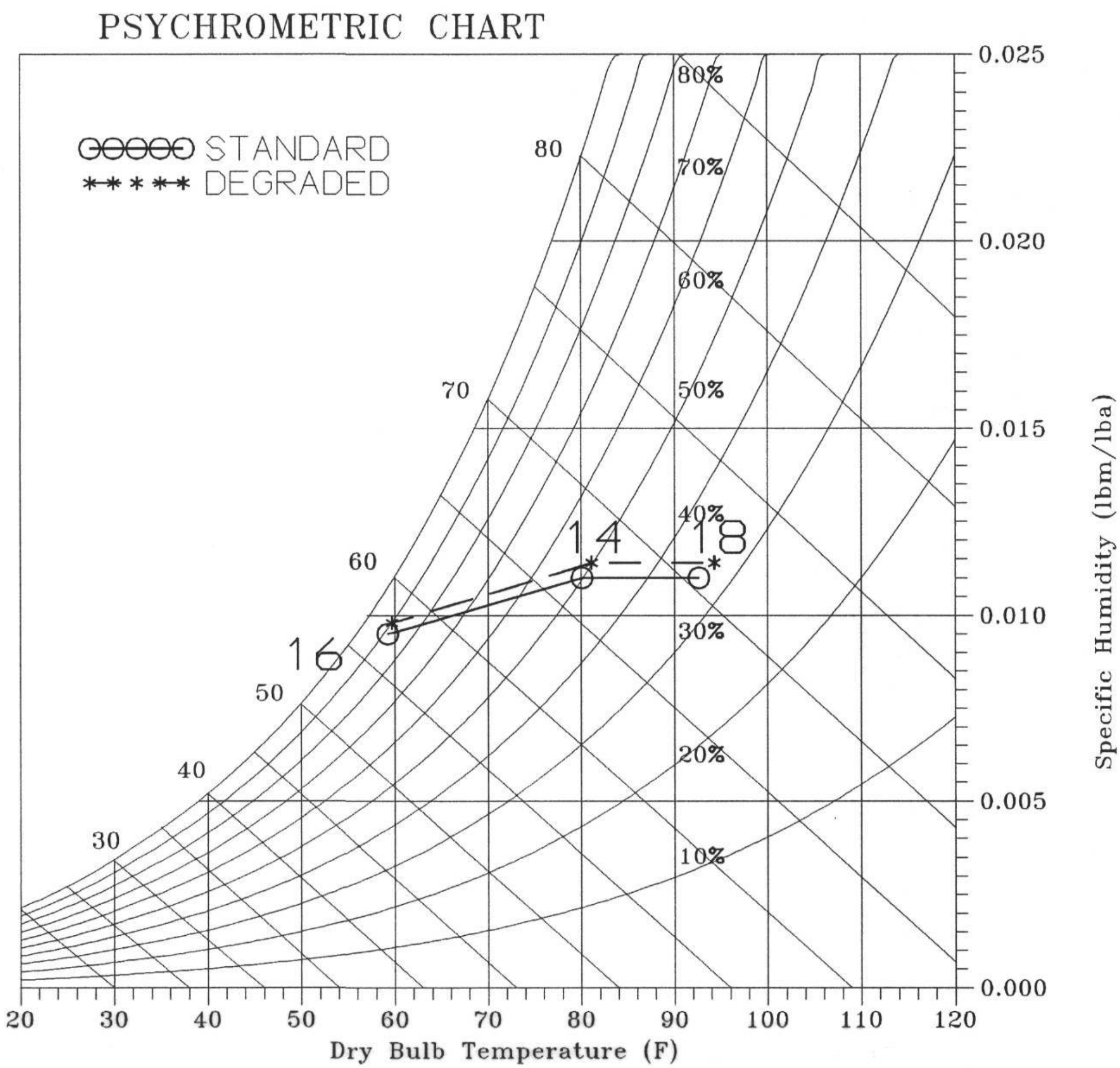


Figure B-6.2.3. Psychrometric chart for 20% overcharge

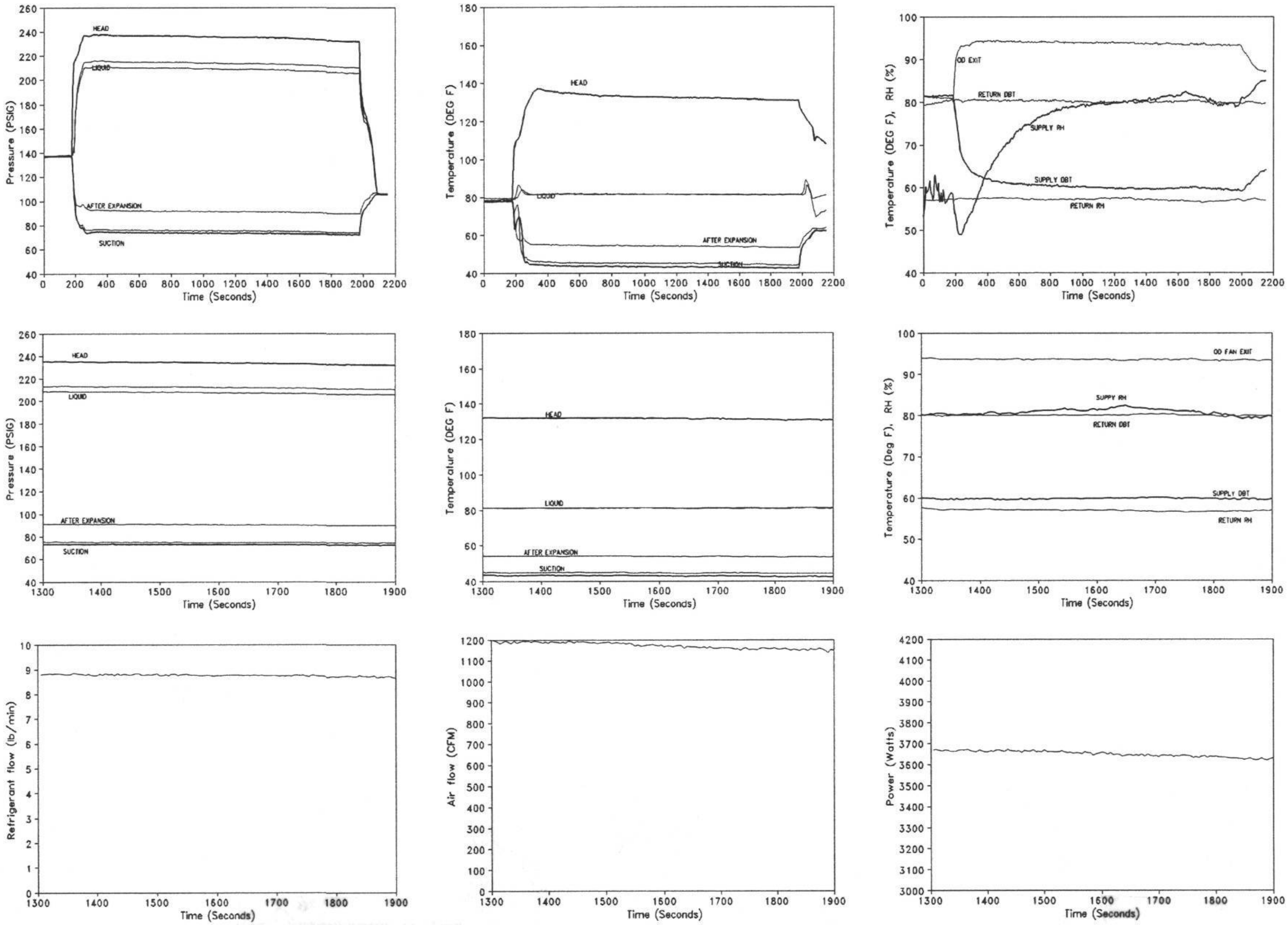


Figure B-6.3.1. Time series plot for 30% overcharge degraded test

	STANDARD RUN			30 % OVERCHARGE		
	P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1.	230	163	124.5	249	132	117
2.	208	83	34.2	227	81	33.5
3.	202	83	34.1	222	81	33.4
4.	101	51	34.1	105	54	33.4
5.	85	43	108.5	89	45	108.6
6.	84	41	108.2	88	43	108.4
REF. CAP			36.8 KBtu/hr (3.07 TONS)			39.51 KBtu/hr (3.29 TONS)
REF. FLOW			8.24 LB/MIN			8.76 LB/MIN

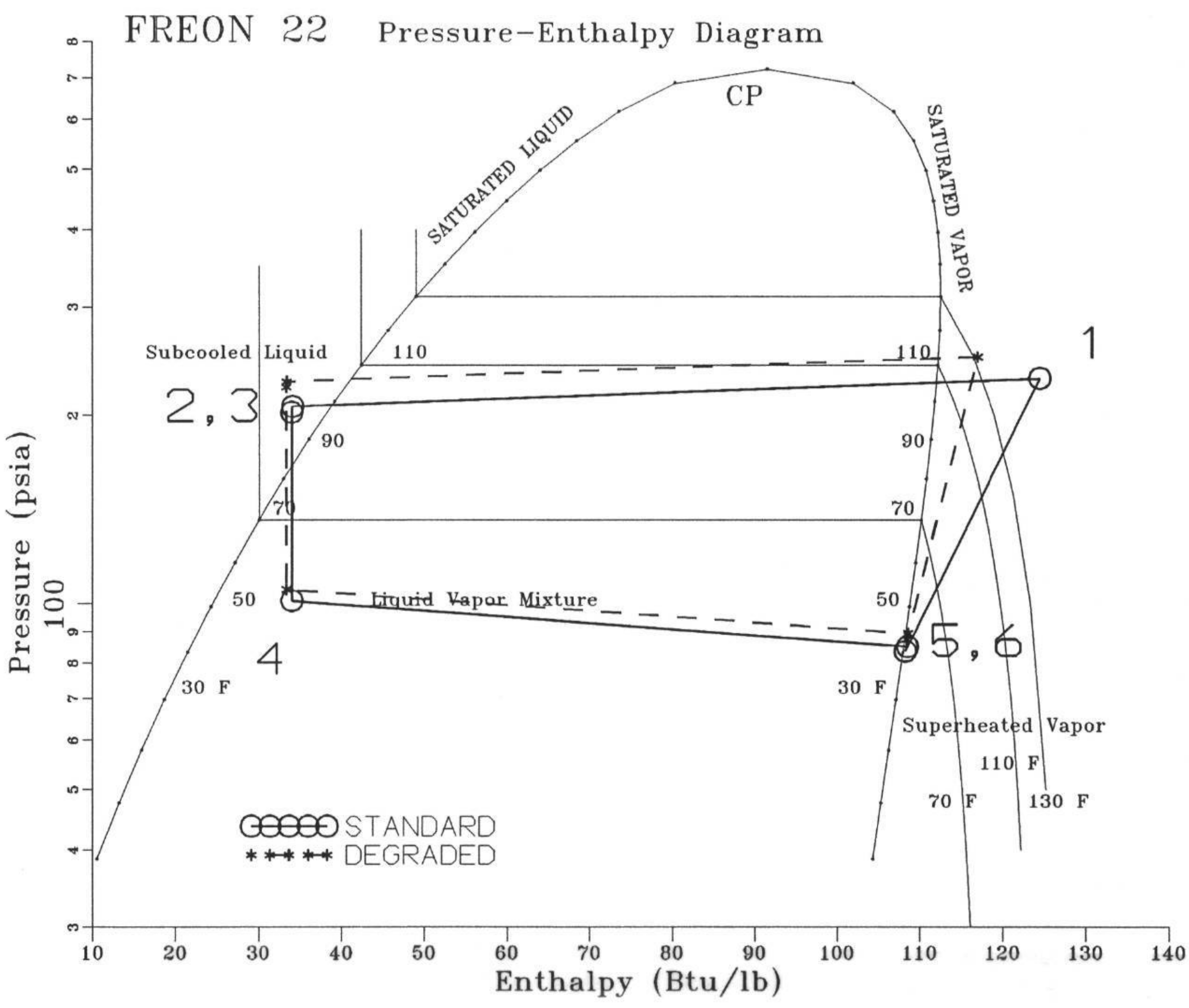


Figure B-6.3.2. P-H chart for 30% overcharge

STANDARD RUN				30 % OVERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	80.0	51	0.0111
15.	59.2	89	0.0095	15.	59.8	90	0.0098
16.	92.6		0.0110	16.	93.5		0.0111
EVAP. AIR FLOW			1137	1156 CFM			
COOLING CAPACITY			35.01	33.7 KBtu/hr			

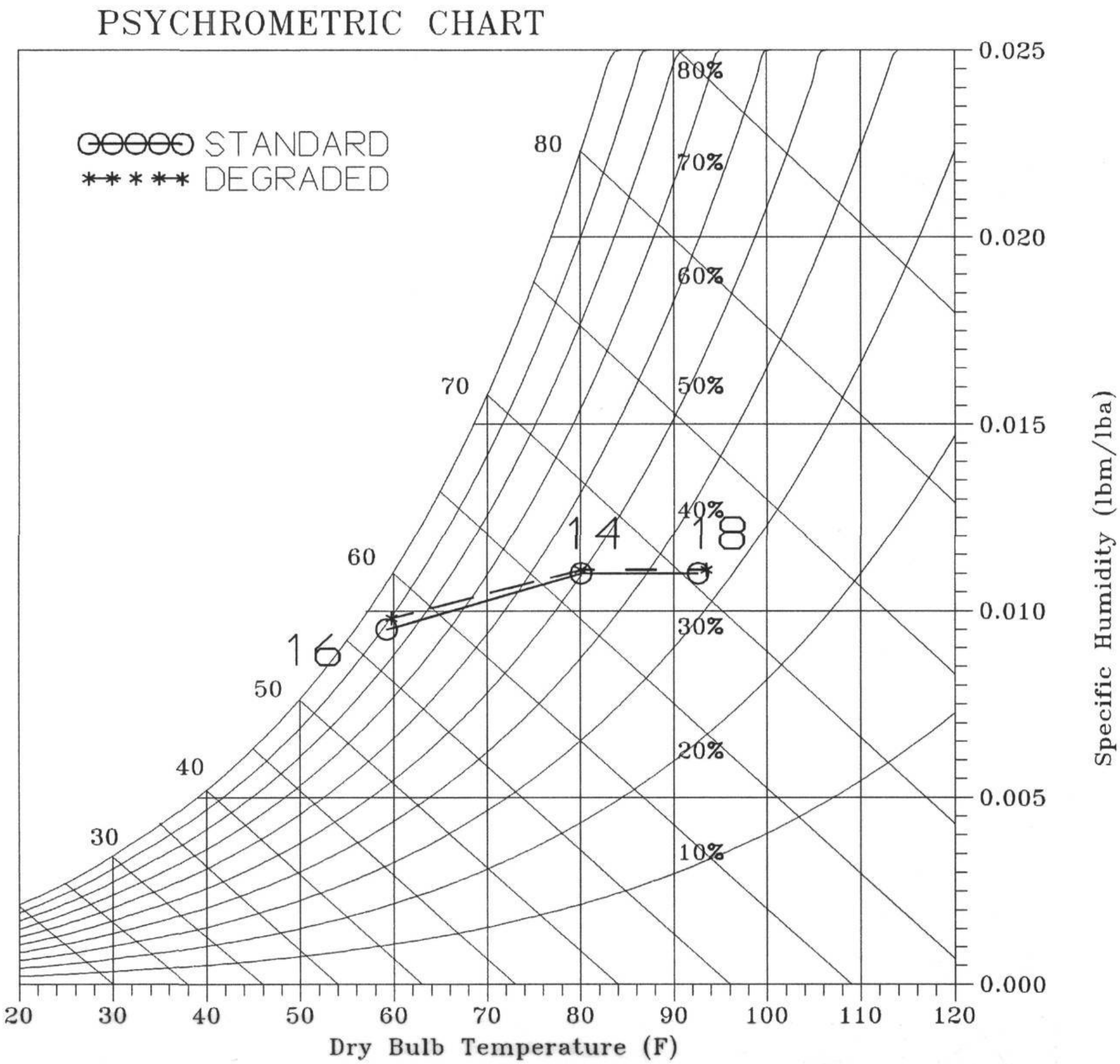


Figure B-6.3.3. Psychrometric chart for 30% overcharge

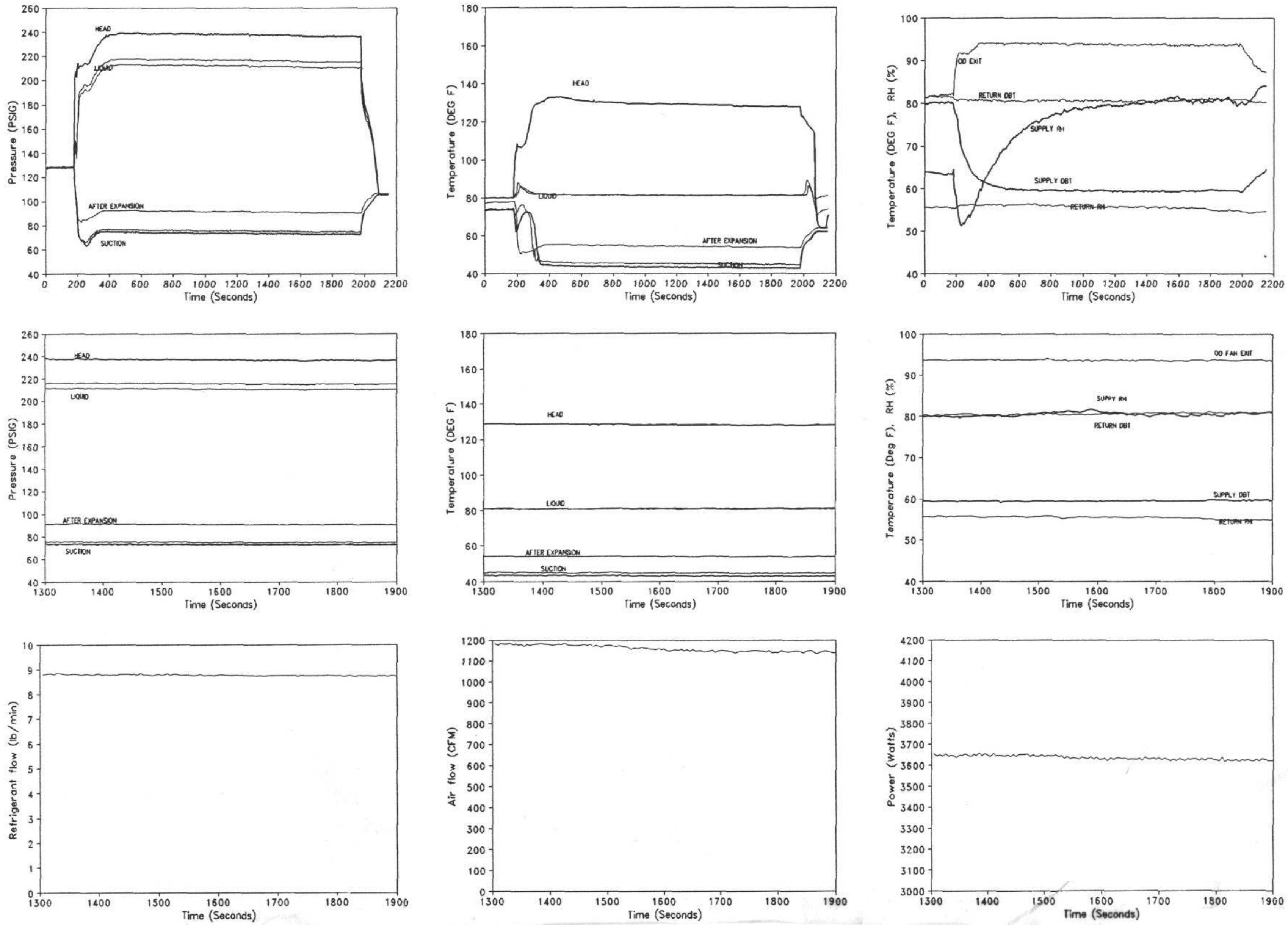


Figure B-6.4.1. Time series plot for 40% overcharge degraded test

STANDARD RUN			40 % OVERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 252	128	116.1
2. 208	83	34.2	2. 230	81	33.4
3. 202	83	34.1	3. 227	81	33.4
4. 101	51	34.1	4. 106	54	33.4
5. 85	43	108.5	5. 90	45	108.6
6. 84	41	108.2	6. 88	43	108.5
REF. CAP		36.8 KBtu/hr (3.07 TONS)			39.58 KBtu/hr (3.30 TONS)
REF. FLOW		8.24 LB/MIN			8.77 LB/MIN

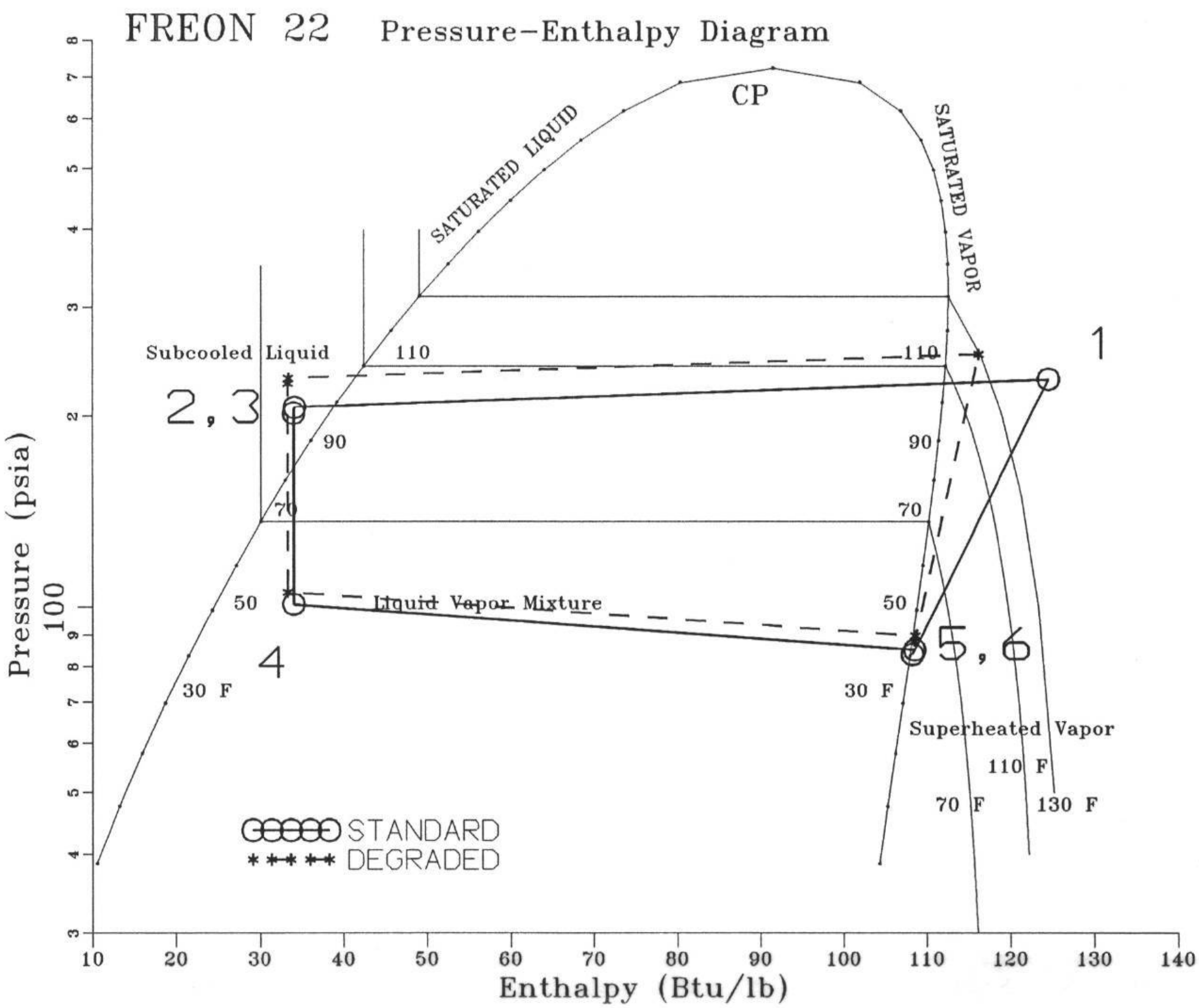


Figure B-6.4.2. P-H chart for 40% overcharge

STANDARD RUN	40 % OVERCHARGE
DBT(F) RH(%) W(lbm/lb air)	DBT(F) RH(%) W(lbm/lb air)
13. 80.1 50 0.0110	13. 80.6 49 0.0110
15. 59.2 89 0.0095	15. 59.4 90 0.0097
16. 92.6 0.0110	16. 93.6 0.0110
EVAP. AIR FLOW 1137	1143 CFM
COOLING CAPACITY 35.01	34.6 KBtu/hr

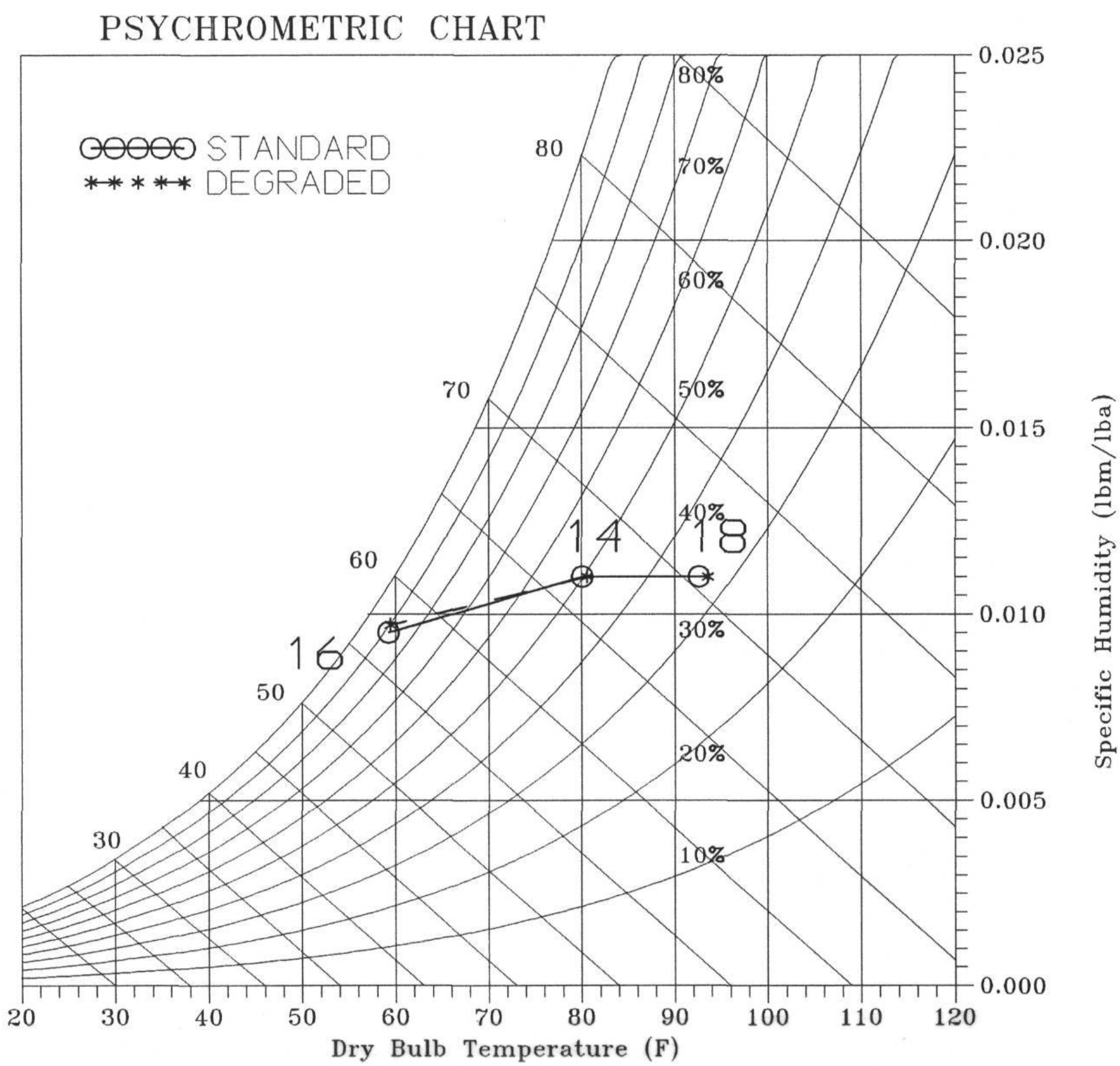


Figure B-6.4.3. Psychrometric chart for 40% overcharge

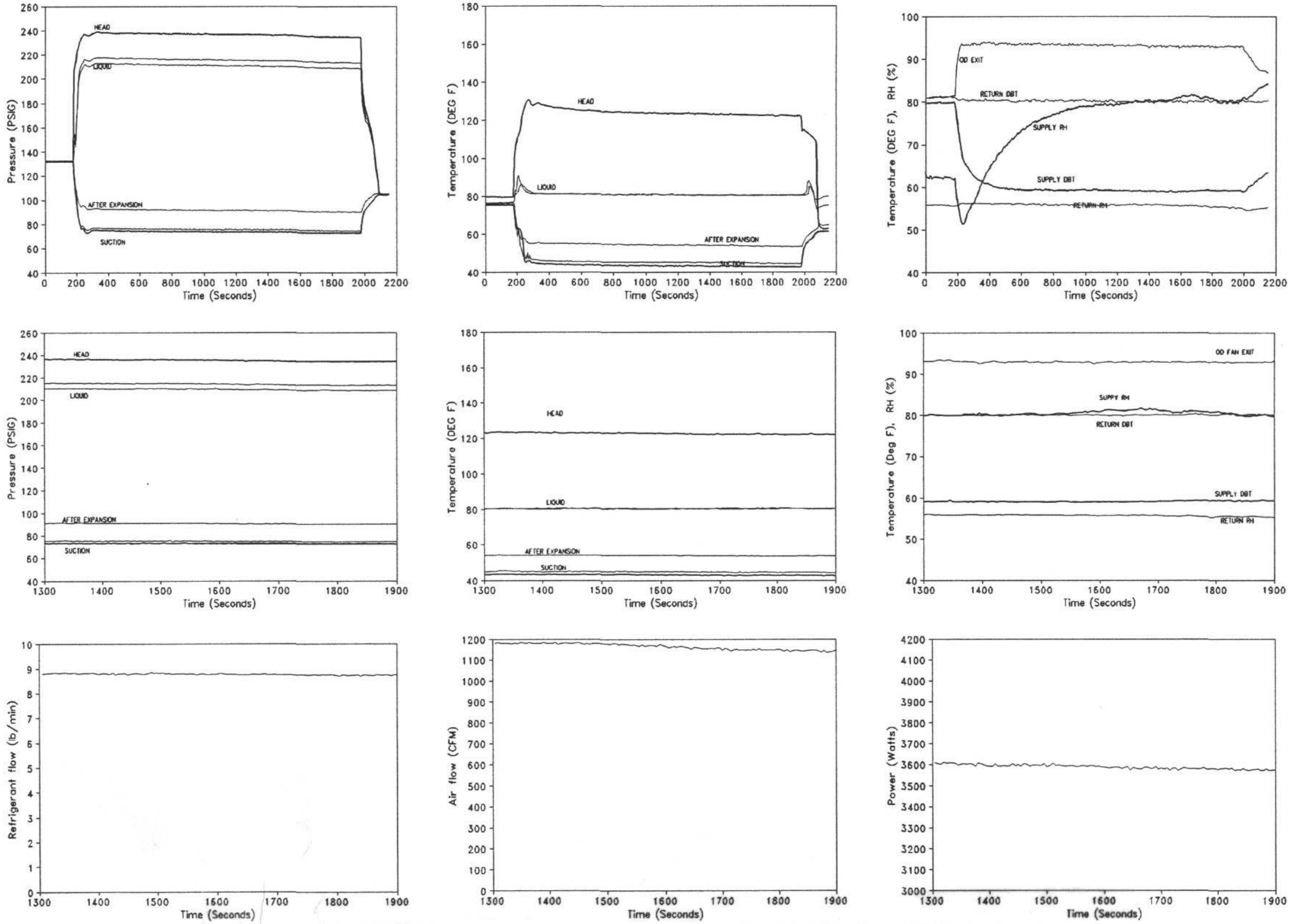


Figure B-6.5.1. Time series plot for 50% overcharge degraded test

STANDARD RUN			50 % OVERCHARGE		
P(psia)	T(F)	H(Btu/lb)	P(psia)	T(F)	H(Btu/lb)
1. 230	163	124.5	1. 250	123	114.9
2. 208	83	34.2	2. 229	81	33.3
3. 202	83	34.1	3. 224	80	33.2
4. 101	51	34.1	4. 105	54	33.2
5. 85	43	108.5	5. 90	45	108.6
6. 84	41	108.2	6. 88	43	108.5
REF. CAP		36.7 KBtu/hr (3.07 TONS)			39.65 KBtu/hr (3.30 TONS)
REF. FLOW		8.24 LB/MIN			8.77 LB/MIN

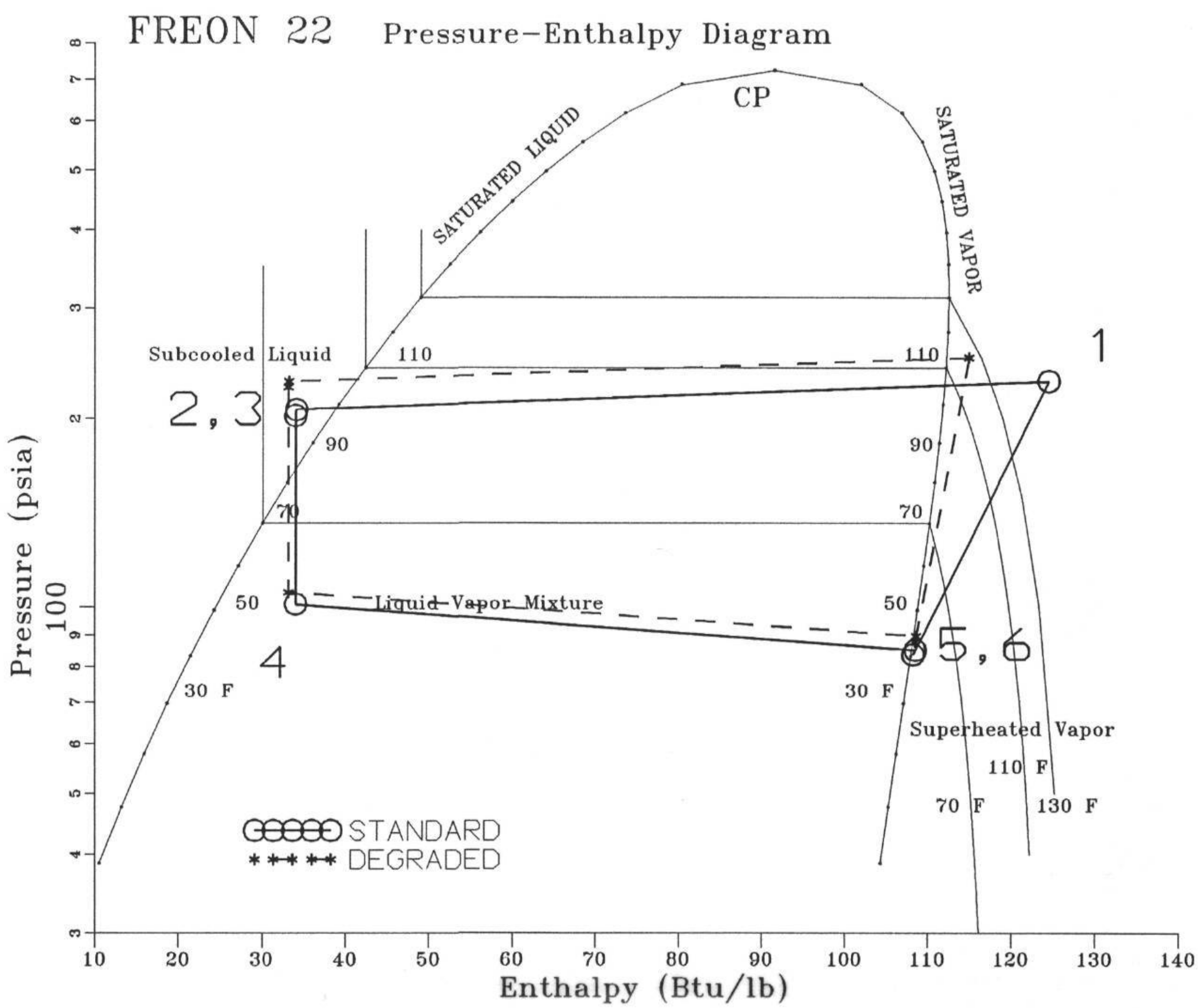


Figure B-6.5.2. P-H chart for 50% overcharge

STANDARD RUN				50 % OVERCHARGE			
	DBT(F)	RH(%)	W(lbm/lb air)		DBT(F)	RH(%)	W(lbm/lb air)
13.	80.1	50	0.0110	13.	80.1	50	0.0109
15.	59.2	89	0.0095	15.	59.1	90	0.0095
16.	92.6		0.0110	16.	92.9		0.0109
EVAP. AIR FLOW			1137	1147 CFM			
COOLING CAPACITY			35.01	34.45 KBtu/hr			

PSYCHROMETRIC CHART

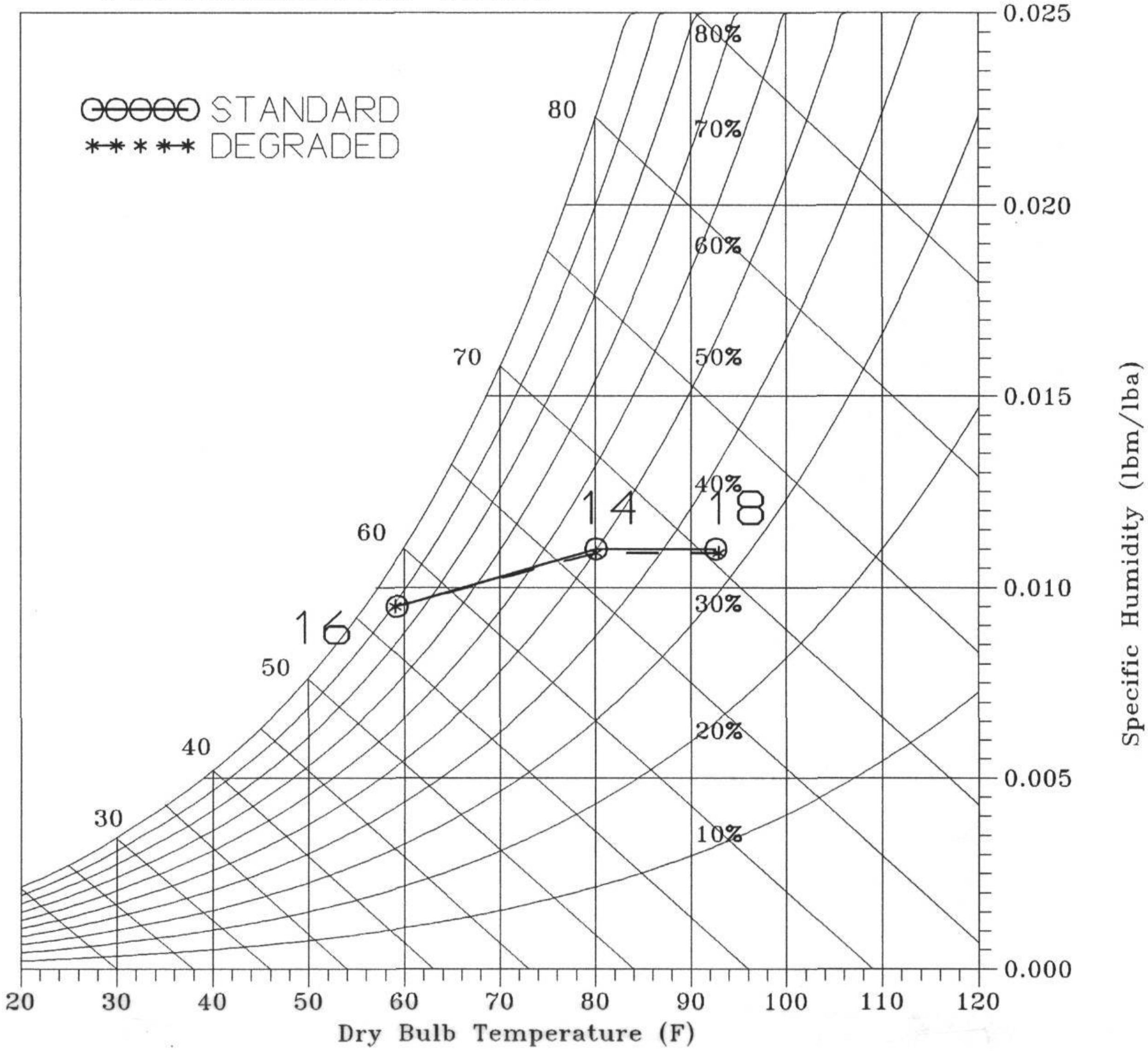


Figure B-6.5.3. Psychrometric chart for 50% overcharge

DIAGNOSTICS PROCEDURE

QUANTITIES TO BE MEASURED

1. Ambient DBT
2. Evaporator supply air temperature at inlet
3. Evaporator supply air temperature at outlet
4. Condenser air discharge temperature
5. Head temperature (Discharge line)
6. Liquid line temperature (Condenser outlet)
7. Refrigerant temperature at evaporator inlet
8. Suction temperature

DATA REDUCTION

1. Measure the drop in air temperature across evaporator

   TEE1 = (Evaporator air inlet temperature - Evaporator air outlet temperature)

2. Measure the rise in air temperature across condenser

   TEE2 = (Condenser air discharge temperature - Ambient DBT)

3. Measure the difference between Head and ambient temperatures

   TEE3 = (Head temperature - Ambient DBT)

4. Measure the difference between Evaporator inlet air temperature and Suction temperature

   TEE4 = (Evaporator air inlet temperature - Suction temperature)

5. Measure the difference between Ambient DBT and Liquid line temperature

   TEE5 = (Liquid line temperature - Ambient DBT)

6. Measure the difference between Supply air temperature at evaporator outlet and refrigerant temperature at evaporator inlet

   TEE6 = (Supply air DBT - Saturated refrigerant temperature at evaporator inlet)

NORMAL TEMPERATURE LEVELS

Normal temperature levels are defined based on standard runs. However, these normals are recommended only at 80 F return and outdoor temperatures.

TEE1 =	26 - 15 F
TEE2 =	15 - 12 F
TEE3 =	100 - 60 F
TEE4 =	42 - 30 F
TEE5 =	7 - 2 F
TEE6 =	12 - 4 F

DEGRADED CONDITION TEMPERATURE LEVELS

FAULT: 1. Reduced Evaporator air flow

Symptoms:
a. TEE1 above normal
b. TEE2 below normal
c. TEE3 below normal
d. TEE4 above normal

FAULT: 2. Reduced OD Fan air flow

a. TEE2 above normal
b. TEE5 above normal

FAULT: 3. Air in System

a. TEE3 above normal

FAULT: 4. Restriction Before Expansion Valve

a. TEE1 below normal
b. TEE2 below normal
c. TEE3 above normal
d. TEE4 below normal

e. TEE5 below normal

f. TEE6 above normal

FAULT: 5. Restriction After Expansion Valve

a. TEE6 below normal

FAULT: 6. Restriction in Suction Line

a. TEE4 below normal

b. TEE6 below normal

FAULT: 7. Restriction in Discharge Line

a. TEE3 above normal

FAULT: 8. Undercharge

a. TEE1 below normal

b. TEE2 below normal

c. TEE3 above normal

d. TEE4 below normal

e. TEE5 above normal

f. TEE6 above normal

FAULT: 9. Overcharge

a. TEE3 below normal

b. TEE5 below normal

RANGE	EVP TEM DROP (DEG F)	CON TEM RISE (DEG F)	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
MINIMUM	26	15	100	42	7	15	15	12
MAXIMUM	15	12	60	30	2	5	5	4

Table C-1.1. Normal temperature range for diagnostic temperature analysis

TEST	PR. RATIO	HEAD T - DBT	DBT - SUCTION	LIQUID - DBT	SUPERHEAT	SUBCOOL	SUPPLY - REF. T
STANDARD	2.79	81	39	5	0.15	16	8
25% CUT IN EVAP AIR	2.78	57	42	4	-0.01	13	7
50% CUT IN EVAP AIR	2.93	58	46	3	0.05	14	6
STANDARD	2.87	96	36	4	3.67	19	10
75% CUT IN EVAP AIR	3.18	34	55	9	-0.24	3	3
90% CUT IN EVAP AIR	3.88	12	70	5	-0.53	2	13

Table C-1.2. Diagnostic temperature analysis for reduced evaporator air flow

TEST	PR. RATIO	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
STANDARD	2.80	90	36	4	4.65	16	9
SHUTOFF OD AIR FLOW	3.27	89	37	28	0.13	10	4

Table C-1.3. Diagnostic temperature analysis for blocked outdoor fan air flow

TEST	PR. RATIO	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
STANDARD	2.75	59	39	4	-0.04	16	5
Non-Condensable Gases	2.90	89	40	2	-0.16	23	4

TEST	PR. RATIO	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
STANDARD	2.86	74	44	6	-0.10	12	6
LIQUID LINE BEFORE EX VALVE	4.28	145	2	0	70.92	13	50
STANDARD	2.81	86	39	4	0.28	16	8
LIQUID LINE AFTER EX VALVE	2.89	104	19	4	23.56	17	-19
STANDARD	2.81	91	36	3	5.18	16	9
SUCTION LINE	2.94	48	49	8	-0.10	5	6
STANDARD	2.82	94	33	3	8.37	16	9
DISCHARGE LINE	3.36	110	38	3	1.62	15	9

Table C-1.5. Diagnostic temperature analysis for restriction in refrigerant lines

TEST	PR. RATIO	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
STANDARD	2.75	82	39	3	0.36	15	8
10% UNDERCHARGE	2.78	85	40	7	0.85	12	8
20% UNDERCHARGE	2.72	96	25	11	16.91	4	9
30%UNDERCHARGE	2.86	113	6	12	41.43	0	21
40% UNDERCHARGE	3.07	123	2	9	52.42	0	30
50% UNDERCHARGE	3.34	133	1	8	59.65	0	36

Table C-1.6. Diagnostic temperature analysis for undercharge

TEST	PR. RATIO	HEAD T - DBT (DEG F)	DBT - SUCTION (DEG F)	LIQUID - DBT (DEG F)	SUPERHEAT (DEG F)	SUBCOOL (DEG F)	SUPPLY - REF. T (DEG F)
STANDARD	2.75	82	39	3	0.36	15	8
10% OVERCHARGE	2.78	63	39	3	0.00	18	6
20% OVERCHARGE	2.78	55	38	2	-0.07	21	6
30% OVERCHARGE	2.84	51	37	1	-0.05	24	6
40% OVERCHARGE	2.87	48	38	0	-0.06	26	5
50% OVERCHARGE	2.85	43	37	0	-0.09	26	5

Table C-1.7. Diagnostic temperature analysis for overcharge