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Steady and Cyclic Performance Testing of Packaged R-410A Units

Submitted to:
Proctor Engineering Group
Southern California Edison Co.

Final Report
November, 2004

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1. INTRODUCTION

1.1 Research Objectives

This project focused on investigating the steady and cyclic performances of two R-410A unitary systems over a range of indoor air flow rates, refrigerant charge levels, outdoor temperatures and indoor humidities. Information about the units (Unit #1 and #2) is presented in Table 1-1.

Table 1-1: Information of the Test Units

Information	Unit #1	Unit #2
Capacity	3 tons	3 tons
Type	Split	Packaged
Refrigerant	R-410A	R-410A
Rated SEER	12	12
Expansion device	Thermal expansion valve	Fixed-area orifice
Compressor	Reciprocating piston Model: H89B303ABC (Bristol)	Scroll Model: ZP32K3E-PFV (Copeland)
Condensing coil	Single row; slit fin; Two paralleled circuits combined to one subcooled circuit; 25 fins/in	Two rows; slit fin; Two paralleled circuits without subcooled circuit; 17 fins/in
Evaporating coil	Three rows; Slit fin; Five paralleled circuits; 14.5 fins/in	Three rows; Slit fin; Four paralleled circuits; 15 fins/in
Condenser fan	1 PH AC; 160 W; 2885 cfm	1 PH AC; 160 W; 2350 cfm
Evaporator blower	1 PH AC; 450 W; 1200 cfm	1 PH AC; 500 W; 1200 cfm
Suction line	Length: 15 ft; OD: 3/4 in	Length: 4.9 ft; OD: 3/4 in
Discharge line	Length: 4.6 ft; OD: 5/8 in	Length: 1.5 ft; OD: 5/8 in
Liquid line	Length: 15 ft; OD: 3/8 in	Length: 4.7 ft; OD: 3/8 in
Factory charge	6.25 lbs	6.9 lbs

1.2 Tasks

Steady-State Tests

Steady-state tests were performed over a range of outdoor temperatures and indoor humidities that are summarized in Table 1-2.

Table 1-2: Steady-State Test Conditions for Units #1 and #2

Test No.	Outdoor DB [˚F]	Indoor DB [˚F]	Indoor WB [˚F]	Charge [%]	Indoor Airflow [cfm/ton]
1	115	80	67	100%	400
2	95	80	67	100%	400
3	115	80	63	100%	400
4	115	80	63	120%	400
5	115	80	63	90%	400
6	115	80	63	80%	400
7	115	80	63	70%	400
8	115	80	63	100%	300
9	115	80	63	100%	350
10	115	80	63	100%	450
11	115	80	63	100%	550
12	125	80	63	100%	400

Cyclic Tests

Cycling tests were performed over a range of outdoor temperatures and indoor humidities that are summarized in Table 1-3.

Table 1-3: Cyclic Test Conditions for Units #1 and #2

Test No.	Outdoor DB [˚F]	Indoor DB [˚F]	Indoor WB [˚F]	Charge [%]	Indoor Airflow [cfm/ton]
1	115	80	<57 (dry)	100%	400
2	82	80	<57 (dry)	100%	400
3 (wet cyclic)	82	80	67 (wet)	100%	400

2. EXPERIMENTAL SETUP AND INSTRUMENTATION

2.1 Test Facilities and Instrumentation

The experiments were carried out in ASHRAE standard psychometric chambers. These chambers consist of two insulated rooms. The temperature and humidity are independently controlled within each room to the desired operating conditions. The indoor room is equipped with an ASHRAE standard air measurement box. The evaporator air stream is connected to this box to yield a measurement of the air flow rate. The evaporator air flow rate is altered by a variable speed blower inside the box. A Dwyer model 603A-2 transducer measures the pressure drop across the nozzle, while a Dwyer model 603A-3 transducer measures the pressure at the nozzle inlets. Three 1000 Ω platinum TEP series probes, RTDs, measure the air temperature near the inlet of the nozzles. A former student analyzed the uncertainty in measuring the air flow rate, which is ±10g/s at a dry air mass flow rate of 1.10 kg/s. Thus, the accuracy of the evaporator air flow rate measurement is assigned to be ±0.91%.

All temperature measurements were performed with Type T (copper-constantan) thermocouples, which are probe-type or wire-type. These thermocouples were calibrated using an ice bath and boiling water to make sure the readings were within 0.5 ˚F of the known values. However, two-point calibration is not enough to guarantee a good linearity and thus, the standard accuracies of T-type thermocouples were still used as references. The accuracies are ±0.5 ˚C for probe-type and ±1 ˚C for wire-type thermocouples, as specified by the manufacturer.

Low pressure transducers and high pressure transducers were installed in the refrigerant flow to measure the refrigerant pressures of interested state points. The low pressure transducers (OMEGA PX176-50S5V) ranged from 0 to 500 psig, which were installed at the evaporator side. The high pressure transducers (OMEGA PX176-50S5V) ranged from 0 to 1000 psig, which were installed at the condenser side. A differential pressure transducer (SENSOTEC HL-Z), having a range from 0 to 50 psig, was installed across the condenser to measure the refrigerant side pressure drop. A second differential pressure transducer (OMEGA PX2300-25DI), having a range from 0 to 25 pig, was installed across the evaporator circuit. The pressure transducers and differential pressure transducers were calibrated by way of a dead weight tester. For both the low pressure transducers and high pressure transducers, the manufacturer claims an accuracy of ±1%. However, the low pressure transducers were calibrated to an accuracy of ±0.7%. The high pressure transducers were calibrated to an accuracy of ±0.8%. The differential pressure transducer across the condenser was calibrated to an accuracy of ±2.4%, which led to a maximum absolute deviation of 0.36 psig. The differential pressure transducer across the evaporator was calibrated to an accuracy of ±6.3%, which led to a maximum absolute deviation of 0.32 psig.

The refrigerant temperatures and pressures were measured at state locations at the inlet and outlet of each component throughout the system, as indicated in Figure 2-2 and Figure 2-6. The refrigerant temperatures were measured using sheathed probe thermocouples inserted into the refrigerant flow, or wire thermocouples soldered on a tube surface. As presented in Figure 2-4 and Figure 2-8, there are multiple branches in each evaporator. The differential pressure transducer, which measured the pressure drop across each evaporator, was connected to the entrance of each parallel evaporator branch by hand valves. In this way, the pressure drop across each branch could be measured separately by opening one hand valve while closing the others.

An array of thermocouples was mounted from top to bottom on the compressor shell. A grid of eight thermocouples was installed at the inlet and exit of the evaporator air flow. A grid of twelve thermocouples was installed at the inlet of the condenser air flow. A grid of thermocouples, eight for the 3-ton split unit or six for the 3-ton packaged unit, was installed at the exit of the condenser air flow. In addition, wire thermocouples were soldered to the tube bends of a condenser and an evaporator as indicated in Figure 2-3, 2-4, 2-7 and 2-8, so as to measure local refrigerant temperatures.

A mass flow meter (Micro Motion model F25 Coriolis-type sensor) was used for measuring the refrigerant mass flow rate. The manufacturer claims an accuracy of ±2%, while the calibration demonstrated that the accuracy is ±0.6%. Two sight glasses were installed upstream and downstream of the Micro Motion mass flow meter to identify the case of two-phase flow.

A General Eastern model D-2 chilled mirror sensor, having an accuracy of ±0.2 ˚C, was used to measure the inlet and outlet dew point of the evaporator air flow. An air valve was used to alter the sensing location. In addition, a General Eastern model M-2 chilled mirror sensor was used to measure the outlet dew point, which has a response rate of 1.5 ˚C/s and an accuracy of ±0.2 ˚C. The extra humidity sensor was especially used for measuring the outlet dew point during a wet cyclic test. Both of these two chilled mirror sensors were calibrated by the manufacturers. Atmospheric pressure was measured using a mercury barometer, with an accuracy of ±0.03 kPa.

The compressor power, outdoor fan power and indoor blower power were measured with Exceltronic Series watt transducers, with an accuracy of ±10 W. It has to be mentioned that the indoor blower power was not measured during the steady-state and cyclic tests because the air flow rate was controlled using an external variable speed fan and the indoor blower was disconnected. However, the performance of each indoor blower was investigated separately without running the compressor and outdoor fan. The voltage and current were measured with a clamp meter. The readings of the clamp meter were not connected to the data acquisition system, which were recorded manually.

The airside pressure drop across each evaporating coil was measured using a Setra model-264 differential pressure transducer, which was installed as indicated by DP1 in Figure 2-1 and 2-5. The differential pressure transducer was calibrated by the manufacturer.

The mass of refrigerant charged to the system was measured with an Acculab model SV-30 digital scale accurate to ±0.005 kg. The condensed water from the evaporator was collected during some tests, and the test duration time was recorded. In this way, the mass flow rate of the condensed water could be determined.

The measured data was recorded using a HP 75000 Series B mainframe data acquisition system. For steady-state performance tests, the sampling interval was within 10 seconds. For cyclic tests, the sampling interval was set within 5 seconds. However, the actual sampling intervals for the cyclic tests could be varied from 4 to 6 seconds, due to the limitation of the acquisition system.

2.2 Test Procedures

During steady-state tests, at first, the temperature and humidity in the psychrometric chambers were controlled at a desired condition for one hour. Then, the data acquisition system was turned on. At the beginning of the data acquisition, the General Eastern model D-2 chilled mirror sensor was switched to measure the evaporator outlet air dew point, and the data was recorded for 10~20 minutes. After that, the humidity sensor was switched to measure the evaporator inlet air dew point for 40~60 minutes, which was the major data recording period. During the data acquisition, the differential pressure transducer across the evaporator was placed across evaporator Branch #1. The pressure drop across other evaporator branches was measured and recorded separately for 100 seconds after the major data recording period. The voltage and current of the compressor were measured four times during a test. The power, voltage and current of each indoor blower were measured in a separate test. The current and voltage of each outdoor fan were investigated in a separate test as well. The steady-state tests simultaneously employed the Air-Enthalpy Method and Refrigerant Enthalpy Method. The values calculated from these two methods have to agree within ±6% to consider a test valid. For steady-state measurements, mass balance of water condensate was performed in some tests for each test unit.

During all cyclic tests, the indoor air flow was always on. After the psychrometric chambers reached a desired condition for one hour, the data acquisition system was turned on. During the following period of data acquisition, each unit ran for four complete cycles, with 6 minutes on and 24 minutes off. The General Eastern model D-2 chilled mirror sensor was used to measure the evaporator inlet air dew point. For a wet cyclic test, a General Eastern model M-2 chilled mirror sensor was used to measure the evaporator outlet air dew point. The cyclic performance tests employed the Air-Enthalpy Method only.

2.3 Experimental Setup

2.3.1 Experimental Setup of 3-ton Split Unit

Figure 2-1 presents the airflow pathway of the 3-ton R-410A split unit. Figure 2-2 presents the system setup. Figure 2-3 and Figure 2-4 present the measurements on the condenser and evaporator, respectively.

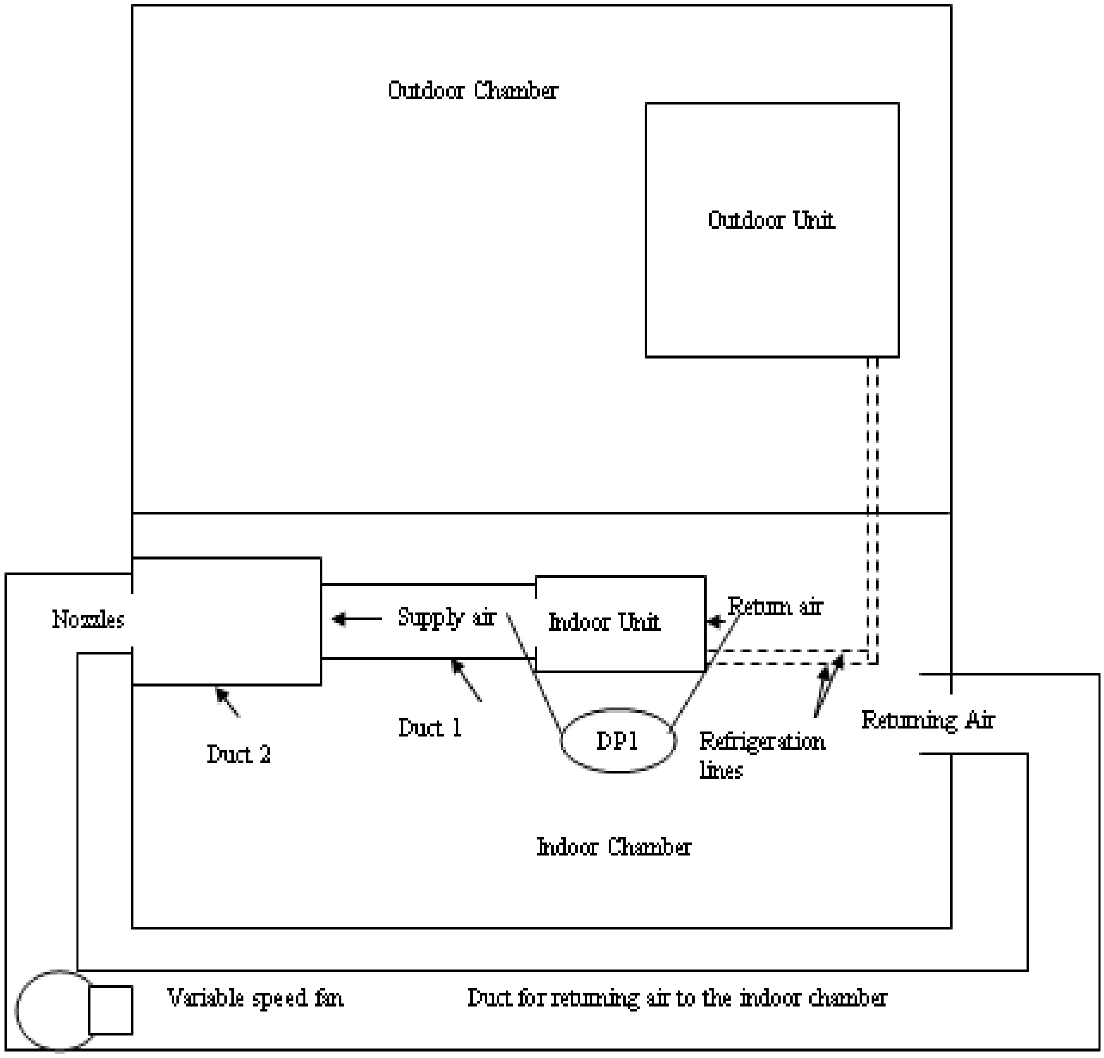
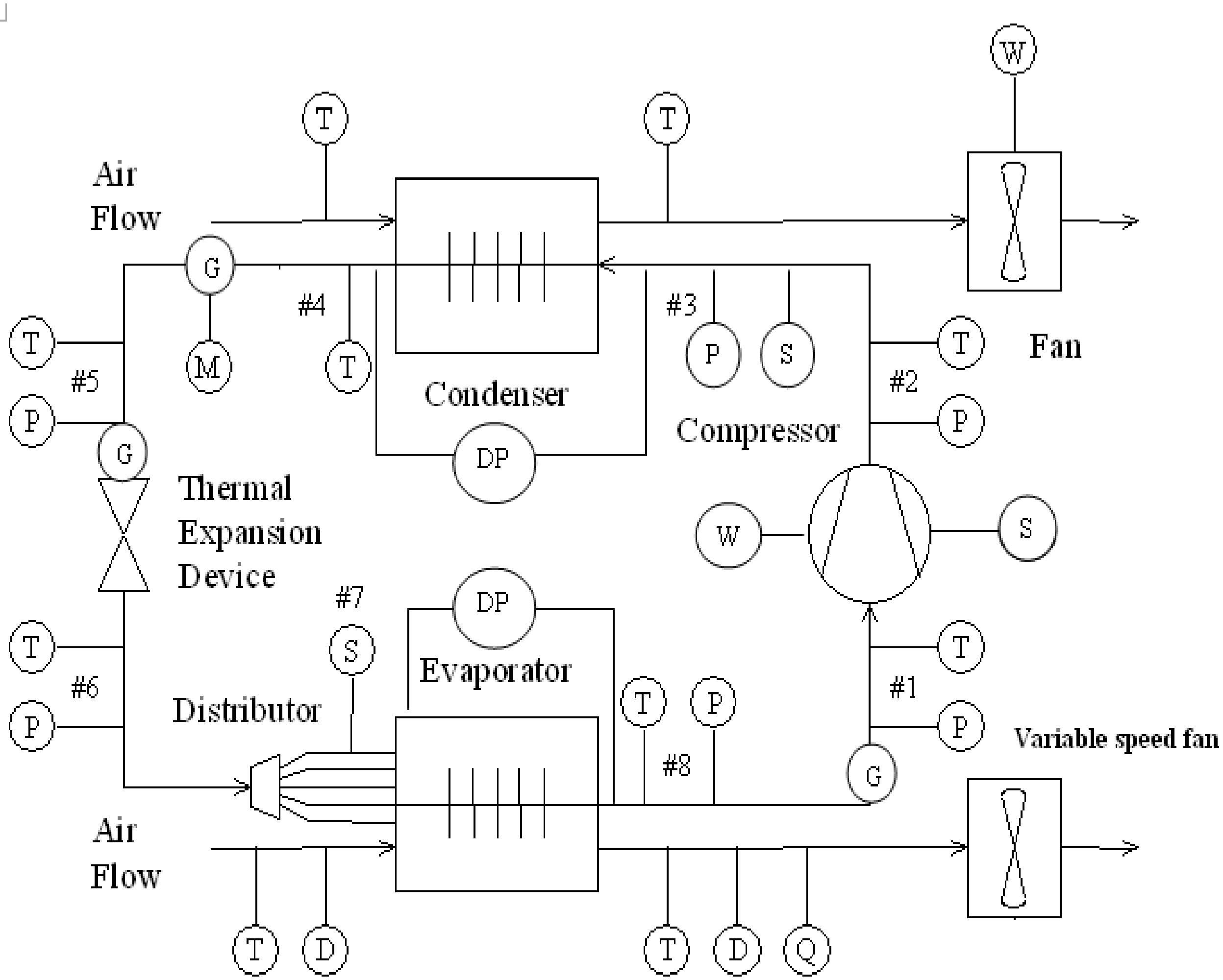


Figure 2-1: Airflow Pathway of the 3-ton R-410A Split Unit within the Psychrometric Chambers



Measurements:
D---Dew point
DP---Differential pressure drop
G---Sight glass
M---Mass flow rate
P---Pressure
Q---Air flow rate
S---Surface temperature
T---Refrigerant or air temperature
W---Power

Figure 2-2: System Setup of the 3-ton R-410A Split Unit

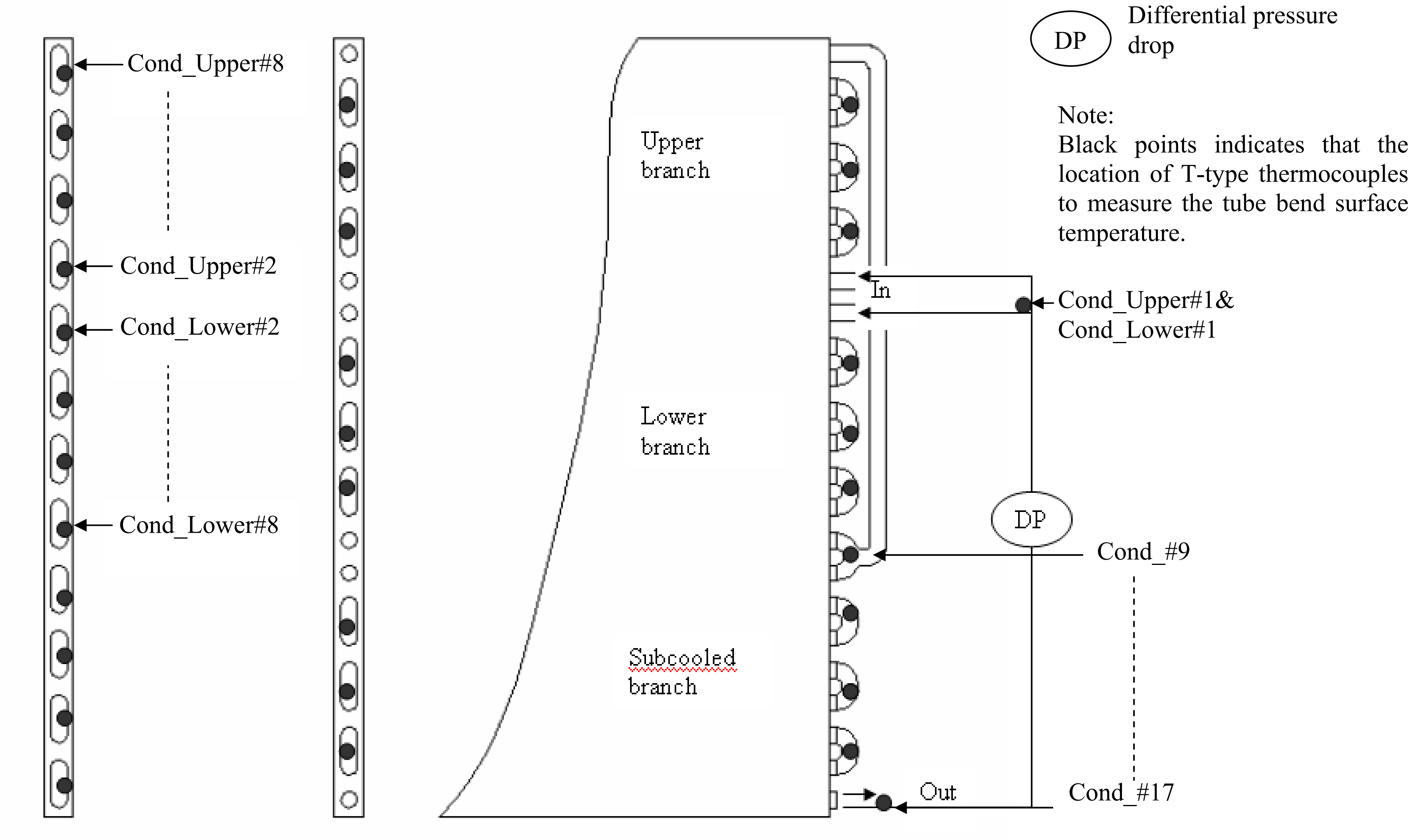
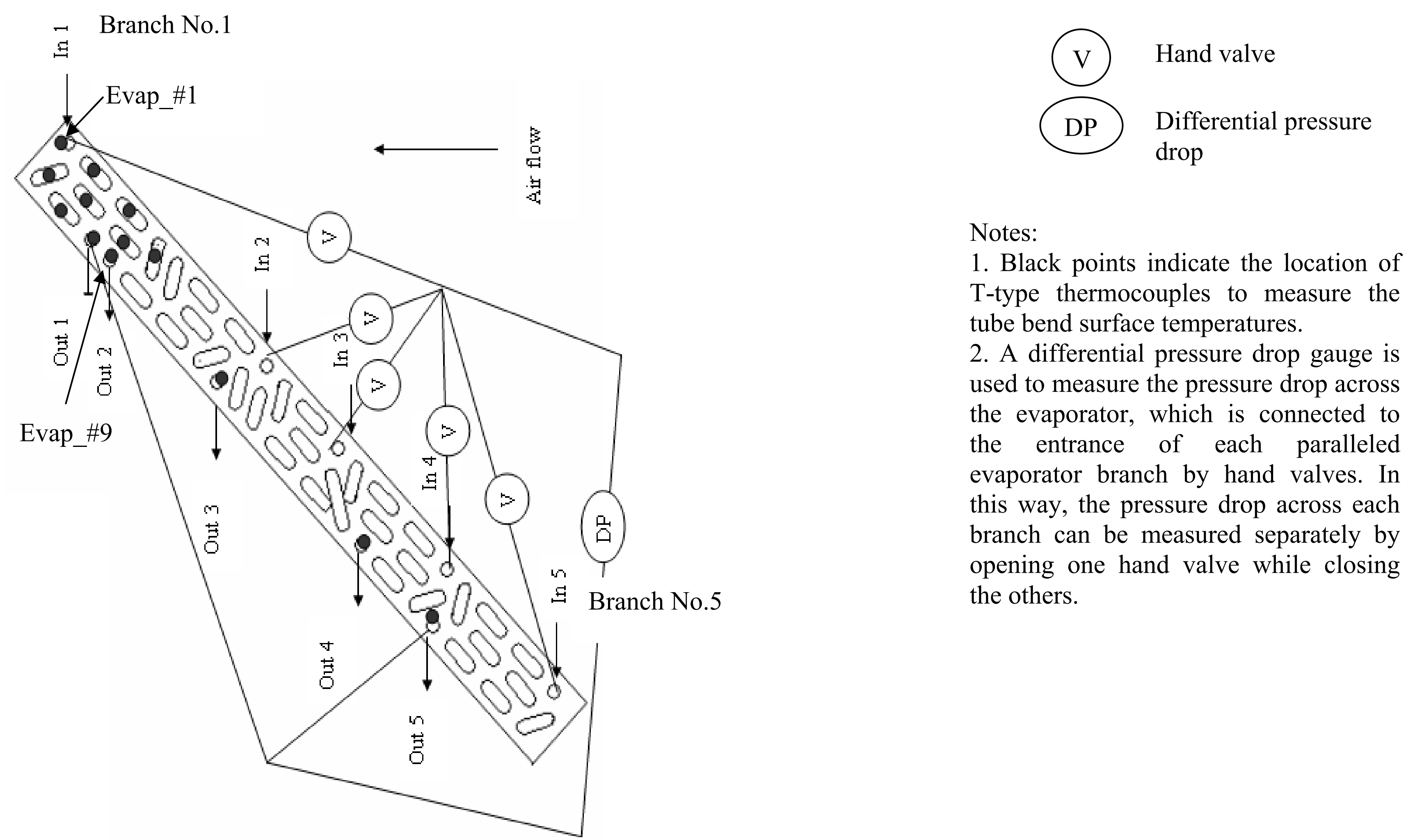


Figure 2-3: Condenser Setup of the 3-ton R-410A Split Unit



Notes:

1. Black points indicate the location of T-type thermocouples to measure the tube bend surface temperatures.
2. A differential pressure drop gauge is used to measure the pressure drop across the evaporator, which is connected to the entrance of each paralleled evaporator branch by hand valves. In this way, the pressure drop across each branch can be measured separately by opening one hand valve while closing the others.

Figure 2-4: Evaporator Setup of the 3-ton R-410A Split Unit

2.3.2 Experimental Setup of 3-ton Packaged Unit

Figure 2-5 presents the airflow pathway of the 3-ton R-410A packaged unit. Figure 2-6 presents the system setup of the unit. Figure 2-7 and Figure 2-8 present the measurements on the condenser and evaporator, respectively.

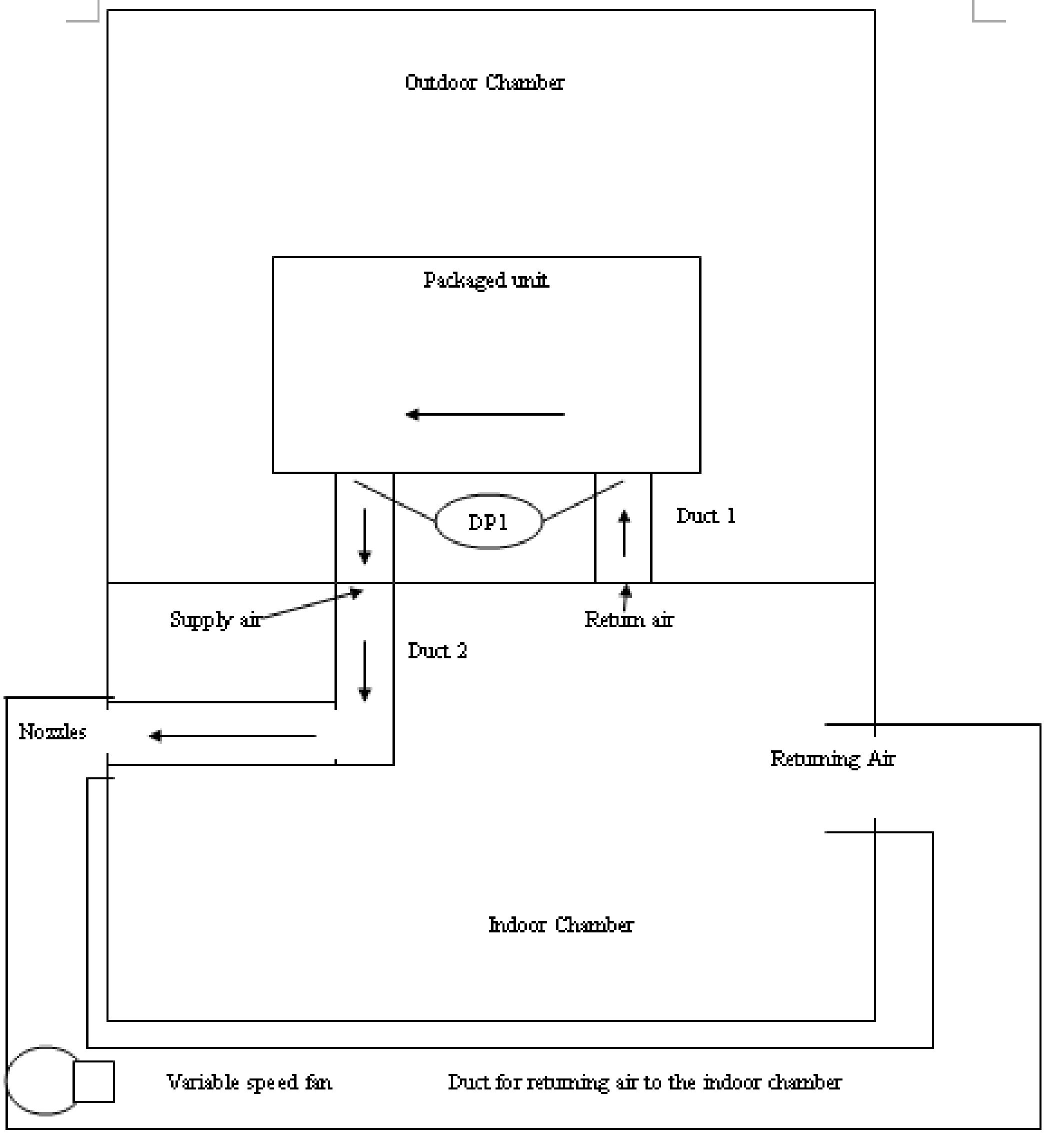


Figure 2-5: Air flow Pathway of the 3-ton Packaged Unit within the Psychrometric Chambers

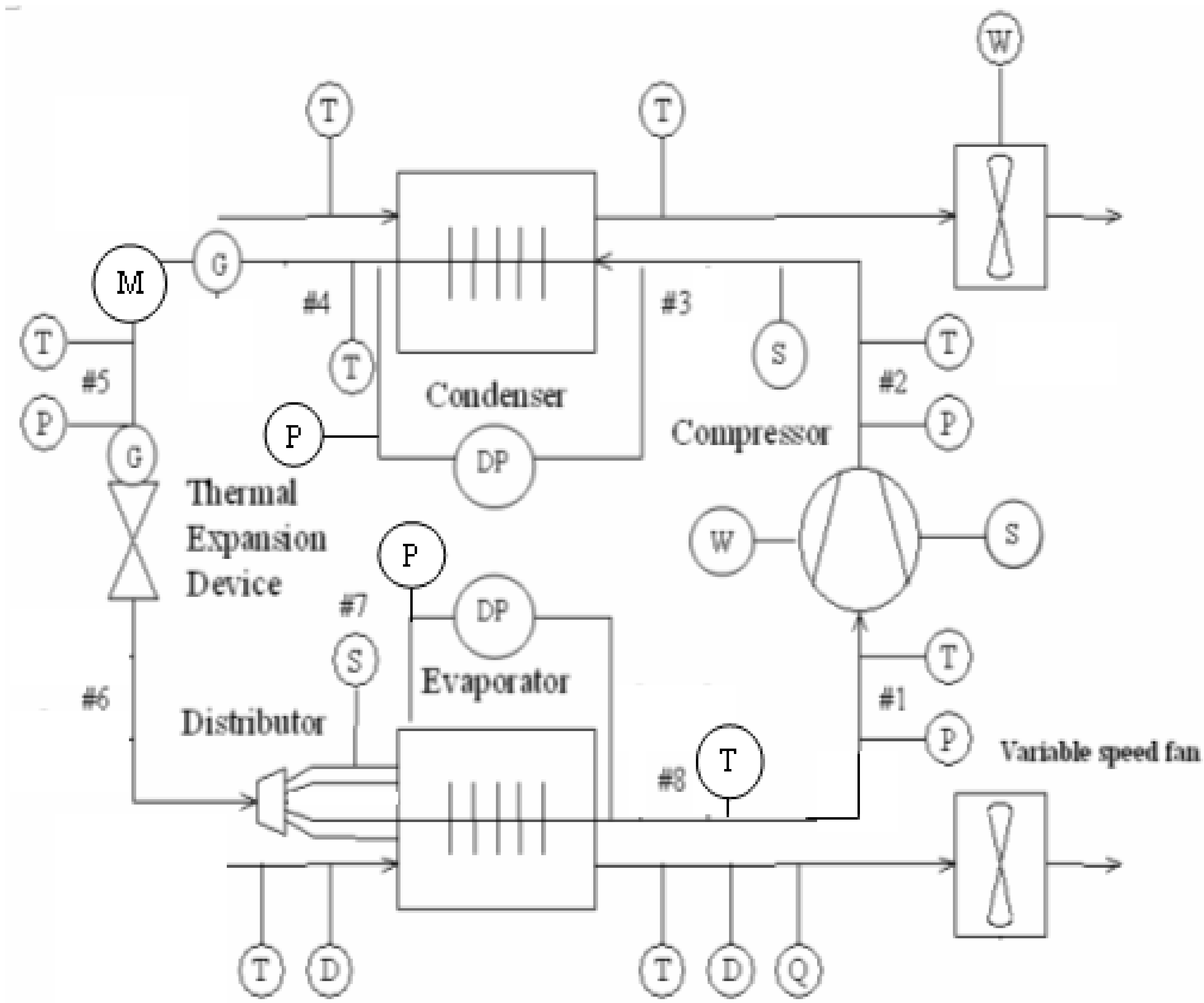


Figure 2-6: System Setup of the 3-ton Packaged Unit

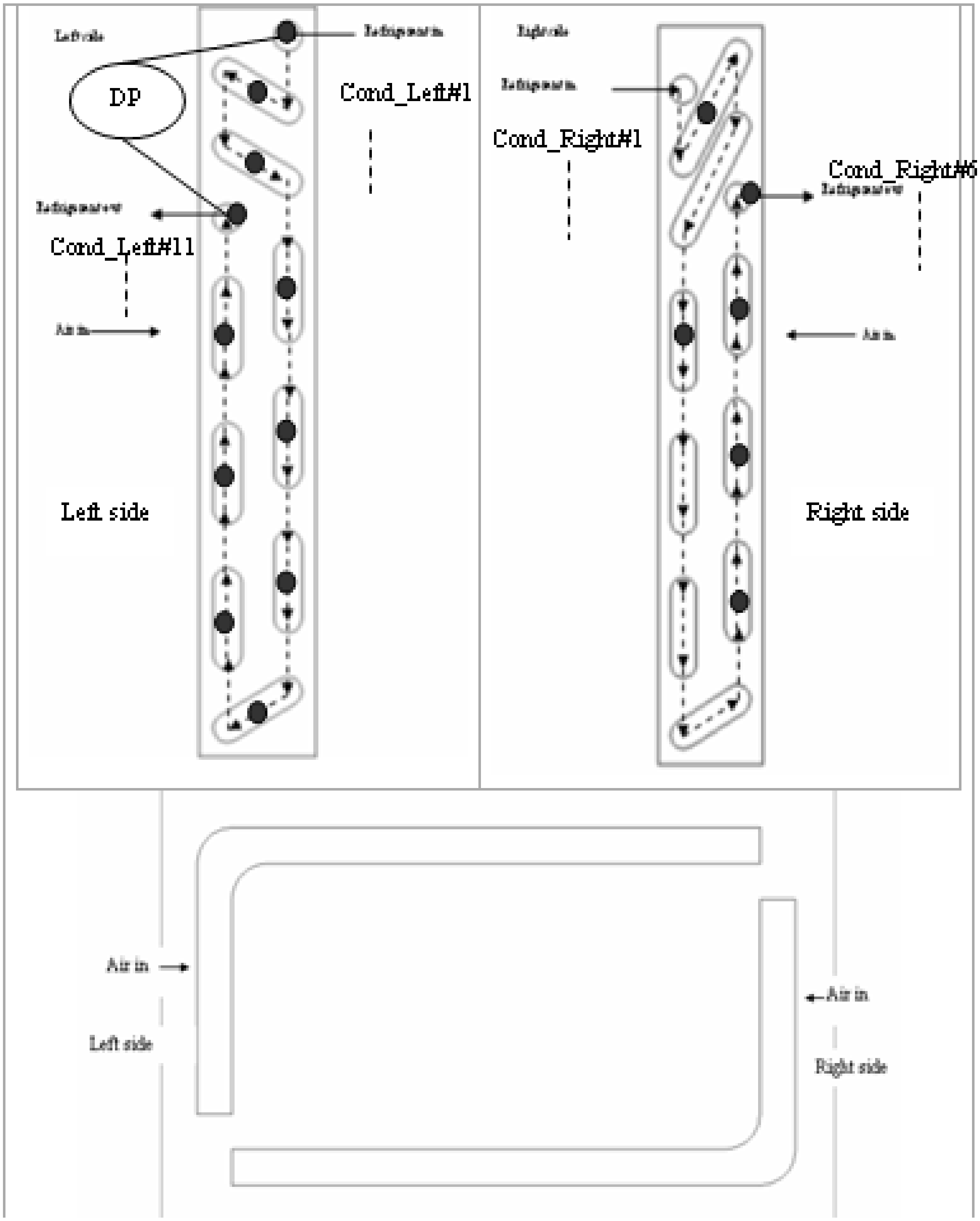


Figure 2-7: Condenser Setup of the 3-ton R-410A Packaged Unit

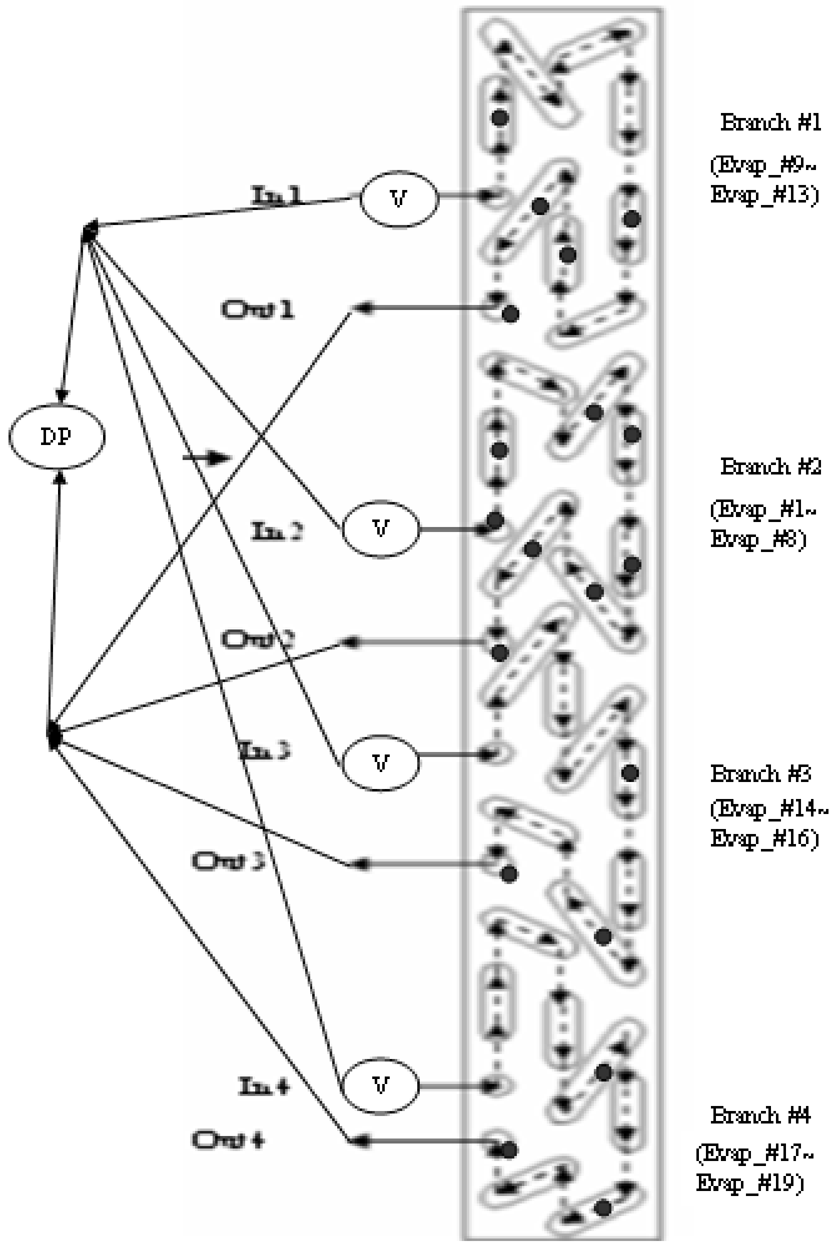


Figure 2-8: Evaporator Setup of the 3-ton R-410A Packaged Unit

2.4 Data Reduction

The air and refrigerant enthalpies are calculated using thermodynamic property functions in EES (Klein 2004). The air enthalpy is determined by the measured dry bulb temperature and dew point at a corresponding location. The refrigerant enthalpy is determined by the local pressure and temperature measurements. However, the refrigerant enthalpy of a two-phase mixture state cannot be determined with the available measurements. In this case, only the air side measurements and calculations are available.

The air-side total cooling capacity is calculated as:

$$\dot{Q}_{cooling,air,tot} = \dot{m}_{air}\,(h_{in,air,evap} - h_{out,air,evap}) \qquad (2\text{-}1)$$

where $h_{in,air,evap}$ and $h_{out,air,evap}$ are determined using the measured inlet and outlet air temperatures and dew points of the evaporator.

The refrigerant-side cooling capacity is calculated as:

$$\dot{Q}_{cooling,ref,tot} = \dot{m}_{ref}\,(h_{out,ref,evap} - h_{in,ref,evap}) \qquad (2\text{-}2)$$

where $h_{in,ref,evap}$ is determined by the pressure and temperature ahead of the expansion device. It is assumed that the enthalpy at state location #5 is the same as the enthalpy at the inlet of the evaporator. $h_{out,ref,evap}$ is determined by the pressure and temperature leaving the evaporator. When two-phase flow occurs at the exit of an evaporator or ahead of an expansion device, the refrigerant side cooling capacity cannot be determined.

The net airside cooling capacity is determined as,

$$\dot{Q}_{cooling,air,net} = \dot{m}_{air}\,(h_{in,air,evap} - h_{out,air,evap}) - W_{blower} \qquad (2\text{-}3)$$

where W_{blower} is evaporator blower power consumption.

The total condensing capacity is determined as,

$$\dot{Q}_{condensing,tot} = \dot{m}_{ref}\,(h_{in,ref,cond} - h_{out,ref,cond}) + W_{fan} \qquad (2\text{-}4)$$

where $h_{in,ref,cond}$ is the enthalpy leaving the compressor, determined by the pressure and temperature at the discharge of the compressor. $h_{out,ref,evap}$ is determined by the pressure and temperature leaving the condenser. When two-phase flow occurs at the exit of the condenser, the total condensing capacity cannot be calculated from the available measurements.

Based on the energy balance at the condenser side, the condenser air flow rate is determined as,

$$\dot{m}_{air,cond} = \frac{\dot{Q}_{condensing,tot}}{(h_{out,air,cond} - h_{in,air,cond})} \qquad (2\text{-}5)$$

where $h_{in,air,cond}$ and $h_{out,air,cond}$ are the condenser inlet and outlet air enthalpies, determined using the measured inlet and outlet air temperatures of the condenser, respectively.

Energy efficiency ratio is given as,

$$EER = \frac{Q_{cooling,air,net}}{W_{comp} + W_{fan} + W_{blower}} \text{ [Btu/W-h]} \qquad (2\text{-}6)$$

The superheat degree is determined using the measurements at state location #1 as follows,

$$T_{super} = T_1 - T_{1,bub} \qquad (2\text{-}7)$$

where $T_{1,bub}$ is the saturation temperature at pressure P_1.

$$T_{1,bub} = f(P_1) \qquad (2\text{-}8)$$

The subcooling degree ahead of an expansion device is determined by the measurements at state location #5 as follows,

$$T_{sub} = T_{5,dew} - T_5 \qquad (2\text{-}9)$$

where $T_{5,dew}$ is the saturation temperature at pressure P_5.

$$T_{5,dew} = f(P_5) \qquad (2\text{-}10)$$

2.5 Nomenclature

A list of nomenclature is given in Table 2-1 to Table 2-6 for all the measured and calculated quantities.

Table 2-1: Refrigerant Side Measurements

P1 [psi]	Suction pressure
P2 [psi]	Discharge pressure
P3 [psi]	Pressure at entrance of condenser
P4 [psi]	Pressure at exit of condenser
P5 [psi]	Pressure upstream of expansion device
P6 [psi]	Pressure ahead of distributor
P7 [psi]	Pressure at entrance of evaporator
P8 [psi]	Pressure at exit of evaporator
T1 [˚F]	Suction temperature
T2 [˚F]	Discharge temperature
T3 [˚F]	Temperature at entrance of condenser
T4 [˚F]	Temperature at exit of condenser
T5 [˚F]	Temperature upstream of expansion device
T6 [˚F]	Temperature ahead of distributor
T7 [˚F]	Temperature at entrance of evaporator
T8 [˚F]	Temperature at exit of evaporator
$T_{sub,mea}$ [˚F]	Measured subcooling degree upstream of expansion device
$X_{sub,cal}$ [%][1]	Quality upstream of expansion device predicted by air side cooling measurements
$T_{sup,mea}$ [˚F]	Measured superheat degree at suction of compressor
$X_{suc,cal}$ [%][2]	Quality at suction of compressor predicted by air side measurements
$m_{ref,mea}$ [lb/min]	Measured refrigerant mass flow rate
$m_{ref,map}$ [lb/min][3]	Refrigerant mass flow rate predicted with compressor map
Comp_top [˚F]	Compressor shell temperature at the top
Comp_middle [˚F]	Compressor shell temperature at the middle
Comp_bottom [˚F]	Compressor shell temperature at the bottom
Charge [lbs]	Refrigerant charge mass

Notes:

1&2: Quality upstream of expansion device determined by air-side cooling capacity is used to investigate the case of two-phase flow entering an expansion device. Quality at the suction of compressor determined by air-side cooling capacity is used to indicate the suction state of a compressor when two-phase flow enters the compressor.

3: Refrigerant mass flow rate predicted with the compressor map is used to as the refrigerant mass flow rate when two-phase flow enters the micro-motion mass flow meter.

Table 2-2: Air Side Measurements

$T_{evap,air,in,drybulb}$ [˚F][4]	Evaporator inlet air dry bulb temperature
$T_{evap,air,in,wetbulb}$ [˚F]	Evaporator inlet air wet bulb temperature
$T_{evap,air,out,drybulb}$ [˚F][5]	Evaporator outlet air dry bulb temperature
$T_{evap,air,out,wetbulb}$ [˚F]	Evaporator outlet air wet bulb temperature
$T_{cond,air,in,drybulb}$ [˚F][6]	Condenser inlet air dry bulb temperature
$T_{cond,air,out,drybulb}$ [˚F][7]	Condenser outlet air dry bulb temperature
$m_{air,cond}$ [CFM][8]	Outdoor air flow rate predicted with the energy balance at condenser side
$m_{air,evap}$ [CFM]	Indoor air flow rate
$m_{dryair,evap}$ [lb/min]	Indoor dry air mass flow rate at entrance of nozzles
Evap Resistance [inH_2O]	Air side pressure drop across evaporating coil
BF [%]	Bypass factor
$m_{water,mea}$ [lb/min][9]	Measured water condensate rate
$m_{water,cal}$ [lb/min]	Calculated water condensate rate based on inlet and outlet air states of evaporator
Dev_{water} [%]	$(m_{water,cal} - m_{water,mea})/ m_{water,mea} \times 100\%$
P_{atm} [atm]	Atmospheric pressure

Notes:

4: $T_{evap,air,in,drybulb}$ is the average temperature of a grid of eight thermocouples at the inlet of the evaporator air flow.

5: $T_{evap,air,out,drybulb}$ is the average temperature of a grid of eight thermocouples at the outlet of the evaporator air flow.

6: $T_{cond,air,in,drybulb}$ is the average temperature of a grid of twelve thermocouples at the inlet of the condenser air flow.

7: $T_{cond,air,out,drybulb}$ is the average temperature of a grid of eight thermocouples at the outlet of the condenser air flow of the 3-ton split unit, or the average temperature of a grid of six thermocouples at the outlet of the condenser air flow of the 3-ton packaged unit.

8: $m_{air,cond}$ is the outdoor air flow rate determined with the condenser energy balance. This calculation is not available when two- phase refrigerant enters the expansion device.

9: $m_{water,mea}$ is the measured water condensate rate determined by collecting the condensate water in a bucket. When the sensible heat ratio is relatively high, the condensate water is not enough to be collected. In this case, this measurement is not available.

Table 2-3: Power Measurements

W_{comp} [W]	Compressor power consumption
V_{comp} [Volts]	Voltage of compressor
I_{comp} [Amps]	Current of compressor
W_{fan} [W]	Condenser fan power consumption
V_{fan} [Volts]	Voltage of condenser fan
I_{fan} [Amps]	Current of condenser fan
W_{blower} [W]	Evaporator blower power consumption
V_{blower} [Volts]	Voltage of evaporator blower
I_{blower} [Amps]	Current of evaporator blower

Table 2-4: Performance

$Q_{cooling,air,net}$ [Btu/h]	Air side net cooling capacity $\dot{Q}_{cooling,air,net} = \dot{m}_{air}(h_{in,air,evap} - h_{out,air,evap}) - W_{blower}$
$Q_{cooling,air,tot}$ [Btu/h]	Air side total cooling capacity
$Q_{cooling,ref,tot}$ [Btu/h]	Refrigerant side total cooling capacity
$Dev_{ref\&air}$ [%]	$(Q_{cooling,ref,tot} - Q_{cooling,air,tot})/Q_{cooling,air,tot} \times 100\%$
$Q_{condensing,tot}$ [Btu/h]	Condensing capacity measured at refrigerant side plus condenser fan power consumption
EER [Btu/Wh]	Energy efficiency ratio = $(Q_{cooling,air,net})/(W_{comp} + W_{fan} + W_{blower})$
SHR [%][10]	Sensible heat ratio

Notes:
10: SHR is determined by the air states upstream and downstream of the evaporating coil. The impact of the evaporator blower is not considered, since it was not on during the tests.

Table 2-5: Steady-state Time Duration Measurements

Duration [min]	Data recording duration time
Sampling interval [s]	Sampling interval of data acquisition system

Notes:
Duration time means a time interval when the Eastern model D-2 chilled mirror sensor was switched to measure the evaporator inlet air dew point, which was usually started after 1 hour at steady-state conditions, and the evaporator outlet dew point measurements were recorded for 10~20 minutes.

Table 2-6: Other Steady-State Measurements

Cond_Upper# [˚C]	Condenser tube bend temperature along the flow path at upper branch of the 3-ton split unit
Cond_Lower# [˚C]	Condenser tube bend temperature along the flow path at lower branch of the 3-ton split unit
Cond_# [˚C]	Condenser tube bend temperature along the flow path at the subcooled branch of the 3-ton split unit
Cond_Left# [˚C]	Condenser tube bend temperature along the flow path at left branch of the 3-ton packaged unit
Cond_Right# [˚C]	Condenser tube bend temperature along the flow path at right branch of the 3-ton packaged unit
Cond_DP [Psi]	Refrigerant side pressure drop across condenser
dp_nozz [Pa]	Air side pressure drop across the measurement nozzles
Evap_# [˚C]	Evaporator tube bend temperature along the flow path of a evaporator branch
Evap_DP# [Psi]	Refrigerant side pressure drop across each evaporator branch
P_nozz [kPa]	Air pressure upstream of the nozzles
$T_{evap,air,in,dew}$ [˚C]	Evaporator inlet air dew point
$T_{evap,air,out,dew}$ [˚C]	Evaporator outlet air dew point
Te_air_in0~ Te_air_in7 [˚C]	Evaporator inlet air temperatures measured at the inlet grid
Te_air_out0~ Te_air_out7 [˚C]	Evaporator outlet air temperatures measured at the outlet grid
Tc_air_in0~11 [˚C]	Condenser inlet air temperatures (each side of the condenser has three thermal couples at the upper, middle, and bottom.)
Tc_air_out0~7 [˚C]	Condenser outlet air temperatures measured at the outlet grid
T_box [˚C]	Air temperature upstream of the nozzles.

Notes:
Other Steady-state Measurements are shown in the Excel data files.

2.6 List of the Measurements for Cyclic Tests

In order to constrict the sampling interval within the required range, only the following necessary measurements were kept for the cyclic tests:

- Evaporator inlet air dry bulb temperature: 8 thermocouples.
- Evaporator outlet air dry bulb temperature: 8 thermocouples.
- Evaporator inlet air dew point: General Eastern model D-2 chilled mirror sensor.
- Evaporator outlet air dew point: General Eastern model HYGRO-M2 chilled mirror sensor for the cyclic test under wet condition.
- Evaporator airflow rate: ASHRAE standard nozzle chamber.
- Condenser inlet air dry bulb temperature: 12 thermocouples.
- Condenser outlet air dry bulb temperature: eight thermocouples for 3-ton split unit, 6 thermocouples for 3-ton packaged unit.
- Refrigerant pressures at interested state locations
- Refrigerant temperatures at interested state locations
- Refrigerant mass flow rate: micro-motion mass flow meter
- Compressor power: power meter
- Condenser fan power: power meter
- Evaporator blower power: measured in a separate test at 1200 CFM and medium blower speed

2.7 Uncertainty Analysis

The quality of the experimental results is determined by estimating the uncertainty of the test results. The Kline and McClintock (1953) method is used, which sums the square of errors:

$$\omega_A = \left[\sum_{i=1}^{j} \left(\frac{\partial A}{\partial z_i} \omega_{z_i} \right)^2 \right]^{1/2} \quad (2\text{-}11)$$

where ω_A is the total uncertainty associated with a dependent variable A, z_i is one of the independent variables which impacts the dependent variable A, ω_{z_i} is the uncertainty associated with an independent variable z_i .

A list of independent measurement uncertainties are given in Table 2-7, as discussed in Section 2.1. These uncertainties affect the calculations in heat transfer and performance parameters.

Table 2-7: Uncertainties of Independent Variables

	Uncertainty (absolute or relative)
Refrigerant side Temperature	± 0.5 ˚C
Air side temperature	± 1.0 ˚C
Refrigerant pressure	± 0.8%
Barometric Pressure	± 0.03 kPa
Dew point	± 0.2 ˚C
Refrigerant mass flow rate	± 0.6%
Power	± 10W
Indoor air flow rate	± 10 g/s

The necessary partial derivatives in Equation 2-11 are given by the analytical equations described in Section 2.4. With the partial derivatives and independent uncertainties, the uncertainties of the dependent variables can be determined using EES. Table 2-8 gives the uncertainties of the dependent variables in steady-state Test #3 of the 3-ton split unit. The data of other steady-state tests have similar uncertainties. The relative uncertainties of cyclic tests should be larger than those in steady-state tests, however, data in Table 2-8 can be used as a reference.

Table 2-8: Uncertainties of Dependent Variables

	Uncertainty (absolute or relative)
Subcooling degree	±0.62 K
Superheat degree	±0.57 K
Airside cooling capacity	±11%
Refrigerant side cooling capacity	±1%
Total condensing capacity (determined with refrigerant side measurements)	±0.9%
EER (determined with air side measurements)	±12%
SHR	±3%
BF (Bypass factor)	±27%

As shown in Table 2-8, the measurements on the refrigerant side are much more accurate than the measurements on the air side.

3. BASELINE TESTING OF 3-TON SPLIT UNIT

3.1 Air-side Pressure Drop and Evaporator Blower Power

For measuring air-side pressure drop, a differential pressure transducer DP1 (as indicated in Figure 2-1 and 2-5) was installed across the indoor unit. It has to be mentioned that the indoor evaporator blower was taken out when measuring the air-side pressure drop. The airflow rate was altered by a variable speed fan downstream of the nozzles. The pressure drop measurements are reported in Figure 3-1 and Table 3-1, where DP_{coil} means the pressure drop measured by DP1**,** the major part of which is the pressure drop across the evaporating coil.

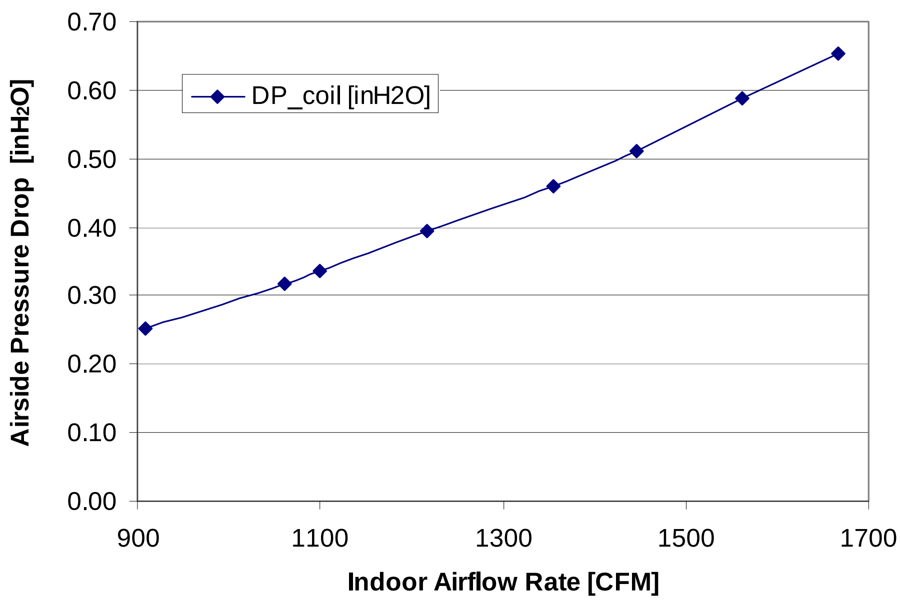


Figure 3-1: Evaporator Air-Side Pressure Drop of 3-ton Split Unit

Table 3-1: Evaporator Airside Pressure Drop of 3-ton Split Unit

Air Flow Rate [CFM]	DP_{coil} [inH2O]
908.1	0.25
1060.8	0.32
1099.9	0.34
1216.9	0.39
1354.7	0.46
1447.0	0.51
1562.0	0.59
1667.0	0.65

The condenser fan has one speed. The evaporator blower has three speeds (low, medium and high). For measuring the evaporator blower power consumption, the blower was installed back into the indoor unit.

During the tests, the airflow rate through the indoor unit was altered at each blower speed. At the beginning of altering the airflow rate, the blower ran at a fixed speed while the variable speed fan behind the nozzles was off. In this way, the smallest airflow rate at each blower speed through the airflow pathway could be obtained. After that, the variable speed fan was turned on while the blower was still running at the speed. Then the airflow rate through the indoor blower at each speed could be altered by adjusting the external variable speed fan.

The electrical measurements of the evaporator blower of the 3-ton split unit are presented in Figure 3-2 and Table 3-2 to 3-4.

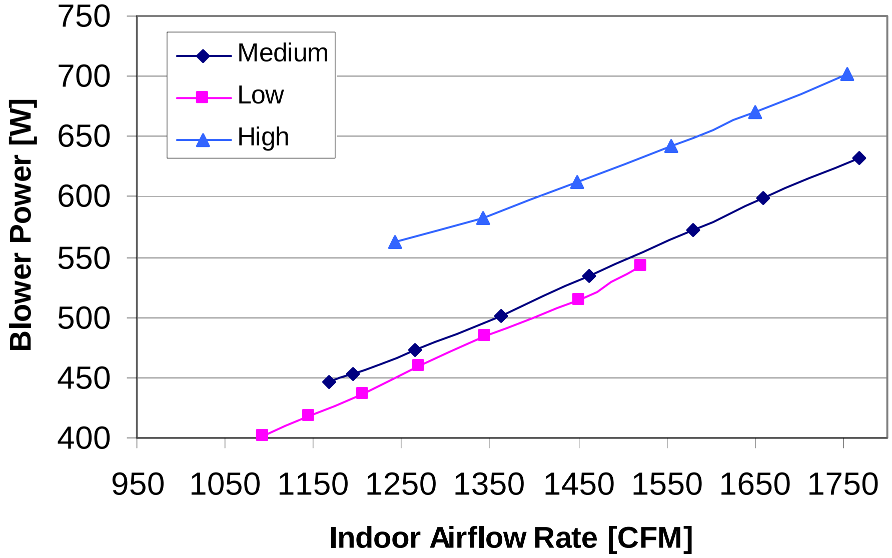


Figure 3-2: Evaporator Blower Power

Table 3-2: Electrical Measurements of the Evaporator Blower at Low Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]
1093.3	402.1	237.3	1.7
1144.0	417.8	238.8	1.8
1205.4	436.7	239.3	1.9
1269.0	458.9	239.8	1.9
1344.9	484.5	240.8	2.0
1451.1	514.5	240.0	2.2
1520.9	542.1	238.8	2.4

Table 3-3: Electrical Measurements of the Evaporator Blower at Medium Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]
1167.8	446.1	237.3	1.8
1194.9	453.6	237.0	1.9
1266.0	472.7	237.0	2.0
1363.4	502.0	236.5	2.1
1462.9	534.5	235.3	2.3
1579.4	571.8	235.8	2.5
1658.5	598.9	236.0	2.6
1768.7	632.0	237.5	2.7

Table 3-4: Electrical Measurements of the Evaporator Blower at High Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]
1243.4	562.8	240.0	2.4
1342.3	581.8	239.0	2.5
1448.6	612.0	238.8	2.6
1554.8	641.5	239.5	2.7
1649.6	670.1	239.0	2.8
1755.5	701.4	239.3	2.9

The electrical measurements for the condenser fan of the 3-ton R-410A split system at the nominal condenser airflow rate are:
Nominal Air Flow Rate: 2885 CFM
Power --150 W
Voltage -- 230 V
Current -- 0.51 A

3.2 Steady-State Measurements of 3-ton Split Unit

Table 3-5: Overall Steady-State Data of 3-ton Split Unit (Required tests)

Steady-state Tests						Test Operating Tolerance							
Test	Outdoor DB	Indoor DB	Indoor WB	Charge	Indoor Airflow	Outdoor DB	Indoor DB	Indoor WB	Net Cooling	Compressor power	EER	SHR	Dev_ref &air
No.	Req/Act [F]	Req/Act [F]	Req/Act [F]	[%]	Req/Act [cfm]	[F]	[F]	[F]	[Btu/h]	[kW]	[Btu/Wh]	[%]	[%]
1	115/115.0	80/79.8	67/66.8	100%	1200/1209	0.71	0.81	0.44	31392	3.37	7.91	75%	-5.1%
2	95/95.2	80/80.3	67/67.3	100%	1200/1199	0.97	1.17	0.45	38123	2.99	10.62	69%	-5.5%
3	115/114.9	80/80.3	63/63.0	100%	1200/1201	1.19	0.80	0.20	28700	3.27	7.42	94%	-3.9%
4	115/114.8	80/79.8	63/62.9	120%	1200/1220	0.77	0.67	0.50	30462	3.34	7.73	91%	-5.7%
5	115/114.9	80/80.1	63/62.8	90%	1200/1155	1.01	0.93	0.60	28333	3.20	7.46	93%	-3.1%
6	115/114.6	80/79.4	63/62.3	81%	1200/1178	0.80	0.46	0.50	26351	3.19	6.95	96%	N/A
7	115/114.8	80/79.6	63/62.7	71%	1200/1191	0.59	0.59	0.50	23650	3.08	6.43	94%	N/A
8	115/115.3	80/79.5	63/62.9	100%	900/919	0.83	0.46	0.10	27267	3.20	7.39	85%	-3.4%
9	115/115.3	80/80.0	63/63.0	100%	1050/1055	0.69	0.72	0.12	27911	3.25	7.35	90%	-3.2%
10	115/115.4	80/80.0	63/62.6	100%	1350/1360	0.73	0.56	0.16	29101	3.30	7.37	97%	-4.4%
11	115/115.4	80/80.0	63/63.0	100%	1650/1675	0.74	0.58	0.14	30616	3.37	7.31	99%	-6.6%
12	127/126.9	80/80.4	63/63.0	100%	1200/1204	0.56	0.44	0.20	26258	3.58	6.28	99%	-3.1%
Allowable Tolerance	±0.5 [F]	±0.5 [F]	±0.3 [F]	---	---	2 [F]	2 [F]	1 [F]	---	---	---	---	±6%

Notes:
Req/Act: required value/actual value.

Detailed data for steady-state tests are given in the following tables. In addition to the required tests listed in Table 3-5, three more steady-state tests were performed to check the mass for balance condensed water and calculate the necessary parameters for transient tests. Tests # 1 to #8 are presented in Tables 3-6 to 3-10 and Tests #9 to #15 are presented from Tables 3-11 to 3-15.

Table 3-6: Refrigerant-Side Measurements of 3-ton Split Unit

Test No.	1	2	3	4	5	6	7	8
P1 [psi]	154.53	152.76	145.26	148.24	146.38	147.08	137.42	139.03
P2 [psi]	531.97	430.87	524.63	554.18	507.18	497.88	485.09	522.90
P3 [psi]	527.34	427.30	519.82	549.29	503.01	493.83	481.00	517.79
P4 [psi]	514.36	409.63	508.34	538.89	488.46	477.76	465.63	507.62
P5 [psi]	520.36	411.28	515.97	546.92	498.09	481.28	465.28	515.22
P6 [psi]	217.49	207.48	203.40	200.07	208.17	215.58	202.89	195.28
P7 [psi]	157.86	156.81	147.78	150.95	150.08	150.66	140.16	141.95
P8 [psi]	156.43	154.50	146.52	149.76	148.27	149.04	139.02	140.76
T1 [˚F]	58.11	55.96	55.26	54.44	54.17	55.75	60.53	52.21
T2 [˚F]	206.15	178.87	208.86	211.29	201.78	200.58	210.03	210.32
T3 [˚F]	186.83	161.29	188.58	189.27	183.10	182.03	188.71	189.51
T4 [˚F]	121.20	102.98	119.97	116.23	124.97	130.03	128.17	119.41
T5 [˚F]	119.65	101.82	118.43	114.81	123.30	128.87	127.01	117.77
T6 [˚F]	68.83	65.16	64.65	65.32	67.80	70.10	66.53	61.86
T7 [˚F]	53.50	52.47	50.19	51.58	51.93	52.52	48.46	47.36
T8 [˚F]	54.90	54.76	50.61	51.97	51.46	53.45	58.11	47.36
$T_{sub,mea}$ [˚F]	15.02	14.25	15.54	23.92	7.83	0	0	16.08
$X_{sub,cal}$ [%]	---	---	---	---	---	0	4.16%	---
T_{sup} [˚F]	9.00	7.54	9.88	7.83	8.34	9.61	18.45	9.45
$m_{ref,mea}$ [lb/min]	8.29	8.83	7.64	7.80	7.94	7.45	6.70	7.22
$m_{ref,map}$ [lb/min]	8.26	8.88	7.58	7.66	7.83	7.91	7.02	7.14
Comp_top [˚F]	115.77	102.87	114.49	113.52	112.50	113.40	117.07	112.42
Comp_middle#1 [˚F]	138.06	120.45	137.98	136.85	136.00	136.62	141.66	138.09
Comp_middle#2 [˚F]	126.34	111.09	126.28	125.17	124.43	125.24	130.86	126.46
Comp_bottom#1 [˚F]	156.43	135.19	157.50	156.43	152.02	152.19	159.53	158.83
Comp_bottom#2 [˚F]	149.16	129.74	148.71	149.00	146.61	147.16	153.72	151.16
Charge [lbs]	6.5[2]	6.5	6.5	7.83	5.87	5.24	4.60	6.5

Notes:
1. All the refrigerant pressures are given as absolute pressure.
2. The 100% charge of 6.5 lbs was obtained by specifying the condenser exit temperature at state location #4 in Test #2 around 102 ˚F, allowing a tolerance of ±3 ˚F. It is larger than the factory charge of 6.25 lbs, since extra sensors were added to the system.

--- means that the measurements or calculations were not conducted.

Table 3-7: Air-Side Measurements of 3-ton Split Unit

Test No.	1	2	3	4	5	6	7	8
$T_{evap,air,in,drybulb}$ [˚F]	79.84	80.30	80.27	79.81	80.12	79.43	79.64	79.54
$T_{evap,air,in,wetbulb}$ [˚F]	66.83	67.31	63.04	62.88	62.78	62.32	62.70	62.89
$T_{evap,air,out,drybulb}$ [˚F]	60.67	59.16	58.34	57.52	57.68	58.18	60.85	55.16
$T_{evap,air,out,wetbulb}$ [˚F]	58.09	56.66	54.32	53.71	53.73	54.0	55.36	52.0
$T_{cond,air,in,drybulb}$ [˚F]	114.96	95.16	114.91	114.76	114.95	114.60	114.78	115.34
$T_{cond,air,out,drybulb}$ [˚F]	130.57	112.31	129.37	129.72	129.79	129.08	127.87	129.15
$m_{air,cond}$ [CFM]	2817	2762	2849	2872	2777	---	---	2847
$m_{air,evap}$ [CFM]	1209	1199	1201	1220	1155	1178	1191	919
$m_{dryair,evap}$ [lb/min]	87.99	87.61	88.40	89.14	84.47	85.94	86.34	68.06
BF [%]	21.55%	21.91%	24.92%	24.18%	24.42%	25.97%	34.17%	20.96%
P_{atm} [atm]	0.991	0.991	0.995	0.988	0.988	0.986	0.985	0.995
Evap Resistance [inH_2O]	0.39	0.39	0.39	0.39	0.39	0.39	0.39	0.25

Table 3-8: Power Measurements of 3-ton Split Unit

Test No.	1	2	3	4	5	6	7	8
W_{comp} [W]	3370	2990	3270	3340	3200	3190	3080	3200
V_{comp} [Volts]	---	---	---	230.8	231.5	233.5	234.0	---
I_{comp} [Amps]	---	---	---	14.12	13.56	13.49	13.03	---
W_{fan} [kW]	150	150	150	150	150	150	150	150
W_{blower} [kW]	450[3]	450	450	450	450	450	450	340[4]

Notes:
3. The blower power of 450 W was obtained from Table 3-3 for the medium speed.
4. The blower power of 340 W was obtained by extrapolating with the data in Table 3-2 for the low speed.

Table 3-9: Performance of 3-ton Split Unit

Test No.	1	2	3	4	5	6	7	8
$Q_{cooling,air,net}$ [Btu/h]	31392	38123	28700	30462	28333	26351	23650	27267
$Q_{cooling,air,tot}$ [Btu/h]	32927	39658	30235	31998	29869	27887	25185	28427
$Q_{cooling,ref,tot}$ [Btu/h]	31277	37521	29080	30210	28951	---	---	27484
$Dev_{ref\&air}$ [%]	-5.1%	-5.5%	-3.9%	-5.7%	-3.1%	---	---	-3.4%
$Q_{condensing,tot}$ [Btu/h]	43408	48420	40840	42287	40553	---	---	38962
EER [Btu/Wh]	7.91	10.62	7.42	7.73	7.46	6.95	6.43	7.39
SHR [%]	75%	69%	94%	91%	93%	96%	94%	85%

Table 3-10: Steady-State Duration of Measurements for 3-ton Split Unit

Test No.	1	2	3	4	5	6	7	8
Duration [min]	47	42	41	40	80	40	40	30
Sampling interval [s]	10 seconds							

Continued

Table 3-11: Refrigerant-Side Measurements of 3-ton Split Unit (continued)

Test No.	9	10	11	12	13	14	15
P1 [psi]	142.39	146.99	152.37	155.94	152.89	147.56	132.59
P2 [psi]	525.10	529.49	533.11	598.59	431.31	371.76	359.01
P3 [psi]	520.03	524.55	528.09	593.53	427.39	366.59	355.16
P4 [psi]	509.29	513.12	515.72	580.60	408.31	345.39	338.81
P5 [psi]	516.55	520.43	523.50	592.61	413.24	348.43	340.59
P6 [psi]	200.05	206.19	213.95	222.46	206.27	193.64	174.82
P7 [psi]	145.46	150.11	155.68	159.00	156.53	150.35	135.48
P8 [psi]	144.20	148.81	154.32	157.35	154.54	148.35	133.65
T1 [˚F]	53.96	56.77	59.13	57.09	55.86	53.99	46.30
T2 [˚F]	209.76	209.97	209.10	220.66	179.16	163.84	164.22
T3 [˚F]	189.23	189.53	189.20	200.25	160.91	145.98	146.11
T4 [˚F]	119.75	120.28	121.01	133.12	103.36	89.60	87.34
T5 [˚F]	118.15	118.70	119.41	131.56	102.24	88.92	86.54
T6 [˚F]	63.43	65.36	67.69	71.01	65.17	60.22	54.00
T7 [˚F]	48.90	50.74	53.28	54.44	52.13	49.21	44.45
T8 [˚F]	49.39	52.10	54.72	54.90	54.55	53.34	43.65
$T_{sub,mea}$ [˚F]	15.91	15.98	15.74	13.81	14.20	14.65	15.35
T_{sup} [˚F]	9.77	10.68	10.86	7.42	7.40	7.67	6.33
$m_{ref,mea}$ [lb/min]	7.44	7.73	8.09	7.99	8.84	8.78	7.77
$m_{ref,map}$ [lb/min]	7.37	7.65	8.03	7.93	8.89	8.88	7.82
Comp_top [˚F]	113.32	115.48	116.67	123.30	102.91	94.28	89.91
Comp_middle#1 [˚F]	138.58	139.05	139.50	146.46	120.75	110.24	108.99
Comp_middle#2 [˚F]	126.82	127.11	127.58	132.96	111.35	102.22	100.35
Comp_bottom#1 [˚F]	158.76	159.01	158.85	166.89	136.02	123.91	123.60
Comp_bottom#2 [˚F]	151.21	150.67	150.89	163.90	129.71	118.17	117.75
Charge [lbs]	6.5	6.5	6.5	6.5	6.5	6.5	6.5

Table 3-12: Air-Side Measurements of 3-ton Split Unit (continued)

Test No.	9	10	11	12	13	14	15
$T_{evap,air,in,drybulb}$ [˚F]	79.95	80.00	79.97	80.37	80.61	80.03	80.24
$T_{evap,air,in,wetbulb}$ [˚F]	63.04	62.62	62.99	62.97	67.32	66.87	54.66
$T_{evap,air,out,drybulb}$ [˚F]	56.82	59.62	61.88	58.93	59.32	57.83	51.91
$T_{evap,air,out,wetbulb}$ [˚F]	53.34	54.76	56.23	54.98	56.55	55.18	41.96
$T_{cond,air,in,drybulb}$ [˚F]	115.30	115.35	115.42	126.86	95.10	82.15	81.99
$T_{cond,air,out,drybulb}$ [˚F]	129.50	130.13	130.64	141.57	112.33	99.68	97.75
$m_{air,cond}$ [CFM]	2842	2825	2841	2820	2771	2750	2767
$m_{air,evap}$ [CFM]	1055	1360	1675	1204	1197	1195	1200
$m_{dryair,evap}$ [lb/min]	77.89	99.82	122.46	87.46	86.70	87.19	88.92
BF [%]	22.23%	29.51%	34.8%	23.93%	23.83%	23.62%	43.7%
$m_{water,mea}$ [lb/min]	---	---	---	---	0.00132	0.00148	---
$m_{water,cal}$ [lb/min]	---	---	---	---	0.00147	0.00164	---
Dev_{water} [%]	---	---	---	---	11.4%	10.8%	---
P_{atm} [atm]	0.995	0.994	0.995	0.984	0.983	0.987	0.984
Evap Resistance [inH_2O]	0.32	0.46	0.65	0.39	0.39	0.39	0.39

Notes:
Test No. 13 and No. 14 were conducted to check if the water condensate rate can be predicted by the air-side humidity measurements. It turns out that the predictions are not accurate enough. The reason could be that the air humidity measurement was conducted only at one position, which could not give an overall humidity distribution. In addition, some condensate water could be carried away by the indoor air flow, since the measured water condensate rate tends to be 10% less than the rate predicted with airside measurements.

Table 3-13: Power Measurements of 3-ton Split Unit (continued)

Test No.	9	10	11	12	13	14	15
$Power_{comp}$ [kW]	3250	3300	3370	3580	3010	2720	2620
$Power_{fan}$ [kW]	150	150	150	150	150	150	150
$Power_{blower}$ [kW]	400[5]	500[6]	670[7]	450	450	450	450

Notes:
5. The blower power of 400 W was obtained by extrapolating with the data in Table 3-3 for the medium speed.
6. The blower power of 500 W was obtained from Table 3-3 for the medium speed.
7. The blower power of 670 W was obtained from Table 3-4 for the high speed.

Table 3-14: Performance of 3-ton Split Unit (continued)

Test No.	9	10	11	12	13	14	15
$Q_{cooling,air,net}$ [Btu/h]	27911	29101	30616	26258	38327	41052	35008
$Q_{cooling,air,tot}$ [Btu/h]	29275	30807	32902	27794	39863	42588	36544
$Q_{cooling,ref,tot}$ [Btu/h]	28359	29485	30787	26949	37454	40241	35731
$Dev_{ref\&air}$ [%]	-3.2%	-4.4%	-6.6%	-3.1%	-6.2%	-5.7%	-2.2%
$Q_{condensing,tot}$ [Btu/h]	39991	41314	42837	39713	48441	50235	45325
EER [Btu/Wh]	7.35	7.37	7.31	6.28	10.62	12.37	10.87
SHR [%]	90%	97%	99%	99%	68%	67%	100%

Table 3-15: Steady-State Duration of Measurements for 3-ton Split Unit (continued)

Test No.	9	10	11	12	13	14	15
Duration [min]	41	41	41	40	65	90	38
Sampling interval [s]	10 seconds						

3.3 Transient Measurements of 3-ton Split Unit

Table 3-16: Overall Cyclic Test Data of 3-ton Split Unit

Cyclic Test	Outdoor DB	Indoor DB	Indoor WB	Charge	Indoor Airflow	Test Operating Tolerance Outdoor DB	Test Operating Tolerance Indoor DB	Test Operating Tolerance Indoor WB
No.	Req/Act [˚F]	Req/Act [˚F]	Req/Act [˚F]	[%]	Req/Act [cfm]	[˚F]	[˚F]	[˚F]
1	82/82.76	80/80.47	<57/56.62	100%	1200/1220	3.06	2.16	0.61
2	115/115.53	80/80.20	<57/55.50	100%	1200/1216	4.73	1.56	0.32
3 (wet cyclic)	82/82.49	80/80.96	67/67.38	100%	1200/1225	2.34	3.62	2.5
Allowable Tolerance	±0.5	±0.5	±0.3	---	---	2.0	2.0	1.0

Notes:

- Due to the limitation of the test facility, some test parameters could not be controlled within allowable tolerances.
- A separate General Eastern model HYGRO-M2 chilled mirror sensor was installed to measure the evaporator outlet air dew point for the wet cyclic test.
- The data sampling interval was within 5 seconds for all the cyclic tests.
- On/off interval: 6 minutes on/24 minutes off, ran for four complete cycles.
- All cycling performance tests employed the air enthalpy method to calculate total cooling capacities.
- During all the cyclic tests, the indoor air flow was always on, driven by an external variable speed fan. However, for calculating the cyclic efficiency, the indoor blower power was considered only for the "on" time.

3.3.1 Cyclic Test #1: Cyclic test under Dry Condition and Outdoor Temperature 82 F

Figures 3-3 to 3-6 present the transient cyclic behavior of the suction and discharge pressure (P1 and P2), refrigerant mass flow rate and compressor power consumption, inlet and outlet dry bulb temperatures of the indoor unit, and inlet and outlet air temperatures of the outdoor unit, respectively.

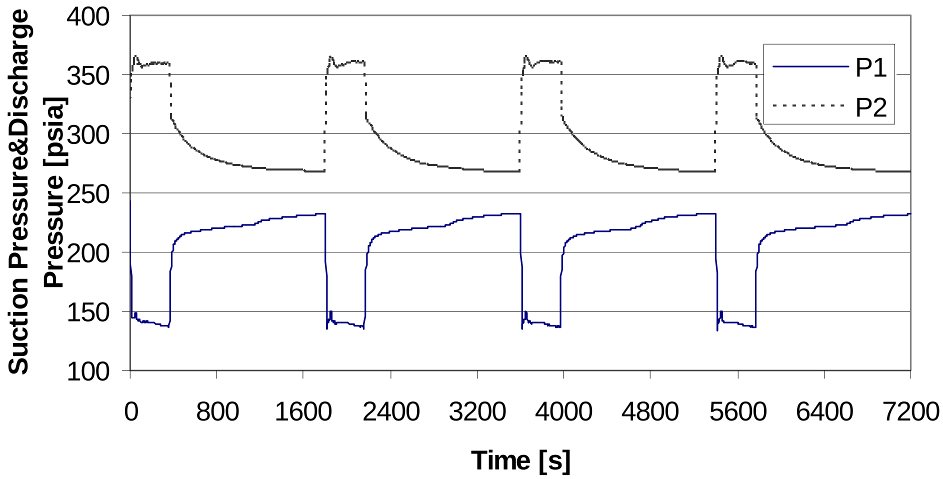


Figure 3-3: Pressure Responses in Cyclic Test #1of 3-ton Split Unit

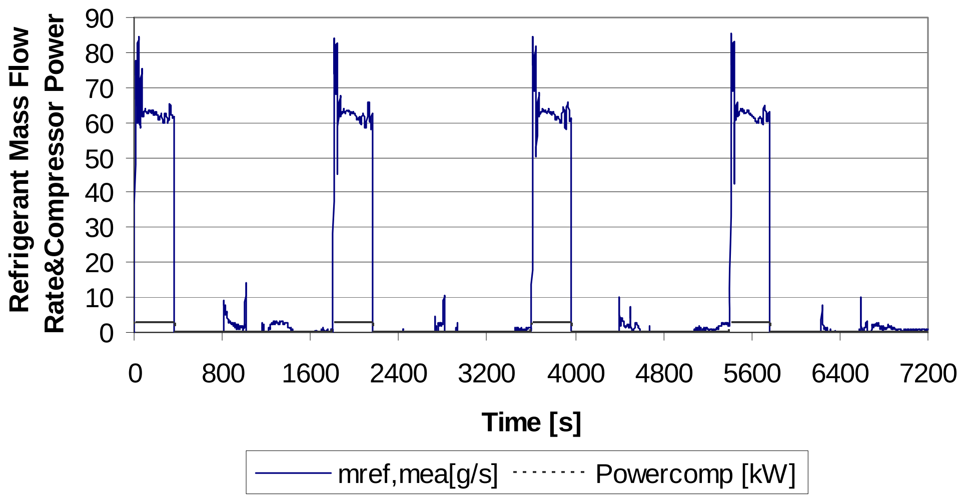


Figure 3-4: Refrigerant Mass Flow Rate and Compressor Power Response in Cyclic Test #1 of 3-ton Split Unit

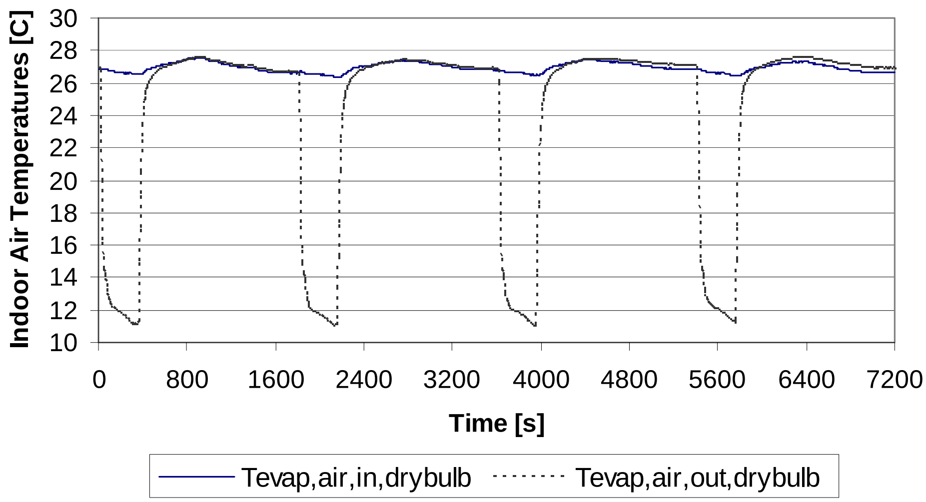


Figure 3-5: Indoor Air Temperature Responses in Cyclic Test #1 of 3-ton Split Unit

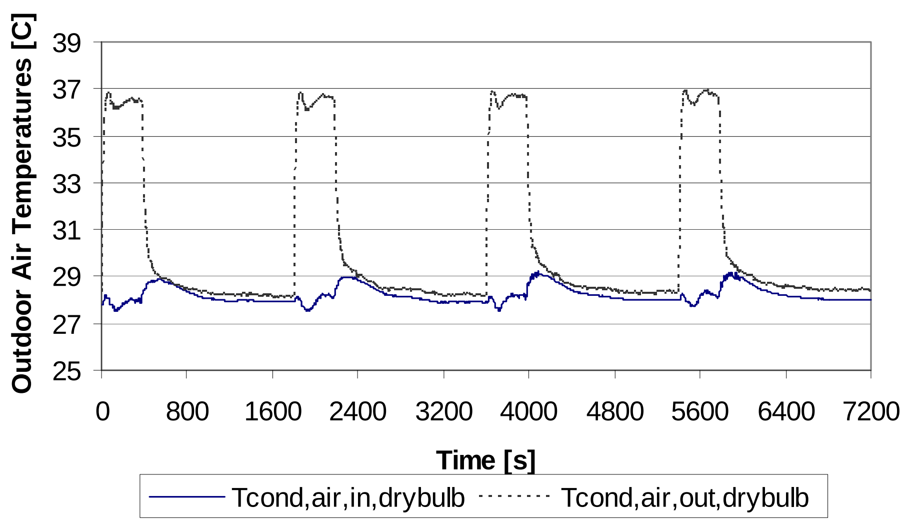


Figure 3-6: Outdoor Air Temperature Responses in Cyclic Test #1 of 3-ton Split Unit

Cyclic Performance

- $Q_{cyc,dry,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles)→ 3421.9 Btu
- $Q_{cyc,dry,net}$ --net cycling cooling capacity during one complete on/off cycle minus evaporator blower power during "on" time (τ =30 min, average among four cycles)
  → $Q_{cyc,dry,net} = Q_{cyc,dry,tot}$ -W_{blower} = 3421.92 Btu – 450 W × 360 Sec = 3268 Btu
- $Q_{ssdry,net} \times \tau$ -- net steady-state cooling capacity from steady-state Test#15 during the time for one complete cycle→35008 Btu/h×0.5 h=17504 Btu
- $CLF = \dfrac{Q_{cyc,dry,net}}{Q_{ss,dry,net} \times \tau}$ --cooling load factor→3268/17504 = 0.187

- W_{tot} ---total power consumption during one cycle → (W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1168770 J, where W_{comp} and W_{fan} are actual measurements, W_{blower} is assigned as 450 W.
- $EER_{cyc,dry}$ --energy efficiency ratio determined from dry cycling test→ $Q_{cyc,dry,net}/W_{tot}$ = 3268×3600/1168770=10.07
- $EER_{ss,dry}$ -- energy efficiency ratio determined from steady-state Test #15→10.87
- $C_D = \dfrac{1-\dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1-CLF}$ --degradation factor→0.0905
- $PLF = 1 - 0.5 \times C_D$ --Partial load performance factor→95.47%
- EER_B -- energy efficiency ratio determined from steady-state Test#14---12.37
- $SEER = PLF(0.5) \times EER_B$ --seasonal energy efficiency ratio→11.81
- Rated SEER---12.00

Discussion

- The TxV in the split unit was closed during "off" times, which led to a static pressure difference of 35 psig between the condenser side and evaporator side. The pressure difference was due to the difference between the outdoor temperature and indoor temperature.
- As shown Figures 3-3 and 3-4, the working difference between the suction pressure and discharge pressure was established within about 15 seconds after starting the compressor in each cycle. In the following "on" time, the compressor power consumption was fairly constant. After that, the suction pressure descended slowly until it reached the minimum pressure at the end of each "on" period.
- As shown in Figure 3-5, the response of the evaporator outlet air temperature was slower than that of the suction pressure, since the refrigerant absorbed the stored heat from the coil at first. The evaporator outlet air temperature reached a minimum value at the end of each "on" period.
- The results for degradation factor and part-load performance factor demonstrate that the reduction due to transient operation was not significant for this operating condition. The evaporator air flow was always on and the response of the refrigerant side was very fast.

3.3.2 Cyclic Test #2: Cyclic Test under Dry Condition and Outdoor Temperature 115 F

Figures 3-7 to 3-10 present the transient responses for the high ambient cyclic test condition with a dry coil.

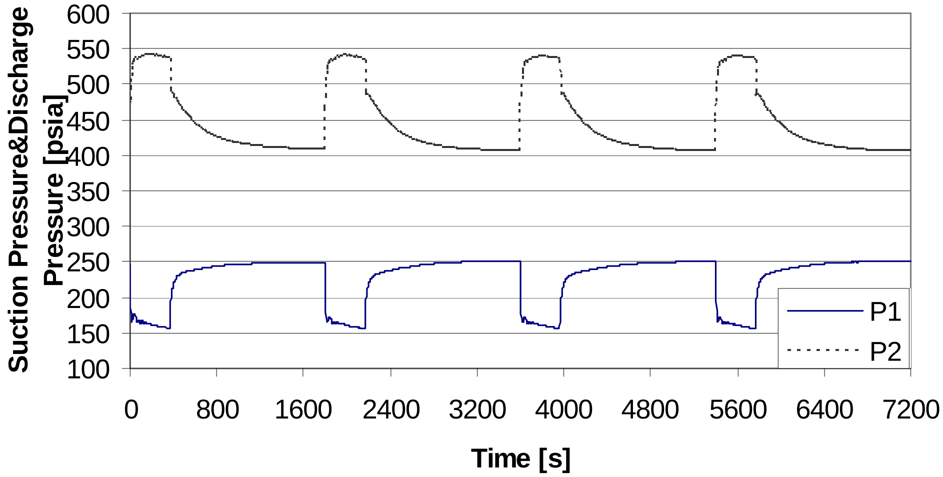


Figure 3-7: Pressure Responses in Cyclic Test #2 of 3-ton Split Unit

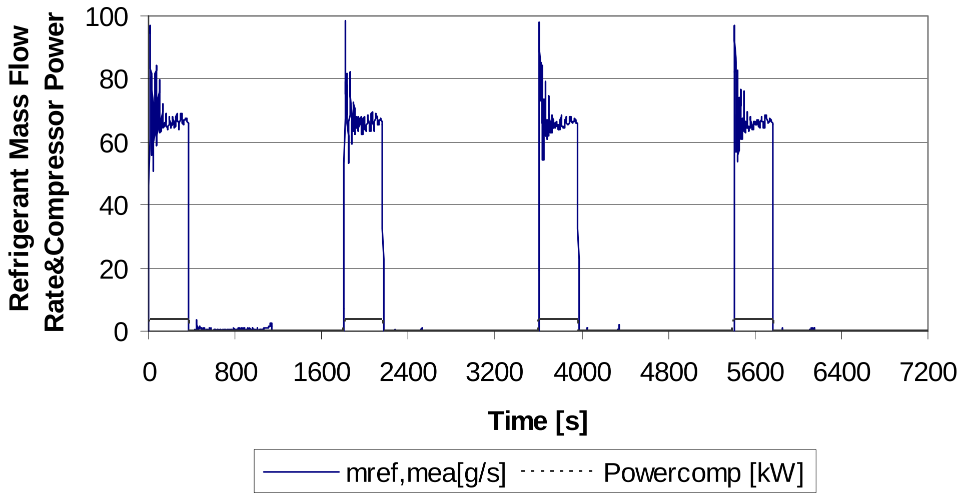


Figure 3-8: Refrigerant Mass Flow rate and Compressor Power Response in Cyclic test#2

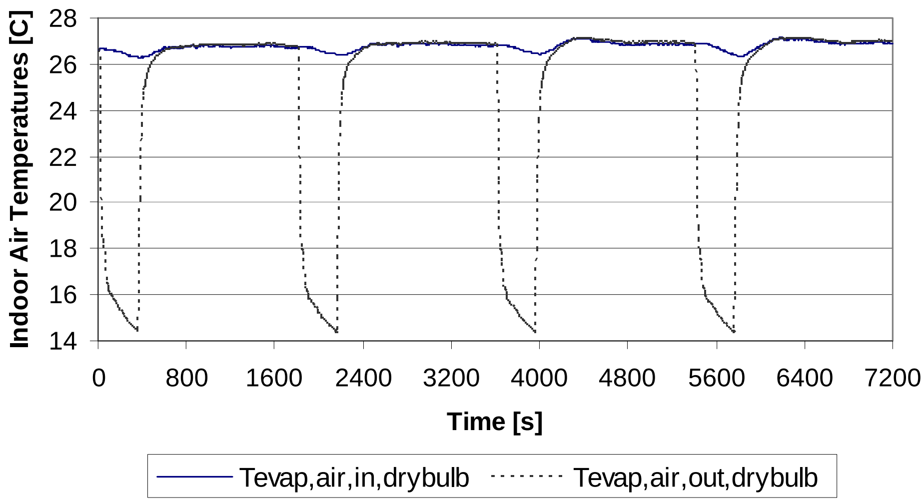


Figure 3-9: Indoor Air Temperature Responses in Cyclic Test #2 of 3-ton Split Unit

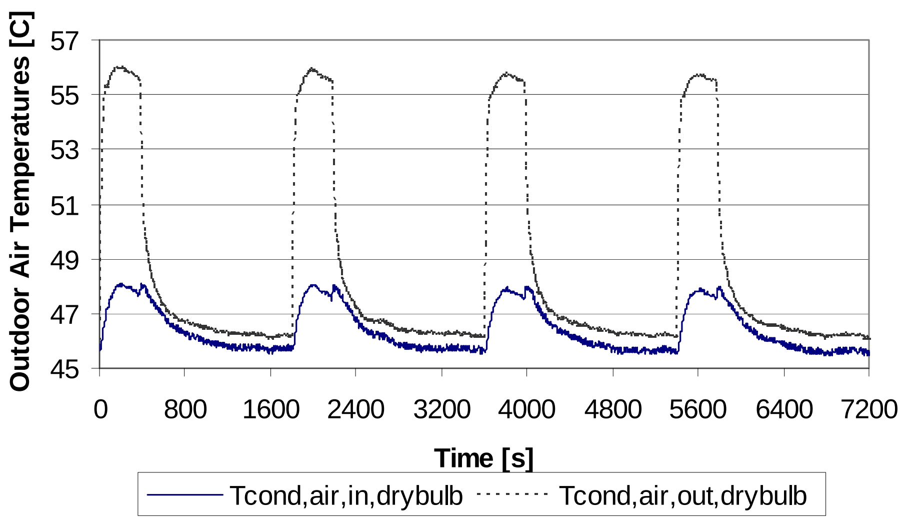


Figure 3-10: Outdoor Air Temperature Responses in Cyclic Test #2 of 3-ton Split Unit

<u>Cyclic Performance</u>

- $Q_{cyc,dry,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles)→ 2657.05 Btu
- $Q_{cyc,dry,net}$ --net cycling cooling capacity during one complete on/off cycle minus evaporator blower power during "on" time (τ =30 min, average among four cycles)
  → $Q_{cyc,dry,tot}$ -W_{blower} = 2657.05 Btu – 450 W × 360 Sec = 2504 Btu
- $Q_{ssdry,net} \times \tau$ -- net steady-state sensible cooling capacity from steady-state Test#3 during the time for one complete cycle→28700 Btu/h×0.5 h × SHR= 14350 Btu×94%=13489
- $CLF = \dfrac{Q_{cyc,dry,net}}{Q_{ss,dry,net} \times \tau}$ --cooling load factor→0.186

- W_{tot} --- Power Consumption During One Cycle → (W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1446804 J, where W_{comp} and W_{fan} are actual measurements, W_{blower} is assigned as 450 W.
- $EER_{cyc,dry}$ --energy efficiency ratio determined from dry cycling test
  → $Q_{cyc,dry,net}/W_{tot}$ = 2504×3600/1446804=6.23
- $EER_{ss,dry}$ -- sensible energy efficiency ratio determined from steady-state Test #3
  →7.42×94%=6.97
- $C_D = \dfrac{1-\dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1-CLF}$ --degradation factor→0.130
- $PLF = 1-0.5\times C_D$ --Partial load performance factor→93.5%

Discussion

- The static pressure difference between the condenser side and evaporator side was about 150 psig during the “off” period for the high ambient transient test. It was much larger than the static pressure difference during the low ambient transient test.
- The transient responses during the high ambient test had similar shapes as those for the low ambient test. However, the degradation factor was increased from 0.09 to 0.13, and the part-load performance factor was decreased from 95.47% to 93.5. The results indicate that the transient performance at high ambient was significantly worse than that at low ambient, since the compressor had to overcome a larger static pressure difference between the condenser side and evaporator side.

3.3.3 Cyclic Test #3: Cyclic Test under Wet Condition and Outdoor Temperature 82 F

Figures 3-11 through 3-15 present transient results for the cyclic test having a wet evaporator.

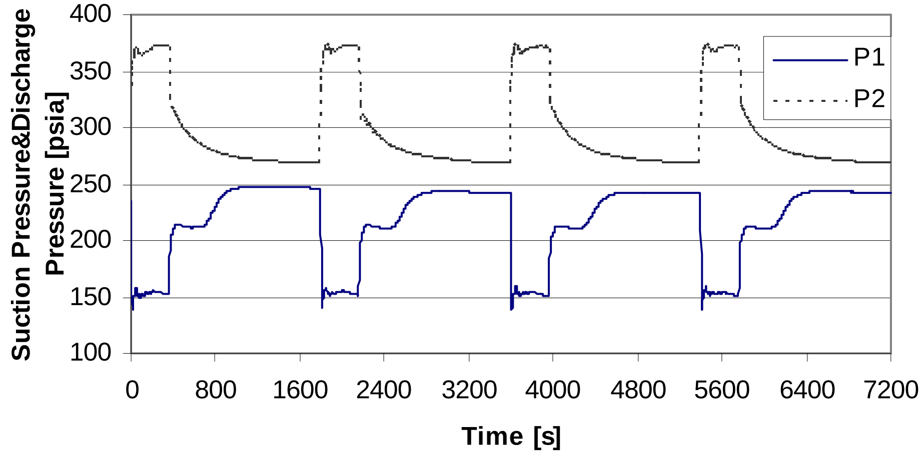


Figure 3-11: Pressure Responses in Cyclic Test #3 of 3-ton Split Unit

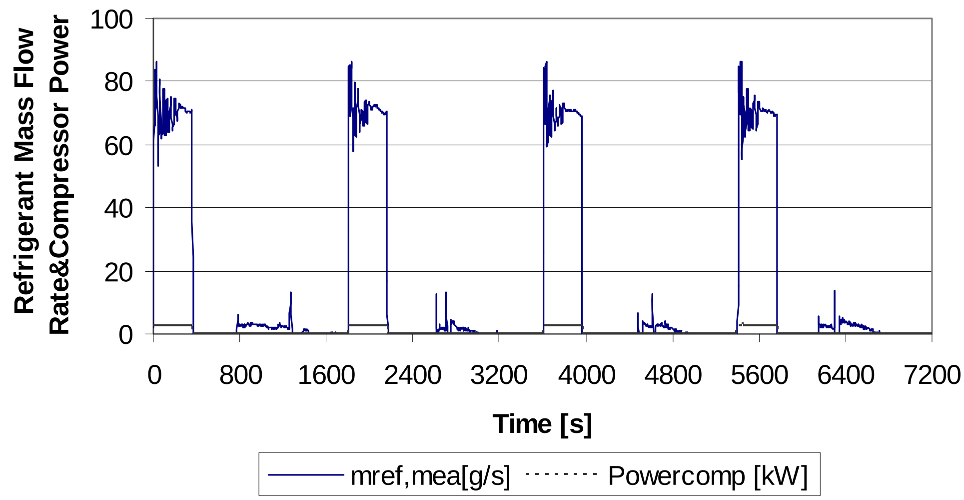


Figure 3-12: Refrigerant Mass Flow Rate and Compressor Power Response in Cyclic Test #3 of 3-ton Split Unit

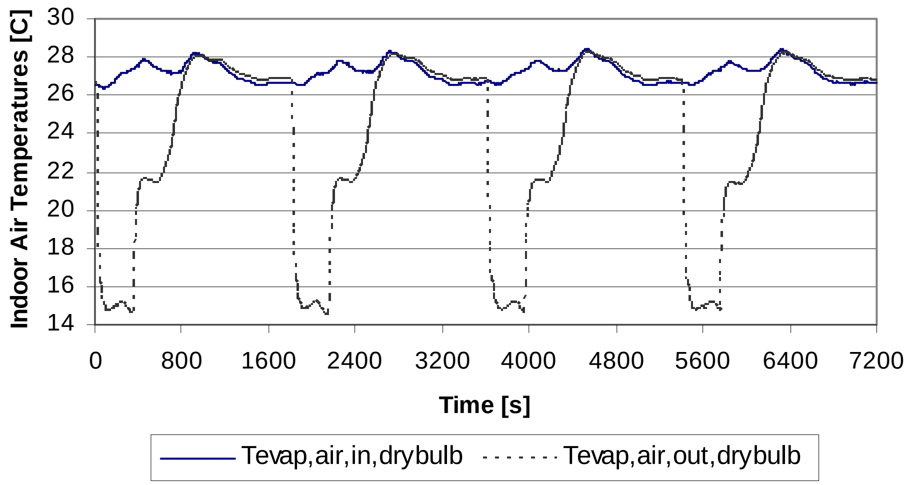


Figure 3-13: Indoor Air Temperature Responses in Cyclic Test #3 of 3-ton Split Unit

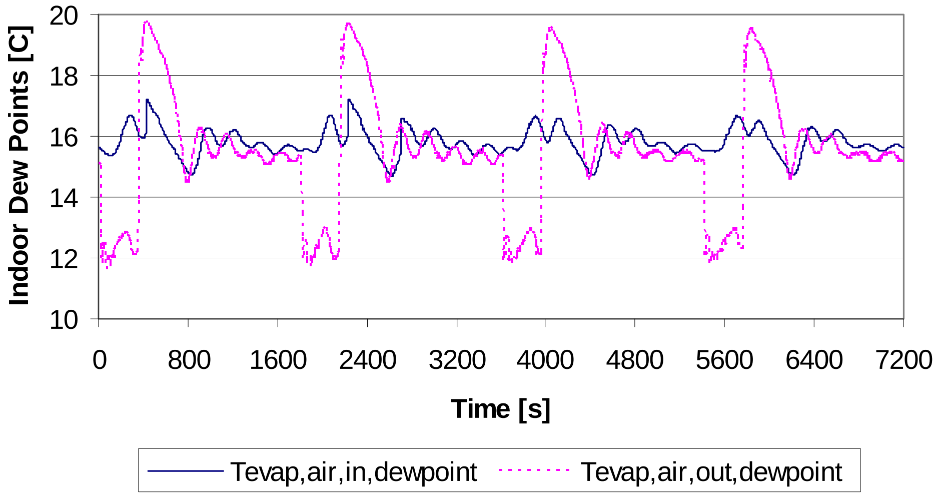


Figure 3-14: Indoor Air Dew Point Responses in Cyclic Test #3 of 3-ton Split Unit

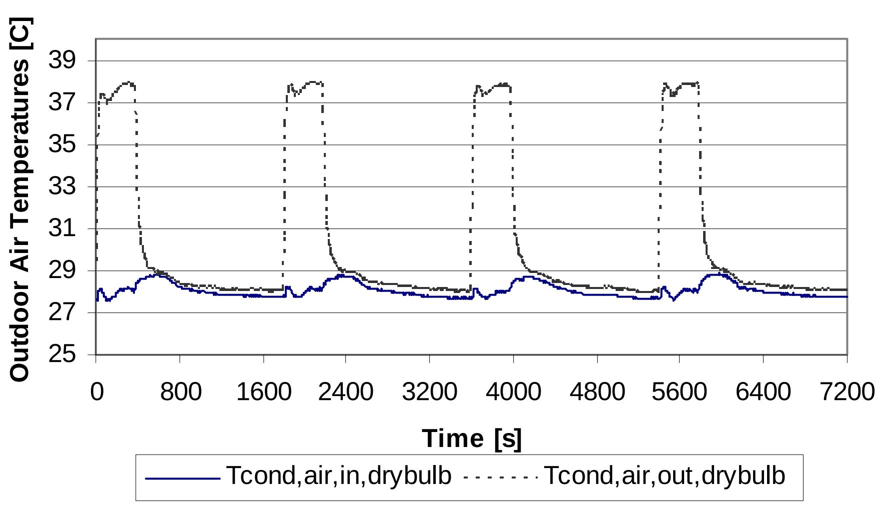


Figure 3-15: Outdoor Air Temperature Responses in Cyclic Test #3 of 3-ton Split Unit

Cyclic Performance

As mentioned before, two humidity sensors, a General Eastern model D2 chilled mirror sensor and a General Eastern model M2 chilled mirror sensor, were installed at the inlet and outlet of the evaporator air flow, respectively. It was found that there was a static deviation in measuring the humidities when the split unit was off. The static deviation, which was obtained from the average of last five minutes data acquisition period at the end of each independent cyclic cycle, is indicated as below,

$T_{evap,air,in,dew} - T_{evap,air,out,dew} = 0.32$ ˚C

The 0.32 ˚C deviation in measuring the outlet air dew point could be due to the matching of different sensors or different sensing locations. In this case, the outlet air dew point measurement needs to be adjusted by adding a 0.32 ˚C to the original measurements. By this way, the wet cyclic performances are calculated as following,

- $Q_{cyc,wet,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles) →3997.46 Btu
- $Q_{cyc,wet,net}$--net cycling cooling capacity during one complete on/off cycle minus the indoor blower power (τ =30 min, average among four cycles)
  → 3997.46 Btu-450 W × (360 Sec) = 3843.72 Btu
- W_{tot} ---Power Consumption during One cycle →(W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1204321 J, where W_{comp} and W_{fan} are actual measurements during the test, W_{blower} is assigned as 450 W.
- $EER_{cyc,wet,net}$ --Energy efficiency ratio determined from wet cycling test →
  $Q_{cyc,wet,net}/W_{tot}$=3843.72 ×3600/1204321 = 11.49
- $Q_{sswet} \times \tau$ --steady-state cooling capacity from Steady-state Test#14 during the time for one complete cycle---41052 Btu/h×0.5 h= 20526 Btu
- $CLF = \dfrac{Q_{cyc,wet,net}}{Q_{ss,wet} \times \tau}$ --cooling load factor→3843.72/20526 = 0.187
- $EER_{ss,wet}$ -- Energy efficiency ratio determined from Steady-state Test#14---12.37.
- $C_D = \dfrac{1 - \dfrac{EER_{cyc,wet,net}}{EER_{ss,wet}}}{1 - CLF}$ --degradation factor→0.0875
- $PLF = 1 - 0.5 \times C_D$ --Partial load performance factor→95.6%

Discussion

- The static pressure difference between the condenser side and evaporator side during the "off" period was about 25 psig during the wet transient test, which was close to the one for the dry transient test at the same ambient temperature.
- As shown in Figure 3-14, at the moment that the compressor was shut off in each cyclic cycle, the evaporator outlet air dew point ascended to a peak, which could be 5 ˚C larger than the evaporator inlet air dew point. The abnormal phenomenon indicates that after the compressor was shut off, some amount of condensed water was left on the evaporating coil. In this case,

the continuous air flow stream allowed the condensed water to re-evaporate into the supply air.

- As shown in Figure 3-11 and 3-13, after the compressor was shut off, the suction pressure and evaporator outlet dry bulb temperature didn't ascend as they did in the dry transient test. The suction pressure and the outlet air dry bulb temperature were still lower than the discharge pressure and the inlet air dry bulb temperature during a transition. It was the condensed water on the evaporating coil that evaporated during this transition, so that the evaporated water absorbed the heat from the air flow stream and the evaporating coil.
- The condensed water that was re-evaporated to the air flow stream led to an increase in the degradation factor and a reduction in the part-load performance factor. However, this effect was relatively small, since the evaporated water increased the sensible cooling capacity at the price of reducing the overall cooling capacity.
- For the humidity sensors with an accuracy of 0.2 ˚C, a static deviation within 0.4 ˚C in dew point measurements of two sensors was unavoidable. The static deviation could lead to a significant error in calculating the cooling capacity during the "off" time.

4. BASELINE TESTING OF 3-TON PACKAGED UNIT

4.1 Air-side Pressure Drop and Evaporator Blower Power Consumption

As shown in Figure 2-5 for the 3-ton packaged unit, one differential pressure transducer DP1 was installed to measure the air-side pressure drop across the indoor unit. The indoor evaporator blower is downstream of the evaporating coil. The airflow rate was altered by a variable speed fan downstream of the nozzles.

The pressure drop across the indoor unit was investigated under dry condition without running the compressor, condenser fan and indoor blower. After all the steady-state tests and cyclic tests were completed, the evaporator blower was moved out of the indoor unit. Thus, the air-side pressure drop across the evaporating coil was investigated. Figure 4-1 and Table 4-1 present results.

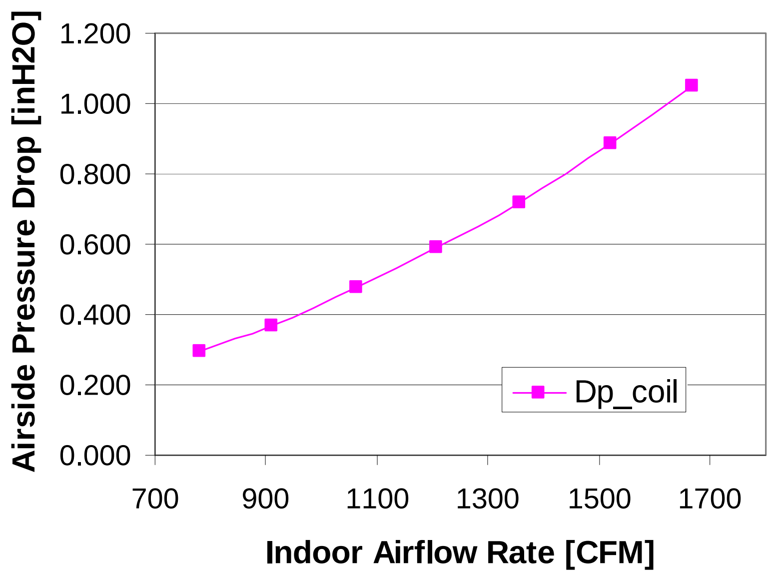


Figure 4-1: Air-Side Pressure Drop versus Indoor Air Flow Rate

Table 4-1: Air-Side Pressure Drop across Evaporating Coil of 3-ton Packaged Unit

Air Flow Rate [CFM]	DP_{coil} [inH2O]
782	0.295
910	0.370
1062	0.476
1208	0.589
1358	0.720
1521	0.888
1666	1.052

The evaporator blower has three speeds (low, medium and high). During the tests, the airflow rate through the indoor unit was altered at each blower speed. When varying the indoor airflow rate, at first, the blower ran at a speed while the variable-speed fan behind the nozzles was off. In this way, the smallest airflow rate at this blower speed was obtained. Afterwards, the variable speed fan was turned on while the blower was still running at the speed. Thus, the airflow rate through the indoor blower at each speed could be altered by adjusting the external variable-speed fan. As a result, the static airside pressure difference across the evaporator blower was varied too, which was measured by the differential pressure transducer DP1.

The electrical measurements of the evaporator blower are presented in Figure 4-2 and Table 4-2 to 4-4.

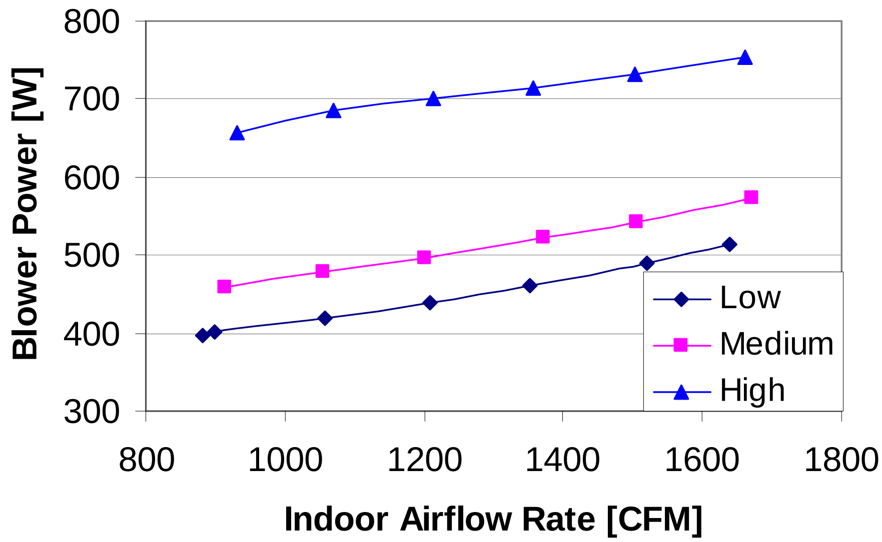


Figure 4-2: Power Consumptions of the Evaporator Blower at Three Speeds

Table 4-2: Electrical Measurements of the Evaporator Blower at Low Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]	Static Pressure Difference across the Blower [inH2O]
882	398	240	1.42	0.531
898	401	242	1.43	0.534
1056	419	241	1.52	0.377
1208	439	242	1.59	0.236
1353	461	241	1.68	-0.004
1520	490	242	1.79	-0.136
1640	513	242	1.88	-0.286

Notes:
In Table 4-2 to Table 4-4, a negative static pressure means that the fan couldn't keep up with the externally driven flow.

Table 4-3: Electrical Measurements of the Evaporator Blower at Medium Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]	Static Pressure Difference across the Blower [inH2O]
914	459	240	1.68	0.561
1054	477	242	1.75	0.433
1201	497	242	1.81	0.296
1372	522	242	1.91	0.140
1506	543	243	1.99	-0.047
1670	573	243	2.10	-0.235

Table 4-4: Electrical Measurements of the Evaporator Blower at High Speed

Airflow Rate [CFM]	Power [W]	Voltage [V]	Current [A]	Static Pressure Difference across the Blower [inH2O]
930	656	239	2.90	0.575
1069	686	243	3.02	0.457
1214	701	242	3.02	0.318
1357	715	243	3.03	0.185
1502	733	243	3.05	-0.016
1661	753	242	3.07	-0.171

The condenser fan has only one speed, which provides a nominal airflow rate of 2350 CFM. The electrical measurements for the condenser fan are,
Nominal Airflow Rate-- 2350 CFM
Power --156 W
Voltage -- 241 V
Current -- 0.58 A

4.2 Steady-State Measurements of 3-ton Packaged Unit

Table 4-5: Overall Steady-State Data of 3-ton Packaged Unit

Test	Outdoor DB	Indoor DB	Indoor WB	Charge	Indoor Airflow	Net Cooling Capacity	Compressor Power
No.	Req/Act [F]	Req/Act [F]	Req/Act [F]	[%]	Req/Act [cfm]	[Btu/h]	[W]
1	115/115.09	80/79.99	67/66.88	100%	1200/1212	29733	3509
2	95/95.05	80/79.99	67/66.87	100%	1200/1199	35903	2714
3	115/115.11	80/80.15	63/62.94	100%	1200/1204	26993	3504
4	115/115.05	80/80.21	63/62.84	118%	1200/1216	26668	3520
5	115/115.03	80/80.16	63/62.89	90%	1200/1213	27437	3384
6	115/115.16	80/80.14	63/62.83	79%	1200/1214	23274	3320
7	115/115.12	80/79.86	63/62.76	69%	1200/1203	18963	3206
8	115/115.14	80/80.17	63/62.92	100%	900/917	25107	3462
9	115/115.07	80/79.93	63/62.88	100%	1050/1072	25896	3489
10	115/115.05	80/80.13	63/62.94	100%	1350/1354	28039	3519
11	115/115.11	80/79.99	63/62.87	100%	1650/1659	29392	3512
12	125/125.13	80/79.88	63/62.81	100%	1200/1216	23734	4011
Allowable Tolerance	±0.5 [F]	±0.5 [F]	±0.3 [F]	---	---	---	---

				Test operating tolerance		
Test	Dev_ref&air	EER	SHR	Outdoor DB	Indoor DB	Indoor WB
No.	[%]	[Btu/Wh]	[%]	[F]	[F]	[F]
1	4.5%	7.15	80%	1.14	0.65	0.30
2	1.1%	10.66	73%	1.4	0.89	0.29
3	N/A	6.49	100%	1.09	0.64	0.23
4	N/A	6.39	100%	1.28	0.31	0.23
5	1.71%	6.79	99%	1.41	0.30	0.31
6	N/A	5.86	100%	1.01	0.43	0.28
7	N/A	4.91	100%	1.24	0.50	0.34
8	N/A	6.25	95%	0.93	0.37	0.12
9	N/A	6.28	99%	1.21	0.83	0.20
10	N/A	6.68	100%	0.94	0.76	0.25
11	3.8%	6.65	100%	1.1	0.99	0.27
12	N/A	5.09	100%	1.32	0.52	0.13
Allowable Tolerance	±6%	---	---	2 [F]	2 [F]	1 [F]

Detailed data for steady-state tests are given in the following tables. Besides the required tests listed in Table 4-5, three more steady-state tests were performed for calculating the necessary parameters for transient tests. Tests #1 to #8 are presented in Tables 4-6 to 4-10, and Tests #9 to #15 are presented in Tables 4-11 to 4-15.

Table 4-6: Refrigerant Side Measurements of 3-ton Packaged Unit

Test No.	1	2	3	4	5	6	7	8
P1 [psi]	163.52	154.95	157.04	158.24	153.99	143.56	122.61	151.13
P2 [psi]	548.34	438.36	541.67	550.58	527.17	511.31	488.31	534.84
P3 [psi]	543.32	432.51	535.42	546.39	520.91	504.98	483.12	530.36
P4 [psi]	535.43	423.17	527.94	537.93	513.03	498.04	477.94	522.93
P5 [psi]	535.62	424.22	531.11	538.05	517.72	503.46	479.52	523.42
P7 [psi]	171.26	162.55	164.52	166.40	161.15	150.21	128.11	158.50
P8 [psi]	165.84	157.24	159.17	160.17	156.34	146.05	125.21	153.17
T1 [˚F]	53.88	60.79	50.75	51.51	60.60	69.95	71.09	48.29
T2 [˚F]	196.63	177.34	188.23	170.84	207.16	222.63	240.92	181.80
T3 [˚F]	183.55	165.14	176.43	161.58	193.70	207.24	222.96	170.69
T4 [˚F]	125.97	105.36	126.57	124.22	130.58	131.29	128.35	126.84
T5 [˚F]	125.02	104.25	125.06	123.45	129.07	132.20	128.91	125.19
T7 [˚F]	55.09	51.72	52.47	53.52	51.40	46.96	37.86	50.30
T8 [˚F]	54.42	60.67	51.14	51.70	58.37	68.64	69.54	48.70
$T_{sub,mea}$ [˚F]	11.99	14.21	11.26	13.92	5.17	0	0	9.95
$X_{sub,cal}$ [%]	---	---	---	---	---	10%	15%	---
$T_{sup,mea}$ [˚F]	1.31	11.50	0	0	11.70	25.27	35.68	0
$X_{sup,cal}$ [%]	---	---	94%	88%	---	---	---	91%
$m_{ref,mea}$ [lb/min]	9.42	8.94	9.19	9.88	8.38	7.40	6.05	9.01
$m_{ref,map}$ [lb/min]	9.24	8.67	8.85	8.90	8.38	7.49	6.14	8.48
Comp_top [˚F]	197.65	177.85	190.36	173.55	207.65	223.05	241.32	184.59
Comp_middle#1 [˚F]	116.81	112.18	106.94	75.02	128.09	143.55	157.31	95.53
Comp_bottom#1 [˚F]	124.58	116.34	115.56	75.57	134.78	149.01	162.42	103.94
Comp_bottom#2 [˚F]	124.85	117.08	115.47	75.18	135.72	150.48	164.48	103.71
Charge [lbs]	7.09[1]	7.09	7.09	8.41	6.38	5.63	4.93	7.09

Notes:
1. The factory charged refrigerant mass is 6.9 lbs. Since extra instrumentation has been added to the system, the updated 100% charge of 7.09 lbs was fixed by specifying T1 in steady-state Test#2 around 60 ˚F, as instructed in the installation manual.
2. All refrigerant-side pressures are presented as absolute pressure.

Table 4-7: Air-Side Measurements of 3-ton Packaged Unit

Test No.	1	2	3	4	5	6	7	8
$T_{evap,air,in,drybulb}$ [˚F]	79.99	79.99	80.15	80.21	80.16	80.14	79.86	80.17
$T_{evap,air,in,wetbulb}$ [˚F]	66.88	66.87	62.94	62.84	62.89	62.83	62.76	62.92
$T_{evap,air,out,drybulb}$ [˚F]	60.55	58.39	57.71	57.65	57.75	60.20	63.42	54.84
$T_{evap,air,out,wetbulb}$ [˚F]	58.64	56.75	54.72	54.76	54.56	55.74	56.87	52.84
$T_{cond,air,in,drybulb}$ [˚F]	115.09	95.05	115.11	115.05	115.03	115.16	115.12	115.14
$T_{cond,air,out,drybulb}$ [˚F]	134.71	115.27	133.37	133.22	132.55	131.18	128.77	132.42
$m_{air,cond}$ [CFM]	2386	2371	2412	2425	2454	---	---	2427
$m_{air,evap}$ [CFM]	1212	1199	1204	1216	1213	1214	1203	917
$m_{dryair,evap}$ [lb/min]	87.93	86.68	88.05	88.40	88.14	87.89	86.50	67.54
BF [%]	15.59%	13.85%	18.30%	17.46%	19.46%	26.9%	39.75%	12.46%
$m_{water,mea}$ [lb/min]	0.105	0.14	0	0	---	0	0	---
$m_{water,cal}$ [lb/min]	0.096	0.15	0	0	0.00309	0	0	0.021962
Dev_{water} [%]	8.69%	-6.68%	0	0	---	0	0	---
P_{atm} [atm]	0.983	0.975	0.985	0.979	0.979	0.979	0.977	0.986
Evap Resistance[3] [inH$_2$O]	0.589	0.589	0.589	0.589	0.589	0.589	0.589	0.370

Notes:
3. Evap Resistance was obtained from the measured pressure drop across the evaporating coil (DP_{coil}), as shown in Table 4-1.

Table 4-8: Power Measurements of 3-ton Packaged Unit

Test No.	1	2	3	4	5	6	7	8
$Power_{comp}$ [W]	3509	2714	3504	3520	3384	3320	3206	3462
V_{comp} [Volts]	231	231	231	232	234	234	235	230
I_{comp} [Amps]	16.21	12.59	16.14	16.09	15.32	14.98	14.46	16.04
$Power_{fan}$ [W]	155	156	155	154	155	156	157	154
$Power_{blower}$ [W]	497[4]	497	497	497	497	497	497	401[5]

Notes:
4. The blower power of 497 W was obtained from Table 4-3 for the medium speed.
5. The blower power of 401 W was obtained from Table 4-2 for the low speed.

Table 4-9: Performance of 3-ton Packaged Unit

Test No.	1	2	3	4	5	6	7	8
$Q_{cooling,air,net}$[Btu/h]	29733	35903	26993	26668	27437	23274	18963	25107
$Q_{cooling,air,tot}$ [Btu/h]	31429	37599	28689	28363	29133	24970	20659	26476
$Q_{cooling,ref}$ [Btu/h]	32883	38041	---	---	29642	---	---	---
$Dev_{ref\&air}$ [%]	4.52%	1.17%	---	---	1.71%	---	---	---
$Q_{condensing}$ [Btu/h]	45990	48298	43462	43019	42173	---	---	41535
EER [Btu/Wh]	7.15	10.66	6.49	6.39	6.79	5.86	4.91	6.25
SHR [%]	80%	73%	100%	100%	99%	100%	100%	95%

Table 4-10: Steady-State Duration of Measurements for 3-ton Packaged Unit

Test No.	1	2	3	4	5	6	7	8
Duration [min]	60	60	60	60	56	60	60	60
Sampling interval [s]	5 seconds							

Continued

Table 4-11: Refrigerant Side Measurements of 3-ton Packaged Unit (continued)

Test No.	9	10	11	12	13	14	15
P1 [psi]	154.26	160.14	165.08	163.49	140.74	149.46	149.21
P2 [psi]	538.43	544.46	549.03	603.77	366.14	374.12	373.90
P3 [psi]	532.61	538.73	543.28	597.55	361.46	370.69	368.61
P4 [psi]	525.12	530.40	535.15	590.56	351.01	359.71	357.86
P5 [psi]	528.66	533.37	536.10	592.38	351.59	357.63	357.52
P7 [psi]	161.60	167.79	173.33	171.32	147.11	156.52	156.96
P8 [psi]	156.26	162.43	167.68	165.83	142.19	151.38	151.74
T1 [˚F]	49.49	52.13	55.11	53.22	46.67	65.60	65.30
T2 [˚F]	185.04	192.29	198.14	202.33	152.86	165.08	165.75
T3 [˚F]	173.43	179.92	185.32	189.67	144.05	154.70	154.00
T4 [˚F]	126.74	126.37	126.15	136.73	91.07	91.21	90.85
T5 [˚F]	125.04	124.96	124.85	135.12	91.77	91.32	90.29
T7 [˚F]	51.36	53.71	55.95	55.10	45.79	49.46	49.33
T8 [˚F]	50.17	51.98	54.50	54.10	44.74	65.14	65.48
$T_{sub,mea}$ [˚F]	10.90	11.72	12.24	10.21	12.46	14.17	15.18
$X_{sub,cal}$ [%]	---	---	---	---	---	---	---
$T_{sup,mea}$ [˚F]	0	0	1.95	0	3.19	18.50	18.30
$X_{sup,cal}$ [%]	92%	95%	---	92%	---	---	---
$m_{ref,mea}$ [lb/min]	9.10	9.28	9.44	9.39	8.18	8.53	8.56
$m_{ref,map}$ [lb/min]	8.68	9.04	9.32	9.11	8.15	8.30	8.29
Comp_top [˚F]	187.68	193.72	199.21	204.90	152.34	164.57	165.65
Comp_middle [˚F]	101.90	111.12	118.78	115.67	95.08	109.57	111.35
Comp_bottom#1 [˚F]	110.45	119.54	126.42	126.61	98.86	111.01	113.55
Comp_bottom#2 [˚F]	110.30	119.65	126.73	126.60	99.97	113.17	114.73
Charge [lbs]	7.09	7.09	7.09	7.09	7.09	7.09	7.09

Table 4-12: Air Side Measurements of 3-ton Packaged Unit (continued)

Test No.	9	10	11	12	13	14	15
$T_{evap,air,in,drybulb}$ [˚F]	79.93	80.13	79.99	79.88	80.21	80.10	80.23
$T_{evap,air,in,wetbulb}$ [˚F]	62.88	62.94	62.87	62.81	58.54	66.91	66.92
$T_{evap,air,out,drybulb}$ [˚F]	56.35	59.32	63.07	59.94	52.43	58.06	58.30
$T_{evap,air,out,wetbulb}$ [˚F]	53.98	62.56	56.83	55.61	47.12	55.91	56.08
$T_{cond,air,in,drybulb}$ [˚F]	115.07	115.05	115.11	125.13	82.24	81.98	82.00
$T_{cond,air,out,drybulb}$ [˚F]	132.98	133.82	134.72	143.29	99.98	101.62	102.12
$m_{air,cond}$ [CFM]	2404	2408	2409	2412	2485	2396	2350
$m_{air,evap}$ [CFM]	1072	1354	1659	1216	1222	1193	1217
$m_{dryair,evap}$ [lb/min]	78.64	98.79	120.37	88.31	89.44	87.05	88.81
BF [%]	14.73%	24.09%	37.83%	26.55%	26.81%	18.6%	18.8%
$m_{water,mea}$ [lb/min]	---	0	0	0	0	---	0.00124
$m_{water,cal}$ [lb/min]	0.0054	0	0	0	0	---	0.00140
Dev_{water} [%]	---	0	0	0	0	---	-13%
P_{atm} [atm]	0.985	0.985	0.983	0.982	0.971	0.981	0.983
Evap Resistance [inH_2O]	0.476	0.720	1.052	0.589	0.589	0.589	0.589

Table 4-13: Power Measurements of 3-ton Packaged Unit (continued)

Test No.	9	10	11	12	13	14	15
$Power_{comp}$ [W]	3489	3519	3512	4011	2301	2320	2310
V_{comp} [Volts]	230	232	232	229	---	---	233
I_{comp} [Amps]	16.14	16.21	16.21	18.40	---	---	10.73
$Power_{fan}$ [W]	155	155	154	154	156	156	158
$Power_{blower}$ [W]	477[6]	522[6]	753[7]	497	497	497	497

Notes:
6. The blower powers of 477 W and 522 W were obtained from Table 4-3 for the medium speed.
7. The blower power of 753 W was obtained from Table 4-4 for the high speed.

Table 4-14: Performance of 3-ton Packaged Unit (continued)

Test No.	9	10	11	12	13	14	15
$Q_{cooling,air,net}$ [Btu/h]	25896	28039	29392	23734	34518	38858	39078
$Q_{cooling,air,tot}$ [Btu/h]	27524	29820	31961	25430	36214	40553	40774
$Q_{cooling,ref}$ [Btu/h]	---	---	33149	---	36219	40113	40418
$Dev_{ref\&air}$ [%]	---	---	3.8%	---	0%	-1.1%	-0.9%
$Q_{condensing}$ [Btu/h]	42540	44606	46441	42491	44902	48590	49063
EER [Btu/Wh]	6.28	6.68	6.65	5.09	11.68	13.07	13.19
SHR [%]	99%	100%	100%	100%	100%	70%	70%

Table 4-15: Steady-State Duration of Measurements for 3-ton Packaged Unit (continued)

Test No.	9	10	11	12	13	14	15
Duration [min]	60	50	60	60	40	40	66
Sampling interval [s]	<10 seconds						

4.3 Transient Measurements of 3-ton Packaged Unit

Table 4-16: Overall Cyclic Test Data of 3-ton Packaged Unit

Cyclic Test No.	Outdoor DB Req/Act [˚F]	Indoor DB Req/Act [˚F]	Indoor WB Req/Act [˚F]	Charge [%]	Indoor Airflow Req/Act [cfm]	Test operating tolerance Outdoor DB [˚F]	Indoor DB [˚F]	Indoor WB [˚F]
1	82/82.22	80/80.00	<57/55.33	100%	1200/1223	2.40	2.14	1.50
2	115/115.20	80/80.07	<57/56.41	100%	1200/1214	3.15	1.98	1.13
3 (wet cyclic)	82/82.10	80/80.10	67/67.07	100%	1200/1222	2.79	1.22	1.46
Allowable Tolerance	±0.5	±0.5	±0.3	---	---	2.0	2.0	1.0

Notes:

- Due to the limitation of the test facility, some test parameters could not be controlled within allowable tolerances.
- A separate General Eastern model HYGRO-M2 chilled mirror sensor was installed to measure the evaporator outlet air dew point for the wet cyclic test.
- The data sampling interval was within 5 seconds for all the cyclic tests.
- On/off interval: 6 minutes on/24 minutes off, ran for four complete cycles.
- All cycling performance tests employed the air enthalpy method to calculate total cooling capacities.
- During all the cyclic tests, the indoor air flow was always on, driven by an external variable speed fan. However, for calculating the cyclic efficiency, the indoor blower power was considered only for the “on” time.

4.3.1 Cyclic Test #1: Cyclic Test under Dry Condition and Outdoor Temperature 82 F

Figures 4-3 to Figure 4-6 present the cycle test transient responses for suction and discharge pressures, refrigerant mass flow rate and compressor power consumption, inlet and outlet dry bulb temperatures of the indoor unit, and inlet and outlet air temperatures of the outdoor unit.

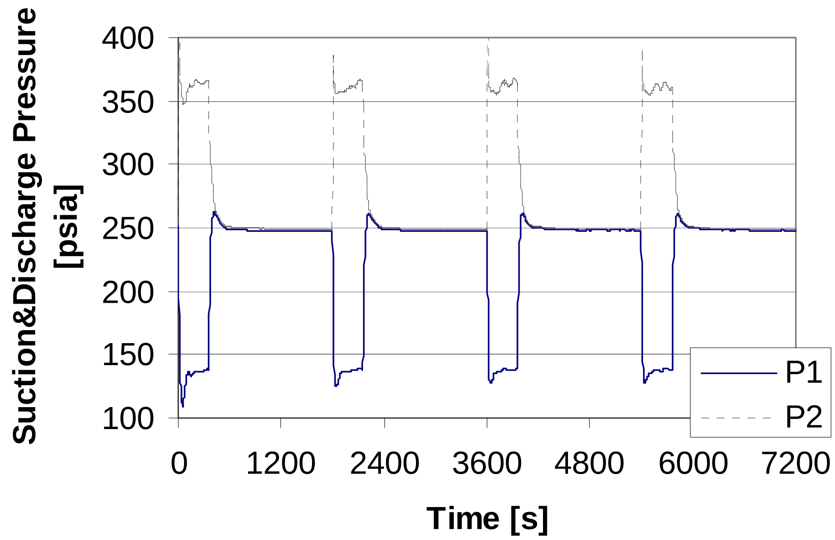


Figure 4-3: Pressure Responses in Cyclic Test # 1 of 3-ton Packaged Unit

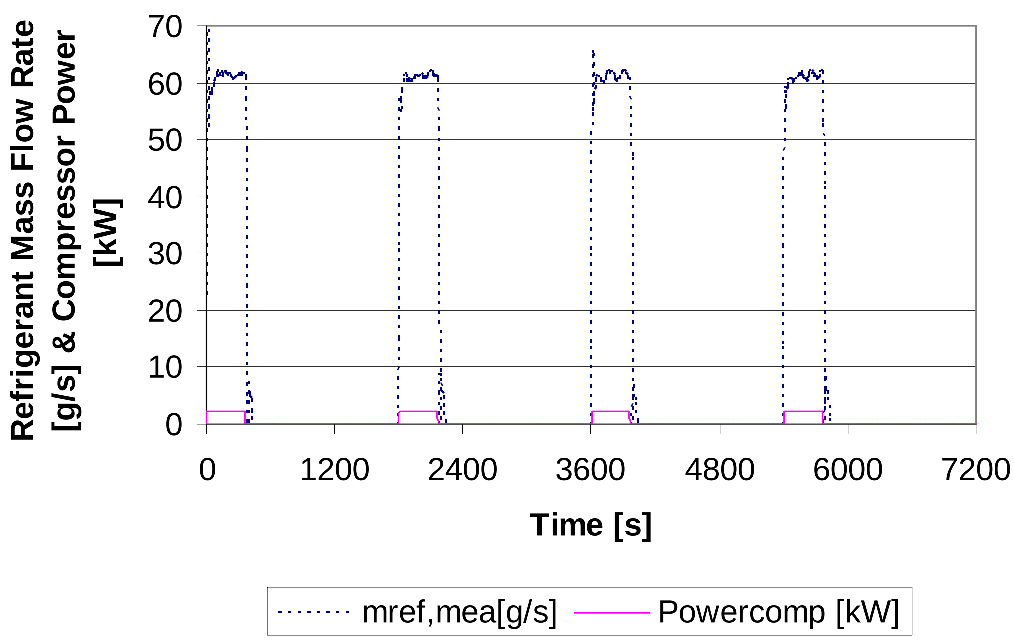


Figure 4-4: Refrigerant Mass Flow Rate and Compressor Power Response in Cyclic Test # 1 of 3-ton Packaged Unit

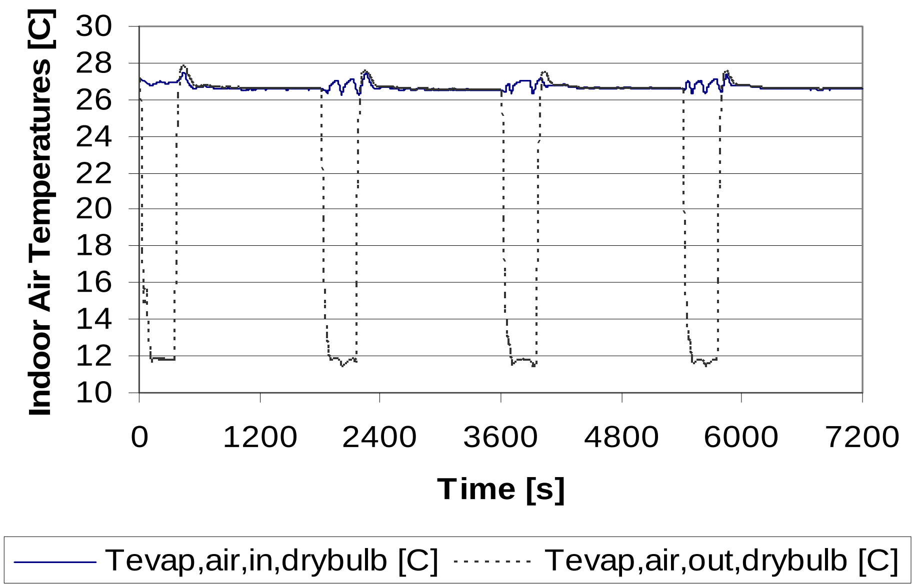


Figure 4-5: Indoor Air Temperature Responses in Cyclic Test # 1 of 3-ton Packaged Unit

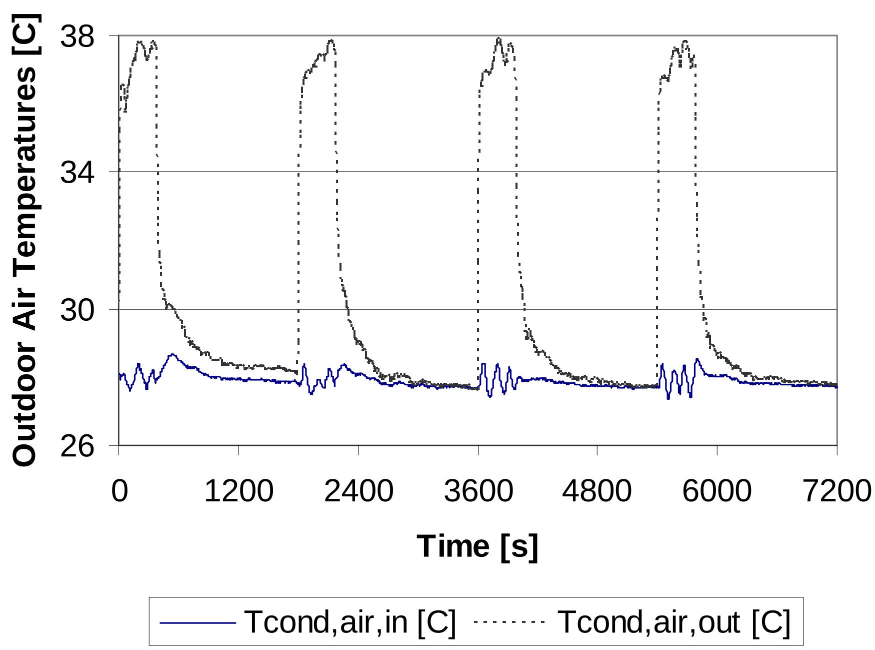


Figure 4-6: Outdoor Air Temperature Responses in Cyclic Test # 1 of 3- ton Packaged Unit

<u>Cyclic Performance</u>

- $Q_{cyc,dry,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles)→ 3404.11 Btu
- $Q_{cyc,dry,net}$ --net cycling cooling capacity during one complete on/off cycle minus evaporator blower power during "on" time (τ =30 min, average among four cycles)
  → $Q_{cyc,dry,net} = Q_{cyc,dry,tot} - W_{blower}$ = 3404.11 Btu – 497 W × 360 Sec = 3234.67 Btu

- $Q_{ssdry} \times \tau$ --net steady-state cooling capacity from Steady-state Test #13 during the time for one complete cycle→ 34518 Btu/h×0.5 h= 17259 Btu
- $CLF = \dfrac{Q_{cyc,dry,net}}{Q_{ss,dry} \times \tau}$ --cooling load factor→ 0.187
- W_{tot} --- Power Consumption During One Cycle) → (W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1052765 J, where W_{comp} and W_{fan} are actual measurements, W_{blower} is assigned as 497 W.
- $EER_{cyc,dry}$ --energy efficiency ratio determined from dry cycling test→ $Q_{cyc,dry,net} / W_{tot}$ = 3234.67×3600/1052765 = 11.06
- $EER_{ss,dry}$ -- energy efficiency ratio determined from Steady-State Test # 13 →11.68
- $C_D = \dfrac{1 - \dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1 - CLF}$ --degradation factor→0.0653
- $PLF = 1 - 0.5 \times C_D$ --Partial load performance factor→96.7%
- EER_B -- energy efficiency ratio determined from Steady-State Test # 14→13.07
- $SEER = PLF(0.5) \times EER_B$ --seasonal energy efficiency ratio→12.64
- Rated SEER---12.00

Discussion

- There was no static pressure difference between the condenser side and evaporator side for the unit using a fixed-area expansion device during the “off” period.
- As shown Figures 4-3 and 4-4, the working difference between the suction pressure and discharge pressure was established very quickly after starting the compressor in each cycle, and then the compressor power consumption was fairly constant. Regarding the unit using a fixed-area expansion device, at the moment of starting the compressor, the suction pressure reached a minimum value, and then it ascended slowly to the final steady-state suction pressure. The discharge pressure reached a maximum value, and then it descended to the final steady-state discharge pressure. These pressure tendencies are reversed to those of the split unit using a TxV.
- The results for degradation factor and part-load performance factor demonstrated that the reduction due to the transient mode was not significant for these conditions, because the evaporator air flow was always on and the response at refrigerant side was very fast. The cyclic penalty of the packaged unit is close to that of the split unit at the low ambient temperature.

4.3.2 Cyclic Test #2: Cyclic Test under Dry Condition and Outdoor Temperature 115 F

Figures 4-7 to 4-10 present the transient responses for the high ambient cyclic test condition with a dry coil.

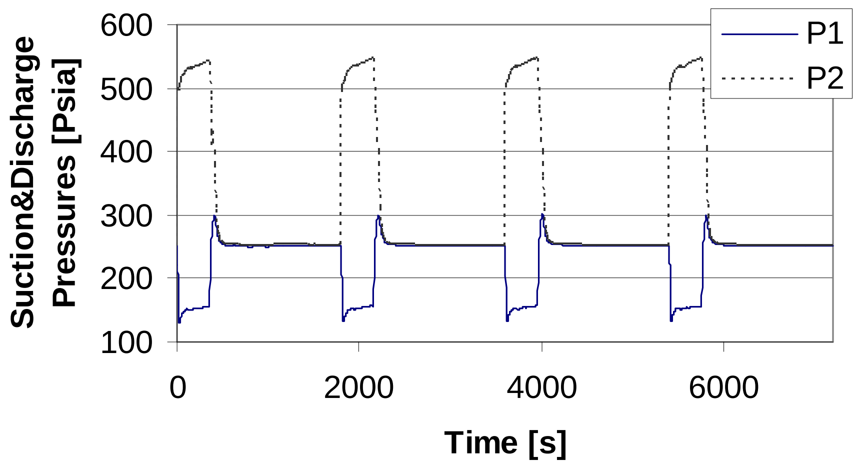


Figure 4-7: Pressure Responses in Cyclic Test # 2 of 3-ton Packaged Unit

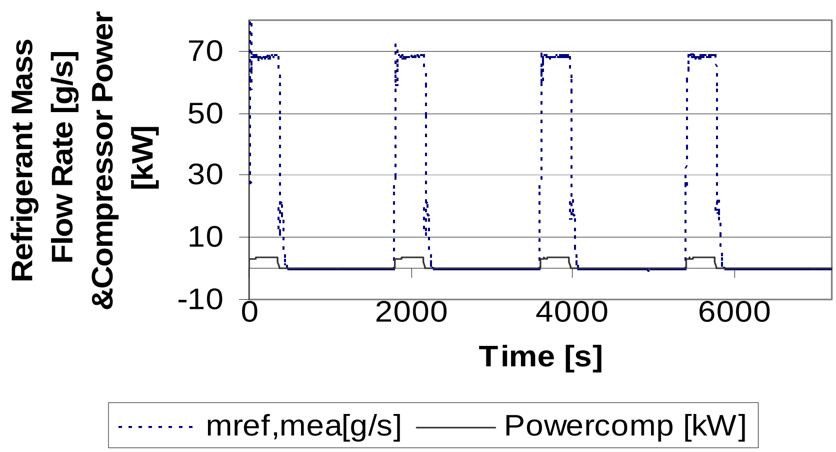


Figure 4-8: Refrigerant Mass Flow rate and Compressor Power Response in Cyclic Test # 2 of 3-ton Packaged Unit

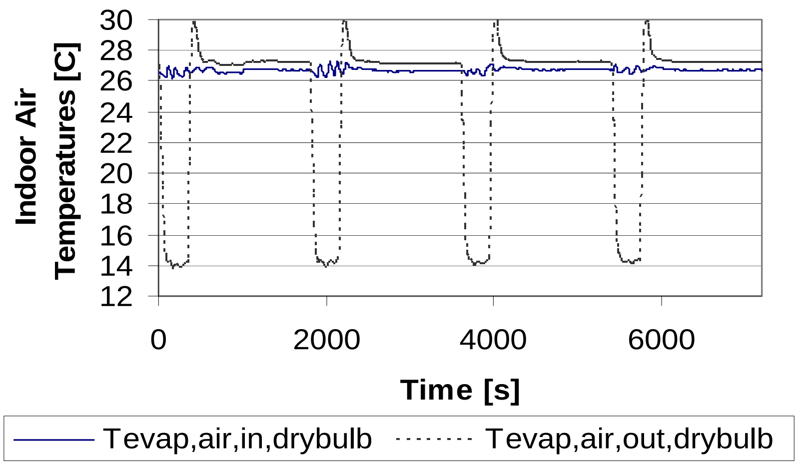


Figure 4-9: Indoor Air Temperature Responses in Cyclic Test # 2 of 3-ton Packaged Unit

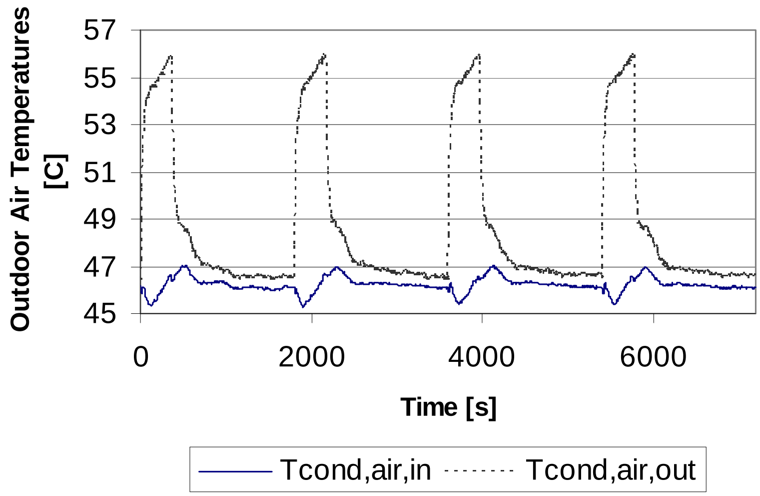


Figure 4-10: Outdoor Air Temperature Responses in Cyclic Test # 2 of 3-ton Packaged Unit

<u>Cyclic Performance</u>

- $Q_{cyc,dry,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles)→2180.49 Btu
- $Q_{cyc,dry,net}$ --net cycling cooling capacity during one complete on/off cycle minus evaporator blower power during "on" time (τ =30 min, average among four cycles)
  → $Q_{cyc,dry,net} = Q_{cyc,dry,tot} - W_{blower}$ = 2180.49 Btu – 497 W × 360 Sec = 2011 Btu
- $Q_{ssdry} \times \tau$ --net steady-state sensible cooling capacity from Steady-state Test # 3 during the time for one complete cycle→ 26993 Btu/h×0.5 h= 13497 Btu
- $CLF = \dfrac{Q_{cyc,dry}}{Q_{ss,dry} \times \tau}$ --cooling load factor→0.1490
- W_{tot} --- Power Consumption during One Cycle) → (W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1459003 J, where W_{comp} and W_{fan} are actual measurements, W_{blower} is assigned as 497 W.

- $EER_{cyc,dry}$ --energy efficiency ratio determined from dry cycling test→
  $Q_{cyc,dry,net} / W_{tot} = 2011 \times 3600/1459003 = 4.96$
- $EER_{ss,dry}$ -- energy efficiency ratio determined from Steady-state Test # 3 → 6.49
- $C_D = \dfrac{1 - \dfrac{EER_{cyc,dry}}{EER_{ss,dry}}}{1 - CLF}$ --degradation factor→0.277
- $PLF = 1 - 0.5 \times C_D$ --Partial load performance factor→86.15%

<u>Discussions</u>

- There was no static pressure difference between the condenser side and evaporator side during the "off" period, although there was a considerable temperature difference between the outdoor temperature and indoor temperature.
- As indicated in Figure 4-9, the hot ambient air heated the evaporator air flow up by 0.9 ˚F during the "off" time, since the indoor air flow pathway of the packaged unit was located at ambient temperature of 115 ˚F.
- As indicated in Figures 4-7 and 4-9, at the moment that the compressor was shut off, hot refrigerant from the condenser side leaked to the evaporator through the fixed-area expansion device. As a result, the refrigerant saturated temperature in the evaporator was higher than the evaporator inlet air temperature, and thus the evaporator turned to heat the indoor air flow.
- The above two heating factors at high ambient temperature led to significant negative cooling capacity during the "off" time. Thus, the cycling penalty of the packaged unit was much larger than that of the split unit, since the indoor part of the split unit was located in the indoor chamber and its TxV stopped the refrigerant flow during the "off" time.

4.3.3 Cyclic Test #3: Cyclic test under wet condition and outdoor temperature 82 F

Figures 4-11 through 4-15 present the transient results for the cyclic test having a wet evaporator.

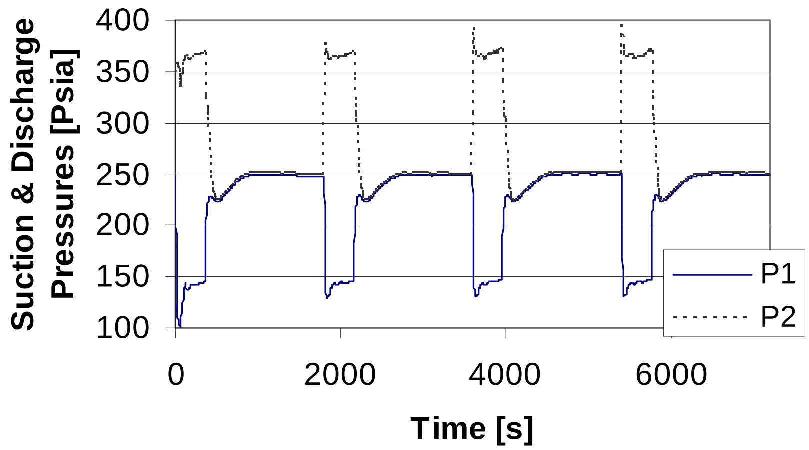


Figure 4-11: Pressure Responses in Cyclic Test # 3 of 3-ton Packaged Unit

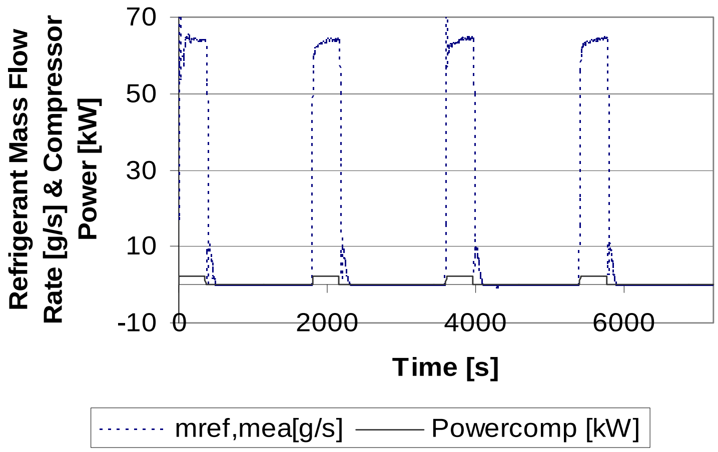


Figure 4-12: Refrigerant Mass Flow rate and Compressor Power Response in Cyclic Test # 3 of 3-ton Packaged Unit

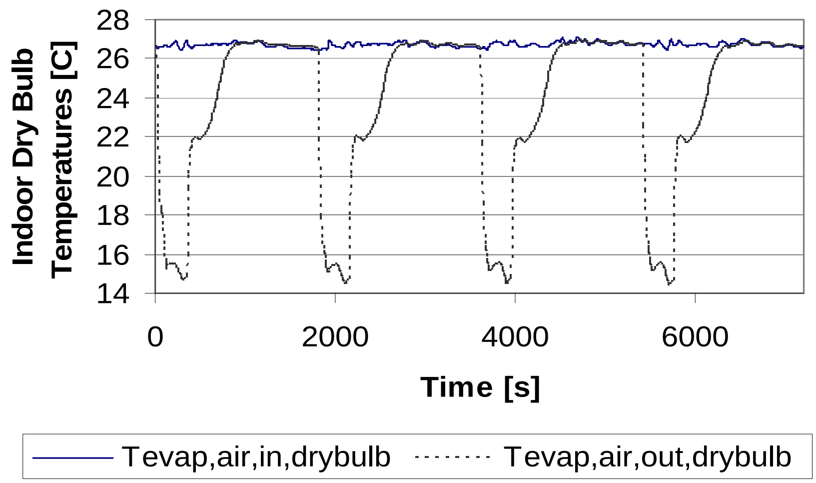


Figure 4-13: Indoor Air Temperature Responses in Cyclic Test # 3 of 3-ton Packaged Unit

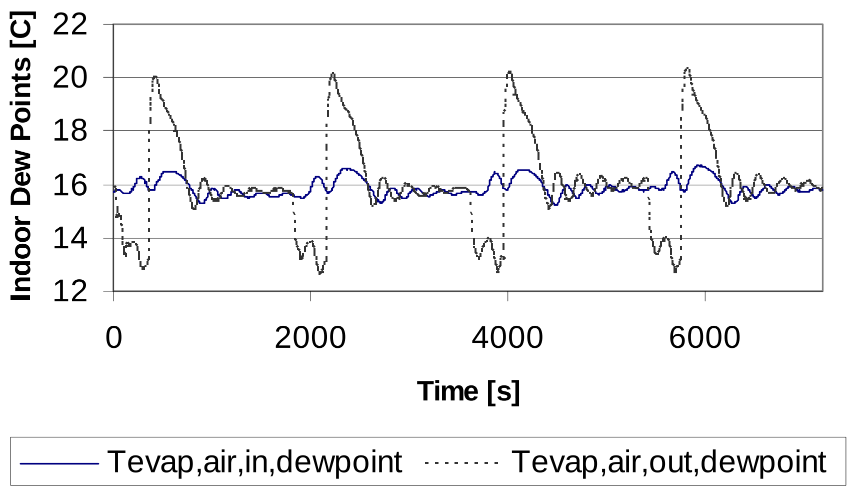


Figure 4-14: Indoor Air Dew Point Responses in Cyclic Test # 3 of 3-ton Packaged Unit

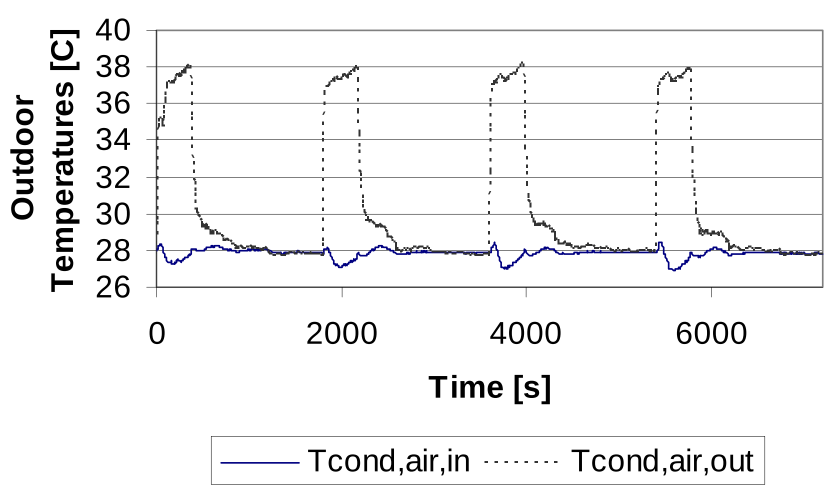


Figure 4-15: Outdoor Air Temperature Responses in Cyclic Test # 3 of 3-ton Packaged Unit

Cyclic Performance

As mentioned before, two humidity sensors, a General Eastern model D2 chilled mirror sensor and a General Eastern model M2 chilled mirror sensor, were installed at the inlet and outlet of the evaporator air flow, respectively. It was found that there was a static deviation in measuring the humidities when the packaged unit was off. The static deviation, which was obtained from the average of the last five minutes of each data acquisition period at the end of each independent cyclic cycle was:

$T_{evap,air,in,dew} - T_{evap,air,out,dew} = -0.13$ ˚C

The -0.13 ˚C deviation could be due to the matching of different sensors or different sensing locations. In this case, the outlet air dew point measurement was adjusted by adding a -0.13 ˚C to the original measurements. By this way, the wet cyclic performances were calculated as follows,

- $Q_{cyc,wet,tot}$ --total cycling cooling capacity during one complete on/off cycle (τ =30 min, average among four cycles) → 3680.40 Btu
- $Q_{cyc,wet,net}$ --net cycling cooling capacity during one complete on/off cycle minus the indoor blower power (τ =30 min, average among four cycles)
  → 3680.40 Btu-497 W × (360 Sec) = 3511.32 Btu
- W_{tot} ---Power Consumption during One Cycle →(W_{comp} + W_{fan} +W_{blower}) × 360 sec = 1051028 J, where W_{comp} and W_{fan} are actual measurements during the test, W_{blower} is assigned as 497 W.
- $EER_{cyc,wet,net}$ --Energy efficiency ratio determined from wet cycling test →
  $Q_{cyc,wet,net}/W_{tot}$ =3511.32×3600/1051028 = 12.03
- $Q_{sswet} \times \tau$ --steady-state cooling capacity from Steady-state Test #15 during the time for one complete cycle→ 39078 Btu/h×0.5 h= 19539 Btu
- $CLF = \dfrac{Q_{cyc,wet}}{Q_{ss,wet} \times \tau}$ --cooling load factor→ 3511.32/19539 = 0.180
- $EER_{ss,wet}$ -- energy efficiency ratio determined from Steady-state Test #15 →13.19
- $C_D = \dfrac{1 - \dfrac{EER_{cyc,wet,net}}{EER_{ss,wet}}}{1 - CLF}$ --degradation factor→0.107
- $PLF = 1 - 0.5 \times C_D$ --Partial load performance factor→94.64%

Discussions

- As indicated in Figure 4-14, after the compressor shuts off, the leftover condensed water on the evaporating coil was re-evaporated to the indoor supply air flow, which was similar to the observation during the wet cyclic test of the split unit.
- As shown in Figures 4-11 and 4-13, after the compressor was shut off, the water on the evaporating coil evaporated by obtaining the heat from the indoor air flow and the evaporating coil. Since the condenser was connected to the evaporator by a fixed-area expansion device, the condensing pressure was pulled down at that point.

- There was condensed water returning to the air flow stream, which increased the sensible cooling capacity at the price of reducing the overall cooling capacity. The cyclic penalty of the packaged unit during the wet cyclic test is similar to that of the split unit.

5. CONCLUSIONS

Some interesting findings are summarized as follow.

<u>Steady-state Tests</u>

1) Regarding the 3-ton split unit that uses a TxV, the TxV could maintain a fairly constant superheat degree in most of the tests, except at the lowest charge of 70% of the normal charge. At the 70% charge level, the superheat degree was significantly larger, since the TxV was fully opened in this case.
2) Regarding the 3-ton packaged unit that uses a fixed-area expansion device, most of the high ambient and dry conditions led to two-phase refrigerant entering the compressor. In this case, the cooling capacity for the refrigerant side could not be determined.
3) The water condensation rate predicted by the air-side humidity measurements did not match the measured water flow rate very well. The reason could be that the air humidity measurement was conducted only at one position, which could not give an overall humidity distribution. In addition, some condensate water could be carried away by the indoor air flow, since the measured water condensate rate tended to be 10% less than the rate predicted with airside measurements.

<u>Transient Tests</u>

1) Regarding the split unit that uses a TxV, during "off" time of the transient tests, the refrigerant in the condenser and evaporator was separated by the closed TxV. In this case, there was a static pressure difference between the condenser and the evaporator, due to the temperature difference between the indoor and the outdoor.
2) Regarding the packaged unit that uses a fixed-area expansion device, there was no static pressure difference between the condenser and evaporator during "off" time, since the refrigerant could mix through the expansion device.
3) At the moment that the compressor was shut off in each wet cyclic cycle, the evaporator outlet air dew point increased, reaching a peak which could be 5 ˚C larger than the evaporator inlet air dew point. This abnormal phenomenon indicates that after the compressor was shut off, some amount of condensed water was still left on the evaporating coil. In this case, the continuous air flow stream re-evaporated some of the condensed water and threw it back into the supply air. The water evaporated by absorbing heat from the indoor air flow and the evaporator coil surface, and thus it could pull down the indoor dry bulb temperature and the suction pressure for a while. The returning condensed water tended to increase the degradation factor and reduce the part-load performance factor. However, this effect was not significant, since the water increased the sensible cooling capacity at the price of reducing the overall cooling capacity.
4) Both units had similar cyclic penalties during the dry and wet cyclic tests at low ambient temperature. However, the cyclic penalty of the packaged unit was noticeably worse at high ambient temperature. There are two reasons to explain this. Firstly, the hot ambient air heated the indoor air flow up by 0.9 ˚F during "off" time, since the indoor air flow pathway of the packaged unit was located at the ambient temperature of 115 ˚F. Secondly, at the moment that the compressor was shut off, hot refrigerant from the condenser side leaked to the evaporator through the fixed-area expansion

device. Consequently, the refrigerant saturated temperature in the evaporator was higher than the evaporator inlet air temperature, and thus the evaporator turned to heat the indoor air flow. These two heating factors at the high ambient temperature led to significant negative cooling capacity during the “off” time. Thus, the cycling penalty of the packaged unit was much larger than that of the split unit, since the indoor part of the split unit was located in the indoor chamber and its TxV stopped the refrigerant flow during the “off” time.

REFERENCES

ASHRAE STANDARD, ANSI/ASHRAE 116-1995, Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps

ARI STANDARD 210/240, 1994, UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT

Engineering Equation Solver (EES), S. A. Klein, 1992-2004, WWW.FCHART .COM

Appendix A: Catalog of the Measurements for Unit #1

Outdoor Section

Compressor

Sensor	Quantity	Location	Objective
Thermocouple	1	Top of the compressor shell (S)	Measure the compressor top surface temperature
Thermocouple	2	Middle of the compressor shell (S)	Measure the compressor middle surface temperature
Thermocouple	2	Bottom of the compressor shell (S)	Measure the compressor bottom surface temperature, for indicating the lubricant temperature
Power meter	1		Measure the compressor power

Note: (S) means the locations at the outside surface for indicating the refrigerant temperature. (T) means the locations inside the tubes for measuring the refrigerant temperature. (A) means the locations in the air flow path for measuring the air temperature.

Discharge line

Sensor	Quantity	Location	Objective
Thermocouple	1	At the compressor exit (T)	Measure the compressor discharge refrigerant temperature
Pressure transducer	1	At the compressor exit	Measure the compressor discharge refrigerant pressure
Pressure transducer	1	At the condenser entrance	Measure the condenser inlet refrigerant pressure
Thermocouple	1	At the exit of the discharge line (S)	Measure the condenser inlet refrigerant temperature

Suction line

Sensor	Quantity	Location	Objective
Thermocouple	1	At the compressor entrance (T)	Measure the compressor suction refrigerant temperature
Pressure transducer	1	At the compressor entrance	Measure the compressor suction refrigerant pressure
Sight glass	1	At the compressor entrance	Judge if two-phase flow enters the compressor

Condenser

Sensor	Quantity	Location	Objective
Thermocouple	1	At the condenser exit (T)	Measure the condenser outlet refrigerant temperature
Differential Pressure transducer	1	Across the condenser, not including the discharge line and the liquid line	Measure the pressure drop across the condenser
Thermocouple	12	Each side of the condenser has three (A)	Measure the condenser inlet air temperature
Thermocouple	8	On the top of the condenser (A)	Measure the condenser outlet air temperature
Power meter	1		Measure the condenser fan power

Liquid line

Sensor	Quantity	Location	Objective
Flow meter	1	After the condenser	Measure the refrigerant mass flow rate
Sight glass	1	Before the flow meter	Judge if two-phase flow enters the mass flow meter
Thermocouple	1	Before the expansion device (T)	Measure the temperature ahead of the expansion device
Pressure transducer	1	Before the expansion device	Measure the pressure ahead of the expansion device
Sight glass	1	Before the expansion device	Judge if two-phase flow enters the expansion device

Summary of the measurements at the outdoor side (totally 41 measurements):
4 pressure transducers
1 differential pressure transducers
3 sight glasses
4 T-type probe thermocouples for measuring refrigerant temperatures
26 T-type wire thermocouples for measuring air temperatures and surface temperatures
1 mass flow meter
2 power meters

Indoor Section

Distributor

Sensor	Quantity	Location	Objective
Pressure transducer	1	At the distributor entrance	Measure the refrigerant pressure after the expansion device
Thermocouple	1	At the distributor entrance (T)	Measure the refrigerant temperature after the expansion device

Evaporator

Sensor	Quantity	Location	Objective
Differential Pressure transducer	1	Across the evaporator, not including the distribution lines and the suction line	Measure the pressure drop across one evaporator branch.
Thermocouple	1	At the exit of the evaporator (T)	Measure the evaporator outlet refrigerant temperature
Pressure transducer	1	At the exit of the evaporator	Measure the evaporator outlet refrigerant pressure
Thermocouple	8	At the evaporator inlet air flow path(A)	Measure the evaporator inlet air dry bulb temperature
Thermocouple	8	At the evaporator outlet air flow path (A)	Measure the evaporator outlet air dry bulb temperature
Dew point measurement	1	At the evaporator inlet air flow path	Measure the evaporator inlet air wet bulb temperature
Dew point measurement	1	At the evaporator outlet air flow path	Measure the evaporator outlet air wet bulb temperature
Airflow rate measurement	1	In the wind tunnel	Measure the evaporator air flow rate
Thermocouples	1	At the entrance of the evaporator (S)	Measure the evaporator inlet refrigerant temperature

Summary of the measurements at the indoor side (totally 25 measurements):
2 pressure transducers
1 differential pressure transducer
2 T-type probe thermal couples
17 T-type wire thermal couples
2 Dew point measurements
1 airflow rate measurement

Appendix B: Catalog of the Measurements for Unit #2

Compressor

Sensor	Quantity	Location	Objective
Thermocouple	1	Top of the compressor shell (S)	Measure the compressor top surface temperature
Thermocouple	1	Middle of the compressor shell (S)	Measure the compressor middle surface temperature
Thermocouple	2	Bottom of the compressor shell (S)	Measure the compressor bottom surface temperature, for indicating the lubricant temperature
Power meter	1		Measure the compressor power

Note: (S) means the locations at the outside surface for indicating the refrigerant temperature. (T) means the locations inside the tubes for measuring the refrigerant temperature. (A) means the locations in the air flow path for measuring the air temperature.

Discharge line

Sensor	Quantity	Location	Objective
Thermocouple	1	At the compressor exit (T)	Measure the compressor discharge refrigerant temperature
Pressure transducer	1	At the compressor exit	Measure the compressor discharge refrigerant pressure
Thermocouple	1	At the exit of the discharge line (S)	Measure the condenser inlet refrigerant temperature

Suction line

Sensor	Quantity	Location	Objective
Thermocouple	1	At the compressor entrance (T)	Measure the compressor suction refrigerant temperature
Pressure transducer	1	At the compressor entrance	Measure the compressor suction refrigerant pressure

Condenser

Sensor	Quantity	Location	Objective
Thermocouple	1	At the condenser exit (T)	Measure the condenser outlet refrigerant temperature
Differential Pressure transducer	1	Across the condenser, not including the discharge line and the liquid line	Measure the pressure drop across the condenser
Pressure transducer	1	At the condenser exit	Measure the pressure at the condenser exit
Thermocouple	12	Each side of the condenser has three (A)	Measure the condenser inlet air temperature
Thermocouple	6	On the top of the condenser (A)	Measure the condenser outlet air temperature
Power meter	1		Measure the condenser fan power
Thermocouple	16	At tube bends, as indicated in Figure 3 (S)	Measure the local refrigerant temperatures to indicate the distribution of two-phase and single-phase heat transfer areas
Pressure transducer	1	At the condenser exit	Measure the pressure at the condenser exit

Liquid line

Sensor	Quantity	Location	Objective
Flow meter	1	After the condenser	Measure the refrigerant mass flow rate
Sight glass	1	Before the flow meter	Judge if two-phase flow enters the mass flow meter
Thermocouple	1	Before the expansion device (T)	Measure the temperature ahead of the expansion device
Pressure transducer	1	Before the expansion device	Measure the pressure ahead of the expansion device
Sight glass	1	Before the expansion device	Judge if two-phase flow enters the expansion device

Evaporator

Sensor	Quantity	Location	Objective
Differential Pressure transducer	1	Across the evaporator, not including the distribution lines and the suction line	Measure the pressure drop across the evaporator. In addition, identify if the refrigerant flow is allocated equally among the four evaporator branches
Pressure transducer	1	At the entrance of the evaporator	Measure the evaporator inlet refrigerant pressure
Thermocouple	1	At the exit of the evaporator (T)	Measure the evaporator outlet refrigerant temperature
Thermocouple	8	At the evaporator inlet air flow path(A)	Measure the evaporator inlet air dry bulb temperature
Thermocouple	8	At the evaporator outlet air flow path (A)	Measure the evaporator outlet air dry bulb temperature
Dew point measurement	1	At the evaporator inlet air flow path	Measure the evaporator inlet air wet bulb temperature
Dew point measurement	1	At the evaporator outlet air flow path	Measure the evaporator outlet air wet bulb temperature
Airflow rate measurement	1	In the wind tunnel	Measure the evaporator air flow rate
Thermocouples	1	At the entrance of the evaporator, as indicated in Figure 4 (S)	Measure the evaporator inlet refrigerant temperature
Thermocouples	18	At the tube bends, as indicated in Figure 4 (S)	Measure the local refrigerant temperatures to indicate the distribution of two-phase and single-phase heat transfer areas
Power meter	1		Measure the evaporator fan power

Summary of the measurements:
5 pressure transducers
2 differential pressure transducers
5 T-type probe thermal couples
74 T-type wire thermal couples
2 Dew point measurements
2 sight glasses
1 mass flow meter
3 power meters
1 airflow rate measurement